Fatigue Life Evaluation of a Grab Ship Unloader

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This case study on a 23-year-old grab ship unloader was performed to evaluate its fatigue life by field tests and numerical analyses. Using the strain gages mounted to the adequate points, the real strain responses were obtained during the static test and long-term monitoring. The static test was performed under different operating conditions including boom lowering and trolley stopped on critical locations with grab full and empty. The measured data were used to evaluate the current safety level according to the national design standard for crane structures. The strains were recorded over one year to obtain the real stress range histograms for fatigue assessment by using the rainflow counting method. The results show that the front tie members were subjected to unexpected moments that induced large enough stresses to threaten safety. This problem arose from end pins that were not free to rotate. Some members with lower safety levels should be monitored until repaired, and some spots should be inspected for fatigue cracks more frequently due to an elevated risk of fatigue damage. Finally, this case study includes recommendations for the design details of new cranes.

1. INTRODUCTION

Port cranes are often subjected to alternating loading mainly due to the movement of cargo, trolley and lifting devices. Besides, for mooring the ships, the waterside parts of the structures above ships are mostly designed to be retractable. These retractable parts of structures have higher probabilities of collapse than the others. Two examples occurred at Southampton Container Terminal in 2008 and 2009; the retractable booms and front tie members fell on the ships and resulted in great damage. Consequently, to ensure the safety of port cranes in use, it is necessary to evaluate the strength safety and fatigue performance of the structure at an appropriate interval with aging, especially for older cranes. The grab ship unloader is a large scale port crane and plays an important role in unloading bulk materials such as iron ore, coal, and stone. However, very little information can be obtained on the structural health assessment of ship unloaders. Liang\(^{(1)}\) carried out an assessment of fatigue life on an 11-year-old grab ship unloader.

To carry out a health assessment of older unloaders, it is necessary to obtain more information about the current stress response of structural members; especially in fatigue check, the representative or real stress range histograms should be known at the positions where the fatigue check needs to be carried out. However, the typical loading history is not available in the current literature, and it is not easily defined due to the different operators, material types unloaded, grab partly or fully loaded, ship types, dimensions of the hatches, etc. To solve these problems, the most direct way is to take the stress measurement from field tests. Consequently, this paper performs a fatigue life evaluation of a 23-year-old grab ship unloader based on measured data obtained from field tests conducted under different operating conditions and long-term monitoring. This protocol was taken to increase the reliability of evaluation results.

2. METHODS AND PROCEDURES

2.1 Unloader Studied

Figure 1 is the grab ship unloader studied herein. The unloader is installed on the quay to unload iron ore, coal, and stone. It is fitted with a grab to remove these bulk materials from the ship to a hopper which controls the flow onto a conveyor belt. It is a type of bridge crane with an outreach of 37 m and a lift height of 39 m. The unloading capacity is rated at 2710 t/h for iron ore and 2590 t/h for coal. The hoisting load (carrying capacity) is 43 t. It has two portal legs which are connected by platforms and diagonal struts. The four corners are supported on the main balancers of travel mechanisms. A trolley runs between double box girder constructions. A stationary track girder is provided between the legs and extends beyond the landside leg. A retractable boom is pin-connected to the fixed track girder and kept in a horizontal position by the front tie
members leading to the pylon on the waterside leg. The front ties are pin-connected at both ends. During the raising and lowering of the boom, the front ties are folded at central pins. The main steel structure is built up from box-type elements except that the front ties are I-shape. Under gravity, the front ties bend around the cross-sectional minor axis which is parallel to the rational axis of the end pin.

2.2 Field Tests

Using strain gages attached to the steel structure, real stress responses were obtained during the static test and long-term monitoring. There were 34 strain gages at specific points mainly on the structural members of retractable parts, as shown in Fig. 2. The gage locations were determined by preliminary static loading tests and structural analyses. Additionally, the practicability of work necessary to attach the gages was also considered. In order to protect the data logger, it was set up in an electric control room, resulting in connecting cables of up to 100m between the strain gages and the data logger. The 4-wire gages and data acquisition system made by the measurement company HBM were used to eliminate cable effects(2).

2.2.1 Static test

The static test was performed under different operating conditions, including during boom lowering and when the trolley stopped at critical locations with the grab empty or loaded. The measured data were used to evaluate the current safety level according to CNS 6426 specification(3) for the design of crane structures.

Boom lowering

At the beginning, the boom was lifted up into the catch hooks, and its vertical angle was 80°. Then, the boom was lowered to the horizontal working position. The ropes were released and not subjected to any tension (i.e. the load was transferred from the ropes to the front ties).

Trolley moving

The trolley with grab, which was empty, loaded by weight block and fully filled by iron ore, was moved into four positions and fixed there, including hopper centre, boom midspan, the location at which the boom was hung by front ties, and maximum outreach. The grab and weight block were 16.6 t and 25.04 t, respectively. The total weight of the full grab could be estimated by test data; and further, the results were used to check the overload level.

2.2.2 Long-term monitoring

This paper carried out a long-term stress monitoring for one year to continuously collect the dynamic data under real working conditions. The recorded data were used to establish the stress range spectra at the positions where the fatigue assessment needed to be carried out.

3. RESULTS AND DISCUSSION

The results show that the boom and back ties pass the strength check and have lower risks of fatigue collapse. However, the front ties have some safety problems. Consequently, only the results of the front ties are discussed in this paper.

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Fig. 1. Bridge type ship unloader crane.

Fig. 2. Strain gage locations.

Note: ^1 → single △ 0°/45°/90° rosettes
^2 All sections facing the sea
3.1 Strength Check

Table 1 shows the measured and calculated stresses at section A of the left front tie close to the lower pin, in the case of the trolley fixed at maximum outreach where the stresses of front ties reached maximum. Four strain gages were attached to the two side flanges at section A as shown in Fig. 3. All gage centres were 3 cm away from the plate edge and designed to be symmetrical with respect to the centroid of the section. Based on the theory of linear elasticity, the measured stresses under different static loading tests were analysed and recombined to calculate the stresses according to the load combinations considered in the CNS 6426 specification. The front ties were of A572 Grade 50 steel with nominal yield strength of 345 MPa and tensile strength of 450 MPa. The maximum permissible tensile stress with respect to the elastic limit, \( \sigma_a \), was 230 \( = \min\{345/1.5, 450/1.8\} \) MPa for the load combination A of ship unloader working without wind.

Based on the measured data, the maximum calculated stress was at the position of gage 3 for the load combination A, and the value was 268.3 MPa, which exceeded the maximum permissible stress of 230 MPa. At the same section, the maximum edge stress was further calculated as 300 MPa, which significantly exceeded the maximum permissible value of 230 MPa and was quite close to the yield strength of 345 MPa. It was evident that the front tie failed in the strength check.

\[ \sigma_a = \min\{345/1.5, 450/1.8\} \]

A bending mode was found from the great difference between the measured stresses of one section. The stresses at the positions of gages 1 and 3 were very close but much less than those at the positions of gages 2 and 4. The direction of the unexpected bending moment was observed around the rotational axis of the end pin. It was caused largely by the end pin which was not free to rotate. This problem could have resulted from the spherical plain bearing due to insufficient lubrication, wear, friction and deformation during the past 23 years. After examining all 8 end pins of front ties from measured data, it was observed that 6 pins, especially two lower and one upper ones, could not rotate freely. The unexpected moments induced stresses that were large enough to damage the safety of the front tie members. Moreover, the results indicated that the discharged material weight of the grab when fully loaded could be up to 23% overload, which also leads to an increase in stress.

3.2 Fatigue Analysis

The fatigue assessment was carried out on the front ties at the butt joint of flange near all 8 end pins, where fatigue cracks have a high probability to initiate and develop. According to the JIB 8821 standard, the fatigue design \( \Delta \sigma - N \) curve for the butt joint is the line marked with “100”, as shown in Fig. 4.

3.2.1 Real stress range histogram

A large amount of stored stress data was processed using the rainflow counting method\(^4\) to determine the distribution of the stress range. Figure 5 is the stress range histogram for gage 3 measured while unloading iron ore from one ship. The number of occurrences, \( n_i \), for the stress range, \( \Delta \sigma_i \), to be above the fatigue limit of 40.5 MPa (specified in JIS B 8821\(^5\)) reached a total of 3,280; however, the total number of recorded hoisting cycles was only 1,719. This difference could be explained by Figure 6 obtained from the measured data. During a hoisting cycle, the front tie was subjected to two stress cycles of a small one within a large one. The small stress cycle occurred while picking up material from...
3.2.2 Cumulative hoisting cycles

To evaluate the remaining fatigue life of the unloader, the cumulative hoisting cycles during the past 23 years should be known. Based on the unloaded material weight corresponding to the recorded hoisting cycles, the average discharged material weights of the grab were calculated as 19.83 t, 20.87 t and 13.0 t for iron ore, coal and stone, respectively. Further, based on the cumulative unloaded material weight, the cumulative hoisting cycles during the past 23 years were calculated as $2.29 \times 10^6$, $1.56 \times 10^6$, $0.80 \times 10^6$ for iron ore, coal and stone, respectively. The predicted hoisting cycles per year in the future were also obtained based on the data during the past 10 years.

3.2.3 Cumulative fatigue damage

The summation of fatigue damage was carried out using the Palmgren-Miner rule as

$$D = \sum \frac{n_i}{N_i}$$

where $D$ is the cumulative damage; $n_i$ is the number of cycles corresponding to the stress range $\Delta \sigma_i$, which can be obtained from real stress range histograms; and $N_i$ is the total number of cycles to failure at the constant stress range $\Delta \sigma_i$, which can be obtained from the fatigue design curves. Theoretically, the fracture occurs when $D$ sums up to 1.0. Based on Palmgren-Miner rule, the equivalent stress range $\Delta \sigma_e$ was calculated for a given number of occurrences to cause the same damage as the recorded stress range histogram. It was considered that the numbers of occurrences for stress ranges at measured points were quite different during the same period of time. Consequently, this paper used the number of hoisting cycles to calculate the corresponding equivalent stress ranges for all gages. For example, for recorded stress range histogram of Fig. 5, the cumulative fatigue damage for gage 3 was $8.54 \times 10^{-4}$, and the equivalent stress range was further calculated as 99.8 MPa corresponding to the hoisting cycles of 1,719. The equivalent stress range was used to obtain the total number of hoisting cycles to failure from the design fatigue curves.

3.2.4 Fatigue life

Tables 2 and 3 show the results of fatigue damage for two different end pins: one free to rotate and the other having the most serious problem with rotation. For the pin free to rotate, the cumulative damage during the past 23 years was 0.507, therefore requiring more 21 years to sum up to 1.0. For the pin not free to rotate, the cumulative damage during one year of monitoring was 0.335. Because the unloader was never subjected to a measurement of stress, there is no information about the front ties in terms of when the unexpected bending moment around the end pins occurred. Nevertheless, there is no doubt that the cumulative damage sums up to 1.0 in three years. Before repair, the maxi-
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Minimum inspection intervals for fatigue cracks should be assessed by crack growth rate based on the fracture mechanics.

4. CONCLUSIONS

Based on the results of the field tests and numerical analyses on a 23-year-old grab ship unloader, it is given that the front tie members have some safety problems:

1. The maximum calculated edge stress of section A for load combination A (working without wind) is 300 MPa, which far exceeds the maximum permissible value of 230 MPa and is quite close to the yield strength of 345 MPa. It is evident that the front tie fails in the strength check.

2. There are 6 end pins which are not free to rotate, inducing the occurrence of unexpected moment. The moment causes an increase in stress which, in turn, damages the safety of the front ties. This problem could arise from spherical plain bearing due to insufficient lubrication, wear, friction and deformation during the past 23 years. These pins need to be re-examined in detail.

3. The discharged material weight of the grab when fully loaded can be up to 23% overload and can also lead to an increase in stress.

4. During a hoisting cycle, the front tie is subjected to two stress cycles, so the number of occurrences for the stress range being above the fatigue limit could be 1.91 times the number of recorded hoisting cycles.

Therefore, based on the long-term recorded data, the results of cycle counting can indeed reflect the real fatigue response.

5. The cumulative damage during past 23 years is 0.507 at the pin that was free to rotate, and it would need more 21 years to sum up to 1.0. However, the value during one year of monitoring is 0.335 at the pin that had most serious problem with rotation, and it would sum up to 1.0 in only three years. Before repair, much attention should be given to the inspection of fatigue cracks.

6. The cross section is suggested as box-type or I-shape with major axis parallel to the rotational axis of the end pin to ensure adequate section modulus to resist lateral bending moment induced by the end pin when it has some problems with free rotation.

REFERENCES


Table 2  Fatigue damage for the pin rotating without problems

<table>
<thead>
<tr>
<th>Unloaded material</th>
<th>Equivalent stress ranges ∆σ_e(MPa)</th>
<th>Total number of cycles to failure N_i</th>
<th>Accumulative value during the past 23 years</th>
<th>Predicted value for one year</th>
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</thead>
<tbody>
<tr>
<td>Iron ore</td>
<td>66.1</td>
<td>8.62×10^6</td>
<td>2,292,679</td>
<td>163,291 0.0190</td>
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<tr>
<td>Coal</td>
<td>69.4</td>
<td>6.75×10^6</td>
<td>1,556,009</td>
<td>24,145 0.0036</td>
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<tr>
<td>Stone</td>
<td>42.7</td>
<td>7.66×10^7</td>
<td>798,763</td>
<td>36,727 0.0005</td>
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<tr>
<td>Total</td>
<td></td>
<td></td>
<td>4,647,451</td>
<td>224,163 0.0231</td>
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</tbody>
</table>

Note:  
*1 Based on the recorded data during past 10 years.

Table 3  Fatigue damage for the pin having problems with rotation

<table>
<thead>
<tr>
<th>Unloaded material</th>
<th>Equivalent stress ranges ∆σ_e(MPa)</th>
<th>Total number of cycles to failure N_i</th>
<th>Predicted value for one year</th>
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</thead>
<tbody>
<tr>
<td>Iron ore</td>
<td>150.4</td>
<td>5.88×10^5</td>
<td>163,291 0.278</td>
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<tr>
<td>Coal</td>
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<td>24,145 0.036</td>
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<tr>
<td>Stone</td>
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<td>1.69×10^6</td>
<td>36,727 0.021</td>
</tr>
<tr>
<td>Total</td>
<td></td>
<td></td>
<td>224,163 0.335</td>
</tr>
</tbody>
</table>

Note:  
*1 Based on the recorded data during past 10 years.
