Research paper

IMPROVEMENTS OF THE CONVEYING MACHINERY IN THERMAL POWER PLANTS – CASE STUDIES

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Received (25.02.2011); Revised (21.03.2011); Accepted (28.03.2011)

Abstract: Bucket wheel boom (BWB) hoisting systems are the vital parts of the bucket wheel excavators (BWE). Failures of the mentioned systems are inevitably cause the BWE collapse followed by a very huge financial loss. This paper discusses the results of the investigation of the stress states of the BWB hoisting system connecting eye-plate. The redesign solution for the connecting eye-plate reconstruction has been chosen based on the comparative stress analysis of alternative solutions. The adopted solution satisfies all design restrictions and strength criterion. Besides that, the stress level is lower for \(\approx 20\%\) compared to the original connecting eye-plate design. After installing the redesigned connecting eye-plates as well as during the operation of the BWE no failures have occurred.

Keywords: bucket wheel boom, hoisting system, connecting eye-plate, FEM, redesign

1. INTRODUCTION

The role of conveying and material handling machines in thermal power plants is twofold. Some of them are used to ensure the supply of boiler fuel – machines and systems for storage and transportation of coal, while others are used during maintenance and repair of the facilities elements – bridge and jib cranes and monorails. Regardless of the purpose, it is necessary that the considered material handling and conveying machines have a high operational readiness. Their eventual failures are dramatically reflected in the production of electricity and cause high financial losses. Following the paper there are examples of improving the design of such machinery in the system of thermal power plan TE „Nikola Tesla“ – Serbia.

2. CASE 1 – BRIDGE CRLJENI

Four unloading bridges with elevators, Fig. 1, are the backbone of the system of coal delivery in the power plant „Kolubara“ Veliki Crljeni [1-4]. During extreme skewing occurs the buckling of rigid legs braces, which resulted in the collapse of the structure of the unloading bridge.

During skewing, the highest calculated stresses occur in the rigid leg braces - in tight brace kN/cm² 16.7, and 17.7 in compressed brace kN/cm², Fig. 2. Maximum displacement occurs in the structure of the shear leg, Fig. 3.

In order to eliminate the causes of failure of supporting structure of unloading bridge, the appropriate reconstruction has been done. Its fundament is installation of diagonals into the existing brace truss and installation of truss that connects (and also stiffen) the brace truss with main girders rigid portal, Fig. 4.
By the reconstruction of structure is achieved a significantly lower level of stress field, with simultaneous dislocation of the zone of maximum stress, Fig. 5. Maximum stresses occur in the elements of vertical truss (11.3 kN/cm² tensile stress) and the main girder (compressive stress kN/cm² 12.8). Stresses in rigid leg braces are much lower than the projected state - the maximum compressive stress is 8.6 kN/cm². Increasing the stiffness of the rigid portal truss, with simultaneous increasing the torsional stiffness of its legs, results in less displacement of the reconstructed structures in the horizontal plane - displacement of the reference node 66 in the direction of axis X of the global coordinate system (direction of crane runway) is 13.2 cm.

3. CASE 2 – BRIDGE CRANE IN THERMAL POWER PLANT “KOLUBARA”

Bridge crane (manufacturer "Lola” Železnik) that serves a mechanical hall, Fig. 6, was reconstructed in 1959. Then the lifting capacity of the main hoist drive was increased from 75 t to 110 t.

Moving trolley ("cat") operates on the runway mounted on the main girders upper, Fig. 7. On them were posted mechanisms of the main and auxiliary hoist, Fig. 8 and 9.
The existing mechanism for auxiliary hoist, Fig. 9, is realized as a kinematic chain consisting of:

- slip-ring motor (1), power $P_{EM} = 22$ kW, number of speed $n_{EM} = 730$ min⁻¹;
- coupling with brake (2);
- two-stage gear reducer (3), gear ratio $i_R = 16,1$;
- outdoor gear set (4), number of teeth of the pinion $z_1 = 18$, number of teeth of high gear $z_2 = 108$, gear ratio $i_Z = i_2/i_1 = 108/16 = 6$;
- drum (5), kinematic diameter $D_D = 475$ mm;
- rope (6), diameter 24 mm;
- lower sheaves of lifting tackle with two sheaves and a hook (7);
- compensating sheave (8).

The lifting tackle setup is done as fourth-part double (double reeving – two ropes coming from the drum), while a compensating sheave is used to equalize the rope parts length.

<table>
<thead>
<tr>
<th>Value</th>
<th>Designation or formula</th>
<th>Intensity</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lifting capacity</td>
<td>$Q$</td>
<td>15 t</td>
</tr>
<tr>
<td>Efficiency of the mechanism</td>
<td>$\eta_M$</td>
<td>0.85</td>
</tr>
<tr>
<td>No. of rope legs on which the load is suspended above tackle</td>
<td>$m$</td>
<td>4</td>
</tr>
<tr>
<td>Mechanical advantage of the lifting tackle</td>
<td>$i_K = \frac{m}{2}$</td>
<td>2</td>
</tr>
<tr>
<td>Drum speed</td>
<td>$n_D = \frac{n_{EM}}{i_Ri_Z}$</td>
<td>7.56 min⁻¹</td>
</tr>
<tr>
<td>Drum circumferential speed</td>
<td>$v_D = \pi D_D n_D$</td>
<td>11.28 m/min</td>
</tr>
<tr>
<td>Main hoist speed</td>
<td>$v_{DIZ} = \frac{v_D}{i_K}$</td>
<td>5.64 m/min</td>
</tr>
<tr>
<td>Required hoist power</td>
<td>$P_{DIZ} = \frac{Qgv_{DIZ}}{\eta_M}$</td>
<td>16,273 W</td>
</tr>
</tbody>
</table>

Given the fact that $P_{DIZ} = 16.3$ kW < $P_E = 22$ kW, it is concluded that the selection of electric motor of the auxiliary hoist drive is properly carried out. However, during operation came the failure of the two-stage gear, position 3, Fig. 9 (a). This is why the reconstruction of the auxiliary hoist is made in 2010 [5], in order to achieve the required parameters of auxiliary hoist - lifting capacity 15 tons, lifting height 14.1 m, the main hoisting speed 5 ... 6 m / min and auxiliary hoisting speed 0.5 ... 0.8 m / min. It consisted of the following:

- from kinematic chain is removed existing electric motor (1), flexible coupling with a brake (2), gear
reducer (3) and pinion of the outdoor set (4) with countershaft;
- it is built in a compact motoreducer (4) with feet and brake, Fig. 10, on whose output shaft is mounted newly built pinion (2) which is coupled with a large hoisting drum gear;
- on the existing trolley frame is fitted metal girder (1) on which is mounted and connected by screws motoreducer (4); on the new motoreducer girder are posted screws (10) on wafers in order to enable a fine lateral motion of motoreducer, i.e., axial distance of the outdoor gear pair, Fig. 10;
- the frequency regulator of the electric motor speed is incorporated in the ratio 1 : 10 (5 Hz : 50 Hz), so that is accomplished a fine hoisting speed \( v_F = 0.1xv_{HOST} \).

Fig. 10. Redesigned auxiliary hoisting mechanism (main parts): 1 – girder; 2 – pinion; 4 – motoreducer 7,8,9 – screw, nut, washer; 10,11 – bolts to adjust the axial distance of the outdoor gear

4. CASE 3 – JIB CRANE ON THE BOILER CROWN IN THE THERMAL POWER PLANT “NIKOLA TESLA” UNIT A4

Arm of the jib crane boom is built on the crown of the boiler unit A4 at elevation 83.0 m, Fig. 11 [6]. It is made of standard rolled sections and tubes, made of material with quality S235. Connections between structural elements have been carried out by welding. Upper support is executed as a radial bearing, and the bottom support, as radiaxial bearing. The arm outreach is 8270 mm.

The projected crane lifting capacity was 3,2 t. In order to enable lifting weight of 5,0 tons, it was necessary to strengthen the bottom - compressed structure chord. Numerical validation of the redesigned structure was based on FEM. The structure is modeled by linear finite element of the rod and the beam type, Fig. 12. So truss chords are considered as beam-type elements, while inner elements are modeled as rod-type elements. Reactions of the supports are determined by using boundary elements.

Fig. 11. Structure of the jib crane

Fig. 12. Finite element model of the jib crane structure

Maximum nodal displacement of the basket structure is 1,23 cm, Figs. 13 and 14.
The field of maximum normal stresses (kN/cm²) of the arm is shown in Fig. 15. The highest tensile stress is 10.5 kN/cm² and is lower than the allowed one (16.0 kN/cm²). The maximum value compressive stress is also lower than the permissible value. In addition, the redesigned lower chord of the arm meets the criterion of elastic stability.

5. CONCLUSION

According to code DIN 22261-2, the impact of the hole on the connecting plate stress-state is taken into account by the factorization of the analytically calculated stress values. The application of FEM, which presents the inevitable stage in the redesign of support structure’s vital parts and mechanisms of the BWE, enables more precise identification of the connecting eye-plate stress field. In the presented case, the comparative stress analysis of the alternative design solutions is done based on the finite element analysis results. The adopted solution satisfies all designing restrictions and strength criterion. Besides that, the stress level is lower for ≈ 20% in comparison to the original connecting eye-plate design. After installing the redesigned connecting eye-plates and during the operation of the BWE no failures have occurred. That finally and unquestionably verifies the results obtained in this paper.

ACKNOWLEDGMENTS

This work is a contribution to the Ministry of Science and Technological Development of Serbia funded project TR 35006.

REFERENCES


