Nonlinear Dynamic Analysis of a Polar Crane

S. Bolourchi, L.E. Malik

URS/Blume Engineers, 130 Jessie Str., San Francisco, California 94105, U.S.A.

ABSTRACT

Linear and nonlinear responses of a containment building polar crane to three orthogonal and simultaneously applied input acceleration time histories are compared. The nonlinearities included are the effects of wheels lifting off the rail and the subsequent impact, as well as cable snapping. It is shown that the linear analysis responses are not always conservative, because they neglect the effects of impact loading which can substantially increase the dynamic responses.

1. INTRODUCTION

The importance of performing nonlinear analysis for pipe structures subjected to impact loading, such as pipe-whip problems, is well established. The objective of this paper is to demonstrate the significance of including the nonlinearity in the analysis of crane structures subjected to seismic loads. The crane analyzed in this study is a massive polar crane which is similar to other PWR containment cranes.

The dynamic analysis addresses the nonlinear behavior of the cable, which can only sustain tension, and the wheels, which can lift off the rail and proceed with subsequent impact.

Response time histories for the linear and nonlinear models were calculated and compared. Both analyses were performed for the identical and simultaneously applied three orthogonal acceleration time-history inputs. The nonlinear responses, to be reported herein, clearly show the dynamic effects of cable snap and wheel truck uplift and subsequent impact on the response of the crane as well as on its dynamic characteristics. A comparison of the responses from the linear and nonlinear analyses shows significant differences in both the amplitudes and frequency content of the response time histories.

2. DESCRIPTION OF THE POLAR CRANE AND MODELLING PROCEDURE

The polar crane is a gantry crane with trolley; it consists primarily of welded box girders and moment-resisting bolted connections. The crane is 65 ft tall and 120 ft long overall, spanning 103 ft between rails in the longitudinal direction with one end of the upper girders cantilevered 13-1/2 ft beyond the legs. The crane travels over a 103-ft-diameter rail by means of four trucks. The rail is placed over a 50-ft-tall cylindrical concrete wall.

- 429 -
The crane nonlinear model, Figure 1, consists of three-dimensional beam elements representing gantry legs, bridge girders, etc. At connections, rigid beam elements are introduced to model the connection between the centerline of the members to the edge of connection. An equivalent three-dimensional beam element is used to model the truck assembly stiffness. The cable is modeled by a nonlinear truss element which buckles under compression. Nonlinear gap elements connect the equivalent beam elements representing the truck assembly to the rail. The gap elements have linear stiffness in compression and zero stiffness in tension. Thus, the model has no restraint at the base of the legs where their trucks uplift off the rail.

The linear model of the crane is identical to the nonlinear model except for the cable, which is modeled by a linear truss element which has identical stiffness in tension and compression, and the base of the model, which assumed the truck wheels to be pinned to the rail.

3. ANALYTICAL PROCEDURE

The linear and nonlinear analyses were performed for the same three orthogonal acceleration time histories applied simultaneously at the base of the crane. The three 20-second-duration time histories input were developed so that they would: (a) have response spectra which closely approximates the input design spectra, and (b) have a low correlation function to approximate statistical independence.

The linear elastic analysis used the SAP IV computer program [1], and is based on a modal superposition time-history analysis. Modal damping of 4.5% of critical was used for all modes, and an integration time step of .005 seconds was employed in this analysis.

The nonlinear analysis used the ANSYS I computer program [2] which employs a Newton-Raphson direct step-by-step time integration formulation. Convergence at each time step is achieved through iteration over equilibrium of member forces and reformation of the stiffness matrix when necessary. An integration time step of .005 seconds was used in this analysis. The stiffness and mass proportional damping was employed such that an equivalent modal damping of 4.5% at the dominant frequencies of the crane was produced.

4. DISCUSSION OF RESULTS

Displacement, acceleration, and member force response time histories from the linear and nonlinear analyses were calculated and compared. The results show that the nonlinearity at the supports influences the dynamic behavior as well as the amplitude of the responses of the crane, while the cable snap primarily influences the amplitude of the response of the trolley and crane bridge girders.

Figure 2 shows the vertical relative displacement response time history at nodes A and B (see Figure 1) for the linear and nonlinear analyses. The linear analysis gives very small vertical relative displacements at a response frequency of about 2.75 Hz. The principal mode contributing to this response is a cantilever mode typical of a frame pinned at the base. The nonlinear vertical relative displacement response shows a significant increase in upward displacements, while downward displacements remain about the same as those from the linear analysis. The upward displacements occur at a frequency of about 1.2 Hz. Furthermore, upward vertical displacement responses of nodes A and B are 180° out of phase, which indicates a rocking mode.
These results indicate that the effects of including the nonlinear boundary condition of the wheel uplift and the subsequent impact is to change the transverse dynamic behavior of the crane to a predominantly rocking mode of lower frequency compared to a higher frequency bending mode for the pinned supports (wheels restrained from lifting off the rail). Figure 3 shows the vertical acceleration response time history for the linear and nonlinear analyses. These results are consistent with the displacement results. These results also show the very high-amplitude, high-frequency acceleration pulses in the nonlinear analysis results due to impact of the wheels on the rail. These pulses occur at the end of an uplift cycle, shown in Figure 3b, and their high frequency and quick decay is characteristic of impact loading.

Figure 4 shows the vertical relative displacement response time history at node C (see Figure 1) for the linear and nonlinear analyses. The linear response is essentially symmetric about the displacement due to gravity with a response frequency of about 3 Hz. This is to be expected from a fairly symmetric structure and a linear modeling of the cable. The vertical displacement response from the nonlinear analysis is more complex to describe. The amplitude of the displacements from the nonlinear analysis are higher than from the linear analysis due to the effects of rocking of the legs described above and due to cable snap. The upward displacements are primarily controlled by the rocking of the crane legs as the cable buckles during upward motion of the load and does not contribute to the upward vertical displacement of node C. The frequency of these upward displacement responses is the same as at nodes A and B (about 1.2 Hz), and they coincide in time with upward displacement pulses at node A. The downward displacement responses of node C are determined by a combination of cable snap and bridge girder dynamic response.

It was also observed that the linear acceleration responses at node C, Figure 5, were fairly symmetrical with a maximum vertical acceleration of about 1.8g. The nonlinear results have a significantly higher amplitude of about 4.0g which is primarily due to cable snap. These high pulses decay quickly, as is expected of impact type loading.

5. CONCLUSIONS

The results of a comparison of a linear analysis of a massive polar crane with a nonlinear analysis which addresses nonlinearities at the boundary conditions due to wheel uplift and the subsequent impact on the rail and also nonlinearity in the cable due to its buckling under compression indicate the following:

a. Nonlinear analyses may yield significantly higher responses due to impact loading and changes in dynamic characteristics of the structure.

b. The dynamic behavior of the crane may be modified by the nonlinear boundary conditions. In the crane reported herein, the fundamental mode in the transverse direction changed from a bending-type mode to a rocking mode.

c. Boundary nonlinearities due to wheel uplift have a global effect on the responses of the crane, whereas cable snap primarily affects the trolley and bridge girders. This observation is true for the trolley placed at center span. In this configuration, cable snap effects are isolated from the legs due to the relative flexibility of the bridge girder when compared to the stiffness of the legs. Cable snap effects on the legs could be different if the trolley was to be placed at one end of the bridge girder.
ACKNOWLEDGEMENT

The authors wish to thank URS/John A. Blume & Associates, Engineers, for their support. Contribution of Johnson K. Wong and Ahmad F. Kabir is gratefully acknowledged.

APPENDIX – REFERENCES


DIAGRAM

FIGURE 1  FINITE ELEMENT MODEL OF POLAR CRANE
FIGURE 2 RELATIVE VERTICAL DISPLACEMENT RESPONSE AT THE TOP OF CRANE LEG(S)

FIGURE 3 VERTICAL ABSOLUTE ACCELERATION AT THE TOP OF CRANE LEG, NODE A
FIGURE 4  RELATIVE VERTICAL DISPLACEMENT AT THE CENTER OF CRANE GIRDER, NODE C

FIGURE 5  VERTICAL ABSOLUTE ACCELERATION AT THE CENTER OF CRANE GIRDER, NODE C