# Seismic Response of a Sliding Polar Crane for a Nuclear Power Plant

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ABSTRACT
A seismic analysis of a design concept which allows a Nuclear Power Plant polar crane to undergo sliding when subjected to seismic vibrations is described. Sliding of the crane system reduces the magnitudes of stresses that would be developed in a restrained bridge crane system, allowing a more economical and practical design. The analysis, herein, predicts the values of the relatively higher displacements and velocities that are expected to occur which replace the higher stresses of a restrained design. The magnitude of these values provide design criteria to assure that sliding permitted does not lead to overall instability or collapse of the system.

In the analysis, the bridge crane design is mathematically modeled in the vertical and lateral directions. The bridge crane system is postulated to vibrate in a linear-elastic fashion, until the dynamic reactions occurring at the crane wheel/support interface exceed the available resisting friction, at which time sliding is initiated. Sliding is postulated to continue until the relative velocity of the crane and supporting structure is zero, at which time a linear-elastic vibration mode is again developed.

The analysis considers the variation in static and dynamic coefficients of friction and the variation of available friction resistance due to the crane vertical response. The initiation of sliding is modeled as an instantaneous event requiring a redescription of the crane system physical properties and coordinate system. Transfer from the vibrating system to the sliding system is governed by maintaining conservation of energy.

Seismic excitation is defined using design floor response spectra appropriate for the crane system location. The design spectra are decomposed into a spectrum of acceleration time history harmonic motions which, when applied to a spectrum of single degree-of-freedom damped spring-mass oscillators, redevelops the original design spectra. The spectrum of acceleration time histories is used as base excitation to the mathematical model.

Analytical results include sliding displacements and velocities, number of times sliding occurs, cumulative sliding displacements, and system kinetic and potential energy. A description of the crane system configuration and the development of the effective mass and stiffness values used in the analysis of the vibrating and sliding systems is presented. The equations of motions coupling the horizontal and vertical responses during the vibrating and sliding phases are presented. A discussion evaluating the applicability of the results, and how the results can be used for design, is also presented.

## 1.0 INTRODUCTION

This report describes the development of a computer analysis method for design of NPP bridge crane systems. Attention is focused upon the effect that sliding between the crane wheels and rail has upon the seismic response of the crane system.

### 1.1 Analytical Approach

The dynamic analysis consists of the development of mathematical models representative of the structural behavior of the crane system when vibrating or sliding and the differential equations of motion connecting the model with the input loading and subsequent model response. The dynamic models consist of coupled single-degree-of-freedom (SDOF) spring-mass models, representative of the horizontal and vertical structural characteristics of a bridge crane system. Non-linear differential equations allow representation of crane system sliding, structural yielding when sliding becomes excessive, and for variation in static and sliding friction due to the vertical response of the system. The use of relatively simplistic models to represent the crane system response is valid due to the large degree of uncertainty associated with non-linear sliding mechanisms. In such cases, the dynamic system response is more sensitive upon the definition of these mechanisms than upon the number of degrees of freedom associated with the dynamic model. Also, bridge crane systems generally have well defined dominant single frequency responses in the vertical and horizontal directions, and higher modes contribute relatively little additional response.

## 2.0 DESCRIPTION OF CRANE SYSTEM

The bridge crane system considered in this study is typical in layout for many power plant designs, although details of hold-down features and restraints may vary greatly. A sketch showing a plan and side view of the system is shown in Figure 1.

### 2.1 Crane System Layout

The polar crane is supported by and travels upon a 40 M diameter crane rail. As shown in the sketch, the polar crane system consists of the following major components:

a. 2 bridge girders, approximately 40M long, set parallel to each other and 10 M apart.

b. a trolley hoist, spanning between the bridge girders and traveling along the length of the girders.

c. 2 wheel bogies, one at each end of the girders, that structurally tie the ends of the girders together and contain the crane system drive motors, bridge wheels, and wheel brakes.

#### 2.1.1 Crane System Restraint and Wheel/Rail Interface

In this study, one end of the crane is longitudinally and laterally restrained while the opposite end of the bridge system has no restraint. The crane wheels do not have flanges. The effect of the end guide restraint allows for out-of-roundness of the 40 M circular rail, as well as for thermal expansion effects of the bridge and rail systems.

## 3.0 GEOMETRIC IDEALIZATION

The following describes the sequence of motions postulated to occur:

a. System vibrates laterally and vertically as a pin-ended beam. Both the horizontal and vertical models vibrate in a linear, elastic fashion. The vibration frequencies are dependent upon the material and geometric characteristics of the system, and upon the system configuration such as the trolley hoist location and weight of the lifted load. The effect of the vertical vibration is to increase and decrease the effect of the normal static weight on the crane wheel/rail interface. The net normal weight determines the amount of static friction (dynamic friction if the horizontal system is sliding) available to resist system sliding. In this manner the vertical and horizontal models are coupled. The longitudinal motion is assumed as being rigidly fixed. This is not a severe restriction and longitudinal effects can be included when the system properties are developed for such a case, and would be included in a complete analysis.

b. System vibrations continue, building up increased reactions and reduced available static friction. The seismic reactions for both the vertical and horizontal vibrating systems are dependent upon the duration, magnitude and frequency content of the seismic loading. When sufficiently severe loadings occur, the vibrating system reactions increase with loading duration. In the vertical system, the reactions alternately increase and decrease the effective weight on the crane rail. In the horizontal system, reactions are built up in a similar manner. The computer analysis continuously compares the horizontal model reactions with the amount of static friction available as determined from the vertical model.

c. Horizontal model reactions exceed available static friction forces and sliding occurs. The horizontal system deformation approximates a horizontally swinging cantilever, restrained at the end having the guide rollers and sliding on the free end. Since the restrained end is still attached to the support, the dynamic loading may continue to excite the sliding system. Sliding friction forces developed have a similar effect as damping forces by providing a means of losing system energy. The sliding frictional forces are still dependent upon the reaction of the vertical vibrating system.

d. Sliding stops, and pin-ended beam vibrations begin. As the system slides, energy is lost through frictional resistance forces. Also, the cantilever system spring stiffness forces may increase to resist continued sliding motion. At some point, the velocity of the sliding system relative to its support becomes zero, and sliding is stopped. The system then once again approximates a vibrating pin-ended beam. The system may now be in a displaced position and may have an effective pre-loading which tends for the system to slide back toward the original undisplaced position.

For strong, continued, vibrational input, the system may undergo several cycles, alternately vibrating and sliding.

## 4.0 DYNAMIC ANALYSIS

The differential equations of motion are solved by direct integration on a time-by-time basis. The equations of motion are general, and allow a time variation of the system mass, damping and stiffness, as well as the non-linear effects of sliding and yielding.

Both the horizontal and vertical motions of the crane system are modeled using SDOF spring-mass systems. The horizontal and vertical systems are dynamically and structurally modeled as being independent. However, the response of the vertical system is coupled into both horizontal systems through the development of applicable friction forces. Sketches of the models are shown in Figure 2.

4.1 Dynamic Models

The horizontal response is modeled using two separate models, one representing the vibrational behavior of the system, and the other the postulated sliding behavior of the system as a horizontal cantilever. The change from one model to the other occurs instantaneously, at the times that sliding is initiated or ended. Although the transition from vibration to sliding represents a discontinuity of the reference frames for system motions, continuity in the form of system energy is preserved. At the time the horizontal reactions are predicted to exceed the available friction force in the vibrating system, the kinetic (based upon total velocity) and strain energy of the vibrating system is calculated. The sliding model is given an initial relative velocity corresponding to the combined strain and kinetic energy of the vibrating model just prior to the initiation of sliding.

The vertical model remains linear and elastic, and the reactions are used to calculate the effective normal weight used in determining available friction by multipling by the static (or dynamic, when sliding) coefficient of friction. When the vertical upward response exceeds the weight of the system, uplift is calculated to occur, and zero frictional resistance is available.

4.2 Generalized Model Properties

A shape function is chosen for each of the models approximating the displacements expected to occur. The shape functions are used to develop SDOF properties equivalent to the distributed properties of the crane system. The procedure involved is well known and can be found in textbooks. A sine function is used as a shape function for the horizontal and vertical vibrating models. The sliding cantilever model uses a linear function, $=\ell/L$. Generalized properties for mass, damping, stiffness and loading input are developed for each model.

4.3 Equations of Motion

The basic generalized SDOF equation of motion is used for each model:

$$M^*\ddot{X} + C^*\dot{X} + K^*X = P^*\ddot{U}_{base} \pm F_r^*(t)$$

$F_r^*(t)$ represents the generalized dynamic friction forces introduced during sliding. $P^*$ is made to be zero in the vertical model during conditions of lift-off. The simplicity of SDOF models allows flexibility in the use of non-linear parameters.

4.5 Crane System Frequencies and Analysis Input

The crane system analyzed has a horizontal vibrating frequency of 2 Hz and a vertical frequency of 3 Hz. During sliding, discounting friction, the crane has a frequency of about 1/2 Hz. The analyses are conducted using a controlled input consisting of independent harmonic base accelerations having a constant amplitude. The input frequencies are incremented throughout the frequency range of crane system. The results of each frequency input are plotted in spectral fashion, as shown in Figures 3 through 9.

## 5.0 DISCUSSION

Parametric studies are conducted using a horizontal input of 0.20 g and an in-phase vertical input of 0.10 g. Figures 3 and 4 show spectral amplifications and maximum sliding velocities obtained for various values of static friction using horizontal input only. Figures 5 and 6 consider the effect of in-phase vertical input.

The coefficient of friction is found to be the most significant parameter affecting the sliding response of the crane system. Other parameters considered include the effect of magnitude and frequency of the base input and the effect of simultaneous vertical input. These parameters are important in predicting the initiation of sliding, but appear to be not as important in terms of affecting the actual dynamic loading the system undergoes, as is seen in comparing Figures 3 and 4 with 5 and 6. Figures 3-6 show how sliding reduces the system structural loading when compared with an elastic analysis, $f_r(\text{static})=\infty$. This is an advantage of allowing a crane system to undergo sliding. Structural design can be governed by the lifted load requirements for the crane rather than by amplified seismic responses or by de-tuning requirements.

The reduction in loading is replaced by the occurrence of large sliding displacements. Although difficult to predict accurately, the analysis indicates displacements exceeding two meters are possible. As shown in Figure 9, sliding may occur many times, introducing a possible fatigue factor. Derailment bumpers such as shown in Figure 1 restrain sliding to within acceptable limits. The most reliable design criteria for the bumper design appears to be the peak sliding velocities. As shown in Figures 4 and 6, the velocities are relatively consistent for static friction values up to 0.4. At higher friction values maximum sliding values peak out at resonance at values of about 20% greater. By using a conservative choice of velocity and crane system effective mass, on impact-momentum expression can be developed. A possible design approach is to allow the derailment bumpers to partially absorb the impact through plastic deformations. Mechanical damping mechanisms could also be used.

A complete analysis would consider various positions of the trolley hoist. Different results can be expected for systems having the trolley hoist located at either end of the bridge girder. In such cases, however, vibrational amplitude effects are expected to be less significant in initiating sliding. It may be expected that with the trolley hoist located at the restrained end, the crane system could undergo rotation during sliding, an example where structural eccentricity may be beneficial in seismic design. The most severe sliding effects may be expected to occur when the trolley hoist is located at the unrestrained end of the crane bridge.

The effect of the addition of a lifted load onto the trolley hoist is similar to an increase in the coefficient of friction. In this study, the vertical frequency of the crane system is reduced to near the horizontal frequency with the added weight. Although more lift-off occurs and sliding is more numerous, the results are similar to those shown in figures 5 and 6.

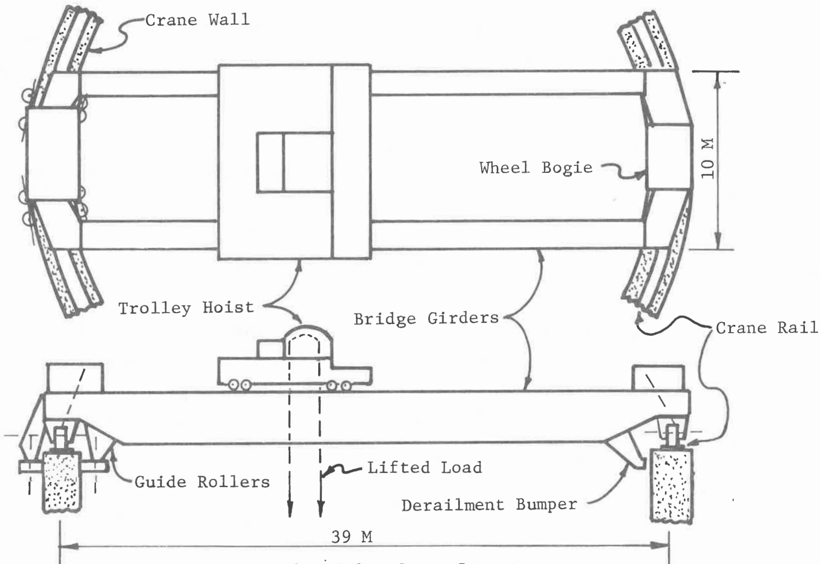


Fig. 1. Polar Crane Layout.

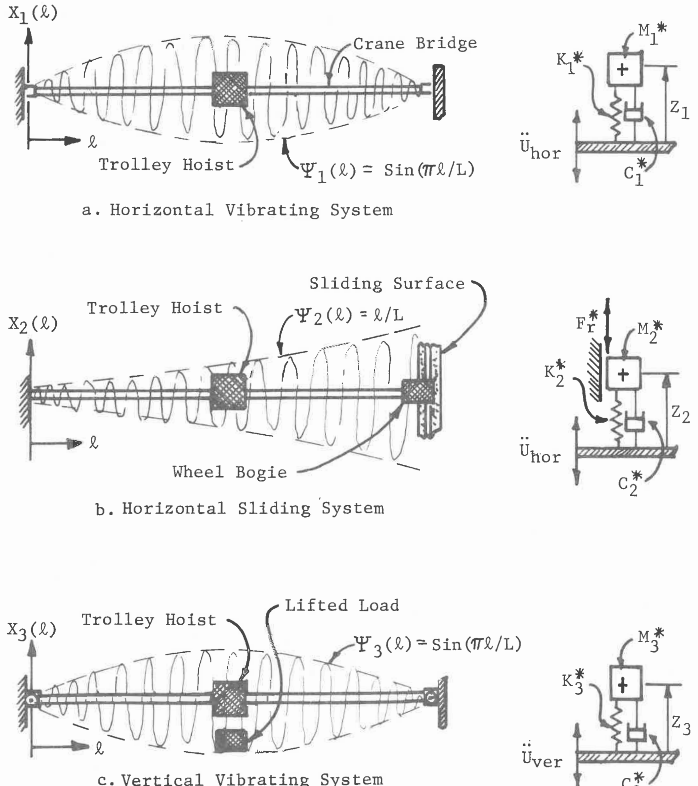


Fig. 2. Idealized Dynamic Models.

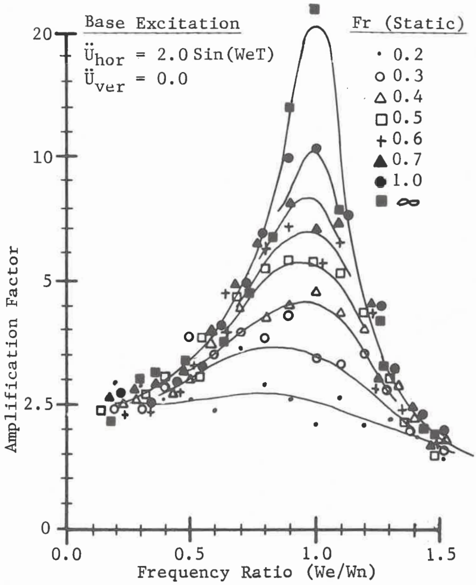


Fig. 3. Spectral Amplification for System Accelerations and Reactions for Horizontal Input Only.

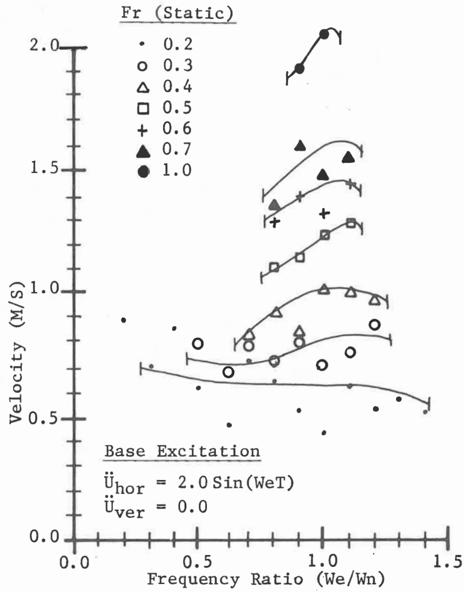


Fig. 4. Spectral Sliding Velocities for Horizontal Input Only.

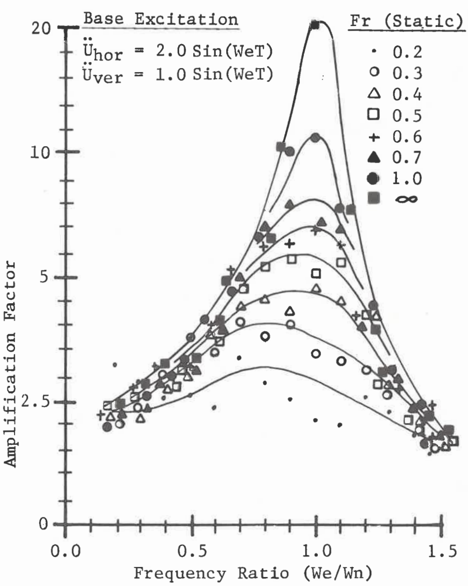


Fig. 5. Spectral Amplification for System Accelerations and Reactions for Horizontal and Vertical Input

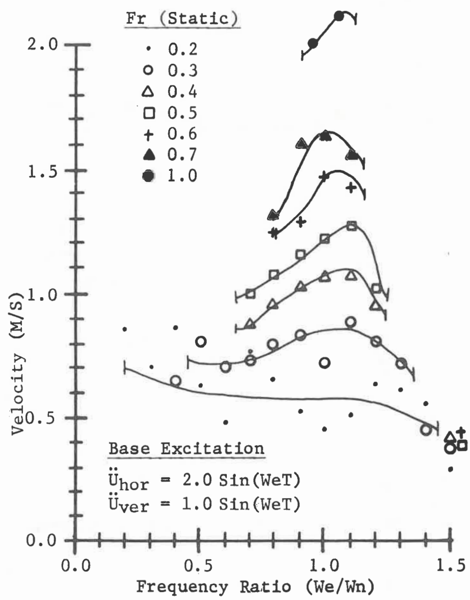


Fig. 6. Spectral Sliding Velocities for Horizontal and Vertical Input

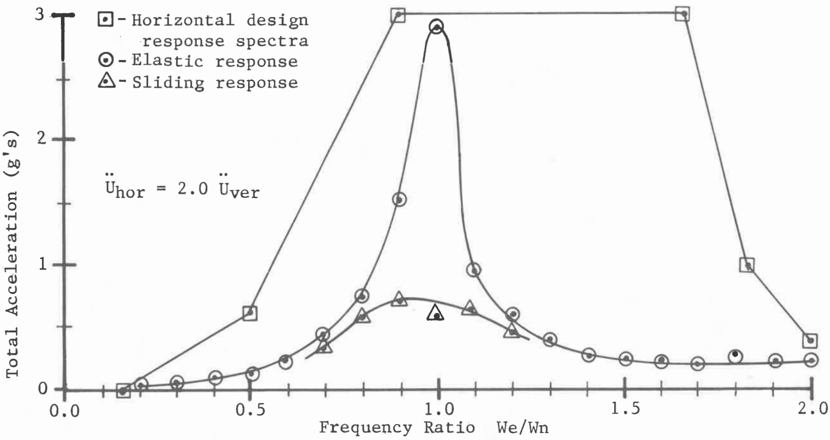


Fig. 7. Polar Crane Spectral Analysis, Acceleration Results. $f_r$(static) = 0.30

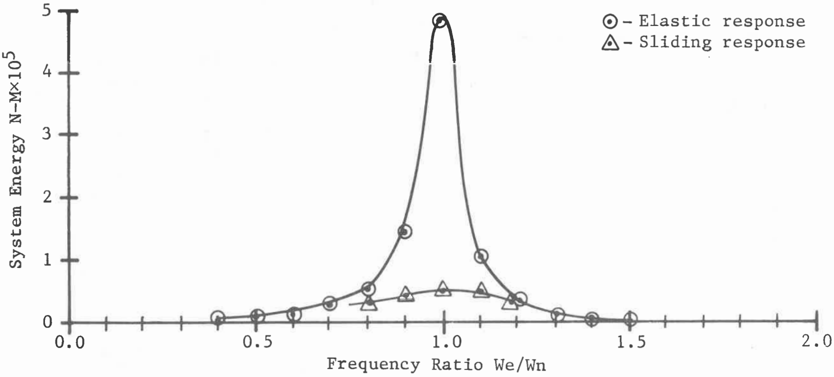


Fig. 8. Polar Crane Spectral Analysis, Maximum System Energy Developed. $f_r$(static) = 0.30

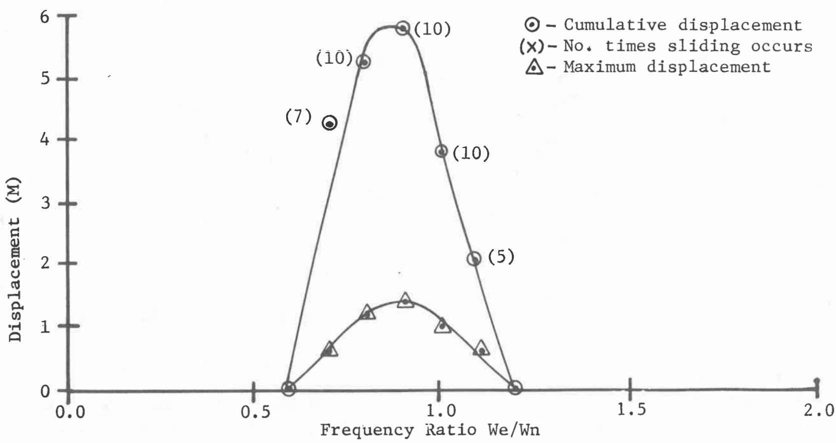


Fig. 9. Polar Crane Spectral Analysis Sliding Displacements. $f_r$(static) = 0.30