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LARGE SODIUM PUMP COASTDOWN DURING AN EARTHQUAKE

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ABSTRACT

Analyses were performed to determine the responses of the main rotary element of a large sodium pump during the simultaneous occurrence of pump coastdown and a seismic event. Analytical procedures are described which enabled reduction of a multi-degree of freedom finite element model of the pump to a representative nonlinear single degree of freedom system with retention of acceptable computational accuracy.

Pump rotor bearing impact forces and stresses, and the bearing rub forces which act throughout the pump coastdown were determined. Bearing material wear depth was calculated and an assessment was made of the effect of rub forces and the associated retarding torque on the shortening of pump coastdown time. Pump coastdown time can be very significantly shortened as a result of loss of rotor bearing stiffness and rubbing at the lower rotor speeds.

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INTRODUCTION

Large pumps which are used to circulate liquid metal in nuclear power plant heat transport systems typically utilize hydrostatic bearings at the impeller end of the rotor to support radial loads. Operating pressure for these bearings is supplied by the pump discharge pressure, therefore the bearing radial stiffness and centering force are directly related to the pump speed. Consequently, the probability of journal to bearing contact sometime during the simultaneous occurrence of a seismic event and pump coastdown is significant. An additional important consideration in the design of hydrostatic bearings for nuclear pumps can be the need to prolong the rotor coastdown during an earthquake for sufficient time to ensure adequate pump flow to the reactor such that fuel cladding temperature limits will not be exceeded before transition to natural circulation decay heat removal.

Under normal pump shutdown conditions, rubbing of the hydrostatic bearing and journal does not usually occur. However, if an earthquake initiates the shutdown or if an earthquake occurs during a normal shutdown transient, rubbing is likely to occur. The resulting frictional retarding torque could significantly hasten the pump coastdown. As a result of this concern, a study was undertaken to better understand the effects of earthquake on bearing integrity and the coastdown time for large sodium pumps and to develop simple design tools and guidelines to assure desired performance.

This paper describes some of the analytical procedures used to evaluate bearing contact stress, bearing material wear, and pump coastdown time predictions.

ANALYSIS OBJECTIVES

The primary objectives of this study were:

- . To determine the hydrostatic bearing contact stress and bearing material wear during the simultaneous occurrence of an earthquake and pump coastdown.

- . To determine the effect of bearing rub forces on the shortening of a large pump coastdown time.
- . To develop analytical procedures whereby simple one-dimensional nonlinear analytical models can be used with acceptable accuracy to predict bearing seismic responses and pump coastdown times for a range of design input parameters.

ANALYTICAL MODELS

The nonlinear characteristics of the hydrostatic bearing dictated that the response computations be carried out using time domain analysis techniques which generally require the use of a relatively large amount of computer time. This requirement, coupled with the need to perform a sufficient number of computer runs to envelope a range of values for the hydrostatic bearing input parameters at various pump speeds over the coastdown profile, necessitated the use of a relatively simple rotor bearing nonlinear spring/mass model. A comparison was made of bearing nonlinear response results obtained from analyses of a detailed pump finite element model and the simplified single degree of freedom rotor/bearing model. The comparison was sufficiently close to verify use of the simple model for the remainder of the computer runs.

Pump Finite Element Model

The pump model, typical for large liquid metal centrifugal pumps, was developed with the specific intent to focus on the response of the hydrostatic bearing and to obtain acceptable computational accuracy with the least complex model. Therefore, the finite element model which was developed for the initial phase of the study excluded axisymmetric shell elements which usually exhibit many natural modes of vibration, none of which affect the seismic response of the rotor/bearing system. The initial pump model was constructed using exclusively beam elements with lumped masses representing the structure and the contained

sodium and linear spring elements supporting the pump tank to ground and connecting the rotor to the support cylinder. A schematic of the pump finite element "stick" model is shown in Figure 1. The finite element model was developed for analysis by the ANSYS [1]* computer code.

The pump hydrostatic bearing and support stiffnesses were represented in the ANSYS finite element model by the combination of a gap element and two linear springs. One of the springs, with a stiffness equal to the support stiffness, was effective only when the gap (bearing clearance) was closed. The other spring, with a stiffness equal to the hydrostatic bearing-centered stiffness, was effective with the gap open. The hydrostatic bearing-centered stiffness was proportional to the square of the rotor speed. The hydrostatic bearing damping was proportional to the square root of the bearing stiffness and therefore directly proportional to the rotor speed.

Simplified Nonlinear Model

The hydrostatic bearing seismic loads for these type pumps were found by a number of pump analyses with detailed finite element models to result primarily from one predominant vibration mode. This mode was characterized by rocking of the pump tank and rocking of the pump rotor about the upper bearing support. Therefore, a simple single degree-of-freedom nonlinear model could be developed.

The effective mass at the hydrostatic bearing was calculated as the rotor pitch moment of inertia about the upper bearing divided by the square of the distance between the upper and hydrostatic bearings. The effective mass was found to be approximately 55% of the total rotor mass.

*Numbers in brackets designate references at the end of this paper.

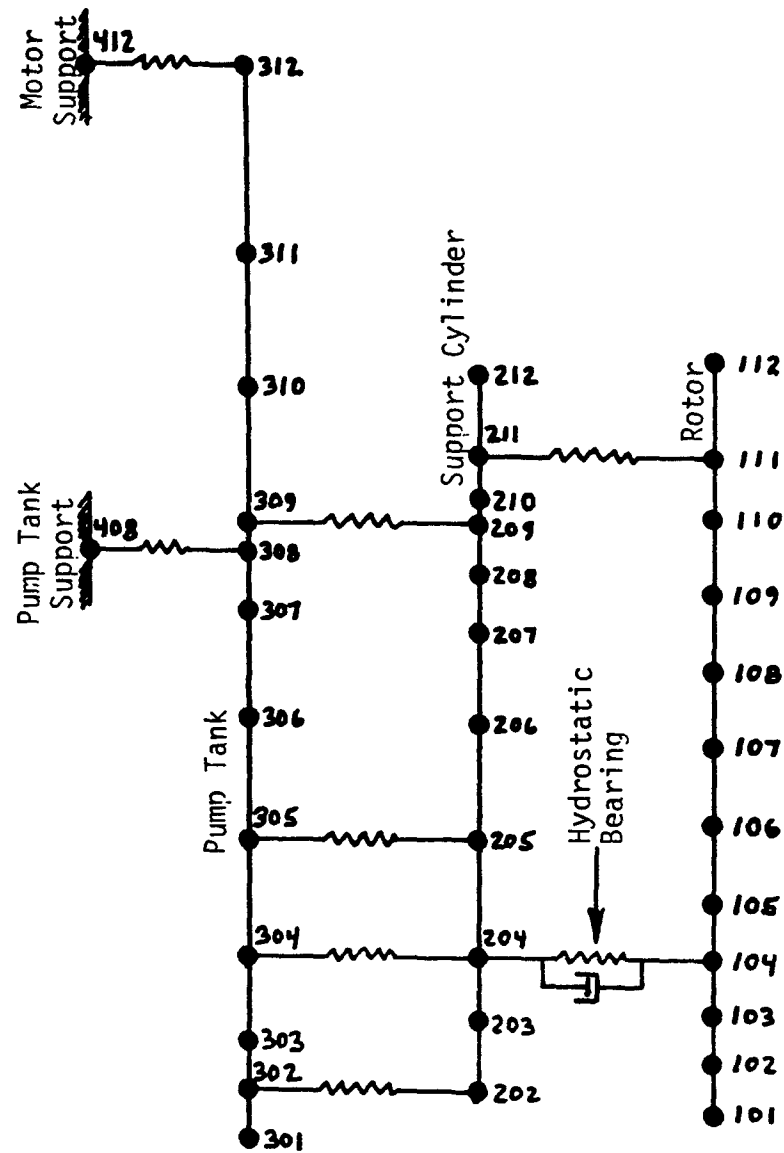


Figure 1. Typical Pump Finite Element Model Schematic

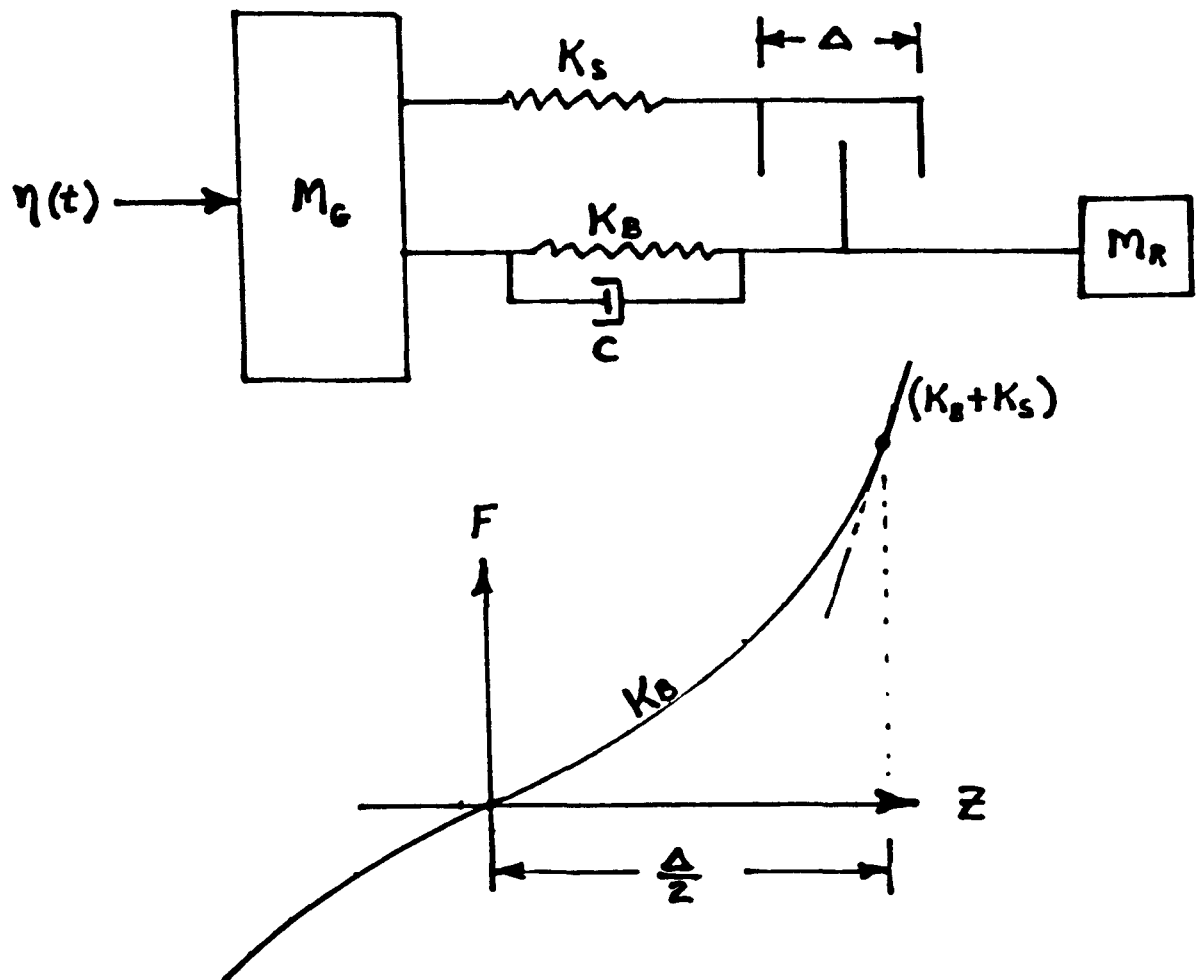
The seismic acceleration time history which was applied to the single degree-of-freedom model was first modified to account for the exclusion of the pump tank and support structure which normally act as a filter of the seismic input to the pump rotor. The original acceleration time history was applied to a linear spring and mass filter tuned to the frequency of the predominant pump tank rocking mode. The acceleration response time history from this filter was stored on file for later application to the simplified nonlinear model.

For the simplified nonlinear model, the bearing and support stiffnesses were represented by a total of 32 stepwise changes (16 in each direction) in the stiffness values. The stiffness values were set inversely proportional to the remaining journal-to-bearing clearance and they ranged from a minimum value equal to the bearing-centered stiffness to a maximum value equal to the support stiffness at journal to bearing contact. A schematic of the simplified nonlinear models is shown in Figure 2. The intent of this representation was to account for the main effect of the squeeze film stiffness by using only the displacement term of the squeeze film fluid equation and thereby avoid the added complications and computational difficulties associated with inclusion of the fluid equation acceleration and velocity terms. It is believed that the conservatism which results from the omission of these terms is slight. The bearing damping was set proportional to the square root of the localized stiffness values and therefore it is also affected by the cushioning effect of the squeeze film. A computer code, NONLIN, was developed for obtaining solutions with the simplified nonlinear method. It will be further discussed later.

ANALYSIS METHODS

Linear Seismic Analysis

Initially, seismic response and pump vibration mode analyses were accomplished using a complete pump linearized finite element model and the ANSYS computer code. During this initial phase of the pump analyses, the pump natural vibration mode frequencies and the associated mode shapes were calculated, and the predominant modes which contributed by any significant amount to the pump



$K_B \sim$ Hydrostatic Bearing Stiffness

$K_S \sim$ Added Stiffness of Support

$C \sim$ Hydrostatic Bearing Damping

$\Delta \sim$ Journal to Bearing Diametral Clearance

$M_R \sim$ Equivalent Rotor Mass

$M_G \sim$ Large Ground Mass

$\eta(t) \sim$ Filtered Seismic Acceleration Time History

Figure 2. Simplified Nonlinear Model

rotor responses were identified. It was established, during the linear analyses, that the rotor seismic response was the result of a combined pump tank and rotor rocking mode with more than 95% of the total response contributed by a single mode. The linear analyses results presented strong evidence that the pump rotor/bearing system could be represented by a simplified single degree of freedom model. More conclusive evidence was later supplied by a comparison of results obtained from time domain nonlinear ANSYS analyses of the finite element model and the NONLIN solutions based on the single degree of freedom representation.

Nonlinear Seismic Response Analysis

For analysis of the pump finite element model, all grounded nodes were tied through constraint equations to one common node point at which a very large, 10^7 lb-sec²/in, mass was placed. The seismic acceleration time history was scaled and applied as a force time history to the large mass to duplicate the original acceleration time history. The ground mass was chosen sufficiently large to preclude interaction with any of the pump structural masses, thereby preserving the pump system natural vibration frequency characteristics. The seismic response spectrum, which corresponds to the acceleration time history used in this analysis, is characterized by an 8.7 G peak acceleration from 5 Hz to 7 Hz, and by an 0.7 G zero period acceleration. The nonlinear finite element model was analyzed using ANSYS to determine the rotor bearing displacement and load responses to the applied seismic acceleration time history. Typical maximum responses are compared in Table 1 to responses obtained using the single degree of freedom nonlinear model. The satisfactory comparison of these responses provided confidence in the simplified model and in the use of the NONLIN program and the simplified rotor models for further response predictions at other pump intermediate speeds, and in assessment of response sensitivity to a range of hydrostatic bearing design input parameters.

The NONLIN computer code is a specialized adaptation of the Phase Plane Delta graphical method [2], [3] for determining successive time domain solutions of a second order differential equation. Its purpose is to provide a rapid

and inexpensive tool to determine responses of a single degree of freedom spring/mass system with stepwise nonlinear elastic restoring force coefficients, with viscous and/or coulomb damping, and acted upon by any describable force/time history. This code was developed in-house at HEDL and has been verified by comparable response computations performed by established computer codes including the ANSYS code. This computer code is ideally suited for representing multi-stepwise changes in the bearing stiffness values. It is also very effective for performing a large parametric study due to the low cost and short turn around time for running the computer program.

TABLE 1

Typical Nonlinear Response Comparison - Finite Element and Simplified Models

BEARING RESPONSE	ANSYS FINITE ELEMENT	NONLIN SIMPLIFIED	($\frac{\text{NONLIN}}{\text{ANSYS}}$)
Displacement	24.9 mil (.632 mm)	22.6 mil (.574 mm)	.908
Load	39850 lb (177253 N)	36170 lb (160884 N)	.908

Coastdown Analysis

A special computer program, COASTDOWN, was prepared to implement the time domain solutions of the following simple torque equation:

$$J\ddot{\phi} + C_0 + C_2 \dot{\phi}^2 + C_F R F(\dot{\phi}) = 0$$

where: $\dot{\phi}, \ddot{\phi}$ ~ Rotor angular velocity and acceleration
 J ~ Rotor moment of inertia
 R ~ Rotor radius
 C_0 ~ Constant drag torque
 C_2 ~ Pumping torque coefficient
 C_F ~ Friction coefficient
 $F(\dot{\phi})$ ~ Seismically induced rub forces, tabulated as a function of the rotor speed

The rotor geometry, constant drag torque and pumping torque were obtained from a pump vendor's report, resistance curves and normal coastdown profiles. Without the application of seismic effects, a typical pump coasts down from 100% design speed to stop in 120 seconds. Coefficients of friction for various bearing hardfacing materials were expected to be in the .30 to .70 range. Thus, coastdown analyses were performed covering a range of friction coefficients from .05 to 1.0. The seismically induced rub forces were determined from the nonlinear response analyses over the pump rotor speed range from 0% to 100% of design speed at speed increments of 20% and also at the 7% pony motor speed. The coastdown computer program interpolates between the values of the rub forces, $F(\phi)$, input to the program as a table.

RESULTS AND DISCUSSION

Typical results of the seismic response analyses for a liquid metal centrifugal pump are presented in Table 2. The maximum impact force and contact stress, the average rub force and rub distance, the bearing material wear depth and the normal coastdown time increments at various rotor speed ratios are listed. The maximum Hertzian stress was determined from the bearing geometry and material properties and from the maximum impact force using from [4] the following:

$$P = .798 \sqrt{\frac{F (D_1 - D_2)}{C_E \ell D_1 D_2}}$$

where: $P \sim$ Hertzian stress

$F \sim$ Impact Force

$D_1, D_2 \sim$ Bearing and Journal Diameters

$\ell \sim$ Bearing land contact length

$E_1, E_2 \sim$ Bearing and journal elasticity modulus

$\nu_1, \nu_2 \sim$ Bearing and journal Poisson's ratio

$$C_E = \frac{(1 - \nu_1^2)}{E_1} + \frac{(1 - \nu_2^2)}{E_2}$$

The rub distance and the wear depth are conservatively based on the occurrence of the earthquake over the entire maximum 120-second coastdown time. Lesser values of these parameters will result in proportion to the actual duration of the seismic event and the length of the actual coastdown which is dependent upon the friction coefficient. The wear depth was determined from the journal to bearing contact stress, the rub distance and the wear coefficient.

The typical rub forces listed at various rotor speeds in Table 2 were combined with the respective pump bearing radii and assumed coefficient of friction to predict the coastdown times. In Figure 3, the typical coastdown speed versus time profile without seismic effects is compared to the coastdown of the pump during an earthquake and assumed value of .45 for the friction coefficient.

SIGNIFICANT FINDINGS AND CONCLUSIONS

- . A comparison of nonlinear response results and pump coastdown times obtained from analyses of a pump finite element model and from a simplified single degree of freedom rotor/bearing model verifies that the simplified model can be used with acceptable accuracy for response predictions and for parametric studies in the design of a pump hydrostatic bearing.
- . The total coastdown time for the pump can be very significantly shortened as a result of loss of bearing stiffness and rubbing at lower rotor speeds.

TABLE 2

Typical Pump Rotor Seismic Response During Coastdown

RESPONSE PARAMETERS	ROTOR SPEED ~ % OF DESIGN							TOTAL COASTDOWN
	100	80	60	40	20	7	0	
Maximum Impact Force (lb)	0	0	59100	113700	163200	188100	193900	Maximum 193900
Maximum Contact Stress (psi)	0	0	3380	4690	5610	6030	6120	Maximum 6120
Average Rub Force (lb)	0	0	120	2050	7310	11240	--	Average 9460
Rub Distance (in)	0	0	10.	170.	950.	1620.	--	2750
Wear Depth (mil)	0	0	.001	.031	.201	.337	--	.569
Coastdown Time Increment (sec)	1.0	1.5	2.5	6.0	26.0	83.0	--	120.

NOTE: 1 lb = 4.448 N
1 psi = 6.894 KPa
1 in = 2.54 cm
1 mil = .0254 mm

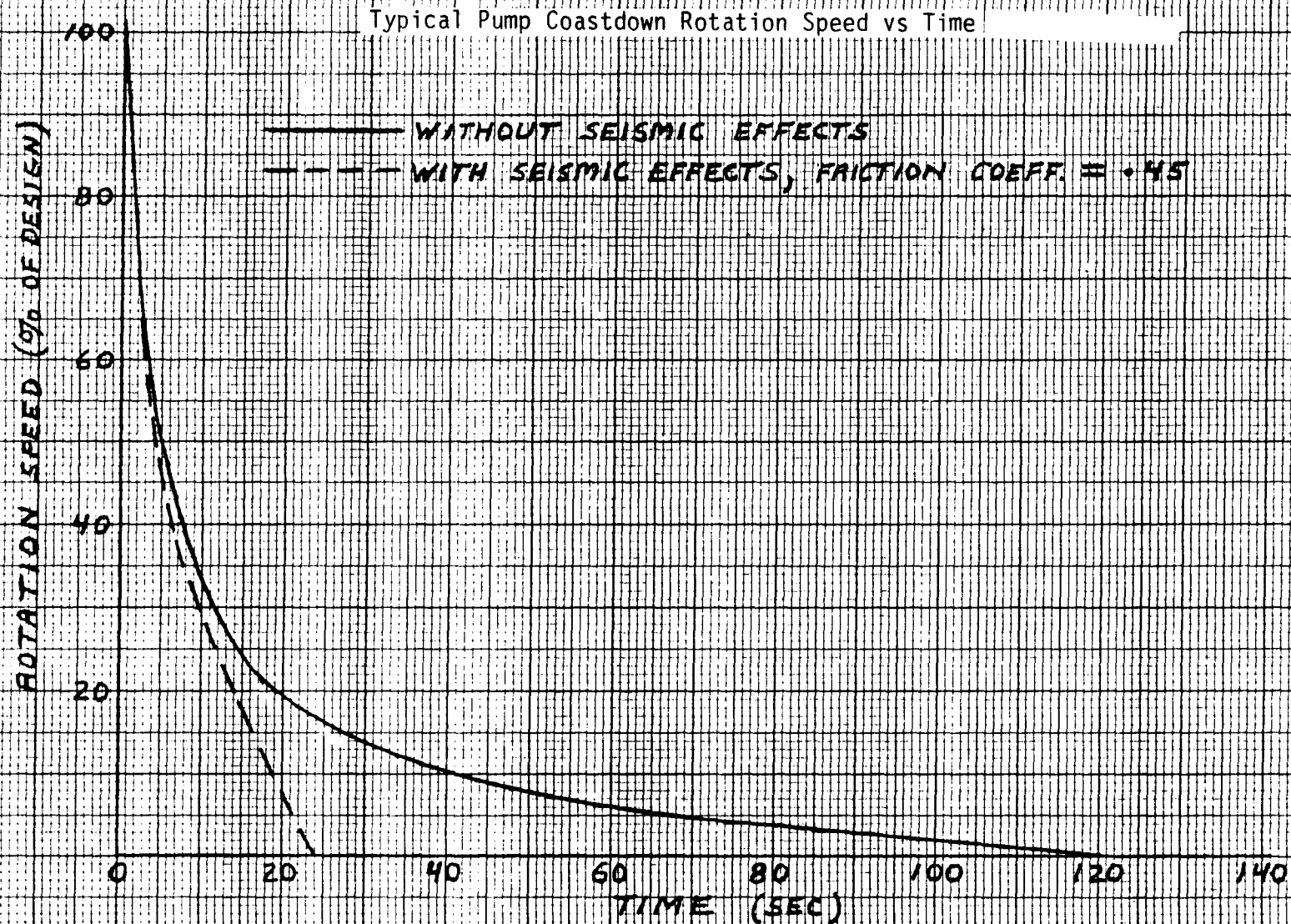


Figure 3. Typical Pump Coastdown Rotation Speed vs Time

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