Development of Advanced Rotor/Bearing Systems for Feed Water Pumps

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NOTICE

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ABSTRACT

Feed pump outages are a significant contributor to power plant forced outages and have been targeted by EPRI as one area where the application of advanced machinery technology can provide design improvements leading to increased power plant availability. Many common feed pump failures are associated with or directly caused by excessive pump vibration. A four-phase EPRI project has been undertaken to develop pump design improvements and new configurations which potentially reduce pump vibration and related failures. Phase - I of this project has been completed and is summarized in this report. The development of design improvements requires the use of advanced engineering analyses to evaluate potential reductions in vibration levels. Types of vibration which must be analyzed in feed water pumps include synchronous and non-synchronous forced vibrations, and rotor/bearing instability vibrations. Vibration analysis methods suitable for evaluation of feed water pumps were developed by upgrading and enhancing general purpose advanced machinery vibration computer codes. These analysis methods were used to study twelve-stage boiler feed pumps and the results correlated with field measurements made on these pumps. The correlation shows why these pumps have had a history of severe vibration problems. It also shows the level of engineering analysis required to study new pump configurations which have the greatest potential for reducing vibration levels.

EPRI PERSPECTIVE

PROJECT DESCRIPTION

As detailed in a recent EPRI report (Survey of Feed Pump Outages, EPRI FP-754, April, 1978), many of the common feed pump failures are associated with or directly caused by excessive pump vibration. As stated in the report, the development of advanced rotor/bearing configurations, which introduce large amounts of damping into the system, would improve pump reliability by reducing damaging vibration levels caused by uncontrolled, large dynamic rotor forces.

This project is directed toward applying advanced machinery vibration computer programs and experimental techniques to analyzing and predicting residual vibration levels of large feed water pumps using existing and new rotor/bearing and inter-stage seal configurations.

PROJECT OBJECTIVES

The overall objective of this four-phase project is to develop and demonstrate by field tests new rotor/bearing and inter-stage seal configurations that reduce pump vibration levels by providing adequate damping over the entire range of operating conditions.

Meeting the objectives of this project will be the basis for establishing tighter utility feed pump bearing/seal specifications necessary for achieving minimum vibration levels.

The objectives of Phase 1, the 12-month effort described in this report, were to (a) develop advanced engineering methods, by upgrading and restructuring general purpose machinery vibration codes, to analyze vibrations that occur in feed water pumps; (b) use this capability to analyze and predict performance of pumps experiencing vibration-related problems; and (c) validate analytic methods and results by correlation with field measurements made on these pumps.

PROJECT RESULTS

From Phase 1, the following significant conclusions are presented: (a) the vibrations of large feedwater pumps can be analyzed and predicted using state-of-the-art machinery vibration computer simulation techniques, (b) typical multistage feed pump rotor/bearing configurations lack sufficient bearing damping to control the lower frequency resonant vibration modes of the system because the bearings are located too close to the vibration nodal points, and (c) correlation of field observations and analyses demonstrated that inter-stage sealing (i.e., wear rings and labyrinth fluid annuli) can substantially influence subsynchronous damping capacity and vibration levels.

Review of the technical literature in Phase 1 revealed that fluid annulus effects on rotor vibration under feed pump conditions have not been thoroughly investigated. This lack of relevant information inhibits development of new inter-stage seal designs that could increase rather than decrease system vibration damping. It is therefore recommended that, as part of Phase 2, a comprehensive analysis (both experimental and analytical) of impeller fluid annuli be performed integrally with new rotor/bearing configurations.

It is planned, based on the successful development of improved rotor/bearing and inter-stage seal configurations in Phases 2 and 3, to involve one or more major feed pump manufacturers with electric utility companies in the design, construction, and field testing of the new configurations in order to validate the predicted vibration level reductions and reliability improvements. Project completion is expected by 1983--Phase 2 in 1980, Phase 3 in 1981, and Phase 4 in 1983.

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The authors appreciate the diligent efforts of Dr. Joseph Padovan who assisted in the job of installing the required computer codes on the University of Akron computer system. He also participated in the preparation of input data during the computer analysis work and deserves particular recognition for his use of the NASTRAN finite element computer code to model feed pump rotors.

Mr. Olaf Szamody is acknowledged for his painstaking work in obtaining all important pump dimensions of the Cane Run No. 6 feed water pumps from one disassembled in the repair shop. This was particularly important to the success of the pump vibration analyses since detail drawings of this design were not available. He also participated in taking pump vibration measurements in the field.

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Section 1

INTRODUCTION AND SUMMARY

A recent EPRI survey of feed pump outages, Reference (1), clearly shows that excessive pump vibration is responsible for many forced outages. Feed pumps are, by necessity, high-head-per-stage fluid movers and therefore add large amounts of energy to the fluid within a very confined space. The present state-of-the-art in pump hydraulic design allows this transfer of energy to be highly efficient only over a relatively narrow flow range which straddles the best efficiency point (BEP) flow capacity of a design. At off-design operating capacities, particularly at part load operation, the reduction in hydraulic efficiency is manifest in the complex secondary fluid flow patterns which are produced by the mismatch of flow-passage and impeller geometries at off-design operating conditions. Most of this lost energy is dissipated in the form of heat in the pumped fluid itself, because the complex secondary flows increase fluid turbulence and shear rate activity within the fluid. However, a small part of this dissipated energy escapes through vibration of the pump. If not adequately controlled, this produces vibration levels which are quite high, resulting in sudden failure or an accelerated wearing out of critical pump components. Seals, wear rings, bearings, shafts, securing bolts and thrust balancers are typical pump components which fall prey to excessive vibration levels. Such components must be designed to withstand some residual vibration which remains even in the "smoothest" running machines. However, present feed pump designs have frequently not been designed to minimize vibration levels. Consequently, numerous presently operating units require frequent shutdowns to replace worn out components, particularly seals. This results in reduced availability, high pump maintenance costs and a substantial spare parts investment.

The purpose of this project is to demonstrate that application of advanced machinery vibration technology to future designs can produce a marked reduction in pump vibration levels and thereby increase feed pump availability and reduce maintenance costs. The overall technical objective of this project is to identify new configurations which minimize vibration levels. Meeting this objective will be the basis for "tighter" standards and specifications for feed water pump vibration levels and will provide designers with techniques to improve their future designs to satisfy these specifications and improve pump availability.

The project is comprised of four (4) sequential phases. This report summarizes the work done on the recently completed Phase - I. The scope of all four phases is summarized below.

PHASE - I: Determine the level of engineering analysis needed to make reliable assessments of feed water pump rotor vibration levels and assemble the necessary computer codes to study advanced rotor/bearing configurations. Also, identify areas where current technology needs to be expanded.

PHASE - II: Explore new rotor/bearing configurations to determine configurations which minimize pump vibration. Also, design a test apparatus to experimentally determine static and dynamic rotor forces.

PHASE - III: Develop new design configurations or components based on the results of Phase - II. Build a test rig to experimentally determine static and dynamic force characteristics of important rotor/stator interactive forces (i.e., fluid annuli, bearings and pump stage).

PHASE - IV: Involve one or more feed pump vendors and electric utility companies to design, build and field test the new pump configuration(s) for extended periods of time to confirm the degree of vibration level reduction and reliability improvements.

Successful completion of this four-phase project will motivate feed pump manufacturers to accelerate their own in-house development programs and to develop a new generation of feed pump designs which are more responsive to the high availability needs of the electric utility industry.

A main portion of the Phase - I effort was devoted to developing the advanced engineering methods needed to analyze the vibration phenomena which occur in feed water pumps. As anticipated, this required extensive upgrading and restructuring of general purpose machinery vibration computer codes to mathematically model and simulate pump vibration phenomena. This effort was concentrated on developing the methodology to study new pump configurations which have the greatest potential for reducing feed pump vibration levels. The types of vibration which are treated in the analyses include, (i) synchronous (rotational speed) unbalance forced vibrations, (ii) non-synchronous (pump hydraulic) forced vibrations, (iii) threshold speeds of self-excited vibrations, (iv) static and dynamic properties of fluid-film journal bearings and inter-stage fluid annuli.

Using the engineering analysis methods and approaches developed in Phase - I, a series of analyses were performed on the Cane Run No. 6 boiler feed pumps (Louisville Gas and Electric Co.). These pumps are a twelve-stage variable speed design which has had a history of severe vibration problems. In parallel with these analyses, vibration measurements were made on these pumps to correlate with the analyses performed. The analyses correlate well with vibration data taken in the field. This shows that the analysis methods developed in Phase - I will give valid results when applied to new pump configurations. The analyses performed on the Cane Run No. 6 pumps also show why these units have had severe vibration problems. Inadequate rotor/bearing system damping is the basic shortcoming of these units and this appears to be a common situation with feed water pumps in general. Although this finding was anticipated, the Phase - I results clearly show why there is insufficient system damping. This additional physical insight therefore gives clear direction as to what types of design improvements have the greatest potential for reducing vibration levels. The thrust of Phase - II will be to exploit these findings to determine new configurations which best reduce feed pump vibration levels over the full range of pump operating conditions.

Section 2

CONCLUSIONS

The successful development of advanced feed pump designs which minimize residual pump vibration levels requires careful attention to a number of factors. First, a sound understanding of why present designs have frequent vibration problems is of foremost importance. Second, based on this understanding, new design configurations must be analyzed to identify design improvements which yield the greatest reductions of residual vibration. Third, those areas where existing technology deficiencies significantly hamper the development of low-vibration designs must be identified and rectified. Fourth, design improvements which analysis results show to best attenuate vibration levels should be thoroughly tested under field operating conditions to confirm the expected vibration reduction and reliability improvements. Based on the findings of Phase - I, the following conclusions can be drawn.

1. The major source of high residual vibration levels is the strong fluid dynamical forces which result from flow conditions in high-head centrifugal pumps. These forces are particularly strong at off-design operating conditions because of the resulting complex secondary flow patterns which are produced by the mismatch of impeller and flow passage geometries.

2. Pump fluid dynamical (hydraulic) forces produce larger than necessary pump vibrations because most present designs do not have adequate damping in the rotor/bearing system. Damping comes primarily from the journal bearings. However, many bearing designs presently used tend to lose damping in the subsynchronous (below rotational speed) frequency range. This is an unfortunate consequence because it is in the subsynchronous frequency range where hydraulic forces are strongest, as field measurements repeatedly show.

3. It is apparent from the correlation of field measurements and analyses that inter-stage sealing (i.e., wear ring and labyrinth fluid annuli) tends to reduce subsynchronous damping capacity and therefore aggravate vibration problems. This conclusion is based on computations which

show a 25 percent reduction of in oil-whip threshold speed when inter-stage fluid annuli are included in the analyses. Work is therefore needed to expand the presently limited amount of available data on the dynamic characteristics of fluid annuli.

4. The computation of resonant rotor mode shapes for the Cane Run No. 6 pumps show that the nodal points (i.e., zero vibration points) of the lowest frequency resonant vibration mode are located near the journal bearings. That is, the rotor vibrates much more between the bearings than at the bearings. This appears to be the usual situation with present multi-stage designs and contributes considerably to the inability of the journal bearings to adequately damp hydraulically induced vibrations. Even the best vibration damper dissipates little energy if located near a vibration nodal point.

5. The application of existing machinery vibration technology can yield a substantial reduction in feed water pump failures and increase considerably the operating periods between major maintenance. This can be accomplished by developing bearing and inter-stage sealing configurations which reduce pump vibration levels by providing adequate damping over the entire range of operating conditions.

Section 3

TECHNICAL SCOPE OF FUTURE WORK

The conclusions stated in the previous section of this report form a basis for a clear set of technical objectives beyond the Phase - I effort. The work that is now completed clearly supports the contention prior to Phase - I that the state-of-the-art in machinery vibration technology (both analytical and experimental) can be successfully applied to produce improvements in feed water pump designs which will significantly reduce vibration related operating problems of current designs and thereby increase feed water pump reliability. The technical objectives and plan of attack for Phase - II are summarized below.

3.1 BEARING CONFIGURATIONS TO INCREASE DAMPING

As previously described in the Conclusions, high levels of vibration in feed water pumps are frequent because of inadequate damping, particularly in the subsynchronous frequency range where hydraulic excitation forces are strongest. A major objective of Phase - II will be to demonstrate by comparative analyses that innovative but practical bearing designs can be devised which will considerably increase effective system damping. In order to accomplish this a prospective bearing configuration will have to alleviate two (2) basic situations which appear to be prevalent in many existing feed water pump designs. These are (i) the use of bearing configurations which inherently tend to lose damping at subsynchronous frequencies, and (ii) the close proximity of the lower resonant-mode nodal points to the bearing location.

Bearing concepts which have worked well in other machinery types will be analyzed to determine their potential benefits if developed for feed water pump applications. For example, squeeze-film damper bearing supports are now used successfully in the newer aircraft jet engine designs. They have also been successfully applied on some large steam turbines where the originally installed bearing configuration did not adequately damp the subsynchronous steam-whirl forces. A squeeze-film damper supported journal bearing developed specifically for feed water pumps would add considerable damping to the system and at the same time effectively move the vibration nodal points away from the bearings. New bearing configurations, employing this concept, will be thoroughly analyzed during the Phase - II effort.

3.2 INTER-STAGE FLUID ANNULUS EFFECTS

The fluid annuli of inter-stage sealing clearances play an important role in setting pump vibration levels. This is substantiated by both measurements in the field and the exploratory analyses of Phase - I. Inter-stage sealing clearances are commonly either a labyrinth or a smooth-bore bushing geometry.

In feed water pumps, the effects of rotational speed as well as high axial pressure gradients (i.e., inter-stage leakage) are both important. A review of the technical literature reveals that the available information on fluid annulus rotor vibration effects has not been geared to feed pump conditions. Currently, there is an unfortunate lack of both experimental and analytical information to guide the development of new inter-stage sealing designs which increase rather than decrease system vibration damping.

To adequately evaluate present annulus component geometries as well potentially better geometries, use of an experimental test apparatus is the most reliable and direct approach. A major objective of the Phase - II effort will be to design a test rig to measure the static and dynamic forces of inter-stage fluid annuli for feed water pumps.

Section 4

PUMP VIBRATION ANALYSIS

4.1 PURPOSE AND PERSPECTIVE

The main objective of Phase-I was to determine the level of engineering analysis required to perform reliable vibration studies of feed water pumps. This required considerable upgrading of general purpose machinery vibration computer codes. It also required verification of the mathematical models on existing feed water pumps. This was accomplished by analyzing operating pumps, making vibration measurements on the same pumps and comparing the analysis results with the measurements. This objective has been completed and has provided an increased understanding of pump vibration problems.

It is a common misconception, even among many who have contributed to the development of advanced computer codes, that rotating machinery vibration technology is now simply a computer user's exercise to get the right answer. This is far from the truth. Most rotating machinery types occasionally experience rotor vibration problems in the field. These field problems are often very costly to correct and in some instances not fully correctable, e.g., some large steam turbines which have been de-rated and cannot be safely operated at the originally contracted output. Although computer codes are often useful tools when attempting to cope with such costly problems, the hard-to-answer aspects of the problem are frequently some of the required inputs to the available computer codes. In the specific case of feed water pumps, the dynamic force levels and frequencies produced by the pump hydraulics are the major unknowns. Also, the static bearing loads can vary considerably depending upon the casing centering-tolerance stack-up, the operating flow rate of the pump and the non-uniform dimensional changes which occur during operating transients. It is therefore misleading to suggest that computer analyses can predict the complete vibration behavior of a particular machine. In the hands of experienced specialists who have good engineering judgement and physical insights, presently available computer codes are nevertheless invaluable tools. In this project, the computer codes assembled provide an effective means to make comparative analyses to test out innovations which can potentially increase rotor/bearing system damping. A comparison of computed vibration levels obtained with different bearing configurations tried on the same pump is a valid method for screening new ideas. In fact, it is in the job of comparing the relative merits of competing ideas where properly applied computerized analyses can make their strongest contribution to the development of improved machinery designs.

All rotating machinery types are not equally amenable to analysis and understanding of their vibration characteristics. For example, electrical machines such as large generators have complex rotors with non-axisymmetric bending stiffness which are not well modeled by circular beam elements, making reliable analyses difficult. Another example is the influence of the support structure of large turbine/generator sets which is not yet well understood because of difficulties involved in adequate dynamic modeling. Feed water pumps have their own idiosyncrasies which demand careful attention if reliable vibration analyses are to be made. The engineering analyses performed in Phase - I are indicative of the development approach which has the greatest potential of leading to new feed pump designs with minimized vibration levels and substantially improved reliability.

4.2 VIBRATION - ANALYSIS MODEL

The approach used in our analyses is based on the theory of linear vibrations which is thoroughly exposed in several available texts on vibration theory, e.g., Reference (2). The rotating and nonrotating components of a feed pump are essentially elastic continuous media bodies with complex geometrical shapes. The dynamic modeling of such structures are commonly made mathematically tractable by "discretizing" the system. That is, modeling the elastic structure by a finite number of masses and inertias inter-connected by a system of linearized "massless" elastic connections (springs) and linearized dampers. Nearly all rotor dynamics computer codes are based on an approach of this type, whether the formulation is the older transfer matrix algorithm* or the newer finite-element method**. As explained in Section 4.2.2, nonlinear dynamic analyses are not considered necessary for the current studies at this time, but have recently been applied in catastrophic turbine failure situations as described in Reference (3).

Provided good judgement is exercised in discretizing the actual machine, then the more masses used (i.e., the more degrees of freedom) the higher in frequency range the discrete model accurately characterizes the actual continuous media body. The

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* Used here to study synchronous and non-synchronous forced vibration.

** Used here to study instability self-excited vibration.

degree of model complexity needed varies considerably with the machine being analyzed and the frequency range within which the dynamic system behavior is sought.

4.2.1 Pump Rotor

The typical multi-stage feed pump rotor has a horizontal rotational centerline and is of fairly long slender construction with mass and inertia concentrations at each impeller stage, the thrust collars and coupling. All necessary dimensions and geometric parameters of the Cane Run No. 6 (CRN6) feed pumps were obtained by taking dimensional measurements on a disassembled unit in the repair shop. From these measurements, a scale layout of the complete rotor was constructed with all pertinent dimensions shown. The CRN6 pump is a twelve-stage design as shown in Figure (1). This rotor was discretized into twenty-six (26) circular beam elements ($4 \times 26 = 104$ degrees of freedom), using appropriate diameters and densities to accurately model the stiffness and mass properties at each element. Furthermore, additional concentrations of mass and inertia were added at each thrust collar and each impeller to account for the effects of these essentially rigid bodies. A tabulation of this input data for the rotor model is given in Table (1).

4.2.2 Journal Bearings

In these studies it is appropriate to consider the journal bearings as linear dynamic elements which is to assume that the stiffness and damping coefficients of the lubricating hydrodynamic oil films do not change with vibration amplitude. This assumption tends to weaken as vibration amplitudes become large and approach the size of the bearing radial clearance. However, the assumption of linearized bearing dynamic properties is appropriate to our analyses because we are searching for designs which make vibration amplitudes small and therefore the bearing dynamic behavior is essentially linear. For the same reason, linearized coefficients are also justified to model inter-stage fluid annulus effects.

The first part of determining the dynamic stiffness and damping coefficients for a journal bearing is a determination of the radial static load on each bearing. This step is important because the dynamic coefficients vary considerably with static load. Important sources of static bearing load are rotor weight, impeller hydraulic radial force and inter-stage fluid annuli. In the exploratory analyses of Phase - I, the impeller and fluid annuli static forces were neglected and only the dead weight of the rotor was included and resulted in 740 lbs. vertically downward radial force on each journal bearing. During the Phase - II effort, the additional static load sources from hydraulic and fluid annulus effects will be included and their influence parametrically studied.

The bearing dynamic coefficients are based on the most frequently used journal bearing dynamics representation --- the 2-degree-of-freedom bearing model. For this model, the dynamic interaction radial force vector between journal and bearing has two (2) components in a radial plane through the center of the bearing at its axial mid-point.

$$\begin{aligned} F_x &= -K_{xx}x-K_{xy}y-B_{xx}\dot{x}-B_{xy}\dot{y} \\ F_y &= -K_{yx}x-K_{yy}y-B_{yx}\dot{x}-B_{yy}\dot{y} \end{aligned} \qquad (4.1)$$

Here, x, y, $\dot{x}$, $\dot{y}$ are the vector components of the relative radial dynamic displacement and velocity between bearing and journal, referenced to the static equilibrium position of the journal. F_x and F_y are the corresponding x and y dynamic rotor force components where the x and y directions are mutually perpendicular and lie in a plane perpendicular to the axis of rotation (z-axis). The y-axis is vertical; x-axis and z-axis are horizontal. Equations (4.1) are commonly written in matrix form,

$$\begin{Bmatrix} F_x \\ F_y \end{Bmatrix} = -\begin{bmatrix} K_{xx} & K_{xy} \\ K_{yx} & K_{yy} \end{bmatrix}\begin{Bmatrix} x \\ y \end{Bmatrix} - \begin{bmatrix} B_{xx} & B_{xy} \\ B_{yx} & B_{yy} \end{bmatrix}\begin{Bmatrix} \dot{x} \\ \dot{y} \end{Bmatrix} \qquad (4.2)$$

where the stiffness, $[K_{ij}]$, and damping, $[B_{ij}]$, matrices contain the standard eight stiffness and damping coefficients of the journal bearing. Computer codes to calculate these eight coefficients are generally based on classical viscous-flow formulations and additionally account for film rupture (cavitation) in the low pressure regions of the lubricating film. As part of the Phase - I effort, the very general FLIMP computer code was assembled for this purpose. It accommodates virtually any journal bearing geometry.

Dynamic coefficients were developed for two (2) different bearing configurations; (i) fixed-arc sleeve bearing, and (ii) pivoted-pad journal bearing. This was done because the CRN6 pumps were originally installed with two (2) sleeve bearings which were later replaced with two (2) pivoted-pad journal bearings in an attempt to reduce vibration levels and thereby extend operating time between repairs. As described in Section 5 of this report, the pivoted-pad bearings control vibrations adequately with new wear rings installed and operating at design clearances. However, as the wear ring clearances open up through excessive wear, high vibration levels gradually return.

The bearing dynamic coefficients used as tabulated in Tables 2 and 3, and cover both synchronous and non-synchronous modes of forced vibration response. These coefficients are also used in the analysis to determine instability self-excited vibration threshold speeds. Note that the sleeve bearing has coefficients which are a function of rotational speed. Regarding pivoted-pad journal bearings, the effective 2-degree-of-freedom bearing dynamic model has coefficients which are a function of vibration frequency in addition to rotational speed. This is because the configuration with a pivoted-pad bearing simply has more than two degrees of freedom, i.e., the pad arcs pivot dynamically as part of the vibration. In order to eliminate the pivoting pad degrees of freedom, while retaining the journal's two degrees of freedom, the vibration is restricted to harmonic motion. This does not limit the applicability of the dynamic coefficients since we are only considering steady-state vibrations, not transient vibrations. This is a standard approach which transforms the bearing/journal combination into a 2-degree-of-freedom system but results in the eight stiffness and damping coefficient, $[K_{ij}]$ and $[B_{ij}]$, becoming a function of vibration frequency. Most dynamic coefficient data available in the literature for pivoted-pad journal bearings is tacitly based on a rotational speed (synchronous) vibration frequency (i.e., $\Omega/\omega=1$), which is applicable only to critical speed and synchronous response computations. However, many rotor vibration investigators in stability and forced non-synchronous response have used these coefficients in their analyses without realizing their error. As a consequence, erroneous results have been generated because the vibration-frequency dependence of pivoted-pad bearing dynamic coefficients is quite strong (See Table 3).

4.2.3 Fluid Annuli

As mentioned in the previous section, presently available bearing computer codes (including FLIMP) are generally based on classical viscous-flow formulations and thus neglect fluid inertia. This assumption is warranted for oil lubricated bearings, but sometimes questionable for water lubricated bearings such as in reactor coolant pumps. In the case of larger clearance fluid annuli such as in feed pump inter-stage sealing, fluid inertia effects are quite important, if not dominant, over viscous effects. For many years, such fluid annuli were not considered to contribute significantly to rotor centering and de-centering forces because classical journal bearing theory (i.e., fluid inertia neglected) predicted so. Recently, analytical treatments which approximate fluid inertia effects have shown that such fluid annuli can produce significant static as well as dynamic interaction forces on the rotor. In the case of multi-stage boiler feed pumps, the large number of interstage fluid annuli compounds their importance.

Available analytical data on fluid annulus forces is very limited and not geared to feed pump conditions. Experimental determination of these forces and their dynamic coefficients is badly needed to determine optimum geometries which maximize rotor system damping. Also, advanced analytical treatment is warranted to account for the combined effect of rotation and high-pressure-gradient axial through flow (i.e., inter-stage leakage).

The dynamic effects of inter-stage fluid annuli can be worked into rotor vibration analyses in the same way as for journal bearings provided the prior determination of the eight linearized stiffness and damping coefficients includes fluid inertia effects. However, the addition of fluid inertia also makes the stiffness coefficients a function of vibration frequency. This can be readily shown since fluid inertia importance means that the dynamic film forces (between rotating and non-rotating boundaries) will be a function of the acceleration vector $(\ddot{x}, \ddot{y})$ in addition to the displacement (x, y) and velocity $(\dot{x}, \dot{y})$ vectors. Therefore, when inertia effects are included, the linearized rotor-stator interactive dynamic forces then have an additional ingredient to those for journal bearings as shown below.

$$\begin{Bmatrix} F_x \\ F_y \end{Bmatrix} = - \begin{bmatrix} K_{xx} & K_{xy} \\ K_{yx} & K_{yy} \end{bmatrix} \begin{Bmatrix} x \\ y \end{Bmatrix} - \begin{bmatrix} B_{xx} & B_{xy} \\ B_{yx} & B_{yy} \end{bmatrix} \begin{Bmatrix} \dot{x} \\ \dot{y} \end{Bmatrix} - \begin{bmatrix} D_{xx} & D_{xy} \\ D_{yx} & D_{yy} \end{bmatrix} \begin{Bmatrix} \ddot{x} \\ \ddot{y} \end{Bmatrix} \tag{4.3}$$

Note that four (4) more dynamic coefficients are needed, $[D_{ij}]$. However, for steady-state harmonic vibrations, acceleration and displacement are directly related because harmonic motion has the following general form; i.e., in the x-direction.

$$\text{Displacement,} \quad x = X \sin (\Omega t + \Phi_x) \tag{4.4}$$

therefore,

$$\text{Velocity,} \quad \dot{x} = \frac{dx}{dt} = \Omega X \cos (\Omega t + \Phi_x) \tag{4.5}$$

and

$$\text{Acceleration,} \quad \ddot{x} = \frac{d^2x}{dt^2} = \frac{d\dot{x}}{dt} = -\,\Omega^2 X \sin(\Omega t + \Phi_x) \tag{4.6}$$

Comparing Equations (4.4) and (4.6), the following well known relationship for harmonic motion is obtained

$$\ddot{x} = -\,\Omega^2 x \tag{4.7}$$

Similarly, the y displacement and acceleration components are related in the same way,

$$\ddot{y} = -\,\Omega^2 y \tag{4.8}$$

where Ω is the frequency of vibration. One can therefore modify the stiffness matrix to contain journal acceleration effects. This facilitates the use of standard rotor vibration codes which do not have specific input options for the four $[D_{ij}]$ coefficients. This is accomplished simply by modifying the stiffness matrix as justified by Equations (4.7) and (4.8) and shown below.

$$[\bar{K}_{ij}] = [K_{ij} - \Omega^2 D_{ij}] \tag{4.9}$$

Realize that $[D_{ij}]$ does not embody all fluid inertia effects, only those which respond to rotor acceleration relative to the pump casing. Proper account of fluid inertia also modifies considerably the numerical value of the $[K_{ij}]$ and $[B_{ij}]$ coefficients.

Fluid annuli can therefore be included in rotor vibration analyses just like journal bearings, provided fluid inertia effects have been properly factored into the prior computation of the linearized stiffness and damping coefficients.

Available data on fluid annulus dynamic coefficients is quite limited and based on very approximate approaches, but adequate to show that fluid annuli effects are quite important and cannot be neglected. The results presented in Reference (4) were used to obtain approximate dynamic coefficients for the inter-stage sealing annuli of the CRN6 boiler feed pumps. Each impeller contains essentially four (4) fluid annuli, three (3) at the impeller eye (i.e., a triple-pass arrangement) and one at the opposite side of the impeller. For each impeller, these were condensed into one set of dynamic coefficients per impeller and included in the various rotor vibration analyses as such. The dynamic coefficient values used are given in Table 4.

4.3 ANALYSES PERFORMED

4.3.1 Synchronous Unbalance Forced Response

This type of analysis simulates the state of rotor/bearing vibration which occurs as a result of residual mass unbalance on the rotor. All rotors of course retain some mass unbalance since perfect dimensions and exact uniform material densities do not exist. Modern balancing machines and techniques have been greatly refined over the last twenty years so that new rotors are routinely well balanced if currently available methods are employed. However, a computation of unbalance induced vibration is still an essential design analysis. Machines frequently go "out" of balance due to wear, thermal distortions and unstable shrink fits. Also important are the so called rotor/bearing "critical speeds", which are the speeds at which vibration amplitudes peak due to a resonance condition excited by the unbalance forces. That is, since the residual imperfection in mass distribution is "locked" in the rotor, the resulting dynamic excitation forces always have a frequency equal to rotational frequency and always vary in magnitude in proportion to rotational speed squared (ω^2). Therefore, as a machine is brought up or down in speed, the mass unbalance produces, in effect, a system of co-rotational synchronous radial forces on the rotor. Keep in mind that the journal bearing dynamic coefficients also change significantly as a function of rotational speed (see Tables 2 and 3).

As the speed frequency sweeps through a rotor/bearing resonance frequency, vibration peaks occur and the speeds at which these peaks occur are commonly called "critical speeds". However, if the rotor is properly balanced, these vibration peaks are usually not sufficiently high to jeopardize the machine. Continuous operation near a critical speed is, however, generally bad design or operating practice since the machine will be highly sensitive to increases in unbalance which occur during normal use.

Direct motor driven (i.e., constant speed) multi-stage boiler feed pumps typically operate above the first (lowest) critical speed, but below the second critical speed. The CRN6 pump design is motor driven, but through a variable speed hydraulic coupling. The maximum operating speed is 3,480 rpm. Large units (above 14,000 hp) are driven by steam turbines and therefore also are variable speed. In general, variable speed operation is good from a pump hydraulics point-of-view since part-load operation can be accommodated by running the pump at reduced speeds rather than throttling back into the hydraulically unstable low-flow region of pump operation. However, accurate critical speed determination becomes more important since variable-speed machines can inadvertently be operated at a critical speed much easier than a constant-speed machine.

The unbalance vibration analysis of the CRN6 pumps was performed using the SANSOR computer program which was upgraded and installed on an IBM 370-178/MVS computer system as part of the Phase - I effort. The SANSOR program uses the transfer-matrix algorithm to solve the steady-state equations of motion for the entire rotor/bearing system. Appropriate unbalances were applied to the CRN6 pump rotor model and its synchronous vibration response was first computed over a speed range from 100 to 5,000 rpm. A computer-printer plot from this series of computations (Figure 2) shows the computed first critical speed to be in the vicinity of 1,500 rpm. To determine the computed first critical speed more precisely, the speed range was then restricted between 1,470 and 1,530 rpm (sort of "zooming in") and the computations repeated. The results (Figure 3) pin-point the computed first critical speed at 1,490 rpm. Both Figures 2 and 3 display the single-peak rotor vibration amplitude in the horizontal radial direction at the rotor mid-span axial location, i.e., directly between the journal bearings. This is where first critical speed rotor vibration amplitude is largest as shown by the plotted rotor resonant-mode shape at 1,490 rpm (Figure 4). Also, note from Figure 4 that the resonant-mode vibration amplitude passes through zero very near both journal bearings. That is, the journal bearings are located very near the vibration nodal points. This is one obvious reason why the CRN6 pumps have suffered frequent vibration problems, because the bearings cannot effectively exert their damping if located so near the vibration nodal points. Keep in mind that the lowest resonant mode is susceptible to excitation from hydraulic forces whether or not the pump is running at the corresponding critical speed.

In this sequence of rotor response computations, pivoted-pad journal bearings were used so as to provide a basis for comparison with field measurements on the actual CRN6 units which have pivoted-pad journal bearings. As described later in Section 5, field measurements taken on these pumps show the actual first critical speed to be near 1,500 rpm, which is an excellent correlation with these computer-analysis results.

The results just described were computed without including any dynamic coefficients for the inter-stage sealing fluid annuli. Therefore, a second sequence of computer runs were made keeping everything the same as in the previous case, but including the fluid annuli dynamic coefficients at each of the twelve (12) impeller stages. The static position of the rotor was assumed to run concentric to each of the fluid annuli non-rotating boundaries. During Phase - II, the effects of non-concentric fluid annuli will be studied. At least with concentric fluid annuli, their effect on critical speed value and unbalance vibration levels is small, comparing the results (Figure 5) with the previous results (Figure 2). It would be premature, based on these results alone, to conclude that fluid annuli do not significantly alter synchronous vibrations, but it appears that this may very well be so. However, as shown later (Section 4.3.3), the fluid annuli have a strong effect on subsynchronous damping, which is particularly important to the objective of controlling subsynchronous hydraulically induced rotor forces.

4.3.2 Non-Synchronous Forced Response

The strongest and most troublesome dynamic forces arise from internal flow conditions within the pump hydraulic passages. Much of the potentially destructive energy in these forces occurs at frequencies other than synchronous. Although these forces are generally stronger in the subsynchronous frequency range, they also can produce significant force magnitudes at above rotational speed frequencies.

The analysis of non-synchronous response requires an independent control over rotor speed and vibration frequency. This is an additional complication to what is required to analyze synchronous vibrations where the forcing frequency is always at speed frequency and the bearing coefficients vary only with the rotor speed. That is, synchronous vibrations are computed as a function of speed only. With non-synchronous vibration studies, one must vary speed and forcing frequency independently. The SANSOR program has been recently upgraded to handle non-synchronous problems. The use of this additional analytic capability will play a major role during the Phase - II analyses when evaluating the ability of new bearing and seal configurations to damp out non-synchronous vibrations.

The approach in using the non-synchronous option of the SANSOR program is somewhat distinct from its use in synchronous unbalance response. Unlike unbalance forces, we generally do not know the actual excitation force frequencies and magnitudes a priori simply by knowing the exact geometric configuration of every pump component. Unfortunately, the present state-of-the-art in pump hydraulics is such that information on these forces for a particular design is only known after it is designed, built and operated in the field. One can therefore not rely on this information during the design-analysis phase. The proper analysis approach must therefore be to "calibrate" the rotor/bearing system by computing its sensitivity to non-synchronous forces over a wide frequency range. This approach will show the particular frequencies where the pump will tend to vibrate excessively if hydraulic forces ultimately occur at these frequencies. This information will facilitate quick diagnosis of pump vibration problems which may subsequently occur when the pump is operated in the field. More important to the present project, this approach provides the tool to determine which bearing and seal designs best control pump vibration levels over the entire frequency range of importance.

The method used to dynamically "calibrate" the rotor/bearing system is to hold the rotational speed constant at various speeds, and to vary the excitation force frequency while holding its magnitude at one pound. The resulting computed vibration amplitudes therefore calibrate the rotor/bearing system by producing plots of vibration amplitude per pound of excitation force as a function of forcing frequency.

To demonstrate this approach, a one pound sinusoidally varying excitation force was applied in the radial horizontal direction (i.e., x-direction) at the next-to-last (11th) impeller stage. This rotor location was chosen to avoid close proximity to vibration nodal points of either of the two lowest frequency resonant modes. That is, to insure adequate excitation of all important resonant vibration modes. The results are shown on Figure 6, which shows single-peak vibration amplitude at the No. 11 impeller versus excitation frequency. Observe that there are two vibration peaks (i.e., resonances) within the frequency range scanned, the lowest at 24 cps (1,440 cpm) and the other at 105 cps (6,300 cpm). The lowest frequency mode at 1,440 cpm is essentially the same mode excited by the synchronous unbalance at the 1,490 rpm critical speed. The second frequency mode at 6,300 cpm would closely correspond to the second critical speed which would be near 6,300 rpm. During Phase - II, studies will be made to determine optimum-damping bearing and inter-stage annulus configurations which minimize these resonant vibration peaks. This approach to optimize rotor/bearing system damping therefore circumvents the problem of not knowing in advance what the actual hydraulic forcing frequencies are.

4.3.3 Instability Threshold Speed

Thus far rotor vibrations which are forced either by rotor mass unbalance or forced by hydraulic flow conditions in the pump have been described. There is another class of rotor vibrations which can occur even if the mass unbalance and radial hydraulic forces are zero. Such vibrations are commonly called self-excited vibrations, thus the term instability is frequently used. Another term frequently applied is "oil whip" since the discovery of such vibrations was first made in machines with oil lubricated sleeve bearings which were found to be the cause. As explained in Reference (3), fixed-arc sleeve bearings tend to lose damping when the vibration frequency goes below some critical cross-over value, typically somewhere below one-half the rotational-speed frequency. If a machine is operating sufficiently above the first critical speed (i.e., above the lowest resonance frequency) such that the ratio of the lowest resonance frequency to the rotational speed, (Ω/ω), is below the zero-damping cross-over point, then the machine will experience high levels of vibration having a frequency of the lowest resonant mode. Such self-excited vibrations are a common occurrence in boiler feed pumps, as well as other rotating machinery types. To avoid this self-excited vibration, an analysis must be performed on prospective designs to determine the lowest speed (so-called threshold speed) at which oil whip first occurs. Unlike critical speeds, oil whip vibrations cannot be passed through by going to higher rotor speeds than the threshold speed. This is so because as the rotor speed is increased above the threshold speed, the vibration-to-speed frequency ratio (Ω/ω) becomes progressively smaller because Ω remains relatively constant while the speed, ω, gets larger. This causes the bearing damping to become even more negative, perpetuating vibration of the rotor at its lowest resonant frequency. Therefore, oil whip limits the safe maximum operating speed in contrast to critical speeds which do not.

To determine threshold speeds on boiler feed pumps, the STABL computer code was upgraded and installed on an IBM 370-178/MVS computer system. STABL was developed to utilize the structural modeling capabilities of modern advanced finite element computer codes such as NASTRAN, as described in Reference (5).

To study oil whip threshold speed with the CRN6 boiler feed pumps, fixed-arc sleeve bearings were used in the analysis to compare with available field data taken prior to the retrofit change to pivoted-pad journal bearings. Two (2) analyses were performed, one without inter-stage fluid annuli and a second with. The results are summarized on the next page (see Tables 5 and 6 for computer summaries). These results show a 25 percent reduction in threshold speed (i.e., the maximum safe speed) due to the additional effects of inter-stage fluid annuli. This boosts the frequency ratio to 0.63. In general, with journal bearings alone, this ratio is below 0.5 as in Case 1. As described later in Section 5, the results of Case 2 correlate well with available field data. The reduction of threshold speed caused by inter-stage fluid annuli indicates that these fluid annuli reduce system damping, at least in the subsynchronous frequency range. These fluid annuli therefore contribute to subsynchronous high vibration levels. Note that the frequency of vibration is basically unchanged between Cases 1 and 2. This is consistent with the results of the synchronous forced response analysis (Section 4.3.1) which showed that the fluid annuli did change the first-critical-speed frequency.

CASE NO.	JOURNAL BEARINGS	INTER-STAGE FLUID ANNULI	COMPUTED THRESHOLD SPEED * (rpm)	COMPUTED FREQUENCY RATIO (Ω/ω)	VIBRATION FREQUENCY Ω, (cps)
1	Sleeve	Not Included	3,060	0.47	24.0
2	Sleeve	Concentric	2,276	0.63	24.1

* Maximum design operating speed is 3,480 rpm

4.3.4 Discussion of Results

The analyses performed thus far are exploratory and were not intended to provide exhaustive parametric studies. The first objective was to assemble and upgrade the required advanced computer codes and to determine the level of mathematical model refinement needed to obtain reasonably good correlation with field measurements. This objective has been met and the results obtained clearly demonstrate that the computer models are quite adequate to simulate various design changes which potentially reduce vibration levels.

As summarized in the Conclusions (Section 2), the results show why the CRN6 pumps, and other similar designs, incur frequent operating difficulties precipitated by excessive vibration. The close proximity of journal bearings to the lowest-mode nodal points and the general lack of system damping allow the strong hydraulically induced dynamic forces to raise havoc with such pumps. Therefore, the Phase - I results establish the needed level of engineering analysis, and give strong direction as to the kinds of new design configurations which warrant further study and development. The findings of Phase - I fortify the conclusion that application of advanced engineering methods in vibration can lead to a marked reduction in feed water pump vibration levels.

Section 5

FIELD DATA ON CANE RUN NO. 6 FEED PUMPS

5.1 BRIEF HISTORY OF OPERATING DIFFICULTIES AND DESIGN FIXES

The Cane Run No. 6 feed pumps are a twelve-stage design supplied by Allis-Chalmers. This same design is also applied in shortened versions which have fewer than twelve (12) stages. A few of the stations (out of a total of 46) where these pumps have been used are summarized below.

STATION	COMPANY	STAGES	TYPE OF SERVICE
Cane Run No. 6	Louisville Gas & Elec.	12	Main feed pump
Palo Seco No. 3 & 4	Puerto Rico W. R. Auth.	8	Main feed pump
Gannon No. 3	Tampa Elec.	10	Main feed pump
Winyah No. 1	S. Carolina Pub. Ser. Auth.	11	Main feed pump
Oak Creek Pine Station	Wisconsin Elec. Power	12	Main feed pump
Eddystone No. 3 & 4	Philadelphia Elec.	8	Start-up feed pump

Everyone of these stations has had operating difficulties with these pumps, typically involving uncontrolled pump vibration levels. Tampa Electric's solution ultimately was to replace the pumps with those of another vendor. Palo Seco continues to have pump failures with these units and is considering different pumps to replace them. The Cane Run No. 6 units have been somewhat improved through a series of design retrofits. A number of changes on the Eddystone units appear to have alleviated the operating difficulties, but these units are only used for start-up.

The CRN6 pumps as initially delivered experienced high vibration levels which made normal operation nearly impossible. Failure of bearings, seals, wear rings and other components frequently resulted after only short periods (as short as one month) of operation. A number of design modifications were made on these units, resulting in the pumps being operable for extended periods of time. However, high levels of vibration still return as the inter-stage sealing clearances open up through accelerated wear. Design changes which improved these pumps are briefly discussed below.

The pumps as originally installed had sleeve bearings, a 12 percent recirc flow valve, floating ring main seals and a pressure-controlled seal injection system. The first, and probably most important, design retrofit was to replace the sleeve bearings with pivoted-pad journal bearings. This change alone seems to control the violent subsynchronous vibrations, at least when the pumps have fairly new inter-stage wear rings. Other improvements were also made but their individual effects cannot be fully evaluated because comprehensive vibration measurements were not made after each change. These other improvements include changing the seal injection system from pressure control to temperature control because the subsynchronous vibration component was found to be sensitive to seal temperature. Also, a change from floating ring seals to serrated bushings was made because the floating ring seals were thought to aggravate the subsynchronous vibration and were also more prone to a short life in the presence of high vibration levels. The original recirc valve was replaced by a larger one to allow a 25 percent recirc flow (instead of original 12 percent) to lessen hydraulically induced vibrations at low flow operation. Finally, the impeller-to-diffuser tip clearance was opened up from 1 percent to 3 percent to reduce the vane-passing-frequency hydraulically induced shocks on the impeller and diffuser vanes; i.e., to reduce the tendency for fatigue failure of impeller and diffuser vanes.

5.2 VIBRATION MEASUREMENTS

Various vibration measurements were taken on the Cane Run No. 6 boiler feed pumps. This data has been studied to determine how well the computer analyses characterize boiler feed pump vibration characteristics. This was a main objective of Phase - I; i.e., to establish sufficiently accurate analysis and modeling details which can be used with confidence to evaluate new configurations.

In order to judge the SANSOR (Synchronous And Non-Synchronous Orbital Response) computer code and the mathematical modeling, measurements to determine the first critical speed were performed. As mentioned previously, new or recently overhauled rotors may be sufficiently well balanced that the critical speed vibration level can be hard to detect from other synchronous run out. In other words, the residual unbalance may be too small to produce an accurately determined vibration spike. A more reliable measurement to determine critical speed is then the synchronous vibration phase angle, especially if the lower resonance frequencies are widely spaced as in this case. Recall from basic vibration theory that passage through a single resonance is accompanied by a 180° phase shift. This occurs even if the resonant peak is small and hard to "capture". The critical speed frequency can be identified by the mid-point speed of this 180° phase shift; i.e., the speed at which the phase angle shift goes through 90°. The results of four (4) separate tests have been plotted on Figure 7, and show the measured critical speed to range from 1,400 rpm to 1,600 rpm. Note that tests were run on two (2) different pumps. This is not a significantly large spread (1,500 ± 100 rpm) and indicates a fairly steady critical speed near 1,500 rpm. The fact that the Cane Run No. 6 feed pumps are variable speed (i.e., speed can be held constant at various values) made these measurements possible. On direct motor driven units, an attempt can be made to pin-point the critical speed during a coast-down but this is considerably less accurate and not always successful.

The next set of field measurements presented (Figure 8) show a vibration amplitude-frequency spectrum for CRN6 #61, while running at 3,117 rpm. Note that there are two (2) prominent vibration frequencies present, the synchronous component (52 Hz) and a subsynchronous component (34 Hz). The practical significance of the sub-harmonic component is that its amplitude subsequently grew to 7 mils as described on Figure 8. The major finding however is that the frequency ratio (Ω/ω) of this subharmonic component is 0.65. This is a particularly important correlation with the computed results for self-induced instability threshold speed analysis.

5.3 CORRELATION WITH ANALYSES

The computer analysis determination of the first critical speed gave a result of 1,490 rpm which agrees very well with the field measurements of 1,500 ± 100 rpm. It is therefore concluded that the SANSOR computer code and the mathematical model are more than sufficiently accurate to study forced vibrations of boiler feed pumps.

Our stability threshold speed analyses and field measurements (Fig. 8) do not correlate as well. This was anticipated because the dynamic coefficients for the inter-stage fluid annuli are quite approximate. Our analysis results predict a threshold speed of 3,060 rpm (Ω/ω = 0.47) without fluid annuli and 2,276 rpm (Ω/ω = 0.63) with fluid annuli as compared to 3,117 rpm (Ω/ω = 0.65) from field measurements (Figure 8). Note that the analysis without fluid annuli gives a good prediction of threshold speed, whereas the analysis with fluid annuli gives a good prediction for the Ω/ω frequency ratio. It must be emphasized that threshold speed determination is considerably more sensitive to mathematical model imperfections than is critical speed determination. The threshold speed occurs when the net system damping (of the lowest resonant mode) goes from positive to negative; i.e., goes through zero. Any small amount of extraneous damping not included in the model can potentially produce a sizeable difference in threshold speed. It is essential that better data (analytical as well as experimental) on inter-stage fluid annuli dynamic forces and linearized stiffness and damping coefficients be obtained during future phases of this project.

CANE RUN NO. 6 ROTOR MODEL

	LG	OD	ID	WA	RG	DENS
1	3.2500	4.5000	0.0	1.27000E+01	3.6000	0.2830
2	7.3750	4.5000	0.0	0.0	0.0	0.2830
3	3.2250	4.5000	0.0	0.0	0.0	0.2830
4	7.1450	5.5000	0.0	0.0	0.0	0.2830
5	4.0950	5.5000	0.0	0.0	0.0	0.2830
6	4.4000	6.5000	0.0	0.0	0.0	0.2830
7	0.0	6.5000	0.0	2.90000E+01	5.0000	0.2830
8	4.1600	6.2500	0.0	0.0	0.0	0.2830
9	6.0000	6.2500	0.0	0.0	0.0	0.2830
10	1.1300	5.3450	0.0	0.0	0.0	0.2830
11	5.0300	5.3610	0.0	3.85000E+01	7.3700	0.2830
12	5.1900	5.3680	0.0	3.85000E+01	7.3700	0.2830
13	5.1900	5.3760	0.0	3.85000E+01	7.3700	0.2830
14	5.1900	5.3840	0.0	3.85000E+01	7.3700	0.2830
15	5.1900	5.3920	0.0	3.85000E+01	7.3700	0.2830
16	5.1900	5.4000	0.0	3.85000E+01	7.3700	0.2830
17	5.1900	5.4080	0.0	3.85000E+01	7.3700	0.2830
18	5.1900	5.4170	0.0	3.85000E+01	7.3700	0.2830
19	5.1900	5.4240	0.0	3.85000E+01	7.3700	0.2830
20	5.1900	5.4320	0.0	3.85000E+01	7.3700	0.2830
21	5.1900	5.4400	0.0	3.85000E+01	7.3700	0.2830
22	5.5400	5.4480	0.0	3.55000E+01	7.6700	0.2830
23	5.2500	5.5000	0.0	0.0	0.0	0.2830
24	8.2500	5.5000	0.0	0.0	0.0	0.2830
25	3.6250	4.5000	0.0	0.0	0.0	0.2830
26	3.9000	4.5000	0.0	0.0	0.0	0.2830
27	8.2250	9.5000	0.0	0.0	0.0	0.2830
28	-1.0000	9.5000	0.0	0.0	0.0	0.2830

TOTAL LENGTH = 132.4998 TOTAL WEIGHT = 1.470521E+03

TABLE 1: ROTOR MODEL INPUT DATA.

S	.562-01	.104+00	.170+00	.394+00	.987+00	.208+01
$\bar{\bar{B}}_{xx}$	.776+01	.698+01	.684+01	.773+01	.117+02	.209+02
$\bar{\bar{B}}_{xy}$	.175+01	.170+01	.177+01	.197+01	.184+01	.167+01
$\bar{\bar{B}}_{yx}$	.175+01	.170+01	.177+01	.197+01	.184+01	.167+01
$\bar{\bar{B}}_{yy}$	.105+01	.129+01	.163+01	.262+01	.423+01	.733+01
$\bar{\bar{K}}_{xx}$	.625+01	.428+01	.322+01	.208+01	.161+01	.155+01
$\bar{\bar{K}}_{xy}$	.394+01	.352+01	.343+01	.384+01	.583+01	.104+02
$\bar{\bar{K}}_{yx}$	.314+00	-.401-01	-.356+00	-.103+01	-.198+01	-.360+01
$\bar{\bar{K}}_{yy}$	.149+01	.147+01	.151+01	.165+01	.168+01	.163+01

TABLE 2: DIMENSIONLESS LINEARIZED STIFFNESS AND DAMPING COEFFICIENTS, SLEEVE BEARING.

$\bar{\bar{B}}_{ij} = C\omega B_{ij}/W$, $\bar{\bar{K}}_{ij} = CK_{ij}/W$, C = Radial Clearance,

ω = Speed, W = Load, S = Sommerfeld Number

Speed (rpm)	K (lb/in)	ωB (lb/in)
100.	1.27 + 06	9.47 + 05
1200.	4.74 + 05	1.13 + 06
2500.	3.36 + 05	1.54 + 06
3500.	3.02 + 05	1.81 + 06
5000.	2.42 + 05	2.22 + 06

(a) Synchronous Stiffness and Damping Coefficients ($\Omega = \omega$)

Frequency Ω (CPM)	K (lb/in)	ΩB (lb/in)
0.	1.44 + 06	0.
1044.	1.20 + 06	4.97 + 04
1740.	8.88 + 05	2.37 + 05
3480.	3.05 + 05	1.81 + 06
6960.	-2.96 + 04	9.83 + 06

(b) Non-Synchronous Stiffness and Damping Coefficients (N = 3480 rpm)

TABLE 3: STIFFNESS AND DAMPING COEFFICIENTS FOR PIVOTED-PAD JOURNAL BEARING.

$K_{xx} = K_{yy} = K$, $B_{xx} = B_{yy} = B$,

ALL CROSS-COUPLED COEFFICIENTS ARE ZERO

Speed (rpm)	K_1 (lb/in)	K_2 (lb/in)	B (lb sec/in)
100.	- 2.09	7.2	0.69
1200.	- 77.5	310.0	2.47
1500.	-103.0	412.5	2.63
3500.	-370.0	1477.5	4.03
5000.	-690.0	2750.0	5.28

(a) Frequency-to-Speed Ratio, $\Omega/\omega = 0.2$

Speed (rpm)	K_1 (lb/in)	K_2 (lb/in)	B (lb sec/in)
100.	- 3.6	7.2	0.69
1200.	- 154.5	310.0	2.47
1500.	- 201.8	412.5	2.63
3500.	- 707.5	1477.5	4.03
5000.	-1380.0	2750.0	5.28

(b) Frequency-to-Speed Ratio, $\Omega/\omega = 0.5$

Speed (rpm)	K_1 (lb/in)	K_2 (lb/in)	B (lb sec/in)
100.	- 5.4	7.2	0.69
1200.	- 228.5	310.0	2.47
1500.	- 297.5	412.5	2.63
3500.	-1030.0	1477.5	4.03
5000.	-2042.5	2750.0	5.28

(c) Frequency-to-Speed Ratio, $\Omega/\omega = 0.7$

TABLE 4: CONCENTRIC FLUID ANNULUS STIFFNESS AND DAMPING COEFFICIENTS FOR EACH PUMP IMPELLER. $K_{xx} = K_{yy} = K_1$, $K_{xy} = -K_{yx} = K_2$, $B_{xx} = B_{yy} = B_{xy} = -B_{yx} = B$.

OUTPUT SUMMARY

THRESHOLD SPEED = 3.0596E+03 RPM
WHIRL RATIO = 4.7234E-01
RESIDUAL GROWTH = 8.9798E-03
WHIRL FREQUENCY = 2.4086E+01 HZ

--------EIGENVALUES--------

REAL	IMAGINARY
-0.1807E+05	0.0
-0.5794E+04	0.0
-0.8691E+04	0.0
-0.2994E+04	0.0
-0.1381E+02	-0.1549E+04
-0.1381E+02	0.1549E+04
-0.4723E+01	-0.1547E+04
-0.4723E+01	0.1547E+04
-0.3561E+01	-0.6608E+03
-0.3561E+01	0.6608E+03
-0.1148E+02	-0.6595E+03
-0.1148E+02	0.6595E+03
-0.1114E+03	-0.1475E+03
-0.1114E+03	0.1475E+03
-0.8130E+02	-0.1585E+03
-0.8130E+02	0.1585E+03
-0.8980E-02	-0.1513E+03
-0.8990E-02	0.1513E+03
-0.1468E+01	-0.1574E+03
-0.1468E+01	0.1574E+03

EIGENVALUE-SOLUTION PERFORMANCE INDEX IS GOOD.

SELF-EXCITED VIBRATION MODE

COORDINATE	AMPLITUDE	PHASE(DEG)
1	0.1000E+01	0.0
2	0.9734E+00	-0.8541E+02
3	0.1382E+02	0.1841E+01
4	0.1239E+02	-0.8652E+02
5	0.1631E+02	0.1924E+01
6	0.1463E+02	-0.8657E+02
7	0.1048E+02	0.2095E+01
8	0.9447E+01	-0.8665E+02
9	0.1143E+01	0.5280E+01
10	0.1127E+01	-0.8791E+02

ROTOR MASS	MAJOR-AXIS	MINOR-AXIS	BETA(DEG)
1	0.1028E+01	0.9442E+00	0.3568E+02
2	0.1384E+02	0.1237E+02	0.7346E+01
3	0.1633E+02	0.1460E+02	0.6777E+01
4	0.1049E+02	0.9435E+01	0.5925E+01
5	0.1167E+01	0.1102E+01	-0.3791E+02

TABLE 5: ROTOR/BEARING INSTABILITY THRESHOLD SPEED COMPUTER ANALYSIS OUTPUT (Fluid Annuli Not Included).

```
OUTPUT SUMMARY
--------------
THRESHOLD SPEED = 2.2760E+03 RPM
WHIRL RATIO = 6.3194E-01
RESIDUAL GROWTH = 6.6881E-03
WHIRL FREQUENCY = 2.3971E+01 HZ

---------EIGENVALUES---------
        REAL           IMAGINARY

-0.1966E+05         0.0
-0.6595E+04         0.0
-0.9732E+04         0.0
-0.2950E+04         0.0
-0.1663E+02        -0.1554E+04
-0.1663E+02         0.1554E+04
-0.1631E+02        -0.1540E+04
-0.1631E+02         0.1540E+04
-0.1210E+02        -0.6653E+03
-0.1210E+02         0.6653E+03
-0.1481E+02        -0.6538E+03
-0.1481E+02         0.6538E+03
-0.6688E-02        -0.1500E+03
-0.6688E-02         0.1500E+03
-0.5569E+02        -0.1189E+03
-0.5569E+02         0.1189E+03
-0.5386E+02        -0.1300E+03
-0.5386E+02         0.1300E+03
-0.1893E+02        -0.1443E+03
-0.1893E+02         0.1443E+03
EIGENVALUE-SOLUTION PERFORMANCE INDEX IS GOOD.

SELF-EXCITED VIBRATION MODE
---------------------------
COORDINATE        AMPLITUDE        PHASE(DEG)

        1         0.1000E+01        0.0
        2         0.1127E+01       -0.9666E+02
        3         0.1050E+02        0.2533E+02
        4         0.9603E+01       -0.6369E+02
        5         0.1236E+02        0.2582E+02
        6         0.1128E+02       -0.6306E+02
        7         0.7979E+01        0.2495E+02
        8         0.7331E+01       -0.6194E+02
        9         0.1013E+01        0.5910E+01
       10         0.1177E+01       -0.9579E+02

ROTOR MASS        MAJOR-AXIS        MINOR-AXIS        BETA(DEG)

        1         0.1150E+01        0.9732E+00       -0.6795E+02
        2         0.1051E+02        0.9595E+01        0.5374E+01
        3         0.1237E+02        0.1127E+02        0.6013E+01
        4         0.7980E+01        0.7331E+01        0.6589E+00
        5         0.1227E+01        0.9510E+00       -0.6327E+02
```

TABLE 6: ROTOR/BEARING INSTABILITY THRESHOLD SPEED COMPUTER ANALYSIS OUTPUT (Fluid Annuli Included).

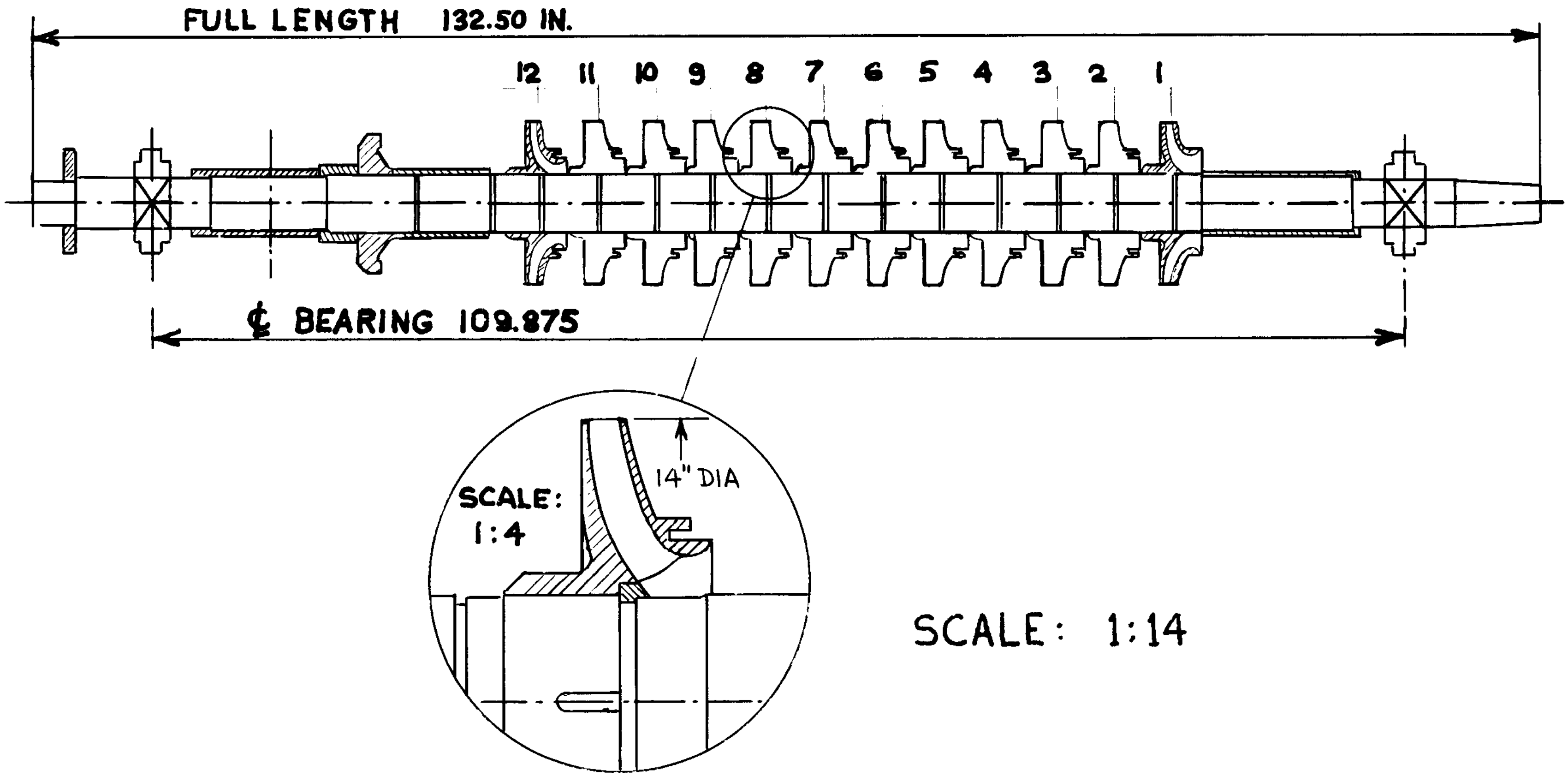
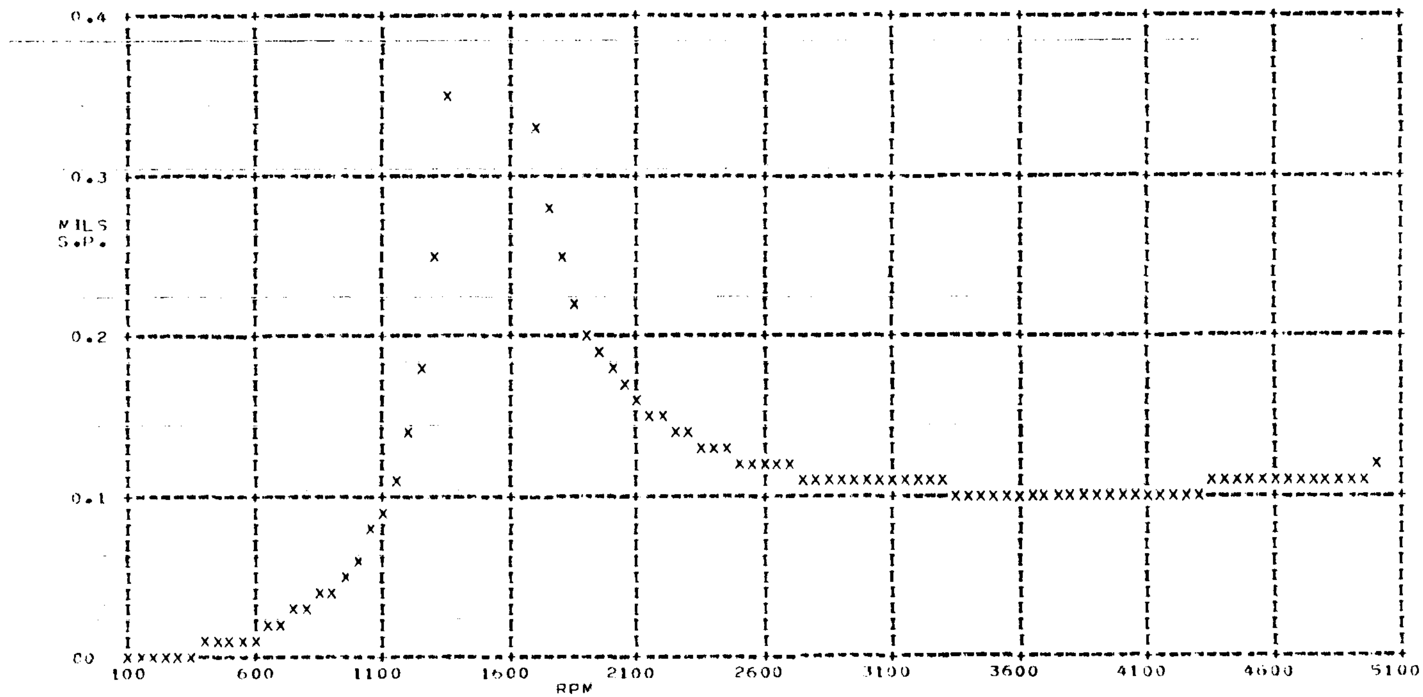


FIGURE 1: ROTOR - CANE RUN NO. 6 (Louisville Gas & Electric) BOILER FEED PUMPS
Mfg. Allis-Chalmers, Type 12x10, Size 35-12 (Twelve-stages) Mathematical Model.



EPRI PROJECT RP-1266-7 ADVANCED ROTOR-BEARING SYSTEMS FOR FEED PUMPS
RUN NO. 1 UNBALANCED RESPONSE WITH PIVOTED-PAD JOURNAL BEARINGS , OCT. 1978 STATION 16

OFF-SCALE POINTS

RPM	X	Y
1400.0000	5.770295E-01	5.770295E-01
1450.0000	1.291867E+00	1.291867E+00
1500.0000	2.969966E+00	2.969966E+00
1550.0000	9.800945E-01	9.800945E-01
1600.0000	5.773039E-01	5.773039E-01
1650.0000	4.187791E-01	4.187791E-01

FIGURE 2: SYNCHRONOUS UNBALANCE VIBRATION, 100 to 5,000 rpm.

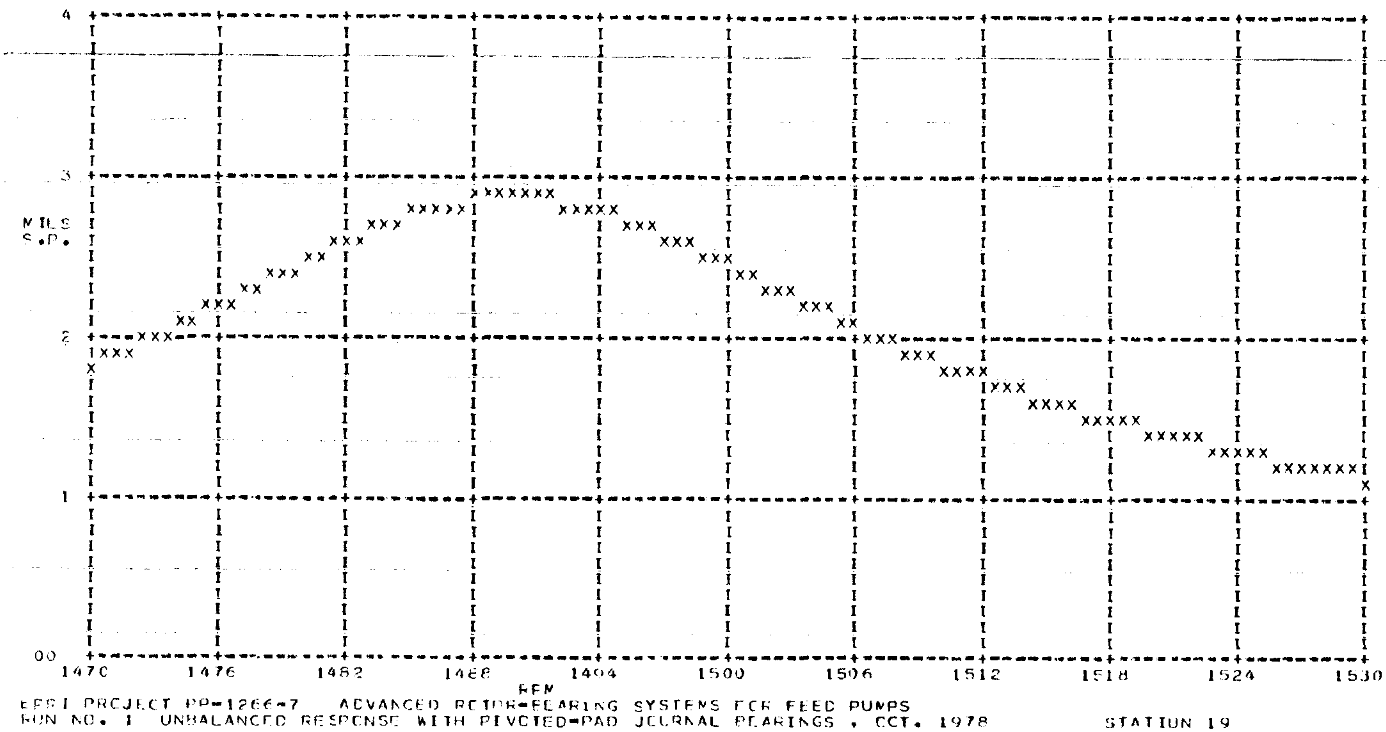


FIGURE 3: SYNCHRONOUS UNBALANCE VIBRATION RESPONSE, 1,470 to 1,530 rpm.

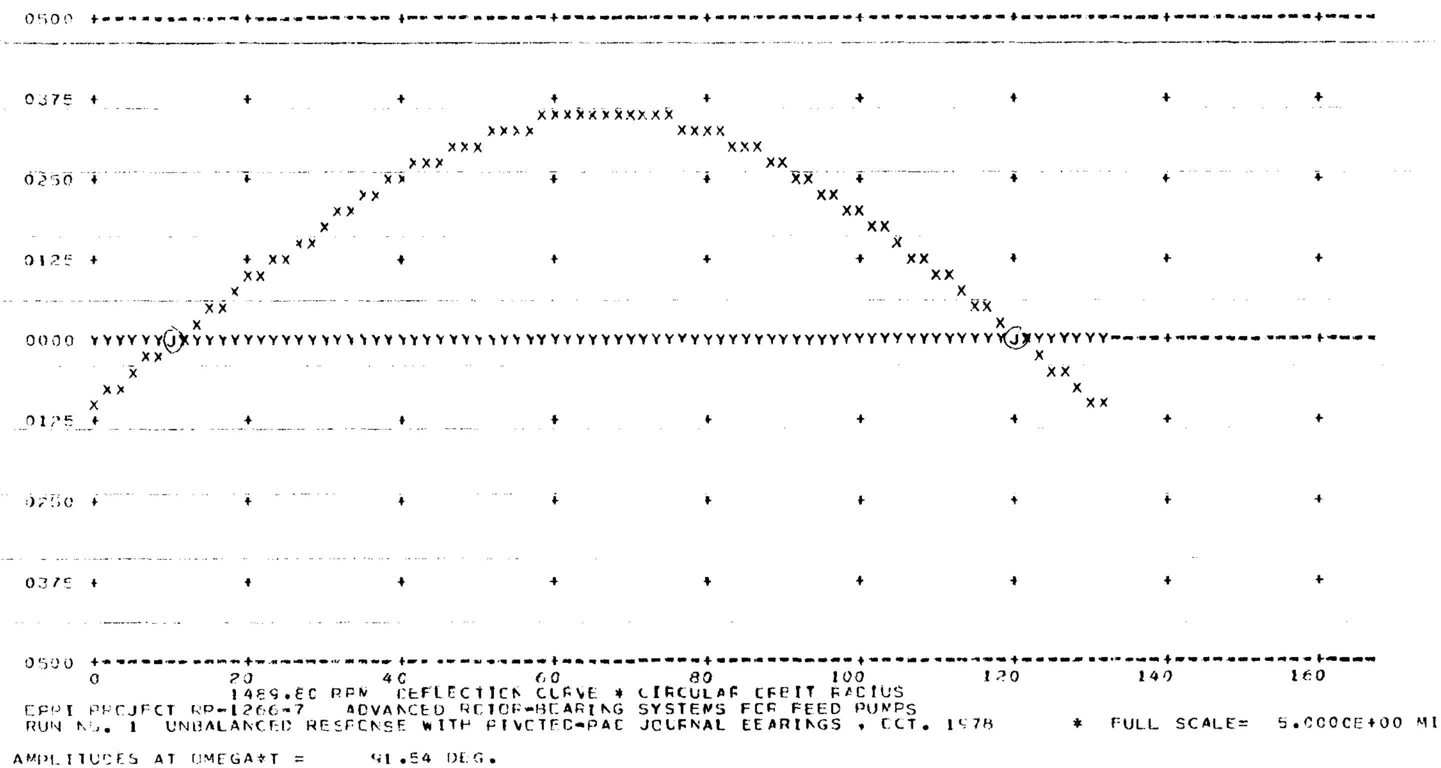
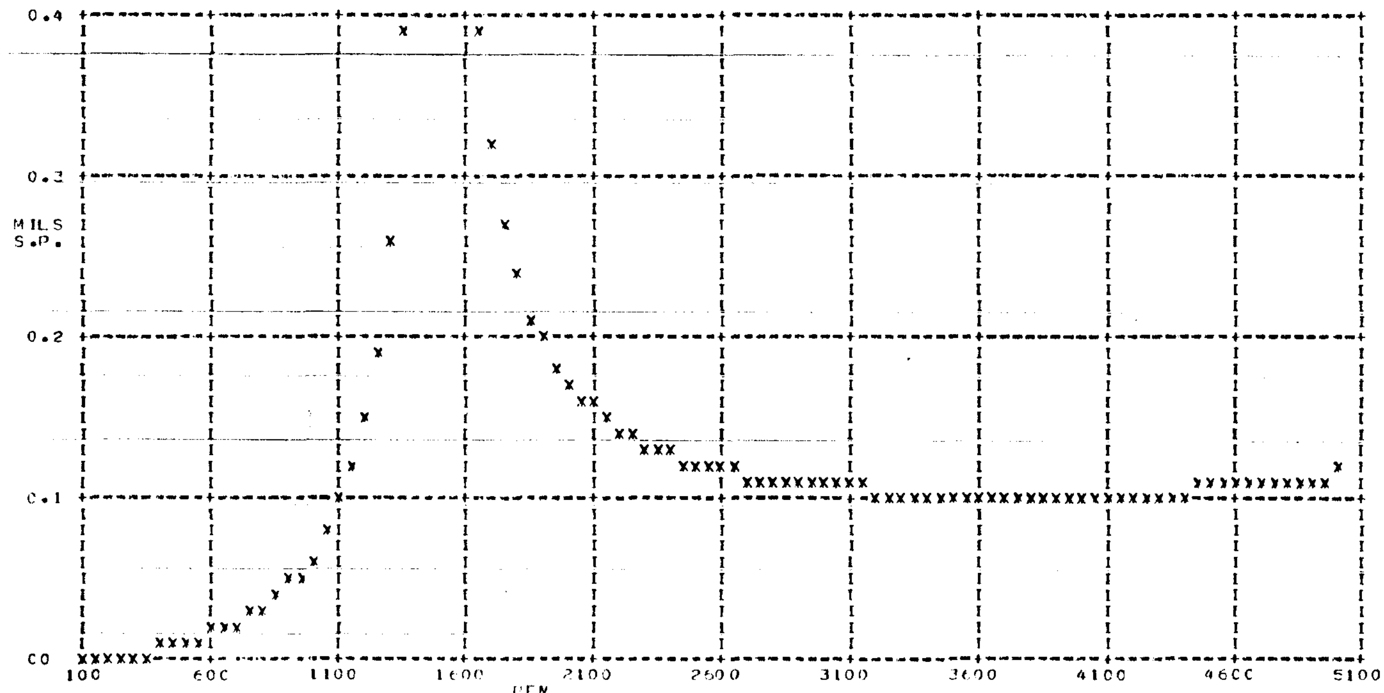


FIGURE 4: FIRST CRITICAL SPEED ROTOR VIBRATION MODE SHAPE.

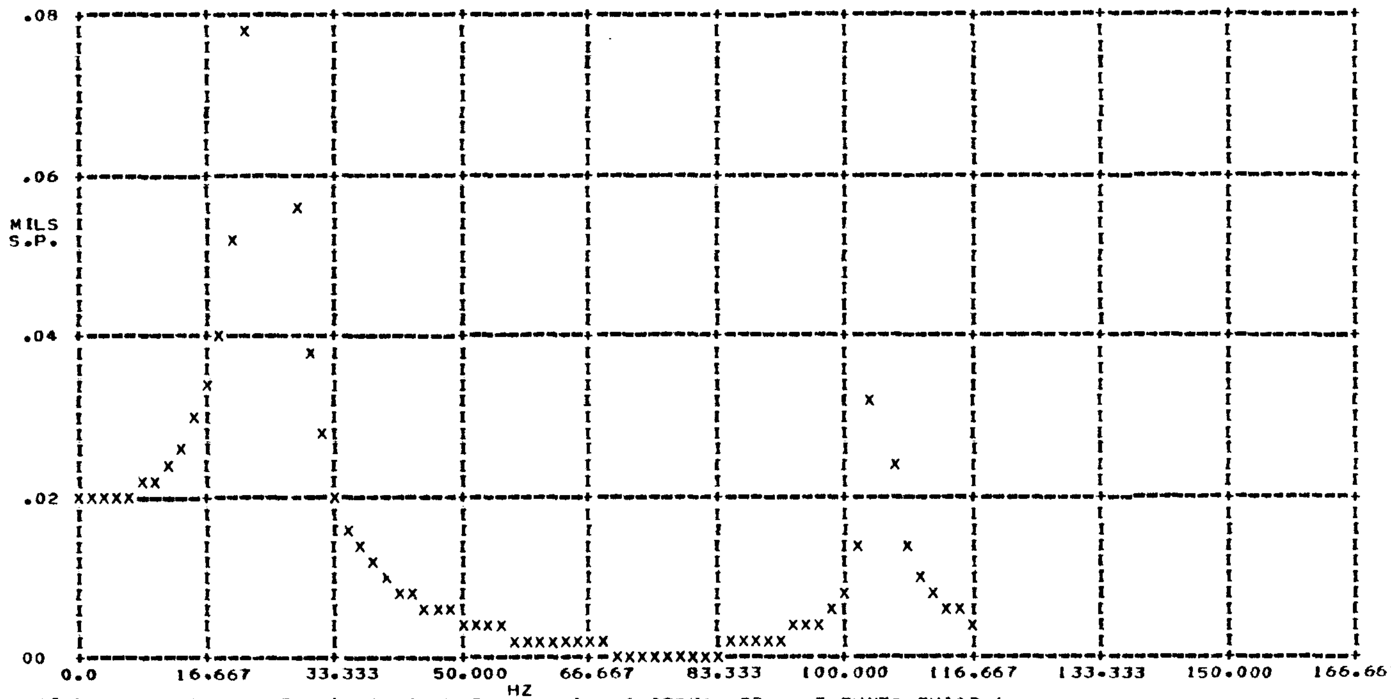


EPRI PROJECT RP-1266-7 ADVANCED ROTOR-BEARING SYSTEMS FOR FEED PUMPS-PHASE I
RUN NO. 2 UNBALANCE,P.P. BEARINGS,DEAD WEIGHT LOAD,CONCENTRIC FLUID ANNULI STATION 16

OFF-SCALE POINTS

RPM	X	Y
1400.0000	6.665516E-01	6.665504E-01
1450.0000	1.742607E+00	1.742605E+00
1500.0000	2.056150E+00	2.056150E+00
1550.0000	8.207642E-01	8.207636E-01
1600.0000	5.195359E-01	5.195366E-01

FIGURE 5: SYNCHRONOUS UNBALANCE VIBRATION WITH FLUID ANNULI INCLUDED
100 to 5,000 rpm.



EPRI PROJECT RP-1266-7 ADVANCED ROTOR-BEARING SYSTEMS FOR FEED PUMPS-PHASE I
RUN 20 NONSYNCHRONOUS RESPONSE WITH PIVOTED-PAD JOURNAL BEARINGS, OCT. 1978 STATION 12

OFF-SCALE POINTS

HZ	X	Y
23.3333	1.626976E-01	1.626976E-01
25.0000	8.470543E-01	8.470543E-01
26.6667	1.099612E-01	1.099612E-01
105.0000	1.126655E-01	1.126655E-01

FIGURE 6: NON-SYNCHRONOUS FORCED VIBRATION, 0 to 117 cps (6,300 cpm).

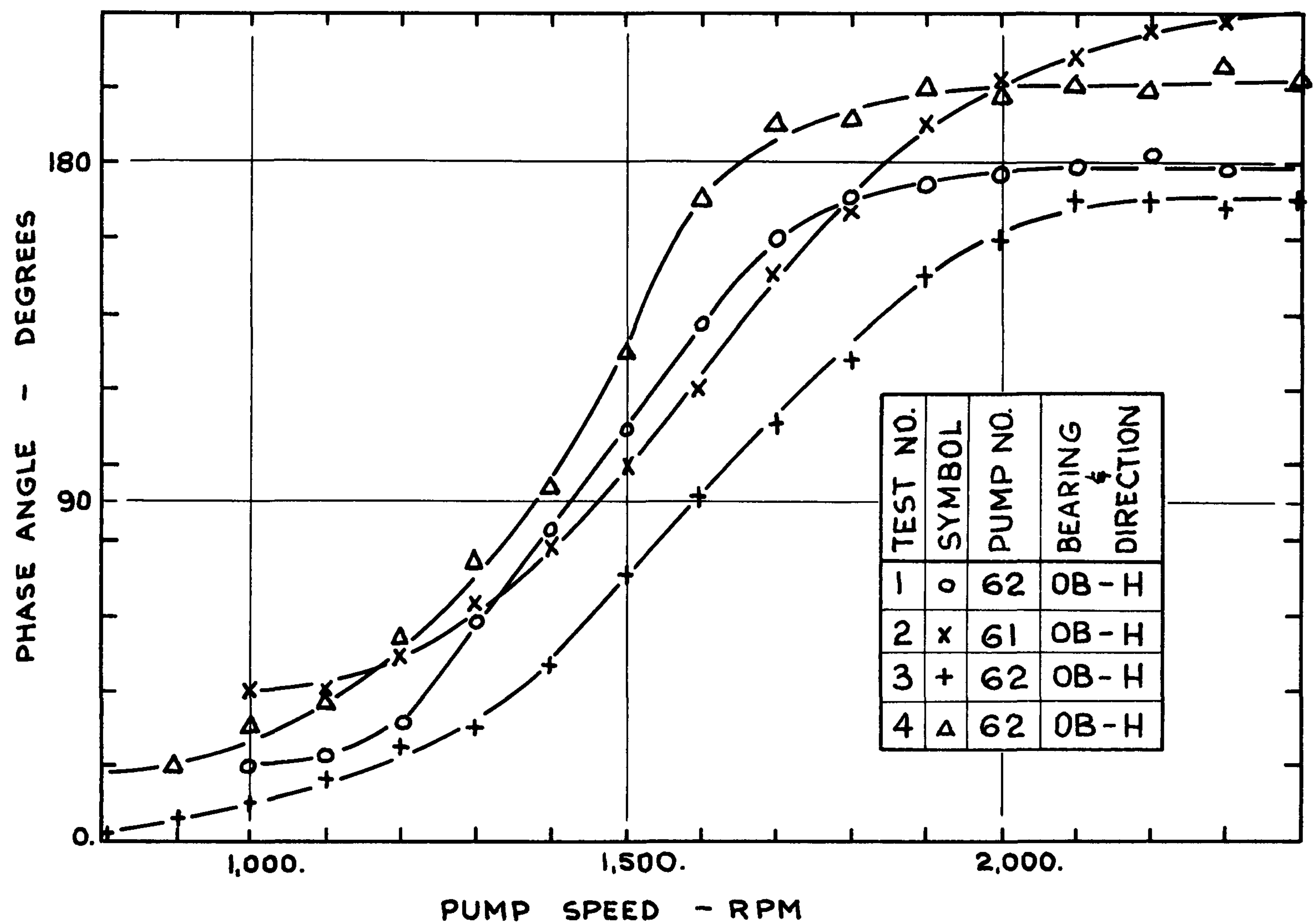


FIGURE 7: FIELD MEASUREMENTS AT CANE RUN NO. 6 FOR THE DETERMINATION OF CRITICAL SPEED.

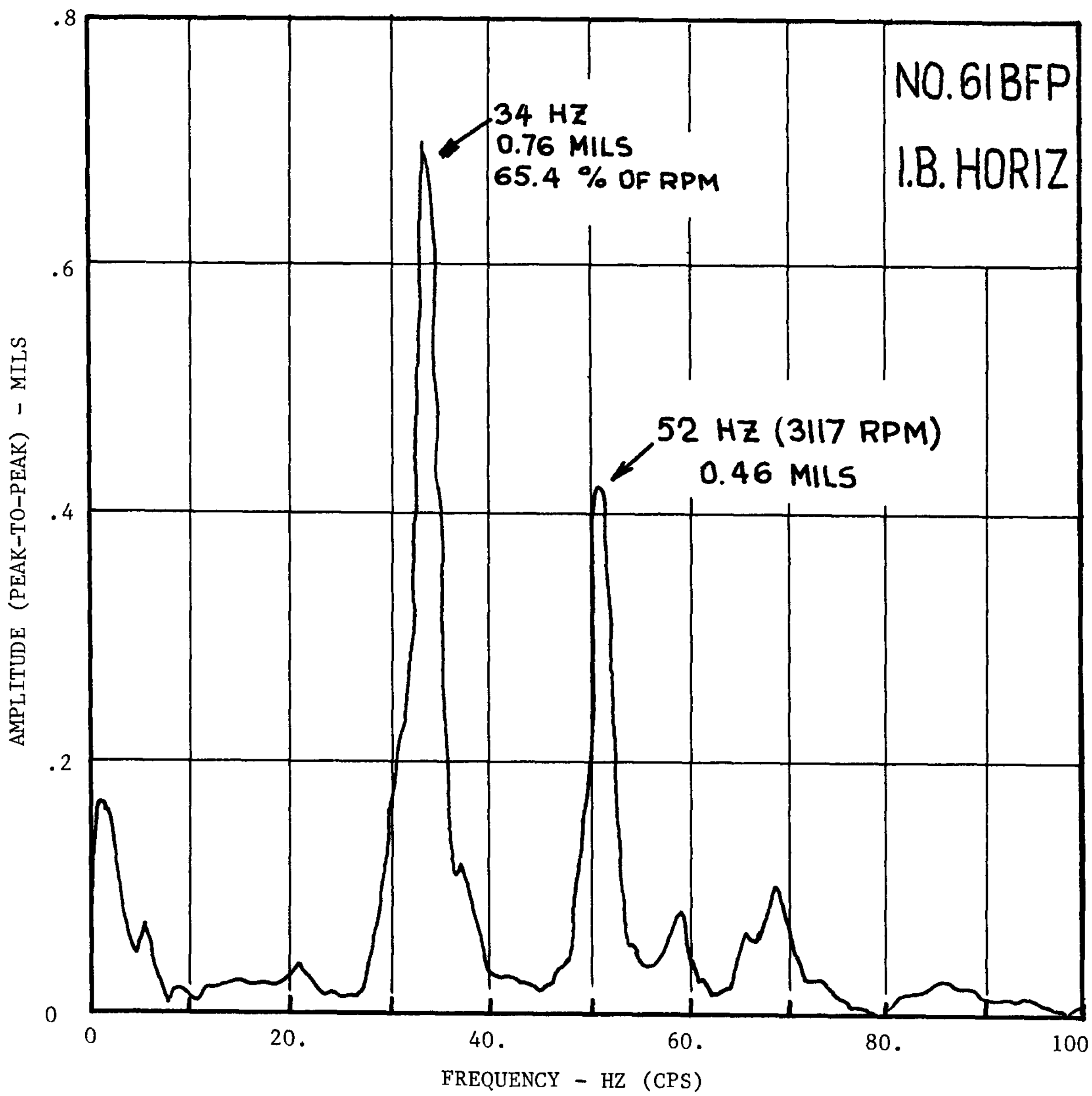


FIGURE 8 : ONSET OF SUBHARMONIC RESONANCE AS DETECTED IN THE I.B. HORIZONTAL COMPONANT OF THE NO. 61 BOILER FEED PUMP AT CANE RUN OF LOUISVILLE G & E CO. THE MAGNITUDE OF THIS SUBHARMONIC COMPONENT RAPIDLY INCREASED TO 7 MILS, NECESSITATING A UNIT SHUTDOWN.

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