MINIMIZING UNBALANCE RESPONSE OF THE CRBRP SODIUM PUMPS

V. GUPTA, F. G. MARRUJO
Byron Jackson Pump Division, Borg-Warner Corporation, P.O. Box 4966, Carson, California 90749, U.S.A.

The unbalance response characteristics of the vertical pumps for the Clinch River Breeder Reactor Plant are investigated. Finite-element shell and beam models representative of the pump-motor structure including the rotating assembly are developed to assess structural stiffnesses of dominant joints as well as the foundation support stiffness so as to exclude the danger of resonant excitation during normal operation. Less than four mils peak-to-peak vibration amplitude at the pump tank discharge nozzle results from just 10% frequency separation between the first rocking mode and the maximum operating speed of 1116 RPM, based on 0.5% modal damping ratio and balance quality grade of ISO/ANSI G2.5 for the rotating components: motor rotor, pump shaft, Bendix diaphragm-type flexible coupling, and centrifugal double-suction impeller.

Several design options are explored for raising shaft critical speed beyond 125% of maximum operating speed. NASTRAN complex eigenvalue solution is performed to take credit for damping in bearings to help meet this requirement. The mitigating effect of increasing shaft critical speed on both unbalance response and seismic response is evaluated. The influence of horizontal foundation stiffness on shaft whip is demonstrated, apart from staggering lateral modes in an effort to minimize dynamic coupling between them; higher the stiffness, less the shaft whip expressed as the peak lateral response along the vertical shaft.

Parametric curves are presented illustrating the extent to which the fundamental rocking mode frequency is influenced by the combined rotational stiffness of the pump foundation and closure flange assembly. The fundamental rocking mode is raised high enough to become less sensitive to the foundation stiffness.

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1. Introduction

Excessive vibration can cause structural fatigue of components in pumps (1), helicopters (2), aircrafts (3,4), rotating machinery (5-8), reactors (9-12), and foundations (13). The major source of vibration particularly in large vertical pumps is the rotor unbalance. Whirling deflections of the rotating shaft due to unbalance cause the otherwise static pump-motor structure to vibrate laterally, most frequently in an elliptical orbit, displacing the hot liquid sodium in the annular space between the lower inner structure and the pump tank. The resulting fluid-solid interactions coupled with the turbulent squeeze-film action in the annular region are a major contributor to hydrodynamic (modal) damping, 2-5% according to some investigators (10-12), while the structural design includes oil-film tilting-pad bearings (14) to dampen the transmission of vibration from the motor rotor to its support structure.

With the objective of balancing flexible rotors, several papers (5,15-19) in recent years have advanced new methods of which some have approached the subject via analytical methods or have employed equivalent geometrical considerations. Standards for rigid and flexible rotor balancing are being developed (20-22).

Experimental investigations of the dynamic response to unbalance of sodium pumps (1,23,24) have given fresh insight into calculating response predictions relevant to damping and nonlinear force-deflection characteristics of sodium hydrostatic bearings (25), in addition to problems of bearing run-outs and thermal bowing. Increasing the bearing stiffness may lower its contribution to modal damping to less than 1% but tends to linearize its linear force-deflection behavior.

In the present paper, a computer-aided structural design method is given which considers a linear structure subjected to synchronous harmonic excitation due to a specified mass unbalance distribution. Finite-element beam/shell models (26-30) representative of the pump-motor structure including the rotating assembly are developed to determine whether or not the natural frequencies are outside the operating speed range. The structural design is then modified to exclude the danger of resonant excitation. While some of the changes in structural design are intuitive or based on geometric or interface dimensional constraints, a more systematic approach will now be presented. The peak-to-peak vibration amplitude at pump tank discharge nozzle (PPN), is examined as a function of various structural parameters such as element mass, stiffness, and strain energy density (strain energy divided by mass). A basic requirement in the optimum dynamic synthesis of the CRBRP Vertical Pump is not to let PPN exceed 0.010 inch (10 mils), in order to limit the piping fatigue damage, as indicated in Table 1, piping loads, and thermal transients.

2. Dynamic Synthesis

Dynamic synthesis generally comprises dynamic analysis (31) and optimum synthesis (32). Design is an iterative process and optimum synthesis seeks to minimize the number of iterations necessary to reach an acceptable design, while parametric studies are conducted to explore feasibility of the design concepts, based on compatible finite-element models.

The problem of structural modification (33) is an optimization problem involving any of a large number of design parameters. The identification of those components and associated stiffnesses which seem to have a marked effect on the dynamic performance of the pump under forced vibration (or seismic excitation) often require not only an understanding of the mechanics of rotor synchronous whirling under the effect of residual unbalance but also recognition of the stiffness paths responsible for transmission of vibration from the motor rotor to pump foundation. Components indicating significant relative deflections in mode shapes demand practical attention. Rotor critical speeds may be raised by reducing thickness or increasing diameter of shaft, lowering operating temperature (higher Young's modulus), and decreasing bearing span. If the optimum structure is a minimum weight structure of specified natural frequencies, a strain energy density approach is often adopted (34); elements possessing maximum strain energy density are stiffened to seek the desired frequency separation between the excitation frequency and the natural frequencies. This and other approaches have been used to identify suitable regions for modification. Regions predicted in the design compared favorable with modifications actually carried out. The method emphases importance of the boundary conditions/interfaces indicated in Fig. 1.

The pump-motor model consists of a large vertical sodium pump (35) (weight 193,000 lbs., C.G. 76.0 in., below ground), a 89,100 lb. motor (C.G. 128 inches above ground), a 24,000 lb. rotating assembly at a maximum speed of 1116 RPM (18.6 cps), and lumped sodium weight (20,000 lbs.) distributed over structural nodes.

2.1 Unbalance Response

Response to rotating unbalance(s) is tantamount to performing frequency response analysis, i.e., to obtaining the steady-state solution to the uncoupled differential equations of motion using the normal mode method (36) for lightly damped systems, assuming that damping matrix is proportional to either mass matrix or stiffness matrix or both. Figure 3 depicts a typical frequency response for a vertical sodium pump, where nodal elevation is measured relative to the nozzle centerline.

Modal analysis using ANSYS resulted in Fig. 2. ANSYS 2-D axisymmetric shell model also provided joint stiffnesses for the 3-D Stardyne colinear beam model used for time-history (modal) seismic analysis. Unbalance response (5, 36-38) procedure was developed for a conservatively preselected mass unbalance distribution using Stardyne beam modes to obtain results of Table III and the frequency response of Fig. 3. In combining the modal responses, modes may be considered either in phase or 180 degrees out-of-phase as goverened by the sign of modal displacement components. Unlike Table III no phase cancellation was allowed for Fig. 3 to obtain an upper bound estimate, nor were the bearing stiffnesses increased beyond those (Table II) prescribed for 18.6 CPS; for Fig. 3, K1 = 70(H), assumed earlier. For speeds less than 18.6 CPS, frequency separation increase rapidly, and the steady-state vibration amplitudes are much lower than Table III values. The effect of modal damping, 0.5% or 2%, on unbalance response at maximum operating speed, i.e., away from resonant modes of Table III and Fig. 3, is relatively insignificant.

2.3 Rocking Mode

Modal dynamic analysis with the Stardyne beam model was used for performing parametric studies such as Fig. 4 to assess the effect of stiffnesses. Table I criteria was met with Table II minimum joint stiffnesses achieved in Fig. 1, by means of systematic design iterations aimed at maximizing stiffness within the spatial constraints. Shell diameters and thicknesses were increased, support ring (K2) was wedged over 21 inch height of the motor support stand and bolted directly to the concrete foundation (K1), to alleviate the overturning moment effect of the 5-inch overhang (radial offset) at K1, among other things. Three dimensional asymmetric aspects of the seal removal window in the motor stand were investigated with an in-house version of SAP IV. Stiffness matrices thus obtained were used in the dynamics model for various different motor stand configurations, to optimally size windows, thickness, and tapers.

Properly tapered wedges, not a spherical mechanism (39), permit one-inch vertical adjustment to maintain controlled clearance (35) in Fig. 1 and thus to compensate for creep over a 30-year life. This is accomplished with a Balance Port concept, without a rigid K5 connection, lowering the rocking mode 0.5 hertz. Without the upper wedge (adjustable shims) and face-to-face contact at the pump tank flange to closure ring interface, K2 drops to less than 500 (R). Then, the radial location of the standoff bolts becomes important from a stiffness and therefore rocking mode frequency standpoint. Placing belleville springs to preload the standoff bolts would have made K2 approach 5 (R).

If stiffnesses K1, K2, K6 in Table II are halved, rocking mode(s) become resonant near 16 CPS and PPN becomes 18 mils, approximately. If K1 (or K2) in Table II is made less than 5 (R), i.e., soft foundation, rocking mode becomes 6.6. CPS and PPN about one mil, offering minimum transmissibility of unbalance moment into pump tank, but required continuous decoupling across the pump tank to inner structure interface, i.e., K5 less than 0.25 (H), which is difficult and impractical because control on alignment without maneuvering response or tuning is lost.

A 10,000 lb/in lateral restraint (equivalent of a seismic snubber) at motor center of gravity will raise the rocking mode to a desired goal of 115% of 18.6 CPS, but may not be an effective solution to the problem for reasons mostly resembling the case of soft foundation.

2.4 Shaft Optimization

Vibratory shaft motion in bearings is minimal, one mil or less, with motor bearings twice as apart as the sodium bearings. Following is a list of few attempts to raise the shaft critical speed, to avoid manufacturing cost and schedule impact.

After each design change follows the result (G, P), where G = CPS (Hertz) gain in shaft critical speed, and P = % of 18.6 CPS. (a) Increase hollow shaft outer diameter from 11" to 13" inches: (3.5, 137%); (b) Decrease bearing span (lower flexible coupling) 4.8": (2.0, 132%) (c) Let shaft seal act as a radial bearing (10,000 lb/in): (0.8, 128%); (d) Model bearings as discrete dampers (NASTRAN complex eigenvalue solution) on grounded supports: (0.5, 127%); (e) Increase KP to 1.0 (H): (0.3, 123%); (f) Bore out shaft another 10" near thermal shield: (0.3, 123%); (g) Decrease middle shaft thickness by 0.25 inch: (0.3, 123%); (h) Lower sodium temperature from 1015 to 650°F: (0.6, 126%); (i) Rigid coupling, KF = 1000(R), 11" for (A): (0.5, 126%), 14" for (A): (1.2, 141%); (j) Soft Bearing, KP = 0.48(H):

(-0.5, 119%); (k) Shaft Mode versus its seismic (OBE) response contribution with Kl = 70(H)

Dia (a)	PPS	P	OBE (MILS)
11"	18	121%	38
13"	10	137%	13

(l) Repeat (d) with Rigid Bearings, KP = 75 (H), KM = 200 (H): (2.3, 136%), generalized stiffness, KG = 93,200 pounds/inch.

Component modes in Table IV, obtained with ANSYS, which share common transmission path, were designed to be well staggered (40,41) to prevent shaft whip, as demonstrated by 2g ($g = 386.4$ in/sec^2) peak time-history shaft acceleration under seismic excitation; it was 20g before the problems with low shear and flexural stiffnesses in the support structure were eliminated. Increasing Kl from 70(H) to 250(H) reduced PPS, defined in Table III, and Fig. 1, from 18 mils to 10 mils, and OBE (Operating Basis Earthquake) shaft displacement relative to mechanical seal from 28 to 18 mils, causing shaft-seal interference to disappear, even for a shaft with only 125% critical speed.

Nevertheless, the importance of higher shaft critical speed, say 135%, cannot be overemphasized, in an effort to facilitate shaft balancing and to alleviate seismic interference, e.g., between shaft and seal, during an OBE; this is most apparent in the foregoing Item (k). The standard one-inch middle shaft thickness is retained when changing outer diameter of the hollow middle shaft shown in Fig. 1 between elevations of upper sodium bearing and thermal shield, in addition to 2-inch diametral enlargement of the solid shaft over 20-inch length above the Bendix Coupling to 10-inches in diameter.

3. Conclusion

Cost-effective unbalance response studies such as this offer great value in structural design. The numerical results reported herein have demonstrated simple means for parameter identification, modification, and optimization, and for enhancing insight into related seismic behavior.

Numerical experimentation with all four computer programs: NASTRAN, ANSYS, SAP IV and STARDYNE may be rationalized on the basis of special economic features and independent model verification to ensure reliable results.

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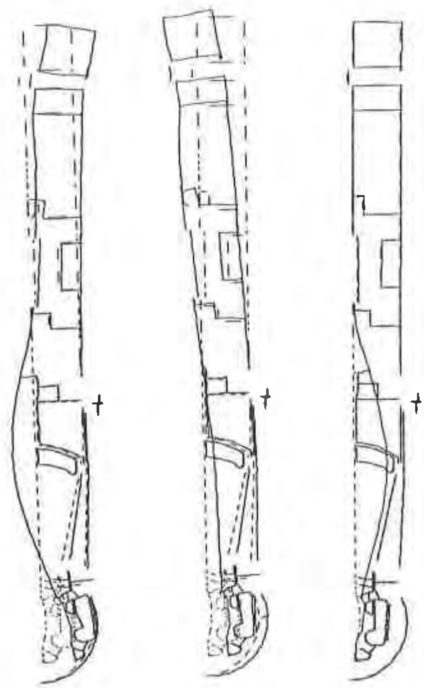

(FLEXURE) (ROCKING) (SHAFT)

Fig. 2 PUMP-MOTOR MODE SHAPES (Ansys)

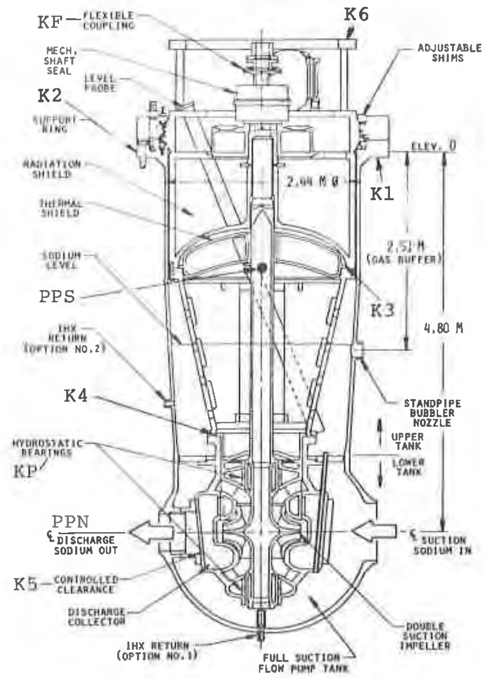

Fig. 1 CRBRP PRIMARY PUMP

TABLE I DESIGN CRITERIA

Operating Speed Range = 40-100 % of 18.6 CPS

Minimum Shaft Critical Speed = 125% of 18.6 CPS

Nozzle (PPN) less than ten Mils

TABLE II DOMINANT STIFFNESSES

R = Rotational Stiffness in 10^9 pound-inch/radian

H = Lateral Stiffness in 10^6 pounds/inch

Spring	Description	Minimum Stiffness
K1	Pump Foundation (ground)	690(R) ,250(H)
K2	Support Ring & Flange	1500(R)
K3	Inner Structure (I.S.)	300(R)
K4	I.S./Hydraulic Assembly	70(R)
K5	I.S./Outer Tank	20(H)
K6	Motor Support Stand Top	1500(R)
KP	Hydrostatic Bearings	0.75(H)
KM	Tilting-Pad Motor Bearings	2.0(H)
KF	Flexible Coupling	10(R)

TABLE III UNBALANCE RESPONSE (AT 18.6 CPS)

PPS = Peak-to-Peak Vibration Amplitude (Mils) of Bulged Shaft

Mode	CPS	PPN	PPS	Component	Unbalance
Rocking	20	2.5	2	Pump Shaft	60
Shaft	23	0.2	13	Impeller	30
Flexure	25	0.5	1	Motor Rotor	235
Vector Sum		3.0	10	Total (ounce-in)	325
Modal Damping		0.5-2%			

TABLE IV COMPONENT MODES (ANSYS)

Component	Spring Fixed	Lateral Mode,CPS
(a) Hydraulic Assembly	K4	55
(b) Pump Tank (P.T.)	K1	42-48*
(c) Inner Structure (I.S.)	K2	33**
(d) (b) + (c)	K1	37 (I.S.),77(P.T.)
(e) (b) + (c)		22
(f) Motor Reed	K6	38
(g) Pump Foundation		
K1 = 70 (H)		30
K1 = 250 (H)		60

* If tank (nozzle) piping is removed.

** If K5 is removed, decoupling I.S. from P.T.

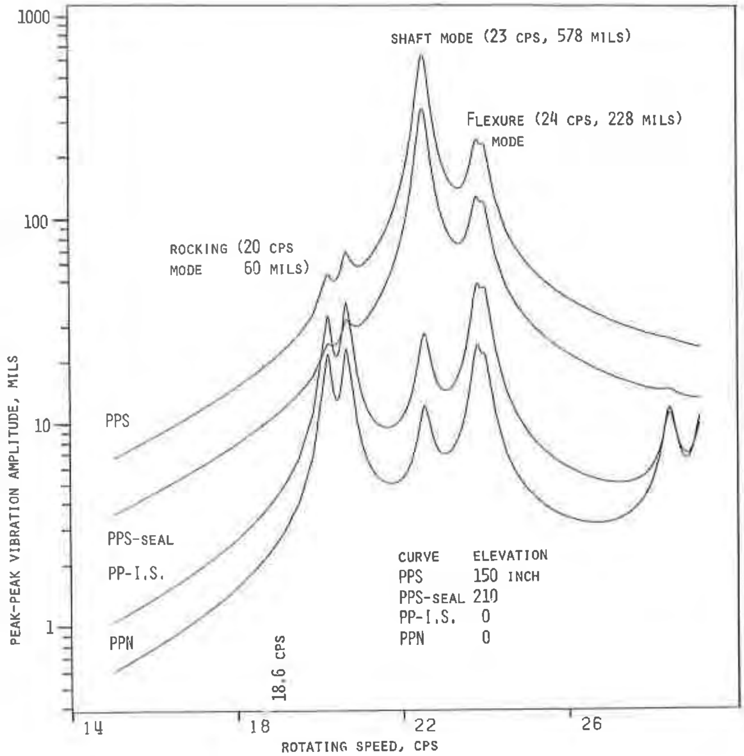

Fig. 3 FREQUENCY RESPONSE - VERTICAL SODIUM PUMP

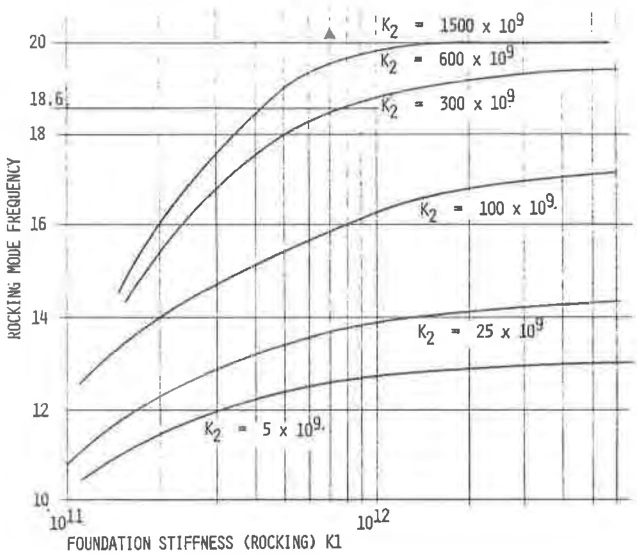

Fig. 4 PARAMETRIC CURVES: ROCKING MODE Versus K1-K2(R)