SITE BALANCING OF A LARGE FLEXIBLE ROTOR CONTAINING UNBALANCE ECCENTRICITY AND PERMANENT RESIDUAL BOW

by

A. S. Maxwell

Maintenance Specialist, Balancing and Vibration

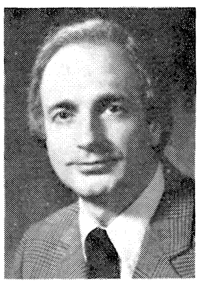
and

A. F. P. Sanderson

Maintenance Engineer, Mechanical System Maintenance Division

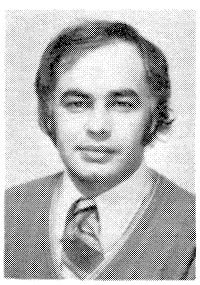
Ontario Hydro

Toronto, Ontario, Canada



Stewart Maxwell was educated at the Royal Technical College, Salford England and is a registered professional engineer of the Province of Ontario. He joined Ontario Hydro four years ago and is presently responsible for providing advice, guidance and assistance to personnel in fossil and nuclear stations and other areas of Ontario Hydro operations, on all aspects of maintaining plant and equipment in satisfactory operating condition from the viewpoint of vibration and balancing. His responsibilities include development and implementation of methods and techniques to ensure the corporation is kept at the state of the art in these areas.

After six years in the aerospace industry in the field of stress and vibration analysis, he spent nine years with Westinghouse Canada as a research and development engineer in experimental stress and analysis, noise and vibration and as a senior design engineer in the Turbine Engineering Department.



Alan F. P. Sanderson was educated at Newcastle-upon-Tyne Polytechnic, England. He is a graduate member of the Institution of Mechanical Engineers. Serving as a research engineer in the Machine Dynamics laboratory of C. A. Parsons and Company for five years, Mr. Sanderson was primarily interested in vibration problems associated with large generator stators and flexible rotor balancing. He was subsequently transferred to Canada to take responsibility for field balancing in North America. Mr. Sanderson now works for Ontario Hydro, where he is involved in the maintenance aspect of vibration analysis on steam turbine generators and rotating equipment.

ABSTRACT

When turbine-generators operating at constant speed require site trim balancing, this is most commonly performed at the synchronous speed using a conventional influence coefficient method. In some cases, the vibration of flexible rotors during run-up through critical speeds is excessive and balance can only be achieved by correcting in turn each model defect throughout the speed range. Difficulties arise in the case of rotors that contain a permanent residual bow. Classical phase/speed relationships through a critical speed no longer apply and conventional balancing techniques are invalid. The theoretical explanation of the unbalance response of bowed flexible rotors has recently been well documented. This paper describes how a flexible rotor containing a significant bend was optimally balanced over its complete speed range, by incorporating bowed rotor response theory into the balance technique.

INTRODUCTION

The primary objective of the authors in the execution of site balancing work is an overall increase in reliability and reduction in maintenance costs and outage times. The work that will be described is part of a more general program to improve accuracy and effectiveness of site balancing flexible steam turbine-generator rotors. It was felt, for example, that more use should be made of computers for such things as optimizing weight corrections between several planes which might have conflicting individual balance requirements. Also, the advent of hand held computers or calculators should make hand graphical computations and constructions on polar graph paper obsolete. Obviously, in such a system, the use of stroboscopic lights would be inadequate for phase measurement. Although the limited number of measuring and balancing planes available would preclude true, in place, modal balancing, it was nevertheless felt that the full potential of site balancing was not being utilized.

SITE BALANCING PHILOSOPHY

If we accept the premise that it is theoretically not possible to exactly site balance flexible rotors, then the practicing engineer who is continually facing this dilemma must address himself to the question: "What is the best that can be done with the tools available?" He does not really care whether his solution is mathematically elegant or not, but whether it is possible for him to correct balance defects at site sufficiently that the machine will be in a completely satisfactory condition for unrestricted operation over its specified speed and load range and duty cycle.

The procedure adopted by the System Maintenance Division of Ontario Hydro to achieve this consists of performing optimum correction of unbalance by operating the machine through its complete speed and load range and dealing with each modal defect in turn.

There are three essential ingredients in this prodedure:

1. A proven and rigorously followed technique with strict adherence to sign convention and repeatable measuring conditions.

2. Precise and convenient measuring instruments and transducers.
3. The ability to identify the model defects affecting each rotor span and to make the appropriate corrections.

In most cases these procedures will produce acceptable results. However, difficulties still arise in the case of rotors that have significant distortion of their elastic center, and it is with the resolution of this problem that this paper will deal.

DISCUSSION OF THE PROBLEM

In general, a bent rotor in a continuous train of solidly coupled rotors will result in at least one journal running eccentrically during slow roll. This eccentricity may be small in comparison to the maximum eccentricity that would be observed at the rotor's mid span if such a measurement were possible, but in the practical case this is usually the only indication of the rotor slow roll shape available for direct comparison with the rotor shape during running. The relationship between the mid span eccentricity and the journal eccentricity will be complex depending on the concentricity of adjacent couplings as well as relative rotor stiffnesses and weights and bearing loadings. In order to completely balance the bent rotor it is not necessary to know what this relationship is. Theoretically, the same balancing logic applies to the quite different cases of:

1. A rotor that is significantly bent because of distortion during service.
2. Straight rotors with significant coupling concentricity errors, causing lateral offset of a rotor span without any imposed bending.
3. Normally straight rotors with imposed bending moments due to coupling face errors (coupling faces not perpendicular to shaft axis).

All three of the above cases result in constant bearing forces related to stiffness as well as dynamic forces caused by unbalance. The forces due to any or all of the above effects cause a constant slow roll eccentricity and the contributing components of each need not be known. The balancing process is aimed at removing only the dynamic forces. This can be done quite successfully even with a combination of the above listed defects.

It is axiomatic among most balancing engineers that attempts to balance a particular rotor with known misalignment will result in either moving the high vibration to an adjacent bearing or will achieve reduction of vibration at the chosen balancing speed at the expense of higher vibration at other speeds, especially at criticals. We believe that the above observations occur because part of the permanent slow roll eccentricity has been compensated for during the attempted balance and that this is done unknowingly and without an understanding of the relationship between slow roll eccentricity and elastic unbalance. This is the case particularly when balancing is done based on bearing housing vibration. The fact is that there is a limit to what can be acheived by balancing a rotor with a permanent slow roll eccentricity. This limit is the removal of dynamic forces only, leaving unaffected the eccentricity that was measured during slow roll at each journal. The ideal result would then be vibration equal to the constant value of slow roll eccentricity throughout the speed range of the machine.

It should always be remembered, however, that in spite of the above assertions, the preferred solution to the problem of slow roll eccentricity is to rectify the fault directly at its source. However, where the only recourse is to balance a machine until the defect can be corrected at an overhaul or where it is judged to be cost effective to operate the machine indefinitely with the slow roll defect, and understanding of how optimally to balance the rotor is of great benefit.

THEORETICAL BACKGROUND

It was as a direct result of the work contained in a course (1) given by Professor E. J. Gunter and his associates at the University of Virginia, that the response of bowed flexible rotors was understood and described sufficiently clearly that this effect was included in the balancing program with confidence.

At this time a computer program was being written to extend the conventional two-plane theory of Thearle, (2) to four planes, and it was to this program that Dr. Gunter's work was adapted.

A brief summary of the theory is provided as follows:

The general expression for rotor response due to unbalance and slow roll eccentricity is shown below where x identifies the bearing number, y the balance plane, and n, the run number.

$$\left\{ z_{xn} \right\} = \left\{ a_{u_{xy}} \right\} \left\{ \hat{e}_{u_y} \right\} + \left\{ a_{r_{xy}} \right\} \left\{ \hat{\delta}_{r_x} \right\} \qquad (1)$$

X = 1, 4 Y = 1, 4
n = 0, 4

$\left\{ a_{u_{xy}} \right\}$ is a 4 x 4 matrix

z_{xn} = Complex shaft amplitude of bearing x during run n

$\hat{e}_{u_y}$ = Complex unbalance eccentricity in Plane y

$a_{u_{xy}}$ = Complex influence coefficient relating unbalance in Plane y to the response at Bearing x

$\hat{\delta}_{r_x}$ = Complex slow roll eccentricity at bearing x

$a_{r_{xy}}$ = Complex influence coefficient relating slow roll eccentricity at Plane y to the response at Bearing x

Study of Dr. Gunter's work shows that in a rotor that has a high shaft stiffness compared to its modal mass, the residual bow or slow roll eccentricity complex influence coefficient approaches unity and becomes independent of speed. It has been our experience that this assumption is valid for large steam turbine rotors with mid span bow eccentricities of up to .005 inch and speeds of up to 3,600 RPM. It agrees with our basic objective in balancing bowed rotors which is to balance only the elastic deflection to zero and not to impose any internal forces or moments on the rotor. Again, the aim is to achieve a condition where the shape of the bowed rotor at all speeds is the same as its shape during slow roll.

Following the assumption that the residual bow coefficient is equal to unity, then the residual bow may be subtracted from the rotor amplitude. Thus for the initial run at balancing speed, from Equation (1).

where $$\left\{ z_{xn} \right\} = \left\{ a_{u_{xy}} \right\} \left\{ \hat{e}_{u_y} \right\} + \left\{ \hat{\delta}_{rx} \right\}$$

then $$\left\{ z_{xo} - \hat{\delta}_{r_x} \right\} = \left\{ a_{u_{xy}} \right\} \left\{ \hat{e}_{u_y} \right\} \qquad (2)$$

x = 1, 4 y = 1, 4

$\hat{\delta}_{r_x}$ is determined experimentally by running the machine at a very low speed where shaft elastic deflection may be considered negligible.

For the calibration runs:

$$\left\{z_{xn} - \hat{\delta}_{r_x}\right\} = \left\{a_{u_{xy}}\right\}\left\{e_{u_y} + \hat{e}_{CAL_y}\right\} \qquad (3)$$

$n = 0, 4 \quad x = 1, 4 \quad y = 1, 4$

where $\hat{e}_{CAL_y}$ = Complex Calibration Weight Eccentricity in Balance Plane Y.

By subtraction: $z_{xn} - z_{xo} = a_{u_{xy}} \hat{e}_{CAL_y}$

or

$$a_{u_{xy}} = \frac{z_{xn} - z_{xo}}{\hat{e}_{CAL_y}} \qquad (4)$$

The Z terms are measured or derived during the calibration runs, $\hat{e}_{CAL_y}$ is known, therefore $a_{u_{xy}}$ can be calculated and equations (2) solved.

In practice, since each term in equation (2) is complex, they are separated out into real and imaginary parts in order to obtain eight linear simultaneous equations for programming convenience.

In extending this theory to four planes and multi-span rotor systems it also has to be extended to accept the concept of modal correction. This is done by maintaining a coherent system of Z terms and $\hat{\delta}_{r_x}$ terms. If balancing is concerned with correcting particular modes that have been isolated from analysis of the vibration vectors from each rotor span, the theory can still be applied if the modal components are used consistently throughout the computation in the same way that individual vectors are normally used. The modal components of the terms used are substituted for the terms themselves, and the solution obtained is a solution for the relevant mode only.

DEVELOPMENT OF THE METHOD

The method described above was first used on a rotor test kit shown in Figure 1 having a deliberately bent and unbalanced rotor, three bearings and three balance planes. The results for Bearing No. 2 are shown in Figure 2; the response at Bearings

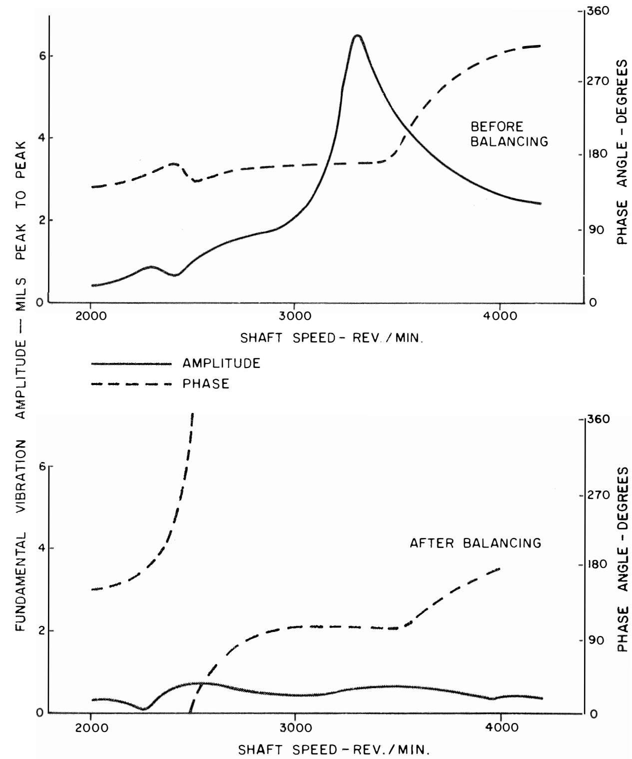


Figure 2. Typical Response of Rotor Test Kit.

No. 1 and 3 were similar. These tests were done primarily to check the computer program and to confirm that Dr. Gunter's theory had been correctly transposed to our instrumentation and computer system. The reason that the results achieved here are not exact is mainly that very small weights were involved and the required accuracy in obtaining the precise weights at the precise angular location was not pursued. Also, there was some problem with resonances in the test kits' foundation.

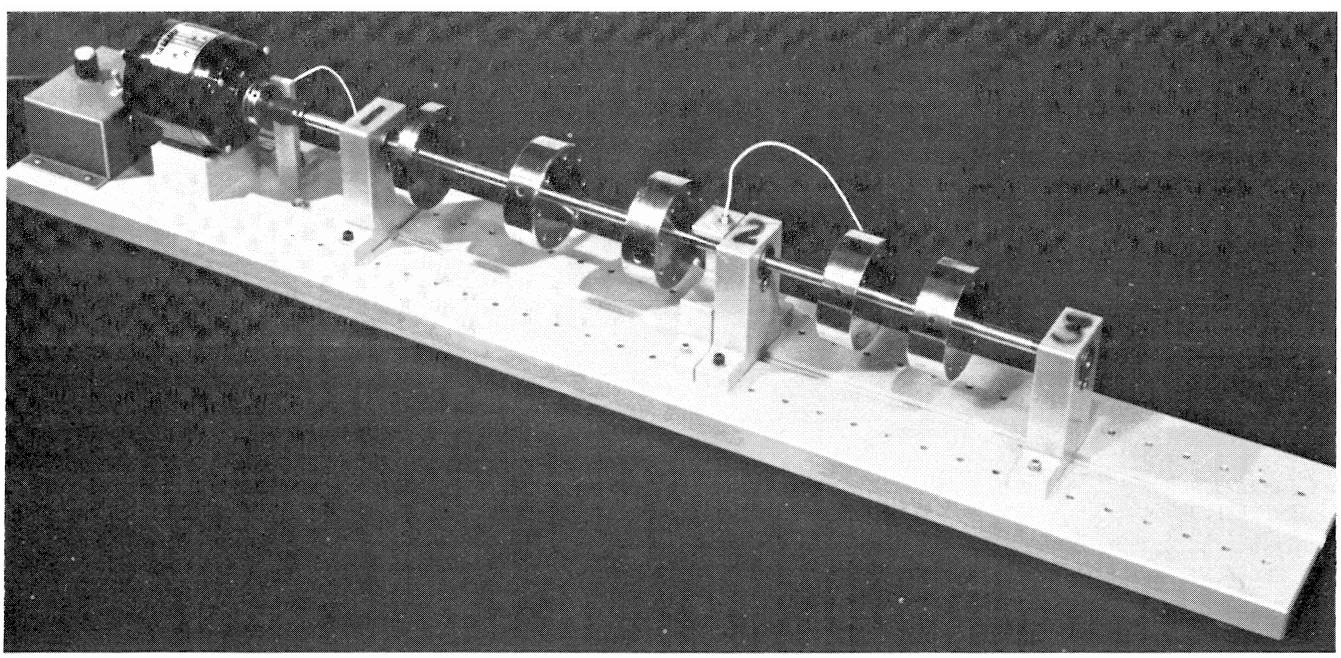


Figure 1. Test Rotor Containing Permanent Bow and Unbalance Eccentricity.

APPLICATION OF THE METHOD TO A LARGE TURBINE ROTOR

The addition of the residual bow effect in the work described was not done solely out of a desire for academic thoroughness. There was the suspicion that a permanently bowed rotor existed in one of Ontario Hydro's large turbine-generator units. Shortly after the test kit work was done it was reported that a 500 MW turbine-generator was experiencing excessive vibration at the intermediate pressure rotor bearings during run up through its first critical speed and that steady state vibration levels at sychronous speed were at the allowable limit. This machine consists of a rigid high pressure rotor, and intermediate pressure rotor operating between its first and second critical speeds, two low pressure rotors in tandem, and a generator rotor each operating above its second critical speed. The intermediate pressure rotor that was suspect, contains a site accessible mid span balance plane as well as rotor body end planes and balance planes in the couplings. The rotor is shown in Figure 3; its weight is 45,000 pounds; the distance between journal centers is 200 inches, the journal diameters are about 16 inches and its maximum diameter is 53 inches.

FIRST SITE BALANCING RUNS

The machine was allowed to cool out and cold datum response curves of amplitude and phase were obtained during a slow, controlled run to speed. The amplitude and phase response curves for both journals are shown in Figure 4. All vibration levels quoted refer to shaft seismic or absolute values. The Nyquist plot, shown in Figure 5, is a polar plot of the vibration vectors at Bearings 3 and 4 over the speed range 1,200 RPM to 3,600 RPM. Only the quadrants 0°, 90°, 180°, and 270° have been shown for clarity. The numbers at the various points refer to the speed of the machine in hundreds of RPM. Thus, point 12 on the full line graph represents a vibration of 2.0 mils at a phase angle of 180° at 1,200 RPM. Similarly, point 22 shows a vibration vector of 12 mils at 280° and 2,200 RPM, and so on. All the information on amplitude, speed and phase for the two bearings that is shown in the more conventional form in Figure 4, is contained in these Nyquist plots. The critical speed can be seen as the maximum vibration vector occurring at the point of maximum rate of change of curvature and at about 90° phase change. The relative magnitudes and phase at each bearing can be compared directly,

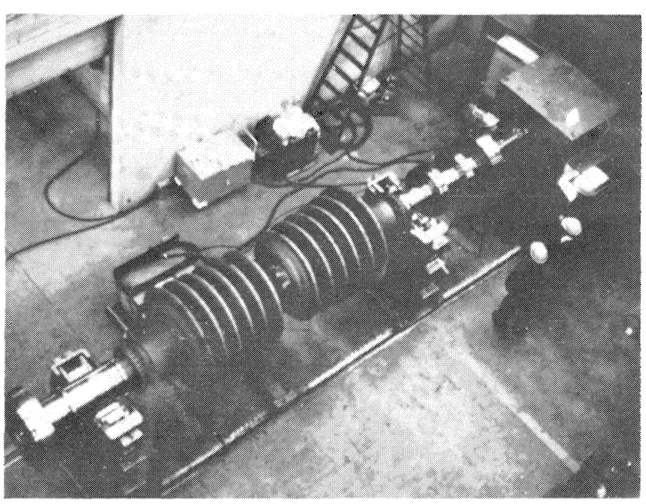


Figure 3. Intermediate Pressure Rotor of a 500 MW Turbine-Generator Containing Permanent Bow and Unbalance Eccentricity.

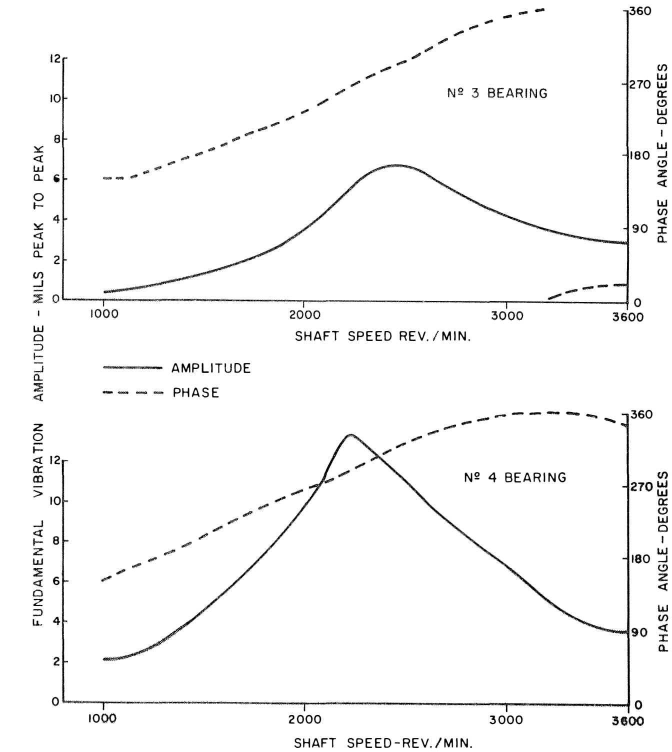


Figure 4. Amplitude and Phase Response with Speed of the Bowed Rotor as Found.

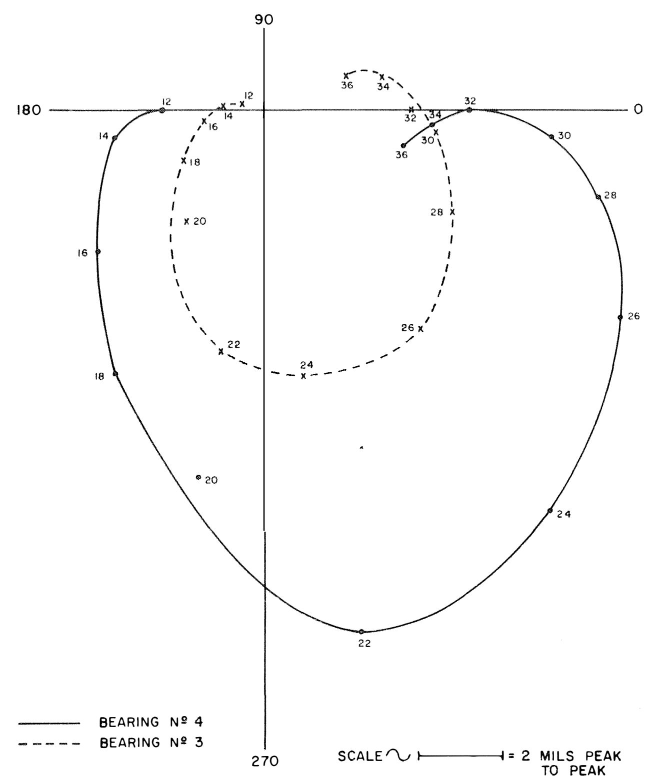


Figure 5. Nyquist Plot of the Bowed Rotor Showing Severe First Mode Defect.

and it is clear that both ends of the rotor are sensibly in phase and that these indicate a considerable first critical speed mode defect.

Prior to this date two balance corrections applied to the mid span balance plane had been made by the turbine manufacturer, but these had been aimed at reducing the response at synchronous speed of 3,600 RPM only. Some balance weights had also been added to a coupling plane outboard of the main rotor span. This latter practice is an expedient that is sometimes preferred because of ease of access, but we will discover later how this approach is incompatible with the concept of modal defect correction, and in fact can distort the rotor response to such a degree that it can make modal separation and optimum correction impossible until these weights are removed. At this stage, the rotor was only suspected of having a bend and an attempt was made to balance what the response indicated to be a severe first mode defect, using the mid span plane.

The result of this exercise is shown in Figure 6. A balancing speed of 1,700 RPM was chosen and mid span corrections for the first mode defect were applied. Here, the very high peak at the critical speeed has been eliminated but the vibration at synchronous speed has increased. At this point the behavior of the rotor was puzzling; logic insisted that the first mode correction should not have upset the balance at 3,600 RPM to the extent that it did. Couple corrections were applied to reduce vibration to acceptable limits at 3,600 RPM and this level further reduced with load. The machine was temporarily left in this condition since it was due for a major overhaul, at which time the extent of any bow would be measured accurately and remedial procedures would be applied at that time. The results obtained were analyzed in the meantime. The shape of the response curve is similar in form to a bowed rotor response curve of Professor Gunter's where the bow is undercorrected, the evidence of this is the dip close to zero amplitude at 2,950 RPM, above the critical speed. Although this condition of an undercorrected bend was suspected at the time of balancing, any further increase in mid span first mode correction weights produced intolerably high vibration at 3,600 RPM. Subsequent analysis of the effect of the coupling weights that had been applied during previous balancing showed that these weights were the reason that a better compromise between first mode defect and operating speed balance defect could not be achieved.

LOW SPEED BALANCING DURING OVERHAUL

Mid span run out readings made while the rotor was in a low speed balancing machine showed a total indicator reading (TIR) of .009 inch and, of course, the angular location of the high spot. At this time the rigid body modes were corrected in the balancing machine with corrections distributed between the mid span and end planes according to a formula provided by the manufacturer. It was not expected that this procedure would completely correct the rotor because its bowed shape and response in its own bearings would be significantly different to that in the balancing machine, and because no elastic first mode deflection could be achieved. Balancing was done at 200 RPM. The low speed balancing was also useful in allowing balance weights to be installed in the factory planes, leaving the field access balance planes available.

The response of the rotor after low speed balancing is shown in Figure 7. The dip in the curve at 3,000 RPM represents a soaking period to allow an increase in rotor bore temperature to reduce stress levels. This was not necessary during other runs. This response curve is included to show the amplitude/phase relationship at 1,950 RPM and because it is

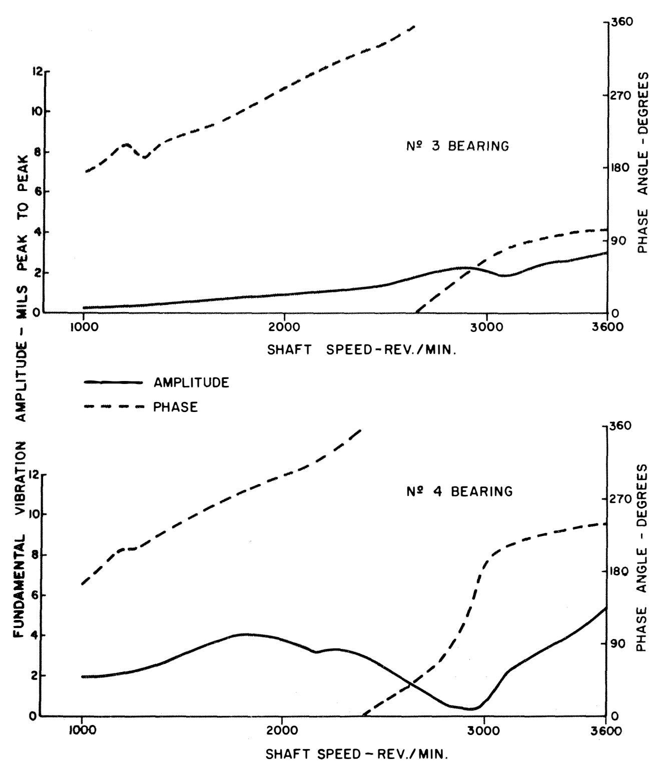


Figure 6. Response of Bowed Rotor After Attempted Conventional Site Balancing Procedure.

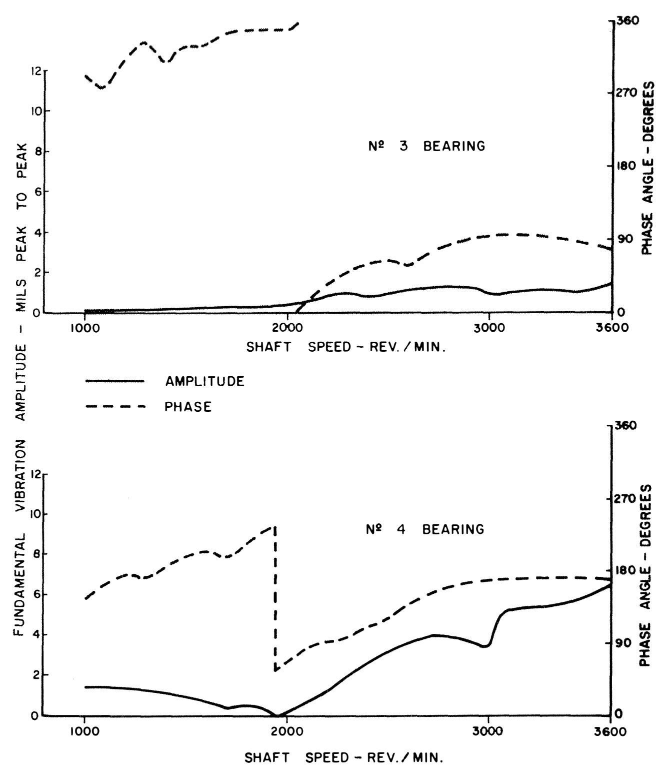


Figure 7. Response of Bowed Rotor After Low Speed Balancing During Overhaul.

the actual run up after low speed balancing. It shows this time, and overcorrected bend according to Gunter's work since the response dips to zero below the critical speed. More importantly, it shows how incorrect it would be to balance only at speeds up to 1,900 RPM because of the subsequent response at higher speeds. The Nyquist plot in Figure 8 shows this dramatically. Data from this run were not used for the balancing computations because of the transient conditions. Repeatable conditions at balancing speeds of 1,700 RPM and 3,600 RPM were established for balancing purposes.

INSTRUMENTATION

In preparation for inclusion of the residual bow effect, proximity probes were mounted at the rotor Bearings No. 3 and 4 (and also No. 5, the inboard low pressure turbine bearing adjacent to Bearing No. 4). The decision and opportunity to include the slow roll eccentricity effect was made only two days before balancing was to commence; consequently the test fixture design is not very sophisticated. Throughout the balance runs, journal seismic response was measured by mounting a velocity transducer on top of the supervisory transducer which observed shaft motion via a spring loaded shaft rider. In the case of the slow roll eccentricity measurement, a proximity probe was mounted on the bearing cap to observe a target that had the same motion as the shaft riding probes. This arrangement is shown in Figure 9.

A commercial digital phase meter containing some minor custom features was the primary read out device. This is shown in Figure 10. This instrument employs two synchronous tracking filters and provides a digital readout of speed, amplitude and phase of the once per turn vibration component. The velocity signals were integrated to read out in mils peak to peak and the phase convention and display was the same for both types of transducer. One advantage of this instrumentation system may not be readily obvious but is of major importance. There is no phase lag introduced into the measuring system by the instrument. It gives true phase lag angle from a predetermined angular datum to the rotor physical high spot at all speeds. This means, for example, that slow roll bend eccentricities which were measured with a dial gauge and referred to the phase datum manually, can be directly compared to the displacement and phase angle shown on the meter. They are the same measurement and will read the same angle at machine speeds that produce negligible elastic shaft deflection. The instrument phase angle reading is an absolute value that allows the physical position of the rotor high spot to be visualized during the tests.

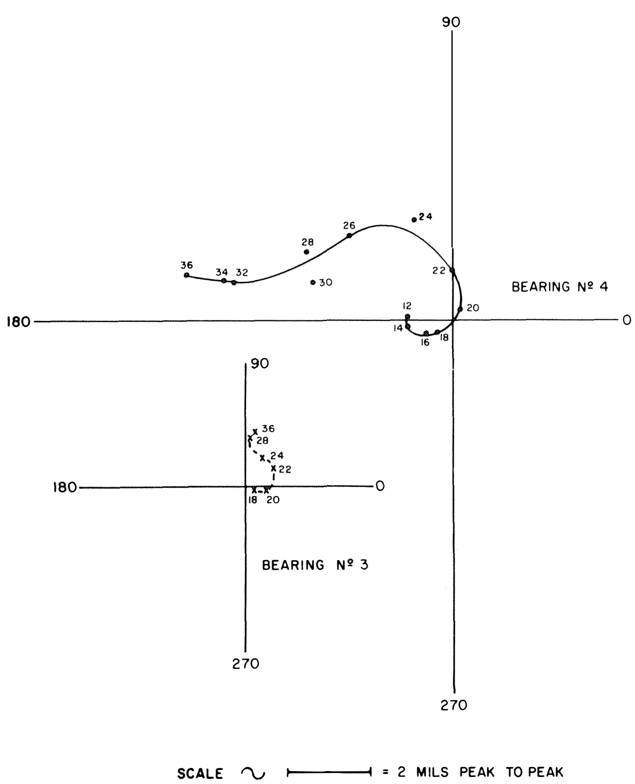


Figure 8. Nyquist Plot of Rotor Response After Low Speed Balancing Showing Divergence of Response Above 2,300 rpm.

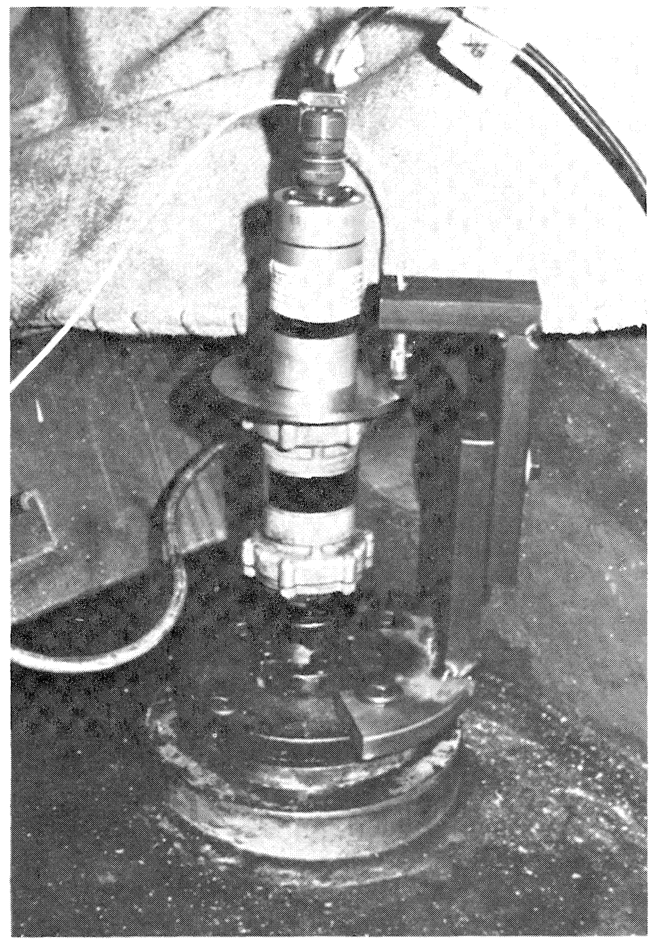


Figure 9. Arrangement of Transducers for Slow Roll Eccentricity and Dynamic Vibration Measurement.

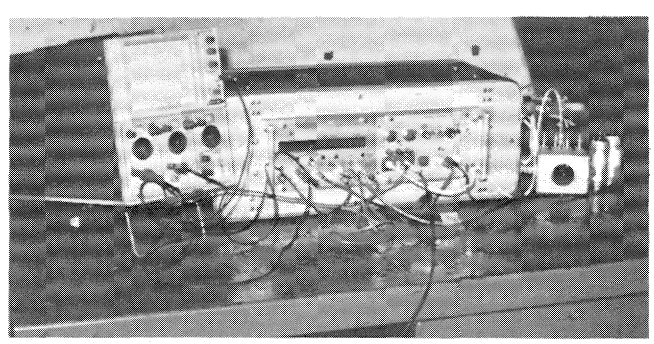


Figure 10. Read Out Instrumentation.

A portable vibration meter was used to provide signal conditioning and power supply to each proximity probe and to allow probe gap setting and gap/voltage calibration. This is shown in Figure 11. In addition to the direct proximity probe inputs, the instrument allowed six bearings to be switch selected and monitored during a run to speed. Phase reference was obtained from the machine's own supervisory phase pulse and this was the datum used during run out checks in the low speed balancing machine, during slow roll, and throughout the tests. The oscilloscope was used to monitor continually the unfiltered vibration wave shape.

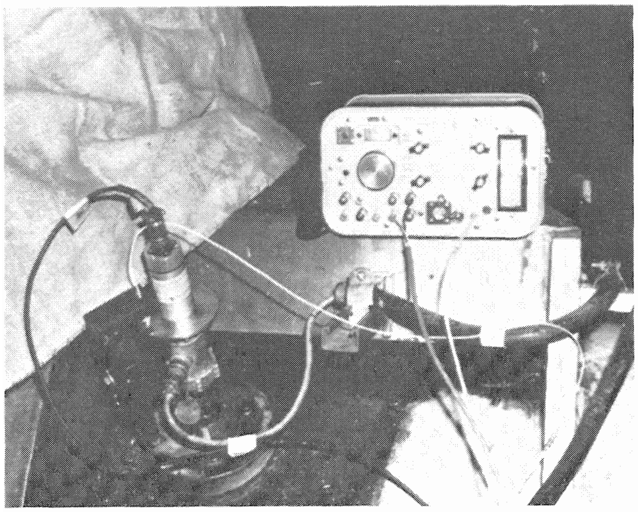


Figure 11. Use of Portable Vibration Meter for Calibration and Power Supply.

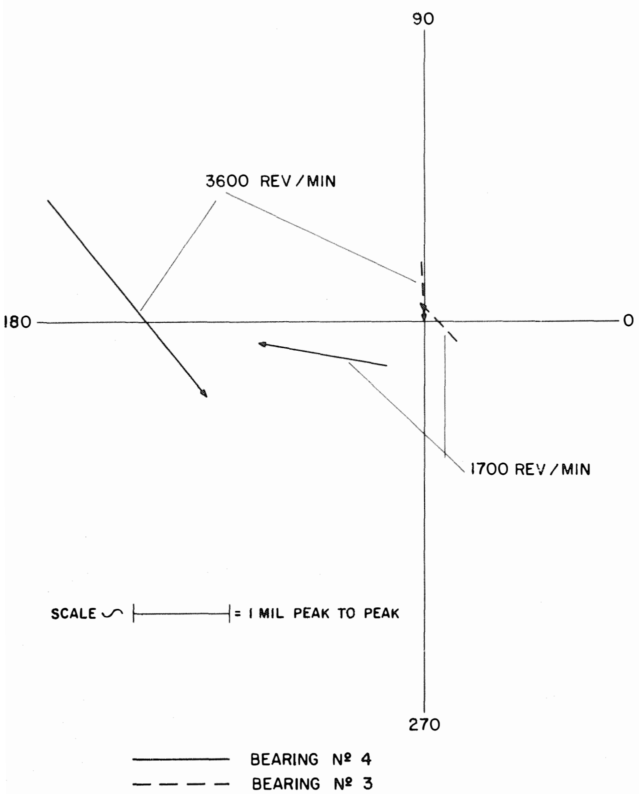


Figure 12. Convergence of Vectors at 1,700 rpm and 3,600 rpm Resulting From the Predicted Single Mid-Span Correction Weight.

FINAL BALANCING PROCEDURE

It was found that only Bearing No. 4, had significant slow roll eccentricity (1.5 mils at 180°) and excessive vibration, so the correction to be applied was based primarily on this response. A mid span calibration weight was applied and its effect at the chosen and repeatable balance speeds of 1,700 RPM and 3,600 RPM was measured. It should be noted here that if these data are put into the computer program directly, along with the slow roll eccentricity vector δ_r a different solution will be predicted for each speed. The reason for this is that the theory is based on the response of a model having a single mass at the center of a flexible shaft. With this configuration, only a first mode can be significantly excited and all vibration data refer to this mode; furthermore, in Gunter's model the response was measured at mid span making the system additionally insensitive to any second mode effect. To make use of the program, therefore, only the in phase components of vibration response at each bearing should be used. The in phase vectors were isolated and these data were used with the slow roll eccentricity vector to obtain the single correction weight predicted by the computer program. Figure 12 shows that the single weight produces an effect that converges to a single point at both speeds and that this point of convergence is close to the ideal value of the slow roll eccentricity of 1.5 mils at 180° for Bearing No. 4 and close to zero for Bearing No. 3.

The final response of the rotor is shown in Figure 13. Number 4 Bearing is seen to be very close to the ideal case of constant slow roll eccentricity vector level over its complete speed range. No evidence of the critical speed at 2,350 RPM can be seen at Bearing No. 4 although this can still be detected in the much lower Bearing 3 response. Again the Nyquist plot in Figure 14 is the most graphic. Here the first critical speed can still be clearly seen at Bearing No. 3 in spite of the low

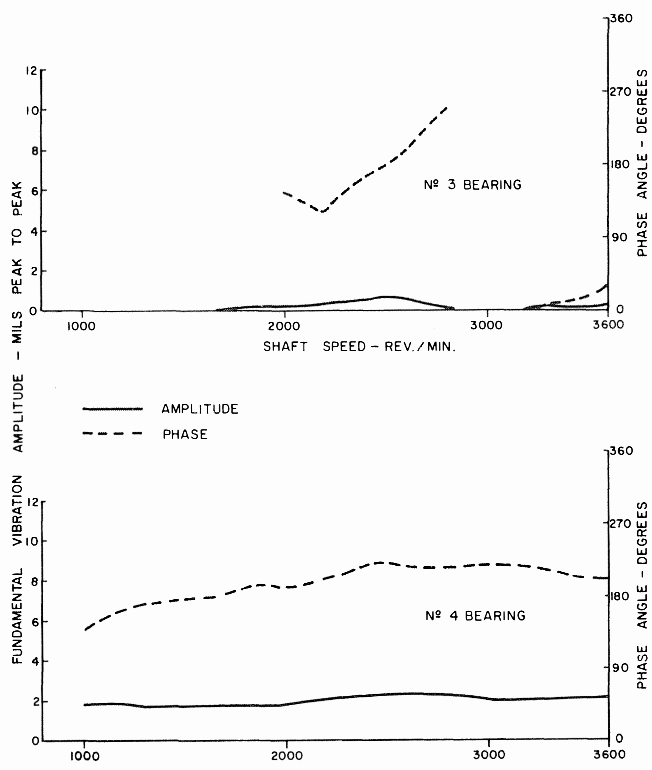


Figure 13. Final Rotor Condition Showing Flat Response at All Speeds.

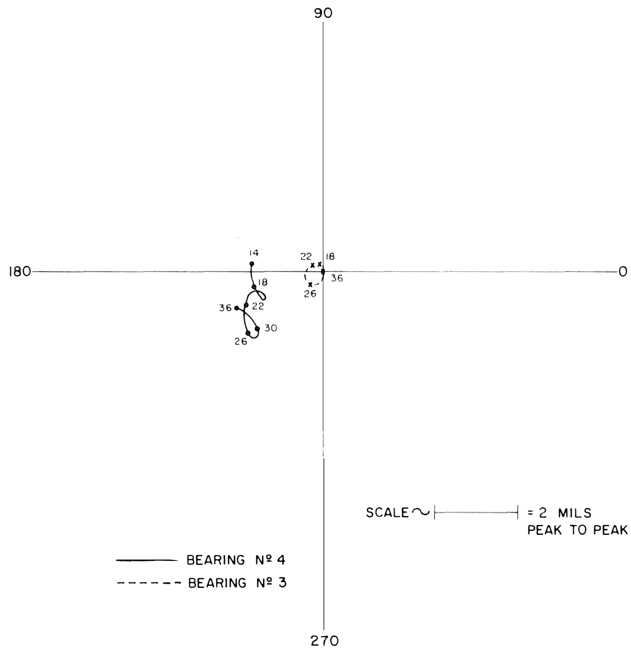


Figure 14. Nyquist Plot of Final Response Showing Residual Small Unbalance at Bearing No. 3 and Dominance of Slow Roll Eccentricity at Bearing No. 4.

level, but Bearing No. 4 is dominated by the constant slow roll eccentricity vector which is in phase with the peak level of No. 3 Bearing.

CONCLUSIONS

A large flexible turbine rotor containing a significant permanent bend has been successfully balanced. Further conclusions regarding the technique for achieving this are as follows:

1. The amplitude/phase/speed response of the rotor must be recorded over its complete speed range.
2. The first mode response must be isolated and only this effect included in the balance computations.
3. A precise measuring system and a meticulously applied method for obtaining reliable and repeatable datum conditions and calibration runs must be used.
4. As Professor Gunter's work predicted, the classical 90° phase change at resonance did not occur for some combinations of unbalance and residual bow. This effect compounds the inadequacy of basing correction weight location on this assumption alone.
5. Balancing corrections based on a single speed are considerably more inadequate in the case of a bowed rotor than they are in the case of a normal flexible rotor.

REFERENCES

1. Gunter E. J., "Dynamic Response and Balancing of Rotor-Bearing Systems," University of Virginia, Short Course. August 1974.
2. Thearle E. L., "Dynamic Balancing of Rotating Machinery in the Field," Transactions ASME Volume 56, 1934, pp. 745, 753.