DYNAMIC ANALYSIS And FIELD BALANCING of 70 MW GAS-TURBINE GENERATORS

E. J. Gunter
R.R. Humphris

Department of Mechanical Engineering & Aerospace Engineering
University of Virginia
Charlottesville, Virginia 22903

I. INTRODUCTION and BACKGROUND

This paper summarizes the multi-plane balancing procedure and dynamic characteristics experienced with several 70 MW gas turbine-generator sets. The gas turbine is a W501 class and weighs approximately 72,000 Lb, and the generator weighs approximately 51,000 Lb. There are two bearings supporting the gas turbine and three bearings supporting the generator. The generator and gas turbine are connected by a rigid coupling.

The W501 gas turbine has had a previous history of high vibrations during cold and hot starts. Thermal bowing of the rotor under load normally occurs with this class of rotor. The bowing effect may be attributed to two sources; the expansion of the copper conductors in the generator and the thermal expansion of the gas turbine. In the later case, re-machining of the torque tube has been reported in the literature to have some benefit.

In the first case study, a utility had a particular 70 MW unit that the manufacturer had attempted to balance for three months without success. However, similar units in the same facility did not experience this difficulty. In the first case, the gas turbine appeared to be operating near a third rotor critical speed. Simple two plane balancing, in this case, is not possible and additional balance planes must be employed.

In addition to the turbine appearing to operate near a third critical speed, the turbine-generator response was effected by the power level. Substantial changes in synchronous amplitude and phase occurred along the turbine-generator when the power level was increased from 0 to 70 MW.

To add to the complexity of the first case, seal rubs were encountered at the exciter under certain unbalance conditions. This would lead to superharmonic oscillations of 3x running speed and cause non-linear jumps in amplitude and phase at the exciter end. Under these conditions, the influence coefficients change dramatically along the entire length of the 60 ton, 55 ft. turbine-generator set.

It was found that proper balancing requires that both no load and full load data be obtained before and after the application of the trial or calibration weights in order to make proper balance predictions. In addition to obtaining data at 3,600 RPM, data should also be obtained in the 1,000 RPM range and 2,400 RPM range. If one, for example, attempts to trim balance the gas turbine at 3,600 RPM with a static balance (in plane) weight set, then one risks bowing the rotor at its first critical speed around 1,000 RPM. This occurred in one of the earlier balancing attempts on the first unit by the OEM field engineer (Original Equipment Manufacturer).

One of the objectives of the second balancing case, performed by E. Gunter and D. Marshall of S.C.E., was to balance the unit efficiently with the fewest number of runs to a suitable level of vibration so that it would not be necessary to strip down the generator to perform an internal balance on the generator. If the generator is taken off-line and degassed, then this procedure is extremely time-consuming and expensive.

Fig 1.1 represents a cross-section of the 70 MW gas turbine. The gas turbine has 17 axial compressor stages and 4 gas turbine stages. The axial compressor is attached to the gas turbine by a bolted flange assembly. The gas turbine proper has 5 balancing planes, at the first stage of the axial compressor, between the 6th and 7th stages, two planes on either side of the bolted flange, and a balance plane at the first gas turbine stage.

I. INTRODUCTION and BACKGROUND

This paper summarizes the multi-plane balancing procedure and dynamic characteristics experienced with several 70 MW gas turbine-generator sets. The gas turbine is a W501 class and weighs approximately 72,000 Lb, and the generator weighs approximately 51,000 Lb. There are two bearings supporting the gas turbine and three bearings supporting the generator. The generator and gas turbine are connected by a rigid coupling.

The W501 gas turbine has had a previous history of high vibrations during cold and hot starts. Thermal bowing of the rotor under load normally occurs with this class of rotor. The bowing effect may be attributed to two sources; the expansion of the copper conductors in the generator and the thermal expansion of the gas turbine. In the later case, re-machining of the torque tube has been reported in the literature to have some benefit.

In the first case study, a utility had a particular 70 MW unit that the manufacturer had attempted to balance for three months without success. However, similar units in the same facility did not experience this difficulty. In the first case, the gas turbine appeared to be operating near a third rotor critical speed. Simple two plane balancing, in this case, is not possible and additional balance planes must be employed.

In addition to the turbine appearing to operate near a third critical speed, the turbine-generator response was effected by the power level. Substantial changes in synchronous amplitude and phase occurred along the turbine-generator when the power level was increased from 0 to 70 MW.

To add to the complexity of the first case, seal rubs were encountered at the exciter under certain unbalance conditions. This would lead to superharmonic oscillations of 3x running speed and cause non-linear jumps in amplitude and phase at the exciter end. Under these conditions, the influence coefficients change dramatically along the entire length of the 60 ton, 55 ft. turbine-generator set.

It was found that proper balancing requires that both no load and full load data be obtained before and after the application of the trial or calibration weights in order to make proper balance predictions. In addition to obtaining data at 3,600 RPM, data should also be obtained in the 1,000 RPM range and 2,400 RPM range. If one, for example, attempts to trim balance the gas turbine at 3,600 RPM with a static balance (in plane) weight set, then one risks bowing the rotor at its first critical speed around 1,000 RPM. This occurred in one of the earlier balancing attempts on the first unit by the OEM field engineer (Original Equipment Manufacturer).

One of the objectives of the second balancing case, performed by E. Gunter and D. Marshall of S.C.E., was to balance the unit efficiently with the fewest number of runs to a suitable level of vibration so that it would not be necessary to strip down the generator to perform an internal balance on the generator. If the generator is taken off-line and degassed, then this procedure is extremely time-consuming and expensive.

Fig 1.1 represents a cross-section of the 70 MW gas turbine. The gas turbine has 17 axial compressor stages and 4 gas turbine stages. The axial compressor is attached to the gas turbine by a bolted flange assembly. The gas turbine proper has 5 balancing planes, at the first stage of the axial compressor, between the 6th and 7th stages, two planes on either side of the bolted flange, and a balance plane at the first gas turbine stage.

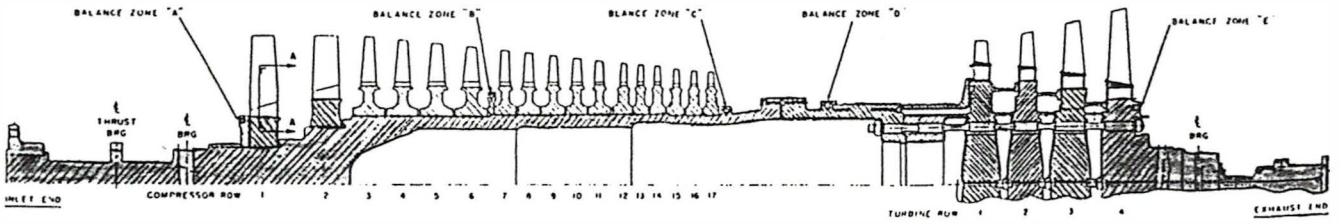


Figure 1.1 Cross Section of 70 MW Gas Turbine

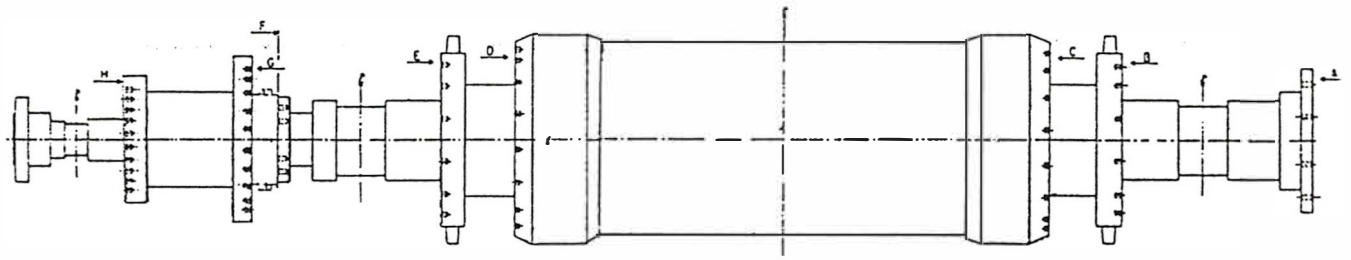


Figure 1.2 Cross Section of Generator

In the first case study, balance weights only were applied at the 1st compressor and gas turbine stages. In field balancing, the 2nd balance plane is never used. Analysis and field experience has shown that the center balance planes 3 and 4 are most useful for controlling the gas turbine 1st critical speed around 1,000 RPM. These planes should be used first, before final balancing using planes 1 and 5 at 3,600 RPM. However, in the first case study, the center balance planes were not available due to piping constraints.

A typical balance weight is approximately 9 - 10 oz. These balance weights are threaded screws and may be inserted into the compressor and turbine end balancing planes by means of a long rod. In the normal multi-plane balancing procedure using influence coefficients, the trial weights are removed after the trial runs. The final balancing calculation is performed and the balance weights are applied. This ideal procedure is not followed when field balancing 70 MW gas turbine-generators. It is the practice never to remove a trial or initial weight if the rotor amplitude reduces. The hot end of the gas turbine is particularly difficult to get at in order to apply balance weights. If one should drop a weight in the hot end of the gas turbine, then it is necessary to disassemble the gas turbine to remove it. Also one only gets one shot a day at the hot end gas turbine location.

Fig. 1.2 represents a typical generator. The generator shows 7 balance planes. In the first case, no balancing weights were placed on the generator except at the coupling, plane A, and at the exciter end, plane H. Balancing corrections at planes D and C require degassing the generator.

In the balancing procedure performed, an HP 9845 desk computer was used in the first case and an HP 200 Series computer was used in the second case to perform the balancing calculations. By means of an advanced desk micro-computer, least squared error and linear programming calculations can be performed using all 5 probes on the turbine-generator, at multiple speeds and power levels. In this way, an optimized balance condition can be computed. The desktop computers were also used to collect the rotor data and to plot polar and Bode diagrams of amplitude and phase. The computer can also be used to predict the new rotor response after application of balance corrections.

In the second balancing case, a successful one-shot balancing of the compressor cold end was performed by using previously obtained balancing data on a similar machine. The one-shot balancing process is based on using the generated influence coefficients of the machine (or a similar machine) along with the current vibration levels. This calculation can only be performed with a modern high-speed computer. The balancing shot was of sufficient accuracy that no other corrections needed to be applied at the compressor station. The authors firmly believe that the use of small high-speed desk computers, such as the HEWLETT PACKARD 200 Series, may have a major impact on balancing of large rotating utility equipment.

In addition to describing the balance procedure and the balancing calculations performed, additional information on the experimental characteristics of a 70 MW turbine-generator are presented. The W501 turbine-generator is a difficult unit to balance because of the high sensitivity of both the gas turbine and exciter end of the generator. The turbine-generator, at running speed, may be operating close to one or more resonance frequencies. Because of this operation at or near a resonant frequency, the turbine-generator is very sensitive to thermal bowing. Extreme care must be taken in the balancing procedure so as not to mix balancing data taken at different power levels. Presented in this paper are some of the response coefficients obtained for correction weights at various planes. It is of particular importance to note that small deviations of weight at the exciter end can cause non-linear changes in the exciter end influence coefficients. This makes the exciter end particularly sensitive to balancing.

II. INSTRUMENTATION

The following is a brief description of the instrumentation and computer equipment used to perform the balancing on the 70 MW turbine-generators. The most essential element for analyzing a 70 MW turbine-generator is instrumentation to measure the vibration amplitude and phase at various planes. Without adequate instrumentation for vibration measurement, accurate balancing is impossible. There are various types of instrumentation used for monitoring turbine-generator vibrations. The instrumentation has either been shaft-rider probes or the BENTLY NEVADA dual probes which measure casing and rotor motion. Fig 2.1 represents a schematic diagram of the 70 MW turbine-generator with 5 probe locations at each bearing. The shaft-rider technique is the method normally used by the OEM field engineer to perform balancing. In this procedure, the field engineer physically applies a shaft-rider stick to the shaft to measure amplitude and phase. Normally, phase is obtained from a phase generator attached to the turbine end.

In both case studies the 70 MW turbine-generators were equipped with the BENTLY NEVADA 7000 SERIES dual probes system as shown in Fig 2.1 (with one dual probe mounted vertically on each of the 5 bearings). The BENTLY dual probe system uses a proximity probe for shaft relative motion and a velocity probe for casing relative motion. An additional probe located at either the coupling (case 1) or the exciter end (case 2) provided the phase reference mark. The system has the capability of showing relative shaft-to-casing motion, casing motion, or the total amplitude of motion by integrating the casing velocity signal and vectorially adding it to the relative shaft motion. Extensive shaft rider data was obtained on the first unit. A comparison of the shaft rider data to the BENTLY displacement data showed considerably different amplitude and phase data. However, balance calculations using shaft rider data or BENTLY displacement probe data predicted similar balancing values. The non-contact BENTLY probes are preferable to the shaft-rider stick method as the BENTLY probes are able to accurately measure a slow roll vector. The balancing calculations should be based upon the compensated displacement data with slow roll vibration subtracted. This is not possible with the shaft-rider sticks because of the frequency response of the device at low speeds.

A BENTLY DIGITAL VECTOR FILTER II was used to determine synchronous amplitude and phase. In addition to determining synchronous motion, the DIGITAL VECTOR FILTER can determine the subsynchronous and total vibration levels. If there is a large discrepancy between the total motion and the synchronous motion, then other effects may be present in the system such as rotor instability, misalignment, or shaft rubs. A tape recorder was used to store the balancing data from 5 probes in the first case for future analysis. In the second case, when the balancing was performed, an operational tape recorder was not available. This, however, was not a serious restriction to accomplishing the balancing procedure.

An HP 9845B and 9816 computer with an HP 2671G printer were used in the two cases to run the least squared error constrained optimization balancing program and the BENTLY ADRE software program for the collection and analysis of the experimental data. A 2-pen analog plotter was also used to plot polar diagrams at the various stations. In order to perform the least squared error balancing procedure using 5 BENTLY probes, the computer is essential for carrying out the computations. A HEWLETT PACKARD 3582 SPECTRUM ANALYZER was also used to examine the vibration spectrum.

In the first case, the synchronous vibration amplitude and phase data was obtained for various speeds and probe locations by means of the field portable TURBOBALANCER developed by E. Shepherd of MECHINTEL, INC. This device allows one to obtain print-outs of up to 6 probes recording the RPM, amplitude and phase on a thermal printer. Data was recorded at the 5 probes at various speeds on spin up and coast down as well as at 3,600 RPM under various power levels. The thermal print-out obtained from the TURBOBALANCER was found to be exceptionally useful in recording the vibration data on start up and coast down. This data allowed the authors to compute the optimum balance correction using vibration data at several critical speeds as well as at running speed under 0 and full power levels. In the second case, the BENTLY ADRE program was used to record two probes and the additional data of the other probes was recorded by hand.

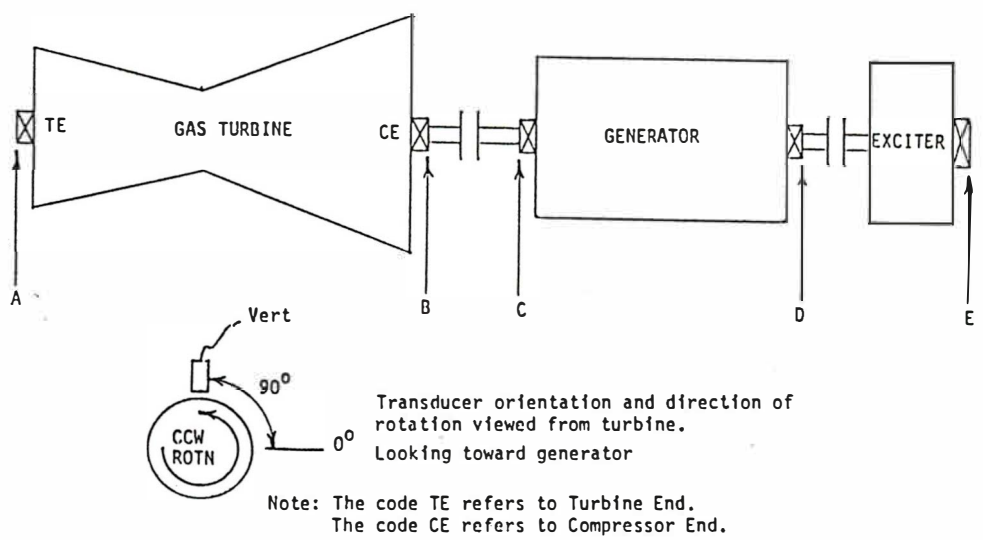


Figure 2.1 Schematic Diagram of Probe Locations for 70 MW Turbine Generator

III. CRITICAL SPEEDS of 70 MW TURBINE-GENERATOR

3.1 Gas Turbine Critical Speeds

In order to understand the planes at which balance weights will be most effective, it is useful to have a critical speed analysis of the gas turbine-generator systems. In the first series of critical speed analyses, the gas turbine was analyzed by itself. A model of the 70 MW gas turbine was developed from a scaled drawing.

Fig 3.1 represents the cross-section of the 54-station gas turbine rotor. The bearings supporting the gas turbine are 4 pad tilting-pad bearings with load between pads. Fig 3.2 represents the first 3 modes of the gas turbine. Fig 3.3 represents the animated 1st critical speed mode shape for the 70 MW gas turbine. From an observation of the rotor mode shape, it is apparent that the center rotor balance planes 3 and 4 will be the most effective for balancing the rotor 1st critical speed. Note that the coupling at station 1 will have very little influence on the first mode.

Static coupling shots are often applied on the gas turbine first compressor and gas turbine stages, at balance planes 1 and 5. In a static balancing shot or correction, two weights are inserted at the two balance planes in phase. This combination of weights has a strong influence on the response characteristics of the gas turbine at the turbine 1st critical speed. If this combination is used to trim balance the turbine at 3,600 RPM, then high excitation or even bowing of the turbine may occur at the 1st critical speed. Therefore, it is preferable to first balance the gas turbine around 1,000 RPM with a center plane correction or with a static correction at the end planes before attempting to trim balance at 3,600 RPM.

Fig 3.4 represents the gas turbine animated second mode. With the assumed bearing stiffness values of 4E6 Lb/in, the rotor second critical speed occurs at 2,793 RPM. In this mode, the turbine and compressor ends are 180° out of phase. If a balance correction is placed at the center plane, it will have no effect on the response of the rotor at the second critical speed. In this case, the balance correction is said to be orthogonal to the second mode.

A single plane balancing correction at either the compressor or turbine end will affect the first mode as well as the second mode. A dynamic balancing shot, in which weights are placed 180° out of phase at the compressor and gas turbine ends, would have little effect on excitation of the first gas turbine mode. Under the circumstances, this weight distribution would be orthogonal to the first mode and hence would not re-excite the first critical speed. This weight combination was not employed by the OEM field engineer for trim balancing at running speed for unit #1.

3.2 Influence of Bolted Section on Gas Turbine Critical Speeds

After extensive experimental data was obtained on the first gas turbine unit, it appeared that the hot end of the turbine was starting to enter a third critical speed. The gas turbine critical speed study showed that the third turbine bending mode should be above 5,400 RPM. However, the analysis also indicated that over 80% of the strain energy of the rotor is concentrated in the area of the bolted-up flange section connecting the axial compressor to the gas turbine. Over an extended operating period, the center span

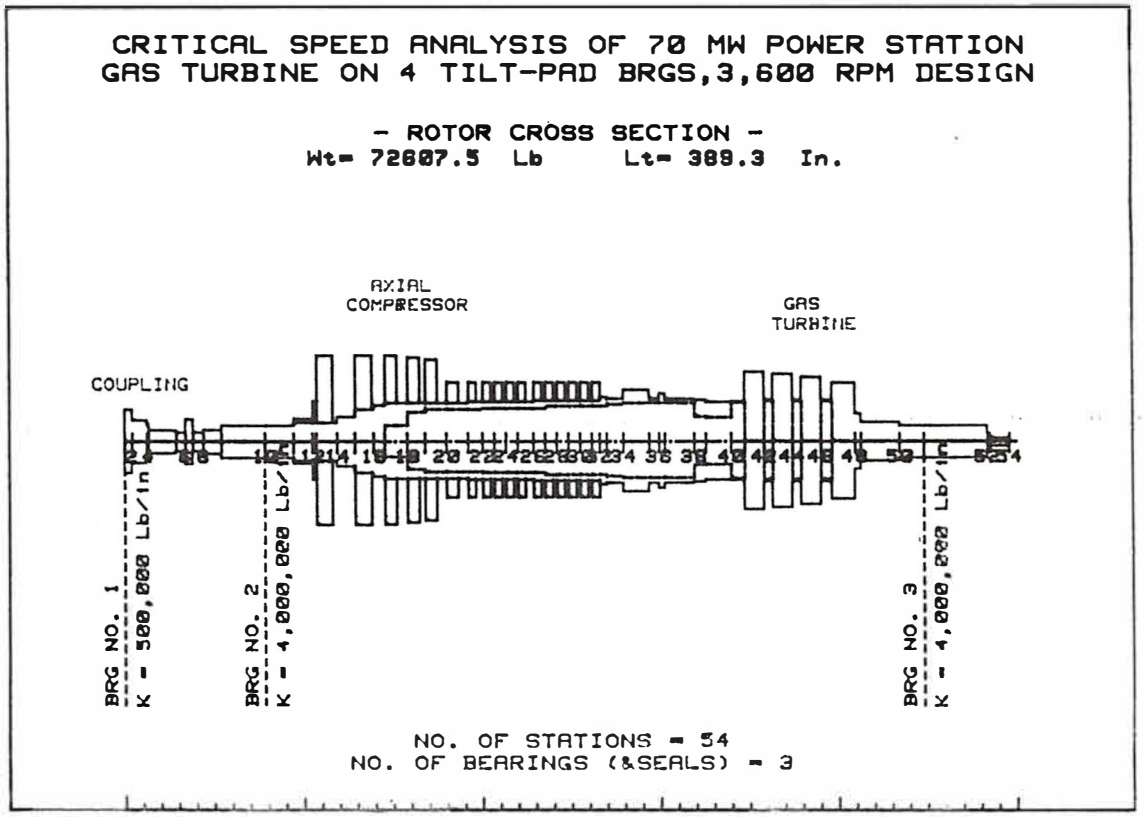


Figure 3.1 Cross-Section of 54 Station 70 MW Gas Turbine

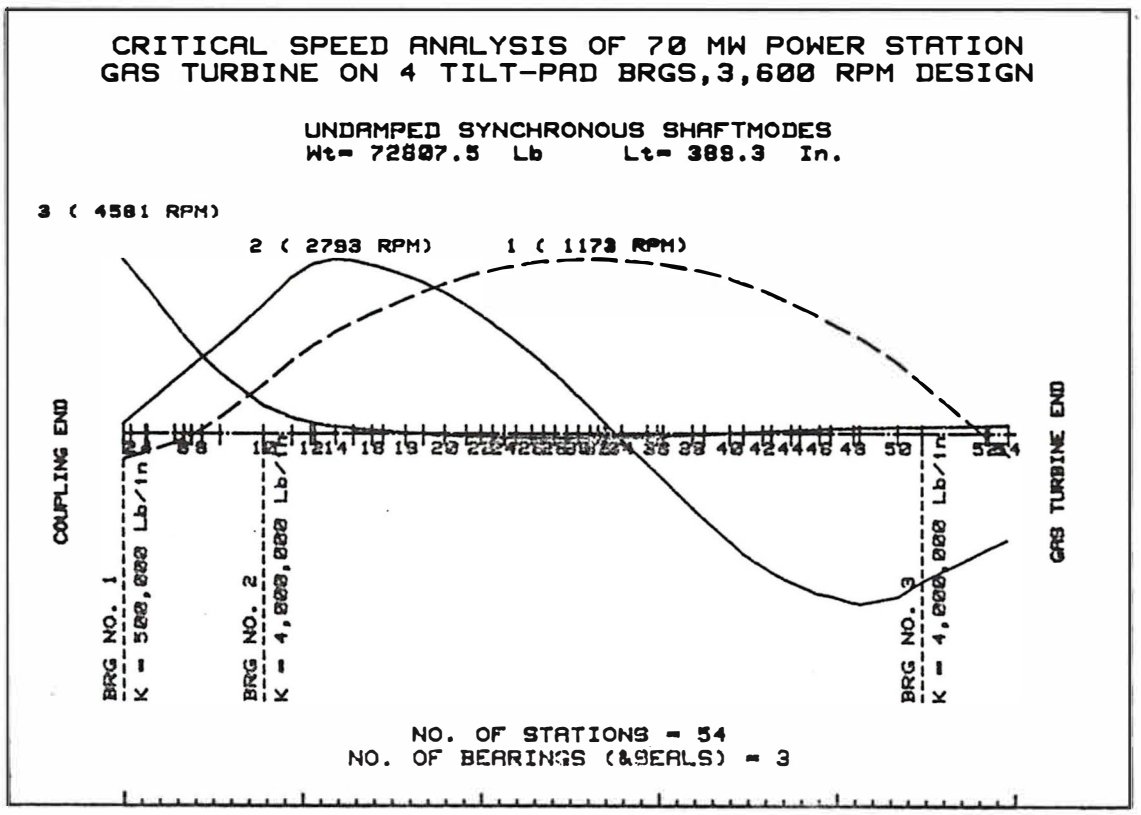


Figure 3.2 Modes of 70 MW Gas Turbine With Symmetric Tilting 4-Pad Bearings

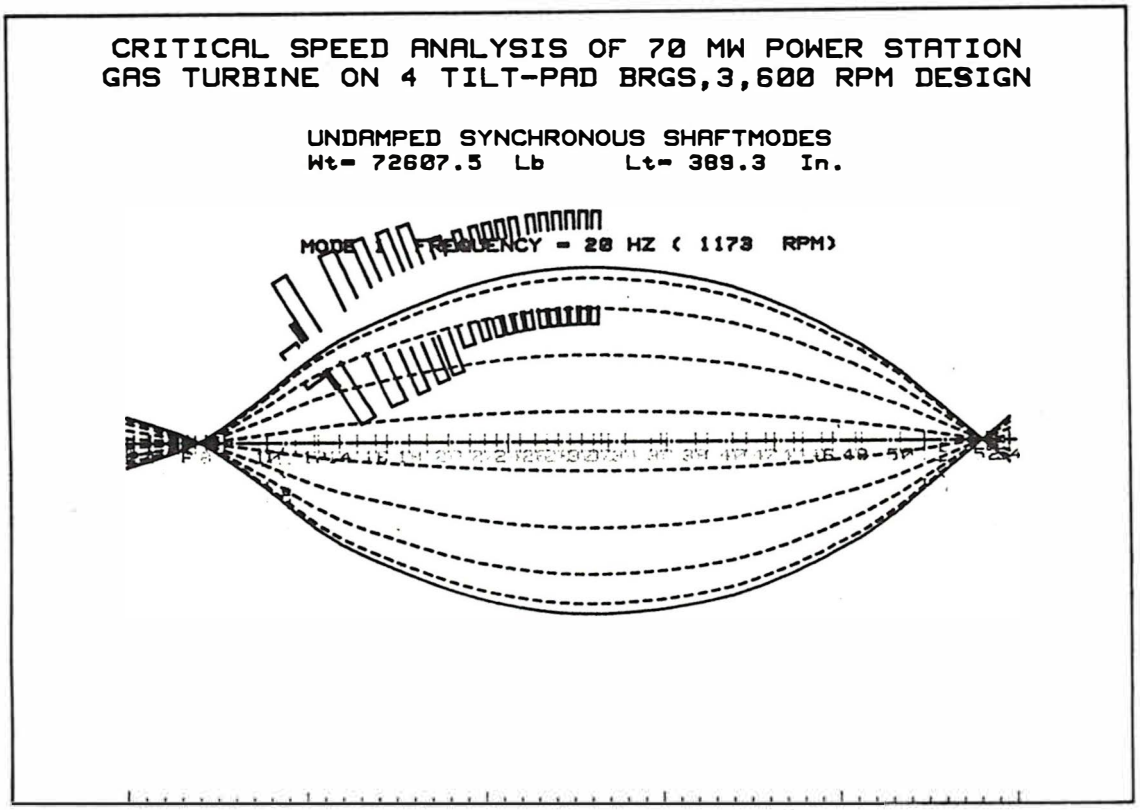


Figure 3.3 Animated 1st Critical Speed Mode of 70 MW Gas Turbine

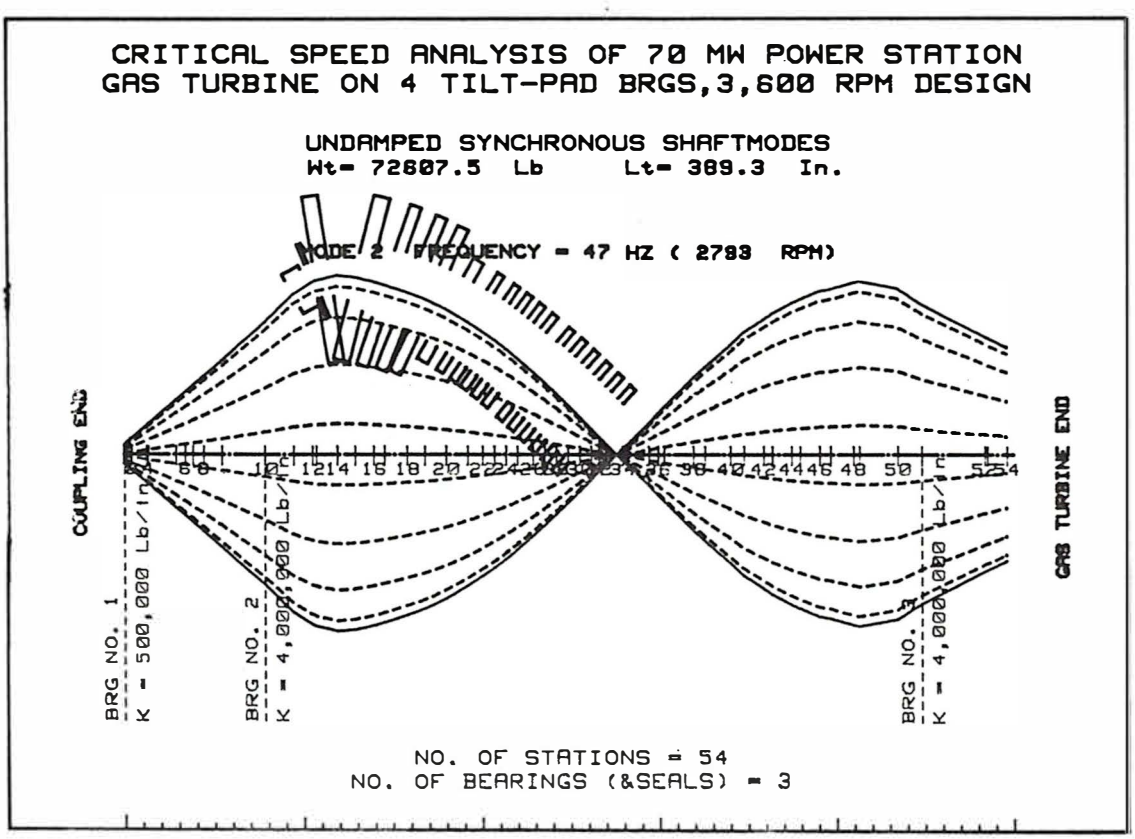


Figure 3.4 Animated 2nd Critical Speed Mode of 70 MW Gas Turbine

bolts will have a tendency to creep. This creep effect is enhanced if the rotor has high unbalance causing bowing in the first mode. The creeping causes a moment release at the bolted-up section.

Fig 3.5 represents the cross-section of the 70 MW gas turbine with a moment release at station 35, the center span bolted connection. In this analysis the compressor end tiltingpad bearing stiffness was taken as 4E6 Lb/in and the gas turbine end bearing was assumed to be 2E6 Lb/in because of the softer bearing support under the turbine bearing. Fig 3.6 represents the first 4 gas turbine critical speeds with the moment release. The second critical speed is unaffected by the moment release at the center span since there is very little strain energy of bending at this position. However, the original 5th mode is reduced in frequency from above 5,400 RPM to 3,812 RPM. The 4th mode, which was the previous mode caused by the coupling and assumed generator impedance of 500,000 Lb/in remains relatively unchanged at 4,308 RPM. Hence it is apparent that a relaxation of the center span bolts may have a significant influence on the dynamics of the gas turbine.

3.3 Influence of Webbed Support on Turbine End Bearing Stiffness and Damping

In the course of balancing the 70 MW gas turbine, it was seen that the gas turbine has an unusually high level of *cross-talk* in the vibration levels experienced at the turbine and compressor bearing locations. That is, the application of a trial weight at either the compressor or turbine location would cause a larger response at the other end of the machine. Thus if high vibrations are experienced on the hot end of the gas turbine, a correction weight is normally placed at the compressor location and a similar procedure is followed for high vibrations at the compressor end.

This high *cross-talk* may be in part due also to the differences in the stiffness characteristics of the tilting-pad bearings at the compressor and turbine locations. One reason for this may be due to the considerable difference in support characteristics of the compressor end and turbine end bearings. The compressor end bearing is mounted on a massive pedestal or foundation.

The turbine end tilting-pad bearing is mounted in a flexible pedestal in which the webs radiate from the support housing. The bearing housing is supported by six webs which are connected tangentially to the bearing housing. The purpose of this flexible support is to allow for thermal expansions. This produces an extremely soft support in comparison to the stiffness of the tilting-pad bearing.

The tilting 4 pad bearing at running speed can have a stiffness value as high as 14E6 Lb/in stiffness, neglecting the shoe pivot support Hertz stress, and a damping of 50,000 Lb-sec/in. The equation of effective bearing impedence was derived assuming synchronous motion in which rotor mass, foundation mass, and foundation stiffness and damping may be included. A computer program was developed to determine the effective bearing stiffness and damping characteristics of the combined tilting-pad-foundation system. The effective bearing characteristics were calculated over a speed range of 500 to 4,500 RPM.

Table 3.1 represents the effective bearing, stiffness and damping coefficients for various support conditions. At running speed, the theoretical bearing stiffness and damping value of the 4 pad bearing under minimum clearance conditions may be as high as 14E6 Lb/in and 5E4 Lb-sec/in damping. In actual practice, the effective stiffness and damping or impedance value experienced by the shaft is substantially reduced. For example, even with a rigid foundation, pad pivot stiffness must be considered. The pivot point effect may be computed from Hertzian or contact stress theory and the pivot stiffness rarely exceeds 5E6 Lb/in. Table 3.1 shows that a support stiffness of 5E6 Lb/in

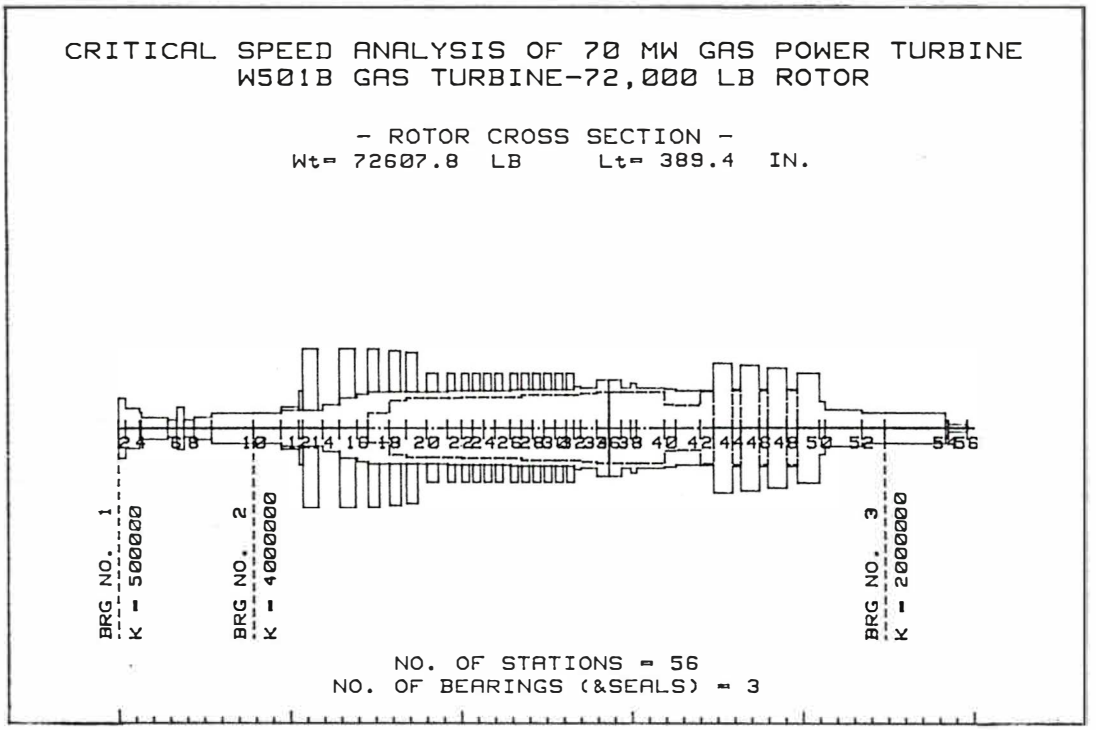


Figure 3.5 70 MW Gas Turbine With Moment Release At St. 35

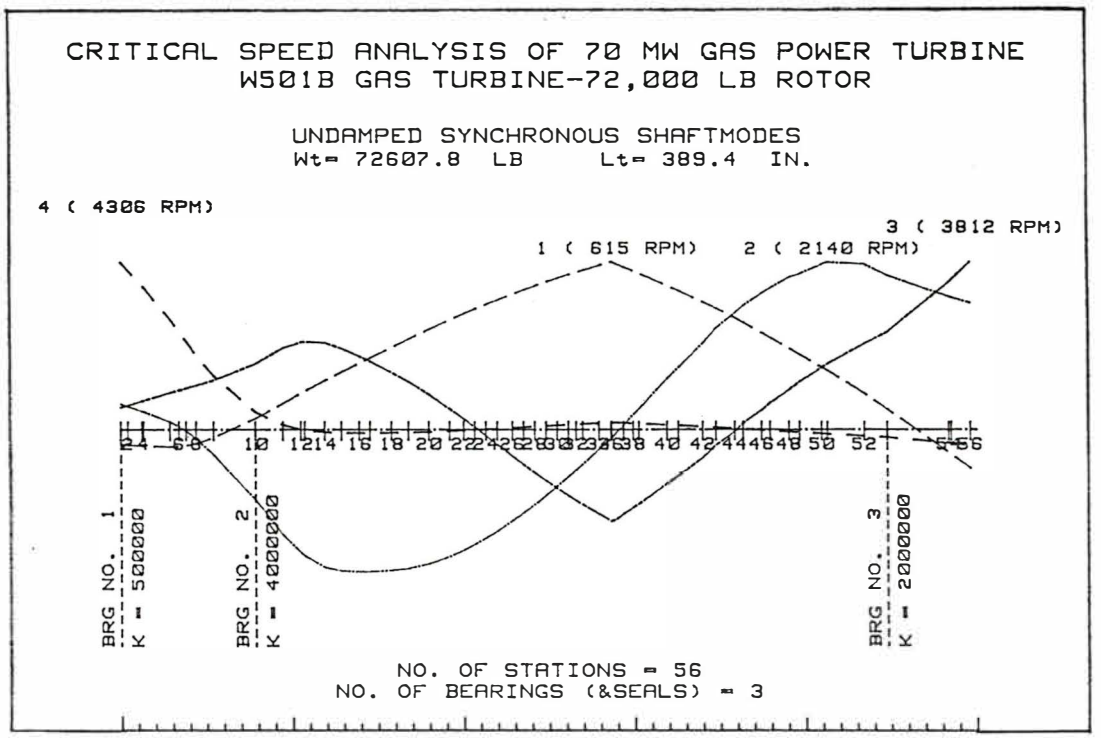


Figure 3.6 Modes of 70 MW Gas Turbine With Moment Release

TABLE 3.1

INFLUENCE OF GAS TURBINE - BEARING SUPPORT HOUSING ON EFFECTIVE BEARING COEFFICIENTS

[N = 3,600 RPM, K_b = 14 x 10^6 Lb/in, C_b = 5 x 10^4 lb-sec/in]

K_f (Lb/in)	W_f (lb) 0	W_f (lb) 50	W_f (lb) 100	W_f (lb) 250
200,000 (C_e)	3.58	2.96	2.4	1.05
(K_e)	.198E6	.180E6	.162E6	.107E6
500,000	22.07	20.53	19.01	14.79
	.493E6	.475E6	.457E6	.403E6
1E6	86.10	83.10	80.09	71.38
	.973E6	.956E6	.939E6	.886E6
2E6	327	321	316	299
	1.894E6	1.878E6	1.862E6	1.813E6
5E6	1,744.9	1,733.9	1,722.8	1,689.7
	4.336E6	4.323E6	4.309E6	4.269E6

K_f = foundation stiffness (Lb/in)
K_b = bearing stiffness (Lb/in)
K_e = effective bearing stiffness

W_f = foundation weight (lb)
C_b = bearing damper (lb-sec/in)
C_e = effective bearing damping

will reduce the bearing stiffness from 14E6 to 4.3E6 Lb/in. A dramatic reduction of bearing damping is experienced by foundation flexibility. The bearing damping at the compressor end is reduced from 50,000 Lb-sec/in to 1,734 Lb-sec/in. At the turbine end, the flexible pedestal may have a support stiffness well below 1E6 Lb/in. In this case, the effective damping at the gas turbine location will reduce from 50,000 Lb-sec/in to below 80 Lb-sec/in. The flexible web required for thermal expansion at the turbine end bearing, in essence, kills the high damping of the tilting-pad bearing. This effect adds to the sensitivity of the gas turbine to unbalance and attributes to its high *cross-talk*. Such dissimilar pedestal effects are not normally experienced with steam turbines.

3.4 Critical Speeds of 70 MW Turbine-Generator System

Discussed in this section are the types of critical speeds that may be expected with a 70 MW turbine-generator. In the original analysis of the gas turbine, in order to simulate the effect of the generator, a simple spring rate of 500,000 Lb/in was assumed at the coupling end. It is the assumption of the coupling impedance that governs the value of the turbine 3rd critical speed with high coupling end motion. However, in order to more fully understand the dynamics of the entire turbine-generator set, the generator was included with the 70 MW turbine to form an 86-station model. Table 3.2 represents the bearing types and dimensions used in the 70 MW turbine-generator set.

A critical speed analysis of the gas turbine rigidly coupled to a typical 70 MW generator was run. In this unit, the exact configuration of the generator and its bearings were not known. In the evaluation of the experimental data for the 70 MW turbine-generator for unit #2, it appeared that the exciter may be operating through several critical speeds. It appeared that at 3,600 RPM the exciter was operating at or near a critical speed on the second unit that was field balanced.

Fig 3.7 represents a cross-section of the 84-station model used. The total weight for the system is estimated to be approximately 123,200 Lb. It would be highly desirable to obtain an exact cross-sectional drawing of the generator and its bearings in order to develop a more accurate model for the turbine-generator system. However, the characteristics developed for this system are very similar to previously observed critical speeds for this class of machinery.

Fig 3.8 represents a summary of the modeshapes determined for the 70 MW turbine-generator. There are 5 modes calculated with the assumed bearing stiffness values. The first mode at 917 RPM is the first critical speed of the turbine. The second mode at 1,735 RPM is the first critical speed of the generator. The third mode at 2,143 RPM is the second mode of the turbine. The fourth mode at 3,440 RPM is the second mode of the generator. The fifth system mode at 4,580 RPM is the third mode of the generator. In this model a low stiffness of 500,000 Lb/in was assumed for the exciter end bearing. It is of interest to note that both modes 4 and 5 show high exciter amplitude. If the exciter spring rate is reduced below 500,000 Lb/in, then the fifth mode will reduce from 4,500 RPM to the 3,600 RPM speed range. Therefore, the dynamic characteristics of the generator modes are highly dependent on the exciter end bearing stiffness values. Details of these modes will be described in the following sections.

Fig 3.9 represents the first mode of the gas turbine. In this analysis, the hot end of the gas turbine was assumed to have a relatively low bearing stiffness value of 1,000,000 Lb/in. The reason for this reduced bearing stiffness at the gas turbine end is the flexible webbed bearing cartridge supporting the tilting-pad bearing at the hot end. This mode should be expected to occur between 900 and 1,100 RPM. An analysis of the strain energy of the system shows that 55% is energy associated with the hot end gas tur-

TABLE 3.2

TURBINE - GENERATOR SYSTEM BEARINGS

[Gulfcrest 32 Turbine Oil.....(~1.2u Reyns)]

Location No.	Type	L/D (dim)	L (in)	D (in)	C_d (in)	C_d/D (mil/in)	W (lb)	Description
1 (E)	Axial-Groove Partial Arc	1 to .75	5.78	5.78	.010	5.88	2,200	Exciter
2 (D)	Relieved Axial-Groove (160° arc brg)	.78	8.2*	10.5*	.024	2.29	23,000	Generator (OB) Outboard
3 (C)	Relieved Axial-Groove (160° arc brg)	.78	8.2*	10.5*	.023	2.29	23,000	Generator (IB) Inboard
4 (B)	Tilting-Pad 4-Pad 0.5 Preload Load Between Pads	.82	11.5*	14.0*	.032 (pad Cl) .020 (brg Cl)	1.14	36,000	Compressor (CE)
5 (A)	Tilting-Pad 4-Pad 0.5 Preload Load Between Pads	.82	11.5*	14.0*	.032 (pad Cl) .020 (brg Cl)	1.14	36,000	Turbine (TE)

* Scaled from photographs

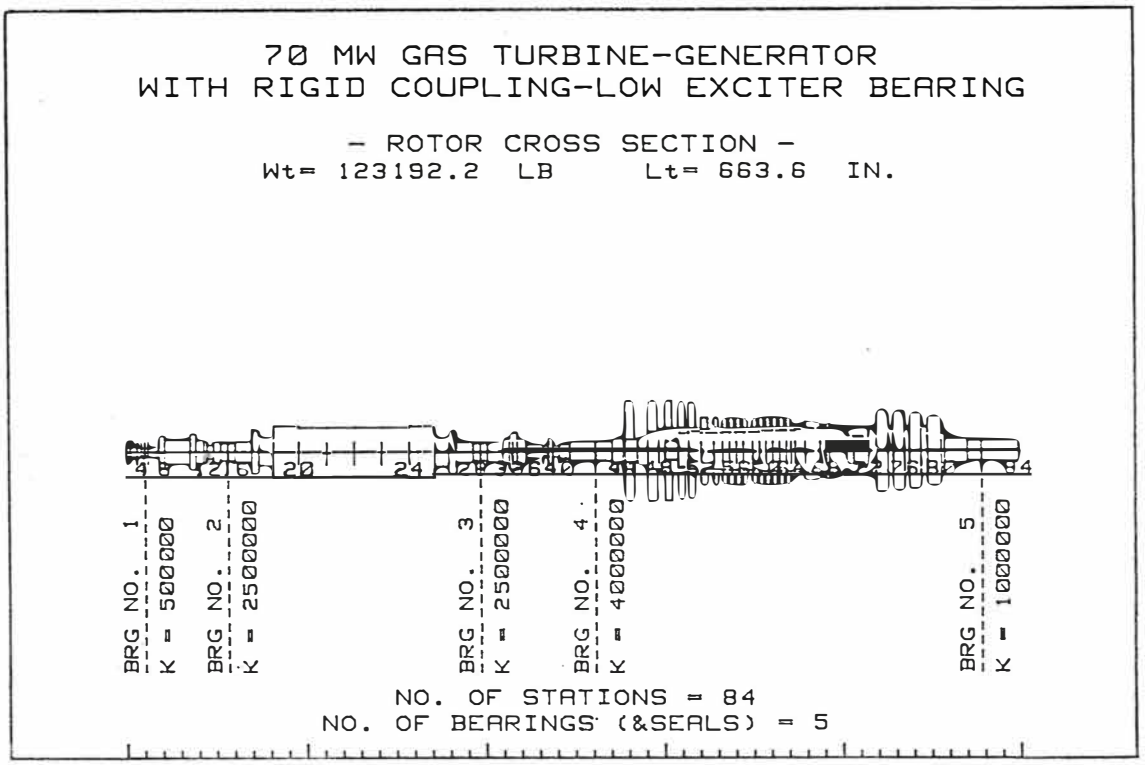


Figure 3.7 Model of 86 St. 70 MW Turbine-Generator With Rigid Coupling

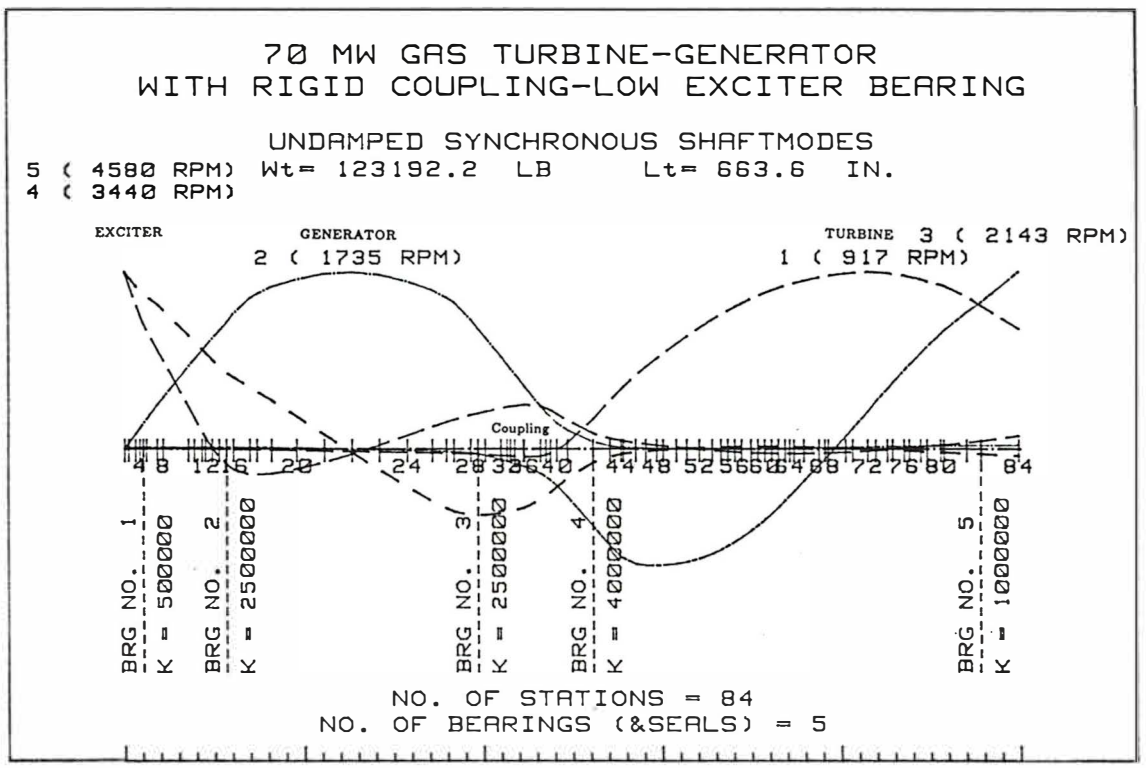


Figure 3.8 Mode Shapes of 70 MW Turbine- Generator With Low Exciter Stiffness

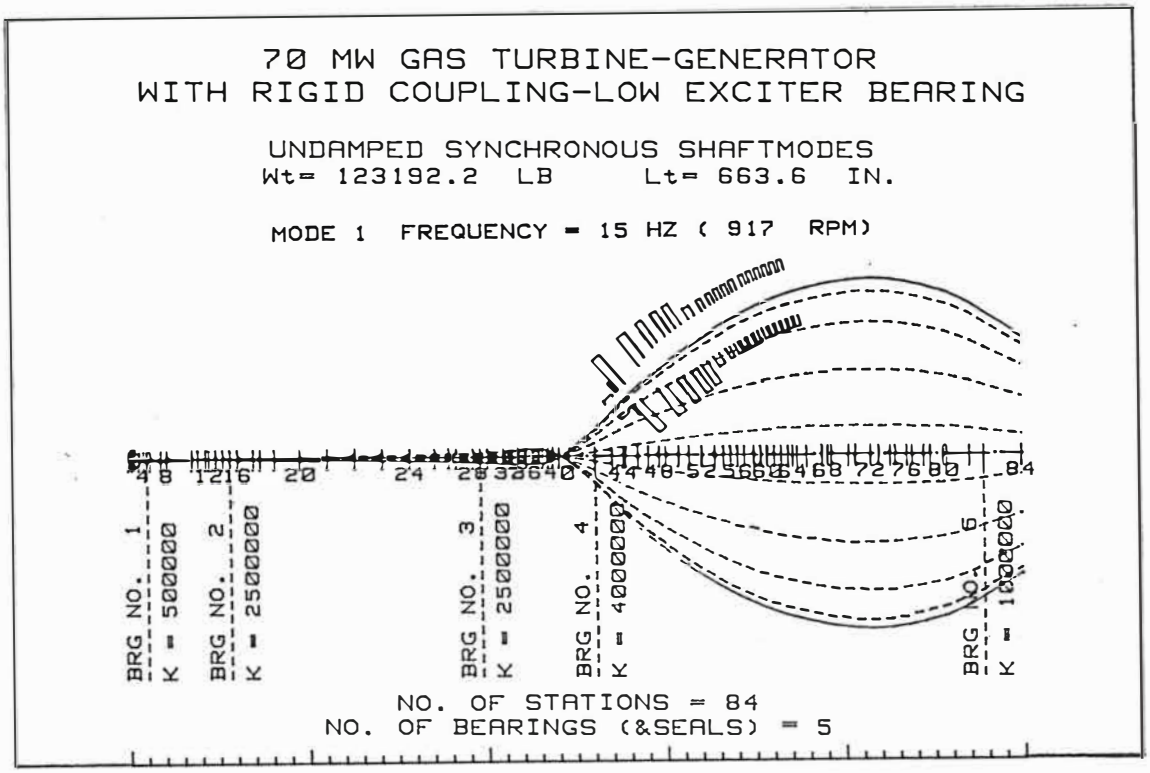


Figure 3.9 Animated 1st Turbine Mode of 70 MW Turbine-Generator

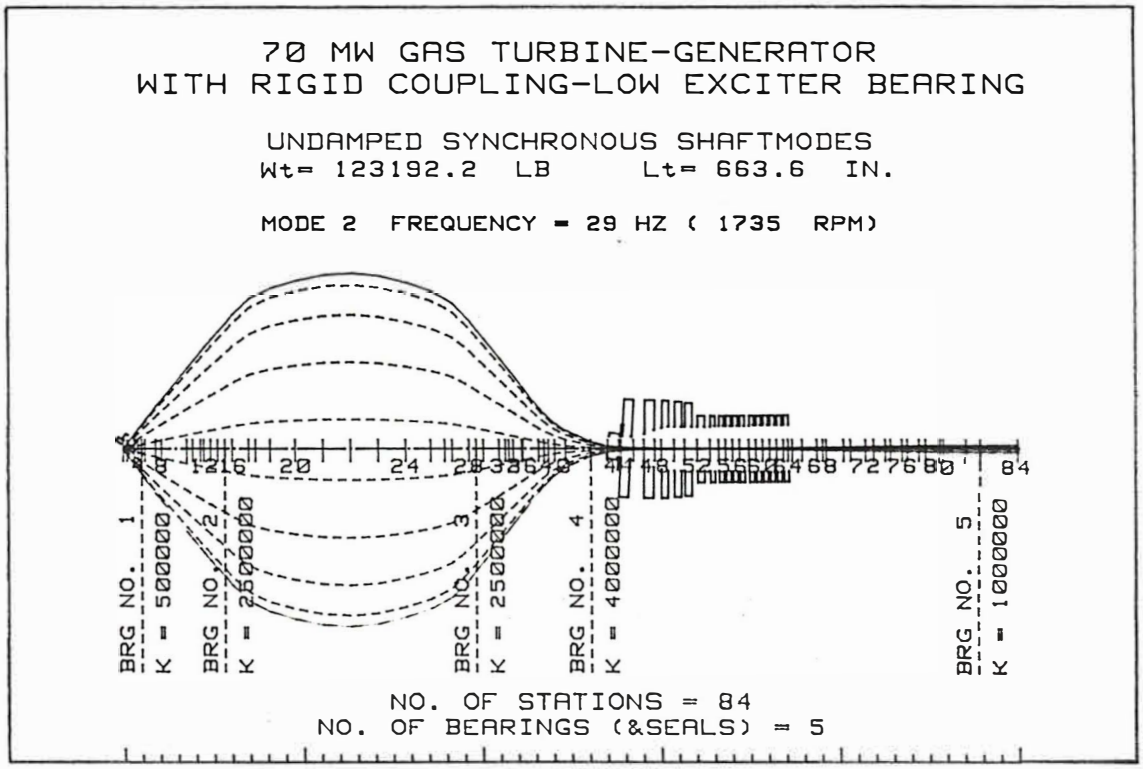


Figure 3.10 Animated 1st Generator Mode of 70 MW Turbine-Generator

bine bearing and 35% is strain energy of bending. Therefore, the hot end turbine bearing stiffness will have a predominant effect on controlling the gas turbine first critical speed.

Fig 3.10 represents the first mode of the generator at 1,735 RPM. Note that in this figure, the exciter end is a node point. There is 74% of the strain energy for this mode associated with the generator inboard and outboard bearings. Therefore, the occurrence of the generator first mode is strongly governed by the inboard and outboard generator bearings.

Fig 3.11 represents the second critical speed of the gas turbine. Note that the center plane of the gas turbine is a node point. Therefore, the center plane is very effective in balancing out the gas turbine first mode, but will not influence the turbine second critical speed. If high vibrations occur around 1,000 RPM on the gas turbine, then the gas turbine center plane will be very effective in balancing out this mode without substantially influencing the gas turbine second critical speed. The strain energy distribution for the gas turbine second critical speed at 2,143 RPM has 61% of the strain energy associated with the gas turbine bearings. Hence, the gas turbine 2nd mode is controlled only by the two tilting-pad bearings of the gas turbine.

Fig 3.12 and Fig 3.13 represent the system 4th and 5th critical speeds. These critical speeds are of particular importance since they occur near the operating speed range. Of major concern, is the 5th mode at 4,500 RPM. If the exciter end bearing stiffness is actually lower than 500,000 Lb/in, then there will be three generator critical speeds. In both of these critical speed modeshapes, the maximum amplitude occurs at the exciter end. This modeshape is controlled predominantly by the inboard and outboard generator bearings and somewhat by the exciter end bearing. The exciter end bearing has 33% of the strain energy. For the third generator critical speed or the fifth system critical speed at 4,500 RPM, a reduction in the exciter end bearing stiffness will cause the third generator critical speed to reduce to the operating speed range of 3,600 RPM. Table 3.3 represents a summary of the 5 critical speeds of the 70 MW turbine-generator.

Analysis of the experimental data of the exciter end polar plots and also of the high magnitude of the exciter influence coefficients for unit #2 indicated that the exciter was operating near a critical speed. It is possible that the exciter is operating at its third critical speed. This made the problem of balancing the unit #2 70 MW turbine-generator extremely difficult because of the high sensitivity and nonlinearity of the exciter end. It may be possible to reduce the sensitivity of the exciter end by a redesign of the bearing support at the exciter location. The current unit #2 exciter had an unacceptably high influence coefficient at running speed. These influcnce coefficients will be described in detail in the later sections.

In unit #1, one of the difficulties experienced in balancing the gas turbine end appeared to be due to the operation near a third turbine critical speed. This behavior was not experienced on the second unit.

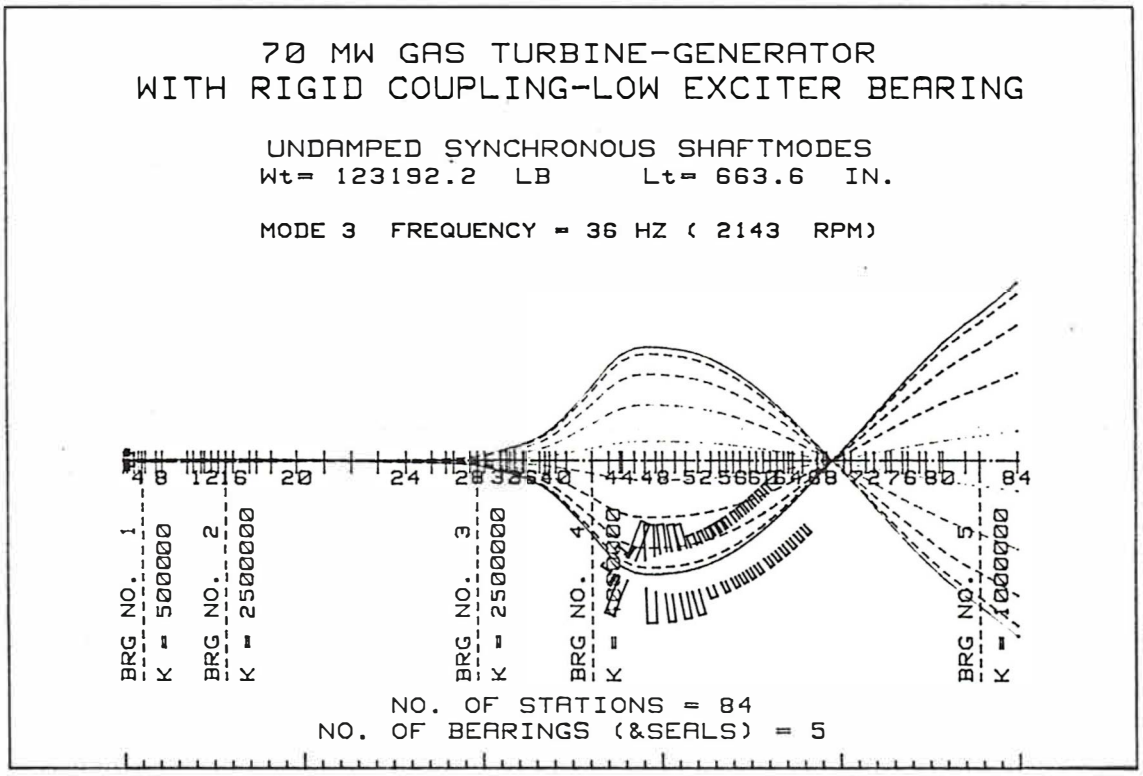


Figure 3.11 Animated 2nd Turbine Mode of 70 MW Turbine-Generator

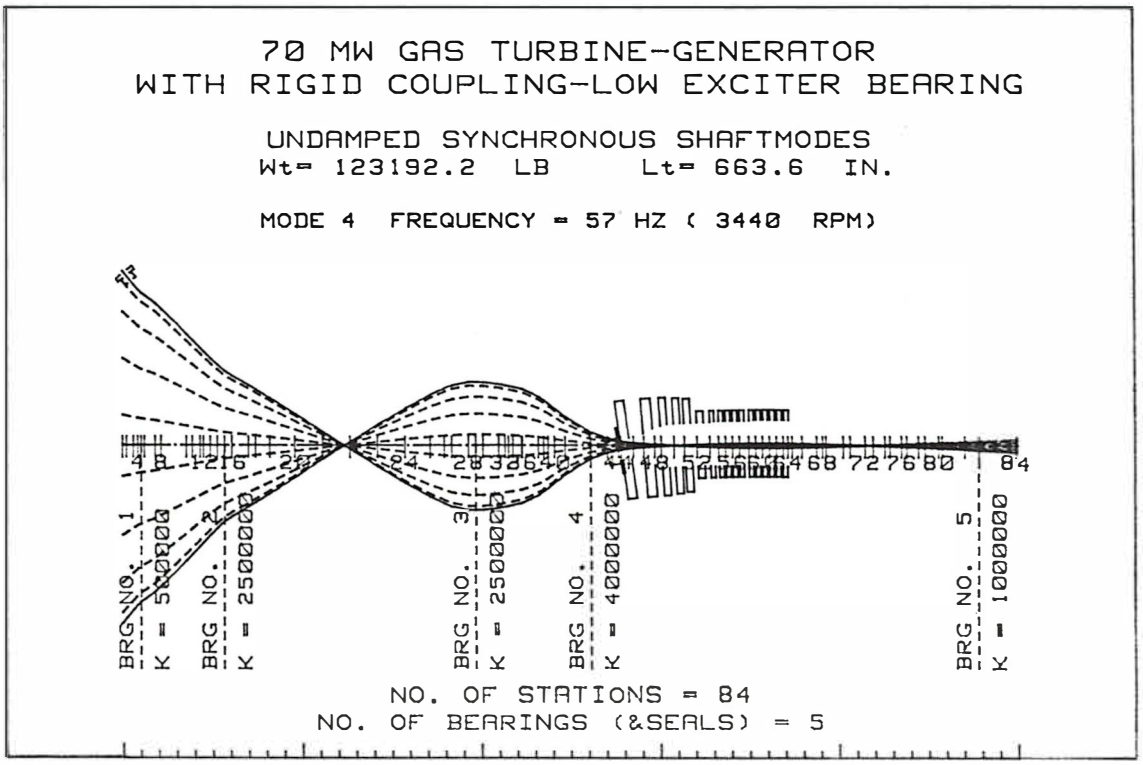


Figure 3.12 Animated 3rd Generator Mode of 70 MW Turbine-Generator

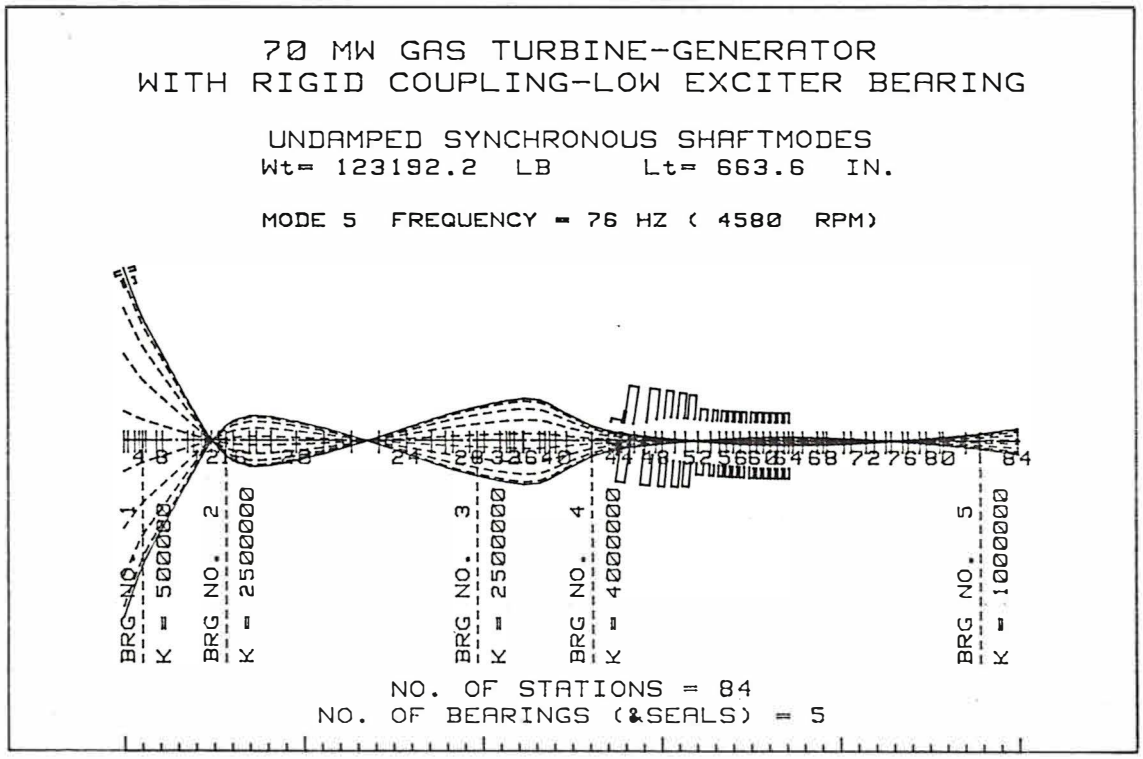


Figure 3.13 Animated 4th Generator Mode of 70 MW Turbine-Generator

IV. SAMPLE EXPERIMENTAL DATA on TYPICAL 70 MW TURBINE-GENERATOR

4.1 Gas Turbine Vibration - Turbine Bearing

This section describes typical experimental vibration data obtained from a 70 MW W501 gas turbine-generator. Fig 4.1 represents the total vertical motion at the hot end of the gas turbine with 38.9 oz applied at the compressor location. The dotted line in the figure represents the absolute motion and the solid line represents the amplitude corrected for the slow roll vector.

The turbine and generator first critical speeds are around 1,100 and 1,400 RPM in this system. A second critical speed is observed at 2,000 RPM. This is a rotor casing mode. The second turbine critical speed is approximately 2,800 RPM. At the hot end of the gas turbine, the amplitude is extremely large at 3,600 RPM. The gas turbine appears to be operating near a critical speed. In this system (unit #1), it was believed that the turbine is operating near its third critical speed due to the lack tightness at the center span bolted connection. The hot end of this particular gas turbine was extremely difficult to balance because of operation in the vicinity of a critical speed.

Fig 4.2 represents the polar diagram of the hot end of the gas turbine. This particular polar diagram was generated using the BENTLY ADRE system in conjunction with a HEWLETT PACKARD computer. From the observation of the peak amplitudes on the polar diagram and from the rate of change of phase angle, one can estimate the location of the various critical speeds. It is apparent, for example, that the gas turbine first critical speed is at 1,100 RPM. As the machine speed is increased from 3,000 to 3,600 RPM, there is a large phase change of approximately 120°. This is an indication that the hot end of the gas turbine is approaching its third critical speed. The turbine end influence coefficients, therefore, will be sensitive to various such effects as thermal bow, unbalance, and power level.

Fig 4.3 represents the relative motion at the hot end of the gas turbine with the run-out vector subtracted from the vibration data. The difference in amplitude between Figs 4.3 and 4.1 represents the casing vibration. In Fig 4.3, the first turbine critical speed is clearly indicated at 1,100 RPM. The gas turbine second critical speed appears to be at 3,000 RPM. However, it is seen that instead of the amplitude decreasing above 3,200 RPM, the amplitude continues to increase. Fig 4.4 represents the polar diagram for the relative rotor casing amplitude at the gas turbine location. The large phase angle change of approximately 120° observed for a speed change of 3,000 to 3,600 RPM is an indication of the gas turbine approaching a third critical speed. This rapid phase angle change near the running speed made it extremely difficult to balance the hot end of the gas turbine. There was also observed a considerable difference in amplitude and phase at 3,600 RPM when the power level was increased to 70 MW. In order to successfully balance this machine, the least squared error constrained linear program was employed with data measurements taken at 10 MW and at the full power level of 70 MW.

4.2 Gas Turbine Vibration - Compressor Bearing

Fig 4.5 represents the absolute motion at the compressor end of the gas turbine with 39 oz applied at the compressor location. The application of the large correction weight at the compressor end of the gas turbine causes an excitation of the gas turbine second critical speed at 2,700 RPM. Above 2,700 RPM, the amplitude reduces at the compressor end, whereas the amplitude increases at the hot end of the gas turbine. The

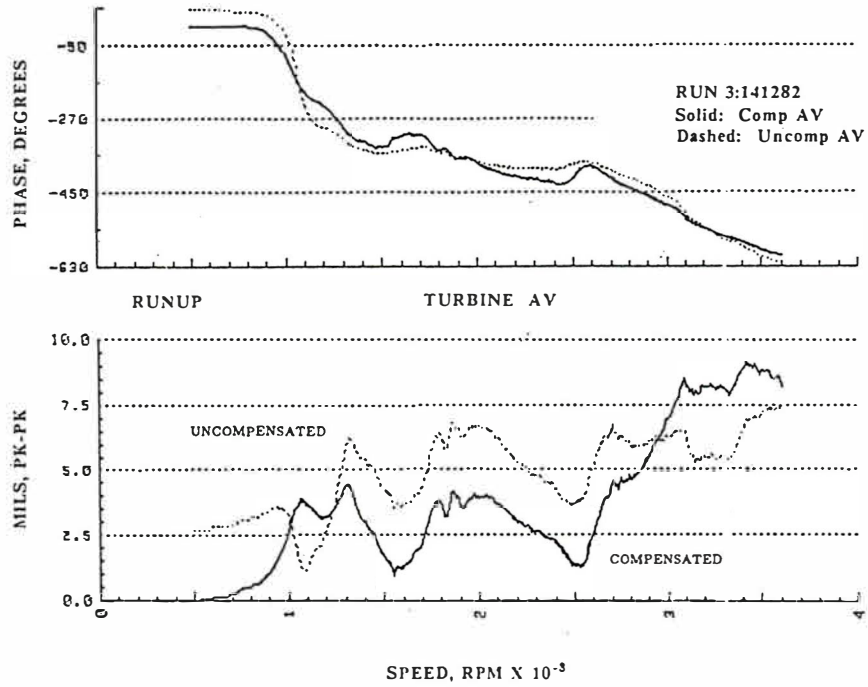


Figure 4.1 Absolute Synchronous Gas Turbine Motion With and Without Compensation With Large Compressor End Unbalance - U_c = 38.9 oz @ 181°, Z_o = 2.70 mils @ -5°

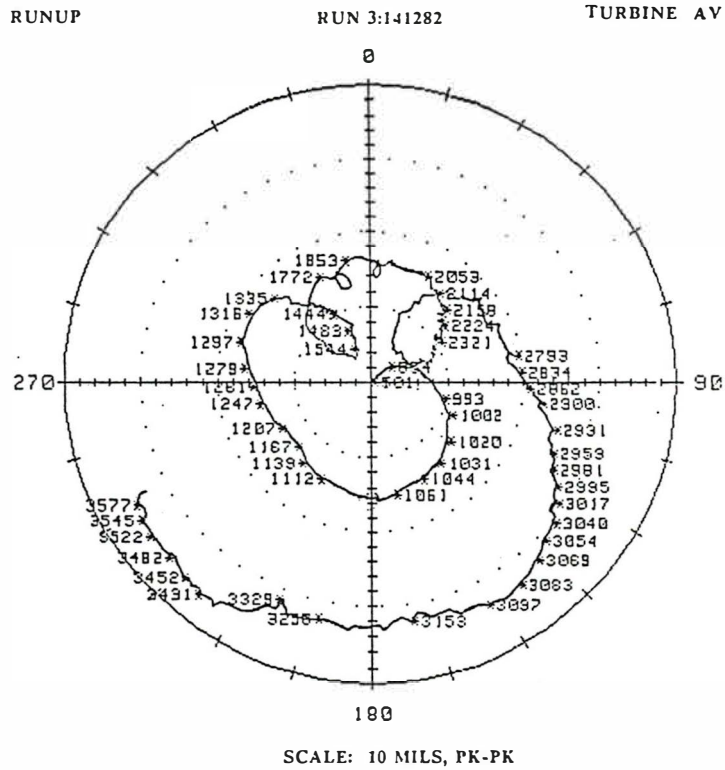


Figure 4.2 Polar Plot of Compensated Absolute Gas Turbine Motion With Large Compressor End Unbalance - U_c = 38.9 oz @ 181°, Z_o = 2.70 mils @ -5°

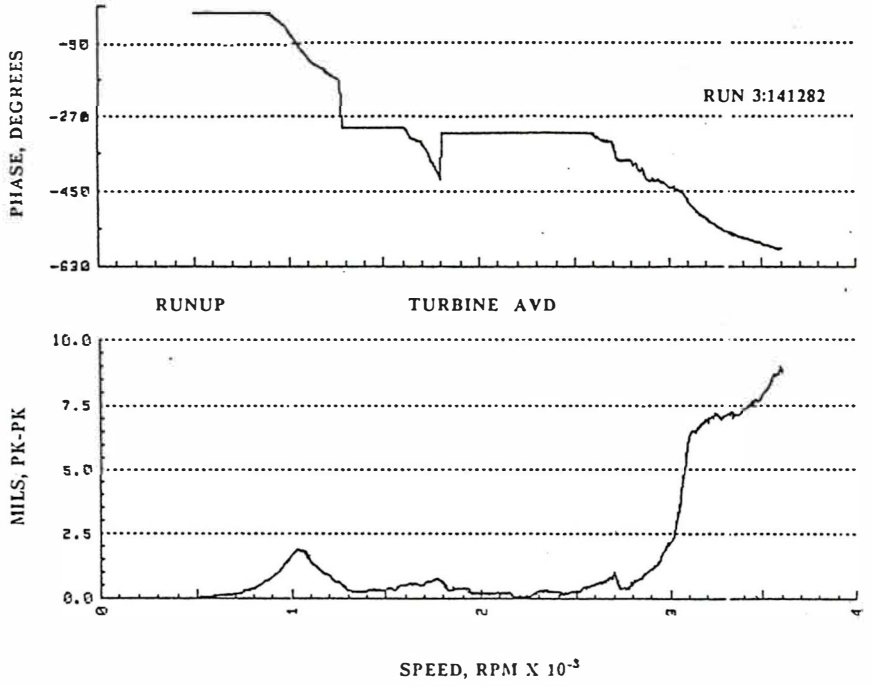


Figure 4.3 Relative Compensated Synchronous Shaft Motion of Gas Turbine With Large Compressor End Unbalance - U_c = 38.9 oz @ 181°, Z_o = 2.40 mils @ -12°

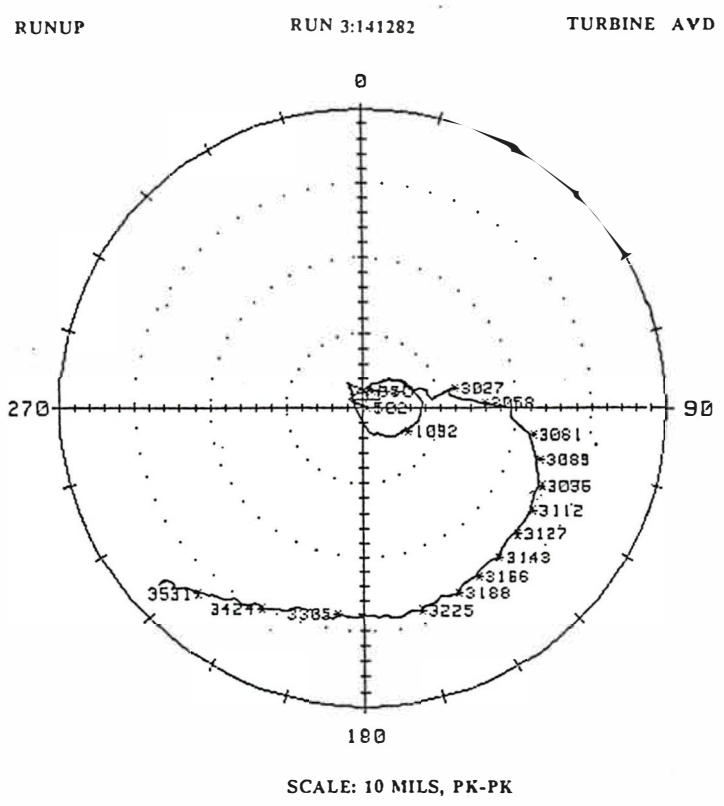


Figure 4.4 Polar Plot of Relative Compensated Synchronous Shaft Motion of Gas Turbine With Large Compressor End Unbalance - U_c = 38.9 oz @ 181°, Z_o =2.40 mils @ -12°

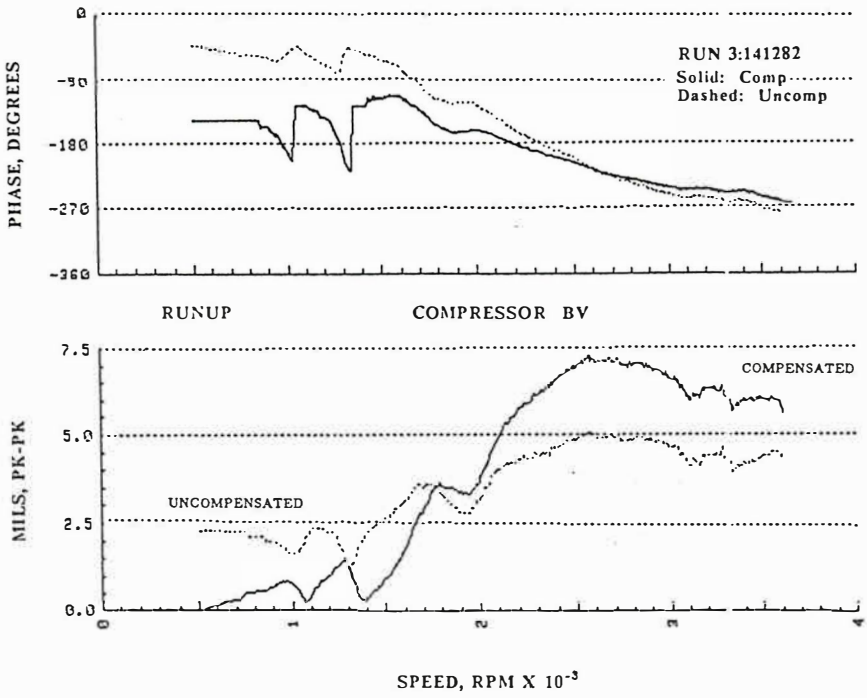


Figure 4.5 Absolute Compressor Synchronous Compensated and Uncompensated Motion With Large Compressor End Unbalance - U_c = 38.9 oz @ 181°, Z_o = 2.2 mils @ -41°

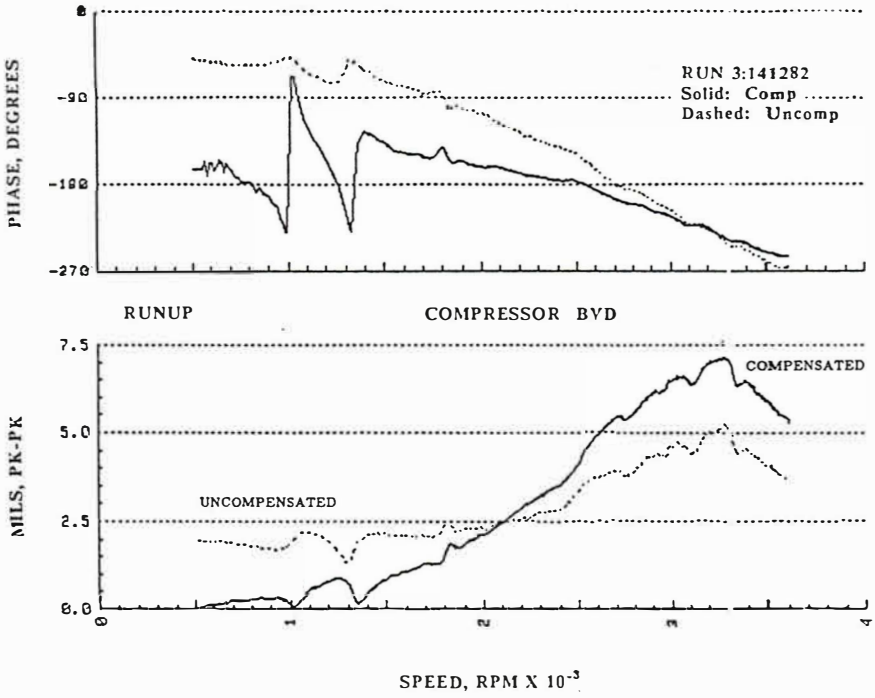


Figure 4.6 Relative Compensated and Uncompensated Compressor Motion With Large Compressor End Unbalance - U_c = 38.9 oz @ 181°, Z_o = 1.96 mils @ -50°

vibration at the compressor end is well behaved in comparison with the characteristics at the hot end. Casing effects are not as clearly evident at the compressor end of the gas turbine.

Fig 4.6 represents the relative shaft-to-casing motion at the compressor end location. The dotted line is the relative rotor casing motion at the compressor bearing location. The solid line is the shaft motion with the low speed run-out subtracted. The slow roll vector is almost 2 mils. This is an indication that the rotor may have been bent during previous balancing attempts by the OEM field engineer. By subtracting the slow roll vector, a considerably different amplitude and phase angle pattern is observed. The large phase change observed with the slow roll vector subtracted is an indication of the presence of the critical speeds. There is a large phase change at 1,000 RPM and at 1,350 RPM. The phase change at 1,000 is the turbine first critical speed and the phase change at 1,350 RPM is the generator first critical speed. The gas turbine second critical speed appears to be at 3,300 RPM when the relative shaft casing motion is examined. Note that the compressor end motion reduced after passing through the second critical speed, whereas the hot end of the gas turbine continues to increase. This characteristic of the W501 gas turbine considerably complicates the balancing procedure. The rotor has its highest sensitivity factors near running speed. It is also important to note that, if a rotor is operating near a critical speed, then it is highly susceptible to thermal bowing.

4.3 Generator Vibration

Fig 4.7 represents the absolute motion of the inboard generator bearing. The large unbalance at the compressor end of the gas turbine causes little excitation of the generator first critical speed at 1,300 RPM. There is an observed rotor casing mode at 2,300 RPM. The application of the large weight at the compressor station causes a high excitation of the generator second critical speed at 3,200 RPM. The amplitude of motion at the generator inboard bearing is well behaved. Above 3,200 RPM, the amplitude reduces as would be expected when passing through the second critical speed. Fig 4.8 represents the polar diagram for the inboard generator bearing.

Fig 4.9 represents the absolute amplitude and phase angle change at the outboard generator bearing with the slow roll vector subtracted. The generator second critical speed is clearly seen at 3,100 RPM. Fig 4.10 represents the generator outboard bearing polar diagram. A comparison of the polar diagrams in Figs 4.8 and 4.10 for the inboard and outboard generator bearings respectively, clearly shows that at 3,100 RPM the motion at these bearings is out of phase. This is an indication that the generator is operating on its second critical speed.

4.4 Exciter - Generator Non-linear Jump Response Due to Seal Rub

Fig 4.11 represents the absolute motion at the exciter end with and without run-out subtraction. A non-linear jump is observed at the exciter end at 2,300 RPM. This may have been due to a seal rub. The exciter end also indicates that a generator critical speed occurs around 3,300 RPM. Fig 4.12 represents the polar diagram for the exciter end. Note that the exciter end is 180° out of phase to the outboard bearing at 3,200 RPM and in phase to the inboard bearing.

Fig 4.13 represents motion at the outboard generator bearing for one of the unbalance runs on the W501 gas turbine. In this particular run, a non-linear jump was observed at the generator and exciter end at 3,200 RPM. This non-linear jump is clearly seen in the polar diagram of Fig 4.14. When viewed in the polar plane, it can be seen that the magnitude of the amplitude change is much more severe. The magnitude of the non-linear jump is 2.7 mils. After the jump occurs, the phase angle change is erratic and in

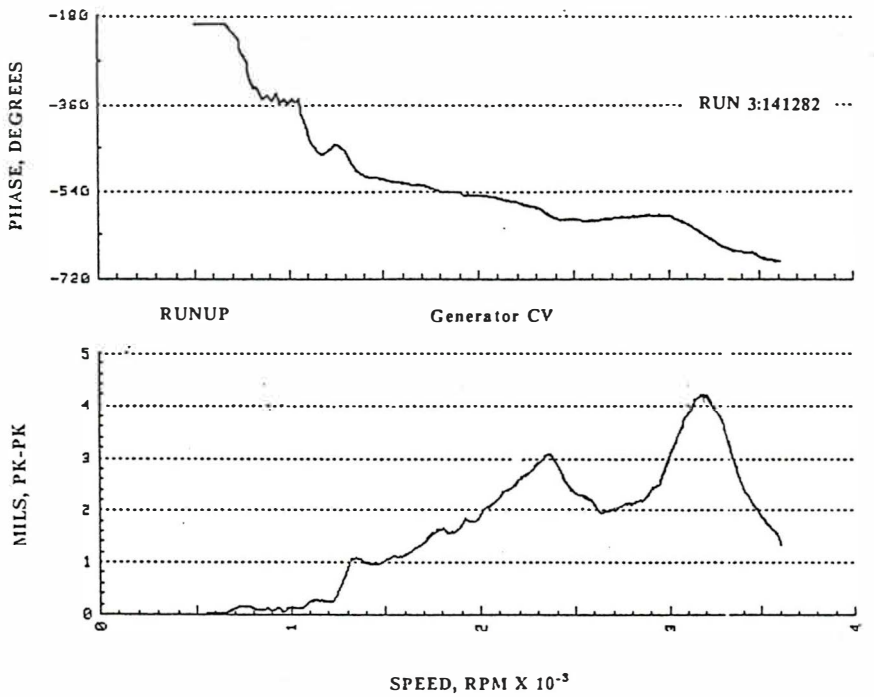


Figure 4.7 Inboard Generator Compensated Motion With Large Compressor End Unbalance
U_c = 38.9 oz @ 181°, Z_o = .12 mils @ -178°

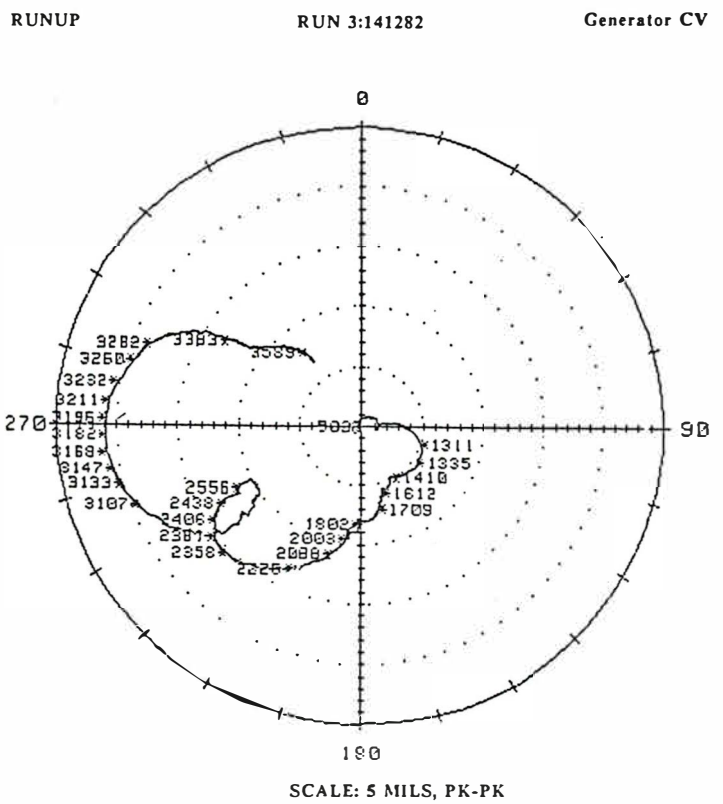


Figure 4.8 Inboard Generator Polar Plot With Large Compressor End Unbalance
U_c = 38.9 oz @ 181°, Z_o = .12 mils @ -178°

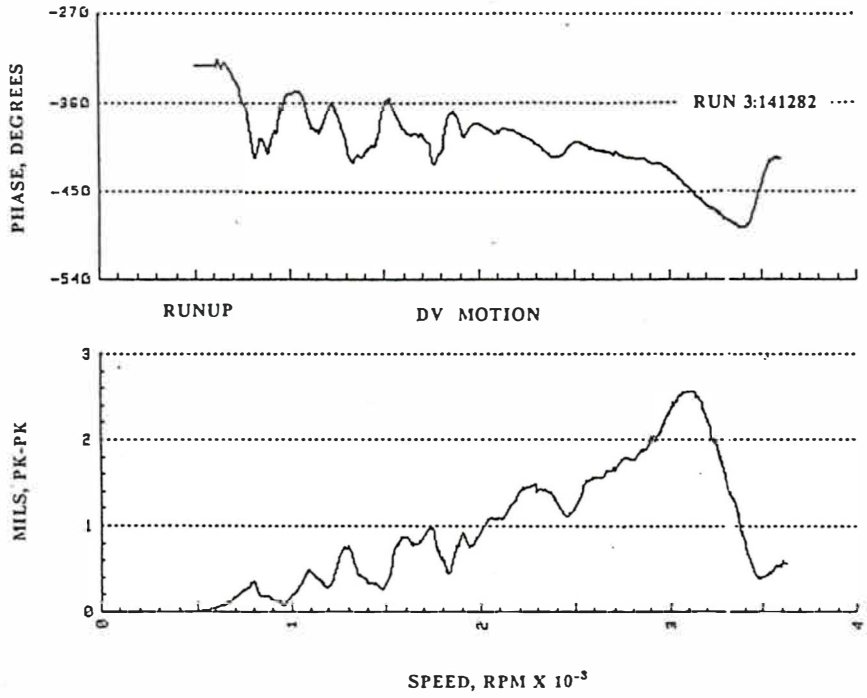


Figure 4.9 Generator Outboard Motion With Large Compressor End Unbalance
U_c = 38.9 oz @ 181°, Z_o = 0.09 mils @ -203°

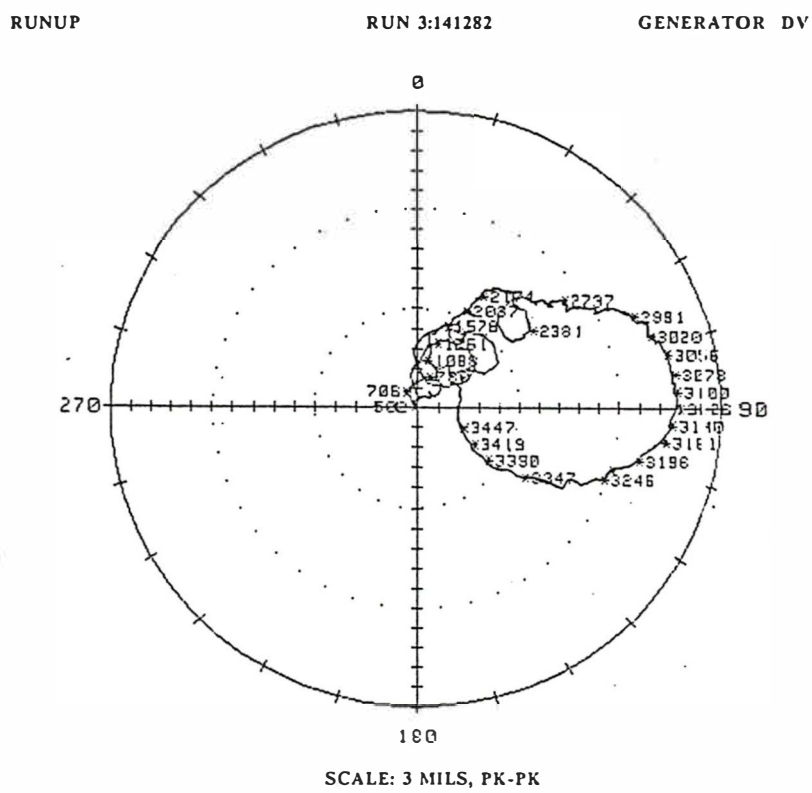


Figure 4.10 Generator Outboard Polar Motion With Large Compressor End Unbalance
U_c = 38.9 oz @ 181°, Z_o = 0.09 mils @ -203°

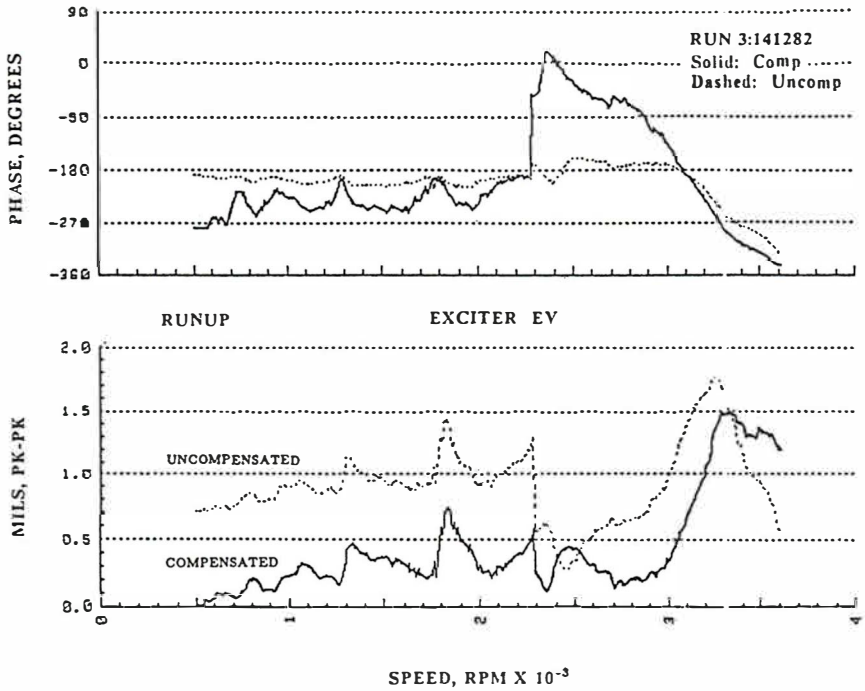


Figure 4.11 Absolute Exciter End Motion and Phase With Large Compressor End Unbalance Showing Seal Rub at 2,300 RPM - U_c = 38.9 oz @ 181°, Z_o = 0.72 mils @ -187°

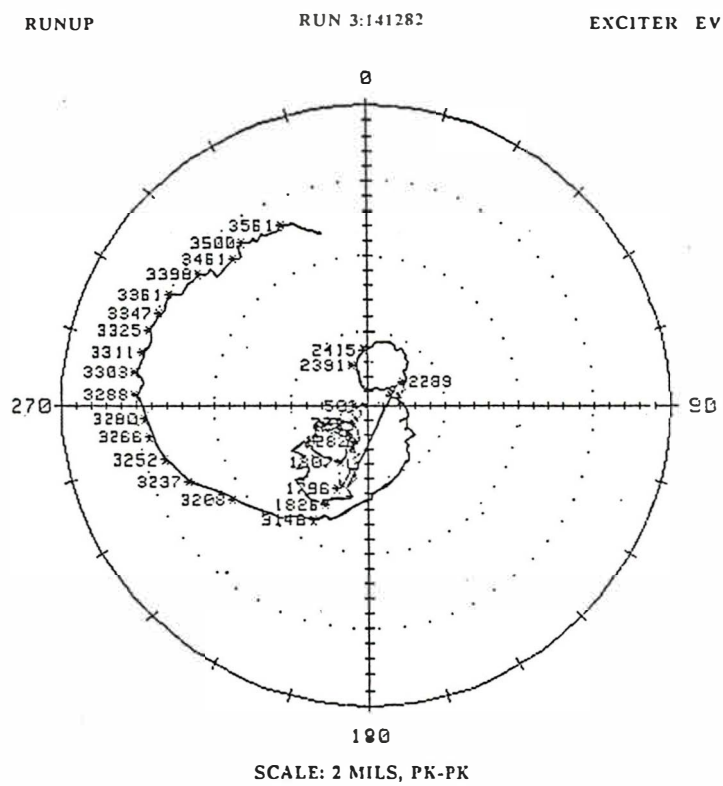


Figure 4.12 Exciter Polar Diagram With Large Compressor End Unbalance Showing Seal Rub at 2,300 RPM - U_c = 38.9 oz @ 181°, Z_o = 0.72 mils @ -187°

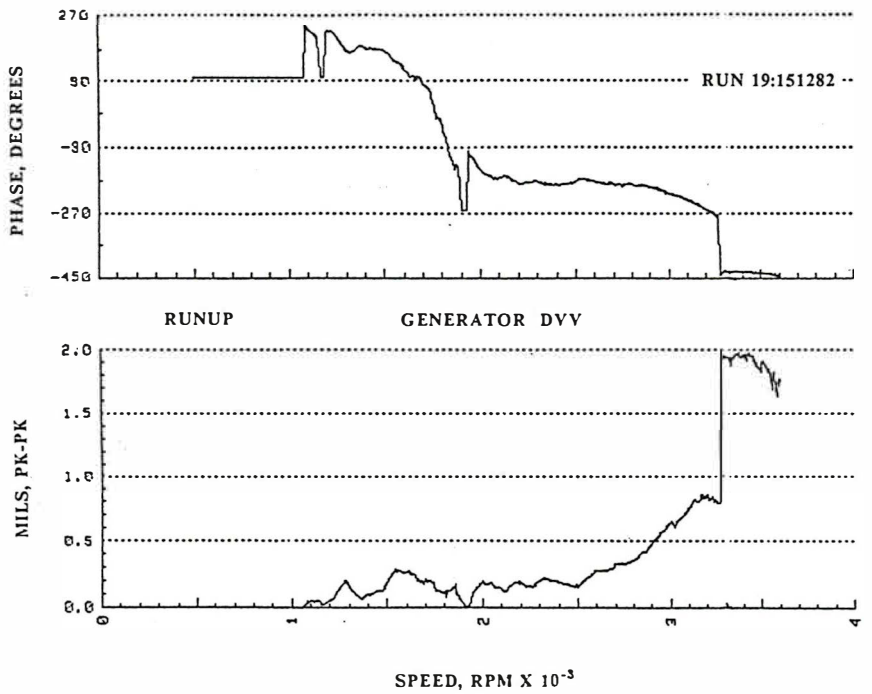


Figure 4.13 Generator Outboard Casing Motion With Turbine and Compressor Unbalance Showing Large Nonlinear Jump at 3,300 RPM - U_c = 10 oz @ 181°, U_t = 10 oz @ 270°

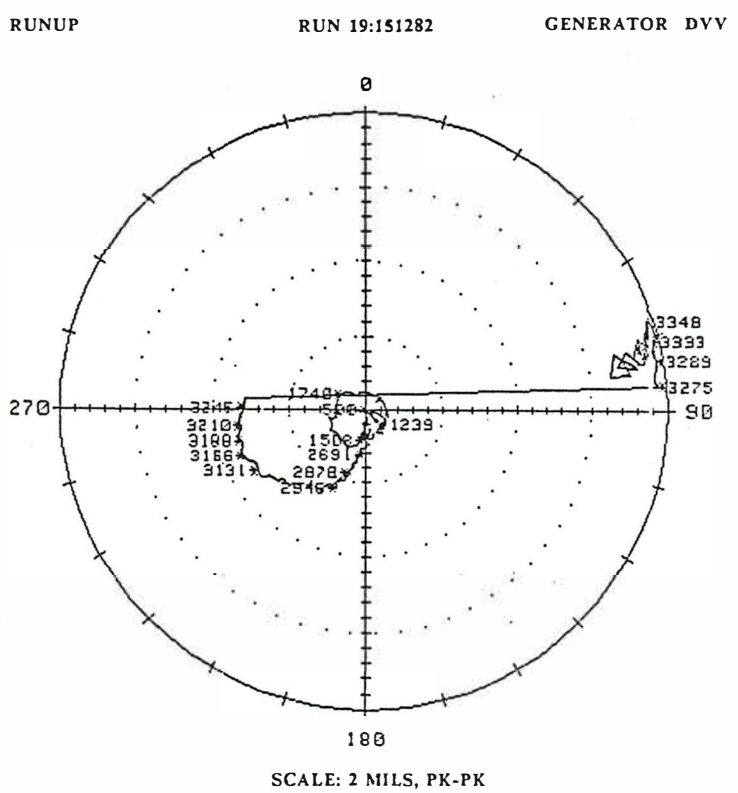


Figure 4.14 Polar Plot of Generator Outboard Casing Motion With Turbine and Compressor Unbalance Showing Large Nonlinear Jump at 3,300 RPM - U_c = 10 oz @ 181°, U_t = 10 oz @ 270°

the direction of rotation. This is an indication of a seal rub at the exciter-generator location. When this occurs, the influence coefficients throughout the entire machine are shifted. The turbine-generator can not be accurately balanced if a seal rub occurs.

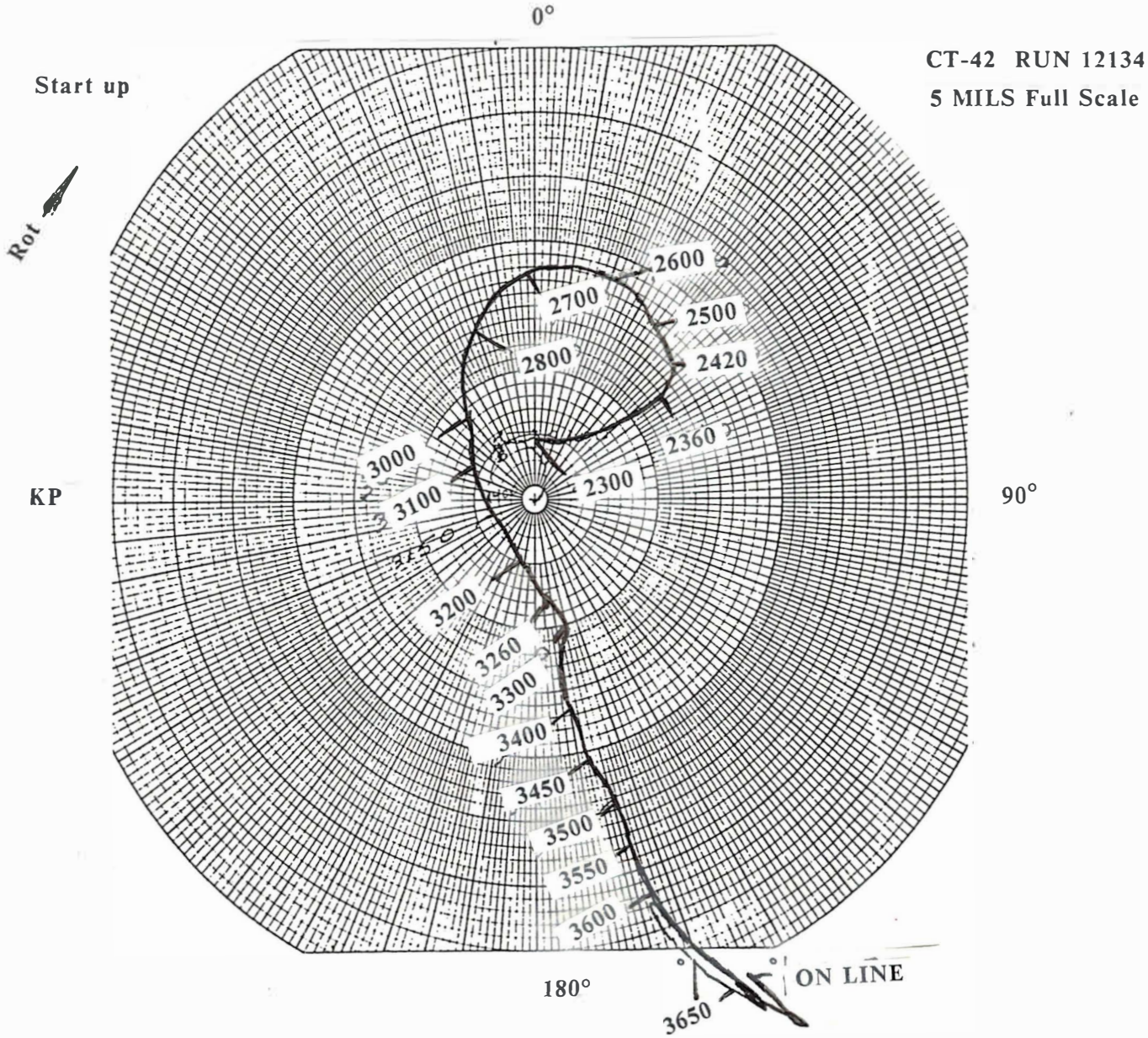
These figures represent typical data obtained on a 70 MW turbine-generator. The turbine-generator was successfully balanced using the least squared error balancing program LSQBAL developed by Dr. Gunter. In order to successfully balance the turbine-generator, the data had to be recorded at several speeds and at various power levels. The OEM field engineer spent over three months attempting to balance unit #1 without success. Accurate balancing of the W501 gas turbine is very difficult to accomplish without the use of a computer. This machine is very sensitive to thermal bowing and seal rubs due to the operation of critical speeds in the operating speed range.

4.5 Non-Linear Response of Exciter End

In the balancing runs performed on unit #2, non-linear response was encountered at the exciter end. In the examimation of the polar plots of the inboard and outboard generator bearings, a critical speed around 3,400 RPM appeared. When the rotor speed was increased above 3,400 RPM, the amplitude of motion reduced. The polar diagram went through a 90° phase shift when passing through the critical speed. However, at the exciter end, the response and the influence coefficients at 3,600 RPM were highly dependent on the unbalance level.

Fig 4.15 represents the polar diagram obtained on the exciter end with unbalance of 93 grams at 65°. Fig (4.15) represents the analogue polar plot of the exciter amplitude and phase. The turbine second critical speed at 2,700 RPM is clearly evident. The generator second critical speed at 3,300 RPM is not apparent. Instead of the rotor amplitude "turning the corner", the motion increases rapidly outward with little phase change. This behavior is highly non-linear and also suggests the presence of a third exciter critical speed above 3,600 RPM. The amplitude of motion increased until the 7 mil limit was reached. At this point the turbine-generator was tripped off line. When this type of non-linear behavior occurs, the exciter end influence coefficient increases by a factor of five over the response characteristics for low levels of unbalance. This makes balancing the exciter end more complex because of the variation of the influence coefficients with unbalance level. In unit #2, it was found that the placement of two washers at the exciter balance plane at a deviation of 45° from the correct balance location could shut down a 60 ton turbine-generator. Redesign of the exciter end bearing may alleviate some of the non-linear characteristics encountered in balancing.

Figure 4.15 Exciter Bearing Motion With 93 Gm @ 65 Deg Showing Nonlinear Response Above 3,300 RPM



V. FIELD BALANCING 70 MW TURBINE-GENERATORS

5.1 Influence Coefficients

The method used to balance the 70 MW turbine-generators was by means of the multi-plane least squared error influence coefficient procedure with constraints. In the influence coefficient procedure, trial weights are placed at various locations along the shaft. A particular weight is placed at a location such as the coupling, turbine or compressor end of the gas turbine, or exciter end of the generator and the resulting response measured at 5 stations and various speeds after the weight has been applied. By subtracting the vector difference of the amplitude at a station before and after the application of the trial weight, an influence coefficient is generated for that station.

The influence coefficients are speed dependent. That is, the influence coefficients at one speed are not the same at a lower or higher speed. This is because the rotor has various critical speeds in its operating speed range. It is therefore important to have an understanding of the critical speed mode shapes of the system. For example, at a given speed, a balance plane may be located at a nodal point and, hence, has very little influence on the balancing at that speed. The coupling plane, for example, has very little influence on the gas turbine first critical speed around 1,100 RPM, but has a substantial influence at 3,200 RPM. In the balancing technique employed by the OEM on unit #1, data only at running speed was used. In the balancing procedure used by the authors, various speed ranges were incorporated, as well as various probe readings. This balancing procedure is called the LEAST SQUARED ERROR BALANCING METHOD. The computer aided balancing procedure has been extended to include one-shot balancing (no trial weights) based on previous runs of the same or similar machines, and also has the capability to balance without removing trial weights. In order to perform this type of calculation, a high speed micro-computer is employed. The balancing procedure employed by the OEM for unit #1 was simple single-plane balancing at the running speed.

In the theory of balancing, the system is assumed to be linear. This implies that if a balance calibration weight is doubled, the influence coefficients remain constant. From an examination of the experimental data, it was observed that the influence coefficients are not constant at a particular speed, but are dependent upon various factors. The non-linearity in the system can be caused by several factors. It is felt that non-linear rotor response has caused some balancing problems with the 70 MW gas turbine-generators in the past. Some of the non-linearities that cause a deviation of the influence coefficients from a linear system are coupling misalignment, generator seal rubs, thermal bowing due to power level, and non-linear fluid film bearings. In spite of these various non-linear effects, successful balance of the various turbine-generators was achieved because of the application of the LEAST SQUARED ERROR BALANCING procedure, which minimizes the response at the various speeds selected.

5.2 Compressor End Balancing Calculations

Figure 4.1 represents the gas turbine hot end motion with approximately 39 oz of balance correction applied to the compressor balance plane. The vibration levels with this large correction at the compressor plane resulted in unacceptably high gas turbine vibrations. In the first balancing run performed by the authors on unit #1, two of the balance weights were removed at the compressor location. Table 5.1 represents a typical calculation of the balancing correction if only one data set is used (simple single-plane balancing). In this example, the turbine end data was used to calculate a balancing

Table 5.1 Compressor End Balance Calculation of 70 MW Gas Turbine Using Turbine End T_e Data only at 3,600 RPM

RUN #7,-20 OZ @359 Deg LEAD AT COMPRESSOR END LOCATION ON GAS TURBINE
SINGLE PLANE BAL; 2 OF 4 BAL WEIGHTS(38.9 OZ) REMOVED AT Ce PLANE

NO. OF PROBES = 1 ; NO. OF SPEEDS = 1 ; NO. OF BALANCE PLANES = 1

THE SPEEDS FOR BALANCING DATA:

SPEED NO.	SPEED (RPM)
1	3600

SLOWROLL DVF2 VECTOR: AMPLITUTE(MILS), PHASE ANG (DEGREES)

PROBE NO.	AMPLITUDE (MILS)	PHASE (DEG)	PROBE LOCATION	PROBE TYPE	PHASE CONV
1	0.00	0.0	Te	S.RIDER	LEAD

INITIAL READINGS OF AMP & PHASE:

SPEED NO.	SPEED (RPM)	PROBE NO.	PROBE LOC	AMPLITUDE (MILS)	PHASE (DEG LEAD)
1	3600	1	Te	6.50	113.0

TRIAL UNBALANCES:

PLANE	MAGNITUDE	PHASE ANGLE (DEG LEAD)	BALANCE LOC	BALANCE (UNITS)	TRIAL WEIGHT AFTER RUN
1	20.00	359.0	COMP.	OZ	WEIGHT LEFT IN

READINGS AFTER A TRIAL WEIGHT HAS BEEN ADDED:

BALANCE PLANE	SPEED NO.	SPEED (RPM)	PROBE NO.	AMPLITUDE (MILS)	PHASE (DEG LEAD)
1	1	3600	1	3.10	99.0

THE BALANCE CORRECTION WEIGHTS:

PLANE	Ux (OZ)	Uy (OZ)	MAGNITUDE ABSOLUTE VALUE	ANGLE (DEG) LEAD-CCW (LAG-CW)	MAGNITUDE TRIM (+TRIAL)	ANGLE (DEG) LEAD (LAG)
1	35.45	-8.26	36.40	346.9(13.1)	17.36	332.9(27.1)

SUMMARY TABLE OF THE SYSTEM INFLUENCE COEFFICIENTS

UNITS OF INFLUENCE COEF:MILS/OZ

BAL #	UBAL MAG	SPD #	SPEED (RPM)	PROBE #	AMP1 (MILS)	PHASE1 (LEAD)	AMP2 (MILS)	PHASE2 (LEAD)	INF. COEF (MAG)	(ANGLE) LEAD(REL LAG)
1	20.00	1	3600	1	6.50	113	3.10	99	.1786	306 53

INITIAL READINGS AND PREDICTED COMP.RESPONSE

SPEED (RPM)	PROBE NO.	PROBE LOC	INITIAL AMP MILS	INITIAL PHASE LEAD	RESIDUAL AMP MILS	RESIDUAL PHASE LEAD (LAG)
3600	1	Te	6.50	113	0.000	297 63

Table 5.2 Compressor End Balance Calculation Using Compressor C_e and Turbine T_e Data at Various Speeds - U_c = 20 oz @ 359° - Initial Vibration Data

TRIAL WEIGHT LEFT IN PLACE AFTER RUN

NO. OF PROBES = 2 ; NO. OF SPEEDS = 6 ; NO. OF BALANCE PLANES = 1

THE SPEEDS FOR BALANCING DATA:

SPEED NO.	SPEED (RPM)
1	1066 TURBINE 1ST CRIT
2	1702 GENERATOR 1ST CRIT
3	2300 TURBINE 2ND CRIT
4	2756 TURBINE-GENERATOR CRIT
5	3600 RUNNING SPEED: COLD OMW
6	3600 RUNNING SPEED: HOT 70MW

SLOWROLL DVF2 VECTOR: AMPLITUTE(MILS), PHASE ANG (DEGREES)

PROBE NO.	AMPLITUDE (MILS)	PHASE (DEG)	PROBE LOCATION	PROBE TYPE	PHASE CONV
1	0.00	0.0	Ce	S RIDER	LEAD
2	0.00	0.0	Te	S RIDER	LEAD

INITIAL READINGS OF AMP & PHASE:

SPEED NO.	SPEED (RPM)	PROBE NO.	PROBE LOC	AMPLITUDE (MILS)	PHASE (DEG LEAD)
1	1066	1	Ce	1.70	339.0
1	1066	2	Te	4.60	54.0
2	1702	1	Ce	2.80	226.0
2	1702	2	Te	6.70	10.0
3	2300	1	Ce	3.90	145.0
3	2300	2	Te	3.70	333.0
4	2756	1	Ce	4.50	103.0
4	2756	2	Te	4.70	302.0
5	3600	1	Ce	5.40	74.0
5	3600	2	Te	6.50	113.0
6	3600	1	Ce	1.98	98.0
6	3600	2	Te	5.70	114.0

correction. The application of the 20 oz causes a reduction from 6.5 mils to 3.1 mils at the gas turbine hot end at 3,600 RPM. A final balancing correction of 36.4 oz at 347° is computed.

In the single-plane influence computation of balancing, the amplitude at the measurement location is reduced to zero. This may cause extremely high vibrations at other probe locations and at other speeds. The vibrations at the lower critical speeds may be of such a magnitude that the turbine-generator set may trip out in attempting to come up to speed. The single influence coefficient method of balancing is highly undesirable.

In order to perform a correct single-plane balancing at the compressor end which will not unset the gas turbine at the lower critical speeds, it is necessary to employ vibration readings at various speeds at both the gas turbine hot end and the compressor end. Table 5.2 represents the vibration data recorded at 5 speeds and 2 power levels at 3,600 RPM. In this balancing calculation, the data was recorded during runup. By means of the TURBOBALANCER, a thermal print-out was obtained at approximately 50 RPM increments on rotor speed, amplitude and phase for the turbine and compressor vertical probes. The first 4 speeds were chosen to correspond to various turbine-generator critical speeds. Two cases were chosen at running speed. The first case corresponds to the gas turbine under 0 MW power level and the second case under the full 70 MW power level. On unit #1, it was found that the change in power level causes a response change at the compressor end of approximately 2.9 mils at 221°, and on the gas turbine end a response change of 0.42 mils 262° was recorded. These thermal changes were fairly repeatable from run to run. However, similar machines of the same class may experience considerably different thermal bows. In order to properly balance the gas turbine, it is essential to obtain vibration data on the turbine under 0 MW and full power conditions. Therefore when one is performing a base run, it is essential to load the generator up to full power in order to get the full power vibration data. The proper base data was not obtained in unit #1. The estimated full power response was computed from previous runs.

In Table 5.2, it is seen that at 1,702 RPM the turbine end vibration is 6.7 mils at 10°. If the vibration should exceed 7 mils, the turbine will trip off line. Table 5.2 represents the balance correction weight to apply at the compressor end considering the turbine end compressor vibration under 6 speed conditions. Note that the computed balance correction has been reduced from 36 oz to 22.6 oz. If the original weight is left place, then a trim weight of 5 oz at approximately 294° is called for instead of 17 oz at 330°. Shown in Table 5.3 are the influence coefficients at the various speed and probe locations. The largest influence coefficient recorded for this case corresponds to the turbine first critical speed at 1,066 RPM. The application of the weight at the compressor location causes an influence coefficient response at the turbine end of 0.21 mils/oz. Note that at this speed the compressor end influence coefficient at 1,066 RPM is only 0.06. Therefore, at the first critical speed, the response at the turbine end due to a weight applied to the compressor end is over 3 times the compressor end response. This large cross coupling effect may be due to the soft tilting-pad bearing webbed support at the gas turbine end. At running speed, the influence coefficients at the compressor and turbine end change slightly under loading. Under full load, the turbine end influence coefficient is 0.14 mils/oz, while the compressor end is only 0.06 mils/oz. Therefore, at running speed, the turbine end response due to a weight applied at the compressor location is over twice the value of the compressor response.

At the lower part of Table 5.3 is the predicted turbine response if the ideal balance correction weight is applied to the compressor location. For the single-plane balancing correction at the compressor location, it is seen that the compressor motion under 0 MW will reduce from 5.4 mils to 3.95 mils while the hot end of the gas turbine will reduce from 6.5 mils to 3.04 mils. Under full load, the hot end of the gas turbine should reduce

Table 5.3 Compressor End Balance Calculations and Influence Coefficients Using Compressor C_e and Turbine T_e Data at Various Speeds

RUN #7 SHOT AT Ce: 20 OZ @ 359 DEG ON COMPRESSOR BALANCE PLANE
TRIAL WEIGHT LEFT IN PLACE AFTER RUN

THE BALANCE CORRECTION WEIGHTS:

PLANE	Ux (OZ)	Uy	MAGNITUDE ABSOLUTE VALUE	ANGLE (DEG) LEAD-CCW (LAG-CW)	MAGNITUDE TRIM (+TRIAL)	ANGLE (DEG) LEAD (LAG)
1	22.09	-4.97	22.64	347.3(12.7)	5.08	294.4(65.6)

SUMMARY TABLE OF THE SYSTEM INFLUENCE COEFFICIENTS

UNITS OF INFLUENCE COEF:MILS/OZ

BAL #	UBAL MAG	SPD #	SPEED (RPM)	PROBE #	AMP1 (MILS)	PHASE1 (LEAD)	AMP2 (MILS)	PHASE2 (LEAD)	INF. COEF (MAG)	(ANGLE) LEAD	(REL LAG)
1	20.00	1	1066	1	1.70	339	2.60	313	.0653	279	80
1	20.00	1	1066	2	4.60	54	5.90	7	.2177	317	42
1	20.00	2	1702	1	2.80	226	3.90	232	.0577	248	111
1	20.00	2	1702	2	6.70	10	4.40	4	.1185	202	157
1	20.00	3	2300	1	3.90	145	4.20	160	.0549	228	131
1	20.00	3	2300	2	3.70	333	2.80	340	.0491	134	225
1	20.00	4	2756	1	4.50	103	3.50	120	.0771	242	117
1	20.00	4	2756	2	4.70	302	5.40	325	.1064	26	333
1	20.00	5	3600	1	5.40	74	4.10	73	.0651	258	101
1	20.00	5	3600	2	6.50	113	3.70	97	.1558	313	46
1	20.00	6	3600	1	1.98	98	1.50	141	.0676	230	129
1	20.00	6	3600	2	5.70	114	3.10	99	.1411	312	47

INITIAL READINGS AND PREDICTED COMP.RESPONSE

SPEED (RPM)	PROBE NO.	PROBE LOC	INITIAL AMP MILS	INITIAL PHASE LEAD	RESIDUAL AMP MILS	RESIDUAL PHASE LEAD	(LAG)
1066	1	Ce	1.70	339	2.567	306	54
1066	2	Te	4.60	54	5.520	357	3
1702	1	Ce	2.80	226	4.095	229	131
1702	2	Te	6.70	10	4.018	10	350
2300	1	Ce	3.90	145	4.479	160	200
2300	2	Te	3.70	333	2.819	345	15
2756	1	Ce	4.50	103	3.729	125	235
2756	2	Te	4.70	302	5.938	325	35
3600	1	Ce	5.40	74	3.948	77	283
3600	2	Te	6.50	113	3.037	104	256
3600	1	Ce	1.98	98	1.820	145	215
3600	2	Te	5.70	114	2.530	108	252

from 5.7 mils to 2.53 mils. It is of interest to note that the application of this correction weight will cause the compressor end turbine motion at the first critical speed of 1,066 RPM to increase slightly. At the turbine-generator critical speed of 2,756 RPM, the turbine amplitude will increase from 4.7 to 5.9 mils.

Since the influence coefficients at the various probes and speed conditions have been generated by the computer, it is possible to compute the new response of the rotor with an arbitrary placement of weight at the compressor location. In Table 5.4, the trim balance weight of 17.36 oz at 333° was computed to see its response on the rotor. This value was predicted using only turbine end data at 3,600 RPM. If this weight were applied to the rotor, then an amplitude of over 7.7 mils would be encountered at the gas turbine first critical speed at the hot end. This would trip the machine off line and it would not be able to come up to speed. Also, a 7 mil amplitude would occur at the turbine-generator critical speed at 2,756 RPM. It is clearly apparent that balancing predictions based on single speed readings are entirely unsatisfactory for a gas turbine-generator.

5.3 Single-Plane Balancing of the 70 MW Gas Turbine Hot End

The hot end of the 70 MW gas turbine is the most difficult to balance. From a practical standpoint, one can only perform one balancing run per day on the hot end. Once a weight is applied at the hot end of the turbine, one must wait until the next day until the machine cools off before another weight can be applied. If the application of a trial weight results in improvement of the overall turbine vibration, then this weight is normally not extracted from the hot end. Once a balance weight is applied at the hot end, it may be very difficult to remove because of thermal growth. Extraction tools have been broken in the process of attempting to remove balance weights. Also if the balance weight is dropped into the hot end section, the gas turbine must be disassembled. This results in a maintenance procedure costing over $100,000. Therefore, from a practical standpoint, balancing procedures involving the hot end of the gas turbine should not require removing the trial weight at the gas turbine. Table 5.5 represents the response and balancing calculations for a 10 oz weight applied at 270° on the hot end of the gas turbine. Note that the application of this weight has in general reduced all of the vibration levels on the gas turbine. The 10 oz weight was conservative. A final trim of 15 oz at 300° is predicted. Therefore the initial trial weight was placed in approximately the correct quadrant. It is of interest to note that at running speed the compressor end influence coefficient is 0.21 mils/oz, while the influence coefficient at the hot end is only 0.07 mils/oz. Therefore, the application of a correction weight at the gas turbine end causes an extremely high response at the compressor end. One thereby encounters the interesting paradox that if high vibrations are encountered at the compressor end at running speed, then a correction weight is called for at the hot end of the gas turbine. High vibrations at the gas turbine hot end require correction weights at the compressor location.

5.4 Multi-Plane Balancing Without Removing Trial Weights

When the weight was applied to the gas turbine end, the overall turbine vibrations were reduced. A single-plane balancing correction was calculated using turbine and compressor end data. The previous data from the compressor end run and the turbine run with the trial remaining in place can be calculated to compute a two plane balancing correction for the turbine and compressor end locations. Table 5.6 represents the speeds used in the balancing location. The two trial unbalance values were 20 oz at the compressor end and 10 oz at the turbine end. A modified form of the LEAST SQUARED ERROR BALANCING program was developed which allows one to leave the trial weights in place. This feature is extremely important when field balancing turbine-generators in which it is difficult to remove the previous trial weights.

Table 5.7 represents the trim balance values to add to the compressor and turbine end. A final trim of 8.2 oz at 319° is predicted for the compressor end and a trim of 10.7 oz at 265° is predicted for the turbine end. Also shown in Table 5.7 are the predicted gas turbine responses if these final trim weights are added to the rotor. Note that at running speed, the turbine and compressor response will be 2 mils or less. The amplitude of the gas turbine while passing through the lower critical speeds will be acceptable.

Table 5.8 represents a summary of the influence coefficients of the turbine and compressor locations for the various speeds. Note that the largest influence coefficient in the system is 0.34 mils/oz at the compressor bearing at a speed of 2,300 RPM due to an unbalance placed at the turbine end. This influence coefficient at 2,300 RPM is 5 to 10 times larger than the influence coefficients obtained at running speed.

Table 5.4 Predicted Gas Turbine Response at Various Speeds With Balance Correction Based on T_e Data Only at 3,600 RPM

ORIGINAL WEIGHT REMAIN IN PLANE(S),TRIM WEIGHT(S) ADDED

ORIGINAL TRIAL WEIGHT IN PLACE,TRIM WEIGHT ADDED
TRIAL UNBALANCES:

PLANE	MAGNITUDE (OZ)	PHASE ANGLE (DEG LEAD)	BALANCE LOC	BALANCE (UNITS)
1	17.36	333.0	COMP END	OZ

INITIAL READINGS AND PREDICTED COMP.RESPONSE

SPEED (RPM)	PROBE NO.	PROBE LOC	INITIAL AMP MILS	INITIAL PHASE LEAD	RESIDUAL AMP MILS	RESIDUAL PHASE LEAD	(LAG)
1066	1	Ce	2.60	313	3.304	296	64
1066	2	Te	5.90	7	7.708	339	21
1702	1	Ce	3.90	232	4.885	230	130
1702	2	Te	4.40	4	2.389	12	348
2300	1	Ce	4.20	160	4.961	167	193
2300	2	Te	2.80	340	2.391	357	3
2756	1	Ce	3.50	120	3.627	142	218
2756	2	Te	5.40	325	7.012	333	27
3600	1	Ce	4.10	73	3.079	81	279
3600	2	Te	3.70	97	1.116	74	286
3600	1	Ce	1.50	141	2.300	168	192
3600	2	Te	3.10	99	.702	79	281

Table 5.5 Turbine End Balance Calculation and Predicted Response Based on T_e and C_e Data at Various Speeds

BALANCE DATA WITH 4 SPEEDS WITH Ce & Te MEASUREMENTS,10 OZ @ 270 DEG
SINGLE PLANE CORRECTION,3600 ,2850 ,1054,1107 RPM- 4 SPEED SETS

THE BALANCE CORRECTION WEIGHTS:

PLANE	Ux (OZ)	Uy	MAGNITUDE ABSOLUTE VALUE	ANGLE (DEG) LEAD-CCW (LAG-CW)	MAGNITUDE TRIM (+TRIAL)	ANGLE (DEG) LEAD (LAG)
1	7.72	-23.37	24.61	288.3(71.7)	15.44	300.0(60.0)

SUMMARY TABLE OF THE SYSTEM INFLUENCE COEFFICIENTS

UNITS OF INFLUENCE COEF:MILS/OZ

BAL #	UBAL MAG	SPD #	SPEED (RPM)	PROBE #	AMP1 (MILS)	PHASE1 (LEAD)	AMP2 (MILS)	PHASE2 (LEAD)	INF. COEF (MAG)	(ANGLE) LEAD	(REL LAG)
1	10.00	1	3600	1	4.10	73	2.60	101	.2178	309	321
1	10.00	1	3600	2	3.70	97	3.40	95	.0325	28	242
1	10.00	2	2850	1	3.40	114	1.40	91	.2181	39	231
1	10.00	2	2850	2	5.50	320	4.30	327	.1339	207	63
1	10.00	3	1054	1	2.60	313	1.80	319	.0831	210	60
1	10.00	3	1054	2	5.90	7	4.30	15	.1748	257	13
1	10.00	4	1107	1	2.50	284	1.80	292	.0760	175	95
1	10.00	4	1107	2	4.50	351	3.40	4	.1412	228	42

INITIAL READINGS AND PREDICTED COMP.RESPONSE

SPEED (RPM)	PROBE NO.	PROBE LOC	INITIAL AMP MILS	INITIAL PHASE LEAD	RESIDUAL AMP MILS	RESIDUAL PHASE LEAD	(LAG)
3600	1	CE#4BRG	4.10	73	1.803	199	161
3600	2	Te#5BRG	3.70	97	3.128	88	272
2850	1	CE#4BRG	3.40	114	3.113	3	357
2850	2	Te#5BRG	5.50	320	2.233	327	33
1054	1	CE#4BRG	2.60	313	.592	295	65
1054	2	Te#5BRG	5.90	7	1.607	12	348
1107	1	CE#4BRG	2.50	284	.631	287	73
1107	2	Te#5BRG	4.50	351	1.431	28	332

Table 5.6 Compressor and Turbine Two Plane Balancing Calculation Without Removing Trial Weights - Speed Values and Trial Unbalances

RUN #7 SHOT AT Ce: 20 OZ @ 359 DEG; RUN #8 SHOT AT Te: 10 OZ @ 270 DEG
TRIAL WEIGHTS LEFT IN PLACE AFTER RUN

NO. OF PROBES = 2 ; NO. OF SPEEDS = 6 ; NO. OF BALANCE PLANES = 2

THE SPEEDS FOR BALANCING DATA:

SPEED NO.	SPEED (RPM)
1	1066 TURBINE 1ST CRIT
2	1702 GENERATOR 1ST CRIT
3	2300 TURBINE 2ND CRIT
4	2756 TURBINE-GENERATOR CRIT
5	3600 RUNNING SPEED: COLD 0MW
6	3600 RUNNING SPEED: HOT 70MW

SLOWROLL DVF2 VECTOR: AMPLITUTE(MILS), PHASE ANG (DEGREES)

PROBE NO.	AMPLITUDE (MILS)	PHASE (DEG)	PROBE LOCATION	PROBE TYPE	PHASE CONV
1	0.00	0.0	Ce	S RIDER	LEAD
2	0.00	0.0	Te	S RIDER	LEAD

TRIAL UNBALANCES:

PLANE	MAGNITUDE	PHASE ANGLE (DEG LEAD)	BALANCE LOC	BALANCE (UNITS)	TRIAL WEIGHT AFTER RUN
1	20.00	359.0	COMP END	OZ	WEIGHT LEFT IN
2	10.00	270.0	TURBINE END	OZ	WEIGHT LEFT IN

Table 5.7 Compressor and Turbine Two Plane Balancing Calculation Without Removing Trial Weights - Balance Correction Weights and Predicted Turbine Response

RUN #7 SHOT AT Ce: 20 OZ @ 359 DEG; RUN #8 SHOT AT Te: 10 OZ @ 270 DEG
TRIAL WEIGHTS LEFT IN PLACE AFTER RUN

THE BALANCE CORRECTION WEIGHTS:

PLANE	Ux (OZ)	Uy	MAGNITUDE ABSOLUTE VALUE	ANGLE (DEG) LEAD-CCW	(LAG-CW)	MAGNITUDE TRIM (+TRIAL)	ANGLE (DEG) LEAD	(LAG)
1	26.24	-5.71	26.85	347.7(	12.3)	8.23	319.3(	40.7)
2	-1.00	-20.62	20.65	267.2(	92.8)	10.67	264.6(	95.4)

INITIAL READINGS AND PREDICTED COMP.RESPONSE

SPEED (RPM)	PROBE NO.	PROBE LOC	INITIAL AMP MILS	INITIAL PHASE LEAD	RESIDUAL AMP MILS	RESIDUAL PHASE LEAD	(LAG)
1066	1	Ce	1.70	339	1.093	310	50
1066	2	Te	4.60	54	2.594	358	2
1702	1	Ce	2.80	226	2.875	257	103
1702	2	Te	6.70	10	1.803	41	319
2300	1	Ce	3.90	145	4.850	66	294
2300	2	Te	3.70	333	.734	317	43
2756	1	Ce	4.50	103	1.610	154	206
2756	2	Te	4.70	302	4.331	341	19
3600	1	Ce	5.40	74	.542	343	17
3600	2	Te	6.50	113	2.112	114	246
3600	1	Ce	1.98	98	1.637	207	153
3600	2	Te	5.70	114	2.086	118	242

Table 5.8 Compressor and Turbine Two Plane Balancing Calculation Without Removing Trial Weights - Summary of Influence Coefficients

RUN #7 SHOT AT Ce: 20 OZ @ 359 DEG; RUN #8 SHOT AT Te: 10 OZ @ 270 DEG
TRIAL WEIGHTS LEFT IN PLACE AFTER RUN

THE BALANCE CORRECTION WEIGHTS:

PLANE	Ux (OZ)	Uy	MAGNITUDE ABSOLUTE VALUE	ANGLE (DEG) LEAD-CCW	(LAG-CW)	MAGNITUDE TRIM (+TRIAL)	ANGLE (DEG) LEAD	(LAG)
1	26.24	-5.71	26.85	347.7(	12.3)	8.23	319.3(	40.7)
2	-1.00	-20.62	20.65	267.2(	92.8)	10.67	264.6(	95.4)

SUMMARY TABLE OF THE SYSTEM INFLUENCE COEFFICIENTS

UNITS OF INFLUENCE COEF:MILS/OZ

BAL #	UBAL MAG	SPD #	SPEED (RPM)	PROBE #	AMP1 (MILS)	PHASE1 (LEAD)	AMP2 (MILS)	PHASE2 (LEAD)	INF. COEF (MAG)	(ANGLE) LEAD	(REL LAG)
1	20.00	1	1066	1	1.70	339	2.60	313	.0653	279	80
1	20.00	1	1066	2	4.60	54	5.90	7	.2177	317	42
1	20.00	2	1702	1	2.80	226	3.90	232	.0577	248	111
1	20.00	2	1702	2	6.70	10	4.40	4	.1185	202	157
1	20.00	3	2300	1	3.90	145	4.20	160	.0549	228	131
1	20.00	3	2300	2	3.70	333	2.80	340	.0491	134	225
1	20.00	4	2756	1	4.50	103	3.50	120	.0771	242	117
1	20.00	4	2756	2	4.70	302	5.40	325	.1064	26	333
1	20.00	5	3600	1	5.40	74	4.10	73	.0651	258	101
1	20.00	5	3600	2	6.50	113	3.70	97	.1558	313	46
1	20.00	6	3600	1	1.98	98	1.50	141	.0676	230	129
1	20.00	6	3600	2	5.70	114	3.10	99	.1411	312	47
2	10.00	1	1066	1	1.70	339	1.80	319	.0831	210	149
2	10.00	1	1066	2	4.60	54	4.30	15	.1748	257	102
2	10.00	2	1702	1	2.80	226	3.10	244	.1081	105	254
2	10.00	2	1702	2	6.70	10	3.40	9	.1055	258	101
2	10.00	3	2300	1	3.90	145	3.20	107	.3421	118	241
2	10.00	3	2300	2	3.70	333	1.90	329	.1003	271	88
2	10.00	4	2756	1	4.50	103	2.40	122	.1105	26	333
2	10.00	4	2756	2	4.70	302	4.40	330	.1087	214	145
2	10.00	5	3600	1	5.40	74	2.40	61	.1822	359	0
2	10.00	5	3600	2	6.50	113	3.50	101	.0321	318	41
2	10.00	6	3600	1	1.98	98	1.02	170	.0784	12	347
2	10.00	6	3600	2	5.70	114	3.11	104	.0244	279	80

VI. SUMMARY AND CONCLUSIONS ON PRACTICAL ASPECTS OF BALANCING

1. The difficulty experienced in balancing 70 MW Gas Turbine-Generators has been due to more than conventional unbalance. Additional effects such as shaft bowing due to thermal effects, seal rubs, misalignment, and bearing non-linearity have been experienced.

2. Critical speeds should always be computed to determine the relative rotor mode shapes and the planes at which the rotor will or will not be sensitive to unbalance.

3. The field engineer supplied by the OEM was not furnished with the data on the combined turbine-generator critical speeds. He therefore did not attempt to balance the rotor using modal weight distributions.

4. The OEM field engineer attempted to balance the gas turbine using only one turbine plane at a time with response data taken only at running speed. This procedure is inadequate to field balance the 70 MW gas turbine-generator.

5. The application of a single correction weight at the turbine or compressor end of the gas turbine will excite both the turbine first and second modes of the system.

6. The gas turbine proper has three balance planes which normally may be used for balancing. In addition to the end planes at the compressor and at the gas turbine end, there is center plane where weights may be added. This center plane in the first installation was never used because the access cover was obscured by piping. In the second installation the center balance location was available for balancing. This center plane is most effective for balancing the rotor motion in the speed range of 1,000 to 1,200 RPM. However, this plane should not be used to add corrections to the rotor motion above 2,500 RPM. This is because the center plane location becomes a node point, and has little useful influence on balancing

7. The application of balance weights at both the compressor and turbine end in phase has an influence on the rotor center. This combination of trial weights will cause a large response at the first critical speed in the range of 900 to 1,200 RPM, but only a small response above 2,500 RPM. If this weight combination is used, then speed data only in the 900 to 1,200 RPM should be used. If this weight combination is used to trim balance the turbine at running speed, then a large response at the first critical may be expected. It is believed that when this weight combination was employed by the field service engineer on unit #1, the gas turbine developed a bow in the first mode.

8. The application of balance weights at both ends of the gas turbine, but out-of-phase, is sometimes referred to as a dynamic correction, or shot, while the two weights in-phase are sometimes referred to as a static shot. The application of two weights at the compressor and gas turbine ends out-of-phase would have little to no effect on the rotor first mode around 1,000 RPM. This weight combination was never applied by the field service engineer. The application of a dynamic couple shot at 3,600 RPM would be most effective, whereas the application of a large static shot could cause rotor bowing, which it did.

9. The use of only two balancing planes is inadequate to properly balance the 70 MW turbine-generator. There appear to be six to seven rotor-casing modes in the system. There are two modes each associated with the turbine and the generator. In addition there are several rotor-casing modes in the system. The third free-free gas turbine mode on unit #1 may have been close to running speed. This conclusion will be discussed in greater detail later. As a result of all of these modes, it is physically impossible to balance the turbine using only two planes.

10. The only additional plane available for balancing on unit #1 was the coupling location. Previous to the balancing work of Gunter and Humphris,, the field service engineer did not attempt to utilize this plane for balancing. Analysis of the system critical speeds showed that the coupling location is not very effective for balancing either the turbine first or second critical speeds. However, the generator critical speed appears around 3,200 RPM. This location, hence, is very effective to use as a single plane correction in the speed range of 3,200 to 3,600 RPM.

11. The generator second critical speed is clearly seen by the reversal in the phase of the polar plots obtained at the two ends of the generator. However, it appears to be highly impractical to field balance the generator except at the exciter end. The trim balancing of the coupling has a substantial influence on the attenuation of the generator second critical speed. No attempt should be make to balance the coupling below 3,200 RPM, as the coupling has little influence on the gas turbine first and second critical speeds.

12. The theory of single plane and multi-plane balancing is based upon the assumption that the rotor bearing system response is a linear combination of the unbalances located along the length of the rotor. The change in response due to the application of a trial or calibration weight is referred to as an influence coefficient. These influence coefficients have the units of response (in mils, velocity per unit of unbalance, oz, grams or oz-in., etc.). In a truly linear system, the influence coefficients are independent of the magnitude of unbalance, and are functions only of speed. For the case of the W501 gas turbine, the influence coefficients were found to be highly non-linear and variable. Some of the various reasons for the non-linearity or wide variations observed in the influence coefficients are non-linear fluid film bearings, thermal bow due to nonuniform expansion of gas turbine and generator under load, rotor acceleration, coupling misalignment, rotor bolted up construction, flexible strut mounting or the turbine end tilting pad bearing, friction in the support legs of gas turbine casing, and rubbing at the seal location at the generator exciter end. All of the above factors cause the influence coefficients to deviate from the ideal linear case.

13. The field service engineer attempted to balance the gas turbine by applying single plane linear theory at running speed to the two balance planes independently. This procedure is not accurate or sufficient to balance the gas turbine for several reasons. First, the total number of planes is insufficient (three must be used), and also one must use more than a single measurement. This requires a least squared error computer program in order to perform the necessary balancing calculations.

14. The unit #1 gas turbine was successfully balanced in two days using a least squared error computer program developed for the HP 9845B computer. By means of the least squared error program, many data sets could be entered into the calculation to determine the best fit over the speed range. These calculations were completed on site and within minutes of the run. Therefore a balancing correction was immediately on hand for the next run.

15. The balancing procedure used by the field engineer on unit #1 was a single or two-plane procedure using only one set of speed data. For a more complicated balancing calculation, it was necessary for him to call the main office in Atlanta to relay the data. Therefore the balancing data was not normally available until the next day. For the cases where data was sent to the Atlanta field office to compute the balance shots, it appears that the field engineer did not relay any lower speed data which would have included the effects of the gas turbine first and second critical speeds.

16. As a result of this balancing procedure, it required almost one week for the field service engineer to get the turbine up to running speed. At this time 4 - 10 oz weights were placed at the compressor balancing location. It is believed that this large concentration of balance weights in a single plane may have resulted in shaft bow at the rotor first critical speed.

17. The least squared error program predicted that 3 of the 4 weights should be removed for best balance over the speed range for unit #1. Balancing was improved when this was done.

18. When balancing using only one plane at a time, there is a limit to the amount of vibration reduction that one can achieve. It was observed in several cases, that the optimum least squared error single plane balance correction value could result in the running speed amplitude being slightly higher with the application of the balance correction. This is because the least squared error method also takes into consideration the vibration amplitude and phase at the lower speeds. Any attempt to further reduce the running speed amplitude by the single plane balancing method will result in higher amplitudes at lower speed. Excessive amounts of single plane correction using only running speed data can bow the rotor at lower speeds.

19. When computing the rotor influence coefficients due to an applied weight at a plane, the influence coefficient is computed by a vector subtraction of the two different runs before and after balancing, divided by the value of the balance weight. The influence coefficient therefore is a complex number. The phase angle of the influence coefficient is as important as the magnitude. For a linear repeatable system, both of these values must be functions of speed only, and not the level of unbalance. The exciter influence coefficients of unit #2 were found to be a found to be a highly non-linear function of the unbalance level at the exciter.

20. The rotor amplitude and phase are not the same on acceleration and deceleration. Therefore runup and rundown data should never be mixed in the balancing computations. It appears that this may have been done on several runs by OEM field engineer.

21. The power level of the turbine-generator set at 3,600 RPM causes a distinct and repeatable bowing of the turbine. Therefore, both balancing data at 0 and at 70 MW power level should be included in order to compute the proper balancing magnitude and phase.

22. Examination of the experimental influence coefficients at running speed on both units #1 and #2 showed a large cross coupling in the gas turbine balance end plane. Such an effect is often observed when operating above the second critical speed. For example, the response due to a weight placed at the compressor end is largest at the turbine end and vice versa. Thus the paradox appears that the machine end with high vibration is reduced by placing a weight at the other end of the rotor.

23. The direct response influence coefficients at 3,600 RPM on unit #1 were not repeatable. The gas turbine end presented the greatest deviation. This non-repeatability may have accounted for much of the difficulty experienced with the previous balancing attempts. This non-repeatability may have been caused by thermal bow, looseness of the center span tie bolts, and seal rubs at the exciter end.

24. The gas turbine end on unit #1 showed a phase shift of 120° in 300 RPM. This indicates that the gas turbine is approaching a third bending critical speed. A computer simulation showed that this is possible if there is slack in the center span bolts. Therefore it is very important to check the center span bolts for tightness. Over a period of time, these bolts may yield.

25. A finite element analysis of the gas turbine shows that there is a free-free bending mode around 5,600 RPM. In this mode almost 80% of the shaft bending strain energy is concentrated at the bolted joint. Even a slight yielding of this section will result in a considerable reduction of this mode towards the operating speed. This will cause a very sensitive gas turbine end with respect to balancing. If this mode is close to the operating speed of 3,600 RPM, the rotor will be difficult to balance in two planes.

26. The turbine-generators were instrumented with 5 planes of probes. The data of unit #1 was extensively analyzed by the BENTLY ADRE program for frequency spectrums, polar and Bode plots of rotor amplitude and phase. The data showed many important characteristics of the turbine-generator system for the various unbalance shots. The data evaluated showed probe relative motion, casing motion, and absolute rotor motion obtained by combining the casing and relative shaft motion. It is preferred to base the balancing calculations on the absolute rotor motion. When balancing using noncontracting probes, the low speed motion should be recorded and subtracted from the response vectors. Serious errors will result if the low speed amplitude is not obtained and compensated for in the balancing calculations.

27. Unit #1 was balanced using a combination of shaft rider measurements and probe data. The amplitude and phase were recorded using the TURBOBALANCER built by MECHINTEL, INC. Later tests comparing the TURBOBALANCER with the BENTLY DVF2 showed excellent agreement. The TURBOBALANCER has the feature that a thermal print-out can be obtained on 6 channels of amplitude and phase.

28. The experimental data indicated that rubs may occur at the generator end of unit #1. This would cause a dramatic shift in the influence coefficients at the gas turbine end. Therefore, to improve the performance and the reliability of this system, the generator should be aligned in the seals. On unit #1, the fluid film bearing at the generator end is a partial arc bearing. This bearing should be changed to a tilting-pad.

29. A successful one-shot balancing was accomplished on the compressor end of unit #2 by using multiple vibration data readings from a similar machine. This eliminates the necessity of removing the trial weight or adding a correction to this plane. Operational costs to perform a balance run on the 70 MW units were estimated at $15,000.

30. In the standard balancing procedure, the trial balance weights are removed after each run. This is not a desirable practice when field balancing actual turbine-generators. If the balance weight results in an overall reduction in vibration, then it is the practice to leave the weight in place. It is feasable to have a multi-plane, multi-speed balancing program in which the trial weights are not removed. The multiple data sets from the various runs may be combined to compute a final trim balance using an advanced microcomputer such as the HP 200 Series. It is essential that data at similar speeds and power levels be recorded for the various balance runs in order to compute the overall multiplane trim balance.

31. To successfully multi-plane balance a turbine-generator it is necessary to record simultaneously the RPM, amplitude and phase from the multiple probes. A two channel analyzer is insufficient for this and a data print-out is essential.