FIELD BALANCING 70 MW GAS TURBINE-GENERATORS

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ABSTRACT

This paper treats the practical aspects of multi-plane field balancing 70 MW gas turbine-generators using a micro-computer. The critical speeds and mode shapes of the turbine-generator were calculated to determine the optimum planes for balancing the various turbine-generator critical speeds. Vibration data was obtained on several 70 MW turbine-generator sets at 5 probe locations. The optimum balance was determined by means of a constrained least squared error multi-plane computer program using vibration data at the various speeds and power levels. A one-shot balancing procedure was developed and successully employed. The one-shot balancing procedure is based on previously developed turbine influence coefficients and does not require the placement of trial weights. This paper also shows some of the experimental characteristics of the 70 MW gas turbine, including non-linear jumps due to seal rubs at the exciter end. The study showed that it is not possible to adequately balance the 70 MW gas turbine-generator system using only simple single or two plane balancing theory.

INTRODUCTION AND BACKGROUND

This paper presents the multi-plane balancing procedure and some of the dynamic characteristics experienced with several power utility 70 MW gas turbine-generator sets. The gas turbine weighs approximately 32,700 Kg (*72,000 Lb*), and the generator weighs approximately 23,180 Kg (*51,000 Lb*). There are two tilting pad bearings supporting the gas turbine and three partial bearings are supporting the generator. The generator and gas turbine are connected by a rigid coupling (*1*). This design 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 turbine center plane torque tube has been reported in the literature to have some benefit.

Fig. 1 represents a cross-section of the 70 MW gas turbine.

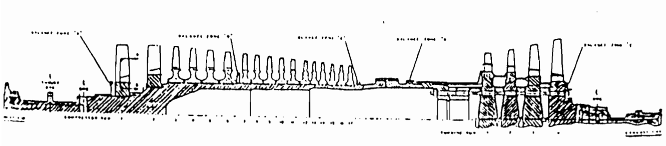

Figure 1 Cross Section of 70 MW Gas Turbine

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.

Fig. 2 represents a typical generator. The generator shows 7 balance planes.

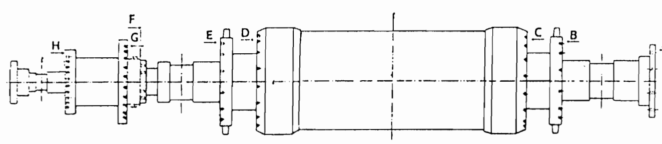

Figure 2 Cross Section of Generator

Balancing corrections at planes D and C require degassing the generator.

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, was not possible and additional balance planes had to be employed. Additionally, 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, 16.8 m (*55 ft.*) turbine-generator set.

In order to make proper balance predictions, both no load and full load data was obtained at operating speed before and after the application of the trial weights. In addition to obtaining data at the operating speed, vibration data should also be obtained at the critical speeds. 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

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 70MW 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 importance to note that small variations of weight at the exciter end caused large non-linear changes in the exciter end influence coefficients. This makes the exciter end particularly difficult to balancing.

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 vibration instrumentation generally in use has either been shaft-rider probes or noncontact proximity probes (*2*). Fig. 3 represents a schematic diagram of the 70 MW turbine-generator with 5 probe locations at each bearing.

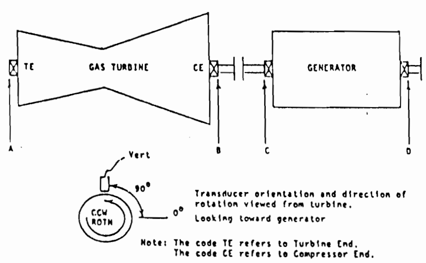

Figure 3 Schematic Diagram of Probe Locations for 70 MW Turbine-Generator

In both case studies , the 70 MW turbine-generators were equipped with the BENTLY NEVADA 7000 SERIES dual probes with one dual probe mounted vertically at 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. The system has the capability of showing relative shaft-to-casing motion, casing motion, or the total amplitude of motion. An additional keyphasor probe located at either the coupling (*case 1*) or the exciter end (*case 2*) provided the phase reference mark. The non-contact proximity probes are preferable to the shaft-rider stick method as the noncontact 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(*3,4*).

A digital tracking filter was used to determine the subsynchronous, synchronous 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 (*5*). 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.

HP 200 series computers were used in the two cases to run the least squared error constrained optimization balancing program (*6,7*), and the BENTLY ADRE© software program (*8*) for the collection and plotting of the experimental data.

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 TURBO-BALANCER developed by MECHINTEL, INC. Amplitude and phase data was simultaneously recorded for the 5 probes at various speeds on spin up and coast down as well as at operation under various power levels. The thermal printout obtained from the TURBOBALANCER was found to be exceptionally useful in recording the vibration data on start up and coast down, allowing rapid field balancing calculations. 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.

CRITICAL SPEEDS OF 70 MW TURBINE-GENERATOR

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.

Fig. 4 represents the cross-section of the 54-station gas turbine rotor model. The bearings supporting the gas turbine are 4 pad tilting-pad bearings with load between pads. Fig. 5 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.

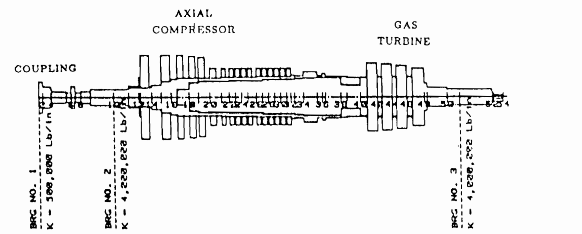

Figure 4 Cross-Section of 54 Station 70 MW Gas Turbine

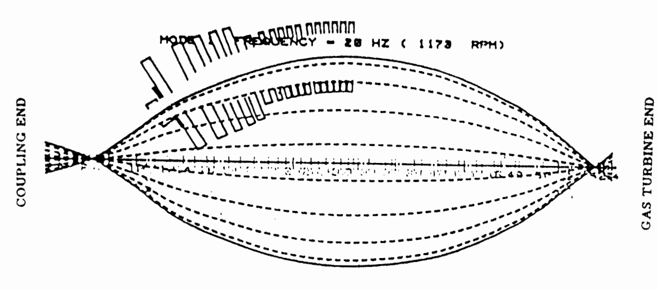

Figure 5 Animated 1*st* Critical Mode of 70 MW Gas Turbine

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. 6 represents the gas turbine animated second mode. With the assumed bearing stiffness values of 7.00E8 N/m (*4.0E6 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.

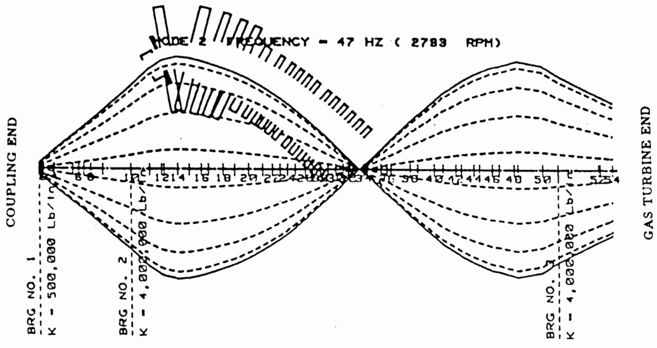

Figure 6 Animated *2nd* Critical Mode of 70 MW Gas Turbine

A suitable balancing correction for 2nd mode vibration is a combination of two weights placed at the compressor and turbine ends 180° out of phase to each other. This combination is referred to as a dynamic balancing shot. With regards to the computer balancing computations, this modal combination may be treated as a single plane correction.

If a balance correction is placed at the center plane, it will have no effect on the response of the rotor at the 2nd critical speed. In this case, the balance correction is said to be orthogonal to the 2nd mode. A single plane balancing correction at either the compressor or turbine end, however, will excite the 1st as well as the 2nd mode. Fig. 6 also shows that balance weights applied to the coupling will have little influence on controlling the 2nd critical speed.

Influence of Bolted Section on Turbine Critical Speeds

After extensive experimental data was obtained on the 1st gas turbine unit, it appeared that the hot end of the turbine was starting to enter a 3rd critical speed (*see Figure 12*). The gas turbine critical speed study showed that the 3rd 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 bolts will have a tendency to creep. This creep effect is enhanced if the rotor has high unbalance causing bowing in the 1st mode. The creeping causes a moment release at the bolted-up section.

The critical speeds of the 70 MW gas turbine with a moment release at the center span bolted connection was analyzed. The original 5th mode is reduced in frequency from above 5,400 RPM to 3,800 RPM. This analysis confirmed that a relaxation of the center span bolts may have a significant influence on the dynamics of the gas turbine.

Critical Speeds of 70 MW Turbine-Generator System

In the original analysis of the gas turbine, in order to simulate the effect of the generator, a simple spring rate of 8.76E7 N/m (*5.0E5 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. This was necessary for the 2nd balancing case where weights were placed at the exciter location.

A critical speed analysis of the gas turbine rigidly coupled to a typical 70 MW generator was run. In the evaluation of the experimental data for the 70 MW turbine-generator for unit No. 2, it appeared that the exciter may be operating through several critical speeds.

Fig. 7 represents the 1st 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 1.76E8 N/m (*1.0E6 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 turbine 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 1st critical speed.

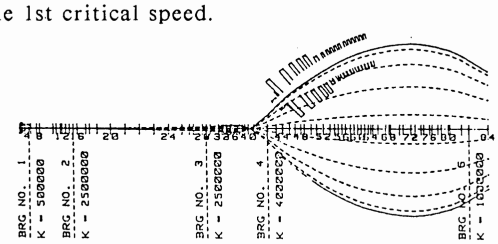

Figure 7 Animated 1st Turbine Mode of 70 MW Turbine-Generator

Fig. 8 represents the 1st 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 1st mode is strongly governed by the inboard and outboard generator bearings.

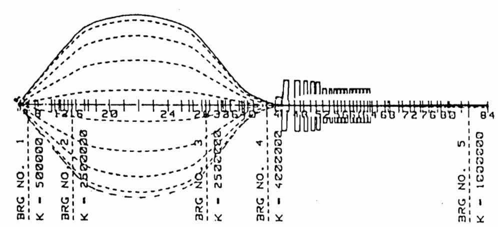

Figure 8 Animated 1*st* Generator Mode of 70 MW Turbine-Generator

Fig. 9 represents the 2nd 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 1st mode, but will not influence the turbine 2nd 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 2nd critical speed. The strain energy distribution for the gas turbine 2nd 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.

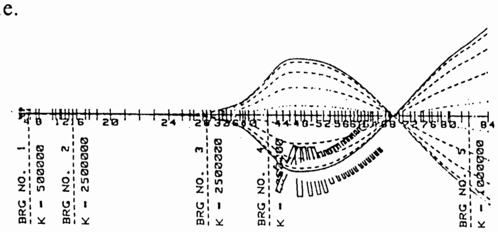

Figure 9 *2nd* Turbine Mode of 70 MW Turbine-Generator

Fig. 10 and Fig. 11 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 system mode (*3rd generator mode*) at 4,500 RPM. If the exciter end bearing stiffness is actually lower than 8.76E7 N/m (*5.0E5 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 3rd generator critical speed or the 5th system critical speed at 4,500 RPM, a reduction in the exciter end bearing stiffness will cause the 3rd generator critical speed to reduce to the operating speed range of 3,600 RPM.

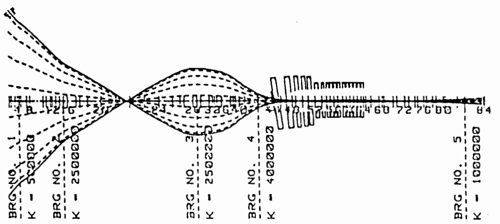

Figure 10 2*nd* Generator mode of 70 MW Turbine-Generator

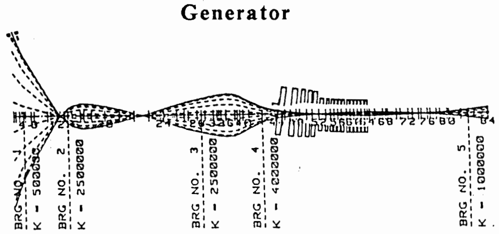

Figure 11 3*rd* Generator Mode of 70 MW Turbine-Generator

Analysis of the experimental data of the exciter end polar plots and also of the high magnitude of the exciter influence coefficients for unit No. 2 indicated that the exciter was operating near a critical speed. It is possible that the exciter is operating at its 3rd critical speed. This made the problem of balancing the unit No. 2 70 MW turbine-generator extremely difficult because of the high sensitivity and nonlinearity of the exciter end.

In unit No. 1, one of the difficulties experienced in balancing the gas turbine end appeared to be due to the operation near a 3rd turbine critical speed. This behavior was not experienced on the 2nd unit.

SAMPLE EXPERIMENTAL DATA ON TYPICAL 70 MW TURBINE-GENERATOR

This section describes typical experimental vibration data obtained from a 70 MW gas turbine-generator. Fig. 12 represents the total vertical motion at the hot end of the gas turbine with 1.10 kg (*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 1st critical speeds are around 1,100 and 1,400 RPM in this system. A 2nd critical speed is observed at 2,000 RPM. This is a rotor casing mode. The 2nd 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 No. 1), it was believed that the turbine is approaching its 3rd critical speed due to the lack tightness at the center span bolted connection. This behavior was not observed in the second case study.

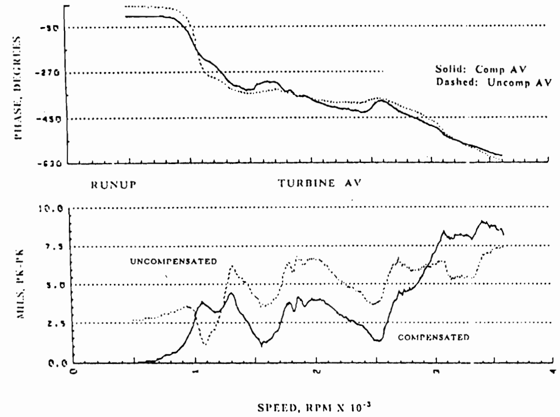

Figure 12 Absolute Synchronous Gas Turbine Motion With and Without Compensation With Large Compressor End Unbalance - U_c = 38.9 oz @ 180°, Z_o = 2.70 mils @ -5°

Fig. 13 represents the polar diagram of the hot end of the gas turbine. One can estimate the location of the various critical speeds from the observation of the peak amplitudes on the polar diagram and from the rate of change of phase angle. 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 3rd critical speed.

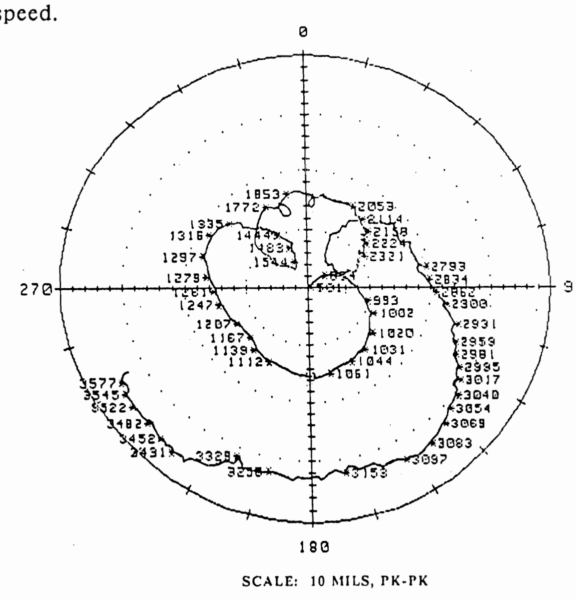

Figure 13 Polar Plot of Compensated Absolute Gas Turbine Motion With Large Compressor End Unbalance - U_c = 38.9 oz @ 181°, Z_o = 2.70 mils @ -5°

Fig. 14 represents the relative motion vs speed at the hot end of the gas turbine with the run-out vector subtracted from the vibration data. The 1st turbine critical speed is clearly indicated at 1,100 RPM, and the 2nd 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. 15 represents the polar diagram for the relative rotor casing amplitude at the gas turbine location (*corresponding to Fig. 14*). The difference in the polar plots between Figures 13 and 15 represents the casing vibration. 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 3rd 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.

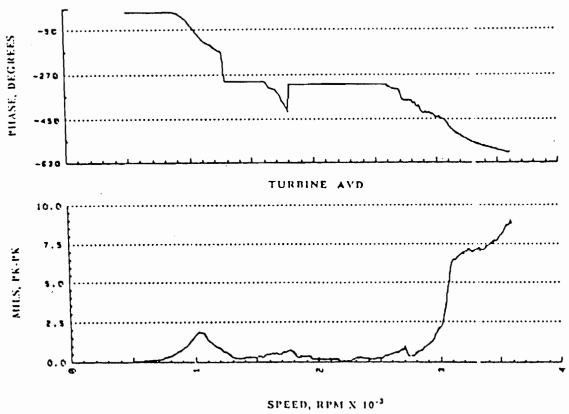

Figure 14 Relative Compensated Synchronous Shaft Motion of Gas Turbine With Large Compressor End Unbalance - U_c = 38.9 oz @ 181°, Z_o = 2.40 mils @ -12°

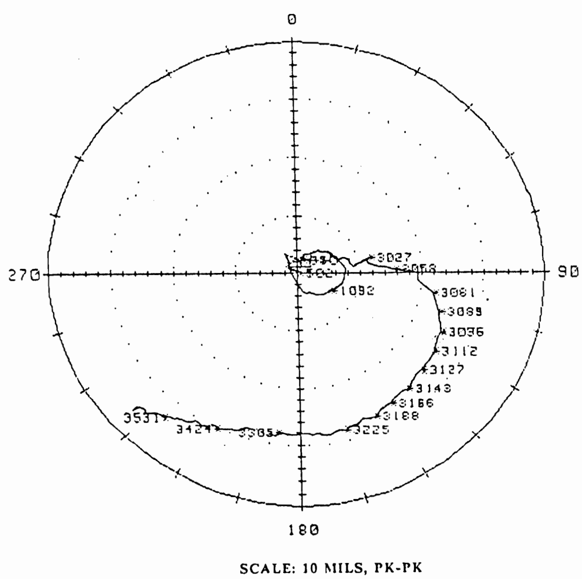

Figure 15 Polar Plot of Relative Compensated Synchronous Shaft Motion of Gas Turbine With Large Compressor End Unbalance - U_c = 38.9 oz @ 181°, Z_o = 2.40 mils @ -12°

Fig. 16 represents the absolute motion at the compressor end of the gas turbine with 1.10 kg (*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 2nd critical speed at 2,700 RPM.

Fig. 17 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 50 μm (*2 mils*). This is an indication that the rotor may have been bent during previous balancing attempts by the

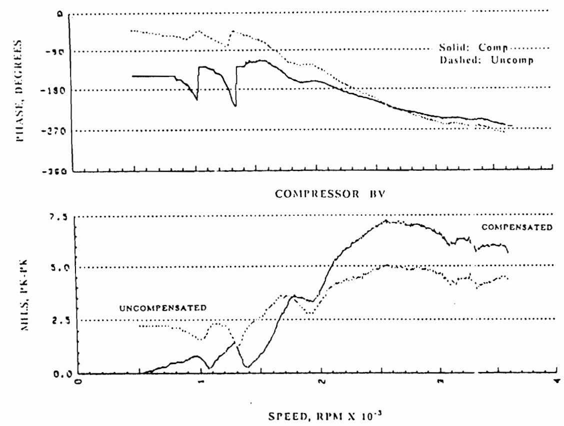

Figure 16 Absolute Compressor Synchronous Compensated and Uncompensated Motion With Large Compressor End Unbalance - U_c = 38.9 oz @ 181°, Z_o = 2.2 mils @ -41°

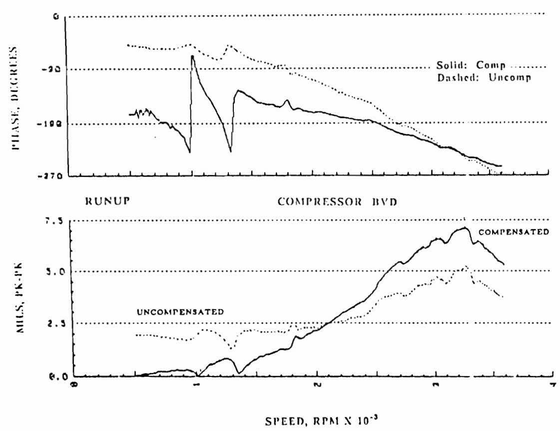

Figure 17 Relative Compensated and Uncompensated Compressor Motion With Large Compressor End Unbalance - U_c = 38.9 oz @ 181°, Z_o = 1.96 mils @ -50°

manufacture's 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 RPM is the turbine 1st critical speed, and the phase change at 1,350 RPM is the generator 1st critical speed. The gas turbine 2nd 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 2nd critical speed, whereas the hot end of the gas turbine continues to increase. This characteristic of the 70 MW gas turbine considerably complicates the balancing procedure.

Generator Vibration

Fig. 18 represents the polar absolute motion of the inboard generator bearing. The large unbalance at the compressor end of the gas turbine causes little excitation of the generator 1st 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 2nd critical speed at 3,200 RPM. Above 3,200 RPM, the amplitude reduces as would be expected when passing through the 2nd critical speed. The generator inboard bearing motion is well behaved.

Fig. 19 represents the generator outboard bearing polar diagram. A comparison of the polar diagrams in

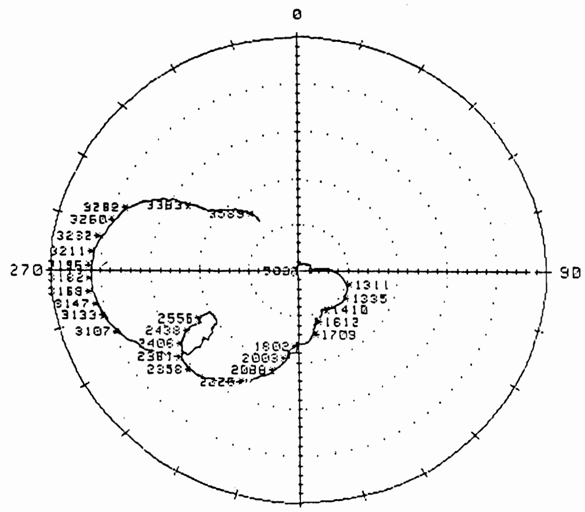

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Figure 18 Inboard Generator Polar Plot With Large Compressor End Unbalance - U_c = 38.9 oz @ 181°, Z_o = .12 mils @ -178°

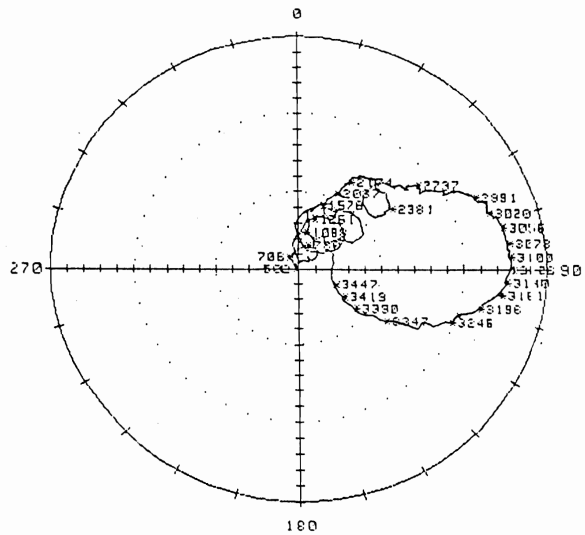

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Figure 19 Generator Outboard Polar Motion With Large Compressor End Unbalance - U_c = 38.9 oz @ 181°, Z_o = 0.09 mils @ -203°

Figures 18 and 19 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 2nd critical speed.

__Exciter - Generator Non-linear Response Due to Seal Rub__

Fig. 20 represents a polar plot of the motion at the outboard generator bearing casing for one of the unbalance runs on the 70 MW gas turbine. In this particular run, a non-linear jump was observed at the outboard generator bearing and the exciter end locations at 3,200 RPM. After the jump occurs, the phase angle change is erratic and in 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 in amplitude and phase. The turbine-generator can not be accurately balanced if a seal rub occurs. Seal alignment must be performed before balancing can be undertaken. Generators with a tilting pad bearing at the exciter location, instead of a partial arc bearing, did not exhibit the nonlinear jump phenomena.

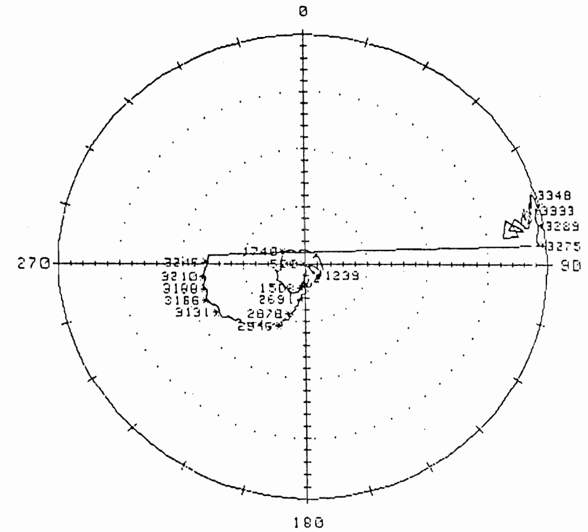

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Figure 20 Polar Plot of Generator Outboard Casing Motion With Turbine and Compressor Unbalance Showing Large Nonlinear Jump at 3,300 RPM - U_c = 10 oz @ 181°, U_t = 10 oz @ 270°

__Non-Linear Response of Exciter End__

In the balancing runs performed on unit No. 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. 21 represents the polar diagram obtained on the exciter end with unbalance of 93 g (*3.3 oz*) at 65°. Fig. 21 was generated directly from the co and quad components of the vibration data using an X-Y analog plotter.

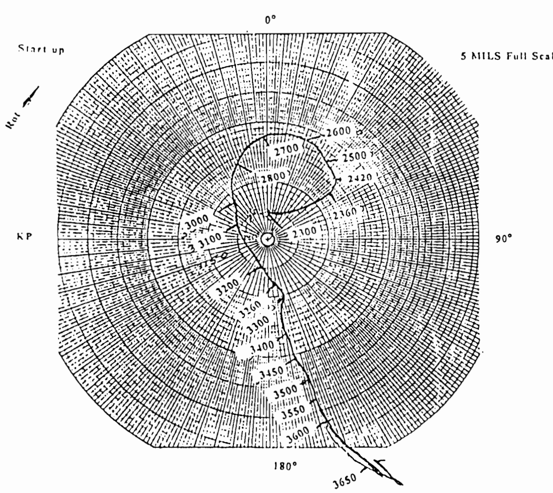

Figure 21 Exciter Bearing Motion With 93 Gm @ 65 Deg Showing Nonlinear Response Above 3,300 RPM

The turbine 2nd critical speed at 2,700 RPM is clearly evident. The generator 2nd critical speed at 3,300 RPM is not apparent. Instead of the exciter amplitude vector "*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 3rd exciter critical speed above 3,600 RPM. The amplitude of motion increased until the 180 μm (*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 No. 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 would alleviate some of the non-linear characteristics encountered in balancing.

FIELD BALANCING 70 MW TURBINE-GENERATORS

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 initially 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 and speed or power condition. For multi-plane balancing, the balance procedure developed does not require the removal of the initial trial weights.

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 1st critical speed around 1,100 RPM, but has a substantial influence at 3,200 RPM. In the balancing procedure used by the authors, various speed ranges were incorporated, as well as various probe readings. 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 must be employed. The previous balancing procedure used on unit No. 1 was simple single-plane balancing at the running speed.

Compressor End Balancing Calculations

Figure 12 represents the gas turbine hot end motion with approximately 1.1 kg (*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 1st balancing run performed by the authors on unit No. 1, two of the balance weights were removed at the compressor location.

In the simple single-plane balancing procedure, the amplitude after balancing at the measurement location is theoretically 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 off line in attempting to come up to speed. The simple single measurement, single plane influence coefficient method of balancing at speed, is highly undesirable and has caused large bows in the turbine. A single plane compressor end correction, based on one probe reading at the hot end at running speed, would result in a correction of 1.0 kg (*35.5 oz*) @ 347°, or a trim of 480 g @ 330°.

In order to perform a correct single-plane balancing at the compressor end which will not upset 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. A single plane compressor end correction was computed, based upon 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 TURBO-BALANCER, 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 No. 1, it was found that the change in power level causes a response change at the compressor end of approximately 74 μm (*2.9 mils*) at 221°, and on the gas turbine end a response change of 10.7 μm (*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 no load 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 70 MW base data, however, was not obtained on unit No. 1. The estimated full power response was computed from previous runs.

The computed compressor end balance correction based on 6 data sets is 640 g (*22.6 oz*) @ 347°. If the original weight is left in place, then a trim weight of 140 g (*5 oz*) at approximately 294° is called for instead of 480 g (*17 oz*) at 330°.

Single-Plane Balancing - 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.

A 284 g (*10 oz*) trial weight was applied at 270° on the hot end of the gas turbine. Using data for four speeds and two probes, a final turbine balance of 697 g (*24.6 oz*) @ 288° was predicted. It is of interest to note that at running speed the compressor end influence coefficient was 0.188 μm/g (*0.21 mils/oz*), while the influence coefficient at the hot end was only 0.063 μm/g (*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.

Multi-Plane Balancing Without Removing Trial Weights

The previous data from the compressor end run and the turbine run with the trial weights remaining in place can be calculated to compute a two plane balancing correction for the gas turbine. The two trial unbalance values were 567 g (*20 oz*) at the compressor end, and 284 g (*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 1 gives the two plane total and trim balance corrections to add to the compressor and turbine end, and also gives a summary of the influence coefficients. Using the two plane data, a total compressor end correction of 761 g (*26.9 oz*) @ 348° is computed as compared to the lower value of 640 g based on single plane balancing. A final turbine correction of 585 g (*20.6 oz*) @ 267° is predicted as compared to 697 g based on single plane turbine balancing. Thus the single plane corrections vary

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

Table 1 Gas Turbine 2 Plane Balance Corrections and Influence Coefficients

approximately 15% from the combined two plane calculations. Table 1 also shows the trim weights of 233 g (*8.23 oz*) @ 319° for the compressor end and 300 g (*10.6 oz*) @ 265° to be added to the turbine end.

Table 1 also contains 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 0.3 μm/g (*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 2 represents the initial and predicted turbine response at the various speeds with the two plane balance correction. Note the high turbine vibration of 170 μm (*6.7 mils*) at 1700 RPM. If a single plane balancing correction is made based only on running speed data, then the turbine vibration at 1700 RPM may exceed 180 μm. This would cause the turbine to trip off line.

INITIAL READINGS AND PREDICTED COMP.RESPONSE

SPEED (RPM)	PROBE NO.	PROBE LOC	INITIAL AMP MILS	INITIAL PHASE LEAD	CURRENT AMP MILS	RESIDUAL AMP MILS	RESIDUAL PHASE LEAD	(LAG)	NET CHANGE MILS
1066	1	Compressor	1.70	339	1.80	1.093	310	50	-.7
1066	2	Turbine	4.60	54	4.30	2.594	358	2	-1.7
1702	1	Compressor	2.80	226	3.10	2.875	257	103	-.2
1702	2	Turbine	6.70	10	3.40	1.803	41	319	-1.6
2300	1	Compressor	3.90	145	3.20	4.850	66	294	+1.6
2300	2	Turbine	3.70	333	1.90	.734	317	43	-1.2
2756	1	Compressor	4.50	103	2.40	1.610	154	206	-.8
2756	2	Turbine	4.70	302	4.40	4.331	341	19	-.1
3600	1	Compressor	5.40	74	2.40	.542	343	17	-1.9
3600	2	Turbine	6.50	113	3.50	2.112	114	246	-1.4
3600	1	Compressor	1.98	98	1.02	1.637	207	153	+.6
3600	2	Turbine	5.70	114	3.11	2.086	118	242	-1.0

Table 2 Gas Turbine Initial and Predicted Compressor and Turbine and Response With Optimum 2 Plane Balancing

Rotor Three Dimensional Mode Shapes

The vibration from five probes has been recorded at the various turbine and generator locations. A computer program was developed using a cubic spline curve fit to generate a 3 dimensional mode shape of the shaft. The 3 dimensional cubic spline curve fit generates a continuous space curve through the various points.

After the balancing shot was applied at the coupling location, the vibration data before and after the coupling correction was vectorially subtracted and divided by the balance trial weight. Fig 22 represents the 3 dimensional mode shape of the shaft response at 3,600 RPM due to a weight applied at the coupling location. The scale of the influence coefficient is 203 μm/kg (*3.63 mils/lb*) of unbalance. The coupling location is between bearings 2 and 3. The application of the weight at the coupling has very little influence at the gas turbine location which is shown on the figure as bearing No. 1.

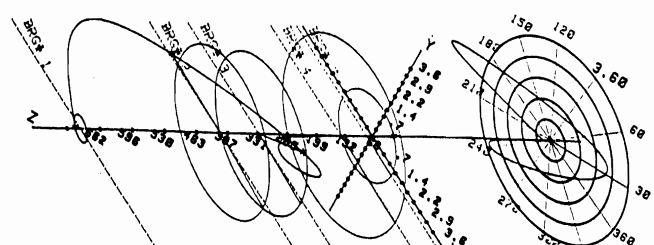

Figure 22 Single Plane Correction on Coupling, Balance Shot: 251 g @ -236° Lead

Fig 23 represents the influence coefficients of the turbine-generator with a balance shot at the turbine cold end (compressor). It is of interest to note that the amplitude at the hot end of the turbine is over 7 times the response at the compressor end where the weight was applied. From examining the various influence coefficient mode shapes, it is apparent that the rotor response is never planar, but is a 3 dimensional space curve.

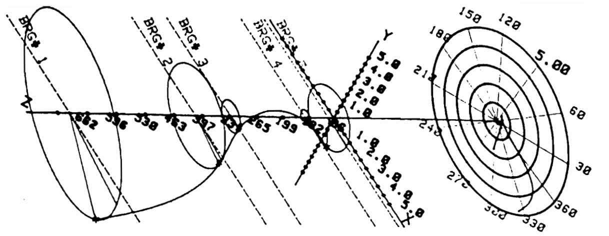

Figure 23 Influence Coefficient Due to Balance Shot on Turbine Cold End, Balance Shot: 827 g @ -72° Lead

SUMMARY AND CONCLUSIONS ON PRACTICAL ASPECTS OF BALANCING

1. Previous balancing procedures attempted to balance the gas turbine by applying single plane linear theory at running speed to the two gas turbine 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 speed measurement. This requires a least-squared error computer program in order to perform the necessary balancing calculations.

2. 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.

3. 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.

4. 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 a center plane where weights may be added. The center balancing plane in the first installation was never used because the access cover was obscured by piping. In the second installation the center 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 which corresponds to the turbine 1st critical speed. 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.

5. The application of in-phase balance weights at both the compressor and turbine ends is referred to as a static correction. This combination of trial weights will cause a large response at the 1st 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 range should be used.

6. The generator 2nd 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 and coupling ends. The trim balancing of the coupling has a substantial influence on the attenuation of the generator 2nd 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 1st and 2nd critical speeds.

7. 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.

8. 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.

9. The power level of the turbine-generator set at 3,600 RPM causes a distinct and repeatable bowing of the turbine. Therefore, data at 0 and at 70 MW power levels should be included in the balancing calculations.

10. Examination of the experimental influence coefficients at running speed on both units No. 1 and No. 2 showed a large cross coupling in the gas turbine balance end planes. To reduce the vibration at one end, the balance weight must be placed at the opposite end of the gas turbine.

11. The direct response influence coefficients at 3,600 RPM on unit No. 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.

12. A successful one-shot balancing was accomplished on the compressor end of unit No. 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.

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