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Unbalance Response and Field Balancing of an 1150-MW Turbine-Generator with Generator Bow

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ABSTRACT

This paper presents some of the theoretical and experimental unbalance response characteristics of an 1150-MW turbinegenerator. These large 11-bearing turbine-generators can operate through as many as 10 distinct critical speeds. The balance condition of these units may change over time due to various influences such as turbine blade erosion, alignment and, in particular, generator bowing. Over a period of time, the generator may developed a bowed condition due to the nature of the electrical conductors. It is difficult to directly field balance a generator which is encased in a hydrogen environment. Removal of the generator for high-speed balancing in a balance facility is exceptionally expensive. This study examines the control of the turbine-generator response due to generator bow and turbine unbalance by various combinations of balance weights placed along the turbines. Some experimental responses of the turbine-generator due to various trial weights along the system are also presented. The 11-bearing turbine-generator, including foundation effects, was analytically modeled using transfer matrix and finite element methods. It was determined that in order to correctly model a large turbinegenerator, the foundation effects must be included. Foundation flexibility may cause as much as an 80-90% reduction in effective bearing damping. Various unbalance response simulations were performed with this model with combinations of turbine unbalance and generator bow. It is shown that system response due to turbine unbalance and generator bow at running speed may be controlled by proper application of unbalance weights and couples placed along the couplings and the low pressure turbines.

KEY WORDS

Turbine-generator response, Multi-plane Balancing, Three-dimensional complex modes

1 INTRODUCTION

The 11-bearing 1150 turbine-generator systems represent some of the largest units in operation. One of the maintenance problems of concern is that of balancing these units in the field. Over a period of time, the generator may develop a bowed condition due to the nature of the electrical conductors. It is difficult to field balance a generator which is enclosed in a hydrogen environment.

The site facility has two identical units. Vibration measurements were recorded at the bearings of both units. Trial balance weights were placed along both rotors to obtain influence coefficients. Although the units are identical, the experimentally obtained influence coefficients were not. The bearings have jacking oil for liftoff and shutdown. After full speed is achieved, the jacking oil is turned off and the bearings become fully hydrodynamic. The bearing hydrodynamic pressure may be measured. Examination of the operating pressures developed in each bearing showed that several of the interior bearings could be underloaded due to pedestal sag. The vertical pedestal alignment thus strongly influences the system dynamics and hence the balancing. The vertical alignment influences the bearing loading and hence the bearing stiffness and damping coefficients. The bearing pedestal flexibility reduces the overall stiffness and may also cause a dramatic reduction in effective damping by as much as 90%.

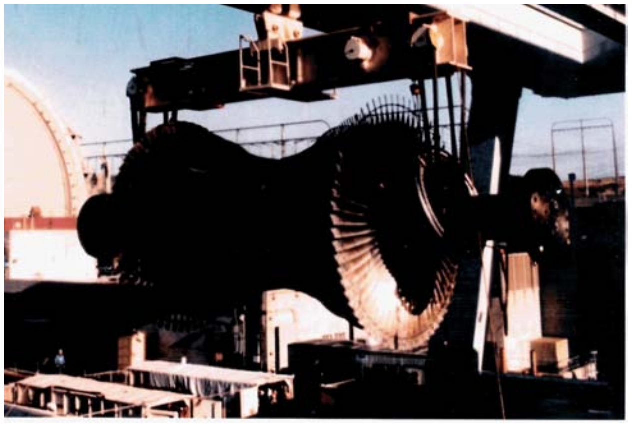

Figure 1a: LP Turbine in Crane

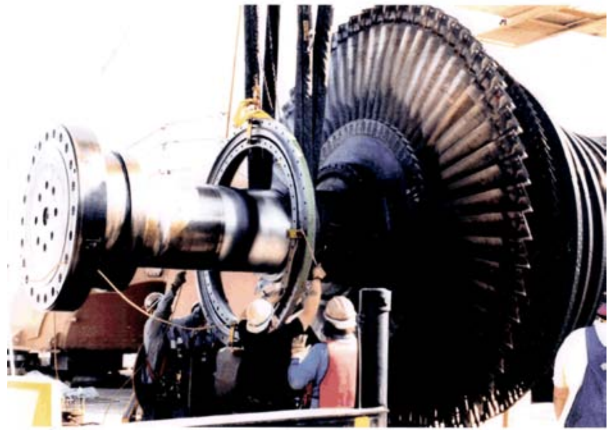

Figure 1b: Closeup of HP Balance Turbine Ring

Figure 1 represents images of the LP turbine in the crane and a close up of the HP rotor. In Figure 1a, note the tiny figure of a man in the lower left corner. These turbines represent some of the largest utility rotors in service. In Figure 1b, the technicians are in the process of installing a balancing ring to the turbine assembly. Also, in this figure, note the coupling at the left side of the rotor. When these couplings are bolted up, they form a rigid connection. Field balancing corrections may also be applied at each coupling location.

2 TURBINE-GENERATOR MODELING

2.1 Transfer Matrix Method

The first method of modeling the complete turbine-generator system was by means of the transfer matrix approach. A modified transfer matrix method was employed with scaling of the matrices to improve computational accuracy. Figures 2a&b represent models of a HP turbine and generator using a transfer matrix program.

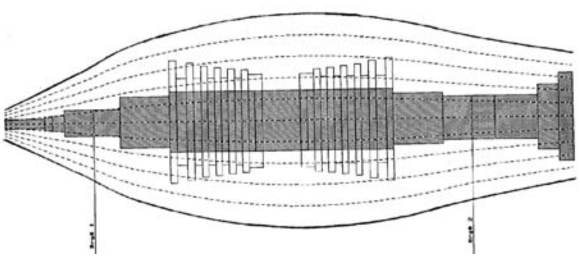

Figure 2a - Model of HP Turbine Showing 1st Mode

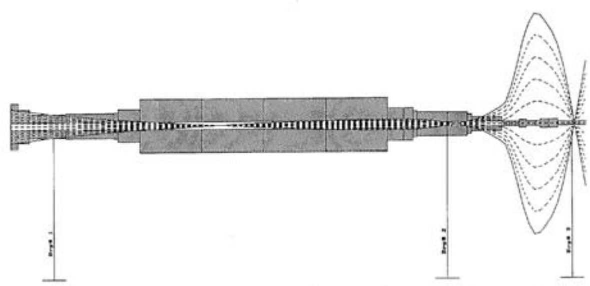

Figure 2b - Generator Showing Exciter Mode

Figure 2a shows the HP turbine, which is the first turbine of the drive system. The HP turbine 1st critical is relatively unaffected by the coupling to the first LP turbine. The individual LP turbine critical speeds are meaningless as they are now part of a system and cannot be computed as individual components.

Figure 2b represents the generator with the attached exciter. In general, the generator is the most flexible component of the system and hence has the lowest critical speed. The attachment of the exciter and the LP turbine have little effect on the generator first critical speed. The mode shown in Figure 2b is a local exciter mode and is unaffected by the LP turbines. The attachment of the LP3 coupling has no influence on the local exciter mode.

The attachment of a very flexible exciter to a massive generator can lead to a very serious situation. It has been demonstrated through numerical time transient analysis that a blade loss on LP3 can cause exciter damage leading to catastrophic system failure. The presence of the very flexible exciter attached to the generator also leads to numerical problems in convergence when computed by the transfer matrix method.

When the entire system of the four turbines and generator was assembled to form the complete model, there were encountered a number of computational difficulties for the determination of all the system modes. The first problem is that of missed or skipped modes. This is caused by the size of the search increment used in the analysis. The second difficulty was that of convergence of the higher frequencies due to the extreme flexibility of the exciter.

The most serious difficulty with the transfer matrix program is the inability to properly handle foundation or pedestal effects with included mass and stiffness. This is due to the generation of local or branched resonances which must be removed from the search range. It was later determined that the pedestal effects are of major importance in the determination of the effective stiffness and damping at a particular bearing location. The transfer matrix method also is not suited for time transient blade loss analysis or the computation of damped eigenvalues including foundation effects. Hence, the transfer matrix method is unsuited for the analysis of multi-bearing turbine-generator systems.

2.2 Finite Element Analysis

A PC-based finite element program was utilized for the computations of the static deflections and undamped system modes.

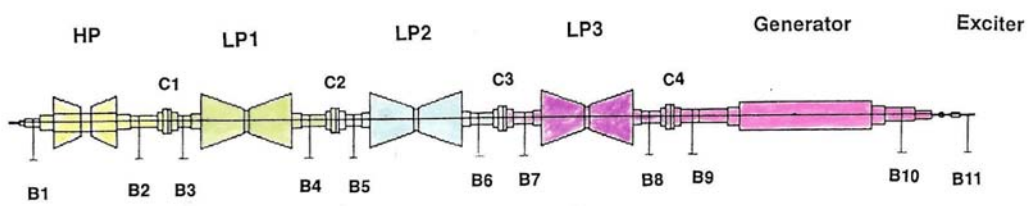

Figure 3: Finite Element Beam Model of 1150-MW Turbine-Generator

A finite element beam model of the turbine-generator system was developed using a standard structural FEM program. In this model, several hundred beam elements were employed. The bearings are represented by simple springs and no disc gyroscopic effects are included in the model. This model was useful in the determination of the static shaft displacements and loads with various assumed alignments and also for the computation of the system modes. Modes are not skipped when computed by a finite element method. However, all the commercial finite element structural codes are unsuited for the

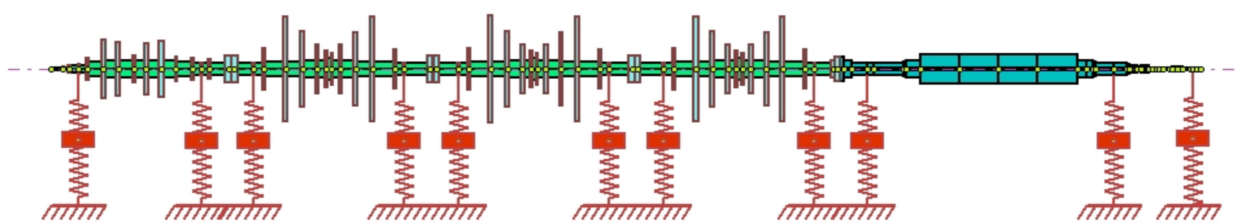

Figure 4: Finite Element Model of 1150- MW Turbine-Generator

complete dynamical analysis of turbine-generators. These programs cannot incorporate generalized bearing stiffness and damping coefficients, disk gyroscopics, or compute complex eigenvalues, synchronous unbalance response with shaft bow or blade loss simulation.

The system was next modeled with a finite element rotor dynamics program that could also include the bearing, as well as foundation effects. A model of the system is shown in Figure 4. Each bearing is represented by 8 bearing stiffness and damping coefficients. These coefficients are determine from a fluid film finite element bearing code once the values of bearing loading are estimated. Of greater difficulty is the determination of the pedestal properties of effective mass and stiffness. Impact testing may be used to determine properties.

Figure 4 represents the turbine-generator model with 110 nodes. Since each node can have two displacements and two rotations, the total model is represented by a system of 440 dof. Some rotor dynamic codes use a polynomial representation for the system and attempt to extract the complex roots from the characteristic polynomial. This model far exceeds the capabilities of such a technique. Transfer matrix methods also are not capable of analyzing such a system as shown in Figure 4. It should be important to note that the turbine-generator may be very accurately modeled with the finite element method. In Figure 4, note that the bearings are connected to pedestal masses. The pedestals have a property of mass, stiffness, and damping. These quantities of the foundation or pedestal effects are difficult to identify initially.

In addition to the pedestal characteristics, the vertical alignment of each pedestal determines the amount of loading on each bearing. The bearing loading in turn, determines the bearing characteristics. Thus, identical bearings do not have identical bearing coefficients. Also, identical units do not have identical vibration responses or influence coefficients. The experiential data obtained from trial unbalance runs can be used to further refine the rotor dynamics model.

3 BEARING LOADING AND FLUID FILM BEARING PROPERTIES

Figure 5 represents the pressure profile and bearing operating characteristics at 1,800 RPM. For the average bearing loading as applied in Figure 5, the bearing is operating at a very high eccentricity ratio of 0.8 at running speed. Because of these high operating eccentricity ratios, jacking oil is required on liftoff. At a suitable speed, the jacking oil is turned off. At the oil feed location at the bottom of the bearing, pressure gauges monitor the oil pressure. At running speed, this oil film pressure is generated by the hydrodynamic bearing action. It was observed that not all of the turbine bearings are generating equal pressures. Table 1 shows the bearing pressures generated in the two units.

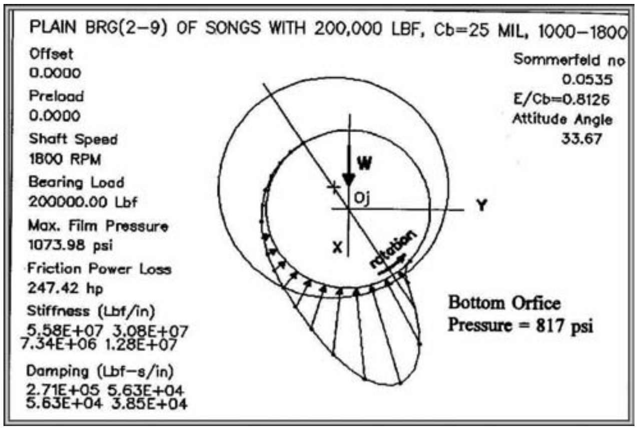

Figure 5: Turbine Bearing Characteristics, 1800 RPM

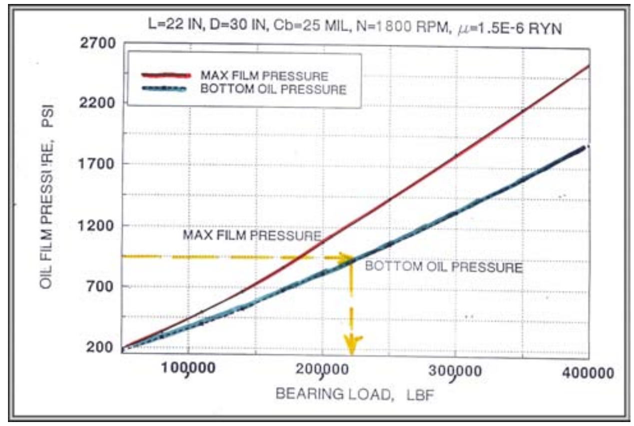

Figure 6: Relationship Between Measured and Maximum Oil Film Pressure

Figure 6 shows the relationship between bottom oil pressure, maximum pressure and bearing loading. This relationship is relatively insensitive to assumed viscosity. By computing the load-pressure characteristics for each bearing, the actual applied loading at each bearing may be estimated. Table 1 shows that the LP3 bearings of unit 2 are underloaded.

Table 1: Bearing Static Loading Versus Orifice Film Pressure Measured
N = 1,800 RPM, = 10 centiPoise

Bearing Locations	Brg No.	Unit 2 kPa	Cb mm	Bearing Load, N	Unit 3 kPa	Cb mm	Brg Load, N
HP Outbrd	1	3,448	.48	.280E6	1,725	.48	.152E6
HP Inboard	2	2,760	.64	.367E6	2,070	.64	.286E6
LP1 Outbrd	3	4,830	.64	.777E6	650	.64	.735E6
LP1 Inboard	4	3,800	.64	.65E6	3,450	.56	.608E6
LP2 Outbrd	5	3,450	.64	.604E6	3,450	.56	.608E6
LP2 Inboard	6	3,450	.64	.604E6	3,100	.56	.551E6
LP3 Outbrd	7	1,380	.64	*.26E6*	1,030	.56	.213E6
LP3 Inboard	8	690	.51	*.15E6*	2,760	.64	.495E6
Gen Inboard	9	2,760	.64	.546E6	3,100	.51	.635E6
Gen Outbrd	10	3,100	.51	.632E6	1,380	.51	.307E6
Exciter	11	1,380	.165	.032E6	1,725	.165	.036E6
			TOTAL	4.90E6		TOTAL	4.63E6

The LP3 bearing no 8 of unit 2 has a pressure of only 690 kPa as compared to the overloaded LP1 outboard bearing no 3 of the same unit of 4,830 kPa. Also note that the LP1 outboard bearing of unit 3 is underloaded with a pressure of only 650 kPa. These differences in measured hydrodynamic bearing pressures are caused by the differences in vertical alignment between the two machines. Hence it may be expected that the experimental influence coefficients between the two units will have difference values of amplitude and phase response to the trial weights.

4 Undamped Natural Frequencies of Turbine-Generator System

The turbine-generator system may have as many as 10 distinct undamped modes in the operating speed range. However, when fluid film bearings and foundation effects are considered, then up to 30 complex eigenvalues must be considered due to bearing asymmetry and the presence of forward and backward modes.

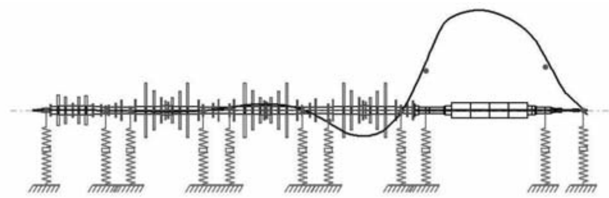

Figure 7: 1st T-G Mode at 637 RPM - Generator

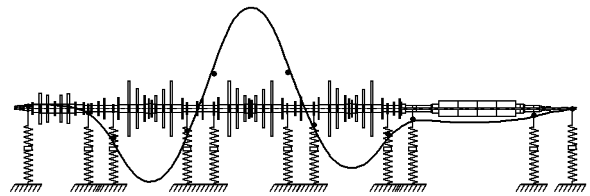

Figure 8: 2^{nd} T-G Mode at 706 RPM

Figure 7 represents the first system mode. Since the generator is the longest unit of the system, the first critical speed is the generator 1^{st} mode. This mode is relatively unaffected by the LP turbines. The mode is primary influenced by the generator design and bearing coefficients. Figure 8 represents the 2^{nd} system mode at 706 RPM. In this mode, the mass centers of the low pressure turbines LP1 and LP3 are out of phase to the second low pressure turbine, LP2. The generator motion in the mode is very quiet.

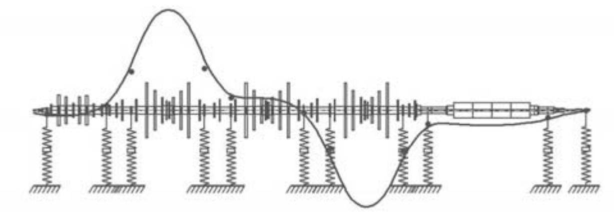

Figure 9: 3^{rd} T-G Mode at 734 RPM

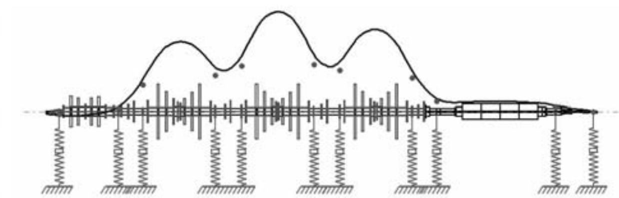

Figure 10: 4^{th} Mode at 757 RPM

In Figure 9 for mode 3, the mass centers of LP1 and LP3 are moving out of phase and the second low pressure turbine is quiet. As the speed is increased a small speed increment of 23 RPM, the mode changes in which the mass centers of the three low pressure turbines are moving in phase. These three modes of the LP rotors take place in only a 50 RPM speed range. When the LP turbines are well balanced before assembly to correct mass center unbalance, then these modes are not usually observed during startup.

It is of considerable interest to note that these closely spaced critical speed mode shapes are not normally detected using a transfer matrix method of calculation. A typical search speed increment of 50 RPM will cause these modes to be completely missed. This situation of missing modes is not encountered when using a finite element. This is only one of many problems encountered with using a transfer matrix method for computation of the dynamics of large turbine-generators.

In general, transfer matrix methods are unsuited for the computation of the dynamical characteristics of large turbine-generators with pedestal effects. Problems with convergence are also encountered. The first four system modes indicate the desirability of initially balancing the generator and turbine in low speed balancing equipment to correct initial mass center unbalance.

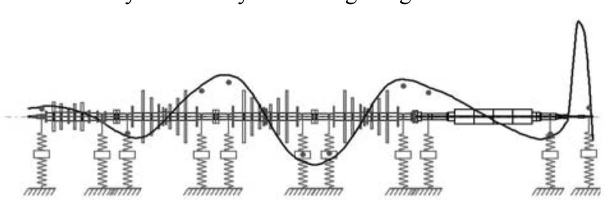

Figure 11: 7^{th} T-G Mode at 1450 RPM

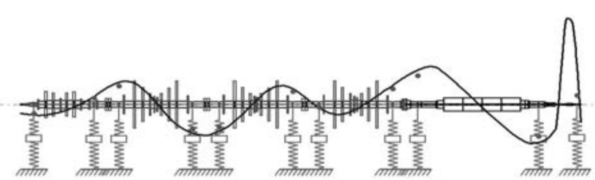

Figure 12: 8^{th} T-G Mode at 1534 RPM

Figure 11 shows the 7th mode of the system at 1450 RPM. In this mode there is high exciter motion The generator has a conical motion in which the ends are out of phase. As the higher system critical speeds are approached, the mass centers of the turbines approach node points with little motion. It is of interest to note that the 7th mode shows 7 crossover points.

Figure 12 represents the system 8th mode at 1534 RPM. This mode is similar to the 7th mode shown in Figure 12 except that there are now 8 crossover or node points along the turbine-generator centerline. The fact that the mass centers of the generator and turbines all indicate node points at the rotor center of gravities has an important conclusion for high-speed field balancing. It would be of little use to apply balance corrections to the rotor mid spans even if these locations could be reached. Midspan corrections are of influence at the lower modes.

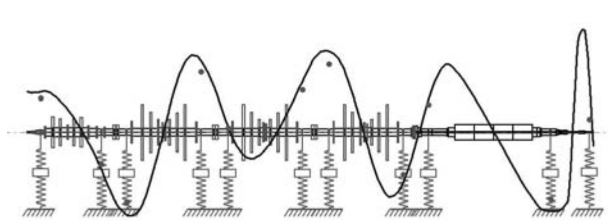

Figure 13: 9th T-G Mode at 1662 RPM

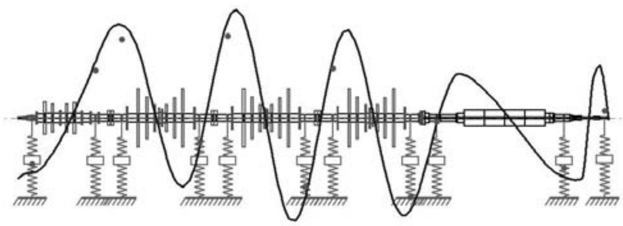

Figure 14: 10th T-G Mode at 1849 RPM

Figure 13 represents the 9th mode at 1662 RPM. Note the high exciter motion that is also seen in this mode. The number of nodal points is 9 which is the order of the critical speed. As each higher critical speed is reached, the nodes or crossover locations of zero amplitude also increase. Figure 14 represents the 10th mode at 1849 RPM. This mode has 10 crossover points. Again it is observed that the turbine mass centers become modal nodal points.

The next higher mode is above 2,400 RPM. This class of turbine -generator was originally designed for 50 Hz operation. The required frequency of 60 Hz for operation in North America places the operating speed at 1,800 RPM. This introduces two additional critical speeds of interest into the system. The observation of the high exciter motion represents a serious situation. A failure of a last stage turbine blade on LP3 has been shown to cause an excitation of the 9th mode leading to exciter failure. This is conjectured to be the cause of failure of the Detroit System on Christmas day of over 10 years ago. The system model here is of similar design to the Detroit unit.

5 Turbine-Generator Complex Influence Coefficients at 1,800 RPM

5.1 Experimental Influence Coefficients For Various Balance Planes

In order to perform field balancing, it is necessary to establish the system influence coefficients under applied unbalances at various locations along the shaft. The measured responses along the shaft divided by the trial weight is referred to as the influence coefficient response. It should be noted that for the case of large turbine-generators, the response is load dependant as well as speed dependant.

The response of the system was recorded for trial weights placed along couplings 2, 3, and 4, and trial weights placed at either end of the third low pressure turbine, LP3. Amplitude and phase measurements were taken at all 11 bearings. An additional measurement between the 10th and 11th bearings on the exciter shaft would have been desirable due to the extreme flexibility of the exciter shaft.

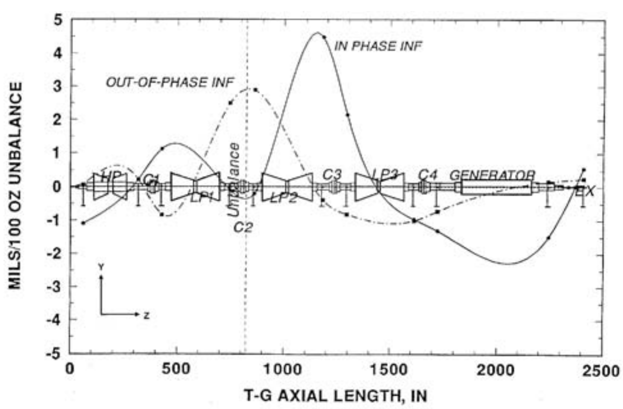

Figure 15: T-G Influence Coefficient Response at 1800 RPM With Trial Weight at 2nd Coupling C2

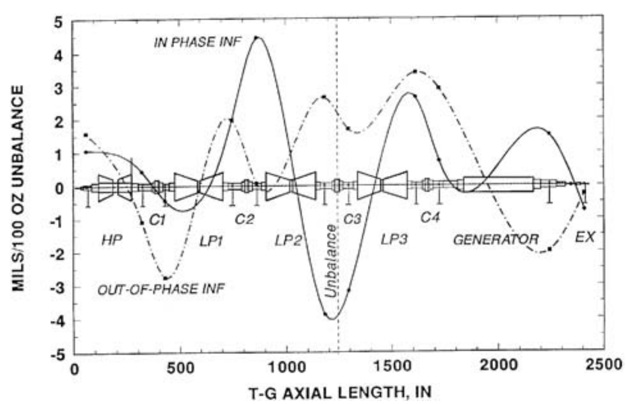

Figure 16: T-G Influence Coefficient Response at 1800 RPM With Trial Weight at 3nd Coupling C3

Figure 15 represents the T-G complex response with a trial weight placed at the second coupling. The complex response was resolved into in-phase and out of phase components. A cubic spline curve fit was used to generate the shaft response. Note the large response obtained at the third coupling C3. Fig 16 represents the complex influence response with a trial weight placed at the third coupling, C3. Note the large responses of the system at couplings C2 and C3. Also note that there is very little response of the mass centers of the turbine rotors.

Figure17 represents the influence coefficient response with the trial weight place at the number 4 coupling. This coupling location connects the generator with the last turbine LP3. The trial weight placed at the generator coupling has a significant influence on the generator response. There is a large out-of-phase 2nd mode type of reaction on the generator. Generator bowing in operation due to conductor shorting tends to create a bow resembling the first mode shape.

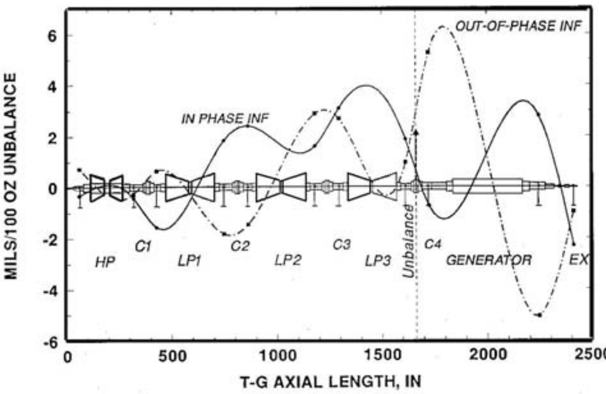

Figure 17: T-G Influence Coefficient Response at 1800 RPM With Trial Weight at 4th Coupling C4

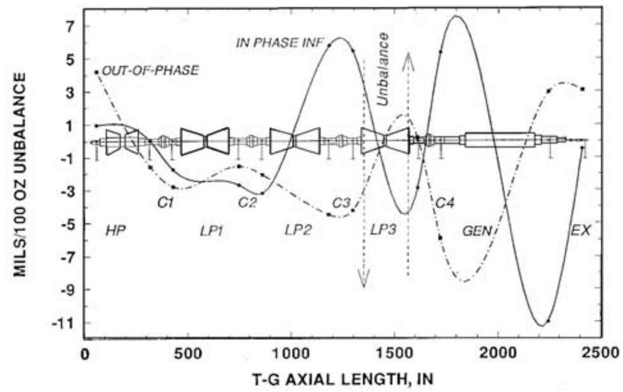

Figure 18: T-G Response With at 1800 RPM With Couple Unbalance on LP3

The influence coefficients were generated for individual trial weights placed on either ends of the LP3 rotor. By vector resolution, the influence response was computed for a pure couple response on the LP3 rotor. As can be seen in Figure 18, there is a substantial response on the generator.

5.2 Analytically Determined Influence Coefficients For Coupling C1 Balance Plane

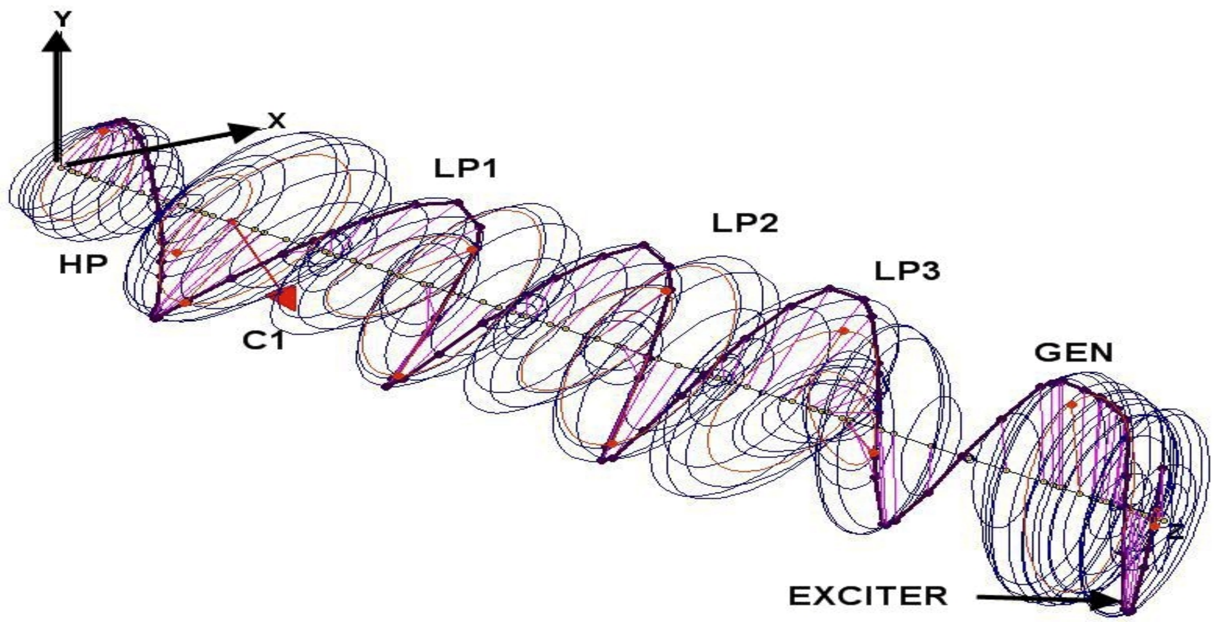

Figure 19: Analytically Determined Influence Response at 1800 RPM For Balance Weight At 1st Coupling - C1

In the balancing procedure, it was determined after many balancing calculations, that an additional balancing plane using the first coupling location could be of benefit in balancing the turbine-generator at speed. The use of 6 or more balancing locations seems logical since the system is theoretically operating through 10 critical speeds. It is normally preferred to have at least the same number of balancing planes as critical speeds required to operate through.

Since no experimental data was available to generate the influence response of the rotor system for a weight placement at the first coupling, the response was generated analytically using a finite element rotor dynamics program. However, before this could be accomplished, numerous simulations of the known influence responses were computed to estimate the pedestal effects. Although the LP turbine bearings are similar, there stiffness and damping coefficients vary due to the applied loads as shown in Table 1.

6 Multi-plane Field Balancing Calculations Based on Experimental and Analytical Influence Responses

Multi-plane balancing corrections were computed assuming various balancing planes. All balancing calculations in this study was performed using a modified least squared error method which could also incorporate weighting values in the data. This provides some flexibility if one desires to emphasize the reduction of response at certain locations such as at or near the generator. Table 2 represents the initial T-G bearing measurements at speed. It is important to note that all measurements and influence coefficients responses must be based on the same generator power conditions. As generator output is varied, so is the thermal bowing of the generator. Reliable data cannot be obtained if taken under various power conditions. After the balancing corrections are computed, the balancing program predicts the new response after the application of the computed weights.

6.1 Three-Plane Balancing With Couplings C2, C3, and C4

The first balancing prediction was based on the use of the influence responses at the second, third and fourth couplings. The use of only these three couplings produces a reasonable system response. With this balancing combination, it is seen that amplitude at the HP turbine bearing no 1 increases as well as bearing no 3 on LP2 rotor and bearing 7. The responses on the LP3 rotor and generator are substantially reduced. It is noticed that the exciter amplitude has increased. Observation of the exciter behavior indicates that it should be balanced last as a single plane. This situation then would represent a 4-plane balancing solution.

Table 2 - Initial and Predicted Rotor Response with Coupling Planes and LP3 Dynamic Couple Balancing

Probe Location	Initial mm	3 Planes CPLS 2, 3, & 4 mm	5 Planes - CPLS 2, 3, 4, & LP3 Dynamic mm	6 Planes CPLS 1-4, & LP3
1	.055	.060	.043	.023
2	.045	.039	.030	.050
3	.124	**.133**	.055	**.076**
4	**.138**	.109	.090	.072
5	.107	.091	**.098**	.022
6	.090	.025	.065	.029
7	.058	.076	.034	.053
8	.108	.055	.083	.067
9	.088	.038	.065	.059
10	.056	.028	.055	.016
11	.075	.087	.070	.070

6.2 Five-Plane Balancing With Couplings C2, C3, C4 and Dynamic Couple on LP3 Turbine

The second balance computation involves the use of the three couplings and the two planes on the LP3 turbine. With this balance calculation, the amplitudes through bearings 1 to 4 are reduced, however the amplitude at the generator coupling and generator-exciter end bearing return to its original value of 0.055 mm. It should be noted that the balance values on the generator may be greatly improved if the exciter motion is ignored in the least-squared-balance calculations. The exciter may be single-plane trim balanced after reduction of generator and turbine vibrations.

In this balance calculation, the largest vibration is at bearing No. 5 with 0.098 mm. The desired vibration amplitude at running speed is 0.076 mm (3 mils). This balance level is achieved at all locations except at bearings 4 and 5. As noted in the previous balancing run with only the 3 coupling planes used, the required balance level may be obtained if the balance calculations are based on ignoring the exciter. This is done by the assumption of a zero weighting factor for the exciter readings. The exciter is then balanced separately by as a single plane. Because of the extreme flexibility of the exciter span, exciter trim balancing has little influence on the response of the generator and turbines.

6.3 Six-Plane Balancing With Couplings C1, C2, C3, C4 and Dynamic Couple on LP3 Turbine

In the last balancing case, as shown in Table 2, the first coupling was added as an additional balance plane. Since no experimental influence responses where measured, the required influence coefficients were numerically computed from the analytic model. Figure 19 shows the theoretical unbalance response with a trial weight placed at the coupling. The first coupling between the HP rotor and the first LP rotor was not originally considered as a possible balance plane to control generator response. At the time, the operating engineers assumed that this location was too far removed from the generator to have an influence on it.

With the addition of the first coupling as an added balance plane, the vibrations of the entire turbine-generator system may be reduced to the desired level of 0.076 mm. When a least-squared-error method is employed in a multi-plane balancing calculation, we often have the situation in which several locations are over corrected and insufficient reduction is achieved at other locations. For example, bearings 5 & 6 are overbalanced with amplitudes of 0.022 and 0.029. The LP3 and inboard generator bearing predicted response is 0.67 and 0.59 respectively. There are several ways to improve and enhance the balancing at the generator and LP3 turbine. The first method is to ignore the influence of the exciter. The second method is the use of weighting functions to place more emphasis on generator vibration reduction. This requires some trial runs but the computations may be rapidly computed on a PC-based balancing program.

In the third method, the balancing is computed using linear programming theory. Further research is being conducted on this technique as it does not require operator intervention with weighting factors applied.

6.4 Balancing Summary With Various Couplings and 2-Plane Moment Balance on LP3 Turbine

Figure 20 represents a plot of amplitude vs bearing number for the initial vibration and of the 3 cases shown with 3, 5, and 6 planes of balance corrections applied to the turbine-generator at running speed. Superimposed on the plot is a dotted red line at 0.076 mm. This represents the desired level of vibration amplitude that it is preferred to operate at. The 6 planes of balance correction clearly meet the desired response level using the 4 couplings and a moment correction on the LP3 turbine. It is apparent from the balancing results that the unmodified least-squared-error balancing method overcorrects the balance of the LP2 turbine. A finer level of balancing of the LP3 turbine and inboard generator bearing may be achieved by the specification of weighting factors to reduce the influence of the amplitudes at the 5th and 6th bearings corresponding to the LP2

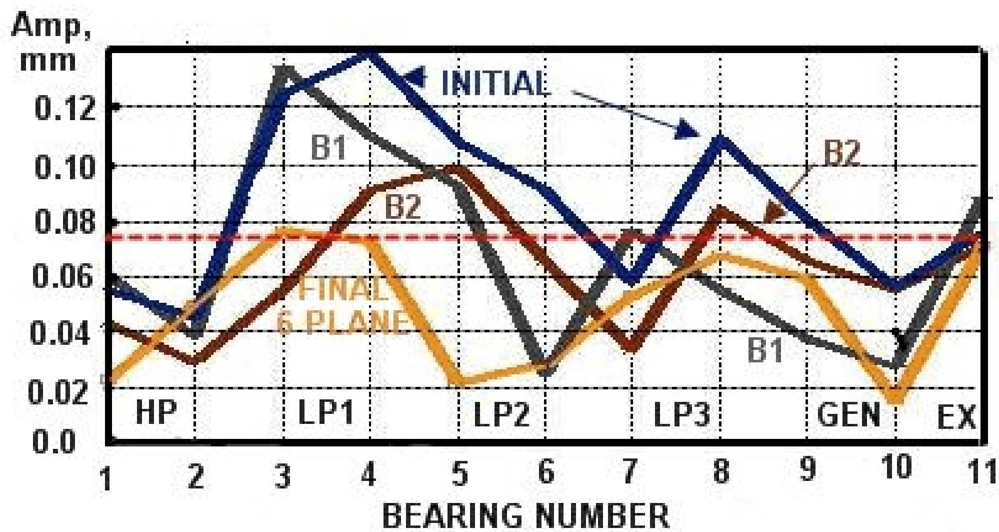

Figure 20 - Summary of Initial And Various Balancing Runs
Amp mm Vs Bearing Number

turbine and weighting factors to increase the influence of the No. 8 LP3 bearing and No. 10 , the inboard generator bearing. In all cases, to adequately reduce the vibration levels of the exciter, a separate single-plane balancing should be performed on the exciter. This is done after balancing of the main turbine-generator is accomplished. The balance along the entire turbine-generator is improved when the exciter vibrations are initially ignored. This will result in a balance state in which the turbines and generator will be well balanced but the exciter amplitude has increased. The exciter amplitude may then be reduced by single-plane balancing.

7 DISCUSSION AND CONCLUSIONS

7.1 Discussion

It is highly desirable to develop field balancing procedures that can assist in balance corrections of large turbine -generators on site. The application of arbitrary balance corrections can be a costly and time consuming process if conducted manually without the aid of a computer. The balancing process is dependent upon the development of accurate response influence coefficients. This requires that the data be taken at the same power levels for the initial and influence response runs. These power levels need to be recorded. If the alignment of the intermediate bearings changes due to pedestal sag, then the experimental influence coefficients will not be accurate.

7.2 Conclusions

1. To properly understand field balancing of large turbine-generators, it is desirable to develop analytic models of the complete system including foundation effects.
2. The transfer matrix method is not suited for the analysis of larger turbine-generators. With the finite element method, it is possible to compute the critical speeds, complex eigenvalues, unbalance response and time transient motion due to blade loss.
3. A valid T-G model must include pedestal effects such as mass and stiffness. The flexibility of the pedestal supports may result in over a 90% reduction of effective damping. Pedestal characteristics may be found by testing and by the simulation of the experimental influence response motion with the analytic model.
4. Calculations indicate as many as 10 undamped modes exist in the operating speed range (and as many as 30 complex eigenvalues, of which only a few are of interest). The undamped modes show that the lower modes involve motion of the turbine and generator mass centers. Thus, low-speed balancing is useful in controlling these modes. For low-speed balancing of the turbines in a balancing machine, the 2 planes of balance correction should be resolved into static and dynamic (couple unbalance).

Half of the computed static correction should be applied at the rotor center with the remainder placed at the end planes. Failure to do this may result in a turbine that is not well balanced at speed.

5. The experimental unbalance response influence coefficients must be developed under repeated T-G load conditions. It was found that for identical T-G units, the influence coefficients may be substantially different. The influence coefficients for two identical units should not be mixed in the balancing computations.

6. The bearings of these large T-G sets are heavily loaded. Hence, jacking oil must be used to support the shafts under startup and shutdown (loss of jacking oil during shutdown will lead to bearing failure). At speed, the jacking oil is turned off. The hydrodynamic pressure generated at each bearing was recorded. This pressure measurement may be used to determine the bearing loading. A settling of some of the interior bearing pedestals may cause these bearings to carry reduced loading. This in turn will effect the influence coefficients. Two identical units may have different influence coefficients due to the differences in vertical alignment and hence bearing loading. For the situation in which several interior bearings are lightly loaded, the vertical alignment should be adjusted. From the measurement of the bottom bearing oil pressure at speed, it is possible to determine the bearing loading and hence predict the bearing 8 stiffness and damping coefficients at speed for use in dynamic analysis such as unbalance response and stability (damped eigenvalues).

7. At operating speeds, the mass centers of the turbines become nodal points. The motion of each turbine is a conical type of motion. This conical behavior occurs after the fourth mode as shown in Figure 10. In Figures 11-14, which represent modes 7 to 10, it is seen that the turbines have conical motion. Therefore, it appears that the placement of static balance shots on the turbine at either end in phase are not indicated. The turbines at speed better respond to dynamic balance couples in which two weights are placed on a turbine at either end out of phase.

8. Satisfactory field balancing is difficult to achieve with fewer than four balance planes. A satisfactory balance condition can be achieved with the four couplings and a two-plane moment correction to LP3 turbine. Using a least- squared-error balancing procedure, it may be necessary to incorporate weighting values to interior measurements so as not to over correct the interior turbines and not adequately reduce the generator vibrations.

9. Due to the extreme flexibility of the exciter, it should be balanced last as a single plane. The excessive flexibility of the exciter not only leads to balancing difficulties, but also places the entire T-G system at risk. The loss of an LP3 turbine blade will lead to exciter failure with catastrophic results.

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