Dynamic Analysis of An 1150 MW Turbine Generator

Edgar J. Gunter
Fellow ASME
RODYN Vibration Inc
Charlottesville, Va. 22903
DrGunter@aol.com

Abstract

This paper presents the dynamical analysis of an 11 bearing 1150 MW turbine-generator system . Original studies of this system yielded information on the system critical speeds and mode shapes. These calculations were first generated by the transfer matrix method. It was found that the transfer matrix method is unsuited for the analysis of large turbine - generators for a number of reasons. The first is the problem of convergence of the modes with a large number of mass stations. The second is that the iteration procedure misses modes that are closely spaced. In the study of the dynamical behavior of large T-G sets, it was determined that it was necessary to include the foundation or bearing pedestal effects. This not possible with the transfer matrix method due to the numerical problems associated with branched elements of the supports.

The system critical speeds were computed using a structural finite element program. This approach could generate the system modes, but is not capable of computing damped eigenvalues, unbalance response, or to perform accurate time transient analysis to evaluate system motion and bearing forces transmitted due to blade loss.

With the recent enhancements to the PC based finite element program ***DYROBES*** , it is now possible to perform both linear and nonlinear time transient studies on large turbine - generator systems as well as damped eigenvalue analysis and unbalance response. In the calculation of the undamped critical speeds, it was observed that there can be as many as 12 undamped modes in the operating speed range. Not all of these modes need to be of concern. A mode is of concern if it has a low log decrement and is in the vicinity of the operating speed, or that it has a negative log dec which indicates that it may be unstable. In order to compute the damped complex 3 dimensional eigenvalues of the system, the 8 bearing stiffness and damping coefficients for the 11 bearings must be known. These values were computed for each bearing and are then used in the calculation of the damped modes. Instead of considering only 12 modes, one must compute the first 30 complex modes to span the frequency range of interest. These modes represent forward, mixed, and backward modes. Only several of these modes are of concern. There are several forward modes that are near the operating speed and have high exciter and LP turbine motion. These modes also have low log decrements which makes them of concern particularly as regards to a suddenly applied unbalance.

A time transient analysis is required in order to access the T-G response at running speed due to a suddenly unbalance such as caused by blade loss. One of the limitations with structural finite element programs is that transient analysis is accomplished by assuming a set of undamped modes. One is then required to assume a percent of modal damping for each mode. This approach may be acceptable for structural systems, but it is not acceptable for a rotor dynamics analysis in which the bearings have high damping in addition to the bearing cross coupling coefficients which structural FEA programs can not handle . A time transient analysis was performed using ***DYROBES*** to simulate 6 cycles of shaft motion. In this simulation of LP3 blade loss at running speed, one of the system modes around 1600 CPM was excited to the extent that exciter damage could occur leading to system failure. It is concluded that exciters of this class may be insufficiently supported and also have insufficient damping to withstand blade loss.

1 Background And Introduction

A schematic representation of the 1150 MW turbine-generator system is shown in Fig. 1. The complete system has 11 bearings with an HP turbine, 3 LP turbines, generator and exciter.

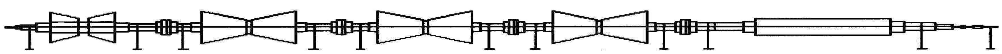


Figure 1 Schematic Diagram of 1150 MW Turbine-Generator

The complete dynamic analysis of a large turbine-generator with multiple bearing supports represents a substantial challenge from a number of standpoints. -

2 Modeling of Turbine Generator System

Component Models

The procedure for modeling the complete turbine - generator system consists of generating first modeling the individual components of the system. The initial modeling of the various rotors was generated by the transfer matrix program ***CRITSPD-PC.*** For example, Fig. 2 represents a model of the HP turbine and shows the first critical speed mode shape. Fig. 3 represents the 3rd mode of the LP1 turbine.

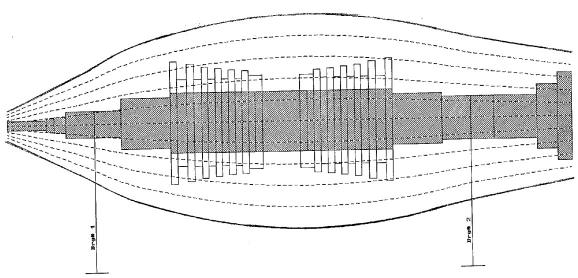


Figure 2 HP Turbine 1st Critical Speed Mode

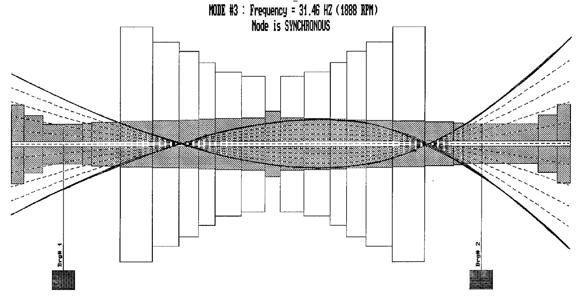


Figure 3 LP Turbine 3rd Critical Speed Mode

In the HP turbine 1st critical speed mode as shown in Fig. 2, the frequency is not greatly effected by the coupling to 1st LP turbine. Fig 3 represents the LP turbine 3rd critical speed free-free mode at 1809 RPM. The coupling of the LP rotors greatly alters the LP modes. The LP2 and LP3 turbines are similar.

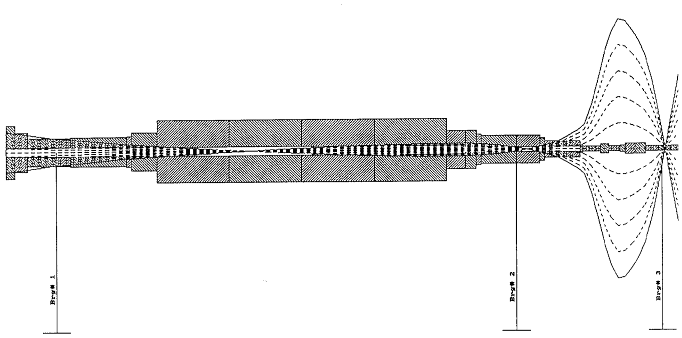


Figure 4 Generator -Exciter Model 3rd Critical Speed at 1,452 RPM (24.2 HZ)

The model of the generator and exciter showing a local exciter mode is shown in Fig. 4. Fig 4 shows the long generator with the attached exciter. It is important to note that the local exciter mode frequency is unaffected by the attached turbines. However, motion at the generator coupling may effect the exciter.

Not shown is the generator first mode shape. Since the generator is the longest component in the system, then it has the lowest 1st critical speed. The mass of the exciter has little influence on the generator 1st critical speed.

Model Reduction For Numerical Convergence

In order to control numerical problems encountered with large rotor dynamic models, the size of the models must be reduced. Models were generated using both the transfer matrix and finite element methods. A large turbine model with 40 or more stations may be easily reduced to 20 or fewer for dynamic analysis purposes. with no loss of accuracy. For example, the nth mode of a uniform beam may be computed with less than 1% error with only 2n +1 nodes. In the transfer matrix method, model reduction is accomplished by the process of lumping stations. The unit critical speeds are recomputed. Satisfactory reduction is accomplished when the critical speeds of the particular unit under reduction does not change in modal properties by more that 1 % for the first 3 modes.

Rotor models were also generated using the finite element program ***MSC/Pal2*** and the finite element rotor dynamics code ***DYROBES***. In the first case, reduction is achieved by the specification of master degrees of freedom. In the second case, model reduction is accomplished by the use of subelements. This process is necessary to insure computational accuracy of system dynamics such as the calculation of 3 dimensional complex mode shapes and time transient analysis due to blade loss.

2 System Critical Speeds

It is very difficult to compute the system dynamics of an 11 bearing turbine-generator system using transfer matrix methods. There are several numerical problems with the transfer matrix method that precludes its use for accurate computations of rotor dynamic behavior of large turbine generators. The first problem is illustrated by Figures 5 and 6. The transfer matrix method will skip closely spaced modes.

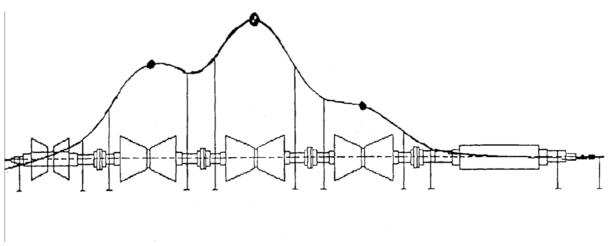


Figure 5 2nd T-G System Mode at 562 RPM

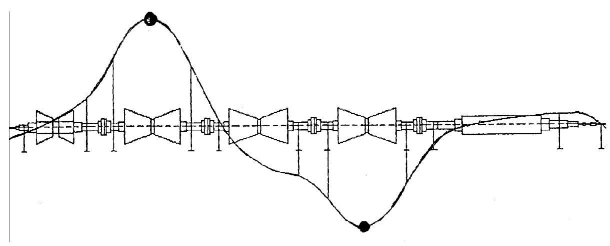


Figure 6 3rd T-G System Mode at 569 RPM

The above modes were computed using the structural dynamics options of the PC based finite element ***MSC/Pal2*** program. Although this program does not skip modes, it is not a true rotor dynamic program in that it can not incorporate generalized bearing coefficients or shaft gyroscopic moments.

Figures 7 and 8 represent the 10th and 11th modes of the system. Note that both modes have high exciter motion. Suddenly applied unbalance at LP2 or LP3 could lead to high exciter amplitudes.

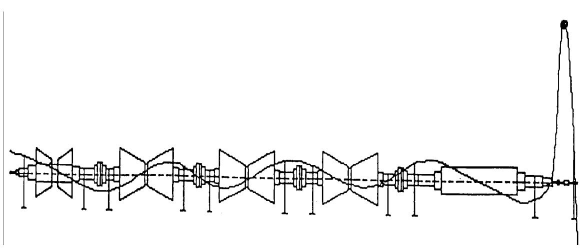


Figure 7 T-G 10th Mode at 1,459 RPM

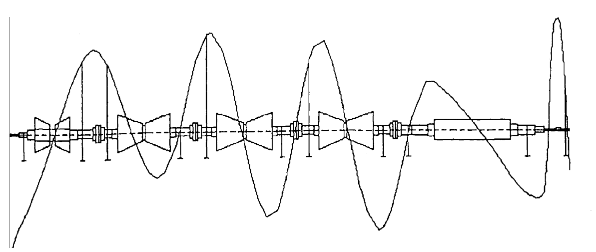


Figure 8 T-G 11th Mode at 1,673 RPM

3 Bearing Stiffness and Damping Characteristics

In the undamped critical speed analysis, only a simple spring is assumed acting at each bearing. It is desirable to know the log decrement of the damped natural frequencies. In order to compute the complex rotor modes, we must first obtain the bearing stiffness and damping coefficients as a function of speed. In this presentation, the value of the bearing coefficients at speed is sufficient. The bearing coefficients are computed from the Reynolds equation of lubrication. Figure 9 represents a typical LP bearing pressure profile and corresponding bearing coefficients.

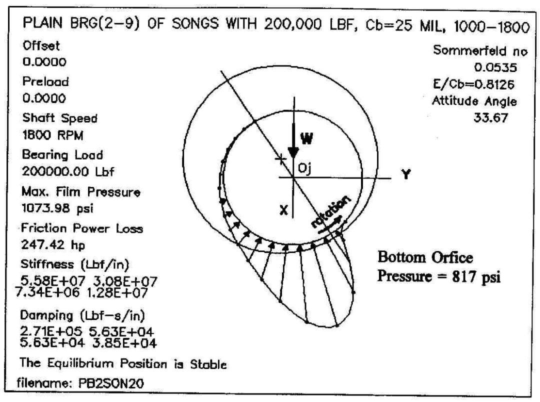


Figure 9 **LP Bearing Pressure Profile And Bearing Coefficients at 1,800 RPM**

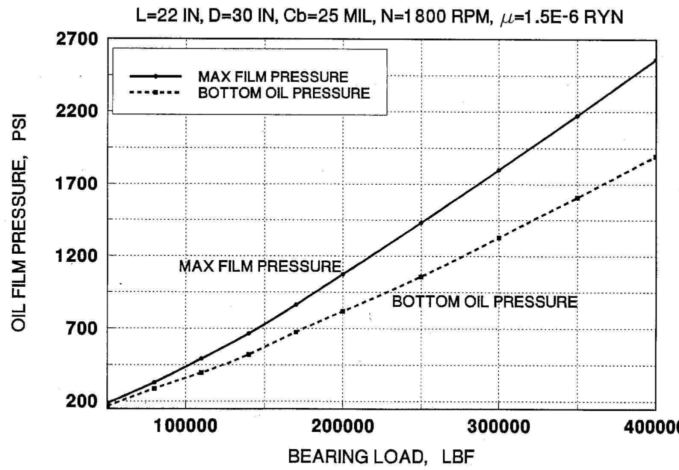


Figure 10 **Orifice and Max Film Pressure Vs Load**

The jacking oil orifice pressure at running speed may be monitored. By measuring this pressure, Fig 10 shows the relationship between the orifice pressure at the bottom of the bearing, maximum pressure and hence the static loading on the bearing. A reduced orifice pressure indicates that the bearing alignment must be adjusted for proper load sharing. Two seemingly identical machines may have different sets of influence coefficients because of changes of vertical alignment in one or more of the bearing supports. Even at running speed, the bearings are operating under high eccentricity ratios. Hence jacking oil is needed for liftoff. To avoid bearing failure during shut down one of the modern synthetic lubricants with boundary lubricant additives is recommended to prevent galling.

In addition to the addition of the fluid film bearing properties, it is essential to include the mass and stiffness properties of the foundation. The effective stiffness of the bearings may be reduced as much as 90% from the original values. Loose grout under a bearing pedestal may result in as much as a 95% reduction in effective damping. This makes field balancing very difficult and ineffective.

$$C_{eq} = \frac{C_b\left(K_s + j\omega C_s\right)}{\left(K_s + K_b\right) + j\,\omega\left(C_s + C_b\right)};$$

$$Where\ K_s = K_{found} - M_s\,\omega^2$$

$$C_b = C_{xx} - \frac{K_{xy}}{\omega}\ or\ C_{yy} + \frac{K_{yx}}{\omega}$$

The transfer matrix method can not be used to compute rotor response including foundation effects. This is due to the problem of numerical convergence with branched elements. This situation is overcome by using a finite element rotor dynamic formulation.

4 Damped Turbine-Generator Modes Including Foundation Effects

Figure 11 represents a finite element model of the entire turbine - generator set including the pedestal effects.

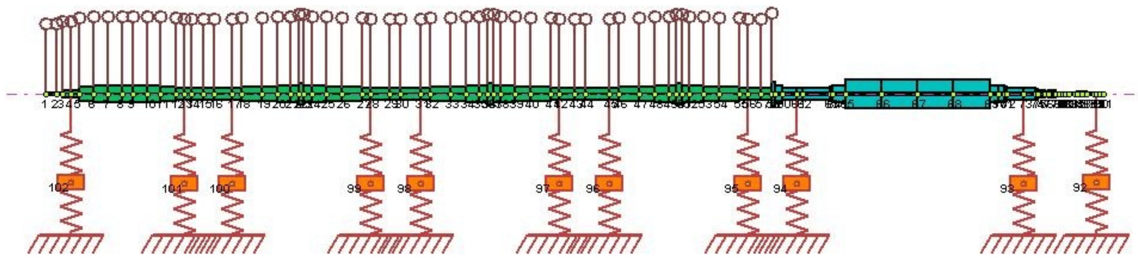


Figure 11 **1150 MW Turbine-Generator Finite Element Model Including Pedestal Effects**

Damped Mode At 1,515 CPM With High Exciter Amplitude

Using the finite element ***DYROBES*** rotor dynamics program, it is now possible to compute the complete system damped eigenvalues including all bearings and pedestal effects. This is important as it shows the sensitivity of the various critical speeds and gives an indication as to what modes may be easily excited by unbalance or blade loss at operating speed.

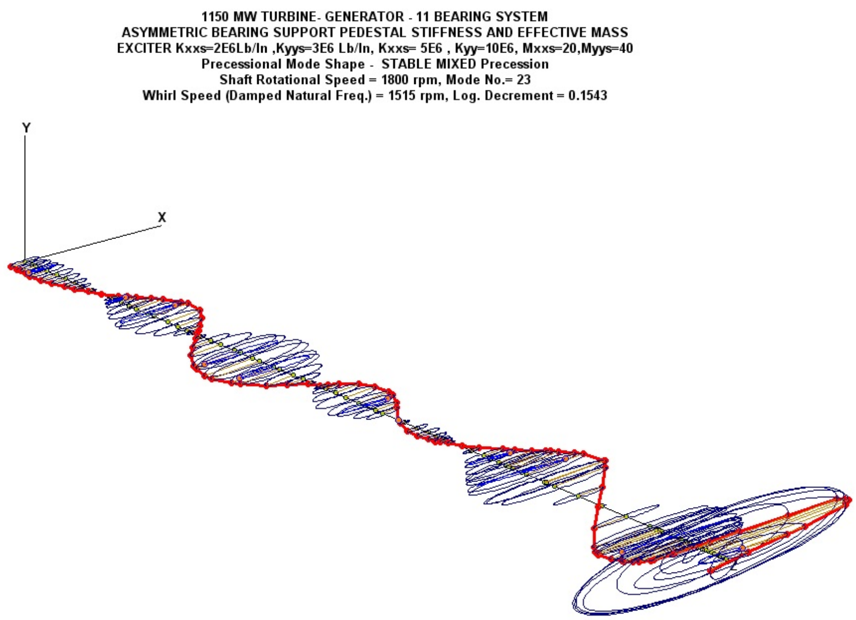


Figure 12 **Turbine-Generator System Mode M23 at 1,515 CPM With Ldec=0.15**

Figure 12 for mode M23 at a frequency of 1,515 CPM (25.25 HZ) shows high exciter motion which could be exciter by blade loss on any of the LP rotors. In order to determine the magnitude of the amplitude at the exciter from blade loss , it is necessary to perform a time transient analysis.

Damped Mode At 1,851 CPM With Log Dec of 0.1-High Amplification Factor Mode

Figure 13 represents complex system mode M26 at 1,851 CPM. This mode is very close to the operating speed of 1,800 RPM. This mode has an even higher amplification factor then does mode M23 This log dec for this mode is 0.103 which translates to an amplification factor of over 30! The LP rotors all exhibit large out of phase conical motion. Hence blade loss on any of the LP rotors could lead to excitation of modes M23 and M26 with resulting exciter failure.

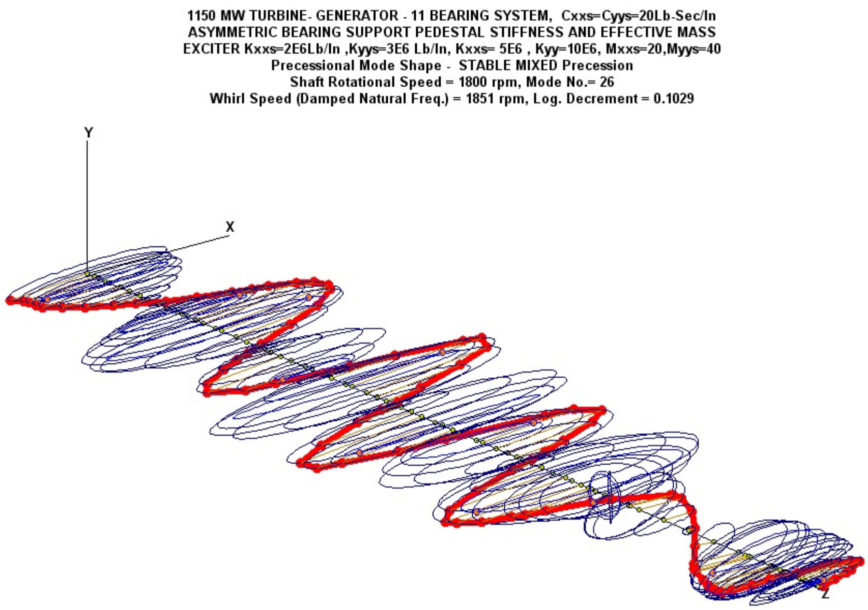


Figure 13 Damped Mode M26 At 1,851 CPM, Log Dec =0.1

5 Time Transient Analysis Turbine-Generator With Suddenly Applied Unbalance

Figure 14 represents the options for a time transient analysis using the ***DYROBES*** program.

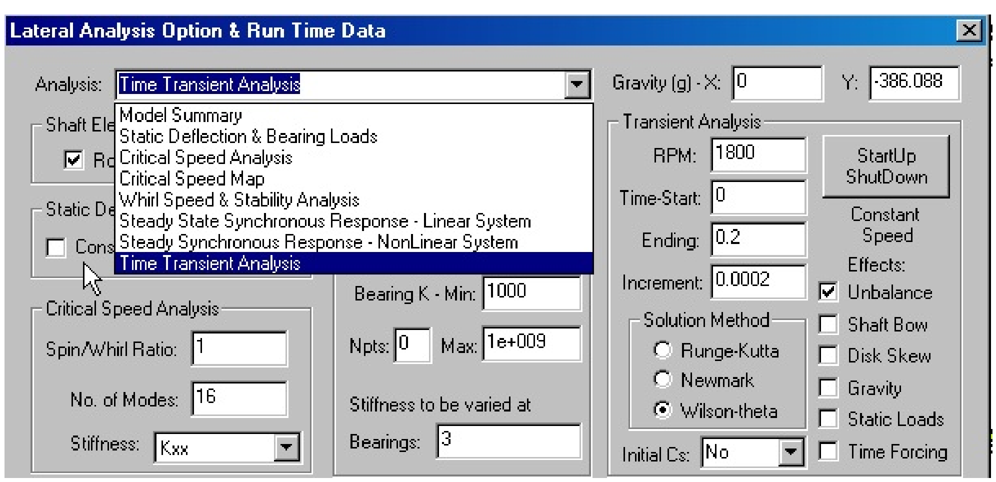


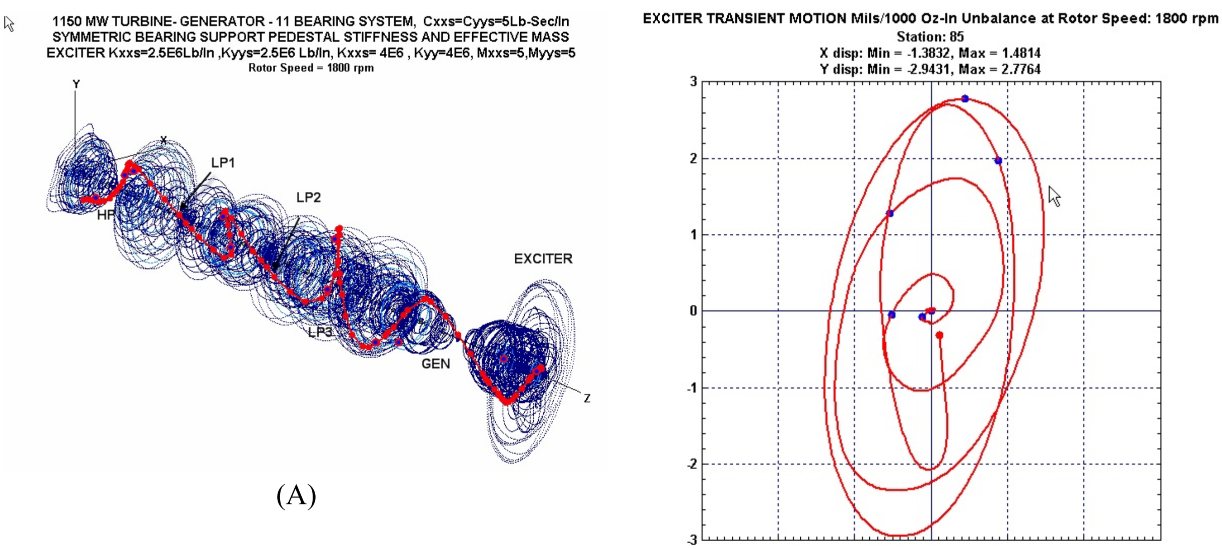
Figure 14 Options For Time Transient Analysis

Time Transient Options

The time transient analysis may be performed using several integration schemes. In this case, the Wilson-Theta method was chosen. The integration was computed for a total time of 0.2 sec with a time step of 0.0002 sec. The time transient may have multiple types of loads specified. In this case, only rotor unbalance was specified. The model used in the transient analysis was slightly modified for symmetric pedestal conditions. This causes a slight altering of the previous modes. Figure 15 represents the time transient motion with suddenly applied unbalance at the LP3 rotor. Figure 15-a represents the motion for 0.2 sec. This represents 6 cycles of shaft motion. It can be seen that the exciter has a large initial transient response in comparison to the motion along the turbine -generator.

Figure 15 b represents the transient response at the exciter location. The scale of the plot is given in terms of mils/1,000 oz-in of unbalance. The maximum amount of the excursion encountered with the exciter occurs after 5 shaft cycles of motion. The maximum exciter motion is almost 3 mils/1,000 oz-in of unbalance.

Transient Shaft Motion With LP3 Blade Loss



(A)

(B) Exciter Orbit

Figure 15 Time Transient Motion Of Turbine - Generator With LP3 Blade Loss

Transient Exciter Bearing Forces

Figure 16 represents the vertical exciter bearing transient forces transmitted for the first 0.2 sec of motion. As can be seen from the figure, the maximum forces occur 0.1 sec after blade loss. The peak exciter force from the initial transient is over 2100 Lb per 1,000 oz-in of LP3 blade unbalance. If a large section of LP3 blading is lost, then exciter bearing failure occurs shortly after this time period.

The LP3 blade loss excites one of the system damped modes. An FFT analysis of this transient wave form indicates a frequency of approximately 1645 CPM. This is one of the modes with high activity of the LP rotors and exciter.

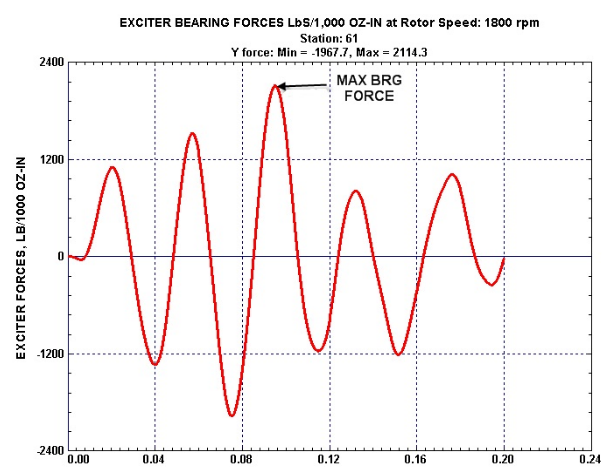


Figure 16 Exciter Bearing Transient Vertical Forces

Failure of the exciter may be sufficiently violent to rupture the bearing housing. This may lead also to the generator rubbing causing a leakage of the hydrogen cooling gas. This may in turn cause a fire with a hard generator rub encountered. Since the turbine shutdown system can not respond in 0.1 sec , the excessive torque at the generator coupling may fail leading to damage of the LP3 rotor.

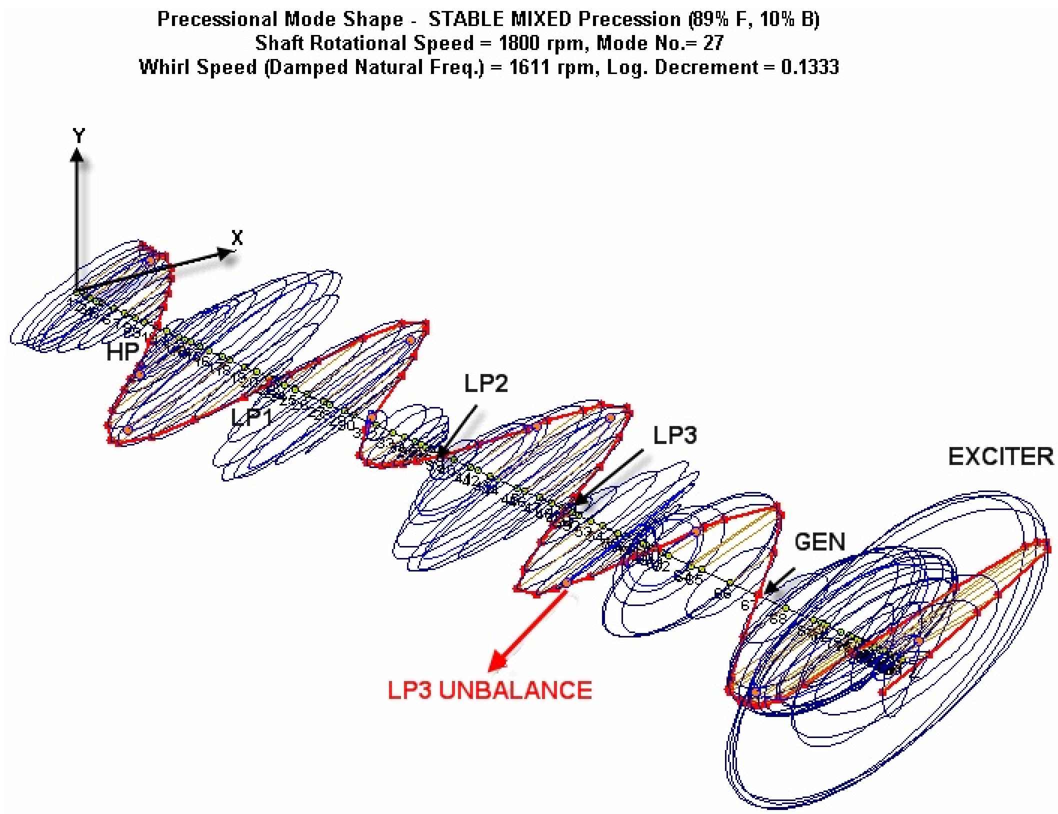


Figure 17 **Turbine Generator M27 at 1611 CPM With Log Dec=0.133**

It appears from examination of the time transient motion, that the blade loss causes an excitation of the M27 forward mode. In this simulation, symmetric bearing pedestal stiffness values were assumed. This mode varies only slightly from previous models and is at 1611 CPM. It can be seen from an examination of Figure 17 that a large excursion of the exciter is encountered. Redesign of the exciter is desirable with a stouter shaft and larger bearing to reduce the exciter response due to blade lo6

6 Discussion and Conclusions-

1. Transfer matrix methods are not suited for the dynamical analysis of large T-G systems.
2. Damped modes and time transient analysis may be computed by suitable finite element based codes.
3. Fluid film bearing characteristics between seemly identical trains may vary due to the differences in vertical bearing alignment. Therefore load sharing may be unequal. Measurements of the orifice pressure at speed is a good indicator of the static bearing load.
4. A valid T-G model must include pedestal effects such as mass and stiffness. The flexibility of pedestal supports may result in over a 90% reduction of the effective bearing damping. Flexibility or softness under the exciter bearing can lead to a very dangerous situation. Pedestal characteristics may be determined by experimental testings.
5. After passing through the lower system modes with high motion of the rotor mass centers, the higher modes above 1,000 RPM have predominately conical motion of the components. Therefore is not effective to place balance corrections at the rotor center spans at speed since these center of gravity locations become nodal points.
6. The damped eigenvalue analysis of an 11 bearing T-G system may compute as many as 30odes within or near the operating speed. Only a few of these are of interest. These are the forward modes with low log

decrements. If an interior bearing should become unloaded due to foundation sag, the lightly loaded bearing characteristics may cause a self excited whirling which shows up as a negative log decrement. Therefore, at running speed, one would observe a subsynchronous component of motion. This condition can not be corrected by balancing and requires that the bearing pedestal be brought back into alignment .

7. Many exciters appear to be designed without regard to the system dynamics. Very flexible exciter shafts with lightly loaded bearings may have excessive respond to transient LP rotor unbalance leading to system failure. Modes such as shown in Figure 17 with the high exciter motion, should have a redesign to reduce the exciter amplitude and increase the modal log decrement.

8. Transient analysis of a large T-G system is possible using the finite element approach. This is not possible using the transfer matrix method. Transient analysis techniques allows for the design of components such as the exciter to resist high unbalance loads without failure. Such design concepts could include a nonlinear squeeze film damper at the exciter bearing for transient load resistance.

9 Analysis of the system modes provides design insight towards balancing as well as stability analysis and sensitivity of the various mode to unbalance.

10.. Analysis of the bearing eccentricity locus vs speed indicates that the LP bearings operate under heavily loaded conditions even at running speed. This implies that during shut sown, jacking oil is required to prevent bearing damage. Conventional ISO 32 and 64 grade turbine oils provide poor boundary lubrication properties. It is recommended that these oils be replaced with the newer synthetic (designer) lubricants that also have EP additives and higher viscosity indexes. A higher quality oil also would provide more protect to the exciter bearing under transient blade loss. With conventional ISO lubricants, high contact exciter bearing forces may cause the exciter shaft to seize and fail. The superior synthetic lubricants could prevent this from occurring.