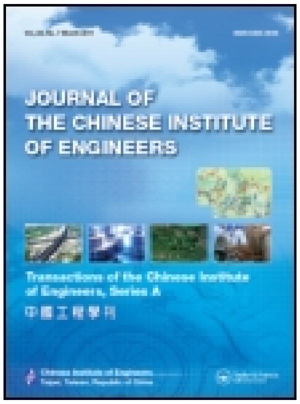

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Analysis of vibration in shaft-disc-blades systems due to power faults using a finite element method

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Analysis of vibration in shaft-disc-blades systems due to power faults using a finite element method

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The calculations of stress distribution and mode frequencies of a turbine-generator mechanical system obtained by using the finite element method (FEM) are more accurate than those obtained by using the lumped parameter program. However, FEM is not capable of analyzing the disturbances coming from the generator. In this article, a modified dynamic program is used to simulate the shaft vibrations resulting from power system faults. The power spectrum density (PSD) method is adopted to assist in the analysis of shaft vibrations occurring at various frequencies. By PSD analysis of the global plant model, the resulting system parameters provide the necessary information on boundary and load conditions needed by FEM. ANSYS, the potent FEM software, is used to propose the prediction steps of the shaft-disc-blades system.

Keywords: vibration; shaft-disc-blades; FEM; PSD

1. Introduction

Due to huge size, complex structure, and high prices, conducting research on turbine-generators, experimentally, is usually difficult and restricted. Currently, only computer simulation software developed for lumped parameter models can thoroughly analyze the components of these machines which are subject to disturbances. Vibration analysis of mechanical systems has been studied by many investigators, e.g., IEEE Working Group on Computer Modeling of Excitation System (1982), Tsao and Chyn (1990), Chyn *et al.* (1996), Yang and Huang (2005), Lim *et al.* (2007), and Sinha (2007). In dynamic analysis using these models, a single inertia term is used to represent one stage of shaft and disc; furthermore, the blade is modeled as an object with one degree-of-freedom. In comparison, the finite element method (FEM) is superior to lumped parameter models because its mesh analysis incorporates more information on the stress and frequency of oscillation (Chen and Tsai 2000, Cha and Zhou 2008).

On the other hand, FEM lacks the capacity of analyzing the electric power disturbances introduced by the generator. In view of the fact that the existing simulation (with lumped parameter models) on the torsional response is quite accurate, while further attempts to include the discs and the blades fall short of accuracy owing to the crude modeling, we choose to adopt conventional torsional response simulation to obtain information essential to FEM analysis, and proceed with further mesh analysis on the discs and the blades. Some papers have been published on the comparison of simulation and test results, e.g., Cudworth and Smith (1990) and He *et al.* (2007).

In this article, an improved dynamic computer program was used to simulate turbine-generator shaft vibration caused by four types of power faults. The contribution of this article is that, this dynamic simulation was developed for state-space models, and is capable of analyzing various kinds of electric power disturbance, including non-symmetric faults (Smith *et al.* 1986, Hammons and Lim 1999, Gopalakrishnan *et al.* 2000). We further utilized power spectrum density (PSD) methods to analyze the vibration spectrum. Boundary and load conditions for FEM were found by using the PSD method to make the analysis of discs and blades more trustworthy (Press *et al.* 1986, Zhang *et al.* 2001, Qian *et al.* 2007).

ANSYS software was used in the FEM analysis to demonstrate prediction methods for dynamics of shafts, discs, and blades. For systems expressed in linear state-space representation, frequency and damping for every vibration mode can be found by applying eigenvalue analysis just once. We also focused on the

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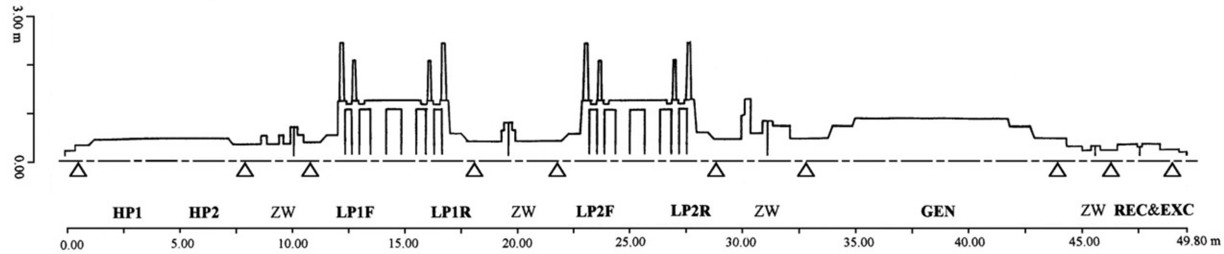

Figure 1. The side drawing of a nuclear generator.

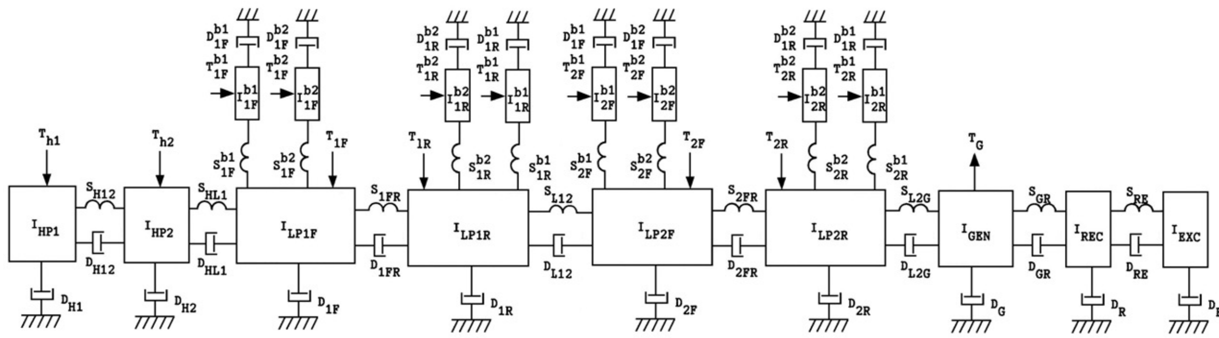

Figure 2. The mechanical system model of a 985 MW turbine-generator.

nuclear power generators of the Taiwan Power Company that are currently in service, and conducted a dynamic analysis.

2. Computer program and analysis

2.1. *Establishment of system model*

Turbine rotors of turbine can be classified into shrunk-on rotor, forging rotor, and drum rotor; categories, depending on whether the discs are attached. Blades of turbines also vary: some are freestanding; some may be connected via tie rods, lancing wires, or shrouds. For instance, welded drum rotors are presently used in the second nuclear power plant of Taiwan. The low-pressure (LP) turbine therein has 11 stages of blades; the first nine come with shrouds while the last two stages are free standing (Liang 1993, Jung *et al.* 2001, Rahmati 2009).

Figure 1 shows the turbine-generator side drawing of a nuclear power plant in Taiwan, of which the total length is 49.8 m. To facilitate observation, its vertical scale is prolonged; further, the details of the last two blades of the LP turbine are also shown. ZW in Figure 1 stands for coupling. Figure 2 shows the mechanical system model of the 985 MW set in the second nuclear power plant, in which there are corresponding momentums for each stage of turbine, generator, exciter, and the last two stages of long blades on the LP rotor that is to be analyzed (Jatskevich *et al.* 2006, Chen and Lee 2008). The torsional model for the shaft consists of a series of cylinders representing momentums. Between two cylinders are the rigid body and mutual damper. The shaft itself has damping; extended from the shaft are the representing momentums, rigidity, and damping for the last two stages of blades. Steam propels the blades of the turbine and outputs mechanical energy to be converted to electrical energy by the generator.

Figure 3 shows the electrical system model used in the accident analysis. When faults occur in one of the power lines, the breakers on the two terminals can jump at the preset time. There is an option to choose recovery automatically.

2.2. *FEM software*

ANSYS is versatile finite element analysis software capable of analyzing problems in mechanics and electromagnetism. It has the capacity to carry out

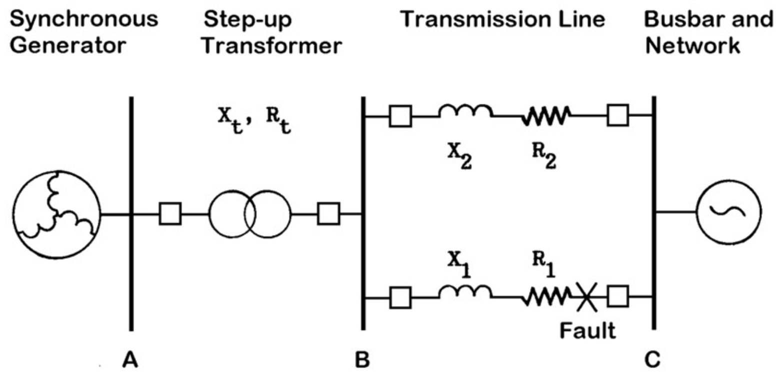

Figure 3. The electrical system model used in the accident analysis.

ordinary linear, static, dynamic, thermal-dynamic, flow-dynamic, and optimization analysis, particularly in the problems of buckling, fatigue, creep, and breakage. Here we used ANSYS/ED 10.0 for PC. Figure 4 shows the flowchart for the dynamic simulation program incorporating ANSYS FEM software.

2.3. *Analysis of PSD*

Fourier analysis techniques have been extensively applied in research fields like linear systems and stochastic processes, and can be implemented in a computer program. For problems which cannot be resolved by continuous Fourier transform, discrete Fourier transform is a viable alternative. It is to be noted that if there are N time-domain data, the computation time will be proportional to N^2, which literally prohibits large time domain data.

In 1965, Cooley and Tukey introduced fast Fourier transform (FFT), of which the computation times are proportional to $N \log_2 N$. The improvement on the computation speed dramatically changes scientific analysis. In this article, PSD analysis, one of the techniques for analysis which resulted from this improvement, transforms the time domain simulation response obtained from the turbine-generator dynamic programs to frequency domain data. This makes the assessment of vibration much easier.

The total power in a torsional vibration is the same whether it is calculated in the time domain or in the frequency domain. The result is known as Parseval's theorem:

$$\text{Total power} \equiv \int_{-\infty}^{+\infty} |h(t)|^2 \mathrm{d}t = \int_{-\infty}^{+\infty} |H(f)|^2 \mathrm{d}f. \quad (1)$$

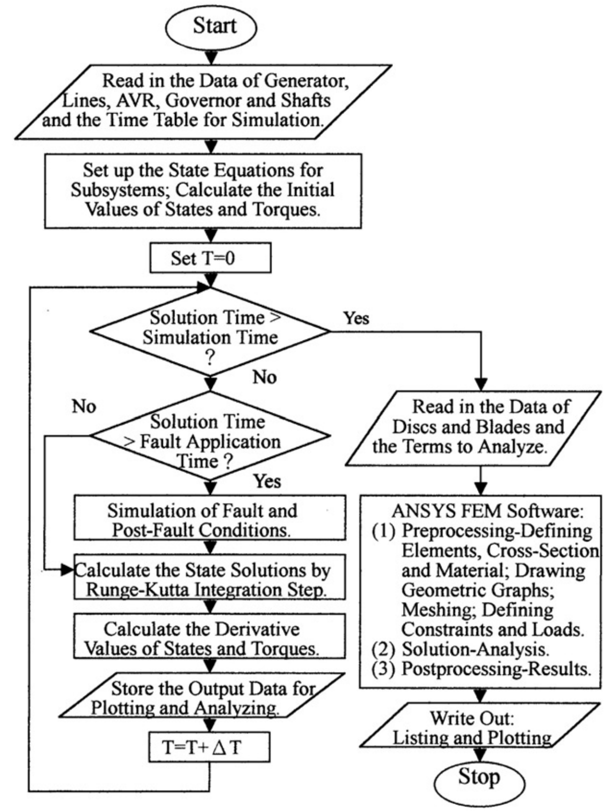

Figure 4. Flowchart of turbine-generator dynamic analysis.

Suppose that the functions of shaft torque $S(t)$ are samples at N points to produce values $S_0 \cdots S_{n-1}$, and these points span a range of time T, $T = (N-1)\Delta$, where Δ is the sampling interval, then the mean-squared amplitude can be written as

$$\frac{1}{T}\int_0^T |S(t)|^2 \mathrm{d}t \approx \frac{1}{N}\sum_{j=0}^{N-1} |S_j|^2. \quad (2)$$

Owing to the irregular nature of the derived torque transient, many apparently random oscillations are perceived so that an averaging procedure, to establish gross characteristics, is required. The frequency composition of the torque function may be described in terms of the spectral density of mean-square values. In this study, the Welch windows are defined by

$$w_j = 1 - \left(\frac{j - \frac{1}{2}(N-1)}{\frac{1}{2}(N+1)} \right)^2, \tag{3}$$

where N is the number of points used, incorporated into the analysis.

2.4. *Eigenvalue analysis*

N-second-order differential equations are used to simulate the mechanical systems of a turbine generator (Martinez 1993, Martinez *et al.* 2005). Or equivalently, linear dynamic systems with n degrees-of-freedom can be expressed in the matrix form as follows:

$$[\boldsymbol{J}]_{n\times n}\{\boldsymbol{\theta}''\}_{n\times 1} + [\boldsymbol{D}]_{n\times n}\{\theta'\}_{n\times 1} + [\boldsymbol{K}]_{n\times n}\{\boldsymbol{\theta}\}_{n\times 1} = \{\boldsymbol{S}(\boldsymbol{t})\}_{n\times 1}. \tag{4}$$

By using the Duncan method and transforming n-second-order differential equations to $2n$-first-order differential equations via auxiliary variable transformation, Equation (4) can be rewritten as

$$[\boldsymbol{M}]_{2n\times 2n}\{\boldsymbol{Y}'\}_{2n\times 1} + [\boldsymbol{N}]_{2n\times 2n}\{\boldsymbol{Y}\}_{2n\times 1} = \{\boldsymbol{Q}(\boldsymbol{t})\}_{2n\times 1}, \tag{5}$$

where

$$[\boldsymbol{M}] = \begin{bmatrix} [0] & [\boldsymbol{J}] \\ [\boldsymbol{J}] & [\boldsymbol{D}] \end{bmatrix}, \quad [\boldsymbol{N}] = \begin{bmatrix} -[\boldsymbol{J}] & [0] \\ [0] & [\boldsymbol{K}] \end{bmatrix},$$
$$\{\boldsymbol{Y}\} = \begin{bmatrix} \{\boldsymbol{\theta}'\} \\ \{\boldsymbol{\theta}\} \end{bmatrix}, \quad \{\boldsymbol{Q}(\boldsymbol{t})\} = \begin{bmatrix} \{0\} \\ \{\boldsymbol{S}(\boldsymbol{t})\} \end{bmatrix}.$$

If the system vibration is free and matrices $[\boldsymbol{J}]$, $[\boldsymbol{D}]$, and $[\boldsymbol{K}]$ are symmetric, a solution can be found through linear transformation

$$\{\boldsymbol{Y}'\}_{2n\times 1} - [\boldsymbol{H}]_{2n\times 2n}\{\boldsymbol{Y}\}_{2n\times 1} = \{0\}_{2n\times 1}, \tag{6}$$

where

$$[\boldsymbol{H}] = -[\boldsymbol{M}]^{-1}\cdot[\boldsymbol{N}] = -\begin{bmatrix} [0] & [\boldsymbol{J}] \\ [\boldsymbol{J}] & [\boldsymbol{D}] \end{bmatrix}^{-1} \begin{bmatrix} -[\boldsymbol{J}] & [0] \\ [0] & [\boldsymbol{K}] \end{bmatrix}.$$

Eigenvalue λ can be found by solving

$$\left|\lambda[\boldsymbol{I}]_{2n\times 2n} - [\boldsymbol{H}]_{2n\times 2n}\right| = [0]_{2n\times 2n}. \tag{7}$$

We can thus obtain $2n$ eigenvalues or $2n$ vibration modes. The real part of eigenvalue corresponds to damping; the imaginary part is the angular velocity ω.

3. Simulations and analysis

Two 985 MW machines in the second nuclear power plant of Tai-power are the largest turbine-generators in Taiwan. Due to the inherent constraints of materials, the component vibration of large generators is particularly worthy of attention. G.E. produced the 2894 MW nuclear steam system used in the second nuclear power plant. The turbines and the 985 MW, 22 kV generator were produced by Westinghouse. Nine sectors of the shaft system are arranged in the following sequence: HP1-HP2-LP1F-LP1R-LP2F-LP2R-GEN-REC-EXC. The mechanical eigenvalues are given in Table 1. Note that there are eight pairs of torsional modes. Three single-phase 350 MVA step-up transformers send electric power to the main bus through dual loop steel core aluminum cables. Various parameters are transformed into per-unit values. The detailed derivation of this technique can be found in the works of Wang and Hsu (1988) and Wang *et al.* (1994). In this article, we consider the most realistic scenario to perform the simulation and obtain the following results.

Four types of conditions are considered in our simulation of power faults. Those are single-line-to-ground, line-to-line, double-line-to-ground, and three-phase-to-ground. Due to the size consideration of this article, only a part of the results for the analysis of the 985 MW set are shown. Figure 5 simulates the three-phase-to-ground fault. The scenario is that an accident occurs at $t = 0.1$ s, the breaker is activated at $t = 0.2$ s and the circuit with accident is disconnected and no longer closed. Frequency responses of two shaft sectors: HP2-LP1F and LP2R-GEN were obtained conducted. We used power density analysis and noticed that in the data from $t = 0.2$ s to $t = 4.295$ s, considerable peaks are observed. Apparently, mechanical vibration modes are the major constituents of the data. This is further confirmed by observing the first six vibration mode frequencies shown in the figure.

The PSD analysis of the span between 0.2 and 4.295 s, gives the peak values at each torsional mode exactly. The natural frequencies coincide with the PSD peak values precisely. The data from the shaft torque simulation is sampled at approximately 1000 samples per second. To avoid large variances per data point, a total number of 4096 sample points have been selected for the PSD analysis. For the time period from 0.2 to 4.295 s, 2048 points may be obtained from the

Table 1. Mechanical mode eigenvalues of 985 MW turbine-generator.

Modes	Eigenvalues	Frequency (Hz)
Mode 1	$-0.018 \pm j\ 45.01$	7.16
Mode 2	$-0.294 \pm j\ 86.52$	13.76
Mode 3	$-0.453 \pm j\ 89.65$	14.27
Mode 4	$-0.042 \pm j\ 102.35$	16.28
Mode 5	$-0.016 \pm j\ 235.82$	37.53
Mode 6	$-0.016 \pm j\ 243.14$	38.69
Mode 7	$-0.050 \pm j\ 546.10$	86.91
Mode 8	$-5.284 \pm j\ 645.43$	102.72

shaft torque oscillation output. These points give the PSD output at frequency intervals of 0.4883 Hz and total frequency up to 500 Hz.

There are six phases in the analysis using ANSYS FEM software:

(1) Defining the elements, cross-section, and material – these elements include 0-D particle and crack, 1-D beam and pillar, 2-D shell and board, and 3-D cylinder and cube.
(2) Drawing geometric graphs.

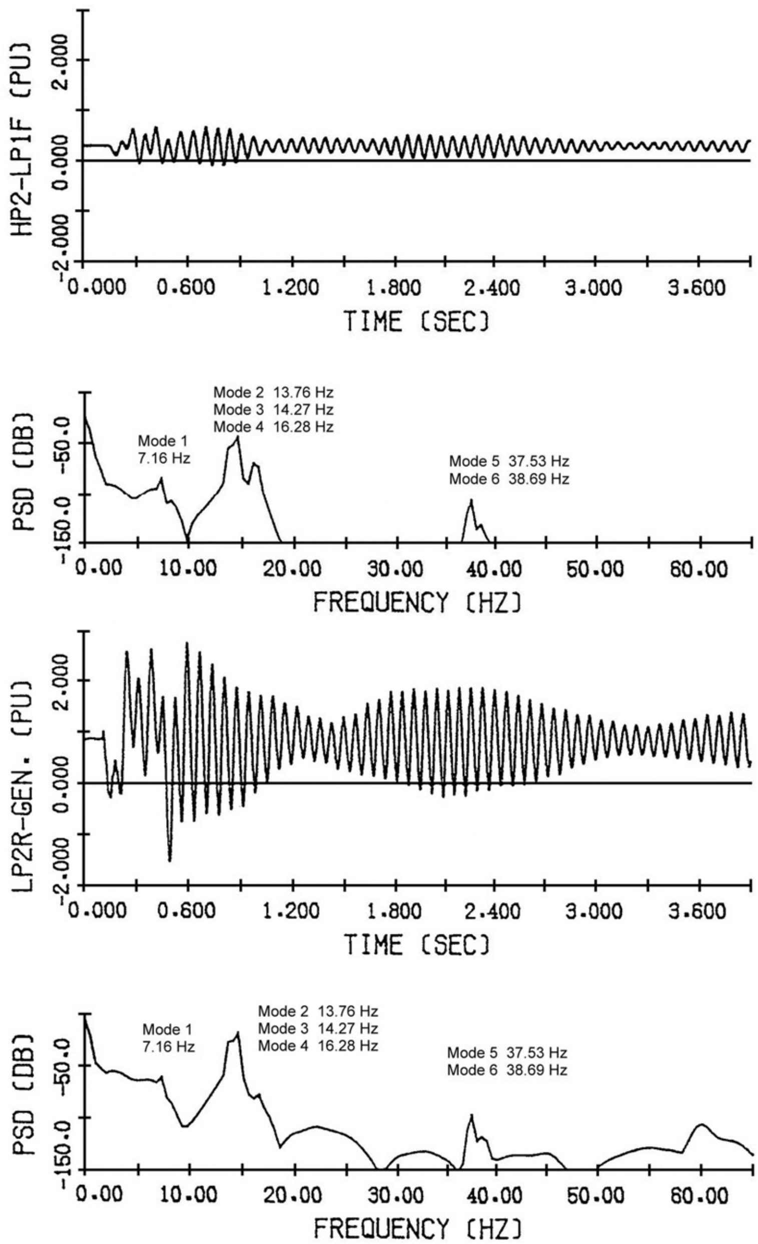

Figure 5. Simulation and PSD analyses.

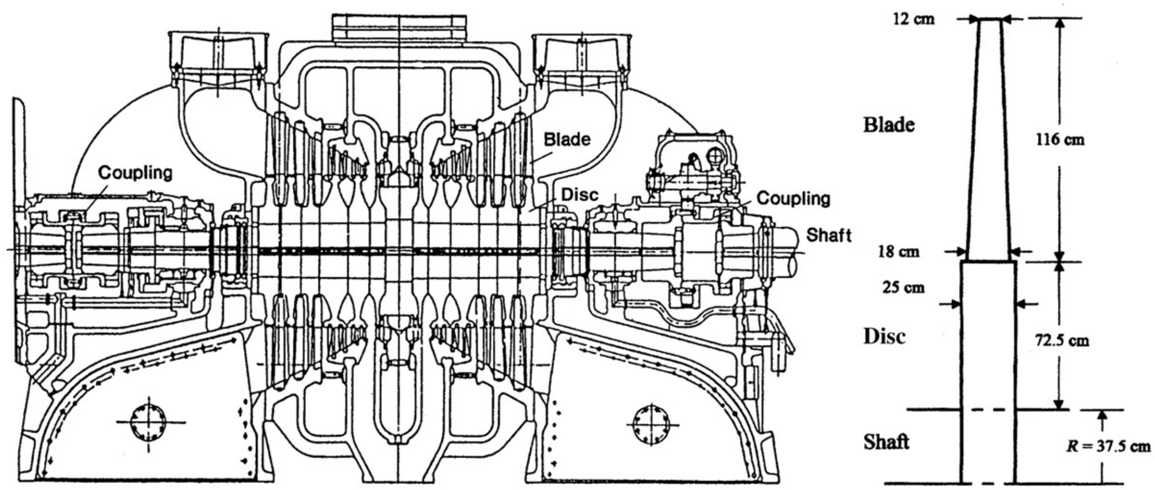

Figure 6. LP turbine and component which have undergone FEM analysis.

(3) Meshing – the object to be analyzed is further divided into meshes. The methods of meshing include direct meshing, self-meshing, and adaptive meshing.
(4) Defining constraints and load – constraints are sometimes referred to as boundary conditions. Load includes forces of concentration and distribution, acceleration, and or preset strain.
(5) Analysis – analysis includes pre-processing and post-processing. It offers information on strain, stress displacement, etc.
(6) Displaying results.

Figure 6 shows the shrunk-on LP turbine with discs. An analysis is performed on this turbine to demonstrate FEM simulation. The diameter of the axis is 0.375 m; the last blade is 46 in. in length and placed constraints on a disc with a 2.2 m diameter. There are 80 blades in the last stage, Figure 7 shows the θ-axis has 80 elements. Owing to the insufficient data, we adopted the simplified model shown next to Figure 8 for analysis. The average thickness of the disc is assumed to be 25 cm; the average width for a blade is 15 cm. Components are made of AISI 304 Chrome Steel.

Figure 7 shows the mesh of the 2-D model of the disc placed on the shaft. The 985 MW generator has four poles. Figure 8 shows the 2-D model analysis of the disc strain at the synchronous frequency $\omega = 188.5$ rad/s with 2.5 mm eccentric magnitude. This is approximately 10 times the alarming value for torsional response.

Figure 9 analyzes the local disc strain with a 3-D model. The PSD value of the torque obtained from the dynamic program gives the load conditions. Figure 10 analyzes the strain on discs with adjacent single-blades in the tail end. There are 80 freestanding blades in that stage. The analysis shown divides the x-axis into 20 elements; the z-axis is also partitioned into 5 elements. The θ-axis has 80 elements. By using a symmetric-axis model, a thorough strain analysis can be done as shown in Figure 11.

The x-axis of the last stage blade of the turbine is divided into five elements; the y-axis is partitioned into 30 elements of 2 cm thickness each. According to our simulation, the centrifugal force and shear stress can be uniformly distributed in each node. Figure 12 shows an example of stress analysis; Figure 13 shows impact analysis. In our simulation, both direct analysis and characteristic model analysis can be chosen. If direct analysis is chosen, and $F_Z = 75$ N is applied; Figure 13 shows the response on the top of the blade. Figure 14 shows the situation when Media Player is used for observation.

4. Conclusions

In the quest for energy efficiency and low-unit price, the capacity of turbine-generators has been consistently increasing. However, due to the constraints on space and material strength of the structures, no design with high-safety factor has been adopted for these high-capacity machines. As a result, cracks frequently develop in discs and blades of LP turbines that are subject to constant stress. The impact of metal fatigue is even more worth noting when electrical power accidents result in instantaneous and strong extra stress, or when persistent excitation is introduced by electric power

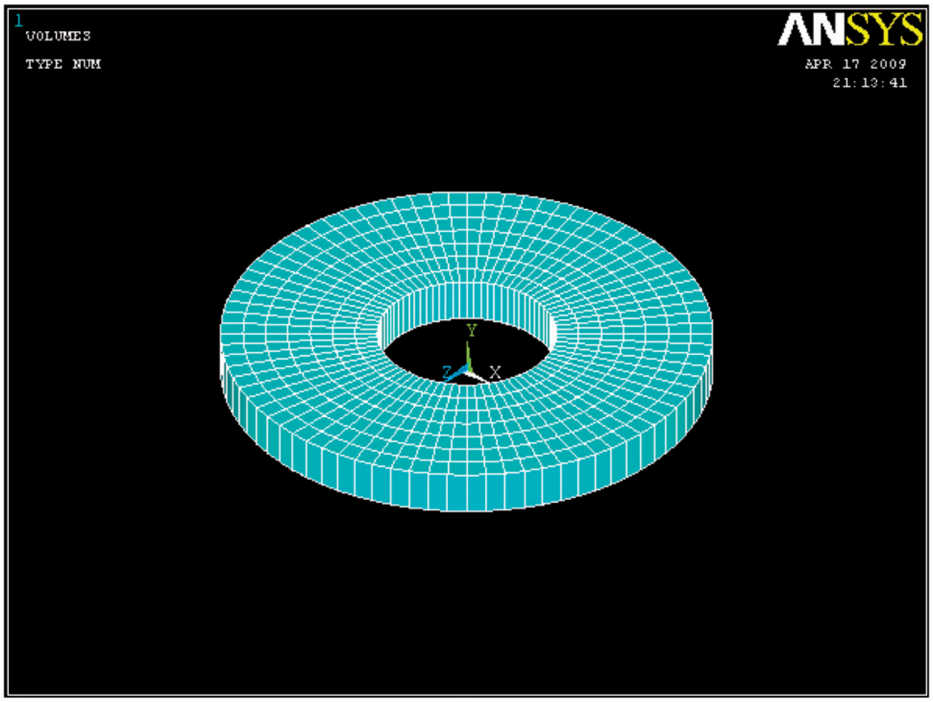

Figure 7. The mesh and constraints of the last disc of the LP turbine.

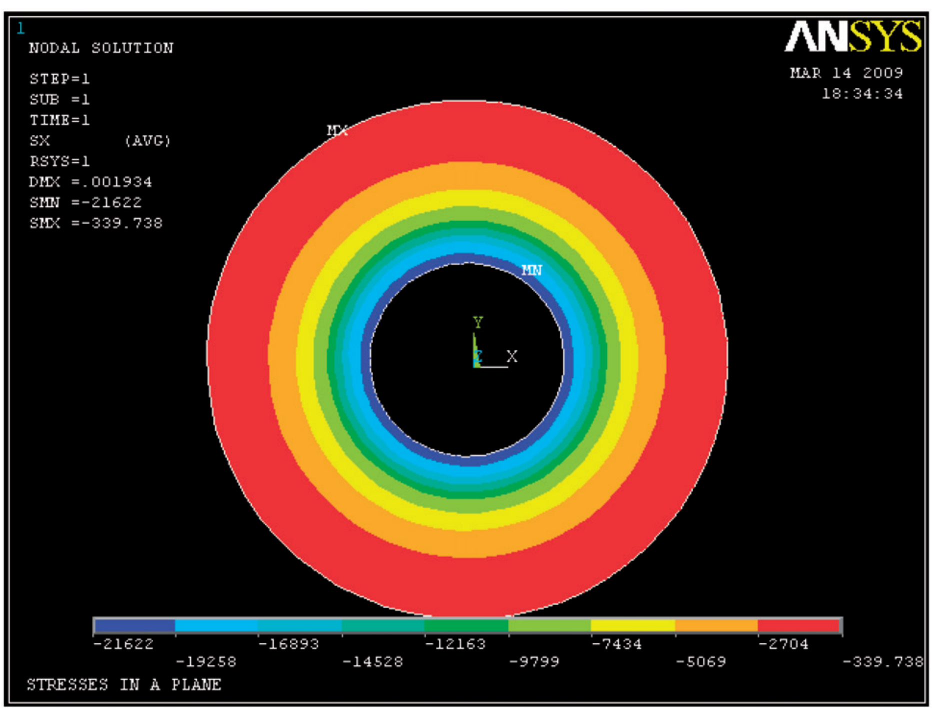

Figure 8. Analyzing the last stage disc strain with FEM.

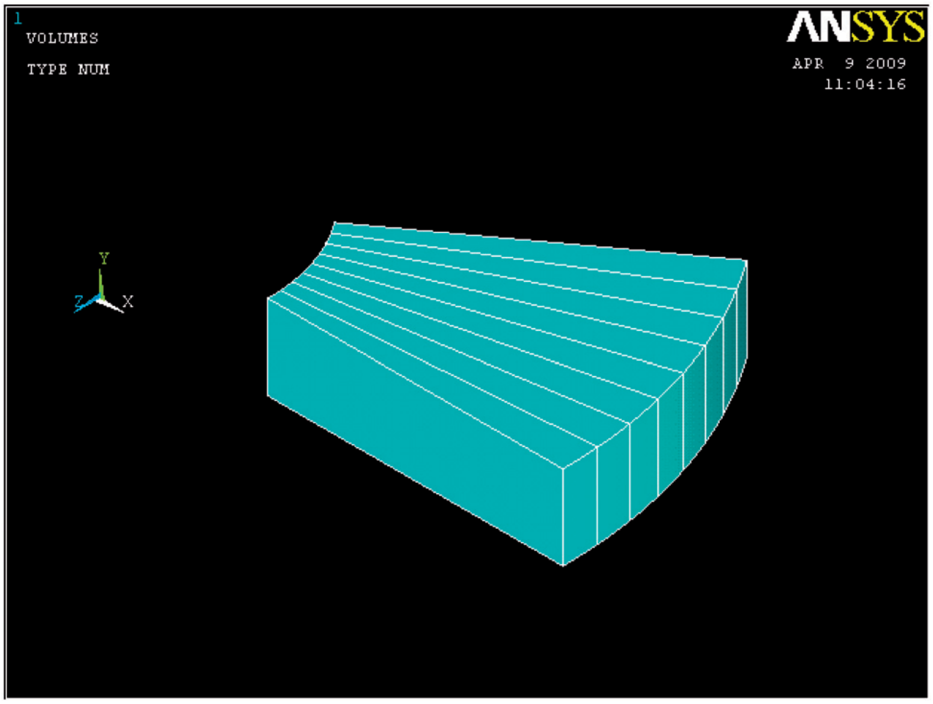

Figure 9. Analyzing the disc strain in a 3-D model with FEM.

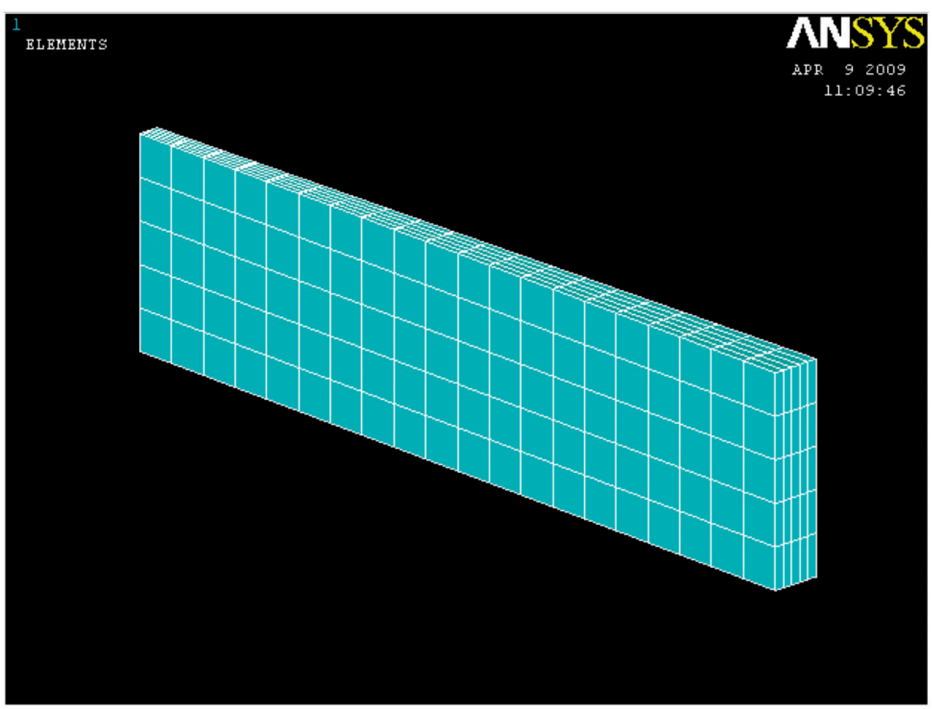

Figure 10. Analyzing the disc and blade root strain with FEM.

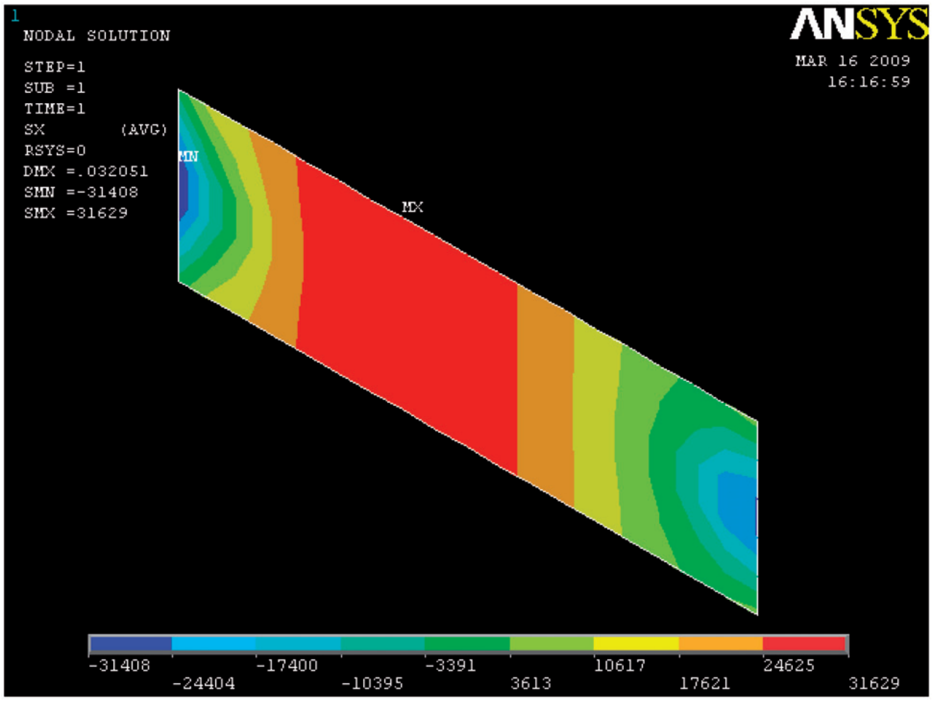

Figure 11. Analyzing the symmetric disc model with FEM.

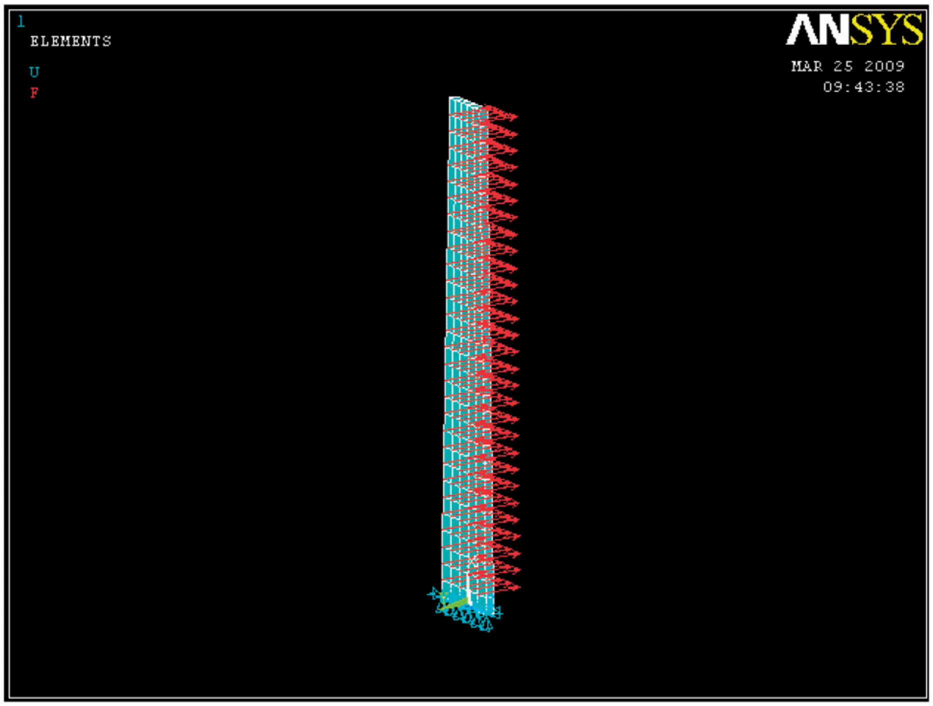

Figure 12. The shear stress and centrifugal stress on the blade.

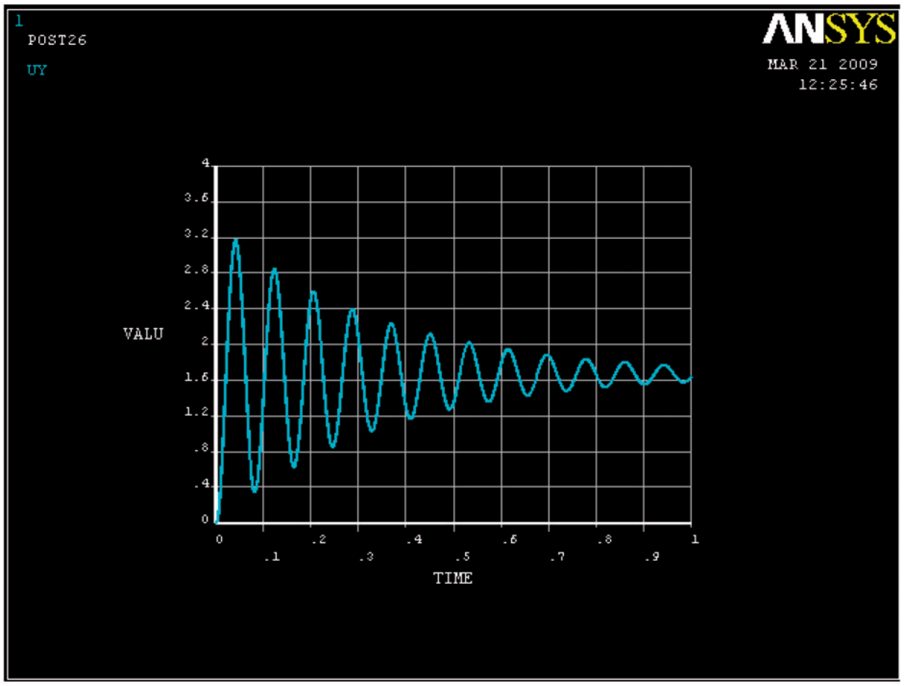

Figure 13. Analyzing the transient response of blades with FEM.

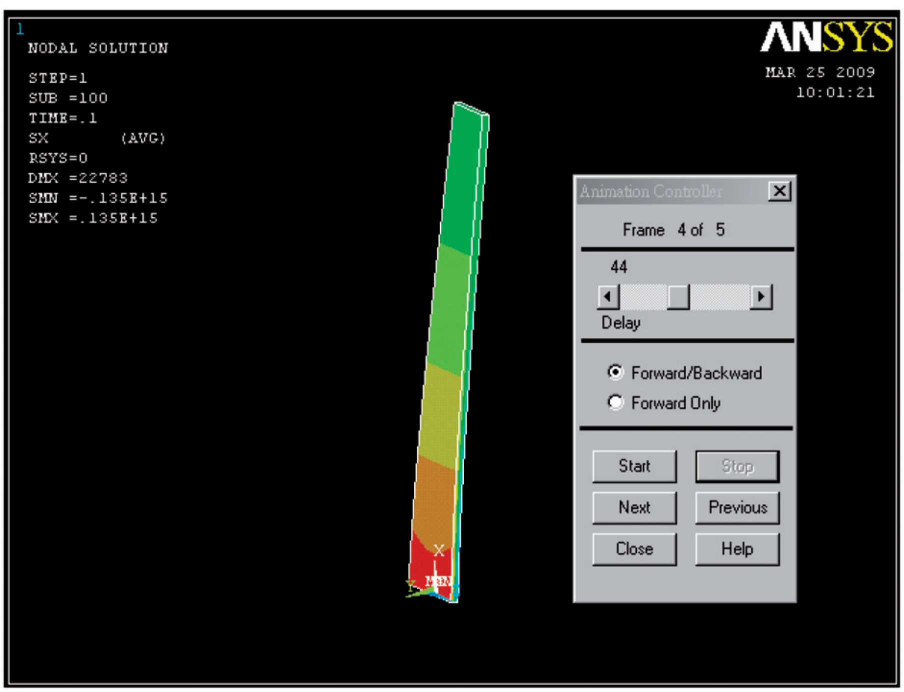

Figure 14. Observing the blade response with Media Player.

harmonics. The traditional models used in the simulation for discs and blades of turbine generators are too simple. FEM software is superior in its more thorough description of the vibration modes and stress on the turbine-generators; however, it is not capable of including power system disturbance in the analysis.

In this article, we focus on a large-scale turbine-generator of the Tai-power Company, currently in service, and perform an analysis of shafts, discs, and blades, considering both regular operation conditions and disturbance while using the results of simulation with lumped-parameter models and incorporating PSD to assist the analysis of the vibration spectrum to give the boundary and load conditions for FEM. Eigenvalue analysis is further utilized to reaffirm our results. This information can be incorporated in analysis using ANSYS FEM software toward a more reliable dynamic response. Because of insufficient data, a very simplified model is used in this article. Nevertheless, it still successfully demonstrates the feasibility and superiority of combining the advantages of those two programs.

Nomenclature

D, J, K	matrices consisting of damping, inertia, stiffness
$H(f)$	torque produced value of frequency f
$h(t)$	torque produced value at time t
N	total number of shaft torque produced values
$S(t)$	shaft torque at time t
T	time range for study
w_j	jth Welch window
θ, θ', θ'',	matrices consisting of angular displacement, angular velocity, and angular acceleration
λ	eigenvalue

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Appendix – parameters of the 985 MW machine

Table A1. Data of mechanical system.

Mass/Torque (%)	Inertia (M) (MW-s/ MVA)	Damping (C) (MW-s/MVA-rad)	Stiffness (K) (MW/MVA-rad)
HP1/17.50	0.08435	0.00009	129.648
HP2/17.50	0.08504	0.00009	16.824
LP1F/16.25	0.6546	0.00022	186.156
LP1R/16.25	0.6489	0.00022	24.069
LP2F/16.25	0.6518	0.00022	184.989
LP2R/16.25	0.6648	0.00022	37.966
GEN	0.8218	0.00012	0.3655
REC	0.00018	0.00002	0.3919
EXC	0.00223	0.00003	

Table A2. Data of electric system.

Synchronous generator (rating: 1095.1 MVA, 22 kV, 60 Hz, 4-pole)
$X_d = 1.55$ $X_{kd} = 0.102$ $X_l = 0.228$ $R_{kd} = 0.0098$ $R_a = 0.0020$
$X_q = 1.51$ $X_{kq} = 0.0576$ $X_{fl} = 0.132$ $R_{kq} = 0.0122$ $R_f = 0.00064$

Step-up transformer (rating: 1095.1 MVA, 22/345 kV)
$X_t = 0.150$ $R_t = 0.00219$

Initial operating conditions
$P_o = 0.86$ $Q_o = 0.12$ $V_t = 1.0$

Transmission line (rating: 1095.1 MVA, 345 kV)
$X_{1A} = X_{1B} = X_{1C} = 0.107$ $R_{1A} = R_{1B} = R_{1C} = 0.0071$
$X_{2A} = X_{2B} = X_{2C} = 0.107$ $R_{2A} = R_{2B} = R_{2C} = 0.0071$

Table A2. Data of control system.

	Gain	Lead	Lag
AVR (IEEE no. 1 Model)			
AVR data			
AVR block 1	1.000	0.0010	0.0010
AVR block 2	50.00	0.0000	0.0200
AVR block 3	−34.483	0.0000	−4.0300
AVR block 4	1.000	0.0460	0.5780
AVR limit (positive and negative)	1.000	−0.950	
Threshold volt (1, 2)	3.7960	5.0620	
SE1, SE2, SE3	0.000	0.0750	0.2230
Governor (IEEE no. 1 Model)			
Governor data			
Governor block 1	18.000	0.000	0.001
Governor block 2	1.000	0.001	0.020
Governor block 3	1.000	0.000	0.700
Governor block 4	0.650	0.000	3.000
Governor limit (positive and negative)	1.200	0.000	