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THE TRANSIENT TORSIONAL VIBRATION BEHAVIOUR OF A TURBINE-GENERATOR SYSTEM UNDER SHORT CIRCUIT EXCITATION

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ABSTRACT

Turbine-generator systems are subjected to sudden short circuits. The sudden change of electrical characteristics in these systems can incur very high excitation torques on the components. The excitation depends on the types of short circuits and the electrical properties of the generator, etc. Transient torsional vibration due to generator short circuit is investigated in this paper. Typical short circuit excitation functions are discussed. A modern gas turbine / generator / steam turbine system is used to illustrate the effects of modelling inaccuracies and parameter variations. The acceptance standards for such transient short circuit conditions are generally based on maximum torque transmitted or some kind of allowable shear stress. They cannot be accurately predicted by models which use 1-2 disks(stations) to represent a major machine in a train. Either a more refined model or a better reduction method is needed. Flexible couplings affect the transient torsional response and they must be included in the system vibration analysis. Their influence on the peak transmitted torque has been examined. The selection of couplings for turbine-generator systems in modern combined cycle power plants should be emphasized.

NOMENCLATURE

[C]	Damping matrix
f_k	Frequency component k
[K]	Stiffness matrix
[M]	Mass(Inertia) matrix
t	Time
T(t)	Transient torque
T_0	Nominal torque
T_k	Torque amplitude for frequency component k
Ω_N	Net frequency
θ	Angular displacement
τ_k	Time constant for frequency component k
ϕ_k	Phase shift for frequency component k

1. INTRODUCTION

During operation in the field, turbine-generator systems are subject to the risk of short circuits occurring in the electrical system. The sudden change of the electrical characteristics in the system can incur very high excitation torques on the components. The excitation depends on many factors such as the types of short circuits (whether it is a 2-phase or 3-phase fault) and the electrical properties of the generator, etc. Nevertheless, the amplitudes of the excitation torques can be considerably higher than the normal operating torque, maybe as large as 12 to 13 times. Thus, it poses a serious threat to the safe operation of turbine-generator systems.

The exact function of how such short circuit excitation torques manifest themselves is system dependent. In most practical cases, the torques are modelled as sinusoidal fluctuations with an exponential decay of their amplitude envelope process. Their maximum amplitudes and the time constant of the decay process are determined by the electrical characteristics of the generator (or motor in the case of electric motor driven systems). Some empirical formulas are provided by the manufacturers. They are mostly based on previous experience of such systems.

A typical modern gas turbine / generator / steam turbine system has been employed for a parametric study of the short circuit response behaviour. The objectives of this study are three-folded:
(1) to investigate the transient torsional vibration of modern turbine-generator systems and compare by a case study different modelling approaches to determine system response;
(2) to study the torsional vibration responses of the system to diverse short circuit excitation functions and the effect of variation in excitation parameters;
(3) to identify important characteristics of the system which most strongly influence on the torsional behaviour in order to provide guidelines for selecting the right component dynamic properties and for optimizing the performance.

The study involves the analysis of the transient response of the turbine-generator system. It is well known that transient analysis incurs large computational effort and resources. Engineering

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judgement and suitable approximations must be used to obtain reasonable and accurate results.

Besides providing the above information, the study reveals the significance of component selection. Especially, the couplings for torque transmission in the system are often critical for safe and reliable operation. We need an accurate evaluation of the possible excitation torques and a thorough understanding of the dynamics of the system so as to ensure the correct dimensioning of these critical components.

2. THEORY

A torsional analysis consists usually of two types of calculations:
- Calculation of natural frequencies and mode shapes
- Calculation of shaft angular displacement amplitudes and stresses, and vibratory torque values.

When systems include an electric motor or generator, a transient analysis may be necessary to determine the motion response (angular displacement, velocity or acceleration) amplitudes and dynamic torques. There are several different causes which can induce transient torsional vibrations in these systems. For instance, transient vibrations due to the start-up of a machine e.g. compressor driven by a synchronous motor [Bogacz et al.(1989)], vibrations of a steam turbogenerator rotor shaft system due to high speed reclosing of the electric network [Bogacz et al.(1992)], turbogenerator sets in operation may be excited to transient torsional vibrations by dynamic electrical moments at the generator due to short-circuits or faulty synchronization [Schwibinger et al.(1989)].

For the solution of the torsional vibration problem, a torsional mass elastic model of the system is needed. There are various methods for building such a model. The simplest way is to represent each part of the system as equivalent torsional springs (stiffnesses) and masses (inertias) and then assemble them together by connecting the appropriate masses and springs. The equivalent inertia and stiffness values are usually calculated from the system drawings and supplied dynamic data. For simple systems the natural frequencies can be calculated by using the well known Holzer method. With the advent of computers, analysis is now often carried out using matrix computer programs which can deal with a large number of masses and complex gearing and multiple prime movers.

Another common approach is to model the torsional system by the finite element method which normally results in a very accurate mechanical model with many degrees of freedom (DOFs). Many general finite element analysis (FEA) programs/packages which can handle this kind of analysis are available. Some of these FEA programs provide elements tailored for torsional analysis such as point mass with rotary inertia and torsional spring. In principle, both the finite element method and the Holzer method are solving the following equation of motion:

$$[K]\theta + [C]\dot{\theta} + [M]\ddot{\theta} = T(t) \tag{1}$$

The damping is normally small and has little effect on the natural frequencies of torsional systems. Thus, damping is often neglected when the torsional natural frequencies and mode shapes are calculated.

The calculated frequencies can be conveniently illustrated on a Campbell diagram [Boyce(1982)]. The diagram shows frequency against speed for the various orders (harmonics) and the critical speeds are identified by the intersections of natural and exciting frequencies in the operating speed range. In contrast with lateral vibration problems, the torsional natural frequencies are usually independent of speed and thus they are represented by horizontal lines in the Campbell diagram. The excitation sources and their frequencies are system dependent. For turbomachinery systems, the main excitation sources often have frequencies corresponding to 1, 2 or 3 times the rotating speeds of the respective system components.

The torsional mode shapes show the relative angular displacement along the length of the shaft string since no absolute values are obtained from a free response analysis. But the mode shapes are not giving directly the twist i.e. the difference between angular displacements at one axial position and at another axial position.

Generally speaking, turbomachine rotors are rather stiff in torsion individually. However, when turbomachines are connected together by shaft couplings or gearboxes etc, the torsional stiffnesses of the couplings are often low enough to bring torsional natural frequencies into the range of excitation frequencies. Thus, couplings are one of the critical elements in a turbomachinery system which can significantly influence the rotordynamic response. Selection of the right type and size of couplings is an important topic in its own right. If rotordynamic (including torsional) calculations indicate that the system is not satisfactory and therefore requires modification, then very often a change of coupling size and type is the most probable course of action to be taken.

In most practical cases, the transient excitation torques due to sudden short circuits are modelled as sinusoidal fluctuations with an exponential decay of their amplitude envelope process. The maximum amplitudes and the time decay constants are system-dependent. Examples of some formulas for specific machines are listed below.

<u>Example 1:</u>
Excitation functions for transient motor loads during short circuit of electric motor running at 30Hz according to manufacturer's data:

- 2 pole short-circuit

$$T(t)/T_0 = 5.0 \times e^{-15t} \times \sin(2\pi \cdot 60t) - 2.5 \times e^{-15t} \times \sin(2\pi \cdot 120t) \tag{2a}$$

- 3 pole short circuit

$$T(t)/T_0 = 5.0 \times e^{-15t} \times \sin(2\pi \cdot 60t) \tag{2b}$$

<u>Example 2:</u>
Excitation functions for transient torques in air gap for a three-phase induction motor running at 55.5Hz according to another manufacturer's data:

- 2 pole short-circuit

$$T(t)/T_0 = -14.089 \times e^{-21.4t} \times \sin(2\pi \cdot 55.5t) + 7.044 \times e^{-13.2t} \times \sin(2\pi \cdot 111t) \tag{3a}$$

- 3 pole short circuit

$$T(t)/T_0 = 14.173 \times e^{-28.3t} \times \sin(2\pi \cdot (55.3t + 175.4/360)) \tag{3b}$$

Example 3:

Excitation function for transient short circuit torque based on DIN 4024 Part 1:

- 2 pole short circuit

$$T(t)/T_0 = 10 \times e^{-t/0.4} \times \sin(\Omega_N t) - 5 \times e^{-t/0.4} \times \sin(2\Omega_N t) - (1 - e^{-t/0.15}) \tag{4}$$

where Ω_N is the net frequency,
$T(t)$ is the transient torque,
T_0 is the nominal torque.

To sum up, the generalized excitation function can be described in the following form:

$$\frac{T(t)}{T_0} = \sum_k \frac{T_k}{T_0} \times e^{-t/\tau_k} \times \sin(2\pi \cdot f_k t + \phi_k) \tag{5}$$

The above formulas are mostly based on previous experience of such systems. It is obvious that there are large variations of the maximum torque values and time decay constants which can only be attributed to factors dependent on the electrical characteristics.

Since the short circuit (and start-up) excitation torques are time-dependent events, the analysis of the torsional response will involve transient vibration calculations. For general cases, the transient analysis has to be carried out by numerical integration of the differential equations of motion. Initial conditions for the variables θ and $\dot{\theta}$ must be known or assumed. The calculation is usually executed on a computer and various numerical integration schemes may be applied. Numerical integration is a versatile method for transient (and nonlinear) dynamic analysis. However, the method suffers from the following disadvantages:
(1) it incurs large computational effort and resources;
(2) numerical quantization errors and truncation errors can occur;
(3) the solutions obtained are specific to a given set of initial conditions.

Therefore engineering judgement is required to limit the scope of the analysis parameters and variables, and suitable approximations should be used to obtain acceptable results under the time and resource limitations. As an example, a method which allows finding an accurate reduced torsional model with discrete masses and springs from a finite element model with many DOFs has been proposed by Schwibinger et al. (1989).

3. CASE STUDY DESCRIPTION

The case study is based on a turbine-generator system in a combined cycle power plant which consists of a gas turbine and a steam turbine both connected to the same generator. This system will be used for a parametric study of the short circuit response behaviour. Two torsional models using different modelling approaches and number of DOFs are employed to represent the system. The first one is a simple discrete model of only eight masses with springs connecting them together. It is used to compare the torsional vibration responses of the system under excitations specified by the formulas as shown in the theory section, and study the effect of variation in excitation parameters. The second is a refined finite element model which should represent the dynamic behaviour to a high accuracy. The objective of analyzing this model is to identify the important properties or components of the system train which most strongly influence on the torsional behaviour. The case study is conducted by using the FEA program ANSYS.

Discrete mass/spring model

A schematic diagram of the simple discrete model is shown in Fig. 1. The system train is composed of a gas turbine on the left side, connected by coupling and gears to the generator in the middle, and further connected through clutch, gears and coupling to the steam turbine on the right side. The system is modelled by discrete masses (moments of inertia) and springs (stiffnesses) and the system parameters are listed in Table 1. It should be noted that the parameters are normalized to the reference speed of the generator, i.e. all the gear (speed) ratio effects have been incorporated.

Station	Moment of inertia ($kg\ m^2$)	Span	Torsional stiffness (Nm/rad)
1	3012	1	1.40E7
2	8	2	1.33E7
3	362	3	8.35E5
4	2276	4	5.20E5
5	2	5	2.36E7
6	66	6	1.50E8
7	2	7	1.33E7
8	355		

Table 1 Parameters of the discrete torsional system in Fig. 1

The system response is investigated by using the generalized formula for short circuit excitation (Eqn. 5) with parameter values suggested in DIN 4024 (Eqn. 4). A time series plot of the excitation function is shown in Fig. 2 to give us an idea of the physical meaning of the respective terms. Obviously, the sinusoidal terms represent time varying functions of the excitation at the respective frequencies Ω_N and $2\Omega_N$. The exponential term $e^{-t/0.4}$ indicates the decay of the excitation and the value 0.4 can be interpreted as the time constant corresponding to the rate of decay. As shown in the plot, the peak excitation torque can be as high as 12.5 times the nominal torque which is 200kNm in this case. This peak excitation torque is usually provided by the manufacturer's data but considerable variation exists among different machines. Nevertheless, the high amplitudes of such excitation torques can give rise to torsional vibration problems if the dynamic behaviour is not carefully tuned.

Let us investigate the response of this system to the excitation. That is, to look more closely at the effects of changing the excitation parameters, i.e. the phase of the excitation and the time decay constant, as well as the torsional damping in the system.

The following scenarios will be calculated by taking the value of nominal torque T_0 as 200kNm and net frequency Ω_N as 50Hz (3000 rpm):

Scenario	Phase (deg)	Time constant (s)	Damping (%)
0 (Base case)	0	0.4	1.67
1	90	0.4	1.67
2	0	0.8	1.67
3	0	0.4	0.83

Table 2 Input data for the different scenarios of investigation

Results are obtained by numerical integration and shown in the form of plots of response vibratory torques at station 2 and station 7. The maximum torque amplitudes are required for the proper selection of couplings which can tolerate such peak torques.

Figs. 3 to 6 show the response plots corresponding to the scenarios 0 to 3 respectively.

Refined finite element model

A schematic diagram of the refined finite element model of the whole system train is shown in Fig. 7. The model is composed of circular beam elements and concentrated mass elements are added to represent the inertial effect of rigid discs/wheels on the shaft. Such a model contains a considerably larger number of DOFs than the simple discrete model and thus requires substantially longer time for modelling and analysis. On the other hand, this refined model can accurately represent the dynamic behaviour of the individual system components. For instance, the variation of angular response within the turbines and generator is obtained, which is not possible with the simple model. Thus, stresses in the shaft string due to torsional vibration can be determined with this refined model.

Results are calculated for the scenarios 0 and 3 as given in Table 2. Response vibratory torques at the stations 90 and 151 corresponding to respectively stations 2 and 7 in the discrete model are plotted in Figs. 8 to 9.

In order to investigate the effect of coupling selection, the stiffness of the flexible gas turbine coupling in the refined model is increased approximately 6 times to simulate a more rigid coupling. This is denoted as scenario A and the same excitation function as for scenario 0 is used. Although the moment of inertia of the coupling has not been changed in the model to reflect this, any modification of coupling inertia is expected to produce negligible effect since it is small compared with the inertias of the connected system. Results for scenario A are calculated and included in Table 6. A plot of the response torques is shown in Fig. 10.

4. RESULTS

From Fig. 2, it can be seen that the maximum peak excitation torque is approximately 2.5MNm. Due to the exponential decay term, the excitation torque decreases with time. The rate of decay depends on the time constant of the excitation function. In this case the excitation decreases relatively slowly because the time constant is equal to 0.4s which means that the excitation will drop to 1/e (37%) of its initial value after 0.4s (20 shaft revolutions). The excitation contains frequency components at 50Hz and 100Hz respectively. Since the 50Hz component is larger, the main period of the excitation torque is clearly 0.02s.

Discrete mass/spring model

The torsional responses at stations 2 and 7 of the discrete model are indicated by the curves MX2S and MX3S respectively in Figs. 3 to 6. It can be seen that the responses (except for scenario 1). appear very different from the excitation. The response at station 2 (gas turbine end) is a wave with a frequency around 24Hz riding on a slow frequency of 3.5Hz. On the other hand, the response at station 7 (steam turbine end) is a wave with a frequency around 50Hz superimposed on a slow frequency of 3.5Hz. These frequencies have obvious connections with the natural and exciting frequencies. The following table gives the calculated natural frequencies of this discrete model:

Mode No.	Natural frequencies (Hz)
1	0.0
2	3.5
3	6.1
4	24.2
5	74.0
6	295

Note: Mode no. 1 is the rigid body mode.

Table 3 Calculated torsional natural frequencies of the discrete model

For scenario 1, the excitation function is substantially changed due to the phase shift and the peak responses are comparatively low. We concentrate now in comparing the effect of the parameters on the other three scenarios, i.e. 0, 2 and 3. The peak amplitudes for the responses are summed up in the following table:

Scenario	Torque station 2 (kNm)	Torque station 7 (kNm)
0	32.2	11.0
2	32.3	11.2
3	34.7	12.6

Table 4 Calculated peak response torques of the discrete model

It is found that the peak responses are quite low in comparison with the nominal torque of 200kNm. But it will be shown later that the response values here should only be used as relative measures. A more refined model is needed to determine the actual peak responses. From Table 4, it can be seen that the change of time constant from 0.4s (scenario 0) to 0.8s (scenario 2) practically does not affect the peak responses. This is expected since the larger time constant implies that the excitation will decay more slowly and its effect on the response is almost completely dominated by the system damping. On the contrary, the system damping has a significant influence on the peak response torques. As expected, a decrease in damping incurs higher response torques. Thus, a reliable estimate of damping is important in such transient analyses. But usually damping in complicated systems like turbine generators or compressor trains cannot be accurately determined. Very often, damping estimates are taken from experience data of previous machine trains by their manufacturers.

Refined finite element model

The calculated natural frequencies of the refined model are given below:

Mode No.	Natural frequencies (Hz)
1	0.0
2	5.3
3	9.8
4	38.4
5	70.3
6	95.9
7	184.9
8	211.4
9	335.3
10	362.1

Note: Mode no. 1 is the rigid body mode.

Table 5 Calculated torsional natural frequencies of the refined model

It can be seen that the calculated natural frequencies differ considerably from those of the simple discrete model. This is also one of the reasons why the transient responses will be different.

Torsional response torques at stations 90 and 151 in the refined model are checked. Their peak response values are summarized in the following table:

Scenario	Torque station 90 (kNm)	Torque station 151 (kNm)
0	173	10.8
3	177	11.4
A	240	10.8

Table 6 Calculated peak response torques of the refined model

As shown in Fig. 8, the response torque at station 90 is composed of a wave of about 38Hz riding on a slow frequency of 10Hz and that at station 151 consists of a wave of about 70Hz superimposed on a frequency of 10Hz. As expected, the responses increase when the damping is reduced in scenario 3. It is interesting to note that the responses at the gas turbine end are very different from those in the discrete model whereas the responses at the steam turbine end are more or less the same as those in the discrete model. This may be explained by the fact that the steam turbine is a relatively light component in the train and it can be represented by a more coarse model than the long and heavy components like the gas turbine and the generator.

From the results of scenario A, in which the gas turbine coupling becomes more rigid, it is obvious that the peak response torque at the gas turbine coupling is significantly increased whereas that at the steam turbine coupling is practically unchanged. This shows clearly the importance of coupling characteristics in determining the system response to transient loadings. Its implication on system component selection is further discussed in the next section.

5. DISCUSSION

Transient excitation torque

The transient torque due to short circuits in generators or motors is dependent on the electrical characteristics of the machines. The excitation torque is produced by dynamic electrical moments across the air gap at the generator/motor rotor. According to the types of short circuits, different frequency components can be induced in the excitation torque. Generally speaking, both 1x Ω_N and 2x Ω_N components exist when it is a 2 phase short circuit and only 1x Ω_N component is contained in a 3 phase short circuit excitation torque. There are other types of faults or start-up conditions where transient and/or pulsating torques can be induced. A discussion of these phenomena can be found in Pollard(1972), Hizume(1976), Bogacz et al.(1992).

The main parameters in the excitation torque functions include:
(1) the ratio of torque component to rated torque T_k/T_0,
(2) the time constant τ_k or its reciprocal,
(3) the phase angle ϕ_k which represents phase difference between the torque components.
Their influences on the torsional vibration response have been studied in this paper.

Torsional vibration model

The torsional vibration model for representing the turbine-generator system can be created by different methods and with varying degrees of complexity. However, the model has a significant impact on the accuracy and efficiency of the analysis. In order to reduce the amount of computational efforts in transient vibration analysis, some kind of reduction is necessary. The most popular methods used for reduction are modal truncation, static condensation and the improved electrical analogous model [Schwibinger et al.(1989)]. They have been shown to produce very accurate results when compared to more refined models.

However, a common way of reducing the number of DOFs is by lumping moments of inertia together. Without proper attention to the change in system stiffness, this can yield large inaccuracies in analysis results. Moreover, a number of manufacturers have used simplified discrete models consisting of only 1-2 disks(stations) to represent each component machine, e.g. a compressor. These models are suspected not to be accurate enough for transient torsional vibration analysis as shown in this paper. Especially when shear stresses in the shaft are needed to determine whether they satisfy the acceptance criteria, it is doubtful that such models can be relied on. But discrete models can produce as accurate results as from finite element models, provided that the system is modelled by an adequate number of stations. The criterion to determine whether the model is adequate depends on the specific system configuration and the purpose of the analysis.

When considering the decrease of computational time by using reduced models, it should also be borne in mind that the calculation of shear stresses often requires an expansion process which usually consumes a considerable amount of time. In case shear stresses in the original (full) model are needed e.g. for acceptance check, the advantage of a reduced model in terms of computational saving is lost substantially.

Damping is one of the parameters which is highly uncertain and difficult to determine. Most turbomachinery manufacturers provide their own estimates of damping usually based on previous experience or empirical data. Damping certainly affects the transient vibration response considerably. But due to the lack of reliable operational/measured data, the best way is to make conservative damping assumption, i.e. using reasonably low values of damping. In addition to system damping, the decay of response amplitudes for short circuit phenomena depends on the time constant of the torsional excitation functions.

Torsional vibration response

The response of a turbine-generator system can be measured in different forms. The two most practical forms are torques and shear stresses. Torques in couplings and shear stresses in the shaft are often used to determine whether the system behaviour is acceptable or not. A commonly used source for specifying acceptance criteria is the "US Navy military standard for mechanical vibrations of shipboard equipment" (MIL-STD-167). The military standard requires that the maximum shear stresses should be lower than the infinite fatigue limit which is 4% of the ultimate tensile strength. However, for short circuits which are not normal conditions, the above requirement is too restrictive since the short circuit excitation torques are very often an order of magnitude higher than the normal rated torque. Thus, different manufacturers often have their own acceptance requirement or design criteria for such conditions. One manufacturer uses only the maximum torque transmitted by the coupling as the design criterion whilst another manufacturer requires that the allowable shear stress for the short circuit condition should be under 50% of the material yield strength. Therefore, both torques and shear stresses are typical outputs obtained from short circuit calculations.

As mentioned above, a representative torsional model and reliable system parameters are crucial to the accuracy of short circuit calculations. But the calculation of shear stresses depends on a correct shaft diameter for evaluating stresses. This is sometimes not so straight forward since the torsional model often consists of lumped inertias and massless torsional springs. If the model is "over-simplified" in that only 1-2 disks are used to represent one component machine, it is very likely that the maximum shear stresses and their actual locations cannot be determined accurately.

System component selection

One of the most important components which affects the torsional behaviour of a turbine-generator system is the couplings. Various types of couplings are employed for transmitting torques, e.g. rigid and flexible couplings. The use of flexible (i.e. torsionally elastic) couplings often implies that the system natural frequencies rely heavily on the couplings' dynamic properties, i.e. their inertias and stiffnesses. It is this interaction of the couplings with the system which renders it essential to take them into account in the design/analysis process. Moreover, the selection of a suitable flexible coupling should consider the interaction and the change of system characteristics or operating conditions induced by couplings.

The selection and design of couplings are dealt with extensively in some references [Mancuso (1986), Neale (1991), Zurn (1994)]. Some types of information typically required to properly select, design and manufacture a flexible coupling include: horsepower, operating speed, torques and potential excitation or critical frequencies - torsional, axial and lateral, etc.

As a rule, the first step is to calculate the selection torque based on the horsepower, the speed and a service factor which has evolved from experience and based on past failures. The initial selection is determined by choosing a continuous torque rating value that is equal to or greater than the calculated selection torque value. Since coupling manufacturers use different design criteria, many service factor charts are found in use. Service factors applied incorrectly will lead to unnecessary increase in coupling size and weight. Torque ratings can, however, also be a confusing term because manufacturers have their own rating system.

A comparison of the following safety factors will allow us to select the safest coupling for the system:
1. For normal operating conditions - safety factor related to endurance strength (for cyclic loading) or yield strength (for steady-state loading):
2. For peak operating conditions - safety factor related to yield strength (for peak loading).
Generator short circuit, electric motor starts and other applications can result in single cycle peak torques. It is crucial that these peak torques be considered when selecting a high performance flexible coupling. This is where a knowledge of the exact peak torque is needed. The problem is that it is difficult to predict this without doing a thorough torsional analysis of the system. Service factors become less important when the load conditions - normal, peak and their duration are clearly known. The coupling characteristics should be used to analyze the system torsionally as well as axially and laterally. The last step in the selection process should contain a check of the peak torque and other capacities and restrictions to confirm the initial selection.

System configuration

The sudden short circuit condition can occur for different types of systems such as electric motor driven compressor trains, steam turbine-generator (commonly called turbogenerator) systems, gas turbine-generator systems and more recently, gas turbine / generator / steam turbine systems in combined cycle power plants. Turbogenerator systems have been the main focus of most transient torsional vibration analyses because of their large size and critical role in power generation. With the increasing power of electric motor driven systems and especially turbine-generator systems in combined cycle plants, these systems are more vulnerable to transient torsional vibration problems. This is not only due to short circuits or faulty synchronization, but also owing to other possible phenomena such as motor startup, compressor surge, acceleration through resonance, high speed reclosing of electric network, etc. Transient torsional vibration can only be accurately analyzed with a good understanding of various aspects of the problem as mentioned in the above discussion.

It is found in this case study that the flexible coupling on the gas turbine side is subjected to a higher transient torque than that on the steam turbine side. This is expected because the steam turbine is relatively small in a combined cycle power plant. However, it is not only the absolute torque value but also the ratio of the peak torque to nominal torque which is the main parameter for coupling selection. This paper provides some insights into the transient analysis and results of a typical modern gas turbine / generator / steam turbine system. It clearly indicates that transient torsional analysis should be carefully considered for such critical turbine-generators.

6. CONCLUSIONS

The issue of transient torsional vibration due to generator short circuit is investigated in this paper. Typical short circuit excitation functions are discussed and they are system-dependent since they are determined by the electrical characteristics of the generators. A modern gas turbine / generator / steam turbine system is used as an example to illustrate various aspects of the problem. Transient torsional vibration analysis has to be based on accurate representations of the excitation torque, system torsional model and its parameters in order to obtain reliable response results to judge the system behaviour.

The effects of modelling inaccuracies and parameter variations have been investigated. The acceptance standards for such short circuit conditions vary from manufacturer to manufacturer, but they are generally based on maximum torque transmitted or some kind of allowable shear stress. It can be concluded that the discrete mass/spring model which uses 1-2 disks(stations) to represent a major machine in a train is not accurate enough to determine the maximum torques and stresses under transient and/or steady state conditions. Either a more refined model or a better reduction method is needed.

One of the most important components which determines the transient torsional response is the flexible couplings. The influence of coupling dynamic characteristics on the peak transmitted torque has been examined in the case study. The selection of couplings for turbine-generator systems in modern combined cycle power plants should be emphasized and the coupling dynamic properties must be included in the system vibration analysis. It is hoped that the results of this investigation will give a better understanding of transient torsional vibration for such turbine-generator systems and provide some guidelines for correct system design and component selection.

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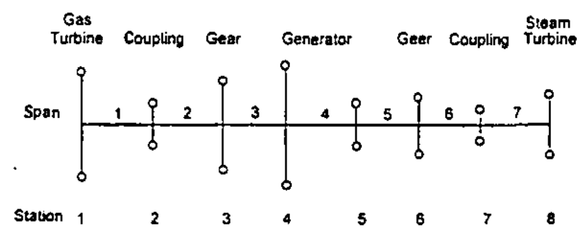

Figure 1 Schematic diagram of the discrete torsional model

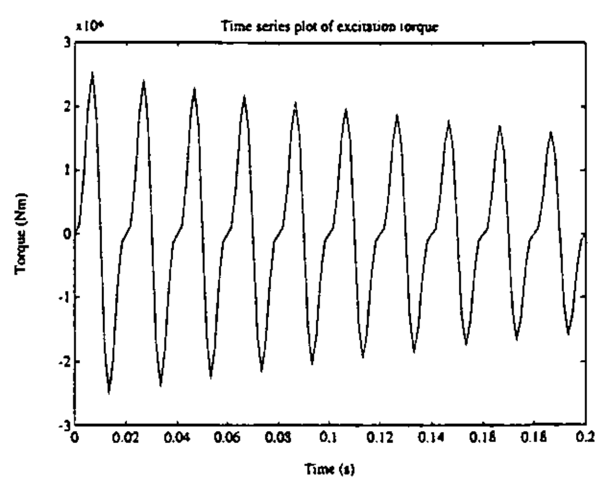

Figure 2 Time series plot of the short circuit excitation torque function

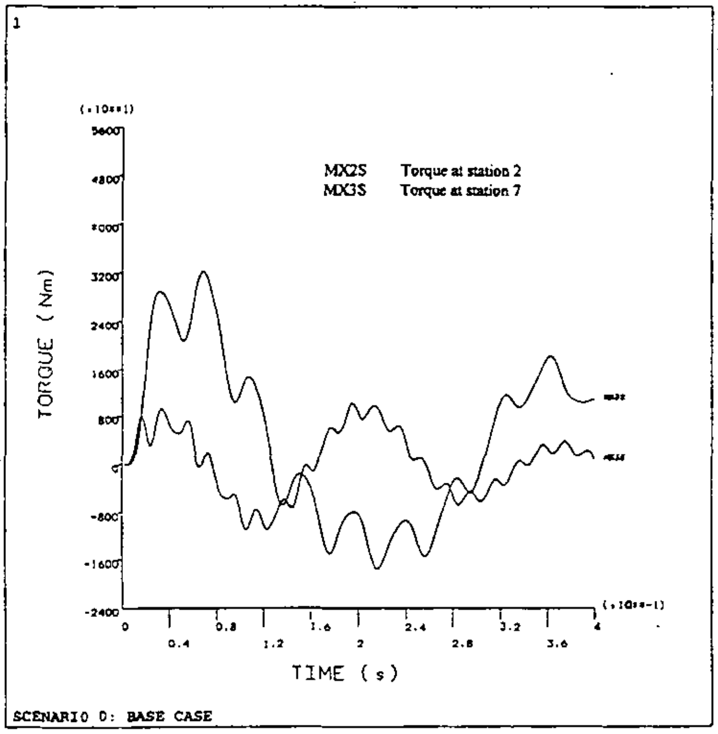

Figure 3 Response vibratory torques of the discrete model for scenario 0

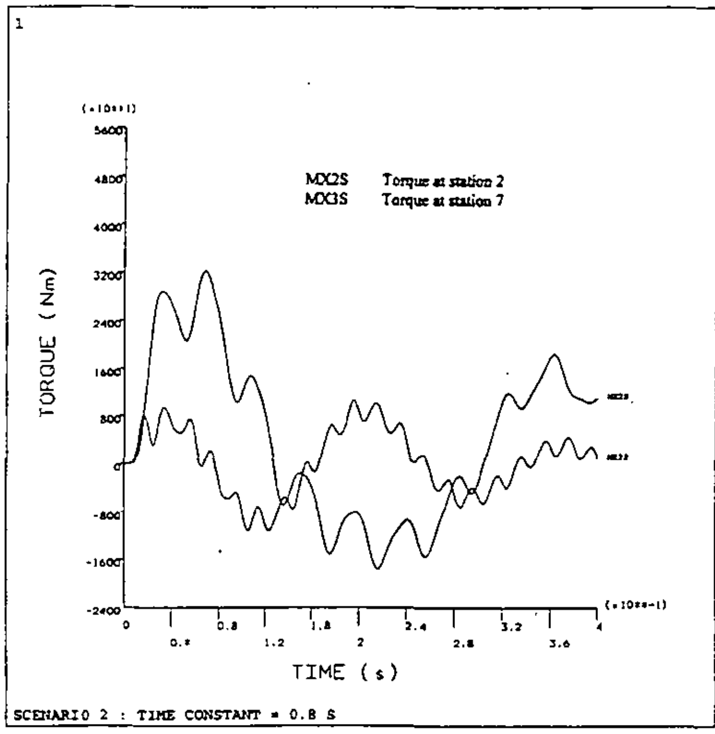

Figure 5 Response vibratory torques of the discrete model for scenario 2

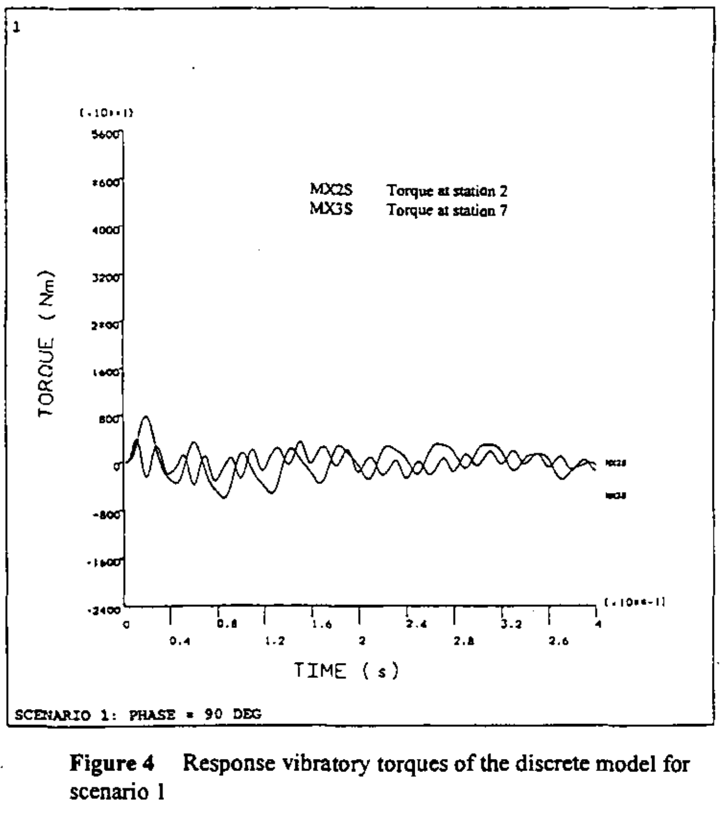

Figure 4 Response vibratory torques of the discrete model for scenario 1

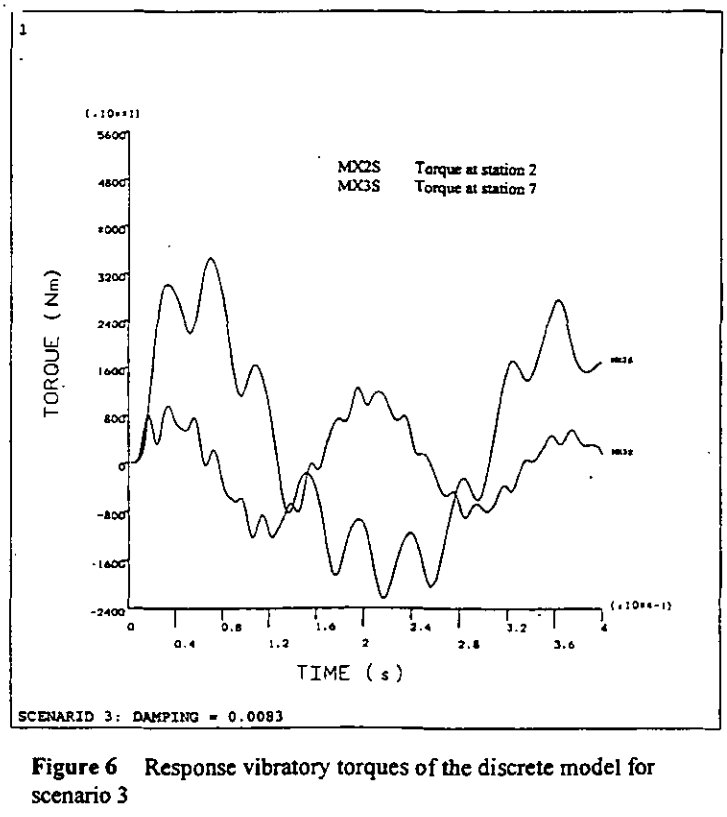

Figure 6 Response vibratory torques of the discrete model for scenario 3

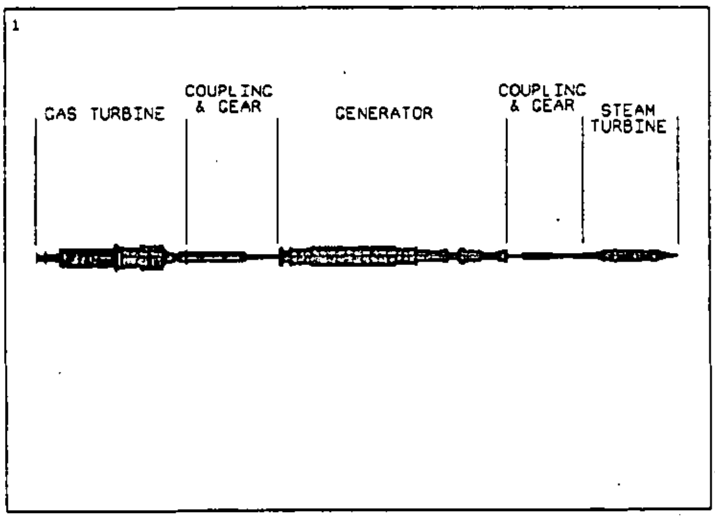

Figure 7 Schematic diagram of the refined finite element torsional model

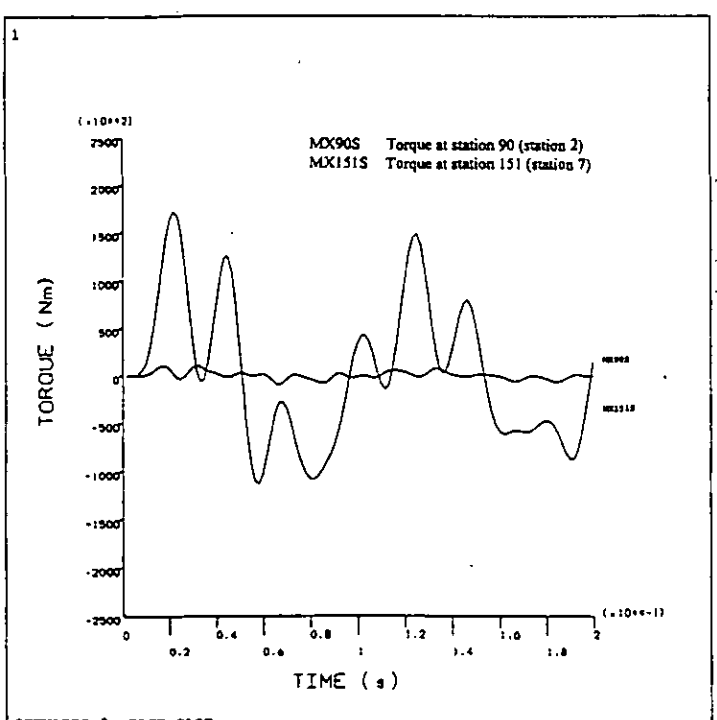

Figure 8 Response vibratory torques of the refined model for scenario 0

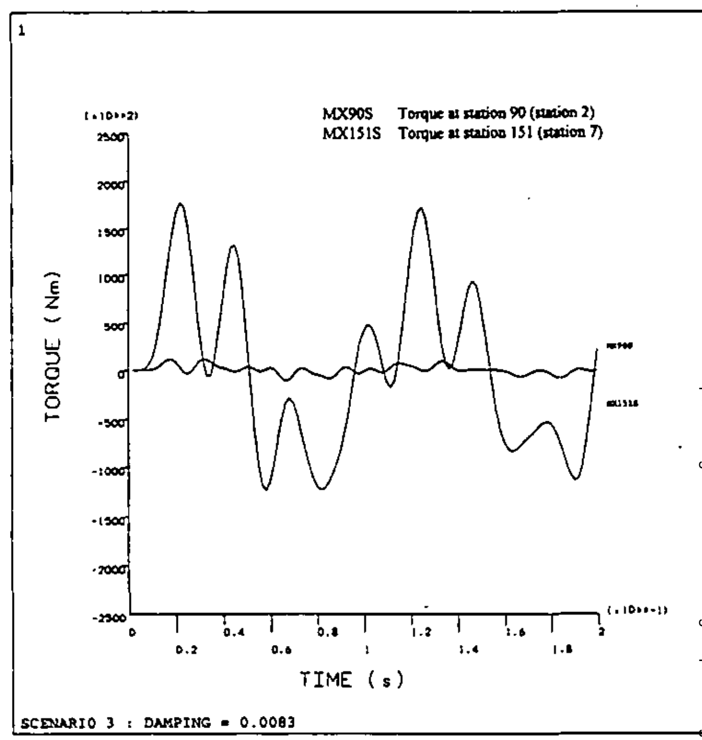

Figure 9 Response vibratory torques of the refined model for scenario 3

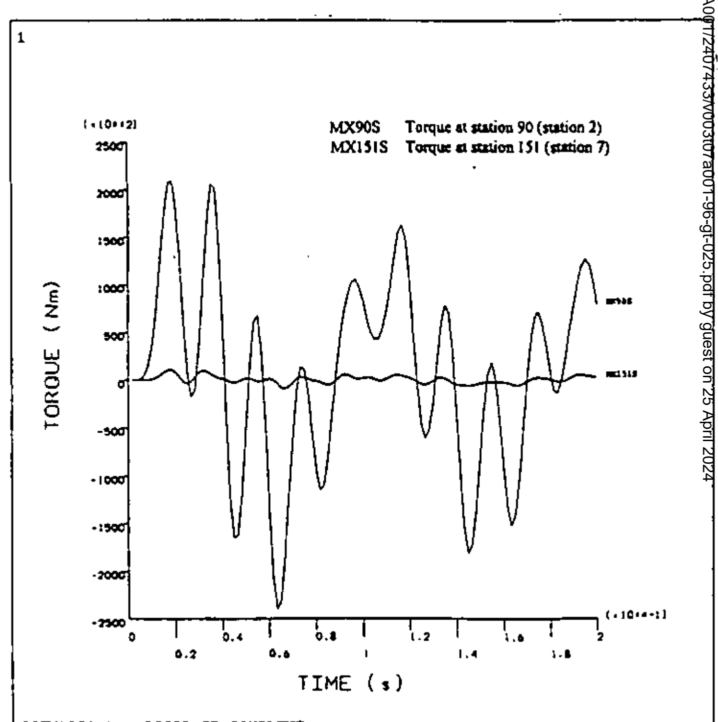

Figure 10 Response vibratory torques of the refined model for scenario A