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FIELD TELEMETRY OF BLADE-ROTOR COUPLED TORSIONAL VIBRATION AT MATUURA POWER STATION NO. 1 UNIT

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At Matuura Power Station No.1 Unit, which is the 700MW coal fired plant of Kyushu Electric Corporation that started commercial operation on June 30, 1989, field telemetry of the last-stage bucket of the steam turbine is executed just after the first steaming in order to verify the blade-rotor (HP, 1P, 2LP's, and Generator) coupled torsional vibration frequency, which is predicted to be near twice the line frequency. The measured frequency is different from the double synchronous frequency, and the steam turbine's reliable operation for the negative sequence current is confirmed. This paper reports the analytical procedure for prediction of coupled vibration incorporated with a single rotor test of blade-rotor coupled vibration at the manufacturer's factory. It also compares the predicted value and the measured results.

1 INTRODUCTION

The effect of torsional excitation in a turbine generator system from the power line is an old but still ongoing topic because of the growing size of electrical networks and larger unit capacity. The electric disturbances which cause torsional excitation of the turbine generator include line accidents such as short circuits and faulty synchronization. Since the mid 1970's, these matters have been widely discussed. A turbine generator was generally modeled as a lumped-mass and spring system. Dynamic simulations of the grid system and shaft system were executed in order to estimate the shaft torque. Stresses in the main journal bearings and coupling were analyzed by using the simulation results. (Ref.1) Finally, fatigue strength was evaluated for shaft life expectancy due to the power line transient.

At the same time, the technology to monitor the torsional frequency of main shaft modes has developed. Generally, toothed gears are machined on each rotor to detect the phase differences between rotors by using a magnet pick-up. The differences in shaft twisting during operation were analyzed using FFT (Fast Fourier Transform) in order to find natural frequencies, mode shapes, and modal damping.

These discussions have focused on the principle lower torsional vibration modes of the rotor system. From the late 1970's, examinations of higher torsional frequency modes coupled with long buckets of low pressure turbines have been reported. The so-called umbrella modes are blade-disk coupled vibration. Since all blades mounted on a disk vibrate in the same direction and the disk vibrates with a zero nodal diameter in the case of an umbrella mode of blade-disk vibration, this mode is hardly excited by steam flow but can be excited by torsional excitation due to negative sequence currents from the power line. (Figure 1) This frequency is a higher torsional mode, and approaches different from the mere shaft system torsional analysis are required in order to accurately estimate the higher torsional modes, because this problem is in the boundary between the blade-disk coupled vibration problem and the rotor torsional vibration problem.

In this paper, the analytical method will be reviewed and estimation of input data for the calculation of a blade-rotor coupled system will be made by measuring the

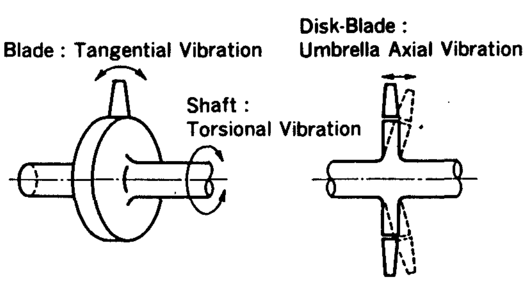


Figure 1 Blade-disk-rotor coupled vibration

umbrella mode of blade vibration coupled with a double-flow low pressure rotor of a 700MW steam turbine during manufacture. Then field telemetry of the last-stage buckets of a 700MW steam turbine at the Matuura P.S. was executed in order to verify the predicted values. The measured results were tabulated and compared with the calculated value.

2 MODELING FOR ANALYSIS

In a coupled shaft-blade system of a turbine generator shaft set, the shaft system and blade system correspond to a main system and to a subsystem, respectively. The main shaft system is more flexible than this subsystem. For example, The first torsional frequency of the shaft system is about 7 to 20Hz compared to the lowest umbrella frequency of blades which is about 50 to 150Hz. Therefore, the coupled model is obtained by combining the standard FEM beam model of the shaft main system with additional branches of a spring-mass model derived from the blade subsystem. A coupled model of the entire system is thus obtained. These additonal branches are actually simple added nodal points with a mass connected to the boss of the shaft by a spring, one for each eigen mode of the blade system.

In case of long blade vibration, two kinds of umbrella modes are generally considered for double synchronous resonance as shown as in Figure 2. The equation of motion can be expressed in the following formula. (Ref.2)

$$\begin{pmatrix} M_1 & & & \\ & m_2+\Delta m & & \\ & & m_{eq1} & \\ & & & m_{eq2} \end{pmatrix} \begin{pmatrix} X_1 \\ x_2 \\ x_{31}^* \\ x_{32}^* \end{pmatrix} + \begin{pmatrix} K_1 & & 0 & 0 \\ & \odot k_{eq1}\ \odot k_{eq2} & -k_{eq1} & -k_{eq2} \\ 0 & -k_{eq1} & k_{eq1} & 0 \\ 0 & -k_{eq2} & 0 & k_{eq2} \end{pmatrix} \begin{pmatrix} X_1 \\ x_2 \\ x_{31}^* \\ x_{32}^* \end{pmatrix} = 0$$

where

$m_{eq1} = a_1^2 \phi_1{}^t M_3 \phi_1$ = equivalent mass of 1st mode
$m_{eq2} = a_2^2 \phi_2{}^t M_3 \phi_2$ = equivalent mass of 2nd mode
$k_{eq1} = \omega_1^2 m_{eq1}$ = equivalent spring of 1st mode
$k_{eq2} = \omega_2^2 m_{eq2}$ = equivalent spring of 2nd mode
$a_1 = 1^t M_3 \phi_1 / \phi_1{}^t M_3 \phi_1$ = 1st coupling factor
$a_2 = 1^t M_3 \phi_2 / \phi_2{}^t M_3 \phi_2$ = 2nd coupling factor
$\odot$ denotes super imposing operation in FEM

Nomenclature is as follows:

X_1 : Absolute disp. vector of the main system
x_2 : Absolute disp. of a bonding point
x_{31}^*: Absolute disp. vector of 1st equivalent mass
x_{32}^*: Absolute disp. vector of 2nd equivalent mass
M_1 : Mass matrices of the main system (shaft)
m_2 : Mass of a bonding point
Δm : Residual mass attached to a bonding point
M_3 : Mass matrices of the blade subsystem
ϕ_1 : 1st eigen umbrella mode of an uncoupled blade subsystem
ϕ_2 : 2nd eigen umbrella mode of an uncoupled blade subsystem
ω_1 : Natural frequency of the 1st eigen mode of an uncoupled blade subsystem
ω_2 : Natural frequency of the 2nd eigen mode of an uncoupled blade subsystem

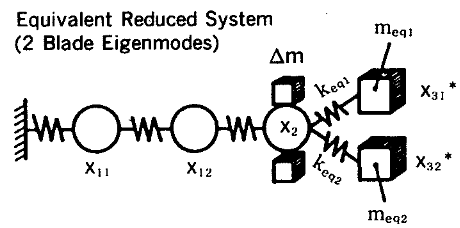


Figure 2 Equivalent Reduced Model (2 Blade Modes)

Table 1

Characteristic Rated		700,000kW
Boiler	Evaporation Quantity Fuel Stack Height	2,300Ton/Hr Coal 200m
Turbine	Main Steam Pressure Main Steam Temperature Reheat Steam Temperature	246kg/cm^2 538°C deg 566°C deg
Generator	Speed Capacity Voltage	3600rpm 778,000kVA 25kV

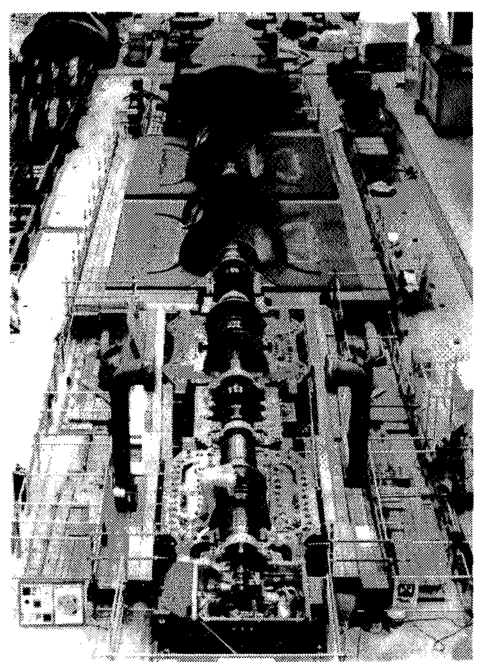
3 COUPLED VIBRATION CALCULATION

3.1 Turbine Plant Description

The Matuura Power Station is a coal fired 700MW power station. Its major specifications are shown in Table 1.

The steam turbine is a tandem-compound type with four cylinders (high pressure turbine, intermediate pressure turbine, two double-flow low pressure turbine, as shown in Picture 1.)

The last-stage buckets of the low pressure turbine



Picture 1 700MW Steam Turbine View

are 33. 5 inches in length, and all blades are continuously connected with loose covers and sleeves for 360 degrees. This construction has very good structural damping and good vibration suppression characteristics for the steam force. The almost 20 years experience with this bucket design has shown its very high reliability.

3.2 Modal quantity calculation of the blade

Due to the pretwisted blade configuration, eigen modes vibrate in tangential and axial directions of the umbrella figure. The two modal masses and modes can be calculated by the differential mesh model of buckets incorporating the cover, sleeve, and dovetail configurations. The details of this procedure were reported by Matsusita et al. (Ref. 3)After the eigenvalue problems of the last stage (L-0) and the last minus one (L-1) stage are solved independently, the two eigen modes, called the tangential and axial modes, are reduced to the equivalent additional masses, as shown in Figure 3.

Concerning the L-0 blade, the equivalent masses of the tangential and axial modes comprise 46.1% and 46.3% of the polar moment of the total inertia of the blades, respectively. The residual polar moment of inertia, 7.6%, is attached to the boss of the last stage (L-0). In the same way, the L-1 stage is reduced, as shown in Figure 4.

3.3 Single LP rotor test

In order to calculate the coupled torsional frequency, it is necessary to first determine the uncoupled umbrella frequency of the blades. In the factory balancing pit, the LP rotor with blade vibration measuring equipment was excited by a torsional shaker during operation. Figure 5 shows the eigen frequency of the coupled umbrella mode of last-stage buckets near 120HZ. Since the measured result is a coupled frequency, the uncoupled frequency can be determined to match the measured results with the calculated result. Table 2 compares the calculated uncoupled umbrella frequency and the corrected uncoupled umbrella frequency. Table 3 shows the umbrella frequency coupled with the single LP rotor and the measured coupled frequency. Although the L-0 tangential umbrella mode has only one mode, the coupled mode with the single rotor splits into two modes.

Figure 6 shows the mode configuration of a single rotor

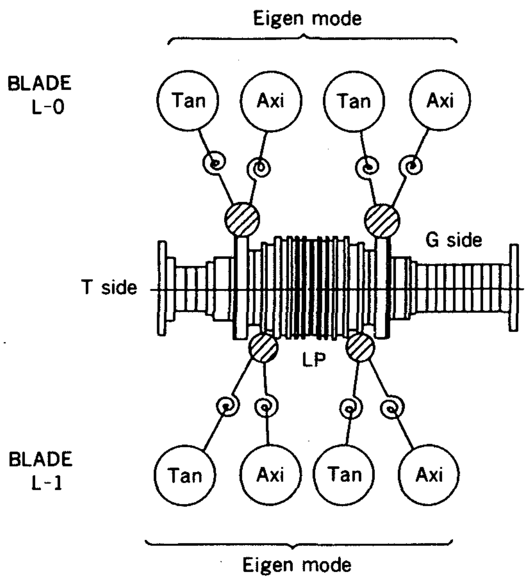


Figure 3 Coupled Model of LP turbine

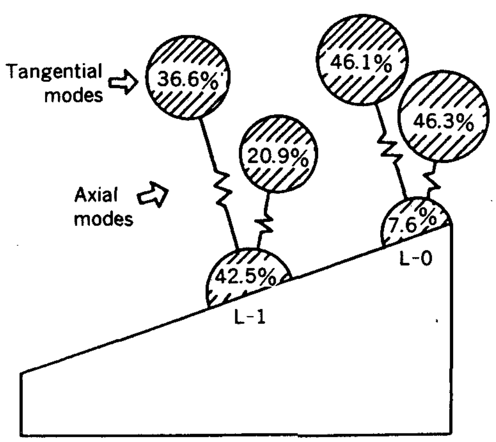


Figure 4 Equivalent Mass Distribution

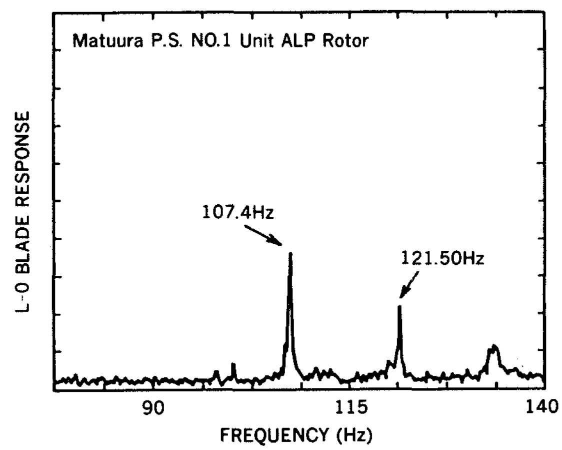


Figure 5 Single rotor umbrella frequency

Table 2 Uncoupled umbrella frequency near 120Hz (L-0 Tangential mode)

Calculated frequency	118.37Hz
Corrected frequency	116.40Hz

Table 3 Coupled umbrella frequency near 120Hz

Calculated frequency*	Measured frequency
107.40Hz	107.40Hz
121.57Hz	121.50Hz

*) Calculation is based on the corrected uncoupled frequency (116.4Hz)

coupled with last-stage blades near 120Hz. In this figure, split modes correspond with the opposite twisting modes of the LP rotor and the blades, where both the turbine (T) and generator (G) side blades twist in the same direction. During the single rotor test, the resonance stress was measured by exciting the torsional exciter. Picture 2 shows the torsional exciter and how it was connected to the LP rotor. Picture 3 shows the LP rotor in the test pit. The measured blade stress is about 1kg/mm² per 5% negative sequence equivalent torque. This means that the stress due to torsional excitation is sufficiently lower than the material fatigue strength.

3.4 Coupled frequency analysis of the rotor train

Based on the single rotor test, the blade-rotor coupled torsional frequency of all rotors connected in a system was analyzed. The shaft system can be modeled by FEM and the blade subsystem can be added using the single rotor test result.

Figure 7 shows the calculated modes and frequencies of blade-rotor coupled vibration near 120Hz. These results are very similar to the single rotor test, because the modes are chiefly related to the L-0 blades and LP rotor stiffness.

The above results also show that there exist two similar modes for each coupled-split result, because there exists two LP rotors (ALP and BLP). Since the 120Hz nearest coupled mode (Mode No.8) is the ALP and L-0 coupled frequency, it was decided to execute a single rotor test for the ALP rotor.

4 FIELD MEASUREMENT OF COUPLED VIBRATION

4.1 Measuring method

In order to verify the coupled frequency under field

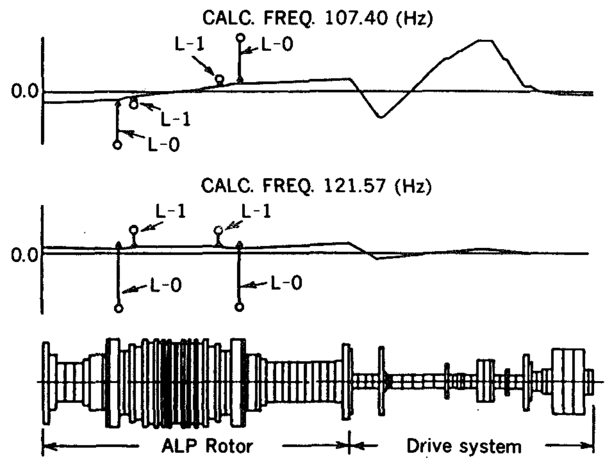
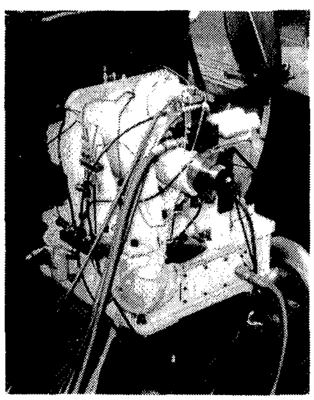
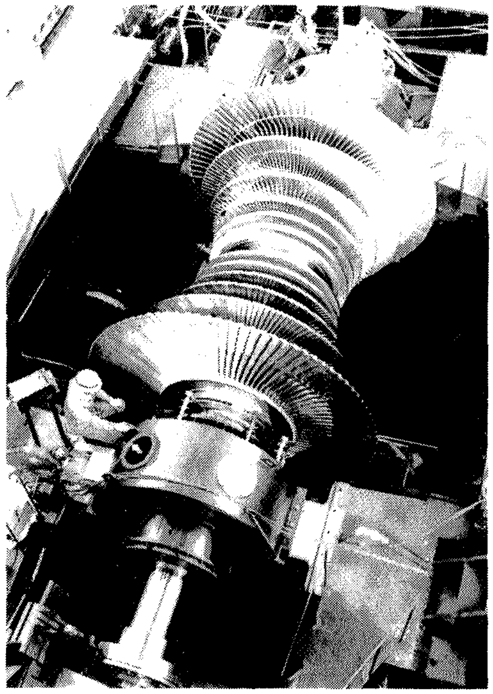


Figure 6 Mode configuration



Picture 2 Torsional Exciter



Picture 3 Single LP rotor test view

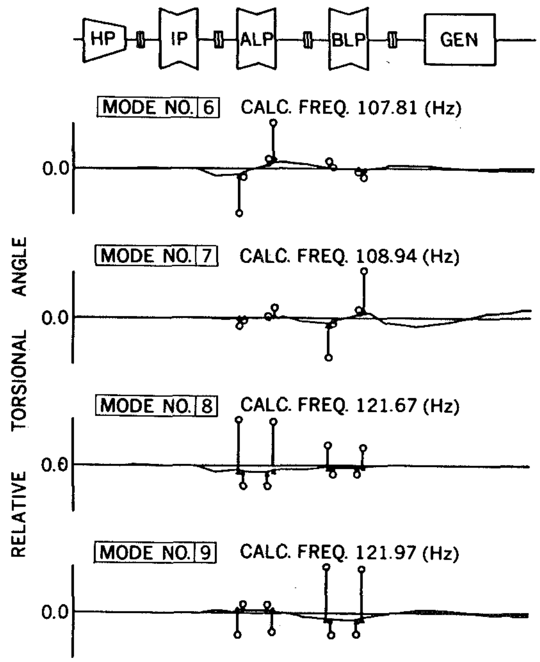


Figure 7 Blade-Shaft Coupled Torsional Modes

conditions, it was decided to measure blade-rotor coupled vibration. Since the mode near the double synchronous frequency is the L-0 blade umbrella mode coupled with the shaft system, strain gauges to measure vibration were attached to the L-0 blades of both a ALP rotor and a BLP rotor. The output of the strain gauges was transmitted by a frequency modulation (FM) telemetry system to a receiving antenna. Figure 8 depicts the telemetry system. Picture 4 shows a transmitter and a receiving antenna under preparation. The transmitter and batteries were set in a balance weight slot of the last stage disk. The receiving antenna was set in a bearing cone of the LP casing.

The test was executed just after initial steaming, considering strain gauge life under a steam flow. Under a no load condition, the turbine was slowly sped up and down near 60Hz and the shaft system was excited from a generator by a one phase grounding circuit at the high voltage side of the main transformer. Figure 9 shows the excitation circuit. The breaker between the main tranformer and the line was open. The measured signal was recorded and analyzed by FFT.

4.2 Field test results

Field test results are shown in Figures 10 and 11 for the ALP rotor and the BLP rotor, respectively. The resonance point with double synchronous frequency nearest to 120Hz is 124.2Hz in the ALP rotor. This frequency is far from twice the expected line frequency variation of 58.5~60.5Hz (117Hz~121Hz).

The eigen frequencies of coupled umbrella modes are tabulated in Table 4. These measured values are also compared with the calculated values. Since the calculated values were incorporated in the single rotor test of the ALP rotor, the differences between the measured values and the analyzed results are smaller for modes dominated by the ALP rotor than those of the BLP rotor. From these results and the single rotor test results, it is necessary to confirm

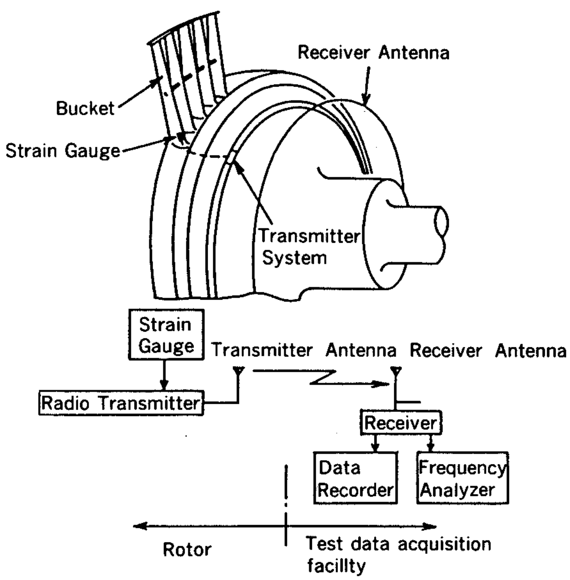


Figure 8 Telemetry System

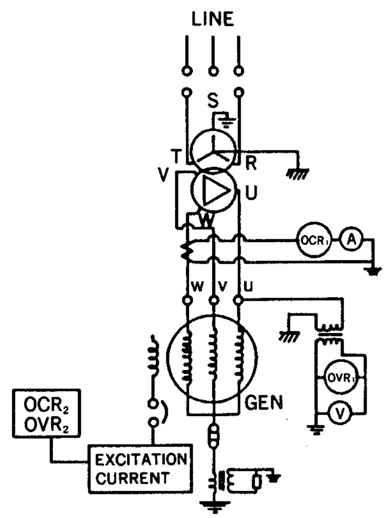


Figure 9 Excitation Circuit

Picture 4 Telemetry Equipment

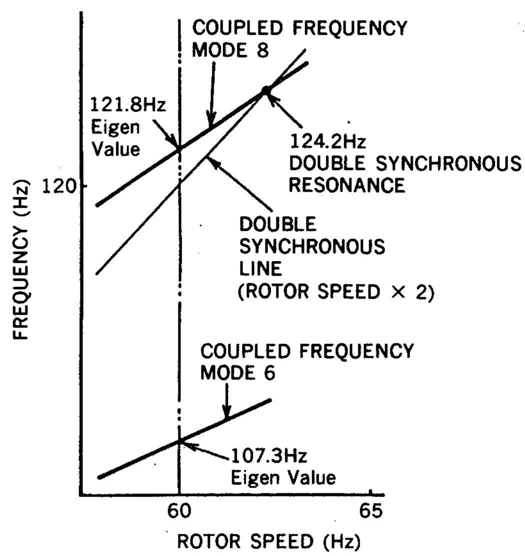


Figure 10 ALP rotor test results

the umbrella frequency and equivalent spring of blades in order to get better accuracy. The input data for blades are one set, but four coupled modes are certain to be very accurate. This means that this analytical procedure (quasi-modal reduction technique) can be applied to estimate the shaft coupled umbrella frequency of an actual turbine shaft.

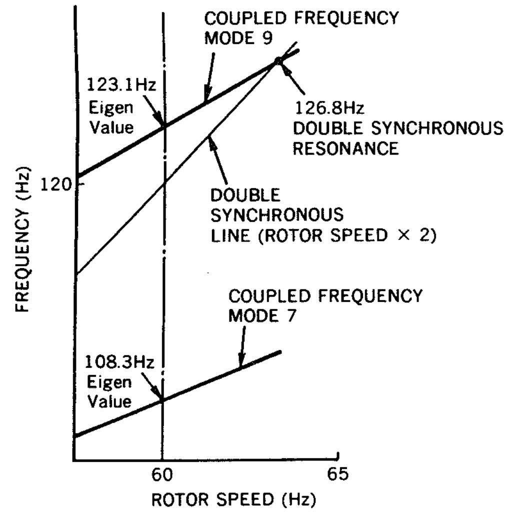


Figure 11 BLP rotor test results

Table 4 Eigen frequencies of test results

Mode	Measured	Calculated	Difference
			Out-of-phase
ALP	107.3Hz	107.81Hz	0.51Hz
BLP	108.3Hz	108.94Hz	0.64Hz
			In-phase
ALP	121.8Hz	121.67Hz	0.13Hz
BLP	123.1Hz	121.97Hz	1.13Hz

4.3 Evaluation at a rated load

The test was executed under a no load condition. Table 5 shows a temperature comparison between the test condition and a rated load condition. The changes in rigidity of the rotor due to this temperature change effect are about 3% for a HP turbine, 4.5% for an IP turbine, and 1.5% for a LP turbine. Since the coupled umbrella modes are mainly dominated by the L-0 buckets and the LP turbine, the change in coupled frequencies is small, as shown in Table 6. Especially, the coupled modes near 120Hz do not change, because these two modes are basically the blade umbrella mode.

Table 5 Temperature condition

	Test condition	Rated load
HP TB. Inlet	397.5°C deg	538.0°C deg
HP TB. Exhaust	218.6°C deg	296.9°C deg
IP TB. Inlet	390.5°C deg	566.0°C deg
IP TB. Exhaust	218.8°C deg	337.8°C deg
LP TB. Exhaust	35.1°C deg	33.0°C deg

Table 6 Frequency change under a rated load

Mode	Measured under no load	Expected change under rated condition
Out-of-phase		
ALP	107.3Hz	−0.2Hz
BLP	108.3Hz	−0.1Hz
In-phase		
ALP	121.8Hz	0.0Hz
BLP	123.1Hz	0.0Hz

5 CONCLUSION

The quasi-modal reduction technique and FEM model were used to construct an analytical model for the blade-rotor coupled torsional vibration of a steam turbine generator of the Matuura Power Station. A single rotor test was executed in order to evaluate umbrella vibration characteristics. Based on the single rotor test results and the quasi-modal procedure, the total rotor system was analyzed to predict coupled torsional frequencies. Finally, field measurement of the vibration of the last-stage buckets was made which confirmed that the double synchronous resonance is 124.2Hz, meaning that the machine can be safely operated. The measured eigen values are very close to the predicted value. The single rotor test and this analytical procedure thus proved to be a valid technique to estimate coupled torsional vibration.

6 ACKNOWLEDGEMENT

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7 REFERENCES

[1] Watanabe, T., et al. : Influence of High-speed Reclosing on Turbine Generators and the Shaft System, Hitachi Review, Vol.27, No.1, PP.33-38 (1978)

[2] Okabe, A., et al. : COUPLED VIBRATION ANALYSIS: TORSIONAL VIBRATION OF TURBINE-GENERATOR-BLADE COUPLED SYSTEM, THE 1989 ASME DESIGN TECHNICAL CONFERENCES-12th BIENNIAL CONFERENCE ON MECHANICAL VIBRATION AND NOISE, Vol. 18-1, PP.135-140 (1989)

[3] Matsushita, O., et al : Torsional Vibration Analysis of Turbine-Generator-Blade Coupled System, Proc. Am. Power Conf. Vol 41 (1989) (to be published)