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증기 터빈 버켓의 회전 진동 시험

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Running Bucket Vibration Test of Steam Turbines

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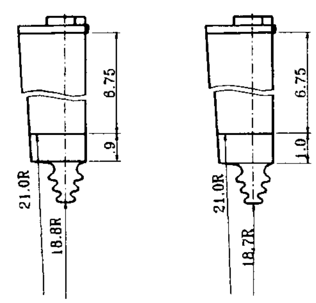
Abstract

A design modification was made on the 9-th stage wheel dovetail of a high-intermediate pressure (HIP) turbine rotor for a fossil power plant that necessitates the use of new long-shank buckets for the row. A bucket vibration test is necessary to verify that the new 9-th stage buckets have adequate frequency margin from a nozzle passing frequency when running at speed. A finite element analysis (FEA) has been performed using a commercial S/W to approximately estimate bucket natural frequencies, and thus to help the vibration test. A row of the new buckets has been assembled on the HIP rotor for the vibration tests using dynamic balancing facilities. The tests have been done during deceleration run with air excitation. The test results are compared with the calculation using our empirical formula, and show that the modified design meets the frequency-margin requirements.

1. INTRODUCTION

A design modification was made on the 9-th stage wheel dovetail of a high-intermediate pressure (HIP) turbine rotor for a 500 MW fossil power plant. Figure 1 shows the geometry of normal and modified buckets. All the dimensions of modified bucket are the same as the normal one but the height of the solid part or platform lengthened by 0.1 inch and the wheel dovetail position moved by 0.1 inch downward for the modified bucket.



(a) Normal Bucket (b) Modified Bucket

Fig. 1 Geometry and Dimension of Buckets

The design modification necessitates the use of new long-shank buckets for the row. A bucket vibration test is necessary to verify that the new 9-th stage buckets have adequate frequency margin from a nozzle passing frequency (NPF). Experience has shown that NPF resonance with bucket vibration modes or mode ranges is to be avoided. The method for calculating bucket frequencies are highly empirical and are based on calibration with test results on very similar buckets. Furthermore, since the buckets are loose in their dovetails when stationary, it will be necessary to do a vibration test to tighten up the dovetails by centrifugal force and then evaluate their vibration characteristics when running at speed. A finite element analysis (FEA) has been performed using a commercial S/W, the BLADE S/W developed by EPRI, to approximately estimate the difference between the modified and the normal bucket natural frequencies at various rotational speeds, and thus to help the vibration test. A row of the new buckets has been assembled on the HIP rotor for the vibration test using dynamic balancing facilities. The test results are compared with the FEA and the calculation using our empirical formula, and are represented in the form of Campbell diagrams.

2. FREQUENCY-MARGIN ANALYSIS OF A BUCKET GROUP

A continuous elastic structure possesses an infinite number of natural frequencies. Its response to an oscillatory force will peak when the forcing frequency coincides with any one of these natural frequencies. Figure 2 shows an example of responses which may be excited in a group of buckets. The response curve is characterized by sets or families of resonance as well as by isolated peaks. These peaks describe the maximum component of vibratory motion such as axial, tangential,

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and torsional. The natural frequencies of group are determined by mass and stiffness distributions of the whole vibrating system. Thus, the bucket twist and taper, number of buckets in a group, tie wires, cover, tenon, wheel configuration and even the pull of centrifugal force helps determine the natural frequencies of the bucket group. Some of these items affect some types of resonances more than others, and a knowledge of these effects is necessary when design limits require the detuning of certain resonances.(1-3)

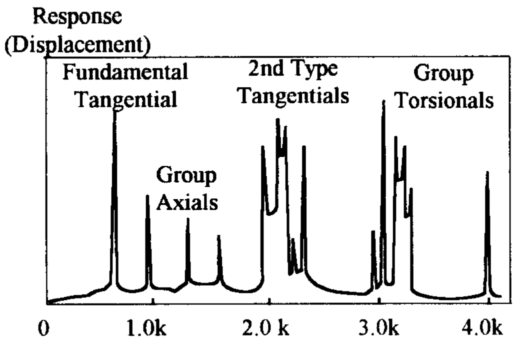

Fig. 2 Typical Response of a Bucket Group

Downstream of each nozzle there is a wake or flow disturbance which is a stimulus to the passing buckets. This stimulus frequency is called nozzle passing frequency (NPF). Since the nozzle force or stimulus pattern will not generally be sinusoidal, stimuli will also exist at integral multiples or harmonics of NPF. In general, it may be expected that energy available to drive the buckets will diminish as the harmonic number increases. It is the practice based on experience in bucket vibration design to be concerned with NPF and the second harmonic, 2NPF. Resonance occurs when a natural frequency of buckets coincides with the stimulus frequency, NPF. With complex vibration mode shapes, it may be difficult to feed large amounts of energy into the vibrating bucket group with the nozzle stimulus. Experience has shown that NPF resonance with the following bucket vibration modes or mode ranges is to be avoided.

i. Fundamental tangential mode (TI0).

ii. Range of modes between the low 2nd type Tangential (TII0) and high torsional (R0) modes.

Grouped bucket natural frequencies for tangential-entry-dovetailed buckets are most easily obtained by vibrating the assembled row while standing. Running vibration tests have shown that for buckets shorter than about 12 inches (3600 RPM) the speed effect on vibration frequencies is negligible. Axial-entry-dovetailed buckets cannot be vibrated while standing due to excessive damping. These buckets must be vibrated while running.

Data from vibration tests have been accumulated for many years. In addition to the raw data, empirical expressions have been derived for estimating TI0, TII0 and R0. For axial entry dovetailed buckets with active length less than 8.5 inches for fossil unit (3600 RPM), the empirical formulas are expressed as

$$\begin{aligned} \mathrm{TI0} &= 7664(\mathrm{R.W.})^{\alpha} / (\mathrm{A.L.})^{\beta} \\ \mathrm{TII0}_5 &= 3.24\mathrm{TI0} \ \ (5 \text{ buckets / cover}) \\ \mathrm{TII0}_4 &= 3.57\mathrm{TI0} \ \ (4 \text{ buckets / cover}) \\ \mathrm{R0} &= 3200 / (\mathrm{A.L.})^{\gamma} \end{aligned} \tag{1}$$

where R.W. and A.L. mean the root width and the active length of the buckets, respectively, and α, β and γ are constant coefficients.

All of the vibration data used in the development of empirical formulas was obtained at room temperature. Bucket natural frequencies will decrease with increased temperature due to the fact that the material modulus of elasticity decreases with temperature. For convenience we have adopted the practice of dividing the stimulus frequencies by the temperature correction factor (T_c=0.933 at 853 °F) instead of correcting each of the bucket frequencies. Thus, the corrected nozzle passing frequencies (CNPF and C2NPF) are written as

$$\mathrm{CNPF} = \mathrm{NPF}/T_c \text{ and } \mathrm{C2NPF} = 2\mathrm{NPF}/T_c \tag{2}$$

where NPF = N_N x RPS, N_N(80) and RPS(60Hz) mean the number of nozzles and the rotational speed, respectively.

The required frequency margins are as follows:

i. Natural frequencies determined by empirical formula : 20 %

ii. Natural frequencies determined from test data : 10 %

The margin is determined as follows:

$$\% \text{ margin} = |f_b - f_n| / f_n \text{ x } 100 \tag{3}$$

where f_b and f_n mean the natural frequencies of running buckets and CNPF or C2NPF, respectively.

Substituting R.W. (2.488 inches) and A.L (6.75 inches) into empirical formulas (1) for the normal buckets, we can obtain the natural frequencies, TI0, $\mathrm{TII0}_5$, $\mathrm{TII0}_4$ and R0 as 672Hz, 2177Hz, 2399Hz, and 3560Hz, and the frequency margins as 666%, 136%, 114%, and 44%, respectively.

A finite element analysis (FEA) has been performed using the commercial S/W to approximately estimate the change of natural frequencies of the modified bucket from the normal one at various rotational speeds. Figure 5 shows the finite element model of the modified bucket. The results of the FEA showed that the differences between the modified and the normal bucket natural frequencies of each mode at the rotational speed range of interest were less than 3 %. From the results of the FEA and the empirical expressions, it is suggestive that this test would produce positive results, and that the vibration characteristics of the modified 9-th stage buckets would satisfy the design criteria for the slight design modification on the bucket.

3. EXPERIMENTAL SET-UP

Figure 4 shows experimental set-up for the running bucket vibration test in the balancing bunker. A row of the modified 9-th stage buckets has been assembled on the HIP rotor for the vibration test. A total of six semi-conductor strain gages has been used. The first and third buckets in two four-cover groups and the first and middle buckets in one five-cover group have been instrumented with one gage each. The data acquisition equipment has been set up to acquire data from 0 to 5500 Hz. The signals from these strain gages have been transmitted from the rotor using telemetry systems (5-6). The buckets were excited using an appropriately designed nozzle injecting air to the tips of the buckets as shown in Fig. 5. The vacuum system in the bunker can pull the air in to provide the excitation force. The test procedure is summarized as follows:

i. Install the rotor assembly in balance cell and prepare for balancing
ii. Balance the rotor per normal procedures up to 20 % overspeed.
iii. Install six telemetries and complete instrumentation checkout and calibration.
iv. Install the air jet excitation provisions on the 9-th stage row 3/4 inches from the leading edge and 1.2 inches from the bucket tip.
v. Decelerate from 4320 to 600 RPM at 60 RPM/min. with the air jet excitation and record gage signals on tape.

4. RESULTS AND SUMMARY

The Campbell diagram shown in Fig. 6 shows that the measured natural frequencies, fundamental tangential TI0, second type tangentials $\mathrm{TII0}_5$, $\mathrm{TII0}_4$, and fundamental torsional R0, are approximately 720 Hz, 2250 Hz, 2550 Hz and 3400 Hz, respectively, and are very similar to the calculated natural frequencies using the empirical formulas for the normal buckets. It suggests that a slight modification on the solid part or platform of the bucket make little change of the bucket natural frequencies.

The corrected nozzle passing frequency (CNPF) for this stage is 5145 Hz. The CNPF must be 10% away from the measured frequencies: TI0, any TII0 and, R0. For this design, the measured frequencies are far less than 5145 Hz and more than 10 % below the CNPF. Therefore, this design meets the standard design frequency margin requirements with respect to CNPF. The higher measured frequency at 4600 Hz is a complex mode. Our design rules do not require a frequency margin between complex modes and CNPF based on successful design experience.

Reviewing the vibration test data or/and Campbell diagram from HIP rotor 9-th stage buckets, it is conclusive that the vibration characteristics of the modified 9-th stage buckets satisfy the design criteria.

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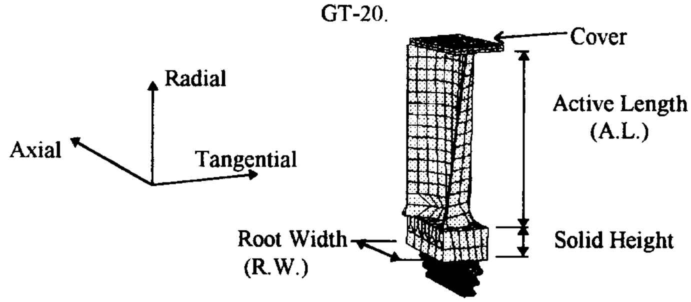

Fig. 3 Finite Element Model of Modified 9-th Stage Bucket

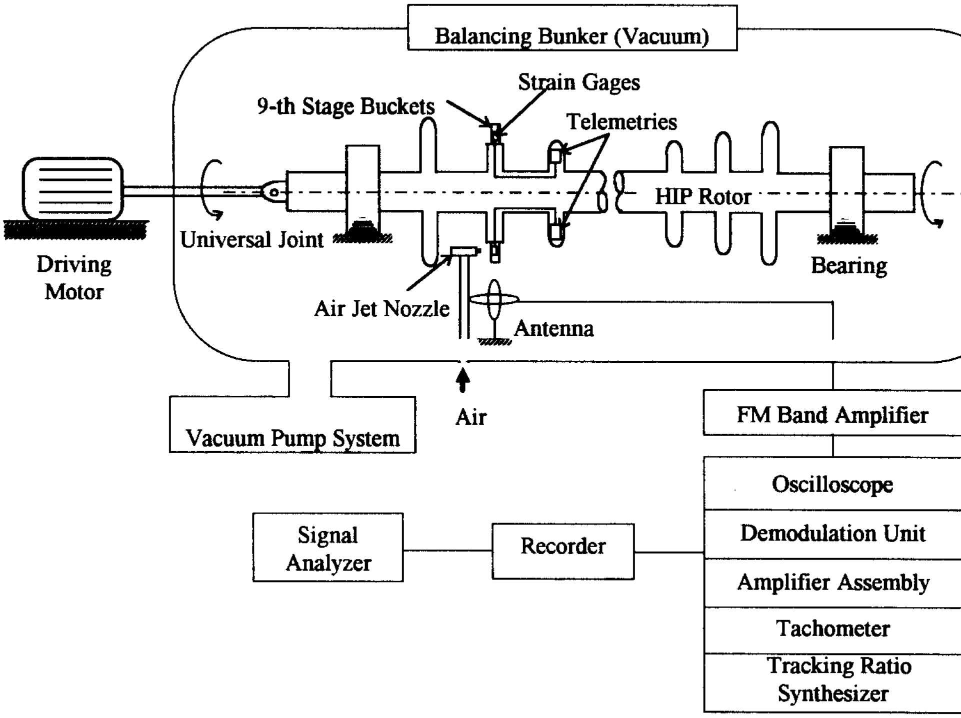

Fig. 4 Experimental Set-up for Bucket Vibration Test

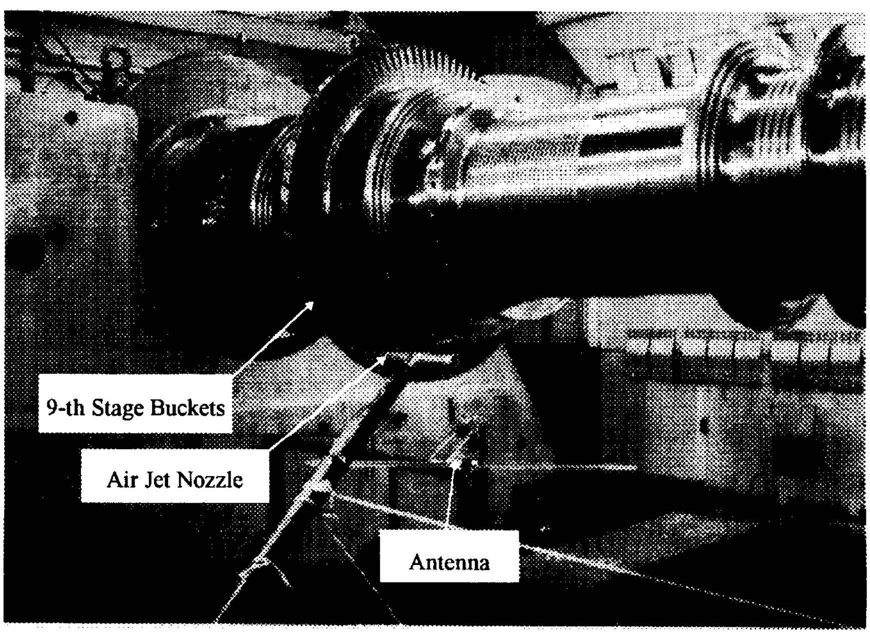

Fig. 5 Bucket-Rotor Assembly Installed in Balancing Bunker

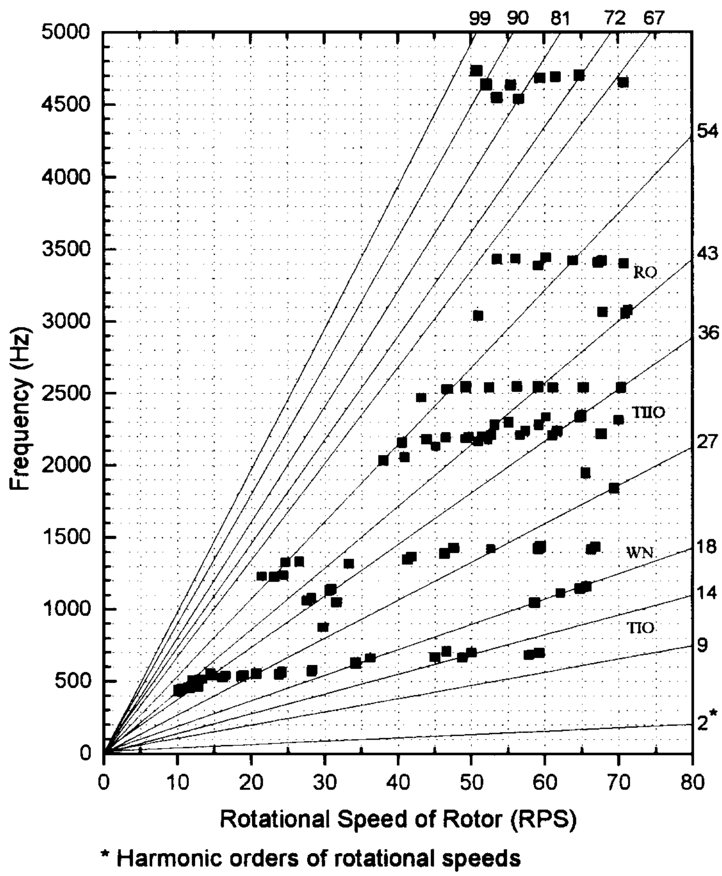

Fig. 6 Campbell Diagram of Modified 9-th Stage Buckets