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In-Situ Running Bucket Vibration Test of an Intermediate-Pressure Steam Turbine

Jong-Po Park*

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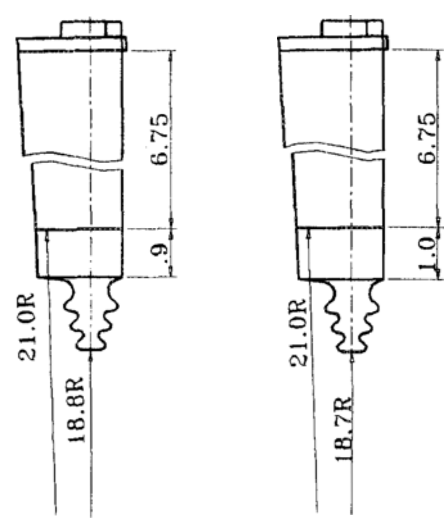
Vibration test and analysis results of running buckets are presented. A design modification is made on the 9th stage wheel dovetail of a high-intermediate pressure (HIP) turbine rotor for a 500 MW fossil power plant that necessitates the use of new long-shank buckets for the row. A bucket vibration test is necessary to verify whether 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 commercial S/W to approximately estimate the difference between the modified and the normal bucket natural frequencies at various rotational speeds. After a row of the new buckets has been assembled on the HIP rotor for the vibration tests using dynamic balancing facilities, deceleration tests have been conducted. The test results are compared with the FEA and our empirical formulas, and show that the modified design meets the frequency-margin requirements.

Key Words : Bucket Vibration, Running Test, Frequency Margin, Campbell Diagram, Nozzle Passing Frequency

1. Introduction

A design modification is made on the 9-th stage wheel dovetail of a high-intermediate pressure (HIP) turbine rotor for a 500 MW fossil power plant that necessitates the use of new long-shank buckets for the row. Figure 1 shows the geometry of the normal and modified buckets.

All the dimensions of the modified bucket are the same as the normal bucket except for height of the solid part or platform which is lengthened by 0.1 inch, and the wheel dovetail position is moved 0.1 inch downward for the modified bucket. A bucket vibration test is necessary to verify whether the new 9th 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 should 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 is necessary to perform a vibration test to tighten the dovetails by centrifugal force, and then evaluate their vibration characteristics when running at



(a) Normal bucket (b) Modified bucket

Fig. 1 Geometry and dimension of buckets.

* Korea Heavy Industries & Construction Co., Ltd. 555 Guygok-Dong, Changwon, Korea

speed. A finite element analysis (FEA) has been performed using a commercial S/W, the BLADE S/W developed by the Electric Power Research Institute (EPRI, 1991), to approximately estimate the difference between the modified and the normal bucket natural frequencies at various rotational speeds. A row of the new buckets has been assembled on the HIP rotor for vibration tests using dynamic balancing facilities. The tests have been conducted during deceleration runs from 4320 to 600 rpm with air excitation on and off, respectively. In this paper, the test results are compared with the FEA and our empirical formulas, and are represented in the form of Campbell diagrams.

2. Vibration Characteristics 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, and torsional. The natural frequencies of the bucket group are determined by the mass and stiffness distributions of the entire vibrating system. This includes the covers, buckets, dovetails, wheel and everything else which is attached. 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 resonance more than others, and a knowledge of these effects is necessary when design limits require the de-tuning of certain resonances (Campbell, 1924; Campbell and Heckman, 1924; Rao, 1991).

Tangential vibration is a term used to describe bucket-wheel frequencies in which the major component of vibratory motion is in the plane of the wheel. In the earliest form of impulse blading, the principal axes of the blade cross-section were truly axial and tangential, and tangential vibration meant just that; tangential motion only. However, as vanes became twisted, tapered and slanted, tangential frequencies acquired axial and torsional components. Slender wheel sections, covers, and tie wires also play a role in coupling the different motions. The form of tangential vibration at the lowest frequency is called the fundamental mode and is characterized by having all of the buckets in a group vibrating in the same phase. However, a number of other modes exist which have all of the buckets vibrating tangentially but with various phase distributions within a group. These are called second type tangential modes. A complete family of these modes is shown in Fig. 3. Theory shows that when there are N buckets in a group, there will be only (N -1) second type tangentials. The order in frequency in which they appear is dependent entirely on the distribution of masses and stiffnesses in the group (Rao, 1995; KHIC, 1995).

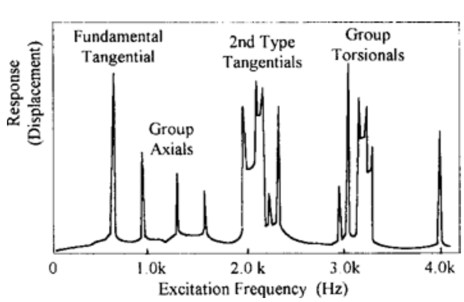


Fig. 2 Typical response of a bucket group.

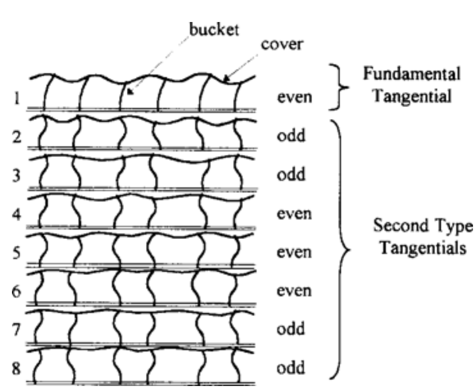


Fig. 3 Tangential modes of a bucket group.

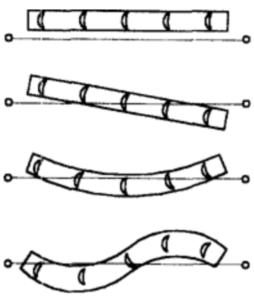


Fig. 4 Axial modes of a bucket group.

Axial vibration was the first form of vibration recognized as dangerous in steam turbine operation. It had resulted in many wheel failures and became the subject of a thorough investigation. Though the investigation primerily covered the thin wheels used in that period (early 1920's), this type of vibration is considered just as serious nowadays when considerably thicker wheels are almost universal (Campbell, 1924; Campbell and Heckman, 1924; Rao, 1991). Other types of axial modes known as group axials have also been found to cause failures. In these modes the wheel plays only a minor role. The action is mostly confined to the vanes and in many ways is quite similar to the tangential family of fundamental and second type modes, in which all the buckets move in phase axially as shown in Fig. 4. The lowest mode is the fundamental axial. This is almost never identified in tests as it is quite indistinguishable from the wheel-bucket axials. The second mode is called the first group axial and it is characterized by a node in the middle of the group. The next mode has two nodes in the group, and so on according to how many buckets there are in a group.

A description for torsional vibration in a group of buckets follows exactly the description given for the second type tangential frequencies, except that the main motion is twisting about a radial line passing more or less parallel to the bucket longitudinal axis. The length of the bucket is the single most important factor in determining torsional frequencies, although tapering buckets and providing wide covers will raise frequencies.

3. Frequency-Margin Analysis

Downstream of each nozzle there is a wake or flow disturbance which is a stimulus to the passing buckets. This stimulus frequency is called the 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. Practical experience in bucket vibration design suggests one should be concerned with the 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 (KHIC, 1995).

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 dovetail buckets are most easily obtained by vibrating the assembled row while standing. Running vibration tests have shown that for buckets shorter than 12 inches (3600 RPM) the effects of speed on vibration frequencies are negligible. Axial-entry dovetail 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 dovetail buckets with active lengths less than 8.5 inches for a fossil unit (3600 RPM), the empirical formulas are expressed as

$$
\begin{aligned}
&\mathrm{T\,I\,0} = 7664(\mathrm{R.W.})^{\alpha}/(\mathrm{A.L.})^{\beta} \\
&\mathrm{T\,II\,0}_5 = 3.24\,\mathrm{T\,I\,0}\ (5\ \text{buckets/cover}) \qquad (1) \\
&\mathrm{T\,II\,0}_4 = 3.57\,\mathrm{T\,I\,0}\ (4\ \text{buckets/cover}) \\
&\mathrm{R0} = 3200/(\mathrm{A.L.})^{\gamma}
\end{aligned}
$$

where R. W. and A. L. denote 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

$$CNPF=NPF/T_C \text{ and } C2NPF=2NPF/T_C \quad (2)$$

where $NPF=N_N \times RPS$, $N_N(80)$ 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:

$$\% \text{ margin}=|f_b-f_n|/f_n \quad (3)$$

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

Substituting R. W. (2.488 inches) and A. L (6.75 inches) into the empirical formulas (1) for the normal buckets, we can obtain the natural frequencies T10, T110$_5$, T110$_4$ and R0 as 672Hz, 2177Hz, 2399Hz, and 3560Hz, and the frequency margins as 666%, 136%, 114%, and 44%, respectively.

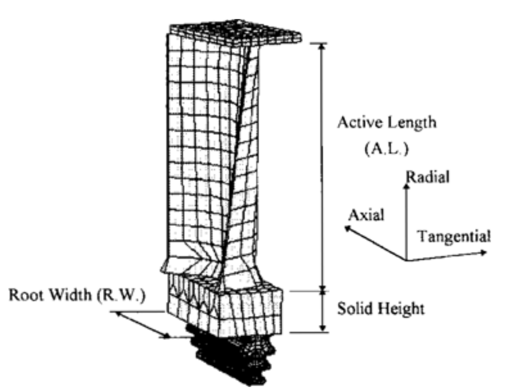


Fig. 5 Finite element model of modified 9th stage bucket.

A finite element analysis (FEA) has been performed using commercial S/W, the BLADE S/W developed by EPRI, to approximately estimate the change in 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 9th stage buckets would satisfy the design criteria for the slight design modification on the bucket.

4. Experimental Set-Up

Figure 6 shows the experimental set-up for the running bucket vibration test in the balancing bunker. A row of the modified 9th 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 as shown in Fig. 7.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 (Donato and Davis, 1973; Gabriel and Donato, 1986). The buckets were excited using an appropriately designed nozzle injecting air to the tips of the buckets as shown in Fig. 8. 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 the balance cell and prepare for balancing

ii. Balance the rotor per normal procedures up to 20 % overspeed.

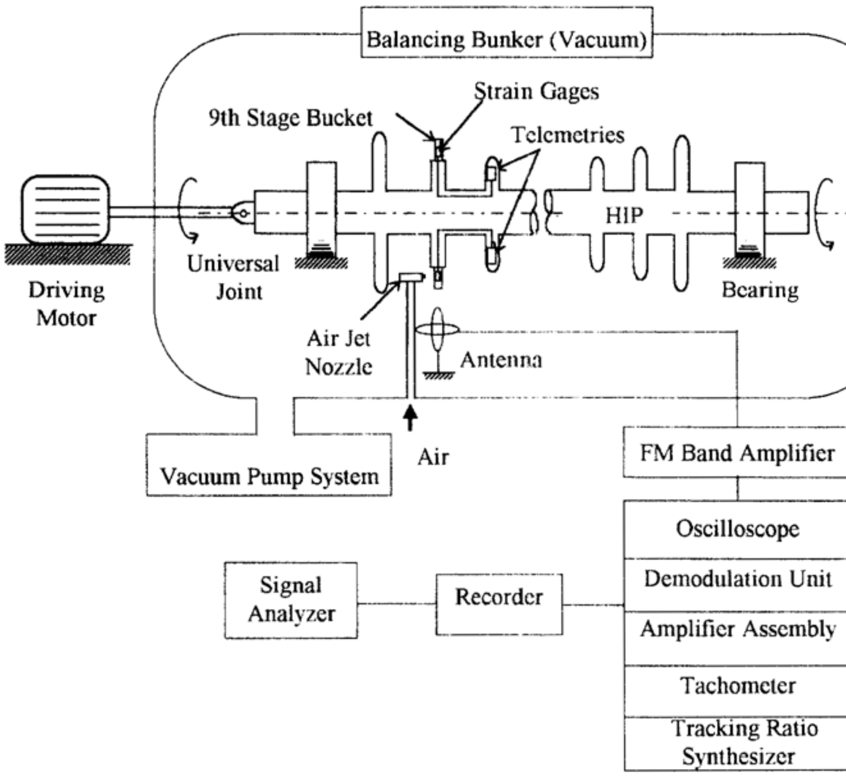


Fig. 6 Experimental set-up for bucket vibration test.

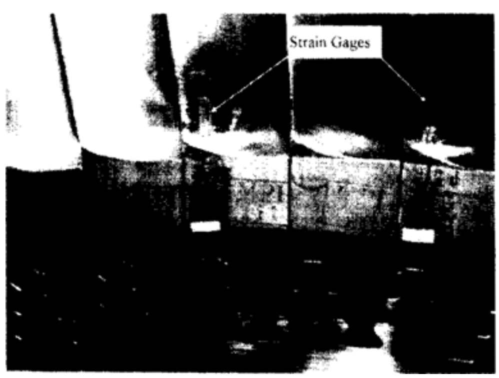


Fig. 7 Instrumented 9th stage buckets and wheel.

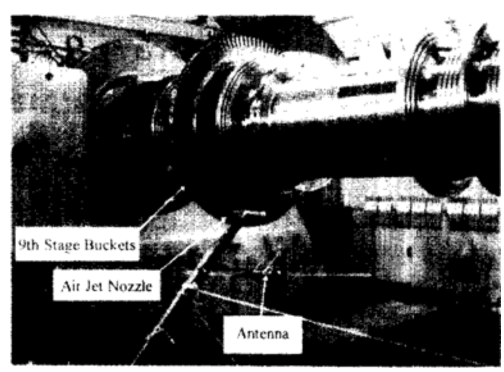


Fig. 8 Bucket-rotor assembly installed in balancing bunker.

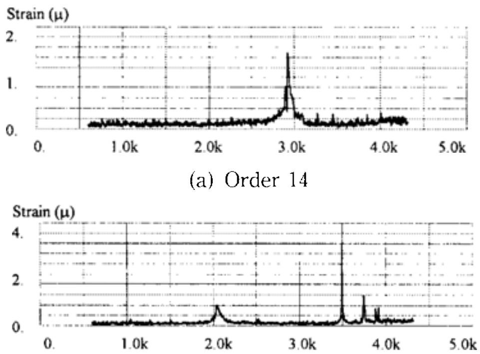
iii. Install six telemetries and complete instrumentation checkout and calibration.

iv. Install the air jet excitation provisions on the 9th 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.

5. Results and Summary

Figures 9 and 10 show the order tracking and the cascade plots of the signals from the strain gages attached on running buckets with the air jet excitation, respectively. The Campbell diagram shown in Fig. 11 as a result of this work shows that the measured natural frequencies, fun-



(a) Order 14

(b) Order 18

Fig. 9 Order tracking plots of modified 9th stage bucket : gage No. 1.

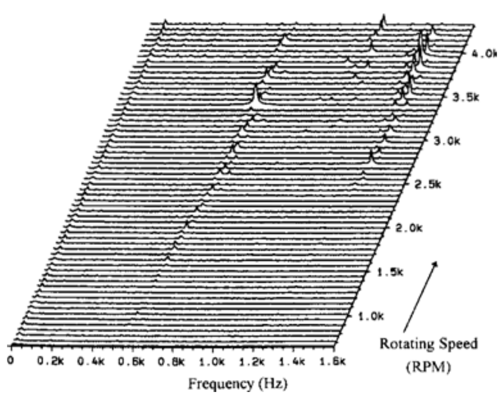
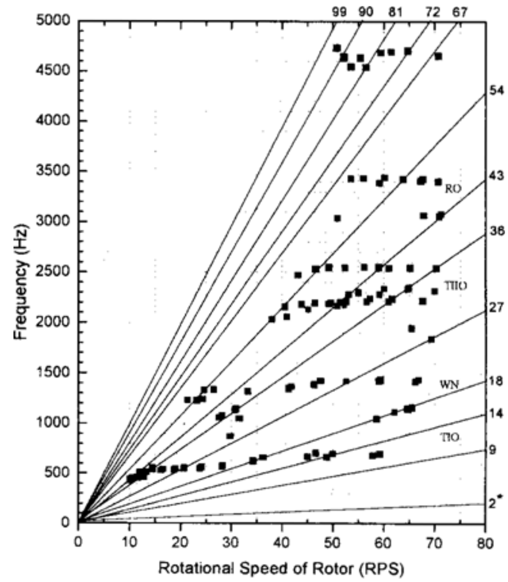


Fig. 10 Cascade plot of modified 9th stage bucket : gage No. 1.

damental tangential TI0, second type tangentials TII05, TII04, 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 cause little change in 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 com-



* Harmonic orders of rotational speeds

Fig. 11 Cambell diagram of modified 9th stage buckets.

plex 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 9th stage buckets, it can be concluded that the vibration characteristics of the modified 9th stage buckets satisfy the design criteria.

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