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Blade vibration triggered by low load and high back pressure

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ABSTRACT

Load and condenser pressure play an important role in overall life of turbine blades especially in the last stages of low pressure (LP) turbine. LP last-stage blades have greatest potential to stall flutter during low steam flow and high backpressure. Exhaust loss refers to the energy lost to the condenser as a result of the steam velocity going into the condenser. The faster the steam leaves the last stage blades, the greater the energy loss. Since the backpressure is a function of the plant cooling system, and the flow is a function of the initial plant design, the only cost-effective way to reduce the velocity is by increasing the area. This is achieved by increasing the length of last stage bucket.

The two primary forces acting on the blades are the steady centrifugal force due to rotation and the fluctuating steam bending force. The best estimation of dynamic stresses in the blades due to the two known sources of vibration works out to be a small portion of the allowable value. But still, incidences of blade failure are regularly reported. Published reports on compilation of blade failures in power plants say that cause for large number of failure is not fully understood. One such cause on which very little is published is self excitation in long blades. Self excitation sustain until the condition remain conducive to vibration inside the turbine. In plants operating at low load and high back pressure the blades are most susceptible to self excitation wherein the blade vibrates in its lower mode. The paper deals with monitoring self excitation in an operating power plant. A Single long blade has been modeled to steady fluid structure interaction by Lagrangian and Eulerian meshing approach during low load and high back pressure. The effect of the steam inlet angle on the blade has been studied. The model simulates low steam flow condition to show increased blade response thus validating observation made in operating power plant.

1. Introduction

It is widely recognized that reliability of low pressure turbine blades decides the limit of technology in building large turbine. Blade resonance, excessive stress and harsh working environment are primary factors that decide the effective life of long blades. The first resonant mode of reasonably long blades appear around 100–150 Hz whereas the blades in the first stage have resonance around 2000–5000 Hz. The blades in the intermediate stages cover the spectrum between these values [1]. Considering the many modes under direct, harmonic and sub-harmonic excitation, there can be hardly any doubt that one or the other frequencies will be under resonance. More importantly, when the most destructive mode is traversed during every startup, good number of high-stress cycles get accumulated in the blades leading to failure in some blades [2,3]. For the

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blades to survive under inevitable resonance, they have to be made strong. Avoiding resonance is the best way but the process is lengthy operation involving many complexities. Blade vibration monitoring is thus important in all types of power plants.

In order to prevent blade vibration and consequent failure, it is important to

- Understand the mechanism and root cause of failure.
- Monitor evolving precursor to take early action to prevent failure.

To achieve the above objective, it is important to adopt online turbine blade vibration monitoring technique. Despite difficulties in measuring blade vibrations, many researchers and manufacturers have invested considerable efforts in developing techniques for blade vibration measurements due to the importance of the problem [4–7]. It is indeed a big challenge to implement in-service measurement by whatever method merely because of the complex behavior of individual blades and entire stage as a whole operating in harsh environment. Some of the non contacting methods have been receiving research attentions since the 1970's and implemented on test rigs and some installed for in-service measurement [8]. Some of the methods are in practice and have been proved technically reliable. Use of casing vibration to monitor blade vibration is gaining popularity in the recent times [8,9].

Blade vibration measurement is required and is very important during development stage to ensure that excessive vibration does not occur and compromise the integrity of machine by the way of impacting with adjacent parts due to large deflections. Known tools like Campbell diagram and direct measurement by use of strain gage based technique may be used [9–11]. That does not mean that the know tools are easy to implement (during development stage). Implementation needs extra ordinary preparation just to verify and validate the design. In-service turbine blade vibration measurement is not just to know if excessive deflection is resulting during operation under different condition but to validate the dynamic characteristics of the blades in different stages at different speeds and propose a robust technique for condition monitoring of turbine engines.

2. Casing vibration on low pressure turbine

It is well known that the casing of a turbine is a pressure vessel which responds to all the dynamics that unfolds when the turbine rotor starts rolling. The method relies on technique which does not need placing of any sensors in the flow path. The turbine casing that envelope the rotating rotor and the blades are strongly coupled in the presence of the working fluid/gas in the turbine. The entire environment inside the casing is filled with general random noise due to steam flow and deterministic steam pressure pulsations corresponding to the rotor speed and its harmonics and pulsations corresponding to different stages of the turbine. All these coexist in the volume of the casing identified as 1X component and its harmonics originating due to rotation of the rotor and the blade passing frequency (BPF) of different stages of blades assembled on the rotor. Each turbine stage generates its own unique frequency and amplitude of pressure pulsation within the casing. The turbine casing enclosing the dynamic pressure environment is subjected to excitation by each of the independently existing pressure signal and as such it responds with appropriate deflection depending on the impedance at each of the deterministic frequencies [8]. The stand alone magnitude of BPF and its trend is directly co relatable to the health of the blades.

2.1. Method of identifying BPF in the casing vibration

Typical frequency spectrum of casing vibration measured by an accelerometer is shown in Fig. 1. The prominent hump around 30 kHz is the mounted resonance frequency of the accelerometer. It is clear from the figure that the steam flow

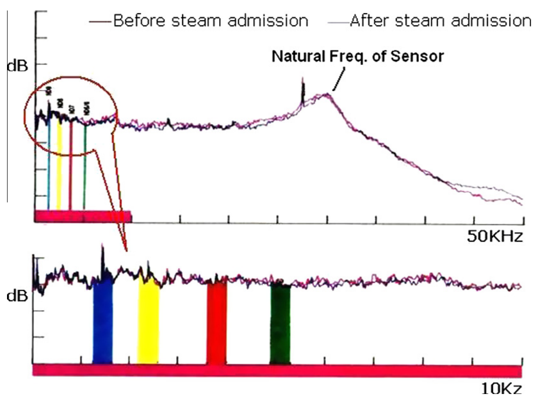

Fig. 1. Typical response of accelerometer placed on turbine casing.