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Vibration Signature Analysis of Shipboard Machinery

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Abstract. A brief description of Vibration Signature Analysis as an effective tool in anticipation and prevention of Shipboard Machinery failures is provided. A case history of marine steam turbine and gearbox vibration signature analysis is given.

1. Introduction

It is well-known that the vibration characteristics of a machinery in operation are indicative of its mechanical condition. Analysis of this vibration signature provides a method to identify and evaluate potential failure in its mechanisms.

With the development of sophisticated and computer based instruments in recent years, the technique of vibration signature analysis has come into prominence. This technology is increasingly finding its use in modern highly stressed high speed machinery. The vibration analysis (i) serves as a diagnostic tool for machinery fault detection and prediction, (ii) provides the maintenance engineer with an early warning of failures and helps in maintenance planning, (iii) is a cost effective non-destructive test technique, (iv) is a quality control tool for new as well as post refit machinery, (v) helps in extending the time between machinery overhauls with confidence, and (vi) helps in assessing vibratory contribution to airborne noise, sonar self-noise and ship-radiated noise, etc.

The effectiveness of vibration signature analysis can be seen in the following case history investigated by the authors :

2. Marine Steam Turbine and Gearbox Vibration Signature Analysis

A 15,000 HP marine steam turbine and gearbox used as propulsion machinery on a ship were reported to have excessive vibration. With a view to investigate and identify the potential sources of vibration, measurements on bearing points were taken on

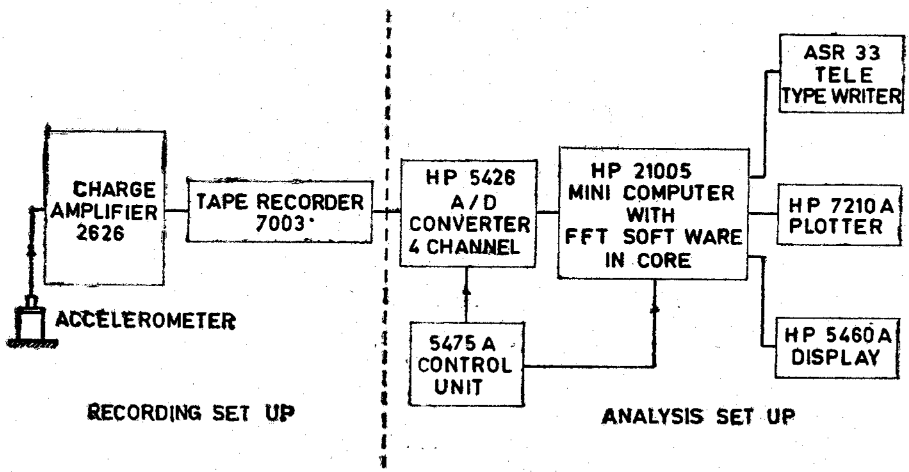

Figure 1. Vibration monitoring set up.

both steam turbine and gearbox for different shaft revolutions. Vibration data was recorded on a portable instrument tape recorder. It was later on replayed in the laboratory for signature analysis.

Fig. 1 shows instrumentation set up used for recording and analysis. Careful analysis of vibration signatures indicated the following defects :

(i) Imbalance of turbine rotor.

(ii) Last stage blade had excessive vibration.

Fig. 2 shows plot of acceleration amplitude *vs* frequency before and after balancing of the turbine. Presence of very high amplitude at fundemental rotational frequency revealed the existence of imbalance of the turbine rotor. Besides, the rise in amplitude of the fundamental frequency component was found to be approximately

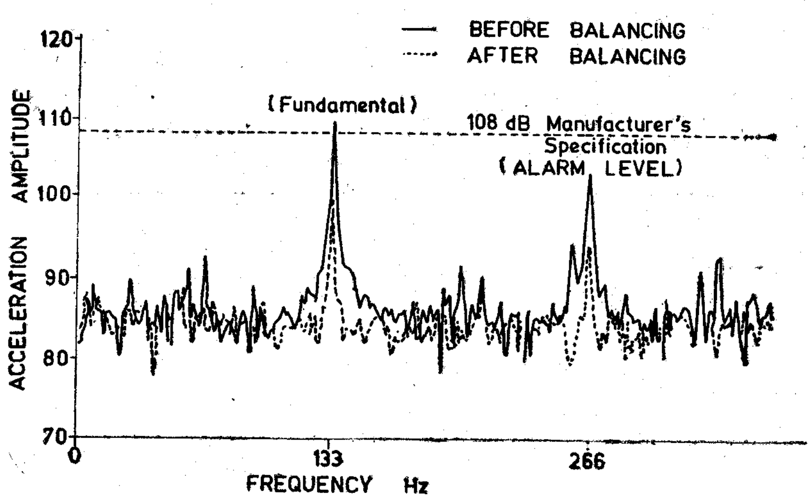

Figure 2. Narrow band analysis of steam turbine vibration signature.

proportional to the square of the turbine speed and the Nyquist plot showed an open ended loop; thus further confirming the imbalance and instability. The imbalance was later on found to be due to inaccurate balancing at the shop floor. On investigation it was found that the machine used for balancing this turbine rotor was meant for balancing heavy duty rotors as compared to this particular small rotor. Hence the imbalance crept in. However, the rotor was once again balanced on a standard balancing machine of rated capacity and the imbalance was reduced to the extent of 66 per cent. Vibration levels were rechecked before acceptance. Though the frequency component corresponding to turbine fundamental frequency was still present after balancing, its amplitude was found to be well within the alarm limits laid down by the manufacturer.

Though trim balancing *in situ* was attempted on aft bearing, it was found that the out of balance forces were fairly high and could not be considered without adversely affecting ford bearing vibration level and also the turbine was not accessible for multiplane trim balancing.

Vibration signature analysis of the two-stage marine gearbox coupled to the turbine indicated two prominent frequencies corresponding to primary and secondary meshing frequencies. On narrow band frequency analysis, these mesh frequencies were found to be somewhat corrupted by extraneous noise signal (Fig. 3), making it difficult to draw any conclusion. However, auto spectrum of these signals (Fig. 3a) clearly brought out the purity of these important frequency components through the absence of any harmonic or sideband frequencies. This indicated that the gearbox was free from eccentric forces, bearing instabilities, hobbing errors or any other manufacturing defect. Considering the large amount of mechanical power being transmitted by this gearbox, the amplitudes of this primary and secondary mesh frequencies were considered to be within limits specified by the manufacturer as shown in Fig. 3. It may be pertinent to mention here that vibration signatures presented in Figs. (3 & 3a) were obtained when the gearbox was run with the turbine as prime mover after balancing the turbine rotor; hence no side bands corresponding to primary pinion rotational frequency were conspicuous in these vibration signatures.

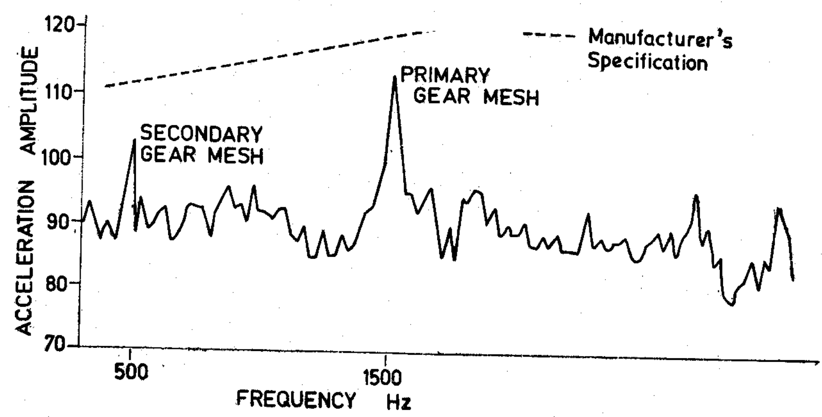

Figure 3. Narrow band analysis of gearbox vibration signature.

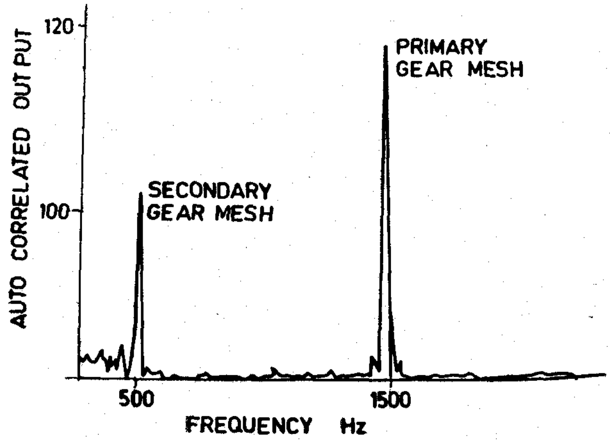

Figure 3(a). Auto spectrum analysis of gearbox vibration signature.

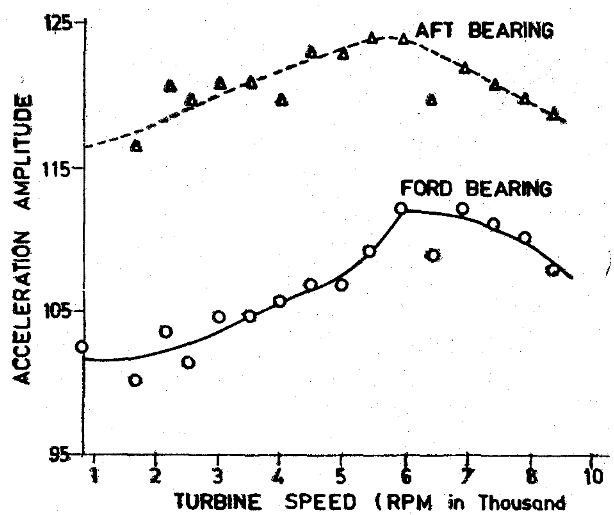

Figure 4. Turbine blade vibration amplitude variation. (Observed value 1.4 khz, Theoretical value 1.65 KHz)

In addition to the imbalance of the turbine rotor which was reflected in the low frequency region, a kind of resonance phenomenon was observed at a frequency 1.4 KHz on the turbine thus showing maximum amplitude at a rotor speed of 6000 rpm, falling down slowly above and below this rpm (Fig. 4). Since this is a fairly high frequency component for a steam turbine, it was assumed that it could be related to either flow induced or nozzle excited vibration of turbine blade. Calculations were made to find out the natural resonance frequency of individual blade in each stage assuming the blade as a cantilever in different modes. The natural frequency value of individual blade was found to be 1.65 KHz in tangential mode after giving corrections for dynamic load effects, temperature effects, etc for the last stage blades (formulae given in appendix–I). For all other stages this value worked out to be fairly high. For calculating these value, constants such as temperature correction factor (α_t), correction for blades shrouding weight (β), correction for blade root softness (K) etc., were obtained from manufacturer's design curves for the specific condition of operation of the turbine. These curves, derived by the manufacturer's as a result of many years of research work, were not available for publication. The formula given at Appendix I when used in conjunction with the appropriate values of the above constants, is expected to give natural frequencies within 10 per cent accuracy.

After careful study, it is felt that steam flow past the blades caused vortex shedding on the blades. This vortex shedding produces alternating forces at a frequency which is determined by the blade cross section dimension/characteristic flow dimension and steam flow velocity at the blade outlet. Large amplitude vibrations are known to occur when the vortex shedding frequency corresponds to the blade's natural frequency in transverse mode. Knowing the dimensions of the blade cross section, flow area and steam flow valocity at turbine rpm 6000, vortex shedding frequency corresponding to Strouhal number of 0.2 was calculated and found to be 1.5 KHz for last stage blades. Except the last stage, a large deviation between vortex shedding frequency and calculated blade natural frequency was observed in all other stages for all rpms. This confirmed that the likely reason for getting a dominant frequency component of 1.4 KHz was due to flow induced last stage blade vibration, particularly at 6000 rpm. Fig. 5 shows the 1.4 KHz dominant frequency component observed at 6000 rpm and its absence at 3000 rpm.

Normally in turbine design, the flow excitation Strouhal frequency is usually kept clear of blade vibration frequency. Calculations were made to find out variation of tangential mode frequency with respect to different blade root fixing clearances. It is observed that even a very minor positive deviation in the blade root housing clearances brings down the frequency value by about 20 per cent. In the present case a repeat inspection check revealed that due to certain machining errors excess tolerances had crept in at the last stage blade root housing thereby bringing down the blade tangential vibration frequency to coincide with flow induced frequency. Knowing the safe level of stress cycle amplitude permitted by the designer and taking the amplitude and

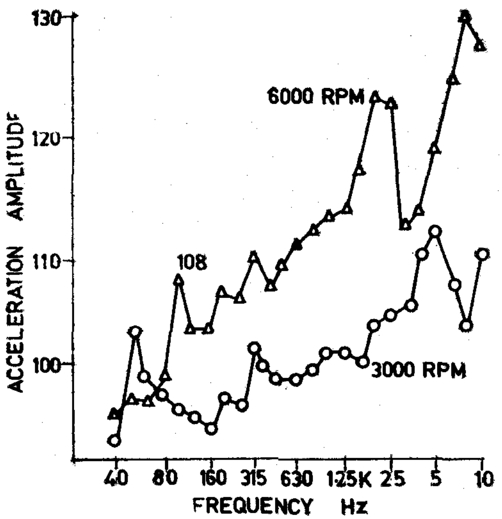

Figure 5. Vibration signature of steam turbine.

frequency of the blade vibration into consideration, a rough estimate of the fatigue life was calculated based on statistical theory and found to be well below the expected life. Formulae and simple theory adopted are given in Appendix II.

3. Conclusion

Based on the above investigations, the following recommendations were made to overcome the vibration problem of the turbine under reference.

1. Shop floor rebalancing of the turbine, as trim balancing *in situ* could not contain the fairly high out-of-balance forces.
2. Removal of last stage turbine blades and replacement of these by blades with oversize root.

It has been possible to overecome the vibration problem effectively by following the above recommendations.

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Appendix I

Estimation of individual blade natural frequency of marine steam turbine in tangential mode

Blade fundamental frequency in cantilever mode

(i) $$f_0 = \alpha \sqrt{\frac{EIg\,\alpha_t}{\beta \rho A h^4}}$$

(ii) Correction for rotary inertia :

$$f_1 = f0\,\frac{(1 - CI)}{Ah^2}$$

(iii) Correction for shear :

$$f_2 = \frac{1}{4h}\,\frac{gG}{\rho s}$$

(iv) Resultant frequency :

$$\frac{1}{f_3^2} = \frac{1}{f_1^2} + \frac{1}{f_2^2}$$

(v) Correction for blade root softness

$f_t = Kf_3$ (K is a function of h and I)

Where f_t is blade natural frequency in fundamental tangential mode.

E = Young's modulus

I = Second moment of inertia

αt = Temp. correction factor

ρ = density of blade material

h = Blade height

A = Blade section area

α = Constant depending on boundary condition

β = Correction for blade shrouding weight

C = Emprical constant

K = Blade root softness correction

G = Modulus of rigidity

S = Shear deflection coefficient.

Appendix II

Failure due to the accumulation of damage in structural members subjected to long duration oscillatory loads can be predicted using peak stress acceleration variation statistics and applying the Random theory. The application of vibration resulted repeated stress cycles to metallic objects like turbine blade will ultimately cause crack development in blade and subsequent fatigue failure; assuming the peak stress to be above a safe limiting value, called the endurance limit of the material. In the present case the response of the turbine blade to the random excitation is dominated by its first cantilever tangential mode. The average number of upward crossings per unit time beyond a critical level 'a' over a narrow band about frequency 'f_n' is given by

$$U_a^+ = f_n \exp\left(\frac{-a^2}{2\sigma_s^2}\right) \tag{1}$$

Where σ_s is the standard deviation of the structural response.

The probability density function of the peak values $p(a)$ is given by

$$P(a) = \frac{1}{\sigma_s}\sqrt{\frac{1-\alpha}{2n}}\exp\left[-a^2/2\sigma_s^2\ (1-\alpha)\right] + \frac{a\sqrt{\alpha}}{2\sigma_s^2}\left\{1 + erf\left[\frac{a}{\sigma_s}\left(\frac{\alpha}{2(1-\alpha)}\right)^{1/2}\right]\right\}\exp\left[\frac{-a^2}{2\sigma_s^2}\right] \tag{2}$$

Where $erf\left\{\frac{a}{\sigma_s}\left(\frac{\alpha}{2(1-\alpha)}\right)^{1/2}\right\}$ = conventional error function

$$\alpha = (U_a^+/M^+)^2 = [\int_0^\infty f^2 G(f)\, df]^2/\sigma_s^2 \int_0^\infty f^4 G(f)\, df \tag{3}$$

and $M+$ = expected number of positive peaks per unit time.

The integral Equation expressing fatigue damage is

$$D = U_a^+ T \int_0^\infty \frac{P(a)}{N(a)}\, da\,;$$ Assuming $N(a) = Ca^{-b}$ from $S-N$ curve relation.

Where $P(a)\, da$ defines the probability that a positive peak will fall between a and da, and b & c are material constants.

Failure is assumed to occur when D reaches approximately unity

The above equation can be further simplified as

$$E(D) = \frac{U_a{}^+T}{C\sigma_s{}^2}\int_0^\infty a^{b+1}\exp\left(\frac{-a^2}{2\sigma_s{}^2}\right) da$$

$$= \frac{U_a{}^+T}{C}\,\sqrt{2}\,\sigma_s)^b\,\Gamma\,(1 + b/2) \tag{4}$$

Eqn. (4) is commonly applied to structural fatigue damage prediction problems. It must be remembered that the result assumes a Rayleigh distribution of stress peaks which is most suitable for present blade fatigue failure problems because of blade's narrow band type of structural response. Although a turbine blade is a continuous structure with an infinite number of degrees of freedom, it is expected that the stress response peaks are distributed much like that of a simple oscillator.