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Vibration Monitoring and Condition Analysis of a 210MW Steam Turbine

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Abstract

In this paper a description is given of the technique of vibration monitoring for condition analysis of steam turbine. Its value as a maintenance tool in the power plant is explained with examples of the type of faults which can be detected at an incipient stage without dismantling or taking the steam turbine off-line. For this purpose a steam turbine of model K-210-130-8 is monitored.
Many power generation steam turbines today are required in service well beyond their intended lifetimes. Dismantling for inspection is expensive and authority need to consider all relevant information in making the decision. Application of condition monitoring in all the applicable methods is justified, with each showing different degradation modes. Data obtained from tests before and after overhaul also reveal whether any restorative work achieved the expected improvements in performance. The paper outlined, with vibration monitoring and condition analysis of a 210MW steam turbine without dismantling it from its working condition.

Keywords: *vibration monitoring, condition analysis, steam turbine.*

1. Introduction
Engineering systems possessing mass and elasticity are capable of relative motion. If the motion of such systems repeats itself after a given interval of time, the motion is known as vibration. Vibration, in general, is a form of wasted energy and undesirable in many cases. It occurs in all mechanical equipment including steam turbines, nuclear power plants [1], pumps etc. Steam turbines are the mainstay of electricity production worldwide. Today's competitive electricity generation market has increased the pressure to keep power generation plant online as and when required. A contributing factor in providing ongoing assurance of acceptable plant condition is the use of vibration monitoring. Methods should be applied according to the modes of degradation expected. Vibration monitoring provides much of this assurance, and has developed such that access to on line vibration data is available to experts who may be located remote from the plant. Some degradation modes can be detected by visual inspection or non-destructive testing, eventually casings do require to be opened for inspection [2]. This is particularly so for the many large power generation machines continuing in service beyond their intended design life. Vibration monitoring can be applied to most plant. It is the one vibration analysis technique, which allows the optimum time for restorative maintenance to be determined, where the deterioration results in increased fuel consumption, or in reduced output, or both. It should be accepted that a turbine outage after a long time in service would probably take longer than if scheduled more frequently, as internal distortion is likely to have occurred. Also, parts such as casing will probably need replacement. Once a casing is opened and clearance measurements made, it is possible to estimate the performance improvements achievable by refurbishment and so justify the expenditure. However, it is clearly preferable to try to determine the internal condition first by testing. The overhaul decision should not be made unless there is a compelling technical or economic reason for opening a casing. Vibration monitoring has been used to extend time between openings of casings to up to 17 years, making its cost/benefits very favorable [3]. Vibration monitoring can also be used to evaluate the effect of maintenance or modification work on the steam path. This is of great benefit when justifying future work.

2. Vibration characteristics of common faults
Rotor imbalance
The key characteristics of the vibration caused by imbalance are (1) it is sinusoidal at a frequency of once per revolution (2) it is a rotating vector and (3) amplitude increases with speed. These characteristics assist in differentiating imbalance from faults that produce similar vibration. The vibration caused by pure imbalance is a once per revolution sine wave sometimes accompanied by low-level harmonics [3]. The faults commonly

mistaken for imbalance usually produce high-level harmonics or occur at higher frequency. In general, if the signal has harmonics above once per revolution, the fault is not imbalance. However, high level harmonics can occur with large imbalance forces, or when horizontal and vertical support stiffness differs by a large amount.

Oil whirl in fluid film bearings

Rotor supported by fluid film bearings are subject to instabilities not experienced with rolling element bearings. A basic difference exists between vibration due to oil film instability and vibration due to other faults, such as imbalance. The latter is a forced response to the imbalance force, occurs at the same frequency and is proportional to the size of the force. Oil film instability, on the other hand is a self-excited vibration that draws energy into vibratory motion that is relatively independent of rotational frequency. This is a highly complex study and further comment is outside the scope of a general paper.

Misalignment

Vibration due to misalignment is usually characterized by a 2 times running speed component and high axial levels. Misalignment takes two basic forms, (1) preload from a bent shaft or improperly seated bearings and (2) offset of the shaft centre-lines of machines or parts of machines in the same train. Flexible couplings increase the ability of the train to tolerate misalignment but are not a cure for serious alignment problems.

Mechanical looseness

Mechanical looseness usually involves bearing mounting housings or bearing caps; and almost always results in a large number of harmonics in the vibration spectrum. Components at integer fractions of running speed may also occur. Looseness tends to produce vibration that is directional, a characteristic that is useful in separating looseness from rotational defects such as imbalance [4]. Measured vibration will be highest in the direction and vicinity of the looseness. One characteristic of looseness is that the basic sinusoidal wave form is truncated or flat-topped, where the looseness is restricted and taken up at the end of travel.

Blades and vanes

Problems with blades and vanes are usually characterized by high fundamental vibration or a large number of harmonics near the blade passing frequency. Some components of blade passing frequency (number of blades X rpm) are always present and levels can vary markedly with load. This is especially true for high speed turbo-machinery and makes recording of operating parameters for historical data critical. It is important to establish base line spectra for several operating levels.

Resonance

Problems with resonance occur when natural frequencies of the shaft, machine housing or attached structures are excited by running speed or harmonics of running speed [7]. These problems are usually easy to identify because levels drop appreciably when running speed is raised or lowered. Spectral maps are especially useful in detecting resonance vibration because the strong dependence on running speed is readily apparent. Piping is one of the most common sources of resonance problems. When running speed coincides with a natural frequency, the resulting vibration will be excessive and strain on the pipe and machine will be excessive and can lead to early failure.

.3. Description of Experimental Facility

Turbine

Specification of the turbine used in the experimental facility is as follows:

- Model No: K-210-130-8
- Type: Condensing type
- Capacity: 210 MW
- Inlet Pressure of Steam: 130kg/cm square
- rpm: 2500-4000

The turbine has total 29 stages. Low pressure cylinder has 6 stages, Intermediate pressure cylinder has 11 and High pressure cylinder has 12.

Vibro-Meter VM600

The Vibro-Meter VM600 is totally flexible both in terms of physical and system configuration. Its working range is from 5-50000Hz. In addition the use of VME hardware, [1] open architecture and industry standard communication protocols for data exchange, VM600 is modular and saleable. VM600 consists of a standard VME chassis (19”, 6U) with four types of Processing/communication cards:

- MPC4 - Machinery Protection Card
- AMC8 – Analogue Monitoring Card
- CPU-M card for configuration/display and network communications to other systems
- CMC16 – Condition Monitoring Card

VM600 is operated with Machinery Protection System 1(MPS1) software.

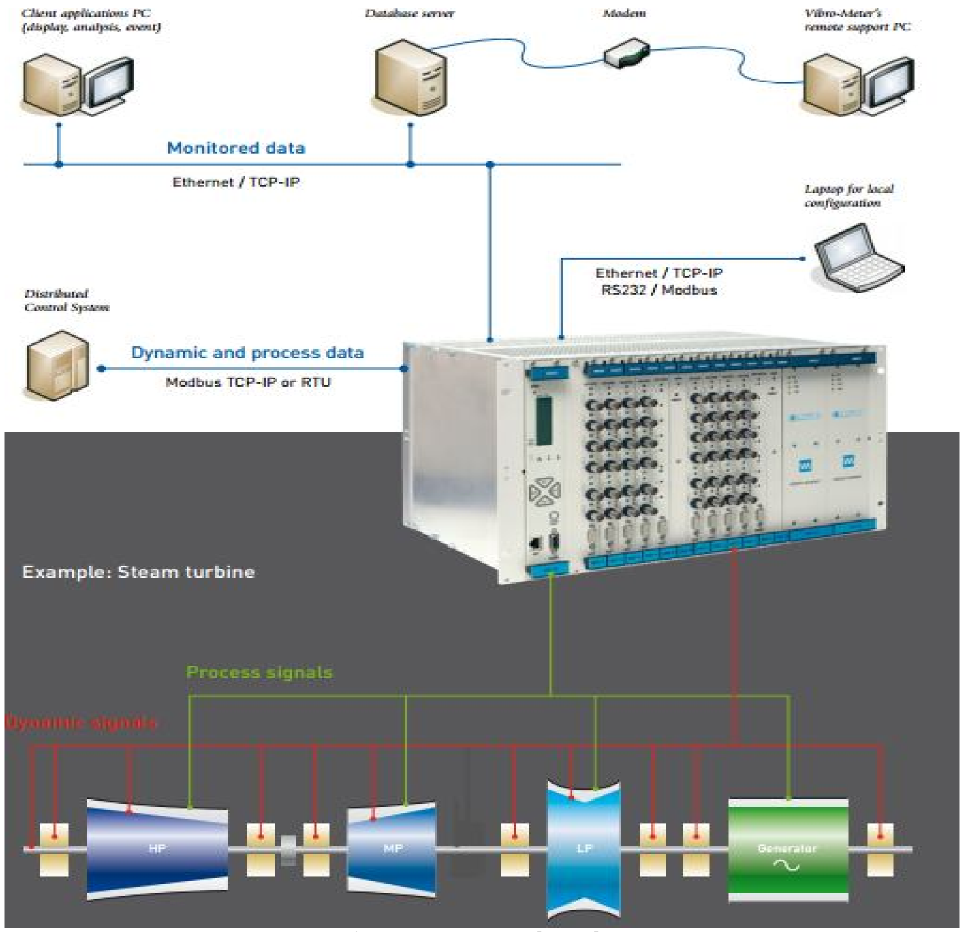

Fig.1 . Experimental Facility

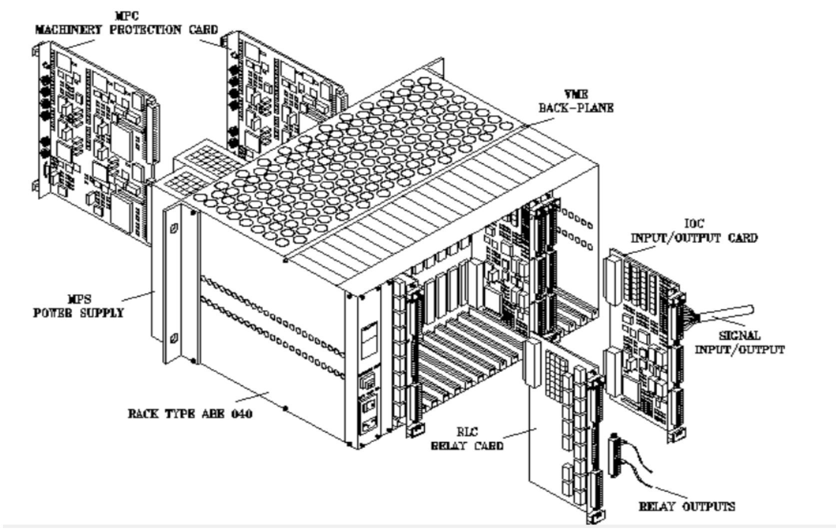

Fig.2 . Vibro-Meter (VM600)

Piezoelectric Accelerometer

Piezoelectric crystals are man-made or naturally occurring crystals that produce a charge output when they are stressed, flexed or subjected to shear forces. In a piezoelectric accelerometer a mass is attached to a piezoelectric crystal which is in turn mounted to the case of the accelerometer [7].

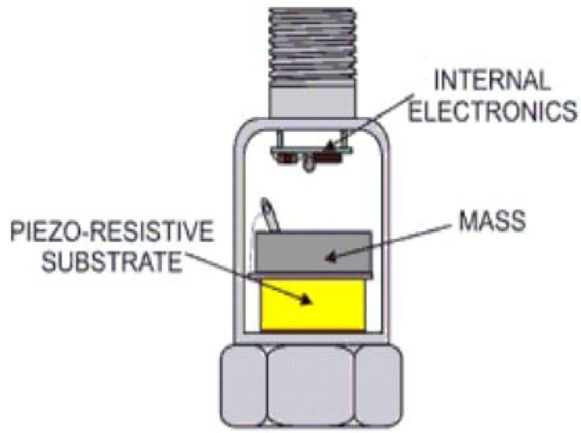

Fig. 3. A single ended compression accelerometer

When the body of the accelerometer is subjected to vibration the mass mounted on the crystal wants to stay still in space due to inertia and so compresses and stretches the piezoelectric crystal. This force causes a charge to be generated. As F=ma, this force is in turn proportional to acceleration. The charge output is either is converted to a low impedance voltage output by the use of integral electronics or made available as a charge output (Pico coulombs /g) in a charge output piezoelectric accelerometer [8]. In this system single ended compression accelerometers have been used. A single ended compression accelerometer is where the crystal is mounted to the base of the accelerometer and the mass is mounted to the crystal by a setscrew, bolt or fastener [8].

4. Methodology

For vibration monitoring three piezoelectric accelerometers have been installed on each of the seven bearings, present in the steam turbine [5]. The piezoelectric accelerometers are installed in three different directions, horizontal, vertical and axial. These accelerometers are used as input sources for Vibro-Meter VM600. So VM-600 is getting total twenty one inputs. VM-600 is getting input as acceleration and giving the output as velocity (mm/s) by integrating the input with respect of time in a definite range [6]. The output result is shown on the computer screen. If the output velocity reaches the value of 4.5 mm/s, the first alarm will be activated. After passing the value of 7.09 mm/s, the second alarm will be activated. After the second alarm, if necessary attempts are not taken for reducing the vibration or taking the output value below 7.09 mm/s; the turbine will shut down automatically. The vibration of the turbine is reduced by reducing or increasing the amount of fuel input and thus reducing or increasing the pressure.

6. Result and Analysis:

Data of vibration of the bearings were taken continuously for 10 days for analysis. From the collected data it has been found that the magnitudes of vibration are changing from time to time. Also the magnitudes are not same for the axial, vertical and horizontal accelerometers installed on the same bearing. For first to third bearings highest magnitudes have been found from axially installed accelerometers. For fourth to seventh bearings, highest magnitudes have been found from vertically installed accelerometers. The magnitudes of vibration of different bearings at a definite time have been provided in the table 1. From the table, the highest magnitude of vibration has been located at bearing no seven, which is also called the tail bearing. As the seventh bearing is located at the end part of the system, maximum instability as well as maximum amount of vibration was found there. So, the seventh bearing needs to be kept on continuous observation. Fig.4 is representing peak vibration vs. day for all the bearings. The peak vibration of a bearing indicates the highest magnitude of vibration which has been got on that day for the bearing. From Fig.4 it can be monitored that the magnitude of vibration is maximum at bearing no seven for each day. On fifth and sixth day the magnitude crossed the limit 4.5 mm/s. So the first alarm was activated on those days.

Table 1. Vibration Measured at a Specific Time

Bearing No	Output Vibration of Bearings (mm/s)		
	Axial (X)	Vertical (Y)	Horizontal (Z)
1	2.10	1.121	1.880
2	3.039	1.465	1.012
3	3.079	2.840	1.045
4	0.553	2.783	1.154
5	1.879	2.216	1.869
6	2.039	2.675	2.645
7	2.112	3.231	2.190

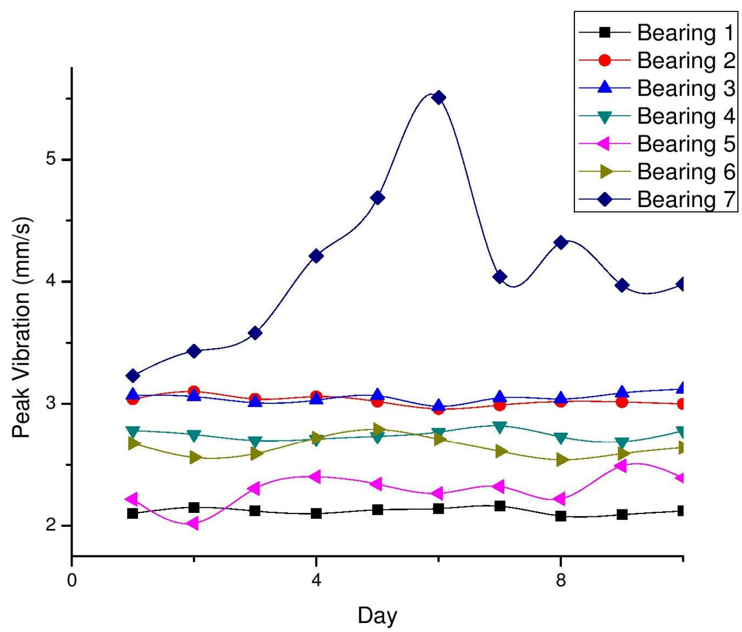

Fig.4 . Peak Vibration vs. Day

7. Summary and Closing Remarks

(1) Incorporation of VM600 system has made the condition monitoring process more reliable and more accurate.

(2) Vibration monitoring process should be imposed in all turbines for condition monitoring which will prevent the efforts of dismantling the turbine and loss of power generation.

(3) The visualization of the actual deformation of a structure under the operational unbalance force excitation gives a better insight in which modes are dominating the behavior at nominal speed. So it can be used in all kind of conditions.

(4) The most important future challenges for vibration monitoring lie in the integration with other condition monitoring techniques towards a reliable “residual lifetime monitoring”, that will enable a safe and durable use of the assets in a flexible power generation portfolio.

(5) As magnitude of vibration is the maximum at seventh bearing, this one should be kept under continuous monitoring and observation.

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