Significance of Vibration Diagnosis of Rotating Machines during Commissioning: Few Case Studies

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ABSTRACT
Satisfactory operation of any machine is always important for plant safety, increased productivity, and low downtime and maintenance overhead. Such requirements could usually be fulfilled by proper design and installation of machines at site. Design part is perhaps more matured in most cases, however installation may play a significant role in the dynamic behaviour even for properly designed machines. Vibration based condition monitoring and codes are well known and widely followed for most of the conventional rotating machines like Pumps, Motors, Turbines, etc. However many rotating machines often used in different process and power plants are not of conventional type wherein condition monitoring is also important, but the codes and the diagnostic techniques may not directly be applicable. The vibration measurements and analysis during commissioning is important to resolve the problems related to the machine installation, if any, and identify the parameters for future condition monitoring of different kinds of machines. It has been experienced that the vibration diagnosis during commissioning gives significant information about the proper installation vis-à-vis the vibration parameters to look for and the locations of measurements for future condition monitoring. The paper presents few such case studies on conventional and unconventional rotating machines.

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1. INTRODUCTION

Rotating machinery used in practice can be classified into two categories – Conventional and Unconventional. The machines often seen, for examples – Electric Motors, Centrifugal pumps, Electric Generators, Turbines, etc. are herein referred as Conventional machines. Any other class of rotating machines used in different processing and power plants are not of conventional types, are referred as Unconventional machines. The proper design, assembly and installation are always needed to cater the optimal design performance as well as to enhance plant safety, increased productivity, and low downtime and maintenance overhead. In fact the vibration based condition monitoring techniques (Goldman and Muszynska, 1999, Sinha, 2002, Rao, 2000) and various international codes giving the guidelines for vibration assessment are widely accepted and used to meet the above objectives for the conventional machines. The Vibration Handbook by Harris (1988) gives a brief summary of different standards and codes on vibration severity limits for machines, and Parkinson and McGuire (1995) give the recent revisions in these standards. However, the application of these techniques and codes may not be straightforward for the unconventional machines. Nevertheless condition monitoring for such machines is also important.

The rotating machine mainly consists of three major components – namely rotor, bearing and foundation that include connecting piping and structures as well. In general, it has been observed that the design of the rotor and bearings are matured enough for the conventional machines, and may be for the unconventional machines for any intended application. So the problem in the machine due to rotor and to an extent in bearing is highly unlikely to exist. Even if some faults occur in the rotor and/or bearing over a period of operation, it can be identified using vibration based signal analysis which is well developed in the field of rotor dynamics for machines (Goldman and Muszynska, 1999, Sinha, 2002). However if the problem is due to improper installation of machine at site then the foundation structure needs to be investigated for the satisfactory performance of complete machine assembly.

Considering these problems, vibration measurement and analysis during commissioning is important to resolve the problems related to the machine installation, if any, and to identify the parameters for future condition monitoring of different kinds of machines. It has been experienced that the vibration diagnosis during commissioning gives significant information about the proper installation vis-à-vis the vibration parameters to look for and the locations of measurements for future condition monitoring. In the present paper, few case studies on both the conventional and unconventional type of rotating machines are discussed.
2. CONVENTIONAL ROTATING MACHINES

The vibration based condition monitoring for the operating conventional rotating machines are matured and usually the following measurements and analysis procedures are adopted for monitoring the health of machines.

(a) Overall vibration measurements as per International Standard Codes (Harris, 1988, Parkinson and McGuire, 1995).

(b) If the vibration amplitudes of machine exceed the severity limit as per codes, usually orbit plot, amplitude and phase plot, waterfall plot, Bode plot, etc. are looked into during normal and transient machine operation to identify the faults so that remedial action could be taken in a planned shut-down.

The above analysis generally identify the rotor related problems like unbalance, mis-alignment, bend, crack, rub, etc. and the fluid bearing related oil whirl and whip. The deterioration in the anti-friction bearings is monitored by crest factor or/and by kurtosis value of the measured responses at bearing pedestals (Bendat and Piersol, 1985, Rao, 1999). The envelope, or more precisely the amplitude demodulation at the carrier frequency (usually the rotating speed of the machine) of the measured responses at bearing pedestal normally locate the exact nature of faults by identifying the bearing characteristic natural frequencies (Harris, 1988, Barkov et al., 1995a & b). These defects may develop due to the change in process parameters and/or over a long period of machine operation. However if the defects are arising due to improper machine installation, more precisely due to foundation dynamics, the above analysis will only identify the defects, and can be rectified before any catastrophic break-down. But such defects will frequently reoccur in machines because the root causes for such recurrence are often not identified adequately by the above analysis. Hence it is good to resolve such problems by conducting modal tests and vibration diagnosis during machine operation at the time of site commissioning. Two case studies on different class of the centrifugal pumps are discussed here.

2.1. CASE STUDY # 1

Figure 1 shows the schematic of the assembly of motor-pump and foundation structure. The motor-pump assembly rests on the top of three steel blocks (attached together by bolts) on either side. The pump is situated at the top of the motor. As shown in Figure 1 the motor and pump units are hung on their support. Total support height is 1.81 m and the length of the pump-motor
unit is 1.75 m which is anchored to the foundation at about 1m from the bottom of the motor. The nominal diameter of the shaft is 100 mm and supported through three bush bearings. The motor-pump shaft has a six bladed impeller at the top. Figure 2 shows the piping layout of the two pumps for both suction and discharge. The pump has an axial suction and a radial discharge pipe of 200mm OD. The suction and discharge lines for Pump 2 are 3 times lengthier than in Pump 1.

*Figure 1 A schematic of the pump assembly*

*Figure 2 Piping layout for Pump 1 & 2*
2.1.1. Vibration Diagnosis
The pump can be operated at 1500 RPM or 3000 RPM depending upon the requirements. During commissioning stage the vibration measurements on the motor-pump assembly were carried out at site when pumps were operating at 1500 RPM. Vibration amplitudes for both the pumps were found to be well within the severity limit as per Hydraulic Institute Standard (1983). However Pump 2 was found to have relatively high overall vibration and with sub and higher harmonics of the rotating speed frequency peak (25 Hz) in the spectrum compared to Pump 1. A typical waterfall spectrum for both pumps at pump casing is shown in Figure 3. The waterfall spectrum consists of two startup events of the pump. The modal tests using the impulse-response method (Ewins, 2000) were then conducted on both the pump assemblies to explain their dynamic behaviour observed during the pump operation. The identified first few natural frequencies for Pump 1 are 23.76Hz, 31.81Hz, 37.76Hz, 45.58Hz and 59.69Hz, and the frequencies at 21.92Hz, 25.80Hz, 32.77Hz, 34.99Hz, 45.14Hz and 49.65Hz are identified modes for Pump 2. Suryam, Sinha and Rao (2003) gave the details of the vibration measurements, modal tests and their results. Since both pumps are supported on similar foundation, the difference in the identified modes for Pump 1 and 2 are suspected to be due to difference in suction and discharge lines.

Figure 3 The waterfall spectrum in x-direction at pump casing ((A) Pump 1, (B) Pump 2)

2.1.3. Discussion
It was observed from the pump dynamics during operation and test modal data that Pump 1 can be operated at 1500 RPM and 3000 RPM as there were no structural modes close to these speeds. However Pump 2 was not safe to operate at either speeds as these speeds are close to its structural modes at 25.80 Hz and 49.65 Hz. Moreover, Pump 2 also showed significant frequency components at 0.5X, 1.5X (near to critical speed 34.99Hz), 2X (near to critical speed
49.65Hz) and 6X (vane passing frequency) other than the operating speed of 25Hz (1X) in comparison to Pump 1. Frequency peak at 12.5Hz (0.5X, where X is rotating speed) is usually due to looseness in bearing assembly or due to gyroscopic effect (Bosmans, 1982). Other cause for 0.5X is due to shaft whirl which is unlikely for the pump, as there is no oil film journal bearing (Sinha, 2002). The possibility of the bearing looseness was unlikely as the pumps were recently assembled. So it could be due to the gyroscopic effect (Bosmans, 1982). The strong component at 0.5X in the Pump 2 indicates more influence of the gyroscopic effect than the Pump 1. This could be explained as the pump is operating close to one of its critical speed 25.80Hz, and so the motor-pump assembly is oscillating at its cantilever type foundation, and hence the combined effect of the shaft rotation and deformation of the entire assembly could result in this pronounced gyro effect. This effect was small in Pump 1 as it was operating slightly above the critical speed. Since in Pump 2 the gyroscopic effect is prominent and operating close to the critical speed 25.80Hz, the varying gap between the impeller and casing will be more that could cause higher hydraulic loading which leads to high vane passing frequency component in Pump 2 compared to Pump 1. High 2X component is also due to the existing critical speed at 49.65Hz compared to Pump 1.

Based on the observation made, it was recommended to either rigidly support the bottom part of the motor or provide rigid support to the discharge and suction lines near pump casing to shift these structural modes for Pump 2. However Pump 1 does not require any modification.

2.2. CASE STUDY # 2

The moderator system of the nuclear power plant under study consists of five vertical pumps with four in operation and one available as a standby. Figure 4 shows the schematic of the layout of the pumps and piping. The pump assembly is shown in Figure 5(a). A vibration survey carried out during commissioning showed normal vibration on top of the motors (i.e., farthest location from the pump base) as per the Hydraulic Institute Standard (1983) but high vibration on the pump casing. The direction of high vibration was specific for each pump: N-S for Pump 2 & 4, and E-W for Pump 1 & 5. Pump 3 had low vibration in both directions. It may be noted that the inlet and outlet piping are similar for Pumps 2 & 4, and for Pumps 1 & 5. If the vibration assessment had been carried out strictly as per the Hydraulic Institute Standard (1983), all the pumps are considered to be healthy, however high casing vibration was not acceptable. This has been investigated and corrected.
Figure 4 Schematic layout of Pumps and Piping

(a) Pump Assembly  
(b) Mode shapes

Figure 5 Schematic of the Pump assembly and its mode shapes
2.2.1. Vibration Diagnosis
The vibration spectrum of the pump consists mainly of the pump speed component (1X) i.e., 3000RPM, and hence not indicating any possible cause for high casing vibration. The modal tests were then conducted on pump assemblies by the impulse-response method (Ewins, 2000). The modal test results indicated that first two cantilever beam modes of the pump assembly were 20.3Hz and 54.7Hz in the N-S direction, and 12.5Hz and 76.5Hz in the E-W direction for Pump 2 & 4, whereas these modes were 17.2Hz and 53.1Hz in the E-W direction, and 11Hz and 76.5Hz in the N-S direction for Pump 1 & 5. The mode shapes for these modes in orthogonal directions for Pump 2 are shown in Figure 5(b). The anti-node for the second mode was observed to be located on the pump casing. Since the operating speed was close to the second mode, high casing vibration was observed during pump operation. Rao, Sinha and Moorthy (1991, 1997) gave the details of measurements and analysis.

2.2.3. Discussion and Solution
The natural frequency close to 3000RPM (50Hz) for the pump assemblies were changed by additional stiffening of the pump stool by welding a thick plate on each side on the stool. The support to the discharge piping of the pump was also strengthened with additional “U” bolts. With these modifications, the casing vibration was reduced significantly. So far none of the pumps reported to have any problems since they have been commissioned in 1991.

3. UNCONVENTIONAL ROTATING MACHINES
Many rotating machines often used in different process and power plants are not of conventional type where condition monitoring is also important, but the codes and the diagnostic techniques may not directly be applicable. In such cases it is important to carry out vibration measurements and diagnosis during commissioning so that the problem related to machine’s natural frequency and RPM can be avoided, and moreover, the measured dynamic behaviour during commissioning can be preserved for future comparison and diagnosis. Two such case studies are discussed below.

3.1. CASE STUDY # 1
The Turbo-dryer is an equipment in which wet chemical in paste form is dried under high temperature and forced air circulation. Figure 6 shows the schematic of the dryer unit. The shaft of the tray unit and the turbo fan are coaxial and are driven by independent drives. They rotate in
opposite direction at 1/6 RPM and 33 RPM respectively. The shafts of the tray unit and fan are supported on bottom foundation through the carbon bush type thrust bearings. The tray shaft carries a number of trays at a number of stages. The tray and fan assemblies are housed in an octagonal casing which is supported independently on horizontal I-beams. The wet charge is continuously fed from the top and removed from the bottom as dry powder. The casing consists of a number of heaters, and a number of levelers/scrapers at each stage of the trays to maintain the level of powder/paste in the tray by sweeping the excess charge to the lower stage trays. Finally the dry powder is taken out from the bottom of the casing.

The dryer unit had a major breakdown after about 10 months of trouble free operation since commissioned in 1990. Large scale damages were seen to its internal structures. The unit was put back into operation after a few modifications. However the operators were cautioned to put-off the dryer whenever a loud noise and vibration occurred.

Figure 6  Schematic of the Turbo-dryer
3.1.1. Vibration Diagnosis

Vibration measurements were carried out in 1991. Vibration levels at different locations were found to be low at the start of the machine, and the operation was quiet and smooth. However with time the vibration level was found to be increasing, as well as the rubbing noise was also heard. The rubbing noise seems to be originating from an area close to the thrust bearings. The FFT analysis of the vibration signal indicates an occurrence of a dominating peak around 12Hz.

Further investigation revealed that the casing with its support had a natural frequency near 12Hz which is close to fan motor speed of 710 RPM. The shaft rubbing with bearings were confirmed by the shaft vibration measurements using proximity probes. Hence the thrust bearings have been suspected to have provided the transmission path to cause the resonance of the casing at the motor RPM. The phase relationship between the lower foundation and the casing supports at 12 Hz was also found to be out-of-phase by 180 degree, hence closing the gap between the trays and levelers/scrapers. Since the casing and its supports are lightly damped structure, the gap closing may cause damage in the tray and leveler if the machine operated for long period. Rao, Sinha and Moorthy (1991) gave the details of measurements and analysis.

3.1.2. Discussion and Solution

A similar machine in another unit of the plant had no such severe history of the failure. There was no rubbing noise and no appreciable vibration level. Detailed investigation revealed that the gap between the tray and leveler was more resulting in minimal chance of direct interference during operation. Secondly the casing structure is directly supported on the floor by eight vertical pillars and so the natural frequency would be much higher eliminating any possibility of resonance.

For the present machine it was observed that there was no severe vibration during first few minutes of the machine start up, and there was no 12Hz component seen in the spectrum shown in Figure 7(a). It only indicates that there was no resonance during this period. However when the machine continued operation for longer period, a dominant peak at 12Hz with its harmonic observed as shown in Figure 7(b) with high rubbing noise indicating the phenomena of resonance. Such phenomena could only occur if there may be small shift in the casing natural frequency. It was suspected that there was no powder hold up at the bottom of the casing during machine start up resulting in a slightly natural higher frequency, however when the operation progresses the powder hold up increases which takes that natural frequency close to the fan motor RPM. The operation personnel further confirmed this suspicion.
One possible solution to reduce the machine vibration level was to increase the gap between the tray and leveler and to change the motor speed to avoid resonance, but these changes were a little involved, and hence not adopted. Alternatively, it was suggested to provide few additional supports like pillars from the floor to increase the frequency of the casing support. The later solution was implemented. After this no major breakdown has been reported in the Turbo-dryer after the support modification.

(a) Just after machine startup

(b) During resonance at 12Hz

Figure 7 Measured dynamic behaviour of the Turbo-dryer
3.2. CASE STUDY # 2

The Pilger mill is a combination of both rotating and reciprocating machine tool for rolling a small length of tube to the required size and wall thickness by applying compressive stresses during each stroke of the machine. The tube is fed progressively during each stroke to form the complete length. The schematic of the Pilger mill is shown in Figure 8. The crankshaft of the mill has five split roller bearings as indicated in figure. Two bearings are mounted on each of the crankpins, and one is mounted on the center pin, which has been deleted in the figure. One of the two bearings mounted on each crankpin is for the connecting arm of the roll-stand (designated as RSB) and the other bearing is for the balancing weight (designated as BWB). The two crankpins are identified as Operator side and Flywheel side.

It was reported that the Operator side BWB has been failing frequently since the machine was commissioned in 1990. It had failed thrice within a period of 2 years. The root cause of this repeated failure was investigated by vibration measurements during normal operating conditions in the year 1992.

![Figure 8 The Pilger Mill](image)

3.2.1. Vibration Diagnosis

Vibration signals were picked up by the piezo-electric accelerometers on all the four bearings (as marked in Figure 8) at the full speed of 277.5 RPM of the crankshaft. The dynamic responses of the balancing weight on the Operator and Flywheel sides are shown in Figure 9 for comparison.
From the time waveforms, it appears that the balancing weight and its connecting arm on the Operator side is not moving freely. The piezo-electric accelerometer could pick up the propagated stress waves due to impact. The vibration signals were processed through an audio system. The audio output supports the above postulation of the obstruction to the free movement of the balancing weight on the Operator side. The bent in the connecting arm was suspected as one of the reason for such time waveform. The site further confirmed this suspicion of the bend. Sinha et al. (1992, 1996) gives the details of the measurements and analysis carried out for this identification.

![Graphs showing dynamic response of balancing weight](image)

**Figure 13** The dynamic response of balancing weight

3.2.2. Discussion and Solution

During maintenance, the bent connecting arm was replaced by new arm, and the machine was put back into service in 1992. Since then, the machine has been working satisfactorily. Few more new machines of similar kinds were installed and commissioned in 1995. They were also put into operation after a certification from the vibration diagnosis by comparing the dynamic behaviour with respect to the Pilger mill (Meher et al., 1995), and none of them reported to have any breakdown so far.
4. CONCLUDING REMARKS
In general, the conventional vibration based condition monitoring techniques used in rotating machines are important and useful in prediction of deterioration in different components so that the defects can be rectified either by replacing the defective components or by repair well in advance before any major breakdown. However if the failure of components is frequent and repeated in nature and particularly when it is occurring due to some kind of resonance, the conventional condition monitoring may not be suitable to identify the root cause in many cases except for identification of deterioration. Hence considering all these limitations, it is always important to carry out details vibration measurements and diagnosis including the dynamic characterization by modal tests during installation and commissioning of machines to avoid many unknown sources of vibration related problems. This has already been brought out through few case studies on different kind of machines. It has been observed that once the root cause of vibration is removed, machines are operating smoothly for years together without any major problem. In fact, each case study is also indicative of the vibration parameters to look for and the locations of measurements for future condition monitoring. For example – Measurement of the natural frequencies of machine assembly over a period of machine operation, vibration measurement at casing for the vertical pump of the Case Study # 2, vibration measurement at bearings of the Pilger Mill, etc. Hence it can be concluded that if a machine is being installed and commissioned with such precautions, then the machine will operate smoothly, and its aging related problems, if any developed over a period of operation, can then be well taken care by the conventional condition monitoring to enhance the safety and availability of the machine.

ACKNOWLEDGEMENTS
All the staff members of the Vibration Laboratory, RED, BARC, Mumbai are acknowledged.

REFERENCES


