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Good vibrations at PARR-1

When Pinstech had to upgrade the reactor coolant system at PARR-1 in Pakistan, the primary pumps were checked by vibrational analysis. How did the technique perform and how were the pumps adjusted to run satisfactorily after initial results showed that their vibration levels were too high?

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Pakistan Research Reactor-1 (PARR-1) – a swimming pool type reactor – was originally designed to use high enriched fuel (90% U-235) and operate at 5 MWt. Its core has been redesigned to operate with low enriched fuel (20% U-235) at 10 MWt. During the conversion and upgrading of the reactor, major changes were made to the cooling system, including installation of a new primary pump, and addition of a heat exchanger and a cooling tower. Vibration analyses were carried out to check that the cooling system components had been installed properly and to investigate any operational abnormalities.

An accelerometer signal was used to measure vibration at various locations on two primary pumps, PW-P1 and PW-P2. The root mean square (rms) value of the vibration signal (g) was converted into velocity (mm/s) and displacement (mm) for comparison with a standard vibration criterion chart for rotating and reciprocating machinery to determine the operating condition of the pumps. The power spectral density (PSD) or frequency spectrum of the accelerometer signal was analysed to identify particular faults such as unbalance, misalignment and bearing instability conditions in the two pumps. General principles of vibrational analysis are given in the panel on p42.

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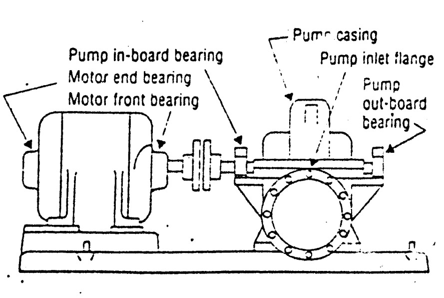
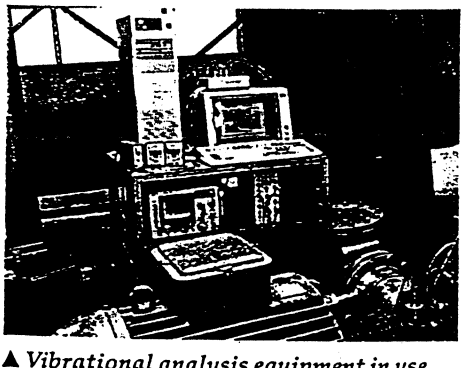
Accelerometer positions



MAKING MEASUREMENTS

A piezoelectric accelerometer was used to measure vibration at various monitoring points on the primary pumps, PW-P1 and PW-P2. It was attached to a heavy-duty magnetic base placed at the pump in-board and out-board bearing housing, and the motor front and end bearing casing. After conditioning, the vibration signal from the accelerometer was analysed with a PC-



▲ *Vibrational analysis equipment in use.*

Table 1. Main characteristics of PW-P1 and PW-P2

Type of pump	horizontally split, single-stage, double suction
Maximum discharge	690 m^3/h
Number of impeller vanes	6
Impeller diameter	343 mm
Motor rpm	1470
Power	100 HP

Table 2. Sensor and power supply specifications

Sensor type	Accelerometer
Maximum input	2000g
Sensitivity	100 mV/g (2%)
Frequency span	2 Hz-10 kHz
Amplitude linearity	1% / 1000g
Sensor excitation current	2 mA at +18 V (dc)

based vibration spectrum analyser (see photograph) and the power spectral densities (PSDs) or frequency spectra were plotted. The diagram on p41 shows the monitoring locations on the primary pumps. Pump and measuring equipment details are given in Tables 1 and 2 above.

The principal results from the vibration analyses were the measured PSDs. Preliminary investigation was made by measuring the root mean square (rms) of the vibration signal from the accelerometer. The rms values converted into velocity and displacement (Table 3) were compared with the vibration criteria given for medium machines on the *VDI 2056* chart, which revealed that the operating condition of both pumps was "not permissible". The reasons for their excessive vibration were further investigated by carrying out a frequency analysis of the vibration signal.

INTERPRETING THE DATA

The frequency spectra of the first run carried out on pump PW-P1 are shown opposite. The spectra in the frequency ranges of 0-100 Hz and 0-500 Hz show distinct resonances at 24, 48, 72 and 96 Hz etc, representing the rotational frequency of the motor and its second, third and fourth harmonics (N, 2N, 3N, 4N etc). The spectra in the frequency range 0-1000 Hz measured at the pump show impeller vane harmonics 6N, 12N, 18N and 24N in addition. These indicate that the pump is unbalanced and misaligned, resulting in a constant centrifugal thrust on the impeller vanes.

The spectra of PW-P1 were compared with those measured on PW-P2, which are shown opposite. The spectra from the first run on PW-P2 within the frequency range 0-500 Hz contains a single resonance at N (24 Hz) and harmonics of the impeller vanes. It is evident that pump PW-P2 was also unbalanced, but the cause of unbalance may not be misalignment.

FINDING THE CAUSES

Physical inspection of both pumps was carried out and revealed that the pump and the motor coupling of PW-P1 were misaligned. The misalignment had badly affected the Wood flexible coupling.

The unbalanced condition of PW-P2 was found to be due to baseplate bolts which needed tightening.

Repairs were made and further vibration

Table 3. RMS of vibration signals from the pumps

Run No.	Acceleration (g)	Velocity (mm/s)	Displacement (mm)	Condition
Pump PW-P1				
1	0.25	16.0	0.22	Not permissible
2 (after first repair)	0.11	6.5	0.09	Just tolerable
3 (after second repair)	0.05	3.0	0.04	Allowable
Pump PW-P2				
1	0.2	13.0	0.18	Not permissible
2 (after first repair)	0.08	4.5	0.07	Just tolerable
3 (after second repair)	0.06	3.9	0.05	Allowable

Sources and analysis of vibration

Common sources of failure in process machinery are unbalance, misalignment, excessive thrust load, bad bearings, bearing instability, excitation of resonances or critical speeds, and seal and coupling problems. Misalignment is an important single factor in machine failure and one aim of vibration analysis on primary pumps is to detect it.

■ Unbalance

Unbalance in rotating and reciprocating machinery is of particular concern in modern equipment, especially when there is a need to couple pump and motor. Balancing such a machine is essential to prevent excessive loading of support bearings, fatigue failure in associated supporting structures, and excessive noise and vibration, as well as to improve the durability and usefulness of the machine in service.

There are many reasons for machine unbalance, but the most important is misalignment of a shaft coupling or rotor. Generally an unbalance results in an increase in the vibration level of the machine and sometimes it also shows up as a change in the frequency spectrum and phase of the vibration signal. Since unbalance exerts a constant centrifugal force upon the impeller vanes of a pump, the frequency spectrum of the system may contain relatively higher harmonics of pump rotational frequency (rpm) multiplied by the number of impeller vanes such as N, 6N, 12N, 18N etc.

■ Misalignment

Misalignment is probably responsible for most machine failures. Pipeline stresses, deformity in foundations, thermal expansion and wrapping of machine casings can misalign a system. In this case, the shaft vibration is transmitted more effectively to the pump and motor bearing assemblies. The frequency spectrum taken at these locations may demonstrate strong resonances at the pump rotational frequency and its first, second and third harmonics (N, 2N and 3N).

■ Bearing instability

Possible bearing failure modes include constant excessive stress, misaligned loads, material flaws, lubricant failure and electrical discharge between balls and races. In any of these situations the malfunction manifests itself as a defect in a race or a ball. Such a failure can effectively be detected by frequency analysis of the vibrating signal obtained from the bearing housing, having peaks between 5 kHz and 8 kHz, depending upon the type of fault in the bearing.

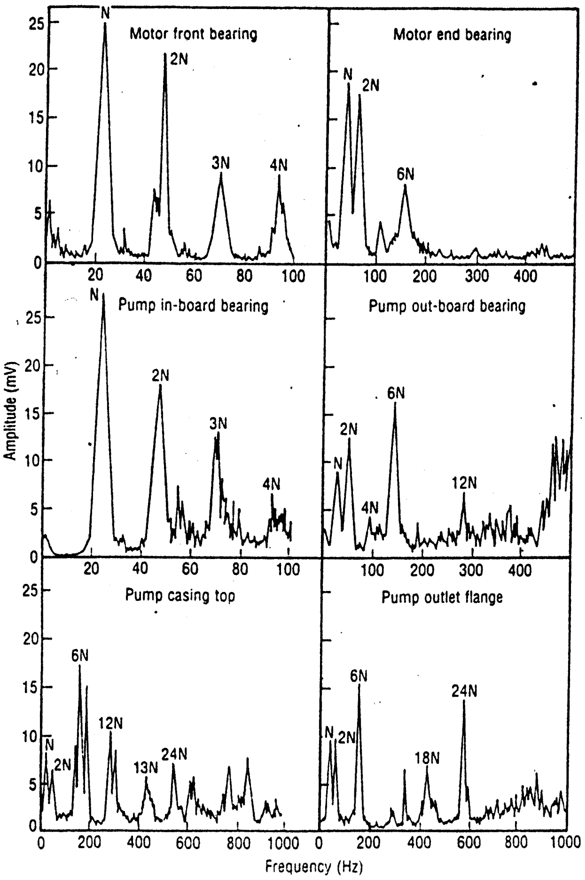
SETTING VIBRATION CRITERIA

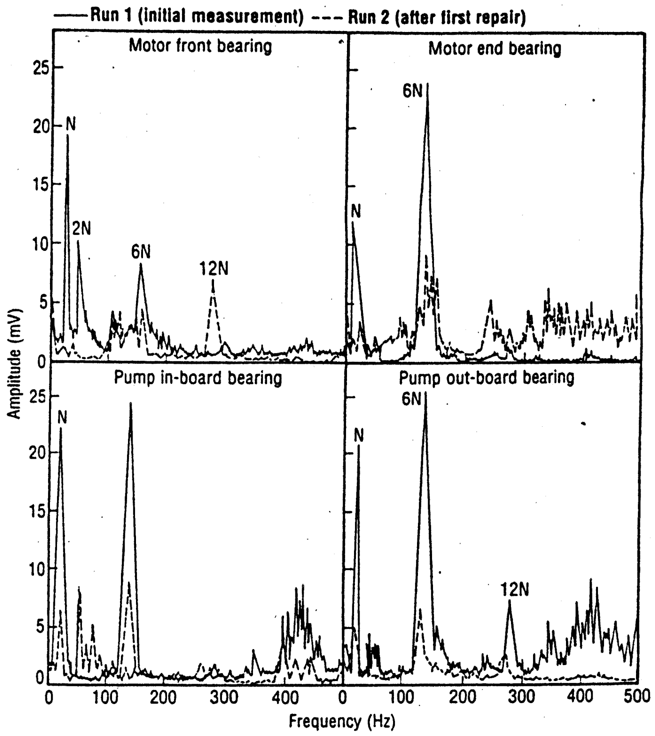
A machine can never achieve a perfect and complete balance owing to machine tolerance, mechanical play, misalignment, distortion and run-out. An appropriate balance quality or vibration tolerance is selected, which depends on the performance required from the machine and also the economics of the balancing process.

Operational criteria represented by vibration boundary levels for satisfactory and unsatisfactory running conditions vary and many depend on the type of machine. The criteria generally used to define machine health were set by Rathbone and Yates, and *VDI 2056*. The Rathbone and Yates criteria were obtained from experience of various types of machinery operating at less than 5000 revolutions per minute, followed by vibration level measurements with relatively crude instruments. *VDI 2056* is one of the most widely accepted vibration criteria for rotating and reciprocating machinery. It places no restriction on the type of vibration measured, but suggests that if the vibration signal is complex, containing several harmonic components in addition to the primary, then analysis of the complex signal is appropriate.

Practical evidence indicates that unless detailed information is available about a particular machinery system, the application of any criterion could be misleading. It is therefore important to obtain some knowledge of the frequency response of a machine to particular dynamic forces. The vibration frequency measured at various monitoring points on the system enables the vibration level to be computed and thus the system's response to dynamic forces under running conditions. This helps considerably in qualifying the relative severity of machine vibration.

Frequency spectra





▲ *Frequency spectra for pump PW-P2 vibration before and after first repair.*

◄ *Initial frequency spectra for pump PW-P1.*

measurements on the two pumps carried out. Measurements made on PW-P2 after repair are shown above right. It can be seen that the peak at 24 Hz diminished and only the hydrodynamic forces acting on the impeller vanes were represented by the harmonics of 6N etc. The rms value measured on PW-P2 indicated that pump vibration had reached the "just tolerable range".

Spectra recorded after repair of PW-P1 demonstrated that the misalignment of pump and motor coupling had reduced to a greater extent, the system was balanced and the magnitude of the vibration level had also decreased to within the "just tolerable" region.

The reasons for the levels of vibration still being too high in both pumps, even after corrective actions, were possibly bearing instability in PW-P1 and system excitation in general.

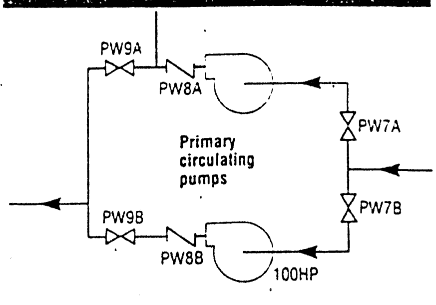
DETECTING BEARING INSTABILITY

Further vibration spectra for PW-P1 were measured. It was believed that the unbalanced and misaligned condition of the pump generated bearing noise because the magnitude of the vibration level at the motor front bearing and the pump in-board bearing was 40% higher than that at the motor end-bearing and pump out-board bearing. The spectra measured indicated the vibrational behaviour of the bearings after alignment of the motor and pump coupling shaft. They indicated that proper alignment of the system had considerably reduced the vibration level of pump in-board and out-board bearing and to some extent that of motor front-bearing. However, the general behaviour of the system was still not satisfactory because of the high level of vibration.

Further investigations were made by measuring the temperature of the motor front-bearing casing. The temperature rose more than 100°C after just two hours of operation. Close examination of all the frequency spectra measured revealed very high frequency components around 2-3 kHz. This indicated a complex system behaviour coupled to the multiple pipe structure, its supports and valves.

The position of the orifice at the outlet of check valves 9A and 9B is near to the common header of the two pumps (diagram below). At maximum flow rate, the high pressure water jet from the orifice strikes the common header, resulting in high vibration feedback to the pumping system, causing stresses on the bearings and adding vibration noise to the system frequency spectrum. It was therefore decided to replace the motor front bearing and to remove the orifice from check valve No 9A and install it at a position beyond check valve 8A. These changes reduced the vibration level of the pump effectively and minimised the load stress on the bearings, as shown in Table 3 (run No 3). The frequency spectrum again demonstrated the absence of high magnitude N, 2N and 3N, indicating that PW-P1 is properly aligned and that the dynamic forces which were acting on the vanes (6N, 12N and 24N) have been reduced to the "allowable" criterion. To conclude, the working condition of both pumps, based on the vibration criteria had become "allowable," so the work was complete.

Configuration of the primary pumps



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