VIBRATION SEVERITY

The most common solution to an inlet cone vortex is the addition of radial dorsal fins. A report published by L & C Steinmuller (50) on pressure pulsations in the discharge ducts for the draught group fans at Duvha revealed the following. Pressure pulsations in the discharge ducts were some 66% lower for fans fitted with dorsal fins than for fans without.
There is little or no connection between bearing vibration and duct vibration. At Tutuka Power Station excessive vibration existed in the ducting from the two ID fans. Vibration tests were performed by the author on the main vertical support channels, the cross struts and on the fans bearings. Figures 8.35, 8.36 and 8.37 show the frequency spectra that were obtained from one of the main vertical support channels, a cross strut and from the non-drive end bearing (horizontal direction), respectively. The overall RMS on the main vertical support channel was 2.83 mm/s, while on the cross beam it was as high as 10.1 mm/s. On the non-drive end bearing of the fan in the horizontal direction it was as low as 1.11. As can be seen large vibration exists in the duct work (10.1 mm/s RMS) with comparatively small scale bearing vibration (1.11 mm/s RMS). There is little structural tie between the two. If both the bearings and the ducts vibrate, then there are two problems to be solved.

8.6.6 Load Contours for PA Fans

Generally the horizontal overall RMS velocity values are higher than in the vertical direction for a particular bearing. This can be expected because the fan is less stiff in the horizontal direction than in the vertical direction.

If we consider only the normal operating conditions i.e. 72% to 94% MCR which corresponds approximately to the overall RMS velocity values at 80% and 100% MCR, then the percentage change varies from 24% to 0.1%. All the values obtained are less than 2 mm/s RMS except for the left hand PA fan, non drive-end horizontal position, whose values range from 2.36 mm/s to 2.32 mm/s RMS.
### M/S 0-P

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<thead>
<tr>
<th>Velocity (m/s)</th>
<th>Frequency (Hz)</th>
</tr>
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<td>16.251</td>
</tr>
<tr>
<td>568.E-6</td>
<td>22.501</td>
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<tr>
<td>1.40E-3</td>
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<tr>
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<tr>
<td>330.E-6</td>
<td>58.752</td>
</tr>
<tr>
<td>341.E-6</td>
<td>87.500</td>
</tr>
</tbody>
</table>

**MAIN VERTICAL SUPPORT COLUMN**

**FIGURE 8.35**
8.6.7 Load Contours for the ID Fans

For the ID fans tested the variation in overall RMS velocity values obtained are more obvious than for the two PA fans. No clear pattern emerges from these contour lines. The horizontal overall RMS velocity values for a particular bearing are higher than the vertical values.

If we consider only the normal operating conditions i.e. 72% to 94% MCR, then the percentage change varies from 25% to 0.8%.

All the values obtained are less than 3 mm/s RMS.

8.6.8 Load Contours for the FD Fans

The load contour lines for the FD fans following a similar pattern as the PA fans. Generally the horizontal overall RMS velocity values are higher than for the vertical, for each bearing.

All the values obtained are less than 3 mm/s except for the lefthand FD fan, drive end vertical position, whose values range from 4.56 mm/s to 2.12 mm/s RMS.

8.6.9 Load Contours for the Steam Feed Pump

The overall RMS displacements from the four permanently installed proximity probes all show a similar trend. Between 55% MCR and 75% MCR the vibration levels tend to fluctuate substantially. From 40% MCR to 55% MCR the vibration levels remain fairly constant and the same applies between 75% MCR to 85% MCR. The vibration levels from the two accelerometers
mounted vertically and horizontally on the drive-end of the turbine fluctuate considerably.

From 40% MCR to 55% MCR the main pump is not supplying the boiler with water and operates on leak off. The speed of pump during leak off only varies from 3180 RPM to 4055 RPM. At 80% MCR and 85% MCR the pump operates at a speed of 4205 RPM and 4311 RPM respectively.

Taking the percentage change of the maximum and minimum vibration levels at 75%, 80% and 85% MCR for each contour, that is, the normal operating conditions, results in the following:- The vibration levels from the drive end x and y proximity probes have a percentage change of 6.6% and 10.4% respectively. The vibration levels from the non-drive end x and y proximity probes have a percentage change of 20.3% and 18.7% respectively. The vibration levels from the accelerometers mounted vertically and horizontally on the drive-end of the turbine have a percentage change of 27.7% and 20% respectively.

8.6.10 Load Contours for the Electric Feed Pumps

The vibration levels from the four permanently installed proximity probes do not fluctuate substantially. For Electric Feed Pump A (EFP A) the overall RMS displacements from the drive end and non-drive end proximity probes vary between 8.5 micrometers and 12.6 micrometers RMS. For EFP B the overall RMS displacements from the drive end proximity probes vary between 18.4 micrometers and 19.2 micrometers RMS, while the non-drive end displacements vary between 3.2 micrometers and 5.7 micrometers RMS. The vibration levels from the two accelerometers mounted axially at the drive-end and non-drive-end of main pump, fluctuate substantially.
Taking the percentage change between the maximum and minimum vibration levels at 75%, 80% and 85% MCR for each contour results in the following:

Electric Feed Pump A

The vibration levels from the drive-end x and y, and non-drive end x and y proximity probes have a percentage change of 11.8%, 1%, 1.2% and 1.2% respectively.

Electric Feed Pump B

The vibration levels from the drive-end x and y, and non-drive-end x and y proximity probes have a percentage change of 2.2%, 2.1%, 17.6% and 49.6% respectively.

The vibration levels from the axial drive end and non-drive end accelerometers have a percentage change of 5.5% and 22.4% respectively.
When formulating guidelines necessary to perform vibration trend analysis two approaches must be considered. The first approach is to formulate unique guidelines for each machine. This approach accounts for the fact that every machine exhibits a unique vibration signature. As one can appreciate this approach is time consuming and is only economically feasible for machines which are critical to the continual operation of plant. The second approach is to formulate general guidelines for different classes of machines. For example, a guideline for fans, pumps, turbines and motors.

This approach is practical in the sense that only a few guidelines exist and not one for every machine. The latter approach was used in this discussion.

It is normal practice when formulating guidelines for trend plots to measure the initial vibration levels and typically allow level changes of a factor of 2.5, that is, 8 dB, for one equality class change. This only holds provided that the initial vibration level is classified as normal in accordance with reputable standards like VDI 2056. The same criteria applies to spectrum cascade trending.

It is therefore pointless and meaningless to trend vibration data that fluctuates more than 250% due to load variations. In this situation it will be impossible to distinguish between fluctuations caused by load variations or increased vibration levels caused by an impending problem.

Judging by the results obtained from all the draught group fans the maximum and minimum percentage change was 51% and 0.1% respectively over the 80% MCR to 100% MCR operating
range. The average percentage change being 14%. For the electric and steam feed pumps the maximum and minimum percentage change was 49.6% and 2.1% respectively over the operating range from 75% MCR to 85% MCR. The average percentage change being 12%. It can therefore be deduced that sensible vibration trending can be performed for the abovementioned machines at the specified operating conditions. When analysing trend data from these machines the fact that up to 50% variation in vibration levels can occur due to load fluctuations, must be borne in mind.

In an article published by Piety and Magette (51) it is demonstrated that variations in speed and load can introduce changes in vibration exceeding those associated with defects. Compensation for such changes stimulated by control variables is necessary for maintaining sensitivity for defect detection and for reducing false alarms. To detect these anomalies they suggest that each vibration signature be tested to determine if its deviations from the baseline signatures are statistically significant.

To recommend a time interval between the acquisition of vibration measurements so that an acceptable lead time before failure can be achieved requires knowledge of the following:

a) The time period between the commencement of a defect to the detection of this defect using vibration analysis techniques.
b) The time period between the commencement of a defect to this defect resulting in catastrophic failure.

c) The time period between the detection of a defect using vibration analysis techniques to this defect resulting in catastrophic failure.

It is very difficult to quantify these lead times because:

a) They will be very dependent on the initial severity of the defect.

b) They will also depend largely on the pertinent failure mechanisms common to the machine.

c) In some cases defects will not necessarily be detected at the onset by vibration analysis techniques alone. Other parameters like seal gas pressure and temperature could indicate the commencement of a defect well before vibration analysis.

d) It is difficult to ascertain when a defect commences, when it will be detected and when this defect will result in catastrophic failure.

The abovementioned lead times could be obtained from well documented and detailed maintenance history files on various machines. Unfortunately, because vibration condition monitoring is relatively new in ESCOM this information is not available.

If the literature on this subject is consulted and from the authors’ experience at SASOL it is recommended that the feed pumps and draught group fans be monitored once fortnightly.
If the trend plots show an increase or decrease then the time between measurements should be decreased so that the defect can be more closely monitored.

Vibration maintenance history files for the abovementioned machines must be kept in view of setting a new or more practical time period between vibration measurements, if necessitated by defects which commence and result in a failure between vibration measurements.

After discussing the pertinent failure mechanisms of the feed pump train and the draught group fans it is possible to recommend a system whereby minimal vibration data acquisition can be performed for acceptable protection against the pertinent failure mechanisms. Generally, the direction that is monitored on a bearing, is that direction in which the machine is the least stiff. For example, turbo-generators are monitored continuously in the vertical direction, while draught group fans are monitored continuously in the horizontal direction.

The fact that the feed pumps have large casing/rotor mass ratios and hence generate more shaft motion relative to the bearing or bearing housing. It is for this reason that shaft relative motion, using proximity probes, be measured. Having discussed the pertinent failure mechanisms of the feed pumps most of the problems manifest themselves in shaft relative movement.

The draught group fans have small casing/rotor mass ratios, and hence most of the vibration will be transmitted from the rotor to the bearing casing. It is therefore suggested that shaft absolute motion for the fans be monitored.
The following systems are recommended for:

A The Draught Group Fans

Vibration measurements should be taken once fortnightly in the horizontal direction at the drive-end and non-drive-end bearings of the motor and fan. Vibration measurements should also be taken in the axial direction at the drive end bearings of the motor and fan. i.e. on either side of the coupling. When horizontal measurements are taken on the fan, these should be taken on the side opposite to which the permanently installed velocity transducer is mounted. Since the periodic measurements will be taken using a velocity transducer with a magnetic base, this magnet could trip the fan if it is brought close to the permanently installed velocity transducer.

By taking measurements in the axial direction on either side of the coupling, misalignment can immediately be detected. Rotor bar defects in the motor will also be detected by measuring in the axial direction of the motor. This is discussed in Chapter 6.

To improve the ability to diagnose defects the following information should also be recorded and trended each time vibration measurements are taken.

   a) Bearing temperatures
   b) Stator current

The FD and ID fans have double inlet impellers with 12 staggered blades for each inlet, and the PA fans have single inlet impellers with 10 blades each. The running speed of
the FD and ID is 12.33 Hz and the PA fan's running speed is 24.83 Hz. The blade passing frequencies are (12 x 12.33) and (24 x 12.33) = 148 Hz and 296 Hz for the ID and FD fans and 10 x 24.83 = 248 Hz for the PA fans. When investigating frequency spectra from the draught group fans a minimum frequency range of 1000 Hz is recommended.

B The Steam Feed Pump

The Main Pump

Vibration measurements should be taken once fortnightly from the 4 proximity probes mounted ± 45° from the vertical at the DE and NDE bearings. Phase measurements from the fifth proximity probe should also be taken, so that phase and shaft orbital analysis can be performed. Axial vibration measurements on the main pump should also be taken using a velocity transducer but preferably an accelerometer. The main pump has 7 vanes per impeller and has a maximum operating speed of 5 228 RPM (87 Hz) and hence its blade passing frequency is 610 Hz. When investigating frequency spectra from the main pump a minimum frequency range of 2 000 Hz is recommended. This frequency range has to be specified but can be changed after some experience has been gained. It is more convenient to have excess information and then reduce it than to have insufficient right from the outset.

The rest of the steam feed pump train

The Steam Turbine

The turbine should be monitored in all three directions at both the DE and NDE bearing with either a velocity transducer or an accelerometer but preferably an accelerometer.
The Steamer's Gearbox

The steamer's gearbox should be monitored in the horizontal and axial direction at the DE and NDE bearing. This is to protect the gearbox from misalignment and gear problems.

C The Electric Feed pump

The main pump should be monitored in the same manner as the steam feed pump.

The rest of the electric feed pump train.

The Voith Gearbox

The 6 journal bearings should be monitored in the horizontal direction, and the 2 thrust bearings should be monitored in the horizontal and axial directions.

Additional parameters that should be monitored on the feed water pumps.
Other parameters that should be monitored and trended each time vibration measurements are recorded on the feed water pumps are:

a) Suction pressure.
b) Discharge pressure.
c) Speed.
d) Load.
e) Lubrication oil temperature.
f) Bearing temperatures.
g) Sealing water temperature.

The abovementioned information is available as a hard copy as shown in Figures 8.38 and 8.39 for the steam and electric steam feed pump respectively.
FIGURE 8.38
FIGURE 8.39
8.8 CONCLUSIONS

1) The vibration levels obtained from the draught group fans at 80% MCR and 100% MCR, have an average percentage change of 14%, a maximum of 51% and a minimum of 0.1%.

2) The vibration levels obtained from the main feedwater pumps, between the ranges 75% MCR to 85% MCR, have an average percentage change of 12%, a maximum of 49.6% and a minimum of 2.1%.

3) Sensible vibration trending can be performed for the above-mentioned machines and specified operating conditions. When analysing overall vibration trend data from these machines an average percentage change of 12-14% and a maximum percentage change of 50% can be expected.

4) To deduce conclusively, for all pumps and fans, what has already been stated above requires similar tests to be performed on numerous fans and pumps of different complexities and load capabilities.

5) To recommend a time interval between the acquisition of vibration measurements so that an acceptable lead time before failure can be achieved, requires knowledge of the maintenance history of the machine. Although the vibration maintenance history of the feed water pumps and draught group fans was unknown a reasonable time interval was recommended.

6) To recommend a procedure whereby minimal vibration data acquisition can be performed requires knowledge of the pertinent failure mechanisms of the machine.
7) Baseline signatures at various loads for the machines in question were obtained.

8) To ensure that vibration data is acquired each time at precisely the same location and orientation on a machine, vibration pads or studs should be attached to the bearing housing. These pads or studs should have an identification number engraved on them for recording purposes.
OTHER IMPORTANT ASPECTS OF A VIBRATION CONDITION MONITORING PROGRAMME

These include:

a) The training requirements of the maintenance personnel.

b) Continuous versus periodic monitoring.

c) Trending of vibration data.

d) Manual and computerised logging.

e) Considerations concerning the purchasing of periodic vibration monitoring equipment.

f) Considerations concerning the purchasing of continuous on-line vibration monitoring equipment.

g) Standardised rules for measurements on rotating equipment.

9.1 TRAINING FOR VIBRATION CONDITION BASED MAINTENANCE

9.1.1 Selection of Personnel

The selection of personnel to carry out a Vibration Condition Monitoring (V.C.M.) system is of paramount importance. They must be dedicated, technically competent, self-disciplined, credible, logical, capable of writing clear and concise reports, and above all, they must not be put off by opposition.
The unbelievers, as with any religion, are capable of producing a formidable opposition. Comments like: "I have run my plant for years without vibration condition based monitoring, why do I need it now?" are regularly heard. This attitude can be overcome by careful persistent logic and proving the point with results: i.e. gaining credibility.

Management must not expect to reap the full benefits of a V.C.M. programme immediately. It is the author's belief that the full economic benefits of such a programme will only become evident two years after the implementation of the programme. Hence, the maintenance staff must be given enough time to gain experience and confidence, and to become proficient in the use of the instrumentation.

The best areas of choice for the maintenance personnel who will perform the V.C.M. lay in the skills band covering a good engineering assistant to an engineer, preferably with machinery experience. The simple reading of a measurement and its interpretation according to predetermined vibration severity charts and tables is not sufficient and often causes evaluation errors. It is therefore necessary that the operator knows how the machines are manufactured, the history of the machines, the more recurring failures, how these reveal themselves within the vibration frequency spectrum and how they have been repaired.
9.1.2 Training

It is the author's view that the training of personnel is best achieved by:

a) **In-house courses in basic vibration analysis techniques and measurements.**

Some vibration analysis equipment manufacturers provide basic courses in vibration analysis and as such can be a very valuable asset to initiating a V.C.M. programme. Unfortunately these courses can become very sales orientated and hence lose their effectiveness as a training programme.

It was therefore decided that ESCOM would provide its own in-house training. The necessary expertise for this training programme was supplied by personnel from Dynamics & Noise and the Central Maintenance Services (CMS).

b) **Continuous on-the-job training.**

This part should involve the use of experienced personnel in guiding, and encouraging the operators at the power stations. It is during the early days of launching a system that most mistakes are made and problems encountered. It is vitally important that the operators be able to rely on specialised groups such as Dynamics & Noise, and CMS for assistance in solving their problems.
c) **Advanced training.**

The purpose of an advanced training course is threefold:

1) To obtain feedback so that the basic vibration analysis course can be refined.

2) To confirm all the points that have been taught in the basic and on-the-job training courses.

3) To upgrade knowledge concerning vibration analysis techniques and measurements by going into more detail concerning diagnostics.

d) **Managerial awareness**

The maintenance superintendents and the power station managers are now involved in problems which were quite unknown or did not concern them. It is therefore vitally important that the abovementioned personnel are aware of the benefits and advantages of a V.C.M. programme.

Irrespective of the sophistication of the vibration monitoring equipment and the comprehensiveness of the training courses a V.C.M. programme will not succeed unless the maintenance personnel involved are motivated and dedicated, and top management support the programme.

9.2 **CONTINUOUS VERSUS PERIODIC MONITORING**

On-line vibration monitoring systems must satisfy certain requirements if they are to provide an effective means of detecting and diagnosing abnormal vibration characteristics for large turbo-generators, feed pumps and draught group fans.
With the advent of computerised real-time systems, vibration monitoring lends itself to continuous on-line-monitoring.

Continuous monitoring requires a relatively large initial capital expenditure, but once installed, cost of operation is quite low. Periodic monitoring on the other hand has a low initial cost, but it is manpower intensive and therefore has a relatively high continuing cost.

9.2.1 Continuous Monitoring

Continuous vibration monitoring is necessary on critical machines that are subject to problems, or where problems can develop rapidly and have severe financial consequences. Continuous monitoring may be dictated by safety considerations. Even if the cost of a failure is small, machines should be continuously monitored if a failure will result in hazards to personnel.

The output of a continuous machinery monitoring system must be highly representative of the machine's condition. It must also be responsive enough to provide warning of impending problems in time to avoid major catastrophic failures.

Continuous monitoring systems should also allow for the analysis of transient vibration data, during the run-up and coastdown of machines so that Bode and Nyquist plots can be generated. This is suggested on the main turbo-generator set for crack detection purposes.

Permanently installed velocity transducers on a FD fan and the turbo-generator can be seen in Figures 9.1 and 9.2 respectively.
Continuous vibration monitoring involves mounting permanently installed accelerometers, velocity probes, proximity probes, dual probes or shaft riders to a machine. The information from these transducers is relayed continuously to a monitor or strip chart recorder or on request in a control room. Typical strip chart recorders can be seen in Figure 9.3.

9.2.2 Periodic Monitoring

Periodic monitoring is typically applied to less critical machinery where advance warning of deteriorating conditions will show a positive return on investment. Periodic monitoring consists of logging measurements at pre-determined intervals using portable hand held meters or portable data collectors. Typical analysis of the data ranges from relatively simple manual logging of overall RMS values to detailed frequency spectrum analysis of dynamic data from machines.

Signals from permanently installed sensors for continuous monitoring can be used for periodic trending. Most permanently installed sensors, for various machines, are connected to a common monitoring panel. This panel consists of individual modules from each of the sensors. These modules normally have buffered outputs whereby signals from the sensors are spectrum analysed and compared against previous baseline signatures using a portable data collector. A typical control box for permanently installed velocity transducers on the draught group fans can be seen in Figure 9.4.
CONTROL BOX FOR PERMANENTLY INSTALLED VELOCITY TRANSDUCERS ON THE DRAUGHT GROUP FANS

PERMANENTLY INSTALLED VELOCITY TRANSDUCER ON DE BEARING OF THE DRAUGHT GROUP FANS
FIGURE 9.3

STRIP CHART RECORDERS IN CONTROL ROOM FOR THE VELOCITY TRANSDUCERS ON THE MAIN TURBO-GENERATOR SET.

PERMANENTLY INSTALLED VELOCITY TRANSDUCER USED FOR TURBO-GENERATOR
The method of acquiring data used for periodic vibration monitoring takes on many forms:

a) A semi-skilled labourer armed with a hand held vibration meter coupled to an accelerometer or a velocity transducer, obtains peak, RMS or overall RMS values from the transducer when it is mounted on an established location and orientation on a bearing casing. The measurements are usually manually logged. The measuring points are usually marked on the machine to ensure that measurements will correlate with previously obtained data. Trending of data is also done manually.

b) An operator is provided with a portable tape recorder and he simply speaks an identification number, location and the measured value into the microphone. Later the tape is played back and the spoken information transcribed onto a permanent record.

c) A tape recorder or data collector is taken to a machine and the vibration spectra from the bearings are recorded or stored in a bubble memory. The data is then downloaded to a mass storage device or computer. If a computer is used, the storing, trending and signature comparison is all performed by the software. This method can be seen diagramatically in Figure 9.5.

Disadvantages of the abovementioned methods. The letters in brackets denotes the method which is applicable.

1) Labour intensive (a).

2) The job becomes routine and monotonous. When this occurs mistakes creep in and interest quickly wanes (a), (b), (c).
CONFIGURATION OF ROUTE

DATA COLLECTER

IEEE

HOST COMPUTER

PRINTOUT OF ROUTE THROUGH PLANT

DATA COLLECTION

BEARING HOUSING

DATA ANALYSIS

LIST OF ALL POINTS THAT HAVE EXCEEDED THE PRE-SET ALARM LEVELS

FIGURE 9.5
3) Efficient trending and analysis of the data becomes increasingly difficult as the data base expands. As the workload increases the time spent correlating and trending data decreases so important aspects of a machines' health is overlooked (a).

4) No detailed diagnosis or trouble shooting can be performed by just examining a trend plot of peak or overall RMS values (a), (b).

Effective monitoring systems contain a combination of continuous and periodic monitoring. Although it may sound absurd, it is the author's view that a significant portion of a machine population does not justify any form of condition monitoring. For example, machines which operate for long periods of time between failure, and whose failures have little or no effect on production and can be inexpensively repaired, do not justify the cost of condition monitoring.

9.2.3 Trending of Vibration Data

Trending of vibration data is one of the most important aspects of a condition monitoring programme. Historic vibration measurements are an excellent reference for severity measurements, because they are specific to the machine in question. While discrete frequency vibration levels for a machine may not be known there is a high probability that large changes in vibration level indicate a problem.

The process of monitoring vibration levels or spectrums for changes is referred to as trend analysis. Since vibration levels in a good machine are variable, it is not always
obvious how much change is tolerable. A typical trend plot can be seen in Figure 9.6 below.

![Trend Plot](image)

9.2.4 Overall Level Monitoring

Irrespective of the complexity of the vibration, overall monitoring expresses the broad level of vibration at a measuring point. It is the common first line of approach to vibration monitoring and provides basic information for trend monitoring.
Overall RMS is defined at the "energy" present in a frequency spectrum.

Mathematically overall RMS = \[ \sum_{x_i=1}^{N} \frac{x_i^2}{N} \]

Where \(x_i\) = some frequency component in a frequency spectrum
\(N\) = total number of values

9.2.4.1 Advantages of Overall Level Monitoring:

a) This method of trending can be used by inexperienced personnel.

b) The equipment used to obtain the overall levels is cheap and compact. It normally consists of a hand held vibration monitor meter coupled to an accelerometer or velocity transducer.

c) Outputs from permanently installed transducers can also be used to obtain overall RMS values.

d) It is only effective in detecting elementary defects such as unbalance, misalignment and mechanical looseness.

e) Minimal data logging is required. Only one value per measuring point is required.

f) Interpretations and appraisals of the data can be based on well established condition acceptability standards.
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