



## **Vibration Analysis of Cooling Tower Cell D**

### Abstract

Analyze and balance, if necessary, the D cell in a cooling tower. The preliminary data indicated that unbalance was not the cause of high vibration observed on the gearbox vibration monitors. Further analysis showed that the vibration was due to a combination of sources generating a high overall vibration. The overall vibration measured on the D cell was 0.2 to 0.3 inches per second. The measurements were taken on the gearbox in the radial direction.

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## Scope

To analyze and balance, if necessary, the D cell in a cooling tower. The scope was expanded to include recording similar data on the B cell for comparative information.

## Background

Personnel stated that continuous gearbox failures have occurred since installation of the cooling tower in 1991. The gear box failures are not restricted to any one cell. The most predominant failure has been the lower thrust bearing in the gearbox. Erratic amplitude readings have been observed on the gearbox monitors by operations personnel. The pitch on the blades has been increased from the factory recommended 10° to 15°.

## Machine Configuration

Foundation Type: Wood / Structural Steel

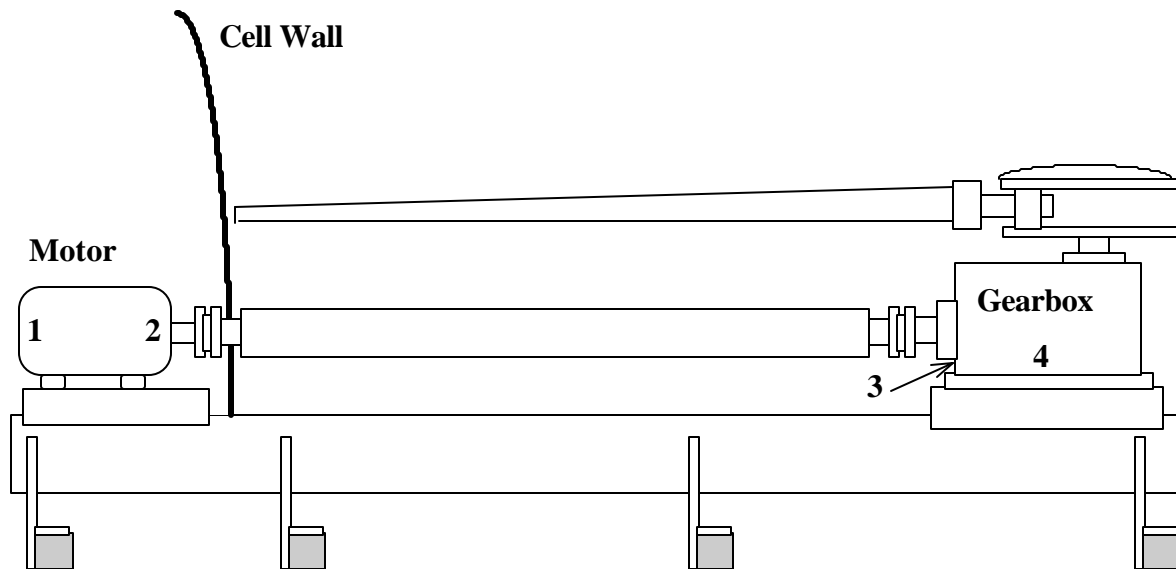
Fan Type: Axial

Number of Blades: 10

Fan Speed: 138 rpm / 2.3 Hz

Motor Speed: 1785 rpm / 29.75 Hz

Motor Horsepower: 250



**Figure 1: Machine Configuration**

## Procedure

The fan structure and rotor were visually observed to check for structural cracks, material buildup on rotor, and any other observable problems.

A low frequency transducer was placed in the radial direction on the gearbox at position 4 (see figure 1), and a standard accelerometer was placed in axial direction on the gearbox at position 3 (see figure 1). Both transducers utilized a magnetic base.

The fan was started and vibration measurements were recorded and analyzed on the fan and motor (see Figure 1). Data was recorded on both the D and B cells.

The fan in D cell was not balanced.

Plant personnel provided defect frequency data, and fundamental gearbox data to facilitate the analysis.

## Results

The visual observation revealed mild material build up on the underside of the blades, and wear in the form of pitting on the leading edges. Loose fasteners were observed connecting the venturi wall sections. The weep holes were clear of obstructions, and the tracking of the blades was < 1.0 inch. The measured pitch at the end of the blades was approximately 13° to 15°

The highest vibration recorded on the D cell gear box was 0.10 inches per second (ips) at 28.0 Hertz (Hz) in the axial direction at position 3. The highest vibration recorded on the D cell motor was 0.20 ips at 30.0 Hz in the horizontal direction at position 1.

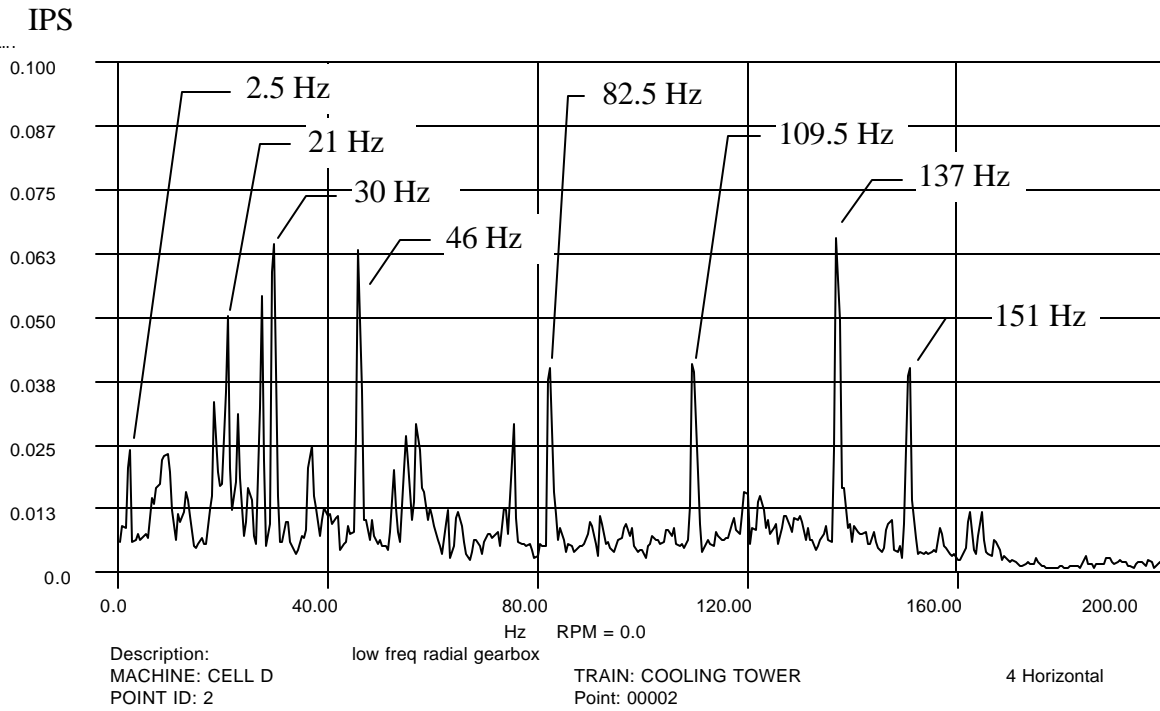
The highest vibration recorded on the B cell gear box was 0.05 inches per second (ips) at 30.0 Hertz (Hz) in the axial direction at position 3. The highest vibration recorded on the B cell motor was 0.35 ips at 30.0 Hz in the horizontal direction at position 1.

Vibration at the running speed of the fan (2.3 Hz) was 0.03 ips on the D cell.

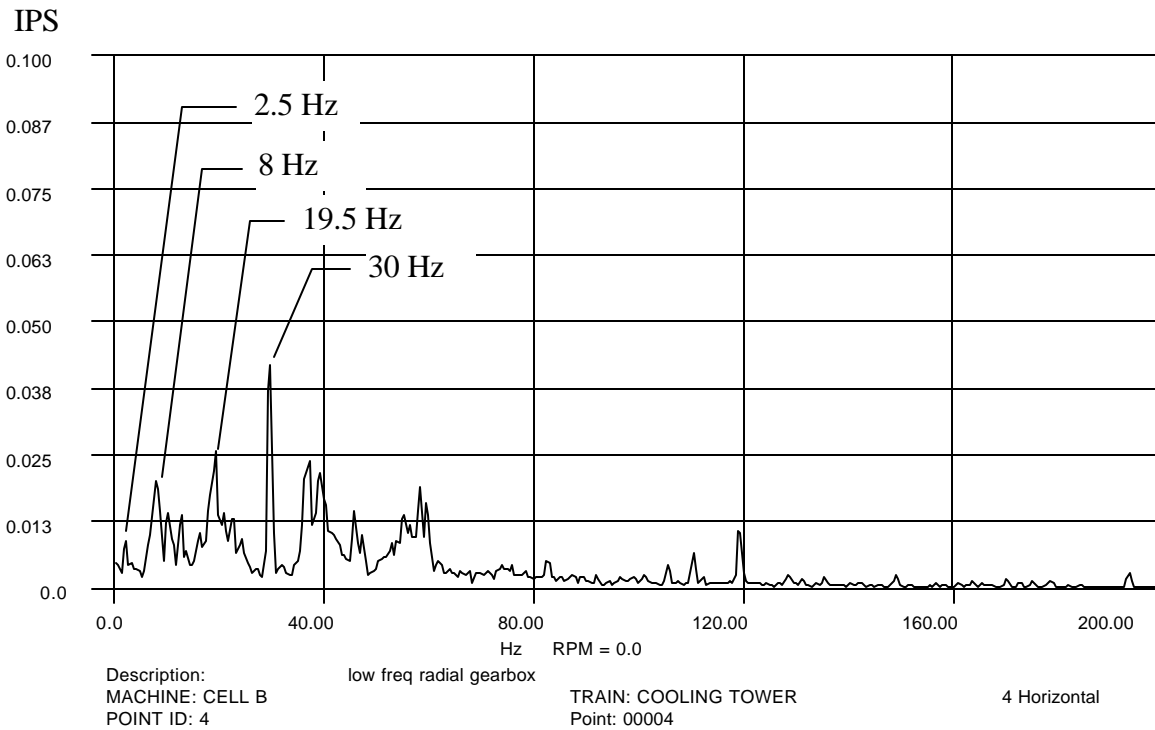
Vibration at the running speed of the fan (2.3 Hz) was 0.02 ips on the B cell.

The time waves were of the multiple frequency type, not the singular frequency type found with unbalance.

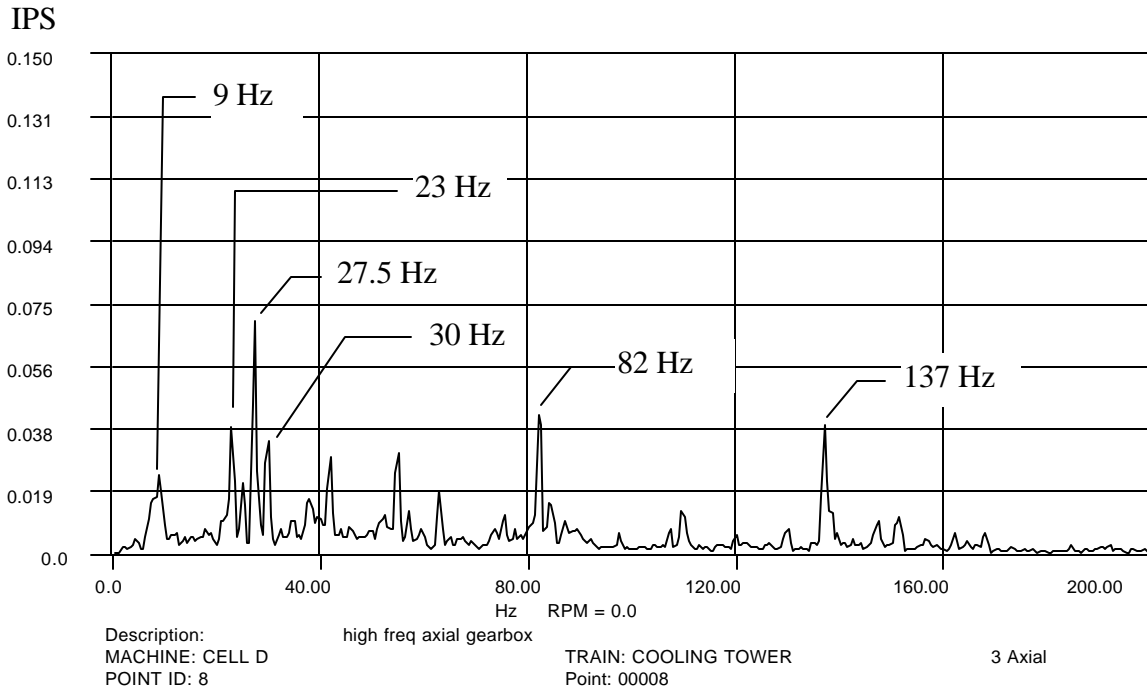
**Other vibration data is located in Appendix B.**



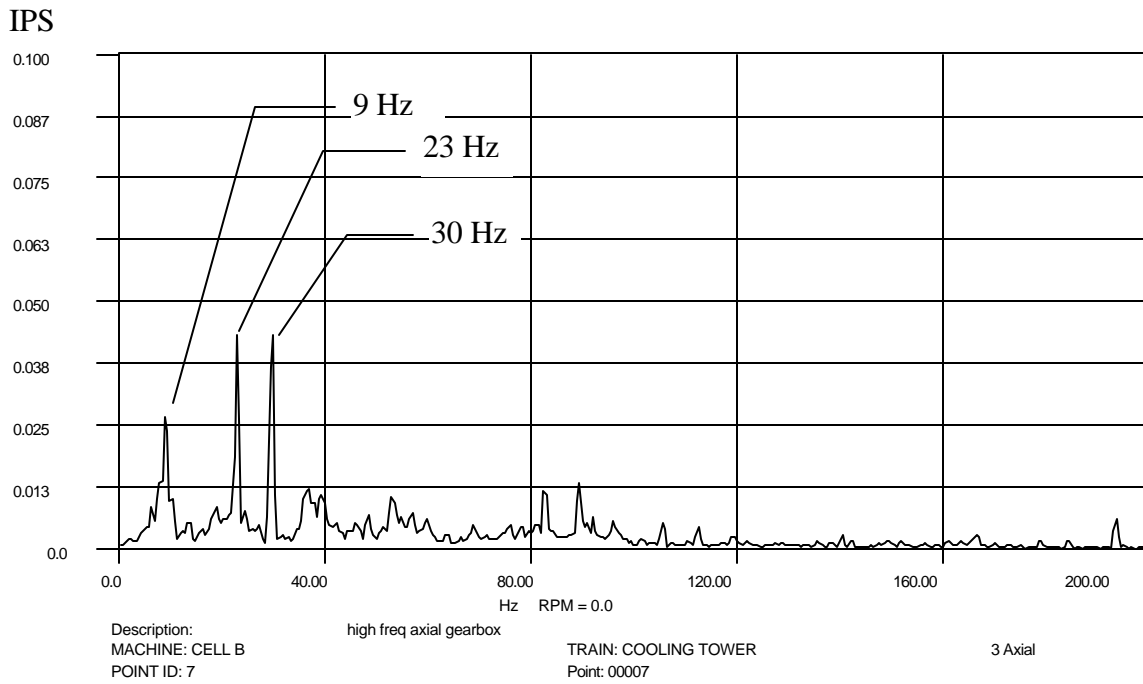
**Figure 2: Vibration Spectrum D Cell (Low Frequency Transducer)**



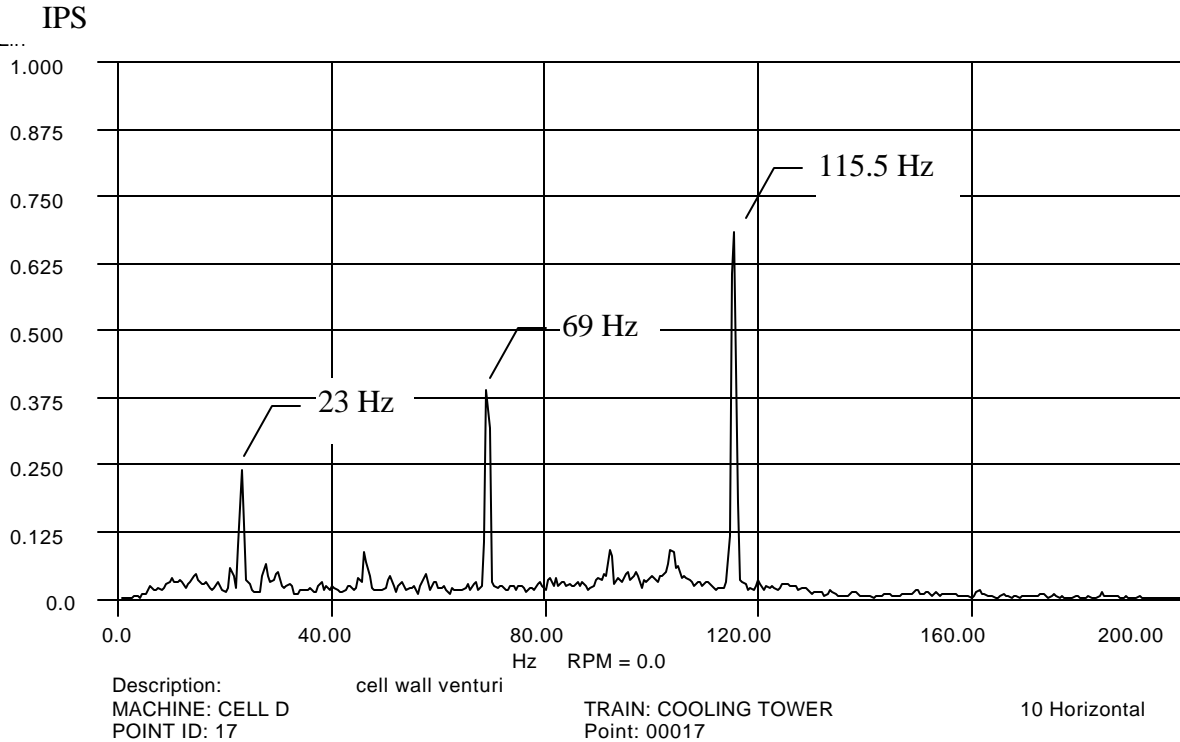
**Figure 3: Vibration Spectrum B Cell (Low Frequency Transducer)**



**Figure 4: Vibration Spectrum D Cell (High Frequency Transducer)**



**Figure 5: Vibration Spectrum B Cell (High Frequency Transducer)**



**Figure 6: Vibration Spectrum D Cell (Venturi Wall)**

## Conclusions

The low frequency spectrum recorded on cell D (see figure 2/pg. 3) shows that the running speed of the fan (2.5 Hz) is low in amplitude compared to other frequencies in the spectrum. The dominant frequencies are the running speed of the motor (30 Hz), 2X the blade pass (46 Hz), and 5X the BPF1 on the lower bearing in the gearbox (137 Hz).

The high frequency spectrum recorded on cell D (see figure 4/pg. 4 ) shows that the dominant frequencies are the BPF1 on the lower bearing in the gearbox (27.5 Hz), 3X the BPF1 on the lower bearing in the gearbox (82 Hz), and 5X the BPF1 on the lower bearing in the gearbox (137 Hz).

The spectrum recorded on the venturi wall on cell D (see figure 6/pg. 5) shows the dominant frequencies are the blade pass and harmonics. The amplitude of 115.5 Hz is greater than 0.50 ips.

The data collected on the motor of the D cell shows a peak at the running speed of the motor and amplitude of 0.09 ips. No defect frequencies were observed.

The data collected on the motor of the B cell shows a peak at the running speed of the motor and an amplitude of 0.24 ips. The time wave is of the uniform singular frequency type normally associated with unbalance or non restrained looseness.

The major cause of vibration on the D cell of cooling tower PB-3302 is not unbalance. It is probably due to loose venturi panels and structural components, gearbox bearing defects, and sympathetic vibration excited by the other cells. All of these vibrations combined generated an overall vibration level of approximately 0.30 ips on the gearbox in the radial direction.

It was observed that the axial monitor for the D cell displayed a vibration amplitude of more than 0.20 ips while the fan was not operating. This could be due to transducer and monitor electrical problems, possibly a ground loop situation. This could also explain some of the erratic readings observed by CHE personnel.

In comparison the D cell spectrums show peaks associated with the gear box defect frequencies, the B cell does not. The overall vibration amplitudes are similar. The highest vibrations recorded on both cells were on the motor.

The manufacturer's recommended initial pitch on the fan blades is 10°. Plant personnel increased the pitch to 15° to increase flow. Due to the fact that the thrust created by the induced draft axial flow fan assists gravity there is a possibility of overloading the lower thrust bearing. One way to determine this is to analyze the forces generated by thrust and the forces from the weight of the rotor assembly, and compare them with the published load capabilities of the bearing at the operating speed.

## Recommendations

The cooling tower should be monitored on a routine basis as part of a regular predictive maintenance program.

The blades should be periodically checked for material buildup and cleaned if necessary.

The loose venturi panel fasteners should be tightened and any missing fasteners replaced.

The venturi panels should be inspected and repaired as necessary.

All structural fasteners should be checked for proper torque and fit.

The welds on the mounting brackets of the drive train/ gearbox structure should be cleaned and checked for cracks and severe corrosion.

The vibration monitoring system should be examined for electrical problems and proper transducer selection.

The loading of the lower thrust bearing should be verified to be within the bearing manufacturers published capacities.

## Appendix A (Instrumentation Used)

IRD 890 Data Collector Analyzer

IRD 970 Accelerometer

IRD 580 Low Frequency Transducer

Miracle Point Inclinator

ENTEK Emontior Analysis Software

## Appendix B (Vibration Data)

