

COOLING TOWER MANUAL



CHAPTER 10

MECHANICAL COMPONENTS FOR COOLING TOWERS

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CHAPTER 10

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TABLE OF CONTENTS

	Page
10.1 Scope	10.3
10.2 Vibration	10.3
10.3 Balancing	10.4
10.4 Fatigue Failure	10.7
10.5 Fans	10.7
10.6 Platforms and Walkways	10.9
10.7 Gear Drives	10.9
10.8 Lubrication	10.13
10.9 Pumping Horsepower and Energy Savings	10.13
10.10 Drive Shafts and Couplings	10.21
10.11 Valves	10.31
10.12 Equipment Removal Systems	10.31
10.13 References	10.31

10.3

10.1 SCOPE

The mechanical equipment in a cooling tower is subjected to a harsh environment and usually a less than ideal inspection program. This chapter deals with the mechanical equipment for a cooling tower, associated equipment problems, and the different ways of access to the equipment for inspection and maintenance.

10.2 VIBRATION

One way rotating equipment can communicate their condition or health is by its mechanical motion or "vibration". By recording and analyzing this motion, a high percentage of problems can be accurately recognized.

10.2.1 DEFINITIONS

MILS	—	0.001 Inch
CPM	—	Cycles per Minute
g's	—	32.2 feet per second ²
Hz	—	Hertz (cycle per second)
RPM	—	Revolutions per Minute
RF	—	Resonant Frequency
BPF	—	Blade Pass Frequency
PWL	—	Sound Power Level
W	—	Weight of Rotor in Pounds
D	—	Vibration Displacement in Inches - Peak to Peak
R	—	Radius at Which Trial Weight is Added in Inches
O	—	Original Vibration Data
O+T	—	Original Vibration Plus Trial Weight Effect
T	—	Trial Weight Vibration Effect
TW	—	Trial Weight

10.2.2 Instrumentation

Many vibration analyzers are available for field testing work. The following is a list of requirements and options for field balance work:

Requirements

1. Amplitude readout in displacement (mils) and velocity (in./second).
2. Filter out plus filtering capabilities to highest reasonable frequency (i.e., 60 gear teeth x 300 rpm = 18,000 cpm).
3. Vibration pickup linear in range of possible frequency (i.e., 300 cpm (5HZ) to 18,000 cpm (300HZ)).

Options

1. Amplitude readout in displacement (mils), velocity (in./second) and acceleration (g's).
2. D.C. and raw outputs for strip chart recorder and raw data tape recording.
3. Transducers - Accelerometer type for mid to high frequency range. Velocity Seismoprobe Type for low frequency range. Transducer selection should be suitable for range of frequencies to be measured.
4. Battery operated.
5. Strobe light capability.

10.2.3 Vibration Data Collection Procedure

Most data for vibration analysis on a routine basis can be done outside the shroud on the motor. Data acquisition should be as follows:

1. Run motor until unit is warm and stable before taking reading. Readings may be taken as unit warms for added information, but final reading should be taken, if possible, at loaded, hot and running.
2. Readings in velocity (displacement), filter out, should be taken in the vertical, horizontal, and axial position on Inboard and Outboard bearings. At initial check each position should then be checked with "filter in" to establish vibration frequencies.

NOTE: After initial test, a comparison of "filter-out" readings may be used for trending. If a gradual increase is detected over a period of time additional "filter in" readings may be acquired for troubleshooting.

10.4

3. If vibration amplitude, filter-out, exceeds 2 mils displacement, a filtered reading shall be taken recording all distinct amplitudes and frequencies.
4. If questions arise as to relation of vibration, a strobe light (phase signal) may be used observing same position on shaft.
5. Care should be taken to record all readings completely and accurately noting all amplitudes, frequencies, and phases.

10.2.4 Diagnostic Procedure

The key to interpretation is accurate data acquired in section 10.2.3. By looking at the frequency, position, and relative amplitudes, most problems can be detected. The following is a breakdown of many, but not all, vibration problems encountered.

Frequency Relative to Machine RPM	Phase Strobe Picture	Amplitude	Cause
1 x RPM	Single	Radial-steady	Unbalance
1 x RPM	Single	Large radial & high axial	Bad sleeve bearing
RPM x # balls in bearing	Erratic	High radial	Bad anti-friction bearing
RPM x # blades	—	High on bearing and on shroud area	Blade pitch problem Blades out of track Fan natural frequency same as blade pass frequency
RPM x 2	Steady 2 marks	High axial	Misalignment - locked coupling
# gear teeth x RPM	—	High velocity	Bad gear
Belt RPM	Erratic	Erratic	Bad or loose belt
1 x RPM	Steady 1 mark	Steady at speed	Electrical unbalance Note: Disappears the minute power is off.

NOTE: In order to do proper diagnostics, the mechanical specifications are needed (i.e., # of teeth on gears, type of bearings, temperature, etc.).

10.2.5 Cooling Tower Vibration Levels

1. Ranges of typical cooling tower vibration levels are provided by figures 5a, 5b and 5c. These vibration levels are based on measurements on operating towers and input from manufacturers and tower owners.
2. Three ranges of vibration severity are superimposed over the typical vibration level fields. These ranges are identified as follows:
 - A — No Faults; Typical of new equipment
 - B — Acceptable; Correction is not necessary
 - C — Marginal; Improvement should be considered for reduced maintenance

The vibration severity ranges were produced from the Southwest Research Institute article "Vibration Limits for Rotating Machinery", Mechanical Engineering, June 1987. Vibration correction factors for equipment rated less than 300 HP, machines with rigid rotors, and soft foundation were assumed for cooling tower mechanical equipment.

10.3 Balancing

The first step in correcting a vibration problem is to determine the cause. An imbalance condition, serious enough for concern, will show up on the motor as motion at speed of imbalance component (i.e., an unbalanced fan hub assembly running at 300 RPM will show up on a motor running 1800 RPM as a 300 RPM displacement).

10.5

10.3.1 General Requirements

Three main requirements are necessary to achieve a field balance correction.

1. The customer must be aware and make arrangement to start and stop the fan several times during balance procedures. This must be done to make adjustment in correction weight location and mount.
2. Weight must be added or taken away at different locations to correct unbalance; therefore, a means of adding or taking away known weights must be found. This may be accomplished by factory furnished dove tail slots, tapped holes, or slotted bars with a set screw.
3. Finally, one part of the rotating equipment to be balanced must be visible at speed. A mark is applied at this point when the unit is down, and then observed at speed with a strobe light, e.g. keyed to the filtered vibration signal, to reference unbalance to pick-up location. If these requirements are met, weight correction can be accomplished.

10.3.2 Instrumentation

Many vibration analyzers are available which are quite adequate for field balance work. The following is a break down of requirements and then options for field balance work only:

Requirements:

1. Amplitude read out in displacement (mils) and/or velocity (in./sec.).
2. Filter out plus filtering capabilities in RPM range of elements to be balanced.
3. Oscillate capability to adjust strobe to fire at filter setting.
4. Output jack to fire strobe.
5. Strobe light.
6. Pickup with long cable-length dependent on instrument to pickup max. location.

Options:

1. Dual pickup inputs with dual pickups and cables.
2. Velocity type vibration pickup with low frequency capability to give more linear response at fan speed.

10.3.3 Balance Procedure

The unit must be run to determine location of unbalance. Identifying areas of unbalance from motor are as follows:

Motor Unbalance - vibration at motor speed, high in radial direction, normally correctable on motor fans and drive sheave.

Sheave Unbalance - vibration at sheave speed - detected on drive unit nearest sheave in radial direction. Weight addition on sheave.

Drive Shaft - vibration at drive speed detected on drive unit - verified by two probe arrangement on motor and gear box, both probes detecting drive speed vibration - correction on coupling bolts with additional washers.

Blade-Hub-Sheave Assembly - vibration at hub speed, detected on driver. Can occur in radial or axial directions. Correction weight on hub in form of washers or plate.

After area of unbalance is found, the following step by step procedure is followed:

1. The fan is shut down allowing a clear, sharp, bright mark to be placed on rotating element to be balanced.
2. Start the unit placing the vibration probes in highest located radial positions by element to be balanced (i.e. if hub, place probes on output side of gear box next to hub. Insure probe and cable is secure.)
3. Record the filtered vibration amplitude and phase mark location as seen by the strobe light. It is helpful to first freeze phase mark in oscillate and then switch to read vibration to insure proper filter location. Henceforth these original readings will be designated as "0".
4. The unit is again shut down cleared for safety and access to correction weight location made. A trial weight is added anywhere, designation of TW in "ounces", approximately equal to $\frac{4WD}{R}$.

R

10.6

Where:

W = weight of rotor in pounds

D = vibration displacement in Inches (change mils to inches)

R = radius at which Trial Weight is added in Inches

5. Start unit, record again amplitude and reference mark. By vectorial plots as shown in example, the position and amount of correction weight can be determined.
6. Continue to change weight and location by comparing to "0" always until desired vibration level is reached.

Example:

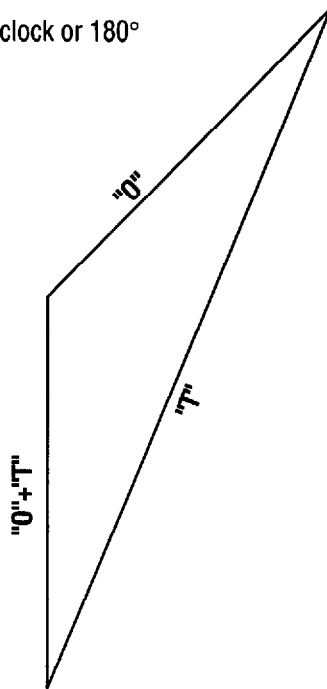
W = 500#

R = 10"

"0" = 12 mils @ 1:30 o'clock or 45°

$$Tw = \frac{4 WD}{R} = \frac{4 (500) (.012)}{10} = 2.4 \text{ oz.}$$

0 + T = 6 mils @ 6:00 o'clock or 180°



Length of "0" = 49mm

Length of "T" = 70mm

$$\text{Correct Weight} = \frac{"0"}{"T"} \times TW \cdot \frac{49}{70} (2.4 \text{ oz.}) = 1.68 \text{ oz.}$$

Move weight angle between "0" and "T" opposite rotation from "0" to "0+T"

Move Indicated 1.68 oz. 15° CCW

"0" = original vibration data

"0+T" = original vibration plus trial weight effect

"T" = trial weight vibration effect

TW = trial weight

10.3.3.1.

Data collection points are defined in Figures 1, 2, 3, and 4.

10.7

10.4 FATIGUE FAILURE

- 10.4.1** Most materials have a known endurance limit, that is the maximum unit stress which can be repeated without structural failure.
- 10.4.2** Under repeated loadings a stress smaller than the yield point of the material can cause failure. First a crack forms at a high concentration point and slowly spreads until failure.

10.5 FANS**10.5.1 Materials of Construction**

Fan blade construction materials are typically fiberglass reinforced epoxy, vinyl ester, or polyester plastic, extruded or cast aluminum, cotton duck reinforced phenolic plastic, or hollow stainless steel. Plastic blades may be solid, hollow, or made of reinforced plastic skin with a rigid urethane foam core. Some hollow plastic blades employ a central metallic load bearing spar as do most hollow stainless steel blades.

Plastic blades usually are lighter and more corrosion resistant than metal blades which may require coatings to resist corrosion. Metal blades are generally less affected by erosion due to water droplets impingement on the blade leading edge than plastic blades. Manufacturers of plastic blades often apply additional leading edge protection of a suitable erosion resistant material which reduces the tendency for leading edge erosion.

10.5.2 Types of Fans

The typical fan used in large industrial cooling towers is the axial flow propeller fan of the adjustable pitch type where pitch changes can be made only when the fan is at rest. Diameters range from smaller than 6' to 40 feet or more. Since the blade angle varies from hub to tip, care must be taken to follow the manufacturer's instruction as to the angle measuring point. The blade angle must be set at the same point on each blade in a fan assembly to within ± 0.5 degrees of each other or the various air pulsations from unequally set blades will cause a vibration problem.

For deicing, fixed pitch fans can be run in reverse rotation for short periods of time to reverse airflow through the tower. This typically should be done at half speed since the air load forces on fan blades are reversed. The fan and gear reducer manufacturer's recommendations concerning reverse speed limitation should be followed.

Controllable automatic pitching fans can be used on cooling towers for power savings and cold water temperature control. Some designs can reverse air flow by operating at a negative blade angle and achieve tower deicing.

There are two basic blade shapes. Extruded aluminum blades are straight, i.e., the width of the blade and thickness of the section are constant from hub to tip. Plastic, cast aluminum, and fabricated blades generally are tapered with the widest part of the blade near the hub and the narrowest part at the tip. The blade thickness varies in the same manner. Blade shape affects the exit airflow uniformity over the plane of the fan and tapered blades generally are more efficient and have better uniformity of flow than straight blades.

10.5.3 Fan Hubs

Fan hubs provide the support and attachment for fan blades. Manufacturers supply fan hubs of various designs but all must retain the fan blades and withstand loadings from centrifugal force and dynamic air pressure. Hubs may be manufactured from suitable materials such as cast iron (gray or ductile), aluminum alloy, or fabricated steel. Finishes such as epoxy or coal tar epoxy paint, urethane paint or other high quality resin based materials are commonly applied to fan hubs. Iron or steel hubs are often hot dip galvanized or coated with an inorganic zinc rich paint. Aluminum hubs do not require coatings unless exposed to aggressive water or air conditions such as salt water.

Fan hub hardware is typically galvanized steel. Austenitic stainless steels and monel are used for corrosive environments. Selection of a suitable grade of stainless steel should be made after consideration of water quality and potential for stress corrosion cracking. The fan blade to hub connection should be designed to limit the loosening of fan hub hardware in normal service.

The fan hub must provide a means for positively locking the hub to the fan shaft. Typical methods are the use of keyed and tapered bores, split tapered bushings and straight bores with the hub retained between a shoulder and retaining plate on top of the hub. Split tapered bushings should never be used on tapered fan shafts because the bushing will not seat properly as it is tightened.

If the fan hub requires welding, full penetration welds shall be used.

Hub seal discs are used to prevent negative airflow at or near the fan hub if the hub is small or of an open spoke design. Hub seal disc diameter should be at least 25% of the fan diameter. The seal disc is typically lightweight and designed for nominal air pressure and centrifugal loading. Axial flow fans with small hubs experience reverse air flow at or near the hub.

10.8

Reverse air flow at the hub will reduce fan efficiency and waste power. Properly sized fan hub seal discs greatly reduce air recirculation at the hub thereby improving fan efficiency.

10.5.4 Critical Speeds

If you were to suspend a fan blade or other slender article and begin vibrating it at varying frequencies, beginning at zero HZ, at some point it would begin to vibrate dramatically as its resonant frequency (RF) was reached. This is the point where its natural frequency to vibrate was excited by an equal applied source and it would resonate. This is a characteristic of any mass but in this case we are discussing fan blades of any size, type, or material of construction. Additionally, the resonant frequency of an operating fan will be somewhat higher than the measured static natural frequency because of the stiffening effect of centrifugal force.

In a fan exciting forces are the aerodynamic loads, typically at a frequency of:

$$F_{\text{(Blade Pass)}} = \frac{\text{No. Blades} \times \text{RPM}}{60} \text{ HZ}$$

This is called Blade Pass Frequency (BPF) and can be a great destructor of fan blades when the Blade Pass Frequency equals the Resonant Frequency. Every fan manufacturer should be able to verify resonant frequency for each size fan blade.

The frequency of blade vibration may differ from the BPF if large beams, drive shafts or other obstructions are within about 2 feet of the blades. In this case the BPF will be modified by "N" times the obstructions. Such as two beams close to the tops may cause a frequency of (2 x BPF)HZ. This frequency would have to be compared to the blade's resonant frequency in the first, second or third modes.

A first resonant peak ideally should not occur between "0" HZ and the fans operating speed. If one does occur during acceleration from rest, it will show up only as a momentary "shudder", since most fans reach full speed within 10 seconds or less.

In the case of a variable speed drive however, a resonant peak between "0" and the operating speed presents a real problem. The fan must be prevented from running steady state at or near the resonance speed by means of an electronic lock-out of a narrow speed range.

Additional fan critical speeds occur when the frequency of an exciting force is in harmony with the fan dynamic frequency. Aerodynamic forces generated by blades passing over structural members, lubrication lines drive shafts, etc. will excite blade vibration at a frequency determined by the number of times per revolution blades pass over obstructions.

These critical speeds occur at 2, 3, 4 ... n cycles per revolution frequency depending on the number of obstructions. The critical speeds can be calculated by the following equation:

$$N = F / \sqrt{a^2 - 1.5}$$

N = Critical Speed (CPS)
F = Fan 1st Mode Static Natural Frequency (CPS)
a = 2, 3, 4 ... n (cycles/revolution)

This approximation from Den Hartog, Mechanical Vibrations, 1947 is applicable for propeller fans with long blades relative to hub diameter. Fans should be applied at speeds no closer than 15% from any critical speed, unless dynamic stress evaluation qualifies operation nearer the critical speed.

10.5.5 Fan Balancing Specifications

There are several fan balance techniques commonly used by manufacturers:

a. Matched Moment Balanced Blades -

- 1) This process requires either static or dynamic balancing of the fan hub separate from the fan blades.
- 2) Each fan blade is moment balanced to a "master blade" standard. The blades thus balanced are interchangeable in the fan assembly and can be replaced with blades moment balanced to the same standard moment weight. The American Petroleum Institute endorses this process.

b. Static Balanced Fan Assembly -

- 1) This process uses the complete fan assembly mounted vertically on low friction bearings. Weight is added or removed from the fan assembly to achieve a static balance within specified limits.
- 2) If blades are to be replaced the entire fan must be rebalanced. This is accomplished dynamically on the cooling

10.9

tower or can be accomplished mathematically if the new blade and original blade moment weights are known.

c. Dynamically Balanced Fan Assembly -

- 1) This process is typically used for small fans such as centrifugal blower wheels. The cost of power and equipment to dynamically balance large fans generally eliminates the process from consideration.
- 2) Dynamic balancing produces the best theoretical balance because the effects of mass imbalance and dynamic forces are resolved.

If a vibration problem occurs in the field there are several options to correct the problem:

- a. Minimize tip track variation which is a major source of dynamic imbalance.
- b. Check the proper assembly of fan including hardware tightness, blades in the proper position, and blades all at equal pitch.
- c. Balance the fan "in place."
- d. Change the fan blades or the entire fan.

"In place" balancing is effective because it takes the mass of the existing machinery mount and dynamic imbalance into consideration.

10.5.6 Fan Noise

Consult CTI Standard ATC 128 for additional specific information concerning noise measurement.

Fan noise is most significantly controlled by reducing fan speeds. OSHA requirements can typically be handled by reducing tip speeds from 12,000 FPM to 11,000 FPM. This reduces noise on the fan deck adjacent to the stacks to 90db (A) or less.

In many cases the total noise emitted by the plant at the plant boundary must be reduced, calling for more stringent measures. Special fans with many wide-chord blades are required to produce the same work at tip speeds as low as 6000 FPM.

It is important to furnish the fan manufacturer all details of the customers specs, tower layout and orientation and the distances to the noise measurement point. In some cases only the general noise energy level, called the Sound Power Level (PWL), is specified. This is sufficient in itself for the fan designer.

10.6 PLATFORMS AND WALKWAYS

In order to inspect and maintain a cooling tower, access to all equipment and the fill area needs to be provided. A stairway from the basin curb to the fan deck is necessary for inspection and maintenance of the hot water basins, flow control valves, fans and drivers. Access doors or covers are necessary in the fan stack and through the fan deck. Platforms and portable walkways under the fan deck are needed for servicing and removal of the fan hub and gearbox. Operating platforms need to be provided for the flow control valves. A walkway above the coldwater basin the length of the cooling tower with access doors through the partition walls is necessary to inspect the internals and support structure. All of the platforms, walkways, ladders, and stairways must be in accordance with the latest OSHA Standards.

10.7 GEAR DRIVES

10.7.1 Purpose

The purpose of gear drives as commonly used in cooling towers is to reduce the speed from the driver to the speed required by the fan. Since the fan is generally mounted directly on the output shaft of the drive, a secondary function is to provide a spindle for the fan. Special design requirements must allow for both the dead load and thrust load and often large and indeterminate radial loads imposed by the fan. Because of these requirements and the severe operating conditions encountered, gear drives used are normally specifically designed for cooling tower service. Rating practice and operating considerations are specified in the CTI Standard 111.

10.7.2 Type of Gear Drives

10.7.2.1 Right Angle Gear Drives

Right Angle Gear Drives are used with horizontal motors. The motor is normally located outside the fan stack unless special motor moisture protection allows the motor to be located inside the wet plenum.

10.7.2.2 Single Reduction

Spiral Bevel - Single reduction spiral bevel gear drives are available up to 9.5:1 reduction ratios. Efficiency is approximately 97% and higher.