CHAPTER 2
THEORETICAL PRINCIPLES CONCERN

Most hydraulic turbines that have been operating for long period probably have some damage part. Its conditions from more twenty years ago are also different from current working conditions.

The hydraulic turbine is a main piece of equipment that transforms the energy of the water into mechanical energy and in turn rotates a water-wheel hydroelectric generator. The turbine runner and the rotor of the water-wheel generator are usually mounted on the same shaft. Modern hydraulic turbines are divided into two groups, (Farhat and Bourdon, 1999) impulse turbine and reaction turbines. An impulse turbine (like pelton turbine) is one which the driving energy is supplied by the water in kinetic from. A reaction turbine (like Francis propeller turbine, Kaplan and diagonal turbine) is one which the driving energy is provided by the water partially in kinetic and partially in pressure from.

The vibration of hydraulic turbine has many causes such as: cavitation under partial load, unbalance between outer circumference pressure and back pressure to the runner, imbalance of the runner itself, looseness of clamping, and others M. Farhat et al. 1999; Prasat, S 1999; the cavitation damage lowers the efficiency of electric generation and thus increases the operation and maintenance cost (Farhat et al. 2004; K. Phet-Asa 2006; X Escaler et al. 2006) for machine monitoring, the vibration is one of prediction maintenance.

2.1 Theory of vibration

The physical movement or motion of a rotating machine is normally referred to as vibration. Since sight or touch, cannot measure the vibration frequency and amplitude a means must be employed to convert the vibration into usable data that can be measured and analyzed. Electronics, mechanics, and chemical physics are closely related. Therefore, it would logically follow that the conversion of the mechanical vibration into an electronic signal is the best solution. The means of the converting the mechanical vibration into an electronic signal is called a transducer. The transducer output is proportionate to how fast the machine is moving (frequency) and how much the machine is moving (amplitude). The frequency describes what is wrong with the machine and the amplitude describes relative severity of the problem. The motion can be harmonic, periodic, and/or random. All harmonic motion is periodic. However, all periodic motion is not harmonic. Random motion means the machine is moving in an unpredictable manner (Tayer, 1994).

Different machinery problems cause different types of vibration. For example, a bent shaft causes a machine to vibrate differently to a worn bearing. Generally, these differences are indistinguishable to the touch or ear. However, with sensors and micro-processors, some condition monitoring equipment coverts' vibration data into various plotted formats that are recorded and analyzed to help diagnose machinery problems. Vibration analysis involves the analysis of these plotted vibration signals to diagnose the causes of abnormal vibration. Popular vibration analysis formats include overall vibration trend plots, time waveforms, spectra analysis and Fast Fourier Transform (FFT)

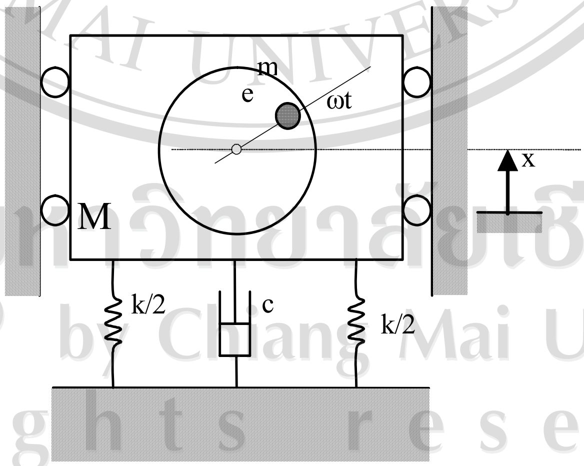

Figure 2.1 Model of unbalanced rotation. (Phet-Asa. 2006)

Vibration due to rotating unbalance in Hydraulic Turbine (Roger, 1998) is shown in Figure 2.1. From this figure, the unbalanced rotating machine is a source of unit vibration. This system consist a mass M that moves up and down in vertical direction. The mass M consists of a small mass m that can rotate as the angle ωt, and e is the distance from center to m. The equation of this motion can be written as

$$M\ddot{x} + c\dot{x} + kx = me\omega^2 \sin \omega t\,. \tag{2.1}$$

The steady state solutions to this equation is

$$x = X \sin(\omega t - \phi). \tag{2.2}$$

Where X is the maximum value of vibration amplitude and defined as

$$X = \frac{me\omega^2}{\sqrt{\left(k - M\omega^2\right)^2 + \left(c\omega\right)^2}}, \tag{2.3}$$

$$\tan\phi = \frac{c\omega}{k - M\omega^2}\,. \tag{2.4}$$

2.1.1 Harmonic motion

Harmonic motion is characteristically a sinusoid or some distorted version, depending upon the harmonic content, as in Figure 2.2. All harmonic motion is periodic, meaning it repeats at some point in time. In a linear system, imbalance in rotating equipment could generate harmonic motion. However, with many variables such as cavitation damages on turbine runner, gear problems, looseness, bearing defects, misalignment, etc., such sinusoids is not often found. It is important to understand that a sine wave is simply a plot of a circle against time. Notice how the circle in Figure 3 can be plotted as a sine wave, proving that linear motion is harmonic. All harmonic motion is repeatable and is just one form of periodic motion.

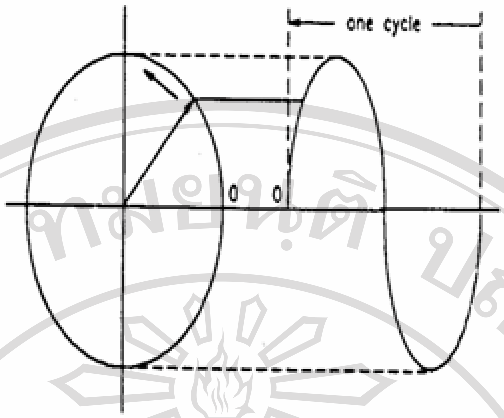

Figure 2.2 Harmonic motion. (Phet-Asa. 2003)

2.1.2 Periodic motion

Periodic motion is all motion that repeats periodically. This includes harmonic motion, pulses, etc. Periodic motion is any that repeats itself in equal time periods. For example, a misaligned turbine-generator coupling that is loose could have a bump one per revolution of the shaft. Although this motion is not harmonic, it is periodic. The time signal will have one pulse every x seconds as indicated in Figure 2.3.

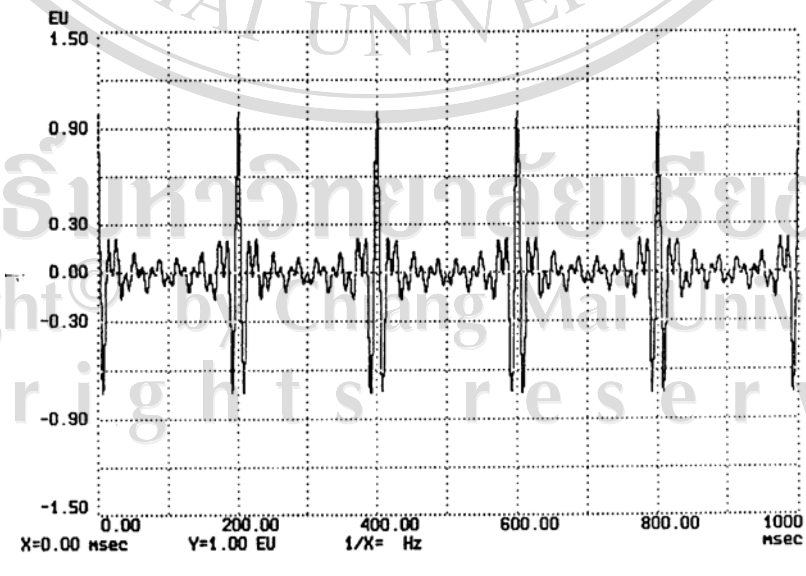

Figure 2.3 Periodic Motions. (Phet-Asa. 2003)

2.1.3 Random motion

Random motion occurs in an erratic manner and contains all frequencies in a particular frequency band. Random motion is any motion that is not repeatable. Random motion is also called noise. A time signal of random noise will contain all frequencies in a given range. The frequency spectra from such time signals will be up off the baseline as indicated in Figure 2.4. Often, random motion in a machine is caused by severe looseness.

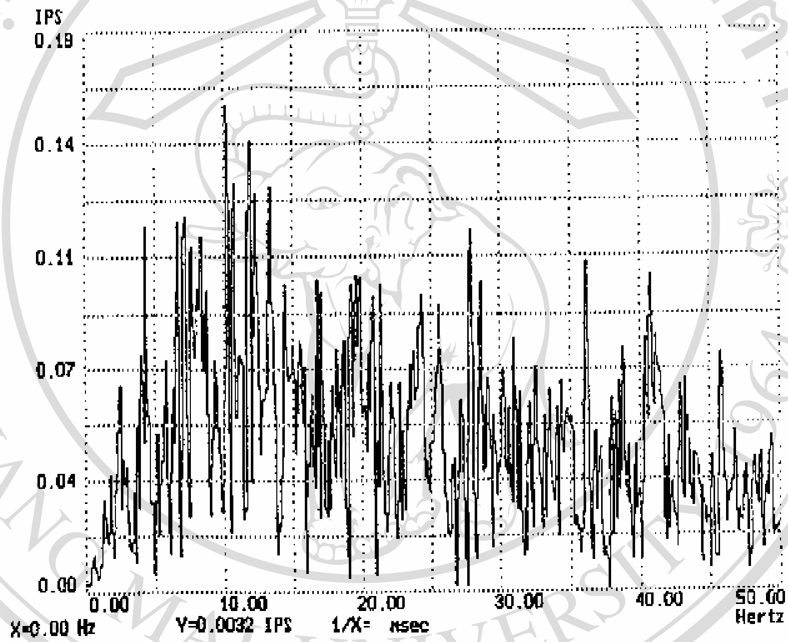

Figure 2.4 Random motion. (Phet-Asa. 2003)

2.2 Amplitude measurement

The four different ways express the vibration amplitude levels are: peak-to-peak, zero-to-peak, RMS, and average. Peak-to-peak is the distant from the top of to positive peak to bottom of the negative peak. The peak-to-peak measurement of the vibration level is shows in figure 2.5. This type of measurement is most often used when referring to displacement amplitude.

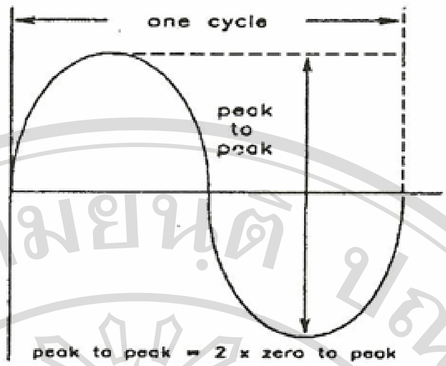

Figure 2.5 Peak to peak. (Prasat, S. 1999)

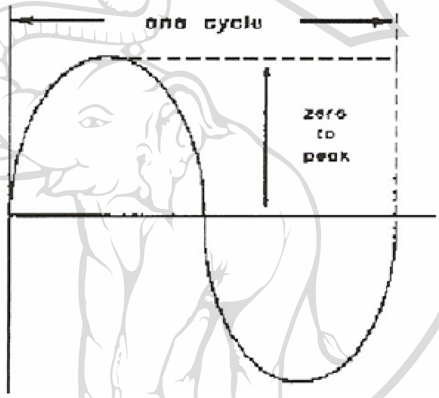

Figure 2.6 Zero to peak. (Prasat, S. 1999)

peak-to-peak is measurement from the zero lines to the top the positive peak or the bottom of the negative peak. The zero-to-peak value of vibration level is shown in the figure 2.6. This type of measurement is used to describe the vibration level from a velocity transducer or accelerometer.

The Root Mean Square (RMS) is the true measurement of the power under the curve. In the figure 2.7, the RMS value is the cosine of 45 degrees time peak (0.707 x peak only applies to pure sin waves). The true RMS value is calculated by the square root of the sum of the squares of give number of point under the curve.

$$RMS = \sqrt{\frac{P_1^2 + P_2^2 + ... + P_n^2}{N}} \tag{2.5}$$

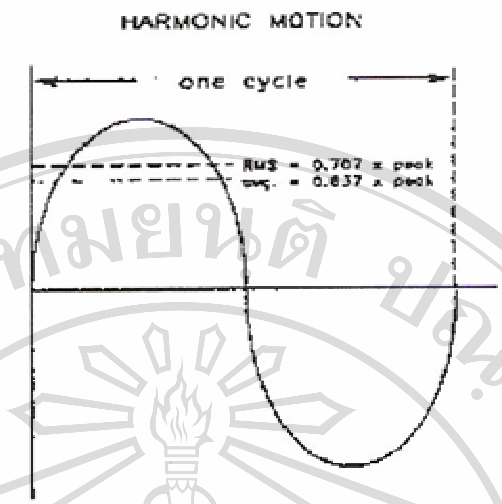

Figure 2.7 RMS average. (Prasat, S. 1999)

2.3 Source of frequency

The three source of frequency in machine are generated frequencies, exited frequencies and frequencies caused by electronic phenomena.

2.3.1 Generated frequency

Generated frequencies, sometime called forcing frequencies, are those frequencies actually generated by the machine. Some examples are imbalance, vane or blade pass frequency (number of vane times speed), gear mesh frequency (number of teeth time speed), various frequencies generated anti friction bearing, ball passing frequency of the outer race, ball passing frequency of the inner race, ball spin frequency, and fundamental train frequency. Generated frequencies are the easiest to identify because they can be calculated if the internal geometry and speed of the machine are known.

In addition to the more general problems, a turbine can have blade or vane pass frequency and cavitation problems.

2.3.2 Excited frequency

Excited frequencies (natural frequencies), are property of the system. Amplified vibration, called resonance, occurs when a generated frequency is tuned to a natural frequency. Natural frequencies are often referred to as a single frequency.

Vibration amplified in a band of frequencies around the natural frequency. The amplitude of the vibration in this band depends on the damping.

When we refer to the natural frequency, we often mean the center frequency. Natural frequencies can be excited by harmonic motion if the harmonic motion is within the haft- power point of the center frequency and contains enough energy.

2.3.3 Frequency caused by electronic phenomena

In the certain situation, false or misleading signals can be present. For example, when a sinusoid is clipped (as occur when input to the Real-time Analyzer (RTD) is overloaded slightly), it can cause a string of harmonics. The amplitude of the harmonic content is normally quite low.

2.4 Forcing function

When a machine vibrated, the cause must be determined. The cause of vibration is often referred to as forcing function. A number of possible problems, any combination of these problems, and an infinite degree of each problem can cause the machine to vibrate. The amplitude of each problem can be either overstated or understated depending on transfer function, resonance, damping, frequency addition, and frequency subtraction. Therefore very low amplitudes can be very serious problems in some causes and very high amplitude can be relatively minor problems in other cases. Frequency measurement and analysis makes vibration analysis work. Each forcing function has its own frequency, as indicated in the figure 2.8. If the problem is present, the frequency is present. The frequency can manifest itself as a discrete frequency or as sum and/or different frequency. Resonance, damping, frequency addition, and frequency subtraction do not affect the frequencies caused by vibration. This fact explains why frequency analysis is accurate, while amplitude measuring and trending are not accurate when diagnosing problems rotating machine.

Defects in rotating machine are caused by normal wear, improper lubrication, overloading or improper operation, installation improper manufacturing, etc. Vibration analysis identifies the problem or problems areas. Therefore, this research will study analysis the damage of hydraulic turbine and find the appropriate solution to the problems.

Vibration analysis of rotating machine involves calculating the frequencies the machine can generated and measuring the frequencies the machine is generating. Then, relating the measured frequencies to calculated frequencies and machine installation identify the problem. If the forcing function is a hit or an impact, the hit and impact may be not measurable. However, the machine response to the impact can be measured. Problem in machine induce vibration and result in wear, malfunction, and /or structural damage.

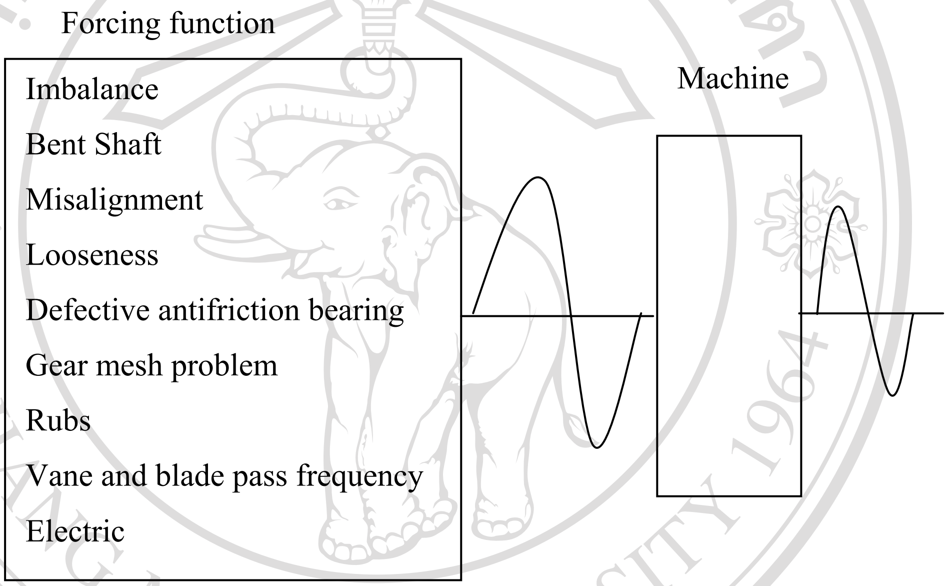

Figure 2.8. Physical nature of vibration. (Prasat, S. 1999)

2.5 Time and frequency domain

The analysis of some problems such as imbalance can be diagnosed in the time domain. However time domain signals from rotating machine are often very complex. Such signals must be analyzed in the frequency domain. If imbalance is diagnosed in only the frequency domain, errors will often occur. In vibration analysis, use of both the time domain signal and frequency domain spectra is required for complete, accurate analysis.

2.6 Fast Fourier Transform

To move from the time domain to frequency domain, we must perform a Fast Fourier Transform on the time domain signal. Mr. Fourier was a French

mathematician who proved all complex waveforms could be broken down in to their individual frequency components mathematically. However, this brilliant technology was not used extensively until the advent of the computer. Utilizing the transformation of the time signal to a frequency spectrum, Mr. Fourier's technology, and the computer's rapid capabilities, can produce and Fast Fourier Transform or FFT. Figure 2.9 contains a time signal and the corresponding FFT. The first (left-most) time signal contains the
fundamental, second, and third harmonic. This statement is true because each time period has three positive going peaks. The fist spectral line is fundamental frequency and its time signal. The second time spectral line is the second harmonic and its time signal. The third spectral line is the third harmonic and time signal, if the time signal of the three frequency components were added together; the result would be the first (left) time signal, as indicated in figure 2.9

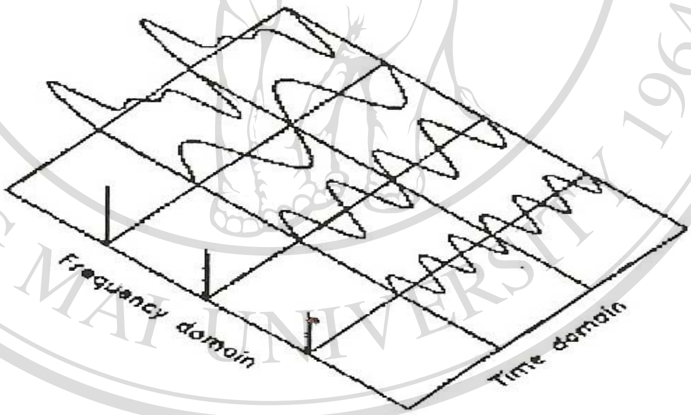

Figure 2.9 Relationship between time and frequency. (Prasat, s. 1999)

Because different type of problems often occur at different frequencies it is very useful to analyze measured vibration signals with respect to frequency. The Fast Fourier Transform can mathematically breakdown the overall vibration signal and arrange it according to frequency. The amount of vibration occurring at any particular frequency is called the amplitude of vibration at that frequency. FFT analysis can be very useful when analyzing the causes of vibration. Vibration of a bent shaft or impeller problem will occur with frequency of rotating of the shaft.

The selections of common detectable failure modes are;

- Imbalance: a peak at shaft speed.
- Misalignment: typically 1x or 2x and 3x shaft speed
- Blade damage: number of blades by shaft speed
- Cracked shaft: typically 2x, 3x shaft speed

2.7 Relationship between velocity, displacement and acceleration

Velocity is the measurement of how fast an object is moving from zero-to – peak and normally measured in tenth of one inch per second (IPS0 or millimeter per second (mm/s). The effective frequency range of most velocity transducers is from about 10 to 2,000 Hz. The velocity is most accurate measurement because it is not frequency relate.

Displacement is the measurement of how fast object is moving from peak-to-peak and is normally measured in thousandths of one mils (inch). Displacement is frequency related. The effective frequency range of non-contacting displacement transducers is from about 0 to 600 Hz. For contacting displacement transducers, the effective frequency range is about 0 to 200 Hz.

Acceleration measures the rate of change of velocity from zero-to-peak and is normally measured in unit of gravitational fore (g's). This means that high frequencies generate high g levels, and acceleration is frequency related. The effective frequency range for low frequency accelerometers is from about 0.2 to 500 Hz. The effective range of high frequency accelerometers is from about 5 to 20,000 Hz.

2.7.1 Velocity transducer

Velocity transducer, shown in the figure 2.10, measure show fast a component is moving in inches per second (IPS) or mils per second (mm/s). The measurement is made in tenths of an inch per second. Most measurements are less than one (1) inch per second.

Figure 2.10 velocity transducer

The velocity transducer is a voltage generator and does not require and external power source. Most of velocity transducers employ a permanent magnet mounted on a stud. A coil of wire surrounds the magnet and the coil is supported by leaf spring. When mounted on a machine, the motion causes the wire coil to move through or intersect the magnetic field. From basic physics any time a conductor is passed through a magnetic field, a voltage is produced. The coil is wound to produce and output proportional to the movement: normally four or five hundred millivolts per inch per second.

2.7.2 Displacement transducer

Displacement transducer measured how far something is moving, normally in thousandths of one in or mils. The displacement transducer is unique because it measures relative motion, i. e. How much one component is moving relative to another.

There are two basic types of displacement transducer such non-contracting and contracting types. Generally, the displacement transducers should be used to measure most low frequencies, below 10 Hz, and all relative motion measurements.

2.7.3 Acceleration transducer

The accelerometer measures the rate of change per time period. The units of measurement are in g's of acceleration. Low frequencies normally have low g levels, or a friction of one g. Higher frequencies can have levels of several g's.

2.7.4 Way of measuring vibration

Displacement, velocity and acceleration can measure the vibration. Transducer, meter, data collection, and Real-time analyzers are some of tools capable of measuring vibration levels.

Velocity measured with a velocity transducer that has a relatively flat frequency response between 10 and 2,000 Hz. The velocity transducer measures the vibration using the stud bolt that is fixed, and a coil of wire mounted on the spring. When a vibration source is applied to transducer, the coil moves over the magnet, producing a signal. In most application, the velocity transducer is the best tools with which to measure and record vibration.

2.8 Analysis of the vibration behavior

One of the main problems when monitoring hydraulic turbine is there is no a standard design. Each machine has specific characteristics and is connected to a specific hydraulic system. Vibration signatures are quite different between a lot head and a high hear machine or between a Pelton and Francis turbine. Moreover hydraulic turbine can operate with different load and vibration behavior can change considerably according to operating conditions. So it is difficult to extrapolate the vibration signatures from one unit to another or to know beforehand which will be the vibration behavior of determined unit.

A systematic approach is necessary for a correct monitoring and diagnostic and before attempting to monitor a turbine it is convenient to analyze the vibration generation chain.

During the operation of the machine, some excitation forces are generated that produce vibrations. There are excitations of different origin that act on different parts of the machine. These excitations are most or less important depending on the design and operating conditions. The excitations generated can be divided into hydraulic, mechanical and electrical and some of them are narrow band synchronously related to the rotating speed and others are of broad band. These excitations and their characteristic frequency have been described several times and can be calculated knowing rotating frequency $f_f = n/60$ (n= rotating speed (rpm)) the number of runner blades $Z_{b,}$ number of stay vanes, number of bearing pads, etc. Some of them are indicated in figure 2.11

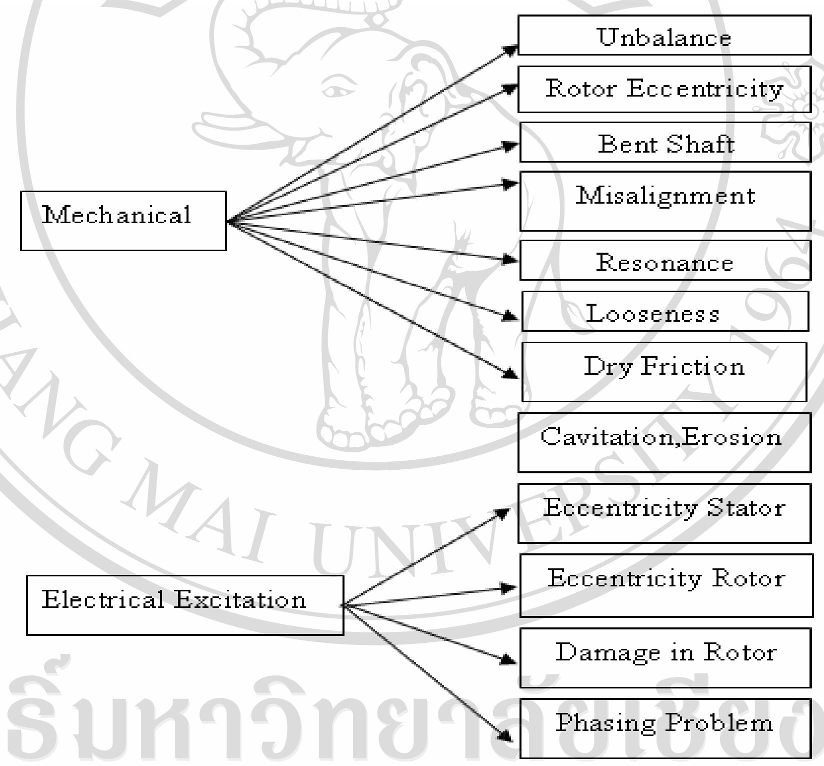

Figure 2.11 Some Excitations of Mechanical and Electrical Original. (Prasat, s. 1999)

These excitations can interact with hydraulic system, structures and rotor and can result in high or low vibration amplitudes figure 2.12. The analysis of the hydraulic system and of the rotor is essential to detect possible a possible resonance that can produce excessive vibration amplitudes in the machine.

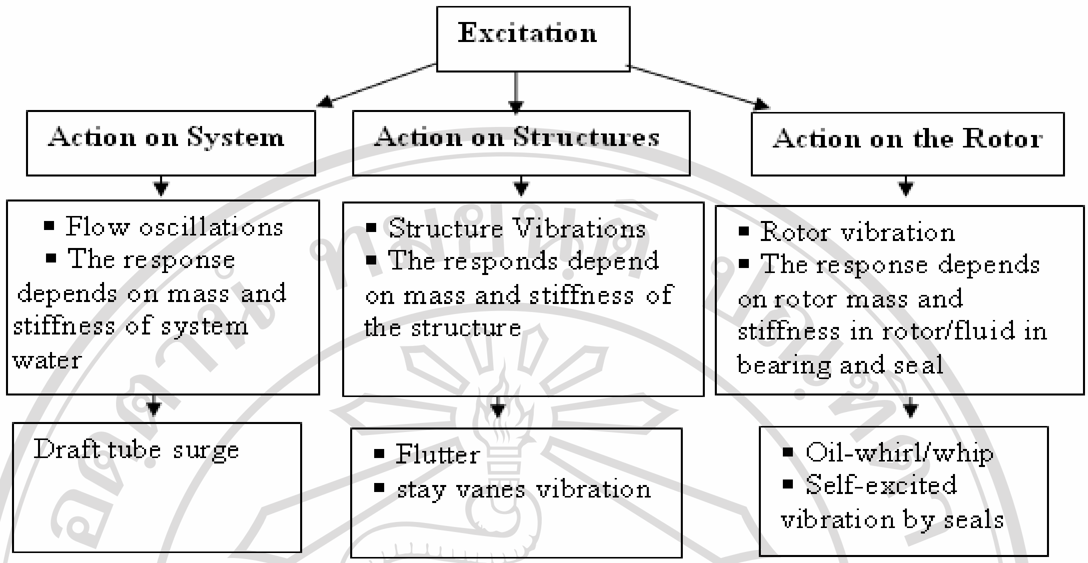

Figure 2.12 Vibration generation chain due to hydraulic excitations. (Prasat, s. 1999)

2.9 Vibration Amplitudes

There is limited information about vibration amplitudes in hydropower plant. Although some standards for rotating machinery are available (ISO 2372, ect.) the consider machinery with rotating speed of 10 Hz or more. Hydraulic turbines generally have lower rotating speed and a vertical shaft. Most of vibration amplitudes are measured in the mean. The amplitude is the overall rms value in mm/s versus de rotating speed in rev/minute. As it can be see there is large scattering of data.

2.10 Signature Variation with damage

For a correct selection of spectral alarm band, the establishment of basic vibration signature of the machine in each measuring point and is vibration with damage necessary. So, for each machine we have determined which can be considered its normal signature will change with the most common type of damage. This can be approached knowing the excitations forces, the rotor response and the basic symptoms of damage, which vibration levels can be considered normal and the basic symptoms of damage. Damage can change excitation, (i.e. Increase in the rotating frequency due to unbalance) it can generate new excitations (appearance of sub-harmonic frequencies in the spectral due to bearing damage) and/or can excite natural frequencies (rubbing, self-excited vibrations) Egusquiza and Robles, 1998.

2.11 Cavitations of Turbine

Daugherty et al (1989) explained the theory of cavitation of hydraulic turbine as follow. Cavitation is undesirable because it results in pitting (Figure 13), mechanical vibration and loss of efficiency. In reaction turbines, the most likely place for occurrence of cavitation is on the back sides of the runner blades near their trailing edges. Cavitation may be avoided by designing, installing and operating a turbine in such a manner that at no point wills the local absolute pressure drop to the vapor pressure of the water. The most critical factor in the installation of reaction turbines is the vertical distance from the runner to the tail water (draft head).

Figure 2.13 Cavitation of hydraulic turbine (Nam Ngum1 HPP).

In comparing the cavitation characteristics of turbines it is convenient to define a cavitation parameter (σ) as

$$\sigma = \left(\frac{p_{atm}}{\gamma} - \frac{p_v}{\gamma} \right) / h \quad (2.6)$$

Where

σ	=	cavitation parameter
P_{atm}	=	atmospheric pressure (Pa)
P_v	=	vapor pressure (Pa)
h	=	gross head (m)
γ	=	specific weight, 9.81 kN/m^3

Z_b = elevation (m)

Note that, Z_B and h are defined in figure 2.14. The term $P_{atm}/\gamma - P_v/\gamma$ represents the height to which water will rise in a water barometer. At sea level with 20 ºC water, $P_{atm}/\gamma - P_v/\gamma$ = 10 m. At higher elevations and at higher water temperatures it is smaller than 10 m. The minimum values of σ At which cavitation occurs is σ_c Its value can be determined experimentally for a given turbine by noting the operating conditions under which cavitation first occurs as evidenced by the presence of noise, vibration, or loss of efficiency.

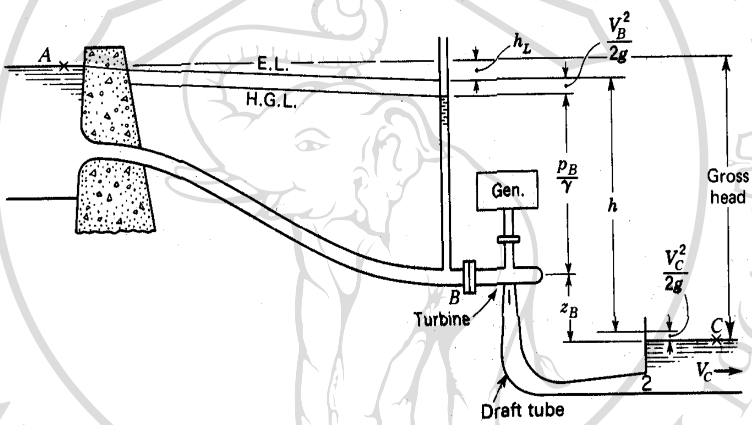

Figure 2.14 Net head on reaction turbine. (Daugherty et al, 1989)

From Eq. (2.6) we see that the maximum permissible elevation of the turbine runner above tail water is given by

$$Z_B = \frac{p_{amt}}{\gamma} - \frac{p_v}{\gamma} - \sigma_c Z_b \quad (2.7)$$

Where

P_{atm} = atmospheric pressure (Pa)

P_v = vapor pressure (Pa)

h = gross head (m)

γ = specific weight, 9.81 kN/m^3

σ_c = coefficient critical

Z_b = elevation (m)

Typical values of σ_c are shown in Table 2.1

Table 2.1 Minimum value of cavitation parameter at various specific speeds. (Daugherty et al, 1989)

	Francis turbines				Propeller turbines		
n_s	160	240	320	400	400	600	800
σ_c	0.10	0.23	0.40	0.64	0.43	0.73	1.5

Inspection of these values shows that a turbine of high specific speed must be set much lower than one of low specific speed. In fact, for a high net head h, it might be necessary to set a high specific-speed turbine below the level of the tail water surface (i.e., with negative draft head). This is a factor which restricts the use of propeller turbines to the low head range, which is fortunately the condition for which they are best suited in other ways also. Figure 2.15 shows recommended limits of safe specific speed of turbines for various heads and settings based on experience at existing power plants.

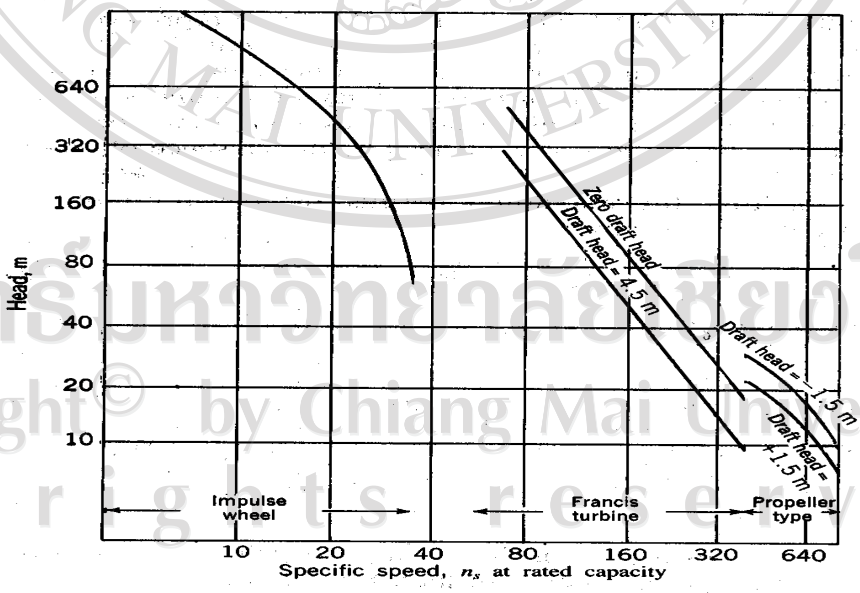

Figure 2.15 Recommended limits of specific speed for turbines under various effective heads at sea level with water temperature at 25 ºC. (Daugherty et al, 1989)