Turbine, Generators and Auxiliaries - Course 134

FACTORS LIMITING STARTUP AND RATES OF LOADING

Modern large steam turbines are exceedingly complex machines. Since they require a large capital investment the basic reliability of the unit is of considerable importance. In addition, the cost of alternate power sources makes the reliability of a nuclear steam turbine/generator particularly important.

Turbine/generator units are designed so that when they are installed, operated and maintained properly they will provide reliable power for essentially the life of the station without serious difficulties. Once a unit is running in a steady state condition, there is relatively little chance that a major problem due to maloperation will occur. However, during startup, warmup or load changes the conditions imposed on the unit are much more severe. Under these transient conditions a great potential exists for considerably shortening the useful life of the turbine and generator. When one considers the infrequency of startups on large nuclear steam turbines the time saved by a less than optimum warmup is insignificant when compared to the potential for long and short term problems.

The major factors which limit the rate at which a large turbine can be started up and loaded fall into the following categories:

(a) low cycle fatigue damage,
(b) high stress in turbine rotor,
(c) low turbine or generator rotor toughness at low temperatures,
(d) excessive vibration,
(e) water induction,
(f) excessive axial expansion,
(g) high stresses in turbine blading, and
(h) low oil temperature.

The avoidance of these conditions are the major determiners of startup procedure and failure to appreciate the consequences of these conditions can lead to premature turbine/generator aging and failure.

LOW CYCLE FATIGUE DAMAGE AND HIGH ROTOR STRESS

Steam temperature changes associated with warmup and loading of a turbine unit can impose significant temperature gradients across the rotor, casing and associated steam piping. The greater the average metal-to-steam temperature differential, the greater will be the imposed thermal stresses. The phenomenon of fatigue cracking is dependent on the magnitude of the peak stress imposed: as the peak stress increases, the number of cycles to produce cracking decreases. This is shown graphically in Figure 5.1. In addition, if the stress imposed is great enough the yield strength of the metal will be exceeded and permanent plastic deformation will occur. The effect of an excessive temperature differential between the average metal temperature and the steam is shown in Figure 5.2. Although the yield point can be exceeded in any turbine component, the combined effects of centrifugal stress and thermal stress make the rotor the most critical component. Exceeding the yield strength of the rotor metal at the bore can have immediate and disasterous effects on the unit since the rotor may well rupture.

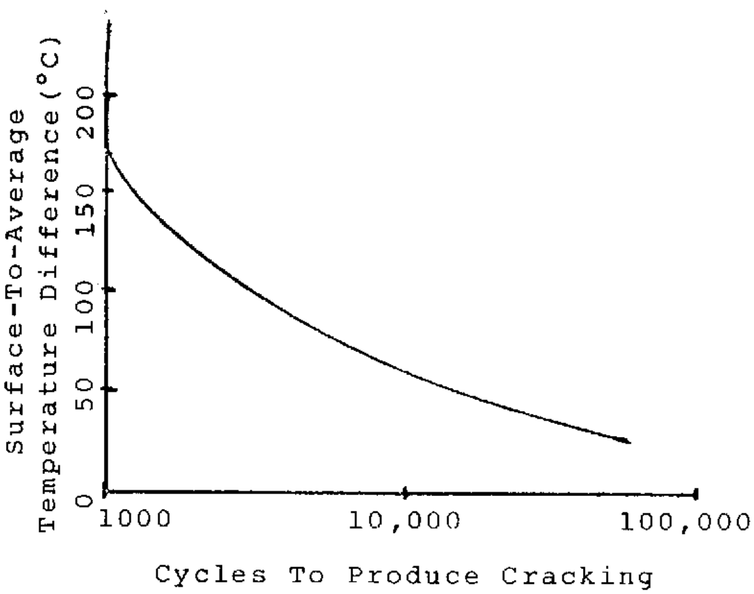


Figure 5.1

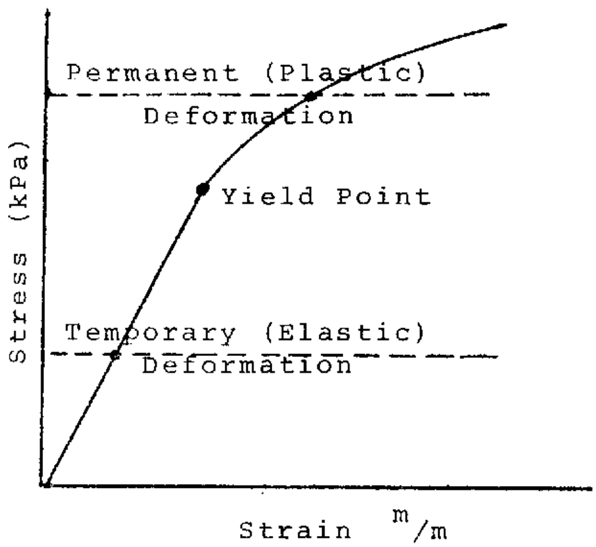
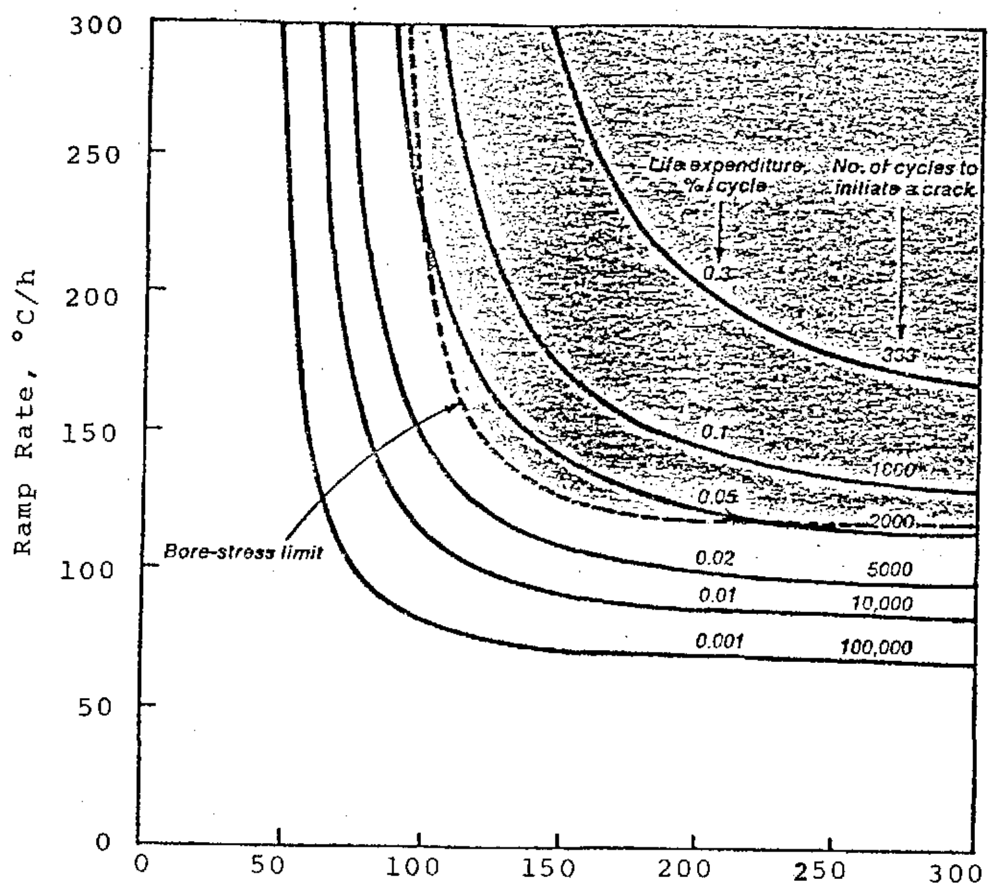


Figure 5.2

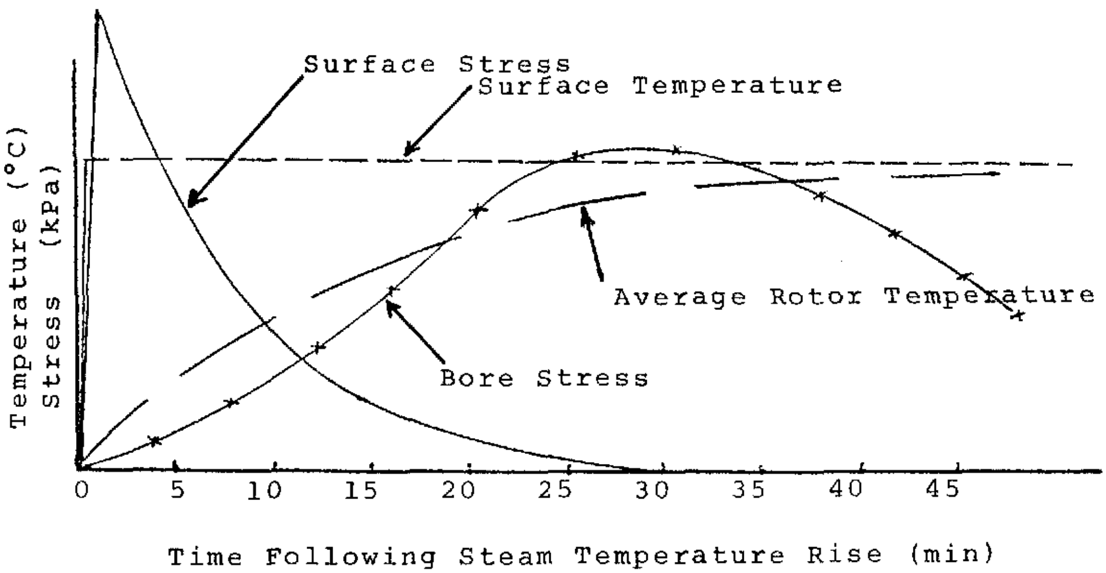
Figure 5.3 shows the effect which thermal cycles have on turbine life. Since the effects of thermal cycling are cumulative, the life expenditure per cycle is additive. For a 200°C warmup, a heatup rate of 100°C per hour (life expenditure of .02% per cycle) is the equivalent of 20 startups with a heatup rate of 70°C per hour (life expenditure of .001% per cycle).



Cycle-Life Expenditure

Figure 5.3

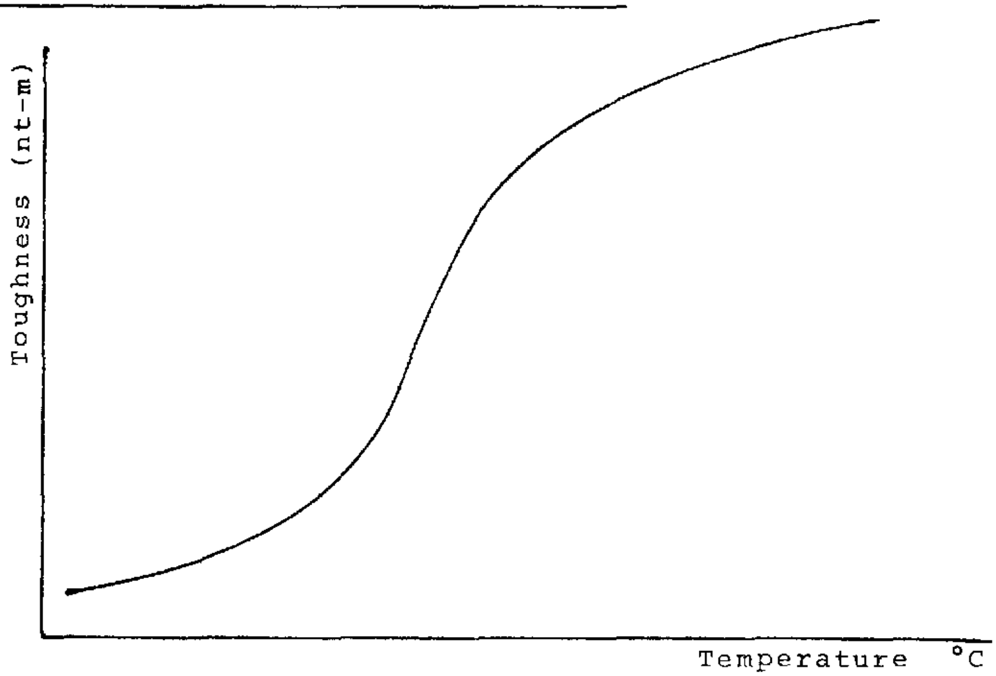
The maximum thermal stress at the bore of the rotor depends on the difference between the average rotor temperature and the bore temperature. On a load change the average rotor temperature lags the steam temperature as the rotor warms up. Thus the maximum bore stress is dependent on the rate at which the rotor heats up (average rotor metal temperature) and may not reach a maximum until 20 to 30 minutes following the load change. Figure 5.4 shows this effect. While the peak surface stress (surface fatigue cracking) occurs almost immediately, the peak bore stress is delayed. The conclusion is obvious: the effect of an excessive heatup rate does not immediately disappear but persists for some time. If the recommended heatup rate is exceeded, an additional soak must be imposed to allow the bore stress to dissipate.



Effect Of Temperature Change

Figure 5.4

LOW ROTOR TOUGHNESS AT LOW TEMPERATURES



Effect Of Metal Temperature On Toughness

Figure 5.5

Figure 5.5 shows the effect which temperature has on metal toughness (toughness is the ability of a metal to absorb energy in both elastic and plastic deformation). The warmer a metal is the less brittle it becomes and the greater the stress it can withstand without cracking. At low temperatures most metals will fail with very little stress in a sudden and catastrophic manner (brittle failure). Because of this phenomenon it is necessary to insure that the turbine and generator rotors are warm before imposing a high stress on them. The temperature at which toughness of a metal rapidly increases is known as the nil-ductility transition temperature because below this temperature the metal exhibits "nil-ductility". Typically the nil-ductility transition temperature for a saturated steam HP turbine rotor is in the range of 50° to 100°C and for generator rotors in the range of 10° to 30°C.

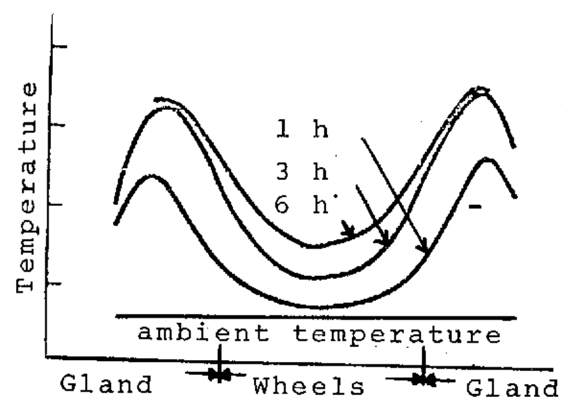
There are several methods of increasing the rotor temperature prior to imposing the high rotor stresses associated with high speeds and loads:

(a) generators normally have electric heaters installed;

(b) generator fields can be energized to provide I^2R heating of the rotor;

(c) turbines may be run at low speed and low steam flow for specified soak times to allow the rotor to come up in temperature while still at low rotor stress levels;

(d) the gland seal system can be operated for several hours while on the turning gear prior to steam admission to the turbine. As shown in Figure 5.6, this raises rotor temperatures. This increase in rotor temperature not only makes the rotor more resistant to cracking but aids in rolling out minor shaft eccentricities on the turning gear.



Effect Of Steel Steam Operation On Rotor Temperature

Figure 5.6

(e) the HP turbine casing can be pressurized with steam at 400-500 kPa(g) for several hours of turning gear operation. This procedure has been recommended by at least one turbine manufacturer (General Electric) as a method of raising HP rotor temperatures. In this method, the intercept valves are shut and steam is admitted to the HP turbine to pressurize it. In a four hour period the steam can raise the average rotor temperature above 150°C. This essentially turns a "cold" startup into a "warm" startup.

The factors of low cycle fatigue failure, rotor bore stresses and rotor transition temperature are the major determiners of the rate at which a turbine can be brought up to operating temperature (heatup rate) and load (loading rate).

The factors which must be considered in assessing the impact of a startup on the health of the turbine are:

(a) the initial metal temperature;

(b) the magnitude of the change between initial and final metal temperature;

(c) the rate of change of metal temperature, and

(d) the mechanical and centrifugal stress (as distinguished from thermal stress) in the rotor material.

Generally a heatup of short duration, at a low heatup rate, on an already hot turbine with low rotor mechanical stresses results in the minimum effect on the turbine. If one of these factors is less than optimum, the others will have to be carefully controlled to prevent unnecessarily shortening turbine life.

In assigning maximum rates of heatup and loading, the initial condition of the turbine is divided into three basic categories COLD, WARM and HOT depending on how long the unit has been shut down. As shown in Figure 5.7 the hotter the turbine, the shorter the time to full load and the greater the allowable loading rate.

Because the loading rate is dependent upon initial temperature, there is a necessity to insure the metal temperature does not decrease after loading commences. This could occur if the unit were brought on line at a very low load and the subsequent loading carried out at a low rate. In this situation the turbine could well cool down as the heat input at low load is less than the losses to ambient temperature from a hot

TYPE OF START	COLD TURBINE	WARM TURBINE		HOT TURBINE	
Period of shut down Hrs.	-	36	12	6	1
Estimated metal temperature °C	21	116	188	216	243
Maximum Condenser Back Pressure kPa(a)	50	30	13.5	8.5	6.5
Time from turning gear to full speed Mins.	40	20	10	5	3
Block load on synchronizing. MW	-	-	50	100	140
Load 0-50 MW @	2½ MW/MIN.	5 MW/MIN.	-	-	-
50-150 MW @	10 MW/MIN.	12½ MW/MIN.	20 MW/MIN.	50 MW/MIN.	100 MW/MIN
150-300 MW @	15 MW/MIN.	20 MW/MIN.	25 MW/MIN.	50 MW/MIN.	100 MW/MIN
300-540 MW @	15 MW/MIN.	20 MW/MIN.	35 MW/MIN.	35 MW/MIN.	100 MW/MIN
Time from synchronizing to full load. Mins.	56	32	18	11	4

Rates of Loading

Figure 5.7

turbine. To prevent this from occurring, a "block load" is specified for a warm or hot turbine. This block load which is brought on the generator at the time of synchronizing draws sufficient steam to keep the unit at the pre-startup temperature.

VIBRATION

The causes of vibration are legion but the effects are usually divisible into four general categories:

(a) fatigue failure due to cyclical loading,
(b) rubbing damage due to component travel beyond design limits,
(c) impact damage due to the pounding effect of bad vibration, and
(d) noise.

The general subject of vibration is treated in detail in the mechanics courses and will be treated here only as it relates to turbine runup.

As can be seen in Figure 5.8, the effect of vibration is generally dependent on the amplitude of the vibration and the frequency of vibration. The higher the speed, the lower the amplitude required to produce unacceptable vibration. With very few exceptions the causes of vibration only intensify as speed increases. There simply is no truth to the popular belief that vibration can be "smoothed out" by raising speed to a high enough value.

Because vibration is the end result of so many turbine/generator problems, excessive vibration is the most obvious warning of a poor runup technique. Figure 5.9 shows some of the major causes of excessive vibration in turbines, the characteristic frequency and probable solutions.

Permanent imbalance to a turbine which has previously run smoothly is generally caused by shaft eccentricity (sag or hog) or loss of material from the rotor (failed blading, shrouds or lacing wire). A rather insignificant change in the center of mass away from the center of rotation can develop large forces. The force created by a one kilogram imbalance located one meter from the center of rotation of a 1800 rpm turbine rotor would be on the order of 10% of the weight of the rotor. The shaft eccentricity necessary to produce sufficient vibration to rapidly destroy the turbine is only on the order of 1-2 mm.

Proper operation of the turning gear on shutdown and while shut down will prevent most eccentricity problems. In addition, there is a limited ability to "roll out" minor eccentricities by operating the shaft on the turning gear for some time prior

GENERAL MACHINERY VIBRATION SEVERITY CHART

For use as a GUIDE in judging vibration as a warning of impending trouble.

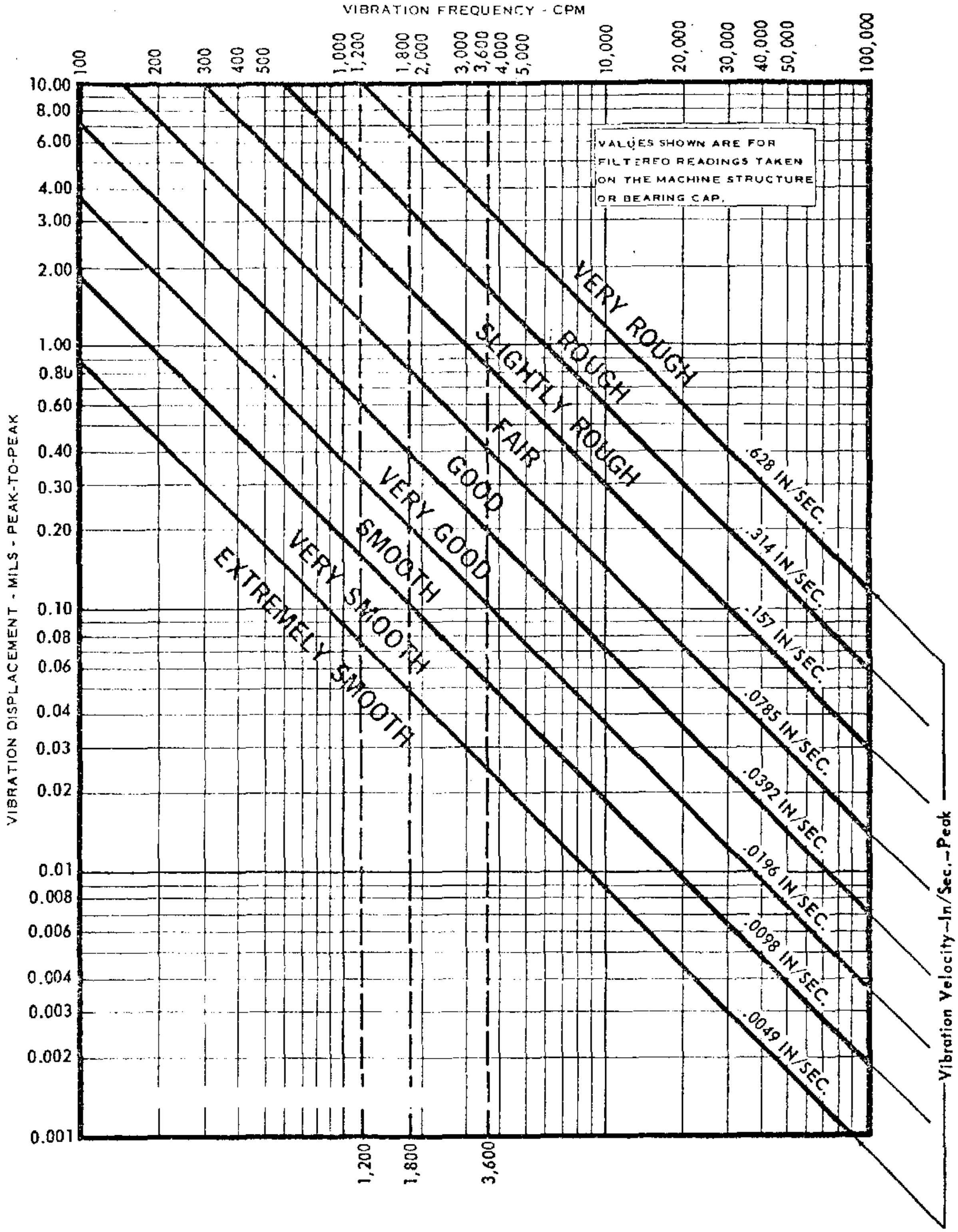


Figure 5.8

Figure 5.9

Possible Causes Of Turbine Vibration

Cause Of Vibration	Identifying Frequency	Possible Solution
Permanent Unbalance	Running Speed Frequency	Repair or Balance Rotor
Temporary Unbalance	Running Speed Frequency	Rotor balancing may be necessary; however, other causes of vibration such as component distortion and rubbing may also require correction.
Impacting	Subharmonic: 1/2, 1/3, or 1/4 of running speed frequency.	Eliminate contact between fixed and moving parts.
Oil Whip	Less than 1/2 of running speed frequency.	Improve bearing parameters (1) increase loading, (2) increase temperatures, (3) design new bearing.
Out of Round Journals	Harmonic: 2, 3 or 4 times running speed frequency.	Machine Journals Round
Rocking Journal Bearings	Harmonic: 2, 3 or 4 times running speed frequency.	Prevent rocking by better fixing of bearing.

to steam admission. The possibility of inducing shaft bending is particularly good if the shaft is left stationary with gland seal steam applied. This results in uneven heating and thermal expansion which can induce permanent distortion.

While severe bending can occasionally be rolled out it is usually uncertain and always time consuming. Failing this, the rotor would have to be sent to the manufacturer for straightening.

Vibration caused by temporary imbalance of the rotor is normally due to some form of thermal distortion. The most frequent causes of this type of uneven heating are:

(a) gland rubbing which produces localized hot spots,

(b) water quenching of the shaft in the area of water glands,

(c) water entry into the turbine.

This latter cause can produce particularly severe vibration if the source of water is in only one of the steam admission lines to the turbine. This can result in a chilling of only part of the rotor and produces an imbalance due to differential contraction of the rotor.

Contact between fixed and moving parts can cause two types of vibration:

(a) Direct contact vibration. Each time the fixed and moving parts contact each other a displacement occurs, and

(b) by acting as a force on the rotor which excites the rotor to vibrate.

This second type of vibration which is similar to striking a rotating piano string sets up a vibration in the rotor which beats against the fundamental frequency of rotation, alternately constructively and destructively interfering with the fundamental frequency. The net effect on the stationary parts of the turbine (what we detect as vibration) is a sum of the natural frequency of the shaft and the rotational frequency of the turbine. While the net frequency of vibration may be at almost any subharmonic of the rotational frequency (1/2, 1/3, 1/4, etc) the usual dominant frequency is at 1/2 the rotational frequency.

These subharmonic vibrations do exist and apart from contact between fixed and moving parts there are a number of causes which have been theorized including oil whip, non-uniform radial blade clearances and non-uniform radial bearing clerances. It should be borne in mind that the cause of excitation forces listed above are only theories and have not been proven to be the only forces present.

Vibration can increase significantly as the turbine speed coincides with the shaft "critical speed". The critical speed of a shaft is that speed at which the shaft is most sensitive to bending or deflecting. This means the internal damping of the shaft is at a minimum as shown in Figure 5.10.

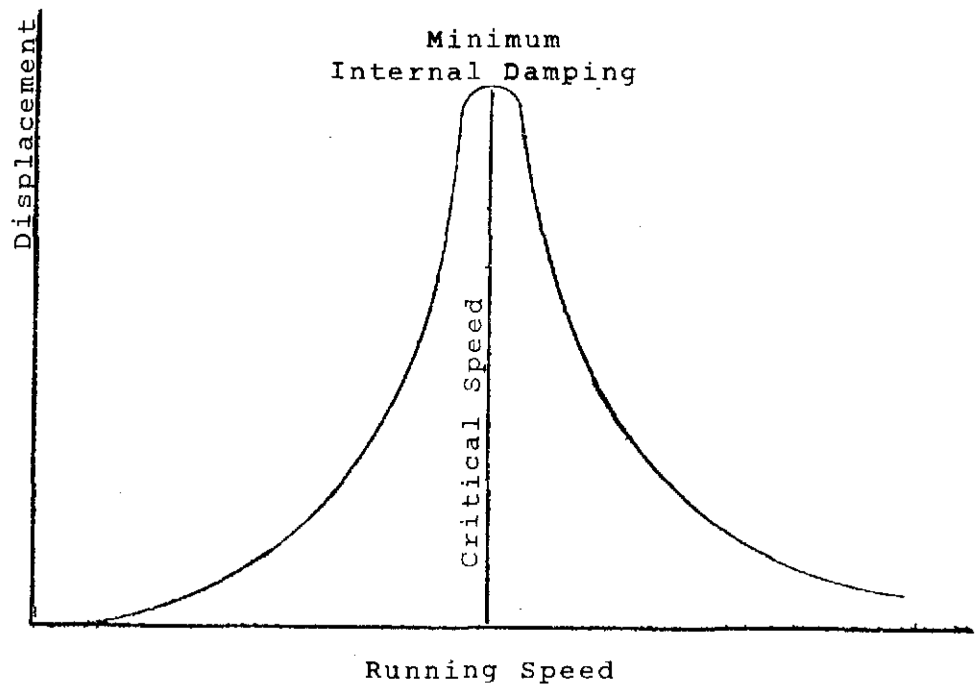


Figure 5.10

It so happens that this critical speed or frequency coincides with the frequency at which the shaft would oscillate if struck. At speeds below 70% of the first critical speed the shaft can be considered as rigid while at speeds above 70% of the first critical speed the shaft is flexible. When the rotational frequency of the turbine coincides with the natural frequency of the shaft (critical speed) the shaft vibrates sympathetically. The greater the vibration, the greater the excitation of the shaft. This resonant condition can rapidly increase the amplitude of vibration until the machine is seriously damaged. For this reason, the turbine should be carefully checked for abnormal vibration prior to passing through the critical speed. In addition, the machine should be accelerated through the critical speed zone as expeditiously as possible.

WATER INDUCTION

The possibility of water induction into the turbine is particularly great during startup. The steam which has condensed in the steam lines and turbine must be removed before any appreciable quantity of steam is directed to the unit. Not only can inadequate draining result in direct impact damage on the turbine blading but steam and water hammering in main and extraction steam lines can cause significant shock loading of piping and valves.

Water lying in steam piping may only move after a sufficient steam flow is established. In saturated steam systems the ability of the steam to "absorb" moisture is rather limited and water left in steam lines during startup may remain there for a considerable period of time until the steam flow is sufficient to move the water. These "water slugs" can cause considerable damage in the turbine. Even if they don't reach the turbine the impact at piping bends or valves can be great enough to deform or even rupture the piping.

Excessive moisture can also enter the turbine through improper steam generator level which decreases moisture separation efficiency. At low steam flows the steam generator level control system tends to be less sensitive to changes than at high flows. Even if the level control system is capable of maintaining level at extremely low flows, the ability to handle rapid steam flow transients is usually diminished. In some plants, the steam generator level control system receives no input from steam flow or feed flow at low power levels and, therefore, functions as a single element controller on steam generator level. For better or worse this makes the system less responsive to changes. For this reason, steam generator levels must be carefully watched while changing load and particularly at low power levels.

Leakage of feedwater regulating valves at low steam flows can make it extremely difficult to maintain boiler level. At low power levels if the leakage through a feedwater regulating valve exceeds the steam flow out of the associated steam generators, the level will increase. This problem may not be apparent while operating as the operating steam flow will be well in excess of even fairly troublesome low power leakage.

Another condition which can lead to high steam generator levels and possible moisture carryover is the effect of adjusting the "level set point" of the steam generator level control system. The level set point is the level which the controller will maintain at minimum power and represents the set point upon which the steam generator level control system determines the desired steam generator level. Basically the level controller adds to level set a gain based on steam generator power to obtained desired level (the level the controller tries to maintain)

$$\text{Desired Level (meters)} = \text{Level Set (meters)} + \text{Gain}\left(\frac{\text{meters}}{\%\ \text{power}}\right)\left(\%\ \text{power}\right)$$

If level set is adjusted high at low power levels the level which the controller seeks to achieve at high power level will be above the design value and may exceed the alarm set point. If the level set is adjusted away from the design value, the operator must be careful to compensate for this on power changes or he may find himself with abnormally high or low steam generator levels.

AXIAL DIFFERENTIAL EXPANSION

Axial differential expansion between the rotor and the casing is a major limitation on the rate at which the turbine unit can be warmed up and loaded. As the turbine warms up the casing and rotor heat up at different rates and therefore expand at different rates. The differential rates at which the two parts expand can cause metal-to-metal rubbing between fixed and moving parts particularly in the glands and blading.

Each turbine unit usually develops a characteristic differential axial expansion pattern in response to normal transients on startup and loading. This pattern should be familiar to the operator because the amount and rate of differential expansion can provide an early diagnosis of problems even before limits are exceeded.

In addition to excessive heat up rates, abnormal differential expansion can be caused by water induction, thrust bearing failure, binding of components (constrained from expanding), and quenching of casings from wet lagging. By understanding the differential expansion pattern for normal conditions, these abnormal conditions can be spotted prior to becoming major casualties.

There are basically three methods of maintaining axial expansion within acceptable limits:

(a) limit the heatup rate so that temperature gradients across the casing and rotor are small enough to keep differential expansion to an allowable value;

(b) design the rotor and casing to limit the temperature differential between fixed and moving parts to a small enough value to force reasonably equal expansion of both;

(c) allow sufficient clearances between fixed and moving parts to accommodate the differential expansion between rotor and casing from cold to hot conditions.

The latter method must be used to accommodate the fact that the rotor expands from its fixed point at the thrust bearing through the entire length of three low pressure turbines while each casing is fixed at one end. This arrangement, which is shown in Figure 5.11, results in the rotor expanding in the order of 25 mm (one inch) more than the casing in the last LP turbine. The only satisfactory method of handling this is to allow sufficient room in the LP turbines to accommodate this growth. Thus the clearance between fixed and moving blades increases in the LP turbines the further one is from the thrust bearing. Typically the clearances are in the order of 5 mm in the HP turbine, 12 mm in the #1 LP turbine, 19 mm in the #2 LP turbine, and 26 mm in the #3 LP turbine.

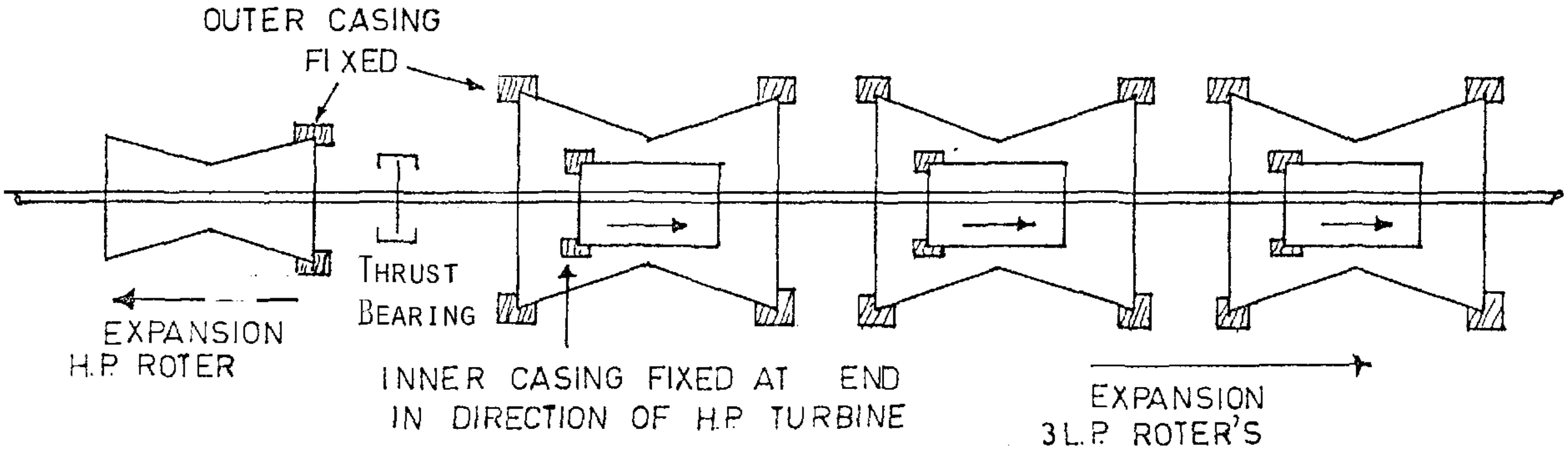
Expansion of the low pressure turbines' outer casings is relatively small compared to shaft expansion and, therefore, the differential expansion detectors are essentially measuring shaft position. Since shaft growth is cumulative as one moves away from the thrust bearing, an expansion in the first LP turbine results in a similar increase occurred in the second and third LP turbines. If this does not occur something is wrong with the indication of differential expansion for the #1 LP turbine.

However, the converse is not true. An increase in the #2 or #3 LP turbine would not necessarily be reflected in the turbines closer to the thrust bearing. A change in the differential expansion of only one LP turbine could occur due to large changes in extraction steam flow from only one turbine or changes in the position of intercept/reheat emergency stop valves.

Since increased clearances reduce turbine efficiency, this method of compensating for differential expansion cannot be used indiscriminately. Such construction techniques as carrier rings, double casings and drum rotors which are discussed in the level 2 course are utilized to equalize the heatup rates of fixed and moving parts.

STRESSES IN TURBINE BLADING

In the long blades of the latter stages of the low pressure turbines the tensile stresses set up in the blades due to centrifugal force are quite impressive. At 1800 rpm the stress in the root of a blade 1 meter long with an overall last stage diameter of 3.6 meters is in excess of 175,000 kPa (25,000 psi). Figure 5.12 shows the distribution of stress along the length of the blade.



Fixed Points Of Shaft And Casings

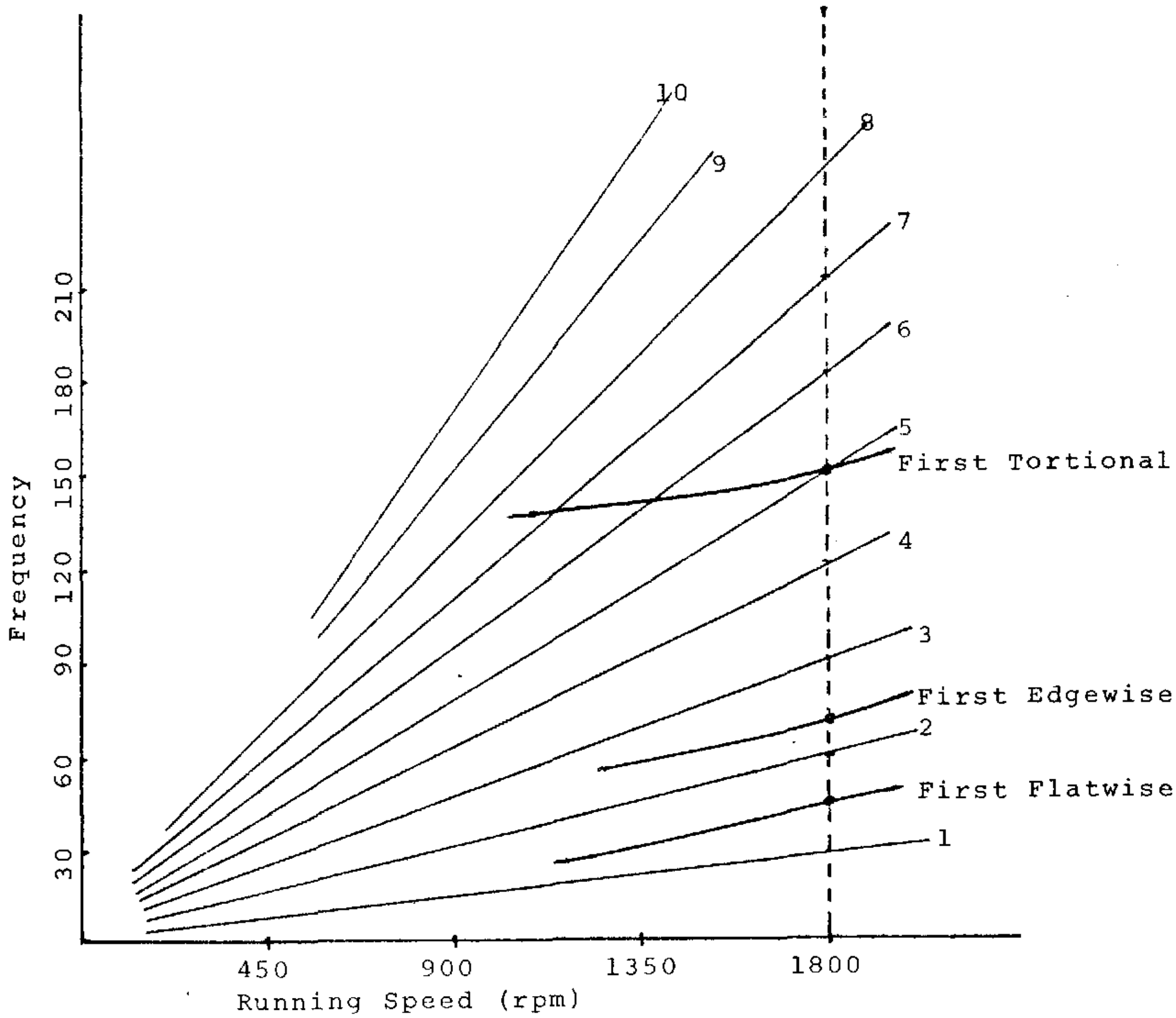
Figure 5.11

Stress Profile In Long Turbine Blade

Figure 5.12

In addition to designing blades which must withstand these stresses for years of operation, the designer must ensure the blades do not come into resonance at the operating speed or any harmonics of this speed. Using computers and advanced experimental facilities the designer develops a "Campbell Diagram" for each stage to insure that the blades do not resonate at operating speed. A typical Campbell Diagram is shown in Figure 5.13. The numbered diagonal lines are harmonics of the fundamental rotational frequency of the turbine. For example, the condition of the first flatwise resonant frequency lying midway between the fundamental rotational frequency (30 Hz) and the 2nd harmonic (60 Hz) is optimal. The coincidence of the first torsional resonant frequency and the 5th harmonic is undesirable and would necessitate stiffening of the blades to prevent undesirable fatigue stresses in the blading. In fact, the coincidence of any harmonic with a resonant frequency of the blade is undesirable.

The point of this discussion is that the blades have a variety of rather complex stimuli with which to deal and further complicating their environment only serves to shorten blade life. Apart from erosion of blading due to poor steam quality, probably the greatest factor in shortening turbine blade life is condenser vacuum. Transient heating of blading under a combination of high speed, low steam flow and low vacuum conditions can appreciably shorten blade life. For this reason vacuum on startup should be the best obtainable and, as turbine speed increases, the minimum acceptable vacuum



Campbell Diagram

Figure 5.13

should increase. In addition, exhaust sprays may be necessary on startup to prevent overheating of the latter stages of the LP turbine.

OIL TEMPERATURE

Generally speaking, the viscosity of oil is a direct function of temperature; the higher the temperature the lower the viscosity. Thus at low temperatures oil does not flow nearly as well as at high. In fact, at low oil temperatures the flow of oil to the bearings may be inadequate to keep the bearings cool and elevated bearing metal temperatures may develop. Even near operating oil temperatures, the effect of decreasing oil temperature out of the oil coolers may be to increase rather than decrease bearing temperatures.

Another phenomenon of low oil temperature is bearing instability caused by oil whip or oil whirl. This oil whip is shown in Figure 5.14 and results from high oil viscosity at low oil temperatures. The wedge of oil which by design should support the shaft becomes unstable and moves around the journal and tends to drive the journal within the bearing at slightly less than half the running speed. The resulting vibration can become quite serious.

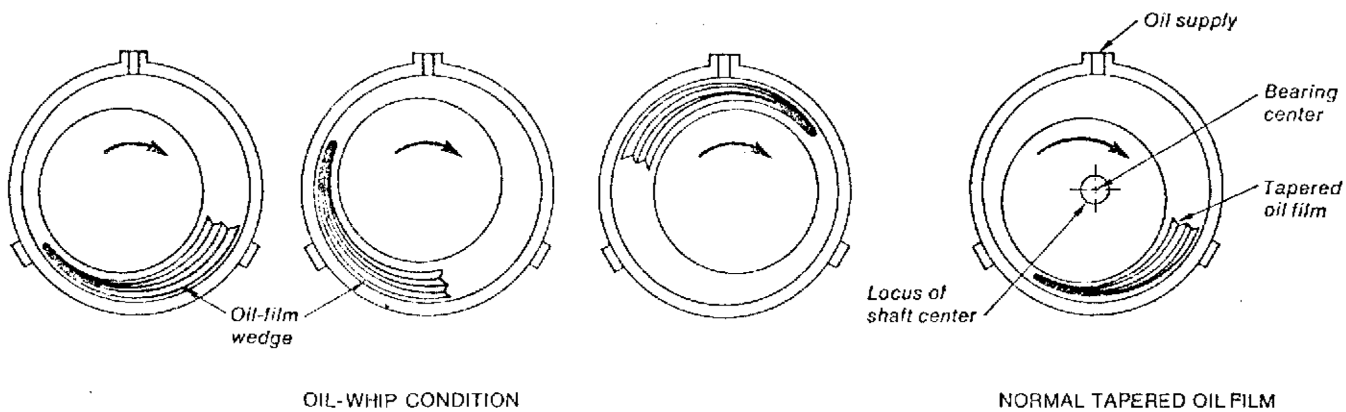


Figure 5.14

There are a number of factors which influence this phenomenon but for a well-designed bearing which has exhibited no instability in the past, the likely cause is an increased oil viscosity caused by low oil temperature. However, the phenomenon of oil whip is intensified by low loading of the shaft in the bearing; the more lightly loaded the bearing the more likely the shaft is to wander about in it. For this reason bearing vibration at normal oil temperature may be an indication of shaft misalignment within the bearings.

ASSIGNMENT

1. Explain the reasons for the following requirements on startup.

   (a) Placing the turbine on the turning gear 24 hours prior to steam admission.

   (b) Having the turbine on the turning gear prior to applying sealing steam to the glands.

   (c) Energizing the generator field prior to steam admission to the HP turbine.

(d) Increasing the minimum acceptable vacuum as turbine speed increases.

(e) Draining steam lines, turbine and extraction steam lines.

(f) Checking vibration.

(g) Checking axial expansion.

(h) Holding speed at 1200 rpm until lube oil temperature is above 39°C and rotor temperature is above 20°C.

(i) Bring speed up quickly through the critical speed range.

(j) Returning to 1200 rpm if HOLD parameters develop in the critical speed range.

(k) Block loads on synchronizing.

(l) COLD, WARM and HOT loading rates.

2. Why is vibration undesirable?

3. What is a "Campbell Diagram" and why is it used by turbine designers?

4. What is oil whirl (oil whip)?

5. You are conducting a startup with a specified loading rate of 15 MW per minute and find you have gone from 150 MW to 300 MW in 7 minutes.

   (a) What problems does this present?

   (b) What action would you take?

R.O. Schuelke