

## Turbine, Generator &amp; Auxiliaries - Course 134

TURBINE OPERATIONAL PROBLEMS

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As steam turbine units increase in size and complexity the operational problems also increase in magnitude. Not only has construction and control become more complex but materials have been pushed closer to their operating limits. As structures have become more massive, thermal gradients and pressure stresses have become more complex. In addition, with the increase in size it is no longer possible to build operational models and test them exhaustively before putting them into commercial operation. Today's turbine units go directly from the drawing board to on-site erection and commissioning.

Problems due to design errors are reasonably rare, but they do occur and because these occurrences are generally catastrophic it is becoming practice to build mathematical models of the system. Computer programs are then used to simulate normal, abnormal and casualty operations so that an assessment of in-service performance can be obtained prior to commissioning. By this method it is often possible to establish operating limitations before the design leaves the drawing board.

Because of the large capital investment in any modern generating station, reliability of the unit becomes a significant concern. This is particularly true of nuclear stations where the cost of alternate electrical energy sources can be truly phenomenal. For this reason it is becoming standard practice to assign a dollar value to estimated unreliability and factor this into the initial cost of the unit. This practice hopes to avoid an initial low price which turns out to be no bargain in service.

Despite these precautions, problems do occur particularly in the first year or so following commissioning. Not only do problems more frequently occur with a new plant but the operating and maintenance personnel require some time to familiarize themselves with the station.

The problems discussed in this lesson are derived from significant event reports and operating experience in nuclear and non-nuclear stations. Particular problems are included either because they occur with some frequency or because they represent a significant hazard to the turbine unit. The comments in this lesson are only of a general nature and are not intended as a substitute for design or operating manuals which constitute the manufacturer's specific recommendations on the operation of a specific turbine unit.

## Overspeed

The hazards of an unterminated overspeed generally fall into one of three categories:

- (a) speed will rise to a level where the centrifugal forces on the largest diameter wheels will cause tensile failure (rupture) of the wheel,
- (b) speed will rise into a critical speed region and remain there long enough for the resulting amplification of vibration to cause failure, or,
- (c) speed will rise to a level where the added stress due to centrifugal force will fail a component which has been weakened through fatigue, erosion or some other long-term phenomena.

The potential for an actual overspeed of the turbine unit occurs from two principle conditions: load rejection and testing of the overspeed trip mechanism. The response of both the mechanical-hydraulic and electrical-hydraulic governing systems to an overspeed following load rejection is discussed in the level 2 course and lesson six of this course.

The periodic testing of the operation of the overspeed bolts to trip the unit on an actual overspeed condition places the unit in a condition which can easily result in damage. Because the operation of the overspeed bolts is the last protective feature which functions to limit overspeed, the testing of this trip requires either the disabling of protective features which operate at lower overspeeds or raising the set-point of these features above the trip point of the overspeed bolts. If the protective features fail to operate properly, the unit speed can be raised to dangerous levels. The testing of overspeed tripping devices is always a hazardous evolution and requires a detailed operating procedure. At least, two independent methods of monitoring turbine speed should be used and personnel conducting the test should be in continuous communication with each other. There should be no question under what conditions the test will be terminated. The raising of speed to the trip point should be smooth and rapid enough to limit the time above operating speed to that required to allow monitors to follow the speed of the unit. Personnel conducting the test should constantly ask themselves if the unit is safe, even if none of the trips function as expected. It should be borne in mind that the vast majority of turbine casualties involving overspeed occur during this type of testing.

### Motoring

When the reactor heat production is lost through a reactor trip, the governor steam valves will shut to prevent the turbine steam consumption from lowering heat transport system temperature and pressure. If the generator output breaker is left shut, the turbine generator unit will motor with the turbine being driven by the generator acting as a synchronous motor. There are certain advantages to maintaining the turbine unit motoring during a reactor trip. Keeping the unit at operating speed shortens the time from steam admission to generator loading on the subsequent startup. This enables a faster recovery: first to avoid xenon poison-out and second to return the generator capacity to the grid.

During motoring the turbine blading is turning through dead steam and the friction between the steam and the blading rapidly overheats the long turbine blades at the exhaust end of the low pressure turbine. The problem is made more severe if the vacuum decreases and the blading encounters higher than design steam densities. The problem can be partially alleviated by an exhaust spray system and a cooling steam system as shown in Figure 3.1.

The exhaust spray system uses water off the discharge of the condensate extraction pump which is sprayed into the exhaust annulus of the turbine. This spray helps cool the dead steam as it is circulated by the rotation of the final low pressure turbine stages. To aid this system, steam is taken from the high pressure steam line ahead of the governor valves and routed to the inlet to the LP turbine. This "cooling steam" keeps a positive direction of steam flow through the LP turbine stages which helps to remove the windage heat.

Even with both cooling steam and exhaust sprays in operation, the final stage LP blading will overheat in something like an hour. This will require stopping the motoring of the unit. However, since the reactor will poison-out in about the same time frame, there would be little advantage in extending this limit.

### Low Condenser Vacuum

When condenser vacuum decreases below design values, the turbine unit is subjected to a variety of unusual stimuli. Heat rate increases as less work is extracted from each kilogram of steam; the turbine internal pressure profile changes; extraction steam pressure and temperature change; the distribution of work between the high and low pressure turbine changes. However, the most immediate problem associated with vacuum decreasing below design is that the condenser will eventually not be able to condense all the steam being exhausted

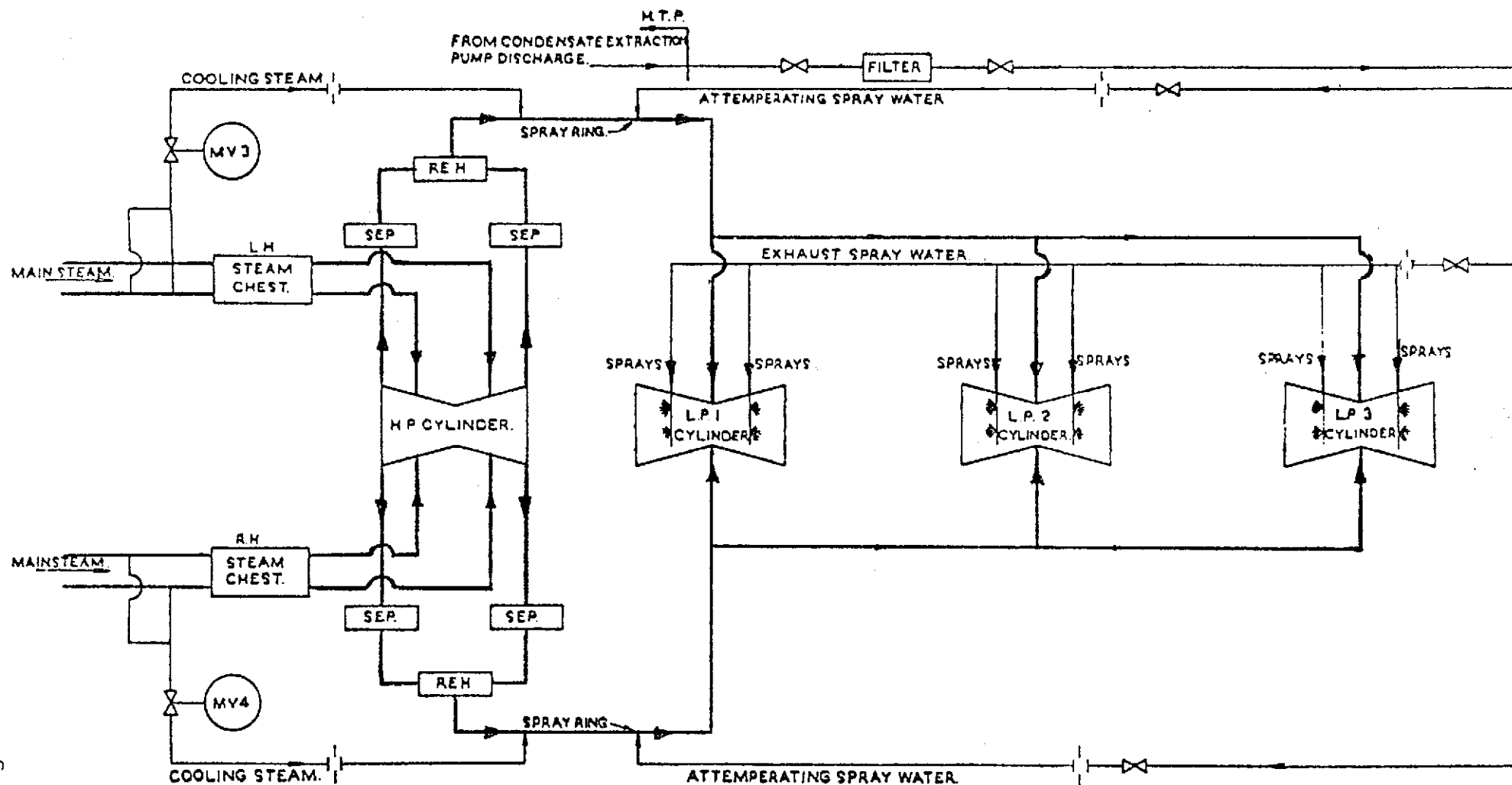


Figure 3.1

SCHEMATIC ARRANGEMENT OF LP CYLINDER EXHAUST SPRAY COOLING SYSTEM  
AND COOLING STEAM SYSTEM.

to it. In order to restore equilibrium to the condenser, the amount of steam rejected to it must be decreased. For this reason when vacuum has fallen below the minimum at which full power can be handled, the turbine will automatically begin to unload to a power level where the condenser can again reach equilibrium. With a design vacuum of 720 mm of Hg [5 kPa(a)], unloading will start at around 705 mm of Hg and will continue until either vacuum stops decreasing or 10% power is reached at 575 mm of Hg. If the vacuum decreases further the emergency stop valve will trip shut at 560 mm of Hg.

The combination of a low power level and low condenser vacuum imposes particularly severe conditions on the low pressure turbine blading. Not only do the long blades have to pass through a high density steam-air mixture but the absence of adequate steam flow through the turbine decreases the rate of heat removal. The adverse effect of low vacuum, low steam flow is the reason for terminating vacuum unloading before the governor steam valve fully shuts off steam. This effect also explains why, on a startup, vacuum should be the best obtainable before rolling the turbine with steam. It is desirable to maintain condenser vacuum on a shutdown until the turbine speed has decreased to about 50-60% of synchronous, to avoid a no steam flow, low vacuum condition.

### Water Induction

Water damage to modern saturated steam turbines can be roughly divided into two categories: long term erosion by wet steam and catastrophic damage due to ingress of large quantities of water. The former cause of turbine damage is covered in the level 2 course and will not be covered here.

Slugs of water can enter the turbine through a number of places, however, the two most common sources of turbine damage are due to water induction through the governor steam valve and through the extraction steam lines. Water induction causes damage in three principle ways:

- (a) direct impact damage on turbine components such as blading, diaphragms and blade wheels,
- (b) excessive thrust caused by water impingement leading to thrust bearing failure or hard rubbing between components, and
- (c) thermal damage to components due to quenching by water which may result in excessive thermal stresses, thermal distortion, or permanent warping. This is particularly true in the superheated section of the low pressure turbine.

Slugs of water which enter a turbine at high velocity will take the shortest path through the turbine, possibly clearing out both fixed and moving blades in the process. Because of the greater fluid density and the resulting impact on the rotor, induction of water from the steam generator may result in thrust loads much higher than design values. A failure of the thrust bearing can result in excessive axial travel of the rotor and subsequent severe rubbing damage to blading, blade wheels, diaphragms, glands and other components. Because of its high heat capacity, water contacting hot turbine parts can cause severe thermal stresses and distortion. This distortion can cause secondary damage if a turbine is restarted before the distortion has dissipated. While thermal distortion is not particularly severe in saturated steam portions of the turbine, it can be a significant cause of damage in the superheated sections.

Prevention of water induction requires both proper operation of protective features and careful avoidance of operating errors. The induction of water from the main steam line is minimized by the steam generator level control system, the high level alarm and by closure of the governor steam valve on high water level. However, improper or inadequate draining of steam lines during startup and subsequent loading can result in slugs of water being accelerated down the steam lines and into the turbine.

Water induction into the turbine can have particularly severe consequences on startup. While running under load, the steam flow can be of some benefit in absorbing water and minimizing thermal distortion, particularly in superheated sections of the turbine. Moreover, damage from rubbing can be increased when rotor speed is in the critical speed range.

If high vibration or other serious problems necessitate shutting down the turbine, the unit should not be restarted until all the water has been drained from the unit and the cause of water entry found and corrected. In addition, sufficient time should be allowed for relief of thermal distortion of the casing and rotor. Experience has shown that the most serious damage from water induction often occurs considerably after the first indication of water induction and attempting to restart may result in extensive damage due to rubbing between fixed and moving parts.

#### Condenser Tube Leaks

The consequences of impurities entering the feedheating system and steam generators through condenser tube leaks is covered in considerable detail in the level 2 Chemistry course. The adverse consequences of feedwater contamination with raw lake water fall into three general categories:

- (1) Introduction of ionic and non-ionic impurities which may cause or accelerate localized corrosion of feed and steam generator system components particularly the steam generator tubes.
- (2) Introduction of impurities which can lead to formation of boiler scale on steam generator tubes which decreases the overall heat transfer coefficient of the tubes, and
- (3) Introduction of impurities which upsets normal system chemistry which can result in increased general corrosion of components. This can be deleterious in its own right and can cause release of the oxide film from the feed system which results in an accumulation of these oxides in the steam generator.

Historically the well being of the steam generators following a condenser tube leak has been maintained through blowdown, pH control and the use of phosphates ( $\text{PO}_4^{-3}$  and  $\text{HPO}_4^{-2}$ ) to minimize boiler scale. While a steam generator which is well laced with phosphates is protected against boiler scale by the phosphates (which can also be used to maintain a buffered pH in the steam generator), the phosphates can cause considerable problems:

- (1) Over the long term the phosphate salts ( $\text{Na}_3\text{PO}_4$  and  $\text{Na}_2\text{HPO}_4$ ) tend to come out of solution and deposit in various locations throughout the steam generators. The deposition of literally hundreds of pounds of steam generator chemicals and corrosion products in the steam generator not only reduces the heat transfer area and alters the design flow distribution around the tube bundle but also acts as a collection point for all manner of corrosive ions which promote localized corrosion, and
- (2) Phosphates which leave the steam generator via blowdown act as a water pollutant through the stimulation of growth of marine life.

To reduce the deposition of boiler chemicals, there has been a gradual decrease in the specified level of phosphates in the steam generators. This has culminated in adaption of "zero solids" or "volatile" boiler chemistry in which phosphates have been eliminated during normal operation and are injected only during tube leaks. Recently, the effects of phosphates on water pollution has raised serious questions about even this use.

There are few problems in a nuclear generating station which present the potential for long term headaches that improper care of the steam generator does. Several nuclear generating stations in the United States have experienced long shutdowns while retubing steam generators. On the other hand the short term effects of steam generator abuse are reasonably undramatic and in the early years of operation there may be a tendency to treat the steam generators as if they will go on forever.

The impurities introduced by condenser tube leakage must be rapidly isolated and the steam generators quickly returned to in specification chemistry. In addition conditions which lead to accelerated erosion or corrosion of condenser tubes must be eliminated. These factors include sand in the circulating water which tends to erode tubes, high circulating water flow which may cause tube cavitation and erosion, tube fretting due to incorrect installation, high steam exhaust velocity or improper steam baffle operation which causes erosion from the steam side, and indiscriminant dumping of live steam to the condenser which accelerates steam side erosion.

#### Blade Failure

If there is a complete failure of a turbine blade in operation the effects may be disastrous as sections of blades get stuck between rows of fixed and moving blades and can strip the blade wheel. The resulting vibration can completely wreck the turbine. This type of failure due to metal failure is extremely rare due to advanced metallurgical developments and methods of blade fixing. However, because of the high stresses imposed on rotating blades and shroud bands, even minor errors during installation or replacement of blading may lead to early blade vibration, cracking and ultimate failure.

Probably the most significant source of blade failure is damage induced by water impact and erosion. Not only is the quality of steam entering the turbine important but in addition the ability of the blade to shed water can influence blade life. Use of cantilevered blades without shrouds is becoming reasonably widespread in nuclear steam HP turbines as the shroud tends to restrict the centrifuging of water droplets off of the blade. There have also been cases of blade tip and shroud band erosion and failure due to inadequately sized stage drains which resulted in standing water in the turbine casing.

Water erosion in the exhaust end of the HP and LP turbines has caused failure of lacing wires and damage to the leading edges of the blading. The erosion of blading causes pieces of metal to break off which may cause damage to fixed and moving blades in subsequent stages. Defects of this kind are minimized by having a very hard stellite or chrome steel



insert welded to the leading edge of LP turbine blades. In cases of extreme water erosion, however, these inserts may become undercut and themselves break loose to become a source of impact damage.

In operation, centrifugal stresses, bending stresses and thermal stresses may ultimately cause fatigue cracking of the blade roots. These cracks can only be detected during shut-down by non-destructive testing. Any evidence of blade cracking should be treated with caution as it is not only indicative of an abnormality within the turbine but also can lead to catastrophic blade failure.

#### Expansion Bellows Failure

Expansion bellows are used extensively in large turbines on LP pipework and between LP turbines and the condenser when the main condenser is being used as a reject or dump condenser.

In practice the bellows develop hairline cracks due mainly to thermal cycling as a result of load changes. Failure may also be caused by overload, for example, if an expansion bellows is fitted between the LP turbine and the condenser, the bellows may become strained if the condenser is over-filled without supporting jacks in position to take the weight.

#### Bearing Failure or Deterioration

Recent experience indicates that approximately half of all major turbine problems involved the bearings and lubricating oil system. With only a few exceptions most bearing problems can be traced directly to malfunctioning or maloperation of the lube oil system. Provided the lube oil system performs its primary function of supplying clean lube oil at the proper temperature and pressure to the bearings at all times when the turbine/generator shaft is rotating, there is usually little problem with the bearings.

Since even a brief failure of the lube oil flow to the bearings can result in considerable damage to the unit, the system is designed to provide continuous oil flow under a variety of pump shutdowns and power failures. The automatic features which provide the backup lube oil supply must be tested frequently to insure satisfactory operation. In particular the pressure switches which indicate low lube oil pressure should be tested not only for proper annunciation but also to insure that they are capable of starting the appropriate backup pump. In addition the response time of backup pumps should be tested to insure that continuity of lube oil flow is maintained. Testing should be conducted with consideration given to the consequences of a failure of the system

to operate as designed. For example, if the starting of the dc emergency lube oil pump is tested by turning off the auxiliary oil pump, with the unit on the turning gear, the shaft will be left rotating with no oil flow if the dc pump fails to start.

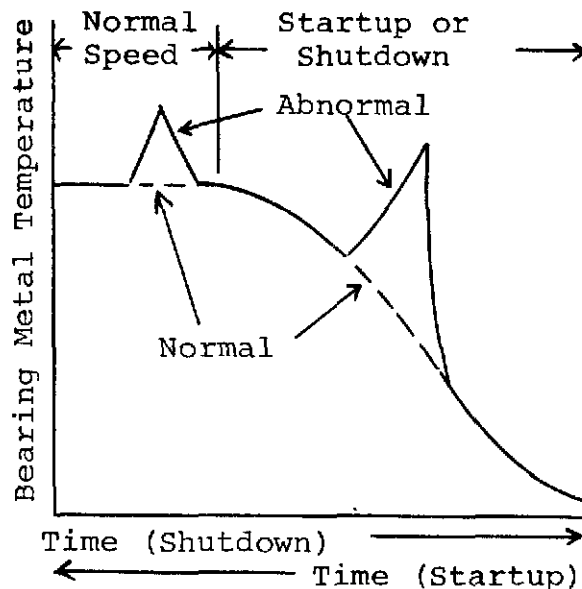
Of almost equal importance to bearing well-being is the cleanliness of the lube oil. Contamination of the turbine lube oil with water, fibers, particulates, dirt, rust and sludge can not only destroy the lubricating properties of the oil but also can cause accelerated bearing wear due to deposition of grit between the bearing and shaft journal. The lubricating oil should be sampled frequently. The results can be used to assess the quality of the oil and the efficiency of the purification system. Sample points should be chosen to insure samples represent not only the oil in the sump but also the oil going to the bearings. Metal particles in either the lube oil samples or the strainers should receive particular attention as they may indicate bearing, journal or pump deterioration.

One of the most effective ways to monitor proper bearing performance is through bearing metal temperature. A gradual increase in metal temperature over a period of several weeks or months can indicate a gradual deterioration of the bearing. Bearing metal temperature is influenced primarily by load, shaft speed and the type of bearing. Of a lesser importance under normal conditions are bearing journal surface, alignment, oil flow and inlet oil temperature. With the shaft at rated speed and oil flow and temperature normal, an upward trend in bearing metal temperature indicates a change in bearing load, alignment or journal surface condition. Temperature spikes of the type shown in Figure 3.2 can be excellent indicators of bearing deterioration. High spots on the journal or bearing can cause metal to metal rubbing until wear has eliminated the contact. This is particularly true on shutdown or startup when the oil film in the bearing is thinner and, therefore, there is more susceptibility to scoring.

Bearings should be inspected for wear and alignment at least each time the turbine unit undergoes a major overhaul. Journals should be checked for smoothness and uniform roundness and diameter from one end to the other. Journals should be inspected for scoring or an uneven surface which occurs from scoring and self-lapping over an extended period. A bearing metal wear pattern such as shown in Figure 3.3 is indicative of journal to bearing misalignment.

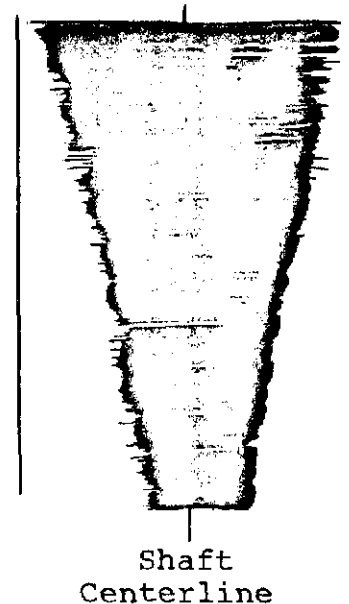
There is a popular and untrue notion that spherical seated journal bearings are self aligning during operation. Bearings must be properly aligned to prevent deterioration.

Additionally, the ball seats must be tight to prevent wear of the ball seats from causing vibration of the bearing.



Abnormal Bearing  
Metal Temperature

Figure 3.2



Misaligned Bearing  
Wear Pattern

Figure 3.3

### Electrical-Hydraulic Governing Systems

Failure of the turbine governing system has been a continuing source of problems. As turbine unit size increases there has been a continuing need for faster acting and more reliable governing systems. This trend has resulted in high pressure, fire resistant fluid (FRF) electrical-hydraulic governing systems replacing more conventional mechanical-hydraulic governors using lubricating oil on many large turbine generators.

While the need for periodic testing of governing systems remains undiminished, the need for system cleanliness and hydraulic fluid purity in high pressure, FRF governing systems is considerably more critical. Removal of impurities which could foul and eventually score the electrical-hydraulic control valves is essential in any high pressure hydraulic system. Control valve clearances are extremely small and the valves are particularly susceptible to sticking, scoring and eventual internal leakage. In addition, electrically actuated valves generally have little reserve power to free galled or sticking stems. The requirement for periodic testing of an electrical-

hydraulic governing system to insure freedom of valve movement is doubly beneficial in that fluid flow through the control valves keeps the valves clean. There is a great amount of practical experience which indicates that if a hydraulic control valve is not exercised for several months it will probably not operate.

All oil and particularly the synthetic, phosphate esters used in FRF systems have a tendency to pick up impurities notably chlorides and water which tend to accelerate system corrosion and eventual failure. The use of water or chlorine based cleaning solvents around these systems is an invitation to problems.

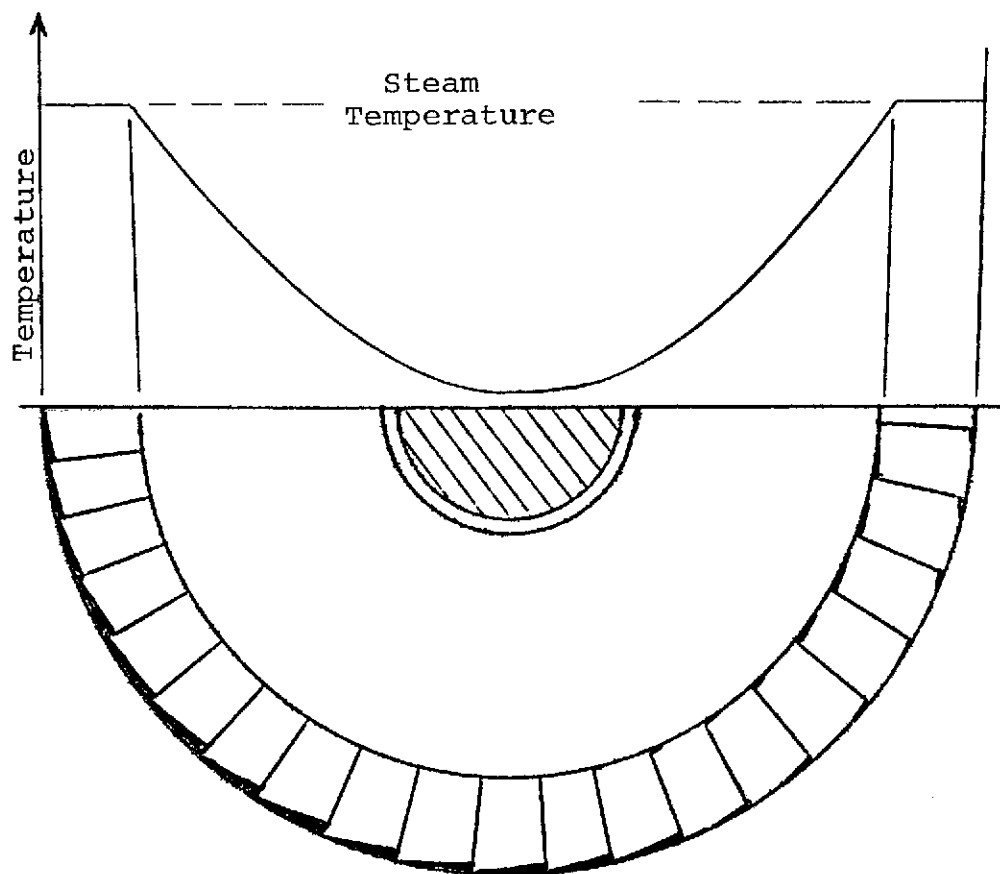
The fire resistant fluid used in most electrical-hydraulic governing systems is a synthetic phosphate ester hydraulic fluid. The fluid looks like and feels like a light mineral oil. It has good lubricating properties and excellent stability. The FRF used by Ontario Hydro has an excellent combination of chemical and physical properties: low particle count, low chlorine content, high electrical resistivity and negligible corrosion of most metals. This makes it a good fluid for use in the close tolerance valves, limit switches and overrides which are used in electrical-hydraulic governing systems. Above all, FRF virtually eliminates the fire hazard associated with conventional petroleum oils leaking onto hot steam lines.

FRF is reasonably non-toxic to the skin and exposure through soiled clothing presents a minimal hazard although FRF entering the eyes can cause a burning sensation and cause subsequent irritation. Phosphate esters can cause fatal poisoning, however, if inhaled in large quantities or if ingested in even moderate amounts. Under usual station operating conditions, inhalation of the vapor is almost impossible due to the low volatility of the fluid. However, when heated to decomposition, the phosphate ester can emit highly toxic fumes of phosphorous oxides. For a personal safety standpoint FRF can be harmful and special care should be taken to prevent ingestion, inhalation or absorption through the skin by persons who handle it. However, it can be handled safely if certain precautions are taken. These include no smoking or eating while handling the fluid, use of rubber gloves and safety glasses and availability of an eye wash fountain.

From an environmental standpoint, FRF can present a potential problem due to its toxicity and relative stability. For this reason disposal procedures for this fluid should be carefully adhered to.

### Low Cycle Fatigue Cracking

In large conventional and nuclear steam turbines the structural components, particularly the rotor, are subjected to extremely high thermal stresses during startup and subsequent loading. These stresses result from differential expansion within the casing and rotor as they are heated to equilibrium operating temperature. Figure 3.4 shows a typical temperature profile across the rotor during startup or loading when the steam temperature rises above the equilibrium temperature of the rotor.

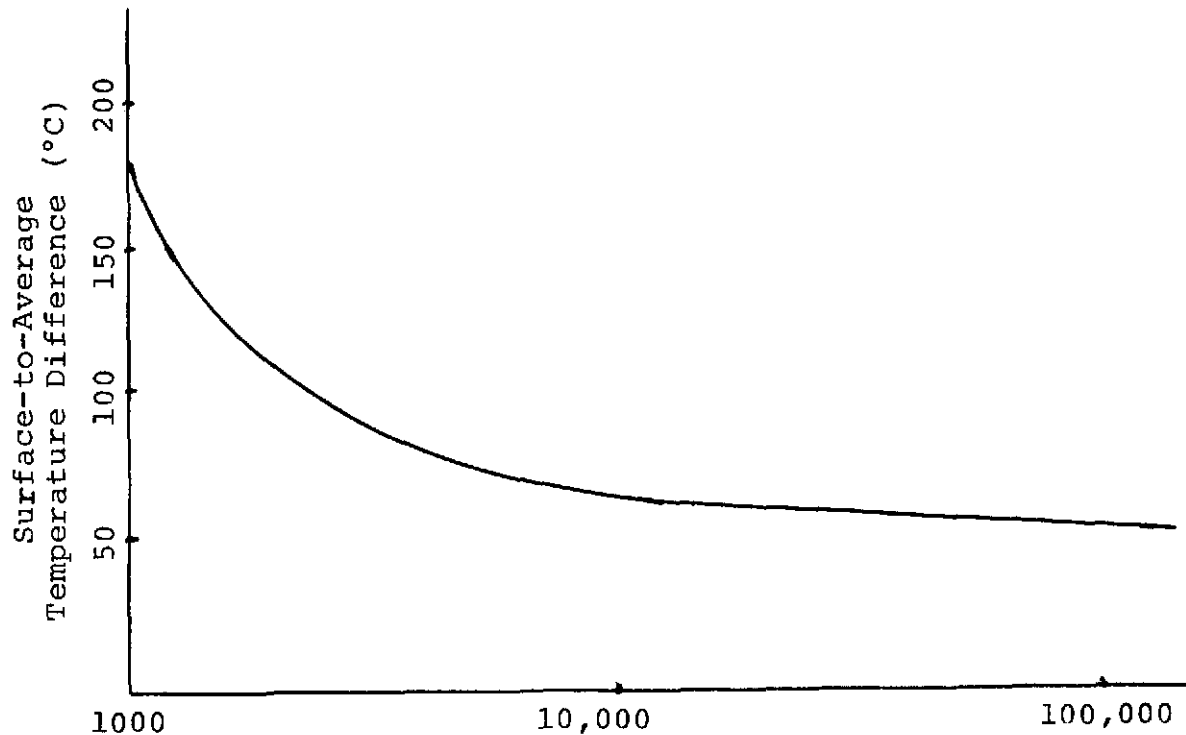


Temperature Profile Across Rotor During Heat-Up

Figure 3.4

The hot, outer surface of the rotor wants to expand relative to the colder shaft centerline. This produces a compressive stress at the surface of the rotor which is constrained from expanding as much as it would like to expand. On the other

hand, the inner portion of the rotor is forced to expand more than it would like to and is therefore under tensile stress. As the average rotor temperature rises to equilibrium, the stresses relax. In being forced to expand at other than the desired rate the metal may be stressed (either in tension or compression) beyond the elastic limit and permanent deformation may occur. Even if the elastic limit is not exceeded, repeated thermal cycling may result in fatigue cracking. As the level of stress rises the number of cycles necessary to induce fatigue cracking decreases. Since it is the temperature gradient across the rotor which produces this stress, the higher this gradient, the higher the stress and the lower the number of cycles to failure.



Cycles to Produce Cracking

Figure 3.5

Figure 3.5 shows the effect that increasing the difference between surface temperature and average rotor temperature has on fatigue cracking.

In addition to the formation of rotor surface cracks, the thermal cycling of the rotor bore can result in cracking which may lead to catastrophic failure of the rotor under the combined effects of centrifugal and thermal stresses.

The phenomena of brittle fracture is dealt with in some detail in the 128 materials course. As it relates to the turbine rotor, catastrophic failure (bursting) of the rotor requires three preconditions:

- (a) a tensile stress,
- (b) a temperature of the metal below the nil-ductility transition temperature (NDTT), and
- (c) a crack of critical length to initiate a brittle crack.

The bore stress limit imposed by designers is based on limiting the rotor bore stress to a level such that a crack of a size that could be conceivably missed during non-destructive testing would not grow to a critical size during the rotor lifetime. It has been verified that to prevent such a crack from growing to critical size in a reasonable number of cycles, the peak bore stress (centrifugal plus thermal) must be limited to approximately 90% of yield strength.

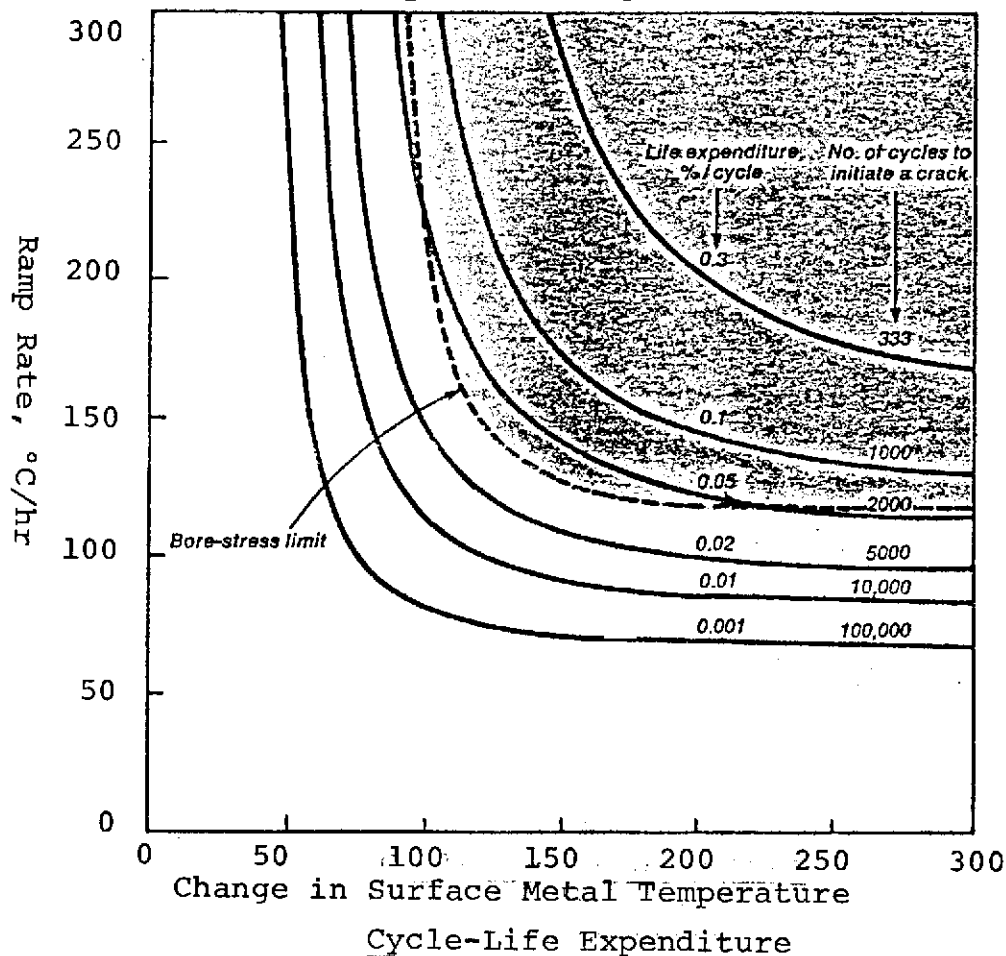


Figure 3.6

Figure 3.6 shows a typical cycle-life-expenditure curve for a large turbine. Although the curves vary from unit to unit depending on size, geometry and material properties the general shape is uniform.

It should be noted how significantly the number of cycles to failure is reduced for only moderate increases in the heat-up rate. For a 200°C heatup, it requires 100,000 cycles to initiate cracking with a 70°C/hr heatup rate. If the heatup rate is increased to 115°C/hr, it requires only 2,000 cycles to initiate cracking.

The area which lies to the right of the Bore-stress Limit should be avoided as the thermal stresses are sufficiently high that when they are combined with centrifugal stresses, catastrophic failure of the rotor may result.

#### ASSIGNMENT

1. Discuss the factors affecting the severity of the following operational problems. Include in your discussion the possible consequences and the design and operational considerations which minimize their frequency or effect.
  - (a) overspeed
  - (b) motoring
  - (c) low condenser vacuum
  - (d) water induction
  - (e) condenser tube leaks
  - (f) blade failure
  - (g) expansion bellows failure
  - (h) bearing failure
  - (i) low cycle fatigue failure
2. What are the advantages of using FRF as a hydraulic fluid for turbine control? What are the precautions which must be exercised?

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