GENERAL PURPOSE STEAM TURBINE RELIABILITY

by

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ABSTRACT

Several aspects of maintaining and operating small steam turbines at the Avon Refinery are discussed. The discussion concentrates on areas that have historically resulted in unreliable performance and repetitive failures. The scope of this discussion is limited to general purpose steam turbines as defined by the American Petroleum Institute in API 611 General Purpose Steam Turbines For Refinery Service; i.e., steam turbines with inlet conditions not exceeding 600 psi at 750°F with speeds below 6000 rpm and horsepowers below 2500. The intent herein is to focus on critical areas of turbine design and share with the reader the methods used at the Avon Refinery to successfully improve the integrity and reliability of turbines in those areas. Areas discussed will be bearing failures, coupling failures, governors, overspeed, and performance monitoring.

INTRODUCTION

Many papers on small steam turbine maintenance discuss general methods used to overhaul a machine, based on a manufacturer's recommended practice. For this presentation, it is assumed the user/reader knows how to overhaul a small steam turbine. Instead, the discussion focuses on areas of turbine design that have habitually been the source of repetitive failures.

What follows is a review of reliability improvement methods initiated at the Avon Refinery to increase the onstream time of small steam turbines, focusing on the areas that have historically caused us the most problems; i.e., bearings, couplings, constant, and overspeed mechanisms. In addition, with steam costs rising, a few comments about performance monitoring are included.

BEARING FAILURES

Contaminated Lube Oil

Bearing failures have historically been, and continue to be, one of the major causes of turbine repairs in the refinery today. When the author entered the Maintenance Department 14 years ago, the refinery was experiencing an average of one bearing failure per month on small steam turbines. This is in a population of approximately 144 steam turbines. Upon further analysis, it was found that the majority of these bearing failures occurred in steam turbines with 600 PSI, 750°F inlet conditions.

In addition to rapid oil discoloration, periodic oil analysis revealed extremely high total acid numbers (TAN) or neutralization numbers. This is an indication that the oil is oxidizing. Oxidation rates increase as the oil is subjected to excessive heat. As oil oxidizes, it also loses its ability to separate water. Free water was seen standing in the lucite reservoirs attached to the bottom of each bearing housing (Figure 1). However, prior to lube oil analysis, it was not known that small amounts of water were also suspended in the oil. Lube oil analysis (Table I) consistently showed water content of more than 1.0 percent. Oil with 0.01 percent dissolved water can shorten antifriction bearing life by as much as 48 percent [1].

The problem of bearing failures due to contaminated and degraded lube oil was attacked on two fronts: 1) additional cooling

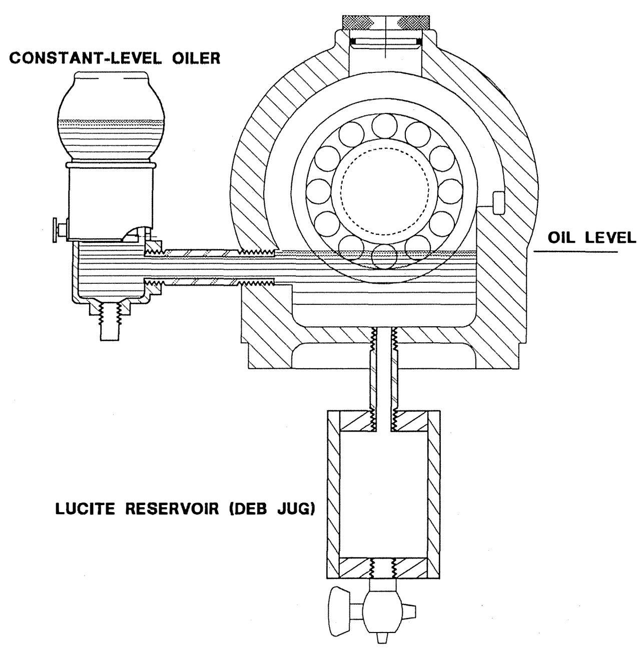


Figure 1. Deb Jug and Constant Level Oiler Arrangement.

Table 1. Lube Analysis.

Unit Description	Recommendations / Comments
CP0298 Wet Gas Compr No 5 Gas East Unocal ISO32 Turbine Oil	ABNORMAL - TEST RESULTS INDICATE UNUSUAL FINDINGS RESAMPLE AT YOUR REGULAR INTERVAL VISCOSITY IS TOO LOW NOTE ACCENTED ITEMS.

Operating Data						Physical Data				
Lab #	Date	Fluid Time	Unit Time	Fluid Added	Date Tested	% Solid	% Water	CS40 Visc	PH	Total Acid
B8712	12/18/90	0	0	0	12/20/90		<0.1	29.1	7.6	0.19
C2727	01/30/91	0	0	0	01/31/91		<0.1	30.6	8.3	0.14
C5866	03/12/91	0	0	0	03/14/91		<0.1	30.6	7.9	0.01
C7438	04/03/91	0	0	0	04/03/91		<0.1	29.6	8.1	0.01
C9208	04/25/91	0	0	0	04/25/91	<0.1	<0.1	28.7	7.5	0.01
C9818	05/02/91	0	0	0	05/02/91	<0.1	<0.1	22.5	7.1	0.01

Spectrochemical Data

Lab #	IRON	ALUMINUM	CHROMIUM	COPPER	LEAD	TIN	NICKEL	SILVER	SILICON	SODIUM	BORON	ZINC	PHOSPHORUS	CALCIUM	MAGNESIUM	BARIUM	MOLYBDENUM	VANADIUM
B8712	<1	<1	<1	<1	<1	<1	<1	<1	<1	<1	2	<1	7	<1	<1	1	<1	1
C2727	1	1	1	<1	<1	2	<1	<1	<1	<1	2	<1	13	<1	<1	2	2	<1
C5866	1	<1	<1	<1	<1	1	<1	<1	<1	<1	2	<1	11	<1	<1	2	<1	<1
C7438	<1	<1	<1	<1	<1	<1	<1	<1	<1	<1	2	<1	3	<1	<1	<1	<1	5
C9208	<1	<1	<1	<1	<1	<1	<1	<1	<1	<1	2	<1	6	<1	<1	1	<1	<1
C9818	1	2	1	<1	<1	1	1	<1	<1	<1	2	<1	9	<1	<1	2	1	<1

Additional Testing

Lab #	---Ferrogram---		
	DL	DS	WPC
B8712			
C2727			
C5866			
C7438	0.1	0.1	0.2
C9208			
C9818	2.0	0.1	2.1

This analysis is intended as an aid in predicting mechanical wear. No guarantee, expressed or implied, is made against failure of this component. This is not an endorsement of any product or service.

to slow down the oxidation rate of the oil; and 2) better methods to keep contamination from entering the bearing housing.

The first problem was addressed by circulating oil through the bearing using a force fed lubrication system (Figure 2). The pressurized oil is provided from a motor driven lube oil pump mounted on top of a 10 gallon stainless steel lube oil reservoir. In addition to cooling the oil, the reservoir also aids in the separation of free water and incorporates a sloped bottom for easy removal through a drain valve. Initially, a small shell and tube heat exchanger was mounted on top of the console to dissipate the heat, but it was discovered through experience that these heat exchangers were not necessary due to the volume of oil in the reservoir.

The specification for the lube oil circulating consoles was written by the Maintenance Engineering Group within the refinery. The specifications were sent to several local manufacturers, and a contract was awarded to a local firm to construct the consoles. Since the oil was now being forced fed through the bearings, a filter element was added to the console.

The filter provides 25 micron filtration and has an integral differential pressure indicator. The filter has a washable element which is easily removed and replaced. Reservoir features include an internal weir and pump suction screen, full height level indicator, and additional plugged drain ports for alternative installation configurations.

Fortunately, the turbine manufacturer's standard design provides ports for injecting the oil into the bearings. The only major modification necessary was the installation of standpipes internal to the bearing housing to maintain an oil level for the oil rings, which remain installed on the shaft (Figure 3). In the event of a power failure, the lube oil consoles circulating capability could be lost. The standpipes will maintain an oil level in the bearing housing, and allow the oil ring to continue to lubricate the bearing. Existing cooling water supplies to the bearing housing jackets are also retained for this purpose. A dramatic decrease in turbine bearing failures has been observed after installing lube oil consoles on the "bad actors" in the refinery.

As a side note, these consoles have been used on hot oil pumps and as backup systems on force fed pumps, and are a storehouse stock item.

Lube Oil Contamination

Although acceptable oxidation rates were achieved with oil circulating consoles, lube oil contamination continued to persist. The primary source of contamination was steam escaping from the

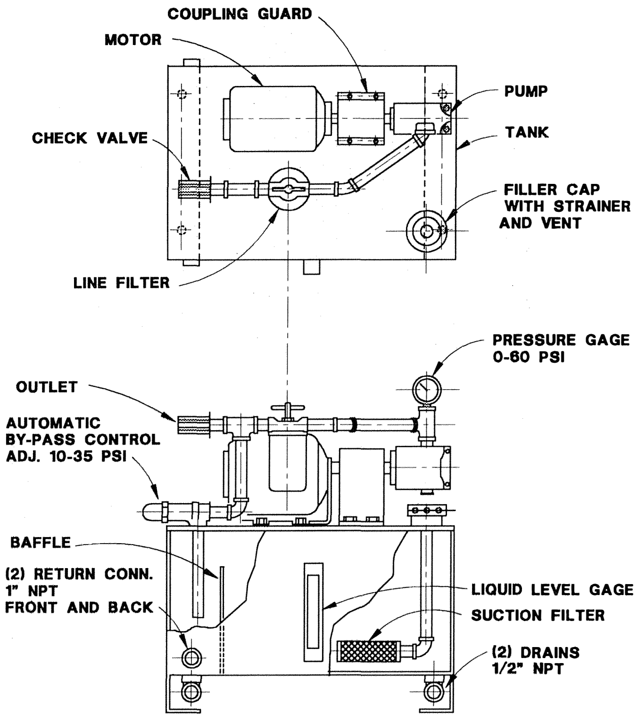


Figure 2. Lube Circulating Console.

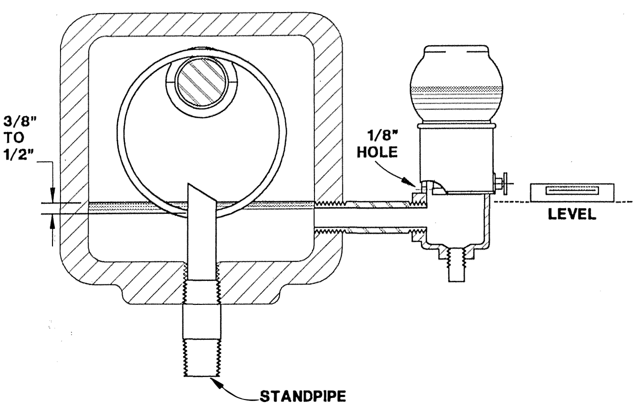


Figure 3. Standpipe Drain Used on Force Fed System.

packing box where the shaft exits the turbine casing. The water vapors then enter the bearing housing through the bearing housing shaft seals. The investigators concluded that to do a proper job of reducing contamination required addressing both of these areas.

Gland Seals

First, consider the stuffing box or gland seal. There are several causes for excessive leakage through the gland seal. The most common types of gland seals consist of a series of segmented carbon rings held in a stuffing box with garter springs (Figure 4).

Sealing is a combination of close clearances between the shaft and carbon ring, and the side of the carbon ring against the machined flat surfaces inside the stuffing box. In order for the rings to seal, they must maintain these close clearances. Things that affect these close clearances are worn carbon rings, carbon rings with excessive design clearances, or carbon rings that do not remain in alignment with the shaft.

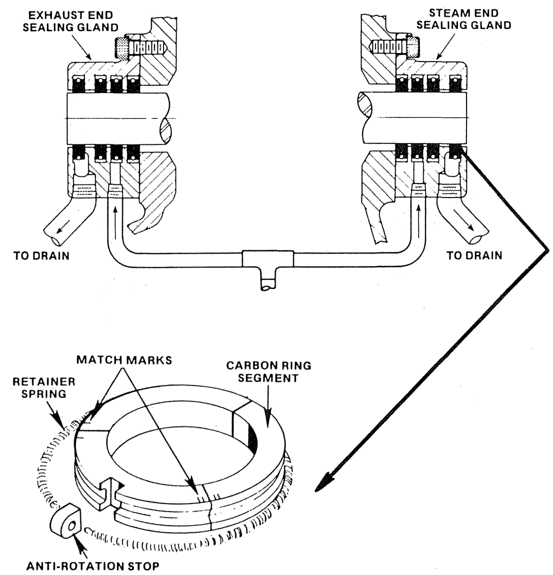


Figure 4. Seal Gland and Carbon Ring Detail.

To assure themselves they were using carbon rings with the proper clearance, an audit was performed on the carbon rings that were being supplied. Carbon rings supplied by the manufacturer have clearances based on the steam inlet temperatures that were specified. The refinery audit revealed that the machine shop was not discriminating between the use of carbon seals for 250 psi steam, 600 psi steam, saturated steam, or super heated steam. One manufacturer's guide for carbon ring selection appears in Table 2.

In addition, aftermarket suppliers had infiltrated the storehouse and were supplying carbon rings that did not distinguish between temperature application. This situation was brought to the Machine Shop Supervisor's and Purchase Manager's attention and has been resolved.

The clearance between the shaft and the carbon ring addressed only one leak path, the other being around the carbon ring to atmosphere. There are usually several segmented carbon rings in each sealing gland. The rings seal against lands machined into the gland housing. These lands must be perpendicular to the axis of the shaft in order to provide a 360 degree seal around the carbon ring. When these glands become worn, they must be periodically machined.

The investigators discovered that between the machining and installation process, these boxes may be cocked after they are mounted on the turbine case. To remedy this situation, more precise machining techniques were used, including dowelling the two halves of the gland together during the machining process and dowelling the stuffing box to the turbine case during installation. In addition, a mandrel was designed for each turbine size. Before the turbine rotor is installed in the bottom half of the turbine case, the mandrel is used to ensure that the bearing housings are concentric and the gland mounting surface of the turbine case is perpendicular to the axis of the shaft.

Steam turbines in condensing service have different sealing requirements than back pressure steam turbines. A condensing turbine requires an external source of low pressure steam on the exhaust end of the turbine along with the inlet end during startup or light load conditions. This is usually accomplished with an external source of makeup steam to inject at the gland seals. Steam leakage at the high pressure end increases with turbine load until the leakage steam pressure exceeds the external sealing pressure. The high pressure packing leakage will then furnish the steam to

Table 2. Carbon Ring Clearances Vs Exhaust Temperature.

Rotor Type / Exhaust Temperature (°F)	AYR Redesign	BYR and Old AYR	CYR and DYR Below 5000 rpm	BYRH Below 5000 rpm
	A-1, A-2, A-3	B-1, B-2, B-3 A-1, A-2, A-3	C-1, C-2 D-1, D-2	BH-1, BH-2, BH-3
300° AND BELOW	2.252 - 2.253	2.252 - 2.253	2.935 - 2.936	2.937 - 2.938
301° TO 400°	2.254 - 2.255	2.254 - 2.255	2.937 - 2.938	2.937 - 2.938
401° TO 500°	2.256 - 2.257	2.256 - 2.257	2.939 - 2.940	2.939 - 2.940

seal the exhaust end and progressively less external steam is required. On older design machines, the pressure to the shaft seals was regulated manually by the use of a globe valve that would bypass excess steam into the exhaust section of the steam turbine. This method turned out to be unreliable on steam turbines with variable load conditions. If the seal steam pressure was regulated at a time when there was light load on the turbine, then there would be excess steam leakage through the seals during a heavy load condition. Conversely, if the leakage rates were set when the machine was under heavy load, there would be a possibility of not having enough sealing steam at the back end of the turbine under lightly loaded conditions.

One particular machine is normally lightly loaded and is extremely sensitive to changes in exhaust pressures. Because the machine is normally lightly loaded, the makeup steam valve would open periodically to maintain set seal steam pressures.

When this happened, condensate that had collected in the dead leg of the makeup line would be injected into the shaft seals and the turbine would go into violent vibrations. Additional condensate knockout pots were added to the external seal steam supply line to prevent this from reoccurring.

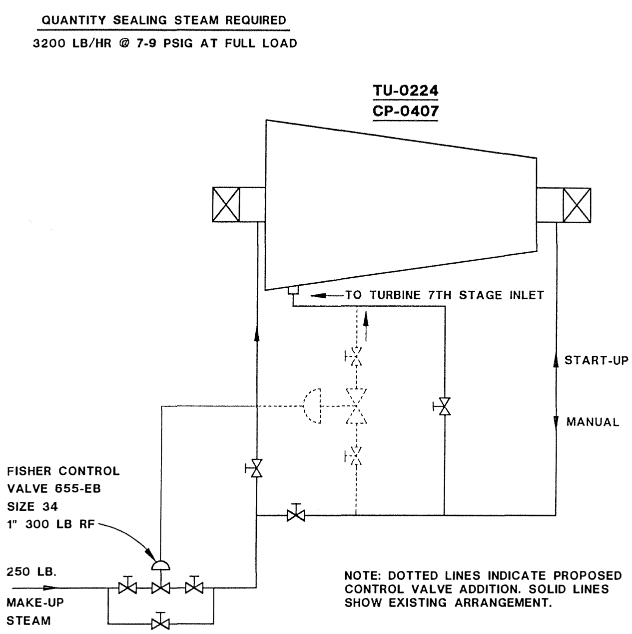


Figure 5. Seal Steam Control Scheme.

An illustration of how the aforementioned problems can be overcome is presented in Figure 5. Notice the original installation had only a globe valve to throttle the excess steam back into the exhaust section of the turbine. The dotted lines show the addition of a control valve to accomplish this task automatically, maintaining a constant seal steam pressure. Since the installation of the additional control valve, problems associated with shaft sealing have been eliminated on this machine.

Bearing Housing Shaft Seals

Preventing excessive amounts of steam from escaping the seal gland did not completely address lube oil contamination. The investigators were aware of methods to improve the efficiency of bearing housing shaft seals, and felt these techniques should be applied broadly and expeditiously in an effort to further improve lube oil cleanliness.

The easiest method for keeping moisture out of bearing housings is to purge it with clean dry air. Purge systems can be installed and commissioned without major disassembly.

Instrument air is fed through an air regulator to each bearing. Pressure is reduced to 2.0 to 3.0 psi and introduced to the housing cavity through a 1/32 in orifice. One word of caution: the oil level will drop about 1/16 in in the housing, so be sure the oil ring has proper submergence.

A more reliable technique to reduce contamination resides in the shaft seal itself. Two designs have been adopted to improve seal performance. One is the torturous path or interlocking labyrinth design, and the other is the air purge design (Figure 6). The latter design is a refinery standard on new machines. *Note:* At least two labyrinth teeth, as shown, are required so that an effective seal is not bypassed by the drain back groove.

Several manufacturers are providing an interlocking labyrinth type. These devices are very effective at keeping dirt and water out of the bearing housing; however, in many cases, they cannot be installed without some machine work on the housing to accept the stator ring.

The air purge labyrinth design is a more common retrofit on machines that have multiple tooth, split labyrinth inserts. These parts can usually be modified by adding a groove near the center of the labyrinth for air distribution, and drilling two air injection ports 90 degrees apart in the top bearing housing cap. The advantages to this method are that existing parts can usually be modified, and a minimum amount of disassembly is required.

The author's company has dramatically reduced the number of bearing failures as a result of the program to isolate bearing housings from the immediate environment. Future plans include oil mist systems; wherein, existing air purge systems could be easily converted, further improving machine reliability.

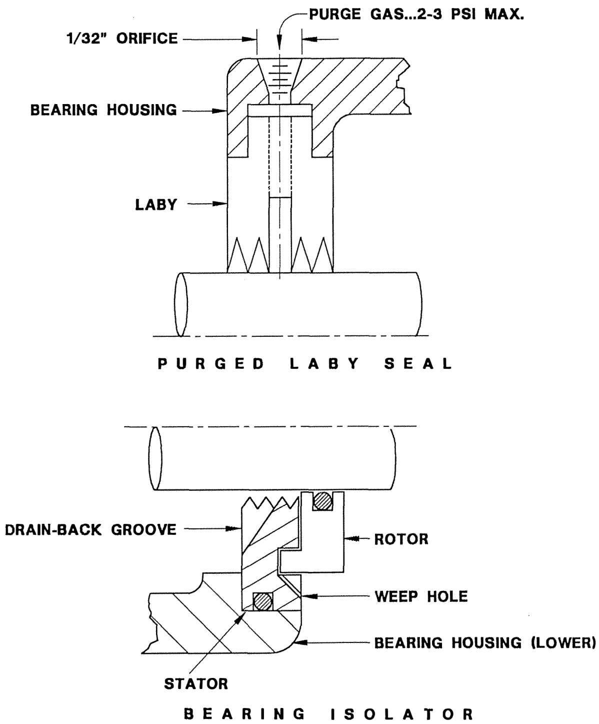


Figure 6. Purge and Isolator Type Bearing Housing Seal Details.

Lack of Lubrication

Up to this point, the discussion has focused on 600 lb to condensing, and 600 lb back pressure steam turbines and their associated bearing contamination problems. Attention should turn to the smaller 250 lb inlet, 435°F back pressure steam turbines.

For the most part, these steam turbines have remained ring oiled and have not been retrofitted with lube oil circulating consoles. The major mode of failure in these turbines has a been lack of lubricating oil to the bearings.

The ring oiler arrangement consists of a large oil ring, usually bronze, resting on the top of the shaft (Figure 3). As the shaft spins, the oil ring dips into the reservoir and lifts oil onto the shaft. Top bearing halves are slotted to allow the oil to distribute through the bearing.

Successful operation of the ring oiler system is dependent on satisfying several operating parameters. Oil viscosity, ring geometry, ring submergence, and shaft speed all impact the performance of the lubrication system. John Dufour and Ed Nelson of Amoco have quantified what the investigators herein have proven empirically through years of experimentation [2]. Their experience indicates that the maximum speed for the ring oilers occurs at journal surface velocities of 2000 to 2500 ft/min. That is, a shaft of 3.0 in in diameter has a maximum speed limitation of approximately 2500 rpm and a shaft of 4.0 in in diameter will have a speed limitation of approximately 1900 rpm.

Users have also found that:

· the amount of oil delivered to the shaft is roughly proportional to the area of the ring.

· at high speed, oil is thrown from the ring by centrifugal force on the upward journey, thus necessitating the use of special grooves to collect the oil and deliver it to the shaft.

· heavy rings deliver more oil than the light ones.

Ring diameters of 1-3/4 to two times the diameter of the shaft will provide adequate lubrication for about 4.0 in on either side of the ring with the help of oil grooving. If the bearing is more than 8.0 in long two rings will be required. The oil in the reservoir must be maintained so that the ring is emersed in at least a 3/8 in above the I.D. of the ring while the machine is running. Oil rings must be perfectly round to function. Note: Excessive submergence of the oil ring, say 3/4 to 1.0" above ring I.D., will impede oil ring rotation, reducing oil delivery to the shaft.

Many of the bearing failures discovered were caused by an inadequate amount of oil being delivered to the bearing by the oil ring. Investigation revealed that oil ring submergence was inadequate due to an improper oil level in the bearing housing. For some unbeknownst reason, some machinery manufacturers have insisted in using artificial methods for maintaining the oil level in bearing housings. That is, they use standpipes or "crows feet" or some mechanical device to keep the constant level oiler above its lowest most setting. These adjusting devices are rarely permanently attached to the oiler, and can be lost or maladjusted during a maintenance period.

Tosco has written into its standards, and the author has proposed to API, that constant level oilers be mounted at the prescribed oil level, with the oiler mounted in its lowest setting without the use of any artificial means to maintain oil levels. The company has also retrofitted most existing machines by drilling and tapping the constant level oiler filler pipe at the proper oil level in the bearing housing, thereby eliminating the use of crows feet or standpipes on the constant level oiler. By adhering to this policy, bearing failures caused by improper oil levels in the bearing housings have been eliminated.

To prevent overfilling a bearing housing, it is a good idea to slowly drain some oil until the constant level oiler "burps" whenever the reservoir is being refilled. Seeing an air bubble assures the operator that the oiler is feeding properly.

One last word on the prevention of bearing failures on ring oil lubricated bearings. Many years ago, the Avon Refinery was fortunate to have in its employ a very innovative machine shop foreman by the name of H. B. "Deb" DeBenedetti. Mr. DeBenedetti developed a clear plastic lucite reservoir that could be fitted on the bottom of bearing housings. He nicknamed these devices "deb" jugs (Figure 1). Every ring oiled bearing housing on every pump, turbine, and motor in the refinery has been outfitted with deb jugs. These clear jugs are an excellent indicator of the condition of oil in bearing housings.

Water, rust, metal particles, emulsion and dirt can be readily detected by drawing a small amount of oil through the jug by opening the cock valve. (Be sure to use a spring loaded, notched cock valve that will not vibrate open. Use a locking compound on the valve threads.) Color photographs of DEB jugs with the most commonly seen types of bearing oil contamination are posted in each control room. The condition of the oil in the jugs is surveyed each shift by the operators and weekly jug condition reports are submitted to operating supervisors for further action on any bearings that are termed suspicious.

Since installing these jugs back in the 1960s, virtually all severe bearing housing failures caused by bearings seizing on the shaft and spinning in the housings have been eliminated. Of all the preventive maintenance techniques that have been developed over the years for use at Avon, the deb jug has probably had the most impact on machinery reliability.

ALIGNMENT

After the primary modes of failure had been remedied, secondary and tertiary weak links in the chain became predominant. In this case, misalignment and resulting coupling failures became the primary modes of failure.

The reverse indicator method of alignment had been introduced to the refinery about 30 years ago, and while the author was in the machine shop, it was the only acceptable method for doing machine alignments. However, with the increasing use of contractors, alternate methods of alignment began to creep in, infiltrate, and influence the machinists. It was not until they began to use vibration analysis as a maintenance tool that the investigators discovered a large number of machine trains in the refinery were out of acceptable alignment tolerance.

By going out into the field and observing the machinists, it was discovered that the alignment techniques being used were rim and face or one indicator attached from the driver to the driven machine.

These methods are inaccurate and can lead to large misalignment tolerances in machine trains. In order to get back to using reverse indicators as the only acceptable method, a comprehensive training program was initiated that included all of the machinists in the refinery, including the contractors that were used regularly. The graph technique was incorporated with the reverse indicator method, so that a permanent record of a machine's alignment could be recorded and maintained. There are several excellent publications on alignment, including Ray Dodds' Total Alignment [3].

Reinstitution of the reverse indicator method of alignment has resulted in significantly reduced vibration levels and coupling failures on all types of machinery. Laser alignment equipment has now displaced dial indicators, but the company still insists that machinists understand the principles of alignment.

COUPLINGS

With the shrinking of the oil industry, there has also been a corresponding shrinking of the maintenance forces within the refinery. The pressure is to do more with less. Since there are fewer mechanics to do preventive maintenance in the refinery, it is imperative that the machinery be designed to reduce the amount of maintenance required. Gear type couplings, which were at one time very popular, require periodic maintenance to inspect and service them. For that reason, dry type couplings have been made the standard in the refinery (Figure 7).

Dry type couplings can be either single diaphragm type, multiple diaphragm, or multiple disc. Other types of dry type couplings are neoprene or rubber ball type (elastomeric). For general purpose turbines, we have specified multiple disc type couplings. Dry type couplings require no maintenance and are very easy to inspect. They are axially compliant and do not transmit high axial forces.

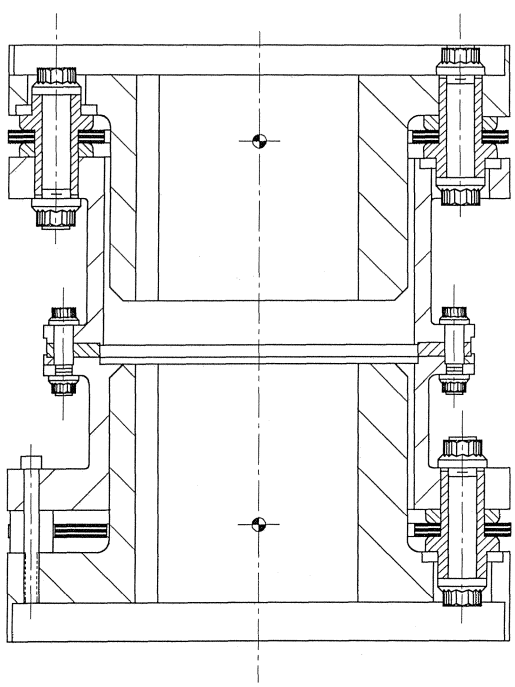


Figure 7. Multiple Disc, Flexible Element Dry Coupling.

The refinery is currently embarking on a program of having retrofit stock bore hubs and spacers for dry couplings available in the machine shop. Then, when a machine comes in with the gear type, it can be readily converted to a dry type coupling.

Several custom designed dry type couplings have been retrofitted on large turbomachines, specifically to deal with dynamic problems that have been aggravated from the use of gear type couplings. The investigators feel that retrofitting these machines with dry type couplings has solved some very serious operating problems and allowed continual operation of these critical machine trains [4].

SPEED CONTROL

The last major retrofit of small steam turbines at the refinery has been the conversion from mechanical governors to hydraulic governors. A typical shaft type mechanical governor is illustrated in Figure 8.

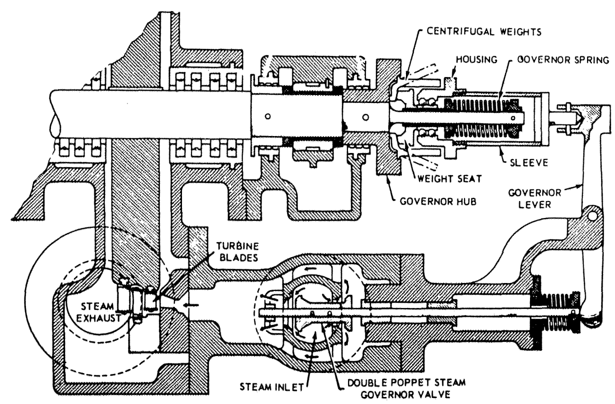


Figure 8. Mechanical Governor Detail.

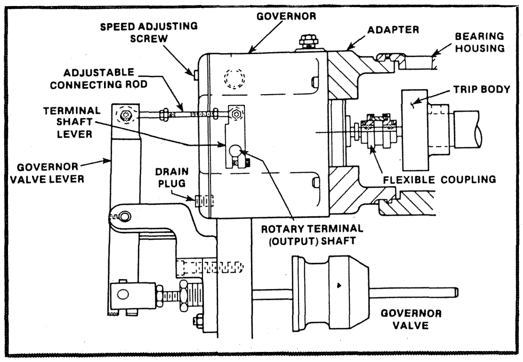
Hydraulic governors have several advantages over mechanical governors. They are more accurate, easier to maintain and adjust, and are inherently more reliable and safe. An example of a T-type hydraulic governor is presented in Figure 9.

The company overhauls and checks these governors in their machine shop. They are tested at running speed and under load on a test stand designed for that purpose. Spare governors in all speed ranges are kept on hand. To the author's knowledge, they have had no catastrophic turbine failures attributable to governor failure on any of the turbines outfitted with hydraulic governors.

OVERSPEED MECHANISMS

Catastrophic failures due to overspeed trip malfunctions have not been a major problem at the Avon Refinery, although they have the potential to do so. Procedures have been established long ago to perform periodic maintenance checks on overspeed mechanisms.

Each time a turbine is taken out of service, it is tripped manually by the operator. Every time a turbine is uncoupled from a driven piece of machinery, the overspeed valve and the overspeed trip mechanism are verified to be in the correct operating order by overspeed tripping the turbine. In addition, any time a turbine is being put back into service, the inlet supply valve is opened against a closed overspeed trip valve. If the turbine begins to turn at all, the steam inlet valve is immediately shut, and tagged off, and a work request is issued to have the overspeed trip valve repaired.



(TOP) EXTERNAL AND (BOTTOM) INTERNAL DETAIL OF A HYDRAULIC GOVERNOR

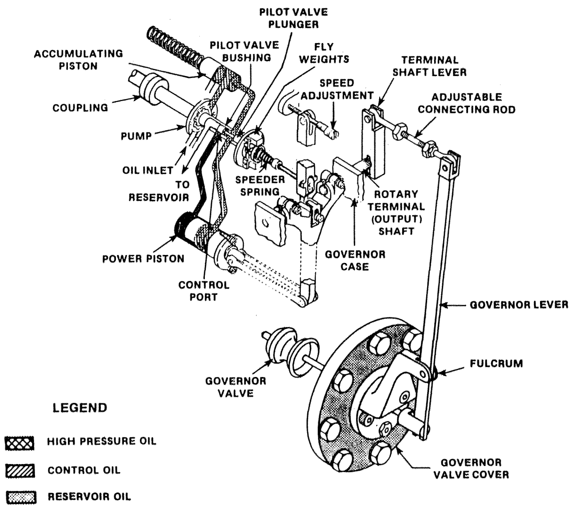


Figure 9. (T●p) External and (Bottom) Internal Detail of a Hydraulic Governor.

Older steam turbines that have not or cannot be retrofitted with hydraulic governors are included in a routine preventive maintenance schedule. These turbines can then be scheduled to have their overspeed trips checked periodically to see that they are in proper working order.

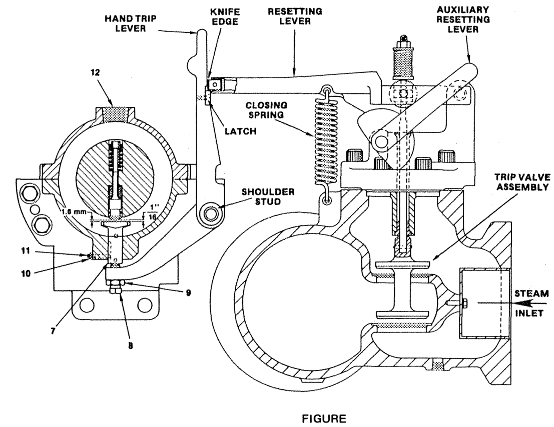
Overspeed mechanisms have two major components (Figure 10). The first component is a spring loaded weight mounted on the shaft. As the shaft begins to overspeed, centrifugal force overcomes spring tension and the weight begins to move outward until it strikes a trip mechanism.

The second component in the overspeed trip system is a valve which shuts the incoming steam off to the nozzle ring, thereby preventing any more energy from being transmitted through the shaft.

Recent design developments on newer steam turbines have made the overspeed trip function much more reliable. The first of these is the Belleville spring type trip or posi-trip (Figure 11).

The posi-trip is a cone shaped disc with a series of weights on the rim of the cone. Centrifugal force acts upon the weights to straighten the cone out and make it a flat disc. As it reaches its "flat position," it goes from a convex to a concave shape and strikes the trip mechanism. The manufacturer claims that these devices are accurate to 1/10 of 1.0 percent. In the investigators' experience, they have never had to readjust this mechanism because of its accuracy and repeatability.

The second innovation has been the introduction of an electronic tripping device. These tripping devices usually consist of two magnetic pickups reading a toothed wheel mounted on the shaft,



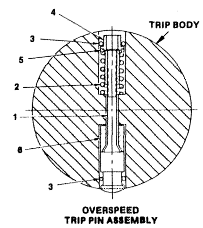


FIGURE ITEM NUMBER	DESCRIPTION	QUANTITY
4-7-1	TRIP PIN	1
2	TRIP SPRING	1
3	"U" LOCK STAPLE	2
4	ADJUSTING NUT	1
5	WASHER	
6	AUXILIARY WEIGHT	1
7	PLUNGER ASSEMBLY	1
8	JACKSCREW	1
9	JAM NUT	1
10	JAM NUT	1
11	SETSCREW	1
12	INSPECTION PLUG	1

Figure 10. Overspeed Trip Mechanism and Shut Off Valve.

connected to an electronic speed sensing device such as a tachometer or an electronic governor. Tripping is achieved by activating a solenoid operated dump valve on the sustaining cylinder on the trip and throttle valve. A simultaneous signal to close can also be sent to the turbine admission valves, as an additional level of safety.

Typically, the electronic overspeed is set 10 percent above maximum continuous running speed, and the mechanical overspeed device is set at 12 percent. (±1.0 percent) These devices can also sense an underspeed condition which can alert the operators when a turbine slows below the minimum speed setting range. The last several large machines purchased at the Avon Refinery have had both electronic and mechanical overspeed trip mechanisms.

PERFORMANCE MONITORING

Most, if not all, maintenance is predicated on measurements taken when the machine is new. The measurements taken on a machine when it is new are termed the baseline readings, and all subsequent readings are usually compared against those. These readings are again taken after each overhaul and a new baseline is established.

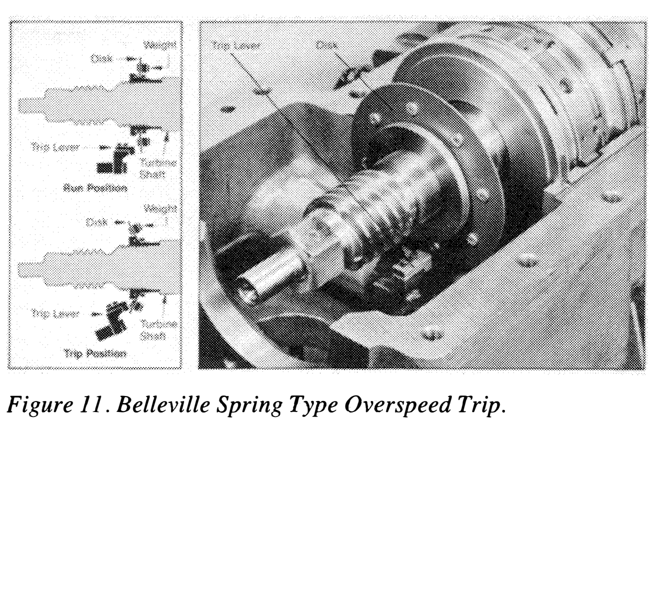


Figure 11. Belleville Spring Type Overspeed Trip.

There are basically two types of baseline measurements that are taken. The first is mechanical baseline information. That is, vibration, temperature, speed, sound, etc. The second performance parameter is the measurement of the energy efficiency of the steam turbine. Most users have been doing performance monitoring, both mechanical and efficiency, on large steam turbines for many years.

Mechanical Performance

With energy costs rising, and refineries operating at near capacity, the importance of keeping small steam turbines running efficiently has become more critical. The refinery herein has dedicated preventive maintenance technicians that periodically check vibration, temperature, and general condition on all general purpose steam turbines and compare them against baseline readings. Computerized data collection systems are being utilized to perform this function. The information is being collected in a hand held data collector. The data is returned to the Reliability Center, where it is entered into a PC based computer system, so that it can be compared against the baseline, analyzed, trended, and reported.

Dynamic Efficiency

The efficiencies of small steam turbines are assessed by monitoring the steam chest pressure. Each small turbine has a gauge mounted in the steam chest after the governor valve. By monitoring the valve position and the steam chest pressure, readings can be compared to baseline and an assessment can be made as to the mechanical condition of the machine. Of course, accurate performance data on the driven machine are also required.

On some larger machines, flow measurement devices have been installed in the inlet lines of the steam turbines and total steam flow to the turbine is being measured. On older 600 lb to condensing steam turbines, the total condensate flow is measured and a back calculation is made to determine how much steam is being consumed by the turbine.

With 144 small steam turbines, it is easy to see how millions of dollars per year in energy savings can be realized by monitoring the performance of these machines, and taking corrective action to restore performance.

CONCLUSION

Several problem areas and how they were addressed have been discussed. The reader's experience may be different. The investigators have been blessed with dry, clean steam and have not had the problems of fouling and blade deterioration that many other papers have discussed.

As an industry, everyone can do a better job of operating and maintaining equipment. There are always new methods to reduce maintenance costs, increase onstream time, and improve reliability. Advances in lubrication, oil mist, speed control and performance monitoring always whet appetites for more reliable machines. It is extremely important that users continue to provide feedback to machinery manufacturers and standard writing organizations so that they can refine and improve machine designs. It is also important that users provide input through the API, ASTM, ASME, and other organizations dedicated to improvements in the reliability and safety of rotating machines.

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