PROBLEMS ENCOUNTERED IN BOILER FEED PUMP OPERATION

by

Dr. Elemer Makay

Technical Director

Energy Research & Consultants Corporation

Morrisville, Pennsylvania

Elemer Makay received his Ph.D. from the University of Pennsylvania. Prior to that, he attended the Technical University of Budapest, where he received his Master's Degree in Mechanical Engineering. Dr. Makay has worked with BBC and Westinghouse in turbine design and development and with DeLaval in design, development and testing, primarily of boiler feed pumps. Since 1970, he has acted as a utility pump advisor and troubleshooter for the electric utility companies. Since 1975, he has been official scorekeeper of large pump failures for the utility companies. He is the first author of a book on "Nuclear Primary Coolant Pumps for Light Water Cooled Nuclear Reactors" and author of several technical papers on centrifugal pumps, bearing and rotor dynamics technology and field measuring techniques.

ABSTRACT

Although mechanical and hydraulic instabilities in centrifugal pumps caused a considerable amount of problems for large Nuclear and Fossil Generating Stations influencing "Power Plant Availability," a relatively limited amount of research work was done in this area. The author makes an attempt to outline problems in general, gives definitions for hydraulic and dynamic "Instabilities," outlines the "Mechanisms" that create them, establishes safe operating speed and flow ranges for large pumps, and discusses how to treat the above subjects when field failures occurred. Interaction between hydraulically induced forces and bearing design parameters, and their influence on rotor vibration characteristics is emphasized. Friction induced partial frequency modes are also discussed. These forces are sizable and can influence rotor and bearing design requirements, can change rotor stability and the appearance of the pump rotor critical speeds. Comparison is made between vertically and horizontally oriented pumps and between volute and diffuser types. The author also outlines the minimum instrumentation necessary to identify problem areas before and after failure occurs and supports this with actual field failure cases. It is outlined how a simple factory (witness) test can uncover potential future field problems. Areas where systematic future research and development work is necessary are pointed out.

INTRODUCTION

As sizes of Nuclear and Fossil Power Generating Stations grow, the unexpected and unexplained number of large pump failures or pump-caused system operating difficulties also grows. As one begins to pay closer attention to these problems, it becomes clear that a large percentage of these failures can be traced to rotor instability (at least in appearance), occurring especially in the low percent flow (or plant load) regimes; in particular, in the "Recirculation" (minimum flow) mode.

Pump problems, as well as pump-caused feed water system problems, can be categorized in the following way:

- Pump types (single or multi-stage, diffuser or volute, horizontal or vertical, etc.).
- Vendors.
- Manufacturing, Q.A. (Quality Assurance).
- Technology (Hydraulics, Lubrication, Rotor-Dynamics, etc.).
- Design concepts.
- Feed water systems.
- Application (Turbine or motor-driven, constant or variable speed).
- Operation (cycling or base load plant).
- Maintenance and Service, etc.

However, at this point the most important objective is to identify basic causes of failures (or operating difficulties) called GENERIC problems that are directly applicable in general to all pump products. The author has been and is conducting a failure survey and failure evaluation for the U.S. Utilities on utility pump applications with special emphasis on boiler feed and reactor feed pumps. The survey this far has allowed us to identify several typical problem areas. Some of these are:

- Pump hydraulic instability (see Figure 1).
- Bearing technology and rotor-dynamics.
- Seals (especially for Nuclear Primary Coolant Pumps).
- Axial balancing devices (balance drum and disk).
- Artificially pushed pump efficiencies.
- Lack of Standards for pressure pulsation amplitudes and frequencies, allowable vibration levels at various frequencies, pump shaft axial vibration levels and frequencies.

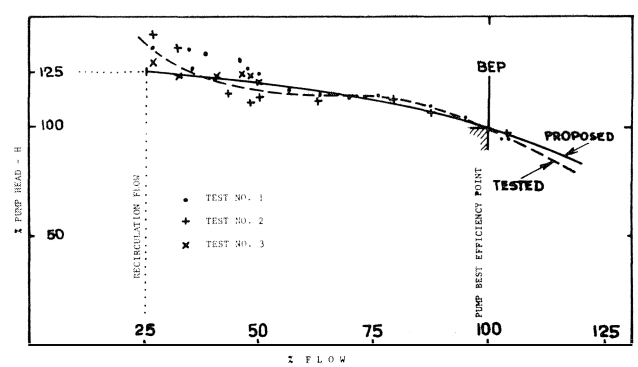


Figure 1. Head-Capacity Curve of a Hydraulically Unstable Multi-Stage Boiler Feed Pump. Parallel operation is difficult at lower flows. Head is unstable below approximately 65 percent flow resulting in various pump or F.W. system malfunctioning. Problem caused by the impeller design.

- Lack of good Specifications for procurement.
- Lack of manufacturing quality assurance.

Explanation of these areas is the next step; namely, what causes these discrepancies, and more important, how do we correct for them, or in case of a new application, how can we avoid them. Reference 1 gives a thorough discussion on how these problems may be avoided when:

- Purchasing equipment.
- Pump bought but not yet tested at the factory.
- Pump delivered to the plant site, but not yet started.
- Pump failed on-line in plant.

RADIAL OR AXIAL ROTOR RESPONSE

Different pump types will exhibit different (in appearance) rotor response. If the pump is a

- Heavy, vertical pump such as a P.C.P. (Primary Coolant Pump for pressurized water reactor), rotor response is usually RADIAL, and appears as a bearing or seal problem.
- Light, horizontal pump without a balancing device, such as a Reactor Feed Pump or a Condensate Booster pump, the dominant rotor response can be both axial and radial in the low-flow regime, such as, at recirculation (minimum) flow.
- Multi-stage boiler feed pump: Rotor response is radial, but often accompanied by frequent balance-device, seal, or thrust bearing failure. For cycling power stations, the result may be fatigue failure of the shaft at the balance-device or at the thrust bearing location.

The cause of these different responses are hydraulically induced dynamic forces, and the resulting responses depend on the pump type. The pump rotor will respond in the weakest direction, such as radially for a Primary Coolant Pump (PCP), and axially for a nuclear feed pump or for a single-stage, double suction booster pump.

Figures 2 to 5 make an attempt to explain what pump-induced responses are. The first impression during actual testing is that only low frequency (N/4) pressure pulsation is present when observing part (a) or part (b) of Figure 3. Part (c) of Figure 3 shows shaft axial vibration as the measuring stick of what the minimum pump flow should be. Part (c) of Figure 4 shows that a very low beat frequency response is also present. In the power station, without a magnetic tape recorder, only low frequencies showed up, using a chart recorder alone. The information was then put on magnetic tape and analyzed with more sophisticated electronic equipment such as frequency analyser, as shown in Figure 4 (b). This tape was played back 24 times slower to see the exact formation of pressure waves at different frequencies. The blade passing frequency is not only clear, but its amplitude is significant, almost half of the overall total amplitude.

The question comes up frequently these days, "What are the 'standards,' if any, for vibration in the axial direction and the peak-to-peak pressure pulsation in the various parts of the pump and the feed water system?" These questions will be answered in the corresponding sections below.

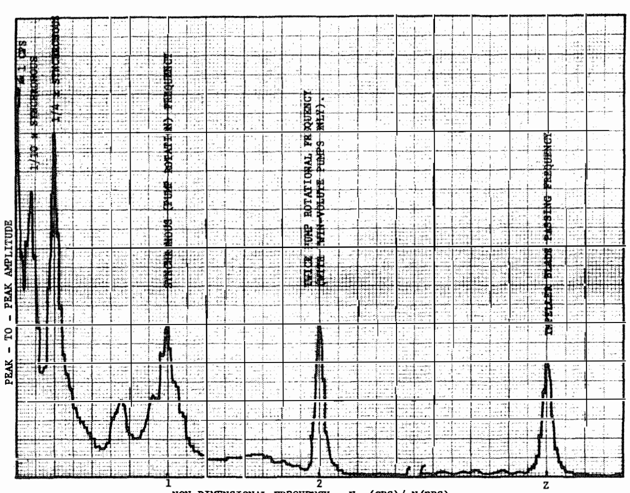


Figure 2. Pressure Pulsation Measured in the Nuclear Feed Pump Discharge Nozzle Shows Sub- and Super-Synchronous Frequency Components.

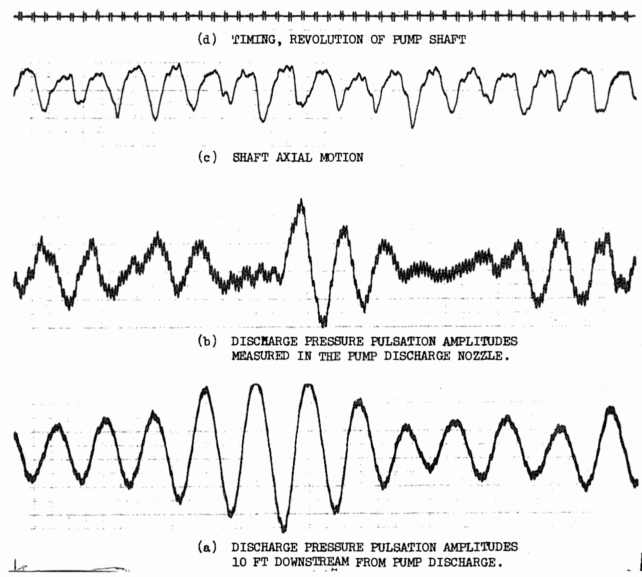


Figure 3. Peak-To-Peak Discharge Pressure Pulsation Measured With Differential Pressure Transducer. Timing (rev. of pump shaft), and shaft axial motion are also shown for comparison. High speed single stage double suction nuclear feed pump.

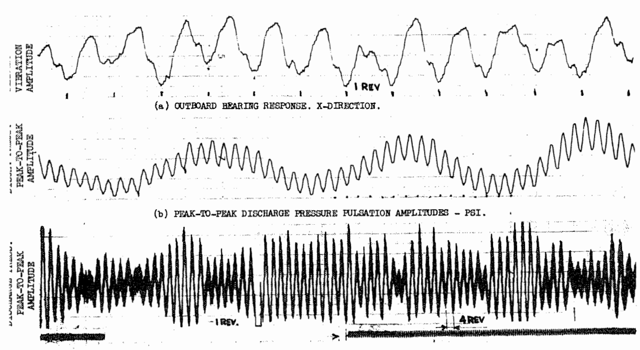


Figure 4. Pressure Pulsation Amplitudes Measured in the Discharge Nozzle of a High Speed, Single Stage, Double Suction Nuclear Feed Pump. Data played back with two different tape speeds to show high and low frequencies. Pump rev. and outboard bearing response are also shown for comparison. Pump on recirculation line.

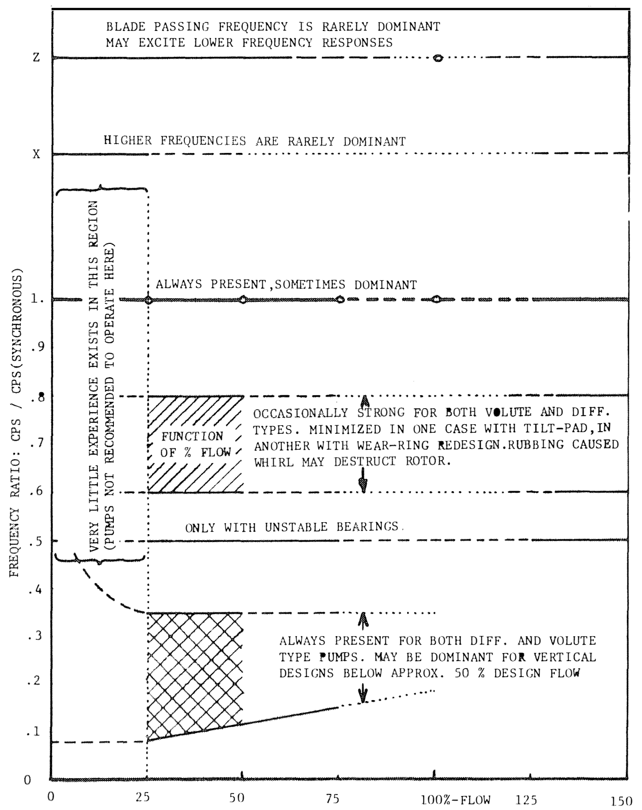


Figure 5. Synchronous and Sub-Synchronous Frequencies of Hydraulically Induced Dynamic Forces.

DYNAMIC AND HYDRAULIC INSTABILITIES

Definition of DYNAMIC instability is given in References 4, 5, 12, and 13 and in many other publications. Subjects such as bearings, seals, wear-rings, rubbing-induced whirl of a shaft, oil whip, etc., are relevant here. Most of them are relatively well defined areas; therefore, discussion of these subjects is neglected here (with the exception of bearing stability).

The definition of HYDRAULIC instability, especially at low percent pump flows, is given in Reference 12, outlining the mechanisms creating them. Hydraulic instability is also classified by pump stage geometry, which is more preferred by pump design engineers. Figures 6 and 7 show, in a simplified way, flow mechanisms at various geometries of a pump stage.

MECHANISMS OF HYDRAULIC INSTABILITY

The following flow mechanisms play an important role in creating pump rotor instability. To what degree they are influential is not yet completely known today. For corresponding geometry see Figures 6 and 7.

- Secondary flows (Figure 6).
- Stall (Figure 7).
- Leakage flow: such as, through wear-rings.
  - Hydro-dynamic (matrix of four coefficients)
  - Hydro-static (matrix of four coefficients)
- Unsteady flow fluctuations
- Wake (blade passing frequencies).
- Turbulence.
- Cavitation.

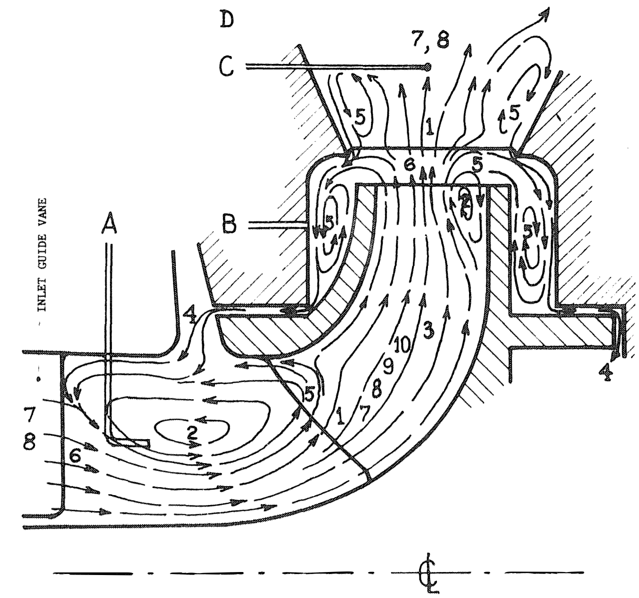
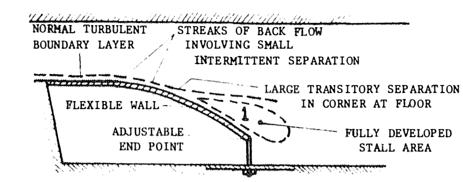
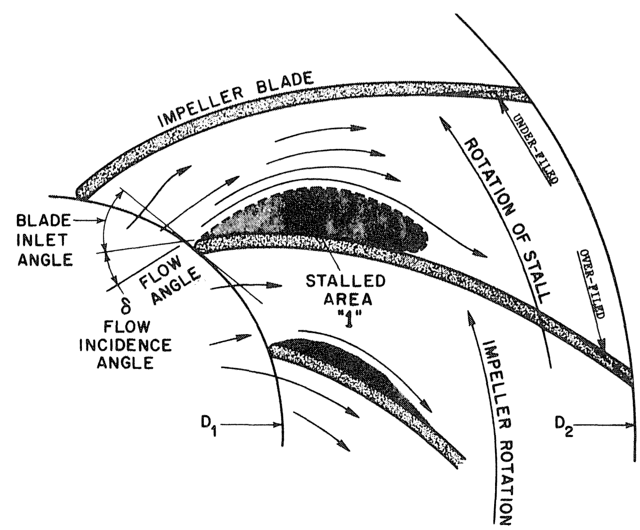


Figure 6. Secondary Flows in and Around a Pump Impeller Stage at Off-Design Flow Operation. Formation of STALL (1) and RECIRCULATION flows (2, 3 & 5) are shown separately in Figure 7.



(a) DEVELOPMENT OF STALL ON A CURVED WALL



(b) FORMATION OF STALL IN AN IMPELLER EYE DUE TO FLOW INCIDENCE ANGLE (VISUALIZED ON EXPERIMENTAL TEST RIG)

Figure 7. Formation of Stall (a) on a Curved Wall, and (b) in an Impeller Eye (Rotating Stall)

- Equivalent hydro-dynamic mass.
- Hydraulic unbalance.

The last two items are not really flow mechanisms, but are important phenomena to be listed. Both may cause many undetectable headaches for centrifugal pumps, especailly, for nuclear primary coolant pumps.

PUMP STAGE GEOMETRY

Most pump designers like to have a classification of pump problems by pump geometry, rather than by flow mechanisms or other classifications. Again, with the aid of Figures 6 and 7, we can categorize our objectives by pump stage geometry, such as:

- Impeller eye (inlet) instability.
- Impeller discharge instability.
- Internal instability (impeller and/or diffuser).
- External instability (gap behind impeller hub and shroud, unusually important for single stage double suction reactor feed pumps).
- Wear-ring interaction.
- Casing interaction.
- Hydrodynamic whirl (when rotor motion sets in, physical motion of the rotor gives rise to additional hydraulic actions).

Hydraulic instability can be detected by examining the pump Head-Capacity curve (see Figure 1). If the curve is flat toward lower flows, or especially if it is "droopy," most likely pump or system vibration will be a problem. Many times the problem comes from the fact that architect-engineers are pushing for higher than possible efficiency, that can be achieved only by designing the pump hydraulic components for the guarantee point alone. This then results in unstable operation at lower flows. In most cases only a new, carefully designed pump impeller can solve the problem when hydraulic instability is present. If a pump is purchased but not yet delivered, insist on a good shop test with flow-wise equally spaced test points. If a point is higher in the mid-capacities than neighboring points, do not accept it, insist on retesting. If it is a new proposal, properly written and executed, specifications can take care of this phenomenon. Most specifications do not press this subject hard enough. Hydraulic instability is the number one "Outage producing" phenomenon, due to the fact that the vibration responses receive endless energy exciting inputs from stall, blade passing, and other sources.

PUMP EFFICIENCY

Artificially inflated high efficiencies are "*chased*" by many architect engineers. Reference 11 (p. 62, first paragraph) gives a clear explanation of this phenomenon. Actually efficiency does not have to be sacrificed to obtain good pump reliability. However, a demand for unreasonably high efficiency will result in lowered machine reliability.

DIFFUSER VERSUS VOLUTE TYPE PUMPS

This question comes up frequently in power stations — which one do you recommend? It really depends on the individual designer's ability, flow range, efficiency and stability.

Formation of instability in the diffuser passages strongly depends on the deceleration rate of the fluid flow. The diffuser decelerates the flow, therefore, it is more receptive to separation and stall formation; hence to part load instability. On the otherhand, at the end of a volute the flow usually is dumped into the discharge nozzle which then creates other problems. This is due to the fact that the accumulated effects of the blades, volute tongue, and diffuser blade passing frequencies, affect the control system, especially for a boiling water reactor system where the reactor water level control is a sensitive item.

SEALS

Available technology for seal design is marginal for high speed application, for any other than labyrinth (break-down bushing) type seals. Also, for the more complex seal types continuous skilled attention is needed at the site which often is not available. Labyrinth seals are the least demanding work horses of utility pumps. In order to avoid future disputes, insist on having labyrinth seals. The only reason for other seal types (mechanical, controlled leakage, etc.) is to bring hydraulic losses (leakage) down, so the apparent efficiency of the machine looks good, and guaranteed commitments can be achieved.

BALANCING DEVICES

Axial balance disk devices are frequent causes of failures. From technology point of view, they belong to the same category as seals. Sometimes lack of space available in a design prevents application of proper dimensions which makes the device receptive to pump internal (pump hydraulic instability at lower flows, dynamic unbalance of components, bearing instability, friction induced shaft whirl, etc.) or external excitations, resulting in unnecessary failures. You may hear from the vendor, that the disk is the safety valve of the pump, namely it will fail before any other component in the pump. It is not so. A properly designed disk can last forever, and can save the pump from destruction.

BEARINGS, ROTOR DYNAMICS, OIL WHIP, FRICTION INDUCED WHIRL

If a problem is traced to a bearing design, or bearing type caused deficiency, selection of proper bearing parameters (L/D ratio, radial clearance, bearing load, spring and damping coefficients, etc.) will in many cases result in a successful solution. In a new machine insist in performing the proper rotordynamics analysis (critical speed, shaft deflection, rotor response, bearing stability) in the speed range of application, and to verify the results during acceptance test. A good coast down test for example can verify the location of critical speeds. If the pump maximum operating speed is above 1.5 times the first lateral critical speed, insist in furnishing the pump with stable type journal bearings. The last page of Reference 4 outlines this.

Figure 8 (also Reference 4) shows in an oversimplified manner the effects of the bearing principal spring coefficients on rotor critical speeds. In general it is recommended for flexible shafts to apply dynamically stable bearings such as tilting-pad type, which are "inherently" stable in theory. A rotor is called rigid if it operates below the first critical speed, above that, it is called flexible. The author finds occasionally the bear-

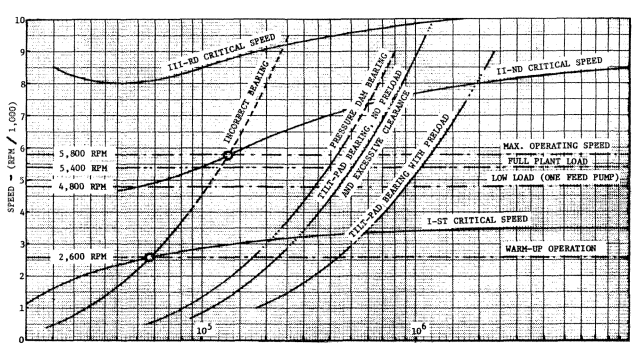


Figure 8. Effects of Bearing Principal Spring Coefficients on Rotor Critical Speeds for a Boiler Feed Pump Turbine Drive with Various Bearing Types. Turbine and feed pump are connected with a flexible coupling, hence the critical speeds represent the turbine only.

ing instability cases with high speed pumps with tilting-pad bearings. Figure 9 shows one such case where the multi-stage boiler feed pump suddenly becomes unstable at twice the first critical speed. The frequency of instability was classical before changing to tilting-pad bearing. The new bearing eliminated the classical half-frequency whirl, but not at the low frequency as shown in Figure 9. The instability sets in only at certain pump flow conditions and only above twice the first critical speed.

Figure 10 (from Reference 6) shows the behavior of a single-stage double suction nuclear reactor feed pump. The original application of these pump types was low speed booster (1800 RPM). Economical reasons dictated a speed-up to 3600 RPM for the booster, and to above 5000 RPM for the nuclear feed pump applications. High speed booster pumps exhibited more than usual field problems but mainly at low flows, especially when minimum flow (recirculation) was set to less than 25% pump design flow. Most cases were looked upon as system design problems, especially piping geometry. Significant amount of money was spent on piping changes with very little success. With constant speed reactor feed pumps (Figure 10 shows one at 5400 RPM) we found serious problems as shown in Figure 10, cases "a" to "d." In case "d" the pump became unstable in the low flow area at 3600 RPM. At 4000 RPM the only remaining stable point was the design point, and the pump was unable to operate above 4000 RPM in any flow range. Redesigned bearings allowed the pump to operate in the range shown by line "c." Partial corrections to the hydraulic components resulted in line "b," while "a" could be achieved only by complete redesign of the pump internals. Present U.S. practice is to apply a "variable speed" drive. If the drive is an electric motor, the "Synchro-Pak" (which has hydraulic coupling with a built-in step-up gear) is a most economical arrangement. Constant speed reactor feed pumps can be found in the U.S. only in smaller and older nuclear units, and with unfavorable experience.

Figure 11 shows shaft vibration amplitudes at various frequencies for case "d" of Figure 10 when the pump was operating at the instability limit line. Driving the pump into the unstable regime by any small amount resulted in rapid increase of the half-frequency component. In two instances the maintenance crew learned that, if such vibration sets in, it will completely destroy the pump rotor. Application of tilting-pad bearings did not completely solve the problem, however sudden destruction of the rotor did not occur again.

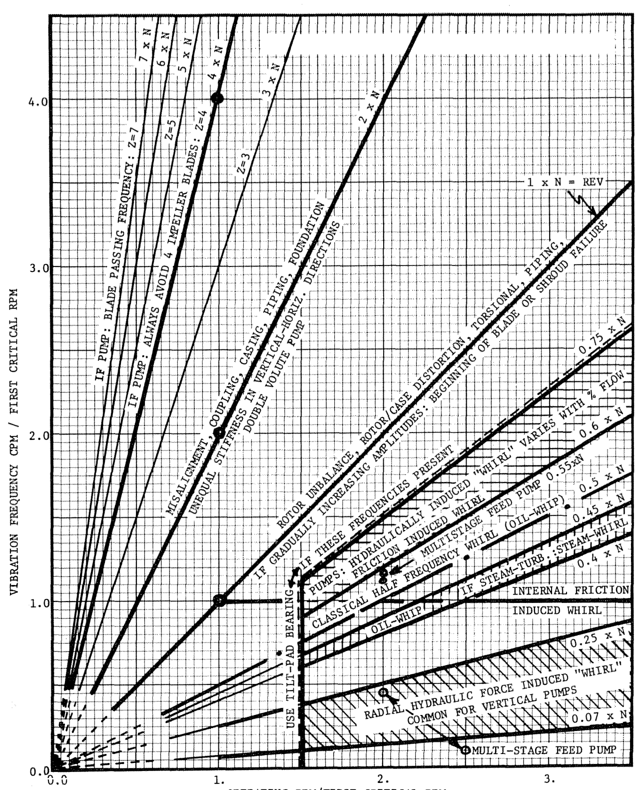


Figure 9. Vibration Frequency — vs — Speed. Causes and cures of common vibration problems of centrifugal pumps, steam & gas turbines, fans & blowers, compressors, and other centrifugal equipment.

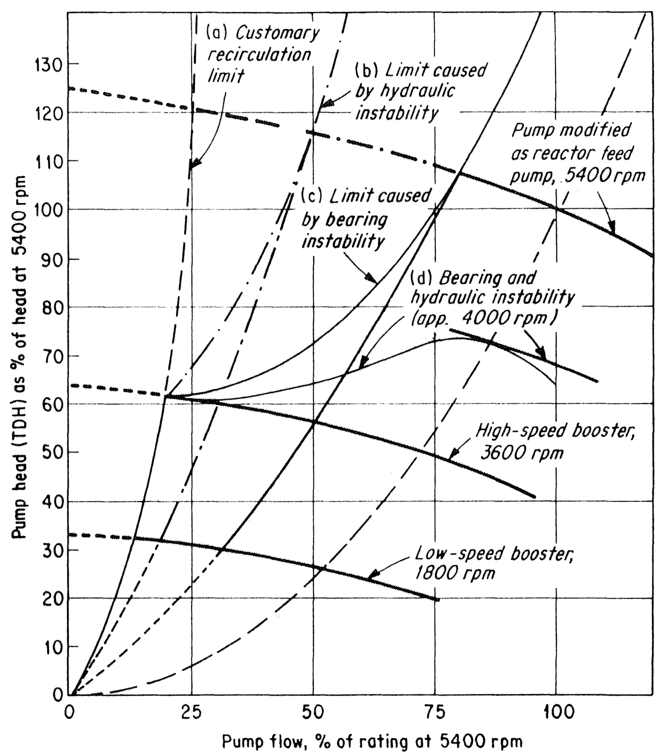


Figure 10. Pump Head-Flow Characteristics at 5400 RPM.

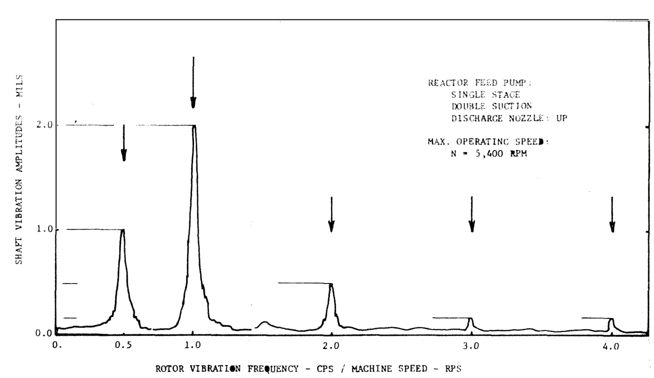


Figure 11. Hydraulically Induced Instability Becomes Strong Enough to Destabilize the Outboard Bearing, Which Then Develops Half-Frequency Whirl (N/2). If bearing is stable, it can still develop an apparent half-frequency whirl when rotor rubbing occurs, but then frequency is function of % pump flow.

Figure 9 (the "Sun-burst" chart) is an experience chart and can be used to diagnose cases where the vibration frequencies can be determined. If vibration amplitudes appear with frequencies lower than the operating speed, the instruction in Figure 9 is: "Use tilting-pad bearings." In general, they bring help in such cases, but one should realize that bearings can cope with hydraulic forces only up to some degree; beyond that, the solution lies in the design of the pump hydraulic compenents. Hydraulic forces, such as wake effect at blade passing frequency, or radial forces induced by impeller to diffuser/volute interaction, can be strong enough to excite the rotor to large amplitudes. If this is coupled with dynamic unbalance of the rotor, rubbing at close clearance surfaces may occur that can induce a counterrotational whirl with a frequency less than rotational. This is a self-exciting type vibration, and if unbounded, will gradually or sometimes violently grow and destruct the rotor. A double frequency component may also appear, giving the indication of misalignment, as shown in Figure 11. Counter-rotational frequency vibrations are unpredictable. The lowest frequency the author measured was 0.08 x rotational speed, and the highest 0.75 x rotational speed. For variable speed machines this frequency is a function of flow and speed, while for a constant speed boiler feed pump it is a function of pump flow alone. The whirl frequency ration usually increases with increasing pump flow, as shown in Figure 9.

RADIAL HYDRAULIC FORCES

References 10 and 13 discuss the subject in great details. One must distinguish between static and dynamic forces, both increasing in magnitude toward lower flows, having a maximum at minimum (recirculation) flow. Unfortunately the angles of the static forces, and the frequencies of the dynamic forces cannot be calculated; they have to be measured for each individual case. More important, each design has its own pattern. Results cannot just be extrapolated from one design to another. This is one area where future work is essential.

STANDARDS

What are vibration and pressure pulsation limits, what is good, what is acceptable, etc., are frequent questions with high energy input pumps in power stations. For all cases a search for amplitudes continuously goes on, but it is most important to find the associated FREQUENCIES. We are usually looking for:

- Dynamic balancing minimums
- Vibration of the
  - Rotor in the
    - Radial direction
    - Axial direction
  - Structure
  - Piping
- Pressure pulsation in the
  - Piping
  - Pump internals
- Safe operating ranges, such as
  - Speeds (Maximum and sometimes also minimum)
  - Flows, expecially minimum (recirculation)
  - Pressures, temperatures, etc.

DYNAMIC BALANCING

The U.S. Navy specifications for allowable dynamic unbalance quantities are clear, precise and adequate for any pump applications.

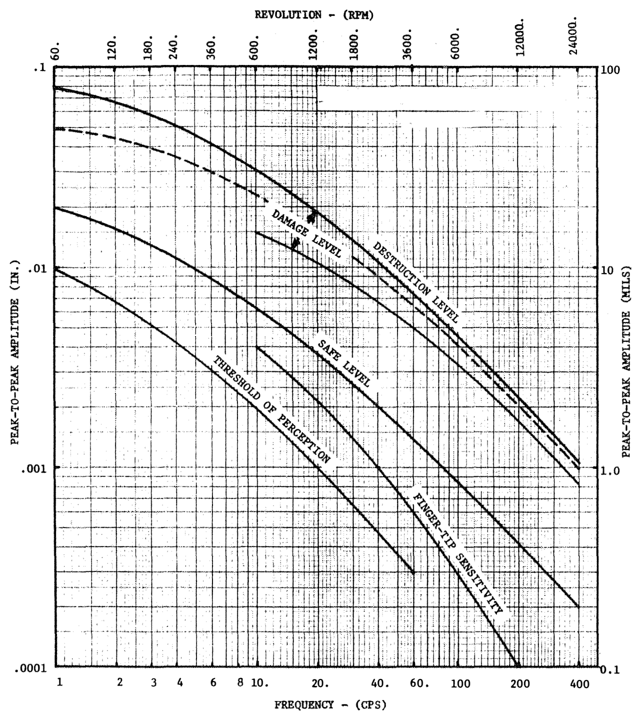


Figure 12. Allowable Rotor Vibration Levels for Centrifugal Machinery, Measured Relative to the Bearing Cap. (Experimental Standards)

VIBRATION

Rotor synchronous vibration amplitude levels are shown in Figure 12 as measured on the shaft relative to the bearing housing. The amplitudes refer to only synchronous speed frequencies (once per rotor revolution). Figure 9 shows frequencies and some simplified explanations for their causes and cures.

Axial vibration level is an excellent indication of pump design quality for single stage double suction pumps, such as booster, and nuclear feed pumps. It can be used to detect hydraulic instability, and can be used to find safe minimum flows (Recirculation flows). There are no established standards among manufacturers or users. Figure 13 is an example, while Figure 14 shows a method how axial vibration data was used to set minimum flow quantities in several nuclear units.

How to measure amplitudes and frequencies is even more important. Figures 15 and 16 (from Reference 4) show how misleading incorrectly measured data can be, while Figure 17 shows how a correctly instrumented machine was saved from destruction.

PRESSURE PULSATION

There are no pressure pulsation standards available at present. Often a 3% maximum of the pump produced head (measured on the discharge side) is quoted, which is satisfactory in some cases. Recently analyzed cases revealed that vendors almost always quote amplitudes at the "Vane passing" frequency (number of impeller vanes "Z" times rotational speed "N," i.e. $Z \times N$). Figure 3 shows insignificant amplitudes at $Z \times N$, but the same data recorded on magnetic tape and played back at a lower speed shows that the amplitude at vane-passing frequency is indeed significant. Figure 20 shows data collected on nuclear feed pumps in installations with pump and system dif-

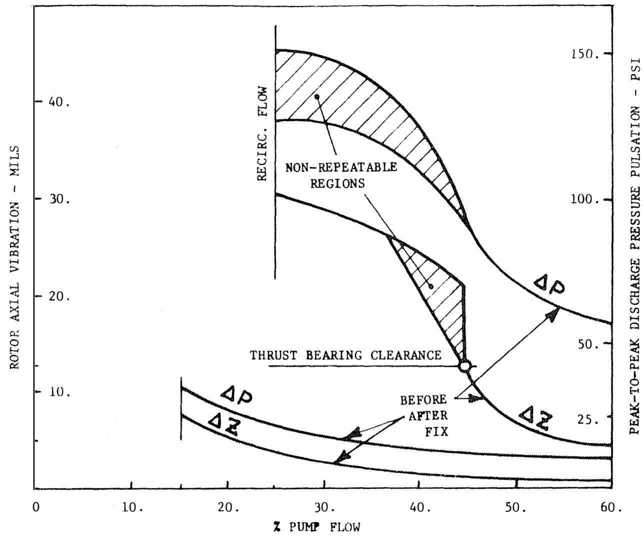


Figure 13. Peak-To-Peak Pressure Pulsation (ΔP) Measured in the Discharge Nozzle, and Rotor Axial Vibration (ΔZ) of a High Speed Reactor Feed Pump With Two Different Impeller Discharge Configurations. Pump Efficiency At 100 Percent Flow Was the Same in Both Cases (~90 %).

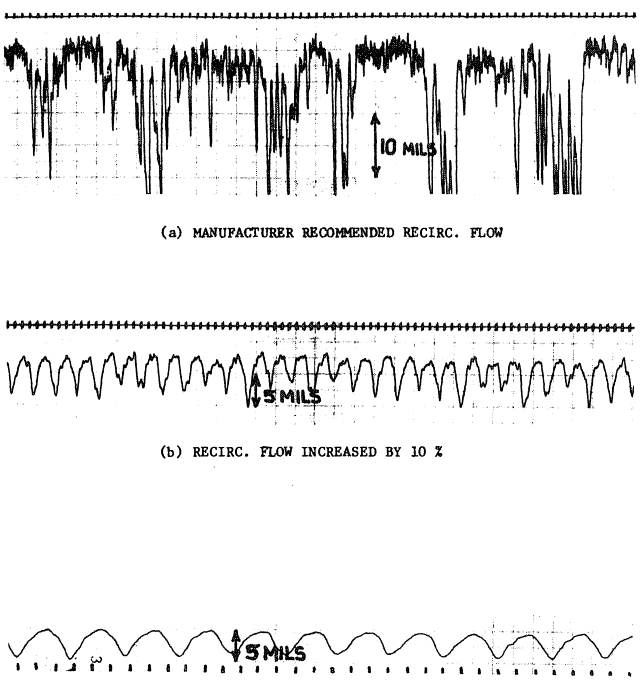


Figure 14. Shaft Axial Vibration of a High Speed, Single Stage, Double Suction Nuclear Feed Pump at Minimum Flow Operation. Directly applicable to any single stage double suction centrifugal pump, such as condensate, booster, or any chemical process pump.

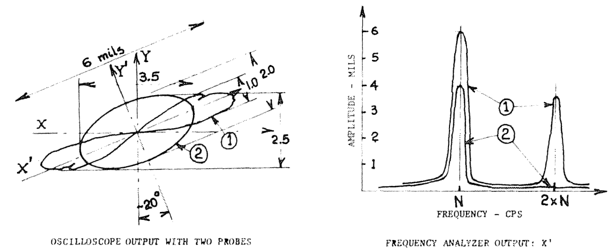


Figure 15. Orbit Shapes at the Turbine Inboard Bearing. Boiler feed pump and turbine drive coupled. Case (1): Before hot alignment with large bearing clearance. Case (2): After hot alignment, bearing clearance reduced by 4 mils on the diameter. Both at the turbine rotor first critical speed. Notice that one vertical probe would not show any change in bearing behavior. One probe 20° off vertical position would show the opposite effect. Two probes perpendicular to each other, oscilloscope, and frequency analyzer give clear picture.

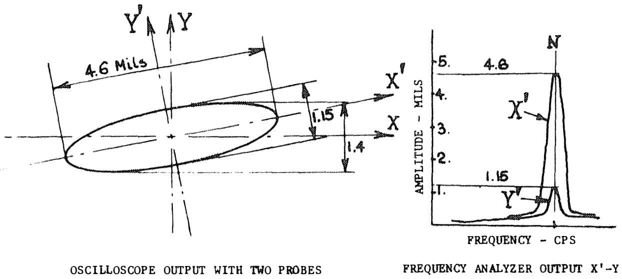
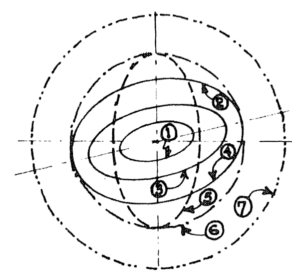


Figure 16. Boiler Feed Pump Uncoupled From the Turbine Drive. Orbit shape at the turbine inboard bearing at the first critical speed. The vibration reading in the control room showed 1.4 mils, while the true amplitude was 4.6 mils. Measuring vibration levels of a high speed rotating machine in a single direction can be not only misleading, but also dangerous.



NO.	RPM	CRITICAL speed:	MAXIMUM orbit: mils
1	2,700		1.45
2	3,100	FIRST	4
3	4,000		3
4	5,000		3.8
5	5,050		4
6	5,100		4
7	3,100	FIRST	~7

Figure 17. Shaft Orbits Measured at the Turbine Inboard Bearing at Various Speeds on June 5, 1975. Drastic increase of amplitudes concluded the testing. Vibration became violent at the first critical speed during coast-down.

ficulties. A recommended "Safe limit line" is also shown, measured in the pump discharge. This limit line is continuously subject to revision as more data is collected. In Figure 5 we make an attempt to explain frequencies and magnitudes of hydraulically induced dynamic forces within the pump.

Figure 2 shows a frequency plot with the aid of an RTA (Real Time Analyzer) of a nuclear reactor feed pump. The measurements were taken at the pump discharge. The amplitude at the frequency of N/4 (N=rotational speed) is very strong. Also a high amplitude was experienced at about one CPS, which can influence the feed water piping, control valve, control system, reactor water level, and the reactor control system as shown in Figure 18. The pulsation with the low frequency can be a dominating response. Its influence on the control system, and the steam turbine itself can be so strong that it may force the turbine into a high load oscillation with heavy vibration of the whole steam cycle. In one particular case when control was put on "Manual," operation became normal, proving that it was caused by the control system, which be-

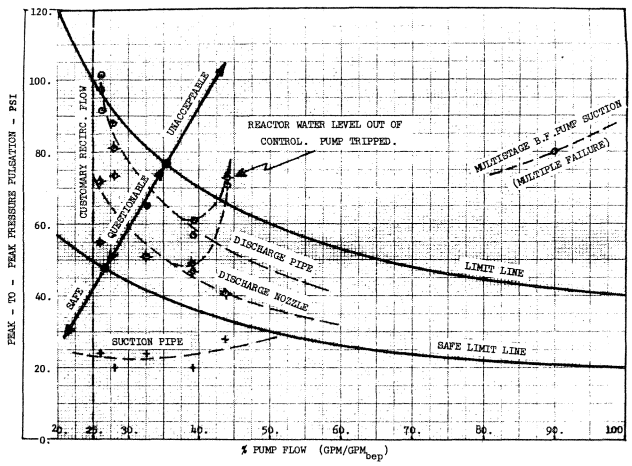


Figure 18; Typical Pressure Pulsation Magnitudes in Large Nuclear Reactor (and Steam Generator) Feed Pumps. Average TDH (head) generated by the pump is 2,500 ft.

came influenced by the pressure pulsation generated by the reactor feed pump proven by measured data (see Figure 2 to 5, and 18).

Figure 3 is a typical example. Pressure pulsation measurements in the reactor feed pump discharge pipe indicate "only" low frequency (N/4) pulsations of high amplitudes, giving a clean bill of health to the pumps, putting the blame on the piping system. Figure 4, however, shows that the forcing function, responsible for the lower frequency amplitudes, indeed, is the blade passing frequency.

To answer additional questions, such as what are the frequencies and the amplitudes inside the different parts of the feed pumps, and, because in one particular case the above-mentioned frequency story resulted in too much outage of several large nuclear stations, such measurements are now taken. All of them show moderate amplitudes at blade passing frequency, however, analyses, similar to that shown in Figure 4, showed that balde passing frequency was the real cause of the system problem. Redesign of the pump impeller with different number of blades, but more important, with very carefully selected impeller design parameters, finally eliminated the disputed problem.

It was observed many times by the author that the amplitudes at blade passing frequencies (Z × N) are steady in magnitudes (forcing function), while at lower frequencies they form, disappear and reform again (excited system responses). Figure 5 is a simplified summary of frequently observed hydraulically induced frequency responses measured on the pump and on the feed water piping. Subsynchronous forces can be detrimental, resulting in similar appearance to "Oil-Whip," often misleading the investigator. One should distinguish between oil-whip, rubbing-induced whirl, and hydraulically induced whirl. Often these three appear alike, and can guide toward false conclusions. The frequency of oil-whip is independent of pump flow, while hydraulic instability is a function of both, speed and flow.

How can we now represent the above mentioned static and dynamic forces when performing rotor-dynamics calculations. The static forces can be represented similarly to bearings, seals, and wear-rings, namely with spring and damping coefficients. Synchronous forces can be represented as dynamic unbalance. Other than synchronous (1 × Rev) and static forces can be represented only when using non-synchronous, or non-linear computer programs.

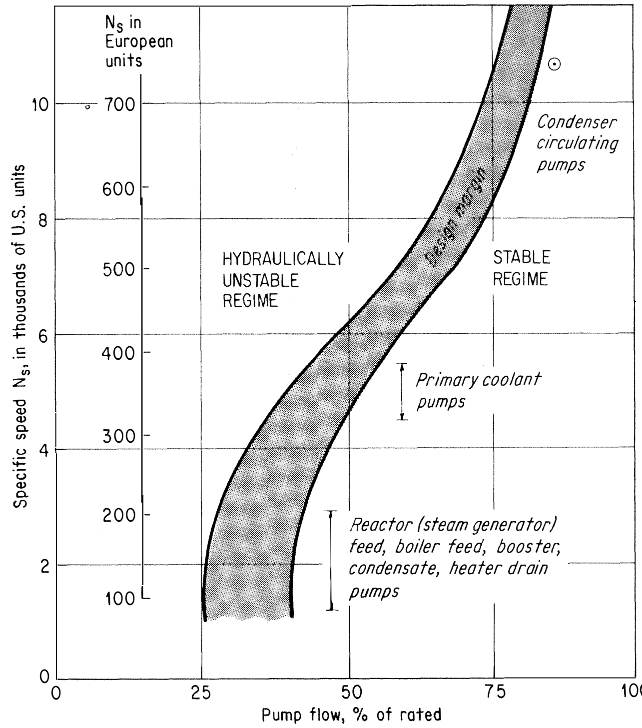


Figure 19. Anticipated Useful Operating Ranges for Pumps Used in Large Nuclear and Fossil Units

SAFE OPERATING RANGES

Here we concentrate on the subject of pump "Minimum" flow, often referred to as "Recirculation," or simply "Recirc" flow. Figure 19 shows minimum flow limits for various pump types and applications. Because of keen competition in the pump market, vendor prices, pump efficiencies, etc. all fall within a narrow range. Consequently, the only attractive — BUT ARTIFICAL — design highlight a marketing man can exploit is minimum flow. For example, instead of guaranteeing 25% recirculation flow, a vendor might promise only 5 to 10%. Then the recirculation line and its control valve can be reduced in size from, say 8 inches to 4 inches, and he can offer a saving in the overall price of the feed water system piping. This seems attractive initially, especially to the architect engineer, but not when you consider the extensive pump damage that resulted from low recirculation flow. The design margin band in Figure 19 represents a regime in which the experience and assumptions of the pump designer come into play. Safe minimum flow for a large reactor feed pump is "NOT LESS" than 25% of design flow, but may be as high as 45% which is not acceptable and the pump impeller design is to be examined and improved: see Figure 13 as an example.

Other quantities such as maximum flow velocities, flows and speeds are well discussed in Reference 1, hence not discussed here. Other standards and measuring guides are outlined in Reference 4. There are many other important standards for centrifugal pumps, one of them is the radial gap between impeller O.D. and diffuser/volute tongue inside diameter. The closer the gap is the higher the pump efficiency

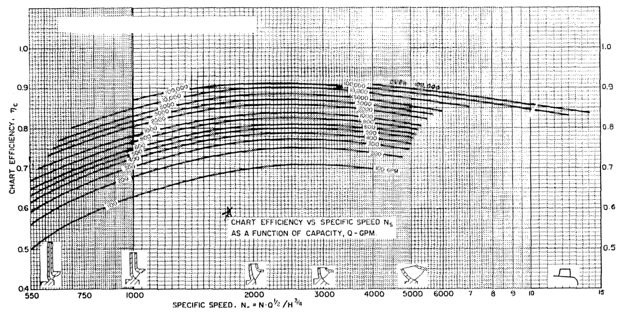


Figure 20. Chart Efficiency vs Specific Speed N_s as a Function of Capacity, Q-GPM.

is, however, beyond a certain limit the pressure pulsation, caused by blade passing phenomenon, becomes strong enough to destroy pump components, but more important, the whole pump itself. For diffuser pump the gap must not be less than 3% of the impeller O. D., for volute type pumps the standard is not to be less than 5%.

WHAT CAN FACTORY TEST TELL ABOUT FUTURE PROBLEMS

Figure 20 is a guide-line for what the expected pump efficiency levels may be. Figure 1 shows an example of a troublesome pump head curve shape. If the head curve is droopy or flat toward decreasing flows, or if it has a "KINK" in it, the pump should not be accepted, because, pump, piping, control valve, control system malfunctioning may follow in the future. Analyze ALL pressure pulsation ranges, do not accept results just at the blade passing frequency. Always ask for proper instrumentation at the bearings (see Figures 15 and 16) and for results at a wide frequency range, not only at operating speed (see Figure 11). If pump has a balance-disk device, always ask for the disk leak-off flow in the complete operating range of the pump, as shown in Figure 21.

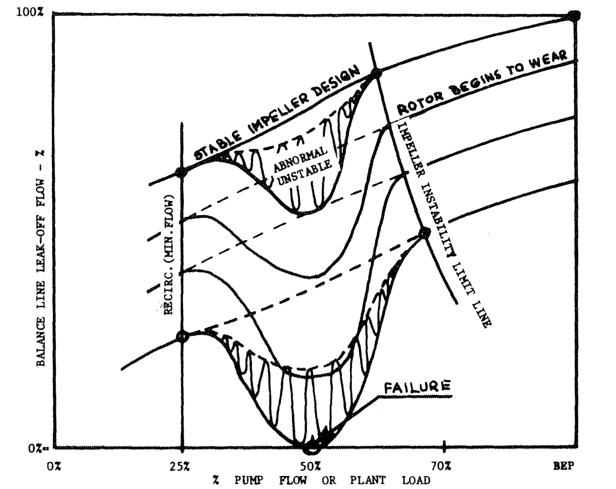


Figure 21. How Boiler Feed Pump Failure Was Intercepted and Degradation of the Pump Was Followed with Simple Instrumentation at an 800 MW Unit. If pump impeller head characteristics was stable most likely none of the failures would have occurred.

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