

CHANGES IN HYDRAULIC COMPONENT GEOMETRIES GREATLY INCREASED POWER PLANT AVAILABILITY AND REDUCED MAINTENANCE COST: CASE HISTORIES

by

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ABSTRACT

A systematic investigation was started within the Texas Utilities Generating Company Power station feedwater pump systems due to unsatisfactory unit availability, frequent equipment breakdown, high maintenance costs, and high replacement parts costs. This action was greatly encouraged by successful corrective actions taken by other utilities [1,2,3], results of failure surveys across the U.S.A. [4,5], and systematic long range research and development, field testing and troubleshooting by the authors, and close cooperation and exchange of ideas and results with at least one oil company [6].

A total of 36 pumps was modified in the Texas Utilities system as described in this paper. This included 16 boiler feed pumps (BFP), 7 high pressure (HP) booster pumps, 6 low pressure (LP) booster pumps, and 7 other pump applications. The operating behavior of all pumps improved exactly as expected, with the exception of one high pressure booster pump, which is described as a special case, with the emphasis on the importance of re-assembly and maintenance procedures. The power plants and the type of pumps involved in this investigation and corrective design changes are reflected in Table 1. Basically, we have to distinguish between multistage boiler feed pumps (or process pumps) and single stage double suction booster pumps (or condensate, ash-sludge, circulating, cooling water, etc.), because the same hydraulically induced instability problem manifests itself in a very different way through the type of vibrations we experience and measure. Single stage pumps usually show a heavy axial vibration mode of the rotor, while multistage pumps show radial response, although the origin of the excitation is the same.

INTRODUCTION

A pump impeller stage, whether diffuser or volute type, single suction or double suction, vertical or horizontal, is always designed for one point which is called the Best Efficiency Point (BEP). Depending on Specific Speed (N_s) and Suction Specific Speed (S_s), the useful operating range of the stage varies

Table 1. Power Generating Units in the Texas Utilities System Where Design Changes Were Introduced in the Hydraulic Components.

Power Station	Plant Size MW	BFP		Drive HP	No. of Booster Rotors		Other Pumps
		No. of BFP	Rotors		HP	LP	
Tradinghouse 1	550	2	3	14,100	2	2	2
Tradinghouse 2	800	1	2	43,750	1	2	2
De Cordova	800	1	1	43,750	1		1
Valley 2	550	2	3	13,500	2	2	
Stryker 2	500	1	2	18,000			
Lake Hubbard	500	1	2	17,000	1		
Mt. Creek 8*	550	2	2	14,000			2
Martin Lake 1, 2 and 3*	750	6	10	22,500			
Total:		16	(25)		7	6	7
Total No. of Pumps:					40		

*Not all rotors altered, in the process of gradually changing them as going through routine maintenance.

(Figure 1). Hydraulic instability sets in at a certain percentage of BEP flow as pump delivery is decreased. Recirculation sets in at impeller eye, at impeller exit, but most importantly, behind impeller sideplates. Instability behind the impeller sideplate can be controlled by gap "A" [1]. This phenomenon shows in various ways on multistage boiler feed pumps, but manifests itself very distinctly as rotor axial shuttling for single-stage double suction type pumps.

The radial gap between the impeller vane exit and the diffuser or volute inlet is emphasized, concluding that it not only controls pump noise level, but also is responsible for minor and

major failures, contributing more than expected to maintenance costs and plant availability.

Material selection and its hardness is also discussed as a major contributor to failures, mainly in large multistage boiler feed pumps.

Energy input selection per pump impeller increased ten times between 1965 and 1970, as power plant sizes increased as shown in Figure 2. The severity of failures also increased with the same ratio, but more importantly, the cost per outage for a large unit increased to the point that a one day outage might equal or exceed the cost of the boiler feed pump itself. Data from similar or duplicate units was used in several examples, with the cooperation of other utilities who experienced more severe problems.

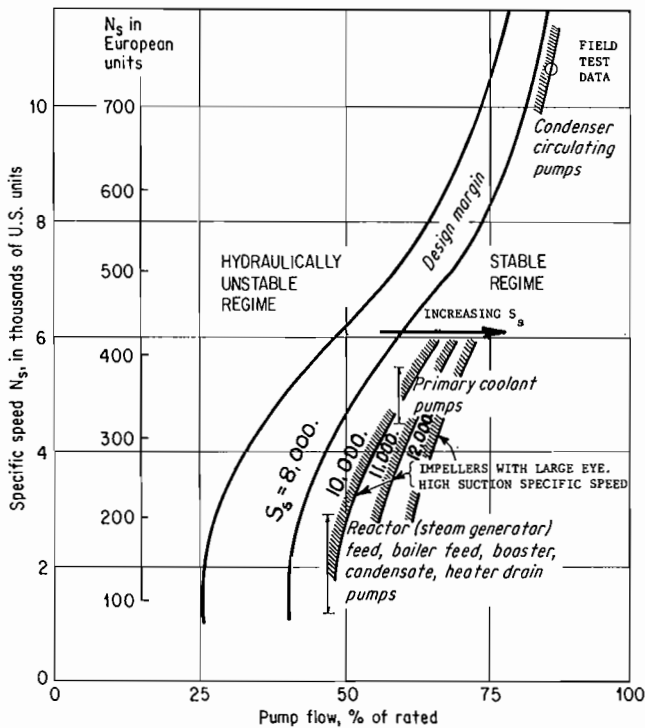


Figure 1. Anticipated Useful Operating Ranges of Centrifugal Pumps Used in Large Nuclear and Fossil Power Generating Units as a Function of Specific Speed (Impeller Shape), and Suction Specific Speed (NPSH).

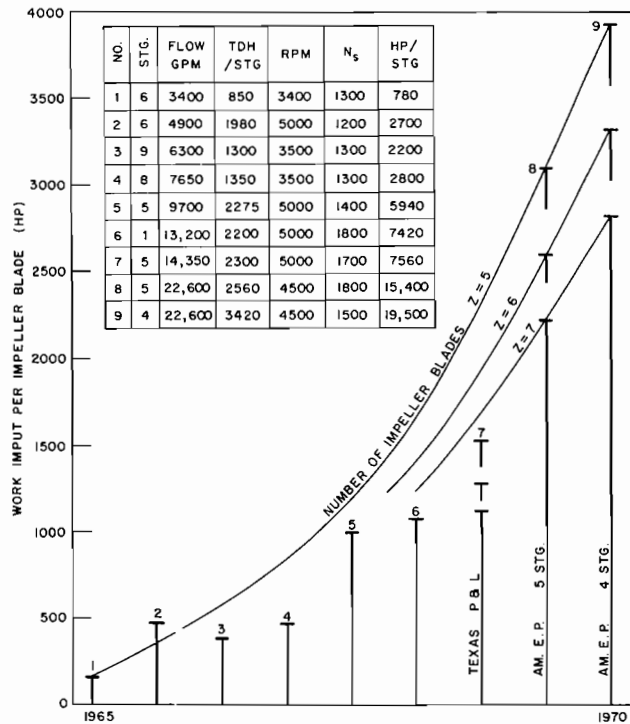


Figure 2. Increase of Horse Power Per Boiler Feed Pump Impeller Blade in 5 Years.

FEED PUMP PROBLEM AREAS AND FAILURE TYPES IN TEXAS

The following problem areas and failure modes were recorded in the Texas Utilities System during the last decade. Many of these causes were already mentioned in literature [10].

- Feedwater (FW) flow fluctuation during partial load operation.
- FW piping vibration at reduced loads (low frequency vibration).
- FW control system instability.
- Boiler Feed Pump (BFP) (multistage pumps) instability at partial loads.
- FW Booster (and other single stage double suction applications) instability, mainly showing as axial shuttling of the rotor.
- Shaft breakage at distinct (repeat) locations.
- Impeller disintegration (Figures 3 and 4).
- Impeller side plate breakage.
- Impeller vane breakage at the exit.
- Diffuser or volute inlet vane breakage.
- Internal bolt breakages.
- Rotor seizure (incorrect material, problem solved).
- High vibration levels at vane passing frequency, or at its multiples.
- High noise level (usually at vane passing frequency).
- Frequent coupling damage.
- Frequent seal failures.
- Head gasket leaking of one BFP type.
- Seal housing gaskets leaks on one BFP time.
- Seal injection line and lube oil piping fatiguing.
- Frequent balancing devise failure.

Apparently, all the above problems were eliminated, or at least greatly reduced, with the incorporation of the geometry changes described in this paper. Improved plant availability and significant reduction in maintenance efforts is the end result of this systematic pump-by-pump corrective effort.

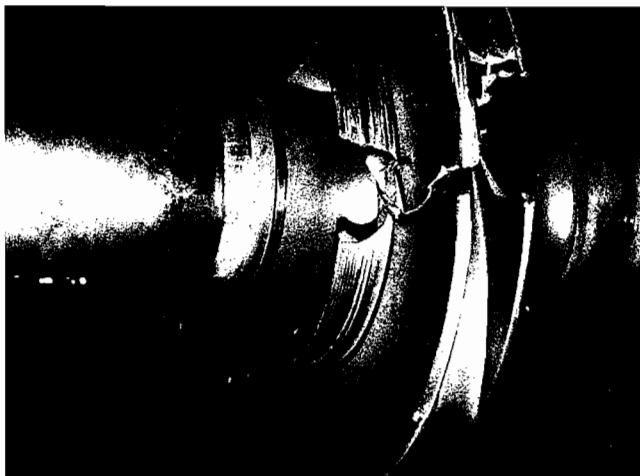


Figure 3. Disintegrated Impeller of a Multi-stage Boiler Feed Pump. Gap "A" Was Only 30 MILS Which Resulted in Contact between the Impeller and Diffuser Sideplates Causing Major Damage of the Entire Rotor.

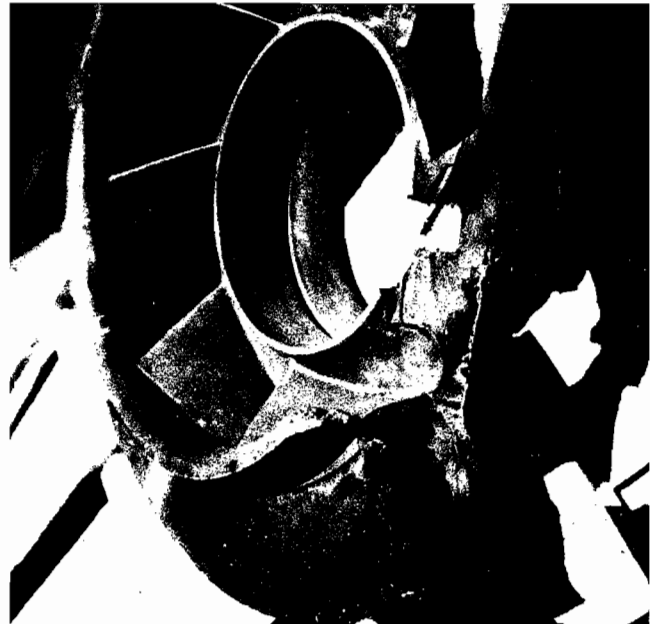


Figure 4. Disintegrated Boiler Feed Pump Impeller Due to Porosity of the Casting. Result: Major Outage of an 800 MW Unit during the Summer of 1982.

MULTISTAGE PUMPS

Two 800 MW Units

One of the biggest operational problems of the Texas Utilities System was the unit instability phenomenon with the two 800 MW units (Tradinghouse Creek No. 2 and DeCordova No. 1). Between the load range of 450 MW and 500 MW, the units were unstable and exhibited most of the problems listed in the preceding paragraph. The original head-capacity curve is shown in Figure 5. Controlling these units in this range was very difficult and, therefore, the units either ran above 500 MW or

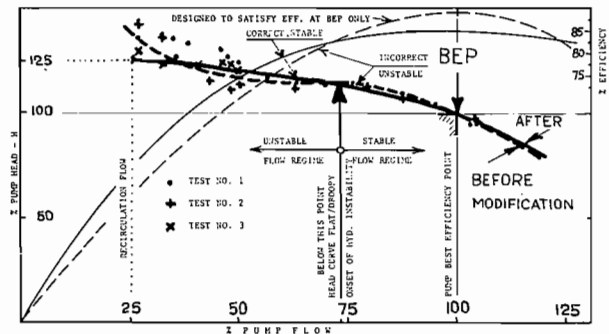


Figure 5. Head Capacity Curve of a Five Stage High Speed Boiler Feed Pump Driven by a 45000 HP Steam Turbine (Single 100 Percent Pump for an 800 MW Unit). The Dotted Line Shows the Original Unstable Head Curve That Caused Severe Control System Instability Below 500 MW (75 Percent of BEP) Unit Load. The Solid Line Shows the H-Curve after Modifying the Impeller Exit to Diffuser Inlet Geometry. The Control System Instability Was Completely Eliminated by This Simple Change of Stage Geometry. (Texas Power & Light Co., Tradinghouse No. 2 (800 MW) and DeCordova No. 1 (800 MW), 1982).

below 450 MW. Both Tradinghouse No. 2 and DeCordova No. 1 were experiencing fluctuation of over 500,000 lb/hr feedwater flow in the 450 MW to 500 MW region and could not load control. This was unacceptable to system operations, because of system demands. Therefore, the decision was made to make any changes that would stabilize the system. The only changes made initially were to increase vane-to-vane clearance between impeller O.D. and diffuser I.D., from the original 2 percent radial gap to 12 percent on the suction impeller and from 2 percent to 7 percent gap on the radial impeller. Also gap "A" (impellers sideplate to diffuser sideplate radial gap) was decreased from the original extreme (approximately 1/4 in) dimension to 100 mils. The results were better than expected. Unit instability control problems, BFP turbine drive control instability, suction and discharge piping vibration, and feedwater flow oscillations between 450 MW all disappeared, so that both units now can be used with the highest confidence for system load control and frequency response between 250 MW AND 800 MW. Rapid load changes of 35 MW/minute is one of the most important requirements of these two units. This is now being done on a daily basis.

The new stable head-capacity curve is shown in Figure 5 as a solid line. The dotted line represents the original head curve. The noise level is down to the point that maintenance personnel can work comfortably in the pump area. However, the most important improvement, of course, is that the feed pumps have required essentially no maintenance during the last two years, and pump availability has been 100 percent.

HIGH NOISE LEVELS

High noise levels generated by many of the boiler feed pumps, coupled with high maintenance items listed earlier, were some of the most annoying things for maintenance personnel while working in feed pump areas in many power plants. Upon examining the radial gap (Gap "B") in the feed pumps listed in Table 1, we found gaps as small as 1/2 percent. Our objective was to open up the gap to 6 percent, if possible. A typical case of diffuser tips being cut back to incorporate larger gaps is shown in Figure 6. Diffuser geometry, channel length, vane inlet angle and other parameters had to be examined carefully in every case. The vanes were cut back before the appropriate radius was put in on the leading edge of diffuser vanes, as is shown in Figure 6. Major reduction in noise level was one of the side effects resulting from increasing gap "B" by recessing diffuser tips.



Figure 6. Diffuser Tips of a Multi-Stage Boiler Feed Pump Cut Back to Enlarge Gap "B" from 1 Percent to 6 Percent (Photo Shows Vane Inlet Tips Before Radius Put on).

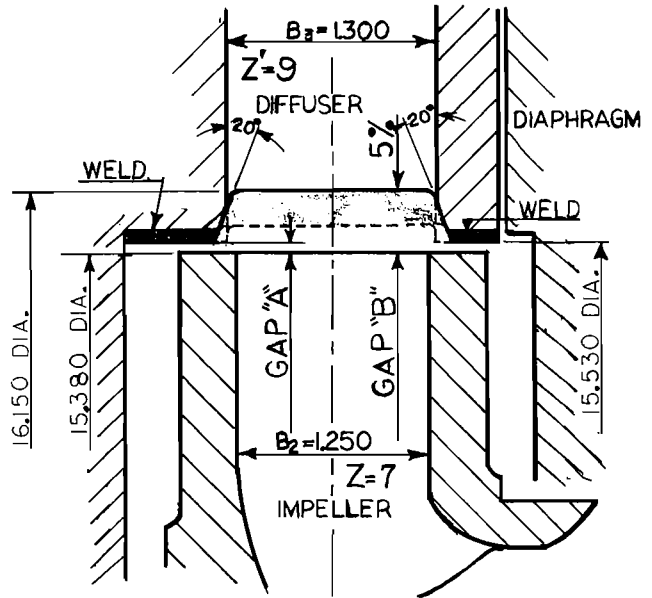


Figure 7. Size 14 In Boiler Feed Pump, First Stage Impeller and Diffuser Modifications. Gap "A" Closed Down to 0.075 in, Gap "B" Increased from 1 Percent to 5 Percent. Diffuser to Impeller Width (b_3/b_2) Ratio Is Almost One, Therefore the 20° Chamfer at the Diffuser Inlet Is Needed for Better Flow Guidance.

SPECIAL CASE

In one particular case, radial gap "B" was increased only to 4 percent, because the diffuser inlet angle was so low (practically 0°) that more than 2 in of the diffuser tips would have had to be removed, in order to gain normal impeller to diffuser vane tip clearance (6 percent). This would have reduced the diffuser channel length to the point that pump efficiency would have deteriorated. Our goal is not to reduce the diffuser channel length (1) to diffuser throat width (d_3) ratio to less than 2.5 from

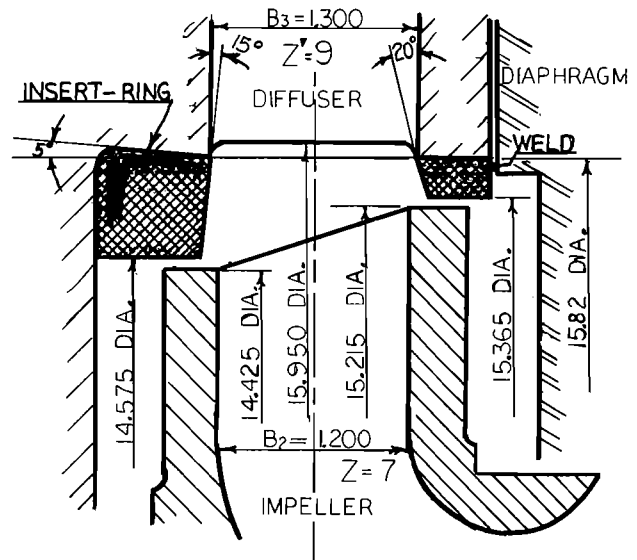


Figure 8. Size 14 In Boiler Feed Pump 2nd to 5th Stages, As Modified. Due to the Irregular Shape of the Impellers, an Insert Ring Was Installed on the Hub Side, While the Shroud Side Was Welded up to Bring Gap "A" to Normal Dimensions. Both Sides Had to Be Slanted to Give Proper Guidance to the Flow into the Diffuser Inlet.

the ideal $1/d_3=4$ ratio. Ideal design practice is $1/d_3=4$, with straight walls and with a diffusion angle of 12° maximum. Noise level remained slightly higher with this pump than desired and is still clearly audible from outside the building, but maintenance activities became normal and fatiguing of minor components ceased to recur. Additionally, this amount of gap "B" eliminated the internal bolt breakages.

SPECIAL GEOMETRY

A special geometry of impeller to diffuser width ratio (b_3/b_2) to be nearly one is shown in Figure 7. If the radial gap "A" is large, the liquid coming from the impeller impacts on the diffuser side plate corner, resulting in a continuously unexplained high (flow induced) vibration level and lower than

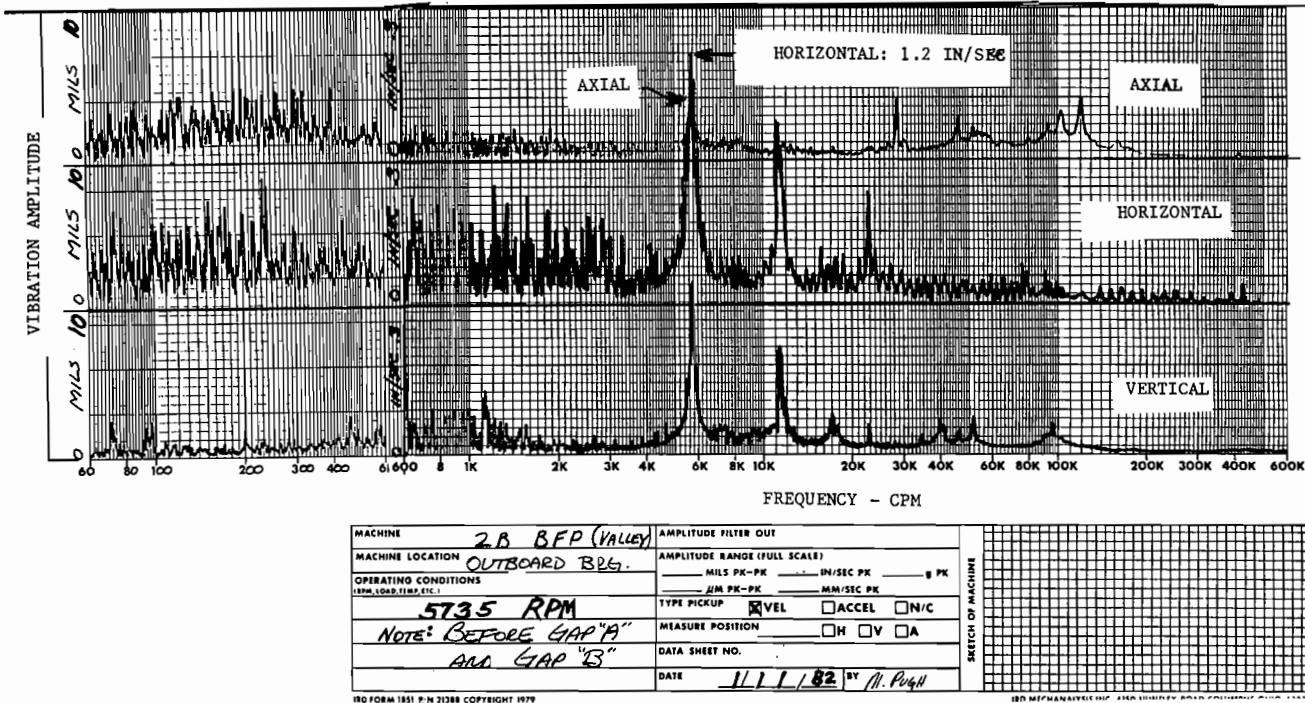


Figure 9. Actual Vibrational Analysis of Angled Impeller Multi-Stage Boiler Feed Pump Before Gap "A" and Gap "B".

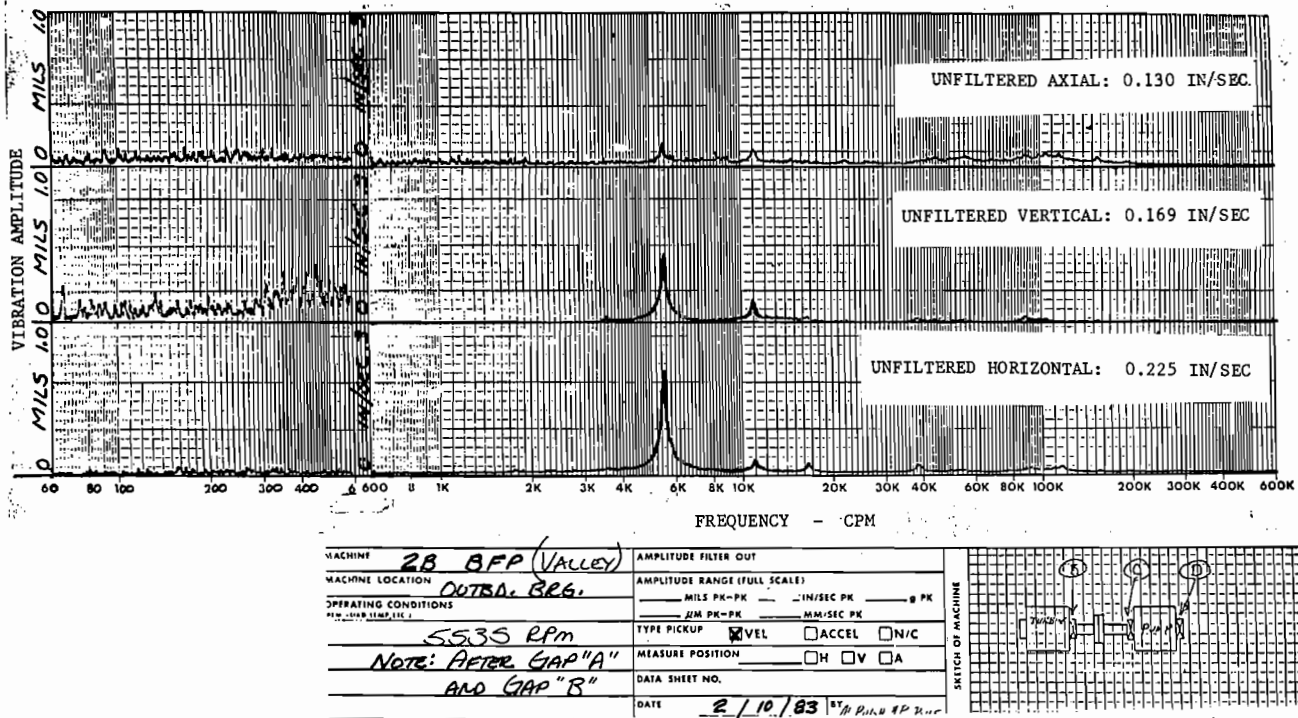


Figure 10. Actual Vibrational Analysis of angled Impeller Multi-Stage Boiler Feed Pump After Gap "A" and Gap "B" Modifications.

expected efficiency. Also, the rotor becomes very sensitive to axial alignment. The corrective actions (shown in Figure 7) required cutting back the diffuser inlet tips, but more importantly, the diffuser side plates were chamfered at a 20° angle to give proper guidance to the water into the diffuser channels, even if slight axial misalignment of the rotor is present.

Several normal stage impellers were cut back by the original equipment manufacturer (OEM) during performance testing, as shown in Figure 8, to control the headrise of the pump. The impeller O.D. was cut at an angle such that the impeller diameter on the hub side was almost an inch smaller than on the shroud (eye) side. Also, the b_3/b_2 ratio was nearly one. This poor combination of geometry gave rise to unexplained vibration problems for many years, as shown in Figure 9. Gap "A" was reduced by the writers by welding up on the shroud side of the diffuser. Welding up on the hub side would have risked thermal deformation. An insert ring of the same material was welded to the hub side to make gap "A".

After the incorporation of geometry changes, vibration behavior of this boiler feed pump became normal in the whole speed and flow operating range. The vibration spectrum analysis after gap "A" was installed, as is shown in Figure 10.

ANOTHER SPECIAL CASE

When diffuser channels are already too short, or the diffuser vane inlet angle is too low, gap "B" cannot open up to the recommended 6 percent by just cutting the diffuser tips back. Also, when closing down gap "A" to the desired dimension, the unfortunate chamfering of impeller side plates at the outside diameter (O.D.) makes the overlap inadequate. The diffuser throat area had to be welded to obtain a 15° angle, as

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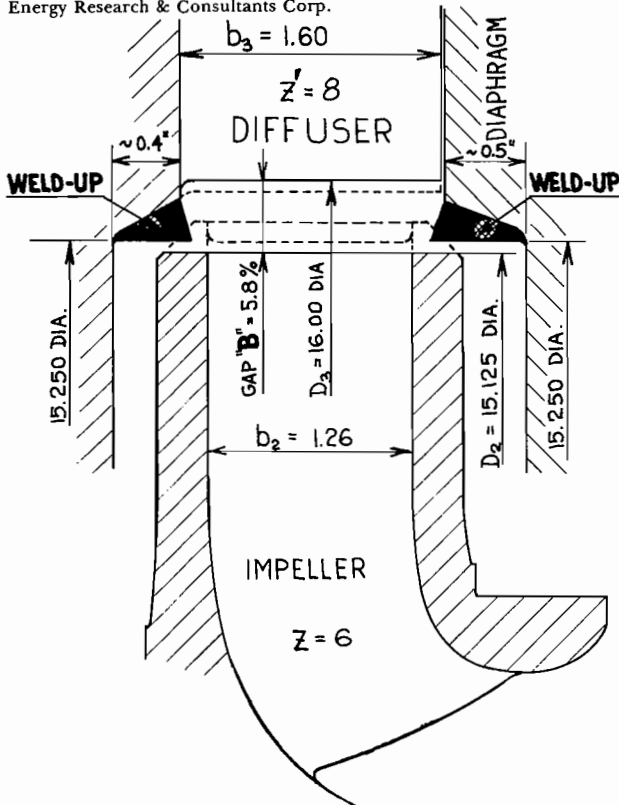


Figure 11. Both Impeller Exit and Diffuser Inlet Were Cut Back to Increase Gap "B", and Especially to Regain the Necessary Overlap for Gap "A". Weld Build-Up with a Chamfer Angle Was Necessary to Make Gap "A" Effective.

shown in Figure 11. A case where reduction of the impeller O.D. allowed enough surface on the impeller sideplates to make gap "A" effective without bringing the diffuser side plate in with an angle is shown in Figure 12. In order to compensate for the head loss that resulted from cutting the impeller O.D., the impeller vanes were underfiled as shown in Figure 13. Depending on the impeller vane exit angle, the number of impeller vanes and specific speed of the impeller (N_s), we can gain 4 to 8 percent flow at BEP on boiler feed pumps. On very large circulating pumps with high N_s (4-5000), as much as 16 percent flow was gained. This was needed to improve condenser capacity to increase plant activity. Underfiling the vanes at the exit also usually results in some increase in pump efficiency. Not less

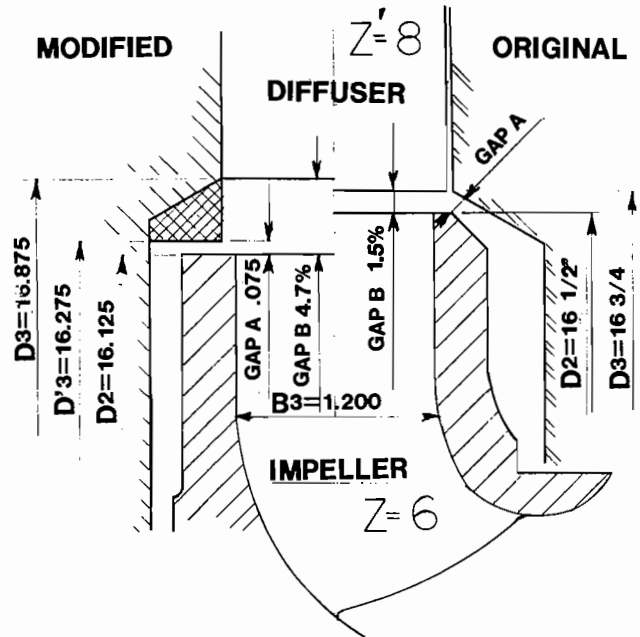


Figure 12. Impeller and Diffuser Modifications to Open Up Radial Gap "B" from 1.5 Percent to Almost 5 Percent, and to Close Down Radial Gap "A" to 75 MILS to Eliminate High Vibration and to Control System Instability at Part Load Operation.

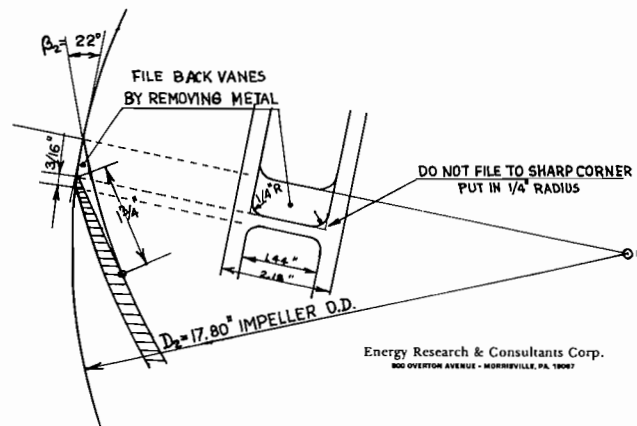


Figure 13. Standard Underfiling of Boiler Feed Pump Impeller Vanes at the Exit to Gain Head and Flow at BEP. The Amount of Gain Is a Function of Vane Exit Angle, Number of Vanes, and Specific Speed (N_s). Head Will Not Change at Shut-Off Flow.

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than 1/4 inch radius is recommended at the corner between the vanes and side plates to avoid crack initiation of the impeller structure, which is a frequent failure mode for high energy input pump impellers.

SINGLE STAGE DOUBLE SUCTION PUMPS

Single stage double suction pumps are applied widely, both in the power and oil industries. When there is a design discrepancy, the pumps have a unique way of showing the problems. It became quite obvious [1,4] that severe axial vibration of a single stage pump results from hydraulic instability in flow conditions below BEP. Alterations made to nine feedwater booster pumps are presented in Figure 14 [1]. Closing down gap "A" to 50 mils completely eliminated pump and feedwater piping vibration at reduced loads. Pumps were equipped with mechanical seals and seal failure was a routine occurrence at that power station. Since the design change, mechanical seal failures have stopped. During inspection of these pumps, insert rings were installed to fill up the large radial gap. A case where a dove-tail ring was used to close down gap "A" to 50 mils is shown in Figure 15.

A case where the gap was welded up and machined, with gap "A" = 50 mils, is displayed in Figure 16. The radius on the impeller sideplates at the O.D. made the modified geometry more complicated. Coming straight down with diffuser side plates did not provide enough overlap between impeller and

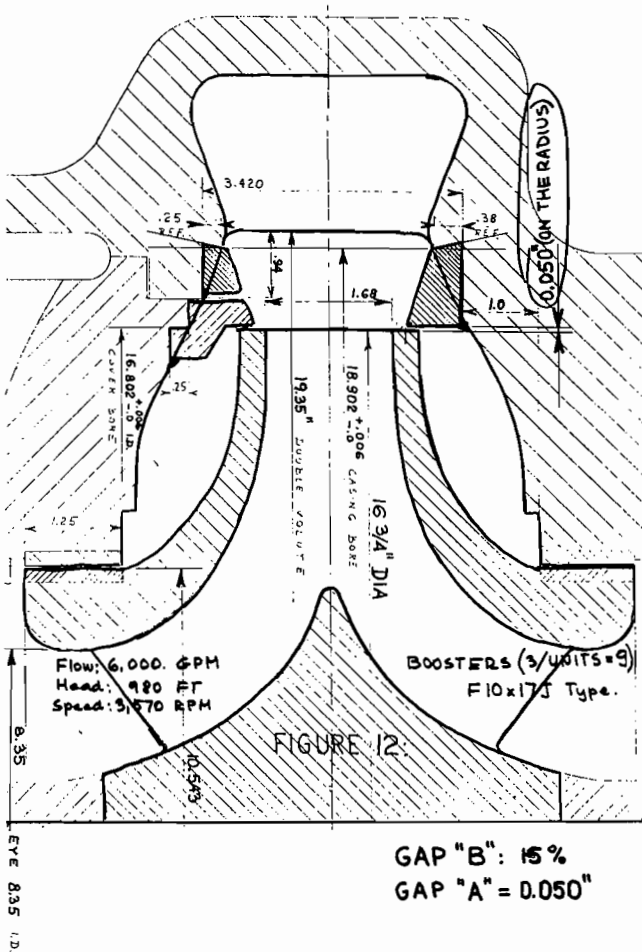


Figure 14. Correcting Gap "A" in a High Speed Booster Pump. 4000 GPM Recirc Flow Was Necessary Prior to the Modifications to Avoid Excess Axial Vibration of the Rotor.

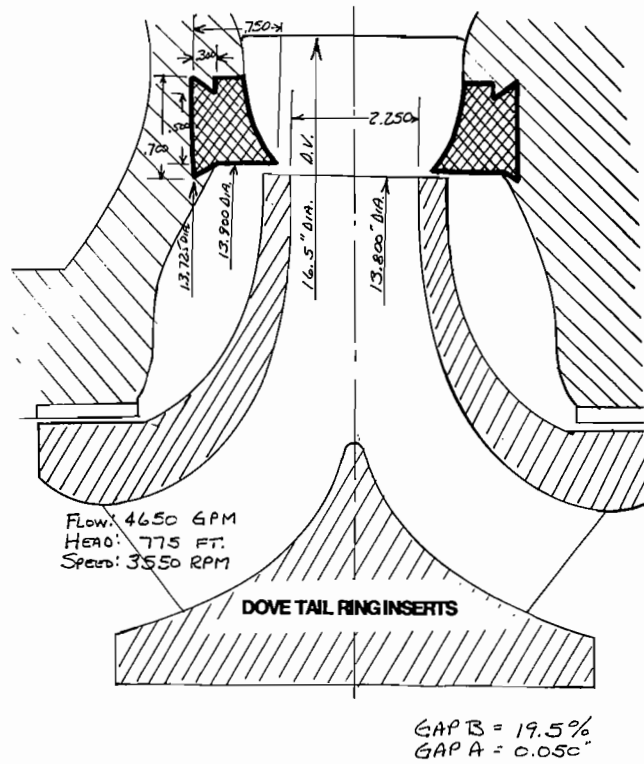


Figure 15. Dove-Tail Ring Inserts Completely Eliminated High Vibration Caused Frequent Failures at Tradinghouse Number 2, 10 x 14 Split Case Booster Pump.

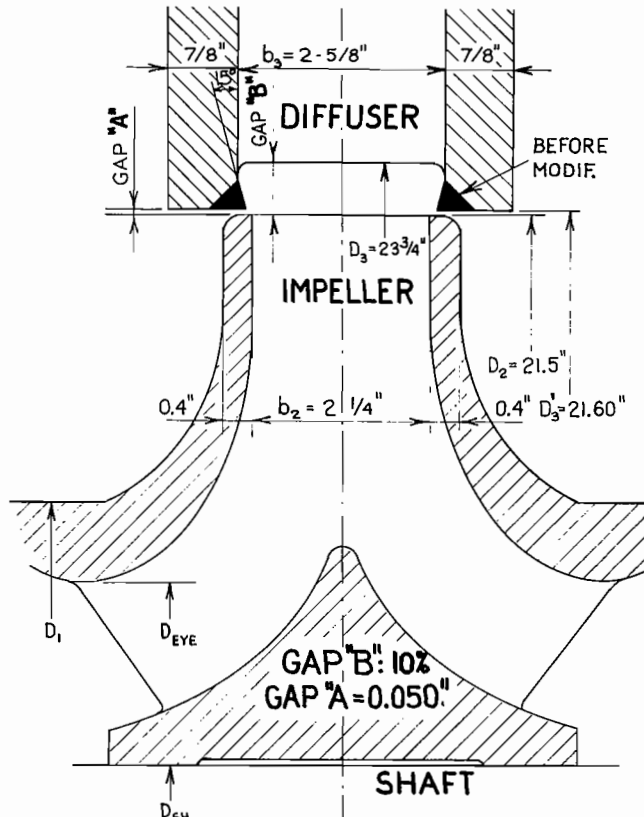


Figure 16. Weld Build-Up and 15° Chamfering Was Necessary to Eliminate Low Flow Instability, Caused by High Speed Booster.

diffuser side plates, and the filter effect of gap "A" proved to be ineffective. The side plate had to be brought in with a 15° angle, to gain proper overlap. Initial 25 percent minimum flow turned out to be inadequate during plant start-up in 1972. Heavy vibration brought upon the whole feedwater system made it essential to add a second recirculation line, bringing the total necessary minimum flow to over 50 percent of BEP. After the recommended modifications were incorporated, the added minimum flow line was removed. Start-up steam flow requirements for the turbine driver were reduced by one-third. Feedwater system instability was completely eliminated and the previous 40 mils axial shuttling of the rotor was reduced to a practically unmeasurable amplitude. The frequency of the rotor vibration before the modification was 4 Hz, the same as the feedwater piping vibration frequency—both of which were completely eliminated after the simple geometry change. Another OEM pump design is shown in Figure 17. Its vibration history is shown in Figure 18, at various flows and speeds. As the flow was reduced below 60 percent of BEP, vibration amplitudes became excessive, even when the pump speed was greatly reduced. The new reduced vibration level at 18 percent flow and full 5200 rpm speed after proper modification of gap "A" is presented in Figure 19.

Three other pump types with basically the same modifications are shown in Figures 20, 21 and 22. The results were equally impressive.

Five major manufacturers' designs were involved in the design modifications. Pump types included diffuser and double volute designs, horizontally split and vertically split casings, motor driven and turbine driven, low speed and high speed applications. The degree of improvement was equally favor-

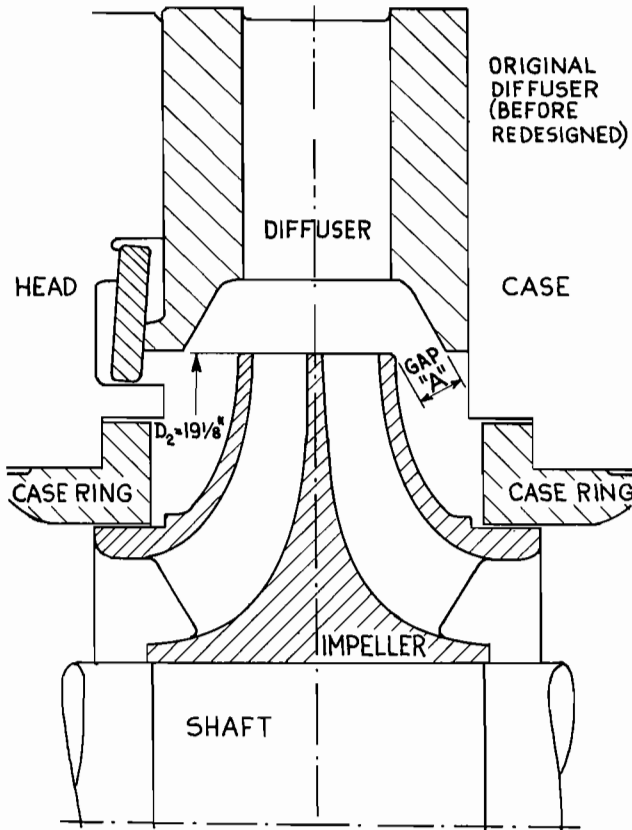


Figure 17. High Speed Reactor Feed Pump Before Diffuser Was Redesigned. Axial Vibration of Rotor Was 42 MILS at 26 Percent of "BEP" at Reduced Speed of 4000 RPM.

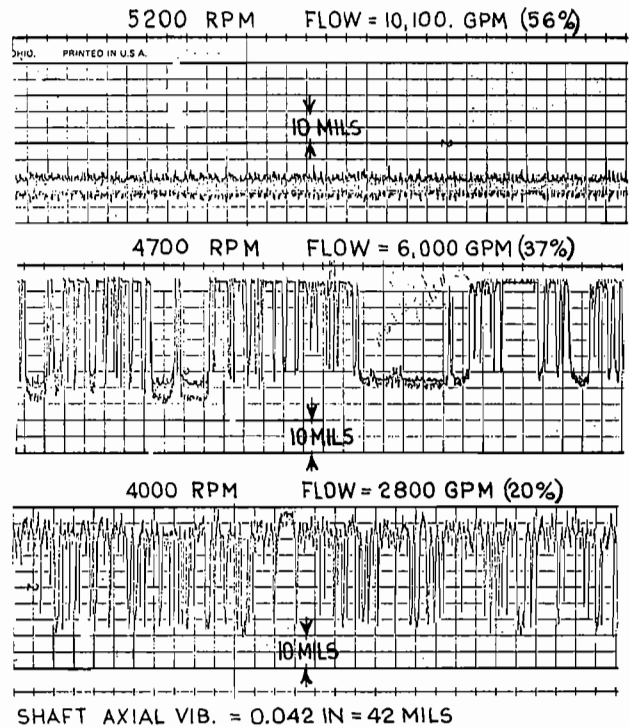


Figure 18. Axial Shuttling of a High Speed Nuclear Feed Pump Rotor. Rotor Vibration Level Was Normal at "BEP" (100 Percent Design Capacity). Maximum Vibration Level Was 42 MILS. Thrust Bearing Was 16 MILS. Shaft Movement Was Clearly Visible at Minimum Flow.

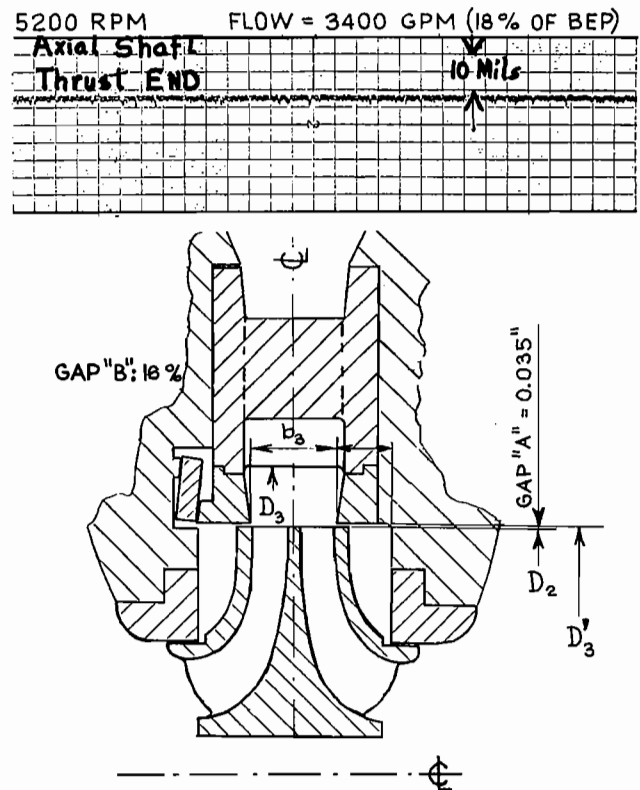


Figure 19. Axial Vibration of a High Speed Nuclear Feed Pump Greatly Reduces (from 42 MILS to 2 MILS Maximum) at 18 Percent of BEP Minimum Flow After the Diffuser Inlet Geometry Was Redesigned with Gap "A" = 35 MILS.

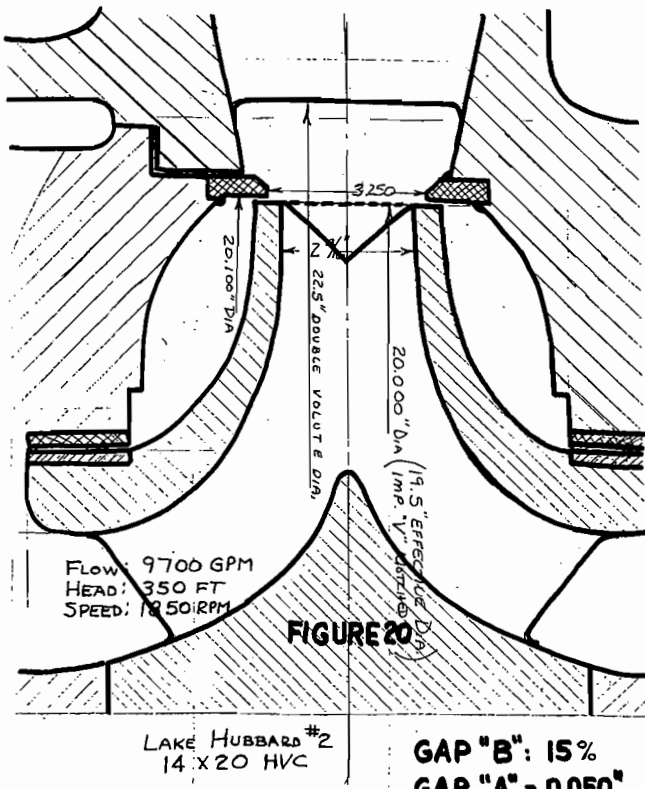


Figure 20. V-Notched Cut Was Applied at the Vane Exit to Further Reduce Amplitudes at Vane Passing Frequency. Insert Rings Were Added to Reduce Gap "A" to Eliminate Instability, at Part Load Capacities.

able, regardless of the make, design or speed, proving the importance of the basic technology behind gap "A," as a flow-instability controlling geometry at reduced capacity.

OTHER DESIGN IMPROVEMENTS

While modifying the previously mentioned 36 pumps, all possible design and material modifications were introduced to improve pump efficiency, reliability and, therefore, to increase plant availability.

CLOSE CLEARANCE SURFACES

Such items which operate at close clearances are shaft seals, impeller and casing wear-rings, balancing drums or disks and center bushings, if the impellers are the opposed type. Regardless of the OEM's practice, a hardness of approximately 10 Rockwell C difference was maintained between rotating and stationary components. Most impellers are 25 to 30 Rockwell C. The wear-rings were kept at 10 Rockwell C harder above that number. Some impellers were flame-hardened to about Rc 50, where mating surfaces were made in the range of Rc 10 softer. In both cases, the ring hardness fell between 37 and 40 Rockwell C.

MATERIALS

All components with close clearances were replaced with "Free Machining" 400 series stainless steel material. Depending on the hardness, different materials were used. When the material hardness required reached Rc 40 or above, AISI 416 SS was avoided. Because of its low carbon content, AISE 416 SS is subject to surface cracks at this hardness. In those cases, a

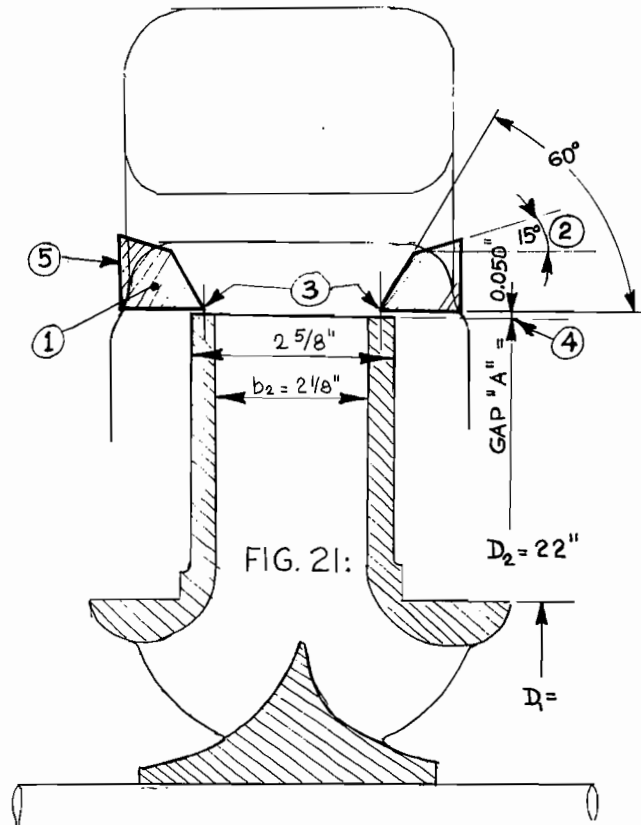


Figure 21. A 15° Taper Angle Is Used to Secure the Gap Reducing Ratings.

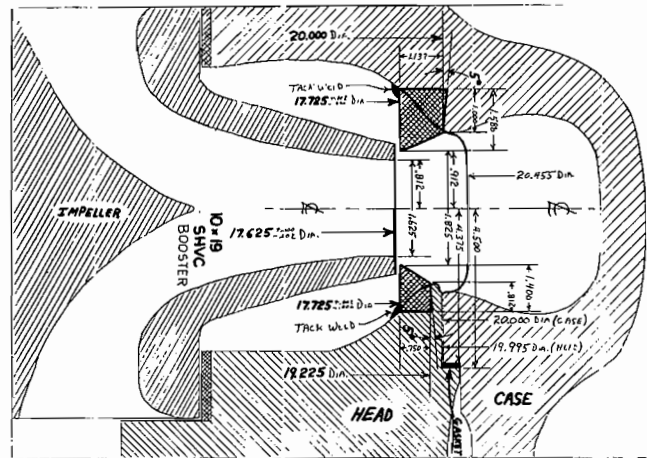


Figure 22. High Speed Booster and System Vibration Problems Were Eliminated With the Added Gap Reducing Rings.

material such as AISI 420F, with not less than 0.2 percent carbon, was applied in situations where a harder free-machining material was required.

GEOMETRY OF LABYRINTH SEALS

On all pumps in which seals were altered, the seals were systematically changed to a saw-tooth geometry, to improve efficiency and to eliminate the possibility of seizures. This labyrinth type geometry, combined with free-machining material, permits application of closer clearances. With this combina-

tion, pump efficiency and, especially reliability, are improved considerably.

IMPORTANCE OF GAP "A" GEOMETRY

Radial Gap "A"

Gap "A" actually acts as a low pass filter. Filtering out interaction between the hydraulic passages and behind the impeller side plate gives rise to a low frequency response (1 to 10 Hz). Determining the right gap was the very first question during the investigation. Various dimensions were tried such as 35, 50, 62, 75, 85, 100 and 125 mils. The effect of gap "A" was identical between 35 and 85 mils, and it lost its effectiveness completely at 125 mils. The effectiveness of gap "A" is the same between 35 and 85 mils, suggesting that the gap need not be smaller than any possible vibration amplitude of the rotor. Long multistage boiler feed pumps, when disassembled for maintenance, showed as much as 60-70 mils wear on close clearance surfaces, proving that internal vibration levels can be that high. An example where the gap was selected on the lower side of the range, resulting in contact between impeller and diffuser side plates during a transient operating mode that destroyed the rotor, is presented in Figure 3. Therefore, there is no need to jeopardize the integrity of the rotor with tight clearances, when the effect of gap "A" is adequate at a larger clearance. Each application was reviewed on an individual basis and the radial gap was selected accordingly. Fifty mils gap was selected for single stage booster pumps, while dimensions between 50 and 75 mils were selected for multi-stage pumps, depending on the flexibility of the rotor. On multi-stage pumps, which have large diameter impellers (100 mils), gap "A" was used with excellent results.

AXIAL OVERLAP

In addition to the radial gap, it was found that the overlap between rotating and stationary surface is equally critical. When final machining impellers, some manufacturers radius the impeller side plate at the O.D. (Figure 16), chamfer it (Figures 11 and 12) or make the impeller side plate too thin, reducing the useful and necessary width of the "low pass filter" surface. A wall thickness of not less than $\frac{3}{8}$ in is now preferred, on both large boiler feed pumps and booster pump applications. On some low pressure pumps, $\frac{1}{4}$ inch side plate proved satisfactory. The effectiveness of gap "A" greatly depends on the overlap. In all applications listed in Table 1, an overlap of not less than $\frac{1}{2}$ of the impeller side plate thickness was preferred.

INFLUENCE OF ASSEMBLY ON THE EFFECTS OF GAP "A"

One of the feedwater booster pumps' (Table 1) vibrational behavior did not improve satisfactorily after modification. The improvement was definite, but at low flows rotor axial vibration remained visible, as well as inducing suction piping low frequency vibration. It was anticipated that the overlap between the impeller and installed rings to develop gap "A" was inadequate on one side, because of improper re-assembly. After investigation, this turned out to be the case—only minimum overlap (edge only) existed on the inboard "low pass filter" surface. After centering the impeller on the gap "A" rings, the pump is operating such that the recirculation line has been taken out of service and main pump recirculation line now protects both the feedwater booster pump and the main boiler feed pump. This reduced the recirculation flow of the system by 2800 gpm and should eliminate first stage cavitation of the main pump. This principal of proper alignment also applies as well to multistage pumps. The centerline of the diffusers must line up with the

centerline of the impellers to get the desired results of gap "A."

Another assembly consideration is to be sure that the impellers are not loose on the shaft. This creates undue vibrations and stresses on impeller and shaft key-ways; it contributes to shaft breakages and rough-running pumps.

DYNAMIC BALANCING OF HIGH SPEED ROTORS

The balance of the pump rotors became a major concern. Without acceptable unbalance tolerance at 1 times rotational frequencies, all pump modifications and proper assembly practices can be negated. Realizing that balancing of rotational elements exhibits different characteristics in air compared to fluids, efforts were made to balance to the best degree practical. Each major rotating component was balanced separately, then assembled as a unit and balanced. This procedure will prove satisfactory in most cases. However, there are cases in which an incremental balance is required to reduce the 1 times rotational unbalance to acceptable levels. The authors have also standardized on an unbalance tolerance in gram-inches:

$$U(\text{gm-in}) = \frac{7\% \text{ Rotor wt}/2}{.0625 \times \left(\frac{\text{rpm}}{1000}\right)^2}$$

is the maximum allowable unbalance in any plane of the assembled rotor. This number is in agreement with the U.S. Navy's MILSPEC 167 using the formula:

$$U(\text{oz-in}) = 4 \times W/N \text{ max.}$$

where W is the total rotor weight in lb and N is the maximum rotor speed in rpm. We usually balance a high speed boiler feed pump rotor to four times better than both above formulas require.

Impeller Casting Quality

The disintegration of one of the impellers at the DeCordova unit (Figure 4) resulted in a major outage. Since then, impeller integrity has become a major concern due to the fact that this 800 MW unit has one 100 percent capacity boiler feed pump with no reserve. X-Ray examination is now required on all new purchased impellers. The authors are specifying that all final machined impeller castings meet ASTM E446 Class 2 on all high energy impellers. All utilities are encouraged to enforce this new requirement on components for all large, high energy input feed pump impellers. The cost of X-Ray examination of an impeller is negligible compared to the cost of a major failure and outage. So far the results of X-Ray examination, especially for regular sand cast impellers, have been very discouraging. This probably explains a large percentage of previous impeller failures. The question remains: What speeds, what hp input per impeller, head per stage and to what sizes is it essential to request X-Ray examination for cast impellers? Also, investment cast quality is preferable, instead of regular sand cast impellers.

Many impeller breakages have been traced to improper impeller exit geometry. For example, when impeller vanes are underfilled at the exit as shown in Figure 13, it is essential to put in a generous radius between the vane and the side plates. Machining that corner to a sharp edge certainly puts in a stress riser, initiating cracks between the vanes and the side plate. This is a very frequent failure mode for high energy input boiler feed pumps.

EXPECTED VIBRATION FREQUENCIES
AND PERMISSIBLE AMPLITUDES IN PUMPS

The most frequent and most important frequency components in centrifugal pumps that must be recognized are featured in Figure 23. The most frequently occurring of those five components shown in Figure 23 is the vibration caused by dynamic unbalance at rotational frequency. This can be quite misleading in centrifugal pumps. If a pump impeller casting is uneven or mismachined, a hydraulic force will develop that rotates with the impeller, giving the impression of dynamic unbalance. It actually is a hydraulically induced dynamic force that cannot be eliminated by fine balancing of the rotor.

Two times rotational frequency vibration component is frequent and can be caused by several items such as a loose bearing or seal bracket, misalignment between the pump and its driver, double volute casings with impellers which have an even number of vanes and many other items. Bently [11] has reported that a cracked rotor will also exhibit the two times rotational frequency component, and can be used as a diagnostic tool for detection of shaft crack propagation. His investigation is concentrated on large turbines.

Subsynchronous vibration components between 0.5 and 0.75 times rotational frequency is common for high speed pumps. That is a special class of vibration mode and is outside of the scope of this paper. When that frequency appears, it has to be taken very seriously; it can destroy a rotor in a matter of minutes, or sometimes in seconds. Loose bearing housings and excessive bearing clearance can excite that extremely dangerous frequency; therefore, those components should always be checked for correctness during assembly of a pump [12].

High vibration amplitude or pressure pulsation at vane passing frequency (or its multiples) is always an indication of inadequate radial clearance between impeller O.D. and diffuser/volute I.D., which is called gap "B" throughout this paper. That frequency is the noise generator of high energy input centrifugal pumps. When that frequency component appears, multiples of that component most likely will also show up. When the radial gap "B" is corrected, most likely the multiple frequency amplitudes will also be eliminated. It cannot over-emphasized that high speed boiler feed pumps (or process pumps) do not have to be noisy.

Very low frequency (0 to 10 Hz) vibration components are usually blamed on the feedwater piping and the feedwater control system. It is feasible that the control system, feedwater piping, and other components would be responsible for low frequency system vibration problems; however, among all troublesome systems investigated, changes to the booster and/or main feedwater pumps completely eliminated the system instability. Similar experiences have been reported in the literature [1,4,5,6,9,10].

On the permissible vibration amplitude levels, reference is often made to the Hydraulic Institute Standards (HIS). This is a misconception, because the HIS standards (1975, 13th Edition, p. 106) only refers to vibration levels caused by dynamic unbalance measured on the bearing housing. The information level concerning permissible vibration levels for high energy input feed pump is extremely limited. The requirements of the API are only slightly more definite.

Throughout the investigation, the investigators' best judgement was used in setting standards. Basically, the guidelines shown by Figure 24 for the synchronous (once per rev) and vane passing frequency ranges were followed; however, there is no proof yet to determine what the permissible levels really are for the vane passing frequency. At other frequency levels such as low (0 to 10 Hz), partial (0.50 to 0.75 times rpm) and two

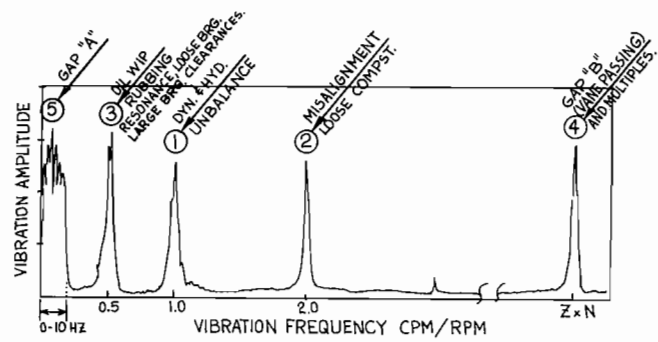


Figure 23. Five Major Hydraulically Induced Vibration Components.

times rpm, it was preferred not to have any trace of vibration at all.

Other frequencies, independent of pump speed and flow, have to be judged independently as they occur. Frequencies may be caused by acoustic resonances, critical speeds, etc., and may change in amplitude as speed and flows change. Those have to be dealt with individually, as they occur. Excessive bearing clearance of looseness between bearing and bearing housing can excite various vibration components, particularly the $\frac{1}{2} \times \text{rpm}$ component, which can destroy the machine. These two items are always to be checked for correctness during assembly of the pump.

RELATIONSHIP BETWEEN IMPELLER EYE
AND EXIT RECIRCULATION

As pump flow decreases from BEP, there is a point at

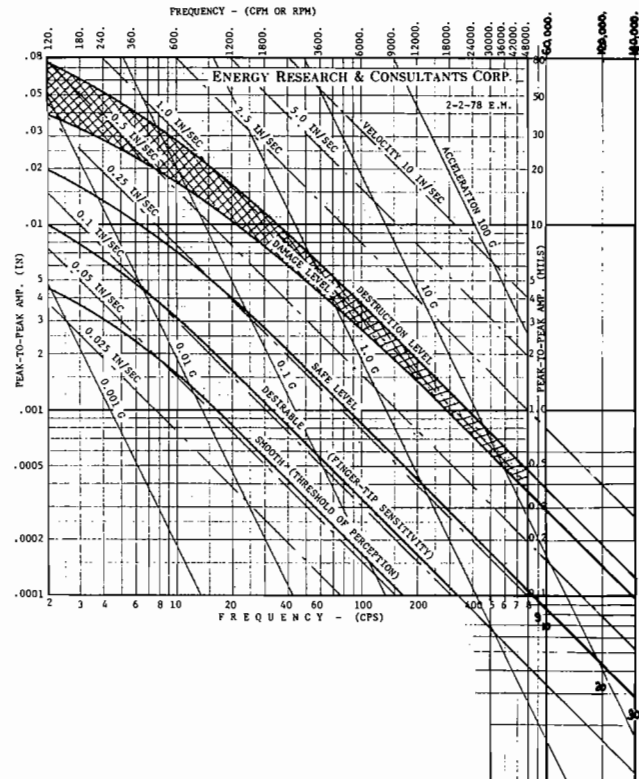


Figure 24. Allowable Rotor Vibration Levels Measured Relative to the Bearing Cap (Values Shown Are Filtered Readings at That Particular RPM or Frequency; Experimental Standards).

which flow instability sets in and causes pump and piping vibration to hinder plant operation. Sloteman, et al. [7] and Fraser [8] discuss flow recirculation in the impeller eye. As discussed in this paper, instability at the impeller exit, particularly behind the impeller side plates is not mentioned in literature other than the writers' [1,4,5,9 and 10]. The relationship between the recirculation in the impeller eye and the impeller exit is a subject for future research work and discussion.

CONCLUSIONS

In several cases, apparent unit system instability was brought under control by change in hydraulic channel geometry. Operation of the two 800 MW units, each with a single 100 percent capacity boiler feed pump, became stable at reduced loads after proper modifications, and both units can now fulfill the 35 MW/minute load change requirements. The changes discussed in this paper were introduced in all pump design types used in the utility industry, in all major manufacturers' products with equal success.

With all pumps modified, a 20 percent recirculation flow can now be maintained. A general recommendation is that 25 percent minimum flow is adequate, but it should not be less than 20 percent for high energy input pumps. More research is needed to make a definite judgement in this area.

Gap "A" proved to be one of the most important system vibration controlling parameters. Elimination of frequent seal failures was an important side result when gap "A" was properly corrected. There are no references to this in world literature, other than the authors'.

It was proven by the authors, again, that boiler feed pumps (of any size) do not have to be noisy. Gap "B" fully controls the noise level in a high speed centrifugal pump. The relative energy level as a function of the radial gap between rotating and stationary vanes is shown in Figure 25. Special attention has to

be paid to cases in which the number of impeller vanes is even, in a double volute case, or when the number of diffuser vanes is even. When both sets of vanes are even, the strength of the dynamic vane passing shock force is doubled as shown in Figure 25. The radial gap "B" then has to be higher than for normal stage designs.

Maintenance practices and procedures for pump reassembly are extremely important to the success of these modifications. Careful measurements must be taken before assembly to ensure that proper clearances and fits are correct. It is extremely important that impeller to diffuser/volute sideplate relationships are maintained.

Loose bearing housing (clearance between bearing upper half and housing) and excessive bearing clearance in cylindrical type journal bearings can excite an extremely dangerous sub-synchronous vibration frequency that can destroy a rotor in a matter of minutes, or sometimes in seconds. Therefore, those dimensions should always be checked very carefully during assembly of a machine.

The proper selection of material and its heat treatment selection proved to be a very important decision for close clearance wear parts and sealing components. Investigations proved that some of the individual pump parts, were in reality annealed, although specifications from the OEMs called for heat treatment. Other parts had been heat treated to their absolute maximum, allowing stress cracks to form. The most helpful material to prevent seizure in these components was the free machining AISI 420-F material.

Poor quality sand cast impellers introduced hydraulically induced forces that gave the appearance of dynamic unbalance. Even the finest degree of balancing could not eliminate those hydraulic forces. Investment casting quality, coupled with X-Ray examination, became the preference for high speed boiler feed pumps.

RECOMMENDED RADIAL GAP "STANDARDS":

DIFFUSER TYPE	4% MINIMUM,	6% PREFERRED
VOLUTE TYPE	6% MINIMUM,	10% IF POSSIBLE
SINGLE VOLUTE TYPE	8% MINIMUM,	12% PREFERRED

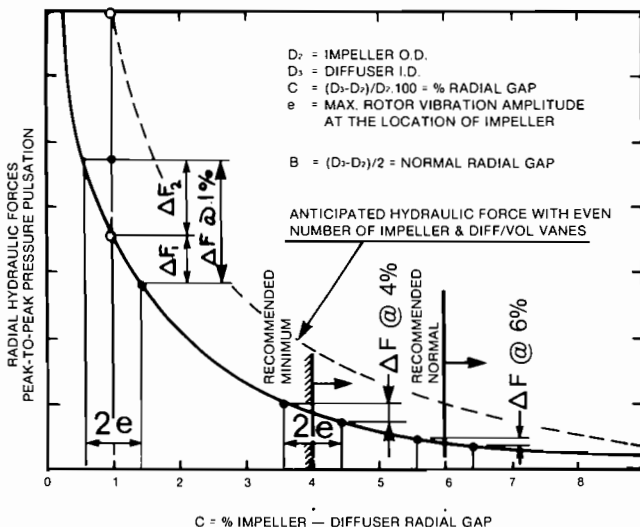


Figure 25. Influence of Impeller-to-Diffuser Radial Gap "B" on Pressure Pulsation and Magnitude of Radial Dynamic Force. Frequency of Forces Is Vane Passing or Its Multiples. If the Number of Impeller Vanes and the Diffuser/Volute Vanes Are Both Even, the Radial Gap "B" Has to Be Considerably Larger.

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