TWENTY-THREE YEARS OPERATING EXPERIENCE
WITH THE WORLD'S LARGEST BOILER FEEDWATER PUMP

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ABSTRACT

On the American Electric Power system, increases in generating unit size progressed from a 225 megawatt (MW) unit to the 1300 MW units, which remain the largest supercritical coal fired electric generating units in the world. Each of these units is equipped with a single boiler feedpump. The utility's experience has proven that the single, turbine-driven feedwater pump concept is extremely reliable and durable under base load conditions. Operating at these conditions, the changeout interval for pumps on the 1300 MW units was as long as five years. Since the 1980s, these units have been operating in the load cycling mode, reducing this changeout interval to as short as one year. Discussed herein are the utility's operational experience and the evolution of pump component design to accommodate the attendant increase in running time at reduced capacities. Changes were made in the hydraulic and mechanical design of the pump, and in overhaul and maintenance procedures. With these improvements, the original changeout interval has been restored for both base load and cycling operation. Continued developments in the area of high energy pump technology promise even greater durability in the future.

INTRODUCTION

American Electric Power (AEP) is a major investor owned electric utility with a total capability of over 23,000 MW, of which approximately 20,000 MW of capacity is generated by coal fired units with single turbine driven feedpump arrangements. AEP currently provides electric service to over seven million people in parts of Ohio, Michigan, Indiana, Kentucky, West Virginia, Virginia, and Tennessee.

Prior to 1957, AEP utilized three motor driven feedpumps for each generating unit, each pump rated at half the maximum boiler feedwater flow requirements. Feedwater flow control on these units was accomplished using a regulating valve that maintains drum levels throughout the unit design operating load range.

Projections of significant load growth for the AEP service territory indicated the need for large-scale expansion of system

facilities to meet expected increases in demand for electricity. The expansion program prompted investigations of plant designs to obtain every possible improvement in heat cycle performance at the lowest evaluated capital cost, while maintaining the high reliability and availability needed for the new generating units.

A major auxiliary, the boiler feedpump, appeared to have a good potential of being an area where improvement could be achieved from the viewpoint of cycle efficiency and reduced overall capital investment. A concentrated study on boiler feedpump arrangement alternatives resulted in adopting the installation of a single turbine driven boiler feedpump on each new electric generating unit. This pump driver concept has several advantages compared to the conventional multipump arrangements as noted below:

• Initial capital investment per kW is reduced.

• Auxiliary power system requirements are reduced.

• For a given main turbine generator rating, plant capability is increased by approximately two to seven percent for subcritical and supercritical units, respectively.

• Variable pump speed for flow control is ideally achieved with the inherently simple single turbine driven pump arrangement, instead of the conventional regulating valve that consumes pump horsepower by throttling feedwater.

The single boiler feedpump concept [1] was initiated on the 225 MW Unit 6 at the Glen Lyn Plant, placed in service in 1957, and continued on progressively larger generating units up to 1300 MW size (Table 1). There are now a total of 29 generating units (10 subcritical and 19 supercritical) constructed by AEP with single turbine driven boiler feedpumps, ranging from 5,650 to 63,200 rated brake horsepower (bhp). This unit design has proven to be an extremely reliable arrangement with a calculated availability of well over 99 percent, established from a total of 814 unit years of operation (as of 1996). All of the boiler feedpumps on these units were supplied by Ingersoll-Dresser Pump Company (IDP).

Table 1. Single Feedpump Operating History.

SINGLE FEED PUMP OPERATING HISTORY AT AEP

UNIT SERIES	PUMP DESIGN SPECIFICATIONS	PUMP MODEL	FIRST YEAR IN SERVICE	TOTAL UNIT - YEARS OF OPERATION
225 MW (9 UNITS)	3,590 GPM 6,050 TDH 5,600 RPM 6,270 HP	8CHTA-4	1957	342
260 MW (1 UNIT)	4,400 GPM 7,000 TDH 5,500 RPM 8,550 HP	10CHTA-4	1963	33
450 MW (2 UNITS)	7,200 GPM 11,170 TDH 3,600 RPM 21,610 HP	12CHTA-7	1960	70
600 MW (4 UNITS)	9,500 GPM 11,600 TDH 4,560 RPM 29,130 HP	120CHTA-5	1964	118
600 MW (6 UNITS)	12,550 GPM 11,300 TDH 4,900 RPM 37,550 HP	130CHTA-4	1969	145
1300 MW (7 UNITS)	21,600GPM 11,300 TDH 4,160 RPM 63,200 HP	20X25CA-4	1973	106
			TOTAL UNIT YEARS	814

A total of seven 1300 MW supercritical units have been designed and constructed by AEP utilizing the single boiler feedpump arrangement. These include the world's first successfully completed conversion of an 800 MW nuclear unit, to a 1300 MW coal fired unit at the William H. Zimmer Generating Station, which was placed into operation in 1991. (AEP jointly owns Zimmer with Cincinnati Gas & Electric and Dayton Power & Light.) The first 1300 MW unit was placed in service at the John E. Amos plant in 1973, with the remaining units following at the General James M. Gavin Plant in 1974 and 1975, the Mountaineer Plant in 1980, and the Rockport Plant in 1984 and 1989. During this period of time, pump modifications and redesigns were incorporated into the original pump design, continuously improving reliability and extending the pump and system operating range.

INSTALLATION FEATURES

The 1300 MW supercritical coal fired generating units are equipped with a single turbine driven boiler feedpump. The purpose of the feedpump is to raise the pressure of the feedwater supplied from the elevated deaerator storage tank through the high pressure heaters, economizer inlet header, and then through the steam generator and turbine. The simplified feedwater flow diagram (Figure 1) and heat cycle diagram (Figure 2), schematically illustrate the pump and feedwater system. The pump is driven by a variable speed steam turbine rated at 66,170 bhp. The output of the feedpump and ultimately feedwater flow is regulated by changing the speed of the drive turbine, which is adjusted by varying the opening of the steam turbine control valves.
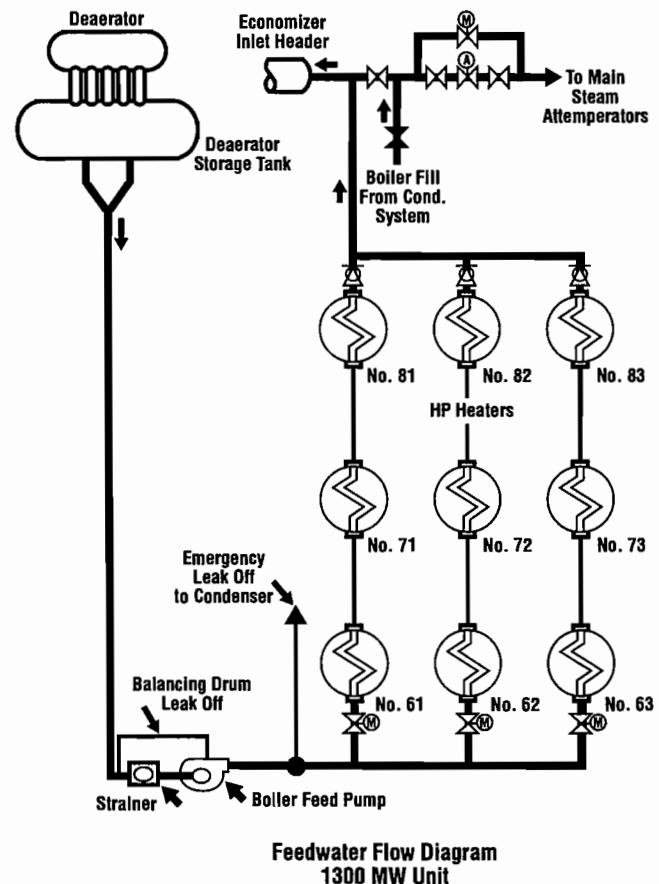

Feedwater Flow Diagram 1300 MW Unit

Figure 1. Feedwater Flow Diagram for 1300 MW Unit. (In the absence of a booster pump, NPSH to the boiler feedpump is provided by elevation of the deaerator in this single pump system.)

Pump suction flow is taken through the bottom of the pump directly after a single basket strainer. The basket strainer is installed in the suction line to prevent the entrance of small
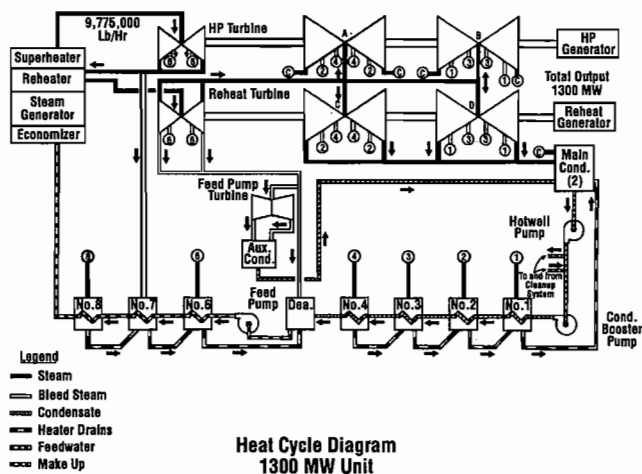
Figure 2. Heat Cycle Diagram for 1300 MW Unit.

particles of dirt, scale, and foreign material into the pump. A gear type flexible coupling, having a disconnect feature, is installed between the pump and turbine. The disconnecting feature is utilized to permit slow speed turning gear operation of the feedpump turbine without turning the feedpump rotor. The turbine requires turning gear operation to maintain uniform rotor temperatures prior to startup, and immediately after shutdown. Since there is no source of warm water in a single feedpump system, a pump warming system has not been provided. Therefore, revolving the feedpump rotor at the very slow turning gear speeds may cause internal contact in the precise running clearances as thermal distortions occur during temperature changes.

Pump discharge flow passes through the top of the barrel (Figure 3) where an emergency leakoff system is provided. The emergency leakoff minimum flow system, designed for approximately one-quarter of the rated feedwater flow (2,500,000 lb/hr or 5,400 gpm) at a developed pressure of 3,500 psig is installed to prevent damage of pump internals, mechanical vibration, and/or flashing caused by operation of the pump at reduced capacities. The reduced pressure emergency leakoff system is operational while the pump is in the startup and shutdown mode, where the pump operates at 3,200 rpm or below.

The feedpump was originally equipped with controlled leakage (floating ring) shaft seals utilizing condensate injection to minimize hot feedwater out leakage. The seal water injection system is a differential temperature type designed to maintain the temperature of the seal water discharge a constant level above the temperature of the seal injection water. This is accomplished using an automatic differential temperature control system that varies the flow of seal injection. The injection water supply is strained to remove any dirt particles or foreign material from the water before it enters the pump, minimizing the potential for contamination and rotor component seizure. To prevent flashing in and across the shaft seals, injection water supply is required any time the feedwater in either the pump or the pump suction line from the deaerator storage tank is above 175°F. The floating ring seals have been upgraded to a serrated labyrinth bushing sleeve and the automatic differential temperature control system remains.

The feedpump bearings are designed for force-feed lubrication with a thrust bearing and split sleeve journal bearings. The split sleeve journal bearings utilize a three element taper land design to stabilize the rotor from hydrodynamic instabilities commonly developed within bearings of high speed pumps. The feedpump and turbine utilize a common lubrication system. The inboard and outboard boiler feedpump bearings are equipped with vibration probes to monitor and record shaft vibration during operation.
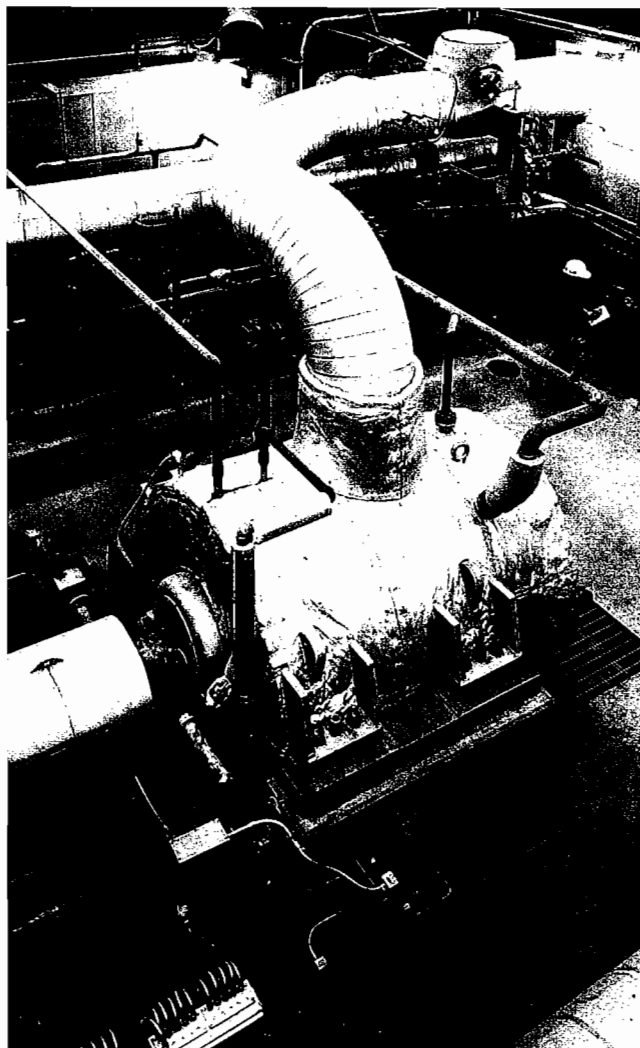

Figure 3. 65,000 HP Boiler Feedpump Installation. (This four-stage pump is driven by a variable speed steam turbine driven.)

PUMP DESIGN FEATURES

While the single boiler feedpump concept was first utilized on the utility company's system in 1957 (Table 1), the performance and cycle efficiency advantages of supercritical power generation became obvious as the demand for power increased rapidly in the USA from the 1950s through the 1970s. With the rapid increase in demand for power, the demand for development and construction of larger and larger boiler feedpumps followed. The increase in pump size over the past 40 years occurred rapidly in the period between the mid 1960s to 1980 (Figure 4). In 1969, the development of the highest power boiler feedpump, nominally 65,000 bhp, was begun. It is important to note that the largest pump size of this type at that time was a 40,000 bhp design that would be placed into service in 1969 at the utility company's 800 MW unit in Louisa, Kentucky. Therefore, the progression in pump design and size was required to proceed on the 1300 MW units without the benefit of proven experience on the nearest size similar design. To minimize risks associated with the development of any new design, the pump engineering team utilized a modelling technique based on design standards maintaining stress levels within proven limits. These techniques are necessary as it is difficult to determine actual working loads associated with operating turbomachinery at

the varied off design flow conditions. While the 1300 MW units were designed for base load operation, single boiler feedpump unit designs require the pump to operate in the startup mode and at reduced capacity for extended periods of relatively low load demand.
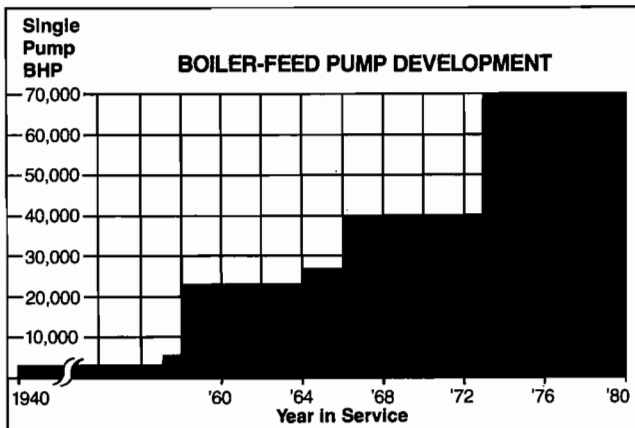

Figure 4. Boiler Feedpump Development.

These requirements became more important in the 1980s and early 1990s as power plants were operating more in the cycling mode. Impending electric utility industry deregulation and the need for design flexibility with improved pump flow rangeability is critical to ongoing competitive positioning of these single pump 1300 MW power plants.

The continuously rising performance characteristics (Figure 5) were developed to support the design conditions specified by the system heat cycle. More specifically, the design conditions for the utility company's 1300 MW unit boiler feedpump are:

- Discharge flow is 21,600 gpm

- Total head is 11,300 ft

- Suction temperature is 333°F

- Suction pressure is 185 psig

- Discharge pressure is 4,625 psig
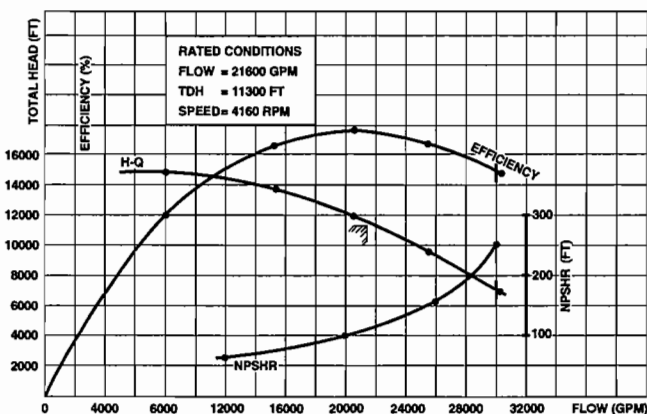
- Speed is 4,160 rpm

- Power is 63,200 bhp

Figure 5. Pump Performance Curves. (Constant-speed characteristics of the 20 X 25CA pump for a 1300 MW unit.)

The pump design (Figure 6) is a double case, multistage configuration with an inline impeller and diffuser arrangement. For a pump of this size and head generation, axial thrust balance must be achieved with a combined disc-drum balance device. Attempts to balance the axial thrust by opposed impellers, or any balance device that is not self compensating, would dictate a thrust bearing beyond the limits of practical design and manufacture. A four stage pump design was selected and developed to capitalize on the experience and design data available at that time. This configuration produced a state-of-the-art hydraulic design, including intermediate stage impellers with 25 in diameters and a stage head generation of approximately 3,000 ft, consuming approximately 15,000 bhp per stage. The inside diameter of the barrel is 40 in with a barrel outside diameter of 80 in resulting in a wall thickness of 20 in. The casing, forged of carbon steel with stainless inlay at high velocity areas, weighs in excess of 62,000 lb. The total pump weight, including casing, rotor elements, and discharge, head is 112,000 lb. The pump discharge head is 18 in thick, incorporating 18 six inch diameter studs to maintain pressure integrity. Martensitic steels of the internal assembly included 13 percent chrome rotating components (shaft, impellers) with five percent chrome stationary components (diffusers, channel rings).
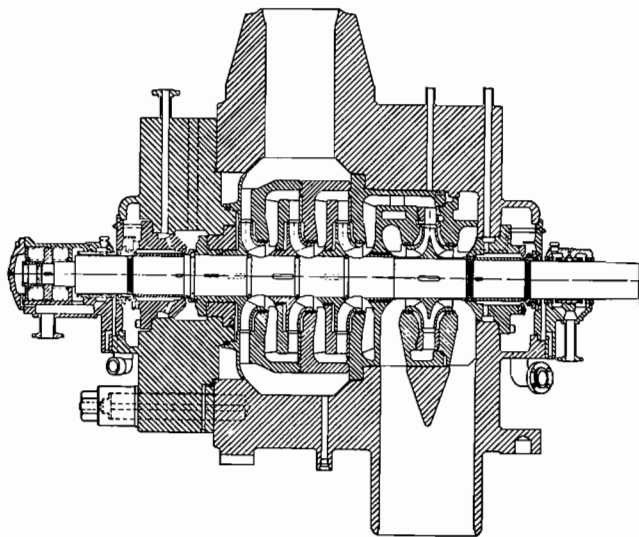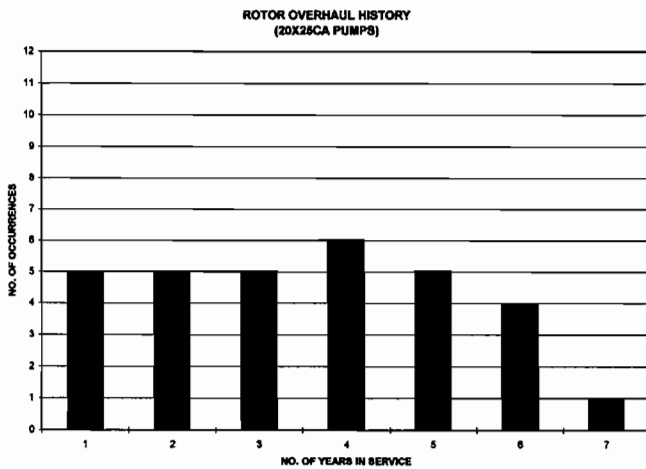

Figure 6. Pump Cross Section.

The uniqueness of this single feedpump arrangement is further complicated by the requirement to operate at supercritical pressures without a boiler feed booster pump providing suction pressure. Elimination of the boiler feed booster pump requires the location of the deaerating heater at an elevation that would provide a suitable net positive suction head (NPSH) margin to prevent first stage impeller cavitation damage and operate through thermal transients. To minimize the elevation of the deaerating heater, a double suction first stage impeller was incorporated into the pump design having a nominal suction specific speed (Nss) of 13,000 per eye (US units).

1300 MW FEEDPUMP
MAINTENANCE/OPERATING HISTORY

The utility company maintains a spare feedpump internal assembly for each of its four plants with 1300 MW units to accommodate scheduled and unscheduled, or forced, changeouts. The jointly owned plant maintains a separate spare internal assembly. The first 1300 MW feedpump rotating element replacement occurred in January 1978, to a plant's pump as a result of an outboard stuffing box seal failure. The pump rotating element had been in service over four years. To date, a total of 31 inner

rotating element replacements have occurred to these pumps following commercial operation. Of this total population of changeouts, nearly 80 percent are directly attributable to first stage impeller cavitation damage and/or failures that required either an unscheduled replacement or set the operating run length for scheduled change outs. Other causes that required unscheduled change outs include stuffing box seal failures, seizure of the rotating element (with the unit out of service), balance drum/sleeve failure, and an intermediate shaft sleeve failure. Based on the number of change out occurrences *vs* years of operating life, the current average rotating element operating life is slightly over three years (Table 2).

Table 2. Rotor Overhaul History.

ROTOR OVERHAUL HISTORY
(20X25CA PUMPS)

[bar chart: vertical axis "NO. OF OCCURRENCES" from 0 to 12; horizontal axis "NO. OF YEARS IN SERVICE" from 1 to 7; bars: year 1 = 5, year 2 = 5, year 3 = 5, year 4 = 6, year 5 = 5, year 6 = 4, year 7 = 1]

Historically, extended periods of minimum load operation have proven to have a detrimental effect on the first stage impeller service life because of accelerated cavitation damage at the lower pump speed and suction temperature. Reduced load operation on these units has increased significantly since the early 1980s and continues today. The 1300 MW units were designed to operate at base load but are capable of daily cycling between full load and a minimum load of 435 MW net. During startup of the unit, the feedpump initially rolls off to approximately 1,300 rpm and maintains minimum feedwater flow up to 3,200 rpm prior to supercritical (once-through) operation. At this minimum load, the pump operates at a point that is approximately 50 percent relative to its best efficiency point (BEP), adjusted for speed, with a feed water suction temperature of 270°F. It is now known that this combination of reduced capacity operation and low temperature accelerates cavitation damage of these high energy first stage impellers.

LIFE EXTENSION AND FLOW RANGEABILITY IMPROVEMENTS

While the utility company's 1300 MW boiler feedpumps amassed operating hours with excellent service and availability records over the last 23 years, it became apparent that elements of the design were in need of development, if improved pump operating life and availability were to be realized under power plant cycling conditions. The first stage impeller was suffering from premature cavitation damage requiring impeller replacement in a one to five (plus) year period, depending on the type of operation. During the original phases of pump design, the effects of high impeller inlet tip speed and material resistance to cavitation were not as well understood as they are today. Industry practice suggested that adding a margin to the performance NPSH would be sufficient to prevent cavitation damage [3]. NPSH was, at that time,

associated with preventing three percent head loss at high pump flows. Since the pump NPSH requirement, with margin included, would ultimately dictate deaerator height in the power plant, a low NPSH design was desirable. The plant and pump were designed with a margin providing impeller life that was sufficient to prevent lost generating time for a base loaded supercritical power plant. The deaerator was placed in a penthouse just above the top of the boiler roof providing an NPSH margin that would be considered adequate, based on past experience. However, increased operation at low plant loads required pump operation at off design conditions resulting in reduced first stage impeller life, and related damage to the five percent chrome stationary components (channel ring/diffuser assemblies, suction ring). As might be expected with high suction specific speed impeller designs, interaction between the pump and plant piping resulted in low frequency vibrations and excessive pipe movement, while operating at minimum feedwater flow and reduced loads.

Since the pumps remain the world's largest boiler feedpumps and operate at very high stage head conditions, the staffs of the utility company and the pump manufacturer have worked closely together in analyzing the operational, mechanical, and metallurgical component related problems. Several upgrades have been introduced into these pumps, improving availability and reliability. The most significant upgrades are:

• Comprehensive nondestructive examination (NDE) of shaft, impellers, and all stationary rotor component castings are performed for each complete rotor rebuild.

• First stage impeller design and material have been modified to resist cavitation damage and reduce low flow pressure pulsations.

• The suction ring design and materials have been improved.

• Smooth impeller hubs have been modified to include shallow grooves running next to the smooth mating surface of the stationary component wear rings (Figure 7). The stationary wear rings are manufactured from Type 420F stainless steel through hardened to 450-500 Brinell hardness number (BHN).

• Balance sleeve bore and active face surfaces are currently laser hardened to 500-550 BHN.

• Gap between impeller blades and diffuser vanes has been increased on the first stage.

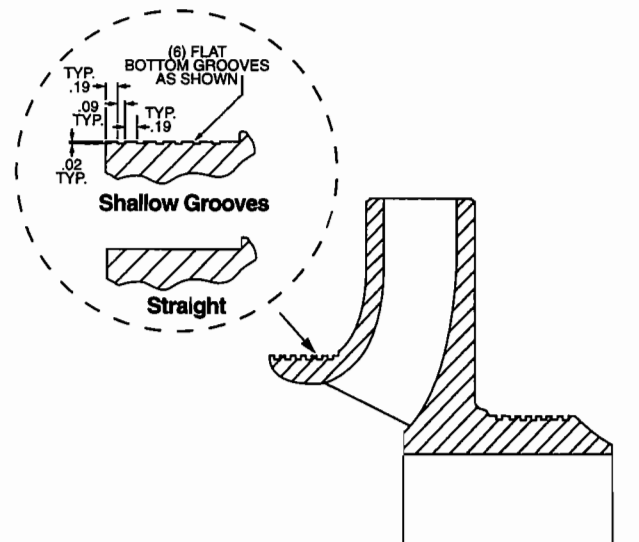• Floating ring seals have been replaced with fixed serrated sleeve bushings.

Figure 7. Shallow-Groove Impeller Wear Surfaces.

Boiler Feedpump Rotor Rebuild Capabilities

The utility company recognized that the lack of adequate control over refurbishing operations contributes to low pump reliability and unit availability [4]. Accordingly, they established a central machine shop (CMS) in 1974. This facility provides power plant equipment inspection and repair capability for all the utility company units. Primary work includes turbine and feedpump rotor rebuilds, electric motor rewinding, general machinery overhauls, and inplant nondestructive examination and related repairs. CMS is centrally located within the utility company's service territory in Charleston, West Virginia. CMS has been instrumental in coordinating and implementing many of the improvements incorporated in the utility company's boiler feedwater pumps. Working with the pump manufacturer, the utility company developed written procedures for repair, assembly, dynamic balancing, and nondestructive examination of this high-energy equipment. The initial 1300 MW feedpump rotor rebuild performed by CMS was in 1982 and has been followed by numerous rebuilds to the pump manufacturer's 20 X 25CA-4 pump. This was part of a program where over 100 rebuilds were done on both turbine driven and motor driven pumps, some of which are the boiler feedpumps for the 1300 MW units. Field installations are supervised by experienced pump manufacturer customer service representatives to ensure compliance with the latest developments and provide guidance as plant personnel install the overhauled rotating elements. The utility company has an installed population of nearly 80 boiler feedpumps, which are being overhauled and maintained using these approaches.

First Stage Impeller Redesign

The desire to extend boiler repair outage intervals from 12 to 24 months (or longer), coupled with increased operation at reduced loads, required a research and development program to evaluate the first stage impeller cavitation characteristics. The low NPSH design dictated a high suction specific speed first stage impeller and problems associated with part load operation. Ultimately these problems were solved using flow visualization testing and an advanced computational technique [5]. In addition to greatly minimizing the cavitation damage problem, the advanced "biased wedge" blade shape improved operation while at part load conditions. Prior to developing the "biased wedge" blade shape, full scale impellers were developed and tested in several 1300 MW boiler feedpumps with mixed results.

On the photo of a damaged blade (Figure 8), cavitation erosion is noticeable on the suction side surface and in the hub fillet area of the impeller. Pressure side surface damage, not visible on the photograph, usually occurred on a single blade of each side of the impeller. It became apparent that a rigorous effort would be required to make any significant improvement in the hydraulic performance of the suction stage. A research and development program was initiated in late 1991, as a cooperative effort between the utility company and the pump manufacturer that was ultimately completed in 1993, producing an improved impeller design. This program focused on the double suction first stage impeller design to improve the cavitation life, and reduce or eliminate the instabilities (caused by excessive cavitation activity in the impeller) that occur in the pump at minimum flow and/or low load operation. A 65 percent scale model of the impeller and suction bay flow visualization test rig were constructed (consisting of half of the double-suction configuration) to perform testing under cavitating conditions. Complementing the experimental program, an advanced analysis utilizing quasi-3D (Q3D) computational methods was initiated to generate new impeller blade designs that minimize the reduction of static pressure on the blade surfaces, and thus discourage the formation of cavitation. A database (CAD) is first utilized to construct a solid model of the impeller. Flow patterns through the impeller are then analyzed at both design and off design flows using the Q3D techniques that are later compared to the results of the flow-visualization test. To avoid trial and error inefficiency in arriving at solutions to difficult hydraulic design problems, the utilization of Q3D computation is considered essential.

Figure 8. Example of Cavitation Damage.

Model testing of the baseline hydraulic design faithfully simulated the cavitation activity existing on the full size units as indicated by the zones of cavitation erosion and flow visualization results. In addition, the model testing reproduced the regions of instability encountered when cycling the pump between full and minimum plant load. The improved impeller blade design, "biased wedge," developed utilizing the Q3D methodology, was found to operate nearly cavitation bubble free (Figure 9). By eliminating the cavitation bubbles on the surface of the blade, the oscillating vapor cloud (Figure 10) formed by the cavitation bubble reaction to off design operation was nearly eliminated, reducing the suction pressure pulsations that are suspected as the driving element for system instabilities [6]. The first full scale "biased wedge" impeller was installed in a boiler feedpump at a unit of the utility company in May 1992. Successful operation and significant improvement in cavitation damage have been verified by periodic video camera inspections over the last four years confirming the design integrity.

Figure 9. Comparison of Original and Biased-Wedge Impeller Designs. (Bubble activity at operational NPSH levels on the suction surface of the original impeller blade is shown on the left. On the right, the biased-wedge impeller is shown at the same operating conditions [5].)

Pump and piping system field performance, vibration, and pressure pulsation testing was conducted by an engineering team from the pump manufacturer and the utility company, before and after the introduction of this state-of-the-art design. Tests at the unit have quantified the improvement in the elimination of cavitation oscillation energy as compared to baseline testing of the original first stage impeller design configuration installed in the first 1300 MW pump.

Figure 10. Unsteady Cavity Behavior on Baseline Impeller. (These three photos were taken at the same flow and NPSH at different instants of time.)

Suction Ring Redesign

The suction ring is a large, fairly complex casting used to direct flow from the first stage impeller to the second stage impeller. Extensive cracking has been experienced on three suction rings around casting contours and shape changes requiring component replacement. These failures are somewhat unique in that the cracking is beginning to show up approximately 15 years after the pumps were placed in service. The cracking is believed to be associated with a change in operating mode producing higher alternating stresses that are now beyond the endurance limit of the five percent chrome martensitic steel used in the construction of these parts.

Initially, the NDE for this large casting included visual and dimensional inspection. The presence of nonvisible surface indications along with factory casting upgrades (weld repairs) during manufacture may now have reached a critical crack length. Based on recently obtained fracture toughness characteristics, five percent chrome material may have a finite life as applied in boiler feedpump applications.

Suction ring design changes have been incorporated to strengthen the part and reduce stress concentration in casting radii. In addition, replacement castings are produced from the higher strength 13 percent chrome martensitic steel increasing the endurance limit by approximately 25 percent. The new 13 percent chrome components are magnetic particle (MT) inspected in all accessible areas to eliminate indications that might develop into larger cracks.

Metallurgy Improvements

The materials selected for the pump construction have proven to perform adequately considering impeller exit velocities exceed 400 ft/sec and inlet vane tip velocities exceed 250 ft/sec. These exit and inlet velocities are nearly the highest in the world; the vane tip velocities of the pump manufacturer's 40,000 bhp pump used on a similar size generating unit with two 50 percent boiler feedwater pumps, exceed 280 ft/sec. In the 1980s, The pump manufacturer developed a patented [7] cavitation resistant copper alloyed 12 percent chrome steel that imparts strength by a precipitation reaction. This material applied to the critical service high energy impellers, in conjunction with the biased wedge inlet vane configuration, provides improved resistance to cavitation.

Internal wear and contact, which occurs during transients, is one of the main failure mechanisms of a utility boiler feedpump. Material hardness control is essential to reduce wear rates, and, therefore, increase pump life and operating efficiency. The pump

manufacturer has developed a laser hardening [8] technology to achieve a hardness range between 500 to 550 BHN on controlled metallurgy martensitic stainless steel (AISI 420). This process introduces a true metallurgical thermally treated surface hardening, without affecting the base material ductility and toughness. This technology had been used successfully for 12 years on smaller multistage boiler feedpumps. In recent years, this process to produce gall resistant material has been applied to these large boiler feedpumps providing a significant improvement in wear life. The wear surfaces of the balance sleeve, which is the stationary component that provides the mating surface for the axial thrust balance device, are currently provided in the laser hardened condition (Figure 11). Originally, the balance sleeve material was a through hardened high carbon, five percent chrome stainless steel (AISI 501), which was provided with a 420 to 460 BHN hardness range. At this hardness level, the material is susceptible to stress corrosion cracking, increasing the potential failure of this critical component in the boiler feedpump. To achieve a comparable wear surface hardness with a commercially available material and to eliminate the potential for stress corrosion cracking of the base material, the Laser hardened process is necessary.
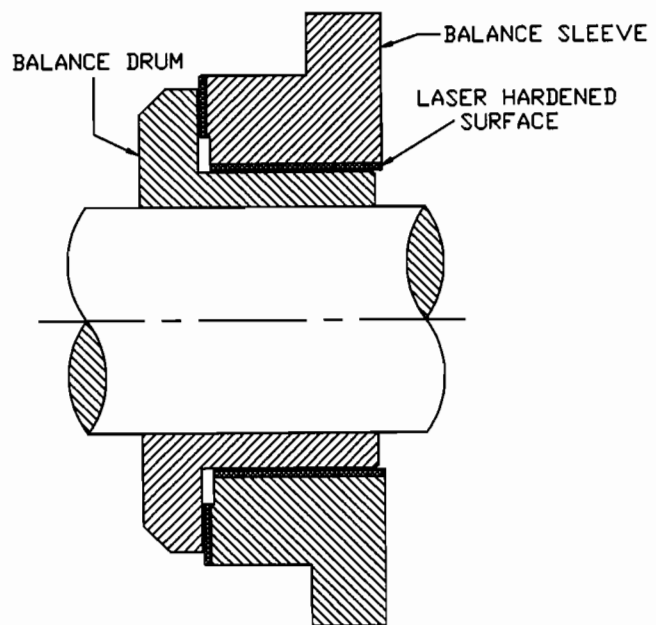

Figure 11. Laser-Hardened Balance Sleeve.

Concern that the five percent chrome materials used for the large stationary components of the pump internals may have a finite life prompted an investigation into the fracture toughness of five percent chrome, as compared to 13 percent chrome (CA6NM), at different operating temperatures. The original five percent chrome is susceptible to corrosion pitting resulting from occasional feedwater oxygen content excursions, over 0.04 ppm (40 ppb). The pitting may sensitize the material to corrosion assisted fatigue. The objective of this program is to predict the life of five percent chrome parts on the utility company's system and plan replacements to minimize lost operating time.

The prediction of life for any component requires an understanding of the loading, the type of flaws in the structure, and the mechanical properties of the material. While loading of the part is difficult to know precisely, estimates and predictions based on the observed failures can be used. The mechanical properties of the material can be obtained more precisely by testing in the laboratory. Originally completed in 1964, rotating beam tests of a

simple notch specimen with Kt $\cong$ 3 suggested an endurance limit for the five percent chrome material of 18,000 psi. The limit was used in establishing stress levels for the diffusers and suction ring. Recent tests for fracture toughness indicated that the five percent chrome material fracture toughness is approximately half of the toughness of 13 percent chrome material. Using this data (Figure 12), it can be projected that a part manufactured of 13 percent chrome material will operate 5.5 times longer under similar loading at a similar flow. A recent strategy has been implemented by the utility company to replace the original five percent chrome components with a higher fatigue strength and corrosion resistant 13 percent chrome material. A new spare rotor element, manufactured with all 13 percent chrome stationary components was delivered in 1996, and is now in service. All new stationary components are magnetic particle (MT) inspected in all accessible areas to ensure casting quality prior to installation. The original five percent chrome pieces will be replaced over the next several years, when significant cracking occurs and weld repair is considered impractical.
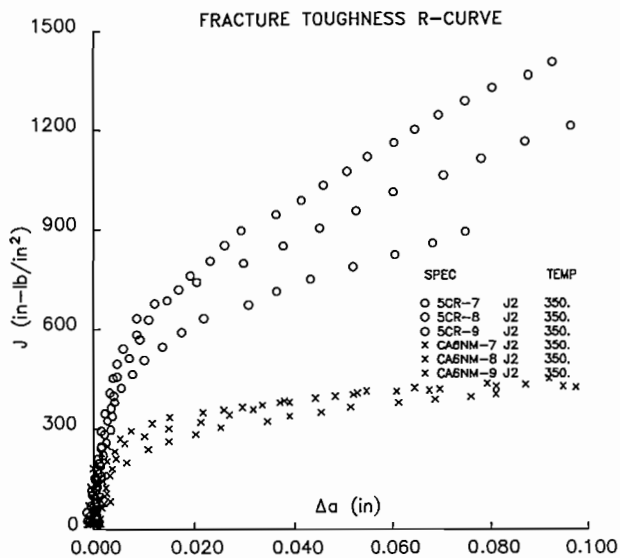

Figure 12. Fracture Toughness R-Curve. (Test data were obtained using test procedures identified in ASTM E-813-89.)

Axial Balance Device

A rotating balance disc-drum operates at a close axial clearance to the balance sleeve face. Like a hydrostatic thrust bearing, pressure developed inside the pump is directed between the drum and sleeve faces, producing the axial thrust balancing force. To compensate for changes in axial load produced as the pump operates at different speeds and flows, the axial clearance changes, balancing the impeller loads exactly. Any residual hydraulic axial load is completely balanced, resulting in a type of "artificial intelligence" for the disc-drum design. During maintenance inspections, minor contact has been observed on the balance drum and sleeve faces. This type of contact occurs during the initial roll off, startup, and coastdown when the axial settings are too close.

Since contact has been minor, installation of a spring loaded thrust bearing has been deferred. The spring loaded thrust bearing consists of a design that incorporates a series of springs that exert an expelling force (opening the axial clearance), during the described transient periods. With the springs installed at higher operating speeds and pressures, the rotating balance disc-drum assumes its optimum axial position, completely balancing the hydraulically developed axial thrust of the impellers.

Impeller-Diffuser Interaction

Impeller-diffuser interaction pressure pulsation, or vane pass energy, is intensified at low pump flows. The hydraulic thrust loads are significantly increased at lower loads, as are the static and dynamic pressures. High pressure pulsation amplitude led to cavitation on the suction side of the diffuser vanes of the first stage (Figure 13). This, in turn, weakened the vanes, resulting in breakage of the leading edges. The original first stage "B" gap (radial gap between diffuser and impeller vanes) was seven percent of the impeller radius. This was increased to 10 percent, eliminating the problem. This problem did not occur on the subsequent stages because of the higher pressure level. Therefore, the "B" gap remains at four percent. This technology also results in improved rangeability and reduces hydraulically induced structural vibrations. The pump manufacturer's "biased wedge" technology is being considered as an improvement to enhance this rangeability. As indicated by the favorable cavitation experience on the impeller, the biased-wedge remains stall-free over a wider range of operation.

Figure 13. Diffuser Leading Edge Cavitation Erosion.

Hydraulic unbalance arising from dimensional nonuniformities is also involved in impeller-diffuser interaction [9]. The original hydraulic elements were manufactured using conventional sandcasting methods, requiring extensive hand finishing to obtain reasonable dimensional uniformity. Investment casting technology, and the ability to produce precise vane shapes via stereolithography, has advanced to the point where large impellers and diffusers can be produced economically. Future pumps of this energy level should, therefore, incorporate precision casting techniques.

Shaft Seal Redesign

The original shaft seal arrangement incorporated an injection-style seal with a multiplicity of floating rings. The floating rings center themselves on the shaft as a consequence of the hydrostatic forces resulting from pressure reductions across the shaft seal. The advantages of floating ring seals include reduced leakage and lower installation and operating costs. As operating conditions varied, the floating rings tended to become unstable on the shaft, causing failure of the sealing system [10]. To accommodate plant cycling, the floating ring seals were, therefore, replaced by fixed, serrated, throttle bushings. This sealing arrangement has the added advantage of being less sensitive to the passage of foreign materials.

CONCLUSIONS

The design of any power plant with a single boiler feedpump requires unique expertise and experience in design with regard to

hydraulic, mechanical, and thermal phenomena. In the past 23 years, considerable time and effort has been invested in optimization of the single boiler feedpumps found in the supercritical power plants at the utility company. Most of this effort has been necessitated by the comparatively recent practice of cycling the power output of electric generating units to meet competitive demand fluctuations. The attendant development process has included plant personnel with first-hand knowledge of the pump and system problems, engineers with experience in high energy pump design, and research and development scientists utilizing state-of-the-art analytical tools. The resulting improved designs include the following features:

- Biased-wedge impeller blading

- Utilization of 13 percent Cr material for all hydraulic components

- Laser hardening of the balance sleeve wearing surfaces

- Shallow grooving on the wear ring surfaces

- Suction ring reinforcement

- 10 percent clearance (gap "B") between first stage diffuser vanes and impeller blades

- Fixed serrated-throttle-bushing shaft seals

Concurrent with these pump improvements, the utility company established control of maintenance, installation, and operation of boiler feedwater pumps on the system. This includes an engineering team that utilizes inputs from the pump manufacturer and develops procedures that ensure repeatable results, when undertaking routine maintenance.

Second and third generation hardware has already demonstrated significant life extension and improved availability. The design of the 1300 MW stations on the utility company's system, and the pumps utilized to provide the boiler feedwater were developed originally for base load operation, are now capable of extended periods of operation at reduced loads. Continued development of the world's largest boiler feedpump permits plant design teams to address the new challenges of the 1990s. Having occurred over a span of many years, this development process continues to address improvements needed for reliable pump operation at all flow conditions.

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