U. S. ATOMIC ENERGY COMMISSION
REGION III
DIVISION OF COMPLIANCE

Report of Inspection

CO Report No. 263/70-18

Licensee:	Northern States Power Company Monticello Nuclear Generating Plant License No. DPR-22 Category B
Date of Inspection:	November 5, 1970
Dates of Previous Inspection:	October 19-21, 1970

Inspected By:	H. D. Thornburg	Sr. Reactor Inspector	11-30-70
	C. D. Feierabend	Responsible Reactor Inspector	11-30-70
Reviewed By:	B. H. Grier	Regional Director	
Proprietary Information:		None	

SUMMARY

A management meeting was conducted with Northern States Power Company, General Electric Company, and Bechtel Corporation management personnel at the corporate offices of Northern States Power to discuss the status of the standby gas treatment and feedwater systems and progress toward resolution of the problems which have occurred within the two systems.

General Electric Company has completed a design review of the standby gas treatment system. The review identified causes for problems that have occurred during system operation and included recommendations for eliminating the causes and preventing future system

maloperations. The licensee is implementing the recommendations and has been operating the system daily to demonstrate increased reliability. Recent system operation has been satisfactory.

Problems associated with the feedwater pumps are not yet solved, although Bechtel Corporation reports progress in identifying causes of pump failures. Bechtel Corporation and DeLaval Turbine Company are placing highest priority in resolution of the problems associated with the feedwater system. The licensee considers that the feedwater system is presently the limiting factor in the critical path for power operation.

DETAILS

I. Scope of Inspection

A management meeting was conducted with Northern States Power Company, General Electric Company, and Bechtel Corporation personnel at the corporate offices of Northern States Power Company in Minneapolis, Minnesota, on November 5, 1970. The purpose of the meeting was to discuss the status of the standby gas treatment (SGT) and feedwater (FW) systems and to determine the status of progress toward resolution of the associated problems.

The following personnel attended the meeting:

Northern States Power Company (NSP)

J. C. Simandl, Manager, Construction
E. C. Ward, Senior Engineer, Plants
R. Duncanson, Superintendent, Steam Plant Operation
C. Larson, Plant Superintendent
M. Clarity, Assistant Plant Superintendent
D. Antony, Test Engineer
K. Gelle, Nuclear Engineer

General Electric Company (GE)

L. Wolf, Project Engineer, APED

Bechtel Corporation (Bechtel)

R. Anderson, Project Engineer

II. Results of Inspection

A. Standby Gas Treatment System (SGTS)

The inspectors reviewed the design review of the SGTS performed by GE. Mr. Wolf discussed the review in detail, describing the actions taken to date to upgrade the system, and the current plans for additional upgrading and testing.

Mr. Skarpelos, the GE engineer who was responsible for upgrading the systems at Oyster Creek and Dresden 2, was assigned to perform a design review of the installed system, to evaluate the system adequacy and reliability, and to make recommendations for providing an adequate and reliable system. The design review included review of the design specifications, review of the system with the responsible Bechtel designs engineer, discussions with GE, Bechtel, and NSP project personnel, and observation of the system in operation.

Mr. Skarpelos found that the system is adequate for the intended purpose, and that reliability could be improved. His report provided recommendations for changes to improve reliability. The following recommendations will be implemented to upgrade the system.

1. Standby Temperature Control

The charcoal filter strip heaters (cause of the fire in the filter described in a previous inspection report[1/]) will be removed or disabled by spot welding the terminal strips to the unit frame. This will be done prior to power operation. The heaters are being removed to eliminate any charcoal ignition source from the SGTS. Condensation will be prevented from forming on the charcoal filters by keeping the SGTS room warm. This will normally be accomplished by the plant heating systems. Supplemental electric heaters will be added to maintain the SGTS room temperature at 70°F. The plant heating system is presently operating.

1/ CO Report No. 263/70-16.

2. Operating Temperature Control

The temperature sensing elements will be relocated to a location between the filters, and the heater element output will be reduced from 36 Kw to 20 Kw (18 Kw will provide the required heat input). Temperature transmitter and control ranges will be changed to improve system response.

3. Fire Dampers

The fire dampers will be removed. Removal of the strip heaters eliminates a source of ignition for the charcoal filters. Removal of the dampers will improve system reliability as actuation of the dampers could prevent actuation of the standby filter train.

4. Cooling Air Intake

The source of cooling air will be changed so that it is taken from the SGTS room instead of from outside air. SGTS room ventilation intake will be modified to allow intake from either the turbine building or outside air.

5. Sample Ports

Five sample ports will be added for testing the HEPA filter. This will provide improved capability for DOP performance testing.

6. Glove Ports

The gloves will be removed for normal operation. This will prevent any possible flow blockage that could occur as a result of future deterioration of the gloves. The glove port covers will be provided with proper gaskets to make adequate seal without the gloves in place.

In addition, charcoal test clips will be added to provide for monitoring filter life, a spring will be replaced in a check valve to provide proper torque, new metallic filter tray labels will be installed to assure permanent identification of filters, the blowout panels will be sealed with soft sealant to reduce air leakage, and an audible temperature alarm will be provided in the control room.

Installation of dilution fan vortex damper stops is complete. The butterfly valves have been inspected and adjusted to eliminate any overtravel. Daily system testing is in progress.

A test program to demonstrate system reliability will be conducted after all planned upgrading is completed. The present plans are that this will consist of a system surveillance test three times a day for five days (15 total) with the criteria that the system must operate satisfactorily 15 of 15 times.

B. Feedwater System

Problems associated with the feedwater system were described in the previous inspection report[2]. The problems and the actions taken to resolve the problems were discussed by Mr. Anderson. Mr. Anderson discussed the subject of feedwater pumps in general, and stated that this problem is not new in the industry. Several pump manufacturers have had similar problems, apparently as a result of demands for high efficiencies. This is accomplished by providing high flow with few stages. Exhibit A, an article extracted from Power Magazine, describes the basic problem.

The pumps were initially tested at design flow rates at the vendor's facility, prior to delivery. These tests, however, were not performed at reduced flows, so the problems with low flow performance were not apparent until the pumps were installed and operated in the feedwater system. Although one of the pumps apparently performed satisfactorily during preoperational tests, subsequent failures of the inlet vanes showed that the pump performance did not meet system requirements.

Mr. Anderson stated that the initial modification to the inlet vanes did not provide satisfactory results. Although the stresses in the vanes may have been reduced, the change appeared to cause an increase in the pump internal flows, so that vibration levels were not reduced. Bechtel obtained consultant services from Pulsation Controls (Pulsco), Santa Paulo, California, to perform vibration measurement and analyses for the purpose of identifying the source of system vibrations. This information will be used to determine what system and/or component changes may be necessary to eliminate vibration problems.

2/ CO Report No. 263/70-17.

Initial analyses (with the modified impellers installed) showed that there were pressure peaks at both pump suction and discharge, however, Pulsco analyses did not indicate any significant contribution from system piping.

The vendor has redesigned the impeller to provide a smaller eye and seven vanes. The new impeller is expected to improve the low flow characteristics, however, this remains to be demonstrated. The first new rotor was expected to be delivered in about a week (approximately November 12, 1970).

Mr. Anderson stated that the pumps as originally furnished apparently performed satisfactorily at design flows, but that they must perform satisfactorily at lower flows also, and that satisfactory performance would be demonstrated.

Mr. Larson stated that satisfactory performance of the feedwater system is a prerequisite for power operation and that he considered this system to be the limiting factor on the critical path for power operation. The inspectors agreed that demonstration of satisfactory feedwater system performance will be required prior to recommending issuance of a full license.

Mr. Thornburg indicated that consideration should be given to the preparation of a written procedure for measuring feedwater pump performance and for inspecting the feedwater pumps as a part of the power testing program. NSP personnel indicated that such a procedure would be prepared prior to power operation.

Attachment:
Exhibit A

Eliminating pump-stability problems

Boiler feed pump vibrations can be reduced considerably, perhaps eliminated, by selecting hydrostatic bearings over conventional hydrodynamic bearings

Competition in the boiler feed pump (BFP) market has created a situation where desired pump efficiencies are exceeding current technological capabilities. Though performance figures appear somewhat inflated now, competitive position in the future demands that higher efficiencies be delivered without creating operational problems.

Many phenomena tax safe operation, especially at partial loads. They include vibration from hydraulic forces, higher noise levels, larger and unpredicted axial thrust, damping effects on critical speeds and nonsynchronous response of the flexible rotor. Further, pressure pulsations in the impeller and hydraulic passages have been greatly magnified in high-output, high-speed pumps. A frequent result of these pulsations is fatigue failure of the impeller.

Although research continues, no concrete solutions for pump-vibration problems have yet been found. The total problem of pump instability involves complex interactions among hydraulic, geometric and mechanical features of a particular unit.

To locate the causes and solve vibratory problems, you must examine carefully both the failure mechanism and pump geometry. Several mechanisms leading to intolerable vibrations are: (1) rotating stall in the impeller, (2) stall in the diffuser and guide channels, (3) secondary flows, (4) cavitation from a low net-positive-suction head (NPSH), and (5) oscillations caused by rotor dynamics.

The first four situations are generally observed during low NPSH conditions and transient part-load operation. The fifth, somewhat independent of flow and NPSH, is strongly dependent on speed variation and bearing characteristics. Pump tests at varying speeds have shown that, at any flow and NPSH condition, there is at least one critical speed where pressure oscillations are most pronounced. Frequency of these oscillations depends on design quality.

A thorough understanding of *stall* is important if you have excessive BFP vibration (figure, lower right). Rotating stall in impellers occurs this way: When the load condition of a centrifugal machine changes, direction of flow also changes in the cascade of blades. That is, as the incidence angle—difference between flow angle and pump-impeller or diffuser-vane inlet angle—increases, you get flow separation first, then stalling. However, because of nonuniform flow upstream of the blades or manufacturing inaccuracies, one channel stalls before the others.

Breakdown of flow in this passage causes a deflection of the inlet fluid stream. Thus, one neighboring passage receives fluid at a smaller incidence angle and another at a larger angle. Result: Passage with the larger incidence angle stalls, and cyclic rotation of the phenomenon begins. For centrifugal pumps with blades bent backward (where inlet angle is less than 90°) stall rotates in the direction of impeller rotation at high loads and in the opposite direction at low loads. Similarly, stall can occur in diffuser channels from a change in incidence angle during low-load conditions. If diffuser is not properly designed, stall also will be evident at full load.

Vibration is strongly influenced by pump-stage geometry (figure in center of column at right). Conclusion based on many laboratory and field tests is that the following areas should be carefully designed if induced vibration, cavitation and general instability are to be reduced:

- Geometry of the inlet guide vanes
- Impeller-eye geometry and shaft size
- Impeller-discharge geometry, exit angle and vanes
- Wearing-ring clearance
- Radial gap between the impeller and the diffuser
- Impeller/diffuser alignment
- Diffuser geometry
- Axial gap between the impeller and stator.

Also, air trapped in suction and discharge piping is an occasional cause of instability. However, this is not of a permanent nature: eventually it is washed out and smooth operation is restored.

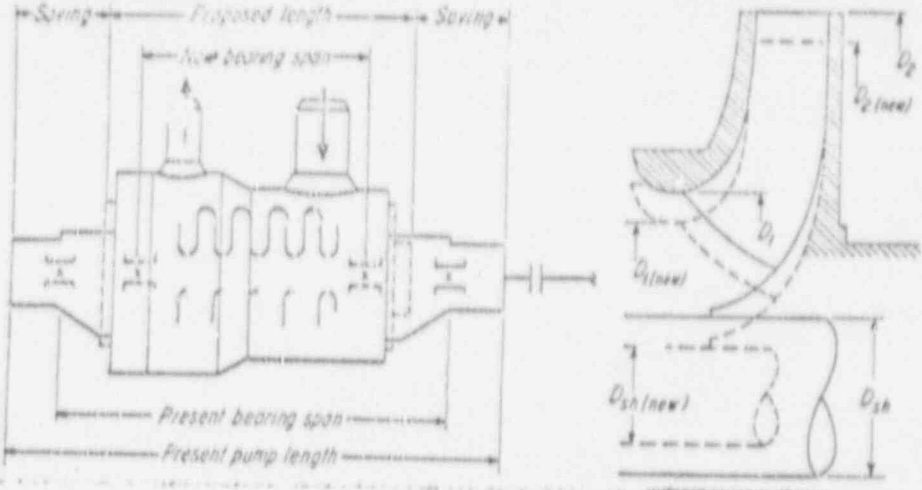
Degree of instability is determined by observing: (1) frequency and amplitude of flow-pressure oscillations, (2) lateral vibration, which induces pressure waves, (3) axial vibration—this, too, induces pressure waves, and (4) variations in axial thrust induced by pressure oscillations.

Poor design or wear at any one of the locations listed above may be responsible for these oscillatory disturbances. Realize that frequency range and peak-to-peak amplitudes depend on magnitude of axial thrust, level of efficiency and head stability.

Pressure pulsations can be defined as low- and high-frequency response of fluid particles to complex, unsteady, nonlinear forces. Low-frequency, high-amplitude pressure pulsations result in visually observable movement of pump

By E Makay, Franklin Institute Research Laboratory

EXHIBIT A



Hydrostatic bearings offer significant reduction in boiler feed pump size and cost. Their load-carrying capacity is maintained by an externally pressurized fluid. Hence, fluid film separates shaft from bearing even at zero speed.

Bearings must be pressurized: Speed of rotation is too slow to give sufficient load-carrying capacity from hydrodynamic action alone —recall, viscosity of water is very low at high temperature.

In hydrodynamic bearings, fluid pressure supporting a load is generated within the bearing by relative motion of bearing and shaft.

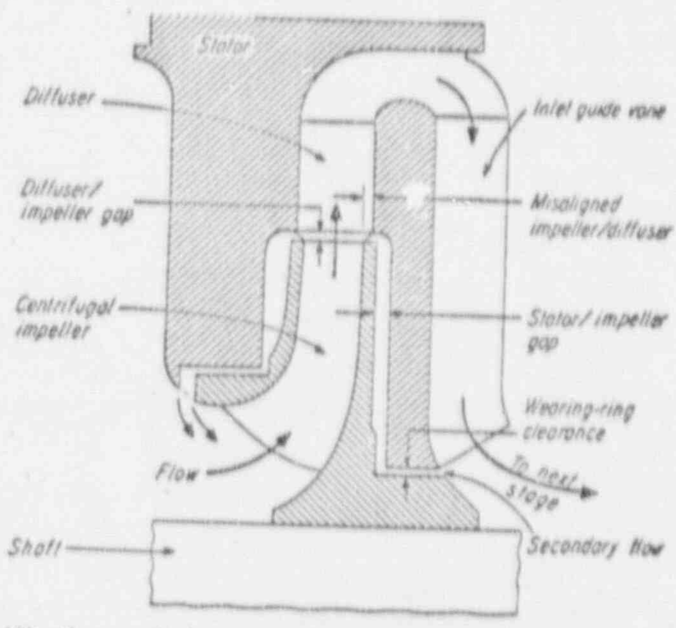
and connecting pipe. They also cause gross fluctuation in discharge flow. High-frequency, high-amplitude pressure pulsation degrades pump performance somewhat; more important, it causes accelerated damage to the rotating and stationary cascades.

Many pump-instability problems can be eliminated, or at least reduced, if hydrodynamic, oil-lubricated journal bearings are replaced by hydrostatic bearings using pressurized boiler feedwater as the lubricant. There are many advantages to be gained by selecting hydrostatic bearings. For example, overall length of a feed pump can be shortened by approximately 40%. Reasons: Water-lubricated bearings can be enclosed within the pump barrel, and seals can be eliminated (figures in tinted area).

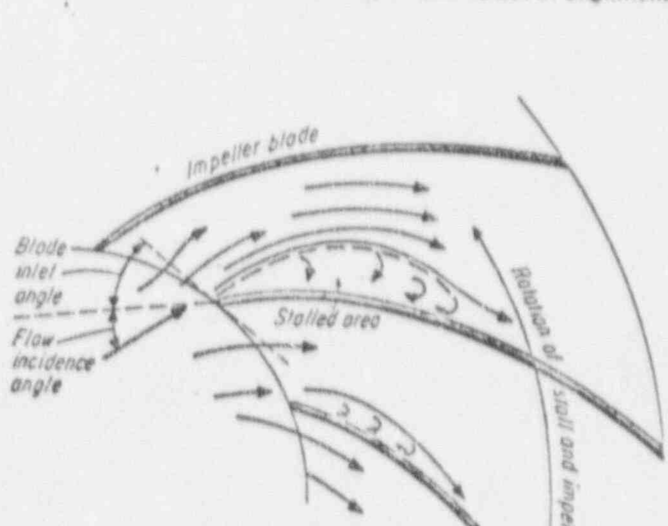
Further, when a shorter bearing span is employed, shaft deflection does not dictate the shaft diameter; it is determined by stress levels alone. To illustrate: For an 80,000-hp, five-stage unit, shaft diameter is estimated at 9.5 in. when hydrodynamic bearings are used. Hydrostatic bearings permit an 8.25-in. diam shaft with its favorable influence on pump performance and life. Other advantages:

- Smaller impeller eye produces a better flow path, yielding increased efficiency and NPSH
- Longer blade path offers more favorable blade loading
- Smaller diameter wearing ring causes less leakage, giving higher pump efficiency
- Smaller clearances at the wearing surfaces give better partial-load performance and less danger of instability
- Secondary flow at the impeller eye during low loads is decreased, reducing vibration caused by pressure pulsation
- Smaller impeller produces the same head, lowers disc friction and reduces barrel stresses
- Higher speeds are possible, reducing unit size
- Oil lubrication system is eliminated
- Capital cost is reduced.

Although most BFP-bearing experience has been with hydrodynamic journals in the laminar regime, the technology for turbulent bearings is well established and has been applied successfully to other demanding applications in rotating machinery. Recently developed computer programs solve hydrostatic bearing problems to improve performance predictions for both dynamic and static loading, and determine the effects of turbulence in the bearing film. These programs have been verified experimentally for general modes of operation. ●



Vibration, cavitation, and instability can generally be traced to one of several possible locations in a centrifugal pump stage. For example, wearing-ring clearance might be excessive or perhaps the impeller/diffuser is out of alignment



Stalling occurs when the incidence angle—difference between flow angle and pump-impeller or diffuser-vane inlet angle—increases above a specific critical value. Stalled area, which eventually washes out, reforms as rotation continues