Feed-Pump Hydraulic Performance and Design Improvement, Phase I: Research Program Design Volume 1

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Prepared by
Borg-Warner Corporation
(Byron Jackson Pump Division and
Borg-Warner Research Center)
Carson, California
and
Massachusetts Institute of Technology
Cambridge, Massachusetts

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**CS-2323, Volume 1
Research Project 1884-6**

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Prepared by

BORG-WARNER CORPORATION
17929 Adria Maru Lane
Carson, California 90746

Byron Jackson Pump Division
Borg-Warner Corporation

Principal Investigators
W. H. Brown, Project Leader
S. Gopalakrishnan
R. Fehla

Consultant
A. J. Acosta

Borg-Warner Research Center
Borg-Warner Corporation

Principal Investigator
W. E. Thompson

MASSACHUSETTS INSTITUTE OF TECHNOLOGY
77 Massachusetts Avenue
Cambridge, Massachusetts 02139

Principal Investigator
D. G. Wilson

Prepared for

Electric Power Research Institute
3412 Hillview Avenue
Palo Alto, California 94304

EPRI Project Manager
I. A. Diaz-Tous

Fossil Plant Performance and Reliability Program
Coal Combustion Systems Division

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Prepared by
Borg-Warner Corporation
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and
Massachusetts Institute of Technology
Cambridge, Massachusetts

ABSTRACT

As a result of prior EPRI-sponsored studies, it was concluded that a research program should be designed and implemented to provide an improved basis for the design, procurement, testing, and operation of large feed pumps with increased reliability and stability over the full range of operating conditions. This two-volume report contains a research plan which is based on a review of the present state of the art and which defines the necessary R&D program and estimates the benefits and costs of the program. The recommended research program consists of 30 interrelated tasks. It is designed to perform the needed research; to verify the results; to develop improved components; and to publish computer-aided design methods, pump specification guidelines, and a troubleshooting manual. Most of the technology proposed in the research plan is applicable to nuclear power plants as well as to fossil-fired plants.

EPRI PERSPECTIVE

PROJECT DESCRIPTION

RP1884-6 is part of an EPRI program to improve the reliability of feedwater pumps for large power-generating units. Problems of feedwater pumps are a leading cause of unscheduled outages, and they are estimated to have cost more than \$408 million for replacement power alone during 1981. The EPRI-sponsored program will be a multiyear effort to provide an improved basis for the design, procurement, operation, and testing of large feedwater pumps with increased reliability and stability over the full range of operation. Phase I of this effort was completed by two independent teams; the result of one team is reported here. The Phase I work was the preparation of a research program design for the remainder of the overall EPRI effort, including both theoretical and experimental research. The work was based on surveys of architect-engineers, utilities, independent research laboratories, and pump manufacturers as well as on the resources of the team that performed the work.

In preparation for RP1884-6, EPRI sponsored several projects, which are reported in the following publications: Survey of Feed Pump Outages (EPRI Final Report FP-754), Centrifugal Pump Hydraulic Instability (EPRI Final Report CS-1445), and Recommended Design Guidelines for Feedwater Pumps in Large Power Generating Units (EPRI Final Report CS-1512). The complementary reports on Phase I work are being published as EPRI Final Reports CS-2322, Volumes 1 and 2.

This two-volume report was prepared to provide utilities, manufacturers, and other interested parties with the results from the Phase I research program design and related activities. The program plan is expected to have a major influence on shaping EPRI work on feedwater pumps, and the program plan will provide an overview of the work that is required to improve the reliability and stability of large feedwater pumps. In addition, the Phase I effort offers a

review of feedwater pump problems, design methods, theory, and research requirements.

PROJECT OBJECTIVES

The general objective of this project was to prepare a research plan for a multiyear effort to improve feedwater pump reliability and stability. The plan was to include resource and facility requirements together with schedules and work descriptions.

Specific objectives were:

- To review feedwater pump design methods
- To survey problems in boiler feed pumps
- To assess feedwater pump experimental research
- To review pump theory
- To formulate a statement of pump research requirement
- To detail plans for the proposed research program

PROJECT RESULTS

A comprehensive research program was formulated to develop a new generation of advanced design feedwater pumps that would be highly reliable and stable throughout their operating range. The proposed research effort would require a five-year period and would cost approximately \$5.2 million (1981 dollars).

A Phase II integrated experimental and analytic effort would be undertaken in the following fields: hydraulics, cavitation, rotor dynamics, and pump systems (interactions with connecting components and piping). Full-scale verification tests would be performed to examine the combined results of the foregoing efforts. Application studies of pump components and economic methods would be made.

In Phase III, design methods for large feedwater pumps would be developed. Computer-aided design methods would be prepared for hydraulic analysis, cavitation, stress analysis, and rotor dynamics. Pump specification guidelines would be prepared on the basis of the

entire program, including monitoring and troubleshooting, materials selection, and code modification.

I. A. Diaz-Tous, Project Manager
Coal Combustion Systems Division

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EXECUTIVE SUMMARY

1.0 INTRODUCTION

Replacement energy costs alone, caused by feed pump problems, were \$408 million in 1981. Plant outages caused by feed pump problems occur at significantly higher rates in large plants than in small plants. Therefore, it was concluded that a research program be designed and implemented to provide an improved basis for the design, procurement, testing, and operation of large feed pumps with increased reliability and stability over the full range of operating conditions.

The purpose of the Phase I effort is to provide a Research Plan that is based on a review of the present state of the art. The plan defines the necessary research and development program and estimates the benefits and costs of the program.

Most of the technology proposed in the Research Plan is applicable to nuclear power plants as well as to fossil-fired plants.

2.0 BENEFIT/COST RATIO

The estimated benefit/cost ratio for the entire research and development program is 154:1. It is more than 100:1 for each research category. These ratios are based on an estimated cost of \$5.2 million 1981 dollars and the assumption that the program starts early in 1983 and continues through 1987.

3.0 FEED PUMP PROBLEMS

Identified major outage-producing feed pump problems include vibration, impeller breakage and cracking, shaft seal failure, rapid wear of wear rings, cavitation damage, axial balancing device failure, broken or damaged shaft, journal bearing failure, thrust bearing

failure, seizure of wear rings and other close-clearance running fits, unstable head-capacity curve, auxiliary system failure, hot misalignment, and lack of flexibility in gear-type couplings. Systematic and thorough analysis of each failure and determination and reporting of the root cause are seldom possible because of the manpower limitations imposed on the typical operating power plant and the typical utility engineering staff. However, it is generally agreed that many of the problems are related to operation of pumps at very low flow or at very high flow.

4.0 CURRENT STATE OF THE ART

Current experimental research directed toward solving feed pump problems and increasing reliability and stability is not being conducted on a large scale, nor are the research efforts being coordinated to achieve these goals.

4.1 Hydraulics

A theory that encompasses the combined effects of the interdependent hydraulic components of a centrifugal pump on the real fluid stream flowing through the pump is not yet available. The present inability to model mathematically the interaction between the hydraulic components (suction, impeller, and volute or diffuser) has thwarted the prediction of performance. Further research in this general area is needed, specifically to improve upon the theory that describes the real fluid stream flowing through a pump. In existing theoretical methods it is generally assumed that the flow in one of the three hydraulic components of the pump can be analyzed with sufficient accuracy, neglecting the influence of the other two. It is also assumed that the flow is inviscid, irrotational, steady, and two dimensional. As a result of these assumptions it is possible to use potential flow theory. This method is important because it forms the basis for more advanced techniques. However, when used alone it does not match experimental results very well.

4.2 Vibration and Rotordynamics

An increasing demand for greater reliability of high-speed rotating machinery and the introduction of specified vibration limits have created renewed interest in the development of techniques to anticipate vibration problems and methods to alleviate them. In the rotordynamics area there are several types of analyses to be considered: critical speeds, instability thresholds, and synchronous and nonsynchronous forced response. Generally, the calculation procedures for these analyses are well founded, but the methods for determining the parameters necessary for the calculations are not. However, there is an EPRI-sponsored program to determine the stiffness and damping coefficients for various wear ring configurations and to study the advantages of squeeze-film damper bearing supports. This program has as its objective the development of advanced rotor-bearing systems for feedwater pumps.

4.3 Cavitation

Cavitation can affect overall pump performance and reliability in many ways. It can cause noise and vibration. Most importantly, cavitation can produce large-scale erosion. Because of its destructive effects, both designers and users have employed various test techniques and basic research to determine the cavitation characteristics of centrifugal pumps. Despite this effort, progress in coping with the problems it creates has been slow. The problem of cavitation combines complex material, chemical, and hydrodynamic relationships. However, empirical predictions as to how and when the onset of cavitation may occur are available. Predictions as to its effects are more difficult. Although research has produced superb scientific studies in bubble dynamics, incipient cavitation, cavity mechanics, noise, material erosion resistance, and ways to alleviate cavitation, there is still no complete and cohesive theory allowing application of this knowledge to feed pump cavitation problems.

4.4 Impeller Forces

With the recent great increase in power-to-weight ratio of all forms of turbomachinery, it has become necessary to evaluate the

consequences of two additional fundamental conditions of the flow of the working fluid. First, the forces arising from the pressure distributions imposed upon both the rotor and the stator must be predicted and/or measured; and second, the energy conversion process must be assessed. It is now well established that the energy conversion process is a function of time-varying or unsteady pressure fluctuations. Thus, we may expect the resultant forces to be unsteady as well. The forces imposed on a pump impeller, arising both from flow within the impeller and from interaction with flow outside the impeller in a volute or diffuser, are not well understood. Therefore, the procedure is first to evaluate the time-averaged conditions of radial and axial thrust. Then, in the area of unsteady flow, the concept of rotating damping, forces due to perturbations of the impeller channel flow, and the impeller/volute or impeller/ diffuser interaction must be considered.

4.5 Flow Instability

Flow instability in a pump system is created by the interaction of an unstable pump head-capacity curve with a system that has unstable characteristics. Power plant feedwater systems have the requisite unstable characteristics, and a stable pump head-capacity curve is necessary for successful operation of the system. Pump designers use empirical methods to produce the needed pump characteristics, but basic knowledge of this subject is incomplete.

4.6 Design Methods and Research by Manufacturers

Most pump manufacturers use design methods that are proprietary and are based on a large empirical data base. The bulk of their research is internally funded, proprietary, and if published, not until long after completion. Pump manufacturers tend not to perform basic research; therefore, this work is generally conducted by research laboratories and universities.

4.7 Feed Pump Specifications

Our survey of architect/engineers indicates that completion of the research and development program will help them in areas such as

the following: better definition of short-duration transient and upset low- and high-flow conditions, as opposed to long-duration operation at minimum or maximum pump flow; better quantification of required minimum flow, maximum flow, and NPSH; suction piping design to avoid undesirable inlet flow profiles; discharge piping design to eliminate discharge piping resonance and water hammer; specification of proper testing, especially cavitation testing; specifications to promote reliability; and proper specification of operating instrumentation.

5.0 RESEARCH PLAN

The Research Plan consists of two parts: Phase II - Research Program, and Phase III - Development of Design Methods. Phase II consists of Tasks 1 through 19 and Phase III of Tasks 20 through 30. An outline of the Research Plan is shown in Figure S-1. A summary of the Research Plan follows.

PHASE II: RESEARCH PROGRAM

HYDRAULICS

Task 1 - Off-Design Flow Studies

An extensive research program is necessary for a better understanding of pump hydraulics at off-design flow and the problems created by operation of feed pumps at off-design flow. An experimental study is needed of flow patterns, impeller/volute (or diffuser) force interactions, cavitation phenomena at impeller eye and discharge, and unsteady pressures and loads in a representative boiler feed pump impeller and volute (or diffuser) geometry. Included in this research will be determination, by flow visualization, of the nature and type of flow at off-design as a function of volute or diffuser-impeller spacing, measurement of impeller/volute (or diffuser) stiffness matrices at off-design flows, and a qualitative assessment of the effects of impeller/volute (or diffuser) geometry on cavitation onset. Transient local loads caused by volute/impeller interaction will be studied by using surface-pressure measurement with a sufficiently high frequency response. Further, special

RESEARCH PLAN

PHASE II - RESEARCH PROGRAM

HYDRAULICS

1. Off-Design Flow Studies
2. Velocity Field Measurements in a Pump Stage
3. Impeller/Volute Hydraulic Analysis
4. Hydraulic Instability

CAVITATION

5. Analytical Study of Cavitation Performance
6. Analytical Study of Cavitation Damage
7. Correlation Between Noise and Cavitation Damage
8. Correlation Between Type of Cavitation and Damage Rate
9. Direct Measurement of Cavitation Damage in Impellers

ROTOR DYNAMICS

10. Development of Advanced Rotor-Bearing Systems for Feed Water Pumps
11. Investigation of Axial Balancing Devices
12. Seal Ring Coefficients
13. Wear Ring Pressure Losses

PUMP SYSTEMS

14. System Interactions
15. Suction Effects on Pump Performance

FULL-SCALE VERIFICATION

16. Full-Scale Tests
17. Shaft Seals

APPLICATION OF RESEARCH RESULTS

18. Pump Component Development
19. Economic Evaluation Methods

PHASE III - DEVELOPMENT OF DESIGN METHODS

COMPUTER-AIDED DESIGN METHODS

20. Hydraulic Analysis
21. Cavitation Analysis
22. Analysis of Annular Pressure-Reducing Devices
23. Stress Analysis of Impeller Vanes, Impeller Shrouds, and Shaft
24. Stress Analysis of Volute Lips and Diffuser Vane Tips
25. Rotordynamic Analysis

PUMP SPECIFICATION GUIDELINES

26. Instrumentation for Monitoring and Trouble-Shooting Operating Feed Pumps
27. Selection of Materials of Construction
28. Review and Updating of Test Codes for Feed Pumps
29. Review and Updating of Standards for Feed Pump Applications

TROUBLESHOOTING AIDS

30. Troubleshooting Manual

FIGURE S-1. Outline of Research Plan

attention will be paid to these measurements at impeller and volute vane tips.

Task 2 - Velocity-Field Measurements in a Pump Stage

It is also necessary to include in the research program quantitative, three-dimensional measurements of flow patterns in a representative boiler feed pump impeller and volute (or diffuser) geometry. This work is expected to provide a means of explaining the origin of the unsteady impeller forces by observing the unsteady velocity and pressure patterns. A laser-doppler velocimeter will be utilized for this task.

Task 3 - Impeller/Volute Hydraulic Analysis

A finite element computer program for boiler feed pump hydraulic analysis will be developed. This program will have three-dimensional analysis capabilities, utilizing steady and unsteady algorithms, and will enable modeling of nonsymmetric volute or diffuser geometry. Classical assumptions regarding the blade exit condition will not be used. A closure condition based on true viscous modeling will be employed instead.

Task 4 - Hydraulic Instability

Also needed is an analytical and experimental study of the onset of recirculation in the eye and at the discharge of a representative boiler feed pump impeller. It is probable that large, unsteady forces are related to recirculation in boiler feed pump impellers and that these forces create hydraulic instability.

CAVITATION

Task 5 - Analytical Study of Cavitation Performance

The development of an analytical method for predicting the performance of cavitating feed pumps is required. An advanced theory for cavity flow in impellers will be developed based on a coordinate transformation in which cavity zones can be explicitly recognized.

This theory must then be correlated to test data showing the effect of advanced cavitation on the deterioration of head and efficiency in a centrifugal pump stage. The analytical results depicting the extent of cavitating volume at any operating condition will be an important input to the development of damage rate prediction.

Task 6 - Analytical Study of Cavitation Damage

An analytical study of cavitation damage is required. A sound theoretical basis for the determination of cavitation damage rate will be established based on dimensional analysis of bubble dynamics, collapse phenomena, and material properties. This will be correlated with the experimental results from the work to be described as Tasks 7, 8, and 9.

Task 7 - Correlation Between Noise and Cavitation Damage

An experimental correlation between noise and cavitation damage rate will be made by using advanced acoustic instrumentation in experimental facilities where cavitation damage rates can be accurately controlled and measured. Any noise correlation that emerges would be an extremely valuable tool for the diagnosis of an operating feed pump and for the possible prevention of cavitation damage.

Task 8 - Correlation Between Type of Cavitation and Damage Rate

Two or more distinct types of cavitation are known to exist. An experimental correlation between type of cavitation and cavitation damage rate is necessary for understanding the cavitation damage problem. These correlations will be attempted by inducing impellers to undergo different types of cavitation in an experimental pump loop.

Task 9 - Direct Measurement of Cavitation Damage in Impellers

An experimental program involving direct measurements of cavitation damage in an impeller eye is necessary to correlate the results of all of the other cavitation research. Methods under consideration

are soft-coating removal, local use of soft metallic material, and use of pressure-sensitive film.

ROTORDYNAMICS

Task 10 - Development of Advanced Rotor-Bearing Systems for Feedwater Pumps

Development of advanced rotor-bearing systems for feedwater pumps is now being sponsored by EPRI projects RP-1266-7 and RP-1884-4. Dr. Maurice L. Adams, Jr., and Dr. Elemer Makay are the principal investigators. This is an experimental study of wear ring force coefficients aimed at determination of coefficients for existing common wear ring geometries and at devising wear ring geometries with better damping coefficients. We recommend an extension of this work to include testing of geometries that represent long interstage bushings, balancing drums, and throttle bushings. Another portion of this work deals with the development of a squeeze-film damper support for feed pump journal bearings.

Task 11 - Investigation of Axial Balancing Devices

Investigation of axial balancing devices is also a necessity. An additional part of the research mentioned as Task 10 is the effort directed at attenuation of axial vibration. An axial squeeze-film damper thrust bearing support is suggested as a possible remedy to the problems caused by the extreme rigidity of balancing disks.

Task 12 - Seal Ring Coefficients

To support the experimental work described as Task 10 and to promote a better understanding of the problems involved and of the experimental results, an analytical study of seal ring coefficients is needed. Perturbation solution techniques will be employed to define force components acting on the rotor due to small motion relative to a centered position in the wear ring or seal. Documentation will cover both the theoretical development of the seal force coefficients and calculated results for a range of currently employed seal geometries and operating conditions.

Task 13 - Wear Ring Pressure Losses

An experimental investigation of pressure-loss coefficients in wear rings and other annular pressure-reducing devices is required in relation to the above research work. It is also needed as an input to one or more of the computer-aided design methods of Phase III. Measurements will be made in an experimental facility where both concentric and eccentric shaft/wear ring configurations can be accommodated. Data will be accumulated for a broad range of Reynolds numbers.

PUMP SYSTEMS

Task 14 - System Interaction

A study of system interaction between the feed pump, the piping, and the feedwater control system is required. To a lesser degree, this study will also consider such structural components as baseplates, foundations, etc. It will include such items as feed pump vibration caused by piping resonance excited by pressure pulsations from the pump, water hammer, determination of maximum allowable pressure pulsations, sources of pump vibration caused by external excitations, interrelationships between the pump and the feedwater control system, and feed pump minimum flow bypass systems.

Task 15 - Suction Effects on Pump Performance

Also required is an investigation of the effects of suction piping configuration and of suction transients on pump performance. This will include analytical and experimental investigations of flow profiles at feed pump inlets and their effect on performance. A definition and quantification of the feed pump inlet transient conditions created by a deaerator transient, a booster pump failure, or by other upset conditions will be included. Experimental investigation of pump first-stage performance during simulated upset suction transients will also be undertaken.

FULL-SCALE VERIFICATION

Task 16 - Full-Scale Tests

Full-scale, full-speed tests of boiler feed pump stages will provide verification of laboratory-scale research. All of the experimental research indicated above (Tasks 1 through 15) will be conducted on scale models. Even though the theory developed would transcend the size effects, it is quite likely that certain obscure but significant scale effects will remain undiscovered. This is particularly true for cavitation-related work. Consequently, full-scale testing is necessary to verify the theories, unearth the obscure mechanisms, and thus close the loop between theoretical and experimental research.

Task 17 - Shaft Seals

Because shaft seal failures have been identified as the most frequent cause of feed pump outages, a full-scale experimental investigation of shaft seals and shaft seal systems for feed water pumps will be undertaken. A more efficient test program will result if the shaft seal tests are separated from the boiler feed pump stage tests described as Task 16.

APPLICATION OF RESEARCH RESULTS

Task 18 - Pump Component Development

Consolidation of the above research results will lead to and give guidance to pump component development. For example, the results of the research will lead to improved designs for wear rings and other annular pressure-reducing devices and to improved impeller design and improved understanding of the various design relationships between impellers and volutes or diffusers. Full-scale development of such devices as squeeze-film radial bearing supports and axial squeeze-film damper thrust bearing support is also likely. Some pump component development will involve large-scale testers, while other developments will utilize the hardware from the full-scale verification testing.

Task 19 - Economic Evaluation Methods

To complete the research phase of the program, a systematic methodology for evaluating pump design elements and performance in terms of total economics is required. The purpose is to provide a basis for engineering and economic tradeoffs between investment and operating costs related to feed pumps.

PHASE III: DEVELOPMENT OF DESIGN METHODS

COMPUTER-AIDED DESIGN METHODS

Task 20 - Hydraulic Analysis

The research results must be translated into design methods that can be utilized by feed pump designers and others. A hydraulic analysis based on the results of the research outlined as Tasks 1 through 4 is required.

Task 21 - Cavitation Analysis

A cavitation analysis of annular pressure-reducing devices based on the research outlined as Tasks 5 through 9 is required.

Task 22 - Analysis of Pressure-Reducing Devices

An analysis of annular pressure-reducing devices based on the research outlined as Tasks 10 through 13 is needed.

Task 23 - Stress Analysis of Impeller Vanes, Impeller Shrouds, and Shaft

A stress analysis of impeller vanes, impeller shrouds, and the pump shaft, taking into account the fatigue stresses caused by unsteady pressures and forces as well as the centrifugal forces on the vanes and shrouds, is necessary.

Task 24 - Stress Analysis of Volute Lips and Diffuser Vane Tips

It is also necessary to perform a stress analysis of the volute lips or diffuser vane tips and the fatigue stresses imposed by unsteady pressure loads.

Task 25 - Rotordynamic Analysis

A rotordynamic analysis that accounts for the stiffness, damping, and virtual mass coefficients of the impellers and the wear rings and other annular pressure-reducing devices is needed. The stiffness and damping coefficients of radial bearings, which may be of an advanced design, must also be known and integrated into the program.

PUMP SPECIFICATION GUIDELINES

Task 26 - Instrumentation for Monitoring and Troubleshooting Operating Feed Pumps

Comprehensive pump specification guidelines will be published as an aid to architect/engineers and users. These guidelines will present the results of the research in a practical and usable manner. A publication on instrumentation for monitoring and troubleshooting operating feed pumps will be required.

Task 27 - Selection of Materials of Construction

Practices for the selection of materials of construction for large feed pumps will be reviewed and updated.

Task 28 - Review and Updating of Test Codes for Feed Pumps

Test codes for feed pumps will also be reviewed and updated.

Task 29 - Review and Updating of Standards for Feed Pump Applications

A review and updating of standards for feed pump applications in view of the completed research, incorporating its results, will be provided.

TROUBLESHOOTING AIDS

Task 30 - Troubleshooting Manual

The research results should provide assistance in troubleshooting of feed pumps. A troubleshooting manual will be prepared. The purpose of this manual will be to aid in the identification, diagnosis, definition, and resolution of problems experienced by existing feed pumps. Included will be methods, analytical tools, and experimental techniques that will be useful in the prevention of problems or their solution when they occur.

6.0 CONCLUSIONS AND RECOMMENDATIONS

The great cost to the utility industry of feed pump problems and the resulting plant outages can be reduced by implementation of the knowledge obtained from a coordinated, large-scale research program. The recommended research program consists of 30 interrelated tasks. It is designed to perform the needed research, verify the results, develop improved components, and publish computer-aided design methods, pump specification guidelines, and a troubleshooting manual.

The estimated benefit/cost ratio for the entire research and development program is 154:1 and is more than 100:1 for each research category.

This program should be started as soon as possible to achieve maximum savings to the utility industry.

7.0 REPORT ORGANIZATION

Volume 1 of this report consists of seven sections. The first four sections outline and update feed pump procurement and design methods, problems of feed pumps, assessment of experimental research, and review of pump theory. These topics are covered in Volume 1 in sufficient detail to support the last three sections which cover feed pump research requirements, the Research Plan, and the determination of benefits.

Volume 2 contains appendices covering the subject matter of the first four sections of Volume 1 in much greater detail. Included are: tabulated and summarized responses, in detail, to the surveys of architect/engineers, utilities, independent research laboratories, and pump manufacturers; a Pro Forma Technical Specification; and considerable detail relating to current experimental research and literature on such topics as hydraulics, vibration in pumps, cavitation, two-phase flow, impeller forces, and flow instability. References and bibliographies are included.

Section 1
FEED PUMP PROCUREMENT AND DESIGN METHODS

1.0 BYRON JACKSON/EPRI SURVEY OF ARCHITECT/ENGINEERS

Six architect/engineers (A/E's) were surveyed. Selection criteria were:

1. The architect/engineering firm must be a leader in the design and construction of large fossil-fired and nuclear power plants.
2. The engineers interviewed must personally be well qualified to answer questions relating to large feed pumps.

There are other A/E's who fulfill these criteria. Recent specifications from these other A/E's were examined, and a significant amount of the information gained from these examinations was incorporated into our Pro Forma Technical Specification (paragraph 2.0 of this section and Appendix B). We were unable to conduct more interviews. However, based on the examination of the specifications of the other qualified A/E's, we believe that our decisions relating to the design of a viable Research Program would not have been influenced to a significant degree by additional interviews.

The detailed "Tabulated and Summarized Responses to Byron Jackson/EPRI Survey of Architect/Engineers" is in Appendix A.

The first question relates to the expected sizes of fossil-fired plants. A summary response was that most fossil-fired plants to be built in the next ten years will range from 300 or 400 MWE to 800 or 1,000 MWE in size. TVA and AEP were mentioned as exceptions. They have built and are expected to build more 1,300-MWE plants. Many responders believe that the trend is toward smaller (400-MWE to 500-MWE) plants. The probability of new supercritical plants is uncertain.

The survey disclosed that the ultimate boiler feed pump, as defined by flow and horsepower, has already been built. The 21,000-gpm, 11,000-foot TDH, 4,600-psi discharge pressure, 65,000-horsepower feed pumps used by AEP in their 1,300-MWE plants are expected to be the maximum size that will ever be required for fossil-fired plants. Higher head and discharge pressure requirements, up to 5,500 psi, are being considered but at lower flow and horsepower. Except for the above, demand for pumps larger than 16,000 to 25,000 horsepower for the next ten years, and ultimately for pumps larger than 25,000 to 45,000 horsepower, will be minimal.

Only one A/E distinguishes between the operating range (minimum flow to maximum flow) for fossil plant boiler feed pumps during normal operation and the operating range for short periods of time during abnormal operation.

FOR BOILER FEED PUMPS, BETTER DEFINITION IS NEEDED OF SHORT-DURATION TRANSIENT AND UPSET LOW- AND HIGH-FLOW CONDITIONS, AS OPPOSED TO LONG-DURATION OPERATION AT MINIMUM OR MAXIMUM PUMP FLOW.

Most A/E's quote minimum flow as 25%.

One response seems to have great merit. It was pointed out that the fuel system in the vast majority of existing and projected fossil-fired plants becomes unstable below 20% load, and operation below 20% load should really not be a significant consideration. The vast majority of large plants normally operate at full load with two boiler feed pumps in parallel. If one pump is shut down when plant load is reduced, then the remaining pump will be at approximately 40% of its design flow at 20% plant load. Except for the minor added cost of the recirculating line piping and valves, there is no justification for a minimum boiler feed pump flow of less than approximately 40% of design flow with this common plant design.

SHOULD MINIMUM LONG-DURATION BOILER FEED PUMP FLOW BE DEFINED AS THE FLOW REQUIRED AT MINIMUM PRACTICAL PLANT LOAD FOR FOSSIL-FIRED PLANTS?

Required maximum flow for feed pumps in fossil plants ranged from 115% of design flow to 140% of design flow.

BETTER DEFINITION OF REQUIRED MAXIMUM BOILER FEED PUMP FLOW IS NEEDED.

For fossil-fired plants, all responders prefer to operate at design load, with two 50% or 60% turbine-driven feed pumps. In addition, a motor-driven start-up pump is installed and is used as a spare when a turbine-driven pump is out of service. The size of the third pump varies from 25% to 50% (20% when the turbine-driven pumps are 60%). There is a difference of opinion relating to the justification of the cost of a variable-speed electric drive for the third pump.

The obvious advantage of having the third pump and its spare parts identical to the 50% turbine-driven pumps is often outweighed by limitations on the size and cost of available electric drive systems. For a given power plant, the electric drive system for a 5,500-rpm variable-speed 50% pump is far more costly than that for a 3,580-rpm constant-speed 25% pump. Also, the 50% pump may require a booster, while the 25% pump could take suction from the deaerator.

Note that a 50% pump plus a 25% pump, both at 133% of pump design flow and at a slightly increased speed, will give 100% feedwater flow to the plant. This relates to the question of maximum flow for long periods of time and why it should ever exceed 133%.

THE PROPER SELECTION OF THE SIZE OF THE START-UP/SPARE BOILER FEED PUMP REQUIRES FURTHER STUDY.

All responders expressed doubt that any new nuclear power plant commitments will be made within the next ten years for construction in the United States. New nuclear plants are expected to appear in foreign countries for many years.

Most responders expect that the range of sizes for new nuclear plants to be built in the next ten years will be 900 to 1,300 MWE. In addition, 600-MWE plants will be built in developing countries.

The largest nuclear feed pumps anticipated within the next ten years would operate at 23,000 gpm, 2,600 feet head, 1,400-psi discharge pressure, and 18,000 horsepower. These figures combine booster pumps

and feed pumps.

The largest anticipated nuclear feed pump would operate at 40,000 gpm, 2,100 feet head, 1,600-psi discharge pressure, and 24,000 horsepower. These figures are for the feed pump only.

Again, only one A/E distinguishes between the operating range (minimum flow to maximum flow) for nuclear feed pumps during normal operation and the operating range for short periods of time during abnormal operation.

FOR NUCLEAR FEED PUMPS, BETTER DEFINITION IS NEEDED OF SHORT-DURATION TRANSIENT AND UPSET AND LOW- AND HIGH-FLOW CONDITIONS, AS OPPOSED TO EXTENDED OPERATION AT MINIMUM OR MAXIMUM PUMP FLOW.

One responder quoted minimum flow as 10% to 25% of design flow. The other responses varied from 25% to 50%.

BETTER DEFINITION OF REQUIRED MINIMUM FLOW FOR NUCLEAR FEED PUMPS IS NEEDED. IS IT DIFFERENT FOR PWR and BWR PLANTS?

Required maximum flow for feed pumps in nuclear plants ranged from 115% to 160%. The majority of responders require 130% or more.

A BETTER DEFINITION OF REQUIRED MAXIMUM FLOW FOR NUCLEAR FEED PUMPS IS NEEDED. ARE THE REQUIREMENTS FOR PWR PLANTS DIFFERENT FROM THOSE FOR BWR PLANTS?

For nuclear plants, all responders prefer two 50% turbine-driven feed pumps for operation at design load. There is slightly more desire for the third pump also to be 50% than was indicated for fossil plants. However, one A/E indicated that 25% to 35% spare is considered adequate, one A/E does not install any spare feed pumps at all, and one A/E does not install any spare feed pumps in PWR plants.

One A/E indicated, as an alternative, four 33% feed pumps. An engineer involved with operation of a nuclear plant (not an A/E) said

that if a 50% feed pump shuts down, the resulting transient makes it difficult to keep the plant operating. He would like to see more and smaller pumps.

THE PROPER SELECTION OF THE SIZE OF THE SPARE NUCLEAR FEED PUMP REQUIRES FURTHER STUDY.

There were wide variations in the responses relating to considerations governing NPSH available to feed pumps.

It was agreed that NPSH is furnished to nuclear feed pumps by booster condensate pumps in the majority of current plant designs. There is no agreement, however, on the question of whether 50% capacity boiler feed pumps in large fossil-fired plants should take suction from an elevated deaerator or should be provided with booster pumps.

There is generally a preference that the 20% to 33% capacity motor-driven boiler feed pumps should take suction from the deaerator and should not have booster pumps.

Half of the responders mentioned NPSH margin. Some spoke of desired margins and some of required margins. Some were concerned with margin at design flow, some with margin at runout flow, some with both. Desired and required margins varied from 80% to 100% at design flow and from 25% to 50% at runout flow. Desired or required margins in some cases were tied to a specific NPSH test method, such as 0% head drop on cold water.

THE SUBJECT OF NPSH MARGIN REQUIRES FURTHER CLARIFICATION.

Feed pump foundation design and piping design as related to suction and discharge nozzle piping loads do not seem to be problems for any of the responders.

Most responders try to provide at least five pipe diameters of straight pipe into the pump suction. Several have unpublished internal specifications governing such things as allowable velocities, specification of long-radius elbows, and avoidance of elbows at a right angle to the plane of rotation of the first-stage impeller

unless additional straight-run pipe is provided.

ADDITIONAL AND MORE SPECIFIC INFORMATION ON THE DESIGN OF SUCTION PIPING TO PROVIDE UNIFORM VELOCITY DISTRIBUTION TO THE PUMP SUCTION WOULD BE WELCOMED BY MANY RESPONDERS.

Most responders believe that discharge piping resonance at vane-passing frequency is not a common problem and, generally, no analysis of it is performed during plant design. Most responders feel that the practical and economical approach is to attend to the few problems as they occur.

The responders do not consider water hammer in feedwater systems to be a pump-related problem.

ALTHOUGH DISCHARGE PIPING RESONANCE AND WATER HAMMER ARE NOT COMMON PROBLEMS, WE BELIEVE THAT FURTHER STUDY OF THESE TOPICS SHOULD BE CONDUCTED.

Seismic requirements imposed on feed pumps, if any, are minor.

No responders are presently requiring dry run of boiler feed pumps. One requires a severe low NPSH test, and others are considering similar requirements.

Only one responder requires a dry run of nuclear feed pumps. Some are considering severe low NPSH testing.

SEVERE LOW NPSH TESTING TO SIMULATE PUMP SUCTION UPSETS AND TRANSIENTS SHOULD BE STUDIED.

Attemperation flows from 5% to 20% of design flow were anticipated by various responders. This is a very wide range of opinions.

Several people mentioned that some coals to be used in the future may cause worse slag formation and fouling. This may increase attemperation requirements.

No re-entry designs are anticipated at this time by the responders.

Almost all responders indicated to some degree that they do not like the concept of giving great weight to quoted pump efficiencies when evaluating pump proposals, except to assess a penalty if the quoted pump is a poor "fit." One will not give credit for any quoted efficiency above 85%. Several others would like to do something similar.

METHODS FOR THE EVALUATION OF QUOTED PUMP EFFICIENCIES SHOULD BE INVESTIGATED.

Three responders evaluate minimum flow and three do not. One evaluates piping cost only, one evaluates power cost only, and one evaluates both.

THE SUBJECT OF MINIMUM FLOW HAS ALREADY BEEN IDENTIFIED AS ONE FOR FURTHER INVESTIGATION.

Anticipated feed pump reliability is evaluated, one way or another, by all responders. There is an increasing desire to do a more formal analysis and, possibly, to credit or penalize pump proposals heavily based on the results of such formal analysis. However, the present data base is inadequate for this purpose.

THE FORMAL ANALYSIS OF FEED PUMP RELIABILITY SHOULD HAVE FURTHER INVESTIGATION.

None of the responders makes a formal analysis of each manufacturer's past service record. However, this item is almost always taken into account, one way or another.

Only one responder evaluates shut-off head. Three responders specify shut-off head not to exceed 130% of design head.

Maximum head per stage, maximum number of stages, or maximum suction specific speed are seldom specified. All responders were reluctant to become involved in pump design.

Maximum rpm is almost always specified, often because of speed limitations imposed by available steam-turbine drives.

Most responders specify instrumentation to monitor both vibration and radial shaft motion (orbit). Instrumentation to monitor axial shaft motion (end float) is specified by three responders for nuclear feed pumps and by two for boiler feed pumps. Only one monitors suction and discharge pressure pulsations, and two specify instrumentation to check hot alignment.

SPECIFICATION OF PROPER OPERATING INSTRUMENTATION SHOULD BE FURTHER INVESTIGATED.

No responder is aware of proven instrumentation that will detect rubbing of internal components in operating feed pumps.

No responder is aware of proven instrumentation that will detect the onset of cavitation in an operating feed pump. Two responders mentioned instrumentation to measure available NPSH.

FURTHER INVESTIGATION OF INSTRUMENTATION FOR CAVITATION DETECTION AND/OR MEASUREMENT OF AVAILABLE NPSH IS IN ORDER.

Four responders believe that analysis of fluid-borne noise can be a valuable tool for monitoring an operating feed pump or for diagnosis of feed pump problems. One responder believes that analysis of air-borne noise can be valuable.

Specification of the throttle bushing type of shaft seals with injection and bleed-off is common. Most prefer temperature control of the injection system. None will allow mechanical seals.

THE SUBJECT OF SHAFT SEALS AND RELATED INJECTION SYSTEMS STILL REQUIRES FURTHER STUDY.

A stable head-capacity curve is specified, and the pumps are witness tested by all responders.

There is a wide variation in specified shop tests and a general feeling that existing test codes are inadequate, at least to some degree. Three responders are not satisfied with vibration standards, two

would like to have pressure pulsation acceptance criteria, and one believes that NPSH tests are not accurate. There was, however, general agreement that the best way to resolve any existing problems is by revision of the Hydraulic Institute Test Code, not by having different requirements for each A/E.

THE QUESTION OF WHAT CONSTITUTES A PROPER ACCEPTANCE TEST FOR A LARGE FEED PUMP REMAINS OPEN.

The right to run field tests is always specified but seldom exercised.

Comments on previously published EPRI reports relating to feed pumps were generally favorable.

2.0 PRO FORMA SPECIFICATION

Appendix B contains the "Pro Forma Technical Specification for Turbine-Driven Boiler Feed Pumps for Large Fossil-Fired Power Plants." Most feed pumps to be purchased during the next ten years will be in this category. A modified specification would be required for the start-up or spare motor-driven pump for a fossil-fired plant or the feed pumps for a nuclear plant. This specification is intended to be typical and illustrative.

This technical specification covers the requirements for the design, manufacture, shop testing, packaging, storage and handling, shipping and technical direction of installation and field testing of feed pumps and accessories for large (600-MWE and larger) fossil-fired power plants.

The pro forma specification is divided into three sections. The body of the specification is intended to be general in nature and to apply to all feed pumps of a given category. Table 1 contains the project requirements that result from plant design considerations, particular preferences of A/E's, and the requirements and preferences of the utility. Addendum 1 is a definition of terms. Addendum 2 is a Technical Bid Form to be filled out by the A/E and the pump manufacturer.

3.0 REVIEW OF THE CURRENT STATE OF THE ART OF FEED-PUMP PERFORMANCE PREDICTION

Only two requirements have to be met to predict the performance of a turbomachine. One is to predict, at the combination of fluid velocity and blade velocity of interest, the outlet flow angle of each streamline flowing through each row of blades. Using Newton's law in the form of the Euler equation, the specific work or enthalpy rise of the fluid along each streamline can then be found.

To predict the head or pressure rise from the specific work or enthalpy rise, the second component of the prediction is required: the losses. The pressure losses occurring at the fluid boundaries and in the body of the fluid will be needed to predict the increase of pressure, and the friction losses occurring because of windage, seal, and bearing friction will be needed to predict the power input required on the shaft.

The traditional design method has been to specify a combination of the impeller vane discharge angle and the number of vanes, to assume a stage efficiency (which means assuming the losses that will occur at best-efficiency point), to refer to previous correlations or experimental results to predict the impeller discharge flow angle (alternatively the deviation or "slip factor"), and thereby to find the impeller peripheral velocity that will give the desired head rise per stage.

The actual impeller diameter and rotational speed at best efficiency will depend on the driver. If the pump is to be turbine driven, the shaft speed can usually be chosen to give an optimum specific speed for a multistage arrangement, taking into account the need to avoid shaft instabilities. If the pump is to be driven by an induction motor running on a fixed frequency, then some compromise in specific speed may be necessary.

Prediction of off-design performance is made from correlations of previous test results. Some correlations, for instance of slip factor, are available in the literature. However, all manufacturers have their own correlations, based on tests of impellers and pumps of their

own manufacture, and very accurate predictions can be achieved in this way.

There is considerable effort in many places to predict pump performance from first principles, using computational fluid mechanics. A principal purpose is to improve the shapes of the impeller blades and of the hub and shroud, so as to be able to produce either a higher design-point efficiency or to ease manufacturing complexity or, preferably, both. It is also hoped that flow patterns at low-flow conditions could be predicted, and measures could be taken that could increase the efficiency, decrease the incremental pressure rise, and/or make for smoother operation in these low-flow conditions.

The state of uncertainty of predictions given by analytical and even empirical methods is illustrated by the fact that some manufacturers still use models to back up their design work. The results of model tests need correction factors before they can be used to predict full-scale performance. However, the correlations of correction factors are well known.

4.0 REVIEW OF THE FEED-PUMP RESEARCH AND DEVELOPMENT

This short review is based solely on the visits and discussions of Professor David Wilson of M.I.T.

The only work specifically directed at boiler-feed pumps was being undertaken in the laboratories of manufacturers of boiler-feed pumps. The manufacturers to which this reviewer made visits or with whom he had discussions were naturally unwilling to reveal all details of what they were involved in, to avoid the transfer of valuable proprietary information.

Very little work was apparently being carried out on hydraulic forces and instabilities. In several independent centers the principal activity was in computational fluid mechanics as applied to centrifugal pumps and compressors. Research and development into cavitation was being carried out with models running at full-scale tip speeds. The use of soft coatings, such as ink and aluminum, was considered to be a successful approach for the evaluation of

cavitation.

Appendix C contains brief notes of the responses received from, first, the independent research laboratories and, subsequently, from manufacturers.

Section 2

PROBLEMS OF FEED PUMPS

1.0 BYRON JACKSON/EPRI SURVEY OF UTILITIES

The detailed "Tabulated and Summarized Responses to Byron Jackson/EPRI Survey of Utilities" is in Appendix D.

In reply to the questions regarding major problems in large boiler feed pumps, failure modes mentioned included balancing device failures, cavitation damage, impeller vane and shroud failure (other than by cavitation), seizures, shaft failures, lack of reliability of auxiliary systems (especially lube-oil systems and bleed-off piping systems), gear-type coupling inflexibility, and hot misalignment.

Failure of balancing devices is frequently due to system upsets, especially emergency trip conditions. Vibration, affecting seals and bearings, is often due to hot misalignment after maintenance. Cavitation damage is mostly due to prolonged service at off-design flow. Seizures often are caused by bypass flow system problems and/or improper start-up/warm-up procedures, mostly at start-up after maintenance.

The corrective actions that have improved reliability include upgrading of operator training, operating instructions, feedwater system maintenance, and instrumentation.

It seems that, in general, failures at "mid-life" are rare. The majority of failures are during "old age" and probably could have been avoided by periodic maintenance, or they occur shortly after inadequate or improper repairs.

With regard to failures in nuclear plants, responders mentioned cavitation on one side of a double-suction impeller, malfunction of the control system, internal wear, wrong shaft material, galling, recirculation line vibration, unreliable floating ring seals, inflexible gear-type couplings, and recirculation system problems.

It is believed that all nuclear plants are in base load service.

All responded that instrumentation is used in the following areas to monitor boiler-feed and nuclear-feed pump performance: discharge and suction pressure, flow, water temperature, bearing vibration, and rpm.

There is no uniformity in scheduling of feed pump maintenance. One responder said that maintenance was performed "every six years or every four fuel cycles." Others replied that they depend upon hydraulic and mechanical monitoring to alert them to the need for maintenance. One said their base load plants are scheduled for two-week outages every four to six months and that repairs are made at those times. One makes an annual check of bearings, seals, and balancing devices; another maintains the pump at the time of turbine inspection, otherwise only in case of failure. The replies stated that maintenance scheduling could be improved by better instrumentation and better records.

The request for comments on EPRI publications CS-1512, NP-1571, and FP-754 elicited positive responses for the most part.

A list of feed pump problems identified by the Byron Jackson/EPRI survey of utilities is as follows:

1. BALANCING DEVICE FAILURES
2. HOT MISALIGNMENT
3. FIRST-STAGE CAVITATION
4. SEIZURES
5. FAILURE OF INTERSTAGE PARTITIONS
6. CAVITATION DAMAGE
7. IMPELLER CRACKING OR BREAKAGE
8. INTERNAL LEAKAGE AND WASHOUT
9. GEAR-TYPE COUPLINGS
10. SHAFT FAILURES
11. VIBRATION
12. AUXILIARY SYSTEM RELIABILITY

The above list is not necessarily in order of severity or frequency of the problems.

2.0 MAJOR OUTAGE-PRODUCING FAILURE CAUSES IDENTIFIED IN EPRI FP-754, "SURVEY OF FEED PUMP OUTAGES"

The ten major outage-producing failure causes identified in EPRI FP-754 are:

1. SHAFT SEAL FAILURE
2. VIBRATION: PUMP, PIPING, FOUNDATION
3. AXIAL BALANCING DEVICE FAILURE
4. JOURNAL BEARING FAILURE
5. CAVITATION DAMAGE
6. IMPELLER BREAKAGE OR CRACKING
7. RAPID WEAR OF WEAR RINGS
8. UNSTABLE HEAD CURVE
9. BROKEN OR DAMAGED SHAFT
10. THRUST BEARING FAILURE

The above items are listed in the order of total number of failures, number 1 indicating the most frequent type of failure.

3.0 PROBLEMS IDENTIFIED IN EPRI NP-1571, "REPETITIVE FAILURE CAUSES FOR FEEDWATER PUMPS"

A list of feed pump problems identified in EPRI NP-1571 is as follows:

1. PROCUREMENT PRACTICES SUCH AS EMPHASIS ON HIGH QUOTED EFFICIENCY, LOW COST, LOW REQUIRED NPSH AND LOW MINIMUM FLOW
2. INCOMPLETE OR INCORRECT INFORMATION GIVEN TO THE VENDORS
3. COMPREHENSIVE TESTING NOT SPECIFIED OR NOT ENFORCED
4. INADEQUATE SPARE ROTATING ELEMENTS AND SPARE PARTS
5. FEEDWATER SYSTEM DESIGN
6. LACK OF SPARE PUMP ON STANDBY
7. FALSE AUTOMATIC FEED TRIPS IN NUCLEAR PLANTS
8. POOR OPERATING PROCEDURES
9. OPERATION ON TURBINE TURNING GEAR
10. FAILURE TO MONITOR AND EVALUATE PUMP OPERATING PARAMETERS

11. LACK OF CONTROL OVER PUMP REPAIRS
12. LACK OF COMMUNICATION AMONG PEOPLE FROM UTILITIES,
ARCHITECT/ENGINEERS, VENDOR SERVICE, AND VENDOR DESIGN
ENGINEERS
13. GENERIC PROBLEMS OF PUMP DESIGN, SUCH AS INTERNAL HIGH-
PRESSURE SEAL WASHOUT, AXIAL BALANCING DEVICE FAILURES,
AND CAVITATION DAMAGE
14. INADEQUATE RESPONSE BY VENDOR SERVICE PEOPLE
15. INADEQUATE TECHNICAL SERVICE MANUALS

The above list is not necessarily in order of severity or frequency of the problems.

Section 3

ASSESSMENT OF FEED PUMP EXPERIMENTAL RESEARCH

1.0 Rotordynamics Testing

It is becoming increasingly clear that one of the major factors affecting the vibration response of centrifugal boiler feed pumps is the stiffening effect of various close-clearance spaces within the pump. Those include annular pressure-reducing devices, such as wear rings, throttle bushings, center-stage pieces, balance drums, etc. These devices play an important part by providing not only a stiffening effect but also a damping effect. In many instances the fluid in the clearance spaces also possesses a virtual inertia, also known as virtual mass, and this too plays a part in determining the vibration response of the system.

Experimental work so far can be broadly classified in two major categories. In the first the shaft is allowed to undergo whirling motion. In the second it is made very stiff, and the forces at the wear rings, etc., are directly measured. The advantage of the first type of test is that one can immediately detect the presence of a critical speed, which is a matter of great practical significance. The difficulty associated with this type of test is that understanding why a particular critical speed is observed is dependent upon the mathematical modeling made for the entire system. Consequently, there is no direct substantiation of the effect of the annular clearances. In the second set of tests, which uses a stiff shaft, the forces are directly measured and, therefore, the sealing ring properties are immediately and directly established.

Recent tests done at Byron Jackson use the flexible-shaft approach. As shown in Figure 3-1, a single-stage, double-suction, horizontal pump was installed in a test rig, wear rings were provided on either side, and the pump was run at variable speed. The pressure difference across the wear rings was changed independently of the speed, and the vibration of the shaft was measured by using two proximity probes.

FIGURE 3-1. Byron Jackson Test Rig

When the tests were first conducted in air, the critical speed was clearly observed as shown in Figure 3-2, which plots the once-per-revolution amplitude versus the speed. It also shows that the phase shift of nearly 180° is observed as well. When water was introduced into the pump, the amplitude response characteristics changed dramatically. For example, Figure 3-3 shows that when smooth wear rings were used with a nominal clearance of .016 inch (diametral clearance) and there was a pressure difference across the rings of 250 psi, no discernable critical speed was evident; also there was scarcely any phase shift. In this case, the critical speed appears to be totally suppressed; the fact that it is can be easily be demonstrated by calculations. However, when the wear ring clearance is opened up or when the wear ring is deeply serrated, critical speed phenomena begin to appear again. It is also interesting to note that the effect of wear on the rings is to reduce the critical speed, particularly for rings that are smooth; this can be seen in Figure 3-4. It shows that as the diametral clearance of the seal rings increases, the stiffening effect is reduced and, therefore, the critical speed decreases. The effect, however, is much weaker for grooved rings, because they produce less stiffness.

Using such experiments, one can determine the performance of the sealing rings. However, since several assumptions are made with regard to the modeling of the system, it is quite conceivable that the indirect measurement of sealing ring properties, as above, could be considerably in error. Therefore the testing using a stiff shaft is more straightforward.

There are several methods by which a stiff-shaft test can be made to determine the sealing ring properties. EPRI is sponsoring stiff-shaft testing in which the journal is set with an offset and then vibrated by orbiting it accurately. In this research it is possible to vary the precession speed with respect to the rotating speed. This testing is being done by M. L. Adams and E. Makay. All twelve dynamic force coefficients, as described in Section 4, paragraph 2.3, of this report, can be determined accurately in this setup.

Recent theoretical work has shown that it will be possible to determine the sealing ring coefficients by means of theoretical

FIGURE 3-2. Air Test

FIGURE 3-3. Water Test

FIGURE 3-4. Critical Speed as a Function of Clearance for Smooth and Grooved Wear Rings

calculations. It will be important to be able to do these calculations. For the calculations to be accurate, the primary requirement is a full understanding of the pressure loss coefficients. They are threefold. One is the entry loss coefficient. The second is the exit loss, or the discharge loss coefficient. The third is the friction coefficient caused by the flow of fluid on the surface. Evidently the Reynolds numbers both for axial flow and circumferential flow play a part. It will be important in the near future to run tests statically to determine all of the pressure-loss coefficients and to use them in conjunction with the theory to predict the dynamic force coefficients of these rings. These predictions then can be compared with accurate test data.

2.0 ADVANCED INSTRUMENTATION IN HYDRAULICS

Recently a great deal of attention has been paid to the possibility of measuring velocity fields in impellers by direct, noninvasive measurement, using laser-doppler anemometry (LDA). Previously, attempts have been made to measure velocity and pressure field by using probes that rotate. However, it is a cumbersome method, and the amount of data obtained is limited. Recent work has shown that considerable improvement in resolution and speed of data taking can be gained by using a laser system. It will be appropriate to consider some of the available systems and how they can be applied to improve the understanding of the flow field in boiler feed pumps.

Two types of laser systems are used for determining the velocity field. The first is called the laser two-focus system. It is quite popular in Europe and has been used to determine the velocity field in advanced high-performance centrifugal compressors. In this method two laser beams are focused in a very small volume. As a particle travels between the two light beams, it produces two pulses of scattered light. The elapsed time between the pulses and the known beam separation yields the velocity component perpendicular to the optical axis.

The more common system is called the laser-doppler anemometer. Its principle is deduced from the Doppler effect, which is often encountered in everyday life. When a source emitting a wave is in motion

with respect to an observer, the frequency perceived by the observer is different from the frequency emitted by the source. The apparent frequency f' as perceived by the observer is different from the original frequency of the source, as shown in the equation below, where W represents the source velocity and c represents the speed of light.

$$f' = \frac{f}{1 - W/c}$$

The laser-doppler anemometer can be operated in three distinct modes. The first is called the reference beam system, in which the scattered light from one beam is mixed with the reference beam. The disadvantage of this system is that the collecting device has to be very carefully aligned with the low-intensity reference beam. The second method is called the differential doppler system, in which only a single beam is used to illuminate the particle and the scattered light is collected and mixed in two directions. The disadvantage of this system is that the signal-to-noise ratio is rather poor, and a very small aperture must be used because the doppler shift can vary across the field stop opening.

The most popular, and perhaps most convenient for work in centrifugal pumps, is the dual-beam system. Here two equal-intensity laser beams are formed by using a beam splitter and a mirror. Using a lens, the beams are focused at a point at which the flow velocity is to be measured. At the crossing volume a fringe pattern is formed; particles crossing this volume will then emit discrete pulses. Since there will be a Gaussian distribution of intensity across the measurement volume and since fringes are formed inside the volume, a typical signal at the photo detector produced by a particle crossing the fringe volume would resemble that shown in Figure 3-5. The low-frequency component can be eliminated by using a high-pass filter. Then the frequency of the pulses will be proportional to the component of particle velocity normal to the fringes divided by the fringe spacing. The fringe spacing depends only on the wavelength of the laser light and the beam crossing angle. Therefore, a fixed linear calibration relationship results, regardless of the angle at which the detecting device is located with respect to the laser beam. The advantages of this system are its noninvasive quality and its ability to measure velocity components instantaneously. Measurement

FIGURE 3-5. Details of Crossover Region

of two components is quite straightforward, achieved by rotating the fringe pattern by 90 degrees. The disadvantages of this system include the necessity for particles to be in the flow and the essentially intermittent nature of the signal that has to be processed. The fact that the particle velocity is measured, and not the fluid velocity, may be important in some instances.

A number of researchers have used such a system for the determination of the velocity field in water pumps. The investigator with the most direct approach used a standard centrifugal pump impeller placed in a volute. He was able to use the back-scattering mode of light detection, and some of his results are extremely intriguing.

Figures 3-6 and 3-7 show the radial velocity profiles measured at the discharge from the impeller in the vicinity of the volute. Near the design mass flow (Figure 3-6) the front station clearly shows a jet on the pressure side, followed by the wake of the blade, and finally high velocity on the suction side. The blade passage therefore exhibits, at least in part of the passage, an isentropic type of flow field with a gradient of velocity equal to twice the angular velocity. Some stations show a much larger wake, and there is even a small portion of reverse flow. Thus, even at design point, we find that for this particular impeller the flow field is extremely complex. At a mass flow of approximately 30% of the best efficiency point flow (Figure 3-7), it can be seen that the highest velocity actually occurs at the corner formed by pressure side of the blade and the front shroud. Normally, by potential flow calculation one would expect that the highest would occur on the suction surface. As a matter of fact, behind the blade on the suction surface there is a strong wake that even shows negative velocity. In other words, the flow is actually going inward. The position nearest the back shroud has a high negative velocity throughout. Therefore, at low flows the complexity increases tremendously and, as a consequence, the determination of blade forces, etc., would also be a difficult undertaking.

3.0 CAVITATION DAMAGE TESTING

Several attempts have been made to determine directly the rate of

FIGURE 3-6. Radial Velocity Fields at Near Design Speed

FIGURE 3-7. Radial Velocity Fields at Low Flow

erosion of the material undergoing cavitation damage. The standard methods, which involve measuring the specimen before and after cavitation and determining the weight loss, are not always successful because of the amount of water that might have been absorbed by the material. More recently, new techniques are being developed to determine damage rate based on use of soft coatings. The soft coatings can be in the form of marking paints, various pressure-detecting films glued to the surface, or thin aluminum plates glued on the blades. The paint erosion test is the easiest and can be evaluated without special measurement. However, the paint method indicates only where the damage is likely to occur and not the rate. The use of a pressure-detecting film gives a quantitative measurement of impact pressure caused by cavitation collapse. However, the relationship between the pressure and the damage rate does not appear to have a simple universal correlation. The most promising procedure may be a combination of paint test and aluminum plate erosion test. The intensity of erosion can be determined by estimating the surface-roughness increase or by counting the number of pits.

Another procedure is also based on using a soft metallic coating. Cadmium was chosen as a suitable coating for accelerated erosion testing and a beta particle back-scatter method for measuring the thickness of the coating. The investigators found that the erosion rate and the coating thickness had a logarithmic relationship. In the measurement of the thickness of the cadmium overlay they used a beta-emitting radioactive source, strontium 90. The source emitted beta particles, which were scattered back by 180 degrees. The amount of scattered particles was related to the amount of the cadmium thickness that was left on the surface. A great advantage of this method is that it discloses local erosion rates; therefore, the propensity to damage at any particular point in the flow field can be accurately ascertained.

There is one other method for the determination of cavitation damage, somewhat older than the methods mentioned above. It is based on the measurement of noise emitted by cavitation. A great deal of work was done in this area when it appeared that the measurement of noise could give a quick and direct assessment of the damage experienced by the impeller. Recent work, however, has indicated that such measure-

ment can be misleading, because noise is emitted whenever a cavitation bubble collapses. It is conceivable that a bubble could collapse in the free stream far from the vane surface without causing damage. It would, however, generate noise. As a consequence, the noise measurement method for erosion rate assessment is in doubt. A recent measurement, based not on direct noise measurement but on the measurement of pulses of acoustic energy situated at very high-frequency bands, has been claimed to produce erosion rate correlations. For example, determination of bearing failures using acoustic detection is well known. It has been suggested that the same method could be applied to the determination of erosion rates. This work is worthy of further study because, if it produces useful results, it would be a valuable tool for pump manufacturers and users.

Other attempts have been made to correlate measured noise intensity with cavitation damage. One approach is to distinguish between bubble collapses having a large amount of energy and those having a small amount. This results in a bubble collapse spectrum, that is, the number of pulses versus the energy per pulse. Recognizing that there is a threshold energy level below which no damage will take place, it is possible to integrate this pulse count spectrum curve and obtain an acoustic energy level that would cause damage. Tests were made with an ultrasonic vibratory test facility. Using this method and comparing the acoustic power measurements against direct weight-loss measurements, it is found that the spectrum area curve generally follows the same trends as the weight-loss curve. The investigator went further to define an erosion efficiency that is obtained by dividing the spectrum area by the weight loss. It was expected that this efficiency would remain a constant, but it was found to vary by about a factor of 5. This work could be extended. If successful, this method would become a very reliable indicator of the cavitation damage rate occurring on a prototype machine.

One last method of measurement in this area is also worthy of mention. It is stated that the volume of cavitation occurring inside a pump can be determined by measuring instantaneous fluctuations in flow between the exit and the inlet of the pump. Presumably, the extent of cavitation will affect these fluctuating levels and, therefore, by measuring these levels one can estimate the volume of

cavitation. However, no attempt has been made to correlate the volume of cavitation bubbles to the erosion rate. Perhaps this concept also is worthy of further study.

4.0 SUMMARY OF EXISTING PUMP TEST FACILITIES, RESEARCH FACILITIES, AND INSTRUMENTATION

Test programs at Byron Jackson Pump Division, Los Angeles Operations, are carried out in three separate test areas, depending on the type of pump: (a) standard types of sizes, (b) large vertical nuclear primary coolant pumps, and (c) large horizontal pumps. A fourth site is reserved for Navy "quiet pump" noise tests only. Additionally, there are the Hydraulics Research Laboratory, the Mechanical Seal Laboratory, and the Dynamics Research Test Facility.

Shielded cable networks, allowing multichannel signal transmission, connect the test areas to the centrally located Dynamics Research Laboratory. Transducer signals can be monitored during a test and simultaneously recorded on a 14-channel instrumentation tape recorder for later analysis. The particular type of analysis utilized depends on the research requirements and can be selected to meet the need. Additional channels can be added and synchronized by means of a time-code generator. Analysis is controlled by a DEC PDP 11 computer as part of a spectral dynamics digital signal processor.

The use of the PDP 11 computer allows a limitless variety of test system controls or readouts to be programmed, depending on the research requirements. The type of hard-copy output that can be obtained is illustrated by three-dimensional plots, Figures 3-8 and 3-9.

Figure 3-8 shows a series of spectrograms plotted at each of a number of operating speeds with rpm plotted on the vertical or Y axis, frequency plotted on the horizontal or X axis, and vibration amplitude plotted in the Z axis (vertically). Some type of mechanical resonance is evident at about 504 Hz, which appears with excitation by the seven-vane impeller frequency at a running speed of 4,320 rpm. Figure 3-9, a three-dimensional plot of pressure in the discharge pipe of a boiler feed pump, shows a very sharply tuned pipe at about 264 Hz

FIGURE 3-8. Three-Dimensional Frequency Response Plot

Example of discharge pipe acoustic resonance excited by the impeller vane frequency.

* The actual peak was probably between scans around 5220 rpm. Other data presentations would have been used had it been necessary to identify the resonance frequency.

FIGURE 3-9. Three-Dimensional Pressure Pulsation Plot

excited by the first-stage three-vane impeller.

Information relating to the test and research facilities of Alden Research Labs, CREARE, Iowa State University, Kyushu University, Northern Research and Engineering Corporation, Penn State University, Ingersoll-Rand, KSB, Sulzer, Westinghouse Electromotive Division R. & D. Center, and Worthington is presented in detail in Appendix C, Volume 2, of this report.

Section 4

REVIEW OF PUMP THEORY

1.0 HYDRAULICS

1.1 Introduction

A theory that encompasses the combined effects of the interdependent hydraulic components of a centrifugal pump on a real fluid stream flowing through the pump is not yet available. The present inability to model the interaction between the hydraulic components (suction, impeller, and volute or diffuser) has thwarted the prediction of performance. Further research in this general area is needed, specifically to improve upon the theory that describes the real fluid stream flowing through a pump.

1.2 Flow in Pumps

In existing theoretical methods it is generally assumed that the flow in one of the three hydraulic components of the pump can be analyzed with sufficient accuracy, neglecting the influence of the other two.

The hydraulic design concept of all centrifugal pumps is a form of Newton's laws of motion applied to the fluid traversing the machine. In the calculation of flow in impellers it is often assumed that the flow is inviscid, irrotational, and steady. In many cases, it is also assumed to be two dimensional. As a result of these assumptions it becomes possible to use the methods of potential flow theory. However, potential flow theory has shown very limited agreement with experimental results.

The method of conformal transformation has been used for analysis of flow through centrifugal pump impellers fitted with thin logarithmic spiral blades. Conformal mapping was employed to obtain the distribution of the flow components. This transformed the given physical plan into one on which the circular system of blades was mapped into

a unit circle. This method was used for calculation of the pressure distribution along the blade surfaces. The calculations were compared with experimental results and showed insufficient agreement.

The basic differences between the potential theory described above and the experimental results are attributed to the effects of inlet turn, viscous effects in the fluid, and the occurrence of stall. These methods are important because they form the basis for more advanced techniques. However, when used by themselves they do not match the experimental results very well. This lack of agreement is one of the major reasons why a one-dimensional theory, with various correction factors, is currently used to design impellers at the best efficiency regime.

1.3 Advanced Numerical Methods

Since the advent of computers, a large number of computer-based calculation methods have become available. The computation methods include finite element methods, singularity methods, and quasi-three-dimensional methods. They give good results, restricted only by the assumptions of circumferential symmetry of the flow field in front of and after the impeller and steady relative velocity field within the impeller. Some methods include corrections for viscosity effects, and even explicit recognition of flow separation.

Recently, several methods were compared with a standard set of test data. The computation procedures included finite element, singularity, and quasi-three-dimensional methods. The two-dimensional methods predicted performance of the pump just as well as the three-dimensional methods as long as the flow in the impeller was not separated. The inclusion of viscous effects did not improve the calculation accuracy of either method. However, when considering the case of an impeller in which the flow field was separated, neither method predicted the performance of the impeller very well.

When flow separation was taken into account, a two-dimensional method showed accuracy. The separation method can predict the velocity field only after the location of separation on the impeller is assumed. The flow separation method, therefore, is not predictive

at all. We conclude that for nonseparated flow, excellent methods are available because the assumption of axisymmetric flow in front of and after the impeller and steady relative flow inside the impeller is valid.

1.4 Off-Design Conditions in Pumps

It is instructive to consider the physical phenomena that are normally observed when a pump operates at off-design conditions. These physical phenomena are:

- a. The flow field behind the impeller becoming nonaxially symmetric
- b. The appearance of reverse flow at the eye of the impeller (see Figure 4-1)
- c. The appearance of recirculation at the discharge between the impeller and the casing (see Figure 4-1)

The impeller pumps against a certain head and, if the head goes up, the flow is reduced in typical centrifugal machines. Therefore, as an impeller passage rotates within the casing, where pressure varies, the impeller passage experiences different static pressures, and it will have to pump different flows as it goes around the periphery. This suggests that the relative flow field within the impeller is unsteady.

By using the known head-flow characteristic of the pump, the flow rate variation around the periphery can be predicted. Flow values must be adjusted by iteration until a solution compatible with the last segment of flow is found. The calculated results have been compared against several sets of test data and a very good comparison with the test data is obtained.

Several theories have been proposed to explain recirculation at the impeller eye. In one hypothesis the occurrence of reverse flow is related to the occurrence of separation at the vane surface. Specifically, as the flow rate reduces, the point is reached at which the flow at the blade surface starts undergoing separation, but as the flow rate is reduced still further the separation becomes larger,

FIGURE 4-1. Reverse Flow and Recirculation in a Pump Impeller

and at some point the loss is so high that it is impossible for the flow to propagate in the normal direction. As the impeller rotates, the flow in the regions of high pressure in the casing will actually be reversing from the casing into the suction, i.e., the flow will be casing driven. At this point the velocity field at suction will not be governed merely by the flow rate and the area but also by the negative flow driven from the casing. Separation is actually fixed in space, because the separation will be determined by the point at which the highest static pressure is reached around the casing. As far as the impeller is concerned, this separation is moving backwards but at the same speed as the impeller rotation. If this hypothesis is correct, the prediction of reverse flow will be intricately connected with the prediction of onset of separation in the various stream surfaces that constitute the impeller. A further extension of this work may lead to a better predictive method for the onset of recirculation at the impeller eye.

The third observation is discharge recirculation. In a standard test setup, if a total pressure probe is used between the impeller and the casing, one would find that the flow begins to have negative components of radial velocity at some flow rate. If the impeller is a mixed-flow impeller, having different tip diameters at the hub and the shroud stream surfaces, it is clear that such a discharge recirculation must appear.

1.5 Volute-Impeller Matching

A very significant concept was put forward in volute-impeller interaction, and was subsequently verified by test data. The concept illustrated is that, on an H-Q curve, the intersection of the "volute characteristic" (increasing with flow) and the "impeller characteristic" (decreasing with flow) defines the best operating condition of the pump.

Two volutes were tested, designated as volutes A and B. Volute A had a wide rectangular area at the volute throat with the cutwater fully cut back. Volute B had a trapezoidal area at the volute throat with the smaller width closely fitting around the impeller outlet.

The test results showed that a wide volute throat has both constructional and hydraulic advantage in comparison to volute cross sections that narrow down to fit the impeller outlet closely. The cutwater for volute B was cut back in stages, a procedure that resulted in a broad, crested efficiency and a best efficiency point at higher flow rate as the cutback increased. A study of the effects of skin friction was also conducted and it found them to have only secondary effects with regard to the hydraulic losses incurred in the volute flow.

2.0 VIBRATION IN PUMPS

2.1 Introduction

An increasing demand for greater reliability of high-speed rotating machinery and the introduction of specified vibration limits have created renewed interest in the development of techniques to anticipate vibration problems and methods to alleviate them. In the rotor dynamics area there are several types of analyses to be considered: critical speeds, instability thresholds, and synchronous and nonsynchronous forced response. Generally, the calculation procedures for these analyses are well founded, and the pertinent literature is fairly substantial. However, the methods for determining the parameters necessary for the calculation methods have not been well established. Substantial effort for determining these parameters is ongoing.

2.2 Critical Speeds

Normally pumps operate filled with liquid, and critical speeds are not a major problem because hydrodynamic and hydrostatic forces act upon the pump rotor. These forces are created in annular clearance spaces, such as wear rings, throttle bushings, and balance drums. The clearance space behaves like a spring having a significant stiffness. Further, the fluid also produces restoring forces that are functions of the velocity and acceleration of the shaft. Most of these effects tend predominantly to raise the critical speed and to promote stability. However, there are some cross-coupling forces, which can cause destabilizing effects.

2.3 Theory

A number of papers provide a basis for calculating the pressures and forces within annular clearances. The dynamic force coefficients of the shaft in the annular clearance are given as:

$$\begin{bmatrix} F_x \\ F_y \end{bmatrix} = \begin{bmatrix} K_{xx} & K_{xy} \\ K_{yx} & K_{yy} \end{bmatrix} \begin{bmatrix} X \\ Y \end{bmatrix} + \begin{bmatrix} C_{xx} & C_{xy} \\ C_{yx} & C_{yy} \end{bmatrix} \begin{bmatrix} \dot{X} \\ \dot{Y} \end{bmatrix} + \begin{bmatrix} M_{xx} & M_{xy} \\ M_{yx} & M_{yy} \end{bmatrix} \begin{bmatrix} \ddot{X} \\ \ddot{Y} \end{bmatrix} \quad (1)$$

It has been demonstrated that inlet swirl reduces the cross-coupled dynamic stiffness coefficients, while slightly increasing direct stiffness and damping. This implies that the capacity of these clearance spaces for promoting dynamic instability is increased. It has been shown that the fluid inertia terms that are normally neglected in the Reynolds equation approach are essential for the correct prediction of cross-coupled stiffness and direct damping terms.

An analysis of long fluid annuli, neglecting axial leakage, was made. Friction factors based on vortex or turbulent flows were employed in the theoretical development, and stiffness, damping, and inertia coefficients in a rotating/coordinate system for small shaft motion about the center were derived. The critical speeds of a test rotor with a long annular fluid clearance were compared to theoretical values and a generally good correlation was found.

During recent theoretical and experimental work, various types of wear ring geometries and clearances were tested. Good correlation was observed between measurements and calculations.

2.4 Experimental Work on Rotordynamics

Researchers, including some sponsored by EPRI, are currently continuing experiments to gather more data to support any conclusions about the adequacy of the theory presented above.

The flow associated with the impeller/volute is considerably different

from that in annular clearances. There have been observations that, due to a nonaxisymmetric flow in an impeller/volute interaction, steady and unsteady lateral forces arise. The unsteady forces and the role they might play in rotordynamics have not been adequately determined. Researchers are presently conducting an experiment to measure these forces and determine their role in rotordynamics. Present work is with a single volute, which is not representative of boiler feed pump geometry.

2.5 Miscellaneous

Several mechanisms have been observed to contribute to rotor instabilities for single-shaft systems. However, there is not very much literature that takes into account the changes in dynamic behavior of the machines after installation. The elements that may influence rotor stability after installation are foundation stiffnesses, mechanical resonances of certain structural parts, and abnormal operating conditions in the plant.

Recent work, supported by EPRI, addresses the influence of other equipment or structures on the vibration performance of auxiliary fans in power plants. A total vibratory system, including supporting soil, foundation, pedestals, fan rotor, housing, motor, etc., was included in the system model. A digital computer program was developed to aid in performing the dynamic response analysis of such systems. The response of several existing installations was determined, and good agreement was obtained between theoretical results and experimental measurements. Such analysis can clearly demonstrate why the vibration in a particular installation may be high in spite of the fact that the calculated critical speeds and stability analysis for the pump are within acceptable limits.

Another matter of interest in the field of pump vibrations is the existence of high-intensity pulsation problems in pump piping systems. Such pulsations have been observed at frequencies that are typically harmonically related to the pump operating speed. Very often the frequency could be the same as the blade-passing frequency. In some installations the frequency could be a small fraction of the operating speed. A typical example is the appearance of pressure pulsations in

the discharge pipe at the vane-passing frequency. The existence of reinforced longitudinal waves can be attributed to the resonant length of piping. In more complex instances, it has been demonstrated that the pump can actively take part in this phenomenon by influencing the acoustic response, and it can also serve as an amplifying element.

It has also been shown that strong vibrations were experienced when the inlet and discharge piping had abrupt transitions. Apparently some low-level vortices were generated at the piping discontinuities. These problems were solved by using a long, smoothly reducing cross section for piping.

3.0 CAVITATION

3.1 Introduction

Cavitation can affect overall pump performance and reliability in many ways. It can cause noise and vibration. Most importantly, cavitation can produce large-scale erosion. Because of its destructive effects, both designers and users have employed various test techniques and basic research to determine the cavitation characteristics of centrifugal pumps. Despite this effort, progress in coping with the problems it creates has been slow.

3.2 Definition of Cavitation Limits

For proper interpretation, it is necessary to define the cavitation conditions that exist at various cavitation levels. (See Figure 4-2.)

Incipient Cavitation: At the earliest stages, cavitation is detected as bubbles that begin to form on the impeller vane surfaces near the leading edges.

Cavitation Damage: As the suction pressure is reduced below that required for cavitation inception, the bubbles grow in size and number. As the bubbles move downstream with the liquid, they enter regions of higher pressure where the bubbles collapse. If this collapse takes place on or near the surface of the vane, it is

FIGURE 4-2. Cavitation Condition at Various Cavitation Levels

likely to cause damage on the material surface due to the high-velocity-liquid impact that results from the imploding bubbles. The detailed mechanisms by which collapsing cavitation bubbles cause surface damage are not well understood.

Cavitation - Break-Off: As the suction pressure is further reduced, the bubbles grow so large in size and number that much of the hydraulic passage front of or within the impeller is blocked off. Consequently the pump ceases to act as a pump and the head drops off.

3.3 Theory

The problem of cavitation combines complex material, chemical, and hydrodynamic relationships. Nonetheless, empirical predictions as to how and when the onset of cavitation may occur are available. Predictions as to its effects are more difficult. Although cavitation research has produced superb scientific studies in bubble dynamics, incipient cavitation, cavity mechanics, noise, material erosion resistance, and ways to alleviate cavitation, there is still much uncertainty.

The standard procedure for calculating incipient cavitation is to apply a method by which detailed static pressure distribution on the vane surface can be computed. Calculation of inception in this method is very straightforward and to a certain extent simplistic, because the method does not include the effects of thermodynamic nonequilibrium and various scale effects.

Over the years, a great deal of research has been done with regard to cavitation inception. Despite much research there is still uncertainty, in a number of areas, e.g., interaction of scale effects, different liquids, and hydrodynamics. There are so many combinations of scale effects, liquid condition, and flow regime that it is obviously difficult, or perhaps impossible, to provide a comprehensive theory for prediction of cavitation inception. However, if the calculations are applied within fairly narrow limits, it is believed that good results can be obtained even with a simplified theory.

Although designers and users must be concerned with the problem of incipient cavitation to know when and where it first appears, it is more important to understand the cavitation regime, during which serious erosion, vibration, and deterioration of hydraulic performance may occur. Prediction of the onset and extent of cavitation erosion has been attempted using theoretical and experimental techniques. One of the classical erosion equations can be written as follows:

$$\frac{I}{\frac{1}{2} \rho V_o^3} = \frac{\sigma}{\delta} (\Delta\sigma)^{\frac{1}{2}} \exp - \left[\frac{2.67}{W \Delta\sigma} \right]$$

In this equation:

I = erosion intensity

ρ = fluid density

V_o = flow velocity

σ = cavitation number

δ = relative nuclei size

$\Delta\sigma$ = degree of cavitation in proximity to cavitation inception

W = Weber number : $1/2 \rho V_o^2 \delta \ell / \gamma$

ℓ = typical length scale

γ = surface tension

By suitable manipulation of the above equation, one can deduce that the primary parameters involved in cavitation damage are the following:

a. Degree of cavitation

- b. The mean cavitation number, i.e., the general pressure level at which implosions are taking place
- c. Velocity
- d. The length scale

We may also identify a few secondary parameters:

- e. Surface tension
- f. Fluid density
- g. Nuclei size
- h. Fluid temperature

Evidently the damage resistance of the material itself is very important. The damage rate varies with some of the parameters mentioned above. It will increase as the amount of cavitation increases, as can be seen from standard figures that show damage rate as a function of NPSH. As the NPSH is reduced, the cavitation damage rate begins to increase at a certain point. It reaches a maximum at an NPSH just slightly higher than that at break-off. Beyond this point the damage rate decreases. A qualitative explanation of this phenomenon can be provided: Once large bubbles are formed, there is a cushioning effect of the shocks experienced by the body; as a consequence, when there is a very large number of bubbles the damage rate tends to decrease.

The effect of the mean cavitation number, however, is not well known. It is clear that pump manufacturers generally provide a greater safety margin when the pump suction pressure is high. That is typical of boiler feed pumps using booster pumps to provide high levels of NPSH at the first stage of the main boiler feed pump.

The effect of velocity on the damage rate has also been researched widely. Several references can be found that demonstrate that the damage rate increases at some high exponent of the velocity. The typical number used is 6.

The effect of length on cavitation damage has also been widely observed, but there appears to be no standard way to describe it. The rate of damage might simply be proportional to the size of the model. However, very recent work indicates that the damage rate could increase much faster than the size of the cavitation model.

For example, in aluminum it was found that the exponent of this rate of increase could be as high as 15. If that is true, the prediction of the damage rate in a full-scale prototype could not easily be made from measured observation or from measurements made in a small-scale model. This area requires further study.

There have been no concerted efforts to elucidate the secondary parameters. It is conceivable that their effects are small and not of major concern to the current project.

Historically, the cavitation break-off point has been considered of paramount importance in pump specification. Recently it was realized that operation without break-off does not by itself guarantee safety from damage. Consequently, even though the prediction of the break-off point is important as a lower limit of NPSH, it is not an adequate value from the standpoint of reliability. It is imperative that a better method be devised to assess the suitable operating value.

4.0 TWO-PHASE FLOW IN PUMPS

4.1 Introduction

The reduced performance of pumps caused by inducing gas or vapor along with liquid in the pump suction appears to be best correlated by a head-loss function at inlet that varies with the pump flow coefficient.

4.2 Definition

A pump can be considered to be operating in two-phase flow when the percentage of vapor or gas in the inlet flow stream (this is a "flow" void fraction, to distinguish it from an "area" void fraction) is between 5 and 95 percent. All known work correlating two-phase flow in pumps has been for conditions in which the pressure rise through the pump is small relative to the inlet pressure. In these conditions the change in the void fraction from inlet to outlet has been small.

4.3 The Problem

Up to the present time the problem has been to correlate, and hence to predict, the head rise that a specified pump will produce when operating at a given flow coefficient with a given inlet flow stream of liquid and vapor (or gas). This implies operation in the first quadrant. For pressurized-water nuclear-reactor systems, the head or pressure loss experienced in the second and third quadrants (reversed flow, forward rotation, and reversed flow, reversed rotation) is also required.

4.4 Past Work

The first quantitative study of two-phase flow in centrifugal pumps fulfilling the above definition covered only a small range of void fractions (well below 10 percent). Later, air-water and steam-water experiments were conducted. In almost all cases, correlations were produced to cover the two-phase test points observed. However, these correlations did not generally cover additional test points.

The current approach is based on the Euler equation applied to two parallel streams (which, for homogeneous two-phase flow, become one stream of mean density). The model shows that head losses should be a function only of flow coefficient and void fraction for any one pump. Experiments with four pump rigs, in air-water and Freon-Freon vapor, confirmed this result for a much wider set of conditions and for all four quadrants of operation.

EPRI has also supported water-steam experiments in steady-state and transient flows in three quadrants of operation, including rotor-torque measurements. The large quantity of data from these experiments has only recently become available, and it is not believed that they have been fully correlated as yet.

5.0 IMPELLER FORCES

5.1 Introduction

By long-established custom, it has been sufficient in studies of

centrifugal pump hydraulics to predict and/or measure the velocity and pressure distributions in the various pump elements and, ultimately, to correlate the efficiency of the energy conversion process that takes place in a pump with those distributions.

In more recent times, however, with the great increase in power-to-weight ratio of all forms of turbomachinery, it has become necessary to evaluate the consequences of two additional fundamental conditions of the flow of the working fluid. First, the forces arising from the pressure distributions imposed upon both the rotor and the stator must be predicted and/or measured; and second, the energy conversion process mentioned above must be assessed. It is now well established that the energy conversion process is a function of time-varying or unsteady pressure fluctuations. Thus, we may expect the resultant forces to be unsteady as well.

In this section, forces imposed on a pump impeller, arising both from flow within the impeller and from interaction with flow outside the impeller in a volute or diffuser, will be reviewed. At first, the time mean conditions of radial thrust and axial thrust will be considered. Then in the area of unsteady flow, the concept of rotating damping, forces due to perturbations of the impeller channel flow, and the impeller/volute or impeller/diffuser interaction will be considered.

5.2 Steady Radial Thrust

A classical reference correlates the steady radial thrust force with the volume flow rate of the pump in the following way:

$$F_r = f \left\{ 1 - \left[(Q/Q_n)^2 \right] \right\}$$

Since Q_n is the normal or design point flow rate, zero radial thrust is predicted as design capacity and a maximum thrust at shutoff. Force increases rapidly at capacities larger than design, as well. The correlation yields the magnitude but not the direction of the radial thrust.

Later, potential flow theory was used to derive a theoretical predict-

ion of radial force magnitude and direction. The effect of an eccentric location of the impeller with respect to the volute was included. Measurements confirmed the trends. Additional experimental measurements used a vaned diffuser and showed a radial force that increased with impeller eccentricity. At very low flow, what was taken to be rotating stall of the diffuser blades resulted in a rotating radial force on the impeller and nonsynchronous vibration of the pump shaft. The significance of the components of the rotating force was not recognized immediately.

Continuing to recognize additional consequences of the impeller/volute flow, investigators described the nonuniform volute pressure distribution that occurs at partial discharge. Individual impeller passages discharge into differing pressure zones, creating fluctuating impulses on the rotor and inducing flow fluctuations in the rotor passages. In additional experimental work, influence of the impeller diameter on radial thrust was measured. Up to an optimum diameter, radial thrust remains constant, and at larger impeller diameters in the same volute the radial thrust increases rapidly. A fluctuating component of the thrust is reported because of proximity to the tongue.

Finally, researchers investigated the impeller/volute reaction in a single-vane pump. It was found that the radial force vector rotates synchronously, indicating that the primary contribution is from the impeller flow, not the asymmetric volute flow. Investigation of the radial thrust and impeller/volute interaction with few impeller blades has not otherwise been carried out.

5.3 Steady Axial Thrust

Classical theory consists of an empirical, pragmatic method of predicting steady axial thrust in a variety of configurations of pumps. For a centrifugal pump stage in which both a hub and a shroud surface enclose the flow passages in the impeller, the axial thrust is the net effect of (1) the difference in pressure from front to back of the hub surface taken over the impeller inlet area, (2) the change of momentum in making the turn from axial to radial in the impeller, and (3) the pressure distribution in the spaces between the impeller

outside surfaces and the stationary casing walls. More modern work has simply served to provide more realistic and complex analysis and measurements of these influences.

In the case of multistage pumps, the additional effect of steps in the shaft diameter contributes to the axial thrust.

Most subsequent work is concerned with the influence of the clearance-space flows on the pressure distribution over the impeller outside surfaces. Using both numerical predictions and experimental measurements, the following effects were documented: (1) the impeller throughflow on the clearance pressure distribution when there is zero radial flow in the clearance space, (2) misalignment of the impeller with respect to the volute, as well as, (3) axial spacing of fixed wall to moving impeller surface, (4) radial throughflow in the clearance spaces, and (5) pump-out vanes as a technique for equalizing pressure distributions on hub and shroud surfaces.

Substantial fluid-mechanical investigations of clearance-space flows were initiated. The research identified the clearance flows as a function of combinations of a Reynolds number and the space/disk radius ratio for the system. For centrifugal pump applications this usually indicated a combined boundary layer on rotor and stator that was less than the clearance spacing, thus including a core flow between the two. As a result of calculations using the momentum integral technique, the ratio of core/disk angular velocity and radial pressure distribution as a function of net through-flow in the clearance space was found.

Approaching practical design needs, researchers experimentally investigated the effects of the impeller axial displacement and wear ring clearance, which induced clearance space through-flow, in two similar pumps. Two important effects were observed. Leakage through the hub side wear ring from the succeeding stage produced an outward flow in the space behind the impeller, while leakage to suction through the impeller wear ring on the eye side produced an inward flow in the space in front of the impeller, and this difference produced greatly different pressure distributions on the impeller surfaces. Further, such effects were quite sensitive to axial

displacement.

5.4 Concept of Rotating Damping

As an introduction to the effect of unsteady flow in creating impeller forces, we consider the concept of rotating damping. The rotor/bearing/pedestal mechanical subsystem, including the hydrodynamic bearing film, is generally positively damped. Including the working fluid to comprise the total system will either further increase or decrease the system damping. If overall damping capacity exceeds excitation, energy will flow out of the system and stability of the rotor will follow. If excitation exceeds the capacity of the system to dissipate the energy, the net influx or accumulation of energy will rapidly produce instability. Some eleven or so vibration-exciting mechanisms are induced by the flow of the working fluid in turbomachine stages. Perhaps four of these mechanisms are pertinent to centrifugal pumps.

In a comprehensive review one author states that ". . . several basic concepts, such as the idea of negative damping which is associated with dynamic instability, will be seen to be common to the different (pumping) systems. . . . In general, it is suggested that efforts should be directed toward obtaining an improved understanding of the transient behavior of the active (instability causing) elements within the system, since it is lack of knowledge of this aspect that currently limits the accuracy of system stability predictions."

Rotating damping in a shaft spinning at a frequency above the undamped natural frequency of the system will "urge on" the motion of the free natural vibration, tending to increase its amplitude without limit. Equations of motion of the shaft include a contribution to the resulting force from the velocity of the vibratory motion and are cross-coupled in the displacement terms.

Later researchers investigated the effect on stability of the rotor of friction of the surrounding medium. If the rotor is displaced, there arises a force perpendicular to the displacement and a moment about the center of the casing that has the same direction as the

rotation of the rotor. This work confirmed the conception of 'urging on" the rotor motion, leading to instability. The specific example of friction between the rotor and the fluid contained in the casing, in which the rotor is immersed, was calculated. Components of the resulting force are proportional to rotor displacement and are cross-coupled, i.e., the y-component of the force arises from the x-component of the displacement. No influence of the velocity of the rotor motion was found in this example.

Recent authors identify, reference, and/or describe a large number of excitation mechanisms pertinent to radial, centrifugal turbomachines.

5.5 Unsteady Impeller Channel Flow

As distinct from steady radial and axial thrust, which has already been considered, the derivation of unsteady impeller forces from unsteady channel flow will be considered in this section. Rotating damping, when combined with the velocity of the vibratory motion, is capable of enhancing or exciting the vibratory shaft motion. Since the velocity of the vibratory motion may increase and decrease as the shaft orbit is traversed, so the resulting exciting force may fluctuate or be unsteady as well. Rotating damping is frequency dependent.

Four investigations of various aspects of unsteady impeller forces are noted. The first investigator conceived that velocity and pressure fluctuations were induced in the channel flow by the velocity of the shaft orbital motion traced by the impeller. This yielded a fluctuating force on each impeller blade. Summation of the force was then made over all blades and resolved into components along, perpendicular, and parallel to the rotor displacement, respectively. Hence, when the orbital velocity is in the direction of shaft rotation (forward whirl), the resulting force plays exactly the same role as rotating damping does in enhancing the amplitude of vibration and acting as a destabilizing influence. A quasi-steady approximation was adopted to expediently calculate the velocity and pressure distributions, and numerous impeller flows were analyzed and compared with operating experiences to yield a threshold of negative damping below which instability would follow. Detailed

results of an actual investigation were reported.

In a second investigation, the unsteady impeller flow is analyzed. The impeller is considered to be surrounded by an unbounded vaneless diffuser. Using unsteady potential flow theory, the authors showed that a positive damping force opposed the whirling velocity of the shaft center for ranges of all parameters investigated. Thus the impeller forces were predicted to have a stabilizing influence. The impeller force was found to be a function of the blade tip angle, the ratio of the whirl angular velocity to rotational angular velocity, and the blade loading, but independent of the number of blades. In the experimental work it was observed that the impeller force varied with both shaft displacement and orbital velocity and that tangential fluid forces damped the whirl motion in most cases. However, at low flow rate and low whirl speed ratio, negative damping did occur. The ranges of parameters studied in the numerical work were not compared with actual radial impeller operating conditions, so that no judgment on their realism can be made.

In an investigation presently under way, a very pragmatic approach to experimentally measuring the stiffness and damping coefficients associated with the unsteady impeller flow is being followed. A fourth study involves the investigators' concept of rotor system stability. It states that (1) a self-exciting mechanism converts rotary energy into translatory energy; (2) a natural frequency of the system is excited by such a mechanism; (3) most self-exciting mechanisms are orthogonal to and linear in the rotor displacement; (4) if the orthogonal component is in the direction of the orbital shaft motion, the influence is destabilizing, if opposite to shaft motion, stabilizing; and (5) several effects can contribute to the "net" orthogonal component of impeller force. Subsequently, they developed quantitative evaluations of the effect of aerodynamic whirl excitation and self-excitation due to labyrinth seals, and concluded the latter were the greater influence in the machinery subject to their investigation. They have now presented their quantitative results on the influence of labyrinth seals.

Several comments can be made. In the compressors evaluated by the

first investigator, floating-ring seals far outnumbered labyrinth seals, and still the fluid dynamic excitation of the unsteady impeller flow led to unstable rotor motion. Other investigators use a potential flow method to predict the impeller force as well as stiffness and damping coefficients. Finally, a method that avoids the limitations of unsteady potential flow analysis, uses a finite element method to compute the inviscid, rotational, unsteady flow through a cascade in which periodic and nonperiodic variations in the flow rate result in the time-variation of the lift and drag. This approach opens a more realistic possibility for the analytical estimation of unsteady impeller forces.

5.6 Impeller Channel/Volute Interaction

Unsteady flow discharged from an impeller will induce unsteady flow in the volute. Unsteady volute flow is a time-varying boundary condition on the impeller channel flow, which may or may not be affected in such a way as to increase its unsteadiness. This subject is at the beginning of its development and established results have not yet been obtained.

6.0 FLOW INSTABILITY

6.1 Introduction

Flow instability in a pump system is created by the interaction of an unstable pump head-capacity curve with a system with unstable characteristics. Power plant feedwater systems have the requisite unstable characteristics, and a stable pump head-capacity curve is necessary for successful operation of the system. Pump designers have empirical methods that produce the needed pump characteristics, but basic knowledge of this subject is incomplete.

6.2 System Instability

Stepanoff (1957) broadly summarizes classical knowledge of pump flow instability, dealing with surge on the one hand and stage geometry on the other. If shut-off head is less than maximum head and head-capacity fluctuations (swings) occur under certain operating

conditions, then part of the head-capacity characteristic is unstable. Head-capacity swings are followed by fluctuations in power and speed.

Three operating conditions must be present for the swings or fluctuations to occur: (1) the mass of fluid in the pump and adjacent piping and vessels must be free to oscillate; (2) the system must include a region, either fluid or structure, that will exchange energy with the working fluid; and (3) external excitation is needed to start the swings (surge). Thus, surge is a system phenomenon.

6.3 The Effect of Pump Design Parameters

Regarding stage geometry, a rising head-capacity characteristic, or lack of one, is dependent on the number of impeller blades as well as the blade geometry designated by the inlet blade angle, the discharge blade angle, and the schedule of change of blade angle between inlet and discharge.

Stable head-capacity curves are more likely to characterize pumps of larger head and smaller capacity, that is, lower specific speed, while higher specific speeds are accompanied by unstable head-capacity curves for flow less than about 70% of design flow. In one of several typical efforts to stabilize high specific speed pumps, Myles (1966) devised a schedule of impeller passage change that delays the instability but at a considerable cost in efficiency.

Makay (1967) qualitatively categorized sources of flow instability in two groups: flow mechanisms or phenomena and pump stage geometry. The author does not give analysis or experimental correlations of the items in his list.

6.4 Theory

The linear relation between flow or capacity, Q , and the theoretical Euler head $H_{th\infty}$, which has a negative slope when backward-curved blades are used ($0 < \beta_2 < 90^\circ$, β measured with respect to the tangential direction), is demonstrated at the outset by Yididiah (1972).

At any Q , the theoretical head is reduced due to a combination of the following factors: slip, hydraulic losses, prerotation at the inlet, rotating stall, and recirculation. The steepness of the theoretical $Q-H_{th\infty}$ curve is crucial, since the head reduction at low flows may be great enough to produce a positive slope of the actual $Q-H$ curve and raise the possibility of pump flow instability. The author continues with a discussion of slip, prerotation, and hydraulic losses and their contribution to the head reduction, all based on one-dimensional, through-flow concepts.

It is noteworthy that a review of Science Citation Index from 1972 to the present showed that, although Yididiah seems to have presented a stimulating proposal for explaining the phenomena that contribute to the actual drooping $Q-H$ curve, no author cites Yididiah in any more recent work. Thus we conclude indirectly that little or no work has been done on this problem since 1972. A likely reason for this is that such work is doubly difficult, because (1) all the phenomena listed by Yididiah are three-dimensional flow effects and (2) all occur at off-design flow conditions. It is only recently that Moore and Moore (1981) report the first, limited success with computation of viscous, three-dimensional radial impeller flows, from which may ultimately follow an analysis and description of Yididiah's factors.

This state of affairs was also confirmed to the writer by Swift (1981) in a comment on his presentation to FDI Summer Course, 1981. In preparing that material, no result unifying published information on pump flow instability was found, and only a few scattered investigations of the individual phenomena and effects seem to exist. These are incomplete, and no study of the interaction of the effects leading to the depression of the actual head-flow curve at low flow has been forthcoming.

Section 5

FEED PUMP RESEARCH REQUIREMENTS

1.0 FEED PUMP PROBLEMS

Feed pump problems were identified in Section 2 of this report, listed under three headings: Byron Jackson/EPRI Survey of Utilities, Major Outage-Producing Causes Identified in EPRI RP-754, and Problems Identified in EPRI NP-1571.

Following is a list of the feed pump problems identified in Task 2, listed in sequence, with what we believe to be the most costly problem at the top of the list and the least costly at the bottom.

1. Vibration
2. Impeller breakage or cracking
3. Shaft seal failure
4. Rapid wear of wear rings
5. Cavitation damage
6. Axial balancing device failure
7. Broken or damaged shaft
8. Journal bearing failure
9. Seizures of wear rings, etc.
10. Thrust bearing failure
11. Unstable head curve
12. Auxiliary system reliability
13. Hot misalignment
14. Gear-type couplings

In addition to the above direct problems, indirect problems relating to specifications, procurement practices, etc., are also identified and listed below. The following list is not in order of severity or frequency. It is an important checklist for Phase 3 activity but not particularly related to the Phase 2 research work.

1. Failure of interstage partitions
2. Internal high-pressure seal washout
3. False automatic trips in nuclear plants
4. Feedwater system design

5. Procurement practices, such as emphasis on high quoted efficiency, low cost, low required NPSH, and low minimum flow
6. Incomplete or incorrect information given to vendors
7. Comprehensive testing not specified or enforced
8. Inadequate supply of spare rotating elements and spare parts
9. Lack of spare pump on stand-by
10. Poor operating procedures
11. Operation on turbine turning gear
12. Failure to monitor and evaluate pump operating parameters
13. Lack of control over pump repairs
14. Lack of communication among people from utilities, architect/engineers, vendor service, and vendor design engineers
15. Inadequate response by vendor service people
16. Inadequate technical service manuals

The following list also falls into the category of an important checklist for Phase 3 activities. It is not necessarily in order of severity or frequency of the problems. It represents the problem areas identified by the Byron Jackson/EPRI survey of architect/engineers.

1. For boiler feed pumps better definition is needed of short-duration transient or upset low- and high-flow conditions, as opposed to long-duration operation at minimum or maximum pump flow.
2. Should minimum long-duration boiler feed pump flow be defined as the flow required at minimum practical plant load for fossil-fired plants?
3. Better definition of required maximum boiler feed pump flow is needed.
4. The proper selection of the size of the start-up/spare boiler feed pump requires further study.
5. For nuclear feed pumps better definition is needed of short-duration transient or upset low- and high-flow conditions, as opposed to extended operation at minimum or maximum pump flow.

6. Better definition of required minimum flow for nuclear feed pumps is needed. Is it different for PWR plants and for BWR plants?
7. Better definition of required maximum flow for nuclear feed pumps is needed. Is it different for PWR plants and for BWR plants?
8. The proper selection of the size of the start-up/spare nuclear feed pump requires further study.
9. The subject of NPSH margin requires further clarification.
10. More specific information on the design of suction piping to provide uniform velocity distribution to the pump suction is needed.
11. Although discharge piping resonance and water hammer are not common problems, we believe that further study of these topics is important.
12. Severe low NPSH testing to simulate pump suction upsets and transients should be studied.
13. Methods for the evaluation of quoted pump efficiency should be investigated.
14. The formal analysis of feed pump reliability should have further investigation.
15. Specification of proper operating instrumentation should be studied.
16. Further investigation of instrumentation for cavitation detection and/or measurement of available NPSH is in order.
17. The subject of shaft seals and related injection systems still requires further study.
18. The question of what constitutes a proper acceptance test for a large feed pump remains open.

2.0 PHASE II: RESEARCH PROGRAM

2.1 Hydraulics at Off-Design Flow Rates

Such problems as axial balancing device failure, impeller breakage or cracking, shaft seal failure, rapid wear of wear rings, vibration of pump, piping, or foundation, broken or damaged shaft, unstable head curve, journal bearing failure, thrust bearing failure, and seizure are often related to operation at off-design flow. An extensive research program is necessary to better understand pump hydraulics at off-design flow and the problems created by operation at off-design

flow.

2.1.1 An experimental study is needed of flow patterns, impeller/volute (or diffuser) force interactions, cavitation phenomena at impeller eye and discharge, and unsteady pressures and loads in a representative boiler feed pump impeller and volute (or diffuser) geometry. Included in this research should be determination, by flow visualization, of the nature and type of flow at off-design as a function of volute or diffuser-impeller spacing, measurement of impeller/diffuser (or volute) stiffness matrices at off-design flows, and a qualitative assessment of the effects of impeller/diffuser (or volute) geometry on cavitation onset. Transient local loads caused by diffuser/impeller interaction should be studied by using surface-pressure measurement with a sufficiently high frequency response. Further, special attention should be paid to these measurements at impeller and volute vane tips.

2.1.2 It is also necessary to have a research program that will produce quantitative, three-dimensional measurements of flow patterns in a representative boiler feed pump impeller and volute (or diffuser) geometry. This work should provide a mechanism for explaining the origin of the unsteady impeller forces by observing the unsteady velocity and pressure patterns. A laser-doppler velocimeter should be utilized for this work.

2.1.3 A finite element computer program for boiler feed pump hydraulic analysis should be developed. This program should have three-dimensional analysis capabilities, utilizing steady and unsteady algorithms and enabling modeling of nonsymmetric volute or diffuser geometry. It is important that classical assumptions regarding the blade exit condition not be used. A closure condition based on true viscous modeling should be employed instead.

2.1.4 Also needed is an analytical and experimental study of the onset of recirculation in the eye and at the discharge of a representative boiler feed pump impeller geometry. It is probable that large, unsteady forces are related to recirculation in boiler feed pump impellers and that these forces create hydraulic instability.

2.2 Cavitation

Cavitation damage, impeller breakage or cracking, rapid wear of wear rings, vibration of pump, piping, or foundation, broken or damaged shaft, journal bearing failure, thrust bearing failure, and seizures may all be related to cavitation in feed pumps. Cavitation research is essential.

2.2.1 The development of an analytical method for predicting the performance of cavitating feed pumps is required. An advanced theory for cavity flow in impellers is needed. This theory must then be correlated to test data showing the effect of advanced cavitation on the deterioration of head and efficiency in a centrifugal pump stage. The analytical results depicting the extent of cavitating volume at any operating condition would be an important input to the development of damage rate prediction in paragraph 2.2.2.

2.2.2 An analytical study of cavitation damage is required. A sound theoretical basis for the determination of cavitation damage rate must be established and correlated with the experimental results from the work to be described in paragraphs 2.2.4 and 2.2.5.

2.2.3 An experimental correlation between noise and cavitation damage rate would be an extremely valuable tool for the diagnosis of an operating feed pump and for possible prevention of cavitation damage.

2.2.4 Two or more distinct types of cavitation are known to exist. An experimental correlation between type of cavitation and cavitation damage rate is necessary for understanding the cavitation damage problem.

2.2.5 An experimental program involving direct measurements of cavitation damage in an impeller eye is necessary to correlate the results of all of the other cavitation research.

2.3 Rotordynamics

Such problems as axial balancing device failure, impeller breakage or cracking, shaft seal failure, rapid wear of wear rings, vibration of

pump, piping, or foundation, broken or damaged shaft, journal bearing failure, thrust bearing failure, and seizures may all be related to rotordynamics. Research in rotordynamics is absolutely necessary, and some of this important research is already being conducted, sponsored by EPRI.

2.3.1 Development of advanced rotor-bearing systems for feed water pumps is now being sponsored by EPRI projects RP-1266-7 and RP-1884-4. Dr. Maurice L. Adams, Jr., and Dr. Elemer Makay are the principal investigators. This is an experimental study of wear ring force coefficients aimed at determination of coefficients for existing common wear ring geometries and at devising wear ring geometries with better damping coefficients. We will recommend an extension of this work to include testing of geometries that represent long interstage bushings, balancing drums, and throttle bushings. Another portion of this work deals with the development of a squeeze-film damper for feed pump journal bearings.

2.3.2 Investigation of axial balancing devices is also a necessity. An additional part of the research mentioned in paragraph 2.3.1 is the effort directed at attenuation of axial vibration. An axial squeeze-film damper is suggested as a possible remedy to the problems caused by the extreme rigidity of balancing disks.

2.3.3 To support the experimental work described in paragraph 2.3.1 and to promote a better understanding of the problems involved and of the experimental results, an analytical study of seal ring coefficients is needed. Perturbation solution techniques should be employed to define force components acting on the rotor due to small motion relative to a centered position in the wear ring or seal. Documentation should cover both the theoretical development of the seal force coefficients and calculated results for a range of currently employed seal geometries and operating conditions.

2.3.4 An experimental investigation of pressure loss coefficients in wear rings and other annular pressure-reducing devices is required in relation to the above research work. It is also needed as an input to one or more of the computer-aided design methods of phase III.

2.4 Pump Systems

Axial balancing device failures, cavitation damage, impeller breakage or cracking, shaft seal failure, rapid wear of wear rings, vibration of pump, piping, or foundation, broken or damaged shaft, journal bearing failure, thrust bearing failure, hot misalignment, and seizures may all be related to pumping systems.

2.4.1 A study of system interaction between pump, piping, and the feed water control system is required. To a lesser degree, this study should also consider such structural components as baseplates, foundations, etc. It should include such items as feed pump vibration caused by piping resonance excited by pressure pulsations from the pump, water hammer, determination of maximum allowable pressure pulsations, sources of pump vibration caused by external excitations, interrelationships between the pump and the feed water control system, and feed pump minimum flow bypass systems.

2.4.2 Also required is an investigation of the effects of suction piping configuration and of suction transients on pump performance. This should include analytical and experimental investigations of flow profiles at feed pump inlets and their effect on performance. A definition and quantification of the feed pump inlet transient conditions created by a deaerator transient, a booster pump failure, or by other upset conditions should be included. Experimental investigation of pump first-stage performance during simulated upset suction transient should also be undertaken.

2.5 Full-Scale Verification

2.5.1 Axial balancing device failure, cavitation damage, impeller breakage or damage, shaft seal failure, rapid wear of wear rings, vibration of pump, piping, or foundation, broken or damaged shaft, journal bearing failure, thrust bearing failure, hot misalignment, and seizures are all problems on which full-scale verification of laboratory-scale research is needed. Full-scale, full-speed tests of boiler feed pump stages are required. All of the experimental research indicated above would be conducted on scale models. Even though the theory developed would transcend the size effects, it is

quite likely that certain obscure but significant scale effects would remain undiscovered. This is particularly true for cavitation-related work. Consequently, full-scale testing is necessary to verify the theories, unearth the obscure mechanisms, and thus close the loop between theoretical and experimental research.

2.5.2 Because shaft seal failures have been identified as the most frequent cause of feed pump outages, a full-scale experimental investigation of shaft seals and shaft seal systems for feed water pumps should be undertaken. A more efficient test program will result if the shaft seal tests are separated from the boiler feed pump stage tests described in paragraph 2.5.1.

2.6 Application of Research Results

Coordinated attacks on the problems of axial balance device failure, cavitation damage, impeller breakage or cracking, shaft seal failure, rapid wear of wear rings, vibration of pump, piping, or foundation, broken or damaged shaft, unstable head curve, journal bearing failure, thrust bearing failure, hot misalignment, seizures, and gear-type coupling problems can be solved only through consolidation of research results and utilization of the research results in pump component development and pump evaluation methods.

2.6.1 Consolidation of the above research results will lead to and give guidance to pump component development. For example, the results of the research outlined in paragraph 2.3 will lead to improved designs for wear rings and other annular pressure-reducing devices. The research conducted as outlined in paragraphs 2.1 and 2.2 will lead to improved impeller design and improved understanding of the various design relationships between impellers and volutes or diffusers. Full-scale development of such devices as squeeze-film radial bearing supports and axial squeeze-film dampers is also likely. Some pump component development should involve large-scale testers, while other developments should utilize the hardware as outlined in paragraph 2.5.

2.6.2 To complete the research phase of the program, a systematic methodology for evaluating pump design elements and performance in

terms of total economics is required. The purpose is that of providing a basis for engineering and economic tradeoffs between investment and operating costs in relation to feed pumps.

3.0 PHASE III: DEVELOPMENT OF DESIGN METHODS

3.1 Computer-Aided Design Methods

The research results must be translated into design methods that can be utilized by the feed pump designers and others.

3.1.1 An hydraulic analysis based on the results of the research outlined in paragraph 2.1 is required.

3.1.2 A cavitation analysis based on the results of the research outlined in paragraph 2.2 is required.

3.1.3 An analysis of annular pressure-reducing devices based on the research outlined in paragraph 2.3 is needed.

3.1.4 A stress analysis of impeller vanes, impeller shrouds, and the pump shaft, taking into account the fatigue stresses caused by unsteady pressures and forces as well as the centrifugal forces on the vanes and shrouds, is necessary.

3.1.5 It is also necessary to conduct a stress analysis of the volute lips or diffuser vane tips and the fatigue stresses imposed by unsteady pressure loads.

3.1.6 A rotordynamic analysis that accounts for the stiffness, damping, and virtual mass coefficients of the impellers and of the wear rings and other annular pressure-reducing devices is needed. The stiffness and damping coefficients of radial bearings, which may be of an advanced design, must also be known and integrated into the program.

3.2 Pump Specification Guidelines

Comprehensive pump specification guidelines should be published as an

aid to architect/engineers and users. These guidelines should present the results of the research in a practical and usable manner.

3.2.1 A publication on instrumentation for monitoring and troubleshooting operating feed pumps will be required.

3.2.2 Practices for the selection of materials of construction for large feed pumps should be reviewed and updated.

3.2.3 Test codes for feed pumps should also be reviewed and updated.

3.2.4 A review and updating of standards for feed pump applications in view of the completed research, incorporating its results, should be provided.

3.3 Troubleshooting Aids

The research results should provide assistance in troubleshooting of feed pumps.

3.3.1 A troubleshooting manual will be prepared. The purpose of this manual will be to aid in the identification, diagnosis, definition, and resolution of problems experienced by existing feed pumps. Included should be methods, analytical tools, and experimental techniques that will be useful in problem solving.

Section 6

PHASE II: RESEARCH PROGRAM HYDRAULICS (1.0 through 4.0)

1.0 OFF-DESIGN FLOW STUDIES

1.1 Objective

The objective is to determine, identify, and quantify, where practicable, fluid dynamic phenomena that are important to the operation and reliability of pumps at off-design (low and high) flows.

1.2 Background

The surveys that were conducted as part of Phase I activity have revealed that operation of boiler feed pumps at lower than design flows, and in some instances at higher than design flows, has caused a number of problems affecting the reliability of the pump. The survey of the literature indicated that this area of operation is not very well understood, both from a fluid dynamic point of view and in terms of the forces generated under these operating conditions. It is felt that a primary research topic for the boiler feed pump reliability improvement program should be an extensive study of the flow phenomena that occur during these off-design operating conditions. To be specific, attempts must be made to understand the detailed fluid mechanics of the impeller and volute (or diffuser) interaction, the cavitation phenomena that appear at both the eye and the discharge of impellers, the appearance of recirculating flows at both the eye and the discharge of the impeller, the forces generated by the volute impeller interactions, the stiffening and damping effects produced by these interaction flows, and, finally, the effect of whirl of the impeller within the volute casing due to lateral vibrations.

1.3 Approach

It is fairly well recognized that flow computations at off-design

conditions do not, in the present state of the art, provide sufficient insight for understanding the mechanisms. Therefore, our approach is to suggest a highly experimental program to recognize the operation of these phenomena and, where practicable, to develop as much of a quantitative understanding as possible. First, we would like to be able to measure directly the impeller/diffuser stiffness matrices at off-design flows by direct determination of the forces (both steady and unsteady) as a function of the flow rate for a given eccentricity of the impeller orbit. Secondly, attempts will be made to determine by flow visualization the nature and type of flow at off-design operation of centrifugal pumps with volutes and diffusers as a function of the spacing between the impeller and the volute or diffuser. Thirdly, possibilities for measuring the impeller blade tip transient loads will also be investigated.

1.4 Equipment

An existing rotor force test facility will be specially modified for this project. This facility will be capable of driving the impeller in an off-centered circular orbit. The impeller itself will be provided by Byron Jackson as a model, approximately 6 inches to 7 inches in diameter, of an existing boiler feed pump impeller. Both a volute model and diffuser model will be provided as needed. The stiffness matrices will be measured over a wide range of flow rates and impeller/diffuser clearance ratios, from say 0.03 to 0.15, to determine principal effects. The corresponding series of flow observations, using tufts, sensing probes, etc., will also be made to determine flow characteristics and to identify the onset of flow recirculation or back-flow phenomena. These observations will provide the basis for practical models to be coupled with volute-to-diffuser steady-state pressure distributions.

Unsteady loads on one or more diffuser inlet sections will be estimated by installing high-frequency, flush-mounted pressure transducers. The possibility of instrumenting the tip section of one or more impeller blades will be investigated. For example, the tip section of an impeller blade could become the live portion of a dynamometer, and the dynamometer could be instrumented with strain gauges and slip ring assemblies. Other ideas will also be considered.

2.0 VELOCITY FIELD MEASUREMENTS IN A PUMP STAGE

2.1 Objective

The objective is to provide a direct measurement of the velocity field that exists in a typical boiler feed pump impeller rotating in a volute or diffuser.

2.2 Background

As has been stated earlier, the velocity field calculations at off-design flow conditions have been found to be decidedly inadequate as far as design predictions are concerned. It is proposed that measurements have to be made directly on typical configurations to establish the velocity fields. Recent experimental advances have suggested that a new technique based on using a laser-doppler anemometric system would be suitable for this kind of measurement. In the literature survey portion of Phase I, it was found that the laser-doppler velocimetric system has been developed to the point of being commercially available.

2.3 Approach

As a result of considerable study into the various available laser systems, it has been found that the most appropriate would be a laser dual-beam system. The principle of the laser dual-beam system is explained elsewhere in this report. It will be sufficient here to add that it can also be used for measuring flow velocities that are negative, by using what is called a frequency-shifting procedure employing a Bragg cell. Determination of velocity components in two mutually perpendicular directions can be achieved by using what is called a two-component fringe pattern system, in which fringe patterns are formed at right angles to each other by using two sets of beam pairs.

An advantage of the laser velocimetric system is that the signal-to-noise ratio of the doppler signal can be estimated by using the Mie scattering theory before the actual set-up is completely worked out. A decision as to whether the back-scattering method or the forward-

scattering method should be used can be based on this theoretical estimate. The selection of tracer particles (to be present in the fluid to scatter the laser beam) is also quite important. In general, the optimum choice of flow seeding in a particular application has to be determined experimentally. It appears that aerosols with nearly uniform particle sizes are preferred. Other particles used are polystyrene latex, silicone carbide, etc.

The following experimental observations will be made on a model of a boiler feed pump impeller to be supplied by Byron Jackson. The flows will be observed at the impeller exits and at the impeller eye. Velocity measurements will be made at several locations around the circumference of the impeller and also at different depths along the axial direction. Measurements will be repeated for several operating speeds and at numerous flow rates for each operating speed, with particular attention paid to off-design operating conditions.

The results obtained from this data-acquisition system will be used to correlate the velocity field to the measured impeller forces of the off-design flow studies.

3.0 IMPELLER/VOLUTE HYDRAULIC ANALYSIS

3.1 Objective

The objective is to develop a finite element computer program for the calculation of the flow field in typical boiler feed pump impellers in conjunction with volute interaction effects.

3.2 Background

It has already been noted in Section 4 of this report that the standard hydraulic computer programs fail in the prediction of velocity fields at other than design flow rates. It was pointed out that several assumptions were made in standard flow calculations. These are: (1) the flow field is axisymmetric, (2) the exit flow angle could be determined by some numerical fixes, for example, specification of smooth velocities at the trailing edge, etc., and (3) the point of separation was assumed either from prior experience or

purely arbitrarily. As a result of these assumptions, it was determined that the computed velocity field was, in general, not in good enough agreement with the measured velocity fields. It was postulated that, without basic breakthroughs with regard to these assumptions, further development of computation is not cost effective. Recent work in the computation of very complex flow fields, not involved in turbomachinery, has shown that some of these basic assumptions can be eliminated. (A.N. Brooks and T. J. Hughes: "Streamline Upwind/Petrov-Galerkin Formulations for Convection-Dominated Flows With Particular Emphasis on the Incompressible Navier-Stokes Equations," to be published in Computer Methods in Applied Mechanics and Engineering). An interesting result of the referenced work is that the separation of the flow from the bounding surface is automatically predicted by the calculation. This is because an accurate turbulence modeling is included as part of the treatment. Further assumptions of axial symmetry are no longer important because the computer programs will be written with algorithm features that facilitate high accuracy without prohibitive computer costs. The fact that the appearance of reverse flow at the eye of the impeller can be predicted a priori is a great advantage to the system and a highly desirable goal.

3.3 Approach

The proposed task is to develop a numerical analysis procedure for calculating incompressible, viscous flows in boiler feed pumps. The computer program developed will possess the following features:

1. The method will be based upon finite element discretization procedures to enable convenient modeling of intricate impeller/volute geometries
2. Axisymmetric and three-dimensional analysis capabilities
3. Steady and unsteady algorithms
4. Turbulence modeling capability
5. The equations will be written with respect to a coor-

dinate system fixed on the impeller, which is assumed to revolve at constant angular velocity

6. A small-data-base, unconditionally stable, implicit method will be employed
7. A Lagrangian-Eulerian mesh will be employed to enable modeling of nonsymmetric volute geometry. This will reduce the discretized domain for this case to the region between two consecutive impeller channels

The computer program will be instilled with algorithm features that facilitate high accuracy in the convection-dominated cases of interest.

4.0 HYDRAULIC INSTABILITY

4.1 Objective

The objective is to develop an understanding of the effects of recirculation on overall performance and to correlate hydraulic instability mechanisms with recirculation.

4.2 Background

Many boiler feed pump problems arise at off-design flows. One of the characteristics of off-design operation is the appearance of reverse flow at the inlet and the discharge of the impeller. Even though this mechanism has been observed in many instances, explanations for it are scarce. Further, its effects are not fully understood. Many experimental studies have revealed the actual flow field to be extremely complex, involving nonaxially symmetric flows as well as time-dependent parameters. However, the importance of the problem necessitates efforts to elucidate the mechanism and to assess its effects on performance.

4.3 Approach

As pointed out in paragraph 3 of this section, there is a high proba-

bility that the prediction of the onset of recirculation might emerge from the development of computer codes involving the solution of the full Navier-Stokes equation. However, if this approach is not successful, due either to conceptual complexity or to prohibitive computer costs, it is desirable to develop alternative simpler methods with greater certainty of success. Two methods of predicting the appearance of reverse flow have been uncovered as a result of the literature search. The first, as developed by Ferrini and his colleagues in Italy, has been fully reviewed in the section on literature search in this report. The second method, by J. Loret, has also been reviewed. We would recommend that both of these approaches be pursued and that a usable computer calculation scheme be developed based on these methods.

More important than the prediction of the onset of recirculation is the assessment of its effects on overall pump performance and the recognition of whether it can create hydraulic instability. A mechanism that has been postulated for the destabilization of the performance is the production, in effect, of an inlet swirl in the presence of recirculating flow. By the Euler equation one can see that the presence of the inlet swirl reduces the overall head of the pump, provided that the inlet swirl is in the direction of rotation. It is well known that at lower than design flows the inlet swirl always rotates in the same direction as the impeller itself. This reduction in head can cause the overall head flow curve to develop points of zero, or even positive, slope. These slope regions can cause overall system instability. It appears that the calculation methods mentioned above cannot conceivably be extended to the point at which the change in the head flow curve can be predicted. It is quite clear that experiments will have to be done in order to appreciate the significance of this effect. A set of experiments can be conducted in continuation with the tests planned in paragraph 1. Using plexiglass inlet pipes, it is possible to observe the onset of recirculation and at the same time measure carefully the forces generated by this mechanism and, also, its effect on the stiffness matrices. Further, it is possible, by controlling the NPSH at the inlet, to make the pump undergo cavitation at the same time as recirculation phenomena begin to make their appearance. In this way the effect of recirculation on cavitation can also be analyzed

concurrently, measuring once again the forces and the stiffness matrices. The effects of combined cavitation and recirculation on the rotating dynamics of the system can be investigated.

It is also proposed that experiments be conducted on a somewhat larger scale. It is possible to run tests in an existing pump laboratory test loop, with variable-speed capacity, up to a power range of about 300 hp. Such a test facility can be used in conjunction with a horizontal-process pump, for example, and measurements of recirculation can be made by either using plexiglass inlet pipes or velocity-traversing probes at the inlet. The overall head vs. flow performance can be quickly measured, and the effects of various modifications (for example, the lengthening of the inlet leading edges, the change of the volute geometry, etc.) can easily be observed. The process can be facilitated by the large number of different impeller and volute combinations that can be obtained in the existing Byron Jackson inventory of pumps.

CAVITATION (5.0 through 9.0)

5.0 ANALYTICAL STUDY OF CAVITATION PERFORMANCE

5.1 Objective

The objective is to develop a method for predicting the performance of cavitating feed pumps.

5.2 Background

The determination of the performance of a pump undergoing severe cavitation involves an analysis of the flow field inside the impeller when a large cavity forms. The analysis of cavity flow even for a single blade is rather complex due to the existence of free streamlines and a highly turbulent wake behind the cavity bubble. The earliest calculations of such flows were made by linearizing the boundary conditions and treating the pressure as constant along the free streamline. Such a method, however, fails when the angle of incidence increases (that is, for low flows in pumps). A nonlinear method for such conditions has been developed for axial pump

impellers using either flat blades or arbitrarily profiled blades. Such theories, by suitable transformation, can be applied for purely radial impellers. However, a typical boiler feed pump impeller has an axial part as well as a radial part; a somewhat modified method has to be developed, therefore, for such a configuration.

5.3 Approach

It is understood that the actual flow field is three dimensional. However, in order to make the problem tractable, it will be analyzed on a set of adjacent stream surfaces. Each stream surface is characterized by two independent variables, the meridional coordinate and the tangential coordinate. Instead of solving the problem by treating these two variables as independent, a stream function definition will be made in such a way that the stream function and the meridional coordinate will become the independent variables and the tangential coordinate will become the dependent variable. The convenience of such a transformation is simply that the stream function along the surface of the cavity, a specified quantity, is the boundary condition of cavity flow. The position of the cavity streamline will be the unknown. Such a transformation has been used previously for separated-flow computations. The cavity length, of course, is not known a priori in the calculations. However, an iterative procedure will be used to determine the cavity length. Further, the wake thickness after the cavity will be chosen to be proportional to the cavity drag. Some success has been reached in using such hypotheses in calculations for propellers. After the calculation of cavity boundary is complete, it is possible to integrate the pressure distribution to obtain the torque and, consequently, the head. Efficiency calculations will also be made as a result of the calculation of the cavity drag. Thus, a complete head versus NPSH curve can be developed for any given flow and NPSH condition. These results will provide an important input into task 6.

6.0 ANALYTICAL STUDY OF CAVITATION DAMAGE

6.1 Objective

The ultimate objective of this program is an analytical model capable of predicting when and where cavitation erosion will occur in a boiler feed pump and the rate at which this erosion will progress.

6.2 Background

Research into cavitation damage has been going on for many years. At the outset, it would appear unlikely that a comprehensive theory of cavitation damage prediction for any given material and hydrodynamic combination can be developed. However, some theoretical guidance must be provided if good cost-effective experiments are to be performed. Therefore, the objective in developing a cavitation damage theory is that of obtaining sufficient understanding of the major parameters controlling the erosion rate to plan and perform appropriate experiments.

6.3 Approach

In order to develop an understanding of cavitation erosion mechanisms, several questions must be asked. Of these the most important, perhaps, are the following: (1) What is the mechanism of cavitation bubble collapse, particularly near a solid boundary? (2) What is the amount of energy imparted to the solid material by the collapsing cavity? (3) What is the material response to such a load? It is unlikely that these questions can be fully answered during the course of this project. However, if certain bubble collapse mechanisms are assumed, a statistical distribution of the cavitation bubble collapse energies can be developed. The damage rate, then, may be considered as a function of this distribution of bubble collapse energies and the material response to the bubble collapse. Of course, it is well known that the damage rate varies with time, particularly during the incubation period. The non-dimensional expression for the damage rate could then be written in terms of several hydrodynamic parameters, material parameters, and the time element.

A general expression for the damage rate would take the form:

$$D^O = F(V_\infty^n, \alpha, \sigma/\sigma_D, T_\infty, D_M, h), G(M, HT, D_G), H(t/t_O) \quad (1)$$

where for hydrodynamic considerations,

V_∞ = the free stream velocity with $n = 6$

α = the total gas content of the liquid

σ/σ_D = normalized cavitation number

T_∞ = the free stream temperature

D_M = the model diameter

h = a flow shape parameter

σ = cavitation number

σ_D = desinent cavitation number

for material response,

M = the test material (includes erosion resistance and corrosion resistance)

HT = any heat treatments of the test materials

D_G = the grain size/microstructure of the material
(due to method of fabrication, etc.)

and for the exposure time,

t/t_O = a dimensionless exposure time

The total damage would be,

$$D_T = \sum_{i=1}^n D_i^O t_i \quad (2)$$

and could be expressed as either a total volume loss or as a mean depth of deformation where the volume loss is divided by the damage

area.

Preliminary attempts can then be made to develop quantitative expressions for these nondimensional parameters, based on the existing theories or using some ad hoc hypotheses. Naturally, experimental data must be gathered continually in order to improve the prediction accuracy of such procedures.

7.0 CORRELATION BETWEEN NOISE AND CAVITATION DAMAGE

7.1 Objective

The objective is to develop an experimental correlation between acoustic emission and cavitation erosion rate.

7.2 Background

As has been pointed out in the section on the assessment of feed pump experimental research, many attempts have been made to correlate acoustic emission to cavitation damage rates. Although qualitative correlations have always been found, a good quantitative correlation has eluded researchers. Recently, they have realized that the reason the correlations are not very close is that noise is generated whenever a bubble collapses, and, of course, a bubble collapse will not cause damage if the collapse occurs in the free stream. Therefore, the objective would be to develop a method to discriminate between collapses that cause damage and those that do not.

7.3 Approach

Recent advances in instrumentation techniques have made it possible to detect high-frequency pulses that are characteristic of material damage. One such concept has recently come to light. The principle of this concept is that when a material undergoes damage by repeated impacts, a distinct noise is emitted in a very high-frequency range, of the order 100 KHz. It has been suggested that the amplitude of this noise may be directly related to the rate of damage. A recently developed instrument measures not only the average R.M.S. energy in this high-frequency range but, also, a new quantity that

is a measure of the energy contained in the intense acoustic bursts generated when a material undergoes damage. The acoustic energy of these intense bursts is measured only above a certain threshold level in order to suppress small spikes. The threshold level itself is considered to be a function of the R.M.S. amplitude. The new measurement is called SAT, to stand for Spike Above Threshold. It is proposed that this experimental avenue be fully explored. In order to provide a cost-effective method for establishing the validity of this unique noise measurement, we propose that a test setup be made in which cavitation can be quickly and conveniently generated. In particular, the test setup will consist of a rotating disk in which several holes are drilled. Behind the holes recesses can be made in which test specimens can be mounted. The wake generated by the rotation of the disk will cause cavitation on the specimens. They can be easily removed and the weight loss measured quite accurately. During this cavitating run, noise measurements can also be made. Because of the simplicity of the setup, various types of cavitating material can be used, and it will be a simple matter to observe whether the spikes above the threshold truly track the actual damage rate. Naturally, in such a setup it is possible to control pressure levels, temperature, speed, degree of cavitation, specimen lengths, fluid properties, and, of course, different materials.

If a good correlation should emerge between the SAT measurement and cavitation damage rate, it is proposed that this type of noise measurement be made on actual cavitating boiler feed pump impellers.

8.0 CORRELATION BETWEEN TYPE OF CAVITATION AND DAMAGE RATE

8.1 Objective

The objective is to establish the correlation, if any, between different types of cavitation and the cavitation damage rate in centrifugal impellers.

8.2 Background

Prediction of long-term performance of cavitating pumps requires an understanding of the mechanism by which the damage is imparted to the

solid surface from the cavitating volume. Several mechanisms have been postulated in the literature. It appears that different types of cavitation can cause different types of damage. An individual cavity closely located to the solid can generate a shock wave on collapse. Also, the collapse of a bubble close to a surface can form a microjet that impinges on the solid and produces pitting. Additionally, when a cloud of bubbles forms and collapses near a surface, it is possible that the individual shocks can combine to provide a large-amplitude single shock wave directed onto the solid surface. It is therefore important to determine whether for the same class of impellers, under similar operating conditions, the same type of cavitation collapse mechanism always exists.

8.3 Approach

It is conceivable that some information with regard to these mechanisms will be generated as a result of the work of paragraph 6. However, it is appropriate to consider suitable experimentation to investigate this point further. It is proposed that single-stage impellers be tested at low speeds in a test facility capable of up to 300 horsepower at variable speed and variable NPSH conditions. Cavitation formation should be observed through plexiglass windows, and high-speed motion pictures should be taken to see if the detailed mechanisms of the bubble collapse are similar in all cases. Attempts will be made to modify the leading-edge shape of the impeller by providing surface roughnesses, protuberances, etc., to change the characteristics of the cavitating bubbles. In conjunction with what is to be described in the next section, the relationship between the type of cavitation and the damage rate will be established.

9.0 DIRECT MEASUREMENT OF CAVITATION DAMAGE IN IMPELLERS

9.1 Objective

The objective is to develop a cost-effective method for the measurement of damage rate in operating impellers.

9.2 Background

Several methods for direct measurement of cavitation damage have been proposed in the literature. These include use of paint, soft metals, pressure-detecting film, and radioactive methods. Of these methods, not all are practical from a cost point of view. It is therefore proposed that some be studied in detail and a promising one be recommended for future use.

9.3 Approach

It is quite obvious that the use of paint or ink is perhaps the simplest method to detect the presence of cavitation damage. The problem with this method is that it provides no means of establishing the rate of cavitation damage. If an impeller could be made entirely of a soft material, like aluminum, it could be run for long enough periods to determine the actual rate of damage. However, due to the lack of mechanical strength, that is not always a feasible alternative. An ingenious way to overcome this difficulty is to glue a small, thin plate of pure aluminum to the blade surface. Recent work in naval hydrodynamics has indicated that the response of a thin glued plate is almost the same as that of a solid impeller of the same material. It is proposed that in the existing test facility some of the single-stage process pumps be investigated under low NPSH conditions by using both paint and glued aluminum plates on the vane surface. The paint can be used to determine the point at which erosion is likely to take place. Then, in that area, a soft aluminum plate can be attached and allowed to undergo cavitation damage. The damage itself can be measured by counting the number of pits. It is also possible that the newly developed pressure-detecting film could be procured. It could then be installed on the appropriate surfaces. This method determines the pressure, acting on the solid surface, that is directly related to the impact erosive intensity. It has been indicated that a good correlation can be obtained between the pressure data and the erosion on the soft aluminum plate. It may be expected that the pressure-detecting film that is at present sold by a company in Japan can be used as a convenient erosion test method in the near future.

It has been suggested that tests be conducted for several impellers and several operating NPSH conditions at different speeds in water. The tests should employ paint or ink and the soft aluminum plate, as well as the pressure-detecting film.

These test results will form an important part of the work on cavitation research. In particular, they will provide an accurate check of the calculations that are developed in paragraph 6. They will establish the effects of the cavitation type as outlined in paragraph 8. And lastly, it might provide direct information on the safe NPSH value for long-life operation of boiler feed pumps.

ROTORDYNAMICS (10.0 through 13.0)

10.0 DEVELOPMENT OF ADVANCED ROTOR-BEARING SYSTEMS FOR FEEDWATER PUMPS

10.1 Objective

The objective of this research is to determine stiffness and damping coefficients for various wear ring configurations, to study the advantages of squeeze-film damper bearing supports, and to lead to the development of advanced rotor-bearing systems.

10.2 Introduction

This essential research effort is already under way. The work is being performed by Dr. Maurice L. Adams, Jr., and Dr. Elemer Makay and is being supported by EPRI. The EPRI project numbers are RP-1266-7 and RP-1884-4. The investigators have recognized that large hydraulic excitation forces probably can never be entirely eliminated, especially at off-design flow. The approach, therefore, is to determine the stiffness and damping coefficients of typical wear ring configurations, to determine which current configuration best contributes to rotor stability, and further, to devise and test other wear ring geometries that may prove to be superior. Also being investigated are two potential journal-bearing configurations that employ squeeze-film damper supports. It is hoped, especially on feed pumps with relatively stiff rotors, that journal bearings

with squeeze-film damper supports will make a significant contribution to rotor stability.

10.3 Interstage Fluid Annuli

The present effort directed toward interstage fluid annuli concentrates on the relatively low length-to-diameter ratio representative of wear rings. This is important, but at a later stage of this work we will request that higher length-to-diameter ratio geometries be investigated so that a better understanding of long interstage bushings, balancing drums, and throttle bushings will be possible. We believe that this additional effort is necessary.

10.4 Squeeze-Film Damper Journal Bearing Supports

At the present time we have no additional suggestions relating to the scope of the work on squeeze-film damper journal bearing supports. However, in the future we may suggest investigation of squeeze-film damper journal bearing supports for other types of journal bearings, perhaps the pressure-dam bearing.

10.5 Relationship to Full-Scale Pump Tests

The first paragraph of Section 2 of EPRI CS-2027, "Development of Advanced Rotor-Bearing Systems for Feedwater Pumps, Phase II," reads in part, "Participation by the feed pump manufacturers is planned to successfully demonstrate this approach in laboratory tests and in the field." Full-scale verification testing of these concepts should be integrated with our full-scale, full-speed tests of representative boiler feed pump stages, as outlined later in this section in paragraph 16.

11.0 INVESTIGATION OF AXIAL BALANCING DEVICES

11.1 Objective

The objective is to improve the state of the art relating to axial balancing devices, and particularly to investigate the advantages of an axial squeeze-film damper.

11.2 Background

Failure of axial balancing devices, especially balancing disks, is one of the most frequent and most costly feed pump failure modes.

11.3 Current Practices

At the present time there are three basic design concepts used to handle the axial pump thrust problem. These three concepts are: (1) opposed stages, (2) in-line stages with balancing drum, and (3) in-line stages with balancing disk. An excellent discussion of this topic is contained in section 5 of EPRI CS-2027 (see paragraph 10.5).

11.4 Axial Squeeze-Film Damper

We agree that further investigation of an axial squeeze-film damper to control axial vibrations and axial dynamic forces is worthy of further investigation. This concept is fully described in paragraph 5.5 of the report mentioned in paragraph 11.3. Additional concepts may emerge from the results of the other research work as it progresses. Our other research work, particularly in the broad categories of the hydraulics of off-design flow, cavitation, pump systems, and full-scale verification testing, will undoubtedly provide additional insight into the phenomena which lead to axial rotor instability.

12.0 SEAL RING COEFFICIENTS

12.1 Objective

The objective is to develop an analytical/computational model for the estimation of the dynamic coefficients of annular close-clearance rings in boiler feed pumps.

12.2 Background

Sophisticated rotordynamic programs are available for critical speed, synchronous response, and rotordynamic stability calculations. The potential accuracy of these computer programs is severely limited by inadequate models for forces that have a marked influence on rotor-

dynamic behavior. These forces arise principally from the fluid/structure interaction that acts on the rotor at impeller and seal locations. The effect of the interaction at the impeller location is being studied in paragraphs 1 through 4 of this section. This task concentrates on the development of calculation methods for the seal locations.

A number of methods exist for the calculation for the seal ring coefficients. Typically, there are three classes of assumptions made in these analyses. The first assumption relates to the length of the bearing. In many cases the bearings are assumed to be short, meaning that the end effects can be neglected. The second assumption relates to what is called the "swirl effect." In high-speed seal rings the fluid can spin at relatively high tangential velocities. A classic assumption, often made, is that the average tangential velocity of the fluid is equal to one-half the peripheral velocity of the shaft. However, when large pressure drops exist, it is conceivable that the fluid will not reach this high peripheral tangential velocity. This effect, if not properly accounted for, will result in overestimation of the cross-coupled terms in the dynamic matrices. The third assumption relates to the variation of the clearance around the circumference and the consequent variations in the Reynolds number. This will affect the friction factors. Further, very few theories exist in the literature for grooved wear rings.

12.3 Approach

We would try to eliminate the assumptions mentioned above and, also, to extend the calculations to geometries of practical interest in boiler feed pumps. Specifically, the calculation for smooth seals will be developed based on the turbulent lubrication equations of Hirs. The rotor forces will be calculated for the finite length of seals, considering that the shaft executes small motions about a centered position. The solutions will account for swirl and influence of clearance perturbation in Reynolds numbers. All first-order inertia terms will be included in the analysis. The solutions to be developed will be consistent with the geometries and operating conditions of current boiler feed pump wear ring and interstage seals.

For grooved seals the solution will be based on either the turbulent lubrication equations of Hirs or the nonlinear turbulence equations of Elrod and Ng ("A Theory of Turbulent Films and Its Application to Bearings," Journal of Lubrication Technology, Volume 89, No. 3, July 1967). The models will account for swirl and the influence of local clearance changes on friction factors. First-order inertia terms will be retained as appropriate. "Fine groove" theory will be employed, in which flows and pressures will be averaged with respect to the grooving width. The solutions will be consistent with the geometries and operating conditions of currently employed boiler feed pump seals.

13.0 WEAR RING PRESSURE LOSSES

13.1 Objective

The objective is to develop estimates and empirical equations for pressure losses in wear rings and other annular pressure-reducing devices found in boiler feed pumps.

13.2 Background

Most of the calculations of the rotordynamic coefficients of wear rings, etc., start from the pressure losses suffered by the leakage flow between concentric rotating cylinders. These pressure losses are of three kinds: the first is an entry loss, the second a friction loss, and the third a discharge loss. Usually the estimates for these coefficients have been developed based on the early work of Yamada, whose equations show the friction loss coefficient as a function of the axial and circumferential Reynolds numbers. The entry loss and discharge loss coefficients are generally arbitrarily assumed to be about 0.5 and 1, respectively. It has been remarked that the accuracy of estimates for the dynamic coefficients depends very strongly upon the assumptions with regard to the pressure losses. Evidently, calculation of these coefficients is virtually impossible. Therefore, an experiment is proposed to determine them.

13.3 Approach

In the test apparatus a stiff shaft carrying the rotating portion of the test seal will be run in the stationary casing, which can be arranged eccentrically with respect to the rotor. Static pressures will be measured at several axial locations and at several circumferential positions with pressure transducers. The eccentricity of the rotor can be measured with displacement sensors at several axial and circumferential positions. The flow rate through the seals will, of course, be measured. Tests can be performed under conditions in which the axial Reynolds number is varied between about 1000 to about 4000, and the rotational Reynolds number between zero and about 5000. These would be typical of boiler feed pump operating conditions. Several width-to-diameter ratios and various grooving configurations would be used. The measured pressure losses will be correlated in terms of Reynolds numbers in the axial and circumferential directions, the local clearance ratios, and the hydraulic diameter ratios of the geometries. We expect that the final correlation will be relatively simple equations that can be conveniently input to rotordynamic seal coefficient computer programs.

PUMP SYSTEMS (14.0 and 15.0)

14.0 SYSTEM INTERACTIONS

14.1 Objective

The objective is to investigate the existence and severity of system interactions between pump, piping, and the feedwater control system.

14.2 Background

This task has been defined to include the following:

- a. The maximum allowable pulsations of discharge pressure at vane-passing and other frequencies, and the piping loads and stresses that can result from these pressure pulsations.

- b. The effects of water hammer in feedwater systems on failure of axial-thrust balancers, impellers, seals, interstage partitions, etc.
- c. Feedwater control-system interactions that might bring about vibrations.
- d. Effect of other external excitations--through the ground, foundations, framework, electric supply, etc.
- e. The characteristics of minimum-flow bypass systems.

14.3 Participants

Research participants in this task will be a major University and a major Architect/Engineering firm.

14.4 Approach

The method of approach to this task will be as follows. The Architect/Engineering firm will search its extensive data for past occurrences of problems in this area and, perhaps also, for situations that can be regarded as examples of good practice. They will make available to and discuss with the University all those data that do not infringe proprietary limits. At the same time, the University will discuss similar data with people at Byron Jackson, at EPRI, and with Dr. Elemer Makay and his co-workers.

We anticipate that after the data-gathering part of the work is completed the succeeding study will be mainly analytical. Computer or other models of the control system, the pump, piping systems, etc., will be set up and subjected to disturbances typical of those to be found in practice. The models will include nonlinear characteristics, nonuniform piping and joint properties, fluid forces, and inertia. As examples of the sophistication with which the models will be set up to simulate real-life situations, trials will be run and compared with practice. When problems are replicated in the model, limits to acceptable practice can be established. Experimental verification can be obtained from the pump test facility to be set up for another task.

The Architect/Engineering firm will be one of the most prominent architect/engineers of power plants and similar large systems in the country, and will include on its staff experts in feedwater systems. The University has probably the widest and most intensive turbo-machinery research programs of any university in the country and has carried out several successive programs on nuclear-PWR circulating pumps and boiler-feed pumps. It is also the acknowledged leader in control system theory.

This task and task 15 will employ a full-time doctoral student and will be supervised by a professor of mechanical engineering who has outstanding qualifications in this field.

15.0 SUCTION EFFECTS ON PUMP PERFORMANCE

15.1 Objective

The objective is to investigate the effects of suction piping configuration and of suction transients on pump performance.

15.2 Background

It has been determined that the effects of different configurations of suction piping and of various suction transients on feed pump performance should be studied. This work would be carried out by the University with the advice and support of the Architect/Engineer.

15.3 Participants

Research participants for this task will be same as those for the task outlined in paragraph 14.

15.4 Approach

The first part of the study will be a survey of past problems to which suction piping or transients have been major contributing factors. Some of these data will be made available from the Architect/Engineer's large body of records, insofar as proprietary restrictions will permit. Other data will be sought from

Byron Jackson, EPRI, and from Dr. Elemer Makay and his co-workers. A literature review (updated from the recent BJ-EPRI review) will also be made.

An investigation of the flow profiles at feed-pump inlets over the whole operating range will then be made. Part of this study can be analytical, but the complexity of the flow will require the setting up of experiments. Existing pump loops at the University will be satisfactory for some studies, and an additional test loop(s) will be erected in any of several available laboratories.

The transient conditions set up by a booster-pump failure, a deaerator transient, or other large-scale changes will be defined and quantified, and the effects of such transients will be investigated in the experimental loop(s).

The effects of various inlet flow conditions and inlet transients on cavitation performance of the first stage of a boiler-feed pump will be investigated, using the results obtained elsewhere under another task.

The results will be produced in a form that will include as precise information as possible on the allowable limits of suction-piping configuration and with recommendations on pump, control-system, and auxiliary-system design to avoid cavitation and other manifestations of poor pump performance in practical conditions.

FULL-SCALE VERIFICATION (16.0 and 17.0)

16.0 FULL-SCALE TESTS

16.1 Objective

The objective is to verify laboratory research results and to determine what obscure but important differences exist (if any) by running full-scale, full-speed tests of representative boiler feed pump stages.

16.2 Background

It is not feasible to perform the experimental portions of the research outlined in paragraphs 1 through 15 of this section utilizing full-scale hardware. Such research must be conducted in research laboratories using highly sophisticated scale-model hardware and advanced laboratory instrumentation techniques. The results exhibit uncertainties, however, such as scale effects in hydraulic and cavitation testing. Full-scale verification of research results is needed.

16.3 Approach

It is recommended that three test pumps be constructed.

Test Pump No. 1: Single-suction impeller, double volute.

Test Pump No. 2: Single-suction impeller, diffuser.

Test Pump No. 3: Double-suction impeller, double volute.

Figures 6-1 and 6-2 show the conceptual designs of test pumps Nos. 1 and 3, respectively. Test pumps Nos. 1 and 2 simulate series stages of multistage double-case boiler feed pumps. The only difference between them is that test pump No. 1 has a double volute and test pump No. 2 a diffuser. With the exception that the diffuser is not shown, Figure 6-1 also represents the conceptual design of Test pump No. 2. Test pump No. 3 simulates the double-suction first stage of a double-case boiler feed pump.

Conventional tilting-pad radial bearings are utilized, as well as conventional pivot-shoe thrust bearings. Although face seals are generally not acceptable on feed pumps, they will be utilized for these test pumps to minimize the radial constraint created by the shaft seals. Throttle bushings would be more reliable, but would introduce more radial constraint and reduce the transmission of hydraulically induced radial forces from the impeller to the radial bearings. High-frequency response load cells will be provided in both radial bearings and in both thrust bearings to measure direction and magnitude of axial and radial reactions at these locations. Vibration of each bearing housing will be measured by accelerometers. Axial and radial shaft motions will be measured by proximeters

FIGURE 6-1. Conceptual Design of Test Pump No. 1

FIGURE 6-2. Conceptual Design of Test Pump No. 3

located near the bearings. Additionally, the radial shaft motion, as near to the impeller as is practical, will be measured by internal proximeters. Suction pressure fluctuations and discharge pressure fluctuations will be measured by high-frequency-response transducers. The test pump with the double-suction impeller will be equipped with a hydrophone to record cavitation noise and an optical system for viewing the cavitation in one side of the impeller eye.

All data will be recorded on magnetic tape and stored at least until the completion of Phase III, because reanalysis of the data in a different manner might be indicated by work performed near the end of the program.

Data reduction and plotting of results will be computerized.

16.4 Test Program

Tests will be run at three speeds (approximately 75%, 85%, and 100% rated speed) to verify affinity relationships.

There will be a series of tests run with an oversize impeller that has a less-than-recommended clearance between the outside diameter of the impeller and the volute lips and diffuser tips.

Most of the testing will be with recommended full-diameter impellers.

Conventional NPSH tests will be run on all test pumps, and extensive special cavitation testing will be conducted on test pump No. 3.

Test pump No. 3 also will be subjected to a simulated deaerator transient test and a simulated booster pump failure test.

All pumps will be tested with the rotor axially displaced the safe maximum distance toward the thrust bearing and again with the rotor axially displaced the safe maximum distance away from the thrust bearing.

All test pumps will be tested with simulated worn wear rings with 150% of normal clearance and 200% of normal clearance.

Test pumps Nos. 1 and 2 will be tested with a radially displaced rotor that is off center and displaced toward one set of volute lips (or diffuser tips).

Test pumps Nos. 1 and 2 will also be tested with an impeller that has been machined with the hydraulic center displaced.

This test hardware may also be used to run a test with deliberately "tuned" discharge piping that resonates at vane-passing frequency.

All pumps will be tested with a series of reduced impeller diameters, probably 95%, 90%, 85%, and 80% of full diameter.

16.5 Test Results

Test results will be compared with the results of the laboratory research. Among the comparison techniques will be a rotordynamic analysis similar to that described in paragraph 25 of this section.

17.0 SHAFT SEALS

17.1 Objective

The objective is to determine the root causes of shaft seal failures, to clarify the importance of shaft seals to rotordynamics, and to improve existing seals and injection systems or establish new concepts.

17.2 Background

The most frequent cause of feed pump outages has been identified in EPRI Report FP-754, "Survey of Feed Pump Outages," as shaft seal failure. Clearly, research into shaft seals and shaft seal systems is necessary. The first step in a shaft seal research program must be to categorize and analyze various types of shaft seal failure. It is recognized that, in some instances, shaft seal failure is the result of such things as vibration caused by operation at very high or low flow, rotor unbalance, improper bearing selection or maintenance, system transients, or other upsets.

A more thorough understanding of root causes can probably be obtained in interviews with people such as Dr. Makay and a number of utility engineers, who have been intimately involved in seal problems. Such a survey could be conducted primarily by mail and telephone, but perhaps with a few selected interviews.

17.3 Approach

The first conclusion from the above analysis will probably be that face-type seals are not sufficiently reliable for use in large feed pumps. This was the conclusion of EPRI Report FP-754, "Survey of Feed Pump Outages." It is borne out by the fact that face seals are not acceptable to any of the architect/engineers interviewed by Byron Jackson during Phase I of this program. The second probable conclusion is that the floating-ring type of seal is not sufficiently reliable in large feed pumps. This again was a conclusion of EPRI FP-754 and was implied by the results of the Byron Jackson/EPRI survey of utilities. However, the Byron Jackson/EPRI survey of architect/engineers indicates that floating-type seals are still acceptable to some of them. It is probable that the shaft seal investigation will consist primarily of a detailed analysis of the proper design of throttle bushings and throttle bushing injection systems. The previously concluded research is expected to provide guidance for the design of throttle bushings, taking into account both the desirability to minimize leakage and the effects of throttle bushings on rotor dynamics.

Also important is the study of throttle bushing injection systems, because injection system failures may prove to contribute to a large percentage of seal failures. Present injection systems are basically either temperature controlled or pressure controlled. The proper selection and design of each type of system should be investigated.

APPLICATION OF RESEARCH RESULTS (18.0 and 19.0)

18.0 PUMP COMPONENT DEVELOPMENT

18.1 Objective

The objective is to develop conceptual designs of components that are likely to have improved performance, and to verify the improved designs by test.

18.2 Background

The goal of the tasks outlined in paragraphs 1 through 17 is to conduct research that will lead to improved understanding of current feed pump problems and produce the basic information required for the design and production of more reliable feed pumps. The next step toward this goal will be the development of improved pump components. The exact nature of this development task cannot be determined until the research is completed. However, we can be sure that certain improved components can be developed as a result of the research. Common failures have been observed in axial balancing devices, first-stage impellers (cavitation damage), impeller vane and shroud tips (breakage or cracking), shaft seals, wear rings, shafts, journal bearings, thrust bearings, internal high-pressure seals, couplings, and auxiliary systems. All of these items are therefore candidates for component development.

18.2 Approach

The full-size pump tests described in paragraph 16 of this section will be completed prior to the beginning of pump component development testing. The hardware from the full-size test pumps, the related instrumentation, and the data-reduction system will undoubtedly be quite valuable and will be adaptable to pump component development. It is anticipated that, except for certain hydraulic development tasks, pump component development will involve testing of full-scale components. At this time, the most likely candidates for pump component development are axial balancing devices, shaft seals, wear rings, shafts, impellers, volutes (and diffusers), journal

bearings, thrust bearings, couplings and auxiliary systems.

19.0 ECONOMIC EVALUATION METHODS

19.1 Objective

The objective is to develop a methodology for the evaluation of the economics of new pump designs and design modifications for existing pumps.

19.2 Background

Pumps of improved configuration resulting from the research may be more costly than the feed pumps currently on the market. An economic justification is required for the development costs of improved pumps and, possibly, for the additional manufacturing costs of the improved pumps. Methods are needed to evaluate the costs of higher priced, more reliable pumps and to justify these costs by the cost savings resulting from greater reliability (lower outage costs, lower repair costs, lower maintenance costs, etc.).

A systematic methodology for evaluating pump design elements and performance in terms of total economic costs is required. This methodology should provide a basis for engineering and economic tradeoffs between investment and total operating costs over the life of the plant.

19.3 Approach

Reliability criteria will be developed for use in the evaluation of design methods and standards for feed pump application (including instrumentation for monitoring and troubleshooting operating feed pumps, practices for the selection of materials of construction for large feed pumps, and test codes for feed pumps).

The majority of the economic evaluation procedures and criteria now standardized by EPRI focus on the total plant and such variables as cycle design, fuels, etc. That is, the greatest concern is on the macroeconomics of the entire plant. This task relates to the

microeconomics of individual plant components, such as feed pumps, and even to individual design features and parts of the feed pumps.

This task will contain economic analyses that relate to the operation of the feed pumps. For example, the additional costs for power and for larger piping and valves in the minimum-flow bypass system might be compared to potential savings in replacement power costs and maintenance costs resulting from the greater reliability of an increased minimum flow in the bypass system.

PHASE III: DEVELOPMENT OF DESIGN METHODS
COMPUTER-AIDED DESIGN METHODS (20.0 through 25.0)

20.0 HYDRAULIC ANALYSIS

20.1 Introduction

The art of hydraulic design to meet particular head and flow specifications is highly developed. The hydraulic research described in paragraphs 1, 2, 3, and 4 of this section may increase our understanding of flow phenomena sufficiently to result in a significant contribution to the state of the art for the best efficiency point. If so, this would be an added benefit. However, the primary purpose of this work is to investigate off-design flow phenomena, particularly as they relate to impeller/volute (or diffuser) force interactions, unsteady pressures and loads, and the onset of recirculation. The goal is to arrange the hydraulic research results in a computerized format usable by design engineers.

20.2 Finite Element Computer Program for Flow Analysis

The output of the research described in paragraph 3 will be a finite element computer program for hydraulic flow analysis. This program is expected to be useful as a basic tool in the transformation of research results to engineering analysis tools.

It should be noted that such transformation may be done in one of two ways. Either a specific computer program may be provided to the designer so that any individual design concept may be analyzed, or

the results of the research may be computerized and analyzed on a parametric basis, utilizing the range of parameters applicable to feed pumps, so that an individual analysis of each pump design that falls within the established parameters need not be performed.

20.3 Impeller/Volute (or Diffuser) Force Interactions

A computer program that will describe the impeller/volute (or diffuser) force interactions is required to furnish the necessary inputs to the stress and rotordynamic analyses described in paragraphs 23, 24, and 25. These may take the form of rather elaborate subroutines, which then feed directly into the above-mentioned stress analysis and rotordynamic analysis programs.

20.4 Unsteady Pressure and Loads

Unsteady pressures and loads, also, are required inputs to the stress analysis programs described in paragraphs 23 and 24. Again, the output of this task may be in the form of an elaborate subroutine that directly feeds the stress analysis.

20.5 Onset of Recirculation

The onset of recirculation may define the dividing line between acceptable and marginally acceptable flow domains. The instabilities that may arise in recirculating flow can very well limit the operation of feed pumps for extended periods of time. The performance effects as a result of recirculation in the impeller eye may also be a necessary input to advanced cavitation analysis.

21.0 CAVITATION ANALYSIS

21.1 Introduction

It is necessary to transform the results of the research described in paragraphs 5, 6, 7, 8, and 9 of this section into a format useful to the designer. A computer program may be provided for analyzing any specific configuration, or the results may be furnished in parametric form, covering the range of parameters commonly found in feed pumps

and presented in such a manner that the designer need not perform individual analyses if the parameters of the design fall within the selected ranges.

21.2 Prediction of the Performance of Cavitating Feed Pumps

The output of the research described in paragraph 5 of this section will be a set of computer programs, which will serve as new analytical tools for determining the flow characteristics of cavitating feed pumps. A complete input-output data manual will be included. These computer programs may provide guidelines on safe NPSH requirements for the long-term operation of pumps.

21.3 Cavitation Damage Rate

The cavitation research results are expected to contribute to a much better understanding of the factors that contribute to cavitation damage rate. The results of this research will be consolidated and presented in a form usable by the design engineer.

21.4 Onset of Recirculation

As mentioned in paragraph 20, the onset of recirculation in the impeller eye may be an important consideration to the designer in the analysis of cavitation.

21.5 Correlation Between Cavitation Damage and Cavitation Noise

The results of the research are likely to lead to a better understanding than now exists of the relationship between cavitation noise and cavitation damage. These results will be correlated and presented in a manner usable by design engineers.

22.0 ANALYSIS OF ANNULAR PRESSURE-REDUCING DEVICES

22.1 Introduction

Annular pressure-reducing devices include wear rings, interstage bushings, balancing drums, and throttle bushings. It is necessary

to transform the results of the research described in paragraphs 10, 11, 12, and 13 of this section into a form usable by design engineers.

22.2 Loss Coefficients

The experimental investigation of pressure losses in wear rings and other annular pressure-reducing devices described in paragraph 13 is expected to produce a set of loss coefficients that take into account factors not now commonly used by design engineers. These loss coefficients will give more accurate prediction of leakage flow through annular pressure-reducing devices. This is a necessary item in itself and, also, one needed as an input for the determination of force coefficients.

22.3 Force Coefficients

The force coefficients for annular pressure-reducing devices that are of primary interest to the design engineer are the stiffness and damping coefficients. Of secondary importance are virtual mass coefficients. Paragraphs 10 and 12 of this section describe the experimental and analytical research that is expected to produce a better understanding of force coefficients than now exists. It is necessary to compute the force coefficients and use them as input in the rotordynamic analysis.

23.0 STRESS ANALYSIS OF IMPELLER VANES, IMPELLER SHROUDS, AND SHAFT

23.1 Introduction

Elastic stresses in the shaft and the impellers, due to discharge pressure pulsations, and axial and radial forces at the impeller tips will be investigated by using a finite element method. Since the shaft geometry is symmetrical about its centerline, a two-dimensional axisymmetric model will be developed for the shaft analysis. The impeller geometry is complex. However, it does have a cyclic symmetry about each vane. Therefore, a sector of the impeller, including the hub, the shroud, and a vane, will be used to develop a three-dimensional model. Thus two separate analyses will

be required, one for the shaft and one for the impeller. A general-purpose finite element computer code such as ANSYS (Swanson Analysis Systems, Inc., Houston, Pennsylvania) will be used for both analyses.

23.2 Stress Analysis of the Shaft

A computer code such as ANSYS, with STIF42 isoparametric solid elements, will be used for this analysis. The model will include the rotating shaft and all axisymmetrical rotating parts. The element is defined by four nodal points having two degrees of freedom at each node: translations in the nodal, radial, and axial directions.

The resulting force (radial and axial) and moment components will be applied on the shaft surface at the points where the impellers are attached. The loading conditions will vary with time as the shaft rotates; they will be calculated using the results of the research to be conducted during earlier phases of the project. The steady and time-dependent elastic stresses will be calculated based on the worst loading condition.

The shaft is rotating at very high speed; therefore the stresses caused by centrifugal force will also be included. Thermal stresses will be very small and hence neglected. The finite element model will have coarse elements everywhere except in the region of critical areas, where fine mesh will be employed. This model will use about 700 elements and 1,000 nodes.

23.3 Stress Analysis of Impeller

Because of the complex nature of the impeller geometry, a three-dimensional detailed finite element model will be required to accurately predict stresses. In order to restrict the impeller model to a manageable size and to minimize computer costs, one repeating sector of the impeller will be modeled. That is, a five-vane impeller will be modeled into five identical cyclic sectors. The finite element model of the impeller will include one vane and the attached portions of the hub and the shroud halfway to the next vane on either side.

A three-dimensional finite element impeller stress model will be developed using a computer code such as ANSYS, with STIF45, isoparametric solid elements. The element is defined by eight nodal points having three degrees of freedom at each node: translations in the axial, the hoop, and the radial directions.

The steady and time-dependent elastic stresses due to pressure and centrifugal loading will be calculated. The pressure field will vary with location. That is, the hub, the shroud, and the vane will each see a different pressure field. The pressure fields will also vary with time.

The pressure fields will be based on the results of earlier phases of the research. At least two or three different loading cases will be analyzed to evaluate reliability under the most pessimistic conditions. The finite element impeller model will use about 400 elements and 900 nodes.

24.0 STRESS ANALYSIS OF VOLUTE LIPS AND DIFFUSER VANE TIPS

The elastic stress analysis of the pump case-volute (or diffuser) will be made by using a general-purpose finite element computer code such as ANSYS. A three-dimensional finite element model will be developed using three-dimensional isoparametric solid elements, such as ANSYS-STIF45. The element is defined by eight nodal points having three degrees of freedom at each node: translations in the nodal axial, radial, and hoop directions. The model will use about two elements across the thickness at most places, and thus it will require about 3,000 elements and 5,000 nodes.

The force and moment components due to crossover configuration, together with pressure loading, will be applied. The pressure loading will be enveloped in a conservative manner. The loadings will be calculated using the results of the research to be conducted during earlier phases of this project. A final detailed report will be prepared.

25.0 ROTORDYNAMIC ANALYSES

25.1 Introduction

The proposed rotordynamic analyses include calculation of critical speeds, system instability, and unbalance response of a pump rotating element without consideration of the pedestal, pump case, or foundation.

A computer program such as ROTSTB (Report No. UVA/464761/ME76/133, Department of Mechanical Engineering, School of Engineering and Applied Science, University of Virginia, Charlottesville, Virginia, December 1976) will be employed to analyze the critical-speed frequency and system instability of the rotating element. For unbalance response analysis, a computer program such as RESP2V3 (Report No. UVA/464761/MAE78/150, Department of Mechanical and Aerospace Engineering, School of Engineering and Applied Science, University of Virginia, Charlottesville, Virginia, March 1978) will be used.

25.2 Lateral Critical Speed Analysis

Lateral critical speed analysis is to be made on the pump shaft-bearing system. The program ROTSTB employs linear transfer functions and uses a modified Myklestad-Prohl method of analysis. The method of analysis assumes a rotor of uniform or nonuniform flexural rigidity to be represented by a large number of massless sections, with concentrated masses located between the massless sections. The method allows each bearing and each support to be represented by eight linear stiffness and damping coefficients. In addition, aerodynamic cross-couplings and hysteretic rotor internal friction damping effects can be included.

25.3 Rotor System Instability

A rotor and bearing system can become unstable as a result of self-excited vibration. Elements that are known to contribute to such self-excited vibrations are the following:

- a. Fluid-film bearings

- b. Aerodynamic components
- c. Shaft hysteresis
- d. Gyroscopic effect

Self-excited vibration phenomena are characterized by an available supply of energy and by a zero or negatively damped system. Rotor/bearing systems that include hydrodynamic fluid film bearings are generally positively damped. However, the ability of such bearings to dissipate energy is a function both of their design (type, dimensions, operating conditions, etc.) and their location along the rotor axis. Thus, positive damping of the system cannot a priori be assured. When the overall damping capacity of the system exceeds excitation, energy flows out of the system and stability of the rotor are assured. When excitation exceeds the capacity of the system to dissipate the energy, the net influx or accumulation of energy rapidly produces rotor instability. For this type of system, a computer program such as ROTSTB is used to determine the logarithmic decrement as a function of the system damping coefficients and frequency. The logarithmic decrement is an indicator of the degree of stability of the system.

25.4 Unbalance Response Analysis

The pump-driver rotor-bearing system will be used for unbalance response analysis by using a program such as RESP2V3. Such programs are capable of calculating the steady-state unbalance response of a linear rotor system consisting of either one or two flexible rotating or nonrotating members. The response amplitude is assumed to be synchronous with the cyclic forces generated by mass unbalance in the rotating member. Since a significant unbalance force may be transferred from the driver rotor, the mathematical model will include the pump rotor-driver rotor-bearing system, using a flexible coupling between pump rotor and driver rotor. Reaction loads of the bearings and wear rings will be evaluated based on their respective actual clearances and allowable loads.

PUMP SPECIFICATION GUIDELINES (26.0 through 29.0)

26.0 INSTRUMENTATION FOR MONITORING AND TROUBLESHOOTING OPERATING FEED PUMPS

26.1 Introduction

The performance of centrifugal pumps handling noncompressible fluids is different from that of centrifugal compressors and gas or steam turbines. The vibration levels for acceptable operation are not comparable. Pump vibrations contain signals caused by hydraulic turbulence and other phenomena that can give rise to a wide range of vibration signals not necessarily associated with the rotating speed or its harmonics. For instance, the turbulence of fluid flowing around elbows in the piping causes pipe reactions that are mechanically telegraphed to the pump case and can show up on pump bearings. Thus vibration analysis of the bearing or shaft orbit can show very low-frequency signals (which might seem excessive), caused by the mechanical resonance of the piping rather than by a problem originating within the pump.

With such a wide range of signals present, simply measuring the overall vibration level is insufficient to realistically determine pump reliability. A complete frequency analysis is required, and trends or changes in the vibration response at different frequencies must be considered in evaluating the condition of the pump. The vibration performance will also vary with system demand and, unless this is considered, unnecessary alarms could result.

26.2 Instrumentation

Signals pointing to possible trouble would be those indicating vibration at the bearings. Such vibration can be detected by means of accelerometers, velocity transducers, or proximity sensors, which sense the relative motion between shaft and bearing. Instrumentation to detect axial vibration, axial motion, and running position is also of value. Bearing temperature is considered by users to be a reliable indicator of bearing performance. Bearings should therefore be monitored, preferably by thermocouples installed in the bearing

material to provide rapid response to bearing distress.

System monitoring is also important to assure that the pump is not run against closed or stuck valves, and that adequate suction pressure is maintained at all times. Also, suction and discharge pressure, flow, and temperature should be monitored so that vibration and bearing temperature data can be correlated with system demand.

26.3 Data Analysis

From a consideration of the above, it is apparent that a computer coupled to a real-time frequency analyzer and trend monitor would be required. Therefore, specific frequencies of interest could be followed and trends and limits established only after the pump has been installed and operated in the complete system.

In the near future the rapid development of electronics and instrumentation will provide even more powerful diagnostic equipment. The proposed research into pump and hydraulic phenomena will provide a better understanding of diagnostic methods for monitoring pump health and for predicting the necessity and timing of maintenance and/or emergency shutdown of the pump.

27.0 SELECTION OF THE MATERIALS OF CONSTRUCTION

27.1 Introduction

As the research produces a better understanding of the mechanisms that produce unsteady forces and of the magnitude of these forces, and as various component development programs advance, it will be necessary to review the selection of the materials of construction for each component.

27.2 Selection Practices

A thorough review of the practices for the selection of materials of construction for large feed pumps must be made on a part-by-part basis by a qualified metallurgical materials specialist.

Information gained from the research, such as the importance of metallurgy to cavitation damage rate, fatigue failures in the highly stressed areas of impellers and diffuser vane tips, shaft failures, erosion of internal high-pressure seals, and wear and/or erosion of wear rings and other annular pressure-reducing devices, must be carefully scrutinized. Selection of materials of construction should be based on a benefit/cost analysis, using the pump evaluation methods described in paragraph 19 of this section. For example, the relative merits of conventional sand castings as compared to ceramic mold castings or investment castings should be thoroughly evaluated.

28.0 REVIEW AND UPDATING OF TEST CODES FOR FEED PUMPS

28.1 Introduction

Test codes both for factory testing and field testing of feed pumps should be reviewed and updated in light of the results of the research.

28.2 Identified Problems

Currently identified problems relate primarily to the validity of prevailing NPSH test methods, the necessity of severe low NPSH testing to simulate pump suction upsets and transients, and the adequacy of present specifications for vibration or dynamics testing.

29.0 REVIEW AND UPDATING OF STANDARDS FOR FEED PUMP APPLICATIONS

29.1 Introduction

A review and updating of standards for feed pump applications, subsequent to the completed research and incorporating the results of the research, are needed.

29.2 Identified Problems

Feed pump problems identified in Section 5, paragraph 1, as important for Phase III activity, and applying to this task, are as

follows:

1. Procurement practices such as emphasis on high quoted efficiency, low cost, low required NPSH, and low minimum flow
2. Incomplete or incorrect information given to vendors
3. Comprehensive testing not specified or enforced
4. Lack of spare pump on stand-by
5. Operation on turbine turning gear
6. Lack of communication among people from utilities, architect/engineers, vendor service, and vendor design engineers
7. Inadequate technical service manuals
8. For feed pumps, better definition of short-duration transient or upset low- and high-flow conditions, as opposed to long-duration operation at minimum or maximum pump flow
9. Need for a study of specified minimum flow as relates to actual practical plant requirements
10. Better definition of required maximum feed pump flow
11. Further study for the proper selection of the size of the start-up/spare feed pumps
12. Further clarification of the subject of NPSH margin

13. More specific information on the design of suction piping to provide uniform velocity distribution to the pump suction
14. More readily available information on discharge piping resonance and water hammer
15. Investigation of methods for the evaluation of quoted pump efficiency
16. Further investigation of formal analysis of feed pump reliability
17. Further study of the subject of shaft seals and related injection systems

29.2 Background Material

Background material to be utilized for this task will include the "Pro Forma Technical Specification, Turbine-Driven Boiler Feed Pumps for Large Fossil-Fired Power Plants," from Appendix B of this report, and EPRI CS-1512, "Recommended Design Guidelines for Feed Water Pumps in Large Power Generating Units." The material described in paragraphs 26, 27, and 28 above has been separated in this report for special emphasis, but will be integrated into this task.

TROUBLESHOOTING AIDS (30.0)

30.0 TROUBLESHOOTING MANUAL

30.1 Introduction

The purpose of the troubleshooting manual is to aid in the identification, diagnosis, definition, and resolution of problems experienced with existing feed pumps. Included should be methods, analytical tools, and experimental techniques that will be useful in problem solving.

30.2 Identified Problems

Problems identified in Section 5, paragraph 1, relating to this task, are as follows:

1. Inadequate spare rotating elements and spare parts
2. Operating procedures
3. Failure to monitor and evaluate pump operating parameters
4. Lack of control over pump repairs
5. Inadequate response by vendor service people
6. Inadequate technical service manuals

30.3 Preparation of Manual

The preparation of the manual will include the integration of all research results as applied to the troubleshooting, maintenance, repair, and service of existing pumps and related systems. The knowledge gained in the research will be presented in a manner that will be understandable and usable by the personnel responsible for operation, maintenance, and modification of pumps and pumping systems. It will include economic evaluation methods, using the methodology described in paragraph 19 of this section, to help the personnel involved appreciate the economic benefits of better operating practices, better maintenance procedures, and modifications that upgrade the feed pumps and related systems. In other words, it will be a trouble-prevention as well as a troubleshooting manual.

Section 7

DETERMINATION OF BENEFITS

1.0 OBJECTIVE

The objective of this task is to predict the anticipated benefits to the utility industry from the completion of the planned research and development tasks, Phase II and Phase III of the contemplated research program.

2.0 APPROACH

The total anticipated benefits derived by the utility industry from the performance of the research and development are computed and compared to the rough-order-of-magnitude estimated costs of the research. A benefit/cost ratio is computed. The base year used is 1987.

A tabulation is prepared, and importance factors are assigned to the interrelationship between each identified feed pump problem and each proposed research topic. Each importance factor relates the relative economic importance of the research topic to the feed pump problem and the importance of that particular feed pump problem to overall feed pump reliability.

A failure mode weighting factor is then computed by dividing the importance factor total for each failure mode by the total of the importance factors for all failure modes.

The estimated cost of each failure mode in the base year of 1987 is then computed by multiplying the failure mode weighting factor by the total estimated cost of feed pump failures during the base year.

For each research category a dependency factor is assigned reflecting the dependence of the later tasks, such as development of design methods, on the more basic research tasks that must provide the required fundamental knowledge. For example, hydraulic research tasks are basic and have a dependency factor of 1.8, while computer-aided design methods are dependent and have a dependency factor of 1.1.

The importance factor previously determined for each research category is multiplied by the probability of success and by the dependency factor to obtain a weighting factor for each research category. The cost-effectiveness index for each research category is then obtained by dividing the estimated research cost in 1981 dollars by the weighting factor for that research category.

The weighted benefit/cost ratio is then obtained by multiplying the overall benefit/cost ratio by the average cost-effectiveness index, and then dividing by the cost-effectiveness index for each research category.

3.0 RESEARCH PROGRAM BENEFIT/COST RATIO

It is expected that Phases II and III of the research program will run from January 1983 through December 1987. Total research costs are estimated to be \$5.2 million 1981 dollars. This is a rough-order-of-magnitude estimate.

The inflation rate in the years from 1980 through 1997 is assumed to be a constant 10% annually. The research is assumed to end in 1987. The economic benefits of the research are assumed to begin in 1987 and last for a period of ten years. The base year for computing benefit/cost ratio is 1987.

The \$5.2 million 1981 dollars research cost is \$9.2 million in 1987 dollars.

Replacement energy costs alone, caused by feed pump problems, were \$408 million in 1981 (source: EPRI, Mr. Isidro Diaz-Tous, 23 September 1981). This would amount to \$723 million per year in 1987 dollars.

It is assumed that 20% of the failure costs will be eliminated through natural progress by 1987, even if EPRI does not sponsor the proposed research. Projected 1987 failure costs, therefore, would be \$578 million, and it is assumed that EPRI-sponsored research will result in a net savings of 40% of that \$578 million per year for ten years starting in 1987. The present value in 1987 of this \$231 million per year for the 10 years from 1987 to 1997 is \$1,419 million.

The research program benefit/cost ratio for Phases II and III is \$1,419 million/\$9.2 million, or 154:1.

4.0 TABULATION OF IMPORTANCE FACTORS

Figure 7-1 is a tabulation of importance factors. An importance factor from 0 to 10 has been assigned to the economic relationship between each research topic and each failure mode, with the maximum possible impact of research on that particular failure mode being rated 10. If the research topic has no relationship to a particular failure mode, the importance factor is then zero. The importance factors for Phase III activity were believed to be constant, and a factor of 5 was assigned in all cases to design methods and a factor of 4 to specification guidelines and to troubleshooting. The importance factor total for each research topic and for each failure mode is the sum of all individual importance factors in that column or line. The failure mode importance factor totals were translated into failure mode weighting factors by dividing each individual importance factor total by the grand total of all importance factors.

5.0 ESTIMATION OF COST FOR EACH FAILURE MODE

The estimated cost of each failure mode is obtained by multiplying the total estimated 1987 failure costs of \$578 million by the failure mode weighting factor, which was calculated as explained in paragraph 4.0 above. Figure 7-2 shows the estimated cost of each failure mode in millions of 1987 dollars.

FAILURE MODES	HYDRAU- LICS	CAVI- TA- TION	ROTOR DYNAM- ICS	PUMP SYS- TEMS	FULL-SCALE VERIFI- CATION	APPLICA- TION OF RESULTS	DESIGN METHODS	SPECIFI- CATION/ LINES	TROUBLE- SHOOTING	IMPOR- TANCE FACTOR TOTAL	FAILURE MODE WEIGHTING FACTOR
	Tasks 1,2, 3 & 4	Tasks 5,6, 7,8,	Tasks 10,11, 12 & 13	Tasks 14,15	Tasks 16 & 17	Tasks 18 & 19	Tasks 20,21, 22,23, 24, & 25	Tasks 26,27, 28, & 29	Task 30		
1. VIBRATION	10	10	9	7	7	7	5	4	4	63	.093
2. IMPELLER BREAKAGE OR CRACKING	10	10	7	4	7	7	5	4	4	58	.085
3. SHAFT SEAL FAILURE	7	7	8	8	7	7	5	4	4	57	.084
4. RAPID WEAR OF WEAR RINGS	8	6	7	7	7	7	5	4	4	55	.081
5. CAVITATION DAMAGE	8	10	2	7	6	6	5	4	4	52	.077
6. AXIAL BALANCING DEVICE FAILURE	8	8	4	6	7	6	5	4	4	52	.077
7. BROKEN OR DAMAGED SHAFT	6	6	7	5	7	7	5	4	4	51	.075
8. JOURNAL BEARING FAILURE	7	4	6	5	7	7	5	4	4	49	.072
9. SEIZURES OF WEAR RINGS, ETC.	9	5	6	4	6	6	5	4	4	49	.072
10. THRUST BEARING FAILURE	9	6	2	5	7	6	5	4	4	48	.071
11. UNSTABLE HEAD CURVE	10	5	2	2	7	7	5	4	4	46	.068
12. AUXILIARY SYSTEM RELIABILITY	3	3	2	7	7	7	5	4	4	42	.062
13. HOT MISALIGNMENT	0	0	6	4	2	6	5	4	4	31	.045
14. GEAR-TYPE COUPLINGS	0	0	3	3	3	4	5	4	4	26	.038
IMPORTANCE FACTOR, TOTAL	95	80	70	74	87	90	70	56	56	679	

FIGURE 7-1. Tabulation of Importance Factors

FAILURE MODES

	<u>FAILURE MODE WEIGHTING FACTOR</u>	<u>ESTIMATED COST IN 1987</u>
		(Millions of Dollars)
1. VIBRATION	.093	53.8
2. IMPELLER BREAKAGE OR CRACKING	.085	49.2
3. SHAFT SEAL FAILURE	.084	48.6
4. RAPID WEAR OF WEAR RINGS	.081	46.8
5. CAVITATION DAMAGE	.077	44.5
6. AXIAL BALANCING DEVICE FAILURE	.077	44.5
7. BROKEN OR DAMAGED SHAFT	.075	43.3
8. JOURNAL BEARING FAILURE	.072	41.6
9. SEIZURES OF WEAR RINGS, ETC.	.072	41.6
10. THRUST BEARING FAILURE	.071	41.0
11. UNSTABLE HEAD CURVE	.068	39.3
12. AUXILIARY SYSTEM RELIABILITY	.062	35.8
13. HOT MISALIGNMENT	.045	26.0
14. GEAR-TYPE COUPLINGS	.038	22.0
	TOTAL	578

FIGURE 7-2. Estimated Cost of Each Failure Mode

6.0 WEIGHTED BENEFIT/COST RATIO

In order to obtain a weighted benefit/cost ratio, the probability of success was estimated for each research category and multiplied by the previously established importance factor and the dependency factor to obtain a weighting factor for each research category. A cost-effectiveness index then was computed by dividing the estimated research cost in 1981 dollars by the weighting factor for that particular research category. The weighted benefit/cost ratio is then the average cost-effectiveness index for all research categories (6.433) divided by the cost-effectiveness index for a particular research category and multiplied by the overall benefit/cost ratio for the entire program (154:1). Figure 7-3 shows the weighted benefit/cost ratio for each research category. The benefit/cost ratio for each category is 103:1 or greater.

7.0 COMMENTS AND CONCLUSIONS

The weighted benefit/cost ratio shown in Figure 7-3 for rotor dynamics is distorted by the fact that tasks 10 and 11 are actually programs already in existence and funded by EPRI. A zero cost is assigned to those tasks in this analysis.

All tasks show excellent benefit/cost ratios, providing ample justification for the research.

RESEARCH CATEGORY	PROBA- BILITY OF SUCCESS	IMPOR- TANCE FACTOR	WEIGHTING FACTOR	DEPENDENCY FACTOR	R.O.M. ESTIMATE OF RESEARCH COST IN 1981 DOLLARS	COST-EFFECTIVENESS INDEX (ESTIMATED COST/WEIGHTING FACTOR)	WEIGHTED BENEFIT/ COST RATIO
HYDRAULICS TASKS 1,2,3, & 4	0.80	95	136.8	1.8	\$1,300k	9.5	104:1
CAVITATION TASKS 5,6,7,8, & 9	0.70	80	100.8	1.8	\$ 900k	8.9	111:1
ROTOR DYNAMICS TASKS 10,11,12, & 13	0.80	70	100.8	1.8	\$ 250k	2.5	396:1
PUMP SYSTEMS TASKS 14 & 15	0.65	74	72.1	1.5	\$ 400k	5.5	180:1
FULL-SCALE VERIFICATION TASKS 16 & 17	0.80	87	104.4	1.5	\$1,000k	9.6	103:1
APPLICATION OF RESEARCH RESULTS TASKS 18 & 19	0.65	90	76.0	1.3	\$ 400k	5.3	187:1
COMPUTER-AIDED DESIGN METHODS TASKS 20,21,22,23,24, & 25	0.85	70	65.4	1.1	\$ 500k	7.6	130:1
PUMP SPECIFICATION GUIDELINES TASKS 26,27,28, & 29	0.90	56	50.4	1.0	\$ 300k	6.0	165:1
TROUBLESHOOTING AIDS TASK 30	0.90	56	50.4	1.0	\$ 150k	3.0	330:1

*NOTE: WEIGHTING FACTOR = PROBABILITY OF SUCCESS X IMPORTANCE FACTOR X DEPENDENCY FACTOR

FIGURE 7-3. Weighted Benefit/Cost Ratio for Each Research Category