NRC FORM 335 (7-77)

U.S. NUCLEAR REGULATORY COMMISSION

BIBLIOGRAPHIC DATA SHEET

1. REPORT NUMBER (Assigned by DOC)
NUREG/CR-2792

4. TITLE AND SUBTITLE (Add Volume No., if appropriate)
An Assessment of Residual Heat Removal and Containment Spray Pump Performance Under Air and Debris Ingesting Conditions

2. (Leave blank)

3. RECIPIENT'S ACCESSION NO.

7. AUTHOR(S)
P. S. Kamath, T. J. Tantillo and W. L. Swift

5. DATE REPORT COMPLETED
MONTH July | YEAR 1982

9. PERFORMING ORGANIZATION NAME AND MAILING ADDRESS (Include Zip Code)
Creare Incorporated
Hanover, New Hampshire 03755

DATE REPORT ISSUED
MONTH September | YEAR 1982

6. (Leave blank)

8. (Leave blank)

12. SPONSORING ORGANIZATION NAME AND MAILING ADDRESS (Include Zip Code)
US Nuclear Regulatory Commission
Ofc. of Nuclear Reactor Regulation
Division of Safety and Technology
Washington, DC 20555

10. PROJECT/TASK/WORK UNIT NO.
A-1237 and A-1296

11. CONTRACT NO.
FIN No. A1296

13. TYPE OF REPORT
Technical Evaluation Report

PERIOD COVERED (Inclusive dates)

15. SUPPLEMENTARY NOTES

14. (Leave blank)

16. ABSTRACT (200 words or less)

This report presents an assessment of the performance of Residual Heat Removal (RHR) and Containment Spray (CS) pumps during the recirculation phase of reactor core and containment cooldown following a Loss-of-Coolant Accident (LOCA). The pumped fluid is expected to contain debris such as insulation and may ingest air depending on sump conditions.

Findings show that for pumps at normal flow rates operating with sufficient Net Positive Suction Head (NSPH), pump performance degradation is negligible if air ingestion quantities are less than 2% by volume. If air ingestion quantities exceed this amount, degradation may be severe depending on pump design, speed and flow rate. Small quantities of air will increase NPSH requirements for these pumps. For the types and quantities of debris likely to be present in the recirculating fluid, pump performance degradation is expected to be negligible. In the event of shaft seal failure due to wear of loss of cooling fluid, seal safety bushings limit leakage rates.

17. KEY WORDS AND DOCUMENT ANALYSIS

17a. DESCRIPTORS

17b. IDENTIFIERS/OPEN-ENDED TERMS

18. AVAILABILITY STATEMENT
Unlimited

19. SECURITY CLASS (This report)
Unclassified

20. SECURITY CLASS (This page)
Unclassified

21. NO. OF PAGES

22. PRICE
$

NRC FORM 335 (7-77)

UNITED STATES
NUCLEAR REGULATORY COMMISSION
WASHINGTON, D.C. 20555

OFFICIAL BUSINESS
PENALTY FOR PRIVATE USE, $300

FOURTH CLASS MAIL
POSTAGE & FEES PAID
USNRC
WASH D C
PERMIT No G 67

120555078877 1 ANA19U9X1S
US NRC
ADM DIV OF TIDC
POLICY & PUBLICATIONS MGT BR
PDR NUREG COPY
LA 212
WASHINGTON DC 20555

NUREG/ CR-2792
CREARE TM-825

An Assessment of Residual Heat Removal and Containment Spray Pump Performance Under Air and Debris Ingesting Conditions

Prepared by P. S. Kamath, T. J. Tantillo, W. L. Swift

Creare, Inc.

Prepared for
U.S. Nuclear Regulatory
Commission

ABSTRACT

This report presents an assessment of the performance of Residual Heat Removal (RHR) and Containment Spray (CS) pumps during the recirculation phase of reactor core and containment cooldown following a Loss-of-Coolant Accident (LOCA). The pumped fluid is expected to contain debris such as insulation and may ingest air depending on sump conditions.

Findings are based on information collected from the literature and from interviews with pump and seal manufacturers. These findings show that for pumps at normal flow rates operating with sufficient Net Positive Suction Head (NPSH), pump performance degradation is negligible if air ingestion quantities are less than 2% by volume. For air ingestion between 3% and 15% by volume, head degradation depends on individual pump design and operating conditions and for air quantities greater than 15% performance of most pumps will be fully degraded. Also, small quantities of air will increase NPSH requirements for these pumps. For the types and quantities of debris likely to be present in the recirculating fluid, pump performance degradation is expected to be negligible. In the event of shaft seal failure due to wear or loss of cooling fluid, seal safety bushings limit leakage rates.

Acknowledgements

The authors would like to acknowledge contributions from several poeple who aided in this study. Much of the information contained in this report was the direct result of personal or telephone interviews with pump and seal specialists. We would like to thank the following for taking the time to help and for the valuable insight they provided: Bill Adams, Durametallic, Inc., Jon Hamaker, Crane Co., Larry Hufford, Mechanical Seal Division of Borg-Warner Corp., John Doolin and Warren Fraser, Worthington Group-McGraw Edison Company, Fred Antumes and Phillip Nagangast, Engineered Pump Division of Ingersoll Rand, Dr. Paul Cooper, Ingersoll Rand Research, Fred Buse of Ingersoll Rand and George Morrisey, Pacific Pumps Division of Dresser Industries, Inc. We would also like to thank Pete Strom and Dr. Gil Weigand of Sandia National Laboratories for their critical comments and helpful suggestions as well as Mr. A. W. Serkiz of the NRC for help in interfacing with other efforts in this program.

CONTENTS

LIST OF FIGURES

LIST OF TABLES

LIST OF SYMBOLS AND ABBREVIATIONS

AF	Air fraction - volume flow rate of ingested air divided by total flow rate, in %
AF_p	Air fraction at pump inlet
AF_s	Air fraction at pump suction
ARL	Alden Research Laboratory
A/W	Air/water
B&R	Burns and Roe, Inc.
B&W	Babcock and Wilcox
CCS	Component Coolant System
C-E	Combustion Engineering
CHRSS	Containment Heat Removal Spray System
CS	Containment Spray
CSS	Containment Spray System
D_i	Diameter of i^{th} piping element (ft)
ECCS	Emergency Core Cooling System
Fr	Froude Number ($u/\sqrt{gs}$)
FSAR	Final Safety Analysis Review
g	Acceleration due to gravity (ft/sec^2)
g_c	Gravational constant (32.2 lbm ft/lbf sec^2)
h	Normalized pump differential head H/H_{bep}
H	Differential total head across pump (ft)
H_R	Rated pump head (ft)
H_s	Head loss through screens at Q(ft)
INEL	Idaho National Engineering Laboratory
K_i	Loss coefficient for i^{th} piping element
L_i	Length of i^{th} piping element (ft)
LOCA	Loss of Coolant Accident
n	Sump suction pipe average velocity (ft/sec)
NPSH	Net Positive Suction Head (Eq. 1-1)
NPSHR	Net Positive Suction Head Required by pump
NRC	Nuclear Regulatory Commission
N_R	Rated pump speed (rpm)
N_s	Pump specific speed (rpm $\sqrt{\text{US gpm}}/(\text{ft})^{3/4}$) (Eq. 2-1)
P_c	Containment pressure (lb/ft^2)

P_{in}	Total absolute pressure at the sump suction pipe inlet (lb/ft^2)
P_v, P_{vp}	Vapor pressure of the fluid at the pump inlet (lb/ft^2)
PWR	Pressurized Water Reactor
P_{loss}	Pressure loss in suction piping components (lb/ft^2)
P_{pa}	Absolute pressure at pump suction flange (lb/ft^2)
P_{sa}	Absolute pressure in sump at centerline of suction pipe (lb/ft^2)
P_{sg}	Absolute pressure inside the inlet of sump suction pipe (lb/ft^2)
q	Normalized pump flow rate Q/Q_{bep}
Q	Volumetric flow rate (gpm)
Q_{bep}	Pump flow rate at best efficiency (US gpm)
Q_{max}	Maximum pump flow rate specified for the plant (US gpm)
Q_R	Rated pump flow rate (US gpm)
RCS	Reactor Coolant System
RHR	Residual Heat Removal
RHRS	Residual Heat Removal System (Eq. 2-1)
RWST	Refueling Water Storage Tank
s	Sump suction pipe submergence (ft)
SL	Sandia Laboratories
S/W	Steam/water
T_w	Water temperature (°F)
USI	Unresolved Safety Issue
u	Average velocity in sump suction pipe (ft/sec)
V_p	Average velocity of pump inlet flange (ft/sec)
Z_p	Elevation of the pump inlet (ft)
Z_s	Elevation of the sump suction pipe inlet (ft)
Z_w	Elevation of liquid surface outside of screens (ft)
α	Void fraction - volumetric ratio of gas or vapor phase to total flow rate assuming no slip between phases
Δp_{loss}	Pressure losses in the pump suction piping (lb/ft^2)
Δp_{20}	Two phase pressure rise (lb/ft^2)
γ	Specific weight of the liquid at the pump inlet (lb/ft^3)
ρ_w	Water density (lb/ft^3)
ρ_a	Air density (lb/ft^3)
ρ_{fl}	Mixture density (lb/ft^3)

EXECUTIVE SUMMARY

This report presents an assessment of the performance of Residual Heat Removal (RHR) and Containment Spray (CS) pumps during operation in the recirculation mode following a postulated Loss-of-Coolant Accident (LOCA) in a Pressurized Water Reactor (PWR). It is the principal report dealing with pump air and debris ingestion. The technical findings with respect to RHR and CS pumps discussed in this report will be incorporated with results of the other related subtasks in the proposed technical resolution of the Unresolved Safety Issue (USI) A-43, "Containment Emergency Sump Performance."

Problem Description

During a postulated LOCA, water from the reactor coolant system flows out a break in the piping. Part of this water flashes into steam and fills the containment atmosphere. The rest spills onto the floor of the containment building and eventually accumulates in the containment sump.

Early in the LOCA transient, the Residual Heat Removal System (RHRS) and Containment Spray System (CSS) are aligned to draw borated water from the Refueling Water Storage Tank (RWST) located outside the containment building. The RHRS provides core cooling capability by pumping this water into the core through the cold legs, and thence out the break into the sump. The CSS sprays water into the containment atmosphere to condense steam and thus maintains the containment pressure within the containment emergency shell design pressure. In some plants, the spray fluid is a dilute solution of sodium hydroxide which serves to reduce the iodine concentration in the containment atmosphere.

When the water level in the RWST reaches a minimum level, the RHR and CS pumps are realigned to the recirculation water drawn from the containment emergency sump. This report deals with factors which may cause degraded performance of the pumps in this recirculation mode of operation.

Two principal issues are addressed. The first deals with debris, mainly from insulation, which is used abundantly on piping and components inside the containment. The concern is that debris, broken loose during a LOCA, could cause blockage of the sump or otherwise adversely affect the operation of the pumps, spray nozzles and valves of the safety systems. A program by Burns and Roe [3, 4] has identified insulation types used in plants and has resulted in a methodology for assessing debris generation and transport to the sump screens. This methodology provides a means for quantifying the impact of insulation debris on sump performance.

The second issue is related to the hydraulic performance of the sump, which may affect the hydraulic performance of the RHR and CS pumps. Adverse flow conditions in the sump resulting either from sump design or extreme blockage may induce the formation of surface vortices which can ingest air.

It is also possible that screen blockage resulting from debris may increase hydraulic losses on the suction side of the pump causing cavitation in the pump. Either of these conditions may degrade pump hydraulic performance leading to insufficient flow for core cooling. In addition, extended operation at low flow or severe cavitation may cause mechanical damage to the pump which can lead to pump failure during the long-term recirculation phase.

An extensive sump evaluation program has been conducted by Alden Research Laboratory and Sandia National Laboratories [2]. A broad range of geometric pump features and flow variables were investigated. The program was aimed at quantifying sump performance in terms of surface vortex formation, air ingestion, inlet pipe swirl and sump losses. Tests conducted under this program show that under most conditions air ingestion levels are very low, less than 0.5% by volume and that swirl in the inlet pipe decays rapidly to negligible values.

Creare Pump Performance Evaluation

The tasks performed by Creare and discussed in this report include the following:

- Survey manufacturers and review technical literature to obtain data on the effects of air and debris on the performance of the pumps.
- Review pump data from a sample of plants to identify important mechanical and hydraulic characteristics of pumps used for RHR and CS service.
- Establish types and concentrations of debris likely to be transported through the sump screens to the pumps.
- Evaluate the data on effects of air and debris on pump performance in the context of RHR and CS pumps.

Mechanical construction details and hydraulic performance characteristics for RHR and CS pumps from 12 PWR systems were evaluated with respect to the effects of air ingestion and debris on performance.

The results of air/water tests on centrifugal pumps from several separate experimental programs were applied to quantify the effects of air on head degradation and NPSH requirements for pumps. The results of this evaluation together with predicted air ingestion quantities for a sump provided by the sump test program can be used to assess RHR and CS pump performance on a plant-by-plant basis.

Estimates of debris quantities and types were used to evaluate the effects of debris on RHR and CS pump operation. Both hydraulic performance degradation and mechanical wear or malfunction were considered. Results from several experimental programs dealing with particulates in pumps were applied to RHR and CS pumps to determine the degradation in performance due to debris.

Interviews with pump and seal manufacturers were conducted to provide supporting data with respect to the effects of air and debris on pump operation. The pump specialists interviewed were:

- Mr. J. H. Doolin, Manager-Engineering, Worthington Pump Group, McGraw Edison Company
- Mr. W. H. Fraser, Chief Hydraulic Engineer, Worthington Pump Group, McGraw Edison Company
- Mr. Fred Antunes, Chief Engineer, Ingersoll-Rand
- Mr. Phillip Nagangast, Manager of Engineering Analysis, Engineered Pump Division, Ingersoll-Rand
- Dr. Paul Cooper, Ingersoll Rand Research, Inc.
- Mr. Fred Buse, Chief Engineer, Standard Pump Division, Ingersoll-Rand

These specialists affirmed that our findings from the technical literature on the air/water performance of RHR and CS pumps supported their experience, although opinions on the level of air ingestion giving negligible degradation varied from 1% to 3%. They also agreed, based on their experience in pumping slurries, sewage and sand/water mixtures and from internal tests on shrouded impellers pumping paper stock, that for the type and concentrations of debris expected in RHR and CS pumps, degradation in pump performance would be negligible.

The seal specialists interviewed were:

- Mr. Bill Adams, Director of Engineering, Durametallic, Inc.
- Mr. Jon Hamaker, Assistant Chief Engineer, Crane Packing Company

Based on test data provided by these specialists on the types of seals commonly used in RHR, CS and other auxillary and cooldown pumps, it was concluded that leakage due to possible seal wear or failure would be limited by seal safety bushings to 0.1% of the flow rate.

Creare Findings on Pump Performance

Air Ingestion

- For a wide range of operating flow rates RHR and CS pumps should handle volumetric air quantities up to 2% with negligible degradation in performance.

- For air quantities greater than 2%, performance degradation of pumps varies substantially depending on design and operating conditions.

- For very low flow rates (less than about 50% of best efficiency point) the presence of air may cause air binding in the pump. However, sump evaluations show that air ingestion is unlikely at low flows.

- Small quantities of ingested air will increase the NPSH requirements for a pump. A correction factor for NPSH requirements is proposed.

- Swirl at the pumps resulting from sump surface vortices will be negligible because of the long suction pipes between the sumps and pump inlets.

- Industrial experience and the technical literature provide corroborative data to support these findings on the behavior of pumps in air/water mixtures.

Section 4.5 of this report identifies the relevant design issues and presents a procedure for the assessment of individual systems with respect to air ingestion effects.

Debris Ingestion

- Conservative estimates of the debris which may reach the pumps in the recirculation mode of operation show that concentrations of debris should be less than 0.5% by volume.

- The debris present consists of fine abrasives and soft, fibrous insulation particles.

- Experimental data and pump and seal manufacturers' experience agree that for the types and quantities of debris present, hydraulic performance degradation of RHR and CS pumps should be negligible.

- Test data on the mechanical wear of pumps indicate that the estimated quantity of debris expected in the recirculating fluid is too small to seriously impair long-term pump operation as a result of material erosion.

- In the event of increased leakage of the shaft seals due to wear, the seal safety bushings limit leakage to less than 0.1% of pump flow rates.

Section 4 of this report provides additional discussion of the technical findings summarized above.

1 INTRODUCTION

1.1 Background

NRC Unresolved Safety Issue (USI A-43) addresses the performance of the containment sump during the recirculation phase following a Loss-of-Coolant Accident (LOCA) in a Pressurized Water Reactor (PWR). Residual Heat Removal and Containment Spray pumps draw their suction from the containment sump during this recirculation phase of operation. The performance of these pumps depends on sump operation.

The principal concerns are interrelated. They involve those factors which have the potential to affect the short or long term ability of the pumps to provide adequate cooling to the core and containment. These factors have been identified as:

- air ingestion resulting from poor sump performance,
- cavitation because of reduced net positive suction head from sump screen blockage by debris,
- mechanical erosion or failure of the pumps caused by debris.

This section of the report provides a description of the design and operation of RHRS and CSS, as well as additional background material relevant to the above topics. Design and operating details of RHR and CSS are plant specific. Thus, we caution the reader that the material on RHR and CS systems in the next two subsections is intended to provide a general description of their structure and operation.

1.1.1 Residual Heat Removal Systems (RHRS)

The purpose of the RHRS is to transfer heat from the Reactor Coolant System (RCS) to the Component Coolant System (CCS) during both normal cooldown following a shutdown and short and long-term cooldown following a LOCA. The RHRS is also used to transfer borated water between the Refueling Water Storage Tank (RWST) and the refueling cavity before and after refueling operations.

A schematic of an RHRS is shown in Figure 1-1. The principal elements of the system are two residual heat removal pumps, two residual heat exchangers and associated piping, valves and instrumentation necessary for operational control. During closed loop RHRS operation, reactor coolant flows from the primary system to the RHR pumps, through the tube-side of the residual heat exchangers, and back to the primary system. The heat is transferred to the component cooling water circulating through the shell-side of the residual heat exchangers.

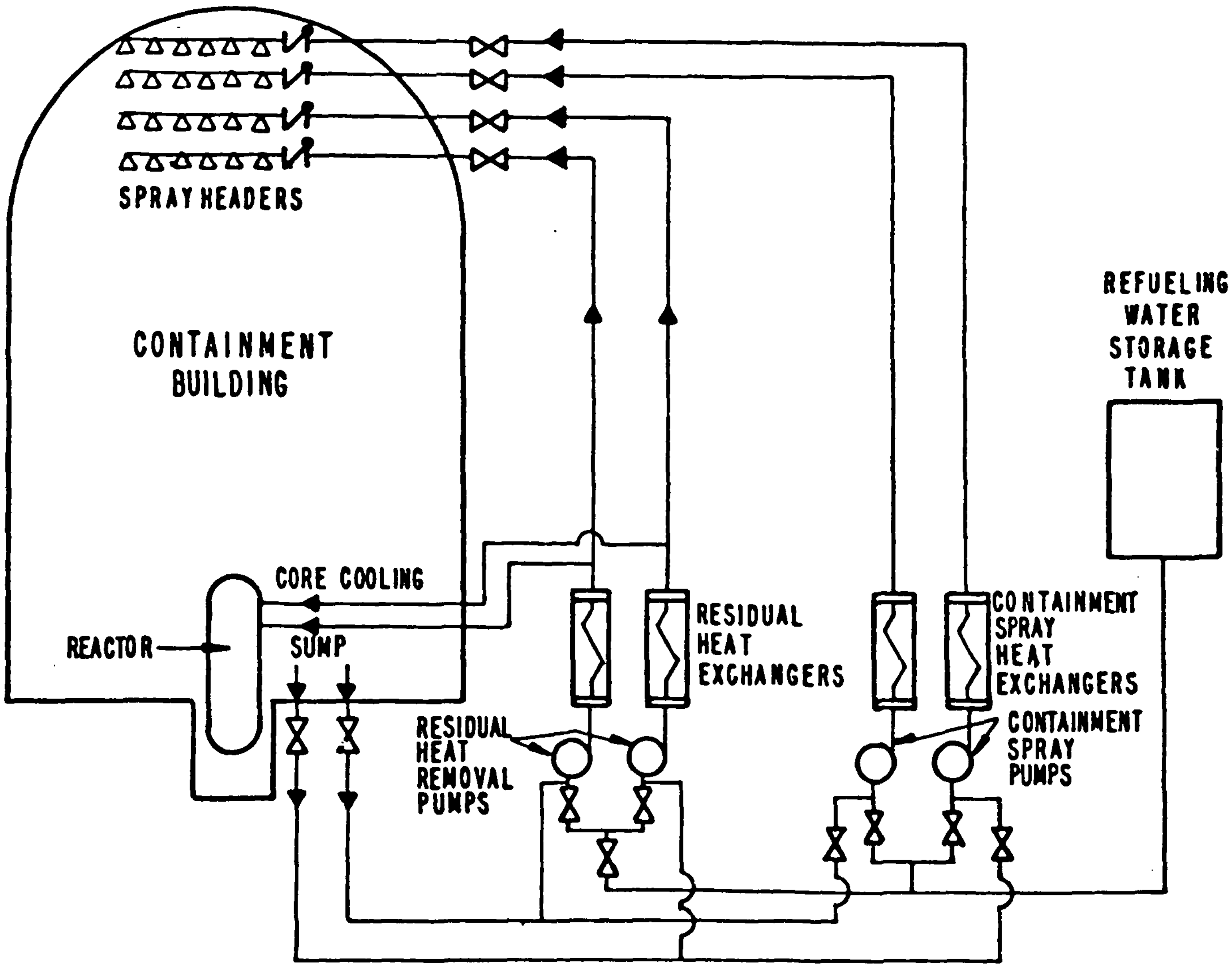

Figure 1-1. SCHEMATIC DIAGRAM OF RESIDUAL HEAT REMOVAL AND CONTAINMENT SPRAY SYSTEMS

For post-shutdown cooling, the RHRS is placed in operation approximately four hours after reactor shutdown when the temperature and pressure of the Reactor Coolant System (RCS) are approximately 350°F and 425 psia, respectively. Assuming that two heat exchangers and two pumps are in service and that each heat exchanger is supplied with component cooling water at design flow and temperature, the RHRS is designed to reduce the reactor coolant temperature from 350°F to 140°F within 16 hours. The heat load handled by the RHRS during the cooldown transient includes residual and decay heat from the core and primary coolant pump heat.

Following a LOCA, the RHRS functions as part of both the high head and low head phases of operation of the Emergency Core Cooling System (ECCS). During the high head phase, the RHRS provides suction flow to the high pressure injection pumps. During the low head phase, when the water level in the RWST has reached a minimum, the RHR pumps are realigned to draw suction from the containment sump, now filled with water that has flowed out the break. In this recirculation mode, which is of relevance to the resolution of USI A-43, the RHR pumps draw fluid from the containment sump, cool it by circulation through the residual heat exchangers and supply it directly to the core. Continuous operation of the RHR pumps in the post-break mode may be as long as one year.

1.1.2 Containment Spray System (CSS)

The CSS is a major component of the Containment Heat Removal Spray System (CHRSS). The CHRSS is designed to provide adequate containment heat removal capability following a LOCA. In the event of a LOCA the CSS sprays cool water from the RWST into the containment atmosphere to condense steam escaping from the break and thereby prevents the containment pressure from exceeding the containment shell design pressure.

The CSS, also shown in Figure 1-1 consists of two separate trains of equal capacity, each capable of independently delivering the required design flow rate. Each train includes a pump, heat exchanger, ring header with nozzles, isolation valves, associated piping, instrumentation and controls. The system which is designed to function only during a LOCA is activated by a high containment pressure signal and possibly others during a LOCA. Operation of the CS system occurs in the following sequence:

1. The containment spray pumps spray a portion of the water from the RWST into the containment atmosphere.

2. After the water level in the RWST has reached a preselected value, water from the containment sump is circulated through the containment spray heat exchangers and sprayed back into the containment atmosphere. It is this recirculation phase of operation of the CSS that is of interest in the resolution of USI A-43.

3. A portion of the recirculation flow from the RHRS may be manually diverted to additional spray headers as an added redundancy to keep containment pressure down.

It has been mentioned that RHR and CS system designs are plant specific. However, in general, the CS pumps are designed to operate only in emergency situations and are expected to operate for up to one month. On the other hand, the RHR pumps may be operated frequently during normal plant operations, and in the event of an emergency, may be required to provide cooling for a year or more.

1.2 Air Ingestion and Debris Effects

Following a LOCA, water flowing from the break accumulates in the containment sump from which the RHR and CS pumps draw their suction while operating in the recirculation mode. Conditions in the containment following a LOCA may be such that debris has migrated to the sump screens. Some debris may in fact pass through the screens and be pumped through the system. Because of the screen opening sizes, typically ¼" or less, only some types of debris are likely to pass through into the sump. Debris which collects outside the sump screens may induce large flow perturbations at the sump and may (because of blockage) increase pressure losses through the screens. Also, if sump surface vortices form, air may be ingested into the piping on the suction side of the RHR and CS pumps. These conditions may result from the combined influence of sump geometry, screen blockage from debris and low sump surface levels.

The full scale tests conducted at Alden Research Laboratory [2] on various sump geometries under a wide range of conditions have shown that under some conditions steady air ingestion rates of about 2% by volume are possible. For very severely perturbed sump inlet flows up to 7% air ingestion have been measured.

The performance of centrifugal pumps is known to degrade with increasing vapor or gas content in the fluid. The amount of degradation is a function of various parameters; the important ones being pump design, specific speed, flow rate, inlet pressure, and fluid properties. At present, the physical mechanism of degradation is not understood well enough to be modeled analytically. However, a general guideline commonly adhered to by the pump industry is that for air ingestion levels less than about 2% by volume, degradation is not a concern at normal flow rates; for air ingestion levels between 2% and 15%, performance is dependent on pump design and for air ingestion greater than 15%, most centrifugal pumps are fully degraded. A wide range of data support this guideline. It is also generally recognized that for NPSH values close to those required by the pump, air ingestion has a noticeable effect on performance.

In this study, two-phase data on pumps with characteristics similar to those of RHR and CS pumps are presented. Based on the available published data, together with manufacturers experience, guidelines for evaluating RHR and CS pump performance for known air ingestion rates are suggested.

For the evaluation of the effects of debris on the RHR and CS pumps, it is necessary to know the types and quantities of debris likely to pass through the screens and the effects of this debris on pump operation. The following are potential effects on operation:

- erosion
- corrosion
- passageway clogging
- increased leakage
- decreased hydraulic performance

Several types of debris were identified:

- Fiberglass and blanket-type insulation which has disassociated into pieces small enough to pass through the screens.
- Hydroxide precipitates--products of borated water and aluminum and zinc used for insulation encapsulation.
- Other miscellaneous suggested debris: paint chips, concrete dust

The concentrations of each type of debris used to assess pump and seal performance are based on conservative estimates. However, actual quantities and characteristics (hardness, size, etc.) are likely to be highly plant specific. For the purpose of our evaluation, volumetric concentrations of less than 1% result from our estimates and consist of hard small (10μ) sized abrasives and of soft, fibrous material. Direct comparison is made with pumping experience from slurry technology and with pumping of fibrous paper stock.

1.3 Cavitation

Cavitation is the formation of vapor-filled cavities in a liquid when the local static pressure falls below the local vapor pressure. In pumps, cavitation is most likely to occur at the inlet to the blades of the impeller where the static pressure is the lowest. Cavitation in the pump is undesirable not only because it can alter the flow pattern and thus degrade pump performance, but also because collapsing cavities cause noise, vibration and mechanical damage to the impeller.

To avoid cavitation in pumps, the Net Positive Suction Head (NPSH) available at the pump inlet should be at least as large as the NPSH required, NPSHR. If it is not, cavitation is sure to occur, erosion may be severe and performance is certain to be degraded by some amount. The available NPSH is determined using the equation,

$$NPSH = \frac{p_{in} - (\Delta p)_{loss} - p_v}{\gamma} + Z_s - Z_p \tag{1.1}$$

where

p_{in} = total pressure (absolute) at the inlet of the sump suction pipe

p_v = vapor pressure of the fluid at the pump inlet

$(\Delta p)_{loss}$ = pressure losses in the suction piping

Z_s = elevation of the suction pipe inlet

Z_p = elevation of the pump inlet

γ = specific weight of liquid at inlet to pump.

As can be seen from Equation (1.1) the available NPSH is a calculated parameter and is subject to error due to the uncertainties in estimating piping losses. The NPSHR for the pump is determined by tests at various flow rates by the manufacturer and/or in the plant.

No standards exist for establishing the NPSHR or the margin between the NPSHR and the calculated NPSH available. However, it is common practice in industry to define the NPSHR as that value of the NPSH at which the head developed by the pump has degraded by some percentage of the non-cavitating head. The percentage varies from 1% to 3% and is generally specified by the purchaser of the pump. Conservatism in the margin between the NPSHR and the NPSH available are left to the system designers.

If the proper design techniques are followed, cavitation, in the absence of air ingestion, is not likely to occur in RHR and CS pumps. Regulatory Guide 1.1 [6] requires that adequate NPSH be provided for the system pumps assuming maximum expected temperatures of pumped fluids and no increase in pressure from that present prior to a postulated LOCA. Preoperational tests of the Emergency Core Cooling Systems (ECCS) and components as specified in Regulatory Guide 1.79 [7] will demonstrate in situ that adequate NPSH is available.

1.4 Combined Effects of Air Ingestion and Cavitation

Air ingestion has been found to affect the NPSHR in pumps. The number of references which provide documented data on the combined effect of air and cavitation are few. In this report, data are evaluated with respect to RHR and CS pumps, and guidelines are suggested for evaluation of pump performance under their combined effects in Section 4.5. At low NPSH values, close to the NPSHR, air ingestion will increase the degradation in performance in comparison to operation in the absence of air. Hence, although a pump has been designed with sufficient suction pressure to operate free of cavitation in the absence of air, the presence of air will induce some degradation in performance. The amount of degradation depends on the quantity of air and on the difference between the available NPSH and NPSH required.

1.5 Structure of the Report

The basic performance characteristics and construction features of RHR and CS pumps are reviewed in Section 2. Pump types used in twelve PWR plants are identified. Although pump specifications vary from plant to plant, several features emerge as fairly common among all plants. The assessment of the effects of air and debris on RHR and CS pump operation is based on data from pumps of similar construction and specific speed.

Information on the behavior of pumps under two-phase flow conditions are reviewed in Section 3. The data base for air/water and vapor/liquid flow situations consists of experimental results from several pumps operating in air/water and in steam/water conditions. These data are reviewed with respect to their applicability to RHR and CS pumps. Section 3 summarizes the methods used to estimate the quantities and types of debris likely to be present. The results of several experimental programs in which the performance of centrifugal pumps under particulate ingesting conditions are presented.

The application of technical findings to RHR and CS pumps is presented in Section 4. The effects of debris, air ingestion and swirl at levels expected during post-LOCA recirculation are discussed. Criteria for acceptable inlet conditions for RHR and CS pumps are suggested and a methodology for assessing pump inlet conditions is outlined. The Conclusions from this study are presented in Section 5.

2 RHR AND CS PUMPS

This section summarizes important characteristics of RHR and CS pumps. The information presented here has been obtained from existing plants, pump specialists and manufacturers of pumps and seals. In general, the data collected is a sample and provides some feeling for both plant-to-plant and manufacturer-to-manufacturer similarities and differences. In no way should the data collected be construed to be representative of all CS or RHR pumps. We have tried to be complete and accurate where adequate information was available and general in areas where our intent was to group similarities between components to be of use in assessing their behavior.

The section is divided into four major topics:

1. Plants Reviewed
2. Mechanical Details of Pumps
3. Hydraulic Performance Characteristics
4. Operating Considerations.

The first section identifies the plants surveyed and the information obtained about the pumps used in each. The second summarizes mechanical construction details of several of the pumps. Major features such as materials used, sealing methods, sizes, clearances are provided. The third section summarizes hydraulic performance characteristics and NPSH requirements of pumps for which this information is available. The final section gives a brief discussion dealing with some of the practical considerations in the operation of these pumps in the RHR and CS systems.

2.1 Reactor Plants Reviewed

Data on the RHR and CS pumps were collected from twelve PWR plants. References [3] and [4] contain information about the plant designs and, in particular, locations and configurations of sumps which serve as intakes for the RHRS and CSS in the recirculation mode. The pump manufacturer and model identification for each pump are listed in Table 2-1 together with rated conditions for each pump. Pump specifications and rating points are plant specific. However, the pumps generally used in these applications are similar. The final column in Table 2-1 lists the specific speed for each pump at rated conditions. Specific speed N_s is defined as:

$$N_s = NQ^{1/2}/H^{3/4} \tag{2.1}$$

where N is shaft speed in rpm, Q is volumetric flow rate in US gpm, and H is the pump differential head in feet. The specific speed value for a pump provides a "type number" conventionally used by the pump industry to roughly characterize pump designs. All RHR and CS pumps evaluated are of relatively low specific speed (800-1600) implying relatively high head centrifugal pumps with radial impellers.

TABLE 2-1

RHR AND CS PUMP DATA

PLANT	MANUFACTURER*/MODEL		RATED CONDITIONS			
	RHR	CS	(RPM) SPEED	(FT) HEAD	(GPM) FLOW	SPECIFIC SPEED
Arkansas Unit #2	I-R/8x20 WD		1780	350	3100	1225
		I-R/6x23 WD	1780	525	2200	760
Calvert Cliffs 1&2	I-R/8x21 AL		1780	350	3000	1205
		B&W/6x8x11 HSMJ	3580	375	1350	1544
Crystal River #3	W/8-HN-194		1780	350	3000	1205
		W/6-HND-134	3550	450	1500	1407
Ginna	Pac/6" LSVC		1770	280	1560	1016
Haddam Neck	Pac/8" LXSVCR		1770	300	2200	1152
Kewaunee	B-J/6x10x18 VDSM		1770	280	2000	1156
		I-R/4x11 AN	3550	475	1300	1257
McGuire 1&2	I-R/8x20 WD		1780	375	3000	1144
		I-R/8x20 WD	1780	380	3400	1205
Midland #2	B&W/10x12x21 KSMK		1780	370	3000	1156
		B&W/6x8x13 SMK	3550	387	1300	1467
Millstone Unit 2	I-R/(No Model #)		1770	350	3000	1198
		G/3736-4x6-13	3560	477	1400	1370
Oconee #3	I-R/8x21 AL		1780	360	3000	1180
		I-R/4x11A	3550	460	1490	1380
Prairie Island 1&2	B-J/6x10x18 VDSM		1770	280	2000	1156
		I-R/4x11 AN	3550	500	1300	1210
Salem #1	I-R/8x20W		1780	350	3000	1205
		G/3415 8X10-22	1780	450	2600	929

* Pac - Pacific
I-R - Ingersoll-Rand
W - Worthington
G - Gould
B&W - Babcock & Wilcox
B-J - Byron Jackson

Specific Speed is defined as $N_s = N_R (Q_R)^{1/2}/(H_R)^{3/4}$

An important feature of CS and RHR pump operations has to do with their relative location with respect to the containment sump. RHR and CS pumps are located outside the containment and are connected to the sump within the containment by piping. For the plants reviewed, pump suction piping is typically about 40' in distance from the sump to the pump with diameters of 14"-16" at the sump, reducing to values of about 6"-8" at the pump inlets. Suction p ping also contains several elbows, reducing sections and at least one valve. None of the pumps evaluated were close-coupled (by suction piping) to the sump. Tests from the ARL sump studies show that swirl from sump vortices decays to a negligible amount within 14 pipe diameters from the pipe inlet. Therefore, for the pumps identified in this study, swirl due to sump surface vortices should have a negligible effect on performance. However, if pumps are either submerged in the sump or very closely-coupled to the sump, inlet swirl to the pump resulting from sump surface vortices should be given serious consideration during design.

2.2 Mechanical Details of RHR and CS Pumps

The previous section has identified that all pumps reviewed had rated operating conditions falling in the range of specific speeds of 800-1600 implying that the pumps were relatively high head designs with radial impellers. In this section, similarities in the mechanical construction for these pumps will be discussed. Detailed mechanical information on the pumps is less complete than are the rated operating conditions because some of the mechanical details (such as seal manufacturer and type) are available only on the original order specifications.

The details which have been accumulated provide a generally consistent picture with respect to several important features. The pumps in use for RHR and CS service are generally of robust construction. They have been designed and manufactured to provide dependable service under relatively severe operating conditions (although the actual operating conditions are not as severe as many process applications in which similar pumps perform). Reference [8] provides insight from one manufacturer on the evolution and development of the type pump commonly specified today for safety system pumps.

2.2.1 Overall Construction Details

Individual pump features are specified and selected by plant reactor manufacturers. These specifications include the hydraulic performance characteristics (rated head, flow, NPSHR, etc.) as well as materials of construction and shaft seal systems. Figure 2-1 shows cross section assemblies of two pumps typical of those identified in the plant survey. Important elements of each are identified in the figure. The two pump assemblies show a horizontal shaft overhung type with oil lubricated ball bearings in the pump frame, and a vertical shaft pump in which the ball bearings are the permanent lubricated type located in the motor drive chassis. All pumps identified in the plant surveys were single-stage vertical and horizontal pumps. All pumps had shrouded (closed) impellers and wearing rings at the impeller inlet. Most impellers were single suction with impeller discharge diameters in the range of 12" to 20". No multistage pumps were identified nor were any mixed flow or axial-flow pumps.

Table 2-2 lists the pump models by plant and the main mechanical features of each. All pumps have mechanical seals and stainless steel impellers and casings. The following paragraphs discuss individual features in more detail.

2.2.2 Materials

The materials used for impellers are listed in Table 2-2(a) and 2-2(b). Interviews with manufacturers revealed that the materials in the table are representative of the class of materials generally used for these pumps. In general, impellers and casings are of austenitic 300-series stainless steels. These materials are chosen for their high resistance to corrosion and to erosion resulting from cavitation.

2.2.3 Shaft Seals

Shaft seals are of the heavy-duty, mechanical type characterized by continuous contact between two mating annular faces. One face is secured to the stationary housing while the other is attached to the rotating shaft. Seal cooling is provided by either closed loop circulation of water through a heat exchanger or by open loop circulation of some water from the pump discharge. In the former configuration, a pumping ring in the seal assembly (fastened to the shaft) drives the cooling fluid through a heat exchanger. No filters or other line obstructions are present in the circuit. In the case of open loop cooling, a cyclone separator may or may not be included in the cooling water line upstream of the seal inlet port. Filters are not used.

Figure 2-2 illustrates two mechanical shaft seal assemblies typical of those used in RHR and CS pump applications. The first, shown in Figure 2-2a, illustrates an assembly in which the coolant is circulated by the pumping ring through an external heat exchanger and through the seal. The spring provides a face load by pushing the stationary washer against the rotating seat. The bellows provides a secondary seal. Seat material is generally tungsten carbide and the washer is a carbon graphite.

Figure 2-2b shows an alternate mechanical shaft seal arrangement in which coolant is extracted from the high pressure discharge end of the pump and recirculated through the seal. In this particular arrangement, the rotating washer is loaded against the stationary seat.

Seals of the types shown are rugged in their construction and capable of operating at elevated temperatures - typically rated at temperatures up to 400°F. Inlet and exit port sizes for coolant flushing are usually 3/16" to 1/4" diameter.

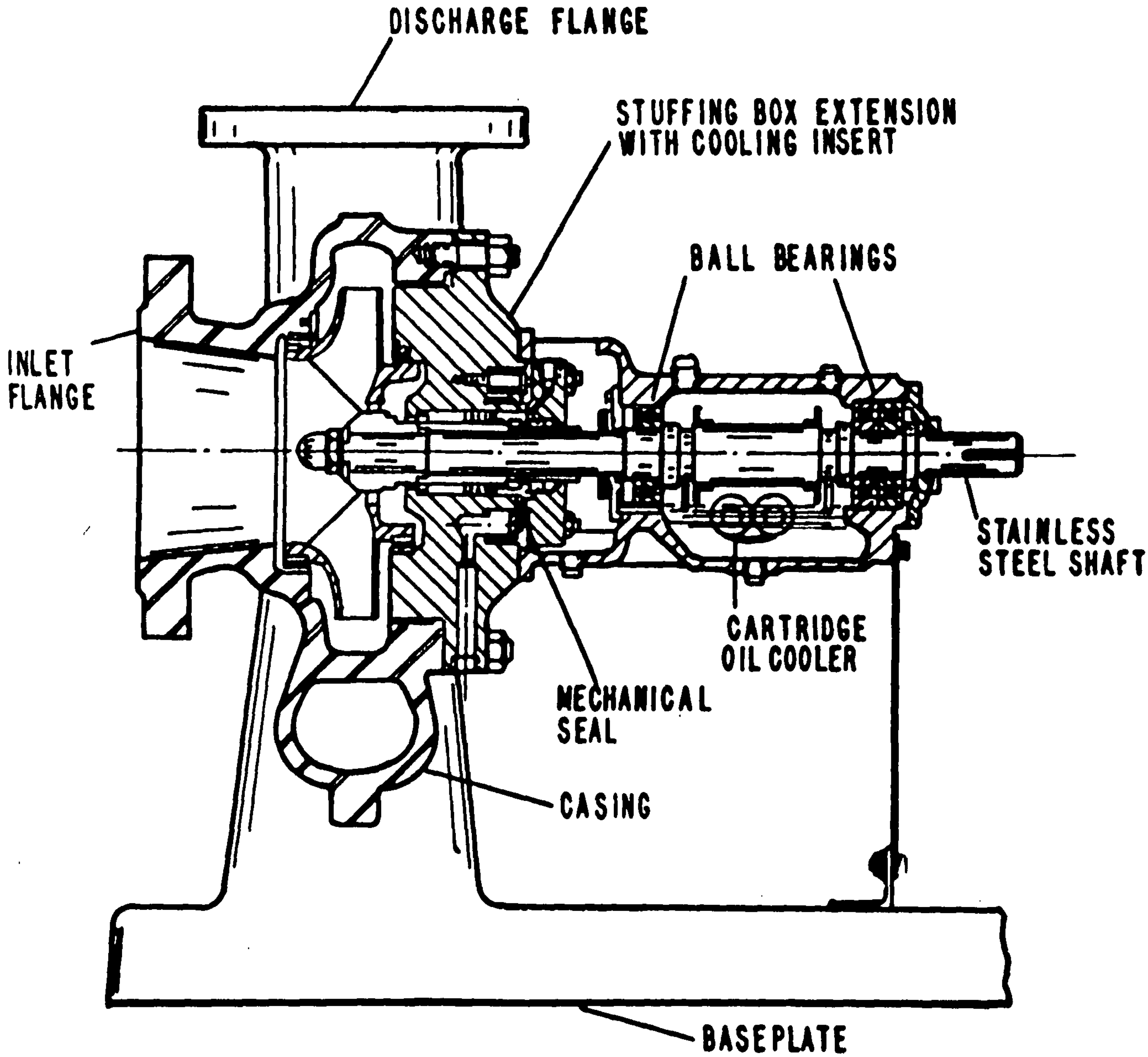

Figure 2-1a. TYPICAL HORIZONTALLY MOUNTED RHR OR CSS PUMP

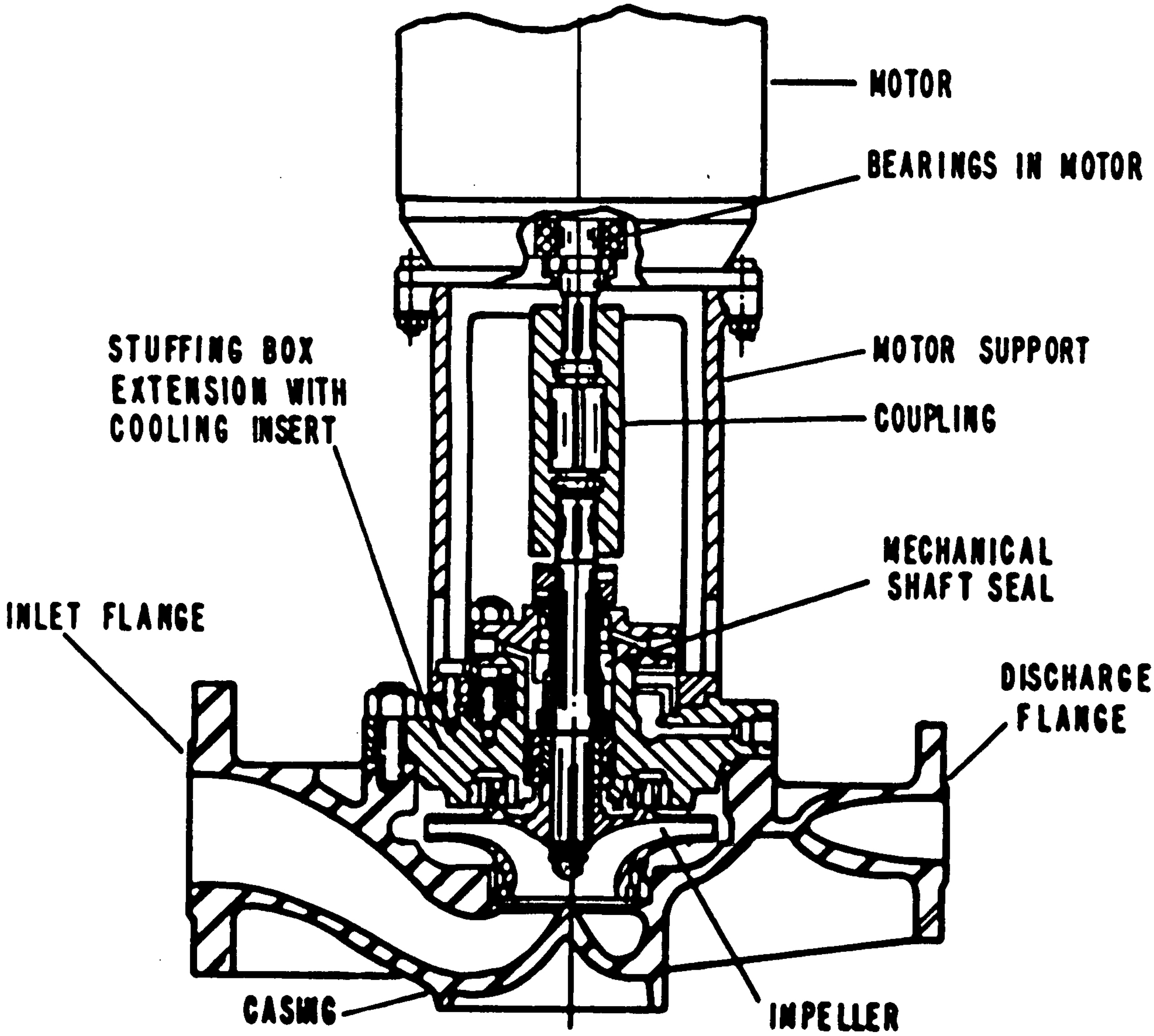

Figure 2-1b. TYPICAL VERTICALLY MOUNTED RHR OR CSS PUMP

TABLE 2-2a

MECHANICAL CHARACTERISTICS OF RHR PUMPS

Unit	Pump Model	Seal Type/ Manufacturer	Impeller Material
Arkansas #2	I-R 8x20 WD	Mechanical	A351 GR-CF8M
Calvert Cliffs #s 1 & 2	I-R 8x21 AL	Mechanical	
Crystal River #3	W 8-HN-194	Mechanical Crane Type 1	18-8SS
Ginna	Pac 6 LSVC	Mechanical Crane Type 1	A358 CF8
Haddam Neck	Pac 8 LXSVCR	Mechanical Crane Type 1	A358 CF8
Kewaunee	B-J 6x10x18 VDSM	Mechanical	
McGuire #s 1&2	I-R 8x20 WD	Mechanical	
Midland #2	B&W 10x12x21 KSMK	Mechanical	
Millstone #2	I-R (No Model #)	Mechanical	
Oconee #3	I-R 8x21 AL	Mechanical	304SS
Prairie Island #s 1 & 2	B-J 6x10x18 VDSM	Mechanical	
Salem #1	I-R 8x20 W	Mechanical	

TABLE 2-2b

MECHANICAL CHARACTERISTICS OF CS PUMPS

Unit	Pump Type	Seal Type/ Manufacturer	Impeller Material
Arkansas #2	I-R 6x23 WD	Mechanical	Austenitic SS
Calvert Cliffs #s 1 & 2	B&W 6x8x1 HSMJ	Mechanical	
Crystal River #3	W 6-HND-134	Mechanical Crane Type 1	18-8 SS
Ginna	Not Available		
Haddam Neck	Not Available		
Kewaunee	I-R 4x11 AN	Mechanical	
McGuire #s 1&2	I-R 8x20 WD	Mechanical	
Midland #2	B&W 6x8x13 SMK	Mechanical	
Millstone #2	G 3736-4x6-13	Mechanical	18-8
Oconee #3	I-R 4x11 A	Mechanical	
Prairie Island #s 1 & 2	I-R 4x11 AN	Mechanical	
Salem #1	C 3415 8x10-22	Mechanical	316 SS

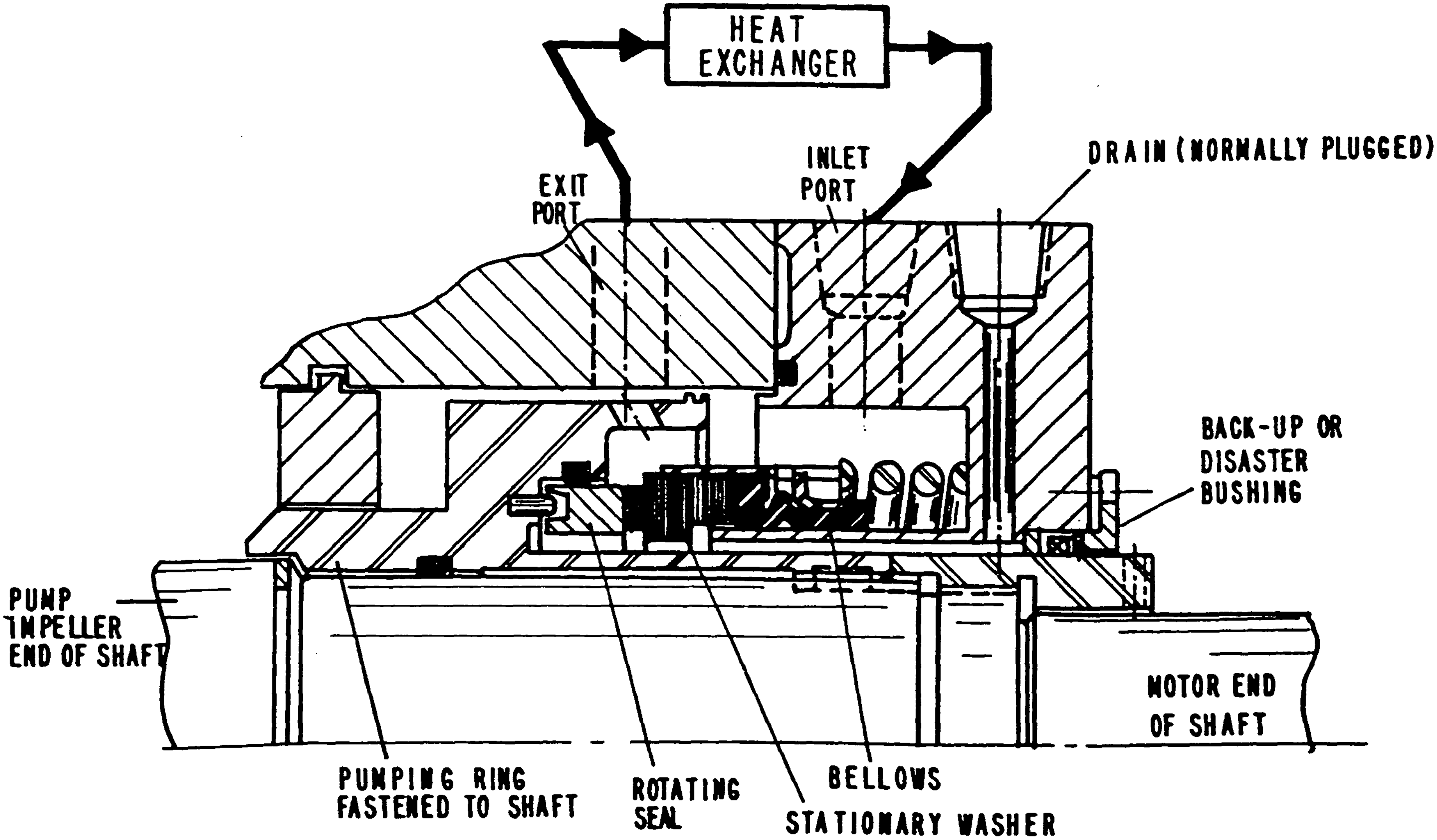

Figure 2-2a. CROSS SECTION OF MERIDIONAL SHAFT SEAL USING PUMPING RING AND CLOSED LOOP COOLING

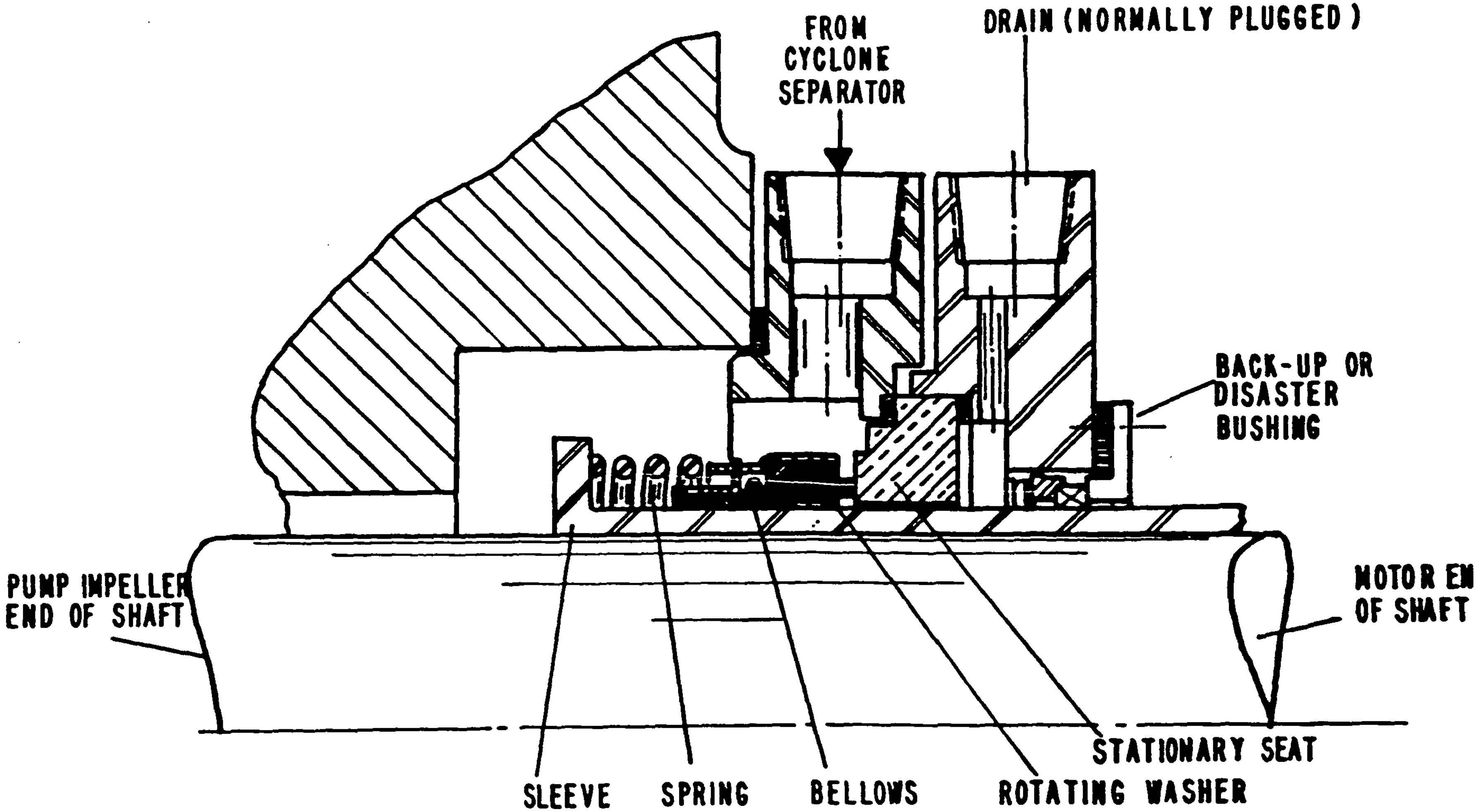

Figure 2-2b. CROSS SECTION OF MECHANICAL SHAFT SEAL USING COOLANT FROM PUMP DISCHARGE

Figure 2-3 shows a schematic of an open loop coolant system for mechanical seals commonly employed in RHR and CS pumps. A cyclone separator is used to separate dense particulates from the fluid stream tapped off the pump exit line. Particulates are separated from the main coolant flow by pressure gradients within the cyclone separator and returned to the low pressure pump inlet. Port lines within the cyclone separator are approximately the same size as the flush lines in the pump seal housing. Seal manufacturers tests have shown that if particulates are large enough, they can accumulate in the cyclone separator and may cause clogging. Although the likelihood of this occurrence is considered small, Crane has recommended since 1970 that seal coolant systems be run without separators on the basis that the likelihood of seal failures due to particulates is less than the likelihood of cyclone separator clogging and subsequent seal failure.

2.2.4 Wearing Rings

Wearing rings are provided in pumps at the impeller inlet and at roughly the same diameter on the backface of the impeller hub. They minimize leakage from the high pressure side of the impeller to the inlet and thereby affect overall pump efficiency. Their diametral position also affects axial thrust loads on the shaft. Design details of wearing rings vary with manufacturer and type of pump. However, all are constructed with appropriate materials to minimize galling in the event of a rub and to minimize material loss due to erosion and corrosion. Typically stainless steels or Monel alloys are employed. Clearances are chosen to minimize leakage, while at the same time they must be large enough to accommodate bearing clearances, shaft deflection, and misalignment due to assembly tolerances and casing distortion. Values of radial clearances are typically 0.008" to 0.012" for RHR and CS pumps.

2.2.5 Bearings

Information on bearing systems was available for only a few of the pumps. Two types are commonly used: 1) oil lubricated bearings mounted in the pump frame, and 2) permanently lubricated bearings within the motor housing. In general these bearings are equipped with three stages of protection against leakage of hot liquid from the shaft seals:

1. shaft seal disaster bushing,
2. shaft slinger,
3. lip seals.

The disaster bushing limits the leakage flow along the shaft in the event of shaft seal failure. The slinger attached to the shaft provides a barrier against a direct jet of leakage from the shaft seal, and by virtue of its rotation, radially slings leakage fluid away from the shaft and bearings. The lip seal ahead of the bearings is designed to prevent low pressure water from seeping along the shaft or housing surface into the bearings.

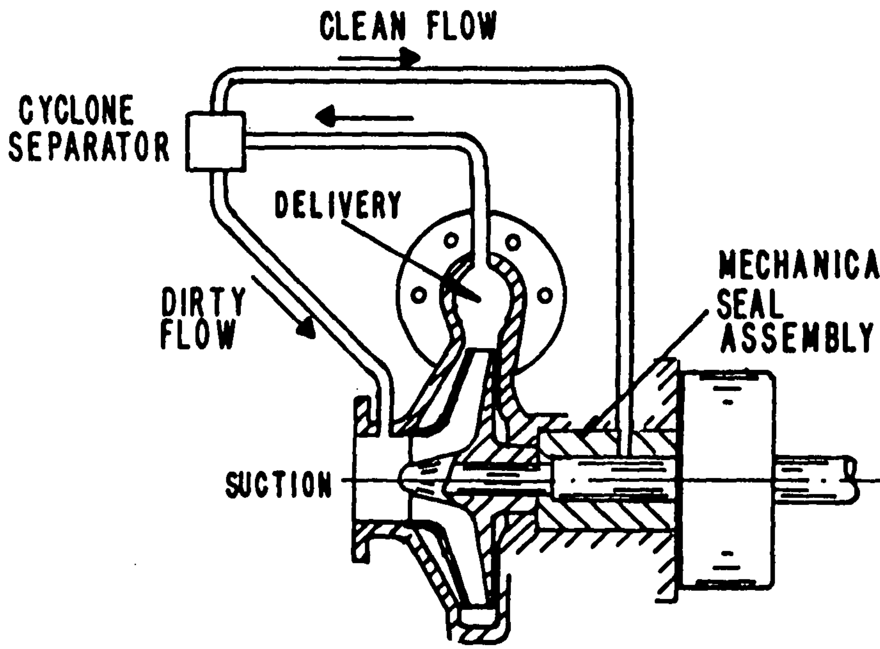

Figure 2-3. OPEN LOOP COOLING FOR MECHANICAL SEAL USING CYCLONE SEPARATOR TO REMOVE PARTICULATES

2.3 Hydraulic Performance Characteristics of RHR and CS Pumps

Table 2-1 from Section 2.1 shows that the pump rating points in CS and RHR service generally fall in the range of specific speeds of 800 to 1600. This small range of specific speed values at rated operating conditions is characteristic of centrifugal pumps with radial impellers. Comparison of the performance of the individual pumps is provided in the following paragraphs.

Table 2-3 summarizes pertinent details about the hydraulic performance characteristics for the pumps surveyed. Rating points for speed N_R, flow Q_R, head H_R, and NPSH requirements are tabulated where the information was available. Actual performance data from manufacturers tests have also been summarized in the table. These data include flow rates at best efficiency point Q_{bep} (which is not necessarily rated flow), maximum flow rates Q_{max}, NPSHR values at maximum flow rate and ratios of rated flows and maximum flows to Q_{bep}. In general, the operating details can be summarized as follows:

For RHR pumps:

- Rated flows are generally 2000 to 3000 gpm and usually 0.7 to 1.0 times the Q_{bep},
- Rated heads are generally 280 to 350 feet,
- Rated speeds are about 1800 rpm,
- Maximum flow rates fall between from 2000 gpm to 5000 gpm and 1.1 to 1.4 times the Q_{bep},
- NPSHR values at rated conditions vary from 8 to 12 feet and
- NPSHR values at maximum flow conditions are about 20 feet.

For CS pumps:

- Rated flows are 1300 to 2000 gpm and 0.8 to 1.1 times Q_{bep},
- Rated heads are 400 to 500 feet,
- Rated speeds are 1800 rpm and 3600 rpm,
- NPSHR values at rated conditions are higher than those for RHR pumps, typically in the range 16 feet to 20 feet.

The differences in rating points noted above must still be considered in regard to similarities in hydraulic design. Again, the range of specific speed values at the rating points is fairly narrow. To further exemplify similarities in the machines studied, the individual performance curves for each of the pumps are compared in Figures 2-4 for RHR pumps and 2-5 for CS

TABLE 2-1

HYDRAULIC PERFORMANCE CHARACTERISTICS OF RHR AND CS PUMPS

PLANT	RATED CONDITIONS				BEP FLOW (GPM)	RATED FLOW / BEP FLOW	MAX. FLOW (GPM)	NPSHRQ MAX. FLOW (FT)	MAX. FLOW / BEP FLOW
	PUMP SPEED (RPM)	FLOW RATE (GPM)	HEAD (FT)	NPSHR (FT)					
RHR									
Arkansas Unit #2	1780	3100	350	12	5430	0.72	5100	25	1.19
Calvert Cliffs	1780	3000	350	12	3130	0.96	4500	19	1.44
Crystal River #3	1780	3000	350	12.5	3000	1.00	*	*	*
Ginna	1770	1560	280	7.8	1750	0.89	2000	11.5	1.14
Haddom Neck	1770	2200	300	10	2640	0.83	*	*	*
Kewaunee	1770	2000	280	8	2000	1.00	*	*	*
McGuire	1780	3000	375	10	4400	0.68	5300	22	1.20
Midland #2	1780	3000	370	8	3130	0.96	4500	*	1.44
Millstone Unit 2	1770	3000	350	13	3800	0.79	4500	19	1.18
Oconee	1780	3000	360	12	3400	0.88	*	*	*
Prairie Island	1770	2000	280	8	2000	1.00	*	*	*
Salem	1780	3000	350	11	4400	0.68	4300, 4600,	18 20	0.98 1.04
CS									
Arkansas Unit #2	1780	2200	525	8	2400	0.92	3200	6	1.33
Calvert Cliffs	3580	1350	375	19	1650	0.82	*	*	-
Crystal River #3	3550	1500	450	19	1660	0.90	*	*	-
Ginna	*	1370	435	*	*	*	*	*	-
Haddom Neck	1770	2200	300	10	2640	0.83	-	-	-
Kewaunee	3550	1300	475	20	1300	1.00	-	*	-
McGuire	1780	3400	380	16	5000	0.68	*	*	-
Midland #2	3550	1300	387	19	1650	0.79	-	*	-
Millstone Unit 2	3560	1400	477	17	1370	1.02	*	*	-
Oconee	3550	1490	460	23	1250	1.19	*	*	-
Prairie Island	3550	1300	500	8.5	1300	1.00	*	*	-
Salem	1780	2600	450	10	3200	0.81	*	*	-

* not available

pumps. In each of the figures, head characteristics are plotted against flow rate in normalized coordinates and NPSHR characteristics are given as a function of normalized flow rate in absolute units.

The performance data have been taken from tests reported in the individual plant Final Safety Analysis Reports (FSAR). The presentation of the head versus flow characteristic is given in terms of normalized head versus normalized flow where head and flow values at best efficiency point are used as the normalizing parameters, i.e.:

$$h = H/H_{bep} \qquad q = Q/Q_{bep} \tag{2.2}$$

However, NPSHR curves for each of the pumps are presented in units of feet as a function of normalized flow rate.

Maximum flow rates, and rated flows have been identified in each of the figures to illustrate the proximity of these values to best efficiency point (q=1).

The figures show that there are strong similarities in the performance characteristics of the pumps. Generally, the limitations of normal operation (from rated flow to maximum flow rate) are from 70% to 140% of bep. Cavitation characteristics are also generally similar with pump inlet requirements in terms of NPSHR of about 20 feet for RHR pumps at maximum flow rates and about 20 feet at rated flows for CS pumps.

These similarities in characteristics then provide a basis for assessing the likely performance of RHR and CS pumps under air ingesting conditions and also provides a basis for discussion of the general operational characteristics of these pumps.

2.4 Operating Considerations for RHR and CS Pumps

It has been noted in the previous section that there are similarities in construction details and performance characteristics of the RHR and CS pumps surveyed. This section will review the operational requirements of RHR and CS pumps and discuss important aspects of these operational details with respect to their performance characteristics.

Several features separate the operation of CS pumps from that of RHR pumps. CS pumps are designed and specified to operate during emergency conditions only. They draw suction from the RWST for the initial period following a LOCA. They may afterward be required to draw suction from the containment sump to continue condensation cooling of vapor in the containment. Their operating cycle is relatively short (about 800 hours or less) compared to RHR pumps during an emergency, which may be required to operate for 10,000 hours or more. In addition, the design flow and head requirements for the CS pumps is based on pumping from either the RWST or sump through distribution piping to the cooling spray nozzles. This is a fairly constant "load" or resistance to the pumps for which they are rated.

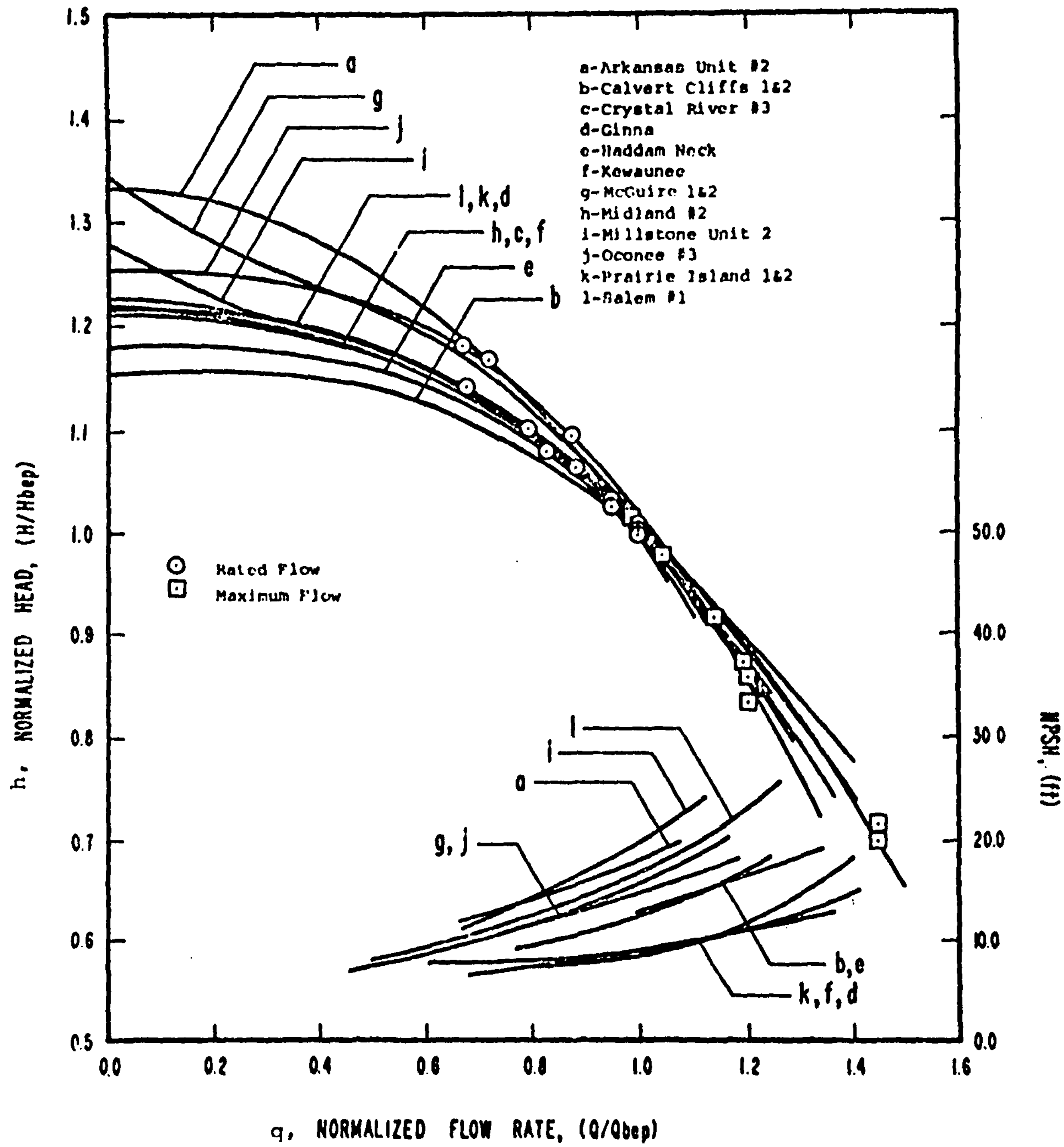

Figure 2-4. HYDRAULIC PERFORMANCE CHARACTERISTICS FOR RESIDUAL HEAT REMOVAL PUMPS

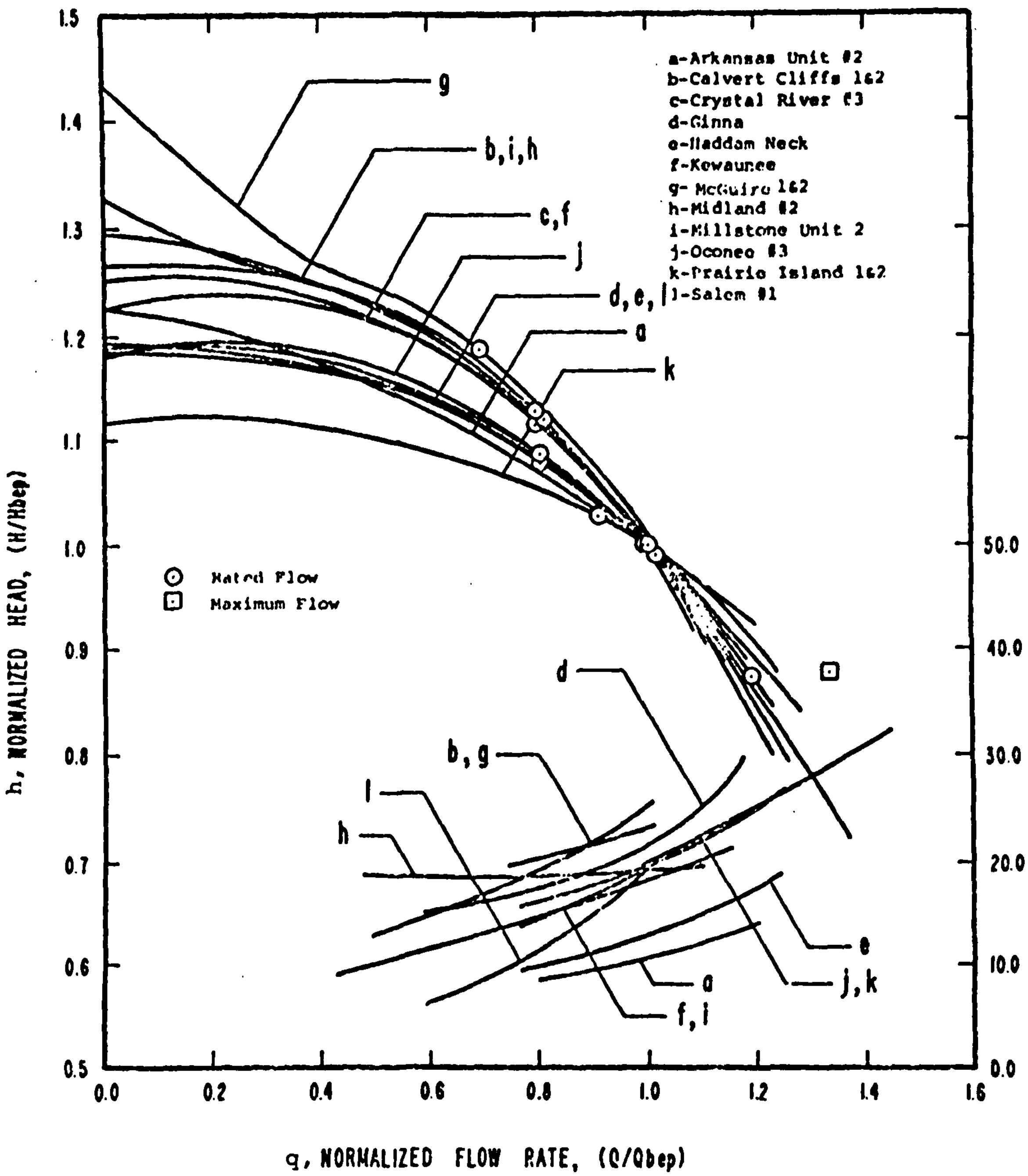

Figure 2-5. HYDRAULIC PERFORMANCE CHARACTERISTICS FOR CONTAINMENT SPRAY PUMPS

RHR pumps, on the other hand, must circulate water through the core during normal cooling cycles and during emergency conditions where a break is located within the piping. Hence the "load" or resistance to flow may vary widely depending on whether a break is present, its size and location. The result is that although the RHR pump has a rating point, it is also specified to meet flow and NPSH requirements to some maximum flow situation where the frictional resistance of the system is a minimum. This establishes the maximum flow rate design basis for the pump. It is the maximum flow situation which is most stringent in terms of cavitation. Suction piping losses will be highest at maximum flow producing the lowest values of NPSH available. It is also clear from Figure 2-4 that the NPSH requirements for pumps increase with increasing flow rates.

Air ingestion characteristics of sumps are such that the likelihood of air ingestion increases with increased flow rate. Hence the maximum flow rate situation is also the conservative condition for evaluating air ingestion effects.

Under some circumstances, low flow rates may be required through RHR pumps. Recirculation in centrifugal pumps at low flows has been a recognized phenomena for some time [44]. Depending on details of the pump design, recirculation at low flows may become severe enough to cause vibration and in some cases flow oscillations. In general, for pumps with relatively larger impeller inlet diameters (to meet relatively low NPSH requirements) recirculation will occur at relatively higher flow rates. Each pump must be evaluated on an individual basis. In general, the pump manufacturer will supply recommended lower flow rate limits and/or methods for operating safely below required limits.

It will be shown in a later section that at low flows, even small air ingestion rates can cause air to accumulate in the impeller causing "air binding" leading to complete degradation in performance. While this situation requires attention, it is also true that sump characteristics are such that at low flow rates, air ingestion is least likely.

In this section the rating points, hydraulic performance characteristics and mechanical details of RHR and CS pumps have been reviewed. Although the pumps have differing rated flows, speeds and heads, their specific speeds fall within a narrow range. Also, the pumps are quite similar in mechanical design and construction. These similarities justify a common assessment of performance of RHR and CS pumps under air and debris ingesting conditions on the basis of the performance of other pumps of similar design. The following section presents a brief review of available literature on two-phase performance of pumps and summary of the data applicable to RHR and CS pumps.

3 SURVEY OF PREVIOUS RESEARCH AND DATA

This section summarizes the sources of data on the behavior of pumps operating with gas/water and vapor/water two-phase flows and with particulate laden flows. Information on two-phase turbomachine behavior is abundant. However, much of it is not directly applicable in assessing RHR and CS pump performance while operating with air or particulate ingestion.

Literature on liquid/gas and liquid/vapor two-phase flow behavior of pumps was thoroughly reviewed. Although the physical mechanisms which cause performance degradation in pumps under these flow conditions is not entirely understood, there is sufficient experimental data available in the literature to provide a sound basis for assessing RHR and CS pump behavior.

A vast amount of information on cavitation in pumps was reviewed but very little was found to have significance to this study. RHR and CS pumps will cavitate if suction conditions are such that there is insufficient NPSH at some operating condition. However, beyond this, the technical literature provides little aid in assessing the behavior of these specific pump types with respect to either erosion , performance degradation, or the combined effects of cavitation and air ingestion. The effects of cavitation, both performance degradation and material erosion, are highly dependent on individual pump design and operating conditions. Most data on these effects were provided by pump manufacturers.

The literature on gas/particulate flows in turbines and compressors was not included in our survey. The differences in density between phases are substantially larger in gas/solid flows than in the solid/liquid flows expected in RHR and CS pumps. Also, fluid velocities in RHR and CS pumps are approximately an order of magnitude less than those in compressors and gas turbines; a factor which is important in both phase separation and erosive behavior.

Information on the behavior of centrifugal pumps under particulate ingesting conditions was obtained from technical literature and from pump manufacturers. Test data dealing with the effects of particulates on the operation of mechanical shaft seals and their filtration systems is not readily available in the open literature. Data on these topics came from both pump and seal manufacturers.

3.1 Data on the Performance of Centrifugal Pumps in Gas/Liquid and Vapor/Liquid Flows

Numerous sources of data on the performance of centrifugal pumps in two-phase, gas/liquid and vapor/liquid flows exist in the literature. These data sets display common trends as illustrated in Figure 3-1, even though the pumps tested and fluid mixtures vary substantially. The pump performance, characterized by head developed at a given flow and speed, degrades with increasing gas or vapor content in the liquid being pumped. The amount of degradation is a function of many variables besides the gas or vapor content, such as pump speed, flow rate, impeller and inlet geometry, suction pressure and pump efficiency. However, the trend of performance degradation with increasing gas or vapor content is present for all data sets.

In the final selection of data sets from the literature to assess the performance of RHR and CS pumps operating with air ingestion, several criteria outlined in Section 4 were applied to the overall collection of gas/liquid and vapor/liquid information. Only well-documented air/water data sets on pumps of designs similar to those of RHR and CS pumps were chosen in the final assessment. However, a sizable number of papers in the literature contain steam/water data, data on pumps with specific speeds outside the range of interest or data on pumps of atypical design. These experimental results for forward flow and forward speed operation are briefly discussed in the following paragraphs and are presented in Figure 3-2 along with the three data sets chosen as the basis for the assessment of RHR and CS pumps in air/water operation (darkened symbols in Figure 3-2). Although there is a wide variation in degradation as a function of pump void fraction caused by the differences in pump design, operating point and fluid mixture, it should be noted that in no case is degradation severe for void fractions less than 2%.

Figure 3-2 shows two-phase test data plotted from several sets of experimental programs. The axes for the plot in Figure 3-2 are similar to those for Figure 3-1, intended to show head or differential pressure at some two-phase flow condition with speed and total flow rate constant. In order to do so, the pump pressure rise values for individual test points have been normalized to the pump pressure rise values under liquid operation. Inlet void fraction is the ratio of volumetric gas or vapor flow rate to the total volumetric flow rate. The individual pumps, together with rating conditions and references are listed in Table 3-1. The range of design point specific speeds for the pumps listed in the table represent impeller geometries varying from radial to mixed-flow. Impeller diameters range from 2" to 12" and inlet pressures range from atmospheric to 1250 psia.

The data show that there is a substantial variation in the initiation and rate of performance degradation among the various tests. It is not possible from the data shown to isolate the separate effects of impeller design (specific speed), fluid mixture, scale and inlet conditions on degradation. However, it is clear that even under the wide range of gas/liquid and vapor/liquid conditions shown, degradation in performance does not occur until the pump inlet void fraction exceeds about 3%.

As the inlet void fraction increases, the scatter in performance increases to the extent that the dependence of degradation on pump design, inlet conditions, fluids, etc. determines the performance of individual pumps. In the following subsections the data sets presented in Figure 3-2 are described in more detail.

3.1.1 Steam/Water Data

A large body of two-phase pump data was generated by the programs funded by the Electric Power Research Institute (EPRI) and conducted at Babcock and Wilcox (B&W), Combustion Engineering (C-E) and Creare. In these programs, scale models of PWR primary coolant pumps (specific speed = 4200 rpm [US gpm]$^{1/2}$/ft$^{3/4}$) were tested in steady and transient steam/water flows and steady air/water flows. The main objective of these programs was to generate a data base for the validation of pump performance analytical models for use in computer codes.

A low specific speed pump [926 rpm (US gpm)$^{1/2}$/ft$^{3/4}$] was tested in steady state steam water flows by the Aerojet Nuclear Company (ANC) [10], [11] (transient tests were also conducted). These data are commonly referred to as Semiscale data since the pump tested was that used in the Semiscale test facility at Idaho National Engineering Laboratory (INEL).

The steady-state data from the EPRI programs and those for the Semiscale pump, shown in Figure 3-2, have been analyzed in detail in [12]. Although both these data sets show degradation in pump performance, they show different extents of degradation with void fraction. The EPRI data show steady but gradual head degradation with complete degradation observed only after 40% void fraction. The Semiscale data show sudden and complete degradation at void fraction about 20%. This difference is attributed to the different specific speeds of the test pumps. Transient data from these programs have not been examined in detail.

Figure 3-2 also shows some GE steam/water data taken at 600 psig on a 1600 specific speed pump. These data were originally reported by Love [13]. They were later presented by Sozzi and Burnette [14] who also presented steam/water and air/water data on other pumps. Runstadler [15] has presented a detailed review of these GE pump data. Burnette and Sozzi suggest that for a given flow coefficient (flow rate) there is a critical threshold value of steam void fraction above which degradation is sudden and complete. It is interesting to note that the Semiscale data, which was also taken on a relatively low specific speed pump, displays a similar behavior.

Heidrick et al. have presented steam/water data on the so-called SAWFT pump of specific speed 500 [16] and on a 2370 specific speed scale model of a CANDU nuclear reactor pump [17]. The lower specific speed (higher head) pump was observed to have the higher critical void fraction.

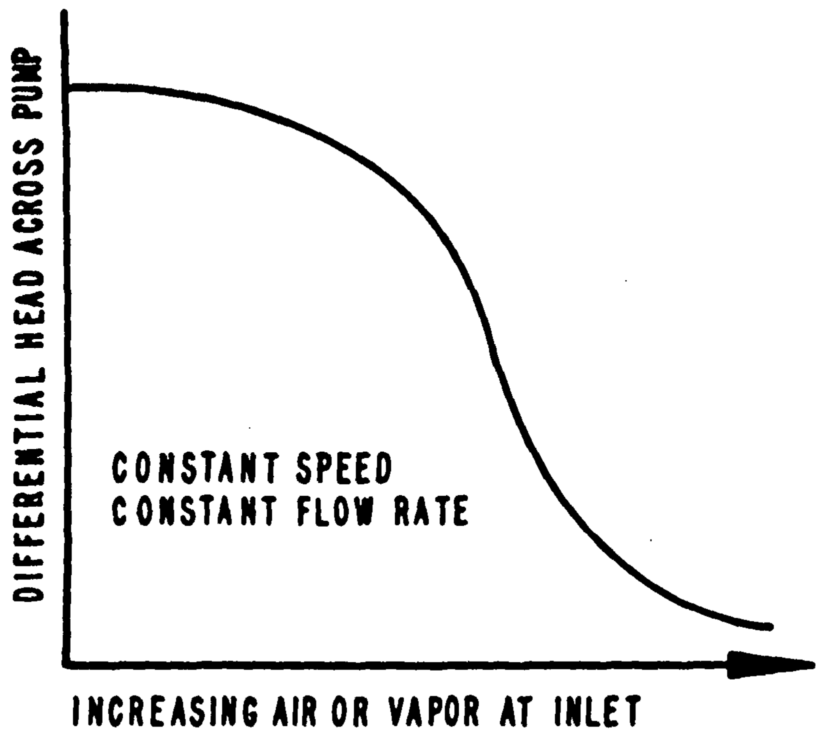

Figure 3-1. A PLOT OF PUMP PERFORMANCE AT SOME FIXED OPERATING CONDITION SHOWING THE EFFECT OF INCREASING AIR OR VAPOR AT THE INLET

TABLE 3-1
TEST CONDITION AND RATED CONDITIONS FROM VARIOUS SOURCES OF TWO-PHASE PUMP DATA

SOURCE	RATED PARAMETERS HEAD (FEET)	FLOW (US GPM)	SPEED (RPM)	SPECIFIC SPEED RPM(US GPM)$^{1/2}$/FT$^{3/4}$	SUCTION PRESSURE (PSIA)	IMPELLER O.D. (INCHES)	POWER (HP)	EFFICIENCY (%)
Creare [12] *A/W and S/W	252	181 (219)	18,000	4,200	A/W at 90	1.94	16.5	>85
B&W A/W [12]	390	11,200	3,580	4,317	20-120	12.33	1297	>85
C-E S/W [12]	252	3,500	4,500	4,200	15-1250	7.75	264	>85
Semiscale S/W [10],[11],[12]	192	180	3,560	926	200-900	7.75	118	-
GE S/W [14]	148	1,750	1,700	1,600	615	-	-	-
Stepanoff A/W [14], [20]	54	1,110	900	1,500	15	11.5	-	-
Stepanoff A/W [14], [20]	188	1,100	1,750	1,130	65	-	-	-
Murakami & Minemura A/W [30]	63	235	1,750	1,200	-	8.8	-	66
Merry A/W [31]	301	697	2,940	1,074	-	-	-	67
Florjancic A/W [32]	357	1,512	2,950	1,397	36.8, 73.5 and 66.2	-	-	-
Arie & Fukusako A/W [32]	23	739	960	2,485	-	10.6	-	-

*A/W - Air/Water
S/W - Steam/Water

3.1.2 Air/Wat‹ r Data

Although experimental information on air/water flows in pumps is as extensive as that for steam/water flows, the data pertinent to pumps in the range of specific speeds typical of RHR and CS pumps is rather limited. Nonetheless, it is instructive to review some of the existing data.

Murakami and co-workers have published data on the air/water performance of pumps of widely different specific speeds including one in the range of interest. The latter is shown in Figure 3-2 and is discussed in subsection 3.1.4.1. In [18], data on two mixed-flow pumps with specific speeds of 7000 and 8000 and an axial flow pump with specific speed of 60,000 are presented. Performance deteriorates continuously with increasing air content until the pumps lose prime at air volume fractions between 7% and 15%. Also, at each flow rate, the maximum void fraction at which the pumps can operate without losing prime increases with speed.

Murakami and Minemura [19] present results of air/water tests on an axial flow pump of specific speed 12,000. A small amount of degradation in head occurs for air volume fractions less than 3%. Above 3%, the rate of degradation increases markedly but still remains less than that for a centrifugal pump.

Air/water data were acquired in the EPRI programs conducted at B&W and Creare. These are shown in Figure 3-2 and show the same general trends as steam-water data. Also shown in Figure 3-2 are Stepanoff's air/water data on two pumps at different suction pressures, in [20]. Although the pumps tested had specific speeds in the same range as those for RHR and CS pumps (500 and 1130), test methods were not well documented. Thus, these data were not used in assessing RHR and CS pump performance. Section 3.1.4.4 has more on Stepanoff's data.

3.1.3 Miscellaneous Data on the Two-Phase Performance of Pumps

This subsection briefly covers references which contain information on the two-phase performance of pumps, but were not considered suitable for the assessment of RHR and CS pump performance and are not shown in Figure 3-2.

Kosmowski [21] presents two-phase performance data of unknown origin. Although the trends in degradation agree with those of others, the paper does not contain sufficient well-documented information on the impeller and its single-phase performance to be useful.

Some two-phase air/water and freon/water data were taken at MIT and are published in [22]. However, the details of the impellers are not given. This EPRI sponsored study was geared more towards developing an empirical correlation for two-phase pump performance as a function of void fraction than towards acquiring performance data.

Chivers has published data on the effects of dissolved air [23], temperature [24] and their combined effect [23] on cavitation inception in a pump of specific speed 1100. No data on undissolved (ingested) air are presented. The data show that the effect of dissolved air up to 30% of saturation, like that of undissolved (ingested) air, increases the NPSH required. Above 30% of saturation, the NPSH required appears to be independent of the amount of dissolved air, unlike the continuously increasing NPSH required for increasing amounts of ingested air. Also, the NPSH required decreases with increased fluid temperature in agreement with the Hydraulic Institute Standards [25]. However, with dissolved air in the loop, the NPSH required first decreases with increasing temperature up to 80°C after which it increases. This reversal in trend was accompanied by the appearance of undissolved air in the loop.

Rothe et al. [26] observed that the presence of air in the fluid loop can lead to pump surge and oscillatory flows. The occurrence of such oscillations is a function of operating conditions, void fraction, and system configuration.

Other studies on two-phase pump performance which were reviewed include those by Patel and Runstadler [27] on an experimental study of the physics of two-phase pump performance, Hunter and Harris [28] on the performance of nuclear reactor primary coolant pumps during blowdown and Grennan [29] on polyphase flow through gas turbine fuel pumps. None had direct application to RHR and CS pump behavior.

3.1.4 Air/Water Data On 800 to 2000 Specific Speed Pumps

In this subsection four sources of air/water data on 800 to 2000 specific speed pumps are reviewed more closely. Three of these sources form the basis of the assessment of air/water performance of RHR and CS pumps. These three sources were selected based on the criteria given in Section 4, one of which required that the test pumps have specific speeds in the range 800 to 2000.

3.1.4.1 Data From Murakami and Minemura [30]

The authors have presented data on the air/water performance of three different pump impellers. Table 3-2 contains information on the impellers. Performance curves are presented in [30] as plots of normalized head coefficient, normalized efficiency and normalized power coefficient vs. normalized flow coefficient. The authors also present the results of an experiment involving visualization of the trajectories of air bubbles in the blade passages. A supporting analysis is also given.

Figure 3-3a shows the test setup. The shape and dimensions of the five bladed impeller used are shown in Figure 3-3b. Gauge pressures at the inlet and discharge legs were measured using manometers. An orifice with 8 holes was located six pipe diameters upstream of the pump to admit air in a

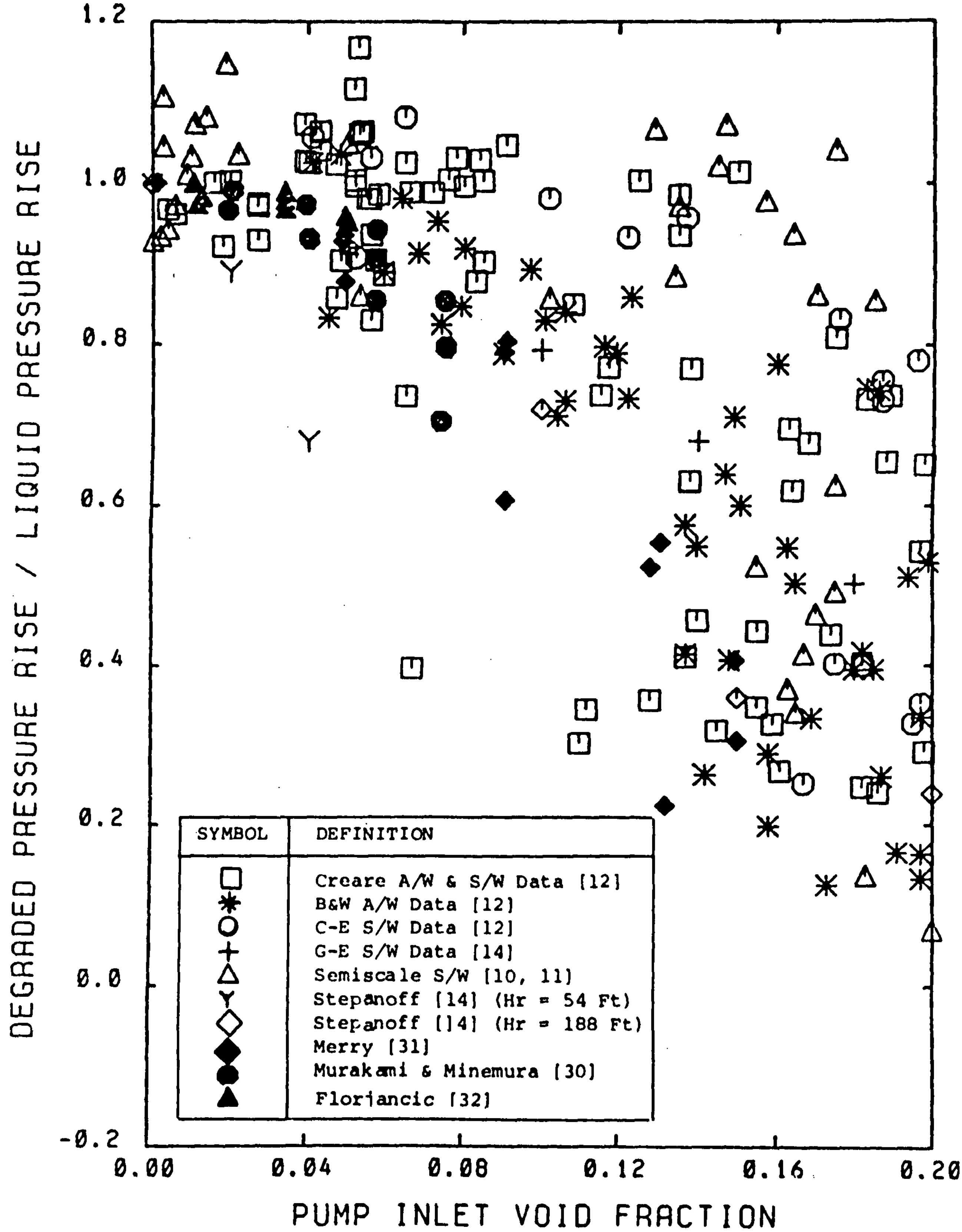

Figure 3-2. PUMP PRESSURE RISE IN AIR/WATER AND STEAM/WATER FLOWS

TABLE 3-2

GEOMETRY AND BEST-EFFICIENCY PARAMETERS OF THREE, FIVE AND SEVEN BLADE IMPELLERS USED BY MURAKAMI AND MINEMURA [30]

Number of Blades on Impeller	Flow Area at Impeller Exit (in^2)	Best Efficiency Parameters				
		Head (ft)	Volumetric Flow Rate (US gpm)	Speed (rpm)	Specific Speed	Maximum Efficiency (%)
3	16.21	51	198	1750	1290	60.2
5	13.94	63	235	1750	1200	65.5
7	11.64	62	222	1750	1180	66.0

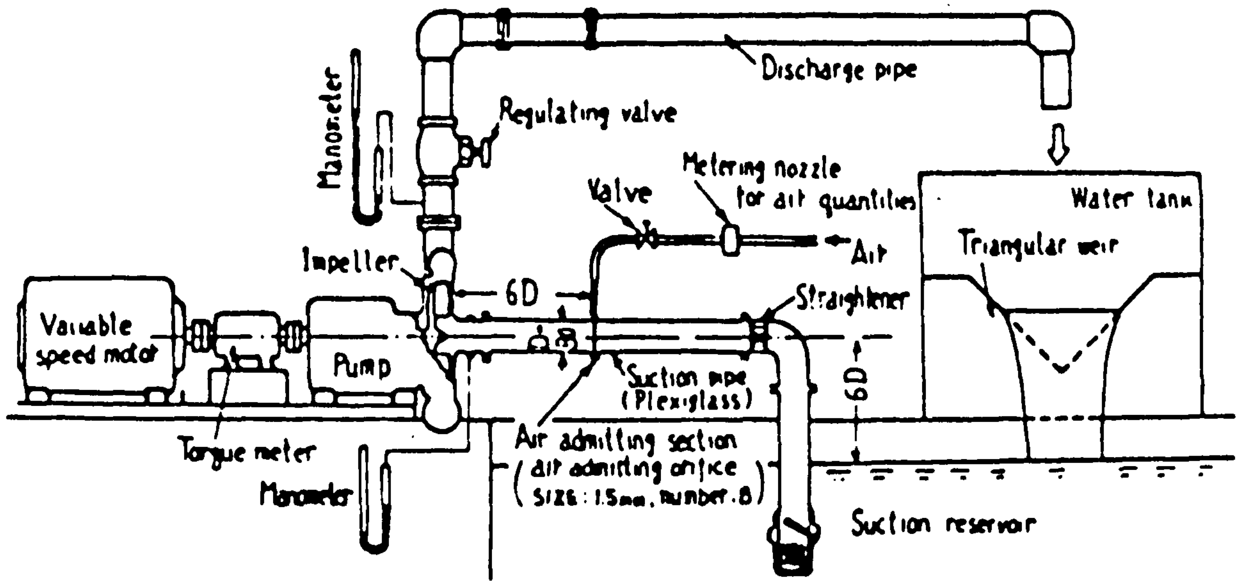

Figure 3-3(a). TEST SET-UP USED BY MURAKAMI AND MINEMURA [30]

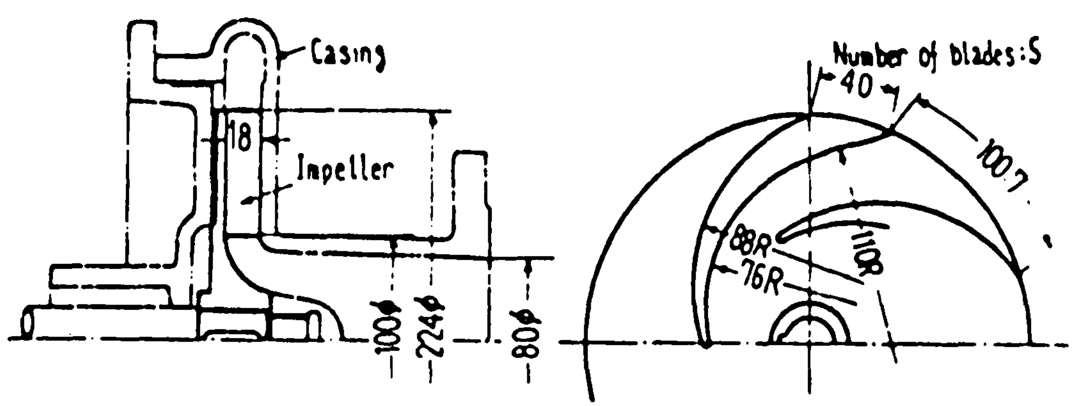

Figure 3-3(b). SHAPE AND DIMENSIONS OF FIVE-BLADE IMPELLER USED BY MURAKAMI AND MINEMURA [30] homogeneous manner. The void fraction was not measured, but was presumably calculated assuming no-slip flow knowing the volumetric flow rates of air and water as,

$$\alpha = (Q_a)/(Q_a+Q_w) \qquad (3.1)$$

Independent tests at Creare have shown that this is an accurate assumption for many air/water flows [12]. The liquid density was used to calculate head. All the tests were conducted at a constant pump speed of 1750 rpm.

The number of blades affected the pump performance in air/water flow. The head developed by the three blade impeller actually increased slightly for less than 2.5% volume fraction of air. The authors contend that this improvement is due to an "improvement in the flow patterns in the impeller". The performance of the five blade impeller was comparable to that of the seven blade impeller. The head decreased continuously with increasing air content until the pumps began to lose prime at volume fractions above 6%.

For air fractions higher than 2.5% the head-flow curve for all impellers were continuous but not smooth. The authors report that minor discontinuities in the head-flow curves were accompanied by changes in flow patterns in the impeller with air accumulation at various locations within the impeller.

The data presented in the reference allows one to define a lower limit of air volume fraction, 2.5%, below which pump performance is unaffected by air. Above this limit, performance is dependent on the number of blades and degrades continuously. The data on the five-blade impeller is shown in Figure 3-2 and is one of those used in Figure 4-1 to assess the air/water performance of RHR and CS pumps.

3.1.4.2 Data From Merry [31]

Data on the air/water performance of an end suction pump of specific speed 1074 are presented in this reference. Some data in oil/air flows and data on a 779 specific speed pump are also presented in the reference but are not discussed here.

The layout of the air/water test rig is shown in Figure 3-4. Water was drawn from the tank into the pump and discharged through an orifice plate and two butterfly valves before returning to the tank. The flow was varied by adjusting the valves. Inlet conditions to the pump wer^ set by applying compressed air or vacuum to the tank.

Pressures were measured at the suction and discharge legs approximately 2 pipe diameters upstream and downstream of the pump by absolute pressure transducers. The orifice plate on the discharge leg, which is claimed to

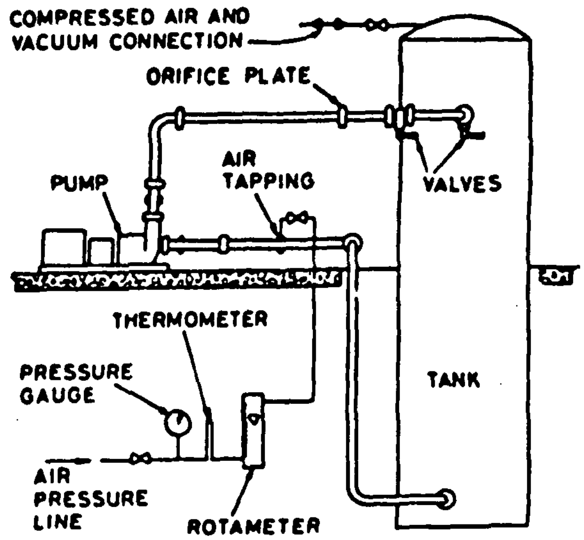

Figure 3-4. TEST SETUP USED BY MERRY [31]

have been calibrated for up to 5% air volume fraction, was used to measure the water flow rate. Air from a pressure line was passed through a regulating valve and a rotameter before being introduced into the rig about 17 pipe diameters upstream of the pump. Other measurements included shaft torque and fluid temperature at the pump inlet.

The data are presented in [31] as plots of total specific energy rise and efficiency vs. water flow rate at different air flow rates. These plots for the 1074 specific speed pump are shown in Figure 3-5a and 3-5b. The head data in normalized form are also shown in Figure 3-2. It is not stated in the reference how the total specific energy rise was calculated knowing only the pressures at the inlet and outlet without velocity or density measurements. The experiments showed that as the percentage of gas was increased, the head, flow and efficiency decreased while the power input remained almost constant. The degradation in performance was lower at the best efficiency flow rate than at lower or higher flow rates.

Data on the combined effect of air ingestion and cavitation are also presented in the reference. These are discussed in Section 3.2.

3.1.4.3 Data From Florjancic [32]

In this reference, the specific speed of the pump tested is 1397. Figure 3-6 shows the schematic layout of the test rig. Pressures were measured at the suction and discharge legs by manometers. The total flow rate was measured at the discharge tank. Power input to the pump was measured by a swivel bearing motor. The inlet pressure to the pump was varied by throttling. All the tests were conducted at a pump speed of 2950 rpm.

Air was introduced about 8 pipe diameters upstream of the pump through a multi-orifice nozzle and its flow rate measured with a calibrated orifice plate. The inlet pressure before the orifice plate was varied with a special reducing valve.

Data in the reference are presented as plots of normalized head, power and efficiency vs. normalized flow rate for various air volume flow fractions. These curves are shown in Figure 3-7. The head data are also shown in Figure 3-2.

Figure 3-7 shows that for 2% air fraction, the head, power and efficiency at rated flow remain unchanged from their single-phase values and are only slightly lower at flow rates higher and lower than the rated flow. At higher air fractions, the characteristics are significantly degraded compared to the single-phase characteristics.

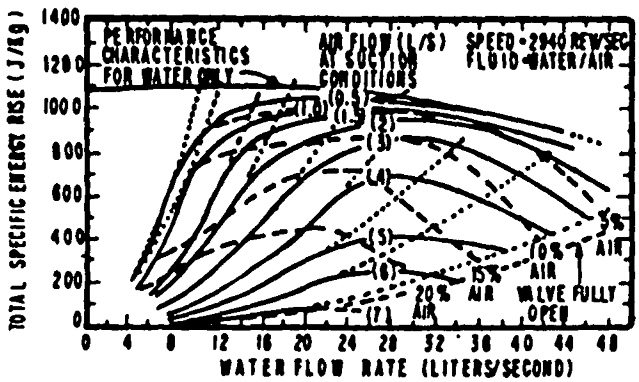

Figure 3-5a. <u>HEAD CHARACTERISTICS IN AIR-WATER FLOW FROM MERRY [31]</u>

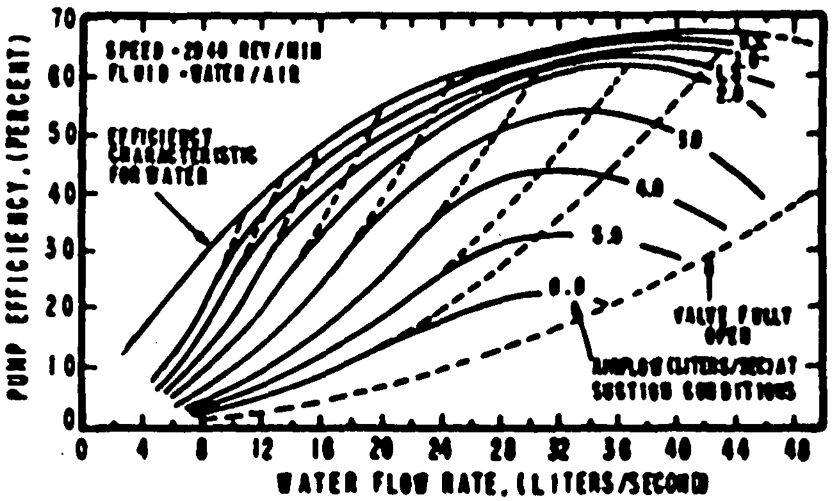

Figure 3-5b. PUMP EFFICIENCY IN AIR/WATER FLOW FROM MERRY (REFERENCE 31)

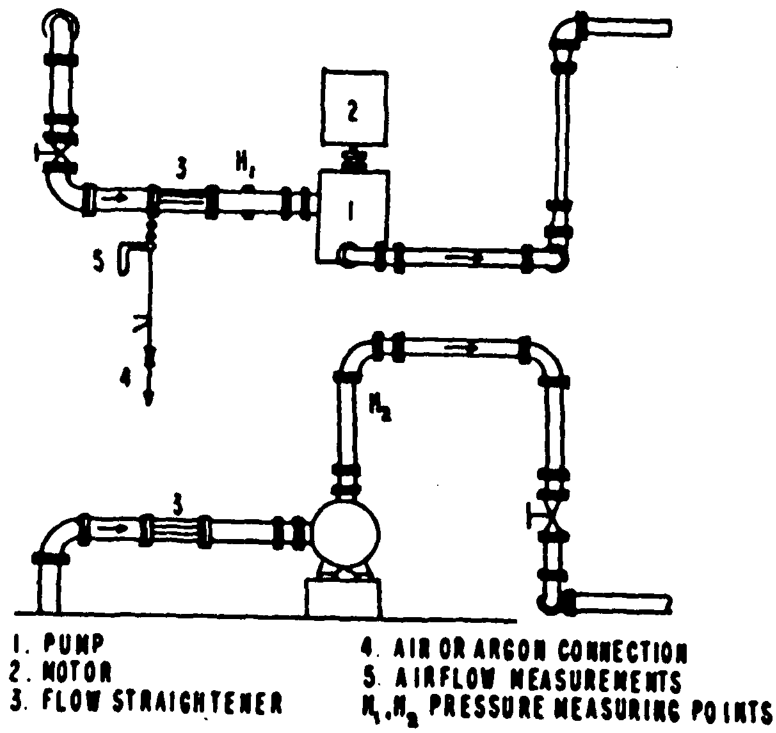

Figure 3-6. LAYOUT OF TEST RIG FROM FLORJANCIC [32]

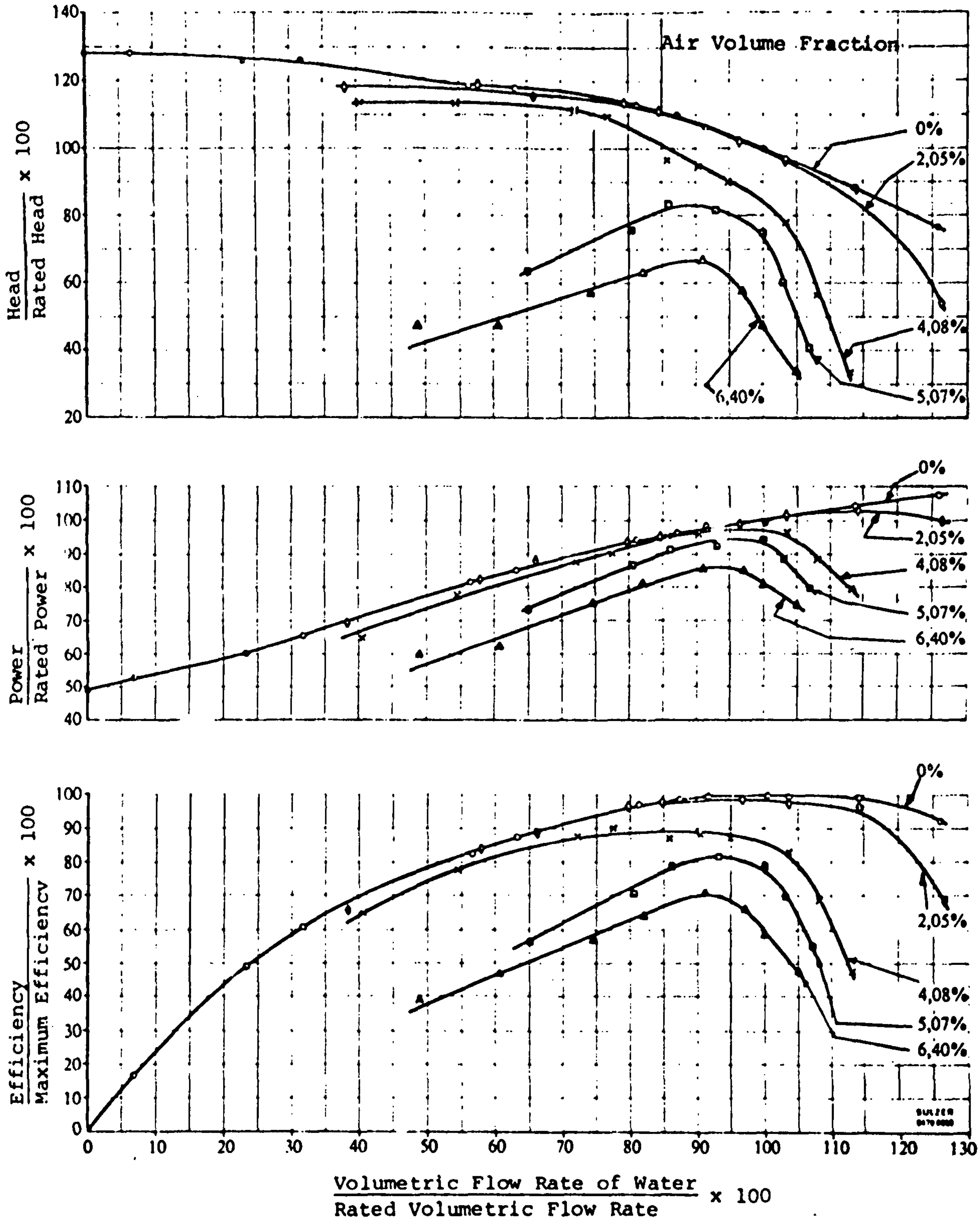

Figure 3-7. NORMALIZED HEAD, POWER AND EFFICIENCY IN AIR/WATER FLOW FROM FLORJANCIC [32]

Reference [32] also contains curves for tests at higher inlet pressures and on multistage pumps. Figures 3-8 and 3-9 also from the reference, show these data in a convenient form. These figures show that at a given air volume fraction, degradation in performance is less pronounced at higher inlet pressures and in multistage pumps. The author of the paper attributes both these trends to the reduced volumetric expansion of air from the air ingestion point to the impeller inlet at higher suction pressures. In multistage pumps, air is raised to a higher pressure at each stage and has less effect on the performance of the next stage.

3.1.4.4 Data From Stepanoff [20]

Stepanoff [20] presents data on the air/water performance of two pumps; one operating at atmospheric suction pressure and the other at a suction pressure of 50 psig. These data, from 1929, were later normalized by Love [13]. The normalized curves presented in Figures 3-10 and 3-11 were reproduced from the report by Sozzi and Burnette [14]. However, the data have not been used to assess RHR and CS pump performance because of insufficient documentation of the test methods employed.

3.1.4.5 Summary of Air/Water Data on 800 to 2000 Specific Speed Pumps

The data on the air/water performance of 800 to 2000 specific speed pumps reviewed here, indicate that pump performance is unaffected by air volume fractions up to 2% at atmospheric suction pressure for flow rates near best efficiency point. For very low flows and for large flow rates, the degradation increases and the increases are pump dependent. Also, the effect of a given air fraction on the performance is less pronounced at higher suction pressures due to lesser expansion of air from the suction to the impeller inlet.

3.2 Data on the Effect of Air Ingestion on NPSH

The literature search revealed only two sources containing data on the effect of air ingestion on the NPSH required. These are reviewed in the following subsections.

3.2.1 Data From Merry [31]

Cavitation performance for the 1074 specific speed pump are presented in Figure 3-12 for various levels of air volume fraction. The performance is given in terms of specific energy rise versus net positive specific energy at the pump inlet. The NPSE was varied by applying compressed air or vacuum to the tank shown in Figure 3-4.

At high values of NPSE, the curves are almost horizontal and show a decline in head rise (specific energy rise) as the air volume fraction increases. At low NPSE values, the curves drop off markedly and the values of NPSE at which the drop begins depends on the air volume fraction through the pump. This behavior demonstrates that as air ingestion rates increase, additional NPSH is required to prevent performance degradation.

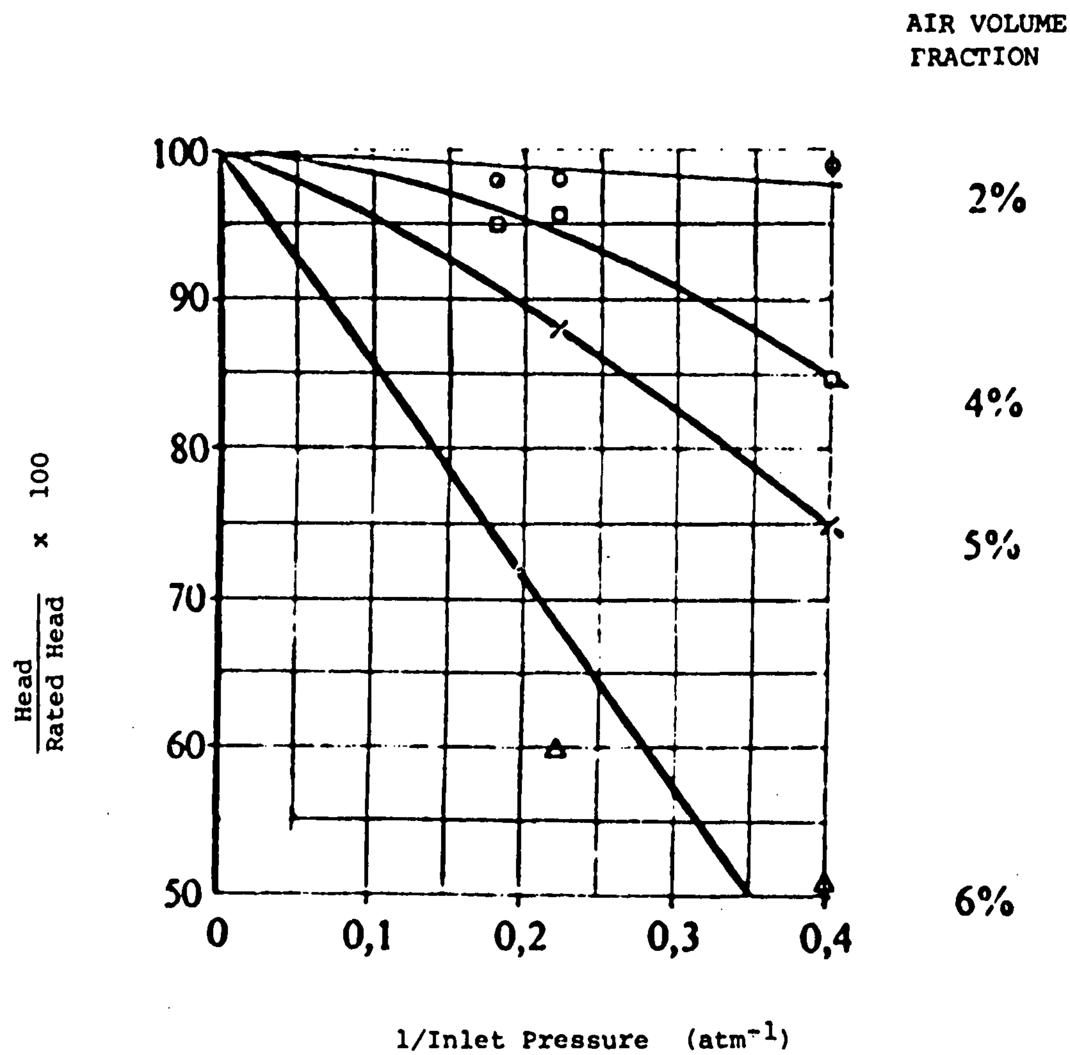

Figure 3-8. EFFECT OF INLET PRESSURE ON AIR/WATER PERFORMANCE FROM FLORJANCIC [32]

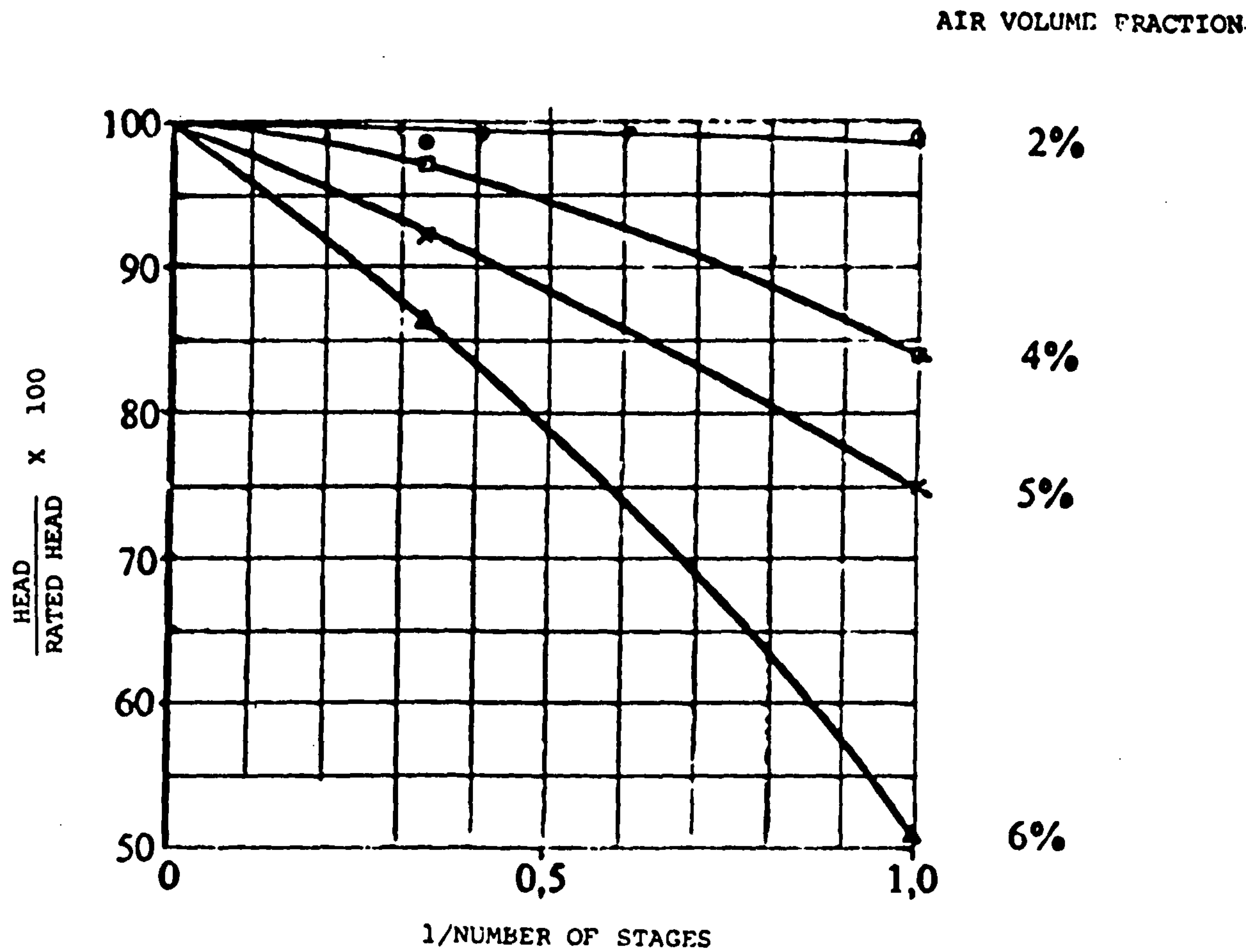

Figure 3-9. EFFECT OF NUMBER OF STAGES ON AIR/WATER PERFORMANCE FROM FLORJANCIC [32]

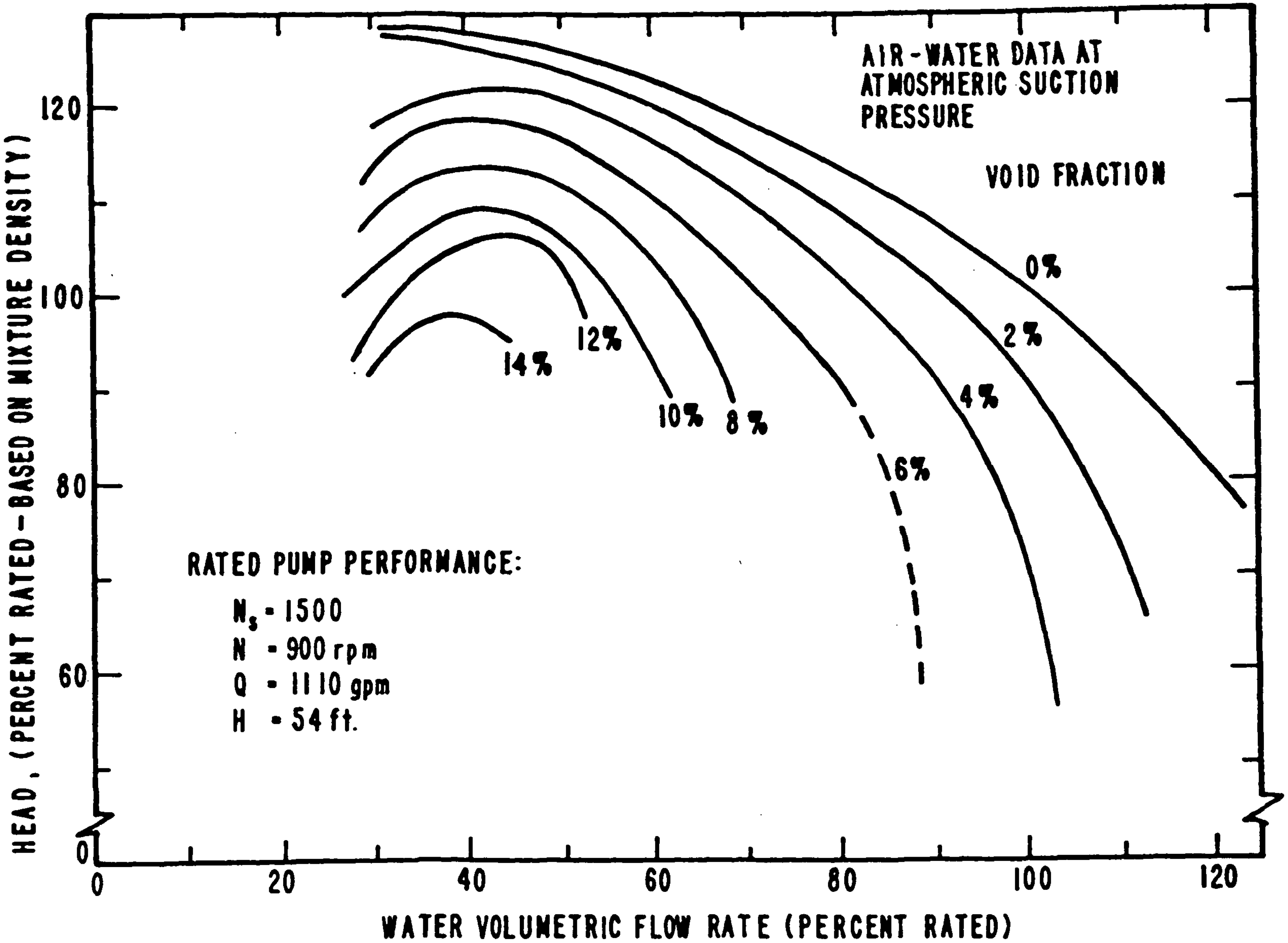

Figure 3-10 AIR/WATER HEAD CHARACTERISTICS AT ATMOSPHERIC SUCTION PRESSURE FROM STEPANOFF [14]

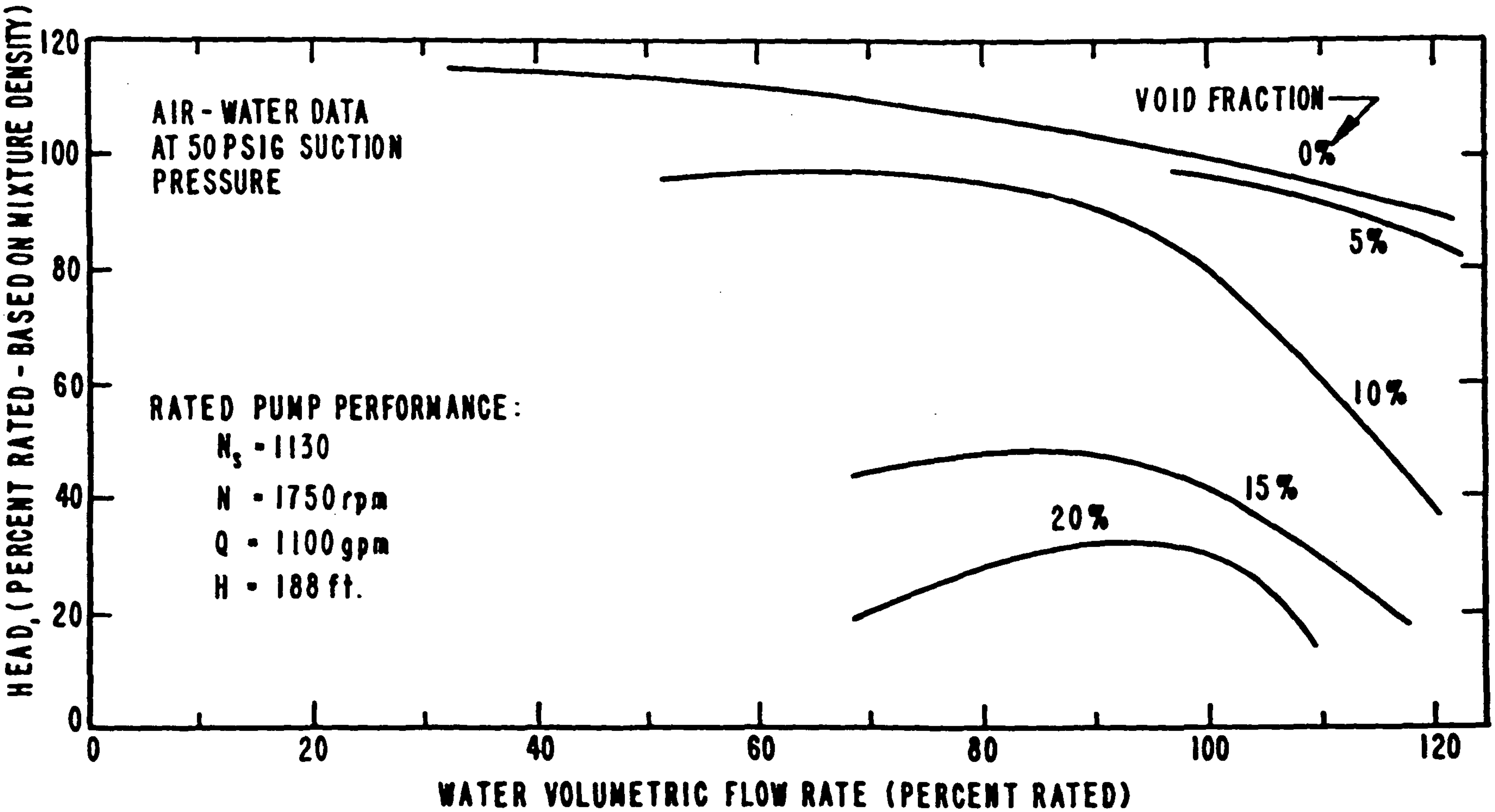

Figure 3-11. AIR/WATER HEAD CHARACTERISTICS AT 50 PSIG SUCTION PRESSURE FROM STEPANOFF [14]

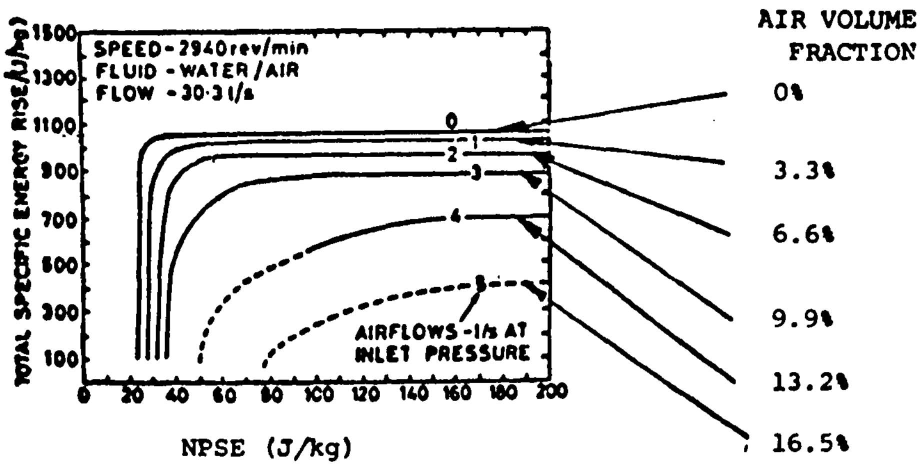

Figure 3-12. CAVITATION CHARACTERISTICS IN AIR/WATER FLOW FROM MERRY [31]

3.2.2 Data From Arie and Fukusako [23]

The experimental setup and details of the impeller are shown in Figure 3-13. The specific speed ot the pump is 2570 which is slightly higher than the range of interest for RHR and CS pumps. Pressures at the suction and discharge legs were measured using manometers and the water flow rate was measured by an orifice meter. The NPSH was varied by applying vacuum to the tank.

Figure 3-14 shows curves of head vs. NPSH. Each curve is for a different air mass flow rate. The liquid flow rate and pump speed are constant at 29.157 kg/sec (460 gpm) and 1200 rpm respectively. The trend in these figures is the same as that described in Section 3.2.1. As the air fraction increases, the pump requires higher NPSH to operate satisfactorily.

3.2.3 Summary of Data on the Effect of Air Ingestion on Cavitation

For pumps operating in single-phase liquid flow at a fixed flow and speed, a change in the available NPSH causes only minor changes in head above a limiting value of NPSH. For NPSH values below this limit, the head drops off abruptly due to vapor formation by cavitation.

The presence of air at the inlet, in addition to causing small or large amounts of degradation in performance, increases the limiting NPSH required for satisfactory operation. The increased degradation at the pump inlet, as inlet NPSH or pressure is lowered, results from the increased volumetric expansion of air between the pump inlet flange and the impeller inlet. Thus, pumps operating with air ingestion will have higher NPSH requirements than those required in single-phase operation.

3.3 Data on Anticipated Debris Through Pumps

Comprehensive assessments of the quantities and types of debris likely to be generated during a LOCA and transported to the sump screens have been reported in [4]. However, estimates of the types and quantities of debris likely to be transported through the screens and into pump suction lines are not well quantified. In practice, several series of screens are located around the containment sumps. They are designed to prevent debris that could clog spray nozzles from reaching the pump suction lines. Spray nozzle orifices typically have diameters of the order of ¼", screens typically have mesh sizes of ¼" or less. Hence, in the assessment of pump performance with respect to debris, only the effects of relatively small particulates and fibers need to be addressed. Potential types of particulates which are likely to pass through the screens have been identified. They include:

- fibers from fibrous insulation
- precipitated hydroxides of aluminum and zinc from components in containment
- paint flakes from "unqualified" coated surfaces
- concrete dust from the floor.

All dimensions are in millimeters.

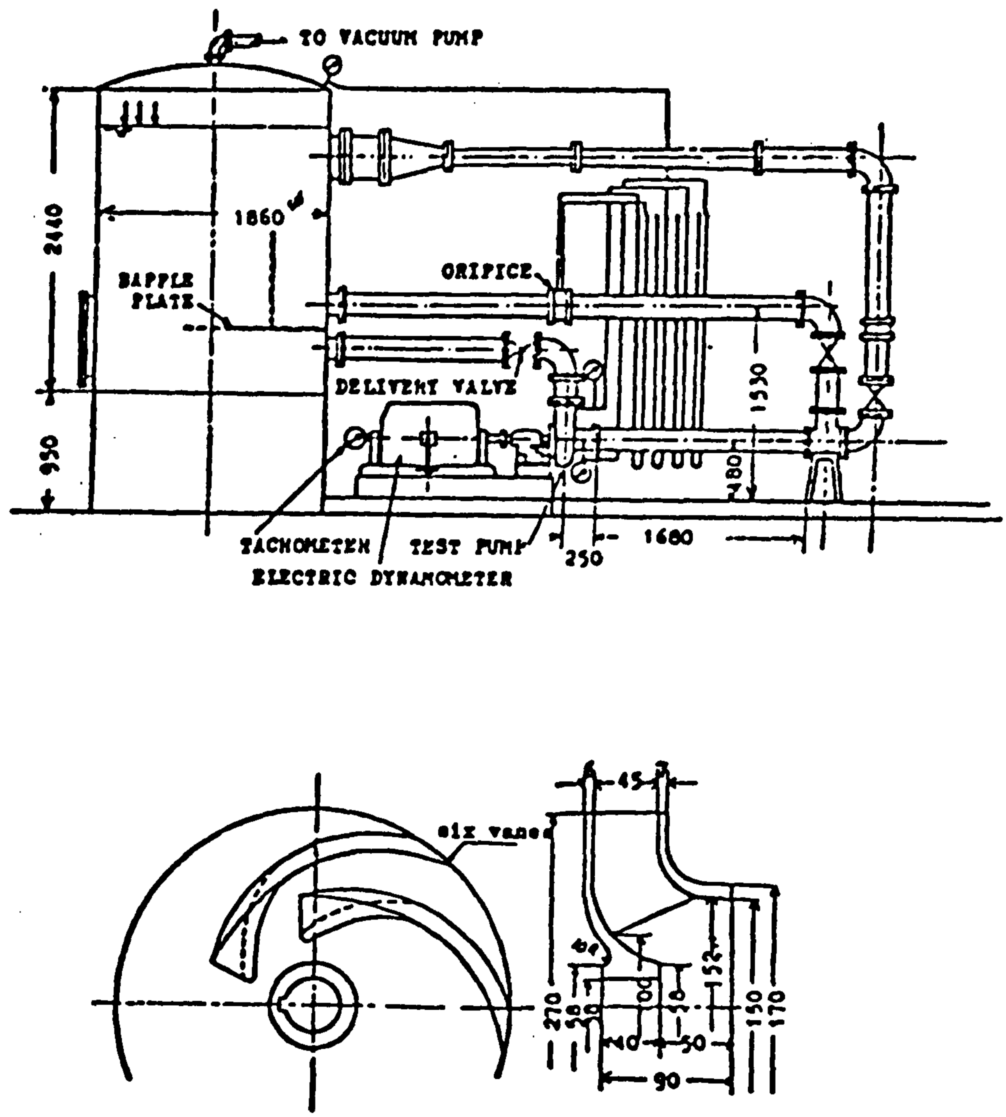

Figure 3-13. <u>EXPERIMENTAL SETUP AND IMPELLER GEOMETRY FROM ARIE AND FUKUSAKO [33]</u>

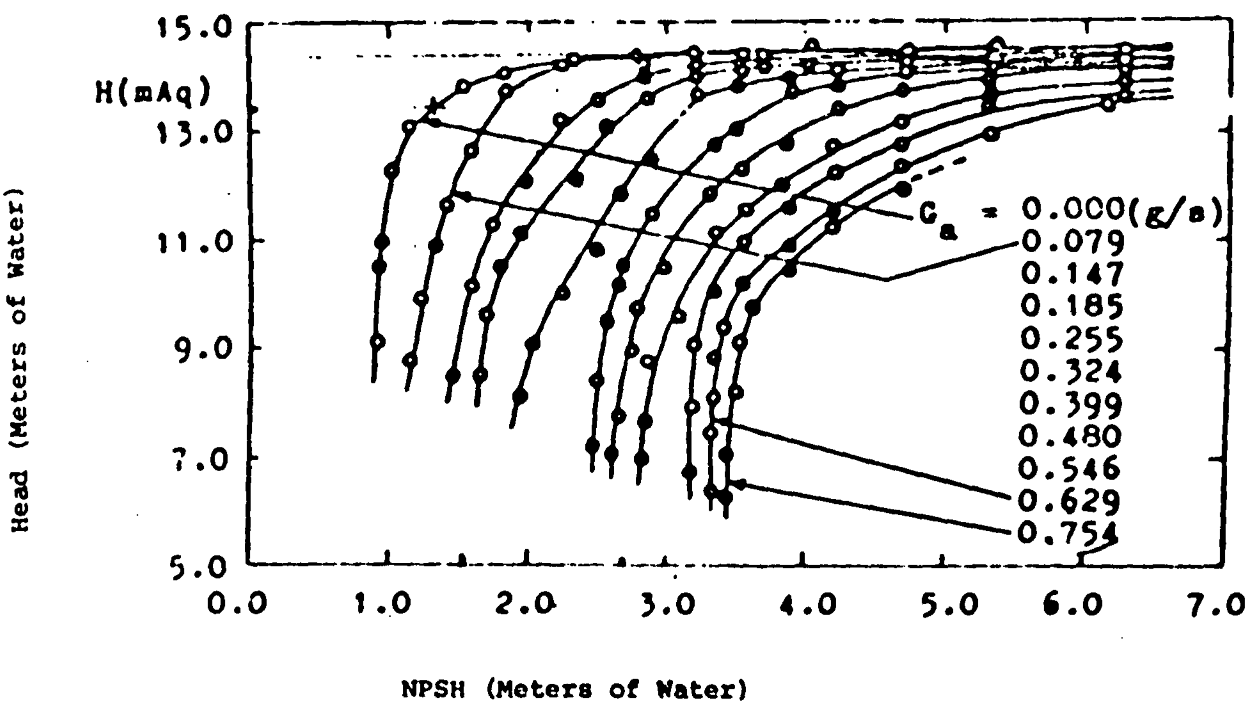

Figure 3-14. CAVITATION CHARACTERISTICS FOR AIR/WATER FLOW FROM ARIE AND FUKUSAKO [33]

The following paragraphs summarize the characteristics of these particulates and provide our estimates for their concentrations.

3.3.1 Insulation Fibers

Fibrous-type insulation materials used in nuclear plants have been identified in [3] and [4]. They are

- o Mineral fiber blankets,
- o Fiberglass insulation.

Little is known about how these types of insulation break up under the forces of jets. However, one study [34] indicates that "fluffy fragments" and "fine suspended fibers" will be formed as a consequence of LOCA jet interaction with fibrous insulation. That study also provides estimates which indicate that these two forms of fibrous debris will constitute 60%-80% of the insulation debris generated by direct interaction with a LOCA jet.

References [3] and [4] describe a methodology for assessing insulation debris generated during a LOCA and provide estimates of the types and quantities of debris generated at several plants. This information is intended to be used in calculating the effects of debris on screen blockage and pressure drop at the containment sump. A major assumption used in that methodology is that fibrous debris which reaches the screens forms a mat surrounding the screens, i.e., none gets through. This is a conservative approach useful for estimating "worst case" pressure losses at the sump screens and is not used, for the purpose of assessing the effects of fibrous debris ingestion on pump performance.

In order to assess the likely consequence of fibrous debris ingested by the pumps, a conservative, bounding estimate of the concentration of debris in the fluid is made. Using an example from [4], in which large quantities of fibrous type insulation are present, conservative estimates of the concentration of fibrous debris can be made by assuming that all fibrous insulation debris generated and transported to the screens following a LOCA (using worst case pipe break results), passes through the screens and pumps. The average concentration is obtained by dividing the transported volume of fibrous insulation by the recirculating water volume inventory. This method of estimating concentration is conservative in that it assumes that all transported fibrous debris is transported through the sump screens. In reality it is more likely that most of the fibrous debris reaching the screens is stopped by them.

Table 3-3 shows estimated concentrations of fibrous debris for a plant with large quantities of fibrous insulation calculated on the basis of the procedure mentioned above. The values in the table show that if <u>all</u> fibrous debris generated during a LOCA and transported to the screens were recirculated through the system in the form of small fibers, the volumetric concentration in the recirculating water would be about 0.3% maximum. For reference, the concentration of fibrous debris (3%) which would result if

100% of the fibrous insulation generated during a LOCA was uniformly mixed with the recirculated water volume is also shown in the table.

TABLE 3-3

ESTIMATES OF FIBROUS DEBRIS CONCENTRATIONS

Total Fibrous Debris at Screens	Total Fibrous Debris Generated
0.3%	3.0%

3.3.2 Aluminum and Zinc Hydroxide Precipitates

A study by United Engineers for Seabrook 1&2 [35] provides calculated estimates for precipitates of hydroxides of aluminum and zinc which are likely to be formed during a LOCA. This is the only source of such information which could be found. The study was based on the corrosion of exposed surfaces within containment by the usable volume of borated water in recirculation at temperatures expected during the recirculation mode. Approximately 1 ton of exposed aluminum and 12 tons of exposed zinc were identified within containment. The total calculated mass of borated water in recirculation was 3.7×10^6 lb (approximately 440,000 gal.). After 30 days, calculations predicted that approximately 3000 lbs of $Zn(OH)_2$ and 2000 lbs of $AlO(OH)$ had precipitated. The projected particle sizes are such that it can be assumed that all precipitated products will be transported to the sump screens.

Properties and mass concentrations of the precipitated products are listed in Table 3-4.

TABLE 3-4

CHARACTERISTICS OF HYDROXIDE PRECIPITATES
(30 day products)

Material	Density (g/cc)	Mass Concentration %	Hardness (Moh)	Particle Size (μ)
$Zn(OH)_2$	3.05	0.08	Soft	<1
Boehmite[AlO(OH)]	3.01	0.05	3.5-4.0	1-10
Diaspore[AlO(OH)]	3.3-3.5	0.05	6.5-7.0	1-10
Gibbsite[Al_2O_3 $3H_2O$]	2.5-3.5	0.05	2.5-3.5	1-10

Within the 30 day period considered, most AlO(OH) will "age" into diaspore which will be the dominant form in recirculation. The mass concentrations result from the estimates of individual products given above. Volume concentrations for $Zn(OH)_2$ and AlO(OH) are 0.025% and 0.015%, respectively.

Long term estimates (~1 year) based on corrosion rates at about 70°F indicate that final precipitate concentrations will be 0.12% $Zn(OH)_2$ and 0.11% AlO(OH) by mass or about 0.04% each by volume.

3.3.3 Paint Flakes

The possibility exists for polymeric coatings of surfaces within the containment to become dislodged and reduced to fairly small particles during a LOCA. If particles are small enough, some may pass through sump screens and be pumped through the recirculation system. Detailed estimates of the quantities or properties of these coatings were not available at the time of this writing. However, a conservative estimate of the concentrations can be obtained by assuming that a "large" surface area of paint is dislodged, that all of it is transported through the screens in particulate form and that it mixes with water volumes of the order of the volume of the RWST (i.e. ~250,000 gal.). Assuming a coating thickness of about 0.020 inches and 5000 ft^2 of coating dislodged (comparable to large quantities of insulation debris [36]) then the volume concentration would be 0.025% by volume. This estimate may be modified as necessary for assessment in individual plants where more information is available about paint surface areas and quantities of debris generated.

3.3.4 Concrete Dust

Our estimate of the quantity of concrete dust is arbitrary. We assume a 0.01 inch thick layer uniformly distributed across the floor of a 160 feet diameter surface to be mixed homogeneously with 250,000 gallons of recirculating water. The resulting concentrations are 0.05% by volume or 0.02% by mass. Particle sizes are estimated to be of the order of 100 μ.

3.4 Data on the Effects of Particulates on Pump Performance

Slurry technology provides the base for most published data on pump performance during particulate ingestion. Published data generally deal with the degradation in performance (or increased power requirements) and occasionally with material erosion which may occur when solids of various types are being pumped. Abrasive solids concentrations of 10-20% and above by volume are typical in slurry applications. As noted in the previous section, conservative estimates indicate that particulate concentrations in RHR and CS pumps during recirculation are expected to be less than 1%. Although pump performance varies substantially depending on the pump design and the characteristics of solids in the fluid, reasonable conclusions can be drawn from the available data base.

3.4.1 Data of Fairbank [37]

Fairbank [37] investigated the effects of two grades of sand and of oil-well drilling mud on the performance of a 6 inch diameter centrifugal pump. The specific speed of the pump at best efficiency point was 1198. Tests were conducted in a closed-loop facility and power, flow rate, speed and differential pressure were recorded. Characteristics of the suspended solids are shown in Table 3-5.

TABLE 3-5

SUSPENDED SOLIDS CHARACTERISTICS - FAIRBANK

Material	Specific Gravity	Median Diameter (mm)	Concentrations Tested (% by volume)
Monterey No. 4 Sand	2.655	0.800	0-20
Crushed Del Monte Sand	2.630	0.034	0-20
Mohave Rotary Mud	2.726	<.001	0-7.7

The results of the tests in the form of head vs. flow rate using drilling mud and Monterey sand are shown in Figure 3-15(a). The test data have been normalized to head and flow rate values at best efficiency point, H_{bep} and Q_{bep}, respectively. Test results using the crushed sand are not shown.

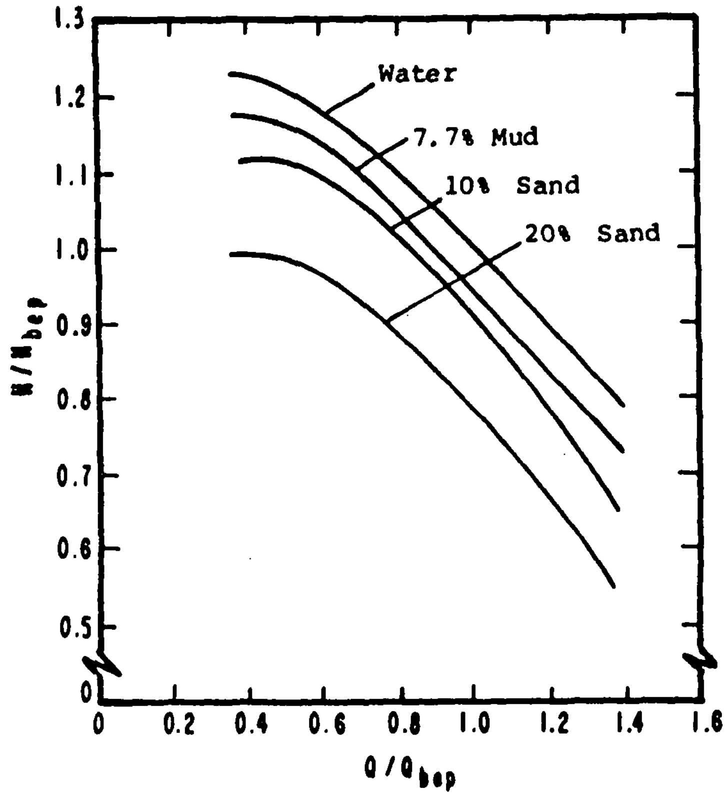

Figure 3-15a. PUMP PERFORMANCE IN WATER, SAND AND DRILLING MUD. CONCENTRATIONS IN PERCENT BY VOLUME [37]

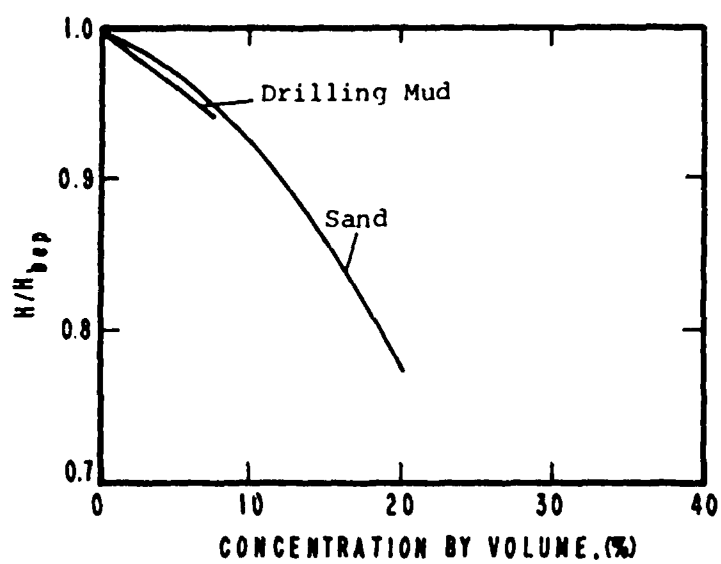

Figure 3-15b. HEAD DEGRADATION AT Q_{bep} DUE TO PARTICULATES [37]

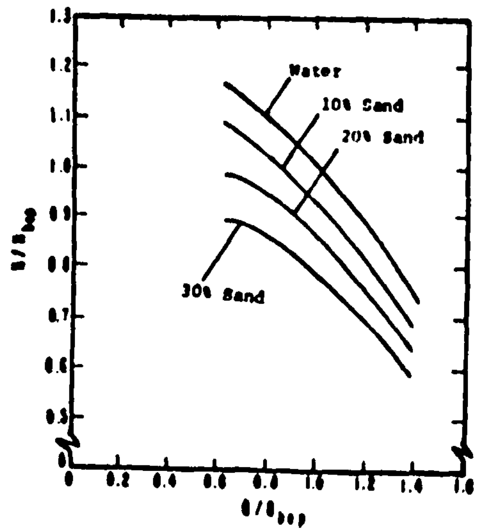

Figure 3-16a. PUMP PERFORMANCE IN SAND/WATER SLURRY: SAND PARTICLE DIAMETER = 0.58 mm, CONCENTRATIONS IN % BY VOLUME [38]

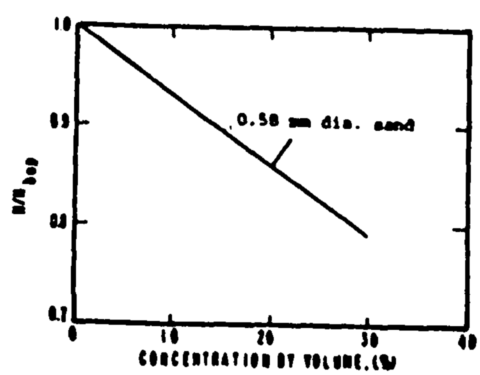

Figure 3-16b. HEAD DEGRADATION AT Q_{bep} DUE TO SAND [38]

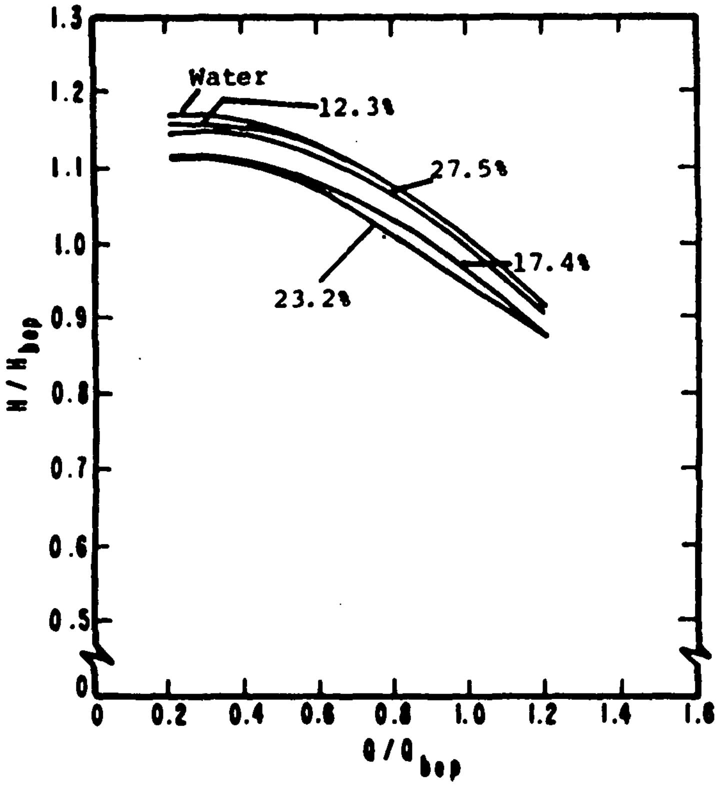

Figure 3-17a. PERFORMANCE OF CENTRIFUGAL PUMP IN WATER AND IN SILT-CLAY-WATER MIXTURES. CONCENTRATIONS IN PERCENT BY VOLUME [39]

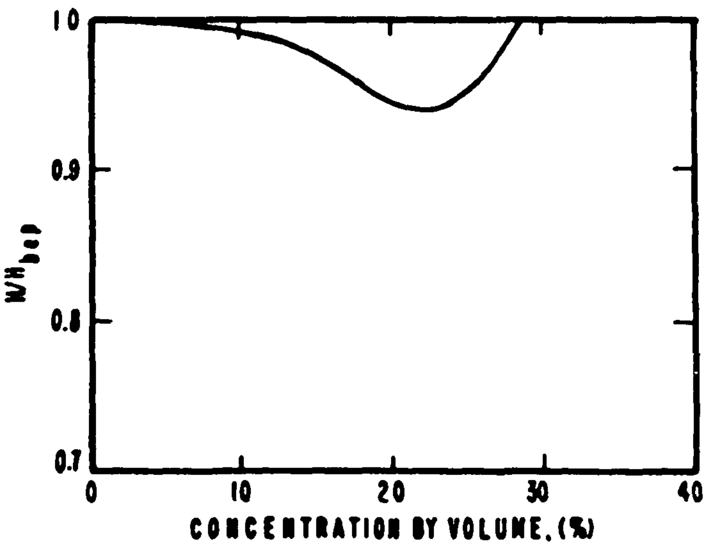

Figure 3-17b. HEAD DEGRADATION OF PUMP AT Q_{bep} IN SILT-CLAY-WATER MIXTURES [39]

The pump performance with crushed sand was identical to clear liquid operation for all test conditions and solids concentrations.

The relationship between head degradation and solids concentration at best efficiency flow rate is shown in Figure 3-15(b). Data for both the drilling mud and Monterey sand are included in the plot. For concentrations by volume of about 1%, performance degradation is about 1%.

3.4.2 Data of Vocadlo et al. [38]

Vocadlo et al. [38] have published test data on pump performance based on test data using four slurry pumps with specific speeds in the range of 1080-1300. The pumps tested had three impeller vanes each and were somewhat different in overall geometry than typical RHR and CS pumps.

Slurry mixtures with several grades of sand (diameters from 0.105 mm to 2 mm) were used in pumps with internal metal components and with rubber liners to provide a data base for a head loss correlation. Among other results, the tests showed that losses increase with concentration and with particulate size. Typical results from the tests on a metal pump are shown in Figure 3-16(a). Normalized head and flow rate are plotted for several concentrations of 0.58 mm diameter sand. The head degradation at the best efficiency flow rate for the same pump is shown as a function of volumetric concentration in Figure 3-16(b). For concentrations of about 1%, degradation is about 1%.

3.4.3 Data of Herbich [39]

The tests in this study were conducted on several 10.5 inch diameter dredge pump impellers with rating points near a specific speed of 1260 in silt-clay-water mixtures. Tests were conducted in a closed loop facility with solids mixtures varying in concentrations up to 27½% by volume. 99.5% of the solids were less than 0.155 mm in diameter.

Typical test results for several mixture concentrations are shown in Figure 3-17(a) in terms of normalized head versus normalized flow rate. The results show that degradation occurs for some concentrations. However, his conclusion on the overall results on all tests conducted was that the observed degradation was within the scatter of the performance data, i.e., virtually no degradation occurs. Figure 3-17(b) shows normalized head degradation at the best efficiency flow rate as a function of mixture concentration. For concentrations of about 1%, degradation is negligible.

3.5 Data on the Effects of Particulates on Pump Wear

The mechanical effects of particulates on impellers, casings, seals, and wearing rings are also considered in assessing the overall performance of CS and RHR pumps. These effects are highly dependant on individual pump designs and particulate characteristics. Much of the information on wear behavior was obtained from manufacturers of RHR and CS pumps and from manufacturers of mechanical face seals.

In general, very little technical information exists in the public domain which can be applied to specific pump or seal designs to predict the effects of particulates in terms of mechanical wear. Seal manufacturers have conducted tests to provide some indication of the effects of particulates, and some of these data have been made available for this report. The results of these tests, together with their opinions based on experience, are provided in the following paragraphs.

3.5.1 Data of Doolin [40]

Doolin [40] reports on the results of comparative tests to evaluate degradation in performance due to abrasive wear on open and closed impellers. The tests were conducted on cast iron impellers using a slurry mixture of diatomaceous earth and water. Concentration of the abrasive solids was about 10% by weight (roughly 5% by volume).

The tests on the closed impeller showed that extensive impeller erosion had occurred after testing had run for about 460 hours. Wearing ring clearances had opened from 0.015 inch to 0.050 inch (diametral) and efficiency had dropped about 10%.

3.5.2 Shaft Seals

Test data on the performance of mechanical shaft seals is sparse, and much of it is proprietary to seal manufacturers. Both Durametallic Corp. [42] and Crane Company [43] have evaluated seal performance at elevated temperatures in boric acid solutions.

Crane Company [43] has performed tests on Type 1 seals (a model commonly used in RHR and CS pumps) in a 2% boric acid solution to evaluate performance data in terms of wear life, leakage and overall operating capability at elevated temperatures. The seals were tested for a range of conditions that simulate emergency conditions in nuclear power plant safety injection systems. The results of projected seal life are as follows.

Operating Condition		Projected Seal Life
Severe	300°F-200 psig	1920 hrs
Less Severe	300°F- 60 psig	2970 hrs
Normal	160°F-400 psig	>3 years

A leakage rate of >100 cc/hr. was chosen as the criterion for seal failure.

Tests on seal safety bushings were also conducted by Crane Company to evaluate leakage under normal and severe operating conditions. In the event of complete seal failure, the clearance between the shaft and the safety bushing ultimately determines leakage quantities. Testing for "normal" conditions included operation at 180°F without fluid for 500 hours and for 100 hours with injection of 2% boric acid solution. Leakage through the shaft-bushing clearance after these tests was 10 gph. "Severe" conditions included 250 hours of operation at 300°F with 15 minutes/day intermittent spray of boric acid and 500 hours operation at 160° to 200°F with continuous injection of 2% boric acid solution. At the conclusion of these tests, leakage had increased to 70 gph.

Durametallic Corp. [42] has also conducted tests of their seals for nuclear power plant auxiliary and cooldown pumps. They report that seal life is shortened due to high temperatures, pressures and the presence of boric acid. They conclude that seal life may be as low as 500 hours under the extreme condition of continuous operation at 350°F and 400 psig. They state that expected seal life appears adequate and should be at least a year (8760 hours under worst anticipated conditions) and may be as much as two to five years under actual operating conditions.

Tests conducted by Durametallic on their safety back-up bushing show that the leakage rates under normal conditions for a 3 inch diameter bushing are about 80 gph at 60 psig for a ½ inch long bushing and 47 gph for a 3/4 inch long bushing.

There have been some isolated tests conducted by individual parties on a proprietary basis to assess the effects of particulates on seal performance. However, these results were not available.

3.6 Technical Input from Manufacturers

An important part of the information gathering for this study consisted of seeking direct input from seal and pump specialists in industry. Personal and telephone interviews were conducted to obtain:

1) statements and opinions on the relevance of data in the technical literature to RHR and CS pump performance,

2) specific nonpublished vendor information on the influence of entrained air and particulates on pump performance, and

3) the benefit of the experiences of many specialists with first-hand knowledge of the operation of these types of pumps.

Except where explicitly noted in this report, the statements of these specialists represent personal opinions based on their experience and do not represent official policies or positions of their respective firms.

Many individuals and firms were contacted to gather information for this project. Of these, those who were most helpful are listed in the Acknowledgements. The following individuals provided the bulk of the key reference information used to corroborate and verify the conclusions of this report.

Pump Specialists

Personal interviews were conducted with:

- Mr. J. H. Doolin, Manager-Engineering, Worthington Pump Group, McGraw Edison Company
- Mr. W. H. Fraser, Chief Hydraulic Engineer, Worthington Pump Group, McGraw Edison Company
- Mr. Fred Antunes, Chief Engineer, Ingersoll-Rand
- Mr. Phillip Nagangast, Manager of Engineering Analysis, Engineered Pump Division, Ingersoll-Rand
- Dr. Paul Cooper, Ingersoll Rand Research, Inc.
- Mr. Fred Buse, Chief Engineer, Standard Pump Division, Ingersoll Rand

Seal Specialists

Telephone interviews were conducted with:

- Mr. Bill Adams, Director of Engineering, Durametallic, Inc.
- Mr. Jon Hamaker, Assistant Chief Engineer, Crane Packing Company

The input from the specialists listed above and from others who supplied information is in the form of internal reports from tests as well as statements of opinion based on experience. The information provided is interspersed within the text of this report by topic, and where generally available documents exist, they have been referenced. In addition to the formal, documented information received, the following paragraphs summarize verbal input regarding the performance of pumps in air and debris ingesting conditions.

Air Ingestion

The pump specialists confirm that the data from the technical literature on the performance of low specific speed pumps under air ingesting conditions confirms their experiences. For low levels of air ingestion performance degradation at flows near best efficiency is negligible. (Opinions on the level of air ingestion giving negligible degradation varied from 1% to 3%.) There was general agreement that for flows less than 50% of best efficiency, the presence of air might cause air binding, depending on pump design. All were aware of the effects of air on increasing NPSH requirements, although no quantitative data on this effect was available. There was also general agreement that for air quantities between 3% and 15% pump degradation depends on individual pump design and operating conditions, and for air quantities greater than 15% performance of most pumps will be fully degraded.

Debris Ingestion

Opinions of pump specialists differ with respect to the long term damage which may occur in impellers or pump casings under particulate ingesting conditions. The minimum quantity identified for which no wear is expected is 100 parts per million of fine abrasives. The maximum quantity identified for which wear should be tolerable over an extended period is 1% by weight. In general, the pump specialists agreed that soft, fibrous debris at volumetric concentrations less than 1% should not impair pump performance. These opinions derived from experience in slurry pumping, the pumping of sewage and sand/water mixtures, and from internal tests on shrouded impellers pumping paper stock [41].

Seal specialists, in addition to providing test information cited in the previous sections also provided details of seal designs. Their data and opinions indicate that seal wear may occur as a result of hard, fine particules in the 3 to 10 m size range (comparable to predicted sizes of Diaspoze). The effect of increased wear is increased leakage up to the limiting values set by the seal safety bushing. The effects of soft debris are to either cause seal flush passage clogging or spring hand-up. In the event of seal failure resulting from either of these mechanisms, the safety bushing again sets the limiting value of leakage. Leakage tests on safety bushings typical of those used in RHR and CS pumps show that leakage rates are less than 100 gph.

4 TECHNICAL FINDINGS

This section presents the technical findings of the Creare study on the effects of air and debris ingestion on the performance of RHR and CS pumps operating in the recirculating mode. These findings should be accepted with an awareness of their limitations, since they are based on empirical information on pumps which are similar in design and performance to RHR and CS pumps. No tests on RHR or CS pumps have been documented in which the effects of air or debris ingestion were studied. The mechanism of performance degradation due to air ingestion is not clearly understood at present. Questions remain as to the effect of various factors such as pump size, specific speed, fluid properties, operating conditions, and other variables on the amount of degradation for a given volumetric fraction of ingested air at the pump inlet. Similarly, the mechanisms which control wear and erosion in pumps due to abrasive particulates or due to cavitation are not well understood. Those factors important to wear include particulate size, density and hardness, material properties, fluid velocities and in the case of cavitation damage, water temperature and inlet conditions.

Lacking proper analytical tools, the approach adopted in this study to assess the effects of air and debris on pump performance was as follows:

- characterize common features typical of RHR and CS pump,
- estimate air and debris quantities and types likely to be present during RHR and CS operation,
- identify experimental results dealing with air and particulates on pumps having characteristics similar to those of RHR and CS pumps,
- gain additional insight through interviews with pump manufacturers, consultants and seal specialists, and
- deduce from the above the effects of air and debris on RHR and CS pumps.

The discussion presented in this section condenses the information presented thus far in the context of RHR and CS pump operation in the recirculating mode. Guidelines are suggested for acceptable levels of air ingestion. A method for calculating pump inlet conditions is outlined. This method can be used with the guidelines for air ingestion to assess RHR and CS pump performance. Conclusions about the effects of debris on pump performance are summarized.

No test data on the air/water performance of RHR and CS pumps were found, either from manufacturers or in the literature. Since the experimental determination of the air/water performance of pumps is difficult, such tests are not routinely conducted by manufacturers. Most air/water performance data available from pump manufacturers are from isolated tests aimed at determining or improving the air-handling ability of pumps used in process plants and those used to pump paper stock, which are similar to RHR and CS pumps. These data were not used to assess the air/water performance of RHR and CS pumps since the test conditions and impeller geometries were not sufficiently well documented. However, the trends observed from many such tests agree with those from better documented sources in the literature.

The conclusions on the air/water performance of RHR and CS pumps presented in this report are based on test data from the literature. Although the literature survey identified many sources of information on two-phase flows in pumps, only three met the following criteria established to determine the applicability of test results to this study:

1. Specific Speed - The test pumps were required to have specific speeds in the same range as that for RHR and CS pumps (800 - 2000). This criterion eliminated a considerable volume of literature on the air/water performance of primary coolant pumps and axial flow pumps; these pumps have higher specific speeds and somewhat different hydraulic performance characteristics.

2. Documentation - A reasonable amount of care should have been demonstrated during experimentation. The results of the tests should have been well documented. This criterion precluded the use of several sets of test results which have appeared in trade journals and in texts.

3. Size - Although recent two-phase pump test results [12] indicate that geometric scale effects may be minor, there is insufficient experimental evidence to warrant generalization. The RHR and CS pumps surveyed by Burns and Roe [3] and [4] have impeller discharge diameters up to about 20". Data on test pumps less than about 1/3 this size were not included.

4. Fluids - The test fluids were required to be air and water. There is a substantial amount of published information on the air handling capabilities of jet aircraft el pumps, petroleum service pumps and paper stock pumps. This information was not used in determining the air/water performance of RHR and CS pumps.

Although the general trends observed in all sources of two-phase pump performance data are similar, the criteria outlined above to screen data were adhered to in order to strive for accuracy. Experimental data from three sources ([30], [31], [32]) met the criteria above, and the conclusions arrived at in this study on the effects of air on pump performance are derived from the data from these studies.

Conclusions regarding the effects of debris on pump performance were based on the estimation of quantities and types of particulates likely to be transported through the screens, and on relevant technical information available from manufacturers and from the open literature. In Section 3, conservative estimates of the nature and quantities of debris show that fine abrasives may be present in concentrations of about 0.1% by volume (about 400 ppm by weight) and that very conservative estimates of fibrous material yield concentrations of less than 1% by volume. Published data on the effects of particulates on pumps generally deal with particulate concentrations at many times these values. Our technical assessment relies heavily on this information coupled with the experience of manufacturers.

4.1 Effects of Air Ingestion

In Figure 4-1, data from the chosen sources [30], [31], and [32] are plotted on common coordinates at three different flow rates, ranging from 60% to 100% of best efficiency point flow for each respective pump to illustrate the effect of air ingestion on developed head. Table 3-1 gives the rated conditions and other relevant data for the three test pumps.

The abscissa of Figure 4-1 represents the ratio of the volumetric flow rate of air to the total volumetric flow rate of the mixture. It is equal to the time-averaged void fraction in flows with no air/water slip. The degradation in pump performance is shown as the ratio of the pressure rise across the pump in air/water flow to that in single-phase liquid flow at the same speed and flow rate. The dashed line in the figure is the curve obtained accounting only for the density difference due to air ingestion. To better appreciate the density effect, note that in two-phase flows the fluid density is given by,

$$\rho_{fl} = \rho_w(1-\alpha) + \rho_a\alpha \tag{4.1}$$

which, for small air volume fractions ρ_{fl} can be approximated by

$$\rho_{fl} \approx \rho_w(1-\alpha) \tag{4.2}$$

since the air density, ρ_a is much smaller than that of water, ρ_w. Thus, the fluid density decreases nearly linearly with increasing air content in the fluid. Similarity relations for turbomachines establish that the head developed by the pump at a given speed and flow rate is independent of the fluid density for dynamically similar operating conditions. Therefore, the differential pressure across the pump, in two-phase flows, given by

$$(\Delta p)_{2\phi} = \frac{g}{g_c}\rho_{fl}H \tag{4.3}$$

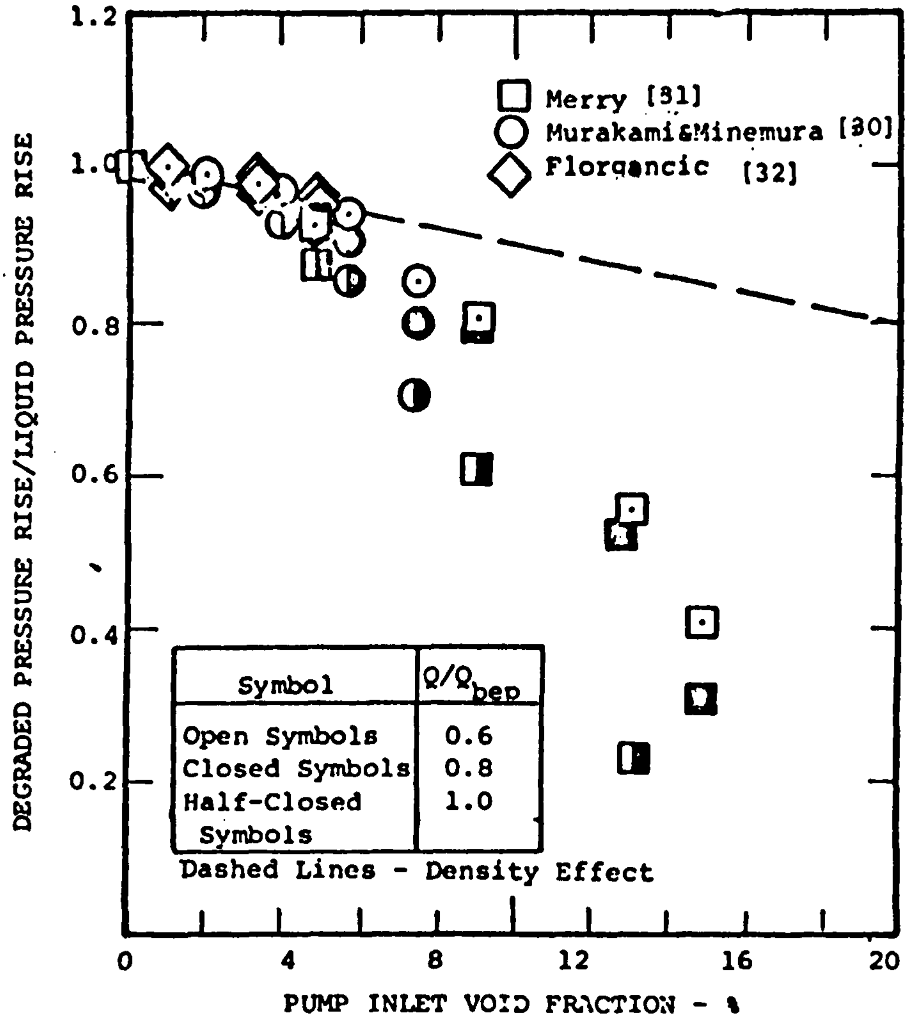

FIGURE 4-1. HEAD DEGRADATION UNDER AIR INGESTING CONDITIONS AS A FUNCTION OF INLET VOID FRACTION (% OF TOTAL FLOW RATE BY VOLUME).

decreases nearly linearly with increasing air volume fraction, α. A similar argument can be used to show that the shaft torque also decreases nearly linearly with increasing air volume fractions. This is true, however, only as long as two-phase effects, such as separation, do not affect flow similarity.

It is evident from Figure 4-1 that for small amounts of air (less than about 3%) the test data closely follow the trend predicted by similarity relations, i.e., there is no degradation in performance except that due to the change in fluid density. However, at increased air quantities, changes in the internal flow occur which violate the "similarity" assumption. The pressure gradients within the impeller combine to produce air cavities which attach themselves to portions of the impeller and pump passages altering the flow pattern from that which exists in single-phase flow. For flow rates near the best efficiency point, the flow distortions due to attached cavities tend to occur along the inlet edges of the blades. For very low flows (less than about 20% of the best efficiency flow rate Q_{bep}) air tends to accumulate at the center of the inlet to the impeller due to strong recirculation [44]. If the flow rate is sufficiently low and if air ingestion occurs over an extended period of time, air can continue to accumulate and the pump ultimately becomes "air bound". Unfortunately, quantitative data on this behavior are not available. What is known is that it is likely to occur at relatively low flow rates and can occur at low air ingestion levels (less than 2% by volume).

The degradation process for air ingestion between 3% and 15% by volume is dependent on operating conditions, pump design and other unidentified variables. However, the trends noted above agree closely with the guidelines commonly adhered to by the pump industry. These state that for air ingestion levels less than about 2%, degradation is not a concern for flows near rated conditions; for ingestion levels in the neighborhood of 5%, performance is dependent on pump design; and for ingestion greater than about 15%, most centrifugal pumps are fully degraded.

Based on the data available, a limit of acceptable air ingestion is established at 2% by volume. All test data show that for ingestion levels up to 2%, negligible degradation occurs. At ingestion rates slightly above (>3%) degradation starts to become pump and operating point dependent. Because of the concern for air binding at very low flows, the 2% applies to pump flow rates at or near best efficiency point. It should be noted that for flow rates at less than 50% of rated flow, chances of air binding are substantial. However, at such low flow rates, sump suction pipe velocities would be half the values at rated conditions (unless the pump is rated at very low flows relative to best efficiency) and the likelihood of air ingestion decreases.

The test data for air/water performance also suggest that the 2% limit be applied only up to flow rates of ≤110% of best efficiency flow rate (not necessarily rated flow). For greater flow rates, the few data that exist indicate that degradation becomes significant. (See Figure 3-7, where Florjancic's results show a 10% head reduction at 120% flow rate).

In addition to the considerations of flow limitations on the 2% allowed air ingestion rate, even small quantities of air affect the NPSH requirements for pumps. The results shown in Figure 4-1 apply to pumps operating with sufficient NPSH to avoid cavitation. The following section deals with the effects of air ingestion on NPSH and the combined effect of low NPSH and air ingestion on head degradation.

4.2 Cavitation and Air Ingestion

There are very few sources of data on the combined effects of cavitation and air ingestion on pump performance. Figure 4-2 shows results from [31] on a pump of specific speed 1074 operating near best efficiency point. The curves have been replotted for Figure 3-12 and head values have been normalized by the non-cavitating liquid head. The curves show cavitation 'breaks' at various levels of air ingestion. For each curve, the flow rate and speed are fixed and inlet pressure (NPSH) is varied. As NPSH decreases, the measured differential head decreases gradually and then abruptly, due to cavitation. The values of head are normalized by the non-cavitating value in liquid with no air.

Applying a commonly (albeit arbitrarily) used criterion of defining the NPSH required as the NPSH value at which head degrades by 3% from the non-cavitating value, one can construct a locus of the required NPSH as a function of the air ingestion level. Figure 4-3 shows four such points obtained by plotting the NPSH values for which head has degraded by 3% from the non-cavitating values. The plotted points are taken from the four curves shown in Figure 4-2 for air fractions of 0%, 3.3%, 6.6% and 9.9%, respectively. In order to establish a guideline for calculating the increased NPSHR in the presence of air, an arbitrary relationship is presented. This relationship is:

$$NPSHR_{air/water} = NPSHR_{water}\ (1+0.5\ AF)$$

where AF is the air volume fraction in percent.

The relationship is shown in Figure 4-3 as a straight line. It is evident from the figure that the equation for NPSH requirements in the presence of air provides a margin above the values obtained by Merry [31]. For example at 2% air volume fraction the NPSH requirement is equivalent to that obtained with 3.3% air volume fraction. The conservatism used in establishing the straight line is arbitrary. However, it is felt necessary because of the limited amount of data available upon which to base such a guideline. It should be noted that the guideline is only intended for use for air volume fractions less than 2%.

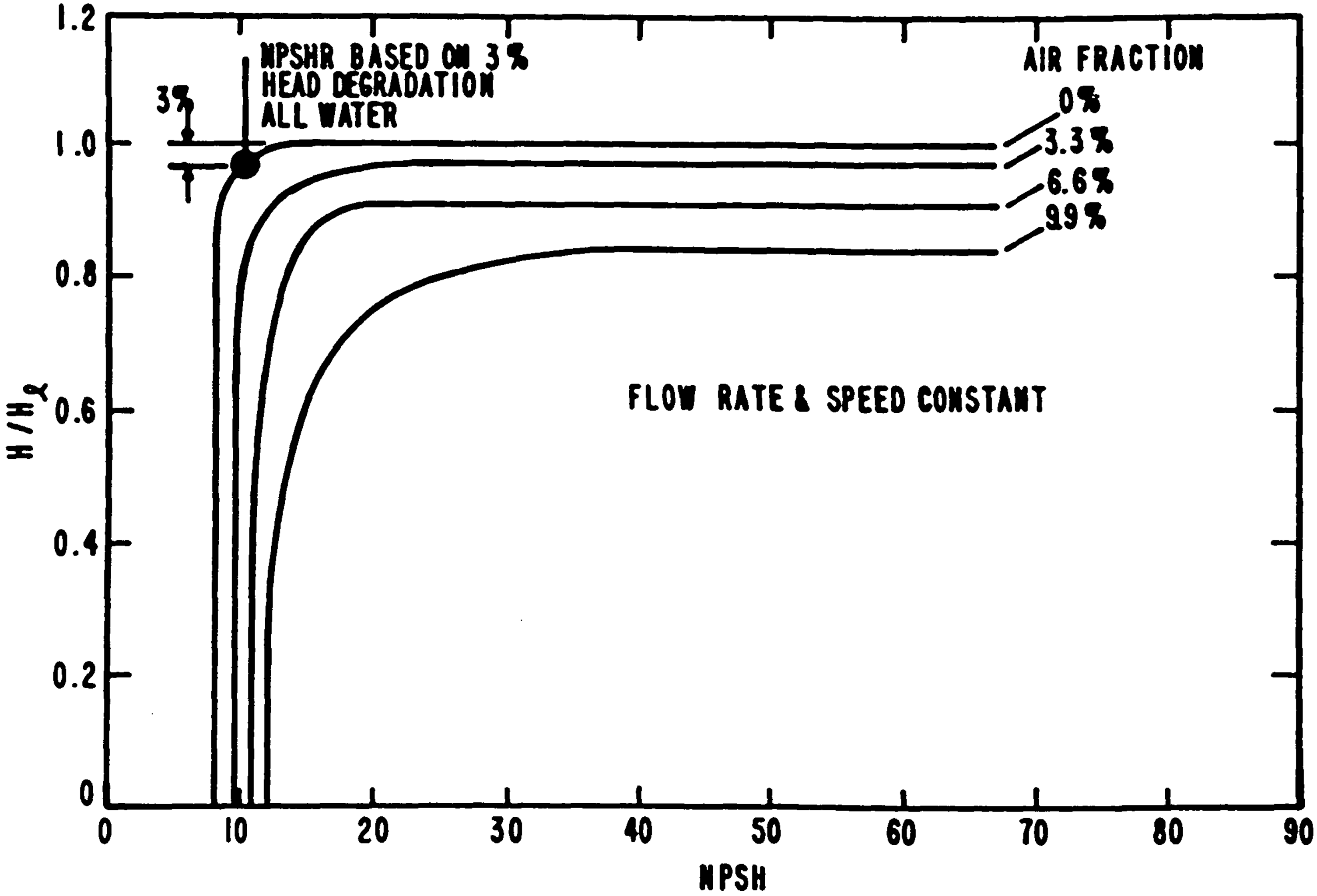

Figure 4-2. NORMALIZED HEAD VS. NPSH AT DIFFERENT VOID FRACTIONS FROM MERRY [31]

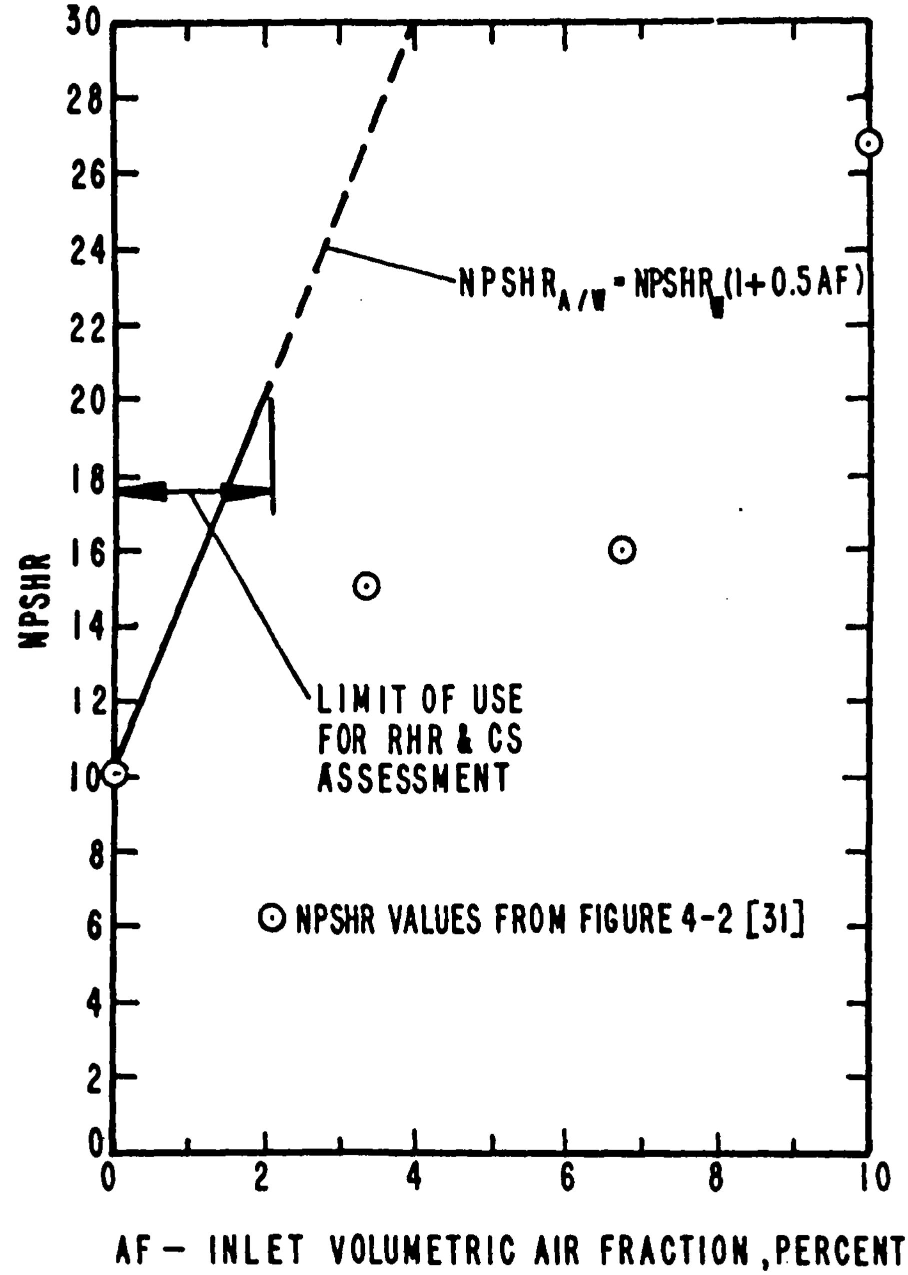

Figure 4-3. NPSH REQUIRED VS. AIR FRACTION BASED ON 3% HEAD DEGRADATION, MERRY [31]

4.3 Cavitation at Elevated Temperatures

This section briefly summarizes the documented behavior of pumps cavitating at elevated fluid temperatures. At high temperatures, cavitation in pumps is influenced by two factors. Firstly, the vapor pressure increases with fluid temperature. Therefore, for a fixed absolute pressure at the sump suction pipe, the NPSH available will drop as the fluid temperature increases. The second effect is that as fluid temperature increases, degradation in pump performance due to cavitation decreases. The trend is illustrated in Figure 4-4 which shows the degradation in head as a function of NPSH for varying temperatures. This effect has been studied extensively in the literature [45], [46], [47] and a curve has been published by the Hydraulic Institute Standards [25] which provides a correction to NPSHR based on water temperature. Figure 4-5 is a reproduction of this correction curve. It is noted that at 300°F, the correction (decrease in NPSHR) applicable to a pump is three feet. It should be noted that Regulatory Guide 1.1 [6], which is used by plant designers, takes into account the decrease in vapor pressure due to increasing temperature but not the decrease in degradation due to temperature increases, thus making such designs conservative.

4.4 Debris

In Section 3, several sources of debris were identified and conservative estimates of the quantities of each type were given. They are summarized in Table 4-1. Assumptions used in these estimates are also listed in the table. In the cases for abrasive debris, it has been assumed that all particles are entrained in the sump flow and that none settle out. For the fibrous and soft debris, it has been assumed that all the fibers (most of which are encapsulated in mats) which reach the screens are somehow shredded to a size capable of passing through the fine mesh screens surrounding the sumps and that all fine soft particulates are transported through the pumps.

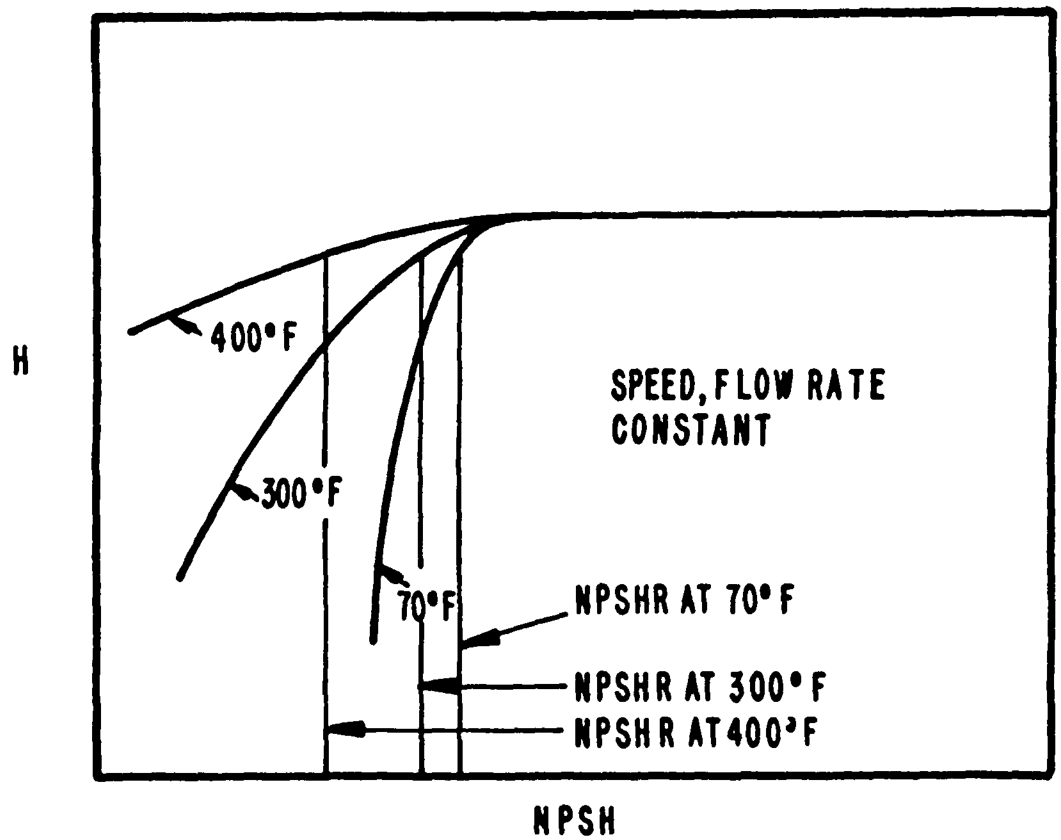

Figure 4-4. RELATIONSHIP BETWEEN PUMP CAVITATION AND WATER TEMPERATURE

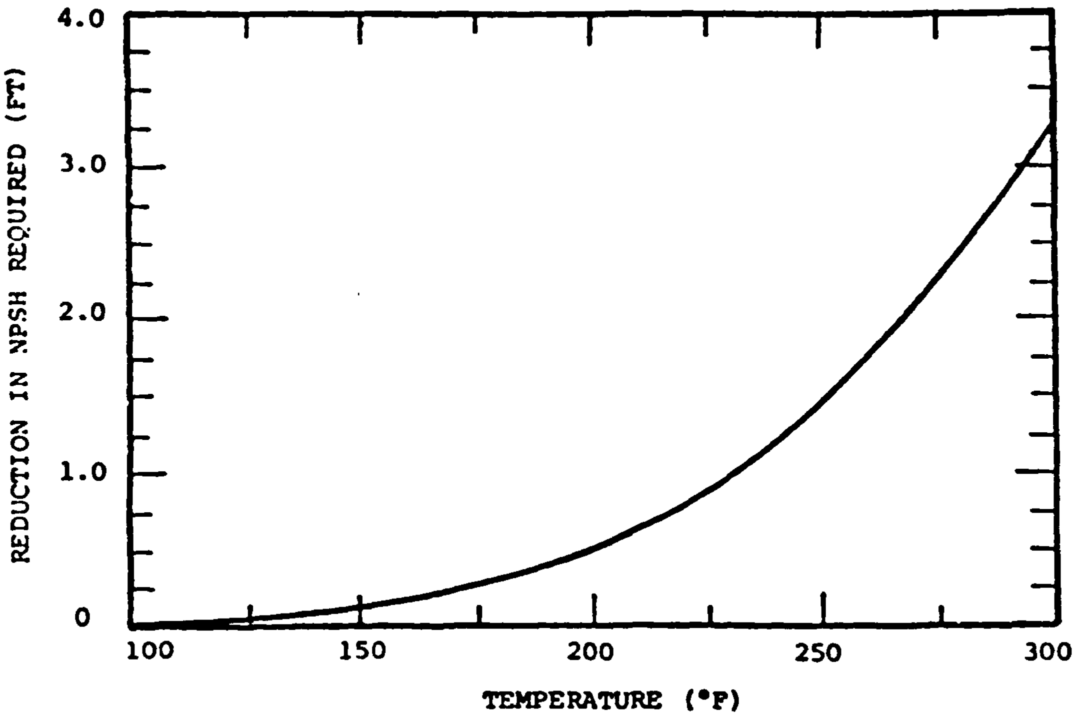

FIGURE 4-5. REDUCTION IN PUMP NPSH REQUIREMENTS AS A FUNCTION OF LIQUID TEMPERATURE (REFERENCE [25]

TABLE 4-1

SUMMARY OF DEBRIS TRANSPORTED TO PUMPS

Type	Size	Hardness	Concentration % mass (% volume)	Assumption
Fibrous	<5000μ	Soft	0.3 (0.3)	Fibrous debris transported to screens passes through (Maine Yankee)
$Zn(OH)_2$	<1μ	Soft	0.12 (0.04)	One year precipitation products in recirculation (Seabrook)
AlO(OH)	1-10μ	6.5-7.0 Moh	0.11 (0.04)	One year precipitation products in recirculation (Seabrook)
Paint Flakes	<5000μ	Soft	0.025 (0.025)	5000 ft^2 dislodged as small chips
Concrete	~100μ		0.05 (0.02)	Assumed 17 cu. ft. mixed with 250,000 gal water

Total volumetric concentration of "soft" products 0.365%
Total mass concentration of abrasives 0.16%

The table shows that two basic types of solids are of concern: fine abrasives ranging from 1 - 100μ in size, and soft particulates and fibrous debris.

A compilation of the data reported in the literature on the effects of solid mixtures on pump performance is given in Figure 4-6. The results demonstrate that for the quantities of solids estimated (up to 1%, total) there is virtually no effect of solids mixtures on performance.

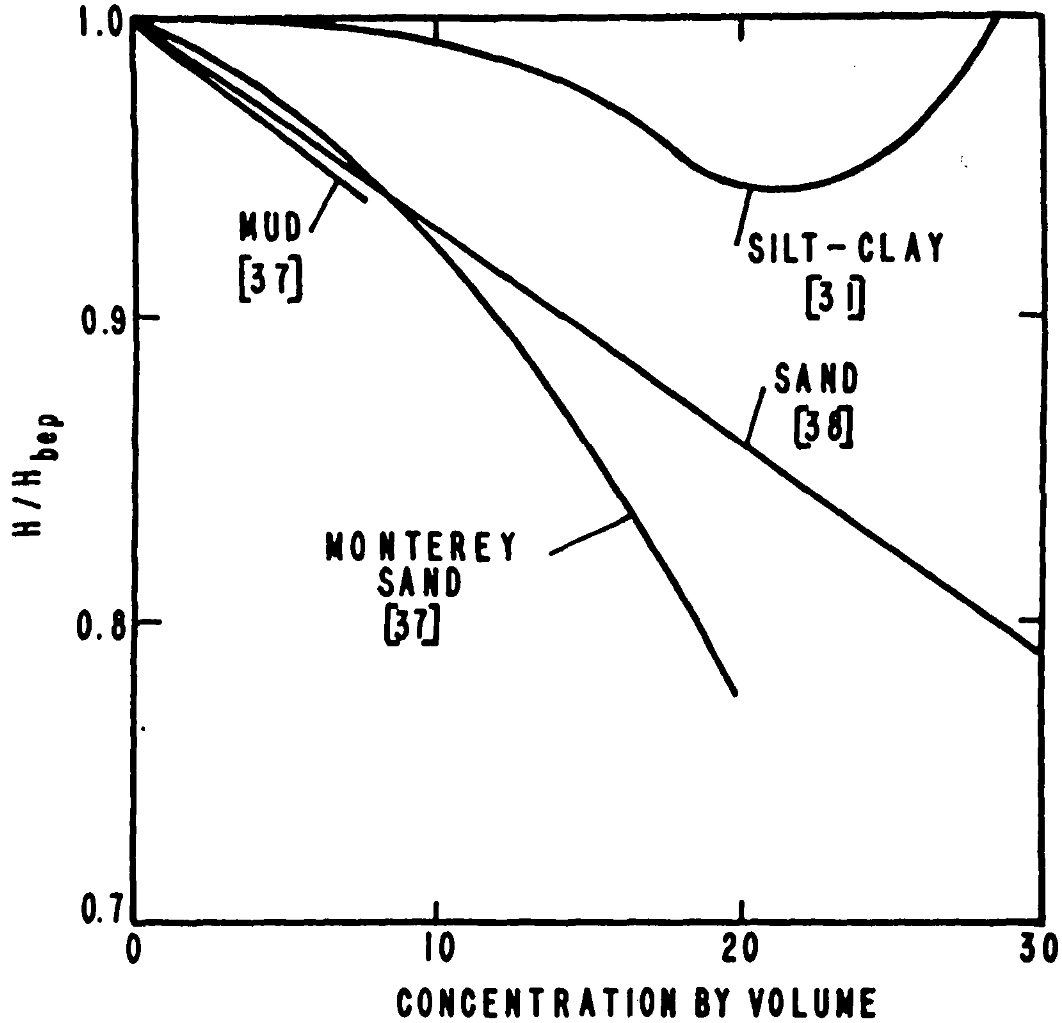

Figure 4-6 COMPILATION OF TEST RESULTS ON THE EFFECTS OF SOLIDS ON CENTRIFUGAL PUMP PERFORMANCE

The effects of debris on the long-term mechanical reliability of the pumps is difficult to quantify. The effects can be divided into several topic areas:

- abrasive wear of impellers, casings, and wearing rings which will slowly degrade performance over a period of time

- the effects of debris in the seals, filtration system and bearings,

- the effects of soft and fibrous debris on clogging the pump passages.

The test results on a shrouded impeller reported by Doolin showed that with concentrations of 10% by mass of abrasive diatomaceous earth in water, pump performance degraded gradually over a 400 hour test. The actual degradation in efficiency was 15% in 458 hours. The fact that the concentrations of abrasives employed in these tests are about one hundred times the worst case estimate for abrasives in the recirculating fluid and that the degradation produced only a 15% decrease in performance indicate that the long term wear of RHR pumps, impellers, or casings should not be serious.

Although some proprietary tests have been conducted on the performance of seals, these data are not easily accessible. Interviews with seal manufacturers provided information about concerns with the presence of debris and possible effects of debris on seal behavior. Current practice by Crane Co., based on their experience, is to recommend against the use of filters and cyclone separators in the seal flush lines. Several damage mechanisms were identified:

- increased leakage due to runner/seat wear,

- increased leakage due to spring cocking or "hang up"

- increased wear/leakage due to high temperatures resulting from blockage of external flush ports.

It is virtually impossible to assess the likelihood of each of the above occurrences although manufacturers opinions indicate that the likelihood of seal failure is low. It is important to note that a "failure" criterion for a seal is typically a "large" leakage rate. The "backup" or "disaster" bushing used in seal assemblies for these pumps severely limits the leakage from the pump in the event of seal failure. Typical test values give leakages of about 70 gph per 100 psi differential through the backup bushing. While this leakage rate is low in comparison to the recirculating flow rate (<0.1%) it does represent a leakage loss to the total inventory.

The bearings identified in the survey were either fully enclosed permanently lubricated bearings in the motor housing or oil lubricated ball bearings mounted in the pump frame. For configurations where a shaft slinger and lip seal are located between the backup bushing and the bearings, likelihood of bearing failure is low. The slinger serves both to deflect direct leakage jets and centrifuge leakage away from the shaft. The lip seal ahead of the bearing should prevent low pressure liquid on the shaft surface from entering the bearings.

The concentration of soft fines given in Table 4-2 should have no effect on overall pump behavior other than possible collection in the seal cavity. Fines accumulation in the seal cavity will most likely affect the spring preload in the seal and cause an increase in seal leakage.

Assessment of the effects of fibrous debris (in which we lump fibers from insulation and paint chips) relies on proprietary test results summarized by J. Doolin [40]. These tests demonstrated that pumps of design similar to those used in RHR and CS service successfully handled 4% concentrations of fibrous paper stock. Note that 4% concentration of fibrous paper stock is approximately ten times the conservatively estimated concentrations for soft debris given in Table 4-2. Thus, the effect of soft particulates and fibrous debris on pump performance is expected to be negligible.

4.5 Method for Calculating Pump Inlet Conditions

This section outlines a procedure for calculating the inlet conditions at the pump. Two important parameters, the NPSH available and the volumetric air ingestion rate, are determined for comparison with the NPSH requirements for the pump and with the criterion of 2% maximum allowable air ingestion. The input conditions to the procedure include details of the system geometry (suction piping elements, sizes, elevations, etc.), the air ingestion rate at the sump suction pipe, flow rate, sump water elevation, pressure losses through the screens, water temperature and containment pressure. The procedure for calculating NPSH follows routine methods, except that steps are also incorporated to allow for air ingestion effects. Figure 4-7 shows a schematic of the pump suction system with appropriate nomenclature.

Input Conditions

Flow Rate	Q (cfs)
Water Temperature	Tw (°F)
Specific Weight of Water at Tw (See Figure 4-9)	γ (lb/ft^3)
Vapor pressure of water at Tw (See Figure 4-8)	P_{vp} (psia)
Containment absolute pressure	P_c (psia)
NPSH required at Q (from pump characteristics)	NPSHR (ft)
Head loss through screens at Q	H_s (ft)
Air ingestion rate at sump suction pipe	AF (% of Q)
Elevation of sump suction pipe	Z_s (ft)
Elevation of pump at impeller centerline	Z_p (ft)
Elevation of liquid surface outside of sump screens	Z_w (ft)
Geometric details of suction piping elements including elbows, pipes, valves, reducers	e.g. D_i, L_i (ft)

Loss coefficient for each suction pipe element, K_i, where:

$$K_i = \frac{\Delta H_i}{V_i^2/2g}$$

can be found in standard handbooks.

Most of the above information is required for conventional evaluation of NPSH requirements for pumps. The air ingestion rate, however, is determined on the basis of a sump evaluation as outlined in [1].

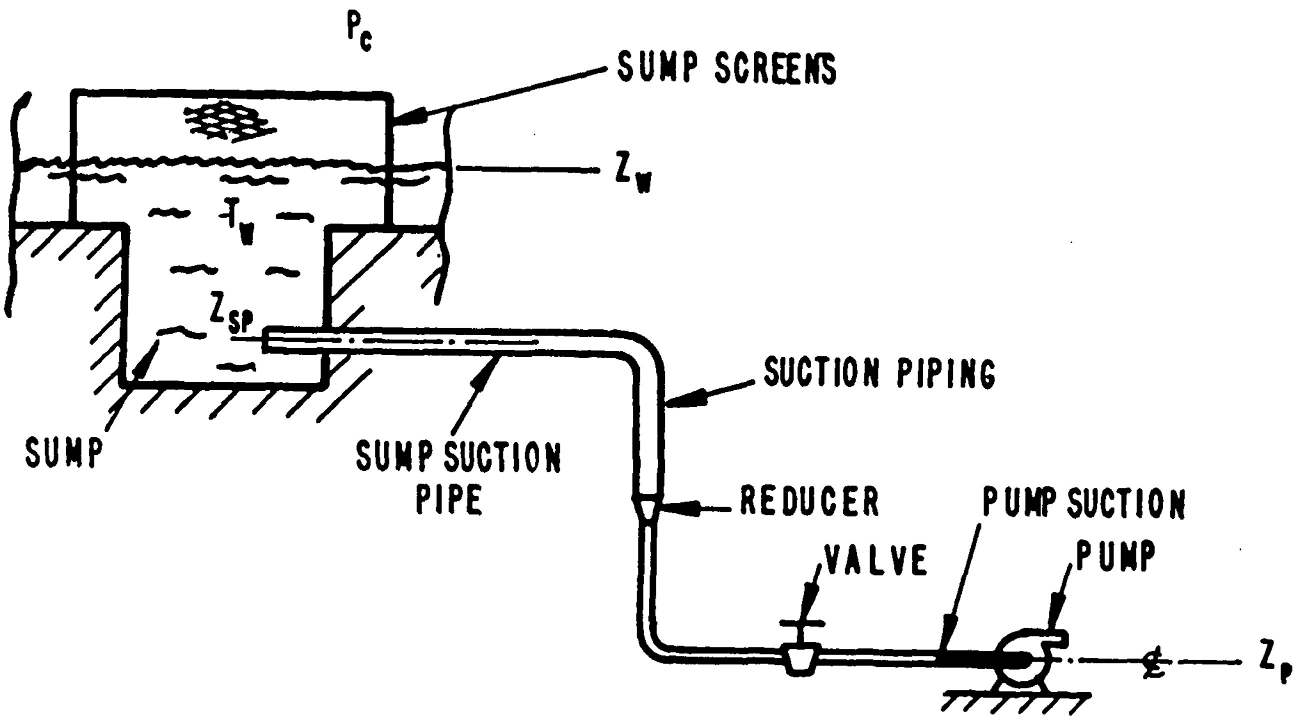

Figure 4-7. SCHEMATIC OF SUCTION SYSTEM FOR CENTRIFUGAL PUMP

The following steps outline the procedure for calculating NPSH at the pump and for calculating pump air ingestion corrected for pressure differences between the sump and pump inlet.

1. Calculate the absolute total pressure in the sump at the elevation of the centerline of the sump suction pipe.

   $$P_{sa} = P_c + \gamma\,(Z_w - Z_s)/144 - \gamma H_s/144 \qquad \text{(psia)}$$

2. Calculate the absolute static pressure just inside the inlet of the sump suction pipe. (This value is required only if air is ingested so that air density changes can be incorporated).

   $$P_{sg} = P_{sa} - (\gamma/144)(1 + K_s)\,u^2/2g \qquad \text{(psia)}$$

   where u is the average velocity at the sump suction pipe inlet, K_s is the loss coefficient for the pipe inlet.

3. Determine the air ingestion rate AF_s at the sump in percent from the sump evaluation methods described in [1].

4. Calculate pressure losses in suction piping components (pipes, elbows, etc.) due to friction.

   $$P_{loss} = \gamma/144\ \Sigma\ K_i V_i^2/2g \qquad \text{(psi)}$$

   where K_i, V_i are the loss coefficients and average velocities in individual piping components.

5. Calculate the absolute static pressure at the pump suction flange. This value is used to correct the volumetric air flow rate for changes in density between the sump suction pipe and the pump. If no air is ingested, Steps 5, 6, and 7 can be ignored.

   $$P_{pa} = P_{sg} - P_{loss} + (\gamma/144)(Z_s - Z_p) + (u^2 - V_p^2)/2g \qquad \text{(psia)}$$

   where u and V_p are the average velocities at the sump pipe and pump inlet flange, respectively.

6. Calculate the air ingestion rate at the pump correcting the volumetric flow for density changes. The correction is based on isothermal, perfect gas relations:

   $$AF_p = (P_{sa}/P_{pa})\ AF_s$$

7. If AF_p is greater than 2%, inlet conditions are not acceptable.

8. Calculate the NPSH at the pump suction flange.

   $$NPSH = (144/u)(P_c + P_{pa} + V_p^2/2g - P_{vp})$$

   where P_{vp} is the vapor pressure of water at T_w (Figure 4-8).

9. If air ingestion AF is not zero, then the NPSHR value at Q from the pump characteristic curves must be modified to account for the effect of air ingestion on pump cavitation.

   $$NPSHR_{air/water} = (1 + 0.5AF)NPSHR_{water}$$

   AF is the air volume fraction in percent at the pump inlet, calculated in step 6.

10. If the calculated available NPSH from step 8 is greater that the NPSH requirements for the pump in step 9, pump inlet conditions are satisfactory.

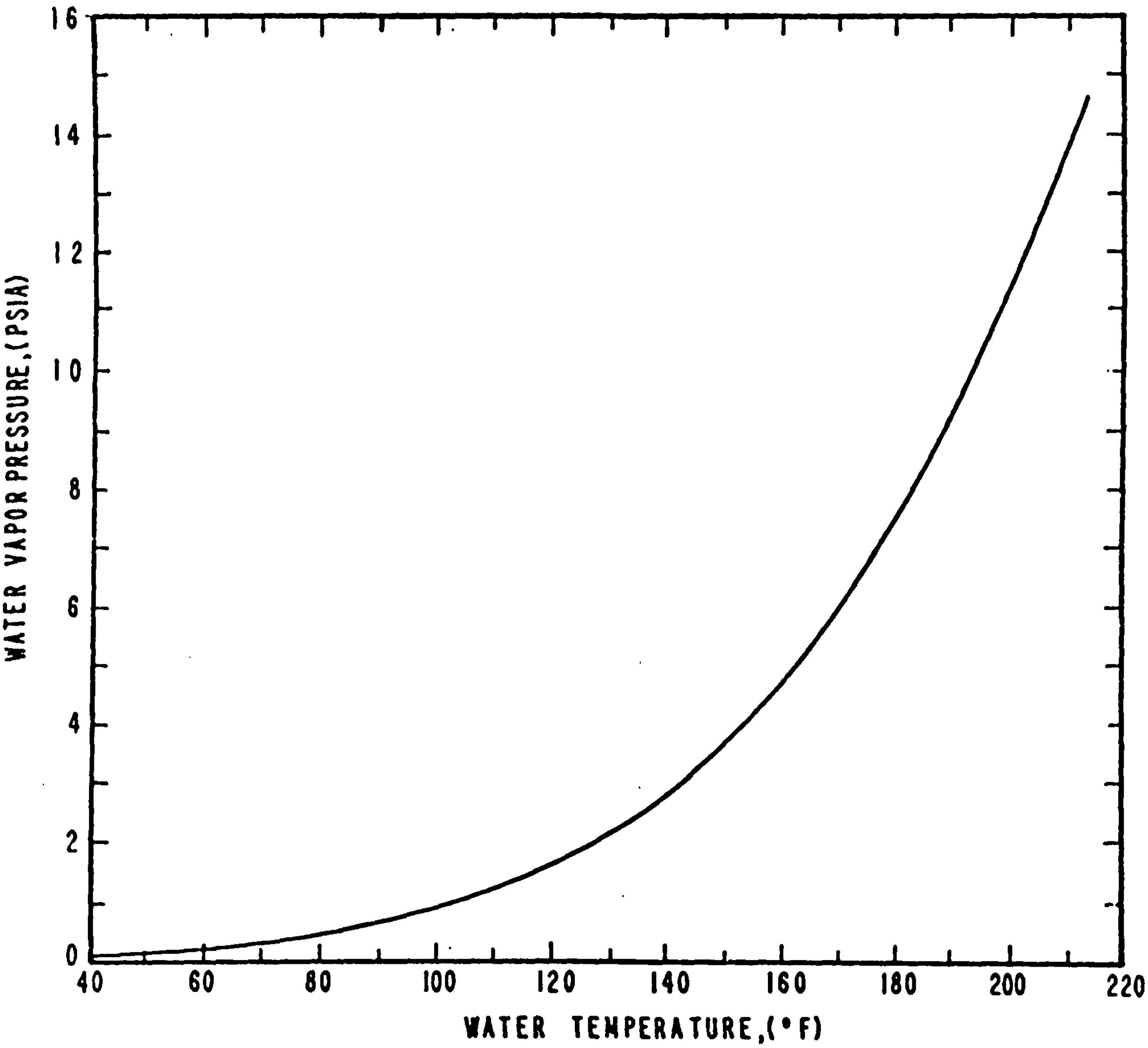

Figure 4-8. VAPOR PRESSURE OF WATER AS A FUNCTION OF TEMPERATURE

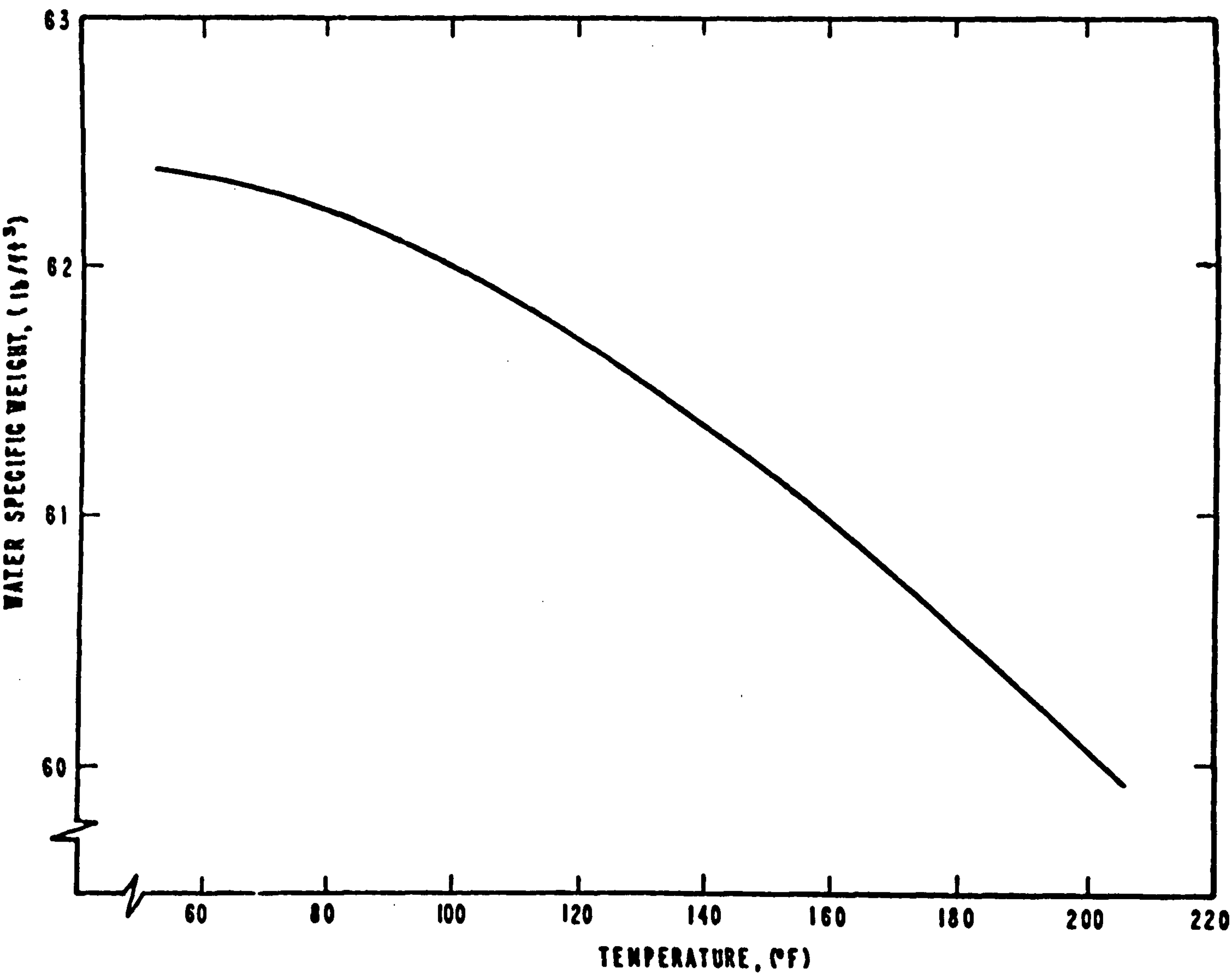

Figure 4-9. SPECIFIC WEIGHT OF WATER AS A FUNCTION OF TEMPERATURE

5 CONCLUSIONS

5.1 Air Ingestion

- There is a substantial amount of experimental air/water pump data in the open literature that can be applied to RHR and CS pump operation. These data derive principally from three independent studies [30], [31] and [32].

- The data show that over a wide range of operating flow rates RHR and CS pumps should be able to handle air ingestion rates up to 2% by volume with negligible degradation in performance.

- For very low flow rates (less than about 50% of best efficiency point) even small air ingestion quantities may accumulate in the impeller inlet and result in "air binding" or loss of prime. However, for low flow rates, sump evaluations show that the likelihood of air being ingested is low.

- Small quantities of ingested air will increase the NPSH requirements for a pump. A correction factor for NPSH requirements to account for ingested air has been proposed.

- Swirl at the pumps resulting from sump surface vortices will be negligible if the suction piping length is greater than 14 pipe diameters. Suction piping configurations (elbows, valves, etc.) are more likely to establish flow at the pump inlet and should be considered using conventional methods.

5.2 Debris Ingestion

- There is sufficient experimental data on the effects of particulates on pump hydraulic performance that can be applied to RHR and CS pumps.

- The data show that pump hydraulic performance degradation is negligible for particulate concentrations less than 1% by volume for a wide range of substances.

- Although data are limited, tests on mechanical wear of pumps indicate that the maximum calculated quantity of debris in the recirculating fluid is too small to impair pump operation as a result of material erosion.

- Among all issues considered with respect to RHR and CS pump performance under debris ingesting conditions, the effect of debris on mechanical face seals systems was the most difficult to quantify.

- Filters in shaft seal systems are generally not recommended because of the likelihood that they will become clogged with debris and inhibit cooling fluid to the seals.

- At least one set of test data on seals show that operation for extended periods without cooling results in only a marginal increase in leakage.

- The presence of "backup" or "disaster" bushings in the shaft seal will minimize shaft leakage in the event of total seal failure. Although this leakage rate is small (<0.1% of flow through the pump) the cumulative effect could deplete recirculating inventory.

REFERENCES

[1] Serkiz, A.W.; RESOLUTION OF UNRESOLVED SAFETY ISSUE A-43; NUREG-0897, (to be published).

[2] Weigand G.G. et al.; A PARAMETRIC STUDY OF CONTAINMENT EMERGENCY SUMP PERFORMANCE; SAND82-06241/NUREG/CR-2758, Sandia National Laboratories, Albuquerque, NM.

[3] Reyer, R., Graham, E. and Riddington, J.W.; SURVEY OF INSULATION USED IN NUCLEAR POWER PLANTS AND THE POTENTIAL FOR DEBRIS GENERATION; NUREG/CR-2403, Burns and Roe Inc., Oradell, NJ, October 1981.

[4] Kolbe, R., and Graham, E.; SURVEY OF INSULATION USED IN NUCLEAR POWER PLANTS AND POTENTIAL FOR DEBRIS GENERATION; NUREG/CR-2403, Supplement No. 1, Burns and Roe Inc., Oradell, NJ.

[5] Regulatory Guide 1.1, USNRC; NET POSITIVE SUCTION HEAD FOR EMERGENCY COOLING AND CONTAINMENT HEAT REMOVAL SYSTEM PUMPS.

[6] Regulatory Guide 1.79, USNRC; PREOPERATIONAL TESTING OF EMERGENCY CORE COOLING SYSTEMS FOR PRESSURIZED WATER REACTORS.

[7] 10 CFR Part 50, Appendix A, General Design Criterion 38; CONTAINMENT HEAT REMOVAL.

[8] Nagengast, P.J.; A REACTOR SAFETY SYSTEM PUMP ITS DESIGN EVOLUTION AND QUALIFICATION; ANS/European Society Topical Meeting, Knoxville, TN, April 1980.

[9] Olson, D.J.; EXPERIMENTAL DATA REPORT FOR SINGLE AND TWO-PHASE STEADY STATE TESTS OF THE 1½ LOOP MOD-1 SEMISCALE SYSTEM PUMP; ANCR-1150, Aerojet Nuclear Company, May 1974.

[10] Olson, D.J.; SINGLE AND TWO-PHASE PERFORMANCE CHARACTERISTICS OF THE MOD-1 SEMISCALE PUMP UNDER STEADY STATE AND TRANSIENT FLUID CONDITIONS; ANCR-1165, Aerojet Nuclear Company, October 1974.

[11] Kamath, P.S. and Swift, W.L.; TWO-PHASE PERFORMANCE OF SCALE MODELS OF A PRIMARY COOLANT PUMP; EPRI NP-2578, Electric Power Research Institute, April, 1982.

[12] Swift, W.L.; MODEL PUMP PERFORMANCE PROGRAM-DATA REPORT: EPRI NP-2379, Electric Power Research Institute, May 1982.

[13] Love, W.J.; CENTRIFUGAL PUMP PERFORMANCE WITH GAS-LIQUID FLOW; S-70-1014, January 30, 1970, Class 2, Research and Development Center, General Electric Company, Schenectady, NY.

[14] Sozzi, G.L. and Burnette, G.W.; TWO-PHASE PUMP PERFORMANCE DURING STEADY STATE OPERATION AND DURING A SIMULATED LOCA BLOWDOWN; NEDE-13239, Class 1, General Electric Company, San Jose, CA, November 1971.

[15] Runstadler, P.W.; REVIEW AND ANALYSIS OF STATE-OF-THE-ART OF MULTIPHASE PUMP TECHNOLOGY; TN-226, Creare Inc., Hanover, NH, November 1975.

[16] Heidrick, T.R., Hancox, W.T. and Nguyen, D.; CENTRIFUGAL PUMP BEHAVIOR IN STEADY AND TRANSIENT STEAM-WATER FLOWS; Polyphase Flow in Turbomachinery, Edited by Grennan, C., Cooper, P. and Runstadler,

[17] Heidrick, T.R., Rajan, V.S.V., and Nguyen, D.M.; THE BEHAVIOR OF CENTRIFUGAL PUMPS IN STEADY AND TRANSIENT STEAM-WATER FLOWS; Proceedings of the ANS/ASME/NRC International Topical Meeting on Nuclear Reactor Thermal Hydraulics held at Saratoga Spring, NY, October 5-8, 1980, pp. 585-607.

[18] Murakami, M., Suehiro, H., Isaji, T. and Kajita, J.; FLOW OF ENTRAINED AIR IN CENTRIFUGAL PUMPS; Paper B8, 13th Congress IAHR, 1969.

[19] Murakami, M. and Minemura, K.; EFFECTS OF ENTRAINED AIR ON THE PERFORMANCE OF A HORIZONTAL AXIAL-FLOW PUMP; in Polyphase Flow in Turbomachinery, edited by Grennan, C., Cooper, P. and Runstadler, P.W., ASME Publication Book No. H00123, 1978, pp. 171-184.

[20] Stepanoff, A.J.; FLOW OF GAS-LIQUID MIXTURES; Pumps and Blowers, Two-Phase Flow, First Edition, Wiley, NY 1964, pp. 246-274.

[21] Kosmowski, I.; BEHAVIOR OF CENTRIFUGAL PUMPS WHEN CONVEYING LIQUIDS WITH GAS INCLUSIONS: Proceedings of the 7th Conference of the British Pump Manufacturers Association, Pumps - The Developing Needs, 1981, pp. 285-291.

[22] Wilson, D.G.; Chan, T.C.; Goldfinsh, A.L. and Zegley, R.D.; ANALYTICAL MODELS AND EXPERIMENTAL STUDIES OF CENTRIFUGAL PUMP PERFORMANCE IN TWO-PHASE FLOW; EPRI NP-170, January 1977.

[23] Chivers, T.C.; AN AIR CONTENT EFFECT ON ADVANCED CAVITATION IN A CENTRIFUGAL PUMP; ASME 1969 Cavitation Forum, pp. 31-32.

[24] Chivers, T.C.; TEMPERATURE EFFECTS ON CAVITATION IN A CENTRIFUGAL PUMP: THEORY AND EXPERIMENT; Proceedings of the Institution of Mechanical Engineers, V 184, Part 1, No. 2, 1969-70, pp. 37-47.

[25] Hydraulic Institute Standards for Centrifugal, Rotary and Reciprocating Pumps, 13th Edition, Hydraulic Institute, Cleveland, OH, 1975.

[26] Rothe, P.H., Runstadler, P.W. and Dolan, F.X.; PUMP SURGE DUE TO TWO-PHASE FLOW; in Polyphase Flow in Turbomachinery, Edited by Grennan, C., Cooper, P., and Runstadler, P.W., ASME Publication, Book No. H00123, 1978, pp. 121-137.

[27] Runstadler, P.W. and Patel, B.R.; INVESTIGATIONS INTO THE TWO-PHASE FLOW BEHAVIOR OF CENTRIFUGAL PUMPS; in Polyphase Flow in Turbomachinery, Edited by Grennan, C., Cooper, P., and Runstadler, P.W.; ASME Publication, Book No. H00123, 1978, pp. 79-100.

[28] Hunter, J.A. and Harris, P.A.; PERFORMANCE OF SMALL NUCLEAR REACTOR PRIMARY COOLANT PUMPS UNDER BLOWDOWN CONDITIONS; In Polyphase Flow in Turbomachinery, Edited by Grennan, C., Cooper, P. and Runstadler, P.W., ASME Publication, Book No. H00123, 1978, pp. 157-170.

[29] Grennan, C.W.; POLYPHASE FLOW IN GAS TURBINE FUEL PUMPS; In Polyphase Flow in Turbomachinery, Edited by Grennan, C., Cooper, P. and Runstadler, P.W.; ASME Publication, Book No. H00123, 1978, pp. 213-221.

[30] Murakami, M. and Minemura, K.; FLOW OF AIR BUBBLES IN CENTRIFUGAL IMPELLERS AND ITS EFFECT ON PUMP PERFORMANCE; 6th Australian Hydraulics and Fluid Mechanics Conference, Adelaide, Australia, December 5-9, 1977.

[31] Merry, H.; EFFECTS OF TWO-PHASE LIQUID/GAS FLOW ON THE PERFORMANCE OF CENTRIFUGAL PUMPS; National Engineering Laboratory, East Kilbride, Glasgow, C130/76, IME Conference on Pumps and Compressors for Offshore Oil and Gas, June 29-July 1, 1976, pp. 61-67.

[32] Florjancic, D.; INFLUENCE OF GAS AND AIR ADMISSION ON THE BEHAVIOR OF SINGLE AND MULTISTAGE PUMPS; Sulzer Research Number 1970, 1970, p. 35-44.

[33] Arie, M. and Fukusako, S.; PERFORMANCE TESTS OF A CENTRIFUGAL PUMP; Proceedings of the US Japan Seminar on Similitude in Fluid Mechanics, September 1967, pp. 13-32.

[34] Loviisa Emergency Cooling System Model Tests (Translated from the Finnish-Preliminary Review Copy) Report Number 93K168FN, Sandia National Laboratories, Report No. RS314018191, printed December, 1981.

[35] Niyogi, K. K. and Lunt, R.; CORROSION IN ALUMINUM AND ZINC IN CONTAINMENT FOLLOWING A LOCA AND POTENTIAL FOR PRECIPITATION OF CORROSION PRODUCTS IN THE SUMP; United Engineers and Constructors, September 1981.

[36] Michelson, C.; CONCERN RELATING TO THE INTEGRITY OF A POLYMER COATING FOR SURFACES INSIDE CONTAINMENT; I.E. Draft Bulletin No. 80-21, USNRC, August 1980.

[37] Fairbank, L.C.; SOLIDS IN SUSPENSION - EFFECT ON THE CHARACTERISTICS OF CENTRIFUGAL PUMPS; ASCE Proceedings, V67, #1, October 1941.

[38] Vocadlo, J.J., Koo, J.K., and Prang, A.J.; PERFORMANCE OF CENTRIFUGAL PUMPS IN SLURRY SERVICE; Hydrotransport 3, May 1974.

[39] Herbich, J.B.; EFFECT OF IMPELLER DESIGN CHANGES ON CHARACTERISTICS OF A MODEL DREDGE PUMP; ASME Paper No. 63-AHGT-33, March 1963.

[40] Doolin, J.; PUMPING ABRASIVE FLUIDS; Reprint from Plant Engineering, November 1972.

[41] Letter dated March 2, 1982 to W. L. Swift, Creare Inc., Hanover, NH from John H. Doolin, McGraw-Edison Company, Harrison, NJ.

[42] MECHANICAL SEALS FOR NUCLEAR POWER PLANT AUXILLARY AND EMERGENCY SERVICES; Durametallic Corp., Kalamazoo, MI, 1969.

[43] SEAL PERFORMANCE TESTING FOR NUCLEAR POWER PLANT SAFETY INJECTION SYSTEMS; JW ABAR Crane Packing Co., Bulletin No. 3472.

[44] Fraser, W.H.; RECIRCULATION IN CENTRIFUGAL PUMPS; Paper presented at the Winter Annual ASME Meeting, Washington, DC, November 15-20, 1981.

[45] Stahl, H.A. and Stepanoff, A.J.; THERMODYNAMIC ASPECTS OF CAVITATION IN CENTRIFUGAL PUMPS; Transactions of the ASME, pp. 1691-1693, November 1956.

[46] Billet, M.L., Holl, J.W. and Weir, D.S.; CORRELATIONS OF THERMODYNAMIC EFFECTS FOR DEVELOPED CAVITATION; Presented at the Polyphase Flow in Turbomachinery Symposium at the Winter Annual Meeting of the ASME, December 10-15, 1978.

[47] Moore, R.D.; PREDICTION OF PUMP CAVITATION PERFORMANCE; Presented at a Symposium on Fluid Mechanics, Acoustics, and Design of Turbomachinery, Pennsylvania State University, sponsored by NASA, NASA SP-304, Part II, 1974.

*U.S. GOVERNMENT PRINTING OFFICE: 1982 361-297/2408 1-3