DOE/SF/98001--T5

LARGE SCALE BREEDER REACTOR PLANT PROTOTYPE MECHANICAL PUMP CONCEPTUAL DESIGN STUDY

**Prepared for GENERAL ELECTRIC Fast Breeder
Reactor Department**

Byron Jackson Pump Division
BORG-WARNER CORPORATION

MASTER

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SPECIFICATION

TITLE

LARGE SCALE BREEDER REACTOR PLANT HOT LEG
MECHANICAL PUMP CONCEPTUAL DESIGN STUDY

1. OBJECTIVE

This specification establishes the requirements for a design study to provide a recommended concept of a hot leg mechanical sodium pump for use in the primary loop of a Large Scale Breeder Reactor Plant. The concept recommendation shall be supported by; technical justification, applicable experience, order of magnitude cost estimates and identification of required development including its order of magnitude cost and schedule.

2. APPLICABLE DOCUMENTS

The following documents form a part of this specification to the extent specified herein. Unless otherwise indicated, the issue in effect on date of or request for proposal shall apply. In the event of conflict between documents referenced herein and the contents of this specification, the contents of this specification shall be considered a superseding requirement.

2.1 CRBRP Pump E-Specification, 22A3444, Revision 8, In Part

For this study all the information contained in the specification shall be used as a reference with the exception of:

3.2.2 Plant Description

3.2.3 Plant Location

3.2.4 Size and Weight

3.3.1.1 Design Point Conditions

3.3.3.1 Design Temperatures and Pressures

3.3.3.2 Mechanical Loads

3.6.3 CRBRP Operating Environment

Appendix B, Para. B3.2 Intermediate Pump Event Description

Appendix E, Intermediate Pump

3. BACKGROUND

Recent planning associated with a Large Scale Breeder Reactor Plant established the near-term need for an LMFBR mechanical pump design concept to meet the requirements of the primary heat transport systems of this plant. Typically the LSBR plant layout can be represented diagrammatically as shown in Figure 1. The heat transport systems consist of a primary system of three parallel loops and an intermediate system of three independent loops. The primary pumps are located in the hot leg side of the loops.

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Preliminary planning for the Large Scale Breeder Reactor has indicated the need of primary hot leg pumps having low net positive suction head requirements.

4. GENERAL APPROACH

4.1 Design Concept

The Large Scale Breeder Reactor Plant will use commercial experience and Clinch River Project and FFTF technology to the maximum extent possible. Consequently, the mechanical pump design will consider design, analysis, development and test experience derived from the Clinch River and FFTF pump programs recognizing that some departures from this technology may be required in certain areas to meet the requirements of this specification. The pump vendor in this Task I effort shall develop a pump design concept which best meets the requirements outlined in Tables 1 and 2, and shall identify and describe any required supporting developments. It is not the intention of this Task I effort to perform a complete detail design, but rather to do that work which is required to insure that the basic pump configurations can meet the system requirements of the Large Scale Breeder Reactor Plant. Additionally, the vendor shall plan for a follow-on task of consultations on pump/systems interface resolution for his recommended concept.

4.2 Consultation on Pump/Systems Problems (Task II)

During and following pump concept submittal the vendor shall provide consultations and technical assistance as needed with LSBRP systems on problems related to pump/systems interfacing. These consultations may be in the form of telecons, correspondence, occasional meetings, and iterations on the conceptual design as required to perform detail trade-off studies in specific reactor concepts.

5. SCOPE OF WORK FOR TASK I

5.1 Requirements

The vendor will include specific consideration of the following aspects; however, it is not to be limited to them.

- a) Pump operation during runout to the two loop minimum impedance operating curve (Figure 2) is not required. If one pump trips off the line, the event will be sensed, the plant will be tripped, and the other two pumps will coast down to pony motor speed. The elapsed time between the first pump trip and the plant trip is TBD seconds, during which time the other two pumps will runout on a transient basis as the first pump coasts down. There is no performance requirement which the pump must meet during this transient, other than the fact that it must withstand the effect of this transient (e.g., incipient or full cavitation if it occurs).

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- b) To the extent required for this study the thermal hydraulic transient descriptions provided in Appendix B, Para. B-3.1 of CRBRP Pump E-Spec 22A3444, Revision 8 are considered applicable for range of temperature change and rate of change; however, the temperature level shall be adjusted to agree with the 935°F nominal operating temperature (70°F shall remain as defined). The descriptions of the seismic events also need to be developed, but to the extent required for this study assume that the seismic activity will be as predicted for the CRBRP pumps as set forth in CRBRP Pump E-Spec 22A3444, Revision 8, Appendix B, Para. B-2.0. More realistic event descriptions for this plant will be developed in the near future, but it is intended that the conceptual design be based on the event descriptions provided by the CRBRP Pump E-Spec 22A3444, Revision 8, Appendix B as indicated. Based on previous experience a preliminary assessment of whether any of the events, as specified, will cause severe design problems will be provided by the vendor.
- c) The NPSH requirement for this pump (Table 2, Item 4, and Figure 3) may require innovative designs in order to minimize weight and cost. The manufacturer may need to explore departures from current design practice in meeting these requirements.
- d) To the extent required by this study the nozzle loads specified in Table 3 are the best information presently available, and are probably conservative. A preliminary assessment of whether the specified nozzle loads will cause severe design problems will be provided in the initial phase of the concept design to be followed by more detailed evaluation in Section 5.3.1.
- e) Table 2 of this specification contains assumed system requirements; an evaluation of these characteristics is to be made as they affect the conceptual pump design of this effort and the pump designer shall recommend to the purchaser any required changes to these assumed values.

5.2 Design Concept Drawings

Prepare design concept drawings for the pump which meet all requirements of this specification. Required development activity or any component verification tests must be noted. The design concept drawings must include:

- a) Pump outline drawing, cross-section view of pump internals, size and location of all nozzles.
- b) Special materials.
- c) Dimensions establishing size, shape, permissible sodium level range, internal sodium passage configuration, fits and clearances critical to the feasibility of the concept.
- d) Approximation of the pump assembly weight, both empty and with full inventory of sodium.

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e) Pump support details.

It is not intended that this information be in final form for inclusion in a design layout drawing, but it should be presented in the depth to which it is available.

5.3 Final Letter Report

The vendor shall prepare a final letter report which will provide adequate documentation in support of the selected pump concept. The following topics must be considered:

5.3.1 Hydraulic Design

Provide conceptual pump performance curves to show pump developed head, torque, efficiency, drive horsepower required, and required NPSH, all versus flow rate for the range of speeds as required to provide the operating characteristics defined in this specification. The estimated performance curves shall be supported by actual performance of similar pumps, model pump data, or analytical studies.

5.3.2 Preliminary Stress

Prepare a rough order of magnitude stress assessment to determine that the recommended concept has no insurmountable stress problems.

5.3.3 Alignment and Critical Dimension

Prepare a brief preliminary evaluation of critical dimensions and the alignment which must be maintained in the pump to assure that the concept does not require any unreasonable dimensions or tolerances. Explain the basis for the evaluation. The study must take cognizance of the pump tank displacements which occur as a result of loop piping reaction forces on the pump tank nozzles.

5.3.4 Dynamics

Identify the natural frequencies of the pump/drive motor rotor and the pump/motor structure. The natural frequencies shall be greater than the maximum pump operating speed and not be in the range of the seismic response spectrum peaks, Pump E-Spec 22A3444, Revision 8, Appendix B, paragraph B-2.0. Any computer codes used for analyses, or to supplement other calculation methods, shall be those for which the vendor has significant experience and shall be adequately identified.

5.3.5 Manufacturability

Prepare a preliminary assessment of the manufacturability including as a minimum the effects of the pump length (considering bowing, misalignment, etc.), unique impeller casting problems resulting from performance requirements.

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5.3.6 Tank Sodium Level Control

The vendor shall provide a technical assessment of the method of liquid level control in the pump tank to include comments and recommendations relating to the effect on design, required development, and reliability of the pump concept.

5.4 Planning for Design, Manufacture and Test

5.4.1 Program Plans and Schedules

Based on the conceptual design developed in compliance with paragraph 5.2 of this specification, prepare preliminary plans and schedules to design, develop, manufacture and test one prototype pump.

These plans and schedules should take into consideration the planning requirements of RDT F2-2 since any subsequent program will be accomplished in accordance with the requirements of that specification. These plans and schedules must show key events, start dates, completion dates; elapsed time of the activity to other activities and interrelationship. Any contingencies must be so identified.

5.4.2 Interfacing and Auxiliary Requirements

Provide a list of major pump-to-plant interfaces with identification of basic conditions which must be satisfied at each interface. Identify all required auxiliaries.

5.5 Development Needs

The vendor shall identify any development program needs in support of the conceptual design including; schedule, required facilities, and estimated cost to complete the work.

5.6 Cost Estimates

Based on the preceding technical studies, prepare order of magnitude estimate of costs necessary to accomplish the prototype pump development. This estimate should be subdivided by major tasks. An expenditure schedule of the costs by task by fiscal year must be shown. The cost estimate should be on the basis that preliminary and detail design will be based on the conceptual design. The design task includes the Quality Assurance and documentation activities which are required to produce and substantiate all the design data packages which constitute the complete pump design. Assuming a successful prototype pump development program, estimate the per unit cost of routine production of multiple unit pump orders.

6. REPORTS & DRAWINGS

The results of the specified studies will be presented in one engineering memorandum or letter report, suitable concept drawings, and by cost estimates prepared in conformance with ERDA cost principles.

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6.1 Technical and Financial Reports

The vendor shall supply:

- a) Brief weekly informal report (Telecon, Telecopy, TWX, etc.) to indicate on-going work, progress, or problem areas.
- b) One monthly technical progress and financial report on August 31, 1976.

6.2 Meeting Schedule

As a minimum, the following meetings will be scheduled at the vendor's offices;

- a) One day progress meeting at the vendor's plant about mid-August, 1976.
- b) Summary meeting to present concept to system design organizations near the end of the program.
- c) Other meetings, as required.

7. SUBMITTALS

Attachment A to this specification lists the required submittals and their scheduled due dates.

PRELIMINARY ATTACHMENT A: DOCUMENT SUBMITTAL REQUIREMENTS

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REV 0 TO ATTACHMENT A 23A2126

EQUIP. TYPE

ITEM	a	b	c	d	e	f	g	h	i	j
PROJECT										
P.O.										
M.P.L.#										

CATEGORY CODE	DOCUMENTS TO BE SUBMITTED	G.E. REVIEW	DOCUMENT FINAL NEED CODE*										COPIES PER DOCUMENT SUBMITTED		CYCLE TIME (WEEKS)	DOCUMENT SCHEDULE			
			a	b	c	d	e	f	g	h	i	j	NUMBER	TYPE		G.E.	APPROVAL FOR USE REQUIRED BY	CERTIFICATION REQUIRED BY	
	5.3 <u>Final Letter Report, to include:</u>	I											20				9/30/76		
	5.2 Design Concept Drawings	-											*						
	5.3.1 thru 5.3.6 Supporting Analyses																		
	5.4.1 Program Plans & Schedules	-																	
	5.4.2 Interfacing and Auxiliary Requirements	-																	
	5.5 Development Needs Memorandum	-																	
	5.6 Cost Estimates	-																	
	6.1 <u>One Monthly Technical and Financial Report</u>	I											20				8/31/76		
	*(1) REPRODUCIBLE REQUIRED OF DRAWINGS																		

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LARGE SCALE BREEDER REACTOR PLANT HOT LEG
MECHANICAL PUMP CONCEPTUAL DESIGN STUDY

TABLE 1 LSBR PLANT PROTOTYPE PUMP CONFIGURATION REQUIREMENTS

	<u>PHTS</u>
1. Pump Type	Mechanical, free surface
2. Location in Sodium Loop	Hot leg
3. Pump Axis Orientation	Vertical
4. Number of Pumps in HTS	3
5. Discharge Nozzle Orientation/ Diameter	Horizontal (side) preferred/ 36 inches O.D.
6. Inlet Nozzle Orientation/ Diameter	Vertical (up) preferred/ 36 inches O.D.
7. In-Sodium Bearing(s)	As required, hydrostatic
8. Cover Gas Seal	As required to retain cover gas and exclude the ambient tmosphere. Seal must be mounted above Reactor Normal Sodium Level.
9. Pump Drive	Electric motor and all auxiliary equipment to turn the pump shaft at the speed demanded by control signals; 4:1 turn down ratio.
10. Pony Motor Drive	Auxiliary electric motor to drive pump to produce some low value of flow when main motor drive is turned off.
11. Sodium Level Control	Sodium Level Control shall be achieved by a vendor specified method so as to preclude sodium intrusion into the thermal and radiation shield region during normal pump operation.

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MECHANICAL PUMP CONCEPTUAL DESIGN STUDY

TABLE 1 (CONTINUED)

PHTS

12. Radiation Shield

Biological protective barrier between radioactive sodium and operating floor to attenuate gamma radiation to permissible biological dose rates.

13. Thermal Shield

Heat barrier in the inert gas space between sodium surface and the pump structure and equipment above.

14. Pump Material

Stainless Steel

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TABLE 2 LSBR PLANT PROTOTYPE PUMP OPERATING REQUIREMENTS

		<u>PHTS</u>	
	<u>Units</u>	<u>Value</u>	
1. Design Flow Rate	gpm	77,000 ^(a)	
2. Head Rise at Design Flow	ft	500 ^(b)	
3. Operating Cover Gas Pressure			
Reactor Cover Gas System	inches H ₂ O	10 max	
4. Total Suction Available at Pump Inlet Nozzle (NPSH)	ft	30 (Figure 4) at 21 ft below normal reactor level	
5. Flow Rate at Pony Motor Speed		(Figure 5) ⁱ	
6. Shut-off Head at Pony Motor Speed			
Maximum	ft	5'-0" ^(c) Figure 5	
7. Head at Three Loop Operation	ft	(Refer to Figure 2)	
8. Maximum Rate-of-Change of Flow (10 Sec by Control System)	%/Sec	1	
9. Temperature			
Design	°F	950	
Nominal	°F	935	
Minimum	°F	300	
10. Availability	%	99	
11. Frequency of Planned Maintenance	Hrs	10,000 ^(g)	

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TABLE 2- LSBR PLANT PROTOTYPE PUMP OPERATING REQUIREMENTS

		<u>PHTS</u>
	<u>Units</u>	<u>Value</u>
12. Free Surface Level Below Operating Floor	Ft	Refer to Figure 3
13. Head vs Flow Characteristic Shall have Negative Slope in the Design Flow Range		(See Figure 2)
14. Maximum Overspeed (% of Design)	%	105
15. Pump Tank Internal Impedance (Min)	psi/gpm ²	3 x 10 ⁻⁶ (d)
16. Pump Tank Sodium Inventory (Max)	ft ³	80 ^(e)
17. Pump Locked Rotor Impedance	ft/ ω^2	≤ 250 (f)
18. Design Life	Yrs	30
19. Transient Events	Use CRBRP Pump E-Spec 22A3444, Revision 8 Appendix B, Para. 3.1 as modified by 5.1.b of this specification.	
20. Seismic Loading	Use CRBRP Pump E-Spec 22A3444, Revision 8, Appendix B, Para. 2.0.	
21. Nozzle Loads	(Refer to Table 3)	
22. Structural Design Requirements		
a) Internal		
Pump Tank & Inlet Nozzle		
Temperature	^o F	950
Pressure	psig	30
Pump Hydraulics & Discharge Nozzle		
Temperature	^o F	950
Pressure	psig	200

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TABLE 2 LSBR PLANT PROTOTYPE PUMP OPERATING REQUIREMENTS

		PHTS	
	<u>Units</u>		<u>Value</u>
22. Structural Design Requirements (Continued)			
b) External (Under Internal Vacuum)			
Temperature	°F		450
Pressure	psia		14.7

Notes for Primary Pump

- (a) The pump shall be designed to operate at the design flow over the sodium temperature range of 400°F to 950°F (Structural Design Temperature).
- (b) Total Dynamic Head Rise from Pump Tank Inlet Nozzle to Pump Tank Exit Nozzle.
- (c) Pony Motor nominal set point in speed range 5% - 10%.
- (d) The value represents the impedance between the tank region and the sump region (See Figure 3) and is required to dampen flow oscillations following a trip and to permit some reasonable pressure difference between the cover gas spaces in the pump tank and reactor vessel following a pipe rupture. This value is to be verified by appropriate analyses after the cover gas system has been designed.
- (e) This value is the sodium inventory change in the pump tank during the transient from 100% to 0% flow.
- (f) This figure is the impedance to flow in the positive direction through the pump when the rotor is locked,

$$H \leq 250 \bar{w}^2$$

where H = pressure drop through pump, feet

$\bar{w}$ = normalized flow, i.e., the ratio of $\frac{\text{flow (system)}}{\text{flow (design)}}$

- (g) All pumps must be available 100% of the time during plant operation. During shutdown and refueling operations, one pump may be down at any time to perform scheduled or corrective maintenance with the other two pumps operating at pony motor speed for decay heat removal.

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TABLE 3. LSBRP PRIMARY PUMP NOZZLE LOADINGS (PRELIMINARY)

Nozzle	Dead Weight Loads				Thermal Expansion Loads				OBE Seismic Loads			
	P (lbs)	V (lbs)	T (ft-lb)	M (ft-lb)	P (lb)	V (lb)	T (ft-lb)	M (ft-lb)	P (lb)	V (lb)	T (ft-lb)	M (ft-lb)
SSE Loads = 1.8 OBE Loads												
<u>Discharge</u>												
Vertical	18,170	0	10,930	15,460	10,490	14,840	21,880	30,940	20,980	29,670	21,880	30,940
Horizontal	0	18,170	10,930	15,460	10,490	14,840	21,880	30,940	20,980	29,670	21,880	30,940
<u>Inlet</u>												
Vertical	18,170	0	10,930	15,460	10,490	14,840	21,880	30,940	20,980	29,670	21,880	30,940
Horizontal	0	18,170	10,930	15,460	10,490	14,840	21,880	30,940	20,980	29,670	21,880	30,940

Above loads are Maximum Loads based on 316 S.S. at 950°F and 1/2" Wall Thickness. These values should be used both as absolute loads and as a load range.

P - Axial V - Shear T - Torsion M - Bending

Inlet Nozzle OD = 36"

Discharge Nozzle OD = 36"

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SCHEMATIC OF LARGE SCALE BREEDER REACTOR PLANT
 HEAT TRANSPORT SYSTEM

FIGURE 1
 (For Information Only)

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Definitions:

- ①
- ④⑤⑥
- ①②④⑤
- ①②④⑥③
- ⑦
- ⑧

Design Point
 Pump Characteristic at 5% Design Speed
 Required Operating Range
 Stable Pump Characteristics (Negative Slope)
 Maximum Runout (transitory - See Para. 5.1a)
 Minimum Shutoff Head Requirements shall be Based on the Ability to Start Up as Defined in Section 3.3.1.4 of Specification 22A3444, Revision 8, except that Minimum Main Drive Motor Speed shall be 25 Percent of Design.

(Typical For Information Only)

FIGURE 2
 PUMP STEADY STATE
 OPERATING BOUNDARIES

(TBD)

- a) IMPELLER ϕ IS OPTIONAL BELOW MSL
- b) PREFERRED SUCTION NOZZLE LOCATION
- c) AZIMUTHAL ORIENTATION OF NOZZLES, OPTIONAL

FIGURE 3
SCHEMATIC OF LSBRP PRIMARY HOT LEG
PUMP INSTALLATION

TITLE

LARGE SCALE BREEDER REACTOR PLANT HOT LEG
MECHANICAL PUMP CONCEPTUAL DESIGN STUDY

SPECIFICATION

Breeder Reactor Department
GENERAL ELECTRIC
Sunnyvale, California

Spec No. 23A2126

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Note: Refer to Figure 3 for
Elevations

Figure 4 NPSH Available at Inlet Nozzle Vs
Design Flow

(Typical, For Information Only) (HOLD)

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FIGURE 5 SYSTEM PRESSURE VS FLOW AT PONY
MOTOR OPERATION

(Typical, For Information Only)

(HOLD)

BYRON JACKSON PUMP DIVISION
BORG-WARNER CORPORATION

761-G-0002

LARGE SCALE BREEDER REACTOR
PROTOTYPE MECHANICAL PUMP
CONCEPTUAL DESIGN STUDY

Cold Leg

PREPARED UNDER CONTRACT 67002
FOR
GENERAL ELECTRIC FAST BREEDER
REACTOR DEPARTMENT
LMFBR PUMP DEVELOPMENT PROGRAM

TASK 1

July 1976

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1.0 SCOPE OF WORK

This final report is a complete conceptual design study of a mechanical pump for a large scale breeder reactor plant. The pumps are located in the cold leg side of the loops. This makes the net positive suction head available - NPSHA - low, and is, in fact, a major influencing factor in the design.

Where possible, experience gained from the Clinch River Project and the FFTF is used in this study. Experience gained in the design, manufacture, and testing of pumps in general and sodium pumps in particular is reflected in this report.

The complete contents of this report are combined in an attempt to satisfy the intention of the Specification, particularly the feasibility of this project. It includes estimated cost and time schedule for design, manufacture and testing. It also includes a recommendation for development needs.

2.0 OBJECTIVE

- 2.1 To provide a design study of a recommended concept for a mechanical sodium pump to be used in the primary loop of a Large Scale Breeder Reactor Plant.
- 2.2 To support the concept recommendation by:
 - 2.2.1 Applicable experience
 - 2.2.2 Technical justification
 - 2.2.3 Order of magnitude cost estimates
 - 2.2.4 Identifying problem areas and concerns
 - 2.2.5 Identifying, describing and justifying required development needs, including their order of magnitude cost and schedule.
- 2.3 To provide documentation in support of the pump design concepts in the form of a report including drawings, schematics, and other data as required.
- 2.4 To provide consultations and technical assistance as needed with LSBRP systems on problems related to pump/systems interfacing, during and following pump concept submittal.
- 2.5 To insure that the basic pump configurations studied can meet the system requirements.

3.0 SUMMARY

This total report is made up of a number of related studies that provide documentation in support of the pump design concept study. It includes engineering memorandums, drawings, key feature descriptions, and other data required by the scope of this conceptual pump study. Some of the reports, such as manufacturability and some stress analysis, were done by consultants for Byron Jackson. It should be emphasized that while these reports are accepted and transmitted in good faith, Byron Jackson experience will be used and not compromised. Where required, our own recommendation or choice will be given prior to the respective report.

A review of this complete report indicates that the design is feasible. The pump can be manufactured to system and Specification requirements. The overall length and weight of some pieces will require special consideration, but is within the scope of equipment and technology available today.

The conceptual pump design is unique in some respects, but is based on sound, proven principles. Many pumps using impellers and pump cases of this size and type have been built and are in use for water applications. The fabricated parts are large and heavy, but can be manufactured and machined. Only the high temperature is unique to this size, since previous sodium pumps were smaller.

SUMMARY (Con't)

Non-destructive tests as required by the Code are described and are feasible. There is of course additional preparation and handling required.

The performance test of the prototype has been studied thoroughly. It is an important part of the work involved. However, it is feasible for a cold water test.

There are some problem areas, better described as "concerns". However, all of them can be solved.

In a study of this scope there are always development needs. These are listed separately. The most important one is for building and testing a small scale model. There is much to be gained in the building and testing of a model that could be projected as a saving in time and money for the prototype.

4.0 EVALUATION FACTORS

4.1 INTRODUCTION

There are many factors which influence the design and performance of pumps in nuclear service, and in particular the sodium pumps required in a power-producing Breeder Reactor. In non-nuclear applications the pump supplier rarely is given more than the desired flow rate and pressure rise for a particular liquid at some temperature. For the majority of such non-critical pumping applications the lack of additional information does not lead to serious problems with the installation. However, the sodium pumps with which we are now concerned are such integral and critical components of the reactor that many factors relating the pumps to their systems must be taken into consideration very early in the conceptual phases of plant and pump design.

4.2 FACTORS INFLUENCING PUMP DESIGN

The factors which influence the design of sodium pumps may be grouped under five major categories. Hydraulic parameters, operating environment, physical requirements, service requirements, and stress analyses cover the areas normally taken into consideration.

These are tabulated in Table 1.

- 4.2.1 Hydraulic parameters include flow rate, total developed head, net positive suction head, pump efficiency, and range or modes of operation.

The desired flow rate, developed head, and suction head combined to determine the maximum design speed of the pump. The design speed determines to a large extent the overall size and weight of the pump assembly, including the driver. With flow rate and developed head, the pump efficiency determines the amount of power required to drive the pump. The pump efficiency, while an important economical consideration, should not be obtained at the sacrifice of reliability and operability. The operating range of the pump in terms of the flow rate may be controlled either by speed or by throttling the system. If more than one point of operation is desired, certain compromises in pump design may be involved. A throttled system wastes energy and may compromise pump performance and efficiency, whereas a variable speed method of flow control usually requires special attention to shaft design to assure that the critical speed is safely outside the operating range.

- 4.2.2 The operating environment includes the properties of the fluid being pumped, the nature and condition of the surrounding ambient, and numerous system-imposed conditions. The chemical properties of the fluid must be considered in making proper selections of compatible materials of construction. The physical properties of the fluid must be known in order to determine net positive suction head available to the pump and the power required to drive it. The nature and condition of the surrounding ambient must be known to determine need of auxiliary

cooling or other protection for pump or driver components. System-imposed conditions include steady-state and transient conditions of temperature, pressure, mechanical and seismic loading and nuclear radiation. These factors all relate to the structural integrity of the pump as opposed to the functional requirements. Often compromises in functional design must be made to accommodate factors dictated by structural considerations.

- 4.2.3 The physical requirements placed on the pump usually are composed of limitations on weight and space, the necessity for radiation shielding, and containment and isolation of the pumped fluid.

The importance of minimization of pump size and weight reach deep into the considerations of plant capital cost, as they affect not only the pump cost, but the size of the plant, the plant structure for pump support, and the size of auxiliary equipment required for installation and maintenance. Radiation shielding may be incorporated as an integral part of the pump for biological reasons and for protection of materials which are susceptible to radiation damage. The containment and isolation of the pumped fluid are significant problems when one considers that conventional pump drivers must be remote from the hostile environment of the pumped fluid, necessitating some type of shaft seal.

- 4.2.4 Pump service requirements normally are specified in quantitative terms for design life and reliability and at least qualitatively for availability. The specified design life of the main pumps in a plant is typically equal to the design life of the systems they are serving, with provision for preventative maintenance and repair or replacement

of components subject to wear or other types of damage. The reliability of the main pumps in a plant must also be high to achieve full output and economical operation. Availability involves life, reliability, and maintenance considerations. Components which have a predictable short life, or poor reliability must be incorporated in such a way as to be easily and quickly removed, repaired, or replaced.

- 4.2.5 Code and Specification requirements must be supported by appropriate stress analyses. This is to determine that the concept has no problems of an insolvable nature. The natural frequencies and critical speed of the pump and motor combination shall be acceptable relative to the pump operating speeds.

The analysis shall take into consideration the loop piping reaction forces on the pump tank nozzles. Seismic response spectrum peaks shown in the specification must also be considered.

The bearings, particularly the thermal effects on the method of support, shall be analyzed. Seismic supports in the form of constant load snubbers may be required. This method of support would require suitable attachments at the pump location in the field.

4.3 DESCRIPTION OF REFERENCE PUMP

This design study provides a recommended concept of a mechanical sodium pump for use in the primary loop of a Large Scale Breeder Reactor Plant. There are three identical sodium pumps, each located in a cold leg of the primary circuit. They are each rated at 127,000 gpm, 500 feet developed head, and each requires over 18,000 brake horsepower.

The three identical sodium pumps are each located in a cold leg of the circuits and take suction from the shell side of one of the primary heat exchangers and discharge into the reactor.

No suitable shaft seal has been developed to date to seal against molten salt. Hence a vertical shaft orientation has been established, incorporating a liquid free surface with an inert cover gas above it. The shaft seal then may be located in a relatively low temperature area free of thermal transients and must only prevent the escape of the cover gas to the atmosphere. The simplest and most conventional type of shaft seal for sealing gas is an oil-lubricated face type mechanical seal of the type proposed.

The hydraulic assembly of the proposed sodium pump is decoupled both hydraulically and mechanically from the outer tank. The mechanical decoupling eliminates structural redundancy and should minimize distortions and misalignment resulting from mechanical or thermal expansion loading. The entire pump assembly with the exception of the drive motor is flexibly supported to allow for horizontal and vertical displacements from the cold to hot conditions. This feature is incorporated to reduce thermal expansion loading on the nozzles of the outer tank. The hydraulic decoupling of the suction piece and volutes and the outer tank will minimize variation of the liquid free surface level in the pump tank during various modes of operation.

The seal assembly, although designed for a long life, contains the least reliable and shortest-lived components in the proposed design, therefore provision has been made for its independent removal from the pump assembly.

This maintenance procedure is made possible by the incorporation of a gas pressure-actuated static shutdown seal above the shield plug to contain contaminated cover gas as the subassembly is removed.

4.4 CONCLUSIONS

The complete report is in part an evaluation of the factors enumerated here. The conclusion is that the pump concept is feasible. There are concerns in some areas and development needs. These are shown in a separate section of this report.

TABLE 1 FACTORS INFLUENCING PUMP DESIGN

HYDRAULIC PARAMETERS	<ol style="list-style-type: none"> 1. FLOW RATE 2. TOTAL DEVELOPED HEAD 3. NET POSITIVE SUCTION HEAD 4. PUMP EFFICIENCY 5. OPERATING MODES AND RANGE
OPERATING ENVIRONMENT	<ol style="list-style-type: none"> 1. FLUID PROPERTIES 2. AMBIENT CONDITIONS 3. SYSTEM IMPOSED CONDITIONS
PHYSICAL REQUIREMENTS	<ol style="list-style-type: none"> 1. WEIGHT AND SPACE 2. RADIATION SHIELDING 3. CONTAINMENT
SERVICE REQUIREMENTS	<ol style="list-style-type: none"> 1. DESIGN LIFE 2. RELIABILITY 3. AVAILABILITY
STRESS REQUIREMENTS	<ol style="list-style-type: none"> 1. DYNAMICS 2. NOZZLE LOADS 3. SEISMIC RESPONSE 4. BEARINGS

5.0 PUMP SELECTION

5.1 ABSTRACT

The method used to select the hydraulic design for this pump is presented here. The reasons^b for selecting this type of pump for the specified service are enumerated and discussed. Through well-known principles such as hydraulic similarity and pump affinity laws, a suitable model was selected. Use of a factor based on size and speed enabled the generation of a complete performance curve.

The specification requirements and design conditions, along with broad experience, led to the basic pump layout. It is from this layout that a sectional drawing showing all the parts and an outline drawing showing sizes are made and submitted.

Materials selected for the different parts will be suitable for the service and in compliance with the required existing codes.

5.2 CONCEPT SELECTION

From the pump operating requirements shown in the specification, it can be seen that the low available NPSH of 20 ft. will probably be the influencing factor in the design selection. This also virtually dictates a double suction impeller, at least for the first stage of a multi-stage pump.

Various combinations were explored, as can be seen from the tabulation of the pump concepts. Five concepts are shown. Work sheets showing the method of selection and the existing model are not included in this report.

For illustration, the work sheet for the concept with two double suction impellers in parallel will be used. The flow for each impeller will of course be half of the total required flow, or about 65,000 gpm. The head remains the same. Using the limiting NPSH of 20 ft. and a speed of 600 rpm, the suction specific speed is found to be satisfactory.

Using the model factor, the ratio of the speeds and the test curve for the model, a performance curve for the prototype is generated. The efficiency is adjusted to compensate for the additional losses in the prototype design. The curve for this concept is shown on PC-32487.

5.3 DISCUSSION

5.3.1 HYDRAULIC SIMILARITY

For two different pumps, hydraulic similarity is established when the specific speed of the model pump is equal to that of the new pump.

The specific speed is defined as:

Equation 1

$$N_s = n \frac{Q^{1/2}}{H^{3/4}} \quad S_s = n \frac{Q^{1/2}}{(NPSH)^{3/4}}$$

where

N_s = specific speed

S_s = suction specific speed

n = rotational speed, rpm,

Q = capacity at the best efficiency point,
gpm,

and

H = head at the best efficiency point, ft.

Also, the head and capacity of the new pump as a function of the head and capacity of the model pump are:

Equation 2

$$Q_2 = Q_1 f^3 (n_2/n_1)$$

Equation 3

$$H_2 = H_1 f^2 (n_2/n_1)^2$$

where subscript 1 and 2 refer to the model pump and the new pump respectively.

Equations 2 and 3 also serve as the definition of the term "f" which is commonly referred to as the model factor or multiplier. The physical significance of "f" is that the dimensions of the new pump can be determined by multiplier. For example, the impeller diameter of the new pump is given as:

Equation 4

$$D_2 = f D_1$$

5.3.2 AFFINITY LAWS

The variation of head, capacity, and brake horsepower with speed follows definite rules known as affinity laws.

Capacity varies directly as the speed.

The head varies directly as the square of the speed..

This is summarized by the following equations, where the subscript "a" refers to the original conditions and subscript "b" refers to the new conditions:

Equation 5

$$\text{Capacity; } Q_b = Q_a \left(\frac{n_b}{n_a} \right) \left(\frac{d_b}{d_a} \right)$$

and

Equation 6

$$\text{Head; } H_b = H_a \left(\frac{n_b}{n_a} \right)^2 \left(\frac{d_b}{d_a} \right)^2$$

The above relationships are commonly referred to as the pump affinity laws.

In addition, by using Equations 1, 5 and 6 it is apparent that for a given pump operating at constant speed, the specific speed change is inversely proportional to the impeller diameter change or:

Equation 7

$$\frac{N_{s,b}}{N_{s,a}} = \frac{d_a}{d_b}$$

5.3.3 SELECTION OF PUMP HYDRAULICS

5.3.3.1 PUMP OPERATING REQUIREMENTS:

127,000 GPM - 500 Ft. - ? RPM

NPSH available at inlet nozzle - 20 Ft.

Look at:

- (a) Two stages in parallel
- (b) Double suction impellers.

Each impeller will supply half the flow and the full head.

For this type of impeller, experience shows that a suction specific speed of about 16,000 is feasible. With this information, we can solve for the speed of the pump:

$$S_s = \frac{N \sqrt{Q}}{(\text{NPSH})^{3/4}} = 16,000$$

$$N = \frac{16,000 \times (20)^{3/4}}{\sqrt{63,500}}$$

$$N = 600 \text{ RPM}$$

There is now enough information to solve for the pump specific speed:

$$N_s = \frac{N \sqrt{Q}}{(H)^{3/4}}$$

$$N_s = \frac{600 \sqrt{63,500}}{(500)^{3/4}}$$

$$N_s = 1440 \text{ (For a double suction impeller)}$$

With this information, we can design the required hydraulics or select a satisfactory model from the vast test data available in our pump test files.

5.3.4 MODEL SELECTION:

Look at a 12" x 12" x 16" double suction pump that has been built and tested. It has been run at 6000 RPM and at 3590 RPM. Use the 6000 RPM test shown on curve T-30948 for performance and the 3590 RPM test shown on curve T-31127 for NPSH.

The specific speed, N_s , and the suction specific speed, S_s , are essentially the same for the model and the prototype.

The model factor, F , is found to be 4.33. Applying the model factor to the test curve T-30948, and correcting for the difference in speed, will generate a performance curve for the prototype pump.

$$Q_P = F^3 \times \frac{N_P}{N_M} \times Q_M$$
$$H_P = F^2 \times \left(\frac{N_P}{N_M}\right)^2 \times H_M$$

Using a prototype motor speed of 600 RPM for calculations:

$$Q_P = (4.33)^3 \times \frac{600}{6000} \times 7900 = 64,000 \text{ GPM}$$
$$H_P = (4.33)^2 \times \left(\frac{600}{6000}\right)^2 \times 2650 = 495 \text{ Ft.}$$

The impeller vane will be modified slightly to give a steeper curve.

Leakage losses in the prototype pump will result in a drop in efficiency of that shown for the model.

5.3.5 SHAFT STRESS

A. At coupling - Solid shaft use H.P. at runout

$$\text{BHP} = \frac{165,000 \times 400}{3,960 \times 0.76} \times 0.84$$

$$= 18,400 \text{ HP}$$

$$\text{Shaft dia.} = 12.5" \quad n = 590 \text{ RPM}$$

$$S = \frac{321,000 \times \text{HP}}{n d^3}$$
$$= \frac{321,000 \times 18,400}{590 \times 12.5^3} = 5,125.6 \text{ PSI}$$

Stress Allowable = 20,000 PSI for ASTM A-182 Gr. F304
at 300° F (ASME Sect. III)

B. At column - Hollow shaft (Hot)

$$d_o = 16" \text{ outside dia.}$$

$$d_i = 12" \text{ inside dia.}$$

$$S = \frac{321,000 \times \text{HP} \times d_o}{n \times (d_o^4 - d_i^4)}$$

$$S = \frac{321,000 \times 18,400 \times 16}{590 \times (16^4 - 12^4)} = 3,575 \text{ PSI}$$

Stress allowable = 15,100 PSI for ASTM A-182 Gr. F304
at 800° F (ASME Sect. III)

C. At impeller - Hollow shaft (Hot)

$$d_o = 19" \text{ outside dia.}$$

$$d_i = 15" \text{ inside dia.}$$

$$\begin{aligned}
 S &= \frac{321,000 \times \text{HP} \times d_o}{n \times (d_o^4 - d_i^4)} \\
 &= \frac{321,000 \times 18,400 \times 19}{590 \times (19^4 - 15^4)} = 2,387 \text{ PSI}
 \end{aligned}$$

Therefore the torsional shear stress at every section of the shaft is well below the allowable stress value for the material used at the appropriate temperature.

5.3.6 TYPICAL BOLT STRESS IN COUPLING
(BOLTS CONNECTING SHAFTS IN SODIUM)

$$\begin{aligned} \text{BHP} &= 18,400 \text{ HP} & \text{N} &= 590 \text{ RPM} \\ \text{TQ} &= \frac{\text{HP} \times 5,250}{\text{N}} \\ &= \frac{18,400 \times 5,250}{590} = 163,728.8 \text{ Ft-Lb.} \end{aligned}$$

$$\text{Bolt Radius} = 10''$$

$$\begin{aligned} \text{Shear Force} &= \frac{163,728.8 \times 12}{10} \\ &= 196,474.5 \text{ Lbs.} \end{aligned}$$

Use 16 - 1½" bolts

$$\text{Bolt Area} = 1.405 \text{ in}^2$$

$$\tau = \frac{196,474.5}{16 \times 1.405} = 8,740 \text{ PSI}$$

$$\text{Bolt Material} = \text{ASTM A-453 Gr. 660}$$

$$S_m \text{ at } 800^\circ \text{ F} = 26,600 \text{ PSI}$$

$$\text{Shear allowable} = 0.6 \times S_m = 15,960 \text{ PSI}$$

Therefore the bolt stress is below the allowable stress.

In addition, the bolts will not have to carry the total torque in shear, since the coupling flanges will transmit a big portion of the load through the clamping force of the bolts.

6.0 KEY FEATURES OF THE PUMP

6.1 Pump Case Assembly

6.1.1 Summary of the Key Features of the Pump Case Assembly

- (1) An effort was made to keep the wall thicknesses uniform in the pump castings for good thermal distribution.
- (2) A multiple volute was used to keep the size to a minimum outside diameter and to obtain a low radial thrust factor.
- (3) The pump case rotating element is of one piece construction. This includes the shaft, impellers, and bearing journals. This allows assembly, final machining, and balancing on one unit.
- (4) The pump case assembly hangs free from the top.
- (5) Locating the suction at the bottom eliminates side loads.
- (6) The pump shaft is hollow, for maximum rigidity with minimum weight. This also avoids thick sections.
- (7) The bearings in the pump end are located adjacent to the impellers, close to the radial thrust load.

6.1.2 Pump Case Parts

6.1.2.1 Double suction Impeller.

As has been stated in the pump concept selection, the low available NPSH of 20 feet led to the use of double suction impellers. The velocity at the inlet or eye of the impeller will determine the NPSH characteristic. For a given flow, the velocity will vary with the eye area. Since the velocity must be low for cavitation-free performance, the eye area must be large.

The design of an impeller that will give good performance places limits on the size and therefore the area of the inlet. The peripheral velocity of the outer shroud and the size of the shaft are limiting factors. This leads to two inlets, or a double suction impeller as a means of increasing - or in fact doubling - the inlet area of an impeller.

Looking at the summary of the five concepts studied, it can be seen that a single stage double suction impeller would require a rather large pump at a relatively low speed - about 360 rpm. It was then decided to look at two stages of double suction impellers in parallel. This doubled the inlet area available to the total flow. It also allowed the use of a speed of 600 rpm and reduced the size of the pump. Since the two stages are in parallel, each stage will give the required head.

Since a double suction impeller is symmetrical about its centerline, no axial thrust exists. This is a desirable feature. There are bound to be some inequalities which will produce small amounts of thrust which are largely indeterminate. These are taken by the thrust bearing, which is required to carry the weight of the rotating element.

6.1.2.2 Suction Piece, Lower

The lower suction piece guides the flow to the lower inlet or suction side of the double suction lower impeller. A series of vanes or ribs in the suction piece guide the flow

from the outer suction tank to the impeller eye. The area at the inlet to the suction piece is about $1\frac{1}{2}$ times that of the impeller eye. Location of the suction nozzle at the bottom of the tank allows a uniform distribution to all of the suction inlets. The suction nozzle size is satisfactory relative to the total impeller eye areas.

The lower hydrostatic bearing is an integral part of the lower suction piece. The bearing pockets are formed by ribs that are part of the casting. The bearing end rings are separate pieces welded to the casting. The bore of these rings has a colmonoy hardfacing. The small size of these rings simplifies the overlay procedure. The bearing pockets are made deep for better thermal distribution.

6.1.2.3 SUCTION PIECE, MIDDLE

The middle suction piece guides the flow into the inner inlets or suction sides of both double suction impellers. This casting is made in two halves, split axially, for assembly purposes.

It is a desirable feature to make the impellers and what would normally be the pump shaft in one piece, including the two bearing journals. This allows the pump portion of the rotating element to be machined and balanced as a single piece. The two journals could be concentric to each other within close limits. The impeller wear rings could also be concentric to the bearing journals and to each other, to close tolerances.

Since the middle suction piece is located between the two impellers, it must be made in two halves for assembly. The two halves can be lightly

bolted at the parting surface. They will be secured at the flange common to the upper and lower volutes. The two halves will be machined as a single piece, after the parting faces have been finished, by the use of a fixture. This will allow the centering fit, bores, and turns to be accurate. Some temporary means of maintaining this position will be provided until the part is assembled with the volutes.

6.1.2.4 SUCTION PIECE, UPPER

The upper suction piece serves the same function as the lower. It guides the flow into the upper inlet or suction side of the double suction upper impeller. The corresponding areas are of course the same.

In addition, however, this casting contains the discharge manifold. The six discharge outlets from the volute are carried to the discharge column assembly. The areas are made to give a smooth flow with uniform change in velocity.

6.1.2.5 VOLUTE CASING

The rotating impeller action gives the liquid a higher pressure and higher velocity as it leaves the impeller than at its entrance. The velocity is partly converted into pressure by the pump volute before it leaves the pump through the discharge manifold. This conversion of velocity into pressure is accomplished in a volute casing. The impeller discharges into a single or multiple channel of gradually increasing area called a volute. Another part of the conversion takes place in the collection chamber of the discharge column, the reducing elbow, and the discharge column.

The total pump capacity passes through the volute throat. The volute areas are gradually increased from the point nearest the impeller exit, called the volute lip or cut-water, toward the discharge manifold. For two stages in parallel, there are two volutes, half the total capacity passing through each.

Due to the large diameter with only half the capacity passing through each volute, a multiple volute is being used for this pump concept. Each volute has six openings. This will keep the radial thrust factor very low, and hence the radial thrust will be low. It will also cut down the length of the shallow radial clearance between impeller exit and volute wall, since there will be six volute lips. Each volute section is carried to the discharge manifold separately, instead of to a circular volute scroll. Multiple volutes divide the required area into smaller sections, keeping the volutes down to a small diameter. This allows the use of a smaller tank.

Sodium pumps with multiple volutes are not unusual. The Fermi pump has five volutes, and the one for Argonne National Lab. has four volutes. These pumps were made and tested. Clinch River is designed with three volutes. The lower stage discharges into a common passage with the upper one. The area is of course made large enough for the total flow.

The suction openings are between the discharge passages. This of course means that there are six entrances from the tank to each impeller eye. Provision is made in the lower volute to feed the lower hydrostatic bearing. This flow must come from the discharge side of the pump.

The volute area is determined by the pumps used as a model. Of course the model factor must be used to give the required area for the prototype. As a check, the average volute velocity can be approximately calculated, using a volute constant based on the specific speed of the pump. Using this method, the average volute velocity is about 67 feet per second. Using the flow and the area at entrance to the volute, the average velocity is about 60 feet per second. This is a close correlation and a good check on the model selection.

6.0 KEY FEATURES OF THE PUMP (Con't)

6.2 Internal Taper Fit for Leakage Control

The design of this pump includes a tapered adjustable clearance to minimize leakage or circulation from discharge pressure back to suction. In terms of pump head at design flow, this could be a discharge head of 500 feet to a suction head of 20 feet. The leakage or circulation affects performance in the form of additional power due to the loss in pump efficiency. It is a design objective, therefore, to keep this leakage down to a minimum flow. The leakage from discharge to suction takes place only around the periphery of the discharge nozzle. The leakage path is short only in the vertical position and keeps increasing in length, reaching a maximum in the horizontal position. Since leakage is a function of pressure, clearance, and length, the design controls clearance and length of the leakage path. This will keep the leakage or recirculation to a minimum.

In a sodium environment it is not practical to use a gasket. The large size and required freedom of movement dictate an opening with a static clearance. The size also discourages the normal straight female bore and male turn to obtain a small clearance, since an adverse tolerance stack-up would make the clearance greater than desired. Size of the parts makes it very difficult to manufacture and inspect to close tolerances.

The leakage or recirculation flow is a function of the pressure drop and the clearance. Since we cannot alter the pressure drop, we chose to control the clearance. This led to a female and male taper

where the clearance could be controlled by axial movement between them. The turn and bore would not require close tolerances, as long as the tapers were essentially close to each other.

For this pump concept, the pump tank has the female taper at the discharge location and remains in the fixed position. The matching taper is on the discharge column and is adjustable axially to give the desired clearance. An arrangement consisting of a series of threaded fasteners is used to move the discharge column axially relative to the pump tank. Another set of cap screws or studs is used to maintain this setting.

Assuming matching tapers, the clearance can be set to a pre-determined value by the relative axial movement between the two parts. The two parts can be allowed to engage lightly for the zero clearance. With the taper in inches on diameter per foot of length known, the desired clearance on diameter can readily be converted to axial movement. This can be measured accurately, and locked in position, thus assuring the diametral clearance.

6.2.1 The use of a taper versus a straight bore enables:

- (1) Use of more liberal tolerances on each piece with no sacrifice of a close clearance.
- (2) An adjustable diametral clearance.
- (3) A means of setting a pre-determined clearance.
- (4) The engagement of the taper to determine the diametral clearance.

6.3 Hydrostatic Bearing

This pump concept cannot be designed with an overhung shaft, therefore some sort of guide bearing - or bearings - is required. There is no lubrication available and the bearing environment is the liquid being pumped. The bearing must be reliable and designed for a long life. It cannot depend on the common-type lubrication film.

For this application a hydrostatic bearing was selected that depends on flow and pressure. It is an integral bearing taking the flow from the fluid being pumped and the pressure from the discharge side of the pump. The bearing is sized and located axially relative to the load to reduce large moments.

This pressurized bearing is basically a non-contact centering device, as shown schematically on Fig. 1. The bearing has a series of pockets located radially around the rotating journal. These pockets have orifice holes that are exposed to pump discharge pressure. Ideally, with the journal in the center of its clearance, the pressure is the same in all of the pockets. Under radial load, the journal moves off center toward one of the pockets. The clearance is decreased, the flow is also decreased, and the pressure is increased. The clearance on the opposite pocket is increased, the flow is also increased, and the pressure is decreased. This combination of events takes the radial load and has a tendency to keep the journal centered. The load is taken by the pressure and during normal operation avoids contact between the journal and the bearing. For slower speeds, the available pressure is lower, but the radial load is also lower, so the bearing will function normally. This description is an over-simplification.

The bearing must be sized properly, clearances and orifice sizes are critical, and the returns must be adequate.

For this pump concept we have chosen two bearings to take the radial loads of the two impellers. The bearings are located adjacent to the impellers. In addition, we have located a third bearing to break the long span from the pump impellers to the motor bearing. This will keep the critical speed within acceptable limits, and those imposed by the Specification.

The normal bearing takes flow from the discharge side of the pump and returns the flow to suction at both ends of the bearing. This is true for the lower bearing shown in the pump. The middle and upper bearings have discharge pressure on one side and suction pressure on the other. The return to suction takes place on only one side of the bearing. This makes the bearing less efficient, and to compensate for this, we have made the bearing larger.

Bearings of this design have been used on a previous sodium pump and on many light water applications. There is no apparent difference in the functioning of the bearing.

At start-up, the journal may contact the bearing surface. This in itself is not harmful. To give a harder bearing surface and insure long life, the journal and the end rings of the bearing are hardfaced with colmonoy. This is described in the special material section.

Bearings of this type have been tested with water at Byron Jackson. They have also been used successfully on sodium pumps in the field. The size selected is now being used on pumps for light water reactors.

If a model for this pump is built, as will be recommended, the bearings could be tested in the model.

HYDROSTATIC BEARING (FUNCTIONAL SCHEMATIC)
BASICALLY A NON-CONTACT CENTERING DEVICE

FIG. 1. HYDROSTATIC BEARING

6.4 Pump Shaft

While the main function of the pump shaft is to transmit torque from the motor, it is by no means the only one. It must maintain good alignment under all thermal conditions. The deflection under load must be limited to stay within the critical speed requirements. The alignment and balancing tolerances must be such that vibration during all modes of operation will be within acceptable limits.

Size of the pump, and length and complexity of the shaft, required manufacture in more than one piece. The complete shaft is made up of four sub-assemblies. The shaft is hollow except in the cold region at the top. A hollow shaft will give maximum rigidity with minimum weight. It will also give better stability under thermal transients than a solid shaft.

- 6.4.1 The lower shaft sub-assembly consists of two short hollow shafts, the two impellers, and the two bearing journals. The journals will be ground concentric to close tolerances, as will the impeller wear rings. The flanged fit to the column shaft will then be machined true to the journals. The complete sub-assembly will be balanced to the limits required. The journal sleeves are hardfaced before mounting on the shaft. This is for the convenience of handling a small, light piece during the colmonoy overlay procedure.

The inside and outside of the hollow stub shafts will be machined, particularly to obtain uniform wall thickness and good balance.

6.4.2 The middle shaft consists of a long tubular piece with a flange at each end. The tube will be a selected grade to obtain uniform wall thickness. The inside diameter will not be machined. The outside diameter will be machined only if inspection shows that uniform wall thickness is not within acceptable tolerances.

A bearing journal is required at the top end of the middle shaft. Again, a sleeve is preferred for the journal.

6.4.3 The top shaft consists of a tube and a solid shaft. The transition from a hollow shaft is made well above the hot sodium level. A stub shaft is added at the top which is in effect a pump coupling, a spacer, and a mounting device for the mechanical seals.

6.4.4 The stub shaft is coupled to the top shaft by means of a self-centering spline. The axial load tending to separate the spline is taken by a large cap screw or a series of fasteners. This type of fixed coupling is described below.

The seals are mounted on the stub shaft in such a manner that the complete assembly can be removed from the pump without disturbing the motor. This is a desirable feature even though the seal is designed to give a very long life.

The seals could be mounted on a sleeve that is removable from the top end of the shaft, thus eliminating the stub shaft. A pump coupling would be required, as well as a coupling spacer used to provide space for seal removal. This would add an extra piece and an additional bolted joint.

6.4.5 The self-centering spline-type coupling used at the top end of the pump shaft gives extreme accuracy and maximum load-carrying capacity. The teeth for this coupling are cut from the solid face at the end of the shaft. The tooth spacing is highly accurate and a fine surface finish is readily attainable.

This design provides an accurate self-contained connection in which the teeth both center and drive. It is easy to assemble and dismantle. There are no close bores or turns, no need for shrink fits, and no keyways. The teeth are machined directly into the face of the part.

This fixed coupling is a face spline for joining two sections of a shaft to form a single operating unit. It combines the features of accurate alignment, precision centering, and positive drive.

For this application, the coupling requires a means of carrying the weight of the rotating element. Also, the two members must be fastened axially to resist any separating force generated by the shape of the teeth. This is readily accomplished by the use of a single large cap screw or a number of small ones. The tension in this fastener must be sufficient to keep the coupling teeth in full engagement under all conditions of operation.

6.4.5 The shafts are coupled rigidly with bolted flanges. They will be centered by male and female fits. The size is such that required machining tolerances, concentricity, and squareness are within manufacturing capability.

6.5 Axial Thrust Requirements

The pump does not have a bearing for carrying an axial thrust load. It depends on the thrust bearing used in the motor. The motor supplier must design the bearing to carry the additional load required by the pump.

The weight of the pump rotating element is estimated to be 39,000 lbs. This is of course an axial downthrust, static and at all speeds.

Double suction impellers are theoretically balanced axially. However, actually an axial thrust is possible. This is due to slight differences in pressure on the upper and lower sides of the shrouds. This is at least in part due to the difference in actual running position between the rotating shroud and the adjacent stationary wall for the two sides of the impeller. With two stages, this could balance out. If assumed to be additive, the axial thrust could be 20,000 lbs. in either direction. This is with the pump running at full speed. Since this load is a function of pressure, it is also a function of speed. The downthrust requirement would be lower for the slow speeds. The motor thrust bearing is therefore sized to take the highest load required by the pump.

The upthrust due to pressure at the impellers will not be high enough to overcome the dead weight of the pump rotating element. Even though the pump requires no axial upthrust on the motor bearing, it is anticipated that the motor thrust bearing will be designed to take some load in that direction. This is for protection due to sudden unforeseen thrust surges either by the pump or the motor rotor.

6.6 Interface with Motor

The motor, though not furnished by the pump supplier, is nevertheless an integral part of the pump motor assembly. Besides supplying the power required to drive the pump, it contains the thrust bearing.

The weight of the pump rotating element, as well as any axial hydraulic thrust, is taken by the motor thrust bearing. A summary showing the axial thrust requirements of the pump under all operating conditions is shown on Fig.

Utilizing the motor thrust bearing to carry the axial pump thrust eliminates the need for one in the pump. This avoids the duplication of a lubrication and cooling system. Since the seal would normally be located below the bearing were it in the pump, it would be much more difficult to remove the seal. It is highly desirable to utilize the motor thrust bearing to carry the additional axial thrust load required by the pump.

If a non-reverse is required for the pump-motor assembly, it should be placed in the motor. Also, if a flywheel is required, it could easily be placed in the motor. The reasons are similar to those enumerated for the thrust bearing.

The interface from motor to pump must be resolved mutually. The squareness and concentricity between the motor shaft and its mounting surface to the pump must be held to close tolerance. Provision shall be made for inspection to the prescribed limits.

The coupling from the pump shaft to the motor shaft is provided by the pump supplier. It is of the rigid type, since it carries the axial

thrust load. Special requirements on the motor shaft extension are made to suit the pump supplier's design.

Provision is generally made to enable some alignment adjustment between the motor and pump shafts. Size and required accuracy make the usual rabbett fit impractical. Some form of doweling in position or fitted face keys to set and maintain alignment will be required.

7. SHAFT SEAL

1.0 OBJECT

The objective of this study is to provide a conceptual design for the shaft seals in the liquid metal recirculating pump for Development 893 Task 1 Large Scale Breeder Reactor Plant.

2.0 SHAFT SEAL CONSTRUCTION AND OPERATION

- 2.1 Details and components of the shaft seal assembly are shown on assembly drawing 1E-3893.
- 2.2 The shaft seal assembly is based upon a modular design in which the mechanical seals, components of the seal cooling system, the pump stub shaft, the pumping ring and the seal flanges and housing are assembled as a complete cartridge. This entire package and the lubricating oil for the mechanical seals provides the sealing system of the argon cover gas.
- 2.3 The modular design permits removal of the assembly from the pump without removing the driver.
- 2.4 The cartridge is equipped with a cooling system which removes heat generated primarily by the seals at the interfaces. The cooling system consists of the two pumping rings, the heat exchanger, and the fan. When the shaft rotates, the pumping rings develop pressure differentials which result in circulation of oil past the seals and through the heat exchanger. The externally mounted fan forces ambient air, which acts as the coolant, through the tubes of the exchanger.
- 2.5 Plates mounted on the upper flange and inserted into the stub shaft lock and locate the stub shaft in the axial direction when the cartridge is out of the pump. These devices prevent excessive axial movement and locate the seal to provide correct spring compression on the sealing surfaces. The devices are removed after the driver is coupled to the shaft.
- 2.6 Refer to sectional drawing 1C-3563 and material of construction sheet SK-0002-45 for identification and location of parts in the cartridge.

3.0 SEAL ELEMENTS

The basic seal assembly with identified components is shown in drawing 1T-5961. The functions of the major components are described as follows:

- 3.1 Rotating Faces: The rotating face is a gasket mounted, pin driven component made of carbon graphite. The front surface is precision lapped and polished and mated against the stationary face to provide the dynamic sealing interface. Static sealing of the rotating face to the retaining and locating component is provided by the rotating face seat gasket. The turning force is transmitted from the shaft by the rotating face drive pin.
- 3.2 Stationary Face: The stationary face has a stainless steel body with a hard metallic overlay which is precision lapped and polished to provide the sealing surface with the rotating face.
- 3.3 U-Cup: The U-Cup is an elastomer gasket which provides static sealing of stationary face.
- 3.4 U-Cup Follower: The U-Cup follower is a stainless steel component which maintains the position of the U-Cup gasket within the stationary face and also transmits the spring load to the stationary face.
- 3.5 Stationary Face Retainer: The stationary face retainer is a stainless steel component which axially holds the stationary face during assembly and disassembly operations. After the assembly is complete, the retainer serves to lock the stationary face against rotation through the stationary face drive key.
- 3.6 Coil Spring: Coil springs are used to maintain interface contact under zero pressure conditions and also to supplement the interface loading by hydraulic pressure.

4.0 BASIC DESIGN CONSIDERATIONS

The basic design configuration and features of the shaft seal are a result of experience and application of mechanical shaft seals in both the laboratory and in the field. The primary design considerations are described as follows:

4.1 Rotating Face

- 4.1.1 The rotating face is made of carbon graphite and is designed to implement the "balance" of the shaft seal.
- 4.1.2 Rotating the component made of carbon provides greater dissipation of heat generated by the seals than if the carbon were the stationary face.
- 4.1.3 The compressive strength of the rotating face is more than adequate for the oil pressures maintained in the seal cavities. The face is designed for applications up to 1000 psi in pressure.
- 4.1.4 Since the carbon sealing face area is important in maintaining the seal balance and the carbon material is subject to wear over a period of time, the length of the face section is significant in establishing the service life of the seal. The initial face length is designed to be 0.10 of an inch and the allowable value of total wear is .05 of an inch. The maximum allowable average wear rate for 20,000 hours or 2-1/4 years is 2.5×10^{-6} inches per hour.
- 4.1.5 Because the seal is designed to provide a low leakage, the sealing surfaces are subjected to a marginal lubricating film. This condition requires that the wearing member be made of a material which can maintain its integrity under this boundary lubricating condition. Experience has shown that carbon graphite satisfies this requirement best, so that this selection was made for the rotating face.
- 4.1.6 Besides being used in many other mechanical seal applications (including shaft seals in all Byron Jackson primary reactor coolant recirculating pumps), carbon material is used as the rotating face for the oil seal in the liquid sodium recirculating pumps supplied to Kawasaki Heavy Industries (K.H.I.). The seals in this application were tested and qualified in the Byron Jackson Seal Testing Laboratory.
- 4.1.7 General physical properties of the carbon material are shown in the Appendix.

- 4.1.8 No problems are anticipated with the carbon operating with some contact with the argon gas. The properly vented and pressurized seal depends on the oil for its lubricating film so that the exposure to the argon occurring on external areas does not affect seal operation. Because of the inert nature of the argon the carbon is not susceptible to chemical or corrosive attack by the gas.
- 4.1.9 The flatness and finish of the sealing surface is very important in the performance of the seal. Flatness of the surface is obtained by lapping and is specified to be two helium light bands total or less. The surface finish is specified to be 10 microinches, root mean square (RMS).

4.2 Stationary Face

- 4.2.1 The stationary face is mounted on the balance diameter and is designed to load against the rotating face by the action of the hydraulic force and the coil springs.
- 4.2.2 To facilitate the assembly and disassembly operations, the stationary face is locked axially against the coil springs by the action of the stationary face retainer as it is fastened to the flange. Installation of the rotating face completes the loading of the springs, as the stationary face is pushed back and the springs are compressed to the design working length.
- 4.2.3 The springs permit the stationary face to experience axial shaft movements and still maintain the sealing interface with the rotating face.
- 4.2.4 The stationary face is held against rotation by the stationary face drive keys which are installed in the stationary face retainer and fit into axially located keyways on the outer diameter of the stationary face.
- 4.2.5 The hard surface overlay that comprises the dynamic sealing surface is Stellite No. 1. This material is commonly used in shaft sealing applications and is used in the KHI oil seal and in the oil seal for the LMFBR sodium pump at the Atomics International Liquid Metal Engineering Center Pump Seal Test Facilities.
- 4.2.6 The hard surfaced sealing area requires a precision lapped finish with a flatness within two helium light bands and a finish of 5 microinches (RMS).

WEAR RATE - 10^{-6} IN/HR

CARBON FACE VS. LENGTH OF OPERATION
WEAR RATE

FIGURE 1

4.3 Coil Springs

- 4.3.1 A multiple number of coil springs are used to provide a pressure of 15 to 20 psi against the rotating face.
- 4.3.2 The spring material is Inconel X-750 and material stress analysis is calculated using normal spring formulae. (See Appendix).

5.0 ANTICIPATED SEAL LIFE AND WEAR RATE

- 5.1 The life of the shaft seal depends primarily on the face area remaining on the carbon member and on the condition of the sealing interface so that leakage is maintained within acceptable values.

5.2 Wear Rate

- 5.2.1 Significant wear is generally associated with one or a combination of the following conditions:
 - 5.2.1.1 Initial run-in period.
 - 5.2.1.2 Number of start-ups and start-up conditions.
 - 5.2.1.3 Change in sealing surface height, flatness and wear pattern from normal wear and tear under prolonged operation.
 - 5.2.1.4 Abrasive solid foreign material in the sealed fluid.
 - 5.2.1.5 Operation of seals in a gaseous environment (improper venting of seal cavity).
 - 5.2.1.6 Abnormal operating conditions (pressure, and temperature, beyond specified values).
- 5.2.2 Correct operating procedures can eliminate the effects of the wearing conditions of Paragraphs 5.2.1.4 and 5.2.1.5. Control of abnormal operating conditions is difficult as they may occur unexpectedly. The remaining conditions contribute to the normal wear throughout the total operating period.
- 5.2.3 Factory tests have shown that immediately after initial start-up and for the subsequent period of about 50 hours, wear measurements are inconsistent, although generally being a relatively high value. This condition is a result of the initial surface characteristics and the wearing away of surface irregularities. After this primary run-in period the wear becomes more stable and the average wear rate, being a function of time, decreases as the period of operation progresses.

- 5.2.4 Calculated values of average wear rate and time of operation of carbon face oil seals for the LMFBR coolant pumps have provided data to generate the curve shown in Figure 1. This data was obtained from factory tests of mechanical shaft seals under various conditions of pressure, temperature, and speed.
- 5.2.5 After approximately 500 hours of operation the wear rate becomes nearly constant at 1.0×10^{-6} inches per hour. Sufficient test evidence has substantiated this wear rate up to 1000 hours of operation and one test has shown that this wear rate was valid to nearly 3000 hours.
- 5.2.6 The extrapolation of the wear rate to determine seal life for 10,000 hours or more may not be entirely valid, although any significant acceleration in wear rate is not anticipated until a large portion of the face length has been worn off because of normal operation. This assumption is based upon test data being substantiated by actual operation in field units where inspection of shaft seals with over 20,000 hours of operation have shown satisfactory face patterns with remaining face lengths sufficient to provide continued operation.
- 5.2.7 The two carbon faces tested for the KHI oil seal under similar operating conditions as the LSBRP oil seal experienced wear rates of 0.2×10^{-6} and 0.6×10^{-6} inches per hour over a test period of 629 hours.
- 5.2.8 Based on the wear rate values of the KHI oil seal and of other shaft seals, it is anticipated that the wear rate for the LSBRP oil seal will be less than the maximum allowable average wear rate of 2.5×10^{-6} inches per hour (Para. 4.1.4) and seal life will be over 20,000 hours or 2-1/4 years.

6.0 SEAL THEORY

- 6.1 Face type mechanical seals, regardless of how they are constructed, are generally classified as unbalanced, where hydraulic pressure acts against the entire seal face, or balanced, where hydraulic pressure acts only on part of the seal face.

6.2 The Unbalanced Seal

- 6.2.1 A typical unbalanced seal configuration is shown in Figure 2. Note that the stationary face is gasketed to the flange and the rotating face is gasketed to the shaft with a spring load to hold the faces together at zero pressure and to prevent separation resulting from shaft end play and vibration.

UNBALANCED SEAL CONFIGURATION

FIG. 2

PRESSURE ACTING ON UNBALANCED SEAL FACE

FIG. 3

PRESSURE GRADIENT ON UNBALANCED SEAL FACE

FIG. 4

OVERBALANCED SEAL

FIG. 5

PRESSURE GRADIENT ON OVERBALANCED SEAL

FIG. 6

BALANCED SEAL WITH PRESSURE GRADIENT

FIG. 7

- 6.2.2 To consider the effects of hydraulic pressure, the seal configuration is reduced to Figure 3 in which the rotating face gasket and spring are removed. Note that the hydraulic pressure, P , acts directly against and across the entire sealing surface. This pressure forces the rotating face against the stationary face. To prevent damage to the faces from excessive heat generation, a lubricating film of fluid must be maintained between the faces. This film enters between the faces at the hydraulic pressure and exits at some lower pressure P' .
- 6.2.3 The lubricating film produces a pressure gradient across the faces which tends to separate the faces. As shown in Figure 4, the gradient exerts considerably less force than the opposing hydraulic pressure. A net force acting to close the face exists and this force increases as the hydraulic pressure increases. If the hydraulic pressure becomes too great, the lubricating film will not form and excessive heat will cause damage to the surfaces and the seal will fail.
- 6.2.4 Generally, unbalanced shaft seals perform satisfactorily up to a pressure differential ($P - P'$) of 100 - 150 psi. Other factors such as type of sealed fluid, rotational speed, face materials, and temperature of sealed fluid may vary the pressure limits.

6.3 The Balanced Seal

- 6.3.1 The efficiency and life of a face type mechanical seal is greatly increased by reducing the effect of the hydraulic loading force on the sealing surface. This reduction is achieved by incorporating a design feature known as seal balance.
- 6.3.2 Consideration is first made of a seal which is overbalanced and is shown in Figure 5. Once again the rotating face gasket and the spring load have been deleted to show only the pressures. Note that a shoulder of diameter "A" has been included on the shaft. Diameter "A" is known as the balance diameter. In this overbalanced seal the hydraulic pressure exerts a load against the rotating face in exactly the same manner as it did on the unbalanced seal. The pressure, however, also acts on the opposite end of the face such that the area beyond the balance diameter "A" is in hydraulic equilibrium.
- 6.3.3 A pressure gradient from P to P' is formed as shown in Figure 6. The net result is a separating force, provided that the spring load is less than the force developed by the gradient. This seal configuration is not acceptable because it usually results in excessive leakage.

- 6.3.4 With balance diameter "A" maintained, a condition can be created to counteract the effects of the pressure gradient. An increase of the outside diameter (O.D.) of the stationary face beyond diameter "A" will provide an area which continues to be loaded by the hydraulic pressure. This configuration is shown in Figure 7. Areas beyond the O.D. of the stationary face are in equilibrium, and thus a resulting force acts only upon the area between diameter "A" and the stationary face O.D.
- 6.3.5 An increase in the loading area will tend to reduce the generation of a lubricating film (seal leakage is decreased). A decrease in the loading area will increase the effect of the separating force (seal leakage is increased). With known operating conditions the loading area is designed to provide the desired face pressure such that a lubricating film is maintained thereby achieving long life with satisfactory seal performance.

6.4 The LSBRP Oil Seal

- 6.4.1 Theory substantiated by both field and test experience has shown that a loading area of 60% to 80% of the sealing area provides the optimum combination of leakage and life in most shaft seal applications. The balance design value of the oil seals for the LSBRP pump falls within this range.
- 6.4.2 Note that in the design of the oil seals the balance diameter is located on the stationary member. This design, however, does not change the hydraulic loading effects and was established to achieve the modular concept with the greatest ease of assembly.

7.0 SEAL LEAK RATES

- 7.1 Balanced, face type mechanical seals for nuclear power coolant pump applications are designed for high reliability and long life. To meet this goal, operation with a finite lubricating film between the rotating and stationary faces is essential. This film must be assured throughout reasonably predictable operating conditions. The balanced face design provides the primary feature in achieving a lubricating film.
- 7.2 The presence of a finite film between the faces implies finite seal leakage. This leakage may be of such low value that it is not measurable without special apparatus and procedures. The term "zero leakage", therefore, indicates that the leakage is too small to be measured or visually observed.
- 7.3 Measurable leakage occurs and varies under many influencing factors as any condition which affects the development of the seal pressure gradient can significantly change the rate of seal leakage.
- 7.4 Primary factors which influence seal leakage are pressure, temperature and speed. Generally, leakage is less at low pressure, high temperature, and low speed. Transient conditions of pressure, temperature and speed may also result in variations of seal leakage. Other factors which may affect leakage are axial and radial shaft movement, seal face eccentricity, and seal face condition - flatness, and surface finish.
- 7.5 Leakage from seals operating under conditions of the LSBRP is in the range of zero to $10.0 \text{ cm}^3/\text{hr}$.
- 7.6 Actual leak rate predictions specifically for the LSBRP oil seal would have to be established from a full-scale component evaluation test of the seal.

8.0 SHAFT SEAL LUBRICANT AND OPERATING TEMPERATURES

- 8.1 The recommended lubricant for the shaft seal is Mobil SHC 824. This selection is based upon the lubricant vendor's recommendation from their testing of lubricants under conditions similar to those of the LSBRP primary pump oil seal. A test program of the seal should include operation with this oil to verify its ability to provide satisfactory lubrication under the operating conditions.
- 8.2 The recommended normal operating temperature is 125°F. or less. The maximum acceptable normal operating temperature is 150°F. These temperatures are recommended for sustained operation in order to achieve the minimum service period of 20,000 hours.
- 8.3 It should be noted that the lower the oil operating temperature, the greater will be the reliability of the seal. It is imperative, therefore, that the oil cooling system be capable of maintaining the oil temperature at or below 150°F. throughout all phases of normal operation.
- 8.4 Temperatures above 150°F. will not cause immediate failure, but the seal life will be significantly decreased by continuous operation at elevated temperatures.
- 8.5 The maximum allowable oil temperature is 250°F. based on the limitation of the elastomer gaskets. Immediate failure is not likely at this temperature, although a controlled shutdown and inspection of the shaft seals is recommended if this temperature is achieved.

APPENDIX

UNLESS TOLERANCE IS GIVEN
MAINTAIN TO WITHIN

DIAMETERS

BORE $\pm 1/64$ TURN $- 1/64$

LENGTH

0 TO 2 FT. $- 1/64$ 2 TO 6 FT. $- 1/32$ 6 FT. AND OVER $- 1/16$

LTR

DESCRIPTION

DATE

APP'D

ORIGINAL

STATIONARY FACE
DRIVE KEY

THIS MATERIAL IS THE PROPERTY OF BORG-WARNER CORPORATION AND IS FURNISHED FOR THE PURPOSE INDICATED. ANY AND ALL CONFIDENTIAL, PROPRIETARY, PATENT AND OTHER RIGHTS, IN THE SUBJECT MATTER BEING RETAINED INCLUDING ANY EXCLUSIVE RIGHTS OF USE AND/OR MANUFACTURE AND/OR SALE. POSSESSION OF THIS MATERIAL DOES NOT CONVEY ANY PERMISSION TO REPRODUCE THIS MATERIAL, IN WHOLE OR IN PART, OR MANUFACTURE THE SUBJECT MATTER SHOWN THEREIN, OR USE THE CONFIDENTIAL OR PROPRIETARY INFORMATION THEREON, SUCH PERMISSION TO BE GRANTED ONLY BY SPECIFIC AUTHORIZATION IN WRITING SIGNED BY AN OFFICER OR OTHER AUTHORIZED AGENT OF BORG-WARNER CORPORATION, ITS DIVISIONS OR SUBSIDIARIES.

UNLESS OTHERWISE SPECIFIED

DIMENSIONS ARE IN INCHES

DECIMAL TOL.

ANGLE TOL.

X $\pm$ $\pm$ XX $\pm$ XXX $\pm$ SURFACE ROUGHNESS
PER MIL - STD 10DRILLED HOLES ∇ AS DRILLED
TAPPED HOLES ∇ AS TAPPEDDO NOT SCALE DRAWING.
REMOVE ALL BURRS AND SHARP EDGES
EQUIVALENT TO .005/.015
ALL MACH. CORNERS/FILLETS .005/.020.

BJ ORDER NO.

REF. NO.

DRAWN BY

DATE

CHECKED BY

DATE

CONTRACT NO.

DESIGN APP'D.

DATE

CUSTOMER APP'D.

DATE

Byron Jackson Pump Division
BORG-WARNER CORPORATION
LOS ANGELES DIVISION

DRAWING TITLE

SEAL ASSEMBLY

CODE IDENT. NO.

DRAWING NO.

1T-5961

SCALE NONE

WEIGHT

SHEET

STRESS CALCULATIONS *

C_1 , spring index
 d , wire diameter
 G , modulus of elasticity in shear
 k , stress concentration factor due to curvature and direct shear
 k_c , stress concentration factor due to curvature
 k_s , stress multiplication factor due to direct shear
 K , spring rate
 l_1 , free length of spring
 N , number of active coils
 P , load
 R , mean radius of helix
 s_s , shearing stress

GIVEN: Spring O.D. = 0.625 in., $d = 0.059$ in., $G = 11 \times 10^6$ psi
 $l_1 = 2.055$ in., $N = 11.5$, Working length = 1.125 in., Material = Inconel X-750,
 Solid height = 0.797 in.

To FIND: ① Spring stress @ solid height
 ② Spring stress @ working length

$$\text{Spring rate, } k = \frac{d^4 G}{64 R^3 N}$$

$$\begin{aligned}
 d^4 &= (0.059)^4 = 1.212 \times 10^{-5} \\
 \text{Spring mean diameter} &= 0.625 - 0.059 = 0.566 \text{ in.} \\
 R &= 0.566 \div 2 = 0.283 \text{ in.} \\
 R^3 &= 0.0227 \text{ in}^3
 \end{aligned}$$

$$k = \frac{1.212 \times 10^{-5} \times 11 \times 10^6}{64 \times 0.0227 \times 11.5}$$

$$k = 7.98 \text{ lb/in}$$

$$\text{① Load at solid height, } P_{SN} = (2.055 - 0.797) 7.98 = 10.04 \text{ lb}$$

$$\text{Spring stress at solid height, } s_{sSN} = \frac{k_s 16 P_{SN} R}{\pi d^3}$$

$$d^3 = (0.059)^3 = 2.054 \times 10^{-4} \text{ in}^3$$

$$k_s = \frac{k}{k_c} \text{ where } k_c = \frac{4C_1 - 1}{4C_1 - 4}$$

$$k = k_c + 0.615/c,$$

$$C = \frac{2R}{d} = \frac{2 \times 0.283}{0.059} = 9.593$$

$$k_c = \frac{4(9.593) - 1}{4(9.593) - 4} = \frac{37.37}{34.37} = 1.087$$

$$k = 1.087 + \frac{0.615}{9.593} = 1.151$$

$$k_s = \frac{1.151}{1.087} = 1.059$$

$$S_{S.H.} = \frac{1.059 \times 16 \times 10.04 \times 0.283}{\pi \times 2.054 \times 10^{-4}}$$

$$S_{S.H.} = \underline{\underline{74,600 \text{ psi}}}$$

② Load at working length, $P_{WL} = (2.055 - 1.125) 7.98 = 7.42.16$

Spring stress at
 working length, $S_{W.L.} = \frac{k_s 16 P_{WL} R}{\pi d^3}$

$$= \frac{1.059 \times 16 \times 7.42 \times 0.283}{\pi \times 2.054 \times 10^{-4}}$$

$$S_{W.L.} = \underline{\underline{55,100 \text{ psi}}}$$

The minimum elastic limit and maximum solid stress for Inconel compression springs made with 0.059 of an inch wire diameter is 90,000 psi. The allowable working stress for severe service is 54,000 psi. *

The calculated solid height stress is within the limit of allowable stress for the solid height and the working stressing capability is near the severe duty stress, although the actual spring service is more closely in the region of average service.

* From Machinery's Handbook, 19th Edition, Industrial Press, New York, N.Y., 1973

7. SHAFT SEAL HEAT EXCHANGER

1.0 OBJECT

The purpose of this study is to provide an estimate of the heat load resulting in the shaft seal assembly of the LMFBR coolant pump for the LSBRP and to determine the fluid flow requirements to maintain acceptable levels of temperature for the operation of the oil lubricated mechanical shaft seals.

2.0 DISCUSSION

2.1 Shaft Seal Design Description: A detailed discussion of the seal conceptual design is given in Part 1.

2.2 Shaft Seal Heat Load

2.2.1 The calculations of the shaft seal heat load are shown in Paragraphs 1.0 and 2.0 of the Appendix.

2.2.2 A basic heat balance diagram was initially established to determine the primary sources of heat input and areas of heat rejection. The major source of heat is anticipated to be the result of friction at the seal faces. Additional sources of heat which were included in the final heat load were energy losses from the oil recirculation pumping ring and heat conduction along the shaft from the liquid metal. The sum of the heat sources, therefore, becomes the total heat load which must be removed by the heat exchanger.

2.2.3 Since the friction effects are reflected in the torque values of mechanical seals, the heat load from friction of the LSBRP seal was based upon known torque values of existing seals. Extrapolation of this data provided the necessary torque figure to determine the heat load occurring at the seal faces.

2.2.4 The assumption was made that 15% of the total heat load was a result of additional sources, so that a final estimated value of 36,940 Btu per hour was developed. For design purposes a value of 40,000 Btu per hour was used.

2.3 Fluid Flow Requirements

- 2.3.1 In this conceptual design, cooling of the lubricating oil is achieved as shown in the Drawing No. 1C-3564 of Part 1. The oil is circulated past and around vertically located tubes mounted around the seal housing. Ambient air is forced through the tubes with heat being transferred from the oil to the air, which is rejected into the surrounding atmosphere.
- 2.3.2 Requirements of oil and air flows are shown in the calculations of Paragraph 3.0 of the Appendix.
- 2.3.3 A diagram of temperatures occurring in the two fluids in the heat exchanger was initially established. This diagram was based on the assumption that the oil enters the heat exchanger at 145°F and exits at 140°F (to the seal cavity), while the cooling air, moving counter to the oil flow, enters at 100°F and exits at 105°F.
- 2.3.4 For the established heat load of 40,000 Btu per hour and with the assumed temperature differentials, a flow of 35.2 gpm is required for the oil and 7835 cfm is required for the air.

3.0 CONCLUSIONS AND RECOMMENDATIONS

3.1 Heat Exchanger Design

- 3.1.1 The heat exchanger as shown in 1C-3564 utilizes a pumping ring to circulate the oil past tubes through which air is forced by a fan mounted on the pump shaft. The major advantage of this system is that the cooling system is always operating when the pump is in operation.
- 3.1.2 The relatively slow speed (600 rpm) of the pump may present problems in achieving the necessary capacities from the shaft-mounted pumping ring and fan. Under pony motor operation the effectiveness of the pumping ring and fan are further diminished, although the heat load from the seals is also decreased.
- 3.1.3 To overcome problems in developing adequate fluid flows at low speed, an alternate arrangement is to use an external blower and an oil pump instead of the shaft mounted fan and pumping ring. This design was used for the seals in the Enrico Fermi LMFBR pump. These external units would provide continuous flows throughout all ranges of shaft speed. The failure of the oil pump or blower would cause loss of cooling to the seals.

- 3.1.4 Utilization of both systems (integral fan and pumping ring combined with external blower and pump) would guarantee sufficient cooling throughout all speeds and during loss of one or both of the external cooling components.
- 3.1.5 Another area of consideration is that for the calculated heat load and with the type of heat exchanger system a large surface area is required in order to maintain acceptable levels of operating temperature. The surface area is directly a function of the number and size of tubes. The required number of tubes will result in a seal cartridge of significant size both in weight and volume.
- 3.1.6 Some reduction in size is possible by using tubes with more effective heat transfer characteristics. Tubes that merit consideration are those having a spiral fluted configuration or tubes having material on the outer diameter raised to provide a spine-like configuration.
- 3.1.7 Even by increasing the effectiveness of the tubes the seal cartridge is anticipated to still be of a size that some problems may be encountered during the installation operations. A further reduction in size could be accomplished by replacing the fan and the tube section with an externally mounted oil cooling heat exchanger system. For the calculated heat load and required operating temperatures heat exchangers for this application are commercially available.

3.2 Future Study and Development

- 3.2.1 The basic cooling concept of extracting the heat load with forced air has been used in both the Enrico Fermi and Kawasaki Heavy Industries LMFBR coolant pumps. This design is also planned for the recirculating coolant pump of the Clinch River Breeder Reactor Plant (CRBRP). The concept is considered to be acceptable for the LSBRP, but the actual means to implement the design requires further evaluation.
- 3.2.2 Results and data from the performance of the seals in the CRBRP will be a valuable source for establishing the design of the seal and heat exchanger components for the LSBRP.
- 3.2.3 Because no data is available from a seal of the size and design anticipated for the LSBRP unit, development and testing in the area of lubricant cooling requirements are recommended to determine fan performance, oil pumping ring performance, and oil and air heat transfer characteristics.

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APPENDIX

BY LIT DATE 21 MAY 76
CHKD. BY _____ DATE _____

SUBJECT LSBRP
1.0 SHAFT SEAL HEAT LOAD
(BASIC HEAT BALANCE)

APPENDIX
SHEET NO. 1 OF 5
JOB NO. 761-G-0002

1.0 SHAFT SEAL HEAT LOAD

1.1 Basic Heat Balance

Requirements of heat exchanger design to maintain acceptable levels of operating temperature of the shaft seal lubricating oil are based upon the heat balance as shown below:

$$\text{Heat balance} = 0 = Q_{SEAL} + Q_{SHAFT} \pm Q_{SURROUNDING} - Q_{COOLING}$$

$$Q_{COOLING} = Q_{SEAL} + Q_{SHAFT} \pm Q_{SURROUNDING}$$

$Q_{COOLING}$, therefore, is the heat load which the heat exchanger system must be capable of eliminating in order to maintain acceptable operating temperatures.

Q_{SEAL} is the heat generated within the seal cartridge and consists primarily of the heat developed at the seating interfaces. If an internal pumping ring is used to circulate the oil, this component and turbulence of the oil contributes to Q_{SEAL} .

Q_{SHAFT} is the heat conducted through the shaft from the liquid metal.

$Q_{SURROUNDING}$ is either heat gained or heat loss, depending upon the temperature of the ambient air and adjacent components.

2.0 SHAFT SEAL HEAT LOAD

2.1 Torque of 14.5 inch LSBRP Seal

The torque from face type mechanical seals of the size (14.5 inches at the balance diameter) and configuration designed for the LSBRP pump is estimated to be 108 ft-lb

2.2 Using 108 ft-lb of torque, the heat load, P , in horsepower is denoted by $P = \frac{TN}{5250}$

where T = torque in ft-lb

N = speed in rpm

= 600 rpm for LSBRP pump

$$P = \frac{108 \times 600}{5250} = 12.343 \text{ hp}$$

or, heat load, Q , in Btu/min = $P \times 42.42$

$$Q = 12.343 \times 42.42 = 523.59 \text{ Btu/min}$$

$$\underline{Q = 31,400 \text{ Btu/hr}}$$

2.3 Additional Heat Sources

The primary source of heat is expected to be generated from the torque effects at the sealing interfaces. This heat load was evaluated in paragraphs 2.1 and 2.2.

Heat input may also result from the turbulence of the oil and from energy losses of an internal pumping ring if this component is used to circulate the oil.

The liquid metal is another heat source, although the LMEC Report TDR-71-3 of the Fermi-type seal indicated that no significant increase in temperature was observed in the seal area as a result of heat conduction along the shaft from the liquid sodium (Table 7, seal annulus temperature at 145°F @ zero r.p.m. with 1200°F sodium - TDR-71-3).

2.0 SHAFT SEAL HEAT LOAD (CONT'D)2.3 Additional Heat Sources (cont'd)

The assumption is made that the ambient air and adjacent components will be at temperatures lower than the oil temperatures so that energy flow will be in the form of heat rejection (-Q). (This rejection is assumed to be negligible compared to the total heat load).

The total heat input, in addition to the seal interface heat, then occurs from oil turbulence, pumping ring losses, and heat conduction along the shaft from the liquid metal. For purposes of this design concept the value of additional heat is estimated at 15% of the total heat load. Thus, the cooling requirement, $Q_{\text{COOLING}} = Q_{\text{SEAL}} + 0.15 Q_{\text{COOLING}}$

$$Q_{\text{COOLING}} = \frac{31,400}{0.85}$$

$$Q_{\text{COOLING}} = 36,940 \text{ Btu/hr}$$

Use 40,000 Btu/hr

BY LIT DATE 8 JUN 76SUBJECT LSBFESHEET NO 4 OF 5CHKD. BY DATE3.0 SHAFT SEAL COOLINGJOB NO 761-G-0002FLUID FLOW REQUIREMENTS3.0 SHAFT SEAL COOLING- FLUID FLOW REQUIREMENTS

3.1 Assumptions

Heat exchanger fluid temperatures occur as shown in the temperature diagram below.

This figure is based upon maintaining the normal operating temperature of the lubricating oil at a maximum of 150°F with an ambient air temperature of 100°F.

3.2 Oil Flow Required

Required heat loss from oil

$$Q = m C \Delta t$$

$$m = \frac{Q}{C \Delta t}$$

Q = heat load = 40,000 Btu/hr

m = oil flow, lb/hr

C = specific heat, Btu/lb-F

Δt = temperature difference, °F

$$\text{Specific heat, } C = \frac{1}{5} (0.368 + 0.00045t) \quad (\text{Kent's M.E. Hdbk})$$

$$C = \frac{1}{0.835} [0.368 + (0.00045 \times 142.5)] \quad \begin{array}{l} S = \text{Spec. Grav.} = 0.835 \text{ for} \\ \text{Mobil SHC-824 (Mobil} \\ \text{Data Sheet)} \end{array}$$

$$C = 0.541 \frac{\text{Btu}}{\text{lb-F}}$$

t = temperature, °F
(Use avg oil temp.)

BY LIT DATE 8 Jun 76 SUBJECT LSBRP SHEET NO. 5 OF 5
 CHKD. BY _____ DATE _____ 0 SHAFT SEAL COOLING - JOB NO. 761-G-0002
FLUID FLOW REQUIREMENTS

3.0 SHAFT SEAL COOLING - FLUID FLOW REQUIREMENTS (CONT'D)

3.2 Oil Flow Requirement (cont'd)

$$m = \frac{40,000}{0.541 (145-140)} = \underline{\underline{14,790 \text{ lb/hr}}}$$

Oil Flow in gpm:

$$\text{Sp. Gr. @ } 142.5^\circ\text{F} = 0.840$$

(Fig. I-4-5 Byr. Jack. Eng.)
Manual

$$\text{Density} = 0.840 \times 62.4 = 52.416 \frac{\text{lb}}{\text{ft}^3}$$

$$\text{Flow} = 14,790 \frac{\text{lb}}{\text{hr}} \times \frac{\text{ft}^3}{52.416 \text{ lb}} \times 7.481 \frac{\text{gal}}{\text{ft}^3} \times \frac{\text{hr}}{60 \text{ min}}$$

$$\underline{\underline{\text{Flow} = 35.2 \text{ gpm}}}$$

3.3 Air Flow Required

$$m = \frac{Q}{C_p \Delta t}$$

C_p = specific heat = 0.240 Btu/lb-F
(Mech. Eng. Laboratory, Doolittle)

$$= \frac{40,000}{0.240 \times (105-100)}$$

$$m = \underline{\underline{33,333 \text{ lb/hr}}}$$

Flow in cfm

$$\text{Density} = 0.0709 \text{ lb/ft}^3$$

(Mech. Eng. Laboratory, Doolittle)

$$\text{Flow} = 33,333 \frac{\text{lb}}{\text{hr}} \times \frac{\text{ft}^3}{0.0709 \text{ lb}} \times \frac{\text{hr}}{60 \text{ min}}$$

$$\underline{\underline{\text{Flow} = 7835 \text{ cfm}}}$$

8.0 TANK SODIUM LEVEL CONTROL

The liquid sodium level in the pump has to be controlled to a predetermined level. If the level variations are greater than those specified for design, damage to pump components may occur. An abnormally high level could damage the pump shaft seal and radiation shield material by high temperature and sodium contamination. An abnormally low level could result in poor hydraulic conditions at the pump suction and severe temperature gradients in components normally submerged in liquid sodium. This level has to be controlled for both static and running conditions. A safe level at elevation of -10 ft. could be chosen for the desired sodium level which is also the elevation of the overflow outlet. For static condition, a cover gas with gage pressure equal to 10 feet of liquid sodium will be applied on top of the sodium to keep it at the design level. This is because the sodium level in the reactor is at zero feet elevation. For running condition, the situation is somewhat more complex. First, at the impeller suction located at elevation -40 feet, the NPSH is 20 feet. That makes the absolute zero pressure point at -20 feet elevation. This is also the location of the discharge column tapered static seal fit. Therefore, if the sodium level is at 10 feet elevation, and the cover gas remains the same as in the static condition, there will be a total of 60 feet of head, 10 feet of elevation and 50 feet of gas plus atmospheric pressure, pushing the liquid sodium through this tapered fit down to suction. With the known clearance and the length of the tapered fit the flow down to suction could be calculated.

Therefore, the total in-flow leakage from the discharge nozzle and the hydrostatic bearing will be regulated to be higher than the down flow. The liquid will be maintained at the design level with the extra flow going out through the overflow outlet.

STEPHEN PATKAY AND ASSOCIATES

MANAGEMENT CONSULTANTS AND INDUSTRIAL ENGINEERS SINCE 1932

9550 FLAIR DRIVE EL MONTE, CALIFORNIA 91733
TELEPHONE 213-686-0900

June 7, 1976

Mr. Don Plapp
Project Manager - AEF .
Byron Jackson Pump Division
Borg-Warner Corporation
2300 East Vernon Avenue
Vernon, California 90058

Dear Mr. Plapp:

The Manufacturability Analysis of the Conceptual Design of a Large Scale LMFBR Pump has been completed as scheduled.

Our report, including the Manufacturing Plan and other related items is enclosed.

As will be seen, the recommended manufacturing tolerances are somewhat "tighter" than those defined for the "CLINCH RIVER" Project. We re-emphasize that our recommendations reflect machine tool capabilities when reasonable disciplines and controls are exercised. They are not intended to reflect functional requirements.

We wish to thank Messrs. Wienstien and Wong as well as the other members of your AEF staff for their fine cooperation and assistance which enabled us to complete the assignment in such a short time period.

Respectfully submitted,

STEPHEN PATKAY AND ASSOCIATES

W. E. Boyer, Chief Engineer

1b

BYRON JACKSON - PUMP DIVISION

Vernon, California

Final Report On:

MANUFACTURABILITY OF LARGE
SCALE BREEDER PUMP

Prepared By:

STEPHEN PATKAY AND ASSOCIATES
El Monte, California

June, 1976

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1.0 SUMMARY

A critical review of all conceptual drawings provided has been completed. These analyses were made to determine on one hand; the manufacturability of the conceptual design, and on the other, the development of a basic production plan which would enable the design to be manufactured with a high level of quality assurance.

Our findings and recommendations are summarized below:

1.1 MANUFACTURABILITY

Although pump size, ie, overall length and weight will warrant special consideration throughout the Fabrication, Machining and Assembly processes, the conceptual design as defined on Drawing No. 1F-7937-A can be manufactured with a high degree of quality assurance using equipment and technology available to industry today.

1.2 DESIGN PARAMETERS

Design Parameters outlined in Section 2.0 of this report are recommended to improve producibility and ease of manufacture. All recommendations have been developed recognizing; available equipment, ease of handling, and the practicality of manufacturing's abilities to satisfy ASME and RDT specifications under controlled conditions in a normal production environment.

1.3 MANUFACTURING PLAN

The Manufacturing Plan as outlined in Section 3.0 defines the basic processes required for Fabrication, Machining and Assembly.

It must be pointed out, that this plan was developed from conceptual drawings and must be updated when final engineered drawings are available.

We do feel, however, that under the plan as outlined the unit can be manufactured with a high degree of quality assurance, provided the final engineering design does not vary drastically from the conceptual design.

The plan has also been developed assuming that the recommendations outlined in Section 2.0 will be incorporated in the final design, and that the facilities as defined in Section 4.0 will be provided.

1.0 SUMMARY, cont.

1.4 FACILITY REQUIREMENTS

The statement made earlier, that "the conceptual design can be manufactured" was not intended to imply that the unit can be manufactured with the existing Vernon Facilities and equipment.

A substantial capital investment will be required to upgrade existing equipment and facilities in order to conform with ASME and RDT specifications as well as providing the capacity for the manufacture of such a large scale pump.

Estimated cost for upgrading equipment and facilities as outlined in Section 4.0 are approximately \$10 Million.

NOTE: The above are estimates only and are based on our best knowledge of 1976 prices.

1.0 SUMMARY, cont.

1.5 AREAS OF FURTHER DEVELOPMENT

Recognizing that any actual manufacture of the conceptual design will be somewhat in the distant future, we have outlined (in Section 5.0 of this report) basic areas of design and manufacturing which warrant further development.

These areas of development, although not mandatory for producing a large scale breeder pump, should be considered as the existing "state of the art" progresses and new technologies are made available.

1.6 ESTIMATED MANUFACTURING CYCLE

Although somewhat beyond the scope of a manufacturability study, we have been requested to provide our estimates for the manufacturing time requirements.

This is not a task that can be performed with a great degree of confidence without the benefit of engineered drawings, defined quality control procedures, etc. However, our best "Guess-ta-mate" based on our past experience in the pump industry, and our intimate knowledge of Byron Jackson operations would be about a 2 to 2½ year flow time from receipt of raw material through the final performance test.

The above estimate is a manufacturing flow time only, and is based on the normal product mix and shipping schedules we have witnessed during our consulting relationship at the Vernon facilities.

It does not include time for:

- Development of:
 - Welding Procedures
 - Test Procedures
 - Q.A., Q.C. Procedures
- Process Planning
- Lead Time for Material Procurement
- Upgrading of Facilities and Equipment

Again, the above time estimate assumes that the majority of the recommendations outlined in Sections 2.0, 3.0 and 4.0 will be incorporated and/or implemented.

1.0 SUMMARY, cont.

1.7 CONCLUSIONS

In conclusion, we re-iterate:

A--The conceptual design can be manufactured using equipment and technology available in industry today.

B--Estimate cost to upgrade the Vernon facilities to handling large scale breeder pumps of this magnitude = \$10,000,000.

C--Estimated manufacturing cycle (flow time) = 2 to 2-1/2 years (24 to 30 months).

2.0 MANUFACTURABILITY

As outlined in Section 1.0, a critical review of the conceptual design as defined on Drawing Number 1F-7937 and 1F-7937, Rev. "A" has been completed.

Even though the present "state of the art" and equipment available on today's market are more than adequate to satisfy the initial design requirements, the recommended design parameters outlined in the following sub sections, if adhered to will assist in keeping manufacturing cost to a practical minimum.

We re-iterate, the conceptual design cannot be manufactured with the existing Vernon facilities. Facility and equipment requirements are defined in Section 4.0 of this report.

2.1 DESIGN PARAMETERS

The following parameters are recommended for ease of manufacturing. They have been established recognizing, available equipment, ease of handling, and the practicality of manufacturing being able to satisfy ASME and R.D.T. specifications under controlled conditions in a normal production environment.

2.1.1 Outer Tank Assembly

- Cylindrical Section Lengths

All barrel sections fabricated from rolled plate should be held to 8 Ft. (96") lengths.

This allows use of standard 96" width plates, which are readily available. This width also allows the use of standard equipment and helps keep material handling problems to a tolerable level.

- Weld Joints

All pressure boundary welds and welds on attachments to same must be of radiograph quality. For ease of "X-ray" interpretations, all weld joints should be double beveled butt joints. "T" joints and/or corner joints of X-ray quality should be eliminated.

Such undesirable joints may be eliminated through the use of machined forgings. All such forgings should have machined weld preps and incorporate straight wall lengths on both sides of the weld joints for adequate access for X-ray film.

2.0 MANUFACTURABILITY, cont.2.1 DESIGN PARAMETERS, cont.2.1.1 Outer Tank Assembly, cont.

Specific components on the tank assembly which should be machined forgings are:

- Item 10-1 Suction Piece
- Item 10-2 Lower Hemisphere Section
- Item 10-4-1 Discharge Port Ring (Barrel Section)
- Item 10-7 Transition Piece
- Item 10-9 Top Flange
- Item 10-5 Discharge Nozzle
- Item 10-6-3 Secondary Discharge (overflow) Nozzle

Overall Sub-Assembly Lengths Requiring Machining

Any sub assembly or cylinder sections, etc., requiring machining, rough machining, or machined weld preps should be kept to a maximum overall length of 14 ft. (168").

The above limitations are recommended recognizing the capacity of standard machine tools. The majority of the machining of tank components will be best accomplished on a Vertical Boring Mill.

Existing equipment now available in house has a working height capacity of 14'-9".

To achieve the capability of machining sub-assemblies greater lengths, the following options are available:

- 1 - Modify existing equipment with riser blocks, extended shafts, etc.

NOTE: This is most undesirable from a production scheduling viewpoint as it would be impractical to leave this machine in a modified state for normal production. This would necessitate a changeover each time a special length section is machined. Such a changeover and back would probably require the unit to be out of service for approximately 5 to 6 shifts (2-3 manufacturing days).

2.0 MANUFACTURABILITY, cont.

2.1 DESIGN PARAMETERS, cont.

2.1.1 Outer Tank Assembly, cont.

- 2 - Purchase a new V.B.M. with an extended rail height. This would most likely be a special order and cost close to \$1,000,000. at 1976 prices.
- 3 - Locate and purchase a used V.B.M. that has an extended rail height. Such a unit will cost between \$200,000. and \$500,000. depending upon condition and accessories.

As can be seen, none of the 3 above options are desirable from an economic point of view. As previously stated, these problems and inflated capital equipment expenditures will be non existent if the overall length of any machined sub-assembly is kept to 14 foot maximum.

2.1.2 Volute Case Assembly

- Basic Design

The basic design of 5 segments lends itself well to manufacturing.

The segmented design offers good access for cleaning of fluid passages and keeps the total weight of any one segment well within the existing capabilities of todays foundries.

Consultation with the Vernon Pattern Shop Manager indicates that no major problems are anticipated.

Some difficulties may be encountered in blocking off the suction nozzles for hydro test. (This problem is discussed in further detail in Section 3.0 of this report.)

Recommendations pertaining to tolerances, surface finish, alignment methods, etc., are covered in Section 2.2 and 2.3.

2.1.3 Rotating Element

The rotating element as defined in this report is composed of the following items:

- Items 7 & 8 -- (2) double suction impellers
- Item 14 -- (1) lower bearing journal
- Item 15 -- (1) upper bearing journal
- (1) intermediate stub shaft
- (1) weld neck type shaft flange

2.0 MANUFACTURABILITY, cont.

2.1 DESIGN PARAMETERS, cont.

2.1.3 Rotating Element, cont.

With this definition, the intermediate and top shafts are not a part of the rotating element.

No major problems in Manufacture are anticipated with the conceptual design.

We do, however, recommend that the bearing journals and the stub shafts be one piece which is machined from a certified forging. This is desirable from a Manufacturing point of view, as it eliminates N.D.T. problems in seal welding a journal sleeve to a stub shaft. On the other hand, however, the ability to replace bearing journals becomes impaired.

2.1.4 Intermediate and Upper Shafts

For ease of welding and non-destructive testing of weld joints, we recommend that all shaft flanges be designed with weld necks rather than the "butt/fillet" approach initially depicted on the first conceptual drawing.

Raw material for shaft tubes is recommended to be seamless tubing and should be purchased with adequate stock to allow for machining on the I.D. and O.D. surfaces without violating minimum wall thickness requirements.

Balancing per ISO/TC 108 DR 1940 specifications are recommended. This will allow for a .005 displacement of uniform mass about a common centerline.

Tolerancing, TIR, etc. are covered in sub-section 2.2.

It may also be desired to divide the intermediate shaft into (2) pieces with another set of flanges that would be accessible in the discharge area for assembly and disassembly. Further detail discussion on this subject as well as the impact on Manufacturing are given in Section 5.0 of this report.

2.0 MANUFACTURABILITY, cont.

2.1 DESIGN PARAMETERS, cont.

2.1.4 Intermediate and Upper Shafts, cont.

The existing design does not specify T.I.R. requirements for both shafts and the rotating element assembly. Also it has been stated that the shafts will be driven by the flange studs. To facilitate a positive driving action and control T.I.R., we recommend the following alternates be considered and incorporated into the design.

The following alternatives are presented in order of preference for ease of Manufacturing.

- 1 - Hold O.D. of flanges concentric to shaft centerline. Utilize (4) body fit studs, or bolts at 90° for positive alignment of shaft centerlines, and facilitate a positive drive between the shaft sections.
- 2 - Utilize (4) drive keys at 90° to facilitate driving power and alignment. In this case the only correction for mis-alignment or excessive T.I.R. would be offset keys.
- 3 - Incorporate male/female rabbet fits at each flange joint specifying a close tolerance, ie $\pm .001$. This approach will increase manufacturing cost appreciably and offer little corrective action for mis-alignment or imbalance conditions.
- 4 - Incorporate male/female rabbet fits at each flange joint, with a semi-close tolerance, ie $\pm .005$ and utilize eccentric studs 4 places at 90° for correcting any mis-alignment.

The first option provides the most economical approach from a Manufacturing point of view.

With this approach, the final sizing of the body fit holes would be machined after the rotating element and shafts have been assembled and aligned. This also allows the mis-alignment at the flange joints to be kept to an absolute minimum thereby assuring good bearing alignment and provides the means of keeping imbalance conditions within an acceptable tolerance.

(Additional detail information is given in Section 3.3)

2.0 MANUFACTURABILITY, cont.

2.1 DESIGN PARAMETERS, cont.

2.1.5 Discharge Column Assembly

2.1.5.1 Lower Column Extension

This component as shown on Drawing Number 1F-7937, Rev. "A" does not present any major problems in fabrication or machining.

- . The recommended method of registering and alignment to mating components are given in Section 2.3.

2.1.5.2 Main Discharge Head Assembly

This component as shown in Drawing Number 1F-7937, Rev. "A" should not cause any problems in manufacture if the recommendations outlined in Section 2.2, 2.3 3.3 are adhered to.

2.1.6 Upper Column Assembly, Item #12

No major problems are anticipated if recommendations presented in Section 1.2 and 1.3 are adhered to.

NOTE: The existing design indicates the section to be approximately 15 Ft. long. As previously stated, maximum overall length for machined sections is 14 Ft. (168"). We recommend that the section be shortened to 168" and the top driver mount assembly be re-engineered to add the 12" difference while still maintaining the insulation box elevation.

2.1.7 Driver Mount Assembly

The latest design on Drawing Number 1F-7937, Rev. "A" does not present any major problems in manufacturing if recommendations presented in sub-sections 2.2 and 2.3 are adhered to.

NOTE: Method of aligning driver to driver mount is assumed to be (4) dowels at 90°.

2.1.8 Insulation Container

No major problems in manufacturing are anticipated.

2.0 MANUFACTURABILITY, cont.

2.1 DESIGN PARAMETERS, cont.

2.1.9 Drive Coupling

Although the conceptual drawing does not show any great detail of the coupling configuration, we do suggest the following parameters for consideration.

- 1 - Both driver and shaft coupling to be machined all over, thereby minimizing imbalance conditions.
- 2 - Alignment between the two pieces be maintained by a close tolerance male/female rabbet fit or a close tolerance keyed taper fit.
- 3 - Both halves be balanced as an assembly per ISO/TC 108 DR 1940.

2.2 RECOMMENDED TOLERANCES

This section presents the recommended tolerance and surface finishes. These recommendations have been developed recognizing; functional requirements, present day technology, and ease of manufacture.

This is not to indicate that closer tolerances cannot be attained with a high degree of quality assurance, however, in doing so manufacturing cost may be increased substantially. Likewise relaxing of manufacturing tolerances may also have an adverse effect on total product cost due to additional assembly requirements.

All recommended tolerances and surface finish requirements are listed below by component and/or sub-assembly.

2.2.1 Outer Tank Assembly

2.2.1.1 Fabrication Tolerance

- overall length of non-machined sections: $\pm 1/8"$ with the anticipated sequence of fabrication, this will present a possible overall length deviation of $1\frac{1}{4}"$.
- roundness and radial mis-match of cylindrical section all tolerances pertaining to the above categories to be in accordance with ASME Section III, Class I, Paragraph NB 4220 and all applicable sub-paragraphs.

2.0 MANUFACTURABILITY, cont.

2.2 RECOMMENDED TOLERANCES, cont.

2.2.1 Outer Tank Assembly, cont.

2.2.1.2 Machining Tolerances

a - Suction, Discharge and Overflow Nozzles:

- Fluid Passage Diameters = $\pm 1/8"$ with 250 RMS finish.

- Machined Weld Preps:

$\pm 1/64$ on diameter and/or wall thickness with 250 RMS finish

$\pm 2^{\circ}-0^{\circ}$ on all angular weld bevels, "J" preps, etc.

b - Tapered Discharge Section (Item 10-4-1):

The tapered diameter should be toleranced $\pm 1/64"$ with basic gage points defined both ways from discharge centerline. Angle or taper per foot should be shown on manufacturing drawing as reference only.

NOTE: It is anticipated that this fit will be machined after final fabrication.

- Surface Finish of Machined Taper Surface = 250 RMS

- Roundness of Finish Machined Taper = .030 inches or equal to fall within diameter tolerance

- Concentricity to Common Centerline Drawn Perpendicular to Top Flange Through Center of Tank to be = .030 inches

2.0 MANUFACTURABILITY, cont.

2.2 RECOMMENDED TOLERANCES, cont.

2.2.1 Outer Tank Assembly, cont.

2.2.1.2 Machining Tolerances, cont.

c - Top Flange and Seal Ring Fit:

- Diameters

- Flange O.D. = $\pm 1/8$
- Seal Ring areas to be in accordance with seal ring manufacture specifications.

NOTE: If seal is to be manufactured in house additional data will be required before tolerance and surface finish requirements may be established.

- Surface Finish on all machined surfaces = 250RMS unless otherwise required for seal.

- Concentricity between seal diameter and taper diameter to be within allowable specifications as dictated by seal ring.

- Perpendicularity to common centerline through top seal diameter and tapered bore at discharge area can readily be held to .015 T.I.R. (See Sections 3.2 and 3.3 for proposed manufacturing methods.)

2.2.2 Volute Case Assembly

- Rabbet Fits = .003 I.D. and O.D. or .006 Total
- Bearing Diameters = $\pm .003$
- I.D. and O.D. of Volute Lips = $\pm 1/64$
- O.D. of Vanes = $-1/8$
- Ring Diameters = $\pm .005$
- Faces adjacent to rabbet fits to be perpendicular to centerline within .005
- Any locating keys, dowel locations, etc. to be within .005 true location

2.0 MANUFACTURABILITY, cont.

2.2 RECOMMENDED TOLERANCES, cont.

2.2.2 Volute Case Assembly, cont.

- Surface Finishes:

- Mating surfaces = 125 RMS
- Ring diameters = 125 RMS
- Bearing surfaces = 63 RMS
- All other machine surfaces = 250 RMS

2.2.3 Rotating Element

a - Tolerances

- T.I.R. between bearings and ring fits = .005 Maximum
- Bearing diameters = .003 total
- Ring diameters = .003 total
- Impeller O.D.s = 0-minus, 1/32
- O.D. of shaft flange = .005 total
- Perpendicularity between flange face and shaft centerline must be held as close as possible (say .002 maximum)

b - Surface Finish

- Ring diameters = 125 RMS
- Bearing diameters = 63 RMS
- Face and O.D. of Shaft Flange = 125 RMS
- All other Machined Surfaces = 250 RMS

2.2.4 Intermediate and Upper Shafts

a - Tolerances

- O.A.L. = $\pm 1/16$
- O.D. of Flanges = $\pm .005$
- I.D. and O.D. of Shaft Tubes = $\pm 1/32$
- Perpendicularity between Flange Faces and shaft centerlines = .002 Maximum
- T.I.R. of each shaft after balancing to be = .005 Maximum

b - Surface Finish

- O.D. and face of Flanges = 125 RMS
- All other Machined Surfaces = 250 RMS

2.0 MANUFACTURABILITY, cont.2.2 RECOMMENDED TOLERANCES, cont.2.2.5 Lower Discharge Column Extensiona - Tolerances

- O.A.L. = $\pm 1/16$
- O.D. of Flanges = $\pm 1/8$
- Perpendicularity of Flanges to centerline = .005 Maximum
- O.D. of Flanges to be concentric to centerline as close as possible (If machined in the same setup as recommended, concentricity between the 2 flanges should be .002 T.I.R. or better)
- Any alignment key, CAM pin slot, etc. to be within .010 of thru location to centerline

b - Surface Finish

- O.D. of Flange and Mating Flange Surfaces = 125 RMS
- All other Machined Surfaces = 250 RMS

2.2.6 Discharge Column Assemblya - Tolerances

- Tapered O.D. = $\begin{smallmatrix} + \\ -1/64 \end{smallmatrix}$ with a basic gage dimension both ways from discharge
- Discharge opening = $\pm 1/8$
- O.A.L. = $\pm 1/8$
- Perpendicularity of lower face to centerline = .005 Maximum
- T.I.R. between bearing journal and taper = .005 Maximum
- Bearing Diameter = $\pm .003$
- All alignment keys, CAM pin locations, etc. to be within .010 of true position to centerline

b - Surface Finish

- All mating surfaces to be 125 RMS
- Bearing surfaces = 63 RMS
- All other machined surfaces = 250 RMS

2.0 MANUFACTURABILITY, cont.2.2 RECOMMENDED TOLERANCES, cont.2.2.7 Upper Column Assemblya - Tolerances

- O.A.L. = $\pm 1/8$
- O.D. of Flanges = $\pm 1/8$ with Maximum Eccentricity to centerline of .005
- Flange faces to be flat and perpendicular to centerline within .005
- Any alignment keys or CAM pin locations to be within .010 true position to centerline

b - Surface Finish

- O.D. and Mating Flange surfaces = 125 RMS
- All others = 250 RMS

2.2.8 Driver Mount Assemblya - Tolerances

- O.D. of Flanges = $\pm 1/8$
- O.A.L. = $\pm 1/8$
- Seal fit = as required by seal ring manufacturer
- Shaft tube clearance bores = minimum clearance-- $+1/8$
- Seal fit to be concentric to shaft bore centerline within .005 T.I.R.
- Lower face and top flange faces to be perpendicular to centerline within .005
- Top flange thickness = Minimum $+1/8$
- Any alignment keys, CAM pin locations, etc. to be within .010 true position to centerline

b - Surface Finish

- Mating Flange Faces and O.D.s = 125 RMS
- Seal Fit = as dictated by seal manufacturer
- All other surfaces = 250 RMS

2.0 MANUFACTURABILITY, cont.

2.2 RECOMMENDED TOLERANCES, cont.

2.2.9 Drive Coupling

a - Tolerances

- Bore to shaft fits = .001 to .0015
- Key ways from shaft to coupling must be within .001 of centerline
- Rabbet fits = $\pm .001$
- Perpendicularity of flange faces to bore centerline should be held as close as possible (.002 Maximum)

b - Surface Finish

- Shaft Bores = 63 RMS
- Mating Surfaces = 125 RMS
- All other surfaces to be 250 RMS

2.3 ALIGNMENT METHODS

2.3.1 Rotating Element and Shaft Assemblies

Method of alignment at flanged joints are recommended to be four (4) body fit bolts at 90° and on a common bolt circle with the retaining bolts.

Body diameter should be .030/.060 larger than the other clearance holes in the bolt circle.

In this manner, all shaft centerlines may be mechanically aligned to the bearing datum and secured. Then four (4) of the clearance holes at 90° may be reamed out to accommodate the body fit bolts, or studs.

By using this method, and adhering to the tolerance specified in Section 2.2, total run out is anticipated to be .005 maximum and imbalance problems kept within an acceptable tolerance band. In fact if the shaft tubes are machined on the I.D. and O.D. and the .005 T.I.R. maintained on each component, balancing by means of adding or removing material will probably not be required.

2.0 MANUFACTURABILITY, cont.

2.3 ALIGNMENT METHODS, cont.

2.3.2 Mating Components of Inner Structure

Due to problems anticipated with thermal expansion due to varying ambient temperatures in a normal production environment, we discourage the application of male/female rabbit fits for registry between mating components.

The four (4) key method recommended for the "Clinch River" unit overcomes the thermal expansion problem. However, this method requires that the joint be assembled, mis-alignment recorded, disassembled, offset keys made and installed and the joint re-assembled and alignment rechecked.

The CAM pin method depicted on the following sheet eliminates the need to disassemble and re-assemble. The mating components may be assembled and mis-match corrected by rotating the (4) four CAM pins. Once correct alignment has been achieved, the pins are locked in place, and components matched marked for orientation. This method will reduce assembly and dis-assembly labor considerably.

The CAM pin method shown is recommended for alignment of the following joints:

- Volute case to lower column extension
- Lower column extension to discharge column
- Upper column to discharge column
- Drive mount assembly to upper column

2.3.3 Driver to Driver Mount

We recommend that the driver be mechanically aligned at assembly secured and held on location by the use of four (4) dowels at 90°.

Should the above method not be acceptable, due to anticipated thermal deviation between the driver and the mount, the CAM pin method outlined in Section 2.3.2 may be utilized.

2.0 MANUFACTURABILITY, Cont.

2.3 ALIGNMENT METHODS, cont.

2.3.2.1 CAM PIN METHOD

3.0 MANUFACTURING PLAN

3.1 MANUFACTURING NETWORK

The Manufacturing Network (See APPENDICES 6.1 and 6.2) has been prepared to graphically depict the basic operations required for each component and sub-assembly. Sub-assemblies are arranged in an indented format showing the anticipated sequence of assembly at the final stack up.

Part identification numbers and basic manufacturing dimensions are shown on conceptual drawing number 1F-7937, Rev. "A", Appendix 6-3.

Note that necessary weld joints and suggested additional forgings are also defined on the above drawing.

Operations such as hydro-testing for internal Q.A. requirements cannot be defined until detail drawings and test specs are available, however, we do assume that test flanges will be utilized and machined off after final hydro-test. More detail on this subject is presented in Section 5.0.

3.2 PROCESS OUTLINES (Components and Sub-Assemblies)

3.2.1 Volute Basic Manufacturing Sequence

The volute component will be semi-finish machined, assembled and finish machined in two major sub-assemblies. This assures a minimized tolerance build-up and reduces fixturing and set-up times. With this method maximum mis-alignment is calculated to .006TIR/over the 15'6" distance and could be adjusted downward. Realistic mis-alignment, tolerance, readily achievable, should be in the .003-.005 TIR range. Should closer tolerance be required, it may be achieved with added manufacturing cost.

The following sub-paragraphs outline by component part the basic manufacturing sequences anticipated.

3.2.1.1 Lower Suction - Piece #1

- Rough machine for overlay
- Clean internal fluid passages
- Non-destructive test, weld repair, etc. and weld core closures
- Stress relieve
- Semi-finish machine and qualify overlay

3.0 MANUFACTURING PLAN, cont.3.2 PROCESS OUTLINES (Components and Sub-Assemblies), cont.3.2.1 Volute Basic Manufacturing Sequence, cont.3.2.1.2 Lower Volute - Piece #5

- Rough machine and drill bleed hole
- Clean all surfaces
- Non-destructive test, weld repairs, weld core closures, bleed holes, etc.
- Stress relieve
- Blend welds
- Finish machine I.D. lower rabbet fits and face, drill and tap
- Assemble to lower suction pipes (#1) and finish machine upper rabbet fit and flange

3.2.1.3 Middle Suction - Piece #2 (Two Halves)

- Mill split line
- Layout and drill outer flange bolt circle
- Drill and tap tie down holes for manufacturing ease
- Grind, clean, weld repair and stress relieve
- Finish machine O.D., lower volute fluid passages and lower flange face
- Assemble to lower volute sub-assembly (Items 5 and 1) and finish I.D. ring fits and all bores
- Drill out tapped holes on flange

3.2.1.4 Upper Volute - Piece #6

- Rough machine for overlay
- Clean internal fluid passages
- Non-destructive test, weld repair, etc., and weld core closures
- Stress relieve
- Semi-finish machine and quality overlay (Note: Rabbet and mating surfaces finished)

3.0 MANUFACTURING PLAN, cont.

3.2 PROCESS OUTLINES (Components and Sub-Assemblies), cont.

3.2.1 Volute Basic Manufacturing Sequence, cont.

3.2.1.5 Upper Suction - Piece #3

- Rough machine and drill bleed hole
- Clean all surfaces
- Non-destructive test, weld repairs, weld core closures, bleed holes, etc.
- Stress relieve
- Blend welds
- Finish machine I.D. lower rabbet fits and face, drill and tap
- Assemble to lower suction piece (#1) and finish machine upper rabbet fit and flange

3.2.2 Rotating Element

The recommended construction of the rotating element differs from the conceptual drawing by showing the bearing journals as one piece forgings rather than fabricated sleeves.

3.2.2.1 Impellers - Items 6 and 7

- Standard machining sequences for nuclear impellers including weld prep on both ends

3.2.2.2 Bottom Stub Shaft and Journal #14 Forging

- Machine I.D. and O.D. and Weld Prep
- Overlay
- Semi-finish and qualify overlay

3.2.2.3 Upper Stub Shaft and Journal #15

Journal Forging

- Machine forging I.D. and O.D. and weld prep
- Overlay
- Semi-finish and qualify overlay

Flange Forging

- Semi-finish machine weld prep and drill

3.0 MANUFACTURING PLAN, cont.

3.2 PROCESS OUTLINES (Component and Sub-Assemblies), cont

3.2.2 Rotating Element, cont.

3.2.2.3 Upper Stub Shaft and Journal #15, cont.

Assembly

- Assembly and weld journal forging to upper shaft
- Qualify weld (N.D.T.) -- X-ray quality
- Assembly and weld flange to upper shaft
- Qualify weld (N.D.T.) -- X-ray quality

3.2.2.4 Rotating Element Assembly

- Weld upper and lower impellers together
- Assemble and weld stub shaft assembly
- Assemble and weld upper shaft assembly
- N.D.T. welds
- Stress relieve
- Turn between centers and finish machine, journals, welds, impeller O.D. and ring fits and flange face
- Balance element
- Grind O.D. of journals

3.2.3 Fabricated Parts (Outer Tank, Discharge Column, Etc.)

All fabrication must comply with applical ASME codes and R.D.T. specifications.

The basic sequence of weldments and machining of all components and sub-assemblies is shown in APPENDICES 6.1 and 6.2.
(Manufacturing Network)

Actual sequence of fabrication may vary due to; type of equipment selected, workload, and/or type of welding process used and method of non-destructive testing.

On most components, the actual sequence is not of any prime concern as long as proper inspection and code hold points are established and adhered to.

3.0 MANUFACTURING PLAN, cont.

3.2 PROCESS OUTLINES (Component and Sub-Assemblies), cont.

3.2.3 Fabricated Parts (Outer Tank, Discharge Column, Etc.), cont.

At the writing of this report it is anticipated that the welding processes used will be automatic "TIG", or "SUBMERGED ARC", or a combination of both.

Thermal cutting is anticipated to be accomplished with "PLASMA ARC" or "IRON POWDER" processes.

3.2.3.1 Shafts

Out of roundness, eccentricity and balancing problems could develop with the use of unmachined seamless tubing. Possible alternatives to this plan would be the addition of "balance rings" on the top and bottom of each section of tubing. Complete machining is of course another answer. It may also be desirable to have an additional joint in the discharge nozzle section. (This is covered in more detail in Section 5.0)

3.2.3.2 Taper Fits

The taper section in the discharge ring of the outer tank will be finish machined after the tank is fabricated complete. The top flange and seal fit will also be finished at this time to insure concentricity and perpendicularity between flange and taper.

For ease of manufacture, we recommend that a flatter taper of 5/8" or 3/4" per foot be considered in lieu of the 1" per foot taper specified.

3.3 METHOD OF ASSEMBLY AND DISASSEMBLY FOR SHIPMENT

Specially designed facilities will provide the means for assembly and disassembly (See Section 4.0)

3.3.1 The Major Assembly Steps Would Include:

- Assembly of the rotating element and shafts. Bearing diameters and all O.D.s will be mechanically aligned in a special fixture using dial indicators. Once aligned four (4) body fit bolts will be installed. (Holes to be reamed at assembly after final alignment.) Once the final assembly is aligned and doweled, a customer buy off for the rotating element should be specified on the route sheet.

3.0 MANUFACTURING PLAN, cont.

3.3 METHOD OF ASSEMBLY AND DISASSEMBLY FOR SHIPMENT, cont.

3.3.1 The Major Assembly Steps Would Include, cont.

- Assembly of the internal structure without the rotating element. (This operation would align all bearing surfaces by the use of the rotary table and dial indicators. Final verification and customer buy off would be made through the use of Lasers. After final inspection all pieces would be match marked and doweled and/or CAM pins secured mechanically and seal welded.)
- Final assembly of complete pump, less the outer tank. Final stack up sequence would be as follows:
 - Secure lower volute sub-assembly (Items #1 and 5 lower suction piece and lower volute) to turn table fixture and indicate true to table centerline.
 - Position rotating element in volute (less intermediate and top shafts).
 - Position middle suction pieces.
 - Lower the upper volute sub-assembly (Items #3 and 6 upper suction and upper volute) over rotating element, align and secure.
 - Position and secure lower shaft.
 - Position and secure discharge extension.
 - Assemble discharge nozzle tank to discharge extension.
 - Assemble and secure upper column.
 - Secure upper shaft.
 - Install reflection box assembly.
 - Position and secure insulator box
 - Install complete inner structure into outer tank set height, etc. and prepare for performance test.

3.0 MANUFACTURING PLAN, cont.

3.3 METHOD OF ASSEMBLY AND DISASSEMBLY FOR SHIPMENT, cont.

3.3.2 Disassembly for Shipment

Shipping procedures for the outer tank will provide many opportunities for further development for the packaging and shipping engineers as with present design the unit must be shipped in one piece. With the flange O.D. being close to 13Ft. diameter and the overall length in excess of 50Ft., this 71.5 ton unit must receive careful attention, and will probably require special permits, etc. if it must be transported via public highways.

The internal pump structure may be disassembled and crated as the following major components:

- Rotating Element
- Volute (2 sub-assemblies or complete)
- Lower Shaft
- Upper Shaft
- Discharge Extension
- Discharge Nozzle Tank
- Upper Column
- Driver Mount
- Seal Assembly
- Reflector Box
- Insulator Box

There are no anticipated problems with the disassembly, packaging and shipping of the internal components. However, assembly on the site could become a major problem if adequate fixturing devices, alignment plugs, etc. are not provided.

3.4 TESTING (Non-Destructive)

Even though the test procedures are not apart of a manufacturability study per se it should be considered in the manufacturing plan to help reduce end product costs.

All hydro-test requirements, procedures and methods of blocking off nozzles, etc. must be clearly defined prior to further development of the manufacturing plan. This would allow any changes in design or sequence of operations to be incorporated to achieve the most economical approach.

3.0 MANUFACTURING PLAN, cont.

3.4 TESTING (Non-Destructive), cont.

3.4.1 Outer Tank

The suction, discharge and overflow nozzles are recommended being machined to accept a flange that would be used during hydro and performance testing. These would later be cut off and the openings finished machined to conform to the final weld joint configuration. (Potential alternates requiring further development are also discussed in Section 5.0)

3.4.2 Volute

The volute components may be tested either individually or as an assembly depending upon the processes dictated by Q.A.

The least costly method for hydro testing volute cross-over would be to cast the suction inlets closed, machine complete as depicted in Section 2.2, assemble complete and hydro-test. After testing the suction openings could be machined out. If this method is not accepted, the costs for fixturing and testing could increase by factors of 10 to 50.

NOTE: This method is now used successfully on "HSB" and "HDB" type volutes.

3.4.3 Discharge Column

Hydro-testing of the discharge column can be most economically accomplished by testing the lower column and discharge column together. If the specs or customer would allow, the discharge column could be fabricated complete and hydro-tested prior to cutting out the discharge opening. If this is not acceptable for a final hydro, we would still recommend that the same sequence be followed, and that the final hydro if required be accomplished in the tank prior to "stack-up". Either method eliminates the need for an elaborate fixture and hours of machining and assembly.

3.4.4 Upper Column

If hydro is required on this part it can be accomplished with modified production components thus eliminating the need for test plates.

3.4.5 Final Testing

Final performance testing with de-ionized water will require a specially designed test pit. The basic dimensions of this pit is shown in APPENDIX 6.7.

4.0 FACILITY REQUIREMENTS

GENERAL

Considerable expense will be incurred preparing the existing facilities and organization to meet RDT specifications. The following requirements are a few examples that would have a significant impact on costs and lead times prior to starting actual production.

- All mercury vapor and fluorescent lamps must be fully contained to prevent the possibility of contamination if a bulb is broken. (Estimated cost to upgrade applicable areas at Vernon is \$25,000.)
- All handling devices must be either of austenitic stainless steel or synthetic fiber rope, woven slings, etc.
- No carbon steel shall come into contact with any finished surface (this could be interpreted to include chuck jaws, fixtures, tools, etc.)
- All markings of parts must meet RDT F7-3T which states no markings to exceed .010 inch including temporary identification and dimensional layout.
- All weld joints must be serialized and documented.
- Coolant, lubricants, cleaning agents, etc. must be non-contaminating or be removed immediately after use. (The control over these items could be quite costly and require the upmost vigilance from manufacturing supervision.)
- Lint free cloth or film gloves should be used when handling final cleaned coolant surfaces or any other surface specified by the purchaser.
- Only qualified riggers and crane operators may be used for handling critical parts. (By definition, all parts could be classified as "Critical".)
- Rigging must be stored in specific manner and usage history maintained, in other words total controls over WHO, WHEN and HOW used.
- Etc., etc.

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- Rigging must be stored in specific manner and usage history maintained, in other words total controls over WHO, WHEN and HOW used.
- Etc., etc.

4.0 FACILITY REQUIREMENTS, cont.

4.1 FABRICATION AND ASSEMBLY

4.1.1 FABRICATION

The changes necessary for handling the large scale sodium pump have been noted in the fabrication facility expansion plans review. The proposed expansion of the Byron Jackson facilities is not included in this report.

We recommend that the outer tank sub-assembly be "X-rayed" in three (3) basis sub-sections and that the final two (2) girth welds, cobalt or iridium methods of non-destructive testing be utilized.

4.1.2 ASSEMBLY

A separate assembly area must be provided that would meet RDT specifications providing a clean working area with limited access.

Major consideration in building this facility would be:

- Crane capacity -- 100 ton minimum (If RDT specifications must be adhered to, a 125 ton hook may be required).
- Hook height -- 65 feet minimum,
- Pit depth -- 50 feet minimum.

A rough sketch has been provided (See APPENDIX 6.7) showing an assembly building meeting the above criteria. This concept would allow complete assembly, testing and disassembly of the conceptual pump design.

4.0 FACILITY REQUIREMENTS, cont.

4.2 MACHINE TOOLS, cont.

4.2.3 ROTATING ELEMENT ASSEMBLY

Individual impellers, stub shafts and journals may be semi-finished machined with 108" V.T.L.'s and 32" engine lathes.

Final machining of the welded assembly should be performed on an engine lathe.

Therefore, we would recommend an engine lathe with a 70" minimum swing capacity over the cross-slide or a gap-bed lathe of similar capacity. Minimum center to center requirements are approximately 192" (16 ft.). This piece of equipment should also be provided with a precision tool post grinder for finishing of barrel journals.

4.2.4 INTERMEDIATE AND UPPER SHAFTS

All machining may be accomplished with engine lathes and drilling equipment.

4.2.5 LOWER DISCHARGE COLUMN EXTENSION, DISCHARGE COLUMN, UPPER COLUMN, INSULATION BOX AND DRIVER MOUNT ASSEMBLY

All of the above items may be machined with 360" and 168" V.B.M.'s, 5" floor mill, radial drills and the portable drill mentioned earlier in this section.

4.2.6 MACHINE TOOL SUMMARY

The tabulation on the succeeding pages summarizes by component part the basic machine tool requirement defining potential utilization of equipment.

4.0 FACILITY REQUIREMENTS, cont.4.2 MACHINE TOOLS, cont.MACHINE TOOLS REQUIRED BY COMPONENT AND SUB-ASSEMBLY

NAME OF PART OR SUB-ASSEMBLY	MACHINE TOOLS	
OUTER TANK ASSEMBLY	360" V.B.M. 86" and 108" V.T.L. 50" H.B.M.	32' Portable Star Feed Bar 7' Portable Radial Drill
VOLUTE CASE ASSEMBLY	86" and 108" V.T.L. 168" and 360" V.B.M. 7' Radial Drills 5" H.B.M.	(1) Precision Grinder adapted to 360" or 168" V.B.M.
ROTATING ELEMENT	108" V.T.L., 32" Engine Lathe, 24" Engine Lathe, 7" Radial Drills, 36" V.T.L.	70" Engine Lathe, Tool Post Grinder Balancing Equip.
INTERMEDIATE AND UPPER SHAFTS	36" V.T.L., 24 and 66" Engine Lathe, 7' Radial Drills, 16 x 120 O.D. Grinder	Balancing Equipment
LOWER DISCHARGE EXTENSION	66" or 86" V.T.L. 360" V.B.M. 7' Radial Drill	
DISCHARGE COLUMN ASSEMBLY	48" V.T.L., 168" and 360" V.B.M. 5" Floor Mill, 7' Radial Drill	7' Portable Radial, Precision Grinder for 360" V.B.M.
UPPER COLUMN ASSEMBLY	168 and 360" V.B.M. 5" H.B.M.	7' Portable Radial Drill
INSULATION BOX	108 or 168" V.B.M.	
TOP COLUMN/DRIVER MOUNT ASSEMBLY	168" and 360" V.B.M. 7' Radial Drill	

4.0 FACILITY REQUIREMENTS, cont.

4.3 FIXTURE AND HANDLING DEVICES

The quantity of pumps to be manufactured will have a minimal influence on the quality of fixturing and tooling required. In order to satisfy ASME and RDT specifications, all fixtures must be well designed and provide an exceptionally high degree of quality assurance when subjected to a production environment.

The tooling defined in this section will be required whether one unit is manufactured or a multiple thereof.

All fixtures, handling devices, etc. defined in this section must conform to RDT specifications. Main areas of concern are:

- 1 - All surfaces contacting finished component surfaces must be of austenetic stainless steel. This could be a major expense if these specification must be adhered to.
- 2- All lifting and handling devices must be designed for a capacity equal to 125% of the anticipated load, and according to RDT, cannot be utilized for lessor non-critical parts.

4.3.1 FIXTURES

4.3.1.1 Fabrication

Normal rings, spiders, etc. with non-contaminating contact points will be required for controlling roundness, etc. during fit up and welding.

Other specific fixtures may only be defined after final manufacturing drawings are released.

4.3.1.2 Machining

1 - OUTER TANK ASSEMBLY

- a - special I.D./O.D. chuck jaws for rounding up and aligning cylindrical barrel section on V.B.M.'s.
- b - Holding fixture for suction and lower hemisphere components for machining on V.T.L. and V.B.M.

4.0 FACILITY REQUIREMENTS, cont.

4.3 FIXTURE AND HANDLING DEVICES, cont.

4.3.1 FIXTURES, cont.

4.3.1.2 Machining, cont.

2 - VOLUTE CASE ASSEMBLY

- a - VBM, VTL holding fixtures for machining all components
- b - VBM alignment fixture (1 each) for finish machining of each half sub-assembly section

3 - LOWER DISCHARGE SECTION

- a - Layout template for flange bolt circles.
(Will also be used on mating faces of discharge column and volute case assembly.)

4 - DISCHARGE COLUMN ASSEMBLY

No special fixtures anticipated

5 - UPPER COLUMN ASSEMBLY

- a - Layout template for flange bolt circles.
(May also be utilized on mating faces of discharge column assembly and top driver mount assembly.)

No other special fixtures are anticipated

6 - DRIVER MOUNT ASSEMBLY

- a - Drill template for bolt circle on top flange which may also be used on mating holes in outer tank flange.

No other special fixtures are anticipated

4.0 FACILITY REQUIREMENTS, cont.

4.3 FIXTURE AND HANDLING DEVICES, cont.

4.3.1 FIXTURES, cont.

4.3.1.3 ASSEMBLY AND STACK-UP

In addition to the turntable defined in Section 4.1 the following assembly fixtures will be required. The actual design of such a fixture will, of course, depend upon the final configuration of the various components.

- a - Alignment and checking fixture for qualifying rotating element and shafts,
- b - Holding and alignment fixture for aligning and securing the volute case assembly to the turntable.

NOTE: This will be a "critical" fixture for "stack-up" operations. It will be used to control the entire internal stack-up of some 100 to 150 tons and must be designed accordingly.

- c - Various supplementary supporting devices, to be incorporated in the assembly pit to retain alignment as the "stack-up" progresses.
- d - Laser equipment, targets, etc. for final customer witness.

4.3.2 HANDLING DEVICES

4.3.2.1 FABRICATION

- a - Special lifting devices will be required for handling the finished outer tank assembly in a horizontal position.
- c - Transfer carts, or rail cars for transferring the outer tank sections from the fabrication area to the "X-ray" chamber,

4.0 FACILITY REQUIREMENTS, cont.

4.3 FIXTURE AND HANDLING DEVICES, cont.

4.3.2 HANDLING DEVICES, cont.

4.3.2.2 MACHINING

No special lifting devices are anticipated, however, all conventional rigging devices must be updated to conform with RDT specifications in terms of non-contaminating surfaces and rated safety factors.

4.3.2.3 ASSEMBLY AND "STACK-UP"

A substantial amount of lifting devices will be required, again the actual requirement, common usage, etc. may only be intelligently defined after the final design has been developed.

5.0 AREAS FOR FURTHER DEVELOPMENT

Recommended areas for further development fall into two (2) basic categories:

- Product Design
- Manufacturing Processes

Although both are closely related, and in some cases inseparable, we have arbitrarily segregated our recommendations as follows:

5.1 PRODUCT DESIGN

5.1.1 ADDITIONAL INTERMEDIATE SHAFT TUBE

As previously recommended, the existing intermediate shaft, if sub-divided into two (2) sections would help to reduce manufacturing problems.

The existing design indicates a intermediate shaft length in excess of 21feet.

Manufacturing control over straightness and T.I.R. is greatly influenced by the diameter to length ratio.

It will be easier to hold the recommended tolerance working with 10 foot sections vs 21 foot sections. We see no problems for access for assembly and dis-assembly as long as the secondary flange joint is placed in the discharge area.

Other potential benefits from this approach will also be gained with the ability to further offset any inherent imbalance of mass at the assembly of the shafts and rotating element.

5.1.2 UPPER COLUMN

As previously stated, maximum overall height of any section requiring machining should be limited to 14 feet (168").

This component as shown is approxiamtely 15 feet (180"). Although shortening this section by 12" is not a major design problem, registry of the insulator box and maintaining the required elevation may require some re-engineering of the "Upper Colum", "Insulator Box", and "Driver Mount" components.

5.0 AREAS FOR FURTHER DEVELOPMENT, cont.

5.1 PRODUCT DESIGN, cont.

5.1.2 UPPER COLUMN, cont.

Should the above present major engineering problems, the best alternative would be a two piece upper column assembly with a flanged joint approximately in the center. Should a flange at this point impair pump performance or otherwise be undesirable we suggest that a flange on the I.D. of the unit be considered.

Should the decision be made to go with the two piece column, the CAM pin method of alignment should be used. All other tolerances, surface finishes recommended for the one piece "Upper Column" would equally apply.

5.1.3 DISCHARGE COLUMN

A straight bore in lieu of the taper bore may be considered.

This would help to reduce manufacturing cost and tooling cost considerably.

However, realistic manufacturing tolerances may not provide an adequate gap for "bleed by" requirements.

Should this idea be pursued the anticipated tolerances for both the I.D. and O.D. would be:

- $\pm .015$ on diamter of both parts or a potential .060 tolerance band.
- concentricity, roundness, etc. would be equal to these specified for the tapered fit.

One disadvantage of this approach will be the elimination of the ability to correct the gap by raising or lowering the inner structure.

Other considerations must also be given to differences in thermal expansion due to the varying cross-sections of the two components.

5.0 AREAS FOR FURTHER DEVELOPMENT, cont.

5.1 PRODUCT DESIGN, cont.

5.1.4 OUTER TANK

The existing design offers some challenging opportunities for a packaging engineer.

As designed, the unit must be shipped in one piece.

We suggest that a two piece tank be considered.

Such a tank could incorporate a flanged joint a minimum of 8' to 10' below the discharge centerline. The flanges would be used for hydro and performance testing. After final performance testing the flanges could be machined off and weld preps generated.

With this concept, the final girth weld would have to be made at the installation site with "Cobalt" or "Iridium" used as the N.D.T. method of qualifying the weld.

5.1.5 METHOD OF SETTING ELEVATION OF THE INNER STRUCTURE

Although not clearly defined on the conceptual drawing, there has been some discussion of using jack screws for adjusting the bleed gap at the taper area.

We suggest that further design analyses be performed in this area.

It would appear that a double set of circular harden and ground wedge plates could be utilized in lieu of jack screws.

Due to the location of this area in relation to the pumpage, it appears that contamination from heat treated alloys would not be a RDT concern. This, of course, would have to be verified.

If this method would be acceptable, the basic wedge thickness could be established during the "stack-up" and the wedge rings finish ground to suit.

Any such wedges should probably provide for some 3/4" to 1" adjustment in elevation.

5.0 AREAS FOR FURTHER DEVELOPMENT, cont.

5.1 PRODUCT DESIGN, cont.

5.1.6 HYDRO-TEST FLANGES

The conceptual design does not define test flanges for the suction and discharge flanges. The manufacturing plan assumes that test flanges will be incorporated and machined off after the final performance test.

An alternative approach may be a "Dresser" type flange coupling. Although not normally used for diameters of such magnitude, we feel that this approach may well offer some interesting possibilities.

Should the above prove feasible, it will certainly reduce manufacturing costs appreciably.

5.1.7 SEISMIC RINGS, SNUBBERS, ETC.

Although not considered at the conceptual design stage, seismic devices, snubbers, etc. could have a substantial impact on assembly procedures and overall manufacturing cost.

As the requirements for these items are defined, an additional producibility analysis would be made in order to assure adherence to RDT specifications and to define the impact on total product cost.

5.2 MANUFACTURING PROCESSES

As previously stated, the equipment and technology required to build the conceptual design is available today in industry. However, since the design will probably be built in the somewhat distant future, an equal number of years of improved technology will more than likely be available. In fact, some of these areas of advanced technology are in existence today in proto-type and/or laboratory environment.

A few of them for future consideration are listed below.

5.2.1 LASER WELDING

Laser techniques whereby full penetration welds may be produced in inaccessible areas are now in the proto-type stage and will more than likely be perfected very shortly.

Such a process would be invaluable in fabricating a large scale breeder pump with an exceptionally high degree of quality assurance. As this technology is made available to industry, it very definitely should be investigated for possible utilization on this project.

5.0 AREAS FOR FURTHER DEVELOPMENT, cont.

5.2 MANUFACTURING PROCESSES, cont.

5.2.2 PLASMA ARC MACHINING

Plasma arc machining has successfully been accomplished in Europe in limited areas.

At present, major firms in the U.S.A. are making further improvements in this area.

When fully developed, this method will afford metal removal rates that are 5 to 10 times greater than those that can be achieved with modern day technology.

The utilization of such a process for machining of austenitic stainless steels would be a major source for reducing manufacturing cost.

5.2.3 FABRICATED IMPELLERS

Recent experimental activities at the Vernon facility indicate the fabricated impellers using electron beam welding processes are quite feasible.

We recommend that these activities be pursued and considered as a possible alternate for cast impellers on the large scale breeder pump.

Should such a process be developed to the point where parts of such size could be considered, the advantages are many:

- cleaning of fluid passages are minimal
- integrity of structure is assured prior to any machining
- pattern costs are eliminated
- long lead times for castings are eliminated (This also would help to reduce W.I.P.)
- impellers may be fabricated to allow minimal machining at cut diameter, rather than full diameter
- engineering has more latitude in vane design which is now limited by pattern and foundry technology

• APPENDICES

FOR

MANUFACTURABILITY OF LARGE
SCALE BREEDER PUMP

6.0 APPENDICES

- 6.1 Manufacturing Network - Drawing Numbers MA-010 and MA-011
- 6.2 Color Coded Raw Material Definition - Drawing Number 1F-7937-1
- 6.3 Basic Dimensional Drawing with Numerical Identification of Sub-Components and Assemblies - Drawing Number 1F-7937-2
- 6.4 Estimated Weight - Sheets 1 thru 3
- 6.5 This section deleted.
- 6.6 Sample Operation Drawings
 - 6.6.1 Suction Nozzle and Lower Hemisphere Assembly - (MA-001)
 - 6.6.2 Typical Outer Tank Section - (MA-002)
 - 6.6.3 Discharge Section of Tank (Original Concept) - (MA-003)
 - 6.6.4 Overflow Nozzle Section of Tank - (MA-004)
 - 6.6.5 Upper Flange and Transition Area of Outer Tank - (MA-005/006)
 - 6.6.6
 - 6.6.7 Lower Discharge Column Extension - (MA-007)
 - 6.6.8 Discharge Head Weldment - (MA-008)
 - 6.6.9 Discharge Section of Tank (Proposed Concept) - (MA-0012)
- 6.7 Rough Sketch of Proposed Assembly and Test Facility

LARGE SCALE FAST BREEDER PUMP

"ESTIMATED" WEIGHTS (Dry and Finish Machined)

All calculations based on .283 lbs/cu" (489 lbs/cu ft)
and assumes drawing number 1F 7937 to be drawn to scale.

- PUMP TANK (Outer Barrel)

- Item 10-1, 10-2, 10-3	=	52,461 lbs	
10-4	=	26,610	
10-5	=	5,422	
10-6	=	44,947	
10-7, 10-8	=	9,208	
10-9	=	<u>4,351</u>	
		142,999	= <u>71.5 TON</u>

- VOLUTE ASSEMBLY

- Item 1, 2, 3, 5 and 6	=	66,580 lbs	= <u>33.3 TON</u>
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- ELEMENT ASSEMBLY

- Item 14 (lower shaft/journal)=	3,041 lbs	
15 (upper shaft/journal)=	4,191	
7 (lower impeller)	= 9,246	
8 (upper impeller)	= <u>9,246</u>	
	25,724	= <u>12.9 TON</u>

- LOWER COLUMN SHAFT

- Item 16	=	8,196 lbs	= <u>4.1 TON</u>
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- UPPER COLUMN SHAFT

- Item 17	=	5,622 lbs	= <u>2.8 TON</u>
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- DISCHARGE COLUMN ASSEMBLY

- Item 11.4	=	26,630 lbs	
11.5	=	2,849	
11.1	=	6,144	
11.2	=	1,280	
11.3	=	7,175	
11.6	=	3,699	
11.7	=	4,608	
11.8	=	<u>1,627</u>	
		54,012 lbs	= <u>27 TON</u>

ESTIMATED WEIGHTS, cont.

- UPPER COLUMN ASSEMBLY

- Item 12.1	=	16,273 lbs
12.2	=	1,792
12.3	=	9,283
12.4	=	9,520
12.5	=	2,811
12.6	=	11,095
12.7	=	3,144
12.8	=	1,850

((4) Gussets between
10.6 - 10.7 2" x 18" x
ABT 60")

= 2,444

58,212

=

29.1 TON

* * * * *

(A) ELEMENT ASSEMBLY WITH UPPER AND LOWER COLUMN SHAFTS

Element Assembly	=	12.9 Ton
Upper Column Shaft	=	2.8
Lower Column Shaft	=	<u>4.1</u>

19.8 TON

(B) TOTAL WEIGHT ON LOWER DISCHARGE COLUMN FLANGE (ITEM 11-2)

Volute Case Assembly	=	33.3 Ton
Element Assembly	=	<u>12.9</u>

46.2 TON

(C) TOTAL WEIGHT ON LOWER FLANGE OF UPPER COLUMN (ITEM 12-2)

Sub Assembly (A) above	=	19.8 Ton
Volute Case Assembly	=	33.3
Discharge Column Assy.	=	<u>27.0</u>

80.1 TON

ESTIMATED WEIGHTS, cont.

- (D) TOTAL WEIGHT ON TOP BARREL FLANGE (ITEM 109-) and SUSPENDED FROM TOP UPPER COLUMN FLANGE (ITEM 12-7) (This also reflects minimum crane capacities)

Sub Assembly "C" above = 80.1 Ton
Upper Column Assembly = 29.1

109.2 TON

- (E) TOTAL PUMP WEIGHT (Including Barrel)

Total Assembly "D" above = 109.2 TONS
Barrel Assembly = 71.5

180.7 TONS

(Does not include driver or
driver mount)

6.7 ASSEMBLY AND TEST FACILITY

BASIC FLOOR LAYOUT

BASIC ELEVATIONS.

LARGE SCALE SODIUM PUMP MANUFACTURING NETWORK

APPENDIX 6.1, SHEET 1 of 2

REVISIONS			
NO.	DESCRIPTION	DATE	APP.

CONT'D ON SNT 2 of 2

DETAIL	REV	SIZE OR PATTERN NO.	MATERIAL
SCALE	5-76	BYRON JACKSON PUMP DIV.	BYRON JACKSON PUMP DIV.
DATE	5-76	BOSS WARNER CORPORATION	TOOL DESIGN
DESIGNED BY	SEED	PART NO.	LARGE SCALE LIFFER PUMP
CHECKED BY	SEED	MANUFACTURING NETWORK	
DESCRIPTION	INDENTURED SEQUENCE of MEQ ASSY.		
FOR INVENTORY	TOOL NO.		
PRODUCT	MA-010		
A	B	C	D

- (RED) ~ MACHINED FORGINGS 304/316 SS
- (GREEN) CASTING ~ 304 SS
- (BLUE) CASTING ~ 316 SS
- (YELLOW) PLATE ~ 316 or 304 SS
- (BROWN) SEAMLESS TUBING ~ 304 SS
- (ORANGE) CARBON STEEL PLATE

10.0 BALANCING AND ALIGNMENT

10.1 DYNAMIC BALANCING REQUIREMENTS

10.1.1 ABSTRACT

The purpose of this report is to summarize the investigation into the requirements and feasibility for establishing dynamic balancing criteria and procedures for the large scale breeder reactor plant prototype mechanical pump.

10.1.2 SUMMARY

The manufacturing tolerances as proposed for the rotating element components will yield acceptable residual unbalance of the individual assemblies, precluding the necessity for extensive balance corrections.

Alignment adjustments incorporated into the design of the rotor will maintain accuracy of assembly that will provide a smoothly operating unit.

10.1.3 INTRODUCTION

As is true in any rotating machine, the need exists to reduce the residual unbalance in this pump's rotating element so that an efficient, smooth running machine will be obtained. An unbalanced rotor will cause vibration and stress in the rotor itself and its supporting structure.

This induced vibration may cause excessive wear in bearings and bearing journals, substantially reducing their service life. In addition, undesirable alternating stresses in structural supports and frames will result, which may eventually lead to their complete failure.

Unbalance may be simply defined as being caused by a non-symmetrical mass distribution about the rotational axis of a rotor. Balancing, therefore consists of the process of redistributing the mass of the rotor (generally by adding or removing weight) so that its mass becomes symmetrically distributed about its rotational axis. Centrifugal forces about the shaft axis, generated when the rotor is spinning, will then be in equilibrium, and the rotor will run without vibration.

The principal causes of unbalance are: (1) eccentricity of components due to machining and/or assembly; (2) variation within materials, (such as porosity or density); and (3) non-symmetry of design. For this pump investigation the primary concern is placed on accurately machining individual components thereby maintaining the concentricity of finished surfaces to established standards. With this approach, the inherent residual unbalance will be minimized, thereby precluding the need for excessive balance corrections. Accurate alignment of the individual components must also be accomplished in order to maintain the prescribed residual unbalance.

10.1.4 PROCEDURE

As is evident from the pump sectional drawing, 1F-8052, it is not practical or feasible to balance the entire rotating element as a completely assembled unit due to its length and weight. In addition, the design of the volute and column structure necessitates a sequential assembly of the rotating element components in conjunction with these items.

The rotor is thus divided into the following assemblies which will be considered first on an individual basis for specific requirements and then on an assembly sequence to note any cumulative effects due to misalignment of the component parts:

<u>Part Number</u>	<u>Description</u>
10-1	Lower Shaft Impeller Assembly
10-2	Middle Shaft Assembly
10-3	Upper Shaft Assembly
12	Mechanical Seal
13-1	Coupling Assembly

10.1.5 BALANCE CRITERIA

The permissible residual unbalance for each component of the rotating element can be determined from the accepted industry standards of the International Standards Organization Technical Committee 108 Draft 1940 Quality Grade 6.3 based on a maximum pump rotating speed of 600 RPM and the individual component weights. The following table lists the acceptable unbalance (in ounce-inches) for the rotating element components:

Description	Weight (lbs)	Maximum Acceptable Residual Unbalance (oz-in)
Lower Shaft Impeller Assembly	25,700	1650 oz-in
Middle Shaft Assembly	8,200	525 oz-in
Upper Shaft Assembly	5,600	360 oz-in
Coupling Assembly	1,300	85 oz-in

It should be noted that the balance criteria tabulated above is equal to 0.004 inch displacement of the center of gravity from the rotational axis for each piece in question. This displacement yields

a total indicator runout (T.I.R.) of 0.008 inch. The balance tolerance for the mechanical seal has been purposely omitted since the entire seal is not a rotating component, and will thus be discussed separately.

10.1.6 LOWER SHAFT IMPELLER ASSEMBLY

The lower shaft impeller assembly consists of the upper and lower impellers, upper and lower stub shafts with journals, and an upper journal support with flange.

The major source of unbalance in this assembly will be the two impellers since they are castings, and variations in metal thickness in both the impeller vanes and shrouds are to be expected. For this reason, each impeller should be "pre-balanced" before being welded into the impeller assembly. This operation will be beneficial in two respects; (1) the primary source of residual unbalance in the entire assembly will be reduced, thereby precluding the necessity for extensive balance corrections; and (2) a serious couple unbalance, which would be undetectable in a typical two-plane balance operation, will be minimized thus reducing the possibility of resultant shaft deflections. Balance corrections should be made by removing material from the impeller shrouds per the standard procedure employed on closed impellers of this type.

Before attempting to balance the entire assembly, the bearing journals as well as the impeller running fits and the middle suction piece clearance should be concentric as well as perpendicular to the upper

stub shaft flange face. A two-plane balance should then be performed utilizing the tolerance presented previously.

10.1.7 MIDDLE SHAFT ASSEMBLY

Since the proposed manufacturing tolerance of the middle shaft tube is 0.005 inch T.I.R. and the I.D. is to be held concentric with the O.D., the residual unbalance should be less than the balancing tolerance requirement, thus precluding the need for balance corrections.

The drilling of the end flanges for the bolting may introduce a slight unbalance due to manufacturing tolerances of the hole diameters and slight variations in location. Because of this possible deviation, the entire assembly should be investigated utilizing a two-plane balancing technique, with the correction planes being the two end flanges. Metal removal can then be made by drilling into the O.D. of the flanges.

An established technique for balancing long, cylindrical objects such as this assembly, entails the use of three correction planes; one at each end of the cylinder with the third plane located in the center of the other two. Since this procedure primarily corrects the assembly for run-out of the O.D. and eccentricity between the O.D. and the I.D. it is unnecessary to employ this method for this application due to the accuracy with which the shaft tube will be machined.

10.1.8 UPPER SHAFT ASSEMBLY

The previous discussion of balancing the middle shaft assembly also pertains to the upper shaft assembly since both units are basically long cylinders with a mass at either end. The upper shaft assembly differs slightly, however, having a flange at one end of the shaft with a slinger at the other. A two-plane balance operation can thus be performed utilizing the slinger and the flange as the correction planes.

10.1.9 COUPLING ASSEMBLY

The coupling assembly consists of the motor half coupling which bolts to the stub shaft which in turn is attached to the upper shaft assembly by means of a self-centering spline. This type of spline has been shown to be very effective in maintaining alignment between shafts, even when subjected to repeated disassembly.

Since the O.D. of the motor half coupling will be machined concentric to the I.D. within 0.005 inch T.I.R., and the stub shaft is basically a solid, fully machined shaft, the residual unbalance of the entire coupling should be acceptable due to the manufacturing tolerances invoked. However, the assembled coupling should be checked for a slight unbalance induced by the drilling of the stud holes with corrections made by drilling into the O.D. of the respective flanges.

10.1.10 MECHANICAL SEAL

The mechanical seal assembly does not require any special balancing considerations since the tolerances and accuracies to which the seal

must be built to function properly will yield a residual unbalance well within acceptable limits.

10.1.11 ASSEMBLY

An accurate assembly and alignment of the rotating element is essential in obtaining a smoothly operating pump. Even though the mating flanges of the rotating element are attached by means of studs and nuts, "weight-matching" of these items does not appear to be necessary based on previous Byron Jackson experience with large nuclear pump couplings. It was found that by maintaining a close tolerance on the length of the studs and obtaining high quality, commercial hex nuts, the variations in weight were insignificant, thus making further controls unnecessary.

Gross misalignment errors in assembly of the pump element are minimized by the close concentricity and perpendicularity tolerances imposed on the individual components. As discussed in the alignment analysis, the maximum mechanical misalignment due to the "stack-up" of machining tolerances is 0.014 inch T.I.R. at the upper journal. This eccentricity combined with the 0.005 inch maximum T.I.R. of the individual components results in a balancing tolerance grade of 9.6 instead of the anticipated grade of 6.3.

From a practical standpoint this tolerance still appears to be acceptable since it is the maximum possible tolerance and a level closer to the target grade will probably be obtained based on the alignment adjustments designed into the components. Imposing closer

machining tolerances is not feasible due to the difficulty and exactness required in attaining even the existing tolerances. Additionally, since this is a slow speed pump, the possibility of inducing a slight unbalance does not appear to be detrimental since successful balancing of a rotor does not require removal of all of the unbalance.

10.2 ALIGNMENT STUDY

10.2.1 Acceptable Alignment Criteria

This alignment study shows that acceptable straightness is achieved with the manufacturing method and tolerances described. Acceptance is based upon centering the end bearings of a "worst condition" pump case. Contact at any other location is then not allowed. This is a more demanding criteria than one that does not allow contact at three points.

10.2.2 Method for Obtaining Alignment

This manufacturing method gives a minimum misalignment by progressively machining the surfaces that determine alignment only after progressively assembling the attachment components. In other words, the typical component is aligned and affixed to the subassembly beneath it before it is final machined on the surfaces that will affect the alignment of the remaining components.

Although this method is cumbersome and expensive, it does result in a minimum misalignment by taking advantage of laser beam techniques. Other methods of greater practicability and lesser cost may later prove to also be acceptable, in which case they would certainly be adopted.

10.2.3 Method of Determining Maximum Misalignment

To find the "worst condition" centerline through the assembled components, a base line is laid out for reference only as a starting point for establishing relative curvature. A straight line through the end bearings is the worst rotation centerline as determined by maximum potential for shaft rubbing. Hydraulic and dynamic loads will make the shaft even less straight but the bearings are then supporting the shaft and contact will not occur.

10.2.4 Detailed Description of Pump Case Alignment Method

The pump case is machined in sub-assembly alignment stages as shown in Figure I. Section 1 is machined by conventional methods except that stock is left on surfaces marked M2. The first joint at the bottom of the pump is assembled, doweled, and laser beam targets are mounted in the locations labeled 1 and 2. This sub-assembly is then aligned with a laser beam so that the upper component of Section 1 can be aligned using laser beam target 3, and temporarily doweled. The surfaces marked M2 are then machined perpendicular to target 1 and 3.

Section 2 is machined on the surfaces marked M1 and stock is left on the surfaces marked M2. Sections 1 and 2 are joined and doweled or fixed by another method. Surfaces M2 on Section 2 are machined after aligning this sub-assembly using targets 1 and 3.

Section 3 is machined as shown with M1 and M2 signifying the same sequence as above. The Section 2 sub-assembly is aligned to a laser beam using targets 1 and 4. Target adapter 4 is removed, Section 3 is assembled by aligning target 5 and the joint is doweled or fixed by another method. Surfaces M2 on Section 3 are machined after aligning sub-assembly Section 1-2-3 using targets 1 and 5.

The same procedure is used for the remainder of the pump upper structure shown in Figure I.

10.2.5 Detailed Description of Pump Shaft Alignment Method

Section 1 of the pump shaft shown in Figure II is constructed by the method outlined above applied to the weld joints shown and pre-balanced. Pilot fits for laser target adapters are machined at each end after assuring there is

clean-up stock at the surfaces marked M2. Surfaces M2 are then machined after Section 1 is aligned to targets A and B; the completed assembly is dynamically balanced.

Bores M1 of shaft Section 2 are machined as concentric to the tube bore as possible using conventional methods, and laser beam targets are mounted in both bores. Surface M2 is machined perpendicular to targets B' and C. Section 1 is then aligned to targets A and B so that target B can be removed to make room for target B' on Section 2. The laser beam is then used to align target B' and the two shaft sections are joined but not yet doweled. Target C is checked for alignment to within 0.005 inches. Adjustment of target C is made by angular indexing shaft Section 2 relative to Section 1, or by re-machining one of the joining faces on the shaft ends. The joint is doweled and surfaces M3 are machined after aligning the sub-assembly to targets A and C.

Shaft section 3 is joined using the same procedure as was used for Section 2. External targets are used since the shaft is not hollow.

10.2.6 Maximum Misalignment of Pump Case and Shaft

Figure III shows the worst condition curvature case centerline and crooked shaft centerline when the described manufacturing procedures are used. Additionally, a total of .004 inches tolerance (0.002 radial) was allowed for mounting the laser beam targets and reading the pump alignment. Another 0.010 inches tolerance (0.005 radial) was allowed for machining, assembling, and doweling or otherwise affixing the pump case points. Perpendicularity of the pump case faces does not affect the worst condition curvature because the manufacturing procedure corrects for this factor as it arises.

The shaft centerline in Figure III is found by using radial tolerances of 0.001 inch for the targets, 0.002 inches manufacturing and doweling concentricity, and .0005 inches perpendicularity of the end faces at the outer diameter of the flanges. The perpendicularity tolerances allows for re-assembly after manufacturing and is in addition to the 0.005" target alignment described in the manufacturing procedure.

10.2.7 Maximum Misalignment "Running Clearance

The worst condition would result if the bearings at the top and at the bottom were centering the shaft and contact at the center bearings is not permitted. Figure IV shows this condition by combining the centerlines through the end bearings for the shaft and for the case from Figure III. Bearing number two in Figure IV shows a shaft orbit of .007 inches radius and is located eccentric from the pump case. The net result is a 0.018 inch radial off-set. Since the radial clearance at the bearings is 0.030 inches, the bearing journal clears the bore by 0.012 inches. Bearing number three has less off-set.

10.2.8 Conclusion

This manufacturing procedure would result in a very favorable alignment to prevent rubbing or binding of the bearing journals in the bearing bores.

LEGEND:

- ☒ - TARGET POINT FOR ALIGNMENT
- M_1 - MACHINE BEFORE ALIGNMENT
- M_2 - MACHINE AFTER ALIGNMENT
- M_3 - MACHINE AFTER M_1 & M_2

FIG. II

- TARGET PL. LOCATION
- x MANUFACTURING & SELLING TOLERANCE LOCATION
- CONSTRUCTION LINES

FIG. IV - TOTAL DEVIATION SHAFT ϕ TO BRG. ϕ
EXCLUDING DEFLECTION FROM DYNAMIC LOADS.

11.0 MATERIAL AND QUALITY ASSURANCE REQUIREMENTS

11.1 General

Based on an evaluation of materials for elevated temperature service and a liquid sodium system, the austenitic stainless steels offer the maximum advantages for the sodium containment member and associated hardware, for maximum low service cost within realistic initial costs.

The existing state of the art includes adequate means to verify materials and fabricated joint integrity. Mechanical integrity can be obtained by sound design principles based upon experience in the design of liquid sodium pumps. Fabrication requirements is one of the most important criteria to be considered and in this area the austenitic stainless steels offer superior values.

Based on the general conclusion that austenitic stainless steels offer the maximum advantages for sodium service, more specific material callouts can be made assuming certain forms in which the materials would be economically available, i.e., plate, forging, bar, or casting. The ASME callouts to which the major components would be procured are

Plate:	ASME SA-240 Type 316 or 304
Forging:	ASME SA-182 Type 316 or 304
Rolled Bar:	ASME SA-461 Grade 660
	ASME SA-193 Grade B7 or B8
	ASME SA-453 Grade 660 and 651
Casting:	ASME SA-613 Grade CF8 (static)
	ASME SA-613 Grade CF8 (centrifugal)
Pipe:	ASME SA-213 Type 316 (seamless)

Material specification will be based upon the requirements as set forth in the ASME Materials Specifications, Section II. Materials may be supplied to ASTM specifications provided the ASTM material is identical as to grade, composition and properties as set forth in the ASME specification. Additional special material requirements for pressure retaining and or permanent attachments to the pressure retaining boundary will be as set forth in ASME Nuclear Power Component Code, Section III, Article NB-2000. Requirements for non-pressure boundary and/or internal pump parts will be as set forth in the Design Specification and by the pump designer. Properties and additional design requirements for temperatures greater than 800°F. shall be based on ASME Code Case No. 1592-7.

11.2 Manufacture

The design and manufacture of a large liquid sodium pump, must of necessity, incorporate welded construction. The importance of materials weldability in obtaining a satisfactory design at an economical cost and with a maximum degree of weldment integrity, therefore, cannot be minimized. Material weldability is no better than the weld filler used for weldments. Electrodes and filler metals should be obtained to produce a range of 4 - 9% ferrite to minimize susceptibility to microfissure cracking in the weld deposit and heat-affected zone in austenitic steels.

The necessity for obtaining sodium containment equipment that will not allow leakage is a major item of importance, but one that can be obtained with existing technology of inspection media and design knowledge.

Final quality, must of necessity, be predicated on initial base metal quality. Radiographic quality castings can be obtained and internal quality verified using established techniques and qualified personnel. Surface quality can be verified by use of liquid penetrant inspection standards. The use of radiography supplemented with liquid penetrant inspection and pressure tightness confirmed

by hydrostatic and helium mass spectrometer testing will provide acceptable castings for sodium containment.

Wrought material quality can be verified by use of ultrasonic and liquid penetrant inspections.

The quality of weldments for liquid metal service has to be of maximum integrity. Control of weld joints for full penetration and use of radiography supplemented by liquid penetrant inspection will give the maximum quality. Pressure tightness must be confirmed by hydrostatic pressure testing. Leak rates of 10^{-8} cc/sec @ 60 - 80°F. as confirmed by helium mass spectrometer testing are easily obtained. Leak testing at higher temperatures has not been considered nor felt to be necessary. The size of equipment would create considerable testing problems not only from the standpoint of heating methods but sealing.

Sections III and IX of the ASME Boiler and Pressure Vessel Code form a basis for welding of equipment; however, it is usually necessary to amplify the welding requirements in procurement documents. Byron Jackson procedures, although based on the ASME Code are more strict for weldments in elevated temperature liquid metal service. Rather than depend on the vendor to set forth weld quality to be offered, procurement specifications should set forth minimum weld quality requirements.

11.3 Quality Assurance

The pump manufacturer shall be responsible for controlling the quality during manufacture in accordance with the requirements of ASME Nuclear Power Plant Components Code, Section III, Div. 1 to assure that all materials and parts,

including subcontracted items and services conform to the requirements of the Code. The responsibility shall include performing or having performed all controls, examinations, and tests necessary to substantiate that the materials, components, parts and appurtenances conform to the requirements of the Code.

ACCEPTANCE TESTS

11.4 HYDROSTATIC TEST

The large scale breeder reactor mechanical pump is designed to ASME Boiler and Pressure Vessel Code, Section III, Class I. Therefore all the pressure containing parts must be subject to a hydrostatic test. The pressure for the hydrostatic test according to NB-6221 of ASME Section III is 1.25 times the design pressure. The design pressure for the pump tank and inlet nozzle is 30 psig, therefore the hydrostatic test pressure is 38 psig. The design pressure for the pump hydraulics and discharge nozzle is 200 psig, therefore the hydrostatic test pressure is 250 psig. All of the test parts will be blocked off by blind flanges, suitable fixtures and gaskets. Test pressure will be maintained a minimum of 15 minutes for each inch of design minimum wall thickness. Indicating pressure test gages will be used, connected directly to the part being tested. Gages will be calibrated as required by the Code. Gage range will be about double the intended maximum test pressure but in no case shall it be less than 1-1/2 nor more than four times that pressure. All hydrostatic tests will be witnessed by code inspector. All parts to be hydrostatic tested will be in finish machined condition except the connection nozzles in the outer tank. Flanges will be welded to these nozzles for hydrostatic test as shown on Drawing No. SK-0002-33. These flanges will also be used for the performance tests. After all tests have been completed, the test flanges will be removed and the nozzle ends prepared for welding. This is shown on Drawing SK-0002-32.

12. PUMP PERFORMANCE TESTING

The size and criticality of these pumps dictate that they should be tested thoroughly and accurately on water before being subjected to the sodium test program. The more important aspects of the water tests are discussed in the following paragraphs.

Tests will be performed in accordance with the requirements of the ASME Power Test Code PTC 8.2 for centrifugal pumps.

12.1 TEST SET-UP

The significant aspects of the physical set-up include; 100 ft. overall height from inlet pipe to top motor, 130 ft. lifting height to install pump into barrel, 150 tons lifting weight of pump. The overall weight of the pump, barrel, and motor to be structurally supported during test is approximately 350 tons. Alternate test arrangements would be investigated in detail for cost and adequacy. Possible alternates are proposed as follows:

(a) Below Ground Test Pit - (see sketch T-1)

This arrangement would require an excavation approximately 50 ft. wide by 150 ft. long by 64 ft. deep. The walls would be lined with reinforced concrete and covered with a supporting deck. Internal cross beams would be suitable for bracing the side walls. Three floor levels would be constructed. The principal advantage of this plan is reducing the above ground height required for lifting out the pump. Lifting facilities would extend upward approximately 60 ft.

(b) Structural Tower (See Sketch T-2)

Essentially the same dimensions are required, however, the extensive excavation would be eliminated. Space is to be provided under the structure for unloading, erecting, assembling and handling the components. Floor levels, stairway, elevator, service and safety features would be included. At this time, selection of a preferred alternate of below ground or above ground is not considered feasible. Such items as zoning ordinances, environmental impact, foundation characteristics, and other siting qualifications indicate that the final selection should be deferred.

12.2 ELECTRICAL POWER

The test with cold water will require approximately 21,000 hp. This value can be reduced by permitting the loop temperature to rise or by running the test at a slightly reduced speed. For example, if a 200°F temperature is selected, the maximum required power could be reduced by 3%. This important test parameter indicates that the test site be located near adequate power facilities.

The test substation at Byron Jackson is rated at 20,000 hp and is therefore approximately of adequate size. Some modifications may be required depending on motor characteristics. 13,000 volts can be provided from this substation which is the most probable motor voltage. New switchgear would be required.

12.3 NPSH TESTING

The configuration of the loop will necessarily be limited by NPSH test requirements. The suppression tank controls suction head conditions and therefore the water level in the tank should not be higher than 10 ft. above the upper pump impeller. The NPSH available to the

pump will be set by pulling a vacuum or by permitting the temperature to rise. In either case, the minimum NPSH is limited by the static level difference.

The NPSH required by the pump is to be determined by setting a constant flow rate and then gradually reducing NPSH until the performance falls off. Several points of data are recorded and plotted so that any change in performance can be detected. This process is repeated at several capacities throughout the operating range of the pump.

12.4 SUPPRESSION TANK

A suppression tank is essential for two reasons; (a) stable control of NPSH setting and (b) re-absorbing of dissolved air. The recommended size is a tank volume in gallons approximately equal to the system flow rate in gallons per minute.

12.5 TEST PIPING

The piping would be designed to code requirements. The 50" discharge pipe would be fabricated from 1/2" plate. Elbow, tees, and fittings can be fabricated as segmented sections. These sizes and shapes are commonly constructed in this manner. The suction pipe may be of lighter weight. Material can be carbon steel with epoxy coated interior surface. A possible alternate material would be aluminum which would also eliminate carbon steel corrosion problems.

12.6 FLOW METER

Two venturi meters are to be used in parallel branches of the discharge system. Byron Jackson has extensive background data on 36"x 24" venturi design which would insure a high level of accuracy of measurement. See attached curve T-33150.

12.7 THROTTLE VALVES

Branches of two 36" discharge lines also permit the selection of throttle valves that are readily available. Motor operated butterfly valves rated at 250 psi would be adequate. Other type throttling valves may also be used such as Control Components "Drag" valves which eliminate valve cavitation at high velocities. It is also possible to design the valve flow control element in a 45° elbow and install the valve outlet directly on the tank to reduce cost.

12.8 LOOP COOLING AND WATER TREATMENT

Power input to the loop water is 20,000 hp or approximately 51 million BTU/HR. Most of this heat must be removed by heat exchanger, cooling tower, or other suitable means. Evaporative cooling would require 100 gpm of makeup water. Direct cycling of test loop water to the cooling tower would cause a rapid build-up of minerals of most available tap waters and is not recommended. A heat exchanger should be provided as the interface between the test loop and the cooling tower. The secondary loop would then absorb the make-up water.

Sizing of the heat exchanger depends on the desired loop operating temperature. The expected design temperature would probably not be less than 150°F but should not exceed 200°F.

De-ionized water may or may not be desired. Generally, austenitic stainless steels are not damaged by typical tap waters at these temperatures, however, some level of treatment is recommended to reduce pump cleaning problems after test. Possibly a combination of partial de-mineralization plus addition of a suitable inhibitor would be adequate.

12.9 EQUIPMENT LIST

A tabulation of equipment required is as follows:

Test Bay - Crane

De-ionizer

CCI Throttling Valve (2)

180° Return - 36" Ø

60 Flanges (6)

50" Flanges (4)

36" Flanges (20)

60" Pipe 150'

36" Pipe 80'

36" 45° El (2)

60" 90° El (1)

Heat Exchanger (at least 2)

Vacuum Pump

Cooling Tower

Cooling Tower Pump

Power Source & Switchgear

Instrumentation

36 x 24 Venturi (2)

SKETCH T-1

AD PAC 2 JULY 76

T-33150

DIFFERENTIAL PRESSURE (PSI)

T-33150

MIAMI

13. AUXILIARY REQUIREMENTS

13.1 Drive Assembly

13.1.1 A driver is required in this conceptual design study of a large scale breeder pump. Drive assembly capabilities and design requirements are of major importance to the selection, design, and application of the pump and driver assembly.

The driver shall be an electrical motor with all the necessary auxiliary equipment and instrumentation. It is for a vertical application, with the pump and driver connected by a rigid, flanged coupling. The pump axial thrust is carried by the motor.

13.1.2 The design requirements are shown in outline form.

13.1.2.1 Rated pump power requirements will range from 300 to 20,000 brake horsepower.

13.1.1.1 The speed shall be 600 RPM with a 4 to 1 turn down ratio. The speed change shall be by control signals.

13.1.2.3 A pony motor drive is required. This shall be an auxiliary electric motor to drive the pump at approximately 9% speed or 54 RPM when the main motor drive is turned off.

13.1.2.4 The pump will require an axial downthrust of 59,000 lbs., which must be carried by the motor thrust bearing.

13.1.2.5 The pump requires a single direction of rotation. There is no requirement for a device to preclude shaft rotation in a reverse direction.

13.1.2.6 There is no requirement for a flywheel in the motor.

13.1.2.7 Current design in commercial use shall be used where possible.

13.1.2.8 The motor located in the reactor containment building shall be of a minimum size and weight.

13.1.2.9 The motor shall give reliable operation with a minimum number of scheduled maintenance tasks.

13.1.2.10 The motor shall be tested separately and with the pump.

13.1.2.11 The interface between motor and pump will be resolved mutually. The shaft extension and method of centering, as well as manufacturing tolerances for mounting dimensions, requires a joint design effort.

13. AUXILIARY REQUIREMENTS (Con't)

13.2 RECOMMENDED INSTRUMENTATION, PUMP AND MOTOR

Recommended instrumentation includes the measurement and/or indication of pressure, temperature, flow, position, speed, vibration and power. Only functional requirements are enumerated.

13.2.1 Pressure should be measured at appropriate locations in the sodium system and in all lube oil and cooling systems to monitor proper function and indicate potential failure.

13.2.2 Temperature should be measured at appropriate locations in all lube oil and cooling systems to monitor proper function and indicate potential failure. Temperature should also be measured in the following locations:

Pump shaft seal

Motor shaft radial bearings

Thrust bearing

Drive motor windings

Liquid rheostat

13.2.3 Flow should be measured at appropriate locations in all lube oil and cooling systems to monitor proper function and indicate potential failure. Because accurate measurements of sodium flow rate are difficult to obtain, flow measurement is not recommended as an indication of failure. Adequate indication of failure or potential failure can be obtained by other means.

13.2.4 Position should be measured for the following components:

- (1) Liquid rheostat electrodes
- (2) Liquid rheostat electrolyte level
- (3) Pump barrel sodium level
- (4) Pump shaft rotating assembly, at motor shaft

13.2.5 Speed should be measured on the pump rotating assembly.

13.2.6 Vibration should be measured on the pump rotating assembly and also in the following locations:

- (1) Pump shaft seal
- (2) Motor shaft radial bearings
- (3) Thrust bearing

13.2.7 Power should be measured to the following components:

- (1) Liquid rheostat
- (2) Drive motor
- (3) Pony motor

14.0 DEVELOPMENT NEEDS

14.1 Development Program

14.1.1 Model Pump and Test

It is strongly recommended that a scale model pump be built and tested. The physical size of the model could be about one-third the size of the prototype. The specific speed of the model must be the same as that of the prototype. The operating speed of the model must satisfy this condition, and should give the same head as the prototype. The attached work sheet shows the deviation of the model speed and expected performance.

Building and testing the model is part of the general development of this type of pump, its behavior, and particularly its NPSH characteristic. The manufacturing and assembling of the model would be highly beneficial. It could be made for the use of additional instrumentation that would not be practical in the prototype.

The design could be improved in some areas as the patterns are being built for castings. Foundries could benefit from making the smaller model castings, using the experience thus gained for the prototype. The so-called "bugs" could be ironed out when the model is being built and assembled.

Models are built generally to obtain the desired information about the hydraulics performance of a given type of pump. In this case the bearings could be modeled and their performance checked. Other information could be gained that would help in the performance test set-up and test of the prototype.

It would of course be ideal to use a variable speed motor. Since availability of such a driver is remote, at least one additional speed could be tested with a gear reducer.

PUMP MODEL FOR TEST

14.1.1.1 Model Requirements

- (1) Model shall have same specific speed, N_s , as prototype.
- (2) Model shall have same head as prototype.
- (3) Model shall be one third, $1/3$, the physical size of the prototype.

14.1.1.2 Model Speed

$$\frac{H_M}{H_P} = \left(\frac{1}{3}\right)^2 \times \left(\frac{N_M}{N_P}\right)^2$$

N_M = Model speed, N_P = Prototype speed

for a head of 500 Ft., $\frac{H_M}{H_P} = 1$

$$N_M = 3 \times N_P = 1,800 \text{ RPM}$$

14.1.1.3 Model Flow Rate (Capacity)

$$Q_M = \left(\frac{1}{3}\right)^3 \times \frac{1800}{600} \times 127,000 = 14,100 \text{ GPM}$$

14.1.1.4 Motor Size

$$\text{HP} = \frac{14,100 \times 500}{3960 \times 0.80} \times 1.0 = 2,220 \text{ HP}$$

Say about a 2500 HP vertical motor.

14.1.1.5 Model Specific Speed

$$N_s = \frac{1800 \times 7050}{(500)^{3/4}} = 1470$$

This is the same as the prototype

14.1.2 HEAT EXCHANGER FOR SEAL

The heat generated by the seal must be carried away to give the desired long life. We have selected to circulate a light oil through the seal. The oil is then air cooled.

Determining the size of the fan, number and size of tube and fins, and other variables that will provide proper cooling for the oil being circulated will require further development. A full size test for the seals and heat exchanger is definitely recommended.

The seal cartridge must be removable without disturbing the motor. It is desirable to make the heat exchanger an integral part of the seal cartridge. This will require careful planning.

The seal should be removable as a cartridge - or complete assembly. This requires either a spacer or a stub shaft. Additional study could explore these alternatives.

14.1.3 OUTER BARREL SUPPORT

The structural frequency of the outer barrel shown in the conceptual design does not satisfy the Specification requirements. It is not practical to increase the thickness of the tank or add ribs.

One method would be the use of a dynamic restraint at the discharge nozzle elevation. The loads would be taken by hydraulic shock struts (snubbers). Brackets would be required on the outer case to take one end of the struts. They would require class 1 welding. The heavy brackets might have an undesirable thermal effect on the relatively thin-walled outer tank.

Space would be required in the field for anchoring the other end of the large snubbers in the proper location to take the required loads. This would of course require the safe use of snubbers in regard to the guard vessel.

Further study of the design and method of attachment of a tank restraint is considered essential.

Feasibility of developing alternative conceptual designs which satisfy frequency requirements without tank restraints has been investigated in this report. However, additional studies are recommended.

14.1.4 PUMP SUPPORT

The method of supporting this pump must be designed jointly by the pump supplier and the user. Carrying the dead weight is only part of the requirement. Thermal transients, shielding, and sealing must also be taken into consideration.

If canopy-type seal welds are required, the support will have to be modified to suit. Size, location, and spacing tolerance for sole plate clamping plate and hold-down bolts will have to be determined jointly.

Further detail design is required in this area.

14.0 DEVELOPMENT NEEDS

14.1.5 Stress Analysis Hydraulic Assembly

The assessment of potentially critical high stress areas indicates that several regions of the hydraulic assembly should be subjected to a detailed stress analysis.

This analysis should investigate the overall thermal response of the hydraulic assembly in addition to determining stresses at critical locations.

It is thus recommended that a three dimensional shell model of the hydraulic assembly be constructed and analyzed. The cyclic symmetry that the structure exhibits about the vertical centerline will permit the analysis to be conducted on only a partial model and yet accurately predict overall thermal response.

14.1.6 Stress Analysis of Tapered Seal Interface.

The effects of increased leakage and tank stresses due to the displacement of the inner support structure within the outer tank should be further evaluated using a three dimensional shell model of the pump in the region of the discharge penetration.

This study should determine the effective contact area between the two structures, in-plane shell stresses, and predicted clearance area under all design loads.

14.2 PROBLEM AREAS AND CONCERNS

14.2.1 WELDING

The specification requires that pump parts which are internal to the containment boundary shall be designed to meet the intent of Section III of the ASME Code for Class 1 Components.

It is difficult to comply with this requirement in some areas. To obtain Class 1 welds, many of the places where plate could be used will require forgings and additional machining to obtain welds that can be radiographed.

Some of the internal attachments should not be designed to this requirement. We have taken exception particularly in the hydrostatic bearing and journal areas. This is all structural and in some areas not subject to pressure differential. The journal has been designed so that it can be removed by grinding out small welds. The hardfacing could more readily be applied to a sleeve that would then be welded to two rings on the shaft. The end rings for the bearing could also be hardfaced and then welded to the large casting containing the integral bearing.

This problem area can be solved, but complicates the design, perhaps unnecessarily.

14.2.2 HYDROTESTING AND HELIUM LEAK TESTING

The work involved in non-destructive testing would not be apparent in the study of this concept, therefore it is included in this section. Handling alone, after major parts are fabricated and

machined, becomes a major undertaking.

Some of the larger parts will have to be designed with this requirement in mind.

Block off plates, fasteners, and seals will require additional design work. Some of the original design may be altered to make possible this non-destructive type test.

Some places, such as suction areas of the pump castings, flow areas of impellers, and others, should not be subject to these tests.

The necessity for acceptance tests is not questioned. Requirements can be met.

- 14.2.3 The performance test for the prototype pump is discussed under component verification tests. This is perhaps a major portion of the complete pump requirements. The performance test itself can be accomplished with no anticipated problems.

The size of the pump and motor and the configuration of the pump, combine to make the test set-up and space requirement a possible problem area. At least it requires additional study. The area or space requirement available now may not be available when the pump will be tested. The first choice for the test site is of course at present facilities.

Electrical power and switch-gear to suit the motor will be required. Crane facilities to handle the pump sub-assemblies and the large,

heavy motor will be needed at the test site. A large supply tank will be required, to give an accurate, reliable NPSH test. Much structural work will be required.

Because of the complexity of this test and its impact on time and cost, it is placed in this section of possible concern area.

PUMP CONCEPTS

[illegible]

REVISIONS		BY	DATE	APPROVED
1	ORIGINAL			
2	REWORK AND DETAILS ADDED: TOP AND BOTTOM			
3	REWORK: TO BE DONE IN 1957

BY: J. BYRON JACKSON DATE: 1/15/57 DRAWING NO.: 1F-7937		CHECKED BY: [blank] DATE: [blank] DRAWING NO.: [blank]	PROJECT NO.: [blank] DRAWING NO.: [blank]
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B- 1/2 INCH THICK RIBS
EQ. SP. ON C
16 RIBS TOTAL

SECTION A-A

DRAWING NO. 5K-0002-20 DATE 10-10-60 BY J. J. JACKSON CHECKED BY J. J. JACKSON APPROVED BY J. J. JACKSON		TITLE UPPER SUCTION & DISCHARGE MANIFOLD		ENGINEER BYRON JACKSON	
SCALE 1" = 1"		WEIGHT 10.0 LBS		PART NO. 5K-0002-20	

5K-0002-20

PARTS LIST AND MATERIALS OF CONSTRUCTION

LARGE SCALE BREEDER REACTOR PROTOTYPE MECHANICAL PUMP
B.J. ORDER NO. 761-G-0002
REFERENCE DRAWINGS: 1F-8052 & 1C-3563

<u>PART NO.</u>	<u>QTY</u>	<u>NAME</u>	<u>MATERIAL SPECIFICATION</u>
1		Outer Tank (Barrel) Assembly	
1-1	1	Suction Nozzle	ASME SA-182, GR. F304
1-2	1	Hemisphere	ASME SA-182, GR. F304
1-3	1	Lower Cylinder	ASME SA-240, Tp. 304
1-4	1	Middle Cylinder	ASME SA-240, Tp. 304
1-5	1	Discharge Nozzle Ring	ASME SA-182, GR. F304
1-6	1	Discharge Nozzle	ASME SA-182, GR. F304
1-7	1	Upper Cylinder	ASME SA-240, Tp. 304
1-8	1	Overflow Nozzle	ASME SA-182, GR. F304
1-9	1	Tapered Ring (Transition Piece)	ASME SA-182, GR. F304
1-10	1	Top Cylinder	ASME SA-240, Tp. 304
1-11	1	Top Flange	ASME SA-182, GR. F304
2		Inner Barrel Assembly	
2-1	1	Cylinder	ASME SA-240, Tp. 304
2-2	1	Lower Flange	ASME SA-182, GR. F304
2-3	1	Upper Flange	ASME SA-182, GR. F304
3		Lower Suction Piece and Bearing Assembly	
3-1	1	Lower Suction Piece	ASME SA-351, GR. CF8
3-2	2	Bearing End Ring	ASME SA-182, GR. F304 with #6 Welded Colmonoy Overlay
4	1	Middle Suction Piece (Two Halves)	ASTM A-351, GR. CF8

<u>PART NO.</u>	<u>QTY</u>	<u>NAME</u>	<u>MATERIAL SPECIFICATION</u>
5		Upper Suction Piece and Bearing Assembly	
5-1	1	Upper Suction Piece and Discharge Manifold	ASME SA-351, GR. CF8
5-2	2	Bearing End Ring	ASME SA-182, GR. F304 with #6 Welded Colmonoy Overlay
6		Lower Volute	
6-1	1	Lower Volute	ASME SA-351, GR. CF8
6-2		Hex. Hd. Cap Screw	ASME SA-193, GR. B8
7		Upper Volute	
7-1	1	Upper Volute	ASME SA-351, GR. CF8
7-2		Hex Hd. Cap Screw	ASME SA-193, GR. B8
7-3		Stud	ASME SA-193, GR. B8
7-4		Nut	ASME SA-194, GR. 6
8		Discharge Column Assembly	
8-1	1	Inner Cylinder	ASME SA-240, Tp. 304
8-2	2	Outer Cylinder (Split)	ASME SA-240, Tp. 304
8-3	1	Lower Flange	ASME SA-182, GR. F304
8-4		Hex. Hd. Cap Screw	ASME SA-193, GR. B8
9		Seal Ring Chamber Assembly	
9-1	1	Reducing Elbow	ASME SA-240, Tp. 304
9-2	1	Flange for Dished Head	ASME SA-182, GR. F304
9-3	1	Tapered Sleeve	ASME SA-240, Tp. 304
9-4	1	Elliptical Dished Head	ASME SA-182, GR. F304
9-5	1	Bearing Support Cylinder	ASME SA-240, Tp. 304
9-6	1	Bearing Outer Cylinder	ASME SA-182, GR. F304
9-7	1	Bearing	ASME SA-182, GR. F304 with #6 Welded Colmonoy Overlay
9-8		Hex. Hd. Cap Screw	ASME SA-193, GR. B8
9-9	8	Rib	ASME SA-240, Tp. 304

<u>PART NO.</u>	<u>QTY</u>	<u>NAME</u>	<u>MATERIAL SPECIFICATION</u>
10		Rotating Assembly	
10-1		Lower Shaft Impeller Assembly	
10-1-1	1	Lower Impeller	ASTM A-351, GR. CF8
10-1-2	1	Upper Impeller	ASTM A-351, GR. CF8
10-1-3	1	Bottom Stub Shaft	ASTM A-182, GR. F304
10-1-4	1	Bottom Journal	ASTM A-213, Tp. 304 with #5 Welded Colmonoy Overlay
10-1-5	1	Upper Stub Shaft	ASTM A-182, GR. F304
10-1-6	2	Upper Journal Support	ASTM A-240, Tp. 304
10-1-7	1	Upper Journal	ASTM A-213, Tp. 304 with #5 Welded Colmonoy Overlay
10-1-8	1	Flange - Upper Stub Shaft	ASTM A-182, GR. F304
10-2		Middle Shaft Assembly	
10-2-1	1	Middle Shaft Tube	ASTM A-213, Tp. 304
10-2-2	2	End Flange	ASTM A-182, GR. F304
10-2-3	2	Journal Support	ASTM A-240, Tp. 304
10-2-4	1	Journal	ASTM A-213, Tp. 304 with #5 Welded Colmonoy Overlay
10-3		Upper Shaft Assembly	
10-3-1	1	Upper Shaft Tube	ASTM A-213, Tp. 304
10-3-2	1	Top End - Upper Shaft	ASTM A-182, GR. F304
10-3-3	1	Flange	ASTM A-182, GR. F304
10-4	1	Slinger	ASTM A-182, GR. F304
10-5	32	Stud	ASTM A-453, GR. 660
10-6	64	Nut	ASTM A-453, GR. 660
10-7	1	Bolt	ASTM A-540, GR. B23 CL. 4
11		Reflector and Insulation Assembly	
11-1	1	Outer Cylinder	ASTM A-240, Tp. 304
11-2	1	Inner Cylinder	ASTM A-240, Tp. 304
11-3	1	Top Plate	ASTM A-240, Tp. 304
11-4	1	Bottom Plate	ASTM A-240, Tp. 304
11-5		Thermal Shield	
11-6		Insulation	

<u>PART NO.</u>	<u>QTY</u>	<u>NAME</u>	<u>MATERIAL SPECIFICATION</u>
10		Rotating Assembly	
10-1		Lower Shaft Impeller Assembly	
10-1-1	1	Lower Impeller	ASTM A-351, GR. CF8
10-1-2	1	Upper Impeller	ASTM A-351, GR. CF8
10-1-3	1	Bottom Stub Shaft	ASTM A-182, GR. F304
10-1-4	1	Bottom Journal	ASTM A-213, Tp. 304 with #5 Welded Colmonoy Overlay
10-1-5	1	Upper Stub Shaft	ASTM A-182, GR. F304
10-1-6	2	Upper Journal Support	ASTM A-240, Tp. 304
10-1-7	1	Upper Journal	ASTM A-213, Tp. 304 with #5 Welded Colmonoy Overlay
10-1-8	1	Flange - Upper Stub Shaft	ASTM A-182, GR. F304
10-2		Middle Shaft Assembly	
10-2-1	1	Middle Shaft Tube	ASTM A-213, Tp. 304
10-2-2	2	End Flange	ASTM A-182, GR. F304
10-2-3	2	Journal Support	ASTM A-240, Tp. 304
10-2-4	1	Journal	ASTM A-213, Tp. 304 with #5 Welded Colmonoy Overlay
10-3		Upper Shaft Assembly	
10-3-1	1	Upper Shaft Tube	ASTM A-213, Tp. 304
10-3-2	1	Top End - Upper Shaft	ASTM A-182, GR. F304
10-3-3	1	Flange	ASTM A-182, GR. F304
10-4	1	Slinger	ASTM A-182, GR. F304
10-5	32	Stud	ASTM A-453, GR. 660
10-6	64	Nut	ASTM A-453, GR. 660
10-7	1	Bolt	ASTM A-540, GR. B23 CL. 4
11		Reflector and Insulation Assembly	
11-1	1	Outer Cylinder	ASTM A-240, Tp. 304
11-2	1	Inner Cylinder	ASTM A-240, Tp. 304
11-3	1	Top Plate	ASTM A-240, Tp. 304
11-4	1	Bottom Plate	ASTM A-240, Tp. 304
11-5		Thermal Shield	
11-6		Insulation	

<u>PART NO.</u>	<u>QTY</u>	<u>NAME</u>	<u>MATERIAL SPECIFICATION</u>
12		Seal Assembly	
12-1		Seal Cartridge	
12-1-1	1	Seal Flange - Upper	ASME SA-182, GR. F304
12-1-2	1	Seal Flange - Lower	ASME SA-182, GR. F304
12-1-3		Seal Housing Assembly	
12-1-3-1	1	Top Flange	ASTM A-240, Tp. 304
12-1-3-2	1	Bottom Flange	ASTM A-240, Tp. 304
12-1-3-3	1	Inner Cylinder	ASTM A-240, Tp. 304
12-1-3-4	1	Outer Cylinder	ASTM A-240, Tp. 304
12-1-3-5	144	Vertical Tube	ASTM A-213, Tp. 304
12-1-3-6	6	Horizontal Tube	ASTM A-213, Tp. 304
12-1-4	1	Pumping Ring	ASTM A-182, GR. F304
12-1-5	1	Stand Pipe	ASTM A-182, GR. F304
12-1-6	2	Retainer	ASTM A-182, GR. F304
12-1-7	2	Rotating Face	Carbon
12-1-8	2	Stationary Face	316 SST with #1 Stellite Overlay
12-1-9	2	U-Cup	Nitrile
12-1-10	2	U-Cup Follower	304 SST
12-1-11	72	Spring	Inconel X-750
12-1-12	2	Seat Gasket	Nitrile
12-1-13	1	O-Ring	Nitrile
12-1-14	2	O-Ring	Nitrile
12-1-15	1	O-Ring	Nitrile
12-1-16	1	Stub Shaft	ASTM A-182, GR. F304
12-1-17	1	Adjusting Cap	ASTM A-182, GR. F304
12-1-18	12	Cap Screw - Hex. Hd.	ASTM A-193, GR. B8
12-1-19	12	Cap Screw - Hex. Hd.	ASTM A-193, GR. B6
12-1-20	8	Cap Screw - Hex. Hd.	ASTM A-193, GR. B6
12-1-21	8	Cap Screw - Hex. Hd.	ASTM A-193, GR. B6
12-1-22	8	Cap Screw - Soc. Hex. Hd.	ASTM A-193, GR. B8
12-1-23	2	Cap Screw - Hex. Hd.	ASTM A-193, GR. B6
12-1-24	2	Lock Plate	ASTM A-240, Tp. 304
12-1-25	1	Drive Key	ASTM A-479, Tp. 304
12-1-26	2	Lock Pin	ASTM A-479, Tp. 304
12-2	2	Fan Cover Half	ASTM A-240, Tp. 304
12-3	8	Cap Screw	ASTM A-193, GR. B8
12-4	8	Lock Washer	Commercial S. Steel

<u>PART NO.</u>	<u>QTY</u>	<u>NAME</u>	<u>MATERIAL SPECIFICATION</u>
13		Coupling Assembly	
13-1	1	Coupling - Motor Half	ASTM A-351, GR. CA-15 HT R/C 23-30
13-2	1	Thrust Disk	ASTM A-479, Tp. 410 HT R/C 26-34
13-3	1	Split Ring	ASTM A-479, Tp. 410 HT R/C 26-34
13-4	16	Stud	ASTM A-193, GR. B6
13-5	32	Nut	ASTM A-194, GR. 6
13-6	32	Lock Washer	Commercial Steel
14		Driver Mount Assembly	
14-1	1	Lower Flange	ASME SA-182, GR. F304
14-2	1	Lower Cylinder	ASME SA-240, Tp. 304
14-3	1	Tapered Ring	ASME SA-182, GR. F304
14-4	1	Middle Ring	ASME SA-240, Tp. 304
14-5	1	Upper Ring	ASME SA-182, GR. F304
14-6	1	Intermediate Plate	ASME SA-240, Tp. 304
14-7	1	Intermediate Flange	ASME SA-182, GR. F304
14-8	1	Transition Piece	ASME SA-240, Tp. 304
14-9	1	Cylindrical Cone	ASTM A-516, GR. 70
14-10	1	Top Flange	ASTM A-105
14-11	1	Inner Cylinder	ASME SA-240, Tp. 304
14-12	1	Stand Pipe	ASME SA-182, GR. F304
14-13	1	Outer Cylinder - Oil Drain	ASME SA-240, Tp. 304
14-14	1	Inner Cylinder - Oil Drain	ASME SA-240, Tp. 304
14-15	1	Middle Plate - Oil Drain	ASTM A-240, Tp. 304
14-16	1	Top Plate - Oil Drain	ASTM A-240, Tp. 304
14-17	1	O-Ring	Nitrile
14-18		Cap Screw - Hex. Hd.	ASTM A-193, GR. B8
14-19		Stud	ASTM A-540, GR. B23 CL 4
14-20		Nut	ASTM A-194, GR. 6
14-21		Stud	ASTM A-193, GR. B6
14-22		Nut	ASTM A-194, GR. 6

REVISIONS			
LTR	DESCRIPTION	DATE	APPROVED
1	ORIGINAL

REFERENCE:
SEAL CARTRIDGE SECTIONAL IC 3563
CONCEPTUAL DESIGN IC 3564

ALL DIMENSIONS UNLESS OTHERWISE SPECIFIED ARE IN INCHES. DIMENSIONS ARE TO BE HONED TO THE CLOSEST TOLERANCE PER A.S. SECTION 10. UNLESS TOLERANCE IS SHOWN DECIMALS TO 0.0005 FRACTIONS TO 1/16		UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES TOLERANCES ON DECIMALS ANGLES .0005 .0005 .0005 .0005 .0005 .0005		BY ORDER NO. 761 G 0202 DRAWN BY A. COLEMAN CHECKED BY L. TACUM CONTRACT NO.		REF. NO. DATE 11/17/76 DATE 12/20/76		BYRON JACKSON PUMP DIVISION BORG-WARNER CORPORATION LOS ANGELES DIVISION	
THIS MATERIAL IS THE PROPERTY OF BORG-WARNER CORPORATION AND IS LOANED TO THE CUSTOMER FOR THE PURPOSE OF TESTING AND EVALUATION. IT IS NOT TO BE USED FOR ANY OTHER PURPOSE. IT IS TO BE RETURNED TO BORG-WARNER CORPORATION AT THE END OF THE TEST. IT IS NOT TO BE REPRODUCED OR COPIED IN ANY MANNER. IT IS NOT TO BE USED FOR ANY OTHER PURPOSE. IT IS TO BE RETURNED TO BORG-WARNER CORPORATION AT THE END OF THE TEST. IT IS NOT TO BE REPRODUCED OR COPIED IN ANY MANNER.				SURFACE ROUGHNESS PER MIL STD 10 DRILLED HOLES V AS DRILLED TAPPED HOLES V AS TAPPED		DESIGN APP'D. W. J. WISE CUSTOMER APP'D.		DRAWING TITLE 14500 SEAL CARTRIDGE LMFBP PUMP	
DO NOT SCALE DRAWING. DIM. A, B, C, D, E, F, G, H, I, J, K, L, M, N, O, P, Q, R, S, T, U, V, W, X, Y, Z, AA, AB, AC, AD, AE, AF, AG, AH, AI, AJ, AK, AL, AM, AN, AO, AP, AQ, AR, AS, AT, AU, AV, AW, AX, AY, AZ, BA, BB, BC, BD, BE, BF, BG, BH, BI, BJ, BK, BL, BM, BN, BO, BP, BQ, BR, BS, BT, BU, BV, BW, BX, BY, BZ, CA, CB, CC, CD, CE, CF, CG, CH, CI, CJ, CK, CL, CM, CN, CO, CP, CQ, CR, CS, CT, CU, CV, CW, CX, CY, CZ, DA, DB, DC, DD, DE, DF, DG, DH, DI, DJ, DK, DL, DM, DN, DO, DP, DQ, DR, DS, DT, DU, DV, DW, DX, DY, DZ, EA, EB, EC, ED, EE, EF, EG, EH, EI, EJ, EK, EL, EM, EN, EO, EP, EQ, ER, ES, ET, EU, EV, EW, EX, EY, EZ, FA, FB, FC, FD, FE, FF, FG, FH, FI, FJ, FK, FL, FM, FN, FO, FP, FQ, FR, FS, FT, FU, FV, FW, FX, FY, FZ, GA, GB, GC, GD, GE, GF, GG, GH, GI, GJ, GK, GL, GM, GN, GO, GP, GQ, GR, GS, GT, GU, GV, GW, GX, GY, GZ, HA, HB, HC, HD, HE, HF, HG, HH, HI, HJ, HK, HL, HM, HN, HO, HP, HQ, HR, HS, HT, HU, HV, HW, HX, HY, HZ, IA, IB, IC, ID, IE, IF, IG, IH, II, IJ, IK, IL, IM, IN, IO, IP, IQ, IR, IS, IT, IU, IV, IW, IX, IY, IZ, JA, JB, JC, JD, JE, JF, JG, JH, JI, JJ, JK, JL, JM, JN, JO, JP, JQ, JR, JS, JT, JU, JV, JW, JX, JY, JZ, KA, KB, KC, KD, KE, KF, KG, KH, KI, KJ, KK, KL, KM, KN, KO, KP, KQ, KR, KS, KT, KU, KV, KW, KX, KY, KZ, LA, LB, LC, LD, LE, LF, LG, LH, LI, LJ, LK, LL, LM, LN, LO, LP, LQ, LR, LS, LT, LU, LV, LW, LX, LY, LZ, MA, MB, MC, MD, ME, MF, MG, MH, MI, MJ, MK, ML, MM, MN, MO, MP, MQ, MR, MS, MT, MU, MV, MW, MX, MY, MZ, NA, NB, NC, ND, NE, NF, NG, NH, NI, NJ, NK, NL, NM, NN, NO, NP, NQ, NR, NS, NT, NU, NV, NW, NX, NY, NZ, OA, OB, OC, OD, OE, OF, OG, OH, OI, OJ, OK, OL, OM, ON, OO, OP, OQ, OR, OS, OT, OU, OV, OW, OX, OY, OZ, PA, PB, PC, PD, PE, PF, PG, PH, PI, PJ, PK, PL, PM, PN, PO, PP, PQ, PR, PS, PT, PU, PV, PW, PX, PY, PZ, QA, QB, QC, QD, QE, QF, QG, QH, QI, QJ, QK, QL, QM, QN, QO, QP, QQ, QR, QS, QT, QU, QV, QW, QX, QY, QZ, RA, RB, RC, RD, RE, RF, RG, RH, RI, RJ, RK, RL, RM, RN, RO, RP, RQ, RR, RS, RT, RU, RV, RW, RX, RY, RZ, SA, SB, SC, SD, SE, SF, SG, SH, SI, SJ, SK, SL, SM, SN, SO, SP, SQ, SR, SS, ST, SU, SV, SW, SX, SY, SZ, TA, TB, TC, TD, TE, TF, TG, TH, TI, TJ, TK, TL, TM, TN, TO, TP, TQ, TR, TS, TT, TU, TV, TW, TX, TY, TZ, UA, UB, UC, UD, UE, UF, UG, UH, UI, UJ, UK, UL, UM, UN, UO, UP, UQ, UR, US, UT, UY, UZ, VA, VB, VC, VD, VE, VF, VG, VH, VI, VJ, VK, VL, VM, VN, VO, VP, VQ, VR, VS, VT, VU, VV, VW, VX, VY, VZ, WA, WB, WC, WD, WE, WF, WG, WH, WI, WJ, WK, WL, WM, WN, WO, WP, WQ, WR, WS, WT, WU, WV, WW, WX, WY, WZ, XA, XB, XC, XD, XE, XF, XG, XH, XI, XJ, XK, XL, XM, XN, XO, XP, XQ, XR, XS, XT, XU, XV, XW, XX, XY, XZ, YA, YB, YC, YD, YE, YF, YG, YH, YI, YJ, YK, YL, YM, YN, YO, YP, YQ, YR, YS, YT, YU, YV, YW, YX, YY, YZ, ZA, ZB, ZC, ZD, ZE, ZF, ZG, ZH, ZI, ZJ, ZK, ZL, ZM, ZN, ZO, ZP, ZQ, ZR, ZS, ZT, ZU, ZV, ZW, ZX, ZY, ZZ.				CODE IDENT NO. DRAWING NO. 1E-3893		SCALE FULL WEIGHT SHEET

SECTION A-A

REVISIONS			
LTR	DESCRIPTION	DATE	APPROVED
6	ORIGINAL		

REFERENCE:
FULL SCALE ASSEMBLY 1E-3893
SEAL CARTRIDGE SECTIONAL 1C-3563

ALL LIMITS OF CONCENTRICITY AND SQUARENESS OF FACES SUBJECT TO BYRON JACKSON INSPECTION STANDARDS PER S. M. SE-T ON 60. UNLESS TOLERANCE IS GIVEN MACHINE TO W. THIN. DIAMETERS: BORE + 1/32, TURN - 1/32. LENGTH: 8 TO 8 FT - 1/32, 8 TO 8 FT - 1/32, 8 FT AND OVER - 1/32.		UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES. TOLERANCES ON: DECIMALS: .005, .010, .015, .020, .025, .030, .035, .040, .045, .050, .055, .060, .065, .070, .075, .080, .085, .090, .095, .100, .105, .110, .115, .120, .125, .130, .135, .140, .145, .150, .155, .160, .165, .170, .175, .180, .185, .190, .195, .200, .205, .210, .215, .220, .225, .230, .235, .240, .245, .250, .255, .260, .265, .270, .275, .280, .285, .290, .295, .300, .305, .310, .315, .320, .325, .330, .335, .340, .345, .350, .355, .360, .365, .370, .375, .380, .385, .390, .395, .400, .405, .410, .415, .420, .425, .430, .435, .440, .445, .450, .455, .460, .465, .470, .475, .480, .485, .490, .495, .500, .505, .510, .515, .520, .525, .530, .535, .540, .545, .550, .555, .560, .565, .570, .575, .580, .585, .590, .595, .600, .605, .610, .615, .620, .625, .630, .635, .640, .645, .650, .655, .660, .665, .670, .675, .680, .685, .690, .695, .700, .705, .710, .715, .720, .725, .730, .735, .740, .745, .750, .755, .760, .765, .770, .775, .780, .785, .790, .795, .800, .805, .810, .815, .820, .825, .830, .835, .840, .845, .850, .855, .860, .865, .870, .875, .880, .885, .890, .895, .900, .905, .910, .915, .920, .925, .930, .935, .940, .945, .950, .955, .960, .965, .970, .975, .980, .985, .990, .995, 1.000, 1.005, 1.010, 1.015, 1.020, 1.025, 1.030, 1.035, 1.040, 1.045, 1.050, 1.055, 1.060, 1.065, 1.070, 1.075, 1.080, 1.085, 1.090, 1.095, 1.100, 1.105, 1.110, 1.115, 1.120, 1.125, 1.130, 1.135, 1.140, 1.145, 1.150, 1.155, 1.160, 1.165, 1.170, 1.175, 1.180, 1.185, 1.190, 1.195, 1.200, 1.205, 1.210, 1.215, 1.220, 1.225, 1.230, 1.235, 1.240, 1.245, 1.250, 1.255, 1.260, 1.265, 1.270, 1.275, 1.280, 1.285, 1.290, 1.295, 1.300, 1.305, 1.310, 1.315, 1.320, 1.325, 1.330, 1.335, 1.340, 1.345, 1.350, 1.355, 1.360, 1.365, 1.370, 1.375, 1.380, 1.385, 1.390, 1.395, 1.400, 1.405, 1.410, 1.415, 1.420, 1.425, 1.430, 1.435, 1.440, 1.445, 1.450, 1.455, 1.460, 1.465, 1.470, 1.475, 1.480, 1.485, 1.490, 1.495, 1.500, 1.505, 1.510, 1.515, 1.520, 1.525, 1.530, 1.535, 1.540, 1.545, 1.550, 1.555, 1.560, 1.565, 1.570, 1.575, 1.580, 1.585, 1.590, 1.595, 1.600, 1.605, 1.610, 1.615, 1.620, 1.625, 1.630, 1.635, 1.640, 1.645, 1.650, 1.655, 1.660, 1.665, 1.670, 1.675, 1.680, 1.685, 1.690, 1.695, 1.700, 1.705, 1.710, 1.715, 1.720, 1.725, 1.730, 1.735, 1.740, 1.745, 1.750, 1.755, 1.760, 1.765, 1.770, 1.775, 1.780, 1.785, 1.790, 1.795, 1.800, 1.805, 1.810, 1.815, 1.820, 1.825, 1.830, 1.835, 1.840, 1.845, 1.850, 1.855, 1.860, 1.865, 1.870, 1.875, 1.880, 1.885, 1.890, 1.895, 1.900, 1.905, 1.910, 1.915, 1.920, 1.925, 1.930, 1.935, 1.940, 1.945, 1.950, 1.955, 1.960, 1.965, 1.970, 1.975, 1.980, 1.985, 1.990, 1.995, 2.000, 2.005, 2.010, 2.015, 2.020, 2.025, 2.030, 2.035, 2.040, 2.045, 2.050, 2.055, 2.060, 2.065, 2.070, 2.075, 2.080, 2.085, 2.090, 2.095, 2.100, 2.105, 2.110, 2.115, 2.120, 2.125, 2.130, 2.135, 2.140, 2.145, 2.150, 2.155, 2.160, 2.165, 2.170, 2.175, 2.180, 2.185, 2.190, 2.195, 2.200, 2.205, 2.210, 2.215, 2.220, 2.225, 2.230, 2.235, 2.240, 2.245, 2.250, 2.255, 2.260, 2.265, 2.270, 2.275, 2.280, 2.285, 2.290, 2.295, 2.300, 2.305, 2.310, 2.315, 2.320, 2.325, 2.330, 2.335, 2.340, 2.345, 2.350, 2.355, 2.360, 2.365, 2.370, 2.375, 2.380, 2.385, 2.390, 2.395, 2.400, 2.405, 2.410, 2.415, 2.420, 2.425, 2.430, 2.435, 2.440, 2.445, 2.450, 2.455, 2.460, 2.465, 2.470, 2.475, 2.480, 2.485, 2.490, 2.495, 2.500, 2.505, 2.510, 2.515, 2.520, 2.525, 2.530, 2.535, 2.540, 2.545, 2.550, 2.555, 2.560, 2.565, 2.570, 2.575, 2.580, 2.585, 2.590, 2.595, 2.600, 2.605, 2.610, 2.615, 2.620, 2.625, 2.630, 2.635, 2.640, 2.645, 2.650, 2.655, 2.660, 2.665, 2.670, 2.675, 2.680, 2.685, 2.690, 2.695, 2.700, 2.705, 2.710, 2.715, 2.720, 2.725, 2.730, 2.735, 2.740, 2.745, 2.750, 2.755, 2.760, 2.765, 2.770, 2.775, 2.780, 2.785, 2.790, 2.795, 2.800, 2.805, 2.810, 2.815, 2.820, 2.825, 2.830, 2.835, 2.840, 2.845, 2.850, 2.855, 2.860, 2.865, 2.870, 2.875, 2.880, 2.885, 2.890, 2.895, 2.900, 2.905, 2.910, 2.915, 2.920, 2.925, 2.930, 2.935, 2.940, 2.945, 2.950, 2.955, 2.960, 2.965, 2.970, 2.975, 2.980, 2.985, 2.990, 2.995, 3.000, 3.005, 3.010, 3.015, 3.020, 3.025, 3.030, 3.035, 3.040, 3.045, 3.050, 3.055, 3.060, 3.065, 3.070, 3.075, 3.080, 3.085, 3.090, 3.095, 3.100, 3.105, 3.110, 3.115, 3.120, 3.125, 3.130, 3.135, 3.140, 3.145, 3.150, 3.155, 3.160, 3.165, 3.170, 3.175, 3.180, 3.185, 3.190, 3.195, 3.200, 3.205, 3.210, 3.215, 3.220, 3.225, 3.230, 3.235, 3.240, 3.245, 3.250, 3.255, 3.260, 3.265, 3.270, 3.275, 3.280, 3.285, 3.290, 3.295, 3.300, 3.305, 3.310, 3.315, 3.320, 3.325, 3.330, 3.335, 3.340, 3.345, 3.350, 3.355, 3.360, 3.365, 3.370, 3.375, 3.380, 3.385, 3.390, 3.395, 3.400, 3.405, 3.410, 3.415, 3.420, 3.425, 3.430, 3.435, 3.440, 3.445, 3.450, 3.455, 3.460, 3.465, 3.470, 3.475, 3.480, 3.485, 3.490, 3.495, 3.500, 3.505, 3.510, 3.515, 3.520, 3.525, 3.530, 3.535, 3.540, 3.545, 3.550, 3.555, 3.560, 3.565, 3.570, 3.575, 3.580, 3.585, 3.590, 3.595, 3.600, 3.605, 3.610, 3.615, 3.620, 3.625, 3.630, 3.635, 3.640, 3.645, 3.650, 3.655, 3.660, 3.665, 3.670, 3.675, 3.680, 3.685, 3.690, 3.695, 3.700, 3.705, 3.710, 3.715, 3.720, 3.725, 3.730, 3.735, 3.740, 3.745, 3.750, 3.755, 3.760, 3.765, 3.770, 3.775, 3.780, 3.785, 3.790, 3.795, 3.800, 3.805, 3.810, 3.815, 3.820, 3.825, 3.830, 3.835, 3.840, 3.845, 3.850, 3.855, 3.860, 3.865, 3.870, 3.875, 3.880, 3.885, 3.890, 3.895, 3.900, 3.905, 3.910, 3.915, 3.920, 3.925, 3.930, 3.935, 3.940, 3.945, 3.950, 3.955, 3.960, 3.965, 3.970, 3.975, 3.980, 3.985, 3.990, 3.995, 4.000, 4.005, 4.010, 4.015, 4.020, 4.025, 4.030, 4.035, 4.040, 4.045, 4.050, 4.055, 4.060, 4.065, 4.070, 4.075, 4.080, 4.085, 4.090, 4.095, 4.100, 4.105, 4.110, 4.115, 4.120, 4.125, 4.130, 4.135, 4.140, 4.145, 4.150, 4.155, 4.160, 4.165, 4.170, 4.175, 4.180, 4.185, 4.190, 4.195, 4.200, 4.205, 4.210, 4.215, 4.220, 4.225, 4.230, 4.235, 4.240, 4.245, 4.250, 4.255, 4.260, 4.265, 4.270, 4.275, 4.280, 4.285, 4.290, 4.295, 4.300, 4.305, 4.310, 4.315, 4.320, 4.325, 4.330, 4.335, 4.340, 4.345, 4.350, 4.355, 4.360, 4.365, 4.370, 4.375, 4.380, 4.385, 4.390, 4.395, 4.400, 4.405, 4.410, 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[illegible]

C.B.R.P. PRIMARY PROTOTYPE PUMP
4000 HP - 1116 RPM

LARGE SCALE BREEDER REACTOR
PROTOTYPE MECHANICAL PUMP
18000 HP - 600 RPM

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES TOLERANCES ARE:		DESIGNED BY CHECKED BY DATE		BYRON JACKSON 761-D-0002	
FINISHES:		FINISHES:		FINISHES:	
SURFACE FINISHES:		SURFACE FINISHES:		SURFACE FINISHES:	
HOLE FINISHES:		HOLE FINISHES:		HOLE FINISHES:	
TITLES:		TITLES:		TITLES:	
SCALE:		SCALE:		SCALE:	
WEIGHT:		WEIGHT:		WEIGHT:	
MATERIAL:		MATERIAL:		MATERIAL:	
MANUFACTURER:		MANUFACTURER:		MANUFACTURER:	
ASSEMBLY:		ASSEMBLY:		ASSEMBLY:	
DRAWING NO.		DRAWING NO.		DRAWING NO.	
SK-0002-5		SK-0002-5		SK-0002-5	

REVISIONS			
LTR	DESCRIPTION	DATE	APPROVED
0	ORIGINAL DRAWING

<small>ALL LIMITS OF CONCENTRICITY AND SQUARENESS OF FACES SUBJECT TO BYRON JACKSON PUMP-TIGHT STANDARDS PER S. M. SECT. ON 98.</small> <small>UNLESS TOLERANCE IS GIVEN</small> <small>FINISHES TO WITHIN</small> <table border="1"> <tr> <th>DIMENSION</th> <th>TOLERANCE</th> </tr> <tr> <td>BORER & 1/8"</td> <td>± .005</td> </tr> <tr> <td>TURN - 1/8"</td> <td>± .005</td> </tr> </table>		DIMENSION	TOLERANCE	BORER & 1/8"	± .005	TURN - 1/8"	± .005	<small>UNLESS OTHERWISE SPECIFIED</small> <small>DIMENSIONS ARE IN INCHES</small> <table border="1"> <tr> <th>DIMENSION</th> <th>TOLERANCE</th> </tr> <tr> <td>DECIMALS</td> <td>± .005</td> </tr> <tr> <td>FRACTIONS</td> <td>± .005</td> </tr> <tr> <td>ANGLES</td> <td>± .005</td> </tr> </table>		DIMENSION	TOLERANCE	DECIMALS	± .005	FRACTIONS	± .005	ANGLES	± .005	<small>BY ORDER NO.</small> <small>DATE</small> <small>BY</small> <small>CHECKED BY</small> <small>DATE</small> <small>CONTRACT NO.</small> <small>DESIGN APP'D.</small> <small>DATE</small> <small>CUSTOMER APP'D.</small> <small>DATE</small>		<small>BYRON JACKSON PUMP DIVISION</small> <small>BOYD-HARRIS CORPORATION</small> <small>LOS ANGELES DIVISION</small> <small>DRAWING TITLE</small> -IDROTTEST ASSEMBLY LARGE SCALE SODIUM PUMP <small>CODE IDENT NO.</small> <small>DRAWING NO.</small> SK-0002-33 <small>SCALE</small> <small>1/2</small> <small>WEIGHT</small> <small>SHEET</small>	
DIMENSION	TOLERANCE																				
BORER & 1/8"	± .005																				
TURN - 1/8"	± .005																				
DIMENSION	TOLERANCE																				
DECIMALS	± .005																				
FRACTIONS	± .005																				
ANGLES	± .005

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ANY OTHER PURPOSES.

✓ 2-SPACE ROUGHNESS
PER MIL STD 18

DIM'D HOLES ✓ AS DRILLED
TAPPED HOLES ✓ AS TAPPED

IN. NO SCALE DRAWING
IN. NO. 10-8-85 AND SHARP EN
IN. ALLOY TO 2024-T3
MACHINING DIMENSIONS OF MILL

DESIGN APP'D BY	DATE
TITLE	
CUSTOMER APP'D

BYRON JACKSON® <small>SCHEFFLER COMPANY, INC.</small>			
DRAWING TITLE			
LARGE SCALE SCOUR PUMP DISCHARGE COLUMN ASSEMBLY			
CONTRACT NO.			
CODE IDENT. NO.		DRAWING NO.	REV.
		SK-0002-23	0
SCALE	REVISION	DATE

PUMP SIZE AND TYPE
 LARGE SCALE SODIUM PUMP
 TWO STAGES IN PARALLEL

RPM
 590

CUSTOMER NO.
 4.33 RC-3783
 G. S. 761-G-0002

IMPELLER NO.
 7-30948
 BASED ON
 DATA BY
 W. S.

DATE
 APR. 2, 76
 DRAWN BY
 S. W.

BYRON JACKSON NUMBER
 PC-32847

PUMP SIZE AND TYPE

LARGE SCALE SODIUM PUMP

TWO STAGES IN PARALLEL

RPM

VARIABLE

CUSTOMER NO.

G. S. 761-G-0002

f = 4.33

IMPELLER NO.

RC-3783

BASED ON

T-30948

DATA BY

MA

DATE

26 MAY 76

DRAWN BY

MA

BYRON JACKSON NUMBER

PC 32847-1

LARGE SCALE SODIUM PUMP
TWO STAGES IN PARALLEL

PUMP SPEED TORQUE
CURVE

DATA BY MA

JOB NO
761-G-0002

DRAWN BY. MA

DATE
26 MAY 76

PC-32847-2

15.1 Weights of Pump Sub-Assemblies

	WT. (lbs)
1. Outer Tank	143,000
2. Inner Barrel	27,500
3. Suction Piece and Volute Assembly	66,600
4. Discharge Column Assembly	79,600
5. Rotating Assembly	39,000
6. Reflector and Insulation Assembly	33,000
7. Seal Assembly	6,000
8. Driver Mount Assembly	54,300
Total	449,000 lbs

Sodium Inventory (Sodium filled to top
of discharge column) 118,000 lbs

Axial Thrust Requirements imposed by Pump:

Pump Speed RPM	Axial Thrust	
	* Pounds	Direction
600	59,000	Down
150	40,000	Down
60	39,000	Down

* All thrusts include weight of pump rotating element, 39,000 lbs.

BYRON JACKSON REPORT TCF-1057-STR

ASSESSMENT AND EVALUATION
OF
POTENTIAL HIGH STRESS AREAS
IN
LSBRP PUMP ASSEMBLY

PREPARED FOR

GENERAL ELECTRIC COMPANY
FAST BREEDER REACTOR DEPARTMENT
SUNNYVALE, CALIFORNIA
LETTER CONTRACT 67002X
BJ JOB NO. 761-G-0002

PREPARED BY Dale Gilson / KMA DATE 25 JUN 76

CHECKED BY Goshio Kawai DATE 28 June, 1976

APPROVED BY Kent A. Blum DATE 30 JUN 76

June 1976

BYRON JACKSON PUMP DIVISION
BORG WARNER CORPORATION
LOS ANGELES, CALIFORNIA

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ADDENDA TO THIS REPORT

ADDENDUM A	Universal Analytics Report "Elastic Screening Analysis Pump Tank Discharge Nozzle" dated 16 June 1976
ADDENDUM B	Basic Technology Report - BTI 76-410-001 dated 25 June 1976 Entitled "Large Scale Sodium Pump Upper Bearing Analysis"
ADDENDUM C	Basic Technology Report - BTI 76-410-002 dated 25 June 1976 Entitled "Large Scale Sodium Pump Middle Bearing Analysis"

1.0 SCOPE

This report discusses a preliminary stress evaluation on a conceptual design of a primary coolant pump for a Large Scale Breeder Reactor. It is not intended to be a detailed analysis but only to identify the most critically stressed areas, determine stress levels for these areas and to make recommendations to bring any unacceptable stress levels into conformance with specification requirements.

2.0 SUMMARY OF RESULTS

The most potentially critical areas of the pump were determined to be the area in which the discharge nozzle is located and the middle and upper hydrostatic bearings. These three areas were analyzed in detail as described in later sections of this report. The results in all these cases indicate that there are no insurmountable stress problems inherent in the design presented by Ref. 12.1. More specific results are discussed in the section of the report dealing with each of these critical areas.

3.0 METHOD OF ANALYSIS

The analysis described in this report was performed in several specific steps which are enumerated generally below.

- a) Loading condition definition
- b) Review of Pump Structure for Function and Loading
- c) Categorization of Regions of Pump Structure into potential high stress areas and detailed design considerations.
- d) Assessment of criticality of potential high stressed areas.
- e) Detailed Analysis of the most critical areas to determine if any insurmountable stress problems exist in the conceptual design.
- f) Results and recommendations.

It should be noted that this report only evaluates the conceptual design. Regions for which the design can be modified with minimal effect on the basic design concept were left for evaluation in the detailed design phase of the project.

4.0 LOADING CONDITION DEFINITION

Loads acting on the conceptual design unit consist of thermal transients, pressure, driver torque, and externally generated nozzle loads.

4.1 Thermal Effects

The entire outer tank of this unit is considered to be insulated. Any thermal effects will therefore be due solely to thermal transients occurring in the liquid sodium. These thermal stresses can be categorized into two basic categories, thermal gradient and thermal discontinuity stresses. The magnitude of thermal gradient stress depends strictly on the thermal gradient through the section which is in turn dependent on the exposure to transients and on heat sink effects. Thus, areas where the transients are mitigated and areas where both surfaces are exposed to the transients are considered to be areas of low stress due to thermal gradients.

Thermal discontinuity stresses are those caused by differences in the average temperature through the cross section between adjacent areas. The thermal gradients causing these stresses are therefore not gradients through the wall but, rather, along the wall. They are due primarily to differences in cross section which cause thermal discontinuities or to heat sink effects of thicker cross sections. Such sections are considered to be areas of high

stress and are even more highly stressed when thermal gradient stresses maximize in the same area.

4.2 Pressure Effects

The unit considered in this report is subject to three basic pressures, atmospheric, suction, and discharge. The cover gas pressure is only slightly above atmospheric and is thus considered to be the same as atmospheric. Only imbalances between these pressures are considered to produce significant pressure stresses.

4.3 Driver Torque

The driver torque is developed within the motor and transmitted through the shafting to the impellers. Thus this loading affects only those portions of the unit which are directly involved with the shafting.

4.4 Nozzle Loads

The nozzle loads are externally generated and are applied to the suction and discharge nozzles by the external piping. These are categorized as deadweight loads, thermal expansion loads and OBE seismic loads. These loads are tabulated in Reference 12.2.

5.0 REVIEW OF PUMP STRUCTURE FOR FUNCTION AND LOADING

Various subregions of the unit were next considered and evaluated structurally and possible loading sources were enumerated. This was done on a piece by piece basis except as otherwise noted.

5.1 Outer Tank

The first major component of the unit as shown in Reference 12.1 to be considered was the outer tank which consists of a cylinder with a hemispherical head at the lower end and the pump support flange located at the top. The tank wall is penetrated by an overflow connection located ten feet below the lower face of the support flange, a discharge nozzle located an additional twelve feet eight inches below the overflow connection and a suction nozzle located on the tank centerline. In the area of the discharge nozzle there is a tapered section that is thicker than the remainder of the tank. Also, near the pump support flange, there is a relatively small change in diameter of the cylinder.

The purpose of the outer tank is to contain the liquid sodium used as pumpage. All the sodium within this tank is at suction pressure unless it is contained within another pump structure inside the tank. The sole exception to this statement is in the area of the discharge nozzle. This area corresponds to that portion of the pump tank with the thicker tapered wall. The purpose of this tapered section is to provide a controlled clearance through which discharge pressure in the nozzle is broken down to suction pressure in the

tank. This controlled clearance holds the leakage between these two regions of differing pressures within acceptable limits. The purpose of the suction and discharge nozzle is to permit the sodium to enter and leave the pump. The overflow connection is provided to prevent the sodium level from becoming too high.

The small change in tank diameter near the top of the tank is intended to prevent any direct radiation escape along the inner surface of the tank. The pump tank flange supports the entire pump assembly, by means of holddown bolts passing through the flange into the foundation.

The outer tank is subjected to pressure loading, thermal loading and to nozzle loads. It is not subjected to any loading whatever as a result of motor torque. Pressure loads above the overflow level are due solely to the pressure of the cover gas which is extremely small. Below the overflow level, with the exception of the discharge nozzle and controlled clearance area, the only pressure loading is due to the liquid level within the tank itself. The design pressure for the entire tank is 30 psi as specified by Reference 12.2.

Thermal loading will be primarily a thermal gradient through the thickness due to the essentially uniform tank wall. Thermal transients will have a greater effect below the sodium surface, due to the contact of the sodium with the outer tank. Above the sodium there will be a longitudinal thermal gradient extending to the pump support flange.

Finally, nozzle loads act at both the suction and discharge nozzles. The main load path is through the wall of the outer tank between the two nozzles. Any unbalanced nozzle loads will be transmitted through the upper portion of the tank, through the pump support flange to the anchor bolts and the foundation.

5.2 Motor Mount and Radiation Shield

The next major piece of the unit is the motor mount and radiation shield. This is bolted to the pump support flange and is not subjected to any of the four defined loadings. Note that the exterior of the radiation shield has a diametral offset to complete the barrier to line of sight radiation at the inner surface of the outer tank and that similar provisions are made where the shaft penetrates the radiation shield. This piece is meant to support the pump drive motor and to provide a protective radiation shield where the pump penetrates the permanent radiation shield of the plant. It also supports the labyrinth shaft seal.

5.3 Inner Structure Upper Support

The inner structure upper support consists essentially of a cylinder flanged at both ends. Its purpose is to provide support for the lower portions of the inner structure and also to support the thermal insulation and reflector assembly.

The inner structure upper support is bolted to the lower surface of the radiation shield. The lower end of this structure is below the sodium level. It is subjected only to pressure and thermal loads. Both internal and external pressures are suction pressure and hence

this loading is essentially zero. The thermal loading is minimal due to the uniform cross section, the lack of discontinuities, and mitigation of thermal transients due to mixing with the static sodium volumes.

5.4 Discharge Elbow Assembly

The discharge elbow assembly is a much more complex structure than the name indicates. It consists of a flat plate which includes a support for the upper hydrostatic bearing, a tapered section which interacts with the controlled clearance portion of the outer tank, a flat plate across the lower portion which supports the discharge elbow and the discharge elbow itself. This entire structure is bolted to the lower end of the inner structure upper support.

The discharge elbow diverts the sodium from an upward flow to a horizontal flow and directs it through a circular penetration in the tapered section of the assembly located at the discharge nozzle. Thus it directs the flow out the discharge nozzle into the external piping system. The discharge elbow has a hole cut into it to permit the pump shaft to pass through. By virtue of this hole and the fact that there is no connection between the outlet of the discharge elbow and the tapered section, discharge pressure fills the entire assembly. The upper hydrostatic bearing assembly is supported by a cylindrical support cantilevered from the top plate of the discharge nozzle assembly. This cylindrical support then is internally subjected to discharge pressure. This support and the hydrostatic bearing itself are more fully discussed in Section 5.9 of this report.

The entire assembly is located below the surface of the sodium and is affected by thermal transient and pressure loading only.

5.5 Inner Structure Lower Support

The inner structure lower support is essentially two concentric cylinders with the gap between cylinders closed at both ends. The inner cylinder provides a flow path for the sodium between the hydraulic assembly and the discharge elbow. The gap between cylinders is filled with sodium at suction pressure by virtue of perforations in the outer cylinder.

This structure is subject only to pressure and thermal transient loads.

5.6 Hydraulic Assembly

Suspended from the inner structure lower support is the hydraulic assembly. This is the heart of the pump and consists of two volute pieces and three suction pieces one of which contains a discharge manifold. This assembly encloses two double suction impellers, connected in parallel for minimum NPSH requirements, and includes two additional hydrostatic bearings. This hydraulic assembly is balanced insofar as pressure loadings are concerned and thermal loadings act on most surface walls of each piece simultaneously insofar as it is possible.

5.7 Motor

The drive motor is mounted on top of the motor support stand. It is furnished and designed by others and is not included in this study. It is mentioned here only in the interest of completeness. The motor

is an electric motor with a vertical shaft. It provides the mechanical energy that drives the pump and absorbs any axial thrust generated in the pump. It is connected to the pump shaft by a rigid coupling.

5.8 Pump Shaft

The pump shaft is the means by which the mechanical energy developed in the motor is transmitted between the motor and the impellers in the hydraulic assembly. In essence, most of the pump shaft is a hollow tube. Due to the extreme length and manufacturing limitations, the shaft, excluding that portion within the hydraulic assembly, has been divided into two sections. These sections must be rigidly connected to each other as well as to the portion in the hydraulic assembly.

The upper end of the pump shaft is solid rather than hollow to provide radiation shielding and to permit the offset necessary to eliminate line of sight radiation escape from the unit. Included on one section of this shafting is the upper hydrostatic bearing journal assembly. This journal is discussed along with the remainder of the bearing in Section 5.9 of this report.

The pump shaft is subject to motor torque loadings, to thermal loading and to pressure loading (suction pressure over the entire inside surface, and discharge pressure over part of the outside surface with suction pressure over the remainder).

The portion of the pump shaft within the hydraulic assembly is constructed of several individual parts welded together to form an integral shaft section. These parts include both impellers, a lower hydrostatic bearing

journal, an upper hydrostatic bearing journal and whatever intermediate sections are required to maintain proper spacing between them. This section of shafting is loaded in the same manner as the other portions of the pump shaft. The two hydrostatic bearing journals are discussed in the next section of this report along with the upper hydrostatic bearing.

5.9 Hydrostatic Bearings

There are three hydrostatic bearings within this pump which have the purpose of maintaining alignment and carrying radial loads. The lower and middle hydrostatic bearings are supported by the hydraulic assembly while the upper hydrostatic bearing is supported by a cylindrical support cantilevered from the upper plate of the discharge elbow assembly. The journal of each hydrostatic bearing is mounted on a hollow portion of the pump shaft using two support rings. Thus the journal is larger than the pump shaft with an annular space between the journal and the outer periphery of the shaft. In each case this annular area is connected to the interior of the pump shaft by perforations in the shaft wall. The external surface of the journal is subjected to a pressure somewhere between suction and discharge pressure due to the pressurization of the various pockets of the bearing from a discharge pressure source through orifices. Sodium flow is therefore from the discharge pressure source through the orifices to the journal surface and then along the surface to an area of suction pressure. The bearings are therefore subjected to thermal transient loading, pressure loading and motor torque loading.

6.0 CATEGORIZATION OF REGIONS OF THE PUMP STRUCTURE

Each of the various regions discussed in the previous section were then classified into one of two categories:

- (a) Potentially high stressed area
- (b) Detailed design consideration

If the particular area being considered is categorized as being for Detailed Design consideration, it indicates that modifications can be made during the detailed design phase of the project that will permit the requirements of Reference 12.2 to be met without altering the basic design concept depicted by Reference 12.1. In general, welds and bolted flanges immersed in liquid sodium can possibly result in thermal problems. The placement of welds and the sizing and bolting arrangements of flanges do not affect the basic design concept and hence are deemed items for consideration during the detailed design phase.

The subsequent categorization does not evaluate strain and deformation limits since these are only significant in regions of high primary or secondary stress. The effect of combined creep and fatigue damage and of creep ratcheting are therefore, evaluated only in the detailed analyses of the potentially high stressed areas.

6.1 Outer Tank

In general, the outer tank is a cylindrical structure of uniform thickness. The thickness can be altered to accommodate thermal, pressure and nozzle loads without any effect on the design concept. Hence the basic tank is deemed to involve detailed design consideration only. Specific areas, which are part of the outer tank structure, where this general categorization may not apply are individually discussed below.

6.1.1 Pump Support Flange

The pump support flange together with the anchor bolting and foundation is the reaction point for any unbalanced nozzle loads. It is also the sole support of the entire unit including driver. As such, it is likely to be a highly stressed area. On the other hand, any modification in the design of this flange has no effect whatever on the design concept and hence this is designated for detailed design consideration only.

6.1.2 Tank Offset at Radiation Shield

This offset is located in a section of the pump tank which is subject only to the unbalanced portion of the nozzle loads and to an axial thermal gradient. The offset introduces discontinuities into the load paths which in turn leads to stress concentration and higher stress levels. The discontinuities are small and any modification required to meet the necessary requirements will not alter the design concept so that this offset is designated for detailed design consideration only.

6.1.3 Overflow Connection

This connection is subject to thermal loading, a minimum of pressure loading and unbalanced nozzle loads. There are discontinuities in this area which tend to increase the stress level. However, these are relatively small and tend to keep such increases to a minimum. The actual design of this connection can be modified as necessary to meet any requirements without altering the conceptual design of the unit.

This connection is therefore designated for detailed design consideration only.

6.1.4 Discharge Nozzle

This nozzle involves thermal, pressure, and nozzle loads. The structural discontinuities are among the largest in the unit. This is therefore an area of potentially high stress and will be further discussed in Section 7.1 of this report.

6.1.5 Suction Nozzle

The same reasoning that was applied to the discharge nozzle applies to this nozzle. It is therefore similarly categorized for further discussion in Section 7.1 despite the fact that the structural discontinuities here are less severe than for the discharge nozzle.

6.1.6 Controlled Clearance Area

The controlled clearance area is also an area of gross structural discontinuity. Subject to both pressure and thermal loads, it must also be categorized as an area of potentially high stress and discussed further in Section 7.3

6.2 Motor Mount and Radiation Shield

In general, this component is not subjected to any external load. The specific design can vary greatly without affecting the conceptual design and the entire component, including the labyrinth seal it supports, is designated for Detailed Design consideration only.

6.3 Inner Structure Upper Support

The inner structure upper support supports the thermal insulation and reflector assembly and is essentially free from structural discontinuities. Since the design of the inner structure upper support as well as the thermal insulation and reflector assembly can be greatly modified without affecting the basic design concept, these items are considered to be in the Detailed Design Category.

6.4 Discharge Elbow Assembly

As previously indicated, this is a fairly complex structure and involves a number of structural discontinuities. Accordingly, in general this assembly must be initially considered to be an area of potentially high stress. By consideration of specific portions of the assembly, however, it is possible to eliminate them from this category and to more specifically pinpoint the areas of maximum concern. These considerations appear in the following sections.

6.4.1 Discharge Elbow

The discharge elbow is of uniform cross section and is essentially free from discontinuities. In addition it is only fixed at one end with the other end free. This has the effect of reducing the stresses in this portion of the assembly. It is subject to the same pressure both internally and externally thus making any stresses due to pressure loading essentially zero. Thermal loading will be from both sides of the wall thus minimizing these stresses. The discharge elbow, except for the area where it is attached to the remainder of the assembly, is immediately categorized as a detailed design consideration.

6.4.2 Flat Plate at Top of Discharge Elbow Assembly

This plate is subjected to a differential pressure loading, and to thermal loading. By virtue of its shape and size, stresses could be high. On the other hand its general shape and size has no impact on the design concept. This plate should receive careful attention in the detail design phase of this project, particularly at the inner and outer edges of the plate where structural discontinuities exist.

6.4.3 Tapered Section of Discharge Elbow Assembly

The tapered section of the discharge elbow assembly matches the taper

of the thicker portion of the outer tank. It is subjected to pressure and thermal loads. In addition, sodium discharging from the discharge elbow causes a lateral reaction which in turn imposes a horizontal force on this particular piece. The location of this force is between 22 and 23 feet below the lower surface of the pump mounting flange where the ultimate reaction occurs. The load path between the point of application of this force and the anchor bolting where it is reacted passes through the inner structure upper support, the radiation shield and the pump support flange. This force also tends to open the gap between the tapered section of the discharge elbow assembly and the discharge nozzle thus increasing the leakage between the main discharge flow path and the outer tank suction pressure areas. This taper section together with the controlled clearance section of the outer tank and its matching taper maintain the gap at the discharge nozzle side within acceptable leakage limits determined from hydraulic considerations. This also permits contact on the side opposite the discharge nozzle and provides an additional load path for this hydraulic force through the outer tank to the anchor bolts which reduces the load on the inner structure upper support. Clearance between the outer tank and this tapered section is controlled by adjusting the bolting between the motor mount and the pump support flange. This controlled clearance section therefore is an area which requires further evaluation as an area of potentially high stress.

6.4.4 Lower Plate of Discharge Elbow Assembly

This plate is bolted to the inner structure lower support and will be categorized in Section 6.5.

6.5 Inner Structure Lower Support

In general, this piece consists of two concentric cylinders. The cross sections are uniform and the entire piece may be redesigned without impact on the conceptual design. Accordingly this piece is classified as involving detailed design consideration only. There are however, areas where discontinuities exist which may alter this general classification.

6.5.1 Upper and Lower Flanges

Both the upper and lower flanges of the inner structure lower support involve structural discontinuities. In addition, the upper flange is bolted to the lower plate of the discharge elbow assembly and the lower flange is bolted to the discharge manifold of the hydraulic assembly. Both of these flanges introduce additional structural and thermal discontinuities which can result in high stress levels. On the other hand, all of the bolting and flange arrangements are subject to review and modification in the Detailed Design phase of this project as has already been indicated in Section 6.0. These two connections are therefore placed in the Detailed Design category with specific attention directed to their design.

6.6 Hydraulic Assembly

Due to the thermal and pressure load balance inherent in the present design, the hydraulic assembly, excluding the hydrostatic bearings, is deemed, generally, to involve detail design considerations only. It is apparent that there are a large number of discontinuities in this assembly but the transient effects are mitigated by the inherent thermal balance and by the symmetry of the component.

Within the hydraulic assembly, there are several areas for concern which are discussed individually in the following paragraphs.

6.6.1 Center Suction Piece

The center suction piece is of concern due to the possibility of thermal discontinuity stresses between the large volume of material in the immediate vicinity of the pump shaft and the bolted joint. Both the pressure loading and the thermal loading are balanced which minimizes the pressure stresses and the thermal gradient stresses but the difference in the amount of material will set up thermal discontinuity stresses which can be high.

6.6.2 Lower Suction Piece

The lower suction piece and lower volute piece are joined with a bolted flanged joint. There are a number of relatively large sections in this area with resulting high thermal discontinuity stresses. While these stresses are minimized by the mitigation of the transient due to mixing in the suction nozzle area and by exposure of all possible surfaces to the transient effects, the stresses may still be high enough to cause problems. The geometry of the flange and bolting has already been categorized as a detail design consideration but it is an area of concern.

6.6.3 Upper Suction Piece

The upper suction piece and upper volute piece are joined with a bolted joint similar to that discussed in Section 6.6.2. As a result, the same considerations are applicable as were pointed out there. In addition, the upper suction piece has another bolted joint where the discharge manifold is connected to the inner structure lower support. This joint is subject to similar considerations.

6.6.4 Risers

The lower volute has six discharge risers which connect to six similar risers on the upper volute. In addition, the six discharge passages in the upper volute connect to these same six risers resulting in a complex joint which is far from symmetrical. This asymmetry can result in thermal stress problems which makes this an area for concern. Again, the transients themselves are mitigated and are applied to both wall surfaces of these passages which tends to minimize thermal stress problems. This must also be considered an area of concern.

6.6.5 Clearances

The hydraulic assembly, by virtue of the fact it is basically a centrifugal pump, must have relatively close running clearances. Accordingly, overall deformation must be evaluated whether it be due to creep effects because of the relatively high operating temperatures or due to deflection from other loads. This overall deformation can be considered cause for concern.

6.7 Motor

The motor is not included in this study as was stated in Section 5.7. The connection between the motor shaft and pump shaft is a rigid coupling which does not affect the design concept and can therefore be relegated to the detailed design phase of the project.

6.8 Pump Shaft

The pump shaft, other than in the areas where the hydrostatic bearings and the couplings are located is of uniform cross section and is symmetrical. It can be readily modified to meet the requirements of Reference 12.2 and is therefore deemed to involve detailed design considerations only.

6.9 Hydrostatic Bearings

In view of the structural discontinuities and close running clearances present in each of the bearing areas, these areas are considered to be potentially high stressed areas and are reviewed further in Section 7.2.

6.10 Summary of Potential High Stress Areas

The preceding portions of this section have reviewed all portions of the conceptual design as depicted by Reference 12.1. Most of the areas reviewed were categorized as detailed design phase considerations. The remaining seven regions were categorized as potential high stress areas which would require additional review and analysis. These seven areas are summarized in the following list and are discussed further in Section 7.0

1. Discharge Nozzle Area
2. Suction Nozzle Area
3. Controlled Clearance Area
4. Hydraulic Assembly
5. Upper Hydrostatic Bearing
6. Middle Hydrostatic Bearing
7. Lower Hydrostatic Bearing

7.0 ASSESSMENT OF CRITICALITY OF POTENTIAL HIGH STRESS AREAS

The seven potentially high stressed areas determined in the preceding section of this report were then reviewed for degree of criticality. That review is described in more detail in the remaining portions of this section.

7.1 Suction and Discharge Nozzles

In view of the similarity of these two nozzles, they are discussed together.

Both nozzles are subject to the same thermal transients. Due to the structural discontinuities present, thermal discontinuity stresses can be expected to be more significant than stresses due to thermal gradients across the wall. The structural discontinuities around the discharge nozzle are more pronounced than those around the suction nozzle hence the thermal stresses in the discharge nozzle are anticipated to be considerably higher than those in the suction nozzle area.

Pressure loadings on the two nozzles are not the same. The discharge nozzle is subjected to discharge pressure which is considerably higher than the suction pressure to which the suction nozzle is subjected. Further, the discharge nozzle is located in a cylindrical section of the outer tank while the suction nozzle is located in a spherical section. All these facts imply that the tank stresses at the discharge nozzle will be higher than those at the suction nozzle. As a result, it is obvious that if the stresses due to nozzle loads are nearly equivalent in both nozzles or if those in the suction nozzle are less than those in the discharge nozzle, the latter will be more critical.

The mechanical loads for the two nozzles as listed in Reference are the maximum that can be applied based on a plastic hinge located at the first elbow in the piping system external to the pump. By inspection, the loading at the suction nozzle is higher than that at the discharge nozzle. Because of this fact, the discharge nozzle cannot be immediately considered to be more critical than the suction nozzle and an assessment of the stresses due to nozzle loads is required.

7.1.1 Assessment of Nozzle Loading Levels

The nozzle loadings tabulated in Reference 12.2 indicate that the suction nozzle is the most heavily loaded nozzle. The most critical nozzle due to the other loadings has already been demonstrated to be the discharge nozzle. If the stress levels due to these loads are approximately the same for both nozzles, the discharge nozzle can be considered to be the more critical nozzle. Accordingly an assessment of stress levels was made. The results are presented in the following paragraphs.

7.1.2 Axial Stress Levels

The most critical stress component was considered to be the axial component in that it involves pressure stresses, stresses due to axial forces, and axial stresses due to the imposed bending moment. A representative axial stress level can be determined from

$$\sigma = (PR/2T) + (Mc/I) + (F/A)$$

where the three terms involved are self-explanatory.

The results of evaluation of these loads are presented in Table 7.1

Table 7.1 Nozzle Axial Stress Levels

Nozzle	PR/2T	Mc/I	F/A	Total
Suction	525	4881	430	5836
Discharge	2600	2759	141	5500

The axial stress levels are therefore approximately equivalent.

7.1.3 Torsional Stress Level

The torsional stress is given by the self explanatory relationship

$$\sigma = MR/I$$

Using the values given in Reference 12.2, the results show low stress levels for both nozzles. Again the levels are approximately equivalent being 105 psi and 157 psi for the discharge and suction nozzles respectively.

7.1.4 Shear Stress

The average shear stress across a section can be determined from the relationship

$$\sigma = F/A$$

The maximum shear stress can vary significantly from this value but due to the similarity in the cross sections of the two nozzles being compared, the variations will be similar. This, therefore, is an acceptable means of determining the relative shear stress levels.

Utilizing this relationship, the average shear stress in the suction nozzle is 386 psi while in the discharge nozzle it is 281 psi. Again, due to the low stress levels involved, these can be considered to be approximately equivalent.

7.1.5 Nozzle Criticality Assessment

The two nozzles are obviously loaded, as a result of nozzle loading, to about the same level. The discharge nozzle is more critical due to thermal and pressure loading. Further, if the suction nozzle area develops significant stress problems in the detailed design phase of this project, its geometry can be altered fairly easily without altering the basic design concept. Due to the necessity for the controlled clearance in the vicinity of the discharge nozzle, this statement cannot be made with respect to the discharge nozzle. The discharge nozzle is therefore the more critical of the two and only this nozzle will be considered for a detailed analysis. The suction nozzle can therefore be categorized as a detailed design consideration.

7.2 Hydrostatic Bearings

There are three hydrostatic bearings in the conceptual design unit. Each is subjected to pressure, thermal transient, and motor torque loading. From the standpoint of the motor torque loading, the full motor torque, less shafting losses, must be transmitted through both the upper and middle bearings. Most of this torque loading is used by the impellers to impart velocity to the sodium so that the torque loading on the lower bearing is minimal.

The middle and the upper bearing are subjected to suction pressure at one end of the journal and discharge pressure at the other end. The lower bearing has suction pressure appearing at both ends of the journal so that the pressure loading is less on the lower bearing than on the upper two bearings.

All three bearings are subjected to the same thermal transients. Since the support construction of the two lower bearings is essentially the same, if the middle bearing proves satisfactory, minor design changes can make the lower bearing satisfactory if such changes become necessary. The lower bearing therefore becomes a detailed design consideration.

The construction of the upper bearing is somewhat different because its support is cantilevered from the upper plate of the discharge nozzle assembly. This bearing should be analyzed separately to investigate possible loss of clearance at the extreme end due to thermal distortion.

7.3 Controlled Clearance Area

Another area of potentially high stress is the controlled clearance area. This is a region where the reaction to fluid flow through the discharge nozzle causes the inner structure to deflect and contact the outer tank. The magnitude of this reaction governs the primary stress level in this particular region. Any increase in the primary stress level results in a corresponding decrease in the allowable secondary stress level. Since this area is subject to thermal shock due to the transients, the secondary stress level which is permitted is an important consideration. The deflection due to this reaction also causes an increase in the gap at the discharge nozzle with resulting increased leakage which is detrimental to pump hydraulic performance. These are not considered critical since creep effects are not significant during normal operation at 850°F and only minimal holding periods occur at higher temperatures.

7.4 Hydraulic Assembly

The hydraulic assembly is basically categorized as a detailed design consideration. In view of the number and type of discontinuities it contains, it must rank high on the list of areas to be analyzed in the detailed design phase of this project. Because of the mitigated transient effect and the exposure of a maximum number of surfaces to these transients, thermal effects are minimized and this assembly is not considered to involve critical stresses.

7.5 Relative Degrees of Criticality

Based on the above discussions, it is felt that for the seven regions of potential high stress, the following is the order of criticality with the first on the list being most critical.

1. Discharge Nozzle Area
2. Upper Hydrostatic Bearing
3. Middle Hydrostatic Bearing
4. Hydraulic Assembly
5. Controlled Clearance Area
6. Suction Nozzle Area
7. Lower Hydrostatic Bearing

The three most critical areas as taken from this list were analyzed further and discussions on these three specific areas make up most of the remainder of the report. If these three areas will meet specification requirements, with or without modification, the intent of this analysis will have been fulfilled in that the conceptual design does not pose any unsolvable stress problems.

Recommendations concerning the fourth and fifth items are included in Section 11.0 of this report. The last two items have been categorized as detailed design considerations and no further mention is made of these areas.

8.0 DISCHARGE NOZZLE ANALYSIS

The pump tank discharge nozzle was determined to be the most critical area in the design concept. Accordingly, the nozzle area was subjected to a detailed analysis. This analysis was performed by Universal Analytics, Inc. and the final report on this analysis is included as Addendum A to this report.

It should be noted that the analysis presented in Addendum A was performed using Revision 0 of Reference 12.1. The differences between the original version of this reference and the version used for the remainder of the report are considered to be insignificant and accordingly to have a negligible effect on the results obtained.

The analysis presented in Addendum A indicates that there is only one area in the discharge nozzle which can be considered a potential problem area due to creep ratcheting strain. It points out that the use of elastic results in a creep ratcheting analysis is extremely conservative. Considering this inherent conservatism and the location of the local region affected, no insurmountable stress problems are anticipated and further work is relegated to a detailed design consideration.

8.1 General Description of Nozzle Analysis Performed

In general, the nozzle analysis was performed using the NASTRAN computer program. It involved both heat transfer analysis and stress analysis prior to evaluation.

8.1.1 Heat Transfer Analysis

The IP-3U and IP-1U transients were analyzed to provide a time history for each. These were examined and the times of peak gradients used to establish four thermal maps for later use in the stress analysis.

8.1.2 Shear Analysis

The same model was used for both the heat transfer and stress analyses. For the stress analysis, thermal loadings as described by the four thermal maps were imposed as well as piping and pressure loads.

8.1.3 Code Evaluation

The stress results were superimposed as necessary for Code evaluation. Load control stresses, use fraction sums and elastic strains were all evaluated and compared with code specified limits. All but the elastic strain limits were satisfied. This necessitated evaluation of creep ratcheting and creep fatigue strain. Comparison of these values with the code specified limits indicated a completely satisfactory design except for the one area already mentioned.

9.0 UPPER HYDROSTATIC BEARING ANALYSIS

The upper hydrostatic bearing was also considered to be one of the most critical areas of the unit. It was accordingly carefully analyzed by Basic Technology, Inc. and the resulting report is included as Addendum B to this report. This analysis indicates that the bearing, as presently designed appears to be satisfactory in all respects. It does point out that a more detailed analysis will be required after specifications are finalized but anticipate no difficulty in meeting all requirements.

In general, this analysis follows the same procedure used in the Nozzle Analysis. The thermal loading is applied first to determine temperatures at times of peak thermal loads. The appropriate loads are then applied to determine stresses under load conditions. Finally, evaluation according to Code Criteria is accomplished. In this instance, due to the critical nature of the bearing clearances, the effects on bearing clearances were also investigated as well as the effect of deformation on functional operability. In no case was there any indication of a design deficiency.

10.0 CENTER HYDROSTATIC BEARING ANALYSIS

The center hydrostatic bearing was considered to be one of the most critical areas of the unit. It was analyzed by Basic Technology, Inc., in a manner similar to that used for the upper hydrostatic bearing. The complete report of this analysis is included as Addendum C to this report. The results of this analysis were similar to those acquired from the analysis of the upper hydrostatic bearing. No design deficiency was noted and a more detailed analysis after the specifications have been finalized is not anticipated to disclose any unsolvable problems.

11.0 CONCLUSIONS AND RECOMMENDATIONS

As a result of the analyses performed and discussed in this report, it can be concluded that the conceptual design presented in Reference 12.1 will result in no unsolvable stress problems.

Only one specific area was noted as being a potential problem area. This occurs partially as a result of the overly conservative analysis method and no insurmountable stress problems are anticipated. Further elastic analyses of alternative nozzle-piping transitions is, however, recommended in the Detailed Design phase.

Two other areas were categorized as areas where high stresses might be found. The magnitude of these stresses was deemed to be such that they were not considered to be critical. Accordingly, no attempt has been made in this report to analyze the two areas. It is specifically recommended that these two areas receive careful consideration in the detailed design phase of this basic design concept. These two areas are the hydraulic assembly and the controlled clearance region.

In addition, two suggestions were made by BTI as to ways in which the bearing design can be improved. These suggestions were:

1. Decrease the volume of sodium above the top plate of the discharge elbow assembly to promote more efficient mixing in this area.

2. Increase the volume of the essentially static sodium within the discharge elbow assembly but outside the discharge elbow itself. This will tend to mitigate the thermal transient effect on the bulk sodium in this area prior to impingement on the bearing surface.

Implementation of these suggestions can be accomplished by replacing the flat plate at the top of the discharge elbow assembly with a dished head. This geometry would also serve to decrease the large pressure stresses anticipated in the present plate geometry as well as to stiffen the upper hydrostatic bearing support.

12.0 REFERENCES

- 12.1 Byron Jackson Conceptual Design Drawing 1F-7937
Rev. A dated 24 May 1976
- 12.2 General Electric Specification 22A3994 Rev. 2
dated 6 Feb. 1976

ELASTIC SCREENING ANALYSIS
PUMP TANK
DISCHARGE NOZZLE

Prepared for
BYRON JACKSON PUMP DIVISION
Vernon, California

June 16, 1976

Prepared by
UNIVERSAL ANALYTICS, INC.
7740 West Manchester Boulevard
Playa Del Rey, California 90291
(213) 822-4422

ABSTRACT

This report documents a preliminary elastic screening analysis of the discharge nozzle on the Large Scale Breeder Reactor Pump as performed by Universal Analytics, Inc. (UAI) for Byron Jackson Pump Division, Vernon, California. Of particular interest were the detailed stresses in and about the nozzle transition to the pump tank caused by mechanical and thermal transient loadings.

These analyses were carried out in two parts. First the transient heat transfer analyses were performed to define the critical thermal loading conditions. Then, with the same structural finite element model, the thermal and mechanical loads were applied to obtain the detailed stresses. The NASTRAN computer program was used to analyze the 3-D solid element axisymmetric model for both the symmetric and antisymmetric loading conditions, using cyclic symmetry.

The detailed stresses were then used to perform the desired elastic screening analysis according to ASME Boiler and Pressure Vessel Code, Section III, requirements including Code Case 1592 and Supplement RDT F9-4 criteria. This screening analysis included evaluation according to primary stress, strain and creep-fatigue damage limits.

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1.0 INTRODUCTION

The purpose of this report is to document a preliminary elastic screening analysis of the Large Scale Breeder Reactor Pump Discharge Nozzle. This effort required both a transient heat transfer analysis and a detailed stress analysis. In addition to four thermal loadings, a set of piping loads and an internal pressure load were applied. A possible "interference" load between the sealing ring and the tank at the level of the discharge nozzle due to temperature differential was found to be negligible.

Included in this report are the analysis results for the pressure, piping, and thermal transient loadings to be used as data for determining compliance of the design with requirements of the ASME Boiler and Pressure Vessel Code, Section III, including supplement Code Case 1592 and RDT F9-4T [Ref. 1]. Specific attention is given in this report to the evaluation of screening limits, strain limits for functional requirements and structural integrity, and creep-fatigue limits, as outlined in the BJPD Request for Proposal [Ref. 2].

Figure 1.0-1 shows a cross-section of the overall pump tank and discharge nozzle. For efficiency of computation, a finite element computer model of an assumed axisymmetric nozzle configuration was produced to study both heat transfer and detailed stress behavior. The NASTRAN (NASA Structural Analysis) program was employed to execute these analyses using concepts of cyclic symmetry (see Appendix A) to provide for processing of the nonsymmetric loading conditions. The same basic model, used throughout, was refined to more fully represent the heat flow phenomena. This model was constructed assuming an effective sphere to represent the thickened segment of the tank which was considered to be the primary load path of interest. A full description of the model is presented in Appendix B.

Figure 1.0-2 presents a computer plot of the finite element model used for the stress analysis. Solid three-dimensional linear strain elements were employed to obtain detailed stress information through the cross-section. The grid density was increased in the area of expected high stresses. A summary of results and the conclusions from this screening analysis are presented in Section 2.0.

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FIGURE 1.0-1 CROSS-SECTIONS OF PUMP TANK
AT DISCHARGE NOZZLE

FIGURE 1.0-2 COMPUTER PLOT OF FINITE ELEMENT MODEL

Analysis methods and details of the finite element model, material properties and loadings are presented in Section 3.0. Included also are descriptions of the features of NASTRAN used and the element stress calculations and assumptions used to obtain the ASME code stress intensities.

The detailed results for both the thermal and the stress analyses are presented in Section 4.0. The corresponding workbook computations showing the linearized stress curves for each loading condition are provided in Appendix C. These data are then used in the evaluation of stress results according to ASME code criteria as presented in Section 5.0.

Section 6.0 contains the pertinent references. Appendix A contains excerpts from the NASTRAN Theoretical Manual describing the features of cyclic symmetry used in the stress analyses. Appendix B presents all the details of the finite element models used. Appendix C contains the stress calculations used and discussed in Sections 4.0 and 5.0 of this report. Appendix D describes the utility computer programs developed to facilitate in the data reduction effort.

2.0 SUMMARY AND CONCLUSIONS

The preliminary elastic screening analysis of the LSBR pump discharge nozzle as documented with this report is summarized here. The first section presents an overview of the analyses and of the assumptions made. The second section summarizes the results of the evaluations made in accordance with ASME Boiler and Pressure Vessel Code, Section III including Code Case 1592 and Supplement RDT F9-4 criteria. The final section offers the conclusions to be drawn from these results.

2.1 SUMMARY OF ANALYSES

This screening analysis was performed in two parts using the NASTRAN (NASA Structural Analysis) computer program for both. First a thermal transient analysis was conducted using two transients, IP-1UT as a down transient and IP-3UT as an up transient. The same finite element model was used here as was to be used in the subsequent stress analysis. To this model was added the effects of the sealing ring cavity and the liquid sodium within, above and below that cavity. The portion of the sealing ring not in contact with the tank was modeled in detail along with the liquid sodium in the gap. The remainder of the system was approximated by a simplified line element model. For conservative stress results, the flow through the support bearing was neglected, thus maximizing temperature differences between the tank and the nozzle.

The results showed, as was expected, that the liquid sodium offers a strong damping effect on the temperature variations in the tank away from the nozzle. The temperature difference between the sealing ring and the tank in the contact zone away from the nozzle never exceeded 15°F.

Temperature differences were monitored both across and along the nozzle to its zone of transition to the tank. Based on these results, the times of peak gradients were isolated for establishing thermal maps to be used in stress analysis. Attention was given to monitoring conditions conducive to both surface stress and global structural discontinuity stresses. Four such thermal maps were selected, three from the long IP-3UT transient and one from the short IP-1UT transient. These provided for an estimate of maximum stress range for all transients to be considered for code evaluation.

For the stress analysis, the discharge nozzle and tank structure was approximated by an axisymmetric sphere with a radially intersecting nozzle. In order to accommodate the unsymmetric piping loads on this axisymmetric model, the "Cyclic Symmetry" capability of NASTRAN was used. In order to obtain detailed stresses through the critical cross-sections, a refined solid 3-D element model was used. Such a model was also considered ideal for use in the thermal transient analyses described above.

Noting that the piping loads could be represented with only a constant, a sine and a cosine force distribution about the nozzle and the cyclic symmetry approach was ideal. Five other loading conditions were also examined: one internal pressure and four thermal. A possible "interference" load due to differential thermal expansion between the sealing ring and tank was found to be negligible.

These stress results were then used for ASME Code evaluation as summarized next.

2.2 SUMMARY OF CODE EVALUATION

The detailed stresses obtained for each of the six loading conditions analyzed were linearized to obtain equivalent membrane and bending stress intensities. As required by the ASME Code, different combinations of these stresses are required according to the conditions being examined. For these purposes, the bending stress component of the mechanical loads was assumed to be a secondary stress caused by the gross structural discontinuity at the nozzle/tank interface. Also, the results for both mechanical loads were superimposed on the thermal load results even though the internal pressure drops during the two transients investigated.

The results for each step in the code evaluation are presented in Table 2.2-1. As can be seen, only one possible critical area was found. This region of concern is on the inside surface of Cut 2 on the nozzle at the beginning of the transition to the tank. Here, the limits for creep ratcheting were exceeded.

The load control stress limits and the use-fraction sum were satisfied on all sections. The elastic strain limits were exceeded as expected on all sections except on Cut 6 which represents the weld to the tank. Except as

TABLE 2.2-1 CODE CASE 1592 EVALUATION SUMMARY

Section		Load Control Stress Limit			Use-Fraction Sum	Elastic Strain Limit	Creep Ratcheting Strain		Creep-Fatigue Evaluation		
Cut	Location	Design P_m	Normal P_m	Upset P_m	$\sum_i \left(\frac{t_i}{t_{im}} \right) \leq B$	$x+y \leq S_a/S_y$ or $x+y \leq 1.0$			$\sum_j \left(\frac{n}{N_d} \right)_j$	$\sum_k \left(\frac{t}{T_d} \right)_k$	Sum $\leq D$
1	Inside	Meets	Meets	Meets	Meets	Did Not Meet	<1%	Meets	<10 ⁻³	0.026	Meets
	Outside					"	<1%	"	<10 ⁻³	"	"
2	Inside	"	"	"	"	"	>1%	Not Met	<10 ⁻³	"	"
	Outside					"	<1%	Meets	<10 ⁻³	"	"
3	Inside	"	"	"	"	"	<1%	"	<10 ⁻³	"	"
	Outside					"	<1%	"	<10 ⁻³	"	"
4	Inside	"	"	"	"	"	<1%	"	<10 ⁻³	"	"
	Outside					"	<1%	"	<10 ⁻³	"	"
5	Inside	"	"	"	"	"	<1%	"	<10 ⁻³	"	"
	Outside					"	<1%	"	<10 ⁻³	"	"
6	Inside	"	"	"	"	Meets	<0.5%		<10 ⁻³	"	"
	Outside					Meets	<0.5%		<10 ⁻³	"	"

noted above, both the creep-ratcheting and creep-fatigue strain limits were also satisfied.

2.3 CONCLUSIONS

This preliminary elastic screening analysis indicates only one area of the nozzle to be a potential problem as noted above. This problem, however, may not be critical. The creep-ratcheting analysis is extremely conservative, as it is based on elastic results. Also, if the stresses due to internal pressure were not to be included as active during the thermal transients, the stress levels in the region of concern would drop sufficiently to meet code at this location. Finally, only 15 events are expected for the IP-3UT transient. Therefore, the overall design analyzed appears to present no serious problems.

Upon review of the overall structural deformations, attention was given to the concern for leakage between the sealing ring and the tank at the discharge nozzle. The combined mechanical loads indicate an 8.5 mill increase in gap due to tank deformation alone. To this should be added the deformation of the sealing ring, including the stiffening effects of the tank/sealing ring interaction.

3.0 ANALYSIS METHODS

The analysis methods used in this screening analysis are discussed below. The NASTRAN program is described. The thermal model and the stress model are defined. Finally, the code stress intensity calculations are presented.

3.1 THERMAL AND STRESS ANALYSIS WITH NASTRAN

The NASTRAN structural analysis program was used for this analysis. NASTRAN is a comprehensive finite element computer program for thermal and structural analysis, and is widely accepted by industry users [Ref. 3, 4].

Based on the finite element displacement method [Ref. 5], NASTRAN allows study of heat transfer and structure behavior using models composed of discrete grid points interconnected by elementary structural elements (e.g., beams, plates, rods, solids, etc.) to simulate the actual structure. Solid 3-D constant strain elements only were used in this discharge nozzle analysis. NASTRAN also provides computer plots of the element connectivities to help the user verify his model.

The general 3-D solid elements of NASTRAN (HEXA2 and WEDGE) were selected because they were designed to simulate the general deformation behavior of arbitrary structural cross-sections. The NASTRAN program is implemented to use the same finite element model to perform both heat transfer transient analysis and elastic stress analysis. For this project, additional modeling capabilities of NASTRAN were employed to extend the heat transfer model to include the properties of the liquid sodium in the discharge nozzle in the sealing ring cavity, and in the reservoir above and below the sealing ring as well as the sealing ring itself. These features include surface boundary layer conductance and equivalent lumped capacitance elements.

Loads can be applied to the model in a variety of ways. For heat transients, temperatures can be specified as a function of time. For stress analyses, mechanical loads can be defined as surface pressures or grid point forces. For thermal stress calculations, the grid point temperatures obtained during the thermal transient analysis at specified points in time can be applied directly to the stress model, an advantage of using the same finite element model for both analyses.

The results printed by NASTRAN vary with the type of analysis performed and according to the specific elements used in the model. The thermal transient solution provides temperature time histories at each grid point. The stress analysis provides grid point displacements, support reaction forces, and element stresses according to the elements used. The 3-D constant strain solid element stresses include all six stress components (direct and shear), the von Mises octahedral shear stress, and the first invariant (average pressure) stress. These stresses are output, not in a local element system but relative to the basic coordinate system of the overall model. The transformation of these stresses to a local coordinate system for selected cross-sections through the model was done outside of NASTRAN. These transformations are described in Section 3.4 following the detailed description of the thermal and stress models presented in Sections 3.2 and 3.3, respectively.

The actual analyses were performed using the unique cyclic symmetry capabilities of NASTRAN. Because the model selected was assumed to be axisymmetric but the loadings were not, the cyclic symmetry approach would be ideally suited. The theory taken from the NASTRAN Theoretical Manual [Ref. 3] and presented in Appendix A, provides for representing any uniformly repetitive model (e.g., fan, pump impeller, longitudinally reinforced shells, or any axisymmetric structure) as a series of identical petals which repeat symmetrically about an axis. An entire structure can be represented by preparing the model for a single petal only. Arbitrary loads are represented by a finite series sum of harmonics to produce exact solutions with increased computer efficiency, the primary advantage of using cyclic symmetry.

The model for the thermal analysis is described next, followed by the presentation of the corresponding stress model.

3.2 THERMAL MODEL

The thermal model selected was designed to be used also as the stress model (see Section 3.3). Using the same model for both analyses improves the reliability of data transfer from the thermal transient results to the input loadings for the critical thermal stress calculations. Figure 3.2-1 shows the basic thermal model, including the solid element representation of the sealing ring. Figure 3.2-2 shows the simplified model (see details in

$$h_A = 297 \text{ Btu/hr ft}^2 \text{ } ^\circ\text{F}$$

$$h_B = 55 \text{ Btu/hr ft}^2 \text{ } ^\circ\text{F}$$

$$h_C = \infty$$

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BJPD GEOMETRY CHECKOUT

UNDEFORMED SHAPE

FIGURE 3.2-1 THERMAL MODEL

Note: Points A, B and C are shown on Figure 3.2-1.
Detail auxiliary model is presented in Appendix B.

FIGURE 3.2-2 AUXILIARY THERMAL MODEL SCHEMATIC

Appendix C) constructed to obtain the gross thermal response of the liquid sodium and structural components not included in the detail model of Figure 3.2-1.

The structural model (see Appendix C) includes 6 inches of the nozzle (1 inch thick) which extends beyond the radius of transition to the tank material. The tank material is represented as a sphere to allow the entire model to be axisymmetric. The 5-inch thickness selected for the tank material represents the average thickness of the tapered section taken at a horizontal plane containing the centerline of the discharge nozzle.

The full thermal model represents one of 240 identical petals, 1.5° arc each. This one petal comprises:

- 1074 - Grid points
- 488 - 3-D Solid elements (Structure)
- 30 - 3-D Solid elements (Liquid Na)
- 82 - Surface conduction elements

Figure 3.2-1 shows how these elements were distributed to obtain refined thermal gradients in the region of primary concern, the transition from the nozzle to the tank. The sealing ring is assumed to be in contact with all but 120° of the tank, 60° on each side of the discharge nozzle centerline. The gap between the tank and the sealing ring was filled with the 30 liquid sodium elements to complete the heat flow path. The sodium conductance was multiplied by 2.5 to approximate the effects of turbulence in that gap.

Referring to Figure 3.2-2, the thermal loading for the IP-3UT up-transient and for the IP-1UT down-transient was imposed on grid point 8001 as a heat source (Figure 3.2-3, 4). In the thermal model, that point was given a very large capacitance. With a correspondingly large heat input, therefore, the desired temperature of the discharge sodium would be obtained. As the transient progressed, the heat would be conducted directly to the inside surface of the discharge nozzle and indirectly, through the entrapped sodium, to the inside of the sealing ring, through the sealing ring, and then into the tank itself. The leakage effects of the flow from the discharge elbow out through the bearing was neglected. This conservative assumption would provide for a maximum temperature differential between the tank and the nozzle.

FIGURE 3.2-3 IP-3UT UP TRANSIENT

FIGURE 3.2-4 IP-1UT DOWN TRANSIENT

The outside boundaries of the tank were assumed to be insulated ($h = 0$). Figure 3.2-2 includes the boundary film conductance coefficients provided by BJPD for the inside surface of the nozzle and the inside surface of the sealing ring. The latter is lower due to the very low flow rate assumed inside the sealing ring cavity. The remaining contact boundaries between the sealing ring and tank were assumed to be fully conductive ($h = \infty$).

The material properties were obtained from Reference 6. They were assumed to be defined at 850 °F. Table 3.2-1 lists all material properties used for both the thermal and the stress models. These properties were reviewed and approved by BJPD prior to performing the analyses.

The solution for each of the two transients provided a time history of not only temperatures at selected grid points, but of critical temperature differences between grid points. The temperature differences were then used to select the worst thermal gradients for use as thermal loads on the stress analysis model described next. These results are discussed in detail in Section 4.1.

3.3 STRESS MODEL

The stress model, shown in Figures 3.3-1 to 11, is identical to the thermal model of the nozzle and tank alone. The auxiliary heat model and the extra sealing ring and liquid sodium elements were removed. Shown in these figures are the element numbers as plotted by NASTRAN. Selected grid point numbers have been added for reference purposes.

The element distribution reflects concern for critical stresses in the transition zone between the nozzle and the tank. Because the remainder of the tank model toward the assumed plane of symmetry is expected to behave as a membrane, the model was greatly simplified for purposes of computing efficiency. The full stress model also represents one of 240 identical petals. This petal comprises:

- 924 - Grid points
- 401 - 3-D Solid elements
- 1383 - Unrestrained degrees of freedom

Three coordinate systems were used for defining the input data and for specifying the orientation of the resulting displacements. Figure 3.3-12

TABLE 3.2-1 MATERIAL PROPERTIES

	Property Code	Page
316SS (850°F) [#] Annealed		
E = 2.369 10 ⁷ psi	2111	1.1
G = 9.109 10 ⁶ psi	2112	1.1
ν = 0.2987	2110	1.1
α = 1.102 10 ⁻⁵ in/°F	3114	1.1 Instantaneous
ρ = 4.855 10 ² lb/ft ³	3304	1.1
C _p = 0.1331 BTU/lb * °F	3108	1.1
K = 11.79 BTU/hr * ft * °F	3112	1.1
Liquid Sodium (850°F) [#]		
ρ = 52.64 lb/ft ³	3304	1.1
C _p = 0.3030 BTU/lb * °F	3108	1.1
K = 39.85 BTU/hr * ft * °F	3112	1.1
[#] 850°F assumed operating temperature		

Reference: Nuclear Systems Materials Handbook, Vol. 1 - Design Data
Revision 2, April 16, 1975.

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BJPD SCREENING ANALYSIS
 GEOMETRY PLOTS WITH ELEMENT NUMBERS
 UNDEFORMED SHAPE

FIGURE 3.3-1 MODEL ELEMENTS 1 THRU 24

BJPD SCREENING ANALYSIS
GEOMETRY PLOTS WITH ELEMENT NUMBERS

UNDEFORMED SHAPE

FIGURE 3.3-2 MODEL ELEMENTS 25 THRU 50

BJPD SCREENING ANALYSIS
 GEOMETRY PLOTS WITH ELEMENT NUMBERS
 UNDEFORMED SHAPE

FIGURE 3.3-3 MODEL ELEMENTS 101 THRU 142

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BJPD SCREENING ANALYSIS
GEOMETRY PLOTS WITH ELEMENT NUMBERS

UNDEFORMED SHAPE

FIGURE 3.3-4 MODEL ELEMENTS 143 THRU 186

FIGURE 3.3-5 MODEL ELEMENTS 187 THRU 226

BJPD SCREENING ANALYSIS
 GEOMETRY PLOTS WITH ELEMENT NUMBERS
 UNDEFORMED SHAPE

FIGURE 3.3-6 MODEL ELEMENTS 227 THRU 264

BJPD SCREENING ANALYSIS
GEOMETRY PLOTS WITH ELEMENT NUMBERS
UNDEFORMED SHAPE

FIGURE 3.3-7 MODEL ELEMENTS 401 THRU 476

8

FIGURE 3.3-8 MODEL ELEMENTS 501 THRU 578

BJPD SCREENING ANALYSIS
GEOMETRY PLOTS WITH ELEMENT NUMBERS

UNDEFORMED SHAPE

FIGURE 3.3-9 MODEL ELEMENTS 621 THRU 628

BJPD SCREENING ANALYSIS
GEOMETRY PLOTS WITH ELEMENT NUMBERS
UNDEFORMED SHAPE

FIGURE 3.3-10 MODEL ELEMENTS 629 THRU 636

BJPD SCREENING ANALYSIS
 GEOMETRY PLOTS WITH ELEMENT NUMBERS
 UNDEFORMED SHAPE

FIGURE 3.3-11 MODEL ELEMENTS 637 THRU 644

Basic Coordinate System
X, Y, Z

Cylindrical Coordinate System
 r, θ, z

Spherical Coordinate System
 r, θ, ϕ

FIGURE 3.3-12 COORDINATE SYSTEMS

shows each relative to the model used. The basic system is the primary system used by NASTRAN. All other systems are referenced to the basic system. The cylindrical system is used primarily in the zone of the nozzle. The spherical system is used primarily in the zone of the tank. Caution is required in the interpretation of the output displacements. Note that for the cylindrical system u_r and u_θ are in the basic X-Y plane, whereas u_r and u_θ for the spherical system are in the basic X-Z plane with u_r normal to the spherical surface.

The support conditions were applied to the tank at the plane of symmetry. The tank was allowed only to displace radially. Because the primary zone of interest is the nozzle/tank intersection and because the symmetry boundary is far away from that region, the same boundary constraints were used for all loading conditions, symmetric and unsymmetric.

The material properties (see Table 3.2-1) were assumed to be defined at 850° F. The same values were used for both the stress and the thermal models.

Three loading conditions were imposed as listed in Table 3.3-1. These were piping, internal pressure, and thermal loads. A fourth condition of "interference" pressure between the sealing ring and the tank due to differential thermal expansion was found to be negligible, assuming the specified 20 mils of initial radial clearance. During the two thermal transients, the maximum mean temperature difference between the tank and sealing ring did not exceed the 30° F required to force contact.

The piping loads were distributed over the end of the nozzle according to the three separate conditions: Dead, Thermal and OBE Seismic. Results were obtained for four combinations specified by BJPD. The internal discharge nozzle pressure load of 200 psi was applied to the nozzle. This pressure was then dropped to a residual pressure of 30 psi on the tank. However, because the membrane stress in a sphere is half that of a cylinder for the same internal pressure, the actual load was adjusted to 60 psi to compensate for the spherical approximation to the tank cylinder.

Four thermal loads were applied, three for the IP-3UT transient and one for the IP-1UT transient. The time slices for each loading were selected based on the maximum thermal gradients obtained from the heat transfer analyses. Special consideration was given to obtain conditions which would cause maximum discontinuity stresses as well as maximum surface stresses. The actual selection of thermal maps used is described in Section 4.1.

TABLE 3.3-1 LOADING CONDITIONS

Internal Pressure Load

Piping Loads (Horizontal)

	P (lb)	V (lb)	T (ft-lb)	M (ft-lb)
Dead	0	-13,290	150,820	71,870
Thermal	7,670	10,850	135,550	191,700
OBE	15,340	21,700	152,450	215,600

Thermal Loads

IP-1UT @ 50 sec down transient
 IP-3UT @ 120 sec up transient
 IP-3UT @ 420 sec up transient
 IP-3UT @ 920 sec down transient

The results obtained with NASTRAN include grid point displacements, applied load vectors, reactions, and element stresses. The element stresses are output in the basic coordinate system (see Figure 3.3-12). Hence, they must be rotated to obtain effective cross-sectional stress patterns for interpretation according to the ASME Code. These transformations are described next.

3.4 CODE STRESS INTENSITY CALCULATIONS

The NASTRAN 3-D solid elements (HEXA2 and WEDGE) provide all six stress components, the octahedral shear stress and the average pressure. The stress components must be converted to equivalent stress intensities for comparison with ASME Boiler and Pressure Vessel Code requirements. Figure 3.4-1 shows the equivalent uniform and linearly varying (across thickness) stress components to be assumed for the stress intensity calculations.

Because the solid element stress components are output in the NASTRAN basic coordinate system (see Figure 3.3-12), they must be rotated to be aligned with any arbitrary cross-section selected for evaluation. A special purpose computer program was developed to perform these transformations. As a verification check, the invariant octahedral shear and average pressure stresses are recomputed for comparison with the original output.

Figure 3.4-2 shows the steps taken to obtain the required intensities for both primary and secondary stresses. The transformed stress components were plotted for each cross-sectional cut in the model (see Appendix C). Effective linearized values were obtained at the inner and outer surfaces for each component for each loading condition. These component results were then tabulated for subsequent superposition according to loading combination. The combined stresses were then taken to obtain principal stresses on each surface and at the midsurface of each cut. Finally, from these tables, the corresponding stress intensities were calculated as the maximum difference between principal stresses and tabulated for evaluation according to code in Section 5.0.

TOTAL STRESS = UNIFORM COMPONENT + LINEARLY VARYING COMPONENT

NOTE: STRESSES SHOWN ON FRONT FACES ONLY.

FIGURE 3.4-1 STRESS COMPONENTS FOR ASME CODE INTERPRETATION

FIGURE 3.4-2 CALCULATIONS REQUIRED TO OBTAIN STRESS INTENSITIES

4.0 ANALYSIS RESULTS

The details for results of both the thermal transient and the stress analyses are presented in Appendix C. A summary of these results is presented here. From the stress summary will be taken the data required for code evaluation as presented in Section 5.0.

4.1 THERMAL TRANSIENT RESULTS

Two thermal transients were investigated, IP-1UT and IP-3UT. The first represents a drop in temperature of approximately 70°F in 50 sec (see Figure 3.2-4). The more severe IP-3UT transient represents a rise of approximately 235°F in 420 sec followed by a drop of more than 285°F in 450 sec (see Figure 3.2-3). The time step for integration was established empirically by executing the IP-3UT transient analysis for 100 sec with three different time steps. Thus, an effective time lag was evaluated justifying selection of the intermediate time step value for the first 780 seconds and the larger time step for the remainder of the transient.

The critical conditions to be determined from these analyses include maximum across the thickness gradients and maximum global thermal gradient from section to section. The first establishes primarily the peak surface stress conditions and the latter establishes gross discontinuity stress conditions. To help select the time slice at which possible peak stresses would occur, the 19 temperature differences were monitored as a function of time. These are plotted in Appendix C. The peaks for the important differences are summarized in Table 4.1-1. The approximate location for each of the temperature differences is also indicated.

From this table, and after examination of the rate of change of temperature with distance through the cross-section, four thermal maps were selected for computation of thermal stresses:

<u>Transient</u>	<u>Time</u>
1. IP-3UT	120 sec Up transient
2. IP-3UT	420 sec Up transient
3. IP-3UT	920 sec Down transient
4. IP-1UT	50 sec Down transient

TABLE 4.1-1 PEAK VALUES OF TEMPERATURE DIFFERENCES

Eqn.	Location	IP-1UT		IP-3UT			
		Time Sec	ΔT °F	Time Sec	ΔT °F	Time Sec	ΔT °F
1	Across knee	60	34.8	420	169.3	920	63.6
2	Near base of nozzle	50	35.5	100	85.4	700	83.2
4	Across nozzle	50	31.8	60	61.0	700	63.0
7	Mid knee up to nozzle			460	154.5	980	62.2
9	Mid high knee to nozzle			420	79.5	900	52.3
10	Across high knee	60	37.0	380	123.4	800	92.7
12	Out - high knee to above knee			480	78.1	980	38.1
13	Out - above knee to knee			600	73.8	1320	16.7
18	In - knee to weld	60	34.6	420	163.7	900	51.5

The third map was selected for the possibility that it may create more severe stresses (stress reversal possibly) than would the IP-1UT transient. This possibility is critical to establishing the maximum secondary stress range used in evaluating strain limits for fatigue and ratcheting effects.

The stress results for these loads as well as the two mechanical loadings are presented next.

4.2 STRESS RESULTS

The details of stress results are presented in Appendix C. Each component was linearized along each of the six cross-sectional cuts shown in Figure 4.2-1. All linearized stresses are summarized in Table 4.2-1 for the sum of mechanical loads and in Table 4.2-2 for the maximum stress range of the thermal loads. Table 4.2-3 presents the maximum range of secondary stress, Q_R , used for evaluating strain limits for creep and fatigue. These tables provide the data required in Section 5.0 for ASME Code evaluation.

It should be noted that in computing Q_R for Table 4.2-3 (see Appendix C.2-130), the bending contributions of the mechanical loads were added as a secondary stress to the thermal stresses at the inner and outer surfaces. These linearly varying components of the mechanical load are considered secondary stresses because the region in which they occur is a gross structural discontinuity at the junction of the nozzle to the tank. Once these stresses were added, then the primary stresses were computed to obtain the required stress intensities.

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FIGURE 4.2-1 CROSS-SECTIONAL CUTS FOR STRESS LINEARIZATION

TABLE 4.2-1 MECHANICAL LOADS SUMMED - LINEARIZED

Section		σ_x	σ_y	σ_z	σ_{yz}	σ_{xz}	σ_{xy}
Cut	Loc	(ksi)	(ksi)	(ksi)	(ksi)	(ksi)	(ksi)
1	I	0	7.98	-0.20	-0.12	0.43	1.19
	M	5.31	9.44	-0.06	-0.12	0.12	1.24
	O	10.62	10.90	0.09	-0.12	-0.20	1.28
2	I	-1.97	7.46	-0.11	-0.35	0.18	0.93
	M	4.52	9.39	0.34	-0.29	-0.90	1.06
	O	11.00	11.32	0.78	-0.23	-1.98	1.19
3	I	-0.60	7.04	0.21	-0.32	-0.56	0.55
	M	2.95	8.13	0.27	-0.20	-0.80	0.70
	O	6.50	9.21	0.32	-0.08	-1.04	0.85
4	I	0.21	5.70	0.15	-0.20	0.24	0.28
	M	1.89	6.45	0.13	-0.09	0.14	0.42
	O	3.57	7.19	0.11	0.03	0.03	0.55
5	I	0.48	4.42	-0.11	-0.13	-0.52	0.17
	M	1.35	5.09	-0.03	0	-0.08	0.26
	O	2.21	5.76	0.05	0.12	0.36	0.34
6	I	1.13	0.78	-0.08		-0.36	0.14
	M	1.03	1.20	-0.08	negl.	-0.53	0.20
	O	0.92	1.62	-0.09		-0.69	0.25

TABLE 4.2-2 MAX THERMAL STRESS RANGE - LINEARIZED

Section		Load* Combination	σ_x (ksi)	σ_y (ksi)	σ_z (ksi)	σ_{yz} (ksi)	σ_{xz} (ksi)	σ_{xy} (ksi)
Cut	Loc							
1	I	D-E	-19.00	-28.30	-0.23	0.37	0.86	negl.
	M	D-E	-0.30	-16.95	0.15	0.24	-0.42	
	O	D-E	18.40	-5.60	0.53	0.10	-1.69	
2	I	D-E	-10.15	-33.00	-0.40	0.38	1.39	negl.
	M	D-E	-0.61	-20.65	0.27	0.25	-1.17	
	O	C-F	17.48	9.35	-1.41	-0.10	-4.94	
3	I	D-E	-0.40	-34.18	-0.88	0.40	0.16	negl.
	M	D-F	-0.64	-17.19	-0.43	0.20	-1.15	
	O	C-E	4.15	12.89	0.18	-0.16	-4.01	
4	I	D-E	2.91	-36.06	-3.43	0.46	3.71	0.14
	M	D-F	-0.38	-13.10	-1.75	0.17	1.37	0.07
	O	C-E	6.86	23.40	-0.04	-0.17	-5.74	-0.18
5	I	D-E	2.26	-32.97	-0.73	0.44	3.93	0.28
	M	D-F	-0.50	-7.85	-1.14	0.08	-1.09	0.04
	O	D-E	-2.99	27.39	-1.58	0.32	-6.86	0.26
6	I	C-E	0.93	9.53	1.09	negl.	-2.69	0.09
	M	D-E	-1.18	8.62	0.78		-0.64	0.05
	O	D-E	-5.53	9.82	-0.22		1.61	0.18

*C: 3UT (120 sec) D: 3UT (420 sec) E: 3UT (920 sec) F: 1UT (50 sec)

TABLE 4.2-3 MAX RANGE OF SECONDARY STRESS

Section		Q_R (ksi)
Cut 1	Inner	29.39
	Outer	27.85
Cut 2	Inner	34.08
	Outer	24.94
Cut 3	Inner	34.33
	Outer	13.75
Cut 4	Inner	38.04
	Outer	24.21
Cut 5	Inner	35.04
	Outer	30.19
Cut 6	Inner	8.08
	Outer	15.88

4.3 DEFLECTION RESULTS

Deflections are particularly important to pump design because of the ever present potential of leakage causing loss of efficiency. In this design, leakage is expected between the pump tank and the sealing ring at the discharge nozzle. The relative deflection between the two bodies dictates that leakage flow rate.

Figures 4.3-1 thru 4.3-6 show the deformation pattern for each of the six loading conditions; pressure, piping (worst case), IP-1UT @ 50 sec, IP-3UT @ 120 sec, 420 sec and 920 sec. The leakage problem is most critical during full flow operations under full hydraulic pressure. From Figure 4.3-1, the pressure load causes the nozzle to deflect 3.6 mils axially along the nozzle. In combination with the piping loads, this will increase to 8.5 mils. The inward deformation of the sealing ring would add to the total increase in gap. To complete the picture, the stiffening effects of tank/sealing ring interaction must be considered.

Though low in magnitude, the piping load deflections in Figure 4.3-2 show a severe deformation pattern. Because the model ignored the effects of a reduced thickness in the pump tank above and below the sealing ring contact zone, these deflections could be larger than indicated. This is important because the piping loads are not symmetric and, hence, they could accentuate the effects of the reduced tank section.

The thermal load deformations of Figure 4.3-3 thru 4.3-6 are shown relative to the 770°F reference temperature. To obtain equivalent total deformation from room temperature (70°F), a $\Delta u_1 = 0.0077 R_1$ must be added where R_1 is the appropriate radius in the direction of Δu_1 of concern.

The up-transients, Figure 4.3-4 and 4.3-5, show similar behavior. However, R_u down-transients, Figure 4.3-3 and 4.3-6 are different, pointing out the residual effects caused by the initial temperature rise of the IP-3UT transient.

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BOPD - Pump Nozzle Deflections
Inside Surface - Pressure Load

FIGURE 4.3-1 PRESSURE LOAD DEFORMATIONS

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BSPD - Pump No. 10 Deflections
Inside Surface - D + T + ϕ (90°)

FIGURE 4.3-2 PIPING LOAD DEFORMATIONS

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BOPD. Pump Nozzle Deflections
Inside Surface - 1UT-50

Scale: 1" = 20 mils

$$\max u_r = -7.64 \times 10^{-3} \text{ in.}$$

@ (11)

$$\max u_z = -3.75 \times 10^{-3} \text{ in.}$$

@ (11)

FIGURE 4.3-3 IP-1UT DEFORMATIONS @ 50 SEC.

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BJPD - Pump Nodgle Deflections
Inside Surface - 3UT-120

FIGURE 4.3-4 IP-3UT DEFORMATIONS @ 120 SEC.

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BJPD - Pump No. 1, Deflections
Inside Surface - 3UT - 420

Scale: 1" = 100 mils

 $\max u_r = 66.961 \times 10^{-3} \text{ in.}$
@ (11) $\max u_z = 58.404 \times 10^{-3} \text{ in.}$
@ (11)

FIGURE 4.3-5 IP-3UT DEFORMATIONS @ 420 SEC.

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BSPD - Pump Nozzle Deflections
Inside Surface - 3UT-920

FIGURE 4.3-6 IP-3UT DEFORMATIONS @ 920 SEC.

5.0 CODE EVALUATION

An inelastic analysis using elastic methods is performed according to Code Case 1592. This requires two types of analysis:

1. Load-controlled stress limit
2. Strain and deformation limit

In Section 5.1, the load-controlled stress limit analysis, stress limits are checked for design conditions, normal and upset conditions, and use-fraction sum. The strain and deformation limit analysis included an elastic strain limit check, Section 5.2, creep ratcheting strain, Section 5.3, and creep-fatigue evaluation, Section 5.4.

The linearized stress data used for these examinations are tabulated in Section 4.2. Each of the six cross-sectional cuts used for this evaluation have been presented in Figure 4.2-1. The mechanical load results are summed separately from the thermal load results. For the secondary stress range Q_R , a separate summary is presented which also includes the bending stress contributions of the mechanical loads. For the creep-fatigue evaluation, the primary and secondary stresses were combined, when required, to obtain a maximum for the strain range,

5.1 LOAD CONTROL STRESS LIMITS

The ASME Code requires that load control stress limits will be estimated due to primary stress. For the conditions given, with BJPD concurrence, only the primary membrane stress intensity is to be considered here. The bending stress contributions due to the mechanical loads are to be considered as gross structural discontinuity stresses (Code Case 1592-3213.9(a)) and hence are to be included in the secondary stress intensity calculations. The stress limits are presented in Table 5.1-1 for the design, normal and upset conditions. The principal stresses for all six cuts meet requirements as tabulated for membrane stress intensity comparison in Table 5.1-2. Table 5.1.3 shows that the Use-Fraction sum associated with general primary membrane stress also is met.

5.1.1 Design Condition

Code Case 1592-3222 specifies the following limitation on primary membrane stress for the Design Condition. The general primary membrane stress intensity (P_m) shall not exceed S_o .

$$P_m \leq S_o$$

5.1.2 Normal and Upset Conditions

Code Case 1592-3223 specifies the limitation for normal and upset conditions as:

$$P_m \leq S_{mt}$$

5.1.3 Use-Fraction Sum

The Use-Fraction sum associated with the general primary-membrane stresses for all increments of primary loadings during normal and upset conditions shall satisfy the following requirements:

(Code Case 1592-3224(b))

$$\sum_i \left(\frac{t_i}{t_{im}} \right) \leq B$$

where $B = 1$

TABLE 5.1-1 LOAD-CONTROLLED STRESS LIMITS: ALLOWABLE QUANTITIES
MATERIAL: 316 SS

Category	Condition	Parameters	Quantity	Derivation Code Case 1592	Value
Design	Design Conditions	$T = 850^{\circ}$	S_o	Table I-14.2	15.7
Normal and Upset	Steady State at 100% Design Flow	$T = 770^{\circ}$ $t = 262,000 \text{ hrs}$	S_{mt}	Table I-14.3B S_m governs	15.8
			S_m	Figure I-14.3B	15.8
	Upset and Using Maximum Sodium Temp.	$T = 1005^{\circ}$ Max. $t < 1 \text{ hr.}$	S_{mt}	Table I-14.3B S_m governs	15.4
			S_m	Figure I-14.3B	15.3

TABLE 5.1-2 STRESS EVALUATION ON DESIGN, NORMAL AND UPSET CONDITIONS

Membrane Stress Intensities					Design	Normal	Upset
Cut	P ₁	P ₂	P ₃	P _m	S _o (15,700) (psi)	S _{mt} (15,800) (psi)	S _m (15,300) (psi)
1	9782	4973	-65	9847	Met	Met	Met
2	9634	4457	154	9840	Met	Met	Met
3	8230	3066	45	8185	Met	Met	Met
4	6483	1864	118	6366	Met	Met	Met
5	5107	1332	-35	5142	Met	Met	Met
6	1400	1037	-298	1698	Met	Met	Met

TABLE 5.1-3 USE-FRACTION SUM

Section	P _m (psi)	t (hr.)	t _{im} (hr.)	$\sum_i \left(\frac{t_i}{t_{im}} \right) \leq B$	
Cut 1	9847	262,800	73 x 10 ⁵	< 1.0	Met
Cut 2	4480	"	"	"	"
Cut 3	8185	"	"	"	"
Cut 4	6366	"	"	"	"
Cut 5	5142	"	"	"	"
Cut 6	1698	"	"	"	"

Total operating hours of primary pump steady-state conditions is 262,800 hours (100% flow). The total transient time is very small compared to time at normal conditions and was therefore neglected. (See Table 5.1-3)

From Figure I-14.4B, allowable time, t_{im} , is more than 3×10^5 hours.

$$\sum_i \left(\frac{t_i}{t_{im}} \right) = \frac{800}{300,000} < 1.0$$

Therefore, the Use-Fraction Sum with primary membrane stress met the code.

5.2 ELASTIC STRAIN LIMIT CHECK

If the conditions of either T-1322 and T-1323 are satisfied, then the strain limit T-1310 is considered to be satisfied, and there is no need to evaluate strain increments due to ratcheting or enhanced creep. The following definitions are made:

$$X \equiv (P_L + \frac{P_b}{K_t})_{\max} \div S_y \quad (\text{Note: } P_b \text{ is assumed} = 0)$$

$$Y \equiv \frac{(Q_R)_{\max}}{S_y} .$$

Where Q_R = the maximum range of the secondary stress during the operating condition.

S_y = the average of the S_y values at the maximum and minimum wall-averaged temperatures during the operating conditions

$(P_L + \frac{P_b}{K_t})_{\max}$ = the maximum values of the primary stress

Note that because the nozzle/tank intersection can be classified as a gross structural discontinuity, the mechanical load bending stresses are considered secondary stresses. Hence, P_b/K_t need not be computed, and $X = P_L/S_y$.

The tests to be applied are as follows:

T-1322

$$X + Y \leq \frac{S_a}{S_y}$$

Where S_a = the lesser of the average value of the minimum yield strength for the highest and lowest average wall temperature occurring during the cycle, of $1.25 S_t$ taken at the highest average wall temperature during the cycle at 10^4 hour.

T-1323

$$X + Y \leq 1$$

Provided that the average wall temperature at one of the stress extremes defining each secondary stress range, Q_R , is below the creep region. This temperature is defined for purposes of this paragraph as that at which S_m equals S_t for 10^5 hour.

This temperature for 316 SS ($T = 1015^\circ\text{F}$, from Figure 1-14.3B) is above all average wall temperatures as required.

Table 5.2-1 shows the data for evaluating σ_y and S_a , followed by Table 5.2-2 listing key stress intensities. Table 5.2-2 also shows that most cross-sections do not comply with T-1322 and T-1323 limits. Hence, creep ratcheting strain limits must be satisfied or investigated next.

TABLE 5.2-1 AVERAGE S_y AND S_a ON EACH CUT

Section	Max. Wall Average Temp.	S_y^* (ksi)	Min. Wall Average Temp.	S_y^* (ksi)	Average S_y^* (ksi)	$1.25 S_t^*$ (ksi)	S_a^* (ksi)
Cut 1	974	17.0	750	17.8	17.4	19.2	17.4
Cut 2	963	17.1	751	17.8	17.4	19.3	17.4
Cut 3	934	17.1	752	17.8	17.5	19.3	17.5
Cut 4	900	17.3	752	17.8	17.6	19.5	17.6
Cut 5	878	17.3	753	17.8	17.6	19.5	17.6
Cut 6	798	17.6	770	17.7	17.6	19.7	17.6

*Code Case 1592 - Figure I-14.5B

TABLE 5.2-2 PRIMARY AND SECONDARY STRESS ON EACH CUT

Section		Max. Wall Average Temp.	P_L (ksi)	S_t (psi)	Q_R (ksi)	$X + Y$	$\frac{S_a}{S_y}$	T-1322 Check	T-1323 Check
Cut	Location								
1	Inside	974°	9.83	15.4	29.39	2.26	1.0	Not Met	Not Met
	Outside				27.85	2.17		"	"
2	Inside	963°	9.48	15.5	34.08	2.50	1.0	"	"
	Outside				24.94	1.98		"	"
3	Inside	934°	8.19	15.5	34.33	2.43	1.0	"	"
	Outside				13.75	1.25		"	"
4	Inside	900°	6.37	15.6	38.04	2.52	1.0	"	"
	Outside				24.21	1.74		"	"
5	Inside	878°	5.14	15.6	35.04	2.29	1.0	"	"
	Outside				30.19	1.99		"	"
6	Inside	798°	1.70	15.8	8.08	0.56	1.0	Meets	Meets
	Outside				15.88	1.00		Meets	Meets

5.3 CREEP RATCHETING STRAIN

Since T-1322 and T-1323 did not meet code, creep ratcheting strain (T-1324) must be investigated to determine whether or not T-1310 is satisfied. For this investigation inelastic characteristics, namely shakedown, plastic flow and creep ratcheting which depend on shakedown region and plastic region must be considered. For this test (T-1324) to apply, the average wall temperature at one of the stress extremes defining each secondary stress range, Q_R , is below the creep regime. This temperature, as for Test T-1323, is defined as that at which S_m equals S_t for 10^5 hours. For 316SS, that temperature is 1015°F which is above all wall averaged temperatures.

The limit of creep ratcheting strain is 1% on parent metal and 0.5% on welded metal (section cut #6 only).

T-1324

$$Z \equiv \frac{\sigma_c}{S_y} \quad \text{in regions } S_1, S_2 \text{ and } P \text{ of Fig. T-1324-1}$$

$$Z = X \cdot Y \quad \text{in regions } S_2, P$$

$$Z = Y + 1 - 2 \sqrt{(1-X)Y} \quad \text{in regions } S_1$$

Creep ratcheting stress is $1.25 \sigma_c$. Creep ratcheting strain can be obtained from the isochronous stress-strain curve of T-1800B using $1.25 \sigma_c$. For this method to be applicable $Z \leq 1$.

Code Case 1592 requires calculation of the maximum primary stress (pressure plus piping loads) for normal and upset conditions. For reasons of simplicity and conservatism, only the thermal stress range between IP-3UT and IP-1UT was used to estimate creep ratcheting strain as listed in Table 5.3-1. Note that the separate inelastic evaluation of selected operating cycles is used. The available curves T-1800-B provide data for total cycle times of 300 hours which is greater than the total time of both the IP-3UT and -1UT transients which are assumed to reach 1000°F. The remaining time is assumed at the design temperature of 850°F. These separate strains are then summed for comparison with allowables.

TABLE 5.3-1 CREEP RATCHETING STRAIN (IP3UT-IP1UT)

Section		P _L (ksi)	Q _R (ksi)	S _y (ksi)	X	Y	Z	1.25 σ_c (ksi)	T-1800		Sum Total Strain	Creep Ratcheting Strain	
Cut	Location								1000° < 300 hr	850° 3x10 ⁵ hr			
1	Inside	9.85	29.39	17.4	0.566	1.689	0.977	21.25	0.3	0.3	0.6	<1%	Meets
	Outside		27.85			1.601	0.934	20.31	0.2	0.2	0.4	<1%	Meets
2	Inside	9.48	34.08	17.4	0.545	1.959	Ratcheting					>1%	Not Met
	Outside		24.94			1.433	0.818	17.79	0.1	0.1	0.2	<1%	Meets
3	Inside	8.19	34.33	17.5	0.468	1.962	0.918	20.08	0.2	0.2	0.4	<1%	"
	Outside		13.75			0.786	0.493	10.78	0.05	0.05	0.1	<1%	"
4	Inside	6.37	38.04	17.6	0.362	2.16	0.782	17.20	0.1	0.1	0.2	<1%	"
	Outside		24.21			1.38	0.503	11.07	0.05	0.05	0.1	<1%	"
5	Inside	5.14	35.04	17.6	0.292	1.99	0.581	12.78	0.1	0.1	0.2	<1%	"
	Outside		30.19			1.72	0.502	11.04	0.05	0.05	0.1	<1%	"
6	Inside	1.70	8.08	17.6	0.097	0.459	Elastic					<0.5%	"
	Outside		15.88			0.902	Elastic					<0.5%	"

As can be seen, only the inside of cut 2 exceeds the limits of T-1324. Note, however, that the most severe thermal transient (IP-3UT) occurs without the internal pressure load. Under these conditions, creep ratcheting strain would be within acceptable limits.

5.4 CREEP-FATIGUE EVALUATION (T-1400)

The combination of normal, upset and emergency condition shall be evaluated for accumulated creep and fatigue damage including hold time and strain rate effects. For this segment of the screening analysis the emergency transient conditions are included.

For a design to be acceptable, the creep and fatigue damage shall satisfy the following relation.

T-1411 Fatigue Damage Sum:

$$\sum_{j=1}^p \left(\frac{n}{N_d} \right)_j + \sum_{k=1}^q \left(\frac{t}{T_d} \right)_k \leq D$$

D = Total creep - fatigue damage (see Figure T-1420-2)

n = Number of applied cycles of loading condition j.

N_d = Number of design allowable cycles of loading condition from Fig. T-1430-1 corresponding to the maximum metal temperature during the cycle.

t = Time duration of loading condition k.

T_d = Allowable time at a given stress intensity. T_d values are obtained by entering the stress-to-rupture curve (Fig. I-14.6) at the stress value equal to the calculated stress divided by the factor K' (≈ 0.9 , Table T-1411-1).

To find N_d , the equivalent strain range must be calculated. The method is given in T-1413. To find T_d , creep damage evaluation must be performed according to T-1433 using the primary and secondary stresses resulting from sustained operating conditions between cycles. For the equivalent strain calculations required for the fatigue damage evaluation, only thermal stress need be used, as described below. This is then followed by an evaluation of the creep damage to complete the Fatigue Damage Sum. As expected, all cross-section cuts meet with code limits.

5.4.1 Equivalent Strain Range (T-1413)

An equivalent strain range is used to evaluate the fatigue damage sum for both elastic and inelastic analysis. Because the sequence of loading

is not defined, then the method of combining cycles described in NB-3222.4(e.5) is applied. The equivalent strain range is computed as follows:

- Step 1. Calculate all strain components for the strain history (ϵ_x , ϵ_y , ϵ_z , ϵ_{xy} , ϵ_{yx} , ϵ_{zx} , versus time) for the complete cycle. Strain concentration factors are added at this step.
- Step 2. Select a point when conditions are at an extreme for the cycle, either maximum or minimum. Refer to this time point by a subscript i.
- Step 3. Calculate the history of the change in strain components by subtracting the values at the time, i, from the corresponding components at each point in time during the cycle.

$$\Delta\epsilon_x = \epsilon_x - \epsilon_{xi}$$

$$\Delta\epsilon_y = \epsilon_y - \epsilon_{yi}$$

Etc.

- Step 4. Calculate the equivalent strain range as for each point in time.

$$\Delta\epsilon_{\text{equiv.}} = \frac{\sqrt{2}}{3} \left[(\Delta\epsilon_x - \Delta\epsilon_y)^2 + (\Delta\epsilon_y - \Delta\epsilon_z)^2 + (\Delta\epsilon_z - \Delta\epsilon_x)^2 + 6(\Delta\epsilon_{xy}^2 + \Delta\epsilon_{yz}^2 + \Delta\epsilon_{zx}^2) \right]^{1/2}$$

Alternately,

$$\Delta\epsilon_{\text{equiv.}} = \sqrt{2} \tau_o / E$$

where τ_o is the octahedral shear stress and E is Young's Modulus.

- Step 5. Use the maximum equivalent strain range calculated as the range of strain to enter the fatigue curves.

- (a) The above procedure may be used regardless of whether principal strains change directions or not.

- (b) An alternative to the above sequence, applicable only when principal strains do not rotate, is given in T-1414.

The equivalent strain calculations were performed using the maximum secondary stress range from the IP3U and IP1U transients combined with mechanical loads to produce maximum stress intensity as calculated in Appendix C. Table 5.4-1 shows equivalent strain on Cuts 1 to 6. The calculations were done according to the method of T-1413 and NB-3222(e.5). These data are then used to estimate the strain range of the other severe thermal transients.

5.4.2 Cumulative Use Factor Including Other Severe Transients (T-1432)

There are other severe transients besides the IP3U and IP1U transient (see Appendix B and Reference 8) and the equivalent strain range of these transients must be analyzed. For simplicity and conservatism, the equivalent strain ranges of the other transients were calculated from the ratio of temperature difference involved to the temperature difference of the IP3U and IP1U transient combination.

Table 5.4-2 shows the ratio of temperature differences for each transient. The equivalent strains for Cuts 1 thru 6 during each transient are listed in Tables 5.4-3. As can be seen, the cumulative usage factors obtained from Fig. T-1430-1B are negligible for all cuts. That is, the sum $\sum \frac{n}{N_d} < 1 \times 10^{-3}$.

5.4.3 Creep Damage Evaluation (T-1433)

Creep damage from primary and secondary stresses resulting from sustained operating conditions between cycles shall be evaluated as required by (a), (b) and (c) below.

The stress intensity, S , due to primary-plus-secondary stresses from load condition, shall be taken as the lesser of:

- a. The minimum specified yield strength of the material under evaluation.
- b. $(P_m + 0.5 S_r)$ where S_r is the maximum range of primary-plus-secondary stress intensity.

TABLE 5.4-1 EQUIVALENT MAX STRAIN FROM IP-3U and IP-1U TRANSIENTS

Section		Max Metal Temp	E x 10 ⁻³ (ksi)	$\Delta\sigma_x$ (ksi)	$\Delta\sigma_y$ (ksi)	$\Delta\sigma_z$ (ksi)	$\Delta\tau_{y3}$ (ksi)	τ_{xz} (ksi)	τ_{xy} (ksi)	τ_o (ksi)	$\Delta\epsilon \times 10^3$ Equiv (in/in)
Cut	Location										
1	Inside	986	22.64	-19.00	-28.30	-0.23	0.37	0.86	0	11.7	0.731
	Outside			29.02	5.30	0.62	-0.02	-1.89	1.28	12.6	0.787
2	Inside	981	22.68	-10.15	-33.00	-0.40	0.38	1.39	0	13.7	0.854
	Outside			28.63	20.67	-0.63	-0.33	-6.92	1.19	13.6	0.848
3	Inside	970	22.76	-0.40	-34.18	-0.88	0.40	0.16	0	15.8	0.981
	Outside			10.55	22.10	0.50	-0.24	-5.05	0.85	9.8	0.609
4	Inside	962	22.83	2.91	-36.06	-3.43	0.46	3.71	0.14	17.3	1.072
	Outside			4.43	30.59	0.07	-0.14	-5.71	0.37	14.3	0.886
5	Inside	960	22.84	2.26	-32.97	-0.73	0.44	3.93	0.28	16.3	1.009
	Outside			-0.76	33.15	-1.53	0.44	-6.50	0.60	16.4	1.015
6	Inside	830	23.84	2.06	10.31	1.01	0	-3.05	0.23	4.9	0.288
	Outside			-4.61	11.06	-0.31	0	0.92	0.43	6.7	0.397

TABLE 5.4-2 TEMPERATURE DIFFERENCE RATIOS

Transient	ΔT_i	ΔT IP3U, IP1U	$\Delta T_i / \Delta T$	Cycles	Maximum Metal Temperature
IP-1UT	70	320	0.219	480	770
IP-3UT	285	320	0.891	15	1005
IP-5UT	80	320	0.250	10	770
IP-6UT	90	320	0.281	5	770
IP-7UT	125	320	0.391	14	770
IP-8UT	150	320	0.469	63	910
IP-1ET	250	320	0.781	8	1000
IP-2ET	155	320	0.484	8	770
IP-5ET	155	320	0.484	8	770

TABLE 5.4-3 EQUIVALENT STRAIN AND USAGE FACTORS FOR TRANSIENTS

Transient		1UT $\Delta T_1/\Delta T=0.219$, $T=770^\circ$, $n=480$				3UT $\Delta T_1/\Delta T=0.891$, $T=1005^\circ$, $n=15$			5UT $\Delta T_1/\Delta T=0.250$, $T=770^\circ$, $n=10$		
Section		$\Delta \epsilon_{1U3U}$ (in/in)	$\Delta \epsilon \times 10^{-3}$ (in/in)	N_d	n/N_d	$\Delta \epsilon \times 10^{-3}$ (in/in)	N_d	n/N_d	$\Delta \epsilon \times 10^{-3}$ (in/in)	N_d	n/N_d
Cut	Location										
1	Inside	0.731	0.160	10^6	Negligible	0.651	2×10^4	Negligible	0.183	10^6	Negligible
	Outside	0.787	0.172			0.701	7×10^4		0.197		
2	Inside	0.854	0.187			0.751	5×10^4		0.214		
	Outside	0.848	0.186			0.756	5×10^4		0.245		
3	Inside	0.981	0.215			0.874	2×10^4		0.245		
	Outside	0.609	0.133			0.543	6×10^5		0.152		
4	Inside	1.072	0.235			0.955	10^4		0.268		
	Outside	0.886	0.194			0.789	10^5		0.221		
5	Inside	1.009	0.221			0.899	3×10^4		0.252		
	Outside	1.015	0.222			0.904	2×10^4		0.254		
6	Inside	0.288	0.063			0.257	10^6		0.072		
	Outside	0.397	0.087			0.354	10^6		0.099		

TABLE 5.4-3 EQUIVALENT STRAIN AND USAGE FACTORS FOR TRANSIENTS (cont'd)

Transient		6UT $\Delta T_1/\Delta T=0.281$, $T=770^\circ$, $n=5$				7UT $\Delta T_1/\Delta T=0.391$, $T=770^\circ$, $n=14$			8UT $\Delta T_1/\Delta T=0.469$, $T=910^\circ$, $n=63$		
Section		$\Delta \epsilon_{1U3U}$ (in/in)	$\Delta \epsilon \times 10^{-3}$ (in/in)	N_d	n/N_d	$\Delta \epsilon \times 10^{-3}$ (in/in)	N_d	n/N_d	$\Delta \epsilon \times 10^{-3}$ (in/in)	N_d	n/N_d
Cut	Location										
1	Inside	0.731	0.205	10^6	Negligible	0.286	10^6	Negligible	0.343	10^6	Negligible
	Outside	0.787	0.221			0.308			0.369		
2	Inside	0.854	0.240			0.334			0.401		
	Outside	0.848	0.238			0.332			0.398		
3	Inside	0.981	0.276			0.384			0.460		
	Outside	0.609	0.171			0.238			0.286		
4	Inside	1.072	0.301			0.419			0.503		
	Outside	0.886	0.249			0.346			0.416		
5	Inside	1.009	0.284			0.385			0.473		
	Outside	1.015	0.285			0.397			0.476		
6	Inside	0.288	0.081			0.113			0.135		
	Outside	0.397	0.112			0.155			0.186		

TABLE 5.4-3 EQUIVALENT STRAIN AND USAGE FACTORS FOR TRANSIENTS (cont'd)

Transient		1ET $\Delta T_1/\Delta T=0.781$, $T=1000^\circ$, $n=8$				2ET $\Delta T_1/\Delta T=0.484$, $T=770^\circ$, $n=8$			5ET $\Delta T_1/\Delta T=0.484$, $T=770^\circ$, $n=8$		
		$\Delta \epsilon_{1U3U}$ (in/in)	$\Delta \epsilon \times 10^{-3}$ (in/in)	N_d	n/N_d	$\Delta \epsilon \times 10^{-3}$ (in/in)	N_d	n/N_d	$\Delta \epsilon \times 10^{-3}$ (in/in)	N_d	n/N_d
Cut	Location										
1	Inside	0.731	0.571	4×10^5	Negligible	0.354	10^6	Negligible	0.354	10^6	Negligible
	Outside	0.787	0.615	3×10^5		0.381			0.381		
2	Inside	0.854	0.667	2×10^5		0.413			0.413		
	Outside	0.848	0.662	2×10^5		0.410			0.410		
3	Inside	0.981	0.766	5×10^5		0.475			0.475		
	Outside	0.609	0.476	10^6		0.295			0.295		
4	Inside	1.072	0.837	3×10^4		0.519			0.519		
	Outside	0.886	0.692	7×10^4		0.429			0.429		
5	Inside	1.009	0.788	3×10^4		0.488			0.488		
	Outside	1.015	0.793	3×10^4		0.491			0.491		
6	Inside	0.288	0.225	10^6		0.139			0.139		
	Outside	0.397	0.310	10^6		0.192			0.192		

- c. The value T_d is obtained from the stress-to-rupture curves (Fig. I-14.6) but the calculated stress intensity is divided by a factor K' from Table T-1411-1.

From T-1433(a) and (b), since all minimum specified yield strengths are less than the $(P_m + 0.5 S_r)$ values for all severe transients, the minimum specified yield strength of the material is used for creep damage evaluation.

The value of the minimum specified yield strength at design temperature 850°F is 17,400 psi.

From Figure I-14.6B, T_d is more than 10^7 hr with $S = \sigma_y/K' = 19,333$ psi. Therefore,

$$\sum_{k=1}^q \left(\frac{t}{T_d} \right)_k = \frac{2.628 \times 10^5}{10^7} = 0.026$$

For all section cuts to meet code limits based on the creep-fatigue damage envelope of Fig. T-1420-2, all the cumulative use factors $\left(\sum_j \frac{n}{N_d} \right)$ must be less than 0.92. Since, from the previous section, the maximum of all cross-sections is less than 10^{-3} , all code limits of paragraph T-1400 are met.

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BASIC TECHNOLOGY, INC.

LARGE SCALE SODIUM PUMP UPPER BEARING ANALYSIS

FOR

BYRON JACKSON PUMP DIVISION
BORG WARNER CORPORATION
2300 E. VERNON, VERNON, CA. 90058

BY

BASIC TECHNOLOGY, INCORPORATED
806 MANHATTAN BEACH BOULEVARD
MANHATTAN BEACH, CALIFORNIA 90266

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BASIC TECHNOLOGY, INC.

1.0 Introduction

This report presents an analysis of a preliminary design of a journal bearing for a proposed large scale sodium pump for a new prototype liquid metal fast breeder reactors (LMFBR). The design incorporates improved features based on existing and demonstrated sodium pump technology and draws considerable experience from previous and current work on FFTF and CRBR sodium pumps.

The journal bearing is located at the upper pump tank, and is used to provide the structural stability for the rotating pump shaft. The structural integrity of the bearing under primary loads and secondary thermal transient loads is essential for the operability of the pump.

The bearing configuration to be analyzed by Basic Technology, Inc. is shown in Byron Jackson's large scale sodium pump layout drawing No. IF-7937, Rev. 0. (1)

2.0 Scope of Work

The analysis entails a two-dimensional axisymmetric finite element elastic model of the upper bearings and bearing support under the most severe thermal transient condition. The temperatures, thermal stresses and distortions are calculated with the thermal and structural finite element models. Particular attention is focused on the bearing gap under the differential thermal expansion and contraction to determine if the relative deflections of the pump shaft and the bearing support might adversely affect the operability of the pump.

The primary loads on the bearing region include pressure, centrifugal force due to the rotating shaft, and torsional effects. The primary and secondary stresses are evaluated for any possible fatigue, creep and/or ratcheting damages, in accordance with the ASME Boiler and Pressure Vessel Code Section III and the high temperature Code Case 1592. (2)

The design and operating conditions are given by General Electric's Specifications (3) No. 22A3994, revision 2, which contains the requirements for the large scale breeder reactor plant prototype mechanical pump conceptual design.

The most severe thermal transient to be analyzed in detail is the IP3UT transient. However, all applicable transients are included in the consideration for fatigue evaluation. (4)

3.0 Finite Element Models

The ANSYS computer program ⁽⁵⁾ is used in the thermal and structural evaluation of the upper journal bearing. A 2-D axisymmetric finite element model is constructed for the inner and outer regions of the bearing. The thermal and structural models are similar except for the pocket of sodium in the inner region, which is included only in the thermal model. The physical dimensions of the bearing region are shown in fig. 3.1. It has been found in a recent ANSYS verification study ⁽⁶⁾ that in most cases, sufficient accuracy in thermal stress analysis can be achieved using only three isoparametric elements across the wall thickness. The results of that study are applicable to the present models and will be used here.

3.1 Thermal Model

The finite element model for the thermal transient analysis is shown in fig. 3.2. It consists of 343 elements, with 170 isoparametric elements (STIF 55) in the inner region and 173 elements in the outer region. Except for the pocket of sodium in the inner region where the primary mechanism of heat transfer is assumed to be conduction, convection is assumed to be the dominant mechanism on all surfaces exposed to the liquid sodium. The bulk temperatures of the sodium in various regions surrounding the bearing are determined as a function of time in separate calculations to be discussed in Section 4.0. Since the bearing is immersed completely in liquid sodium during the operation cycles of the pump, no thermal radiation effects need be included in the analysis.

3.1.1 Convection Surfaces

The inner surface of the rotating shaft wall convects to the sodium

THERMAL MODEL
UPPER BEARING
FIGURE 3.2

UPPER BEARING

FIGURE 3.3
THERMAL MODEL
UPPER BEARING

inside the shaft. The temperatures of the sodium within the shaft is determined not only by the transient conditions around the bearing region, but also by those in the lower portion of the pump tank, in which the liquid sodium moves up through the discharge nozzle. The outer surface of the rotating shaft and the inner surface of the feed cylinder convect to the transient sodium which flows up directly from the lower pump tank. The outer surfaces of the bearing and the feed cylinder convect to the large tank of sodium in the upper tank. The bulk temperatures of the sodium in this region will be calculated in Section 4.0 assuming 1) heat transfer of transient sodium to the bulk sodium through the walls of the bearing and support and 2) mixing of transient sodium through the bearing gap with the bulk sodium.

3.1.2 Stagnant Sodium

The pocket of sodium in the inner region of the bearing is assumed to be stagnant in the thermal model. The sodium in this region is modelled with isoparametric elements, in direct contact with the elements representing the stainless steel. This makes possible the heat conduction through the sodium in this region.

3.1.3 Transient flow and leakage paths

The flow paths of high-pressure sodium through the upper bearing region are shown by arrows in fig. 3.1. A flow rate of 75 GPM is used for the sodium at pony motor speed. ⁽⁷⁾ The effect of transient mitigation through the feed cylinder is small as shown in the next section. Therefore the full transient will be applied in the high-pressure sodium flowing through the bearings. The transient sodium leaks out through the gap between the inner and outer regions of the upper bearing into the bulk sodium in the upper pump tank.

3.1.4 Insulated surfaces

The cut boundaries of the thermal model are assumed to be thermally insulated as shown in fig. 3.3.

3.2 Structural model

The finite element model for the structural analysis is shown in fig. 3.4. The structural axisymmetric isoparametric elements (STIF 42) are used in place of STIF 55 in the thermal model. The elements representing the stagnant sodium in the thermal model have no counterpart in the structural model. Instead, pressure boundary conditions are applied on the walls surrounding the stagnant sodium, as well as on all other wall surfaces.

3.2.1 Pressure boundary

The high pressure regions where sodium flows at 75 GPM apply a pressure of 200 PSI on the walls. These regions include the feed cylinder, the pocket area in the outer bearing region, and the bearing gap. A pressure of 30 PSI is applied to all other walls, which include the inside shaft region, the pocket area in the inner bearing region, and the outer walls. A linear pressure drop is assumed at the bearing gap where the high-pressure sodium flows out into the upper pump tank.

3.2.2 Weldments and Gaps

The locations of welds in the upper bearing are shown in fig. 3.1. It is seen that a gap exists between the journals and the outer cylinder at the outer region of the bearing. Since the scope of the present analysis is limited to linear elastic analysis (except for the temperature dependent material properties) this gap is not included in the finite element model.

The effect of gapping is studied with hand calculations in a later section. It will be shown that the distance of separation under the most severe transient conditions is small, therefore the linear elastic model is adequate for the present analysis in estimating the transient effects. It should also be noted that the primary pressure load does not cause separation of the gap.

3.2.3 Coupled degrees of freedom

The structural model of the upper bearing is extended at the lower boundaries to a distance larger than $2\sqrt{Rt}$ beyond the area of structural discontinuity. These boundaries are confined to move in the radial direction only. The upper boundary of the inner shaft is also extended to a distance larger than $2\sqrt{Rt}$ beyond the discontinuity area. A plane strain boundary condition is applied at this section. Thus all nodes lying in the upper boundary plane are coupled as far as the axial displacement degree of freedom is concerned. Fig. 3.5 depicts the pressure and displacement boundary conditions for the structural model.

50.0

44.9

39.8

34.7

29.6

24.5

19.4

14.3

9.2

4.1

-1.0

0.0

5.1

10.2

15.3

20.4

25.5

30.6

35.7

40.8

STRUCTURAL MODEL

UPPER BEARING

FIGURE 3.4

UPPER BEARING

FIGURE 3.5

STRUCTURAL MODEL

UPPER BEARING

4.0 Boundary Conditions

4.1 Thermal hydraulic boundary conditions

Considerable attention is given to the determination of temperature boundary conditions in the present analysis, since the thermal stresses and deformations are largely determined by the temperature gradients through various sections of the bearing walls. Three sets of temperature-time curve are required to fully describe the thermal transient boundary condition. These are $T_{trans}(t)$, the input sodium transient temperature; $T_{in}(t)$, the bulk temperature inside the shaft; and $T_{out}(t)$, the bulk temperature in the upper pump tank. These temperatures are listed in Table 4.1 and are discussed below.

4.1.1 Thermal transients

The thermal transients considered in this analysis are listed in Table 4.2. The most severe transient is the IP3UT which rises 180°F in 120 seconds, and then drops 265°F in 250 seconds. This transient will be analyzed in detail using the finite element model described in the previous section. Secondary thermal stresses as well as thermal deformations for the IP3UT transient are calculated in the computer analysis. Other transients will be investigated with hand calculations and their effects included in the creep-fatigue evaluation.

4.1.2 Effect of transient mitigation

Since the transient sodium has to travel about 5 feet in the vertical feed cylinder before it reaches the upper bearing, it is possible that some heat loss may incur along the way. The effect of mitigation is estimated

4.0 Boundary Conditions

4.1 Thermal hydraulic boundary conditions

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4.1.2 Effect of transient mitigation

Since the transient sodium has to travel about 5 feet in the vertical feed cylinder before it reaches the upper bearing, it is possible that some heat loss may incur along the way. The effect of mitigation is estimated

with a steady state model of one-dimensional flow in a long cylinder, including the heat loss through the cylinder wall and the effects of convection at both sides of the wall. The temperature of sodium in the cylinder, $T(x)$, is given by (see Appendix I)⁽⁸⁾:

$$T = (T_1 - T_0) \exp \left\{ \frac{1}{2} \left[\frac{\rho c v}{k} - \left(\left(\frac{\rho c v}{k} \right)^2 + \frac{8H}{kd} \right)^{1/2} \right] x \right\} + T_0$$

where T_1 is the original temperature of sodium at $X=0$, T_0 is the bulk temperature outside the wall, V is the velocity of the sodium flow; k, ρ, c are respectively the conductivity, density and specific heat of sodium; d is the radius of the cylinder and H is given by:

$$H = h_i h_o k_w / (k_w h_o + l h_a h_o + k_w h_i)$$

where k_w is the conductivity in the wall, l is the thickness and h_i, h_o are the film coefficients on the inside and outside wall surfaces. respectively.

The applicable parameters are:

$$T_0 = 770^\circ$$

$$T_1 = 1005^\circ$$

$$d = (r_2^2 - r_1^2)^{1/2} = (14^2 - 8^2)^{1/2} = 11.5 \text{ in}$$

$$l = 2 \text{ in}$$

$$X = 64 \text{ in}$$

$$V = 75 \text{ GPM} = 288.75 \text{ in}^3/\text{sec} = \frac{288.75}{(11.5)^2} \text{ in/sec} \\ = .695 \text{ in/sec}$$

we find $T(x = 64 \text{ in}) = 964^\circ$. The steady - state analysis shows that the maximum temperature drop along the feed cylinder is 41° . Since the IP3UT transient temperature ⁽⁹⁾ rises gradually from 770° to 1005° and then decreases gradually to 730° , the actual transient temperature variation along the cylinder

is less than the maximum calculated from the steady-state model. As a conservative measure, the full IP3UT transient shown in fig. 4.1 is applied to the convective surfaces in the bearing region through which the high-pressure sodium flows.

4.1.3 Transient response inside pump shaft

The temperature of the stagnant sodium inside the pump shaft is calculated with ANSYS using a one-dimensional model with three isoparametric elements across the thickness of the wall, with $h\Delta T$ applied at both the inner and outer surfaces of the wall. The initial temperature inside the bulk sodium is 770° . The change of the shaft temperature due to the transient sodium outside the shaft also affects the bulk sodium temperature inside the shaft. The transient response lags behind the applied transient as shown in fig. 4.1. Since the volume of sodium inside the shaft is relatively small, its temperature response vs time is significant, and the maximum temperature difference reaches 915°F at 620 sec. The calculated temperatures, T_{in} , as shown in Table 4.1 are used as thermal boundary condition on the inner wall of the pump shaft in the full finite element model.

4.1.4 Transient response in the upper pump tank

The bulk temperatures of sodium in the upper pump tank are determined by convection and heat conduction through the walls, and by the mixing of the transient effluents through the bearing gap with the bulk sodium.

The effects of convection and conduction are calculated in a similar way as described in the previous section. A one-dimensional model with three isoparametric elements across the wall thickness is used in the ANSYS program to determine the transient response of the sodium in the upper tank. The

transient $T_{trans}^{(t)}$, in this case is applied at the inner surface of the wall. A large conductivity is assumed for the outside bulk of sodium. The film coefficients are taken to be 0.002 BTU/sec/in²/°F for the transient convective surface, and 0.001 BTU/sec/in²/°F for the outer convective surface. The results of the transient response are shown along with the original transient in fig. 4.2. The rise of temperature is considerably slower in this case since the volume of the bulk sodium is much larger than that of the sodium inside the shaft.

4.1.5 Mixing contribution to the bulk temperature at upper pump tank

The transient sodium leaks from the upper bearing gap at the rate of 75 GPM and enters into the bulk sodium in the pump tank. The effect of mixing is calculated with an adiabatic mixing model (see Appendix II). In this model, piece-wise linear thermal transient flow enters an insulated volume V at a flow rate of $\dot{q}$. The transient fluid is assumed to mix thoroughly with the bulk fluid in V , whose response temperature as a function of time can be expressed in terms of the parameter $\gamma = \frac{\dot{q}}{V}$. A FORTRAN computer program is written for the calculation of the mixing response temperature, with γ and the piece-wise linear transient as input. There is no restriction on the number of piece-wise linear segments of the input transient temperatures. For $\gamma = 0.000287$ and an input transient as shown in fig. 4.2, the temperature history of the upper tank bulk sodium due to the mixing effect is computed with this program, and the results are shown in fig. 4.2.

The combined effects of mixing and convection/conduction through the walls are included in the temperature history curve, $T_{out}^{(t)}$, (Fig. 4.1) which is used as the input thermal boundary condition for the surfaces exposed to the pump tank sodium in the finite element model.

4.1.6 Convection film coefficients

Sodium convection coefficients for heat transfer between exposed structure and liquid sodium can be estimated from the following formulae:

Forced convection (Lyon-Martinelli Equation)⁽¹⁰⁾ —

$$h = \frac{k}{D} [7.0 + .025 (Pe)^{.8}] = \frac{.0634}{D} [1 + .02268 (V \cdot D)^{.8}]$$

Natural convection (McDonald - Connally Equation)⁽¹¹⁾ —

$$h = .0785 \frac{k}{D} [Gr \cdot Pr]^{.32} = .000751 \Delta T^{.32} / D^{.4}$$

where D is the effective hydraulic diameter, V the velocity, k the conductivity; Pe, Gr, Pr are respectively Peclet's number, Grashof's number and Prandtl's number⁽¹²⁾; ΔT is the temperature difference between the structural surface and the bulk sodium.

We adopt the following values for the film coefficients in this analysis:

$$h = 0.01 \text{ BTU/sec} - \text{in}^2 - ^\circ\text{F} \quad \text{for the bearing gap region}$$

$$h = 0.001 \text{ BTU/sec} - \text{in}^2 - ^\circ\text{F} \quad \text{for outside surfaces exposed to the sodium in the pump tank}$$

$$\text{abd} \quad h = 0.002 \text{ BTU/sec} - \text{in}^2 - ^\circ\text{F} \quad \text{for all other convective surfaces}$$

These values of film coefficients are obtained after representative calculations for typical regions are made using the aforementioned equation.⁽¹³⁾

4.2 Structural boundary conditions

The boundary conditions for the structural finite element model are discussed in Section 3.2. The effect of gapping is discussed in Section 9.5.

t (SEC)	$T_{TRANSIENT}$	T_{MIX}	$\Delta T_{MIX} \equiv T_{MIX} - 770$	T_{I-D}	$\Delta T_{ID} \equiv T_{I-D} - 770$	$T_{OUT} \equiv 770 + \Delta T_{MIX} + \Delta T_{I-D}$	$T_{TRANS} - T_{IN}$	T_{IN}
70	920	772	2	770	0	772	150	770
120	950	774	4	770	0	774	178	772
220	955	779	9	772	2	781	156	799
380	1005	788	18	776	6	794	155	850
470	1005	793	23	778	8	801	128	877
620	810	798	28	783	13	811	-99	909
720	740	798	28	784	14	812	-164	904
820	720	796	26	784	14	810	-158	878
1000	730	792	22	783	13	805	-95	825

UPPER BEARING
TRANSIENT BOUNDARY CONDITIONS

TABLE 4.1

Page 17

TRANSIENT	ΔT_i	CYCLES	MAXIMUM METAL TEMPERATURE
IP-1UT	70	480	770.
IP-3UT	285	15	1005.
IP-5UT	80	10	770
IP-6UT	90	5	770
IP-7UT	125	14	770
IP-8UT	150	63	910
IP-1ET	250	8	1000
IP-2ET	155	8	770
IP-5ET	155	8	770

TABLE 4.2TRANSIENTS FOR THELARGE SCALE SODIUM PUMP

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BY-----DATE-----
CHKD.-----DATE-----
SHEET NO.-----OF-----
P.J.-----

FIG. 4.1 TRANSIENT RESPONSE VS TIME
UPPER BEARING

BY _____ DATE _____ SHEET NO. _____ OF _____
CHKD. _____ DATE _____ PJ. _____

FIGURE 4.2

TEMPERATURE HISTORY OF BULK SODIUM
IN UPPER PUMP TANK

5.0 Material Properties

The thermophysical properties of SS316 are summarized in Table 5-1, and of sodium in Table 5-2. Values for these physical constants at various temperatures are taken from Nuclear Systems Materials Handbook, Vol. 1 & 2: Design Data (TID6666)⁽¹⁴⁾. A polynomial fit is obtained for each set of physical constants in the temperature range 400°F - 1100°F. The lowest degree of polynomial which meets the following criterion is used:

$$1 - D_0 < 10^{-4}$$

where D_0 is the coefficient of determination for a given degree of polynomial. The coefficients and the degree of the polynomials are found using a small computer program. They are summarized in Tables 5-3 and 5-4, for SS316 and sodium, respectively. Only static values of the sodium thermal conductivity are used.

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T °F	E PSI	ν	α /°F	ρ LB/IN ³	K BTU/ SEC-IN-°F	C BTU/ LB-°F
400	.2664E8	.2795	9.601E-6	.28495	2.2762E-4	.1255
500	.2607E8	.2838	9.710E-6	.28409	2.3773E-4	.1278
600	.2544E8	.2881	9.822E-6	.28322	2.4769E-4	.1296
700	.2477E8	.2923	9.942E-6	.28229	2.5787E-4	.1310
800	.2406E8	.2966	10.07E-6	.28142	2.6782E-4	.1323
900	.2331E8	.3009	10.19E-6	.28056	2.7801E-4	.1338
1000	.2253E8	.3051	10.30E-6	.27963	2.8796E-4	.1355
1100	.2172E8	.3094	10.39E-6	.27876	2.9815E-4	.1375

PHYSICAL PROPERTIES OF SS316 VS TEMPERATURE

TABLE 5-1

T °F	P LB/IN ³	K BTU/SEC-IN-°F	C BTU/LB-°F
400	.032604	1.0910E-3	.3195
500	.032135	1.0519E-3	.3146
600	.031661	1.0137E-3	.3105
700	.031181	.97639E-3	.3070
800	.030700	.94028E-3	.3042
900	.030220	.90509E-3	.3020
1000	.029740	.87083E-3	.3006
1100	.029253	.83750E-3	.2998

PHYSICAL PROPERTIES OF SODIUM VS TEMPERATURE

TABLE 5-2

BASIC TECHNOLOGY, INC.

	E PSI	ν	α /°F	ρ LB/IN ³	K BTU/SEC-IN-°F	C BTU/LB-°F
N	2	2	3	2	2	4
C_0	.286111E8	.262363	9.436	.288454	1.87502E-9	.102262
C_1	-4.12385E3	4.29415E-5	-2.7098E-4	-8.6885E-6	1.00213E-7	1.015E-4
C_2	-1.95235	-1.78253E-10	2.11104E-6	-1.187E-10	3.15717E-13	-1.4453E-7
C_3	—	—	-9.7765E-10	—	—	9.8976E-11
C_4	—	—	—	—	—	-2.2724E-14

COEFFICIENTS OF POLYNOMIAL FIT-SS316

TABLE 5-3

	ρ LB/IN ³	K BTU/SEC-IN-°F	C BTU/LB-°F
N	2	2	2
C ₀	3.44764 E - 2	1.25747 E - 3	.345607
C ₁	-4.63469 E - 6	-4.35789 E - 7	-7.88338 E - 5
C ₂	-1.03528 E - 10	4.91145 E - 11	3.38099 E - 8

COEFFICIENTS OF POLYNOMIAL OF FIT-SODIUM

TABLE 5-4

6.0 Thermal transient solutions

The temperature solutions at all nodes shown in the thermal finite element model of fig. 3.2 are obtained using the ANSYS computer program. A total of 9 load steps and 36 iterations are used in the transient analysis (fig. 6.1). The temperature histories for all convective boundaries as calculated in Section 4.0, as well as the input transient temperatures, are inputted at each time point during the iterations. Three axisymmetric isoparametric elements (STIF 55) are used through all wall thicknesses, as this mesh size has been previously verified to be adequate.⁽⁶⁾ The physical properties of SS316 and sodium are temperature dependent. The calculated nodal temperatures, blocked according to the element identification numbers, are saved on tapes for all 36 time points at the completion of the thermal analysis.

ANSYS LOAD INPUT FOR IP-3U TRANSIENT

FIGURE 6.1

7.0 Thermal stress calculations

7.1 Selection of time points

As the criteria for ratchetting, creep-fatigue and operability evaluations are based on the maximum linearized and peak stresses, it is necessary in the stress analysis to use only the temperatures calculated at the time points which would give these maximum values. Two effects may play major roles in the selection of these time points. The first is the variation of temperatures through the wall thickness where the stresses are expected to be the highest. The locations of the highly stressed areas generally correspond to relatively thick walls with differential transients applied on both sides of the wall. The second is the " $T_a T_b$ effect",⁽¹⁵⁾ which estimates the discontinuity stresses by the correlation of temperature differences at points across the boundaries of the discontinuity. Since the geometry of the bearing region is regular and the walls are of fairly uniform thickness, it is expected that the first effect is the more important in this case. Therefore, a plot of (T surface - T mean) vs. time through the wall at critical regions should give an indication of when the maximum stresses may occur. Figures 7.1 to 7.6 show that for most sections chosen, the peaks occur at $t = 120$ sec, 380 sec and 720 sec. A more detailed stress history calculation from a one-dimensional calculation with the corresponding temperature histories applied at either side of the wall also show that the stress levels are the highest at these time points. (Fig. 7.7 and 7.8) While the $T_a T_b$ effect may have slightly different peak time locations, its magnitudes are significantly lower than the T surface - T mean effect. We therefore use the three time points chosen for all thermal stress calculations.

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BY _____ DATE _____ SHEET NO. _____ OF _____
CHKD. _____ DATE _____ PJ. _____

$\Delta T (^{\circ}F)$

UPPER BEARING, INNER REGION. AT A

FIGURE 7.1

BY _____ DATE _____ SHEET NO. _____ OF _____
CHKD. _____ DATE _____ PJ. _____

$\Delta T(^{\circ}F)$

110

100

80

60

40

20

0

-20

-40

-60

-80

-100

100

200

300

400

500

600

700

800

900

1000

t (SEC)

UPPER BEARING INNER REGION AT B

FIGURE 7.2

BY _____ DATE _____ SHEET NO. _____ OF _____
CHKD. _____ DATE _____ PJ. _____

BY _____ DATE _____ SHEET NO. _____ OF _____
CHKD. _____ DATE _____ P.J. _____

$\Delta T, (\%)$

110
100
90
80
70
60
50
40
30
20
10
0
-10
-20
-30
-40
-50
-60
-70
-80
-90

100 200 300 400 500 600 700 800 900 1000 1100
t (SEC)

UPPER BEARING, OUTER REGION AT A

FIGURE 7.4

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BY-----DATE----- SHEET NO-----OF-----
CHKD.-----DATE----- PJ.-----

UPPER BEARING OUTER REGION AT B

FIGURE 7.5

BY _____ DATE _____ SHEET NO. _____ OF _____
CHKD. _____ DATE _____ PJ. _____

BY-----DATE----- SHEET NO-----OF-----
CHKD.-----DATE----- PJ-----

BASIC TECHNOLOGY, INC.

BY _____ DATE _____ SHEET NO _____ OF _____
CHKD. _____ DATE _____ PJ. _____

STRESS
INTENSITY (ksi)

1-D STRESS HISTORY IN INNER SHAFT WALL

FIGURE 7.7

BY-----DATE----- SHEET NO-----OF-----
CHKD.-----DATE----- PJ-----

7.2 Finite element results

Thermal stresses and deflections are calculated at the selected time points using the nodal temperatures stored on tape from the thermal transient analysis. The thermal elements representing the stagnant sodium in the inner region of the bearing are taken out in the stress analysis. Other thermal isoparametric elements are replaced by the equivalent structural isoparametric elements (STIF 42). Computer outputs of the finite element analysis include printed tables of nodal deflections and stress components at four nodes as well as at the centroid of each of the elements. In addition, post-processing plots of deflected shapes, hoop stress contours and stress intensity contours are obtained at each of the selected time points. These are shown in fig. 7.9 to 7.27. The contour plots identify locations of the highest and lowest stresses, as well as areas of high stress concentrations. Sections with high peak stresses and/or high membrane stresses are selected from these contour plots and are analyzed in detail in Section 9.

7.3 Hand Calculations

All other transients except the IP3UT are analyzed with hand calculations to obtain the maximum thermal stresses for the fatigue evaluation.

STEP=

1 ITERATION= 1

DEFLECTION AT $t = 120 \text{ SEC}$

FIGURE 7.9

UPPER BEARING INNER REGION THERMAL STRESS POSTPROCESSING PLOT

STEP=

1

ITERATION= 1

STRESS INTENSITY AT $t=120$ SEC.
INCREMENT = 2000 PSI
FIGURE 7.10

UPPER BEARING INNER REGION THERMAL STRESS POSTPROCESSING PLOT

STEP=

1

ITERATION=

1

HOOP STRESS AT $t = 120$ SEC
INCREMENT = 4000 PSI
FIGURE 7.11

UPPER BEARING INNER REGION THERMAL STRESS POSTPROCESSING PLOT

STEP= 1 ITERATION= 1

DEFLECTION AT $t = 120$ SEC
FIGURE 7-18

UPPER BEARING OUTER REGION THERMAL STRESS POSTPROCESSING PLOT

STEP= 1 ITERATION= 1

STRESS INTENSITY AT $t = 120 \text{ SEC.}$
INCREMENT = 2500 PSI
FIGURE 7.20

UPPER BEARING OUTER REGION THERMAL STRESS POSTPROCESSING PLOT

STEP= 1 ITERATION= 1

HOOP STRESS AT $t = 120 \text{ SEC.}$
INCREMENT = 4000 PSI
FIGURE 7.19

UPPER BEARING OUTER REGION THERMAL STRESS POSTPROCESSING PLOT

STEP=

2

ITERATION=

1

DEFLECTION AT $t = 380$ SEC.

FIGURE 7.12

UPPER BEARING INNER REGION THERMAL STRESS POSTPROCESSING PLOT

STEP=

2

ITERATION=

1

STRESS INTENSITY AT $t=380$ SEC
INCREMENT = 2000 PSI
FIGURE 7.13

UPPER BEARING INNER REGION THERMAL STRESS POSTPROCESSING PLOT

STEP= 2 ITERATION= 1

0-31691 HOOP STRESS AT $t = 380 \text{ SEC.}$
INCREMENT = 4000 psi
FIGURE 7.14

X 25212

UPPER BEARING INNER REGION THERMAL STRESS POSTPROCESSING PLOT

STEP= 2 ITERATION= 1

DEFLECTION AT $t = 380$ SEC.

FIGURE 7.21

UPPER BEARING OUTER REGION THERMAL STRESS POSTPROCESSING PLOT

STEP= 2 ITERATION= 1

STRESS INTENSITY AT $t=380$ SEC
INCREMENT = 4000 PSI
FIGURE 7.23

UPPER BEARING OUTER REGION THERMAL STRESS POSTPROCESSING PLOT

STEP= 2 ITERATION= 1

HOOP STRESS AT $t = 380 \text{ SEC}$

INCREMENT = 5000 PSI

FIGURE 7.22

UPPER BEARING OUTER REGION THERMAL STRESS POSTPROCESSING PLOT

STEP=

3 ITERATION= 1

DEFLECTION AT $t=720$ SEC.
FIGURE 7.15

UPPER BEARING INNER REGION THERMAL STRESS POSTPROCESSING PLOT

STEP=

3

ITERATION=

1

STRESS INTENSITY AT $t=720$ SEC
INCREMENT = 2000 PSI
FIGURE 7.16

1769

UPPER BEARING INNER REGION THERMAL STRESS POSTPROCESSING PLOT

STEP =

3

ITERATION =

1

HOOP STRESS AT $t = 720$ SEC.

INCREMENT = 4000 PSI

FIGURE 7.17

UPPER BEARING INNER REGION THERMAL STRESS POSTPROCESSING PLOT

STEP= 3 ITERATION= 1

DEFLECTION AT $t = 720 \text{ SEC}$
FIGURE 7.24

UPPER BEARING OUTER REGION THERMAL STRESS POSTPROCESSING PLOT

STEP= 3 ITERATION= 1

HOOP STRESS AT $t = 720$ SEC.
INCREMENT = 4000 PSI
FIGURE 7.25

UPPER BEARING OUTER REGION THERMAL STRESS POSTPROCESSING PLOT

STEP= 3 ITERATION= 1

STRESS INTENSITY AT $t=720$ SEC.
INCREMENT = 2000 PSI
FIGURE 7.26

UPPER BEARING OUTER REGION THERMAL STRESS POSTPROCESSING PLOT

8.0 Mechanical loads

The primary loads of the bearing region consist of pressure, centrifugal force, torsion and weight in the inner region; and of pressure only in the outer region. The centrifugal force and torsion loads are imposed on the inner bearing region as a result of the shaft rotation.

8.1 Pressure

The pressures applied at various boundaries of the bearing walls are indicated in fig. 3.5. The displacement boundary conditions of the finite element model are the same as those of the thermal stress model discussed in Section 3.2.3.

8.2 Centrifugal force

The rotating shaft imparts a centrifugal force to the inner region of the bearing, in addition to the pressure loads. This effect combined with the pressure load in the ANSYS run by specifying an angular velocity in the axial direction. The rotational speed of the shaft at normal operation ⁽¹⁶⁾ is $\omega = 600 \text{ RPM} = 20\pi \text{ rad/sec}$. The hoop stress and stress intensity contour plots for the upper bearing region are included in fig. 8.1 - 8.4.

8.3 Torsion

The torsional stress in the shaft can be calculated with the following formula⁽¹⁷⁾:

$$\text{Max } \sigma = \frac{2Tr_1}{\pi(r_1^4 - r_0^4)} \quad \text{at the outer boundary}$$

where T is the torque; r_1 and r_0 are, respectively, the outer and inner radius of the shaft.

During normal operation, the pump operates⁽¹⁶⁾ at 18000 HP and the

shaft rotates at a speed of 600 RPM, thus

$$T = 1.89 \times 10^6 \text{ in-lb.}$$

With $r_1 = 8 \text{ in.}$ and $r_0 = 6 \text{ in.}$, we find $\text{Max } \sigma = 3.4 \text{ ksi.}$

8.4 Weight

The axial stress in the shaft due to its own weight is calculated with $\sigma = P/A = \rho l$ where $\rho = .28 \text{ lb/in}^3$ is the density of the shaft, A is the cross sectional area and $l \approx 800 \text{ in}$ represents the equivalent length of the shaft, including the impellers. The axial stress σ is thus 224 PSI.

Table 10.1 summarizes the maximum primary stresses in the inner and outer regions of the upper bearing and support.

HOOP STRESS

INCREMENT = 150 PSI

FIGURE 8.1

STEP =

1 ITERATION = 1

STRESS INTENSITY
INCREMENT = 100 PSI
FIGURE 8.2

1871

46

UPPER BEARING INNER REGION - PRESSURE AND ROTATION

STEP= 1 ITERATION= 1

HOOP STRESS
INCREMENT 150 PSI
FIGURE 8.3

UPPER BEARING OUTER REGION - PRESSURE LOADING

STEP= 1 ITERATION= 1

STRESS INTENSITY
INCREMENT 150 PSI
FIGURE 8.4

UPPER BEARING OUTER REGION - PRESSURE LOADING

9.0 Data Reduction

This section contains the calculations necessary to reduce the raw data into the desired form for comparison with allowables. All stress results are summarized for various cross-sections in tables 9.1 through 9.7.

9.1 Membrane Stresses

Membrane stresses through a wall section are obtained by averaging the σ_x , σ_y , τ_{xy} and τ_z components, calculating principal stresses from the averaged components and then determining stress intensities from the largest difference in principal stresses. They are summarized for each thermal stress and primary load run in tables 9.1 through 9.7.

9.2 Linearized Stress

Linearized stresses are calculated for the surfaces of each cross-section. First, linearized components are determined from the following equations.

$$\sigma_a = \bar{\sigma} - \frac{6}{t^2} \sum_j \sigma_j x_j d_j$$

$$\sigma_b = \bar{\sigma} + \frac{6}{t^2} \sum_j \sigma_j x_j d_j$$

where σ_a , σ_b = linearized components at surfaces A and B

x_j = distance from centroid

d_j = width of j th element

t = width of cross-section

N = number of elements through section

Then, principal stresses are calculated at the surfaces from the linearized components σ_x , σ_y , τ_{xy} and τ_z . Finally, stress intensities are determined by the largest difference of the principal stresses. They are

summarized for each thermal stress and primary load run in tables 9.1 through 9.7.

9.3 Peak Stress

Surface stress intensities are calculated by ANSYS at each free surface, thus eliminating the need for extrapolation to the surfaces. They are used directly as peak stress intensities and are summarized at each surface in tables 9.1 through 9.7.

9.4 Bearing Clearances

Radial displacements at adjacent nodes on the inner and outer bearing regions are summarized at 120,380 and 720 seconds in tables 9.8 through 9.10. The clearances between adjacent nodes are then calculated and plotted in figures 9.1 and 9.2. These figures assume an initial clearance of 0.030 inches and show the gap size between adjacent nodes of the inner and outer regions of the bearing. The maximum closure in the this region during the IP-3UT occurs between nodes 390 and 420 at time = 120 seconds. Thus, assuming an initial 0.030 inch radial bearing clearance, the minimum clearance during transient conditions is 0.024 inches.

In addition, the gapping effect, discussed in Section 9.5, may further reduce the clearance up to 0.004 inches, leaving a net minimum clearance of 0.020 inches.

This analysis does not cover the severe negative transients such as the IP-1ET. The effect of this transient is discussed in the Conclusions.

9.5 Gapping Effects

The journal of the upper outer bearing region is not integral with the

support. The journal is welded to the support at the top and is considered a slip fit inside the support at room temperature. This condition of no prestress/no initial clearance is maintained at all steady state uniform temperatures since the material properties are the same.

The finite element model, however, comprises an integral journal. Figures 7.7 and 7.26 show the contour plots of hoop stress and stress intensity in this region at three transient times. From these plots, and from the detailed computer output, the stresses and displacements do not vary significantly in the axial direction except for end effects. Thus, a study of a representative cross-section will give results which are valid for most of the length of the journal. Therefore, the gapping effect may be studied by hand calculations using the long cylinder assumption. The temperatures across the thickness of the long cylinders are taken from the finite element results. The section chosen is nodes 446 through 596 of the outer region.

Table 9.11 summarizes thermal and stress results for each node through the thickness. Average wall temperatures are then calculated for the journal and support. Since the average inner wall temperature for stress iterations 1 and 12 is greater than the average outer wall temperature the walls are considered to be in contact at these times. At stress iterations, the outer wall is hotter, thus causing separations. The amount of separation based on the radius, is: $\text{clearance} = \alpha r \Delta T = (10 \times 10^{-6})(12)(31) = 3.72 \times 10^{-3} \text{ inch} = 0.00372 \text{ in.}$

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ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
4	.165	18.276	.013	16.833
5	1.887	4.697	.142	2.835
6	1.106	-19.164	.020	-19.67

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	σ_1 (KSI)	σ_2 (KSI)
AVERAGE SECTION	1.253	1.270	.058	0	1.320	1.203
LINEARIZED: FACE A	1.026	26.23	.053	24.335	26.230	1.026
FACE B	1.480	-23.69	.063	-24.335	1.480	-23.690

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	25.204	23.579
FACE B	25.815	33.799
SECTION AVERAGE	0.66	

NOTES

$$1. \sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

SECTION LOCATION

UPPER BEARING INNER REGION
STRESS AT T = 120 SEC.
TABLE NO. 9.1 (A)

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BASIC TECHNOLOGY, INC.

ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
4	0.715	16.796	0.11	15.428
5	1.668	2.683	0.131	1.098
6	0.917	-16.135	0.021	-16.529

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	σ_1 (KSI)	σ_2 (KSI)
AVERAGE SECTION	1.10	1.115	-0.087	0	1.195	1.020
LINEARIZED: FACE A	0.965	23.069	0.146	21.305	23.07	0.964
FACE B	1.23	-20.839	0.028	-21.305	1.230	-20.839

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	22.106	23.202
FACE B	22.535	26.764
SECTION AVERAGE	1.195	

NOTES

$$1. \quad \sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

SECTION LOCATION

UPPER BEARING INNER REGION
STRESS AT T = 380 SEC.
TABLE NO. 9.1 (B)

Page 66

BASIC TECHNOLOGY, INC.

ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
4	-0.704	-16.619	-0.027	-15.428
5	-1.725	-4.037	-0.172	-2.432
6	-0.989	17.217	-0.029	17.719

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	σ_1 (KSI)	σ_2 (KSI)
AVERAGE SECTION	-1.139	-1.146	-0.076	-0.047	1.066	-1.219
LINEARIZED: FACE A	-0.949	-23.703	-0.075	-22.145	-0.949	-23.703
FACE B	-1.329	21.411	-0.077	22.051	21.411	-1.329

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	22.754	22.187
FACE B	23.38	30.352
SECTION AVERAGE	2.285	

NOTES

1. $\sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$

SECTION LOCATION

UPPER BEARING INNER REGION
STRESS AT T = 720 SEC.
TABLE NO. 9.1 (C)

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BASIC TECHNOLOGY, INC.

ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
34	-25.989	-7.974	-4.892	-27.293
35	8.942	-1.764	-2.115	3.014
36	30.641	4.055	-4.080	19.075

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	σ_1 (KSI)	σ_2 (KSI)
AVERAGE SECTION	4.531	-1.894	-3.696	-1.735	6.215	-3.578
LINEARIZED: FACE A	-33.222	-9.913	-4.237	-32.647	-9.167	-33.968
FACE B	42.284	6.125	-3.154	29.177	42.557	5.852

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	24.801	52.386
FACE B	36.705	36.821
SECTION AVERAGE	9.793	

NOTES

$$1. \sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

SECTION LOCATION

UPPER BEARING INNER REGION
STRESS AT $T = 120$ SEC.
TABLE NO. 9.2 (A)

BASIC TECHNOLOGY, INC.

ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
34	-20.867	-6.789	-9.625	-23.546
35	7.691	-1.422	-2.214	-0.154
36	28.579	3.368	-3.605	14.127

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	σ_1 (KSI)	σ_2 (KSI)
AVERAGE SECTION	5.134	-1.614	-3.481	-3.191	6.608	-3.088
LINEARIZED: FACE A	-27.83	-8.385	-4.161	-28.306	-8.870	-30.02
FACE B	38.098	5.157	-2.801	21.924	38.334	4.921

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	21.15	42.117
FACE B	33.413	40.028
SECTION AVERAGE	9.696	

NOTES

$$1. \sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

SECTION LOCATION

UPPER BEARING INNER REGION
STRESS AT T = 380 SEC.
TABLE NO. 9.2 (B)

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BASIC TECHNOLOGY, INC.

ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
34	26.955	7.909	5.278	26.749
35	-8.534	1.359	2.555	-3.011
36	-30.837	-4.541	4.564	-18.931

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	σ_1 (KSI)	σ_2 (KSI)
AVERAGE SECTION	-4.139	1.576	4.132	1.602	3.742	-6.305
LINEARIZED: FACE A	34.389	9.876	4.608	32.055	35.227	9.038
FACE B	-42.667	-6.724	3.656	-28.851	-6.356	-43.035

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	26.189	53.922
FACE B	36.679	43.520
SECTION AVERAGE	10.047	

NOTES

$$1. \quad \sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

SECTION LOCATION

UPPER BEARING INNER REGION
STRESS AT T = 720 SEC.
TABLE NO. 9.2 (C)

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BASIC TECHNOLOGY, INC.

ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
139	2.304	.371	4.444	-12.093
140	1.708	-1.715	4.136	-22.269
141	.623	-6.223	1.886	-35.131

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	σ_1 (KSI)	σ_2 (KSI)
AVERAGE SECTION	1.545	-2.522	3.489	-23.164	3.550	-4.527
LINEARIZED: FACE A	NOT APPLICABLE					
FACE B	.424	-6.918	1.784	-38.523	.835	-7.33

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	NOT APPLICABLE	NOT APPLICABLE
FACE B	39.358	43.55
SECTION AVERAGE	26.719	

NOTES

$$1. \sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

SECTION LOCATION

UPPER BEARING INNER REGION
STRESS AT T = 120 SEC
TABLE NO. 9.3 (A)

BASIC TECHNOLOGY, INC.

ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
139	1.945	-1.722	3.216	-16.428
140	1.567	-1.512	2.808	-25.062
141	.572	-3.688	1.123	-36.069

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	σ_1 (KSI)	σ_2 (KSI)
AVERAGE SECTION	1.361	-1.974	2.382	-25.853	2.601	-3.214
LINEARIZED: FACE A	NOT APPLICABLE					
FACE B	.446	-3.951	.987	-38.947	.657	-4.162

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	NOT APPLICABLE	NOT APPLICABLE
FACE B	39.604	43.847
SECTION AVERAGE	28.454	

NOTES

$$1. \quad \sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

SECTION LOCATION

UPPER BEARING INNER REGION
STRESS AT T=380 SEC
TABLE NO. 9.3 (B)

BASIC TECHNOLOGY, INC.

ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
139	-1.643	.201	-4.307	9.943
140	-1.325	1.444	-3.624	16.175
141	-.483	5.435	-1.664	24.740

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	σ_1 (KSI)	σ_2 (KSI)
AVERAGE SECTION	-1.150	2.36	-3.198	16.953	4.253	-3.043
LINEARIZED: FACE A	— NOT APPLICABLE —					
FACE B	-3.77	5.849	-1.436	26.818	6.164	-.692

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	— NOT APPLICABLE —	
FACE B	27.51	30.774
SECTION AVERAGE	19.996	

NOTES

1. $\sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$

SECTION LOCATION

UPPER BEARING INNER REGION
STRESS AT T = 720 SEC
TABLE NO. 9.3 (C)

BASIC TECHNOLOGY, INC.

ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
4	-494	-19.971	-127	-19.864
5	-847	3.362	-026	4.301
6	-406	14.913	-027	15.841

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	σ_1 (KSI)	σ_2 (KSI)
AVERAGE SECTION	-582	-565	-06	-093	-513	-634
LINEARIZED: FACE A	-641	-23.821	-127	-23.71	-640	-23.822
FACE B	-523	22.691	0	23.896	22.691	-523

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	23.182	35.188
FACE B	24.419	19.138
SECTION AVERAGE	.727	

NOTES

$$1. \quad \sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

SECTION LOCATION

UPPER BEARING OUTER REGION
STRESS AT T = 120 SEC
TABLE NO. 9.4 (A)

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BASIC TECHNOLOGY, INC.

ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
4	-478	-20.653	-0.059	-20.268
5	-935	-761	-198	3.070
6	-630	17.960	0.075	21.06

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	U_1 (KSI)	U_2 (KSI)
AVERAGE SECTION	-.681	-.644	.0713	1.287	-.589	-.736
LINEARIZED: FACE A	-.709	-26.386	-0.018	-26.265	-.709	-26.386
FACE B	-.782	25.098	.161	28.839	25.099	-.783

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	25.677	33.797
FACE B	29.622	29.847
SECTION AVERAGE	2.023	

NOTES

$$1. \sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

SECTION LOCATION

UPPER BEARING OUTER REGION
STRESS AT T=380. SEC.
TABLE NO. 9.4 (B)

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BASIC TECHNOLOGY, INC.

ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
4	.262	9.544	-.128	9.786
5	.371	-2.937	.188	-2.401
6	.065	-5.890	.091	-4.809

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	σ_1 (KSI)	σ_2 (KSI)
AVERAGE SECTION	.233	.239	.136	-.857	.372	-.010
LINEARIZED: FACE A	.364	10.528	-.161	10.587	10.531	.361
FACE B	-.102	-10.05	-.111	-8.873	-.103	-10.051

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	10.226	18.90
FACE B	10.154	5.861
SECTION AVERAGE	.847	

NOTES

1. $\sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$

SECTION LOCATION

UPPER BEARING OUTER REGION
STRESS AT T = 720 SEC.
TABLE NO. 9.4 (C)

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BASIC TECHNOLOGY, INC.

ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
24	12.249	-1.460	2.860	19.084
25	5.280	-3.667	6.756	9.820
26	-22.431	-7.334	-1.466	-16.599

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	σ_1 (KSI)	σ_2 (KSI)
AVERAGE SECTION	-1.634	-4.154	2.717	4.102	.101	-5.889
LINEARIZED: FACE A	21.486	-2.238	5.601	27.891	22.837	-1.589
FACE B	-24.754	-8.07	-1.167	-19.687	-8.068	-24.756

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	29.480	23.790
FACE B	16.688	45.071
SECTION AVERAGE	9.991	

NOTES

$$1. \sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

SECTION LOCATION

LIPPER BEARING OUTER REGION
STRESS AT T = 120 SEC.
TABLE NO. 9.5 (A)

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BASIC TECHNOLOGY, INC.

ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
24	22.892	-1.592	7.113	31.256
25	2.351	-2.433	13.698	12.504
26	-34.311	-6.284	1.289	-17.537

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	σ_1 (KSI)	σ_2 (KSI)
AVERAGE SECTION	-3.023	-3.436	7.367	8.741	4.140	-10.599
LINEARIZED: FACE A	35.112	-3.308	11.250	41.270	38.383	-3.579
FACE B	-41.158	-6.564	3.484	-23.788	-6.217	-41.503

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	44.849	43.201
FACE B	35.288	61.845
SECTION AVERAGE	19.34	

NOTES

$$1. \sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

SECTION LOCATION

LIPPER BEARING OUTER REGION
STRESS AT T = 380 SEC.
TABLE NO. 9.5 (B)

BASIC TECHNOLOGY, INC.

ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
24	.995	.656	1.465	-2.250
25	-4.329	2.543	1.689	-3.343
26	3.557	4.860	3.194	7.558

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	σ_1 (KSI)	σ_2 (KSI)
AVERAGE SECTION	.074	2.686	2.116	.655	3.867	-1.107
LINEARIZED: FACE A	-1.634	-1.117	.963	-5.884	.350	-2.101
FACE B	1.782	5.489	3.269	7.194	7.393	-.122

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	6.234	5.966
FACE B	7.515	15.843
SECTION AVERAGE	4.974	

NOTES

$$1. \sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

SECTION LOCATION

UPPER BEARING OUTER REGION
STRESS AT T = 720 SEC.
TABLE NO. 9.5 (C)

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BASIC TECHNOLOGY, INC.

ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
96	-153	-22.903	2.012	-18.244
97	3.946	-2.436	2.570	3.754
98	8.095	8.759	.142	17.774
99	6.359	13.077	-.994	20.963
100	3.569	5.620	-2.040	11.205
101	-.849	-17.104	-1.250	-12.136

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	σ_1 (KSI)	σ_2 (KSI)
AVERAGE SECTION	3.795	-2.078	-.142	4.141	3.798	-2.081
LINEARIZED FACE A	4.372	-3.787	2.347	3.279	5.000	-4.415
FACE B	3.218	-.369	-2.632	5.003	4.609	-1.760

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	9.415	38.978
FACE B	6.763	32.242
SECTION AVERAGE	6.222	

NOTES

$$1. \sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

UPPER BEARING OUTER REGION
STRESSES AT T = 120 SEC.
TABLE NO. 9.6 (A)

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BASIC TECHNOLOGY, INC.

ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
96	-0.078	-12.819	1.512	-9.677
97	1.883	-5.191	2.159	-0.801
98	4.587	-1.487	0.488	4.99
99	3.924	0.395	-0.01	6.413
100	2.102	-2.276	-1.072	2.678
101	-5.49	-10.159	-1.052	-6.316

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	σ_1 (KSI)	σ_2 (KSI)
AVERAGE SECTION	2.188	-5.078	.187	-2.55	2.193	-5.083
LINEARIZED FACE A	2.366	-5.993	2.081	-1.300	2.855	-6.482
FACE B	2.010	-4.163	-1.706	.790	2.450	-4.603

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	9.337	20.931
FACE B	7.053	15.951
SECTION AVERAGE	7.276	

NOTES

$$1. \sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

UPPER BEARING OUTER REGION
STRESSES AT T = 380 SEC.
TABLE NO. 9.6 (B)

BASIC TECHNOLOGY, INC.

ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
96	-1.157	19.656	-1.572	14.938
97	-3.718	-1.810	-1.755	-6.953
98	-6.74	-12.525	.055	-19.258
99	-5.122	-16.007	1.379	-21.699
100	-3.051	-8.970	2.266	-12.834
101	-1.718	14.534	1.194	10.115

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_1 (KSI)	σ_2 (KSI)
AVERAGE SECTION	-3.210	-1.230	0.455	-6.262	-1.130
LINEARIZED FACE A	-3.786	-1.086	-1.690	-5.793	-4.440
FACE B	-2.634	-2.373	2.599	-6.731	-5.106

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	6.363	35.239
FACE B	6.831	30.639
SECTION AVERAGE	5.132	

NOTES

$$1. \sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

UPPER BEARING OUTER REGION
STRESSES AT T = 720 SEC.
TABLE NO. 9.6 (C)

BASIC TECHNOLOGY, INC.

ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
168	-19.566	-12.508	-7.896	-18.515
169	5.334	-7.282	-4.462	8.162
170	11.185	-1.579	-0.042	18.774

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	σ_1 (KSI)	σ_2 (KSI)
AVERAGE SECTION	-1.016	-7.123	-2.80	2.807	-0.73	-8.212
LINEARIZED: FACE A	-21.517	-14.409	-8.036	-22.052	-26.75	-9.176
FACE B	19.485	1.163	2.436	27.666	19.787	-1.139

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	17.574	42.794
FACE B	27.805	22.639
SECTION AVERAGE	11.019	

NOTES

$$1. \sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

SECTION LOCATION

UPPER BEARING OUTER REGION
STRESS AT T=120 SEC
TABLE NO. 9.7 (A)

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BASIC TECHNOLOGY, INC.

ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
168	-38.397	-19.626	-15.426	-21.224
169	4.917	-10.869	-5.156	14.000
170	24.233	-2.429	-2.197	35.488

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	σ_1 (KSI)	σ_2 (KSI)
AVERAGE SECTION	-3.08	-10.975	-7.593	9.421	1.530	-15.585
LINEARIZED: FACE A	-44.833	-22.440	-16.412	-28.387	-13.769	-53.504
FACE B	38.673	.490	1.226	47.229	38.712	.451

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	39.735	76.127
FACE B	46.778	46.297
SECTION AVERAGE	25.006	

NOTES

1.
$$\sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

SECTION LOCATION

UPPER BEARING OUTER REGION
STRESS AT T = 380 SEC.
 TABLE NO. 9.7 (B)

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BASIC TECHNOLOGY, INC.

ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
168	-4.738	1.29	-1.460	7.915
169	-2.920	1.079	-3.315	-4.46
170	4.052	0.194	-1.768	2.753

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	σ_1 (KSI)	σ_2 (KSI)
AVERAGE SECTION	-1.202	0.854	-2.181	3.705	2.237	-2.585
LINEARIZED: FACE A	-7.062	1.585	-1.976	7.146	2.015	-7.492
FACE B	4.658	0.123	-2.386	0.264	5.682	-0.901

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	14.638	19.808
FACE B	6.583	9.993
SECTION AVERAGE	6.29	

NOTES

$$1. \quad \sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

SECTION LOCATION

UPPER BEARING OUTER REGION
STRESS AT T = 720 SEC.
 TABLE NO. 9.7 (C)

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BASIC TECHNOLOGY, INC.

INNER REGION		OUTER REGION	
NODE	DISPLACEMENT UX (INCH)	NODE	DISPLACEMENT (INCH)
395	.08863	425	.10084
394	.09226	424	.09904
393	.09535	423	.09819
392	.09805	422	.09773
391	.10118	421	.09723
390	.10318	420	.09705
384	.10391	414	.10063
383	.10302	413	.10104
382	.10144	412	.10117
381	.09927	411	.10093

UPPER BEARING
THERMAL TRANSIENT AT T=120SEC.
TABLE NO. 9.8

BASIC TECHNOLOGY, INC.

INNER REGION		OUTER REGION	
NODE	DISPLACEMENT UX (INCH)	NODE	DISPLACEMENT (INCH)
395	.09495	425	.11477
394	.09952	424	.11228
393	.10344	423	.11077
392	.10688	422	.10976
391	.11086	421	.10898
390	.11344	420	.10873
384	.11519	414	.11578
383	.11460	413	.11526
382	.11338	412	.11469
381	.11150	411	.11404

UPPER BEARING
THERMAL TRANSIENT AT T=380 SEC.
TABLE NO. 9.9

BASIC TECHNOLOGY, INC.

INNER REGION		OUTER REGION	
NODE	DISPLACEMENT UX (INCH)	NODE	DISPLACEMENT (INCH)
395	.09929	425	.09503
394	.09711	424	.09547
393	.09519	423	.09542
392	.09350	422	.09523
391	.09163	421	.09526
390	.09054	420	.09526
384	.09096	414	.09550
383	.09164	413	.09450
382	.09294	412	.09386
381	.09492	411	.09362

UPPER BEARING
THERMAL TRANSIENT AT T=720SEC.
TABLE NO. 9.10

Page 85-C

BASIC TECHNOLOGY, INC.

BY _____ DATE _____ SHEET NO. _____ OF _____
CHKD. _____ DATE _____ PJ. _____

BY _____ DATE _____ SHEET NO. _____ OF _____
CHKD. _____ DATE _____ PJ. _____

UPPER BEARING

VARIATION OF GAP BETWEEN OUTER & INNER REGION

FIGURE 9.2

BASIC TECHNOLOGY, INC.

BY _____ DATE _____ SHEET NO. _____ OF _____
CHKD. _____ DATE _____ PJ. _____

TRANSIENT NODAL TEMPERATURES
TABLE 9.11

ITERATION NUMBER	NODE NUMBER	NODAL TEMPERATURE	AVERAGE SECTION TEMPERATURE
1	446	945	881
	476	875	
	506	830	
1	506	830	849
	536	815	
	566	851	
	596	930	
2	446	1003	976
	476	973	
	506	956	
2	506	956	963
	536	950	
	566	964	
	596	996	
3	446	744	807
	476	815	
	506	856	
3	506	856	838
	536	869	
	566	836	
	596	760	

10.0 Code Calculations

This section contains all calculations and comparisons necessary to justify feasibility of the bearing design with regard to Code and Specification requirements.

Code Case 1592 sets allowable values as the limits for primary membrane and bending stresses. Allowable limits are also set on total accumulated ratchetting strain and creep-fatigue damage. In addition, deformation limits must be established and met to ensure proper component functioning.

Table 10.1 summarizes the maximum stresses from the data reduction performed in Section 9. The maximum stresses for both the inner and outer regions are given for maximum primary (conservative for $P_L + P_b$), thermal membrane, linearized thermal (Q) and peak thermal (Q + F) stress intensities.

10.1 Primary Stress Evaluation

The Specification defines only Design Conditions and Normal plus Upset Operating Conditions. Additional conditions, with the exception of three Emergency transients, have not been specified at this time.

The Code criteria and allowable stresses for Design Conditions are given below.

$$P_m \leq S_o$$

$$P_L + P_b \leq 1.5 S_o$$

where $S_o = 15.7$ ksi for SS316 at the Design Temperature of 850°F

The Code criteria and allowable stresses for Normal plus Upset Conditions are:

$$P_m \leq S_{mt}$$

$$P_L + P_b = \text{lesser of } \begin{cases} K_t S_t \\ 1.5 S_m \end{cases}$$

where $S_m = 15.9$ ksi at 770°F

$S_t = 20.8$ ksi at 800°F for 3×10^5 hours

$S_{mt} = 15.9$ ksi

and $1.0 \leq K_t \leq 1.25$

Since Design Conditions consist only of internal pressure loading, and since the bearings are not on the pressure boundary, the criteria for Design Conditions are automatically met.

Primary stresses during Normal plus Upset Operating Conditions are calculated in Section 9 and summarized in table 10.1. The primary loading is considered to be:

bearing pressure + 200 psig

tank pressure = 30 psig

temperature = 770°F

rotational speed = 600 RPM

and shaft torque = 158,000 ft. lbs.

Since stresses for these conditions are generally low, only the largest of the resultant stresses is considered here. Thus, conservatively letting

$$P_m = P_L + P_b:$$

$$P_m = 5.6 \text{ ksi} < S_{mt} = 15.9 \text{ ksi}$$

$$\text{and } P_L + P_b = 5.6 \text{ ksi} < 1.5 S_m = 23.8 \text{ ksi}$$

Therefore, primary stress allowables are easily met.

10.2 Ratchetting Evaluation

The deformation and strain limits of paragraph T-1300 of Code Case 1592 are intended to limit the accumulated ratchetting strain to ensure structural integrity. The elastic method used in the Code is the O'Donnell-Porowski

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	INNER REGION	OUTER REGION
PEAK STRESS INTENSITY	54	76
LOCATION (ELEMENT)	34	168
TIME (SECONDS)	720	380
LINEARIZED STRESS INTENSITY	45	52.9
LOCATION (ELEMENT)	141	170
TIME (SECONDS)	120	380
THERMAL MEMBRANE STRESS	28	25
LOCATION (ELEMENT)	136 - 144	168 - 170
TIME (SECONDS)	380	380
PRESSURE & ROTATION STRESS	1.6	2.2
TORSIONAL STRESS + WEIGHT	3.6	0.0
PRIMARY STRESS INTENSITY	5.2	2.2

NOTE: All stresses are in ksi.

The linearized stress intensity at inner region is obtained by multiplying the calculations using the equations in Section 9.2 by the factor 1.112 for three elements across the thickness.

UPPER BEARING

STRESS SUMMARY

TABLE 10.1

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TRANSIENT	ΔT	NUMBER OF CYCLES (n_i)	$Q+F$	STRAIN FROM: P_L+P_b+Q+F	N_d	$\frac{n_i}{N_d}$
IP-3UT	285	15	76 KSI	3.48×10^{-3}	35	0.429
IP-1UT	70	480	18.7	0.93×10^{-3}	$> 10^6$	0.001
IP-5UT	80	10	21.3	1.04×10^{-3}	$> 10^6$	—
IP-6UT	90	5	24.0	1.16×10^{-3}	$> 10^6$	—
IP-7UT	125	14	33.3	1.58×10^{-3}	$> 10^6$	—
IP-8UT	150	63	40.0	1.88×10^{-3}	1400	0.045
IP-1ET	250	8	66.7	3.06×10^{-3}	43	0.186
IP-2ET	155	8	41.3	1.93×10^{-3}	5×10^5	—
IP-5ET	155	8	41.3	1.93×10^{-3}	5×10^5	—

NOTES: $P_L + P_b = 2.2$ KSI

$$\Sigma = 0.661$$

$$E = 22.5 \times 10^6 \text{ PSI}$$

N_d FOR IP-3UT AND IP-1ET IS

CALCULATED BY 1000-1200°F CURVES;

IP-8UT BY 950°F CURVE; AND OTHERS

BY 800°F CURVES

THESE RESULTS ARE CONSERVATIVE BY COMPARISON
WITH THE METHODS USED IN CODE CASE
1592, SECTION T-1413.

FATIGUE DAMAGE - UPPER BEARING

Page 90

TABLE 10.2

method which determines total ratchetting strain based on a calculated effective creep stress. The limiting resultant values are 1% for the parent metal and 0.5% for the weld metal.

Primary and secondary stress ranges, from table 10.1 are 5.2 ksi and 40.0 ksi for the inner region and 2.2 ksi and 47.0 ksi respectively for the outer region.

Yield stress, S_y , is taken as the average values of S_y at 720°F and 1015°F. Thus:

$$S_y = 0.5 (18.0 + 16.8)$$

$$S_y = 17.4 \text{ ksi}$$

Using Code notation, the primary stress parameter, x , is :

$$x_{\text{inner region}} = \frac{5.2}{17.4} = 0.299$$

$$x_{\text{outer region}} = \frac{2.2}{17.4} = 0.126$$

Likewise, the secondary stress parameter, y , is:

$$y_{\text{inner region}} = \frac{45.0}{17.4} = 2.59$$

$$y_{\text{outer region}} = \frac{52.9}{17.4} = 3.04$$

Both locations are in the "P", plastic cycling, regime. Thus, the effective creep stress parameter, Z , is:

$$\text{inner region: } Z_i = 0.774$$

$$\text{outer region: } Z_o = 0.383$$

Thus:

$$\sigma_{c,i} = 13.5 \text{ ksi}$$

$$\sigma_{c,o} = 6.66 \text{ ksi}$$

The total service life is divided into 100 hours at 1050°F (conservative for both time and temperature) and 2.6×10^5 hours at 800°F (also conservative).

The effective stresses for entering the isochronous curves are: $1.25 (13.5) = 16.9$ ksi and $1.25 (6.66) = 8.33$ ksi. The resultant strains are given below.

	inner region	outer region
100 hours at 1050°F	0.21	0.10
3×10^5 hours at 800°F	0.08	0.04
Total strain (%)	0.29	0.14

These strains, conservatively calculated, are well within the allowable strains of 1% for parent metal and 0.5% for weld metal.

10.3 Creep - Fatigue Evaluation

An analysis must be performed to ensure an acceptable level of accumulated creep-fatigue damage. Limits for elastic analysis are shown in figure T-1420-2 of Code Case 1592.

The creep damage contribution to the total damage is $\sum \frac{t}{T_d}$. Stresses from sustained operating conditions are low, less than 6 ksi. Sustained normal operating temperature are also low, 770°F. The isochronous curve for SS316 at 800°F shows no difference between the hot tensile curve and the 3×10^5 hour curve. Thus creep damage is neglected for all subsequent calculations.

The fatigue damage contribution to total damage is $\sum \frac{n_i}{N_d}$. Peak thermal stresses have been calculated for the IP-3UT transient. The peak stress intensity is conservatively taken as the sum of the maximum primary stress intensity and the peak thermal stress. Thus the peak stress range is 59.2 ksi for the inner region and 78.2 ksi for the outer region. The strain ranges are taken as:

$$\epsilon_R = \frac{\sigma_R}{E}$$

since the finite element results include all stress concentration factors. The modulus of elasticity is conservatively taken as $E = 22.5 \times 10^6$ psi at 1000°F . Thus $\epsilon_R = 2.63 \times 10^{-3}$ for the inner region and $\epsilon_R = 3.48 \times 10^{-3}$ for the outer region. The elastic fatigue curves for $1000^\circ\text{F} - 1200^\circ\text{F}$ are used to calculate N_d since the maximum temperature during the cycle is 1015°F .

The number of cycles to failure, N_d , for the inner region is $N_d = 62$ and for the outer region is $N_d = 35$ cycles. The maximum fatigue damage occurs in the outer region. In order to calculate fatigue damage for all transients, the peak thermal stress is modified by the ratio of ΔT for that transient to the $\Delta T (=320^\circ\text{F})$ for the IP-3UT transient. The sum of the individual damages for the outer region is calculated in table 10.2. It is:

$$\sum \frac{n_i}{N_d} = 0.661$$

Since creep damage is 0.0, the total accumulated damage is 0.661. This is less than the allowable damage of 1.0.

10.4 Deformation Limits for Functional Operability

In addition to ratchetting requirements for structural integrity, T - 1300 of Code Case 1592, additional deformation limits must be established and satisfied to ensure proper component functioning, T - 1200.

These limits will probably be more stringent than the ratchetting requirements for the bearing regions which essentially allow an accumulated 1% non-transient deformation. This limit is not acceptable when applied to radial deformation, circumferential runout (out-of-roundness), centerline offset, etc.

The present analysis considers only the bearing clearance during transient conditions, see Section 9.4. Additional criteria must be established and met to ensure operability for the entire design life.

11.0 Conclusions

This analysis shows that the design of the upper bearing is feasible for the conditions given in the Design Specification. Any potential problem areas which exist appear to be minor in nature and seem to be relatively easily solvable by either a minor redesign or more detailed analysis.

The primary stresses meet the Code requirements by wide design margins since the membrane thermal stress is considered to be secondary. The region is conservatively designed for the primary loads consisting of pressure, shaft rotational stresses and shaft transmitted torque.

The Code limits on stresses for structural integrity, i.e. ratchetting limits, are met on an elastic basis for both parent metal and weld regions. Maximum primary stresses of $P_L + P_b = 5.2$ ksi coupled with a maximum linearized secondary stress range of $Q_R = 53$ ksi. Since most of the pump operation is at 770°F , the accumulated strain is only 0.29%.

Creep damage is negligible since the operating temperature is low. The only time spent above 850°F occurs during thermal transient conditions. Since this time is less than 100 hours over the entire operating life the creep damage may be neglected.

Allowable fatigue damage is 1.0 due to the insignificant creep damage. The fatigue damage contribution from 15 cycles of the IP-3UT transient is 0.429. The fatigue damage from all other transients, calculated by the ratio of maximum transient temperature difference to that of the IP-3UT times the peak stress of the Ip-3UT, is only 0.232, giving a total accumulated fatigue damage of 0.661.

Positive clearance at the bearing surfaces are shown to be maintained

during the IP-3UT.

Thus, the additional Code requirement of functional operability is met for the bearing clearances for this transient. The minimum clearance for the IP-3UT transient is 0.020 inches from an initial clearance of 0.030 inches.

The largest change in bearing clearance during the IP-3UT causes an increase in clearance. Thus care must be taken when calculating clearance for a negative transient, in particular the IP-1ET which has a transient of -250°F. The gapping effect, see Section 9.5, is most crucial during a negative transient. An estimate of the clearance during the IP-1ET is

$\delta = 0.030 - \left(\frac{250}{285}\right) (0.020) - 0.004 = 0.008$ inches. Since the accuracy of these hand calculations is acceptable only for order of magnitude calculations, they are not intended to show acceptability for this transient. Further analysis is required to prove that a positive clearance is maintained. This, however, does not affect the conclusion that the design is feasible.

These results have been obtained by considering thermal membrane stress as a secondary stress. The maximum thermal membrane stress is 25 ksi on the outer structure and 28 ksi on the inner structure. Some inelastic strain will occur since this is higher than yield stress.

Further consideration must be given to the classification of membrane thermal stress to ensure structural and functional integrity in accordance with the intent of the Code. This subject is discussed in more detail in Section 12.0

12.0 Discussion and Suggestions

Code Case 1592 criteria are inherently easier to pass for this pump than for the Clinch River Pump since Design and Operating Temperatures are lower and thermal transients have lower temperature ranges while primary stresses remain about the same. The Code comparisons which are summarized in Section 11.0 show a high probability that the bearing may be shown acceptable by elastic analysis.

The results therefore show the design to be feasible for the required conditions. Although this analysis does not constitute a full Code report it shows that with minor design modifications and with further analysis all Code requirements are expected to be met.

Further analysis must be performed to demonstrate the effects of gapping on these results, to determine whether the membrane thermal stress affects functional operability and/or ratchetting and to analyze detailed areas not covered by this analysis such as loss of preload in the bolts.

A finite element model which includes non-linear gap elements is suggested as the most appropriate method of analyzing the effects of gaps in detail. This would include a mesh of only the local region, with the proper boundary conditions obtained from hand calculations or from a larger finite element model.

A simplified inelastic finite element model is suggested to analyze the effects of membrane thermal stress. The inelastic results would then be used to calculate accumulated ratchetting strain and to determine whether bearing clearance are affected. For example, if shakedown to elastic cycling can be shown, and if the bearing surface clearances are not changed then these may be used to help justify the high thermal membrane stress.

13.0 References

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2. ASME Boiler and Pressure Vessel Code, Section III - Division 1, Rules for Construction of Nuclear Power Plant Components, 1974 Edition with the summer, 1975 addenda; ASME B & PV Code Case 1592, Class 1 Components in Elevated Temperature Service, Section III.
3. General Electric Breeder Reactor Department Document No. 22A3994, Rev. 2, Large Scale Breeder Reactor Plant Prototype Mechanical Pump Conceptual Design Study; and GE Document No. NE56490, dated Feb. 5, 1976.
4. BJ memo from Y. Usui to D. Plapp, 761-G-0002 Transients of Fatigue Analysis, dated May 14, 1976.
5. "ANSYS Engineering Analysis System Users Manual", Swanson Analysis Systems, Inc. (1975).
6. BTI unpublished study, Verification and Optimization of ANSYS Elements For Transient Thermal Stress Problems, completed March 16, 1976.
7. BJ memo from D. Gibson to K. Huber, The First Meeting of the Large Scale Breeder Reactor Upper Bearing Analysis, dated May 21, 1976.
8. J.W. Rizika, "Thermal Lags in Flowing Systems Containing Heat Capacitors", Trans. ASME Vol. 76, 1954 pgs. 411-420
9. G.E. Specification No. 22A3444, Rev. 2 as modified by G.E. for the large scale sodium pump, confirmed by BJ memo from D. Gibson to K. Huber, the Second Meeting of the Large Scale Breeder Reactor Upper Bearing Analysis, dated May 27, 1976.

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10. "Liquid Metal Heat Transfer", Sodium & Nak Supplement to the Liquid Metals Handbook, Vol. 2, Chapter 1, January, 1969.
11. Same as above.
12. See, for example, "Marks' Standard Handbook for Mechanical Engineers", Seventh Edition, McGraw-Hill (1958).
13. The representative values of the film coefficients are chosen on the basis of previous work BTI has done on the liquid sodium pumps.
14. "Nuclear Systems Materials Handbook, Vol. 1: Design Data", Hanford Engineering Development Laboratory, Richland, Washington.
15. S. P. Timoshenks & S. Woinowsky-Krieger, "Theory of Plates and Shells", Second Edition, McGraw-Hill (1959)
16. BJ memo from D. Gibson to BTI dated May 28, 1976, confirming the horsepower and rotation speed of the pump shaft at normal operation.
17. R. J. Roark, "Formulas for Stress and Strain", McGraw-Hill (1975).

14.1 Appendix I - Heat loss of fluid through a cylinder

Consider a fluid whose temperature is initially T_1 flowing with a velocity V through the axial direction of a cylinder of radius d and wall thickness l . The cylinder is immersed in a bulk fluid of temperature T_0 . The flowing fluid convects heat to the inside wall of the cylinder. Heat is conducted through the thickness of the cylinder to the outside wall, to which the bulk fluid convects. The temperature $T(x)$ of the flowing fluid along the axis of cylinder is determined by the following differential equation:

$$\frac{d^2T}{dx^2} - \frac{\rho cv}{k} \frac{dT}{dx} - \frac{2HT}{kd} = - \frac{2HT_0}{kd}$$

where ρ , c , k are the density, heat capacity and conductivity of the transient fluid and H is given by

$$H = \frac{h_i h_o k_w}{k_w h_o + l h_i h_o + k_w h_i}$$

where k_w is the conductivity of the cylinder wall, h_i and h_o are the film coefficients on the inside and outside walls of the cylinder, respectively. The solution of the differential equation is readily obtained and is given by

$$T = (T_1 - T_0) e^{\frac{1}{2} \left\{ \frac{\rho cv}{k} - \left[\left(\frac{\rho cv}{k} \right)^2 + \frac{8H}{kd} \right]^{1/2} \right\} x} + T_0$$

where x is the distance along the axis of the cylinder, with T satisfying the initial boundary condition of $T = T_1$ at $x=0$.

14.2 Appendix II - Adiabatic mixing with piecewise linear transient fluid

Consider a large insulated volume V of fluid initially at temperature

T_0 . Transient fluid with piecewise linear temperature vs. time $T_1(t)$, enters the volume with a flow rate of $\dot{q}$. The temperature of the bulk fluid as a function of time satisfies the following differential equation

$$\dot{T} = \gamma (T_1 - T)$$

where $\gamma = \frac{\dot{q}}{V}$.

Assume $T_1(t) = a + bt$ for $t_1 < t < t_2$. The solution to the differential equation is

$$T(t_2) = \frac{1}{e^{\gamma t_2}} \left\{ T(t_1) e^{\gamma t_1} + \left(a - \frac{b}{\gamma}\right) (e^{\gamma t_2} - e^{\gamma t_1}) + b \left(t_2 e^{\gamma t_2} - t_1 e^{\gamma t_1}\right) \right\}$$

The constants a , b are determined by the initial and final temperatures of the piecewise linear transient. Assuming $T = T_1$ at $t = t_1$ and $T = T_2$ at $t = t_2$, we find

$$a = T_1 - (T_1 - T_2) \left(\frac{t_1}{t_2 - t_1} \right)$$

$$b = \frac{T_1 - T_2}{t_1 - t_2}$$

The boundary conditions are $T = T_1 = T_0$ at $t = 0$. The calculated temperature $T(t_2)$ for the first leg of the transient becomes the initial temperature for the second leg of the transient, and so on. Therefore, in general,

$$\text{with } a_i = T_i - (T_i - T_{i+1}) \left(\frac{t_i}{t_{i+1} - t_i} \right) \text{ and } b = \frac{T_i - T_{i+1}}{t_i - t_{i+1}}$$

A FORTRAN computer program is written to solve the above set of equations.

The input quantities to the program are γ , T_0 and the sets of temperature & time points $\{t_i, t_{i+1}, T_i, T_{i+1}\}$ to describe the piecewise linear transient curve. A listing of the FORTRAN is included herewith.

```

PROGRAM MIX(INPUT,OUTPUT,TAPE5=INPUT,TAPE6=OUTPUT)
READ(5,10) GAMMA,T1
WRITE(6,20) GAMMA
WRITE(6,30)
5 READ(5,10) TI,TF,TEMPI,TEMPF
IF(EOF(5)) 999,6
6 DT=TF-TI
DTEMP=TEMPF-TEMPI
A=TEMPI-DTEMP*TI/DT
R=DTEMP/DT
E1=EXP(GAMMA*TI)
E2=EXP(GAMMA*TF)
AB=A-B/GAMMA
F1=T1*E1
F2=AB*(E2-E1)
F3=B*(TF*E2-TI*E1)
T2=(F1+F2+F3)/E2
WRITE(6,20) TI,TF,TEMPI,TEMPF,A,B,T1,T2
T1=T2
GO TO 5
999 STOP
10 FORMAT(4F10.2)
20 FORMAT(8F15.4)
30 FORMAT(1H1,9X,2HTI,13X,2HTF,11X,5HTEMPI,11X,5HTEMPF,
112X,1HA,13X,1HB,13X,2HT1,13X,2HT2/)
END

```

BASIC TECHNOLOGY, INC.

LARGE SCALE SODIUM PUMP MIDDLE BEARING ANALYSIS

FOR

**BYRON JACKSON PUMP DIVISION
BORG WARNER CORPORATION
2300 E. VERNON, VERNON, CA. 90058**

BY

**BASIC TECHNOLOGY, INCORPORATED
806 MANHATTAN BEACH BOULEVARD
MANHATTAN BEACH, CALIFORNIA 90266**

JUNE 25, 1976

BTI 76-410 -002

BASIC TECHNOLOGY, INC.

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1.0 Introduction

This report presents an analysis of a preliminary design of a journal bearing for a proposed large scale sodium pump for a new prototype liquid metal fast breeder reactor (LMFBR). The design incorporates improved features based on existing and demonstrated sodium pump technology and draws considerable experience from previous and current work on FFTF and CRBR sodium pumps.

The journal bearing is located at the lower pump tank, and is used to provide the structural stability for the rotating pump shaft. The structural integrity of the bearing under primary loads and secondary thermal transient loads is essential for the operability of the pump.

The bearing configuration to be analyzed by Basic Technology, Inc. is shown in Byron Jackson's large scale sodium pump layout drawing No. IF-7937, Rev. 0. (1)

2.0 Scope of Work

The analysis entails a two-dimensional axisymmetric finite element elastic model of the middle bearings and bearing support under the most severe thermal transient condition. The temperatures, thermal stresses and distortions are calculated with the thermal and structural finite element models. Particular attention is focused on the bearing gap under the differential thermal expansion and contraction to determine if the relative deflections of the pump shaft and the bearing support might adversely affect the operability of the pump.

The primary loads on the bearing region include pressure, centrifugal force due to the rotating shaft, and torsional effects. The primary and secondary stresses are evaluated for any possible fatigue, creep and/or ratcheting damages, in accordance with the ASME Boiler and Pressure Vessel Code Section III and the high temperature Code Case 1592. ⁽²⁾

The design and operating conditions are given by General Electric's Specification ⁽³⁾ No. 22A3994, revision 2, which contains the requirements for the large scale breeder reactor plant prototype mechanical pump conceptual design.

The most severe thermal transient to be analyzed in detail is the IP3UT transient. However, all applicable transients are included in the consideration for fatigue evaluation. ⁽⁴⁾

3.0 Finite Element Models

The ANSYS computer program ⁽⁵⁾ is used in the thermal and structural evaluation of the middle journal bearing. A 2-D axisymmetric finite element model is constructed for the inner and outer regions of the bearing. The thermal and structural models are similar except for the pocket of sodium in the inner region, which is included only in the thermal model. The physical dimensions of the bearing region are shown in fig. 3.1. It has been found in a recent ANSYS verification study ⁽⁶⁾ that in most cases, sufficient accuracy in thermal stress analysis can be achieved using only three isoparametric elements across the wall thickness. The results of that study are applicable to the present models and will be used here.

3.1 Thermal Model

The finite element model for the thermal transient analysis is shown in fig. 3.2. It consists of 403 elements, with 198 isoparametric elements (STIF 55) in the inner region and 205 elements in the outer region. Except for the pocket of sodium in the inner region where the primary mechanism of heat transfer is assumed to be conduction, convection is assumed to be the dominant mechanism on all surfaces exposed to the liquid sodium. The bulk temperatures of the sodium in various regions surrounding the bearing are determined as a function of time in separate calculations to be discussed in Section 4.0. Since the bearing is immersed completely in liquid sodium during the operation cycles of the pump, no thermal radiation effects need be included in the analysis.

3.1.1 Convection Surfaces

The inner surface of the rotating shaft wall convects to the sodium

DIMENSIONS OF MIDDLE BEARING

FIGURE 3.1

46.0

41.1

36.2

31.3

26.4

21.5

16.6

11.7

6.8

1.9

-3.0

THERMAL MODE
MIDDLE BEARING
FIGURE 3.2

5.0 9.9 14.8 19.7 24.6 29.5 34.4 39.3 44.2

MIDDLE BEARING, GEOMETRY PLOT 6/9/76

THERMAL MODEL
MIDDLE BEARING
FIG 3.3

inside the shaft. The temperature of the sodium within the shaft is determined not only by the transient conditions around the bearing region, but also by those in the middle portion of the pump tank, in which the liquid sodium moves up through the discharge nozzle. All other surfaces in the middle bearing region are exposed to the transient sodium, except for the upper conical surface in the outer region of the bearing support, which convects to the bulk sodium in the lower pump tank. The bulk temperatures of the sodium in the pump tank region above the impellers will be calculated in Section 4.0 assuming 1) heat transfer of transient sodium to the bulk sodium through the walls of the bearing support and 2) mixing of transient sodium with the bulk sodium.

3.1.2 Stagnant Sodium

The pocket of sodium in the inner region of the bearing is assumed to be stagnant in the thermal model. The sodium in this region is modelled with isoparametric elements, in direct contact with the elements representing the stainless steel. This makes possible the heat conduction through the sodium in this region.

3.1.3 Transient flow and leakage paths

The low-pressure transient sodium from the suction nozzle enters the impeller region at the lower portion of the pump. After being whipped around by the impellers, the high-pressure sodium enters the middle feed tank. Some of the high-pressure sodium is also fed into the pocket in the outer region of the bearing support. The flow then leaks through the lower gap at the journal bearing and mixes with the low-pressure sodium. The flow paths of the sodium through the middle bearing region are shown by arrows in

fig. 3.1. A flow rate of 75 GPM at pony motor speed ⁽⁷⁾ is used for the sodium flow in the outer pocket region.

3.1.4 Insulated surfaces

The cut boundaries of the thermal model are assumed to be thermally insulated as shown in fig. 3.3.

3.2 Structural model

The finite element model for the structural analysis is shown in fig. 3.4. The structural axisymmetric isoparametric elements (STIF 42) are used in place of STIF 55 in the thermal model. The elements representing the stagnant sodium in the thermal model have no counterpart in the structural model. Instead, pressure boundary conditions are applied on the walls surrounding the stagnant sodium, as well as on all other wall surfaces.

3.2.1 Pressure boundary

The high-pressure sodium in the middle bearing regions exert a pressure of 200 PSI on the walls. These regions include the feed cylinder, the pocket area in the outer bearing region, and the bearing gap. A pressure of 30 PSI is applied to all other walls, which include the inside shaft region, the pocket area in the inner bearing region, and the outer walls. A linear pressure drop is assumed at the bearing gap where the high-pressure sodium flows out into the impeller region.

3.2.2 Weldments and Gaps

The locations of welds in the middle bearing are shown in fig. 3.6. It is seen that possible gapping exists at three areas of the bearing support as circled in the figure. Since the scope of the present analysis is

limited to linear elastic analysis (except for the temperature dependent material properties) this gap is not included in the finite element model. The effect of gapping is studied with hand calculations in a later section. It will be shown that the distance of separation under the most severe transient conditions is small, therefore the linear elastic model is adequate for the present analysis in estimating the transient effects. It should also be noted that the primary pressure load does not cause separation of the gap.

3.2.3 Coupled degrees of freedom

The structural model of the middle bearing is extended at the lower boundaries to a distance larger than $2\sqrt{Rt}$ beyond the area of structural discontinuity. These boundaries are confined to move in the radial direction only. The upper boundary of the inner shaft is also extended to a distance larger than $2\sqrt{Rt}$ beyond the discontinuity area. A plane strain boundary condition is applied at this section. Thus all nodes lying in the upper boundary plane are coupled as far as the axial displacement degree of freedom is concerned. On the outer region of the model, a plane is cut through the wall of the hemispherical feed tank so that all nodes on the boundary of the plane are confined to move in the tangential direction.

3.2.4 Modelling of the ribs in the outer bearing support

The middle bearing support contains a rib providing structural stiffness for the pocket in the outer region, as shown in Fig. 3.7 -3.8. Since a 2-D axisymmetric model is used in the finite element analysis, the effect of the rib is simulated with several tension-compression-only spar elements (STIF 1). The areas of the vertical and crossed spars are calculated so that both the stiffness in the axial direction and in the horizontal

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direction are represented. The total areas of the vertical and horizontal components of all the spar elements are equated separately to the averaged vertical and horizontal cross sections of the rib.

FIG 3.5
STRUCTURAL MODEL
MIDDLE BEARING

AREAS OF GAPPING :

FIG 3.6

SECTION

FIG 3.7
MIDDLE BEARING RIB

FIG 3.8

MIDDLE BEARING RIB

4.0 Boundary Conditions

4.1 Thermal hydraulic boundary conditions

Considerable attention is given to the determination of temperature boundary conditions in the present analysis, since the thermal stresses and deformations are largely determined by the temperature gradients through various sections of the bearing walls. Three sets of temperature-time curves are required to fully describe the thermal transient boundary condition. These are $T_{trans}(t)$, the input sodium transient temperature; $T_{in}(t)$, the bulk temperature inside the shaft; and $T_{out}(t)$, the bulk temperature in the pump tank. These temperatures are listed in Table 4.1 and are discussed below.

4.1.1 Thermal transients

The thermal transients considered in this analysis are listed in Table 4.2. The most severe transient is the IP3UT which rises 180°F in 120 seconds, and then drops 265°F in 250 seconds. This transient will be analyzed in detail using the finite element model described in the previous section. Secondary thermal stresses as well as thermal deformations for the IP3UT transient are calculated in the computer analysis. Other transients will be investigated with hand calculations and their effects included in the creep-fatigue evaluation.

4.1.2 Effect of transient mitigation

The effect of transient mitigation has been shown to be small in the upper bearing analysis. ⁽¹¹⁾ It is expected that the same conclusion applies to the middle bearing region. As a conservative measure, the full

IP3UT transient shown in fig. 4.1 is applied to the convective surfaces in the bearing region through which the high-pressure sodium flows.

4.1.3 Transient response inside pump shaft

The temperature of the stagnant sodium inside the pump shaft is calculated with ANSYS using a one-dimensional model with three isoparametric elements across the thickness of the wall, with ΔT applied at both the inner and outer surfaces of the wall. The initial temperature inside the bulk sodium is 770°. The change of the shaft temperature due to the transient sodium outside the shaft also affects the bulk sodium temperature inside the shaft. The transient response lags behind the applied transient as shown in fig. 4.1. Since the volume of sodium inside the shaft is relatively small, its temperature response vs time is significant, and the maximum temperature difference reaches 915°F at 620 sec. The calculated temperatures, T_{in} , as shown in Tables 4.1 are used as thermal boundary condition on the inner wall of the pump shaft in the full finite element model.

4.1.4. Transient response in the middle pump tank

The bulk temperatures of sodium in the upper pump tank are determined by convection and heat conduction through the walls, and by the mixing of the transient at suction pressure with the bulk sodium in the tank.

The effects of convection and conduction are calculated in a similar way as described in the previous section. A one-dimensional model with three isoparametric elements across the wall thickness is used in the ANSYS program to determine the transient response of the sodium in the middle tank. The transient $T_{trans}(t)$, in this case is applied at the inner surface of the wall. A large conductivity is assumed for the outside bulk of sodium. The film

coefficients are taken to be 0.002 BTU/sec/in²/°F for the transient convection surface, and 0.001 BTU/sec/in²/°F for the outer convection surface. The results of the transient response are shown along with the original transient in fig. 4.2. The rise of temperature is considerably slower in this case since the volume of the bulk sodium is much larger than that of the sodium inside the shaft.

4.1.5 Mixing contribution to the bulk temperature at middle pump tank

The effect of mixing is calculated with an adiabatic mixing model (see Appendix II). In this model, piecewise linear thermal transient flow enters an insulated volume V at a flow rate of $\dot{q}$. The transient fluid is assumed to mix thoroughly with the bulk fluid in V, whose response temperature as a function of time can be expressed in terms of the parameter $\gamma \equiv \frac{\dot{q}}{V}$. A FORTRAN computer program is written for the calculation of the mixing response temperature, with γ and the piecewise linear transient as input. For $\gamma = 0.00035$ and an input transient as shown in fig. 4.2, the temperature history of the upper tank bulk sodium due to the mixing effect is computed with this program, and the results are shown in fig. 4.2. It is assumed in this calculation that 4% of the transient flow entering into the impeller region closest to the middle bearing diverts upward and mixes with the bulk sodium.

The combined effects of mixing and convection/conduction through the walls are included in the temperature history curve, $T_{out}^{(t)}$, (Fig. 4.1) which is used as the input thermal boundary condition for the surfaces exposed to the pump tank sodium in the finite element model.

4.1.6 Convection film coefficients

Sodium convection coefficients for heat transfer between exposed structure and liquid sodium can be estimated from the following formulae:

Forced convection (Lyon-Martinelli Equation)⁽¹⁰⁾ -

$$h = \frac{k}{D} [7.0 + 0.025 (Pe)^{.8}] = \frac{.00634}{D} [1 + .02268 (V \cdot D)^{.8}]$$

Natural convection (McDonald - Connally Equation)⁽¹⁰⁾ -

$$h = .0785 \frac{k}{D} [Gr \cdot Pr]^{.32} = .0751 \Delta T^{.32} / D^{.64}$$

where D is the effective hydraulic diameter, V the velocity, k the conductivity; Pe, Gr, Pr are respectively Peclet's number, Grashof's number and Prandtl's number⁽¹²⁾; ΔT is the temperature difference between the structural surface and the bulk sodium.

We adopt the following values for the film coefficients in this analysis:

$$h = 0.01 \text{ BTU/sec} - \text{in}^2 - ^\circ\text{F} \quad \text{for the bearing gap region}$$

$$h = 0.001 \text{ BTU/sec} - \text{in}^2 - ^\circ\text{F} \quad \text{for outside surfaces exposed to the sodium in the pump tank}$$

$$\text{abd } h = 0.002 \text{ BTU/sec} - \text{in}^2 - ^\circ\text{F} \quad \text{for all other convective surfaces}$$

These values of film coefficients are obtained after representative calculations for typical regions are made using the aforementioned equation.⁽¹³⁾

4.2 Structural boundary conditions

The boundary conditions for the structural finite element model are discussed in Section 3.2. The effect of gapping is discussed in Section 9.5.

t (SEC)	T _{TRANSIENT}	T _{MIX}	$\Delta T_{MIX} \equiv T_{MIX} - 770$	T _{I-D}	$\Delta T_{I-D} \equiv T_{I-D} - 770$	$T_{OUT} \equiv 770 + \Delta T_{MIX} + \Delta T_{I-D}$	T _{AVR.}	T _{IN}
70	920	771.82	1.81	769.99	0	772	851	770
120	950	774.65	4.65	770.07	0.07	775	902	772
220	955	780.77	10.77	773.25	3.25	784	944	799
380	1005	791.63	21.63	781.79	11.79	803	990	850
470	1005	798.25	28.25	787.47	17.47	816	1001	877
620	810	803.79	33.79	796.76	26.76	831	867	909
720	740	802.80	32.80	799.45	29.45	832	779	904
820	720	800.29	30.29	799.69	29.69	830	736	878
1000	730	795.70	25.70	796.80	26.80	823	728	825

MIDDLE BEARING
TRANSIENT BOUNDARY CONDITIONS

TABLE 4.1

Page 20

TRANSIENT	ΔT_i	CYCLES	MAXIMUM METAL TEMPERATURE
IP-1UT	70	480	770
IP-3UT	285	15	1005
IP-5UT	80	10	770
IP-6UT	90	5	770
IP-7UT	125	14	770
IP-8UT	150	63	910
IP-1ET	250	8	1000
IP-2ET	155	8	770
IP-5ET	155	8	770

TABLE 4.2

TRANSIENTS FOR THELARGE SCALE SODIUM PUMP

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SHEET NO. _____ OF _____
P.J. _____

FIGURE 4.1 TRANSIENT RESPONSE V.S. TIME
MIDDLE BEARING

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BY _____ DATE _____ SHEET NO. _____ OF _____
CHKD. _____ DATE _____ PJ. _____

FIGURE 7.2

TEMPERATURE HISTORY OF BULK SODIUM
IN MIDDLE PUMP TANK

5.0 Material Properties

The thermophysical properties of SS316 are summarized in Table 5-1, and of sodium in Table 5-2. Values for these physical constants at various temperatures are taken from Nuclear Systems Materials Handbook, Vol. 1 & 2: Design Data (TID6666)⁽¹⁴⁾. A polynomial fit is obtained for each set of physical constants in the temperature range 400°F - 1100°F. The lowest degree of polynomial which meets the following criterion is used:

$$1 - D_0 < 10^{-4}$$

where D_0 is the coefficient of determination for a given degree of polynomial. The coefficients and the degree of the polynomials are found using a small computer program. They are summarized in Tables 5-3 and 5-4, for SS316 and sodium, respectively. Only static values of the sodium thermal conductivity are used.

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T °F	E PSI	ν	α /°F	ρ LB/IN ³	K BTU/SEC-IN-°F	C BTU/LB-°F
400	.2664E8	.2795	9.601E-6	.28495	2.2762E-4	.1255
500	.2607E8	.2838	9.710E-6	.28409	2.3773E-4	.1278
600	.2544E8	.2881	9.822E-6	.28322	2.4769E-4	.1296
700	.2477E8	.2923	9.942E-6	.28229	2.5787E-4	.1310
800	.2406E8	.2966	10.07E-6	.28142	2.6782E-4	.1323
900	.2331E8	.3009	10.19E-6	.28056	2.7801E-4	.1338
1000	.2253E8	.3051	10.30E-6	.27963	2.8796E-4	.1355
1100	.2172E8	.3094	10.39E-6	.27876	2.9815E-4	.1375

PHYSICAL PROPERTIES OF SS316 VS TEMPERATURE

TABLE 5-1

T °F	P LB/IN ³	K BTU/SEC-IN-°F	C BTU/LB-°F
400	.032604	1.0910E-3	.3195
500	.032135	1.0519E-3	.3146
600	.031661	1.0137E-3	.3105
700	.031181	.97639E-3	.3070
800	.030700	.94028E-3	.3042
900	.030220	.90509E-3	.3020
1000	.029740	.87083E-3	.3006
1100	.029253	.83750E-3	.2998

PHYSICAL PROPERTIES OF SODIUM VS TEMPERATURE

TABLE 5-2

	E PSI	ν	α /°F	ρ LB/IN ³	K BTU/SEC-IN-°F	C BTU/LB-°F
N	2	2	3	2	2	4
C_0	.286111E8	.262363	9.436	.288454	1.87502E-9	.102262
C_1	-4.12385E3	4.29415E-5	-2.7098E-4	-8.6885E-6	1.00213E-7	1.615E-4
C_2	-1.95235	-1.78253E-10	2.11104E-6	-1.187E-10	3.15717E-13	-1.4453E-7
C_3	—	—	-9.7765E-10	—	—	9.8976E-11
C_4	—	—	—	—	—	-2.2724E-14

COEFFICIENTS OF POLYNOMIAL FIT-SS316

TABLE 5-3

	ρ LB/IN ³	K BTU/SEC-IN-°F	C BTU/LB-°F
N	2	2	2
C ₀	3.44764E-2	1.25747E-3	.345607
C ₁	-4.63469E-6	-4.35789E-7	-7.88338E-5
C ₂	-1.03528E-10	4.91145E-11	3.38099E-8

COEFFICIENTS OF POLYNOMIAL OF FIT-SODIUM

TABLE 5-4

6.0 Thermal transient solutions

The temperature solutions at all nodes shown in the thermal finite element model of fig. 3.2 are obtained using the ANSYS computer program. A total of 9 load steps and 36 iterations are used in the transient analysis (fig. 6.1). The temperature histories for all convective boundaries as calculated in Section 4.0, as well as the input transient temperatures, are inputted at each time point during the iterations. Three axisymmetric isoparametric elements (STIF 55) are used through all wall thicknesses, as this mesh size has been previously verified to be adequate.⁽⁶⁾ The physical properties of SS316 and sodium are temperature dependent. The calculated nodal temperatures, blocked according to the element identification numbers, are saved on tapes for all 36 time points at the completion of the thermal analysis.

ANSYS LOAD INPUT FOR IP-3U TRANSIENT

FIGURE 6.1

7.0 Thermal stress calculations

7.1 Selection of time points

As the criteria for ratchetting, creep-fatigue and operability evaluations are based on the maximum linearized and peak stresses, it is necessary in the stress analysis to use only the temperatures calculated at the time points which would give these maximum values. Two effects may play major roles in the selection of these time points. The first is the variation of temperatures through the wall thickness where the stresses are expected to be the highest. The locations of the highly stressed areas generally correspond to relatively thick walls with differential transients applied on both sides of the wall. The second is the " $T_a T_b$ effect",⁽¹⁵⁾ which estimates the discontinuity stresses by the correlation of temperature differences at points across the boundaries of the discontinuity. Since the geometry of the bearing region is regular and the walls are of fairly uniform thickness, it is expected that the first effect is the more important in this case. Therefore, a plot of $(T_{\text{surface}} - T_{\text{mean}})$ vs. time through the wall at critical regions should give an indication of when the maximum stresses may occur. Figures 7.1 to 7.4 show that for most sections chosen, the peaks occur at $t = 120$ sec, 380 sec and 720 sec. A more detailed stress history calculation from a one-dimensional calculation with the corresponding temperature histories applied at either side of the wall also show that the stress levels are the highest at these time points. (Fig. 7.5 and 7.6) While the $T_a T_b$ effect may have slightly different peak time locations, its magnitudes are significantly lower than the $T_{\text{surface}} - T_{\text{mean}}$ effect. We therefore use the three time points chosen for all thermal stress calculations.

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MIDDLE BEARING, INNER REGION AT B

FIG 7.2

BY _____ DATE _____ SHEET NO. _____ OF _____
CHKD. _____ DATE _____ P.J. _____

BY _____ DATE _____ SHEET NO. _____ OF _____
CHKD. _____ DATE _____ PJ. _____

MIDDLE BEARING OUTER REGION AT A

FIG. 7.4

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CHKD. _____ DATE _____ PJ. _____

1-D STRESS HISTORY IN OUTER SHAFT WALL

FIGURE 7.6

7.2 Finite element results

Thermal stresses and deflections are calculated at the selected time points using the nodal temperatures stored on tape from the thermal transient analysis. The thermal elements representing the stagnant sodium in the inner region of the bearing are taken out in the stress analysis. Other thermal isoparametric elements are replaced by the equivalent structural isoparametric elements (STIF 42). Computer outputs of the finite element analysis include printed tables of nodal deflections and stress components at four nodes as well as at the centroid of each of the elements. In addition, post-processing plots of deflected shapes, hoop stress contours and stress intensity contours are obtained at each of the selected time points. These are shown in fig. 7.7 to 7.27. The contour plots identify locations of the highest and lowest stresses, as well as areas of high stress concentrations. Sections with high peak stresses and/or high membrane stresses are selected from these contour plots and are analyzed in detail in Section 9.

7.3 Hand Calculations

All other transients except the IP3UT are analyzed with hand calculations to obtain the maximum thermal stresses for the fatigue evaluation.

STEP= 1 ITERATION= 1

DEFLECTION AT $t = 120 \text{ SEC}$
FIGURE 7.7

MIDDLE BEARING INNER REGION THERMAL STRESS POSTPROCESSING PLOT

STEP 1 ITERATION= 1

HOOP STRESS AT $t = 120$ SEC

INCREMENT = 5000 PSI

FIGURE 7.8

MIDDLE BEARING INNER REGION THERMAL STRESS POSTPROCESSING PLOT

STEP- 1 ITERATION= 1

STRESS INTENSITY AT $t = 120 \text{ SEC}$
INCREMENT = 2500 PSI
FIGURE 7.9

MIDDLE BEARING INNER REGION THERMAL STRESS POSTPROCESSING PLOT

STEP 1 ITERATION= 1

STRESS INTENSITY AT $t = 120 \text{ SEC}$

INCREMENT = 2500 PSI

FIGURE 7.9

MIDDLE BEARING INNER REGION THERMAL STRESS POSTPROCESSING PLOT

STEP= 1 ITERATION= 1

DEFLECTION AT $t = 120 \text{ SEC.}$
FIGURE 7.10

MIDDLE BEARING OUTER REGION THERMAL STRESS POSTPROCESSING PLOT

STEP= 1 ITERATION= 1

HOOP STRESS AT $t = 120 \text{ SEC}$
INCREMENT = 4000 psi
FIGURE 7.11

STEP= 1 ITERATION= 1

STRESS INTENSITY AT $t = 120 \text{ SEC}$
INCREMENT = 2500 PSI
FIGURE 7.12

STEP= 1 ITERATION= 1

TEMPERATURE AT $t = 120 \text{ sec.}$
INCREMENT = 10°F
FIGURE 7.13

STEP: 2 ITERATION= 1

DEFLECTION AT $t = 380$ SEC

FIGURE 7.14

MIDDLE BEARING INNER REGION THERMAL STRESS POSTPROCESSING PLOT

STEP

2 ITERATION: 1

HOOP STRESS AT $t = 380$ SEC

INCREMENT = 4000 PSI

FIGURE 7.15

MIDDLE BEARING INNER REGION THERMAL STRESS POSTPROCESSING PLOT

STEP. 2 ITERATION. 1

STRESS INTENSITY AT $t = 380$ SEC.
INCREMENT = 2000 PSI
FIGURE 7.16

MIDDLE BEARING INNER REGION THERMAL STRESS POSTPROCESSING PLOT

STEP= 2 ITERATION= 1

DEFLECTION AT $t = 380$ SEC.

FIGURE 7.17

MIDDLE BEARING OUTER REGION THERMAL STRESS POSTPROCESSING PLOT

STEP= 2 ITERATION= 1

STEP= 2 ITERATION= 1

STRESS INTENSITY AT $t=380$ SEC
INCREMENT = 4000 PSI
FIGURE 7.19

STEP= 2 ITERATION= 1

TEMPERATURE AT $t = 380$ SEC.

INCREMENT = 15°F

FIGURE 7-20

STEP= 3 ITERATION= 1

DEFLECTION AT $t = 720$ SEC
FIGURE 7.21

MIDDLE BEARING INNER REGION THERMAL STRESS POSTPROCESSING PLOT

5111

INTERMITTENT

HOLD AT + 7.0 SE

INCREMENT = 400 PSI

FIGURE 7.2.

MIDDLE BEARING INNER REGION THERMAL STRESS TO THE DEFORMING FLUID

0744

INTERSECTION

720 SEC
INCIDENTAL
FIGURE 1

STEP= 3 ITERATION= 1

DEFLECTION AT $t = 720 \text{ SEC}$
FIGURE 1

MIDDLE BEARING OUTER REGION THERMAL STRESS POSTPROCESSING PLOT

STEP= 3 ITERATION= 1

HOOP STRESS AT $t = 720$ SEC.
INCREMENT = 4000 PSI
FIGURE 7.25

STEP= 3 ITERATION= 1

STEP- 3 ITERATION 1

8.0 Mechanical loads

The primary loads of the bearing region consist of pressure, centrifugal force, torsion and weight in the inner region; and of pressure only in the outer region. The centrifugal force and torsion loads are imposed on the inner bearing region as a result of the shaft rotation.

8.1 Pressure

The pressures applied at various boundaries of the bearing walls are indicated in fig. 3.5. The displacement boundary conditions of the finite element model are the same as those of the thermal stress model discussed in Section 3.2.3.

8.2 Centrifugal force

The rotating shaft imparts a centrifugal force to the inner region of the bearing, in addition to the pressure loads. This effect combined with the pressure load in the ANSYS run by specifying an angular velocity in the axial direction. The rotational speed of the shaft at normal operation ⁽¹⁶⁾ is $w = 600 \text{ RPM} = 20\pi \text{ rad/sec}$. The hoop stress and intensity contour plots for the middle bearing region are included in Fig. 8.1 to 8.4.

8.3 Torsion

The torsional stress in the shaft can be calculated with the following formula ⁽¹⁷⁾:

$$\text{Max } \sigma = \frac{2T r_1}{\pi (r_1^4 - r_0^4)} \quad \text{at the outer boundary}$$

where T is the torque; r_1 and r_0 are, respectively, the outer and inner radius of the shaft.

During normal operation, the pump operates ⁽¹⁶⁾ at 18000 HP and the

shaft rotates at a speed of 600 RPM, thus

$$T = 1.89 \times 10^4 \text{ in-lb.}$$

with $r_1 = 9.5 \text{ in.}$ and $r_0 = 7.5 \text{ in.}$, we find $\text{Max } T = 3.2 \text{ ksi.}$

8.4 Weight

The axial stress in the shaft due to its own weight is calculated with $\sigma = P/A = \rho l$ where $\rho = .28 \text{ lb/in}^3$ is the density of the shaft, A is the cross sectional area and $l \approx 800 \text{ in}$ represents the equivalent length of the shaft, including the impellers. The axial stress σ is thus 224 PSI.

Table 10.1 summarizes the maximum primary stresses in the inner and outer regions of the middle bearing and support.

STEP= 1 ITERATION= 1

HOOP STRESS
INCREMENT 150 PSI
FIGURE : 8.1

STEP= 1 ITERATION= 1

STRESS INTENSITY
INCREMENT 150 PSI
FIGURE 8.2

STEP= 1 ITERATION= 1

STEP= 1 ITERATION= 1

MIDDLE BEARING OUTER REGION - PRESSURE LOADING

9.0 Data Reduction

This section contains the calculations necessary to reduce the raw data into the desired form for comparison with allowables. All stress results are summarized for various cross-sections in tables 9.1 through 9.4.

9.1 Membrane Stresses

Membrane stresses through a wall section are obtained by averaging the σ_x , σ_y , τ_{xy} and σ_z components, calculating principal stresses from the averaged components and then determining stress intensities from the largest difference in principal stresses. They are summarized for each thermal stress and primary load run in tables 9.1 through 9.4.

9.2 Linearized Stress

Linearized stresses are calculated for the surfaces of each cross-section. First, linearized components are determined from the following equations.

$$\sigma_A = \bar{\sigma} - \frac{6}{t^2} \sum_j \sigma_j x_j d_j$$

$$\sigma_B = \bar{\sigma} + \frac{6}{t^2} \sum_j \sigma_j x_j d_j$$

where σ_a, σ_b = linearized components at surfaces A and B

$$\bar{\sigma} = \frac{1}{N} \sum_{j=1}^N \sigma_j$$

x_j = distance from centroid

d_j = width of j th element

t = width of cross-section

N = number of elements through section

Then, principal stresses are calculated at the surfaces from the linearized components σ_x , σ_y , τ_{xy} and σ_z . Finally, stress intensities are determined by the largest difference of the principal stresses. They are

summarized for each thermal stress and primary load run in tables 9.1 through 9.4.

9.3 Peak Stress

Surface stress intensities are calculated by ANSYS at each free surface, thus eliminating the need for extrapolation to the surfaces. They are used directly as peak stress intensities and are summarized at each surface in tables 9.1 through 9.4.

9.4 Bearing Clearances

Radial displacements at adjacent nodes on the inner and outer bearing regions are summarized at 120, 380 and 720 seconds in tables 9.5 through 9.7. The clearances between adjacent nodes are then calculated and plotted in figures 9.1 and 9.2. These figures assume an initial clearance of 0.030 inches and show the gap size between adjacent nodes of the inner and outer regions of the bearing. The maximum closure in the this region during the IP-3UT occurs between nodes 466 and 980 at time = 380 seconds. Thus, assuming an initial 0.030 inch radial bearing clearance, the minimum clearance during transient conditions is 0.014 inches.

In addition, the gapping effect, discussed in Section 9.5, may further reduce the clearance.

This analysis does not cover the severe negative transients such as the IP-1ET. The effect of this transient is discussed in the Conclusions.

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ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
97	-71.608	-12.356	-9.593	-31.095
100	10.345	-1.513	-3.555	2.774
103	33.712	7.191	-7.383	19.189

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	σ_1 (KSI)	σ_2 (KSI)
AVERAGE SECTION	4.150	-2.220	-6.845	-3.044	8.515	-6.585
LINEARIZED: FACE A	-39.397	-15.237	-8.322	-36.567	-41.986	-12.648
FACE B	47.696	10.799	-5.368	30.479	48.461	10.034

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	29.338	67.576
FACE B	38.427	50.820
SECTION AVERAGE	15.10	

NOTES

$$1. \sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

SECTION LOCATION

MIDDLE BEARING INNER REGION
STRESS AT T=120 SEC.
TABLE NO. 9.1A

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BASIC TECHNOLOGY, INC.

ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
97	-27.195	-11.291	-9.575	-28.329
100	7.751	-1.577	-3.387	-2.022
103	31.905	6.254	-6.554	13.590

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	U_1 (KSI)	U_2 (KSI)
AVERAGE SECTION	4.154	-2.205	-6.68	-5.597	8.373	-6.424
LINEARIZED: FACE A	-35.246	-13.901	-8.706	-33.533	-38.347	-10.80
FACE B	43.554	9.492	-4.651	22.359	44.178	8.868

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	27.547	57.505
FACE B	35.31	49.974
SECTION AVERAGE	14.797	

NOTES

$$1. \sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

SECTION LOCATION

MIDDLE BEARING INNER REGION
STRESS AT T = 380 SEC.
TABLE NO. 9.1B

BASIC TECHNOLOGY, INC.

ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
97	29.001	11.797	9.076	28.562
100	-9.552	1.721	3.373	-2.441
103	-30.984	-6.560	6.690	-17.230

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	σ_1 (KSI)	σ_2 (KSI)
AVERAGE SECTION	-3.878	2.054	6.232	2.964	6.126	-7.950
LINEARIZED: FACE A	36.112	14.025	7.968	33.422	38.046	11.451
FACE B	-43.868	-9.977	4.795	-27.559	-9.253	-44.532

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	27.235	62.773
FACE B	35.279	46.760
SECTION AVERAGE	14.076	

NOTES

$$1. \sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

SECTION LOCATION

MIDDLE BEARING INNER REGION
STRESS AT $T = 7.20$ SEC
TABLE NO. 9.1C

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BASIC TECHNOLOGY, INC.

ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
124	35.342	7.119	1.516	20.300
127	11.904	-0.884	3.462	4.801
130	-30.889	-10.573	4.260	-28.964

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	U_1 (KSI)	U_2 (KSI)
AVERAGE SECTION	5.452	-1.393	6.879	-1.255	9.713	-5.654
LINEARIZED: FACE A	49.606	10.509	5.717	31.652	-0.425	9.690
FACE B	-38.702	-13.294	8.042	-34.163	-10.763	-46.033

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	40.735	51.739
FACE B	30.07	66.483
SECTION AVERAGE	15.367	

NOTES

$$1. \sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

SECTION LOCATION

MIDDLE BEARING INNER REGION
STRESS AT T = 120 SEC.
TABLE NO. 9.2A

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BASIC TECHNOLOGY, INC.

ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
124	33.568	6.458	6.777	14.732
127	9.062	-8.25	4.149	-0.334
130	-26.555	-9.361	9.131	-26.389

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	U_1 (KSI)	U_2 (KSI)
AVERAGE SECTION	5.358	-1.243	6.686	-3.997	9.514	-5.399
LINEARIZED: FACE A	45.440	9.303	5.116	23.417	46.150	8.593
FACE B	-34.724	-11.789	8.255	-31.411	-9.127	-37.386

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	37.557	51.006
FACE B	28.259	56.098
SECTION AVERAGE	14.913	

NOTES

$$1. \sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

SECTION LOCATION

MIDDLE BEARING INNER REGION
STRESS AT T = 380 SEC.
TABLE NO. 9.2B

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BASIC TECHNOLOGY, INC.

ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
124	-32.495	-6.65	-6.824	-18.391
127	-11.092	0.711	-3.648	-4.347
130	28.381	9.695	-0.742	26.548

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	σ_T (KSI)	σ_z (KSI)
AVERAGE SECTION	-5.069	1.251	-6.405	1.27	5.233	-9.051
LINEARIZED: FACE A	-45.652	-9.647	-5.126	-28.689	-8.931	-46.368
FACE B	35.515	12.149	-7.683	31.229	37.815	9.849

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	37.437	47.59
FACE B	27.966	61.794
SECTION AVERAGE	14.284	

NOTES

$$1. \sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

SECTION LOCATION

MIDDLE BEARING INNER REGION
STRESS AT T = 720 SEC.
 TABLE NO. 9.2C

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BASIC TECHNOLOGY, INC.

ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
69	-4.749	-21.876	-2.974	-15.342
85	1.464	8.035	-1.455	15.717
86	5.656	8.422	1.182	20.817
87	6.193	9.555	1.966	20.405
88	4.164	7.873	2.276	14.669
89	1.342	-18.896	1.358	-15.436

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	σ (KSI)	σ_z (KSI)
AVERAGE SECTION	2.484	-3.61	.331	8.35	25.4	.311
LINEARIZED FACE A	-4.89	1.095	-9.412	10.486	2.842	-2.236
FACE B	5.457	-3.373	3.075	6.215	6.779	-1.695

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	12.772	45.434
FACE B	8.474	41.763
SECTION AVERAGE	8.039	

NOTE

$$1 \quad \sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

MIDDLE BEARING OUTER REGION
STRESSES AT T = 120 SEC
TABLE NO. 9.3A

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BASIC TECHNOLOGY, INC.

ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
69	-4589	-29362	-4189	-14998
85	2.663	7.649	-1.657	22.882
86	9.424	16.697	1.770	37.980
87	9891	13.965	2.534	32.628
88	6.551	5.440	2.099	15.336
89	1.953	-22077	-977	-15.871

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	σ_1 (KSI)	σ_2 (KSI)
AVERAGE SECTION	4.439	.121	.215	14.795	4.45	.110
LINEARIZED FACE A	.912	.525	-2.876	19.193	3.601	-2.164
FACE B	7.966	-.283	3.306	10.397	9.127	-1.444

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	21.357	54.524
FACE B	11.841	40.118
SECTION AVERAGE	14.685	

NOTE

$$1 \quad \sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

MIDDLE BEARING OUTER REGION

STRESSES AT $T = 380$ SEC

TABLE NO 9.3B

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BASIC TECHNOLOGY, INC.

ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
69	3.867	12.444	3.115	10.191
85	-1.117	-5.532	1.565	-8.521
86	-2.265	-3.737	-0.977	-9.459
87	-3.231	-5.956	-1.896	-11.422
88	-2.439	-6.826	-2.500	-10.773
89	-0.865	13.042	-1.569	10.083

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	σ_1 (KSI)	σ_2 (KSI)
AVERAGE SECTION	-0.921	-1.371	-0.304	-4.227	-0.236	-1.056
LINEARIZED FACE A	1.490	-1.371	2.703	-4.165	3.118	-2.999
FACE B	-3.332	0.630	-3.11	-3.893	2.336	-5.038

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	7.283	28.116
FACE B	7.374	31.227
SECTION AVERAGE	3.991	

NOTE

$$1. \sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

MIDDLE BEARING OUTER REGION
STRESSES AT T=720 SEC
TABLE NO. 9.3C

BASIC TECHNOLOGY, INC.

ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
157	.671	-7.09	.177	9.967
162	5.242	4.475	-.097	13.803
167	9.416	5.326	-.323	15.428
172	9.594	6.060	-.413	12.938
177	5.066	-2.602	-.109	-4.427
182	-.137	-25.108	-.205	-33.298

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	σ_1 (KSI)	σ_2 (KSI)
AVERAGE SECTION	5.52	-.479	-.170	4.476	5.525	-.484
LINEARIZED FACE A	5.855	-9.278	-.322	-14.208	5.862	-9.285
FACE B	5.185	8.320	-.018	23.160	8.320	5.185

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	20.07	47.986
FACE B	17.975	13.482
SECTION AVERAGE	6.009	

NOTES

$$1. \sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

MIDDLE BEARING OUTER REGION
STRESSES AT T = 120 SEC
TABLE NO. 9.4 A

BASIC TECHNOLOGY, INC.

ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
157	1.463	5.136	-.669	18.059
162	5.776	6.248	-.065	16.682
167	8.528	3.067	-.494	11.419
172	7.209	-.053	-.358	1.343
177	3.314	-5.454	.020	-15.503
182	-.173	-16.003	-.236	-34.098

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	σ_1 (KSI)	σ_2 (KSI)
AVERAGE SECTION	4.687	-.562	.017	.718	4.687	-.562
LINEARIZED FACE A	3.695	-10.991	-.227	-26.94	3.698	-10.994
FACE B	5.679	9.867	-.193	28.376	9.876	5.670

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	30.638	42.768
FACE B	22.706	17.653
SECTION AVERAGE	5.249	

NOTES

$$1. \sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

MIDDLE BEARING OUTER REGION
STRESSES AT T = 380 SEC
TABLE NO. 9.4B

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BASIC TECHNOLOGY, INC.

ELEMENT NUMBER	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)
157	.118	7.487	-.916	-1.603
162	-3.935	-3.361	.259	-11.103
167	-7.676	-7.073	.836	-16.795
172	-8.293	-8.067	.779	-17.276
177	-4.549	-.448	.174	-4.234
182	-.306	21.802	-.132	20.538

STRESSES	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	σ_1 (KSI)	σ_2 (KSI)
AVERAGE SECTION	-4.550	.226	.243	-7.025	.238	-4.562
LINEARIZED FACE A	-4.603	4.300	-.639	.585	4.346	-4.649
FACE B	-4.417	-3.850	-.153	-14.635	-3.811	-4.456

LOCATION	LINEAR STRESS INTENSITY (KSI)	PEAK STRESS INTENSITY (KSI)
FACE A	5.054	33.489
FACE B	10.824	15.912
SECTION AVERAGE	7.263	

NOTES

$$1. \sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

MIDDLE BEARING OUTER REGION
STRESSES AT T = 720 SEC
TABLE NO. 9.4C

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BASIC TECHNOLOGY, INC.

INNER REGION		OUTER REGION	
NODE	DISPLACEMENT UX (INCH)	NODE	DISPLACEMENT UX (INCH)
471	.097912	985	.09385
470	.099986	984	.09326
469	.101933	983	.09296
468	.103346	982	.09241
467	.104078	981	.09174
466	.104381	980	.09144
461	.103982	511	.09369
460	.103268	510	.09591
459	.101916	509	.09812
458	.09997	508	.09981
457	.09779	507	.10148

MIDDLE BEARING
THERMAL TRANSIENT AT T=120SEC
TABLE 9.5

BASIC TECHNOLOGY, INC.

INNER REGION		OUTER REGION	
NODE	DISPLACEMENT UX (INCH)	NODE	DISPLACEMENT UX (INCH)
471	-.10989	985	.10236
470	.11199	984	.10141
469	-.11372	983	.10081
468	.11487	982	.10025
467	.11540	981	.09977
466	.11557	980	.09961
461	.11533	511	.10436
460	.11480	510	.10668
459	.11367	509	.10897
458	.11190	508	.11104
457	.10970	507	.11308

MIDDLE BEARING
THERMAL TRANSIENT AT T=380SEC
TABLE 9.6

BASIC TECHNOLOGY, INC.

INNER REGION		OUTER REGION	
NODE	DISPLACEMENT UX (INCH)	NODE	DISPLACEMENT UX (INCH)
471	.09623	985	.09575
470	.09431	984	.09598
469	.09256	983	.09602
468	.09130	982	.09636
467	.09064	981	.09689
466	.09036	980	.09715
461	.09069	511	.09781
460	.09134	510	.09660
459	.09255	509	.09538
458	.09431	508	.09460
457	.09633	507	.09384

MIDDLE BEARING
THERMAL TRANSIENT AT T=720SEC
TABLE 9.7

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BY _____ DATE _____ SHEET NO. _____ OF _____
CHKD. _____ DATE _____ PJ. _____

BY-----DATE----- SHEET NO-----OF-----
CHKD.-----DATE----- PJ-----

9.5 Gapping Effects

Gapping effects in the middle bearing may occur at the bolted junction and at the bearing surface rings which are welded to the outer structure. These locations are shown in figure 9.3.

Temperatures in the bolted junction region are given in figures 7.13, 7.20 and 7.25 at the three chosen time points. For the first two time steps the inner material is at a higher temperature than the outer material. This tends to cause radial compression between the two vertical faces. The horizontal faces are originally preloaded in compression by the bolt. For the third time step chosen, temperatures on both sides of the vertical line of separation appear to be of the same magnitude. Thus the gap is expected to be relatively small for this case. The effect of the gap in this region is expected to result in slightly lower stresses since it essentially removes a constraint. Since this region is not the region of resultant highest membrane, linearized or peak stress (see Section 9) any errors induced by the gap are considered to be small.

Temperature distributions in the bearing surface ring regions are also given in figures 7.13, 7.20 and 7.25 at the three time points. Both the upper and lower rings are at a higher temperature than the surrounding metal at 120 and 380 seconds. This shows that contact is expected to be maintained through the first part of the transient since the rings expand radially more than the surrounding metal. Thus there will be compression on the vertical faces at these times. Since the lower surface of the upper ring and the upper surface of the lower ring are in contact with the flange surface, there will also be compression on the horizontal faces. Thus the model is expected to adequately represent the state of stress and displacement in these regions

at these times.

Mean temperatures in the rings at 720 seconds are slightly lower than that of the surrounding metal. Thus, separation is expected to occur at that time. The results of this analysis are not expected to be adversely affected for the following reasons. One reason is that even though separation will occur, the bearing clearance has opened at this time, see figures 9.1 and 9.2. If the mean temperature difference is conservatively estimated to be 50°F, then the relative closure will be on the order of: $\delta = \alpha r \Delta T = (10^{-5}) \cdot (10.0)(50) = 0.005$ inches. This is acceptable since the relative clearance is on the order of 0.030 inches. Secondly, stresses are not a maximum at 720 seconds, see the results of the tables 9.3 through 9.4. If the gap were included in the model, stresses in the bulk metal cross-sections would be lower due to the reduction in constraint. The possibility exists, however, of a stress concentration in the weld since its structural response may be strain-controlled by the thicker sections. Suggestions for further analysis or redesign are given in Section 12.0.

MIDDLE BEARING - GAPPING LOCATIONS

FIGURE 9.3

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10.0 Code Calculations

This section contains all calculations and comparisons necessary to justify feasibility of the bearing design with regard to Code and Specification requirements.

Code Case 1592 sets allowable values as the limits for primary membrane and bending stresses. Allowable limits are also set on total accumulated ratchetting strain and creep-fatigue damage. In addition, deformation limits must be established and met to ensure proper component functioning.

Table 10.1 summarizes the maximum stresses from the data reduction performed in Section 9. The maximum stresses for both the inner and outer regions are given for maximum primary (conservative for $P_L + P_b$), thermal membrane, linearized thermal (Q) and peak thermal (Q + F) stress intensities.

10.1 Primary Stress Evaluation

The Specification defines only Design Conditions and Normal plus Upset Operating Conditions. Additional conditions, with the exception of three Emergency transients, have not been specified at this time.

The Code criteria and allowable stresses for Design Conditions are given below.

$$P_m \leq S_o$$

$$P_L + P_b \leq 1.5 S_o$$

where $S_o = 15.7$ ksi for SS316 at the Design Temperature of 850°F

The Code criteria and allowable stresses for Normal plus Upset Conditions are:

$$P_m \leq S_{mt}$$

$$P_L + P_b = \text{lesser of } \begin{cases} K_t S_t \\ 1.5 S_m \end{cases}$$

where $S_m = 15.9$ ksi at 770°F

$S_t = 20.8$ ksi at 800°F for 3×10^5 hours

$S_{mt} = 15.9$ ksi

and $1.0 \leq K_t \leq 1.25$

Since Design Conditions consist only of internal pressure loading, and since the bearings are not on the pressure boundary, the criteria for Design Conditions are automatically met.

Primary stresses during Normal plus Upset Operating Conditions are calculated in Section 9 and summarized in table 10.1. The primary loading is considered to be:

bearing pressure = 200 psig

tank pressure = 30 psig

temperature = 770°F

rotational speed = 600 RPM

and shaft torque = 158,000 ft. lbs.

Since stresses for these conditions are generally low, only the largest of the resultant stresses is considered here. Thus, conservatively letting

$$P_m = P_L + P_b:$$

$$P_m = 4.2 \text{ ksi} < S_{mt} = 15.9 \text{ ksi}$$

$$\text{and } P_L + P_b = 4.2 \text{ ksi} < 1.5 S_m = 23.8 \text{ ksi}$$

Therefore, primary stress allowables are easily met.

10.2 Ratchetting Evaluation

The deformation and strain limits of paragraph T-1300 of Code Case 1592 are intended to limit the accumulated ratchetting strain to ensure structural integrity. The elastic method used in the Code is the O'Donnell-Porowski

method which determines total ratchetting strain based on a calculated effective creep stress. The limiting resultant values are 1% for the parent metal and 0.5% for the weld metal.

Primary and secondary stress ranges, from table 10.1 are 4.2 ksi and 46.1 ksi for the inner region and 2.5 ksi and 31.0 ksi respectively for the outer region.

Yield stress, S_y , is taken as the average values of S_y at 720°F and 1015°F. Thus:

$$S_y = 0.5 (18.0 + 16.8)$$

$$S_y = 17.4 \text{ ksi}$$

Using Code notation, the primary stress parameter, x , is :

$$x_{\text{inner region}} = \frac{4.2}{17.4} = 0.241$$

$$x_{\text{outer region}} = \frac{2.5}{17.4} = 0.144$$

Likewise, the secondary stress parameter, y , is:

$$y_{\text{inner region}} = \frac{46.1}{17.4} = 2.65$$

$$y_{\text{outer region}} = \frac{31.0}{17.4} = 1.78$$

Both locations are in the "P", plastic cycling, regime. Thus, the effective creep stress parameter, Z , is:

$$\text{inner region: } Z_i = 0.639$$

$$\text{outer region: } Z_o = 0.257$$

Thus:

$$\sigma_{c,i} = 11.1 \text{ ksi}$$

$$\sigma_{c,o} = 4.5 \text{ ksi}$$

The total service life is divided into 100 hours at 1050°F (conservative for both time and temperature) and 2.6×10^5 hours at 800°F (also conservative).

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The effective stresses for entering the isochronous curves are: $1.25 (11.1) = 13.9$ ksi and $1.25 (4.5) = 5.6$ ksi. The resultant strains are given below.

	inner region	outer region
100 hours at 1050°F	0.16	0.06
3×10^5 hours at 800°F	<u>0.06</u>	<u>0.03</u>
Total strain (%)	0.22	0.09

These strains, conservatively calculated, are well within the allowable strains of 1% for parent metal and 0.5% for weld metal.

10.3 Creep - Fatigue Evaluation

An analysis must be performed to ensure an acceptable level of accumulated creep-fatigue damage. Limits for elastic analysis are shown in figure T-1420-2 of Code Case 1592.

The creep damage contribution to the total damage is $\sum \frac{t}{T_d}$. Stresses from sustained operating conditions are low, less than 6 ksi. Sustained normal operating temperature are also low, 770°F. The time-to-rupture curves for SS316 show essentially zero damage from extended operating conditions at 770°F or from the short time spent above 800°F. Thus creep damage is neglected for all subsequent calculations.

The fatigue damage contribution to total damage is $\sum \frac{n_i}{N_d}$. Peak thermal stresses have been calculated for the IP-3UT transient. The effective strain range for this transient is calculated according to Paragraph T-1413 in Code Case 1592, using the stress components from the finite element analysis. The maximum stresses are taken at $t = 120$ second. The range is taken to be those at $t = 720$ seconds where both the radial and hoop stresses reverse their directions. All other transient effective strain ranges are

~~scaled according to the temperature differences from the IP-3UT transient~~

since the finite element results include all stress concentration factors. The modulus of elasticity is conservatively taken as $E = 22.5 \times 10^6$ psi at 1000°F . Thus $\epsilon_R = 3.57 \times 10^{-3}$ for the inner region.

The elastic fatigue curves for $1000^\circ\text{F} - 1200^\circ\text{F}$ are used to calculate N_d since the maximum temperature during the cycle is 1015°F .

The number of cycles to failure, N_d , for the inner region is $N_d = 33$, for the temperature range $1000^\circ\text{F} - 1200^\circ\text{F}$. The maximum fatigue damage occurs in the inner region. In order to calculate fatigue damage for all transients, the peak thermal stress is modified by the ratio of ΔT for that transient to the $\Delta T (=285^\circ\text{F})$ for the IP-3UT transient. The sum of the individual damages for the outer region is calculated in table 10.2. It is:

$$\sum \frac{n_i}{N_d} = 0.690$$

Since creep damage is 0.0, the total accumulated damage is 0.690. This is less than the allowable damage of 1.0.

10.4 Deformation Limits for Functional Operability

In addition to ratchetting requirements for structural integrity, T - 1300 of Code Case 1592, additional deformation limits must be established and satisfied to ensure proper component functioning, T - 1200.

These limits will probably be more stringent than the ratchetting requirements for the bearing regions which essentially allow an accumulated 1% non-transient deformation. This limit is not acceptable when applied to radial deformation, circumferential runout (out-of-roundness), centerline offset, etc.

The present analysis considers only the bearing clearance during transient conditions, see Section 9.4. Additional criteria must be established and met to ensure operability for the entire design life.

	INNER REGION	OUTER REGION
PEAK STRESS INTENSITY	68	55
LOCATION (ELEMENT)	97	69
TIME (SECONDS)	120	380
LINEARIZED STRESS INTENSITY	46.1	31
LOCATION (ELEMENT)	124	182
TIME (SECONDS)	120	380
THERMAL MEMBRANE STRESS	15	15
LOCATION (ELEMENT)	124-130	69-89
TIME (SECONDS)	120	380
PRESSURE & ROTATION STRESS	1.7	2.5
TORSIONAL STRESS+WEIGHT	2.5	0.0
PRIMARY STRESS INTENSITY	4.2	2.5

NOTE: All stresses are in ksi.

The linearized stress intensity at inner region is obtained by multiplying the calculations using the equations in Section 9.2 by the factor 1.112 for three elements across the thickness.

MIDDLE BEARING

STRESS SUMMARY

TABLE 10.1

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LOCATION = ELEMENT 98 ; NODE 204 ; MIDDLE R'IA 119
INNER REGION

TIME (SEC)	σ_x (KSI)	σ_y (KSI)	τ_{xy} (KSI)	σ_z (KSI)	ϵ_x	ϵ_y	ϵ_z	γ_{xy}
120	-39.3	6.7	-4.0	-41.6	-1.28×10^{-3}	1.38×10^{-3}	-1.41×10^{-3}	-0.46×10^{-3}
380	-29.8	3.9	-4.2	-34.2				
720	36.0	-4.2	3.9	38.4	1.14	-1.18	1.28	0.45

120 SEC VS 720 SEC

$$\Delta \epsilon_x = -2.42 \times 10^{-3}$$

$$\Delta \epsilon_y = 2.56$$

$$\Delta \epsilon_y = -2.69$$

$$\Delta \gamma_{xy} = -0.91$$

$$\Delta \epsilon_{EQUIV} = \frac{\sqrt{2}}{3} \sqrt{(4.98)^2 + (5.25)^2 + (0.27)^2 + 6(0.91)^2}$$

$$= 3.57 \times 10^{-3}$$

TRANSIENT	ΔT	n_i	$\Delta \epsilon_{EQUIV}$	N_d	n_i/N_d
IP-3UT	285	15	3.57×10^{-3}	33	0.455
IP-1UT	70	480	0.88	$> 10^6$	—
IP-5UT	80	10	1.00	$> 10^6$	—
IP-6UT	90	5	1.13	$> 10^6$	—
IP-7UT	125	14	1.57	$> 10^6$	—
IP-8UT	150	63	1.88	1400	0.045
IP-1ET	250	8	3.13	42	0.190
IP-2ET	155	8	1.94	5×10^5	—
IP-5ET	155	8	1.94	5×10^5	—

$$\sum \frac{n_i}{N_d} = 0.690$$

MIDDLE BEARING - INNER REGION

FATIGUE DAMAGE SUMMARY

TABLE 10.2

Page 94

11.0 Conclusions

This analysis shows that the design of the middle bearing is feasible for the conditions given in the Design Specification. Any potential problem areas which exist appear to be minor in nature and seem to be relatively easily solvable by either a minor redesign or more detailed analysis.

The primary stresses meet the Code requirements by wide design margins since the membrane thermal stress is considered to be secondary. The region is conservatively designed for the primary loads consisting of pressure, shaft rotational stresses and shaft transmitted torque.

The Code limits on stresses for structural integrity, i.e. ratchetting limits, are met on an elastic basis for both parent metal and weld regions. Maximum primary stresses of $P_L + P_b = 4.2$ ksi coupled with a maximum linearized secondary stress range of $Q_R = 46.1$ ksi. Since most of the pump operation is at 770°F, the accumulated strain is only 0.22%.

Creep damage is negligible since the operating temperature is low. The only time spent above 850°F occurs during thermal transient conditions. Since this time is less than 100 hours over the entire operating life the creep damage may be neglected.

Allowable fatigue damage is 1.0 due to the insignificant creep damage. The fatigue damage contribution from 15 cycles of the IP-3UT transient is 0.455. The fatigue damage from all other transients, calculated by the ratio of maximum transient temperature difference to that of the IP-3UT times the peak stress of the IP-3UT, is only 0.235, giving a total accumulated fatigue damage of 0.690.

Positive clearance at the bearing surfaces are shown to be maintained

during the IP-3UT.

Thus, the additional Code requirement of functional operability is met for the bearing clearances for this transient. The minimum clearance for the IP-3UT transient is 0.014 inches from an initial clearance of 0.030 inches.

These results have been obtained by considering thermal membrane stress as a secondary stress. The maximum thermal membrane stress is 15 ksi on both the outer structure and the inner structure. This would not meet the primary limits of thermal membrane stress were considered to be primary in nature.

Further consideration must be given to the classification of membrane thermal stress to ensure structural and functional integrity in accordance with the intent of the Code. This subject is discussed in more detail in Section 12.0.

12.0 Discussion and Suggestions

Code Case 1592 criteria are inherently easier to pass for this pump than for the Clinch River Pump since Design and Operating Temperatures are lower and thermal transients have lower temperature ranges while primary stresses remain about the same. The Code comparisons which are summarized in Section 11.0 show a high probability that the bearing may be shown acceptable by elastic analysis.

The results therefore show the design to be feasible for the required conditions. Although this analysis does not constitute a full Code report it shows that with minor design modifications and with further analysis all Code requirements are expected to be met.

Further analysis must be performed to demonstrate the effects of gapping on these results, to determine whether the membrane thermal stress affects functional operability and/or ratchetting and to analyze detailed areas not covered by this analysis such as loss of preload in the bolts.

A finite element model which includes non-linear gap elements is suggested as the most appropriate method of analyzing the effects of gaps in detail. This would include a mesh of only the local region, with the proper boundary conditions obtained from hand calculations or from a larger finite element model.

A simplified inelastic finite element model is suggested to analyze the effects of membrane thermal stress. The inelastic results would then be used to calculate accumulated ratchetting strain and to determine whether bearing clearance is affected. For example, if shakedown to elastic cycling can be shown, and if the bearing surface clearances are not changed then these may be used to help justify the high thermal membrane stress.

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A suggestion is made here to redesign the outer bearing surface region in order to eliminate the welded rings. Although separation of the rings from the retainers is expected to occur during thermal transients, the bearing clearance is not as crucial a problem as failure of the weld by either fatigue or ratchetting. This problem was not studied in detail in this analysis since the mesh was intended for overall thermal/stress representation.

The ring weld, since it is only a partial penetration weld, represents a local reduction in size of the cross section. Thus the ring will be subjected to plastic strain concentrations at the weld due to elastic follow-up in the surrounding structure. Since this region was not studied in detail in this analysis, further analysis, possibly including inelastic analysis, is required to demonstrate structural adequacy. Alternatively, the region may be redesigned to eliminate the expected high stresses at the weld.

13.0 References

1. Byron Jackson's drawing No. IF-7937, Rev. 0, Large Scale Sodium Pump, Two stages in Parallel; Byron Jackson's purchase order No. V222073 to Basic Technology, Inc., and change order dated May 25, 1976.
2. ASME Boiler and Pressure Vessel Code, Section III - Division 1, Rules for Construction of Nuclear Power Plant Components, 1974 Edition with the summer, 1975 addenda; ASME B & PV Code Case 1592, Class 1 Components in Elevated Temperature Service, Section III.
3. General Electric Breeder Reactor Department Document No. 22A3994, Rev. 2, Large Scale Breeder Reactor Plant Prototype Mechanical Pump Conceptual Design Study; and GE Document No. NE56490, dated Feb. 5, 1976.
4. BJ memo from Y. Usui to D. Plapp, 761-G-0002 Transients of Fatigue Analysis, dated May 14, 1976.
5. "ANSYS Engineering Analysis System Users Manual", Swanson Analysis Systems, Inc. (1975).
6. BTI unpublished study, Verification and Optimization of ANSYS Elements For Transient Thermal Stress Problems, completed March 16, 1976.
7. BJ memo from D. Gibson to K. Huber, The First Meeting of the Large Scale Breeder Reactor Upper Bearing Analysis, dated May 21, 1976.
8. J.W. Rizika, "Thermal Lags in Flowing Systems Containing Heat Capacitors", Trans. ASME Vol. 76, 1954 pgs. 411-420
9. G.E. Specication No. 22A3444, Rev. 2 as modified by G.E. for the large scale sodium pump, confirmed by BJ memo from D. Gibson to K. Huber, the Second Meeting of the Large Scale Breeder Reactor Upper Bearing Analysis, dated May 27, 1976.

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10. "Liquid Metal Heat Transfer", Sodium & Nak Supplement to the Liquid Metals Handbook, Vol. 2, Chapter 1, January, 1969.
11. "Large Scale Sodium Pump Upper Bearing Analysis", BTI 76-410-001, 6/25/76.
12. See, for example, "Marks' Standard Handbook for Mechanical Engineers", Seventh Edition, McGraw-Hill (1958).
13. The representative values of the film coefficients are chosen on the basis of previous work BTI has done on the liquid sodium pumps.
14. "Nuclear Systems Materials Handbook, Vol. 1: Design Data", Hanford Engineering Development Laboratory, Richland, Washington.
15. S. P. Timoshenk & S. Woinowsky-Krieger, "Theory of Plates and Shells", Second Edition, McGraw-Hill (1959)
16. BJ memo from D. Gibson to BTI dated May 28, 1976, confirming the horsepower and rotation speed of the pump shaft at normal operation.
17. R. J. Roark, "Formulas for Stress and Strain", McGraw-Hill (1975).

14.1 Appendix I - Heat loss of fluid through a cylinder

Consider a fluid whose temperature is initially T_1 flowing with a velocity V through the axial direction of a cylinder of radius d and wall thickness l . The cylinder is immersed in a bulk fluid of temperature T_o . The flowing fluid convects heat to the inside wall of the cylinder. Heat is conducted through the thickness of the cylinder to the outside wall, to which the bulk fluid convects. The temperature $T(x)$ of the flowing fluid along the axis of cylinder is determined by the following differential equation:

$$\frac{d^2T}{dx^2} - \frac{\rho cv}{k} \frac{dT}{dx} - \frac{2HT}{kd} = - \frac{2HT_o}{kd}$$

where ρ , c , k are the density, heat capacity and conductivity of the transient fluid and H is given by

$$H = \frac{h_i h_o k_w}{k_w h_o + l h_i h_o + k_w h_i}$$

where k_w is the conductivity of the cylinder wall, h_i and h_o are the film coefficients on the inside and outside walls of the cylinder, respectively.

The solution of the differential equation is readily obtained and is given by

$$T = (T_i - T_o) e^{\frac{1}{2} \left\{ \frac{c\rho v}{k} - \left[\left(\frac{c\rho v}{k} \right)^2 + \frac{8H}{kd} \right]^{1/2} \right\} x} + T_o$$

where x is the distance along the axis of the cylinder, with T satisfying the initial boundary condition of $T = T_1$ at $x=0$.

14.2 Appendix II - Adiabatic mixing with piecewise linear transient fluid

Consider a large insulated volume V of fluid initially at temperature

T_0 . Transient fluid with piecewise linear temperature vs. time $T_1(t)$, enters the volume with a flow rate of $\dot{q}$. The temperature of the bulk fluid as a function of time satisfies the following differential equation

$$\dot{T} = \gamma (T_1 - T)$$

where $\gamma = \frac{\dot{q}}{V}$.

Assume $T_1(t) = a + bt$ for $t_1 < t < t_2$. The solution to the differential equation is

$$T(t) = \frac{1}{\gamma} \left\{ T(t_1) e^{\gamma(t-t_1)} + \left(a - \frac{b}{\gamma}\right) (e^{\gamma t_2} - e^{\gamma t_1}) + \frac{b}{\gamma} (e^{\gamma t} - e^{\gamma t_1}) \right\}$$

The constants a, b are determined by the initial and final temperatures of the piecewise linear transient. Assuming $T = T_1$ at $t = t_1$ and $T = T_2$ at $t = t_2$, we find

$$a = T_1 - (T_1 - T_2) \left(\frac{t_1}{t_2 - t_1} \right)$$

$$b = \frac{T_1 - T_2}{t_1 - t_2}$$

The boundary conditions are $T = T_1 = T_0$ at $t = 0$. The calculated temperature $T(t_2)$ for the first leg of the transient becomes the initial temperature for the second leg of the transient, and so on. Therefore, in general,

$$\text{with } a_i = T_i - (T_i - T_{i+1}) \left(\frac{t_i}{t_{i+1} - t_i} \right) \text{ and } b = \frac{T_i - T_{i+1}}{t_i - t_{i+1}}$$

A FORTRAN computer program is written to solve the above set of equations.

The input quantities to the program are γ, T_0 and the sets of temperature and time points $\{t_i, t_{i+1}, T_i, T_{i+1}\}$ to describe the piecewise linear transient curve. A listing of the FORTRAN is included herewith.

```

PROGRAM MIX(INPUT,OUTPUT,TAPE5=INPUT,TAPE6=OUTPUT)
READ(5,10) GAMMA,T1
WRITE(6,20) GAMMA
WRITE(6,30)
5 READ(5,10) TI,TF,TEMPI,TEMPF
IF(FOF(5)) 999,6
6 DT=TF-TI
  DTEMP=TEMPF-TEMPI
  A=TEMPI-DTEMP*TI/DT
  B=DTEMP/DT
  E1=EXP(GAMMA*TI)
  E2=EXP(GAMMA*TF)
  AB=A-B/GAMMA
  F1=T1*E1
  F2=AB*(E2-E1)
  F3=B*(TF*F2-TI*E1)
  T2=(F1+F2+F3)/E2
  WRITE(6,20) TI,TF,TEMPI,TEMPF,A,B,T1,T2
  T1=T2
  GO TO 5
999 STOP
10 FORMAT(4F10.2)
20 FORMAT(8F15.4)
30 FORMAT(1H1,9X,2HTI,13X,2HTF,11X,5HTEMPI,11X,5HTEMPF,
112X,1HA,13X,1HB,13X,2HT1,13X,2HT2/)
END

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BYRON JACKSON REPORT TCF-1017-DYN
TORSIONAL VIBRATION ANALYSIS
of
LSBRP PUMP ASSEMBLY

PREPARED FOR

GENERAL ELECTRIC COMPANY
FAST BREEDER REACTOR DEPARTMENT
SUNNYVALE, CALIFORNIA
LETTER CONTRACT 67002X
BJ JOB NO. 761-G-0002

PREPARED BY

DATE 25 June 1976

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DATE 24 June 1976

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DATE 30 JUN 76

June 1976

BYRON JACKSON PUMP DIVISION
BORG WARNER CORPORATION
LOS ANGELES, CALIFORNIA

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APPENDIX A Model Details (Separate Table of Contents)

APPENDIX B Detailed Calculations (Separate Table of Contents)

APPENDIX C TORVIB Run

1.0 SCOPE

This report describes the torsional analysis performed in accordance with the requirements of Reference 6.2 for the pump assembly of Reference 6.1. It includes a detailed description of the model used, the results obtained and draws appropriate conclusions.

2.0 METHOD OF ANALYSIS

The torsional analysis described in this report was conducted using the proprietary Byron Jackson computer program "TORVIB" which is based on the "Holzer" Method. References 6.3 and 6.4 contain the details of this method and of the program capability.

In general, TORVIB uses a mathematical model consisting of a number of inertial masses, separated by sections of massless shafting which provide torsional rigidity. The TORVIB model begins and ends with an inertial mass. The shafting between inertial masses can involve multiple sections of shafting of varying diameters and lengths.

The program generates successive Holzer tables for the entire model each having an angular velocity one radian per second greater than the preceding table. The initial table is formed for an angular velocity of one radian per second. The residual torques for each table are examined until two successive tables are found where the

residual torque differs in sign. Linear interpolation between these two frequencies to locate the point where the residual torque is zero then determines the torsional critical frequency. The Holzer table for this frequency is then printed by the computer along with the critical frequency and the search for the next critical frequency is resumed. The program continues to determine torsional critical frequencies until all possible frequencies have been found or a frequency of 100,000 cycles per minute is reached. On special occasions, the 10^5 cpm limit may be exceeded by entering another upper limit in the input data.

The model used in this analysis is detailed in Appendix A and the resulting computer output using the normal 100,000 cycle per minute upper limit is presented in Appendix C.

3.0 RESULTS

The results of this TORVIB run were nineteen of the twenty-one possible torsional critical frequencies. The first of these frequencies was found to be 599.59 cycles per minute. A rough manual calculation using a single disc model is presented in Appendix B, which predicts a first torsional critical frequency of 567.27 cycles per minute. This agrees with the calculated value within six percent and substantiates the accuracy of the model.

The first nine natural frequencies determined are shown in Table 3.1. The other ten frequencies so determined exceeded 43,000 cycles per minute and are so far from any possible exciting frequency that they need not be considered. They are included in Appendix C for purposes of providing complete information.

TABLE 3.1 Torsional Natural Frequencies

Mode	Frequency (cpm)
1	599
2	5,664
3	6,849
4	13,198
5	17,032
6	21,331
7	25,145
8	31,403
9	36,599

4.0 COMPARISON WITH POSSIBLE EXCITING FREQUENCIES

The major exciting frequency developed by the pump occurs due to pressure pulses generated when impeller vanes pass volute lips (cutwaters). Each impeller will be installed in a portion of the hydraulic assembly with six volute lips. These volute lips are therefore angularly spaced at sixty degree intervals.

This unit has two identical double suction impellers, of five vane design. The vanes for each half of the impeller are staggered with respect to the other half which in effect makes each impeller a ten vane impeller. Thus successive vane tips are displaced from each other by thirty-six degrees.

Every twelve degrees of rotation of the pump shaft will result in two opposing vanes passing two opposing volute lips. This will generate a pressure pulse for every twelve degrees of revolution or thirty pulses per revolution. The design rotating speed of the unit is 600 RPM. Thus each impeller will generate pressure pulses at a rate of 18,000 pulses per minute. This is the internally generated frequency that can excite the unit torsionally. The first four torsional natural frequencies are well below this possible exciting frequency and thus no torsional resonance is expected at these frequencies.

The fifth and sixth natural frequencies straddle the possible excitation frequency. The fifth torsional critical is less than 95 percent of the possible exciting frequency. The sixth torsional critical is more than 118 percent of the exciting frequency which is far enough removed so that no resonance at this natural frequency need be considered.

Inasmuch as there are two impellers and each is to be enclosed in its own portion of the hydraulic assembly, and since the volute lips will be in alignment for the two stages, the excitation frequency can be doubled by installing the two impellers with the vanes offset by eighteen degrees. This results in a possible excitation frequency of 36,000 cycles per minute which is far removed from all torsional natural frequencies except the ninth which is within two percent of this possible exciting frequency.

It is also noted that the first mode torsional natural frequency corresponds almost exactly with the 600 RPM operating speed of the unit. This mode is due to the two impellers vibrating, in phase, against the inertial mass of the motor. The other modes occur at frequencies in excess of 5,000 cpm and their proximity to possible excitation frequencies has already been discussed.

Because the driver is an electric motor, there is little likelihood that the first mode natural frequency, which corresponds to the motor operating speed, will ever be excited. On the other hand, the frequency of this mode can easily be increased since it is very sensitive to the impeller moment of inertia and to the torsional rigidity of the shafting. As an example, changing the shaft dimensions to 18" O.D. and 13" I.D. increases the first mode natural frequency to 727 cpm. A shaft of these dimensions can be easily accommodated in the conceptual design. Similar results can be obtained by decreasing the impeller moment of inertia, a significant portion of which is due to the impeller vanes (See Appendix B). Due to the fact that impeller hydraulic drawings are not available, the moment of inertia was calculated from estimated vane dimensions. Final impeller drawings will optimize material thickness throughout the impeller as well as providing more definitive vane dimensions. In view of these anticipated modifications, optimization of shaft dimensions was not attempted.

5.0 CONCLUSIONS

The torsional vibration analysis has shown that the specification requirements can be satisfied within the existing conceptual design. An optimization of shafting and impeller parameters will be required. However this optimization is deemed to be a detailed design consideration.

6.0 REFERENCES

- 6.1 Byron Jackson Drawing 1F-7937 Rev. A dated 24 May 1976
- 6.2 General Electric Specification 22A3994 Rev. 2 Dated 6 Feb. 1976
- 6.3 Byron Jackson Report TCF-1009-PGM, "Documentation for Program TORVIB"
- 6.4 Handbook on Torsional Vibration, Compiled by E. J. Nestorides of the BICERA Research Laboratory published by the SYNDICS of the Cambridge University Press, 1958.
- 6.5 Telephone Conversation between Hal Gerbe (G.E.) and K. Huber (Byron Jackson) 4 June 1976
- 6.6 Byron Jackson Report TCF-1016-DYN

APPENDIX A
MODEL DETAILS

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APPENDIX A

A-1.0 GENERAL

This appendix describes the model used in this Analysis. As described in the report, the model is required to begin and end with an inertial mass. Between these two inertial masses, additional inertial masses may be placed. Connections between all inertial masses are modeled as one or more sections of massless shafting (multiple sections are connected in series).

The model used for this analysis is basically described in Figure A-1 and consists of 22 inertial masses interconnected by

Figure A-1 TORVIB Model of Rotating Element

21 sections of shafting with torsional stiffness properties only. The identifiers shown in Fig. A-1 are Identifiers only. The values used as input to the program were derived as shown in the following two sections of this Appendix. Section A-2.0 details the derivation of the inertial mass values while Section A-3.0 shows how the torsional stiffnesses were derived. Section A-4.0 includes other calculations.

A-2.0 INERTIAL MASSES

Program TORVIB requires that the inertial masses be input in the form of polar weight moments of inertia (WK^2). The units as input are pound-inch squared.

Relating the model shown in Figure A-1 to the drawing from which it was taken (Reference 6.1) the inertial masses indicated represent the actual parts as indicated in Table A-1 which also indicates the section of the Appendix where the origin of the appropriate values may be found. Most of the moments of inertia required were calculated using a supplementary program called IMPWT which was primarily written for determination of Impeller weights and moments of inertia. The appropriate runs of this program are included in Appendix B and are referenced by number in the appropriate section

A-2.1 Motor Rotor

The moment of inertia of the motor rotor was estimated by the motor manufacturer (G.E.) to be 36,000,000 lb-in². This value was provided to B.J. via Reference 6.5. It is input as M22.

Table A-1 Model Mass Identifiers vs Pump Parts

Masses	Pump Part	Section Number
M1, M2	Lower Hydrostatic Bearing	A-2.6
M3, M4	Impellers	A-2.2
M5, M6	Center Hydrostatic Bearing	A-2.7
M8, M16	Flanged Shaft Couplings	A-2.5
M7, M9, M10, M11, M12, M13, 2M16, M17 M18, M19, M20	Hollow Shafting	A-2.3
M21	Curvic Coupling	A-2.8
M22	Motor Rotor	A-2.1
M14, M15	Upper Hydrostatic Bearing	A-2.4

A-2.2 Impellers

Both impellers are identical hence a single calculation sufficed for both torsional masses. The impellers were considered to be torsionally rigid between wear ring extremities. This calculation was performed using TIMPWT. This was a temporarily changed version of IMPWT which would permit the output of a large moment of inertia. The value of the moment of inertia thus determined was 4,791,514 lb-in². The Computer Run is contained in Appendix B as Run B-1. The dimensions used in this run were scaled from Revision 0 of Reference 6.1. Between the original issue and Reference 6.1 itself, no changes were made to the impeller area and hence the two are identical. M3 and M4 represent the impellers. Moment of Inertia due to the vanes was calculated manually and added to the results of Run B-1 as shown in B-9.

A-2.3 Hollow Shafting

The hollow portion of the shafting was divided into individual sections. The mass of the hollow shafting was calculated from the drawing dimension (16" OD x 2" thick) as a part of the calculation on the upper hydrostatic bearing (See A-2.4). The results of this calculation indicated that the moment of inertia of the shafting was 1238.54 lb-in^2 per inch of length. The lengths of the individual sections of hollow-shafting were taken as being three sections 33 inches long between the wide end of the internal taper at the top of the shaft and the normal sodium level at elevation 10' as shown in Reference 6.1. An additional 22-1/2" section is between the normal sodium level and the upper flanged coupling while a 21-1/2" section is located between this coupling and the upper hydrostatic bearing. Between the upper hydrostatic bearing and the lower flanged coupling the shaft was divided into four sections 34" long, another 33" long and one 32" long in the order specified here. Below the lower flanged coupling, a 21" section was included which ends at the beginning of the external taper on the shaft. The inertial mass of the remaining sections of shafting was disregarded as being relatively insignificant. The only exceptions to this are in the case of the three hydrostatic bearings when the inertial mass of the shafting within the bearings was considered to be part of the bearings.

For each section of shafting, the inertial mass was calculated by multiplying the moment of inertia, ($1238.54 \text{ lb in}^2/\text{in}$) by the length of the section and applying half the resulting mass at each end of

the section. Thus at the junction between two sections of shafting, half the inertial mass of each adjacent section was applied. Where another inertial mass exists at the end of a section, the appropriate portion of the shaft moment of inertia was added to the moment of inertia due to that mass. The results of these calculations are tabulated in Table A-2.

Table A-2. Moments of Inertia Hollow Shaft Section

Length (in)	Total Moment of Inertia (lb-in ²)	Moment of Inertia Applied at each end (lb-in ²)
33	40871.82	20435.91
22-1/2	27867.15	13933.58
21-1/2	26628.61	13314.31
34	42110.36	21055.18
32	39633.28	19816.64
21	26009.34	13004.67

Accordingly the inertial masses input to represent the shafting were calculated as indicated in Table A-3.

Table A-3 Moments of Inertia Hollow Shafting

Mass Number	Inertia Above Location	Inertia Below Location	Total Shafting Inertia	To be Added
M20	0	20435.9	20435.9	---
M19	20435.9	20435.9	40871.8	---
M18	20435.9	20435.9	40871.8	---
M17	20435.9	13933.6	34369.5	---
M16	13933.6	13314.3	27247.9	Flgd Cplg
M15	13314.3	0	13314.3	Upper Hydro Brg.
M14	0	21055.2	21055.2	Upper Hydro Brg.
M13	21055.2	21055.2	42110.4	---
M12	21055.2	21055.2	42110.4	---
M11	21055.2	21055.2	42110.4	---
M10	21055.2	20435.9	41491.1	---
M9	20435.9	19816.6	40252.5	---
M8	19816.6	13004.7	32821.3	Flgd Cplg.
M7	13004.7	0	13004.7	---

A-2.4 Upper Hydrostatic Bearing

Torsionally speaking, the upper hydrostatic bearing is symmetrical about its transverse centerline. The dimensions of the bearing were scaled from Reference 6.1 and used as input to IMPWT. The results of this run are presented as run B-2 in Appendix B. The total moment of inertia of each half of the upper hydrostatic bearing was calculated to be 60330.04 lb-in⁴. This value when added to the shafting values shown in Table A-3 for M14 and M15 yield the values 81385.2 and 73644.3 lb-in² respectively which are included in the program input.

A-2.5 Flanged Couplings

The flanged couplings were calculated using dimensions scaled from Reference 6.1. They were considered to be single torsional discs and to be torsionally rigid. The calculation was performed by IMPWT and showed the moment of inertia to be 26048.37 lb in². This run is included in Appendix B as Run B-3. Torsional masses M8 and M16 were then determined by adding this value to the values for these masses tabulated in Table A-3. The resulting inertial masses as input to the computer are therefore 58869.7 and 53296.3 lb in² for masses M8 and M16 respectively.

A-2.6 Lower Hydrostatic Bearing

The lower hydrostatic bearing is symmetrical about its transverse centerline and hence a single calculation suffices for both inertial masses. This calculation was performed by IMPWT using dimensions scaled from Reference 6.1. The run is included in Appendix B as run B-4. The moment of inertia determined by this run was 63356.8 lb in² for each half of the bearing. These were input to the program as M1 and M2.

A-2.7 Center Hydrostatic Bearing

The center hydrostatic bearing is not quite symmetrical due to the internal taper of the shafting extending through the plane of the upper support disc. Accordingly, two separate calculations were required. The lower portion was calculated first using IMPWT and dimensions scaled from Reference 6.1. This run resulted in a moment of inertia of 78067.56 lb in² for the lower portion of this bearing. The run is included in Appendix B as Run B-5. This is the value of M5 as input to the computer.

Using the results of Run B-5 as a starting point, IMPWT was run again with the additional material due to the internal taper added. Unfortunately, IMPWT permits an added weight and volume but does not incorporate the moment of inertia. This is obvious from the run on this portion of this bearing (See Run B-6 in Appendix B). The additional moment of inertia due to the added material within the shaft was 349.16 lb in^2 which when added to the 78067.56 lb in^2 calculated for the lower portion of this bearing yields 78416.7 lb in^2 which was used as M6 in the TORVIB input.

A-2.8 Curvic Coupling

Those portions of the curvic coupling which involve the bolted flanges and the hub were considered to be torsionally rigid and contribute to the inertial mass of the system. That portion of the curvic coupling which extends to the self centering spline as depicted in Reference 6.1 was considered to provide no inertial mass.

Accordingly, the dimensions were scaled from Reference 6.1, run through IMPWT and the results were found to be 172030.0 lb-in^2 as shown in Run B-7 in Appendix B. Note in B-7 that in order to get this result, it was necessary to alter the output format to locate the decimal point so that the entire value would be printed. At the same time, the spelling of the word "inertia" was corrected. This was a temporary insertion into the program for this run only. This value was used as M21 in the input to TORVIB

A-3.0 Torsional Stiffnesses

TORVIB permits torsional stiffnesses to be input in two different ways. These are after direct calculation or by means of shaft geometry. When the shaft geometry method is used for input, the program first calculates an equivalent length of 2" diameter shafting for that particular geometry. The equivalent length of successive sections of shafting are added to determine a single equivalent length from which is determined the torsional stiffness between inertial masses.

The equivalent length calculation is described in References 6.3 and 6.4. Most of the geometries included in Reference 6.1 are included in program TORVIB.

The method chosen for input of torsional stiffness to TORVIB for this analysis was the shaft geometry method. Accordingly, the sections of shafting between inertial masses were scaled from Reference 6.1 and input directly. The logic involved in the inputs for torsional stiffness is explained briefly in the following sections.

A-3.1 Hydrostatic Bearings

In the three hydrostatic bearings, the shaft torsional stiffness between the two inertial masses is considered to be the stiffness of the shaft between the two masses. The effects of the journal portion of the bearing were disregarded. Hence the torsional stiffnesses represented by S_1 , S_5 and S_{14} as defined by Figure A-1 were input by the use of the shaft dimensions only. These were scaled from Reference 6.1.

A-3.2 Shafting Inside the Hydraulic Assembly

The remaining shafting inside the hydraulic assembly, represented by S2, S3, and S4 as defined by Figure A-1 are represented to the computer by the geometric dimensions as scaled from Reference 6.1.

A-3.3 Hollow Shafting

Sections S7 , S8, S9, S10, S11, S12, S13, S15, S16, S17, S18 and S19 as defined by Figure A-1 all represent sections of hollow shafting. These were input to the computer program TORVIB by the use of their geometry. Note that the lengths of the physical inertial masses were not included as these portions were considered to be torsionally rigid.

A-3.4 Motor Shafting

Reference 6.1 indicates a motor shaft diameter of 13". This was superseded by later information included in Reference 6.6 which provided a motor shaft diameter of 14" and a length of 24" from the motor mounting face to the rotor. The input to TORVIB shows a motor shaft 14" in diameter and a length of 29.333". This length of the motor shaft was obtained by adding the 24" between the rotor proper and the mounting face, the additional inch between the mounting face and the upper face of the curvic coupling hub, plus an additional 4.333 inches which is one-third the 13" coupling penetration. This convention is described and justified in Reference 6.4. This motor shaft is represented by S21.

A-3.5 Curvic Coupling to Hollow Shaft

This section of shafting is represented in Figure A-1 as S20. This is a composite section made up of several different geometries. From the lower face of the flange of the curvic coupling downward, the shafting was considered to be solid at 12-1/2" diameter as shown by Reference 2.1.

The next section (the threaded section) was scaled to determine dimensions. Scaling was used for the next two sections of the same diameter (this was made two sections for thermal reasons as discussed in Section A-4.0). The next short section is a solid section with an external taper, and then two more sections were used which require additional explanation. All of the above mentioned sections which were not assigned dimensions by Reference 6.1 were scaled from that reference. The last two sections, immediately adjacent to M₂₀, were approximated by extending the internal taper beginning at M₂₀ for a total length of 8" and then using a plane perpendicular to the axis of the shaft at this point. Above this plane, the shaft was considered to be solid. It is felt that the error introduced by this approximation is negligible for the purposes of this analysis.

A-3.6 Immediately above Center Hydrostatic Bearing

The only other section of shafting is that represented by S6 in Figure A-1. This again is a composite section of shafting, made up of two separate sections. The upper section of the two is a section with a straight bore and an external taper which was input directly from the scaled dimensions. The remaining section is a section with both internal and external taper. TORVIB does not calculate this geometry and hence estimated dimensions using straight bore and turn were used. For the purposes of this analysis, the possible error introduced by this approximation was considered to be negligible.

A-4.0 Other Calculations

The only other items required for input to TORVIB are the moduli of elasticity of the various shaft sections. In accordance with Reference 6.2, the temperature of the sodium is 850°F. and all material below normal sodium level (Elevation 10' on Reference 6.1) was assumed to be at that temperature. Modulus of elasticity for Austenitic stainless at this temperature is 23.7×10^6 psi. This value was used for all shafting between M1 and M17 which was purposely placed at this point for this reason.

All shafting material above the lower face of the top plate of the cover gas cavity was considered to be at 200°F. with a modulus of elasticity of 27.7×10^6 psi. This point is the point at which the division between two shaft sections of the same diameter was made as mentioned in Section A-3.5.

Between these two points, the modulus of elasticity was considered to vary linearly with length. Hence the moduli of elasticity as input were obtained by linear interpolation at the center of length of each section. The results of this interpolation are listed along with the remaining computer input and the Holzer tables at the torsional critical frequencies in Appendix C.

In addition, a preliminary check calculation was performed manually using a simplified model. For this calculation, the motor moment of inertia was assumed to be infinite, the total impeller moment of inertia (the sum of the inertias for the two impellers) was concentrated at the midpoint between the two actual impeller locations and the shafting was assumed to be uniform (16" O.D. x 12" I.D.) over the entire length.

Using this simplified model, the 1 node critical frequency was determined to be 659.72 RPM. This calculation is included in Appendix B as B-8. The purpose of this calculation was to provide some verification of the results from the program.

APPENDIX B

DETAILED CALCULATIONS

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B-1

TIMPWT IMPELLER CALCULATION

3-1

ENTER A CODE INDICATING THE SHAPE OF THE CROSS-SECTION.

CODE.	SHAPE
0	OTHER THAN INDICATED BELOW
1	HOLLOW DISC (EITHER RECTANGULAR OR PARALLELOGRAM)
2	TRAPEZOIDAL TOROID (TWO FACES PARALLEL TO SHAFT)
3	TRIANGULAR TOROID (FACE PARALLEL TO SHAFT ON OD)
4	NO ADDITIONAL ENTRIES

21

ENTER: INSIDE DIAMETER/OUTSIDE DIAMETER/LENGTH

737.68,1

VOLUME OF THIS SECTION = 0.25564678E+04 CUIN

WEIGHT OF THIS SECTION = 0.719901123E+03 LBS

MOMENT OF INERTIA THIS SECTION = 0.539295812500E+06LB-IN²

ENTER: NUMBER OF IDENTICAL SECTIONS

22

IS THE NEXT SECTION SHAPED SIMILARLY? ENTER 0 FOR YES, 1 FOR NO

20

ENTER: INSIDE DIAMETER/OUTSIDE DIAMETER/LENGTH

732.52,1

VOLUME OF THIS SECTION = 0.13194668E+04 CUIN

WEIGHT OF THIS SECTION = 0.371561768E+03 LBS

MOMENT OF INERTIA THIS SECTION = 0.173147812500E+06LB-IN²

ENTER: NUMBER OF IDENTICAL SECTIONS

22

IS THE NEXT SECTION SHAPED SIMILARLY? ENTER 0 FOR YES, 1 FOR NO

20

ENTER: INSIDE DIAMETER/OUTSIDE DIAMETER/LENGTH

728.5,36,1.5

VOLUME OF THIS SECTION = 0.56990381E+03 CUIN

WEIGHT OF THIS SECTION = 0.160484909E+03 LBS

MOMENT OF INERTIA THIS SECTION = 0.422927617188E+05LB-IN²

ENTER: NUMBER OF IDENTICAL SECTIONS

22

IS THE NEXT SECTION SHAPED SIMILARLY? ENTER 0 FOR YES, 1 FOR NO

20

ENTER: INSIDE DIAMETER/OUTSIDE DIAMETER/LENGTH

724.5,30.5,2

VOLUME OF THIS SECTION = 0.51836206E+03 CUIN

WEIGHT OF THIS SECTION = 0.145970749E+03 LBS

MOMENT OF INERTIA THIS SECTION = 0.279260234375E+05LB-IN²

ENTER: NUMBER OF IDENTICAL SECTIONS

22

IS THE NEXT SECTION SHAPED SIMILARLY? ENTER 0 FOR YES, 1 FOR NO

20

ENTER: INSIDE DIAMETER/OUTSIDE DIAMETER/LENGTH

721.26,2

B-1

VOLUME OF THIS SECTION = 0.36913672E+03 CUIN
WEIGHT OF THIS SECTION = 0.103948898E+03 LBS
MOMENT OF INERTIA THIS SECTION = 0.145138632813E+05LB-IN:2
ENTER: NUMBER OF IDENTICAL SECTIONS

?? /

IS THE NEXT SECTION SHAPED SIMILARLY? ENTER 0 FOR YES, 1 FOR NO

?0 /

ENTER: INSIDE DIAMETER/OUTSIDE DIAMETER/LENGTH

?19,22.5,2 /

VOLUME OF THIS SECTION = 0.22815790E+03 CUIN
WEIGHT OF THIS SECTION = 0.642492523E+02 LBS
MOMENT OF INERTIA THIS SECTION = 0.696501953125E+04LB-IN:2
ENTER: NUMBER OF IDENTICAL SECTIONS

?? /

IS THE NEXT SECTION SHAPED SIMILARLY? ENTER 0 FOR YES, 1 FOR NO

?0 /

ENTER: INSIDE DIAMETER/OUTSIDE DIAMETER/LENGTH

?17,20.5,4 /

VOLUME OF THIS SECTION = 0.41233350E+03 CUIN
WEIGHT OF THIS SECTION = 0.116113098E+03 LBS
MOMENT OF INERTIA THIS SECTION = 0.102941484375E+05LB-IN:2
ENTER: NUMBER OF IDENTICAL SECTIONS

?? /

IS THE NEXT SECTION SHAPED SIMILARLY? ENTER 0 FOR YES, 1 FOR NO

?0 /

ENTER: INSIDE DIAMETER/OUTSIDE DIAMETER/LENGTH

?15.5,19.4,75 /

VOLUME OF THIS SECTION = 0.45047437E+03 CUIN
WEIGHT OF THIS SECTION = 0.126853577E+03 LBS
MOMENT OF INERTIA THIS SECTION = 0.953383593750E+04LB-IN:2
ENTER: NUMBER OF IDENTICAL SECTIONS

?? /

IS THE NEXT SECTION SHAPED SIMILARLY? ENTER 0 FOR YES, 1 FOR NO

?0 /

ENTER: INSIDE DIAMETER/OUTSIDE DIAMETER/LENGTH

?47.5,68,1.5 /

VOLUME OF THIS SECTION = 0.27894365E+04 CUIN
WEIGHT OF THIS SECTION = 0.785505127E+03 LBS
MOMENT OF INERTIA THIS SECTION = 0.675558937500E+06LB-IN:2
ENTER: NUMBER OF IDENTICAL SECTIONS

?? /

IS THE NEXT SECTION SHAPED SIMILARLY? ENTER 0 FOR YES, 1 FOR NO

?0 /

ENTER: INSIDE DIAMETER/OUTSIDE DIAMETER/LENGTH

?44,47.5,1.75 /

VOLUME OF THIS SECTION = 0.44016602E+03 CUIN
WEIGHT OF THIS SECTION = 0.123950745E+03 LBS

MOMENT OF INERTIA THIS SECTION = 0.649540312500E+05LB-IN²
ENTER: NUMBER OF IDENTICAL SECTIONS
72 ✓

IS THE NEXT SECTION SHAPED SIMILARLY? ENTER 0 FOR YES, 1 FOR NO
70 ✓

ENTER: INSIDE DIAMETER/OUTSIDE DIAMETER/LENGTH
740, 44, 1.75 ✓

VOLUME OF THIS SECTION = 0.46181348E+03 CUIN
WEIGHT OF THIS SECTION = 0.130046661E+03 LBS
MOMENT OF INERTIA THIS SECTION = 0.574806132813E+05LB-IN²
ENTER: NUMBER OF IDENTICAL SECTIONS
72 ✓

IS THE NEXT SECTION SHAPED SIMILARLY? ENTER 0 FOR YES, 1 FOR NO
70 ✓

ENTER: INSIDE DIAMETER/OUTSIDE DIAMETER/LENGTH
738, 40, 2.25 ✓

VOLUME OF THIS SECTION = 0.27567432E+03 CUIN
WEIGHT OF THIS SECTION = 0.776298824E+02 LBS
MOMENT OF INERTIA THIS SECTION = 0.295381640625E+05LB-IN²
ENTER: NUMBER OF IDENTICAL SECTIONS
72 ✓

IS THE NEXT SECTION SHAPED SIMILARLY? ENTER 0 FOR YES, 1 FOR NO
71 ✓

ENTER NEXT CODE
73 ✓

ENTER: OUTSIDE DIAMETER/LENGTH ALONG THIS DIAMETER/INSIDE DIAMETER
738, 4, 33.5 ✓

VOLUME OF THIS SECTION = 0.51600562E+03 CUIN
WEIGHT OF THIS SECTION = 0.145307175E+03 LBS
MOMENT OF INERTIA THIS SECTION = 0.485183398438E+05LB-IN²
ENTER: NUMBER OF IDENTICAL SECTIONS
72 ✓

IS THE NEXT SECTION SHAPED SIMILARLY? ENTER 0 FOR YES, 1 FOR NO
71 ✓

ENTER NEXT CODE
71 ✓

ENTER: INSIDE DIAMETER/OUTSIDE DIAMETER/LENGTH
732.75, 36, 4.875 ✓

VOLUME OF THIS SECTION = 0.85550000E+03 CUIN
WEIGHT OF THIS SECTION = 0.240908798E+03 LBS
MOMENT OF INERTIA THIS SECTION = 0.713258750000E+05LB-IN²
ENTER: NUMBER OF IDENTICAL SECTIONS
72 ✓

ENTER NEXT CODE

?4 —

TOTAL VOLUME = .2352577E+05 CUIN TOTAL WEIGHT = 6624.352 LBS
I = 0.35426840E+07LB-IN²
STOP

Vane Moment of Inertia is calculated and added to this total in B-9

B.2 IMPWT UPPER HYDROSTATIC BEARING CALCULATION

ENTER A CODE INDICATING THE SHAPE OF THE CROSS-SECTION.

CODE	SHAPE
0	OTHER THAN INDICATED BELOW
1	HOLLOW DISC (EITHER RECTANGULAR OR PARALLELOGRAM)
2	TRAPEZOIDAL TOROID (TWO FACES PARALLEL TO SHAFT)
3	TRIANGULAR TOROID (FACE PARALLEL TO SHAFT ON OD)
4	NO ADDITIONAL ENTRIES

?1 .

ENTER: INSIDE DIAMETER/OUTSIDE DIAMETER/LENGTH

?12,24.5,2

VOLUME OF THIS SECTION = 716.675 CUIN
WEIGHT OF THIS SECTION = 201.8156128 LBS
MOMENT OF INERTIA THIS SECTION = 18775.148438LB-IN²
ENTER: NUMBER OF IDENTICAL SECTIONS

?1 .

PROGRAM LINE 560 LIST DIRECTED INPUT ERROR
RETYPE LAST INPUT LINE STARTING WITH ITEM 1

?1 .

IS THE NEXT SECTION SHAPED SIMILARLY? ENTER 0 FOR YES, 1 FOR NO

?0 .

ENTER: INSIDE DIAMETER/OUTSIDE DIAMETER/LENGTH

?23,24.5,12

PROGRAM LINE 320 LIST DIRECTED INPUT ERROR
RETYPE LAST INPUT LINE STARTING WITH ITEM 1

?23,24.5,12

VOLUME OF THIS SECTION = 671.515 CUIN
WEIGHT OF THIS SECTION = 189.0985107 LBS
MOMENT OF INERTIA THIS SECTION = 26692.421875LB-IN²
ENTER: NUMBER OF IDENTICAL SECTIONS

?1 .

IS THE NEXT SECTION SHAPED SIMILARLY? ENTER 0 FOR YES, 1 FOR NO

?0 .

ENTER: INSIDE DIAMETER/OUTSIDE DIAMETER/LENGTH
?12,16,1

VOLUME OF THIS SECTION = 87.964 CUIN
WEIGHT OF THIS SECTION = 24.7707825 LBS
MOMENT OF INERTA THIS SECTION = 1238.539307LB-IN²
ENTER: NUMBER OF IDENTICAL SECTIONS
?12

PROGRAM LINE 560 LIST DIRECTED INPUT ERROR
RETYPE LAST INPUT LINE STARTING WITH ITEM 1
?12

IS THE NEXT SECTION SHAPED SIMILARLY? ENTER 0 FOR YES, 1 FOR NO
?1...

ENTER NEXT CODE
?4...

TOTAL VOLUME = 2443.7631836 CUIN TOTAL WEIGHT = 688.163 LBS
I = 60330.039063LB-IN²
STOP

B-3 IMPWT FLANGED COUPLING CALCULATION

ENTER A CODE INDICATING THE SHAPE OF THE CROSS-SECTION.
CODE SHAPE

- 0 OTHER THAN INDICATED BELOW
- 1 HOLLOW DISC (EITHER RECTANGULAR OR PARALLELOGRAM)
- 2 TRAPEZOIDAL TOROID (TWO FACES PARALLEL TO SHAFT)
- 3 TRIANGULAR TOROID (FACE PARALLEL TO SHAFT ON OD)
- 4 NO ADDITIONAL ENTRIES

?1 :

ENTER: INSIDE DIAMETER/OUTSIDE DIAMETER/LENGTH
✓ ?12,22.5,4

VOLUME OF THIS SECTION = 1138.040 CUIN
WEIGHT OF THIS SECTION = 320.4719238 LBS
MOMENT OF INERTA THIS SECTION = 26048.367188LB-IN²
ENTER: NUMBER OF IDENTICAL SECTIONS
?1

IS THE NEXT SECTION SHAPED SIMILARLY? ENTER 0 FOR YES, 1 FOR NO
?1

ENTER NEXT CODE
?4

TOTAL VOLUME = 1138.0400391 CUIN TOTAL WEIGHT = 320.472 LBS
I = 26048.367188LB-IN²
STOP

B.4 IMPWT LOWER HYDROSTATIC BEARING CALCULATION

ENTER A CODE INDICATING THE SHAPE OF THE CROSS-SECTION.

CODE	SHAPE
0	OTHER THAN INDICATED BELOW
1	HOLLOW DISC (EITHER RECTANGULAR OR PARALLELOGRAM)
2	TRAPEZOIDAL TOROID (TWO FACES PARALLEL TO SHAFT)
3	TRIANGULAR TOROID (FACE PARALLEL TO SHAFT ON OD)
4	NO ADDITIONAL ENTRIES

?1

ENTER: INSIDE DIAMETER/OUTSIDE DIAMETER/LENGTH

?15,24,3

VOLUME OF THIS SECTION = 827.023 CUIN

WEIGHT OF THIS SECTION = 232.8897247 LBS

MOMENT OF INERTIA THIS SECTION = 23318.078125LB-IN²

ENTER: NUMBER OF IDENTICAL SECTIONS

?1

IS THE NEXT SECTION SHAPED SIMILARLY? ENTER 0 FOR YES, 1 FOR NO

?0

ENTER: INSIDE DIAMETER/OUTSIDE DIAMETER/LENGTH

?15,20,7

VOLUME OF THIS SECTION = 962.112 CUIN

WEIGHT OF THIS SECTION = 270.9304199 LBS

MOMENT OF INERTIA THIS SECTION = 21166.445313LB-IN²

ENTER: NUMBER OF IDENTICAL SECTIONS

?1

IS THE NEXT SECTION SHAPED SIMILARLY? ENTER 0 FOR YES, 1 FOR NO

?0

ENTER: INSIDE DIAMETER/OUTSIDE DIAMETER/LENGTH

?22,24,7

VOLUME OF THIS SECTION = 505.796 CUIN

WEIGHT OF THIS SECTION = 142.4320526 LBS

MOMENT OF INERTIA THIS SECTION = 18872.230469LB-IN²

ENTER: NUMBER OF IDENTICAL SECTIONS

?1

IS THE NEXT SECTION SHAPED SIMILARLY? ENTER 0 FOR YES, 1 FOR NO

?1

ENTER NEXT CODE

?4

TOTAL VOLUME = 2294.9304199 CUIN

TOTAL WEIGHT =

646.252 LBS

I = 63356.753906LB-IN²

STOP

B-5 IMPWT CENTER HYDROSTATIC BEARING CALCULATION

ENTER A CODE INDICATING THE SHAPE OF THE CROSS-SECTION.

CODE	SHAPE
0	OTHER THAN INDICATED BELOW
1	HOLLOW DISC (EITHER RECTANGULAR OR PARALLELOGRAM)
2	TRAPEZOIDAL TOROID (TWO FACES PARALLEL TO SHAFT)
3	TRIANGULAR TOROID (FACE PARALLEL TO SHAFT ON OD)
4	NO ADDITIONAL ENTRIES

?1

ENTER: INSIDE DIAMETER/OUTSIDE DIAMETER/LENGTH

?15,24.5,2

VOLUME OF THIS SECTION = 589.440 CUIN

WEIGHT OF THIS SECTION = 165.9864197 LBS

MOMENT OF INERTIA THIS SECTION = 17122.535156LB-IN²

ENTER: NUMBER OF IDENTICAL SECTIONS

?1

IS THE NEXT SECTION SHAPED SIMILARLY? ENTER 0 FOR YES, 1 FOR NO

?0

ENTER: INSIDE DIAMETER/OUTSIDE DIAMETER/LENGTH

/ ?22.5,24.5,12

VOLUME OF THIS SECTION = 885.928 CUIN

WEIGHT OF THIS SECTION = 249.4773102 LBS

MOMENT OF INERTIA THIS SECTION = 34505.820313LB-IN²

ENTER: NUMBER OF IDENTICAL SECTIONS

?1

IS THE NEXT SECTION SHAPED SIMILARLY? ENTER 0 FOR YES, 1 FOR NO

?0

ENTER: INSIDE DIAMETER/OUTSIDE DIAMETER/LENGTH

/ ?15,19,12

VOLUME OF THIS SECTION = 1281.768 CUIN

WEIGHT OF THIS SECTION = 360.9455566 LBS

MOMENT OF INERTIA THIS SECTION = 26439.261719LB-IN²

ENTER: NUMBER OF IDENTICAL SECTIONS

?1

IS THE NEXT SECTION SHAPED SIMILARLY? ENTER 0 FOR YES, 1 FOR NO

?1

ENTER NEXT CODE

?4

TOTAL VOLUME = 2757.1359863 CUIN

TOTAL WEIGHT =

776.409 LBS

I = 78067.562500LB-IN²

STOP

B.6 IMPWT CENTER HYDROSTATIC BEARING CALCULATION

ENTER A CODE INDICATING THE SHAPE OF THE CROSS-SECTION.

CODE	SHAPE
0	OTHER THAN INDICATED BELOW
1	HOLLOW DISC (EITHER RECTANGULAR OR PARALLELOGRAM)
2	TRAPEZOIDAL TOROID (TWO FACES PARALLEL TO SHAFT)
3	TRIANGULAR TOROID (FACE PARALLEL TO SHAFT ON OD)
4	NO ADDITIONAL ENTRIES

?0

ENTER VOLUME IN CUBIC INCHES AND WEIGHT IN POUNDS
?2757.1359863,776.409

ENTER A CODE INDICATING THE SHAPE OF THE CROSS-SECTION.

CODE	SHAPE
0	OTHER THAN INDICATED BELOW
1	HOLLOW DISC (EITHER RECTANGULAR OR PARALLELOGRAM)
2	TRAPEZOIDAL TOROID (TWO FACES PARALLEL TO SHAFT)
3	TRIANGULAR TOROID (FACE PARALLEL TO SHAFT ON OD)
4	NO ADDITIONAL ENTRIES

?3

ENTER: OUTSIDE DIAMETER/LENGTH ALONG THIS DIAMETER/INSIDE DIAMETER
?15.,2,14

VOLUME OF THIS SECTION = 23.038 CUIN
WEIGHT OF THIS SECTION = 6.4875889 LBS
MOMENT OF INERTIA THIS SECTION = 349.157471LB-IN²
ENTER: NUMBER OF IDENTICAL SECTIONS
?1

IS THE NEXT SECTION SHAPED SIMILARLY? ENTER 0 FOR YES, 1 FOR NO
?1

ENTER NEXT CODE

?4

TOTAL VOLUME = 2780.1740723 CUIN TOTAL WEIGHT = 782.896 LBS
I = 349.157471LB-IN²
STOP

B-7 IMPWT CURVIC COUPLING CALCULATION

ENTER A CODE INDICATING THE SHAPE OF THE CROSS-SECTION.

CODE	SHAPE
0	OTHER THAN INDICATED BELOW
1	HOLLOW DISC (EITHER RECTANGULAR OR PARALLELOGRAM)
2	TRAPEZOIDAL TOROID (TWO FACES PARALLEL TO SHAFT)
3	TRIANGULAR TOROID (FACE PARALLEL TO SHAFT ON OD)
4	NO ADDITIONAL ENTRIES

?1

ENTER: INSIDE DIAMETER/OUTSIDE DIAMETER/LENGTH

?15,30,6.5

VOLUME OF THIS SECTION = 3445.930 CUIN

WEIGHT OF THIS SECTION = 970.3737793 LBS

MOMENT OF INERTIA THIS SECTION = 136458.7500LB-IN²

ENTER: NUMBER OF IDENTICAL SECTIONS

?1

IS THE NEXT SECTION SHAPED SIMILARLY? ENTER 0 FOR YES, 1 FOR NO
?0

ENTER: INSIDE DIAMETER/OUTSIDE DIAMETER/LENGTH

?14,19,14

VOLUME OF THIS SECTION = 1814.268 CUIN

WEIGHT OF THIS SECTION = 510.8977051 LBS

MOMENT OF INERTIA THIS SECTION = 35571.2539LB-IN²

ENTER: NUMBER OF IDENTICAL SECTIONS

?1

IS THE NEXT SECTION SHAPED SIMILARLY? ENTER 0 FOR YES, 1 FOR NO
?1

ENTER NEXT CODE

?4

TOTAL VOLUME = 5260.1953125 CUIN

TOTAL WEIGHT = 1481.271 LBS

I = 172030.0000LB-IN²

STOP

B-8 CHECK CALCULATION FOR FIRST TORSIONAL CRITICAL

LUMP BOTH IMPELLERS

$$I = \frac{2 (4791519)}{386.4} = 24800.823 \text{ lb in sec}^2$$

Shaft Stiffness

$$K = \frac{\pi}{32} G \frac{D_o^4 - D_i^4}{L}$$

$$= \frac{\pi}{32} (11.8 \times 10^6) \left(\frac{16^4 - 12^4}{593} \right)$$

$$= \frac{11.8 \pi (16^4 - 12^4)}{32 (593)} \times 10^6 = 8.75196 \times 10^7$$

Critical Frequency

$$\omega = \sqrt{\frac{K}{I}} = \left[\frac{8.75196 \times 10^7}{24800.823} \right]^{\frac{1}{2}} = [3528.89]^{\frac{1}{2}} \checkmark$$

$$\omega = 59.40 \text{ cps}$$

$$f = \frac{\omega}{2\pi} = \frac{59.40}{2\pi} = 9.45 \text{ RPS}$$

$$f = 60 \times \text{or } 9.45 = 567.27 \text{ RPM}$$

B-9 VANE moment of Inertia

The vanes were assumed to be rectangular parallelepipeds with the following dimensions

45" Long
20" Wide
1" thick

The centroidal Moment of Inertia is then given by

$$I = Wk^2 \quad \text{where } W = \text{weight}$$

$$K = \text{Radius of Gyration} = \frac{h}{\sqrt{12}}$$

$$\therefore I = \rho V \frac{h^2}{12} \quad \text{where } \rho = \text{density}$$

$$V = \text{Volume}$$

The polar Moment of inertia is the sum of two perpendicular centroidal Moments of inertia.

$$\therefore I_p = \frac{\rho V}{12} (h_1^2 + h_2^2)$$

Choosing the proper values for h_1 & h_2 , there become 45 & 1, the extra value is negligible in that

$$45^2 = 2025 \quad \text{and} \quad 1^2 = 1$$

and the dimensions are assumed

$$I_p \approx \frac{\rho V h^2}{12}$$

Using the parallel axis theorem then,

$$I_p \approx \frac{\rho V h^2}{12} + \rho V R^2 = \rho V \left(\frac{h^2}{12} + R^2 \right)$$

where R = the distance between the actual axis and the centroidal axis.
(Assumed to be 18")

$$I_p \approx \rho V \left(\frac{45^2}{12} + 18^2 \right) = \rho V (492.75)$$

$$V = Lwt = 45(20)(1) = 900 \text{ in}^3 \quad \rho = .2816 \text{ #/in}^3$$

$$\therefore I_p = 900(.2816)(492.75) = 124883 \text{ # in}^2$$

there are 10 Vanes for the impeller thus

$I_p = 1245830$ For the entire Impeller ($\# \text{ in}^2$)

Impeller moment of inertia without Vanes is $3542684 \# \text{ in}^2$
(See B-1)

thus the total Moment of inertia is

$4791514 \# \text{ in}^2$

APPENDIX C

TORVIB

COMPUTER RUN

BYRON JACKSON PUMP DIVISION
BORG-WARNER CORPORATION

TORSIONAL VIBRATION CALCULATION

PUMP SIZE AND TYPE
LARGE SCALE SODIUM PU

PURCHASER
MP **DWG IF-7937 REV A**

DATE
24 JUN 19

TORVIB-R2

- INPUT DATA -

MOM INERTIA LB-IN ²	TORSIONAL STIFF LB-IN/RAD F6	OD INCH	LENGTH INCH	OD END 2 IF TAPERED	ID INCH	MOD OF ELAST PSI E6
63356.8	0.000	20.000	14.000	0.000	15.000	23.70
63356.8	0.000	19.000	6.000	0.000	15.000	23.70
4791519.0	0.000	19.000	22.500	0.000	15.000	23.70
4791519.0	0.000	19.000	6.000	0.000	15.000	23.70
78067.6	0.000	19.000	24.000	0.000	15.000	23.70
78416.7	0.000	17.500	4.000	0.000	12.500	23.70
0.0	0.000	17.000	1.000	16.000	12.000	23.70
13004.7	0.000	16.000	21.000	0.000	12.000	23.70
58869.7	0.000	16.000	32.000	0.000	12.000	23.70
40252.5	0.000	16.000	33.000	0.000	12.000	23.70
41491.1	0.000	16.000	34.000	0.000	12.000	23.70
42110.4	0.000	16.000	34.000	0.000	12.000	23.70
42110.4	0.000	16.000	34.000	0.000	12.000	23.70
42110.4	0.000	16.000	34.000	0.000	12.000	23.70
81385.2	0.000	16.000	24.000	0.000	12.000	23.70
73644.3	0.000	16.000	21.500	0.000	12.000	23.70
53296.3	0.000	16.000	22.500	0.000	12.000	23.70
34369.5	0.000	16.000	33.000	0.000	12.000	24.26
40871.8	0.000	16.000	33.000	0.000	12.000	25.39
40871.8	0.000	16.000	33.000	0.000	12.000	26.52
20435.9	0.000	16.000	8.000	-8.000	12.000	27.22
0.0	0.000	16.000	1.500	0.000	0.000	27.38
0.0	0.000	16.000	1.500	14.500	0.000	27.43
0.0	0.000	14.500	7.000	0.000	0.000	27.58
0.0	0.000	14.500	12.500	0.000	0.000	27.70
0.0	0.000	14.250	6.000	0.000	0.000	27.70
0.0	0.000	12.500	22.000	0.000	0.000	27.70
172030.0	0.000	14.000	29.333	0.000	0.000	27.70
36000000.0	0.000	0.000	0.000	0.000	0.000	30.00

BYRON JACKSON PUMP DIVISION
BORG-WARNER CORPORATION

TORSIONAL VIBRATION CALCULATION

PUMP SIZE AND TYPE
LARGE SCALE SODIUM PU

PURCHASER
MP **DWG IF-7937 REV A**

DATE
24 JUN 19

1 NODE FREQUENCY IS 599.59 RPM

NO	MOM INERTIA LR-IN-SEC ²	X LR-IN/RAD	BETA RADIAN	TORQ IN-LB	SUM IN-LB	SPRG CNST LR-IN/RAD	PFL ANGLE RADIAN
1	.1640E+03	.6464E+06	.1000E+01	.6464E+06	.6464E+06	.6991E+10	.9246E-04
2	.1640E+03	.6464E+06	.9999E+00	.6464E+06	.1293E+07	.1189E+11	.1088E-03
3	.1240E+05	.4889E+08	.9998E+00	.4888E+08	.5017E+08	.3170E+10	.1583E-01
4	.1240E+05	.4889E+08	.9840E+00	.4810E+08	.9827E+08	.1189E+11	.8268E-02
5	.2020E+03	.7965E+06	.9757E+00	.7772E+06	.9905E+08	.2972E+10	.3333E-01
6	.2029E+03	.8001E+06	.9424E+00	.7540E+06	.9981E+08	.1170E+11	.8534E-02
7	.3366E+02	.1327E+06	.9338E+00	.1239E+06	.9993E+08	.1909E+10	.5234E-01
8	.1524E+03	.6006E+06	.8815E+00	.5295E+06	.1005E+09	.1253E+10	.8018E-01
9	.1042E+03	.4107E+06	.8013E+00	.3291E+06	.1008E+09	.1215E+10	.8296E-01
10	.1074E+03	.4233E+06	.7184E+00	.3041E+06	.1011E+09	.1179E+10	.8573E-01
11	.1090E+03	.4297E+06	.6326E+00	.2718E+06	.1014E+09	.1179E+10	.8596E-01
12	.1090E+03	.4297E+06	.5467E+00	.2349E+06	.1016E+09	.1179E+10	.8616E-01
13	.1090E+03	.4297E+06	.4605E+00	.1979E+06	.1018E+09	.1179E+10	.8633E-01
14	.2106E+03	.8304E+06	.3742E+00	.3107E+06	.1021E+09	.1670E+10	.6112E-01
15	.1906E+03	.7514E+06	.3130E+00	.2352E+06	.1023E+09	.1865E+10	.5488E-01
16	.1379E+03	.5438E+06	.2582E+00	.1404E+06	.1025E+09	.1782E+10	.5751E-01
17	.8895E+02	.3507E+06	.2006E+00	.7036E+05	.1026E+09	.1244E+10	.8245E-01
18	.1058E+03	.4170E+06	.1182E+00	.4929E+05	.1026E+09	.1302E+10	.7883E-01
19	.1058E+03	.4170E+06	.3937E-01	.1642E+05	.1026E+09	.1359E+10	.7548E-01
20	.5289E+02	.2085E+06	-.3612E-01	-.7531E+04	.1026E+09	.6131E+09	.1674E+00
21	.4452E+03	.1755E+07	-.2035E+00	-.3572E+06	.1023E+09	.1370E+10	.7465E-01
22	.9317E+05	.3673E+09	-.2781E+00	-.1022E+09	.9540E+05	.0000E+01	.0000E+01

BYRON JACKSON PUMP DIVISION
BORG-WARNER CORPORATION

TORSIONAL VIBRATION CALCULATION

PUMP SIZE AND TYPE
LARGE SCALE SODIUM PUI

PURCHASER
MP **DWG IF-7937 REV A**

DATE
24 JUN 19

2 NODE FREQUENCY IS 5663.68 RPM

NO	MOM INERTIA LB-IN-SEC ²	X LB-IN/RAD	BETA RADIAN	TORO IN-LB	SUM IN-LB	SPRG CNST LB-IN/RAD	REL ANGLE RADIAN
1	.1640E+03	.5768E+08	.1000E+01	.5768E+08	.5768E+08	.6991E+10	.8250E-02
2	.1640E+03	.5768E+08	.9918E+00	.5720E+08	.1149E+09	.1189E+11	.9665E-02
3	.1240E+05	.4362E+10	.9821E+00	.4284E+10	.4399E+10	.3170E+10	.1388E+01
4	.1240E+05	.4362E+10	-.4056E+00	-.1769E+10	.2629E+10	.1189E+11	.2212E+00
5	.2020E+03	.7107E+08	-.6268E+00	-.4455E+08	.2585E+10	.2972E+10	.8698E+00
6	.2029E+03	.7139E+08	-.1497E+01	-.1068E+09	.2478E+10	.1170E+11	.2119E+00
7	.3366E+02	.1184E+08	-.1709E+01	-.2023E+08	.2458E+10	.1909E+10	.1287E+01
8	.1524E+03	.5359E+08	-.2996E+01	-.1606E+09	.2297E+10	.1253E+10	.1834E+01
9	.1042E+03	.3664E+08	-.4829E+01	-.1770E+09	.2120E+10	.1215E+10	.1745E+01
10	.1074E+03	.3777E+08	-.6575E+01	-.2483E+09	.1872E+10	.1179E+10	.1587E+01
11	.1090E+03	.3834E+08	-.8162E+01	-.3129E+09	.1559E+10	.1179E+10	.1322E+01
12	.1090E+03	.3834E+08	-.9484E+01	-.3636E+09	.1195E+10	.1179E+10	.1014E+01
13	.1090E+03	.3834E+08	-.1050E+02	-.4025E+09	.7929E+09	.1179E+10	.6725E+00
14	.2106E+03	.7409E+08	-.1117E+02	-.8276E+09	-.3468E+08	.1670E+10	-.2076E-01
15	.1906E+03	.6704E+08	-.1115E+02	-.7475E+09	-.7822E+09	.1865E+10	-.4195E+00
16	.1379E+03	.4852E+08	-.1073E+02	-.5206E+09	-.1303E+10	.1782E+10	-.7312E+00
17	.8895E+02	.3129E+08	-.9999E+01	-.3129E+09	-.1616E+10	.1244E+10	-.1299E+01
18	.1058E+03	.3721E+08	-.8700E+01	-.3237E+09	-.1939E+10	.1302E+10	-.1490E+01
19	.1058E+03	.3721E+08	-.7210E+01	-.2683E+09	-.2208E+10	.1359E+10	-.1624E+01
20	.5289E+02	.1860E+08	-.5586E+01	-.1039E+09	-.2312E+10	.6131E+09	-.3770E+01
21	.4452E+03	.1566E+09	-.1816E+01	-.2844E+09	-.2596E+10	.1370E+10	-.1895E+01
22	.9317E+05	.3277E+11	.7922E-01	.2596E+10	.3635E+06	.0000E+01	.0000E+01

PAGE NO 3

BYRON JACKSON PUMP DIVISION
ROG-WARNER CORPORATION

TORSIONAL VIBRATION CALCULATION

PUMP SIZE AND TYPE
LARGE SCALE SODIUM PU

PURCHASER
MP **DWG IF-7937 REV A**

DATE
24 JUN 19

3 NODE FREQUENCY IS 6848.86 RPM

NO	MOM INERTIA LR-IN-SEC2	X LR-IN/RAD	RFTA RADIAN	TORQ IN-LB	SUM IN-LB	SPRG CNST LR-IN/RAD	RFL ANGLE RADIAN
1	.1640E+03	.8434E+08	.1000E+01	.8434E+08	.8434E+08	.6991E+10	.1206E-01
2	.1640E+03	.8434E+08	.9879E+00	.8333E+08	.1677E+09	.1189E+11	.1411E-01
3	.1240E+05	.6379E+10	.9738E+00	.6212E+10	.6379E+10	.3170E+10	.2013E+01
4	.1240E+05	.6379E+10	-.1039E+01	-.6626E+10	-.2464E+09	.1189E+11	-.2073E-01
5	.2020E+03	.1039E+09	-.1018E+01	-.1058E+09	-.3522E+09	.2972E+10	-.1185E+00
6	.2029E+03	.1044E+09	-.8995E+00	-.9390E+08	-.4461E+09	.1170E+11	-.3814E-01
7	.3366E+02	.1731E+08	-.8614E+00	-.1491E+08	-.4610E+09	.1909E+10	-.2415E+00
8	.1524E+03	.7837E+08	-.6199E+00	-.4858E+08	-.5096E+09	.1253E+10	-.4067E+00
9	.1042E+03	.5359E+08	-.2132E+00	-.1142E+08	-.5210E+09	.1215E+10	-.4288E+00
10	.1074E+03	.5523E+08	.2156E+00	.1191E+08	-.5091E+09	.1179E+10	-.4317E+00
11	.1090E+03	.5606E+08	.6473E+00	.3629E+08	-.4728E+09	.1179E+10	-.4009E+00
12	.1090E+03	.5606E+08	.1048E+01	.5877E+08	-.4140E+09	.1179E+10	-.3511E+00
13	.1090E+03	.5606E+08	.1399E+01	.7845E+08	-.3356E+09	.1179E+10	-.2846E+00
14	.2106E+03	.1083E+09	.1684E+01	.1824E+09	-.1531E+09	.1670E+10	-.9166E-01
15	.1906E+03	.9804E+08	.1776E+01	.1741E+09	.2096E+08	.1865E+10	.1124E-01
16	.1379E+03	.7095E+08	.1764E+01	.1252E+09	.1461E+09	.1782E+10	.8202E-01
17	.8895E+02	.4575E+08	.1682E+01	.7698E+08	.2231E+09	.1244E+10	.1794E+00
18	.1058E+03	.5441E+08	.1503E+01	.8178E+08	.3049E+09	.1302E+10	.2342E+00
19	.1058E+03	.5441E+08	.1269E+01	.6903E+08	.3739E+09	.1359E+10	.2751E+00
20	.5289E+02	.2721E+08	.9937E+00	.2703E+08	.4010E+09	.6131E+09	.6540E+00
21	.4452E+03	.2290E+09	.3397E+00	.7779E+08	.4788E+09	.1370E+10	.3495E+00
22	.9317E+05	.4792E+11	-.9813E-02	-.4703E+09	.8468E+07	.0000E+01	.0000E+01

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BYRON JACKSON PUMP DIVISION
BORG-WARNER CORPORATION

TORSIONAL VIBRATION CALCULATION

PUMP SIZE AND TYPE PURCHASER
LARGE SCALE SODIUM PU MP **DWG IF-7937 REV A**

DATE
24 JUN 19

4 NODE FREQUENCY IS 13198.10 RPM

NO	MOM INERTIA LB-IN-SQC2	X LB-IN/RAD	BETA RADIAN	TORQ IN-LB	SUM IN-LB	SPRG CNST LB-IN/RAD	RFL ANGLE RADIAN
1	.1640E+03	.3132E+09	.1000E+01	.3132E+09	.3132E+09	.6991E+10	.4480E-01
2	.1640E+03	.3132E+09	.9552E+00	.2992E+09	.6124E+09	.1189E+11	.5152E-01
3	.1240E+05	.2369E+11	.9037E+00	.2141E+11	.2202E+11	.3170E+10	.6946E+01
4	.1240E+05	.2369E+11	-.6043E+01	-.1431E+12	-.1211E+12	.1189E+11	-.1019E+02
5	.2020E+03	.3859E+09	.4147E+01	.1600E+10	-.1195E+12	.2972E+10	-.4022E+02
6	.2029E+03	.3877E+09	.4436E+02	.1720E+11	-.1023E+12	.1170E+11	-.8749E+01
7	.3366E+02	.6429E+08	.5311E+02	.3415E+10	-.9890E+11	.1909E+10	-.5180E+02
8	.1524E+03	.2910E+09	.1049E+03	.3053E+11	-.6837E+11	.1253E+10	-.5457E+02
9	.1042E+03	.1990E+09	.1595E+03	.3174E+11	-.3663E+11	.1215E+10	-.3015E+02
10	.1074E+03	.2051E+09	.1896E+03	.3890E+11	.2267E+10	.1179E+10	.1922E+01
11	.1090E+03	.2082E+09	.1877E+03	.3908E+11	.4134E+11	.1179E+10	.3506E+02
12	.1090E+03	.2082E+09	.1527E+03	.3178E+11	.7312E+11	.1179E+10	.6201E+02
13	.1090E+03	.2082E+09	.9064E+02	.1887E+11	.9199E+11	.1179E+10	.7802E+02
14	.2106E+03	.4023E+09	.1262E+02	.5079E+10	.9707E+11	.1670E+10	.5811E+02
15	.1906E+03	.3641E+09	-.4549E+02	-.1656E+11	.8051E+11	.1865E+10	.4318E+02
16	.1379E+03	.2635E+09	-.8866E+02	-.2336E+11	.5715E+11	.1782E+10	.3207E+02
17	.8895E+02	.1699E+09	-.1207E+03	-.2051E+11	.3664E+11	.1244E+10	.2946E+02
18	.1058E+03	.2021E+09	-.1502E+03	-.3035E+11	.6291E+10	.1302E+10	.4833E+01
19	.1058E+03	.2021E+09	-.1550E+03	-.3132E+11	-.2503E+11	.1359E+10	-.1841E+02
20	.5289E+02	.1010E+09	-.1366E+03	-.1380E+11	-.3883E+11	.6131E+09	-.6334E+02
21	.4452E+03	.8504E+09	-.7327E+02	-.6231E+11	-.1011E+12	.1370E+10	-.7384E+02
22	.9317E+05	.1780E+12	.5682E+00	.1011E+12	-.2377E+08	.0000E+01	.0000E+01

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BYRON JACKSON PUMP DIVISION
BORG-WARNER CORPORATION

TORSIONAL VIBRATION CALCULATION

PUMP SIZE AND TYPE
LARGE SCALE SODIUM PU

PURCHASER
MP **DWG IF-7937 REV A**

DATE
24 JUN 19

5 NODE FREQUENCY IS 17032.39 RPM

NO	MOM INERTIA LR-IN-SFC2	X LR-IN/RAD	BETA RADIAN	TORQ IN-LR	SUM IN-LR	SPRG CNST LR-IN/RAD	RFL ANGLE RADIAN
1	.1640E+03	.5216E+09	.1000E+01	.5216E+09	.5216E+09	.6991E+10	.7461E-01
2	.1640E+03	.5216E+09	.9254E+00	.4827E+09	.1004E+10	.1189E+11	.8449E-01
3	.1240E+05	.3945E+11	.8409E+00	.3317E+11	.3418E+11	.3170E+10	.1078E+02
4	.1240E+05	.3945E+11	-.9941E+01	-.3922E+12	-.3580E+12	.1189E+11	-.3012E+02
5	.2020E+03	.6428E+09	.2018E+02	.1297E+11	-.3450E+12	.2972E+10	-.1161E+03
6	.2029E+03	.6456E+09	.1363E+03	.8799E+11	-.2570E+12	.1170E+11	-.2198E+02
7	.3366E+02	.1071E+09	.1583E+03	.1695E+11	-.2401E+12	.1909E+10	-.1258E+03
8	.1524E+03	.4847E+09	.2840E+03	.1377E+12	-.1024E+12	.1253E+10	-.8176E+02
9	.1042E+03	.3314E+09	.3658E+03	.1212E+12	.1879E+11	.1215E+10	.1547E+02
10	.1074E+03	.3416E+09	.3503E+03	.1197E+12	.1385E+12	.1179E+10	.1174E+03
11	.1090E+03	.3467E+09	.2329E+03	.8075E+11	.2192E+12	.1179E+10	.1859E+03
12	.1090E+03	.3467E+09	.4700E+02	.1629E+11	.2355E+12	.1179E+10	.1997E+03
13	.1090E+03	.3467E+09	-.1527E+03	-.5295E+11	.1826E+12	.1179E+10	.1548E+03
14	.2106E+03	.6701E+09	-.3075E+03	-.2061E+12	-.2352E+11	.1670E+10	-.1408E+02
15	.1906E+03	.6063E+09	-.2935E+03	-.1779E+12	-.2015E+12	.1865E+10	-.1080E+03
16	.1379E+03	.4388E+09	-.1854E+03	-.8137E+11	-.2828E+12	.1782E+10	-.1587E+03
17	.8895E+02	.2830E+09	-.2671E+02	-.7557E+10	-.2904E+12	.1244E+10	-.2335E+03
18	.1058E+03	.3365E+09	.2068E+03	.6957E+11	-.2208E+12	.1302E+10	-.1696E+03
19	.1058E+03	.3365E+09	.3764E+03	.1267E+12	-.9415E+11	.1359E+10	-.6925E+02
20	.5289E+02	.1683E+09	.4456E+03	.7498E+11	-.1917E+11	.6131E+09	-.3126E+02
21	.4452E+03	.1416E+10	.4769E+03	.6755E+12	.6563E+12	.1370E+10	.4791E+03
22	.9317E+05	.2964E+12	-.2213E+01	-.6560E+12	.2547E+09	.0000E+01	.0000E+01

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BYRON JACKSON PUMP DIVISION
BORG-WARNER CORPORATION

TORSIONAL VIBRATION CALCULATION

PUMP SIZE AND TYPE PURCHASER
LARGE SCALE SODIUM PII MP **DWG IF-7937 REV A**

DATE
24 JUN 19

6 NODE FREQUENCY IS 21331.08 RPM

NO	MOM INERTIA LB-IN-SEC ²	X LB-IN/RAD	BETA RADIAN	TORQ IN-LB	SUM IN-LB	SPRG CNST LB-IN/RAD	RFL ANGLE RADIAN
1	.1640E+03	.8182E+09	.1000E+01	.8182E+09	.8182E+09	.6991E+10	.1170E+00
2	.1640E+03	.8182E+09	.8830E+00	.7224E+09	.1541E+10	.1189E+11	.1296E+00
3	.1240E+05	.6188E+11	.7534E+00	.4662E+11	.4816E+11	.3170E+10	.1519E+02
4	.1240E+05	.6188E+11	-.1444E+02	-.8934E+12	-.8453E+12	.1189E+11	-.7111E+02
5	.2020E+03	.1008E+10	.5667E+02	.5713E+11	-.7881E+12	.2972E+10	-.2652E+03
6	.2029E+03	.1013E+10	.3219E+03	.3259E+12	-.4622E+12	.1170E+11	-.3952E+02
7	.3366E+02	.1679E+09	.3614E+03	.6069E+11	-.4015E+12	.1909E+10	-.2103E+03
8	.1524E+03	.7602E+09	.5717E+03	.4346E+12	.3313E+11	.1253E+10	.2645E+02
9	.1042E+03	.5198E+09	.5452E+03	.2834E+12	.3166E+12	.1215E+10	.2606E+03
10	.1074E+03	.5358E+09	.2847E+03	.1525E+12	.4691E+12	.1179E+10	.3978E+03
11	.1090E+03	.5438E+09	-.1131E+03	-.6152E+11	.4076E+12	.1179E+10	.3456E+03
12	.1090E+03	.5438E+09	-.4588E+03	-.2495E+12	.1581E+12	.1179E+10	.1341E+03
13	.1090E+03	.5438E+09	-.5928E+03	-.3224E+12	-.1643E+12	.1179E+10	-.1393E+03
14	.2106E+03	.1051E+10	-.4535E+03	-.4766E+12	-.6409E+12	.1670E+10	-.3837E+03
15	.1906E+03	.9510E+09	-.6984E+02	-.6642E+11	-.7073E+12	.1865E+10	-.3793E+03
16	.1379E+03	.6882E+09	.3095E+03	.2130E+12	-.4943E+12	.1782E+10	-.2774E+03
17	.8895E+02	.4438E+09	.5869E+03	.2605E+12	-.2338E+12	.1244E+10	-.1880E+03
18	.1058E+03	.5278E+09	.7749E+03	.4090E+12	.1752E+12	.1302E+10	.1346E+03
19	.1058E+03	.5278E+09	.6404E+03	.3380E+12	.5131E+12	.1359E+10	.3775E+03
20	.5289E+02	.2639E+09	.2629E+03	.6938E+11	.5825E+12	.6131E+09	.9501E+03
21	.4452E+03	.2222E+10	-.6872E+03	-.1527E+13	-.9441E+12	.1370E+10	-.6892E+03
22	.9317E+05	.4649E+12	.2030E+01	.9436E+12	-.5881E+09	.0000E+01	.0000E+01

BYRON JACKSON PUMP DIVISION
BORG-WARNER CORPORATION

TORSIONAL VIBRATION CALCULATION

PUMP SIZE AND TYPE PURCHASER
LARGE SCALE SODIUM PU MP **DWG IF-7937 REV A**

DATE
24 JUN 19

7 NODE FREQUENCY IS 25145.48 RPM

NO	MOM INERTIA LR-IN-SEC2	X LR-IN/RAD	BETA RADIAN	TORQ IN-LR	SUM IN-LR	SPRG CNST LR-IN/RAD	RFL ANGLE RADIAN
1	.1640F+03	.1137F+10	.1000F+01	.1137F+10	.1137F+10	.6991F+10	.1626F+00
2	.1640E+03	.1137F+10	.8374F+00	.9520F+09	.2089E+10	.1189F+11	.1757F+00
3	.1240F+05	.8598E+11	.6616F+00	.5689F+11	.5898E+11	.3170E+10	.1861F+02
4	.1240F+05	.8598F+11	-.1794F+02	-.1543F+13	-.1484F+13	.1189E+11	-.1248F+03
5	.2020F+03	.1401F+10	.1069E+03	.1498F+12	-.1334F+13	.2972E+10	-.4490F+03
6	.2029F+03	.1407E+10	.5559E+03	.7822F+12	-.5520E+12	.1170E+11	-.4720E+02
7	.3366E+02	.2334E+09	.6031F+03	.1407E+12	-.4113E+12	.1909E+10	-.2154F+03
8	.1524E+03	.1056E+10	.8185E+03	.8647F+12	.4534F+12	.1253E+10	.3619F+03
9	.1042F+03	.7223F+09	.4566E+03	.3298E+12	.7832F+12	.1215E+10	.6447E+03
10	.1074F+03	.7446F+09	-.1881F+03	-.1400F+12	.6432F+12	.1179E+10	.5455E+03
11	.1090E+03	.7557E+09	-.7335E+03	-.5543E+12	.8888F+11	.1179E+10	.7538F+02
12	.1090F+03	.7557F+09	-.8089F+03	-.6113F+12	-.5224E+12	.1179E+10	-.4430E+03
13	.1090F+03	.7557E+09	-.3659F+03	-.2765F+12	-.7989E+12	.1179E+10	-.6775F+03
14	.2106E+03	.1460E+10	.3116E+03	.4551E+12	-.3438F+12	.1670E+10	-.2058E+03
15	.1906F+03	.1322E+10	.5174E+03	.6838E+12	.3400E+12	.1865F+10	.1823F+03
16	.1379F+03	.9564F+09	.3351E+03	.3205F+12	.6605E+12	.1782F+10	.3707F+03
17	.8895E+02	.6168F+09	-.3556F+02	-.2193E+11	.6385F+12	.1244E+10	.5134F+03
18	.1058F+03	.7334E+09	-.5489E+03	-.4026E+12	.2359F+12	.1302E+10	.1813F+03
19	.1058E+03	.7334F+09	-.7302E+03	-.5355E+12	-.2996E+12	.1359E+10	-.2204E+03
20	.5289F+02	.3667F+09	-.5098E+03	-.1869E+12	-.4866F+12	.6131E+09	-.7936F+03
21	.4452F+03	.3087E+10	.2838E+03	.8762F+12	.3896F+12	.1370E+10	.2844E+03
22	.9317E+05	.6460E+12	-.5972E+00	-.3858E+12	.3820F+10	.0000E+01	.0000E+01

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RYRON JACKSON PUMP DIVISION
BORG-WARNER CORPORATION

TORSIONAL VIBRATION CALCULATION

PUMP SIZE AND TYPE
LARGE SCALE SODIUM PU

PURCHASER
MP **DWG IF-7937 REV A**

DATE
24 JUN 19

R NODE FREQUENCY IS 31402.69 RPM

NO	MOM INERTIA LR-IN-SFC2	X LR-IN/RAD	RETA RADIAN	TORQ IN-LB	SUM IN-LR	SPRG CNST LR-IN/RAD	RFL ANGLE RADIAN
1	.1640E+03	.1773E+10	.1000E+01	.1773E+10	.1773E+10	.6991E+10	.2536E+00
2	.1640E+03	.1773E+10	.7464E+00	.1323E+10	.3097E+10	.1189E+11	.2605E+00
3	.1240E+05	.1341E+12	.4859E+00	.6516E+11	.6825E+11	.3170E+10	.2153E+02
4	.1240E+05	.1341E+12	-.2105E+02	-.2822E+13	-.2754E+13	.1189E+11	-.2317E+03
5	.2020E+03	.2185E+10	.2106E+03	.4602E+12	-.2294E+13	.2972E+10	-.7719E+03
6	.2029E+03	.2195E+10	.9825E+03	.2156E+13	-.1375E+12	.1170E+11	-.1176E+02
7	.3366E+02	.3640E+09	.9943E+03	.3619E+12	.2244E+12	.1909E+10	.1175E+03
8	.1524E+03	.1648E+10	.8768E+03	.1445E+13	.1669E+13	.1253E+10	.1332E+04
9	.1042E+03	.1127E+10	-.4553E+03	-.5129E+12	.1156E+13	.1215E+10	.9515E+03
10	.1074E+03	.1161E+10	-.1407E+04	-.1634E+13	-.4776E+12	.1179E+10	-.4051E+03
11	.1090E+03	.1179E+10	-.1002E+04	-.1181E+13	-.1658E+13	.1179E+10	-.1406E+04
12	.1090E+03	.1179E+10	.4045E+03	.4767E+12	-.1181E+13	.1179E+10	-.1002E+04
13	.1090E+03	.1179E+10	.1406E+04	.1658E+13	.4761E+12	.1179E+10	.4038E+03
14	.2106E+03	.2278E+10	.1003E+04	.2284E+13	.2760E+13	.1670E+10	.1652E+04
15	.1906E+03	.2061E+10	-.6495E+03	-.1339E+13	.1421E+13	.1865E+10	.7622E+03
16	.1379E+03	.1492E+10	-.1412E+04	-.2106E+13	-.6844E+12	.1782E+10	-.3841E+03
17	.8895E+02	.9619E+09	-.1029E+04	-.9885E+12	-.1673E+13	.1244E+10	-.1345E+04
18	.1058E+03	.1144E+10	.3173E+03	.3630E+12	-.1310E+13	.1302E+10	-.1006E+04
19	.1058E+03	.1144E+10	.1324E+04	.1514E+13	.2042E+12	.1359E+10	.1502E+03
20	.5289E+02	.5719E+09	.1173E+04	.6711E+12	.8753E+12	.6131E+09	.1428E+04
21	.4452E+03	.4815E+10	-.2542E+03	-.1224E+13	-.3487E+12	.1370E+10	-.2546E+03
22	.9317E+05	.1008E+13	.3281E+00	.3306E+12	-.1815E+11	.0000E+01	.0000E+01

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RYRON JACKSON PUMP DIVISION
BORG-WARNER CORPORATION

TORSIONAL VIBRATION CALCULATION

PUMP SIZE AND TYPE PURCHASER
LARGE SCALE SODIUM PU MP **DWG IF-7937 REV A**

DATE
24 JUN 19

9 NODE FREQUENCY IS 36599.29 RPM

NO	MOM INERTIA LR-IN-SEC2	X LR-IN/RAD	BETA RADIAN	TORQ IN-LR	SUM IN-LR	SPRG CNST LR-IN/RAD	REL ANGLE RADIAN
1	.1640E+03	.2409E+10	.1000E+01	.2409F+10	.2409F+10	.6991F+10	.3445F+00
2	.1640E+03	.2409E+10	.6555E+00	.1579F+10	.3987F+10	.1189F+11	.3354E+00
3	.1240F+05	.1822E+12	.3200F+00	.5830F+11	.6229F+11	.3170E+10	.1965E+02
4	.1240F+05	.1822E+12	-.1933F+02	-.3521E+13	-.3459F+13	.1189F+11	-.2910E+03
5	.2020F+03	.2968F+10	.2715E+03	.8062E+12	-.2653F+13	.2972E+10	-.8926E+03
6	.2029F+03	.2981F+10	.1164F+04	.3471F+13	.8182F+12	.1170E+11	.6996F+02
7	.3366E+02	.4944F+09	.1094E+04	.5410E+12	.1359F+13	.1909E+10	.7119E+03
8	.1524E+03	.2238F+10	.3823E+03	.8557E+12	.2215E+13	.1253E+10	.1768E+04
9	.1042F+03	.1530E+10	-.1385E+04	-.2120F+13	.9472E+11	.1215E+10	.7797F+02
10	.1074F+03	.1577F+10	-.1463F+04	-.2308F+13	-.2214E+13	.1179E+10	-.1877F+04
11	.1090E+03	.1601F+10	.4138F+03	.6625F+12	-.1551F+13	.1179F+10	-.1315F+04
12	.1090E+03	.1601E+10	.1729E+04	.2768F+13	.1217F+13	.1179F+10	.1032F+04
13	.1090E+03	.1601E+10	.6970E+03	.1116F+13	.2333E+13	.1179E+10	.1979E+04
14	.2106E+03	.3094E+10	-.1282E+04	-.3965F+13	-.1632E+13	.1670E+10	-.9769F+03
15	.1906E+03	.2800F+10	-.3046E+03	-.8528F+12	-.2485E+13	.1865E+10	-.1333E+04
16	.1379E+03	.2026F+10	.1028E+04	.2083E+13	-.4021F+12	.1782E+10	-.2257E+03
17	.8895E+02	.1307F+10	.1254F+04	.1638F+13	.1236F+13	.1244E+10	.9935E+03
18	.1058E+03	.1554E+10	.2601F+03	.4041F+12	.1640E+13	.1302F+10	.1260F+04
19	.1058E+03	.1554E+10	-.9998E+03	-.1553E+13	.8643E+11	.1359E+10	.6357F+02
20	.5289F+02	.7769E+09	-.1063E+04	-.8261E+12	-.7397F+12	.6131F+09	-.1206F+04
21	.4452E+03	.6540E+10	.1431E+03	.9358F+12	.1962E+12	.1370E+10	.1432E+03
22	.9317F+05	.1369E+13	-.1088E+00	-.1489F+12	.4725E+11	.0000F+01	.0000F+01

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BYRON JACKSON PUMP DIVISION
BORG-WARNER CORPORATION

TORSIONAL VIBRATION CALCULATION

PUMP SIZE AND TYPE
LARGE SCALE SODIUM PU

PURCHASER
MP **DWG IF-7937 REV A**

DATE
24 JUN 19

10 NODE FREQUENCY IS 43264.49 RPM

NO	MOM INERTIA LB-IN-SEC ²	X LB-IN/RAD	BETA RADIAN	TORQ IN-LB	SUM IN-LB	SPRG CNST LB-IN/RAD	RFL ANGLE RADIAN
1	.1640E+03	.3366E+10	.1000E+01	.3366E+10	.3366E+10	.6991E+10	.4814E+00
2	.1640E+03	.3366E+10	.5186E+00	.1745E+10	.5111E+10	.1189E+11	.4300E+00
3	.1240E+05	.2545E+12	.8861E-01	.2255E+11	.2766E+11	.3170E+10	.8728E+01
4	.1240E+05	.2545E+12	-.8639E+01	-.2199E+13	-.2171E+13	.1189E+11	-.1827E+03
5	.2020E+03	.4147E+10	.1740E+03	.7217E+12	-.1450E+13	.2972E+10	-.4878E+03
6	.2029E+03	.4166E+10	.6618E+03	.2757E+13	.1307E+13	.1170E+11	.1118E+03
7	.3366E+02	.6909E+09	.5500E+03	.3800E+12	.1687E+13	.1909E+10	.8839E+03
8	.1524E+03	.3127E+10	-.3338E+03	-.1044E+13	.6434E+12	.1253E+10	.5136E+03
9	.1042E+03	.2138E+10	-.8474E+03	-.1812E+13	-.1169E+13	.1215E+10	-.9619E+03
10	.1074E+03	.2204E+10	.1145E+03	.2524E+12	-.9162E+12	.1179E+10	-.7770E+03
11	.1090E+03	.2237E+10	.8915E+03	.1994E+13	.1078E+13	.1179E+10	.9143E+03
12	.1090E+03	.2237E+10	-.2279E+02	-.5098E+11	.1027E+13	.1179E+10	.8711E+03
13	.1090E+03	.2237E+10	-.8939E+03	-.2000E+13	-.9724E+12	.1179E+10	-.8247E+03
14	.2106E+03	.4323E+10	-.6916E+02	-.2990E+12	-.1271E+13	.1670E+10	-.7611E+03
15	.1906E+03	.3912E+10	.6920E+03	.2707E+13	.1436E+13	.1865E+10	.7699E+03
16	.1379E+03	.2831E+10	-.7794E+02	-.2207E+12	.1215E+13	.1782E+10	.6819E+03
17	.8895E+02	.1826E+10	-.7598E+03	-.1387E+13	-.1723E+12	.1244E+10	-.1385E+03
18	.1058E+03	.2171E+10	-.6213E+03	-.1349E+13	-.1521E+13	.1302E+10	-.1169E+04
19	.1058E+03	.2171E+10	.5474E+03	.1189E+13	-.3327E+12	.1359E+10	-.2447E+03
20	.5289E+02	.1086E+10	.7921E+03	.8600E+12	.5273E+12	.6131E+09	.8600E+03
21	.4452E+03	.9139E+10	-.6789E+02	-.6204E+12	-.9312E+11	.1370E+10	-.6798E+02
22	.9317E+05	.1912E+13	.9331E-01	.1784E+12	.8533E+11	.0000E+01	.0000E+01

BYRON JACKSON PUMP DIVISION
BORG-WARNER CORPORATION

TORSIONAL VIBRATION CALCULATION

PUMP SIZE AND TYPE PURCHASER
LARGE SCALE SODIUM PU MP **DWG TF-7937 REV A**

DATE
24 JUN 19

11 NODE FREQUENCY IS 46253.34 RPM

NO	MOM INERTIA LB-IN-SEC ²	X LB-IN/RAD	RFTA RADIAN	TORO IN-LB	SUM IN-LB	SPRG CNST LB-IN/RAD	RFL ANGLE RADIAN
1	.1640E+03	.3847E+10	.1000E+01	.3847E+10	.3847E+10	.6991E+10	.5502E+00
2	.1640E+03	.3847E+10	.4498E+00	.1730E+10	.5577E+10	.1189E+11	.4692E+00
3	.1240E+05	.2909E+12	-.1940E-01	-.5644E+10	-.6654E+08	.3170E+10	-.2099E-01
4	.1240E+05	.2909E+12	.1593E-02	.4635E+09	.3969E+09	.1189E+11	.3339E-01
5	.2020E+03	.4740E+10	-.3180E-01	-.1507E+09	.2462E+09	.2972E+10	.8285E-01
6	.2029E+03	.4761E+10	-.1146E+00	-.5459E+09	-.2997E+09	.1170E+11	-.2562E-01
7	.3366E+02	.7896E+09	-.8902E-01	-.7029E+08	-.3699E+09	.1909E+10	-.1938E+00
8	.1524E+03	.3574E+10	.1048E+00	.3744E+09	.4486E+07	.1253E+10	.3580E-02
9	.1042E+03	.2444E+10	.1012E+00	.2473E+09	.2518E+09	.1215E+10	.2072E+00
10	.1074E+03	.2519E+10	-.1060E+00	-.2672E+09	-.1540E+08	.1179E+10	-.1306E-01
11	.1090E+03	.2557E+10	-.9299E-01	-.2378E+09	-.2532E+09	.1179E+10	-.2147E+00
12	.1090E+03	.2557E+10	.1217E+00	.3112E+09	.5801E+08	.1179E+10	.4920E-01
13	.1090E+03	.2557E+10	.7250E-01	.1854E+09	.2434E+09	.1179E+10	.2064E+00
14	.2106E+03	.4941E+10	-.1339E+00	-.6617E+09	-.4183E+09	.1670E+10	-.2504E+00
15	.1906E+03	.4471E+10	.1165E+00	.5209E+09	.1026E+09	.1865E+10	.5502E-01
16	.1379E+03	.3236E+10	.6147E-01	.1989E+09	.3015E+09	.1782E+10	.1692E+00
17	.8895E+02	.2087E+10	-.1077E+00	-.2248E+09	.7667E+08	.1244E+10	.6164E-01
18	.1058E+03	.2482E+10	-.1694E+00	-.4204E+09	-.3437E+09	.1302E+10	-.2640E+00
19	.1058E+03	.2482E+10	.9465E-01	.2349E+09	-.1088E+09	.1359E+10	-.8003E-01
20	.5289E+02	.1241E+10	.1747E+00	.2167E+09	.1079E+09	.6131E+09	.1761E+00
21	.4452E+03	.1045E+11	-.1370E-02	-.1431E+08	.9363E+08	.1370E+10	.6835E-01
22	.9317E+05	.2186E+13	-.6972E-01	-.1524E+12	-.1523E+12	.0000E+01	.0000E+01

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BYRON JACKSON PUMP DIVISION
BORG-WARNER CORPORATION

TORSIONAL VIBRATION CALCULATION

PUMP SIZE AND TYPE PURCHASER
LARGE SCALE SODIUM PU MP **DWG IF-7937 REV A**

DATE
24 JUN 19

12 NODE FREQUENCY IS 46299.31 RPM

NO	MOM INERTIA LB-IN-SEC2	X LB-IN/RAD	RETA RADIAN	TORQ IN-LB	SUM IN-LB	SPRG CNST LB-IN/RAD	RFL ANGLE RADIAN
1	.1640E+03	.3854E+10	.1000E+01	.3854E+10	.3854E+10	.6991E+10	.5513E+00
2	.1640E+03	.3854E+10	.4487E+00	.1729E+10	.5584E+10	.1189E+11	.4698E+00
3	.1240E+05	.2915E+12	-.2107E-01	-.6142E+10	-.5585E+09	.3170E+10	-.1762E+00
4	.1240E+05	.2915E+12	.1551E+00	.4522E+11	.4466E+11	.1189E+11	.3757E+01
5	.2020E+03	.4749E+10	-.3602E+01	-.1711E+11	.2755E+11	.2972E+10	.9272E+01
6	.2029E+03	.4771E+10	-.1287E+02	-.6142E+11	-.3386E+11	.1170E+11	-.2896E+01
7	.3366E+02	.7912E+09	-.9978E+01	-.7894E+10	-.4176E+11	.1909E+10	-.2187E+02
8	.1524E+03	.3581E+10	.1189E+02	.4260E+11	.8424E+09	.1253E+10	.6723E+00
9	.1042E+03	.2449E+10	.1122E+02	.2748E+11	.2832E+11	.1215E+10	.2331E+02
10	.1074E+03	.2524E+10	-.1209E+02	-.3052E+11	-.2198E+10	.1179E+10	-.1864E+01
11	.1090E+03	.2562E+10	-.1023E+02	-.2620E+11	-.2840E+11	.1179E+10	-.2408E+02
12	.1090E+03	.2562E+10	.1386E+02	.3550E+11	.7100E+10	.1179E+10	.6021E+01
13	.1090E+03	.2562E+10	.7836E+01	.2007E+11	.2717E+11	.1179E+10	.2305E+02
14	.2106E+03	.4951E+10	-.1521E+02	-.7531E+11	-.4813E+11	.1670E+10	-.2881E+02
15	.1906E+03	.4480E+10	.1360E+02	.6095E+11	.1282E+11	.1865E+10	.6873E+01
16	.1379E+03	.3242E+10	.6731E+01	.2182E+11	.3464E+11	.1782E+10	.1944E+02
17	.8895E+02	.2091E+10	-.1271E+02	-.2658E+11	.8064E+10	.1244E+10	.6483E+01
18	.1058E+03	.2487E+10	-.1919E+02	-.4772E+11	-.3966E+11	.1302E+10	-.3047E+02
19	.1058E+03	.2487E+10	.1128E+02	.2804E+11	-.1162E+11	.1359E+10	-.8549E+01
20	.5289E+02	.1243E+10	.1982E+02	.2465E+11	.1303E+11	.6131E+09	.2125E+02
21	.4452E+03	.1047E+11	-.1420E+01	-.1486E+11	-.1838E+10	.1370E+10	-.1342E+01
22	.9317E+05	.2190E+13	-.7817E-01	-.1712E+12	-.1730E+12	.0000E+01	.0000E+01

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BYRON JACKSON PUMP DIVISION
RORG-WARNER CORPORATION

TORSIONAL VIBRATION CALCULATION

PUMP SIZE AND TYPE PURCHASER
LARGE SCALE SODIUM PU MP **DWG IF-7937 REV A**

DATE
24 JUN 19

13 NODE FREQUENCY IS 51086.67 RPM

NO	MOM INERTIA LB-IN-SEC ²	X LB-IN/RAD	BETA RADIAN	TORQ IN-LB	SUM IN-LB	SPRG CNST LR-IN/RAD	REL ANGLE RADIAN
1	.1640E+03	.4693E+10	.1000E+01	.4693E+10	.4693E+10	.6991E+10	.6712E+00
2	.1640E+03	.4693E+10	.3289E+00	.1543E+10	.6236E+10	.1189E+11	.5246E+00
3	.1240E+05	.3549E+12	-.1958E+00	-.6949E+11	-.6326E+11	.3170E+10	-.1996E+02
4	.1240E+05	.3549E+12	.1976E+02	.7013E+13	.6950E+13	.1189E+11	.5847E+03
5	.2020E+03	.5782E+10	-.5649E+03	-.3267E+13	.3683E+13	.2972E+10	.1239E+04
6	.2029E+03	.5808E+10	-.1804E+04	-.1048E+14	-.6797E+13	.1170E+11	-.5812E+03
7	.3366E+02	.9632E+09	-.1223E+04	-.1178E+13	-.7975E+13	.1909E+10	-.4177E+04
8	.1524E+03	.4360E+10	.2954E+04	.1288E+14	.4906E+13	.1253E+10	.3916E+04
9	.1042E+03	.2981E+10	-.9619E+03	-.2868E+13	.2038E+13	.1215E+10	.1678E+04
10	.1074E+03	.3073E+10	-.2640E+04	-.8113E+13	-.6074E+13	.1179E+10	-.5151E+04
11	.1090E+03	.3119E+10	.2511E+04	.7833E+13	.1759E+13	.1179E+10	.1492E+04
12	.1090E+03	.3119E+10	.1020E+04	.3180E+13	.4939E+13	.1179E+10	.4189E+04
13	.1090E+03	.3119E+10	-.3169E+04	-.9885E+13	-.4946E+13	.1179E+10	-.4194E+04
14	.2106E+03	.6028E+10	.1025E+04	.6179E+13	.1233E+13	.1670E+10	.7383E+03
15	.1906E+03	.5455E+10	.2867E+03	.1564E+13	.2797E+13	.1865E+10	.1500E+04
16	.1379E+03	.3948E+10	-.1213E+04	-.4790E+13	-.1993E+13	.1782E+10	-.1118E+04
17	.8895E+02	.2546E+10	-.9507E+02	-.2420E+12	-.2235E+13	.1244E+10	-.1797E+04
18	.1058E+03	.3027E+10	.1702E+04	.5151E+13	.2917E+13	.1302E+10	.2241E+04
19	.1058E+03	.3027E+10	-.5391E+03	-.1632E+13	.1285E+13	.1359E+10	.9449E+03
20	.5289E+02	.1514E+10	-.1484E+04	-.2246E+13	-.9617E+12	.6131E+09	-.1569E+04
21	.4452E+03	.1274E+11	.8453E+02	.1077E+13	.1154E+12	.1370E+10	.8424E+02
22	.9317E+05	.2666E+13	.2920E+00	.7785E+12	.8939E+12	.0000E+01	.0000E+01

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BYRON JACKSON PUMP DIVISION
BORG-WARNER CORPORATION

TORSIONAL VIBRATION CALCULATION

PUMP SIZE AND TYPE
LARGE SCALE SODIUM PU

PURCHASER
MP **DWG IF-7937 REV A**

DATE
24 JUN 19

14 NODE FREQUENCY IS 54108.37 RPM

NO	MOM INERTIA LR-IN-SEC ²	X LR-IN/RAD	RFTA RADIAN	TORQ IN-LR	SUM IN-LR	SPRG CNST LR-IN/RAD	RFL ANGLE RADIAN
1	.1640E+03	.5264F+10	.1000E+01	.5264F+10	.5264F+10	.6991E+10	.7530E+00
2	.1640E+03	.5264F+10	.2470F+00	.1300F+10	.6565F+10	.1189F+11	.5523E+00
3	.1240E+05	.3981E+12	-.3052F+00	-.1215E+12	-.1150E+12	.3170E+10	-.3627F+02
4	.1240E+05	.3981E+12	.3596F+02	.1432F+14	.1420E+14	.1189E+11	.1195F+04
5	.2020E+03	.6487F+10	-.1159F+04	-.7517F+13	.6686E+13	.2972F+10	.2250F+04
6	.2029E+03	.6516E+10	-.3409F+04	-.2221F+14	-.1552E+14	.1170E+11	-.1327F+04
7	.3366F+02	.1081F+10	-.2081E+04	-.2249F+13	-.1777E+14	.1909E+10	-.9309E+04
8	.1524E+03	.4891E+10	.7228E+04	.3536E+14	.1758E+14	.1253F+10	.1403E+05
9	.1042E+03	.3345E+10	-.6807E+04	-.2277E+14	-.5182E+13	.1215F+10	-.4265E+04
10	.1074E+03	.3447F+10	-.2542E+04	-.8762E+13	-.1394E+14	.1179E+10	-.1183E+05
11	.1090E+03	.3499E+10	.9284E+04	.3248F+14	.1854E+14	.1179F+10	.1572E+05
12	.1090E+03	.3499E+10	-.6439F+04	-.2253E+14	-.3989E+13	.1179E+10	-.3383F+04
13	.1090E+03	.3499E+10	-.3055F+04	-.1069E+14	-.1468E+14	.1179E+10	-.1245E+05
14	.2106E+03	.6762F+10	.9394E+04	.6353E+14	.4885E+14	.1670E+10	.2924E+05
15	.1906E+03	.6119E+10	-.1985E+05	-.1214E+15	-.7260E+14	.1865E+10	-.3893E+05
16	.1379E+03	.4428E+10	.1908E+05	.8452E+14	.1192E+14	.1782E+10	.6689E+04
17	.8895E+02	.2856F+10	.1240E+05	.3540E+14	.4732E+14	.1244F+10	.3804F+05
18	.1058F+03	.3396E+10	-.2565E+05	-.8710E+14	-.3978E+14	.1302E+10	-.3056F+05
19	.1058F+03	.3396E+10	.4915E+04	.1669F+14	-.2309E+14	.1359E+10	-.1698E+05
20	.5289E+02	.1698F+10	.2190F+05	.3719F+14	.1410F+14	.6131F+09	.2299E+05
21	.4452E+03	.1429E+11	-.1090E+04	-.1559E+14	-.1491E+13	.1370E+10	-.1089E+04
22	.9317F+05	.2991E+13	-.1679F+01	-.5024F+13	-.6515E+13	.0000E+01	.0000E+01

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BYRON JACKSON PUMP DIVISION
BORG-WARNER CORPORATION

TORSIONAL VIBRATION CALCULATION

PUMP SIZE AND TYPE PURCHASER
LARGE SCALE SODIUM PU MP **DWG IF-7937 REV A**

DATE
24 JUN 19

15 NODE FREQUENCY IS 57642.82 RPM

NO	MOM INERTIA LB-IN-SEC ²	X LR-IN/RAD	BETA RADIAN	TORQ IN-LR	SUM IN-LR	SPRG CNST LR-IN/RAD	PFL ANGLE RADIAN
1	.1640E+03	.5975E+10	.1000E+01	.5975E+10	.5975E+10	.6991E+10	.8546E+00
2	.1640E+03	.5975E+10	.1454E+00	.8690E+09	.6844E+10	.1189E+11	.5757E+00
3	.1240E+05	.4518E+12	-.4303E+00	-.1944E+12	-.1876E+12	.3170E+10	-.5918E+02
4	.1240E+05	.4518E+12	.5875E+02	.2654E+14	.2636E+14	.1189E+11	.2217E+04
5	.2020E+03	.7362E+10	-.2159E+04	-.1589E+14	.1047E+14	.2972E+10	.3522E+04
6	.2025E+03	.7395E+10	-.5680E+04	-.4200E+14	-.3154E+14	.1170E+11	-.2697E+04
7	.3366E+02	.1226E+10	-.2984E+04	-.3659E+13	-.3520E+14	.1909E+10	-.1844E+05
8	.1524E+03	.5551E+10	.1545E+05	.8578E+14	.5059E+14	.1253E+10	.4038E+05
9	.1042E+03	.3796E+10	-.2492E+05	-.9461E+14	-.4402E+14	.1215E+10	-.3623E+05
10	.1074E+03	.3913E+10	.1131E+05	.4425E+14	.2303E+12	.1179E+10	.1953E+03
11	.1090E+03	.3971E+10	.1111E+05	.4414E+14	.4437E+14	.1179E+10	.3763E+05
12	.1090E+03	.3971E+10	-.2651E+05	-.1053E+15	-.6091E+14	.1179E+10	-.5165E+05
13	.1090E+03	.3971E+10	.2514E+05	.9984E+14	.3893E+14	.1179E+10	.3302E+05
14	.2106E+03	.7675E+10	-.7875E+04	-.6044E+14	-.2150E+14	.1670E+10	-.1287E+05
15	.1906E+03	.6945E+10	.4998E+04	.3471E+14	.1320E+14	.1865E+10	.7081E+04
16	.1379E+03	.5026E+10	-.2083E+04	-.1047E+14	.2734E+13	.1782E+10	.1534E+04
17	.8895E+02	.3241E+10	-.3618E+04	-.1172E+14	-.8991E+13	.1244E+10	-.7228E+04
18	.1058E+03	.3854E+10	.3611E+04	.1392E+14	.4926E+13	.1302E+10	.3784E+04
19	.1058E+03	.3854E+10	-.1735E+03	-.6687E+12	.4257E+13	.1359E+10	.3131E+04
20	.5289E+02	.1927E+10	-.3305E+04	-.6369E+13	-.2112E+13	.6131E+09	-.3444E+04
21	.4452E+03	.1622E+11	.1396E+03	.2264E+13	.1521E+12	.1370E+10	.1111E+03
22	.9317E+05	.3395E+13	.2849E+02	.9672E+14	.9687E+14	.0000E+01	.0000E+01

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BYRON JACKSON PUMP DIVISION
BORG-WARNER CORPORATION

TORSIONAL VIBRATION CALCULATION

PUMP SIZE AND TYPE
LARGE SCALE SODIUM PU

PURCHASER
MP **DWG IF-7937 REV A**

DATE
24 JUN 19

16 NODE FREQUENCY IS 61926.19 RPM

NO	MOM INERTIA LR-IN-SQC2	X LR-IN/RAD	BETA RADIAN	TORQ IN-LR	SUM IN-LR	SPPG CNST LR-IN/RAD	RFL ANGLE RADIAN
1	.1640E+03	.6895E+10	.1000E+01	.6895E+10	.6895E+10	.6991E+10	.9863E+00
2	.1640E+03	.6895E+10	.1373E-01	.9467E+08	.6990E+10	.1189E+11	.5881E+00
3	.1240E+05	.5215E+12	-.5743E+00	-.2995E+12	-.2925E+12	.3170E+10	-.9228E+02
4	.1240E+05	.5215E+12	.9171E+02	.4782E+14	.4753E+14	.1189E+11	.3999E+04
5	.2020E+03	.8496E+10	-.3907E+04	-.3320E+14	.1434E+14	.2972E+10	.4824E+04
6	.2029E+03	.8534E+10	-.8731E+04	-.7452E+14	-.6018E+14	.1170E+11	-.5146E+04
7	.3366E+02	.1415E+10	-.3585E+04	-.5075E+13	-.6526E+14	.1909E+10	-.3418E+05
8	.1524E+03	.6407E+10	.3060E+05	.1960E+15	.1308E+15	.1253E+10	.1044E+06
9	.1042E+03	.4381E+10	-.7378E+05	-.3232E+15	-.1925E+15	.1215E+10	-.1584E+06
10	.1074E+03	.4516E+10	.8464E+05	.3822E+15	.1897E+15	.1179E+10	.1609E+06
11	.1090E+03	.4583E+10	-.7626E+05	-.3495E+15	-.1598E+15	.1179E+10	-.1355E+06
12	.1090E+03	.4583E+10	.5925E+05	.2715E+15	.1118E+15	.1179E+10	.9478E+05
13	.1090E+03	.4583E+10	-.3553E+05	-.1628E+15	-.5106E+14	.1179E+10	-.4330E+05
14	.2106E+03	.8858E+10	.7778E+04	.6890E+14	.1783E+14	.1670E+10	.1067E+05
15	.1906E+03	.8015E+10	-.2897E+04	-.2322E+14	-.5385E+13	.1865E+10	-.2888E+04
16	.1379E+03	.5801E+10	-.9003E+01	-.5222E+11	-.5437E+13	.1782E+10	-.3051E+04
17	.8895E+02	.3741E+10	.3042E+04	.1138E+14	.5943E+13	.1244E+10	.4778E+04
18	.1058E+03	.4448E+10	-.1736E+04	-.7721E+13	-.1778E+13	.1302E+10	-.1366E+04
19	.1058E+03	.4448E+10	-.3696E+03	-.1644E+13	-.3422E+13	.1359E+10	-.2517E+04
20	.5289E+02	.2224E+10	.2148E+04	.4777E+13	.1355E+13	.6131E+09	.2209E+04
21	.4452E+03	.1872E+11	-.6167E+02	-.1155E+13	.2000E+12	.1370E+10	.1460E+03
22	.9317E+05	.3918E+13	-.2077E+03	-.8138E+15	-.8136E+15	.0000E+01	.0000E+01

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BYRON JACKSON PUMP DIVISION
BORG-WARNER CORPORATION

TORSIONAL VIBRATION CALCULATION

PUMP SIZE AND TYPE PURCHASER
LARGE SCALE SODIUM PU MP **DWG IF-7937 REV A**

DATE
24 JUN 19

17 NODE FREQUENCY IS 67367.82 RPM

NO	MOM INERTIA LR-IN-SEC2	X LR-IN/RAD	BETA RADIAN	TORQ IN-LB	SUM IN-LB	SPRG CNST LR-IN/RAD	REF ANGLE RADIAN
1	.1640E+03	.8161E+10	.1000E+01	.8161E+10	.8161E+10	.6991E+10	.1167E+01
2	.1640E+03	.8161E+10	-.1672E+00	-.1365E+10	.6796E+10	.1189E+11	.5717E+00
3	.1240E+05	.6172E+12	-.7389E+00	-.4560E+12	-.4493E+12	.3170E+10	-.1417E+03
4	.1240E+05	.6172E+12	.1410E+03	.8701E+14	.8656E+14	.1189E+11	.7282E+04
5	.2020E+03	.1006E+11	-.7141E+04	-.7181E+14	.1475E+14	.2972E+10	.4965E+04
6	.2029E+03	.1010E+11	-.1211E+05	-.1223E+15	-.1075E+15	.1170E+11	-.9194E+04
7	.3366E+02	.1675E+10	-.2912E+04	-.4879E+13	-.1124E+15	.1909E+10	-.5888E+05
8	.1524E+03	.7583E+10	.5596E+05	.4244E+15	.3120E+15	.1253E+10	.2490E+06
9	.1042E+03	.5185E+10	-.1930E+06	-.1001E+16	-.6888E+15	.1215E+10	-.5670E+06
10	.1074E+03	.5344E+10	.3740E+06	.1998E+16	.1310E+16	.1179E+10	.1111E+07
11	.1090E+03	.5424E+10	-.7367E+06	-.3996E+16	-.2686E+16	.1179E+10	-.2278E+07
12	.1090E+03	.5424E+10	.1541E+07	.8360E+16	.5674E+16	.1179E+10	.4812E+07
13	.1090E+03	.5424E+10	-.3270E+07	-.1774E+17	-.1206E+17	.1179E+10	-.1023E+08
14	.2106E+03	.1048E+11	.6961E+07	.7297E+17	.6091E+17	.1670E+10	.3646E+08
15	.1906E+03	.9486E+10	-.2950E+08	-.2798E+18	-.2189E+18	.1865E+10	-.1174E+09
16	.1379E+03	.6865E+10	.8790E+08	.6034E+18	.3845E+18	.1782E+10	.2158E+09
17	.8895E+02	.4427E+10	-.1279E+09	-.5661E+18	-.1816E+18	.1244E+10	-.1460E+09
18	.1058E+03	.5264E+10	.1814E+08	.9550E+17	-.8612E+17	.1302E+10	-.6616E+08
19	.1058E+03	.5264E+10	.8430E+08	.4438E+18	.3577E+18	.1359E+10	.2631E+09
20	.5289E+02	.2632E+10	-.1788E+09	-.4706E+18	-.1130E+18	.6131E+09	-.1842E+09
21	.4452E+03	.2216E+11	.5433E+07	.1204E+18	.7430E+16	.1370E+10	.5424E+07
22	.9317E+05	.4637E+13	.8933E+04	.4142E+17	.4885E+17	.0000E+01	.0000E+01

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BYRON JACKSON PUMP DIVISION
HORG-WARNER CORPORATION

TORSIONAL VIBRATION CALCULATION

PUMP SIZE AND TYPE PURCHASER
LARGE SCALE SODIUM PU MO **DWG TF-7937 REV A**

DATE
24 JUN 19

1R NODE FREQUENCY IS 70677.35 RPM

NO	MOM INERTIA LR-IN-SEC ²	X LR-IN/RAD	BETA RADIAN	TORQ IN-LR	SUM IN-LR	SPRG CNST LR-IN/RAD	REL ANGLE RADIAN
1	.1640E+03	.8982E+10	.1000E+01	.8982E+10	.8982E+10	.6991E+10	.1285E+01
2	.1640E+03	.8982E+10	-.2847E+00	-.2557E+10	.6425E+10	.1189E+11	.5405E+00
3	.1240E+05	.6793E+12	-.8252E+00	-.5606E+12	-.5541E+12	.3170E+10	-.1748E+03
4	.1240E+05	.6793E+12	.1740E+03	.1182E+15	.1176E+15	.1189E+11	.9896E+04
5	.2020E+03	.1107E+11	-.9722E+04	-.1076E+15	.1003E+14	.2972E+10	.3376E+04
6	.2029E+03	.1112E+11	-.1310E+05	-.1456E+15	-.1356E+15	.1170E+11	-.1159E+05
7	.3366E+02	.1844E+10	-.1505E+04	-.2775E+13	-.1384E+15	.1909E+10	-.7247E+05
8	.1524E+03	.8346E+10	.7097E+05	.5923E+15	.4539E+15	.1253E+10	.3623E+06
9	.1042E+03	.5707E+10	-.2913E+06	-.1663E+16	-.1209E+16	.1215E+10	-.9948E+06
10	.1074E+03	.5882E+10	.7035E+06	.4138E+16	.2929E+16	.1179E+10	.2484E+07
11	.1090E+03	.5970E+10	-.1781E+07	-.1063E+17	-.7702E+16	.1179E+10	-.6532E+07
12	.1090E+03	.5970E+10	.4751E+07	.2836E+17	.2066E+17	.1179E+10	.1752E+08
13	.1090E+03	.5970E+10	-.1277E+08	-.7624E+17	-.5558E+17	.1179E+10	-.4713E+08
14	.2106E+03	.1154E+11	.3436E+08	.3965E+18	.3409E+18	.1670E+10	.2041E+09
15	.1906E+03	.1044E+11	-.1697E+09	-.1772E+19	-.1431E+19	.1865E+10	-.7674E+09
16	.1379E+03	.7556E+10	.5977E+09	.4516E+19	.3085E+19	.1782E+10	.1731E+10
17	.8895E+02	.4873E+10	-.1134E+10	-.5524E+19	-.2439E+19	.1244E+10	-.1961E+10
18	.1058E+03	.5794E+10	.8270E+09	.4792E+19	.2353E+19	.1302E+10	.1808E+10
19	.1058E+03	.5794E+10	-.9810E+09	-.5684E+19	-.3331E+19	.1359E+10	-.2450E+10
20	.5289E+02	.2897E+10	.1469E+10	.4256E+19	.9253E+18	.6131E+09	.1509E+10
21	.4452E+03	.2439E+11	-.4018E+08	-.9800E+18	-.5471E+17	.1370E+10	-.3994E+08
22	.9317E+05	.5104E+13	-.2476E+06	-.1264E+19	-.1318E+19	.0000E+01	.0000E+01

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BYRON JACKSON PUMP DIVISION
BORG-WARNER CORPORATION

TORSIONAL VIBRATION CALCULATION

PUMP SIZE AND TYPE
LARGE SCALE SODIUM PU

PURCHASER
MP **DWG IF-7937 REV A**

DATE
24 JUN 19

19 NODE FREQUENCY IS 84135.73 RPM

NO	MOM INERTIA LR-IN-SEC2	X LR-IN/RAD	RETA RADIAN	TORQ IN-LR	SUM IN-LR	SPRG CNST LR-IN/RAD	REL ANGLE RADIAN
1	.1640E+03	.1273E+11	.1000E+01	.1273E+11	.1273E+11	.6991E+10	.1821E+01
2	.1640E+03	.1273E+11	-.8206E+00	-.1044E+11	.2284E+10	.1189E+11	.1921E+00
3	.1240E+05	.9626E+12	-.1013E+01	-.9748E+12	-.9726E+12	.3170E+10	-.3068E+03
4	.1240E+05	.9626E+12	.3058E+03	.2944E+15	.2934E+15	.1189E+11	.2468E+05
5	.2020E+03	.1568E+11	-.2438E+05	-.3823E+15	-.8893E+14	.2972E+10	-.2993E+05
6	.2029E+03	.1575E+11	.5548E+04	.8741E+14	-.1523E+13	.1170E+11	-.1303E+03
7	.3366E+02	.2613E+10	.5679E+04	.1484E+14	.1331E+14	.1909E+10	.6973E+04
8	.1524E+03	.1183E+11	-.1295E+04	-.1531E+14	-.1999E+13	.1253E+10	-.1595E+04
9	.1042E+03	.8087E+10	.3007E+03	.2432E+13	.4330E+12	.1215E+10	.3564E+03
10	.1074E+03	.8336E+10	-.5570E+02	-.4643E+12	-.3128E+11	.1179E+10	-.2653E+02
11	.1090E+03	.8460E+10	-.2917E+02	-.2468E+12	-.2780E+12	.1179E+10	-.2358E+03
12	.1090E+03	.8460E+10	.2066E+03	.1748E+13	.1470E+13	.1179E+10	.1247E+04
13	.1090E+03	.8460E+10	-.1040E+04	-.8799E+13	-.7329E+13	.1179E+10	-.6215E+04
14	.2106E+03	.1635E+11	.5175E+04	.8461E+14	.7729E+14	.1670E+10	.4627E+05
15	.1906E+03	.1480E+11	-.4109E+05	-.6079E+15	-.5307E+15	.1865E+10	-.2846E+06
16	.1379E+03	.1071E+11	.2435E+06	.2607E+16	.2076E+16	.1782E+10	.1165E+07
17	.8895E+02	.6905E+10	-.9218E+06	-.6365E+16	-.4289E+16	.1244E+10	-.3448E+07
18	.1058E+03	.8211E+10	.2526E+07	.2074E+17	.1645E+17	.1302E+10	.1264E+08
19	.1058E+03	.8211E+10	-.1011E+08	-.8305E+17	-.6660E+17	.1359E+10	-.4899E+08
20	.5289E+02	.4106E+10	.3887E+08	.1596E+18	.9300E+17	.6131E+09	.1517E+09
21	.4452E+03	.3456E+11	-.1128E+09	-.3899E+19	-.3806E+19	.1370E+10	-.2779E+10
22	.9317E+05	.7232E+13	.2666E+10	.1928E+23	.1928E+23	.0000E+01	.0000E+01

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COMPUTATION HAS BEEN TERMINATED AT UPPER LIMIT OF 100000. RPM

BYRON JACKSON REPORT TCF-1016-DYN
LATERAL MODAL CHARACTERISTICS
of
LARGE SCALE LMFBR PUMP ASSEMBLY

PREPARED FOR

GENERAL ELECTRIC COMPANY
FAST BREEDER REACTOR DEPARTMENT
AEC CONTRACT AT (04-3) 893

PREPARED BY Yoshio Usui DATE 27 June 76
Harry B. Boswell DATE 27 JUN 76
APPROVED BY Kent A. Huber DATE 28 JUN 76
APPROVED BY R. T. Da C. DATE 28 JUN 76

June 1976

BYRON JACKSON PUMP DIVISION
BORG WARNER CORPORATION
LOS ANGELES, CALIFORNIA

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1.0 SCOPE

The purpose of this study is to identify potential areas of concern in the large scale LMFBR cold leg pump conceptual design of Reference 2.1. Specifically, the reported analyses determine whether lateral modal characteristics and seismic response of the rotating element and of the assembled pump unit meet the requirements of Reference 2.2.

Details of the foundation, attached piping, and driver were not available. Consequently, these analyses are based either on "typical" dimensions and weights, approximate the effect of the component with conservative mass and spring data, or use information available from the Clinch River Breeder Reactor hot leg pump. (Reference 2.10)

As required, the analysis investigates design modifications required to achieve the desired natural frequencies.

2.0 REFERENCES

- 2.1 Byron Jackson Engineering Drawing 1F-7937, Revision A.
- 2.2 General Electric Specification 22A3994, Revision 2.
- 2.3 "Structural Response of CRBRP Primary and Intermediate Prototype Pumps", J. Chrostowski and L. T. Lee, J. H. Wiggins Company, Technical Report No. 75-1225-3.
- 2.4 Nuclear Systems Materials Handbook, Volume 1, "Design Data", Hanford Engineering Development Lab. TID 26666.
- 2.5 "SAP IV, A Structural Analysis Program for Static and Dynamic Response of Linear Systems", Earthquake Engineering Research Center, College of Engineering, University of California, Report No. EERC 73-11.
- 2.6 "Analysis of the Deformation Behavior of a Pump Tank Structure", Universal Analysis, Inc. July 15, 1974
- 2.7 "Lateral Modal Characteristics of CRBRP Pump Assembly", Y. Usui, Byron Jackson Report No. TCF-TM-1025.
- 2.8 "Design Analysis of Shafts and Beams" by R. B. Hopkins, McGraw-Hill, 1970.
- 2.9 MTI letter report dated 5 December 1975 to J. Ogg "Preliminary Results of Analyses of 24" x 17" and 16" x 12" Bearings".

3.0 DESCRIPTION OF LMFBR PUMP

The pump is comprised of six major subassemblies, the motor rotor, the pump shaft, the motor housing, the motor mount, the internal structure (includes radiation shield, hydrostatic bearing and hydraulic assembly), and the outer tank. Figure 3.1(a) identifies these subassemblies.

The outer tank is rigidly mounted to the foundation. The motor, motor mount, and pump internals are assembled and lowered into place as a complete unit. Attachment between the two structures consists of a series of jack-screws used to obtain a prescribed vertical placement of the pump internal assembly and a series of hold-down bolts.

For analysis purposes, the various components, including the hydraulic assembly, are represented by collinear shear beams with thin walled circular cross sections. Figure 3.1(b) is a simplified description of the collinear beam model used to represent the LMFBR pump assembly for the dynamic analysis.

IDEALIZATION OF LMFR PUMP

FIGURE 3.1

4.0 STRUCTURAL MODELING

4.1 General Modeling Considerations

The modal characteristics of the LMFBR pump were determined using the colinear shear beam model shown in Figure 4.1.

The analysis uses the SAP IV Structural Analysis Program, Reference 2.5, and accounts for the following effects:

- Shear deformations,
- Temperature dependent material properties
- Mass of entrained sodium, and
- Mass of attached piping.

The basic assumptions for the analysis are:

- The pump structure is axisymmetric
- Tapered portions of the structure are represented by graduated cylindrical sections,
- Only planar motion is considered (i.e., torsional motion is not allowed),
- Attached piping has zero stiffness,
- The steel balls comprising the radiation shield behave as a lumped mass,
- The shaft bearings can be treated as linear elastic springs,
- The foundation can be represented by a set of linear springs,
- No shell deformation occurs,
- Bolted joints in the shaft and inner structure components are rigid.

Figure 4.1 LMFBF - Colinear Shear Beam Model

- The mass moment of inertia can be neglected.
- Gyroscopic effects in the rotating element can be neglected, and,
- Dynamic coupling due to the sodium between the shaft, inner structure, and tank is neglected.

The effect of lateral sloshing of the liquid sodium was not considered in the present analysis. However, Reference 2.3 presents an analysis of this effect using an equivalent mechanical system for a prototype pump of similar design. The results predict the slosh frequency to occur well below the pump operating range.

Figure 4.1 shows the location of the lumped masses (nodes) used to represent the pump structure. The node locations were selected to provide a proper mass and stiffness representation along the length of the pump shaft, inner structure, and tank. The nodes were also placed to provide enough elements to define a linear temperature distribution in material located above the liquid sodium level. The origin of the coordinate system used to define the nodes is located at an elevation of -51'2" (614" below the foundation elevation of 0').

Due to the collinear model, there is no coupling between the lateral and vertical degrees of freedom. Therefore the analysis of the lateral pump modal characteristics is considered separate from the vertical by defining only a lateral displacement and an in-plane rotation at each node.

The mass and stiffness properties are calculated for each element defined between the nodes shown in Figure 4.1. For circular cylinders with inside and outside diameters of D_i and D_o , respectively, these calculations are performed by the proprietary Byron Jackson computer program INSAP using the following equations from Reference 2.8:

1. Cross Sectional Area, A ;

$$A = \frac{\pi}{4} (D_o^2 - D_i^2)$$

2. Shear coefficient, K_s :

$$K_s = 6(1+\nu)[1 + (D_i/D_o)^2]^2 / \{ (7+6\nu)[1 + (D_i/D_o)^2]^2 + (20 + 12\nu)(D_i/D_o)^2 \}$$

where Poisson's ratio, ν , is assumed to be 0.3.

3. Shear Area, A_S ;

$$A_S = AK_S$$

4. Weight, W ;

$$W = (\text{Length} * \text{density} * A) + \text{concentrated load}$$

5. Modulus of Elasticity, E ;

$$E = E(T_{ave})$$

where T_{ave} is temperature at center of element.

Unfortunately, the program cannot process complex geometries and the shear beam properties for these components must be calculated manually. These calculations may be found in Appendix A.

The modulus of elasticity is determined as a function of temperature from Reference 2.4. A computer printout of the complete model may be found in Appendix D.

4.2.1 Pump Shaft

With the exception of the impellers and hydrostatic bearings, the mass and stiffness properties of the pump shaft can be calculated by program INSAP.

Each impeller was modeled as a concentrated load acting on a shaft section of 19 inch O.D. and 15 inch I.D. Weight of the axisymmetric portions of the impeller were determined using the IMPWT Program (See Appendix A). The non-symmetric vanes were assumed to contribute an additional weight of fifteen percent.

The total weight was found to be 7,792 lbs. This value was increased by 1,422 lbs to account for the entrained sodium. The modeled shaft section contributes 1,107 lbs of this total weight. Thus, the equivalent concentrated load acting at the impeller centerline is 8,106 lbs.

Figure 4.2 presents the parameters of the shear beams used to represent the pump shaft, impellers, and hydrostatic bearings.

4.2.2 Hydraulic Assembly

The complex geometry of the hydraulic assembly was represented as a uniform cylindrical section with an inside diameter of 70 inches and a constant thickness of 2 inches. The weight of the volutes, suction guide pieces, and discharge manifold (82,189 lbs) and the appropriate sodium volumes (10,990 lbs) were calculated manually (See Appendix A) and added to the model as concentrated loads. This simplified shear beam model is consistent with the method of construction. Also, the proper distribution of mass at the tip of a cantilever type structure and the stiffness characteristics at the base are of primary interest in modeling. This is because during vibrations most of the strain energy ($U = \frac{1}{2}k\Delta^2$) will be stored at the base of the structure while the tip will contribute the largest portion of the kinetic energy ($K.E. = \frac{1}{2}Mv^2$) in the system.

4.2.3 Internal Support Structure

The structural members supporting the hydraulic assembly exhibit, with two exceptions, an axis of symmetry and the equivalent shear

PUMP SHAFT
FIGURE 4.2

INNER STRUCTURE
FIGURE 4.4

OUTER TANK
FIGURE 4.5

beam properties may be calculated using the INSAP program. The first of the two exceptions is the discharge elbow which contributes weight to the inner structure but provides insignificant stiffness.

The second non-symmetric area occurs at the discharge opening in the outer cylinder. Figure 4.3 shows a planar view of this section. The moments of inertia about the two horizontal axes are different. For conservatism, the moment of inertia used to represent this region assumed rotation about the yy-axis and

Figure 4.3 Planar View of Discharge Opening in Inner Support Structure

was calculated using the equation;

$$I = R^3 t (\theta + \sin\theta \cos\theta - 2 \sin^2\theta/\theta)$$

where $\theta = \pi - \alpha$ (Reference 2.8).

A similar calculation is performed to determine the moment of inertia of the outer tank in this region. Both calculations are shown in Appendix A.

In the calculation of these equivalent beam properties the weight of the liquid sodium between the shaft and the internal structure was added to the metal weight of the structure. An additional weight was also added to account for the shot contained in the radiation shield plug. The density of the shot was taken to be 283 lbs. per cubic foot (0.1638 lbs. per cubic inch).

Figure 4.4 shows the equivalent beam properties used to represent the hydraulic assembly and internal support structures for the modal analysis.

4.2.4 Outer Tank

Figure 4.5 shows the equivalent shear beam model used to represent the outer tank. It is the longest of the three components which extend below the foundation. Consequently, it determines the minimum elevation of the colinear beam model.

The hemispherical portion at the bottom of the tank is represented by a series of thin walled cylindrical sections of graduated diameters.

The weight of sodium between the internal support structure and the outer tank is assumed to be added to the weight of the outer tank. A similar assumption is made in the hydraulic assembly region.

4.2.5 Motor Mount

The motor mount was modeled as a conical section without access windows. The actual structure will thus have a smaller moment of inertia for the same geometry. This representation is considered adequate since the only load acting on the structure is due to the driver weight, and the geometry can be readily modified in the Detailed Design Phase to provide the required stiffness.

The equivalent beam properties for the motor mount are shown in Figure 4.4.

4.3 Motor Components

The pump driver will be an 18,000 horsepower, 6600 volt unit running at 600 RPM. Total weight is estimated to be 185 Kips.

4.3.1 Motor Shaft

Detail drawings of the motor rotor were not available. Consequently a "typical" motor was constructed for analysis purposes. This mathematical model was based on estimates of significant rotor characteristics. These data are:

Rotor weight	60 Kips
Coupling shaft diameter	14 inches
Bearing span	12 feet
Natural Frequency	25 cps (approx.)

An equivalent diameter was calculated to represent the stiffness of the length of shafting beneath the motor laminations. This was accomplished by factoring the 14 inch coupling shaft diameter by the ratio of equivalent shaft diameter to coupling shaft diameter obtained from an 8000 horsepower motor. The calculated equivalent diameter of 19 inches is shown in Figure 4.2 which illustrates the driver constructed for the analysis.

4.3.2 Motor Housing

Detailed engineering drawings of the motor housing were not available at the time the modal analysis was performed. Consequently, a mass-elastic model was developed based on the following criteria:

1. The fundamental lateral mode of the motor housing was to be equal to or slightly greater than 30 Hz (on a fixed base).
2. The estimated weight of approximately 125,000 pounds was to be distributed uniformly.
3. The overall geometric dimensions of length and diameter were 15 feet and 12 feet, respectively.

It was found that the stiffness of the motor housing could be represented by a shear beam of 144 inches outside diameter and one inch thickness. These parameters were used in the present analysis.

Hydrostatic bearing stiffness parameters were obtained from Reference 2.9 which determines the stiffness range to be $0.74 \times 10^6 \leq K \leq 1.35 \times 10^6$ lbs/in. These calculations were based on a bearing with 24" OD and 17" length and with suction pressure at both ends. Thus these data are directly applicable to the lower hydrostatic bearing in the conceptual design and a mean value of $K(\text{lower hydrostatic}) = 1.00 \times 10^6$ lbs/in was used.

The middle and upper hydrostatic bearings, on the other hand, have suction pressure at one end and discharge pressure at the other. As a result, these bearings are only 50 percent efficient and the stiffness was assumed to be

$$K_{(\text{upper hydrostatic})} = 0.49 \times 10^6 \text{ lbs/in}$$

$$K_{(\text{middle hydrostatic})} = 0.50 \times 10^6 \text{ lbs/in}$$

The additional decrease in stiffness for the upper bearing results from the flexibility of the bearing support arrangement as calculated in Appendix B.

4.5.2 Inner Structure

The "inner structure" beam is comprised of the hydraulic assembly, internal support structure, motor mount, and motor housing. It is connected to the outer tank at the foundation by a jack-screw arrangement which permits raising and lowering of the structure to control leakage at the discharge nozzle location.

The connection to the outer tank at the foundation level is assumed to be pinned in the lateral direction and a rotational spring is used to represent the bolting flexibility.

The analysis also assumes that a rigid connection exists between the inner structure and outer tank in the area of the discharge nozzle. This is based on the assumption that the pressure differential across the inner structure in the area of the discharge causes it to expand and make metal-to-metal contact with the outer tank. It is also assumed that this contact occurs for 270° of the circumference of the tank which is verified by Reference 2.6.

The stiffness assumed between the inner structure and the outer tank was:

$$\begin{aligned}K_{\text{horiz}}(\text{foundation}) &= 1 \times 10^{10} \text{ lbs/in} \\K_{\text{rot}}(\text{foundation}) &= 3.41 \times 10^{12} \text{ in-lb/rad.} \\K_{\text{horiz}}(\text{nozzle}) &= 1 \times 10^{10} \text{ lbs/in}\end{aligned}$$

4.5.3 Outer Tank

Two boundary conditions were studied between the tank and the foundation. The first assumes a completely rigid support. The second investigates the effect of a compliant foundation on the modal characteristic of the structure. The latter analysis assumes the following stiffness between the outer tank and foundation:

$$K_{\text{horiz}} = 1.59 \times 10^8 \text{ lbs/in}$$

$$K_{\text{rot}} = 7.66 \times 10^{12} \text{ in-lb/rad}$$

The structure is also completely restrained against vertical motion.

5.0 MODAL CHARACTERISTICS

5.1 Lateral Characteristics

A series of computer runs were made to optimize the fundamental frequencies of the individual components and of the entire assembled unit. Table 5.1 summarizes the major results of the analysis performed on the conceptual design of Reference 2.1 Appendix D contains the computer output for each run. Results of analyses of other configurations are not presented in this report but are available for review at the Byron Jackson Engineering offices.

Table 5.1 Summary of Lateral Modal Characteristics

Run	Structure	Boundary Conditions	File No.	Frequency of Modes (cps)			
				1	2	3	4
1	Motor Shaft Only	Pinned Supports	21	28.05	105.5	212.0	287.5
2	Shaft	Pinned Supports	22	25.21	29.86	39.24	47.75
3	Shaft	Spring Supports	20	13.02	17.91	20.91	23.37
4	Pump Assembly	Rigid Foundation without Snubber	34	7.04	12.54	15.32	19.39
5	Pump Assembly	Compliant Foundation without Snubber	29	6.80	12.47	15.32	19.39
6	Pump Assembly	Rigid Foundation with Rigid Snubber	24	12.54	12.94	15.82	19.43
7	Pump Assembly	Compliant Foundation with Rigid Snubber	23	12.44	12.82	15.77	19.42
8	Pump Assembly	Compliant Foundation with Spring Snubber	26	11.35	12.47	15.54	19.41
9	Pump Assembly	CRBRP Foundation with Spring Snubber	30	11.15	12.28	15.52	19.39

Runs 8 and 9 use a stiffness for the snubber of 5×10^7 lbs/in. Also the runs which have a compliant foundation use the values listed in Section 4.5.3 except for run Number 9 which uses the following stiffness:

$$\begin{aligned}K_{\text{horiz}} &= 69.5 \times 10^6 \text{ lbs/in} \\K_{\text{rot}} &= 0.5 \times 10^{12} \text{ in-lb/rad}\end{aligned}$$

These values are the same values used in Reference 2.7 and were included to investigate the effect of decreasing the foundation stiffness.

Plots of the mode shapes for each of the runs described in Table 5.1 may be found as Figures 5.1 through 5.11 and in Appendix E.

5.2 Vertical Characteristics

The vertical modal characteristics were calculated separately from the lateral modes and the results are given below. The calculations are presented in Appendix C.

Mode	Natural Vertical Frequency
1	65.64
2	276.80
3	430.94

RRFD*7CIG0002*WHOLE SHAFT RIGID*P22

07-06-76 12:46:23

MODE 1

25.71 CPS

BRPD*761G0002*R20*WHOLE SHAFT SOFT

07-06-76 12:49:18

MODE 1

13.02 CPS

Figure 5.3 Shaft on Five Flexible Bearings
Second Mode Deflected Shape

BRPD*761G0002*R20*WHOLE SHAFT SOFT

07-06-76 12:49:18

MODE 2

17.91 CPS

BRPD*761G0002*R29*COMP FOUND*REV A*SNUBER K=0

07-06-76 21: 5:57

MODE 1

6.80 CPS

BRPD*761G0002*R29*COMP FOUND*REV A*SNUBBER K=0

07-06-76 21: 5:57

MODE 2

12.47 CPS

BRPD*761G0002*R29*COMP FOUND*REV A*SNUBBER K=0

07-06-70 21: 5:57

MODE 3

15.32 CPS

Figure 5.7 Assembled Unit on Complaint Foundation. Fourth Mode Deflected Shape

BRPD*761G0002*R29*COMP FOUND*REV A*SNUBBER K=0

07-06-76 21: 5:57

MODE 4

19.39 CPS

BRPD*761G0002*R26*COMP FOUND*REV A*SNUBBER K=5E7

07-06-76 21: 2:39

MODE 1

11.35 CPS

BRPD*761G0002*R26*COMP FOUND*REV A*SNUBBER K=5E7

07-06-76 21: 2:39

MODE 2

12.47 CPS

BRPD*761G0002*R26*COMP FOUND*REV A*SNUBBER K=5E7

07-06-76 21: 2:39

MODE 3

15.54 CPS

BRPD*761G0002*R26*COMP FOUND*REV A*SNUBBER K=5E7

07-06-76 21: 2:39

MODE 4

19.41 CPS

6.0 DISCUSSION OF LATERAL MODE RESULTS

The results presented in Section 5.0 are all based on the assumption that bolted-flanged connections immersed in liquid sodium remain rigid. This assumption is considered reasonable since the operating temperature is 850°F. where creep strain is negligible and there are no significant holding periods at higher temperatures.

6.1 Shaft Frequencies:

The fundamental frequency of the motor shaft on rigid bearings is 28.10 cps. This value is reasonably close to the target value of 25 cps desired for the "typical" motor.

The first mode frequency for the combined pump and motor shaft is 25.21 cps on rigid bearings (See Figure 5.1). When the flexible bearings of Section 4.5.1 are used to support the shaft, this value becomes 13.02 cps. The second mode frequency is 17.91 cps. Mode shape plots for these frequencies are shown in Figures 5.2 and 5.3.

Inspection of Figure 5.2 shows the lowest frequency is a resonance in the motor portion of the shaft. This resonance is due to the stiffness values assumed for the motor rotor bearings. Although this frequency is greater than the 125% specification requirement, significant improvement would result by using stiffer bearings.

Inasmuch as no information could be obtained from the driver vendor, this optimization of motor bearings was not attempted. The frequency obtained should thus be considered a lower bound.

The second shaft mode is a resonance of the pump shaft between the upper and lower hydrostatic bearing (See Figure 5.3. From the deflected shape, it is seen that this mode loads the middle hydrostatic bearing.

6.2 Structural Frequencies

The results of Run 5, Table 5.1, show the lowest structural frequencies of the assembled unit to be 6.80 cps and 15.32 cps. Modes 2 and 4 are motor shaft and pump shaft resonances, respectively. This value is below the operating speed of 10 cps and does not meet the Reference 2.2 requirements. The upper bound of the lowest frequency of a cantilever structure is obtained with a rigid connection at the base. For the pump structure this upper bound was found to be 7.04 cps which was still less than the requirements of Reference 2.2

To satisfy the design requirements, it was thus necessary to consider both seismic restraints and design modifications.

6.2.1 Seismic Restraints

Attachment of seismic restraints were considered at three locations, at the top of the motor, to the heavy-walled section of the tank at the discharge nozzle location, and to the bottom of the tank.

Inspection of Figure 5.4 for the lowest structural mode shows an in-phase resonance of the inner structure and the pump tank. Very little deflection occurs at the top of the motor. Such a result

can be expected when one considers the relative weight and center of gravity locations of the tank and motor structures. As a result, locating a seismic restraint at the top of the motor would be ineffectual in increasing the lowest frequency.

On the other hand, locating a seismic restraint at either the suction or discharge nozzle will aid in decreasing the deflection of the tank with a resulting increase in frequency. Table 6.1 summarizes the predictions for a rigid seismic restraint placed at each of these locations.

Table 6.1 Effect of Seismic Restraint Location

Location	File No.	First Mode Frequency
Motor	N/A	Ineffectual
Discharge Nozzle	24	12.54 cps
Suction Nozzle	9	10.82 cps

Since the discharge nozzle location provides the greatest improvement and the attachment to the tank may be achieved with ease and with minimum regard to the thermal transient conditions, the suction nozzle location was not investigated further.

The minimum required stiffness for the seismic restraint at the discharge nozzle location was next determined by plotting the first mode frequency as a function of restraint stiffness. This plot is presented as Figure 6.1 and indicates a minimum spring rate of 5×10^7 lbs per inch is required for the seismic restraint.

Figure 6.1 Optimization of Seismic Restraint Stiffness

The spring rate may be reduced by further optimizing the location of the seismic restraint. It is shown in Table 6.2 that 1 cps may be gained by locating the restraint thirty inches below the discharge nozzle centerline.

Table 6.2 Optimization of Seismic Restraint Location

Location	File No.	First Mode Frequency
Discharge Centerline	26	11.4 cps
Discharge Centerline - 30"	36	12.4 cps
Discharge Centerline - 50"	39	12.4 cps

In contrast to the free structure, the assumed values for foundation stiffness are no longer a significant parameter. This is seen in Runs 8 and 9, Table 5.1, in which two drastically different foundations yield only a 2% change in frequency.

6.2.2 Design Modifications:

In an effort to increase the lowest structure frequency without adding seismic restraints, four models were analyzed each with a specific design change to give insight into possible design modifications.

The first of these runs assumed an internal seismic restraint placed between the tank and the lower sodium bearing in an attempt to increase the stiffness of the lower tank without adding significant mass.

The second run relocated the suction nozzle from the bottom of the tank to a location near the discharge centerline. This tank design removes 23,000 lbs piping weight from the tip of the structure.

Finally, the top portion of the inner support structure and tank were increased in diameter to increase the stiffness. The dimensions of those components below the discharge nozzle were assumed unchanged from the straight tank design to minimize weight at the tip of the structure. To satisfy design requirements, the top of the tapered portion of the tank at the foundation flange requires a diameter of 304 inches.

Results for these analyses are presented in Table 6.3 and Figure 6.2

Figure 6.2 Lowest Structural Frequency for Various Design Modifications

7.0 STRUCTURAL RESPONSE TO SEISMIC INPUT

There are two main methods for the analysis of structures subjected to base accelerations caused by earthquake type motion. The first is a time history solution which evaluates the response of the entire structure to an actual or artificial earthquake base acceleration record. This involves evaluating the displacements and/or forces in all the elements defining the structure at a specified time increment for the entire record. The second method is the Response Spectrum Analysis. In this method, it is assumed that the structure can be represented dynamically with sufficient accuracy by its first few characteristic modes of vibration. Thus, only maximum response (member forces, displacements) for each of the chosen modes is found and summed to give the total response.

Both of these methods employ the normal mode method of structural dynamics. This involves generating the structure's overall stiffness matrix from the individual element stiffnesses previously described and also the system mass matrix. The system mass and stiffness matrices are then used to solve for the structure's natural (free vibration) periods and mode shapes of vibration with the use of the SAP IV structural analysis program, Reference 2.5.

The spectral and time history analysis procedures differ mainly in their methods of finding the total structural response from the response in the individual modes of vibration. The time history analysis exactly combines the responses at each instant in time whereas the spectral analysis does not retain the time character of response. The spectral analysis employs

the Root Sum Square (RSS) procedure for combining the maximum response in each mode. This technique yields results consistent with a time history analysis.

For the present analysis, the spectral response analysis was performed using the operating base earthquake (OBE) response spectra from the CRBR Equipment Specification. These are reproduced in Figure 7.1 for convenience.

The response spectra, and the natural frequencies and mode shapes from Section 5.0 are sufficient information to calculate modal participation factors and determine the structural response. The SAP4 computer program was used to perform these calculations for both OBE spectra.

The conceptual design was analyzed with the seismic restraint attached to the tank at the discharge nozzle elevation. Thus the finite element model used was the same as for Run 8 of Table 5.1. Seismic response was determined for the first six structural modes. Appendix D contains the complete computer output.

The results of the spectral response analysis are forces, moments, and deflections at each node in the mathematical model. Table 7.1 summarizes these results for critical locations in the pump assembly. The points for which data are presented are defined in Figure 7.2. Rather than presenting only the raw data from the computer output, Table 7.1 shows the stresses resulting from the node loadings. Notice also that the deflections are calculated in a global coordinate system and relative deflection between components must be obtained by subtracting the global deflection of each component.

FIGURE 7.1 OBE RESPONSE SPECTRA

FIGURE 7.2 LOCATION OF STRESS
CALCULATION POINTS

TABLE 7.1 SUMMARY OF STRUCTURAL RESPONSE

Component	Location (Fig.7.2)	Node	Deflection (in)		Bending Stress (psi)		Shear Stress (psi)	
			IP (1)	PP (2)	IP	PP	IP	PP
Shaft	1	150	.053	.100	1148	2175	192	362
	2	143	.062	.118	1480	2992	24	45
	3	136	.046	.086	576	1092	200	379
	4	131	.038	.072	412	780	20	30
	5	84	.007	.010	443	843	41	61
	6	37	.077	.153	348	653	241	470
	7	13	.093	.194	13	27	226	438
Inner Structure	8	153	.009	.017	0	0	1147	9
	9	124	.001	.002	398	754	167	316
	10	67	.007	.014	969	1929	146	284
	11	47	.030	.060	1705	3388	495	984
	12	12	.084	.167	0	0	1	2
Tank	13	118	.001	.002	263	524	135	268
	14	75	.004	.008	925	1836	169	323
	15	2	.066	.131	0	0	74	146

NOTES: (1) Response Spectrum for CRBR Intermediate Pump

(2) Response Spectrum for CRBR Primary Pump

8.0 DISCUSSION OF SEISMIC RESPONSE RESULTS

Figure 7.1 shows where the first structural mode of the standard pump configuration falls out the respective response spectra. It can be seen that the pump natural frequencies lie beyond the peak of each response spectrum. It can be anticipated that response to the primary pump spectrum will be more significant than for the intermediate pump spectrum due to the relative acceleration levels. Thus the seismic response of the pump assembly is minimized when the unit is mounted on a flexible foundation.

All modes higher than the sixth mode, which has a frequency of 25.15 cps, fall on the constant acceleration part of the spectra. Their contributions to the total response are negligible since their modal participation factors are significantly less than those for the first six modes. Consequently, including only the first six modes in the analysis is considered adequate.

Since the response spectra represent the CRBRP structure, they are not directly applicable to the present large scale commercial unit. Thus no conclusions will be drawn. Rather some discussion of general trends will be presented to give insight into how the structure responds to two different response spectra. It is hoped that such a presentation will aid future plant design.

8.1 Shaft Response

Inspection of the deflections for each mode, shows that the maximum deflection of the motor shaft occurs for the second mode. The frequency of this motor bearing mode is relatively close to the first structural

mode and to the maximum acceleration frequency of the response spectrum. To reduce the seismic response of the shaft to the primary pump response spectrum, stiffer motor bearings, which will increase the second mode frequency, should be supplied.

8.2 Structure Response

As expected from the mode shapes, the maximum deflection of the tank and inner structure occurs for the lowest cantilever frequency. Stresses for both structures are largest in the region of the discharge nozzle where the structure is the weakest and where the maximum moments occur. The inner structure also shows larger stress levels at the discharge manifold. The magnitude of the stresses shown do not appear to be excessive but should be considered as a Detailed Design consideration.

8.3 Bearing Response

Table 7.2 shows the loads and relative journal displacement for linear elastic bearings supporting the rotating element. For the primary pump response spectrum, the gap in the middle hydrostatic bearing closes and contact is predicted between the journal and bearing surfaces. Surface contact under a seismic load is not a critical concern and will be investigated further as a Detail Design consideration.

Table 7.2 Bearing Response

Bearing Location	Load (lbs)		Deflection (in)	
	IP	PP	IP	PP
Top Motor	26,392	49,991	.045	.085
Bottom Motor	29,960	56,671	.043	.081
Top Hydrostatic	2,995	4,345	.006	.008
Middle Hydrostatic	17,467	34,010	.035	.070
Lower Hydrostatic	13,625	26,441	.019	.029

9.0 CONCLUSIONS

9.1 Shaft Frequencies

The pump shaft fundamental frequency of 13.02 cps is only five percent above the minimum allowed by Reference 2.2. This frequency is due entirely to the estimated motor bearing stiffness used for the analysis. The first shaft bending mode occurs at 17.91 cps. Thus it is recommended that motor bearings with stiffness greater than 0.7×10^6 lbs/in be supplied to increase the lowest shaft frequency.

9.2 Structural Response

The conceptual design of Reference 2.1 has a minimum structural frequency of 6.80 cps which is less than the operating speed of 10 cps and does not satisfy the requirements of Reference 2.2.

Structural frequency requirements are satisfied by the design of Reference 2.1 if a dynamic restraint is provided on the tank at the discharge nozzle elevation. An upper bound for the stiffness of this tank restraint is estimated to be 5×10^7 lbs/in. Further study of the design and method of attachment of a tank restraint is considered essential.

Feasibility of developing alternative conceptual designs which satisfy frequency requirements without tank restraints has been investigated in this report. However, additional studies are recommended.

9.3 Seismic Response

Evaluation of pump response to the seismic data of Reference 2.2 is intended only to provide guidance in the development of floor response spectra for the actual large scale commercial LMFBR plant.

Development and analyses of these specific spectra are required before any conclusions can be drawn with respect to structural response of the conceptual design.

20. PROGRAM PLANS, SCHEDULES AND COST

20.1 Schedules

The conceptual design has been detailed to establish engineering, procurement, and manufacturing time spans. This program overview shows all major events necessary to design, develop, procure materials, manufacture components, assemble, water test and ship one prototype pump and four production plant pumps.

Detail schedules of facility modification tool and fixture design and fabrication have not been included. The major effort has been directed into the technical overview. With long lead times until start of fabrication there is adequate time for facility improvement or construction.

20.2 Cost

Order of magnitude cost estimates place the cost of the prototype pump at \$30 million requiring eight years from award to delivery.

Additional units are approximately \$12 million each requiring approximately three and one half years from release of material to delivery.

Prototype cost includes development engineering, manufacturing and test facilities and fixtures.

Costs are based on 1976 dollar estimates and material deliveries.

LARGE SCALE BREEDER REACTOR MECHANICAL PUMP
PROGRAM SCHEDULE

PLANT PUMPS

RFP - REQUEST FOR PROPOSAL
SUPP ENG - SUPPORT ENGINEERING
RPT - REPORT