**This document was too large to scan
as a single document. It has
been divided into smaller sections.**

Section 1 of 2

Document Information			
Document #	RPP-5117	Revision	0
Title	DESIGN ACCEPTANCE SUMMARY REPORT FOR THE NEW GENERATION TRANSFER PUMP (NGTP)		
Date	09/28/1999		
Originator	IRONS J	Originator Co.	COGEMA
Recipient		Recipient Co.	
References			
Keywords			
Projects	RPP		
Other Information			

SEP 28 1999

4

ENGINEERING DATA TRANSMITTAL

Page 1 of 1

1. EDT

625009

2. To: (Receiving Organization) 241-SY-101 Rapid Mitigation		3. From: (Originating Organization) 241-SY-101 Rapid Mitigation		4. Related EDT No.: n/a	
5. Proj./Prog./Dept./Div.: 241-SY-101 Rapid Mitigation		6. Design Authority/Design Agent/Cog. Engr.: W.J. Powell		7. Purchase Order No.: n/a	
8. Originator Remarks:				9. Equip./Component No.: P-350	
				10. System/Bldg./Facility: 241-SY-101	
				12. Major Assm. Dwg. No.: VI #50071	
11. Receiver Remarks: <i>USQ is not necessary according to HNF-TP-0842 IV, 5.4, @ Row H b.</i>				13. Permit/Permit Application No.: n/a	
11A. Design Baseline Document? ☒ Yes ☐ No				14. Required Response Date: Sept. 20, 1999	

15. DATA TRANSMITTED					(F)	(G)	(H)	(I)
(A) Item No.	(B) Document/Drawing No.	(C) Sheet No.	(D) Rev. No.	(E) Title or Description of Data Transmitted	Approval Designator	Reason for Transmittal	Originator Disposition	Receiver Disposition
1	RPP-5117	n/a	0	Design Acceptance Summary	S,Q	1	1	1
				Report for the New				
				Generation Transfer Pump				
				(NGTP)				

16. KEY		
Approval Designator (F)	Reason for Transmittal (G)	Disposition (H) & (I)
E, S, Q, D OR N/A (See WHC-CM-3-5, Sec. 12.7)	1. Approval 2. Release 3. Information 4. Review 5. Post-Review 6. Dist. (Receipt Acknow. Required)	1. Approved 2. Approved w/comment 3. Disapproved w/comment 4. Reviewed no/comment 5. Reviewed w/comment 6. Receipt acknowledged

17. SIGNATURE/DISTRIBUTION (See Approval Designator for required signatures)									
(G) Reason	(H) Disp.	(J) Name	(K) Signature	(L) Date	(M) MSIN	(G) Reason	(H) Disp.	(J) Name	(K) Signature
1	2	Design Authority W.J. Powell	<i>W.J. Powell</i>	9/20/99	1	1	1	Ind. Rev.: J. Irons	<i>J. Irons</i>
1	1	Design Agent G.A. Leshikar	<i>G.A. Leshikar</i>	9/20/99	1	1	1	F.M. Hauck	<i>F.M. Hauck</i>
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1	1	Cog. Mgr. R.W. Reed	<i>R.W. Reed</i>	20 Sep 99	1	1	1	C.P. SHAW	<i>Craig Shaw</i>
1	1	QA ⁵⁷⁻⁰⁷ M.L. McElroy	<i>M.L. McElroy</i>	9/20/99					
1		Safety ^{C.D. JACKSON} L.B. Krogerud	<i>L.B. Krogerud</i>	9/20/99					
		Env.							

18. <i>G.A. Leshikar</i> G.A. Leshikar Signature of EDT Originator Date: 9/20/99		19. <i>M.F. Erhart</i> M.F. Erhart Authorized Representative for Receiving Organization Date: 9/20/99		20. <i>R.W. Reed</i> R.W. Reed Design Authority/Cognizant Manager Date: 20 Sep 99		21. DOE APPROVAL (If required) Ctrl No. _____ ☐ Approved ☐ Approved w/comments ☐ Disapproved w/comments	
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S

Design Acceptance Summary Report for the New Generation Transfer Pump (NGTP)

J. Irons

Cogema Engineering

Richland, WA 99352

U.S. Department of Energy Contract DE-AC06-96RL13200

EDT/ECN: 625009

UC: 2000

Org Code: LK940000

Charge Code:

B&R Code:

Total Pages: **557**

Key Words: NGTP, WEMD, transfer pump, new generation transfer pump, Westinghouse Electromechanical Division, Rapid Mitigation, submersible, pump, 2A-513-A1,

Abstract: This report documents design review of the New Generation Transfer Pump versus the functions and requirements of the SY-101 Rapid Mitigation Project. Previously unpublished documentation for the pump is included in support of the design.

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Release Approval

9/28/99
Date

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Design Acceptance Summary Report for the New Generation Transfer Pump (NGTP)

1.0 INTRODUCTION AND SUMMARY

BACKGROUND

The tank 241-SY-101 transfer system was conceived and designed to address the immediate needs presented by rapidly changing waste conditions in tank 241-SY-101. Recently, the waste in this tank has exhibited unexpected behavior in the form of rapidly increasing crust growth. The crust growth rate is such that by September 1999, the waste level within the tank may violate regulatory definitions of a double-shell tank (DST).

The first activity of near-term mitigation is to perform a small transfer of waste from tank 241-SY-101 into tank 241-SY-102. An existing waste transfer pump, known alternately as the new generation transfer pump (NGTP) or pump P-350, has been installed in 241-SY-101 to accomplish the transfer.

The general concept of the initial transfer is to dilute a volume of tank 241-SY-101 wastes with an equal volume of water and to transport the combined volume to tank 241-SY-102. The water dilution is specified to minimize operational risk in the form of solids precipitation in and potential plugging of the transfer line.

The waste transfer pump has been designed by the Westinghouse Electro-Mechanical Division (WEMD). Primary objectives of the pump design were to optimize performance, increase reliability, and to minimize the decontamination and disposal effort required at the end of the pump service life. The pump has a minimum 10,000 hour service life over 10 years at the design point and is capable of complete decontamination and inspection, with a minimum quantity of parts requiring disposal. The pump is repairable and can be restored for reuse.

SCOPE

The scope of this document is to determine the various major design and operational requirements of the transfer pump and to reference the source of these requirements. In addition, the requirements will be reviewed to ascertain the means by which they have been resolved, i.e. analysis, testing, etc.

The documents which support the requirement resolutions will be reviewed for completeness and accuracy to the extent reasonably possible. Those resolution documents which have not been previously issued as Hanford documents will be included by appendix to this document.

DISCUSSION

The major source of requirements for the transfer pump used in this analysis was the following document:

HNF-3885, Revision 0, Functional Requirements and Technical Criteria for the 241-SY-101 Rapid Mitigation System, February 24, 1999.

The following specific requirements were reviewed:

- Pump performance in design waste conditions of density and viscosity
- Pump NPSH requirements
- Pump performance in waste conditions of density and viscosity in excess of design
- Effects of suspended and dissolved solids on pump start-up and mitigation of these effects by flushing
- Effects of large solid particles on pump operation
- Operational life
- Waste chemistry and materials selection
- Performance testing

The following Hanford issued documents were used for requirement resolution:

- HNF-4512, Test Report for the 241-SY-101 Transfer Pump, Revision 0, August 1999, as updated by Sulzer Bingham corrected discharge head curves, 9/7/99.

In addition, the following documents were reviewed and used for requirement resolution, and are appended to this document:

- Westinghouse Electric Corporation, Model 2A-513-A1 Lead Unit, Final Design Report, EM 6673, Revision 4, October 6, 1997.
- Westinghouse Electric Corporation, EMD, Model 2A-513-A1 Transfer Pump Lead Unit Test Report, October 1, 1997.
- Westinghouse Government Services Company, EMD, Model 2A-513-A1 Transfer Pump High Viscosity Operability Evaluation, EM 6910, June 29, 1999.
- Interoffice Memo 74B50-99-090, D. A. Reynolds to C. Hanson, Comparison of Composition of Tank 241-SY-101 to Pump Specifications, September 22, 1999.

CONCLUSION

The above specific requirements were individually reviewed to determine the extent to which each requirement has been satisfied by design, analysis and/or testing.

The review results are summarized as follows:

- Design performance (head/flow with diluted waste) was confirmed to be acceptable by testing.
- High viscosity performance was confirmed to be acceptable by analysis.
- The effects of suspended and dissolved solids on pump start-up were found to be acceptable based on the result of model testing and on the operational requirements for flushing immediately following pump shutdown and for periodic flushing during

shutdown. In particular, the purge flow through the lower wear ring area was found to be adequate during flushing. The operational requirement to periodically flush the pump is strongly concurred with as an essential preventative measure.

- The effects of large solid particles on pump operation were found to be acceptable to the extent reasonably possible by analysis.
- Pump operational 5 year project life was confirmed by analysis.
- Pump materials selection and physical design were found to be acceptable and optimized to the extent reasonably possible for ease of decontamination and decommissioning.

With regard to those requirements reviewed in this report, the transfer pump is found to be acceptable for transfer of waste from tank 241-SY-101 to tank 241-SY 102 under the design conditions anticipated in these tanks.

2.0 REQUIREMENTS AND RESOLUTION

Requirement #1 and Source

Pump Flow.

HNF-3885, Functional Requirements and Technical Criteria for the 241-SY-101 Rapid Mitigation System, February 24, 1999, Revision 0, para. 4.2.2.1, indicates that the pump shall be able to transfer 1140 m³ of diluted waste in 20 days. This corresponds to an average flow of 10.7 gpm.

HNF-3885, Revision 0, para. 4.2.4.1, indicates that the diluted waste transfer line velocities shall be not less than 6 ft/sec. This corresponds to a flow of 60 gpm in a 2 inch line.

Waste Viscosity and Density.

HNF-4264, Process Control Plan for Tank 241-SY-101 Surface Level Rise Remediation, Revision 0, Table 2-4, indicates that viscosity will be 7.4-15 cp for 1:1 diluted waste with 1.3 SG and 18-36 cp for 1:2 diluted waste with 1.4 SG. It is further stated that only in the extreme of low dilution at the high viscosity bound does the viscosity exceed 30 cp, and at the mean dilution the expected maximum viscosity will be 15 cp. This same data is presented in HNF-3885, Revision 0, Appendix A.

Pump operations in conditions of density and viscosity in excess of HNF-4264 design conditions with dilution are reviewed later.

Pump Head.

Preliminary Calculation HNF-4359, Revision 0-E, indicates the following pump head requirements for waste transfer from SY-101 to SY-102 in the maximum design conditions above:

166 ft head @ 140 gpm, 1.4 SG, 36 cp

Resolution of Requirement #1

WHC-S-0238, Specification for Prototype Tank Farm Transfer Pumps with Submersible Motor, Revision 0, specified the pump performance as:

450 ft head, 140 gpm, 1.5 SG, 30 cp.

The design, analysis, and development activities performed to ensure the design are documented in the following design report and are presented in Appendix A:

- Westinghouse Electric Corporation EMD Final Design Report E.M. 6673 Revision 4, October 6, 1997.

The performance was verified by the following pump tests:

- Westinghouse Electric Corporation EMD test commencing June 25, 1997. The test report is Appendix B of this document.
- Sulzer Bingham test commencing April 27, 1999, reported in HNF-4512, Revision 0, August 1999, as updated by Sulzer Bingham corrected discharge head curves, 9/7/99.

The pump is variable speed and will clearly meet the performance requirements which are considerably less than design.

Requirement #2 and Source

NPSH

HNF-3885, Revision 0, Table 1-1 indicates that the pump must operate with an available NPSH of 10.6 ft.

Resolution of Requirement #2

Westinghouse Electric Corporation EMD Final Design Report E.M. 6673 Revision 4, October 6, 1997, section 5.8, provided as Appendix A to this document, describes the cavitation testing performed by Southwest Research Institute. The test was conducted on the first stage impeller in a special test loop.

The NPSH required at design flow of 140 gpm was 10.6 ft minimum, which is acceptable. The NPSH required at lower flows is less, as shown on Figure 5-20.

Requirement #3 and Source

Westinghouse EMD was contracted to perform a study and analysis to assess the upper bounds of density and viscosity under which the transfer pump could continuously operate with an acceptable reliability and life expectancy. The operating conditions specified were:

60 gpm, 100 ft head, 1.00 to 1.70 SG, 1 to 1000 cp viscosity.

The results of the study would become the upper bounding operating conditions for the pump.

The study, which is presented in Appendix C of this document, concluded the following:

Maximum viscosity acceptable for continuous operation at above design point is 250 cp. In this condition the operating life of the motor is approximately 8900 hrs.

The de facto high viscosity maximum operating conditions for continuous operation are therefore:

60 gpm, 100 ft head, 1.00 to 1.70 SG, 250 cp viscosity.

In addition, HNF-3885, Revision 0, Functional Requirements and Technical Criteria for the 241-SY-101 Rapid Mitigation System, February 24, 1999, section 4.2.1.1, is being revised to require a capability of pumping 200 cp, 1.6 SG fluid. This represents the undiluted tank waste conditions.

Resolution of Requirement #3

Westinghouse EMD produced a report, titled: Model 2A-513-A1 Transfer Pump High Viscosity Operability Evaluation, S.O. V008, June 29, 1999, E.M. 6910, Original Issue. The report is presented in Appendix C of this document.

The following topics were considered in the evaluation:

- Radial and thrust bearing design.
- Hydraulic design.
- Motor electrical design.
- VFD design
- Motor temperatures
- Motor internal flow
- Mechanical loads
- Fluid frictional losses
- Pump, motor and column vibrations

The viscosity study indicated there are two items limiting operation of the pump in a high viscosity fluid. One is the torque output of the motor and the second is the operating life

of motor electrical winding. The required operating torque approaches the maximum motor torque (breakdown torque) at approximately 440 centipoise. Above this viscosity the motor is not capable of coming up to operating speed and will stall. The operating life of the motor is limited to approximately 8900 hours at 250 centipoise and 10 hours at 375 centipoise. Therefore, of the two limiting items, motor winding life is the controlling parameter.

WEMD recommended that for continuous operation the maximum viscosity of the pumped fluid should be 250 cp at the new design point. WEMD also recommended that the pump should be operated in fluids with viscosities between 250 and 375 cp for no longer than 4 hours total to ensure at least 5000 hours of design life.

Preliminary Calculation HNF-4359, Revision 0-E, indicates the following pump head requirements for waste transfer from SY-101 to SY-102 in the high viscosity conditions above:

72 ft head @ 60 gpm, 1.7 SG, 250 cp

Therefore the pump will be capable of transferring waste, on a continuous basis, from tank 241-SY-101 to tank 241-SY 102 at a minimum flow of 60gpm @ 1.7 SG and 250 cp.

This case also envelopes the off-design case of transferring 60 gpm of undiluted waste @ 1.6 SG and 200 cp

Requirement #4 and Source

The pump shall be capable of starting following an extended period of shutdown. The concern is that suspended or dissolved solids may settle or precipitate in the various internal pump annular spaces and cavities to the extent that pump startup would be precluded.

Resolution of Requirement #4

Westinghouse Electric Corporation EMD Final Design Report E.M. 6673 Revision 4, October 6, 1997, section 5.9, Appendix A of this document, discusses the internal collection and circulation of particles. Section 12 of the document describes fill and drain testing on full scale plastic and metal models.

The model testing showed that after being filled with simulated waste containing suspended solids, the model would self drain cleanly if allowed to immediately. Only when the waste was held in the pump to allow settling of solids did the model plug and not drain. The application of purge water at this point allowed the waste to drain, but did not remove all collected solids.

When installed in SY-101 the pump cannot self drain and a purge flow is required to simulate this condition.

The model testing resulted in modifications to the purge system and modification of some internal features to preclude plugging and to improve drainage.

Westinghouse recommends that the pump should be flushed immediately after operation to prevent the settling of solids.

Table 5-14 of the above Westinghouse Design Report provides the internal circulation flow resulting from the pump flush at various purge flow rates. An area of concern has been the ability to flush the lower wear ring area at the pump inlet. The internal circulation flow analysis indicates that during a 10 gpm flush of the pump, more than 1 gpm of the purge flow will pass through the impeller lower shroud area and through the wear ring gap. Thus the wear ring area is adequately flushed during the purge operation.

Per Transfer Procedure TO-430-100 (Draft) the pump will be flushed after operation, and the clean warm flush water will be held in the pump. This will also preclude the possibility of dissolved solids from the waste precipitating inside the pump. Initially, the pump flush will be repeated once per week, per HNF-4264 Revision 0, Process Control Plan for Tank 241-SY-101 Surface Level Rise Remediation, Section 8.1. The flush frequency will be re-evaluated periodically to reflect operating experience.

Requirement #5 and Source

WHC-S-0238, Revision 0, B-1 indicates that the pump must be able to pass an occasional particle of up to 0.25 inches in diameter.

Also, the maximum size of particle that could enter the motor rotor-to-stator gap should be addressed.

Resolution of Requirement #5

Westinghouse Electric Corporation EMD Final Design Report E.M. 6673 Revision 4, October 6, 1997, Appendix A of this document, addresses these requirements as follows:

- Section 3.3.1.4. indicates that the main pump hydraulic passages are sized to pass particles up to 0.25 inch diameter.
- Section 5.9 discusses the maximum steel particle size that could be introduced into the 0.142 inch annular motor rotor-to-stator gap.

The particle size is calculated which would have a terminal velocity in the waste equal to the waste flow velocity entering the motor. The maximum particle size calculated was 0.075 inch with a waste flow of 140 gpm, 1.5 SG and 30 cp. The waste flow rate in this application will be less than 140 gpm and hence the particle size will be less than 0.075 inch.

In addition, the motor inlet holes are offset circumferentially from the main pump discharge holes by 20° to 38°, which will further mitigate the migration of particles into the motor.

Westinghouse has demonstrated to the extent reasonably possible by analysis that the pump will function correctly with the anticipated solid particles present in the waste.

Requirement #6 and Source

HNF-3885, Revision 0, Table 1-1 indicates that the pump is designed for an operating life of 10,000 hours over 10 years. Project requirements, Section 4, specify a 5 year life.

Resolution of Requirement #6

Westinghouse Electric Corporation EMD Final Design Report E.M. 6673 Revision 4, October 6, 1997, section 13.0, Appendix A of this document, addresses this requirement.

The maintenance-free life expectancy of the pump included an analysis of the following items, which are considered the primary modes of pump failure:

- Bearing wear
- Rotor and stator can wear
- Hydraulics wear
- Impeller seal wear
- Motor winding insulation life
- Column assembly fatigue
- Rotor shaft fatigue

The results of the analyses indicate that the minimum life of the pump is 10,000 hours.

Requirement #7 and Source

HNF-3885, Revision 0, paragraph 4.4.1 states "Selection of materials shall consider operations, maintenance, and decontamination and decommissioning in addition to ensuring compatibility with interfacing components and environments".

Resolution of Requirement #7

Westinghouse Electric Corporation EMD Final Design Report E.M. 6673 Revision 4, October 6, 1997, section 4.0, Appendix A of this document, addresses this requirement. The physical and material design of the pump is optimized for ease of decontamination and decommissioning.

For the chemical environment presented by the tank waste, the Design Report references WHC-S-0238, Specification for Prototype Tank Farm Transfer Pumps with Submersible Motor, Revision 0. Table 1 of the specification lists the chemical constituents expected to be found globally in the tank farms.

The following references characterize the specific 241-SY-101 waste.

- Herting, D. L., Welsh, T. L., Lambie, R. W., Tran, T. T., "Tank Characterization Report for Double-Shell Tank 241-SY-101", WHC-SD-WM-ER-409, Rev. 0, July 1995
- Estey, S. D., Barton, W. B., "Process Control Plan for Tank 241-SY-101 Surface Level Rise Remediation", HNF-4264, Rev. 0, June 1999

Interoffice Memo 74B50-99-090 from D. A. Reynolds to C. Hanson September 22, 1999, Appendix D of this document, compares the composition of waste in SY-101 to that in the pump specification, and concluded that the only significantly higher chemical concentration found in the SY-101 tank waste when compared with the pump design specification was for Total Na. The memorandum further concluded that this should not affect the pumpability of the waste nor would the materials of construction be harmed by the higher sodium values.

Requirement #8 and Source

HNF-3885, Revision 0, paragraphs 5.2.2 and 5.2.3 require pump performance testing, including a 24 hour run-in test at pump design flow and pressure.

Resolution of Requirement #8

Pump performance and run-in test were conducted successfully. The results are reported in the following documents:

- Westinghouse Electric Corporation EMD test commencing June 25, 1997. The test report is Appendix B of this document.
- Sulzer Bingham test commencing April 27, 1999, reported in HNF-4512, Revision 0, August 1999, as updated by Sulzer Bingham corrected discharge head curves, 9/7/99.

APPENDIX A

WEMD
E.M. 6673

WESTINGHOUSE ELECTRIC CORPORATION
CHESWICK, PENNSYLVANIA

MODEL 2A-513-A1 LEAD UNIT
FINAL DESIGN REPORT ADDENDUM

S.O. V001

October 6, 1997

ENGINEERING MEMORANDUM 6673
REVISION 4

APPROVED BY:

PE

J. A. Drake, Manager
Pump & Propulsion &
Design & Technology

COORDINATED BY:

R. F. Guardani, Principal Engineer
Pump Design & Technology

NOTICE

Revision 4 of E.M. 6673 contains only the specific pages revised (and their facing pages). Refer to Revision 1 and Revision 3 of E.M. 6673 for all other pages which were not revised.

KEYWORDS: Transfer Pump
Pump
2A-513-A1

RECORD OF REVISIONS

Rev Number	Date	Revision	Written or Revised by:	Approved by:
Original	5/8/95	Original	R. F. Guardiani	J. A. Drake
1	12/14/95	<p>Revised Sections 3.1.5, 3.3.1.1, 3.3.1.2, 3.3.1.3, 3.3.1.4, 3.3.1.5, 3.3.2.1, 3.3.2.3, 3.3.2.4, 3.3.3, 3.3.4.1, 4.0, 5.4.1, 5.4.2, 5.4.3, 5.4.4, 5.4.5, 5.5.1, 5.5.2, 5.5.3, 5.5.4, 5.7.1, 5.7.2, 5.7.3, 5.7.4, 5.7.5, 5.7.6, 5.7.7, 8.6.4, 8.8, 8.10.2, 9.1, 9.2, 11.1, 11.1.1, 11.1.2, 11.1.3, 12.1, 12.2, 13.0, 13.1, 13.2, 13.3 and 15.0.</p> <p>Revised Figures 15-2, 15-4, 15-7, 15-9, 15-11, 15-13 and 15-15.</p> <p>Add new Sections 6.5, 6.5.1, 6.5.2, 6.5.3, 6.5.4, 6.5.5, 6.6, 6.6.1, 6.6.2, 6.6.3, 6.6.4.</p>	R. F. Guardiani	J. A. Drake
2	10/10/96	<p>Revised Sections 3.0, 3.1.2, 3.1.3, 3.1.4, 3.2.1, 3.3.1.1, 3.3.1.2, 3.3.1.3, 3.3.1.4, 3.3.2.1, 3.3.2.3, 3.3.2.4, 3.3.3, 3.3.4.1, 5.6, 5.7.5, 5.7.6, 7.1.2.1, 7.1.3, 7.1.4, 8.6.2, 10.0 (all) and 11.1.3.</p> <p>Revised Figures 5-10, 5-11, 5-15, and 15-1, 15-2, 15-4 thru 15-8, 15-10 thru 15-14, and 15-16.</p> <p>Added new section 5.8 and 5.9.</p>	R. F. Guardiani	J. A. Drake

RECORD OF REVISIONS

Rev Number	Date	Revision	Written or Revised by:	Approved by:
3	4/30/97	Revised Sections 3.3.1.1, 3.3.1.2, 3.3.1.3 Revised Section 4.0 and Table 4-1. Revised Sections 5.7.1, 5.7.2, 5.7.3, 5.7.4, 5.7.5, 5.7.6 Revised Section 7.2.3. Revised Section 9.0 entirely. Revised Section 10.0 entirely. Revised Section 13.1. Revised Section 15.0 entirely. Added new drawings 15-17 through 15-20. Revised Section 17.0	R. F. Guardiani	J. A. Drake
4	10/6/97	Revised Section 5.7.6, Figures 5-18 and 5-19, and Table 5-15. Revised Section 15.0. Revised Figures 15-4, 15-9, 15-11, 15-13, 15-14, 15-17, 15-19 and 15-20.	R. F. Guardiani	J. A. Drake

ABSTRACT

This design report presents the performance curves, analyses and procedures required to document the final design of the lead unit Westinghouse EMD Model 2A-513-A1 Submerged Canned Motor Hazardous Waste Transfer Pump for the Westinghouse Hanford Company (WHC) Tank Farm. It is prepared under WHC Purchase Order MRB-SXP-342098 in compliance with the requirements of Section 5.2 of WHC-S-0238, Revision 1, Final Design Review Data.

ACKNOWLEDGEMENTS

This engineering memorandum is a report of the final design of the Tank Farm Transfer Pump. The calculations and technical content are compiled from information provided by the participants on the Transfer Pump design team. The following listing acknowledges the contributions of those participants.

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Contributing Authors -

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ANALYSIS VERIFICATION

A listing of all of the subjects of this report that contain transfer pump design calculations is presented below. All of these calculations have been independently reviewed and verified by a second person. The engineer that performed the calculations and the engineer that verified the calculations have signed for the corresponding analysis section.

Title	Prepared by:	Verified by:
Hydraulic Performance	<u><i>A. Morris</i></u> A. Morris	<u><i>R. Guardiani</i></u> R. Guardiani
Motor Electrical Performance	<u><i>Jeffrey A. Zastawniak</i></u> J. Zastawniak	<u><i>P. Gaberson</i></u> P. Gaberson
Motor Thermal	<u><i>J. Raymond</i></u> J. Raymond	<u><i>P. Gaberson</i></u> P. Gaberson
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Bearing Performance	<u><i>R. Guardiani</i></u> R. Guardiani	<u><i>R. Gronholm</i></u> R. Gronholm
Dynamic Evaluation	<u><i>M. Lamantia</i></u> M. Lamantia	<u><i>D. Deddo</i></u> D. Deddo
Structural	<u><i>L. E. Parsons</i></u> L. Parsons	<u><i>R. Guardiani</i></u> R. Guardiani

RECORD OF REVISIONS	i
ABSTRACT	ii
ACKNOWLEDGEMENTS	iii
ANALYSIS VERIFICATION	iv
TABLE OF CONTENTS	v
1.0 INTRODUCTION	1-1
2.0 SPECIFICATION REQUIREMENTS	2-1
3.0 DESIGN REQUIREMENTS	3-1
3.1 Electrical Design	3-5
3.1.1. Rotor Assembly	3-5
3.1.2 Stator Assembly	3-6
3.1.3. Power Leads and Cables	3-9
3.1.4 Terminal Box	3-10
3.1.5 VFD Interface	3-10
3.2 Hydraulics	3-11
3.2.1 Impellers and Diffusers	3-11
3.2.2 Internal Circulation	3-17
3.3 Mechanical Design	3-18
3.3.1 Motor Unit Assembly	3-18
3.3.1.1 Rotating Assembly	3-18
3.3.1.2 Bearings	3-21
3.3.1.3. Stator Assembly	3-23
3.3.1.4 Static Hydraulics	3-25
3.3.1.5 Purge and Dilution Lines	3-28
3.3.2 Column Assembly	3-31

TABLE OF CONTENTS

RPP-5117, Rev 0
Addendum A-8b

3.3.2.1	Column	3-31
3.3.2.2	Column Extension Assembly	3-32
3.3.2.3	Transition Piece	3-32
3.3.2.4	Power Cable Conduit	3-33
3.3.2.5	Discharge Nozzle	3-33
3.3.3	Purge and Dilution Line Network	3-33
3.3.4	Pressure and Accelerometer Instruments	3-34
4.0	MATERIAL OF CONSTRUCTION	4-1
5.0	HYDRAULIC EVALUATION	5-1
5.1	Hydraulic Design Basis	5-1
5.1.1	Hydraulic Adjustments for Dilution Lines	5-5
5.2	Predictions of BHP Requirement	5-6
5.3	Hydraulic Loads	5-6
5.4	Development Air Test Program	5-10
5.4.1	Introduction	5-10
5.4.2	Program Scope	5-10
5.4.3	Results/Conclusions	5-14
5.4.4	Sample Calculation	5-23
5.5	Hydraulic Performance	5-27
5.5.1	Introduction	5-27
5.5.2	Performance Scaleup at Design Point	5-27
5.5.3	Performance Scaleup at Other Operating Points	5-32
5.5.4	Radial Load Scaleup	5-35
5.6	Reverse Flow (Windmilling) Analysis	5-37a

TABLE OF CONTENTS

5.7	Internal Circuit Analysis-Introduction	5-38
5.7.1	Description of Internal Circulation Flowpaths	5-38
5.7.2	Analysis of Normal Operating Points	5-41
5.7.3	Analysis of the Reverse Flow Condition	5-53
5.7.4	Analysis of the Shutoff Head Condition	5-53
5.7.5	Analysis of the Purge Line System	5-57
5.7.6	Analysis of the Dilution Line System	5-57
5.8	NPSH Requirements	5-66
5.9	Internal Collection and Circulation of Particles	5-68
6.0	MOTOR EVALUATION	6-1
6.1	Motor Design Characteristics	6-1
6.2	Motor Materials Compatibility with Environment	6-1
6.3	Motor Performance Characteristics	6-3
6.3.1	Normal Operating Points	6-3
6.3.2	VFD Start and Operation	6-3
6.3.3	Reverse Flow Start	6-5
6.3.4	Locked Rotor Analysis	6-6
6.4	Motor Protection Requirements	6-6
6.5	Variable Frequency Drive (VFD) Information	6-18
6.5.1	General Overview	6-18
6.5.2	Hardware Overview	6-18
6.5.3	Control Modes	6-21
6.5.4	Drive Features	6-22
6.5.5	VFD Technical Specifications	6-24

TABLE OF CONTENTS

RPP-5117, Rev 0
Addendum A-10

6.6	Variable Frequency Drive (VFD) Sizing and Operation	6-25
6.6.1	VFD Sizing and Selection	6-25
6.6.2	VFD Operation	6-25
6.6.3	Drive Enclosure	6-27
6.6.4	Power Draw	6-28
7.0	THERMAL EVALUATION	7-1
7.1	Motor Thermal Analysis	7-1
7.1.1	Design Requirements	7-3
7.1.2	Thermal Analysis	7-3
7.1.2.1	Pump Thermal Characteristics	7-5
7.1.3	Locked Rotor Thermal Analysis	7-6
7.1.4	Loss of Flow Condition	7-10
7.2	Machine Thermal Growth Analysis	7-13
7.2.1	Column Insertion Length	7-13
7.2.2	Required Rotating Assembly Axial Clearance	7-14
7.2.3	Radial Bearing Clearances	7-15
7.2.4	Electrical Cable Axial Growth	7-17
8.0	STRUCTURAL EVALUATION	8-1
8.1	Assumptions	8-1
8.2	Handling Analysis	8-3
8.2.1	Horizontal Case	8-3
8.2.2	Vertical Case	8-3
8.3	Jet Impingement	8-6
8.3.1	Column Stress for 150 lb Impingement Load	8-6
8.3.2	Lateral Jet Impingement Load for 10^4 Cycles	8-6

TABLE OF CONTENTS

8.4	Seismic Analysis	8-8
8.5	Code Section VIII-1 Analysis	8-11
8.5.1	Column Discharge Pipe (3" Schedule 40S)	8-11
8.5.2	Pump External Wall	8-14
8.5.3	Stator ID Housing Wall	8-15
8.5.4	Purge/Dilution Lines	8-17
8.6	Bolting Analysis	8-18
8.6.1	Pump Mounting Flange Bolt Loads	8-18
8.6.2	Transition Piece Bolts	8-19
8.6.3	Diffuser Bolts	8-21
8.6.4	Shaft-Impeller Bolt	8-21
8.7	Shaft Fatigue Analysis	8-30
8.7.1	Shaft Stresses	8-30
8.7.2	Stress Concentrations at Critical Shaft Location	8-31
8.7.3	High Cycle Fatigue Analysis	8-32
8.8	Thermal Shock Analysis	8-34
8.9	Locked Rotor Analysis	8-36
8.10	Miscellaneous Analyses	8-36
8.10.1	Rotor End Ring Overhang	8-36
8.10.2	Rotor and Stator Can Analyses	8-37
9.0	BEARING EVALUATION	9-1
9.1	Radial Bearings	9-2
9.2	Thrust Bearing	9-50

TABLE OF CONTENTS

RPP-5117, Rev 0
Addendum A-12

10.0	DYNAMIC EVALUATION	10-1
10.1	Model Description	10-1
10.2	Classical Critical Speed Analysis	10-6
10.3	Undamped Natural Frequency Analysis	10-8
10.4	Damped Rotordynamic Analysis	10-42
11.0	ABRASION TESTING	11-1
11.1	Test Description and Test Plan	11-1
11.1.1	Bearing Testing	11-2
11.1.2	Can Testing	11-14
11.1.3	Diffuser Impingement Testing	11-20
12.0	FILL AND DRAIN TESTING	12-1
12.1	Test Description	12-1
12.2	Test Results	12-2
13.0	MAINTENANCE-FREE LIFE EXPECTANCY EVALUATION	13-1
13.1	Bearings	13-1
13.2	Motor Cans	13-4
13.3	Hydraulics	13-7
13.4	Motor Windings	13-9
13.5	Fatigue	13-9
14.0	DECONTAMINATION AND MAINTAINABILITY	14-1
14.1	Pump Assembly Decontamination	14-1
14.2	Pump Assembly Handling	14-1
14.3	Column Assembly Maintainability	14-2
14.4	Motor Unit Assembly Maintainability	14-3

TABLE OF CONTENTS

15.0	DRAWINGS	15-1
Figure 15-1	Outline	15-2
Figure 15-2	General Assembly	15-3
Figure 15-3	Electrical Schematic	15-5
Figure 15-4	Motor Unit Assembly	15-6
Figure 15-5	Column Assembly	15-8
Figure 15-6	Column Extension	15-11
Figure 15-7	Rotor Assembly	15-13
Figure 15-8	Stator Assembly	15-16
Figure 15-9	Impeller Casting	15-18
Figure 15-10	First Stage Impeller	15-19
Figure 15-11	Second Stage Impeller	15-20
Figure 15-12	First Stage Diffuser	15-21
Figure 15-13	Stator Modification	15-23
Figure 15-14	Second Stage Diffuser	15-26
Figure 15-15	Suction Adapter Casting	15-30
Figure 15-16	Suction Adapter Machining	15-31
Figure 15-17	Thrust Bearing	15-32
Figure 15-18	Thrust Runner Assembly	15-33
Figure 15-19	Lower Radial Bearing Assembly	15-34
Figure 15-20	Upper Radial Bearing Assembly	15-35
16.0	PROCEDURES	16-1
17.0	DISTRIBUTION LIST	17-1

1.0 INTRODUCTION

The lead unit Submerged Canned Motor Hazardous Waste Transfer Pump, Model 2A-513-A1, has been designed in accordance with Specification WHC-S-0238, Revision 1 (Reference 1) by the Westinghouse Electro-Mechanical Division (WEMD) for Westinghouse Hanford Company (WHC) under Purchase Order MRB-SXP-342098. Primary objectives of the Transfer Pump design were to optimize performance, increase reliability, and to minimize the decontamination and disposal effort required at the end of the pump service life. The lead unit Transfer Pump has a minimum 10,000 hour service life over 10 years at the design point and is capable of complete decontamination and inspection, with a minimum quantity of parts requiring disposal. The Model 2A-513-A1 lead unit Transfer Pump is repairable and can be restored for reuse.

This Final Design Report has been prepared to document the Model 2A-513-A1 lead unit Transfer Pump design in accordance with Section 5.2 of Reference 1. The Final Design Report contains drawings which present the overall design configuration and wiring diagrams/schematics of the lead unit Transfer Pump. A Bill of Material which specifies all materials and their standards/specifications is included in the report. The Final Design Report provides the projected motor and hydraulics performance curves, calculations (structural, dynamics, thermal and hydraulics) and procedures required to document and support the design of the lead unit Transfer Pump. In addition, the report provides a narrative description of design features of the Transfer Pump including decontamination, maintenance, and assembly. A description of plans for assembling, disassembling, and testing the Model 2A-513-A1 lead unit Transfer Pump is also included.

WEMD has the principal responsibility for review of the pump design and has accordingly conducted internal design reviews on the motor unit assembly, the column assembly and the electrical motor.

WEMD has explicit controls on computer software used for design of the transfer pump as identified in this document. These are stated in WEMD Interdepartmental Procedure (IDP) E-23, titled "Control of Computer Programs." This procedure defines all aspects of computer program control for software used in the design and analysis of products. It

includes, but is not limited to, the development, validation, configuration control, maintenance, error reporting/resolution, control, and protection of computer programs. It also defines the methods for local and off-site computer files backup. This procedure satisfies ASME NOA-2a, Part 2.7 requirements for commercial nuclear safety related components and is applied to products such as the transfer pump. This procedure exceeds the requirements of ISO 9000.

The procedure is composed of six areas as listed below:

Part A defines the general process for control of life cycle of computer programs.

Part B addresses the aspects of configuration control.

Part C addresses error reporting and resolution.

Part D defines computer security administration.

Part E applies to storage of workstation computer data.

Part F addresses off-site storage of computer data.

Documentation in support of the execution of IDP E-23, including applicable forms are maintained in a Computer Code Library, and Software Control Engineers are responsible for performing computer software control activities via control and maintenance of the library.

2.0 SPECIFICATION REQUIREMENTS

The-design basis requirements for the Westinghouse Model 2A-513-A1 Hazardous Waste Transfer Pump are contained in the following technical documents:

WHC-S-0238, Revision 1 - Specification for Prototype Tank Farm Transfer Pumps with Submersible Motor (Reference 1)

API-610, Seventh Edition, February, 1989 - Centrifugal Pumps for General Refinery Service (Reference 2)

Table 2-1 summarizes the key design requirements for the lead unit Transfer Pump.

Table 2-1 Design Requirements	
Design Head	450 ft at the discharge flange
Design Flow	140 gpm
Critical Speed, Minimum	125% of maximum speed
Fluid Specific Gravity	1.0 - 1.5
Fluid Viscosity	1.0 - 30 Cp
Solids Volume	0 - 50%
Solids Size	0 - 1000 microns 95% < 50 microns
Miller Number (ASTM G75-82)	≤ 50
Fluid Temperature	200°F maximum
Minimum Available NPSH	10 ft
Power Required	460V, 3-Phase, 60 Hz to Variable Frequency Drive
Pump Weight	10,000 lbs. maximum
Insertion Length	49 ft
Radiation	5.87 x 1.0 ⁷ rad maximum
Dilution Water Supply Purge Water Supply	105 gpm at 100 psig 10 gpm at 90 psig
Operating Life, Minimum	10,000 hours over 10 years

1000
Megarad

3.0 DESIGN DESCRIPTION

The Westinghouse Model 2A-513-A1 lead unit pump for the Tank Farm Submersible Motor Transfer Pump Program is shown in Figures 15-1 and 15-2. The lead unit Transfer Pump design incorporates the following key features:

- A 60 horsepower, submerged canned motor with integral hydraulics which is passively cooled by waste tank process fluid.
- A two stage, centrifugal pump with one crossover diffuser and one axial diffuser.
- A single piece, welded column assembly with hermetically sealed internals to preclude the entry of process fluid.
- A field adjustable column assembly which allows overall insertion length changes between 47 and 52 feet by cutting one pipe section and making one circumferential pipe weld.
- No dynamic seals in the motor.
- No dedicated external lubrication system for the pump bearings.
- No dedicated external cooling system for the submerged canned motor.
- No shafting, coupling, or dynamic penetrations of the pump mounting flange.
- Rugged, reliable and maintenance free below the mounting flange over the design life of the pump.

The lead unit Model 2A-513-A1 Transfer Pump is a two stage, centrifugal pump delivering 140 gpm at 450 feet of head at the pump discharge flange above the tank mounting plate with a minimum available NPSH of 10.6 feet. The pump design speed is 3860 rpm,

delivered by a 60 HP two-pole motor driven at 71 Hz by a variable frequency drive (VFD) supplied with 460 volt, three phase, 60 Hz power. The lead unit Transfer Pump design characteristics are summarized in Table 3-1. The analyses contained in the following sections indicate the lead unit Model 2A-513-A1 Transfer Pump will meet or exceed all specification requirements. The following summarizes the results of the analyses contained herein:

- The hydraulics will meet the 140 gpm and 450 feet of head design requirement at the maximum 1.5 specific gravity and 30 Centipoise viscosity conditions within the allowable 11.0 inch diameter space envelope.
- The motor will produce the 60 horsepower required for operation at the design point within the 11.00 inch diameter space envelope.
- Motor cooling with 200°F process fluid is acceptable for continuous operation over the 10,000 hour design life of the pump.
- The rotor assembly classical critical speed is 160% of the pump design speed.
- The performances of the thrust bearing and radial bearings are acceptable.
- The pump is acceptable with a 150 lb jet impingement force for 10,000 cycles.
- The pump can be lifted from horizontal to vertical without damage.
- The Transfer Pump will maintain integrity during the specified seismic event.

The Model 2A-513-A1 Transfer Pump incorporates some design features that are not typical of WEMD canned motor pumps. These features occur because of the small size of the pump and the desire to make the pump easily disassembled. To ensure the transfer pump can be easily manufactured, mock-ups of key features are being produced. This mock-up development effort is delineated in Table 3-2.

Table 3-1 Design Characteristics	
Design Head	450 ft at the discharge flange
Design Flow	140 gpm
Operating Speed	3860 rpm
Critical Speed	6175 rpm (160% operating)
Fluid Specific Gravity	1.5 maximum
Fluid Viscosity	30 Cp maximum
Fluid Temperature	200°F maximum
Required NPSH	10.6 ft minimum
Power Required	460V, 3-Phase, 60 Hz to variable frequency drive
Motor Cooling	Process Fluid
Bearing Lubrication	Process Fluid
Dilution Water Supply	105 gpm at 100 psig
Purge Water Supply	10 gpm at 90 psig
Operating Life	10,000 hours minimum over 10 years
Weight	Rotating Assembly - 161 lbs Dry Motor Unit Excluding Rotating Assy - 1239 lbs Dry Column Assembly - 2390 lbs Total Dry Weight - 3790 lbs Motor Unit Fluid at 1.5 SG - 44 lbs Column Assembly Fluid at 1.5 SG - 262 lbs Total Wet Weight - 4096 lbs

Table 3-2

**TRANSFER PUMP
MOCK-UP DEVELOPMENT**

Title	Objective	New or Different
Braze Mock-Up	Develop a brazing technique for the rotor bar to the connection ring using puddle brazing.	Puddle braze of a small unit. Induction brazing.
End Turn Mock-Up	Verify that the wound stator end turns can be maintained within preliminary drawing tolerances and develop winding process.	Size of winding end turns.
Piping Weld Mock-Up	Verify accessibility for welding piping at the top end of the pump.	Piping configuration.
Impeller/Rotor Shaft Fit	Determined the maximum allowable fit clearance that will maintain component runout and balance parameters.	Impellers not shrunk onto shaft.
Wound Stator Mock-Up	Develop manufacturing processes for a completed wound stator.	Mush winding of a small bore, 18 inch length stator.
Rotor Assembly Mock-Up	Develop manufacturing processes for a completed rotor assembly.	Size of rotor slots and rotor can.

E.M. 6673

3-4

3.1 Electrical Design

3.1.1 Rotor Assembly

General

The rotor assembly for the transfer pump consists of a solid 410 stainless steel shaft and a squirrel cage winding (See Figure 15-7). The squirrel cage rotor winding contains 17 copper alloy bars which are brazed at both ends to a copper connection ring. Stainless steel end rings are then assembled to each end of the rotor core to protect the connection rings. A Hastelloy can is then shrunk onto the outer diameter of the rotor core and welded at both ends to complete the assembly. The overall length of the rotor assembly is 22.75 inches. The total weight of the entire canned rotor assembly is approximately 85 pounds.

Rotor Core

The outer diameter of the stainless steel rotor shaft is 3.728 inches and the active length of the rotor core is 18.5 inches. The inner diameter of the rotor is considered to be zero because of the solid steel rotor shaft. The rotor core itself contains 17 milled slots. The nominal size of the rotor slots is 0.338 inches wide by 0.590 inches high. The flux density in the rotor core is 70 KL/in², with the maximum flux densities in the rotor teeth being 95 KL/in². The approximate weight of the active rotor iron is 65 pounds.

Rotor Winding

The squirrel cage rotor winding consists of 17 copper alloy rotor bars which are brazed to pure copper connection rings at each end of the assembly. The nominal size of the rotor bars is 0.330 inches wide by 0.560 inches high. The rotor bars are made from UNS Alloy C70200 which contains 1.70% Nickel and

0.15% Manganese. The nominal dimensions of the connection ring are 0.800 wide by 0.600 high. The connection rings are made from electrolytic tough-pitched (ETP) copper which is 99.90% pure copper. The approximate weight of the squirrel cage winding is 20 pounds.

The rotor bar and connection ring material was chosen to meet the performance requirements of the machine. Copper alloy bars, with a resistivity of $1.695 \mu\text{ohm-inches}$, were selected because of the high torque requirements of the motor. Although these high resistivity bars decrease the efficiency of the machine, they allow the motor to operate at an increased slip thus increasing the output torque. The machine is capable of withstanding a locked rotor condition of more than 10 seconds without sustaining any damage to the rotor winding or the rotor and stator cans. However, the variable frequency drive and corresponding breaker system will prohibit a locked rotor condition of more than 5 to 6 seconds from occurring.

Rotor Can

The rotor can, like the stator can, is made from 0.033 inch thick Hastelloy C-276 material. This material was chosen because of its high electrical resistivity and its high resistance to abrasion. The can will be shrunk on to the outer diameter of the rotor core and welded at both ends to complete the hermetic seal of the rotor cavity.

3.1.2 Stator Assembly

General

The transfer pump stator assembly contains coils which are connected together in a two parallel wye or star configuration. The stator core itself has a 4.15 inch inside diameter, a 9.00 inch outside diameter, and a core length of 18.0 inches. The stator core contains 24 slots which are 0.680 inches in width and 1.625 inches deep. The overall length of the stator assembly, including the active core, finger

plates, end plates, and endturns is approximately 28 inches. The total weight of the entire wound stator assembly is approximately 250 pounds (See Figures 15-3 and 15-8).

Stator Core

The stator core consists of an 18.0 inch stack of electrical laminations which are sandwiched between two stainless steel finger plates. The laminations are laser cut out of 0.025 inch thick silicon steel (10502DW), deburred and recoated to provide additional interlaminar resistance. The finger plates are made from 304 stainless steel and will be either laser or water-jet cut to final dimension. Once stacked, the stator core will be pressed under load and welded to maintain the required stacking pressure. After the stacking process is completed, the outer diameter of the stator core will then be machined to its final dimensions. Upon completion of the machining operations, micarta endplates will be assembled to the fingerplates (via machine screws) to provide an insulating barrier between the stator core and the random wound coils. The total weight of the stator core is approximately 190 pounds.

Stator Winding

The random wound stator winding consists of 24 coils which are connected in a two parallel wye configuration. The coils themselves are eleven turn coils consisting of seven strands of 0.0508 inch diameter (AWG #16) wire per turn. Each strand is coated with a polyimide coating (heavy ML) which provides both strand and turn insulation. The stator winding has a 9/12 coil pitch which corresponds to a 1-10 slot throw. This coil pitch was selected to minimize the endturn length in this small diameter, two pole stator core. After the core is wound and the coils wedged into place, the coils will be brazed together to form the two parallel star winding. The brazed connections will then be insulated with mica tape and the wound stator assembly shrunk into its inner housing and sand potted. The total weight of the stator winding is approximately 60 pounds (less potting).

Insulation Materials

The insulation system used in the design of the transfer pump motor is a high temperature (Class N) insulation system with which WEMD has hundreds of thousands of hours of operating experience. This insulation system is capable of providing 40 years of continuous operating life at 200°C (392°F). Materials such as mica tapes, mica papers, kapton tape, glass tape, and silicon based bonding resins are examples of materials in this insulation class. Use of this insulation system allows the proposed canned motor pump to employ 200°F process fluid through the rotor/stator gap for cooling the motor. Initial thermal analysis of the motor cooling indicates that the maximum winding surface temperature will be approximately 205°C (401°F) (indicating a 20-year design life). In addition, the WEMD insulation system has been tested in radiation environments of up to 1000 Megarads, far in excess of the 58.7 Megarads expected in the transfer pump operational environment.

In summary, the Class "N" insulation system was chosen because its wide temperature range allows for continuous, long-term operation at winding temperatures above 200°C (392°F).

Stator Can

The internals of the machine are hermetically sealed from the process fluid by metal rotor and stator cans made from Hastelloy C-276 material. Hastelloy was chosen primarily for its corrosion resistant properties. Another benefit of the Hastelloy material is its high resistivity (44.8 μ ohm-inches) which helps minimize can losses and thus increases the efficiency of the motor. The stator can is 0.033 inches thick and is welded to end rings on both ends of the stator assembly to complete the hermetic seal of the stator cavity.

Cooling

The primary source of cooling for the machine is the 200 °F (maximum) process fluid which flows over the outer diameter of the stator housing. In addition, cooling is also provided by the process fluid that is pumped through the rotor/stator gap. The endturns of the stator winding are cooled via conduction. The stator coils transfer the heat from the end turn areas of the winding, through the copper, to the stator core. The heat in the stator core is then dissipated through the stator housing to the surrounding process fluid. Sand potting will be used to increase the cooling in the endturn region.

3.1.3 Power Cable

The transfer pump power cable connects the variable frequency drive (VFD) to the terminals of the motor. This cable is run in an underground conduit and the pump column to protect it from the outside environment. The cable itself is comprised of three separate 2/0 AWG stranded conductors with a solitary stranded ground.

The cable is constructed with a silicone rubber insulation jacket specifically designed for high temperature applications within nuclear generating facilities. It is intended for use in harsh environments where temperature extremes preclude the use of standard electrical cables. In addition, this cable is nuclear qualified with a minimum 40-year thermal life expectancy at 200 °C (392 °F) and is radiation resistant up to 200 megarads.

3.1.4 Terminal Box

There is no terminal box provided by WEMD for the transfer pump. The power cable is designed to connect the motor leads directly to the VFD.

3.1.5 VFD Interface

Input power to the transfer pump will be supplied via a variable frequency drive (VFD) to allow for variable speed operation. The VFD motor controller will convert the 460 volt, three-phase, 60 hertz utility power to adjustable voltage and frequency, three-phase, AC power for stepless motor speed control. The VFD will utilize a Pulse Width Modulation (PWM) scheme for the power conversion. This space vector modulation strategy is implemented using IGBT (Insulated Gate Bipolar Transistor) technology to deliver efficient and cost-effective drive control.

The controller itself will have a continuous output current rating of 100% of the rated motor current. In addition, the drive can also supply 150% of rated current for 60 seconds. Sinusoidal current will be provided to the inverter section of the drive with minimal harmonic flow back to the incoming power source. Operation of the VFD will not add more than 3% total harmonic voltage distortion, and no more than 5% total harmonic current distortion (conforming to IEEE 519), to the utility power bus. The VFD will also have an output voltage regulator to maintain the correct output volts/Hz ratio despite variations in the incoming voltage.

The VFD controller contains a user-friendly keypad and display interface which will be mounted on the front of the drive enclosure. The display consists of a 2-line by 24-character, alphanumeric LCD that is used for displaying menu text, help text, data logging messages, and parameter information in easy-to-understand English text. In addition, a meter display screen shows dynamic values of four system variables and is completely programmable. The meter display also shows the current mode of operation and is backlit for easy readability. The keypad and display interface also contain three diagnostic indicators that are located above the

LCD display: Power On, Fault, and Run. The Power On indicator is lit when power is being supplied to the system. The Run indicator lights to show when the drive is in operation, and the Fault indicator is lit when one or more system errors have occurred (e.g., boot-up test failure, overvoltage fault, etc.).

The entire VFD system will be housed in a NEMA-3R, environmentally controlled, Hoffman enclosure. The enclosure will contain a thermistor which will control separate heating and air conditioning systems contained within the drive enclosure. This climate control system will allow a constant temperature to be maintained during the varying weather conditions that the drive will be exposed too. See Sections 6.5 and 6.6 for a more complete description of the VFD and its operation.

3.2 Hydraulics

3.2.1 Hydraulic Component Description

The hydraulics is a two stage centrifugal pump consisting of a perforated inlet plate, a suction entrance to the first stage impeller consisting of a suction adapter and a streamlined impeller bolt, a first stage impeller which discharges into a first stage diffuser, and a second stage impeller which discharges into an axial diffuser cast as the thrust bearing support plate. All hydraulic components are completely self-draining, and all hydraulic flowpaths were sized to pass a foreign object to 0.25 inch diameter.

The pump hydraulics will deliver the required 140 gpm and 450 feet of head at the discharge flange on top of the mounting flange at the design speed of 3860 rpm when operated in process fluid with a viscosity of 30 Centipoise and a specific gravity of 1.5. At this operating condition the second stage impeller discharge head is approximately 512 feet and about 55 shaft horsepower is required.

The suction specific speed of the first stage impeller at the minimum available NPSH of 10.6 feet is 7775. Westinghouse has produced impellers with no head fall off at suction specific speeds in excess of 9500. Refer to Sections 5.0 and 5.8 for a detailed discussion of the hydraulic design and NPSH prediction.

Refer to Sections 3.3.1.1 and 3.3.1.4 for a description of the hydraulic components. See Figure 3-1A and 3-1B for a view of the second stage impeller. The first stage diffuser is shown in Figure 3-2 and the second stage diffuser is shown in Figure 3-3.

Figure 3-2 -

8 - Vane First Stage Diffuser

Figure 3-3

27 Slot Second Stage Diffuser and Outside Closure

3.2.2 Design Description - Internal Circulation

The internal hydraulic analysis of the Transfer Pump is used to define the magnitude of the internal flows and pressures within the motor and hydraulic assemblies as well as calculate the axial thrust characteristic of the pump assembly. The analysis of the circuit flows is calculated in the manner of electrical circuitry. Resistances represent pressure losses due to friction or geometrical area changes. Batteries represent pressure increases or decreases due to fluid vortex rotation or pressure increases due to auxiliary impeller effects. A complete map of the pump internals is established by outlining all of the detailed flowpaths in each component subassembly. A complete listing of all resistance and vortex elements is added to a list of mass continuity equations to form a solution matrix. This matrix is analyzed using a Fortran computer code call "PCIRCUIT". The output from the code is a listing for the internal flows, pressures, and the summation of the axial thrust pressures. Of particular importance for this pump is the design and operation of the hydraulic seal assemblies that are used to reduce the recirculation flow from the high pressure side of the impeller to the low pressure side. A design study of seal types and clearances was done to optimize the seal performance.

3.3 Mechanical Design

This section discusses the mechanical design features of the motor unit assembly, the column assembly, the purge and dilution line systems and the instrumentation system planned for the Model 2A-513-A1 Transfer Pump lead unit. Refer to the figures in Section 15.0 as identified below.

3.3.1 Motor Unit Assembly

The Model 2A-513-A1 Transfer Pump motor unit assembly contains the submerged canned motor and the pump hydraulics. The motor unit assembly consists of the rotating assembly, the radial and thrust bearings, the stator assembly and the static hydraulics. Refer to Figure 15-4, the Motor Unit Assembly drawing.

3.3.1.1 Rotating Assembly

The rotating assembly contains the rotor assembly, the thrust runner, the two impellers, the impeller spacer and the impeller bolt.

The rotor assembly (Figure 15-7) consists of the rotor shaft, the squirrel cage assembly, the rotor end rings, the rotor can and the rotor journal assemblies. The rotor shaft is a solid, magnetic 410 stainless steel (SST) shaft that has 17 copper alloy bars, 19.5 inches long and two connection rings brazed together to form a traditional squirrel cage rotor assembly. The rotor end rings, made of Inconel 600, cover the connection rings and are welded to the shaft. The end rings are shrunk onto the shaft with a 0.002 to 0.005 inch diametral interference to minimize the load on the end ring to shaft welds. Inconel 600 is used for the end rings because the material has the required non-magnetic properties for compatibility with the motor, the material is compatible with the process fluid, and the material has thermal expansion properties similar to the 410 SST shaft. The end rings contain vent plugs which are welded closed after Helium leak testing which is performed to ensure integrity of the pressure boundary. A 0.033 inch thick Hastelloy C276 can

is welded to the rotor end rings to hermetically isolate the squirrel cage components from the process fluid environment. Westinghouse conducted abrasion wear tests on the Hastelloy C276 can material to confirm that the can thickness chosen for the rotor is adequate for the 10,000 hour life of the pump. Refer to Section 11.0 for a detailed discussion of the can abrasion testing and the results of the testing. The hermetically sealed rotor eliminates the decontamination and inspection effort of the rotor assembly internals after the pump is removed from the waste tank. The rotor squirrel cage components are cooled by process fluid flowing over the shaft and can as identified in Section 3.2.2. The overall length of the rotor is approximately 46 inches.

The top end of the rotor assembly contains the upper radial bearing journal and the upper journal clamp nut assembly. The lower end of the rotor assembly contains the lower radial bearing journal, the lower journal holder assembly, the thrust runner, the first and second stage impellers, the impeller spacer and the impeller bolt.

The two journals are made of direct sintered Silicon Carbide (SiC) to obtain high hardness, high strength and good chemical resistance to the process fluid. The SiC journals provide the required abrasion resistance for process fluid operation. The journals are clamped axially in compression by the 410 SST lower journal holder and the upper and lower 304L SST journal holder nuts. This clamping action also secures the journals against radial movement. The lower journal holder is shrunk onto the shaft with a 0.0010 to 0.0025 inch diametral shrink to preclude rotation of the journal holder relative to the shaft when subjected to a torque in excess of the maximum motor torque (110 lb-ft is used). The upper journal is located axially by the journal shoulder ring, and the lower journal assembly is located axially by a shoulder on the shaft. The journals are assembled onto the shaft/journal holder and clamped by nuts with all components at a temperature of 250 °F. The large difference in coefficient of expansion between the Silicon Carbide journals (2.2×10^{-6} /°F) and the 410 SST shaft and lower holder (6.1×10^{-6} /°F) ensures that sufficient axial clamping load will be applied to the journals when they cool to pump operating temperatures to prevent journal rotation relative to the holders at a torque in excess of 110 lb-ft. The large axial clamping load developed

will also prevent radial motion of the journals under operating loads. A 0.0045 to 0.0065 inch room temperature diametral clearance is provided between the lower journals and journal holder to ensure a slip fit of the lower journal onto the holder exists at the 250 °F assembly temperature. The diametral clearance between the upper journal and shaft is 0.0015 to 0.003 inch to ensure a slip fit at the 250 °F assembly temperature.

The thrust runner insert is also Silicon Carbide. The thrust runner insert is attached to the runner holder to form the thrust runner assembly. The thrust runner assembly seats against the shaft shoulder adjacent to the lower radial bearing holder. The thrust runner is keyed to the rotor shaft with a 304L SST pin and secured axially with the second stage impeller hub. The thrust runner assembly is assembled with a 0.001 to 0.003 inch diametral clearance on the rotor shaft to ensure that the runner insert is not put into tension at operating temperatures due to the differences in coefficient of expansion between the Silicon Carbide runner and the 410 SST shaft and to allow easy removal of the runner from the shaft.

The two CF3 SST impellers are keyed to the shaft. The second stage impeller seats against the thrust runner and uses the same key as the thrust runner. The 410 SST impeller spacer axially locates the first stage impeller on the shaft. The upper and lower seals of the first stage impeller are Chromide coated to minimize wear, as is the lower seal of the second stage impeller. The impeller spacer is also coated with Chromide to prevent significant wear during pump operation. A 304L SST pin prevents both the first stage impeller and the impeller spacer from rotating. The two pins used as keys (one for the thrust runner and second stage impeller, and the other for the first stage impeller and impeller spacer) are sized to preclude rotation of the components relative to the shaft at greater than 225 lb-ft of torque. The impellers have a 0.001 to 0.004 inch diametral clearance on the shaft to minimize the assembled residual unbalances of the impellers and to allow installation and removal of the impellers without significant heating. The impeller spacer is assembled on the shaft with a 0.002 to 0.004 inch diametral clearance to minimize residual unbalance and ensure the spacer can be easily installed and removed. The first stage impeller is secured axially with a 0.625-11UNC-2A impeller bolt. The impeller bolt clamps the assembly of

impellers, impeller spacer and thrust runner axially to the rotor at all operating temperatures. The impeller bolt has a right-hand thread which tends to tighten when the pump is operated in the proper rotational direction (clockwise when viewed from the top of the pump). This bolt would tend to loosen when the pump is operated in reverse or windmills in reverse due to backflushing the discharge line. The impeller bolt is positively locked after tightening by two lockpins which are inserted through the head of the bolt into the first stage impeller. The lockpins are secured by welding.

The classical critical speed analysis indicates that the critical speed of the two-bearing rotating assembly is a minimum of 60% above design operating speed of 3860 rpm. Refer to Section 10.0 for further details relative to the rotor dynamics.

3.3.1.2 Bearings

The radial bearings are pivoted pad (4 pads) hydrodynamic bearings designed and manufactured by WEMD as detailed in Section 9.1. The upper radial bearing housing is shrunk into the stator assembly lower closure. The lower radial bearing housing is bolted into the stator assembly lower closure with a 0.001 to 0.005 inch diametral clearance. The pivoted pad radial bearing pad inserts (static) and journals (rotating) are made of Silicon Carbide to provide the necessary hardness, toughness and process fluid compatibility.

The pivoted pad radial bearings have full self-alignment capability as discussed in Section 9.1.

The motor process fluid passes around and through the bearings for cooling and lubrication. There are four bypass holes around the lower bearing in the stator assembly lower closure to improve flushing capability around the lower radial bearing. The Silicon Carbide radial bearing pad inserts run against journals which are locked in compression onto the rotor shaft as discussed in Section 3.3.1.1.

The thrust bearing assembly is a six pad Kingsbury-type thrust bearing designed and manufactured by WEMD. Refer to Section 9.2 for a description of the thrust bearing performance. The thrust bearing pads seat on two rows of links which seat on the second stage diffuser. The thrust bearing features six cylindrical crowned Silicon Carbide thrust shoe inserts running against the Silicon Carbide thrust runner insert previously described. The thrust shoes pivot on and are supported by two rows of six links. The links deflect when all the shoes are not subject to the same load, thereby correcting for misalignment between the thrust runner and thrust bearing. The thrust bearing misalignment capability is discussed in Section 9.2.

The radial and thrust bearings are film riding, hydrodynamic bearings which utilize waste tank process fluid, pumped through the motor, as the bearing lubricant and cooling media. Refer to Section 3.2.2 for a discussion on process fluid flow through the bearings. The viscosity of the process fluid is between 1.0 and 30.0 Centipoise and adequately supports the applied operating loads. The bearing film thicknesses associated with operation in the process fluid will be much greater than that realized in water. Refer to Section 9.0 for a detailed discussion of the bearing designs and performances.

Silicon Carbide is an excellent bearing material to use with the waste tank process fluid. Silicon Carbide is not only compatible with the composite properties of the process fluid but also exhibits excellent wear properties in abrasive environments like the waste tanks. Westinghouse has conducted wear tests with Silicon Carbide in a prototypic type medium to confirm the acceptable performance of the bearings over the 10,000 hour pump operating life. Refer to Section 9.0 for an evaluation of the bearing operating life based on the abrasion wear testing.

3.3.1.3 Stator Assembly

The stator assembly comprises a stator core, a stator shell, two stator end closures, and a stator can. Refer to Figures 15-4 and 15-8. The stator core consists of an 18 inch stack of 0.025 inch thick silicon steel insulated laminations compressed between two 304 SST fingerplates and welded together. The stator core contains a set of random wound (mush) coils to construct a two-pole induction motor. Refer to Section 3.1.2 for a detailed electrical description of the motor. The coils are insulated with a heavy ML (Kapton) insulation system which allows continuous motor operation for over 20 years of service at winding temperatures of 210°C (410°F) and at radiation levels expected over the 10 year minimum life of the pump. Refer to Section 13.4 for a detailed discussion of the motor insulation life based on operating temperature and radiation.

The stator core is shrunk into the stator shell with a 0.005 to 0.011 inch diametral interference which not only prevents the stator assembly from rotating in the 0.40 inch thick stator shell with a torque in excess of 225 lb-ft but also provides a good thermal conductivity path between the core and the stator shell to help cool the motor windings with process fluid flowing in the annular space between the outside of the stator shell and inside of the stator jacket.

The stator shell has two through holes in the vicinity of the motor electrical end turns which are welded closed with plugs after a sand-silicone resin potting mixture is installed. The stator is completely potted to the top of the power lead tubes with the sand-silicone resin mixture to improve the heat transfer from the winding end turns and to minimize the entry of air or moisture into the stator winding cavity. A similar potting has been used on other WEMD canned motor pump and Control Rod Drive Mechanism applications. As described in Section 13.4 the potting material is acceptable for use in the expected radiation field over the 10 year minimum life of the pump.

To prevent process fluid from entering the stator winding cavity, the 304L stainless steel top and bottom closures are welded to the 304L stator shell. Fluid passing through the internal cooling circuit of the motor is prevented from entering the stator winding cavity by a 0.033 inch thick Hastelloy C276 stator can which is welded to the top and bottom closures. The inside diameter of the stator core and the inside diameter surfaces of the top and bottom closures, where they butt up against the stator core, support the stator can from the fluid pressures generated by the pump. The rotor cavity formed by the stator can, top closure and bottom closure is Nitrogen leak tested to ensure integrity of the stator can and associated welds is acceptable.

The stator can, the stator shell and the top and bottom closures are designed to withstand process fluid pressures up to 660 psi. This hermetically sealed stator arrangement facilitates the decontamination and inspection of the stator assembly when the pump is removed from the tank since process fluid is prevented from entering the electrical parts of the motor; refer to Section 14.4 for a discussion of motor unit assembly maintainability.

Westinghouse conducted abrasion wear tests on the Hastelloy C276 can material to confirm that the can thickness chosen for the stator is adequate for the 10,000 hour life of the pump. Refer to Section 11.0 for a detailed discussion of the can abrasion testing and the results of the testing.

The motor leads are brought out of the motor winding cavity inside a single 304L SST tube which is welded to the top closure. A double o-ring seal is welded to end of the motor lead tube. The o-ring seals provide a secondary sealing mechanism

between the process fluid in the tank and discharge line and the motor electrical lines. The primary sealing mechanisms between the process fluid and the motor electrical lines are: a carbon gasket compressed between the top of the motor unit assembly and the bottom of the column assembly around the electrical tube; a gold coated stainless steel C-ring which is also compressed between the motor unit and the column assembly which fits around the electrical tube. Refer to Section 3.3.2 for a description of the column assembly. The stator top closure provides holes for purge lines and dilution lines as discussed in Section 3.3.1.5. As described in Section 3.2.2 the main process fluid flow is from the outside of the stator shell to the column assembly discharge line located at the top of the stator assembly rotor cavity. The stator top closure contains four 1.12 inch diameter radially oriented through holes to provide the main discharge flow path from the outside of the stator shell to the rotor cavity.

The stator bottom closure contains dilution line holes which are discussed in Section 3.3.1.5. The bottom closure also contains nine axial 0.375-16UNC-2B tapped holes for attachment of the second stage diffuser described in Section 3.3.1.4.

Eight 304L SST channel dilution lines are attached to the outside of the stator shell. These lines transport dilution water from the stator assembly top closure to the static hydraulics. Refer to Section 3.3.1.5 for an additional description of the dilution lines.

3.3.1.4 Static Hydraulics

The static hydraulics for the two stage centrifugal pump consists of an inlet screen, a suction adapter, a first stage diffuser, a second stage diffuser and the stator

jacket. Refer to Figure 15-4 and Figures 15-11 through 15-16. The rotating hydraulic components, the first and second stage impellers, are described in Section 3.3.1.1. All hydraulic components are completely self-draining. Refer to Section 5.0 for a detailed discussion of the hydraulic design.

The inlet screen is a 304L SST sheet 0.051 inch thick (16 gauge) perforated with 0.25 inch diameter holes oriented to provide a 58 percent area opening. The inlet screen is sized to prevent particles larger than 0.25 inch in diameter from entering the hydraulics region and potentially blocking the hydraulic passages which are sized to pass particles up to 0.25 inch in diameter. The inlet screen flow area is very large to minimize both the head loss across the screen and the through-flow velocity. It is noted that the inlet screen will also help reduce vortexing at low tank levels. The inlet screen is attached to the suction adapter with the same eight bolts which secure the suction adapter to the first stage diffuser.

The suction adapter is a CF3 SST casting. Four radial vanes at the suction inlet channel act as guides during pump installation and support the inlet screen. The four vanes also reduce vortexing when the pump is operated at low tank fluid levels. The suction adapter is clamped to the bottom of the first stage diffuser with eight 0.375-16UNC-2A hex head bolts which go completely through the first stage diffuser and are threaded into the second stage diffuser. The bolts are secured against rotation with lockplates. There are four dilution lines cast into the suction adapter which discharge through the four vanes into the suction inlet and against the screen to dilute the process fluid and backflush the inlet screen. Refer to Section 3.3.1.5 for a discussion of the dilution flow path in the motor unit assembly. The suction adapter is designed such that when the bottom of the pump is less than 1.50 inches above the bottom of the waste tank the upper shroud of the first stage impeller is located less than six inches above the bottom of the tank. This ensures the pump is self-priming to empty the tank to below a six inch level. The suction adapter impeller seal is chromide coated to minimize wear.

The first stage diffuser is made from two pieces of 304L SST. This crossover diffuser uses eight vanes to turn the flow from the first stage impeller discharge into

the second stage impeller suction. See Figure 3-2. The minimum passageway clearance is 0.25 inch. The eight vanes are machined into one piece of the diffuser. The second piece of the diffuser forms the upper shroud. Both pieces are bolted together by eight 0.375-16UNC-2A bolts which are locked by welding. The bolts penetrate a low velocity flow region of the diffuser and do not significantly affect pump performance, as documented in Section 5.0. The bolts are chromide coated to minimize wear during operation. The first stage diffuser has four dilution lines that feed from the second stage diffuser into the suction adapter. The bolted joint interfaces between the suction adapter, first stage diffuser and second stage diffuser are not gasketed around the circumference. Some weepage of pressurized process fluid back into the tank is expected. However, the interface surface finish and bolt preload will be such that no joint separation will occur and gross leakage of fluid (and atomization) will be prevented. The first stage diffuser impeller discharge surface and impeller upper seal are chromide coated to minimize wear.

The 304L SST second stage diffuser turns the second stage discharge flow axially upward into the annular gap between the stator shell and stator jacket through 27 angled slots arrayed circumferentially. Refer to Figure 3-3. The minimum passageway clearance is 0.35 inch. These holes act as an axial diffuser for the second stage impeller. This axial diffuser arrangement has two advantages over a conventional crossover diffuser. One, the reduced axial length improves the rotor critical speed. Two, the inspection and decontamination activity is significantly reduced with the elimination of hard-to-inspect and reach diffuser vanes and passages found in a standard crossover diffuser. The second stage diffuser also consists of two pieces which are shrunk together and pinned with six 0.500 inch diameter pins. These pins are not in the flow path.

The second stage diffuser is bolted to the bottom of the stator assembly with nine 0.375-16UNC-2A socket head capscrews which are locked in place with lockcups swaged into both the diffuser and the capscrews. The thrust bearing pads fit on the top of the second stage diffuser.

The second stage diffuser incorporates four dilution line holes in the diffuser which interface with the eight dilution channels on the stator shell. The second stage diffuser also locates and axially secures the stator jacket. The second stage impeller discharge surface and impeller lower seal are chromide coated to minimize wear. The 304L SST stator jacket is shrunk onto both the stator upper closure and the second stage diffuser with a 0 to 0.010 inch diametral interference to form an annulus through which the fluid is pumped by the impellers to the discharge pipe. The shrink fit joints are not designed to be leak-tight and may weep slightly (but not atomize). The stator jacket can be removed to decontaminate the outside of the stator shell and inside of the stator jacket.

3.3.1.5 Purge Water and Dilution Water Lines

Purge water is required to be supplied to the motor unit assembly to flush solids out of the internals of the motor and to soften hardened solid deposits in the motor unit assembly. Purge water flow is required immediately prior to starting a pump installed in a hazardous waste tank and immediately after stopping a pump. Also, purge water flow may be required periodically while a pump is inoperative in a tank. Purge water flow is also required when starting a pump if the tank level is below the upper radial bearing, approximately 50 inches above the bottom of the tank. The Installation and Operations Manual, to be provided with each Transfer Pump, will identify the frequency that purge water flow must be supplied to an inoperative pump. It is expected that the periodic flow intervals will vary from tank to tank and that this will be a learning experience based on changes in torque (current) required to start the pump after inoperative time intervals. The Installation and Operations Manual will also identify the maximum frequency (speed) at which the motor can be started while purge water flow is on (due to the fluid tank level being below the upper radial bearing) to ensure no process water will flow up a 90 psig supply purge water line.

The available purge water is 10 gpm at 90 psig. Since the head-flow curve of the purge water supply system is not known and probably varies between tank farms and pump purge water resistance will vary between tanks based on waste fluid level and specific gravity, it cannot be identified exactly what the purge water flow will be to the pumps. However, refer to Section 5.7.2 for a nominal flow estimate.

The purge water is supplied to the motor through a 1/2-inch pipe in the column assembly. Refer to Sections 3.3.2 and 3.3.3 for a description of the purge water system associated with the column assembly. At the bottom of the column transition piece a purge water supply header is provided. The supply header mates with a hole in the top of the motor unit assembly stator top closure. See Figures 15-4 and 15-8. The hole leads directly to the annular gap between the top closure inside diameter and the outside diameter of the upper bearing housing. This purge line will clear solids from the top of the upper radial bearing and direct process fluid flow through the upper radial bearing down the motor discharge annulus between the stator shell and jacket, past the rotor and stator cans, into the lower radial bearing, and into the hydraulics region.

Only one source of purge water can be provided in the motor to ensure displacement of solids. If more than one source is used, the purge water flow will bypass a plugged area and "short-circuit" to only the low resistance (clean) location, thereby leaving the plugged area undisturbed.

WHC has identified that backflushing of the main discharge line after each pumping operation is complete will be performed. It is expected that this will be very effective in removing the larger, heavier solids from the motor unit internals.

Model testing was performed to simulate accumulation of solids in the motor assembly and to simulate flushing of the solids with purge water lines. Refer to Section 12.0 for a discussion of the testing.

Dilution water is required to be supplied to the pump suction area to reduce the viscosity and specific gravity of fluids in some tanks into which the Model 2A-513-A1 Transfer Pump will be installed. The required flow rate is 105 gpm at approximately 100 psig supply pressure to the pump. Refer to Section 5.7.2 for an evaluation of the supply pressure required to produce 105 gpm dilution flow to the pump suction.

The dilution water is supplied by a single 1 1/2-inch pipe in the column assembly. Refer to Sections 3.3.2 and 3.3.3 for a discussion of the dilution water network associated with the column assembly. The pipe exits at a header in the bottom of the column assembly transition piece. The header supplies four holes in the top of the stator assembly top closure. Refer to Figures 15-4 and 15-8. Each top closure dilution hole goes to two channels attached to the outside of the stator assembly. Each of the eight channels attaches to a close-fitting slot in the second stage diffuser. The four 0.500 inch diameter dilution holes in the second stage diffuser go to four matching dilution holes (0.500 inch diameter nominal) in the first stage diffuser. The first stage diffuser dilution holes go to four matching cast-in holes in the suction adapter. The suction adapter holes terminate at the four vanes in the suction adapter below the pump suction. The holes are 0.500 inch in diameter and are oriented radially so that the NPSH requirements of the first stage impeller are not degraded by an adverse entrance velocity profile. In addition to the 0.500 inch diameter dilution line hole at the inside diameter of each vane below the suction, there are four 0.25 inch diameter holes angled downward toward the inlet screen. These 16 holes (four each vane) provide dilution flow to backflush the inlet screen perforations. All leakage into the dilution lines will simply flow out the dilution holes in the suction adapter.

3.3.2 Column Assembly

The column assembly, shown in Figure 15-5, is an assembly consisting of a column extension assembly and a column and transition piece which suspends the Transfer Pump from the waste tank pump mounting flange. The column assembly section includes power cable, purge water, dilution water and process fluid discharge lines to the pump motor. The outside diameter of the column assembly is 8.63 inches at the pipe section and 11.0 inches at the transition piece. A 63 RMS finish will be supplied on all outer surfaces of the column assembly which contact the waste tank process fluid. The column assembly is designed to hermetically seal the internals of the assembly from the process fluid.

3.3.2.1 Column

The outer section of the column assembly, illustrated in Figure 15-5, is fabricated from an eight-inch welded 304L schedule 40 SST pipe. The 1/2 inch purge line and 1 1/2 inch dilution line run inside of the column along with the three-inch discharge line, all which are welded 304L schedule 40 SST pipe. The electrical cable is located in a 304L SST two inch schedule 40 pipe that also runs the entire length of the column. Each of the internal lines are supported by baffles spaced as needed for support during assembly. After the internal pipes are in place, the column is filled with polyurethane foam to provide lateral support, stiffness and damping. The bottom end of the column is welded to the transition piece.

The top of the column section is closed with a welded plate to support the internal lines during shipping and also to aid in the welding of the column extension assembly. Above the column closure plate is an eight inch pipe section which is cut to length, to provide the proper pump insertion, and welded to the column extension assembly.

3.3.2.2 Column Extension Assembly

The column extension assembly, shown in Figure 15-6, is designed to allow adjustment of the column assembly length by mating with the adjustable length, welded eight-inch 304L schedule 40 SST pipe on the end of the column piece as previously mentioned. The column extension assembly consists of the pump mounting flange and a short section of eight inch pipe. The pump mounting flange contains holes to accept the process fluid, purge, dilution and power cable pipes which penetrate it upon assembly.

3.3.2.3 Transition Piece

The transition piece, depicted in Figure 15-5, is 304L SST with machined passages for mating the column dilution water, the purge water, the process fluid discharge and the electrical lines with the motor unit assembly. Scallops are machined into the transition piece for the SST bolts which secure the column assembly to the motor unit assembly. A graphite gasket is provided for sealing between the motor unit electrical lines and the transition piece. A gold coated metal C-ring fits around the electrical extension tube and is compressed between the transition piece and the motor unit to ensure no fluid contacts the electrical lines. The graphite gasket is outside the metal C-ring surrounding the electrical extension tube. The pressure across this gasket is near the tank submergence pressure (23 psig). While it is extremely unlikely that such low pressure fluid will leak across the gasket, any leakage that is produced will encounter the metal C-ring which is gold coated to ensure zero leakage of process fluid. The top of the transition piece is welded to the column assembly to hermetically seal the column internals from the process fluid. The electrical conduit passage that is machined into the transition piece holds the brazed connections between the motor leads and the power cable.

3.3.2.4 Power Cable Conduit

The pump electrical leads, discussed in Section 3.1.4, exit from the motor unit stator housing through an extension tube. The extension tube has an o-ring seal assembly at the end. The extension tube design shown in Figure 15-2, permits the extension tube to radially flex so the o-rings will engage with the bore of the column transition piece where the electrical connections will be made. This provides a tertiary seal (the graphite gasket is the primary seal and the C-ring is the secondary seal) from the process fluid. The power cable conduit is made from 304L schedule 40 SST pipe and the o-rings are made from an ethylene-propylene compound that will perform satisfactory in the caustic and high radiation environment. Thus the electrical lines are sealed from the process fluid with the o-rings a graphite gasket and a metal C-ring between the transition piece and the motor unit.

3.3.2.5 Discharge Nozzle

The discharge nozzle, illustrated in Figure 15-5, is located on top of the pump mounting flange and accepts the Customer supplied purex fitting with modification. The purex fitting must be modified to shorten the extension below the kickplate on the weld end so the discharge piping can be attached. The discharge nozzle is a standard three inch schedule 40 304L SST 90° elbow with a 45° elbow section attached to meet the required interface.

3.3.3 Purge and Dilution Line Network

One 1/2 inch purge line and one 1 1/2 inch dilution line are supplied for the Transfer Pump. Each line is a 304L schedule 40 SST welded pipe that leads through the column assembly and connects to the appropriate transition piece passage. These passages network a series of channels to lead the fluid to the various lines that supply the motor unit assembly. The purge line and dilution line terminate at a 2-inch purex fittings above the pump mounting plate.

The purge line is designed for 10 GPM of water at 90 PSIG and the dilution line is designed for 105 GPM of water at 100 PSIG.

3.3.4 Pressure and Accelerometer Instruments

3.3.4.1 Accelerometer

The top flange of the column assembly has an accelerometer, shown in Figure 15-5, to monitor pump vibration. The accelerometer is enclosed in a SST box to prevent any handling damage that may occur. The box and cable penetration are sealed with RTV. Sufficient cable will be supplied for connection of required electronics. The accelerometer and cable will exceed the 4.38×10^6 Rad requirement of the pit environment.

The lead unit will be water tested with an accelerometer, that is mounted outside the pump, just above the transition piece. This will allow for additional vibration monitoring of the motor unit assembly during water test operation. Sufficient cable will be provided for appropriate connection of electronics. The accelerometer is provided to allow better correlation of motor vibration to pump mounting bracket accelerometer readings during operation is a hazardous waste tank.

4.0 MATERIALS OF CONSTRUCTION

The materials list for the Model 2A-513-A1 lead unit Transfer Pump components is provided in the assembly drawings identified in Section 15. The following list summarizes the materials to be used with major Transfer Pump components.

The major components in contact with the hazardous waste process fluid are delineated in Table 4-1.

The metals used in the Model 2A-513-A1 Transfer Pump which are in contact with the process fluid, 304 SST, 304L SST, 316 SST, 410 SST (H1350), 17-4 PH SST (H1100), Inconel 600, Hastelloy C-276 and CF3 castings, are all compatible with both the radiation and the chemical environments presented by the hazardous waste tanks as delineated in the Reference 1 Design Specification. The metals used in the Transfer Pump which do not contact the process fluid, Silicon Steel, Copper and Copper-Nickel alloys are all compatible with the radiation environment of the hazardous waste tanks. Also, based on WEMD experience with these metals in the pH range of 9 to 14 there should be no galvanic corrosion problem. The metals will tend to passivate (develop an oxide film) to preclude galvanic corrosion problems.

The Transfer Pump radial and thrust bearings are made with direct sintered Silicon Carbide (SiC). The SiC bearing material is compatible with both the radiation environment and the high pH chemical environment of the waste tank process fluids.

The Carbon gasket material used between the motor unit assembly and the column assembly is manufactured from graphite flakes that have been acid washed, expanded under heat, then compressed into sheets. The material is normally used in fluid applications of extreme temperature and pressure. The graphite composition of this sheet material provides low creep relaxation to maintain pressure boundary integrity for the

required application life. The Carbon gasket is resistant to the caustic hazardous waste environment. Also, pure carbon products are generally assumed to be radiation resistant to 10^{10} Rads.

The metal C-ring, used as a secondary electrical lead pressure boundary at the top of the motor unit assembly is stainless steel with a gold coating. The gold is compatible with the radiation environment and is inert to the waste tank chemical environment.

The O-rings identified for use as the back-up pressure boundary to prevent contact of process fluid with the electrical leads at the top of the motor unit assembly are ethylene propylene elastomers. The specific material listed is Parker compound E740-75. This material resists chemical attacks in a wide variety of alkaline and acid solutions and is considered acceptable for the hazardous waste tank chemical environment. Parker compound E740-75 is reasonably radiation resistant. The material will begin to take a permanent compression set at 10^7 Rad, and exposure to 10^8 Rad will reduce the ultimate tensile strength and ductility by approximately a factor of 2.

The materials used in the stator insulation system include ML (Kapton) coated magnet wire and sand potting. The acceptability of these materials in the hazardous waste tank radiation environment is discussed in Section 3.1.2.

ITEM	NOMENCLATURE	MATERIAL	FORM	STATUS
Hydraulics				
1	1st Stage Impeller	A 351 Gr CF3	Casting	Wetted
2	2nd Stage Impeller	A 351 Gr CF3	Casting	Wetted
3	1st Stage Impeller Key	A 276 Gr 304L Cond A Cold Finish	Bar	Wetted
4	2nd Stage Impeller Key	A 276 Gr 304L Cond A Cold Finish	Bar	Wetted
5	1st Stage Diffuser	A 276 Gr 304L	Bar	Pressure Boundary
6	2nd Stage Diffuser	A 276 Gr 304L	Bar	Pressure Boundary
7	Impeller Spacer	A 276 Gr 410 H1350	Bar	Wetted
8	Suction Adapter	A 351 Gr CF3	Casting	Pressure Boundary
9	Impeller Bolt	A 193 Class 1 B8 (304 SST)	Bolt	Wetted
10	Suction Screen	Commercial 304 SST	Sheet	Wetted
11	Suction Adapter Bolts	A 193 Class 2 B8M (316 SST)	Bolt	Pressure Boundary
12	Suction Adapter Bolt Lockplates	A 167 Gr 304L	Sheet	Wetted
13	2nd Stage Diffuser Bolts	A 193 Class 2 B8M (316 SST)	Bolt	Wetted
14	2nd Stage Imp Bolt Lockcups	A 167 Gr 304L	Sheet	Wetted
Bearings				
15	Upper Radial Bearing Housing	Commercial 304 SST	Bar	Wetted
16	Upper Radial Bearing Pads	Direct Sintered Silicon Carbide	Sintered	Wetted
17	Upper Radial Brg Pad Holders	Commercial 410 SST H1350	Forgings	Wetted
18	Lower Radial Bearing Housing	Commercial 304 SST	Bar	Wetted
19	Lower Radial Bearing Pads	Direct Sintered Silicon Carbide	Sintered	Wetted
20	Lower Radial Brg Pad Holders	Commercial 410 SST H1350	Forgings	Wetted
21	Radial Bearing Pivot/Lock Pins	Commercial 304 SST	Bar	Wetted
22	Upper Radial Bearing Journal	Direct Sintered Silicon Carbide	Sintered	Wetted
23	Lower Radial Bearing Journal	Direct Sintered Silicon Carbide	Sintered	Wetted
24	Upper Journal Shoulder Ring	A 276 Gr 410 H1350	Bar	Wetted
25	Lower Journal Holder	A 276 Gr 410 H1350	Bar	Wetted
26	Upper Journal Holder Nut	A 276 Gr 304L	Bar	Wetted
27	Lower Journal Holder Nut	A 276 Gr 304L	Bar	Wetted
28	Thrust Bearing Shoe Insert	Direct Sintered Silicon Carbide	Sintered	Wetted
28	Thrust Shoe Holder	Commercial 410 SST H1350	Plate	Wetted
30	Upper and Lower Links	Commercial 304 SST	Bar	Wetted
31	Thrust Runner Insert	Direct Sintered Silicon Carbide	Sintered	Wetted
32	Thrust Runner Holder	Commercial 410 SST H1350	Bar	Wetted
33	Thrust Bearing Pins	Commercial 304 SST	Bar	Wetted
Stator Assembly				
34	Stator Shell	A 240 Gr 304L	Plate	Pressure Boundary
35	Top Stator End Closure	A 351 Gr CF3	Casting	Pressure Boundary
36	Bottom Stator End Closure	A 351 Gr CF3	Casting	Pressure Boundary
37	Stator Can	B 575 Hastelloy C-276	Sheet	Pressure Boundary
38	Stator Punchings	Modified M-36 Silicon Steel	Sheet	Non Wetted
39	Stator Windings	NEMA MW 16-C (Heavy) Magnet Wire	Wire	Non Wetted
40	Stator Finger Plates	A 167 Gr 304	Plate	Non Wetted
41	Stator Jacket	A 240 Gr 304L	Plate	Pressure Boundary
42	Stator Vent Plugs	A 276 Gr 304L	Bar	Pressure Boundary
43	Stator Purge Water Lines	A 269 Gr 304	Tube	Wetted
44	Stator Dilution Channels	A 276 Gr 304	U-Shape	Wetted
45	Stator Electrical Lead Tube	A 269 Gr 304	Tube	Non Wetted
46	Electrical Tube C-Ring	Commercial 304 SST Gold Coated	Sheet	Non Wetted
47	Pump Closure Gasket	Carbon Graphite	Gasket	Pressure Boundary
48	Electrical Tube O-Rings	Ethylene Propylene E740-75 (Parker)	Molded	Non Wetted
Rotor Assembly				
49	Rotor Shaft	A 276 Gr 410 H1350	Bar	Wetted
50	Rotor Bars	B 152 Cu/Ni Modified C70200	Bar	Non Wetted
51	Connection Rings	B 152 Cu C11000 Temper H02	Plate	Non Wetted
52	End Rings	B 166 Inconel Alloy 600	Bar	Wetted
53	Rotor Can	B 575 Hastelloy C-276	Sheet	Wetted
54	Rotor Vent Plugs	A 276 Gr 304L Cond A Hot Finished	Bar	Wetted
Column Assembly				
55	Transition Piece	A 276 Gr 304L	Bar	Pressure Boundary
56	Pump Mounting Flange	A 240 Gr 304L	Plate	Non Wetted
57	Main Discharge Pipe	A 312 TP 304L	Pipe	Pressure Boundary
58	Outer Closure Pipe	A 312 TP 304L	Pipe	Pressure Boundary
59	Discharge Elbows	A 312 TP 304L	Pipe	Pressure Boundary
60	Purge Water Line	A 312 TP 304L	Pipe	Pressure Boundary
61	Dilution Line	A 312 TP 304L	Pipe	Pressure Boundary
62	Electrical Conduit	A 312 TP 304L	Pipe	Non Wetted
63	Motor Support Bolts	A 193 Class 2 B8M (316 SST)	Bolts	Pressure Boundary
64	Motor Support Bolt Lockplates	A 167 Gr 304L	Sheet	Wetted
65	Foam, "Last-A-Foam" FR-3704	Polyurethane (General Plastics Mfg)	Foam	Non Wetted
66	Electrical Power Cable	Silicone Rubber/Glass Braid/Aramid	Cable	Non Wetted
		Fiber Covering Copper per B 8 & B 33		

Table 4-1 Transfer Pump Materials List

5.0 HYDRAULIC EVALUATION

The analysis of the transfer pump design involves a number of criteria and parameters based on WEMD experience - backed up by test results and substantiated by pump theory, found in most text books. The hydraulic evaluation for the transfer pump is summarized as follows: Section 5.1 scopes-out the design, the type of stage, and the number of stages using the parameters of suction specific speed and pumps specific speed as guidelines. The well-known "Worthington Pump Curve" of efficiency versus pump specific speed is dominant in the selection of pump hydraulic efficiency and hence the motor's brake horsepower. Section 5.1.1 describes the hydraulic adjustments made to accommodate the dilution lines feature. Section 5.2 discusses the brake horsepower requirements. The hydraulic load estimates are based on an empirical formula from the Hydraulic Institute Standards for the radial forces and Westinghouse test results for the axial thrusts; hydraulic loads are reported in Section 5.3.

The design parameters selected in this report are biased on the side of conservatism.

5.1 Hydraulic Design Basis

The specification requirements of this transfer pump are:

flow (Q) = 140 gpm

head (H_p) = 450 feet, delivered at the pump discharge nozzle

max. elevation (Z) = 49 feet

NPSH = 10 feet, minimum

The required pump head is:

$$\text{(Eqn. 1)} \quad H = H_p + Z + \text{Pipe Friction Loss}$$

Neglecting the pipe friction loss for this sizing analysis, the pump head is:

$$H = 450 + 49 = 499 \text{ feet}$$

Based on experience, a suction specific speed of 7000 is initially selected which results in the determination of the motor speed (N).

$$(Eqn. 2) \quad \text{suction specific speed } (S_N) = 7000 = \frac{N\sqrt{Q}}{(NPSH)^{3/4}}$$

$$N = \frac{7000 (10)^{3/4}}{\sqrt{140}} = 3327 \text{ rpm}$$

Thus, a 2 pole motor design (3600 rpm = synchronous speed) is selected. (This speed will be increased to accommodate the dilution line feature. See Section 5.1.1.)

The hydraulic type of pump, the number of stages, and the hydraulic efficiency are determined using the concept of pump specific speed and a Worthington curve. The specific speed, N_s , of a pump's stage is defined as:

$$(Eqn. 3) \quad N_s = \frac{N\sqrt{Q}}{\left(\frac{H}{N_s}\right)^{3/4}} \quad \text{where } N_s = \text{number of stages}$$

Solving for N_s as a function of stage number:

$$N_s = \frac{3327\sqrt{140}}{\left(\frac{499}{N_s}\right)^{3/4}} = 373 (N_s)^{3/4}$$

Figure 5-1 shows the Worthington Pump Curve, which relates N_s and hydraulic efficiency (a sketch of the impeller shape is also included below the N_s values).

Table 5-1 shows the solution of Equation 3 and the resulting hydraulic efficiency from Figure 5-1. Note from this table that a single stage pump is disregarded since its efficiency is so low and is outside Figure 5-1 parameters. Also, the impeller diameter for a single stage would slightly exceed the required pump diametral envelope of eleven inches. WEMD has little experience for the specific speed ranges of the 3 and 4 stage types. A specific speed of 2400 would optimize efficiency but the 12 stage mechanical complexity of such a pump would not be sensible. The 2 stage, $N_s = 627$ was chosen since WEMD has a

comparable impeller design (called the A150 pump) which had a sufficient number of diameters tested that included the range of operation for the transfer pump. A nominal diameter of 9 1/2 inches was selected for both impeller stages. Figure 5-2 depicts this impeller-type. A cross-over set of diffuser vanes is needed between impellers, and an axial second stage diffuser will then deflect the exiting flow from the second stage impeller to an annular collector. Pump-out ribs behind the second stage impeller are employed to force a strong vortex behind the impeller to reduce circulation of large slurry particles in the motor/bearing area.

TABLE 5-1
Stage Number and Efficiency Selection

<u>No. Stages</u>	<u>N.</u>	<u>Hyd. Eff.</u>	<u>Comment</u>
1	373	Off-Curve	N _s lower than conventional
2	627	55%	<u>W</u> design experience here
3	850	60	No <u>W</u> design experience here
4	1055	63	No <u>W</u> design experience here
12	2400	70	Optimum efficiency, but mechanically too complex

Figure 5-1 Worthington Pump Curve

5.1.1 Hydraulic Adjustments for Dilution Lines

One modification to the original workscope was the inclusion of dilution lines, running along the periphery of the hydraulics and parallel to the pump axis. Since the insertion diameter is fixed at eleven inches, the hydraulic dimensions had to be reduced in order to provide radial space for the dilution lines; simultaneously with the impeller size reduction, the impeller speed would have to be increased to maintain the pump head, according to the specifications. The hydraulic adjustment to effect such a change was to keep the product of impeller diameter and rpm a constant. The advantages of such an adjustment are that it hydraulically preserves the pump head and entrance velocity triangles to the diffuser vanes along with an improvement in hydraulic efficiency. The disadvantages of the increased speed are: the impeller vane entrance angles are slightly stalled, the erosion velocities have increased, and the NPSH required by the impeller has slightly increased.

In order for the hydraulics to fit the intended dilution line envelope, the first stage impeller had to be reduced to 8.25, and to maintain tangential velocities, the rpm had to be increased to 3840 rpm from the 2-pole slip speed of 3400 rpm; this higher speed will be obtained using a variable frequency drive motor control.

The main impact of speed increase and exit diameter reduction is on the hydraulic efficiency estimation described in Section 5.1 and the predicted NPSH, described in Sections 5.8. The specific speed for the dilution line pump is now calculated to be (see eqn. 3):

$$N_s = \frac{3860\sqrt{140}}{(499/2)^{3/4}} = 728$$

And the resulting hydraulic efficiency from Figure 5-1 is now:

$$\eta_H = 57\%$$

And the new power requirements are calculated in Section 5.2.

5.2 Prediction of BHP Requirements

The required brake horsepower is:

$$(Eqn. 4) \quad BHP = \frac{H \times Q \times SG}{\eta_H} \times C_{vis}$$

where η_H is the hydraulic efficiency = 57%

SG is the specific gravity of the pumped fluid = 1.5

C_{vis} is the viscosity correction factor

Q and H have been previously defined

The viscosity correction factor can be calculated from charts in the Hydraulic Institute Standards as (using a viscosity of 30 cp):

$$C_{vis} = \frac{0.995 \times 0.96}{0.81} = 1.18$$

Equation 4 now becomes:

$$BHP = \frac{499 \times 140 \times 1.5}{3960 \times 0.57} \times 1.18 = 55 \text{ HP}$$

This value also represents the maximum pump power required by the motor since it is calculated at the maximum flowrate and the pump's power characteristic is one that monotonically increases with flow.

5.3 Hydraulic Loads

Hydraulic loads are composed of axial (along the shaft axis) and radial (normal to impeller discharge) forces which are estimated as follows.

The axial hydraulic load is estimated from Westinghouse's A150 pump tests, adjusted to the 8.25 and 8.75 inch diameter impellers and also to the maximum specific gravity (1.5).

$$F_{x1} = 225\# \times 1.5 = 338\#, \text{ Impeller No. 1}$$

$$F_{x2} = 245\# \times 1.5 = 367\#, \text{ Impeller No. 2}$$

The radial impeller loads are estimated using an expression found in the Hydraulic Institute Standards:

$$(Eqn. 5) \quad F_R = \frac{K \times H \times SG \times DIA \times B}{2.31}$$

where H = pump design head
DIA = impeller diameter
B = impeller exit width, including shrouds
K = experimental factor = 0.05 for circular casings and diffuser vanes

Equation 5 then becomes,

$$F_R = \frac{0.05 \times \frac{(499)}{2} \times dia. \times 0.75 \times 1.5}{2.31}$$

$$F_{R_1} = 50\# / \text{impeller no. 1 @ 8.25 inches diameter}$$

$$F_{R_2} = 53\# / \text{Impeller no. 2 @ 8.75 inches diameter}$$

Figure 5-2 Transfer Pump Impeller Type

E.M. 6673

The assumption is also made that the impeller radial loads are circumferentially separated by the wrap angle of the intervening diffuser, assumed to be 115° , as identified in Figure 5-3. It should be noted that the hydraulic forces do not include the weight of the rotating elements nor the hydraulic forces in the canned motor.

Figure 5-3 Radial Load Orientation

5.4 Developmental Air Test Program

5.4.1 Introduction

The use of air testing during hydraulic development on new canned motor pumps has been a standard practice at WEMD for over 20 years. The information gained during this type of testing is useful to optimize the hydraulic design prior to actual water testing of a prototype pump. Past experience has shown good agreement between air and water testing for determination of total head versus flow performance.

For the transfer pump design, a multistage hydraulic arrangement was selected which utilized two radial flow impellers separated by vaned/slotted diffusers. Due to the multiple operation points the hydraulic design was required to perform to, a comprehensive test program was conducted to quantitatively determine stable head versus flow performance and hydraulic radial loads.

The following head/flow points define operation of the transfer pump per the equipment specification.

Design Point	450 ft @ 140 gpm
Operating Points	19 ft @ 53 gpm
	19 ft @ 132 gpm
	305 ft @ 132 gpm
	394 ft @ 53 gpm
	366 ft @ 138 gpm

5.4.2 Program Scope

For the air test program, a hydraulic model was created as depicted in Figure 5-4. The hydraulic components were fabricated from a combination of aluminum, and a rapid prototyping material defined as ABS. The remaining test rig parts were fabricated from plexiglas and steel. The testing was conducted at a full speed value of 3420 rpm versus the final design speed of 3860 rpm. The test results were extrapolated to 3860 rpm using the affinity laws as guidelines.

Figure 5-4 Air Test Assembly of Two-Stage Transfer Pump

The test rig did not prototype the column geometry and its associated fluid losses. These losses were analytically subtracted from the pump total head performance after the scaleup to full speed operation was done. Refer to Section 5.5 for more details.

The test program was divided into nine test series. Initial testing was to screen various diffuser designs and perform optimization on the hydraulics for maximum performance. Subsequent testing was used to determine the effect on hydraulic performance of different design changes as the pump design continued to evolve as a function of time. Upon completion of the test program, the best hydraulic configuration from a performance and pump functionality standpoint was achieved. The following paragraphs detail the individual test series:

Series #1 - This test series was used to verify the basic integrity of the hydraulic design and to test the first of two second stage diffuser designs. Design #1 incorporated a 27 slot diffusion configuration which turned the tangential discharge from the second stage impeller into the axial plane before the flow entered the motor cavity. The testing was done at a speed of 3420 rpm.

Series #2 - This test series was used to determine the performance of the second diffuser design. Design #2 incorporated a 19 vane diffusion configuration which performed the same function of turning the discharge flow from the second stage impeller. This testing was also done at 3420 rpm.

Series #3 - This test series was done with the slotted diffuser design, chosen as the best design from the first two test series. An optimization of the approach angle on the front of the diffuser slot was done to open the diffuser throat by "overfiling" the leading edge. This design change resulted in a significant performance improvement. This testing was done at 3420 rpm.

Series #4 - This test series added a prototypic suction screen to the first stage impeller inlet. Only minor performance differences were noted, as was expected. This testing was done at 3420 rpm.

Series #5 - This test series consisted of testing the current rig configuration at the other five operating points outlined in section 5.4.1. The results of this test series showed stable operation at all conditions considered. The test speed was varied as required to achieve the head and flow.

Series #6 - This test series used the Series #4 geometry, but added three sets of additional pump out vanes to the remaining shroud surfaces of the two impeller assemblies. All four impeller shrouds were now configured with pump out vanes. The recirculation flow decreased as was expected and the head/flow performance improved. This testing was done at 3420 rpm.

Series #7 - This test series was a separate study to determine the radial loads on each impeller stage at all six operating points outlined in section 5.4.1. The test rig was extensively modified to add approximately eighty additional pressure taps at key locations around the outer diameter of the impellers. Running at a selected rotor speed and at a flowrate close to the specification value, circumferential pressure data was taken and scaled appropriately to determine the load and phase angle shift between the two impellers. This testing was done at the same speed as the performance testing.

Series #8 - The configuration of test series #6 was used with the addition of eight pins directly in the flowstream of the first stage diffuser passage. These pins will be cap screws in the lead unit design, acting to hold the pieces of the first stage diffuser together. This testing was done at 3420 rpm.

Series #9 - This test series was used to mimic the final impeller seal design which was chosen for the lead unit pump. The close clearance labyrinth seals shown in Figure 5.4 were machined out to a diametrical clearance of 0.060 inch at both locations on the first stage impeller and at the suction end of the second stage impeller.

5.4.3 Results/Conclusions

- 5.4.3.1 The final test configuration (series #9) provided a prototypic hydraulic configuration of the final design chosen for the lead unit transfer pump. A tabulation of the tested total head versus flow values for test series #8 (0.010 inch seal clearance) and test series #9 (0.060 inch seal clearance) is presented in Table 5-2 and illustrated in Figure 5-5. The degradation in performance due to increased seal clearance is evident. Both curves illustrate stable operation over the entire flow range tested.
- 5.4.3.2 The initial test series (#1 and #2) were used to determine the best design for the second stage diffuser. The tested head versus flow characteristics are illustrated in Figure 5-6. For nearly identical performance, the slotted diffuser design was chosen due to its thicker "vanes" which will provide better wear characteristics in the erosive environment of the process fluid.
- 5.4.3.3 The additional operating point testing done during test series #5 showed stable operation at all five non-design operational points that the transfer pump will be subjected to. A tabulation of the test values is provided in Table 5-3 and is illustrated in Figure 5-7. It should be noted that this test series was done in the middle of the test program. As such the following design characteristics were not in place at the time of testing: 1) the addition of three more sets of pump out vanes to the impellers, 2) the addition of pins in the flow stream of the first stage diffuser, and 3) the remachining of the impeller seals from 0.010 inch clearance to 0.060 inch clearance.

5.4.3.4 Test series #7 on hydraulic loads showed that for all operational modes, the radial loads generated in the hydraulics are relatively small (< 100 lbs per stage). A tabulation of the radial load values for each operational mode is provided in Table 5-4. It should be noted that this test series was done in the middle of the test program. As such the following characteristics were not in place at the time of the testing: 1) the addition of pins in the flow stream of the first stage diffuser, and 2) the remachining of the impeller seals from 0.010 inch clearance to 0.060 inch clearance.

5.4.3.5 The progression of the test program from initial screening tests to optimization and radial load tests was successful in developing a pump design that meets all customer requirements. For a summary of influence of each design test series on the head/flow performance of the pump, Table 5-5 lists each test series applicable to the 140 gpm/450 ft total head design point. For each test series the increase or decrease in total head as a function of the tested characteristic is tabulated.

TABLE 5-2
Transfer Pump - Tested Head versus Flow
Influence of Seal Clearance on Performance

	Design Curve	.010" Seal Clearance	.060" Seal Clearance
Tested Flow (GPM)	Tested Head (FT)	Tested Head (FT)	Tested Head (FT)
0	567	586.4	578.2
10	565	588.5	578.2
20	565	589.7	577.5
30	565	590.0	575.9
40	562	589.4	573.6
50	560	587.8	570.4
60	559	585.3	566.5
70	555	581.9	561.8
80	550	577.6	556.4
90	545	572.4	550.1
100	540	566.2	543.1
110	533	559.1	535.3
120	527	551.2	526.7
130	520	542.3	517.3
140	512	532.4	507.1
150	504	521.7	496.2
160	494	510.0	484.5
170	485	497.5	472.0
180		484.0	458.7
190		469.6	444.6
200		454.2	429.8
240		383.8	362.5
280		298.6	282.8
325		185.3	178.2

Test Speed = 3420 RPM

$Q_{\text{DESIGN}} = 140 \text{ GPM}$

$H_{\text{DESIGN}} = 450 \text{ Ft. (Delivered Head)}$

Notes: 1) Delivered Head = Tested Head - Column Losses

TRANSFER PUMP TESTED HEAD VS FLOW INFLUENCE OF SEAL CLEARANCE

N=3420 RPM

FIGURE 5-5

TRANSFER PUMP TESTED HEAD VS FLOW DIFFUSER SCREENING TESTS

N=3420 RPM

FIGURE 5-6

TABLE 5-3
Transfer Pump - Tested Head versus Flow
Hydraulic Performance at Additional Operating Points

	OP #1	OP #2	OP #3
Tested Flow	Tested Head	Tested Head	Tested Head
105	382.9	100.2	--
110	379.7	95.2	449.4
115	376.1	90.2	445.0
120	372.2	85.1	440.5
125	368.0	80.0	435.7
130	363.3	74.8	430.7
135	358.4	69.5	425.6
140	353.1	64.2	420.2
145	347.4	58.7	414.6
150	341.4	53.3	408.9
155	--	--	402.9

	OP #4	OP #5
Tested Flow	Tested Head	Tested Head
30	455.0	79.2
35	454.8	77.8
40	454.2	76.1
45	453.1	74.1
50	451.7	71.9
55	449.9	69.4
60	447.6	66.6
65	445.0	63.6
70	442.0	60.3
75	438.6	56.7

OP #	Test Rig Speed	Flow (GPM)	Delivered Head (FT)
1	2885	132	305.0
2	1715	132	19.0
3	3100	138	366.0
4	3005	53	394.0
5	1290	53	19.0

- Notes:
- 1) OP# ⇒ Operating Point #
 - 2) Delivered Head = Tested Head - Column Losses

TRANSFER PUMP TESTED HEAD VS FLOW FIVE ADDITIONAL OPERATING POINTS

DELIVERED HEAD-TEST HEAD - COLUMN LOSSES

FIGURE 5-7

TABLE 5-4
Transfer Pump - Radial Load Summary from Test Data

Test Flow (GPM)	Test Head (FT)	Test Speed (RPM)	1st Stage Thrust (LBS)	2nd Stage Thrust (LBS)	Delta Phase Angle (Degrees)	SG Correction
145.1	529.1	3421.5	27.8	15.4	41.91	1.0
54.5	71.0	1291.5	4.2	3.1	23.00	1.0
134.7	71.3	1719.0	5.0	1.2	27.63	1.0
133.6	361.0	2883.0	19.6	10.9	42.41	1.0
140.0	424.8	3100.5	26.2	10.2	39.46	1.0
53.2	449.7	3004.5	23.0	20.2	63.97	1.0

TABLE 5-5
Transfer Pump - Performance Summary at Design Point

Test Series #	Tested Head (FT) @ 140 GPM	Configuration
Series #1	510.8	Diffuser #1
Series #2	512.7	Diffuser #2
Series #3	528.5	Diffuser #1 with Overfiled Inlet
Series #4	527.9	Added Suction Screen
Series #6	533.3	Added 3 more sets of pump out vanes
Series #8	532.4	Added Diffuser Pins
Series #9	507.1	Increased Seal Clearance

Notes: Speed = 3420 RPM

5.4.4 Sample Calculation

- The following sample calculation illustrates the data reduction methodology utilized to determine the volumetric flow and total head of the test rig.

Using the sample data set from point #11 of Test 9-A for the slotted diffuser,

The following environmental conditions apply:

Temperature = 82.8° F

Barometric Pressure = 30.0 in HG

Relative Humidity = .321

The following test rig conditions apply:

ΔP Flow Cell Indication (ΔP_q) = .7155" H₂O

Rotor Speed = 227.9 rev./sec. (indicated)

Static Pressures:

P_1 = -.044" H₂O

P_2 = -.043" H₂O

P_{out} = 7.133" H₂O

I. Calculate Air Density, ρ_A (lb_m/ft³)

For a temperature of 82.8°F, the vapor pressure (P_v) is 0.5530 PSIA.

The partial pressure of vapor (P_v) is calculated as follows:

$$P_v = (\text{relative humidity}) \times P_d = 0.321 \times 0.5530 = 0.1775 \text{ PSIA}$$

The partial pressure of air (P_A) is calculated as follows:

$$P_A = P_{\text{atm}} - P_v = 14.7360 - .1775 = 14.5585 \text{ PSIA}$$

where, $P_{\text{atm}} = 30.00 \text{ in. HG} = 14.7360 \text{ PSIA}$

The specific humidity of mixture(s) is calculated as follows:

$$S = 0.62 \frac{P_v}{(P_{\text{atm}}) - P_v} = .62 \frac{.1775}{14.5585} = 0.0076$$

The volume of mixture per pound of dry air (V_A) is calculated as follows:

$$V_A = .370 \frac{(\text{Temp} + 460)}{P_{\text{ATM}} - P_v} = .370 \frac{(82.8 + 460)}{14.5585} = 13.7951$$

The air density is calculated as follow:

$$\rho_A = \frac{1 + S}{V_A} = \frac{1 + .0076}{13.7951} = .0730 \text{ lb}_m/\text{ft}^3$$

II. Calculate the Volumetric Flowrate, q_v (ft³/sec)

The following equation defines the volumetric flow rate for the test rig:

$$q_v = CF_A K_u \sqrt{\Delta h}$$

where,

C = Discharge coefficient = .5439

K_m = Meter Constant = .0833

F_A = Thermal Expansion Factor = 1.000

$$\Delta h = \frac{\Delta P_q}{12} \times \frac{62.3377}{\Delta_A}$$

ΔP_q = Measured Differential Pressure at Flow Meter

for the same data set:

$$q_v = 0.5439 \times 1.000 \times 0.0833 \times \sqrt{\frac{.7155}{12} \times \frac{62.3377}{0.0730}} = .323 \frac{\text{ft}^3}{\text{sec}}$$

$$q_v = 145.1 \text{ gal/min.}$$

III. Calculate the Pump Head, H_T (ft.)

The following equation defined the pump head for the test rig.

$$H_T = \frac{P_{out} - \frac{(P_1 + P_2)}{2}}{12} \times \frac{62.3377}{\rho A} + \frac{V_2^2 - V_1^2}{2g} - \text{Screen loss}$$

where:

P_{out} = Measured discharge pressure, in H_2O

P_1, P_2 = Measured inlet pressures, in. H_2O

ρA = Calculated air density, lb_m/ft^3

V_2 = Discharge velocity, ft./sec.

V_1 = Inlet velocity, ft./sec.

$$V_2 = \frac{q_v}{A_2} \quad V_1 = \frac{q_v}{A_1}$$

q_v = Calculated volumetric flow, ft^3/sec

A_2 = Discharge area, ft^2

A_1 = Inlet area, ft^2

$$H_T = \frac{7.133 - \frac{(-.044 + (-.043))}{2}}{12} \times \frac{62.3377}{0.0730} + \frac{(6.3019)^2 - (11.7138)^2}{2(32.174)} = 3.1$$

$$\text{Screen Loss} = \left[\left(P_{atm} - \left(\frac{P_1 + P_2}{2} \right) \right) / 12 \times \frac{62.3377}{\rho A} \right] - \frac{0 - \frac{(-.044 + (-.043))}{2}}{12} \times \frac{62.3377}{.0730} = 3.1 \text{ ft}$$

$$A_1 = \frac{\pi}{4} (2.25)^2 = 3.976 \text{ in}^2 = 0.0276 \text{ ft}^2$$

$$A_2 = \frac{\pi}{4} (3.068)^2 = 7.393 \text{ in}^2 = 0.0513 \text{ ft}^2$$

$$V_2 = \frac{.323}{.0513} = 6.3019 \text{ ft./sec.}$$

$$V_1 = \frac{.323}{.0276} = 11.7138 \text{ ft./sec.}$$

$$H_t = 505.8 \text{ ft.}$$

IV. Calculate The Pump Speed, RPM_T

Indicated Speed = 227.9 revolutions/second

$$\text{Actual Speed} = \text{Indicated Speed} \times 60 \times \frac{1}{4} = 227.9 \times 60 \times \frac{1}{4} = 3418.5 \text{ RPM}$$

V. Ratio Head and Speed to 3420 RPM

$$\begin{aligned} H_T (\text{at } 3420 \text{ RPM}) &= H_T (\text{Tested}) \times \left(\frac{3420}{RPM_T} \right)^2 \\ &= 505.8 \times \left(\frac{3420}{3418.5} \right)^2 = 506.3 \text{ ft.} \end{aligned}$$

$$\begin{aligned} Q_v (\text{at } 3420 \text{ RPM}) &= Q_v (\text{Tested}) \times \left(\frac{3420}{RPM_T} \right)^1 \\ &= 145.1 \times \left(\frac{3420}{3418.5} \right) = 145.1 \text{ GPM} \end{aligned}$$

5.5 Hydraulic Performance

5.5.1 Introduction

The Model 2A-513-A1 Transfer Pump is designed to deliver 140 gpm at 450 feet of total head at the top of the column assembly. The operating conditions are 3860 rpm with fluid conditions of 30 centipoise and a specific gravity of 1.5. The final impeller diameters selected will be slightly larger (+ 1.5%) to account for the effect of 30 CP viscosity, which amounts to 3% in pure water head. This value of head adjustment was determined using the graphical approach found in the Hydraulic Institute Standards. The following sections summarize the analytical extrapolations for geometry and speed which were applied to the test data from the air test. An analysis of the final head versus flow curves for each operating condition shows a stable curve over the applicable flow range. Per API 610, the delivered head at rated flow should be within a tolerance of -2% to +5%. The final performance at all conditions is within this tolerance. Review of the scaled up hydraulic radial loads also shows acceptable values.

5.5.2 Performance Scaleup at Design Point

Using the information contained in Table 5-2 and illustrated in Figure 5-5, a total head versus flow curve was analytically calculated for the lead unit pump under actual operating conditions. The results of that analysis are presented in Table 5-6 and illustrated in Figure 5-8. The air test data is manipulated by applying a diameter scale ratio and a speed ratio applicable between the air test impellers and the lead unit impellers. The following sample calculation illustrates the procedure:

From Table 5-2, use the following data: $Q = 150 \text{ GPM}$
 $H = 496.2 \text{ FT}$

Applicable Test Conditions:	Rotor Speed 3420 RPM
	Seal Clearance 0.060"
Applicable Lead Unit Conditions:	Rotor Speed 3860 RPM
	Seal Clearance < 0.060"

A diameter scale ratio, β , of 0.9 will be used; therefore, the diameter scale of the lead unit impeller = 0.9 (Air Test Impeller).

The following equations define the tested performance condition for the lead unit pump.

Note: Subscript _{LU} denotes Lead Unit Pump
 Subscript _{AT} denotes Air Test Pump

$$\begin{aligned}\text{Head}_{\text{LU}} &= \text{Head}_{\text{AT}} \times \beta^2 \times \left(\frac{3860}{3420}\right)^2 \\ &= 496.2 \times (0.9)^2 \times \left(\frac{3860}{3420}\right)^2 = 512.0 \text{ FT}\end{aligned}$$

$$\begin{aligned}\text{Flow}_{\text{LU}} &= \text{Flow}_{\text{AT}} \times \beta^2 \times \left(\frac{3860}{3420}\right) \\ &= 150 \times (0.9)^2 \times \left(\frac{3860}{3420}\right) = 137.1 \text{ GPM}\end{aligned}$$

In order to determine the final delivered head at the top of the column assembly, the appropriate elevation head and pipe friction losses must be subtracted from the Head_{LU} term. The following assumptions apply:

- 1) Pipe friction factor assumed as .03
- 2) Elevation Head = 52 FT
- 3) Pipe Internal Diameter = 3.068 IN
Pipe Internal Area = 7.393 IN²

For the case listed above: Fluid velocity in column = $.321 \frac{\text{Q}}{\text{A}}$

$$V_c = .321 \left(\frac{137.1}{7.393}\right) = 5.953 \text{ FT/SEC}$$

The total head at the top of the column is calculated as follows:

$$\text{Head}_{(\text{FINAL})} = \text{Head}_{\text{LU}} - \text{Elevation Head} - f\left(\frac{L}{D}\right) \frac{V^2}{2g}$$

Where: $f\left(\frac{L}{D}\right) \frac{V^2}{2g}$ represents the pipe friction losses

$$\text{Head}_{(\text{FINAL})} = 512.0 - 52 - (.03) \left(\frac{52 \times 12}{3.068} \right) \frac{(5.953)^2}{2g} = 456.6 \text{ FT}$$

A similar procedure was applied to all the test data in TABLE 5-2, substituting the appropriate flow, head, and speed values. The results of the analysis are presented in TABLE 5-6.

TABLE 5-6
Transfer Pump - Delivered Head versus Flow
Influence of Seal Clearance on Performance

	Design Curve	.010" Seal Clearance	.060" Seal Clearance
Delivered Flow (GPM)	Delivered Head (FT)	Delivered Head (FT)	Delivered Head (FT)
0.0	531.9	553.1	544.6
9.1	532.0	555.3	544.6
18.3	531.5	556.4	543.8
27.4	530.4	556.6	542.1
36.6	528.5	555.9	539.6
45.7	526.0	554.1	536.2
54.9	522.9	551.4	532.0
64.0	519.0	547.7	527.0
73.1	514.5	543.0	521.1
82.3	509.3	537.4	514.4
91.4	503.5	530.7	506.9
100.6	497.0	523.1	498.5
109.7	489.8	514.5	489.3
118.8	481.9	505.0	479.2
128.0	473.4	494.4	468.3
137.1	464.2	482.9	456.6
146.3	454.4	470.4	444.1
155.4	443.9	457.0	430.7
164.6		442.5	416.4
173.7		427.1	401.4
182.8		410.7	385.5
219.4		335.4	313.5
256.0		244.4	228.1
297.1		123.4	116.1
140		479.0 (106.4%)	452.7 (100.6%)

Speed = 3860 RPM

$Q_{\text{DESIGN}} = 140 \text{ GPM}$

$H_{\text{DESIGN}} = 450 \text{ FT (Delivered Head)}$

TRANSFER PUMP DELIVERED HEAD VS FLOW

N=3860 RPM, $f=0.030$

FIGURE 5-8

5.5.3 Performance Scaleup at Other Operating Points

Using the information contained in Table 5-3 and illustrated in Figure 5-7, a total head versus flow curve was analytically calculated for the lead unit pump for the remaining operating conditions outlined in the equipment specification. A similar procedure to that outlined in Section 5.5.2 was utilized with the addition of numerical corrections to account for design changes that were not present in the test rig when the original test data was taken. These design changes were discussed in Section 5.4.3.3. The results of that analysis are presented in Table 5-7 and illustrated in Figure 5-9.

TABLE 5-7

Transfer Pump - Delivered Head versus Flow
Hydraulic Performance at Additional Operating Points

OP #1	N = 3350 RPM
Delivered Flow (GPM)	Delivered Head (FT)
98.8	349.9
103.5	345.9
108.2	341.6
112.9	336.8
117.6	331.7
122.3	326.3
127.0	320.5
131.7	314.3
136.4	307.7
141.1	300.8
132	313.8 (102.9%)

OP #2	N = 2063 RPM
Delivered Flow (GPM)	Delivered Head (FT)
102.3	57.9
107.2	51.8
112.1	45.7
116.9	39.4
121.8	33.1
126.7	26.7
131.5	20.2
136.4	13.7
141.3	7.0
146.2	0.3
132	19.6 (103.2%)

OP #3	N = 3600 RPM
Delivered Flow (GPM)	Delivered Head (FT)
103.5	419.8
108.2	414.5
112.9	409.1
117.6	403.4
122.3	397.5
127.0	391.5
131.7	385.2
136.4	378.7
141.1	372.0
145.8	365.2
138	376.4 (102.9%)

OP #4	N = 3410 RPM
Delivered Flow (GPM)	Delivered Head (FT)
27.6	415.7
32.2	414.7
36.8	413.2
41.4	411.4
46.0	409.2
50.6	406.6
55.2	403.7
59.7	400.3
64.3	396.5
68.9	392.3
53	405.0 (102.8%)

OP #5	N = 1490 RPM
Delivered Flow (GPM)	Delivered Head (FT)
28.1	32.2
32.7	30.4
37.4	28.4
42.1	26.1
46.8	23.4
51.5	20.5
56.1	17.3
60.8	13.9
65.5	10.1
70.2	6.1
53	19.5 (102.6%)

Notes: 1) OP# → Operating Point #

TRANSFER PUMP DELIVERED HEAD VS FLOW FIVE ADDITIONAL OPERATING POINTS

0.060" SEAL CLEARANCE, $f=0.030$

FIGURE 5-9

5.5.4 Radial Load Scaleup

- Using the information contained in Table 5-4, the tested radial load data was analytically extrapolated to conditions for the lead unit pump. The results of that analysis are presented in Table 5-8. The test data was manipulated by applying a speed ratio and a fluid specific gravity ratio. The following sample calculations illustrate the procedure. There were two design changes that were not incorporated into the test rig at the time the radial load data was taken. These changes were discussed in Section 5.4.3.4. No numerical corrections were applied to the test data to compensate for the omission of these design features.

From Table 5-4, use the following data:

- Q = 145.1 GPM
- H = 529.1 FT
- N_{TEST} = 3421.5 RPM
- 1st Stage Thrust = 27.8 LBS
- 2nd Stage Thrust = 15.4 LBS
- Phase Angle = 41.91°
- SG = 1.0

Applicable Lead Unit Conditions:

- Rotor Speed 3860 RPM
- SG = 1.5

Using the same scaling methods as outlined in Section 5.5.2 for Head and Flow:

$$\text{Head}_{LU} = 529.1 \times (0.9)^2 \times \left(\frac{3860}{3421.5} \right)^2 = 545.5 \text{ FT}$$

$$\text{Flow}_{LU} = 145.1 \times (0.9)^2 \times \left(\frac{3860}{3421.5} \right) = 132.6 \text{ GPM}$$

The following equation defines the hydraulic load scaleup for the lead unit pump

$$\text{Thrust}_{LU} = \text{Thrust}_{AT} \times \left(\frac{3860}{3421.5} \right)^2 \times \left(\frac{1.5}{1.0} \right)$$

Where: $\text{Thrust}_{\text{LU}}$ is the final hydraulic thrust for the individual lead unit components.

$\text{Thrust}_{\text{AT}}$ is the load value calculated in TABLE 5-4.

$\left(\frac{3860}{3421.5}\right)$ is the appropriate speed ratio.

$\left(\frac{1.5}{1.0}\right)$ is the appropriate specific gravity ratio.

For the first stage impeller

$$\text{Thrust}_{\text{LU}} = 27.8 \times \left(\frac{3860}{3421.5}\right)^2 \times \left(\frac{1.5}{1.0}\right) = 53.1 \text{ LBS}$$

For the second stage impeller

$$\text{Thrust}_{\text{LU}} = 15.4 \times \left(\frac{3860}{3421.5}\right)^2 \times \left(\frac{1.5}{1.0}\right) = 29.4 \text{ LBS}$$

The phase angle between the load vectors is unchanged due to scaleup calculations.

A similar procedure was applied to all the test data in TABLE 5-4, substituting the appropriate head, flow, and speed values. The results of the analysis are presented in TABLE 5-8.

TABLE 5-8
Transfer Pump - Radial Load Summary
Scaleup of Test Data

Ratioed Flow (GPM)	Ratioed Head (FT)	Final Speed (RPM)	1st Stage Thrust (LBS)	2nd Stage Thrust (LBS)	Delta Phase Angle (Degrees)	Final SG
132.6	545.5	3860	53.1	29.4	41.91	1.5
50.9	76.5	1490	8.3	6.2	23.00	1.5
130.9	83.2	2063	10.8	2.5	27.63	1.5
125.7	394.8	3350	39.7	22.1	42.41	1.5
131.7	463.9	3600	52.9	20.7	39.46	1.5
48.9	469.2	3410	44.3	39.1	63.97	1.5

5.6 Reverse Flow Analysis

The reverse flow analysis of the transfer pump consists of the calculation of two events: reverse flow windmilling due to an impressed head and the locked rotor (rpm = 0) torque due to the resulting impressed, reverse flow.

The reverse flow is due to the 85 feet (see Hydraulic Basis, Section 5.1) of tank elevation when the pump is de-energized. On a per stage basis, this elevation head, as a percent, is:

$$\frac{85}{(499/2)} \times 100 = 34.1\% \text{ impressed head}$$

In order to calculate the windmilling speed, published Worthington Pump data* can be cross-plotted to show the relationship between percent head and percent design speed; this was done and shown in Figure 5-10. From this figure, the windmilling speed is 58% or:

$$0.58 \times 3860 = 2240 \text{ rpm}$$

Similarly, this Worthington Pump data can be used to predict locked impeller torque due to a 34.1% impressed head. Figure 5-11 was drawn for this prediction, and this figure shows that the locked torque is 41% of design, or:

$$0.41 \times 75 \text{ ft.-lbs.} = 31 \text{ ft.-lbs.}$$

These values of windmilling speed and locked torque were considered acceptable for pump re-start conditions.

*Pump Handbook, Karassik, Editor, McGraw-Hill, N.Y., 1976, p. 9-17.

Figure 5-10 Windmilling Speed

Figure 5-11 Locked Rotor Torque

5.7 Internal Circulation Analysis - Introduction

The required calculations for the internal circulation analysis of the transfer pump are the determination of the steady state internal motor flow and pressure distribution, as well as calculation of the hydraulic thrust load. The hydraulic thrust load is a summation of the axial thrust components of the motor and hydraulic components. This load added to the buoyant rotating assembly weight determines the total thrust load which must be counteracted by the thrust bearing assembly. Several modes of pump operation were examined to demonstrate adequate internal motor flows and provide calculational estimates of total thrust load.

Added to this scope of work is an evaluation of the auxiliary flushing systems used to maintain proper operation of the transfer pump. The purge system is used to inject clean water into the motor cavity prior to pump operation. The dilution line system is used to inject clean water at the suction inlet of the first stage impeller to dilute the process fluid before it is transferred into the pump internals.

5.7.1 Description of Internal Circulation Flowpaths

The motor internal circulation paths are illustrated in Figure 5-12. The corresponding analytical circuit model is shown in Figure 5-13. The tank process fluid, Q16, enters the hydraulic area by passing through the suction screen and into the first stage impeller, Q1. After being pressurized by passage through the impeller, the majority of the flow enters into the first stage diffuser, Q4. Two recirculation flows exist at the discharge of the first stage impeller. A portion of the impeller flow recirculates back to the low pressure region of the impeller suction, Q2. Also a small portion of the first stage diffuser flow, Q3, recirculates around the top shroud of the impeller and joins the Q1 flow at the impeller exit.

The main diffuser flow, Q4, then proceeds to the second stage hydraulics and is pressurized by passage through the second stage impeller, Q5. As before, a portion of the impeller flow recirculates back to the low pressure side of the impeller, Q6. The remaining flow is joined with the motor recirculation flow, Q37, and passes through the second stage diffuser and into the motor stator annulus, Q7.

FIGURE 5-12 Transfer Pump Fluid Circulation Paths

FIGURE 5-12 (Continued) Transfer Pump Fluid Circulation Paths

FIGURE 5-13 Transfer Pump Hydraulic Circuit Model
Normal Operation

At the top of the motor stator annulus, the majority of the flow, Q16, is directed into the transition piece and column assembly for transport out of the tank. A small portion of the annulus flow, Q19 and Q17, is directed radially inward to provide lubrication and motor cooling. The flow passes through several paths in and around the upper radial bearing, Q9, Q11, Q15, and Q17 through Q23. The flows exiting the upper bearing region join to form the rotor/stator annulus flow, Q12. The process fluid then flows in and around the lower radial bearing assembly, Q25 through Q32, and Q34. Finally upon exit from the lower radial bearing assembly, the flow, Q10, proceeds around the thrust bearing assembly, Q35 and Q36, and back to the discharge of the second stage impeller (Q37).

5.7.2 Analysis of Normal Operating Points

As was previously mentioned in Section 5.4, the transfer pump has six operating points that must have stable operation. The results of the hydraulic air test study were used to define the final rotor speeds and pump head values at the equipment specification flow points. This information, along with the motor internal geometry was used to construct the detailed circuit model shown in Figure 5-13. The input fluid conditions and rotor speeds used for the analysis are listed in Table 5-9. The results of the analysis at the six operating points are presented in Table 5-10. A sample file listing for the 140 gpm, 3860 rpm case is presented in Table 5-11. The resistance, vortex, and continuity equations are shown as well as the flow, pressure, and thrust results. The thrust presented in Table 5-11 differs from the total thrust load presented for the same case in Table 5-10 by the "wet weight" of the rotating assembly. The calculated hydraulic thrust is added to the "wet rotor weight" to give the total thrust load as presented in Table 5-10.

TABLE 5-9
Internal Circulation - Input Conditions
Normal Operating Points Analysis

Equipment Specification		Rotor Speed (RPM)	Process Fluid	
Flow (GPM)	Delivered Head (Ft.)		Density (LBM/Ft. ³)	Absolute Viscosity (CP)
140	450	3860	93.507	30
53	394	3410	93.507	30
53	19	1490	93.507	30
132	305	3350	93.507	30
132	19	2063	93.507	30
138	366	3600	93.507	30

TABLE 5-10
Transfer Pump Internal Circulation Results
Normal Operating Points Analysis

	Flow No.	Flow GPM	Flow GPM	Flow GPM	Flow GPM	Flow GPM	Flow GPM
Q1-#1 Impeller	1	168.99	81.99	60.07	154.23	134.56	162.74
Q2-#1 Impeller-Lwr Shr	2	28.99	28.99	7.07	22.23	2.56	24.74
Q3-#1 Impeller-Upr Shr	3	20.27	17.88	4.61	16.51	5.49	18.28
Q4-#1 Diffuser	4	160.27	70.88	57.61	148.51	137.49	156.28
Q5-#2 Impeller	5	162.85	75.21	57.83	149.17	134.96	157.35
Q6-#2 Impeller-Lwr Shr	6	22.85	22.21	4.83	17.17	2.95	19.35
Q7-Stator Annulus Flow	7	170.47	79.26	64.43	158.25	147.82	166.33
Q8-Upper Cavity Recirc	8	30.47	26.26	11.43	26.25	15.82	28.33
Q9-Upper Bearing Housing Annulus	9	6.31	5.44	2.34	5.44	3.25	5.87
Q10-Thrust Runner Flow	10	30.47	26.26	11.43	26.25	15.82	28.33
Q11-Upper Rad Bearing Pad Area	11	15.07	13.28	6.48	13.27	8.60	14.16
Q12-Rotor/Stator Annulus	12	30.47	26.26	11.43	26.25	15.82	28.33
Q15-Upper Housing Bypass Flow	15	1.50	1.30	0.58	1.30	0.79	1.40
Q16-Design Flow	16	140.00	53.00	53.00	132.00	132.00	138.00
Q17-Purge Vent Flow	17	7.81	6.73	2.93	6.73	4.04	7.26
Q18-Top of Rotor Flow	18	24.16	20.82	9.08	20.82	12.57	22.46
Q19-Upper Cavity Flow	19	22.66	19.52	8.50	19.52	11.78	21.06
Q20-Upper Bearing Annulus	20	9.09	7.54	2.61	7.54	3.97	8.30
Q21-Upper Bearing Radial Holes	21	2.21	1.75	0.41	1.75	0.76	1.97
Q22-Upper Bearing Housing Annulus	22	8.52	7.19	2.75	7.19	4.01	7.84
Q23-Upper Bearing Annulus	23	6.88	5.79	2.20	5.79	3.21	6.32
Q25-Lower Bearing Bypass Flow	25	10.77	9.53	4.78	9.53	6.27	10.14
Q26-Lower Bearing Flow	26	19.70	16.73	6.64	16.73	9.55	18.18
Q27-Lower Bearing Pad Area	27	11.93	10.35	4.52	10.34	6.29	11.13
Q28-Lower Bearing Annulus	28	6.79	5.60	1.88	5.60	2.88	6.18
Q29-Lower Bearing Housing Annulus	29	0.98	0.78	0.24	0.78	0.37	0.88
Q30-Lower Bearing Radial Holes	30	0.27	0.19	0.02	0.19	0.05	0.23
Q31-Lower Bearing Housing Annulus	31	0.71	0.59	0.22	0.59	0.32	0.65
Q32-Lower Bearing Annulus	32	7.06	5.79	1.90	5.79	2.94	6.41
Q34-Lower Housing Axial Holes	34	11.48	10.12	5.00	10.12	6.59	10.79
Q35-Thrust Bearing Assembly Holes	35	68.22	60.28	26.35	59.18	36.65	63.61
Q36-Thrust Runner Flow	36	98.69	86.53	37.78	85.43	52.47	91.93
Q37-Thrust Runner Flow	37	30.47	26.26	11.43	26.25	15.82	28.33
Total Thrust Load		1063	1107	247	701	81	824
Rotor Speed (RPM)		3860	3410	1490	3350	2063	3600
Flowrate (GPM)		140	53	53	132	132	138
Pump Head (FT) w/o Column Losses		521.5	454.5	71.4	368.1	78.2	430.8

TABLE 5-11
Sample File Listing for Q = 140 GPM, H = 450 Ft. Case

TITLE - TRANSFER PUMP FINAL CIRCUIT ANALYSIS - 4 PAD BEARINGS 4/23/97

DENSITY = 93.51 LBS/CU FT
VISCOSITY = 0.020160 LBS/FT-SEC
ROTOR O.D. = 3.794 INCHES
PUMP SPEED = 3860.0 RPM

CONTINUITY EQUATIONS AT TEES

+ Q7 - Q8 - Q16 = 0.
+ Q8 - Q17 - Q19 = 0.
+ Q17 - Q9 - Q15 = 0.
+ Q19 + Q15 - Q18 = 0.
+ Q18 - Q20 - Q11 = 0.
+ Q22 + Q23 + Q11 - Q12 = 0.
+ Q21 + Q9 - Q22 = 0.
+ Q20 - Q21 - Q23 = 0.
+ Q12 - Q25 - Q26 = 0.
+ Q26 - Q27 - Q28 - Q29 = 0.
+ Q28 + Q30 - Q32 = 0.
+ Q29 - Q30 - Q31 = 0.
+ Q25 + Q31 - Q34 = 0.
+ Q32 + Q27 + Q34 - Q10 = 0.
+ Q35 + Q10 - Q36 = 0.
+ Q16 + Q2 - Q1 = 0.
+ Q1 - Q2 + Q3 - Q4 = 0.
+ Q4 - Q5 + Q6 - Q3 = 0.
+ Q5 - Q6 - Q7 + Q37 = 0.

TABLE 5-11 (Continued)
Sample File Listing for Q=140 GPM, H=450 Ft. Case

Mon Apr 28 15:08:45 1997
4/23/97

INPUT DATA - TRANSFER PUMP FINAL CIRCUIT ANALYSIS - 4 PAD BEARINGS

	NODE-I	NODE-J	FLOW NO.	MAT. NO.	K-FACTOR	LENGTH INCHES	AREA INCH**2	DIA. INCHES
#1IMP.LS E/F/E	2	3	2	4	1.019	0.133	1.3180	0.100
#1IMP.LS BEND	3	4	2	0	0.800	0.000	1.3180	0.100
#1IMP.LS BEND	6	7	2	0	0.800	0.000	1.6120	0.360
#1IMP.LS C/F/E.	7	8	2	4	1.002	0.550	0.1973	0.046
#1IMP.LS BEND	8	9	2	0	0.800	0.000	0.1973	0.046
#1IMP.LS EXIT	10	1	2	0	1.000	0.000	0.7450	0.200
#1IMP.US E/F/E.	20	19	3	4	0.604	0.320	0.4790	0.158
#1IMP.US BEND	19	18	3	0	0.800	0.000	0.4790	0.158
#1IMP.US BEND	17	16	3	0	0.800	0.000	0.7410	0.200
#1IMP.US C/F/E.	16	15	3	4	1.013	1.160	0.1560	0.040
#1IMP.US BEND	15	14	3	0	0.800	0.000	0.1560	0.040
#1IMP.US BEND	12	11	3	0	0.800	0.000	5.5020	0.420
#1IMP.US C/F/E	11	2	3	4	1.370	0.140	1.3180	0.100
#1 DIFFUSER	2	20	4	0	11.843	0.000	5.0000	1.000
#2IMP.LS E/F/E	21	22	6	4	0.968	0.160	1.5540	0.108
#2IMP.LS BEND	22	23	6	0	0.800	0.000	1.5540	0.108
#2IMP.LS BEND	25	26	6	0	0.800	0.000	1.9600	0.400
#2IMP.LS C/F/E	26	27	6	4	1.121	0.540	0.1600	0.034
#2IMP.LS BEND	27	28	6	0	0.800	0.000	0.1600	0.034
#2IMP.LS EXIT	29	20	6	0	1.000	0.000	0.7854	0.200
STATOR ANN. FRIC.	30	31	7	1	0.000	1.900	11.3730	0.800
STATOR ANN. BEND	31	32	7	0	0.062	0.000	11.3730	0.800
STATOR ANN. FRIC.	32	33	7	1	0.000	1.450	11.6240	0.800
STATOR ANN. S/C	33	34	7	0	0.222	0.000	8.2840	0.692
STATOR ANN. CONTR.	34	35	7	0	0.040	0.000	7.8720	0.504
STATOR ANN. FRIC.	35	36	7	1	0.000	37.000	7.8720	0.504
COLLECTOR FRIC.	36	44	8	1	0.000	1.100	7.8720	0.504
COLLECTOR CONTR.	44	45	19	0	0.508	0.000	26.4000	2.568

TABLE 5-11 (Continued)
Sample File Listing for Q = 140 GPM, H = 450 Ft. Case

Mon Apr 28 15:08:45 1997
4/23/97

INPUT DATA - TRANSFER PUMP FINAL CIRCUIT ANALYSIS - 4 PAD BEARINGS

	MODE-I	MODE-J	FLOW NO.	MAT. NO.	K-FACTOR	LENGTH INCHES	AREA INCH**2	DIA. INCHES
COLLECTOR CONTR.	45	46	19	0	0.320	0.000	9.9688	1.577
COLLECTOR CONTR.	46	47	19	0	0.500	0.000	0.4501	0.379
COLLECTOR BEND	47	48	19	1	0.000	4.430	0.4501	0.379
COLLECTOR FRIC.	48	49	19	1	0.000	1.000	0.4501	0.379
ORIFICE CONTR.	49	123	19	0	0.380	0.000	0.0755	0.155
ORIFICE FRIC.	123	124	19	1	0.000	0.500	0.0755	0.155
ORIFICE EXPAN.	124	125	19	0	0.693	0.000	0.0755	0.155
COLLECTOR FRIC.	125	126	19	1	0.000	1.000	0.4501	0.379
COLLECTOR EXIT	126	50	19	0	1.000	0.000	0.4501	0.379
ST.ANN-ADD.FRIC.	44	90	17	1	0.000	1.250	7.8720	0.504
ST.ANN-MITER BEND	90	91	17	0	0.312	0.000	7.8720	0.504
PURG.VENT E/F	91	92	17	1	0.500	2.000	0.0491	0.250
PURG.VENT BEND	92	93	17	0	2.813	0.000	0.0491	0.250
PURG.VENT EXPAN.	93	94	17	0	0.700	0.000	0.0491	0.250
PURG.VENT FRIC	94	95	17	1	0.000	6.313	0.3070	0.625
PURG.LINE M. BEND	95	96	17	0	1.250	0.000	0.3070	0.625
PURG.LINE CONTR.	96	97	17	0	0.100	0.000	0.2500	0.488
PURG.LINE EXIT	97	98	17	0	1.000	0.000	0.2500	0.488
BY-PASS ENTR.	98	87	15	0	0.500	0.000	0.0573	0.191
BY-PASS F/EXIT	87	50	15	1	1.000	0.145	0.0573	0.191
UPR BRG NSG ANN ENTR.	98	60	9	0	0.500	0.000	0.1700	0.202
UPR BRG NSG ANN F/EXP.	60	61	9	1	0.500	0.360	0.1700	0.202
UPR BRG NSG ANN FRIC.	61	62	9	1	0.000	0.486	0.7250	0.098
UPR BRG PAD AREA-E/F	50	53	11	1	0.500	0.100	1.0500	0.488
UPR BRG PAD AREA-C/F	53	57	11	1	0.370	1.750	0.5606	0.213
UPR BRG PAD AREA-EXIT	57	66	11	0	1.000	0.000	0.5606	0.213
UPR BRG ANN-ENTR.	50	55	20	0	0.500	0.000	0.5298	0.162
UPR BRG ANN-FRIC.	55	54	20	1	0.000	0.600	0.5298	0.162
UPR BRG RAD HOLES-ENTR.	54	63	21	0	0.590	0.000	0.5070	0.284
UPR BRG RAD HOLES-FR.	63	64	21	1	0.000	0.315	0.5070	0.284
UPR BRG RAD HOLES-EXIT	64	62	21	0	1.000	0.000	0.5070	0.284
UPR BRG NSG ANN-FRIC	62	65	22	1	0.000	1.250	0.7250	0.098

TABLE 5-11 (Continued)
Sample File Listing for Q=140 GPM, H=450 Ft. Case

Mon Apr 28 15:08:45 1997
4/23/97

INPUT DATA - TRANSFER PUMP FINAL CIRCUIT ANALYSIS - 4 PAD BEARINGS

	NODE-I	NODE-J	FLOW NO.	MAT. NO.	K-FACTOR	LENGTH INCHES	AREA INCH**2	DIA. INCHES
UPR BRG NSG ANN-EXIT	65	66	22	0	1.000	0.000	0.7250	0.098
UPR BRG ANN-FRIC	54	56	23	1	0.000	1.250	0.5298	0.102
UPR BRG ANN-EXIT	56	66	23	0	1.000	0.000	0.5298	0.102
UPR BRG FRIC.	66	67	12	1	0.000	0.750	5.8580	0.690
UPR BRG EXP	67	68	12	0	0.006	0.000	5.8530	0.690
UPR BRG FRIC.	68	69	12	1	0.000	0.275	6.3480	0.792
UPR BRG EXIT	69	70	12	0	1.000	0.000	6.3480	0.792
R/S ANN ENTR	71	72	12	0	0.500	0.000	1.7557	0.284
R/S ANN FRIC	72	73	12	4	0.000	23.500	1.7557	0.284
R/S ANN EXIT	73	74	12	0	1.000	0.000	1.7557	0.284
LWR BRG BYPASS-ENTR	77	78	25	0	0.500	0.000	0.4418	0.375
LWR BRG BYPASS-FRIC	78	79	25	1	0.000	1.130	0.4418	0.375
LWR BRG BYPASS-BEND	79	80	25	0	0.312	0.000	0.4418	0.375
LWR BRG BYPASS-FRIC	80	81	25	1	0.000	1.280	0.4418	0.375
LWR BRG-ENTR	77	82	26	0	0.500	0.000	5.5130	0.943
LWR BRG-FRIC	82	83	26	1	0.000	0.150	5.5130	0.943
LWR BRG-EXP	83	84	26	0	0.240	0.000	5.5130	0.943
LWR BRG PAD AREA-ENT	84	85	27	0	0.500	0.000	0.5442	0.203
LWR BRG PAD AREA-FRIC	85	108	27	1	0.000	1.920	0.5442	0.203
LWR BRG PAD AREA-EXIT	108	107	27	0	1.000	0.000	0.5442	0.203
LWR BRG ANN-ENTR	84	88	28	0	0.500	0.000	0.6466	0.100
LWR BRG ANN-FRIC	88	86	28	1	0.000	0.400	0.6466	0.100
LWR BRG NSG ANN-ENT	84	99	29	0	0.500	0.000	0.3748	0.045
LWR BRG NSG ANN-FRIC	99	100	29	1	0.000	0.400	0.3748	0.045
RAD HOLES-ENT	100	102	30	0	0.500	0.000	0.5070	0.284
RAD HOLES-FRIC	102	101	30	1	0.000	0.315	0.5070	0.284
RAD HOLES-CONV TEE	101	86	30	0	1.000	0.000	0.5070	0.284
LWR BRG NSG ANN-FRIC	100	103	31	1	0.080	0.850	0.3748	0.045
LWR BRG NSG ANN-BEND	103	104	31	0	1.250	0.000	0.3748	0.045

TABLE 5-11 (Continued)
Sample File Listing for Q = 140 GPM, H = 450 Ft. Case

Mon Apr 28 15:08:45 1997
4/23/97

INPUT DATA - TRANSFER PUMP FINAL CIRCUIT ANALYSIS - 4 PAD BEARINGS

	NODE-I	NODE-J	FLOW NO.	MAT. NO.	K-FACTOR	LENGTH INCHES	AREA INCH**2	DIA. INCHES
LWR BRG NSG ANN-EXP	104	81	31	0	0.435	0.000	0.3748	0.045
LWR BRG ANN-FRIC	86	106	32	1	0.000	1.320	0.6466	0.100
LWR BRG ANN-EXIT	106	107	32	0	1.000	0.000	0.6466	0.100
AX HOLES-ENTR	81	109	34	0	0.500	0.000	0.6120	0.442
AX HOLES-FRIC	109	110	34	1	0.000	0.630	0.6120	0.442
AX HOLES-EXIT	110	107	34	0	1.000	0.000	0.6120	0.442
THR BRG NSG HOLES-ENT	122	121	35	0	0.500	0.000	0.7540	0.400
THR BRG NSG HOLES-FRIC	121	120	35	1	0.000	0.500	0.7540	0.400
THR BRG NSG HOLES-EXIT	120	114	35	0	1.000	0.000	0.7540	0.400
TH BRG NSG-E/F	119	116	36	1	0.500	0.500	3.1340	0.700
TH BRG NSG-BEND	116	117	36	0	0.800	0.000	3.1340	0.700
TH BRG NSG-CONTR	117	58	36	0	0.140	0.000	2.3500	0.427
SCREEN LOSS	118	1	16	0	1.185	0.000	3.8010	2.200
DIS.PIPE SPLIT	36	37	16	0	0.500	0.000	3.9760	1.125
DIS.PIPE FRIC.	37	38	16	1	0.313	4.000	3.9760	1.125
DIS.PIPE EXP.	38	39	16	0	0.150	0.000	3.9760	1.125
DIS.PIPE BEND	39	40	16	0	0.019	0.000	6.4920	2.875
DIS.PIPE FRIC.	40	41	16	1	0.000	8.700	6.4920	2.875
DIS.PIPE BEND	41	42	16	0	0.019	0.000	6.4920	2.875
DIS.PIPE FRIC.	42	43	16	1	0.000	1.500	7.3950	3.068

TABLE 5-11 (Continued)
Sample File Listing for Q = 140 GPM, H = 450 Ft. Case

Mon Apr 28 15:08:45 1997

INPUT DATA - FLUID VORTICES

	NODE-I	NODE-J	FLOW NO.	EQU. NO.	OUTER DIA. INCHES	INNER DIA. INCHES	FACTOR
#1 IMPELLER	2	1	1	1	7.502	0.000	1.0000
#1 IMP-LWR.SHR.	4	5	2	1	8.350	7.600	0.5000
#1 IMP-LWR.SHR.	5	6	2	1	7.600	2.708	0.7791
#1 IMP-LWR.SHR.	9	10	2	1	2.708	2.370	0.5000
#1 IMP-UPR.SHR.	12	13	3	1	8.340	7.600	0.5000
#1 IMP-UPR.SHR.	13	14	3	1	7.600	2.708	0.7857
#1 IMP-UPR.SHR.	17	18	3	1	2.464	2.100	0.5000
#2 IMPELLER	21	20	5	1	7.881	0.000	1.0000
#2 IMP-LWR.SHR.	23	24	6	1	8.705	7.600	0.5000
#2 IMP-LWR.SHR.	24	25	6	1	7.600	2.970	0.7791
#2 IMP-LWR.SHR.	28	29	6	1	2.970	2.480	0.5000
#2 DIFFUSER	30	21	7	1	2.423	0.000	1.0000
UPPER ROTOR CAVITY	50	51	18	1	2.400	0.000	0.3000
LOWER BEARING	71	70	12	1	3.794	2.128	0.2000
LOWER BEARING	74	75	12	1	3.794	2.128	0.2000
LOWER BEARING	76	75	12	1	2.680	2.128	0.2000
LOWER BEARING	77	76	12	1	3.260	2.680	0.9030
LOW THR.RUNNER	107	111	10	1	3.110	2.080	0.5000
LOWER THR.RUNNER	112	107	10	1	7.000	3.110	0.5000
LOWER THR.RUNNER	112	113	10	1	7.000	6.630	0.0000
LOWER THR.RUNNER	113	114	10	1	6.630	4.720	0.0000
LOWER THR.RUNNER	114	115	36	1	4.720	2.900	0.0000
LOWER THR.RUNNER	115	119	36	1	2.900	2.460	0.0000
#2 IMP-UPR.SHR.	59	122	37	1	7.600	4.720	0.6935
#2 HUB VORTEX	122	58	36	1	4.720	3.400	0.6935
#2 IMP-UPR.SHR.	21	59	37	1	8.700	7.600	0.5000

FOR EQU. NO. 1 $HEAD = (DO^{**2} - D1^{**2}) * (FAC * RPM)^{**2}$
FOR EQU. NO. 2 $HEAD = ((DO/D1)^{**2} - 1.0) * (RPM^{**2} * DO * FAC)^{**2}$
FOR EQU. NO. 3 $HEAD = (1.0 - (D1/DO))^{**2} * (RPM^{**2} * D1 * FAC)^{**2}$
FOR EQU. NO. 4 $HEAD = (D1^{**2} - (0.5 * DO^{**2})) * RPM^{**2}$

TABLE 5-11 (Continued)
Sample File Listing for Q = 140 GPM, H = 450 Ft. Case

FLOW RESULTS - TRANSFER PUMP FINAL CIRCUIT ANALYSIS - 4 PAD BEARINGS Mon Apr 28 15:08:45 1997
6/23/97

	FLOW NO.	FLOW GPM
Q1-#1 IMPELLER	1	168.99
Q2-#1 IMPELLER-LWR.SNR.	2	28.99
Q3-#1 IMPELLER-UPR.SNR.	3	20.27
Q4-#1 DIFFUSER	4	160.27
Q5-#2 IMPELLER	5	162.85
Q6-#2 IMPELLER-LWR.SNR.OLD	6	22.85
Q7-STATOR ANNULUS	7	170.47
Q8-COLLECTOR AREA	8	30.47
Q9-UPR BRG NSG ANN	9	6.31
Q10-THRUST RUN FLOW	10	30.47
Q11-UPR BRG PAD AREA	11	15.07
Q12-ROTOR/STATOR ANNULUS	12	30.47
Q15-UPR NSG BYPASS FLOW	15	1.50
Q16-DESIGN FLOW	16	140.00
Q17-PURGE VENT FLOW	17	7.81
Q18-TOP OF ROTOR FLOW	18	24.16
Q19-UPR CAVITY FLOW	19	22.66
Q20-UPR BRG ANNULUS	20	9.09
Q21-UPR BRG RAD HOLES	21	2.21
Q22-UPR BRG NSG ANN	22	8.52
Q23-UPR BRG ANNULUS	23	6.88
Q25-LWR BRG BYPASS	25	10.77
Q26-LWR BRG FLOW	26	19.70
Q27-LWR BRG PAD AREA	27	11.93
Q28-LWR BRG ANN	28	6.79
Q29-LWR BRG NSG ANN	29	0.98
Q30-LWR BRG RAD HOLES	30	0.27
Q31-LWR BRG NSG ANN	31	0.71
Q32-LWR BRG ANN	32	7.06
Q34-LWR NSG AX HOLES	34	11.48
Q35-TWR BRG ASSEM.HOLES	35	68.22
Q36-THRUST RUNNER FLOW	36	98.69
Q37-THRUST RUNNER FLOW	37	30.47

TABLE 5-11 (Continued)
Sample File Listing for Q = 140 GPM, H = 450 Ft. Case

Mon Apr 28 15:08:45 1997
4/23/97

PRESSURE RESULTS - TRANSFER PUMP FINAL CIRCUIT ANALYSIS - 4 PAD BEARINGS

NODE	PRESSURE PSI	NODE	PRESSURE PSI	NODE	PRESSURE PSI	NODE	PRESSURE PSI	NODE	PRESSURE PSI
1	12.71	2	174.08	3	173.41	4	173.01	5	164.43
6	76.67	7	76.40	8	33.46	9	15.51	10	14.28
11	174.53	12	174.54	13	166.09	14	76.83	15	90.86
16	159.10	17	159.73	18	158.54	19	160.02	20	161.44
21	339.52	22	339.08	23	338.90	24	325.99	25	240.81
26	240.70	27	181.19	28	164.23	29	162.32	30	356.36
31	356.33	32	356.32	33	356.30	34	356.20	35	356.18
36	354.49	37	353.85	38	353.27	39	353.08	40	353.07
41	353.02	42	353.01	43	353.01	44	354.48	45	354.48
46	354.48	47	353.16	48	352.33	49	352.14	50	237.77
51	236.28	53	237.66	54	237.08	55	237.61	56	236.25
57	236.82	58	262.96	59	326.67	60	238.17	61	237.28
62	237.04	63	237.07	64	237.06	65	236.21	66	236.07
67	236.07	68	236.07	69	236.07	70	236.04	71	237.17
72	237.02	73	230.78	74	230.46	75	229.33	76	229.64
77	237.69	78	237.38	79	237.28	80	237.08	81	236.96
82	237.68	83	237.68	84	237.68	85	237.43	86	237.40
87	238.53	88	237.62	90	354.48	91	354.48	92	333.34
93	259.45	94	241.06	95	240.84	96	240.00	97	239.90
98	238.89	99	237.68	100	237.40	101	237.40	102	237.40
103	236.97	104	236.97	106	236.51	107	236.38	108	236.88
109	236.78	110	236.75	111	232.55	112	264.57	113	264.57
114	264.57	115	264.57	116	264.04	117	263.22	118	14.38
119	264.57	120	273.08	121	273.49	122	277.74	123	316.57
124	305.45	125	240.59	126	240.40				

TABLE 5-11 (Continued)
Sample File Listing for Q = 140 GPM, H = 450 Ft. Case

	THRUST LBS.		EQU.	O.D.	I.D.	NODES	P80.D. PSI	P81.D. PSI	AREA INCH**2
#1 IMP - SHROUD	-1585.18	UP	1	8.350	7.600	4 5	173.01	164.43	9.40
#1 IMP - SHROUD	-4774.42	UP	1	7.600	2.708	5 6	164.43	76.67	39.61
#1 IMP - SHROUD	-20.08	UP	1	2.708	2.370	9 10	15.51	14.28	1.35
#1 IMP - SHROUD	-56.07	UP	1	2.370	0.000	1 1	12.71	12.71	4.41
#1 IMP - HUB	207.63	DOWN	1	2.464	2.100	17 18	159.73	158.54	1.30
#1 IMP - HUB	4810.36	DOWN	1	7.600	2.708	13 14	166.09	76.83	39.61
#1 IMP - HUB	1577.83	DOWN	1	8.340	7.600	12 13	174.54	166.09	9.26
#1 IMP - HUB	268.95	DOWN	1	2.100	1.500	18 18	158.54	158.54	1.70
#2 IMP - SHROUD	-494.54	UP	1	2.480	1.500	20 20	161.44	161.44	3.06
#2 IMP - SHROUD	-4681.53	UP	1	8.700	7.600	23 24	338.90	325.99	14.08
#2 IMP - SHROUD	-10892.91	UP	1	7.600	2.970	24 25	325.99	240.81	38.44
#2 IMP - SHROUD	-342.45	UP	1	2.970	2.480	28 29	164.23	162.32	2.10
TOP OF ROTOR	1072.26	DOWN	1	2.400	0.000	50 51	237.77	236.28	4.52
UPPER CORE	1833.41	DOWN	1	3.794	2.128	71 70	237.17	236.04	7.75
LOWER ROTOR	-1781.42	UP	1	3.794	2.128	74 75	230.46	229.33	7.75
LVR.BRG.JOURN-TOP	478.35	DOWN	1	2.680	2.128	76 75	229.64	229.33	2.08
LVR.BRG.JOURN-TOP	632.26	DOWN	1	3.260	2.680	77 76	237.69	229.64	2.71
THRUST RUNNER-TOP	7736.81	DOWN	1	7.000	3.110	112 107	264.57	236.38	30.89
THRUST RUNNER-BTM	-1047.94	UP	1	7.000	6.630	112 113	264.57	264.57	3.96
THRUST RUNNER-BTM	-4504.71	UP	1	6.630	4.720	113 114	264.57	264.57	17.03
THRUST RUNNER-BTM	-2881.80	UP	1	4.720	2.900	114 115	264.57	264.57	10.89
THRUST RUNNER-BTM	-490.07	UP	1	2.900	2.460	115 119	264.57	264.57	1.85
#2 IMP - HUB	7533.95	DOWN	1	7.600	4.720	122 98	277.74	262.96	27.87
#2 IMP - HUB	2544.04	DOWN	1	4.720	3.400	59 122	326.67	277.74	8.42
#2 IMP - HUB	4690.73	DOWN	1	8.700	7.600	21 59	339.52	326.67	14.08
#2 IMP - HUB	1096.67	DOWN	1	3.400	2.500	58 58	262.96	262.96	4.17
TOTAL NO-LOAD THRUST	930.12	DOWN							

5.7.3 Analysis of the Reverse Flow Condition

This operating condition simulates the reversal of the process fluid from the top of the column, through the pump, and back into the tank. The reverse impressed flow and rotor speed were calculated at 26 gpm and 1225 rpm, respectively. The circuit diagram for this condition is illustrated in Figure 5-14. The flow configuration is very similar to Figure 5-13, but the reverse flow through the hydraulic components is shown as a resistance loss instead of a pressure rise (battery). The fluid density and viscosity from Table 5-9 were used as input conditions. The resultant flow distribution under these conditions is presented in Table 5-12. As can be seen, two-thirds of the flow is distributed to the outer stator annulus while the remaining one third is directed through the motor internals.

5.7.4 Analysis of the Shutoff Head Condition

This operating condition simulates the slow speed operation of the pump to generate only enough pressure head from the hydraulics to match the required elevation head of the column at the top of the pump. The circuit diagram illustrated in Figure 5-13 and the fluid properties from Table 5-9 were used for this analysis. An elevation head of 52 feet above suction pressure (node 118) was selected for node 43 of the circuit diagram. Since the pump only matches the elevation head at node 43, there is no through flow up the column. An iterative procedure was utilized to determine that a pump speed of 1190 rpm resulted in the required head differential of 52 feet. A summary of the analysis results is presented in Table 5-13.

FIGURE 5-14

Transfer Pump Hydraulic Circuit Model
Reverse Flow Condition

TABLE 5-12
Transfer Pump Internal Circulation Results
Reverse Flow Condition

Flow Results - Transfer Pump Final Circuit Analysis

	Flow No.	Flow GPM
Q1-#1 Impeller	1	20.89
Q2-#1 Impeller-Lwr Shr	2	5.31
Q3-#1 Impeller-Upr Shr	3	3.54
Q4-#1 Diffuser	4	22.48
Q5-#2 Impeller	5	21.90
Q6-#2 Impeller-Lwr Shr	6	4.10
Q7-Stator Annulus Flow	7	16.97
Q8-Upper Cavity Recirc	8	9.03
Q9-Upper Bearing Housing Annulus	9	1.85
Q10-Thrust Runner Flow	10	9.03
Q11-Upper Rad Bearing Pad Area	11	5.26
Q12-Rotor/Stator Annulus	12	9.03
Q15-Upper Housing Bypass Flow	15	0.47
Q16-Design Flow	16	26.00
Q17-Purge Vent Flow	17	2.32
Q18-Top of Rotor Flow	18	7.18
Q19-Upper Cavity Flow	19	6.71
Q20-Upper Bearing Annulus	20	1.92
Q21-Upper Bearing Radial Holes	21	0.25
Q22-Upper Bearing Housing Annulus	22	2.10
Q23-Upper Bearing Annulus	23	1.68
Q25-Lower Bearing Bypass Flow	25	3.93
Q26-Lower Bearing Flow	26	5.10
Q27-Lower Bearing Pad Area	27	3.54
Q28-Lower Bearing Annulus	28	1.39
Q29-Lower Bearing Housing Annulus	29	0.17
Q30-Lower Bearing Radial Holes	30	0.01
Q31-Lower Bearing Housing Annulus	31	0.16
Q32-Lower Bearing Annulus	32	1.39
Q34-Lower Housing Axial Holes	34	4.09
Q35-Thrust Bearing Assembly Holes	35	21.62
Q36-Thrust Runner Flow	36	30.65
Q37-Thrust Runner Flow	37	9.03
Rotor Speed (RPM)	-1225	
Flowrate (GPM)	-26	
Pressure Drop Across Pump (FT)	51.9	

TABLE 5-13
Transfer Pump Internal Circulation Results
Shutoff Head Condition

	Flow No.	Flow GPM
Q1-#1 Impeller	1	5.96
Q2-#1 Impeller-Lwr Shr	2	5.96
Q3-#1 Impeller-Upr Shr	3	3.36
Q4-#1 Diffuser	4	3.36
Q5-#2 Impeller	5	3.93
Q6-#2 Impeller-Lwr Shr	6	3.93
Q7-Stator Annulus Flow	7	9.11
Q8-Upper Cavity Recirc	8	9.11
Q9-Upper Bearing Housing Annulus	9	1.87
Q10-Thrust Runner Flow	10	9.11
Q11-Upper Rad Bearing Pad Area	11	5.30
Q12-Rotor/Stator Annulus	12	9.11
Q15-Upper Housing Bypass Flow	15	0.48
Q16-Design Flow	16	0.00
Q17-Purge Vent Flow	17	2.34
Q18-Top of Rotor Flow	18	7.25
Q19-Upper Cavity Flow	19	6.77
Q20-Upper Bearing Annulus	20	1.95
Q21-Upper Bearing Radial Holes	21	0.25
Q22-Upper Bearing Housing Annulus	22	2.12
Q23-Upper Bearing Annulus	23	1.69
Q25-Lower Bearing Bypass Flow	25	3.96
Q26-Lower Bearing Flow	26	5.16
Q27-Lower Bearing Pad Area	27	3.58
Q28-Lower Bearing Annulus	28	1.40
Q29-Lower Bearing Housing Annulus	29	0.17
Q30-Lower Bearing Radial Holes	30	0.01
Q31-Lower Bearing Housing Annulus	31	0.17
Q32-Lower Bearing Annulus	32	1.41
Q34-Lower Housing Axial Holes	34	4.12
Q35-Thrust Bearing Assembly Holes	35	20.97
Q36-Thrust Runner Flow	36	30.08
Q37-Thrust Runner Flow	37	9.11
Rotor Speed (RPM)	+ 1190	
Flowrate (GPM)	0	
Pressure Drop Across Pump (FT)	52.0	

5.7.5 Analysis of the Purge Line System

As referenced in Section 5.7, the purge line system is an auxiliary system used to inject clean water into the motor internals prior to pump operation to clean out or purge the motor cavity of any solid particles which have settled out of solution while the pump was not being operated. A one-half inch schedule 40 pipe extends down the column and connects to the pump internals at the upper bearing assembly. Refer to Figure 5-15 for an illustration of the upper bearing cross-section. The injected flow then splits with part of the flow going through the motor cavity and part of the flow extending to the stator annulus. Both flowpaths connect together in the hydraulic end of the pump. The purge flow then winds through the hydraulic components and finally flushes into the tank.

The purpose of the analysis was to determine the internal flow distribution and overall system pressure drop for various input flowrates; thereby creating a system resistance line which can be matched up with the purge line supply pump. The intersection of the supply pump head/flow curve with the system resistance line will determine the amount of flow injected into the pump. In the case of a variable speed supply pump, the internal flow distribution for the selected injection flowrate can be seen. The mathematical circuit model for the purge piping is illustrated in Figure 5-16. The resultant flow distribution and overall pressure drop are tabulated in Table 5-14. The pressure drop as a function of the supply flowrate is illustrated in Figure 5-17. As can be seen from the flow results, approximately two-thirds of the injected purge flow flushes the motor internals while one third flushes the outer stator annulus. This flow distribution is evident over the entire range of flows analyzed. Note that 10 GPM total flow is achieved with a supply pressure of 39.60.

5.7.6 Analysis of Dilution Line System

As referenced in Section 5.7, the dilution line system is an auxiliary system used to inject clean water into the suction end of the first stage impeller to dilute the process fluid in the tank before it is transported into the pump internals. A one and one-half inch schedule 40 pipe extends down the column and connects to the top of the stator. The dilution lines then extend down the stator annulus finally connecting

FIGURE 5-15
Upper Radial Bearing Cross-Section Showing Purge Line Connection

FIGURE 5-16
Transfer Pump Hydraulic Circuit Model Purge Line Operation

TABLE 5-14
Transfer Pump Internal Circulation Results Purge Line Analysis

	Flow No.	Flow GPM	Flow GPM	Flow GPM	Flow GPM
Q1-#1 Impeller	1	8.97	7.21	5.45	3.68
Q2-#1 Impeller-Lwr Shr	2	1.03	0.79	0.55	0.32
Q3-#1 Impeller-Upr Shr	3	0.28	0.19	0.12	0.05
Q4-#1 Diffuser	4	9.72	7.81	5.88	3.95
Q5-#2 Impeller	5	9.26	7.45	5.63	3.80
Q6-#2 Impeller-Lwr Shr	6	0.74	0.55	0.37	0.20
Q7-Stator Annulus Flow	7	2.90	2.35	1.80	1.23
Q9-Upper Bearing Housing Annulus	9	6.06	4.84	3.63	2.41
Q10-Thrust Runner Flow	10	7.10	5.65	4.20	2.77
Q11-Upper Rad Bearing Pad Area	11	1.57	1.27	0.97	0.67
Q12-Rotor/Stator Annulus	12	7.10	5.65	4.20	2.77
Q14-Purge Vent Flow	14	2.22	1.78	1.33	0.89
Q15-Upper Housing Bypass Flow	15	1.72	1.38	1.04	0.70
Q16-Design Flow	16	10.00	8.00	6.00	4.00
Q17-Purge Vent Flow	17	7.78	6.22	4.67	3.11
Q18-Top of Rotor Flow	18	1.04	0.81	0.58	0.36
Q19-Upper Cavity Flow	19	0.68	0.57	0.46	0.34
Q20-Upper Bearing Annulus	20	0.53	0.46	0.40	0.31
Q21-Upper Bearing Radial Holes	21	2.13	1.75	1.35	0.95
Q22-Upper Bearing Housing Annulus	22	3.93	3.10	2.28	1.47
Q23-Upper Bearing Annulus	23	1.80	1.28	0.95	0.63
Q25-Lower Bearing Bypass Flow	25	1.85	1.53	1.24	0.88
Q26-Lower Bearing Flow	26	5.25	4.12	2.96	1.90
Q27-Lower Bearing Pad Area	27	2.51	2.03	1.53	1.04
Q28-Lower Bearing Annulus	28	2.17	1.69	1.19	0.73
Q29-Lower Bearing Housing Annulus	29	0.56	0.40	0.24	0.13
Q30-Lower Bearing Radial Holes	30	0.26	0.18	0.11	0.06
Q31-Lower Bearing Housing Annulus	31	0.30	0.22	0.13	0.08
Q32-Lower Bearing Annulus	32	2.43	1.87	1.30	0.77
Q34-Lower Housing Axial Holes	34	2.15	1.75	1.38	0.95
Q35-Thrust Bearing Assembly Holes	35	1.39	1.10	0.82	0.53
Q36-Thrust Runner Flow	36	5.71	4.55	3.39	2.24
Q37-Thrust Runner Flow	37	7.10	5.65	4.20	2.77
Rotor Speed (RPM)		0	0	0	0
Purge Line Supply Flow (GPM)		10	8	6	4
Pressure Drop Across Pump and Piping (FT)		39.20	26.05	15.40	7.42
(PSI)		16.96	11.27	6.67	3.21

Note: Purge water @ 70°F

FIGURE 5-17
Purge Line Operation Pressure Drop versus Purge Line Flowrate

to four pipes which direct the dilution fluid to the pump intake. At this point the dilution water mixes with the process fluid before it is transported into the pump internals.

The purpose of the analysis was to determine the internal flow distribution and overall system pressure drop for various input flowrates; thereby creating a system resistance line which can be matched up with the dilution line supply pump. The intersection of the supply pump head/flow curve with system resistance line will determine the amount of flow injected into the pump. In the case of a variable speed supply pump, the internal flow distribution for the selected injection flowrate can be seen. The mathematical circuit model for the purge piping is illustrated in Figure 5-18. The resultant flow distribution and overall pressure drop are tabulated in Table 5-15. The pressure drop as a function of the supply flowrate is illustrated in Figure 5-19. As can be seen from the flow results, the distribution of the supply flow to the four individual dilution lines is uniform over the entire flow range analyzed. Note that 105 GPM total dilution flow is achieved with a supply pressure of 98.55 PSIG.

FIGURE 5-18

Transfer Pump Hydraulic Circuit Model

Dilution Line Operation

TABLE 5-15
Transfer Pump Internal Circulation Results
Dilution Line Analysis

	Flow No.	Flow GPM	Flow GPM	Flow GPM	Flow GPM
Q1-Dilution Line #1	1	29.77	25.50	19.80	14.12
Q2-Branch 1-2	2	75.23	64.50	50.20	35.88
Q3-Dilution Line #2	3	25.26	21.66	16.86	12.05
Q4-Branch 2-3	4	49.97	42.84	33.34	23.83
Q5-Dilution Line #3	5	25.43	21.80	16.96	12.12
Q6-Dilution Line #4	6	24.54	21.04	16.38	11.71
Q7-Suply Flow	7	105.00	90.00	70.00	50.00
Q8-Dilution Line #5/#1	8	26.25	22.50	17.50	12.50
Q9-Dilution Line #6/#2	9	26.25	22.50	17.50	12.50
Q10-Dilution Line #7/#3	10	26.25	22.50	17.50	12.50
Q11-Dilution Line #8/#4	11	26.25	22.50	17.50	12.50
Rotor Speed (RPM)		0	0	0	0
Dilution Line Supply Flow (GPM)		105	90	70	50
Pressure Drop Across Pump and Piping (FT)		174.58	129.67	79.95	41.97
(PSI)		75.53	56.10	34.59	18.16

Note: Purge water @ 70°F

FIGURE 5-19

Dilution Line Operation Pressure Drop versus Dilution Line Flowrate

5.8 NPSH Requirements

- Cavitation testing of the first stage Transfer Pump impeller was conducted, under sub-contract, by the Southwest Research Institute, San Antonio, Texas. The Institute prepared a special test loop facility to accommodate the low NPSH hydraulic design of the transfer pump first stage impeller.

The test article consisted of the prototype impeller with the screened inlet and wear rings. The test medium was cold water, and the impeller was operated at the design speed of 3860 rpm.

NPSH data were taken between 80 and 180 gpm in 10 gpm increments. The design flow rate is 140 gpm. Cavitation characteristics of the impeller were established with the impeller operating at a constant flow rate. Eleven series of tests were conducted, one each for constant flow rates starting at 80 gpm and ending at 180 gpm. While maintaining a given flow, the pump head was recorded for successive reductions of the impeller suction pressure. During any one test series, at least seven different sets of data, each of which was considered a pump operating condition, were recorded where no sign of head fall-off was apparent. A sufficient number of test conditions were recorded to establish the occurrence of 3% head fall-off performance degradation due to cavitation. The dwell time under cavitating performance was minimized during each test series so as to prevent pitting of the impeller.

Figure 5-20 shows the NPSH required by the impeller for various flow rates, bounding the range of operation of the Transfer Pump. The NPSH at design flow was measured to be 10.6 feet minimum. It is recommended that 25% NPSH margin be used for actual operation of the Transfer Pump in the hazardous waste tanks (i.e., 13.25 feet of NPSH at the design flow point, 140 gpm).

Figure 5-20 Required NPSH

5.9 Internal Collection and Circulation of Particles

The Transfer Pump process fluid could contain a maximum of 50% solids by volume. When the pump is stopped the solids in the motor, particularly the rotor cavity, will settle out over time. Large particles will settle out in a matter of seconds or minutes while very small particles will take days or weeks to settle. The two major rotor cavity locations for particulate settlement are above the upper radial bearing and above the lower radial bearing.

The total volume above the upper radial bearing is approximately 20.90 cubic inches. A 50% solids mixture in this volume contains 10.45 cubic inches of particulates. Assuming none of these solids pass through the upper bearing bypass holes, this results in a solid, settled-out mass from the top of the radial bearing bypass holes to an elevation approximately 0.10 inch above the upper journal retaining nut. Initiation of purge water flow immediately after pump shut-down will flush substantial amounts of solids out of the motor cavity. It is estimated that flushing 10 motor cavity volumes of fluid (35 gallons) will reduce the solids content by 90%. This would leave only 1.0 cubic inch of solids above the upper radial bearing. This indicates the significant benefit of flushing the motor with purge water immediately after operation. Note that the fill and drain model testing identified in Section 12.0 did not indicate a significant accumulation of solids above the upper radial bearing.

The total volume between the upper and lower radial bearings is 101.7 cubic inches. A 50% solids mixture in this volume produces a 50.9 cubic inch solid mass above the lower radial bearing, assuming no solids go down the lower radial bearing bypass holes. This mass would occupy a space from the top of the lower bearing to an elevation of 4.6 inches up into the rotor/stator annulus (9.8 inches above the top of the lower radial bearing). With an estimated 90% reduction in solids due to flushing 10 motor cavities of fresh water through the motor cavity, the volume of solids between the upper and lower bearings could be reduced to 5.1 cubic inches. This is a solid mass that extends only 1.4 inches above the lower bearing. Again, this indicates the significant benefit of flushing the motor immediately after operation. The fill and drain model testing identified in Section 12.0 revealed that accumulation of solids in the motor cavity was a primary concern. In addition to flushing with purge

water, the effects of solids collection above the lower radial bearing are mitigated by the "grinder" built into the lower journal nut which will break up and disperse agglomerated solids.

The motor cooling inlet is designed to minimize the quantity and size of particles entering the motor cavity. There are four pairs of dilution water channels spaced 90° apart which break the process fluid discharge flow annulus into four quadrants. The quadrants are connected hydraulically at the discharge of the second stage diffuser and at the annular space above the dilution channels. Each quadrant has one 1.12 inch diameter main discharge hole (4 main discharge holes total) and one 0.38 inch diameter motor cooling inlet hole (4 motor cooling inlet holes total). The motor cooling inlet holes are at an elevation 1.75 inches higher than the main discharge holes, and the motor cooling inlet holes are separated circumferentially from the main discharge holes by 1.70 to 3.23 inches (20° to 38°).

The design point main discharge annulus flow rate is approximately 172 gpm total or 43 gpm per quadrant (Q_7 of Figure 5-12). At the design point the motor cooling flow rate is about 32 gpm or 8 gpm per quadrant (Q_8 of Figure 5-12). Since two dilution tubes occupy the volume of each quadrant, the area of each dilution tube must be subtracted from the main discharge flow area of the quadrant. The dilution tube area is:

$$A_{dt} = (W \times (L - W)) + (\pi / 4) \times W^2$$

where: W is the width of the tube, 0.25 inch
 L is the overall length of the tube, 1.25 inches
 A is the area of a tube, in²

$$A_{dt} = (0.25 \times (1.25 - 0.25)) + (\pi / 4) \times 0.25^2$$

$$A_{dt} = 0.30 \text{ in}^2$$

The flow area of a main discharge quadrant is calculated as follows:

$$A_q = (90 / 360) \times (\pi / 4) \times (OD^2 - ID^2) - 2 \times A_{dt}$$

where: OD is the stator jacket ID, 10.38 inches
ID is the stator shell OD, 9.75 inches
Aq is a quadrant flow area, in²

$$Aq = (90 / 360) \times (\pi / 4) \times (10.38^2 - 9.75^2) - (2 \times 0.30)$$

$$Aq = 1.89 \text{ in}^2$$

The main discharge flow average velocity in each quadrant is:

$$V_{qm} = 0.321 \times Q_q / A_q$$

where: Qq is the quadrant flow rate, 43 gpm
Vqm is the quadrant main flow velocity, ft/s

$$V_{qm} = 0.321 \times 43 / 1.89$$

$$V_{qm} = 7.30 \text{ ft/s}$$

At design conditions 35 gpm flows out the main discharge holes in each quadrant (140 gpm total). Therefore, as previously noted the flow continuing up each annulus quadrant to the motor cooling inlet holes reduces to 8 gpm. The average flow velocity in this region reduces to:

$$V_{qc} = 0.321 \times 8 / 1.89$$

$$V_{qc} = 1.36 \text{ ft/s}$$

The size of particles which will be lifted up in the flow stream can be estimated by evaluating the terminal velocity of the particle settling in still fluid under the influence of gravity, buoyancy and drag forces. When the terminal velocity exceeds the fluid upward velocity through the main flow annulus, then the particle will not flow up the annulus with the fluid. When the terminal velocity is less than the fluid velocity, then the particle will flow up the annulus. Note that this is a simplification that does not account for vortex currents or localized turbulence which can boost local fluid velocities above the average velocity. The simplified situation is described by the following equations:

$$V_g = ((D_p / 12)^2 \times (12^3 \times \rho_p - 62.39 \times SG)) / (18 \times \mu_f)$$

or

$$D_p = ((2592 \times V_g \times \mu_f) / (12^3 \times \rho_p - 62.39 \times SG))^{0.5}$$

where: V_g is the particle terminal velocity, ft/s
 D_p is the particle diameter, inch
 ρ_p is the particle weight density, lb/in³
 SG is the fluid specific gravity, 1.00 to 1.50
 μ_f is the fluid viscosity, 2.09×10^{-5} to 6.27×10^{-4} lb-s/ft²
 (1 to 30 Centipoise)

The largest heavy, steel particle (0.283 lb/in³ density) that can be lifted up in the main discharge flow stream (7.30 ft/s velocity) which has the minimum specific gravity (1.00) and viscosity (2.09×10^{-5} lb-s/ft² or 1.00 Centipoise) is 0.030 inch diameter as calculated below:

$$D_p = ((2592 \times 7.30 \times 2.09 \times 10^{-5}) / (1728 \times 0.283 - 62.39 \times 1.00))^{0.5}$$

$$D_p = 0.030 \text{ inch}$$

In contrast, when the fluid velocity decreases to 1.36 ft/s at the entrance to the internal cooling holes, the particles that can be carried into the motor decrease from the 0.030 inch diameter to 0.013 inch diameter as shown below:

$$D_p = ((2592 \times 1.36 \times 2.09 \times 10^{-5}) / (1728 \times 0.283 - 62.39 \times 1.00))^{0.5}$$

$$D_p = 0.013 \text{ inch}$$

If the fluid viscosity increases to 30 Centipoise (6.27×10^{-4} lb-s/ft²) and the fluid specific gravity increases to 1.50, the particle size which can be carried up in the main discharge flow stream (7.30 ft/s) increases to 0.173 inch diameter as shown below:

$$D_p = ((2592 \times 7.30 \times 6.27 \times 10^{-4}) / (1728 \times 0.283 - 62.39 \times 1.50))^{0.5}$$

$$D_p = 0.173 \text{ inch}$$

With a decrease in fluid velocity to that entering the motor (1.36 ft/s), the particle size that can be carried into the motor decreases to 0.075 inch as calculated below:

$$D_p = ((2592 \times 1.36 \times 6.27 \times 10^{-4}) / (1728 \times 0.283 - 62.39 \times 1.50))^{0.5}$$

$$D_p = 0.075 \text{ inch}$$

Examination of the above results indicates that for a given process fluid, the maximum size of the particles entering the motor is only 43 percent of the maximum size of the particles that can be pumped out the tank. Not only does the orientation of the motor inlet holes relative to the main discharge holes reduce the concentration of suspended particles entering the motor, but the orientation also precludes large metallic particles which can span the rotor-to-stator can gap (0.142 inch) from entering the motor under all fluid viscosity and specific gravity conditions.

6.0 MOTOR EVALUATION

6.1 Motor Design Characteristics

The transfer pump design incorporates canned motor technology with which WEMD has more than 40 years of design experience. The motor for the transfer pump, designated Series 513, is a two pole, 460 volt, three-phase, 60 hertz, squirrel cage induction type machine which operates at a synchronous speed of 3600 RPM. The motor is cooled by the surrounding process fluid which flows over the outer diameter of the stator core as well as through the rotor/stator gap. The motor is designed to meet the requirements set forth in the transfer pump design specification while conforming to the NEMA MG-1 standard for electrical machines. The input power to the motor is supplied via a variable frequency drive (VFD). The design data for the transfer pump at rated horsepower conditions (460 volts, 71 hertz) is contained in Table 6-1.

6.2 Motor Materials Compatibility with Environment

The insulation system used in the design of the transfer pump motor is a high temperature insulation system with which WEMD has hundreds of thousands of hours of operating experience. In addition, the WEMD Class N insulation system has been successfully tested in radiation environments of up to 1000 Megarads; far in excess of the 58.7 Megarads expected in the transfer pump operating environment. To confirm these test results, the following materials were reanalyzed by Westinghouse's Science and Technology Center (WSTC): glass-mica-glass tape, untreated glass tape, Kapton polyimide film, several silicon bonding and potting resins, and NEMA-G7 silicon based micarta. The analysis verified that the materials in the WEMD insulation will perform satisfactorily in the radioactive environment. It was also discovered that these materials will perform adequately at temperatures above the 200°C (392°F) Class N temperature rating. However, because the ML coated magnet wire has a temperature rating of 220°C (428°F), the above mentioned materials can only be subjected to temperatures at or below 220°C (428°F) without degrading their dielectric performance or the

Table 6-1

DESIGN DATA FOR TRANSFER PUMP		
RATING:	460 Volts, 71 Hz, 3 Phase, 60 Hp, .70 PF, 2 Pole, 3860 RPM, 4260 RPM (Synchronous). Input Power = 67 KW, Breakdown Torque = 141% FLT, Locked Rotor Torque = 87% FLT. Rated Current = 120 Amps, No-Load Current = 50 Amps, Locked Rotor Current = 255 Amps.	
ARMATURE:	Type Winding - Throw - Strand Size (Bare) - (ML Insulated) - Number of Slots - Slot Size - Series Turns/Phase - Resistance/Phase - Gap Density - Core Density - Tooth Density - Current Density - Grade of Iron - Can Material - Can Thickness -	2 Parallel Wye (Star) 1 to 10 (3/4 pitch) 2 Coils/Slot 11 Turns/Coil 7 Strands/Turn 0.0508 in. Diameter (AWG #16) 0.0560 in. Diameter 24 0.680 x 1.625 in. (Round Bottom) 44 0.06115 Ohms @ 25°C 30 KL/in² 79 KL/in² 83 KL/in² 4200 Amps/in² 0.025" Thick (10502DW) Hastelloy C-276 0.033 in.
FIELD:	Rotor Winding - Number of Slots - Slot Size - Rotor Bar Dimensions - Rotor Bar Material - Connection Ring Material - Cage Resistance - Grade of Iron - Can Material - Can Thickness -	Squirrel Cage 17 0.338 to 0.254 x 0.590 in. (Trapezoidal) 0.330 to 0.246 x 0.560 in. (Trapezoidal) UNS Alloy C70200 (1.70% Ni, 0.15% Mn) ETP Copper (99.90% Cu) 0.14606 Ohms @ 25°C 0.025" Thick (10502DW) Hastelloy C-276 0.033 in.
LOSSES:	Stator I²R - Rotor I²R - Fluid & Friction - Stator Can - Rotor Can - Load - Total Efficiency -	3.8 KW 5.6 KW 8.1 KW 3.7 KW 0.6 KW 0.4 KW 22.2 KW 67%
COOLING:	Process fluid over outer diameter of stator and through rotor/stator gap.	
REACTANCES:	X1 - 0.632 Ohms (Operating Conditions) X2 - 0.608 Ohms (Operating Conditions) XM - 5.751 Ohms	

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21-Apr-95

40 year life of the insulation. The final report documenting the compatibility of the motor insulation materials with the high radiation operating environment, as well as the thermal characteristics of these materials, is attached in the Appendix.

6.3 Motor Performance Characteristics

6.3.1 Normal Operating Points

The transfer pump motor was designed to operate at a speed of 3455 RPM, 3600 RPM synchronous, under normal operating conditions (460 volts, 60 hertz). The primary basis for selecting this operating point was the assumption that the pump will run at this speed for approximately 80% of its design life. The input power, line current and other parameters describing the performance of the motor at this design point are shown in Table 6-2.

For the remaining 20% of its life, it was estimated that the motor will operate at a speed of 3860 RPM. This 71 hertz operating point is where the pump attains maximum flow and, because of the increased frequency and thus higher line current, reaches maximum temperature. The parameters describing the performance of the motor at this design point can also be found in Table 6-2. To further illustrate the capabilities of the motor at the 71 hertz operating point, various plots depicting the performance of the machine are shown in Figures 6-1 thru 6-11. In addition to the 60 and 71 hertz operating points, Table 6-2 also shows the performance of the motor at a few additional points where the pump may be required to operate.

6.3.2 VFD Start and Operation

Input power to the transfer pump will be supplied via a variable frequency drive (VFD) to allow for variable speed operation. The VFD motor controller will convert the 460 volt, three-phase, 60 hertz utility power to adjustable voltage and frequency, three-phase, AC power for stepless motor speed control. See Section 3.1.5 for a more complete description of the capabilities of the VFD.

Table 6-2

TRANSFER PUMP

Variable Speed Performance

Parameter	24 Hz	30 Hz	57 Hz	60 Hz	62 Hz	71 Hz
Load (Hp) **	2.3	4.5	31	36	40	60
Shaft Speed (RPM)	1420	1770	3290	3455	3550	3860
Slip	1.3%	1.7%	3.7%	4.0%	4.6%	9.3%
Input Power (KW) **	4	6.6	36	41	45	67
Terminal Voltage (Volts)	184	230	437	460	460	460
Line Current (Amps)	52	53	76	81	85	120
Power Factor	.24	.31	.62	.64	.67	.70
Efficiency	44%	51%	64%	65%	66%	67%
Full-Load Torque (Ft-lbs)	8.5	13	49	55	59	82
Starting Torque (Ft-lbs)	147	139	110	108	100	71
(% of Full Load Torque)	(1730%)	(1069%)	(224%)	(196%)	(169%)	(87%)
Pull-Out Torque (Ft-lbs)	154	160	167	167	156	116
(% of Full Load Torque)	(1812%)	(1231%)	(341%)	(304%)	(264%)	(141%)
Current Density (Amps/in ²)	1830	1885	2685	2845	2985	4200
Stator MWST (°C)	16	17	36	40	44	91

J. A. Zastawnik

21-Apr-85

** Based on process fluid with the following properties:
Viscosity: 30 Centipoise.
Specific Gravity: 1.50.

The VFD has two basic modes of operation: a constant torque mode, and a constant voltage mode. The VFD operates in the constant torque mode when the input frequency ranges from 0 to 60 hertz or rated frequency. When operating in this mode, the volts/hertz ratio of the machine is held constant. By holding the volts/hertz ratio constant, the magnetic flux of the machine is also fixed thus enabling the motor to produce a constant torque throughout this range. The VFD operates in the constant voltage mode whenever the input frequency is increased beyond 60 hertz or rated frequency. In this mode the terminal voltage is held constant thus decreasing the volts/hertz ratio of the machine. When operating in this mode, the performance of the motor decreases because of the increased losses due to the higher line current and increased motor temperatures.

When starting the transfer pump with the VFD, motor operation will be initiated at a low frequency, 20 to 30 hertz, and the frequency increased at a selected rate until the desired operating speed is obtained. For speeds less than 3600 RPM, the VFD will operate in the constant torque mode. At speeds greater than 3600 RPM, the VFD will operate in the constant voltage mode. For the transfer pump, the VFD will only operate in the constant voltage mode when the 3860 RPM (71 hertz) operating point is selected.

6.3.3 Reverse Flow Start

A reverse flow condition exists when pump operation is terminated and suction is broken in the line. When this scenario occurs, the process fluid reverses direction and flows back through the line thus causing the impeller to rotate in the opposite direction. In order to overcome this condition, the starting torque of the motor needs to be increased. The torque needed to overcome this reverse flow condition is 19 ft-lbs (See Section 5.6 for the hydraulic evaluation). Since the motor develops 139 ft-lbs. of torque when started at 30 hertz, the reverse flow condition is not seen as a problem. Furthermore, the motor also develops a sufficient amount of torque when line started at 60 hertz to overcome this condition.

6.3.4 Locked Rotor Analysis

A locked rotor event will occur when the rotor is blocked or kept from rotating. This situation will arise if there is any foreign material blocking the impeller, clogging the bearings, or lodged in the rotor/stator gap. The input power, locked rotor current, and locked rotor torque at the 71 hertz design point are 56 KW, 255 amps, and 70 ft-lbs. respectively. See Sections 7.1.2 and 7.2 for the thermal profile and other details pertaining to the locked rotor analysis.

6.4 Motor Protection Requirements

To prevent the motor from being damaged during a locked rotor event, power surge, loss of power, or some other unforeseen transient, the VFD must be programmed to correctly detect and remedy any of these conditions. Because there is not a breaker system defined at this time, the VFD will act as the primary circuit breaker/protection device for the transfer pump. The VFD can be programmed to terminate the power to the machine at a specified time limit after a locked rotor event is detected, or if a voltage spike or power surge occurs. In addition, circuit breaker trip settings, fault detection settings, frequency ramp times, normal operating points, etc., can all be programmed into the VFD to provide added protection during operation. The exact operational modes of the VFD will be defined at a later date when the full capabilities of the drive are understood and can be utilized to their fullest advantage. See Section 3.1.5 for a more complete description of the capabilities of the VFD.

Transfer Pump - S.O. V001 - Final Design Run (71hz) - Case JAZ05 - 4/21/95

Motor Performance at Rated Voltage

Plot No. 3

Fri Apr 21 14:01:02 1995

Speed (RPM)
Figure 6-1

Transfer Pump - S.O. V001 - Final Design Run (71hr) - Case JAZ05 - 4/21/95

Motor Performance

Figure 6-2

Plot No. 4
Fri Apr 21 14:01:02 1995

Transfer Pump - S.O. V001 - Final Design Run (71hz) - Case JAZ05 - 4/21/95

Motor Performance

Plot No. 6

Fri Apr 21 14:01:02 1995

Horsepower (HP)

Figure 6-3

Transfer Pump - S.O. V001 - Final Design Run (71hz) - Case JAZ05 - 4/21/95

Motor Performance

Horsepower (HP)
Figure 6-4

Plot No. 6

Fri Apr 21 14:01:02 1995

Transfer Pump - S.O. V001 - Final Design Run (71hz) - Case JAZ06 - 4/21/95

Motor Performance

Plot No. 7

Fri Apr 21 14:01:02 1995

Horsepower (HP)

Figure 6-5

Transfer Pump - S.O. V001 - Final Design Run (71hz) - Case JAZ05 - 4/21/95
Starting Curve at 90.0% of Rated Voltage

Figure 6-6

Plot No. 8
Fri Apr 21 14:01:02 1995

Transfer Pump - S.O. V001 - Final Design Run (71hz) - Case JAZ05 - 4/21/96

Starting Curve at 80.0% of Rated Voltage

Plot No. 9

Fri Apr 21 14:31:02 1996

Time (Seconds)

Figure 6-7

Transfer Pump - S.O. V001 - Final Design Run (71 Hz) - Case JAZ05 - 4/21/86

Locked Rotor Set. Curve

Line Volts
Figure 6-8

Printed Apr 21 14:01:02 1986

Print No. 10

Transfer Pump - S.O. V001 - Final Design Run (71hz) - Case JAZ05 - 4/21/95

Locked Rotor Sat. Curve

Plot No. 11

Fri Apr 21 14:01:02 1995

Line Volts
Figure 6-9

Transfer Pump - S.O. V001 - Final Design Run (71hz) - Case JAZ05 - 4/21/96

No Load Sat. Curve

Line Volts
Figure 6-10

Plot No. 12
Fri Apr 21 14:01:02 1996

Transfer Pump - S.O. V001 - Final Design Run (71hz) - Case JAZ05 - 4/21/95

No Load Sat. Curve

Plot No. 13

Fri Apr 21 14:01:02 1995

Line Volts
Figure 6-11

6.5 Variable Frequency Drive (VFD) Information

6.5.1 General Overview

Input power to the transfer pump will be supplied via a variable frequency drive (VFD) to allow for variable speed operation. The VFD motor controller will convert the 460 volt, three-phase, 60 hertz utility power to adjustable voltage and frequency, three-phase, AC power for stepless motor speed control. The VFD will utilize a Pulse Width Modulation (PWM) scheme for the power conversion. This space vector modulation strategy is implemented using Insulated Gate Bipolar Transistor (IGBT) technology to deliver efficient and cost-effective drive control.

6.5.2 Hardware Overview

There are several key components that are common to all IGBT drives. These components include the following:

a. voltage input connectors	e. inputs and outputs
b. rectifier circuit	f. keypad/LCD display
c. gate driver board	g. IGBT's
d. microprocessor board	h. drive output connectors

These key elements and their interrelationships are shown in Figure 6-12. This illustration also shows several common options that can be added to the drive system. To understand how the hardware components of the drive work, it is helpful to understand the functions of each of the key system elements as well as their relationships to each other.

The voltage input connectors are used to connect an external 3-phase, 480 VAC, 60 Hz signal to the drive system. This is the source of power that operates the VFD and ultimately drives the motor. The rectifier circuit is responsible for converting the external fixed frequency input voltage into a DC voltage which is then stored in large

capacitors within the unit. This DC voltage is then passed to the gate driver board which in turn feeds a special group of transistors called IGBT's. The IGBT's convert the DC voltage into a 3-phase output voltage that varies in frequency from 0 to 120 hertz as determined from the microprocessor board. Lastly, the 3-phase drive output is sent to the three drive output connectors of the system. The motor to be controlled, the transfer pump in this case, is then connected to these three terminals for operation.

Figure 6-12

The keypad and LCD display, shown in Figure 6-13, provide a local operator interface to the system. From this interface the operator can configure the system, set the internal clock, program control parameters, perform diagnostic functions, program four display meters, and view fault (error) messages, diagnostic messages, and a historical list of internal data (such as parameter values) for troubleshooting or testing. The keypad/LCD display is mounted on the front of the drive enclosure. In addition to the keypad/LCD interface, a 34-connection terminal block is located on the

microprocessor board of the system. Control inputs and outputs are connected to these terminals. The functions of these inputs and outputs are programmable by the operator, although a standard configuration is programmed at the factory by default. Furthermore, an RS232 connector will be mounted on the front panel of the drive enclosure, near the keypad, to allow serial communication between the drive and a personal computer (for the uploading or downloading of system information), or between the drive and a printer (for report printouts), to be easily accomplished.

Figure 6-13

6.5.3 Control Modes

The drive system has several default control modes of operation. These control modes include a stop mode, a local manual mode, a remote manual mode, an automatic mode, a jog mode, and a safety override mode. A brief description of each of the control modes is as follows:

- 1) The stop mode is used to shut down the drive in a controlled manner regardless of its current state. To enter this mode, the operator must either press the manual stop button on the system keypad, or press a user-supplied digital input switch that is programmed as a manual stop input.
- 2) The manual mode allows the operator to manually control the speed of the motor in one of two forms: local manual mode or remote manual mode. To enter the local manual mode, the manual start button on the system keypad must be pressed. In this mode, the desired motor speed is selected manually using the up and down arrow buttons on the system keypad. To operate the drive in the remote manual mode, a contact closure connected to the manual start digital input must be provided. Once in this mode, the desired velocity is selected manually using a user-supplied potentiometer connected to the drive system.
- 3) In the automatic mode, the operator chooses an input to be used as the desired velocity input. This input can be one of two analog inputs, a PID control loop input, or any one of eight programmable setpoints that correspond to the eight digital inputs to the system. In this mode, the drive utilizes a feature called speed profiling which uses the selected input signal, along with other programmable parameters, to provide a new desired velocity input signal. This new velocity signal, which is a function of the original signal, provides minimum, maximum, and proportional output control based on a selectable reference input. To enter the automatic control mode, the automatic button on the system keypad must be selected, or a user-supplied digital input switch,

programmed as an auto run input, must be pressed.

- 4) The jog mode is similar to automatic mode except that a programmable jog speed parameter defines the desired velocity and there is no speed profiling or auto on/off control. The jog speed setpoint is used to directly control the desired velocity with the speed profiling software routine being completely bypassed. To enter this mode, a user-supplied digital input switch, programmed as a jog mode input, must be pressed.
- 5) The safety override is a fail safe feature that takes priority over all other operational modes including the manual and automatic modes. In the safety override mode, the velocity setpoint is obtained from a pre-programmed safety override parameter setting. To enter this mode, a contact closure to the pre-programmed safety override mode digital input must be provided.

6.5.4 Drive Features

In addition to the key components discussed in the previous sections, the IGBT drive systems contain additional options, performance enhancements, and technological innovations that make these drives efficient, cost-effective, and easy to use. Some of these features are as follows:

- Modular design that provides lower cost and easier addition of "pre-engineered" options
- Built-in keypad and 2x24 LCD interface provides English language messages for configuration, programming, and diagnostics functions.
- Programmable inputs and outputs
- Menu monitoring capabilities for all inputs and outputs
- Programmable on/off feature for normal speed demand, PID command, or PID output
- Standard line reactor to add inductance for controlled communication of power devices and to reduce voltage harmonics

- Programmable serial port communication for PC file transfers, printing, etc.
- Enhanced "catch-a-spinning-load" feature
- Full-load and no-load boost features
- Built-in heat sink thermistor sensing to control operations as a function of the heat sink temperature
- Enhanced power loss ride through feature won't trip if power is lost and there is no need for an uninterruptable power supply (UPS)
- Undervoltage operation
- Built-in underload function to automatically shut off pumps if insufficient fluid levels exist
- Built-in real time clock
- Standard historic and diagnostic log features
- Slip compensation function
- Resonance (or critical speed) avoidance feature for the mechanical system
- Eight preset operational speeds are built-in
- Programmable loss of input signal feature with programmable hysteresis
- Programmable ramp stopping or coast stopping feature
- Four programmable meter displays which are programmed via the keypad

6.5.5 VFD Technical Specifications

Specification	Value
HP Range	125 HP
Input	460 VAC +10%, -5% at rated output, 57-63 Hz
Input Power	Non-phase sequence sensitive
Input Power Factor	0.95 Displacement power factor at all operating speeds
Frequency Drift	$\pm 0.5\%$
Voltage Regulation	$\pm 1\%$ Rated value, no-load to full-load
Speed Regulation	3% (motor dependent)
Speed Range	40:1 (motor dependent)
Control Circuits	24 VDC
Design	Voltage source, pulse width modulated
Ambient Temperature	0-40°C
Altitude	Up to 3,300 feet above mean sea level (MSL). Drive is derated at altitudes above 3,300 feet.
Current Limit	50%-150% adjustable
Acceleration Rate	2-600 sec, selectable
Deceleration Rate	2-600 sec, selectable
Overload Capacity	100% continuous, 150% for 1 minute
Input Speed Command	4-20 mA (isolated), 0-10 VDC (nonisolated)
Output Frequency	0-120 Hz
Carrier Frequency	3 KHz
Enclosure	Open Chassis
Chassis Dimensions (H x W x D)	37.5" x 16.6" x 19.7"
UL Listing	Adheres to UL code 508C for power conversion equipment. The enclosure is panel approved to UL code 508A.

6.6 Variable Frequency Drive (VFD) Sizing and Operation

6.6.1 VFD Sizing and Selection

Selecting the size of a VFD to operate a motor is usually not a difficult task since the drives are rated to deliver a specific amount of power. This rating is usually based on the assumption that the VFD will be driving a NEMA type B (high power factor, high efficiency) polyphase induction motor. By classifying the drives in this manner, a drive rated at 60 HP should sufficiently operate a 60 HP NEMA type B motor. However, because of the poor power factor associated with canned induction motors, a VFD with a larger output rating will be necessary to operate the transfer pump.

The VFD for the transfer pump was selected based on the rating of the transfer pump motor at the 71 hertz operating point (the operating point with the largest line current). Because of the low power factor of the motor, the rated current of the motor at this operating point is 125 amps. In order for the motor to operate satisfactorily at this design point, a 125 HP drive (156 amps rated output current) is required to supply the motor. In addition, the 125 HP drive can supply 150% rated current (approximately 188 amps) to the motor for 60 seconds thus allowing the motor to produce 125 ft-lbs of torque (at the 71 hertz operating point) when needed. However, the drive circuitry will limit an overcurrent condition to 5 seconds thus protecting the motor from exhibiting a locked rotor condition of more than 5 seconds (even though the motor can handle a locked rotor condition in excess of 10 seconds).

6.6.2 VFD Operation

The VFD has two basic ranges of operation: a constant torque range, and a constant voltage or constant horsepower range. The VFD operates in the constant torque range when the input frequency is between 0 and 60 hertz (rated frequency). When operating in this range, the volts/hertz ratio of the motor is held constant. By holding the volts/hertz ratio constant, the magnetic flux of the machine is fixed thus enabling the motor to produce a constant torque throughout this range. The VFD operates in

the constant voltage or constant horsepower range whenever the input frequency is increased beyond 60 hertz or rated frequency. In this range the terminal voltage is held constant thus decreasing the volts/hertz ratio of the machine. When operating in this range, the performance of the machine is decreased because of the increased motor losses due to the higher line current and increased motor temperatures. In addition, since the frequency is increased and the horsepower held constant, the developed torque of the motor is reduced. For the transfer pump, at speeds less than 3600 RPM, the VFD will operate in the constant torque range with the voltage being boosted at the lower speeds. At speeds greater than 3600 RPM, the VFD will operate in the constant voltage or constant horsepower range. The VFD will only operate in this range when the 3550 (62 hertz) or the 3860 RPM (71 hertz) operating points are selected.

When starting the transfer pump with the VFD, motor operation will be initiated at a frequency of 14 hertz. At this frequency, the terminal voltage will be boosted 12 percent by the VFD (to approximately 116 volts) to increase the developed torque of the motor. The voltage boost (the amount of voltage, above and beyond the voltage needed to keep a constant volts/hertz, that is needed to maintain rated torque) will then be decreased as the line frequency is increased until the original volts/hertz ratio is once again obtained (this should occur at approximately 30 hertz). The voltage and frequency will then be increased, at a constant volts/hertz ratio, until the desired operating speed is obtained.

6.6.3 Drive Enclosure

Because of the severe weather conditions that are present at the work site, the VFD enclosure must be able to withstand the following conditions: air temperatures from -30 to 50°C, relative humidity from 10 to 100 percent, average wind velocity of 12.4 km/hr, peak wind gust velocity of 129 km/hr, and the VFD enclosure must withstand wind driven rain, snow, and sand. To accomplish this, the drive will be enclosed in a NEMA-3R enclosure. The enclosure will be equipped with separate heating and air conditioning systems to provide a climate controlled environment for the drive to operate in. To provide power for the HVAC components, a 10 KVA stepdown transformer will be included within the enclosure to reduce the 460 volt incoming voltage to a usable voltage (230 VAC) for the HVAC components. In addition to the drive and HVAC components, the VFD enclosure will also be equipped with line reactors and circuit breakers as shown in Figure 6-14. The line reactors will be used to add inductance to the outgoing voltage signal to aid in the reduction of voltage harmonics and the circuit breaker/disconnect switch will serve as the main breaker for the entire VFD system. An alarm beacon has also been added to the top of the enclosure housing to indicate any faults that may occur.

The basic dimensions of the enclosure are 72" x 40" x 24" (H x W x D) with the overall dimensions being 84" x 42" x 28" (including the heat shield and alarm beacon). These dimensions are based on a preliminary drawing of the VFD which was supplied to us by the vendor. The size of the enclosure may change based upon the final size of the drive components. The enclosure itself also contains a heat shield which is mounted to the top of the cabinet, drip shields which are mounted over the cabinet door and the VFD heat sink, and window which will house the remote keypad and RS232 output jack.

6.6.4 Power Draw

The transfer pump VFD is rated at 125 HP, with a continuous current rating of 156 amps, at an input voltage of 460 volts. The maximum current that the drive can supply is 234 amps. In addition, the accompanying 10 KVA stepdown transformer and ventilation components draw about 21 amps for a total power draw of approximately 255 amps. To protect the VFD and the additional system components from during an overload condition, a 300 amp circuit breaker has been added to the input of the system.

The total power draw of the system is based on this 300 amp circuit. The KVA requirement needed to operate the system is therefore 240 KVA based on a 460 volt supply voltage. It is therefore recommended that power be supplied to the transfer pump VFD system via a transformer with a minimum KVA rating of 250 KVA. This value excludes any addition equipment (additional pumps, motors, utility power, etc.) that may already be connected to, and drawing power from, the respective transformer or supply bus.

Figure 6-14

7.0 THERMAL ANALYSIS EVALUATION

7.1 MWST

During the initial design phase of pump motors it is important to have an accurate knowledge of motor temperature distribution and maximum winding insulation temperature. Such knowledge is important for two reasons. First, to design the motor stator to utilize all possible modes of cooling effectively and maintain maximum motor efficiency. Second, to assure the motor meets the thermal design requirement of a 200°C maximum winding surface temperature.

In the past, the maximum winding surface temperature, MWST, for canned motor designs were determined using an analog computer. Under Westinghouse contract, the analog computer was designed and built in 1957/1958 by Professor G. W. Penney of Carnegie Institute of Technology. The analog computer is a resistive electrical network which is representative of a steady state thermal circuit. Variable resistors and variable current inputs can be set to represent a various number of thermal situations over a wide range. The potential differences between points on the electrical network are metered which represent direct temperature differences. By 1959 the analog computer proved to be a reliable and useful tool. During the following years, several improvements were made which further improved the motor temperature results.

Since the analog computer is employed only for new motor temperature design analysis, it is used intermittently over the years. Due to the size of the analog and the availability of space, during times of inactivity it is dismantled and reassembled as needed. Upon completion of the most recent motor design analysis, the analog was dismantled and inadvertently stored in an area open to shop dust. The dust fouled the sensitive variable resistors to the point where replacement is necessary. Rather than rebuild the analog computer, the digital computer was verified as an adequate replacement of the analog computer.

WEMD Engineering Memorandum No. 4603 presents the evaluation to verify the digital computer supersedes the analog computer for motor temperature design analysis. The maximum winding surface temperature, MWST, of three canned motor designs are determined by analog computer, digital computer and test. The motor sizes selected are 300, 1071, and 3020 horsepower. The results establish the digital computer method as a reliable and useful analytical tool for motor temperature analysis.

This section of the report presents the maximum winding surface temperature, MWST, evaluation of the Transfer Pump motor. The analysis used for this motor is the same analytical procedure used and verified for all motor designs at EMD, Cheswick. A discussion of the thermal analysis is shown by paragraph 7.1.2.

The MWST occurs in the winding end turn knuckle during maximum load condition with 200°F, 93°C, as the ambient fluid temperature. Based on the results of this analysis, it is concluded the MWST of the Transfer Pump Motor Design Case JAZ05, dated 3-20-95, is within the insulation Class N specification limit for 10,000 hours of operation at the maximum motor load condition and is considered acceptable. The calculated MWST is,

$$112^{\circ}\text{C} + 93^{\circ}\text{C} = 205^{\circ}\text{C} (401^{\circ}\text{F})$$

7.1.1 Design Requirements

The thermal design requirements of the electrical insulation is specified by WHC-S-0238, Revision 1, paragraph 3.5.1.3i. This paragraph requires the unit to operate for 10,000 hours. For the Class N insulation system used in the Transfer Pump, the MWST calculated at the design point should be less than 225°C (437°F) to maintain an adequate life safety margin. Under all transient conditions, the MWST must be less than 270°C (518°F) to prevent rapid breakdown of insulation properties.

7.1.2 Thermal Analysis

The MWST is determined using the computer program, WECAN_PLUS. Since the thermal path is analogous to an electric path;

$$q \approx i$$

$$R_t \approx R_e$$

$$\Delta \approx \Delta E$$

A network of resistances and appropriate current inputs are set to represent the Transfer Pump motor core and electrical parameters.

The heat flow assumes two paths, radial and axial. Heat flows radially inward to the bearing water and outward to the external fluid surface of the stator shell.

The axial heat flow is from the end turns to the middle of the stator core.

First, consider radial heat flow only. All the heat generated in the copper flows through thermal resistances to the bearing fluid or to the pump discharge fluid.

All the heat generated in one slot flows into one tooth due to the symmetrical configuration. This is the same as considering 1/2 the heat at each side of a full tooth.

The amount of heat generated in the copper conductor is calculated from:

$$q = I^2RA$$

where I = current density
 R = electrical resistance of copper
 A = copper cross-sectional area

The additional heat generated, namely core loss, is also considered by:

$$\text{Core Loss} = KW_{\text{loss}}/NL$$

where N = number of teeth
 L = core length

Since the heat flow network is known, the thermal resistances which comprise this network can be calculated from:

$$R_t = l/kA$$

where l = distance between nodes
 k = thermal conductivity
 A = flow path cross-sectional area

Using the heat generated and the calculated resistances, the temperature rise of radial heat flow is measured for both the inner and outer coil.

The results of the radial analysis are used as input to the heat transfer analysis in the axial direction.

The axial heat flow determines the MWST. The axial dissipation resistances, as determined from the radial flow considerations and the axial currents are used as inputs to the computer to obtain a solution.

Based on the good agreement of calculations to test of prior motor designs, this method of analysis is considered an accurate evaluation of the motor temperature.

Using 200°F, 93°C, as the ambient fluid temperature, the resulting MWST at the design point is,

$$112^{\circ}\text{C} + 93^{\circ}\text{C} = 205^{\circ}\text{C} (401^{\circ}\text{F})$$

7.1.2.1 Pump Thermal Characteristics

The Transfer Pump Thermal Characteristics are approximated for various running speeds. First, the thermal characteristics are calculated for the design point of 3860 rpm. The motor temperature, MWST, is determined by computer analysis using electrical case No. JAZ05, dated 3/20/95. The power losses, including friction and bearing heat losses are considered heat gains to the can annulus flow. Using the heat gain and the can annulus flow from the lower bearing cavity to the upper bearing cavity, the temperature rise of the upper bearing water is determined. The process fluid temperature is taken as 200°F. The results are shown by Table 7-1.

The running speeds of 3550, 3290, 1770 and 1420 rpm are approximated from the 3860 rpm operating condition by direct ratio from the design point. The input power for these four running speeds is less than the design point. The current density is nearly proportional to the can annulus axial flow. The can annulus water is fully turbulent for all running speeds as shown by the can annulus peripheral velocity. The stator flow velocity is nearly constant for all speeds except 1420 rpm. At this speed the power losses are also small. In summary, based on the above discussion, approximating the pump thermal characteristics by direct ratio from the design point is considered valid.

TABLE 7-1
Approximated Thermal Characteristics

Running Speed - RPM	3860	3550	3290	1770	1420	3860
Frequency - Hz	71	62	57	30	24	71
Input Power - KW	67.0	45.0	36.0	6.6	4.0	67.0
Current Density - amp/in ²	4200	2985	2685	1885	1830	4200
Can Annulus Axial Flow - GPM	31.2	29.7	27.3	13.7	10.7	33.0
Can Annulus Perphial Velocity - ft/sec	33.6	31.0	28.6	15.4	12.3	33.6
Stator OD Flowrate - gpm	171.2	158.5	146.6	77.9	62.2	33.0
Stator OD Velocity - ft/sec	6.6	6.1	5.6	3.0	2.4	1.3
Process Fluid Temp - °F	200.0	200.0	200.0	200.0	200.0	200.0
Upper Brg Water Temp - °F	203.2	202.4	202.0	200.7	200.5	---
Lower Brg Water Temp - °F	206.9	204.9	204.3	201.5	201.2	---
MWST - °C	203	~147	~136	~115	~111	270°C @ 3.3 minutes

7.1.3 Locked Rotor Thermal Analysis

A locked rotor condition exists when full voltage is applied to the pump motor and the rotor does not rotate. The locked rotor endurance time limit is that which the pump can withstand without damage to the motor rotor squirrel cage. Good design practice is to provide a minimum of 10 seconds as a locked rotor endurance time. The variable frequency drive (VFD) will not only limit locked rotor time to less than 10 seconds but will also preclude a full voltage line start as analyzed herein. Therefore, the following analysis is conservative.

The pump internal coolant temperature could be as high as 93°C (200°F). The following electrical parameters represent full voltage locked rotor;

Electrical Case JAZ01

Voltage	460
Amps	300
Stator Can Loss	1.0 KW
Rotor Can Loss	1.0 KW
Rotor Copper Loss	55.0 KW
Total Rotor Bar Resistance	.185 ohms
Total Rotor Ring Resistance	.009 ohms

The rotor bar and connection ring temperatures during the assumed 10 seconds of locked rotor condition are calculated.

First, calculate the rotor bar temperature using the semi-empirical equation;

$$\text{Heat gen} = \text{Heat}_{\text{stored in bar}} + \text{Heat}_{\text{stored in rotor core}}$$

$$Q_{\text{gen}} = M C_p \frac{\Delta T}{\Delta \Theta} + \frac{k A \Delta T}{2(\sigma \Delta \Theta)^{1.5} \pi}$$

By re-arrangement,

$$\Delta T_{\text{bar}} = \frac{Q_{\text{gen}}}{\frac{\sum M C_p}{\Delta \Theta} + \frac{k A}{\frac{2(\sigma \Delta \Theta)^{1.5}}{\pi}}}$$

Assuming conservatively that all the heat generated is stored in the bar. Neglect heat into the rotor core. Let:

$$\frac{k A}{\frac{2(\sigma \Delta \Theta)^{1.5}}{\pi}} = 0$$

Where:

$$Q_{gen} = \frac{36 \times 3412}{17_{\text{bar}} \times 18.5 \text{ in.}}$$

$$Q_{gen} = 390 \frac{\text{BTU}}{\text{hr.-in. of bar}}$$

$$\sum MC_p = \text{Bar}$$

$$\text{Bar area} = 0.153 \text{ in}^2$$

$$\text{Bar specific heat} = 0.09 \text{ BTU/lb-}^\circ \text{F}$$

$$\text{Bar length} = 1.0 \text{ inch}$$

$$\text{Bar density} = 558 \text{ lb/ft}^3$$

$$\text{Bar} = \frac{558 \times .153 \times .09 \times 1}{1728}$$

$$\text{Bar} = .0044 \frac{\text{BTU}}{\text{degrees F}}$$

In order to simplify the analysis, assume all the heat is stored in the bar and the heat stored in the tooth is neglected. this assumption is considered conservative.

and,

$$\Delta T_{\text{bar}} = \frac{390}{\frac{.0044 \times 3600}{\Delta \Theta}}$$

$$\Delta T_{\text{bar}} = 24.5 \text{ degrees/gen} \times \Delta \Theta$$

at $\Delta\Theta = 10 \text{ seconds}$

$$\Delta T_{br} = 245 \text{ degrees F}$$

and finally; at $T_{ambient} = 200 \text{ degrees F}$

$$T_{br} = \Delta T_{br} + T_{ambient}$$

$$T_{br} = 245 \text{ degrees F} + 200 \text{ degrees F} = 445 \text{ degrees F}$$

Secondly, calculate the rotor connection ring temperature from:

$$Q_{gen} = Q_{stored}$$

$$\text{where: } Q_{gen} = 1.3KW \times 3412 = 4435 \frac{BTU}{hr.}$$

$$Q_{stored} = CppV \frac{dT}{d\Theta}$$

Connection ring specific heat = 0.091 BTU/lb-°F

Volume of connection ring = 5.603 in₃

Density of connection ring = 0.323 lb/in³

$$Q_{stored} = \frac{.091 \times 5.603 \times .323 \times 3600}{d\Theta} dT$$

$$Q_{stored} = 592 \frac{dT}{d\Theta}$$

by substitution;

$$dT = \frac{4435 d\theta}{582}$$

$$dT = 75 d\theta$$

or $dT_{avg} = 75 ^\circ F$ at $d\theta = 10 \text{ seconds}$

and finally, at $T_{ambient} = 200 ^\circ F$

$$T_{avg} = T_{ambient} + dT$$

$$T_{avg} = 75 \text{ degrees } F + 200 \text{ degrees } F = 275 \text{ degrees } F$$

In conclusion, the approximated locked rotor copper temperatures at the endurance time limit of 10 seconds is 445°F bar temperature and 275°F connection ring temperature. Good design practice for WEMD canned motor pumps is to limit squirrel cage temperatures to no more than 500°F. Therefore, the pump is considered acceptable for a 10 second locked rotor.

7.1.4 Loss of Flow Condition

During normal pump operation, the pumped fluid removes the heat generated by the electrical losses from the motor assembly. In the event of loss of flow condition, heat transfer from the assembly to the pumped fluid is zero. The pump internal flow is zero, but, fluid remains in the rotor annulus. All heat generated is stored in the assembly and is expressed as,

$$Q_{gen} = Q_{stored}$$

where Q_{gen} is the total electrical loss. At 67 KW input power, 44 KW is shaft power and the remaining 23 KW represents the total electrical power loss.

$$Q_{gen} = 23 \times 3412 \frac{KW}{1} \times \frac{BTU}{KW \text{ hr}}$$

$$Q_{gen} = 78,500 \text{ BTU/hr}$$

And Q_{stored} is expressed as,

$$Q_{\text{stored}} = \Sigma c p \rho V \frac{dT}{d\theta}$$

where

$$E c p \rho V = (c p \rho V)_{\text{iron}} + (c p \rho V)_{\text{copper}}$$

And

$$(\rho V)_{\text{iron}} = (183 \text{ lbs})_{\text{stator}} + (69 \text{ lbs})_{\text{rotor}}$$

$$(\rho V)_{\text{iron}} = 252 \text{ lbs}$$

$$(c p)_{\text{iron}} = .11 \text{ BTU/lb-}^{\circ}\text{F}$$

And

$$(\rho V)_{\text{copper}} = (69 \text{ lbs})_{\text{stator}} + (18 \text{ lbs})_{\text{rotor}}$$

$$(\rho V)_{\text{copper}} = 87 \text{ lbs}$$

$$(c p)_{\text{copper}} = .09 \text{ BTU/lb-}^{\circ}\text{F}$$

Solving for $E c p \rho V$;

$$E c p \rho V = (.11 \times 252 + .09 \times 87) \frac{\text{BTU}}{\text{lb-}^{\circ}\text{F}} \times \frac{\text{lb}}{1}$$

$$E c p \rho V = (27.7 + 7.9) \text{ BTU/}^{\circ}\text{F}$$

$$E c p \rho V = 35.6 \text{ BTU/}^{\circ}\text{F}$$

By substitution and rearrangement from;

$$Q_{\text{gen}} = Q_{\text{stored}}$$

$$Q_{\text{gen}} = \Sigma c p \rho V \frac{dT}{d\theta}$$

Solve for $d\theta$;

$$\Delta\theta = \Sigma c p \rho V \frac{\Delta T}{Q_{\text{gen}}}$$

$$\Delta\theta = 35.6 \times \frac{\Delta T}{78500} \times \frac{60}{1} \frac{\text{BTU}}{^{\circ}\text{F}} \times \frac{\Delta T}{1} \times \frac{\text{Hr}}{\text{BTU}} \times \frac{\text{min}}{\text{Hr}}$$

$$\Delta\theta = .027 \times \Delta T \frac{\text{minutes}}{^{\circ}\text{F}}$$

Assume the motor temperature limit is 225°C (437°F) and motor assembly temperature is 200°C (392°F) at the time loss of flow condition occurs; this assumption is considered conservative. Solve for the ΔT limit,

$$\Delta T = (225^{\circ}\text{C} - 200^{\circ}\text{C}) = 25^{\circ}\text{C} \Rightarrow 45^{\circ}\text{F}$$

Finally, solve for the transient time at which the motor temperature approaches 225°C (437°F) during the loss of flow condition;

$$\Delta\theta = .027 \times 45 \frac{\text{minutes}}{^{\circ}\text{F}} \times \frac{^{\circ}\text{F}}{1}$$

$$\Delta\theta = 1.2 \text{ minutes}$$

For an assumed 270°C (518°F) temperature limit, i.e.,

$$\Delta T = (270^{\circ}\text{C} - 200^{\circ}\text{C}) = 70^{\circ}\text{C} \Rightarrow 126^{\circ}\text{F}$$

The transient time is determined by direct ratio,

$$\Delta\theta = 1.2 \left(\frac{126^{\circ}\text{F}}{45^{\circ}\text{F}} \right)$$

$$\Delta\theta = 3.3 \text{ minutes}$$

In summary, the pump assembly running time during a loss of flow condition without exceeding 225°C (437°F) is approximated as 1.2 minutes. This approximation is considered conservative because the total temperature rise used in the calculation is only 25°C (45°F). Whereas, the motor core temperature is expected to be less than the MWST, i.e., 200°C (392°F) as compared to 150°C (302°F) core temperature. Also, thermal conduction heat transfer into pump parts without electrical losses is neglected, namely, the stator shell.

7.2 Machine Thermal Growth Analysis

The Transfer Pump components are manufactured to dimensions based on 68 °F. The pump is operated at temperatures up to 200 °F. If all components were manufactured from the same material then the relative fits between pump components would remain the same over the entire temperature range the pump experiences. However, as identified in Section 4.0 the pump is composed of many different materials. These materials have different coefficients of thermal expansion. The analyses in this section document the expected changes in fits between various components between the 68 °F manufacturing/assembly temperature and the 200 °F maximum ambient temperature. These analyses assume that the waste tank process fluid and atmosphere are both at a uniform 200 °F so it does not matter what the actual liquid level is in the tank.

7.2.1 Column Insertion Length

The lead unit insertion length is 49 feet (588 inches). All but approximately 24 inches of that length is made of 304L material for the structure. The remaining structural material is CF3 which has a coefficient of expansion very close to that of 304L. Therefore, the entire structural length is assumed to be 304L.

The thermal growth of the column is as follows:

$$dL = L \alpha \Delta T$$

where: dL is the thermal growth, inch

L is the column length, 588 inches

α is 304L expansion coefficient, $9.6 \times 10^{-6} / ^\circ\text{F}$

ΔT is the temperature change, °F

$$dL = 588 \ 9.6 \times 10^{-6} (200 - 68) = 0.745 \text{ inch}$$

E.M. 6673

Therefore, the transfer pump will grow
almost $\frac{3}{4}$ -inch when inserted in a tank at
200 °F.

7.2.2 Required Rotating Assembly Axial Clearance

The rotating assembly structure is primarily 410 SST and the surrounding static structure is primarily CF3 SST. The rotating assembly is supported axially at the underside of the thrust runner. Axial fits occur between the rotating and static components at the upper shroud, lower shroud, and inlet seal bottom face of the second stage impeller. Axial fits also exist at the hub top face, upper shroud, lower shroud and inlet seal bottom face of the first stage impeller. Because the static hydraulics will expand more than the rotating hydraulics (the coefficient of expansion is higher for CF3 SST than 410 SST) the axial clearance will decrease on heat-up only at the second stage impeller upper shroud and the first stage impeller hub top face and upper shroud. Axial clearances will increase on heat-up at the first and second stage impellers lower shrouds and inlet seal bottom faces so these locations need not be evaluated. All the impeller to diffuser axial fits are about the same magnitude, 0.20 inch. The axial fit of interest farthest from the thrust runner is the first stage impeller upper shroud, which is 7.38 inches below the bottom face of the thrust runner. This fit will experience the greatest loss in axial clearance. The change in axial clearance between the first stage impeller and the first stage diffuser at this location is:

$$dL = L \Delta \alpha \Delta T$$

where: dL is the clearance decrease, inch
 L is the length, 7.38 inches

$\Delta\alpha$ is the difference in expansion coefficient /°F

between 304L and 410 SST

ΔT is the temperature change, °F

$$dL = 7.38 (9.6 \times 10^{-6} - 6.1 \times 10^{-6}) (200 - 68)$$

$$dL = 0.0034 \text{ inch}$$

Since the axial clearance is approximately 0.200 inch at this location, this clearance decrease is minimal and can be ignored.

7.2.3 Radial Bearing Clearances

The bearings are assembled at 68 degrees F. The maximum operating temperature is approximately 200 degrees F. The change in diameter is given by:

$$D' = D(1 + \alpha \Delta T)$$

where D' is the increased dimension, inch

D is the original dimension of interest, inch

α is the coefficient of thermal expansion:

2.2×10^{-6} /degrees F for Silicon Carbide

6.1×10^{-6} /degrees F for 410 SST

9.6×10^{-6} /degrees F for 304 SST

For the upper bearing journal:

$$D'_{uj} = 2.5[1 + 2.2 \times 10^{-6} (200 - 68)]$$

$$D'_{uj} = 2.500726$$

For the upper radial bearing, at the pins

$$D'_{up} = 3.500[1 + 9.6 \times 10^{-6} (200 - 68)]$$

$$D'_{up} = 3.504435$$

For the upper and lower pad holder, the thickness increase is:

$$T'_{ph} = 0.25[1 + 6.1 \times 10^{-6}(200-68)]$$

$$T'_{ph} = 0.25021$$

For the upper and lower pads, the thickness increase is:

$$T'_p = 0.25[1 + 2.2 \times 10^{-6}(200-68)]$$

$$T'_p = 0.250073$$

The increased diameter of the upper radial bearing is:

$$D'_{ub} = D'_{up} - 2(T'_{ph}) - 2(T'_p)$$

$$D'_{ub} = 3.504435 - 2(0.25021) - 2(0.250073)$$

$$D'_{ub} = 2.503870$$

The increase in upper bearing diametral clearance is a maximum of:

$$d_{Cu} = D'_{ub} - D'_{uj}$$

$$d_{Cu} = 2.50387 - 2.500726$$

$$d_{Cu} = 0.00314 \text{ inch}$$

For the lower radial bearing:

$$D'_{lj} = 3.25 [1 + 2.2 \times 10^{-6}(200-68)]$$

$$D'_{lj} = 3.250944$$

For the lower radial bearing at the pins:

$$D'_{lp} = 4.25[1 + 9.6 \times 10^{-6}(200-68)]$$

$$D'_{lp} = 4.255386$$

The increased diameter of the lower radial bearing is:

$$D'lb = 4.255386 - 2(0.25021) - 2(0.250073)$$

$$D'lb = 3.254820$$

The increase in lower radial bearing diametral clearance is a maximum of:

$$dCl = D'lb - D'lj$$

$$dCl = 3.254820 - 3.250944$$

$$dCl = 0.003876 \text{ inch}$$

7.2.4 Electrical Cable Axial Growth

The axial thermal growth of the electrical cable from the motor to the pump pit is in the order of 1.0 inch. This is very small as compared to the overall length of the column. The cable electrical pipe has sufficient room to permit the electrical cable to sag inside. The thermal growth is based on an average cable temperature of 230°F over a 50 foot length.

The radial expansion of the cable is insignificant.

The thermal growth calculation is as follows:

$$\alpha_{cu} = \frac{9.8 \times 10^{-4}}{^{\circ}F}$$

$$\Delta L = L \alpha_{cu} (\Delta T)$$

$$\Delta L = (50)(12)(9.8 \times 10^{-4})(230 - 68)$$

$$\Delta L = 0.95 \text{ inches}$$

8.0 STRUCTURAL EVALUATION

8.1 Assumptions

Various assumptions are used for the transfer pump and column structural evaluation. Since the bulk of the column length is presently 8 inch Schedule 40S pipe and other components are short by comparison, it is assumed that deflections/stresses are governed by a column of 588 inches of said pipe. The internal pipes (3 inch, 2 inch, 1 1/2 inch and 1 inch) contribute little to lateral stiffness and hence are neglected. The welds have been designed to be as strong or stronger than the pipe and thus pose no local weakness. Upper joint flexibility or stiffness of any flanges/adapters/bolts/customer support is neglected.

Other assumptions will be given as used. Pipe data used in the analyses for the column assembly is given in Table 8.1. The vertical pump cases assume fixed at the top, free at the bottom, even though fluid restriction may be significant, especially for time dependent load applications. The pump-column in the horizontal position is assumed simple-simple supports at the ends, although it is recommended to support the column assembly at some internals when kept horizontal. The analysis cases were run on a personal computer with the DE/CAASE program (Desktop Engineering/Computer Aided Analytical Solutions for Engineers), which compared favorably with classical beam equations. The column plus pump assembly is taken as 588 inches long with the stiffness of the purge/dilution/electrical pipes and any foam filler neglected.

Only pump head pressure/hydro-pressure loaded components are evaluated for pressure loading. Pit head pressure is considered trivial. Due to the unique situation of the pump being submerged in the pumping fluid, joint integrity against leakage is not analytically investigated and gaskets/o-rings are not used at bolted closures except for the column-pump joint.

TABLE 8-1
Column Pipe Data

	<u>Purge Line</u>	<u>Dilution Line</u>	<u>Electrical Line</u>	<u>Waste Line</u>	<u>Column</u>
	<u>1" Sch 40S</u>	<u>1.5" Sch 40S</u>	<u>2" Sch. 40S</u>	<u>3" Sch. 40S</u>	<u>8" Sch. 40S</u>
OD (in.)	1.315	1.900	2.375	3.50	8.625
ID (in.)	1.049	1.610	2.067	3.068	7.981
t (in.)	0.133	0.145	0.154	0.216	0.322
A _{pipe} (in. ²)	0.494	0.799	1.075	2.228	8.40
A _{fluid} (in. ²)	0.864	2.036	--	7.39	--
I (in. ⁴)	0.0874	0.31	--	3.02	72.5
Z (in. ³)	0.1329	0.326	--	1.724	16.81
Wt. (lb/ft)	1.679	2.718	3.653	7.58	28.55
Wt. _{water} (lb/ft)	0.374	0.882	--	4.80	--
Wt. _{waste} (lb/ft)	--	--	--	7.20 *	--
Wt. _(pipe + waste) (lb/ft)	--	--	--	14.78	--

*Using a waste specific gravity of 1.5.

8.2 Handling Analysis

8.2.1 Horizontal Case

The horizontal handling case is the assembled pump/column assembly of 588 inches simply supported at the two ends with a distributed pipe load of

$$\omega = \frac{1}{12} (28.55 + 7.58 + 1.679 + 2.718 + 3.653) = 3.7 \text{ lb/in.}$$
 The pump weight would

be supported at one end and not contribute much to the center deflection. The results (see Figure 8-1) are as follows accounting for the 8 inch pipe stiffness only.

Center Deflection = 2.80 in.

Maximum Moment = 1.60 (10⁵) in. lb.

Center Bending Stress = 9518 psi

This is less than the yield strength of 25 ksi.

Any center supports will keep this stress lower.

Buckling is no problem on lifting (see Figure 8-2).

8.2.2 Vertical Case

The pump and column weight is in the range of 3640 to 3790 lbs.

For the weight to be carried by outer 8" pipe only:

8" Schedule 40S

Area (in. ²)	8.40
Tensile Stress (psi)	451

Thus column tensile stresses due to weight are minor for vertical lifting. The weight 3790 lbs. was used above, but it is conservative to use the lower value of 3640 in all subsequent evaluations.

Module B-7: Lateral Buckling of Beams

Figure 8-2 Buckling Limit for Lifting

8.3 Lateral Impingement Analysis

8.3.1 Column Stress for 150 lb. Impingement Load

Paragraph 3.5.10.1 of Specification WHC-S-0238 states that the pump withstands 10^4 cycles of 150 lbs. lateral impingement load at the suction inlet. Local effects of fluid impingement are not considered significant, but the pump/column assembly can be pushed laterally by the 150 lb. load. For the case of:

Length = 588 in.

Lateral Load = 150 lb. at bottom end

Axial Load = 3640 lbs.

Young's Modulus = $28.3 (10^6)$ psi for 304L stainless steel

Moment of Inertia = 72.5 in^4 (min.)

The lateral deflection is 3.98 inches (see Figure 8-3). The resultant bending stress at the top support is ± 4385 psi. Conservatively using the weld fatigue reduction factor of 4 for a fillet weld, the alternating stress is:

$$S_a = \frac{1}{2} (4)(4385) = 8770 \text{ psi}$$

Per the fatigue curve for stainless steel, an infinite number of such stress cycles are allowed.

8.3.2 Lateral Jet Impingement Load Limit for 10^4 Cycles

The weld of the pump mounting flange to the 8-inch Schedule 40S pipe column is the limiting stress item. Now consider the weld is a full penetration weld with a fatigue reduction factor of 2.0. At 10^4 cycles S_a equals 63 ksi for 300 series stainless steels per the ASME Code, Figure 5-110.2.1 of Section VIII-2. Since an impingement force acts on only one side of the pump, the stress range is $2S_a$, or 126 ksi. The allowed bending stress is:

$$\sigma = (2 S_a / K_f) = 63 \text{ ksi}$$

Module B-1: Single Span Beams Under Transverse and Axial Loads

Figure 8-3 Lateral Impingement = 150

For the 8-inch Schedule 40S pipe, Z is 16.81 inch³. The allowed moment is:

$$M_a = 16.81 (63,000) = 1,059,030 \text{ inch-lbs}$$

DE/CAASE and the following data was used to calculate the allowable lateral load (see Figure 8-4):

$$\text{Length} = 588 \text{ inches}$$

$$\text{Axial load} = 3,640 \text{ lbs}$$

$$E = 28.3 \times 10^6 \text{ psi}$$

The allowed lateral jet impingement load for 10⁴ cycles is 2,155 lbs. The maximum pump deflection for this load is 57.2 inches.

The bending stress required to limit the jet impingement cycles to 10⁴, 63 ksi, exceeds the ambient yield strength for 304L stainless steel of 25 ksi. Thus, the above calculated jet impingement load of 2155 lbs. could produce a column assembly permanently bent off straight vertical, which may be undesirable for removing the pump from the pit. Using the 25 ksi yield stress as the limiting stress, the allowed moment caused by the jet impingement force is:

$$M_a = 16.81 (25,000) = 420,250 \text{ inch-lbs}$$

The lateral jet impingement force which causes yielding in the outer column to pump mounting flange weld is 856 lbs. This load produces a maximum deflection of 22.7 inches (see Figure 8-5). This value is less than the agreed upon deflection limit of 24 inches.

8.4 Seismic Analysis

Per Specification WHC-S-0238 (paragraph 3.5.10.2):

$$\text{Horizontal Seismic Force} = F_p = ZIC_pW_p$$

$$\text{where: } Z = 0.12$$

$$I = 1.25$$

Module B-1: Single Span Beams Under Transverse and Axial Loads

Figure 8-4 Limiting Impingement for $\sigma = 63$ Ksi

Module B-1: Single Span Beams Under Transverse and Axial Loads

Figure 8-5 Impingement Limit for Yield Stress

$$C_p = 1.5 \text{ (per dynamics analysis, Section 10.0)}$$

$$W_p = 3790 \text{ lbs. (or less)}$$

Thus:

$$F_p = 0.12(1.25)(1.5)(3790)$$

$$F_p = 853 \text{ lbs.}$$

$$\text{Use } F_p = 860 \text{ lbs.}$$

Now F_p acts at the pump/column c.g., which is above the center of pump or -28" from the aft end; thus it is conservative to use:

$$L = 588 - 28 = 560 \text{ in. (location of load)}$$

Results from DECAASE (see Figure 8-6);

$$\delta_{max} = 21.0 \text{ in.}$$

$$\sigma_{max} = \frac{3790}{8.4}$$

$$\sigma_t = 451 \text{ psi}$$

$$\sigma_{bend} = \frac{M_b}{Z} = \frac{401,862}{16.81}$$

$$\sigma_b = \pm 23,906 \text{ psi}$$

Thus:

$$\sigma_{max} = 451 + 23,906$$

$$\sigma_{max} = 24,357 \text{ psi}$$

The 8-inch pipe is not a pressure boundary component and hence is limited to a yield strength of 25 ksi.

8.5 Code Section VIII-1 Analysis

8.5.1 Column Discharge Pipe (3" Schedule 40S)

Use for Design Conditions: $P_d = 400 \text{ psi}$

$$T_d = 250^\circ\text{F}$$

where Operating $P_{max} = 369 \text{ psi}$, $T_{max} = 200^\circ\text{F}$.

Module B-1: Single Span Beams Under Transverse and Axial Loads

Figure 8-6 Seismic Analysis

Per UG-99 - Hydro-Test:

$$P_{hydro} = 1.5P_d \left(\frac{S_{max}}{S_{design}} \right)$$

For 304L SS; Allowable (S) in ksi per the ASME Code Section VIII:

<u>304L S.S.</u>	<u>-20 to 100°F</u>	<u>200°F</u>	<u>300°F</u>
SA-312 (Welded Pipe)	14.2	14.0	13.0
SA-312 (Smls. Pipe)	16.7	16.5	15.3
SA-351 (CF3 Casting)	17.5	16.6	15.3

Thus, for the 3 inch welded discharge pipe:

$$P_{hydro} = 1.5(400) \left(\frac{14.2}{13.5} \right)$$

$$P_{hydro} = 631 \text{ psi}$$

And for pump castings:

$$P_{hydro} = 1.5(400) \left(\frac{17.5}{15.95} \right)$$

$$P_{hydro} = 660 \text{ psi}$$

Section VIII-1, Paragraph UG-27, gives the thickness required for a given pressure.

$$t = \frac{PR}{SE - 0.6P}$$

where, for the discharge pipe of 3" Schedule 40S:

	<u>Design</u>	<u>Hydro</u>
P (psi)	400	660
T (°F)	250	70
S (psi)	13,500	14,200
R (in.)	1.534	1.534 (3" Sch. 40)
E (Table UW-12)	0.45	0.45 (min.)
	E.M. 6673	
	8-13	

The weld efficiency factor E is conservatively taken as the minimum shown of 0.45 although higher values apply for the column design.

For Design Case:

$$t_{min} = \frac{400(1.534)}{13500(.45) - (.8)(400)}$$

$$t_{min} = 0.105 \text{ in.}$$

Actual thickness is 0.216 in. which leaves 0.111 in for erosion loss.

The normal operating stress for 369 psi (or less) is:

$$\begin{aligned}\sigma &= \frac{PR}{t} + 0.6P \\ &= \frac{369(1.534)}{.216} + .6(369) \\ \sigma &= 2840 \text{ psi (or less)}\end{aligned}$$

For Hydro Case:

$$t_{min} = \frac{660(1.534)}{14200(.45) - (.8)(660)}$$

$$t_{min} = 0.169 \text{ in.}$$

Actual thickness = 0.216 in. exceeds these minimums.

The nominal pipe stress at 660 psi is:

$$\begin{aligned}\sigma &= \frac{PR}{t} + 0.6P \\ &= \frac{660(1.534)}{.216} + .6(660) \\ \sigma &= 5083 \text{ psi}\end{aligned}$$

8.5.2 Pump External Wall

The fluid flows about the pump and is contained at the pump OD by a tube with:

$$ID = 10.375 \text{ in.}$$

$$OD = 11.0 \text{ in.}$$

$$t = 0.3125 \text{ in.}$$

The minimum required thickness is:

$$t_m = \frac{PR}{SE - 0.6P}$$

where: R = 5.188 in.

E = 1.0 (no welds)

Design Case:

$$t_m = \frac{400(5.188)}{13500 - .6(400)}$$

$$t_m = 0.156 \text{ in.}$$

The actual thickness of this outer shell is 0.3125 in. This leaves 0.1565 in. for erosion. Now the ends of this shell are slip joints and hence no axial loads are anticipated. The half thickness at the end joints can carry the pressure since the allowable is about half of yield, but at worst the inner lip would deform until the outer lip thickness is effective and full thickness would again exist. The lip

bending stress for operating pressure (330) is only $\sigma = \frac{6M}{t^2} = \frac{6(10.26)}{(1.15)^2} = 2736 \text{ psi}$

where, M is obtained from Figure 8-7 from DECAASE.

Hydro Case:

$$t_m = \frac{660(5.188)}{14200 - .6(660)}$$

$$t_m = 0.248 \text{ in.}$$

So, the actual thickness of 0.3125 in. is adequate.

This analysis conservatively bounds the motor stator shell also.

8.5.3 Stator ID Housing Wall

The stator windings are sealed against the fluid at the ID by a stator can. The structural support for the can is furnished by the punchings at the center length and by tube sections at the ends. The tube sections are called backup sleeves with the following dimensions.

OD = 4.45 in.

ID = 4.15 in.

t = 0.15 in.

Module D-3: Cylindrical Shells-Thin Walled

Figure 8-7 Outer Shell Lip Analysis

The minimum thickness for the backup sleeves is as follows:

$$t_m = \frac{PR}{SE - 0.6P}$$

where: R = 2.075
E = 1.0 (no weld)
S = 13.5 ksi (Design), 14.2 ksi (Hydro)

Design Case:

$$t_m = \frac{400(2.075)}{13500 - .6(400)}$$

$$t_m = 0.063 \text{ in.}$$

Hydro Case:

$$t_m = \frac{660(2.075)}{14200 - .6(660)}$$

$$t_m = 0.099 \text{ in.}$$

So, the actual thickness of 0.15 in. exceeds these values even excluding the additional support provided by the 0.033 inch thick can.

8.5.4 Purge/Dilution Lines

One purge line of one inch schedule 40S pipe and a dilution line of 1.5 inch schedule 40S pipe carry filtered river water into the pump assembly. Using the same pressure values as for the discharge pipe:

	<u>Design</u>	<u>Hydro</u>
P (psi)	400	660
T (°F)	250	70
S (psi)	13,500	14,200
E (Table UW-12)	0.45	0.45 (min.)

For Design Case:

Purge Line: R = 0.5245 in.

Dilution line: R = 0.805 in.

$$t_{min} = \frac{400(.5245)}{13500(.45) - .6(400)}$$

$$t_{min} = \frac{400(.805)}{13500(.45) - .6(400)}$$

$$t_{min} = 0.038 \text{ in.}$$

$$t_{min} = 0.055 \text{ in.}$$

The purge pipe thickness is 0.133 in. The dilution pipe thickness is 0.145 in.

For Hydro Case:

Purge Line

Dilution Line

$$t_{min} = \frac{660(.5245)}{14200(.45) - .6(660)}$$

$$t_{min} = \frac{660(.805)}{14200(.45) - .6(660)}$$

$$t_{min} = 0.058 \text{ in.}$$

$$t_{min} = 0.089 \text{ in.}$$

The purge pipe thickness is 0.133 in. The dilution pipe thickness is 0.145 in.

8.6 Bolting Analysis

8.6.1 Pump Mounting Flange Bolt Loads

The upper square pump mounting flange supports the pump on the customer's flange. This is a 16.5 x 16.5 in. square flange with four corner bolts. Normally, these bolts see no significant loading except preload, and the customer will select the bolt and the preload. The worst loading case will be for lateral seismic where the plate will want to tip up on one side and the two opposite bolts are loaded (length from bolts to opposite edge is 15 in.).

Using a seismic load of 860 lbs. and neglecting fluid restraint during the seismic event, the seismic moment at the plate is the 401,862 in. lb. from section 8.4.

Thus, neglecting the near side bolts which contribute little, the bolt load P is:

$$2P(15) = 401,862$$

$$P = 13,395 \text{ lbs.}$$

The top flange bolt may see a maximum tensile load of 13.4 Kips per bolt during the seismic event.

8.6.2 Transition Piece Bolts

The transition piece bolts to the pump proper with a gasket used but there is metal to metal contact inside and outside of the gasket and bolts. Thus, Appendix 2 of ASME VIII-1 does not apply and flange bending does not exist by design. The bolts merely have to support the pressure and gasket compression force, which is considered small. For conservatism; the following pressure load is calculated in keeping with Section VIII bolt design load calculations.

$$P = 400 \text{ psi (Design Pressure)}$$

$$D \leq 10 \text{ in. (Pressure is localized only)}$$

$$\begin{aligned} \text{Max. Pressure Load} &= \frac{\pi}{4} D^2 P \\ &= \frac{\pi}{4} (10)^2 (400) \\ &= 31,400 \text{ lbs.} \end{aligned}$$

Using SA-193, B8M, Class 2 for 3/4 inch diameter or less, the Table UHA-23 allowable is 25 ksi. Thus, the required bolts area is:

$$A = \frac{31.4}{25} = 1.26 \text{ in.}^2$$

For the following sizes, the required bolt number follows:

Bolt size	=	0.375	0.50	0.625	0.75 in.
Bolt area	=	0.0773	0.1416	0.2256	0.334 in. ²
No. Bolts	=	17	9	6	4

Nine 0.500 bolts are used. Now if a yield strength of 95 ksi is used as the limiting criteria, then the required area is:

$$A = \frac{31.4}{95} = 0.33 \text{ in.}^2$$

For 0.500-13 in. bolt with a stress area of 0.1416 in.², then

$$\text{no.} = \frac{0.33}{0.1416} = 3 \text{ bolts needed}$$

Thus, the nine bolts are adequate for pressure and the stress is $\frac{31.4}{9(.1416)} = 24.6$

ksi. Increasing the bolt number or size has space limitations, the disassembly time will increase, radiation exposure will increase, and additional complications will occur. Thus, Code sizing is considered secondary if less than nine bolts are used.

Now to check for stress due to imposed joint moment which is the seismic load (860 lbs.) which is assumed here to act at the pump middle or $(54.6/2) = 27.3$ inches from the joint. Note this is conservative since the 860 includes the column weight.

$$M = 27.3(860) = 23,478 \text{ in. lb.}$$

The maximum load on a bolt is given by:

$$F = \frac{2M}{nR}$$

Reference: "Eccentrically Loaded Joints", by R. J. Berger,

Machine Design, pp. 185-188, 8-17-67.

where:

$$n = 9 \text{ bolts}$$

$$R = \text{bolt circle radius} = 4.75 \text{ in.}$$

$$F = \frac{2(23478)}{9(4.75)}$$

$$F = 1098 \text{ lbs.}$$

Thus:

$$\sigma_s = \frac{1098}{.1416}$$

$$\sigma_s = 7757 \text{ psi}$$

Therefore, for pressure plus seismic on nine 1/2-13 bolts:

$$\sigma_{\text{max}} = 24600 + 7757$$

$$\sigma_{\text{max}} = 32,357 \text{ psi}$$

This is 34% of the yield strength of 95 ksi.

8.6.3 Diffuser Bolts

The diffuser bolts are 3/8 in. - 16 and see negligible loading except discharge pressure. Using a pressure diameter of 11 inches, the 31.4 Kip design pressure load of 8.6.2 is ratioed up to 38 Kips. With a yield strength of 95 ksi, the no. of bolts required is $n = \frac{38}{95(.0773)} = 5$. Since eight bolts are used, the joint is adequate for pressure loading.

8.6.4 Shaft-Impeller Bolt

The impellers are retained axially on the shaft by a 0.5 inch bolt and are keyed to the shaft with circular cross section keys or pins of 0.3127 in. diameter. The 410SS shaft from the lower journal/thrust runner shoulder to the impeller bolt head is 9.19 inches. The shaft OD is filled with the following.

<u>Component</u>	<u>Material</u>	<u>Length (in.)</u>
Thrust Runner	Tungsten Carbide	1.00
Sec. Stage Impeller	CF3 SS	2.67
Spacer Sleeve	410SS	3.25
First Stage Impeller	CF3 SS	<u>1.83</u>
		9.19 in.

A PC program on TK Solver was prepared to study the retention bolt stress in detail and is given on the following pages. The bolt has a free length of 1.4 inches before thread engagement and is locked in place with a lock pin. The combined stress in the bolt is 12,991 psi including differential thermal expansion for heatup from 68° to 200°F and 7,970 psi in storage from 68°F assembly to 40°F. This leaves margin for any potential temperature differential upon insertion.

<u>St</u>	<u>Input</u>	<u>Name</u>	<u>Output</u>	<u>Unit</u>	<u>Comment</u>
					IMPBOLT.TK
					Transfer Pump Impeller Bolt Analysis
					Dated 1-4-95
					by R.F. Guardiani
					Initial Preload Analysis
L	2000	Fi		lbs	Initial bolt preload
		Sigmat	8848	psi	Bolt axial stress due to preload
		TDEFi	0.00133	in	Total joint deformation due to preload
		DELabi	0.00069	in	Initial bolt plus shaft elongation
		DELbi	0.00033	in	Initial bolt elongation
		DELai	0.00037	in	Initial shaft elongation
		DELrci	-0.00064	in	Initial clamped parts total elongation
		DELtri	-0.00009	in	Initial thrust runner elongation
		DELimp2i	-0.00007	in	Initial impeller #2 elongation
		DELspi	-0.00043	in	Initial spacer elongation
		DELimp1i	-0.00004	in	Initial impeller #1 elongation
					Thermal Analysis
		Fi	936	lbs	Bolt load due to thermal affects
		Sigmat	4142	psi	Bolt axial stress due to thermal loads
L	200	T		F	Temperature of interest
	88	Ta		F	Assembly temperature
		TDEFt	0.00061	in	Total joint thermal deformation
		DELabt	0.00032	in	Thermal stress bolt & shaft elongation
		DELbt	0.00015	in	Thermal stress bolt elongation
		DELst	0.00017	in	Thermal stress shaft elongation
		DELrct	-0.00029	in	Thermal stress clamped parts elongation
		DELrtt	-0.00004	in	Thermal stress thrust runner elongation
		DELimp2t	-0.00003	in	Thermal stress impeller #2 elongation
		DELspt	-0.00020	in	Thermal stress spacer elongation
		DELimp1t	-0.00002	in	Thermal stress impeller #1 elongation
		DELt	0.00802	in	Total thermal growth

Combined Preload & Thermal Results

L	F	2836	lbs	Net bolt load, preload plus thermal
L	Sigma	12891	psi	Net bolt axial stress from all loads
	JCOMP	0.00194	in	Total joint compression

Dimensional Data for Bolt Analysis

L	1.40	Lb	in	Bolt free length (head to shaft thds)
	0.625	Dnom	in	Nominal bolt diameter
	11	n	thd/in	Bolt threads per inch
		Dbtd	0.836 in	Bolt diameter at root of threads
		Abmin	0.226 in ²	Bolt cross-section area at minor dia
	0.625	Db	in	Bolt shank diameter
		Ab	0.307 in ²	Bolt area at shank
		La	7.35 in	Shaft length
	3.55	La1	in	First shaft section length
	1.601	Da1	in	First shaft section diameter
	2.20	La2	in	Second shaft section length
	1.250	Da2	in	Second shaft section diameter
		La3	1.60 in	Third shaft section lg to end of thds
	1.250	Da3o	in	Third shaft section outside diameter
	0.625	Da3i	in	Third shaft section inside diameter
	1.00	Ltr	in	Thrust runner thickness
	2.67	Limp2	in	Impeller #2 hub length
	3.25	Lsp	in	Spacer length
	1.83	Limp1	in	Impeller #1 hub length
	2.380	Dtre	in	Thrust runner effective outside diameter
	1.603	Dtri	in	Thrust runner inside diameter
	2.500	Dimp2o	in	Impeller #2 hub outside diameter
	1.602	Dimp2i	in	Impeller #2 hub inside diameter
	1.500	Dspo	in	Spacer outside diameter
	1.258	Dspi	in	Spacer inside diameter
	2.500	Dimp1o	in	Impeller #1 hub outside diameter
	1.251	Dimp1i	in	Impeller #1 hub inside diameter
		As1	2.013 in ²	First shaft section area
		As2	1.227 in ²	Second shaft section area
		As3	0.920 in ²	Third shaft section area
		Atr	2.431 in ²	Thrust runner effective area
		Aimp2	2.893 in ²	Impeller #2 hub area
		Asp	0.525 in ²	Spacer area
		Aimp1	3.680 in ²	Impeller #1 hub area

Bolt Analysis Material Properties			
T304	Bolt		Bolt material: T304 or T316
	Es	2.80E7 psi	Bolt modulus
	Alphas	9.6E-6 1/F	Bolt thermal expansion coefficient
T410	Shaft		Shaft material: T304 or T410
	Es	2.80E7 psi	Shaft modulus
	Alphas	9.1E-6 1/F	Shaft thermal expansion coefficient
WC	Runner		Runner material: T304, T410 or WC
	Etr	8.98E6 psi	Thrust runner modulus
	Alphasr	2.6E-6 1/F	Thrust runner thermal exp coefficient
CF3	Impeller		Impeller material: CF3 or CD4MCu
	Eimp	2.80E7 psi	Impeller modulus
	Alphasimp	9.0E-6 1/F	Impeller thermal expansion coefficient
T410	Spacer		Spacer mtl: T304/T410/CD4MCu/WC
	Esp	2.80E7 psi	Spacer modulus
	Alphasp	9.1E-6 1/F	Spacer thermal expansion coefficient

Joint Stiffnesses

Ksb	2.9E6 lb/in	Combined stiffness, shaft and bolt
Kb	8.1E6 lb/in	Bolt stiffness
Ks	5.5E6 lb/in	Shaft stiffness
Ks1	1.8E7 lb/in	First shaft section stiffness
Ks2	1.8E7 lb/in	Second shaft section stiffness
Ks3	1.7E7 lb/in	Third shaft section stiffness
Kro	3.1E6 lb/in	Combined clamped parts stiffness
Ktr	2.2E7 lb/in	Thrust runner stiffness
Kimp2	3.0E7 lb/in	Impeller #2 stiffness
Ksp	4.7E6 lb/in	Spacer stiffness
Kimp1	5.6E7 lb/in	Impeller #1 stiffness
5.0E6 Kimp2f	lb/in	Impeller #2 flange stiffness
1.4E6 Kimp1f	lb/in	Impeller #1 flange stiffness

<u>St</u>	<u>Input</u>	<u>Name</u>	<u>Output</u>	<u>Unit</u>	<u>Comment</u>
					IMPBOLT.TK
					Transfer Pump Impeller Bolt Analysis
					Dated 1-4-95
					by R.F. Guardiani
					Initial Preload Analysis
L	2000	Fi		lbs	Initial bolt preload
		Signal	8849	psi	Bolt axial stress due to preload
		TDEFi	0.00133	in	Total joint deformation due to preload
		DELSbi	0.00069	in	Initial bolt plus shaft elongation
		DELbi	0.00033	in	Initial bolt elongation
		DELSi	0.00037	in	Initial shaft elongation
		DELrci	-0.00064	in	Initial clamped parts total elongation
		DELtri	-0.00009	in	Initial thrust runner elongation
		DELimp2i	-0.00007	in	Initial impeller #2 elongation
		DELSpi	-0.00043	in	Initial spacer elongation
		DELimp1i	-0.00004	in	Initial impeller #1 elongation
					Thermal Analysis
		Ft	-199	lbs	Bolt load due to thermal effects
		Signal	-879	psi	Bolt axial stress due to thermal loads
L	40	T		F	Temperature of interest
	68	Ta		F	Assembly temperature
		TDEFt	-0.00013	in	Total joint thermal deformation
		DELSbt	-0.00007	in	Thermal stress bolt & shaft elongation
		DELbt	-0.00003	in	Thermal stress bolt elongation
		DELSbt	-0.00004	in	Thermal stress shaft elongation
		DELrct	0.00006	in	Thermal stress clamped parts elongation
		DELtrt	0.00001	in	Thermal stress thrust runner elongation
		DELimp2t	0.00001	in	Thermal stress impeller #2 elongation
		DELSpt	0.00004	in	Thermal stress spacer elongation
		DELimp1t	0.00000	in	Thermal stress impeller #1 elongation
		DELt	-0.00170	in	Total thermal growth

Combined Preload & Thermal Results				
L	F	1801	lbs	Net bolt load, preload plus thermal
L	Sigma	7870	psi	Net bolt axial stress from all loads
	JCOMP	0.00120	in	Total joint compression
Dimensional Data for Bolt Analysis				
L	1.40	Lb	in	Bolt free length (head to shaft thds)
	0.625	Dnom	in	Nominal bolt diameter
	11	n	thd/in	Bolt threads per inch
		Dbtd	0.536 in	Bolt diameter at root of threads
		Abmin	0.226 in ²	Bolt cross-section area at minor dia
	0.625	Db	in	Bolt shank diameter
		Ab	0.307 in ²	Bolt area at shank
		Ls	7.35 in	Shaft length
	3.55	Ls1	in	First shaft section length
	1.601	Ds1	in	First shaft section diameter
	2.20	Ls2	in	Second shaft section length
	1.250	Ds2	in	Second shaft section diameter
		Ls3	1.60 in	Third shaft section lg to end of thds
	1.250	Ds3o	in	Third shaft section outside diameter
	0.625	Ds3i	in	Third shaft section inside diameter
	1.00	Ltr	in	Thrust runner thickness
	2.67	Limp2	in	Impeller #2 hub length
	3.25	Lsp	in	Spacer length
	1.83	Limp1	in	Impeller #1 hub length
	2.380	Dtre	in	Thrust runner effective outside diameter
	1.603	Dtri	in	Thrust runner inside diameter
	2.500	Dmp2o	in	Impeller #2 hub outside diameter
	1.602	Dmp2i	in	Impeller #2 hub inside diameter
	1.500	Dspo	in	Spacer outside diameter
	1.258	Dspi	in	Spacer inside diameter
	2.500	Dimp1o	in	Impeller #1 hub outside diameter
	1.251	Dimp1i	in	Impeller #1 hub inside diameter
		As1	2.013 in ²	First shaft section area
		As2	1.227 in ²	Second shaft section area
		As3	0.920 in ²	Third shaft section area
		Atre	2.431 in ²	Thrust runner effective area
		Aimp2	2.683 in ²	Impeller #2 hub area
		Asp	0.525 in ²	Spacer area
		Aimp1	3.680 in ²	Impeller #1 hub area

Bolt Analysis Material Properties			
T304	Bolt		Bolt material: T304 or T316
	Eb	2.80E7 psi	Bolt modulus
	Alphab	9.6E-6 1/F	Bolt thermal expansion coefficient
T410	Shaft		Shaft material: T304 or T410
	Es	2.80E7 psi	Shaft modulus
	Alphas	6.1E-6 1/F	Shaft thermal expansion coefficient
WC	Runner		Runner material: T304, T410 or WC
	Etr	8.96E6 psi	Thrust runner modulus
	Alphatr	2.6E-6 1/F	Thrust runner thermal exp coefficient
CF3	Impeller		Impeller material: CF3 or CD4MCu
	Eimp	2.80E7 psi	Impeller modulus
	Alpheimp	9.0E-6 1/F	Impeller thermal expansion coefficient
T410	Spacer		Spacer mtl: T304/T410/CD4MCu/WC
	Esp	2.80E7 psi	Spacer modulus
	Alphasp	6.1E-6 1/F	Spacer thermal expansion coefficient
Joint Stiffnesses			
	Ksb	2.8E6 lb/in	Combined stiffness, shaft and bolt
	Kb	6.1E6 lb/in	Bolt stiffness
	Ks	5.5E6 lb/in	Shaft stiffness
	Ks1	1.6E7 lb/in	First shaft section stiffness
	Ks2	1.6E7 lb/in	Second shaft section stiffness
	Ks3	1.7E7 lb/in	Third shaft section stiffness
	Krc	3.1E6 lb/in	Combined clamped parts stiffness
	Ktr	2.2E7 lb/in	Thrust runner stiffness
	Kimp2	3.0E7 lb/in	Impeller #2 stiffness
	Ksp	4.7E6 lb/in	Spacer stiffness
	Kimp1	5.6E7 lb/in	Impeller #1 stiffness
5.0E9	Kimp2f	lb/in	Impeller #2 flange stiffness
1.4E8	Kimp1f	lb/in	Impeller #1 flange stiffness

RULE SHEET

S Rule

;Initial Preload Analysis

$$Ks1=As1*Es/Ls1$$

$$Ks2=As2*Es/Ls2$$

$$Ks3=As3*Es/Ls3$$

$$1/Ks=(1/Ks1)+(1/Ks2)+(1/Ks3)$$

$$Kb=Ab*Eb/Lb$$

$$1/Ksb=(1/Ks)+(1/Kb)$$

$$Dbtd=Dnom-0.974/n$$

$$As1=0.7854*Ds1^2$$

$$As2=0.7854*Ds2^2$$

$$As3=0.7854*(Ds3o^2-Ds3i^2)$$

$$Ab=0.7854*Db^2$$

$$Abmin=0.7854*Dbtd^2$$

$$Ktr=Atr*Etr/Ltr$$

$$Kimp2=Aimp2*Eimp/Limp2$$

$$Ksp=Asp*Esp/Lsp$$

$$Kimp1=Aimp1*Eimp/Limp1$$

$$1/Krc=(1/Ktr)+(1/Kimp2)+(1/Ksp)+(1/Kimp1)+(1/Kimp2f)+(1/Kimp1f)$$

$$Atr=0.7854*(Dtre^2-Dtri^2)$$

$$Aimp2=0.7854*(Dimp2o^2-Dimp2i^2)$$

$$Asp=0.7854*(Dspo^2-Dspi^2)$$

$$Aimp1=0.7854*(Dimp1o^2-Dimp1i^2)$$

$$Ls=Ls1+Ls2+Ls3$$

$$Ls+Lb=Ltr+Limp2+Lsp+Limp1$$

$$Fi=Ksb*DELSbi$$

$$Fi=Ks*DELSi$$

$$Fi=Kb*DELBi$$

$$Fi=-Krc*DELRci$$

$$Fi=-Ktr*DELtri$$

$$Fi=-Kimp2*DELimp2i$$

$$Fi=-Ksp*DELSpi$$

$$Fi=-Kimp1*DELimp1i$$

$$TDEFi=DELSbi-DELRci$$

;Thermal Analysis

$$F=Fi+Ft$$

$$DELst=Ft/Ks$$

$$DELBt=Ft/Kb$$

$$DELSbt=DELst+DELBt$$

$$DELtrt=Ft/Ktr$$

$$DELimp2t=Ft/Kimp2$$

$$DELSpt=Ft/Ksp$$

$$DELimp1t=Ft/Kimp1$$

$$DELRct=DELtrt+DELimp2t+DELSpt+DELimp1t$$

```

DELT=DELebt+(Le*Alphas*(T-Ta))+(Lb*Alphab*(T-Ta))
DELT=DELRct+(T-Ta)*(Ltr*Alphatr+Limp2*Alphaimp+Lsp*Alphas+Limpi*Alphaimp)

-TDEFt=DELebt-DELRct

;Combined Analysis

JCOMP=TDEFi+TDEFt

;Stress Analysis

* if Abmin => Ab then Sigmai=Fi/Ab
* if Abmin => Ab then Sigmat=Ft/Ab
* if Abmin => Ab then Sigma=F/Ab
  if Abmin < Ab then Sigmai=Fi/Abmin
  if Abmin < Ab then Sigmat=Ft/Abmin
  if Abmin < Ab then Sigma=F/Abmin

;Material Selection

if Bolt = 'T304 then Eb=2.8E7
if Bolt = 'T304 then Alphab=9.6E-6
* if Bolt = 'T316 then Eb=2.8E7
* if Bolt = 'T316 then Alphab=8.9E-6

* if Shaft = 'T304 then Es=2.8E7
* if Shaft = 'T304 then Alphas=9.6E-6
  if Shaft = 'T410 then Es=2.9E7
  if Shaft = 'T410 then Alphas=6.1E-6

* if Runner = 'T304 then Etr=2.8E7
* if Runner = 'T304 then Alphatr=9.6E-6
* if Runner = 'T410 then Etr=2.9E7
* if Runner = 'T410 then Alphatr=6.1E-6
  if Runner = 'WC then Etr=8.96E6
  if Runner = 'WC then Alphatr=2.6E-6

if Impeler = 'CF3 then Eimp=2.8E7
if Impeler = 'CF3 then Alphaimp=9.0E-6
* if Impeler = 'CD4MCu then Eimp=2.9E7
* if Impeler = 'CD4MCu then Alphaimp=6.5E-6

* if Spacer = 'T304 then Esp=2.8E7
* if Spacer = 'T304 then Alphasp=9.6E-6
  if Spacer = 'T410 then Esp=2.9E7
  if Spacer = 'T410 then Alphasp=6.1E-6
* if Spacer = 'CD4MCu then Esp=2.9E7
* if Spacer = 'CD4MCu then Alphasp=6.5E-6
* if Spacer = 'WC then Esp=8.96E6
* if Spacer = 'WC then Alphasp=2.6E-6

```

8.7 Shaft Fatigue Analysis

8.7.1 Shaft Stresses

The shaft will see the following stresses below the lower bearing which are the stresses at the thrust bearing location (d = 1.60 in.) for fatigue.

$$\begin{aligned} \text{Axial Stress} &= \frac{\text{Axial Thrust}}{\text{Area}} \\ &= \frac{338 + 367}{\pi \left(\frac{1.60}{2} \right)^2} \end{aligned}$$

$$\sigma_a = 350 \text{ psi}$$

$$\text{Bending Stress} = \pm \frac{32M}{\pi d^3}$$

The two impeller radial thrust values are at 115° apart with the first stage impeller radial thrust of 50 lbs. at 9.3 in. from the shoulder and the second stage impeller thrust of 53 lbs. at 4.3 in. from the shoulder. Thus, the moment at the shoulder is:

$$M = [MX^2 + MY^2]^{1/2} = 423 \text{ in. lb.}$$

$$MX = 50 (9.3) - 53 (\sin 25)(4.3) = 369 \text{ in. lb.}$$

$$MY = 0 + 53 (\cos 25)(4.3) = 207 \text{ in. lb.}$$

$$\sigma_b = \pm \frac{32(423)}{\pi (1.60)^3}$$

$$\sigma_b = \pm 1052 \text{ psi}$$

Thus, the combined axial stress is 1402 psi, which is less than the 410 stainless steel yield strength of 38 ksi at 200°F.

$$\text{Torsional Stress} = \frac{16T}{\pi d^3}$$

The maximum output torque is 165 ft-lbs while operating torque is about half of this. Thus, at the thrust bearing location ($d = 1.60$ in.), the torsional stress is:

$$\tau = \frac{16(165)(12)}{\pi(1.60)^3}$$

$$\tau = 2462 \text{ psi}$$

And at the impeller location ($d = 1.25$ in.):

$$\tau = \frac{16(165)(12)}{\pi(1.25)^3}$$

$$\tau = 5163 \text{ psi}$$

Using the allowable for shear stress as half of the yield strength, the allowable for annealed 410 stainless steel is $0.5(38) = 19$ ksi at 200°F.

8.7.2 Stress Concentrations at Critical Shaft Location

The shaft below the lower bearing is the critical location for high cycle fatigue due to the effect of the impeller radial and axial thrust. The thrust bearing location has two stress concentrators, namely the Keyway and a fillet. The lower bearing lip and the thrust runner are not shrunk-on at the area since this complicates the shaft stresses at this critical location. An estimate of the two stress concentrations are as follows, using Peterson's "Stress Concentration Factors".

$$\text{Fillet:} \quad d = 1.60 \text{ in.} \quad D = 2.135 \text{ in.} \quad r = 0.060 \text{ in.}$$

$$r/d = \frac{.06}{1.6} = 0.038 \quad D/d = \frac{2.135}{1.6} = 1.33$$

$$\text{From Figure 78, bending: } K_t = 2.24$$

$$\text{From Figure 79, torsion: } K_{ts} = 1.68$$

Keyway: Note this is a circular Keyway with a round pin key, which is not in Peterson. Thus, the reference "Don't Guess on Non-circular Shafts" by R. Isakower in the November 1980, Machine Design will be used.

Circular Keyway $r = 0.1565$ in.

Shaft $R = 0.80$ in.

$r/R = 0.2$

$K_{ts} = 1.2$

Since this is less than the fillet value and the Keyway is further from the lower bearing (i.e., lower bending stress), the fillet values will be considered limiting.

8.7.3 High Cycle Fatigue Analysis

The shaft factor of safety (FS) in high cycle fatigue is given by the following equation adapted from Peterson's "Stress Concentration Factor's" equation (38).

$$FS = \left[\left(\frac{\sigma_a}{\sigma_f} + \frac{K_f \sigma_a}{\sigma_f} \right)^2 + 3 \left(\frac{\tau_a}{L\sigma_f} + \frac{K_{ts} \tau_a}{\sigma_f} \right)^2 \right]^{-1/2}$$

There is nil steady bending stresses in the shaft and the radial thrust values are static with respect to the pump and, thus, cause a cyclic bending stress in the rotating shaft. The terms are as follows.

- σ_a = Steady axial stress = 440 psi
- σ_y = Yield strength = 38 ksi
- K_f = Conservatively use $K_f = 2.24$
- K_{ts} = Conservatively use $K_{ts} = 1.68$
- σ_a = Cyclic axial stress, assuming a 20% cyclic σ_a addition to the cyclic bending = $1052 + .2(350) = 1122$ psi
- σ_f = Fatigue strength of shaft, assumed to be 33 ksi
- τ_a = Steady torsional stress = 2462 psi
- τ_a = Cyclic torsional stress, assumed to be 20% of $\tau_a = 492$ psi
- L = Torsional limit factor = 1.33 for a round bar

Thus:

$$FS = \left[\left(\frac{350}{38000} + \frac{2.24(1122)}{33000} \right)^2 + 3 \left(\frac{2482}{1.33(38000)} + \frac{1.68(492)}{33000} \right)^2 \right]^{-1/2}$$

$$FS = 6.5$$

The other step in the shaft below the lower bearing is the shoulder from $d = 1.60$ in. to $d = 1.25$ in. Using $d = 1.25$, $D = 1.6$, and $r = 0.06$ in., then:

$$\frac{r}{d} = 0.048 \quad \frac{D}{d} = 1.28 \quad \frac{d}{D} = 0.78$$

$$K_t = 2.05 \text{ (bending)} \quad K_{ts} = 1.58 \text{ (torsion)}$$

The bending stress is:

$$M = [MX^2 + MY^2]^{1/2} = 264.3 \text{ in. lb.}$$

$$MX = 50(5.48) - 53 (\sin 25)(.48) = 263$$

$$MY = 0 + 53 (\cos 25)(.48) = 23$$

$$\sigma_b = \frac{32(264.3)}{\pi(1.25)^3} = 1378 \text{ psi}$$

The axial stress is:

$$\sigma_t = \frac{338 + 367}{\pi \left(\frac{1.25}{2} \right)^2} = 575 \text{ psi}$$

The torsional stress from section 8.7.1 is 5163 psi. Thus, the shaft factor of safety in high cycle fatigue at the second stage impeller is:

$$FS = \left[\left(\frac{575}{38000} + \frac{2.05(1378 + .2(575))}{33000} \right)^2 + 3 \left(\frac{5163}{1.33(38000)} + \frac{1.58(.2)(5163)}{33000} \right)^2 \right]^{-1/2}$$

$$FS = 3.5$$

Thus, the shaft everywhere will have a $FS > 2.0$ in high cycle fatigue.

8.8 Thermal Shock Analysis

Any differential thermal growth can only come from differences in the axial temperature profile of the various components. Very little differential thermal expansion exists due to different materials since most items are 304L SS. An exception is the rotor/stator cans which are covered separately in Section 8.10.2. In general, the axial temperature in the assembly should be nearly identical so thermal stresses are nil. Two exceptions are the thermal shock of insertion of a cold pump into a hot tank and the use of cold purge/dilution water into a hot pipe/pump. The resultant thermal shock stress is given by:

$$\sigma = KE\alpha\Delta T$$

where for 304 in the range of 70-200°F:

$$E = 28 (10^6) \text{ psi}$$

$$\alpha = 9.2 (10^{-6}) \text{ in/in/}^\circ\text{F}$$

and per Roark, "Formulas for Stress and Strain":

$$K = 1.0 \text{ (pipe axially restrained)}$$

$$= \frac{1}{1-\mu} = 1.43 \text{ (local shock)}$$

$$= \frac{1}{2(1-\mu)} = 0.714 \text{ (Pipe with linear radial gradient)}$$

$$= \frac{1}{1-\mu} \left[\frac{A_1}{A_1 + A_2} \right] \text{ (2 pipes joined at the ends with different areas and temperatures.)}$$

$$= 1.82 \text{ (pipe bending stress at fixed end)}$$

Ref. Hetenyi, "Beams on Elastic Foundation"

Now any or all of these cases may occur for the two thermal shock cases considered. The ASME Code fatigue waiver requirements bounds these cases by using $K=2$ as will be done herein. Thus, the limiting temperature shock will be calculated assuming various allowable thermal shock stresses (σ_t).

$$\Delta T = \frac{\sigma_t}{2(28)(9.2)} = 0.00184 \sigma_t$$

$$\text{or } \sigma_t = 515.2 \Delta T$$

$\sigma_t(\text{psi}) =$	20,000	25,000	30,000	40,000	60,000
$\Delta T(^{\circ}\text{F}) =$	39	48.5	58.2	77.6	116.5

Thus, for insertion of a pump at an ambient temperature of say 50°F into a tank at say 180°F, a $\Delta T = 180 - 50 = 130^\circ\text{F}$ will exist and the thermal shock stress potential is:

$$\sigma_t = 515.2(130) = 66,976 \text{ psi}$$

This is well above yield strength (~25 ksi) for 304LSS hence unless localized yielding is acceptable, care should be used to minimize thermal shock ΔT .

Localized distortion is the only real concern, since the worst case fatigue analysis would be to consider from -10°F to 200°F.

$$\Delta T = 210^\circ\text{F}$$

Using a code worst case fatigue reduction factor of $K_f = 5$, the cyclic stress would be:

$$\sigma_c = \frac{K_f}{2}(515.2)(210)$$

$$\sigma_c = 241 \text{ ksi}$$

A Section VIII-2 fatigue curve for stainless steel gives an allowed cycles of 150 for this worst case scenario. Assuming that this occurs once a month, the minimum life due to thermal fatigue is 12.5 years, more than the 10 years life of the component. Note that the Code has a factor of safety of 20 on cycles for low cycle fatigue.

In the case of purge/dilution water of say 50°F being injected in a pump in the tank with the pipe at say 180°F, then:

$$\begin{aligned}\sigma_{\max} &= E\alpha T \\ &= 257.6 (180-50) \\ &= 33,488 \text{ psi}\end{aligned}$$

Now this may exceed the yield strength, although the actual yield strength may be higher. Hence, some yielding may occur on the initial shock. But, then a residual stress will be set up such that elastic cycling will then occur thereafter. The fatigue life under this condition is an infinite number of cycles.

A review was made to determine the thermal shock capability of Tungsten Carbide and SiC, either of which may be used in the bearings. Although no information was found on Tungsten carbide, Coors Ceramic Company has identified a minimum shock ΔT of 570°F for SiC. Since Tungsten Carbide is generally considered to be "tougher" than SiC, it is estimated that Tungsten Carbide has at least the same thermal shock capability of SiC. A ΔT of 570°F is acceptable for the transfer pump application.

8.9 Locked Rotor Analysis

The motor rotor bars are joined at the ends by a connecting ring to form a squirrel cage assembly. During a locked rotor event (limited to 10 sec.), the electrical input is all converted to heat in the copper bars and connecting rings. An EMD PC program was used to evaluate the stresses in the bars which are restrained radially by the shaft while the two connecting rings grow radially with the increase in temperature of the locked rotor event. The bars heat up to 440°F (or less) during a locked rotor event while the connecting ring heats up to 270°F (or less). The resulting bar and ring stresses were found to be low; e.g., less than 1 ksi.

8.10 Miscellaneous Analyses

8.10.1 Rotor End Ring Overhang

The rotor assembly uses an end ring to support the rotor can over the rotor bar/connection ring junction area. The bending stress in this area is approximated by a fixed end cylinder.

$$\sigma_r = \pm 1.54 \frac{Pr}{t} \quad \text{(From "Theory of Plates and Shells", by Timoshenko and Woinowsky-Krieger)}$$

where using $r = 1.72$, $t = 0.09$ in.,

$$\begin{aligned}\text{Design Case: } \sigma_s &= \pm 1.54 \frac{(400)(1.72)}{(0.09)} \\ &= 11,772 \text{ psi}\end{aligned}$$

$$\begin{aligned}\text{Hydro Case: } \sigma_s &= \pm 1.54 \frac{(660)(1.72)}{(0.09)} \\ &= 19,425 \text{ psi}\end{aligned}$$

which is less than yield strength so is acceptable.

8.10.2 Rotor and Stator Can Analyses

Input: Rotor Can D = 3.75 in.

Can shrink-fit interference = 3 mils max. (to 2 mils clearance)

Can thickness, $t = 0.033$ in.

Hastelloy C-276 Material:

Use $E = 29.8 (10^6)$ psi (Young's Modulus)

$\nu = 0.3$ (Poisson's ratio)

$\sigma_y = 52$ ksi (yield strength)

Slot Width:

Stator I = .18 in., Rotor I = 0.125 in.

(a) Bending stress in a Can over unsupported length I:

$$\begin{aligned}\sigma &= \frac{P}{2} \left[\frac{I}{t} \right]^2 \\ &= \frac{660}{2} \left[\frac{.18}{.033} \right]^2 \\ \sigma &= 9818 \text{ psi}\end{aligned}$$

This is low against a yield of 52 ksi

$$\text{Elastic Deflection} = 0.0284 \frac{PI^4}{Et^3} = 1.84(10^{-6}) \text{ in} \sim 0$$

(b) Shrink-fit stress (rotor can):

$$\sigma = \frac{\Delta D}{D} E$$

$$= \frac{3(10^{-3})}{3.75} (29.8)(10^6)$$

$$\sigma_{max} = 23,840 \text{ psi (tension, maximum)}$$

Yield strength is 52 ksi

(c) Temperature effect on shrink:

$$\text{Use } \alpha_{can} = 7.285 (10^{-6})$$

$$\alpha_{rotor} = 5.423 (10^{-6})$$

$$\Delta D = D \Delta \alpha \Delta T \quad \Delta T = 200 - 70 = 130^\circ F$$

$$\Delta D = 3.75 (7.285 - 5.423) 10^{-6} (130)$$

$$\Delta D = 0.908 (10^{-3}) \text{ in.}$$

$$\Delta D = 0.9 \text{ mils shrink lost}$$

(d) Can axial stress on heatup:

Can must see all differential growth.

Steady State:

$$\sigma = E(\Delta \alpha)(\Delta T)$$

$$= 29.8(10^6)(1.862)(10^{-6})(130)$$

$$= 7213 \text{ psi (compression in Can due to restrained axial growth)}$$

Shock:

$$\sigma = E \alpha \Delta T = 29.8(7.285)(130)$$

$$= 28,222 \text{ psi (54% of yield)}$$

(e) Hoop Pressure stress (if Can is not supported, i.e., clearance)

Rotor Can:

$$\sigma = \frac{PD}{2t}$$

$$= \frac{660(3.75)}{2(0.033)}$$

$$\sigma = 37,500 \text{ psi}$$

But this could be deflection limited. Thus, yield is not exceeded due to pressure load.

Stator Can:

$$\sigma = \frac{660(4.15)}{2(.033)}$$

$$\sigma = 41,500 \text{ psi}$$

Thus, both Cans will carry the pressure elastically (except for buckling).

(f) Centrifugal Can stresses:

$$\text{Tensile stress in Can} = \frac{\rho V^2}{g}$$

$$V = R\omega$$

$$R = \frac{3.75}{2}$$

$$\omega = 3860 \text{ rpm} \left(\frac{2\pi}{60} \right)$$

$$V = \frac{3.75(3860)\pi}{60}$$

$$V = 758 \text{ in/sec}$$

$$\sigma = \frac{\left(0.304 \frac{\text{lb}}{\text{in}^3} \right) \left(758 \frac{\text{in}}{\text{sec}} \right)^2}{386.4 \frac{\text{in}}{\text{sec}^2}}$$

$$\sigma = 452 \text{ psi}$$

Negligible effect at design speed and lower.

(g) Stator Can Collapse Pressure (Gas pressure from potting compound emission)

Ratio data from another pump.

$$P = \left(\frac{.033}{.017} \right)^{2.6} (4.9)$$

$$P = 27.5 \text{ psi}$$

(h) Rotor/Stator Can Thermal Buckling:

Temperature: Upon insertion of the pump from a Cold atmosphere to the pit ($\leq 200^\circ\text{F}$), the pump must heat up slowly.

That is, rapid insertion of the 0°F pump into 200°F pit fluid can result in the Can heating up faster than the supports. No radial offsets of the Can are considered.

Analysis: Critical buckling temperature differential.

Transfer Pump Stator Can:
(Also Rotor Can)

$$R = \frac{4.15}{2} = 2.075 \text{ in.}$$

$$t = 0.033 \text{ in.}$$

From a test on a 7.9 in. radius Can for a different pump.

$$\frac{R_r}{R} = \frac{7.9}{2.075} = 3.807$$

R_r = radius of reference test can

$$\text{Assuming } \sigma_{cr} = CE \left[\frac{t}{R} \right]$$

For $t = 0.033$ in. can thickness, $\Delta T_{cr} = 300^\circ\text{F}$ was obtained.

Thus, for Transfer Pump:

$$\Delta T_{cr} = 3.807(300) = 1142^\circ\text{F}$$

Thus, thermal buckling is not considered a problem for the rotor/stator cans.

If the can sees uniform axial compression:

$$\sigma_{\text{compression}} = E \frac{\Delta l}{l}$$

Stator Can $l = 29$ in.

Use $E = 29 (10^6)$ psi

Then: $\sigma_c = \Delta l(10^6)$

So:

$\Delta l =$	.1	1	2	5	10 (mils)
$\sigma_c =$	100	1000	2000	5000	10,000 (psi)

If something causes the Can to not seek uniform compression stress (axially and circumferentially), then local Can stresses will be higher.

In summary, the rotor/stator Cans should survive operational and shock temperatures and pressures.

9.0 BEARING EVALUATION

The Model 2A-513-A1 Transfer Pump is equipped with two radial bearings and a single acting thrust bearing. A static analysis of these bearings, shown in Figure 15-4 is provided in this section. Refer to Section 10.0 for an evaluation of bearing performance in the dynamic analysis.

The Transfer Pump must operate in the following environment:

- Radioactive hazardous waste (up to 600 Rads per hour)
- Solids content up to 50 percent volume, with particle sizes from 1 to 1000 microns (95 percent of the particles are less than 50 microns)
- Fluid viscosity between 1.0 and 30.0 centipoise
- Fluid specific gravity between 1.00 and 1.50
- Fluid temperatures up to 200 °F

The Transfer Pump bearings use the hazardous waste tank process fluid for cooling and lubrication. The process fluid main discharge flow from the second stage diffuser travels up the annulus outside the stator shell. Some of the process fluid is bled off into the upper rotor cavity for motor cooling and bearing lubrication. The process fluid which circulates through the motor flows down past the upper radial bearing, down through the annulus between the rotor and stator cans, down past the lower radial bearing, down past the thrust bearing, and into the hydraulic region above the second stage impeller upper shroud where it mixes with the impeller discharge flow and becomes part of the main discharge flow. Refer to Section 5.0 for a complete description of the flow paths in the Transfer Pump.

The following sections provide descriptions and analyses of the radial and thrust bearings.

9.1 Radial Bearings

The radial bearings are film riding hydrodynamic bearings which are cooled and lubricated by the waste tank process fluid which is pumped through the motor. The radial bearings are pivoted pad bearings. The pivoted pad bearings utilize four pads which individually pivot for self-alignment with the rotating assembly. A cross-section of the radial bearings is shown in Figure 9-1.

The pivoted pad radial bearings have Silicon Carbide (SiC) journals and bearing inserts to minimize abrasive wear and to maximize life. The SiC is the direct sintered alpha form which is extremely hard and highly resistant to corrosion, erosion, sliding wear, high temperature and thermal shock. The SiC radial bearing inserts are retained in 410 stainless steel pad holders. Each pad holder pivots about a 410 stainless steel pivot pin which is pressed into the bearing housing. The housings of both the upper and lower radial bearings are 304 stainless steel.

Refer to Figure 9-2 for a visual description of the nomenclature used for the pivoted pad bearings. The upper pivoted pad bearing is 2.50 inches in diameter, is 2.50 inches long, and has a radial pivot clearance of 0.0025 to 0.0030 inch ($r_b - r_j$ in Figure 9-2). The upper bearing has a nominal 0.500 preload ratio, which means the radial clearance between the journal and the pads ($r_b - r_j$ in Figure 9-2) at the pivot pins is nominally 50% of the difference between the radius machined into the pads and the radius of the journal ($r_p - r_j$ in Figure 9-2). Note that in a concentric bearing, the preload ratio $\{(r_b - r_j) / (r_p - r_j)\}$ per Figure 9-2 is equal to 1.00. Preloading is accomplished to preclude spragging of the unloaded pad(s). The upper pivoted pad bearing housing is attached to the stator with a 0.001 to 0.005 inch diametral shrink fit and is tack welded to the stator shell for locking.

The lower radial bearing is 3.25 inches in diameter, is 2.30 inches long, and has a radial pivot clearance ($r_b - r_j$ in Figure 9-2) of 0.0030 to 0.0035 inch. The lower bearing also has a nominal 0.500 preload ratio $\{(r_b - r_j) / (r_p - r_j)\}$ per Figure 9-2. The lower radial bearing housing is fitted into the stator with a 0.001 to 0.005 inch

diametral clearance and secured to the stator with four 0.375 inch capscrews and lock cups.

Figure 9-1 Radial Bearing Cross-Section

Figure 9-2 Pivoted Pad Radial Bearing Nomenclature

Radial Bearing Misalignment Potential

The static angular misalignment in the radial bearings is caused by the following lateral misalignments between the pump radial bearings which could tilt the shaft rotational axis as well as shift it laterally:

- A 0.040 inch runout in the stator shell between the upper and lower radial bearing housing fit diameters.
- An interference fit between the upper radial bearing housing and the stator (0 clearance).
- A 0.005 inch maximum diametral clearance between the lower radial bearing housing fit diameter and the stator housing.
- A 0.005 inch maximum runout in the upper and lower radial bearing housings between the stator fit diameters and the bearing bores.
- A 0.006 inch design maximum diametral pivot clearance in the upper radial bearing with the rotor having an eccentricity ratio of 1.41 relative to the pivot clearance.
- A 0.007 inch design maximum diametral pivot clearance in the lower radial bearing with the rotor having an eccentricity ratio of 1.41 relative to the pivot clearance.

The maximum runout between the upper and lower radial bearings, as assembled in the stator, is the direct sum of the above runouts and diametral clearances, or:

$$RO_{max} = 0.040 + 0 + 0.005 + 0.005 + 1.41 * (0.006 + 0.007)$$

$$RO_{max} = 0.0684 \text{ inch}$$

The maximum eccentricity of the upper radial bearing relative to the lower bearing is $\frac{1}{2}$ the runout, or:

$$E_{\max} = 0.0684 / 2$$

$$E_{\max} = 0.0342 \text{ inch}$$

The distance between the upper and lower radial bearing centerlines is 34.13 inches. The maximum angular misalignment between the upper and lower radial bearings is, therefore:

$$A_{\max} = 0.0342 / 34.13$$

$$A_{\max} = 1.00 \times 10^{-3} \text{ radian}$$

Since the above results are worst case with an almost zero probability of occurring, a three-dimensional tolerance stack-up analysis was performed on the as-assembled Transfer Pump using a Monte Carlo technique to link angular misalignments with probabilities. Using the runouts and clearances identified above, the tolerance stack-up analysis predicts a 95 percent probability that the as-assembled angular misalignment in either radial bearing will not exceed 0.63×10^{-3} radian and a 99 percent probability that the misalignment will not exceed 0.71×10^{-3} radian.

It is expected that some wear will occur in the bearings due to the presence of particulates in the process fluid. As a worst case evaluation, it is assumed that the maximum wear will be 0.006 inch from each component (i.e., the journals will decrease 0.012 inch in diameter and the bearing bores will increase 0.012 inch in diameter). The effective pivot diametral clearance will increase to 0.030 inch at the upper radial bearing ($0.006 + 0.012 + 0.012 = 0.030$) and will increase to 0.031 inch at the lower radial bearing ($0.007 + 0.012 + 0.012 = 0.031$).

As previously calculated for the as-assembled condition, the maximum runout between the upper and lower radial bearings, after maximum wear, is the direct sum of the runouts and diametral clearances, or:

$$RO_{max} = 0.040 + 0 + 0.005 + 0.005 + 1.41 * (0.030 + 0.031)$$

$$RO_{max} = 0.136 \text{ inch}$$

The maximum eccentricity of the upper radial bearing relative to the lower bearing is $\frac{1}{2}$ the runout, or:

$$E_{max} = 0.136 / 2$$

$$E_{max} = 0.068 \text{ inch}$$

The distance between the upper and lower radial bearing centerlines is 34.13 inches. The maximum angular misalignment between the upper and lower radial bearings in the maximum wear condition is, therefore:

$$A_{max} = 0.068 / 34.13$$

$$A_{max} = 1.99 \times 10^{-3} \text{ radian}$$

Since the above results are worst case with an almost zero probability of occurring, a three-dimensional tolerance stack-up analysis was performed for the maximum wear condition using the same Monte Carlo technique previously discussed to link angular misalignments with probabilities. Using the runouts and clearances identified above, the tolerance stack-up analysis predicts a 95 percent probability that the as-assembled angular misalignment in either radial bearing will not exceed 1.32×10^{-3} radian and a 99 percent probability that the misalignment will not exceed 1.40×10^{-3} radian.

Based on the above analyses, the radial bearings should be capable of compensating for angular misalignment of no less than 2.00×10^{-3} radian.

Pivoted Pad Radial Bearing Misalignment Capability

The individual pads of both the upper and lower radial bearings independently pivot axially and circumferentially about pivot pins attached to the inside of the bearing housings. Each of the pivot pins has a spherical radius machined onto the head of the pin to allow the pad to pivot. The pivot pins seat on a flat, central recess in the back of the pads. The recess prevents the pin from slipping out of the pivot location while allowing the sleeve to pivot. The pivot pins for both radial bearings are identical. The pin head diameter is 0.600 ± 0.005 inch, and the spherical radius on the head of the pin is 18 ± 1 inch. The equation which governs the alignment capability of the pads on the pin spherical radii is as follows:

$$d = 2 * R * \sin \Theta$$

or,

$$\Theta = \sin^{-1} (d / (2 * R))$$

where: R is the spherical radius, inch

Θ is the misalignment capability, radian

d is the diameter over which the spherical radius acts, inch

The minimum misalignment capability occurs when the spherical radius is a maximum (19 inches) and the pin diameter is a minimum (0.595 inch). The results are as follows:

$$\Theta = \sin^{-1} (0.595 / (2 * 19))$$

$$\Theta = 15.66 \times 10^{-3} \text{ radian}$$

The results indicate that both the upper and lower radial bearings can accommodate a minimum angular misalignment of 15.66×10^{-3} radian. This is 7-½ times the maximum expected misalignment. Therefore, the alignment capability of the pivoted pad radial bearings is acceptable.

Pivot Pin Stresses

The contact stresses due to axial loading of the spherical seat of a pivot pin against the flat recess of a pad holder were solved using Roark's Formulas for Stress & Strain (6th Edition), Table 33, Case 1a (sphere on a flat plate). The 410 stainless steel pins and pad holders have a modulus of 29×10^6 psi and a Poisson's ratio of 0.30. For the upper radial bearing calculations, a maximum radial bearing load of 340 lbs was estimated to occur axially on a single pivot pin which has the minimum spherical radius, 17 inches. The maximum compressive stress was 38565 psi, which is less than the 45000 psi yield strength of 410 stainless steel. The maximum combined elastic deformation of the pin and pad holder was 0.00025 inch.

For the lower radial bearing calculations, the spherical radius of the pin was also 17 inches. The maximum axial load on a single pin was estimated to be 500 lbs. The maximum compressive stress was 43985 psi. This is less than the 45000 psi yield strength of the 410 material. The maximum combined elastic deformation of the pin and pad holder was 0.00032 inch.

In the event that particulates bridged the gap between the journal and the pads on the radial bearings, the pivot pins would be subject to side contact stresses and shear stresses when the motor was started. For the analysis of this condition only the stresses in the upper pivoted pad radial bearing will be evaluated because the pins are identical for each bearing and the loads will be higher on the smaller diameter upper bearing. It is also conservatively assumed that the entire motor torque acts on only a single one of the four pins in the upper bearing.

The maximum torque a radial bearing pad could be subject to is the total 110 ft-lbs of torque. The upper radial bearing pad is at a radius of 1.25 inches. The maximum applied tangential load at one of the pins is:

$$F = T / R$$

$$F = 110 \times 12 / 1.25$$

$$F = 1056 \text{ lbs}$$

The contact stresses between the side of the pin and the side of the axial recess in the pad holder due to the 1056 lb tangential force loading were solved using Roark's Formulas for Stress & Strain (6th Edition), Table 33, Case 2c (cylinder in a cylindrical socket). The 410 stainless steel pins and pad holders have a modulus of 29×10^6 psi and a Poisson's ratio of 0.30. The contact length was 0.15 inch. The minimum diameter of the pin is 0.595 inch, and the maximum diameter of the socket was 0.628 inch. The results indicated a maximum contact stress of 79425 psi. This exceeds the minimum yield strength of the pin and pad holder (45000 psi). It is expected that this condition would cause local yielding of the pin and holder where contact occurred. This is not expected to preclude acceptable operation of the bearing after the bearing breaks free.

The minimum diameter of the pin, where it is relieved under the head, is 0.318 inch. The cross-section area of the pin at this location is 0.079 in². A radial (relative to the pin) load of 1056 lbs would produce the following maximum shear stress on the pivot pin:

$$\tau = F / A$$

where: F is radial force on the pin, lbs

A is the cross-section area of the pin, in²

τ is the shear stress, psi

$$\tau = 1056 / 0.079$$

$$\tau = 13367 \text{ psi}$$

The calculated shear stress is less than the shear stress which could cause yielding for 410 stainless steel, 22500 psi, and is therefore acceptable.

SiC Pad to Pad Holder Attachment

The SiC pads are attached to the 410 stainless steel pad holders with Locktite (catalog number 085). Additionally, the SiC pads are locked against the pads by 0.125 inch thick endplates which are clamped onto the leading and trailing edges of the pads. Each endplate is secured to the pad holder by two 0.164 - 32UNC x 0.375 long 316 stainless steel flat head screws. The endplates secure the SiC pads to the pad holders both circumferentially and radially.

In the event that particulates bridged the gap between the journal and the pads on the radial bearings, the SiC pads would be subject to tangential forces when the motor was started. Under normal conditions the Locktite between the SiC pads and the 410 stainless steel pad holders can withstand a shear loading of

2000 psi at room temperature and 1500 psi at 200 °F without failing, per the Loctite technical data sheet. The Loctite shear area of a single upper radial bearing pad is 4.70 in² and the Loctite shear area of a single lower radial bearing pad is 5.65 in². The shear force required to break the Loctite bonds is 7050 lbs on a upper pad (1500 * 4.70 = 7050) and 8475 lbs on a lower pad (1500 * 5.65 = 8475). Since the maximum tangential force that can be applied to a pad is 1056 lbs (110 ft-lbs of torque at a 1.25 inch upper pad radius) the Loctite should secure the pads to the pad holders.

In the event that the Loctite loses its adhesion strength the endplates will secure the pads to the pad holders. Tangential forces on the pads would load the endplates and put the flat head capscrews in tension. For the analysis of this condition only the stresses in the upper pivoted pad radial bearing is evaluated because the endplates and flat head screws are identical for each bearing and the loads will be higher on the smaller diameter upper bearing. It is conservatively assumed that the entire motor torque acts on only a single pad in the upper bearing.

The maximum torque a single radial bearing pad is subjected to is 110 ft-lbs. The upper radial bearing pad is at a radius of 1.25 inches and the maximum tangential force that can be applied to the pad is 1056 lbs (110 * 12 / 1.25 = 1056). The maximum length over which the tangential force acts is approximately 0.10 inch (from the SiC insert to the base of the head of the screws); therefore the maximum moment on the endplate is 106 in-lbs (1056 * 0.10 = 106). The active length of the endplate is 2.10 inches long, so the moment of inertia of the plate area is:

$$I = 2.10 * (0.125)^3 / 12$$

$$I = 0.000342 \text{ in}^4$$

The maximum bending stress in the endplate is:

$$\sigma = 106 * (0.125 / 2) / 0.000342$$

$$\sigma = 19370 \text{ psi}$$

The yield strength of 304 stainless steel sheet is 30000 psi, minimum.
Therefore, the bending stress in the endplate is acceptable.

On the endplate, the distance from the back edge of the plate (pivot location) to the applied tangential force location is 0.45 inch, and the distance from the back edge of the plate to the centerline of the screws is 0.35 inch. With a 1056 lb tangential force on the endplate, the axial load on both endplate screws is:

$$F_{ax} = 1056 * 0.45 / 0.35$$

$$F_{ax} = 1358 \text{ Lbs}$$

The load on each screw is 679 lbs (1358 / 2 = 679). The stresses in each screw are calculated as follows:

$$A_s = (\pi / 4) * (D - (0.9743 / n))^2$$

where: A_s is the tensile stress area of the screw, in^2

D is the basic diameter of the screw, 0.164 inch

n is the number of threads per inch, 32

$$A_s = (\pi / 4) * (0.164 - (0.9743 / 32))^2$$

$$A_s = 0.014 \text{ in}^2$$

$$\sigma = F / A_s$$

$$\sigma = 679 / 0.014$$

$$\sigma = 48500 \text{ psi}$$

The yield stress of the screw material is 95000 psi. Therefore, the axial stresses in the screw are acceptable.

The yield stress of the 410 stainless steel pad holder material, 45000 psi, is substantially less than the yield stress of the screw. Therefore, even though the bolt (external) thread shear area is somewhat smaller than the pad holder (internal) thread shear area, the pad holder internal thread shear area is limiting relative to thread engagement length. The internal thread shear area is calculated as follows:

$$AS_n = \pi * n * Le * D_{s,min} * [(1 / (2 * n)) + 0.57735 * (D_{s,min} - E_{n,max})]$$

where: AS_n = minimum thread shear area for internal thread, in²

n = number of threads per inch, 32

Le = length of engagement, 0.20 inch

$D_{s,min}$ = minimum major diameter of external thread, 0.1631 inch

$E_{n,max}$ = maximum pitch diameter of internal thread, 0.1475 inch

$$AS_n = \pi * 32 * 0.20 * 0.1631 * [(1 / (2 * 32)) + 0.57735 * (0.1631 - 0.1475)]$$

$$AS_n = 0.081 \text{ in}^2$$

$$\tau = F / AS_n$$

$$\tau = 679 / 0.081$$

$$\tau = 8383 \text{ psi}$$

A guideline for thread shear stress is 25 percent of the tensile stress (50 percent of the tensile stress is considered yielding in shear). An acceptable limit is, therefore, 11250 psi ($45000 / 4 = 11250$). The shear on the threads is acceptable.

Radial Bearing Journal Attachment

The journals cannot be shrunk onto the rotor shaft because tensile stresses may cause the SiC ceramic to fracture. The journals must maintain a clearance with temperatures as high as 220 °F, accounting for internal motor cooling fluid temperature increases above the 200 °F maximum ambient temperature. Additionally, the journals must maintain a clearance at assembly temperatures as high as 300 °F. The upper journal is a slip fit onto the 410 stainless steel shaft with a required minimum diametral clearance of :

$$DC_u = 1.66 \text{ in} * (6.1 \times 10^{-6} \text{ }^{\circ}\text{F}^{-1} - 2.2 \times 10^{-6} \text{ }^{\circ}\text{F}^{-1}) * (300 \text{ }^{\circ}\text{F} - 68 \text{ }^{\circ}\text{F})$$

$$DC_u = 0.00150 \text{ inch}$$

The lower journal is a slip fit onto the 410 stainless steel journal holder with a minimum diametral clearance of:

$$DC_l = 2.57 \text{ in} * (6.1 \times 10^{-6} \text{ }^{\circ}\text{F}^{-1} - 2.2 \times 10^{-6} \text{ }^{\circ}\text{F}^{-1}) * (300 \text{ }^{\circ}\text{F} - 68 \text{ }^{\circ}\text{F})$$

$$DC_l = 0.00233 \text{ inch}$$

The journals are assembled onto the shaft and journal holder with all parts at a temperature of 250 to 300 °F. The journal axial clamp nuts are hand-tightened, and the assembly is allowed to cool. The difference in coefficients of thermal

expansion between the 410 stainless steel shaft / journal holder ($6.1 \times 10^{-6} \text{ }^{\circ}\text{F}^{-1}$) and the SiC journals ($2.2 \times 10^{-6} \text{ }^{\circ}\text{F}^{-1}$) load the journals axially in compression.

A TKSolver program was developed to perform the appropriate calculations. The results indicate that at the maximum tank fluid temperature of 200 °F, the axial clamping force on the upper journal is 8158 lbs and can resist a torque of 141 ft-lbs (with a 0.10 coefficient of friction), which is more torque than the motor can produce. The compressive stress on the upper journal under this loading condition is 2972 psi, and the clamp nut thread shear stress is 3018 psi. At 40 °F the axial clamping force on the upper journal increases to 33942 lbs, and the journal can resist a torque of 588 ft-lbs. The journal axial compressive stress is 12367 psi. Also, the shear stress on the clamp nut threads is 12556 psi. At the yield stress of the clamp nut threads, 15000 psi, the temperature is -1 °F. This is the limiting low temperature storage condition for the pump.

At the maximum process fluid temperature of 200 °F, the axial clamping force on the lower journal is 7750 lbs and can resist a torque of 188 ft-lbs (0.10 coefficient of friction), which is more torque than the motor can produce. The compressive stress on the lower journal under this loading condition is 2493 psi, and the shear stress on the clamp nut threads is 1818 psi. At 40 °F the axial clamping force on the lower journal increases to 32230 lbs, and the journal can resist a torque of 782 ft-lbs. The journal axial compressive stress is 10369 psi. The shear stress on the clamp nut threads is 7561 psi. At the limiting condition for the upper journal assembly, -1 °F, the clamping force on the lower journal is 38502 lbs and the journal can resist a torque of 934 ft-lbs. The shear stress on the clamp nut threads is 9033 psi, less than the 15000 psi yield strength of the material. Therefore, the stresses in the upper journal assembly are limiting.

Wear of SiC Radial Bearing Material

Direct sintered alpha SiC radial bearings were tested at WSTC with two abrasive simulants as lubricants to determine the rate of wear of the SiC

material in process fluid. The properties of each slurry are listed below in Table 9-1:

Table 9-1 - SiC Bearing Test Simulants

<u>Simulant</u>	<u>Miller</u> <u>Number</u>	<u>% Solids</u>	<u>Constituents</u>	<u>Weight</u> <u>% of Solids</u>	<u>Weight</u> <u>% Slurry</u>	<u>DPH</u> <u>Hardness</u>	<u>Particle</u> <u>Size,</u> <u>µm</u>
#1	87	30	alumina	5.00	1.5	1900	10-63
			silica	95.00	28.5	1100	2-8
#2	112	20	alumina	0.03	0.006	1900	10
			silica	0.54	0.108	1100	2-8
			zeolite	2.58	0.516	800	20-24
			albite	32.93	6.586	800	10-11
			goethite	63.92	12.784	450	4-5

Note that Simulant #1 had a higher solids content than Simulant #2 (30% versus 20%) and a larger quantity of harder materials (as measured by the Vicker's Diamond Pyramid Hardness (DPH) test, with units of kg/mm²), yet the ASTM-G-75 Miller Number, a measure of slurry abrasiveness, was lower (87 versus 112). A partial explanation for this may be that for a given particle hardness many small particles cause greater wear than few large particles because the overall contact area over a given time period is larger in the Miller Number test rig.

The SiC radial bearings tested in the simulants were 3.94 inches in diameter by 1.80 inches long with diametral clearances of 0.005 inch. The bearings were operated at 3200 rpm. With Simulant #1 the load was 200 lbs, and with Simulant #2 a 91 lb load was maintained. The simulant temperature was maintained at less than 185 °F in Simulant #1 and at 85 °F in Simulant #2.

With Simulant #1 the diameter of the SiC bearing bore increased 0.001 inch in only 5 hours with a load of 200 lbs. The wear of the journal could not be accurately measured because an axial crack developed in the journal during testing due to thermal growth interference with the shaft; however the diametral wear of the journal was estimated to be 0.001 inch, also. The estimated

average component radial wear with this Simulant #1 was 0.0005 inch. The average component wear rate, corrected to the 91 lb load of the Simulant #2 test (since wear is assumed to be proportional to load), was:

$$WR_{\#1} = 0.0005 * 1000 * (91 / 200) / 5$$

$$WR_{\#1} = 0.0455 \text{ mil/hour of component radial wear}$$

The bearing was operated for 300 hours in Simulant #2 with a 91 lb load. The journal diameter decreased 0.001 inch, and the bearing bore diameter increased 0.001 inch over the test period. This resulted in a average component radial wear of 0.0005 inch. The average component wear rate over the 300 hours is:

$$WR_{\#2} = 0.0005 * 1000 / 300$$

$$WR_{\#2} = 0.00167 \text{ mil/hour of component radial wear}$$

The Simulant #1 SiC wear rate was over 27 times the wear rate with Simulant #2 when normalized for the same load, even though the Miller Number was lower. One of the reasons for this can be found in relative hardness comparisons. Per ASTM-G-75 (Miller Number) the wear material is a chrome alloy steel with a DPH hardness of approximately 600. All materials in Simulants #1 and #2 had hardnesses close to or greater than the chrome steel hardness test material (i.e., the test piece hardness was 600 and the hardness of the materials in the simulants varied from 450 to 1900). On the other hand, the SiC bearing material had a DPH hardness of approximately 2800. There is only one simulant material which is hard enough to have a significant wear effect on the SiC no matter how small the amount in the simulant – alumina with a hardness approaching 1900. The rest of the simulant materials are so soft relative to the SiC (450 to 1100 DPH versus 2800 DPH) that their impact on SiC

bearing wear is very small and very large quantities are needed to cause significant wear. The affect of simulant materials on SiC wear can therefore be separated into two discrete groups: 1) the affect of alumina on SiC and 2) the affect of all the other minerals in the simulants on SiC. The Miller Number can be used to simulate the wear affect of all the minerals other than alumina. Since the affects of Miller Number and alumina would be additive, a function that describes the wear rate of the SiC bearings is:

$$WR = \alpha \cdot M + \beta \cdot C_{\text{alumina}}$$

where: WR is the wear rate with a 91 lb load, mils/hr

α is a constant to be determined based on Miller Number

M is the Miller Number

β is a constant to be determined based on alumina concentration

C_{alumina} is the weight percent of alumina in the simulant

Using the results of the wear testing on the two SiC bearings, where: 1) with Simulant #1 the wear rate was 0.0455 mil/hour with a 1.5% weight concentration of alumina and a Miller Number of 87 and 2) with Simulant #2 the wear rate was 0.00167 mil/hour with a 0.006% weight concentration of alumina and a Miller Number of 112, the wear rate equations become:

$$0.0455 = \alpha \cdot 87 + \beta \cdot 1.5$$

$$0.00167 = \alpha \cdot 112 + \beta \cdot 0.006$$

Solving the two equations for α and β results in:

$$\alpha = 1.33 \times 10^{-5}$$

$$\beta = 0.0296$$

The overall bearing wear is not only a function of the Miller Number and the amount of alumina in the lubricant, but it is also a function of the relative velocity of the journal to the sleeve, and the pressure load on the bearing. These last two are the "PV" values commonly used in bearing wear evaluations. Therefore, for various bearing sizes, operating speeds, and fluid abrasiveness, the diametral wear of a direct sintered alpha SiC bearing can be estimated by:

$$W = (1.33 \times 10^{-5} * M + 0.0296 * C_{\text{alumina}}) * (\pi * D * N * P / PV_0) * T$$

where: W is the total diametral wear, mils

M is the Miller Number of the abrasive lubricant

C_{alumina} is the concentration of alumina in the slurry, weight %

D is the diameter of the bearing, inches

N is the operating speed of the bearing, rpm

T is the total operating time, hours

P is the projected pressure loading of the journal on the bearing
(F / L * D), psi

F is the force on the bearing, lbs

L is the length of the bearing, inches

PVo is the PV rating of the bearings tested, 500000 lb/min-in for Simulant #1 with a 91 lb load on a 3.94 inch diameter x 1.80 inch long bearing at 3200 rpm

As noted in the section following this on bearing loads, the load produced on the bearing is a function of eccentricity of the journal in the bearing - the greater the eccentricity, the greater the load. This is caused by the can annulus and motor magnetic pull forces which increase as the eccentricity increases. Therefore, as wear on the SiC bearing parts progress, the journal eccentricity increases. As eccentricity increases, the load increases and the wear rate increases, accelerating the total wear. This was accounted for in a Monte Carlo analysis which was performed to estimate the life of each bearing for a given allowable wear. A Monte Carlo analysis builds a large number of models by drawing variables at random from statistically described lots. The Monte Carlo analysis was performed via a BASIC program which performed the following:

1. The number of transfer pumps to be built by the computer program was selected. The number was 1000 pumps, for a good statistical sample.
2. The minimum Miller Number which could be encountered in any tank was selected. A minimum Miller Number of 15 was used for conservative results.
3. The maximum Miller Number was selected. The number used was 50 since this is the specification maximum.
4. The maximum allowable radial clearance due to wear was selected to evaluate bearing life. Generally the clearance selected varied between 0.002 and 0.012 inch.

5. The diameter of the bearing was selected. For the upper bearing 2.50 inches was used, and for the lower bearing 3.25 inches was used.
6. The length of the bearing was selected. For the upper bearing 2.50 inches was used, and for the lower bearing 2.30 inches was used.
7. The bearing location, upper or lower, was selected. This identified the can annulus, motor magnetic pull, hydraulic and unbalance load influence coefficients.
8. The hydraulic load direction, between zero and 360°, was randomly calculated.
9. The manufactured eccentricity of the bearing center relative to the stator can center (for can annulus loads) was randomly picked between 0.002 and 0.0075 inch. The 0.002 inch value is the minimum expected, and the 0.0075 inch value is the maximum allowed eccentricity per the drawings. The angle at which the eccentricity acts, between zero and 360°, was also randomly selected.
10. The manufactured eccentricity of the bearing center relative to the stator core center (for motor magnetic loads) was randomly picked between 0.002 and 0.0075 inch. The 0.002 inch value is the minimum expected, and the 0.0075 inch value is the maximum allowed eccentricity per the drawings. The angle at which the eccentricity acts, between zero and 360°, was also randomly selected.
11. The Miller Number for a particular tank hazardous waste fluid was randomly selected between the minimum and maximum values previously identified.

12. The concentration of alumina in a particular tank hazardous waste fluid was randomly selected between 0.001% weight and 0.006% weight. The 0.006% maximum value was selected because this concentration was used in Simulant #2, and was agreed upon as the worst case alumina concentration that would be encountered by transfer pumps.
13. For a particular hazardous waste tank, the specific gravity of the fluid to be pumped was randomly picked between 1.00 and 1.50, the specification lower and upper limits.
14. The motor magnetic load magnitude was calculated based on the eccentricity previously randomly selected.
15. The can annulus load was calculated based on the eccentricity and the specific gravity, which were both previously randomly selected.
16. The hydraulic load magnitude was calculated based on the specific gravity previously selected.
17. The unbalance force magnitude was calculated.
18. The can annulus, hydraulic and motor magnetic loads were combined by vector addition. The unbalance force was added to the resultant of the vector sum to get the net load on the bearing.
19. One of the six main head-flow-speed operating points, listed in Table 9-2, was selected at random for pumping fluid out of a particular waste tank.
20. The pump was operated for the amount of time required to pump one million gallons of fluid out of the tank, based on the flow rate previously selected.

21. For the given bearing (length and diameter) the radial clearance increase (wear) over the time required to pump one million gallons was calculated using the selected Miller Number, alumina concentration, operating speed, and net load on the bearing.
22. If the total bearing radial clearance increase exceeds the allowable radial clearance increase input, the total hours of pump operation is calculated, and the program is terminated. If total radial clearance increase does not exceed the allowable, the can annulus and motor magnetic eccentricities are increased by the calculated radial clearance increase, and the program goes back to step 11 to calculate additional wear associated with pumping out a million gallons of new fluid in a tank.

Table 9-2 - Transfer Pump Operating Points

<u>Head (feet)</u>	<u>Flow (gpm)</u>	<u>Speed (rpm)</u>
19	53	1462
19	132	1985
305	132	3324
394	53	3381
366	138	3549
450	140	3860

The results of the Monte Carlo wear analyses for both Transfer Pump SiC radial bearings are as follows in Tables 9-3 and 9-4 using a 1000 pump evaluation:

Table 9-3 - Upper Radial Bearing (2.50 inches diameter) Life

<u>Allowable Radial Increase or Wear (mils)</u>	<u>Minimum Life (hours)</u>	<u>Average Life (hours)</u>	<u>Maximum Life (hours)</u>	<u>10,000 Hour Probability (%)</u>
2.00	5,616	12,764	40,907	57.60
2.50	6,847	16,304	52,343	82.60
3.00	8,560	18,975	58,817	96.60
3.50	9,679	22,192	68,151	99.90
4.00	11,102	24,911	76,532	100.00

Table 9-4 - Lower Radial Bearing (3.25 inches diameter) Life

<u>Allowable Radial Increase or Wear (mils)</u>	<u>Minimum Life (hours)</u>	<u>Average Life (hours)</u>	<u>Maximum Life (hours)</u>	<u>10,000 Hour Probability (%)</u>
7.00	6,891	11,793	18,902	72.70
7.50	8,068	12,509	20,781	83.40
8.00	7,767	13,421	22,043	92.00
8.50	8,511	14,182	23,477	96.70
9.00	9,341	15,071	24,031	99.10
9.50	9,443	15,875	26,307	99.40
10.00	9,770	16,798	28,467	99.70
10.50	10,333	17,457	29,194	100.00

A 10,000 hour bearing wear life probability of 99+ percent is considered to be acceptable for the Transfer Pump application. Although some of the results in Tables 3 and 4 indicate 100 percent probability of exceeding 10,000 hours of life, a larger population sample, say 100,000 pumps, would probably reveal that a few of these pumps would not reach 10,000 hours. For the upper radial bearing, an allowable 0.004 inch increase in radial clearance due to wear will provide an average life of about 25,000 hours with a 99+ percent probability of exceeding 10,000 hours. For the lower radial bearing, an allowable 0.0105 inch increase in radial clearance will provide an average life of about 17,500 hours with a 99+ percent probability of exceeding 10,000 hours. Therefore, the upper radial bearing must be evaluated with a 0.014 inch diametral clearance (add twice the 0.004 inch radial wear to the design maximum diametral clearance of

0.006 inch), and the lower radial bearing must be evaluated with a 0.028 inch diametral clearance (add twice the 0.0105 inch radial wear to the design maximum diametral clearance of 0.007 inch). Note that the wear calculated is less than the wear assumed for the angular alignment capability calculations. Therefore, the alignment calculations are conservative.

Radial Bearing Loads

The following forces were considered to be acting on the rotating assembly:

- Hydraulic radial thrust from each of the two impellers
- Can annulus radial forces due to the rotor operating eccentric to the stator
- Motor magnetic pull radial forces due to the rotor core operating eccentric to the magnetic center of the motor
- Rotating assembly unbalance forces

Steady-state can annulus forces can be approximated by the following equation:

$$F_{ca} = \pi * \rho * sg * (R_{avg})^3 * L * (2 * \pi * N / 60)^2 * \Delta_{ca} / (4 * C)$$

where: F_{ca} is the can annulus force, lbs

ρ is the fluid density in a 1.00 specific gravity fluid, 9.355×10^{-5} lb-s²/in⁴

sg is the specific gravity of the fluid

R_{avg} is the average radius of the annular gap, 1.968 inches

C is the nominal can annulus radial gap, 0.142 inch

L is the effective can annulus length, 23.25 inches

N is the rotor rotational speed, rpm

Δ_{ca} is the radial eccentricity of the rotor can relative to the stator can, inch

$$F_{ca} = 1.005 \times 10^{-3} \cdot sg \cdot N^2 \cdot \Delta_{ca}$$

The center of the rotor can is 16.20 inches from the centerline of the upper bearing, and the bearing span is 34.13 inches. Therefore, the can annulus forces on the upper and lower bearings (WU_{ca} and WL_{ca} , respectively) are:

$$WU_{ca} = (1 - (16.20 / 34.13)) \cdot F_{ca}$$

$$WU_{ca} = 5.280 \times 10^{-4} \cdot sg \cdot N^2 \cdot \Delta_{ca}$$

$$WL_{ca} = (16.20 / 34.13) \cdot F_{ca}$$

$$WL_{ca} = 4.770 \times 10^{-4} \cdot sg \cdot N^2 \cdot \Delta_{ca}$$

Steady state motor forces can be approximated by the following equation:

$$F_{mp} = BG_{ave}^2 \cdot A \cdot \Delta_m / (72 \times 10^6 \cdot CK \cdot C_{sg})$$

$$A = \pi \cdot (ID_s - 2 \cdot C_{sg}) \cdot Lc$$

$$BG_{ave} = 2 * BG / \pi$$

where: F_{mp} is motor radial magnetic pull, lbs

BG_{ave} is the average air gap flux density, lines/in²

BG is the maximum design point air gap flux density, 30523 lines/in²

A is the rotor surface area, in²

CK is the number of parallel circuits/phase, 1

C_{ag} is the nominal radial air gap, 0.211 inch

Δ_m is the rotor radial eccentricity, inch

ID_s is the stator punching inside diameter, 4.150 inch

L is the active stator core length, 18.00 inches

$$F_{mp} = 5240 * \Delta_m$$

The center of the stator iron is 16.20 inches from the centerline of the upper bearing, and the bearing span is 34.13 inches. Therefore, the motor magnetic forces on the upper and lower bearings (WU_m and WL_m , respectively) are:

$$WU_m = (1 - (16.20 / 34.13)) * F_{mp}$$

$$WU_m = 2752 * \Delta_m$$

$$WL_m = (16.20 / 34.13) * F_{mp}$$

$$WL_m = 2487 * \Delta_m$$

Based on air testing the Transfer Pump hydraulics (see Section 5.0), the impeller hydraulic radial loads are estimated to be present at design head and flow conditions at the magnitudes identified in Table 9-5:

Table 9-5 - Design Point Impeller Hydraulic Radial Loads

	1.00 sg	1.50 sg
<u>Impeller</u>	<u>Fluid</u>	<u>Fluid</u>
1 st Stage	35 lbs	53 lbs
2 nd Stage	19 lbs	29 lbs

As indicated in Table 5, the hydraulic forces are directly proportional to the specific gravity of the pumped fluid and are also proportional to the square of the operating speed. The air testing indicated that the hydraulic load vectors for the first and second stage impellers were 42° apart; for a conservative calculation for bearing loads it will be assumed that the hydraulic load vectors are in-line. The following equations therefore approximate the radial hydraulic loads at the impellers:

$$F_{hy1} = 35 * sg * (N / 3860)^2$$

$$F_{hy2} = 19 * sg * (N / 3860)^2$$

where: F_{hy1} is the radial hydraulic load at the first stage impeller, lbs

F_{hy2} is the radial hydraulic load at the second stage impeller, lbs

sg is the specific gravity of the fluid

N is the operating speed of the pump, rpm

Since the bearing span is 34.13 inches, the distance from the lower bearing centerline to the #1 impeller discharge centerline is 10.89 inches, and the distance from the lower bearing centerline to the #2 impeller discharge centerline is 5.78 inches, the hydraulic loads at the upper and lower radial bearings (WU_{hy} and WL_{hy} , respectively) are:

$$WU_{hy} = (10.89 / 34.13) * F_{hy1} + (5.78 / 34.13) * F_{hy2}$$

$$WU_{hy} = 9.655 \times 10^{-7} * sg * N^2$$

$$WL_{hy} = (1 + (10.89 / 34.13)) * F_{hy1} + (1 + (5.78 / 34.13)) * F_{hy2}$$

$$WL_{hy} = 4.590 \times 10^{-6} * sg * N^2$$

The unbalance forces were calculated using the commercial software "TKSolver". A balance program, "TRANRAUB.TKW", was written using TKSolver and is enclosed in this section. The program contains all equations used in the analysis. Dynamic amplification is not accounted for in this static analysis, but it is accounted for in the Section 10.0 dynamic analysis.

The weights used to calculate the unbalance forces on the rotating assembly radial bearings are as follows:

- Weight of the rotor assembly, 115.0 lbs
- Weight of the thrust runner, 10.0 lbs

- Weight of the second stage impeller, 13.2 lbs
- Weight of the first stage impeller, 12.0 lbs
- Weight of the impeller sleeve, 1.0 lb
- Weight of the impeller bolt, 0.45 lb

The allowable residual unbalance of each of the components is as follows:

- The rotor assembly is allowed 1.67 g-inches at the upper bearing and 1.71 g-inches at the lower bearing per API 610, but only 0.46 g-inch is allowed at each bearing per the manufacturing drawing.
- API 610 allows 0.29 g-inch unbalance at the thrust runner, but the manufacturing drawing allows only 0.25 g-inch.
- API 610 allows 0.39 g-inch unbalance at the second stage impeller , but the manufacturing drawing allows only 0.26 g-inch.
- The first stage impeller is allowed 0.35 g-inch unbalance per API 610, but the manufacturing drawing allows only 0.25 g-inch.
- The sleeve unbalance is assumed to be a maximum of 0.50 g-inch.
- The impeller bolt unbalance is assumed to be a maximum of 0.01 g-inch.

The TIR's of the shaft sections at the impellers and thrust runner are assumed to be within 0.001 inch relative to the bearing journals. The thrust runner and impellers are assumed to have a 0.001 inch diametral clearance on the shaft.

Based on API 610 balance guidelines, the maximum rotating assembly residual unbalance forces should be 1.35 lbs at 3860 rpm (1.45 g-inch) for the upper bearing and 2.80 lbs at 3860 rpm (3.01 g-inch) for the lower radial bearing. The TRANSRAUB.TKW analysis of the rotating assembly indicates that the residual unbalance at the upper radial bearing is a maximum of 4.45 g-inches (assuming a worst case tolerance stack-up of balanced parts) which would produce a force of 4.15 lbs at 3860 rpm on the upper bearing (use 4.0 lbs). The analysis also revealed that the maximum residual unbalance at the lower radial bearing is 23.48 g-inches which would produce a force of 21.89 lbs at 3860 rpm on the lower radial bearing (use 24.0 lbs). The unbalance forces are proportional to speed squared.

The equations governing the unbalance forces on the upper and lower bearings (WU_{ub} and WL_{ub} , respectively) simplify to:

$$WU_{ub} = 4 * (N / 3860)^2$$

$$WU_{ub} = 2.685 \times 10^{-7} * N^2$$

$$WL_{ub} = 24 * (N / 3860)^2$$

$$WL_{ub} = 1.611 \times 10^{-6} * N^2$$

St	Input	Name	Output	Unit	Comment
					TRANRAUB.TK
					Transfer Pump Rotating Assy Unbalance

					Rotor Assembly Allowable Unbalance

	115.00	Wr		lbs	Rotor weight
		Wru	56.84	lbs	Rotor upper bearing weight
		Wrl	58.16	lbs	Rotor lower bearing weight
		UBrua	3.68E-3	lb*in	Rotor allowable unbalance at upper brg
		UBrug	1.67	g*in	Rotor allowable unbalance at upper brg
		UBrla	3.77E-3	lb*in	Rotor allowable unbalance at lower brg
		UBrlg	1.71	g*in	Rotor allowable unbalance at lower brg
		Frua	1.56	lbs	Rotor allowable force at upper brg
		Frla	1.59	lbs	Rotor allowable force at lower brg

					Thrust Runner Allowable Unbalance

	10.00	Wtr		lbs	Thrust runner weight
		UBtra	6.48E-4	lb*in	Thrust runner allowable unbalance
		UBtrag	0.29	g*in	Thrust runner allowable unbalance
		Ftra	0.27	lbs	Thrust runner allowable balance force

					Impeller #2 Allowable Unbalance

	13.20	Wl2		lbs	Impeller #2 weight
		UBl2a	8.55E-4	lb*in	Impeller #2 allowable unbalance
		UBl2ag	0.39	g*in	Impeller #2 allowable unbalance

<u>St</u>	<u>Input</u>	<u>Name</u>	<u>Output</u>	<u>Unit</u>	<u>Comment</u>
		Fi2a	0.36	lbs	Impeller #2 allowable unbalance force
<hr/>					
Impeller #1 Allowable Unbalance					
<hr/>					
12.00	W1			lbs	Impeller #1 weight
		UBi1a	7.77E-4	lb*in	Impeller #1 allowable unbalance
		UBi1ag	0.35	g*in	Impeller #1 allowable unbalance
		Fi1a	0.33	lbs	Impeller #1 allowable unbalance force
<hr/>					
Sleeve Allowable Unbalance					
<hr/>					
1.00	Ws			lbs	Sleeve weight
		UBsa	6.48E-5	lb*in	Sleeve allowable unbalance
		UBsag	0.03	g*in	Sleeve allowable unbalance
		Fsa	0.03	lbs	Sleeve allowable unbalance force
<hr/>					
Impeller Bolt Allowable Unbalance					
<hr/>					
0.45	Wb			lbs	Impeller bolt weight
		UBba	2.91E-5	lb*in	Impeller bolt allowable unbalance
		UBbag	0.01	g*in	Impeller bolt allowable unbalance
		Fba	0.01	lbs	Bolt allowable unbalance force
<hr/>					
Rotating Assembly Allowable Unbalance					

SI	Input	Name	Output	Unit	Comment
		Wra	151.65	lbs	Rotating assembly weight
		Wrau	49.40	lbs	Rotating assembly upper bearing weight
		Wral	102.25	lbs	Rotating assembly lower bearing weight
		UBraua	3.20E-3	lb*in	Rotating asm allow unbal at upper brg
		UBrauag	1.45	g*in	Rotating asm allow unbal at upper brg
		UBraia	6.62E-3	lb*in	Rotating asm allow unbal at lower brg
		UBraieg	3.01	g*in	Rotating asm allow unbal at lower brg
		Fraua	1.35	lbs	Rotating asm allow force at upper brg
		Fraia	2.80	lbs	Rotating asm allow force at lower brg
<hr/>					
Rotor Assembly Actual Unbalance					
		UBru	1.01E-3	lb*in	Rotor mfg unbalance at upper brg
0.46		UBrug		g*in	Rotor mfg unbalance at upper brg
		UBrl	1.01E-3	lb*in	Rotor mfg unbalance at lower brg
0.46		UBrlg		g*in	Rotor mfg unbalance at lower brg
		Fru	0.43	lbs	Rotor mfg force at upper brg
		Frl	0.43	lbs	Rotor mfg force at lower brg
<hr/>					
Thrust Runner Actual Unbalance					
		UBtr	5.51E-4	lb*in	Thrust runner manufactured unbalance
0.25		UBtrg		g*in	Thrust runner manufactured unbalance
0.0010		TIRtrs		in	TIR runner shaft to rotor bearings
0.0010		DCtrs		in	Diametral clearance, shaft to runner
		Ftr	4.47	lbs	Assembled thrust runner mfg unbal force
		Ftir	2.12	lbs	Runner mfg unbal force from TIR
		Ftrdc	2.12	lbs	Runner mfg unbal force from dia clear

St	Input	Name	Output	Unit	Comment
		Ftru	0.35	lbs	Runner mfg unbal force at upper brg
		Ftrl	4.82	lbs	Runner mfg unbal force at lower brg
<hr/>					
Impeller #2 Actual Unbalance					
<hr/>					
		UBi2	5.73E-4	lb*in	Impeller #2 manufactured unbalance
	0.26	UBi2g		g*in	Impeller #2 manufactured unbalance
	0.0010	TIRi2s		in	TIR impeller #2 shaft to rotor bearings
	0.0010	DCi2s		in	Diametral clearance, shaft to imp #2
		Fi2	5.83	lbs	Assembled impeller #2 mfg unbal force
		Fi2u	1.04	lbs	Imp #2 mfg unbal force at upper brg
		Fi2l	6.87	lbs	Imp #2 mfg unbal force at lower brg
<hr/>					
Impeller #1 Actual Unbalance					
<hr/>					
		UBi1	5.51E-4	lb*in	Impeller #1 manufactured unbalance
	0.25	UBi1g		g*in	Impeller #1 manufactured unbalance
	0.0010	TIRi1s		in	TIR impeller #1 shaft to rotor bearings
	0.0010	DCi1s		in	Diametral clearance, shaft to imp #1
		Fi1	5.31	lbs	Assembled impeller #1 mfg unbal force
		Fi1u	1.74	lbs	Imp #1 mfg unbal force at upper brg
		Fi1l	7.05	lbs	Imp #1 mfg unbal force at lower brg
<hr/>					
Sleeve Actual Unbalance					
<hr/>					
		UBs	1.10E-3	lb*in	Sleeve manufactured unbalance
	0.50	UBsg		g*in	Sleeve manufactured unbalance

St	Input	Name	Output	Unit	Comment
	0.0010	TIRss		in	TIR sleeve shaft to rotor bearings
	0.0010	DCss		in	Diametral clearance, shaft to sleeve
		Fs	0.89	lbs	Assembled sleeve mfg unbal force
		Fsu	0.20	lbs	Sleeve mfg unbal force at upper brg
		Fsl	1.09	lbs	Sleeve mfg unbal force at lower brg

Impeller Bolt Actual Unbalance

		UBb	2.20E-5	lb*in	Bolt manufactured unbalance
	0.01	UBbg		g*in	Bolt manufactured unbalance
	0.0100	TIRbs		in	TIR bolt shaft to rotor bearings
	0.0030	DCbs		in	Diametral clearance, shaft to bolt
		Fb	1.25	lbs	Assembled bolt mfg unbal force
		Fbu	0.38	lbs	Bolt mfg unbal force at upper brg
		Fbl	1.63	lbs	Bolt mfg unbal force at lower brg

Rotating Asm Max Assembled Unbalance

		UBrau	9.80E-3	lb*in	Mfg rotating assy unbalance upper brg
		UBraug	4.45	g*in	Mfg rotating assy unbalance upper brg
		UBrel	5.17E-2	lb*in	Mfg rotating assy unbalance lower brg
		UBraig	23.48	g*in	Mfg rotating assy unbalance lower brg
		Frau	4.15	lbs	Rotating asm max force at upper brg
		Fral	21.89	lbs	Rotating asm max force at lower brg

Rotating Assembly Speed/Dimensions

3860	Nmax			rpm	Maximum pump speed
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<u>St</u>	<u>Input</u>	<u>Name</u>	<u>Output</u>	<u>Unit</u>	<u>Comment</u>
	3860	N		rpm	Operating speed of interest
	34.130	L		in	Bearing span
	17.260	Lru		in	Upper brg centerline to rotor asm CG
	2.700	Ltr		in	Thrust runner CG overhang
	6.080	Li2		in	Impeller #2 CG overhang
	7.700	Ls		in	Sleeve CG overhang
	10.520	Lb		in	Impeller bolt CG overhang
	11.190	Li1		in	Impeller #1 CG overhang
		Lrau	23.012	in	Upper brg centerline to rotating asm CG

S Rule

;Rotor assembly allowable unbalance calculations

$$Wr = Wru + Wrl$$

$$Wrl = Lru \cdot Wrl$$

$$UBrua = 4 \cdot Wru / (16 \cdot N_{max})$$

$$UBrla = 4 \cdot Wrl / (16 \cdot N_{max})$$

$$UBrug = 454 \cdot UBrua$$

$$UBrlg = 454 \cdot UBrla$$

$$Frua = UBrua \cdot ((\pi) \cdot N)^2 / (900 \cdot 386)$$

$$Frla = UBrla \cdot ((\pi) \cdot N)^2 / (900 \cdot 386)$$

;Thrust runner allowable unbalance calculations

$$UBtra = 4 \cdot Wtr / (16 \cdot N_{max})$$

$$UBtrg = 454 \cdot UBtra$$

$$Ftra = UBtra \cdot ((\pi) \cdot N)^2 / (900 \cdot 386)$$

;Impeller #2 allowable unbalance calculations

$$UBi2a = 4 \cdot Wi2 / (16 \cdot N_{max})$$

$$UBi2g = 454 \cdot UBi2a$$

$$Fi2a = UBi2a \cdot ((\pi) \cdot N)^2 / (900 \cdot 386)$$

;Impeller #1 allowable unbalance calculations

$$UBi1a = 4 \cdot Wi1 / (16 \cdot N_{max})$$

$$UBi1g = 454 \cdot UBi1a$$

$$Fi1a = UBi1a \cdot ((\pi) \cdot N)^2 / (900 \cdot 386)$$

;Sleeve allowable unbalance calculations

$$UBsa = 4 \cdot Ws / (16 \cdot N_{max})$$

$$UBsag = 454 \cdot UBsa$$

$$Fsa = UBsa \cdot ((\pi) \cdot N)^2 / (900 \cdot 386)$$

;Impeller bolt allowable unbalance calculations

$$UBba = 4 \cdot Wb / (16 \cdot N_{max})$$

$$UBbag = 454 \cdot UBba$$

$$Fba = UBba \cdot ((\pi) \cdot N)^2 / (900 \cdot 386)$$

;Rotating assembly allowable unbalance calculations

$$Wra = Wr + Wtr + Wi2 + Wi1 + Ws + Wb$$

$$Wra = Wru + Wrl$$

S Rule

$$Wrai = Lrau * Wra / L$$

$$UBraua = 4 * Wra / (16 * Nmax)$$

$$UBraia = 4 * Wrai / (16 * Nmax)$$

$$UBraug = 454 * UBraua$$

$$UBraig = 454 * UBraia$$

$$Fraus = UBraua * ((\pi() * N)^2) / (900 * 386)$$

$$Frai = UBraia * ((\pi() * N)^2) / (900 * 386)$$

;Rotor manufactured unbalance calculations

$$UBrug = 454 * UBr$$

$$UBrig = 454 * UBri$$

$$Fru = UBr * ((\pi() * N)^2) / (900 * 386)$$

$$Fri = UBri * ((\pi() * N)^2) / (900 * 386)$$

;Thrust runner manufactured unbalance calculations

$$UBtrg = 454 * UBtr$$

$$Ftr = (UBtr + Wtr * (0.5 * TIRtrs + 0.5 * DCtrs)) * ((\pi() * N)^2) / (900 * 386)$$

$$Ftir = 0.5 * Wtr * TIRtrs * ((\pi() * N)^2) / (900 * 386)$$

$$Ftrdc = 0.5 * Wtr * DCtrs * ((\pi() * N)^2) / (900 * 386)$$

$$Ftrl = Ftr * (L + Ltr) / L$$

$$Ftru = Ftr * Ltr / L$$

;Impeller #2 manufactured unbalance calculations

$$UBi2g = 454 * UBi2$$

$$Fi2 = (UBi2 + Wi2 * (0.5 * TIRi2s + 0.5 * DCi2s)) * ((\pi() * N)^2) / (900 * 386)$$

$$Fi2l = Fi2 * (L + Li2) / L$$

$$Fi2u = Fi2 * Li2 / L$$

;Impeller #1 manufactured unbalance calculations

$$UBi1g = 454 * UBi1$$

$$Fi1 = (UBi1 + Wi1 * (0.5 * TIRi1s + 0.5 * DCi1s)) * ((\pi() * N)^2) / (900 * 386)$$

$$Fi1l = Fi1 * (L + Li1) / L$$

$$Fi1u = Fi1 * Li1 / L$$

;Sleeve manufactured unbalance calculations

$$UBsg = 454 * UBs$$

$$Fsg = (UBs + Ws * (0.5 * TIRss + 0.5 * DCss)) * ((\pi() * N)^2) / (900 * 386)$$

$$Fsl = Fsg * (L + Ls) / L$$

S Rule

$$F_{su} = F_s \cdot L_s / L$$

;Impeller bolt manufactured unbalance calculations

$$UB_{bg} = 454 \cdot UB_b$$

$$F_b = (UB_b + W_b \cdot (0.5 \cdot TIR_{bs} + 0.5 \cdot DC_{bs})) \cdot ((\pi \cdot N)^2) / (900 \cdot 386)$$

$$F_{bl} = F_b \cdot (L + L_b) / L$$

$$F_{bu} = F_b \cdot L_b / L$$

;Rotating assembly maximum assembled unbalance calculations

$$F_{rau} = F_{ru} + F_{rlu} + F_{i2u} + F_{i1u} + F_{su} + F_{bu}$$

$$F_{ral} = F_{rl} + F_{rlu} + F_{i2l} + F_{i1l} + F_{sl} + F_{bl}$$

$$F_{rau} = UB_{rau} \cdot ((\pi \cdot N)^2) / (900 \cdot 386)$$

$$F_{ral} = UB_{ral} \cdot ((\pi \cdot N)^2) / (900 \cdot 386)$$

$$UB_{rau} = 454 \cdot UB_{rau}$$

$$UB_{ral} = 454 \cdot UB_{ral}$$

;Dimensional relationships

$$L_{rau} = (W_r \cdot L_{ru} + W_r \cdot (L + L_r) + W_{i2} \cdot (L + L_{i2}) + W_{i1} \cdot (L + L_{i1}) + W_s \cdot (L + L_s) + W_b \cdot (L + L_b)) / W_{ra}$$

To determine the maximum loads on each bearing for conservative minimum film thickness calculations, the hydraulic forces, can annulus forces, motor magnetic forces and unbalance forces are directly summed to obtain the radial bearing loads. The bearings centers are assumed to have a manufactured radial eccentricity of 0.010 inch from the stator bore. The eccentricities of the journals in the bearings are then added directly to the initial offset to obtain the total radial offset for can annulus and motor magnet loads. Therefore the radial offsets for can annulus and motor magnetic loads are:

$$\Delta_{ca} = \Delta_m = \epsilon + 0.010$$

where: ϵ is the offset of the journal in the bearing, inch

The total upper and lower radial bearing loads (W_U and W_L , respectively) are given by the equations:

$$W_U = W_{U_{ca}} + W_{U_m} + W_{U_{hy}} + W_{U_{ub}}$$

$$W_U = \{5.280 \times 10^{-4} * sg * N^2 * (\epsilon + 0.010)\} + \{2752 * (\epsilon + 0.010)\} \\ + \{9.655 \times 10^{-7} * sg * N^2\} + \{2.685 \times 10^{-7} * N^2\}$$

$$W_L = W_{L_{ca}} + W_{L_m} + W_{L_{hy}} + W_{L_{ub}}$$

$$W_L = \{4.770 \times 10^{-4} * sg * N^2 * (\epsilon + 0.010)\} + \{2487 * (\epsilon + 0.010)\} \\ + \{4.590 \times 10^{-6} * sg * N^2\} + \{1.611 \times 10^{-6} * N^2\}$$

The bearing load evaluation covers both upper and lower radial bearings over a speed range of 1280 to 3860 rpm, at the minimum and maximum specific gravity conditions. The loads on the upper and radial bearings under these conditions are identified in Table 9-6.

Table 9-6 - Radial Bearing Loads as a Function of Journal Eccentricity
(ϵ in inches)

<u>Speed (rpm)</u>	<u>Specific Gravity</u>	<u>WL</u> <u>Upper Bearing Load (lbs)</u>	<u>WU</u> <u>Lower Bearing Loads (rpm)</u>
1280	1.00	38 + (3619 * ϵ)	43 + (3272 * ϵ)
2570	1.00	71 + (6242 * ϵ)	98 + (5645 * ϵ)
3860	1.00	125 + (10624 * ϵ)	189 + (9607 * ϵ)
1280	1.50	43 + (4052 * ϵ)	51 + (3663 * ϵ)
2570	1.50	92 + (7988 * ϵ)	128 + (7222 * ϵ)
3860	1.50	172 + (14560 * ϵ)	258 + (13165 * ϵ)

Radial Bearing Performance Characteristics

The pivoted pad radial bearings were analyzed with a proprietary and verified program called TILTPAD. TILTPAD consists of two subprograms, STILT and ASEMB. For a particular pivoted pad radial bearing TILTPAD produces the static load, the journal eccentricity, the minimum film thickness, the static pressure distribution on the pads, and the dynamic stiffness and damping values. The subprogram STILT generates performance of a single tilting pad operating with incompressible fluid in either the laminar or turbulent regime. Output of STILT is used as the input to the subprogram ASEMB to develop performance of multi-pad pivoted pad radial bearings.

Both the upper and lower radial bearings were analyzed at the nominal machined dimensions and the maximum wear dimensions identified in the above section. For both the nominal and maximum wear dimensions, both bearings were analyzed at the low viscosity (1.00 centipoise), low specific gravity (1.00) condition and the high viscosity (30.0 centipoise), high specific gravity (1.50) condition. To determine the maximum wear dimensions of the

upper pivoted pad radial bearing, the following was performed to simulate the 0.004 inch total radial wear expected over 10,000 hours of operation: 1) the journal diameter was decreased by 0.004 inch from its minimum material condition and 2) 0.002 inch of material was removed from the pads, increasing the maximum pivot diameter by 0.004 inch and the pad maximum machined diameter by 0.004 inch. This results in an upper bearing whose diametral clearance has increased from 0.006 inch to 0.014 inch (at the pivot pins) and whose preload ratio has increased from 0.500 to 0.718, approaching a concentric bearing with a preload ratio of 1.00.

To determine the maximum wear dimensions of the lower pivoted pad radial bearing, the following was performed to simulate the 0.0105 inch total radial wear expected over 10,000 hours of operation: 1) the journal diameter was decreased by 0.0105 inch from its minimum material condition and 2) 0.00525 inch of material was removed from the pads, increasing the maximum pivot diameter by 0.0105 inch and the pad maximum machined diameter by 0.0105 inch. This results in an upper bearing whose diametral clearance has increased from 0.007 inch to 0.028 inch (at the pivot pins) and whose preload ratio has increased from 0.500 to 0.812, approaching a concentric bearing with a preload ratio of 1.00.

The results of the analyses are presented below in Tables 9-7 through 9-14. Tables 9-7 through 9-10 describe results for the upper radial bearing, and Tables 9-11 through 9-14 delineate results for the lower radial bearing. For the upper radial bearing, Tables 9-7 and 9-8 provide performance data on the nominal bearing in a low viscosity, low specific gravity fluid and a high viscosity, high specific gravity fluid. Tables 9-9 and 9-10 provide performance data on the upper radial bearing at the maximum wear dimensional conditions in a low viscosity, low specific gravity fluid and a high viscosity, high specific gravity fluid. For the lower radial bearing, Tables 9-11 and 9-12 provide performance data on the nominal bearing in a low viscosity, low specific gravity fluid and a high viscosity, high specific gravity fluid. Tables 9-13 and 9-14 provide performance data on the lower radial bearing at the maximum wear dimensional conditions in

a low viscosity, low specific gravity fluid and a high viscosity, high specific gravity fluid. All film thickness values calculated are acceptable. At end of bearing life (maximum wear) the minimum film thickness is 0.29 mil for the upper bearing in water and 0.17 mil for the lower bearing in water. The minimum end of life film thickness for SiC bearings is considered to be 0.10 mil.

Table 9-7 - Upper Radial Bearing (2.50 Dia x 2.50 L x 0.0055 DC with 0.500 Preload Ratio)

1.00 Centipoise Viscosity and 1.00 Specific Gravity

<u>Speed (rpm)</u>	<u>Load (lbs)</u>	<u>Film (mils)</u>	<u>Horsepower</u>	<u>Pad Flow (gpm)</u>
1280	48	0.58	0.008	0.12
2570	87	0.64	0.029	0.27
3860	153	0.60	0.080	0.39

Table 9-8 - Upper Radial Bearing (2.50 Dia x 2.50 L x 0.0055 DC with 0.500 Preload Ratio)

30.0 Centipoise Viscosity and 1.50 Specific Gravity

<u>Speed (rpm)</u>	<u>Load (lbs)</u>	<u>Film (mils)</u>	<u>Horsepower</u>	<u>Pad Flow (gpm)</u>
1280	44	2.23	0.106	0.68
2570	95	2.22	0.429	1.36
3860	178	2.17	0.972	2.03

Table 9-9 - Upper Radial Bearing (2.50 Dia x 2.50 L x 0.0140 DC with 0.718 Preload Ratio)

1.00 Centipoise Viscosity and 1.00 Specific Gravity

<u>Speed (rpm)</u>	<u>Load (lbs)</u>	<u>Film (mils)</u>	<u>Horsepower</u>	<u>Pad Flow (gpm)</u>
1280	70	0.26	0.012	0.12
2570	125	0.31	0.046	0.28
3860	218	0.29	0.122	0.42

Table 9-10 - Upper Radial Bearing (2.50 Dia x 2.50 L x 0.0140 DC with 0.718 Preload Ratio)

30.0 Centipoise Viscosity and 1.50 Specific Gravity

<u>Speed (rpm)</u>	<u>Load (lbs)</u>	<u>Film (mils)</u>	<u>Horsepower</u>	<u>Pad Flow (gpm)</u>
1280	62	3.22	0.060	0.47
2570	130	3.15	0.245	0.93
3860	248	2.81	0.592	1.29

Table 9-11 - Lower Radial Bearing (3.25 Dia x 2.30 L x 0.0065 DC with 0.500 Preload Ratio)

1.00 Centipoise Viscosity and 1.00 Specific Gravity

<u>Speed (rpm)</u>	<u>Load (lbs)</u>	<u>Film (mils)</u>	<u>Horsepower</u>	<u>Pad Flow (gpm)</u>
1280	53	0.77	0.012	0.23
2570	115	0.78	0.061	0.49
3860	219	0.71	0.169	0.71

Table 9-12 - Lower Radial Bearing (3.25 Dia x 2.30 L x 0.0065 DC with 0.500 Preload Ratio)

30.0 Centipoise Viscosity and 1.50 Specific Gravity

<u>Speed (rpm)</u>	<u>Load (lbs)</u>	<u>Film (mils)</u>	<u>Horsepower</u>	<u>Pad Flow (gpm)</u>
1280	52	2.61	0.192	1.02
2570	131	2.56	0.774	2.05
3860	265	2.47	1.758	3.07

Table 9-13 - Lower Radial Bearing (3.25 Dia x 2.30 L x 0.0280 DC with 0.812 Preload Ratio)

1.00 Centipoise Viscosity and 1.00 Specific Gravity

<u>Speed (rpm)</u>	<u>Load (lbs)</u>	<u>Film (mils)</u>	<u>Horsepower</u>	<u>Pad Flow (gpm)</u>
1280	104	0.22	0.024	0.29
2570	201	0.21	0.114	0.74
3860	363	0.17	0.316	1.16

Table 9-14 - Lower Radial Bearing (3.25 Dia x 2.30 L x 0.0280 DC with 0.812 Preload Ratio)

30.0 Centipoise Viscosity and 1.50 Specific Gravity

<u>Speed (rpm)</u>	<u>Load (lbs)</u>	<u>Film (mils)</u>	<u>Horsepower</u>	<u>Pad Flow (gpm)</u>
1280	102	3.27	0.104	0.78
2570	231	3.05	0.441	1.50
3860	454	2.58	1.245	2.10

As noted in Table 9-8 for the upper radial bearing and Table 9-12 for the lower radial bearing, at the minimum operating clearance, maximum viscosity/specific gravity condition, the bearing frictional horsepower to pad flow ratio is a maximum. This provides the maximum lubricant temperature rise for each bearing. Because the loads on the lower radial bearing are higher than the loads on the upper radial bearing, and because the frictional power to pad flow ratios for the lower radial bearing are greater than for the upper radial bearing ,

the lower radial bearing is considered limiting relative to bulk temperature rise. To evaluate the maximum bulk temperature rise in the lower pivoted pad radial bearing, the following assumptions were made:

- The viscosity was 30 centipoise (4.35×10^{-6} Reyn), the maximum expected in the hazardous waste tanks.
- The specific gravity was 1.50
- The fluid specific heat was 0.55 BTU/lb_m-°F, the minimum expected in the hazardous waste tanks.

The following equations were used:

$$Q_{btu} = m * Cp * \Delta T$$

$$m = 8.345 * sg * GPM$$

$$Q_{hp} = 0.02356 * Q_{btu}$$

where: Q_{btu} is the frictional heat generated in the bearing, BTU/minute

Q_{hp} is the frictional heat generated, horsepower

m is the mass flow rate, lb/minute

Cp is the specific heat of the fluid, 0.55 BTU/lb-°F

sg is the specific gravity of the fluid, 1.50

ΔT is the temperature rise, °F

GPM is the flow rate through the bearing pad, gpm

The lower bearing frictional power generation and pad flows shown in Table 9-12 were input into the above equations. The results of the analysis are presented in Table 9-15 for the lower radial bearing.

Table 9-15 - Lower Radial Bearing (3.25 Dia x 2.30 L x 0.0065 DC with 0.500 Preload Ratio)

<u>30.0 cP and 1.50 sq</u>				
<u>Speed (rpm)</u>	<u>Load (lbs)</u>	<u>Horsepower</u>	<u>Pad Flow (gpm)</u>	<u>Temperature Rise (°F)</u>
1280	52	0.192	1.02	1.16
2570	131	0.774	2.05	2.33
3860	265	1.758	3.07	3.53

The lower radial bearing has a maximum bulk fluid temperature rise of only 3.5 °F at design conditions when the bearing clearances are a minimum. At all other operating conditions the bulk temperature rise will be less. The bulk temperature rise in the upper radial bearing will be less than the temperature rise in the lower bearing because power to flow ratios are lower. The maximum fluid temperature rise will be higher than the bulk temperature rise. However, since the maximum bulk temperature rise is so small, the maximum fluid temperature rise will not be a problem. The upper and lower pivoted pad bearings are acceptable from a temperature rise and film thickness viewpoint.

9.2 Thrust Bearing

The thrust bearing is a film riding hydrodynamic bearing which is cooled and lubricated by the waste tank process fluid which is pumped through the motor. The thrust bearing is a single-acting (downward) Kingsbury-type bearing with links and pins for self-alignment with the rotating thrust runner. The thrust bearing has SiC thrust shoes set in 410 stainless steel holders. The SiC thrust shoes are attached to the holders with Loctite and are prevented from sliding out of the holders by mechanical interference. The thrust runner contains a SiC thrust runner insert

attached to a 410 stainless steel runner holder with Loctite. The runner holder and SiC runner insert are both close fitting with the shaft and are both keyed to the shaft. The SiC used in the thrust bearing is the direct sintered alpha form discussed in the radial bearing evaluation.

The thrust bearing has six shoes and six sets of links and pins. The working inside radius of the SiC shoe insert is 1.680 inches, the pivot radius is 2.465 inches, and the working outside radius is 3.260 inches. The SiC shoe insert has a tangential cylindrical crown with a nominal radius of curvature of 1900 inches (0.00041 inch crown at the pivot radius). The effective pad sector angle, at the 2.465 inch radius is 53.6° (the pad is a 60° sector with a 0.150 inch kerf). The thrust bearing assembly is held in place by six radial pins located between the shoes. The nominal clearance between the pins and the shoes is 0.050 inch.

Thrust Bearing Misalignment Potential

The static angular misalignment in the thrust bearing is caused by the following out-of-perpendicular conditions between the radial bearings and thrust bearing which tilt the shaft rotational axis:

- The static angular misalignment of the upper radial bearing relative to the lower radial bearing. In the radial bearing section this was calculated to be a maximum of 1.99×10^{-3} radian in the fully worn condition.
- A 0.002 inch parallelism between the lower radial bearing seat and the thrust bearing seat. This is an angular misalignment of 0.25×10^{-3} radian over the 8.00 inch thrust bearing fit diameter ($0.002 / 8 = 0.25 \times 10^{-3}$).
- An estimated 0.016 inch maximum possible stack-up difference between links, pins and shoes in the thrust bearing. Over the 6.52 inch working diameter of the thrust bearing, this is an angular misalignment of 2.45×10^{-3} radian.

The maximum static angular misalignment between the rotating assembly and the thrust bearing is the direct sum of these angular misalignments, or:

$$AM_{tb} = (1.99 + 0.25 + 2.45) \times 10^{-3}$$

$$AM_{tb} = 4.69 \times 10^{-3} \text{ radian}$$

A three dimensional tolerance stack-up analysis was performed on the Transfer Pump using a Monte Carlo technique. Using the runouts and clearances identified above, the tolerance stack-up analysis predicts a 95 percent probability that the thrust bearing angular misalignment will not exceed 3.41×10^{-3} radian and a 99 percent probability that the misalignment will not exceed 3.89×10^{-3} radian. Based on the above analyses, the Kingsbury-type thrust bearing self-alignment feature should be capable of correcting for an angular misalignment of no less than 4.70×10^{-3} radian.

Thrust Bearing Misalignment Capability

The thrust shoes are held in place by radial pins located between the shoes. The nominal clearance between the pins is 0.050 inch. Assume with tolerance stack-ups the minimum clearance is 0.035 inch. The minimum misalignment capability of a shoe is the ability to tilt tangentially until the outside corner of the shoe contacts a pin. The chordal length of a shoe at the outside diameter corners is 3.20 inches, nominal. One-half the chordal length is 1.60 inches. A tilt of 0.035 inch over a distance of 1.60 inches, from the shoe radial centerline to an outside corner is 21.88×10^{-3} radian ($0.035 / 1.60 = 21.88 \times 10^{-3}$). This is more than 4-½ times the 4.70×10^{-3} radian required misalignment capability.

The shoes and links are supported on pins with spherical heads. The spherical heads allow the links and pins to pivot. The pins must be sized so the shoes and links do not "roll over" the edge of the spherical head. To achieve a 4.70×10^{-3} radian misalignment capability, the pin spherical radii are sized as follows:

$$d = 2 * R * \sin \Theta$$

or,

$$R = d / (2 * \sin \Theta)$$

where: d is the diameter of the head of the pin, inch

R is the spherical radius, inch

Θ is the misalignment angle, radian

Table 9-16 lists the maximum allowable spherical radii to achieve 4.70×10^{-3} radian misalignment capability for various pin sizes:

Table 9-16 - Allowable Spherical Radii for 4.70×10^{-3} Radian Misalignment

<u>Pin Diameter (inch)</u>	<u>Maximum Spherical Radii (inch)</u>
0.250	26
0.375	39
0.500	53
0.625	66
0.750	79
0.875	93
1.000	106

Wear of Thrust Bearing SiC Material

Although it is reasonably straight forward to quantitatively estimate wear due to abrasive material in radial bearing lubricants, the task is more difficult for thrust bearings. There are an infinite number of ways in which wear could occur on

thrust bearing shoes and runners. Thrust bearing wear must be evaluated predominantly in a qualitative manner, considering the major failure modes.

Centrally pivoted thrust shoes are extremely tolerant to crown profile, if it is convex. A convex crown of even 0.00005 to 0.0001 inch is generally enough to generate sufficient film to prevent failure. On the other hand, a concave shoe surface profile will cause bearing failure. Therefore, thrust bearing failure due to abrasive wear can occur under the following circumstances:

- Sufficient shoe material is worn away to cause structural failure.
- The shoe operating surface is worn perfectly flat (better than 0.00005 inch)
- The runner surface is worn so rough that a hydrodynamic film cannot be developed.
- The shoe operating surface is worn concave.

The bearing abrasive testing performed at WSTC and the analysis performed on the radial bearings indicates that the following wear could occur on the SiC thrust bearing shoes and runner using the wear equation developed for the radial bearings:

$$W = (1.33 \times 10^{-5} * M + 0.0296 * C_{\text{alumina}}) * (\pi * D * N * P / 500000) * T$$

where: M is the Miller number, 15 to 50

C_{alumina} is the alumina concentration, 0.001 to 0.006 weight percent

D is the average runner diameter, 4.77 inches

N is the operating speed, 1462 to 3860 rpm depending upon operating point listed in Table 9-2

P is the pressure load on the bearing which varies with axial thrust as identified in the next section

A Monte Carlo analysis of thrust bearing wear was performed in a manner similar to that done on the radial bearings. A population of 10,000 pumps was used for each run. The results of the analysis are identified below in Table 9-17:

Table 9-17 - Thrust Bearing Life

<u>Axial Wear (mils)</u>	<u>Minimum Life (hours)</u>	<u>Average Life (hours)</u>	<u>Maximum Life (hours)</u>	<u>10,000 Hour Probability (%)</u>
20.00	4,314	7,369	11,479	0.49
25.00	5,884	9,154	13,491	20.33
30.00	7,472	10,945	15,713	79.95
35.00	9,233	12,753	18,676	99.37
40.00	10,522	14,592	20,501	100.00

Using the same wear probability criteria as for the radial bearings, 99+ percent probability of achieving 10,000 hours of life, the expected wear over the life of the transfer pump is expected to be less than 0.035 inch. Axial wear up to 0.085 inch is mechanically acceptable in the transfer pump. Metal to metal contact between the thrust shoe holders and the thrust runner holder will not occur with 0.0175 inch wear ($0.035 / 2 = 0.0175$) from each component. At 0.085 inch total axial wear of the thrust bearing, contact may occur between the lower shroud of one of the impellers and the static hydraulics. Therefore, the expected wear of the SiC thrust bearing parts is acceptable.

There is little probability that all six SiC thrust shoes, which are crowned, will be worn flat within 0.00005 inch by the abrasive fluid. The thrust shoes are free to

tilt to form fluid film wedges with the SiC runner, and therefore, a uniform wear across the faces of the shoes is unlikely.

The particle sizes in the hazardous waste fluid vary from 1 micron (0.00004 inch) to 1000 microns (0.0394 inch) in size; 95 percent of the particles vary between 1 and 50 microns (0.0019 inch). The particle population is such that the surface finish of the thrust runner and thrust shoes should remain acceptable from a bearing function viewpoint.

Assuming that the particle density is relatively uniform in the thrust bearing fluid lubricant, it is difficult to postulate a condition whereby abrasive wear preferentially occurs in the center of the pads and a "hole" / "slot" develops. This condition is, therefore, also considered unlikely to occur.

Thrust Bearing Loads

Hydraulic axial thrust has been calculated in the motor circulation analysis for the six operating points identified in Table 9-2. The axial hydraulic thrust was assumed, for simplification, to be the wet weight of the rotating assembly plus a function of the square of the operating speed. Based on the calculated thrust at each operating point, an equation was developed which conservatively calculates axial thrusts equal to or greater than those predicted in the motor circulation analysis. The axial thrust equation is:

$$T_{ax} = 161 - (21 * sg) + (5.607 \times 10^{-5} * sg * RPM^2)$$

where: T_{ax} is the axial thrust, lbs

sg is the specific gravity of the fluid

RPM is the operating speed of the motor, rpm

Table 9-18 identifies the axial thrust on the Transfer Pump as a function of operating speed and specific gravity.

Table 9-18 - Transfer Pump Axial Thrust

<u>Speed (rpm)</u>	<u>Hydraulic Thrust (lbs)</u>	
	<u>sg = 1.00</u>	<u>sg = 1.50</u>
1280	233	267
1710	305	375
2140	398	515
2570	511	685
3000	646	886
3430	801	1119
3860	976	1383

Thrust Bearing Performance Characteristics

Thrust bearing performance was calculated using the WEMD proprietary program "SECPAD" which has been verified and is under configuration control. SECPAD can compute the steady-state characteristics of sector-shaped thrust bearing pads. These characteristics include load capacity, minimum film thickness, friction loss, and leakage flow. The model used for the fluid film is based on a finite difference representation of the lubrication equations for isothermal, incompressible flow in either the laminar or turbulent regime.

Based on an effective inside radius of 1.680 inches, an effective outside radius of 3.250 inches, and an included sector angle of 53.6°, the centrally pivoted shoe was optimized for maximum film thickness in 1.00 centipoise fluid (cold water) at the design speed condition as follows: six shoes per bearing was chosen based on available space envelope, tangential cylindrical crowns (δ) of 0.0006 inch nominal (1300 inches crown radius nominal) and 0.0003 inch nominal (2600 inches crown radius nominal) were initially chosen, and then the radial pivot location which maximized the film thickness was selected. The results are shown in Tables 9-19 and 9-20.

Table 9-19 - Thrust Bearing Minimum Film Thickness

**For $\delta=0.0006$ inch, 1.0 Centipoise Viscosity, 1.00 Specific Gravity,
and 3860 rpm**

<u>Shoe Load (lbs)</u>	<u>Pivot Radius (inch)</u>	<u>Hmin (mils)</u>
163	2.445	0.646
163	2.450	0.646
163	2.455	0.646
163	2.460	0.646
163	2.465	0.646
163	2.470	0.646
163	2.475	0.646

Table 9-20 - Thrust Bearing Minimum Film Thickness

**For $\delta=0.0003$ inch, 1.0 Centipoise Viscosity, 1.00 Specific Gravity,
and 3860 rpm**

<u>Shoe Load (lbs)</u>	<u>Pivot Radius (inch)</u>	<u>Hmin (mils)</u>
163	2.445	0.634
163	2.450	0.634
163	2.455	0.634
163	2.460	0.634
163	2.465	0.634
163	2.470	0.633
163	2.475	0.633

Based on the results presented in Tables 9-19 and 9-20, a pivot radius of 2.465 was chosen. Next, the acceptability of this nominal design over the operating speed/load range for both the low viscosity/specific gravity and high viscosity/specific gravity conditions were evaluated. The results are presented in Tables 9-21 and 9-22.

Table 9-21 - Thrust Bearing Performance Results (1.0 cP and 1.00 sq)
 $R_i = 1.680$, $R_{pivot} = 2.465$, $R_o = 3.250$, $\beta = 53.6^\circ$, $\beta_{pivot} = 26.8^\circ$, $R_c = 1300$

<u>Speed (rpm)</u>	<u>Pad Load (lbs)</u>	<u>Minimum Film (mils)</u>	<u>Pad Horsepower</u>	<u>Leading Edge Leakage (in³/s)</u>
1280	39	0.761	0.010	0.420
1710	51	0.769	0.017	0.564
2140	66	0.756	0.028	0.718
2570	85	0.729	0.042	0.868
3000	108	0.698	0.061	1.012
3430	134	0.671	0.084	1.149
3860	163	0.646	0.113	1.280

Table 9-22 - Thrust Bearing Performance Results (30.0 cP and 1.50 sq)
 $R_i = 1.680$, $R_{pivot} = 2.465$, $R_o = 3.250$, $\beta = 53.6^\circ$, $\beta_{pivot} = 26.8^\circ$, $R_c = 1300$

<u>Speed (rpm)</u>	<u>Pad Load (lbs)</u>	<u>Minimum Film (mils)</u>	<u>Pad Horsepower</u>	<u>Leading Edge Leakage (in³/s)</u>
1280	45	3.382	0.071	0.286
1710	63	3.315	0.129	0.690
2140	86	3.193	0.209	2.051
2570	114	3.059	0.315	2.378
3000	148	2.920	0.448	2.673
3430	187	2.796	0.610	2.950
3860	231	2.684	0.803	3.212

Tables 9-21 and 9-22 indicate that the minimum film thickness will occur at the maximum pump operating speed. Also, the film thickness is a minimum at the low viscosity (1.0 centipoise) condition, and increases substantially with viscosity increases. The minimum film thickness condition is approximately 0.65 mil, which is acceptable for water or other low viscosity fluids. Because the minimum film thickness occurs at the minimum viscosity, design speed condition, this is the proper operating point for which to maximize the film thickness.

A thrust shoe crown sensitivity study was performed at the minimum film thickness operating point to confirm the acceptability of the crown initially chosen for the shoes. The results of this study, performed with the SECPAD program are delineated in Table 9-23.

Table 9-23 - Thrust Bearing Crown Affect at 163 lbs Load (1.0 cP and 1.00 sq)
Ri = 1.680, Rpivot = 2.465, Ro = 3.250, β = 53.6°, β_{pivot} = 26.8°, RPM = 3860

<u>Crown (mils)</u>	<u>Crown Radius (in)</u>	<u>Minimum Film (mils)</u>	<u>Pad Horsepower</u>	<u>Leading Edge Leakage (in³/s)</u>
1.0	775	0.608	0.116	1.662
0.9	861	0.620	0.115	1.570
0.8	969	0.631	0.114	1.480
0.7	1107	0.640	0.113	1.379
0.6	1292	0.646	0.113	1.280
0.5	1550	0.647	0.113	1.172
0.4	1938	0.643	0.113	1.058
0.3	2584	0.634	0.113	0.935
0.2	3876	0.612	0.117	0.803
0.1	7752	0.563	0.129	0.645

The above crown study reveals that the low viscosity, design speed maximum film thickness occurs at a crown radius of 1550 inches (0.0005 inch crown at a tangent to a radius of 2.465 inches). An acceptable crown radius range is 969 inches (0.0008 inch crown) to 7752 inches (0.0001 inch crown). As a design basis a crown of 0.0005 ± 0.0002 inch, or a crown radius of 1850 ± 750 inches, is acceptable. Note that a crown radius of 7752 inches (0.0001 inch crown) provided a film thickness over 0.56 mil; this substantiates the statement in the abrasive wear section that a centrally pivoted thrust shoe would be extremely tolerant of crown loss.

Thrust Bearing Links and Pins

The thrust bearing contains six sets of lower links and six sets of upper links. The maximum thrust load is 1383 lbs as listed in Table 18, and the average operating load on any one link is 1/6 of the thrust load, or 231 lbs. The maximum span between load carrying pins on a link is 1.32 inches. The maximum bending stress in the links is as follows, considering the links to be supported and the maximum load and bending moment mid-way between the supports:

$$\sigma = M \cdot h / (2 \cdot I)$$

$$M = F * L / 4$$

$$I = b * h^3 / 12$$

where: σ is the maximum bending stress, psi

M is the maximum bending moment, in-lb

L is the span between supports, 1.32 inches

b is the mid-point width of the links, 2.00 inches

h is the thickness of the links, 0.375 inch

$$M = 231 * 1.32 / 4$$

$$M = 76 \text{ in-lbs}$$

$$I = 2.00 * (0.375)^3 / 12$$

$$I = 0.00879 \text{ in}^4$$

$$\sigma = 76 * 0.375 / (2 * 0.00879)$$

$$\sigma = 1626 \text{ psi}$$

The links are 304 stainless steel with a yield strength of 30000 psi. The maximum bending stress is acceptable.

There are two sizes of pins used in the thrust bearing assembly, large pins and small pins. The large pins each support 1/6 the total load (231 lbs maximum),

and the small pins each support 1/12 (115 lbs maximum) the total load. One set of large pins are shrunk into the second stage diffuser/thrust bearing housing; each lower link is supported in the center by one of these large pins, which allows each link to tilt in response to load differentials. Each of the lower links contains two integral small pins, one at each end; these pins support the upper links at the ends and allow the links to tilt. The other set of six large pins are machined into and integral with each of the six upper links; these links support the thrust shoes. If one shoe is subject to a larger load than the adjacent shoes, the higher load on the upper link directly under the shoe causes the lower links which support it to pivot, lifting the adjacent upper links and shoes upward, increasing their load. This allows the bearing to take large angular misalignments while equalizing the loads on all the thrust shoes. The lower large pins and the links are 304 stainless steel. The shoe holders are 410 stainless steel.

To preclude the spherical radii on the ends of each pin from permanently deforming under load, the radius of each pin must be a certain minimum value. These spherical radii were calculated using Roark's Formulas For Stress and Strain (6th Edition), Table 33, Case 1a (sphere on a flat plate). The maximum allowed stress was 30000 psi, the yield strength of the 304 stainless steel material (the 410 stainless steel has a higher yield stress, 45000 psi). The results of the analysis is presented in Table 9-24.

Table 9-24 - Minimum Allowable Link Pin Spherical Radius

<u>Pin Location/Size</u>	<u>Material</u>	<u>Load (lbs)</u>	<u>Minimum Spherical Radius (inch)</u>
Support plate to lower link, large	304/304	231	19.8
Lower link to upper link, small	304/304	115	14.0
Upper link to shoe holder, large	304/410	231	20.2

Comparing the results of Table 9-24 which identifies the minimum spherical radii required to meet bearing stress requirements with the results of Table 9-16

which identifies the maximum spherical radii required to meet angular misalignment criteria, results in the following pin spherical radii choices:

- The six thrust bearing support plate to lower link pins are 0.715 inch nominal diameter with a spherical radius of 35 ± 2 inches.
- The two integral, machined pin heads on each of the six lower links which seat against the upper links are 0.465 inch nominal diameter with a 25 ± 2 inch spherical radius.
- The integral, machined pin head on each of the six upper links which seat against the thrust shoes are 0.715 inch nominal diameter with a spherical radius of 35 ± 2 inches.

Thrust Runner

The thrust runner is an assembly of two parts. The operating surface of the runner assembly which runs against the thrust shoes is the SiC insert. The thrust runner insert is nominally 0.350 inch thick and is backed up axially by the 410 stainless steel thrust runner back-up ring. The thrust runner insert is precluded from radial motion by a close fit at the inside diameter with the shaft. The runner insert is precluded from torsional motion by the shaft anti-rotation pin which extends from the second stage impeller hub through the thrust runner back-up ring. The insert is held axially against the runner by two methods: one is a 304 stainless steel ring which is fit onto the outside diameter of the shaft and is preloaded against the inner diameter of the insert by the second stage impeller (the second stage impeller is ultimately held axially by the first stage impeller lock bolt), and the other is a Locktite bond between the runner and insert. It is recognized that the Locktite may be ineffective in the high radiation, alkaline environment of the hazardous waste tanks. However, the main purpose of the Locktite is to secure the insert to the runner during handling and assembly operations.

The following minimum diametral clearance must be maintained between the inside diameter of the insert and the locating outside diameter of the 410 stainless steel shaft:

$$\Delta D = D * (\alpha_{410} - \alpha_{SiC}) * (T_{max} - T_{ref})$$

where: ΔD is the diametral difference in expansion, inch

D is the applicable diameter, 1.604 inches

α_{410} is the coefficient of thermal expansion of 410 stainless steel,
 $6.1 \times 10^{-6} \text{ } ^\circ\text{F}^{-1}$

α_{SiC} is the coefficient of thermal expansion of SiC, $2.2 \times 10^{-6} \text{ } ^\circ\text{F}^{-1}$

T_{max} is the maximum operating temperature, 220 °F estimated

T_{ref} is the reference assembly temperature, 68 °F

$$\Delta D = 1.604 * (6.1 \times 10^{-6} - 2.2 \times 10^{-6}) * (220 - 68)$$

$$\Delta D = 0.00095 \text{ inch}$$

Therefore, the maximum shaft fit diameter at the thrust runner must be 0.001 inch less than the minimum runner insert diameter.

Under normal conditions the Loctite between the SiC runner insert and the 410 stainless steel runner holder can withstand a shear loading of 2000 psi at room temperature and 1500 psi at 200 °F without failing, per the Loctite technical data sheet. The Loctite shear area of the thrust runner insert is 32.3 in^2 . The shear force required to break the Loctite bond is 48450 lbs ($1500 * 32.3 =$

48450). Since the maximum tangential force that can be applied to the SiC insert is 1650 lbs (110 ft-lbs of torque at a 0.80 inch inside radius of the insert) the Locktite should secure the thrust runner insert to the thrust runner holder. If the Locktite degrades due to radiation and cannot perform this function, the impeller/thrust runner key, sized for motor breakaway torque, will prevent rotation of the insert within the holder.

A finite element analysis of the thrust runner assembly with the maximum design load of 1383 lbs at 3860 rpm indicates maximum tensile stress of 1715 psi will occur on the SiC insert at the inside diameter. Although it is desirable to preclude any tensile stresses in SiC, some minor tensile stresses cannot be avoided, and the maximum tensile stress level is under 3 percent of the 67000 psi flexural strength of the SiC material.

10.0 DYNAMIC EVALUATION

10.1 Model Description

A classical critical speed analysis of the Transfer Pump rotating assembly was performed. The pump rotor assembly was modeled for stiffness and inertia to obtain the first lateral natural frequency assuming pinned lateral (radial) bearings (i.e., infinitely stiff bearings). Also, it was assumed that the thrust bearing was pinned in the axial direction and that rocking could occur.

The dynamic model of the pump rotor and stator, as well as the column is included in the complete dynamic analysis of the transfer pump system. This model is the basic model used for all the dynamic analysis of the transfer pump system assembly.

Rotor Modeling Description

The rotor assembly was modeled using a finite element computer code WECAN. To form the model inputs used were the different stages of the rotor section diameters, length, material density, and modulus of elasticity. A special in house code (PHYTRI) is used to calculate equivalent elements from detailed input for the finite element model. The left side of Figure 10-1 shows the detailed geometric values used as input into the PHYTRI code. The right side of Figure 10-1 shows stiffness elements calculated by the PHYTRI code, that were then entered into the finite element model.

Figure 10-1

Rotor Model -- Detailed Geometric Input Used

Rotor Model- Half of cross-section of Equivalent Stiffness Elements Calculated

Rotor, Stator, and Pipe Column Modeling Description

The entire transfer pump system assembly (Figure 15-2) was modeled using the finite element computer code WECAN. To form the model inputs used were the different stages of the rotor and stator section diameters, length, material density, and modulus of elasticity. A special in house code (PHYTRI) is used to calculate equivalent elements from detailed input for the finite element model. The left side of Figure 10-2 shows the detailed geometric values used as input into the PHYTRI code. The right side of Figure 10-2 shows stiffness elements calculated by the PHYTRI code, that were then entered into the finite element model. The resulting model was then modified to add the calculated bearing and seal stiffness values, and the pipe column elements for representation of the entire transfer pump system assembly.

Stiffness parameters for the radial and thrust bearings, and the impeller seals were all modeled. The radial bearings were modeled using a special in house computer code (TILTPAD) for tilting pad bearings. The computer code TILTPAD was used to compute the bearing stiffness and damping for the tilting pad bearings. The thrust bearing was not modeled for this system since there is presumed no coupling between axial and lateral motions of the pump. A value of thrust bearing axial stiffness was used that was calculated previously for the previous thrust bearing design. This value should be a good estimate since there are no known excitations in the system axially that could cause problems. Seal stiffness values were calculated based on a plain seal using friction factors, Reynolds numbers, and pressure drops that were calculated from the fluid internal circulation analysis (Section 5.7).

The piping column assembly (Figure 15-5) was added to the dynamic model by using pipe elements in the finite element code (WECAN) with the proper diameter and wall values to represent the various pipes used. To simplify the model the outer pipe was modeled directly and all the inner pipes were lumped together as one equivalent pipe using the PHYTRI computer code. This should be somewhat representative of how the actual piping will act due to the design construction of the piping system. The top of the piping was considered to be ground with the other end connected to a pipe element that modeled the transition piece. The transition piece is connected at the other end to the top of the pump stator (node 100 in the model shown Figure 10-2). The tank fluid was

not included in the modeling to represent the worse case of damping to the piping column when the pump is running with little or no water next to the piping.

This transfer pump system finite element dynamic model is used to perform modal analysis of the entire system. Results from the modal analysis are then used to create a damped harmonic response model for stability and vibration prediction using the in house computer codes DYSTAB and HARRESX, respectively.

Most cases analyzed were for the pump running in water instead of the process fluid. This is due to the fact that water should be the worst case for vibration consideration due to the higher damping effects of the more viscous process fluid.

Figure 10-2

Rotor & Stator Model -- Detailed Geometric Input Used

Rotor & Stator Model -- Half of cross-section of Equivalent Stiffness Elements Calculated

10.2 Classical Critical Speed Analysis

Results for the classical critical speed analysis showed the first rotor lateral bending mode to be around 96 Hz. This is the traditional lateral pinned-pinned rotor critical speed; it neglects any rocking stiffness from the axial bearing. This is about 1.5 times the highest running speed of the pump, which is approximately 64 Hz (3860 rpm).

The first rotor lateral bending mode shape is shown in Figure 10-3. This mode is the first mode that occurred using the finite element code (WECAN) model described previously, which is part of the definition of the classical critical speed value. No other modes are included in these results since the modal analysis used neglects many other stiffness factors such as the bearings, the magnetic and can annulus stiffness, the stator, and the piping column.

Displ. Magnif. = 4.69601
Mode number = 2 Frequency = 95.628
Neutral file = TPRM_RES

WEDANPLUS Thu Apr 24 1997 15:39:06 TPRM Ver:95

Figure 10-3

Rotor Model -- Pinned-Pinned Lateral Bearings with Axial Constrained Thrust Bearing

10.3 Undamped Natural Frequency Analysis

The transfer pump system finite element dynamic model was used to perform an undamped modal analysis of the entire system. The entire system includes the pump rotor and stator, and the piping column. This analysis is based on the pump running at some steady state position so that constant rotor bearing stiffness values were assumed. The non-linear effects of the rotor lateral bearings are not included in this modal analysis calculation, since this is a linear calculation. Results from the modal analysis do not include any damping or dynamic loading effects. Also, this modal analysis model does not include the effects of the motor can annulus stiffness and damping. The motor can annulus effects are included in the damped natural frequency analysis (Section 10.4).

The results of the undamped modal analysis are used to evaluate the system undamped frequencies and are input into the damped frequency analysis model used in Section 10.4. Figure 10-4 to Figure 10-36 show the undamped modal analysis results for the pump running in water at the highest speed (3860 rpm) for the various frequencies of interest. Note that the first frequency calculated for this system was zero since the analysis allowed for rotation of the rotor about its axis, so this frequency was ignored. Also, lateral mode frequencies often occur in pairs due to symmetry in the model. Thus only one of the similar modes is shown for simplicity.

One case was examined for the pump running in water at the highest running speed of 3860 rpm. This is just one case of rotor lateral bearing stiffness values. Various bearing stiffness values are possible due to the non-linear stiffness of the lateral bearings and the different potential loading of the bearings. The case presented here is for a stator can offset of zero. This case is somewhat representative of the cases possible for the stator can offset, which can be in different directions that result in different bearing dynamic stiffness values. (Stator can offset is the radial offset of the bearing center line to the motor stator can center line.)

Displ. Magnif. = 50.5337
Mode number = 2 Frequency = 0.4645
Neutral file = TPM1W0_RES

MODE2Z

WECANPLUS Tue, Apr 22 1997 15:05:05 PLOT Ver:85

y-axis

x-axis

Figure 10-4
Undamped Pump & Column Model -- First Column Mode

Displ. Magnif. = 43.3619
Mode number = 4 Frequency = 3.8045
Neutral file = TPM1W0_RES

MODE4Z

WECANPLUS Tue, Apr 22 1997 15:05:11 PLOT Ver:85

Figure 10-5
Undamped Pump & Column Model -- Second Column Mode

Displ. Magnif. = 42.5578
Mode number = 6 Frequency = 11.382
Neutral file = TPM1W0_RES

MODE6Z

WECANPLUS Tue, Apr 22 1997 15:05:17 PLOT Ver:95

Figure 10-6
Undamped Pump & Column Model -- Third Column Mode

Displ. Magnif. = 42.5687
Mode number = 8 Frequency = 22.74
Neutral file = TPM1WQ_RES

MODEBZ

W -WECANPLUS Tue, Apr 22 1997 16:05:23 PLOT Ver:06

Figure 10-7
Undamped Pump & Column Model -- Fourth Column Mode

Displ. Magnif. = 0.
Mode number = 10 Frequency = 33.132
Neutral file = TPM1W0_RES

MODE10X

WE CAN PLUS Tue. Apr 22 1997 15:05:28 PLOT Ver:95

y-axis

z-axis

Figure 10-8

Undamped Pump & Column Model -- First Torsional Column Mode

Displ. Magnif. = 58.1616
Mode number = 11 Frequency = 35.224
Neutral file = TPM1W0_RES

MODE11Z

WECANPLUS Tue, Apr 22 1997 16:06:32 PLOT Ver:95

Figure 10-9
Undamped Pump & Column Model -- First Rotor Rocking Mode (X Plane)

Dipl. Magnif. = 10.
Mode number = 11 Frequency = 35.224
Neutral file = TPM1W0_RES

MODE11Z

WECANPLUS Tue, Apr 22 1997 15:10:51 PLOT Ver:85

Figure 10-10

Undamped Pump & Column Model -- Pump View of First Rotor Rocking Mode (X Plane)

Dipl. Magnif. = 58.0886
Mode number = 12 Frequency = 36.169
Neutral file = TPM1W0_RES

MODE12X

WECANPLUS Tue, Apr 22 1997 15:05:34 PLOT Ver:05

Figure 10-11
Undamped Pump & Column Model -- First Rotor Rocking Mode (Z Plane)

Displ. Magnif. = 10.
Mode number = 12 Frequency = 36.169
Neutral file = TPM1W0_RES

MODE12X

WECANPLUS Tue, Apr 22 1997 15:10:53 PLOT Ver:95

Figure 10-12

Undamped Pump & Column Model -- Pump View of First Rotor Rocking Mode (Z Plane)

Displ. Magnif. = 47.9563
Mode number = 13 Frequency = 37.778
Neutral file = TPM1W0_RES

MODE13Z

WECAWPLUS Tue, Apr 22 1997 15:05:38 PLOT Ver:95

Figure 10-13
Undamped Pump & Column Model -- Fifth Column Mode & Second Rotor Rocking Mode
(X Plane)

Displ. Magnif. = 10.
Mode number = 13 Frequency = 37.778
Neutral file = TPM1W0_RES

MODE13Z

W WECANPLUS Tue, Apr 22 1997 15:10:56 PLOT Ver:95

Figure 10-14
Undamped Pump & Column Model -- Pump View of Second Rotor Rocking Mode (X Plane)

Displ. Magnif. = 52.0506
Mode number = 14 Frequency = 38.02
Neutral file = TPM1W0_RES

MODE14X

W -WECANPLUS Tue, Apr 22 1997 15:05:40 PLOT Ver:06

Figure 10-15
Undamped Pump & Column Model -- Fifth Column Mode & Second Rotor Rocking Mode
(Z Plane)

Displ. Magnif. = 10.
Mode number = 14 Frequency = 38.02
Neutral file = TPM1WD_RES

MODE14X

WECAPLUS Tue, Apr 22 1997 15:10:57 PLOT Ver:85

Figure 10-16
Undamped Pump & Column Model -- Pump View of Second Rotor Rocking Mode (Z Plane)

Displ. Magnif. = 42.4267
Mode number = 15 Frequency = 53.547
Neutral file = TPM1W0_RES

MODE15X

WECANPLUS Tue, Apr 22 1997 15:05:43 PLOT Ver95

Figure 10-17

Undamped Pump & Column Model – Third Rotor Mode (Z Plane)

Displ. Magnif. = 10.
Mode number = 15 Frequency = 53.547
Neutral file = TPM1W0_RES

MODE15X

W -WECANPLUS Tue, Apr 22 1997 15:11:00 PLOT Ver:95

Figure 10-18
Undamped Pump & Column Model -- Pump View of Third Rotor Mode (Z Plane)

Displ. Magnif. = 42.4347
Mode number = 16 Frequency = 56.545
Neutral file = TPM1W0_RES

MODE16Z

W -WECANPLUS Tue, Apr 22 1997 15:05:48 PLOT Ver:06

Figure 10-19
Undamped Pump & Column Model -- Sixth Column Mode & Third Rotor Mode (X Plane)

Displ. Magnif. = 10.
Mode number = 16 Frequency = 56.545
Neutral file = TPM1W0_RES

MODE16Z

W -WECANPLUS Tue, Apr 22 1997 15:11:03 PLOT Ver:95

Figure 10-20
Undamped Pump & Column Model -- Pump View of Third Rotor Mode (X Plane)

Displ. Magnit. = 58.544
Mode number = 17 Frequency = 57.124
Neutral file = TPM1W0_RES

y-axis

x-axis

MODE17Z

WECANPLUS Tue, Apr 22 1997 15:05:51 PLOT Ver.95

Figure 10-21
Undamped Pump & Column Model -- First Axial Column Mode

Displ. Magnif. = 42.4491
Mode number = 18 Frequency = 59.244
Neutral file = TPM1W0_RES

MODE18X

-WECANPLUS Tue, Apr 22 1997 15:06:53 PLOT Ver:95

Figure 10-22

Undamped Pump & Column Model – Column Mode & Fourth Rotor Rocking Mode (Z Plane)

Displ. Magnif. = 10.
Mode number = 18 Frequency = 59.244
Neutral file = TPM1WD_RES

MODE18X

WECANPLUS Tue, Apr 22 1997 15:11:07 PLOT Ver:85

Figure 10-23

Undamped Pump & Column Model -- Pump View of Fourth Rotor Rocking Mode (Z Plane)

Displ. Magnif. = 42.5395
Mode number = 19 Frequency = 70.16
Neutral file = TPM1W0_RES

MODE19X

W -WECANPLUS Tue, Apr 22 1997 15:06:56 PLOT Ver:95

Figure 10-24
Undamped Pump & Column Model – Column Mode & Rotor Mode (Z Plane)

Displ. Magnif. = 10.
Mode number = 18 Frequency = 70.16
Neutral file = TPM1W0_RES

MODE19X

W -WECANPLUS Tue, Apr 22 1997 15:11:08 PLOT Ver:85

Figure 10-25
Undamped Pump & Column Model – Pump View of Rotor Mode (Z Plane)

Displ. Magnif. = 42.6205
Mode number = 20 Frequency = 80.995
Neutral file = TPM1W0_RES

MODE20Z

WECANPLUS Tue, Apr 22 1997 15:06:00 PLOT Ver:95

Figure 10-26
Undamped Pump & Column Model -- Column Mode & Rotor Mode (X Plane)

Displ. Magnif. = 10.
Mode number = 20 Frequency = 80.995
Neutral file = TPM1W0_RES

MODE20Z

W -WECANPLUS Tue, Apr 22 1997 15:11:12 PLOT Ver:95

Figure 10-27
Undamped Pump & Column Model -- Pump View of Rotor Mode (X Plane)

Displ. Magnif. = 42.6307
Mode number = 21 Frequency = 81.491
Neutral file = TPM1W0_RES

W -WECANPLUS Tue, Apr 22 1997 15:09:02 PLOT Ver:95

Figure 10-28
Undamped Pump & Column Model – Column Mode & Rotor Mode (Z Plane)

Displ. Magnif. = 10.
Mode number = 21 Frequency = 81.491
Neutral file = TPM1W0_RES

MODE21X

W WECANPLUS Tue, Apr 22 1997 15:11:13 PLOT Ver:95

Figure 10-29

Undamped Pump & Column Model -- Pump View of Rotor Mode (Z Plane)

Displ. Magnif. = 10.
Mode number = 22 Frequency = 109.94
Neutral file = TPM1W0_RES

MODE22Z

W -WECANPLUS Tue, Apr 22 1997 15:11:17 PLOT Ver:35

Figure 10-31
Undamped Pump & Column Model -- Pump View of Rotor Mode (X Plane)

Displ. Magnif. = 42.657
Mode number = 23 Frequency = 109.98
Neutral file = TPM1W0_RES

MODE23X

MECANPLUS Tue, Apr 22 1997 15:06:08 PLOT Ver:86

Figure 10-32
Undamped Pump & Column Model – Column Mode & Rotor Mode (Z Plane)

Displ. Magnif. = 10.
Mode number = 23 Frequency = 109.98
Neutral file = TPM1W0_RES

MODE23X

W -WECANPLUS Tue, Apr 22 1997 15:11:18 PLOT Ver:95

Figure 10-33

Undamped Pump & Column Model -- Pump View of Rotor Mode (Z Plane)

Displ. Magnif. = 0.
Mode number = 24 Frequency = 124.46
Neutral file = TPM1W0_RES

y-axis

z-axis

MODE24X

WECANPLUS Tue, Apr 22 1997 15:08:12 PLOT Ver:95

Figure 10-34
Undamped Pump & Column Model -- Second Torsional Column Mode

Displ. Magnif. = 44.9796
Mode number = 25 Frequency = 138.24
Neutral file = TPM1W0_RES

y-axis

MODE25Z

-WECANPLUS Tue, Apr 22 1987 15:06:16 PLOT Ver:95

Figure 10-35

Undamped Pump & Column Model -- Column Mode & Rotor Mode (X Plane)

Displ. Magnif. = 10.
Mode number = 25 Frequency = 136.24
Neutral file = TPM1W0_RES

MODE25Z

WECANPLUS Tue, Apr 22 1997 15:11:24 PLOT Ver:35

Figure 10-36
Undamped Pump & Column Model -- Pump View of Rotor Mode (X Plane)

10.4 Damped Rotor Dynamic Analysis

A damped dynamic analysis of the pump rotor in the entire transfer pump system was performed. The damped harmonic response program uses the undamped modal analysis mode shape results, as described in Section 10.3. It then applies the corrections to the model for bearing stiffness and damping values; as well as adding the effects of the can annulus stiffness, damping and total inertia, as described in Section 10.1. The resulting complete dynamic model was then analyzed for stability and vibration prediction.

Different cases were analyzed at various operating speeds of the pump with water as the operating fluid and a zero stator can offset. (Stator can offset is the radial offset of the bearing center line to the motor stator can center line.) The various cases of rotor speed were analyzed to show how speed affects bearing stiffness values and thus rotor modes. Similar results were calculated for the pump operating in the process fluid for the highest speed case only.

Natural frequency results of the stability analysis for the various operating rotor speeds are shown in Table 10-1. The mode shape results show the damped modes are predicted in both lateral planes. This is why most lateral modes occur in pairs. The mode numbers do not necessarily correspond to any other mode numbers from the previous calculation of Section 10.3. The stability analysis shows that all modes are stable for any of the cases analyzed. The highly damped rotor modes shown in Table 10-1 have at least ten percent critical damping. This means there is very little influence on the vibration results due to these modes.

To predict vibration of the rotor in the pump relative to the stator the maximum unbalance values allowed by the pump specification were applied to the pertinent locations along the rotor. The results of this vibration prediction for the various operating speeds in water are shown in Table 10-2 and Figure 10-37. Since none of the vibration values for the rotor relative to the stator are greater than 0.002 inch radial, rotor vibration for the various speeds at a stator offset of zero should not be a problem.

Examination of the results shows that rotor speed does have some effect on the results, but none that are significant to the safe operation of the pump.

Table 10-1 -- Transfer Pump Rotor Stability Results for Various Pump Speeds
System Damped Natural Frequencies (Pump in Water, Zero Stator Can Offset)

Rotor Speed (RPM)	3860	3600	3410	3350	2063	1490
Damped Mode	Frequency (Hz)	Frequency (Hz)	Frequency (Hz)	Frequency (Hz)	Frequency (Hz)	Frequency (Hz)
1	0.46	0.46	0.46	0.46	0.46	0.46
2	0.46	0.46	0.46	0.46	0.46	0.46
3	3.80	3.80	3.80	3.80	3.80	3.80
4	3.80	3.80	3.80	3.80	3.80	3.80
5	* 6.05	11.38	* 9.85	11.38	* 5.71	* 7.38
6	11.38	11.38	11.38	11.38	11.38	11.37
7	11.38	* 17.39	11.38	* 11.56	11.38	11.38
8	22.75	22.75	22.74	22.75	22.73	* 20.79
9	22.76	22.75	22.75	22.75	22.73	22.73
10	* 22.89	33.13	* 28.64	33.13	* 23.16	22.74
11	33.13	37.80	33.13	* 33.28	* 31.38	* 23.66
12	37.81	37.81	* 35.35	* 37.32	33.13	33.13
13	37.82	* 40.55	37.81	37.81	* 34.52	* 34.92
14	* 45.46	* 46.41	37.84	37.83	37.84	37.83
15	56.83	* 52.95	* 48.35	* 39.64	37.88	37.95
16	56.94	56.89	56.87	56.91	57.08	57.02
17	57.12	56.95	57.01	57.00	57.09	57.08
18	* 60.25	57.12	57.12	57.12	57.12	57.14
19	80.44	80.50	80.48	80.49	80.63	80.59
20	80.53	80.58	80.57	80.57	80.66	80.65
21	107.61	109.75	108.04	108.86	109.79	109.80
22	109.75	109.80	109.76	109.75	109.82	109.81
23	109.80	112.82	109.80	109.80	112.06	111.85
24	114.20	122.69	114.69	115.35	118.00	116.14
25	124.46	124.46	124.46	124.46	124.46	124.46

Note: All Modes were stable.

* Highly damped rotor modes below the highest running speed.

Table 10-2

Vibration Response Results for Various Pump Speeds

Radial Displacements of Rotor Relative to Stator & Bottom of Pump Relative to Ground

Pump Running in Water, Zero Stator Can Offset

Rotor Speed (RPM)	3860	3600	3410	3350	2063	1490
Location	Displacement (inches)	Displacement (inches)	Displacement (inches)	Displacement (inches)	Displacement (inches)	Displacement (inches)
Upper Bearing	0.000483	0.000473	0.000806	0.000614	0.000480	0.000278
Can Midpoint	0.000187	0.000156	0.000349	0.000230	0.000169	0.000204
Lower Bearing	0.000817	0.000721	0.000975	0.000929	0.000818	0.000605
2nd Stage Impeller Seal	0.001329	0.001133	0.001310	0.001353	0.001136	0.000804
1st Stage Impeller Upper Seal	0.001600	0.001338	0.001503	0.001567	0.001268	0.000884
1st Stage Impeller Lower Seal	0.001888	0.001557	0.001714	0.001797	0.001409	0.000970
Bottom of Pump (Relative to ground)	0.000294	0.000410	0.002393	0.000427	0.000021	0.000303

Figure 10-37

Vibration Response Results for Various Pump Speeds

Radial Displacements of Rotor Relative to Stator & Bottom of Pump Relative to Ground
Pump Running in Water, Zero Stator Can Offset

The final case analyzed was for the transfer pump operating in the process fluid at the highest rotor speed with zero stator can offset. The damped natural frequency results of the stability analysis for the pump in process fluid are shown in Table 10-3. The results of the vibration prediction for the pump in process fluid are shown in Table 10-4. Results are compared for the pump running in water to the pump running in the process fluid. These results are at the same rotor speed and stator can offset. The highly damped rotor modes shown in Table 10-3 have at least ten percent critical damping. This means there is very little influence on the vibration results due to these modes.

Reduced vibration occurs with the process fluid in the pump due to the added damping and changes to the other fluid effects. Similar results with the pump operating in the process fluid at other speeds, instead of water, are expected.

Table 10-3 -- Transfer Pump Rotor Stability Results -- System Damped Frequencies
Comparison of Pump in Water to Pump in Process Fluid, Rotor speed = 3860 rpm

Type of Pump Fluid Stator Can Offset	Pump in Water Zero Offset	Pump in Process Fluid Zero Offset
Damped Mode	Frequency (Hz)	Frequency (Hz)
1	0.46	0.46
2	0.46	0.46
3	3.80	• 0.70
4	3.80	3.74
5	• 6.05	3.74
6	11.38	• 7.06
7	11.38	11.17
8	22.75	11.17
9	22.76	22.35
10	• 22.89	22.35
11	33.13	33.10
12	37.81	37.18
13	37.82	37.18
14	• 45.46	55.87
15	56.83	55.87
16	56.94	56.61
17	57.12	• 56.91
18	• 60.25	78.79
19	80.44	78.87
20	80.53	88.93
21	107.61	107.57
22	109.75	107.76
23	109.80	124.11
24	114.20	140.37
25	124.46	140.54
Note: All Modes were stable. • Highly damped rotor modes below the highest running speed.		

Table 10-4

Vibration Response Results

Radial Displacements of Rotor Relative to Stator & Bottom of Pump Relative to Ground
Comparison of Pump in Water to Pump in Process Fluid, Rotor speed = 3860 rpm

Type of Pump Fluid & Stator Can Offset	Pump in Water & Zero Offset	Pump in Process Fluid & Zero Offset
Location	Displacement (inches)	Displacement (inches)
Upper Bearing	0.000483	0.000093
Can Midpoint	0.000187	0.000418
Lower Bearing	0.000817	0.000086
2nd Stage Impeller Seal	0.001329	0.000533
1st Stage Impeller Upper Seal	0.001600	0.000718
1st Stage Impeller Lower Seal	0.001888	0.000908
Bottom of Pump (Relative to ground)	0.000294	0.000214

11.0 ABRASION TESTING**11.1 Test Description and Test Plan**

Westinghouse Electro-Mechanical Division (WEMD) contracted with the Westinghouse Science and Technology Center (WSTC) to perform abrasion testing on various critical features of the Transfer Pump. These features are the radial bearings, the stator/rotor cans, and the impeller/diffuser areas. WSTC constructed test rigs to operate with a simulated non-radioactive waste tank process fluid. This process fluid has a 20 percent by weight mixture of solids in water as delineated in Table 11-1. The Miller number represented by this mixture is 100 with a 20 percent solids mixture and 112 with a 50 percent solids mixture. The "true" Miller number per ASTM G75-82 is calculated with a 50 percent mixture. All Miller number measurements were performed by White Rock Engineering, Inc. of Garland, Texas.

Table 11-1 Simulant Constituents

Constituent	Trade Name	Average Particle Size, microns	Constituent Concentration, grams/gallon H ₂ O
Silica	Min-U-Sil 5	2	2.50
Silica	Min-U-Sil 30	8	2.50
Alumina	#5 Aluminum Oxide Powder	10	0.25
Zeolite	Micronized Clinoptilolite	20 - 24	24.40
Albite	Sodium Feldspar	10 - 11	311.70
Goethite	Ochre Pigment	4 - 5	604.90
Total Solids			946.25

11.1.1 Bearing Testing

The bearing test rig has a stationary bearing sleeve and a rotating journal configuration. The bearing design is a standard 360° sleeve bearing. The sleeve inside diameter is 3.940 inches with a sleeve/journal diametral clearance of 0.006 inch. The total bearing length is 2.0 inches (1.80 inches operating length). Refer to Figure 11-1 for a view of the test rig.

Three bearing materials were tested: Silicon Carbide, diamond-coated Tungsten Carbide, and Chromide-coated 403 stainless steel. The bearings were tested as Silicon Carbide on Silicon Carbide, diamond-coated Tungsten Carbide on diamond-coated Tungsten Carbide, and Chromide-coated 403 stainless steel on Chromide-coated 403 stainless steel. The Silicon Carbide bearing was direct sintered and the Tungsten Carbide bearing has a 6% nickel binder. The Silicon Carbide bearing has four slots, 90° apart, positioned in the bearing sleeve inside diameter. The slots are 0.312 inch wide by 0.140 inch deep, and are arranged in a helical pattern, spiraling 90° from entrance to exit. The diamond-coated Tungsten Carbide bearing and the Chromide-coated bearing had no axial slots in the sleeves.

The stationary bearing sleeve and rotating journal were assembled in the test rig such that a simulation of a typical wedge-type fluid film is formed during bearing operation. The bearings were operated at 3200 rpm with the simulant temperature maintained at 85 °F. The bearing was loaded to 91 lbs to simulate the expected minimum film thickness (approximately 0.6 mil) in the pump bearings during actual operating conditions. WEMD considered that most bearing wear would probably occur as simulant particles tried to push through the converging fluid wedge formed by the bearing and insert; therefore, relative to abrasive wear in a radial bearing it is more important to simulate the operating minimum film thickness than the operating loads.

Bearing wear was measured on both the journals and the bearing sleeves. Journal wear was obtained by a set of 4 axial profile traces spaced 90° apart on the outside diameter. The accuracy of the profile measurement instrumentation is within 5

Figure 11-1 Bearing Test Rig

microinches. The outer 0.10 inch of both ends of the journals protruded beyond the bearing sleeves in the test rig. These outer end sections were not subjected to abrasive wear and were used as reference datums to compare profile traces and measure wear in the film-riding section of the journal. Bearing sleeve wear was obtained by a set of 3 circumferential profile traces at the fluid inlet region, the center, and the fluid outlet region. The roundness of the bearing sleeves were determined based on the traces, and the changes in roundness were considered to be wear. Changes in roundness were considered to be an appropriate measure of wear because the wear test rig provides a unidirectional, constant load which fixes the minimum film location. This results in most of the bearing sleeve wear occurring at only one location. Sleeve wear therefore is proportional to changes in roundness.

The Chromide-coated bearing was operated for 25 hours in a simulant that had only one-quarter of the solids content identified in Table 11-1 (i.e., the simulant had only a 5 percent by weight solids content). Inspections of the Chromide-coated bearing revealed that the journal diameter had decreased an average of 1.88 mils and a maximum of 2.70 mils, and the bearing sleeve out-of-roundness had increased an average of 2.10 mils and a maximum 2.42 mils. Because of the high wear rate, no additional wear testing of the Chromide-coated bearing was performed.

The Silicon Carbide bearing was operated for 300 hours in the simulant identified in Table 11-1. Dimensional inspections were performed on the bearing prior to operation and at the following operating times: 25 hours, 80 hours and 300 hours. The Silicon Carbide journal wear as a function of operating time is delineated in Table 11-2. The Silicon Carbide sleeve roundness as a function of operating time is listed in Table 11-3.

Table 11-2 Silicon Carbide Journal Radial Wear (Mils)

Hours	Loc #1 Inlet Wear	Loc #1 Outlet Wear	Loc #2 Inlet Wear	Loc #2 Outlet Wear	Loc #3 Inlet Wear	Loc #3 Outlet Wear	Loc #4 Inlet Wear	Loc #4 Outlet Wear
0	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
25	0.60	0.10	0.30	0.10	0.05	0.05	0.20	0.10
80	0.65	0.20	0.40	0.15	0.20	0.10	0.30	0.10
300	0.90	0.40	0.55	0.30	0.35	0.15	0.45	0.15

Table 11-3 Silicon Carbide Sleeve Roundness (Mils)

Hours	Inlet Roundness	Center Roundness	Outlet Roundness	Average Roundness
0	0.530	0.445	0.430	0.468
25	0.950	0.540	0.450	0.647
80	0.925	0.555	0.475	0.652
300	0.920	0.700	0.650	0.757

Table 11-2 reveals that the maximum Silicon Carbide journal wear was 0.90 mil radial, or 1.80 mils ($2 \times 0.90 = 1.80$) on the diameter, after 300 hours of operation. The journal average diametral wear was 0.81 mils ($2 \times [0.90 + 0.55 + 0.35 + 0.45 + 0.40 + 0.30 + 0.15 + 0.15] / 8 = 0.81$). Table 11-3 indicates that the maximum diametral wear on the Silicon Carbide sleeve after 300 hours was 0.39 mil ($0.920 - 0.530 = 0.390$), and the average diametral wear was 0.29 mil ($0.757 - 0.468 = 0.289$). Because the Silicon Carbide bearing wear rate was considered to be excessive (an estimated 60 mils wear over 10000 hours), the testing was terminated after 300 hours.

The diamond-coated Tungsten Carbide bearing was operated for 1000 hours in the simulant identified in Table 11-1. Dimensional inspections were performed on the bearing prior to operation and at the following operating times: 25 hours, 185 hours, 500 hours, and 1000 hours. The diamond-coated Tungsten Carbide journal measured wear was judged to be less than 0.10 mil over the entire 1000 hours. Axial profile traces of the diamond-coated journal prior to simulant testing are shown in Figures 11-2 through 11-5. Axial profile traces of the journal after 1000 hours of operation are shown in Figures 11-6 through 11-9. Comparing the traces indicates that the estimated 0.10 mil of wear is conservative and that most of the "wear" is caused by an improvement in the surface finish over the film-riding area of the journal. The diamond-coated Tungsten Carbide sleeve roundness as a function of operating time is listed in Table 11-4.

Figure 11-2 Diamond-Coated Journal Profile Location 1 Prior to Simulant Test

Figure 11-3 Journal Profile Location 2 Prior to Simulant Test

Figure 11-4 Diamond-Coated Journal Profile Location 3 Prior to Simulant Test

Figure 11-5 Diamond Coated Journal Profile Location 4 Prior to Simulant Test

Figure 11-6 Diamond Coated Journal Profile Location 1 After 1000 Hours Simulant Test

Figure 11-7 Diamond Coated Journal Profile Location 2 After 1000 Hours Simulant Test

Figure 11-8 Diamond Coated Journal Profile Location 3 After 1000 Hours Simulant Test

Figure 11-9 Diamond Coated Journal Profile Location 4 After 1000 Hours Simulant Test

Table 11-4 Diamond-Coated Tungsten Carbide Sleeve Roundness (Mils)

Hours	Inlet Roundness	Center Roundness	Outlet Roundness	Average Roundness
0	0.375	0.305	0.275	0.318
25	0.355	0.270	0.295	0.307
185	0.335	0.275	0.260	0.290
500	0.360	0.335	0.345	0.347
1000	0.325	0.290	0.255	0.290

Table 11-4 indicates that there is effectively no wear on the diamond-coated Tungsten Carbide sleeve after 1000 hours of operation. Roundness readings reveal improvements in roundness at the inlet end, the center, and the outlet end of the sleeve. These readings are indicative of a surface finish improvement rather than any significant wear. This tends to confirm the conclusion reached above, that the 0.10 mil maximum journal wear was mainly an improvement in the surface finish.

Based on these wear test results, WEMD considers that diamond-coating on Tungsten Carbide is the only acceptable bearing surface for the Transfer Pump application. The expected bearing life is discussed in Section 13.0.

Incorporation of diamond-coated Tungsten Carbide bearings has necessitated several design features in the transfer pump. One feature is the use of full 360° sleeve bearings with no spiral grooves. The lack of spiral grooves requires an upper radial bearing housing with bypass holes for internal circulation. The lower radial bearing also has bypass holes in the stator for internal circulation of fluid.

11.1.2 Can Testing

The can test rig consisted of a short length mock-up of the rotor/stator interface of the Transfer Pump. The stator can was welded to the inside diameter of a stationary

pipe with two circumferential end welds. The rotor can was welded to the outside diameter of a mandrel with two circumferential end welds. The rotor mandrel was rotated at 3600 rpm. Both cans were 0.021 inch thick Hastelloy C-276 sheet and each was formed into a cylinder with a single longitudinal seam weld. The stator can has an inside diameter of 4.11 inches, and the rotor can has an outside diameter of 3.83 inches. The nominal radial gap between the stator and rotor cans is 0.145 inch which is prototypic of the Transfer Pump. The mock-up cans are six inches long. Refer to Figure 11-10 for a view of the can test rig.

The simulated process fluid identified in Table 11-1 was pumped through the can gap at 15 gpm. The fluid was heated to 200 °F. The tangential velocity of the rotor can is 60.16 feet/second at 3600 rpm, and the average tangential velocity in the can annulus is 30.08 feet/second ($0.5 \times 60.16 = 30.08$). The 15 gpm flow rate through the can annulus produces an axial fluid velocity of 2.76 feet/second. The net velocity of the fluid spiraling through the can annulus region is 30.21 feet/second ($[(30.08^2 + 2.76^2)^{1/2} = 30.21]$).

The cans were inspected prior to testing, and after the following hours of operation: 13.5, 104, 302, 622, and 1000. The pre-test can material thickness was measured. Axial profile traces at 4 locations, 90° apart were also obtained on the cans as welded to the test rig components prior to testing and after all inspection points identified above. The accuracy of the thickness measurement instrumentation is within 100 microinches. The accuracy of the profile measurement instrumentation is within 5 microinches. After completion of testing the cans were cut from the test rig components at the circumferential welds and the thickness of the cans were remeasured. These measurements were obtained at the same locations as the axial profile measurements. Based on the known wear over the 1000 hours of testing, the profile traces were then used to estimate wear at specific time and locations on the cans.

Figure 11-11 depicts the rotor can wear at 4 circumferential locations over the can length, and Figure 11-12 depicts the stator can wear over the can length at 4 axial

Figure 11-10 Can Test Rig

Figure 11-11 Rotor Can Wear Over 1000 Hours

Figure 11-12 Stator Can Wear Over 1000 Hours

locations. Referring to Figure 11-11, the maximum rotor can wear is 5.0 mils over 1000 hours at the fluid entrance (inboard end). The rotor can wear reduces to 2.0 to 2.50 mils within 0.25 inch of the inboard end and remains relatively constant at 2.0 to 2.5 mils over the remaining length of the rotor can. The high inlet wear is considered to occur because of the accelerating particles as they enter the can annulus region. The actual Transfer Pump design incorporates a long (0.25 inch) circumferential can weld to compensate for the increased wear in this region.

Therefore, in the Transfer pump rotor this high wear region will occur only in an excess weld region over the rotor end ring where no leakage will take place even if the weld material is worn completely away. Because the rotor can wear is relatively uniform over the entire can length as shown in Figure 11-11, the changes in rotor can wear rate cannot be estimated with the profile traces at intermediate inspection times; however, assuming a constant wear rate of 2.5 mils maximum per 1000 hours or 0.0025 mils/hour maximum, the wear over 10000 hours is estimated to be less than 25.0 mils in the Table 11-1 simulant. Refer to Section 13.0 for a discussion of pump life based on the can wear rates.

Figure 11-12 identifies that the wear on the stator can is between 0.5 and 1.0 mil at the inboard end, increases to 3.5 to 4.0 mils in the middle, and then decreases to 0.25 to 0.5 mil at the outboard end. The wear at both ends of the stator can are less than in the center because the stator can overhangs the rotor can at both ends and the fluid velocities at these ends are significantly less than in the annular gap between the cans. Because of the low wear (compared to the can mid-region) at both ends, the axial profile traces at intermediate inspection points could be used to estimate the wear that occurred between inspections. The results of these intermediate wear estimates are delineated in Table 11-5.

Table 11-5 Stator Can Wear (Mils)

Hours	Loc #1 Wear	Loc #2 Wear	Loc #3 Wear	Loc #4 Wear	Average Wear	Wear Rate mils/hour
0	0	0	0	0	0	-
13.5	0.073	0.051	0.072	0.020	0.054	0.00400
104	0.867	0.547	0.466	0.946	0.707	0.00722
302	1.698	1.281	1.130	1.071	1.295	0.00297
622	2.511	2.721	2.994	3.117	2.836	0.00482
1000	3.771	3.215	3.840	3.622	3.612	0.00205

Table 11-5 identifies that the maximum stator can wear after 1000 hours of operation in the simulant was 3.84 mils. The average stator can wear was 3.61 mils. Note also that the rate of stator can wear is generally decreasing, from 0.00722 mils/hour at 104 hours of operation to 0.00205 mils/hour at 1000 hours. Based on the wear data presented in Table 11-5, the estimated test rig stator can wear over 10000 hours in the Table 11-1 simulant is 22.1 mils ($0.00205 \times [10000 - 1000] + 3.61 = 22.1$). The stator can wear is greater than the rotor can wear (2.50 mils maximum) in the 1000 hour test. This is expected since the forced fluid vortex caused by the spinning rotor would tend to throw larger, heavier particles outward against the stator can and away from the rotor can. Because the rotor and stator cans are the same thickness in the transfer pump (33 mils), the stator can wear is considered to be controlling relative to pump operating life. Refer to Section 13.0 for a discussion of estimated stator can wear relative to Transfer Pump life.

A design feature shown to be important for can life is the 0.25 inch long can to end ring welds on the rotor assembly.

11.1.3 Diffuser Impingement Testing

The diffuser impingement test rig simulates the impingement of impeller flow on the diffusers. The rig consists of test coupons that are held in position relative to a fixed

nozzle that permits simulated process fluid to strike the coupons at a 10° angle. The fluid nozzle velocity was maintained at 80 feet per second and 200 °F. The nozzle exit was 4.0 inches from coupons. The specimens were cast plates of corrosion resistant stainless steel (CF3) and Chromide coated CF3 material. Refer to Figure 11-13 for a view of the diffuser impingement test rig.

The uncoated CF3 test coupon was inspected for wear after 35, 104 and 302 hours of impingement flow. The Chromide coated CF3 coupon was inspected for wear after 43, 112, 302, and 607 hours of testing. Inspection for wear consisted of taking profile traces across the coupon in the zone impinged upon by the nozzle flow and comparing the measurements with previous traces. The accuracy of the profile measurement instrumentation is within 5 microinches. The wear to the uncoated CF3 coupon is summarized in Table 11-6.

Figure 11-13 Diffuser Impingement Test Rig

Table 11-6 Uncoated CF3 Diffuser Impingement Wear (Mils)

Hours	Loc 0.4 Wear	Loc 0.6 Wear	Loc 0.8 Wear	Loc 1.0 Wear	Maximum Wear	Wear Rate mils/hour
0	0	0	0	0	0	-
35	0.30	0.20	0.20	0.70	0.70	0.0200
104	1.40	1.30	1.20	1.30	1.40	0.0101
302	3.00	3.20	3.00	2.90	3.20	0.0091

Based on the testing performed, the estimated 10000 hour wear on the uncoated CF3 coupon in the test rig with the Table 11-1 simulant is 91.5 mils ($0.0091 \times [10000 - 302] + 3.20 = 91.5$).

Inspection of the Chromide coated CF3 coupon revealed no measurable wear after 607 hours of testing. Since the accuracy of the measuring instruments is about 0.05 mil, it is assumed that this amount of wear actually did occur on the Chromide coating. The estimated 10000 hour wear of the Chromide coating is therefore less than 1.0 mil ($0.05 \times 10000 / 607 = 0.82$). Refer to Section 13.0 for a discussion of the Transfer pump life based on hydraulic and impeller seal wear.

The impingement wear testing has revealed the necessity to chromide coat the first and second stage diffuser surfaces directly impinged by the impeller discharge flows. The desirability of coating impeller seal surfaces was also revealed from this testing; therefore, the following impeller seal surfaces will be chromide coated:

1. The suction adapter impeller seal.
2. The first stage impeller upper and lower seals.
3. The first stage diffuser impeller seal.
4. The second stage diffuser impeller seal.
5. The second stage impeller lower seal.

In addition, the eight first stage diffuser axial bolts will be chromide coated since these bolts are directly in the flow stream. The impeller spacer will also be chromide coated.

12.0 FILL AND DRAIN TESTING

12.1 Test Description

Westinghouse EMD had two full-scale plastic and metal models built, that simulated the internal passages of the Transfer Pump. One model had an internal rotor/bearing cavity simulation while the other model depicted the diffuser and outer stator annulus passage of the process fluid. The internal rotor/bearing cavity had an outer shell of clear plastic with a solid brass rotor body. The radial bearings were clear plastic with the four flow slots. The journals were machined on the brass rotor body. The thrust runner was made from stainless steel. The thrust bearing assembly is simulated by stainless steel disks mounted on all-thread supported by a steel base. Purge line locations were placed prototypically above the top radial bearing and at the thrust bearing. These models depicted the design of the Transfer Pump as of January 1995. The full-scale models are shown in Figures 12-1 and 12-2. The models permitted the viewing of the passages and settling out of the simulated process fluid. Also, the flushing ability of purge water was observed.

The test sequence for the internal rotor/bearing cavity model began with the filling of the cavity with simulated process fluid which contains 30 weight percent of Grade #110 silica sand with water. The cavity was filled from the bottom and was permitted to settle. Rotor shaft breakaway torque was measured. The cavities were purged. Rotor breakaway torque was again measured. Videotape of the filling/torque check/purging actions was recorded. The characteristics of the

simulated process fluid action, on how it visually flowed and settled out, was evaluated.

The model that depicted the diffuser and outer stator annulus permitted the viewing of the simulated process fluid through the annulus. The test sequence for this annulus began with filling the annulus with the 30 percent weight silica and water simulation of the process fluid, and observing how it flowed and settled out. The annulus was then back flushed. This testing was over time periods that permitted the process fluid to settle out, so that any evidence of a clogging action was observed. Videotaping of the tests was done to record the characteristics of the process fluid in this annulus.

12.2 Test Results

The testing has been completed on the internal rotor/bearing cavity simulation test model. The following are the results and observations that were made.

- The rotor breakaway torque in the clean dry state was 10 ft-lbs. The breakaway torque with the model filled with clean water was 7.5 ft-lbs.
- The test model, when filled with simulated process fluid and permitted to allow particle settling in the can annulus above the bearing and thrust bearing area, was observed to contain particles in the lower can annulus and lower radial bearing slots. Also particles collected underneath the thrust bearing shoes.

This had a choking affect that did not permit drainage out.

- The breakaway torque of the rotor was not measurable as the settled out particles froze the shaft to keep it from turning. The resisting torque measured was up to 150 ft-lbs.
- The use of purge water through the purge lines dislodged the settled particles to permit drainage out of the bottom of the model. The area underneath the thrust bearing shoes did not fully drain of settled particles. This is a crud trap area in the model. This area could more adequately be flushed depending on the positioning and quantity of the purge water line locations.
- The model drains cleanly if there is no particle settling permitted.
- The breakaway rotor torque, with the model filled with simulant and no particle settling permitted, was approximately 25 ft-lbs.

The testing is also completed on the diffuser and outer stator annulus model. The following are the results and observations that were made.

- The annulus was immediately self draining if the process fluid was not permitted to settle out.
- The process fluid was permitted to settle out in the annulus for a time period

of 15 minutes. The fluid would not immediately self drain. The water would percolate through the settled silica and then eventually discharged out over a time period of three to five minutes. When the fluid discharged it cleared the annulus of any settled particles.

- The process fluid settling operations were carried out for the settling times of 15 minutes, 30 minutes and 45 minutes. The self draining capability of the model for all settling times was the same as described for the 15 minute duration.

Video taping of the model test was done and was made available to WHC.

The test results were used to improve the design and operation of the transfer pump as follows:

The volume between the lower radial bearing and the can annulus region of the motor was increased to ensure that particles will settle out above the lower bearing and below the can annulus. The top of the thrust bearing support plate was sloped to eliminate the flat surface that would collect waste fluid particles. The rotor end ring surfaces were also sloped. Other areas of the pump internals were modified to eliminate flat surfaces such as the second stage diffuser exit, which should preclude plugging of the diffuser passages during a pump shutdown.

The washout grooves in the bearing bores were eliminated and replaced by large by-

pass flow regions to aid in waste drainage. The small motor cooling annular entrance was eliminated to preclude plugging, and larger motor cooling holes were added with a redirected flow pattern.

The two purge line locations were removed in favor of one purge location at the upper bearing. This one purge location, which is still fed from one common water connection, ensures that no short-circuit of the purge flow would by-pass clogged areas that could be present if a two purge location system was used.

The back flushing and purge operations must be performed after every shutdown of the transfer pump to minimize solid particle collection in the pump motor. This flushing and purging will become part of the pump operating requirements.

Figure 12-1 Inner Flow Model Rotor Cavity Can Annulus

Figure 12-2 Outer Flow Model Diffuser/Jacket Annulus

13.0 MAINTENANCE-FREE LIFE EXPECTANCY

The maintenance-free life expectancy evaluation of the Transfer Pump included an analysis of the following items, which are considered the primary modes of pump failure:

- Bearing wear
- Rotor and stator can wear
- Hydraulics wear
- Impeller seal wear
- Motor winding insulation life
- Column assembly fatigue
- Rotor shaft fatigue

The results of the analyses, which are delineated below, indicate the minimum life of the Transfer Pump is 10,000 hours. WEMD therefore concludes that the Transfer Pump design/materials meet the specification minimum life requirement. It is noted that this conclusion is based on wear test results which used a simulant with a 100 Miller number abrasiveness. The pump specification, WHC-S-0238, Revision 1, requires a maximum Miller number of only 50. Since wear is directly proportional to Miller number, it can also be concluded that the actual life of the Transfer Pump, when installed in a hazardous waste tank with a fluid Miller number abrasiveness of 50 or less, will greatly exceed 10,000 hours.

13.1 Bearings

The maintenance-free life expectancy of the bearings considers the items delineated below in the analysis of the radial bearings. The thrust bearing is not considered the limiting bearing because the pads are expected to wear relatively evenly and, therefore, will not produce concave pockets to reduce capacity or increased clearance to cause contact between the close-running rotating and static components.

- The design bearing clearances (0.0055 inch diametral upper bearing and 0.0065 inch diametral lower bearing)
- The bearing capacities at minimum process fluid viscosity (1.0 Centipoise)
- The bearing loads at the minimum process fluid specific gravity (1.0)
- The rate of increase of the bearing clearances at design operating conditions based on the abrasion wear testing results identified in Section 11.0 and the bearing wear calculations in Section 9.0.
- The minimum allowable film thickness considered practical for continuous operation without rubbing contact (estimated to be about 0.0001 inch in 1.0 Centipoise fluid) for pivoted pad radial bearings
- The maximum allowable bearing clearance which precludes unacceptable wear between rotating and static components based on a tolerance stack-up analysis

As identified in Section 9.0 the design loads on the radial bearings in a water environment were calculated to be a function of journal eccentricity in the bearing. Based on the loads and wear rates observed during abrasion testing, bearing wear over 10,000 hours of life was calculated. For the upper radial bearing wear would increase the diametral clearance from 0.0055 inch nominal to 0.014 inch worst case. For the lower radial bearing, wear over 10,000 hours would increase the diametral clearance from 0.0065 inch nominal to 0.028 inch worst case. The bearing film thicknesses in a 1.0 Centipoise fluid environment were found to exceed 0.0001 inch at the maximum wear condition and were therefore acceptable.

Based on an operability evaluation all impeller seals are designed to provide a running radial clearance no less than 0.00075 inch in 95 percent of the Transfer Pump motor unit assemblies under all specification operating conditions. This operability evaluation includes a 3-dimensional tolerance stack-up analysis using design bearing clearances and design component tolerances, an operational load shaft deflection analysis, an assembly deflection analysis (due to an eccentric loading on the impeller bolt from component non-parallelisms), and various operating combinations of fluid viscosity and specific gravity.

The operability evaluation was repeated with increasing values of bearing clearances to determine impeller seals clearance/contact as a function of bearing clearance. Two events happen before bearing wear becomes unacceptable. The first is that bearing wear reduces the minimum impeller seal operating clearance from 0.00075 inch radial to zero. When the bearing wear reduces an operating seal clearance to zero, the pump is still operable, but will experience an increase in power draw that may (or may not) be noticeable. Bearing wear (and seal wear) can acceptably continue until clearances get so large that the pump can no longer produce the required head and flow; this is the second and limiting event at which failure is arbitrarily considered to occur. Based on an internal circulation sensitivity study, the limit is considered to be an increase of 0.025 inch radial clearance to a 0.050 inch radial (0.100 inch diametral) total clearance at each impeller seal. With a 0.050 inch radial clearance at all seals the pump must operate at 3900 rpm to produce the design head and flow. This 40 rpm increase is acceptable. The analysis indicates that impeller seal rubbing contact may occur with an increase in bearing clearance (wear) as little as 0.0005 inch radially. However, with the 10,000 hour radial clearances of 0.007 inch (upper bearing) and 0.014 inch (lower bearing) the maximum expected radial seal wear is only 0.018 inch. This analysis was performed with the 3-dimensional tolerance stack-up program used to calculate design clearance. This is less than the 0.025 inch allowable wear and is acceptable. This conservatively assumes no load carrying capacity in the seal.

13.2 Motor Cans

The maintenance-free life expectancy of the motor rotor and stator cans is based on the following items in the evaluation:

- The average design wall thickness of the rotor and stator cans (0.033 inch)
- The minimum allowable thickness of the cans to ensure integrity of their hermetic seals to prevent the entry of process fluid into the rotor squirrel cage and into the stator windings.
- The rate of wear of the rotor and stator cans based on the abrasion testing described in Section 11.0

Loss of integrity of the stator/rotor cans due to abrasive wear is expected to occur first in the stator can due to the higher wear rate, as identified in Section 11.0. The expected failure mode is rupture of the can in a very localized area over a stator

punching slot due to pressure. This is expected to occur when the ultimate stress of the can material is reached.

Over the stator slots, the minimum required can thickness is:

$$\sigma = 0.5 P \left(\frac{b}{t} \right)^2$$

or

$$t = b \sqrt{\frac{0.5P}{\sigma}}$$

Where: P is pressure, 325 psi at design

σ is minimum ultimate stress, 115,000 psi for Hastelloy C-276 material

b is the maximum stator slot width, 0.173 inch

$$t = 0.173 \sqrt{\frac{(0.5)(325)}{115,000}}$$

$$t = 0.0065 \text{ inch can thickness}$$

The minimum acceptable wall thickness is 0.0065 inch. With a 0.033 inch can, the maximum allowable wear is 0.0265 inch ($0.033 - 0.0065 = 0.0265$).

In Section 11.1.2 the estimated maximum can (rotor or stator) wear was identified as 0.0221 inch in the simulant test environment. It is estimated that the can wear is proportional to speed squared. This is based on diffuser impingement testing performed with simulant different from that specified in Table 11-1. Because the can wear testing was performed at 3600 rpm, the expected 10000 hour wear at the 3860 pump design speed is 0.0254 inch ($0.0221 \times [3860 / 3600]^2 = 0.0254$). This is less than the 0.0265 inch allowable can wear ($0.033 - 0.0065 = 0.0265$). The estimated can life at design operating conditions is 10,433 hours (10000×0.0265

/ 0.0254 = 10433) based on the tested simulant. This meets the 10,000 hour design life of the pump. Note that with a slurry which has a Miller number of 50 in lieu of the 100 that the test simulant had, the expected can wear over 10,000 hours is only 0.0127 inch ($.0254 \times 50 / 100 = 0.0127$).

Failure of a rotor can in a localized spot due to material erosion will have no operational consequences for the pump over the short term. Over the long term, the waste tank chemical environment may attack the copper bars and connection rings sufficiently to cause an open circuit in the squirrel cage and failure of the motor to turn. Decontamination of the rotor would be more difficult because the rotor can would have to be removed. In any event, a rotor can failure presents less of a concern than a stator can failure.

A localized failure of the stator can due to erosion would allow process fluid to enter the stator cavity. The Transfer Pump stator is potted with a material that is similar to a soft concrete. Assuming there are only small, local voids in the potting material, the process fluid may take a considerable amount of time (months) to reach the windings. If the fluid collects in a void which contains turns of wire in the same electrical phase, a low voltage turn-to-turn short will occur which will cause a phase current unbalance in the motor during operation. This will probably result in a shut-down of the motor by the variable frequency drive system. This type of failure would probably not result in significant sparking or melting of copper.

A collection of process fluid in a stator potting void which causes a phase-to-ground or phase-to-phase short will cause sparking and copper melting, possibly violently. However, even if an explosive gas were present in the void, the damage would be localized within the motor because any voids in the potting will be small. There would be nowhere for an explosive gas flame-front to propagate to, and the motor shell/stator can/potting would contain the damage to a small part of the stator winding.

Another possible result of a stator can leak is that the process fluid will by-pass the

potting and find its way up to the connection between the motor leads and the power cable in the column assembly. This would cause a phase-to-phase short which would produce sparks and molten copper. The void space in this region is much larger (a 1.50 inch diameter cable in a 2.00 inch diameter pipe). The presence of an explosive gas in this region during the short circuit would cause more damage than in the stator winding area, but propagation into the tank region is not likely because the flame/overpressure would be contained with both a 2.00 inch pipe and an 8.00 inch pipe. A more detailed evaluation of the column assembly relative to an explosive gas event is beyond the scope of this effort.

13.3 Hydraulics

The maintenance-free life expectancy of the pump hydraulics which include the diffuser high velocity regions and the impeller seals is based on the following items:

- The allowable amount of diffuser wall that can be eroded away before the diffuser wear coating is removed (0.015 inch minimum)
- The allowable amount of impeller seal wear over the operating life of the pump to ensure adequate hydraulic performance (0.006 inch minimum as identified in Section 13.1)
- The rate of wear of the diffuser hydraulics based on the abrasion testing described in Section 11.0

As identified in Section 11.1.3 the expected wear of the Chromide coating over 10,000 hours is less than 0.001 inch based on diffuser impingement testing. The diffuser discharge areas which sustain high direct impingement velocities will be coated with the Chromide. Assuming an allowable wear of 0.015 inch in these locations, the expected life at these locations is 150,000 hours minimum ($0.015 \times 10000 / 0.001 = 150000$). In the impeller seal regions the minimum allowable

wear is 0.003 inch each side, or 0.006 inch total and, therefore, the minimum expected wear life is 30,000 hours ($0.003 \times 10000 / 0.001 = 30000$) assuming impingement flow is involved in seal wear.

If it is assumed that the impeller seals act more like the can annulus region with swirling through flow, the following analysis applies:

The impingement testing indicated that the Chromide coating was 128 times more abrasion resistant than the base CF3 material (0.05 mils of wear over 607 hours versus 3.2 mils of wear over 302 hours). Assume that the abrasion resistance of CF3 is similar to the abrasion resistance of the Hastelloy C276 can material because the hardnesses are similar. The estimated wear on the Hastelloy cans is 0.0254 inch over 10,000 hours. The 10,000 wear on the Chromide coating under similar circumstances would be 0.0002 inch over 10,000 hours ($0.0254/128 = 0.0002$). The allowable wear of the seals is 0.003 inch each side. Therefore, the expected life of the impeller seals is 150,000 hours ($10,000 \times 0.003/0.0002 = 150,000$) assuming annulus flow-type wear.

Under certain circumstances, when an impeller seal clearance is very small (less than several mils), the seal may act somewhat like a bearing with some of the particulates being driven into a rotating, converging wedge of fluid in lieu of a jet of fluid impacting a surface. Based on the bearing wear studies identified in Section 11.0, the Chromide coating does not hold up very well as a bearing compared to the diamond coating. However, since the seals are not designed to provide load-carrying capacity, the abrasive action on the seal will be self-limiting. That is, once part of the seal has worn somewhat due to the abrasive particles, the damage will stop because the minimum seal clearance will get large and pass the particles through. Conservatively assuming that up to 0.004 inch of wear can occur at a seal due to bearing-like action, at least 0.002 inch of Chromide-coating remain to be worn away by impact-like wear (0.006 inch minimum can be worn away before head-flow performance may be affected). At the established impact wear rate of 0.001 inch per 10,000 hours of operation, the seal life is expected to exceed 20,000 hours

$(0.002 \times 10000 / 0.001 = 20000)$.

13.4 Motor Windings

The maintenance-free life expectancy of the motor windings is based on the following information:

- The calculated maximum winding surface temperature (MWST) of the motor stator insulation at design point operation.
- The minimum expected insulation life at that temperature based on testing. For the Class N insulation system used in this application the insulation life is 40 years at an MWST of 200 °C (392 °F) and the life roughly decreases by half for every 10 °C (18 °F) increase above 200 °C (392 °F).

As identified in Section 7.1.1 the design point MWST of the transfer pump stator winding insulation system is calculated to be 205 °C (401 °F). Based on test results, the expected life of the insulation system is approximately 22 years (190,000 hours) of continuous operation at the design point.

13.5 Fatigue

The maintenance-free life expectancy based on fatigue considers the two components subjected to substantial cyclic loading, the column assembly and the rotor shaft. The column assembly cyclic load is induced by the lateral jet impingement force. Section 8.3 identifies that the column can withstand an infinite number of cycles with the 150 lb specification load. This section also notes that the specification limit of 10,000 cycles is reached with a substantially increased jet impingement force of 2,155 lbs. The column assembly, therefore, has an infinite

fatigue life with the specified jet impingement force.

Rotor shaft fatigue is analyzed in Section 8.7. The minimum factor of safety (FS) for the shaft loads is determined to be 3.5. An FS greater than 2.0 is normally considered acceptable for high cycle fatigue. Therefore, the shaft is considered to also have an infinite fatigue life.

14.0 DECONTAMINATION AND MAINTAINABILITY

14.1 Pump Assembly Decontamination

The pump/column assembly will be back flushed with clean water through the process fluid 3-inch schedule 40 by the Customer upon completion of process fluid transfer. The purge and dilution lines will be flushed with water also. These actions will disperse the radioactive contaminants back into the waste tank. It is recommended that at least 100 gallons of water be back-flushed through the motor after each use. After this the pump/column assembly can be removed from the waste tank. While the column is being removed from tank, high pressure water is sprayed on the outside surfaces of the assembly to wash all radioactive waste from the outer surfaces. All of the wetted surfaces have a 63 rms surface finish which permit cleaning with high pressure water. There are no significant pockets or traps within the pump/column assembly that could retain radioactive waste. The column assembly has no flanges to concentrate solid waste and most of the exposed surface is a round eight inch pipe. Also, the motor unit assembly has no protruding flange to trap radioactive waste. The only exposed bolts are the bolts between the column and motor unit assemblies and the bolts on the bottom of the motor unit hydraulics. Thus most of the outer pump surfaces are smooth and relatively easy to flush.

14.2 Pump Assembly Handling

The transfer pump will be shipped to WHC as one unit, completely assembled except the suction adapter screen. The pump will also be shipped with a rotor

locking device installed. The top end of the pump column is tack welded in place to permit field column length adjustment. Plastic cap plugs are placed in all openings:

These cap plugs may be additionally secured with weather-resistant pressure sensitive tape. The pump will be enclosed on polyethylene and crated. The shipping container shall be identified with lifting points and center of gravity.

The pump can be lifted from the container using two nylon slings equally spaced about the pump center of gravity. To prepare the pump for field column length adjustment, place pump horizontally in rolls or V-blocks. Instructions for field column length adjustment will be provided in the operation manual. After column length adjustment is complete, remove all plastic cap plugs. Remove the rotor locking device and assemble the suction adapter screen. (Ref. para 5.3 of WEMD spec. 85310XX.)

To rotate the pump to a vertical position, first assemble the group two lifting ring assembly of the WEMD transport cart 05-50093 to the bottom end of the pump. (Ref. para 6.3 of WEMD spec. 85310XX.) Assemble pump into transport cart with a support block at the upper end of the column. Remove the nylon slings and assemble WHC lifting bails to the top end adaptor flange. Lift the pump from a horizontal position to a vertical position, allowing the transport cart to roll. When the pump is in the vertical position, remove the cart lift ring assembly.

14.3 Column Assembly Maintainability

The column assembly is effectively maintenance free. The outer wall cylinder is welded to the pump mounting flange and transition piece. These welded joints preclude process fluid from entering the column assembly. The other pipes are not welded to the mounting flange but are sealed with RTV to prevent contamination of the column internals. Therefore, no decontamination of the column assembly internals is required.

If the motor unit assembly is removed during the life of the column, the graphite gasket between the motor and the transition piece must be replaced. The o-rings on the motor unit electrical conduit must also be replaced. When reassembling, lockplates must be used after bolt tightening to prevent loss of preload.

14.4 Motor Unit Assembly Maintainability

The Model 2A-513-A1 Transfer Pump is designed to operate in a hazardous waste tank environment maintenance-free for over 10,000 hours in a 10 year period. In the event that pump refurbishment is desired after the full operating life is achieved or after an incident occurs which renders the pump inoperable, the Transfer Pump is capable of being refurbished. From a normal wear viewpoint, the most probable causes for refurbishment are worn bearings and/or worn impeller seals. The scenarios for these wear modes are discussed in the following paragraphs.

Worn transfer pump bearings would be characterized by higher rotating assembly vibration which normally can be picked up by an accelerometer on the pump. Worn bearings will probably not result in significantly higher motor power/current draw until contact is made at the impeller seals. The impeller seal clearances are slightly larger than the bearing clearances (two to three times) and contact can be foreseen after bearing wear occurs. In contrast, the cans are highly unlikely to ever contact each other because can clearance is almost 50 times the bearing clearance. In the case of the Transfer Pump these increased vibration signals might be masked until rubbing contact with the impeller seals occurs for the following reasons. The accelerometer will be installed on the top of the pump mounting flange, more than 40 feet from the rotating assembly. The transmissibility of rotating assembly frequencies through the column assembly to the mounting flange accelerometer will not be known until lead unit testing. Therefore, the ability of the mounting flange accelerometer to pick up subtle changes in vibration due to increases in bearing clearance is not known at this time. The constituency of the process fluid will vary from tank to tank and

from layer to layer within a tank. These density/viscosity/solids content changes will affect the vibration level of the rotating assembly and may mask any vibration level changes due to bearing clearance changes. Also, the process fluid changes can cause power/current changes in the motor far greater than bearing wear would. Therefore, it is highly unlikely that any instrumentation will pick up significant bearing wear, thrust or radial, until metal-to-metal contact is made in the motor unit, and this contact is most likely at the impeller seals.

Contact between the impellers and the static hydraulics will occur after significant bearing wear has occurred as previously mentioned. The pump design is such that contact between the rotor and stator cans cannot occur due to bearing wear. The first contact between rotating and static parts are designed to occur at the impeller seals. Impeller seal contact will not only cause significantly higher vibration levels in the pump but will also cause higher current/power draw from the pump. A spectrum analysis of a Transfer Pump operating with impeller seal contact will show a "picket fence" affect in a plot of vibration level versus frequency with vibration spikes at pump operating frequency intervals (i.e., 3600 rpm would reveal spikes at 60 Hz, 120 Hz, 180 Hz, etc.). There may also be noise spikes at fractions of pump operating frequency intervals, depending on the number of times the impeller seals contact in one revolution. For example, if contact occurs twice during each revolution for a pump operating at 3600 rpm, then noise spikes would occur at frequencies of 30 Hz, 60 Hz, 90 Hz, etc.

When seal contact occurs current/power will increase due to larger frictional loads. These current increases will be noticeable but may not be significantly different from normal background fluctuations if the process fluid being pumped has large variabilities in viscosity and density. However, the current fluctuations should be readily apparent when process fluid with relatively uniform properties is pumped.

As seal rubbing occurs the clearance between the static and rotating components of the impeller seal will increase due to wear. As bearing wear increases more impeller seals will contact and seal contact area will increase. The end result of increased bearing and seal wear (in addition to increased noise and current/power) is increased fluid leakage around the seals. For a while this increased leakage can be compensated for by increasing the pump operating speed. This can be accomplished because the pump is equipped with a variable frequency drive (VFD). However, after significant seal wear has occurred, it will become impossible to increase the pump speed sufficiently (due to frequency limitations built into the VFD to protect the pump) to produce the required flow. At this point the pump will require replacement. It is noted that the bearing and seal wear required to render the pump inoperative (around 0.010 inch) is an order of magnitude less than the bearing and seal wear required to cause rotor-to-stator can contact. Therefore, no bearing wear failure can cause loss of integrity of a rotor or stator can.

While the above discussion has involved mainly the radial bearings, thrust bearing wear would have a similar effect. Wear of the deflection pad thrust bearing film riding surfaces, which are ground flat, will have little effect on performance until the pads erode down to the metal pad holders. At this time bearing film will be disrupted and lost, and contact will occur between the pads and the thrust runner. When this occurs noise and power levels will jump, and the VFD logic will probably trip the pump off-line. Restart of the pump may be possible, but for only a short period as power demand rises from a degrading thrust bearing. Radial bearing loads may increase during thrust bearing failure, but probably not enough to cause significant radial bearing wear. Impeller seals may also contact during a thrust bearing failure and some wear may occur. Because the radial bearings will likely sustain little if any damage during a thrust bearing failure, no rotor or stator can failure will occur.

In addition to contact wear, impeller seals can wear by process fluid erosion. The wear will result in larger seal clearances and leakage. The symptoms and results are

the same as discussed above for contact wear. The pump speed must be increased to obtain the required flow. Power draw increases as hydraulic efficiency drops and friction losses increase (due to higher operating speeds). Eventually, the required flow cannot be achieved within the allowable operating frequency range of the VFD and the pump must be replaced.

There are several types of incidents which can cause the pump to be rendered inoperable in the waste tank. One possible cause is a locked rotor condition due to solids build-up in the rotor and/or hydraulics cavities or ingestion of a foreign object. This failure mode would have to be non-responsive to purge water flow, backflushing, and rotor "rocking", reversing the attempted start direction by changing the position of a phase rotation switch in the VFD unit. These corrections could be applied individually or concurrently, but if they were unsuccessful the pump would have to be replaced. The potential for this type of incident to occur can be minimized by following the Installation and Operation Manual to be provided with the Model 2A-513-A1 Transfer Pumps. The manual will recommend that the following should be accomplished to while the pump is installed in the waste tank to avoid rotor lockup.

When the pump is first installed in the tank, purge water flow should be initiated as soon as practical. The pump should be operated for a short time after installation is complete to characterize the current draw (motor torque). The purge water should be turned on periodically, and the motor should be turned on for a short time shortly thereafter to evaluate current draw (torque) versus time. If current draw (torque) is increasing with time the purge water interval should be shortened as necessary to stabilize current (torque required). Immediately after the pump has completed a transfer of process fluid, the pump should be backflushed through the main discharge line with as high a flow rate for as long as waste tank conditions permit. This backflush should discharge most of the heavy particulates from the motor unit assembly. The purge water flow and short pump operating periods should be re-initiated as identified above. The purge/operating periods should be adjusted to ensure motor current (torque) remains in the acceptable zone. It is noted that these short

operating periods do not require process fluid to be discharged from the tank. It will be acceptable to operate the pump at low speeds which do not produce sufficient head to pump fluid out of the tank. This shutoff head speed will be sufficient to circulate process fluid in the motor and will not damage the motor if performed for only short periods of time.

After the pump has been removed from the tank and separated from the column assembly for maintenance the pump should be completely disassembled into the following components: stator assembly, rotor assembly, thrust bearing assembly, second stage diffuser, first stage diffuser, suction adapter, inlet screen, thrust runner, second stage impeller, first stage impeller, impeller spacer, impeller bolt, second stage diffuser bolts, suction adapter bolts, lockplates, lockcups, purge water lines and the column gasket. Note that to disassemble the motor unit assembly the impeller bolt lockpin had to be drilled out, the lockplate tabs must be bent and the lockcups must be unswaged from their housings.

The lockplates, lockcups and column gasket are not reusable and must be replaced. The impeller seals can be inspected remotely with no-go gages. If the seals are too worn the impeller should be replaced. The seals on the diffusers can also be inspected remotely with no-go gages. If the seals are worn but the rest of the diffuser is acceptable, seal inserts can be installed (machining and heating may be required), and the diffuser can be reused. The rotor radial bearing journals can be remotely inspected for wear and replaced if desired. Journal replacement would require heating to approximately 250-300 °F. The radial bearings in the stator assembly can be inspected remotely and replaced if unacceptable. Heating to approximately 400 °F would be required. If the pump had not operated beyond the estimated wear life of the cans (see Section 13.2) the rotor assembly and stator assembly can be reused. As an alternate, the can thicknesses can be evaluated by inspection and a determination made whether to reuse the rotor and/or stator based on estimated remaining can life. The thrust bearing pads and thrust runner can be inspected for wear and reused or replaced as appropriate.

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been divided into smaller sections.**

Section 2 of 2

Document Information			
Document #	RPP-5117	Revision	0
Title	DESIGN ACCEPTANCE SUMMARY REPORT FOR THE NEW GENERATION TRANSFER PUMP (NGTP)		
Date	09/28/1999		
Originator	IRONS J	Originator Co.	COGEMA
Recipient		Recipient Co.	
References			
Keywords			
Projects	RPP		
Other Information			

15.0 DRAWINGS

<u>Figure</u>	<u>Drawing Number</u>	<u>Title</u>
15-1	4D97586 Revision 1	Outline - Lead Unit
15-2	4D97585 Revision 3	General Assembly
15-3	4D97539 Revision 1	Winding Diagram
15-4	4D97587 Revision 7	Motor Unit Assembly
15-5	4D97536 Revision 3	Column Assembly
15-6	4D97594 Revision 2	Column Extension Assembly
15-7	4D97525 Revision 3	Rotor Assembly and Final Machining
15-8	4D97534 Revision 4	Final Stator Assembly
15-9	4D97580 Revision 3	Impeller Casting
15-10	4D97529 Revision 2	First Stage Impeller Machining
15-11	4D97530 Revision 2	Second Stage Impeller Machining
15-12	4D97579 Revision 4	First Stage Diffuser Machining
15-13	4D98151 Revision 3	Stator Modification
15-14	4D97528 Revision 5	Second Stage Diffuser Machining
15-15	4D97584 Revision 1	Suction Adapter Casting
15-16	4D97794, Revision 3	Suction Adapter Machining
15-17	4D97588, Revision 4	Thrust Bearing Assembly
15-18	4D98083, Revision 1	Thrust Runner Assembly
15-19	4D98081, Revision 2	Lower Radial Bearing Assembly
15-20	4D98089, Revision 2	Upper Radial Bearing Assembly

DATE	11-1-76	BY	J. D. BARTO	CHKD	J. D. BARTO
REV	1	DATE	11-1-76	BY	J. D. BARTO
REV	2	DATE	11-1-76	BY	J. D. BARTO
REV	3	DATE	11-1-76	BY	J. D. BARTO
REV	4	DATE	11-1-76	BY	J. D. BARTO
REV	5	DATE	11-1-76	BY	J. D. BARTO
REV	6	DATE	11-1-76	BY	J. D. BARTO
REV	7	DATE	11-1-76	BY	J. D. BARTO
REV	8	DATE	11-1-76	BY	J. D. BARTO
REV	9	DATE	11-1-76	BY	J. D. BARTO
REV	10	DATE	11-1-76	BY	J. D. BARTO
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REV	20	DATE	11-1-76	BY	J. D. BARTO
REV	21	DATE	11-1-76	BY	J. D. BARTO
REV	22	DATE	11-1-76	BY	J. D. BARTO
REV	23	DATE	11-1-76	BY	J. D. BARTO
REV	24	DATE	11-1-76	BY	J. D. BARTO
REV	25	DATE	11-1-76	BY	J. D. BARTO
REV	26	DATE	11-1-76	BY	J. D. BARTO
REV	27	DATE	11-1-76	BY	J. D. BARTO
REV	28	DATE	11-1-76	BY	J. D. BARTO
REV	29	DATE	11-1-76	BY	J. D. BARTO
REV	30	DATE	11-1-76	BY	J. D. BARTO
REV	31	DATE	11-1-76	BY	J. D. BARTO
REV	32	DATE	11-1-76	BY	J. D. BARTO
REV	33	DATE	11-1-76	BY	J. D. BARTO
REV	34	DATE	11-1-76	BY	J. D. BARTO
REV	35	DATE	11-1-76	BY	J. D. BARTO
REV	36	DATE	11-1-76	BY	J. D. BARTO
REV	37	DATE	11-1-76	BY	J. D. BARTO
REV	38	DATE	11-1-76	BY	J. D. BARTO
REV	39	DATE	11-1-76	BY	J. D. BARTO
REV	40	DATE	11-1-76	BY	J. D. BARTO
REV	41	DATE	11-1-76	BY	J. D. BARTO
REV	42	DATE	11-1-76	BY	J. D. BARTO
REV	43	DATE	11-1-76	BY	J. D. BARTO
REV	44	DATE	11-1-76	BY	J. D. BARTO
REV	45	DATE	11-1-76	BY	J. D. BARTO
REV	46	DATE	11-1-76	BY	J. D. BARTO
REV	47	DATE	11-1-76	BY	J. D. BARTO
REV	48	DATE	11-1-76	BY	J. D. BARTO
REV	49	DATE	11-1-76	BY	J. D. BARTO
REV	50	DATE	11-1-76	BY	J. D. BARTO
REV	51	DATE	11-1-76	BY	J. D. BARTO
REV	52	DATE	11-1-76	BY	J. D. BARTO
REV	53	DATE	11-1-76	BY	J. D. BARTO
REV	54	DATE	11-1-76	BY	J. D. BARTO
REV	55	DATE	11-1-76	BY	J. D. BARTO
REV	56	DATE	11-1-76	BY	J. D. BARTO
REV	57	DATE	11-1-76	BY	J. D. BARTO
REV	58	DATE	11-1-76	BY	J. D. BARTO
REV	59	DATE	11-1-76	BY	J. D. BARTO
REV	60	DATE	11-1-76	BY	J. D. BARTO
REV	61	DATE	11-1-76	BY	J. D. BARTO
REV	62	DATE	11-1-76	BY	J. D. BARTO
REV	63	DATE	11-1-76	BY	J. D. BARTO
REV	64	DATE	11-1-76	BY	J. D. BARTO
REV	65	DATE	11-1-76	BY	J. D. BARTO
REV	66	DATE	11-1-76	BY	J. D. BARTO
REV	67	DATE	11-1-76	BY	J. D. BARTO
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REV	79	DATE	11-1-76	BY	J. D. BARTO
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REV	81	DATE	11-1-76	BY	J. D. BARTO
REV	82	DATE	11-1-76	BY	J. D. BARTO
REV	83	DATE	11-1-76	BY	J. D. BARTO
REV	84	DATE	11-1-76	BY	J. D. BARTO
REV	85	DATE	11-1-76	BY	J. D. BARTO
REV	86	DATE	11-1-76	BY	J. D. BARTO
REV	87	DATE	11-1-76	BY	J. D. BARTO
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REV	90	DATE	11-1-76	BY	J. D. BARTO
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REV	94	DATE	11-1-76	BY	J. D. BARTO
REV	95	DATE	11-1-76	BY	J. D. BARTO
REV	96	DATE	11-1-76	BY	J. D. BARTO
REV	97	DATE	11-1-76	BY	J. D. BARTO
REV	98	DATE	11-1-76	BY	J. D. BARTO
REV	99	DATE	11-1-76	BY	J. D. BARTO
REV	100	DATE	11-1-76	BY	J. D. BARTO

NOTES:

- 1- THIS DRAWING HAS NO MAJOR OR CRITICAL CHARACTERISTICS AS DEFINED BY Q510062.
- 2- CONDUCT PUMP GENERAL ASSEMBLY ELECTRICAL TEST PER 348A55.
- 3- TAPING TERMINOLOGY PER 160A670.
- 4- LUBRICATE THREADS WITH LUBRICANT (ITEM 5). TIGHTEN IN ACCORDANCE WITH TORQUE TABLE.
- 5- LEAK TEST POWER LEAD SEALS PER 348A52 PRIOR TO INSTALLING CAP SCREW (ITEM 26).
- 6- ROTOR LOCKING DEVICE (ITEM 18) INSTALLED FOR TRANSPORTATION ONLY. REMOVE ROTOR LOCKING DEVICE PRIOR TO PUMP OPERATION.
- 7- AFTER REMOVING ROTOR LOCKING DEVICE (ITEM 18), INSTALL SCREEN (ITEM 32), LOCKPLATES (ITEM 31) AND CAPSCREWS (ITEM 30). LOCK SCREEN ASSEMBLY LOCKPLATES (ITEM 31).

TORQUE TABLE			
ITEM	THREAD SIZE	TORQUE FT-LBS.	TORQUE IN-LBS.
4	.500-13	60-65	---
17	.190-32	---	8-10
22	.375-16	6-8	---
26	.190-24	---	8-10
30	.250-20	---	70-75

Figure 15-2

E.M. 6673, Rev. 3
15-3

A- EPDM MATERIAL PER SEALS EASTERN INC. COMPOUND NUMBER 7331. EPDM COMPOSITION SHALL CONTAIN NO NATURAL RECLAIMED RUBBER OR OTHER SYNTHETIC ELASTOMER. PROCESSING AIDS AND PLASTICIZERS SHALL NOT EXCEED 5 BY WEIGHT PER HUNDRED RUBBER. COMPOUND SHALL BE PEROXIDE CURED.

O-RINGS SHALL BE PACKAGED WITH CURE DATE MARKED. SEALS EASTERN INC. RED BANK NEW JERSEY OR EQUIVALENT.

B- BOLT TO BE PER ASTM-A-193, CLASS 2, GRADE B8M.

C- WELD FILLER MATERIAL SIZE SPECIFIED IN WELD SPEC.

D- B.O.: FROM McMASTER-CARR CAT. NO. 5520K69 OR EQUIV.

E- ENDEVCO MODEL 7704A-50 ISOSHEAR ACCELEROMETER WITH A MODEL 2981-3 STAINLESS STEEL MOUNTING STUD 10-32 TO 10-24 SIZE. B.O. FROM ENDEVCO CORPORATION, SAN JUAN CAPISTRANO, CA 92675.

F- B.O. FROM McMASTER-CARR CAT. NO. 92314A827 OR EQUIV.

G- B.O. FROM McMASTER-CARR CAT. NO. 92146A011 OR EQUIV.

H- B.O. FROM McMASTER-CARR CAT. NO. 92146A031 OR EQUIV.

J- B.O. FROM McMASTER-CARR CAT. NO. 92196A240 OR EQUIV.

K- MATERIAL PER 8715A24H01 MAY BE USED AS AN ALTERNATE FOR ITEM 9.

L- B.O. FROM ELECTOLOCK INC., 16938 PARK CIRCLE DRIVE, P.O. BOX 368 CHAGRIN FALLS, OHIO 44022, OR EQUIV.

M- B.O. FROM HELICOFLEX COMPONENTS DIVISION, COLUMBIA, SC OR EQUIVALENT.

N- B.O. FROM McMASTER-CARR CAT. NO. 92198A628 OR EQUIV.

P- ORDERING INFORMATION:
(1) MATERIAL: TYPE L
(2) TEMPER: H58 (HARD DRAWN)
(3) PURCHASE IN STRAIGHT LENGTHS.

Q- CUT TO FIT AT ASSEMBLY.

R- TWISTED SHIELDED THERMOCOUPLE EXTENSION WIRE B.O. FROM OMEGA ENGINEERING, INC., STAMFORD, CT 06907, MODEL EXFF-J-20-TW5H

S- B.O. FROM ANIXTER CO., 10 PARKWAY VIEW DRIVE, PITTSBURGH, PA 15205, OR EQUIVALENT.

QTY	DESCRIPTION	UNIT	QTY	DESCRIPTION	UNIT
1	500 FT. SPOOL OF THERMOCOUPLE EXTENSION WIRE	4 Q5	1	500 FT. SPOOL OF THERMOCOUPLE EXTENSION WIRE	4 R
1	FT. OF .375 I.D. X 0.030 WALL COPPER TUBING	4 P0	1	FT. OF .375 I.D. X 0.030 WALL COPPER TUBING	4 P0
1	LOCKPLATE	8579C47G01	1	LOCKPLATE	8579C47G01
1	SCREEN BOLT	250-20 X .62 L.G. HEX HD	1	SCREEN BOLT	250-20 X .62 L.G. HEX HD
1	KAPTON TAPE	YDS. OF 0.75 WIDE KAPTON TEMP-R-TAPE (K350)	1	KAPTON TAPE	YDS. OF 0.75 WIDE KAPTON TEMP-R-TAPE (K350)
1	CAP SCREW	(.190-24 X .375 L.G. SOC HD)	1	CAP SCREW	(.190-24 X .375 L.G. SOC HD)
1	O-RING	PART NO. H-304822, TYPE HN 220 (GOLD PLATED)	1	O-RING	PART NO. H-304822, TYPE HN 220 (GOLD PLATED)
1	PACKING PLUG	(EPDM)	1	PACKING PLUG	(EPDM)
1	LOCKWASHER	(.375 DIA. HELICAL SPRING)	1	LOCKWASHER	(.375 DIA. HELICAL SPRING)
1	CAP SCREW	(.375-16 X 1.50 L.G. HEX HD)	1	CAP SCREW	(.375-16 X 1.50 L.G. HEX HD)
1	PLATE WASH	(.190 DIA.)	1	PLATE WASH	(.190 DIA.)
1	LOCKWASHER	(.190 DIA.)	1	LOCKWASHER	(.190 DIA.)
1	REFRIG LK6 DVC	(.190-32 X .375 L.G.)	1	REFRIG LK6 DVC	(.190-32 X .375 L.G.)
1	ACCELEROMETER	PIEZOELECTRIC	1	ACCELEROMETER	PIEZOELECTRIC
1	NAMEPLATE		1	NAMEPLATE	
1	WELD METAL		1	WELD METAL	
1	POWER CABLE	FT. OF 3-CONDUCTOR	1	POWER CABLE	FT. OF 3-CONDUCTOR
1	VARNISH	0.25 GAL. DC 804 (BOND)	1	VARNISH	0.25 GAL. DC 804 (BOND)
1	GLASS BINDER	YDS. OF 1.005 X .750 WIDE UNTREATED GLASS TAPE	1	GLASS BINDER	YDS. OF 1.005 X .750 WIDE UNTREATED GLASS TAPE
1	O-RING	PARKER SIZE NO. 2-216 EPDM 80 DUREMETER	1	O-RING	PARKER SIZE NO. 2-216 EPDM 80 DUREMETER
1	GROUND INSUL	YDS. OF 1.005 X .500 WIDE GLASS-MICA-GLASS TAPE	1	GROUND INSUL	YDS. OF 1.005 X .500 WIDE GLASS-MICA-GLASS TAPE
1	BRAZE ROD	0.500 X 0.375 COPPER REDUCER	1	BRAZE ROD	0.500 X 0.375 COPPER REDUCER
1	LOCKPLATE	FEL-PRO N-5000	1	LOCKPLATE	FEL-PRO N-5000
1	CAP SCREW	.500-13 X 6.00 L.G. HEX HD	1	CAP SCREW	.500-13 X 6.00 L.G. HEX HD
1	CARBON BASKET	GRAPHITE	1	CARBON BASKET	GRAPHITE
1	COLUMN ASSY		1	COLUMN ASSY	
1	MOTOR UNIT ASSY		1	MOTOR UNIT ASSY	
1	PART NAME		1	PART NAME	
1	GROUP CLASS		1	GROUP CLASS	
1	IDENT NOTE		1	IDENT NOTE	
1	TOLERANCE & MACHINING NOTES		1	TOLERANCE & MACHINING NOTES	
1	STANDARDS, PRACTICES & INTERPRETATIONS		1	STANDARDS, PRACTICES & INTERPRETATIONS	
1	UNSPECIFIED GEOMETRIC TOLERANCES TO FALL WITHIN LIMITS OF SIZE DIMENSIONS.		1	UNSPECIFIED GEOMETRIC TOLERANCES TO FALL WITHIN LIMITS OF SIZE DIMENSIONS.	
1	ALL UNFINISHED SURFACES TO BE CHAMFERED		1	ALL UNFINISHED SURFACES TO BE CHAMFERED	
1	RADIUS OR CHAMFER ALL EDGES		1	RADIUS OR CHAMFER ALL EDGES	
1	FILLET RADIUS		1	FILLET RADIUS	
1	ANGULARITY		1	ANGULARITY	
1	SURFACES		1	SURFACES	
1	CHAMFERS		1	CHAMFERS	
1	MAXIMUM SURFACE ROUGHNESS 250 IN. (1/4) AA		1	MAXIMUM SURFACE ROUGHNESS 250 IN. (1/4) AA	
1	THROUGH-HOLE DIMENSIONS TO BE CHAMFERED		1	THROUGH-HOLE DIMENSIONS TO BE CHAMFERED	
1	EXT. & INT. THROTS TO BE CHAMFERED		1	EXT. & INT. THROTS TO BE CHAMFERED	
1	SPECIFIED DEPTH OF THREADED HOLES		1	SPECIFIED DEPTH OF THREADED HOLES	

PARTS LIST

Q510062	348A55	160A570	82127P0083	82018K2-3	348A2033	348A52
TOLERANCE & MACHINING NOTES (UNLESS OTHERWISE SPECIFIED) STANDARDS, PRACTICES & INTERPRETATIONS BASED ON ANSI Y14 SERIES - UNLESS OTHERWISE SPECIFIED, GEOMETRIC TOLERANCES TO FALL WITHIN LIMITS OF SIZE DIMENSIONS. DIMENSIONS ARE UNLESS OTHERWISE SPECIFIED. ALL UNFINISHED SURFACES TO BE CHAMFERED. RADIUS OR CHAMFER ALL EDGES. FILLET RADIUS. ANGULARITY. SURFACES. CHAMFERS. MAXIMUM SURFACE ROUGHNESS 250 IN. (1/4) AA. THROUGH-HOLE DIMENSIONS TO BE CHAMFERED. EXT. & INT. THROTS TO BE CHAMFERED. SPECIFIED DEPTH OF THREADED HOLES.						
DRAWING REF. FEATURES TO BE MEASURED & RECORDED IN THIRD ANGLE PROJECTION						
WESTINGHOUSE ELECTRIC CORPORATION ELECTRO-MECHANICAL DIV. - CHESTER, PA. U.S.A. GENERAL ASSEMBLY LEAD UNIT TRANSFER PUMP						
SIZE FROM NO. D04808						4D97581

Figure 15-2

GENERAL NOTES:

- 1- BEFORE ASSEMBLY FOR TEST AND FOR SHIPMENT, CLEAN ALL PARTS PER 8331TB053.
- 2- CONDUCT HYDROSTATIC TESTS PER 348A52.
- 3- CONDUCT MOTOR UNIT FINAL ACCEPTANCE ELECTRICAL TESTS PER 348A55.
- M4- ROTOR END PLAY SHALL BE .050 MINIMUM.
- 5- LUBRICATE THREADS WITH LUBRICANT. (ITEM 26). TIGHTEN IN ACCORDANCE WITH TORQUE TABLE.
- M6- AFTER ASSEMBLY, MEASURE AND RECORD THE DRY ROTOR BREAKAWAY TORQUE WITH THE ROTOR IN THE VERTICAL POSITION. MAXIMUM ALLOWABLE TORQUE IS 25 LB-FT.
- 7- LOCK LOCKCUPS, (ITEM 9) PER 160A273.
- 8- MINIMUM 7.0 GROUND WIRE, PHASE LEADS AND THERMOCOUPLE WIRE LENGTHS.
- 9- LOCK IMPELLER BOLT LOCK PINS. (ITEM 27) BY PEENING IMPELLER BOLT. (ITEM 17) FOR TEST. LOCK BY WELDING PER DETAIL -Y- FOR SHIPMENT TO CUSTOMER.
- 10- THE SYMBOL M DENOTES A MAJOR CHARACTERISTIC AS DEFINED BY QS10062.
- 11- ROTOR LOCKING DEVICE, (ITEM 16) INSTALLED FOR TRANSPORTATION ONLY. REMOVE ROTOR LOCKING DEVICE PRIOR TO PUMP OPERATION.
- 12- FIT UP ITEM 31 TO THIS DIMENSION FOR WELDING ITEMS 7 AND 18 TO ITEM 31. ITEMS 7 AND 18 TO BE MADE FLUSH WITH ITEM 31 AS SHOWN IN DETAIL T PRIOR TO WELDING.

VIEW V-V
ENLARGED
(SHEET 2)
SEE NOTE 8
FOR WIRE LENGTHS

TORQUE TABLE		
ITEM	THREAD SIZE	TORQUE FT-LBS.
8	.375-16	23-25
17	.625-11	33-36
21	.375-16	23-25
29	.375-16	23-25

Figure 15-4

E.M. 6673, Rev. 4
15-6

- F- MATERIAL: EPDM DUROMETER 75+5-15.
- E- WELD FILLER MATERIAL SIZE SPECIFIED IN WELD SPEC.
- D- B.O. FROM McMASTER-CARR PART NO. 92373A245, OR EQUIVALENT.
- C- BOLT TO BE PER ASTM-A-193, CLASS 2, GRADE 88M.
- B- GO2 WITHOUT STATOR WINDING THERMOCOUPLES.
- A- GO1 WITH STATOR WINDING THERMOCOUPLES.

1	FIRST STAGE IMP	4097523H01	36
1	SECOND STAGE IMP	4097530H01	35
1	O-RING	SPECIAL SIZE: 10.385±.055 I.D. X .103±.003 C/S	4 F 34
1	O-RING	PARKER SIZE NO. 2-177 (5.482±.055 I.D. X .103±.003 C/S)	4 F 33
8	O-RING	PARKER SIZE NO. 2-020 (.864±.005 I.D. X .070±.003 C/S)	4 F 32
1	OIL LINE HEADER	4098257H02	31
1	THR RUNNER WSHR	5371C80H02	30
4	LWR RAD BRG BOLT	4544860H04	29
1	PUMP NAMEPLATE	131P586H02	28
2	LOCK PIN	131P586H02	27
AR	LUBRICANT	(.188 DIA. X .50 L.G. SPRING PIN)	4 D 27
AR	WELD METAL	FEL-PRO N-5000	26
1	UPR RAD BRG ASSY	4942A24	E 25
1	2ND STAGE IMP PIN	4098089G01	24
4	LOCKPLATE	4544858H04	23
8	SUCT ADPTR BOLT	8580C06H02	22
1	LWR RAD BRG ASSY	.375-16 X 6.00 L.G. HEX HD.	4 C 21
1	SUCTION ADAPTER	4098081B01	20
4	DILUTION TUBE ASSY (SHORT)	4097794H01	19
1	IMPELLER BOLT	8575C83G02	18
1	ROTOR LKG DEVICE	4543859H02	17
1	1ST STG IMP MOD	56-08346	16
1	IMPELLER SLEEVE	4098265H01	15
1	1ST STG DIFFUSER	4543854H03	14
1	1ST STAGE IMP PIN	4097579G01	13
1	2ND STG IMP MOD	4544858H02	12
1	SPACER	4098265H02	11
13	LOCKUP	(.375)	10
9	2ND STG DIFF BOLT	(.375-16 X 2.75 L.G. SOC. HD. CAP SCREW)	9
4	DILUTION TUBE ASSY (LONG)	360A348H03	8
1	STATOR JACKET	4544860H02	7
1	THRUST BRG/OIFF	8579C83G01	6
1	THR RUNNER ASSY	4097588G01	5
1	FINAL ROTOR ASSY	4098083G01	4
1	FINAL STATOR ASSY	4097525G01	3
1	FINAL STATOR MOD	4097534G02	2
402	GO1	4098151B01	1

TOLERANCE & MACHINING NOTES (UNLESS OTHERWISE SPECIFIED) STANDARDS, PRACTICES & INTERPRETATIONS BASED ON ASME Y14.5 SERIES UNSPECIFIED DIMENSIONS TO FALL WITHIN LIMITS OF SIZE DIMENSIONS DIMENSIONS IN INCHES BASED ON 88°F (32°C) ALL UNTOLERANCED DIMENSIONS RADIUS OR CHAMFER ALL EDGES FILLET RADIUS ANGULARITY SURFACES CHAMFERS MAXIMUM SURFACE ROUGHNESS 250 THRU-OUT 24-28 EXT. & INT. THUS TO BE CHAMFERED SPECIFIED DEPTH OF THREADED HOLES IS MINIMUM GAGING LENGTH OF FULL THREAD		PARTS LIST WESTINGHOUSE ELECTRIC CORPORATION ELECTRO-MECHANICAL DIV. - CHESTER, PA. U.S.A. MOTOR UNIT ASSEMBLY TRANSFER PUMP MODEL 2A-513-A1 SIZE FROM NO. D04808 SCALE 1:1 SHEET 1 OF 2	
DRAWING REF. FEATURES TO BE MEASURED & RECORDED THIRD ANGLE PROJECTION		IDENTIFICATION OF PART PER IDENT. CLASS NO. IN LIST OF MATL. PER PS 595620.	

REVISION

7

VT
0.03/.25
TACK EACH CORNER
(82127PQ083 (25)

28

1 2

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26

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23

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12

21

22

26

SEE NOTE 5

SEE NOTES 5 AND 7

SEE NOTES 5 AND 7

26

9

29

X1

Z

82127PQ083
TYP. 4 PLACES
90° APART
(25)
VT
0.05/.38
SX NO CRACKS

DETAIL W
GOI ONLY

SECTION X-X
STATOR COILS NOT SHOWN

82127PQ083
TYP. 8 PLACES
VT
0.11(.03)
SX NO CRACKS

DETAIL T

DETAIL Y

2 X Ø.188 ±.008 ±.002
USE 2 EXISTING Ø.188 HOLES 180° APART
IN IMP. BOLT IT. 17 AS PILOTS

SEE NOTE 9

VT
SX NO CRACKS

Figure 15-4

E.M. 6673, Rev. 4
15-7

Westinghouse Electric Corporation
ELECTRO-MECHANICAL DIV. - CHESWICK, PA. U.S.A.
SIZE FROM NO. D 04808
SCALE N.T.S.
SMT. 2 OF 2

REVISION

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-GROUP 2 -

③

Figure 15-6

E.M. 6673, Rev. 3

15-11

	PIN	4.38±0.3 OF 1.50 DIA. BAR	(30AL SST)	ASTM-A-276	E F
1	TOP PLATE	5.00±0.6 X 2.75±0.6 OF .50 THK. PLATE	(30AL SST)	ASTM-A-240	A D
1	GUSSET	12.50±0.6 X 2.75±0.6 OF .50 THK. PLATE	(30AL SST)	ASTM-A-240	A D
2	SIDE PLATE	20.00±0.6 X 5.00±0.6 OF .50 THK. PLATE	(30AL SST)	ASTM-A-240	A D
AR	WELD METAL			4942A24	C
AR	WELD METAL			4942A23	C
---	LIFT BAIL ASY			4DS759G02	
1	PIPE	6.2±1 OF 8.0C SCH. 40. WELDED	(30AL SST)	ASTM-A-312	B
---	PLATE	26.5±5 DIA. OF 2.50 THK. PLATE	(30AL SST)	ASTM-A-240	A D
502 G01	PART NAME	SIZE AND REFERENCE INFORMATION			
502 G01				MATERIAL PART NUMBER	REVISIONS
502 G01					NOTES

DRAWING REF. FEATURES TO BE MEASURED & RECORDED THIRD ANGLE PROJECTION	IDENTIFICATION OF PART PER IDENT. CLASS NO. IN LIST OF MATL. PER PS 598620.	TOLERANCE & MACHINING NOTES (UNLESS OTHERWISE SPECIFIED) STANDARDS, PRACTICES & INTERPRETATIONS BASED ON ANSI Y14 SERIES - UNSPECIFIED GEOMETRIC TOLERANCES TO FALL WITHIN LIMITS OF SIZE DIMENSIONS. DIMENSIONS IN INCHES BASED ON 68°F.(20°C) ALL UNTOLERANCED DIMENSIONS ---.02 MAX. RADIUS OR CHAMFER ALL EDGES ---.04 MAX. PILET RADI - SURFACES ---.04 & .02 ANGULARITY: SURFACES --- 2° 30' CHAMFERS --- 2° MAXIMUM SURFACE ROUGHNESS 250 μ IN. (1/4) AA TWO SURFACES TO BE JUST A LAST THIRD OF EACH & REMAINING TO BE CHAMFERED OR ROUNDED. SECTIFIED DEDITH OF THREARD 10 HOLE IN.	IDENT CLASS GROUP NOTE	PARTS LIST		WESTINGHOUSE ELECTRIC CORPORATION ELECTRO-MECHANICAL DIV. - CHESBEE, PA. U.S.A. COLUMN EXTENSION ASSEMBLY	SIZE 7/8CH NO. D 04808 DWG NO. 4D97594
				OS10062 8212/P0035 8212/P0083 3AB2033 ASME B.31.3 ASME IX SNT-1C-1A	DATE D. PARTO 1-5-56 DES APP APP APP APP APP

Figure 15-6

E.M. 6673, Rev. 3

15-12

(FOR LIFTING PURPOSES)

Westinghouse Electric Corporation
ELECTRO-MECHANICAL DIV. - CHESWICK, PA. U.S.A.

SIZE 43 CM NO. 04808 4D9759 OVER NO.

-GROUP 1-

NOTES

- 1- THE SYMBOLS M AND B DENOTE MAJOR AND CRITICAL CHARACTERISTICS, RESPECTIVELY, AS DEFINED BY OSI0062.
- 2- 125/ ALL MACHINED SURFACES UNLESS OTHERWISE SPECIFIED.
- 3- HEAT END RINGS, JOURNAL SHOULDER RING AND LOWER JOURNAL HOLDER (ITEMS 5, 9 AND 12) TO 600°F MAX. TO ASSEMBLE ON SHAFT.
- 4- INSPECT ROTOR CAN (ITEM 6) BEFORE AND AFTER ASSEMBLY PER 8579C77. TRIM ENDS OF CAN FLUSH WITH END RING WELD PREPS. HEAT THE CAN TO 900°F MAX. FOR ASSEMBLY. WHEN CAN IS ASSEMBLED, THE WELD SEAM SHALL BE BETWEEN ANY TWO SLOTS.
- 5- HELIUM LEAK TEST THE ROTOR ASSEMBLY PER 348AS2.
- 6- LUBRICATE EXTERNAL THREADS WITH ITEM 17 PRIOR TO ASSEMBLY OF MATING PART. WIPE OFF EXCESS SO THAT ONLY A THIN FILM REMAINS. TIGHTEN IN ACCORDANCE WITH TORQUE TABLE.
- 7- LOCK LOWER JOURNAL NUT (ITEM 14) BY PEENING THREADS 2 LOCATIONS 180° APART.
- 8- BALANCE THE FINAL MACHINED ROTOR ASSEMBLY TO 0.25 GRAM AT THE 1.842 RADIUS OF EACH END RING. BALANCING IS TO BE ACCOMPLISHED BY REMOVING METAL FROM EACH END RING PER DETAIL Y. SEE VIEW Z FOR GRINDING LIMITATIONS IN VICINITY OF THE LEAK TEST PLUGS.
- 9- THESE DIMENSIONS ARE FOR RE-MACHINING THE .75 DIA. COUNTERSINK AFTER MACHINING END OF SHAFT.

TORQUE TABLE		
ITEM	THREAD SIZE	TORQUE
8	.190-24	8-10 IN-LBS.
11	1.625-8 LH	HAND TIGHT AT 250°F
14	2.500-8 LH	HAND TIGHT AT 250°F
15	.500-13	200-225 IN-LBS.

Figure 15-7

E.M. 6673, Rev. 3

15-13

- D- BRAZE MATERIAL SIZES SPECIFIED IN BRAZE SPEC.
- C- B.O. FROM McMASTER-CARR. CAT. NO. 92198A716, OR EQUIV.
- B- B.O. FROM McMASTER-CARR. CAT. NO. 92196A240, OR EQUIV.
- A- WELD MATERIAL SIZE SPECIFIED IN WELD SPEC.

AR WELD METAL	FEL-PRO M5000	43-42A24	A
AR LUBRICANT		9224676H04	
AR WELD METAL		908A682	A
1 LOCK BOLT	(.500-13 X 1.50 L.G. HEX HD CAP SCREW) (18-8 SST)	4097525H15	4 C
1 LWR JOURNAL NUT		8590C48H02	
1 UPR JOURNAL NUT		5371C2H04	
1 JNL SHOULDER RING		8590C42H02	
1 UPR JOURNAL NUT		8579C52H02	
1 LOWER JOURNAL		9371C2H02	
1 LWR JOURNAL HLDR		8579C79H02	
2 ROTOR VENT SCREW	(#10-24 X .375 L.G. HEX SOC HD CAP SCREW) (18-8 SST)	4097525H08	4 B
AR WELD METAL		908A782	A
1 ROTOR CAN		8579C77H01	
2 END RING		8579C50H02	
AR BRAZE ALLOY		360A611	D
17 ROTOR BAR		4544857H02	
2 COMM RING		8579C76H02	
1 ROTOR SHAFT MACH		4097525G01	
ROLL REGR			
SIZE AND REFERENCE INFORMATION			
PART NAME			
IDENT CLASS			
GROUP NOTE			

OSI0062 3482033 348AS2 82018KZ 80165 82127P5098 82127P5099 82127P0083		TOLERANCE & MACHINING NOTES UNLESS OTHERWISE SPECIFIED STANDARDS, PRACTICES & INTERPRETATIONS BASED ON ANSI Y14 SERIES - UNSPECIFIED GEOMETRIC TOLERANCES TO FALL WITHIN LIMITS OF SIZE DIMENSIONS. ALL UNTOLERANCED DIMENSIONS IN INCHES BASED ON 68°F (20°C) DIMENSIONS IN MILLIMETERS BASED ON 68°F (20°C) FRACTIONAL DIMENSIONS - .005 MAX DECIMAL DIMENSIONS - .005 MAX ANGULARITY SURFACES - 8°-30° MAXIMUM SURFACE ROUGHNESS 250 A IN (1/4) AA THUS - UNLESS OTHERWISE SPECIFIED, ALL DIMENSIONS SHALL BE TO THE CLOSEST TOLERANCE PER THE LIST OF MATERIALS.		OSI0062 3482033 348AS2 82018KZ 80165 82127P5098 82127P5099 82127P0083	
DRAFTING REF. FEATURES TO BE MEASURED & RECORDED THIRD ANGLE PROJECTION		IDENTIFICATION OF PART PER IDENT. CLASS NO. IN LIST OF MATL. PER P3 595620.		OSI0062 3482033 348AS2 82018KZ 80165 82127P5098 82127P5099 82127P0083	
PARTS LIST		Westinghouse Electric Corporation ELECTRO-MECHANICAL DIV. - CHESTER, PA. U.S.A.		PARTS LIST	
E.D. REPLER		ELECTRO-MECHANICAL DIV. - CHESTER, PA. U.S.A.		E.D. REPLER	
ROTOR ASSEMBLY AND FINAL MACHINING		ROTOR ASSEMBLY AND FINAL MACHINING		ROTOR ASSEMBLY AND FINAL MACHINING	
SIZE FROM NO.		SIZE FROM NO.		SIZE FROM NO.	
D 04808		D 04808		D 04808	
4D9752E		4D9752E		4D9752E

Figure 15-7

E.M. 6673, Rev. 3
15-14

Figure 1:

E.M. 6673, F
15-15

NOTES:

- 1- THE SYMBOL M DENOTES A MAJOR CHARACTERISTIC AS DEFINED BY Q510062.
- 2- ²⁵⁰✓ ALL OVER UNLESS OTHERWISE SPECIFIED.
- 3- ELECTRICAL TEST STATOR ASSEMBLY PER 348A55.
- 4- LUBRICATE THREADS OF CAPSCREW (ITEM 16) WITH ITEM 22, AND TIGHTEN TO 20-24 INCH-LBS.
- 5- LEAK TEST THE STATOR ASSEMBLY IN ACCORDANCE WITH 348A52.
- 6- INSPECT STATOR CAN BEFORE AND AFTER WELDING PER 8579C78.
- 7- HEAT UPPER AND LOWER STATOR HOUSINGS (ITEMS 5 AND 7), TO 375°-400°F TO ASSEMBLE THE UPPER BEARING ASSEMBLY (ITEM 6) AND LOWER BEARING INSERT (ITEM 8).
- 8- PLUG (ITEM 13) AND WELD MUST NOT PROTRUDE ABOVE TOP SURFACE OF UPPER STATOR HOUSING (ITEM 5). LOCAL GRINDING OF WELD, PLUG AND STATOR HOUSING IS ACCEPTABLE.
- 9- NO TRACES OF SILICONE RTV ARE PERMITTED 0.12 BEYOND THE O.D. OF THE ELECTRICAL STANDPIPE. SILICONE RESIDUE COULD PREVENT PROPER SEALING OF THE MATING SURFACES.

Figure 15-8

E.M. 6673, Rev. 3

15-16

- A- 601 WITH THERMOCOUPLES.
- B- 602 WITHOUT THERMOCOUPLES.
- C- TRIM TO SUIT.
- D- BOUGHT OUTSIDE FROM MCMASTER-CARR CATALOG NO. 92198A537 OR EQUIVALENT.
- E- BOUGHT OUTSIDE FROM MCMASTER-CARR CATALOG NO. 92141A029 OR EQUIVALENT.
- F- WELD FILLER MATERIAL SIZE SPECIFIED IN WELD SPEC.
- G- BOUGHT OUTSIDE FROM ANIXTER CO., 10 PARKWAY VIEW DRIVE, PITTSBURGH, PENNSYLVANIA 15205 OR EQUIVALENT.

QTY	DESCRIPTION	UNIT	QTY	DESCRIPTION	UNIT
1	SILICONE RTV	100W CORNING 734	1	SILICONE RTV	100W CORNING 734
1	CATALYST	PINT OF PLATINUM CATALYST	1	CATALYST	PINT OF PLATINUM CATALYST
1	POTTING RESIN	GALLON OF SILICONE POTTING RESIN	1	POTTING RESIN	GALLON OF SILICONE POTTING RESIN
AR	LUBRICANT	FEL PRO N-5000	AR	LUBRICANT	FEL PRO N-5000
AR	VARNISH	PINT OF DC804 (BOND)	AR	VARNISH	PINT OF DC804 (BOND)
AR	WELD METAL		AR	WELD METAL	
AR	WELD METAL		AR	WELD METAL	
AR	WELD METAL		AR	WELD METAL	
1	WASHER	.250 STANDARD FLAT	1	WASHER	.250 STANDARD FLAT
1	HEX HD SCREW	.250-20 X .50 LONG	1	HEX HD SCREW	.250-20 X .50 LONG
3	GROUND WIRE	FT. OF AWG NO. 4 THIN COPPER BLDG. WIRE (PART NO. 60-0401)	3	GROUND WIRE	FT. OF AWG NO. 4 THIN COPPER BLDG. WIRE (PART NO. 60-0401)
1	TERMINAL LUG	AWG NO. 4 AMP SOL. STRAND TERMINAL LUG (PART NO. 33469)	1	TERMINAL LUG	AWG NO. 4 AMP SOL. STRAND TERMINAL LUG (PART NO. 33469)
7	POTTING PLUG		7	POTTING PLUG	
18	POTTING	LBS. OF FLINT SILICA COARSE SAND	18	POTTING	LBS. OF FLINT SILICA COARSE SAND
2	MICA RING		2	MICA RING	
4	MICA RING		4	MICA RING	
4	MICA RING		4	MICA RING	
1	LWR BRG INSR		1	LWR BRG INSR	
1	LWR STTR MSG		1	LWR STTR MSG	
1	UPR BRG ASSY		1	UPR BRG ASSY	
1	UPR STTR MSG		1	UPR STTR MSG	
1	STATOR SHELL		1	STATOR SHELL	
1	STATOR CAN		1	STATOR CAN	
1	WLD STTR ASSY		1	WLD STTR ASSY	
1	WLD STTR ASSY		1	WLD STTR ASSY	

PARTS LIST	
QTY	DESCRIPTION
1	STATOR ASSEMBLY
1	STATOR CAN
1	STATOR SHELL
1	UPR STTR MSG
1	UPR BRG ASSY
1	LWR STTR MSG
1	LWR BRG INSR
4	MICA RING
2	MICA RING
18	POTTING
7	POTTING PLUG
1	TERMINAL LUG
3	GROUND WIRE
1	HEX HD SCREW
1	WASHER
AR	WELD METAL
AR	VARNISH
AR	LUBRICANT
1	POTTING RESIN
1	CATALYST
1	SILICONE RTV

WESTINGHOUSE ELECTRIC CORPORATION
ELECTRO-MECHANICAL DIV. - PITTSBURGH, PA. U.S.A.
FINAL STATOR ASSY.

Figure 15-8

VANE PLAN VIEW

MARK SERIAL NO. PART NO. -
AND REVISION 1 TO .4 HIGH
PER 595620-1

NOTES:

- 1- THIS DRAWING HAS NO MAJOR OR CRITICAL CHARACTERISTICS AS DEFINED BY QS10062.
- 2- ALL DIAMETERS TO TO BE CONCENTRIC TO -A- WITHIN 0.03 TIR. AND ALL FACES TO BE PERPENDICULAR TO -A- WITHIN 0.03 TIR.
- 3- ALL FILLET RADII TO BE .10±.03 UNLESS OTHERWISE SPECIFIED. ALL OUTSIDE CAST CORNERS TO BE .10 RADIUS MAX.
- 4- DATA FILE OF VANE DEVELOPMENT IS AVAILABLE FOR CASTING PATTERN.

[illegible]

E.M. 6673, Rev. 3

PARTS LIST

DETAIL -Y-
TYPICAL IMPELLER UPPER SIDE
BALANCE CORRECTION MATERIAL
.. (SEE NOTE 2)

TYPICAL IMPELLER UPPER SHROUD
BALANCE CORRECTION MATERIAL REMOVAL
.. (SEE NOTE 2)

5-COAT WEAR SURFACES WITH CERAMIC COATING PER 3A82031.
COATING TO BE APPLIED BY BANNING SPECIALTY CO., SARVER, PA.

PARTS LIST

MARK SERIAL NO., PART NO. AND REVISION
.2 TO .4 HIGH PER 595620-1

-GROUP 1-

Figure 15-12

E.M. 6673, Rev. 3

15-21

ITEM	DESCRIPTION	QTY	UNIT	REMARKS
1	SPRING PIN	1	EA	
2	WELD METAL	1	LB	
3	CAP SCREW	1	EA	
4	DIF LO MACH	1	EA	
5	DIF UP MACH	1	EA	
6	DIF LO MATL	1	EA	
7	DIF UP MATL	1	EA	

REVISIONS

NO.	DESCRIPTION	DATE
1	AS SHOWN	6-21-86

NOTES:

- THE SYMBOL M DENOTES A MAJOR CHARACTERISTIC AS DEFINED BY Q10062.
- 250/ ALL MACHINED SURFACES UNLESS OTHERWISE SPECIFIED.
- MACHINE VANS AND SURFACE -X- PER 1-DEAS PART.
- PROJECT: transfer-pump
- LIBRARY: hydraulics
- NAME: nc-hub
- VERSION: 1

APPLY COATING .015 $\pm$.010 PER NOTE 4 ON SHEET 1

WESTINGHOUSE ELECTRIC CORPORATION
ELECTRO-MECHANICAL DIV. - CHENEE, PA. U.S.A.

WESTINGHOUSE
FIRST STAGE DIFFUSER
MACHINING

SIZE: 4D97579

Figure 15-12

E.M. 6673, Rev. 3

15-22

E.M. 6673, Rev. 4
15-24

REVISION
3

Figure 15-13

E.M. 6673, Rev. 4
15-25

E.M. 6673, Rev. 4
15-26

NOTES:

- 1- THE SYMBOL M DENOTES A MAJOR CHARACTERISTIC AS DEFINED BY Q510062.
- 2- $\sqrt{\quad}$ ALL MACHINED SURFACES UNLESS OTHERWISE SPECIFIED.
- 3- COAT WEAR SURFACES WITH CERAMIC COATING PER 3A82031. COATING TO BE APPLIED BY BANNING SPECIALTY COMPANY, SARVER, PA.

A- B.O. FROM McMASTER-CARR OR EQUIVALENT.

B- CONDITION A, HOT FINISHED.

C- MATERIAL FURNISHED SHALL CONFORM TO ASTM-A-484 INCLUDING THE FOLLOWING OPTIONAL REQUIREMENT: NONE

[illegible]

3A82031 Q510062		IDENT CLASS GROUP NOTE		PARTS LIST		IDENT CLASS GROUP NOTE	
DRAFTING REF A FEATURES TO BE MEASURED & RECORDED THIRD ANGLE PROJECTION		IDENTIFICATION OF PARTS & IDENT. CLASS NO. IN LIST OF MATL. PER PS 935420.		TOLERANCE & MACHINING NOTES UNLESS OTHERWISE SPECIFIED) STANDARDS, PRACTICES & INTERPRETATIONS BASED ON ANSI Y14.5 SERIES UNSPECIFIED GEOMETRIC TOLERANCES TO FALL WITHIN LIMITS OF SIZE DIMENSIONS. DIMENSIONS IN INCHES BASED ON 68°F (20°C) ALL UNTOLERANCED DIMENSIONS --- ±.02 RADIUS OR CHAMFER ALL EDGES --- .03 MAX FILLET RADIUS --- .04 ±.02 SURFACES CHAMFERED --- 6°-30° MAXIMUM SURFACE ROUGHNESS 500 IN. 1/4 THRS.-J.L.C. 24-80 --- 1ST & LAST THRD. OF EXT. & INT. THRS. TO BE CHAM. 50° INCL. ANGLE. SPECIFIED DEPTH OF THREADED HOLES IS WITH MIN. BASIC LENGTH OF FULL THREAD		TOLERANCE & MACHINING NOTES UNLESS OTHERWISE SPECIFIED) STANDARDS, PRACTICES & INTERPRETATIONS BASED ON ANSI Y14.5 SERIES UNSPECIFIED GEOMETRIC TOLERANCES TO FALL WITHIN LIMITS OF SIZE DIMENSIONS. DIMENSIONS IN INCHES BASED ON 68°F (20°C) ALL UNTOLERANCED DIMENSIONS --- ±.02 RADIUS OR CHAMFER ALL EDGES --- .03 MAX FILLET RADIUS --- .04 ±.02 SURFACES CHAMFERED --- 6°-30° MAXIMUM SURFACE ROUGHNESS 500 IN. 1/4 THRS.-J.L.C. 24-80 --- 1ST & LAST THRD. OF EXT. & INT. THRS. TO BE CHAM. 50° INCL. ANGLE. SPECIFIED DEPTH OF THREADED HOLES IS WITH MIN. BASIC LENGTH OF FULL THREAD	
SIZE FROM INO. D 04808		SIZE FROM INO. D 04808		SIZE FROM INO. D 04808		SIZE FROM INO. D 04808	
SCALE N.T.A. POINT 70 LBS. SHT. 1 OF 4		SCALE N.T.A. POINT 70 LBS. SHT. 1 OF 4		SCALE N.T.A. POINT 70 LBS. SHT. 1 OF 4		SCALE N.T.A. POINT 70 LBS. SHT. 1 OF 4	

E.M. 6673, Rev. 4
15-27

REVISION
5

Figure 15-14

E.M. 6673, Rev. 4
15-28

Ø9.450±.001
[.005]
[N]
AFTER COATING

Ø9.430 ±.010
PRE-COATING

.015 ±.010
COATING
SEE NOTE 3
(SHT. 1)

$\varnothing .201^{+.008}_{-.002}$ $\nabla .25$
 $2 \times 180^\circ$
 $\varnothing .030 \text{ HEM}$
 ANTI-ROTATION HOLES

E.M. 6673, Rev. 3

15-30

**1- THIS DRAWING HAS NO MAJOR OR CRITICAL CHARACTERISTICS
AS DEFINED BY OSI0052.**

- 2- ALL DIAMETERS TO TO BE CONCENTRIC TO -A- WITHIN 0.03 TIR
AND ALL FACES TO BE PERPENDICULAR TO -A- WITHIN 0.03 TIR.

A. MATERIAL FURNISHED SHALL CONFORM TO ASTM-A-703, INCLUDING THE FOLLOWING SUPPLEMENTARY REQUIREMENTS: NONE.

[illegible]

Figure 15-1

E.M. 6673, R€
15-31

- NOTES:
- 1- THE SYMBOL M DENOTES A MAJOR CHARACTERISTIC AS DEFINED BY QSI0062.
- 2- 250/ ALL OVER UNLESS OTHERWISE SPECIFIED
- 3- COAT WEAR SURFACES WITH CERAMIC COATING PER 3A82031. COATING TO BE APPLIED BY BANNING SPECIALTY COMPANY, SARVER, PA.

(ø9.50 CASTING)

[D-]

NON-COATED
COATING I.D.

[C-]

DETAIL-B

SECTION Z-Z

COATING

.015
+.010
-.000

SEE NOTE 3

(2.00)

.97

(40°)

(45°)

[D-]

[A]

[B]

[E]

[F]

[G]

[H]

[I]

[J]

[K]

[L]

[M]

[N]

[O]

[P]

[Q]

[R]

[S]

[T]

[U]

[V]

[W]

[X]

[Y]

[Z]

REVISIONS	
1	S.O. VOOI
4	IT. 1 IN B/A. 2ND STG DIFF MOO. WAS 2ND STAGE DIF. AND 4098259601. WAS 4097528601 - DETAIL OF IT. 1 REVISED. EDH 8-26-97/ <i>R. J. J. J.</i> 8-27-97

MARK SERIAL NO., PART NO.
AND REVISION NO.
.2 TO .4 HIGH PER 595620-1

Figure 15-17

E.M. 6673, Rev. 4
15-32

NOTES:

- 1- THIS DRAWING HAS NO MAJOR OR CRITICAL CHARACTERISTICS AS DEFINED BY QSI0062.
- 2- MACHINE LENGTH OF LOCK PIN ITEM 7 TO SUIT.
- 3- RECORD SERIAL NUMBERS OF SHOE ASSEMBLIES ITEM 3 PRIOR TO ASSEMBLY.

7 SEE NOTE 2

A- B.O. FROM McMASTER-CARR, CAT. NO. 90145A546, OR EQUIV.

6	LOCK PIN	(.25 DIA. X 1.50 LG. DOWEL PIN)	(18-8 SST)	4	A
6	BRG LINK PIN		4098259602		
6	UPR BRG LINK		4098259604		
6	LWR BRG LINK		4098259602		
6	SHOE ASSY		4098259601		
6	REFINER RUN ASSY		4098259601		
1	2ND STG DIFF MOO		4098259601		
601	PART NAME	SIZE AND REFERENCE INFORMATION			
601	IDENT CLASS	PART NO.			
601	GROUP NOTE	PART NO.			

PARTS LIST

QSI0062	Westinghouse Electric Corporation ELECTRO-MECHANICAL DIV. - CHESTER, PA. U.S.A.
THRUST BEARING/ SECOND STAGE DIFFUSER ASSEMBLY	SIZE (FROM IN.) D04808 SCALE: NTS. 1:1 77 LBS. SMT. 1 OF 1
QSI0062	4D97588

REVISIONS	
1	S.O. V002

MARK SERIAL NO., PART NO. AND REVISION
NO. .1 TO .2 HIGH PER 595620-1

Figure 15-18

E.M. 6673, Rev. 3
15-33

NOTES:

- 1- THE SYMBOLS M AND A DENOTE MAJOR AND CRITICAL CHARACTERISTICS, RESPECTIVELY, AS DEFINED BY QS10062.
- 2- APPLY A THIN FILM OF LOCTITE, ITEM 3, BETWEEN BACK-UP RING, ITEM 1, AND INSERT, ITEM 2, AND COMPRESS WITH 2500 LBS. MINIMUM LOAD UNTIL CURED.
- 3- BALANCE RUNNER ASSEMBLY TO 0.07 GRAMS AT 3.50 RADIUS BY REMOVING MATERIAL AS SHOWN.
- 4- RECORD THE MIX/BATCH NO. OF ITEM 2 AT TIME OF ASSEMBLY.

1	LOCTITE	50 ML	908A536H11	
1	INSERT		937IC20H02	
1	BACK-UP RING		937IC25H02	
1	PART NAME	SIZE AND REFERENCE INFORMATION	MATERIAL PART NUMBER QTY REQ. PER	LOCTITE

PARTS LIST

QS10062	TOLERANCE & MACHINING NOTES (UNLESS OTHERWISE SPECIFIED) STANDARDS, PRACTICES & INTERPRETATIONS BASED ON ANSI Y14 SERIES - UNSPECIFIED GEOMETRIC TOLERANCES TO FALL WITHIN LIMITS OF SIZE DIMENSIONS. DIMENSIONS IN INCHES BASED ON 88°F (32°C) ALL UNTOLERANCED DIMENSIONS ±.02 RADIUS OR CHAMFER ALL EDGES FILLET RADIUS .03 MAX. ANGULARITY: CHAMFERS .03 ±.02 CHAMFERS .03 ±.02 THIRD ANGLE PROJECTION	IDENTIFICATION OF PART PER IDENT. CLASS NO. IN LIST OF MATL. PER PS 98560.	DRAWING BY: PLANNING & RECORDS THIRD ANGLE PROJECTION	WESTINGHOUSE ELECTRIC CORPORATION ELECTRO-MECHANICAL DIV. - CHESTER, PA. U.S.A. THRUST RUNNER ASSY.	
				SIZE FROM NO. D 04808	QTY REQ. PER 4D98083

TORQUE TABLE	
ITEM	TORQUE (INCH-LBS)
6	20 ± 2
7	30 ± 3

4	PIN	.125+.001-.000 DIA. X .90' L.G.	(304 SST)	4
1	LOC TITE	50 ML	908A636H11	
AR	LUBRICANT	FEL-PRO N-5000	922A576H04	
4	RETAINING PIN		454A566SH02	
16	LOCK SCREW	(164-32 X .375 L.G. HEX 50C FLAT HD CAP SCREW)	4098089H06	
8	END PLATE		9371C82H02	
			4545960H02	
4	LOWER PAD		9371C81H01	
4	LWR PAD HLDR		4098130H02	
1	LWR BRG HSG		4098082H02	
GC1	PART NAME		MATERIALS PART NAME REV. 10/27/2006	NOTES
REC'D		SIZE AND REFERENCE INFORMATION		

PARTS LIST

DRAFTING REF. FEATURES TO BE MEASURED & RECORDED THIRD ANGLE PROJECTION	QSI0062	TOLERANCE & MACHINING NOTES (UNLESS OTHERWISE SPECIFIED) STANDARDS, PRACTICES & INTERPRETATIONS BASED ON ANSI Y14 SERIES UNLESS SPECIFIC DIMENSIONS TO FALL WITHIN TOLERANCE LIMIT OF SIZE DIMENSIONS. DIMENSIONS IN INCHES BASED ON 68°F (20°C) ALL UNTOLERANCED DIMENSIONS — .02 RADIUS OR CHAMFER ALL EDGES — .03 MAX. FILLET RADIUS — .04 & .02 ANGULARITY SURFACES — 0.04 & .02 MAXIMUM SURFACE ROUGHNESS 250 μ IN (✓) AA THOS — U.N.C. 2A-2B — 1ST & LAST THRD. OF EXTS. & INT. THDS. TO BE CHAM 50° INCL. ANGLE. SPECIFIED DEPTH OF THREADED HOLES IS MINIMUM GAUGING LENGTH OF FULL THREAD	DATA E. QUEPLER DESIGNED CHECKED BY DATE APP APP APP APP BY	3-2-57	Westinghouse Electric Corporation ELECTRO-MECHANICAL DIV. — CHESTER, PA. U.S.A.
LOWER RADIAL BEARING ASSEMBLY — PIVOTED PAD BEARING
					SIZE FROM NO. D04808 SHOP NO.
					4D9808
					SCALE: N.T.S. INCHES 10 LBS. SHT. 1 OF 1

1- THE SYMBOLS M AND B DENOTE A MAJOR AND CRITICAL CHARACTERISTIC, RESPECTIVELY, AS DEFINED BY QSI0062.

2- RECORD THE MIX/BATCH NUMBER OF ITEM 3 AT TIME OF ASSEMBLY.

- 3- THE PIVOT PIN, ITEM 4, IS TO BE SEATED AGAINST THE BEARING HOUSING PIVOT PIN SHOULDER, ITEM 1 WITHIN .0025.
- 4- LUBRICATE THREADS WITH LUBRICANT, ITEM 8, AND TIGHTEN IN ACCORDANCE WITH TORQUE TABLE.

5- LOCK END PLATE FLAT HEAD CAP SCREWS, ITEM 6, BY PEENING
END PLATES, ITEM 6, AT TWO LOCATIONS FOR EACH SCREW.

6- LOCK RETAINING PINS, ITEM 7, BY PEENING HOUSING, ITEM 1,
AT TWO LOCATIONS FOR EACH PIN.

7- THIS DIMENSION APPLIES ONLY AT THE FOUR LOCATIONS IN LINE WITH THE PIVOT PINS.

Ø 8- EACH PAD IS TO BE CONTOURED TO A 3.2625+/-0.0005 DIAMETER.

9- THE BORE SIZE AND RUNOUT OVER THE PINS ARE APPLICABLE TO THE GRINDING OPERATION WHEN THE BEARING PADS ARE IN THE SECURED POSITION.

10- APPLY A THIN FILM OF LOCTITE, ITEM 9, BETWEEN LOWER PAD, ITEM 3, AND LOWER PAD HOLDER, ITEM 2, AND COMPRESS WITH 250 LBS. MINIMUM LOAD UNTIL CURED.

11- MARK PIVOT PIN, PAD HOLDER, AND HOUSING (ITEMS 1, 2 AND 4) LOCATION LETTERS "A", "B", "C" OR "D" IN LINE WITH EACH OTHER.

12- ASSEMBLE PIVOT PIN LOCK PINS, ITEM 10. ASSURE PIN IS FLUSH OR BELOW FACE OF HOUSING, ITEM 1. PEEN (2) PLACES 180° APART TO LOCK PIN.

- MARK ASSEMBLY LOCATION LETTERS
"A," "B," "C" OR "D" ON ITEMS 1
AND 2: MARK WITH LETTERS 1 TO
.2 HIGH PER 595620-1. SEE NOTE 11

SECTION Z-Z

12- ASSEMBLE PIVOT PIN LOCK PINS, ITEM 10. ASSURE PIN IS FLUSH OR BELOW FACE OF HOUSING, ITEM 1. PEEN (2) PLACES 180° APART TO LOCK PIN.

TORQUE TABLE	
ITEM	TORQUE (INCH-LBS)
6	20 ± 2
7	30 ± 3

	PIN	.125+-.001--000 DIA. X .90 L.G.	(304 SST)		A
I	LOC TITE	50 ML		S08AS36H11	
AR	LUBRICANT	FEL-PRO N-5000		92ZA67EH04	
4	RETAINING PIN			4S45B6SH02	
IS	LOCK SCREW	.164-.32 X .375 L.G.	(316 SST)		A
B	END PLATE			937ICQZHO2	
4	PIVOT PIN			4S45B6SHO2	
4	UPPER PAD			937ICB IHO2	
4	UPPR PAD HLR			4D9BI23HO2	
I	UPR BRO HSU			4D9BO9OH02	
SDI RECOR	PART NAME	SIZE AND REFERENCE INFORMATION		MATERIAL: S08AS36H11 92ZA67EH04 4S45B6SH02	NOTES:

PARTS LIST

DRAFTING REF. △ FEATURES TO BE MEASURED & RECORDED THIRD ANGLE PROJECTION	IDENTIFICATION OF PART PER IDENT. CLASS NO. IN. LIST OF MATL. PER PS SPECED. SPECIFICATION LIST	TOLERANCE & MACHINING NOTES (UNLESS OTHERWISE SPECIFIED) STANDARDS, PRACTICES & INTERPRETATIONS BASED ON ANSI Y14 SERIES - UNLESS SPECIFIC TOLERANCES TO FALL WITHIN LIMIT OF SIZE DIMENSIONS DIMENSIONS IN INCHES BASED ON 68°F (20°C) ALL UNTOLERANCED DIMENSIONS - ±.02 RADIUS OR CHAMFER ALL EDGES - .03 MAX. FILLET RADI! SURFACES - .04 & .02 ANGULARITY: SURFACES - 2° 30' CHAMFERS - 2° 30' MAXIMUM SURFACE ROUGHNESS 250 - 2.5 AA THUS - UNLESS SPECIFIED TO BE CHAMFERED, EXT. & INT. THO. TO BE CHAMFERED INCL. HOLE. SPECIFIED DEPTH OF THREADED HOLES IS MINIMUM TAPPING LENGTH OF FULL THREAD	DATE EDI/WHM 3-18-77		WESTINGHOUSE Electric Corporation ELECTRO-MECHANICAL DIV. - CHESTER, PA. U.S.A.	
			UPPER RADIAL BEARING ASSEMBLY - PIVOTED PAD BEARING	SIZE FROM IN. DOW NO. D 04808	SCALE N.T.S. SHEET 8 OF 85.	SMT. 1 OF 1

16.0 PROCEDURES

<u>Page</u>	<u>Procedure Number</u>	<u>Title</u>
16-2	348A53	Production Testing of Transfer Pumps
16-5	83311TB053	Cleaning Procedure for Hazardous Waste Pumps
16-11	348A52	Hydrostatic and Helium Leakage Testing of Transfer Pumps
16-19	85310XX	Packaging, Shipping, Handling and Storage of the 2A513-A1 Transfer Pump
16-30	82121PQ006	Welding Procedure
16-31	PQR21001	Procedure Qualification Records
16-35	82127PQ006	Welding Procedure
16-36	PQR 934D064G02	Procedure Qualification Test
16-37	PQR 27045	Procedure Qualification Record
16-40	CMM070805A	Welder Qualification Test
16-41	CTM070804A	Welder Qualification Test

TITLE: PRODUCTION TESTING OF TRANSFER PUMPS

1.0 SCOPE:

- 1.1 This specification describes the minimum test requirements for pump acceptance testing of tank farm transfer pumps. This specification complies with WHC-S-0238, "SPECIFICATION FOR PROTOTYPE TANK FARM TRANSFER PUMPS with Submersible Motor," and API - 610, "Centrifugal Pumps for General Refinery Service." Westinghouse EMD may perform additional testing to assure quality and operability at its discretion.
- 1.2 The objective of the testing defined by this specification is to:
- 1.2.1 Demonstrate the operability of the pump at the operating conditions specified by the customer,
 - 1.2.2 Demonstrate the hydraulic performance of the pump over the range of heads and flows represented by full speed operation,
 - 1.2.3 Measure the vibration demonstrated by the pump at various operating conditions.
 - 1.2.4 Measure the electrical performance of the motor at the hydraulic conditions of interest.

2.0 TEST EQUIPMENT

- 2.1 Testing will be conducted in a test circuit consisting of:
- 2.1.1 A tank with an appropriate free surface to draw pump suction from
 - 2.1.2 Discharge piping or hose attached to the pump discharge to support test apparatus and return the pumped fluid to the tank.
 - 2.1.3 A piezometer ring for measuring discharge pressure.
 - 2.1.4 A bubbler and differential pressure transmitter(s) to measure the head difference from the suction submergence to the pressure at the pump discharge.
 - 2.1.5 A flow metering system capable with measuring flow from 20 - 200 gpm.
 - 2.1.6 Temperature detectors to establish the average temperature of the pumped fluid.
 - 2.1.7 Vibration detectors mounted in 2 axes at the location of the motor upper stator housing.

¹ These specifications are referenced for information only. All of the requirements of these specifications have been integrated into this document. It is not necessary to be familiar with either document to perform the required testing.

TITLE: PRODUCTION TESTING OF TRANSFER PUMPS

- 2.1.8 A real time analyzer to establish pump speed on the basis of accelerometer readings.
- 2.1.9 Vibration monitors to support the test detectors and any permanently mounted pump vibration instrumentation.
- 2.1.10 Electrical power metering for voltage, current, KW and frequency

3.0 GENERAL REQUIREMENTS

- 3.1 Data logging required by this specification shall be submitted in tabular form. Plots where required will be prepared and submitted. Logs of the testing will be prepared and maintained and submitted with the final data release. All data will include the identification of the test article, the time and date taken and the signature of the operator recording the data.
- 3.2 Vibration measured at the lower accelerometer location (attached to the upper stator housing) shall be less than 2 mils, filtered, at any flow rate above 20% of rated flow at any operating speed of the pump.
- 3.3 The supports for the pump in the test facility are expected to be considerably more limber than the tank installation and will possess low frequency resonances, that may be excited by low speed operation. Traces of the frequency spectrum shall be recorded for any condition which results in exceeding the limits of paragraph 3.2. Operation at flow and speed conditions exceeding 5 mils should not be continued without consulting the cognizant design or test engineer.

4.0 24 HOUR RUN - Lead Unit Pump Only

The pump will be erected in a suitable tank and operated for 24 hours at rated speed (60 Hz) and rated flow $\pm 5\%$. All operating parameters will be recorded every half hour. Flow may be adjusted periodically to assure that operation is maintained within the required range of flows throughout the mechanical run.

5.0 PERFORMANCE AT RATED SPEED - 60 Hz

Head, flow, voltage, current, KW, pump speed and frequency shall be recorded at the following nominal flow rates: 20, 40, 60, 80, 100, 120, 140, and 160 gpm (achieving these rates ± 3 gpm for each measurement is acceptable). The fluid medium shall be water at a temperature below 120°F. Each test point shall consist of the average of a minimum 5 sets of data recorded at the test condition.

TITLE: PRODUCTION TESTING OF TRANSFER PUMPS

This data may be manually logged and averaged or may be computer collected and averaged. Computer data shall be averaged over a period of 10 minutes. Data plots of head, current, KW, overall efficiency vs. flow shall be prepared and submitted with the test data. The curve shall fall between 441 ft-H₂O and 472.5 ft-H₂O at 140 GPM in water.

6.0 OFF SPEED DEMONSTRATION TESTING

6.1 Record the head, current, voltage, KW for operation at the following speeds and flow rates.

Pump Speed	Flow Rate
3550 RPM (59.2 Hz [°])	138 GPM
3290 RPM (54.8 Hz [°])	132 GPM
1815 RPM (30.3 Hz [°])	132 GPM
3510 RPM (58.5 Hz [°])	53 GPM
1435 RPM (23.9 Hz [°])	53 GPM

* This frequency is the mechanical running speed of the pump, not the electrical in put frequency.

Record this data at the specified speed and flow rate ± 2 GPM. Adjust flow to $+10 \pm 2$ GPM and -10 ± 2 GPM of the nominal value in the table and record a complete set of mechanical, electrical and hydraulic data.

7.0 APPROVALS:

Design	<u>R.F. Gardiner</u>	Date	<u>4/5/95</u>
Q.A.	<u>C. C. Currello</u>	Date	<u>4/12/95</u>
Mfg Engn.	<u>Clifford H. Cameron</u>	Date	<u>4/4/95</u>
Test Engn.		Date	
Other		Date	

Revision A - Initial Revision

WESTINGHOUSE ELECTRIC CORPORATION
ELECTRO-MECHANICAL DIVISION
CHESWICK, PA 15024

Number 83311TB053
Revision A
Date March 23, 1995

TITLE: CLEANING PROCEDURE FOR HAZARDOUS WASTE PUMPS

TABLE OF CONTENTS

- 1.0 GENERAL REQUIREMENTS
- 2.0 APPLICABLE DOCUMENTS
- 3.0 EQUIPMENT AND CLEANING AGENTS
- 4.0 PRECLEANING
- 5.0 GENERAL CLEANING
- 6.0 FINAL CLEANING
- 7.0 DRYING

TITLE: CLEANING PROCEDURE FOR HAZARDOUS WASTE PUMPS

1.0 GENERAL REQUIREMENTS

- i.1 The procedures specified herein govern the cleaning of part surfaces of Pump Model 2A-513-A1.
- 1.2 The following precautions apply at all times throughout product manufacture.
 - 1.2.1 WARNING - The use of chemical cleaning agents required for cleaning operations can be hazardous to health and requires careful handling for safe application. All solvents shall be used in minimum quantities and in well ventilated areas. In the event of spillage remove all sources of ignition and clean up immediately. If spillage cannot be cleaned up immediately, evacuate the area and notify the area supervisor. Use protective clothing/equipment as necessary. In particular, high concentrations of N-Heptane are potentially lethal. However, used in minimum quantities in a well-ventilated area, N-Heptane can be used safely.
 - 1.2.2 The use of tape or tape adhesive on part surfaces is prohibited except where allowed by specific routing instruction and the part is subsequently cleaned.
 - 1.2.3 Machining of stainless steel shall be performed with cutting tools specifically designated for stainless steel material. New or re-sharpened tools may also be used. Previously used grinding wheels shall be dressed prior to use on stainless steel material. Wire brushing of stainless steel shall be done with stainless steel brushes that are new or are designated for use on stainless steel only. The use of components containing chlorides is prohibited.
 - 1.2.4 Handling devices and fixturing shall be made as to not induce foreign material onto stainless steel parts. Handling devices and fixturing shall be made of stainless steel or other materials that are stainless steel clad or chrome plated at point of contact with the product.

WESTINGHOUSE ELECTRIC CORPORATION
ELECTRO-MECHANICAL DIVISION
CHESWICK, PA 15024

Number 83311TB053
Revision A
Date March 23, 1995

TITLE: CLEANING PROCEDURE FOR HAZARDOUS WASTE PUMPS

- 1.3 Clean the pump parts prior to assembly for test as required by the specific instructions that follow. Parts will be considered clean when visual evidence of foreign materials such as grit, metal or other particles, oil, slag, scale, rust, dirt, stains, or fibers have been removed. The steps may be repeated as often as necessary to provide clean components. Protect cleaned components by placing in polyethylene bags, or by covering with polyethylene when the parts are not in work.

When necessary, scotch brite may be used when the listed cleaning method is inadequate for preliminary cleaning. All residues from preliminary cleaning methods including those from the use of Scotch Brite must be completely removed. Final cleaning of the component must then be performed prior to final assembly. In addition, forced air may be used to assist in the drying of components by evaporation to the atmosphere at both the preliminary and final cleaning operations.

2.0 APPLICABLE DOCUMENTS

- 2.1 ASTM A-380 Cleaning and descaling stainless steel parts, equipment and systems.

3.0 EQUIPMENT AND CLEANING AGENTS

- 3.1 Equipment, solvents and agents needed for cleaning pump parts or assemblies are as defined below:

3.1.1 The equipment required includes:

Plastic bottles

Immersion tub

Fixtures for introducing fluids

Lint free cloths

Steam genny

Power hand tools

WESTINGHOUSE ELECTRIC CORPORATION
ELECTRO-MECHANICAL DIVISION
CHESWICK, PA 15024

Number 83311TB053
Revision A
Date March 23, 1995

TITLE: CLEANING PROCEDURE FOR HAZARDOUS WASTE PUMPS

3.1.2 Solvents permitted for use are as follows:

Acetone (MOD 9907A41)

** Denatured alcohol (MOD 9907A42)

N-Heptane (MOD 9909A20)

** The use of pure grain ethyl alcohol is PROHIBITED.

3.1.3 Water and cleaning solutions permitted for use are:

Grade B water (demineralized water max chloride ion content 1.0 ppm).

Grade C water (tap water max chloride ion content 35 ppm).

Ultrasonic cleaning solution (Bransonic Industrial Strength Cleaning Solution)

Nonionic detergent solution (IGEPAL or TERGITOL) MOD 9907A43).

3.1.4 Mechanical cleaning material:

Abrasive cloth - aluminum oxide

Grinding wheel - aluminum oxide

Abrasive pad - scotchbrite

Wire Brush - stainless steel

4.0 PRECLEANING

4.1 Precleaning

Precleaning is the removal of grease, oil, paint, soil, grit, and other gross contamination preparatory to a fabrication process or final cleaning. Precleaning is generally not as thorough as subsequent cleaning operations. Materials should be precleaned

WESTINGHOUSE ELECTRIC CORPORATION
ELECTRO-MECHANICAL DIVISION
CHESWICK, PA 15024

Number 83311TB053
Revision A
Date March 23, 1995

TITLE: CLEANING PROCEDURE FOR HAZARDOUS WASTE PUMPS

before high-temperature operations (250 degrees or greater), and before any finish-cleaning operation where the parts will be immersed or where the cleaning solutions will be reused. Precleaning may be accomplished by vapor degreasing; immersion, spraying, wiping or swabbing with alkaline or emulsion cleaners, steam, or high-pressure water-jet.

5.0 GENERAL CLEANING

- 5.1 General cleaning includes all operations required for the removal of surface contaminations to ensure corrosion resistance and prevention of product contamination. General cleaning is an intermediate cleaning operation performed during manufacturing and prior to final cleaning. General cleaning may be accomplished by mechanical cleaning, immersion, wiping or swabbing with a solvent, water, steam or detergent cleaners.
- 5.2 Solvent and detergent cleaning is a process for removing contaminants from metal surfaces by immersion, spraying, wiping or swabbing, using an approved solvent or detergent. Reference paragraphs 3.1.2 and 3.1.3. Detergent cleaning is to be followed by a water rinse.
- 5.3 Mechanical cleaning using an abrasive or wire brush is desirable for removing surface contaminants and rust. Reference paragraphs 1.2.3 and 3.1.4. Mechanical cleaning should be followed with other general cleaning or final cleaning method.
- 5.4 Steam cleaning, used mostly for cleaning large items, may be used with cleaning agents such as solvents and detergents.

6.0 FINAL CLEANING TO BE PERFORMED AS REQUIRED PRIOR TO ASSEMBLY

- 6.1 Final cleaning may be accomplished by immersion, wiping or swabbing with a solvent, water or detergent cleaners. Detergent cleaning is to be followed by a water rinse.

TITLE: CLEANING PROCEDURE FOR HAZARDOUS WASTE PUMPS

7.0 DRYING

- 7.1 All components may be dried by wiping with clean lint free cloth or by evaporation to atmosphere or clean forced air.

APPROVAL:

Design	<u>R.F. Gardner</u>	Date	<u>3/23/95</u>
PQI	<u>[Signature]</u>	Date	<u>3/28/95</u>
Mfg Engrg	<u>[Signature]</u>	Date	<u>3/23/95</u>
Others	<u>N/A</u>	Date	

Revision A - Original Rev.

WESTINGHOUSE ELECTRIC CORPORATION
ELECTRO-MECHANICAL DIVISION
CHESWICK, PA 15024

Number 348A52
Revision A
Date April 2, 1995

TITLE: HYDROSTATIC AND HELIUM LEAKAGE TESTING OF TRANSFER PUMPS

1.0 SCOPE:

- 1.1 This specification defines the procedures and limits applicable to the hydrostatic testing and helium leakage testing of tank farm transfer pumps. The hydrostatic testing is governed by API-610, "Centrifugal Pumps for General Refinery Service," 7th edition February 1989¹.

The transfer pumps contain many features that are unusual for pumps covered by hydrostatic tests performed under API-610. The pump design includes a number of gasketless joints in the casing that permit leakage. The design includes the presence of features requiring a hydrostatic test that are inaccessible for inspection when the pump is fully assembled. The point in the process where hydrostatic testing and inspection are conducted are defined by the detailed drawings of the pump components. The drawings provide for the hydrostatic test of individual components or sub assemblies selected to assure that inspections can be conducted. The combination of all hydrostatic tests defined on the individual component or sub assembly drawings results in satisfying the hydrostatic test requirements for the complete pump.

- 1.2 Helium leakage testing as specified is governed by good commercial practice and the limits of sensitivity of helium leakage detection technology. Helium leakage testing is used to assure hermetic seals of sensitive areas in the motor design, which are uninspectable by other techniques. These tests are conducted in addition to the mandatory hydrostatic test. These tests are specified on the sub assembly drawings that define the rotor and stator assemblies of the pump.

2.0 GENERAL REQUIREMENTS

- 2.1 This specification is applicable to transfer pump components when specified on the pump detail drawings, manufacturing drawings or manufacturing routings.

- 2.2 The Test Department shall maintain records of all tests performed in accordance with this specification. These records must show the following:

- 2.2.1 Identification of the component or assembly

¹ This information is for reference only. The requirements have been integrated into this specification. It is not necessary to be familiar with API - 610 to perform the required testing.

WESTINGHOUSE ELECTRIC CORPORATION
ELECTRO-MECHANICAL DIVISION
CHESWICK, PA 15024

Number 348A52
Revision A
Date April 2, 1995

TITLE: HYDROSTATIC AND HELIUM LEAKAGE TESTING OF TRANSFER PUMPS

- 2.2.2 The serial number of the component or assembly
 - 2.2.3 The name of the person performing the test.
 - 2.2.4 The results of the test
 - 2.2.5 The date and time of the test
 - 2.2.6 The serial number of the test gage or leak detector and the date of last calibration.
 - 2.2.7 A certification that the test was conducted in accordance with this specification.
- 2.3 The equipment for testing, i.e., mass spectrometer, pressure gages, valves and piping shall be kept in good repair and shall be checked and calibrated in accordance with the procedures contained in the Test Procedures (TP) Manual and the appropriate Instrument Calibration Procedures (ICP'S).
- 2.3.1 The mass spectrometer shall have a sensitivity sufficient to detect a leak of 2×10^{-4} cc/sec STP of helium.
 - 2.3.2 A calibrated hydrostatic test gage shall be a mirrored dial bourbon tube pressure gage in good repair and calibrated within 20 calendar days. Gages shall be calibrated with a suitable deadweight tester or transfer standard traceable to the National Bureau of Standards. See the hydrostatic test requirements for limitations on accuracy and full scale range.
- 2.4 Helium Leakage Test Acceptance - Detectable changes in the leak rate indication shall be investigated. Any repeatable increase in indications larger than 4×10^{-9} mbar liter/sec. above the initial background reading shall be cause for rejection.
- 2.5 Hydro Static Test Acceptance - There shall be no evidence of leakage from the component undergoing the hydrostatic test.
- 2.6 Cleanliness - Vacuum sealing substances such as glypthal and apiezon Q are prohibited from contact with any surface of the components being tested. These materials may be used on test connections and fixtures are required to achieve the required leakage integrity for the leak test systems.
- 3.0 HYDROSTATIC TEST
- 3.1 Hydrostatic tests for the transfer pump design is nominally 660 psig. The limits of test are 670-683 psig, indicated. The hydrostatic test shall be conducted with a calibrated gage

WESTINGHOUSE ELECTRIC CORPORATION
ELECTRO-MECHANICAL DIVISION
CHESWICK, PA 15024

Number 348A52
Revision A
Date April 2, 1995

TITLE: HYDROSTATIC AND HELIUM LEAKAGE TESTING OF TRANSFER PUMPS

of full scale range between 1000 psig and 2000 psig (nominally 150% - 300% of the test pressure). Gages shall be accurate to within 1/2% of full scale. That is no correction recorded on the calibration sticker shall exceed 5 psig on a 1000 psig gage, or 10 psig on a 2000 psig gage or the gage has been calibrated in accordance with ICP-10.

- 3.2 The hydrostatic test gage is to be connected directly to the test article. Intervening valves are not permitted. Direct connection means connected to the test article or test fixtures by an integral connection, or connected to the test article or test fixtures by a short length of hose or tubing of a suitable pressure rating. Note that the elevation of the gage relative to the test article is not critical. In general, pressure testing of the long assemblies of the transfer pump column will be conducted in a horizontal position. In the event that a special test has to be conducted with the pump in a vertical position, contact the test engineer for guidance.
- 3.3 The test shall be conducted for a minimum of 30 minutes regardless of feature size or wall thickness. Note that pressure is maintained during this period, there is no requirement for a component or assembly to "hold" pressure.
- 3.4 At the end of 30 minutes the test pressure shall be lowered to 500-550 psig indicated and the component or assembly shall be inspected for leakage. Special attention shall be paid to welds and section transitions in the material of the test article. Evidence of leakage shall be cause for rejection.
- 3.5 Leakage from temporary test fixtures may be permitted provided that features or surfaces of interest are not masked by the leakage of the test fixture or associated fittings. Leakage from test fixtures that mask the component being hydroed must be corrected and testing repeated to permit evaluating the acceptability of the component.
- 3.6 Demineralized water corresponding to WEMD grade B or better is required for use during component hydrostatic test. In special circumstances tap water can be used for hydrostatic test and flushing (WEMD tap water meets the API requirements on chloride - <50 ppm). However, many components undergo welding operations subsequent to in process hydro's and the chlorides are to be avoided. Contact the test engineer prior to filling the pump with tap water.

TITLE: HYDROSTATIC AND HELIUM LEAKAGE TESTING OF TRANSFER PUMPS

4.0 STATOR HELIUM LEAK TESTING

- 4.1 General Description - Helium leakage testing is performed on the motor rotor and the motor stator to evaluate the effectiveness of the can welds that provide a hermetic seal. While the cans and welds are hydrostatically tested, they cannot be visually inspected during the test to evaluate the results. The hydrostatic test stresses the material and welds. The high pressure helium leakage test provides for an evaluation of the leakage integrity of these components. An initial low pressure helium leak test verifies that there are no leaks prior to the initiation of the hydrostatic test.
- 4.2 Vacuum Drying The Stator Windings - In order to provide an adequate vacuum within the stator winding cavity, it is necessary to dry the punching insulation and winding materials. These materials will have absorbed moisture from the atmosphere during the fabrication process. With the leak test connections in place bake the stator at 220-250°F, until a vacuum can be maintained sufficient to permit leakage testing (typically 24-48 hours). A cold trap shall be installed between the stator winding and the vacuum pumping system. This trap avoids back streaming of oil from the vacuum system and prevents contamination of the vacuum pumping system with water and other aromatics in the punchings and windings.
- 4.3 Stator Helium Leakage Test - Seal the terminal hole and the locations for filling with the sand/silicon potting material and evacuate the stator winding cavity. The pressure in the stator winding cavity shall be reduced to a pressure sufficiently low to permit unthrottled operation of the leak detector. The leak detector shall be calibrated prior to test per the manufacturer's instructions using a 1×10^{-4} atm cc/sec range standard leak. Carefully spray with a fine helium jet, all hoses, fittings, etc., to ensure a leak tight system. Use helium identified as UN1046 on PDS specification 52102AA.
- 4.4 Background Check - With the leak detector on line, record the ambient (background) helium leak rate indication as recorded from the leak detector system leakage rate meter. This reading establishes the threshold of detection for the system. Changes in the reading of the leakage rate meter above this level determine whether a leak exists during the helium leak test examination.
- 4.5 System Sensitivity and Response Test - With a 1×10^{-4} atm cc/sec standard leak installed in the line connecting the

TITLE: HYDROSTATIC AND HELIUM LEAKAGE TESTING OF TRANSFER PUMPS

vacuum system to the test components, open the standard leak and record the time to reach a constant value on the leak detector. Record the leak rate observed and determine the system sensitivity by subtracting the ambient level recorded in paragraph 4.4. Isolate the standard leak and verify that the ambient leakage indication returns the level obtained in paragraph 4.4.

4.6 Pre-Hydro Leak Test - Slowly spray with a fine jet of helium the following:

- 4.6.1 Upper and lower shell welds. These welds are on the exterior of the stator assembly and join the stator shell to the upper and lower stator housing
- 4.6.2 The stator can welds, top and bottom and seam if one is present.
- 4.6.3 All drilled holes in the stator assembly.

Flood the stator bore with helium at atmospheric pressure. Since helium is lighter than air, begin all spraying from the top surfaces and work down. In that way rising helium will not give an indication from a surface that has not been evaluated and the location of a leak can proceed in a systematic manner.

Fluctuations in the leakage rate meter output shall be investigated. Any repeatable increases larger than 4×10^{-9} mbar liter/sec above the initial background shall be cause for rejection.

5.0 STATOR HYDROSTATIC TEST

Install the hydrostatic test fixtures and perform a hydrostatic test per the requirements of paragraph 3.0 with the following exceptions:

- 5.0.1 The stator leak test ports may be either capped to exclude moisture, or they can be maintained connected to the leakage test system and a vacuum maintained.
- 5.0.2 The only pressurized external surface that will be visible and inspectable is the exterior of the upper stator housing. All other leakage, should it exist, will be into the stator winding cavity.

TITLE: HYDROSTATIC AND HELIUM LEAKAGE TESTING OF TRANSFER PUMPS

- 5.0.3 While it is not a requirement of this specification to "hold" the hydrostatic test pressure, caution should be exercised when performing this test. If it is necessary to constantly pump to maintain the hydrostatic pressure, and there is no indication of permissible leakage from the test fixtures and fittings, then there may be leakage into the stator winding cavity. There will be some change in the internal volume of the fixtures and stator shell as deformation of the can occurs and as the temperature of the fixtures stator and test water come to an equilibrium. A continuous addition of water during the hydro may indicate a leak. If there is a clear indication of an internal leak, the test should be suspended and the fixtures drained. An MRR should be generated and the indication investigated.

6.0 STATOR HIGH PRESSURE HELIUM LEAK TEST

- 6.1 Vent and drain the hydrostatic test fixturing. If the leak testing equipment has been removed from the stator cavity during the hydrostatic test, reconnect it, evacuate the cavity and establish the integrity of the leak test system and its sensitivity per paragraph 4.1 through 4.5.
- 6.2 Pressurize the hydrostatic test fixturing with helium, identified as UN1046 in PDS 52101AA. Pressurize to 335-342 psig in 100 psig increments. Record at each increment on the appropriate report form: the helium leak rate, the mass spectrometer total pressure, the rotor cavity pressure and the time of day. Upon reaching the required test pressure hold this condition for 30 minutes. Record the above data every 10 minutes.

Fluctuations in the meter output leak rate shall be investigated and any repeatable increase larger than 4×10^{-9} mbar liter/sec, above the initial background shall be cause for rejection.

in the event of test rejection further analysis and leak location will be performed per internal procedures.

7.0 ROTOR ASSEMBLY HYDROSTATIC TEST

Hydrostatically pressurize the rotor assembly using the rotor autoclave. The leak test plugs shall be sealed but not welded prior to this test. Pressurize the rotor assembly to the

TITLE: HYDROSTATIC AND HELIUM LEAKAGE TESTING OF TRANSFER PUMPS

hydrostatic test pressure and duration specified in section 3.0. Reduce the autoclave pressure to atmospheric pressure and remove the rotor from the autoclave. Examine the can for bulges and/or foreign material under the can. Remove the test plug caps. Dry thoroughly the external surfaces with a clean lint free cloth and examine the external welds for any indication of weepage. If there is any visual indication of internal moisture following hydrostatic test, report the condition to the cognizant test engineer. Any indication of internal moisture shall be cause for rejection.

8.0 ROTOR HELIUM LEAK TEST

After the successful completion of the hydrostatic test, the rotor core cavity will be baked and evacuated to remove moisture and to provide a helium leak test of the can.

8.1 As a preliminary system check, install the rotor in the drying oven and connect the rotor leak test plugs to the helium leak detection system. Evacuate the vacuum system and rotor until the leak detector can be operated. Establish the leak integrity and sensitivity of the helium leak detection system as described in paragraphs 4.1-4.5.

8.2 Bake the rotor in an oven for 24 hours, while evacuating. The oven temperature shall be $250 \pm 20^\circ\text{F}$ and the rotor cavity shall be less than 10 microns for a minimum of 12 hours during that time.

8.3 Carefully spray the can surface, upper and lower can welds and seam weld (if present), rotor end ring welds and all fittings with a fine jet of helium. Movement of the spraying nozzle should not exceed 3 ft./min. The oven may be flooded with helium in place of the spraying procedure. Maintain vacuum on the system until it is determined that there are no leaks. Any repeatable increase in indications larger than 4×10^{-9} mbar liter/sec. above the initial background reading shall be cause for rejection.

9.0 ROTOR CAVITY PURGE

After completion of the drying operation and satisfactory acceptance of the helium leak test, paragraph 8.0, and with the rotor still hot and evacuated, a controlled flow of dry nitrogen shall be admitted to the rotor cavity.

9.1 Isolate the rotor from the vacuum system and connect a nitrogen supply equipped with a suitable low pressure regulator to test system.

TITLE: HYDROSTATIC AND HELIUM LEAKAGE TESTING OF TRANSFER PUMPS

- 9.2 Establish the nitrogen pressure at 2-3 psig and gradually open the valve between the rotor and the nitrogen source.
- 9.3 Before removing the test connections identify the permanent rotor plugs and have them available for the next activity.
- 9.4 After establishing an equilibrium pressure of 2-3 psig in the rotor and interconnecting piping, isolate the nitrogen and disconnect the rotor from the test fittings.
- 9.5 Allow the rotor to reach atmospheric pressure. Install the permanent rotor plugs and torque to the value shown on the drawings. The rotor will be in a hot condition so caution should be exercised when handling the rotor.

10.0 APPROVALS:

Design	<u><i>R.F. Gardiner</i></u>	Date	<u>4/3/95</u>
Q.A.	<u><i>C. K. Kelton</i></u>	Date	<u>4/3/95</u>
Mfg Engr.	<u><i>M. L. Jackson</i></u>	Date	<u>4/3/95</u>
Test Engr.	<u><i>W. H. Campen</i></u>	Date	<u>4/3/95</u>
Other	<u></u>	Date	<u></u>

Revision A - Initial Revision

WESTINGHOUSE ELECTRIC CORPORATION
ELECTRO-MECHANICAL DIVISION
CHESWICK, PA 15024

Number 85310XX
Revision A
Date March 29, 1995

TITLE: PACKAGING, SHIPPING, HANDLING, AND STORAGE
OF THE MODEL NO. 2A-513-A1 TRANSFER PUMP

TABLE OF CONTENTS

1.0	SCOPE
2.0	PACKAGING
3.0	SHIPPING
4.0	STORAGE
5.0	UNCRATING
6.0	HANDLING

WESTINGHOUSE ELECTRIC CORPORATION
ELECTRO-MECHANICAL DIVISION
CHESWICK, PA 15024

Number 85310XX
Revision A
Date March 29, 1995

TITLE: PACKAGING, SHIPPING, HANDLING, AND STORAGE
OF THE MODEL NO. 2A-513-A1 TRANSFER PUMP

1.0 SCOPE

- 1.1 This specification describes the packaging, shipping, storage, uncrating, and handling procedures for the Model No. 2A-513-A1 transfer pump. It encompasses these activities at WEMD, the manufacturer, and Westinghouse Hanford Company (WHC), the purchaser of the equipment.
- 1.2 Requirements and procedures specified may be modified as necessary by means of the Manufacturing Instructions (routing) or Engineering Special Instruction (ESI).

2.0 PACKAGING

- 2.1 Packaging shall follow the general guidelines of ASME NQA-2-1989, "Quality Assurance Requirements for Nuclear Facility Applications," Level C. Any exceptions to the Level C requirements shall be detailed in this procedure.
- 2.2 Cleanliness of all pump components throughout the manufacturing and assembly process will be governed by WEMD specification 83311TB053, "Cleaning Procedure for Hazardous Waste Pumps."
- 2.3 A final cleanliness inspection of all accessible surfaces shall be performed immediately prior to packaging to verify removal of all grease, oil, soil, grit, and foreign material. Any conditions found unsatisfactory will be corrected by cleaning per 83311TB053.
- 2.4 Contact preservatives applied directly to bare metal surfaces (e.g. oil, Cosmolene, Rust-Veto) are not required.
- 2.5 Connection flanges at the top end of the pump assembly shall be sealed using cap plugs. In addition to the friction fit of the plug in the opening, the plug will be taped in place using an approved pressure-sensitive tape to prevent the plug from falling into or out of the opening.

WESTINGHOUSE ELECTRIC CORPORATION
ELECTRO-MECHANICAL DIVISION
CHESWICK, PA 15024

Number 85310XX
Revision A
Date March 29, 1995

TITLE: PACKAGING, SHIPPING, HANDLING, AND STORAGE
OF THE MODEL NO. 2A-513-A1 TRANSFER PUMP

- 2.6 Following the final cleanliness verification, the suction adapter screen shall be removed from the impeller end of the pump to allow assembly of a rotor locking device (see Figure 1). The locking device shall be made of the same grade stainless steel as the pump. The visible portion of the locking device shall be tagged to alert the installer to remove the device. The suction adapter screen shall be shipped as a separate item inside the same shipping container as the pump.
- 2.7 The pump shall be covered with 6-mil thick (minimum) clear polyethylene to provide a waterproof, vaporproof barrier material. The edges of the polyethylene shall be heat sealed where practical. Adjoining sections that cannot be readily heat sealed will be taped together using a durable, weather-resistant pressure sensitive tape. The polyethylene and tape are capable of withstanding temperatures in the -10°F to 110°F range, as specified by the Buyer.
- 2.8 The unit shall be packaged with desiccant inside the polyethylene covering. The amount of desiccant shall be calculated according to the formula in Para. 3.6.2 of ASME NQA-2-1989. The desiccant shall be packaged in 16-unit ruptureproof bags. Humidity indicators shall be packaged with the desiccant in a location that is visible through the polyethylene. Location of humidity indicators shall be marked on the outside of the shipping container.
- 2.9 The pump shall be packaged in a wooden shipping container. The lid will be nailed in place. Runners made from 4" x 4" timbers will be used under the container to allow clearance for forklift tines and lifting devices. See Figure 2 for an illustration of the shipping container.
- 2.10 The outside of the shipping container shall be identified with the following information:
- 2.10.1 Shipping Address
 - 2.10.2 Return Address

WESTINGHOUSE ELECTRIC CORPORATION
ELECTRO-MECHANICAL DIVISION
CHESWICK, PA 15024

Number 85310XX
Revision A
Date March 29, 1995

TITLE: PACKAGING, SHIPPING, HANDLING, AND STORAGE
OF THE MODEL NO. 2A-513-A1 TRANSFER PUMP

- 2.10.3 Purchase Order Number
- 2.10.4 Package Number Out of Total Number (if required)
- 2.10.5 Shipping Weight
- 2.10.6 Handling Instructions (lifting points, center of gravity)
- 2.10.7 Location of Humidity Indicators

- 2.11 Information shall be marked using stencils and paint and/or waterproof marking pen. Character height to be .5" minimum. Crate shall be marked on the top and one side.

3.0 SHIPPING

- 3.1 Westinghouse EMD form PA 2217 will be used as shipping documentation (see Figure 4). This form shall contain the following information:

- 3.1.1 Name of Seller
- 3.1.2 Name of Buyer
- 3.1.3 Shipping Address
- 3.1.4 WEMD Shop Order Number
- 3.1.5 Purchase Order Number and Item Number
- 3.1.6 Itemized List of Supplies or Services Furnished (includes part number, revision, serial number if applicable, description, and WEC stock number).
- 3.1.7 Quantity of Each Item
- 3.1.8 Shipment Date
- 3.1.9 Verification of Information Accuracy by the Production Planner, Packer, and Inspector

- 3.2 The lead unit shall be shipped fully assembled, with the exception of the suction adapter screen, in the wooden shipping container described in Section 1. The unit shall be shipped horizontally on a flatbed trailer as long or longer than the shipping container. Subsequent production units will be shipped in two or more shipping containers; one for the pump/column assembly and the remaining containers for the top mounting flange and connection piping to be fit up and welded in the field.

WESTINGHOUSE ELECTRIC CORPORATION
ELECTRO-MECHANICAL DIVISION
CHESWICK, PA 15024

Number 85310XX
Revision A
Date March 29, 1995

TITLE: PACKAGING, SHIPPING, HANDLING, AND STORAGE
OF THE MODEL NO. 2A-513-A1 TRANSFER PUMP

- 3.3 The pump will be loaded on the trailer using nylon web slings (or equivalent lifting devices) rated for a 10,000 lbs. lift. Two slings shall be used in a choke or basket hitch around the shipping container. Slings are to be located about the center of gravity at the lifting points marked on the shipping container. One crane may be used for lifting the load, provided that it is carefully balanced and remains horizontal. Two cranes are recommended for added stability. See Figure 2.
- 3.4 The shipping container shall be secured on the trailer using chains or straps. A tarpaulin shall be used over the container for weather protection and security. The unit shall remain on the same trailer for the duration of the shipment.
- 3.5 The pump shall be unloaded from the trailer at the buyer's facility using the lifting devices and methods described in Para. 3.3.

4.0 STORAGE

- 4.1 The pump is packaged to allow outdoor storage inside the wooden shipping container for the life of the shipping container. As required in the Buyer's specification, the packaging can withstand rain, snow, wind, blowing sand, and temperature ranges in excess of -10°F to 110°F for an extended period of time. This time period is dependent upon the weather to which the crate is subjected. Outdoor storage time may be extended by covering the shipping container with a tarpaulin and providing as much protection from the elements as possible.
- 4.2 During long-term storage, desiccant shall be replaced by the Buyer when humidity indicators (provided) show that the desiccant is saturated. Location of humidity indicators shall be marked on the shipping container.

WESTINGHOUSE ELECTRIC CORPORATION
ELECTRO-MECHANICAL DIVISION
CHESWICK, PA 15024

Number 85310XX
Revision A
Date March 29, 1995

TITLE: PACKAGING, SHIPPING, HANDLING, AND STORAGE
OF THE MODEL NO. 2A-513-A1 TRANSFER PUMP

- 4.3 Periodic maintenance (i.e., shaft rotation) of the transfer pump during long term storage is not required.

5.0 UNCRATING

- 5.1 Remove lid from the top of the shipping container.
- 5.2 Lift pump free of the shipping container using two nylon lifting straps rated for a 10,000 lb. lift in a basket or choke hitch. Straps are to be located about the center of gravity of the unit at the lifting points (see Figure 3). Lift the pump horizontally above the shipping container as required to allow removal of the overwrap material. Remove packaging material, desiccant, and humidity indicators and discard.
- 5.3 Disassemble the rotor locking device from the impeller end of the assembly, assemble the suction adapter screen, and lock the lockplate tabs for the .375" dia. screen retaining hardware per the instructions provided. Return locking device to WEMD.
- 5.4 Remove cap plugs, covers, and tape from all sealed openings. Remove tape residue by cleaning with a WEC-approved solvent.

6.0 HANDLING

- 6.1 Transfer the pump to the tank where it will be installed by trucking on a flatbed trailer as described in Para. 3.3 of this procedure, except the lifting straps are to be used directly around the column assembly. Use blocking under the pump on the trailer to prevent shifting. The WEMD-designed transportation cart (see Figure 3) may be used as an alternate.
- 6.2 After field welding (not required on lead unit) has been performed on the assembly in the horizontal position, the top adapter flange can be used for lifting. Assemble the WEC lifting bail on the adapter flange.

WESTINGHOUSE ELECTRIC CORPORATION
ELECTRO-MECHANICAL DIVISION
CHESWICK, PA 15024

Number 85310XX
Revision A
Date March 29, 1995

TITLE: PACKAGING, SHIPPING, HANDLING, AND STORAGE
OF THE MODEL NO. 2A-513-A1 TRANSFER PUMP

- 6.3 To rotate the pump assembly to a vertical position, first assemble the group 2 lift ring assembly of WEMD transport cart 05-50093 to the bottom end of the pump. See Figure 3. Lift the pump assembly horizontally as described in para. 5.2. Assemble pump to transport cart per Figure 3. Remove the nylon straps and connect the hoist to the WHC lifting bail on the adapter flange. Lift the pump from a horizontal to vertical position, allowing the transport cart to roll. When the pump is in the vertical position, remove the cart lift ring assembly.

APPROVAL:

Design	<u>R. F. Garbin</u>	Date	<u>3/31/95</u>
PQI	<u>J. C. Casella</u>	Date	<u>3/31/95</u>
Mfg Engrg	<u>M. Kinney</u>	Date	<u>3/31/95</u>
Others	_____	Date	_____

FIGURE 1 - ROTOR LOCKING DEVICE

NOTE: SUCTION ADAPTER SCREEN WILL BE REMOVED FOR SHIPMENT

FIGURE 3 - TRANSPORTATION CART

FIGURE 2 - SHIPPING CONTAINER / LIFTING METHOD

FIGURE 4 - SHIPPING DOCUMENTATION

WESTINGHOUSE ELECTRIC CORPORATION - EMD - Cheswick, PA
SMAW OF SST BASE METAL, P No. 8*, USING FILLER METAL,
F NO. 5*, A No. 8*, (E308).

Welding Procedure
82121PQ0 06
Rev. D

Procedure Qualification Code Nos.
21001 (1/4" & 1-1/2")

WELDING PARAMETERS

PROCESS	SMAW-MANUAL**
Electrical Characteristics	DCRP**
Base Material Type	304 or 316 SST
Electrode Type	SFA 5.4**, E308
Preheat Temperature	60°F Minimum*
Interpass Temperature	350°F Maximum.
Post Weld Heat Treatment	None*
Position of Welding	6G **

Typical Joint Designs**
Backing Material - P No. 8**

Thickness Range*:
Base Metal; .06" to 8"
Deposited Weld Metal; 8" Max.

SOLID ROD ELECTRODE

DIA.**	AMPS**	VOLTS**
5/64	30-60	21-23
3/32	60-90	22-24
1/8	90-125	22-24
5/32	120-160	22-25
3/16	150-190	23-26
1/4	225-300	24-27

STOODY FABRICATED ELECTRODE

DIA.**	AMPS**	VOLTS**
3/32	75-95	22-24
1/8	95-115	22-24
5/32	130-150	22-25
3/16	170-190	23-26

GENERAL NOTES

1. All welding and inspection shall be in accordance with ASME BAPV Code, Section III.
2. Welders welding to this procedure shall be qualified in accordance with the ASME BAPV Code, Section IX.
3. Nondestructive testing shall be performed as instructed on applicable manufacturing drawing.
4. Arrangement of parts to be welded with spacing and details of the welding groove, if used, shall be shown with appropriate Engineering Drawing or GP Sketch.**

REPAIR NOTES

This procedure may also be used to repair weld metal or base metal defects caused by grinding or machining errors or other related causes. The following steps must be followed:

1. Prepare the defective area to a suitable contour for welding.
2. Clean the surface to be welded and adequately protect the surrounding surfaces.
3. Weld per above parameters.
4. Machine or grind to restore acceptable surface contour or dimensions.
5. Perform NDT as specified on drawing or applicable repair instructions.
6. No limit on minimum weld deposit.

WELDING NOTES

1. Clean assembly with alcohol or acetone dampened cloth prior to welding.**
2. All starts and stops shall be visually checked for excessive build-up or crater cracks and ground, if necessary.
3. Each weld bead shall be cleaned by wire brushing or grinding to remove oxides and slag prior to depositing subsequent beads.**
4. Measure preheat and interpass temperature with contact pyrometer.
5. Back gouging shall be performed by grinding or machining.**
6. Peening shall not be performed unless instructed by drawing, GP Sketch, or other instructions.**
7. Stringer bead applied on first and second layers. Weave bead may be applied on other layers with max. weave of 3 electrode dia.**
8. No single pass may exceed 1/2" in depth.*
9. Vertical position use upward progression.**
10. Nonmetallic retainers shall be used only as given by other instructions.**

*Essential Variable

**Nonessential Variable

R. Vtch 10-12-84
J. Geralt 10-12-84
D. P. 10/12/84

WESTINGHOUSE ELECTRIC CORPORATION
ELECTRO-MECHANICAL DIVISION

Procedure Qualification Record No. 21001 Date June 10, 1976
WPS No.(s) 82121PQ003
Welding Process(es) SMAW Types Manual
(Manual, Automatic, Semi-Aut.)

JOINTS (QW-402) Groove Design Used

Drawing No. 1748C07 G02 Rev. 2

BASE METALS (QW-403)

Material Spec. ASME SA 240
Type or Grade 304 To 316
P No. 8 to P No. 8
Thickness 1/4"
Diameter _____
Other Plate

FILLER METALS (QW-404)

Weld Metal Analysis A No. 8
Size of Electrode 3/32" & 1/8"
Filler Metal F No. 5
SFA Specification 5.4
AWS Classification 5.4 E308-16 R. Smith 5/1/95
Other Stoody Fabricated Electrode

POSITION (QW-405)

Position of Groove Flat
Weld Progression _____
(Uphill, Downhill)
Other _____

PREHEAT (QW-406)

Preheat Temp. 60°F
Interpass Temp. 350°F Max.
Other _____

POSTWELD HEAT TREATMENT (QW-407)

Temperature NA
Time NA
Other _____

GAS (QW-408)

Type of Gas or Gases NA
Composition of Gas Mixture NA
Other NA

ELECTRICAL CHARACTERISTICS (QW-409)

Current DC
Polarity Reverse
Amps. 80-105 Volts 22-24
Travel Speed _____
Other _____

TECHNIQUE (QW-410)

String or Weave Bead String
Oscillation None
Multipass or Single Pass Multiple
(per side)
Single or Multiple Electrodes Single

TENSILE TEST (QW-150)

SPECIMEN NO.	WIDTH	THICKNESS	AREA	ULTIMATE TOTAL LOAD LB.	ULTIMATE UNIT STRESS PSI	CHARACTER OF FAILURE & LOCATION
1	1.509	0.228	.3441	31,700	92,100	Base
2	1.500	0.230	.3450	31,850	92,300	Base

FERRITE TESTS

SPECIMEN	FERRITE	TYPE OF TEST
Weld Face	8.75 - 11.0	Ferrite Scope
Weld Face	9.0 - 11.6	Magne Gage

GUIDED BEND TESTS (QW-160)

TYPE AND FIGURE NO.	RESULT	TYPE AND FIGURE NO.	RESULT
Face	No Defects	Root	No Defects
Face	No Defects	Root	No Defects

TOUGHNESS TESTS (QW-170)

SPECIMEN NO.	NOTCH LOCATION	NOTCH TYPE	TEST TEMP.	IMPACT VALUES	LATERAL EXP.		DROP WEIGHT	
					%SHEAR	MILS	BREAK	NO BREAK

Type of Test _____
Deposit Analysis _____
Other _____

FILLET WELD TEST (QW-180)

Result - Satisfactory NA Penetration into Parent Metal NA
Yes, No
Type and Character of Failure NA Macro-Results NA
Yes, No

Welder's Name E. F. Pleva Clock No. _____ Stamp No. AE 7
Tests Conducted By: PTL Laboratory Test No. 762509
Per: ASME Section IX Weld Record No. 37135

We certify that the statements in this record are correct and that the test welds were prepared, welded, and tested in accordance with the requirements of Section IX of the ASME Code. Addenda Summer, 1976

Signed Westinghouse Electric Corporation
Electro-Mechanical Division
(Manufacturer)

Date July 27, 1976 By P. Stok

QW-483 Procedure Qualification Record (PQR)
(See QW-201.2)

RPP-5117, Rev 0
Addendum A-445

WESTINGHOUSE ELECTRIC CORPORATION
ELECTRO-MECHANICAL DIVISION

Procedure Qualification Record No. 21001 Date June 10, 1976
WPS No.(s) 82121PQ003
Welding Process(es) SMAW Types Manual
(Manual, Automatic, Semi-Aut.)

JOINTS (QW-402) Groove Design Used

Drawing No. 1748C07 601 Rev. 2

BASE METALS (QW-403)

Material Spec. ASME SA 240
Type or Grade 304 To 316
P No. 8 to P No. 8
Thickness 1-1/2"
Diameter _____
Other Plate

FILLER METALS (QW-404)

Weld Metal Analysis A No. 8
Size of Electrode 3/32, 1/8, 5/32, & 3/16
Filler Metal F No. 5
SFA Specification 5.4
AWS Classification A5.4 E308-16 R *5/1/93*
Other Stoody Fabricated Electrode

POSITION (QW-405)

Position of Groove Flat
Weld Progression _____
(Uphill, Downhill)
Other _____

PREHEAT (QW-406)

Preheat Temp. 60°F
Interpass Temp. 350°F Max.
Other _____

POSTWELD HEAT TREATMENT (QW-407)

Temperature NA
Time NA
Other _____

GAS (QW-408)

Type of Gas or Gases NA
Composition of Gas Mixture NA
Other NA

ELECTRICAL CHARACTERISTICS (QW-409)

Current DC
Polarity Reverse
Amps. 80 - 180 Volts 22-25
Travel Speed _____
Other _____

TECHNIQUE (QW-410)

String or Weave Bead String
Oscillation None
Multipass or Single Pass Multiple
(per side)
Single or Multiple Electrodes Single

TENSILE TEST (QW-150)

SPECIMEN NO.	DIA.	THICKNESS	AREA	ULTIMATE TOTAL LOAD LB.	ULTIMATE UNIT STRESS PSI	CHARACTER OF FAILURE & LOCATION
1A	0.505		0.2003	17.600	87.900	Base
1B	0.506		0.2011	17.600	87.500	Base
2A	0.506		0.2011	17.300	86.000	Base
2B	0.505		0.2003	17.250	86.100	Base

FERRITE TESTS

SPECIMEN	FERRITE	TYPE OF TEST
Weld Face	10.2 - 12.4	Magne Gage
Weld Face	10.5 - 12.0	Ferrite Scope

GUIDED BEND TESTS (QW-160)

TYPE AND FIGURE NO.	RESULT	TYPE AND FIGURE NO.	RESULT
Side Bend	No Defects	Side Bend	No Defects
Side Bend	No Defects	Side Bend	No Defects

TOUGHNESS TESTS (QW-170)

SPECIMEN NO.	NOTCH LOCATION	NOTCH TYPE	TEST TEMP.	IMPACT VALUES	LATERAL EXP.		DROP WEIGHT	
					%SHEAR	MILS	BREAK	NO BREAK

Type of Test _____
Deposit Analysis _____
Other _____

FILLET WELD TEST (QW-180)

Result - Satisfactory NA Penetration into Parent Metal NA
Yes, No
Type and Character of Failure NA Macro-Results NA
Yes, No

Welder's Name E. F. Pleva Clock No. _____ Stamp No. AE 7
Tests Conducted By: PTL Laboratory Test No. 762509
Per: ASME Section IX Weld Record No. 37144

We certify that the statements in this record are correct and that the test welds were prepared, welded, and tested in accordance with the requirements of Section IX of the ASME Code. Addenda Summer, 1976

Signed Westinghouse Electric Corporation
Electro-Mechanical Division
(Manufacturer)

Date July 27, 1976 By R. Lth

WESTINGHOUSE ELECTRIC CORPORATION - EMD - CHESWICK, PA
GTAW OF SST BASE METAL, P NO. 8* WITH FILLER METAL,
F NO. 6*, A NO. 8*. (ER308)

Welding Procedure
82127P0006
Rev. K

Procedure Qualification Code No.
934D064G02 and 27045

PARAMETERS

Typical Joint Designs**

Consumable Insert - P No. 8*
Backing Material - P No. 8**

Thickness Range**

Base Metal; .06" to 8"
Deposited Weld Metal; 8" Max.

TUNGSTEN ELECTRODE

Process	GTAW-Manual***
Base Metal	Type 304 or 316
Filler Metal	SFA 5.9* E308
Preheat Temperature	60°F Min.*
Interpass Temperature	350°F Max.
Post Weld Heat Tr.	None*
Position	6G **
Gas Orifice Size	.25"-0.5" I.D.**
Shielding/Backing Gas	Argon, 99.99*
Shielding Gas Flow	13-18 CFH**
Backing Gas Flow	4-10 CFH**
Electrode Type	EWth-2**
Electrode Extension	3/8" Max.
Electrical Character.	DCSP**
Method of Arc Initiation	High Frequency
Welding Voltage	8-16 Volts**

Electr. Dia. In.**	Filler Metal** Dia. In.	Amperage**
1/16	.020-.045	30-80
1/16	EB Insert	75-110
1/16	1/16	75-115
3/32	1/16	100-150
3/32	3/32	100-170

GENERAL NOTES

- All welding and inspection shall be in accordance with the ASME Boiler and Pressure Vessel Code, Section III.
- Nondestructive testing shall be performed as instructed on applicable manufacturing drawing.
- Arrangement of parts to be welded with spacing and details of the welding groove, if used, shall be shown with appropriate Engineering drawing or GPX.**
- This procedure may be used to repair weld metal or base metal defects as outlined by the following:
 - Prepare the defective area to a suitable contour for welding.
 - Clean the surface to be welded and adequately protect the surrounding surfaces.
 - Weld per above parameters with single pass permitted.**
 - Machine or grind to restore acceptable surface contour or dimensions.
 - Perform NOT as specified on drawing or applicable repair instructions.
 - No limit on minimum weld deposit.

WELDING NOTES

- Clean assembly with alcohol to acetone dampened cloth prior to welding.**
- Decay weld current prior to extinguishing arc to minimize crater size.
- Examine craters and grind, if necessary, to remove cracks.
- Clean each weld bead to remove any oxide scale prior to depositing the next bead. Grinding or wire brushing may be employed.**
- Measure preheat and interpass temperatures with contact pyrometer.
- Required only for full penetration welds.
- Peening shall not be performed unless instructed by drawing, GPX or other instruction.**
- String bead applied on first and second layers. Weave bead may be applied on other layers with max. weave of 3 electrode dia.**
- Sack gauging shall be performed by grinding or machining.**
- Vertical position use upward progression.**
- No single pass may exceed 1/2" in depth.*
- Multipass per side preferred but single pass permitted.**
- Pulsing current not to be applied.**
- Nonmetallic retainers shall not be used.**

*Essential Variable
**Nonessential Variable

R. V. 10-12-84
J. J. 10-12-84
K. J. 10/13/84

MANUFACTURER'S RECORD OF WELDING PROCEDURE QUALIFICATION TESTS

RPP-5117, Rev 0
Addendum A-448

Specification No. 82127PQ - 934D064G02 Date November 6, 1972
 Welding Process Gas Tungsten Arc Manual or Machine Manual
 Material Specification Equiv. SA312, Equiv. SA312 P-No. 8 to P-No. 8
 Thickness (if pipe, diameter and wall thickness) Pipe 2.25" O.D. x 0.30" wall
 Thickness Range this test qualifies .062" to .60"
 Filler Metal Group No. F- 76 R. Evelyn 4/20/95 FLUX OR ATMOSPHERE
 Weld Metal Analysis No. A- 76 R. Evelyn 4/20/95 Flux Trade Name or Composition NA
 Describe Filler Metal if not included in Table Q-11.2 Inert Gas Composition Argon
 or QN-11.2 NA Trade Name Air Products Flow Rate 15-20 cfm
 For oxyacetylene welding—State if Filler Metal is sil- Is Backing Strip used? No
 icon or aluminum killed. Preheat Temperature Range 70° F
 WELDING PROCEDURE Interpass Temperature Range 250° F
 Single or Multiple Pass Multiple Pass Postheat Treatment NA
 Single or Multiple Arc Single Arc
 Position of Groove 2G (See Pars. & Figs. Q-2 & Q-3, or QN-2 & QN-3)
 (Flat, horizontal, vertical, or overhead; if vertical, state whether upward or downward)

FOR INFORMATION ONLY

Filler Wire—Diameter 1/8" EB Insert, 1/16" wire WELDING TECHNIQUES
 Trade Name NA Joint Dimensions Accord with 934D064G02
 Type of Backing None Filler Metal Amps (DC) Volts (SP)
 Forehand or Backhand Forehand EB Insert 75-110 9-13
 1/16" Dia. 75-115 9-13

REDUCED SECTION TENSILE TEST (Figs. Q-6 and QN-6)

Specimen No.	Dimensions		Area	Ultimate Total Load, lb.	Ultimate Unit Stress, psi	Character of Failure and Location
	Width	Thickness				
1 - B	O.D. 2.246	I.D. 1.630	1.875	152,500	81,350	Base Metal
2 - B	O.D. 2.248	I.D. 1.650	1.830	150,000	81,950	50% Weld 50% Base Metal

GUIDED BEND TESTS (Figs. Q-7.1, Q-7.2, QN-7.1, QN-7.2, QN-7.3)

Type and Figure No.	Result	Type and Figure No.	Result
6 Face Q-7.2	No Defects		
6 Root Q-7.2	No Defects		

Results of Fillet Weld Tests, Fig. Q-9(c) NA
 Welder's Name J. Rosepink Clock No. NA Stamp No. AE14
 Who by virtue of these tests meets welder performance requirements.
 Test Conducted by Pittsburgh Testing Lab Laboratory—Test No. 732273
 per Earl Gallagher

We certify that the statements in this record are correct and that the test welds were prepared, welded and tested in accordance with the requirements of Section IX of the ASME Code.

Westinghouse Electric Corp.

Signed Electro-Mechanical DivisionJ. W. Sindley, P.E.

1-19-73

Date

WESTINGHOUSE ELECTRIC CORPORATION
ELECTRO-MECHANICAL DIVISION

Procedure Qualification Record No. 27045 Date March 8, 1981
WPS No.(s) 82127PQ007, Rev. L
Welding Process(es) GTAW Types Mach.
(Manual, Automatic, Semi-Aut.)

BASE METALS (QW-403)
Material Spec. SA240
Type or Grade 304
P No. 8 to P No. 8
Thickness 1-7/8" Plate
Diameter _____
Other _____

FILLER METALS (QW-404)
Weld Metal Analysis A No. 8
0.125" EB Insert
Size of Electrode 0.045" Dia. Wire
Filler Metal F No. 6
SFA Specification 5.9
AWS Classification E308
Other Electrode 5/32" Dia., EWTh-2

POSITION (QW-405)
Position of Groove 1G
Weld Progression _____
(Uphill, Downhill)
Other _____

POSTWELD HEAT TREATMENT (QW-407)
Temperature None

PREHEAT (QW-406)
Preheat Temp. 60°F Min.
Interpass Temp. 350°F Max.
Other _____

ELECTRICAL CHARACTERISTICS (QW-409)
Elec. Characteristic - DCSP

PASS	ARC AMPS	ARC VOLTS	WORK SPEED IPM	WIRE FEED IPM
Seal	60	8	2	---
Fusion	100	8-9	2	---
Third	200	12	5-8	75
Fourth	250	12	5-8	75
Bal.	300	14	5-8	90

GAS (QW-408)
Type of Gas or Gases Argon
Composition of Gas Mixture _____
Other Backing Gas - Argon

TECHNIQUE (QW-410)
String or Weave Bead String
Oscillation N/A
Multipass or Single Pass Multi
(per side)
Single or Multiple Electrodes Single

TENSILE TEST (QW-150) QW 462.1(a)

SPECIMEN NO.	WIDTH	THICKNESS	AREA	ULTIMATE TOTAL LOAD LB.	ULTIMATE UNIT STRESS PSI	CHARACTER OF FAILURE & LOCATION
F1	0.997	0.764	0.7617	67,800	89,000	Weld, Ductile
R1	1.000	0.789	0.7890	71,500	90,600	Weld, Ductile
F2	1.000	0.775	0.7750	68,800	88,800	Weld, Ductile
R2	0.999	0.774	0.7732	69,400	89,800	Weld, Ductile

FERRITE TESTS (PERFORMED BY WEMD)

SPECIMEN	FERRITE	TYPE OF TEST
Weld Face	10 < FN < 12.5	Severn Gage
Weld Root	5 < FN < 7.5	Severn Gage
Six Measurements Were Taken Along The Face And Also Along The Root of the Weld		

GUIDED BEND TESTS (QW-160)

TYPE AND FIGURE NO.	RESULT	TYPE AND FIGURE NO.	RESULT
QW462.3(a) F1	1/16" Fissure Edge	QW462.3(a) R2	No Defects
QW462.3(a) R1	No Defects		
QW462.3(a) F2	No Defects		

TOUGHNESS TESTS (QW-170)

SPECIMEN NO.	NOTCH LOCATION	NOTCH TYPE	TEST TEMP.	IMPACT VALUES	LATERAL EXP.		DROP WEIGHT	
					ZSHEAR	MILS	BREAK	NO BREAK

Type of Test ASTM 262 Practice E: Four Bends, Weld and HAZ, Face and Root, Mandrel 4T, Acceptable Per Specification

FILLET WELD TEST (QW-180)

Result - Satisfactory _____ Penetration into Parent Metal _____
Yes, No Yes, No

Type and Character of Failure _____ Macro-Results _____

Welder's Name A. M. Delenne Clock No. _____ Stamp No. _____
Tests Conducted By: I.T.L.S. Laboratory Test No. 30798-1, 2
Per: Dr. G. T. Horne Weld Record No. QA073 No. 72934

We certify that the statements in this record are correct and that the test welds were prepared, welded, and tested in accordance with the requirements of Section IX of the ASME Code. 1980

Westinghouse Electric Corporation
Signed Electro-Mechanical Division
(Manufacturer)

Date March 19, 1981

By F. Robert Altch

SPECIMEN NO.	WIDTH IN.	THICK. IN.	AREA IN. ²	ULTIMATE TOTAL LOAD LB.	ULTIMATE UNIT STRESS PSI	CHARACTER OF FAILURE & LOCATION

TYPE AND FIGURE NO.	RESULT	TYPE AND FIGURE NO.	RESULT
QW 462.2(a) SB-1	1/16" Fissure, Edge		
QW 462.2(a) SB-2	No Defects		
QW 462.2(a) SB-3	No Defects		
QW 462.2(a) SB-4	No Defects		

[illegible]

Page 2 of 2

Form No. 302 WE
Date: Feb. 23, 1989

Welder's Qualification Code No. (s)
CMM070805A-

QW-484 WESTINGHOUSE ELECTRO-MECHANICAL DIVISION
RECORD OF WELDER OR WELDING OPERATOR QUALIFICATION TEST

RPP-5117, Rev C
Addendum A-45

Welder Name D. Majka Check No. _____ Stamp No. 0062

Using WPS No. 82121PQ003 Rev. L
The Above Welder Is Qualified For The Following Ranges

VARIABLE	RECORD ACTUAL VALUES USED IN QUALIFICATION	QUALIFICATION RANGE
Process	<u>SMAW</u>	<u>SMAW</u>
Process Type	<u>Manual</u>	<u>Manual</u>
Welding [Metal, Weld Metal, Flux, Etc. (QW-402)]	<u>Metal</u>	<u>Metal, Weld Metal</u>
Material Spec. (QW-403)	<u>P No. 8 to P No. 8</u>	<u>P No. 1 to P No. 1</u>
Thickness	<u>0.77"</u>	<u>All Thicknesses</u>
Groove	<u>Not Applicable</u>	<u>All Thicknesses</u>
Fillet	<u>4" O.D.</u>	<u>2 7/8" and over</u>
Weld Diameter	<u>Not Applicable</u>	<u>Any Diameter</u>
Groove	<u>SFA 5.4</u>	<u>SFA 5.4</u>
Fillet	<u>E308</u>	<u>A No. 8</u>
Weld Metal (QW-404)	<u>F5</u>	<u>F5</u>
Spec. No.	<u>2G</u>	<u>1G & 2G</u>
Class	<u>Not Applicable</u>	<u>Not Applicable</u>
-No.	<u>Not Applicable</u>	<u>Not Applicable</u>
Position (QW-405)		
Weld Progression (QW-410)		
Type (QW-408)		
Physical Characteristics (QW-409)		
Weld Joint		
Polarity	<u>DC</u>	<u>DC, AC</u>
Additional P No. Combinations Qualified	<u>RP</u>	<u>RP, SP</u>

GUIDED BEND TEST RESULTS QW-462.2(a), QW-462.3(a), QW-462.3(b)
Type & Figure No. _____ Result _____

Radiographic Results: For alternative qualification of groove welds by radiography in accordance with QW-304 and QW-305: Film No. 11340 Result Acceptable

Test Conducted By WEMD Laboratory Test No. WPQ 89-120
Per ASME Sect. IX Weld Record No. 114095

Additional Comments _____

I certify that the statements in this record are correct and that the test welds were prepared, welded, and tested in accordance with the requirements of Section IX of the ASME Code.

Applicable Code Addenda 1986 Edition thru 87A

Welder's Original Date June 22, 1989

Signed Electro-Mechanical Division
Westinghouse Electric Corp.
(Organization)

By R. Ntch

Inspector Verified

Form No. 302 WE
Date: July 29, 1982

Welder's Qualification Code No.(s)

CTM070804A-

QW-484 WESTINGHOUSE ELECTRO-MECHANICAL DIVISION
RECORD OF WELDER OR WELDING OPERATOR QUALIFICATION TEST

RPP-5117, Rev 0
Addendum A-453

Welder Name Majka Check No. _____ Stamp No. AE61

Using WPS No. 82127P0006 Rev. K
The Above Welder Is Qualified For The Following Ranges

VARIABLE	RECORD ACTUAL VALUES USED IN QUALIFICATION	QUALIFICATION RANGE
Process	GTAW	
Process Type	Manual	
Welding [Metal, Weld Metal, Flux, Etc. (QW-402)]	Metal	
Material Spec. (QW-403)	SA479 to SA479	to
Base Material P No.	P No. 8 to P No. 8	*P No. 8 to P No. 8
Thickness	0.39"	0.78" Max.
Root Face		Unlimited
Root Fillet		
Root Penetration	4" O.D.	Dia. > 2 7/8"
Root Fillet		All Dia.
Weld Metal (QW-404)		
Spec. No.	SFA 5.9	
Class	ER 308	
-No.	F6	
Position (QW-405)	2G	2G, 1G
Shielding Gas Progression (QW-410)	NA	
Shielding Gas Type (QW-408)	Argon	
Physical Characteristics (QW-409)		
Current	DC	
Polarity	SP	
Additional P No. Combinations Qualified		P8 - P8

GUIDED BEND TEST RESULTS QW-462.2(a), QW-462.3(a), QW-462.3(b)
Type & Figure No. Result

Radiographic Results: For alternative qualification of groove welds by radiography in accordance with QW-304 and QW-305: Film No. 9618 Result Accept

Test Conducted By Westinghouse Electric Corp. Laboratory Test No. WPQ 85-161
Per ASME Sect. IX, 1983, Su 85 Weld Record No. 84934

Additional Comments _____

I certify that the statements in this record are correct and that the test welds were prepared, welded, and tested in accordance with the requirements of Section IX of the ASME Code.

Welder's Original Date 7-31-85 Signed _____
Cable Code Addenda Su 85 Electro-Mechanical Division
Westinghouse Electric Corp.

By R. Stoltz 9-19-85
(Organization)

Off Verified

17.0 DISTRIBUTION LIST

**RPP-5117, Rev 0
Addendum A-454**

Mr. K. S. Hoeft, WHC (7 Total, 1 Unbound)

Mr. A. E. Hribar

Mr. J. A. Drake

Mr. R. F. Guardiani

Mr. W. B. Hellman

Mr. C. P. Horomanski

Mr. M. A. Lamantia

Mr. L. C. McNutt

Mr. A. L. Morris

Mr. J. R. Raymond

Mr. J. A. Zastawniak

Internal File

APPENDIX B

Westinghouse Electric Corporation
Electro-Mechanical Division

**Model 2A-513-A1
Transfer Pump
Lead Unit Test Report**

Customer:	PHMC
Pump Serial Number:	001
Motor Unit Serial Number:	201
Shop Order Number:	V002
Contract Number:	MKH-SXP-342098
General Assembly Drawing Number:	4D97585G01, Rev. 3
Motor Unit Assembly Drawing Number:	4D97587G01, Rev. 7
Outline Drawing Number:	4D97586, Rev. 1
Test Specification:	348A53, Rev. C

Written by:

R. Guardian

Date:

9-26-97

Approved by:

James A. Drake PE

Date:

10-1-97

INTRODUCTION

Water testing of the lead unit (S/N 001) Model 2A-513-A1 transfer pump was performed in two separate locations.

The 24-hour test was performed at the Westinghouse Savannah River Site (WSRS) on June 25, 1997 and June 26, 1997. The testing was performed on the complete pump, which consists of the motor unit assembly and the column assembly. The pump was suspended from the top of the column assembly and the motor unit assembly plus two feet of column assembly was submerged in an open tank of water.

The Rated Speed Test and the Off-Speed Tests were performed at WEMD in P-Loop on September 15 and 16, 1997. These tests involved only the motor unit assembly (short-set); the column assembly was not attached. The motor unit assembly was completely submerged in an open tank of water during testing.

Test Date: 6/25/97 through 6/26/97
 Test Location: WSR

24 Hour Run - Specification 348A53-C Section 4.0

Beginning Time	Ending Time	Avg. Flow (gpm)	Avg. Head (feet)	Water Temperature (°F)	Variable Frequency Drive Output			Vibration (mils pk-pk)
					Frequency (Hz)	Voltage (Volts)	Current (amps)	
1707	2245	140	338	80.2	67	412	90	0.4-1.3
2255	2332	141	333	77.5	67	412	90	.056-.21 ⁽²⁾
2335	0046	136	305	77.9	67	412	95	.031-.10 ⁽³⁾
0052	0800	140	330	78.3	67	452	84	.07-.22 ⁽⁴⁾
0816	0840	140	334	78.3	67	452	84	.066-.24 ⁽⁵⁾
0900	1558	141	333	78.4	67	456	83	0.4-2.1
1604	1808	142	333	78.7	67	456	83	0.4-2.0

Total Operation Time: 24 Hours
 Total Start-Stops: 6

- (1) Instrumentation not working properly.
 (2) Reading is in inches/s. This translates to 0.3-1.1 mils pk-pk at 3600 rpm.
 (3) Reading is in inches/s. This translates to 0.2-0.5 mils pk-pk at 3600 rpm.
 (4) Reading is in inches/s. This translates to 0.4-1.2 mils pk-pk at 3600 rpm.
 (5) Reading is in inches/s. This translates to 0.4-1.3 mils pk-pk at 3600 rpm.

Comments/Evaluation - 24 Hour Run Per 348A53-C, Section 4.0

The 24-hour test run was started at 17:07 on June 25, 1997 and was completed on June 26, 1997 at 18:08. The test sequence was as follows:

17:07	start 24-hour test on 67 hz, 140 gpm flow
22:45	stop pump
22:55	start pump on 67 hz to resume 24-hour run
23:32	stop pump
23:35	start pump on 67 hz to resume 24-hour run
00:46	stop pump
00:52	start pump on 67 hz to resume 24-hour run
08:00	slow pump speed to 30 hz
08:06	change pump speed to 40 hz
08:10	change pump speed to 50 hz
08:14	change pump speed to 60 hz
08:16	increase pump speed to 67 hz to resume 24 hour run
08:40	stop pump
09:00	start pump on 67 hz to resume 24-hour run
15:58	stop pump
16:04	start pump on 67 hz to resume 24-hour run
18:08	stopped pump to complete 24-hour run at 67 hz

The 24-hour run was conducted at 67 hz input frequency from the VFD at 140 gpm flow and 330 ft. of head at the discharge flange in lieu of the required 140 gpm flow and 450 feet of head at the discharge flange. At the time of the test, 450 feet of head could not be achieved. Subsequent testing at WEMD revealed an approximate 80 gpm leakage back to pump suction from 2nd stage diffuser discharge through the dilution lines. The actual flow, though the hydraulics during the 24-hour run was, therefore, approximately 220 gpm (at 330 feet of head).

During the testing the VFD parameters were changed to increase the 67 Hz supply voltage from 412 volts to 456 volts maximum. This resulted in a decrease in supply current from 90 amps to 83 amps minimum.

There were a total of 6 start-stops and 5 speed changes during the 24-hour run. Operation at less than 67 hz was not included in the 24-hour test run.

**TRANSFER PUMP P-LOOP HEAD-FLOW TESTING AT 63 HZ (450 FEET @ 140 GPM DESIGN POINT)
IN ACCORDANCE WITH SPECIFICATION 348A53-C SECTION 5.0**

Test Date: 9-15-97

Test Location: WEMD

Delivered Flow (gpm)	Motor Unit Head (ft)	Delivered Head (ft)	Operating Speed (rpm)	Input Frequency (Hz)	Motor Slip (%)	Water Temp (F)	Input Power (KW)	Line Voltage (volts)	Line Current (amps)	Power Factor	Motor Unit Efficiency (%)	Overall Efficiency Delivered (%)
38.6	558.1	506.1	3681	63.1	2.77	78.8	26.1	465.8	58.7	0.552	15.5	14.1
60.2	552.0	500.0	3667	63.2	3.30	78.0	28.8	465.1	61.5	0.581	21.7	19.6
80.5	545.1	493.1	3652	63.2	3.69	79.0	31.4	465.1	64.3	0.606	26.3	23.7
98.5	538.1	486.1	3643	63.3	4.08	79.4	33.8	465.0	67.0	0.625	29.8	26.9
120.2	525.5	473.5	3628	63.3	4.48	79.6	36.2	464.6	69.9	0.642	32.8	29.5
140.7	507.8	455.8	3614	63.3	4.84	80.4	38.3	463.4	72.6	0.657	35.0	31.5
159.9	489.0	437.0	3600	63.4	5.36	81.3	40.2	463.6	75.1	0.666	36.5	32.7
182.1	467.0	415.0	3590	63.4	5.63	82.7	42.2	463.8	77.8	0.676	37.9	33.6
220.3	419.1	367.1	3566	63.4	6.26	84.1	45.0	462.6	81.8	0.687	38.5	33.8

NOTES: Motor Unit Head is head developed by the pump at the top of the motor. Delivered Head is head developed by the pump at the discharge header at the top of the column.

Motor Unit Overall Efficiency is the efficiency of the machine based on head developed at the top of the motor unit (traditional method). Delivered Overall Efficiency is the efficiency of the machine based on head developed at the column discharge header.

Rated Speed (63 Hz VFD Input) Head-Flow Curves
Design Point: 450 Feet Delivered Head at 140 GPM

Comments/Evaluation - Rated Speed Test Per 348A53-C, Section 5.0

The rated speed test was performed in WEMD's P-Loop. The motor unit was suspended in the P-Loop casing by a 2-inch thick circular plate which was bolted to both the motor unit and the casing. The casing was filled to overflowing with water, thereby keeping the motor unit completely submerged.

Because the motor unit was not attached to the column assembly and the attachment method was extremely rigid, it was deemed that accelerometer vibration data was meaningless, and therefore, was not taken. The vibration data obtained during the 24-hour run was considered to show the acceptability of the pump in this regard.

Performance data indicates no instabilities over the flow range and that the pump can easily meet design point performance. There were no unusual noises and pump operation was very smooth.

The pump meets the requirements of Specification 348A53-C, Section 5.0.

TRANSFER PUMP P-LOOP HEAD-FLOW TESTING AT 57 HZ (366 FEET @ 138 GPM OPERATING POINT)
IN ACCORDANCE WITH SPECIFICATION 348A53-C SECTION 6.0

Test Date: 9-16-97

Test Location: WEMD

Delivered Flow (gpm)	Motor Unit Head (ft)	Delivered Head (ft)	Operating Speed (rpm)	Input Frequency (Hz)	Motor Slip (%)	Water Temp (F)	Input Power (KW)	Line Voltage (volts)	Line Current (amps)	Power Factor	Overall Efficiency Motor Unit (%)	Delivered Efficiency (%)
128.7	425.4	373.4	3312	57.3	3.7	69.9	30.6	454.3	66.9	0.581	33.6	29.5
137.1	419.4	367.4	3312	57.3	3.7	68.3	31.2	454.8	67.5	0.587	34.6	30.3
147.9	410.0	358.0	3306	57.3	3.8	69.0	32.1	454.8	68.6	0.595	35.5	31.0

NOTES: Motor Unit Head is head developed by the pump at the top of the motor. Delivered Head is head developed by the pump at the discharge header at the top of the column.

Motor Unit Overall Efficiency is the efficiency of the machine based on head developed at the top of the motor unit (traditional method). Delivered Overall Efficiency is the efficiency of the machine based on head developed at the column discharge header.

Off-Speed Head-Flow Curves at 57 Hz VFD Input
Operating Point: 366 Feet Delivered Head at 138 GPM

TRANSFER PUMP P-LOOP HEAD-FLOW TESTING AT 56 HZ (394 FEET @ 53 GPM OPERATING POINT)
IN ACCORDANCE WITH SPECIFICATION 348A53-C SECTION 5.0

Test Date: 9-16-97

Test Location: WEMD

Delivered Flow (gpm)	Motor Unit Head (ft)	Delivered Head (ft)	Operating Speed (rpm)	Input Frequency (Hz)	Motor Slip (%)	Water Temp (F)	Input Power (KW)	Line Voltage (volts)	Line Current (amps)	Power Factor	Overall Efficiency Motor Unit (%)	Delivered Efficiency (%)
43.5	446.5	394.5	3298	56.2	2.2	73.9	21.0	446.9	57.8	0.470	17.4	15.3
52.8	444.3	392.3	3294	56.2	2.3	72.6	22.0	447.0	58.6	0.484	20.0	17.7
61.9	442.1	390.1	3288	56.2	2.5	73.0	22.9	447.0	59.5	0.497	22.4	19.8

NOTES: Motor Unit Head is head developed by the pump at the top of the motor. Delivered Head is head developed by the pump at the discharge header at the top of the column.

Motor Unit Overall Efficiency is the efficiency of the machine based on head developed at the top of the motor unit (traditional method). Delivered Overall Efficiency is the efficiency of the machine based on head developed at the column discharge header.

Off-Speed Head-Flow Curves at 56 Hz VFD Input
Operating Point: 394 Feet Delivered Head at 53 GPM

TRANSFER PUMP P-LOOP HEAD-FLOW TESTING AT 53 HZ (305 FEET @ 132 GPM OPERATING POINT)
IN ACCORDANCE WITH SPECIFICATION 348A53-C SECTION 6.0

Test Date: 9-16-97

Test Location: WEMD

Delivered Flow (gpm)	Motor Unit Head (ft)	Delivered Head (ft)	Operating Speed (rpm)	Input Frequency (Hz)	Motor Slip (%)	Water Temp (F)	Input Power (KW)	Line Voltage (volts)	Line Current (amps)	Power Factor	Overall Efficiency Motor Unit (%)	Overall Efficiency Delivered (%)
122.4	364.3	312.3	3078	53.0	3.2	71.6	25.0	422.2	63.0	0.543	33.5	28.7
131.7	357.0	305.0	3072	53.0	3.4	70.4	25.7	422.3	63.7	0.550	34.4	29.4
142.3	349.6	297.6	3066	53.0	3.6	70.9	26.4	422.2	64.5	0.559	35.4	30.1

NOTES: Motor Unit Head is head developed by the pump at the top of the motor. Delivered Head is head developed by the pump at the discharge header at the top of the column.

Motor Unit Overall Efficiency is the efficiency of the machine based on head developed at the top of the motor unit (traditional method). Delivered Overall Efficiency is the efficiency of the machine based on head developed at the column discharge header.

Off-Speed Head-Flow Curves at 53 Hz VFD Input
Operating Point: 305 Feet Delivered Head at 132 GPM

TRANSFER PUMP P-LOOP HEAD-FLOW TESTING AT 28 HZ (19 FEET @ 132 GPM OPERATING POINT)
IN ACCORDANCE WITH SPECIFICATION 348A53-C SECTION 6.0

Test Date: 9-16-97

Test Location: WEMD

Delivered Flow (gpm)	Motor Unit Head (ft)	Delivered Head (ft)	Operating Speed (rpm)	Input Frequency (Hz)	Motor Slip (%)	Water Temp (F)	Input Power (KW)	Line Voltage (volts)	Line Current (amps)	Power Factor	Overall Efficiency Motor Unit (%)	Overall Efficiency Delivered (%)
122.7	75.7	23.7	1686	28.7	2.1	69.7	6.2	228.9	51.1	0.305	28.1	8.8
132.2	69.6	17.6	1686	28.7	2.1	70.5	6.3	229.0	51.2	0.310	27.4	6.9
140.8	63.2	11.2	1686	28.7	2.1	70.0	6.4	229.0	51.3	0.314	26.1	4.6

NOTES: Motor Unit Head is head developed by the pump at the top of the motor. Delivered Head is head developed by the pump at the discharge header at the top of the column.

Motor Unit Overall Efficiency is the efficiency of the machine based on head developed at the top of the motor unit (traditional method). Delivered Overall Efficiency is the efficiency of the machine based on head developed at the column discharge header.

Off-Speed Head-Flow Curves at 28 Hz VFD Input
Operating Point: 19 Feet Delivered Head at 132 GPM

TRANSFER PUMP P-LOOP HEAD-FLOW TESTING AT 24 HZ (19 FEET @ 53 GPM OPERATING POINT)
IN ACCORDANCE WITH SPECIFICATION 348A53-C SECTION 6.0

Test Date: 9-16-97

Test Location: WEMD

Delivered Flow (gpm)	Motor Unit Head (ft)	Delivered Head (ft)	Operating Speed (rpm)	Input Frequency (Hz)	Motor Slip (%)	Water Temp (F)	Input Power (KW)	Line Voltage (volts)	Line Current (amps)	Power Factor	Overall Efficiency Motor Unit (%)	Overall Efficiency Delivered (%)
42.7	76.3	24.3	1422	24.0	1.3	70.9	3.3	191.3	49.3	0.203	18.5	5.9
53.7	74.0	22.0	1416	24.0	1.7	71.6	3.5	191.3	49.4	0.214	21.3	6.3
62.7	70.8	18.8	1416	24.0	1.7	71.2	3.6	191.3	49.5	0.222	23.2	6.2

NOTES: Motor Unit Head is head developed by the pump at the top of the motor. Delivered Head is head developed by the pump at the discharge header at the top of the column.

Motor Unit Overall Efficiency is the efficiency of the machine based on head developed at the top of the motor unit (traditional method). Delivered Overall Efficiency is the efficiency of the machine based on head developed at the column discharge header.

Off-Speed Head-Flow Curves at 24 Hz VFD Input
Operating Point: 19 Feet Delivered Head at 53 GPM

**Delivered Head-Flow Performance Curves
At Design and Off-Speed VFD Input Frequencies**

Comments/Evaluation - Offspeed Demonstration Testing Per 348A53-C, Section 6.0

The off-speed demonstration testing was performed in WEMD's P-Loop using the set-up described for the rated speed test.

Performance at the five off-speed points indicates no instabilities over the flow and head ranges tested. No instabilities were found going between the five points.

There were no unusual noises and pump operation was very smooth over the head and flow ranges tested. The pump easily met the performance requirements of the five points.

The pump meets the requirements of Specification 348A53-C, Section 6.0.

Hot Shutdown Resistance After Design Point Testing to Determine Average Core/ Winding Temperature Rise

Calculations of Core Temperature

$$R1 = 0.1398 \text{ ohms @ } 19.4 \text{ deg C}$$

$$R2 = 0.1587 \text{ ohms @ } t = 0$$

$$R2 / (234.5 + T2) = R1 / (234.5 + T1)$$

$$T2 = (R2 / R1) (234.5 + T1) - 234.5$$

$$T2 = (.1587 / .1398) (234.5 + 19.4) - 234.5$$

$$T2 = 53.7 \text{ deg C (128.7 deg F)}$$

$$T_{\text{water}} = 87.6 \text{ deg F}$$

$$T_{\text{rise}} = 128.7 - 87.6$$

$$T_{\text{rise}} = 41.1 \text{ deg F (22.8 deg C)}$$

The results indicate that the core and windings experienced an average temperature rise of 22.8 C above the ambient water temperature during steady state testing at the design point head and flow. This is acceptable.

Pump S/N 001

Motor Unit S/N 201

Test Date: 9/23/97

Electrical Conduit Seal Pressure Drop: 8 psi (10 psi allowed)

Test Date: 9/23/97

Ambient Temperature 22°C (72°F)

Terminal-to-Terminal Resistance	T1-T2	<u>0.1519 @ 25°C</u>
	T2-T3	<u>0.1519 @ 25°C</u>
	T3-T1	<u>0.1518 @ 25°C</u>

Terminal-to-Ground Resistance 24,000 Megohms

Lead Unit Test Summary Comments/Evaluation

All head-flow test data is acceptable. The pump can easily meet the design point requirements (450 feet of delivered head at 450 gpm flow). Power draw is acceptable at design point, requiring only 72.6 amps in water. Estimated current draw in 1.50 specific gravity process fluid is less than 120 amps. The allowable limit is 156 amps; this is the VFD output limit.

All off-speed testing was acceptable. There were no flow instabilities at any tested point.

The electrical conduit seal pressure drop (between the motor unit and column assembly) was less than 10 psi and is acceptable.

The final assembly terminal-to-terminal and terminal-to-ground resistance values are acceptable.

All testing was acceptable and the pump is therefore acceptable for delivery to the Customer.

APPENDIX C

WESTINGHOUSE GOVERNMENT SERVICES COMPANY LLC
ELECTRO-MECHANICAL DIVISION
1000 CHESWICK AVENUE
CHESWICK, PENNSYLVANIA

MODEL 2A-513-A1 TRANSFER PUMP
HIGH VISCOSITY OPERABILITY EVALUATION

S.O. V008

JUNE 29, 1999

E.M. 6910

Original Issue

**Westinghouse Government Services
Company LLC
Electro-Mechanical Division**

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ELECTRO-MECHANICAL DIVISION
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Revision Number	Date	Revision	Written or Revised by:	Approved by:
Original	6/29/99	Original	R. F. Guardiani	F. J. Orehowsky

ABSTRACT

This design report presents the performance curves and analyses required to document the viscosity study of the Westinghouse Government Services Company LLC Electro Mechanical Division Model 2A-513-A1 New Generation Transfer Pump (NGTP). The viscosity study evaluates the maximum viscosity in which the Model 2A-513-A1 NGTP is capable of operating. This document is prepared under Lockheed Martin Hanford Corporation Purchase Order 00002947.

TABLE OF CONTENTS (continued)

	<u>PAGE</u>
4.6 OTHER MOTOR POWER LOSSES	4-7
4.7 IMPELLER AXIAL THRUST	4-9
4.8 MOTOR AXIAL THRUST	4-10
4.9 IMPELLER RADIAL THRUST	4-10
4.10 MOTOR INTERNAL COOLANT CIRCULATION RATE	4-11
4.11 MINIMUM NPSH.....	4-12
 5.0 THRUST BEARING ANALYSIS	 5-1
 6.0 MOTOR ANALYSIS	 6-1
 7.0 THERMAL ANALYSIS.....	 7-1
 8.0 DYNAMIC ANALYSIS	 8-1
8.1 MODEL DESCRIPTION.....	8-1
8.2 ROTOR MODELING DESCRIPTION	8-1
8.3 ROTOR, STATOR, AND PIPE COLUMN MODELING DESCRIPTION	8-3
8.4 UNDAMPED NATURAL FREQUENCY ANALYSIS DESCRIPTION	8-6
8.5 DAMPED ROTOR DYNAMIC ANALYSIS DESCRIPTION AND RESULTS.....	8-7
 9.0 REFERENCES.....	 9-1
 10.0 ACKNOWLEDGEMENTS	 10-1
 11.0 DISTRIBUTION LIST	 11-1

TABLE OF CONTENTS

	<u>PAGE</u>
RECORD OF REVISIONS	i
ABSTRACT	ii
TABLE OF CONTENTS	iii
LIST OF TABLES	v
LIST OF FIGURES	vi
1.0 INTRODUCTION.....	1-1
2.0 CONCLUSIONS AND RECOMMENDATIONS	2-1
2.1 CONCLUSIONS	2-1
2.2 RECOMMENDATIONS.....	2-3
3.0 DESIGN AND ANALYSIS SUMMARY	3-1
3.1 DESIGN SUMMARY	3-1
3.2 ANALYSIS SUMMARY	3-4
4.0 HYDRAULIC ANALYSIS	4-1
4.1 REFERENCE PUMP PERFORMANCE	4-2
4.2 VISCOUS FLUID EXTRAPOLATION	4-4
4.3 DELIVERED HEAD	4-5
4.4 SPEED CONVERGENCE.....	4-6
4.5 PUMP HYDRAULIC POWER	4-7

LIST OF TABLES

	<u>PAGE</u>
TABLE 3-1 ORIGINAL DESIGN CHARACTERISTICS	3-2
TABLE 3-2 NEW DESIGN CHARACTERISTICS	3-3
TABLE 3-3 TRANSFER PUMP VISCOSITY PERFORMANCE AT 60 GPM FLOW AND 100 FEET HEAD	3-5
TABLE 4-1 REFERENCE MODEL 2A-513-A1 NGTP PERFORMANCE	4-3
TABLE 4-2 ESTIMATION OF FLOW, HEAD, AND EFFICIENCY FOR 250 Cp VISCOSITY LIQUID	4-4
TABLE 4-3 DELIVERED FLOW, DELIVERED HEAD, AND POWER ADJUSTED FOR SPEED, DENSITY, AND VISCOSITY	4-6
TABLE 4-4 COMBINED MOTOR AND PUMP HYDRAULIC POWER LOSSES	4-8
TABLE 4-5 AXIAL AND RADIAL THRUSTS AT NEW DESIGN POINT	4-11
TABLE 4-6 INTERNAL COOLANT FLOW RATE AT NEW DESIGN POINT	4-11
TABLE 5-1 THRUST BEARING PERFORMANCE AT 60 GPM FLOW AND 100 FEET OF HEAD	5-1
TABLE 6-1 TRANSFER PUMP VISCOSITY PERFORMANCE AT 60 GPM FLOW AND 100 FEET HEAD	6-2
TABLE 7-1 THERMAL ANALYSIS RESULTS AT 60 GPM FLOW AND 100 FEET HEAD	7-3
TABLE 8-1 TRANSFER PUMP ROTOR STABILITY RESULTS – SYSTEM DAMPED FREQUENCIES COMPARISON OF PUMP IN WATER TO PUMP IN PROCESS FLUID	8-8
TABLE 8-2 VIBRATION RESPONSE RESULTS RADIAL DISPLACEMENTS OF ROTOR RELATIVE TO STATOR & BOTTOM OF PUMP RELATIVE TO GROUND COMPARISON OF PUMP IN WATER TO PUMP IN PROCESS FLUID	8-9

LIST OF FIGURES

	<u>PAGE</u>
FIGURE 4-1 FLUID POWER LOSS VERSUS VISCOSITY AT 60 GPM AND 100 FEET OF HEAD	4-9
FIGURE 4-2 MOTOR INTERNAL CIRCULATION VERSUS VISCOSITY AT 60 GPM AND 100 FEET OF HEAD	4-12
FIGURE 6-1 MOTOR MECHANICAL PERFORMANCE VERSUS VISCOSITY	6-3
FIGURE 6-2 MOTOR PERFORMANCE CURVES (32.659 Hz)	6-4
FIGURE 6-3 MOTOR PERFORMANCE CURVES (34.091 Hz)	6-5
FIGURE 6-4 MOTOR PERFORMANCE CURVES (37.5 Hz)	6-6
FIGURE 6-5 MOTOR PERFORMANCE CURVES (39.682 Hz)	6-7
FIGURE 7-1 MOTOR INSULATION LIFE VERSUS VISCOSITY	7-7
FIGURE 8-1 ROTOR MODEL - DETAILED GEOMETRIC INPUT USED ROTOR MODEL - HALF OF CROSS-SECTION OF EQUIVALENT STIFFNESS ELEMENTS CALCULATED	8-2
FIGURE 8-2 ROTOR & STATOR MODEL - DETAILED GEOMETRIC INPUT USED ROTOR & STATOR MODEL - HALF OF CROSS-SECTION OF EQUIVALENT STIFFNESS ELEMENTS CALCULATED	8-5

1.0 INTRODUCTION

The EMD Model 2A-513-A1 New Generation Transfer Pump (NGTP) was designed in accordance with Westinghouse Hanford Company specification WHC-S-0238, Revision 1, under Purchase Order MRB-SXP-342098. The pump was designed for 10,000 hours of service life over 10 years at the following design point:

- 140 gpm flow rate
- 450 feet of delivered head
- 1.50 specific gravity
- 30 centipoise viscosity

Lockheed Martin Hanford Corporation (LMHC) is considering the Model 2A-513-A1 NGTP for a special mission (the SY-101 tank) that falls outside the original design parameters. LMHC has requested EMD to evaluate the Model 2A-513-A1 NGTP at the following design conditions:

- 60 gpm flow rate
- 100 feet of delivered head, maximum
- 1.00 to 1.70 specific gravity
- 1 to 1000 centipoise viscosity

This report was prepared to document the Model 2A-513-A1 NGTP capabilities at the new design conditions. The following analyses/designs were considered in the evaluation:

- Radial and thrust bearing design
- Hydraulic design
- Motor electrical design
- VFD design
- Motor temperatures
- Motor internal flow
- Mechanical loads
- Fluid frictional losses
- Pump, motor and column vibrations

2.0 CONCLUSIONS AND RECOMMENDATIONS

2.1 Conclusions

The EMD Model 2A-513-A1 NGTP viscosity analyses indicate that for a given head, flow and specific gravity the following occur as fluid viscosity is increased from 1 to 1000 centipoise:

- Hydraulic efficiency decreases with higher viscosities, thereby requiring more shaft input power and higher impeller rotational speeds.
- Bearing loads increase and bearing film thicknesses increase. Bearing power dissipation increases due to the bearing load increases, the viscosity increase and the operating speed increase.
- Vibration of the rotating assembly, motor, and column assembly decrease as viscosity increases due to increased damping.
- Rotating component "windage" losses increase due to both the viscosity increase and the operating speed increase.
- Fluid flow through the motor and bearings decrease due to the increased fluid viscosity.
- The Reynolds Number for the fluid flowing down the can annulus between the rotor and stator cans decreases due to both the increase in fluid viscosity and the decrease in motor flow. The increase in motor operating speed tends to increase the Reynolds Number but this effect is overwhelmed by the viscosity increase.
- The decrease in Reynolds Number into the laminar flow region (below 2000) decreases the heat transfer film coefficient and decreases the heat removal capability of the fluid flowing down the can annulus region.

- The increase in motor output power due to increased hydraulic losses, bearing losses, and "windage" losses and the increase in required operating speed increases the motor slip, thereby increasing the input frequency and current supplied to the pump by the variable frequency drive (VFD).
- The increase in motor current coupled with the decrease in heat removal capability in the can annulus region increases the maximum winding surface temperature (MWST) of the motor electrical windings.
- The increase in losses due to increased hydraulic losses, bearing losses, and "windage" losses increase the required output torque of the motor. The increase in required operating speed has a minor effect in partially offsetting this torque increase.
- The motor has a fixed maximum output torque capability. When the required torque exceeds the maximum torque capability of the motor, the motor will fail to come up to operating speed and will stall.
- The increase in MWST decreases the operating life of the motor windings.

The viscosity study indicates there are two items limiting operation of the Model 2A-513-A1 NGTP in a high viscosity fluid. One is the torque output of the motor and the second is the operating life of motor electrical winding. The required operating torque approaches the maximum motor torque (breakdown torque) at approximately 440 centipoise. Above this viscosity the motor is not capable of coming up to operating speed and will stall. The operating life of the motor is limited to approximately 8900 hours at 250 centipoise and 10 hours at 375 centipoise. Therefore, of the two limiting items, motor winding life is the controlling parameter.

Based on the viscosity study, EMD concludes that at the new design point (60 gpm flow with 100 feet of delivered head at a specific gravity of 1.70) the Model 2A-513-A1 NGTP operation is limited to a maximum viscosity of 250 centipoise to ensure at least 5000 hours of operating life. EMD also concludes that at the special mission design point the 2A-513-A1 NGTP is capable of operation in fluids with viscosities up to 375 centipoise, albeit with a substantially reduced life.

It is noted that for operating conditions that require less hydraulic power and lower operating speeds than the new design point identified above, the pump can operate in fluids with viscosities higher than those identified. However, the limiting viscosity must be calculated for each operating condition.

2.2 Recommendations

EMD recommends that on a continuous basis the Model 2A-513-A1 NGTP should be operated in fluids with viscosities that do not exceed 250 centipoise at the new design point. EMD also recommends that the Model 2A-513-A1 NGTP should be operated in fluids with viscosities between 250 and 375 centipoise for no longer than 4 hours total to ensure at least 5000 hours of design life. For non-Newtonian, shear-thinning fluids, EMD recommends the Model 2A-513-A1 NGTP should be started in fluids with an initial viscosity less than 440 centipoise.

3.0 DESIGN AND ANALYSIS SUMMARY

3.1 Design Summary

The EMD Model 2A-513-A1 NGTP incorporates the following key features:

- A 60 horsepower, submerged canned motor with integral hydraulics, passively cooled by hazardous waste tank process fluid
- A two stage centrifugal pump with one crossover diffuser and one axial diffuser
- A single piece, welded column assembly with hermetically sealed internals to preclude the entry of process fluid
- No dynamic seals in the motor
- No dedicated external lubrication system for the pump bearings
- no dedicated external cooling system for the submerged canned motor
- No shafting, coupling, or dynamic penetrations of the pump mounting flange
- Rugged, reliable and maintenance free below the mounting flange over the 10 year, 10000 hour design life of the pump

A detailed design description of the Model 2A-513-A1 NGTP and the analyses conducted to verify the pump's performance are documented in EM 6673, Revisions 1 through 4.

The original design characteristics of the Model 2A-513-A1 NGTP are summarized in Table 3-1. The new design characteristics of the pump and the fluid properties used in the viscosity study are listed in Table 3-2.

TABLE 3-1
Original Design Characteristics

Design Head	450 feet at the discharge flange
Design Flow	140 gpm
Operating Speed	3614 rpm
Fluid Specific Gravity	1.00 to 1.50
Fluid Viscosity	1.0 to 30.0 centipoise
Fluid Temperature	50 °F to 200 °F
Required NPSH	10.6 feet minimum
Power Required	460 Volt, 3-Phase, 60 Hz to variable frequency drive
Motor Cooling	Process Fluid
Bearing Lubrication	Process Fluid
Dilution Water Supply	105 gpm minimum at 100 psig
Purge Water Supply	10 gpm minimum at 90 psig
Operating Life	10000 hours minimum over 10 years
Weight, dry	3790 lbs
Weight, wet	4096 lbs maximum

TABLE 3-2
New Design Characteristics

Design Head	100 feet at the discharge flange
Design Flow	60 gpm
Operating Speed	To be determined in analysis
Fluid Specific Gravity	1.00 to 1.70
Fluid Viscosity	30.0 to 1000 centipoise
Fluid Temperature	120 °F
Fluid Specific Heat	0.750 Btu / lbm - °F
Fluid Conductivity	0.091 Btu / hr - ft - °F
Required NPSH	To be determined in analysis
Power Required	460 Volt, 3-Phase, 60 Hz to variable frequency drive
Motor Cooling	Process Fluid
Bearing Lubrication	Process Fluid
Dilution Water Supply	105 gpm minimum at 100 psig
Purge Water Supply	10 gpm minimum at 90 psig
Operating Life	5000 hours minimum over 10 years
Weight, dry	3790 lbs
Weight, wet	4137 lbs maximum

3.2 Analysis Summary

The analyses are contained in Sections 4.0 through 8.0. Section 4.1 describes the methodology for extrapolating the Model 2A-513-A1 NGTP water test data to the new design point (see Table 3-2) at high viscosities (up to 1000 centipoise) and high specific gravities (up to 1.70). Sections 4.2 through 4.5 provide calculations for determining operating pump speeds and hydraulic power losses at the new design point with 30 to 1000 centipoise viscosity fluids. In Section 4.6 other fluid power losses, including the radial bearings, are delineated at viscosities between 30 and 1000 centipoise. Sections 4.7 and 4.8 calculate new design point axial thrust in the hydraulics and motor, while Section 4.9 evaluates hydraulic radial loads at the new design point. In Section 4.10 the motor internal circulation flow is evaluated at fluid viscosities between 30 and 1000 centipoise. Section 4.11 provides an NPSH evaluation for the new design point.

The thrust bearing analysis at the new design point for fluids with viscosities between 30 and 1000 centipoise is contained in Section 5.0. The motor electrical analysis is described in Section 6.0; the analysis reveals the motor is not capable of starting the pump with fluid viscosities above 440 centipoise. Section 7.0 contains the thermal analysis of the motor which indicates the insulation life degrades significantly when pumping fluids with viscosities greater than 250 centipoise. Section 8.0 describes the dynamic analysis which reveals that pump vibration levels are acceptable and that the rotating system is stable for design point operation at 250 centipoise.

A summary of the results of all analyses is delineated in Table 3-3.

TABLE 3-3
Transfer Pump Viscosity Performance at 60 gpm Flow and 100 Feet Head

Fluid Properties						
Specific Gravity	1.0	1.70	1.70	1.70	1.70	1.70
Viscosity, Centipoise	1.0	30	250	375	600	1000
Temperature, °F	120.0	120.0	120.0	120.0	120.0	120.0
Specific Heat, Btu/lbm-°F	1.000	0.750	0.750	0.750	0.750	0.750
Conductivity, Btu/hr-ft-°F	0.364	0.091	0.091	0.091	0.091	0.091
Operating Speed, rpm	1918	1942	2020	2060	2151	2262
Pumped Fluid Flow Rates, gpm						
Hydraulics Discharge Flow	90.9	88.3	74.4	70.3	66.3	63.5
Pump Discharge Flow	60.0	60.0	60.0	60.0	60.0	60.0
Motor Internal Flow	30.9	28.3	14.4	10.3	6.3	3.5
Reynolds No., Can Annulus	37504	2130	257	173	113	71
Mechanical Losses, bhp						
Thrust Bearing	0.2	1.3	5.4	7.3	10.7	16.5
Thrust Runner	0.2	0.7	2.0	2.5	3.5	5.5
Chopper	0.4	0.7	0.9	1.0	1.2	1.5
Lower Radial Bearing	0.0	0.4	3.9	6.2	10.7	19.8
Rotor-Stator Annulus	0.2	1.0	3.2	4.2	5.8	10.0
Upper Radial Bearing	0.0	0.2	2.2	3.4	6.0	11.0
Miscellaneous	0.0	0.1	0.2	0.3	0.4	0.6
Hydraulics	6.8	12.0	16.1	18.3	22.9	29.1
Total Mechanical Losses	7.8	16.4	33.9	43.2	61.2	94.0
Motor Performance						
Maximum Torque, ft-lbs	149.5	146.1	132.7	126.1	∞120	∞115
Terminal Voltage, volts	240	250	275	291	stall	stall
Supply Frequency, Hz	32.7	34.1	37.5	39.7	stall	stall
Line Current, amperes	52.3	63.2	97.4	118.6	stall	stall
MWST, °C	75	90	253	351	stall	stall
Insulation Life, hours	2.03E+09	7.18E+08	8894	10	stall	stall
Input Power, KW	8.2	15.6	32.6	42.6	stall	stall
Total Electrical Losses, KW	2.4	3.4	7.3	10.4	stall	stall
Efficiency, %	61.8	57.4	36.8	32.0	stall	stall
Fluid Temperature Rise, °F						
Impeller Discharge Fluid	0.2	0.3	0.8	1.1	stall	stall
Motor Circulation Fluid	1.4	2.3	8.9	15.8	stall	stall
Axial Thrust, lbs						
First Stage Impeller	56	97	105	109	119	132
Second Stage Impeller	60	105	114	119	129	143
Motor	278	469	595	621	665	702
Rotating Assembly Weight	140	125	125	125	125	125
Total Axial Thrust	534	796	939	974	1038	1102
Radial Loads, lbs						
First Stage Impeller	4.6	7.9	7.9	7.9	7.9	7.9
Second Stage Impeller	2.6	4.4	4.4	4.4	4.4	4.4

4.0 HYDRAULIC ANALYSIS

The hydraulic performance of the Model 2A-513-A1 NGTP operating at the new design point (60 gpm flow at 100 feet delivered head) in high viscosity fluids with viscosities up to 1000 centipoise was calculated to determine the maximum fluid viscosity that can be pumped, consistent with motor power limits. A parametric analysis was performed to determine hydraulic performance for various values of viscosity. The pump hydraulic performance was calculated at viscosities higher than water using the viscosity coefficients from the Hydraulic Institute Standards (Reference 9-1). The column assembly friction losses at the higher viscosities were modeled using the Moody friction factor. The prediction process consists of the following six steps:

1. Define the pump performance in low viscosity water.
2. Extrapolate the pump performance (flow rate, head, efficiency) to that in a higher viscosity and higher specific gravity fluid and adjust for a guess of the lower operating speed. (Subsequent iterations will produce a converged value of the operating speed guess).
3. Calculate the column head loss and combine with the pump head to get the delivered head between the pump suction and the column outlet.
4. Check for the required delivered head of 100 ft at 60 gpm flow. Revise the speed guess in Step 2, and redo Steps 3 and 4 until the delivered head is 100 ft at 60 gpm.
5. Determine the maximum pump power requirement.
6. Calculate the other motor parasitic power losses that change with viscosity and add these to the pumping power from Step 5 to get the total hydraulic power loss (motor required developed power). The correction factors from the Hydraulic Institute Standards account for only the impeller performance, whereas the rotating friction losses in the canned motor must be accounted for separately.

Sections 4.1 through 4.6 discuss these six steps in detail, using a 1.70 specific gravity (sg), and 250 centipoise (cP) liquid as an example. Similar calculations were performed for four other

viscosities (30, 375, 600, and 1000 cP) at 1.70 sg, and for cold water (1.0 sg and 1.0 cP). These six cases parametrically cover a range of viscosities that provide the input used to determine the highest viscosity fluid that can be pumped within the motor power limit. The calculated pumping power, and viscous power losses for all six cases are presented in Section 4.6.

4.1 Reference Pump Performance

The reference performance of the Model 2A-513-A1 NGTP was defined by the measured values of delivered flow rate, motor unit head, rpm, and overall motor unit efficiency presented in the EMD water test report (Reference 9-2) and listed in Table 4-1. The delivered flow rate is the flow out the discharge pipe at the top of the column, and does not include the cooling/lubricating fluid that flows through the motor. The motor unit head is defined as the head developed from the pump suction to the top of the motor (inlet to the column assembly). The additional head loss over the length of the column is modeled separately as described in Section 4.3. The measured overall motor unit efficiency, η_{OA} , and an estimated motor efficiency, η_M , of 72% were used to calculate the hydraulic efficiency, $\eta_H = \eta_{OA} / \eta_M$. The 72% motor efficiency is an estimated value based on water test results. The impeller power was calculated from the equation:

$$P = \frac{\rho Q H}{\eta_H} \quad (\text{Eq. 4-1})$$

where: ρ = density

Q = volumetric flow rate

H = head

η_H = hydraulic efficiency

TABLE 4-1
Reference Model 2A-513-A1 NGTP Performance

Measured Data				Assumed	Calculated		Adjusted to Average rpm		
Flow (gpm)	Motor Head (ft)	Speed (rpm)	η_{OA}	η_M	η_H	Power (bhp)	Flow (gpm)	Motor Head (ft)	Power (bhp)
38.6	558.1	3681	15.5%	72.0%	21.5%	43.0	38.0	541.8	41.1
60.2	552.0	3667	21.7%	72.0%	30.1%	47.3	59.5	540.0	45.8
80.5	545.1	3652	26.3%	72.0%	36.5%	51.6	79.9	537.6	50.5
99.5	538.1	3643	29.8%	72.0%	41.4%	55.5	99.1	533.3	54.8
120.2	525.5	3628	32.8%	72.0%	45.6%	59.5	120.2	525.1	59.5
140.7	507.8	3614	35.0%	72.0%	48.6%	63.1	141.2	511.4	63.8
159.9	489.0	3600	36.5%	72.0%	50.7%	66.2	161.1	496.3	67.7
182.1	467.0	3590	37.9%	72.0%	52.6%	69.4	184.0	476.6	71.5
220.3	419.1	3566	38.5%	72.0%	53.5%	74.1	224.1	433.5	78.0
		3627					300*	250*	

*Estimated from air tests (Reference 9-3, Section 5.4) to extend the data range.

Each data set (flow, head, and power) corresponds to a slightly different speed. Consequently, the data were adjusted based on the well-known pump affinity relationships to define performance at a single speed. The selected single speed is the average value of the rpm measurements, 3627 rpm. The affinity relationships for flow rate, head, and power are the following, respectively:

$$Q = Q_o \frac{f}{f_o} \quad H = H_o \left(\frac{f}{f_o} \right)^2 \quad P = P_o \left(\frac{f}{f_o} \right)^3 \quad (\text{Eqs. 4-2a/b/c})$$

where f is the pump rotation speed, and the o subscript implies reference (test) values, such as at a test point. The values of flow rate, head, and power adjusted for rotation speed are presented as the last three columns of Table 4-1.

4.2 Viscous Fluid Extrapolation

A procedure for extrapolating pump performance in water to pump performance in higher viscosity liquids is presented in Reference 9-1, page 100. The procedure uses coefficients to reduce the flow, head, and efficiency to account for viscous operation. The flow and efficiency coefficients are single-valued for viscosity, but the head coefficient varies with flow rate. Consequently, the four head coefficient values given by Figure 63 of Reference 9-1 was curve fitted, and the curve was then used to determine the value of the flow coefficient corresponding to the flow rates of the test data. Table 4-2 presents the results of the viscous corrections for liquid with a viscosity of 250 cP and an sg of 1.70 (147.1 centistokes).

TABLE 4-2
Estimation of Flow, Head, and Efficiency for 250 cP Viscosity Liquid

Water Flow, gpm	79.9	120.2	141.2	161.1	224.1	300.0
Water Head, ft	537.6	525.1	511.4	496.3	433.5	250.0
Water Efficiency	36.5%	45.6%	48.6%	50.7%	53.5%	53.0%
Flow Coeff., C_Q	0.93	0.93	0.93	0.93	0.93	0.93
Head Coeff., C_H	0.951	0.933	0.915	0.893	0.790	0.599
Efficiency Coeff., C_E	0.61	0.61	0.61	0.61	0.61	0.61
Viscous Flow, gpm	74.3	111.7	131.3	149.8	208.4	279.0
Viscous Head, ft	511.5	490.0	468.1	443.4	342.6	149.7
Viscous Efficiency	22.3%	27.8%	29.7%	30.9%	32.6%	32.3%
Specific Gravity	1.70	1.70	1.70	1.70	1.70	1.70
Viscous Power, bhp	73.3	84.6	89.0	92.2	94.0	55.5

The first three rows are water values calculated at 3627 rpm from Table 4-1 (note the transposition of rows and columns). Rows 4, 5, and 6 of Table 4-1 are the correction factors, and rows 7, 8, and 9 of Table 4-1 are the extrapolated values of flow, head, and efficiency. The viscous flow, head and efficiency are simply the products of the water values and the coefficients. A fluid specific gravity of 1.70 is used to calculate the viscous power from Equation 4-1.

4.3 Delivered Head

The delivered head calculation begins with a guess of the pump speed that is expected to result in a delivered head of 100 ft at 60 gpm. The speed guess for this iteration (250 cP) is 2020 rpm (this is the correct value as verified in Section 4.4). The viscous flow rates from Table 4-2 are adjusted by the affinity relation, Equation 4-2a, to get the flow rates in the first column of Table 4-3. In this case the speed ratio is:

$$f / f_o = 2020 / 3627 = 0.557 \quad (\text{Eq. 4-3})$$

The Reynolds number is calculated from the definition $Re = Vd/v$, where the velocity, V , and the diameter, $d = 3.068$ in, are those of the column, and $v = 147.1$ centistokes. The values of the Reynolds numbers indicate laminar flow at this high viscosity. The turbulent flow friction factor is the Moody friction factor for pipes, and the laminar flow friction factor is:

$$f = 64 / Re \quad (\text{Eq. 4-4})$$

The delivered head presented in Table 4-3 is the viscous head (for the motor) adjusted for speed according to Equation 4-2b, minus the additional height of the column (48.375 ft), minus the form ($K = 2.6$) and friction losses in the column and the main flow path around the stator. The delivered head is calculated:

$$H = H_M (0.557)^2 - 48.375 - [2.6 + f \times (40 / 0.5 + 43.61 \times 12 / 3.068)] V^2 / 2g \quad (\text{Eq. 4-5})$$

The friction factor in the approximately 40 inch long stator flow path is nearly the same as in the 43.61 ft column, so the column friction factor is applied to both flow paths. At 60 gpm, the form and friction losses are only 6.1 ft, so the friction and form losses are not significant. Consequently, the approximated friction loss does not introduce a significant error.

TABLE 4-3
Delivered Flow, Delivered Head, and Power
Adjusted for Speed, Density, and Viscosity

Flow gpm	Reynolds Number	Friction Factor	Head ft	Hydraulic Efficiency	Power bhp
41.4	290	0.2205	107.4	22.3%	12.7
62.2	436	0.1467	99.2	27.8%	14.6
73.1	513	0.1248	91.5	29.7%	15.4
83.4	585	0.1094	83.0	30.9%	15.9
116.1	814	0.0787	49.1	32.6%	16.2
155.4	1089	0.0588	-14.2	32.3%	9.6

The hydraulic efficiency is the viscous efficiency value from Table 4-2. The motor hydraulic power is calculated according to Equation 4-1 using the flow, head, and hydraulic efficiency values in Table 4-3, and a specific gravity of 1.70.

4.4 Speed Convergence

In this step, the head-flow curve relationship defined in Table 4-3 is curve fitted with a second-order polynomial and evaluated at 60 gpm. The polynomial is:

$$H = -0.006993 Q^2 + 0.3121 Q + 106.4 \quad (\text{Eq. 4-6})$$

This has a value of 99.95 ft at $Q = 60$ gpm. Consequently, the speed guess of 2020 rpm in Section 4.3 produced the goal value of head. If the head value was not sufficiently close to 100 ft, a different value of speed would be guessed, the values in Table 4-3 adjusted as described in Section 4.3, and a new polynomial fitted and evaluated at 60 gpm.

4.5 Pump Hydraulic Power

The maximum power from Table 4-2 is used rather than the value corresponding to 60 gpm since the actual liquid viscosity and flow rate are uncertain. (The maximum power is no more than 2 hp greater than the power at 60 gpm for all viscosities examined). The maximum value of pump hydraulic power is 16.2 bhp from Table 4-3. This power will produce a flow rate of 60 gpm at a head of 100 ft and includes power losses and inefficiencies in the impeller (including rotating losses in the hubs and shrouds) and main flow path through the motor and column. Other losses due to the circulating cooling flow in the motor and rotation drag due to the fluid viscosity are considered in Section 4.6.

4.6 Other Motor Power Losses

The rotor of the canned motor rotates in the same viscous fluid that is pumped, and therefore does work on the fluid. The rate of work is a power loss that can be calculated from the product of the rotation speed and drag. Drag correlations for rotating disks and cylinders are available in the technical literature (References 9-4 to 9-7) for laminar and turbulent flows.

The rotating assembly above the hydraulics (from the thrust bearing to the top of the rotor) was modeled as a system of rotating disks and cylinders. The frictional drag and power loss for each was calculated. In addition, detailed models of the Kingsbury-type thrust bearing, upper radial bearing, and lower radial bearing were run to calculate the bearing drag and power losses. The radial bearings were evaluated using the proprietary program "TILTPAD" which is verified and under configuration control. The thrust bearing evaluation is presented in Section 5.0, based on loads calculated in Section 4.7.

The power losses for the three bearings are listed in Table 4-4 for a parametric range of liquid viscosities, at a specific gravity of 1.70. The reference case of water at 1.0 cP and 1.00 sg is included for comparison. In addition to the bearings, major rotating losses for the thrust runner, the lower radial bearing chopper, the rotor/stator annulus, and a group of all "other" minor rotating power losses are listed in Table 4-4. Power losses for components other than the hydraulics and bearings were calculated using References 9-4 through 9-7.

TABLE 4-4
Combined Motor and Pump Hydraulic Power Losses

Viscosity, cP	1.0	30	250	375	600	1000
Specific Gravity	1.0	1.70	1.70	1.70	1.70	1.70
Speed, rpm	1918	1942	2020	2060	2151	2262
Motor Losses, hp						
Thrust Bearing	0.2	1.3	5.4	7.3	10.7	16.5
Lower Radial Brg	0.0	0.4	3.9	6.2	10.7	19.8
Upper Radial Brg	0.0	0.2	2.2	3.4	6.0	11.0
Thrust Runner	0.2	0.7	2.0	2.5	3.5	5.5
Chopper	0.4	0.7	0.9	1.0	1.2	1.5
Rotor Stator Ann	0.2	1.0	3.2	4.2	5.8	10.0
Others	0.0	0.1	0.2	0.3	0.4	0.6
Pump Power Loss, hp	6.8	12.0	16.1	18.3	22.9	29.1
Required Total Motor Power Input, hp:	7.8	16.4	33.9	43.2	61.2	94.0

For viscosities of 250 cP and greater, the rotating flow is laminar throughout the motor, so power losses are proportional to the rotational speed squared. The speed squared relationship was used after the initial calculation of the component power losses to account for small (less than 3%) changes in speed. Figure 4-1 is a chart which shows the trends of fluid power losses as viscosity is increased. Note that the Figure 4-1 data points at 1.0 cP are not continuous with the data points at 30 to 1000 cP because the 1.0 cP data points are at 1.0 sg while the others are at 1.70 sg.

Figure 4-1
Fluid Power Loss Versus Viscosity
at 60 gpm and 100 feet of head

4.7 Impeller Axial Thrust

The axial thrust loads for the two impellers were calculated in Reference 9-3, Section 5.3. The values for 1.50 sg fluid at 3860 rpm were 338 lbf for the first stage impeller and 367 lbf for the second stage impeller. Since the axial thrust is proportional to speed squared and specific gravity, the extrapolation formulae are:

$$1^{\text{st}} \text{ Stage Impeller: } T_1 = 338 \times (n / n_0)^2 (\text{sg} / 1.50) \quad (\text{Eq. 4-7})$$

$$2^{\text{nd}} \text{ Stage Impeller: } T_2 = 367 \times (n / n_0)^2 (\text{sg} / 1.50) \quad (\text{Eq. 4-8})$$

The resulting axial thrusts for the different parametric cases considered are listed in Table 4-5.

4.8 Motor Axial Thrust

The motor axial thrust calculation was performed with the "CIRCUIT" code as described in Reference 9-3, Section 5.7.2. The same "CIRCUIT" model was used to calculate thrust for the six parametric cases, changing only the liquid density, liquid viscosity, and rotor speed. The predicted motor thrust loads are listed in Table 4-5. All thrust loads are positive downward; therefore, the use of a single-acting thrust bearing is acceptable. Note that the rotor buoyant weight is included separately.

4.9 Impeller Radial Thrust

The impeller radial loads were estimated in Reference 9-3, Section 5.3 and extrapolated from air-test data in Reference 9-3, Section 5.5.4. The calculation of the radial loads at the six parametric conditions is based on using the Hydraulic Institute Standards calculation in Reference 9-3, Section 5.3 to scale the test loads of 27.8 lbf (1st stage impeller) and 15.4 lbf (2nd stage impeller) documented in Section 5.5.4 of Reference 9-3. The resulting equations are as follows (the subscript o indicates test conditions):

$$1^{\text{st}} \text{ stage impeller: } F_1 = 27.8 \times (H / H_o) \times (sg / sg_o) \times (D / D_o) \quad (\text{Eq. 4-9})$$

$$2^{\text{nd}} \text{ stage impeller: } F_2 = 15.4 \times (H / H_o) \times (sg / sg_o) \times (D / D_o) \quad (\text{Eq. 4-10})$$

where: H = head

sg = specific gravity

D = impeller diameter

The test head is 529.1 ft, the sg is 1.0, and the impeller diameters are $D_o = 9.37$ inches (1st stage impeller) and $D_o = 9.93$ inches (2nd stage impeller). The corresponding transfer pump head is 100 ft and the diameters are 8.25 and 8.75 inches. The resulting radial thrusts are listed in Table 4-5.

TABLE 4-5
Axial and Radial Thrusts at New Design Point

Viscosity, cP	1.00	30	250	375	600	1000
Specific Gravity	1.00	1.70	1.70	1.70	1.70	1.70
Speed, rpm	1918	1942	2020	2060	2151	2262
Impeller 1 Axial, lbf	56	97	105	109	119	132
Impeller 2 Axial, lbf	60	105	114	119	129	143
Motor Axial, lbf	278	469	595	621	665	702
Rotor Wet Weight, lbf	140	125	125	125	125	125
Total Axial Thrust, lbf	534	796	939	974	1038	1102
1 st Impeller Radial, lbf	4.6	7.9	7.9	7.9	7.9	7.9
2 nd Impeller Radial, lbf	2.6	4.4	4.4	4.4	4.4	4.4

4.10 Motor Internal Coolant Circulation Rate

The flow rate of fluid through the motor internals affects motor cooling. Consequently, parametric calculations were performed to determine the internal circulation rate expected with the higher liquid density and viscosity. The same CIRCUIT model documented in Reference 9-3, Section 5.7.2 was used with only density, viscosity, and rotation speed changed for the new calculations. Q12, the rotor/stator flow rate from the "CIRCUIT" model is the flow rate of interest. The values of Q12 corresponding the six parametric conditions are listed in Table 4-6. The flow rates are used in the thermal analysis presented in Section 7.0 to verify acceptable thermal performance. A graphical representation of the decrease in motor internal coolant flow with increase in fluid viscosity is shown in Figure 4-2.

TABLE 4-6
Internal Coolant Flow Rate at New Design Point

Viscosity, cP	1.0	30	250	375	600	1000
Specific Gravity	1.0	1.70	1.70	1.70	1.70	1.70
Speed, rpm	1918	1942	2020	2060	2151	2262
Motor Internal Flow, gpm	30.9	28.3	14.4	10.3	6.3	3.5

Figure 4-2
Motor Internal Circulation Versus Viscosity
at 60 gpm and 100 feet of head

4.11 Minimum NPSH

The minimum net positive suction head (NPSH) required for the transfer pump operating in the 250 cP fluid at 2020 rpm was calculated to be 3.2 ft. Cavitation tests of the first stage impeller were conducted by the Southwest Research Institute and presented in Section 5.8 of Reference 6-3. The tests were performed in cold water at 3860 rpm. Extrapolation to the 250 cP fluid at 2020 rpm was based on the suction specific speed as follows (there was no correction for viscosity):

At 2020 rpm in 250 cP fluid the transfer pump delivers 60 gpm at 100 ft head. The NPSH calculation was performed assuming a constant value for the suction specific speed, defined in Reference 9-1 as:

$$S = \frac{N\sqrt{Q}}{(\text{NPSH})^{3/4}} \quad (\text{Eq. 4-11})$$

where: S = suction specific speed

N = Rotation speed in rpm

Q = Volumetric flow rate in gpm

NPSH = net positive suction head required in feet

The transfer pump operating at its original design condition (3860 rpm, 140 gpm) has a net positive suction head of 10.6 ft (Reference 6-3, Figure 5-20) and a suction specific speed, $S = 7774$. Consequently, the transfer pump operating at 2020 rpm and 60 gpm has a net positive suction head required of 2.54 ft. It was recommended in Reference 9-3 to maintain a 25% NPSH margin. That margin raises the required NPSH to 3.2 ft for the Model 2A-513-A1 NGTP operating in 250 cP fluid at 2020 rpm.

5.0 THRUST BEARING ANALYSIS

The 6-pad, Kingsbury-type thrust bearing used in the Model 2A-513-A1 NGTP was analyzed in the same manner as performed in Section 9.2 of Reference 9-3. Thrust bearing performance was calculated using the proprietary program "SECPAD" which has been verified and is under configuration control. The steady-state characteristics of interest for this investigation are minimum film thicknesses and friction losses. The operating axial thrust loads on the bearing are provided from Table 4-5. The results of the analyses are presented in Table 5-1.

TABLE 5-1
Thrust Bearing Performance at 60 gpm Flow and 100 Feet of Head

Viscosity, cP	1.00	30.0	250	375	600	1000
Specific Gravity	1.00	1.70	1.70	1.70	1.70	1.70
Speed, rpm	1918	1942	2020	2060	2151	2262
Axial Thrust, lbf	534	796	939	974	1038	1102
Minimum Film Thickness, mils	0.7	2.5	5.7	6.7	8.1	9.7
Power Loss, hp	0.2	1.3	5.4	7.3	10.7	16.5

The thrust bearing minimum film thicknesses identified in Table 5-1 are significantly larger than the 0.50 mil that EMD generally uses as a guideline for acceptable operation. Minimum film thickness is not a concern for high viscosity operation, and the results are acceptable for operation of the Model 2A-513-A1 NGTP. The power losses identified in Table 5-1 were used in Table 4-4 for use with calculating required output power of the motor.

6.0 MOTOR ANALYSIS

The Model 2A-513-A1 NGTP was analyzed for operation in high viscosity fluids using the same "AA102" canned motor design code and the same electrical model as was used in the original design analysis, Reference 9-3. The analysis was performed for water (a 1.0 sg and 1.0 cP viscosity liquid) and 1.70 sg fluids with viscosities of 30, 250, 375, 600 and 1000 cP. The analyses determined motor performance using the six motor output power and operating speed requirements delineated in Table 4-4.

For each of the six cases, voltage and frequency were varied using a constant 7.333 volts/Hz ratio ($440 \text{ volts} / 60 \text{ Hz} = 7.333$). The solution for each case was obtained when the required speed and shaft horsepower converged to their required values while iterating with fluid and friction factors. The results of the motor analysis are presented in Table 6-1. Note that motor efficiency is based on hydraulic (impeller) required power not total (hydraulic plus friction) required power.

No solution was obtained in the "AA102" code for the 600 cP and 1000 cP conditions analyzed. This indicates that the motor can not come up to operating speed and will stall when trying to reach new design point operation in viscosities above 375 cP. Figure 6-1, which plots required torque versus available motor torque, indicates that required torque exceeds available torque around 440 cP, and the pump is not capable of operating at the new design point above this viscosity.

The speed-torque curve for the 1.0 cP viscosity water operating condition (32.7 Hz supply frequency) is shown in Figure 6-2. The speed torque curves for the 30 cP (34.1 Hz), 250 cP (37.5 Hz), and 375 cP (39.7 Hz) operating conditions are shown in Figures 6-3 through 6-5, respectively.

TABLE 6-1

Transfer Pump Viscosity Performance at 60 gpm Flow and 100 Feet Head

Fluid Properties						
Specific Gravity	1.0	1.70	1.70	1.70	1.70	1.70
Viscosity, Centipoise	1.0	30	250	375	600	1000
Temperature, °F	120.0	120.0	120.0	120.0	120.0	120.0
Hydraulics Shaft Power, hp	6.8	12.0	16.1	18.3	22.9	29.1
Total Mechanical Losses, hp	7.8	16.4	33.9	43.2	61.2	94.0
Operating Speed, rpm	1918	1942	2020	2060	2151	2262
Required Motor Torque, ft-lbs	21.4	44.3	88.1	110.1	148.9	218.2
Motor Performance						
Maximum Torque, ft-lbs	149.5	146.1	132.7	126.1	>126	>126
Terminal Voltage, volts	240	250	275	291	stall	stall
Supply Frequency, Hz	32.7	34.1	37.5	39.7	stall	stall
Line Current, amperes	52.3	63.2	97.4	118.6	stall	stall
Input Power, KW	8.2	15.6	32.6	42.6	stall	stall
Total Electrical Losses, KW	2.4	3.4	7.3	10.4	stall	stall
Efficiency, %	61.8	57.4	36.8	32.0	stall	stall

Figure 6-1
Motor Mechanical Performance Versus Viscosity

Figure 6-2

Transfer Pump - S.O. V008 - Viscosity Study Run (32.659hz) - 5/17/99

Motor Performance at 90.0% of Rated Voltage

Figure 6-3
Transfer Pump - S.O. V008 - Viscosity Study Run (34.091hz) - 5/18/99
Motor Performance at 90.0% of Rated Voltage

Figure 6-4

Transfer Pump - S.O. V008 - Viscosity Study Run (37.5hz) - 5/18/99

Motor Performance at 90.0% of Rated Voltage

Figure 6-5

Transfer Pump - S.O. V008 - Viscosity Study Run (39.682hz) - 5/18/99

Motor Performance at 90.0% of Rated Voltage

7.0 THERMAL ANALYSIS

The thermal analysis of the Model 2A-513-A1 NGTP motor was performed in the same manner as identified in Reference 9-3, Section 7.1.2. The motor was analyzed at the new design point (see Table 3-2) for water and for fluids with a specific gravity of 1.70 and viscosities of 1.0, 30, 250, and 375 cP.

Pump operation in fluids with viscosities higher than 375 were not specifically evaluated. Motor stall occurs around 600 cP as identified in Section 6.0, and data could be extrapolated as necessary between 375 and 600 cP based on results at 30, 250, and 375 cP.

The primary interest in the motor thermal analysis is the maximum winding surface temperature (MWST). The MWST determines the operational life of the motor insulation system. The Model 2A-513-A1 NGTP is equipped with a Class "N" insulation system which conservatively has a 40 year (350,000 hours) continuous operating life with an MWST of 200 °C. For every 10 °C temperature rise of MWST above 200 °C, the continuous operating life of the insulation system is halved (i.e., with an MWST of 230 °C, the insulation life is 5 years, or 44,000 hours).

The major factors affecting the Model 2A-513-A1 NGTP motor winding MWST are the following:

- Pumped fluid ambient temperature. The fluid ambient temperature plus the motor winding maximum temperature rise equals the MWST.
- Fluid flow rate through the motor rotor/stator can annulus. The can annulus flow rate has a minor affect on the Reynolds Number and a major affect on the temperature rise of the fluid passing through the can annulus region. Turbulent flow (high Reynolds Number) in the can annulus region will provide significantly higher heat transfer film coefficients than laminar flow (low Reynolds Number) on the stator can.
- Heat transfer film coefficient on the stator can. The film coefficient magnitude is proportional to the heat removal rate from the stator can. This affects the heat removal rate from the wound stator.

- The thermal diffusivity of the pumped fluid. Thermal diffusivity is the ratio of thermal conductivity to the product of specific heat times density. Thermal diffusivity determines how fast heat can move through a material and affects the heat removal capability of the fluid passing through the rotor/stator can annulus region.
- The operating speed of the rotor. This has a major affect on the Reynolds Number of the fluid passing through the can annulus region.
- The motor line current. The motor line current is the major heat source of the motor windings.

The thermal analyses were performed at the new design point (60 gpm flow at 100 feet of head at the discharge) for water and 1.70 specific gravity fluids with viscosities of 30, 250 and 375 cP. Analyses were not performed at viscosities higher than 375 cP because the motor stalled at 600 cP (see Section 6.0), and MWST data could be extrapolated between 375 and 600 cP if necessary. The results of the thermal analyses are delineated in Table 7-1.

TABLE 7-1

Thermal Analysis Results at 60 gpm Flow and 100 Feet Head

Fluid Properties						
Temperature, °F	120	120	120	120	120	120
Specific Gravity	1.00	1.70	1.70	1.70	1.70	1.70
Viscosity, cP	1.00	30.0	250	375	600	1000
C _p , Btu/lbm-°F	1.00	0.75	0.75	0.75	0.75	0.75
K, Btu/hr-ft-°F	0.364	0.091	0.091	0.091	0.091	0.091
Operating Speed, rpm		1918	1942	2020	2060	2151
Motor Flow, gpm		30.9	28.3	14.4	10.3	6.3
Annulus Reynolds No.		37504	2130	256	172	110
Mechanical Losses, hp		7.8	16.4	33.9	43.2	61.2
Motor Performance						
Supply Voltage, volts	240	250	275	291	stall	stall
Supply Freq, Hz	32.7	34.1	37.5	39.7	stall	stall
Line Current, amps	52.3	63.2	97.4	118.6	stall	stall
Input Power, KW	8.2	15.6	32.6	42.6	stall	stall
MWST, °C	75	90	253	351	stall	stall
Insulation Life, hours	2x10 ⁹	7x10 ⁸	8894	10	stall	stall
Fluid Temperature Rise						
Discharge Fluid, °F	0.2	0.3	0.8	1.1	stall	stall
Motor Fluid, °F	1.4	2.3	8.9	15.8	stall	stall

The Table 7-1 values for operating speed, motor circulation flow and mechanical losses are obtained from results provided in Section 4.0, Tables 4-4 and 4-6. The motor supply voltage, frequency, current and input power values provided in Table 7-1 are obtained from Section 6.0.

The Reynolds Number for flow through the can annulus region is calculated as follows:

$$Re = (D_E / 12) V_R \rho / \mu$$

$$D_E = OD - ID$$

$$V_R = (V^2 + U_F^2)^{1/2}$$

$$V = 0.321 Q / A$$

$$U_F = 0.5 N D_{RC} / 229$$

$$\rho = 62.4 \text{ sg}$$

$$\mu = 12 \bullet 386.4 \bullet 1.45 \times 10^{-7} \text{ cP}$$

where: D_E is the effective annulus diameter, inch

OD is outside diameter of the annulus, 4.143
inches

ID is the inside diameter of the annulus, 3.732
inches

V_R is relative velocity of the annulus fluid, ft/s

V is the axial fluid velocity, ft/s

Q is the flow rate through the annulus, gpm

A is the annulus area, 2.542 in²

U_F is the forced vortex velocity in the annulus, ft/s

N is the rotor rotational speed, rpm

D_{RC} is the outside diameter of the rotor can, 3.732
inches

ρ is the fluid density, lb/ft³

sg is the fluid specific gravity

μ is the fluid absolute viscosity, ft/lb-s

cP is the fluid absolute viscosity, centipoise

The temperature rise of the fluid pumped by the impellers and the fluid pumped through the motor are calculated as follows: (Based on the Reference 9-3 thermal analysis it is assumed that one-third of the heat generated by the motor is removed in the main discharge annulus between the stator jacket and stator shell and two-thirds of the heat generated is removed by the fluid passing through the rotor-stator can annulus in the motor.)

$$\Delta T = H_{\text{GEN}} / m C_p$$

$$m = 8.345 \cdot 60 \text{ sg } Q$$

where: ΔT is the temperature rise of the fluid, °F

H_{GEN} is the generated in the motor, Btu/h

m is the mass flow rate, lbm/h

C_p is the specific heat of the fluid, Btu/lbm-°F

sg is the specific gravity of the fluid

Q is the flow rate of the fluid through the annulus,
gpm

Based on the results of the thermal analyses which calculated MWST for various viscosities, the motor insulation operating lives were calculated using the following:

$$L = 40 / 2^n$$

$$L_H = 8760 L$$

$$n = (T - 200) / 10$$

where: L is insulation operating life, years

L_H is insulation operating life, hours

n is the de-rating factor

T is the MWST, °C

The insulation operating life of the Model 2A-513-A1 NGTP is graphed in Figure 7-1.

Figure 7-1
Motor Insulation Life Versus Viscosity

Using a minimum required insulation life of 5,000 hours for the Model 2A-513-A1 NGTP, Figure 7-1 indicates that the maximum viscosity capability is approximately 250 cP for operation at the new design point (see Table 3-2). Although the pump is capable of operation at 375 cP, the insulation operating life at this viscosity is only about 10 hours. In summary, the Model 2A-513-A1 NGTP is capable of operating at the new design point with no reduction in a 5,000 hour design life with fluid viscosities up to 250 cP. The pump is also capable of operating in fluids with viscosities between 250 and 375 cP with significant reductions in operating life due insulation degradation caused by high electrical winding temperature.

8.0 DYNAMIC ANALYSIS

8.1 Model Description

This section documents the dynamic behavior of the Model 2A-513-A1 NGTP at a single point of operation. The previous dynamic analysis performed on this pump is documented in Section 10.0 of Reference 9-8.

The point of operation evaluated is the highest recommended fluid viscosity (250 cP) at the new design point (refer to Table 3-2). Only this operating point is documented herein for the dynamic evaluation. All other documentation for the dynamic evaluation is in Reference 9-8. Also, as in the previous report, the fluid properties are assumed to be Newtonian.

The dynamic model of the pump rotor and stator, as well as the column is included in the dynamic analysis of the transfer pump. This system model is used for all the dynamic analyses performed in Section 8.0. The math model was similar to that used in Reference 9-8. Differences are related to the higher viscosity of the fluid in the can annulus (250 cP versus 30 cP).

8.2 Rotor Modeling Description

The rotor assembly was modeled using the finite element computer code "WECAN". The different stages of the rotor section diameters, length, material density, and modulus of elasticity were used to form the model inputs. A proprietary in house code ("PHYTRI") is used to calculate equivalent elements from detailed input for the finite element model. This code is verified and under configuration control. The left side of Figure 8-1 shows the detailed geometric values used as input into the PHYTRI code. The right side of Figure 8-1 shows stiffness elements calculated by the PHYTRI code, that were then entered into the finite element model.

Figure 8-1
Rotor Model -- Detailed Geometric Input Used
Rotor Model- Half of cross-section of Equivalent Stiffness Elements Calculated

8.3 Rotor, Stator, and Pipe Column Modeling Description

The entire system assembly (Figure 15-2 of Reference 9-8) was modeled using the finite element computer code "WECAN". The different stages of the rotor and stator section diameters, length, material density, and modulus of elasticity were used to form the model inputs. The "PHYTRI" code is used to calculate equivalent elements from detailed input for the finite element model. The left side of Figure 8-2 shows the detailed geometric values used as input into the "PHYTRI" code. The right side of Figure 8-2 shows stiffness elements calculated by the "PHYTRI" code, that were then entered into the finite element model. The resulting model was then modified to add the calculated bearing and seal stiffness values, and the pipe column elements for representation of the entire system assembly.

Stiffness parameters for the radial and thrust bearings, and the impeller seals were all modeled. The radial bearings were modeled using the in-house computer code "TILTPAD" for tilting pad bearings. The computer code "TILTPAD" was used to compute the bearing stiffness and damping for the tilting pad bearings. The thrust bearing was modeled using the special in house computer code "SECPAD" for thrust bearings. For this system there is presumed no coupling between axial and lateral motions of the pump. The "SECPAD" program computes a dynamic axial stiffness value for each pad. Then a total value of thrust bearing axial stiffness was computed to account for all six thrust bearing pads. This value should not be significant, since there are no known excitations in the system axially that could cause problems. Seal stiffness values were calculated based on a plain seal using friction factors, Reynolds numbers, and pressure drops that were calculated from the fluid internal circulation analysis.

The dynamic evaluation of the Model 2A-513-A1 NGTP was performed at one point of operation. The case analyzed was for the pump running in the process fluid with a viscosity of 250 centipoise and a specific gravity value of 1.70. Values for pump motor performance needed to do the analysis included the pump speed (2020 rpm), all electrical parameters such as motor torque (502 in-lb), and the radial hydraulic load for each impeller. The values for this point of operation were needed before a dynamic evaluation could be performed. A basic assumption made for the analysis was that the fluid is Newtonian, since the current calculations can not properly address non-Newtonian fluids.

The calculations included the motor can annulus stiffness, damping, and mass calculations. Viscous and inertial effects were both considered to properly calculate the motor can annulus fluid properties. These fluid properties were then added to the magnetic properties of the motor can annulus to calculate the total stiffness, damping and mass of the motor can annulus. Other calculations included definition of the single pad dynamic parameters, and the pad assembly dynamic parameters. Together these bearing calculations define the bearing properties. Next the calculation of bearing steady state loads was performed. Also, the calculation of the seal fluid properties of stiffness, damping, and mass were updated for the viscous process fluid.

The piping column assembly (Figure 15-5 of Reference 9-8) was added to the dynamic model by using pipe elements in the finite element code ("WECAN") with the proper diameter and wall values to represent the various pipes used. To simplify the model the outer pipe was modeled directly and all the inner pipes were lumped together as one equivalent pipe using the "PHYTRI" computer code. This should be somewhat representative of how the actual piping will act due to the design construction of the piping system. The top of the piping was considered to be ground with the other end connected to a pipe element that modeled the transition piece. The transition piece is connected at the other end to the top of the pump stator (node 100 in the model shown in Figure 8-2). The tank fluid was not included in the modeling to represent the worse case of damping to the piping column when the pump is running with little or no fluid next to the piping.

This transfer pump system finite element dynamic model is used to perform modal analysis of the entire system. Results from the modal analysis are then used to create a damped harmonic response model for stability and vibration prediction using the proprietary in-house computer codes "DYSTAB" and "HARRESX", respectively. These codes are under configuration control and are verified.

Figure 8-2

Rotor & Stator Model – Detailed Geometric Input Used

Rotor & Stator Model – Half of cross-section of Equivalent Stiffness Elements Calculated

8.4 Undamped Natural Frequency Analysis Description

The Model 2A-513-A1 NGTP system finite element dynamic model was used to perform an undamped modal analysis of the entire system, as described previously in Reference 9-8. The entire system includes the pump rotor and stator, and the piping column. This analysis is based on the pump running at some steady state position so that constant rotor bearing stiffness values were assumed. The non-linear effects of the rotor lateral bearings are not included in this modal analysis calculation, since this is a linear calculation. Results from the modal analysis do not include any damping or dynamic loading effects. Also, this modal analysis model does not include the effects of the motor can annulus stiffness and damping. The motor can annulus effects are included in the damped natural frequency analysis (Section 8.5).

The results of the undamped modal analysis are used to evaluate the system undamped frequencies and are input into the damped frequency analysis model used in Section 8.5.

One case was examined for the pump running in the process fluid at the a viscosity of 250 centipoise. This is just one case of rotor lateral bearing stiffness values. Various bearing stiffness values are possible due to the non-linear stiffness of the lateral bearings and the different potential loading of the bearings. The case analyzed was for a stator can offset of zero. This case was selected since it provides the smallest loading on the bearings and is a potentially less stable condition for the pump.

8.5 Damped Rotor Dynamic Analysis Description and Results

A damped dynamic analysis of the pump rotor in the entire transfer pump system was performed. The damped harmonic response program uses the undamped modal analysis mode shape results, as described in Section 8.1 of this report and in Section 10.3 of Reference 9-8. It then applies the corrections to the model for bearing stiffness and damping values. Also it adds the effects of the can annulus stiffness, damping and total inertia, as described in Section 8.1 of this report. The resulting complete dynamic model was then analyzed for stability and vibration prediction. The case analyzed was for the pump operating in the process fluid at 250 centipoise and a zero stator can offset. To predict vibration of the rotor in the pump relative to the stator the maximum unbalance values allowed by the pump specification were applied to the pertinent locations along the rotor.

The damped natural frequency results of the stability analysis for the pump in process fluid are shown in Table 8-1. The stability analysis shows that all modes are stable for the case analyzed. The results of the vibration prediction for the pump in process fluid are shown in Table 8-2. Results are compared for the pump running in water to the pump running in the process fluid. These results are at the same rotor speed and stator can offset.

Reduced vibration occurs with the process fluid in the pump due to the added damping and changes to the other fluid effects. The results with the pump operating in the process fluid at the highest viscosity show very low vibration response due to unbalance.

TABLE 8-1 - Transfer Pump Rotor Stability Results -- System Damped Frequencies
Comparison of Pump in Water to Pump in Process Fluid

Pump Fluid Viscosity Rotor Speed Damped Mode	Pump in Water 1 cP 3860 rpm Frequency (Hz)	Process Fluid 30 cP 3860 rpm Frequency (Hz)	Process Fluid .250 cP 2020 rpm Frequency (Hz)
1	0.46	0.46	* 0.14
2	0.46	0.46	0.46
3	3.80	* 0.70	0.46
4	3.80	3.74	3.74
5	* 6.05	3.74	3.74
6	11.38	* 7.06	* 4.56
7	11.38	11.17	11.17
8	22.75	11.17	11.17
9	22.76	22.35	22.35
10	* 22.89	22.35	22.35
11	33.13	33.10	33.10
12	37.81	37.18	37.18
13	37.82	37.18	37.18
14	* 45.46	55.87	54.43
15	56.83	55.87	55.88
16	56.94	56.61	55.89
17		57.12 * 56.91	56.26
18	* 60.25	78.79	71.97
19	80.44	78.87	78.88
20	80.53	88.93	78.92
21	107.61	107.57	107.56
22	109.75	107.76	107.61
23	109.80	124.11	124.11
24	114.20	140.37	136.47
25	124.46	140.54	139.16
Note: All Modes were stable. * Highly damped rotor modes below the running speed.			

TABLE 8-2 - Vibration Response Results

**Radial Displacements of Rotor Relative to Stator & Bottom of Pump Relative to Ground
Comparison of Pump in Water to Pump in Process Fluid**

Type of Pump Fluid Viscosity Rotor Speed	Water 1 cP 3860 rpm	Process Fluid 30 cP 3860 rpm	Process Fluid 250 cP 2020 rpm
Location	Displacement (inches)	Displacement (inches)	Displacement (inches)
Upper Bearing	0.000483	0.000093	0.000003
Can Midpoint	0.000187	0.000418	0.000077
Lower Bearing	0.000817	0.000086	0.000004
2nd Stage Impeller Seal	0.001329	0.000533	0.000126
1st Stage Impeller Upper Seal	0.001600	0.000718	0.000188
1st Stage Impeller Lower Seal	0.001888	0.000908	0.000252
Bottom of Pump (Relative to ground)	0.000294	0.000214	0.000028

9.0 REFERENCES

- 9-1 *Hydraulic Institute Standards*, Hydraulic Institute, 1975.

- 9-2 R. F. Guardiani, "Model 2A-513-A1 Transfer Pump Lead Unit Test Report", WEMD, October 1, 1997.

- 9-3 R. F. Guardiani, "Model 2A-513-A1 Lead Unit Final Design Report", EM 6673, Revision 4, October 6, 1997.

- 9-4 H. Schlichting, *Boundary Layer Theory*, Sixth Edition, 1968, McGraw-Hill.

- 9-5 J. W. Daily and R. E. Nece, "Chamber Dimension Effects on Induced Flow and Frictional Resistance of Enclosed Rotating Disks, *Journal of Basic Engineering*, March 1960, pp217-232.

- 9-6 T. Theodorsen and A. Regier, "Experiments on Drag of Revolving Disks, Cylinders, and Streamline Rods at High Speeds", NACA-TR-793.

- 9-7 E. Bilgen and R. Boulos, "Functional Dependence of Torque Coefficient of Coaxial cylinders on Gap Width and Reynolds Numbers", *Journal of Fluids Engineering*, March 1973, pp. 122-126.

- 9-8 R. F. Guardiani, "Model 2A-513-A1 Lead Unit Final Design Report", EM 6673, Revision 3, April 30, 1997.

10.0 ACKNOWLEDGEMENTS

This engineering memorandum is a report on the operability capability of the Model 2A-513-A1 NGTP in high viscosity environments. The calculations and technical content are compiled from information provided by the participants on the viscosity evaluation team. The following listing acknowledges the contributions of those participants.

Coordinating Author - R. F. Guardiani, Principal Engineer
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J. R. Raymond, Contract Thermal Analysis Engineer

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APPENDIX D

INTEROFFICE MEMO

LOCKHEED MARTIN

RPP-5117, Rev. 0
Addendum D-1

From: Process Control 74B50-99-090
Phone: 373-3115 R2-11
Date: September 22, 1999
Subject: COMPARISON OF COMPOSITION OF TANK 241-SY-101 TO PUMP SPECIFICATIONS

To: C. E. Hanson S7-70

cc: W. B. Barton R2-11
J. M. Conner R2-11
S. D. Estey R2-12
J. Irons H1-19
N. W. Kirch R2-11 *AWK*
W. J. Powell S7-70
DAR File/LB

- References:
- (1) HNF-4264, "Process Control Plan for Tank 241-SY-101 Surface Level Rise Remediation," Rev. 0, dated June 1999.
 - (2) WHC-SD-WM-ER-409, "Tank Characterization Report for Double-Shell Tank 241-SY-101," Rev. 0, dated July 1995.
 - (3) WHC-S-0238, "Specification for Prototype Tank Farm Transfer Pumps with Submersible Motor," Rev. 1, dated August 5, 1994.

The Tank 241-SY-101 transfer pump was designed per specification documented in Reference 3. Table 1 (attached) shows tank 241-SY-101 (SY-101) waste composition and parameters for those specifications. There are three columns. The low value represents the lowest value found in SY-101 anywhere in the tank. The high column is the highest value found anywhere in the tank. The expected value is the estimate for the convective layer waste. The values came from the tank characterization report (Reference 2).

While the convective layer is the layer that will be pumped, the pump internals will see a diluted waste. Current plans call for a 1 to 1 dilution with water with a minimum of 1 part water to 2 parts waste. Table 2 (attached) shows the diluted waste compared with the pump specifications. Some of the data comes from the process control plan (Reference 1).

The values that exceed the pump specifications are shaded. The total sodium will be higher than the pump specifications at all expected dilutions. This should not affect the pumpability

C. E. Hanson

Page 2

September 22, 1999

74B50-99-090

RPP-5117, Rev. 0
Addendum D-2

of the waste nor is the material of construction expected to be harmed by the higher sodium values. Other places where the undiluted waste exceeds the specifications will come back into specifications with the dilution.

If you have any questions, please call me on 373-3115.

A handwritten signature in black ink, appearing to read 'D. A. Reynolds', with a long horizontal flourish extending to the right.

D. A. Reynolds

dar/mjg

Attachment

Table 1 241-SY-101 Composition				
Constituent or Property	Lowest Measured in Tank	Convective Layer	Highest Measured in Tank	Reference
NaOH, <u>M</u>	1.28	2.13	2.93	Table 4-5 & Appendix (Ref. 2)
NaAlO ₂ , <u>M</u>	1.4	2.15	2.42	Table 4-5 & Appendix (Ref. 2)
NaNO ₂ , <u>M</u>	1.83	4.25	5.14	Table 4-5 & Appendix (Ref. 2)
NaNO ₃ , <u>M</u>	2.13	3.55	6.36	Table 4-5 & Appendix (Ref. 2)
Na ₂ CO ₃ , <u>M</u>	0.152	0.692	4.25	Table 4-5 & Appendix (Ref. 2)
Na ₃ PO ₄ , <u>M</u>	0.068	0.085	0.311	Table 4-5 & Appendix (Ref. 2)
Na ₂ SO ₄ , <u>M</u>	0.0062	0.018	0.28	Table 4-5 & Appendix (Ref. 2)
NaF, <u>M</u>	0		0.16	Table 4-5 & Appendix (Ref. 2)
Total Na, <u>M</u>	12.55	13.83	17.44	Table 4-5 & Appendix (Ref. 2)
Fe(OH) ₃ , <u>M</u>	0.00046	0.0031	0.148	Table 4-5 & Appendix (Ref. 2)
ZrO ₂ ·2H ₂ O, <u>M</u>				Not measured in SY-101
pH	12.9		13.7	Appendix (Ref. 2)
Specific Gravity, g/mL*	1.43	1.5	1.93	Table 4-4 (Ref. 2)
Absolute Viscosity, cP	12	30	1000	Figure 4-1 and 4-2 (Ref. 2)
H ₂ O %	18.1	38.4	40.5	Table 4-5 & Appendix (Ref. 2)
Solids (Vol%)*	8		78	Table 4-5 (Ref. 2)
Solids Size (microns)	0	Sauter Mean ranging from 1.41 to 15.3	300	Table 4-6 (Ref. 2)
Miller Number				Not measured in SY-101

*Depends on temperature and location.

Table 2 Comparison of the Diluted Waste with the Range Specified for the Pump					
Constituent or Property	Desired Range	Convective Layer	1 part water to 2 part waste dilution	1 part water to 1 part waste dilution	Reference
NaOH, <u>M</u>	0 to 2.5	2.13	1.42	1.07	
NaAlO ₂ , <u>M</u>	0 to 2.0	2.15	1.43	1.07	
NaNO ₂ , <u>M</u>	0 to 3.0	4.25	2.83	2.13	
NaNO ₃ , <u>M</u>	0 to 4.0	3.55	2.37	1.77	
Na ₂ CO ₃ , <u>M</u>	0 to 5.0	0.692	0.461	0.346	
Na ₃ PO ₄ , <u>M</u>	0 to 1.0	0.085	0.057	0.047	
Na ₂ SO ₄ , <u>M</u>	0 to 1.0	0.018	0.12	0.009	
NaF, <u>M</u>	0 to 0.2				
Total Na, <u>M</u>	≤ 6.0	13.83	9.22	6.92	
Fe(OH) ₂ , <u>M</u>	0 to 2.0	0.0031	0.0021	0.0015	
ZrO ₂ *2H ₂ O, <u>M</u>	0 to 2.0				Not measured in SY-101
pH	≥ 9.0	13.7			Diluted values not measured
Specific Gravity, g/mL	1.0 to 1.5	1.5	1.49	1.34	Table 2-2 (Ref. 1)
Absolute Viscosity, cP	1.0 to 30	30	10.5	2.9	Table 2-2 (Ref. 1)
H ₂ O %	50 to 100	38.4	59	69	Calculated
Solids (Vol%)	0 to 50	5% to 25%	2% to 17%	2% to 17%	Page 2-1 (Ref. 1)
Solids Size (microns)	1 to 1000 (95% <50)	Sauter Mean ranging from 1.41 to 15.3	Sauter Mean ranging from 1.41 to 15.3	Sauter Mean ranging from 1.41 to 15.3	Table 4-6 (Ref. 2)
Miller Number	<50				Not measured in SY-101

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