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STATE OF TECHNOLOGY STUDY--PUMPS
EXPERIENCE WITH HIGH TEMPERATURE
SODIUM PUMPS
IN NUCLEAR REACTOR SERVICE
AND THEIR APPLICATION TO FFTF

December 1969

AEC RESEARCH &
DEVELOPMENT REPORT

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EXPERIENCE WITH HIGH TEMPERATURE SODIUM PUMPS
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By

W. Babcock

Engineering Department
FFTF Division

December 1969

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ABSTRACT

A state of technology survey has been made defining design features, development problems, and operating experience for high-temperature sodium pumps that have some extrapolation value for the FFTF Reactor Plant.

It has been found that Fermi type pumps are best suited to meet FFTF pump requirements, for both main and closed loop systems.

The total analysis indicates that FFTF could go "on line" with maximum predictable reliability with the Fermi type of main heat transport and closed loop systems pumps.

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INTRODUCTION

This report was prepared by Battelle-Northwest under Contract No. AT(45-1)-1830 for the Atomic Energy Commission, Division of Reactor Development and Technology.

The study described herein, which relates to sodium pumps as components in nuclear reactor systems, is one of several investigations being carried out by Battelle-Northwest to provide data for design of the FFTF heat removal system and closed loops, and for application in LMFBR design planning.

I. SUMMARY

This report describes work performed by Battelle-Northwest for the purposes of (a) defining the types of sodium pumps which offer greatest potential for immediate application in FFTF, and (b) defining the types of research and development work which should be carried out to provide pumps for ultimate FFTF and LMFBR operating conditions. While this study necessarily considered the effects of variations in systems design and operating parameters on pump requirements and performance, it was not within the scope of the study to define the optimum design configuration for the FFTF heat removal system.

A. PUMPS FOR FFTF INITIAL CONDITIONS

1. Pump Types

It was found that no existing sodium pumps can meet all of the FFTF initial requirements for service in the FFTF primary and secondary main heat transport system, and in the primary systems of the FFTF closed loops. For these applications it was concluded that the "Fermi-type pump" (mechanical centrifugal vertical pump with the inlet located in a sump) should be modified to meet FFTF needs. Fermi-type pumps would require the least amount of modification to meet FFTF needs. They would provide the best long range reliability least systems complexities, and fewer hazards problems. The Fermi pumps are considered superior to canned motor centrifugal pumps, electric magnetic pumps, and the "Hallam type" of mechanical centrifugal vertical pumps.

For closed loop secondary pumps, FFTF requirements can be met by either the Rapsodie single stage pump or the LASL multistage pump. Both are Fermi-type pumps. The LASL multistage pump is preferred because of its greater compactness, simplicity and lower cost.

Other considerations favoring Fermi-type pumps are as follows:

- Fermi designs best accept a double-suction impeller, which will resolve many of the critical problems relating to shaft length, Net Positive Section Head (NPSH), sub-atmospheric inlet piping pressures, emergency cooling, impeller stress, and axial thrust loads.
- Pump component parts, as universally used, represent no departure from the state of technology for FFTF. These state-of-technology component parts are also best represented in size and configuration by those used in Fermi pumps. Some improvement in seal life would be desirable. In general, FFTF should not depart from these concepts.

2. Effect of Pump Location

As part of this study the author was requested to examine the effect, on primary pump requirements and performance, of location of the pump in the primary system. This evaluation was limited to the effects on the pump only. It was found that location of the pump in the hot leg of the FFTF primary system would permit a minimum (6 ft) extrapolation of pump shaft length (increase from 18 ft length to 24 ft length) over current proven practice, while location of the pump in the cold leg would require an extrapolation of 12 ft (increase from 18 ft to 30 ft). While the reduced shaft length in the hot leg location would be beneficial in simplifying pump design, the hot leg location would also result in an increase of 300 °F in FFTF pump operating temperature (from ~900 to ~1200 °F). This increased temperature would result in a 65% reduction in "allowable" materials protection. Based on these facts, the writer has concluded that a system design which placed the pumps in the hot leg of the primary circuit is not advantageous insofar as the pumps are concerned. However, as stated above,

this study did not include an evaluation of the total effect on the heat transport system of such a hot leg design.

B. DEVELOPMENT REQUIRED TO MEET FFTF AND LMFBR ULTIMATE CONDITIONS

The long term FFTF pump will be a 1200 °F pump for 600 to 700 ft Thermal Dynamic Head (TDH). This combination of operating requirements is well beyond current technology for a single stage pump because impeller stresses caused by the high operating speeds and larger diameter will most likely exceed ASME code allowables. Therefore, the next stage of development for a high temperature, high TDH pump should be a multistage pump which has the greatest potential for reducing operating speed and impeller stress while still remaining with proven hydraulic state of technology. Battelle-Northwest has recommended development of such a pump as a backup primary pump for the FFTF, and for LMFBR applications.

II. PRIMARY SODIUM PUMPS FOR INITIAL FFTF SERVICE

A. COMPARISON OF FFTF REQUIREMENTS WITH STATE-OF-THE-ART

FFTF initial requirements for primary sodium pumps cannot be met by any one existing pump.

In general, as shown in Tables I, II, III, sodium pumps have been developed and successfully applied for the following limitations of parameters:

- To 13,800 gpm in the U.S. (Fermi), and to 16,500 gpm in the BN-350 in the U.S.S.R.⁽²⁾
- To 610 ft TDH in the CANEL project.⁽⁵⁾
- To 1500 °F operating temperatures in the CANEL project.⁽⁵⁾
- To 17 ft abs NPSH (Hallam).⁽⁶⁾
- With up to 40 ft vertical shaft length (Fermi).⁽⁶⁾
- With up to 5 stages (Fermi).⁽⁷⁾
- FFTF requirements thus represent moderate extension of state-of-the-art in pumps for handling sodium in the primary system.

B. SELECTION OF TYPE OF PRIMARY PUMP FOR FFTF

1. Basic Pump Types Evaluated

The types of pumps included in this survey were:

- Mechanical free-surface, centrifugal, vertical-shaft sump pumps (single stage)
- Mechanical free-surface, centrifugal, vertical-shaft sump pumps (multistage)
- Mechanical free-surface, centrifugal, vertical-shaft piped pumps
- Canned rotor centrifugal pumps
- Electromagnetic and magnetically driven pumps.

Operating experience with the above types of pumps has overwhelmingly favored the mechanical free-surface centrifugal pumps--both single and multistage--for reliable sodium service. Since the resolving of initial bearing clearance and alignment

problems, failure modes have been primarily concerned with mechanical seals, motor bearings, and eddy current couplings. Large pumps of these types have logged impressive numbers of relatively trouble-free hours of service.

Canned rotor centrifugal pumps have demonstrated shorter life and more failures in high temperature service and are deemed unsuitable for FFTF primary reliability goals. Further details are given in Exhibit A.

Electromagnetic or magnetically driven pumps have some advantages for small pump services, but consistent failures of pump ducts from burning and cracking indicate that the state-of-the-art has not advanced enough to make them suitable for near term FFTF goals. See also Exhibit A.

2. Description of Mechanical Centrifugal Pumps of Fermi and Hallam Type

Because the Fermi and Hallam primary pumps are best known, and because they represent the two basic types of pump concepts used in vertical free-surface designs, this part of the survey will classify all such designs as either a Fermi* (Figures 1 and 2) or Hallam* (Figures 3 and 4) type. Fermi-type pumps are defined as sump pumps where the pump inlet is not piped but is oriented to contact the sump fluid. The Hallam-type pumps, however, have inlets directly piped to a heat exchanger or reactor vessel with the sump fluid in contact with the discharge side of the pump.

The basic differences in the Fermi and Hallam pump designs are that the Fermi design takes suction from a sump and has the discharge piped directly to the system piping, while the Hallam design has the pump suction piped directly to the system piping, with the pump discharge in association with both the sump and system discharge piping.

* *Other Fermi and Hallam type pumps are discussed in Exhibit A.*

TABLE I. Parameters for Primary Sodium Pumps

Plant	Capacity, gpm	TDH, ft	NPSH	Design Temp, °F	Motor, hp	Speed, rpm	Driver Type	Speed Control	Shaft, ft	Seal Type	Sodium Bearing Type	Impeller Type	Basic Material	Basic Pump Type	Pump Classification	Supplier	Approx. Hours of Operation
Fermi	11,800	360	<35	1000	1000	900	Wound Rotor	Liq. Rheostat	18	Double Oil	Hydrostatic	S.S.S.S. (a)	304 S/S	Mech. FS	Fermi	Byron Jackson	27,000
Hallam	7,200	160	17	1000	350	900	ac Induction	Eddy Current	14						Hallam		21,000
EBR-II	5,500	200	<35	800	300	1075	ac Induction H.S.	Var. Freq. and Volt.							Fermi		14,000
SCTI	3,200	242	<35	900	250	1750	ac Motor	None							Hallam		44,000
SRE-PEP	2,700	180	35	1200	150	1075	ac Motor	Mag. Coupling									44,000
CANEL	4,000	610	25 - 40	1300	700	2450	ac Motor			Mech. Seal	None			Canned Rotor	Fermi		>16,000
Rapsodie	1,650	105	<35	1200	73	1100	Ward Leonard System			Double Oil	Hydrostatic		316 S/S	Mech. FS			
PFR	18,000	320	51	750	1900	960	Var. Speed Motor	Hyd. Coupling				D.S.S.S. (b)	S/S			England	
PFR Backup	18,000	320	<51	750	1900	960	Var. Speed Motor	Hyd. Coupling				T.S.S.S. (c)				England	
Phenix	17,600	290				1000	Var. Speed Motor		16.4							France	
Pompes Guinard	3,170	302	<35	1200	300	1450										Byron Jackson	
BN-350	14,200	360			2280	1000	2 Speed Motor				None					Russia	
FFTF Conc.	12,000	450	20 - 25	1200	1500	1050	Wound Rotor	Liq. Rheostat 25 - 30			Hydrostatic	D.S.S.S.	304 S/S		Fermi	Byron Jackson	
PFR Proto	7,100	160	<35	750	475	980		Ind. Reg.			Hydrostatic	S.S.S.S.	S/S			England	
KKK	2,970	115	<35	1040	134	1450		Eddy Current			Hydrostatic	S.S.S.S.	347 S/S			Germany	

a. Single Suction, Single Stage

b. Double Suction, Single Stage

c. Top Suction, Single Stage

TABLE II. Parameters for Secondary Sodium Pumps

Plant	Capacity, gpm	TDH, ft	NPSH	Design Temp, °F	Motor, hp	Speed, rpm	Driver Type	Speed Control	Shaft, ft	Seal Type	Sodium Bearing Type	Impeller Type	Basic Material	Basic Pump Type	Pump Classification	Supplier	Approx. Hours of Operation
Fermi	13,000	100	<35	1000	350	900	Ind. Squirrel Cage	Eddy Current	17	Double Oil	Hydrostatic	S.S.S.S.	304 S/S	Mech. FS	--	Byron Jackson	27,000
Hallam	7,200	160	17	1000	350	900	ac Induction	Eddy Current	17	Double Oil	Hydrostatic	S.S.S.S.		Mech. FS	Hallam	Byron Jackson	21,000
EBR-II	6,500	140	<35	610	500MG	1800MG	ac Linear Ind.	Elec. Mag.		None	None	None		EM	EM		14,000
SCTI	3,400	280	<35	1200	350	1200	ac Motor H.S.	Eddy Current		Double Oil	Hydrostatic	S.S.S.S.		Mech. FS	Fermi	Allis Chalmers Mod. B. J.	
SRE-PEP	2,500	247	<35	800	200	1800	ac DC	Mag. Coupling		Double Oil	Hydrostatic		S/S	Canned Rotor	Canned Rotor	Byron Jackson	
CANEL	2,000	460	25 - 40	1500	300	3200	ac-0.1 Filled	Var. Speed		Mechanical	None						
Rapsodie	1,675	50	<35	1200	73	875	Ward Leonard System			Double Oil	Hydrostatic	S.S.S.S.	316 S/S	Mech. FS	Fermi	Byron Jackson	>16,000
PFR	15,000	120	69	700	700	960	Var. Speed	Hyd. Coupling			Hydrostatic		S/S		Fermi	England	
BN-350	17,000	230			1470	1000	2 Speed				None				--	Russia	
FFTF	1,000	250	47	1100	900	900	Wound Rotor	Liq. Rheostat	17		Hydrostatic		304 S/S			Byron Jackson	
KNK	647	165	<35	1040	40	2950	ac Induction	Eddy Current			Hydrostatic		321 S/S		Fermi	Germany	

TABLE III. Parameters for Miscellaneous Sodium Pump for Closed Loop Comparison

Plant	Capacity, gpm	TDH, ft	NPSH	Design Temp, °F	Motor, hp	Speed, rpm	Driver Type	Speed Control	Shaft, ft	Seal Type	Sodium Bearing Type	Impeller Type	Basic Material	Basic Pump Type	Pump Classification	Supplier	Approx. Hours of Operation
Fermi O'Flow	300 300	100	Low								Hydrostatic	S.S.M.S. (a)	S/S	Mech. FS	Fermi	Byron Jackson	27,000
CANEL	2,000	610	<35	1400	350	3200	ac Motor-0.1.F	Var. Speed		Mechanical	None			Canned Rotor	Canned Rotor	Byron Jackson	
LASL	500	115	Low	1200	25	1710				Double Oil	Hydrostatic	S.S.M.S.		Mech. FS	Fermi	Byron Jackson	13,474
EBR-I	500	20	Low				dc Electro- magnetic					S.S.M.S.					
FFTF Primary	600	600		1200													
FFTF Secondary	1,000	100- 135		1200													
CCTL	800	320		1200	100	1700				Double Oil	Hydrostatic	S.S.S.S.	S/S	Mech. FS	Hallam	Byron Jackson	3,000

a. Single Suction, Multistage

FIGURE 1. Primary Sodium Pump, Enrico Fermi Atomic Power Plant (EFAPP)

FIGURE 2. *Typical Arrangement with Fermi Pump*

FIGURE 3. Sodium Pumps, Hallam Nuclear Power Facility (HNPF)

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FIGURE 4. Typical Arrangement with Hallam Pump

Both units have a barrel that can be classed as a sump, but the barrel is in communication with the suction of the Fermi types and with the discharge on Hallam types.

3. Evaluation of Operating Experience with Mechanical Centrifugal Pumps

a. Fermi Type Pumps

(1) EFAPP (Enrico Fermi Atomic Power Plant). There are three primary and three secondary sodium pumps in this system. The three primary pumps have two sodium bearings; the secondary pumps have one. The upper end of the primary shaft is attached to the motor shaft by a draw bolt. A double, oil lubricated shaft seal is used in all pumps. The overflow pump is a multi-stage centrifugal with five stages and a low speed, and has straight hydrodynamic bearings and double oil seals.

During the initial start-up of these units, some motor ball-bearing and pump sodium-bearing problems were encountered, along with a severe check valve slam. Since the correction of these problems, the units have operated for nine years without difficulty except for periodic mechanical seal replacement.

(2) EBR-II (Experimental Breeder Reactor - II). The EBR-II has two centrifugal pumps in the primary circuit and one AC linear induction pump (Figure 5a) in the secondary circuit. The primary pumps are vertical free-surface pumps of the Fermi type, with the shaft supported by the motor bearings.

A Hallam-type centrifugal pump was also designed for the secondary circuit, but no operating history is available for it.

The EBR-II primary pumps experienced binding between the pump shaft and lower labyrinth seals, but have operated satisfactorily since corrections were made.

b. Operating Experience with Hallam Type Pumps

(1) HNPF (Hallam Nuclear Power Facility). The Hallam system has three sodium pumps in both the primary and secondary systems. These units contain the same basic seal and bearing

- (a) Cross Section of Annular Linear Induction Pump
- (b) Helical Induction Pump
- (c) 10,000 gpm Sodium Pump Cutaway View
- (d) Alternating-Current Conduction Pump

FIGURE 5. Several Types of EM Pumps

designs as the Fermi units, except a ball thrust bearing supports the upper end of the pump shaft instead of being in the motor. The sump on the Hallam design contains an overflow line.

An initial inability to control the liquid level in the sumps resulted in gas entrainment. This problem was relieved by plugging half of the axial thrust balancing holes in the impeller. The units operated one year after these problems were corrected and were then shut down when the program terminated.

(2) SRE-PEP (Sodium Reactor Experiment -Power Expansion Program). The SRE-PEP has Hallam-type pumps with overflow loops. The overflow loops are undersize which produces a serious, gas entrainment problem. Gas is entrained by high velocity induction from the gas interface.

Elaborate series of baffles and a change in overflow pipe size were used as correction measures, but little was recorded on subsequent operating history. A throttling valve was installed on the overflow line to help reduce the entrainment problem.

4. Comparison of Extrapolation Requirements of Fermi and Hallam Pumps

a. Extent of Extrapolation

Table IV shows the relative extrapolations required for the Fermi and Hallam pumps to meet FFTF initial requirements.

In all respects, the Fermi primary pump most closely matches FFTF requirements and represents the least deviation from the state-of-the-art. This would automatically imply that the greatest reliability for FFTF would be obtained from the Fermi design concept. The extensions of parameters for the Hallam design are 50% or greater in all significant areas including NPSH. Although the Hallam pump required only 17 ft abs NPSH, it was a smaller pump (7200 gpm) and extrapolations show that the same unit would require 40 ft NPSH extrapolated to 13,000 gpm.

TABLE IV. *Primary Heat Transport System Pump Comparisons*

Parameter	Cold Leg			Hot Leg		
	Fermi	Hallam	FFTF	Fermi (a)	Hallam (a)	FFTF
gpm	12,000	7,200	12,000	12,000	7,200	13,000
TDH, ft	360	160	440	360	160	500
NPSH, ft Na abs	35	35 (a)	25	35	35 (b)	22
Design Temp, °F	1,000	1,000	900	1,000	1,000	1050/1200
Design Transient	30°/sec-10	10°/sec-10	35°/sec-10	30°/sec-10	10°/sec-10	35°/sec-10
Net Specific Speed	1,370	1,720	1,240	1,370	1,720	1,260
Net Suction Specific Speed	6,800	6,800 (b)	10,400	6,800	6,800 (b)	12,5000
Shaft Length	18	14	30	18	14	24
Motor, hp and Speed Control	1,000	350	2,000 (d)	1,000	350	2,500 (d)
Speed, rpm	900	900	1,050	900	900	1,110

a. Using same parameters as actual pumps as built.

b. Extrapolated to the same flow and NPSH as Fermi in order to compare on same basis.

c. Design temperature is lower than Fermi or Hallam.

d. Estimated non-overloading brake-horsepower.

The extrapolation of a TDH of 500 ft from the Fermi 360 ft is not an unreasonable one, as already verified by the Phase I Pump Studies. ^(8,9) The development of a single-stage pump for 440 ft TDH is significant to the pump only in regard to strength of the impeller material at elevated temperatures. BN-350 had a rated 360 ft TDH at 14,200 gpm. If it is assumed that the conventional stable sloping curve is required, the TDH at 12,000 gpm may approach 400 ft, and the shut-off head may approach 400 to 450 ft.

b. Pump Shaft Lengths and NPSH

Extrapolations of pump shaft lengths are closely related to NPSH. The greater problem area exists with the NPSH deviation since a reduction of shaft length will accompany a reduction of NPSH, and shaft length is a mechanical problem which can be resolved by usage of a correct number of bearings and by usage of an appropriate shaft diameter.

The Fermi pump had two sodium bearings and no antifriction guide bearing above the mechanical seal. Most other sodium pumps have had one sodium bearing and one antifriction bearing. The Fermi pump had alignment problems because of the rigid coupling used and the clearance of the upper bearing had to be opened to resolve the alignment problem so that it may not in effect be acting as a bearing. From this point of view, it might be argued that the Fermi pump has one effective bearing between the lower motor bearing and the impeller while most other pumps have at least two. Some sodium pumps such as the LASL and Fermi overflow multistage pumps have several sodium bearings of the hydrodynamic type.

It has been shown by study contractors that FFTF pump shafts can be built in lengths longer than that of the Fermi pump by using one sodium bearing near the impeller and one

antifriction bearing above the floor. Study contractors have also suggested that two sodium bearings of the self-aligning type could be mounted with a third antifriction bearing. By placing a combined radial and thrust bearing just below the pump coupling, a flexible coupling can be used which helps relieve alignment problems. In addition, the self-aligning sodium bearings would assist in initial and running alignment. It may be argued that the pads of such self-aligning bearings could be frozen to a stationary position by oxide deposits, but since this would occur after the bearing has already settled to its running dynamic position, no harm has been done.

Extrapolation of a 25 to 30 ft shaft length from the Fermi length of approximately 18 ft is significant in the following areas:

- Number of sodium bearings required
- Critical speeds and vibration
- Thrust
- Fatigue and distortion
- Required tolerances and manufacturing state-of-the-art.

Extrapolation of a 25 to 30 ft shaft for FFTF from a Hallam-type pump has still greater significance than from a Fermi because the Hallam design is limited to a single suction impeller which is more NPSH limited than a double suction impeller. Also, requirements for more than one sodium bearing involve considerably more overflow capacity and excessively large overflow loops.

The difference in shaft length requirements between a Fermi and Hallam design has very small significance when dealing with near atmospheric cover gas pressures. The shaft length may only be appreciably shortened on the Hallam pump by increasing the cover gas pressure. (See Exhibit B)

An increase in diameter of the inlet piping system will decrease the shaft length of both pumps. The significance of this analysis is as follows:

- When considering near atmospheric cover gas pressures, there is no significant difference in shaft length between either concept.
- Suction line ΔP affects both concepts equally. The Hallam concept is affected because of NPSH considerations induced by inlet ΔP changes while the Fermi type is equally affected by sump drawdown caused by ΔP changes.
- The Fermi always has a significantly greater available NPSH and thus provides a greater spread between available and required NPSH to permit some run-out on the pump curve to assist core cooling during accidents.

If a double-ended pipe break in one of the heat transfer loops occurs, it is most desirable, for emergency cooling purposes, not to have the remaining loop pumps NPSH limited. With no such limits, maximum flow from these pumps may be obtained to replace the flow lost from the pump on the ruptured loop. The Hallam pump is NPSH limited and has only 5 ft reserve available to it from the system, some of which will be used up in heat transients. This means the pump will not produce greatly more flow than the design value.

The Fermi pump with double-suction impeller is not NPSH limited. It has 25.8 extra feet available which means each remaining pump will be capable of supplying 40 to 50% more flow for emergency cooling with a pipe rupture on one loop.

To produce an equivalent run-out flow capability for a Hallam pump, the NPSH available would have to be raised by making the shaft up to 20 ft longer than the Fermi shaft, or by increasing the cover gas pressure. Both of these options are undesirable.

c. Other Comparisons

The Fermi type of pumps is least sensitive to NPSH and cavitation problems because the system inlet piping and IHX losses, and velocity head are not charged against the available NPSH. The inlet piping is not attached directly to the pump suction, but to the sump barrel, so that NPSH available is a function only of pressure on the sump fluid interface and the submergence on the pump suction. The problem that arises with the Fermi type design is that the shaft length is governed by the system head losses between the pump barrel inlet and the reactor outlet because the sump level in the pump is located below the reactor fluid level by this factor. This is the limiting factor, not the NPSH, that influences Fermi-type pump shaft lengths.

In the Hallam types, NPSH is the limiting factor on pump shaft length; therefore with the restriction to single-suction impellers in Hallam designs, shaft lengths of larger pumps become comparable between the two types. With the tendency to high-flow velocities, shaft lengths tend to be more favorable to the Fermi types because double suction impellers can be used in Fermi designs with considerably less required NPSH. As pump sizes increase, shaft lengths become more greatly dependent upon NPSH problems than upon inlet system pressure drop.

The Fermi designs do not require an overflow; an initial pump-down of the fluid level equals in height the ΔP in the inlet piping system, and equilibrium is then reached. The remaining submergence level must be high enough to prevent gas entrainment into the pump section, and also to take care of pressure and capacity transients. Normally, suction bells are used on the pump suction to increase submergence on the entrance to the pump. They are also used to reduce the entering fluid to a very low velocity to assist in preventing vortexing types of gas entrainment.

While it might appear that Fermi types of design would produce gas entrainment difficulties, actual experience has shown that gas entrainment in Hallam designs has been much more of a problem, as experienced at both Hallam and SRE-PEP.

Other significant differences between the Hallam and Fermi concepts are in the internal sealing structures. In the Hallam designs, discharge leakage into the sump is controlled by close fitting sealing members between the discharge side of the pump and the upper sump containing the overflow. These sealing members would indicate a more serious distortion problem than that in the Fermi types having no such close fitted members. Because there are three heavy nonsymmetrical piping loads on the Hallam designs and two on the Fermi, the barrel will be subject to greater distortions at high temperatures. If metal-to-metal contact is made between the sealing member and pump barrel, a distortional load can be placed on the pump rotating assembly; whereas, the Fermi designs are not susceptible because contact between the barrel and rotating assembly cannot be made. This may explain why the Hallam designs have had more sodium bearing and labyrinth failures.

The Fermi type designs do not incorporate a sump discharge plenum and are, therefore, not subject to problems arising from this type of design.

Other pump problems are described in Exhibit C.

5. Effect of Primary Pump Location in System

As part of this study an evaluation was made of the effects on the pumps of location in the system. This evaluation was limited to the effects on the pumps-only needs. The conclusions of this study relating to primary pumps at FFTF initial conditions are summarized below under the headings of (a) Pump Shaft Length, (b) Material Mechanical Properties, (c) Pump Reliability, and (d) Total Dynamic Head. The effect of

location on primary pumps at FFTF ultimate conditions is discussed in Section III of this report and the effect on secondary pumps of primary pump location is given in Section IV.

- (a) Pump Shaft Length - For a Fermi-type pump, location in the cold leg of the FFTF system would require extrapolation of shaft length from the proven length of 28 ft to a new length of 30 ft; location in the hot leg would require only a 24 ft long shaft.
- (b) Material Mechanical Properties - As compared to location in the cold leg, hot leg location would increase the operating temperature of the FFTF primary pumps by ~ 300 °F (from ~ 900 to ~ 1200 °F). This results in a 65% loss of allowed material mechanical properties.

This appears not to be desirable for a large dynamic machine which is subject to a much greater number of failure modes than any other component of the system.

- (c) Pump Reliability - When located in the cold leg, the pump operates at lower temperatures and within the range of proven technology. Stress allowables are higher and a greater degree of certainty of reliability is obtained, as well as greater economy. Temperature transients within the pump are less than for hot leg mounting.
- (d) Total Dynamic Head - When the pump is located in the hot leg, the full IHX pressure drop must be added to the pump TDH. This results in an increase of 30 to 45 ft of pump TDH required. The increased head would necessitate use of a larger pump, and would also result in increased costs for operating power from the pump.

6. Conclusions

It is true that concepts such as Hallam may be successfully expanded, but reason dictates that too great parametric

extensions of any one design produces unknowns and less expected reliability. (It is also undesirable to extrapolate the greatly increased overflow and gas entrainment problems.) This is particularly true of hydrodynamic machinery in that turbulent flow is nonpredictable in its behavior. It is irrational to state that separate design features of several designs may be integrated into a single new design and that reliability will not be affected. For this reason, it would appear that FFTF interests are best served by usage of the Fermi pump designs. Fermi designs have operated with minimal difficulties for at least 27,000 hr.⁽¹⁰⁾ Hallam pumps of approximately one-half FFTF size have operated ~21,000 hr.

Fermi pumps incorporate well-proven concepts representing the state-of-the art closest to FFTF requirements. Analyses overwhelmingly favor the Fermi designs in all areas. Reference 9 states that "Because of their size and configuration, the primary sodium pumps at this facility (EFAPP) simulate many requirements anticipated for LMFBR pumps." Reference 10 states that "A universal preference for the mechanical centrifugal pump has been expressed."

7. FFTF Design Parameters

Table V shows conceptual design parameter for FFTF pumps.

TABLE V. Conceptual Design Parameters

Parameter	Primary HTS Coolant	Secondary HTS Coolant	Closed Loop Primary ^(a)	Closed Loop Secondary ^(a)
Capacity, gpm	13,000 ^(b)	13,000 ^(b)	500	500
TDH, ft	500	260	600	75
Max Shutoff Hd, ft	625	325	660	100
Required NPSH in ft Abs.	22	22	25	25
Sodium Design Temp. (Process), °F	900 ^(c)	700	1,200	1,100
Structural Design Steady-State Temperature, °F	1,200	900	1,200	1,100
Upset Design Temperature, °F	1,200	1,100	1,400	1,200
Transient Design Temperature, °F	±35 °F/sec for 10 sec	±35 °F/sec -10 sec	±35 °F/sec -10 sec	±35 °F/sec -10 sec
BHP at Design Point	1,620	880		
Non-Overloading BHP	3,000	1,000 Max	150 Max	75 Max
Design Speed, rpm	1,110	875	1,150-1,750	875-1150
Shaft Length, ft	24	24	~15	~15
Steady-State Design Pressure, psig	200	250	275	100
Upset Design Pressure, psig	250		300	125
Inlet Size, in.	26	18	4	4
Discharge Size, in.	18	18	4	4
	Free Surf.	Free Surf.	Free Surf.	Free Surf.
	Double Suction Sump Phase I	Sump Phase I	Multi- Stage Sump	Multi Stage Sump
Mechanical Seals	Dble.-Oil	Dble.-Oil	Dble.-Oil	Dble.-Oil

TABLE V. (contd)

Parameter	Primary HST Coolant	Secondary HST Coolant	Closed Loop Primary ^(a)	Closed Loop Secondary ^(a)
Sodium Bearings	Hydro- Static	Hydro- Static	Hydro- Static	Hydro- Static
Motor Concept	Wound Rotor	Wound Rotor	Sq. Cage Ind.	Sq. Cage Ind.
Speed Control	Liq. Rheo	Liq. Rheo	EC Coupling	EC Coupling
Stages	1	1	3-5	1-3
Auxiliary Drive	Pony Motor	Pony Motor		
Motor Voltage	2,400	2,400	2,400	2,400
Pony Motor Voltage	480	480	-	-
Vibration-Peak to Peak	1 Mil Maximum	1 Mil Maximum	1 Mil Maximum	1 Mil Maximum
Ball & Roller Bearings	ABEC5 or RBEC5	ABEC5 or RBEC5	ABEC5 or RBEC5	ABEC5 or RBEC5
Cover Gas Pressure, psia Normal	15-16	16.5	20-50	35-40
Pony Motor Flow, gpm	2,000	2,000		
Pony Motor Head, ft	5	5		
Max Shut Off Head Pony Motor, ft	8	8		

a. Closed Loop parameters are not firmly fixed at this time.

b. Final design flow for these pumps may be changed

c. The current steady-state design temperature for these pumps is 1050 °F based upon state-of-technology considerations as determined by the Westinghouse Phase II low capacity pump studies. The actual FFTF process design temperature is 900 °F.

III. PRIMARY SODIUM PUMPS FOR ULTIMATE FFTF CONDITIONS

Most of the text in this report is equally applicable to the FFTF whether the pump be a cold leg pump (900 °F) or a near-term hot leg pump (1050 °F). The state of technology is about the same for either condition.

Long-term or ultimate goals require a pump capability for operation to 1200 °F in the hot leg. For a single stage pump for 500 ft TDH, technology is reasonably established although some consideration must be given until proven unnecessary to the problem of impeller stresses at these temperatures since very little proven technology has been developed for components above 800 °F. In the range of 1050 °F to 1200 °F, no reliable technology exists to make possible a meaningful evaluation. Materials properties are degraded significantly in this range. In addition, the combination of high pressure and higher hot leg transients provide problems in regard to the capability of the heavy wall thicknesses required for the pressure to take the levels of transients which are predicted. A further problem exists in that no bearing hard-facing material has yet been developed or proven at temperatures over 1050 °F. Past sodium pumps have used colmonoy which can be applied as a hard facing material to Types 304 or 316 stainless steel. However, the hardness of colmonoy deteriorates very readily after 1000 °F and its useful limit is placed at this temperature. Stellites apparently retain the desired hardness, but difficulties presently exist in the application to stainless steels because of the differences in thermal expansion coefficients.

The ultimate pump requirements for the FFTF are also perturbed by the possible need to operate against a TDH system higher than 500 ft. The ultimate conditions may be 600 to 700 ft TDH at 1200 °F. A large uncertainty would then exist for impeller integrity in a single stage design. From data

gathered in current design studies, it appears that total impeller stresses for a single suction impeller exceed the allowables for 1200 °F. Centrifugal stresses alone approach the allowable. The double suction impeller design provides more assurance since these same studies show stresses between 1/3 to 1/2 the allowable. These are based, however, on a 440 ft TDH. The change from a cold leg to a hot leg pump increased the system TDH to 500 ft which aggravated this condition by placing a total uncertainty on the single suction impeller and by pushing the centrifugal stresses of the double suction at greater than 1/2 the code allowables. Therefore a 600 to 700 ft TDH would place some greater uncertainty on even the double-suction impeller.

The long term FFTF pump may be a 1200 °F pump for 600 to 700 ft TDH. This combination of operating requirements is well beyond current technology for a single stage pump because impeller stresses caused by the high operating speeds and large diameter will most likely exceed ASME code allowables. The near term 500 ft TDH pump is best suited to a double suction impeller, from the stress standpoint, when restricted to state of technology single stage requirements. Therefore, the next stage of development for a high temperature, high TDH pump would be a multistage pump which has the greatest potential for reducing operating speed and impeller stress while still remaining with proven hydraulic state of technology. BNW has recommended development of such a pump as a backup primary pump for the FFTF

In addition to the backup prototype pump, a long-term sodium bearing development and testing program should be initiated to prove such a bearing for ultimate FFTF operation.

Extrapolations for LMFBR pumps of 20,000 gpm and higher have more serious implications than FFTF requirements, and state-of-the-art considerations for the two programs should be separated.

Further development work for FFTF requirements is needed in the following areas.

- Vortexing and gas entrainment
- Total vibration reduction
- Inertia
- NPSH reduction
- Mechanical seal improvements.

Methods of improving pump efficiency are described in Exhibit D.

IV. SODIUM PUMPS FOR FFTF SECONDARY COOLANT SYSTEM AND FOR CLOSED LOOPS

A. COMPARISON OF FFTF REQUIREMENTS WITH STATE-OF-THE-ART

Table II (See Part II) is an accumulation of operating parameters for secondary sodium pumps, for both U.S. and foreign applications, including conceptual parameters for FFTF.

Table III (See Part II) is an accumulation of operating parameters for smaller types of sodium pumps suitable for closed loop applications, including conceptual parameters for FFTF closed loop pumps.

B. SECONDARY COOLANT SYSTEM PUMPS

1. Fermi-Type Pumps

From Table VI, it is apparent that the Fermi secondary pump design closely matches FFTF requirements, with deviations only in the area of TDH. However, the Fermi primary pump developed a TDH greater than required for FFTF, so the FFTF requirement lies between the two pumps. It is therefore reasonable to assume that the state-of-the-art for FFTF secondary heat transport system pumps is entirely covered by Fermi extrapolations.

2. Hallam-Type Pumps

At Hallam the secondary pumps experienced binding of the rotating elements. The binding was caused by galling wear at the close-running clearances of the impeller wear rings, the sodium bearings, and the impeller rim on the casing and the wear was caused by distortion, close clearances, and metal chips. The pump casings were also thermally distorted.

3. Secondary Pump Seal Requirements

For a design which places the primary pump in the hot leg, a 250 psia mechanical seal must be developed for the secondary

TABLE VI. Secondary Heat Transport System Comparisons

Parameter	Cold Leg or Hot Leg			% Deviation	
	Fermi	Hallam	FFTF	Fermi	Hallam
gpm	13,000	7,200	13,000	0	81
TDH, ft	100	160	260	160	63
NPSH, ft Na abs	<35	17	50	43	194
Design Temp, °F	1,000	1,000	1,100	10	10
Design Transient	30°F/sec ⁻¹⁰	10°F/sec ⁻¹⁰	35°F/sec ⁻¹⁰	17	250
Net Specific Speed	3,210	1,700	1,670	47	2
Net Suction Specific Speed	7,080	8,140	5,410	24	34
Shaft Length, ft	17	17	17	0	0
Motor, hp and Speed Control	350	350	1,000	186	186
Speed, rpm	900	900	900	0	0

pump. This seal is required to permit an increase in the heat exchanger secondary side pressure to a value of 15 to 25 psi above the primary side in order to prevent secondary system contamination in the event of a failure which would cause leakage from one side to the other.

Based on proven technology trends, such a seal should be a double, balanced oil seal. One study contractor has suggested the usage of high pressure gas seals. Gas seals have not acquired proven technology and would require a longer range development and testing program than the near-term FFTF schedule could allow. Double, liquid-lubricated, balanced seals have been used for many years by the pump industry for high-temperature, high-pressure chemical service and are

developed to a good state of technology. Adaptation of this type of seal to sodium pump service appears to be more realistic at this time.

Another method of sealing which has been suggested is the use of a pressurized inert gas containment about the pump driver and driver support. Such a system has several disadvantages. Electrical insulation breaks down in an inert gas atmosphere as demonstrated by the Allis Chalmers pumps at LMEC. Driver power and torque is also reduced. If a mechanical seal is used between the pump cover gas and the driver containment gas, nothing is gained, since the seal would still have to be designed to withstand high pressure in the event of failure of gas on either side. Failure from one direction would cause sodium vapor to enter the driver containment where it would condense and cause driver failure. Failure from the other direction would cause carbon, metal dust, and other undesirable impurities to enter the sodium system. For these reasons, inert gas containment is not recommended.

C. PRIMARY AND SECONDARY CLOSED LOOP PUMPS

1. Closed Loop Primary Pumps

No pump design of the type desired for FFTF (i.e., free-surface multistage centrifugal) approaches a 600 ft TDH as required for primary closed loop service. (Table VII)

A preliminary evaluation of a multistage free-surface pump for the primary closed loop system indicates that a three-or four-stage pump running at 1750 rpm would be most suitable for developing the 600 ft TDH. Pumps with up to five stages have been built and operated in sodium reactor systems with very excellent success. A second preference would be the CANEL pump converted to a vertical free-surface design. It adequately covers all the parameters of the primary closed loop system. The disadvantages of the present CANEL pumps are

TABLE VII. Closed Loop Comparisons Primary and Secondary Pumps

Parameter	Fermi Overflow	Cold or Hot Leg							Deviation Overall
		LASL	CCTL ANL	SRE-PEP	Rapsodie	CANEL	Helicoil	FRTF	
gpm	300	500 max	800	1,200	1,650	2000-4000	2,200	500	0
TDH, ft	100	115	320	80	105	610	100	600	0
NPSH	Low	Low	Low	Low	<35ft	60ft	--	Low	0
Design Temp, °F	1,000	1,200	1,200	1,200	1,200	1,400	600	1,200	0
Shaft Length, ft	40							15-20	0
No. Stages	5	3,4	1	1	1	1	1	1-5	0
Motor, hp		25	100	150	73	300-700		150 max	0
Speed, rpm		1,710	1,700	1075-1800	1,100	3,200		1,750 max	0

that they are all canned rotor types with oil filled motors, with the oil separated from the sodium only by a mechanical seal. The pumps also run at higher speeds than are consistent with good reliability for a 20 yr life.

Helical rotor and electromagnetic pumps may be parametrically extended for FFTF, but the extensions represent a major problem in size, cost, weight, availability, and reliability. The state-of-the-art in these types of pumps is very soft, and much development work is required to resolve all the many consistent failures being experienced.

The apparent most reliable choice for FFTF pumps is a multistage free-surface centrifugal such as the Fermi and LASL pumps.

2. Closed Loop Secondary Pumps

The Rapsodie single-stage pump adequately covers the requirements of the secondary pumps, and no deviation exists here. The Rapsodie pumps are Fermi type. The LASL multistage pump also satisfies the closed loop secondary pump requirements. The preference would be the LASL multistage pump because of its great compactness, simplicity, and lower cost.

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APPENDICES

EXHIBIT "A"

ADDITIONAL INFORMATION ON SODIUM PUMPS

A. ILLUSTRATIONS

Figures A-1 through A-9 show various types of sodium pumps (See also Figures 1-4, Pages 11-14).

B. OPERATING EXPENSES

Data on operating expenses with CCTL, Electromagnetic, LASL, EFAPP Overflow, Canned Return and EBR II Centrifugal Pumps are listed in the subsequent pages.

(a) PFR Prototype Pump
(7,100 gpm)

(b) PFR Primary Pump
(18,000 gpm)

(c) Phenix Primary Pump
(17,800 gpm)

(d) BN-350 Primary Pump
(14,200 gpm)

FIGURE A-1. Foreign Pumps--Prototype and Primary

FIGURE A-2. PFR Double Suction Pump

FIGURE A-3. *PFR Secondary Pump*

FIGURE A-4. Primary Pump for Rapsodie Reactor

FIGURE A-5. *Primary Sodium Pump, Experimental Breeder Reactor-2 (EBR-II)*

FIGURE A-6. *Fermi Sodium Overflow Pump*

FIGURE A-7. Primary Pump for Sodium Test Facility (LASL)

FIGURE A-8. *Main Primary and Secondary Pumps,
Sodium Reactor Experiment (SRE-PEP)*

FIGURE A-9. *Secondary Sodium Pump, Enrico Fermi Atomic Power Plant (EFAPP)*

CCTL PUMP (ARGONNE NATIONAL LABORATORY)

The CCTL pump developed a seal leak shortly after startup. The cause of the failure was attributed to a cocked seal face caused by one drive pin hole not being machined deep enough. After correction, this pump has run satisfactorily at temperatures between 1000 and 1100 °F.

An operating fault encountered with this unit is that sodium vapor condenses or "freezes" at the joint between the upper end of the pump between the seal plate structure and shaft. Eventually the solidified sodium builds up across the gap between the shaft and seal plate structure and interferes causing erratic starting and decay of the pump rotor.

A new problem has developed with the pump overflow line. It is too small to carry away the sodium pumped through leakage paths into the upper tank and the tank is flooding.

ELECTROMAGNETIC PUMPS

Electromagnetic pumps are still in the research stage for sodium service, with no large high-head units having been built or proof-tested. The following operational problems have been found in the units built to date:⁽²⁾

- Loss of pump cooling increases coil resistivities, reduces magnetic material performance, and deteriorates electrical insulation.
- Loss of flow in the pump duct is caused by void formation; damage results from the high current density at the duct-to-bus bar connection.
- A common type of failure in electromagnetic pumps is sudden local temperature excursions caused by high ohmic heating at the joint between the duct and bus-bar induced by fluctuations in the flow. This type of failure is

common with electromagnetic pumps operating at high temperatures and low NPSH, and the usual result is duct leakage.

- Fatigue of the pump duct wall is caused by pressure fluctuations created by rapid variations between internal and external pressures. Because of the thin duct wall existing about the rotor field, electrical and mechanical vibration frequencies also cause fatigue.
- Electromagnetic pumps with rotating magnet structures fail with loss of bearing lubricant.
- Reference (3) states that "So far as ORNL experience is concerned, electromagnetic pumps were unacceptable for the liquid metal applications involved and not applicable to low conductivity salts. The weight-to-capacity ratio is high, and is aggravated not only by the power supply required but also by the low efficiency of the pump."^(a)
- Major objections to this type of pumping equipment include the extremely low efficiency (10 to 20%) and the large, excitation-power and electrode current that would require special generating equipment. Objections also include the large pump dimensions, weight, and cost (as compared to centrifugals), and the extensive development work and cost for pumps of closed loop sizes.⁽⁴⁾

LASL MULTISTAGE CENTRIFUGAL (LOS ALAMOS SODIUM LOOP)

The primary and secondary pumps at LASL are both multi-stage free-surface centrifugals, with the primary having two stages and the secondary having four stages. Both units have hydrodynamic radial sodium bearings and double oil lubricated seals, with the pump suctions in association with the sump.

a. *A primary HTS EM or helical rotor pump would weigh about 20 to 25 tons, while a closed loop unit would weigh 7 to 10 tons.*

The pumps were operated with no problems for approximately 12,000 hr at the end of which, gas lubricated seals were installed in place of the oil lubricated seals. The gas seals failed during the first week of operation, and the tests were discontinued.

EFAPP OVERFLOW

The Fermi overflow pumps are multistage free-surface centrifugals having five stages. They are furnished, also, with hydrodynamic sodium bearings and double oil lubricated seals, with the pump suctions in association with the sump.

The only problem encountered was a seizure, after four years of service, between the shaft and a labyrinth seal. The units have operated with good reliability since correction.

CANNED ROTOR CENTRIFUGAL

The mechanisms of failure are well recorded history, and pumps of this type (Figure A-10) are presently deemed completely unsuitable for high temperature sodium service.

The basic problems lie in the low fatigue life of the thin metal shell membrane separating the rotor and stator in the motor. In units such as CANEL where a mechanical or mercury seal is used to separate the oil filled motor [Figure A-10 (a)] from the sodium in the pump, other areas for failure are evident and need not be explained.

EBR-II SECONDARY PUMPS

The EBR-II secondary linear induction electromagnetic pumps developed fatigue cracks in the pump duct from vibration.

(a) OIL FILLED MOTOR
 Mercury Seal---400HP
 Mech. Seal-----200HP

(b) CANNED MOTOR-Low PR.3
 Full Pr. Liner-Not Supported
 Liquid Sodium 1000 °F

FIGURE A-10. Two Types of Canned Rotor Pumps

EXHIBIT "B"

SHAFT LENGTH AND NPSH EVALUATION

It is commonly assumed that a Hallam type pump may be very short as compared to a Fermi type. This is a misconception, as shown by the following analysis.

1. NPSH and Shaft Length Hallam Design - Cold Leg

- $NPSH = H_p \pm H_z - H_{vp} - H_f - H_h.$
- At 400 ft elevation, $H_p = 14.48$ psia.
- H_{vp} 900 °F = 0 psia.
- $H_f = 5.74$ psi inlet system friction losses.
- $H_h =$ velocity head not charged to pump = 1.4 ft.
- Neglecting submergence, NPSH available = 14.48 psia
- 5.74 psi - 1.4 ft = 8.74 psia - 1.4 ft.
- $8.74 \text{ psia} = \frac{8.74 \times 2.31}{0.8} - 1.4 \text{ ft} = 23.8 \text{ ft abs available.}$

The limitations on a single suction impeller for required NPSH disregarding velocity head are 34 ft. An available NPSH must allow for at least five feet more available than is required to take care of system transients. Therefore, the total available should equal at least 39 ft.

- Then $39 = 23.8 + \text{submergence } H_z$ and $H_z = 39 - 23.8$
= 15.2 ft between reactor fluid level and pump suction.
- The distance from the operating floor to the reactor fluid level is 10 ft.
- The shaft length minimum for a Hallam pump is then
 $10 + 15.2 = 25.2$ or 26 ft abs.

However, it is very undesirable to have the pump suction line under subatmospheric pressure since gas may be entrained through the dip leg arrangement of the overflow line and because any crack or hole developing in the suction pipe will pull ambient gas from the containment into the pump. Therefore, the shaft should be long enough to ensure a submergence that will keep the pump suction line at atmospheric or slightly higher pressure. Fifteen pounds per square inch is probably a safe suction line pressure requirement that will allow for variations of barometric pressure. This is equivalent to 42 ft of sodium absolute at 900 °F.

Then:

$$42 = 23.8 + \text{Submergence } H_z \text{ and } H_z = 42 - 23.8 = 18.2 \text{ ft.}$$

Thus, the shaft length equals $18.2 + 10 = 28.2$ ft.

This value does not allow for uncertainties at the impeller eye caused by nonuniform distribution of velocity across the eye. The nonuniform distribution is induced by high flow velocities in the suction piping, the numbers of elbows required for expansion loops, and the elbow which is normally near the pump inlet.

2. NPSH and Shaft Length Fermi Design - Cold Leg

The available NPSH is taken from the surface level of the sump of the pump rather than the reactor. The pressure here is also 14.48 psia at 400 ft elevation.

- NPSH available is 14.48 + submergence.
- Neglecting submergence NPSH available = 14.48 psia

$$= \frac{14.48 \times 2.31}{0.8} = 41.8 \text{ ft abs.}$$
- Three to four feet of submergence is required on the pump suction to prevent vortexing, so the total available is $41.8 + 4 = 45.8$ ft NPSH available.

- A Fermi double suction pump can be built with a 20 ft required NPSH. By adding five feet for transients, the NPSH available need only be 25 ft. The system is then not NPSH limited as in the case of the Hallam design since the available exceeds the required by 25.8 ft.
- The factor that most influences the shaft length on the Fermi is not NPSH but, instead, the system losses in the inlet piping. These losses are 16.6 ft, and the pump sump level will draw down below the reactor vessel level 16.6 ft. The shaft length then becomes 10 ft + 16.6 ft + 4 ft = 30.6 ft.

3. NPSH and Shaft Length Hallam Design - Hot Leg

- $NPSH = H_p \pm H_z - H_{vp} - H_f - H_h$
- $H_p = 14.48$ psia at 400 ft elevation
- $H_{vp} = 1.03$ psia at 1200 °F
- $H_f = 2$ psig
- $H_h =$ velocity head not changed to pump = 5.4 ft
- Neglecting submergence, NPSH available = 14.48 psia - 1.03 psia - 2 psi - 5.4 ft = 11.45 psia - 5.4 ft
- $11.45 \text{ psia} = \frac{11.74 \times 2.31}{0.8} = 34 \text{ ft}$
- 34 ft - 5.4 ft = 28.6 ft NPSH available.

To obtain atmospheric pressure on the pump suction the submergence H_z should be:

- $H_z = 42 - 28.6 = 13.4 \text{ ft}$
Shaft length is then 13.4 ft + 10 ft = 23.4 ft.

4. NPSH and Shaft Length Fermi Design - Hot Leg

The NPSH available for a hot leg Fermi pump is the same as for the cold leg Fermi pump except for the difference in vapor pressure between the two temperatures.

- $H_{vp} = 1.03 \text{ psia} = 3 \text{ ft sodium abs}$

- $\text{NPSH available} = 45.8 - 3 = 42.8 \text{ psia}$

The shaft length is not determined by the available NPSH however, but by pump inlet line ΔP . For the FFTF hot leg Fermi pump:

- $\Delta P \text{ inlet} = 6 \text{ ft}$

The shaft length must then be: $10 \text{ ft} + 6 \text{ ft} + 4 \text{ ft} = \underline{\underline{20 \text{ ft}}}$.

EXHIBIT "C"MAJOR POTENTIAL PROBLEM AREAS FOR PUMPSA. EVALUATION OF PROBLEMS WITH FERMI PUMPS

The large Fermi pumps have had seal problems which have required replacement of seals every six to nine months. Byron Jackson is replacing the Fermi seals with a modified design. The Fermi pump is different from other pumps in that there is no guide bearing close to the seal. Other pumps which have used guide bearings in the seal area have not experienced the degree of problem which exists at Fermi.

The Fermi pump had two sodium bearings and no antifriction guide bearing above the mechanical seal. (Most other sodium pumps have had one sodium bearing and one antifriction bearing.) The Fermi pump had alignment problems because of the rigid coupling used, and the clearance of the upper bearing had to be opened to resolve the alignment problem so that it may not in effect be acting as a bearing. From this point of view, it might be argued that the Fermi pump has one effective bearing between the lower motor bearing and the impeller while most other pumps have at least two.

The Fermi check valves, as well as some others have experienced check-valve slam problems.

These problems have been corrected by modifying the check valves and installing dash pots.

B. EVALUATION OF PROBLEMS WITH HALLAM-TYPE PUMPS

Because the barrel is associated with the discharge of the Hallam pump types, fluid could rise to the mechanical seals and cause failure. To prevent this, an overflow line is established just below the gas interface to carry away excess fluid. However, such an overflow line is undesirable because it creates additional weight on the pump barrel which can cause distortion.

It is also another source for gas entrainment, as experienced at SRE-PEP, and represents an additional containment problem at the pump.

Hallam-type pumps are also sensitive to gas entrainment problems through the impeller axial thrust balance holes, as experienced in the Hallam installation where the holes had to be plugged to stop entrainment. This is also not desirable for sizes of FFTF pumps since these pumps are much higher head than Hallam and have considerably larger impeller eye diameters. If balance holes cannot be used for FFTF pumps, excessive axial thrust will result, and failure modes in the pump thrust bearings will develop.

Leakage into the sump in Hallam-type pumps comes from the discharge through the hydrostatic bearings, which require a large flow for FFTF pump sizes, and through other internal sealing members. An additional large flow exists at the seal plate because of the annular seal clearance which must be used to avoid metal-to-metal contact. Because the FFTF pumps are much larger in diameter than those previously used, the annular clearance area is large. These two sources of leakage represent a very appreciable amount so that the overflow line must be sized very large (24 to 30 in.) to reduce line velocity to minimize gas entrainment. To stop gas entrainment completely, a fluid submergence should be established above the overflow line, but this is very difficult to accomplish or control.

Because Hallam-type pumps are directly piped from the pump suction to the inlet piping from the Immediate Heat Exchanger (IHX) or reactor, the pumps are quite sensitive to suction line velocities and inlet pipe sizing caused by NPSH implications. The entire piping and IHX losses, and the velocity head must be charged against the system available NPSH. The Hallam types are also limited to single suction impellers

requiring the highest NPSH. Therefore, where critical suction conditions exist, use of the Hallam pumps is not desirable.

In installations where the primary pump is to be mounted in the cold leg location, the NPSH and cavitation problems become increasingly difficult as reactor sizes increase. In the cold leg position, the pump is mounted with the IHX as part of the pump inlet system. As reactors get larger, flow velocities will be increased to minimize pipe sizes and piping problems. IHX's will also tend to increase flow velocities to minimize size. The net result would be an increase of friction losses in the pump inlet piping system which, in turn, would mean less available NPSH to a Hallam-type design. At the same time, the required NPSH of the Hallam design would be increasing so that critical situations would always exist. Such situations are only resolved by very long shafts or higher cover gas pressures, neither of which is desirable.

The practical limitation to single-suction impellers for Hallam pumps also aggravates the overall pump problems in that an inherent characteristic of these impellers in radial types is an unstable head curve. This curve droops to shut off head and makes them unsuitable for closed loop or parallel service.

Gas entrainment problems occurred with the initial testing of the Hallam pump. When the flow exceeded 5300 gpm⁽²⁰⁾ erratic overflow developed and cover gas was drawn into the pump suction pressure than was estimated. The overflow line attachment was relocated to the pump suction line with no evident improvement. The submergence level on the overflow line in the pump tank was increased until the level was sufficiently high to prevent vortexing in the overflow line entrance.

When the Hallam primary pumps were run in their final position in the system, gas entrainment problems again occurred. The system installation also had a lower suction line pressure than anticipated. Level control of all pumps became erratically

low because of a system head. The overflow line submergence could not be held, and when the tank level dropped, gas was entrained through probably both the overflow line and the impeller balance holes. The vacuum on the pump suction line pulled gas into the impeller eye from the tank sodium-gas interface. Level control was also difficult because the system resistance was much lower than anticipated so that the pressure on the pump internal leakage paths restricted flow to the point where flow into the tank was less than backflow through the impeller balance holes. Plugging of half of the eight balance holes was chosen as a method of balancing internal flow. However, plugging of the balancing holes increased the pump axial thrust. The operating rangeability of the pump was probably affected also. Operation at reduced speed could upset the internal flow balance by further reduction of the pressure on the leakage joints and gas entrainment could result. AI engineers have reported that it was never resolved that gas entrainment was completely eliminated, but that if any existed, it was at a sufficiently low level to cause no appreciable problems.

The SRE experience was similar to that of Hallam. Gas entrainment was discovered when visual observations through a corescope revealed gas bubble eruptions in the reactor sodium surface. As flow rates increased, the entrainment became more severe and resulted in large flow oscillations.⁽¹⁾ The cause of the gas entrainment was attributed to a combination of a low sodium level in the pump tank and high velocity in the overflow line. The lack of submergence on the overflow line permitted gas entrainment, and the high line velocity did not permit the gas to separate. In addition, the falling of the high velocity sodium into the dip leg entrained additional gas. To produce a submergence on the overflow line, a throttling valve was installed and throttled until the submergence was at a high enough level to prevent vortexing.

This valve had to be very carefully controlled, and the pump was not able to operate off its empirically set operating point so that rangeability of the SRE pump was also completely lost. These problems have led AI engineers to state⁽¹⁾ that such concepts are not recommended for future reactors.

These problems are basically the result of the concept. A dip leg concept, such as one that requires returning the overflow back to the pump suction, is vulnerable in several ways. When fluid is to flow into a pump suction or overflow line below a liquid-gas interface, submergence must be accomplished at a level dependent upon the line velocity. The Hydraulic Institute has established relationships between submergence and line velocity to prevent gas vortexing. At 1 to 2 ft/sec entrance velocity, a submergence of 3 to 4 ft of fluid is suggested. This is more difficult to achieve on an overflow line than on a pump suction since the overflow line must make its pump tank connection considerably higher than the pump suction. Also, suction bells may be attached to the pump suction to decrease entrance velocities to a low level and to increase submergence. This cannot be accomplished on the overflow line.

In addition, the pump concept requiring an overflow line has a piped suction, which at low cover gas pressures is most likely to operate at subatmospheric pressures as did both Hallam and SRE. This vacuum on the suction line will induce gas through a fluid column if a fluid-gas interface exists, which is what occurred at Hallam on preoperating tests as indicated by the report that the suction pressure was lower than expected. This phenomenon is quite common with water well point systems if they are not buried to a depth somewhat deeper than the number of feet equivalent to the line vacuum produced. Consequently, when the sodium level in the pump tank became low enough, suction line vacuum pulled gas through the impeller

balance holes. Raising of the submergence level in the tank provided the correction for both the suction line vacuum problem and the overflow line submergence problem. The SRE pump was small, and the amount of vacuum induced by the impeller eye was less than that of the Hallam pump. As pumps increase in size, the problem becomes more severe, and shaft length of piped suction pumps must be increased to permit both larger overflow lines to decrease overflow line velocity, to supply proper submergence to the overflow line, and to decrease suction line vacuum by increasing submergence on the pump suction. Pump suction line vacuum is also undesirable from other viewpoints since small cracks or holes that might develop in the suction pipe will permit entrance of ambient containment gas into the pump suction. Such cracks are not easily detected because the vacuum can prevent sodium from leaking out, and the pump and core can be gas bound before the problem is discovered. An additional hazard exists if there is any air or water in the containment atmosphere.

For the FFTF pump, the overflow capacity, obtained by the high pump head, for a Hallam-type design would approach 4000 gpm. An overflow line sized to keep the line velocity down to 1 to 2 ft/sec, as suggested by the Hydraulic Institute, would require a diameter of 30 to 42 in. and a submergence of 3 ft to ensure against gas entrainment.

A medium capacity LMFBR pump may require overflow lines as large as 36 to 54 in., which would take care of the overflow line only, but would not resolve the problem of pump suction vacuum. Only shaft length or high cover gas pressure will resolve the vacuum problem. This resolution would result in very high costs for installation of dip leg type overflow lines and would be very impractical from other considerations. From the pump tank stress and distortion viewpoint, such an extra

large nozzle, which is nearly 50% of the pump tank ID is most undesirable for high temperature service.

One study contractor has suggested that the overflow line problem can be resolved by limiting a piped suction pump to hot leg installation and piping the overflow back to the reactor. He suggests that the overflow should run partially full. The size of such an overflow pipe would be much larger than the dip leg requirements, since reliable partial flow could only be ensured with 60 in. or larger piping. This really places undue hardships on the pump tank and reactor vessel, both of which would now require these additional large penetrations. In addition, the thermal mixing problems would be severe at the reactor outlet plenum.

When the reasons are evaluated as to why overflow concepts are proposed (mainly to provide what is assumed to be better level control in the pump tank) it becomes obvious that the desired level control capability has not been achieved in those installations where this has been used. In fact, more severe problems have been created. The sump type Fermi pumps have not had level control or gas entrainment problems. Further realistic evaluations revealed that the need for overflow line level control is created by the overflow line itself, and that level fluctuations in pump tanks are not a problem when the tank is properly baffled. In Phase II of the FFTF pump study, it was discovered that the pump tank wall was essentially at the same temperature as the sodium when argon cover gas was used. Thus, level fluctuations did not cause thermal transients as originally thought.

Therefore the overflow line piped suction concept:

- Created its own level-control and gas-entrainment problems
- Limited the pump concept to one not particularly desirable for large capacity-high head-high temperature pumps
- Created appreciably high pump installation costs

- Created severe stress problems for the pump tank and reactor vessel
- Results in undesirably high operating costs by virtue of the poor pump efficiency caused by pump internal leakage losses.

These are all very poor trade-offs for some presumed gain that cannot even be defined.

One of the major problem areas existing in the Hallam types of pumps is in the discharge sump or plenum. This discharge plenum will be subject to full pump discharge pressure. In the case of the FFTF pumps and LMFBR large sodium pumps, this discharge pressure will be significantly higher than has been experienced in past Hallam designs.

In addition, the plenum size will be quite large because of the large diameter of the impeller and diffuser ring required to achieve the high TDH of the pump. Thus, the large size of the plenum and the high internal pressure will produce high temperature stress problems and abnormal distortion problems which will result in a decrease of expected reliability.

Another possible problem with Hallam pumps is that the pump inlet may be under subatmospheric pressure. The Hallam design is particularly susceptible to this because it has a lower available NPSH for the same shaft length than the Fermi, and because the pump suction is attached directly to the system inlet piping. If a crack develops in the inlet piping, the subatmospheric pressure will pull air into the sodium system. Since sodium will not leak out in operation, the crack and ensuing damages will not be readily detected. The inlet piping on the Fermi design is not attached to the pump suction and is always under a positive pressure. The Fermi design also has an available NPSH of 43 ft of sodium which is slightly above atmospheric pressure. To give a Hallam pump a positive

pressure on the inlet piping, the cover gas pressure would need to be increased or the pump shaft would need to be increased to at least 35 ft.

C. GENERAL PROBLEMS

The major potential problem areas for FFTF consideration would be:

- Reduction of NPSH and cavitation
- TDH available in a single-stage impeller
- Vortexing and gas entrainment
- Inertia of pump and motor rotors
- Reduction of vibration levels in both the pump structure and fluid
- Improvement of mechanical seals, bearings, and impellers.

1. NPSH and Cavitation

Because cavitation (incipient) problems tend to increase as pump sizes increase and because it may be one of the most damaging phenomena to large sodium systems, an approach to NPSH reduction must be undertaken which is not solely involved with old state-of-the-art concepts such as specific speed, suction specific speed, and Thoma coefficients. Present state-of-the-art covers means of reducing cavitation and **required** NPSH below that predicted by the above parameters.

There is a relationship between low blading inlet angles and low-meridional and low-relative velocities as they affect cavitation and NPSH reduction. Because these relationships imply larger eye diameters and wider impeller entrance widths, they should be evaluated against other hydraulic design factors such as stability of head-capacity curves, head generated per stage, entrance vortexing and turbulent recirculation in impeller entrance for single radial, single Francis, and double suction radial and Francis vanes. The study will probably include mixed-flow impellers.

In general, the ratio of eye diameter to impeller outside diameter (OD) is an important hydraulic criterion which affects the diameter of impeller required (head coefficient), surge and pulsation characteristics, and efficiency.

It is particularly essential that cavitation be avoided entirely in large high-temperature sodium pumps because sound or vibration tends to be at an increased magnitude and will thus fatigue pump and systems components more rapidly.

Modern state-of-technology studies (11-12) indicate a conservative reduction of NPSH required by large pumps of at least 8 to 10 feet abs over that indicated by old conservative parameters.

Empirically, radical departures from NPSH expectations may be obtained through usage of old parameters, because results are too greatly determined by empirical skills of independent designers. Therefore, general evaluation of all the latest empirical knowledge is essential so that a general set of rules may be established to guide any designer of large sodium pumps.

2. Total Dynamic Head

Series 300 stainless steels in high-temperature service exhibit greatly altered strength, creep, and fatigue properties. Because offered pump concepts for high capacity, high head, high-temperature sodium pumps are all single stage, greater impeller diameters and weights are required than have previously been used. This, in turn, implies greater unbalances and higher centrifugal forces. A conclusion can then be reached that life expectancy (reliability) will be decreased. Studies that are indicated to improve reliability are an evaluation of creep and fatigue life versus impeller diameter for high-temperature sodium service to 1200 °F for single-stage heads to 500 feet TDH, the homogeneity of corrosion and/or scaling in single suction and

double suction impellers, and the effect on increase of dynamic unbalance with time. The latter study should also relate to:

- The effect of increased vibration on fatigue life
- What initial balancing requirements will be
- At what time of operation (life) will critical vibrations develop.

Critical vibration levels will need to be determined to develop safety maintenance and also to determine the feasibility of the designs offered.

It is expected that the more symmetrically designed double suction impeller will represent a lesser problem.

3. Vortexing

All the pump designs offered are susceptible to vortexing problems, whether single or double suction. Single suction impellers, of necessity, must have higher impeller inlet velocities that can be obtained in double suction impellers. In addition, the single suction designs have balancing mechanisms that can induce vortexing through the fluid level in the pump column to either the impeller suction or volute channel.

The top inlet of the double suction impeller has an inherently greater capability for gas entrainment through vortexing than the single suction. However, this top inlet may be designed with 50% or less impeller inlet velocities over a single suction design without materially reducing hydraulic parameters. This implies that a lower submergence may be considered than for a normal single or double suction design. The following studies are indicated:

- Minimum submergence requirements for single suction and double suction impellers to prevent vortexing.

- Features that will prevent vortexing from the pump column through the axial balancing means on single suction pumps.
- If pressurizing of the balancing chamber is required, thrust increases must be defined along with the effect on seal and bearing life.
- Methods to be used to prevent vortexing with low submergence. These studies should relate to design of sumps, bell entrances, vortex breakers, surface floats, etc.

4. Inertia

System fluid-hammer, pressure rise, and valve slam in the new concept large sodium pumps will be potentially more serious than in previous systems. Potential problems could arise partly from an increase of flow energy caused by the system size and higher TDH, and partly from the reduction in size and inertia of present-day motors. The reduction in size and inertia of present-day motors is made possible by the usage of new insulating materials permitting high-temperature rises. A qualified review should be made of pump decay time as compared to system decay time, and of valve response as compared with system response. The use of thin-walled piping, high operating temperatures, and low inertia motor rotors cause potent vulnerability from either reverse pump flow or check valve slam.

A study of inertia requirements must also be concerned with temperature decay transients and with inertia evaluations, taking into account both temperature and system pressure decay.

5. Vibration Problems

a. Shaft Length Versus Number of Bearings

Because of the relatively large bearing clearances and long length of proposed shafting, a possibility of conical pendulum oscillation exists which may dictate the usage of one or more additional "stiff" hydrostatic bearings.

An evaluation should be made of this phenomenon as regards length of shaft versus number of bearings.

No critical speeds will be permissible in the entire range of operational speeds. Pump study contractors often evaluate critical speeds based on purely theoretical concepts and may fail to include:

- The "softening" effect of the motion of the sodium in the sump
- The relative "stiffness" of the bearings in the structure; particularly, the lower hydrostatic bearing with its wide clearances
- The influence of dynamic unbalance
- The influence of radial thrust with different casing concepts
- The influence of numbers of bearings versus shaft length, and the stabilizing influence of gyroscopic effects
- The influence of creep and unbalance caused by corrosion and/or scaling with operational time.

Conclusions to be obtained from these studies must define maximum shaft lengths for an entire range of LMFBR pumps versus the number of pump shaft bearings, and the point in operational life at which effects of creep and corrosion might produce criticals in the range of pump operation for various pump sizes.

It is possible that these studies might show a limitation in sizes of large vertical pumps and that, also, other pump concepts may need to be derived and evaluated.

b. Vibration Versus Fluid-Borne Noise

In general, the amplitude of structure-borne and fluid-borne noise or vibration increases with the size of pump and pump TDH. Because of the low operating speeds used for large pumps, the spectrum will be composed of high amplitude,

low-to-medium frequency vibrations in the structure and fluid. All circuit components will absorb these fluid frequencies and, thus, may create serious fatigue problems.

The frequency of fluid-borne vibrations is a direct function of the operating speed, number of impeller blades, number of diffuser blades and/or number of volute tongues. The amplitude of fluid-borne frequencies is largely related to the clearance between the impeller periphery and the diffuser entrance tips or volute tongue. It is also related to the dynamic balance of the rotor.

Since the lower frequencies will be more critical to fatigue because of their inherently greater amplitudes, an evaluation of methods to reduce fluid-borne noise in high capacity sodium pumps operating at low rotational speeds is indicated. The following paragraphs suggest specific criteria for such an evaluation

Blade and balance frequencies should be evaluated against motor frequencies to ensure against compounding effects. For instance, a pump with six vanes operating at 1200 rpm will have a blade frequency of 120 cps. Magnetic frequencies in the motor may also be of the order of 120 cps. Rotational vane frequency will be 20 cps, the third harmonic of which is 60 cps--also an electrical motor frequency. Impellers with six vanes should be avoided. Past experience with ultraquiet pumps indicates that seven blades in the impeller may be optimum.

An evaluation should be made of optimum clearances and the effect that larger clearances will have on overall efficiency. Perhaps a vaned diffuser with a vaneless diffuser section would be indicated. Amplitude rises as the volute tongue clearance or diffuser clearance with the impeller OD is reduced.

If diffusers are used, the evaluation should concern itself with both first- and second-order vibrations as related to the number of blades in both diffuser and impeller.

Studies of types of impellers and their relationship to vibration are suggested. A double suction impeller design may lend itself more ideally to reduction of fluid-borne and structure-borne vibrations. Again, the ratio of the impeller eye to impeller outside diameter (OD) is a significant parameter. Large impeller eyes tend to produce more vibration and fluid-borne noise because of higher entrance velocities and hydraulic transients in the eye. The single suction impeller will have a greater tendency to incipient cavitation which produces high amplitude vibration.

It may also be possible to stagger the blading in each half of a double suction impeller in a manner which will produce some destructive interference in the fluid-borne noise or vibrations.

Vibration from all sources, including drivers, should be evaluated in large sodium pumps, since they will have a very pronounced effect on the life of all components in a high-temperature system.

c. Flow Velocity and Vibration

Another departure from the state-of-the-art is in the piping sizes and flow line velocities used in the conceptual FFTF system. The effects on the inlet piping have already been described.

Reference 13 indicates that vibration induced fatigue from flow of coolant in loops accounts for 29.1% of failure cases. Considering that flow velocities from 5 to 15 ft have been conventional, any departures from these flow velocities must involve an even higher incidence of failure. Some FFTF systems will have line velocities up to 22 ft/sec. Since the recorded failures have primarily occurred in temperature ranges of 300 to 1000 °F, the elevation to 1200 °F will aggravate this problem still further. An incidence rate of 29.1%

failure is already too high, so vibration must be reduced. The only means of reduction available are in production of lower balance levels in the pump rotating assembly, reduction of fluid-borne vibration produced by the pump impeller, and in reduction of line velocities in the piping.

6. Component Parts

It is desirable to define optimum types of seals, bearings, and impellers that produce the best reliability, and after their selection, endurance tests are desirable.

Seals should be tested extensively in the atmosphere and the other ambient conditions in which they will be used. Leakage should be measured as a function of time; face wear should be measured periodically as well as the condition of other seal parts such as secondary sealing members. Life expectancy plots should be developed from these tests. Bearings should also be tested under the simulated conditions in which they will operate. Impellers should be rotated under simulated conditions that will compare vibration with time of service, submergence required to prevent vortexing, cavitation margins, creep and fatigue properties, etc.

A long shaft of the length and design proposed for usage should be set up under all simulated operating conditions to determine critical speeds, oscillatory effects, creep effects, etc. The shaft can be weighted on the end to simulate radial- and axial-thrust and unbalance loads.

a. Universal Usage

Universally accepted concepts for the usage of critical parts for sodium pumps are as follows: (5,6,8,9,11,15,16)

- Hydrostatic sodium pump bearings
- Oil lubricated, double, mechanical shaft seals
- Oil lubricated motor ball bearings

- Hard facing of wearing parts: to 1000 °F - Colmonoy 5 and 6; to 1200 °F - Stellite 6B.

The most successful driver and speed control systems used in the U.S. for large sodium pumps for ^(8,9,17,18) motor sizes of 1000 hp or over is the wound rotor motor with liquid rheostat. Smaller motor sizes are induction squirrel cage with eddy-current couplings.

References 8, 9, 10, 18 and 19 state that the wound rotor motor with liquid rheostat is the most preferred system for sodium pumps and represents the greatest liability at the least installed cost. References 3, 10, and 19 give many instances of eddy current coupling failure. Reference 3 shows that the maximum number of pump failures occurred with eddy current couplings. Thirty failures are reported with an incidence rate of 5,600 hr per accident. Motor bearing failures accounted for 17 accidents at an incidence rate of 9,900 hr per failure, and pump sodium bearings showed 2 failures in 290,000 hr at an incidence rate of 145,000 hr per failure. No evidence of mechanical seal failure is shown.

This evidence is supported by experience with large reactor systems where there are more recorded failures of motor bearings (Fermi) and eddy current couplings than any other pump failure except the mechanical seals at Fermi.

7. Parametric Extensions of Component Parts for Pumps

Usage of component parts, as listed above represents no parametric extensions, and FFTF pumps should not depart from these proven concepts. The usage of eddy current couplings for 1500 hp primary and 900 hp secondary pumps would represent an appreciable departure from the state-of-the-art. It would also be in direct disagreement with LMFBR Pump Study contractors and all other experts in the sodium pump field. The base concept for primary- and secondary-main heat transport pumps should be wound rotor motors with liquid rheostat.

However, there is some justification for usage of eddy current couplings for closed loop pumps because state-of-the-art covers couplings for these sizes of motors and provides greater speed control ratios.

In general, component parts that may present problem areas are as follows:

- Hard Facing for Bearing Journals

Ultimate FFTF operation for a hot leg pump will require hard-face materials which will give extended life in 1200 °F sodium. To date, colmonoys have been used successfully to temperatures of 1050 °F. However, Phase I Pump Study contractors have pointed out that colmonoys are not satisfactory for temperatures exceeding 1050 °F and that stellites will have to be used. Stellites have a different coefficient of expansion than the base metal 300 series SS which they must be applied to, and techniques will have to be developed for stellite facing of these materials.

- Seal Problems

The large Fermi pumps have had seal problems which have required replacement of seals every 6 to 9 months. Byron Jackson is replacing the Fermi seals with a modified design. The Fermi pump is different from other pumps in that there is no guide bearing close to the seal. Other pumps which have used guide bearings in the seal area have not experienced the degree of problem which exists at Fermi. It is believed that the FFTF pumps should have a radial antifriction bearing at the mechanical seal to inhibit run-out and wobble.

In addition, the hot leg mounting of the primary pump perturbs the secondary pump as previously described and necessitates the use of a high pressure (250 psi) mechanical seal. Such a high pressure seal should be of

the balanced double oil lubricated type similar to that used in commercial high temperature, high pressure chemical pumps. Such a seal has not yet been adapted for sodium service and long term developmental testing will be required.

- Pump Axial Thrusts

The rangeability requirements (700 Ns to 3,000 Ns) of the FFTF pumps mean that the pumps will operate at both ends of the pump curve where axial and radial thrusts will be greatest. ORNL data (Reference 3) shows that motor bearings have had the second highest incidence of failure rate in sodium pumps. Fermi had motor bearing change-outs during the shakedown period. Hallam had a potential for motor bearing failure when the pump axial thrust balancing holes were plugged, but did not run long enough after correction to evaluate. The pump concept which has the best potential for reducing thrusts in a high TDH pump such as the FFTF is a double suction impeller. The only thrust unbalanced area in a double suction pump is the shaft area of the lower eye. The thrust on this area is upward and opposes the downward thrust of the rotating assembly due to its weight.

- Pump Shaft Length and Numbers of Bearings

The FFTF pump shaft length for a cold leg pump is 30 ft and 24 ft for a hot leg pump. Because of the high TDH, relatively high speed, high thrust loads, elevated temperature, and other factors relating to these pump shafts, evaluations of shaft length versus numbers of bearings must be undertaken in extensive detail in order to ensure against critical speeds in the entire range of pump operation. The Fermi pump initially had two sodium bearings for an 18 ft shaft length and no bearing at the

pump seal. Seal problems developed that are not yet resolved. It appears that one sodium bearing and one antifriction bearing at the shaft seal are the minimum bearing requirements that the FFTF pump can accept. With this configuration, the FFTF shaft must be very large in diameter and very heavy, but this aggravates the axial thrust problem. In addition, the pump seal gets very much larger than the Fermi seal, and with higher pump speed the face rubbing pressure and the speed become higher. Perhaps a configuration of two sodium bearings and one antifriction bearing is required to relieve these problems. If so, the two sodium bearings would present problems in shaft alignment as experienced at Fermi.

- Eddy Current Couplings

No liquid cooled eddy current coupling has yet been used in a sodium pump. Very few have been used in commercial practice. Eddy current couplings which have been used for sodium pumps have been gas cooled and limited to 350 hp. FFTF motor sizes will be 1,000 to 2,500 hp. The slip which is natural to this type of device causes generation of a considerable amount of heat in these larger sizes. This heat would be exhausted to the containment atmosphere unless liquid cooling is used. Liquid cooling provides several modes of failure in the mechanical seals which must be added, location of bearings below or above these seals, auxiliary equipment such as oil pumps, etc. ORNL reports in Reference 3 that eddy current couplings used for sodium pumps have given the highest incidence of failure of any component part. Two current users of large liquid cooled couplings have experienced brush, seal, and bearing failures, and had occasions in which the couplings locked and would not start.

D. MAJOR POTENTIAL PROBLEM AREAS FOR PUMP COMPONENT PARTS

- Hard facing materials for 1200 °F need to be evaluated.
- Mechanical seals require improvement.
- A high-pressure secondary pump seal needs to be developed.
- Reduction of pump axial thrust needs to be achieved to relieve the motor bearing problems. (The logical solution is the usage of double-suction impellers.)
- Pump shafting lengths and number of bearings require careful consideration.
- Eddy current couplings need state-of-the-art improvements.

EXHIBIT "D"IMPROVING PUMP EFFICIENCY

Because of the numbers of large pumps to be used in large power generating sodium systems for reactors, the operating costs will be significantly influenced by the overall pump and motor efficiencies. Modern state-of-the-art indicates that higher efficiencies than have been offered might be obtained. (Similar pumps are operating in water service in ranges of 85 to 80% overall efficiency, while peaks of up to 81% "hydraulic" have been offered.)

The efficiencies offered by pump study contractors are again related to older specific speed parameters. A. J. Stepanoff ⁽¹²⁾ offers a departure from specific speed parameters, and supports a statement that peak efficiency may be constant for all specific speeds. Modern state-of-the-art designs do tend to verify this since axial flow pumps in specific speed ranges of 8,000 to 12,000, for instance, have been built with efficiencies comparable to pumps in specific speed ranges of 2,500 to 3,000. Again, obtaining high efficiencies in pumps is largely a function of the individual empirical skill of the designer.

While Stepanoff does establish some general criteria as to blading and impeller geometry to produce high efficiencies, there is a lack of definition of all parameters. Different designers use different types of blade curvature and obtain varying results. Casing geometry is more significant than generally realized. Improvements affecting NPSH reduction usually lead to efficiency improvement.

These factors need to be evaluated and general criteria established which will make the design of large sodium pumps consistent in expectancy, and efficiency less dependent upon

individual skills. This evaluation should include the latest state-of-technology, including some of the following parameters:

- The types of blade curvatures that result in the best efficiency
- Optimum blade entrance and exit angles
- The influence of improved inlet characteristics on efficiency
- The optimum relationship of impeller eye diameter to impeller outside diameter (OD)
- The true efficiency potentials between single- and double-suction and radial- and Francis-vane impellers for large low NPSH pumps
- Types of casings that will best produce expected efficiencies such as diffuser ring concentric, diffuser ring volute, straight volute--constant (no diffusion) and decreasing (more diffusion) velocity-- and concentric casings with no diffuser
- Relationship of magnitude of meridional and relative velocities and effects of linear increase or decrease through impeller
- Value of scalloped shrouds to decrease disk friction (This will also be valuable in studying maximum impeller diameters)
- Other factors such as finish of surfaces in impeller and volute, and/or coatings to reduce friction
- How internal circulation through wearing ring and bearing clearances may be reduced.

There is reason to believe that double suction designs have greater efficiency potential than single suction designs by virtue of the following:

- Better entrance characteristics
- More head developed per stage, or less diameter required for a fixed head.

- Better eye diameter to impeller outside diameter (OD) ratios
- Less axial loading of bearings and seal faces, as well as smaller axial thrust bearing requirements.

A. CURVE CHARACTERISTICS

The curve characteristics of large single-suction, low NPSH impellers tend to be unstable to quite flat, at best. This is partly caused by the less favorable ratio of impeller eye diameter to impeller OD, and the consequent lesser vane overlap which leads to poorer fluid support and more circulatory flow in the impeller channels.

These types of unstable-to-flat curves, whether single or parallel, are not desirable in large sodium pump loops. Mixed flow impellers are excluded from this assumption except that, while they have steep curves, they tend to have dips and unstable regions in the curves. Francis vanes are more ideal than straight radials. Figure D-1 demonstrates the general curve characteristics for the types of impellers discussed.

Pump studies should evaluate design methods to be used for ensuring stable steep curve characteristics in single suction impeller pump designs.

FIGURE D-1. Performance Characteristics of Vertical Pumps

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