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ELECTRO-MECHANICAL DIVISION
CHESWICK, PENNSYLVANIA

CONCEPTUAL DESIGN STUDY

LIQUID-METAL PUMPS FOR
LARGE-SCALE BREEDER REACTORS
(Hot Leg Prototype Pump)

ENGINEERING MEMORANDUM 4958

General Electric Contract 67001

WEMD Shop Order No. 4Q454

November 1976

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ABSTRACT

A limited time/funding study was performed to extrapolate the knowledge gained during the 127,000 gpm LSBR cold leg pump study to a 77,000 gpm hot leg pump application.

Two conceptual designs are presented with the recommendation that the spherical casing diffuser pump, with an inducer/impeller, be used as the preferred design. However, further analysis is required to verify the acceptability of this concept for the specified thermal transients. The alternate design, which is heavier, larger, and more costly, would meet all thermal transient requirements, based on the extensive analysis and experience with the FFTF pump.

For either hydraulic concept, a most expeditious development effort is recommended to verify the inducer/impeller design.

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I. INTRODUCTION

In Engineering Memorandum (E.M.) 4858 dated July, 1976, EMD presented a conceptual design for a 127,000 gpm, 500 feet sodium head, cold leg pump. The design temperature was 850°F and the impeller centerline was specified at 40 feet below the sodium level, which provides 20 feet available NPSH. Although EMD considered that the pump concept presented in E.M. 4858 could be fabricated and operated successfully (based on current technology), it was recommended that additional analysis and/or key feature development studies be performed to achieve a higher level of confidence in the design.

It was realized that, while the developed cold leg pump concept is technically feasible, it is economically undesirable; therefore, G.E. issued Amendment No. 1 to Contract 67001 for the purpose of extrapolating the cold leg design (E.M. 4858) to a 77,000 gpm, 500 feet head, 950°F hot leg pump.

The study for this (LSBR) hot leg pump was performed on a limited time/funding basis in recognition of a similar study being performed for the Prototype Large Breeder Reactor (PLBR) hot leg pump for the Westinghouse Advanced Reactors Division (WARD). The initial results of the PLBR hot leg pump study were presented in EMD Engineering Memorandum (E.M.) 4911.

E.M. 4911 recommended (as did the LSBR cold leg report, E.M. 4858) a volute type hydraulics, with certain qualifications. Subsequent PLBR/LSBR studies showed that the volute concept presented in E.M. 4911 (Drawing 900J786) could not meet the specified thermal transient requirements and the hydraulic end was redesigned, as depicted in this report.

This report presents a preferred hydraulic design designated as the spherical casing design (Figure 1) and an alternate designated as the cylindrical casing design (Figure 2). The mechanical end of the pump (i.e., shaft, bearing, etc.) is identical to that presented in the WARD PLBR study (E.M. 4911, copies of which are also attached to be used in conjunction with this E.M.) with the exception that, at the upper end, the tank shot cavity is eliminated by the reduction of the casing diameter.

Figure 1
LSBR Hot Leg Pump
Spherical Casing

Figure 2
LSBR Hot Leg Pump
Cylindrical Casing

The spherical casing is the preferred concept and has a bottom suction and side discharge. However, further analysis is required to verify its acceptability for the specified thermal transients. The cylindrical casing design represents a more conservative approach and EMD has a very high degree of confidence that this concept would meet all thermal transient requirements. This statement is based on the extensive analysis and experience with the FFTF pump.

EMD will continue to analyze and modify the preferred (spherical casing) design during a Phase 2 contract to be placed by WARD in December, 1976. It is believed that this preferred design can be qualified for all presently known specification requirements for the PLBR plant.

II. CONCLUSIONS

Based on studies conducted in extrapolating the LSBR Cold Leg Prototype Pump Concept (E.M. 4858) for a hot leg application, and on the Westinghouse ARD PLBR hot leg pump (E.M. 4911), the following conclusions have been reached:

1. The inducer centrifugal impeller design is the best approach to achieve a low NPSH hot leg pump design for the LSBR.
2. The volute-type cold leg pump design presented in E.M. 4858 has a very low likelihood of meeting G.E. Contract 67001 thermal transient requirements at an operating temperature of 950°F.
3. The preferred design concept for the hot leg application is shown in Figure 1 (spherical casing design). However, EMD recommends verification of this concept by additional more detailed thermal transient analysis.
4. The alternate design shown in Figure 2 (cylindrical casing) will meet specification thermal transient requirements, based on the FFTF pump analysis and experience.

III. RECOMMENDATIONS

1. EMD recommends that Figure 1 be considered the preferred design for the LSBR hot leg application. However, further analysis is recommended to assure that the design meets specification thermal transient requirements.
2. EMD recommends further analytical effort in the following areas, plus the developments noted below:

	<u>Preferred Design (Figure 1)</u>	<u>Alternate Design (Figure 2)</u>
a. Conduct a detailed thermal and stress analysis of the suction and discharge nozzle, plus other selected areas.	Yes	No
b. Conduct detailed dynamic and seismic analyses.	Yes	Yes
c. Conduct detailed radiation analysis.	Yes	Yes

In addition, the following development tests and/or studies should be conducted for both concepts: [Refer to Section 1.6 of E.M. 4858 (Cold Leg Pump Study) for details of these recommended areas.]

- 1) Development, Optimization and Validation of an Inducer/Centrifugal Impeller Combination for Low Available NPSH Applications

This effort is key to verifying the design concept and should be initiated most expeditiously.

- 2) Prototype Model Test

- 3) Optimization of Close Clearance Running Fits
- 4) Sodium Testing of MTI Self-Aligning Bearing
- 5) Evaluation of Flexible Coupling Designs
- 6) Development of Volumetric In-Service Inspection Capability
- 7) Development of Long Term Surveillance/Diagnostic Type Instrumentation
- 8) Simplification of MTI Self-Aligning Bearing

IV. DISCUSSION

For purposes of identification, the two designs are identified as follows:

a. Preferred Design - Figure 1

Inducer/centrifugal impeller with diffuser and spherical casing -
bottom suction and side discharge.

b. Alternate Design - Figure 2

Inducer/centrifugal impeller with diffuser and cylindrical casing -
side suction and bottom discharge.

It is noted that both hydraulic designs are based on a two-piece inducer/centrifugal impeller which was presented in Engineering Memorandum 4858. This WEMD innovative approach allows the inducer portion to raise the available head to the conventional (centrifugal) portion of the impeller, thereby allowing the centrifugal impeller to operate at a suction specific speed of about 10,000. This value is considered state-of-the-art.

As discussed in E.M. 4858, for the cold leg application the impeller discharged the pumped sodium into a volute which converted the velocity head into pressure head. EMD considers the volute a more optimum hydraulic design because it would be more efficient and could be operated at low-flow conditions without introducing the hydraulic instability phenomenon encountered in the FFTF pump. In addition, the volute would result in a pump tank diameter of 74 inches. However, the complex configuration of a volute results in a shape that cannot be extrapolated for specification thermal transients for a hot leg pump.

As the cold leg volute design showed no promise for hot leg application, it became necessary to utilize a diffuser design. Two alternates have been developed.

A. Diffuser Design with Spherical Casing - Preferred Design

In an attempt to minimize cost and optimize pump efficiency, the preferred design shown on Figure 1 was developed.

The pump is similar to the volute-type cold leg design in that it uses bottom suction with flow directly into the inducer and thence into the centrifugal impeller.

Discharge from the impeller flows into an eleven-vane diffuser and then into a spherical casing and out the side discharge nozzle.

The internal flow to the bearing is exactly the same as for the cold leg pump design.

This design however, due to the limited time available, has not been completely evaluated for the thermal transient conditions required by the specification. A preliminary assessment of the spherical casing has been made (see Appendix A) and, although the major portion of the casing is exposed to discharge pressure sodium, it appears that the casing would meet the specification thermal transients. However, a more detailed assessment is required on the casing as well as the nozzles.

This design speed is 873 rpm with an impeller diameter of 50 inches. The pump tank inside diameter is 78.6 inches (to allow the diffuser to fit inside the tank during assembly). The opening in the foundation is about 108 inches.

As the design does not make effective use of the discharge flow (dumps into the casing which would result in some slight loss of velocity head), the design would be slightly less efficient than the volute design but would be more efficient than the hydraulic design shown in Figure 2 (alternate design). This design would also be less costly and weigh less than the alternate design.

Based on the apparent advantages of this design, it is recommended that the diffuser with spherical casing design be considered the preferred design, and further evaluation of the thermal transient capability of this concept should be continued.

B. Diffuser Design with Cylindrical Casing - Alternate Design

The alternate design concept shown on Figure 2 is a side suction, bottom discharge pump. Sodium enters the pump tank through a nozzle on the side of the pump and enters a collecting chamber, then into a suction plenum (with two inlets - one directly in line with the suction nozzle, the other 180° opposite). The inlet sodium then flows through a reducer section (to assure a uniform velocity profile) into the suction side of the inducer, where the available head is raised to about 70 feet. The flow then proceeds into the suction side of the centrifugal impeller where the velocity is increased. The flow out of the impeller is into an eleven-vane diffuser, then into a turning vane section and down over the inlet plenum nozzles and out the discharge.

A secondary flow pattern exists where sodium at discharge head flows up past the outer portion of the turning vanes into a bearing support housing, which supports the hydrostatic bearing and provides a medium for supplying sodium at discharge head to the self-aligning hydrostatic bearing. Bearing flow is divided in two. About one-half flows down past the lower land of the hydrostatic bearing, past the bearing thermal barrier (through a clearance between the barrier and the shaft), and into the back side of the impeller. The flow then proceeds through the equalizing holes in the impeller shroud into a low-pressure zone of the impeller.

The upward flow of about one-half of the bearing flow is past the upper hydrostatic bearing land into the support cylinder inner chamber and thence down between the tank and the pressure casing to the pump inlet nozzle.

This design is optimum from a thermal transient standpoint. All structures which are exposed to full discharge to suction pressure differential are exposed to two-sided thermal shocks or attenuated thermal shocks, except for the discharge nozzle which sees full discharge pressure on one face, suction pressure on the other face, and full thermal shock on the discharge area. However, this nozzle configuration was evaluated for the specification transients, and it was ascertained that the nozzle design was acceptable.

Therefore, WEMD considers that this design is fully acceptable under thermal transient conditions. However, it must be noted that, to achieve this insensitivity to thermal transients, several compromises are necessary.

First, the pump length (below the impeller centerline) increases by a factor of three to accommodate the static hydraulic structures needed to achieve the thermal transient insensitivity.

Second, the more complex inlet path to the inlet side of the inducer means a loss of about 4 feet of the 30 feet NPSH available at the inlet nozzle. With 26 feet available at the inducer inlet, the pump speed must be reduced to 830 rpm (versus 873 rpm for the preferred design) to facilitate an inducer suction specific speed of 19,912. The reduced speed necessitates an increase in impeller diameter to 52 inches (versus 50 inches for the preferred design), which in turn results in an increase in pump tank diameter.

Third, the use of the internal pressure-containing structure causes an increase in pump tank diameter.

These compromises result in a slight reduction in pump efficiency and a significant increase in cost (due to the longer and larger diameter pump resulting from this design) but the concept will meet all thermal transient specification conditions.

C. Other Features of the Pump

Assessment of the other significant pump components, such as hydrostatic bearing, the shaft, the shield plug/support cylinder assembly, concepts developed for the cold leg pump (E.M. 4858) would not be affected by the implementation of either the preferred or the alternate concept. Their sizes, however, would be "scaled to fit". This "scaled to fit" mechanical design is depicted in Figure 2-1 of the WARD PLBR hot leg pump study (E.M. 4911, to be used in conjunction with this report). However, as discussed earlier, the tank shot cavity will be eliminated for the spherical casing design, due to reduction in the diameter to be passed through the support floor.

COMPARISON OF EMD PUMP DESIGN OPTIONS
LSBR Hot Leg (Primary) Pumps

	Preferred Design Diffuser <u>With Spherical Casing</u>	Alternate Design Diffuser <u>With Cylindrical Casing</u>
NPSH Available @ Suction Nozzle (Feet)	30	30
NPSH Available @ Inducer (Feet)	30	26
S_n (Suction Specific Speed @ Inducer)	18,812	19,912
N (rpm)	873	830
Impeller Diameter (Inches)	50	52
Tank Outer Diameter	82.6 (lower portion) 92.4 (upper portion)	130

LSBR Hot Leg (Primary) Pumps
EMD RANKING OF PUMP DESIGN OPTIONS
(On Scale of 0 - 10)

	<u>Preferred Design Diffuser With Spherical Casing</u>	<u>Alternate Design Diffuser With Cylindrical Casing</u>
Hydraulic Design	9	7
Cost	9	6
Weight/Size	9	6
Thermal Transient Capability	Not Determined At This Time	9

It is recommended that WEMD work with the system designers to develop a decision chart on selecting the optimum design.

APPENDIX A

PRELIMINARY STRESS ANALYSIS OF SPHERICAL CASING FOR LSBR HOT LEG (PRIMARY) PUMP

Note: This is a preliminary analysis
of the preferred design shown
in Figure 1.

- OBJECTIVES : (1) DETERMINE REQUIRED THICKNESS OF
SPHERICAL PORTIONS OF CASING
FROM A PRESSURE STANDPOINT
- (2) DETERMINE THERMAL STRESS IN
THE WALL OF THE SPHERICAL PORTION
OF THE CASING DUE TO THE SPECIFIED
THERMAL TRANSIENTS.

DESCRIPTION OF ANALYSIS: THE TENTATIVE THICKNESS OF
THE SPHERICAL PORTION IS DETERMINED
FIRST USING THE APPROPRIATE "CODE"
FORMULA FOR REQUIRED THICKNESS OF
A SPHERICAL VESSEL. ONCE THE THICKNESS
IS CHOSEN, THERMAL STRESSES DUE TO
THE SPECIFIED TEMPERATURE TRANSIENTS
ARE DETERMINED. THEN THE FATIGUE
DAMAGE IS CALCULATED ON THE BASIS
OF THE SPECIFIED NUMBER OF CYCLES
OF EACH OF THE TRANSIENTS.

ANALYSIS :

(A) DETERMINATION OF WALL THICKNESS OF SPHERICAL PORTION OF CASING (IN REGIONS REMOTE FROM DISCONTINUITIES).

$$t = \frac{p R}{2 S_m - p}$$

p = INTERNAL PRESS. (DESIGN)
PSI

R = INSIDE RADIUS (IN.)

$$t = \frac{200.48}{2 \times 14,300 - 200}$$

S_m = DESIGN STRESS INT.
= 14300 PSI FOR
304 SS AT 950°F

$$t = 0.338''$$

SINCE DISCONTINUITIES DO EXIST AS WELL AS OTHER STRESSES, THE DESIGN THICKNESS WILL BE TENTATIVELY SELECTED TO BE 1.0''

(B) DETERMINATION OF THERMAL STRESS IN THE SPHERICAL PORTION OF THE CASING (REMOTE FROM DISCONTINUITIES) DUE TO THE SPECIFIED THERMAL TRANSIENTS OF THE SODIUM.

THE FOLLOWING ASSUMPTIONS ARE MADE :

(1) INFINITE HEAT TRANSFER COEFF. AT INNER SURFACE.

(2) INSULATED OUTER SURFACE

THE FOLLOWING FORMULA GIVES THE THERMAL STRESS
DUE TO TRANSIENTS

$$\sigma = \frac{E \alpha m}{15 A (1 - \nu)} \left(-2R^2 - \frac{5b^3}{2R} + \phi + \psi \right) \quad \leftarrow \text{SEE PAGE 378, FORMULAS FOR STRESS AND STRAIN BY ROARK (4TH ED.)}$$

E = MODULUS OF ELASTICITY = 24.1×10^6 PSI

α = COEFF. OF THERMAL EXPANSION = $10.07 \cdot 10^{-6}/^\circ\text{F}$

A = COEFF. OF THERMAL DIFFUSIVITY = $0.10 \text{ IN}^2/\text{SEC}$

ν = POISSON'S RATIO = 0.3

R = RADIUS AT POINT OF INTEREST = 49"

b = INNER RADIUS = 48"

c = OUTER RADIUS = 49"

$$\phi = \frac{c^5 + 5c^2b^3 - 6b^5}{c^3 - b^3} = 11478$$

$$\psi = \frac{c^5b^3 - 6c^2b^5 + 5c^2b^4}{R^3(c^3 - b^3)} = -2286$$

m = RATE OF TEMPERATURE CHANGE AT SURFACE

SUBSTITUTING,

$$\sigma = 25190 m$$

FOR THE UPSET TRANSIENT, $m = \frac{205}{85} = 2.41 \text{ } ^\circ\text{F}/\text{SEC}$

$$\therefore \sigma = 25190 \times 2.41 = 60708 \text{ PSI}$$

FOR THE EMERGENCY TRANSIENT, $m = \frac{437}{150} = 2.91 \text{ } ^\circ\text{F}/\text{SEC}$

$$\therefore \sigma = 25190 \times 2.91 = 73303 \text{ PSI}$$

THE SPECIFICATION INDICATES THAT 400 CYCLES
OF THE UPSET TRANSIENT AND 10 CYCLES OF THE
EMERGENCY TRANSIENT MAY OCCUR.

PLASTIC STRAIN CONCENTRATION FACTOR OF 3.333 WILL BE USED

<u>TRANSIENT</u>	<u>m</u>	<u>STRESS RANGE</u>	<u>σ_a^*</u>	<u>N^{**}</u>	<u>m/N</u>
UPSET	400	60708 PSI	101170 PSI	1200	.333
EMERGENCY	10	73303	122159	800	.012

$$* \frac{3.333}{2} \times \text{STRESS RANGE} \quad \Sigma \frac{m}{N} = .345$$

** FROM ACCOMPANYING CURVE, ATTACHMENT A

THUS WE SEE THAT IN THE AREAS OF THE SPHERICAL PORTION OF THE CASING WHERE THE THICKNESS IS UNIFORM, THE THERMAL STRESSES ARE ACCEPTABLE. FURTHER ANALYSIS WILL BE REQUIRED IN THE NOZZLE REGIONS WHEN THE DESIGN IS MORE DEFINED.

ATTACHMENT A

NOTE
 $E = 26.0 \times 10^6 \text{ psi}$

DESIGN FATIGUE CURVE FOR AUSTENITIC STEELS, NICKEL-CHROMIUM-IRON
ALLOY, NICKEL-IRON-CHROMIUM ALLOY AND NICKEL-COPPER ALLOY.