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85,000-GPM, SINGLE-STAGE, SINGLE-SUCTION
LMFBR INTERMEDIATE CENTRIFUGAL PUMP

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LARGE LFMBR PUMPS AND COMPONENT

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ABSTRACT

The mechanical and hydraulic design features of the 85,000-gpm, single-stage, single-suction pump test article, which is designed to circulate liquid-sodium coolant in the intermediate heat-transport system of a Large-Scale Liquid Metal Fast Breeder Reactor (LS-LMFBR), are described. The design and analytical considerations used to satisfy the pump performance and operability requirements are presented. The validation of pump hydraulic performance using a hydraulic scale-model pump is discussed, as is the feature test for the mechanical-shaft seal system.

1.0 INTRODUCTION

The Liquid Metal Fast Breeder Reactor (LMFBR) Intermediate Vertical Centrifugal Pump (IVCP) is currently under construction by the Byron Jackson Pump Division for the Argonne National Laboratory. The IVCP was designed to be used in the intermediate sodium heat transport system of a large-scale fast breeder reactor power plant (1,300 MWe). The IVCP will generate the necessary pump discharge head (300 ft) to recirculate 85,000 gpm of 650°F sodium from the COLD LEG side of the steam generator to the intermediate heat exchanger (IHX) for reheat. The IVCP is of bottom-piped suction (Hallam type) construction with a side discharge nozzle. The pump tank contains an overflow sodium/gas system for maintaining a relatively constant liquid sodium level at the stand pipe gas bubbler nozzle elevation. The overflow system returns the internal sodium leakage to the intermediate loop.

The hydraulic end (wet end) of the pump is similar to the secondary pump supplied to the FERMI project. The pump case is a stainless steel double voluted casting.

The single-stage, single-suction impeller is welded to a hollow, gas-evacuated pump shaft in an overhung configuration, with a large sodium-lubricated hydrostatic bearing directly above the impeller. The motor/pump coupling is described as a rigid device, allowing for some flexibility in the bending mode. The motor shaft is supported radially by two oil-lubricated journal bearings straddling the rotor, with a Kingsbury-type thrust bearing sized to carry the static and running axial thrust loads. These thrusts may be resultant loads caused by gravity, seismic, dynamic, and hydraulic forces.

There is a cover gas buffer zone between the liquid sodium level and the oil-lubricated mechanical shaft seal cartridge. The purpose of the seal cartridge is to prevent the catastrophic loss of cover gas. The cartridge contains two balanced-type face seals mounted back to back, with an internal recirculating oil pump for seal face cooling.

The IVCP was designed, analyzed, and engineered to meet the stringent requirement of the equipment specifications (E-Spec). The E-Spec references the ASME Boiler and Pressure Vessel Code, Section III, Class 2, and all applicable RDT standards and Code cases. Current technological findings gained from the high-temperature tests on the FFTF and Clinch River pumps at ETEC were also made available to the pump designers and the analytical team. Such things as material densification and cellular convection generation were important considerations in finalizing the centrifugal pump design.

The total pump, i.e., tank, case, rotating assembly, inner structure, hydrostatic bearing, and seal cartridge, was thoroughly and rigorously analyzed for structural integrity. It was also shown that the pump operability will not be impaired or jeopardized under the worst-case thermal transient and/or during a safe shutdown earthquake (SSE). The IVCP is defined as an active component.

To verify the predicted hydraulic performance and the NPSH margin, and to meet the E-Spec requirement ". . . either no visible cavitation or cavitation bubble collapse on pumping element surfaces or static hydraulic surfaces . . . ," extensive hydraulic testing was conducted on a vertically mounted, one-third (1/3) hydraulic scale model (HSM) pump and loop. Hydraulic design conditions were met, and there was no dye removal from cavitation. A special dye and its proven method of application were used in the investigations, and the results were recorded on film, using high-speed photographic equipment.

Development tests were performed on a full-size (11.125-inch), oil-lubricated mechanical shaft seal and the internal oil pump. The design concept is a thin-film, hydrodynamic, balanced-type face seal, which after several design iterations met a self-imposed leakage criterion that covers the entire gas/system pressure range.

The purpose of this paper is to present in some detail the various aspects of the IVCP, such as design selection and considerations, development testing, and structural and hydraulic analytical approaches.

2.0 VERTICAL CENTRIFUGAL PUMP SELECTION AND DESCRIPTION

2.1 System Description

The basic system of the large-scale LMFBR is the loop-type heat transport design in which the primary vertical centrifugal pump operates as a HOT LEG unit recirculating liquid sodium between the IHX and the breeder reactor.

The intermediate (secondary) heat transport loop contains a COLD LEG vertical pump (IVCP) recirculating liquid sodium between the IHX and the steam generator.

The difference between HOT LEG and COLD LEG liquid sodium pumps refers to whether the pump takes its suction directly from the reactor (HOT LEG) at a relatively high temperature or pumps into the reactor (COLD LEG) from an intermediate heat exchanger at a somewhat reduced temperature. While obviously a COLD LEG pump, handling liquid sodium at a lower temperature, presents a less severe service condition, another very important factor compromises this seemingly obvious choice: relative suction conditions.

The COLD LEG pump in the intermediate heat transport loop takes its suction from the steam generator in which high friction losses reduce pump suction pressure available. Adequate suction pressure, plus margin, is critical to long, successful pump service life, but suction pressure excesses do not exist in either arrangement.

The suction conditions for the large-scale IVCP of 150-ft minimum to 236-ft maximum appear adequate when compared to 35 to 45 feet NPSHa (avalailable NPSH) for the HOT LEG primary pump. But large NPSHa does not necessarily preclude cavitation bubble formation. A brief discussion on this subject will be presented later in this paper.

2.2 IVCP Design Choices for Loop-Type Systems

Depending on suction-pressure conditions, both sump and piped-suction pump arrangements have been used in breeder reactors (Reference No. 1). They are typified by the name of the plant in which they were first installed (Figure 1). The "Hallam" pump is a piped-suction type named after the Hallam Nuclear Power Facility; the "Fermi" pump is a sump design named after the Enrico Fermi Fast Breeder Reactor Plant.

In the Hallam pump (Figure 2), internal leakage into the upper pump barrel occurs around the relatively large internal case labyrinth seal and the hydrostatic bearing. An overflow or return line to the sodium system is provided, and balancing holes in the upper impeller shroud permit sodium flow from the upper pump barrel down into the impeller eye. Thus, the liquid sodium level in the upper pump barrel is controlled by carefully sizing the holes in the impeller shroud and using an effective overflow system design. In addition, leakage around the suction inlet seal is relatively high because of the necessarily large diameter of the suction pipe.

The classification of the IVCP is the Hallam type: a vertical shaft design with a vertical, bottom-piped suction and horizontal, side discharge. Byron Jackson, in the early 1960's, supplied both the primary and secondary pumps for the Enrico Fermi breeder test program. Thousands of test hours were successfully logged in the test program. The Fermi secondary pump, at 13,500 gpm and 100 ft head and operating at 300° to 770°F, was the Hallam type, but with a cast stainless steel double-volute pump case (see Figure 3). The design team processing the ANL contract chose the Hallam/Fermi secondary-type design.

2.3 IVCP Description

The pumping design conditions are 85,000 gpm of liquid sodium at 650°F and 300 ft head. The nominal brake horsepower is 7,060 BHP (Na). The pump contains a single-suction, single-stage (SS), 40-inch impeller with a 36-inch suction nozzle and 36-inch discharge nozzle. The pump case (hydraulic passageways) is a standard double-voluted (DV) scroll construction. Therefore, the IVCP designation is 36x36x40 DVSS.

The major material of construction is stainless steel of the 316 series, chosen because of its resistance to corrosion, relative ease of machining/fabrication, and its structural properties at elevated temperatures.

There are ten major subassemblies that make up IVCP (see Figure 4). They are 1) motor mount, 2) shaft seal assembly, 3) seal oil supply tank, 4) closure flange, 5) solid coupling, 6) level sensor, 7) pump tank, 8) rotating assembly, 9) hydraulic bearing assembly, and 10) inner structure assembly.

The pump tank will be welded into the plant piping system, thus imposing the requirement that the pump internals be vertically removable. The inner structure, bearing assembly, and rotating assembly are extractable as a single unit. The inner structure also contains a thermal shield for high-temperature isolation, a seal oil leakage compartment, a sodium/oil vapor barrier, a cover gas purge system, and two level-probe sensors. The hydrostatic bearing assembly is bolted to the lower end of the inner structure.

The rotating assembly consists of a long, hollow shaft with a wall thickness of 1.25 inches and a length of approximately 135 inches. The diameter is 14.5 inches. There is a curvic coupling at the motor end and a welded-on single-stage, single-suction impeller at the "wet" end. Immediately behind the impeller, which is overhung, is the hydrostatic bearing journal for radial support. The hollow shaft is evacuated on final assembly to negate thermal distortions caused by convection cells within the shaft.

The pump shaft is connected through the curvic to an axially rigid, solid spacer coupling, then to the motor shaft. The solid coupling transmits the resultant axial thrust to an oil-lubricated Kingsbury-type thrust bearing contained in the motor. The solid spacer coupling is flexible enough in the bending mode to accommodate shaft radial motion when the hydrostatic bearing is recentering the rotating assembly. Therefore, because of the servo action of the hydrostatic bearing and the quasi-rigid, solid coupling, any pump tank distortion caused by thermal transients will not impair pump operability.

As a result of the two oil-lubricated journal bearings straddling the rotor within the motor, the single 24x17-inch sodium-lubricated hydrostatic pump bearing, and the design-optimized hollow shaft, there is sufficient design margin between the shaft critical speed and the operating frequency. The margin at 105% speed is 5.43 Hz. There will be further discussion on the subject of pump operability as this paper continues.

Internal leakage of several kinds, from the lower static seal, the partition between the pump discharge region and the pump tank sump, the hydrostatic bearing flow, and the upper wear ring, is directed between the return holes to the suction of the impeller and overflow system (Fig. 4). The overflow, less than 3,000 gpm, is returned to the intermediate loop system upstream of the suction nozzle. The overflow system also incorporates a standpipe gas bubbler network that ensures a fixed pump tank sodium level at the standpipe gas bubbler nozzle. The major benefit of the network is to minimize the severe effects of thermal transients in the vicinity of the sodium bearing and static seal area. It also mitigates the pump tank thermal deformations that may be caused by large sodium level excursions during up-transients as high as 900°F in a relatively short time.

A purge-gas system is used to flush out oil vapors to retard sodium vapor diffusion between the shaft and thermal shield.

3.0 MECHANICAL SHAFT SEAL ASSEMBLY DESCRIPTION

The cover gas, which provides a buffer gas zone between the shaft seal assembly and the sodium free surface, is contained within the pressure boundary of the IVCP and the intermediate heat transport system by a hydrodynamically balanced face-type mechanical shaft seal assembly, referred to as a seal cartridge. The cartridge is a self-contained module, completely preassembled, and can be removed from the IVCP through a spacer coupling opening without removing the main drive motor.

The mechanical shaft seal assembly (Figure 5) consists of two back-to-back, oil-lubricated, face-type balanced mechanical seals. The contacting faces, or rings, one stationary and one rotating, are separated by a very thin hydrodynamic oil film (about 40 microinches). The film is developed by the sealing pressure gradient and the hydrodynamic pumping action of the rotating face.

The stationary seal face is machined from a very hard and stable material (Stoody No. 1). The rotating face is machined from a softer material (carbon). Both faces are precision lapped to within two light bands of flatness. The faces, highly polished, are therefore conducive to a low-friction, film-forming operation. The seal faces are optimally configured to minimize pressure and thermally induced deforming stresses. Deformations causing a divergent gap in the direction of film flow would cause localized contact, resulting in a high wear rate.

The rotating and stationary rings are so designed that the balance between the face hydrodynamic separating forces and the hydraulic-plus-mechanical closing forces will support the thin film at a low face unit loading. The balance ratio is 65%.

This type of mechanical shaft seals will provide 10,000 hours mean time between failures (MTBF) at a nominal leakage rate. In this design the acceptable leakage is 210 ml/hr total.

The lower seal experiences a minimum differential pressure of about 30 psig, whereas the upper seal to atmosphere experiences, at the same time, an operating differential pressure of 160 psig. The resolution of this situation required an extensive seal development program, which is discussed later.

The heat generated by the seal internals and that transmitted up the pump shaft (about 16 Hp total) are being removed by a built-in, oil-to-air heat exchanger. Inside the cartridge is a pump-shaft-driven, viscous-shear, recirculating oil pump. There is also an oil make-up tank that is pressurized to cover gas pressure.

The secondary seals, U-cups, and O-rings are made of Viton to combat age hardening (loss of flexibility) caused by high temperature and the lubricant.

4.0 MAIN MOTOR AND SPEED CONTROL DESCRIPTION

4.1 Main Motor and Pony Motor

The IVCP will be driven by a nuclear-grade type of solid-shaft, vertical, variable-frequency, synchronous motor. The motor will be capable of transmitting over 7,000 brake horsepower (BHP) at design and of operating over a variable-speed range of 531 rpm to 1,171 rpm, holding speed constant within ± 6 rpm. The design of the motor will provide a calculated critical reed frequency of 37 Hz and will also withstand the imposed SSE requirements (see Figure 6 for relative sizes and weights).

The main motor will contain two oil-lubricated journal bearings, a Kinsbury-type thrust bearing, a flywheel, a brushless exciter, and an oil lube and lift system. Sitting on top of the main motor is a 100-Hp, 4-pole, variable-speed, induction motor pony motor. The pony motor, through a gear box and clutch assembly, will power the IVCP over a speed range of 2.5% to 7% of design speed. At 40% design speed, the main motor speed control will take over and operate the IVCP, satisfying the plant system head and flow requirements.

The design and analytical qualifications verifications needed for meeting the stringent requirements of the equipment specification ~~(Reference~~ 5) were accomplished employing a proposed nuclear-grade motor that is approximately twice the physical size of the commercially procured test motor to be used for water/sodium design. The variable-speed capability is provided by a coordinated, solid-state, static-converter, adjustable-frequency control system.

4.2 A Brief Description of the Adjustable-Frequency Controller

An adjustable-frequency control system controls the speed of a synchronous motor by converting a given power source to the voltage and frequency required by the motor running at the speed necessary for the process for which it is being used.

This is accomplished in two stages by a power system consisting of two three-phase thyristor bridges with their DC terminals connected through a reactor. The first-stage, three-phase bridge is connected to the 60-Hz power supply and operates as a phase-controlled rectifier to supply power to a DC link. The second bridge is connected to the synchronous motor and inverts the power from the DC link. The second bridge is connected to the synchronous motor and inverts the power from the DC link to the stator of the motor. The frequency of the output of the inverter is determined by the motor speed. The speed of the motor is approximately proportional to the voltage of the DC link as well as the motor frequency. The torque is approximately proportional to the current in the DC link.

The conversion of the AC power source begins in the first stage by means of a static power converter, which rectifies the alternating voltage and current to produce a DC voltage and current. A full-wave, six-pulse, single converter is used. The converter utilizes thyristor devices which are gate controlled. The controlled gating of the thyristor converter allows the DC output to be controlled to any value of voltage desired. In summary, the first-stage converter in an adjustable-frequency system converts the customer's AC power supply source to a controlled, adjustable-voltage DC power source.

The second stage of the adjustable-frequency system is a static-power, full converter which inverts the DC voltage and current in the DC link, between the first- and second-stage converters, to produce AC power on the output of the converter. The output AC power source connects directly to the synchronous motor. The voltage and frequency at the terminals of a synchronous motor are proportional to the speed of the motor. The DC voltage is controlled in the first converter to be proportional to the motor speed desired, since the output second-stage converter voltage is proportional to the DC link voltage. The second converter then changes the controlled voltage to be an AC voltage with a frequency also proportional to the motor speed desired.

The effect of the two converters on the synchronous motor is to connect it to a power source that provides power of a voltage and frequency proportional to the motor speed and that is essentially constant in volts per Hz.

The final result is a variable-speed motor/pump system that is very efficient and reliable. The efficiency of the motor and controller varies from 96.5% to 97.3% for loads of 50% to 100%.

5.0 IVCP SIGNIFICANT DESIGN PARAMETERS

5.1 Hydraulic Performance

The hydraulic performance as defined by the equipment specification requires the selected pump to operate in the regions shown in Figures 7 and 8. They are the IVCP performance with the main pump drive and the operational boundaries with the pony motor drive, respectively. The proposed IVCP hydraulic performance curve selected to meet the requirements of Figures 7 and 8 is shown in Figure 9. The curve was generated from a pump design used in reactor circulating pumps for both PWR and BWR nuclear plants.

The NPSH available, 150 feet for 50 hours of abnormal operation and 236 ft at design, plus the cavitation damage criteria, did place a design constraint when determining the impeller suction eye configuration, namely the inlet vane profile and angle.

A hydraulic analysis was made of the ANL intermediate pump to predict its cavitation performance and to judge qualitatively the impeller hydraulics. Two in-house computer programs were used for this purpose. They were the Jacobs and the Impel programs.

In the cavitation study it was found that the available NPSH of 150 feet would be enough to totally suppress cavitation over a flow range from 66,000 gpm to 83,000 gpm. The design flow rate of 85,000 gpm is slightly higher than this range, and one specified low-flow operating point (59,500 gpm) is slightly lower. Therefore, the selected impeller was an excellent compromise over this operating range. However, operation at flow rates at lower than design point will be required only for one hour during the 40-year life of the pump. Therefore, to favor the steady-state operating range, which is higher than design point, an alternate impeller, having a small leading edge modification, was also tested during the hydraulic scale-model testing. The final impeller leading edge geometry was based on the results of hydraulic scale-model tests. The hydraulic scale model was part of the development program.

The pressure and velocity-field computations were made using a detailed, quasi-three-dimensional analysis computer program. The calculations indicate very light loading in the inlet regions, a condition that will give good cavitation performance. The loading elsewhere in the impeller is quite conservative, except for a small region near the hub towards the outlet. By comparing previous Byron Jackson test data for impellers having different blade loadings, it was concluded that there would be no measurable efficiency penalty associated with this blade-loading effect. Several alternative designs were briefly examined and discarded because they tend to reduce performance in other areas. It was concluded that the present design was the optimum selection over the whole range of performance requirements.

The following displays several significant hydraulic parameters:

Flow - 85,000 gpm

Head - 300 ft

NPSH

Normal Minimum - 236 ft

Abnormal Minimum - 150 ft (50 hrs)

Specific Speed (Ns) - 4510

Suction Specific Speed (Ss)

Normal - 5399

Abnormal - 7584

Radial Hydraulic Thrust - 2,200 lbs (max)

Axial Hydraulic Thrust - 23,300 lbs (down)

3,800 lbs (up)

Eye Peripheral Velocity -

At Shroud - 131 fps

At Hub - 58 fps

5.2 MECHANICAL DESIGN CONSIDERATIONS AND DIMENSIONS

There were two major high-temperature design considerations that deserve mentioning, which if not properly handled would impact the operability of the IVCP. They are densification and cellular convection.

1. Densification

Dimensional changes in austenitic stainless steel castings exposed to high temperature for long periods have been considered to be a partial cause of the initial bearing failure of the FFTF pump. Further investigation into the failure disclosed that 45% loss of bearing clearance was caused by densification (casting shrinkage) and stress relaxation of the massive cast bearing support.

Additional test reports by Westinghouse (Reference 3) and Rockwell International (Reference 4) have shown that cast austenitic stainless steel does shrink when subjected to high temperature (1,000°F) for long periods of time (1,000 hours), and that dimensional stabilizing by specified heat treating has no significant effect on tensile properties of casting alloy CF8 or its counterpart, wrought 304 stainless steel.

Therefore, conservative design guards against material densification were taken. They were:

- o Heat treating the castings to provide dimensional stability
- o Using similar materials in critical areas
- o Designing all operating clearances to accommodate misalignment effects resulting from high-temperature operation and mechanical loads.

2. Cellular Convection

The formation of cellular convection (thermal convection cell pairs) in large vertical pumps, found during sodium testing of both the FFTF and the CRBRP prototype pumps, resulted in rigorous studies and verification testing. The findings and developed analytical methods resulting from that experience will be used for justification of the IVCP design configuration.

There is a potential for cell formation in the gap between the upper inner structure diameter (90.5 inches) and the pump tank inside diameter (91.0 inches). It can occur also in the lower tank region. The potential for cell formation appears greater in the gap region, but both areas will be investigated. If necessary, last-minute minor design modifications aimed at preventing cell formation (such as adding baffles to the inner structure) can be made prior to sodium testing of the IVCP.

The following is a partial list of significant mechanical dimensions:

Shaft Lenth from Motor Coupling	184.5 in.
Shaft Diameter	14.5 in.
Bearing Size (Diameter Length)	17.0 in.
Wear Ring Clearances (dia.)	
Top	100 mils
Bottom	140 mils
Journal/Bearing Clearance (dia.)	70 mils

6.0 DEVELOPMENT PROGRAMS

There were two significant development programs conducted during the design phase of the IVCP. They were: 1) the mechanical shaft development tests and 2) the hydraulic scale-model development tests. A brief description of the two programs follows.

6.1 The Mechanical Shaft Seal Development Tests

The primary purpose of these tests was to optimize 1) the seal face balance ratio, 2) the seal face(s) configurations, and 3) the seal face(s) unit loading for meeting an acceptable leakage rate (210 ml/hr total) and an acceptable wear rate (10,000 hrs MTBF).

The testing was conducted using a horizontal test fixture containing an inboard seal stage and an outboard seal stage. The two seals are full-size replicas of the prototype seal. The tester also contained an internal oil circulating pump.

Originally the hardface (stationary member) material was selected to be a bimetallic ring of stainless steel with a stellite hard-surface overlay. The softer face (rotating member) was made from a carbon composition.

There were many development tests conducted for face design optimizations, culminating in a 500-hour endurance test. The 500-hour test verified the seal design configuration, the leakage rate, and the wear rate at design temperature and pressure.

During the design optimization phase, two vital changes were made. They were:

1) The bimetallic stationary face was changed to a monometallic material, Stoody No. 1, for increased thermal stability. Thermal distortions when using the stainless steel ring with stellite overlay caused large variations in leakage and wear patterns. Special grooving behind the mating surface also mitigated the effects of thermal distortion on seal performance.

2) Development testing and earlier research conducted by the Borg-Warner Research Center disclosed that the selected secondary seal (U-cups and O-rings) material, Nitrile, was shrinking and losing its resiliency (loss of recovery) when in Mobil DTE oil at relatively high temperatures, above 200°F. Tensile, swell, and Lucas compressive stress relaxation property measurements were made, with the conclusion that seals manufactured from Viton were more appropriate. Therefore, a material change was made.

Two significant results from the 500-hour test were:

1) Cumulative leakage rate at 160 psig

	100 Hrs	500 Hrs
Inboard Seal:	45 ml/hr	25 ml/hr
Outboard Seal:	5 ml/hr	5 ml/hr

2) Cumulative wear rate at 160 psig after 500 hrs

Inboard Seal:	0.7×10^{-6} in/hr
Outboard Seal:	0.3×0^{-6} in/hr

6.2 Hydraulic Scale-Model Development Tests

A 1/3 (0.3115) hydraulic scale model of the IVCP was fabricated that included the hydrostatic bearing, an overflow system, and a cast double-volute pump case. The HSM was vertically mounted in a closed-loop test facility and tested for hydraulic and bearing performance. In addition, three clear plastic viewing ports were installed in the suction nozzle for observing and photographing bubble formation that occurred during both critical NPSH testing and normal operation. Prior to each cavitation test, Roller System Ink S-1, a black stenciling ink manufactured by Diagraph Bradley Industries, Inc., was applied to both sides of each impeller vane, to the inside surfaces of both impeller shrouds, to the suction splitter, and to both volute cutwater tips.

Cavitation-caused coating removal occurred on all vanes during the test at point 1, 150 feet NPSH (Figure 7). The amount of coating removal was moderate and was evident on the front (nonworking) side of all five vanes. The minor amount of coating removal that did occur indicates a relatively innocuous condition and, furthermore, that cavitation damage will not occur in 50 hours or less of operation at this point during the 40-year life of the pump.

There was no cavitation-caused coating removal when performing tests at the remaining flow points, indicating impeller/hydraulic design margins.

The results of the NPSH testing are shown in Figure 10. The design flow for the HSM was 8,246 gpm.

All other objectives were fulfilled by the HSM. Pump efficiency was approximately 84%, well above the efficiency goal of 80%. The HSM had a constantly rising head characteristic throughout the range tested, which included the entire operating flow range and flows far less and far greater than the operating range. The overflow rate was less than the maximum allowed. The discharge pressure fluctuations were well within the allowable limits.

The hydrostatic bearing loads were well within the load-carrying ability of the hydrostatic bearing. The inlet velocity profile was satisfactory.

7.0 TECHNICAL EVALUATIONS

7.1 Structural Integrity

Extensive structural analyses were conducted on the entire pump. Such regions as the upper tank flange and closure, the hydrostatic bearing assembly, the suction and discharge nozzles, the impeller and cast stainless steel volute were modeled and analyzed, employing 2D and 3D finite element analysis to evaluate primary and secondary stresses. The results when compared against the Code allowables were acceptable, with sufficient design margins.

An analytical effort involving the 316 series (CF8M) stainless steel pump case/volute (Figure 11) was undertaken.

A global 3D finite element analysis of the cast pump case was performed. The purpose of the analysis was to demonstrate that the pump case satisfies the rules of ASME Code Section III and elevated-temperature Code Case N253 for Class 2 pumps when subjected to specified design and operational loads. Code Case N253 was issued January 7, 1980, containing rules for materials and design of Class 2 and 3 components, parts, and appurtenances experiencing metal temperatures that exceed those covered by the rules and stress limits of ASME subsections NC and ND. There was concern expressed regarding the use of CF8M as a pressure-boundary material at elevated temperature. To establish CF8M fatigue properties, controlled laboratory tests were made on coupons sized 1"x1"x6". The data collected are shown in Figure 12.

The Equipment Specification defines thirty thermal transient events and categorizes them by service levels. For determining the secondary stresses in the 3D global model of the case, the heat transfer and thermal stress solution was performed only for the B-16 event.

In general, the B-16 transient is considered the most significant event because of its severe up-ramp rate and temperature excursion. Stresses for the other events are established by the stress ratio method, using transient severity factors.

The following is a summary of the findings:

For the suction nozzle, all stresses were well below Code limits. The most critical stress was the membrane stress because of the design loadings for which the margin of safety was .32.

The volute portion of the case satisfied all stress limits for both design and service level conditions. The minimum margin of safety, .30, occurred for membrane stress caused by Level A loadings.

The crotch, splitter, and static seal regions also satisfied all applicable stress limits. Again, the most critical stress was the membrane stress caused by the design loadings for which margins of safety were .18, .69, and 1.17 respectively.

7.2 IVCP Operability and Dynamic Considerations

7.2.1 Operability

On large, vertical, liquid sodium centrifugal pumps, the maintenance of bearing/shaft alignment during all phases of pump manufacturing, during normal operating conditions, and during severe abnormal plant events is essential to successful plant operation (Reference 5), commonly referred to as "operability."

To demonstrate operability, e.g., following a safe shutdown earthquake (SSE), especially at low pony motor speeds, bearing loads and clearance losses were computed with a finite element model that considered the static deflections of the otherwise statically indeterminate rotating and nonrotating structure. The operability analysis considered the worst-case effects of manufacturing tolerance stack-up, thermal transients, cellular convection, piping loads, hydraulic and impeller radial thrust, shaft unbalance, and permanent plastic and creep deformation.

Performance of the IVCP hydrostatic bearing, a four-pad, deep-pocket bearing with two pockets per pad (see Figure 13), was evaluated at 4% and 40% of design speed. The significant results were:

	Bearing Radial Clearance Loss (mils)
	4%
Manufacturing	13.17
Deadweight and Thermal Piping Loads	4.94
Thermal Effect	14.27
Hydraulic Radial Impeller Thrust	0.539
Rotating Unbalance	0.3697
Densification	0.
Absolute Sum	32.96
(RSS)	20.04

The absolute sum exceeds the nominal bearing clearance by 2.96 mils. The resultant rubbing load was estimated to be 5.4 lbs. This minor amount of heat generation (rubbing) will not impair the 40-year life expectancy of the IVCP.

The maximum displacements and impact loads at the lower and upper lands of the bearing during 30 seconds of a SSE time-history event were:

	Pump Speed %	Metal-to-Metal Contact Interface in.	Max. Impact Load lb.	Contact Duration sec.
Lower	4%	.00532	19520	.009
Upper	4%	.002943	12730	.006
Lower	40%	.00190	6970	.0075
Upper	40%	.000506	2190	.005

Further analytical studies disclosed no major bearing damage would occur affecting IVCP operability.

7.2.2 Maximum Suction and Discharge Nozzle Deflections

The pump motor assembly was analyzed for the effects of mass unbalance and seismic events on the amplitudes of vibration at the suction and discharge nozzles. Three computer runs were performed at 4%, 40%, and 105% of full pump speed. The significant results were:

		Mils Calc.	Mils Allowable
Suction	4%	.0106	1.0
	40%	.013765	1.0
	105%	.013976	0.05
Discharge	4%	.0074379	1.0
	40%	.0096565	1.0
	105%	.0098061	0.05

It is obvious from the above that the pump motor/foundation system is more than adequate for meeting the requirements of the E-Spec and are suitable for long, successful pump performance.

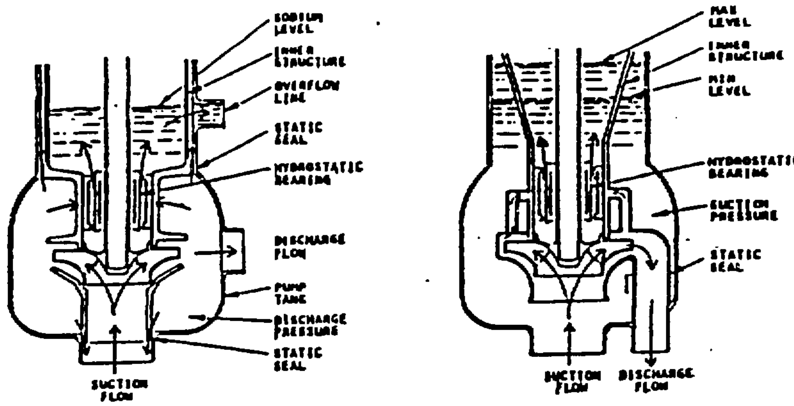
8.0 CONCLUSIONS

The analytical evaluation and the successful completion of the development testing have resulted in the release of long lead materials for the fabrication of the IVCP. The prototype test article will be water tested at Byron Jackson in preparation for sodium tests at ETEC.

The present design configuration is readily scalable to meet the requirements of CDS, 62,500 gpm at 300 ft head.

Meeting the stringent requirements of the E-spec, ASME Design Codes, and 10CFR Part 50, Appendix A, will ensure the IVCP owner plant licensability. This paper has presented a design/analytical overview of the IVCP, and we believe licensing will not be a problem.

HALLAM & FERMI
ARRANGEMENT COMPARISON



PIPED SUCTION (HALLAM TYPE) SUMP SUCTION (FERMI TYPE)

FIG. 1

HALLAM

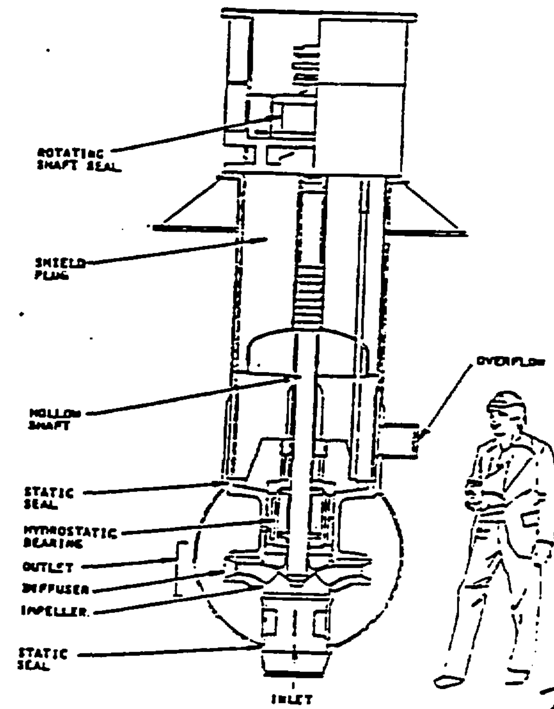


FIG 2

FERMI SECONDARY SODIUM PUMP

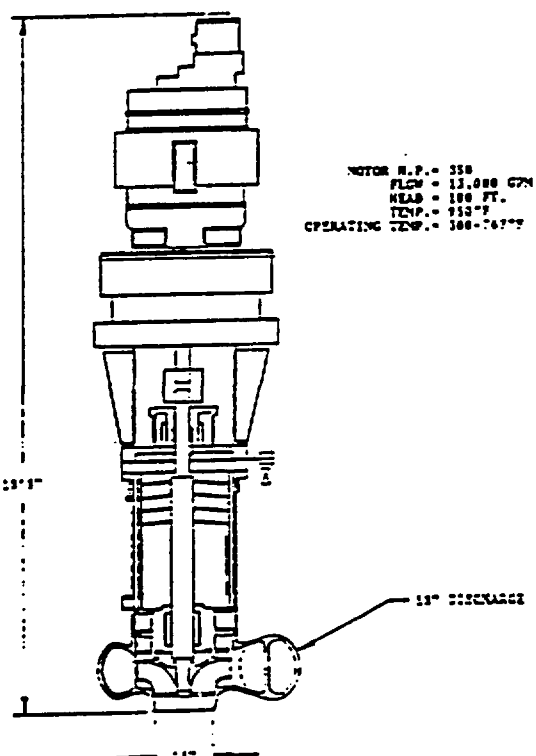
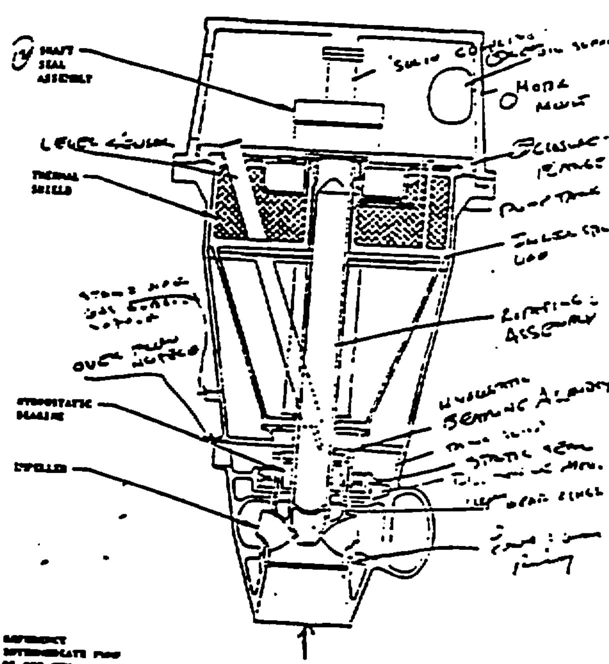


FIGURE 3



REFERENCE
INTERMEDIATE PUMP

FIGURE 4

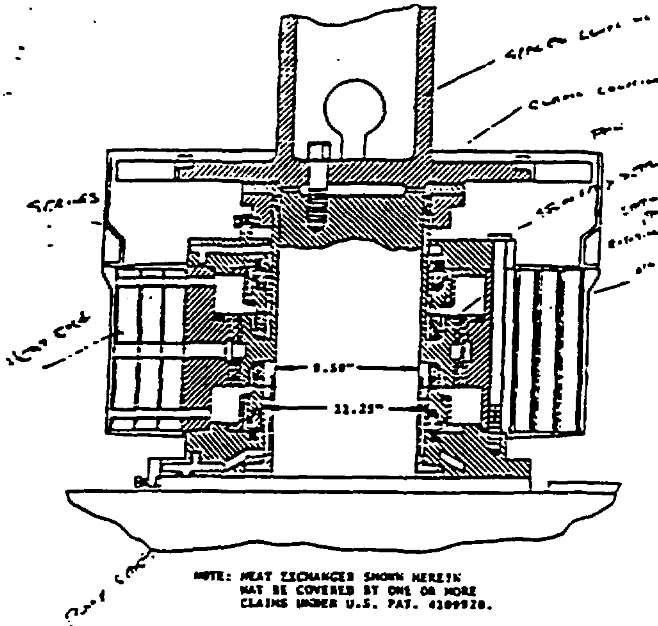
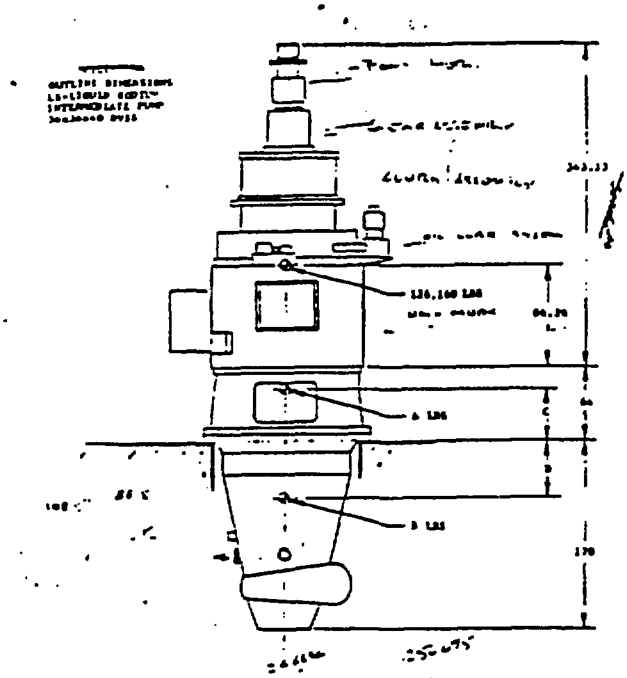


FIG. 5



ITEM	DRY	NORMAL	FULL
A (LBS)	[illegible]	[illegible]	242,098
B (LBS)	[illegible]	[illegible]	176,730
C (INCHES)	[illegible]	[illegible]	57.6
D (INCHES)	[illegible]	[illegible]	42.6

FIG 6

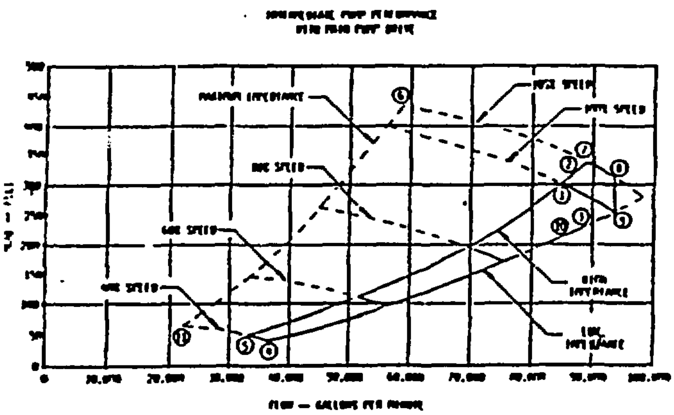


FIG 7.

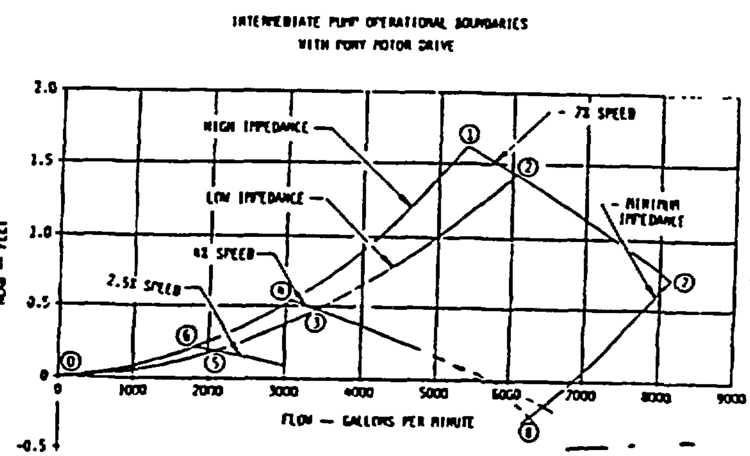


FIG 8

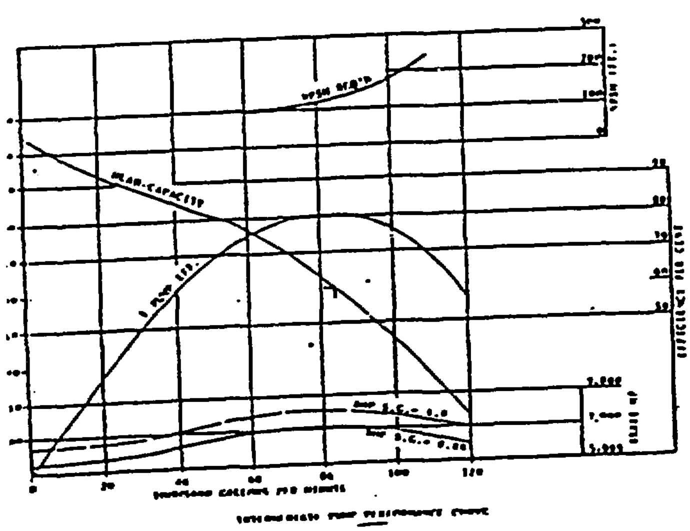


FIG 9

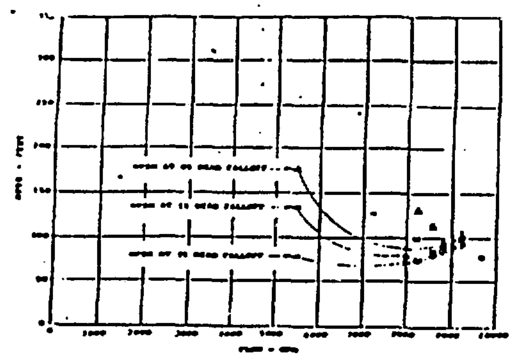


Figure NPSH vs. Flow for 0%, 1%, and 3% Head Falloff

FIG 10

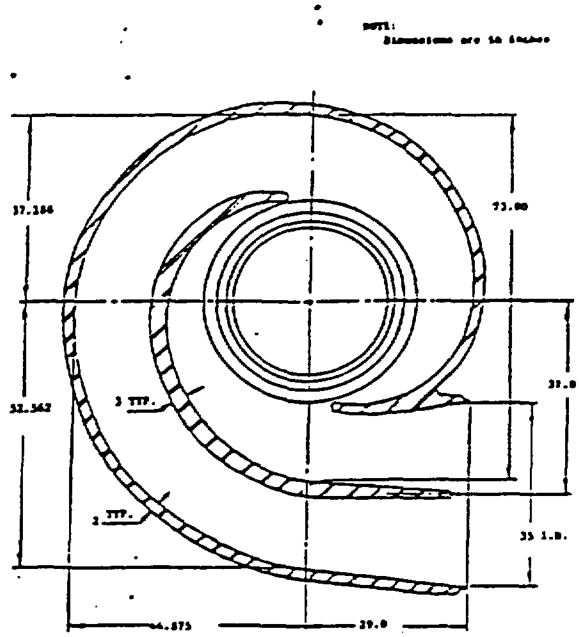


Figure Plan View of Pump Case at Centerline

FIG 11.

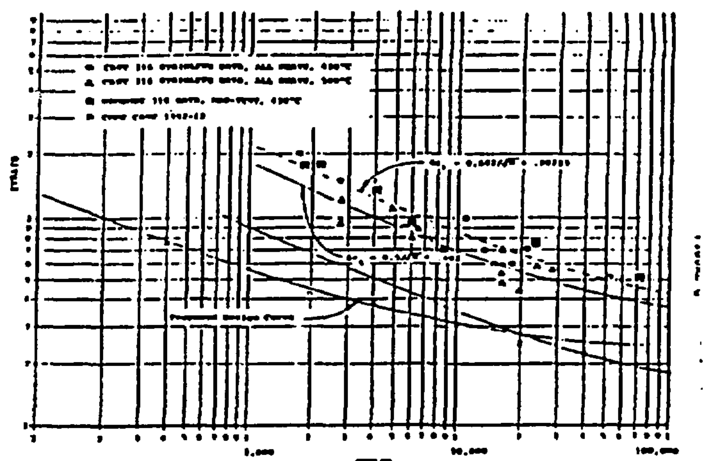


FIG. 12

Fig 12 WROUGHT 316 VS. CAST 316 FATIGUE DESIGN CURVE

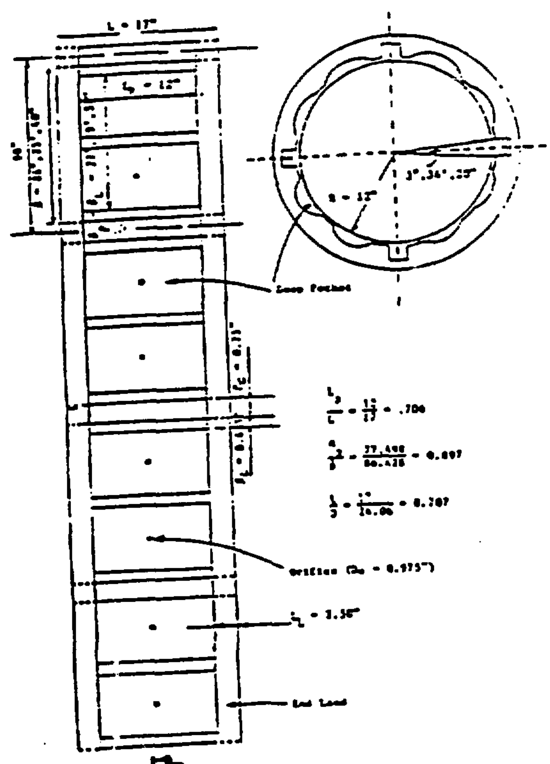


FIGURE 13 Four - Pad, Deep - Pocket, Hydrostatic Bearing With Two Pockets Per Pad

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1. "History of Liquid Metal Reactor Coolant Pumps in the United States," C. E. Fair, 1978 ASME Winter Annual Meeting, 15 December 1978.

2. "Internal Fluid Flow Management Analysis for Clinch River Breeder Reactor Plant Sodium Pumps," S. M. Cho, H. L. Zury, M. E. Cook, C. E. Fair, ASME 78-WA/NE-4, 1 September 1979.

3. "Dimensional Instability of CF8 Stainless Steel Castings at Elevated Temperatures," F. C. Hull, ASME Winter Annual Meeting, 2 December 1979, NYC.

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5. "Alignment and Operability Analysis of a Vertical Sodium Pump," V. K. Gupta, C. E. Fair, Paper E 1/7 Smirt, 17 August 1981.