PERFORMANCE OF HNPf PROTOTYPE
FREE-SURFACE SODIUM PUMP

By
R. W. ATZ

ATOMICS INTERNATIONAL

A DIVISION OF NORTH AMERICAN AVIATION, INC.
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ABSTRACT

A free-surface centrifugal pump, incorporating a hydraulic bearing running in sodium, was operated at the conditions required for service in the HNPF. After difficulties arising from inadequate shaft clearances were alleviated, the pump performed properly at a flowrate of 7200 gpm of 945°F sodium at 150-ft head. Results indicate that this pump should perform satisfactorily in the Hallam plant.

I. INTRODUCTION

Centrifugal pumps, capable of pumping 7200 gpm of 950°F sodium against 150-ft head, are to be used for circulation of sodium coolant in both primary and secondary systems of the HNPF. Because capacity of these pumps is substantially larger than for those previously used in sodium service, a test program was devised to demonstrate the performance of prototype units. The objectives of the project were to test two pumps of different design and to develop auxiliary components needed. This report describes the testing of a free-surface type pump in which there is no shaft seal.

A separate report^{*} has been written covering the tests of the other pump which used a shaft freeze seal.

II. DESCRIPTION OF APPARATUS

A. PUMP

The pump[†] used for this test program was a diffuser-type free-surface centrifugal pump having a nominal impeller diameter of 22 in. Because the drive motor was required to be mounted above radiation shielding, the pump was built with a vertical shaft 18 ft long. Power was transmitted from the motor shaft to the pump shaft by a variable-speed eddy-current coupling.[§] Shown in Figure 1 are the motor and coupling housings as well as a cross section of the entire pump.

The term "free surface" derives from the fact that this type of pump has no shaft seal to contain the pumped fluid. Leakage past the wear ring clearance on the discharge side of the impeller was allowed to accumulate in a pool in the pump case. The free surface of this pool rose to a level governed by the height of an overflow line which provided for return of this leakage flow back to the pump suction. The conventional radial bearing used near the impeller end of the pump shaft and the impeller suction side wear ring were both replaced by a hydrodynamic type bearing.

*R. W. Atz, "Testing of HNPF Freeze-Seal Pump," NAA-SR-4387 (to be published)

†Manufactured by Allis-Chalmers Mfg. Co., Milwaukee, Wisc.

§Manufactured by Dynamatic Div., Eaton Mfg. Co., Kenosha, Wisc.

Figure 1. Cross Section of Pump

The housing for this hydraulic bearing, supported in the diffuser casting, contained four pressure-pad chambers which were supplied with sodium from the impeller discharge during pump operation. The bearing journal was formed as an extension of the inlet eye of the impeller. A radial displacement of the journal resulted in an increase in pressure in the pad toward which the journal moved, and similarly a decrease in pressure on the diametrically opposite side; this action resulted in a centering force. The load-carrying capacity of the bearing increased with pump discharge pressure. When the pump was started, stopped, or run slowly, rubbing occurred between the bearing and journal surfaces. These surfaces were faced with stellite to resist wearing action. Construction of this bearing and the surrounding structure is illustrated in Figure 2.

Figure 2. Detail of Hydrodynamic Bearing

This pump was similar in many respects to the freeze-seal pump previously reported.* One of these similarities was the removable internal assembly. The entire inner structure was supported by a heavy flange which mated with the floor plate of the outer case. The internal assembly consisted of two principal parts. The upper portion, made of carbon steel, was an annular container for the lead-shot shielding required in the primary pumps. This shielding container surrounded the upper end of the pump shaft. Supported below the shield container was the thermal barrier, which was made of 304 stainless steel. The thermal barrier consisted of four circular baffle plates, dished to permit drainage. A cylindrical spool piece between the lower two plates and surrounding the shaft, served as an anti-whirl baffle in the free-surface pool. A photograph of the inner assembly is shown in Figure 3.

The impeller wear ring was mounted in the plate that formed the lower end of the thermal barrier structure. This plate also supported the diffuser casting. A seal ring, placed between the wear-ring support plate and the diffuser, located the inner assembly within the outer case. All parts in contact with sodium were made of 304 stainless steel.

The pump shaft was supported by conventional ball bearings mounted above the floor plate. The upper section of the shaft was solid, the lower section hollow, forming a continuous welded structure. A flange on the lower end of the shaft provided the surface to which the impeller was bolted. Sodium was directed vertically upward into the pump suction, through a 16-in. pipe, and discharged horizontally through a 14-in. pipe. Design operating conditions for the pump were 7200 gpm of 945°F sodium at a 150-ft head.

B. PUMP DRIVE AND SPEED CONTROL

The pump was driven by an induction motor connected through a variable-speed eddy-current coupling (trade name: Dynamatic Drive). Full-load speed for the motor was 1185 rpm; pump shaft speed was continuously variable from 0 to a maximum of about 1135 rpm. The Dynamatic Drive comprised a driving rotor mounted on the drive-motor shaft and a driven rotor, mounted on the top of the pump shaft. Within an annular space in the driven rotor was mounted a

*NAA-SR-4387

Figure 3. Pump Being Installed

stationary coil; dc power to this coil effected a magnetic coupling force between the two rotors. Pump speed was changed by varying power input to the excitation coil. Heat developed in this coupling was rejected to the nitrogen atmosphere in the casing by a blower attached to the driving rotor. A heat exchanger, mounted around the periphery of the coupling housing, transferred heat from the nitrogen to ambient air which was circulated through the exchanger by two blowers.

Automatic control of the input power to the coupling was provided by a magnetic-amplifier controller system. A tachometer generator mounted integrally with the eddy-current coupling generated an ac voltage used by the magnetic-amplifier for speed-regulation purposes. Manual control of pump speed was done by using a rheostat switched into the control circuitry. A provision, included in the speed controller, to use a voltage from a magnetic flowmeter during reactor operation, was not utilized during these tests.

C. TEST LOOP AND INSTRUMENTATION

The pump was tested in a loop made of 12-in. 304 stainless steel piping. Included in the discharge leg of the loop were a 12-in. ball-type throttling valve* and a magnetic flowmeter. An expansion tank was located near the pump suction. The total length of 12-in. piping was about 60 ft. A carbon steel storage tank of 28,000-lb capacity was connected to the loop through 4-in. stainless steel pipe. The overflow line from the pump casing was connected to the pump suction line to return sodium from the free-surface pool above the impeller to the system. Flow in the overflow line was monitored with a 3-in. magnetic flowmeter. A vent line connected the gas space in the pump case with the gas space in the expansion tank; because both vessels were essentially at pump suction pressure, the sodium levels were also the same.

Coolant for the valve-stem freeze-seals was supplied by a circulating system which initially contained tetralin but later contained kerosene. This coolant-supply temperature was maintained at 95°F. If the coolant-supply pump (or its standby) stopped, an alarm would sound and an electrical interlock would stop the sodium pump. The heat absorbed by the coolant was given up to a cooling-water system through a shell-and-tube heat exchanger.

*P-K Paul Co.

Nitrogen was used as a cover gas in the sodium pump, the pump-loop expansion tank, over the coolant system, and for pressurizing the sodium storage tank during loop filling operations.

Preheating of the complete test loop (i. e., storage tank, drain and fill lines, vent lines, pump casing, and loop expansion tank) was done with tubular heaters attached to the piping and the walls of the vessels.

Conventional variable-area flowmeters and Bourdon tube pressure gages were used in the inert gas and coolant systems. Power to the pump drive-motor and the eddy-current coupling was monitored by standard electrical meters. The sodium pressures at the pump suction and discharge were indicated by high-temperature, double-bellows, pressure transmitters.* Sodium height in the expansion tank was indicated by an inductance-coil liquid-level indicator. Flow was shown by a potentiometer reading output from a magnetic flowmeter. All temperature measurements were made with chromel-alumel thermocouples; thermocouples inside the pump were sheathed in stainless steel.

*Manning, Maxwell, and Moore, Model 185-X

III. PROCEDURE

In preparation for pump performance testing, the loop piping, pump, and expansion tank were filled to an appropriate level with sodium at 350°F.

It was originally intended that the operating level of the sodium in the pump be at the overflow line on the pump casing. During the test program it was necessary to raise this level above the pump overflow line. The need for this change in procedure is discussed in Section IV. After the system was filled, a nitrogen gas pressure of 5 psig was established in both the expansion tank and the pump case. Because power to the coolant pumps, heat-exchanger blower motors, and main drive-motor was interlocked with the Dynamatic Drive circuitry, all three of these interlocks had to be properly closed before the pump could be driven. With these conditions satisfied, excitation to the eddy-current coupling effected pump rotation at a speed determined by the operator. Flow-rate could be varied by adjusting either pump speed or throttle valve position, or both.

Heat losses from the test were such that, under average ambient conditions, energy transferred to the flowing sodium by the pump running at 800 rpm would maintain the loop at a temperature of about 945°F; operation of the pump at full-speed caused the temperature to rise above this value. Bulk sodium heating above 350°F was routinely accomplished by running the pump at full speed.

During operation in the reactor, the pump will be stopped and started many times; it will pump sodium at temperatures ranging from 350°F to 945°F; and pump speeds will vary from zero to 1100 rpm. A test schedule (Table I) was followed to simulate all reactor operating conditions (except radiation). Data were obtained so that head-capacity curves could be derived for the pump at the operating conditions described in Table I. Simultaneously, equilibrium data were taken of the temperatures of the thermal barrier, thrust bearing, and heat-exchanger inlet and outlet. Intermittently recorded were the temperatures of the motor windings, pump shaft temperature (at the floor-plate level), and nitrogen temperature at the heat-exchanger inlet.

TABLE I
TEST SCHEDULE

Test No.	Sodium Temperature (°F)	Pump Speed (rpm)
1	350	227
2	350	350
11	600	227
12	600	350
13	600	550
15	600	1100
22	900	350
23	900	550
24	900	1000
25	900	1100
26	900	1135
32	945	350
33	945	550
34	945	1000
35	945	1100
36	945	1135
42	1000	350
43	1000	550
44	1000	1000
45	1000	1100
46	1000	1135

Performance of an equilibrium test was accomplished by:

- a) Heating the sodium to the specified temperature (by varying the pump speed)
- b) Adjusting the pump speed to the proper value
- c) Varying the sodium flow over the range from zero to the maximum attainable within the capability of the pump.

Generally, six to eight different flowrates were recorded during a particular test.

Recorded continuously during each test were the sodium bulk temperatures and pump bearing temperatures. Information was taken from the following indicating (but nonrecording) instruments at each value of sodium flow:

- a) Sodium pump suction and discharge pressure transmitters
- b) Sodium flowmeters (12-in. and 3-in.)
- c) Sodium bulk temperature indicator
- d) Drive motor kilowatt meter, ammeter, and voltmeter
- e) Eddy-current-clutch excitation voltage and current meters
- f) Tachometer indicator.

IV. RESULTS AND DISCUSSION

Initial performance testing was thwarted by two operating difficulties; first, seizure of the pump shaft, and second, entrainment of cover gas into the pump suction line.

Two seizures of the pump shaft occurred at the bottom of the radiation barrier. An aluminum-bronze bushing that had been originally installed at this point was removed to increase the running clearance over the shaft. At the same time the annular clearance around the shaft was increased from 1/8-in. to 1/4-in. to help prevent shaft seizure.

Alignment of subassemblies within the pump proved to be a major problem. In an effort to determine cause of shaft seizure, a series of measurements of shaft deflection was made. Maximum deflection measured was approximately 0.040 in. at the midpoint of the shaft. This deflection, being less than the 0.060-in. annular clearance beneath the radiation-barrier bushing, suggested that misalignment of the overall pump assembly might have been a contributing factor to the seizure of the shaft.

The pump was hung as (previously described) from, and supported by, the floor plate. During the sodium filling-and-draining operations, the lower end of the pump casing was observed to deflect. The magnitude and direction of these movements were measured to try to relate the effect of this motion to the seizure of the shaft. It was concluded that this displacement did not contribute to the cause of shaft seizure because it was determined that the entire pump assembly moved as a unit. The locating fit at the bottom of the pump case was found to be displaced from the design centerpoint by 0.168 in.; this amount was not sufficient to have caused the shaft binding. The reason for compound deflections large enough to result in shaft seizure was not determined.

Sodium collected in the free-surface pool overflowed through a 6-in. line to be returned to pump suction. The amount of this recirculatory flow was dependent upon the pump operating conditions (speed, flow, ΔP) but never exceeded 185 gpm. During initial test runs it was not possible to circulate the full rated flow of 7200 gpm with the pump. At flowrates in excess of 5300 gpm the flow in the overflow line became erratic, and cover gas was drawn into pump suction.

An analysis of the hydraulic design of the pump test loop showed that the pressure at the pump suction was much lower than had been estimated, with the result that the sodium in the overflow line was being withdrawn at a rate much greater than it was being replaced from the pool in the casing. The point of attachment for the overflow line to the pump section line was relocated to reduce its pressure drop. The flow through the line was unchanged, and it was still not possible to circulate 7200 gpm through the test loop.

Satisfactory stable flow in the overflow line was achieved only when the loop expansion tank and the pump casing were filled to a point high enough to submerge the overflow line. When filled to this level, overflow became stable and the pump ran smoothly, with loop flowrates as high as 9000 gpm. The only effect of operating the pump with the surface of the sodium at the higher level was to raise the temperature axially along the casing. These changes in temperatures are shown in Figure 4.

After modifications of pump and piping were completed, hydraulic tests were performed according to the schedule of temperatures and speeds shown in Table I. Results of these tests are shown in the head-capacity curves of Figures 5 and 6. It was not possible to get satisfactory head-capacity data at high-speed conditions because the pressure developed by the pump exceeded the capacity of the pressure measuring instruments. The total running time of the test series was about 800 hours.

No difficulty was experienced with operation of the hydraulic bearing. This bearing was examined on each of three disassembly and cleaning operations of the inner assembly. Very slight wear marks were observed on the journal.

Disassembly and cleaning of the pump was found to be tedious. The pump shaft was a delicate part and was not provided with any means for handling. Removal of sodium from the discharge-side wear-ring labyrinth was not possible; this necessitated mechanical removal of the part prior to cleaning. Large forces were needed to accomplish this removal; the provision of tapped holes for jack screws would have facilitated this work. It was also found necessary to plug instrument thimbles in the inner assembly to prevent the flow of sodium vapor up into the space surrounding the driven rotor of the eddy current coupling. Design modifications to surmount these difficulties have been incorporated into pumps to be used in the HNPF.

Figure 4. Temperature Distribution in Pump

Figure 5. Pump Performance at Full Speed
(a. at 1000°F, b. at 600°F)

Figure 6. Pump Performance at Half Speed

V. CONCLUSIONS

The test pump operated satisfactorily over the range of required operating conditions. Satisfactory operation of the hydraulic bearing and the resolution of the overflow problem resulted in confirmation of the desirability of the free-surface pump concept for liquid-metal service.

APPENDIX

Derivation of the head-versus-capacity and pump efficiency curves are described in this section. Variables measured, their means of measurement, and their probable accuracy are shown in Table II.

A sample calculation is given below. The point described is plotted on Figure 5-a. Similar calculations were performed for all plotted points on the pump performance curves.

Taking the sodium flow as the independent variable, the head developed by the pump and the pump efficiency were plotted on the coordinate axes in Figure 5-a.

TABLE II
TEST INSTRUMENTS

Item	Read Out	Estimated Accuracy (%)
Pump shaft speed (rpm)	Strobotac	1/4
Sodium temperature (°F)	Thermocouple	2
Sodium pressures (psig)	Manning, Maxwell & Moore Transmitters	2
Sodium flow (read as mv)	Rubicon Potentiometer	5
Pump motor power (kw)	G. E. Panel Meter	2

Using the 7000 gpm flow from Figure 5-a as an example, the following laboratory data were used:

Indicated pump shaft speed (rpm)	1100
Sodium bulk temperature (°F)	985
Pump discharge pressure (psig)	55.0
Pump suction pressure (psig)	0.10
Main sodium flowmeter (mv)	38.0
Overflow sodium flowmeter (mv)	6.5
Power to drive motor (kw)	230

The actual pump speed, taken from the tachometer calibration curve, is 1085 rpm.

With sodium at 985°F, a main flowmeter signal of 38.0 mw equals 6900 gpm.

The overflow line was carrying 6.5 mv x 15.0 gpm/mv = 97.5 gpm.

The total dynamic head represents the sum of the differential head, the elevation head, and the contraction loss across the pump.

The differential head, P_1 , equals the pump discharge pressure less the pump suction pressure;

$$P_1 \text{ psi} = P_{\text{disch}} - P_{\text{suct.}}$$

$$55.0 - 0.10 = 54.9 \text{ psi}$$

and
$$P_1 (\text{ft}) = 54.9 \frac{\text{lb}}{\text{in.}^2} \times 144 \frac{\text{in.}^2}{\text{ft}^2} \times \frac{1}{51.6} \frac{\text{lb}}{\text{ft}^3} = 153.2 \text{ ft},$$

where 51.6 lb/ft^3 = density of sodium at 985°F.

The elevation head, P_2 , is the distance between the centerline of the pump discharge nozzle and the centerline of the pump suction line. In this instance, P_2 equals 2.25 ft.

The contraction head loss, P_3 , is credited to the head developed by the pump. The pump had a 16-in. suction nozzle and a 14-in. discharge nozzle. The contraction head is derived as follows:

Pipe Size (nominal) (in.)	Head Loss $V^2/2g$ (ft/1000 gpm)
14	0.0874
16	-0.0512
	0.0362

The contraction loss, P_3 , at 6900 + 97.5 gpm becomes

$$0.0362 \left(\frac{6997}{1000} \right)^2 = 1.78 \text{ ft.}$$

Therefore, the total dynamic head is $153.2 + 2.25 + 1.78 = 157.3$ ft at 6900 gpm.

The pump efficiency is the ratio of the hydraulic horsepower developed by the pump to the brake horsepower input to the pump.

The hydraulic horsepower developed by the pump is calculated as follows:

$$\begin{aligned} \text{HP} &= \frac{6900 \text{ gpm} \times 157.3 \text{ ft}}{33,000 \frac{\text{ft-lb}}{\text{min}}/\text{hp} \times \frac{1 \text{ gal}}{8.33 \text{ lb}}} \times \frac{51.6 \text{ lb/ft}^3}{62.4 \text{ lb/ft}^3} \\ &= 227 \text{ hp.} \end{aligned}$$

The horsepower input to the pump is calculated as follows:

$$\text{hp to drive motor} = \frac{\text{kw to drive motor}}{0.746 \text{ kw/hp}} \times \text{motor multiplier of 1.0.}$$

For this point

$$\text{hp to drive motor} = \frac{230 \text{ kw}}{0.746 \text{ kw/hp}} = 308 \text{ hp}$$

At 308 hp input the motor efficiency equals 90.7% and the motor speed equals 1183 rpm. The combined efficiency of the eddy-current coupling and the main drive motor is computed as follows:

$$\begin{aligned} \text{Efficiency} &= \text{Motor efficiency} \times \frac{\text{Pump speed}}{\text{Motor speed}} \\ &= 90.7 \times \frac{1085}{1183} = 83.2\%. \end{aligned}$$

The horsepower input to the pump becomes

$$308 \times 0.832 = 256.5 \text{ hp,}$$

and the pump efficiency equals

$$\frac{227}{256.5} \times 100 = 88.5\%.$$

For plotting purposes the head developed, and flowrates will be converted to a speed of 1100 rpm.

This is done as follows:

$$\frac{Q_1}{Q_2} = \frac{S_1}{S_2}, \frac{H_1}{H_2} = \left(\frac{S_1}{S_2}\right)^2$$

where Q_1 represents the flow, gpm at the shaft speed, S_1 in rpm.

H_1 describes the head developed in feet at the shaft speed, S_1 , in rpm.

$$Q_1 = 6900 \times \frac{1100}{1085} = 7000 \text{ gpm}$$

$$H_1 = 157.3 \times \left(\frac{1100}{1085}\right)^2 = 162.0 \text{ ft.}$$

This point is plotted on Figure 5-a.