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THE SNAP II POWER CONVERSION
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SODIUM PUMP DESIGN AND TESTING
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ABSTRACT

SNAP II is the designation for a 3 KW nuclear auxiliary power unit for utilization in a satellite vehicle. The SNAP II system consists of a reactor source which utilizes a mercury Rankine engine for power conversion. Design details and test data for the primary fluid reactor coolant pump used to transfer heat from the reactor to the mercury boiler are described in this report. The work was performed under a subcontract to Atomics International as part of the Atomic Energy Commission Contract AT(11-1)-Gen. 8.

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1.0 SUMMARY

The purpose of this report is to show the design procedure, calculations and test data for the permanent magnet axial gap centrifugal pump utilized in the SNAP II power conversion system reactor coolant loop.

The pump requirements and design points are:

1. Operating temperature 1000°F
2. Flow rate 45.2 lb/min on Na, 71.2 lb/min on NaK
3. Pressure rise 3 psi
4. Rotor speed 40,000 rpm

Test results of five pumps operating on Na and NaK are presented and compared to analytical performance estimates.

2.0 INTRODUCTION

The SNAP II auxiliary power unit consists of a reactor heat source and a power conversion system. The power conversion system is composed of a mercury boiler heated by a sodium transfer loop, a turbine, a condenser, a mercury pump and an alternator. The source of thermal energy is a nuclear reactor which uses liquid sodium as a coolant. A means of flowing the liquid sodium from the reactor, where it is heated, through the heat exchanger and back to the reactor was required. A number of different designs were investigated to meet this requirement and the advantages and disadvantages of these designs are discussed in the following paragraphs.

2.1 Separate Low Speed Drive to Rotate Electromagnetic Sodium Pump

As the efficiency of a high rotational speed E.M. Pump is very low, it is considered desirable to place the pump on a separate low-speed shaft rather than on the high-speed shaft of the power conversion unit. The efficiency of the pump could be increased from approximately 2 to 60 percent if the rotational speed was reduced to 2000 rpm. Low speed could be obtained by a low-speed liquid mercury turbine. The liquid mercury would be supplied by a high-speed centrifugal mercury pump operating on the PCS shaft. A schematic of such a system appears in Figure 1.

An investigation of a system of this type showed that the low-speed turbine, bearings, E.M. pump, and the efficiency of the high-speed mercury pump required three times more power from the PCS turbine than was required when the E.M. pump was placed directly on the high speed shaft. The disadvantages of this system included: increase in power required of the high-speed turbine; overall unfavorable effect on the PCS even though the E.M. pump efficiency was increased; a need for a separate system which added to the complexity of the overall system and reduced reliability. The weight and size of the pump was also a disadvantage.

2.2 Radial Gap Permanent Magnet Pump

Pumps of this type are attractive for operating at high temperatures and are the preferred electromagnetic types since there are no field windings. The magnetic field is produced by a rotating permanent magnet rotor placed on the high-speed shaft of the power conversion unit which induces eddy currents in the liquid metal surrounding the rotor. The induced currents react on the applied field and develop the driving force. The fluid passage consists of a thin annular space made from thin gauge stainless steel which surrounds the O.D. of the rotor. A laminated cylinder encloses the fluid passage for a suitable magnetic return path. A drawing of this pump is shown in Figure 2.

An analysis of the pump showed the efficiency to be very low, approximately 1 to 2 percent. The weight, size, and the need for heat shielding of the other components due to close proximity of the 1000°F liquid sodium would also present a problem if the pump were placed on the high speed shaft.

LOW SPEED LIQUID MERCURY TURBINE DRIVE FOR E. M. PUMP

FIGURE 1

RADIAL GAP PERMANENT MAGNET PUMP

FIGURE 2

2.3 Centrifugal Pump with Bladed Rotor

A conventional centrifugal pump utilizing a rotor with blades operating in the liquid metal was also investigated. In this design the rotor could be placed on the end of the high-speed shaft which would keep the high-temperature liquid metal away from the other components, however the difficulties of sealing liquid sodium from the rest of the system would present a sizable problem.

A magnetic coupling in conjunction with a centrifugal pump was also considered but the investigation showed that bearing losses alone would amount to approximately 600 watts which would be prohibitive.

2.4 Traveling - Field Induction Pump

A rotating field to pump the liquid metal through a centrifugal pump by the use of polyphase windings in a stationary slotted stator placed against the walls of the pump channel was investigated. With an arrangement of this type, additional alternator power would be required since the power for the pump stator would be drawn from the alternator. Therefore the overall system would be affected for a 1 to 6 percent efficient pump.

2.5 Axial-Gap Permanent Magnet Pump

This pump has a magnetic impeller which replaces the mechanical impeller of a conventional centrifugal pump. The fluid is rotated tangentially in an annulus by means of a perpendicular rotating magnetic field which produces a static head corresponding to the centrifugal force of the rotating liquid, and a velocity head recoverable as a pressure head in the volute. The efficiency of this type of pump at high shaft speeds is 2 to 6 percent. Its advantage is the absence of moving parts in direct contact with the liquid metal, therefore, eliminating potential seal or bearing problems. The small size and weight, and the suitability of placing the permanent magnet rotor on the end of the high-speed shaft of the combined rotating unit which locates the high temperature liquid Na heat away from the other components on the shaft is also advantageous. As analysis proved this pump the most favorable of any considered for recirculating the liquid sodium through the reactor and boiler, a permanent magnet axial gap pump was designed for the SNAP II power conversion system.

During the time of design and testing of the axial gap pump, the inherent difficulties of using sodium as the reactor coolant became apparent. As sodium is cooled and solidified, sodium oxides develop which form deposits in the lines. These deposits are very difficult to dissolve, requiring fluid temperatures in excess of 1200°F to break through the plugged lines.

NaK has a melting temperature of about 10°F as compared to 210°F for sodium. As NaK is always liquid at room temperature, plugs are not apt to develop in the lines. The use of NaK therefore, not only facilitates testing but would make the final PCS more reliable and trouble free.

Originally both sodium and NaK were considered for the reactor coolant. NaK was the preferred liquid metal for the reasons previously listed, but as the high resistivity of NaK further decreased the efficiency of the radial gap pump selected to pump the liquid metal, the decision was made to employ sodium instead. The promising results obtained from the investigation and tests of the axial gap permanent magnet pump encouraged the change from sodium to NaK.

3.0 DESCRIPTION

The operation of the axial gap permanent magnet liquid metal pump is similar to that of a conventional centrifugal type. The mechanical impeller of the centrifugal pump is replaced by a rotating magnetic field which induces eddy currents within the surrounding liquid metal. The induced currents in the liquid metal react on the applied field so that a driving force is developed. The construction of this pump is shown in Figure 3.

The magnetic flux produced by a two pole permanent magnet rotor emerges from the permanent magnet north pole, travels through the magnetic pole piece, crosses the fluid gap, enters into the laminated stator core and emerges opposite the rotor south pole. After again crossing the fluid gap, the flux travels through the magnetic pole piece and returns to the south pole of the permanent magnet. Finally the magnetic circuit is closed by connecting both permanent magnet bars to the end plate of the rotor.

A search coil of 10 turns is wound around the laminated stator core. The search coil is used to determine the flux density or flux density changes in the gap due to the demagnetization effects or eddy currents which are induced in the liquid metal. The rotating field produced by the permanent magnet induces EMF in the search coil. The induced EMF is proportional to the rotor speed and flux density in the gap. A millivoltmeter connected to the search coil indicates these changes.

PUMP STATOR
AXIAL GAP ELECTROMAGNETIC CENTRIFUGAL SODIUM PUMP FOR SNAP II

FIGURE 3

4.0 DISCUSSION

4.1 Design Procedure

Figure 4 represents the pumping annulus and the forces acting on the liquid metal.

- F_1 - force produced by rotating magnetic field on liquid metal in pumping annulus
- F_3 - tangential friction force
- F_7 - inertia force required to accelerate the fluid from its entrance to exit velocity

The retarding forces acting on the rotor are the sum of forces $F_1 + F_5$ where:

$F_1 = F_3 + F_7$ = the tangential force imparted to the liquid metal being pumped.

F_5 = the tangential force produced on the pump shell by the rotating field.

The useful head of the pump (H_T) is equal to the ideal head less the following losses:

- H_4 - head loss due to a fluid passing radially through a magnetic field which is directly related to P_4 .
- H_5 - head loss due to radial friction which is directly related to P_5 .
- H_6 - head loss due to friction in volute and inlet and exit of the fluid disc which is directly related to P_6 .

Efficiency is the major factor in the design of liquid metal pumps. The pump efficiency for the high-speed shaft depends largely on minimizing the liquid metal slip, which is the difference between the angular velocity of the magnetic field and the angular velocity of the liquid metal. For maximum efficiency the angular velocity of the liquid metal must therefore be high.

The procedure for the design of the axial gap liquid metal pump is listed below.

1. Select the optimum ratio of annulus I.D. to gap length for the desired liquid metal inlet velocity at rated flow. The permanent magnet rotor design must also meet this ratio.
2. Assume theoretical head (H_i) and solve for the liquid metal velocity.

FORCES ACTING ON FLUID IN THE PUMPING ANNULUS

FIGURE 4

3. The inertia (F_7) and tangential friction (F_3) forces are then determined by the liquid metal velocity, and the addition of these two forces is the total tangential force (F_1) which must be produced by the rotating field.
4. Determine the required flux density to produce the total tangential force (F_1). From this required flux density in the gap, the magnetic circuit is established and the total required ampere turns are calculated.
5. Determine the force produced on the shell by the rotating field (F_s).
6. With the liquid metal velocity in the pumping annulus known, a scroll is designed to convert velocity head to pressure head.
7. Permanent magnet rotor dimensions are selected to meet the preceding magnetic circuit requirement. Rotor flux permeances are calculated and the maximum available gap density for the selected rotor is determined.
8. The secondary losses -- magnetic retarding, radial friction, inlet, outlet, and scroll losses are calculated and the efficiency of the pump is determined.

4.2 Design Formula

4.2.1 Stator

4.2.1.1 Magnetic Design

Design equations are derived by assuming:

1. The moving fluid in the pumping section rotates as a disc.
2. The radius of the disc is large compared to the width of pumping annulus.
3. Fluid gap length in the annulus is small compared to pole pitch.
4. Sinusoidal flux distribution occurs in the gap.

The design calculations are based on References 1 and 2.

The calculation of tangential force F_1 produced by the rotating field is derived by R. Rudenberg: (1)*

$$F_1 = \frac{1}{4} \frac{\delta_1}{\rho} V_s B^2 \frac{\lambda^2 W}{\left(\frac{V_s}{\rho} \ell\right)^2 + w^2}$$

where:

F = developed force/pole (dynes)

δ_1 = thickness of rotating disc (cm)

V_s = velocity (cm/sec)

B = gap density (gauss)

λ = pole length (cm)

ρ = resistivity (abohm-cm)

t = pole pitch (cm)

$$w = \frac{\lambda}{t} + \frac{t}{\lambda}$$

$$\ell = 4 \frac{\delta_1}{\delta} \lambda$$

δ = total gap length (cm)

1. By switching from absolute CCS system to joule-inch-sec system and further substitution, R. Klecker obtains the following equation:

$$F_1 = \frac{n_s B^2}{K_i n_s + K_r} \quad (\text{lbs})$$

where:

n_s = field slip to the fluid (rpm)

B = flux density in fluid (lines/sq. inch)

*Figures in parenthesis pertain to the references appended to this report.

$$K_r = \frac{4.33 \times 10^{16} \sigma}{[N (r_2 - r_1) + 4r]^2}$$

$$\sigma = \frac{w \rho N}{r \delta_1}$$

N = number of poles

ρ = electrical resistivity (Ω -in)

δ_1 = fluid gap length (in)

r = mean radius of annulus (in)

$$w = \frac{\lambda}{t} + \frac{t}{\lambda}$$

Corrected conductor length

$$\lambda = r_2 - r_1 + \frac{2t}{\pi} \quad (\text{in.})$$

Pole pitch

$$t = \frac{2\pi r}{N} \quad (\text{in})$$

$$K_i = \frac{0.0489}{\sigma \delta^2}$$

Constants K_i and K_r represent reactance and resistance effect on the eddy currents.

K_i decreases with increasing ratio δ/δ_1 and with increased number of poles.

2. Tangential force (F_s) produced on shell by rotating field (2)

$$F_s = \frac{n_p B^2}{K_{is} n_r^2 + K_{rs}} \quad \text{Lbs}$$

where:

$$K_{is} = \frac{0.0489}{\sigma_s \delta^2}$$

$$\sigma_s = \frac{w_s \rho_s N}{r \delta_{is}}$$

$$w_s = \frac{N(r_2 - r_1) + 4r}{2\pi r} + \frac{2\pi r}{N(r_2 - r_1) + 4r}$$

ρ_s = electrical resistivity of walls (Ω - in)

δ_{is} = wall thickness (in)

$$K_{rs} = \frac{4.33 \times 10^{16} \sigma_s}{[N(r_2 - r_1) + 4r]^2}$$

This force can be minimized by selecting a high electrical resistivity material for the shell and by using a minimum shell thickness.

3. Inertia force (F_7) of fluid:

By applying the principle of angular momentum to the mass of liquid passing through the magnetic impeller:

Torque

$$T = \frac{Qw}{g} (v_{t2} r_2 - v_{t1} r_1)$$

Inertia force

$$\begin{aligned} F_7 = \frac{T}{r} &= \frac{Q62.4\gamma}{g r} \frac{2\pi r_2^2 n_t}{60 \times 12^2} - \frac{2\pi r_1^2 n_t}{60 \times 12^2} \\ &= 3.76 \times 10^{-5} \frac{\gamma Q n_t}{r} \left[r_2^2 - r_1^2 \right] \end{aligned}$$

4. Radial head (H_2) due to centrifugal force:

From Euler head equation:

$$H_2 = \frac{V_2^2 - V_1^2}{2g} = \frac{(2\pi n_t)^2 (r_2^2 - r_1^2)}{60^2 \times 2 \times 32.2 \times 144}$$

$$H_2 = 1.182 \times 10^{-6} (r_2^2 - r_1^2) n_t^2 \quad (4)$$

4.2.1.2 Secondary Losses

Hydraulic losses of the sodium flowing tangentially in the pump annulus are accounted for by consideration of forces F_3 and F_7 . The remaining hydraulic losses of the pump are accounted for by the following equation:

$$H_T = H_i - (H_4 + H_5 + H_6)$$

$$H_6 = H_{in} + H_{ex} + H_v$$

where H_{in} = pumping annulus inlet loss

H_{ex} = pumping annulus exit loss

H_v = volute friction head loss

H_4 = head loss due to fluid passing radially thru the magnetic field

4.2.2 Rotor

The rotating magnetic field in the pumping annulus is produced by an air stabilized permanent magnet rotor. For a minimum rotor weight, permanent magnets with a maximum energy product are used. Figure 5 shows the cross sectional design of this rotor.

To determine the flux density in the gap, the following flux permeances are calculated:

1. Total flux permeance, when the rotor is stabilized in air is represented by line OP_{st} in Figure 6. Since air stabilization applies the maximum demagnetization effect on the rotor, the position of the minor hysteresis loop is determined by the intersection of the permeance line with the demagnetization curve of the permanent magnet.
2. Leakage permeance when the rotor is in operating position is represented by line OP_e . The slope of this line depends on the ratio of the magnet length to the cross sectional area of the magnet and to the distance between both magnet bars.

PERMANENT MAGNET ROTOR

FIGURE 5

P.M. ROTOR DEMAGNETIZATION & PERMEANCE CURVES

FIGURE 6

3. Total flux permeance when rotor is in operating position is represented by line OP_t .

From Figure 6 the flux density in the gap at standstill is

$$\frac{AC \times \text{Cross Section area of magnet}}{\text{Area of pole face}}$$

F_g represents the required mmf to maintain this flux density in the gap. Under armature reaction, operating point A on the minor hysteresis loop will move towards A_1 and flux density in the gap will decrease. F_d represents the maximum demagnetization ampere turns for this air stabilized rotor and the flux density in the gap at this point will be $A_1 C_1$.

4.2.3 Pump Efficiency

Efficiency is defined as the ratio of the pump output power to the power applied to the shaft. The actual efficiency of the pump depends on the degree of departure of the assumptions from actual conditions. If $r_2 - r_1$ is small, the pump annulus narrow, and the flux distribution nearly sinusoidal, the calculated efficiency may be within 10 percent of test values. But if these are not the true conditions, the discrepancy between calculated and test values may be as high as 25 or 30 percent.

4.2.3.1 Determination of Efficiency

1. Required power to compensate shell losses

$$Hp_s = \frac{F_s N_p r_a}{5250 \times 12}$$

2. Power necessary to overcome tangential friction

$$Hp_f = \frac{F_3 N_p r_a}{5250 \times 12}$$

3. Power necessary to accelerate fluid in annulus

$$Hp_t = \frac{F_7 N_p r_a}{5250 \times 12}$$

4. Total power required

$$Hp_t = Hp_s + Hp_f + Hp_t$$

5. Total head

$$H_T = H_i - (H_4 + H_5 + H_6)$$

6. Pump output

$$Hp_o = \frac{H_T \quad Q \quad r}{3960}$$

7. Efficiency (neglecting rotor windage losses)

$$\eta = \frac{Hp_o \times 100}{Hp_t}$$

4.3 Design Calculations

The design calculations were based on the following requirements:

$$Q = 6.6 \text{ gpm}$$

$$\Delta P = 3 \text{ psi}$$

$$\text{Rotor speed} = 40,000 \text{ rpm}$$

$$\text{Liquid metal temp} = 1000^\circ \text{ F}$$

The geometric parameters are shown below:

$$\text{Fluid disc inside radius } r_1 = 0.375 \text{ inches}$$

$$\text{Fluid disc outside radius } r_2 = 0.625 \text{ inches}$$

$$\text{Fluid gap thickness} = 0.060$$

$$\text{Total gap thickness} = 0.115$$

The calculated values meeting the above design conditions are presented in Table I.

TABLE I

STATOR

Required Liquid Metal Disc Speed (n_t)	5370 rpm
Inertia Force (F_7)	0.554 lb
Tangential Friction Force (F_3)	0.0226 lb
Total Tangential Force Required by Rotating Field (F_1)	0.576 lb
Required Flux Density in Liquid Metal Gap (B)	14600 Lines/in ²
Slip Speed (n_s)	34630 rpm
Tangential Force Produced on Shell by Rotating Field (F_s)	0.167 lb

ROTOR

Permeance for 0.115 inch Gap Length (P_{eg})	4.52
Total Permeance when Rotor is Removed from Stator (P_e)	8.06
Leakage Permeance when Rotor is in Operating Position (P_{est})	6.78
Total Permeance when Rotor is in Operating Position	11.3

SECONDARY LOSSES

Head Loss Due to Fluid Passing Radially through a Magnetic Field (H_4)	0.37 ft
Radial Friction Head Loss in the Annulus (H_5)	0.0983 ft
Inlet Head Loss (sudden contraction) (H_{ic})	0.903 ft
Inlet Head Loss (turning) (H_{iT})	0.361 ft
Exit from Annulus Head Loss (Sudden Expansion) (H_{EE})	1.0624 ft
Exit from Annulus Head Loss (Turning - Radial) (H_{ER})	0.2127 ft
Exit from Annulus Head Loss (Turning - Tangential) (H_{ET})	2.2457 ft
Volute Friction Head Loss (H_{v_f})	0.3981 ft
Total Secondary Head Loss (H_T)	5.752 ft
Efficiency (Does not Include Friction and Windage) (η)	6.61 %

4.4 Test Apparatus

4.4.1 Dynamometer

An air turbine drive type dynamometer was chosen because of its adaptability of operation at various speeds. Additional factors leading to this choice were: (1) The only facility required for powering this device is plant air, which results in maximum mobility of the dynamometer; (2) Experience in proven turbine designs increases the machine performance reliability; (3) It affords a simple precision spindle design for this high-speed, high-temperature application.

The sodium pump rotor is mounted on the dynamometer turbine shaft. The dynamometer stationary housing is piloted and bolted to the sodium pump stator and axial clearance is set between sodium pump rotor and stator. (An assembly drawing showing the dynamometer design appears in Figure 7). Air is supplied to the turbine radially through a gap, via a stationary housing, then through nozzles on a movable housing. Air leakage through the gap between stationary and movable housings is minimized by labyrinth seals. The lubrication system is an oil mist system.

Reaction torque of the pump is transmitted from the cradled movable housing to a force transducer via a tee bar which is affixed to the movable housing. The force transducer is a linear variable differential transformer which has a movable core mounted on cantilevered springs. The output signal of the force transducer is fed to an amplifier. The amplifier output is applied to an indicator which is calibrated in torque. Speed is sensed by a magnetic pickup and a counter.

This dynamometer design features no restraining torque due to air inlet line or oil mist line stiffness. The air inlet is connected to the stationary housing and the air passed radially through an air gap to the nozzles in the movable housing. The air is exhausted from the turbine wheel axially from the movable housing through straightening vanes thereby eliminating any torque component. The oil mist line is also attached to the stationary housing. The oil mist is played from the stationary housing through a gap to holes in the movable housing to the shaft spindle which is profiled to direct the lubricant to the bearings.

Torque calibration is achieved by hanging various precision weights on an accurately located notch of the tee bar arm and recording transducer outputs.

4.4.2 Na Pump Test Rig

The sodium loop section and instrumentation of the test rig, which is described in detail in Topical Report No. 2, was used to test the sodium pump described in this report.

SODIUM PUMP DYNAMOMETER

FIGURE 7

4.4.3 Test Procedure

4.4.3.1 Pump Performance

The object of these tests was to determine the developed pressure, efficiency and characteristics of the liquid metal pump at various flows, temperatures and rotor speeds. The experimental pump was installed in a sodium loop with pressure gages on the inlet and outlet of the pump. An auxiliary E.M. pump was in the loop so that Na flow to the experimental pump could be varied. A dynamometer with the pump rotor mounted on the shaft was installed to measure torque from which efficiency could be calculated.

4.4.3.2 Friction and Windage Losses

A demagnetized rotor was spun and the torque recorded at various speeds.

4.4.3.3 Shell Losses

With the liquid metal removed from the pump, the input power was recorded at various rotor speeds. The losses in this test also include friction and windage losses.

4.4.3.4 Tangential Friction Losses in the Pumping Annulus

The pump was filled with liquid Na (no flow) and the input power recorded at various speeds. The losses from this test are a combination of friction and windage, shell and tangential friction.

4.4.3.5 Pressure Drop Due to Fluid Passing Through the Magnetic Field

With the rotor in operating position but not spinning, flow was established through the pump with the auxiliary E.M. pump and the pressure drop recorded. The losses include both the hydraulic and magnetic retarding forces.

4.4.3.6 Hydraulic Losses

In an effort to simulate the hydraulic losses obtained during the performance testing, the rotor was removed from the pump and flow established by a separate E.M. pump. The pressure drops through the pump at various flows were recorded.

This test did not give a complete picture of losses since the flow paths of the liquid will probably be different when the liquid is actually pumped through the pump. It gave, however, an approximate value which can be used to compare the different pumps tested.

4.4.4 Water Tests

After the first Na pump had been tested, and the results analyzed, it was felt that more information was needed regarding hydraulic losses, therefore water tests on each pump modification were incorporated into the program. The water loop and instrumentation used to conduct these tests is shown in Figure 8.

The results of the water tests at various flows for the five experimental pumps appear in Figure 9. The losses on the first three pumps were reduced slightly by changing the fluid gap width. The greatest decrease in losses occurred in pump No. 4 which had the original fluid gap width (.06 inch) but an increased volute size. This pump had considerably reduced hydraulic losses. When the fluid gap was increased in Pump No. 5 the losses decreased only slightly, as was proven in the first three pumps tested.

An effort was made to correlate the pressure drops obtained from the water tests with the losses received on the Na and NaK tests.

An approximate mathematical correlation can be made with these three fluids by means of the densities and Reynolds numbers as shown below.

$$\Delta P_{Na} = \Delta P_{H_2O} \times \frac{f_{Na}}{f_{H_2O}} \times \frac{\rho_{Na}}{\rho_{H_2O}}$$

$$\frac{f_{Na}}{f_{H_2O}} = \frac{K}{(N_{RE})_{Na}^{.25}} \times \frac{(N_{RE})_{H_2O}^{.25}}{K} = \frac{(N_{RE})_{H_2O}^{.25}}{(N_{RE})_{Na}^{.25}}$$

$$\frac{(N_{RE})_{H_2O}^{.25}}{(N_{RE})_{Na}^{.25}} \approx \left(\frac{V_D}{v} \right)_{H_2O}^{.25} \times \left(\frac{v}{V_D} \right)_{Na}^{.25} \approx \frac{(v)^{.25}_{Na}}{(v)^{.25}_{H_2O}}$$

$$\therefore \Delta P_{Na} = \Delta P_{H_2O} \times \frac{\rho_{Na}}{\rho_{H_2O}} \times \frac{(v)^{.25}_{Na}}{(v)^{.25}_{H_2O}}$$

$$\text{And } \Delta P_{NaK} = \Delta P_{H_2O} \times \frac{\rho_{NaK}}{\rho_{H_2O}} \times \frac{(v)^{.25}_{NaK}}{(v)^{.25}_{H_2O}}$$

$$\therefore \Delta P_{H_2O} = 1.6 \Delta P_{Na}$$

$$\text{And } \Delta P_{H_2O} = 1.89 \Delta P_{NaK}$$

WATER PRESSURE DROP TEST RIG

FIGURE 8

SNAP II Na PUMP WATER FLOW TEST PRESSURE DROP VS. FLOW

FIGURE 9

Table II shows the water hydraulic losses obtained from the tests. Losses on Na and NaK were calculated from the water test values by applying the factors mentioned previously and the actual hydraulic losses obtained from the tests conducted with Na and NaK

TABLE II

Pump No.	Description	Hydraulic Losses at Design Flow (psi)				
		Tests H ₂ O	Calculated		Tests	
			Na	NaK	Na	NaK
1	.060 Gap	5.65	3.53		*10	
2	.095 Gap	4.60	2.88		-	
3	0.120 Gap	4.90	3.06		*15	
4	.060 Gap Large Volute	1.40		0.74		0.4
5	.095 Gap Large Volute	0.85		0.45		0.3

* Extrapolated Values

As shown by the Table, the losses changed only slightly with increase in fluid gap, the greatest change occurring with the larger scroll. There appears to be no correlation between the Na and H₂O tests, but the NaK tests seemed to correlate with the expected NaK values which were based on the water tests. The wide discrepancy in the Na tests can be accounted for by undissolved Na plugs in the lines when the tests were conducted.

4.5 Test

The pump developed pressure (both calculations and test results) at the design condition of 6.6 gpm, 1000°F liquid metal temperature and a rotor speed of 40,000 rpm is shown in Table III. The only changes made in the pump were in the stators, the rotors all had the same size and shape of magnets.

4.5.1 Na Pump No. 1

This pump was theoretically designed to deliver 3.7 psi. This design objective was used to compensate for some of the unknown conditions that might exist in the pump.

From the Table it can be seen that the pump developed 0 psi at the design conditions, but a 3 psi pressure was developed at approximately 1/3 the design flow as shown in Figure 10. Losses such as friction, and windage, shell, tangential friction, magnetic retarding and hydraulic losses are shown in Figures 11, 12, 13, 14 and 15.

From the analysis of the tests made on this pump, it appeared that the theoretical magnetic design was correct since the voltage induced in the search coil at design conditions during test was 0.82 volts (17,000 lines/in² flux density) which compared favorably with the calculated value of 0.70 volts (14,500 lines/in² flux density). Hydraulic losses, however, were much higher than had been expected.

Since the hydraulic losses could not be individually separated by tests, it could not be ascertained whether the largest losses were occurring in the pumping annulus or in the scroll; therefore, it was decided to increase the annulus thickness to see if this would increase the pump performance.

4.5.2 Na Pump No. 2

As the gap width increased, two variables changed simultaneously: (1) the flux density in the gap decreased, and (2) as the width of liquid metal increased, the resistance of the fluid decreased which in turn increased the driving force. Also as the gap increased, the radial friction loss was reduced. The original pump was cut apart and an .035 inch spacer inserted which made the total fluid gap thickness equal to .095 inches. The spacer also produced a small increase in the scroll area. The greatest apparent benefit, however, was in the pumping annulus. Pump test results appear in Table III and in Figure 16.

SNAP II COMPONENT Na PUMP #1 FLOW VS ΔP VS SPEED

FIGURE 10

SNAP II COMPONENT Na PUMP #1
FRICTION AND WINDAGE LOSSES VS ROTOR SPEED

FIGURE 11

FIGURE 12

SNAP II COMPONENT Na PUMP #1
 FRICTION & WINDAGE, SHELL & TANGENTIAL FRICTION LOSSES VS ROTOR SPEED

FIGURE 13

SNAP II COMPONENT Na PUMP #1
LOSS DUE TO FLUID PASSING RADially THROUGH A MAGNETIC FIELD
AND HYDRAULIC LOSSES VS. FLOW

FIGURE 14

FIGURE 15

SNAP II COMPONENT Na PUMP #2 (.035 SPACER)
FLOW VS. ΔP VS. SPEED

FIGURE 16

Design pressure was not developed at design flow, but 3 psi was obtained at one-half the desired flow which showed an increase in capacity of 1 gpm. Hydraulic loss data was not taken due to test conditions, so these losses could not be compared to Na Pump No. 1 losses. The friction and windage losses remained the same for both pumps since the rotor was not changed. The shell loss and tangential friction losses are shown in Figures 17 and 18.

4.5.3 Na Pump No. 3

Since the performance was increased by widening the fluid gap on Pump No. 2 it was apparent that there was an optimum gap width at which the pump would operate at maximum performance; consequently the pump was cut apart again and a .060 inch spacer inserted. This change made the total fluid gap width 0.120 inches. Pump No. 3 test results appear in Table III and Figure 19.

Although the pump developed no pressure at design flow it did achieve 3 psi at one-half the flow (the same as pump No. 2). The calculations show that performance decreased at design flow and that the fluid gap had increased past the optimum point. Figure 19 demonstrates that the performance curve rapidly changes as the flow is increased. When the pump was removed from the test housing, it was found that sodium had leaked at the point where the pump halves had been welded together as a result of depletion of the weld stock from two previous rewelds. This condition may have caused some sodium oxide to form inside the pump, thus making the data unreliable. Hydraulic losses, which were very high, may also have been caused by the leakage.

4.5.4 Pump No. 4

Results of the first two tests indicated that the pumping annulus was not the area in which the greatest losses were occurring. Since a change in gap width increased the performance of the pump only slightly, it was assumed that the largest losses were occurring in the scroll.

To verify this theory, a larger scroll was designed to better diffuse the flow and reduce the hydraulic losses. The fluid gap was set at the original gap width of .06 inch so only the increased performance caused by the scroll would appear in the results. As the change in working fluid from Na to NaK occurred at this time, Na Pump No. 4 was tested in a NaK loop rather than a Na loop for which it was designed. Theoretical calculations were made to convert NaK performance to Na performance.

The anticipated pressure at 6.6 gpm, 1000°F and a rotor speed of 40,000 rpm, with NaK as the working fluid, was 0.433 psi. The pressure developed on test, 0.4 psi, indicated that the scroll change increased the capacity of the pump greatly since the calculated pressure agreed well with the test data. Full pump performance vs.

SNAP II COMPONENT Na PUMP #2
FRICTION & WINDAGE & SHELL LOSSES VS ROTOR SPEED

FIGURE 17

SNAP II COMPONENT Na PUMP #2
FRICTION & WINDAGE, SHELL & TANGENTIAL FRICTION LOSSES VS ROTOR SPEED

FIGURE 18

SNAP II COMPONENT Na PUMP #3 (.06 SPACER)
FLOW VS. ΔP VS. SPEED

FIGURE 19

TABLE III

Na Pump No.	Description	Liquid Metal	Flow Gpm	Rotor Speed Rpm	Liquid Metal Temp °F	Dev. Pressure psi		Hydraulic Losses psi	
						Calc.	Test	Calc.	Test
1	.060 Gap	Na	6.6	40,000	1000	3.708	0	1.855	*10
2	.095 Gap	Na	6.6	40,000	1000	5.088	0	1.695	--
3	.0120 Gap	Na	6.6	40,000	1000	4.336	0	1.49	*15
4	.060 Gap Large Volute	NaK	6.6	40,000	1000	0.433	0.4	0.911	0.4
5	.095 Gap Large Volute	NaK	6.6	40,000	1000	1.197	0.9	0.755	0.3

*Extrapolated Values

speed is shown in Figure 20. From Table III and Figure 21 it can be seen that the hydraulic losses had been reduced considerably.

4.5.5 Na Pump No. 5

To check the change in performance with the use of the larger scroll, a .035 inch spacer was inserted in Pump No. 4 making the fluid gap .095 inches wide. This pump was tested in the NaK loop. Results at design conditions are shown in Table III. The developed pressure of 0.9 psi compares favorably with the calculated pressure of 1.1. The increase in fluid gap width increased the performance of this pump by 0.5 psi, proving that the original gap of .06 was not the optimum width. Complete performance curves at various speeds and flows are shown in Figure 22.

The hydraulic losses for Pumps No. 4 and 5 were considerably reduced from the first three pumps tested. The separate losses are shown in Figures 21 and 23.

4.5.6 Summary of Test Results

Five pumps of different configurations have been tested. A comparison of these configurations is presented in Figure 24. The first three pump configurations which were designed and tested on sodium did not meet the expected performance. When the scroll design was changed, sodium pumps No. 4 and No. 5 performed on NaK as predicted according to the calculations. The performance on Na would be approximately 2-1/2 times better than on NaK due to the change in resistivity, density, and kinematic viscosity. Therefore the performance of Pump No. 5 would have very nearly met design requirements if it had been tested on Na. A tabulation of reduced data from Pump No. 5 testing appears in the Appendix. A comparison of pump performance on all five pumps tested appears in Figure 25.

The highest efficiency of 2.5 percent was achieved by Pump No. 2 at half-flow conditions. Pump No. 5 achieved approximately a two percent efficiency at the Na design flow of 6.6 gpm on NaK. Figure 26 shows the efficiency at design speed and temperature of the pumps tested.

The calculated hydraulic losses and the actual test data vary considerably especially for the pumps tested on Na. The losses were calculated with the use of maximum coefficients wherever possible to arrive at a conservative pump design.

The test conducted to obtain the hydraulic losses is a static test and does not completely represent actual losses when the pump is in operation. This is due to the fact that the disc of liquid metal rotates faster during the pumping process than when the pump is not operating. The speed of the rotating disc affects the radial friction, sudden expansion into the volute, turning and volute friction losses; therefore, the data taken during tests will be only an approximation of the hydraulic losses and should be lower than the calculated values. This did not occur in the pumps tested on Na as can be seen in Table III. The Na pumps tested on NaK did have lower losses than were calculated,

FIGURE 20

SNAP II COMPONENT Na PUMP #5
DEVELOPED PRESSURE VS FLOW

FIGURE 21

SNAP II COMPONENT Na PUMP #4
HYDRAULIC LOSSES VS FLOW

FIGURE 22

FIGURE 23

NA PUMP STATOR CONFIGURATIONS

Pump#	A	B	C	D
1	.060	.020	1.360	1.238
2	.095	.020	1.360	1.238
3	.120	.020	1.360	1.238
4	.060	.015	1.538	1.988
5	.095	.015	1.538	1.988

FIGURE 24

SNAP II COMPONENT Na PUMPS
FLOW VS. ΔP

FIGURE 25

SNAP II COMP. Na PUMPS/ EFFICIENCY VS FLOW

FIGURE 26

but the overall performance, both calculations and test values, did agree closely.

It can therefore be assumed that the calculated hydraulic losses were correct for these two pumps since the magnetic design had been checked previously.

Since the tests run on Na did not produce the expected results but the test data obtained with the use of NaK compared favorably with the calculations, it must be assumed that when Na was used, the lines and pumps were partially clogged with sodium oxide which would account for the high losses obtained during testing.

5.0 CONCLUSIONS

On the basis of comparative analysis a shaft driven electromagnetic pump was selected as the most promising pump concept for the SNAP II application. The development effort has significantly advanced the state of the art for this type pump, particularly for the high speed applications. Analysis coupled with promising test results on the axial gap centrifugal version of this pump allowed a change in primary system fluid from Na to eutectic NaK. Because of NaK's high resistivity, the efficiency of an induction pump will be less with NaK than with sodium, but the efficiency which is possible with the axial gap pump will permit the utilization of NaK with approximately the same shaft power originally predicted for pumping sodium. The use of NaK results in improved system operational reliability.

An evaluation of pump test results indicates that the original Na pump objectives were met by the No. 5 pump. Test confirmation of the design analysis indicates that the NaK pump objectives can be achieved with pumps currently being designed.

6.0 RECOMMENDATIONS

The future program will be directed towards a NaK pump development. Tests should be conducted to determine the optimum fluid gap at design flow conditions. Methods of reducing hydraulic losses which will increase the performance and efficiency will also be studied. Off design information on the characteristics of the pump should be gained by running tests at various temperatures, rotor speeds and flows.

APPENDIX

PUMP NUMBER 5 PERFORMANCE DATA

Performance TestPump #5

<u>Rotor Speed RPM</u>	<u>NaK Temp. °F</u>	<u>NaK Flow GPM</u>	<u>ΔP PSI</u>	<u>Torque IN - OZ</u>
10000	1000	1.70	0.45	0.72
		3.40	0.30	0.62
		4.38	0.10	0.62
		6.60	-0.10	0.62
		8.70	-0.40	0.64
		10.92	-0.90	0.64
		12.00	-1.00	0.64
20000	1000	1.70	0.85	1.90
		3.40	0.70	1.94
		4.38	0.55	1.94
		6.60	0.20	1.96
		8.70	-0.20	2.00
		10.92	-0.70	2.00
		12.00	-0.80	2.00
30000	1000	1.70	1.25	3.10
		3.40	1.00	3.16
		4.38	0.80	3.20
		6.60	0.60	3.20
		8.70	0.05	3.22
		10.92	-0.40	3.24
		12.00	-0.50	3.24
40000	1000	1.70	1.50	4.30
		3.40	1.30	4.30
		4.38	1.20	4.30
		6.60	0.90	4.30
		8.70	0.45	4.32
		10.92	-0.20	4.32
		12.00	-0.30	4.32
44000	1000	1.70	1.75	4.70
		3.40	1.40	4.74
		4.38	1.30	4.74
		6.60	0.95	4.80
		8.70	0.50	4.80
		10.92	-0.10	4.80
		12.00	-0.25	4.82

Secondary Losses
Pump #5

<u>Rotor Speed RPM</u>	<u>NaK Temp. °F</u>	<u>NaK Flow GPM</u>	<u>ΔP PSI</u>
0	1000	1.70	0
		3.40	0
		4.38	-0.05
		6.60	-0.40
		8.70	-0.50
		10.92	-1.10
		12.00	-1.30

Shutoff Head
Pump #5

<u>Rotor Speed RPM</u>	<u>NaK Temp. °F</u>	<u>NaK Flow GPM</u>	<u>ΔP PSI</u>	<u>Torque IN - OZ</u>
10000	1000	0	0.70	0.4
20000	940	0	0.90	1.90
30000	910	0	1.30	3.10
40000	875	0	1.70	4.4
44000	850	0	1.90	4.8

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