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SUBJECT:

NaK Free Convection Cooled Shaft Freeze Seal for SRE Pumps

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I STATEMENT OF PROBLEM

Determine the feasibility of using a NaK free-convection cooled shaft freeze seal on the SRE Main Sodium Pumps. The objective is to eliminate the carburization hazard present should the seal coolant (tetralin) leak into the sodium system.

If the concept is feasible, prepare a conceptual design of a prototype system for the SRE Main Sodium Pumps.

II SUMMARY OF RESULTS AND RECOMMENDATIONS

1.0 Results*

A NaK₅₆ free-convection cooled shaft freeze seal is feasible. A design has been made of a prototype system for the SRE Main Primary Pump (see Figure 1). The NaK system has the following advantages;

- (a) Does not depend on components mechanically moved
- (b) It is inexpensive as compared to the present tetralin cooled shaft freeze seal system.

* These results are based on the specified ground rules and assumptions outlined in Part III of this report.

** Sodium-Potassium alloy composed of 56 weight percentage of Potassium.

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Improvement to the existing freeze seal system was also made by reducing by 13% the flow of heat into the seal. This was accomplished by a heat barrier and a hollow pump shaft as shown in Figure 2. The computed contribution by each of the heat sources during maximum heat load (1250 RPM and 1000°F sodium) is tabulated below.

	BEFORE MODIFICATION	AFTER MODIFICATION	MODIFICATION MADE
TOTAL HEAT LOAD	3.0KW	2.6KW	-
	<u>% of total load</u>		
Convection in Na annulus	40.0	57.0	none
Shaft friction	23.0	33.0	none
Conduction through shaft	10.7	4.0	Hollow shaft
Conduction through Na annulus	1.3	2.0	none
Conduction through throat bushing wall	25.0	4.0	Reduced wall width & heat barriers

2.0 Recommendations

- The performance of the designed system be experimentally proven.
- A very low capacity heater be attached to the "Hot Legs" for use during the warm-up period. It is believed that the NaK flow will start in the required direction during the equipment warm-up period, before filling the system with sodium, but the heater will assure this.
- If the designed system performs in accordance with the freeze seal requirements, an experimental design optimization program should be undertaken.

III METHOD OF SOLUTION AND SAMPLE CALCULATIONS

1.0 Method of Solution

The NaK cooled freeze seal system has been investigated based on the following specified assumptions and ground rules:

Assumptions:

The most severe shaft freeze seal operation condition is at a sodium bulk temperature of a 1000°F, a pump speed of 1250 RPM, and an ambient air temperature of a 110°F.

Ground Rules:

- (1) If the system maintains the frozen seal at the most severe steady state condition it will at any other condition .
- (2) Overcooling the sodium frozen seal (This could seize the pump shaft) is not a problem.
- (3) The system should work the first time tested, therefore conservative safety factors should be used.
- (4) The system should not depend on components mechanically moved.
- (5) The system final goal is the SRE Main Primary Pump.

A trial-and-error method of solution was used which is summarized as follows:

(a) Assume a pump configuration (see Figure 2). Assume the sodium temperature at the bottom of the seal. Compute heat flow into the seal.

(b) Assume all of this heat is transferred to the NaK. Assume a NaK loop, a freeze seal, and a NaK-air heat exchanger configuration (see Figure 1 and 3). Assume an inlet NaK temperature into the freeze seal. Assume that no temperature drop occurs in either hot leg or cold leg of the NaK loop. Compute the NaK flow rate and ΔT required.

(c) Compute the size of NaK-to-air heat exchanger required. Compare this computed size with the assumed configuration used to compute flow rate.

(d) Assume a desired sodium temperature at the top of the NaK coolant jacket. Using the NaK inlet temperature, NaK ΔT , and NaK flow rate from above, determine the sodium temperature at the bottom of the NaK jacket. Compare with (a) above.

In the sample calculations presented below the configurations and dimensions finally arrived at (Figures 1, 2, 3) are used.

Nomenclature of all symbols is given on pages 16-18.

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2. Loop Configuration

The basic design configuration used is shown in Figure 1. A sodium-to-NaK heat exchanger freezes sodium to form the freeze seal. The NaK, in turn, is cooled by NaK-to-air heat exchangers in which the air flow is by convection.

The NaK portion of the shaft freeze seal, shown in Figure 3, has two inlet and two outlet NaK fittings arranged so that NaK flow can be from either inlet fitting to either exit fitting. The remainder of the NaK circulation system contains two identical loops, each containing a NaK-to-air heat exchangers.

As shown in Figure 2, the basic configuration contains several modifications to the present SRE pump design. Chief among these is the incorporation of a heat barrier and hollow pump shaft to reduce heat flow to the seal, and the relocation of the seal to a higher location. These changes are possible in the present SRE pump without removing the pump case.

3. Sample Calculations *

A. Heat Flow into the Seal

- (1) Assume that the sodium temperature at the bottom of the freeze seal is 400°F and that the temperature of the pump shaft at this level is also 400°F.

- (2) Shaft friction heat. ⁽³⁾

$$f = 1.41 \times 10^{-10} \mu N^2 D^3 \left(\frac{L_{SA}}{t} \right)$$

$$= 1.41 \times 10^{-10} (0.580) (1250)^2 (3.00)^3 \frac{10}{0.4 \times 10^{-4}}$$

$$= 865 \text{ watts}$$

$$f = \underline{2950} \text{ BTU/hr.}$$

- (3) Conduction through Hollow Shaft Wall

$$q_{D,HS} = \frac{T_{BS} - T_{BT}}{\frac{L_2}{K_{SS} A_1} + \frac{L_1}{K_{SS} A_1}}$$

* Nomenclature is given on page 16.

$$q_{D,HS} = \frac{(1000 - 400)}{\frac{3.5(4)144}{12(11)\pi[(3)^2 - (2.5)^2]} + \frac{(1.5)(4)(144)}{(12)(11)(\pi)(3)^2}}$$

$$q_{D,HS} = \underline{300} \text{ BTU/hr.}$$

(4) Radiation Through Hollow Shaft

For simplification purposes, the heat transferred through the shaft by conduction and radiation has been considered separately. A 1000°F is being assumed at the bottom of the hollow shaft for the calculation of the radiated heat. This is not true, but the difference will be negligible and it will be on the safe side.

$$q_{R,HS} = \sigma F_a F_e A''_{HS} (T_{1000}^4 - T_{600}^4)$$

$$F_a = 1 \text{ and } F_e = \frac{1}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1} = 0.375$$

where $\epsilon_1 = 0.600$
 $\epsilon_2 = 0.500$

$$q_{R,HS} = 0.173 \times 10^{-8} (1)(0.375) \frac{\pi}{4} \left(\frac{2.50}{12}\right)^2 [(1460)^4 - (860)^4]$$

$$q_{R,HS} = \underline{89.0} \text{ BTU/hr.}$$

(5) Conduction Through Sodium Annulus

$$q_{SA} = \frac{K_{Na}}{L_1 + L_2} A^1_{SA} (T_{BS} - T_{BT})$$

$$q_{SA} = \frac{43}{5/12} \frac{\pi}{4} \left[\frac{(3.090)^2 - (3.0)^2}{144} \right] (1000 - 400) = \underline{170} \text{ BTU/hr.}$$

008 006

(6) Convection in Sodium-Filled Annulus

The heat transferred to the seal by convection in the sodium-filled annulus is expected to be very large as compared to that by conduction. This has been experimentally proven⁽⁷⁾, but the configuration and operating conditions of the sodium-filled annulus that was analyzed and tested⁽⁷⁾ were different and the results are not directly applicable here.

Since a thorough analysis of thermo-convection for our condition will be extremely complicated an estimate was made. Data on the present SRE shaft freeze seal configuration⁽⁸⁾ shows the ratio of convection to conduction in the sodium-filled annulus to be about 30. This value is used to compute the convection in the sodium-filled annulus in the modified design presented herein.

$$q_{\text{conv.}} = 30 q_{\text{SA}}$$

$$q_{\text{conv.}} = 30 (170) = 5100 \text{ BTU/hr.}$$

(7) Conduction Through Throat Bushing Wall

$$q_w = \frac{K_{SS}}{L_1 + L_2} A_w^1 (T_{BS} - T_{BT})$$

$$q_w = \frac{11.0}{5/12} \frac{\pi}{4} \frac{(3.59)^2 - (3.09)^2}{144} (1000 - 400)$$

$$q_w = \underline{285} \text{ BTU/hr.}$$

(f) Total Heat Flux into Seal

$$q_T = f + q_{D,HS} + q_{R,HS} + q_{SA} + q_w + q_{\text{conv.}} = 8894 \text{ BTU/hr.}$$

$$q_T = \underline{2.6} \text{ kw}$$

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For laminar, free-convection flow under steady state condition,

Head available = system pressure losses

$$H \beta \rho_H \Delta T_{NaK} = \Delta P_L + \Delta P_{NA} + \Delta P_{KA}$$

For the calculation of pressure loss in annuli under isothermal laminar flow, Mac Adams⁽⁴⁾ uses:

$$\Delta P = \frac{32 \mu G L N A}{\rho g \left[D_2^2 + D_1^2 - \frac{D_2^2 - D_1^2}{\ln \frac{D_2}{D_1}} \right]}$$

where $G = \rho V$

The physical properties of the NaK flowing through the jacket annuli are averaged to use the above equation.

For pressure losses in the loop where the conduits will be of circular cross section, Darcy's Equation is used⁽⁶⁾:

$$\Delta P_L = \rho \frac{64}{R_e} \frac{L_p}{D_1} \frac{V_p^2}{2g}$$

In this computation the ΔP_{KA} term is combined with the ΔP_L term.

Then,

$$H \beta \rho_H \Delta T_{NaK} = \rho \frac{64}{R_e} \frac{L_P}{D_1} \frac{V_P^2}{2g} + \frac{32 \mu_{NA} V_{NA} (2L_{NA})}{\rho g \left[D_2^2 + D_1^2 - \frac{D_2^2 - D_1^2}{\ln \frac{D_2}{D_1}} \right]}$$

Substituting $R_e = \frac{\rho V_P D_1}{\mu}$, $V_P = \frac{4W}{\rho \pi D_1^2}$, $V_{NA} = \frac{8W}{\rho_{NA} \pi (D_2^2 - D_1^2)}$

in the above equation, cancelling equal terms and solving for W we get:

$$W = \frac{2.45 \times 10^{-2} \rho_{ave}^H \beta \rho_H g}{\left\{ \frac{L_P}{D_1^4} + \frac{4L_{NA}}{(D_2^2 - D_1^2) \left[D_2^2 + D_1^2 - \frac{D_2^2 - D_1^2}{\ln \frac{D_2}{D_1}} \right]} \right\}} \Delta T_{NaK}$$

The assumed system dimensions are,

$L_P = 70\text{ft.}$ this includes equivalent length for bends, expansion, contraction, length of the fin-tubing, etc., for either of the two NaK loops.

$$D_1 = 1/2"$$

$$L_{NA} = 5"$$

$$D_2 = 3.6524"$$

$$D_1 = 3.340"$$

$$H = 15 \text{ ft.}$$

It is also assumed that the inlet NaK temperature at the freeze seal is 130°F.

The above dimensions and temperature were used to solve the equation, $W = f(\Delta T_{NaK})$. The parameters μ and β , were adjusted to agree with the temperatures used in each solution, and a graph of the results is shown in Figure 4.

Substitution of the above equation, $W = f(\Delta T_{NaK})$ into the basic heat equation, $Q = WC_p \Delta T$, gives an expression for Q as a function of ΔT_{NaK} . A set of values of this relationship was also found (μ and β were again varied with temperature) and plotted in Figure 4.

The ΔT_{NaK} and W_{NaK} in each loop is obtained from figure 4 by using half of the value of $Q = 2.6KW$ computed in section B above. This means, that it has been assumed that each loop of the two-loop-freeze seal design will remove half of the total heat load.

$$\Delta T_{NaK} = \underline{70^\circ F}$$

$$W_{NaK} = \underline{250 \text{ \#/hr.}} \text{ (for each NaK loop)}$$

C - Size of NaK-Air Heat Exchanger

The NaK temperature profile throughout the NaK air heat exchanger section will be approximately the following,

So, by using the equation;

$$Q = [UA''(\Delta T)_{\ln}]_{K-A}$$

where $A''_{K-A} = \pi D_o L_{FP}$, and where $(\Delta T)_{\ln}$ is the difference in the maximum and minimum ΔT divided by the difference in the natural logarithm of these maximum and minimum. The length of fin-tubing (L_{FP}) required to reject the heat into the atmosphere can be obtained.

The heat transfer characteristics of NaK and the thermal conductivity of the stainless steel pipe wall are large compared with the heat transfer film coefficient on the air side of this NaK-air heat exchanger. Thus, the latter is the predominant factor in the value of U_{K-A} . For this reason an extended surface (Fin-Tubing) in contact with the air is used to improve heat transfer.

Selecting a Fin-Tubing as follows:

1/2" pipe-sched. 40
7/16 Fin height
0.012 Fin thickness
8 Fin/inch (Pitch)
0.471 Fin width

The total area in contact with the air will be equal to the fin surface area plus the bare tube area.

$$\begin{aligned}\text{Fin surface area}^{(5)} &= 0.523(1 + \frac{t}{w})h_f(D_o)(P) \\ &= 0.523(1 + \frac{0.012}{0.171})7/16(0.84)(8) = 1.65 \text{ ft}^2/\text{ft}\end{aligned}$$

$$\begin{aligned}\text{Bare tube surf. area}^{(5)} &= 0.262 D_o P(\frac{1}{P} - t) \\ &= 0.262(0.84)(8)(1/8 - 0.012) = 0.198 \text{ ft}^2/\text{ft}\end{aligned}$$

$$\text{Total air-side area} = 1.85 \text{ ft}^2/\text{ft}$$

$$\text{Area in contact with NaK (inside area)} = 0.262 D_i = 0.163 \text{ ft}^2/\text{ft}$$

$$\text{Area Ratio} = \frac{\text{Air-Side Area}}{\text{NaK-Side Area}} = 11.4$$

Higher area ratios can be obtained by selecting bigger fin dimensions. The above dimensions were chosen because they are shelf items.

Reference (5) contains a graph relating air heat transfer film coefficients, the temperature difference between the average tube wall temperature and the ambient air temperature, and tube inside diameter.

For the assumed configuration it predicts,

$$h_a = 2.5 \frac{\text{BTU-ft.}}{\text{hr-ft}^2 \cdot ^\circ\text{F}}$$

Then,

$$U_{KA} = h_a \times \text{area ratio} = 28.5 \frac{\text{BTU}}{\text{hr-ft}^2 \cdot ^\circ\text{F}}$$

$$Q_R = 28.5 A''_{KA} \frac{(200 - 110) - (130 - 100)}{\ln \frac{90}{20}} = 2.6(3413) \frac{\text{BTU}}{\text{hr}}$$

$$A''_{KA} = 6.20 \text{ ft.}^2$$

$$\text{Since } A''_{KA} = D_o L_{KA} = \pi \left(\frac{0.840}{12} \right) L_{KA} = 6.00 \text{ ft}^2$$

$$L_{KA} = 28.2 \text{ ft.}$$

Since the system proposed has two identical NaK loops, the heat exchanger for each loop requires 14.1 ft. of finned tube.

The system proposed allows 30 ft. of finned tube for each loop. Therefore, a safety factor of above two in heat exchanger size is provided..

D. Check on Assumed Sodium Temperature at Bottom of Seal.

As a requirement of the design fix the sodium temperature at the top of the freeze seal at 150°F

The NaK inlet temperature into the coolant jacket has been fixed at 130°F and the ΔT_{NaK} was computed to be 70°F at a flow rate of 250 #/hr for each of the two NaK loops.

For the proposed system the NaK-Na Heat Exchanger is of the counterflow type. The temperature profile in the NaK and sodium sides, from the bottom of the coolant jacket to the top of the frozen sodium (seal), will approximate the following;

The basic heat transfer equation is,

$$Q_A = [A''(\Delta T)_{lm}] S-K$$

where

$$U_{S-K} = \frac{1}{\frac{a}{K_{Na}} + \frac{b}{K_{SS}} + \frac{1}{h_{NaK}}}$$

To calculate h_{NaK} no theoretical or empirical equation could be found which is directly applicable. Other equations partially applicable were tried; and of these equations the one below yields the lowest value for h_{NaK} (i.e., it predicts the smallest amount of heat will be transferred). For laminar free convection flow about a vertical cylinder at constant heat flux: ⁽²⁾

$$NU_{NA} = \frac{0.546 PR^{\frac{1}{2}} GR^{\frac{1}{4}}_{NA}}{(0.800 + PR)^{\frac{1}{4}}}$$

where

$$NU_{NA} = \frac{h_{NaK} L_{NA}}{K_{NaK}} \quad \text{and} \quad L_{NA} = 5''$$

NaK Physical Properties

	130°F	200°F	Ave.
K	12.4	13.0	12.7
μ	1.60	1.35	1.48
ρ	54.5	54.0	54.3

$$PR = \frac{\mu C_p}{K} = 0.020$$

$$GR_{NA} = \frac{\rho^2 L_{NA}^3 \beta g \Delta T_{NaK}}{\mu^2} = 4.5 \times 10^8$$

where

$$\beta = \frac{(\rho_c - \rho_H)}{\rho_H \Delta T}$$

So,

$$NU_{NA} = \frac{0.546(0.020)^{1/2}(4.7 \times 10^8)^{1/4}}{(0.800 + 0.020)^{1/4}} = 12.2$$

$$h_{NaK} = \frac{(12.2)(12.7)(12)}{5} = 370 \frac{\text{BTU-ft}}{\text{hr-°F-ft}^2}$$

$$U_{S-K} = \frac{1}{\frac{0.045}{12(45)} + \frac{0.125}{12(11)} + \frac{1}{370}} = 268 \frac{\text{BTU-ft}}{\text{hr-°F-ft}^2}$$

$$(\Delta T_{ln})_{S-K} = \frac{\Delta T_{max} - \Delta T_{min}}{\ln \frac{\Delta T_{max}}{\Delta T_{min}}} = \frac{(T_{BT}-200)-20}{\ln \frac{T_{BT}-200}{20}}$$

$$Q = U \pi D L (\Delta T)_{\ln}$$

$$2.6(3413) = (268) \pi \left(\frac{3.59}{12} \right) \left(\frac{5}{12} \right) \frac{(T_{BT} - 200) - 20}{\ln \frac{T_{BT} - 200}{20}}$$

$$84.0 = \frac{T_{BT} - 220}{\ln \frac{T_{BT} - 200}{20}}$$

$$T_{BT} = 423^{\circ}\text{F}$$

This temperature is 23 °F higher than the 400°F originally assumed as a basis for calculations. If the heat exchangers are sized for a given amount of heat transfer at 423°F, they can transfer more heat at 400°F. If a given amount of conductive and convective heat flow from the pump impeller region occurs at 400°F, less will occur at 423°F. Therefore, the assumption of 400°F is conservative.

F. Safety Factors

The ultimate criteria of successful seal operation may be taken as the ability of the seal to maintain the upper end of the sodium annulus at 150°F. In addition to numerous minor assumptions the following major design safety factors will each tend to make the upper end of the sodium annulus run colder than 150°F:

- (1) Heat transfer from the sodium annulus into areas other than the NaK coolant is ignored.
- (2) The NaK-air heat exchanger is (twice) larger than necessary to reject the computed flow of heat into the seal.
- (3) The temperature assumed to exist at the lower region of the seal is lower than a back-check predicts.

308 015

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References

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Nomenclature

Subscripts: a - air
BS - Bulk sodium in pump impeller region
BT - Sodium in the shaft annulus at the level
of the bottom of the coolant jacket
c - cold
conv.- convection
FP - Finned Tubing
H - Hot
HS - Hollow Shaft
KA - NaK to air (heat exchanger section)
L - NaK loop piping
NA - NaK annuli jacket
Na - sodium
P - Piping
R - Radiation
S - Solid Shaft
SA - Sodium Annulus
SK - Sodium to NaK (heat exchanger section)
SS - Stainless Steel
T - Total
W - Throat bushing wall

A - Fluid Flow Area, Ft.²
A¹ - Heat Transfer Cross Sectional Area, Ft.²
A" - Heat Transfer Surface Area, Ft.²
a - Radial Sodium Annuli Gap, Ft.
b - Coolant Jacket Wall Thickness, Ft.
C_p - Heat Capacity, BTU/#-°F
D - Shaft Diameter, inches
D₂ - Coolant Jacket Annuli Outside Diameter, Ft.
D₁ - Coolant Jacket Annuli Inside Diameter, Ft.
D_o - Outside Pipe Diameter, inches
D_i - Inside Pipe Diameter, inches
f - Shaft Friction Heat, BTU/hr.

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- G - Mass Velocity #/sec-ft.²
H - Vertical Distance from Freeze Seal to NaK-Air Heat Exchanger, Ft.
h - Heat Transfer Film Coefficient, BTU/Hr-Ft.²-°F
h_f - Fin Height, inches
K - Coefficient of Thermal Conductivity, BTU/Hr-°F-Ft
L - Axial Length, inches
N - Shaft Speed, RPM
ΔP - Pressure Loss, #/Ft.²
ΔP_H - Thermal Head Available, #/Ft.²
Q - Heat Transferred in KW
q - Heat Transferred in BTU/Hr.
T - Temperature, °F
T^o - Temperature, Degree rankine
t - Film of Liquid Shaved by Shaft (usually the shaft surface roughness)-ins.
t¹ - Fin Thickness, inches
ΔT - Temperature difference across a Region as Shown by Subscript,
U - Overall Heat Transfer Coefficient, BTU/Hr.-°F-Ft.²
V - NaK velocity, Ft/Hr.
W - NaK flowrate, #/Hr.
w - Fin Width, inches
β - Coefficient of volumetric Expansion, °F⁻¹
ρ - Fluid density, #/Ft.³
μ - Absolute Viscosity #/Hr-Ft.
μ_i - Viscosity of Sodium at Melting temperature - Centipoise
σ - Stephon-Boltzman Constant = 0.173 x 10⁻⁸ BTU/Hr-°R⁴-Ft.²

Dimensionless Numbers

- Re - Reynolds
Nu - Nusselts
Pr - Prandtl's
Gr - Grashof's
L/D - Equivalent Diameter
Fa - Radiation Configuration Factor for the Hollow Shaft.

F_e - Radiation Emissivity Factor for the
Hollow Stainless Steel Shaft.

K^1 - Fluid Flow Resistant Coefficient for
Expansion or Contraction

P - Fin Pitch

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TECHNICAL DATA RECORD

TDR NO. 3984

DATE June 18, 1959

PROGRAM

PROJECT

PAGE 19 OF 22

NaK - Convection Cooled Shaft Freeze Seal System
(SRE Main Primary Pump)

NOTES

- 1-This sketch shows only the overall assembly of the system. For the relative position of the system components and their dimensions see Figures 2 & 3.
- 2-The coolant jacket lines (inlet and outlet) are located radially 90° apart.
- 3-Loop lines should be insulated as shown.

FIGURE 1

310 220

Heat Barriers for Shaft Freeze Seal

NOTE: The configuration shown below modifies the present SRE Pumps as follows: (1) Seal moved to a higher elevation, (2) Gas space heat barrier, (3) Hollow shaft.

FIGURE 2

3(a) Annuli Jacket (Seal), Assembly and Dimensions

3(b) NaK-Air Heat Exchanger, Assembly and Dimensions

Cooling Load - KW

FIGURE 4