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DESIGN AND DEVELOPMENT OF A CANNED-MOTOR PUMP FOR HIGH TEMPERATURE NaK SERVICE IN SNAP-8

By

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AEROJET-GENERAL CORPORATION
Azusa, California

prepared for

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

NASA Lewis Research Center
Contract NAS 5-417
Martin J. Saari, Project Manager

NOTICE

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TOPICAL REPORT

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FOREWORD

The work described in this report was performed by Aerojet-General Corporation, Azusa, California, as part of the SNAP-8 electrical generating system contract being conducted within the Power Systems Department. The work was directed under NASA Contract NAS-5-417, with Mr. Martin J. Saari as NASA Program Manager, and Dr. W. F. Banks as Aerojet-General Corporation Program Manager. Acknowledgement is given to Messrs. A. Stromquist, James Dunn, and H. O. Slone, of the Space Power Division, NASA-Lewis Research Center for their valuable assistance in the NaK pump motor development program.

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ABSTRACT

A pumping system for high temperature (1170°F) service in a nuclear-electrical power conversion system (SNAP-8) for use in space was designed and developed. The service fluid was a eutectic mixture of liquid sodium and potassium (NaK 78). The pumping system consisted of a pump-motor combination operating on NaK-lubricated hydrodynamic bearings, and a cooling-purification system which circulated filtered and cooled NaK lubricant. The lubricant NaK is drawn from the primary pumped fluid. Tests totaling 31,136 hours (as of December 1968), including an endurance run of 10,362 hours with 786 startups, have been conducted. The pumping system has met the design objectives, including the 10,000-hour life requirements.

Examination of the detailed parts of the pump motor after the endurance test showed that the only damage on the pump-motor components were minor surface scratches on the running surfaces of the tool steel radial bearings. The scratches were believed to be caused by the passage of small debris particles due to inadequate loop cleanliness. It should be noted that the tungsten carbide thrust bearings which were exposed to the same liquid environment as the radial bearings have shown no evidence of scratch damage. With the resolution of this problem there appears to be no life limiting features of the pump-motor.

SUMMARY

A liquid NaK pump was designed and developed for the heat transfer loop in the SNAP-8 nuclear turbo-electric space power system. Design of the NaK pump was started in 1963 with development continuing to the present.

The fluid pumped by the unit was a eutectic mixture of sodium and potassium (NaK) at temperatures of up to 1170°F. The life requirement of the unit was to operate continuously for 10,000 hours with no maintenance.

The pump motor combination consists of a centrifugal pump driven by a 400 Hz squirrel cage induction motor. The rotating parts are mounted on a single shaft supported by hydrodynamic NaK lubricated bearings.

Several NaK pumps were built and tested for a total of 31,136 hours with the maximum time accumulated on one unit of 10,362 hours with 786 start cycles. In addition to this endurance test, other pumps were operated in SNAP-8 system test facilities at Aerojet-Azusa and NASA-LeRC, Cleveland.

Detailed examinations of the pumps, which were performed following the various tests, confirmed their ability to meet the design objectives, including the extended life requirement of 5 years without maintenance, this being the later life goal for the SNAP-8 system.

Problems encountered during the development process included leaks through the pump housing walls due to casting porosity. The casting design was later changed to a pump housing design machined from forging blanks.

Also some problems in the canned stator winding end turns causing winding failures were experienced. This was mainly due to space restrictions in the end turn area. It was resolved by allowing more room for the end turns and an improvement in the insulation techniques. Operational problems were also encountered, sometimes resulting in gas entering the motor cavity, thus causing loss of recirculation flow and consequent overheating.

Inadequate venting of the pump during NaK filling at the test facilities was also responsible for ineffective cooling of the motor. More stringent operating procedures were applied and these problems were then resolved. The NaK pump is now considered to be developed to the extent that all the SNAP-8 performance and endurance requirements have been adequately demonstrated.

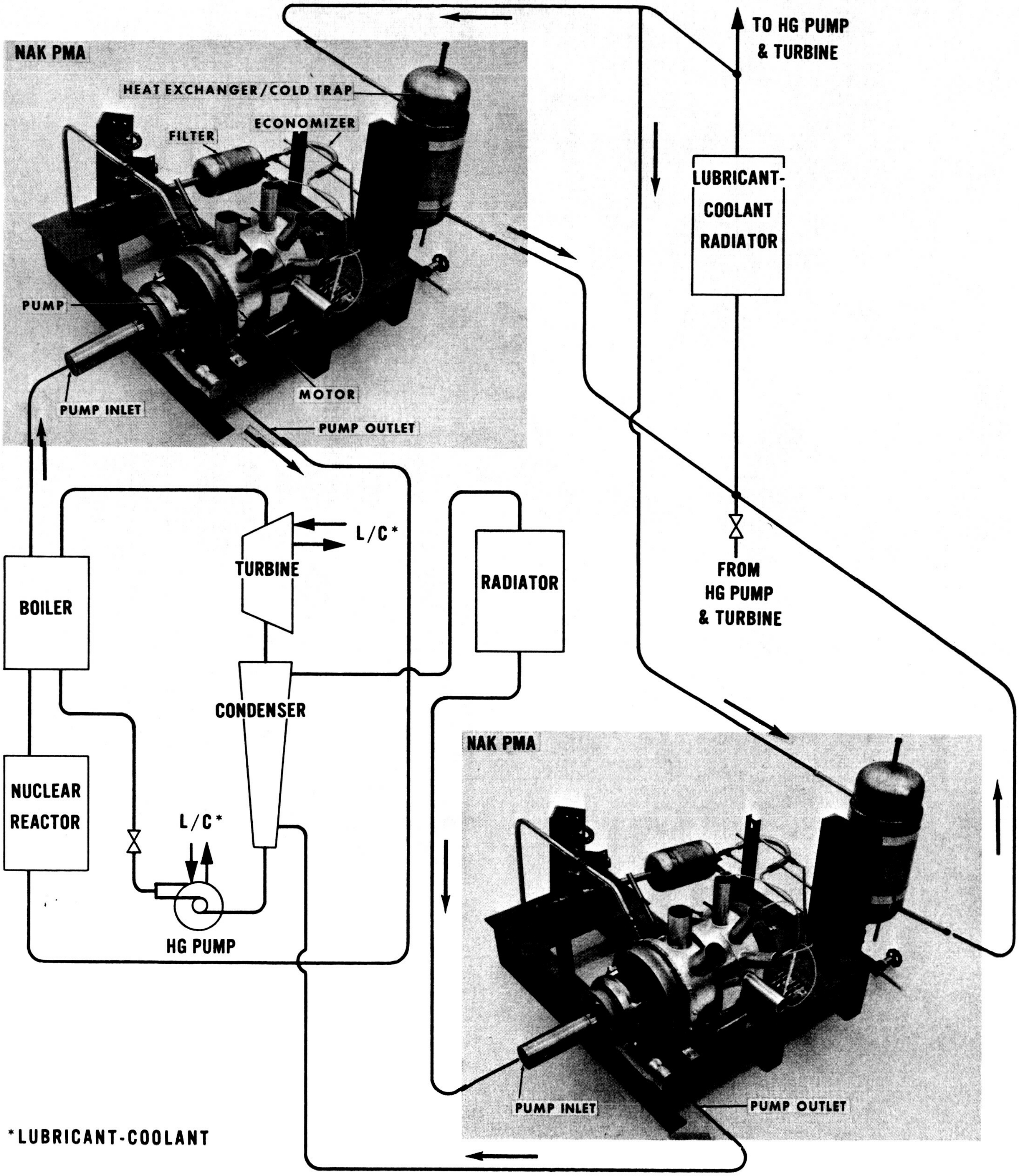
I. INTRODUCTION

The System for Nuclear Auxiliary Power (SNAP-8) program was initiated by the National Aeronautics and Space Administration to develop power systems for use in outer space. SNAP-8 was the designation of the system that was awarded to the Power Systems Department of the Aerojet-General Corporation (AGC) under Contract NAS 5-417. Under the guidelines for SNAP-8, AGC was to design and develop a system which would convert heat from a nuclear reactor into 35 kw of useful electrical power. The components were to be designed for a life of 10,000 hours (unattended) and to be within current technology.

The SNAP-8 system has two loops which use a eutectic mixture of sodium and potassium (NaK 78) as the heat transport fluid (Figure 1). One loop is used to transport heat from the heat source (a nuclear reactor) to the mercury boiler at a temperature between 1100 and 1300°F; the other loop is used to transport heat from the condenser to the radiator at temperatures between 465 and 680°F. The pump described in this report is used to circulate the NaK in these loops, one pump, (common design) being provided for each loop.

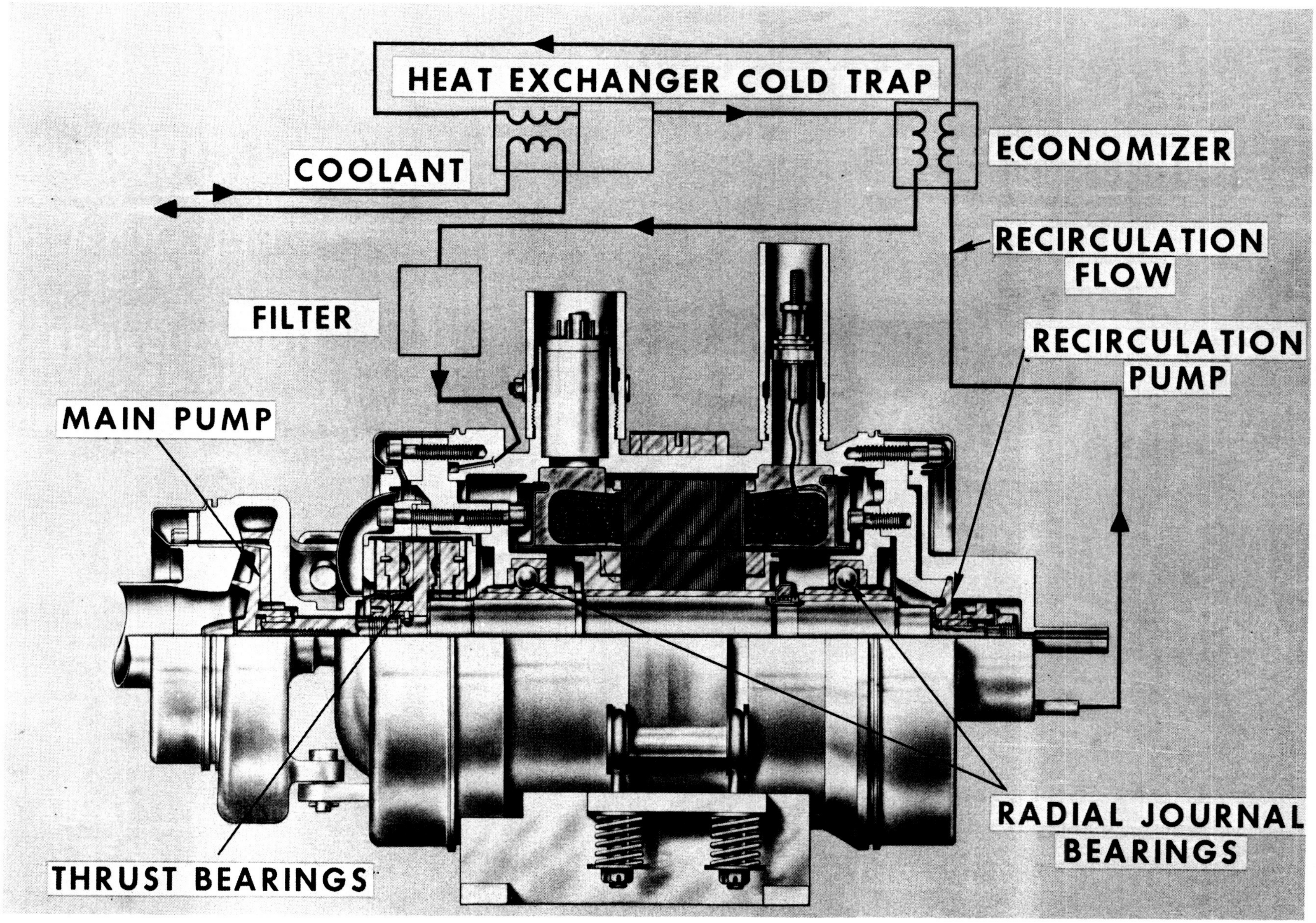
The initial requirements of the two loops was considered sufficiently different to warrant two pump designs. However, as the system evolved into the present configuration, it became evident that one common pump design would perform both NaK loop functions.

In the design of the pump-motor (Figure 2), the main problems were the designing of a system which would keep the motor sufficiently cool and thermally isolated from the hot NaK to pump NaK at temperatures up to 1170°F, the designing of NaK-lubricated bearings, and the designing of reliable components that would perform without failure for 10,000 hours. During the development of the pump-motor, facilities were also developed which provided the proper environment for the testing of the pump.



Schematic of Nak Pump Motors in the SNAP-8 System

Figure 1



Cross Section Drawing of the Nak Pump-Motor Assembly

Figure 2

As of December 1968, pump-motors have been tested for a total of 10,362 hours. The major problems attributed to the component itself have been resolved. The only outstanding deficiency is related to maintaining adequate cleanliness in the component and the facility in order to prevent damage to the radial bearings.

The NaK pump-motor can be considered as qualified for the SNAP-8 system with a life potential far exceeding the 10,000-hour contract requirement.

II. DESIGN

A. SYSTEM REQUIREMENTS

General design requirements of the pumping system are:

Fluid: Eutectic mixture of sodium and potassium (NaK 78)

	Pump Systems	
	Heat Rejection Loop	Primary Loop
Flow, gal/min.	98.7	125
Head rise across pump, ft	118	99
Temperature at pump suction °F	495	1170
Net positive suction head (NPSH) available, ft	42.5	42.5
Minimum efficiency, %	36	38
Unattended life, hr	10,000	
Design approach	1963 Technology	
AC Voltage, rms, line-to-line (3 phase)	208	
Frequency, Hz	400	

The following system design requirements were imposed:

1. Minimum Weight with Maximum Reliability

High reliability and low weight were the prime requirements for the pumping system because the pump-motor assembly must operate unattended in outer space. A centrifugal pump-motor combination was selected because of the light weight and high reliability. Electromagnetic pumps were not considered because of weight factors and inefficiency. The design requirements also eliminated the positive-displacement pump because sliding parts in positive placement pumps will wear, which was considered life limiting.

2. Leakage Allowed from the Pumping System

The design requirements stressed that leakage of the pumped fluid from the pumping system would not be permitted. Canned motor construction and use of NaK lubricated bearings was therefore indicated.

3. No Intermixing of the Fluids in the Two Loops

The intermixing of the fluids of the two NaK loops would not be acceptable because the primary loop NaK would have been made radioactive by the nuclear reactor, and radioactivity would not be acceptable in the heat rejection loop. By having separate pumps for each of the loops, intermixing cannot occur. This requirement eliminated the possibility of using one pump for both loops, or using one motor with two pumps.

B. PUMP DESIGN

1. General

The pump design began with the selection of a pump speed. The selection was made on the basis of the type of power available, the motor design requirements, and high pump efficiency. The pump speed was optimized at 6,000 rpm based on the 400 Hz power supply, number of motor poles, and bearing design drag losses. At the applied torque, the motor slip gave an actual shaft speed of 5840 rpm. This speed, together with the head and flow requirements of the pump, resulted in a "specific speed" of approximately 1,800.

The specific speed is defined as:

$$N_s = \frac{NQ^{1/2}}{H^{3/4}}$$

where N is the pump speed in rpm

Q is the pump flow in gallons/min.

H is the head rise across the pump in feet

The specific speed parameter is used by designers to classify pumps by their similarity in hydraulic characteristics. For a specific speed of 1800, the best type pump design would be one with a mixed flow impeller. The net positive suction head required for this pump was calculated to be 15 feet. Since the available NPSH to this pump is 42.5 feet, a very conservative suction performance margin existed. The pump was designed and the pump performance predicted using a design approach based on Reference 1.

2. Impeller

a. Number of vanes

The number of vanes was selected on the basis of maximum fluid flow control with minimum friction losses. For maximum fluid control, a high number of vanes would be desirable. However, a large number of vanes would cause high friction losses. A trade-off was made of these factors and giving consideration to tooling and manufacturing limitations, a 5-vane design was selected.

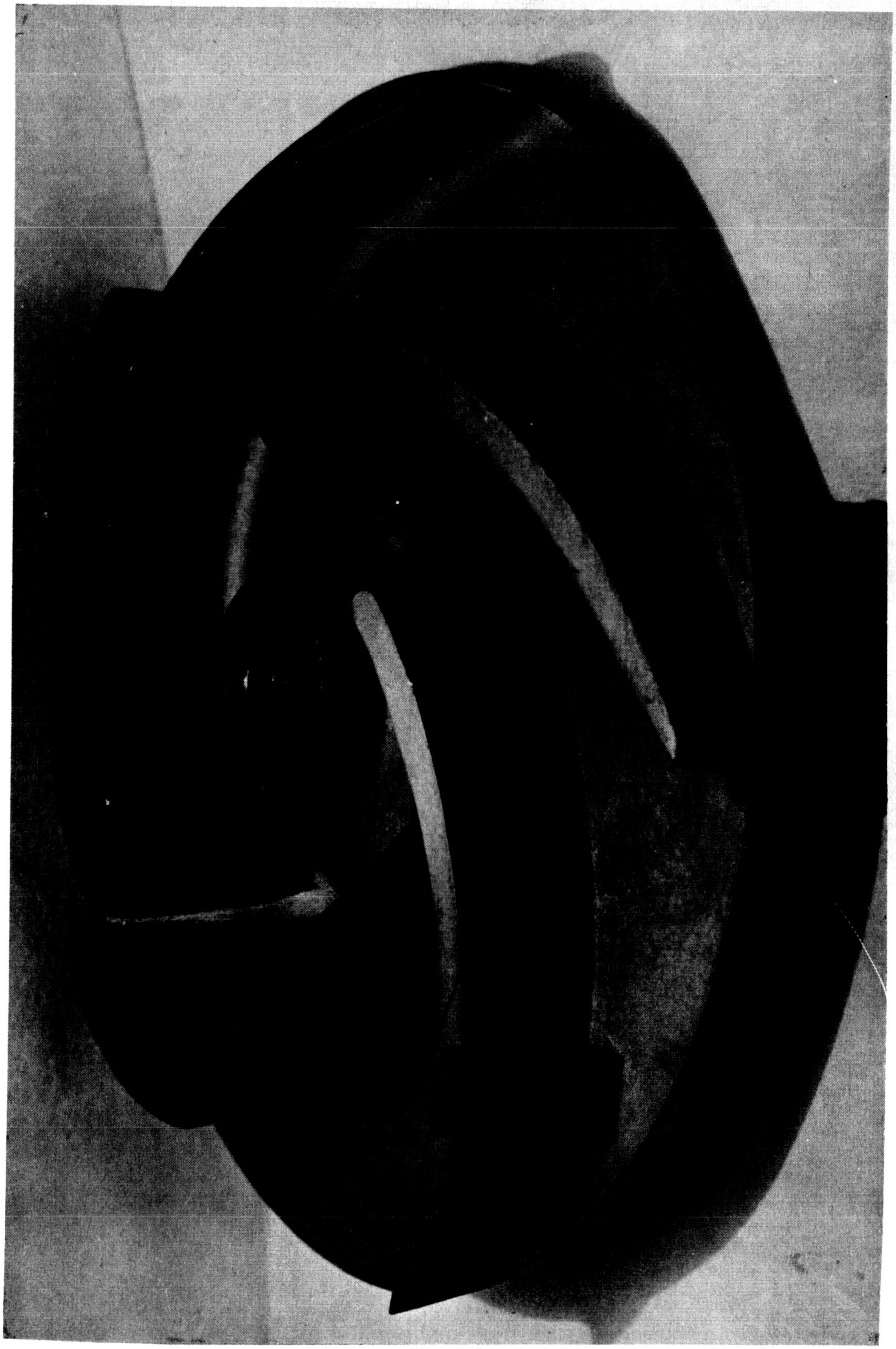
b. Impeller shrouding

Mixed flow impellers for pumps can be shrouded or semi-open. The semi-open impeller design tends to have a high axial thrust due to pressure imbalance across the impeller and would require back vanes to neutralize it. The shrouded impeller design is sensitive to radial clearance, and it must operate at clearances higher than optimum because of the possibility of galling or fouling due to oxide particles. With these considerations in mind, a semi-open impeller design was selected.

c. Impeller hydraulic design

The inlet was designed in accordance with a free vortex pattern with an allowance for 115% prerotation of the fluid before it hits the impeller to maximize the impeller efficiency. Details (Figure 3) of the impeller are as follows:

• Type	Semi-open, mixed flow
• Inlet eye diameter, in.	1.75
• Mean discharge diameter, in.	3.54
• Front seating angle	25°
• Vane entrance angle	24° 30'
• Vane discharge angle	22° 30'
• Vane height at discharge, in.	0.30



Pump Impeller

Figure 3

d. Back Vane Design

Back vanes were incorporated to balance the axial thrust loads due primarily to forces imposed by front vane pressures. The balanced load was approximately zero net thrust at rated flow with back vanes incorporated. Without back vanes, the net thrust would have been approximately 160 lb.

Techniques for back vane design described in Reference 1 were applied with the resulting design being as follows:

• Number of back vanes	24
• Back vane height	0.13 inches
• Back vane clearance	0.040 inches*

C. MOTOR DESIGN

The motor must provide driving power to both the main pump and the recirculation pump during the system startup and steady-state operation. The motor must operate in a radioactive, high-temperature environment, use NaK as a bearing lubricant, and operate with power inputs that vary from 95 Hz and 19 volts for initial startup to 400 Hz and 208 volts for steady-state running. The motor design selected to perform this function was an 8-pole, squirrel-cage induction machine. This selection was based on the overall pump-motor optimization which took into account such factors as frequency, viscous losses, and allowable space for fabricating the required poles.

1. General Approach

The design parameters of an electrical motor such as electrical power generation, speed of rotation, bearing type and configuration, and cooling methods are interrelated. The speed of rotation and the bearing design are fixed by system consideration. That leaves the electromagnetic sizing and cooling requirements to be established.

An induction motor consists of a laminated stator and rotor. The laminated stacks are made of electrical sheet steel and are slotted for windings. The cylindrical volume determined by the outside diameter and the stack length is the volume of a motor; and it is within this volume that the torque is generated. The amount of torque that can be generated in a given

*This is the final value selected as described later in this report.

volume depends on the geometry and electromagnetic properties of the laminations.

The heat rejection effectiveness of a motor is equal in importance to the electromagnetic design when determining the final machine size. Where light weight and small size are desirable, effective cooling must be provided so that the temperatures in the bearings and the insulation are within their limits. The NaK pump-motor is to be used in an outer space environment where air cooling is not available, therefore an active cooling system must be provided.

a. Motor

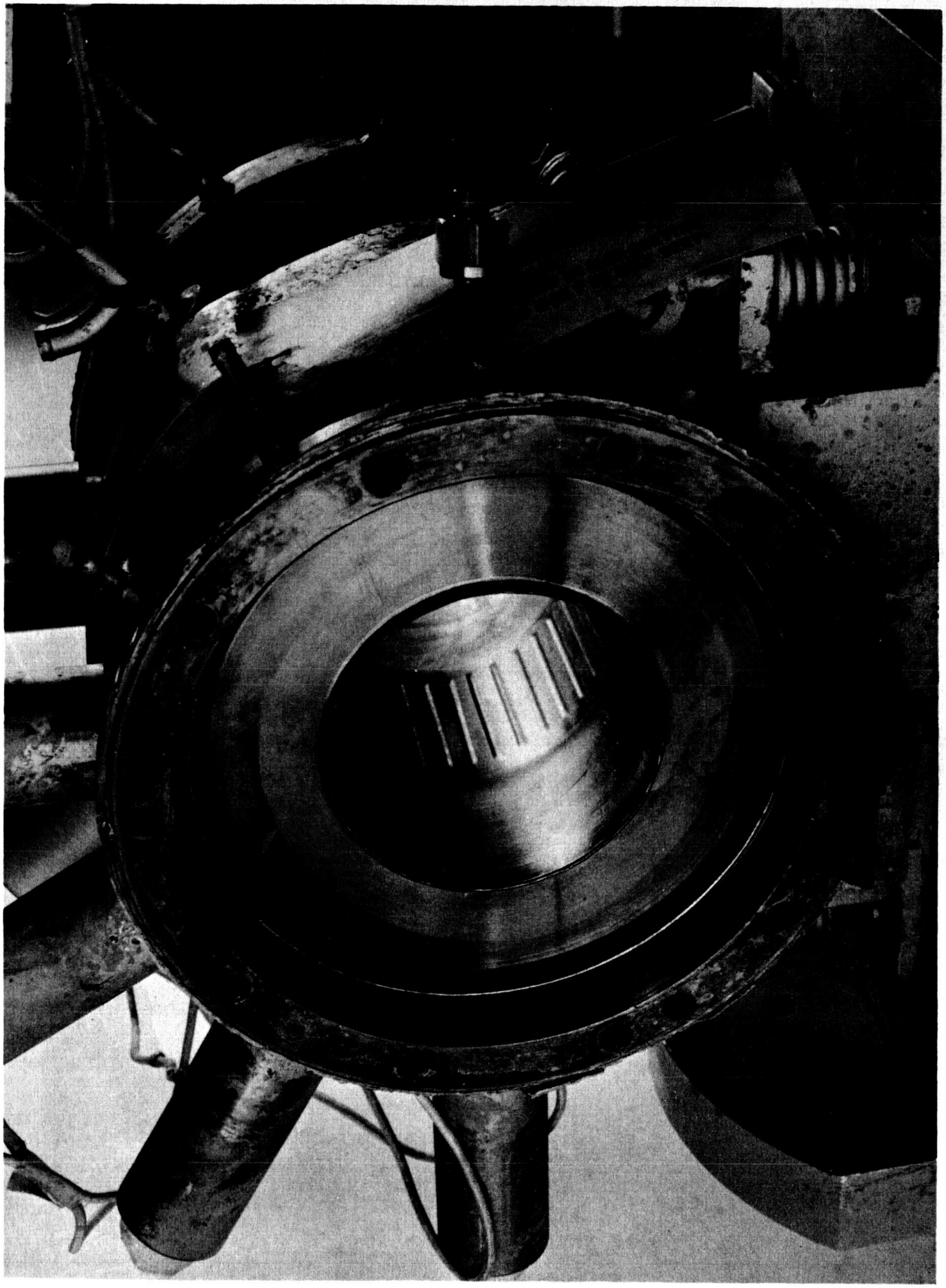
The motor design was established on the basis of standard motor design practice. The diameter and length of the rotor were first optimized. Then the number of slots in the rotor and the stator and their dimensions, and the end ring dimensions were determined. The factors which describe the type of winding are optimized with respect to the copper areas, magnetic circuit cross section, and the number of turns required for a predetermined magnetic density. The air gaps were set by the mechanical design requirements and the fabrication limits of the cans.

The startup torque required of the motor was assumed to be low because the motor would drive a mixed-flow, impeller pump; the essential characteristic of centrifugal pumps is that the torque required is proportional to the square of the operating speed.

b. Stator

The stator was designed to be sealed by an Inconel can. It consists of a 36-slot laminated core of AISI M-22 steel handwound with copper wire (Figure 4). The stator windings were series wound for higher winding reliability when it was determined that the resulting magnetic caused bearing load would be 5 pounds or less.

Conventional organic insulation such as glass silicon or ML cannot withstand both high temperatures and the nuclear radiation from the nuclear reactor so an inorganic insulation (ceramic) was selected.



Pump Motor Stator Assembly

Figure 4

Insulation life rating for the inorganic insulation for temperatures up to $600^{o}F$ is greater than 10,000 hours. The stator can is pressure formed onto the ceramic core; the housing is evacuated, purged with nitrogen, and hermetically sealed.

The three-phase electrical connections are wye-connected. Electrical instrumentation connections are made through separately sealed ceramic terminals. The instrumentation consists of chromel-alumel thermocouples embedded at various locations in the stator and end turns to monitor the motor temperature.

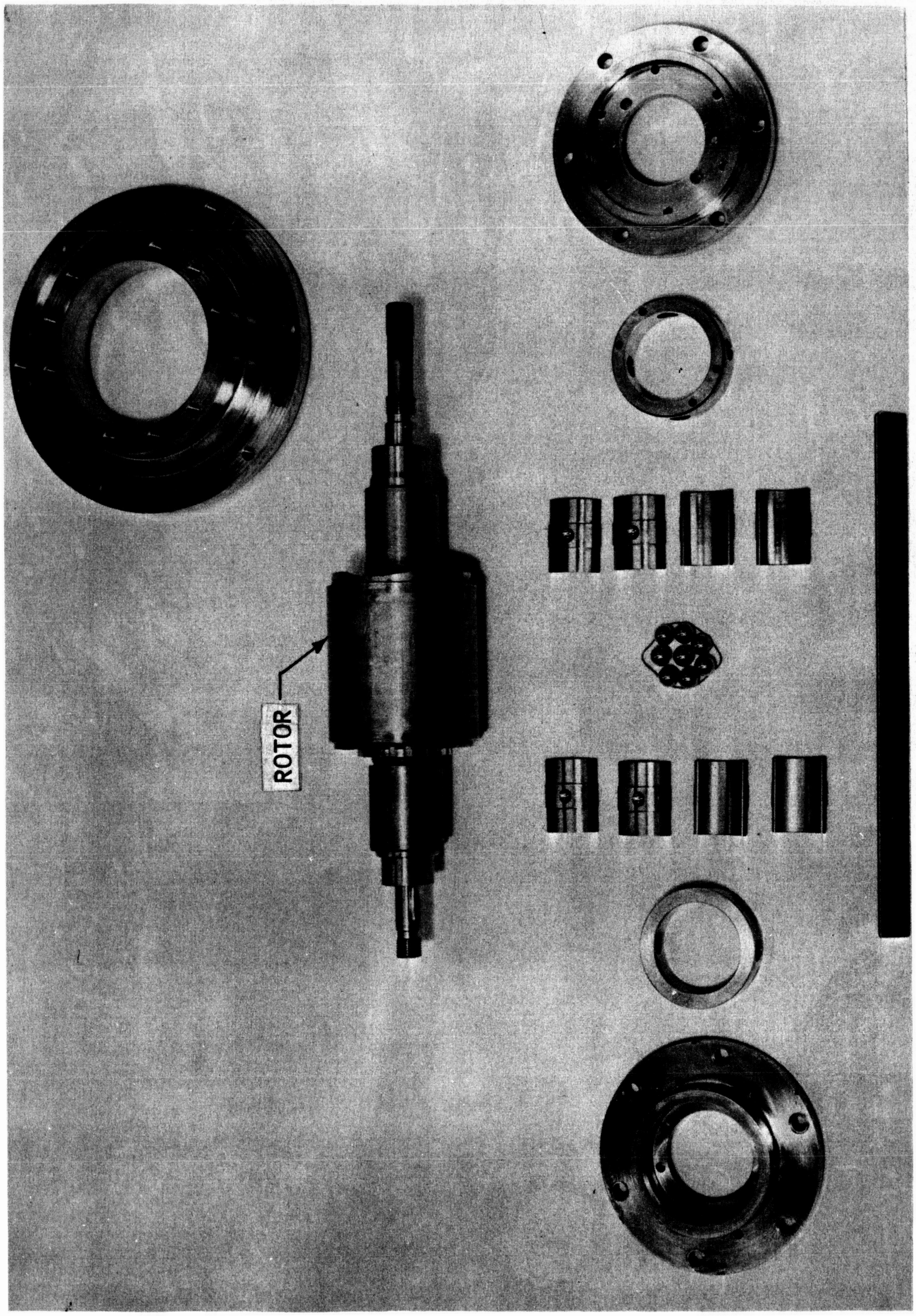
c. Rotor

The rotor is a squirrel-cage design with gross core dimensions of 3.5 inches diameter by 2.5 inches long, see Assembly, Figure 5. It consists of 46 semi-deep slot laminations made of AISI M-22 steel. Rotor conductor bars and end rings were made of oxygen-free electrical grade copper. To keep the rotor materials from being attacked by NaK, the rotor is completely sealed by a nonmagnetic Inconel can. The canning consists of a shrink fit inner and outer sleeve welded to end plates. The end plates have enough excess material for balancing the rotor. The rotor is shrunk on to the shaft and positively located with a nut and lock washer during assembly.

D. RECIRCULATION SYSTEM DESIGN

Successful operation of a hydrodynamic bearing system is significantly dependent on providing a clean and oxide-free fluid film for the bearings. The bearing film fluid for the NaK pump motor used the pump working fluid (NaK) which also functioned as a coolant for the motor. Since it would have been an almost impossible task to maintain the total quantity of system fluid at an adequate cleanliness level for the bearings, a much smaller volume of fluid was isolated and contained within the motor and bearing cavity. This fluid was provided with its own recirculation, cooling and filtering system as shown in Figure 2.

A small centrifugal pump overhung from the bearings at the end opposite the main pump circulates the NaK through an economizer which is a tube-in-tube counter flow heat exchanger. The NaK then passes through an



Rotor and Radial Bearings

Figure 5

oil-to-NaK heat exchanger where oxide particles are trapped out due to the lowering of the temperature. The oil for this heat exchanger is 4P3E which is externally supplied by the SNAP-8 organic loop pump. The NaK then passes through the other side of the economizer where it picks up heat from the incoming side of the economizer. The fluid then passes through a sintered metal filter which collects any debris before the NaK is finally returned to the pump-motor and bearing cavity. The purpose of the economizer is to heat the NaK slightly (about $10^{\circ}F$) to preclude oxide precipitation at the filter.

Some further details of the major elements in the Recirculation Loop are as follows:

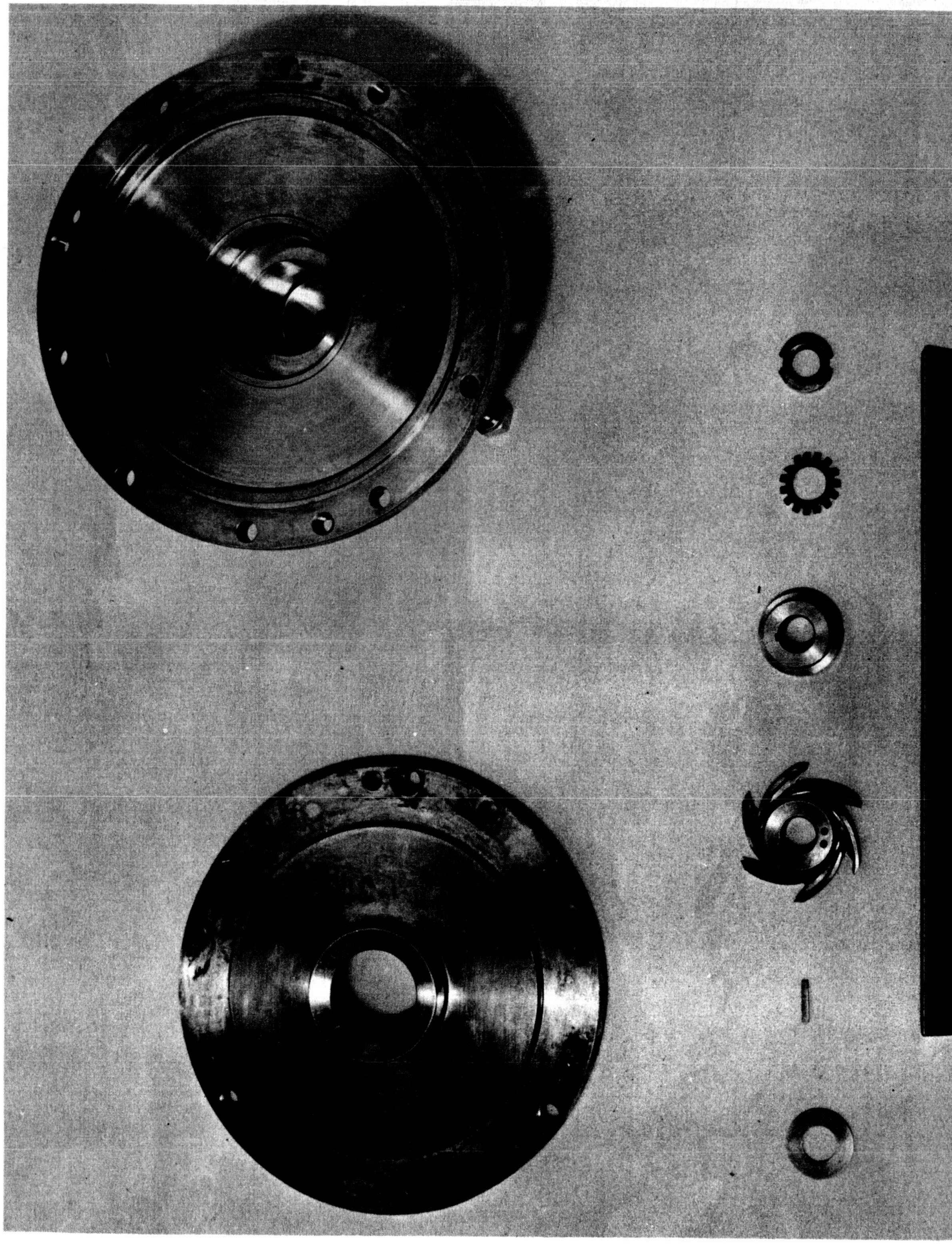
1. Recirculation Pump

The recirculation pump (Figure 6) is required to pump 1.6 gpm of NaK and produce a head rise of 40 feet with a pump speed of 5800 rpm; the calculated specific speed (N_s) is then 465. A radial flow, open vane impeller design was selected to perform this function. Suction pressure for this pump is controlled by the main loop pressure control. Under the normal operating conditions, the pump will have more suction pressure than the minimum required for a pump of this type.

a. Impeller

The impeller is a conventional design and is shown in Figure 3. A large allowance for suction prerotation was made because of the proportionally large suction eye diameter dictated by the diameter of the shaft on which the impeller is mounted. The impeller characteristics are as follows:

• Type impeller	open, radial flow
• Inlet eye diameter, in.	1.58
• Discharge diameter, in.	2.4
• Number of vanes	6
• Front seating angle, degrees	14-1/2
• Vane entrance angle, degrees	15
• Vane discharge angle, degrees	25
• Vane height at discharge, in.	0.10



Recirculation Pump

Figure 6

The impeller is keyed on to the shaft. Its axial location is adjusted with shims to give 0.011 inch front vane and 0.023 inch back vane clearances.

b. Pump Housing Design

The power requirements for the recirculation pump are low (hydraulic output is 10 watts). Therefore, the pump housing does not require maximum efficiency in design and simplicity and ease of fabrication were given prime consideration.

The selected design is an annular volute with a flow-limiting exit diffuser of 0.11 inch diameter. The flow-limiting exit diffuser controls the axial thrust. The annulus has dimensions of 0.06 inch x 0.10 inch on a diameter of 2.4 inches.

2. Cold Trap Heat Exchanger

The cold trap heat exchanger rejects the heat gathered by the lubricant-coolant NaK as it flows through the pump, see Figure 1. The heat collected is estimated to be 1925 watts. The heat exchanger, being the coldest in the recirculation system, can also be used as a cold trap to remove excess oxide particles which can harm the bearings. The selected design is a counterflow heat exchanger. In operation, the polyphenyl ether (Mix 4P3E) is directed into the outer shell at the top and exits at the bottom. The required heat exchanger effectiveness is low; therefore baffling is not needed. The NaK is first directed from the bottom upward between an annulus formed by the NaK container and a wire-mesh filled inner can. After the flow reaches the top of the annulus, it is reversed and then flows through the inner can and then out of the heat exchanger. The wire mesh inside the inner can presents a large area in which oxides can be collected.

The wire-mesh filled inner can is made from a section of Schedule 5 piping, 5 inches in diameter and 10-5/8 inches long. This is encapsulated in a 6-inch schedule 40 pipe, 10 inches long welded to end caps. The capsule is then enveloped by a shell formed by 7-inch tubing.

3. Filter

The recirculation system filter removes solid particles of foreign matter which could result in damage to the hydrodynamic bearings or cause possible interference in small clearance areas. The filter is constructed of sintered 304 CRES elements in a 4-inch, schedule 10 pipe housing. A 5-micron sintered filter is used which consists of sixteen disc shaped elements with approximately 250 square inches of filtering surface.

E. BEARING REQUIREMENTS

(NOTE: A more detailed description of these bearings is provided in NASA Report CR 72824.)

1. Radial Bearings

The requirements established for the radial bearings are:

Steady state load, lb.	0 - 50
Rotating load, lb.	0 - 5
Speed, rpm	5800
Lubricant	NaK 78 at 600°F max.
Bearing stiffness, lb/in.	18,000 min. @ 6000 rpm
Normal environment	zero 'g'

a. Bearing Selection

The high temperature operation requires a self-aligning type bearing to allow for shaft and housing thermal growth and distortions. The low lubricant feed pressure available cannot be made to support hydrostatic bearings, therefore a hydrodynamic bearing design was selected. Tilting-pad type journal bearings were selected because they comply with these considerations (see Figure 5). This type of bearing is one of the most stable bearings under light load, particularly against half-frequency whirl vibrations; it can also tolerate small magnitudes of misalignment and can be designed to have reasonable stiffness even at zero load, a significant consideration for startups. The tilting-pad type bearings offer a simple means of lubricant introduction and efficient shaft and bearing cooling with practically no pressure drop.

b. Number of pads

A four-pad design was chosen for the pump motor because the loading on this type design is more uniform than a three-pad design and it is less difficult to fabricate and assemble than bearings with additional pads. The inertia of the pads is sufficiently low so that shaft motions can be traced effectively. To assure stability of the unloaded shoes and increase the stiffness of the bearing, a preloaded design was selected (Reference 2).

c. Bearing Characteristics

The low viscosity of the NaK results in very low power consumption within the bearings; therefore the bearing size is limited only by package design and assembly considerations. The actual bearings have a length to diameter ratio of one; each of the four pads cover an arc of 80°.

The bearing characteristics are as follows:

Type	Four pad, tilting pad bearings
Pad length, in.	1.75
Bearing diameter, in.	1.75
Angular extent of each pad	80°
Angular distance from leading edge to pivot	46° (58% of total angular extent of pad.)
Pad radial clearance, in.	$0.0015^{+0}_{-0.0002}$
Bearing radial clearance, in.	$0.0010^{+0.0003}_{-0.0002}$

d. Bearing Materials

The selection of a material for the bearings must take into account the problems of starting the hydrodynamic bearings. At start and during low speed operation, a metal-to-metal contact frequently exists due to the lack of hydrodynamic forces. This means that to minimize galling, the bearing materials must be compatible both dry and operating in hot (600°F) NaK lubrication.

The pads are made of molybdenum-tungsten M-2 steel and the sleeves are made of tungsten T-1 tool steel. The ball pivots, also made of tungsten T-1 tool steel, are located at the 58% offset points from the leading edge of the pads. The balls are centered and located with the housing-bore as the reference. The bearing sleeve is shrunk on to the shaft and then ground to a 2 microinch, rms, finish.

2. Performance Estimates

a. Film Thickness

The performance of the four-pad bearing was calculated as a function of pad and bearing clearances for a bearing load of 55 lb (design load is 50 lb), NaK lubricant at 600°F (viscosity 3.5×10^{-8} lb-sec/in.^{2}) and a pump speed of 6,000 rpm. For the range of clearances defined by the manufacturing tolerances that were specified in the design, the calculated film thickness is approximately 0.00035 inch. The detailed bearing design and preliminary test evaluations in oil were conducted by M.T.I., Latham, New York.

b. Bearing Stiffness

The zero load stiffness of the bearings is very sensitive to bearing-to-shaft clearance, but not to pad-to-gimbal clearance. Assuming the most severe conditions, the lowest value of zero load stiffness is 19,000 lb/in. Based on critical frequencies of bearings and rotor shaft, an ample safety margin exists.

c. Natural Frequency and Resonance

The natural frequency of the pads was calculated to see if they would track shaft precessions. The results show that, over the range of loads and clearances considered, the natural frequency range for the lower pads is 710 Hz to 3680 Hz and the natural frequency for the upper pads is 210 Hz to 1700 Hz. The pads will thus track the shaft motions without difficulty, and the design also shows that there is no resonance in operating speed range.

3. Thrust Bearings

The selection of a thrust bearing (Figure 7) for the pump motor involves the same considerations required for the selection of the radial bearings; i.e., self-alignment and stability (Reference 2). Because of this, the thrust-bearing equivalent of the tilting-pad for the radial bearing was selected. This is the Kingsbury-type, self-aligning bearing. Each bearing pad is an individual plate which is free to pivot. The pads are made to pivot radially so that each pad can be inclined in a circumferential direction to provide a tapered lubricant film. Lubricating fluid is normally fed to the center of the housing near the shaft and pumped through the bearings by centrifugal action. A relatively small portion of the lubricant flows over the working bearing surfaces and the remainder passes through the spaces around the pads.

The thrust bearing characteristics are:

Load, lb	0-80
Speed, rpm	5,800
Lubricant	NaK 78 at 600°F maximum

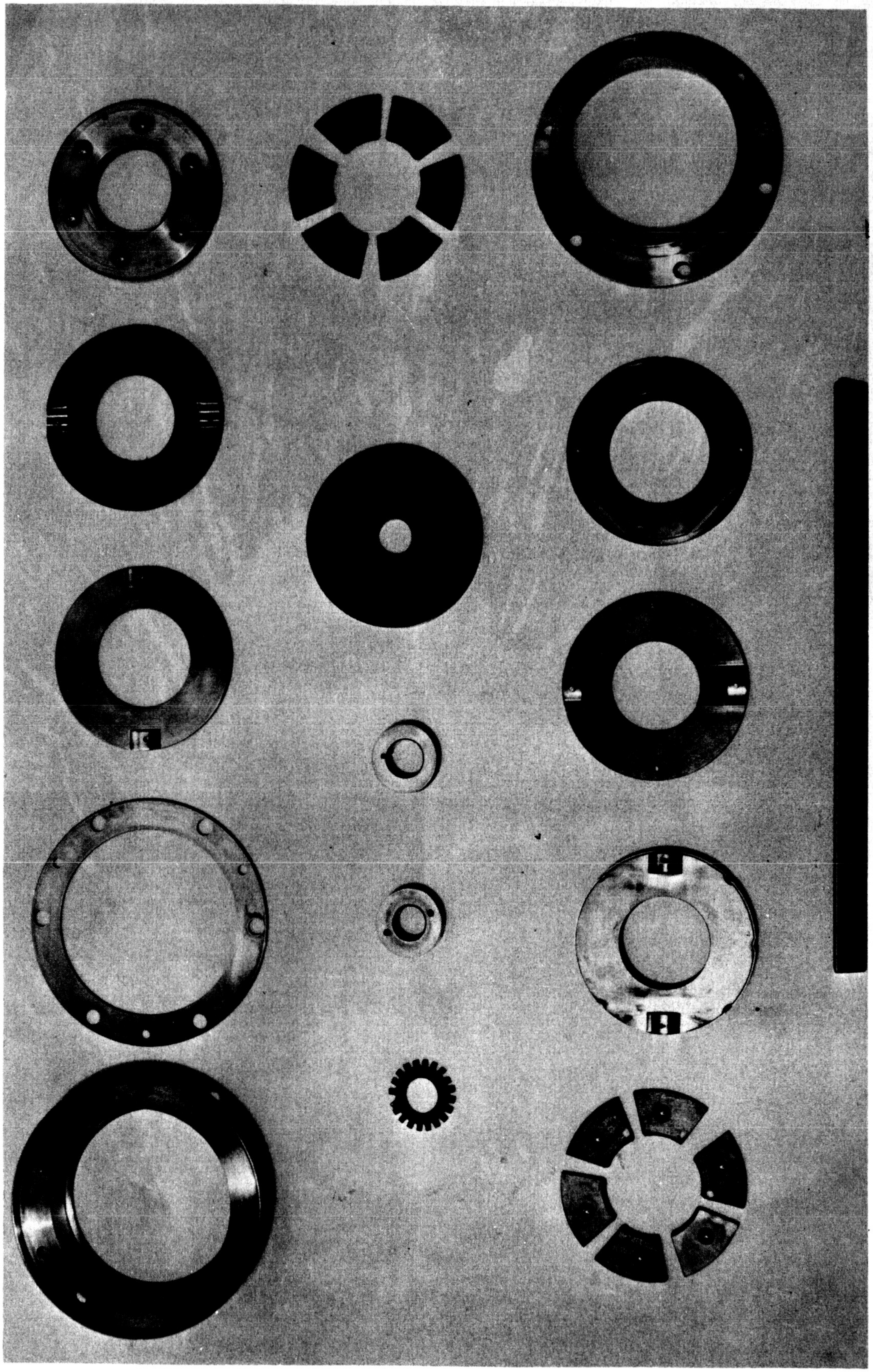
a. Design

The selected bearing design has six pads, see "Thrust Bearing Component," Figure 7. The pivot location is 58% of the distance from the leading edge to the trailing edge of the pad. This pivot location is optimum for machines with unidirectional rotation (Reference 2). The pads are supported on gimbal plates to provide alignment. The gimbal arrangement was selected because it fits into the small axial space available and is easier to fabricate from carbide parts than the more complex leveling link system. The bearing design details are as follows:

Type bearing	Kingsbury-Type, self aligning, tilting-pad bearings
Number of pads	6
Outside diameter, in.	3.75
Inside diameter, in.	1.875
Angular extent of pad, degrees	50
Ratio of angular distance from pivot to pad leading edge/angular extent of the pad	0.58

F. MECHANICAL DESIGN

The central problem in the mechanical design of the pump motor assembly involves thermal stresses and the need to keep the motor cool and isolated thermally from the pump while the pump is circulating the 1170°F NaK. Other problems, while significant, do not require special techniques.



Thrust Bearing Components

Figure 7

1. Pump-Motor Interface

The motor must remain at a temperature not higher than 600°F because of the degeneration of the magnetic properties of the motor lamination at temperatures higher than 600°F. To do this, the pump volute housing is necked down where it is attached to a closure plate on the main pump housing. The pump housing is then braced for rigidity by means of small cross section lugs which are pinned to similar lugs on the motor closure end plate (Figure 2). Thus, cross sections transmitting heat from the pump are minimized.

A computer analysis was made using the finite element method to check the effectiveness of the thermal design. For a pumped fluid temperature of 1170°F, the results showed that the motor temperature can be effectively kept below 400°F (Figure 8).

2. Critical Speed

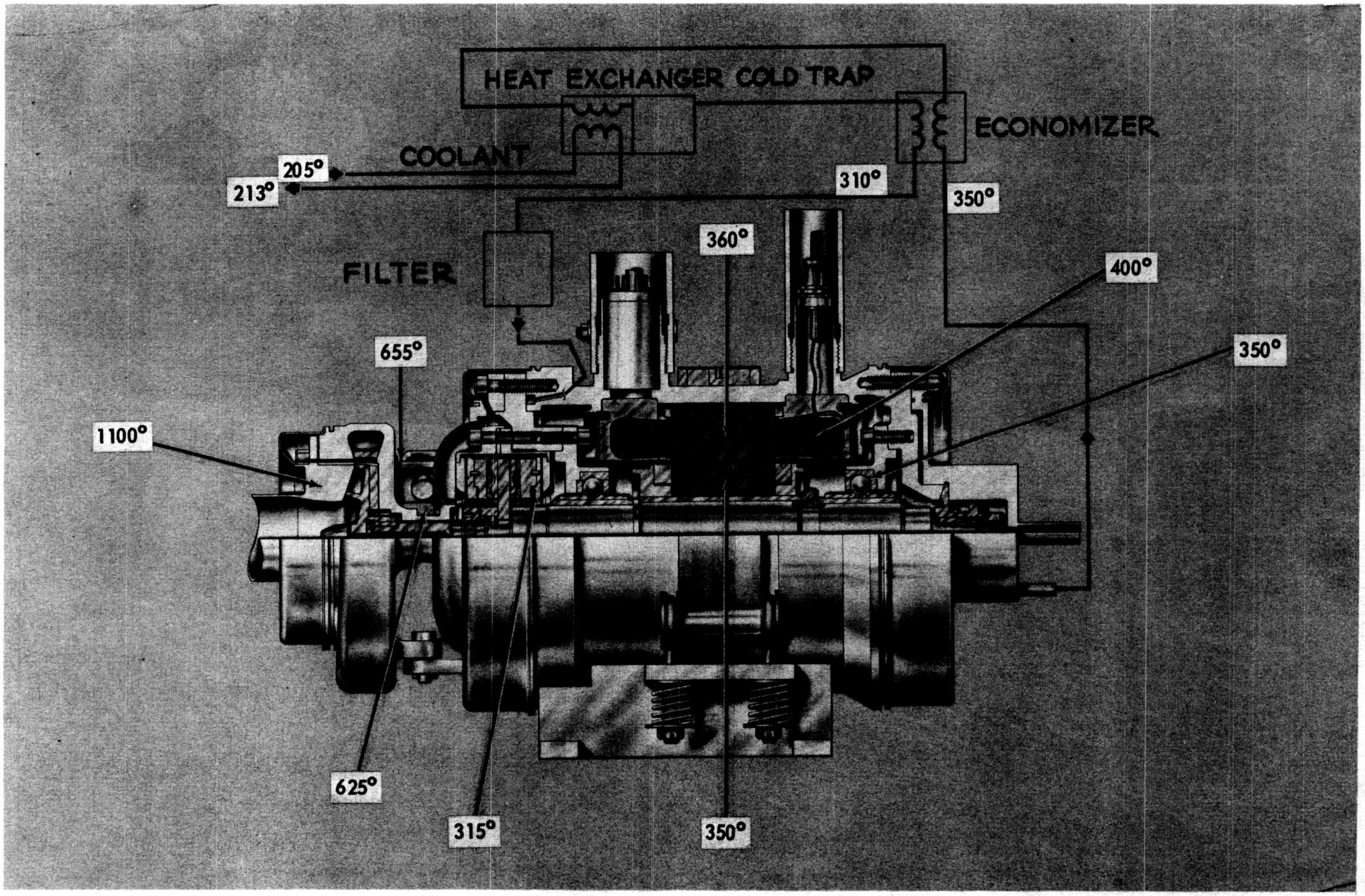
Using the conventional Dunkerly approach, the analysis of the rotor critical speed based on the stiffness of the rotor alone indicated that it was approximately 25,000 rpm.

The minimum calculated bearing stiffness was 19,000 lb/in. based on running the bearings unloaded and with maximum clearance at 6,000 rpm. A bearing stiffness of 19,000 lb/in. yields a critical speed of 9,000 rpm, an acceptable value for the pump-motor. Since an unloaded bearing or a bearing running with maximum clearance (zero eccentricity) is not to be expected in normal running, the most severe case of 9,000 rpm will be conservative, and the margin of safety is more than ample.

3. Materials

The following is a summary of the materials used in pump-motor development:

Part	Material
Motor Housing	9Cr-1Mo
Pump housing	304SS
Journal bearings	
Sleeves & Ball	Tool Steel T-1
Pads	Tool Steel M-2
Thrust Bearing	Carboloy 44A
Impeller (s)	410SS
Shaft	9Cr-1Mo
Rotor and stator cans	Inconel 600



Temperature Distribution in Nak Pump-Motor Assembly

Figure 8

III. TEST PROGRAM

A. RESULTS OF THE EARLY TESTS

1. Insulation Tests

Life tests were conducted to determine if the inorganic insulation could withstand the high temperatures to be expected for the pump-motor. Tests of 10,000 hours were conducted with a heated stator while monitoring the insulation resistance. The insulation tests showed that the winding resistance remained at the original value for the duration of the 10,000-hour test. There were no indications of any breakdown or potential breakdown.

2. Bearings

a. Hydrodynamic Performance

Bearing tests were conducted with the bearings in a silicon oil. The silicon oil was chosen on the basis of its similarity in viscosity to NaK. The tests were used to define minimum bearing thicknesses, minimum bearing stiffness, and general bearing performance.

The measured film thicknesses and the measured load capacities of the journal bearings agreed with the calculated values. However, the results of the tests on the thrust bearings did not show as close an agreement; the measured film thicknesses were as much as 0.0002 inch less than the calculated values. The decrease meant that the bearing's load carrying ability was slightly less than expected, but was still adequate for the loading expected for the pump-motor.

b. Bearing Stability

No bearing whirl or other instabilities were indicated by the tests. The inductance proximity probes showed that the load pads tracked the motions of the shaft accurately.

c. Bearing Flutter

Flutter tests were conducted with the journal bearing clearances at a maximum because this was where the maximum amplutides of flutter were expected. The data indicated that the flutter amplitudes increased with load and decreased with speed. Peak flutter amplutides of 0.0018 inch were measured. This is relatively small, and it is not expected that the flutter amplitude would ever become large enough to cause the pad edges to touch the shaft.

d. Wringing of the Thrust Bearing Pads

The first tests on the thrust bearings showed that one of the pads did not lift until a high shaft speed (about 3,000 rpm) was reached. The pad was apparently wringing to the thrust collar. Bench tests indicated that the wringing was caused by the extremely high surface finish and close conformity of the pads and runner as well as contaminant films on the bearing surface. Subsequent tests, made after changing the surface finishes from 2 to 4 microinches, rms and after properly cleaning the bearing pads, eliminated the wringing effect.

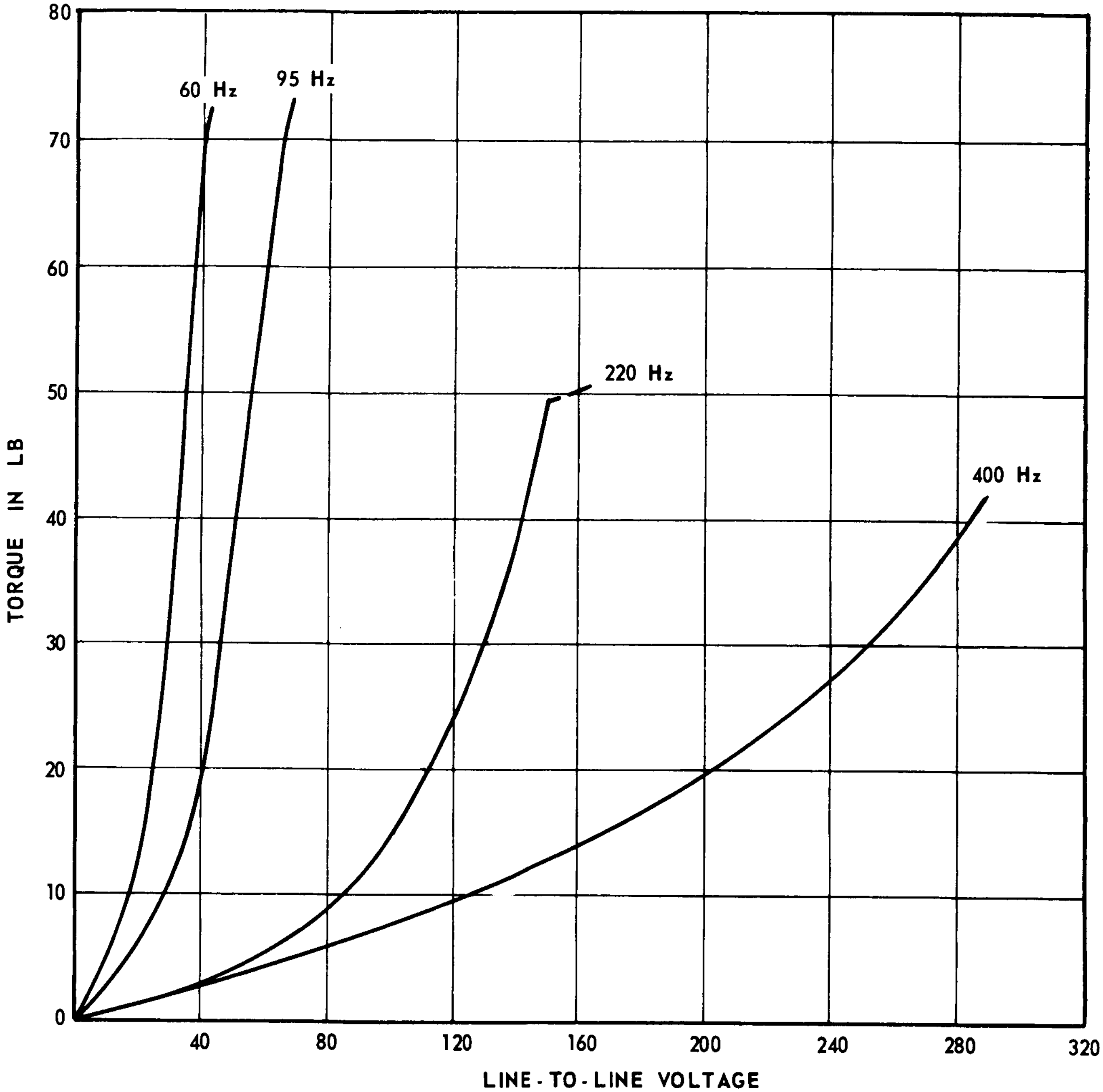
e. Bearing Pivot Fretting

Fretting tests were performed on the ball pivots of the bearings to determine if there were any problems with fatigue of the pivot points. Both oil and NaK were used as a lubricant, and different material combinations were used. The pivots that were tested showed some damage, both macroscopic and microscopic. Because of the limitations of the tests, it was not possible to extrapolate the data to 10,000 hours of operation in the pump-motor with any kind of assurance that the extrapolation would be valid. However, in the assessment of materials, it was found that the Carboloy 44A specimens showed superior wear characteristics when compared to tool steel.

3. Motor Tests

a. Motor Speed-Torque Relationship

Tests were made on a motor to determine the locked rotor torque (or starting torque), input power, and current at frequencies of 60, 95, 220, and 400 Hz and various input voltages (Figure 9).



Motor Starting Torque

The speed-torque relationship for the motor is shown in Figure 10. The results indicate that there is approximately 15% torque margin over the pump-motor requirements at 400 Hz. There could be startup difficulties if there should be any abnormally high drag resistance in the pump-motor. For instance, on 400 Hz power a 15% increase in the running torque requirement by the pump motor can cause the motor to stall at 4,000 rpm (Figure 10). The current is higher at this crossover point, and would cause higher line losses and lower the voltage to the motor, thus aggravating the situation. The starting torque at 95 Hz at 38 volts is much more favorable than at 400 Hz and 208 volts.

b. Motor Losses

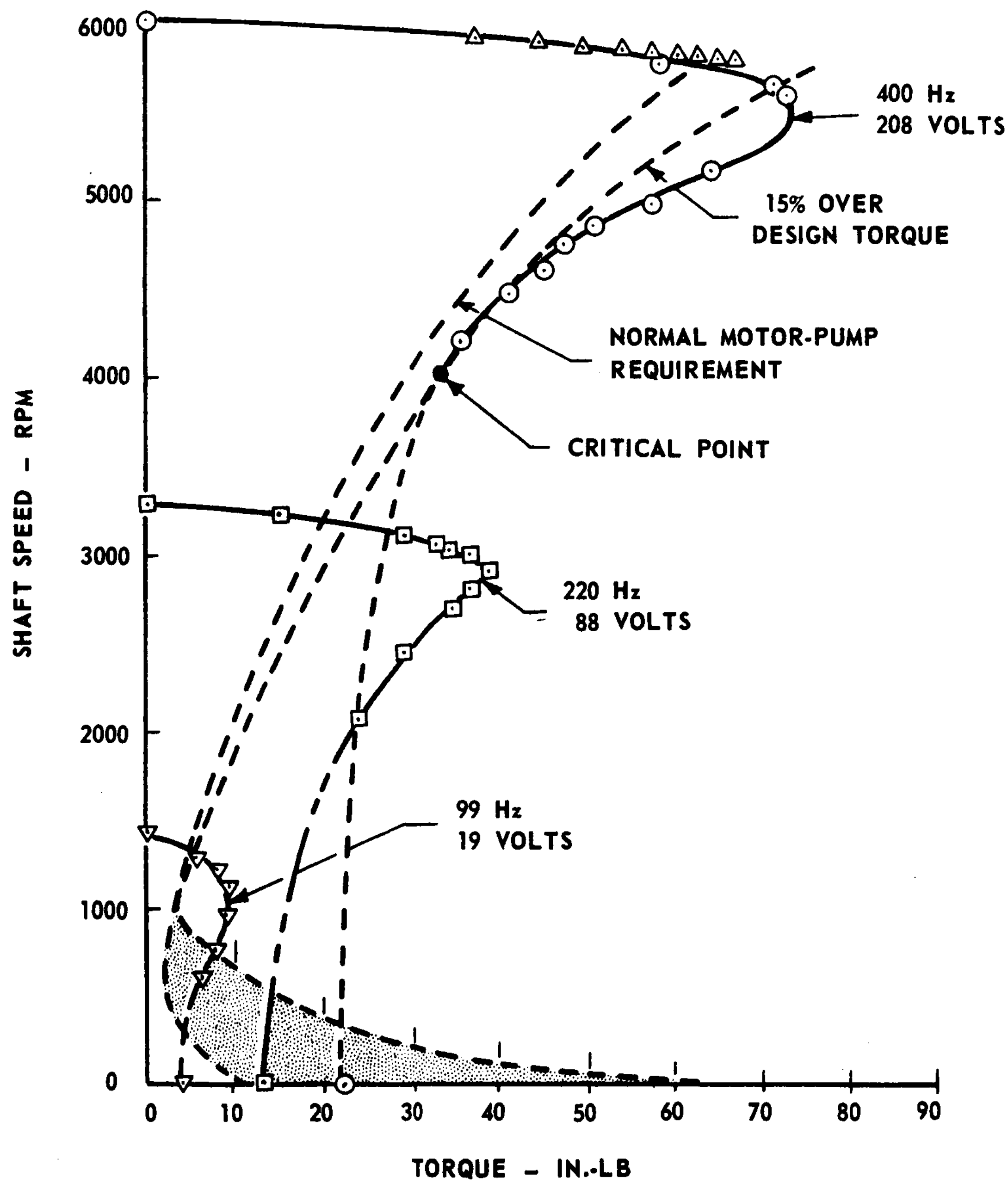
Motor losses can be divided into air-gap losses, copper I^2R losses, and friction and windage losses. In the case of the NaK motors, the air-gap losses consist of core loss, stray load loss, can loss, and NaK eddy current loss; the copper losses consist of the stator and rotor I^2R losses; the hydraulic losses consist of rotor hydraulic losses, bearing losses, and the recirculation pump output and losses. These losses were determined by various combinations of "in-air" motor tests, no-load saturation tests, full-load tests, and hot shutdown tests.

(1) Stator I^2R Loss

This loss was derived from the stator resistance (per phase) and the current. The actual stator resistance, a temperature-dependent variable, was calculated by using the ambient resistance of the winding and a temperature correction factor. The temperature correction factor was found through the correlation of winding temperature and thermocouple readings.

(2) Rotor I^2R Loss

The rotor I^2R loss can be derived from the electrical or air-gap power and the slip factor. The electrical power can be determined by summing up the shaft power and the hydraulic losses. On this basis the rotor I^2R losses were calculated to be 210 watts. Another method used was to evaluate the input power, the stator copper and air-gap losses and the slip; the slip being determined from the difference between the actual speed and the synchronous speed. On this basis the resultant I^2R loss was calculated to be 120 watts. This latter value was close to the design values and thought to be more likely to be correct.



Motor Speed-Torque Characteristics

Figure 10

(3) Core and Can Losses

The core and can losses were determined by the use of the no-load saturation tests on both uncanned and canned motors. The core loss was determined by subtracting the known losses (stator I^2R, rotor I^2R, friction, and windage) from the input power in an uncanned motor. The added losses in the canned motor were then presumed to be can losses.

(4) Hydraulic and Windage Losses

The hydraulic and windage losses can be determined either by measuring the rate of deceleration of the rotor or by a no-load saturation test. In the deceleration test, the average torque needed to decelerate the rotor at a finite speed during a finite time was determined and the power loss was found by multiplying the average speed and torque by a constant. In the no-load saturation test, the power input was plotted against the applied voltage and the resultant plot was then extrapolated to zero volts to determine the hydraulic and windage losses. The hydraulic and windage losses derived and extrapolated to NaK operation were 960 watts at 5800 rpm.

(5) Stray Load Loss

The stray load loss is the difference between the power input and the known power consumption.

4. <u>Testing the Complete Pump Motor Assembly in Water</u>

Once the characteristics of the motor were determined, it was used as the reference for establishing the performance of the pump. For the purpose of simplification, the first complete pump tests were run with water. The pump losses and the thrust on the impeller were established, and the thrust on the impeller was calculated by means of pressure profile measurements taken in the pump housing.

a. Hydraulic Performance

The head-capacity performance of the pump with an axial, front-vane clearance of 0.009 inch to 0.011 inch (set to simulate the valve for operating the pump with the circulating fluid at 500°F) was determined.

The measured head rise was about 6% less than expected; but the hydraulic efficiency was 68% or approximately 5% better than expected. The discrepancy between the test value and the expected pump head generation was due primarily to the pump's sensitivity to the front-vane clearance.

b. Cavitation Performance

The start of cavitation was defined as a 2% decay in head generation of the pump. This criterion is the standard used by the pump industry to guarantee pumps against excessive cavitation damage for one year's continuous operation. At the design flow conditions, the 2% head decay occurred at an NPSH of 16.6 feet. This is close to the calculated value of 15 feet and is well within the design requirements.

c. Pump Axial Thrust

The axial thrust of the pump impeller was calculated from the measured pressure profiles with trimmed back vanes as discussed in 4.e. on the front and the back of the impeller. The results showed that for an impeller back vane clearance of 0.021 to 0.023 inch, the axial thrust was 95 pounds toward the pump at shutoff, zero at 95 gpm, and 43 pounds away from the pump at 150 gpm. The uncertainty in these thrust values is about 10 pounds.

d. Pump Radial Force

The radial load on the pump impeller was calculated with the pressure measurements taken on the volute along the impeller tip. The measurements were taken with four pressure taps located 90° apart. The results showed that the direction of the force varied for varying flows. The force was a maximum of 6 pounds at rated flow.

e. Motor Cavity Pressure

The motor cavity pressure is set by the difference in the pumping capability of the front and back vanes of the impeller. To minimize the migration of NaK from the main loop into the motor cavity and to prevent the formation of vapors it is necessary to maintain motor cavity pressure equal to or slightly greater than the suction pressure of the main pump. This was accomplished by slightly trimming the back vanes at the hub as substantiated by test results.

5. Recirculation Pump

The actual head-capacity performance of the recirculation pump is shown in Figure 11, and is within the range of the projected values. No attempt was made to determine the exact suction requirements or the efficiency of the pump; but the results from the pump-motor tests indicated that they were approximately equal to the projected values.

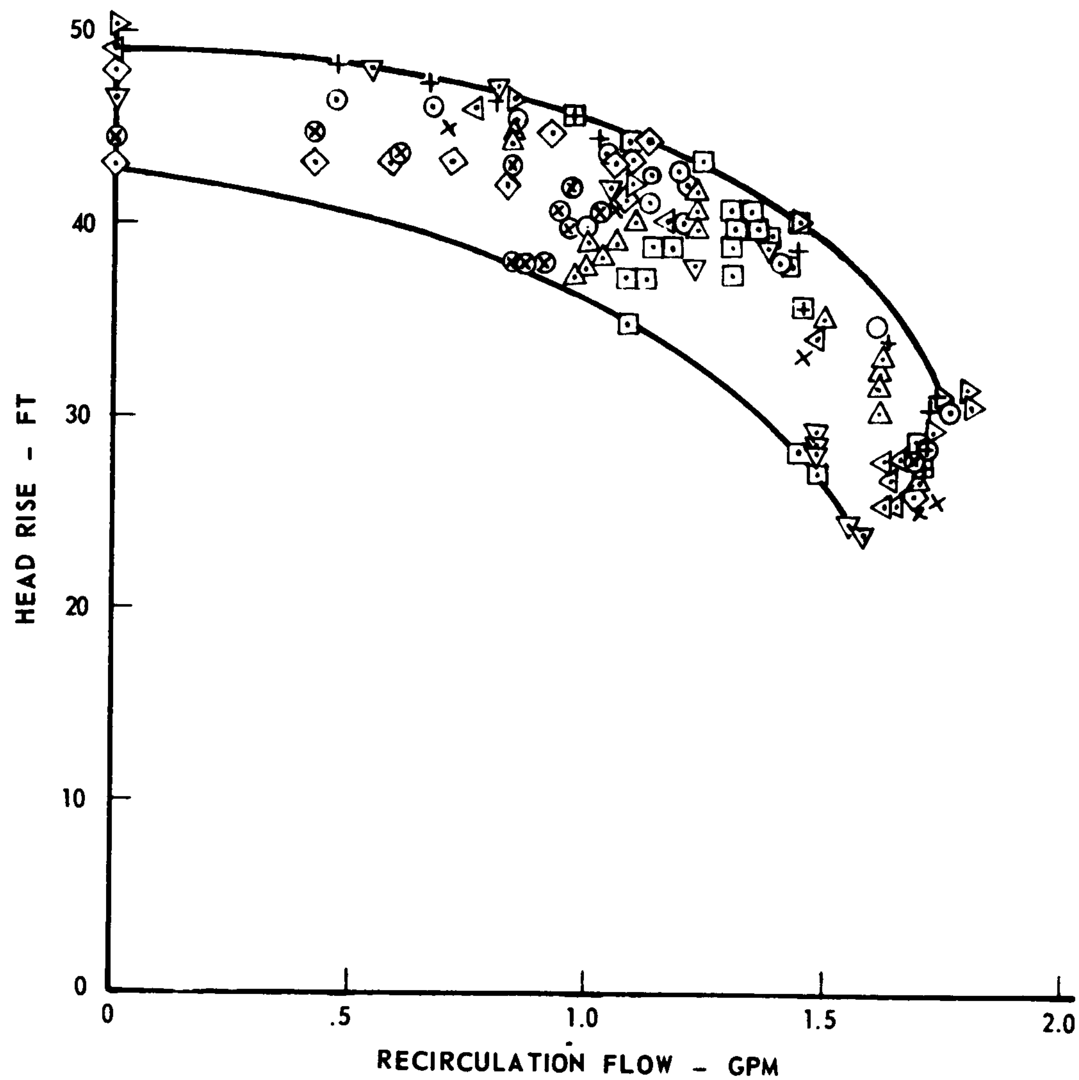
B. TESTING IN NAK

1. Performance Evaluation

The preliminary test results on the pump motor assembly and components had to be confirmed. The tests were performed with the working fluid (NaK) under the conditions specified for the design (1170°F). In these tests, the effect of NaK and the effect of temperature were established.

a. Pump

The results of the NaK tests on the pump showed that it confirmed the results obtained in the water testing. The head generation capability of the pump decreased slightly with increasing pump-fluid temperature because of the slight increase in the front vane clearance of the impeller. The pump performance did not change over a period of 13,442 hours of testing at 1170°F. Some rubbing between the impeller back vanes and the housing occurred during the early tests. The rubbing occurred when the pump was tested at extremely high temperatures, and a closer study of the thermal gradients indicated that it was a differential thermal expansion effect. The assembly clearance of the back vanes on the subsequent pumps was increased from the original 0.011 inch minimum to 0.021 inch minimum. The increased back vane clearance did not affect the hydraulic performance of the pump significantly. However, rubbing still occurred in one of the subsequent tests. This was caused by severe heating brought about by abnormal operating conditions. The back vane clearance was then increased to .040 inches to alleviate this and no further problems were experienced.



Recirculation Pump Performance

Figure 11

b. Motor

The motor performance (with water) did not change significantly when the complete pump-motor assembly was tested in NaK. It was calculated that there would be a NaK eddy current loss of 351 watts added to the losses found in the air and water testing. In the NaK tests, several hundred watts increase in the power input was noted; but it was not possible to determine whether the increase was due to the eddy current loss, or to an increase in the motor hydraulic and windage losses.

c. Bearings

The bearing performance conformed generally to expectations for steady-state running, but some difficulty was encountered in starting.

(1) Steady-State

The bearings were designed for 10,000 hours of continuous operation. The best test of these bearings would be to run them in a complete pump-motor assembly under the design conditions for as long a duration as possible. In the longest test conducted to date, a pump-motor was run for 10,362 hours and 786 starts with the same bearings. During the test period the bearings performed their functions in accordance with the design requirements. When the pump-motor was disassembled and inspected, it was found that the bearing surfaces were in excellent condition and their dimensions did not change significantly. On the basis of this test, it can be stated that the bearings performed all of their steady-state design requirements and would operate satisfactorily for a much longer period than the original goal of 10,000 hours.

(2) Startup

The bearings contributed to the starting problems encountered in the pump-motor. Because of the marginal starting torque available from the motor, any increase in bearing drag caused startup problems.

(3) Bearing Operational Summary

(a) For the first 200 to 400 hours, the bearing drag was low. Starting torque was generally 11 to 20 inch-pounds.

(b) After 500 hours of operation, the starting torque required was about 14 to 16 in.-lb if the torque was applied as a step transient. If the torque-time profile was too gradual, the motor would not start. After an unsuccessful start, the torque required to overcome the drag on the rotor was between 43 and 70 in.-lb. An input voltage greater than 66 volts at 95 Hz. was required to insure a motor start every time. At this time a modification was made to the thrust bearings by pinning the gimbal pivots as described in (4) and (5).

(c) After some 7,000 hours of testing in the 10,362-hour test, the pump required very high torque for starting. With the input power at 60 Hz, up to 100 volts at the motor terminals are required to assure that the pump would start every time.

(4) Causes of Starting Problems

Two causes of the "hard start" problems were considered initially to be most likely; namely the start up friction forces on the pads were too high and/or the pivots on the thrust bearing pads created a wedging action. The friction forces of the bearings could change due to removal of contamination from the bearing surfaces by NaK; when two contaminant-free metal surfaces come into contact (there is no hydrodynamic force during startup), the friction force can increase by as much as a factor of seven. The wedging effect of the thrust bearing pads was considered possible due to a peculiarity in the design of the pivots. With the friction forces on the bearing pads sufficiently high, the gimbal pivots can be drawn circumferentially up their inclined sockets. This causes a wedging effect against shaft rotation, and the only way to free the rotor is to back it off the wedge.

(5) Solutions

The problems of wedging at the thrust bearing were resolved by pinning the gimbal pivots so that the circumferential motion was restricted. After this change was made on the bearings, some wedging did occur, however, the wedging effect was considerably less than before. The friction problem if it is significant however, cannot be so easily solved. Contaminant-free bearing surfaces are inherent parts of the NaK-lubricated bearings because clean NaK tends to cleanse the metal surfaces of the oxide. The only solution is to provide sufficient torque to overcome the resistance. Recent studies indicate that the radial journal bearings may also be subject to a wedging action at their pivots and some corrective action similar to the pinning of the thrust bearing pads may be initiated. Consideration is also being given to the use of higher voltage at 95 Hz to increase the motor starting torque capabilities.

d. Recirculation System

The recirculation system was tested only to the extent of the determination of whether the motor and bearings were kept at the required temperatures. The results showed that the system was effective; NaK oxide and foreign particles were kept out of the bearing areas and the motor and bearings were kept below 400°F.

e. Minimum Pump Suction Pressure

Neither the main pump nor the recirculation pump normally operated at suction conditions low enough to cause cavitation problems. However, there were instances where gas ingestion into the motor cavity had caused loss of flow from the recirculation impeller due to it cavitating. These conditions however were associated with operator errors or facility deficiencies as mentioned in 3.

The minimum NPSH required for the overall pumping system was found to be 42.2 ft. at 98.7 gpm.

f. Axial Thrust

The axial thrust on the pump-motor varied as a function of the flow in the pump. In the tests, the pressures around the various parts of the pump-motor were measured and the resultant axial forces calculated. The results showed that the axial thrust varied from -62 lb to +76 lb when the pump flow was varied from zero to 150 gpm. The positive direction is from the pump toward the motor.

g. Radial Loads

The radial loads on the pump-motor consisted of the rotor weight, the motor magnetic load, and the radial loads on the pump impellers. The motor magnetic load was 5 pounds, and the radial load on the recirculation pump impeller was 2.5 pounds. This means that the major portion of the radial loads was imposed by the 18-lb rotor and the 12-lb radial load on the main pump impeller. These loads resulted in reaction loads of 46 pounds on the pump-end bearing and 14 pounds on the motor-end bearing.

h. Critical Speed

There were no indications of dynamic instabilities or critical speed problems in any of the tests. Accelerometers mounted on the housings showed readings of less than 0.5 'g'.

2. Development Problems

Some problems were encountered in the operation of the pump-motor in NaK. These development problems included leaks in the welds and the cast pump housings and gas trapped in the motor cavity.

a. Leaks

NaK has a propensity for leaking due to its low surface tension, and it will tend to penetrate the slightest imperfection or crack in the containment material. Furthermore, NaK has a very high affinity for oxygen; it will leach out any oxide particle with which it comes into contact. In the case where castings have oxide inclusion, the leaching out of the oxide will result in leaks.

(1) Weld Leaks

In the development testing, the leaks around welds had always been through imperfections that would pass gross inspections.

By tightening the inspection procedures and minimizing the number of welds to be used on the pump-motor, the problem appears to have been solved. The 13,442-hour test was conducted without failure.

(2) Leaks Through the Cast Housing

The first pump housings were made from castings. These cast pump housings leaked during pump-motor tests. The castings had previously passed dye-penetrant and helium leak checks with no indications of possible leaks.

It might have been possible to have developed casting and inspection techniques which would have made the cast pump housings usable for NaK service. However, it was decided that this course was too uncertain, and the pump housing was redesigned to be machined from bar stock. After this design change, there was no further problem of leaking.

b. Noncondensible Gas in the Motor Cavity

The motor cavity is the lowest pressure point in the NaK pump system. There was a tendency for gas to form in the motor cavity during operation. This gas was trapped in the motor cavity and caused erratic or loss of flow in the recirculation pump.

This problem was solved by designing a vent for the motor cavity; thus the gas was vented out of the motor cavity during system filling. Also, stringent requirements were imposed relative to the gas content of the NaK fluid used in the pump motor.

3. Pump Motor Operational Problems

There were operational problems with the pump motor, primarily due to improper operation of the testing facilities.

a. Improper Bleeding of the Recirculation Loop

On several occasions overheating of motors, bearings, etc., occurred due to loss of coolant flow in the recirculation loop. In some instances, a loss of radial bearing clearance occurred causing seizure of the pump. The loss of coolant flow was invariably traced to gas forming or passing into the motor cavity and then to the eye of the recirculation impeller causing it to cavitate with a consequent loss of flow. A rapid heating up of the rotor parts then occurred with the resultant seizure.

b. Mass Transfer in the Heat-Transfer Loop

In some cases the heat transfer loop was in operation while the pump was not operating. With the pump in this mode; i.e., bypassed and idle, the whole pump-motor unit is naturally colder than the rest of the heat transfer loop, and there is then a tendency for NaK oxides to precipitate in the pump and motor cavity. In many such instances starting difficulties were experienced, and on some occasions it was impossible to start the pump at all.

IV. CONCLUSIONS

A. PERFORMANCE

1. Steady-State

The performance of the pump-motor as it exists at the present time is shown in Tables I and II for operating temperatures of 495°F and 1170°F. Variation from design point performance is shown in Figures 12 and 13. The pump motor performed as expected and the design life of 10,000 hours has been achieved.

2. Startup

Start problems were encountered when a combination of several adverse factors existed. This has been a continuing problem even when the pumped NaK is clean and free of contaminants. The startup problems and proposed solutions will be documented in a subsequent report.

B. MATERIAL

The change to a harder material such as tungsten carbide could increase the tolerance for contaminants in the radial bearings. It is noted that while the NaK oxides have caused varying degrees of damage to the tool steel radial bearings, there was virtually no damage on the thrust bearing surfaces, which are made of tungsten carbide.

C. PRESENT STATUS

As of December 1968, pump-motors have been tested for a total of 31,136 hours, with the longest single unit test lasting 10,362 hours. Following is a summary of operational experience:

	1965	1966	1967	1968	
Test Time, Hours	3,550	2,252	14,426	10,923	31,151 Total

All of the major problems have been resolved, and the pump motor can be considered as qualified for the SNAP-8 system with a life potential far exceeding the original goal of 10,000 hours. In fact, there is every indication that an endurance goal of 40,000 hours can be achieved.

TABLE I

PUMP-MOTOR PERFORMANCE WITH 495°F AND 1170°F NaK

Components and Parameters	Note	Test Results 495°F	Test Results 1170°F
Pump			
Flow, gal/min		98.7	125
Speed, rpm		5,810	5,815
Inlet temperature °F		495	1,170
Density, lb/ft^3		50.8	45.8
Inlet pressure, psia		15.0	30.8
Outlet pressure, psia		55.5	59.1
Pressure rise (ΔP) psi		39.8	28.3
Head rise, feet		113	89
Shaft input power, hp (kw)		3.36 (2.51)	3.21 (2.39)
Hydraulic power, hp (kw)		2.30 (1.72)	2.06 (1.54)
Pump efficiency, %	(1)	68.5	64.2
Motor			
Input power, kw		4.6	4.52
Power factor		0.56	0.56
Total electrical losses (watts)	(5)	1,130	1,156
Motor electrical efficiency, %	(2)	75.5	74.4
Total mechanical and electrical losses (watts)	(5)	2,090	2,131
Motor overall efficiency, %	(3)	54.6	53
Pump Motor Assembly			
Overall efficiency, %	(4)	37.3	34.1

NOTES:

(1) $$\text{Pump Efficiency} = \frac{\text{Hyd. Power Output}}{\text{Shaft Input Power}} \times 100$$

(2) $$\text{Motor Elect. Efficiency} = \frac{\text{Input Pwr. - Total Elect. Losses}}{\text{Input Power}} \times 100$$

(3) $$\text{Motor Overall Efficiency} = \frac{\text{Input Pwr. - (Elect. Losses \& Mech. Losses)}}{\text{Input Power}} \times 100$$

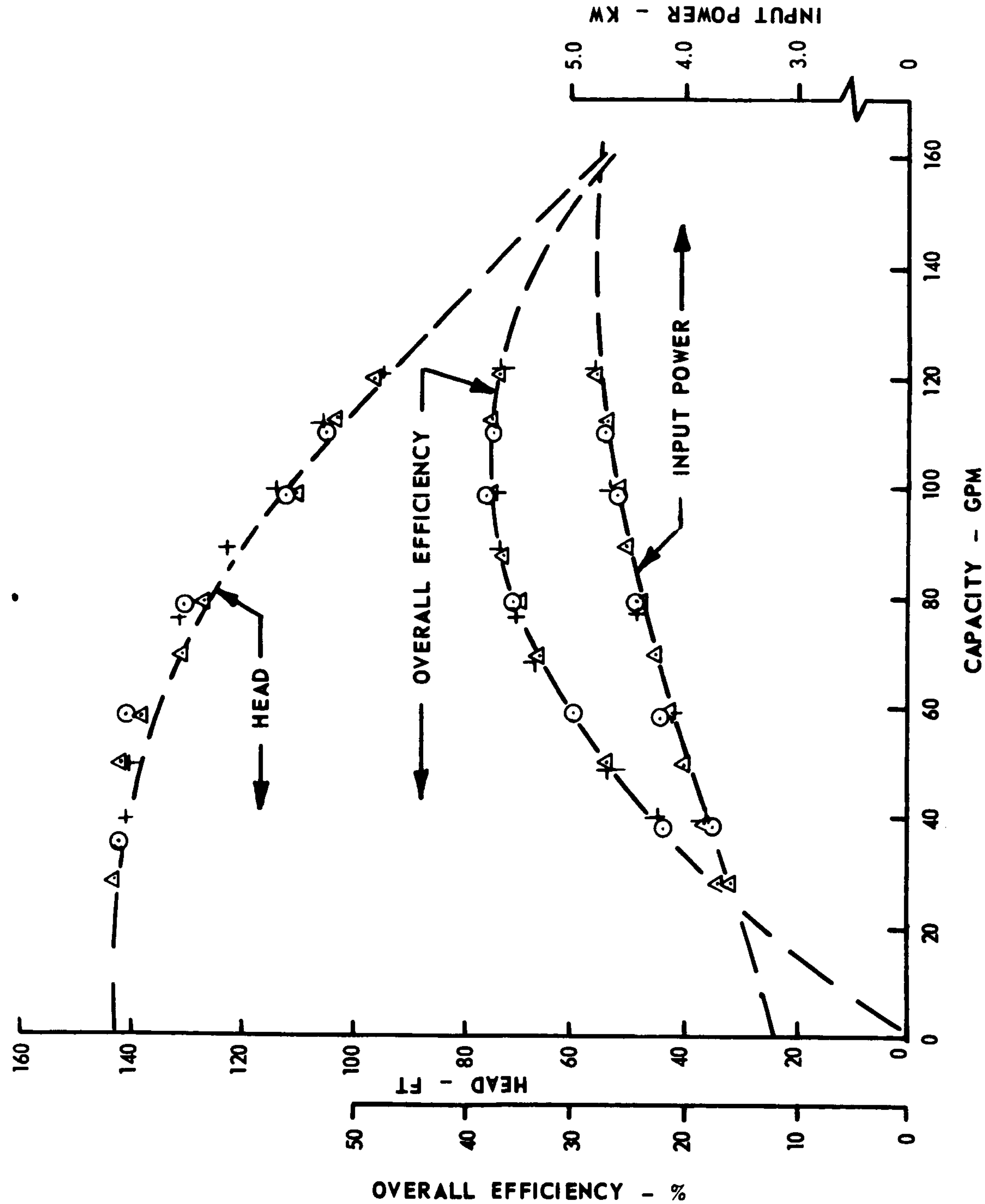
(4) $$\text{PMA Overall Efficiency} = \frac{\text{Input Power}}{\text{Hyd. Power Output}} \times 100$$

(5) See Table II for Breakdown of Motor Losses.

TABLE II

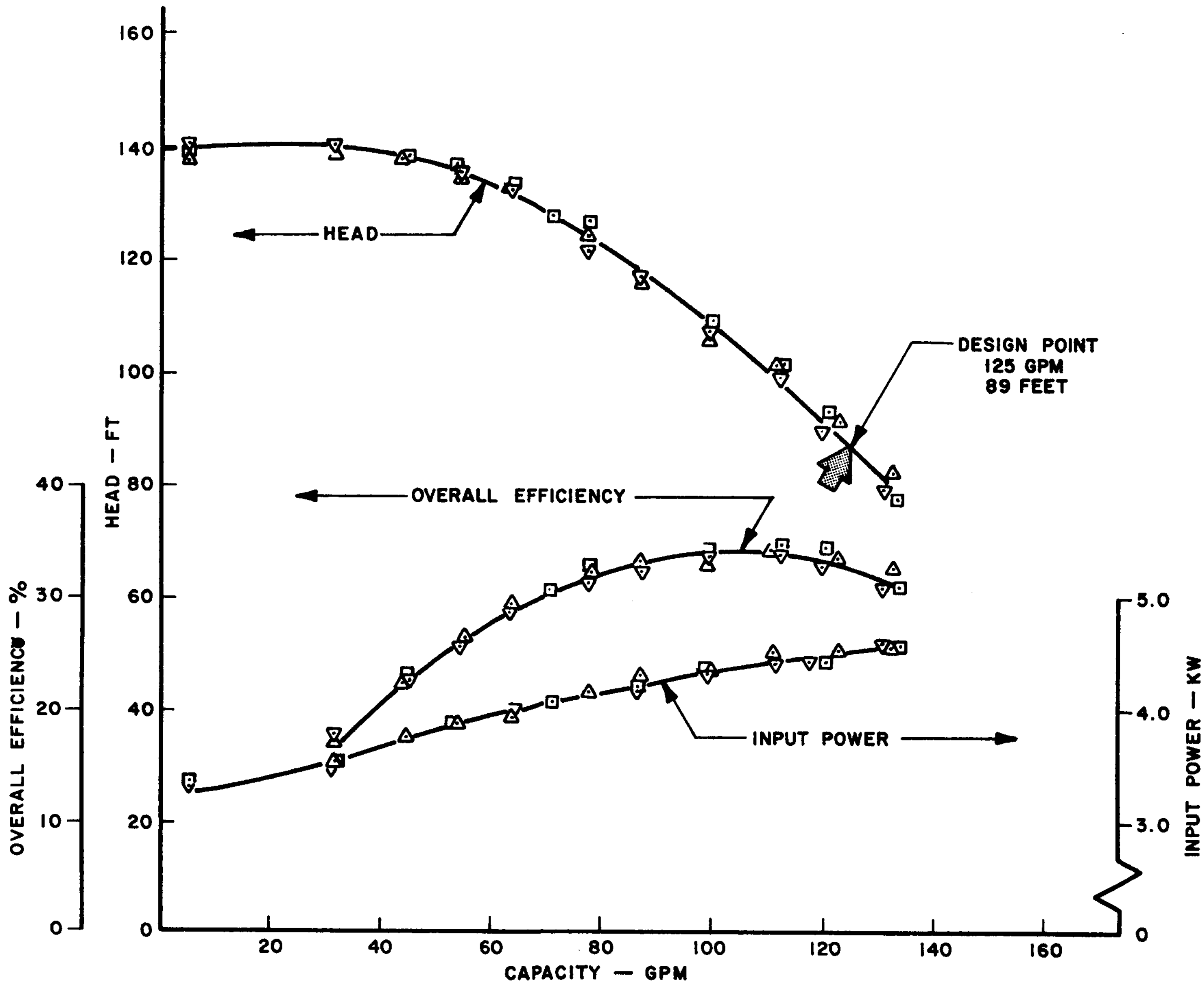
MOTOR LOSSES WITH 495°F AND 1170°F NaK

Summary of Losses	Test Results 495°F	Test Results 1170°F
Electrical (watts)		
Stator iron	112	110
Stator copper	323	312
Rotor copper	110	102
Rotor can	20	20
Stator can	154	158
NaK eddy	351	389
Stray load	60	65
Mechanical (watts)		
Rotor hydraulic (windage), Bearings, Recirculation pump	960	975
Total mechanical losses (watts)	960	975



Pump-Motor Performance with 495°F NaK

Figure 12



Pump Performance with 1170°F NaK

Figure 13

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