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REPORT No. ER-4101

**THE SNAP II POWER CONVERSION SYSTEM
TOPICAL REPORT No. 5 MERCURY PUMP DESIGN**

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STAFF RESEARCH AND DEVELOPMENT

THOMPSON PRODUCTS DIVISIONS

Thompson Ramo Wooldridge Inc.

CLEVELAND, OHIO

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REPORT NO. ER-4101
THE SNAP II POWER CONVERSION SYSTEM
TOPICAL REPORT NO. 5
MERCURY PUMP DESIGN

PREPARED BY:

M.A. Keresman
M.A. Keresman
Design Engineer

CHECKED BY:

George Y. Ono
G.Y. Ono
Senior Project Engineer

APPROVED BY:

W.R. Chapman
W.R. Chapman
Manager - Engineering

D.L. Southam
D.L. Southam
Senior Project Engineer

J.E. Taylor
J.E. Taylor
Manager,

New Devices Laboratories

DATE

June 10, 1960

DEPARTMENT

8868

Thompson Ramo Wooldridge Inc.

CLEVELAND, OHIO, U. S. A.

01-2

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ABSTRACT

SNAP II is the designation for a 3 KW nuclear auxiliary power unit to be used in a satellite vehicle. The SNAP II system consists of a reactor heat source, a mercury Rankine engine, and an alternator. Design details and test results of the jet-centrifugal mercury pump are presented in this report. This work was performed under a subcontract to Atomics International as part of Atomic Energy Commission Contract AT(11-1)-GEN-8.

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1.0 SUMMARY

The SNAP II system is a 3 KW nuclear auxiliary power unit intended to be used in a satellite vehicle. The SNAP II power plant utilizes a mercury vapor Rankine-cycle with a reactor as a heat source. Useful electrical power output is obtained from an alternator coupled directly to the turbine shaft.

The work presented in this report was performed under a subcontract to Atomic International with three prime objectives:

1. Select the type of mercury pump that will best suit SNAP II requirements.
2. Design, fabricate, and test such a pump to evaluate its performance at both design and off-design operating conditions.
3. Determine areas of possible improvement.

A preliminary analysis conducted on various types of pumps has indicated that a centrifugal pump plus a jet boost pump is best suited to the requirements of SNAP II.

Experimental results have confirmed the pump design approach which gives pump overall efficiencies of 30 percent and still allows the pump suction to be at 6 psia. Further modification of the pump will be directed towards system integration.

2.0 INTRODUCTION

The SNAP II auxiliary power unit for 3 KW of electrical output consists of an Atomic International reactor heat source and a Thompson Ramo Wooldridge power conversion system. The power conversion system is a mercury Rankine engine which is composed of a mercury boiler heated by a NaK heat transfer loop, an axial flow impulse turbine which extracts energy from the superheated mercury vapor, a condenser which returns the vapor to liquid state, and a mercury pump which returns the condensate to the boiler. The turbine drives the alternator, the mercury pump, and a permanent magnet induction NaK pump which is designed to circulate the NaK between the mercury boiler and the reactor.

The Tapco Group of Thompson Ramo Wooldridge Inc. has been actively engaged in the development and manufacture of centrifugal, axial flow, turbine, gear, and vane type pumps for a number of years. Experience with fluids has included various hydrocarbon fuels, water, alcohol, liquid oxygen, nitric acid, ethylene oxide and hydrazine. Interest in booster pumps was initiated with the development of a booster pump for the original B-17 bomber. Since then Tapco has been a major manufacturer of fuel booster pumps. Thompson Ramo Wooldridge Inc. also maintains a research group in Portland, Oregon, whose purpose is the continuous study of the state of the art of pumping mechanisms.

Considerable information for the current program was obtained from the SNAP I program. The lower power SNAP I system utilizes a similar mercury Rankine engine with an isotope heat source instead of a reactor. Several types of pumps were tested for SNAP I, and the jet-centrifugal combination was selected as optimum. All available information applicable to the SNAP II pump was incorporated in its design.

Selection of the cycle conditions, which established the SNAP II pump operating requirements, are reported in SNAP II Topical Report No. 1 titled, "Design Point Determination" (Thompson Ramo Wooldridge Engineering Report No. 3619, Document AE-61 [A]).

The following sections of this report describe the selection, design, test apparatus and test analysis of the SNAP II jet-centrifugal pump.

3.0 PUMP DESIGN

3.1 Pump Specifications and Requirements

The boiler feed pump for the SNAP II system is required to take fluid from the low pressure condenser and supply the boiler at high pressure. The pump must not only supply a continuous flow to the boiler, but must also supply lubricating flow to the bearings. The initial system was so designed that the pump discharge pressure must remain relatively constant for the flow variations present in the system. The limit on pressure variation is set by the boiler and bearings. The pump performance curve will have to be matched to boiler and system requirements. Figure 3.1-1 shows the representative flow characteristics

GENERAL HEAD VS. FLOW CHARACTERISTICS- SNAP II

FIGURE 3.1-1

of the SNAP II system. From a system standpoint, it was desirable to have a flat head versus flow curve from the pump, thus making pressure relatively independent of flow. This would eliminate the need of a control of the pump discharge. However, this type of performance is difficult to achieve and still have an efficient pump design. The original design requirements for the pump were:

Speed	40,000 rpm
Inlet Pressure	6 psia
Inlet Temperature	450°F

Flow and pressures were related as follows:

<u>Hg Flow Lb/Min</u>	<u>Pump Discharge Pressure (psia)</u>
25	129.5 to 142.5
37	129.5 to 135.5
45	122.5 to 135.5

As components developed, the pumping requirements changed. For example, PCS I system will require the pump to provide 34 to 40 lb/min at 260 to 280 psia with speed and inlet as listed before.

In addition to these specifications the pump must meet the following requirements:

3.1.1 Reliability

The major design consideration is reliability. The pump is expected to run continuously for one year without any appreciable loss in performance.

3.1.2 Weight

Since the unit is designed for orbital operation, weight must be held to a minimum. A higher weight design could only be accepted if a greater degree of reliability or performance would be obtained.

3.1.3 Efficiency

The efficiency of the pump in itself is a relatively unimportant design criteria since mercury pumping power is insignificant in the system. Reliability, performance, weight and package integration take precedence over efficiency.

3.1.4 Cavitation

Pump cavitation must be considered very carefully. Cavitation will not only reduce performance and efficiency, but it can also cause damage to the pump which will impair reliability and performance. Steps must be taken to reduce cavitation effects to a minimum and to keep performance and reliability at a high level.

3.1.5 Leakage

Leakage from the system cannot be tolerated. The use of a make-up feed system would impose a high weight penalty and would definitely limit the life of the system. External leakage must be zero.

3.2 Pump Selection

There are three sources of power available to drive the pump: a high-speed shaft; high-temperature mercury vapor; and electrical power from the alternator.

High-temperature mercury vapor could be used to drive a vapor jet pump, a piston pump, or an auxiliary turbine. The vapor jet pump is simple and reliable, but its efficiency is very low (3 to 4 percent). A high-temperature vapor motor requires extensive development and is of doubtful reliability due to the high-temperature valving. An auxiliary turbine provides a good drive, but introduces the complexities of a second rotating unit.

Electrical power is available to drive a motor, but this also requires a second rotating unit. Pumping problems could be minimized, however, the addition of a hermetically sealed unit introduces a host of new problems.

The use of the main drive shaft of the turbo-alternator package unit to drive the pump simplifies the leakage problem, increases reliability by reducing the number of component parts, and increases the efficiency of the power conversion system by eliminating a second rotating unit with an inefficient motor or turbine drive and a separate set of bearings.

Several types of pumps were investigated to determine their power requirements, NPSH requirements, weight, reliability, simplicity, and efficiency.

3.2.1 Piston Pumps

This type of pump appears advantageous from the standpoint of cavitation, start up, and control. The problems encountered in obtaining a suitable high speed drive and valving appear to offset any gain in performance and control. For a direct drive off the main drive shaft, a cam or slight eccentric would have to be used, and then wear and lubrication would become a problem. If the piston pump was driven at a sufficiently low speed to make valving feasible, a separate drive or gearing would be required which complicates the overall design.

3.2.2 Gear Pumps and other Rotating Positive Displacement Pumps

These pumps have relatively high NPSH requirements which practically eliminates them from consideration. In addition, if driven at 40,000 rpm, dynamics and wear problems develop on the gears and other moving parts.

3.2.3 Diaphragm Pumps

Diaphragm pumps present much the same difficulties as piston type pumps. They have the added disadvantage of doubtful reliability for the life required.

3.2.4 Centrifugal Pumps

Centrifugal pumps present the problem of introducing the inlet flow at only 5 psi above the saturation pressure, into a 40,000 rpm impeller without causing cavitation. They are advantageous, however, from the standpoint of simplicity, size, external leakage control and reliability.

3.3 Design Procedure and Assumptions

In order to solve the centrifugal pump problem, a study was made as to the type of pump required. The determination of a pump type, and its approximate proportions can be found by the use of specific speed (N_s). The original design point specific speed for this system was:

$$N_s = \frac{\text{rpm} \sqrt{\text{gpm}}}{H^{3/4}} = \frac{40,000 \sqrt{\frac{40}{810} \times 7.5}}{\left(\frac{140 \times 144}{810}\right)^{3/4}} = 2180$$

This type number indicates that the required pump will have proportions of a Francis type impeller (i.e. ID = 1/2 OD). With this established, it is then possible to determine if it is feasible to design the pump. A check of standard pump references was used to determine the expected performance of a pump with a specific speed of 2200. It was found that the SNAP II pump had to extend the state of the art for small high speed pump design.

The following program was established to develop the required pump:

1. Pump requirements were checked for feasibility using existing pump literature.
2. This data was compared with SNAP I test results.
3. The Snap II pump was designed and tested using the information obtained from 1 and 2.
4. Pump was redesigned according to SNAP II test results.

To better understand the problem and to determine the state of the art of pump development, a literature search was conducted. This survey revealed two major problem areas: cavitation and miniturization. A check of references revealed that a minimum inlet pressure of 22.5 psia would be required. This is approximately 4.5 times the available absolute inlet pressure. The difference would have to be eliminated if the pump was to operate without cavitation.

The second problem, that of miniturization would not only make the pump difficult to manufacture, but it was doubtful if empirical pump constants determined for large flows (10, 100 or more gpm) could be used for a pump with 0.37 gpm flow rate.

The information obtained from literature was integrated with SNAP I test results. The composite of this is the basis for No. 1 pump design. This design was built, tested, and results obtained from the tests were then used to modify the design and to develop a more suitable unit (No. 3 pump). An intermediate design option was also incorporated in the first series of tests. An impeller with a 90° discharge angle (No. 2 pump) was designed to determine the effect on performance of the larger discharge angle.

3.4 Pump Design

The experience gained from the SNAP I pump development program showed that it was possible to design a pump to operate at very low inlet pressures. The test data indicated that a jet pump was needed to raise the impeller inlet pressure high enough to eliminate cavitation. Figure 3.4-1 shows a comparison of the SNAP I impeller operation with and without a jet pump. This figure also clearly demonstrates the overall improvement in performance given by the jet pump. The first SNAP II pump design was to be a direct copy of the successful SNAP I pump.

3.4.1 Impeller Design

The original pump specifications were as follows:

Speed	40,000 rpm
Discharge Pressure	140 psia
Inlet Pressure	5.5 psia at 420°F
Maximum Power	240 watts
Flow	40 lb/min.

The pump performance was to fall in the area shown in Figure 3.4.1-1.

Original Pump Requirements

FIGURE 3.4.1-1

SNAP I PUMP PERFORMANCE WITH AND WITHOUT PUMP BOOST

FIGURE 3.4-1

Designing for the maximum conditions, the design point was established at:

Speed	40,000 rpm
Discharge Pressure	140 psia
Inlet Pressure	5 psia
Useful Flow	40 lb/min = 0.368 gpm
Inlet Temperature	420°F
Jet Flow	10 lb/min
Impeller Flow	50 lb/min = 0.46 gpm
Maximum Power	240 watts
Fluid	Liquid mercury

The available net positive suction head to this pumping system is:

$$\begin{aligned} \text{NPSH}_a &= H_1 - H_v - H_{vp} - H_Z \\ &= 0.88 - 0 - .092 - 0 \\ &= 0.788 \text{ ft} \end{aligned}$$

where

$$H_Z = \text{Elevation Head} = 0 \text{ FT}$$

$$H_v = \text{Velocity Head} = \frac{V^2}{2g} \approx 0$$

$$H_1 = \text{Pressure Head} = 5 \text{ psia} = 0.88 \text{ FT}$$

$$H_{vp} = \text{Vapor Pressure of Liquid at Sub-Cooler Temperature} = .092 \text{ FT}$$

IMPELLER INLET NOMENCLATURE

FIGURE 3.4.1-2

With only this small NPSH available to the pumping system, it would be impossible to have an excessive velocity head in the pump inlet. Just how much of the NPSH_a could be used for velocity was unknown. In the SNAP I system, 40 percent of the available head (H_1) was converted into velocity. By using a jet pump ahead of the impeller, and retaining the same velocity, it was felt that the jet pump would overcome all the additional

losses. These additional losses are friction in the pump inlet line from the subcooler to the pump inlet, and dynamic depression on the pump inlet blades.*

Using these assumptions, the inlet to the pump was designed:

$$H_v = 0.4 (H_I) = 0.4 (.88) = 0.352 \text{ ft}$$

The inlet velocity would be:

$$V_{IN} = \sqrt{2g H_v} = \sqrt{64.4 \times .352} = 4.76 \text{ ft/sec}$$

The inlet diameter is determined by

$$A_I = \frac{Q^{**}}{A}$$

$$D_I = \sqrt{\frac{4}{\pi} \times \frac{50 \times 144}{815 \times 60 \times 4.76}} = 0.2$$

The next design consideration, and a very important one, is the impeller OD. To establish a base for comparison, specific speed will have to be determined for the impeller.

$$N_S = \frac{40,000 \sqrt{.46}}{24^{3/4}} = 2500$$

Once specific speed is established, a discharge blade angle, β_2 , is selected. Two β_2 angles were chosen, one to match SNAP I (22.5°) and the other to obtain the flat performance required by SNAP II (90°). The test of these two impellers would provide an evaluation of pump performance with a normal design discharge blade angle of $\beta_2 = 22.5^\circ$ and a 90° discharge blade angle.

The impeller O.D. was sized and then modified to agree with SNAP I data. The impeller tip speed (u_2) required is obtained by use of a head coefficient ψ , where

* Discussed in Section 3.5, Cavitation.

** 10 lb/min added to supply jet pump.

$$\psi = H / \frac{u_2^2}{g}$$

$$u_2 = \sqrt{\frac{g H}{\psi}} = \sqrt{\frac{32.2 \times 24}{0.45}}$$

$$u_2 = 41.5 \text{ fps}$$

$$\text{for } \beta_2 = 22.5^\circ$$

$$N_3 = 2500$$

$$\therefore \psi = 0.45$$

then

$$D_2 = \frac{u_2}{\pi N} = \frac{41.5 \times 229}{40,000} = 0.238 \text{ in.}$$

Using the pump laws and data from SNAP I, D_2 should be,

$$\left(\frac{D_{2 \text{ II}}}{D_{2 \text{ I}}} \right)^2 = \frac{H_{\text{II}}}{H_{\text{I}}}$$

$$D_{2 \text{ II}} = \sqrt{\frac{24}{51} \times 0.4^2} = 0.3 \text{ IN}$$

$$u_2 = 51.7 \text{ fps}$$

$$\psi = \frac{24 \times 32.2}{51.2} = 0.292 = \frac{C u_2}{u_2}$$

$$C u_2 = \psi u_2 = 0.292 \times 51.2 = 15 \text{ fps}$$

The blade height at impeller discharge can be calculated only after a fair knowledge of the impeller configuration is known. In Figure 3.4.1-3 the pertinent dimensions are shown.

IMPELLER NOMENCLATURE

FIGURE 3.4.1-3

Once S_2 is known and the number of blades (z) established, b_2 can be found by:

$$C \quad b_2 = \frac{Q}{(\pi U_2 - z S_2) C_{m2}} = \frac{50 \times 12}{(0.3\pi \times 4 \times .047) (6 \times 8.5 \times 60)}$$

$$b_2 = .042 \text{ IN}$$

$$\begin{aligned} \text{where: } C_{m2} &= \phi u_2 & \phi &= \frac{C_{m2}}{u_2} = 0.13 \\ &= 0.13 \times 51.2 \\ C_{m2} &= 6 \text{ fps} \end{aligned}$$

The discharge velocity triangle for this design is shown in Figure 3.4.1-4 with anticipated velocities indicated.

DISCHARGE VELOCITY TRIANGLE FOR NO. 1 PUMP
 22.5° DISCHARGE BLADE ANGLE

	BASED ON EULER'S HEAD	BASED ON IDEAL HEAD	DESIRED RESULTS	TESTED RESULTS
u_2^*	51.2	51.2	51.2	51.2
C_{u2}	36.7	21.4	15	13.8
C_2	37.2	22.6	16.2	14.5
w_2	15.7	30.0	39.6	37.6
C_{m2}	6	6	6	4.5
B_2	22.5°	11.5°	9.4°	6.8°
α_2	9.3°	15.4°	21.8°	18°

* velocities in fps

FIGURE 3.4.1-4

The inlet configuration of the impeller is established by the following relationships which are also a duplication of the SNAP I method.

$$V_{in} = C_{m1} \quad \text{to keep entrance losses at a minimum}$$

$$b_1 = \frac{Q}{\pi D_1 C_{m1}} \quad \text{blade thickness } S_1 \text{ assumed negligible}$$

$$b_1 = \frac{50 \times 12}{\pi \times 0.2 \times 6 \times 815 \times 60} = .05 \text{ in}$$

b_1 should be increased slightly to make up for possible cavitation losses - booster pump style.

$$b_1 = .06 \text{ in. which makes } C_{m1} = 4.76 \text{ fps}$$

Enough information is now available to draw the entrance velocity triangle (Figure 3.4.1-5), where:

$$u_1 = \frac{40000 \times 0.2}{229} = 35 \text{ fps}$$

$$\beta_1 = \tan^{-1} \frac{C_{m1}}{u_1} \quad - \text{ assumes no pre-rotation}$$

$$\beta_1 = \tan^{-1} \frac{4.76}{35}$$

$$\beta_1 = 8^\circ$$

NO. 1 PUMP ENTRANCE VELOCITY TRIANGLE

FIGURE 3.4.1-5

With this data available, the impeller layout is drawn. Table I gives the design dimensions for this pump, as well as for the other models designed and tested. The procedure for determining these dimensions, once the proper design constants are chosen, is the same for each of the pumps.

3.4.2 Volute Design

A volute (Figure 3.4.2-1) must be designed as a collector to remove the fluid from the impeller and deliver it to the pump discharge line with a minimum of losses.

VOLUTE NOMENCLATURE

FIGURE 3.4.2-1

The main parameter which must be established is the average volute velocity, C_3 . Where:

$$C_3 = K_3 \sqrt{2gH}$$

and K_3 is a function of specific speed.

Figure 3.4.2.2 is a plot of K_3 and other important volute parameters (D_3 , P_1 , α_v) vs specific speed.

$$C_3 = 0.34 \sqrt{64.4 \times 24} = 13.35 \text{ fps}$$

TABLE I
PUMP GEOMETRY
(DESIGN)

No.	Item		No. 1	No. 2	CRU I	No. 3	No. 4
<u>IMPELLER</u>							
1	N_s		2500	2500	2500	1500	2310
2	β_2		22.5°	90°	22.5°	90°	30°
3	β_1		8°	8°	8°	13°	5°
4	D_2	IN.	0.3	0.3	0.3	0.34	0.3
5	D_1	IN.	0.2	0.2	0.2	0.174	0.170
6	u_2	fps	51.2	51.2	51.2	59.2	51.2
7	u_1	fps	35	35	35	25.6	23.6
8	cm_2	fps	6.5	6.5	6.5	8.0	6.5
9	cm_1	fps	5.3	5.3	4.8	6.1	6.5
10	b_2	IN.	.037	.037	.042	.026	.037
11	b_1	IN.	.05	.05	.06	.062	.054
<u>VOLUTE</u>							
1	N_s		2500	2500	2500	1500	2500
2	K_3		0.34	0.34	0.34	0.32	0.34
3	C_3	fps	13.35	13.35	13.35	12.85	13.35
4	D_3	IN.	0.354	0.354	0.354	0.408	0.354
5	A_v	IN. ²	0.011	0.011	0.011	.0114	0.011
6	α_v		11°	11°	11°	11.5°	11°
<u>JET PUMP</u>							
1	d_2	IN.	0.0318	0.0318	0.0318	0.0318	0.0318
2	d_1	IN.	0.2	0.2	0.2	0.174	0.170
3	d_3	IN.	0.2	0.2	0.2	0.174	0.170

VOLUTE CONSTANTS *

FIGURE 3.4.2-2

The base circle diameter D_3 is found by:

$$\alpha = \frac{D_3 - D_2}{D_2} = 0.18$$

$$D_3 = 0.354 \text{ in}$$

The best modern pumps are designed to have a constant average velocity (C_3) for all sections of the volute. This means that volute areas are proportional to their angular advancement from this cut-water. The total capacity (50 lb/min) must pass through the throat section. The other sections have only partial capacity. The throat area can be found by the following expression:

$$A = \frac{Q}{C_3} = \frac{50 \text{ lb/min}}{13.35 \text{ ft/sec} \times 815 \times 60} = .011 \text{ in}^2$$

In order to have continuity of volute elements, the volute shape must fit the following equation

$$A = b_3 D_3 \pi \sin \alpha_v$$

where α_v is the volute angle and also the angle at which the fluid enters the volute.

To simplify fabrication and to provide minimum deviation from the average volute velocity, a rectangular section, expanding radially in an Archimedes spread, was used.

Once the fluid has been collected in the volute, it can be removed in a straight diffusing section. The only velocity, convertible into pressure in the diffuser is the volute velocity C_3 (13.35 fps). The fluid was diffused to a velocity of 5 ft/sec which amounts to only 2.5 psi velocity head.

A tabulation of this volute's dimensions as well as the other models designed and tested, is shown in Table I.

3.4.3 Jet Pump

The successful operation of the jet pump in this application depends upon the centrifugal pump meeting its design requirements. This is obvious when it is realized that the jet driving power comes directly from the centrifugal pump discharge. An optimization must be made to determine how much centrifugal pump flow can be taken from the output and recirculated to drive the jet. This optimization is based on:

1. Centrifugal pump output

2. NPSH required by the impeller
3. Developed pressure of the jet pump
4. Cavitation limit on the jet pump

The optimization of these parameters indicated that 20 percent of the impeller discharge should be recirculated. This value was chosen because the calculated jet pump discharge pressure was approximately the same as the jet pump output in the SNAP I system.

The performance of the jet pump was calculated from the impulse and momentum equations and is expressed as follows:

$$V_1 Q_1 + V_2 Q_2 = V_3 Q_3$$

(see Figure 3.4.3-1)

JET PUMP NOMENCLATURE

FIGURE 3.4.3-1

$$\text{where } V_1 = 5 \text{ ft/sec} \quad (\text{assumed})$$

$$V_2 = C \sqrt{2gH_2} = 37 \text{ fps}$$

$$Q_1 = 40 \text{ lb/min}$$

$$Q_2 = 10 \text{ lb/min}$$

$$Q_3 = Q_1 + Q_2 = 50 \text{ lb/min}$$

$$V_{3T} = \frac{V_1 Q_1 + V_2 Q_2}{Q_3}$$

$$V_{3 \text{ total}} = 11.4 \text{ ft/sec} \quad H_3 \text{ velocity} = \frac{(V_{3T})^2}{2g} = 2.0 \text{ ft}$$

$$H_{3T} = H_{1 \text{ static}} + H_{3v}$$

$$H_{3T} = 0.33 + 2.0 = 2.33 \text{ ft} = 28 \text{ in. hg}$$

H_{3T} then can be divided into velocity and pressure as desired. Since the impeller inlet velocity is set for Q_3 , V_3 must be equal to V in which 5 fps or 0.39 ft of velocity head; the remainder of H_{3T} is converted into pressure. The static pressure at Section 3 of the jet pump is then 11.6 psia. The total pressure at the point would be H_{3T}

$$H_{3T} = H_{3 \text{ static}} + H_{3p}$$

$$H_{3T} = 0.38 \text{ ft} + 11.6 = 2.33 \text{ ft} = 14 \text{ psi}$$

Cavitation must also be considered in a jet pump design. The problem is not eliminated with the use of the jet pump, but moved to a different spot. The location of this cavitation point is in the throat of the jet pump. The pump fluid must pass this point before it receives additional energy from the high velocity jet. As this area is the smallest through which the flow must pass, maximum flow is defined as:

$$Q_1 = (A_2 - A_1) \sqrt{2g(H_1 - h_s)}$$

where h_s represents the loss in the suction pipe, and other symbols are as defined in Figure 3.4.3-1. Previous calculations for the centrifugal impeller showed that V_1 maximum is equal to $\sqrt{2gH_1}$ available, and A_1 is set at the same diameter as the impeller eye. This gives the jet pump a constant diameter mixing tube. A summation of these designed dimensions is shown in Table I.

3.5 Cavitation

Since cavitation presented a difficult problem in the design of the SNAP II pump, it is discussed separately. Basically, the problem is as follows:

1. The exact prediction of unworkable cavitating conditions is doubtful
2. Cavitation limits a design; it is not an absolute design factor.
3. The overall pumping requirements must be optimized against cavitation limits to obtain a suitable pump.

Cavitation is difficult to predict because it can occur at any point in the system where the pressure of the liquid reaches the vapor pressure of the liquid at the corresponding temperature.

Low pressures can be caused by turbulence, vortex motion of the fluid, rapid changes in the velocity of the fluid either in direction or magnitude, or hot spots in the system. Therefore, to reduce the possibility of cavitation, all flow passages must be smooth, without sharp turns, or sudden changes in area. In addition, the inlet blade loading of the impeller should be gradual to reduce the pressure drop (dynamic depression) caused by the blades changing the fluid velocity.

Two factors must be known to determine the cavitation limit of a pump. One is the net positive suction head available (NPSH_a); the other is the net positive suction head required (NPSH_r). The determination of the NPSH_a is relatively simple (see Section 3.4.1); however, determination of the NPSH_r parameter is difficult to make. One method available for predicting the start of cavitation for low specific speed pumps is based on the inlet velocities of the pump and an experimental coefficient. Since pressure is converted into velocity to produce flow into the pump, the proper velocities, if determined, could then be used to estimate incipient cavitation, and in turn, give the NPSH required to overcome cavitation. The empirical expression for this relationship is:

$$\text{NPSH}_r = \frac{C_1^2}{2g} + \lambda \frac{w_1^2}{2g} \quad \text{Equation 3.5.1}$$

where C_1 = average absolute fluid velocity through the impeller eye

w_1 = average relative fluid velocity at the impeller entrance

λ = an experimental coefficient (Figure 3.5-1 λ vs. Q)

This Equation was applied to No. 1 and 2 pumps to determine the impeller requirements after the inlet velocity C_1 was established from SNAP I correlations. After test data showed that Equation 3.5.1 gave good results in determining the NPSH_r, the information

DYNAMIC DEPRESSION COEFFICIENT λ (STEPENOFF)

FIGURE 3.5-1

was applied to No. 3 pump to determine the diameter D_1 for minimum $NPSH_r$.

3.6 System

Although pump output performance is an important characteristic, three secondary characteristics require consideration in order to ascertain the effect of the pump on the system. These are:

1. Power needed to drive the pump
2. Thrusts resulting from the pump
3. Fabrication and assembly of pump parts.

3.6.1 Pump Power Requirements

The power requirement of the pump can be estimated by an analysis of the losses associated with pumping. These losses are hydraulic, volumetric, and mechanical. The summation of these losses, added to the pump hydraulic output, gives the power needed to drive the pump (Equation 3.6.1).

$$\text{Equation 3.6.1 } P_{IN} = P_{out} + P_h + P_v + P_m$$

Overall efficiency of the unit can be stated as:

$$e_o = \frac{P_o}{P_{IN}} = \frac{QH}{P_{in}}$$

The overall efficiency is composed of three efficiencies which are related to the previously mentioned losses. They are defined as:

1. mechanical efficiency $e_m = \frac{Q_i H_i}{P_{in}}$
2. volumetric efficiency $e_v = \frac{Q}{Q_i} = \frac{Q}{Q + Q_L}$
3. hydraulic efficiency $e_h = \frac{H}{H_i} = \frac{H_i - \text{Hyd. losses}}{H_i}$

Algebraic manipulation of the three efficiencies will give the following relationship for overall efficiency:

$$e_o = e_v e_h e_m$$

The most important of these and the most difficult to determine is hydraulic efficiency which establishes leakage and mechanical losses. A lower hydraulic efficiency means a higher ideal head and in turn a larger impeller diameter for a given discharge head. Since leakage is a function of ideal head, and mechanical losses are a function of diameter, these losses are indirect functions of hydraulic efficiency. Once hydraulic efficiency is established, the other losses and overall efficiency of the pump can be estimated. The problem is complicated by possible cavitation which tends to lower the hydraulic output of the pump by an undeterminable amount.

In this design, it was assumed that the SNAP II pump would have the same hydraulic efficiency as the SNAP I pump (see Section 3.4). With a known diameter, the mechanical losses can be calculated from the expression,

$$\text{mechanical losses} = P_m = K_m N^3 D^5$$

$$\text{where } K_m = 28.1 \times 10^{-12} \text{ (from SNAP I data)}$$

$$\text{for No. 1 pump } P_m = 28.1 \times 10^{-12} \times 4 \times 10^4 \times (0.3)^5 = 4.6 \text{ watts}$$

The estimated hydraulic efficiency was calculated by using an ideal head of $\frac{u_2^2}{2g}$. This gives an ideal head of

$$H_i = \frac{(46.5)^2}{64.4} = 34 \text{ ft}$$

And a hydraulic efficiency of,

$$e_n = \frac{24}{34} = 70.5\%$$

The estimated hydraulic efficiency is then used to determine actual velocity triangles from the relationship of:

$$H_x = \frac{C_{u2} U_2}{g}$$

H determines the velocity triangle to be drawn (shown in Figure 3.6-1). The triangles are defined as follows:

1. Euler's Triangle

This triangle is drawn using actual blade angles. Fluid does not follow this path as the impeller does not impart energy into the fluid.

2. Ideal Triangle

Presents the path the fluid will take in absence of hydraulic losses. The ideal head is obtained from this triangle.

3. Desired Triangle

This velocity triangle must be obtained if designed (desired) performance is obtained.

DISCHARGE VELOCITY TRIANGLES
NO. 1 IMPELLER

FIGURE 3.6-1

4. Actual Triangle

Computed from information obtained from test data and the use of the head equation.

$$H_{act} = \frac{C_{u2} u_2}{g}$$

Since leakage is considered as being only a function of ideal head, extrapolation from SNAP I data gives the following results.

$$A_1 = f(H_i)$$

$$\text{SNAP I } H_i = 118 \text{ ft}$$

$$Q_L = 4 \times \frac{34}{118} = 1.15 \text{ lb/min}$$

$$Q_L = 4 \text{ lb/min}$$

An additional leakage over the pump O.D. of 7 lb/min was calculated by the use of the general Reynolds number and friction factor relationships. The total flow lost to recirculation and leakage is 8.15 lb/min. The power lost to this leakage is:

$$P_v = Q_v H_i = \frac{8.15 \times 34}{49.3} = 6.3 \text{ watts}$$

Aside from leakage over the O.D. of the impeller, the commonly used relationship of $P_v = 1/2 P_m$ would hold.

Once the mechanical and volumetric losses are known, the input power can be calculated by:

$$e_m = \frac{P_{in} - P_m}{P_{in}} = \frac{Q_i H_i}{P_{in}}$$

$$P_{in} - P_m = Q_i H_i$$

$$P_{in} = 4.6 + \frac{58.15 \times 34}{44.3}$$

$$P_{in} = 4.6 + 46 = 50.6 \text{ watts}$$

$$e_m = \frac{50.6 - 4.6}{50.6} = 91\%$$

$$e_o = e_h e_v e_m = 70.5 \times 86 \times 91$$

$$e_o = 54\%$$

Basing the overall efficiency on useful pump output

$$e_o = e_h e_v e_m = 70.5 \times 69 \times 91$$

$$e_o = 44\%$$

3.6.2 Pump Thrust

The axial thrust in a centrifugal pump is entirely a function of pressure and area. Figure 3.6.2-1 shows the pertinent areas and pressure distributions.

AXIAL THRUST NOMENCLATURE

FIGURE 3.6.2-1

H_v = average pressure before entrance into volute

$$H_v = H_{Dev} - H_{vel}$$

$$H_v = 24 \text{ ft} - \frac{C_3^2}{2g} = 24 - \frac{13^2}{2g} = 20 \text{ ft}$$

$$T_1 = (A_2 - A_1) \frac{H_v}{2} \rho = 2.74 \times 10^{-4} \times \frac{20}{2} \times 810$$

$$T_1 = 2.2 \text{ lb}$$

For smooth back

$$T_2 = (A_2 - A_s) \left[H_v - 1/8 \frac{u_2^2 - u_s^2}{2g} \right] \quad \text{for } 1/8 \text{ in. shaft}$$

$$= (4.07 \times 10^{-4}) \left[20 - 1/2 \frac{(51.2^2 - 21.8^2)}{28} \right] 810$$

$$T_2 = 5.2 \text{ lb}$$

It was felt that in CRU designs the ease of machining would overcome the penalty of added thrust, if the pump shaft was of the same diameter as the pump O.D. The thrust will then be:

$$T_1 = 2.2 \text{ lb}$$

3.6.3 Fabrication and Assembly

The size of the parts imposed a difficult problem in the fabrication of the impeller and volute. (Figure 3.6.3-1) To allow sufficient room for a cutter, the distance between adjacent impeller blades, and the thickness of the blades, required careful planning. The minimum practical cutter size is approximately 0.030 in. diameter. To keep good impeller proportions, blade thickness was tapered from 0.030 at the O.D. to 0.006 at the ID. As machining tolerances (10 percent of controlled dimensions) would not allow for a mixed flow blade, a plain (constant radius curve) blade was used.

Inspection of these parts also presented a problem. After trying several methods without success, the impeller and volute curves were set upon a grid system and measurement was taken from a fixed reference point.

The size of the impeller made its attachment to the main shaft difficult. In order to have stock for threading and pinning, the shaft size was made equal to the diameter of the impeller. Although this arrangement solves the mounting problem satisfactorily, it adds the penalties of thrust and pump leakage to the system.

Since the pump has no backface to compensate for the thrust exerted on the front-face, more thrust was added to the combined rotating unit. Complicating this problem is the fact the thrust was set in the direction of the thrust from the NaK pump. This added thrust is tolerable as long as the total thrust in the CRU is within bearing capacity.

Pump capacity was also handicapped by added leakage. Again, as long as the impeller can handle this extra capacity, the penalty is not too severe. If these penalties become severe during the development of the CRU's, the shaft can be reduced in diameter and back vanes added to the pump, which would reduce both pump thrust and external leakage.

PUMP PARTS SHOWING RELATIVE SIZE

4.0 PUMP TEST PROGRAM

4.1 Test Objectives

The test program for the SNAP II mercury pump was developed to evaluate the performance of the pump at conditions which simulate, as closely as possible, the actual system. The overall objectives of the program were to determine actual pump performance and methods for improvement of design. Specifically, the tests were directed towards determining:

1. Head, power, and efficiency versus flow relationships
2. Inlet pressure requirements (cavitation curves)
3. Performance versus speed relationships
4. Geometry effect on performance (front face clearance, jet nozzle spacing).
5. General operation behavior - start-up ability, pressure fluctuations, and time responses.

4.2 Test Rig Description

The basic pump test facility was built and developed for the SNAP I mercury pump. Only minor changes were required, primarily to accommodate a larger flow rate. Figure 4.2.1 shows the pump test rig schematically. Actual photographs of the test rig can be seen in Figures 4.2-2 and 4.2-3. A description of the test rig is as follows:

1. The driver for the pump is a precision built motor with an integrated rigid spindle in which the pump shaft is held by a collect chuck. Motor speed can be regulated by a rheostat control and indicated by frequency signals from a magnetic pick-up at the chuck.
2. Power measurements were obtained by electrical calibration of the motor, using a dynamometer. An electrical input versus brake power output calibration curve was obtained (Figure 4.2-4). A check of the method of power measurement was made on a transmission dynamometer designed and built for this application. Figure 4.2-5 shows the general arrangement of this dynamometer.
3. Flow measurements were made with sharp-edged orifice plates and indicated on 30 in. inverted "U-tube" manometers. The orifices were calibrated by weighed flow checks.
4. High pressure measurements were made using stainless steel bourdon tube gages. U-tube manometers were used to measure low and vacuum pressures.

PUMP TEST RIG HG FLOW SCHEMATIC

FIGURE 4.2-1

EXTERIOR VIEW - PUMP TEST RIG

INTERIOR VIEW - PUMP TEST RIG

TYPICAL MOTOR CALIBRATION CURVE

FIGURE 4.2-4

PUMP DYNAMOMETER

5. Temperature measurements were made with iron-constantine thermocouples. The read-out of the thermocouples was either on a 24-station potentiometer, a single-point controlling-indicating pyrometer, or a single-point strip chart recorder-controller.
6. High-temperature operation was obtained by heating the mercury with electrical resistance immersion heaters in the supply reservoir. Temperature control was accomplished by an indicating-controlling pyrometer.
7. Accurate control of pump inlet conditions was obtained through the use of a specially designed separator tank. Mercury was pressure pumped to the top of the tank and passed by gravity over a series of baffled trays. Various vacuums were drawn into the vessel by a vacuum pump. The liquid level in the tank could be observed through a sight gage. The liquid head above the pump centerline, plus the vapor pressure, controlled pump inlet pressure. This method also insured that the mercury was degasified. An added feature of the system is that it simulates the actual subcooler and pump inlet line, with the exception of gravity.

4.3 Test Procedure

Testing, in the pump test rig (PTR) was conducted by running five types of tests on four different pumps. The pumps were composed of four impellers, two volutes and three jet-centrifugal housings, mounted as shown in Figure 4.3-6. Table II gives the designation and numbering of these pumps. The following tests were conducted:

1. Head, Power and Efficiency versus Flow Tests - These tests were run by keeping a constant speed and inlet pressure and varying flow. Pressure, flow, and power were recorded. This was done at various speeds and inlet pressures. In order to evaluate the actual impeller and jet pump performance, these tests were run with and without the jet pump.
2. Cavitation Tests - Data for cavitation characteristics was obtained by keeping speed and flow constant, and varying inlet pressure. This was done at various speeds and flows. The data was plotted as H versus NPSH for each speed and flow run. By running cavitation tests with and without the jet, jet pump performance was obtained.
3. Performance vs. Speed - Obtained by setting design conditions of flow, pressure and speed on the pump and then varying speed.
4. Geometry Effects - The effects of geometry on pump performance was determined by running the test stated above on various,

S/N - 1 IMPELLER LAYOUT

FIGURE 4.3-1

S/N - 2 IMPELLER

FIGURE 4.3-2

FIGURE 4.3-3

FIGURE 4.3-4

CRU NO. 1 PUMP

FIGURE 4.3-5

GENERAL PUMP TEST CONFIGURATION

IMPELLER
8 x 90°

DIFFUSER PLATE

IMPELLER
8 x 22.5°

FIGURE 4.3-6

TABLE II

PUMP NUMBERING AND DESIGNATIONS							
Pump No.	Impeller		Volute		JET-Centrifugal Housing		Impeller Figure No.
	S/N	Designation	S/N	Drawing No.	S/N	Drawing No.	
1	1	8° x 22.5° D-27450	1	D-27451	1	D-27449	4.3-1
2	2	8° x 90° D-27635	1	D-27451	1	D-27449	4.3-2
3	3	12° x 90° D-22635	2	D-28467	2	D-28466	4.3-3
4	4	5° x 30° D-29026	1	D-27451	3	D-29027	4.3-4
CRU I	1-A	8 x 22.5 D-27681	1	D-27451	1	D-27792	4.3-5
*CRU II	3	12° x 90°	2		2		4.3-3
*PCS I	3	12° x 90°	2		2		4.3-3

* CRU II & PCS I have a No. -3 Pump

- 1) Front-face clearances
 - 2) Nozzle to throat spacing of the jet
 - 3) Volute and housing irregularities.
5. Start-up Tests - The starting ability of the pump was determined by priming the pump and stabilizing it at or near design. The inlet pressure was maintained constant and the pump was stopped by cutting power. After approximately five seconds of standstill, power was applied to the pump at various rates. Brush traces of discharge pressure and speed were obtained to record the starts.

5.0 DISCUSSION OF TEST RESULTS

5.1 No. 1 and 2 Pumps

As shown in Table II, the only difference between No. 1 and 2 pumps is the exit blade angle of the impeller. No. 1 pump was a direct copy from SNAP I, and No. 2 pump was an attempt to obtain the flat performance required by SNAP II.

5.1.1 No. 1 Pump Performance

The general performance obtained by No. 1 pump is shown in Figure 5.1.1-1. This pump gave relatively poor performance. It did not have the flow or pressure required. Cavitation was so severe that the jet pump could not produce a net gain in performance. The test results dictate that the inlet to the impeller must be redesigned to reduce NPSH requirements of the impeller. A layout of this pump which is similar to the other models tested is shown in Figure 5.1.1-2.

5.1.2 No. 2 Pump Performance

The performance of pump No. 2 was an improvement, but still did not meet requirements. Although cavitation was severe, the jet pump was able to overcome a large portion of this difficulty and produce a net gain in overall pump performance. The calculated $NPSH_r$ is 48 psia based on equation 3.5.1. However, the impeller was able to operate at a value less than the calculated $NPSH_r$ (Figure 5.1.2-1). The results of the cavitation tests on the impeller showed that an inlet pressure of 14.5 psia, plus the jet, flattened out the cavitation curves at zero useful flow (5.1.2-2). This amounted to an impeller inlet pressure of approximately 37 psia, the jet pump developing 23 psi.

5.1.3 CRU-I Pump Test Results

The analysis of the performance of No. 1 and 2 pumps, using the actual measured dimensions of the impeller, gave a better picture of how and why these pumps performed as they did. Even with high inlet pressure, they did not have the flow capacity required. Close examination of these impellers after test showed that the blade heights had been reduced by rubbing. This rubbing was caused by an assembly error. The smaller blade heights reduced the flow area and in turn the capacity of the pumps. Table I shows the actual dimensions of the various impellers. The blade heights (b_1 & b_2) for No. 1 and 2 pumps were designed to be about 10 percent higher than given in Table I.

The pump used in CRU I had the full blade height, and gave the desired flow, when sufficient inlet pressure was available.

The performance of the CRU I pump is compared with No. 1 pump in Figure 5.1.3-1. The increase in flow performance can be accounted for by the increase in flow area, and a reduction of front face clearance from .004 to .002 inches. This pump was also used in the

NO. 1 PUMP PERFORMANCE

FIGURE 5.1.1-1

TYPICAL PUMP LAYOUT

FIGURE 5.1.1-2

NO. 2 PUMP GENERAL PERFORMANCE

FIGURE 5.1.2-1

NO. 2 PUMP CAVITATION CURVE

FIGURE 5.1.2-2

CRU I PUMP VS. NO. 1 PUMP PERFORMANCE

FIGURE 3.1.3-1

487 hour endurance run of CRU-1. The performance was satisfactory as long as inlet pressure remained high.

Since the impeller was destroyed in disassembly of the CRU, close examination for erosion damage could not be made. There was, however, no noticeable damage to the impeller when viewed with the naked eye. Some erosion damage did occur on the front wall of the impeller casing (see Figure 5.1.3-2). This type of damage is most likely to occur when the pump is operated at capacities above normal, destroying the streamlining of the inlet, causing a discrepancy between the angles of incoming flow and the vanes. In this case, the prerotation is in a direction opposite to the impeller rotation. Figure 5.1.3-3 shows a typical velocity triangle for this type of inlet flow.

EROSION DAMAGE - CRU I PUMP

FIGURE 5.1.3-2

INLET VELOCITY TRIANGLE NORMAL CAPACITY AND OVER CAPACITY

FIGURE 5.1.3-3

High leakage was not present in the first run of CRU-I; it was within reasonable limits (3 to 5 lb/min). The second assembly of CRU-I had the same impeller, but the radial clearance was increased from .003 in. to .005 in., causing the leakage to increase to 10 to 15 lb/min.

5.2 Analysis of Pump No. 1 and 2 to Obtain No. 3 Pump

The tests of pumps 1 and 2 reveal one very important design procedure error: the method of sizing the pump inlet. The direct copy of the SNAP I technique of allowing a certain percentage of the $NPSH_a$ for velocity head in the pump inlet resulted in an impeller $NPSH_r$ higher than the amount given by both the jet pump and the inlet pressure available. Equation 3.5.1 shows that the minimum $NPSH_r$ for the SNAP II specifications is 4 ft at an inlet impeller diameter of 0.143 inches. Since the condenser can supply only 1 ft (6 psia) to the pump, a jet pump must supply the rest. The jet pump for No. 1 and 2 pump could only produce approximately 2 ft of head. The additional inlet pressure could be obtained if more flow was bypassed or a higher pressure was used to recirculate the flow.

At the time this problem developed, there were several indications that the inlet pressure required by the boiler would be higher than the original 140 psia, and since added recirculating flow would tend to increase the $NPSH_r$, a higher pressure pump was designed and tested (No. 3 pump). An estimate of this additional pressure requirement gave the following pump design point:

$$\begin{array}{ll} \Delta P & = 320 \text{ psi} \\ \text{Useful flow} & 40 \text{ lb/min.} \end{array}$$

This high pressure was chosen to meet system contingencies, and to insure adequate jet pump output. Although the pressure is high, it was felt that a combination of the pump not meeting this new design point and the anticipated increase in system pressure requirements would result in a pressure which could be trimmed slightly by a throttling valve or orifice to tune in each system for exact flow and pressure requirements.

The design specific speed for this impeller ($S/N-3$) is:

$$N_s = \text{rpm} \frac{\sqrt{gpm}}{H^{3/4}} = 40,000 \frac{0.46}{55^{3/4}}$$

$$N_s = 1500$$

A discharge blade angle of 90° was chosen to obtain a flat head versus flow curve.

5.3 No. 3 Pump Test Results

The No. 3 pump showed a marked improvement in performance over the previous designs. Although the output of the pump was very good, an even more significant accomplishment was suction performance. An overall suction specific speed $S = \frac{\text{rpm} \sqrt{\text{gpm}}}{(\text{NPSH})^{3/4}}$ of 27,000 was obtained.

The design inlet performance of the pump is shown in Figure 5.3-1. The performance was stable and reproducible. The impeller performance, without the jet pump, appears in Figure 5.3-2. This curve indicates that with sufficient NPSH, the efficiency of the impeller is very respectable (50 percent).

The inlet pressure requirements are shown in Figures 5.3-3 and 5.3-4. These curves not only demonstrate what the pump can do, but can be used for comparison purposes. The equation $\phi = \frac{\text{NPSH}}{N_s}$, with values of ϕ taken from Hydraulics Institute data, set the accepted standard^{fi} (Figure 5.3-5). For a N_s of 1500 the Institute gives a value of $\phi = .07$. This means that the operating NPSH of the impeller should be,

$$\text{NPSH} = \phi (H) = .07 (280) = 19.6 \text{ psia}$$

The operating NPSH for the No. 3 pump was only 6 psia. The jet pump produces a 10 psi ΔP , giving the impeller inlet 16 psia. The ϕ for the impeller is 0.055, which means the pump is operating with a NPSH considerably lower than it should as indicated by standard pump references. This added inlet performance is obtained by having a high inlet velocity (C_{in}) and a small inlet diameter (d_1). The efficiency of the jet pump is increased, because it does not have to convert most of its output to pressure. This high velocity is used by the impeller, thereby eliminating losses incurred if full diffusing was used.

The performance of the pump at various speeds is shown in Figure 5.3-6 and 5.3-7. These curves deviate only slightly from the pump laws. This deviation, however, is an overall characteristic caused by leakage and the recirculating jet flow. Broken down to actual impeller performance, the pump laws would hold. This again proves that there is no discontinuity in the pump design. The data on these curves is used to determine pump performance during starting and stopping (assuming the pump is primed).

Front face clearance of the impeller has a definite and discernible effect on the performance of the impeller. In fact, it can be used to adjust the performance of the pump. A larger clearance produces a lower, flatter H-Q curve; however, as shown in Figure 5.3-8, an increase in clearance beyond .010" does not reduce performance appreciably. The H-Q curves at various front face clearances are shown in Figures 5.3-9 and 5.3-10. The system has a choice as to which performance curve it will use.

NO. 3 PUMP PERFORMANCE -
21 PSIA INLET - NO JET PUMP
DESIGN SPEED

FIGURE 5.3-1

S/N - 3 IMPELLER PERFORMANCE

FIGURE 5.3-2

FIGURE 5.3-3

FIGURE 5.3-4

HYDRAULIC INSTITUTE VALUES OF SINGLE SUCTION, DOUBLE SUCTION

FIGURE 5.3-5

PUMP NO. 3
PERFORMANCE VERSUS SPEED

FIGURE 5.3-6

FIGURE 5.3-7

PERFORMANCE VERSUS FRONT FACE CLEARANCE

FIGURE 5.3-8

FIGURE 5.3-9

PUMP NO. 3
HEAD VS. FLOW AT VARIOUS FRONT FACE CLEARANCES
INLET 3.0 PSIA - SPEED 40,000 RPM - FULL JET

FIGURE 5.3-10

The start-up ability of the pump was investigated in the PTR. (Pump Test Rig) These tests were run to determine the limits surrounding pump start-up. Due to PTR limitations, the lowest inlet pressure which could be obtained for a start-up tests was 8 psia. The best the pump could do is shown in Figure 5.3-11. This result breaks down to a maximum acceleration of 416 rev/sec^2 for the first 0.2 seconds of running. Once the pump was primed, acceleration of 1000 rev/sec^2 to 40,000 rpm still allowed the pump to run.

5.4 No. 4 Pump Test Results

Pump No. 4 was designed to meet the original pump requirements. It was an attempt to correct the inlet configuration of the No. 1 pump to minimum NPSH requirements. The tested performance characteristics of the pump are shown in Figure 5.4-1, which indicate that although the pump performance passed through the design area, it did not meet full system requirements. The slope of the H-Q curve was too steep. Low inlet pressure operation was obtained, which was a big improvement over the No. 1 pump, but flow output of the pump was limited. Enlarging the flow passages of the impeller would aid in getting more flow out of the pump.

The power requirement of this design should be the same as in No. 1 and 2 pumps, because of size and hydraulic output. The predicted power of 50 watts is in good agreement with the tested value.

Jet pump performance was good. Output, estimated from its effect on centrifugal pump performance, was very close to design. An analysis of Figure 5.4.2 shows that the jet pump gave the impeller an equivalent inlet pressure of 15 psia when driven with 140 psia fluid. The calculated jet pump output was 14 psia. The jet pump efficiency was then:

$$e_j = \frac{Q_1 (H_3 - H_1)}{Q_2 (H_2 - H_3)} = \frac{30 (15 - 5)}{10 (140 - 15)} = 22\%$$

This is a relatively high efficiency for a jet pump of this type, but it must be remembered that this efficiency is defined as jet pump effectiveness (impeller performance increase) over the energy put into the jet pump by the centrifugal pump.

TYPICAL BRUSH TRACE OF PUMP START-UP NO. 3 PUMP

ΔP ACROSS IMPELLER = 2.0 PSIA

ΔT 0 - 20,000 RPM = 5.0 SEC.

ΔT 20,000 - 30,000 RPM = 0.1 SEC.

ACCELERATION DURING FIRST 0.2 SEC. = 4.16 REV/SEC²

PUMP INLET PRESS. = 8.0 PSIA

FIGURE 5.3-11

SNAP II PUMP

FIGURE 5.4-1

JET PUMP EVALUATION NO. 4 PUMP

FIGURE 5.4-2

6.0 CONCLUSIONS

6.1 Pump Testing

The experimental results obtained demonstrate the capability of each pump design. The influence of the various systems problems was also investigated. Examination of the H-Q curves of each design indicates pump capabilities, and can be used to evaluate the pump in terms of system requirements. The more successful pump designs (No. 3 and 4) came very close to meeting their design parameters.

6.2 Pump Design

The design procedures used in these tests is confirmed within reasonable limits by the test data. This procedure is based on the facts that:

1. Specific speed dictates the general proportions of the pump.
2. Inlet sizing must be an optimization of NPSHa, speed and flow. A minimum NPSHr value can be used if this value is made available by either the NPSHa or a boost pump.

6.3 Pump Performance

The success of a pump design is strictly a function of its ability to meet system requirements. From this standpoint, the No. 3 pump is the most successful design. It can supply enough flow and pressure at the proper inlet conditions to supply the boiler and the bearings.

This pump design is not only adequate in meeting system requirements, but it also runs within the acceptable limits of power and thrust. The overall efficiency of 30 percent indicates that the pump design in itself is sound.

7.0 RECOMMENDATIONS

7.1 Pump Design

1. Increase blade height to allow maximum pump flow and thereby provide increased flow capacity and a flatter pump characteristic curve.
2. If leakage over the pump is excessive, back vanes will have to be installed.
3. Pump vapor swallowing capacity should be increased to minimize the effects of air in the system, and thereby obtain more reliability.

7.2 Pump Testing

1. Vapor swallowing capacity of the pump must be investigated to determine limits of operation.
2. Start tests should be run in a combined rotating unit so actual dynamic conditions can be simulated.

8.0 APPENDIX

8.1 List of Symbols - General

N	=	Rotating speed	RPM
H	=	Developed head	ft ¹
N _s	=	Specific speed	$\text{RPM} \frac{\text{GPM}}{H^{3/4}}$
Q	=	Flow rate	lb/min
NPSHa	=	Net positive suction head available	ft, psi, in-Hg
NPSHr	=	Net positive suction head required	ft, psi, in-Hg
H _v	=	Velocity head	ft
H _{vp}	=	Vapor pressure of liquid at the temperature of that liquid	ft or psi or in-Hg
H _z	=	Static elevation head	ft
H _s	=	Shut-off head	ft
ρ	=	Density of fluid	lb/ft ³
ψ	=	Dimensionless head coefficient	$\frac{H}{u^2/g}$
ϕ	=	Dimensionless capacity coefficient	$\frac{C_m^2}{u^2}$
K _{u2}	=	Speed constant	$\frac{u_2}{\sqrt{2gH}}$
Q _I	=	Ideal flow - Q _{act} plus Q _L	lb/min
Q _{act}	=	Act flow	lb/min
Q _L	=	Leakage flow	lb/min
e _o	=	Overall efficiency =	e _h e _v e _m
e _h	=	Hydraulic efficiency	$= \frac{H_{act}}{H_I}$
e _m	=	Mechanical efficiency	$\frac{Q_I H_I}{P_{in}}$
P _{in}	=	Power Input to pump	watts

P_o = Hydraulic work out
 g = Acceleration due to gravity

watts
 fps²

IMPELLER

BASIC IMPELLER SYMBOLS

ENTRANCE VELOCITY TRIANGLE

DISCHARGE VELOCITY TRIANGLE

u = peripheral velocity of impeller
 w = relative velocity of flow
 c = absolute velocity of flow

fps
 fps
 fps

Subscript 1 refers to the entrance, subscript 2 to the discharge. Tangential components of relative and absolute velocities have u for a subscript. The meridional velocities are denoted by C_{m1} , and C_{m2} for entrance and discharge respectively.

D_1	- Inlet Diameter	Inch
D_{ave1}	- Average Inlet Blade Diameter	Inch
D_2	- Impeller O.D.	Inch
b_1	- Inlet blade height	Inch
b_2	- Discharge blade height	Inch
β_1	- Inlet blade Angle	Degree
β_2	- Discharge blade angle	Degree
Z	- Number of blades	
S	- Blade thickness	Inch

VOLUTE

α_v - Volute angle $\approx \alpha_2$

D_3 - Base circle dia. Inch

D_2 - Impeller O.D. Inch

$$\alpha = \frac{D_3 - D_2}{D_2}$$

- A_v - Largest volute area in^2
 r_v - Radius at 360° from tongue inch
 C_3 - Average volute velocity = $K_3 \sqrt{2gh}$ fps
 K_3 - Volute design factor = $f(N_s)$

JET-PUMP

- A_1 = Throat Area in^2
 A_2 = Nozzle Area in^2
 H_1 = Inlet Pressure to Jet Pump psia
 H_3 = Developed Pressure psia
 H_{T3} = Total Head at Impeller Entrance ft
 H_2 = Driving Pressure psia
 Q_1 = Pumped Flow lb/min
 Q_2 = Driving Flow lb/min
 Q_3 = $Q_2 + Q_1$ - Impeller Flow lb/min
 V_1 = Inlet Velocity fps
 V_2 = Driving Velocity fps
 V_3 = Velocity After Mixing fps

$$R = \text{Area Ratio} = \frac{A_2}{A_1}$$

$$M = \text{Capacity Ratio} = Q_1/Q_2$$

$$N = \text{Head Ratio} = \frac{H_3 - H_1}{H_2 - H_3}$$

$$e_j = \text{Jet Pump Efficiency} = M N$$

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SNAP II

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