LEVEL II (12)

ADA084920

CR 80.015

CIVIL ENGINEERING LABORATORY
Naval Construction Battalion Center
Port Hueneme, CA

Sponsored by
NAVAL SEA SYSTEMS COMMAND

DEVELOPMENT OF A SEAWATER HYDRAULIC VANE MOTOR FOR DIVER TOOLS

April 1980

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An Investigation Conducted by
WESTINGHOUSE ELECTRIC CORPORATION
Oceanic Division
P. O. Box 1488
Annapolis, MD

N00123-78-C-1057

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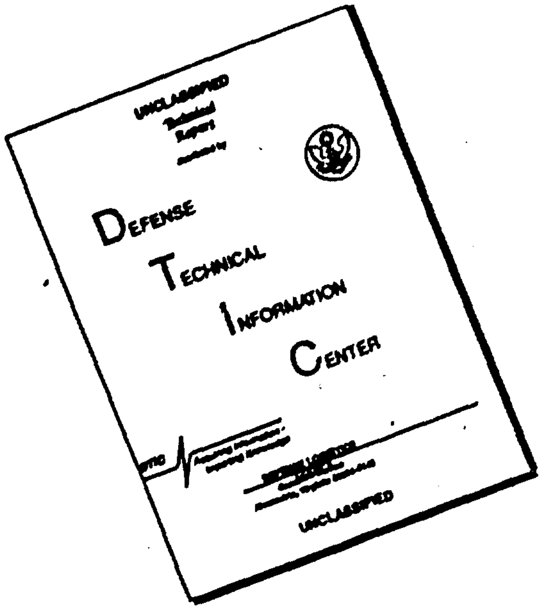

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REPORT DOCUMENTATION PAGE		READ INSTRUCTIONS BEFORE COMPLETING FORM
1. REPORT NUMBER: 19 CR-80.015	2. GOVT ACCESSION NO. AD-A084 920	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle): 6 DEVELOPMENT OF A SEAWATER HYDRAULIC VANE MOTOR FOR DIVER TOOLS		5. TYPE OF REPORT & PERIOD COVERED: 9 Contract Report. Oct 78- Jan 80
		6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(s): Westinghouse Oceanics Division		8. CONTRACT OR GRANT NUMBER(s): 15 N00123-78-C-1057
9. PERFORMING ORGANIZATION NAME AND ADDRESS: Westinghouse ~~Electric Corp.,~~ Oceanic Div. P. O. Box 1488 Annapolis, MD 21404		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS: 3.1450-1
11. CONTROLLING OFFICE NAME AND ADDRESS: Civil Engineering Laboratory Naval Construction Battalion Center Port Hueneme, CA 93043		12. REPORT DATE: 11 Apr [redacted] 80
		13. NUMBER OF PAGES: 115
14. MONITORING AGENCY NAME & ADDRESS (if different from Controlling Office): Civil Engineering Laboratory Naval Construction Battalion Center Port Hueneme, CA 93043		15. SECURITY CLASS. (of this report): UNCLASSIFIED
		15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

Approved for publice release; distribution unlimited.

12 119

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

Diver tools, Hydraulics, Seawater, Materials, Corrosion, Underwater tools, Vane motors, Hydraulic equipment, Underwater salvage, Working diver, Hydraulic motors, Seawater Hydraulics

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

A compact efficient hydraulic vane motor capable of operating with pressurized seawater as the working fluid has been successfully developed for use with diver tools. The motor occupies a volume of 23 $in.^3$ and weighs less than five pounds. With 1,000 psi seawater a six GPM flow rates the motor delivered 3.3 hp at 1,585 RPM with 80% overall efficiency

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The motor operated for 50 hours without component failure. Results of the design and development effort are presented.

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SECTION 1

INTRODUCTION

Most of the power tools in use by Navy divers are powered by positive displacement oil hydraulic motors. Dual hoses supply and return pressurized hydraulic oil from a power supply located on a surface support craft to the tool in the divers hand. These tools have greatly increased the diver's capabilities and effectiveness to perform assigned tasks. As with most any system selected for use in the ocean, oil hydraulic systems have disadvantages, some of which are: (1) leakage of working fluid contaminates the environment, (2) leakage of seawater into the system readily damages precision components, (3) from the diver standpoint, the need for two hoses burdens tool handling in areas where heavy surge and strong currents exists, and (4) logistics problems are created by shipping large quantities of oil. The use of filtered seawater in place of hydraulic oil eliminates most of the disadvantages of oil hydraulic systems.

A relatively compact and efficient hydraulic motor capable of operating with pressurized seawater as the working fluid has for some time been considered the missing link in seawater tool development. The desired physical and operational requirements for such a motor are summarized in Table 1. A summary of past efforts to develop seawater motors is contained in Reference 1. The major obstacle to the successful development of seawater motors has been the selection of materials for motor components to operate under both the harsh mechanical environment internal to the motor and the low lubricity, low viscosity and highly corrosive nature of the seawater working fluid.

Based on these requirements a comprehensive analysis of the mechanical/chemical requirements of components of positive displacement motors was conducted under contract to the U.S. Navy Civil Engineering Laboratory (CEL) in 1977. It is described in detail in Reference 2. As a result of this analysis and a series of materials screening tests, the study concluded that materials are available (see Table 2) to meet the requirements of heavily loaded critical motor components in both double entry vane and piston motors. In the context of overall suitability to seawater operation (including matching operational requirements as stipulated in Table 1, complexity, solutions to material problems, life cycle, ruggedness, fabricability, and other factors) top rating was given to the development of the double entry vane type motor.

PROGRAM OBJECTIVE

Following completion of the materials study for seawater hydraulic tool motors, CEL contracted the Oceanic Division of Westinghouse Electric Corporation to develop and evaluate an experimental, seawater hydraulic motor. Conforming to the findings of the materials study, CEL selected a positive displacement, double entry, vane-type motor operating in an open circuit mode. In

1. S.A. Black, "Preliminary Analysis of Seawater Hydraulic Tool Motors", TM 43-77-06, December 1976.

2. "Material Study for High Pressure Seawater Tool Motors", Contract Report No. CR 78.012, Mechanical Technology, Inc. (MTI) Latham, NY, April 1978.

this mode, the working fluid had to be exhausted at an outlet port on the motor and not returned directly to the supply pump, as is the case with oil hydraulic tools. The use of a fluid other than seawater for power, lubrication, etc. was not allowed.

The physical and operational characteristics cited in Table 1 are intended to be practical, yet span the motor requirements of a variety of diver tools. Thus, the development of the demonstration motor need not be complicated by additional requirements unique to a particular tool. The power level sought is ample for most diver hand tools, and the size and weight are compatible with the need to be ultimately handled by a diver. Requiring low flow rates and efficiency comparable to that of an oil hydraulic motor is intended to minimize the size and stiffness of the motor supply hose.

The purpose of this development program was to demonstrate that not only are the materials needed for such a motor available but also the design technology required is within the state-of-the-art. Moreover, proving in laboratory tests that a seawater hydraulic motor can provide performance equivalent to that of an oil hydraulic motor helps demonstrate that seawater powered hydraulic tools for divers the ultimate goal of this program can be successfully developed.

Table 1. Desired Motor Characteristics

Feature	Contract Objective
Power (HP)	≥ 3
Speed (RPM)	1,000 - 15,000
Overall Efficiency (%)	≥ 70
Supply Pressure (PSI)	$\geq 1,000$
Flow Rate (GPM)	≤ 10
Submerged Weight (Lb)	≤ 15
Volume (In.^{3})	≤ 50
Operational Life (Hrs) (With Sparing)	≥ 50

Table 2. Recommended Materials for a Seawater Vane Motor

Component	First Choice	Alternate
Shaft	Inconel 625	Inconel 625
Sleeve Bearing	Torlon 4301/Nitronic 50	Torlon 4301/Nitronic 50
Vanes	Torlon 4301/Nitronic 50	Tungsten Carbide/Nitronic 50
Rotor	Inconel 625	Al_2O_3/Inconel 625
Ring Track	Inconel 625	Al_2O_3/MP35N
Thrust Plates	Torlon 4301/Nitronic 50	Torlon 4301/Nitronic 50

SCOPE OF WORK

To show that a seawater hydraulic vane motor having a useful operating life could be developed, Westinghouse was required to first conduct a comprehensive preliminary design and material analysis. The primary purpose of preliminary design was to establish motor characteristics, to determine material requirements, to investigate techniques for reduction of bearing and vane tip loads, and to select materials for use in motor design and fabrication. In essence, the motor characteristics which affected material requirements had to be established.

Then, using this information and the data from the earlier materials study,[3] as background, Westinghouse was required to define material requirements for the motor components and to select candidate materials. For component areas where material data were insufficient, material tests which closely represented the loading and motion expected within the motor were required. The purpose of this effort was the selection of materials for the demonstration motor.

Upon completion of the preliminary design steps, Westinghouse was required to prepare a detailed motor design and build and evaluate the motor. As shown in the next section, virtually all of the design effort was actually accomplished during the preliminary design and materials selection stages of this project. As shown earlier in Table 1, the design life of the motor was to be 50 hours. A demonstration test lasting five hours, of which at least 1/2 hour was at maximum power, completed this phase of development.

REPORT ORGANIZATION

Section 1 has defined the scope of work that was required to be accomplished by this project. How that work was done is described in Section 2. That section covers the details of the preliminary design effort and selection of materials for the motor. Materials selection was a two-step process, in which material candidates were first identified by matching known material characteristics to requirements defined during preliminary design. Then a final selection was made via actual tests, which are described in Section 5.

The program through these initial selection tests and life tests is identified in this report as Phase I. At the conclusion of this phase, it became apparent that additional improvement in motor performance could be achieved through a modest extension of the initial program. The improvement program is identified in this report as Phase II.

Details of the motor which ultimately evolved are contained in Section 3. Section 4 discusses the test program and test results. Finally, Section 5 states the significant results of the whole program and recommends several continuation projects which are logical sequels.

To keep the reader from becoming lost in detail in the main body of this report, ample use has been made of appendices.

3. Ibid

SECTION 2

DEVELOPMENT APPROACH

As indicated in Section 1, Westinghouse was contracted to develop and evaluate an open circuit, balanced, vane-type hydraulic motor using seawater as the working fluid. In designing such a motor, three characteristics of seawater (see Table 3) presented major technical problems not encountered when a more traditional petroleum-based hydraulic fluid is used. Seawater causes corrosion; oils retard it. Seawater is a poor lubricant, while oils are quite good. Worse yet, seawater has much lower viscosity than oils, which means that to achieve equivalent volumetric efficiency, the design clearances of a seawater system must be an order of magnitude smaller than those found in an oil system.

As shown in the next section of this report, the solution to the corrosion problem was to use materials which are inherently corrosion resistant. This often meant that costly materials and difficult machining problems had to be faced. Solution of the lubrication problem required the selection of materials which exhibit low friction and wear when moving relative to each other in a seawater environment. The very nature of a vane motor helped resolve the smaller clearance issue, for its vanes are self-adjusting at their tips. The selection of self-adjusting, flexible sideplates also helped minimize the number of areas in which the tight running clearances had to be achieved through high precision machining.

The selection of suitable combinations of materials was basic to the solution of each of these three problems. With this in mind, Westinghouse undertook the development program depicted in Figure 1. (The four steps in Figure 1 should be viewed only as the four basic program tasks; they are only broadly indicative of task sequence, for the tasks are strongly interrelated.)

Table 3. Technical Obstacles

• **Low Lubricity**
• **Corrosiveness**
• **Low Viscosity**

PRELIMINARY DESIGN

To improve the possibility of attaining the contract objectives, Westinghouse set for itself design goals which are more stringent than the technical goals of the contract. The relationship between them is depicted in Table 4. As shown therein, a higher output power goal was set, but the overall efficiency desired of the motor was not relaxed. To ease pressure-velocity (PV) loading on bearings, vanes, and other moving parts, relatively low motor speed and water supply pressure were chosen. Likewise, a compact and relatively lightweight motor was specified at the outset. Selection of such design goals forced development of a motor whose characteristics are as close to those needed for a seawater hydraulic tool motor as could then be anticipated.

Development of the preliminary design for the motor was then relatively straightforward analytical engineering. (Details of this design process, along with pertinent calculations, are contained in Appendices B and C. Moreover, details of the motor itself are described in the next section of this report.) The process itself is iterative, i.e., performance calculations for the initial design led to a revision of that design to more closely meet the design goals which had been set earlier. This process is iterative in a macroscopic as well, for considerable modifications to the design of the motor were made as a result of actual motor test performance. For these reasons, the development sequence illustrated in Figure 1, while generally correct, is not precise.

Among the results of preliminary design is the fact that the compactness of this motor resulted in relatively high tip loading of its vanes. At maximum power, the theoretical PV value of a vane tip can be as high as 7,000,000 psi x fpm; however, wear-in was expected to reduce that maximum value to approximately 3,270,000 psi x fpm. This relatively high maximum loading level enhances the versatility of the motor as a materials test bed for its allows a wide PV operating range, a characteristic whose importance is explaned below.

Table 4. Westinghouse Design Goals

Feature	Contract Objective	Design Goal
Power (HP)	$\geq$ 3	4
Speed (RPM)	1,000 - 15,000	1,500
Overall Efficiency (%)	$\geq$ 70	70
Supply Pressure (PSI)	$\geq$ 1,000	1,500
Flow Rate (GPM)	$\leq$ 10	6.1
Submerged Weight (Lb)	$\leq$ 15	5
Volume (In.^{3})	$\leq$ 50	20
Operational Life (Hrs) (With Sparing)	$\geq$ 50	50

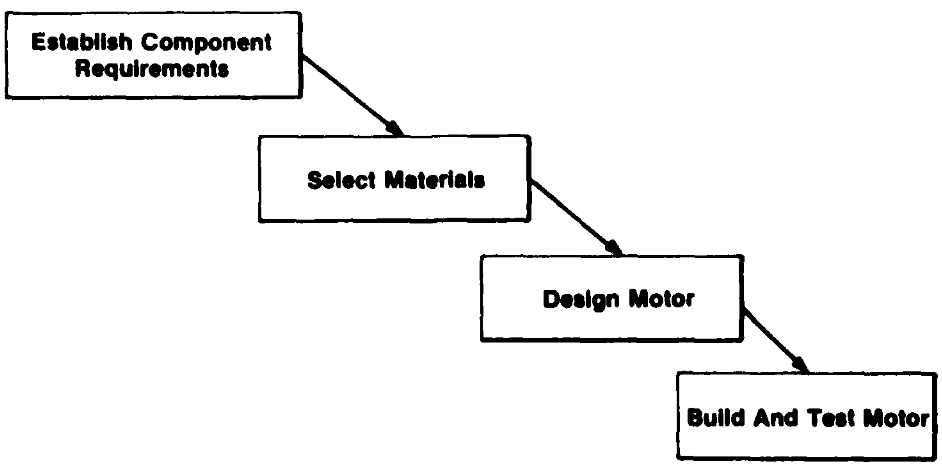

Figure 1. Development Program

MATERIALS SELECTION

The selection of materials for the motor was a two-step process. First, materials which had characteristics which seemed suitable for each of the motor components were identified and ranked in order of preference. This process is described in detail in Appendix A. Basically, though, materials data were obtained from the final report of the aforementioned materials study,[4] from materials suppliers, from seawater systems manufacturers and users, and from Westinghouse's own experience. These were compared to component material requirements which evolved during preliminary design.

Table 5 lists the more important parameters which were used to isolate the materials which would actually be tested later. In contrast to the criteria used during the earlier study, this list is relatively brief; however, it does contain those items critical to the selection of satisfactory materials.

Notably missing from this list is wear rate. Analysis had shown that few materials would be rejected due to unsatisfactory wear rate for a motor which had a life expectancy of 50 hours. Thus, wear rate was excluded from the selection criteria so attention could be focused on chacteristics more critical to the success of this motor.

Material availability was considered as important to the success of this project as the listed technical items. Indeed, as indicated later, some otherwise desirable materials, such as Stellite 6B, were eliminated from consideration because they could not be obtained in a reasonable time.

A list of the materials which were, at this state, most preferred for the components of the motor are indicated in Table 6. Generally speaking, preference increases as one moves toward the left of this table.

The second step of materials selection was the test evaluation of candidate materials from Table 6. Here, a problem was foreseen, viz., that tests usually employed for screening materials might not be sufficiently representative of the operating conditions which later would be actually encountered in the motor. Figures 2 and 3 suggest the reason.

Table 5. Material Selection Criteria

- **High Corrosion Resistance**
- **Low Friction**
- **Availability**
- **Mechanical Strength**
- **Low Swelling**

4. Ibid

Table 6. Preferred Materials

End Pieces	Inconel 625	Titanium 6Al - 4V		
Ring Track	Inconel 625	Stellite 6B	Titanium 6Al - 4V	Inconel 718
Rotor And Shaft	Inconel 625	Titanium 6Al - 4V		
Side Plates	Torlon 4301	Metcar M-271	Vespel SP-1	Torlon 4275
Vanes	Torlon 4301	Metcar M-271	Vespel SP-1	Torlon 4275 Ryton R4 UHMW Polyethylene
Bearings	Torlon 4301	Metcar M-271	Vespel SP-1	Torlon 4275
Springs	Elgiloy	MP35N	17-7 PH	

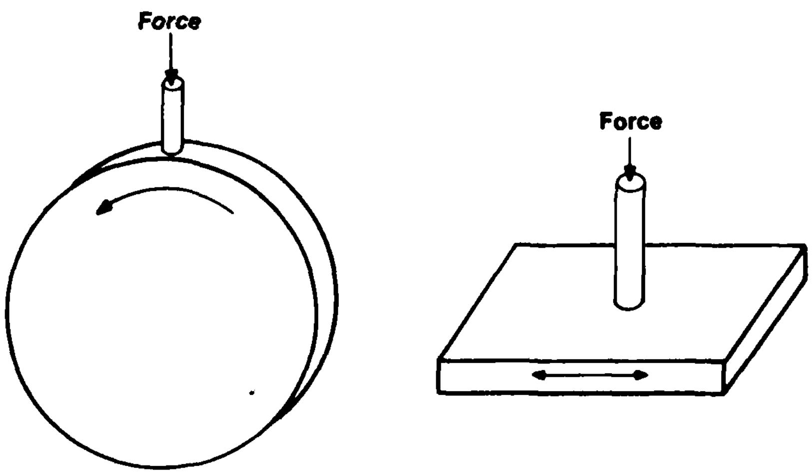

Figure 2. Typical Wear-Friction Tests

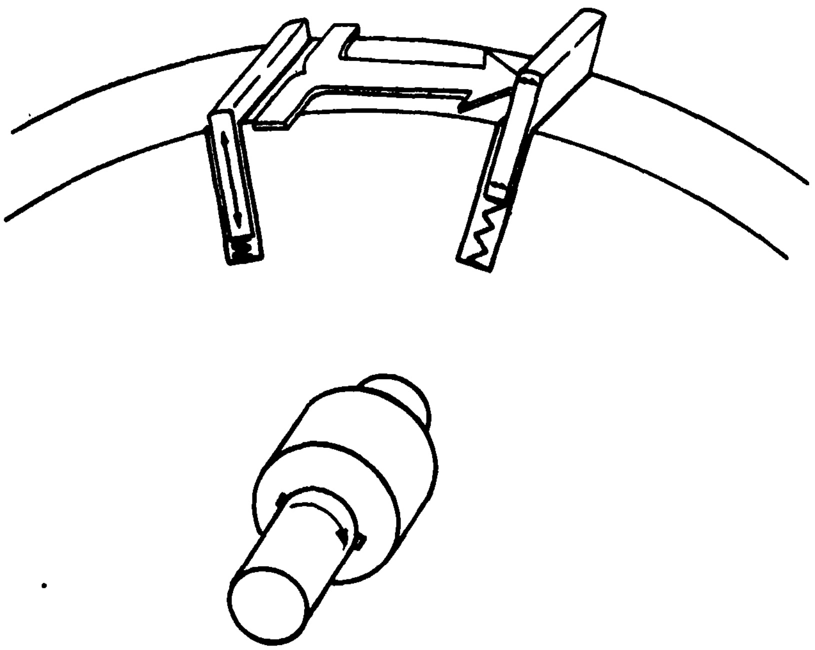

Figure 3. Complex Motion in a Motor

Wear rates and coefficients of friction are commonly measured for materials pairs using one or both of the methods depicted in Figure 2. A pin made of one of the two materials being evaluated is forced against an oscillating plate or the edge of a rotating disk, using loads and speeds appropriate to the test conditions. These are very good screening tests, but, as shown in Figure 3, they are not representative of some of the motions encountered in a hydraulic vane motor. There, one might observe that the relationship of the illustrated shaft and journal bearing is similar to that of the pin and disk of Figure 2. A similar relationship does not exist between the vane tips and the ring track (not shown) it follows, for the cam-like shape of the track causes cyclic variation in contact area on the vanes. Moreover, the vane wobbles back and forth in its slot as pressure is applied first to one side and then the other, thus superimposing a rolling motion of the tip on the track. Tests which better represented such complex motion were felt needed.

Tests which more closely represent the environmental conditions within the motor were also needed, for those of Figure 2 cannot be used to identify conditions which might cause precipitation of inorganic scales from seawater. Lastly, experience has shown that corrosion rates in seawater vary considerably with water velocity; a test which would indicate abnormally high corrosion rates under normal motor running conditions would be very desirable.

Westinghouse's common solution to all of these problems was to use the motor itself as the material test-bed. That meant that the motor design had to allow ready substitution of components made of different materials. As shown in Section 3, the motor design is well suited for this use.

MOTOR DESIGN MODIFICATIONS

During early stages of development of hydraulic vane motors, vane instability is frequently observed. This phenomenon is characterized by oscillation of the vanes against the ring track. It has various causes, such as (1) water hammer (described later) and (2) improper balance of the forces which hold the vanes against the track and those which tend to depress them. Such instability usually occurs as supply pressure is increased, and it sets the upper limit at which a motor so affected can be driven. Thus, vane instability can seriously limit the torque and power which a motor can produce.

Vane instability can be readily detected, for it produced a loud, low frequency knocking sound which closely resembles that of a jackhammer. Its effect can be devastating to the components of a vane motor.

Vane instability was encountered during development of this motor, and Westinghouse used a variety of means to control it. V-shaped notches were cut into the leading and trailing edges of water supply and exhaust ports to reduce water acceleration into and out of the power chamber between the vanes and thereby control water hammer. Both the leading and trailing edges of the ports supplying high pressure water beneath the vanes were extended to increase the rotational angle through which hydraulic pressure is used to force the vanes outward. By altering pressure distribution between the sideplates and rotor, such actions also increased water leakage away from the base of the vanes. Consequently, they produced only moderate improvements in motor performance, and sometimes they were actually counterproductive.

The most productive design alterations were the shift from single to double springs for each vane, shimming the vane springs, widening the clearance between the vanes and their slots, and increasing sideplate stiffness. The first two alterations increased and better balanced the outward-pushing spring force on the vanes. The widened clearance provided an alternate high pressure water path to the base of the vanes to increase vane force without adversely affecting leakage. Increased sideplate stiffness, which resulted from an increase in thickness, helped better control leakage between the rotor and the two sideplates. As shown in Figure 4, such changes resulted in progressively improved motor performance.

As indicated earlier, this motor also served as the test bed for material evaluation. A description of the developed motor is the subject of the next section of this report, and detailed drawings of it are contained in Appendix D.

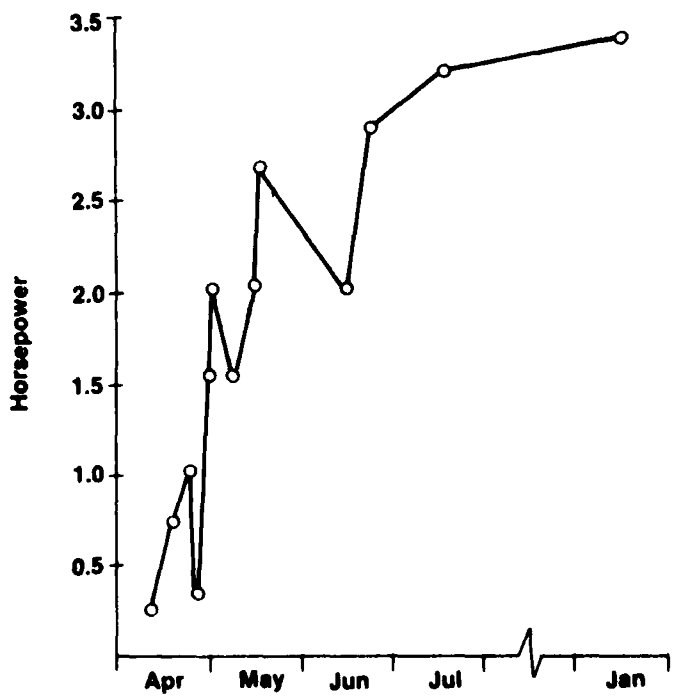

Figure 4. Power Goal Achieved

SECTION 3

MOTOR DESIGN DESCRIPTION

GENERAL DESCRPITION

The seawater motor is shown assembled in Figure 5 and partially disassembled in Figure 6. Figure 7 shows critical parts after the motor was disassembled to replace springs during its 50-hour test.

The motor was the result of an extensive development program, the sequence of events for which is shown by Figure 8. The sequence shows an iterative design process with testing to determine the merit of each design modification, as described earlier. The preliminary design calculations and related design considerations are contained in Appendices B and C.

The seawater motor is a hydraulic vane-type with pressure balanced rotor and sideplates. It is lubricated only with the solid surfaces of its components and the filtered seawater used to power it. Physical and operational characteristics of the model are given for two test conditions in Table 7.

Features of the developed model are listed in Table 8. The first three features are requirements specified by CEL. Several of the features are unique design concepts, and others are interim design concepts such as the four supply ports and four return ports. In a future model, there will no doubt be compact internal manifolding with one supply port and dirt-excluding return ports venting directly to the ambient.

OPERATION

The motor operates in the same general manner as an oil hydraulic vane motor. The fluid (seawater) entering the supply ports flows through internal porting to apply high pressure across the vanes. The vanes follow and seal against a ring track with a major and minor diameter. Pressure drop across vanes on the major diameter produces torque on the rotor and output shaft. The rotor rotates as high pressure fluid forces the vanes circumferentially away from the high pressure port and toward the low pressure port. The high pressure flow enters the space between the rotor and ring track at the ramps where the vanes follow the ring track from the minor to major diameter. The low pressure flow exits the space between the rotor and ring track at the ramps where the vanes follow the ring track from the major to minor diameter. The low pressure fluid then flows through internal porting and out the return ports. The ramp shape is an approximation of a cycloidal shape tangent to the minor and major diameter surfaces of the ring track.

VANE, SIDEPLATE, AND BEARING DETAILS

Among the most critical components in the seawater motor are the vanes. The vanes must slide against the ring track with high velocities and high tip stresses in seawater, a fluid with poor lubricity. The vanes are also subjected to high side stresses in bending and compression when they have a high pressure drop across them and are sliding on the major diameter of the ring track. They must operate with reasonably low wear, friction, leakage, and chance of brittle failure. In addition, the vanes must not be forced away from the ring track in spite of high pressure forces at the vane tip and some leakage near the base of the vane between the rotor and sideplates.

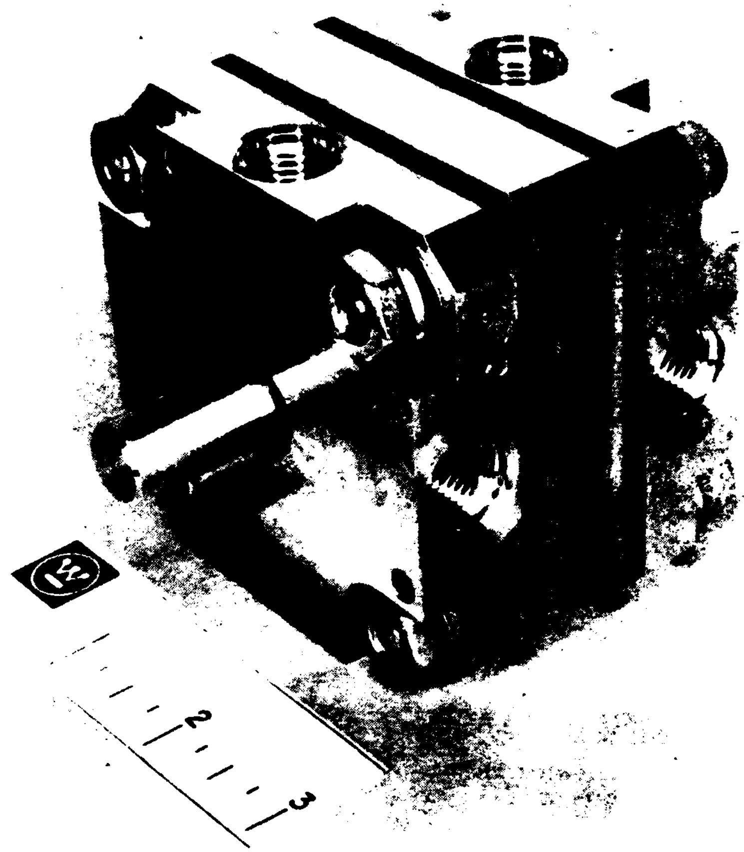

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Figure 5. Phase I Developed Engineering Model

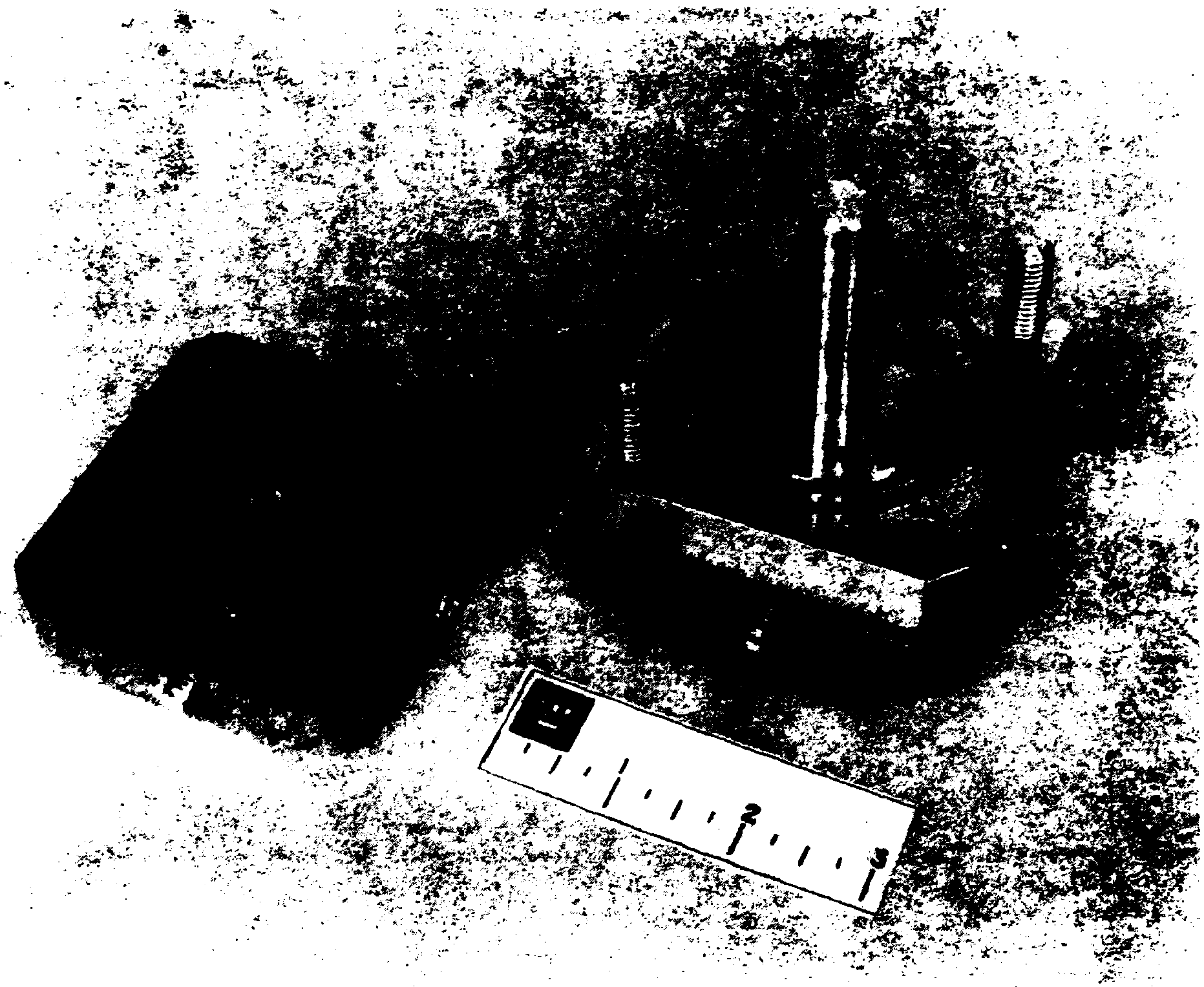

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Figure 6. Developed Model, Partially Disassembled

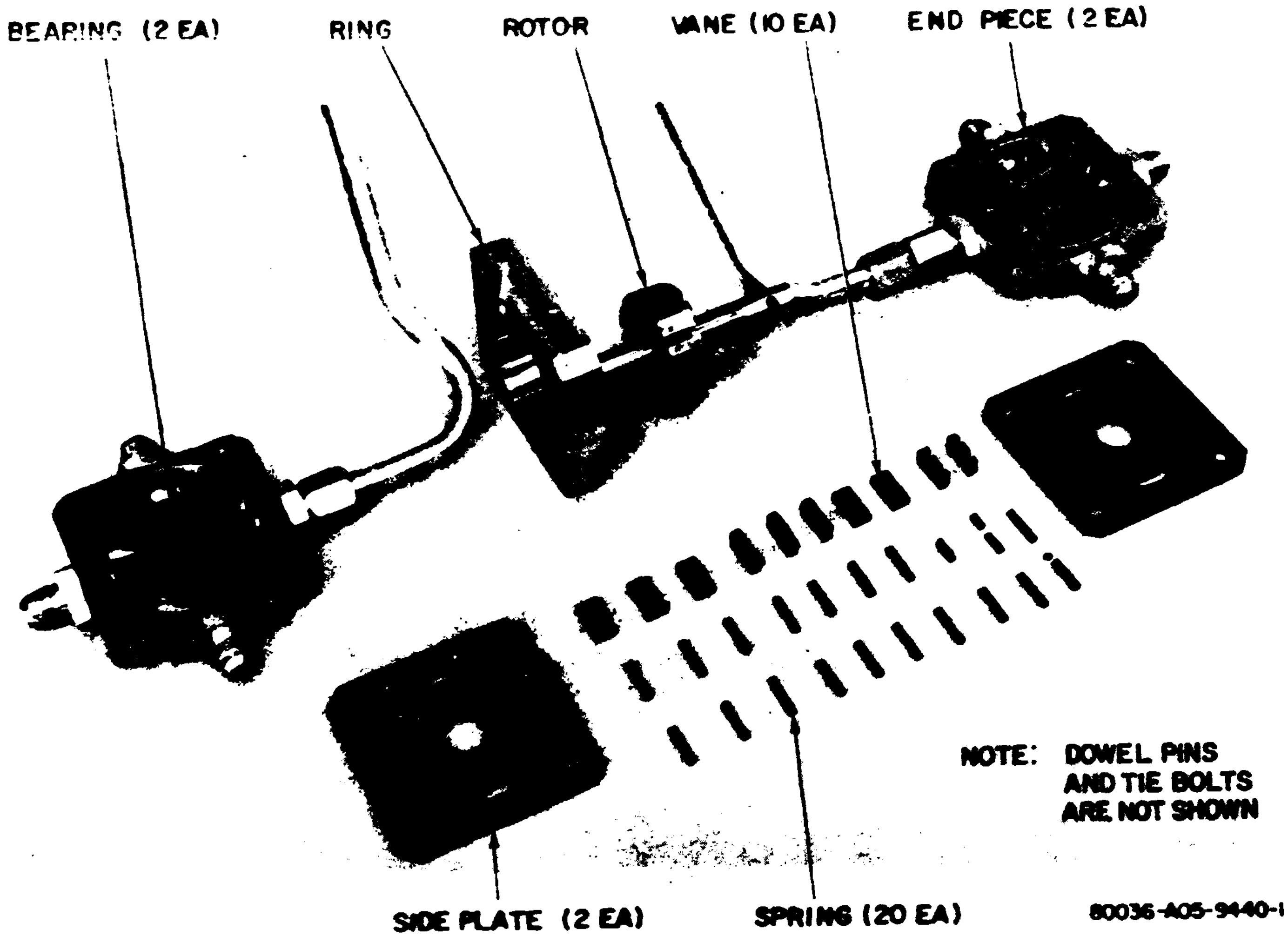
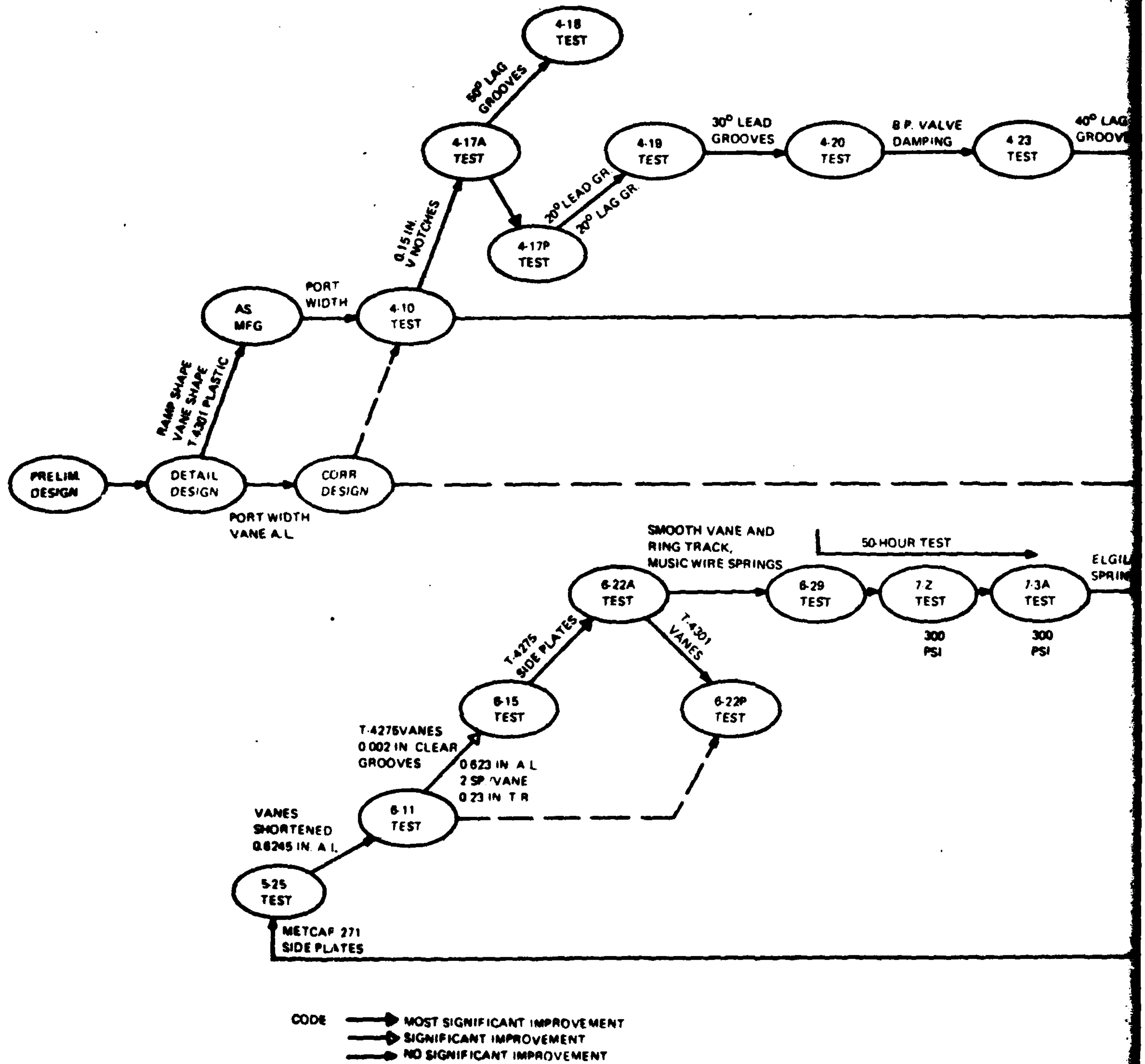


Figure 7. Developed Model Disassembled During Its 50-Hour Test



PRELIM DESIGN
DETAIL DESIGN
CORR DESIGN
PORT WIDTH
VANE A.L.
RAMP SHAPE
VANE SHAPE
T-4301 PLASTIC
AS MFG
PORT WIDTH
4-10 TEST
0.15 IN. V NOTCHES
4-17A TEST
50° LAG GROOVES
4-18 TEST
4-17P TEST
20° LEAD GR.
20° LAG GR.
4-19 TEST
30° LEAD GROOVES
4-20 TEST
B.P. VALVE DAMPING
4-23 TEST
40° LAG GROOVE
METCAF 271 SIDE PLATES
5-25 TEST
VANES SHORTENED 0.6245 IN. A.L.
6-11 TEST
T-4275VANES 0.002 IN. CLEAR GROOVES
0.623 IN. A.L.
2 SP./VANE
0.23 IN. T.R.
6-15 TEST
T-4275 SIDE PLATES
6-22A TEST
T-4301 VANES
6-22P TEST
SMOOTH VANE AND RING TRACK, MUSIC WIRE SPRINGS
6-29 TEST
50-HOUR TEST
7-2 TEST
300 PSI
7-3A TEST
300 PSI
ELGIL SPRIN
CODE
MOST SIGNIFICANT IMPROVEMENT
SIGNIFICANT IMPROVEMENT
NO SIGNIFICANT IMPROVEMENT

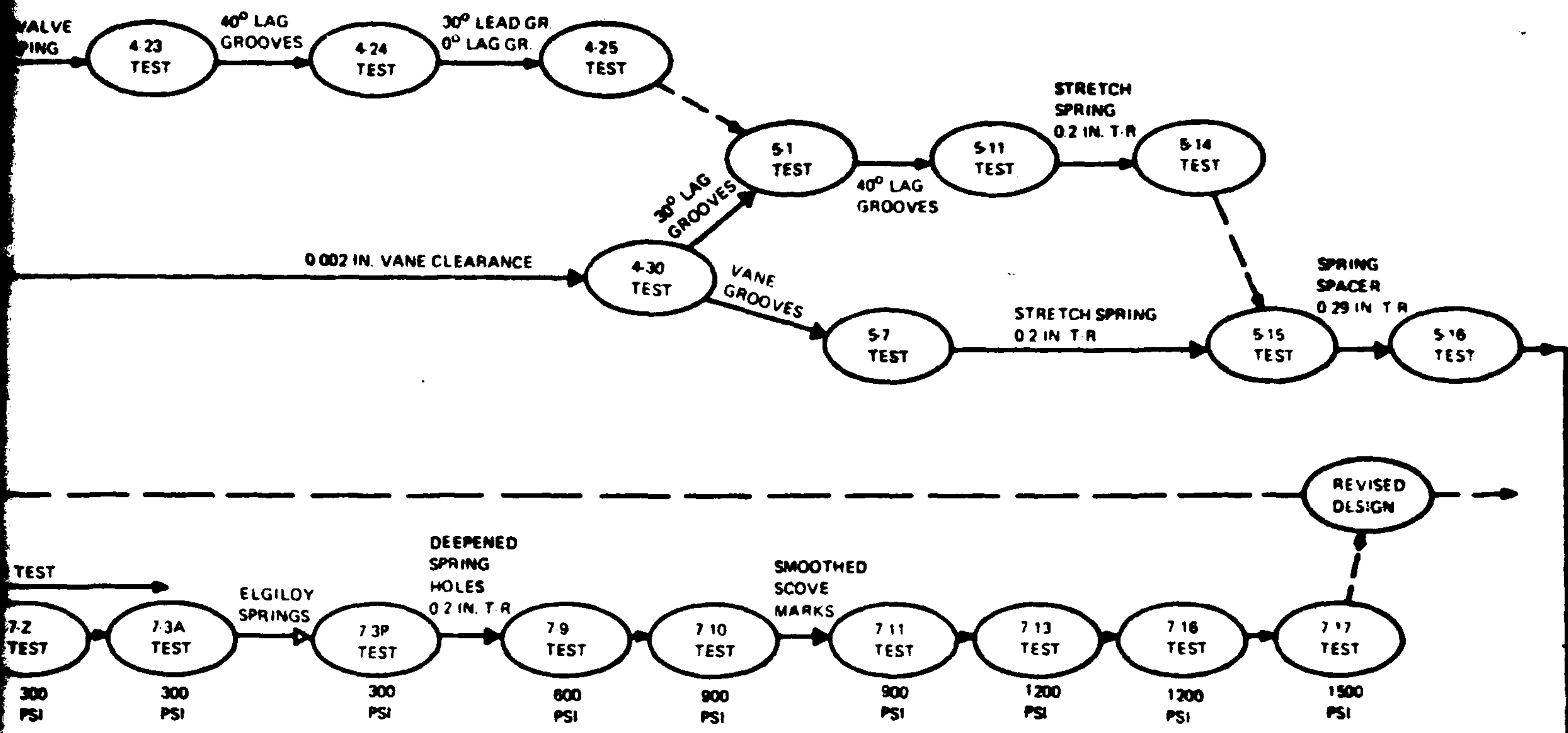


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Figure 8. Model Development Sequence With Significant Changes and Improvements

Table 7. Physical and Operational Characteristics

PARAMETER	VALUES		COMMENTS
Power Output (hp)	3.2	2.8	Slightly exceeds the goal of 3 hp at 1500 psi. (Should achieve higher power at higher flow rates.)
Speed, (rpm)	1150	1240	Meets the goal range at 1000 to 15,000 rpm.
Overall Operational Efficiency (%)	65	69	Nearly meets the goal of 70% at 3 hp. (Should exceed 70% efficiency at higher flow rates.)
Operational Pressure (psi) at Full Power	1500	1200	Exceeds the minimum goal of 1000 psi.
Maximum Flow Rate (gpm)	5.65	5.85	Exceeds the goal of flow rates less than 7-10 gpm.
In-Water Weight (lb)	*9 w/ manifolds 5 w/o manifolds		Meets the goal of weight less than 5 to 15 lb. (Should decrease to 5 lb with compact manifolds.)
Volume (in.^{3})	*45 w/ manifolds 22 w/o manifolds		Meets the goal of volume less than 10 to 50 in.^{3}. (Should decrease to 25 in.^{3} *with compact manifolds.)*
Maximum Length (in.)	3.5	3.5	Meets the goal of length less than 3 to 6 in. (Should increase to 4 in. with compact manifolds.)
Maximum Width (in.)	*19 w/ manifolds 3 w/o manifolds		Does not meet the goal of width less than 2 to 4 in. (Should decrease to 3 in. with compact manifold.)
Operational Life Expectancy (Hours)	50	50	Does not meet the goal of minimum life of 50 hours. (May exceed 50 hours with further development.)

* The manifolds are large test manifolds and should not be considered part of the motor.

80036T06

Table 8. Features of Phase I Developed Model

- Vane type hydraulic motor - to minimize wear effects on performance.
- No lubricants other than filtered seawater and solid surfaces of components; to reduce hazards.
- Pressure balanced rotor - to minimize bearing load, friction, and wear.
- Vanes spring loaded and pressure loaded against the ring track - to allow operation over a wide range of pressures.
- Pressure balanced, flexible bearing material, end plates - to minimize leakage and damage from contamination, misalignment, and expansion. Also to minimize friction torque loss and act as a limited thrust bearing for motor shaft axial loads.
- Grooves in end plates and sleeve bearings directing filtered seawater past bearing surfaces - to cool, help lubricate, and carry contamination particles away from the surfaces.
- Dirt seal on shaft - to exclude dirt from the motor when it is not running.
- Four inlet and four outlet ports - for easier modifications during engineering model development.
- Small size per displacement ratio - to minimize size and weight.
- Design based on properties of the best recently available bearing material combinations, as indicated by tests at WEC and MTI - to maximize performance and life.

80036T07

To minimize wear at the tip of the vane, the tip shape has a symmetrical radius as large as the maximum slope of the ramps would permit. Also the material for the vane was selected based on its high strength and low modulus of elasticity, friction coefficient, and wear rate. The material, Torlon 4275, was proven in recent tests to be the best material for bearings operating in seawater at high PV values. The low modulus of elasticity along with large tip radius minimizes the Hertz stress at the tip. Wear-in also helps reduce Hertz stress by making the vane tip conform to the ring track.

There is a tendency for the vanes to be forced away from the ring track by high fluid pressures acting over part of the vane tip and by low pressure at the base of the vane caused by leakage away from the base between the rotor and sideplates. To reduce this tendency, a vane design was conceived with a high pressure leakage path from near the vane tip to its base. The leakage path results from clearance grooves and slots on the side of the vane (see Figure 9). The leaning of the vane also reduces the pressure force on the tip of the vane because the high pressure acts over a smaller area at the tip.

Although flexible sideplates have been used before with vane pumps, the flexible sideplates in this motor are novel in that they have critical pressure balance areas and flexibility range to allow them to act as thrust bearings as well as practical dynamic seals.

Radial grooves in the sideplates (see Figure 6) act as seawater flow paths to help lubricate, cool, and carry contamination particles away from the sideplates. Axial grooves are located in the shaft sleeve bearings to help lubricate, cool, and carry particles away from the sleeve bearings. Lubricating seawater in the flow paths originates at the inner return ports in the sideplates, flows radially inward through the grooves in the sideplates, flows axially outward through the grooves in the shaft sleeve bearings, then flows out of the motor.

MATERIALS

Critical components and their preferred materials are listed below.

Components	Material
Vanes (10 each)	Torlon 4275
Springs (20 each)	Elgiloy
Rotor	Inconel 625
Ring	Inconel 625
Side Plates (2 each)	Torlon 4275
End Pieces (2 each)	Inconel 625
Bearings (2 each)	Torlon 4301 (Torlon 4275 Preferred)
Dowel Pins (2 each)	Case Hardened Steel (Inconel 718 Preferred)

These materials were selected for their high resistance to corrosion, reasonable probability of long wear, low coefficient of friction, and availability. A detailed discussion of the material selection process is included in Appendix A.

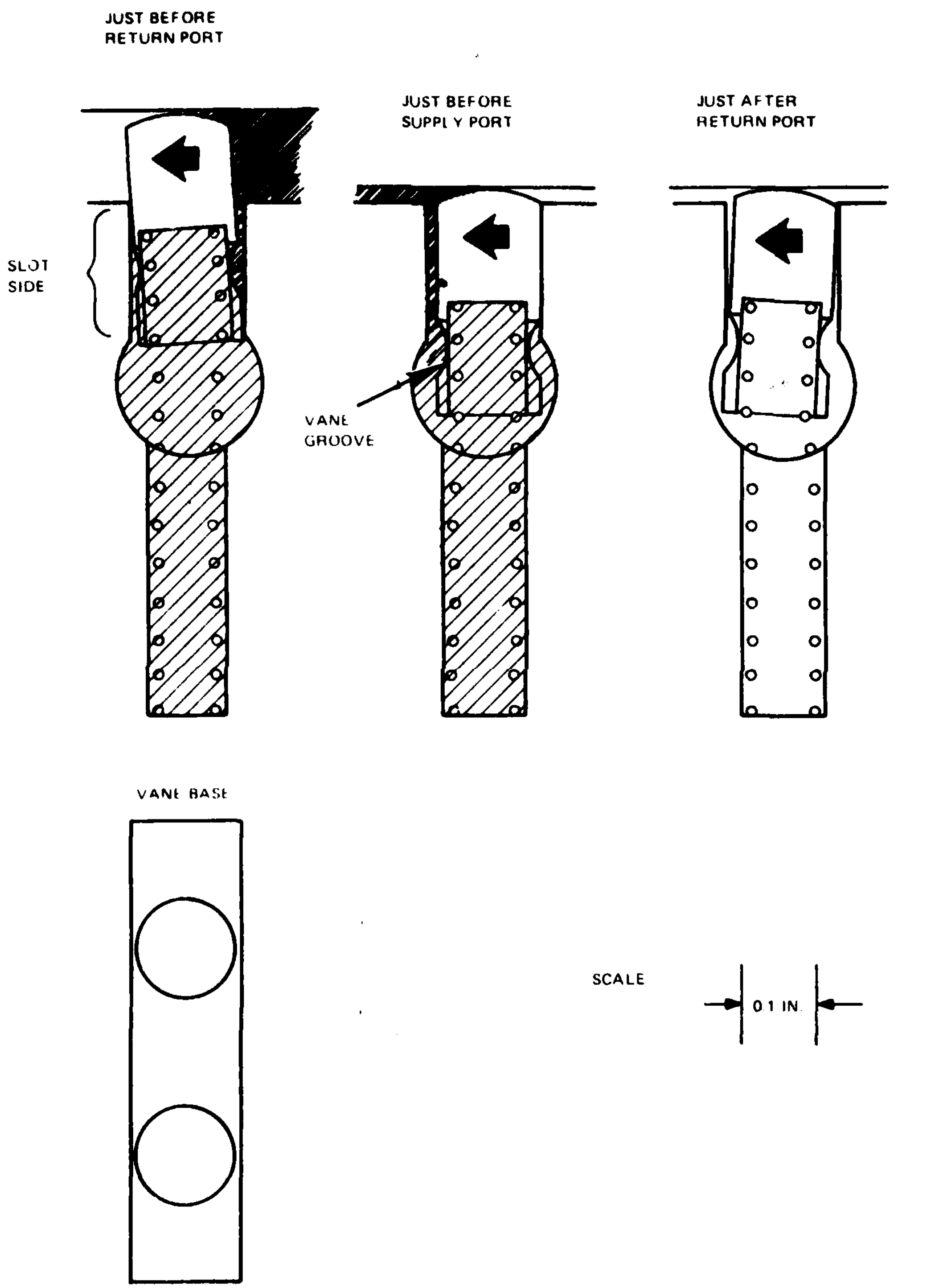

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Figure 9. Present Vane and Slot Configuration

During our preliminary material selection, delivery time of Torlon 4275 in sheet form for sideplates and vanes was estimated to be too long for use in this contract. As a result, in the early stages we selected the more available alternate, Torlon 4301, as a candidate material. WEC had successfully used Torlon 4275 in rod form for turbine and pump sleeve bearings in another development effort, the hydraulic heater. Compared to Torlon 4301, tests at Amoco Chemicals Corporation and WEC indicated Torlon 4275 used against Inconel 625 in seawater would have:

- a lower coefficient of friction
- a lower wear rate
- a higher operating PV value
- a slightly lower strength, and
- a slightly greater modulus of elasticity.

Midway in the development/test program, Torlon 4275 sheet became available and the change from Torlon 4301 to 4275 was made for vanes and sideplates. The result was lower wear rate of the vane tips and sides and higher output torques for the same operating pressure, indicating less friction at the vane tips. End plates of 4275 resulted in a lower wear rate and less leakage and a wider operating range of pressure and flow. Part of the reason for these increases was that the Torlon 4275 end plates were intentionally made about twice as stiff by increasing plate thickness from 0.150 to 0.180 inch and relying on the slightly higher modulus of elasticity. Slight wear patterns on the sideplates and good motor performance over a wide range of operating pressures and flows indicate sideplate stiffness is about optimum.

SECTION 4

TEST PROGRAM AND RESULTS

As indicated earlier, the motor itself was designed to be used as the materials test bed. It was mounted for this purpose in a test system which is described in detail in Appendix E. With this test setup, water under controllable pressure and flow rate was pumped from a large (approximately 800-gallon) tank to the test motor via a water supply manifold. Water exhausted from the motor was returned directly to the tank via a short gap open to the atmosphere.

The test motor was directly coupled to an ordinary oil hydraulic vane pump which served as the load for the motor. This pump was mounted in a gimballed stand which allowed the output torque of the motor to be monitored electrically. The magnitude of the load presented by the pump was varied by throttling the pump output. Motor speed was monitored with a strobe light.

MATERIALS EVALUATION

The intent of the materials test program was to evaluate the materials which were identified in Table 6 to have highest potential for satisfactory performance in this motor. The combination which produced the best motor performance (i.e., highest output power) would then be subjected to a 50-hour test to assess life expectancy.

Figure E-1 (see Appendix E) indicates the motor configurations which were planned for evaluation. The planned sequence was from top to bottom in that table, the most promising material combinations being first listed. Unavailability of several listed materials and schedule constraints led to curtailment of some of these tests. Nonetheless, the necessary materials evaluation tests were successfully completed.

During the selection tests, the material combinations listed in Table 9 were actually evaluated. The sequence in which these tests were performed was shown earlier, in Figure 8. Pertinent results are shown in Table 10, where the data are related to the Figure 8 tests via the common test date. In all, over twenty hours of actual motor running time were used to conduct these tests, and they clearly showed the material combination of the next to last column of Table 9 to be superior. This combination was selected for a 50-hour life test.

50-HOUR OPERATING TEST

The 50-hour test was conducted in a manner similar to those just described. The principal differences were that synthetic seawater, rather than fresh water, was used as the hydraulic fluid and detailed measurements of motor components were made at 5-hour intervals. The test was run only during the normal working day, with no special attempt being made to purge the motor of seawater during quiescent periods. In fact, the motor was intentionally left filled with seawater over one nonoperating weekend during this test to encourage corrosion.

Table 9. Materials Evaluated

Material Combination Selected For Life Test (marked column: 4th)

End Pieces	**Inconel 625**	**Inconel 625**	**Inconel 625**	**Inconel 625**	**Inconel 625**
Ring Track	**Inconel 625**	**Inconel 625**	**Inconel 625**	**Inconel 625**	**Inconel 625**
Rotor And Shaft	**Inconel 625**	**Inconel 625**	**Inconel 625**	**Inconel 625**	**Inconel 625**
Side Plates	**Torlon 4301**	**Metcar 271**	**Metcar 271**	**Torlon 4275**	**Torlon 4275**
Vanes	**Torlon 4301**	**Torlon 4301**	**Torlon 4275**	**Torlon 4275**	**Torlon 4275**
Bearings	**Torlon 4301**	**Torlon 4301**	**Torlon 4301**	**Torlon 4301**	**Torlon 4301**
Springs	**Elgiloy**	**Elgiloy**	**Elgiloy**	**Elgiloy**	**17-7PH**

Table 10. Material Evaluation Test Results

TEST DATE	SUPPLY PRESSURE (psi)	SUPPLY FLOW RATE (gpm)	MOTOR SPEED (rpm)	MOTOR TORQUE (in./lb)	OUTPUT* POWER (hp)
4/18	470	6.2	1200	39	0.74
4/24	515	6.0	1180	56	1.05
4/25	135	5.7	1500	12	0.29
4/30	740	6.2	1480	66	1.55
5/1	780	6.2	1360	93	2.01
5/7	560	6.2	1420	61	1.37
5/11	1100	6.2	800	128	1.62
5/14	1100	6.2	1000	112	1.78
5/15	900	6.2	1320	98	2.05
5/16	1200	6.2	1260	134	2.68
5/25	570	6.0	1540	46	1.12
6/11	600	6.0	1450	53	1.22
6/15	1300	6.0	1055	122	2.04
6/22A	1500	5.7	1170	156	2.90
6/29	1200	5.85	1200	141	2.68
7/16	1200	5.85	1240	144	2.83
7/17	1500	5.65	1150	176	3.21

*Output powers are maximum hp for test date.

Component Wear (50-Hour Test)

The 50-hour test was designed to determine component failures and wear at operating pressure of 300, 600, 900, 1200, and 1500 psi, and at an operating flow of about 6 gpm. Component failures included those of springs and one vane. Appreciable component wear was noticed only on vanes and sideplates, and some surface roughness noted on the ring track and rotor ends.

Wear and spring break patterns on critical components are shown in Figure 10. A close-up view of wear and damage on a sideplate and vanes is shown in Figure 11.

Elgiloy spring failures shown in Figure 10 occurred shortly after 5 hours of testing. The nonductile type breaks normally occurred between the spring holes in the rotor and vanes, indicating that the spring is more highly stressed in this region. The springs also show wear marks where they rub against the outer side of the hole in the rotor.

Analyses and measurements indicate the cause of spring failure is cyclic fatigue in the more highly stressed region of the spring. The higher stress is attributed to friction force added to the normal compression force, plus bending stresses caused by misalignment between the spring holes in the vanes and rotor. Additional friction force in the holes is also attributed to axial flow of water from under the vanes, pushing the springs against the sides of the holes.

Vane wear is most appreciable at the tip and on one side as shown by the depression on the vane at the far right in Figure 11. After 44 hours of the 50-hour test, vane tip wear was about 0.001 inch and the vane side wear was about 0.003 inch. Most of this wear occurred in the first hour of testing and during the last 4 hours of testing at 1500 psi.

The vane break shown in Figure 11 occurred after 44 hours of testing. The break originated at an inside sharp corner of the side groove of the vane. This is a stress concentration point when the vane is being highly stressed in bending. This failure indicates the need to reduce stress concentration by rounding the inside corner of the groove. A higher strength material will not be required.

Side plate wear was a maximum of 0.0005 inch at the long shaft (drive) end of the motor and 0.0013 inch maximum at the short shaft end. The plate for the short shaft end is shown in Figure 11; it is the plate that restrains considerable axial force caused by the shaft coupling. The deep gouge damage shown between inner supply and return ports occurred about 5 hours into the 50-hour test when a piece of wire spring broke loose and wedged between the rotor and side plate on its way out of the motor. The gouge and wear resulted in a slight decrease in motor volumetric efficiency, but not enough to prevent continuation of the 50-hour test.

Surface roughness on the ring track and rotor ends appears to result from contamination caused by "knocking". Knocking occurs when outward force on the vane is insufficient to keep the vane tip in contact with the ring track. Water leakage past the opening at the vane tip is suddenly shut off with another vane moving into the leakage path. This is believed to cause water hammer when the water travels away from the vane forming a cavitation bubble, then returns to the vane, collapsing the bubble and causing a high pressure knock. The collapsing cavitation bubble causes the very high local pressures associated with cavitation damage. Cavitation damage particles of ring track material are then embedded in the vanes and side plates causing scoring of the ring track and rotor ends. Without knocking, the vanes tend to polish the ring track, even at 1500 psi operating pressures.

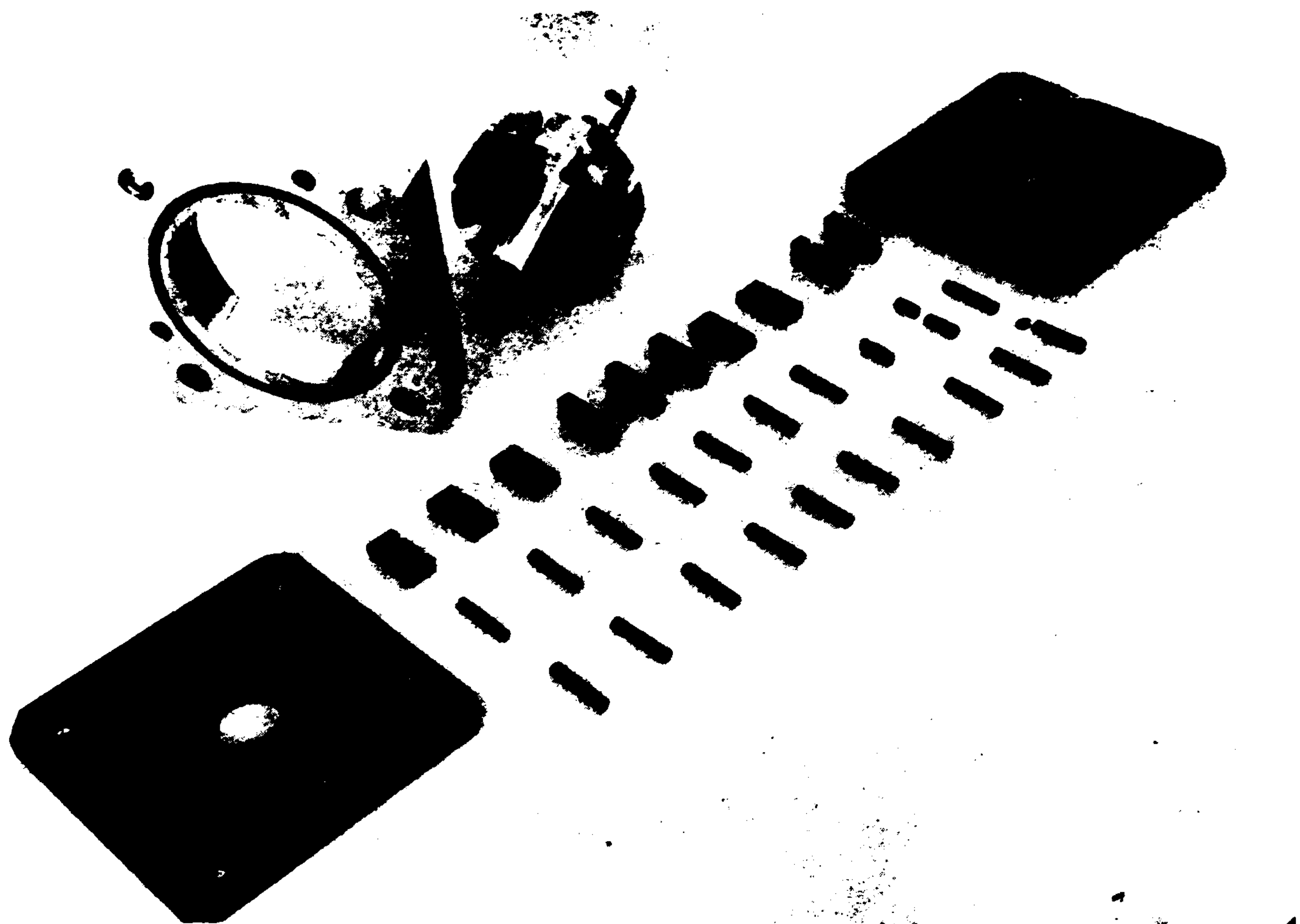

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Figure 10. Wear and Spring Break Patterns

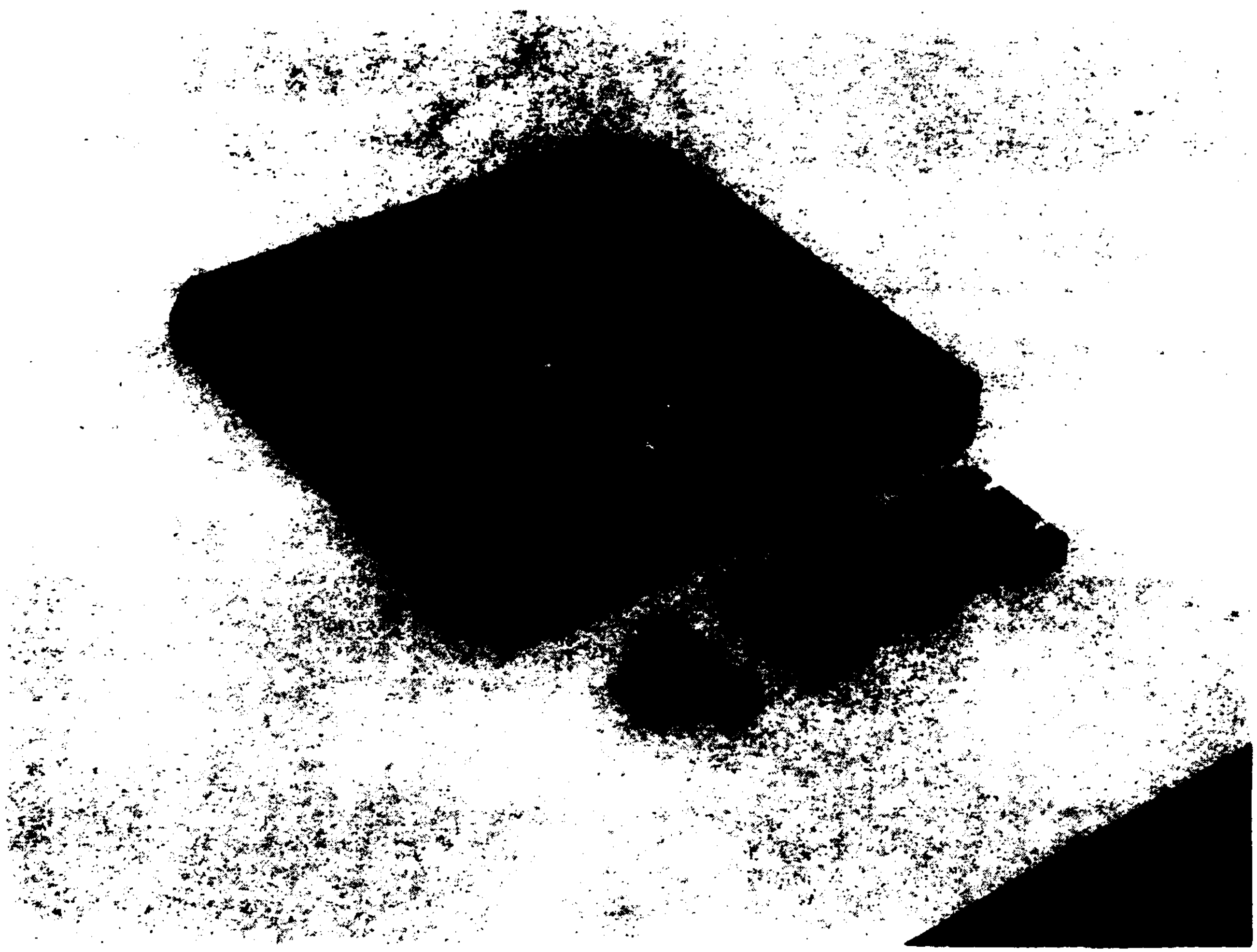

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Figure 11. Wear and Damage On Side Plate and Vanes

Motor Performance

Performance checks were made periodically during the 50-hour test. Performance is summarized by the curves of Figure 12. The sloped curves are for flow vs. motor speed and the nearly horizontal curves are for torque vs. motor speed. The perfectly straight lines are theoretical flow and torque curves for the motor with 100% volumetric efficiency and 100% mechanical efficiency respectively.

Flow curves become increasingly higher than the theoretical flow curve as pressure increases indicating that volumetric efficiency decreases with increased pressure. This is expected since leakage increases with pressure. The leakage is not greatly affected by increased shaft speed, which indicates higher flows and speeds could produce higher volumetric efficiency.

Torque curves fall under the theoretical torque curves for various pressures indicating less than 100% mechanical efficiency. The mechanical efficiency is highest at the higher pressures and at the higher flows, but tends to level off at the highest pressures and flows. This, along with the shape of the flow curves, indicates that higher power output and slightly higher overall efficiency can be expected at higher flow rates, especially if the pressure is limited to 1200 psi.

Test data during the last two days of the 50-hour test are shown in Tables 11 and 12. Table 11 is for the second part of the 1200-psi test and includes performance checks of the motor at lower pressures and flows. Table 4-12 is for the 1500-psi test and includes two performance checks at 1200 psi. Note that motor speed and torque increase slightly with time indicating that motor wear-in at one pressure and flow increases both volumetric and mechanical efficiency.

PARAMETRIC ANALYSIS

Test results showed that, upon completion of the just described 50-hour test, the motor had attained nearly all performance goals set by the contract. Although it had not quite met the goal of producing 3 shaft horsepower at 70% overall efficiency, it had come close. When supplied with 5.64 gpm of seawater at 1500 psi, it had produced 3.2 horsepower at 1150 rpm with an efficiency of 65%. Likewise, 5.85 gpm at 1200 psi had resulted in 2.8 horsepower at 1240 rpm and an efficiency of 69%. Analysis suggested that these figures could be improved by operating the motor at higher speed than the test system then existing permitted.

Continuous operation had been limited to slightly more than 5 hours by spring fatigue failure and to 45 hours by vane breakage. Yet, both of these constraints could be corrected without difficulty. Spring life could be improved by reducing spring stress. This could be accomplished by such actions as lengthening the springs and deepening the spring holes. Vane life could be improved by reducing stress concentration at the edge of the vane groove, and that required only a minor redesign of the vane, rounding the groove and displacing the end of the spring hole from it.

Such improvements were subsequently undertaken in an extension to the basic program, termed Phase II development in this report. The purpose of this extended effort was to seek improvement in output power, efficiency, and life without otherwise degrading motor characteristics.

This extension is described in detail in Appendix F. It was quite successful in producing the desired results. The motor operated 50 hours without failure, and, as indicated in Table 13, all technical goals of the contract were met. Moreover, as Figure 13 illustrates, wear rates of the vanes — the most susceptible component to wear — were well within reason.

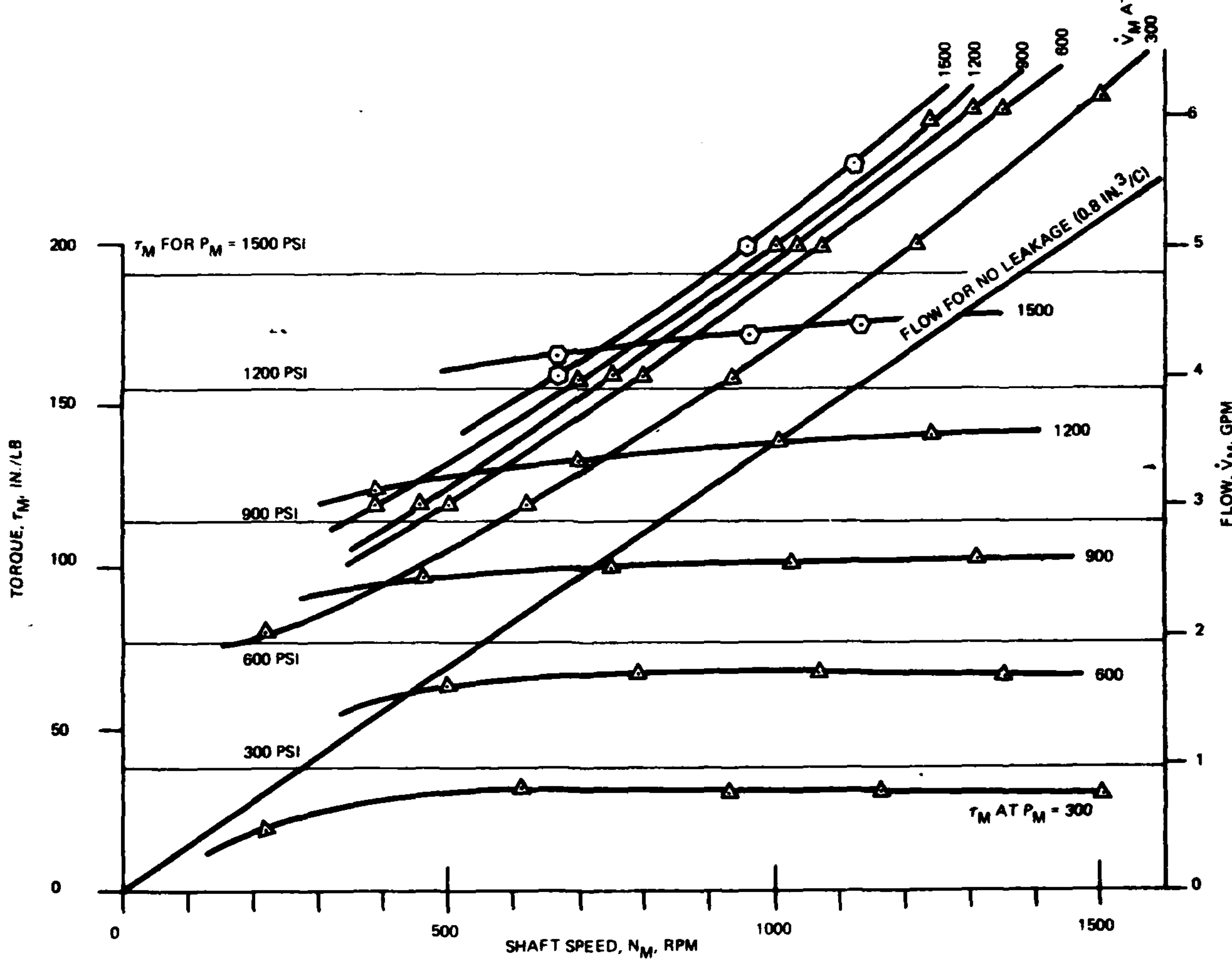

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Figure 12. Seawater Motor Test Data

Table 11. Test Data for 50-Hour Test, 1200 Psi Part 2
Test Date: 16 July 1979, Start Time: 8:17 a.m.

TIME	$\dot{V}_M$, gpm	P_M, psi	N, rpm	τ, mni	τ, in./lb	COMMENTS
8:30 AM	6.0	1200	1235	32.2	145	Water Temp. 80°F
9:25 AM	5.95	1200	1240	32.8	148	
10:15 AM	6.0	1200	1240	33.0	149	
10:52 AM	5.95	1200	1245	33.0	149	
	6.05	900	1310	25.3	112	
	6.05	600	1355	17.7	75	
	6.15	300	1500	9.8	39	
	5.15	135	1630	5.2	21	
	5.0	115	1335	4.6	18	
	5.0	300	1220	9.5	38	
	5.0	600	1075	17.4	74	
	5.0	900	1035	25.0	110	
	5.0	1200	1010	32.3	146	
	4.0	100	1065	4.2	18	
	4.0	300	935	9.5	38	
	4.0	600	795	17.5	75	
	4.0	900	755	24.7	109	
	4.0	1200	700	31.7	143	
	3.0	90	770	4.0	16	
	3.0	300	620	10.0	40	
	3.0	600	500	16.9	72	
	3.0	900	464	23.9	105	
	3.0	1200	395	29.7	133	
	2.0	90	454	3.8	15	
	2.0	300	221	5.7	27	
	2.0	≈600	–	–		Stopped
11:32 AM	5.9	1200	1220	32.9	148	Water Temp. 85°F
12:15 PM	5.85	1200	1235	33.3	150	
12:43 PM	5.85	1200	1240	33.5	152	
12:18 PM	5.85	1200	1240	33.5	152	Water Temp. 87°F

NOTE: Torque readings are 8 in./lb too high because of oil hose torque.

80036T01

Table 12. Test Data for 50-Hour Test, 1500 Psi Part 1
Test Date: 17 July 1980, Start Time: 8:15 a.m.

TIME	$\dot{V}_M$, gpm	P_M, psi	N, rpm	τ, min.	τ, in./lb	COMMENTS
8:20 AM	5.75	1500	1100	38.8	177	
8:45 AM	5.75	1500	1125	39.6	182	
9:10 AM	5.75	1500	1140	39.6	182	
	5.8	1200	1225	32.6		
	5.0	1200	1000	32.3		
	5.0	1500	935	39.1	180	
	4.0	1500	635	37.7	173	
	3.3	1500	–	–		Stopped
9:25 AM	5.6	1500	1095	39.8	183	Water Temp. 86°F
9:45 AM	5.65	1500	1120	39.8	183	
10:15 AM	5.65	1500	1130	40.0	184	
10:40 AM	5.65	1500	1130	40.0	184	Water Temp. 88°F
	3.0	1500	960	39.3	181	
	4.0	1500	670	38.0	174	
	5.65	1500	1130	39.3	181	
11:24 AM	5.6	1500	1140	40.0	184	
11:45 AM	5.65	1500	1145	40.0	184	Water Temp 90°F
12:15 PM	5.65	1500	1150	40.0	184	
12:20 PM	–	1500	–	–	–	Stopped Suddenly

NOTE: Torque readings are 8 in./lb too high because of oil hose torque.

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Table 13. Motor Characteristics Achieved

Feature	Contract Objective	Design Goal	Actual Performance
Power (HP)	≥3	4	3.3
Speed (RPM)	1,000 - 15,000	1,500	1,585
Overall Efficiency (%)	≥70	70	80
Supply Pressure (PSI)	≥1,000	1,500	1,000
Flow Rate (GPM)	≤10	6.1	7
Submerged Weight (Lb)	≤15	5	5
Volume (In.^{3})	≤50	20	23
Operational Life (Hrs) (With Sparing)	≥50	50	50

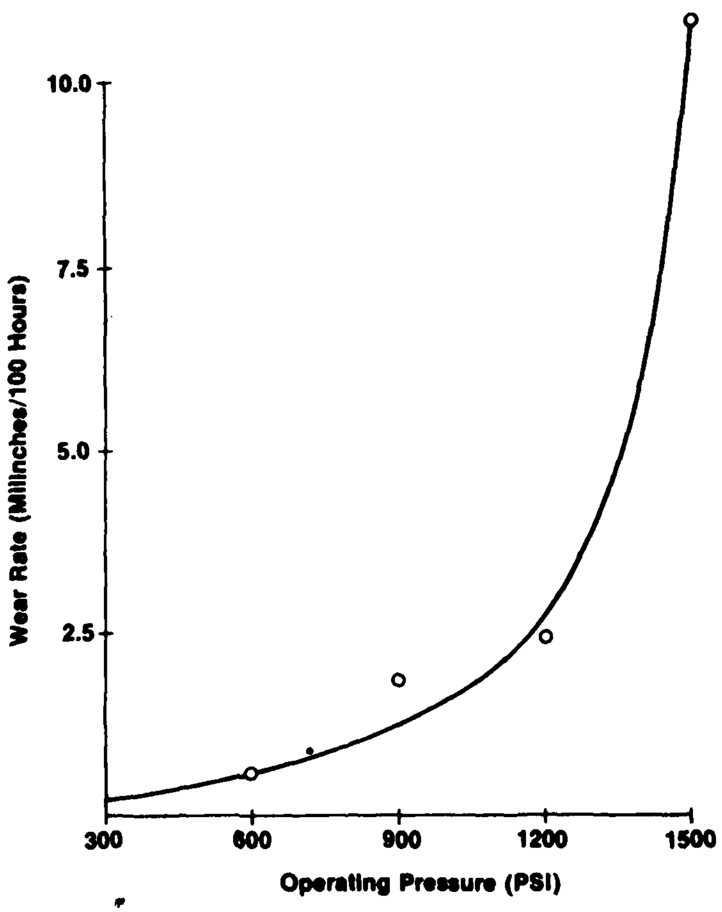


Figure 13. Acceptable Wear Rates When Operating Below 1200 Psi

SECTION 5

CONCLUSIONS AND RECOMMENDATIONS

The primary purpose of this project was to prove the technical feasibility of a seawater hydraulic vane motor through laboratory demonstration of a working motor. To this end, it was an unqualified success. The results show unequivocally that materials needed for such a motor are available and that the design technology required is at hand.

The motor which evolved during this project is compact and light enough, even at this early stage of development, to power a diver-held tool. Moreover, it can rival its more traditional oil-powered counterpart in output power, speed, and overall efficiency. It has shown no observable signs of corrosion, and wear is well within manageable limits.

Consequently, **Westinghouse strongly recommends that further development of this motor for use in a diver-held tool be undertaken.**

We have noted that, although it fulfilled all contract obligations, life expectancy is the principal limitation of this motor. Its components have not failed by wearing out; instead they have failed catastrophically by fatigue. Dramatic improvement in the parts so affected is mandatory if a reasonable improvement in motor life is to be obtained.

Success to the contrary, the reader should note that the scope of this project was intentionally restricted. Of the many types of hydraulic motors known today, only a single type — a double entry vane motor — was investigated. We believe that the success achieved with this motor suggests that similar success might be achieved with other types. We therefore suggest that **investigation intended to convert other types of hydraulic motors (and pumps, as well as hydraulic actuators) to use seawater as the working fluid is technically warranted.**

The scope of the materials evaluation was also intentionally limited during this project. The reason for this was that a comprehensive materials evaluation program could not have been undertaken without substantially increasing the required cost and schedule. The results of this project have confirmed that the additional effort was unnecessary, even though a better combination of materials for the motor might have been discovered. Nonetheless, a number of promising material candidates, such as those listed in Table 14, were identified both in this project and the one which preceded it[5]. We recommend that **investigation of such materials for use in seawater hydraulic systems should be actively pursued.** The absence of such information can only inhibit the rapid development of seawater hydraulic power systems.

Table 14. Materials With Potential

• **Chromium Plating**	• **Graphites (Metallic Filled)**
• **Stellite 6B**	• **Ceramics**
• **T1 - 6 Al - 4V (Hardened)**	• **Ultra High Molecular Weight Polyethylene**
• **MP35N**	• **Vespel SP-1**
• **Inconel 718**	• **Riton R4**

5. Ibid

APPENDIX A

Preliminary Materials Selection

APPENDIX A

PRELIMINARY MATERIALS SELECTION FOR THE SEAWATER MOTOR

A.1 Introduction and Background

The critical materials selection problems for the seawater motor are those concerned with the wear surface combinations: vane/ring track, side plate/rotor, and the sleeve/shaft. The other areas of the motor will be exposed to less demanding conditions so that materials requirements are less restrictive.

An important fundamental decision on materials selection was made early in the program which establishes the context of the subsequent discussion. It was decided that the wear surface materials combinations would comprise a metallic material vs. a relatively soft nonmetallic material. This type of combination is compatible with the broad conclusions of the MTI Report and has been adopted in many demanding wear surface mechanisms developed by Westinghouse for seawater applications. Specifically excluded were:

a. **Metal-to-Metal Combinations.** Westinghouse experience and the MTI Report indicated that these combinations would be unacceptable because of high wear rate and high coefficient of friction.

b. **Combinations Involving Hard Ceramics.** These materials are possible candidates for the seawater motor but were considered unsuitable for inclusion in the short time frame of this program. Design with these materials is more complicated than with other materials and the possibility of catastrophic failure (because of the hard and brittle character of the ceramics) appeared greater. Machining of ceramic parts to meet the schedule also appeared to be a problem.

The background considerations which were used in selecting actual materials and combinations for the critical location comprised:

- Data contained in the MTI Report
- Materials availability
- WEC experience with marine materials
- Discussion with pump manufacturers and operators of seawater handling systems

With regard to the latter source, an especially relevant program has recently been carried out at NSRDC (Annapolis). In that program a vane pump has been modified to handle seawater and seawater-oil mixtures. Although the PV conditions for the NSRDC pump are somewhat lower than for the prime candidates in the present program, the conditions are sufficiently similar so that the results represent a valuable source of information for our current program. Relevant details of the NSRDC pump are described in Section A.6.

Before proceeding with a detailed description of the candidate materials combinations, some comments on general requirements of both metallic and nonmetallic materials are made.

A.2 General Comments, Metals

Metallic materials were considered for all components except the vanes, side plates, and bearing sleeves. These materials must have both high corrosion resistance in seawater and good friction and wear characteristics. In the critical parts of the motor, high corrosion resistance must be maintained under conditions of high relative water motion and abrasive contact. The material must also have good resistance to crevice corrosion and pitting. This last requirement is necessary since the motor will have to survive storage periods between uses, during which time some residual seawater will remain trapped in its interior. In addition to high corrosion resistance, the material for wear surfaces such as the vane track and the bearing must also have good friction and wear characteristics when used in combination with other candidate materials.

When the above requirements are considered along with the constraint of availability, a fairly small number of candidate materials remain: Inconel 625, Inconel 718, certain other high nickel alloys, and titanium alloys. After also considering the data from the MTI report it appeared that Inconel 625 alloy would be the best choice as the prime metallic material. The MTI results showed that the friction and wear performance of this material, when used in combination with several nonmetallics, was slightly better than the other metallic materials.

Backup metallic materials were felt to be desirable for the ring track and were selected to be harder and more wear-resistant than the Inconel 625 alloy, since there was some concern that abrasive particulate matter would inevitably be present in the motor (in spite of filtration). The prime backup material was Stellite 156, applied as a weld-overlay to an Inconel 625 ring. This decision was made because of the favorable Westinghouse experience with Stellite in various seawater mechanisms as well as the experience of NSRDC in the seawater pump program. MP-35N alloy was another possible backup for the ring track; it would have a hardness comparable to Stellite but the fabrication would be simpler since overlaying would be unnecessary. Neither Stellite or MP-35N could be adopted as test candidates because of lack of availability within schedule constraints.

Titanium alloys are also regarded as possible materials of construction and may become very desirable at a later stage in motor development when weight reduction becomes important. For the ring-track it would be necessary to apply a hard or galling-resistant finish to its wear surface. Several coating, conversion-coating, and anodize type finishes have been developed and are commercially available.

Elgiloy was selected for the vane springs. This material is virtually immune to seawater corrosion, has high fatigue strength, and is used extensively for high performance springs which must function in seawater.

A.3 General Comments, Non-Metals

Nonmetallic materials were considered as candidate materials for the vanes and bearing inserts.

Chief properties which were considered in their selection were: strength (particularly for the vanes), modulus, thermal expansion coefficient, and compatibility with seawater. Strength is important in determining maximum operating pressure, and modulus is important since it is used in calculating anticipated PV conditions and distortions. Thermal expansion data and seawater absorption are also important since they indicate susceptibility to dimensional change during operation and the possibility of seizing up.

In the case of the vane material, it was anticipated that considerable wear would be experienced. On the other hand it was decided early in the program to accept relatively high wear rates on these components since they would be inexpensive and easily replaceable. Comparative wear rate data was, therefore, assigned a relatively low priority in the selection of vane material.

The important characteristics and properties of the candidate materials are summarized in Table A-1. The selection procedure could not be made on entirely quantitative information since the complete set of desired data has not been determined for many of the materials. Furthermore, much of the test data on individual materials was obtained by test procedures which differed between various manufacturers. For this reason, the selection was influenced by data from applications in which material has been successfully used in seawater.

A.4 Candidate Materials Combinations

Candidate material combinations were selected by two rationales: (a) by reference to the comparative evaluations contained in the MTI Report, and (b) from successful application experience with seawater mechanisms using materials which appeared to be applicable to the seawater motor.

Individual metallic and nonmetallic materials were, of course, subject to the requirements outlined in Sections A.2 and A.3 for use in seawater. For each combination the PV conditions for the vane tips were calculated as described in Section C. The calculations showed that even the best two candidate combinations yielded initial PV values of 7×10^6 and 23×10^6 psi ft/min at the vane tips. Such high values would be regarded as totally unacceptable in most applications. On the other hand, the cooling effect of the ambient water was considered as possibly allowing operation at unusually high PV conditions. Furthermore, the ability to accept high wear rates on the vanes suggested that very high PV conditions might be acceptable in the seawater motor. This viewpoint was supported to some extent by a study of the Seawater Vane Pump Program at NSRDC. The PV conditions had not been derived by NSRDC but sufficient information was obtained by Westinghouse to show that this pump operates at an initial PV of 5×10^6 psi/ft/min.

The candidate material combinations selected are shown in Table A-2. In each case the same material is used for each of the nonmetallic parts. The vane/ring track is, of course, the most demanding situation and the selection of the nonmetallic was made from consideration of this area. Schedule permitted testing of only two baseline combinations: Inconel 625/Torlon 4301 (Torlon 4301 replaced with Torlon 4275 during testing) and Inconel 625/Metcar 271 (Metcar side plates only. Vanes of Torlon 4301). Materials were also procured for three alternate configurations which may be regarded as Candidates 3, 4, and 5.

The prime candidate combination (Inconel 625/Torlon 4301 or 4275) was selected because of its good comparative rating in the MTI test series and because it operates at the lower initial PV of 7×10^6 psi ft/min. The second candidate combination (Inconel 625/Metcar 271) was selected primarily because of the favorable results in the NSRDC seawater pump program with Metcar 271. This combination has the negative feature of a very high PV (23×10^6 psi ft/min). Also, the graphite base Metcar 271 may have the ability to induce galvanic corrosion in the metallic materials, although it is believed that Inconel 625 would be immune to this type of corrosion. To some extent this possibility exists also for the prime candidate combination since Torlon 4301 contains 12% graphite and Torlon 4275 contains 20% graphite. Also, since Metcar 271 is harder than Torlon 4301 it may cause more ring wear.

Table A-1. Data Summary, Nonmetallic Candidate Materials

MATERIAL	DESCRIPTION	STRENGTH	ELASTIC MODULUS (10^6 PSI)	THERM. EXP. COEFF. ($10^{-6}/^{o}F$)	SEAWATER COMPATIBILITY	RELEVANT APPLICATION OR TEST DATA
Torkon 4301	Poly (amide-imide), 12% graphite, 3% PTFE	19 Ksi (tension)	*	15	Approx. 1% linear expansion after 1 month immersion	Preferred material as indicated by MTI tests.
Vespel SP-1	Polyimide	12 Ksi (tension)	0.5	30	0.5% linear expansion at $73^{o}F$ and 100% humidity	Satisfactory seawater use by WEC in heavily loaded sliding contacts.
Metcar 271	Carbon-graphite impregnated with copper-lead	9 Ksi (tension) 43 Ksi (compression)	4.2	3.8	*	Satisfactory performance in NSRDC seawater vane pump.
Ryton R-4	Polyphenylene sulfide, 40% glass fiber	21 Ksi (compression)	2.1 – 2.4	11	No change in dimensions after 168 hr. in water	Previously used as vane material in NSRDC pump. A second priority material in MTI study.
Hercules 1900	Ultra-high molecular weight polyethylene	6.3 Ksi (tension)	*	100	0.13% weight increase after 30 days in seawater	Several satisfactory seawater applications on USN ships. UHMW composite material a first priority material in MTI study.
Torlon 4275	Poly (amide-imide) 20% graphite, 3% PTFE	18 Ksi	1.24	13	Same as for Torlon 4301	Good seawater bearing performance in WEC equipment. Preferred over Torlon 4301 if it is available in time for test.

*Data not available.

Table A-2. Test Configurations

CATEGORY	CONFIGURATION	RING	ROTOR AND SHAFT	END PIECES	VANE	SIDE PLATES	BEARING
Configurations to be tested	Baseline 1	In625	In625	In625	T-4301	T-4301	T-4301
	Baseline 2	In625	In625	In625	M-271	M-271	M-271
Configurations not presently planned for testing	Alternate 1	In625(H)	In625	In625	M-271	M-271	M-271
	Alternate 2	In625	In625	In625	Ryton	Ryton	Ryton
Material on Hand	Alternate 3	In625	In625	In625	Vespel	Vespel	Vespel
Configurations not presently planned for testing	Alternate 4	MP35N	MP35N	In625	Best of above materials by testing		
Material Not on Hand	Alternate 5	Ti(H)	Ti(H)	Ti	M-271	M-271	M-271

NOTES: 1. For Baseline 1, T-4301 to be replaced by T-4275 if material is available within schedule constraints.

2. In625 = Inconel 625
   T-4301 = Torlon 4301
   T-4275 = Torlon 4275
   M-271 = Metcar 271 Bronze Graphite from Metallized Carbon Corp.
   Ryton = Ryton R4 Plastic Extrusion
   Vespel = Vespel SP1 Isostatically pressed
   In625 (H)= Inconel 625 with surface hardened by Stellite weld overlay
   MP35N = MP35N
   Ti = Titanium 6Al-4V alloy
   Ti (H) = Titanium 6Al-4V with surface hardended by conversion coat

3. Alternate 1 will become Baseline 2 if hard surfacing process can be implemented without schedule delay. The hardening is recommended to preclude chance for the hard Metcar 271 to score the Inconel 625 surfaces.

The following comments pertain to the three alternate combinations:

a. Alternate 1 is the same as the second choice candidate combination except that the Inconel 625 ring track surface is overlayed with Stellite 156. This combination duplicates more exactly the conditions in the NSRDC Pump and is expected to offer superior performance if wear of the Inconel 625 ring track is significant.

b. Ryton R-4 (in Alternate 2) gave adequate performance in the NSRDC Pump provided no particulate matter was present. This type of plastic was also one of the better performers in the MTI evaluation (against several metals).

c. Vespel SP-1 has been used by WEC with satisfactory results in several highly loaded sliding mechanisms operating in seawater.

A.5 Materials for Future Consideration

While selecting candidate materials for the seawater motor, information has also been obtained on other materials which for reasons of availability or lack of complete data could not be identified as firm candidates in this contract, but which merit consideration for future extensions of this work. The more attractive of these materials are identified below.

A.5.1 Metallic Materials

Chromium-plated surfaces have attractive combinations of friction and wear characteristics and are also low cost and available. The chromium itself is also highly corrosion-resistant. However, many organizations, including WEC, have found chromium plating to be unreliable in seawater. Failures have occurred because of corrosion of the substrate material at discontinuities in the plated layer. The fallacy in these previous applications has been the assumption that chromium plating provides corrosion protection to the substrate material. It has, therefore, been applied to materials such as carbon steels, austenitic, and high-strength stainless steels. Although some stainless steels are resistant to corrosion when all of their exposed surfaces are in contact with aerated seawater, they will (crevice) corrode at sites such as are provided at pores and fissures in plated layers. Thus, it appears that chromium plating, when intended for seawater applications, should be applied only to the most highly corrosion-resistant materials. Examples would be Inconel 625 and titanium alloys, materials which are highly resistant to crevice corrosion. On the other hand, these materials are not easily electroplated because the highly stable passive oxide films on their surfaces interfere with adhesion of the plating. Recently, specialized procedures have been developed for chromium plating these alloys which overcome this difficulty. Applications are mainly in the aerospace industry. WEC has recently initiated an R&D program to investigate the behavior of these combinations in seawater. If they are shown to be satisfactory they would appear to be logical candidates for use on the ring track of the seawater motor.

Other methods for improving the surface friction and wear characteristics of titanium alloys should also be investigated. Titanium has relatively poor tribological properties so that considerable effort has been applied to developing special surface finishes with improved galling resistance and frictional properties. Several conversion-coating and anodize type finishes have been developed and are commercially available. Some of these have also been evaluated for seawater applications. These finishes are also candidates for the ring-track surface, and might permit more extensive use of titanium along with simplicity of manufacture.

A.5.2 Expendable Type Nonmetallic Materials

There is much scope for further experimentation with nonmetallic materials, given the many proprietary variants of individual materials and the many types of composites. Two types of material which seem to be especially worthy of further investigation are discussed below.

Ultra-high molecular weight polyethylene has been found to have excellent frictional properties in seawater and has been selected for use in critical sliding contact components on U.S. Navy ships. We note also the good static and kinetic friction results shown for this type material (in combination with metals) in the MTI Report. The leading commercial variant (Hercules H-1900) has such low strength (2000 psi) that it was thought unsuitable as a candidate for the vanes of the seawater motor. The strength of this material could be improved by incorporating a strong reinforcing material. Such composite materials are not yet commercially available but will probably be in the future. A small special development program could be initiated to produce vanes for the seawater motor.

The encouraging results obtained with the bronze graphite composite suggest another direction for additional experimentation. There are many types of such composite available, made by different manufacturing processes and incorporating different metals. Some additional development work to select the optimum material is an area for useful future activity.

A.5.3 Ceramics

Ceramics may be excellent candidates for certain components of the seawater motor, notably for the ring-track wear surface. Although not considered in the current program for reasons outlined earlier, it would be worthwhile to investigate their use in a longer time frame program.

A.6 Seawater Vane Pump Under Development at NSRDC (Annapolis)

This pump is being developed by the Pollution Control Branch at NSRDC. The program is based on modification of an existing pump which has previously been used for fuel oil transfer and is supplied by Blackmer Pump Division of Dover Corporation. Pump characteristics are: 50-100 gpm, 60 psi, 270 rpm, rotor diameter approximately 7-1/4 in. diameter. Vane dimensions are 4-3/8 in. x 1-5/8 in. x 7/16 in. Seawater is not filtered. Original materials for the critical vane/ring track combination were: melamine asbestos vanes and a chromium plated track. Problems occurred with this pump in less than 100 hours of operation, caused by the entrained particulate matter. Some improvement was effected by changing to Ryton R-4 vanes and a Stellite ring track. Later, the vanes were changed to Metcar 271 (which run against the Stellite track). This combination has run for 200 hours without problems.

APPENDIX B

Preliminary Design

APPENDIX B

PRELIMINARY DESIGN

DESIGN OVERVIEW

The Preliminary Design for the seawater motor was a result of:

- Concept tradeoffs of vane motor variations to establish a conceptual design with a high probability of success consistent with the broad specification requirements of the contract.
- Preliminary analyses and design iterations to minimize major problems.
- Selection of one set of materials based on the preliminary material selection and loads identified in the preliminary analyses.

The conceptual design was based on judgment gained from a strong background in hydromechanical equipment, as well as knowledge of commercial vane motor and pump designs. The goal was to define a motor within the CEL requirements and specifications that would have the highest chance of success. Nominal operating speed and pressure were kept to the low end of the specification to minimize wear. Size was kept small to provide a motor close to the ultimate design — a small lightweight motor easily handled by a diver. The CEL specifications and the Westinghouse design goals are summarized in Table B-1. These goals are conservative in that we can reduce pressure to 1000 psi and allow flow and speed to increase to 7.3 gpm and 1800 rpm respectively to reach the minimum specified power of 3 hp.

Having established the conceptual design for the motor, analyses of performance, stress, strain, and wear related parameters were made for the motor and its critical components. Design refinements and material iterations resulting from this analysis led to the Preliminary Design shown by Figure B-1. Design features of the motor Preliminary Design are presented in Table B-2.

This preliminary design led to the engineering test model designs discussed in Section 3 of this report.

DESIGN ANALYSES

Sketches and analyses for the preliminary design are included later in this Appendix. The critical components analyzed include the rotor, shaft, vanes, vane springs, side plates, and bearings. As with many complex devices in the research and development stage, certain areas of the design have not been analyzed because of the high complexity of the analysis required. These areas include:

- accurate force values of vanes, side plates and bearings on their rubbing surfaces.
- Wear rates resulting from rubbing surfaces of vanes, side plates, and bearings.

Final design as affected by these areas is best addressed by trial and error testing, analyzing, modifying and retesting.

Table B-1. Physical and Operational Characteristics for a Vane Type Seawater Hydraulic Tool Motor

PARAMETER	CEL SPECIFICATIONS (DESIRED) VALUE	COMMENTS	WEC DESIGN GOAL VALUE
Power Output (hp)	3 minimum	adaptable to higher power development	3.8
Speed - Optional (rpm)	1,000-15,000	selection depends on developmental difficulty	1500
Overall Operational Efficiency (%)	70	high to maintain overall system efficiencies	70
Absolute Minimum Operational Pressure (psi) at Full Power	1,000	higher desirable to minimize system flow rates	1500
Maximum Flow Rate (gpm)	7-10	low values desirable to minimize hose diameter and maintain high transmission line efficiencies	6.2
In-Water Weight (lb)	5-15	without using buoyancy materials	5*
Volume (cu. in.)	10-50	smaller desirable for ease of diver handling	20
Maximum Length (in.)	3-6	smallest desirable for ease of diver handling	3.4
Maximum Width (in.)	2-4	smallest desirable to prevent restriction of diver visual field	3.0
Minimum Operational Life Expectancy (hr)	50		50

*Weight would be about 2.5 lb for model with Titanium 6 AL-4V in place of Inconel 625.

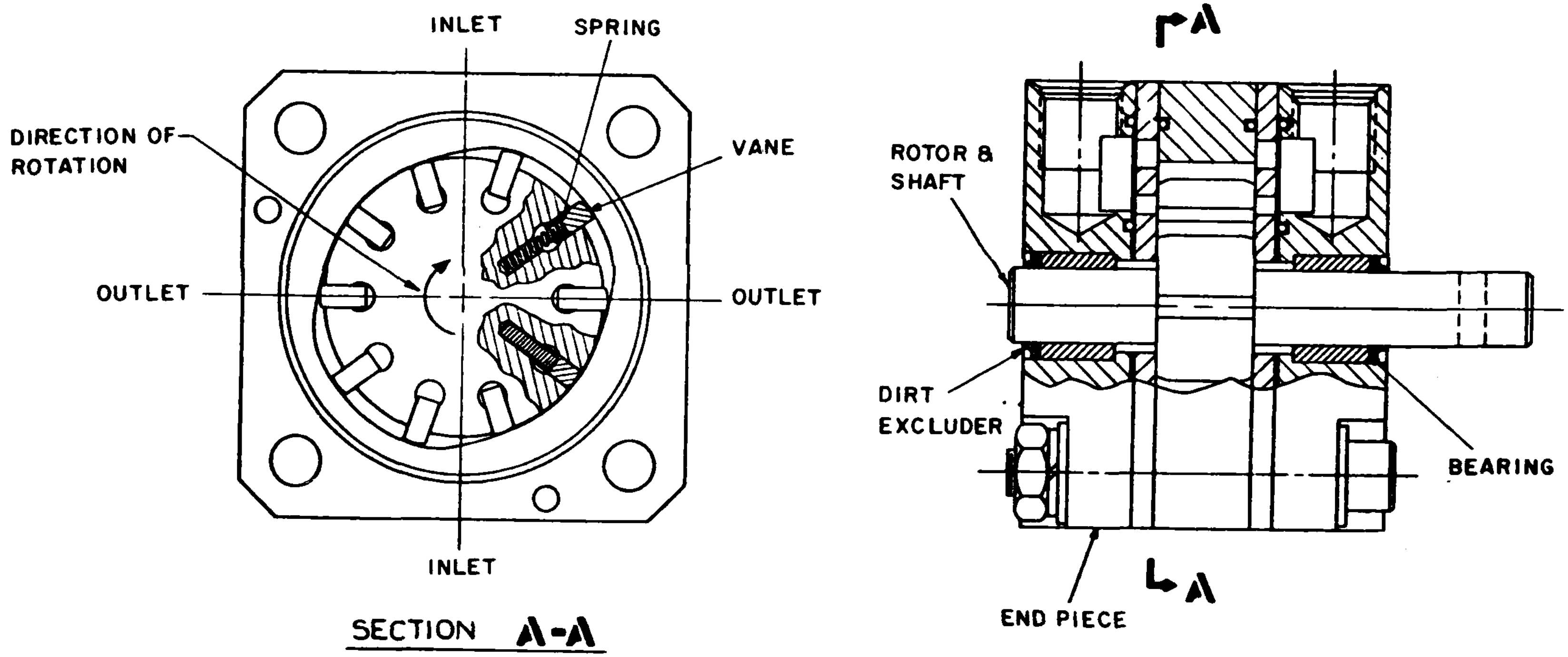


Figure B-1. Motor Assembly

Table B-2. Design Features Preliminary Design Seawater Hydraulic Motor

- Pressure balanced rotor to minimize bearing load.
- Pressure and spring loaded vanes to minimize wear and leakage.
- Elgiloy vane springs for high fatigue strength and corrosion resistance.
- Pressure balanced flexible side plates to minimize leakage and damage from contamination, misalignment and expansion.
- Grooved side plates and bearings to minimize damage from contamination and to allow adequate cooling flow for bearing surfaces.
- Dirt seals to partially exclude external dirt.
- Four inlet and four outlet ports for ease of machining and modification.
- Design features similar to those in presently used oil hydraulic motors, for high probability of success.
- Materials similar to those with the least wear in recent wear test in seawater, for high probability of success.
- Small size per displacement ratio as a basis to minimize size and weight in future models.
- Design goal physical and operational characteristics are within desired ranges. (See Table B-1.)
- Design permits testing over wide PV loading range.

Accurate force values of vanes, side plates, and bearings are difficult to analyze quickly because of the complex, transient pressure and flow fields around the rotor.

Rates resulting from rubbing surfaces of vanes, side plates, and bearings depend greatly on widely varying stresses and PV values. Vane tip PV values, for example, were calculated to be 7.0×10^6 ft/min based on initial instantaneous Hertz stress. Less conservative calculations based on average stress over the tip result in only 0.6×10^6 psi ft/min. These PV values cannot be directly compared to MTI's steady PV value of 3.3×10^6 psi ft/min that resulted in low wear rates for the same materials.

POTENTIAL PROBLEM AREAS

Potential problem areas identified in the Preliminary Design are listed below. The potential problem areas will be closely watched during the test with further analysis to be performed on a case by case basis.

1. Uneven wear occurring on the vanes, side plates, rotor or ring could cause pressure unbalance of the rotor. An oscillating pressure unbalance would occur which might cause cavitation between the shaft bearing surfaces, thereby reducing bearing life. The bearings are designed to withstand steady loads greater than the maximum estimated unbalance load, 590 lb.

2. Uneven wear or improperly sized V-grooves in the side plates could cause excessively high loads or inward loads on the vanes. The high loads would result in excessive vane tip wear, and inward loads could cause vane tip lift-off and high impact loads resulting in high leakage and high tip wear.

3. Uneven wear or improperly sized V-grooves or O-ring pads could cause high unbalance forces on the side plates. High unbalance force on the side plates toward the rotor could prevent the rotor from turning and high unbalance force away from the rotor could cause high leakage. The O-ring pads were designed so the mean force between the side plates and rotor is moderately positive when small V-grooves are cut into the side plates on the rotor side. This force, composed of pressure forces as well as O-ring squeeze forces, varies as regions between vanes change pressure.

4. Too much flexibility of side plates could cause excessive side plate wear in the region where pressure changes between vanes. The wear is caused by the tendency of the side plates to flex inward and outward in this region. Excessive wear would cause increased leakage.

5. The rigid joint between the rotor and shaft will result in wobble of both portions of the rotor-shaft due to the nonparallel axes of the portions. The wobble is not likely to be appreciable, but should be considered as a possible cause of wear and leakage.

6. Bearing wear from contamination caused by vane wear particles is unlikely because water used for bearing lube water is exposed only to the radially inward part of the vane. Wear of this part of the vane should be small compared to wear of the tip of the vane.

7. Vane wear at the tip of the vane is expected because of initially high PV values, 7.0×10^6 psi ft min. Wear flats on the vane tips should occur where maximum stresses occur, thereby helping to reduce the PV value to close to the 3.3×10^6 psi ft/min, a value that resulted in low wear in MTI tests. Also wear tendency is minimized because wear location changes on the tip and ring surface is polished to 4 μ inch rms.

8. Spring fatigue is possible because the maximum spring stress of 94,300 psi is close to the fatigue strength, 100,000 psi at 10^7 cycles for Elgiloy strip in reverse bending. The spring is in torsion and not reverse bending which might result in a fatigue strength less than 94,300 psi.

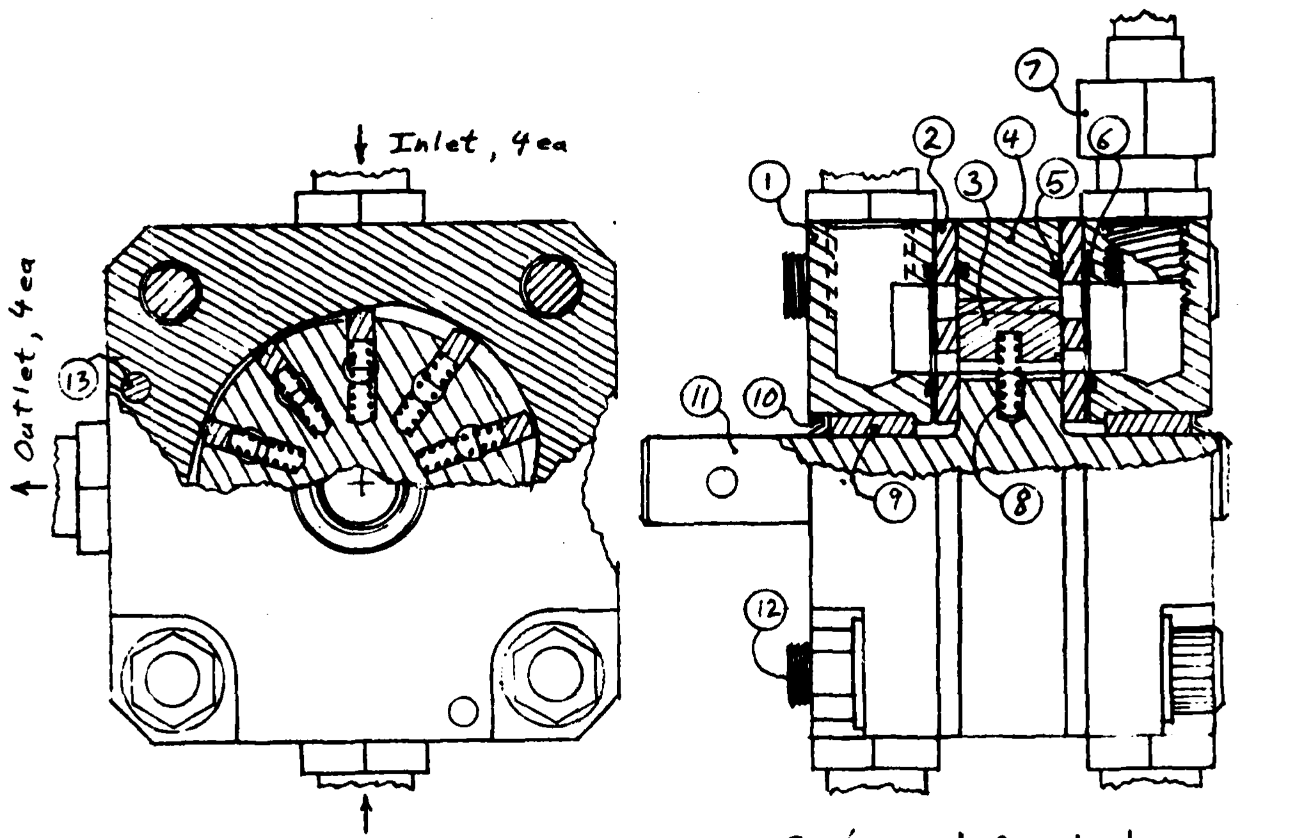
9. Fluid flow restriction and inertia between the rotor and the ring track is marginally high. Possible effects of this should be considered during analyses of test results.

10. The length of the large V-groove should be analyzed based on leakage occurring and considered in detail design changes of test models.

11. A method of polishing the vane groove and ring track surface to 4 μ inch rms is yet to be devised.

PRELIMINARY DESIGN SKETCHES AND PARTS SPECIFICATIONS

(See following pages.)



Scale: 1 inch

Parts List

1. End Piece, 2ea
2. Side Plate, 2ea
3. Vane, 10ea
4. Ring
5. Large O-Ring, Viton A, 70 durometer, 2½" OD, 1/16" W, 2ea
6. Small O-Ring, Viton A, 70 durometer, 7/8" OD, 1/16" W, 8ea
7. Male Connector, CPV H859-6-6-SS with O-Rings, 8ea
8. Spring, 10 ea
9. Bearing Sleeve, 2ea
10. Dirt Seal, Shamban No. 25E 00500A46 without O-Ring, 2ea
11. Rotor with Shaft
12. Tie Bolt with Nut and Washers, Inconel, 5/16" D, 2¼" L, 4ea
13. Dowel Pin, 3/16" D, 2" L, 2ea

Seawater Motor Assembly, Prelim. Design
WEC JC-1B

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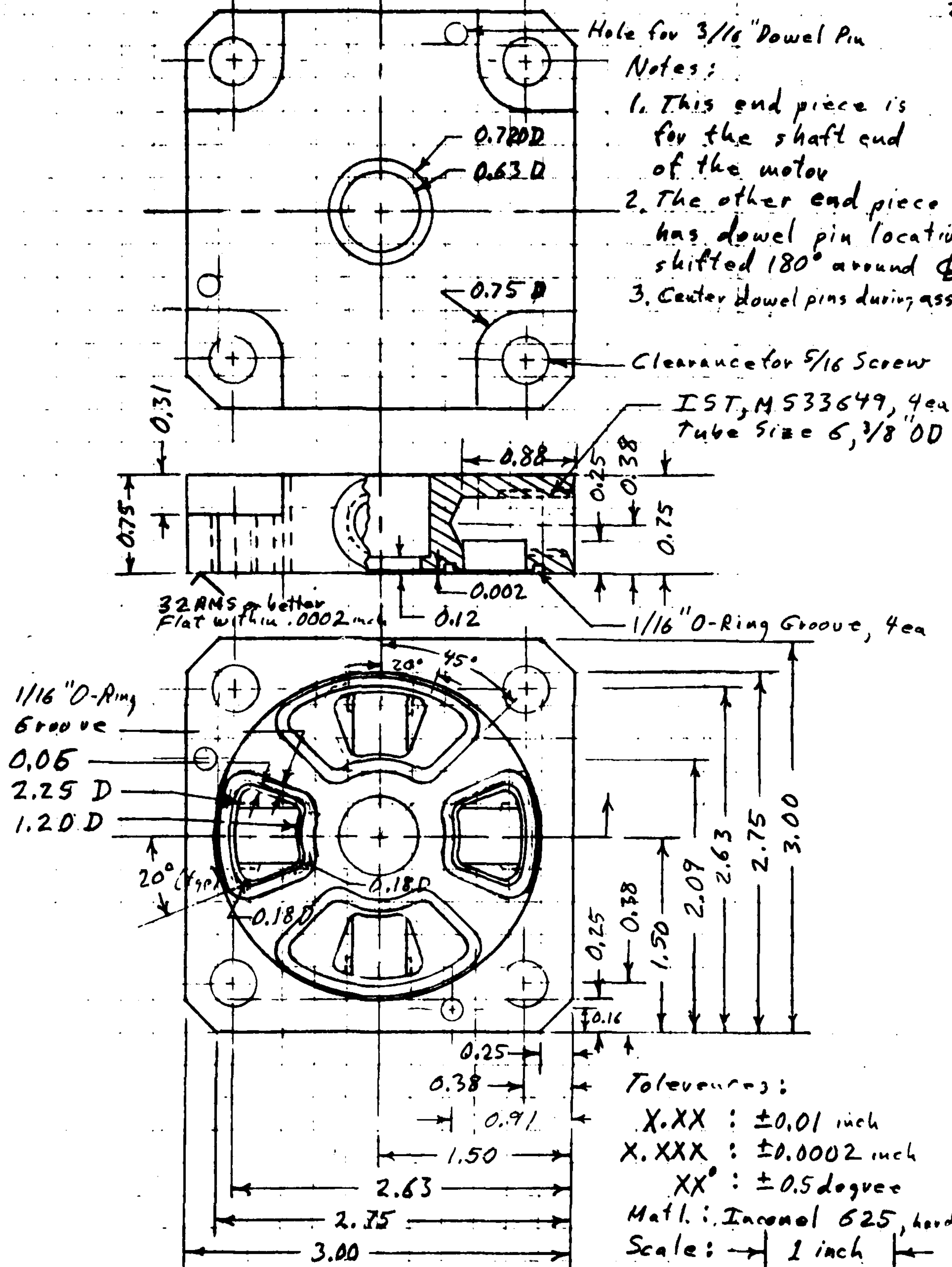

Notes:

1. This end piece is for the shaft end of the motor
2. The other end piece has dowel pin locations shifted 180° around ℄
3. Center dowel pins during assembly

Tolerances:

X.XX : ±0.01 inch
X.XXX : ±0.0002 inch
XX° : ±0.5 degree
Matl.: Inconel 625, hard.
Scale: → 1 inch ←

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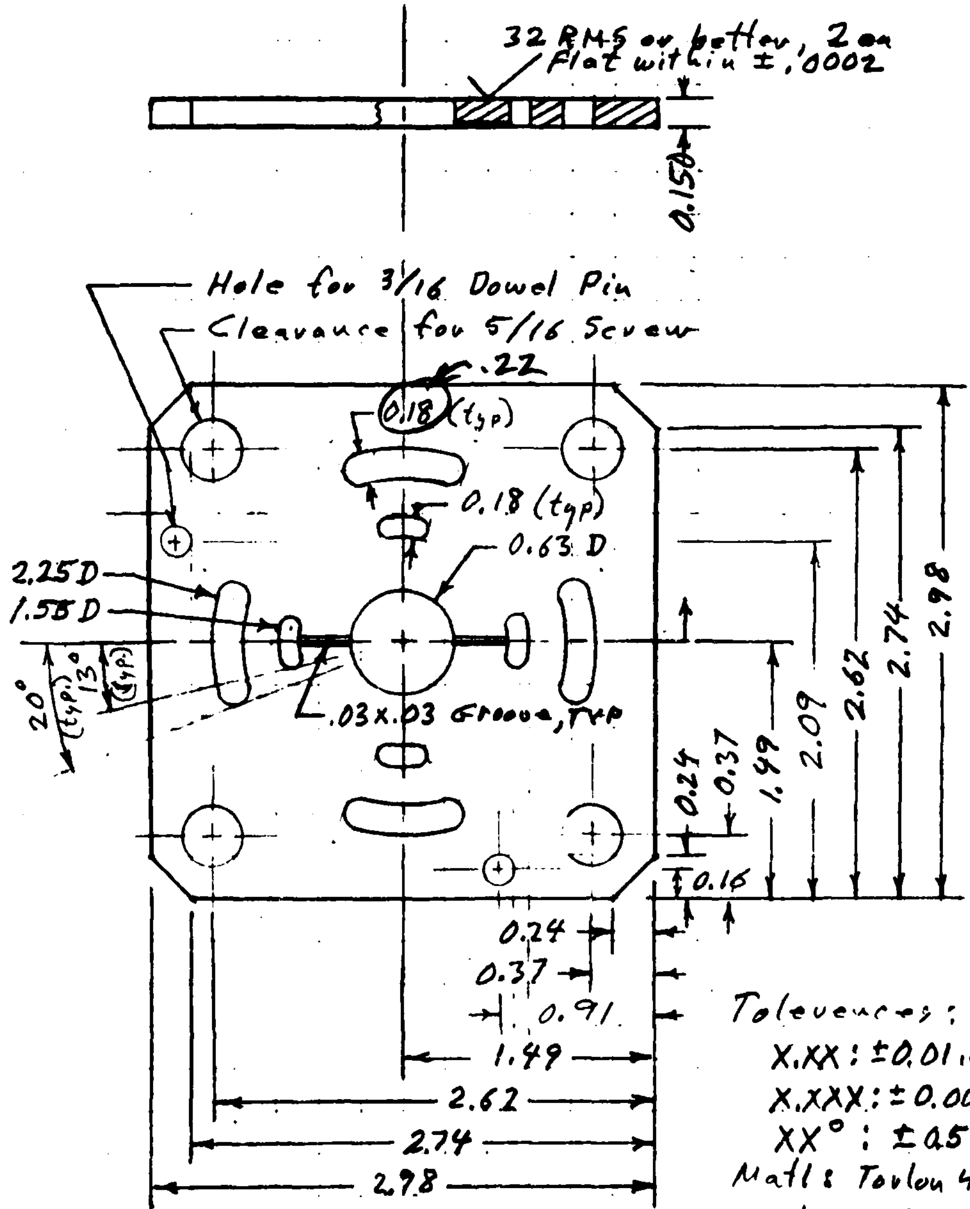
End Piece, Prelim. Design

WEC JG-2B

Note: V-slots are to be added to face toward rotor after first test.



Tolerances:
X.XX : ±0.01 inch
X.XXX : ±0.0002 inch
XX° : ±0.5 degree
Mat'l: Torlon 4275, postcured
Scale: → 1 inch ←

Side Plate, Prelim. Design
WEC JC-3A

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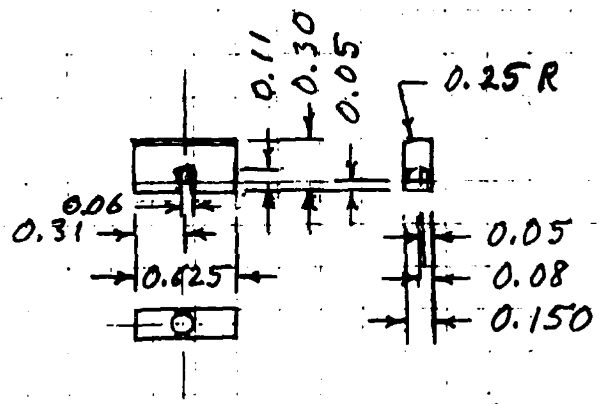
Rev: 4-3-79, JRC

Sealing Surfaces

Surface Finish: 64 RMS except for hole

Flatness within 0.0002 inch

Perpendicularity of adjacent flat surfaces $\pm \frac{.0002}{1}$



Tolerances:

X.XX : ± .01 inch

X.XXX: ± .0002^{5} inch

Matl.: Torlon 4275, post cu[illegible]

Scale: →| 1 inch |←

Vane, Prelim. Design

WEC JC-4A

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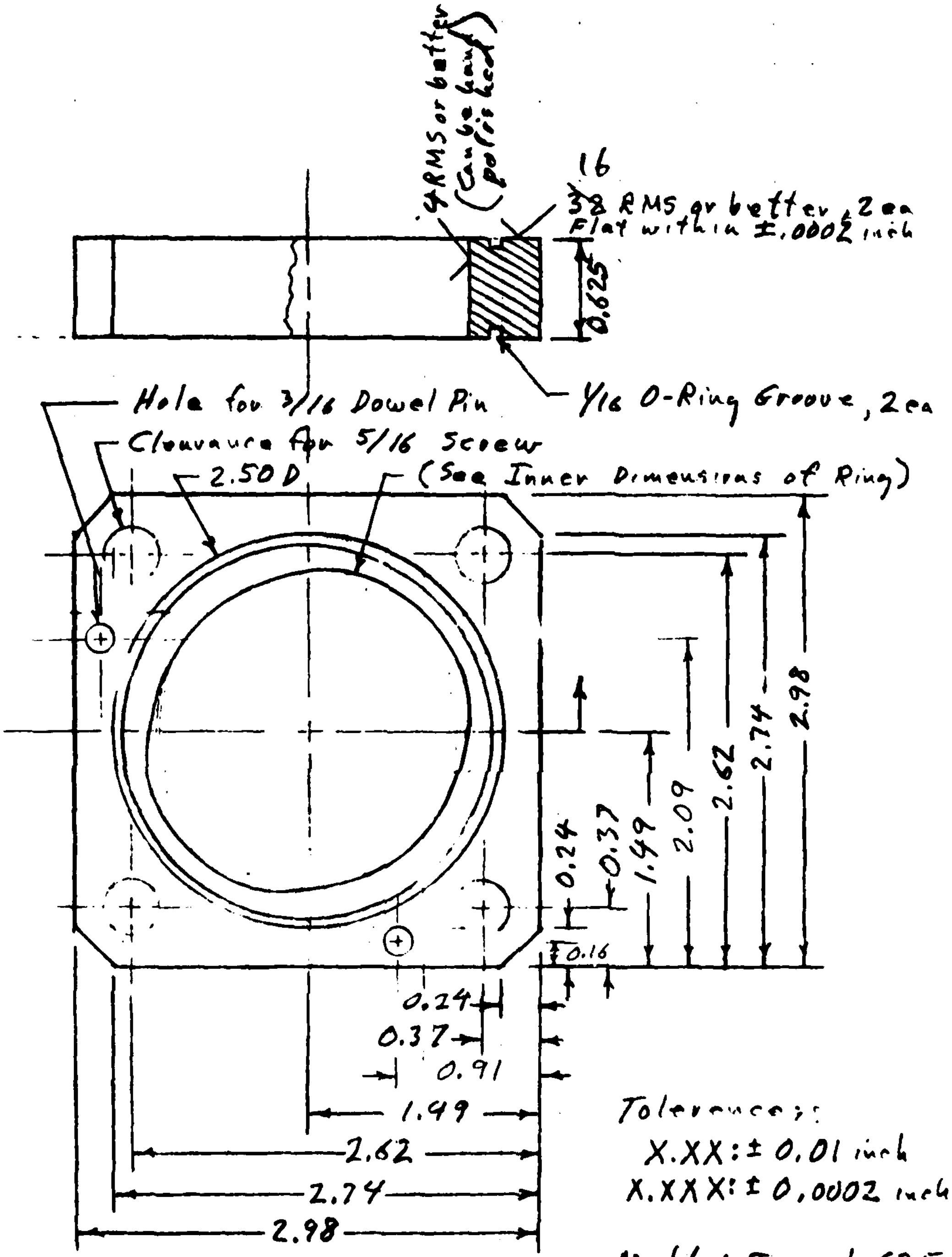

Tolerances:
X.XX: ± 0.01 inch
X.XXX: ± 0.0002 inch

Matl.: Inconel 625, hard
Scale: → 1 inch ←

Ring, Prelim. Design
WEC JC-5B

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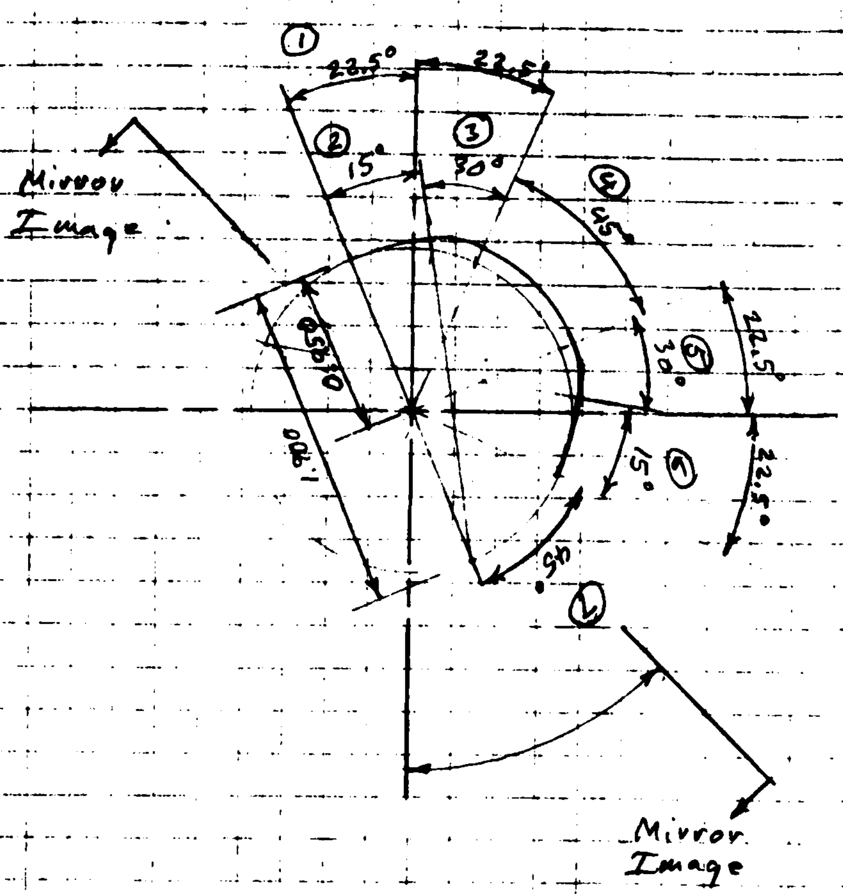

Tolerances:
X.XX : ± 0.01 inch
X.XXX : ± 0.0002 inch
XX° : ± 0.5 degree
XX.X° : ± 0.5 degree
Scale: 1 inch

Inner Dimensions of Ring, Prelim. Design
WEC JC-6

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Type: Compression with ends squared and ground

Dimensions and Tolerences: Same as Lee

Compression spring LC-018A-7

Outside diameter: 0.120 in

Wire diameter: 0.018 in

Free length: 5/8 in

Solid height: 0.29 in

Material: Elgiloy wire cold worked and spring heat treated for maximum fatigue strength.

Vendors:

Elgiloy Co., Elgin IL, (312) 695-1900, Don Nasielak

Kirk-Habicht Co, Baltimore MD, (301) 485-7200

Lee Spring Co., Brooklyn, NY, (212) 855-7163

Spring Specifications, Prelim. Design
WEC JC-7A

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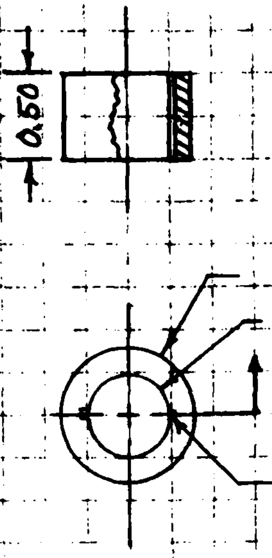
Notes:

1. Install bearing into end piece with grooves oriented toward the outlet ports.
2. Fine machine (bore, ream or etc) inside diameter of bearings after assembly of End Pieces, Side Plates, Ring, Dowel Pins and Tie Bolts.



Size for light press fit into 0.720 D hole

0.5015 D, after installation

Groove, .04 W, .04 deep, 2ea

Tolerances

Diameters: ±0.0002 inch (includes concentricity)

Length and groove: ±0.01 inch

Mat'l: Torlon 4275, post cured

Scale: →| 1 inch |←

Bearing Sleeve, Rolim, Design

WEC-JC-8A

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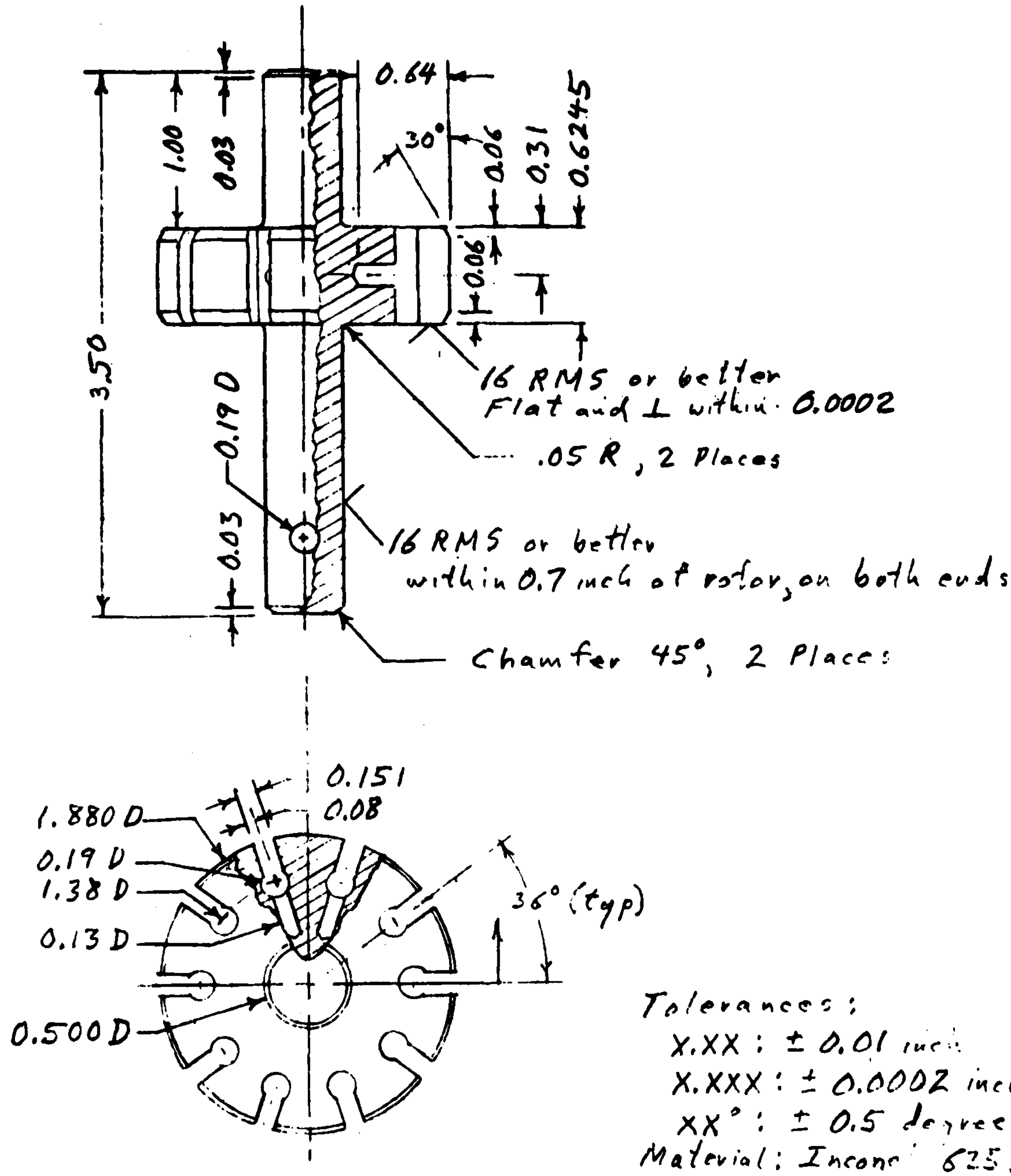

Tolerances:
X.XX : ± 0.01 inch
X.XXX : ± 0.0002 inch
XX° : ± 0.5 degree
Material: Inconel 625, ...
Scale:

Rotor with Shaft, Prelim. Design
WEC-JC-9A

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Dirt Seal 1¢

Shamban Series 25E Light Duty Excluder

Shamban No. 25E 00500A46

Shaft diameter: 1/2 inch

Fluid: seawater

Vendor; W. S Shamban & Co. Fort Wayne, Ind.

(219) 749-9631

Dirt Seal, Specifications, Prelim. Design
WEC JC-10

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APPENDIX C

Preliminary Design Calculations

Rotor and Vane Calculations

Pressure: 1500 psi (assume)
This is a pressure midway between the 1000 to 2000 psi extremes of desired pressures. The higher pressure causes more wear and the lower pressure more flow and larger umbilicals

Shaft power: 4 HP (assume)
This is midway between the 3 to 5 HP extremes of desired shaft power

Motor efficiency: 70% (assume)
This seem reasonable for a seawater motor of the type being considered

Motor speed: 1,500 c/min (assume)
This is near the lower end of the desired speed range of 1000 to 15000 RPM

Motor torque: $\dfrac{\left(33{,}000 \frac{ft\,lb/min}{HP}\right)(4\,HP)\left(12 \frac{in}{ft}\right)}{\left(1500 \frac{c}{min}\right)\left(2\pi \frac{rad}{c}\right)} = 168\ in\,lb$

Volumetric efficiency: 85% (assume)

Mechanical efficiency: $(100\%)(70\%)/(85\%) = 87.5\%$

No loss torque: $(168\ in\,lb)/(.875) = 192\ in\,lb$

Displacement per radian: $(192\ in\,lb)/(1500\ psi) = .128\ in^3/rad$

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Displacement per cycle: $\left(.128 \frac{in^3}{rad}\right)\left(2\pi \frac{rad}{c}\right) = .804\ in^3/c$

Mean inside radius of ring: 1.00 in (assume)

Pump cycles per shaft cycle: 2 P/c
Based on 2 lobe rotor balance.

Mean velocity of vane tip: $\frac{(1500 \frac{c}{min})(1 \, in)(2\pi \frac{rad}{c})}{(60 \frac{s}{min})} = 157.1 \frac{in}{s}$

Width of vane: .625 in (assume)
Based on width to diameter ratios of known hydraulic vane pumps and motors

Displacement flow: $(.804 \frac{in^3}{c})(1500 \frac{c}{min}) = 1,206 \frac{in^3}{min}$

or $(1206 \frac{in^3}{min})/(60 \frac{s}{min}) = 20.1 \; in^3/s$

or $(1206 \frac{in^3}{min})/(231 \frac{in^3}{gal}) = 5.22 \frac{gal}{min}$

or $\dot{V} = \underset{vel}{U} \; \underset{width}{w} \; \underset{stroke}{S} \; \underset{pump\ cycles/shaft\ cycle}{2 \, P/c}$

Stroke of the vane: $S = \dot{V}/2Uw$

$(20.1 \frac{in^3}{s})/(2 \frac{pc}{c})(157.1 \frac{in}{s})(.625 in) = .102$ in

This results in a relatively large stroke to rotor radius compared to other vane pumps and motors used for 1500 psi hydraulics. Vickers' PF4-05? vane pump has about 1/2 the stroke and about the same rotor diameter. The larger stroke results in larger ramp slope, smaller tip diameter and higher tip stress.

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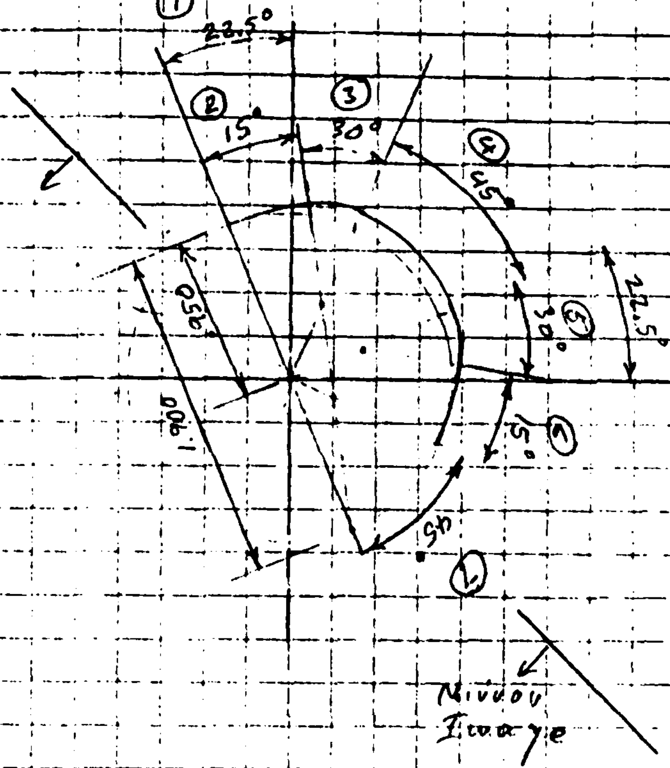

Inner Dimensions of Ring

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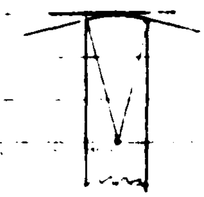
Max slope range of vane tip: ± .25 (see ring dim.)

Thickness of vane: .15 in (assume)



~~Vane tip radius: $R = \frac{w}{2} / \sin \frac{\theta}{2} =$~~

$$R = \left(\frac{.15\text{ in}}{2}\right) / (.25) = .3 \text{ in}$$

Vane tip diameter: $2(.3\text{ in}) = .6\text{ in}$

Major diameter of ring: $2(1.05\text{ in}) = 2.10\text{ in}$

Force on vane, radially: $(1500 \frac{lb}{in^2})(.625\text{ in})(.15\text{ in})/2 = 70.3\text{ lb}$

Vane properties (assume Torlon 4275)
(From AT-7 by Amoco Chemicals Corp.)
Compressive modulus: $.31 \times 10^6$ psi
Poisson's ratio: .45
Compressive strength: 20,000 psi

This seems low by a factor of 1/20. Jim Hooker of Amoco is checking it

Ring properties (assume Inconel 625)
(From Matl. Engrg, Nov '77)
Tensile modulus: 29.7×10^6 psi
Poisson's ratio: .3
Tensile yield stress: 53,000 psi
Tensile ultimate str: 124,000 psi

Hertz stress (From Roark, p 320)

$$S = .798 \sqrt{\frac{\frac{(70.3\text{ lb})}{(.625\text{ in})} \cdot \frac{(2.1\text{ in}) - (.3\text{ in})}{(2.1\text{ in})(.3\text{ in})}}{\frac{1-(.3)^2}{29.7 \times 10^6 \text{ psi}} + \frac{1-(.45)^2}{.31 \times 10^6 \text{ psi}}}} = .798 \sqrt{\frac{321.4 \text{ psi}}{.0306 \times 10^{-6} \frac{1}{psi} + 2.57 \times 10^{-6} \frac{1}{psi}}}$$

$$= 8,870 \text{ psi}$$

This is much higher than the fatigue strength of Torlon after 10^7 cycles (4200 psi in flex). It may be too high because of cylindrical tip assumption though, or too low from too low compressive modulus, CS

J.R. Coleton
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PV value: $(8870\ \text{psi})(157.1\ \tfrac{\text{in}}{\text{s}})(60\ \tfrac{\text{s}}{\text{min}})(12\ \tfrac{\text{in}}{\text{ft}}) = 6.97 \times 10^6\ \text{psi ft/min}$
This is about twice the value tested by MTI. Our assumptions might be conservative overall, resulting in a PV value closer to the max test value of $(2180\ \text{psi})(1500\ \text{ft/min}) = 3.27 \times 10^6\ \text{psi}\ \tfrac{\text{ft}}{\text{min}}$ for Torlon 4301 vs Inconel 625, resulting in 2.1×10^{-3} in/100 hr over a 30 hour test. Torlon 4275 is a newer material than 4301 with longer wear characteristics. (per Amoco)

Vane friction
Vane high load: 70.3 lb
Number of vanes under high load: 5
Dynamic friction coef.: .02
Static friction coef.: .2
Dynamic friction torque: $(1\ \text{in})(70.3\ \text{lb})(.02)(5) = 7.03\ \text{in lb}$
Static friction torque: $(1\ \text{in})(70.3\ \text{lb})(.2)(5) = 70.3\ \text{in lb}$
No loss torque: $(1500\ \tfrac{\text{lb}}{\text{in}^2})(.128\ \tfrac{\text{in}^3}{\text{rad}}) = 192\ \text{in lb}$
The static friction is only 37% of no load friction and the dynamic friction is only 4% of no load friction

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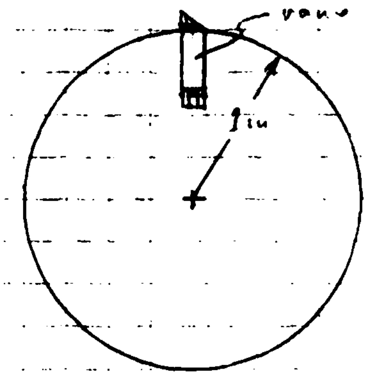
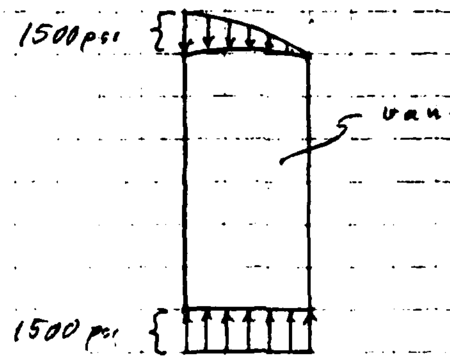
Non-Conservative PV Value for Vane

Vane Tip radius : 1inch



Pressure distribution on vane:



Tip area : $(.15\ in)(.625\ in) = .094\ in^2$

Fluid pressure force on vane:

$$(.094\ in^2)\left[(1500\ psi) - \tfrac{1}{2}(1500\ psi)\right] = 70.5\ lb$$

Mean velocity of vane tip:

$$(1500\ \tfrac{r}{min})(2\pi\ \tfrac{in}{r})(1\ in)/(12\ \tfrac{in}{ft}) = 785\ \tfrac{ft}{min}$$

Normal stress on vane tip:

$$(70.5\ lb)/(.094\ in^2) = 750\ psi$$

PV value : $(750\ psi)(785\ \tfrac{ft}{min}) = \underline{\underline{589{,}000\ psi\ ft/min}}$

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PV values based on initial Hertz stress are about 7×10^6 psi ft/min and after some wear are about 2×10^6 psi ft/min

Vane Acceleration Effects

Volume of vane : $(.625\,in)(.15\,in)(.30\,in) = .0281\,in^3$

Mass of vane : $\dfrac{(.0281\,in^3)\left(.05\,\frac{lb}{in^3}\right)}{\left(386\,\frac{in}{s^2}\right)} = 3.64\times10^{-6}\,\frac{lb\,s^2}{in}$

Rotational frequency : $\left(1500\,\frac{c}{min}\right)\left(2\pi\,\frac{r}{c}\right)\left(60\,\frac{s}{min}\right) = 157.1\,\frac{r}{s}$

Centrifugal force of vane from rotational acceleration:

$$\underbrace{(.9\,in)}_{\text{mean radius of vane}}\left(157.1\,\frac{r}{s}\right)^2\left(3.64\times10^{-6}\,\frac{lb\,s^2}{in}\right) = \underline{\underline{.0809\ lb}}$$

Very small

Angle of 1/2 cycle while vane is on ramp : $45°$

Assume ramp is sinusoidally shaped

Vane on ramp frequency : $\dfrac{(360°)}{2(45°)}\left(1500\,\frac{c}{min}\right)\left(2\pi\,\frac{r}{c}\right)\left(60\,\frac{s}{min}\right) = 628\,\frac{r}{s}$

Vane on ramp position amplitude : $.05\,in$

Vane on ramp vel. amplitude : $(.05\,in)\left(628\,\frac{r}{s}\right) = 31.4\,\frac{in}{s}$

Vane on ramp acc. ampl. : $\left(31.4\,\frac{in}{s}\right)\left(628\,\frac{r}{s}\right) = 19,719\,\frac{in}{s^2}$

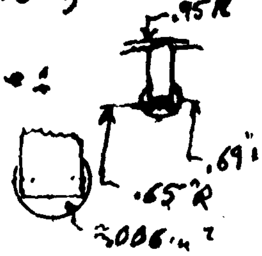
Vane on ramp acc. force ampl. : $\left(19,719\,\frac{in}{s^2}\right)\left(3.64\times10^{-6}\,\frac{lb\,s^2}{in}\right) = \underline{\underline{.0718\,lb}}$

Very small

Area of vane projected radially : $(.625\,in)(.15\,in) = .0938\,in^2$

Max velocity of water to inner side of vane:

$\left(31.4\,\frac{in}{s}\right)\left(.0938\,in^2\right)/\left(.006\,in^2\right)2 = 245.4\,\frac{in}{s}$



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Max velocity pressure of water to inner side of vane;

$$\left[\frac{\left(245.4 \frac{in}{s}\right)}{\underbrace{.7}_{\text{assumed flow coefficient}}}\right]^2 \frac{1}{2}\left(93.5\times10^{-6} \frac{lb\,s^2}{in^4}\right) = \underline{\underline{5.75 \text{ psi}}}$$

OK

Max acceleration of water to inner side of vane;

$$\left(\left(9,719 \frac{in}{s^2}\right)\left(.0938 \, in^2\right)\right) / \left(\left(.006 \, in^2\right) 2\right) = 154,100 \frac{in}{s^2}$$

Max acc. pressure of water to inner side of vane

$$\Delta P = L\ddot{V} = \frac{\rho l}{\cancel{A}} \cancel{A} \ddot{x} = \rho l \ddot{x}$$

$$\left(93.5\times10^{-6} \frac{lb\,s^2}{in^4}\right)\left(\frac{.625 \, in}{4}\right)\left(154,100 \frac{in}{s^2}\right) = \underline{\underline{2.25 \text{ psi}}}$$

OK

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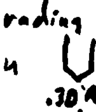
Spring Calculation

Material Properties:
~~Elgiloy Strip~~ ~~85% cold work reduction in cross section, heat treated 900°F for 5hr~~
~~Ult. tensile str: 368,000 psi~~
~~Tensile yield str: 280,000 psi~~
~~Tensile modulus: 30 x10^6 psi~~
~~Reverse bending fatigue str: 100,000 psi @ >10^7 cycles~~
~~Torsional modulus: 11.5 x10^6 psi~~

→ Elgiloy Wire: 45-48% cold reduction, 980°F for 5hr after forming,
Ult. tensile str: 340,000 psi
Tensile yield str: 310,000 psi
Shear yield: 210,000 psi
Tensile modulus: 28.5×10^{-6} psi
Torsional modulus: 11.2×10^{-6} psi
Fatigue str: 100,000 psi @ >10^7 cycles {assume same as for reverse bending of Elgiloy strip}

Compression spring dimensions:
Outside diameter: 0.12 in
Wire diameter: 0.018 in
Mean diameter: 0.12 in − 0.018 in = .102 in
Free length: .625 in
Number of turns: 16 (16−2 = 14 active turns)
Max spring compression: .17 in (assume)
This requires that the spring hole inner radius is
$.95\text{ in} - .19\text{ in} - (.63\text{ in} + .17\text{ in}) = .30\text{ in}$



Max spring force:

$$P = \frac{(.17\text{ in})(11.2\times10^6\text{ psi})(.018\text{ in})^4}{8(.102\text{ in})^3(14)} = 1.68\text{ lb}$$

Minimum spring force: $(1.68\text{ lb})(.07\text{ in})/(.17\text{ in}) = .69\text{ lb}$

Max stress in torsion

$$S = \frac{8(1.68\text{ lb})(.102\text{ in})}{\pi(.018\text{ in})^3}(1.26) = 94,300\text{ psi}$$

Just under 100,000 psi

Wahl factor 1.26 based on $\frac{.102\text{ in}}{.018\text{ in}} = 5.67$

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4 JAN 79

Volume in spring mat'l: $\frac{\pi}{4}(.018\,in)^2\pi(.102\,in)(16) = .013\,in^3$

Mass of spring: $\frac{(.0013\,in^3)(.3\,lb/in^3)}{(386\,\frac{in}{s^2})} = \frac{(.00039\,lb)}{(386\,\frac{in}{s^2})} = 1.0\times10^{-6}\,\frac{lb\,s^2}{in}$

Spring natural freq: $\frac{1}{2\pi}\sqrt{\frac{(10\,lb/in)\,4}{(1.0\times10^{-6}\,\frac{lb\,s^2}{in})}} = 1{,}000\,\frac{c}{s}$

This is much higher than the rotational freq, $25\,\frac{c}{s}$

Volume of vane: $(.625\,in)(.15\,in)(.30\,in) = .0281\,in^3$

Mass of vane: $\frac{(.0281\,in^3)(.05\,lb/in^3)}{(386\,\frac{in}{s^2})} = \frac{(.001405\,lb)}{(386\,\frac{in}{s^2})} = 3.64\times10^{-6}\,lb\,s^2/in$

Natural freq of spring constant and vane mass:

$$\frac{1}{2\pi}\sqrt{\frac{(10\,lb/in)}{(3.64\times10^{-6}\,\frac{lb\,s^2}{in})}} = 264\,\frac{c}{s}$$

This is about 10 times the rotational freq, $25\,\frac{c}{s}$, and about 5 times the vane oscillating freq, $50\,\frac{c}{s}$, and about 2.6 times the frequency of vane motion during outward and inward motion, $100\,\frac{c}{s}$. This higher natural frequency and the varying spring force will cause the force between the vane and ring to be at least 1 lb, excluding fluid pressure forces on the vane which could be much larger.

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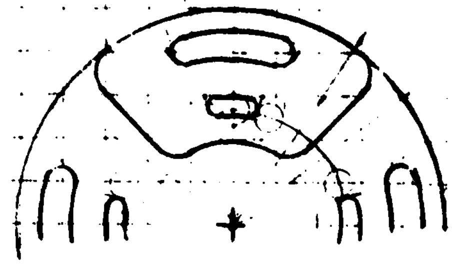
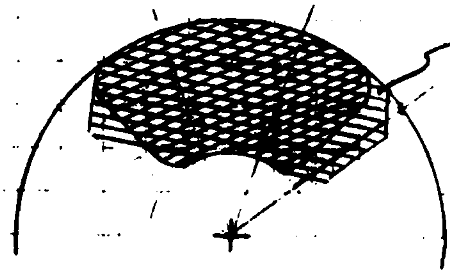
Side Plate Pressure Balance



Vane on minor diameter just after increased pressure

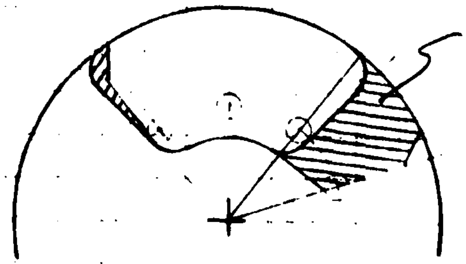


Unbalanced area : .10 in^2 (est)

Unbalanced force:
$(.10\ in^2)(1500\ \frac{lb}{in^2}) = 150\ lb$

This could be conservative because force reduced with leakage etc.

Vane on major diameter just before decrease pressure



Unbalanced area : .30 in^2 (est

Unbalanced force : $(.3\ in^2)(1500\ psi) = 450\ lb$

This could be very conservative because force reduced with leakage caused by plate deflection

Vane on major diameter just after decreased pressure
Unbalance area and unbalanced force : very small

Vane on minor diameter just before increased pressure
Unbalanced area : .20 in^2 (est)
Unbalanced force : $(.20\ in^2)(1500\ psi) = 300\ lb$
This could be conservative

J.R. Colston
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Friction force : $(300\ lb)(.02) = 6\ lb$
Friction torque : $(6\ lb)(1\ in)(4) = 24\ in\ lb$
(for both sides and both inlet ports)
Start-up friction torque : 240 in lb (may be problem)

Side Plate O-Ring Forces

Finding torque caused by O-ring around ports

Depth of O-ring groove: .052 ±.002 in
Carlson p 7-8

Gap between End Piece and Side Plate: .002 in

O-ring thickness: .070 ±.003 in
Parker p 7-8

O-ring squeeze: $(.070 \pm .003\,\text{in}) - (.054 \pm .002\,\text{in}) = .016 \pm .005\,\text{in}$

O-ring compression %: $(100\%)(.016 \pm .005\,\text{in})/(.070\,\text{in}) = 23 \pm 7\%$

O-ring Hardness: Shore A 70

Force per length: 4 ± 3 lb/in (Carlson)

O-ring length around one port: 3 inch

Force per port: $(3\,\text{in})(4 \pm 3\,\frac{\text{lb}}{\text{in}}) = 12 \pm 9\,\text{lb}$

Force for 4 ports: $4\,(12 \pm 9\,\text{lb}) = 48 \pm 36\,\text{lb}$

both sides

Static friction torque: $2(.9\,\text{in})(48 \pm 36\,\text{lb})(.2) = 17 \pm 13\,\text{in lb}$
Static coef of friction

Dynamic friction torque: $2(.9\,\text{in})(48 \pm 36\,\text{lb})(.02) = 1.7 \pm 1.3\,\text{in lb}$
Dynamic coef of fric

These are small compared to the maximum torque of 168 in lb.

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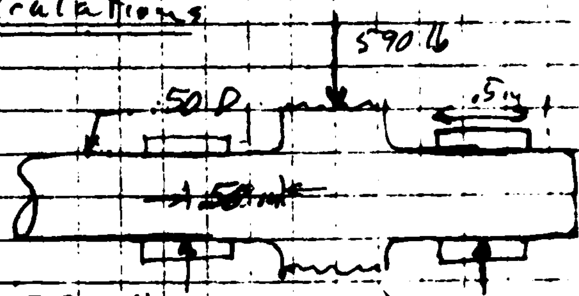
8 DEC 78

Shaft Deflection Calculations

Configuration:



Rotor unbalance force: 590 lb (assume)

Bearing force: (590 lb)/2 = 295 lb

Max shaft moment: (295 lb)(.56 in) = 165.2 in lb

Shaft radius: (.50 in)/2 = .25 in

Shaft area inertia: $\pi(.50\text{ in})^4/64$ = .00307 in^4

Max flex stress: (165.2 in lb)(.25 in)/(.00307 in^4) = 13,450 $\frac{\text{lb}}{\text{in}^2}$

Max shear stress: (295 lb)4/π(.5 in)^2 = 1,502 $\frac{\text{lb}}{\text{in}^2}$ (small)

Max torsion stress: (not calculated yet)

Max flex deflection: $(295\text{ lb})(.56\text{ in})^3/3(29.5\times10^{+6}\frac{\text{lb}}{\text{in}^2})(.00307\text{ in}^4)$

= 1.907×10^{-4} in (small)

Max slope at bearing: $(295\text{ lb})(.56\text{ in})^2/2(29.5\times10^{+6}\frac{\text{lb}}{\text{in}^2})(.00307\text{ in}^4)$

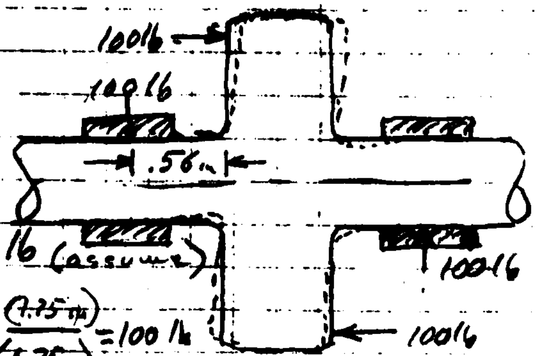
= 5.11×10^{-4}

Max misalignment in bearing: $(.5\text{ in})(5.11\times10^{-4})$ = .000256 in

This is small enough

J. R. Colston
12 DEC 78

Configuration:



Forces on rotor: 100 lb (assume)

Forces on bearings: $(100\,lb)\frac{(1.75\,in)}{(1.75\,in)} = 100\,lb$

Max shaft moment: $(100\,lb)(.56\,in) = 56\,in\,lb$

Shaft radius: $(.50\,in)/2 = .25\,in$

Shaft area inertia ($\sum A r^2$): $\pi(.50\,in)^4/64 = .00307\,in^4$

Max shaft deflection: $(100\,lb)(.56\,in)^3/3(29.5\times10^6\,\frac{lb}{in^2})(.00307\,in^4)$

$= \underline{\underline{.0000646\,in}}$

This deflection is small indicating the shaft is too stiff to accomodate much non-perpendicularity between the rotor and shaft.

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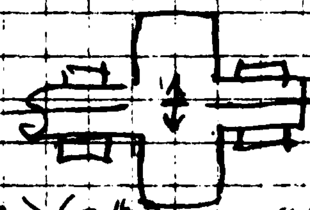
Natural Frequency of Rotor and Shaft

Configuration:



Mass of rotor: $\frac{\pi}{4}(1.8\,in)^2(.625\,in)(.3\,\frac{lb}{in^3}) = .477\,lb$
equivalent dia

or $(.477\,lb)/(386\,\frac{in}{s^2}) = .001236\,\frac{lb\,s^2}{in}$

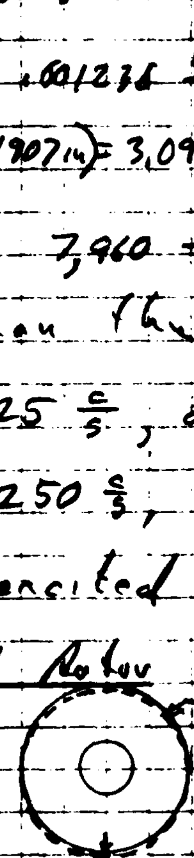
Spring constant of shaft: $(590\,lb)/(.0001907\,in) = 3.09\times10^6\,\frac{lb}{in}$
12-12-78 Calc.

Natural Frequency: $\frac{1}{2\pi}\sqrt{\frac{3.09\times10^6\,lb/in}{.001236\,\frac{lb\,s^2}{in}}} = 7,960\,\frac{c}{s}$

This is so much higher than the rotation frequency, $\frac{(1500\,\frac{c}{min})}{(60\,\frac{s}{min})} = 25\,\frac{c}{s}$, or the pump vane frequency, $250\,\frac{c}{s}$, that it probably will not be excited.

Acceleration Force from Unbalanced Rotor

Configuration: →



Unbalanced mass:
$2(.0005\,in)(.85\,in)(.625\,in)(.3\,\frac{lb}{in^3})/(386\,\frac{in}{s^2}) = .413\times10^{-6}\,\frac{lb\,s^2}{in}$

Rotation freq.: $(25\,\frac{c}{s})(2\pi\,\frac{r}{c}) = 157.1\,r/s$

Centrifugal acceleration: $(.95\,in)(157.1\,\frac{r}{s})^2 = 23450\,\frac{in}{s^2}$

Radial acc. force: $(.413\times10^{-6}\,\frac{lb\,s^2}{in})(23,450\,\frac{in}{s^2}) = \underline{\underline{.0097\,lb}}$

very small

V. A. Carlson
14 DEC 78

Bearing Calculations

Area between vanes on rotor: $\frac{2\pi\,(1\,in)(.625\,in)}{(10\ vanes)} = .393\,\frac{in^2}{vane}$

Maximum unbalance force: $(.393\,in^2)(1500\,psi) = 590\,lb$

Projected area of one bearing: $(.5\,in)(.5\,in) = .25\,in^2$

Mean pressure in bearing: $(590\,lb)/2(.25\,in^2) = 1{,}180\,psi$

Velocity of bearing: $(.25\,in)(1500\,\frac{c}{min})(2\pi\,\frac{r}{c}) / (12\,\frac{in}{ft}) = 196\,\frac{ft}{min}$

PV value: $(1{,}180\,psi)(196\,\frac{ft}{min}) = 231{,}000\,psi\,\frac{ft}{min}$
This should result in negligible bearing wear over 50 hours as long as the bearing is wetted with water for lubrication. MTI study indicates PV can be as high as $3\times10^6\,psi\,ft/min$

Hersey Number: $(1.0\,cp)(1500\,RPM)/(1180\,psi) = 1.27\,cp\,RPM/psi$
room temp value

Coef. of friction: .02 (see Fig 4.26, MTI Study)
The figure is for Torlon 4301 with Inco 625, Torlon 4275 should result is slightly less friction

Friction torque for both bearings: $(.25\,in)(590\,lb)(.02) = 3.0\,in\,lb$
This is negligible compared to the no loss torque of 192 in lb. A static friction coef. of .2 results in a static friction torque of 30 in lb which is acceptable for start up.

Heat power for both bearings: $\frac{(2\pi\,\frac{r}{c})(1500\,\frac{c}{min})(3.0\,in\,lb)}{(33000\,\frac{ft\,lb/min}{HP})(12\,\frac{in}{ft})} = .0714\,HP$

or $(.0714\,HP)(746\,\frac{W}{HP}) = 53.3\,W$

or $(53.3\,W)(3.413\,\frac{Btu/hr}{W}) = 182\,Btu/hr$

This is an unlikely worst case

J.A. Olsson
15 NOV 78

Heat transfer from one bearing through shaft to rotor

Heat power : $(182 \frac{Btu}{hr})/2 = 91 \frac{Btu}{hr}$

Length of shaft : 1.0 in (approx.)

Area of shaft : $\frac{\pi}{4}(.5 in)^2 = .196 in^2$

Thermal conductivity of Inco 625 : $5.67 \frac{Btu}{hr\,ft\,F}$
(Titanium is about the same) (Matl Engrg, NOV 77)
(Torlon 4275 is about .2 Btu/lb ft F

Heat transfer eq : $Q = KA\Delta T/L$

Temperature drop in shaft : $\Delta T = \dot{Q}L/KA$

$(91 \frac{Btu}{hr})(1.0 in)(12 \frac{in}{ft})/(5.67 \frac{Btu}{hr\,ft\,F})(.196 in^2) = 983\ F$

This indicates heat conduction is not adequate for the heat transfer.

Heat Transfer by cooling water flow through one bearing

Water Heating eq : $\dot{Q} = C_p \dot{M} \Delta T$

Temperature rise in water : 20 F° (assume)

Heat power : $91 \frac{Btu}{hr}$

Specific heat : $1 \frac{Btu}{lb\,F}$

Mass flow : $\dot{M} = \dot{Q}/C_p\Delta T$

$(91 \frac{Btu}{hr})/(1 \frac{Btu}{lb\,F})(20 F) = 4.55 \frac{lb}{hr}$

Volume flow : $(4.55 \frac{lb}{hr})/(.0361 \frac{lb}{in^3})(231 \frac{in^3}{gal})(60 \frac{min}{hr}) = .00909$ gal/min
(Structure in)

or $(231 \frac{in^3}{gal})(.00909 \frac{gal}{min}) = 2.1\ in^3/min$

This is reasonably low flow, but IMPORTANT flow

J.W. Glista
15 Nov 78

Thermal expansion of bearing material

Temperature rise (mean): 50 F (assume)

Radial dimension: .125 in

Coef of linear thermal expansion: $13 \times 10^{-6}\ 1/F$

Radial increase: $3(.125\text{ in})(50F)(13\times10^{-6}\tfrac{1}{F}) = .000244\text{ in}$

This is about .0005 in. decrease in ID.

Thermal expansion of shaft

Temperature rise (mean): 50F

Diameter of shaft: .5 in

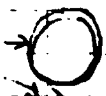
Coef of lin. thermal expansion: $7.1\times10^{-6}\ \frac{1}{F}$ (Inco 625)

Ti-6-4 is about $5.3\times10^{-6}\ 1/F$

Diametral increase: $(.5\text{ in})(50F)(7.1\times10^{-6}\tfrac{1}{F}) = .0018\text{ in}$

Diametral clearance in bearing: .0015 in (assume)

This is sufficient for termal expansions and is within machining tolerance.

$\varepsilon = 1$


Flow in viscous sleeve

Flow equation for viscous sleeve: $\dot{V} = \frac{\pi h^3 D}{12 \mu l}\left(1+\frac{3}{2}\varepsilon\right)\Delta P$

Pres. drop for minimum required flow: $\Delta P = \frac{12\mu l \dot{V}}{\pi h^3 D \frac{5}{2}}$

$$\frac{24(.145\times10^{-6}\frac{lb\,s}{in^2})(.5\text{ in})(2.1\frac{in^3}{min})}{\pi(.00075\text{ in})^3(.5\text{ in})\,5\,(60\frac{s}{min})} = 1.8\text{ psi}$$

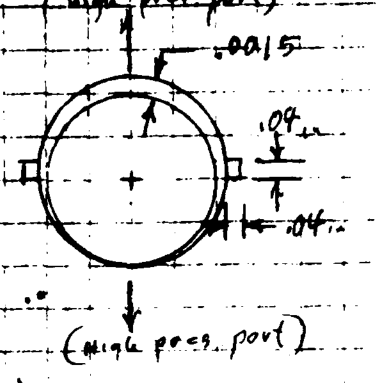
For a centered shaft, the pressure is $\frac{5}{2}1.8 = 4.5$ psi

Pressure drops are 28 times greater at max expansions

This indicates slots in sleeves are necessary.

J.A. Colston
15 Nov 78

Required Flow in bearing sleeve slots



Slot depth: .04 in

Slot width: .04 in

Slot length: .5 in

Number of slots: 2 ea

Area of slot: $(.04\,in)(.04\,in) = .0016\,in^2$

Flow per slot: $\left(21 \frac{in^3}{min}\right) / \left(2\,slot\right)\left(60 \frac{s}{min}\right) = .0175 \frac{in^3}{s\ slot}$

Flow velocity: $\left(.0175 \frac{in^3}{s}\right) / \left(.0016\,in^2\right) = 10.94 \frac{in}{s}$

Reynolds number: $\left(645 \frac{s}{in^2}\right)\left(.04\,in\right)\left(10.94 \frac{in}{s}\right) = 282$

This indicates viscous flow

Friction factor: $64/282 = .227$

Velocity pressure: $\frac{1}{2}\left(93.5 \times 10^{-6} \frac{lb\,s^2}{in^4}\right)\left(10.94 \frac{in}{s}\right)^2 = .00560\,psi$

Pressure drop from wall friction: $\Delta P = f \frac{L}{D} q$

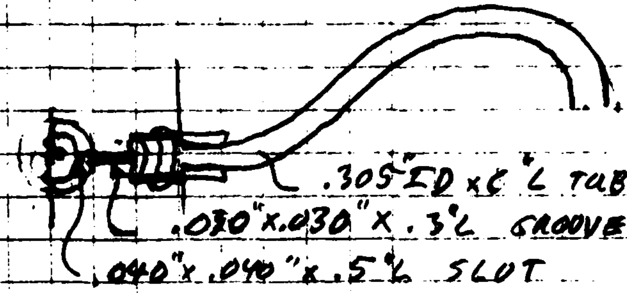
$$.227 \frac{(.5\,in)}{(.04\,in)}(.0056\,psi) = \underline{.0159\,psi}$$

This is so small it indicates the slots can be smaller by a factor of 2 in width and depth.

J. R. Colston
15 NOV 78

Bearing Flow Calculations

Flow paths:



Flow through tube: $\frac{(5.22 \frac{gal}{min})}{(.85)\,4} = 1.535 \frac{gal}{min}$

η_v

or $(1.535 \frac{gal}{min})(231 \frac{in^3}{gal})/(60 \frac{s}{min}) = 5.91 \ in^3/s$

Flow area of tube: $\frac{\pi}{4}(.305\ in)^2 = .0731\ in^2$

Flow vel in tube: $(5.91 \frac{in^3}{s})/(.0731\ in^2) = 80.8 \frac{in}{s}$

Velocity pressure in tube: $\frac{1}{2}(93.5 \times 10^{-6} \frac{lb\,s^2}{in^4})(80.8 \frac{in}{s})^2 = .0305$ psi

Reynolds number in tube: $(645 \frac{s}{in^2})(.305\ in)(80.8 \frac{in}{s}) = 15,900$

Friction factor in tube: .029 (for $\frac{\epsilon}{D} = .001$)

Wall friction of tube: $.029 \frac{(6\ in)}{(.305\ in)}(.0305\ psi) = .0174$

Increase in head: $(1\ in)(.0361 \frac{lb}{in^3}) = .0361$ psi

Total pres drop in tube: .0305 + .0174 + .0361 = .084 psi

Pressure drop in groove and slot: $\Delta P = \frac{32\,L\mu}{d^4}\dot{V}$

$\Delta P = 32\mu\left(\frac{L_1}{d_1^4} + \frac{L_2}{d_2^4}\right)\dot{V}$

Flow through groove and slot: $\dot{V} = \Delta P / 32\mu\left(\frac{L_1}{d_1^4} + \frac{L_2}{d_2^4}\right)$

[illegible] = .032 $\frac{in^3}{s}$

[illegible] .0175

Bearing Hydrodynamic Effects

Fluid viscosity: $(.145 \times 10^{-6} \frac{lb\,s}{in^2\,cp})(.85\,cp) = .123 \times 10^{-6} \frac{lb\,s}{in^2}$
for water at 80°F

Shaft speed: $(1500 \frac{r}{min})/(60 \frac{s}{min}) = 25 \frac{r}{s}$

Bearing length: .5 in

Bearing diameter: .5 in

Shaft radius: $(.5\,in)/2 = .25\,in$

Mean radial clearance: $(.0015\,in)/2 = .00075\,in$

Max load force: $(590\,lb)/2 = 295\,lb$

Min. Sommerfeld Number: $S = \frac{\mu N D L}{W}\left(\frac{R}{C}\right)^2$

$$\frac{(.123 \times 10^{-6} \frac{lb\,s}{in^2})(25 \frac{r}{s})(.5\,in)(.5\,in)}{(295\,lb)}\left(\frac{.25\,in}{.00075\,in}\right)^2 = \underline{\underline{.000029}}$$

Since this number is << .1, the shaft will not be supported by just a hydrodynamic force

Min. load force (from weight): $(.5\,lb)/2 = .25\,lb$

Max Sommerfeld Number: S

$$\frac{(.123 \times 10^{-6} \frac{lb\,s}{in^2})(25 \frac{r}{s})(.5\,in)(.5\,in)}{(.25\,lb)}\left(\frac{.25\,in}{.00075\,in}\right)^2 = \underline{\underline{.342}}$$

This number indicates the bearings will act as ideal hydrodynamic bearing with an eccentricity ratio of .6. Overloaded bearings occur at $S < .01$ and unstable bearings occur at $S \geq 1.0$. (fluid whip instability)

J. R. Colston
14 DEC 78

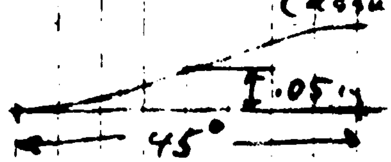
Finding Water Entrance and Exit Pressures Between the Rotor and Ring Next to a Vane on a Ramp

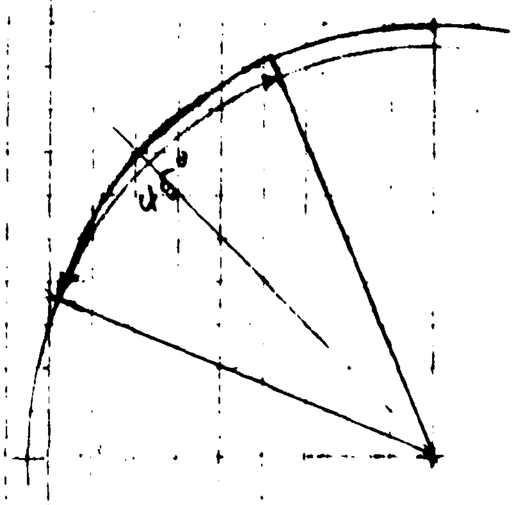
Ramp Configuration:

Ramp arc angle: 45°

Ramp ampl: .05 in

Ramp shape: sinusoidal (assumed)





Motor speed: 1500 $\frac{r}{min}$

Ramp cycle freq: $(1500 \frac{r}{min})(4 \frac{c}{r}) = 6000 \frac{c}{min}$

or $(6000 \frac{c}{min})(2\pi \frac{r}{c})/(60 \frac{s}{min}) = 628.3 \frac{r}{s}$

Vane velocity ampl.: $(.05 in)(628.3 \frac{r}{s}) = 31.4 \frac{in}{s}$

Vane acc. ampl.: $(31.4 \frac{in}{s})(628.3 \frac{r}{s}) = 19{,}740 \frac{in}{s^2}$

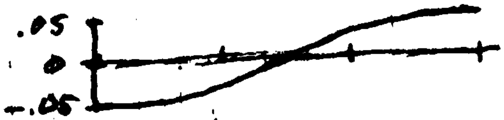
A slightly different approach follows

J.A. Colston
2 FEB 79

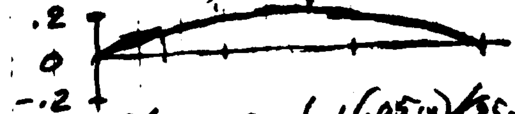
Functions of vane position on ramp

Shaft angle, deg.	0	15	30	45
Ramp angle, deg.	0	60	120	180

Radial position, in, X_r



Slope, $\frac{in}{in}$



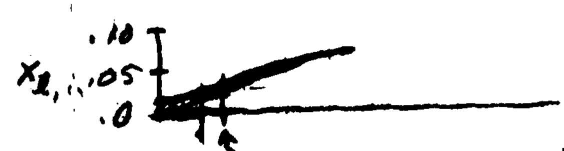
Ramp radian length: $1^{in}/4 = .25\,in$

slope ampl: $(.05\,in)/(.25\,in) = .2$

Circumferential vel of vane: $\frac{(1500\,\frac{r}{min})(2\pi\,\frac{rad}{r})(1\,in)}{(60\,\frac{s}{min})} = 157.1\,\frac{in}{s}$

Effective Gap axial dim: $\left(\frac{.625\,in}{2}\right) - (.06\,in) = .25\,in$

Effective Gap height ~~length~~: $2(.01\,in) + (X_r + .05) = X_l$, in



Assume worst case angle for flow, 30° Ramp angle

Volumetric flow: $(.03\,in)(.25\,in)(.1)(157.1\,\frac{in}{s}) = .1178\,\frac{in^3}{s}$

Flow area: $(.03\,in)(.015\,in) = .00045\,in^2$

Flow velocity: $(.1178\,\frac{in^3}{s})/(.00045\,in^2) = 261.8\,\frac{in}{s}$

Velocity pressure drop: $\frac{1}{2}(93.5\times10^{-6}\,\frac{lb\,s^2}{in^4})(261.8\,\frac{in}{s})^2 = 3.2\,psi$

Reynolds number: $(645\,\frac{s}{in^2})(.02\,in)(261.8\,\frac{in}{s}) = 3377$

Total friction pres drop: $\left[(.05)\frac{(.1\,in)}{(.02\,in)} + 1\right](3.2\,psi) = \underline{4\,psi}$

Not too high

J. R. Colston
2 FEB 79

Assumed worst case angle for acceleration, 0°

Volumetric flow acc: $\frac{(.1178 \frac{in^3}{s})(15.71 \frac{in}{s})}{(.13\,in)} = 142 \frac{in^3}{s^2}$

Path length : .1 in

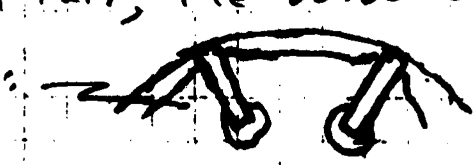
Path area : $(.01)(.05\,in) = .0005\,in^2$

Fluid inductance: $\frac{\Delta P}{\Delta \dot{V}} = L = \frac{\rho l}{a} = \frac{(93.5 \times 10^{-6} \frac{lb\,s^2}{in^4})(.1\,in)}{(.0005\,in^2)} = .0187 \frac{psi}{\frac{in^3}{s^2}}$

Acc pres drop: $(142 \frac{in^3}{s^2})(.0187 \frac{psi}{in^3/s^2}) = \underline{\underline{2.66\ psi}}$

Not too high

These results indicate the design is marginally OK for ~~not~~ preventing cavitation at 0, FSW. Wear in the bearing or tolerance build up could cause less than .01 in radial clearance between the rotor and cam ring, which in turn could cause higher flow velocity and flow acceleration pressure drops. To prevent this, the rotor could be reshaped like:



J. R. Colston
2 FEB 79

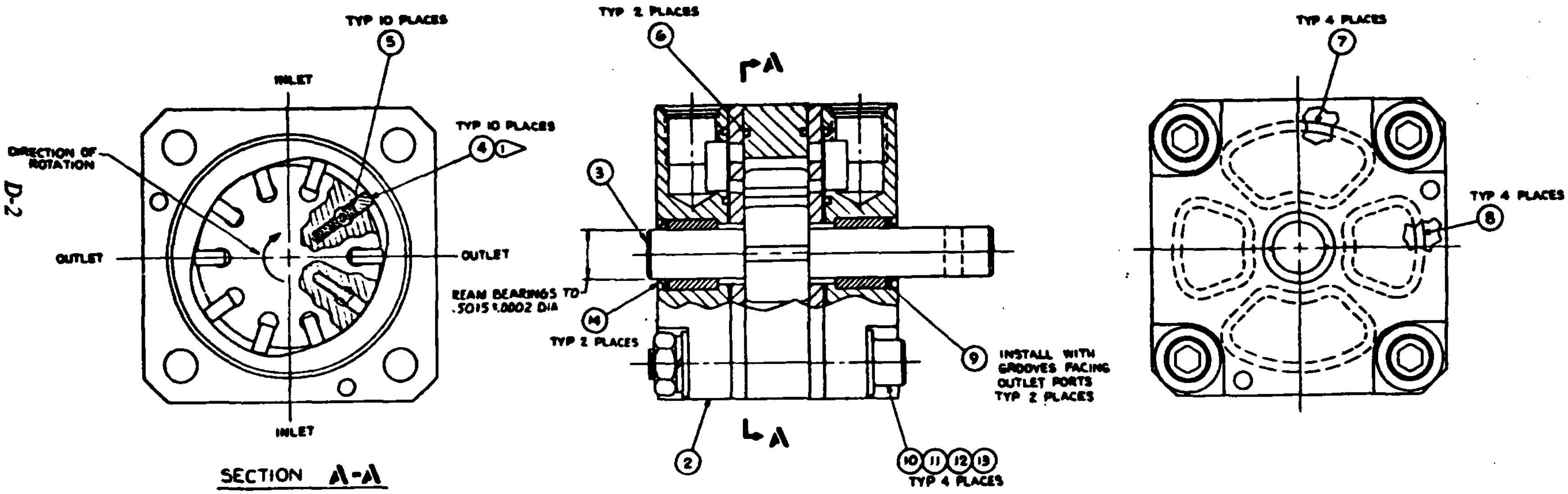
APPENDIX D

Drawing Package

Title	Drawing Number
Seawater Hydraulic Motor	
• Assembly	SKD 381839
• Parts List, Sheets 1 and 2	PL SKD 381839
• Body, Matched Set, Sheet 1	SKD 381840
• End Plate and Side Piece, Sheet 2	SKD 381840
• Ring, Sheet 3	SKD 381840
• Pin	SKB 376737
• Rotor	SKD 381841
• Spring	SKB 376735
• Vane	SKB 376734
• Bearing	SKB 376736

NOTES

1. INSURE VANES ARE INSTALLED IN THEIR PROPER ORIENTATION - THE LEADING FACE OF THE VANE HAS THE LARGER VERTICAL DIMENSIONS.



SECTION A-A

REVISIONS

A REVISED ... ORIENTATION

APPLICATION

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES
DO NOT SCALE

2 PLACE | 3 PLACE | ANGLES

PARTS LIST

Westinghouse Electric Corporation

SEAWATER HYDRAULIC MOTOR ASSEMBLY

18 DEC 78

SIZE	CODE IDENT NO.	DWG NO.
D	31442	SKD381839

PARTS LIST	Westinghouse Electric Corporation Undersea Division OCEAN RESEARCH AND ENGINEERING CENTER ANNAPOLIS, MARYLAND 21404	CONTRACT NO.	31442	PL SKD381839
		ORIGINAL DATE OF PL 78-12-19	CODE IDENT	SHEET 1 OF 2 SHEETS
		DRAFTSMAN R. MILLER / CHECKER		SEAWATER HYDRAULIC MOTOR ASSEMBLY
		ENGINEER / APPD		
		DESIGN ACTIVITY APPROVAL		A6

APPLICATION	
NEXT ASSY	USED ON
	SW HYDR MOTOR

SYMBOLS: V–VENDOR ITEM B–BULK MATERIAL TC NC & BC FOR Ⓦ REF ONLY

LTR	REVISION DESCRIPTION	DATE	APPD	LTR	REVISIONS (CONTINUED)	DATE	APPD
–	PARTS LIST RELEASED	1/23/79					

SHEET REVISION STATUS

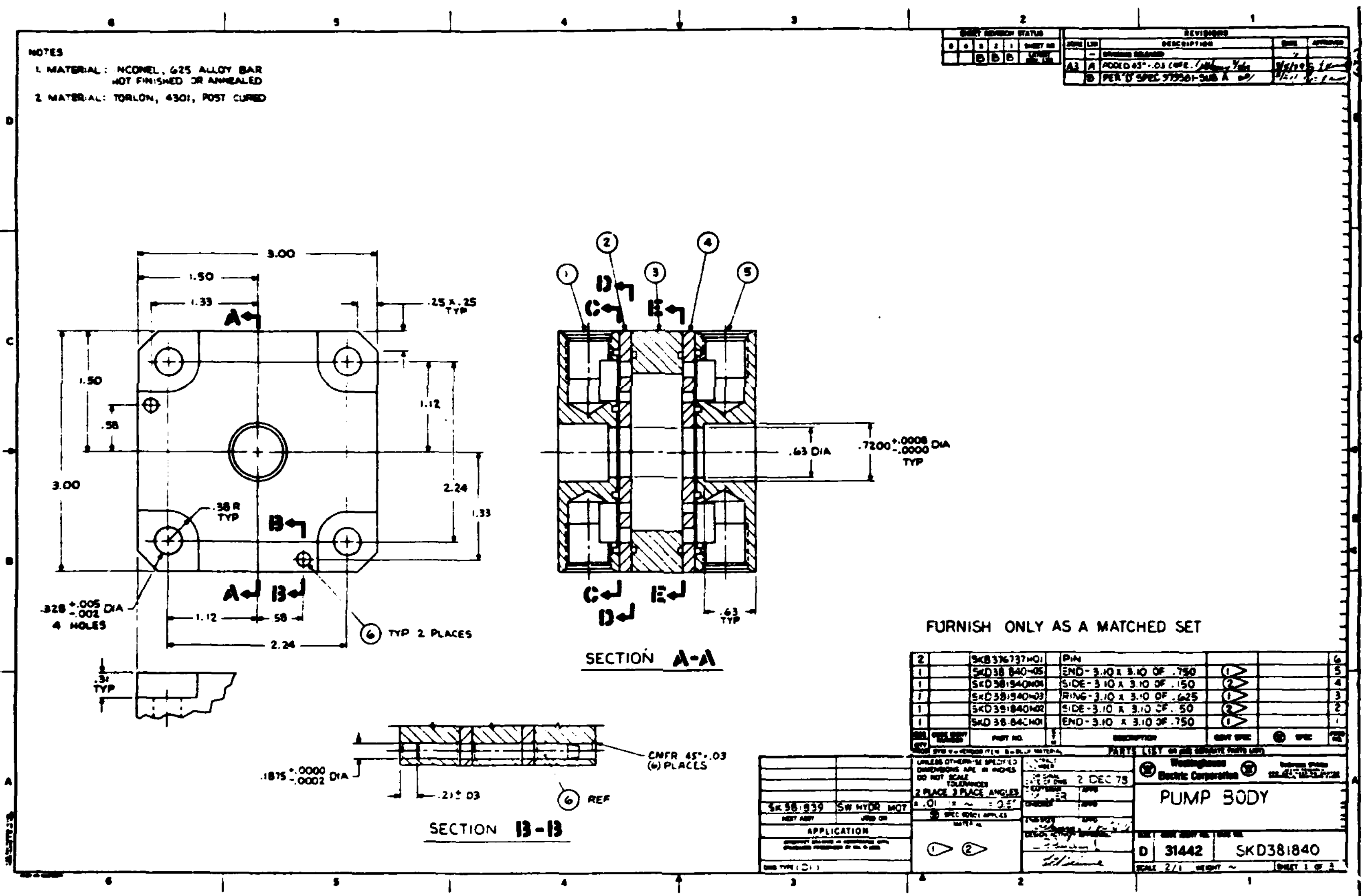
LATEST REV LTR	-	-																																				
SH NO.	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39

FIND NO.	QTY. REQ'D.	UM	CODE IDENT. NO.	PART NUMBER	SPEC NUMBER	NAME / DESCRIPTION / MAT'L	WT.	TC	NC	BC	SYM
001	REF			SKD381839G01		SEAWATER MOTOR ASSY					
002	1			SKD381840G01		PUMP BODY, MATCHED SET					
003	1			SKD381841H01		ROTOR					
004	10			SKB376734H01		VANE					
005	10			SKB376735H01		SPRING					
006	2		02697	2-036 V747-75	PARKER	O-RING					
007	4		02697	2-023 V747-75	PARKER	O-RING					

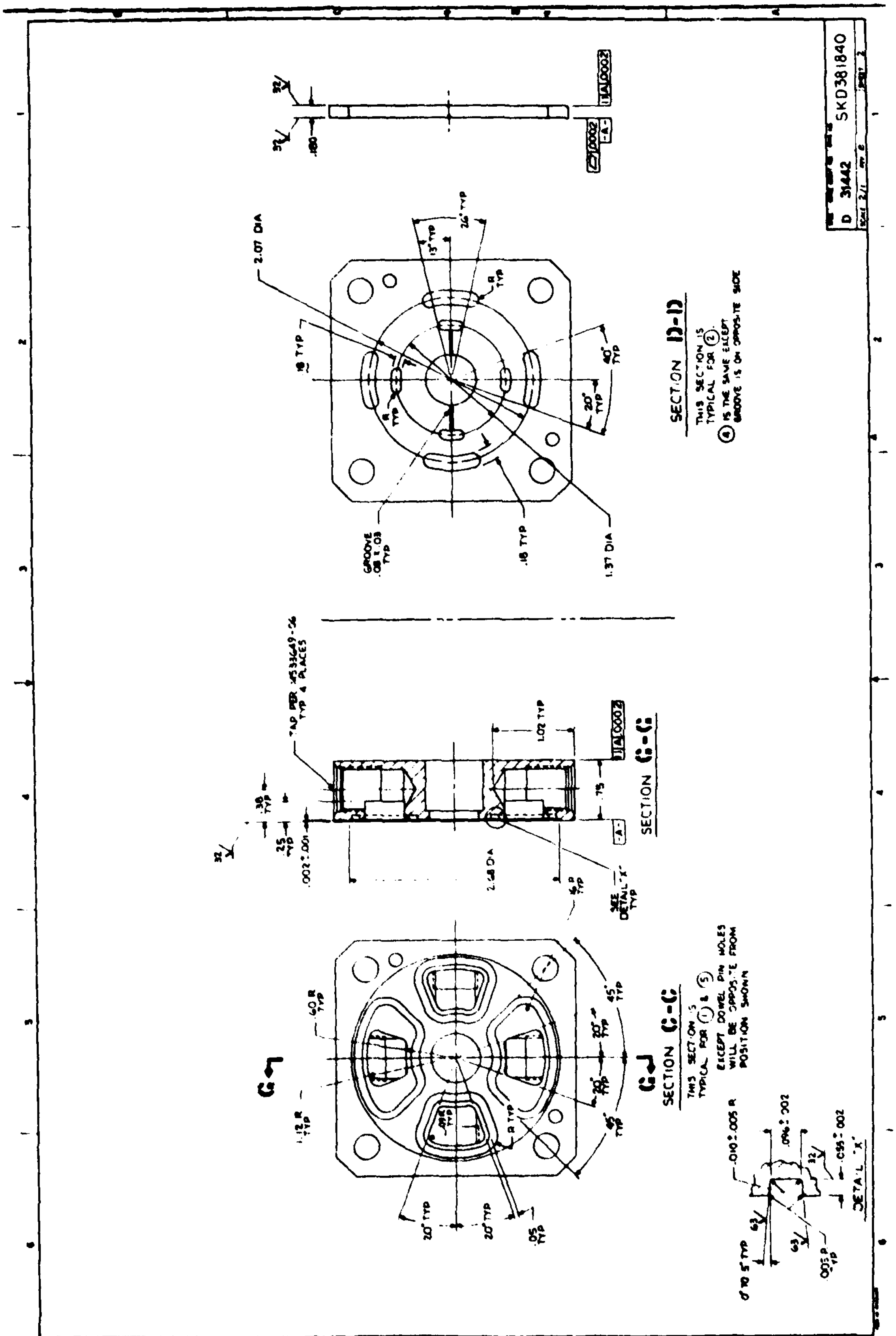
D-3

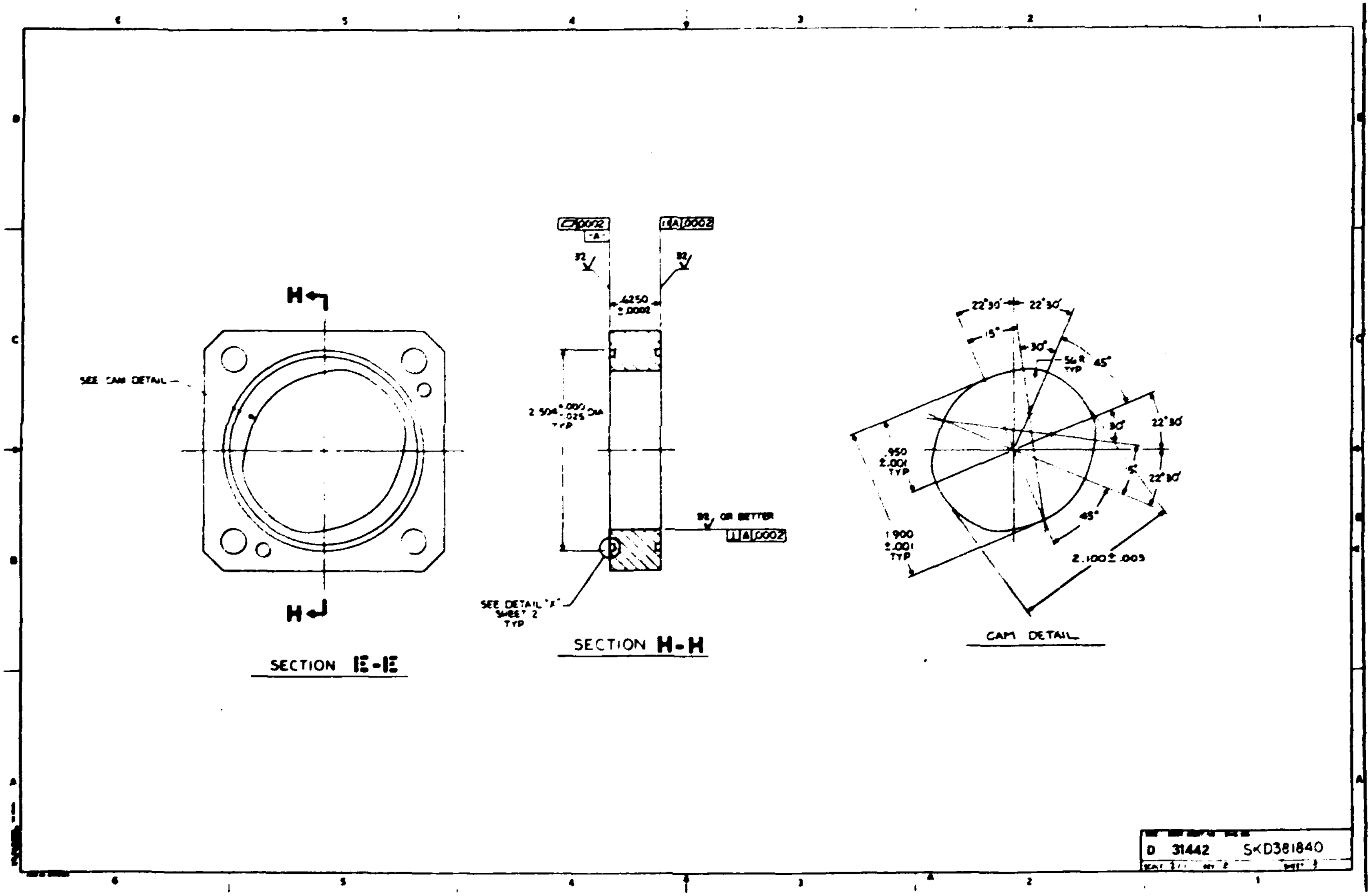
PARTS LIST CONTINUATION SHEET	CONTRACT NO.	SHEET NO. 2	CODE IDENT. NO. 31442	PL SKD381839	REV. -

FIND NO.	QTY. REQ'D.	UM	CODE IDENT. NO.	PART NUMBER	SPEC NUMBER	NAME / DESCRIPTION / MAT'L	WT.	TC	NC	BC	SYM
008	4		02697	2-019 V747-75	PARKER	O-RING					
009	2			SKB376736H01		BEARING					
010	4			MS24674-41		SCREW, CAP, .312-18					
011	8			MS15795-812		WASHER, FLAT					
012	4			MS35338-140		WASHER, LOCK					
013	4			MS35649-2314		NUT					
014	2		09257	25E00500A46	W.S. SHAMBAN & CO.	EXCLUDER, LIGHT SERIES					



NOTES
1. MATERIAL: INCONEL, 625 ALLOY BAR HOT FINISHED OR ANNEALED
2. MATERIAL: TORLON, 4301, POST CURED
3.00
1.50
1.33
.25 X .25 TYP
1.12
.58
2.24
1.33
.38 R TYP
.328 +.005 -.002 DIA 4 HOLES
6 TYP 2 PLACES
.31 TYP
.63 DIA
.7200 +.0008 -.0000 DIA TYP
.63 TYP
SECTION A-A
.1875 +.0000 -.0002 DIA
.21 ± .03
CMFR 45° ×.03 (6) PLACES
6 REF
SECTION B-B
FURNISH ONLY AS A MATCHED SET
PARTS LIST
PIN
END - 3.10 x 3.10 OF .750
SIDE - 3.10 x 3.10 OF .150
RING - 3.10 x 3.10 OF .625
SIDE - 3.10 x 3.10 OF .50
END - 3.10 x 3.10 OF .750
2 DEC 75
Westinghouse Electric Corporation
PUMP BODY
D
31442
SKD381840
SK38939
SW HYDR MOT
APPLICATION

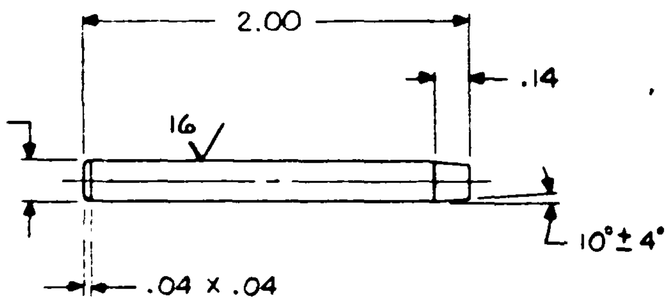

SECTION D-D
THIS SECTION IS TYPICAL FOR (2).
(4) IS THE SAME EXCEPT GROOVE IS ON OPPOSITE SIDE
2.07 DIA
1.37 DIA
13° TYP
26° TYP
40° TYP
20° TYP
.18 TYP
GROOVE .08 X .03 TYP
R TYP
SECTION C-C
THIS SECTION IS TYPICAL FOR (1) & (5)
EXCEPT DOWEL PIN HOLES WILL BE OPPOSITE FROM POSITION SHOWN
TAP PER MS33649-56 TYP 4 PLACES
1.02 TYP
.38 TYP
.25 TYP
.75
2.58 DIA
.002 ± .001
SEE DETAIL "X" TYP
.60 R TYP
1.12 R TYP
45° TYP
20° TYP
.05 TYP
R TYP
DETAIL "X"
.010 ± .005 R
.094 ± .002
.035 ± .002
.005 R TYP
0° TO 5° TYP
.180
-A-
D 31442
SKD381840
SCALE 2/1
SHEET 2


SEE CAM DETAIL
H
H
SECTION E-E
.6250
±.0002
-A-
32
32
.0002
.0002
2.504 +.000 -.025 DIA
TYP
32 OR BETTER
.0002
SEE DETAIL "X"
SHEET 2
TYP
SECTION H-H
22°30'
22°30'
15°
30°
.56 R
TYP
45°
30°
22°30'
5°
22°30'
45°
.950
±.001
TYP
1.900
±.001
TYP
2.100±.005
CAM DETAIL
D 31442 SKD381840

D-8

NOTES

1. MATERIAL: INCONEL, 718 HARD
   PROCURE MATERIAL AS INCONEL 718
   PLATE OR ROD PER AMS-5596, -5662, 5663
   ANNEALED CONDITION.
2. MACHINE TO NEAR FINAL DIMENSIONS
3. PARTS TO BE GIVEN AGING HEAT-TREATMENT
   PER ABOVE SPECS, GRIND TO FINAL DIMENSIONS.



SHEET REVISION STATUS

5	4	3	2	1	SHEET NO
					LATEST REV. LTR

REVISIONS

LTR	DESCRIPTION	DATE	APPROVAL
—	DRAWING RELEASED	1-3-79	

QTY REQD	CODE IDENT NUMBER	PART NO	SYM	DESCRIPTION	GOVT SPEC	SPEC	FIND NO
G01		SKB376737H01		2.00 OF .250 DIA	1		

SYM V - VENDOR ITEM B - BULK MATERIAL

PARTS LIST OR (SEE SEPARATE PARTS LIST)

NEXT ASSY	USED ON
SKD38 84C	SW HYDR MOT

APPLICATION

INTERPRET DWG IN ACCORDANCE WITH STDS PRESCRIBED BY MIL-D-1000

DWG TYPE (D)

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES. DO NOT SCALE

TOLERANCES

2 PLACE	3 PLACE	ANGLES
± .01	± ~	± ~

SPEC 90501 APPLIES

MATERIAL 1

CONTRACT NUMBER

ORIGINAL DATE OF DWG 18 DEC 78

DRAFTSMAN MILLER APPD

CHECKER Miller APPD

ENGINEER APPD

DESIGN ACTIVITY APPROVAL

Westinghouse Electric Corporation

Underseas Division
OCEAN RESEARCH AND ENGINEERING CENTER
ANNAPOLIS, MARYLAND 21404

PIN

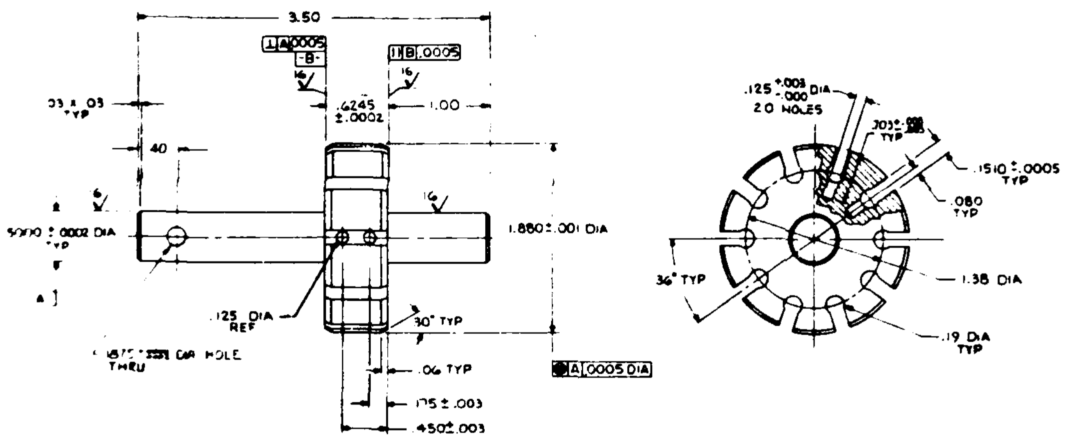
SIZE	CODE IDENT NO	DWG NO
B	31442	SKB376737

SCALE 2/1 WEIGHT ~ SHEET 1 OF 1

FORM NO. 06C3001F1

NOTES
1. MATERIAL: INCONEL 625, HOT-FINISHED OR ANNEALED
2. DEBURR ALL SHARP CORNERS.

REV	DESCRIPTION
–	DRAWING RELEASED
A	.312 ±.003 WAS .317 ±.003
B	ADDED ... .1875 DIA HOLE
C	PER D SPEC 379561-SUB A



SKD 381841H01	3.50 OF 2.000 DIA	
PART NO	DESCRIPTION	

PARTS LIST

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES DO NOT SCALE

ORIGINAL DATE OF DWG 2 AUG 78

SK 38 939 — SW HYDR MOT

APPLICATION

Westinghouse Electric Corporation

ROTOR

D 31442 SKD 381841

SCALE 2 SHEET 1 OF

SHEET REVISION STATUS					
5	4	3	2	1	SHEET NO.
					LATEST REV. LTR

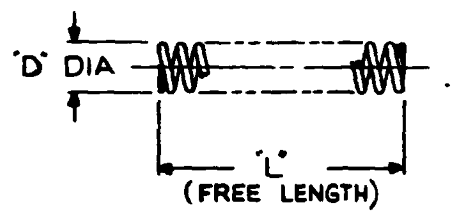
REVISIONS

LTR	DESCRIPTION	DATE	APPROVAL
-	DRAWING RELEASED	[illegible]	[illegible]
A	PER "D" SPEC 979581	[illegible]	[illegible]

NOTES

1. MATERIAL : ELGILOY WIRE, COLD WORKED AND SPRING HEAT TREATED FOR MAXIMUM FATIGUE STRENGTH
2. MATERIAL; 17-7PH STAINLESS STEEL WIRE CONDITION CH900
3. TYPE : COMPRESSION WITH ENDS SQUARED AND GROUND

SPRING	"L" FREE LENGTH	"D" DIAMETER	SOLID HEIGHT	NO. OF TURNS
H01	.62	.120	.290	16
H02 H04	.69	.120	.306	17
H03 H05	.75	.120	.342	19



WIRE DIA : .018 DIA

QTY REQD	CODE IDENT NUMBER	PART NO.	SYM	DESCRIPTION	GOVT SPEC	SPEC	FIND NO.
-		SKB376735H05		6.60 OF .018 DIA	1		5
-		SKB376735H04		5.90 OF .018 DIA	1		4
-		SKB376735H03		6.60 OF .018 DIA	2		3
-		SKB376735H02		5.70 OF .018 DIA	2		2
-		SKB376735H01		5.50 OF .018 DIA	1		1
G01							

SYM: V - VENDOR ITEM, B - BULK MATERIAL

PARTS LIST OR (SEE SEPARATE PARTS LIST)

NEXT ASSY	USED ON
SK361839	SW HYDR MOT

APPLICATION

INTERPRET DWG IN ACCORDANCE WITH STDS. PRESCRIBED BY MIL-D-1000.

DWG TYPE (DI)

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES. DO NOT SCALE.

TOLERANCES

2 PLACE	3 PLACE	ANGLES
±.02	±.004	± ~

SPEC 90901 APPLIES

MATERIAL: 1

CONTRACT NUMBER

ORIGINAL DATE OF DWG: 18 DEC 78

DRAFTSMAN: MILLER

CHECKER: Miller

ENGINEER

APPD

DESIGN ACTIVITY APPROVAL

Westinghouse Electric Corporation

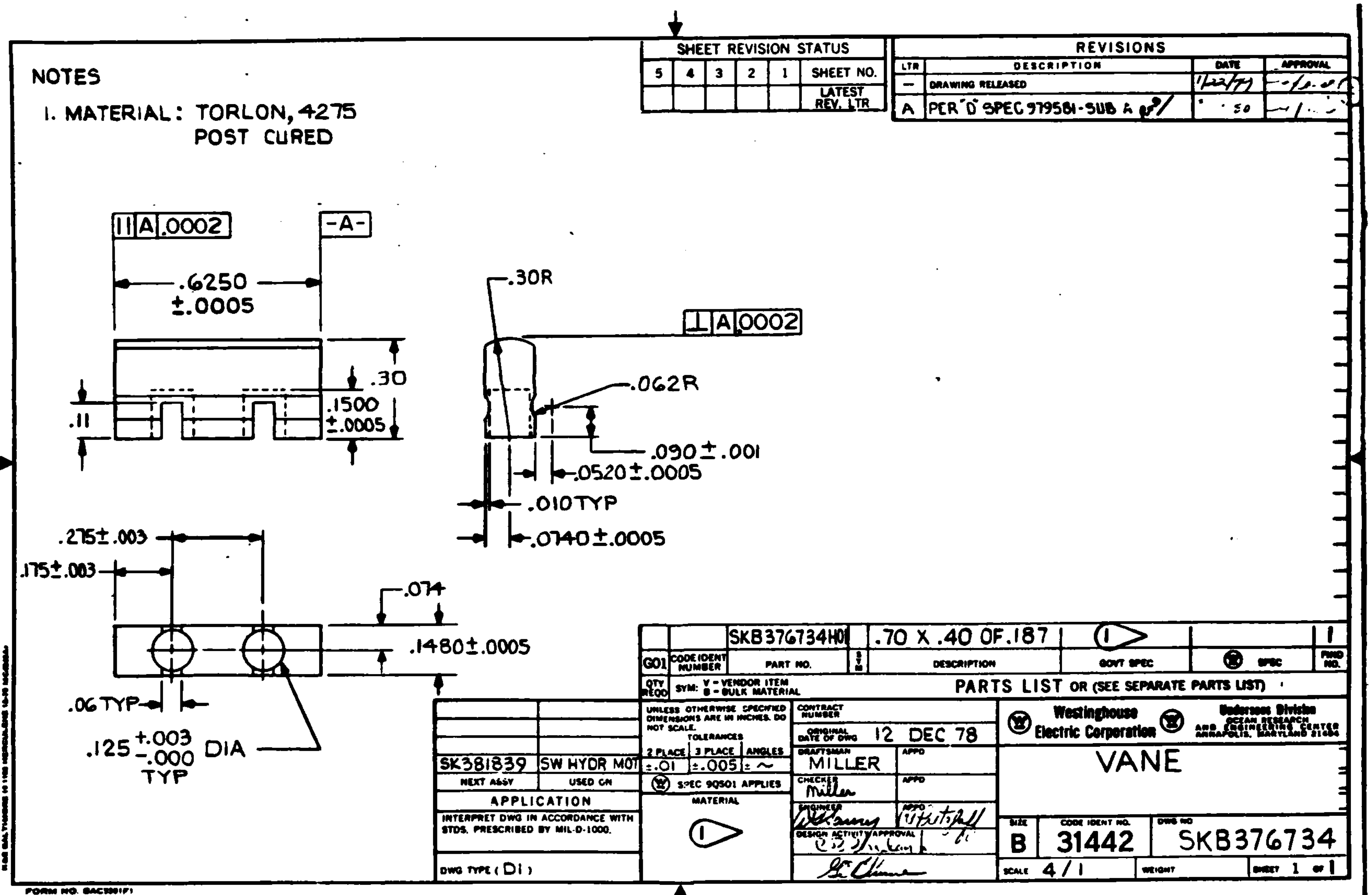
Undersea Division, Ocean Research and Engineering Center, Annapolis, Maryland 21404

SPRING

SIZE	CODE IDENT NO.	DWG NO.
B	31442	SKB376735

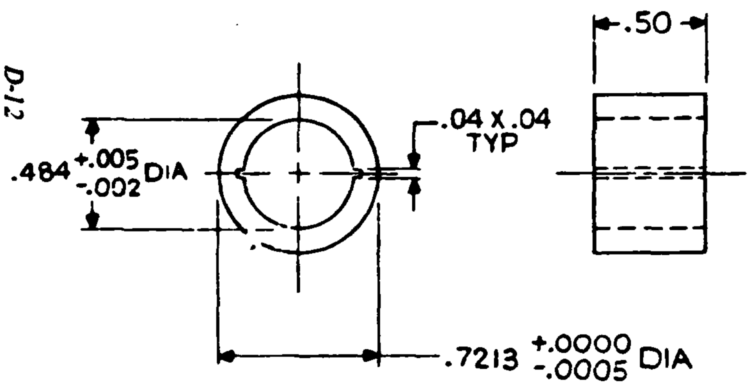
SCALE 4/1 WEIGHT SHEET 1 OF 1

FORM NO. BAC2301F1


NOTES
1. MATERIAL: TORLON, 4275 POST CURED
SHEET REVISION STATUS
5 4 3 2 1 SHEET NO.
LATEST REV. LTR
REVISIONS
LTR DESCRIPTION DATE APPROVAL
— DRAWING RELEASED
A PER "D" SPEC 979581-SUB A
.6250 ±.0005
-A-
.30
.1500 ±.0005
.11
.30R
.062R
.090 ±.001
.0520 ±.0005
.010 TYP
.0740 ±.0005
.275±.003
.175±.003
.074
.1480±.0005
.06 TYP
.125 +.003 -.000 DIA TYP
SKB376734H01
.70 X .40 OF .187
G01
CODE IDENT NUMBER
PART NO.
DESCRIPTION
GOVT SPEC
SPEC
FIND NO.
QTY REQD
SYM: V - VENDOR ITEM B - BULK MATERIAL
PARTS LIST OR (SEE SEPARATE PARTS LIST)
UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES. DO NOT SCALE.
TOLERANCES
2 PLACE ±.01
3 PLACE ±.005
ANGLES ± ~
SPEC 90501 APPLIES
MATERIAL
SK381839
SW HYDR MOT
NEXT ASSY
USED ON
APPLICATION
INTERPRET DWG IN ACCORDANCE WITH STDS. PRESCRIBED BY MIL-D-1000.
DWG TYPE (D1)
CONTRACT NUMBER
ORIGINAL DATE OF DWG 12 DEC 78
DRAFTSMAN MILLER
CHECKER Miller
ENGINEER
DESIGN ACTIVITY APPROVAL
APPD
Westinghouse Electric Corporation
Undersea Division
OCEAN RESEARCH AND ENGINEERING CENTER
ANNAPOLIS, MARYLAND 21404
VANE
SIZE B
CODE IDENT NO. 31442
DWG NO SKB376734
SCALE 4/1
WEIGHT
SHEET 1 OF 1
FORM NO. 8AC5001F1

NOTES

1. MATERIAL: TORLON, 4301
   POST CURED



D-12

SHEET REVISION STATUS					
5	4	3	2	1	SHEET NO.
					LATEST REV. LTR

REVISIONS			
LTR	DESCRIPTION	DATE	APPROVAL
—	DRAWING RELEASED	1/ /79	

QTY REQD	CODE IDENT NUMBER	PART NO.	SYM	DESCRIPTION	GOVT SPEC	SPEC	FIND NO.
G01		SKB376736H01		.50 OF .750 DIA	1		1

SYM: V - VENDOR ITEM, B - BULK MATERIAL

PARTS LIST OR (SEE SEPARATE PARTS LIST)

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES. DO NOT SCALE.

TOLERANCES		
2 PLACE	3 PLACE	ANGLES
± .01	± ~	± ~

SPEC 90501 APPLIES

MATERIAL: 1

NEXT ASSY	USED ON
SK381839	SW HYDR MOT

APPLICATION

INTERPRET DWG IN ACCORDANCE WITH STDS PRESCRIBED BY MIL-D-1000.

DWG TYPE (D1)

CONTRACT NUMBER

ORIGINAL DATE OF DWG: 7 DEC 78

DRAFTSMAN: MILLER

CHECKER: Miller

ENGINEER

DESIGN ACTIVITY APPROVAL

APPD

Westinghouse Electric Corporation — Undersea Division, Ocean Research and Engineering Center, Annapolis, Maryland 21404

BEARING

SIZE	CODE IDENT NO.	DWG NO
B	31442	SKB376736

SCALE 2/1 | WEIGHT | SHEET 1 OF 1

APPENDIX E

Test Plans and Setup

- Initial Test Approach
- Test Specification
- Inspection Data Sheets
- Test Setup Detailed Description

Initial Test Approach

The number of materials selected for analysis offers many potential combinations for testing. From the list of candidate materials, we selected for fabrication, assembly, and test two baseline configurations made up of the material combinations judged to offer the highest probability of success. Materials for three alternates to the baseline configurations are on hand, but schedule and funding precluded fabrication and testing of these alternates. Table E-1 presents the list of test configurations and also includes two configurations (4 and 5) for which material is not on hand.

The basic procedure used for testing the baseline configurations is outlined on the following pages. The procedure was optimistically based on few problems occurring during tests and was intended only as a basic guideline. The procedure was modified as required to accommodate problems, solutions, limitations of motor operation, and program cost and schedule. Synthetic seawater was used for the tests.

Test Procedure Outline

1. Performance test of Inconel 625 ring track and rotor with Torlon 4301 (or 4275) vanes, side plates, and bushing.

a. Record critical as-built dimensions.
b. Test for no load speed and pressure flow for speeds increasing to 1500 rpm then decreasing to 0 rpm. Repeat several times until variations in curves are negligible.
c. Test for speed and flow vs. torque at motor supply pressures of 50, 100, 250, 500, 750, 1000, 1250, and 1500 psig. Obtain increasing and decreasing torque curves for each supply pressure. Repeat several times until variations in curves are negligible.
d. Diassemble and inspect parts for wear patterns and dimensions.

2. Performance test of Inconel 625 and Torlon 4301 (4275) model with new vanes and side plates, and large V-grooves in the side plates.

a. File V-grooves into the rotor side of the side plates at each end of the outer slots. V-grooves are tetrahedron in shape and are to be about 0.15 inch long, 0.07 inch deep, and 0.10 inch wide at the slot.
b. Same as in Item 1 (a through d).

3. Performance test of Inconel 625 and Torlon 4301 (4275) model with new vanes and side plates, and small V-grooves in the side plates.

a. File V-grooves in the side plates as indicated by results of performance tests described in Items 1 and 2.
b. Same as in Item 1 (a through d).

Table E-1. Test Configurations

CATEGORY	CONFIGURATION	RING	ROTOR AND SHAFT	END PIECES	VANE	SIDE PLATES	BEARING
Configurations to be tested	Baseline 1	In625	In625	In625	T-4301	T-4301	T-4301
	Baseline 2	In625	In625	In625	M-271	M-271	M-271
Configurations not presently planned for testing	Alternate 1	In625(H)	In625	In625	M-271	M-271	M-271
	Alternate 2	In625	In625	In625	Ryton	Ryton	Ryton
	Alternate 3	In625	In625	In625	Vespel	Vespel	Vespel
Material on Hand							
Configurations not presently planned for testing	Alternate 4	MP35N	MP35N	In625	Best of above materials by testing		
	Alternate 5	Ti(H)	Ti(H)	Ti	M-271	M-271	M-271
Material not on Hand							

NOTES: 1. For Baseline 1, T-4301 to be replaced by T-4275 if material is available within schedule constraints.

2. In625 = Inconel 625
T-4301 = Torlon 4301
T-4275 = Torlon 4275
M-271 = Metcar 271 Bronze Graphite from Metallized Carbon Corp.
Ryton = Ryton R4 Plastic Extrusion
Vespel = Vespel SP1 Isostatically pressed
In625 (H)= Inconel 625 with surface hardened by Stellite weld overlay
MP35N = MP35N
Ti = Titanium 6Al-4V alloy
Ti (H) = Titanium 6Al-4V with surface hardened by conversion coat

3. Alternate 1 will become Baseline 2 if hard surfacing process can be implemented without schedule delay. The hardening is recommended to preclude chance for the hard Metcar 271 to score the Inconel 625 surfaces.

4. Life test of Inconel 625 and Torlon 4301 (4275) model with the best performance.

a. Use parts from the previous model with the best performance.

b. Test for speed and torque vs. time with supply pressure set at 1500 psi and flow varied to maintain 1500 rpm. Continue test for 50 hours or until speed decreases by 50%.

c. Disassemble motor and inspect parts for wear patterns and dimensions.

5. Performance test of Inconel 625 and Metcar M-271 model and small V-grooves in the side plates with the same dimensions as in Item 3.

a. Same as in Item 3 (a through d)

6. Life test of Inconel 625 and Metcar M-271 model of Item 5.

a. Same as in Item 4 (a through c).

Inspection Data Sheet Test Motor Wear

The dimensions taken for determining wear patterns for each piece will be the same as those shown on the test motor as-built dimensions data sheets. Several readings for each parameter will be made to show variances in each surface. Additionally, photographs will be taken to show the wear patterns.

Test Setup Detailed Description

The test setup is designed to provide filterd, artificial seawater under pressure to the test motor with controls, instrumentation, and a method of providing a load to the shaft of the motor necessary to accomplish the testing as defined in Westinghouse Test Specification SKA 926551.

A schematic diagram of the test setup is shown in Figure E-1. Table E-2 is the list of components in the test setup keyed to the circled numbers on the schematic. Figure E-2 is a photograph of the setup during testing.

The artificial seawater, mixed with tap water to a specific gravity of 1.025, as per ASTM D-1141-52, is pumped from the reservoir through 10 micron nominal filters and the low pressure flowmeter to the test motor by the motor-driven, 3-piston pump. The low pressure plumbing to the pump is 3/4-inch, schedule 40, plastic pipe. The pump has a constant flow output of about 6.2 gpm as long as the pressure set by the adjustable pressure regulator on the pump is not exceeded and flow bypasses through the regulator. Maximum output pressure of the pump is 2000 psig.

The seawater is pumped through the surge damper, in-line flowmeter, temperature probe, and manifold to the test motor. The high pressure plumbing is 3/4-inch stainless steel tubing up to the manifold. At the manifold, the flow is split into four, 3/8-inch equal length tubes connected directly to the inlet ports of the test motor. The in-line flowmeter is connected to a Gould Brush 2400 Recorder. The four outlet ports of the test motor have short, 3/8-inch tubes that exhaust directly to the reservoir.

Table E-2. Test Setup Parts List

1.	Synthetic seawater	ASTM D-1141-52, Formula A
2.		
2.	Seawater reservoir	Approximately 800 gallons
3.	Flowmeter	Schulte and Koerting Co. 0-10 gpm
4.	Seawater pump	Gail Ind. Model 10,000, Series II 0-2000 psig, 6.3 gpm
5.	GN_2 source	
6.	Pulsation dampener	Westinghouse Air Brake Co. Fluid Power Div. No. PC: 162.3002
7.	Seawater sample valve	
8.	In-Line flowmeter	National Tank Co., Instruments, Inc. Division Type W3/0750
9.	Temperature probe	Omega Engineering Inc. 0-110-UUA-35J3
10.	Manifold	
11.	Test Motor	
12.	Spectrum analyzer	
13.	Plotter	
14.	Accelerometer	
15.	Strobe	
16.	Hydraulic pump (test motor load)	Abex Corp., Dennison Division TMB004-21R-4
17.	Force transducer	Lebow Assoc., Inc., Model 3397-200 0-200 lbf
18.	Hydraulic fluid reservior	
19.	Heat Exchanger	30 ft of 5/8 in. OD, 0.06-in. W copper tubing

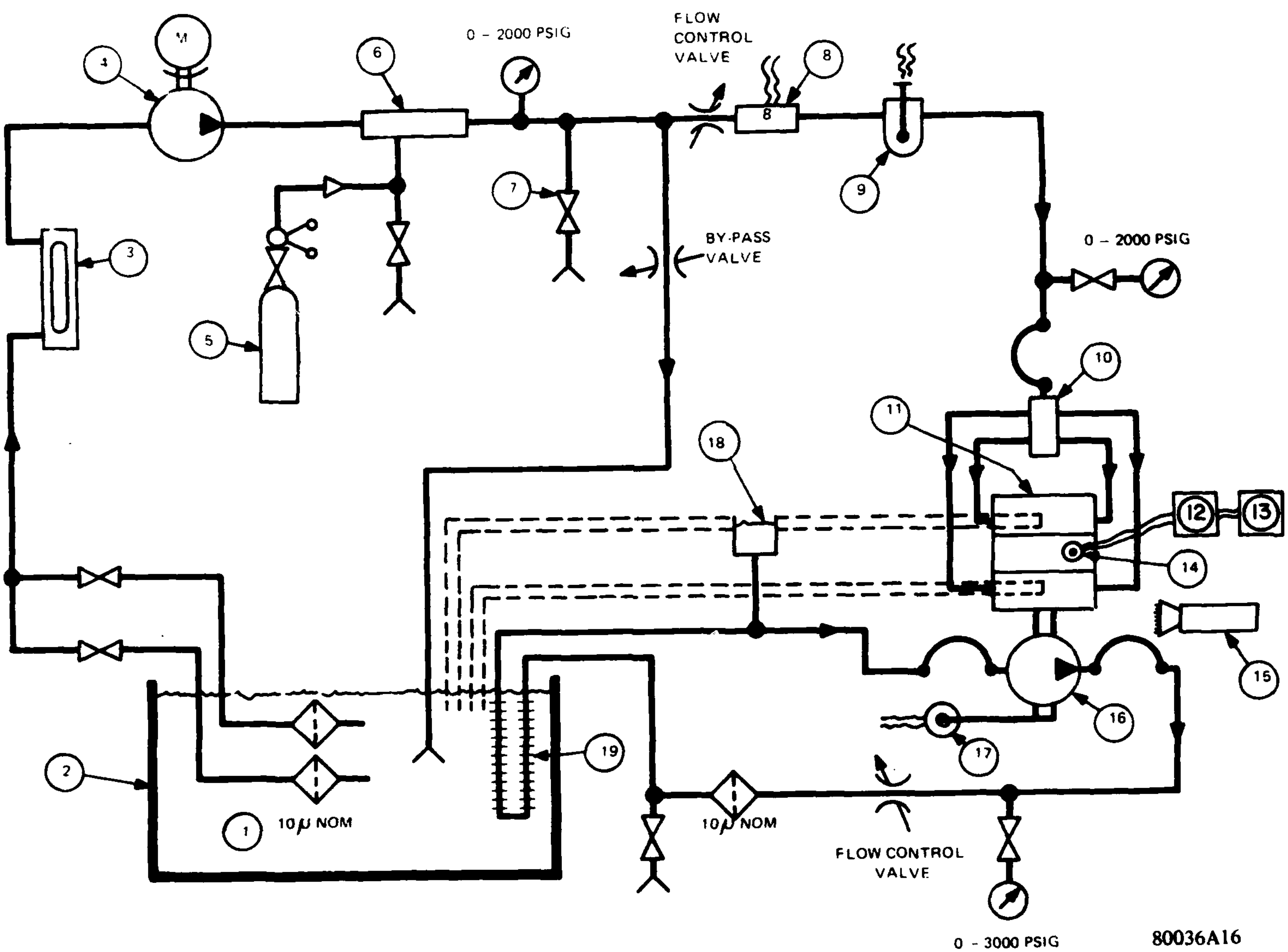
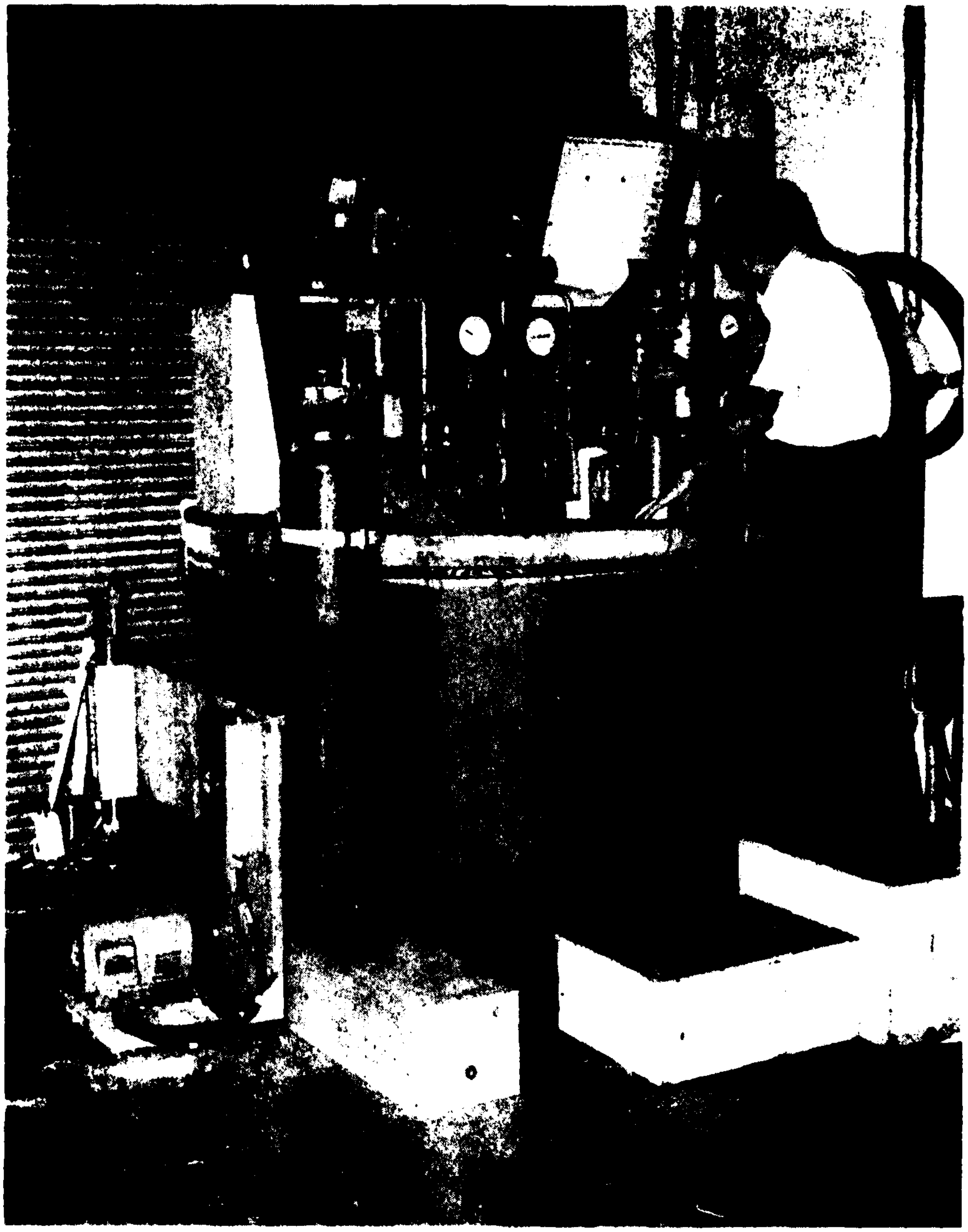


Figure E-1. Seawater Motor Test Setup



80036A17

Figure E-2. Test Setup During Testing

The test motor is mounted rigidly in the test fixture that also holds the Dennison Pump (load) and force transducer. The output shaft of the test motor is coupled to the shaft of the load pump by a short shaft with Magnaloy couplings at each end. The load pump is held in the fixture by a bracket that is supported in the main fixture by ball bearings. These bearings allow freedom of motion about the pump shaft axis to allow output torque measurement.

A rigid arm connected to the bracket holding the load pump rests on the force transducer which is connected to the Gould Brush 2400 Recorder and used to measure output torque of the test motor. The force transducer was calibrated statically with known weights.

The Dennison Pump (load pump) has its own hydraulic circuit which consists of a flow control valve, filter, heat exchanger, and reservoir. A flow control valve on the output from the pump is used to throttle the output flow to produce the torque load.

APPENDIX F

Parametric Test Analysis

- Introduction
- Design
- Test Plan
- Test Results
- Conclusions

INTRODUCTION

At the conclusion of the Phase I development program for the seawater hydraulic motor, Westinghouse proposed a Phase II program to improve the motor design. The goal of this program was to improve motor performance and motor life through a parametric test program. Parameters to be varied were vane taper, spring length, flow rate, and pressure. Two different spring lengths, both increased from the previous spring length, and four vane tapers were planned for the test program. The program called for a total of eight performance tests; each vane taper would be tested with both spring lengths. For each performance test, maximum pressures and flows would be varied. The program would conclude with a 50-hour life test for the most promising candidate.

In addition to the parametric test program, other changes were also planned. These included polishing the ring track and lateral rotor surface, deepening the rotor spring holes to accommodate the longer springs, and modifying the test setup to increase flow to the test motor.

The Phase II program was successfully accomplished. Funding and schedule constraints limited this program, but results were favorable: increased motor performance and increased life.

DESIGN MODIFICATIONS

Design changes in the seawater motor components included modifications to the springs, rotor, and vanes plus polishing of the ring track. Details are as follows:

1. Springs and Rotor — The springs were lengthened and the rotor spring holes deepened to increase minimum spring force and decrease maximum stress. Two spring lengths, 0.69 and 0.75 inch, were selected to test effects of spring length on motor performance. Spring and rotor drawings are shown in Figures F-1 and F-2.

2. Vanes — Stress concentration on the vanes caused problems in the previous stages of motor development. To relieve the stress concentration, the squared side grooves were changed to rounded side grooves as shown on the vane drawing (see Figure F-3), and the spring holes were deepened past the side groove so that the stress concentration at the hole corner occurred in the thicker material. The vanes were to be modified by changing taper on the leading side of the vane, then tested to check effects on performance. Tapers were chosen to be 0, 0.002, 0.004, and 0.006 inches (see Figure F-4) — a range of tapers believed to be sufficient to determine an optimum taper based on performance tests.

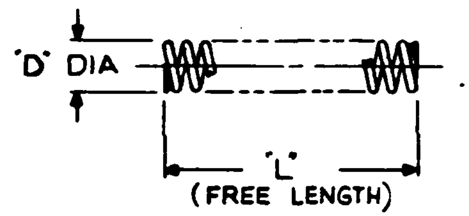
3. Ring Track — The ring track was smoothed with 600-grit emery paper using diesel oil as a lubricant, then polished with semi-chrome polish to decrease friction between the vanes and ring track.

SHEET REVISION STATUS					
5	4	3	2	1	SHEET NO.
					LATEST REV. LTR

REVISIONS			
LTR	DESCRIPTION	DATE	APPROVAL
—	DRAWING RELEASED	[illegible]	[illegible]
A	PER "D" SPEC 979581	[illegible]	[illegible]

NOTES

1. MATERIAL: ELGILOY WIRE, COLD WORKED AND SPRING HEAT TREATED FOR MAXIMUM FATIGUE STRENGTH.
2. MATERIAL: 17-7PH STAINLESS STEEL WIRE CONDITION CH900
3. TYPE: COMPRESSION WITH ENDS SQUARED AND GROUND



WIRE DIA : .018 DIA

SPRING	"L" FREE LENGTH	"D" DIAMETER	SOLID HEIGHT	NO. OF TURNS
H01	.62	.120	.290	16
H02 H04	.69	.120	.306	17
H03 H05	.75	.120	.342	19

QTY REQD	CODE IDENT NUMBER	PART NO.	SYM	DESCRIPTION	GOVT SPEC	SPEC	FIND NO.
—		SKB376735H05		6.60 OF .018 DIA	1		5
—		SKB376735H04		5.90 OF .018 DIA	1		4
—		SKB376735H03		6.60 OF .018 DIA	2		3
—		SKB376735H02		5.70 OF .018 DIA	2		2
—		SKB376735H01		5.50 OF .018 DIA	1		1

G01

SYM: V = VENDOR ITEM, B = BULK MATERIAL

PARTS LIST OR (SEE SEPARATE PARTS LIST)

NEXT ASSY	USED ON
SK381839	SW HYDR MOT

APPLICATION

INTERPRET DWG IN ACCORDANCE WITH STDS. PRESCRIBED BY MIL-D-1000.

DWG TYPE (D1)

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES. DO NOT SCALE.

TOLERANCES		
2 PLACE	3 PLACE	ANGLES
±.02	±.004	± ~

SPEC 90901 APPLIES

MATERIAL: 1

CONTRACT NUMBER

ORIGINAL DATE OF DWG: 18 DEC 78

DRAFTSMAN: MILLER — APPD

CHECKER: Miller — APPD

ENGINEER — APPD

DESIGN ACTIVITY APPROVAL

Westinghouse Electric Corporation — Underseas Division, Ocean Research and Engineering Center, Annapolis, Maryland 21404

SPRING

SIZE	CODE IDENT NO.	DWG NO.
B	31442	SKB376735

SCALE 4/1 — WEIGHT — SHEET 1 OF 1

FORM NO. BAC3001F1

Figure F-1. Lenghthened Springs

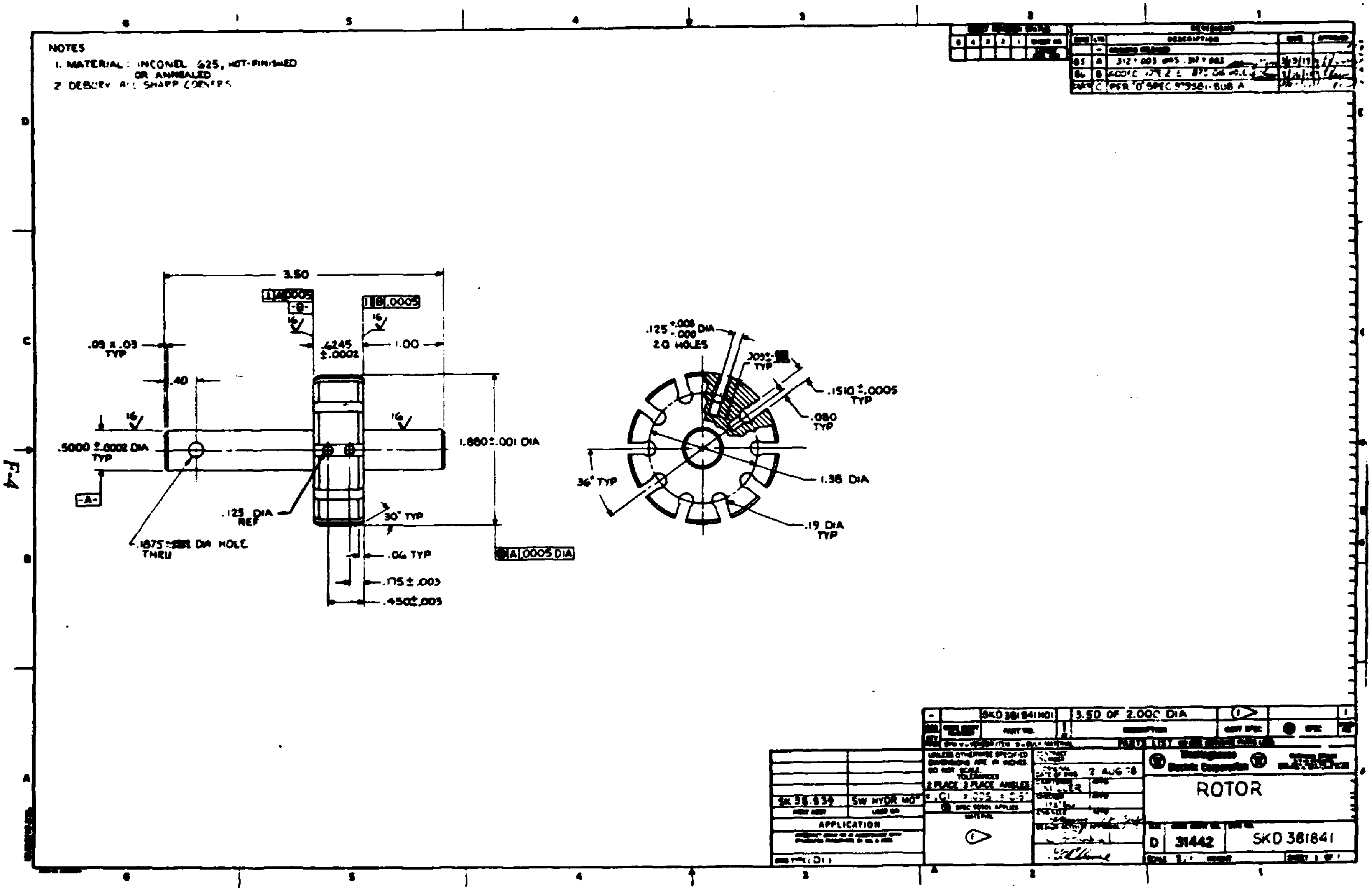


Figure F-2. Modified Rotor

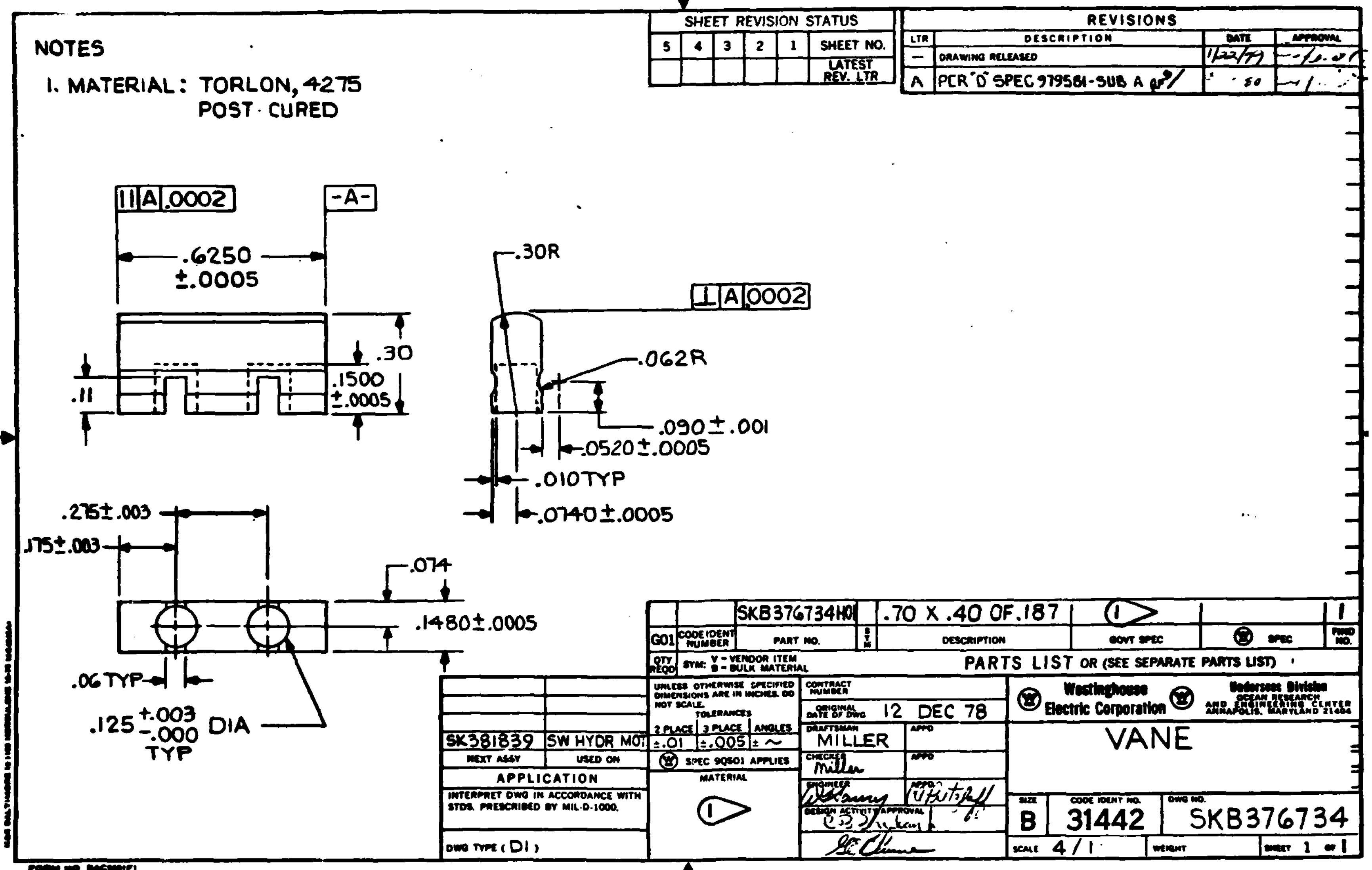


Figure F-3. Seawater Motor Vanes

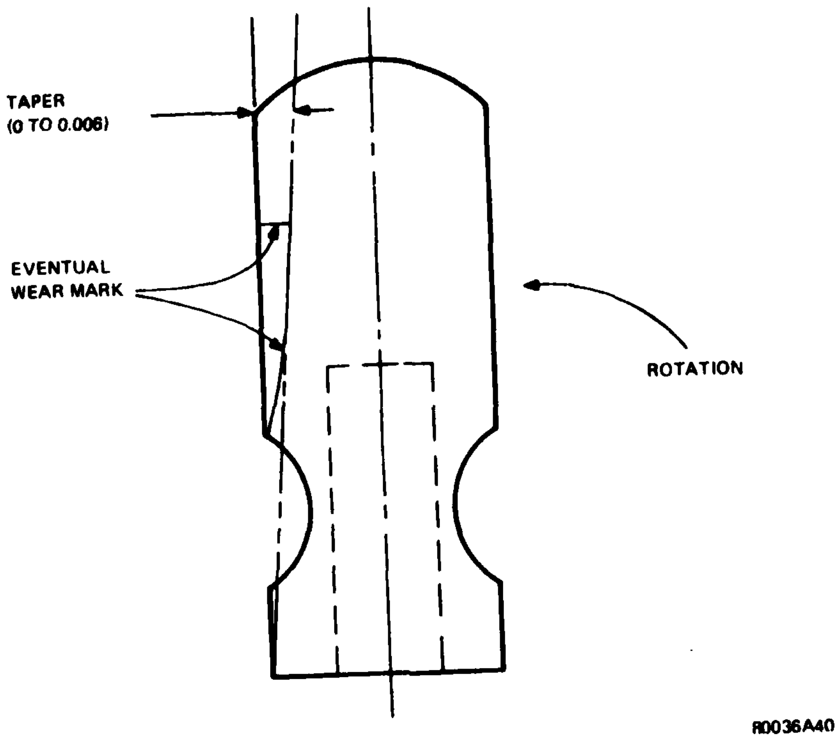


Figure F-4. Vane Taper

TEST PLAN

For this stage of the seawater motor development, performance tests with varying parameters and 50-hour life test were to be performed. Two different spring lengths (both increased from the previous length) and the four vane tapers were to be among the varying parameters for the performance tests, along with flow rate and inlet supply pressure to the motor. A performance test consists of varying the flow to the motor, while at each flow changing the inlet supply pressure from a minimum to a maximum. The minimum supply pressure is the zero flow condition and the maximum pressure is the point where vane is instable, "knocking", would occur.

Eight performance tests were planned for motor optimization, each vane taper tested with both spring lengths. These results would indicate the appropriate vane taper and spring length for motor optimization. The 50-hour life test would be run with those parameters at a flow rate and pressure delivering approximately 3 horsepower and 70% efficiency. Detailed measurements of sideplates and vanes would be taken before and after the 50-hour test.

TEST RESULTS

The results of the first performance tests indicated that the long springs gave higher output power and performance than the shorter springs with the zero taper vanes. Long springs produced 3.40 horsepower with 69% efficiency running at 7 gpm and 1200 psi inlet supply pressure, while short springs produced 2.52 hp with 70% efficiency at 6.9 gpm and 900 psi. Insufficient pump pressure caused maximum flow to be below the desired 7 gpm. Higher motor pressures were prevented by "knocking", which occurs when the spring force holding the vanes against the ring track is not sufficient to compensate for the water pressure pushing the vanes back into the rotor slots.

The second set of performance tests were done with the 0.002 inch taper on the leading vane side. One vane was tapered the wrong way, i.e., narrow at the base and wide at the tip. These results yielded a poorer performance for the motor. In this case the long springs produced 1.59 hp with 61% efficiency at 5 gpm and 900 psi, while the short springs produced 1.13 hp with 65% efficiency at 5 gpm and 600 psi. Higher pressures could not be reached due to knocking.

After each performance test, the motor was disassembled to check vane and spring conditions. No unexpected wear patterns occurred. However, in each case there were a number of broken springs. Previous testing of the seawater motor resulted in Elgiloy springs' breaking after six hours running time. For this phase, Elgiloy springs were not available, therefore 17-7PH stainless steel springs were used. The mechanical properties of these two materials vary only slightly, with the Elgiloy being more corrosive resistant and having a slightly higher strength. Springs were expected to break somewhat sooner with the 17-7PH material substituting for Elgiloy; however, the performance tests with 17-7PH springs resulted in broken springs after only 20 minutes running time. An investigation into the unusually short spring life uncovered a machining error: the spring holes in the rotor were not aligned with the holes in the vanes. Remachining of the rotor holes corrected the error.

Funding and schedule limitations following the rotor rework resulted in curtailment of the additional vane-taper parametric tests. We moved directly to the 50-hour test preparation. Based on previous test results, we selected the long spring, zero vane taper configuration for this test.

Before the 50-hour test, several performance tests were run to check the motor's output. Three horsepower was attained at 74% efficiency from 7 gpm and 1000 psi. The goal for the life test was to run the motor for 50 hours at 3 horsepower; the flow and the pressure, therefore, were held to 7 gpm and 1000 psi respectively.

The 50-hour test was run over a ten-day period (with shutdowns each night) from February 29 through March 9. The first day allowed for only three hours running time due to spring force problems and frozen water in two short sections of the umbilical hose which were external to the plant. Investigation showed that the spring holes in the rotor were drilled deeper than planned causing the long springs to be too short for high pressures. Glass beads of 0.12 in. diameter were placed behind each spring to increase spring force. The motor was idle over the weekend, but run for 5-1/2 hours Monday, March 3. Again below freezing temperatures froze the water in the hose causing delays for thawing. Tuesday, March 4 the motor ran for 8 hours, 8-1/2 on Wednesday, and 10 hours on Thursday and Friday, completing the 50-hour test with 5 hours on Sunday, March 9. Motor performance for this 50-hour life test is shown in Figures F-5 through F-7.

Observations during and after the 50 hour test may explain part of the decrease in performance:

1. Supply water temperature varied from approximately 50° F to a high of 109° F. Operating at high water temperature can cause decreases in performance and life. The water viscosity is lower at higher temperatures, resulting in more leakage friction and vane wear. Also axial thermal expansion of vanes causes larger friction loss and larger gaps between the rotor and side plates.

2. After several hours of running, there was an increase in water leakage around the bearings. This could be caused by the O-ring seals' flattening and not filling the groove.

3. Measurements taken after disassembly indicated that the vanes had absorbed moisture due to the high temperature water. The vane swelling would cause the side plates to be pushed away from the ring track seal, also allowing higher leakage through the bearings. The swelled vanes also explains the increased tightness of the shaft noted near the end of the test.

4. Further vane examination pointed out the nonuniform wear of the Torlon. The vanes with a uniform graphite content were more wear resistant than those vanes with marbled pattern of graphite.

The motor ran for 50 hours on one set of 17-7 PH stainless steel springs, surpassing the calculated spring life. Spring hole alignment and fresh water contributed to the prolonged spring life. One spring was discovered broken at disassembly. It was believed to have broken at that time since no knocking occurred while running and because it occurred cleanly in a highly worn area. When a spring breaks while running, it becomes twisted in on itself. The broken spring from the life test was not twisted at all.

CONCLUSIONS

The purpose of the parametric test analysis was to test design changes for motor optimizaton and run it at those conditions producing 3 hp and 70% efficiency for 50 hours. These terms were essentially met with satisfactory results. No spring failure occurred while running after the spring holes were aligned. Moreover, new vane grooves producing lower stress concentration prevented vane failure. It is felt that the seawater motor worked harder than required due to the high water temperatures, indicating this to be a good life test.

For further efforts, longer springs are recommended, along with better quality control on the Torlon to attain more uniform grain. When improving the finish on the ring track, simply smoothing it with 600 grit emery paper and diesel oil is sufficient; polishing with semi chrome polish leaves particles embedded in the ring track. These eventually become embedded in the vanes.

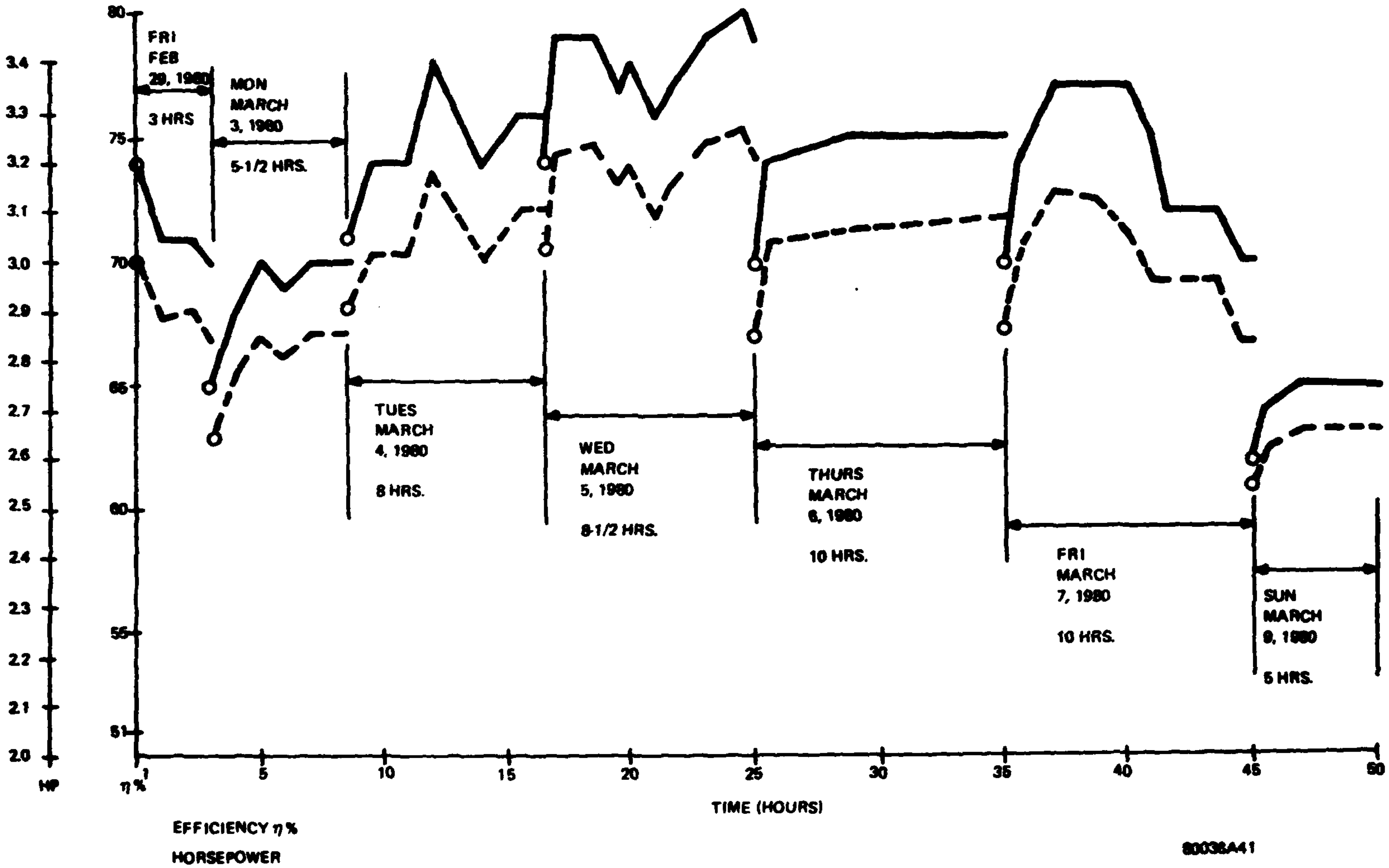


Figure F-5. Seawater Motor Horsepower and Efficiency 50-Hour Test

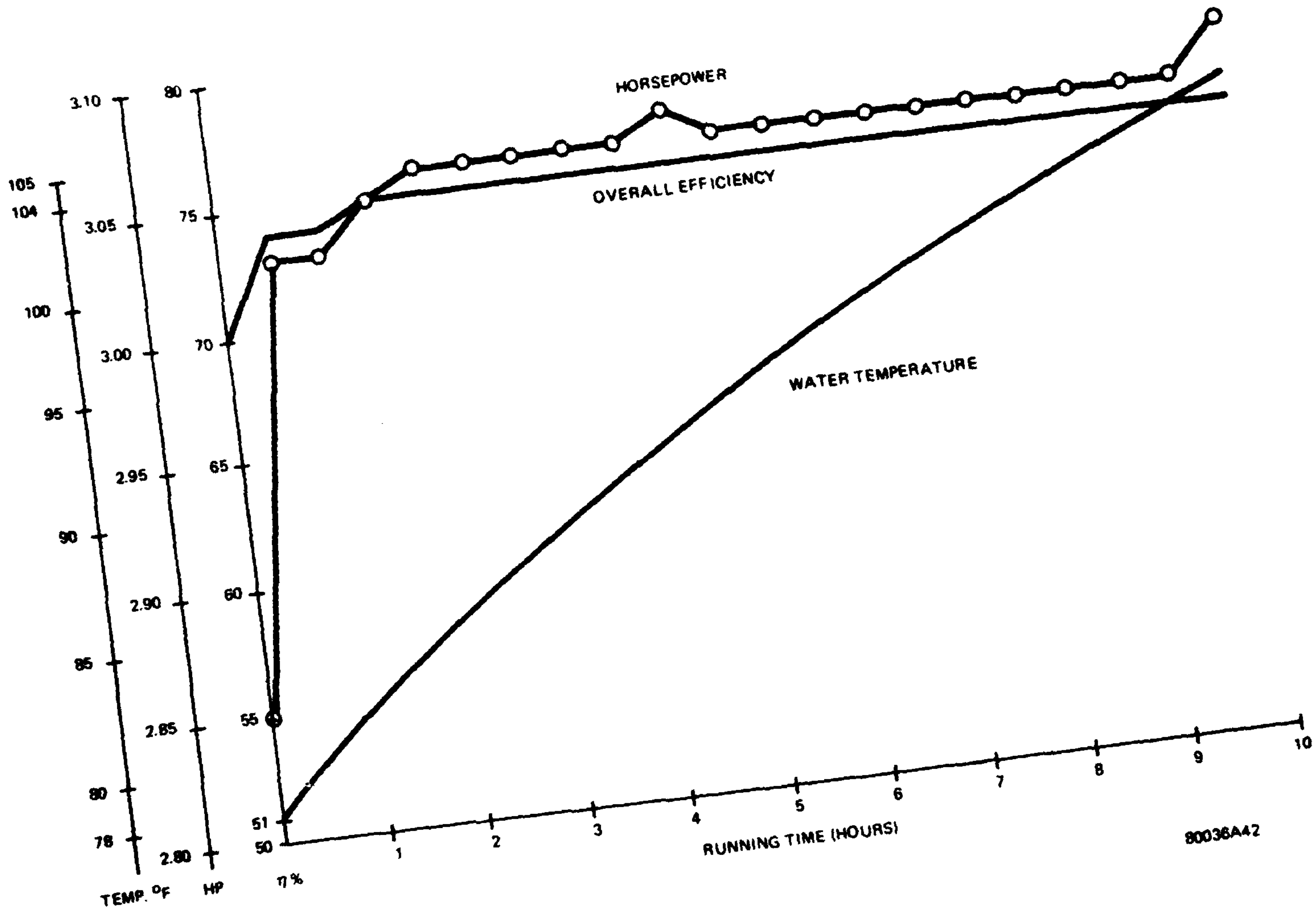


Figure F-6. 50-Hour Test (Hours 25 to 35) Thursday, March 6, 1980

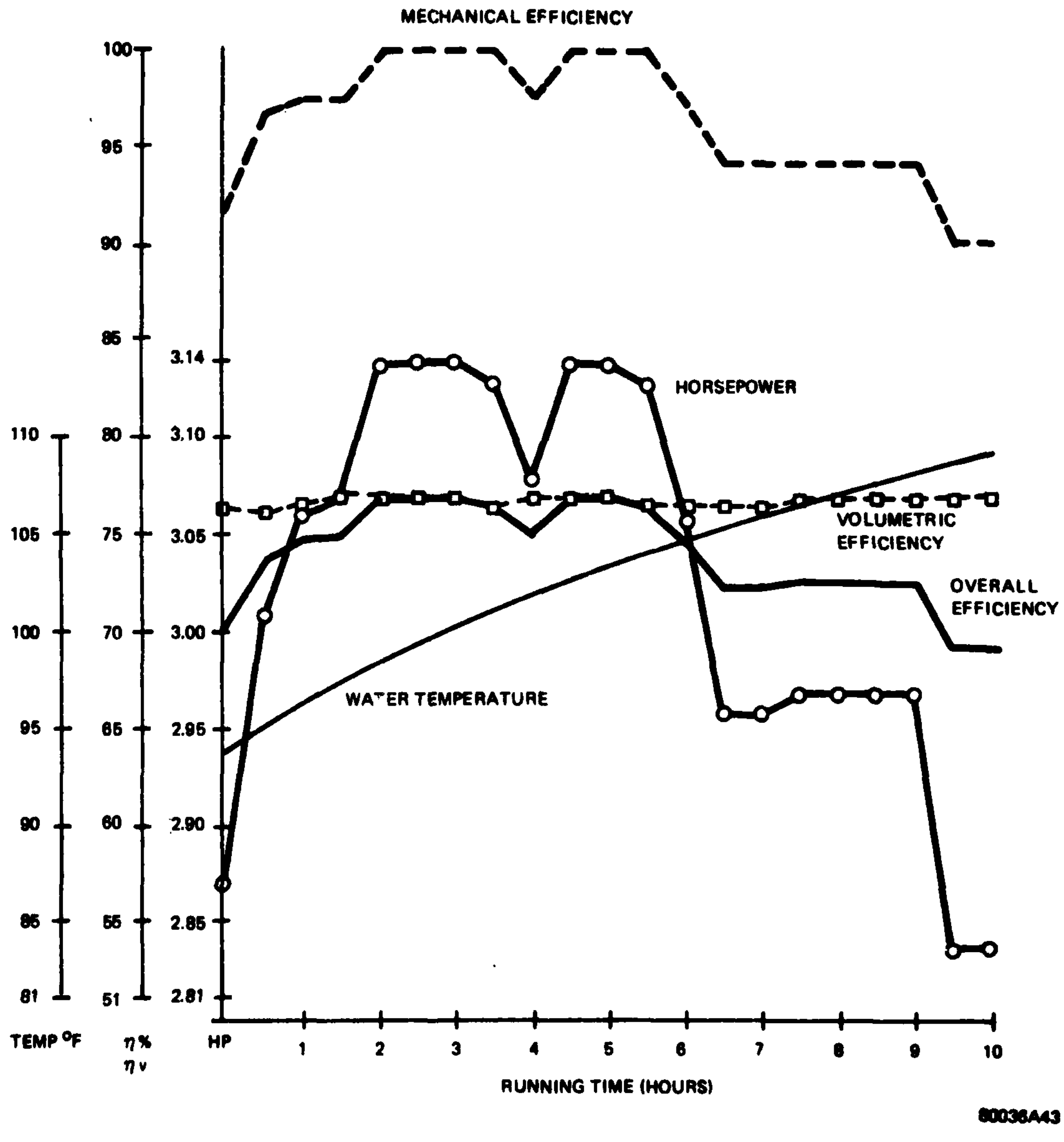


Figure F-7. 50-Hour Test (Hours 35 to 45) Friday, March 7, 1980