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DEVELOPMENT OF A SEAWATER HYDRAULIC TOOL SYSTEM.(U)
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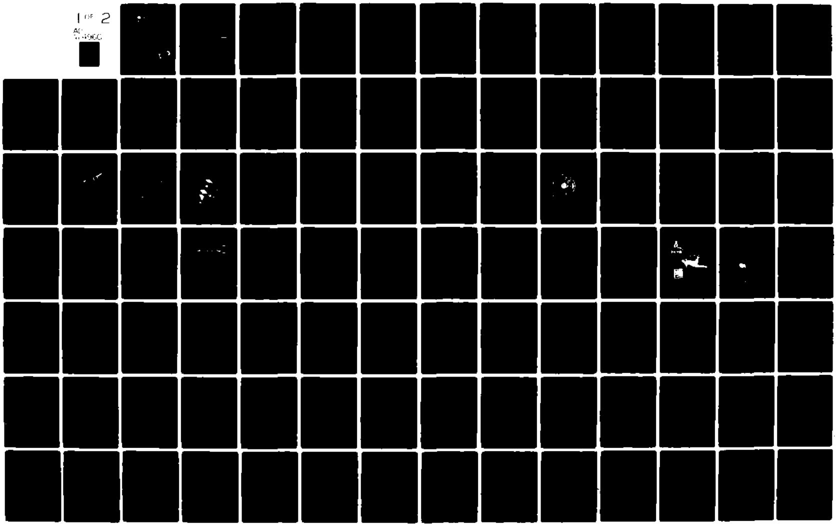

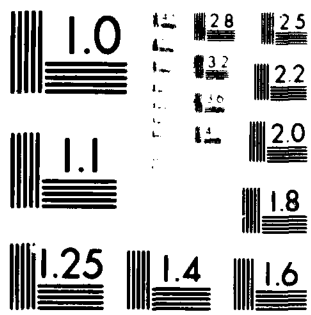

MICROCOPY RESOLUTION TEST CHART

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NAVAL CIVIL ENGINEERING LABORATORY
Port Hueneme, California

Sponsored by
NAVAL FACILITIES ENGINEERING COMMAND

AND
NAVAL SEA SYSTEMS COMMAND

DEVELOPMENT OF A SEAWATER HYDRAULIC TOOL SYSTEM

May 1982

An Investigation Conducted by
WESTINGHOUSE ELECTRIC CORP.
Oceanic Division
P. O. Box 1488
Annapolis, Maryland

N00123-80-C-1271

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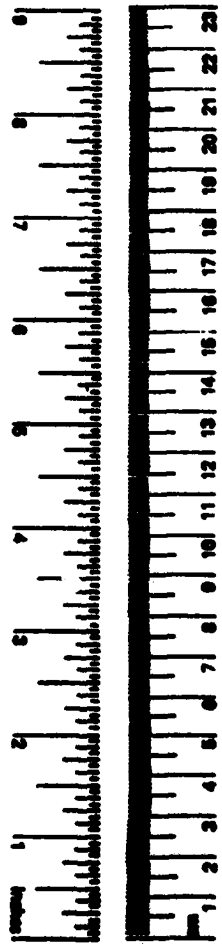
METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

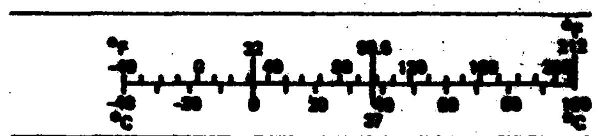
Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2,000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C

*1 in = 2.54 (exactly). For other exact conversions and more detailed tables, see NBS Misc. Publ. 286, Units of Weights and Measures, Price $2.25, SD Catalog No. C13.10:286.



Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1,000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



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Seawater; tools; diver equipment; motors; hydraulic equipment

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

The Naval Civil Engineering Laboratory is developing a diver-operated hydraulic tool system that uses seawater as the working fluid. The program was initiated in 1976 with the subsequent development of an experimental balanced vane by 1980. Late in 1980 a contract was awarded to improve the performance of the experimental vane motor and make it reversible; to couple

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SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered) the motor to a rotary impact mechanism with appropriate control to form a diver-operated impact wrench and to develop a diesel-operated portable power supply for operating the tool. This report describes the successful development of this experimental seawater hydraulic tool system.

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1.0 INTRODUCTION

The need for better and more reliable underwater tools has long been recognized. The introduction of oil hydraulic powered diver tools resulted in a marked improvement in diver work efficiency; however, oil hydraulic systems exhibit several disadvantages in the underwater environment. The working fluid is hazardous to men and the environment. Components built of materials quite suitable for land use ultimately corrode in underwater use. The necessity to return low pressure oil to a reservoir on the surface results in bulky, dual hose umbilicals between the surface and the diver. Tool systems designed specifically for underwater use are needed to optimize diver performance.

Use of seawater as a working fluid in diver tool systems is an attractive alternative to oil based systems. The working fluid is free and nonhazardous. Exhaust water vents directly to sea at the diver tool, reducing the diver umbilical from two hoses to one. The potential for improving the safety and efficiency of diver work is significant.

Under contract from the U.S. Naval Civil Engineering Laboratory, Westinghouse has developed a prototype seawater hydraulic diver tool system.

1.1 BACKGROUND

In April, 1980, Westinghouse delivered a seawater hydraulic vane motor to the U.S. Navy Civil Engineering Laboratory. This motor constituted the most significant single advance to date toward development of a complete seawater hydraulic tool system for deep sea divers. Based upon our experience in developing this 3.3 hp seawater vane motor, Westinghouse has now developed an engineering model diver impact wrench system to demonstrate the useful work capabilities of seawater hydraulics. This report addresses the development of that diver tool system.

1.2 SCOPE OF WORK

The objective of this effort was to develop and demonstrate a seawater hydraulic tool system capable of doing useful underwater work for an extended period of time. An important implied objective was the development of a system which makes a strong case for the practicality and desirability of continued seawater hydraulics development.

Development of the diver tool system consisted of three major tasks. The first was to improve the original seawater motor by extending its life, increasing its operating pressure, making it reversible, and making it compatible with an impact mechanism. The second was to design and fabricate an impact wrench assembly with capabilities similar to the oil hydraulic Stanley IW06 wrench currently used by Navy divers. The third was to develop a seawater hydraulic power source capable of providing dependable service for the impact wrench and other seawater hydraulic tools to be developed in the future. The power supply includes a 250-foot long umbilical to allow diver operation of the tool from the surface-operated power source.

Improved Motor

To improve the original seawater motor for use in a diver impact wrench, Westinghouse first conducted a comprehensive preliminary design and material analysis program. By using the existing motor as a starting point, it was possible to define the basic design component materials early in the program. Also, development tests performed by NCEL on the original motor made it possible to analyze improvement modifications without an extensive in-house testing program. Reversibility was accomplished by equalizing the pressure loaded areas of the side plates and rerouting bearing water supply. The Ingersoll-Rand Model 2910 impactor was selected and the motor/impactor interfaces were specified early in the program, allowing parallel development of the motor and wrench assembly. The most demanding goals were the extended life and increased operating pressure. Basically, spring life and wear were improved by geometric and material charges. Also, the spring force was increased to allow motor operation at higher pressures. A complete and detailed description of motor improvements is contained in Sections 2 and 3, with final drawings contained in Appendix H.

Impact Wrench Assembly

The entire wrench assembly consists of three major components: the impact mechanism, the handle (with valves), and the coupling.

The Ingersoll-Rand 2910 impactor was analyzed by Westinghouse and then tested at NCEL during the tool motor performance and evaluation tests. This was the impactor selected for the final design.

The handle is an off-the-shelf design by Fairmont Hydraulics, a division of Fairmont Railway Motors, Inc., in Fairmont, Minnesota. Modifications were incorporated to make the handle compatible with the seawater tool motor, the impact mechanism, and the seawater environment.

The coupling design proved to be the most difficult. At first it appeared that a flexible coupling would be required, but from tests performed at NCEL, we found that the seawater motor would drive the impactor directly. A design for a flexible coupling, which had been pursued in parallel, is contained in Appendix C.

A detailed description of the entire wrench assembly development and design can be found in Sections 4 and 5.

Seawater Hydraulic Power Source

The major tasks in the development of a high performance, flexible seawater power source for diver use was the design of suitable control valving, development of appropriate filtration, and packaging of the power source hardware on a trailer suitable for shipboard use. The pump/diesel engine/trailer assembly was specified and procured from a company specializing in pumping systems — Hydra Tech Pumps, Inc., Mt. Holly, N.J. The system used off-the-shelf components, changing materials as required for compatibility with seawater. A complete, detailed description of the SWHPS development and design is contained in Sections 6 and 7, with documentation in Appendix F.

1.3 SYSTEM SAFETY

The diver tool system was reviewed for safety as a prototype for experimental use. No safety problems were found for the impact wrench assembly. For the seawater hydraulic power source, one potential area of concern was noted, the reservoir vent filter. This filter must be kept clean to prevent possible overpressurization of the reservoir. Details of the safety review are included in Appendix G. Further analysis will be required before the system can be considered safe for fleet use.

1.4 REPORT ORGANIZATION

Section 1.2 has described the scope of the work that was required to be accomplished by this project. Sections 2 and 3 cover the seawater tool motor development and final design, respectively. Sections 4 and 5 describe the development and design of the impact mechanism, the coupling, and the wrench handle. The development and the design of the hydraulic power source are included in Sections 6 and 7. Section 8 covers the developmental and final tests and results of the seawater tool motor and the impactor. Also included in Section 8 are the test plans for the wrench assembly and hydraulic power source. Finally, Section 9 states the significant results of the whole program and makes recommendations for continuing projects.

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

2.0 MOTOR DEVELOPMENT

The major part of the Diver Tool System contract was to develop a reversible seawater hydraulic motor to provide useful and reliable motive power for a diver impact wrench. This section will discuss:

1. The major technical issues involved in that development
2. The two design approaches taken to meet the seawater tool motor requirements and goals
3. The developmental tradeoffs
4. The final design selection

2.1 MOTOR DESIGN REQUIREMENTS AND GOALS

There are three primary requirements for the seawater tool motor: It must be double entry and reversible; it must use seawater as the sole working fluid; and it must be capable of driving an impact mechanism similar to Ingersoll-Rand Model 2910 impactor.

The design goals for the motor included the following:

● Power	3 hp
● Efficiency	75%
● Minimum Operating Pressure	1200 psi
● Maximum Water Weight	7 pounds
● Service Life	250 hours
● Maximum Flow	7 gpm
● Maximum Volume	25 in.^{3}

The original (previous contract) Westinghouse/NCEL vane motor had met or exceeded all but the following:

● Reversibility	The original design was not reversible
● Pressure	The original motor was limited to 1000 psi
● Life	The maximum life demonstrated was 50 hours
● Impactor Drive	The original motor was not tested driving an impactor

Two developmental design approaches were pursued to meet these requirements and goals: an improved spring design approach and a pressurized vane approach. The spring design approach built directly upon the original motor design, making only those changes required for reversibility, increased pressure, extended life, and impactor drive. The pressurized vane approach modified the motor design to eliminate the springs — the weakest links in satisfying the motor life requirement. Both approaches were evaluated, and some developmental testing was accomplished prior to selecting the spring design model for fabrication.

2.2 SPRING DESIGN APPROACH

The spring design approach was to improve the original Westinghouse/NCEL version of the seawater motor. Using the existing motor configuration as a starting point, the first step was to identify the motor modifications with the greatest potential to meet the requirements and goals while maintaining low overall program risk. The proposed and final modifications identified are listed in Table 2-1.

2.3 PRESSURIZED VANE APPROACH

The pressurized vane approach eliminated the springs by replacing the spring force with a fluid pressure force. This parallel effort had the same requirements and goals as the spring design approach.

Basically, inlet fluid pressure was directed through the shaft bearings and through the center of the rotor to the underside of the vanes. The differential pressure that developed between the top and bottom of the vanes was intended to keep the vanes in contact with the cam track. See Figure 2-1 for a graphic representation of this flow concept. However, during developmental tests, we found that too much pressure under the vanes and on the face of the rotor caused a braking effect. These and other problems encountered during tests forced abandonment of the pressurized vane approach. See Appendix A for the design and test results of the pressurized vane motor.

On the positive side, the knowledge gained by this effort was used in the final design of the seawater tool motor, particularly in the bearing lubrication scheme.

2.4 MOTOR DESIGN TRADEOFF AND DESIGN SELECTION

As can been seen from Table 2-1, the final modifications for each design area did not necessarily match those proposed. The final ones resulted from design studies made during the development stages of the improved seawater motor with respect to the reversibility, pressure, life, and impactor drive requirements. The following paragraphs summarize the design studies for each design area.

Reversibility

To make the original motor reversible required equalizing the pressure loaded areas of the side plates and rerouting the bearing water supply. The reversibility configuration change was accomplished by enlarging the "outlet port" pad areas, slightly reducing the "inlet port" pad areas, and increasing all four pad areas radially inward (see Figure 2-2). These changes allowed for more equal distribution of pressure pad loading and O-ring loading on the bearing plates. This concept was tested during the development phase with no degradation in motor performance. See Section 8 for a complete test description.

Motor reversibility required a further change to the lubrication grooves in the side plates and a new way to port lube water to the shaft bearings. The first approach taken was to use discharge flow from the handle, and port the seawater through the forward bearing, through the center of the rotor, and finally through the aft bearing to ambient (see Figure 2-3). This concept was found to be impractical because of the small diameter hole required in the handle and because of the risk of allowing suspended contaminants into the motor from the outlet ports in the bearings. Because flow is sensitive to pressure drops and restrictions, and high pressure flow provides a steady lubricating flow, high pressure flow from the motor inlets was passed through restrictors to supply a constant steady flow to the bearings. This high pressure flow concept separated the lubrication system from the handle and made the motor more versatile (see

Table 2-1. Motor Modifications

REQUIREMENTS AND GOALS	PROPOSED MODIFICATIONS	FINAL MODIFICATIONS
Reversibility	● Make inlet and outlet ports equal in area ● Change bearing lube system via compact manifolding within motor	● Ports are equal in area ● Bearing lube flow via restrictors in inlet ports and with flow through center port of rotor
Increased Operating Pressure	● Make springs longer to increase spring force and consequently eliminate "knocking" ● Increase spring hole depth in vanes and rotor for longer springs	● Optimum spring force determined ● Springs have larger O.D. and wire diameter ● Vanes wider for larger spring O.D. ● Vane slot in rotor wider for increased vane width
Life	● Make springs longer to maximize fatigue life ● Change spring material to Eligiloy for better fatigue life and corrosion resistance ● Use dry film lubricants or incorporate spring guide of bearing material to reduce spring wear ● Change vane, bearing, and side plate material to Torlon 4275 or Torlon 7130 for better wear ● Change axial port in rotor to circumferential axial port to reduce vane tipping	● Redesigned spring for 250 hr life and nonbuckling ● Spring material is Eligiloy ● One set of springs are coated with dry film lubricant — tiodize compound ● Vane, bearing, and bearing plate material is Torlon 4275 ● Eliminated rotor axial port for wider vanes ● Increased inner annular slots in bearing plates for better timing
Impactor Drive	● Design elastomeric coupling ● Optimize power requirements ● Supply spring sets of various forces	● Direct drive tested and direct solid coupling designed ● Increased rotor shaft O.D. for increased loads and stress ● Porting from one side of motor

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NOTES:
1. SHIM AS REQD. SO CAM (4) IS FLUSH TO .001 BELOW SURFACE OF (5).
2. RETAIN USING LOCTITE, GRADE 242
3. VANES TO SLIDE FREELY IN GROOVES OF ROTOR (5)
4. ROTOR (5) TO ROTATE FREELY BEFORE & AFTER (8)(9)(10)(11) ARE SATURATED WITH WATER.

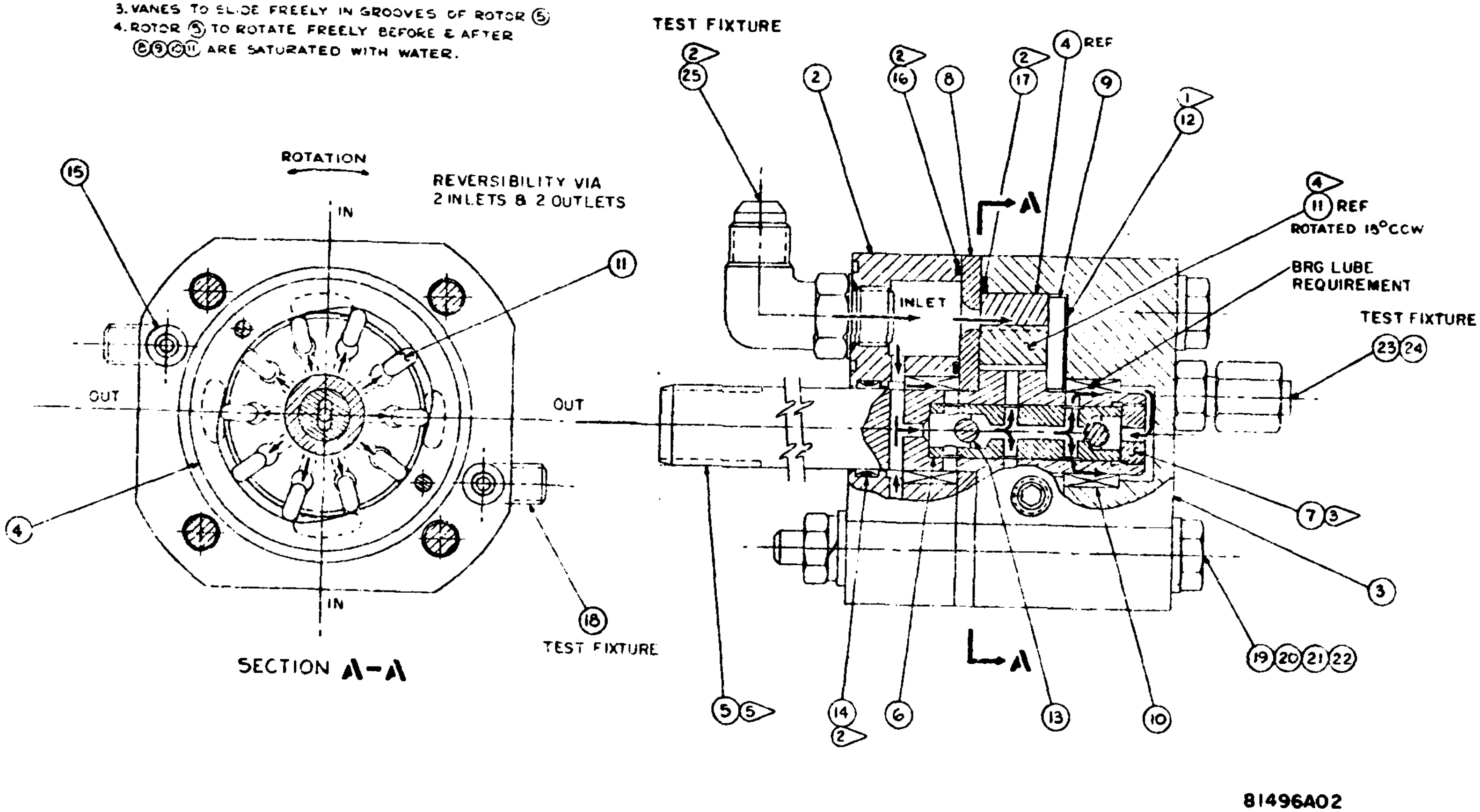

Figure 2-1. Pressurized Vane Motor

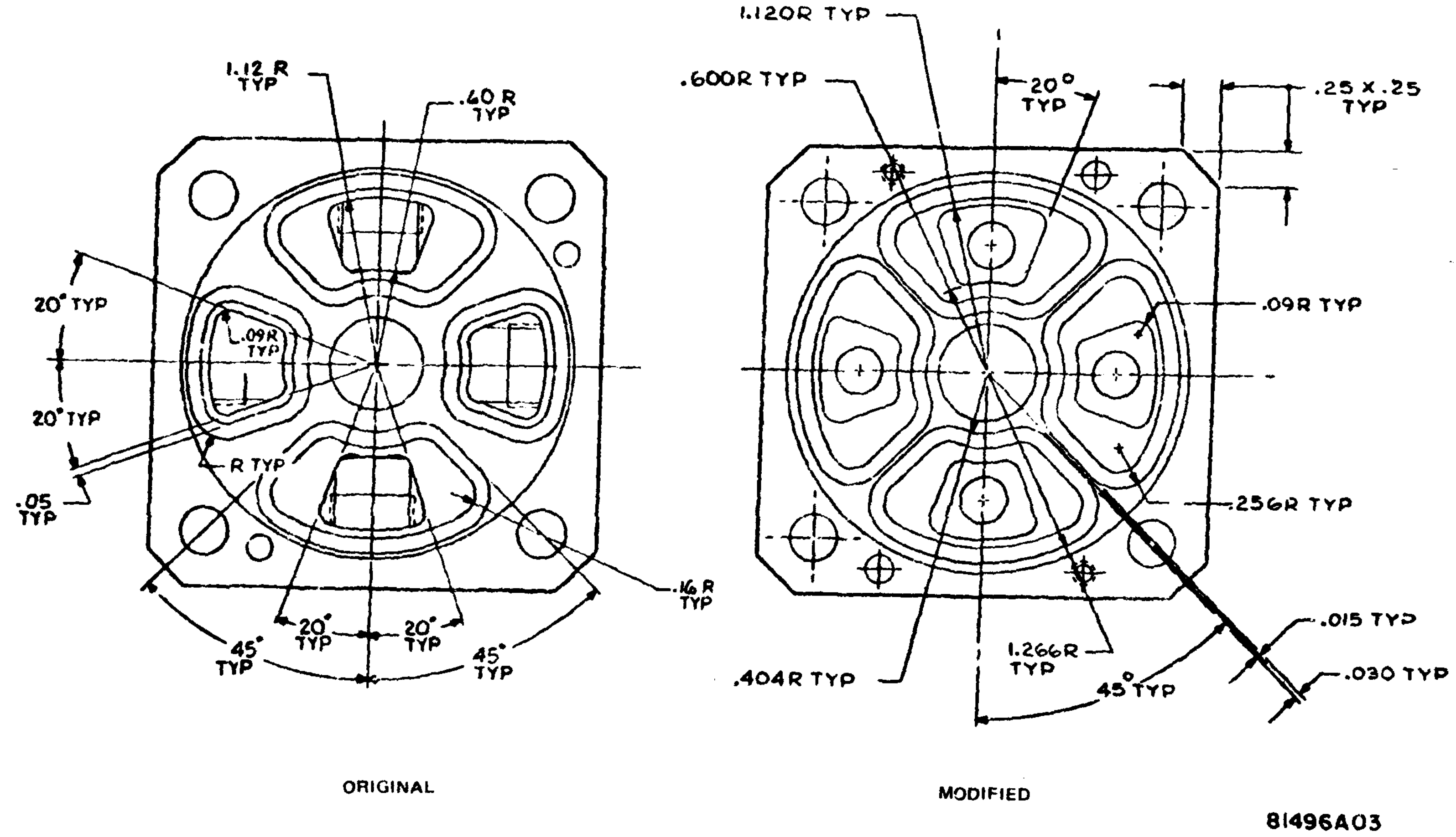

Figure 2-2. Reversibility Modification

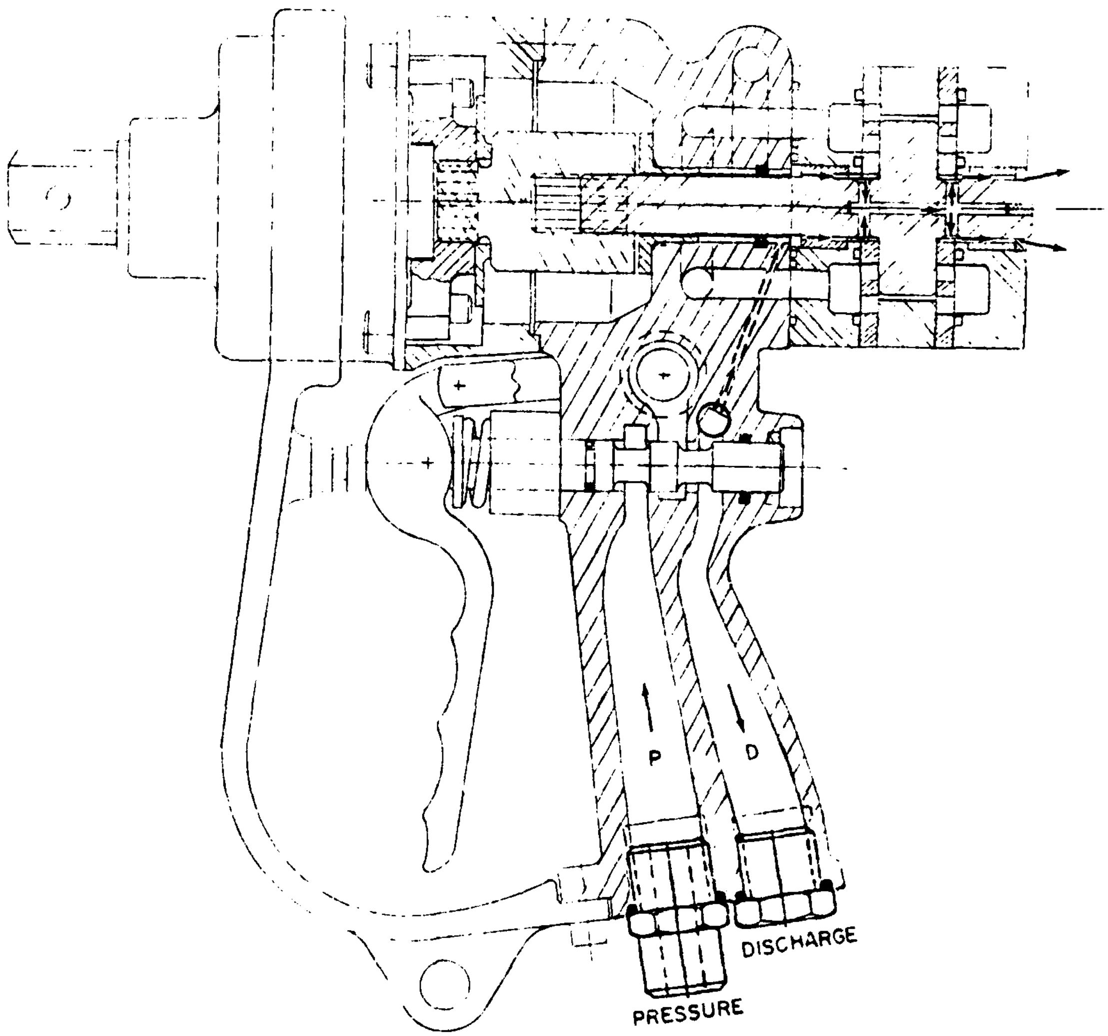

Figure 2-3. First Approach to Bearing Lube

Figure 2-4). The bearing lube flow was also used to cool the coupling support bearings and the impactor bushing.

Increased Operating Pressure

We found during the prototype motor development and testing phases, that the primary limitation in motor operating pressure was vane instability or "knocking". This instability was due to dynamic forces on the vanes causing them to lift away from the cam track and oscillate; vane resealing caused water hammer and cavitation. This effect was pressure dependent, increasing in severity as pressure increased.

A proposed solution was to increase the spring force so that the vane tip remained in contact with the cam track. To reduce the stress levels within the spring due to this increased force and to prolong spring life, the feasibility of deepening the spring holes in the rotor and vane was investigated. However, a

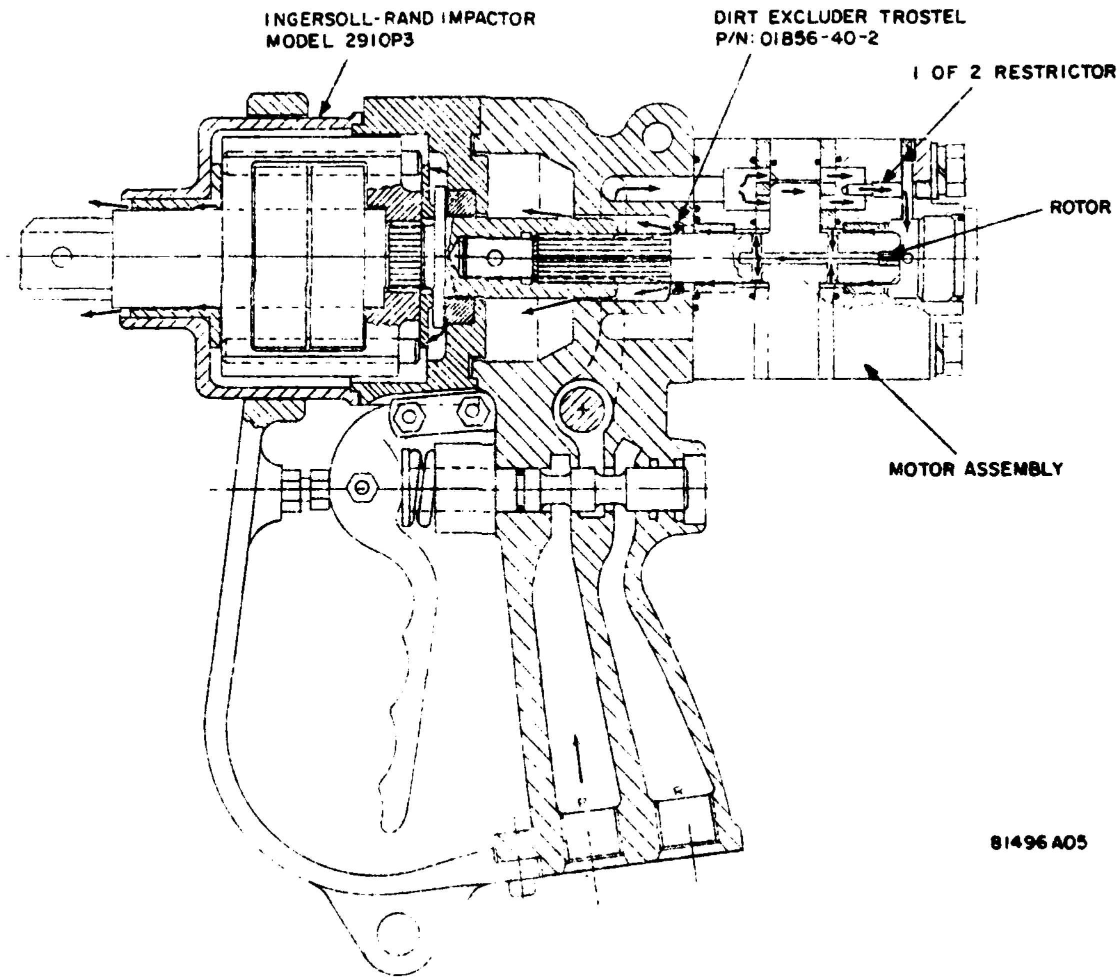

Figure 2-4. Final Approach — Rotor Shaft Bearing Lube Flow

detailed layout showed that a deeper hole in the rotor was not possible. It was also found that increasing the spring length without increasing the spring outside diameter would cause greater buckling while the spring was in compression. This buckling would create increased spring wear because the spring would rub the edge of the rotor spring holes and the side of the rotor vane slots.

Therefore, two consecutive design approaches were taken: (1) to design a nonbuckling spring using a larger wire diameter and a larger outside diameter for the 250-hour life goal; and (2) to test the original Westinghouse/NCEL motor using various spring sets of different spring forces to determine the optimum spring force required for a "no-knock" increased pressure operation with minimum vane tip wear. Ultimately, a compromise was struck between spring geometry to meet the motor life goal and spring force to meet the peak pressure goal. See Appendix B for spring design analysis and Appendix D for a description of the spring force determination tests.

Life

The most demanding design goal was the 250-hour motor life target. Two approaches were taken to extend motor life: (1) to make a detailed study of the critical motor components and modifications while maintaining the desired performance goals; and (2) to optimize the motor power requirements to drive an impact mechanism.

The critical components studied were the springs; the vane, shaft bearing, and bearing side plate materials; and the rotor vane slot configuration.

Springs

After determining that the optimum mean spring force to meet the increased motor pressure requirement was 0.75 pounds, a study was initiated to extend spring life by changing the spring geometry and material. The spring studied had a larger wire size and outside diameter for better fatigue resistance, with a shorter free length to prevent column buckling. To compensate for the larger spring outside diameter, the vane width had to be increased proportionally, while the optimum spring force was increased to 1.0 pound for the wider vanes. The spring material was changed, as proposed, from 17-7PH stainless steel to elgiloy, a cobalt-nickel spring alloy with better fatigue life and much better corrosion resistance than 17-7PH. See Appendix B for the spring design analysis.

To reduce spring friction and related wear, the practicality for either installing spring guides made of a bearing material in the rotor spring holes or coating the springs with a dry film lubricant was studied. The incorporation of the spring guides made of "karon" (a polyester-teflon composite) proved to be too costly and time consuming for this phase of development. Also, a set of 17-7PH stainless steel springs was coated with tiolube 460 (a dry film lubricant) and overcoated with tiolon E20 (a solid film lubricant). This set of "tiodized" springs was not tested for a long enough period to assess wear characteristics. Therefore, a set of "tiodized" springs made of elgiloy will be supplied with the improved tool motor for further evaluation and tests.

Materials

One of the major factors in limited motor life was the wear of the Torlon 4301 vanes, bearing side plates, and shaft bearings. Two grades of Torlon were selected for evaluation and developmental testing: Torlon 4275 (proposed for its better wear properties than 4301) and Torlon 7130 (recommended by the Torlon manufacturer, Amoco Chemical Corp., if higher strength vanes were required). The higher modulus, the good dimensional stability in water, and the equal wear resistance (to Torlon 4275) were the reasons behind the Torlon 7130 recommendation. However, tests conducted by NCEL on 7130 vanes showed a high vane tip wear rate (approximately 1 mil per hour), scoring of cam track and bearing plates (probably due to graphite fibers), and low mechanical efficiency. Therefore, no further investigations were made using Torlon 7130. Torlon 4275 was selected for the vanes and bearings.

During the development phase of the original Westinghouse/NCEL motor, we found that color and texture of Torlon surfaces varied considerably, indicating nonuniformity in material distribution and porosity. We also found that the wear resistance of Torlon improved with post curing. Therefore, working with Amoco Chemical Corp., stringent inspection requirements were imposed on the raw material to ensure uniformity. The marble appearance of Torlon 4275 was acceptable, while any "grainy" surface texture was cause for rejection. Finally, post curing of all Torlon raw material was specified on the drawings to ensure the needed wear resistance.

Vane Slot Configuration

In the original motor, when the vane tip rode on the cam track minor diameter, the vane side below the vane grooves extended radially inward beyond the slot walls into the slot ports. This resulted in excessive angular motion of the vane which caused unnecessary spring wear due to bending and eventual spring failure (see Figure 2-5 a). To allow the vane sides to remain in contact and to slide on the slot walls, we proposed to elongate the slot walls, replace the axial ports with axial circumferential ports (see Figure 2-5 b), and relocate the inner annular slots in the side plates to accommodate the new rotor ports. During developmental tests at NCEL, we found that the circumferential ports were difficult to manufacture. Therefore the proposed ports were replaced with circular axial ports that were moved radially inward as far as possible to provide enough sliding surface for the vane while not affecting motor timing. This concept was tested in the original motor and did not affect motor performance (see Figure 2-5 c). Further investigations showed that these ports were not needed with the wider vanes and were eliminated (see Figure 2-5 d). Also eliminated was the need to relocate the annular slots in the side bearing plates. However, to improve motor timing, the inner annular slots in the side bearing plates were increased from 26 degrees to 30.6 degrees. This angular increase allows the bearing plate slots to be tangent to the rotor vane slot when the vane tip is tangent to the ramp starting point on the cam track (see Figure 2-6).

Impactor Drive

Early in the development phase it was demonstrated that the seawater motor could directly drive the Ingersoll-Rand Model 2910 impact mechanism without the need for an elastomeric coupling as proposed. Therefore, the only modifications to the tool motor, with respect to impact drive, were the porting arrangement change and the rotor shaft diameter increase.

To change the porting arrangement only required reducing the number of inlet and outlet ports from four to two and restricting the porting to one side of the motor. This concept was tested by blocking off the aft (short shaft side) two inlet and outlet ports in the original motor and running performance tests. The tests showed no change in motor performance.

With respect to the rotor shaft diameter change, calculations were based on heavy shock loads being transmitted to the rotor via direct coupling to the impactor. These calculations indicated a needed increase in diameter from 0.500 inches to 0.609 inches (see Appendix C for actual calculations).

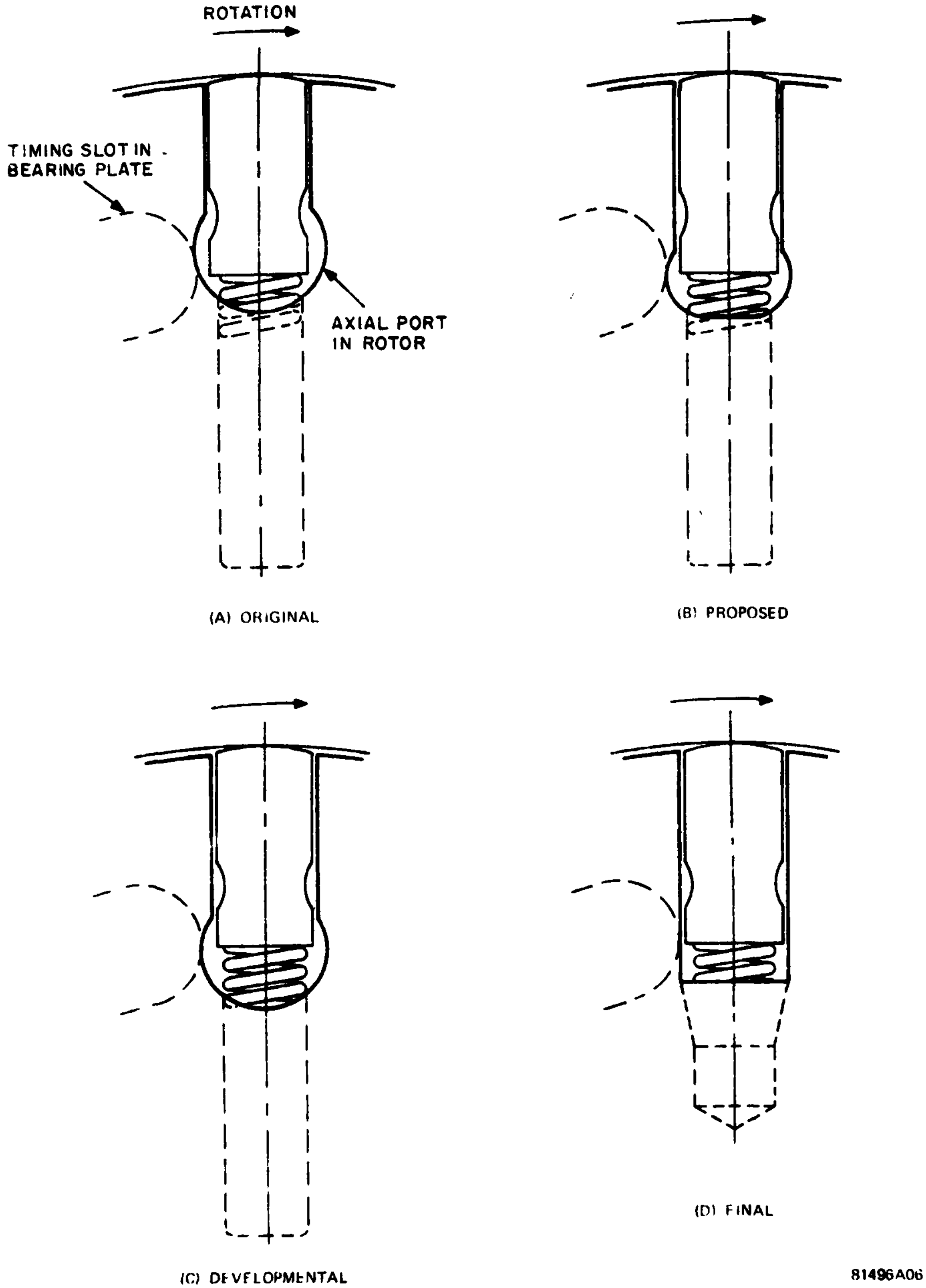

Figure 2-5. Rotor Vane Slot Configurations

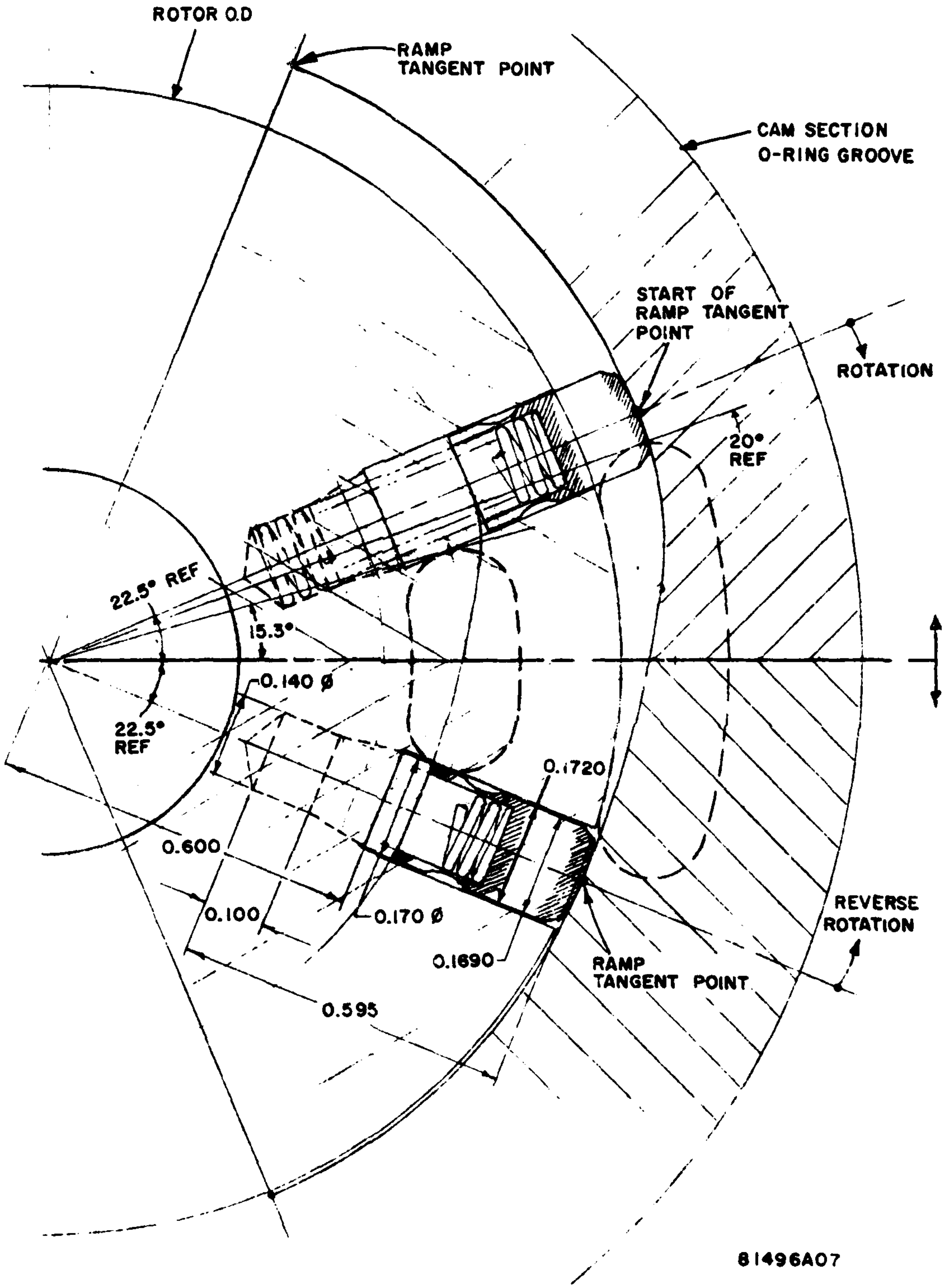

Figure 2-6. Rotor Spring Hole Detail

We proposed that optimization of motor design for the reduced power required by the impactor could lead to a significant increase in motor life. However, all motor modifications were based on calculations independent from the impact mechanism power requirements except for the rotor shaft size and porting arrangement. For example, calculations on the springs were based on a 250-hour fatigue life at 1500 rpm, while calculations on the vanes were based on 1500 psi operation (see Appendix B).

Other Materials and Processes Investigated

During the beginning of the program, an intensive materials investigation task was initiated. Various materials and processes which seemed suitable for motor components were evaluated. See Table 2-2 for a list of the materials and processes investigated.

It was ascertained during the preliminary motor design review that the major portion of these materials and processes could not be tested within the time frame of this program, and that the cost was prohibitive. Therefore, this study was postponed but should be considered for future development and value engineering programs.

As indicated from the previous discussion, many tradeoffs were made during the development stages of the improved seawater tool motor. A description of the final design is the subject of the next section of this report with detail drawings found in Appendix H.

Table 2-2. Preliminary Materials/Processes Investigation

MOTOR PARTS	MATERIALS/ PROCESSES	VENDOR/ SUPPLIER	REASONS INVESTIGATED	REASONS NOT INCORPORATED
Cam & Cam Surface	Macor Ceramic Cam (Dwg. SKHAH1010)	Accuratus Ceramic Corp. Washington, NJ (201) 689-0880	Easily machinable to 16 rms surface finish; to provide a hard cam track wear surface	Time & Cost
	Stellite 6B weld overlay on 316 SST cam track	Cabot Corp. Kokomo, IN (317) 457-8411	To reduce manufacturing costs and provide excellent wear surface for cam track	Time & Cost
	Triboloy cam casting	Cabot Corp. Kokomo, IN	New material with superior corrosion resistance and wear properties	Time & Cost
	Karamite DST plasma sprayed coating on 316 SST cam track	Kamatics Corp. Bloomfield, CT (203) 243-9704	To reduce machining cost and provide hard wear surface on cam track	Time & Cost
	Precision machining of cam track	Surface Finishes, Inc. Addison, IL (312) 543-6682	To provide better than 16 rms surface finish on cam track	Time & Cost
Springs	Karon spring cups (Dwg. SKHAH1008)	Kamatics Corp. Bloomfield, CT	To reduce friction & wear between springs and rotor slots	Time & Cost
	Tiodized springs (Dwg. SKHAH1008)	Tiodize Co., Inc. Huntington Beach, CA (213) 594-0971	To provide low friction coating on springs to reduce wear and increase life	Incorporated but not tested thoroughly
	Cryotech 302 spring wire	Al Tech Steel Corp. Watervliet, NY (518) 273-4110	Alternate less expensive spring wire equivalent to Eligiloy	Wire diameter too large
	Alloy 6x spring wire		Alternate spring wire	Time & Cost
Vanes & Side Plates	Torlon 7130	AMOCO Chem Corp. Chicago, IL (800) 621-4557	To provide higher strength with equal wear resistance to 4275	Tested — scored cam surface

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3.0 MOTOR DESIGN DESCRIPTION

The final seawater tool motor design incorporated the proposed modifications described in Section 2 plus additional performance/life modifications determined during evaluation testing. We also incorporated some engineering changes to reduce machining costs for production. This section describes the seawater tool motor operation and the final detailed design.

To keep the reader from becoming lost in details, frequent references to the appendices are made for supporting calculations and tests.

3.1 DESCRIPTION AND OPERATION

The seawater tool motor is a reversible, double entry, hydraulic vane-type with a pressure balanced rotor and bearing (side) plates. The surfaces of its components are lubricated only with the seawater used to power it. The tool motor is shown assembled (without the test manifold) in Figure 3-1, partially disassembled in Figure 3-2, and fully disassembled in Figure 3-3. The physical and operational characteristics of the motor for various test conditions are listed in Table 3-1 with supporting motor performance curves shown in Figure 3-4. Features of the motor are described in Table 3-2.

Operation

The tool motor operates in the same general manner as the original Westinghouse/NCEL vane motor except that the tool motor is reversible. 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 cam 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 pressure flow enters the space between the rotor and cam track at the ramps where the vanes follow the cam track from the minor to major diameter. The low pressure flow exits the space between the rotor and cam track at the ramps where the vanes follow the ring track from the major to minor diameter. The low pressure fluid then flows back through the rotor and cam and out the return ports. See Figure 3-5 for graphic representation of flow through motor.

The tool reversibility comes from the equal area pressure pads on the forward and aft housings and from a unique method of lubricating the shaft bearings (see Figures 3-6 and 3-7). For either rotational direction, bearing lube flow initiates from the high pressure ports, flows across the rotor to the cavity in the aft housing, passes through the Lee Visco Jet Restrictor to the aft housing center cavity, flows through the shaft bearing axial slots, is forced through the rotor central ports to forward shaft bearing, passes through the bearing axial slots, and exists the motor via a modified dirt excluder. See Appendix B for calculations and preliminary design of the Bearing Lube Concept.

Table 3-1. Physical and Operational Characteristics

PARAMETER	GOALS	VALUES				COMMENTS
		MAX	AFTER 50 HR	FROM CURVES	DURING TEST	
Power Output (hp)	3.0	3.92	2.93	3.00	2.00	Exceeds the goal of 3 hp at 1500 psi.
Speed (rpm)	1000-15,000	1519	1628	1445	1416	Meets the goal range at 1000 to 15,000 rpm
Overall Operational Efficiency (%)	75	64	68	66	70	Close to goal.
Operational Pressure (psi) at Full Power	1200	1500	1044	1200	822	Exceeded the minimum goal of 1200 psi initially. (Should exceed 1200 psi with no spring wear (new springs) and increased force springs.)
Maximum Flow Rate (gpm)	7.0	6.96	7.03	6.42	6.00	Meets the goal of 7 gpm flow rate.
Operational Life Expectancy	250	75	—	—	100	Does not meet the goal of minimum life of 250 hours. (May exceed 250 hours with further development.)
In-Water Weight (lb)	7.0	5.86	—	—	—	Meets the goal of weight less than 7 lb. Should decrease with compact aft housing.
Dry Weight (lb)	—	6.85	—	—	—	Should decrease with compact aft housing
Volume ($in.^3$)	25 w/o Manifold*	27.5	—	—	—	Slightly exceeds the goal of volume less than 25 $in.^3$. (Should decrease to 25 $in.^3$ with compact aft housing.)
Reversiblity	Reversible					

*The manifold is a test manifold and should not be considered part of the tool motor.

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Table 3-2. Features of Seawater Tool Motor

● Vane type hydraulic motor - to minimize wear effects on performance.
● No lubricants other than 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 cam track - to allow operation over a wide range of pressures.
● Pressure balanced, flexible bearing 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.
● Hydraulic restrictor valves in high pressure ports and internal rotor porting to direct seawater through bearings - 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.
● Two inlet and two outlet ports - for pressure balancing.
● Small size per displacement ratio - to minimize size and weight.
● Design based on properties of the best available bearing material combinations - to maximize performance and life.
● Equal area inlet and outlet pressure parts for reversibility.
● Eligiloy springs for better corrosion resistance and longer fatigue life.
● Internal manifolding to mate with wrench handle and provide bearing lubrication and cooling.
● Splined rotor shaft impactor drive to minimize thrust loads on rotor during operation.
● Larger diameter shaft for shock loads transmitted from impactor.

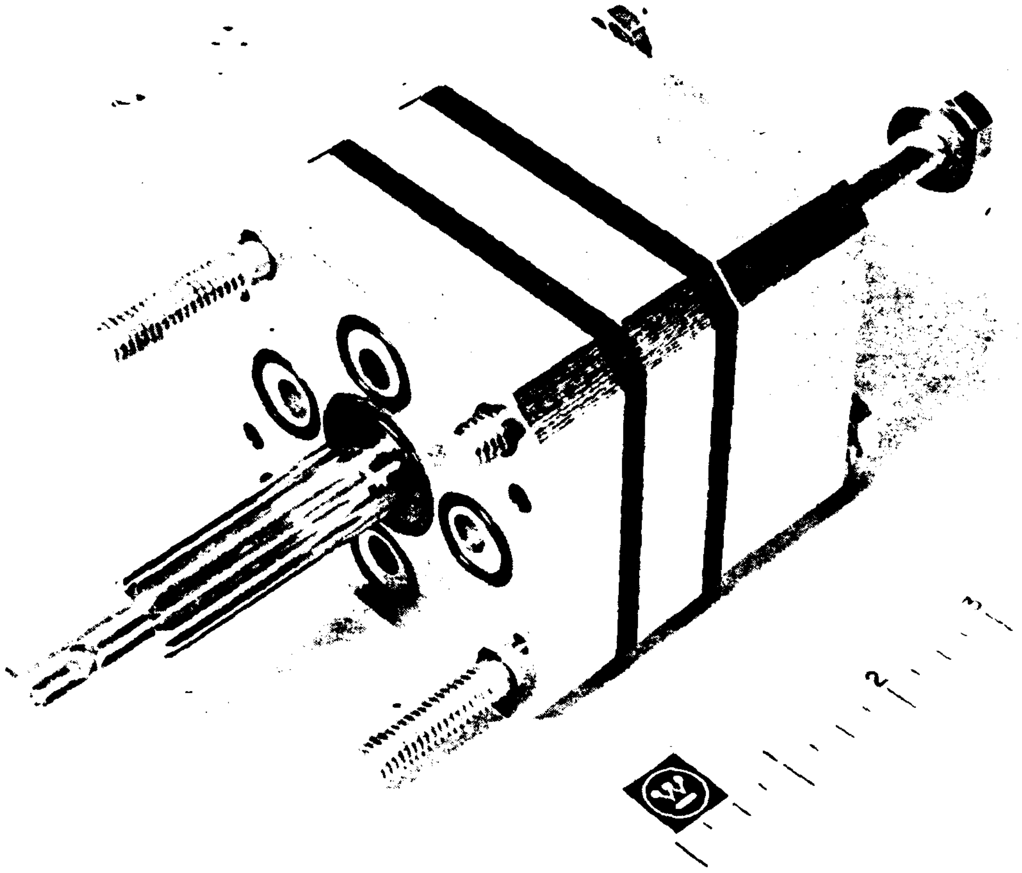

Figure 3-1. Seawater Tool Motor Assembled

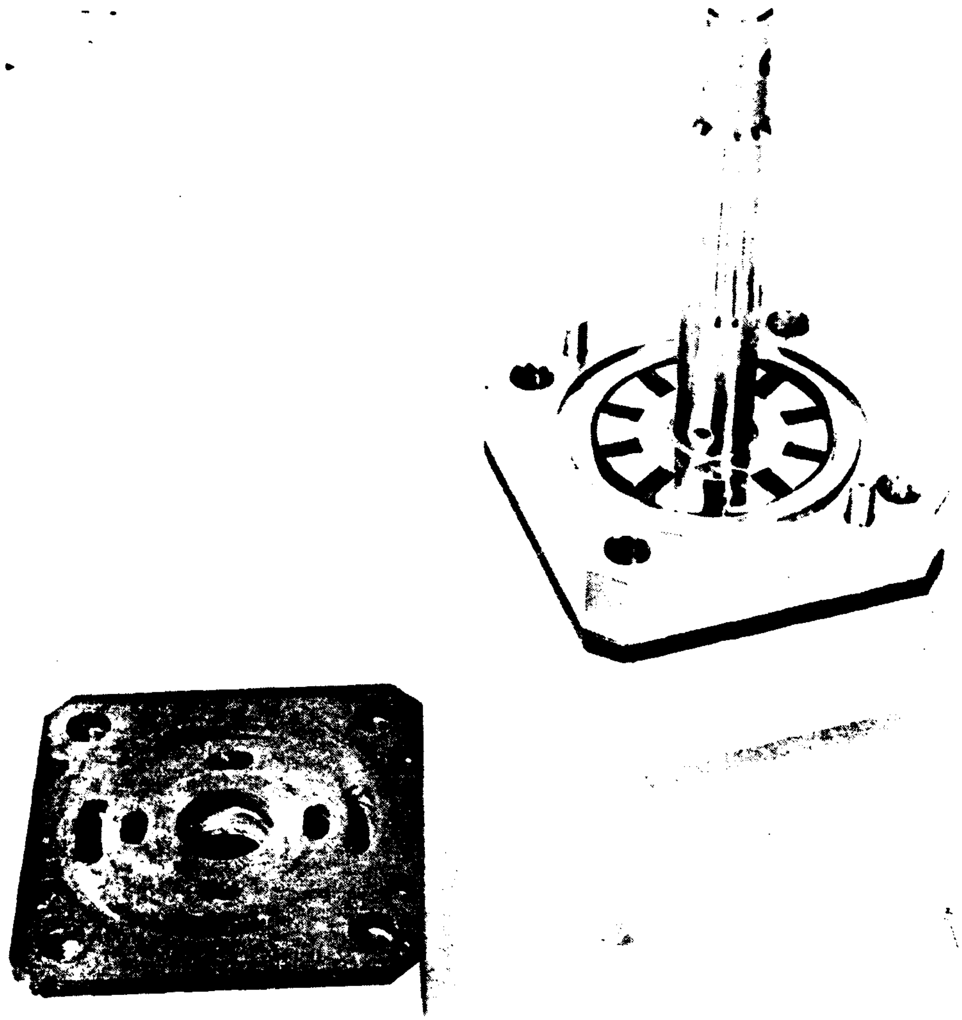

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Figure 3-2. Seawater Tool Motor Partially Disassembled

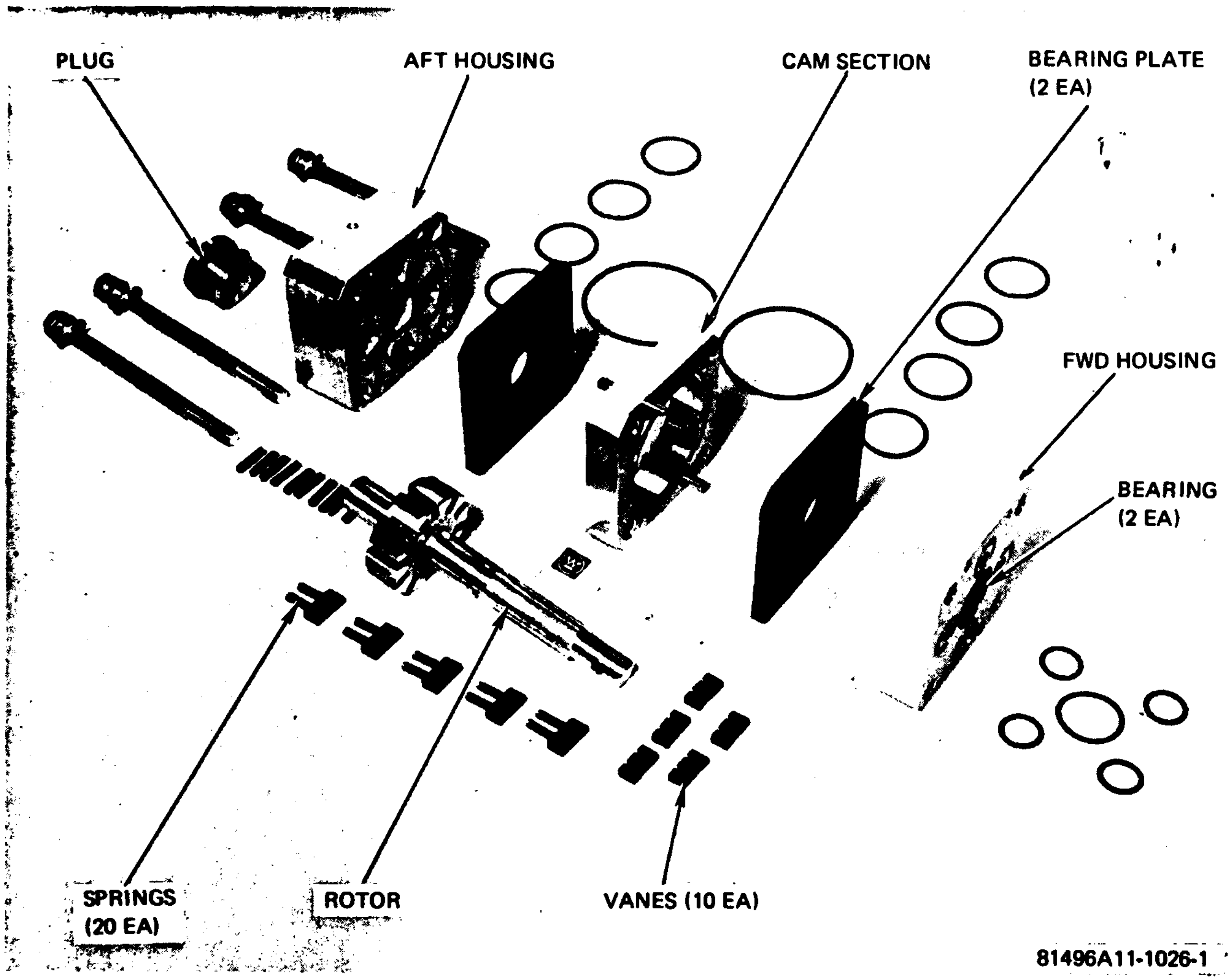

Figure 3-3. Seawater Tool Motor Completely Disassembled

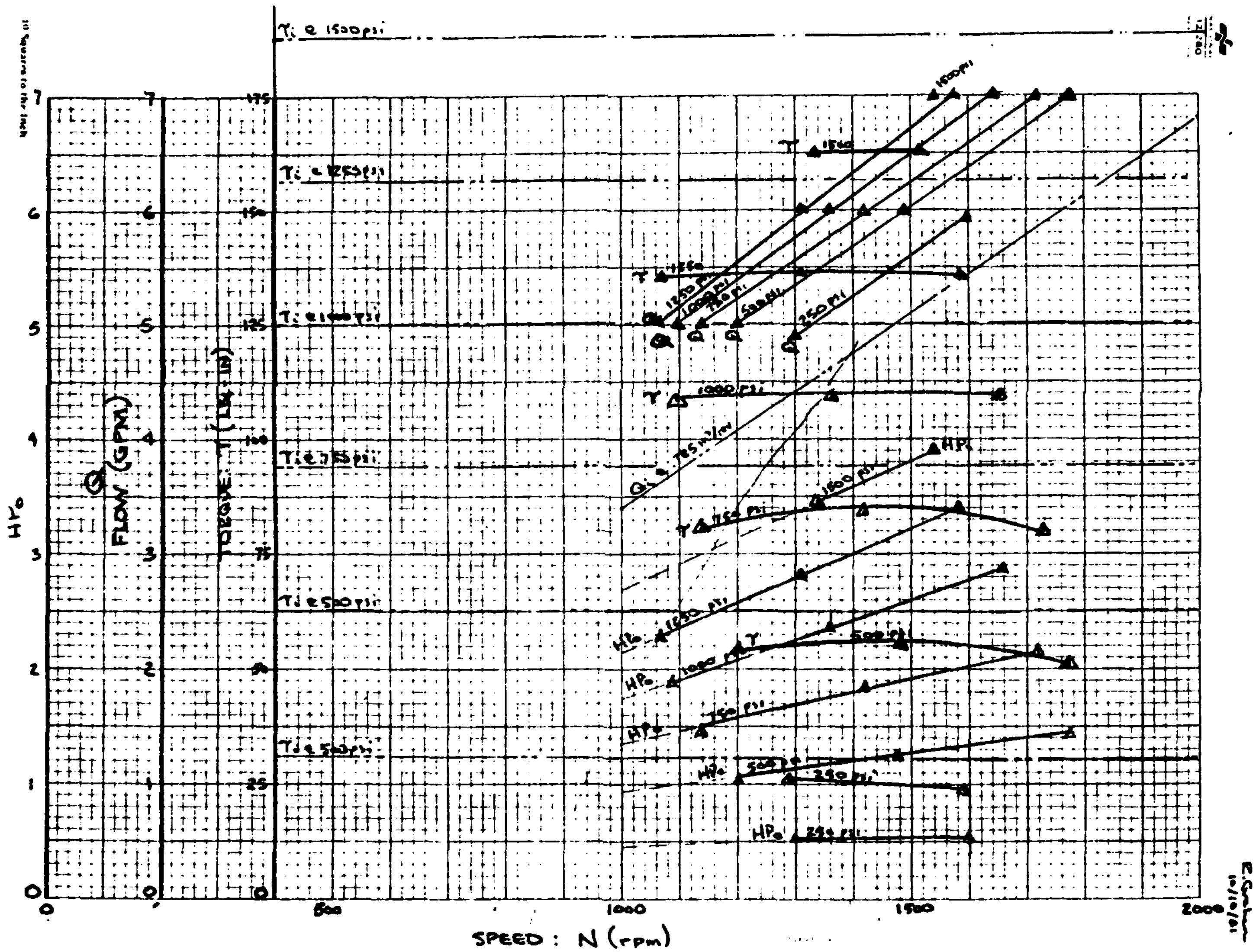

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Figure 3-4. Overall Tool Motor Performance After 16 Hours

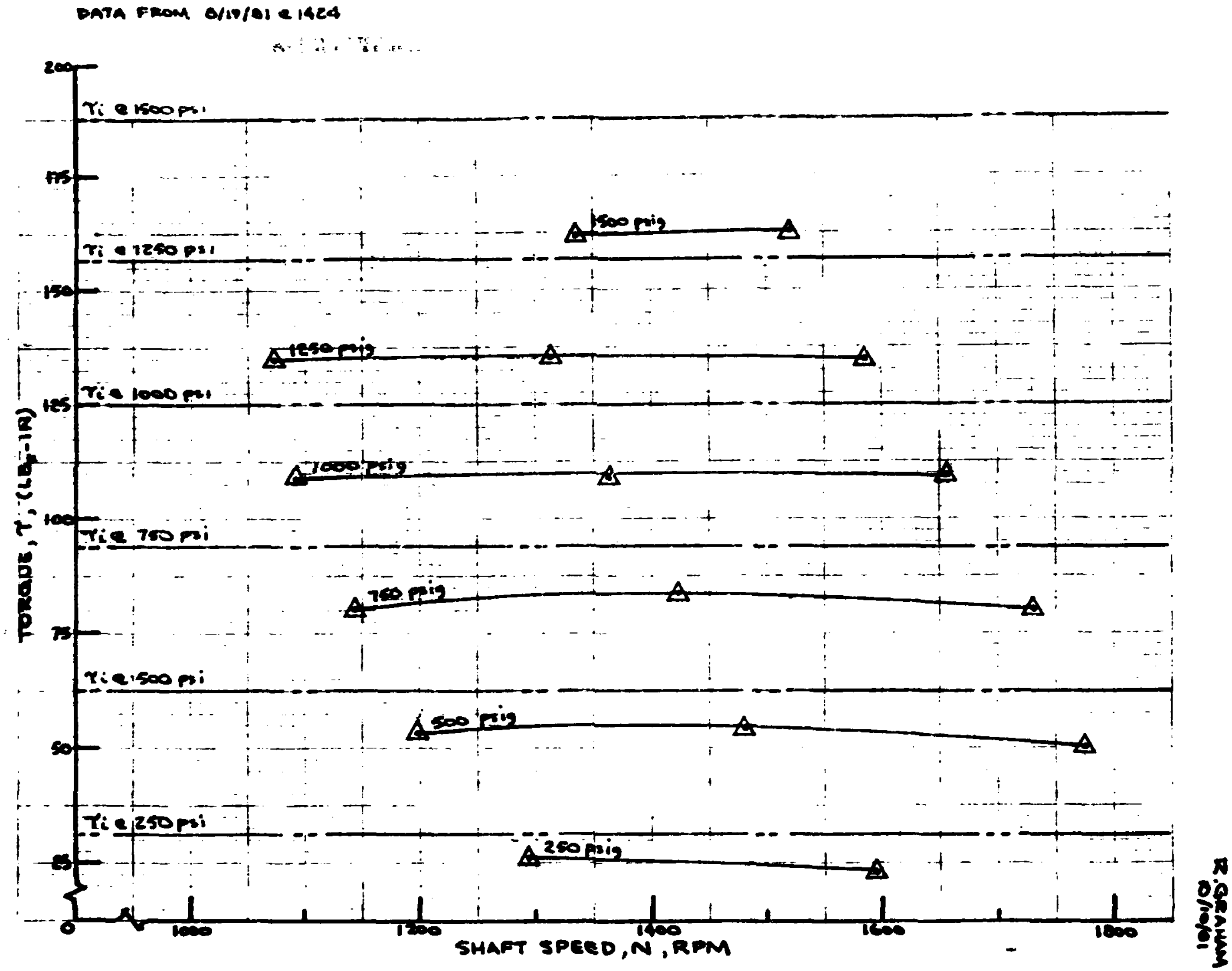

Figure 3-4 a . Torque

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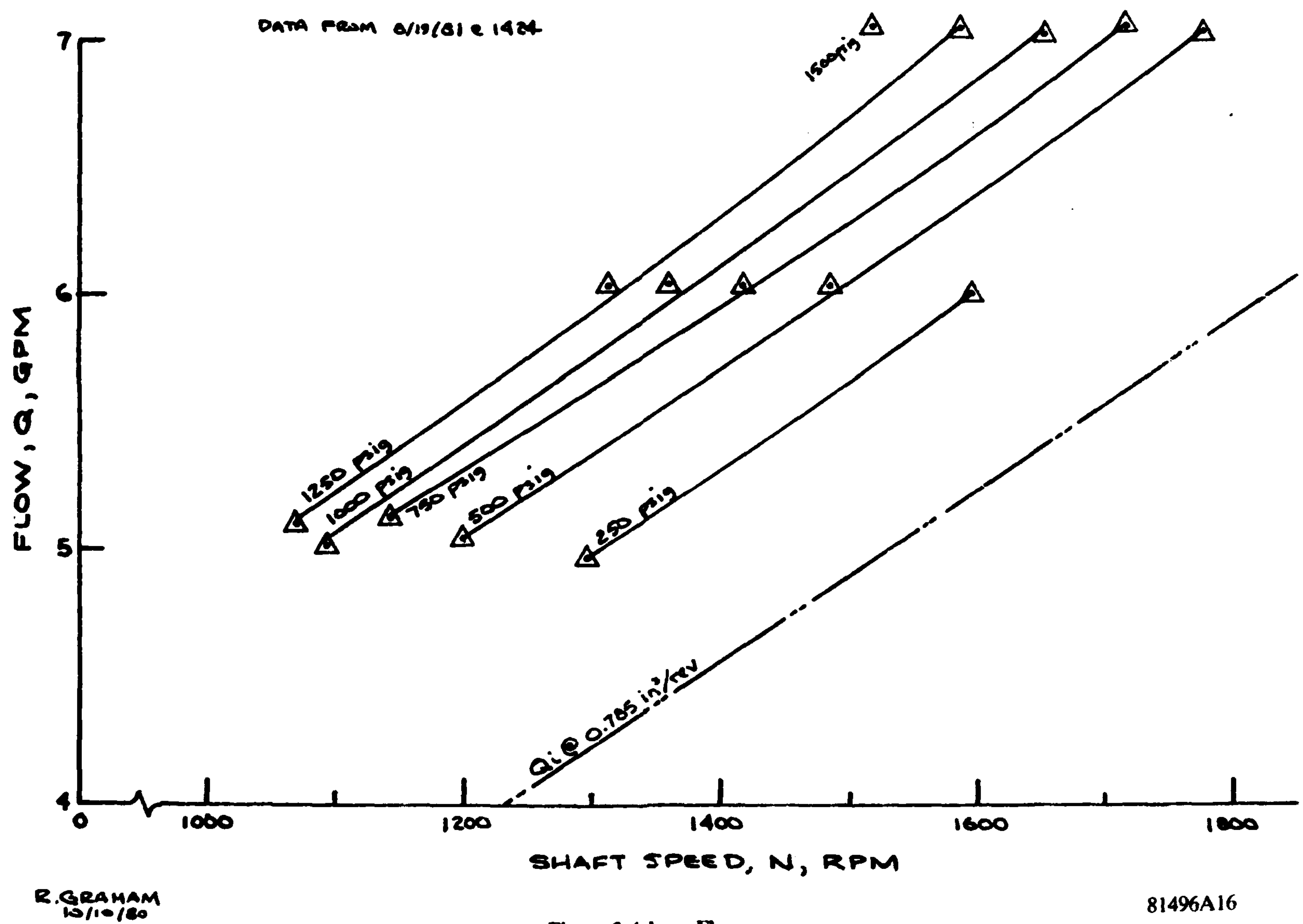

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Figure 3-4 b. Flow

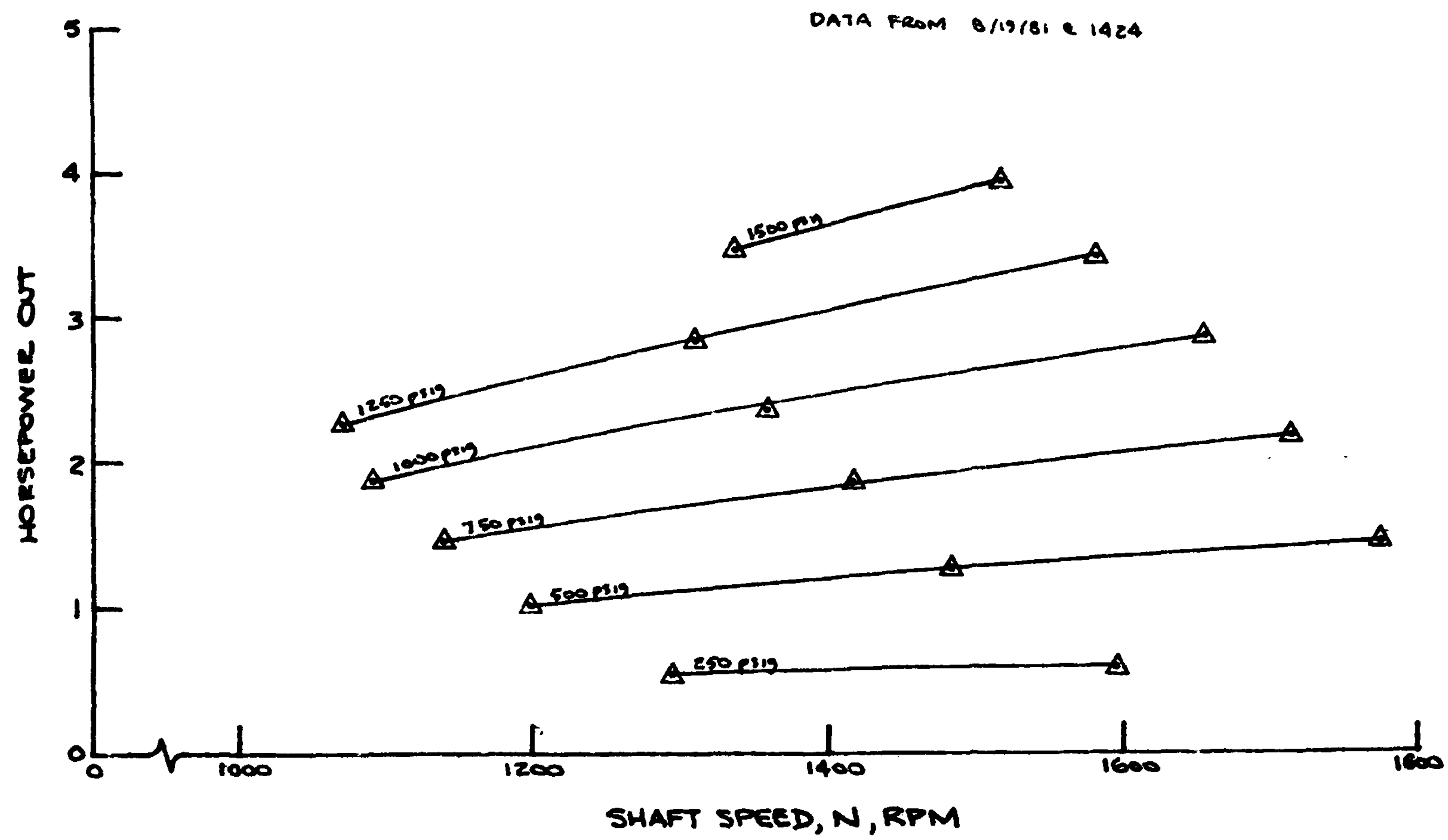

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Figure 3-4 c . HP_0

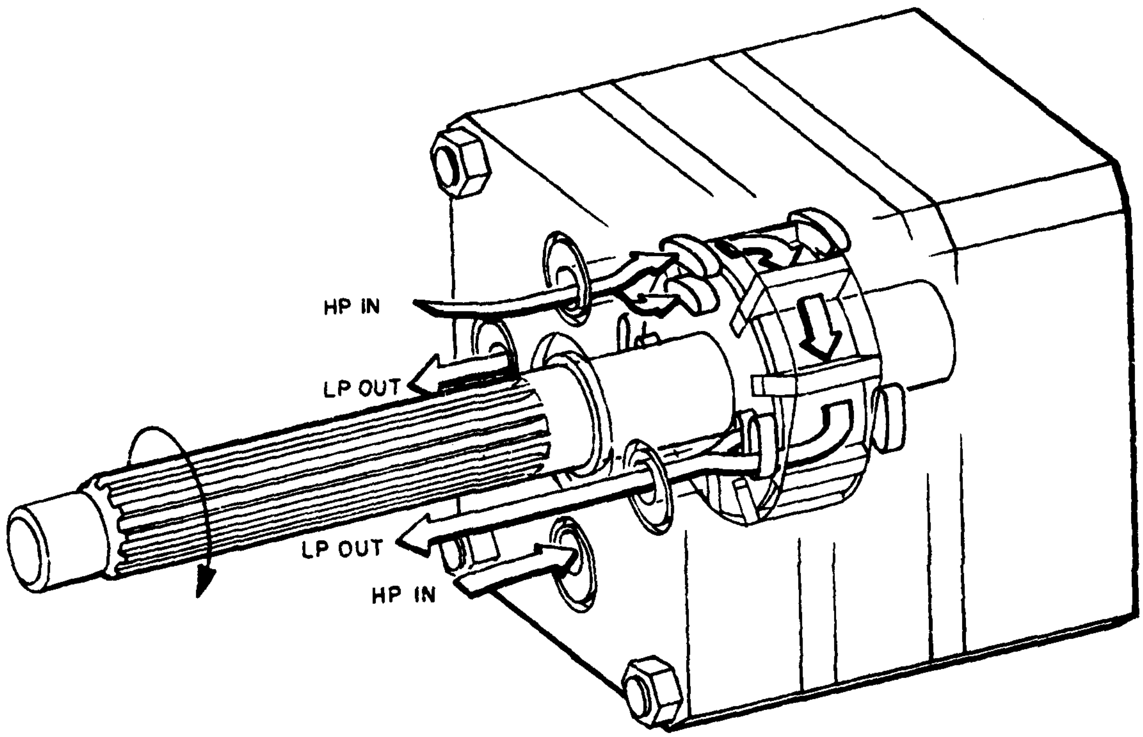

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Figure 3-5. Operational Flow

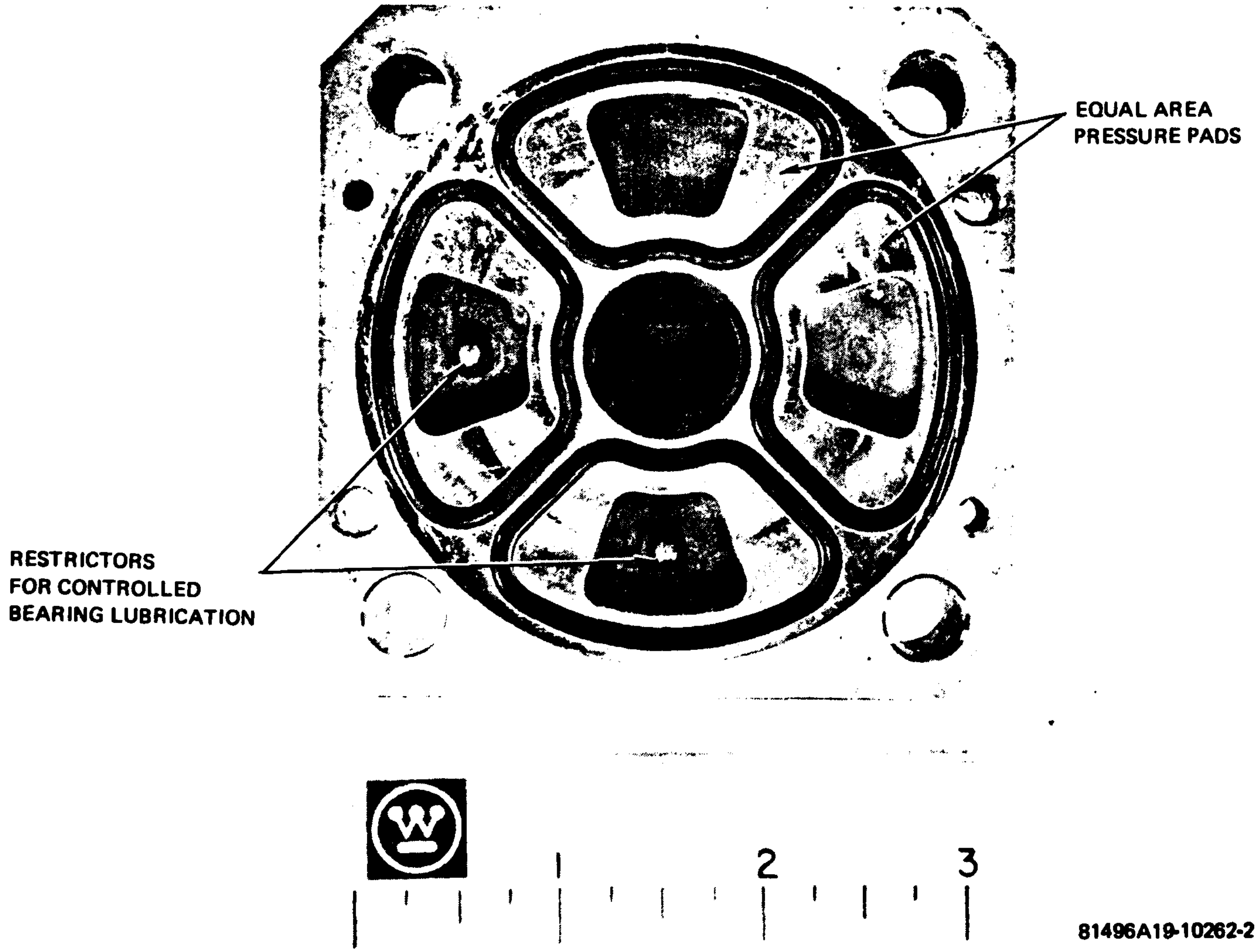
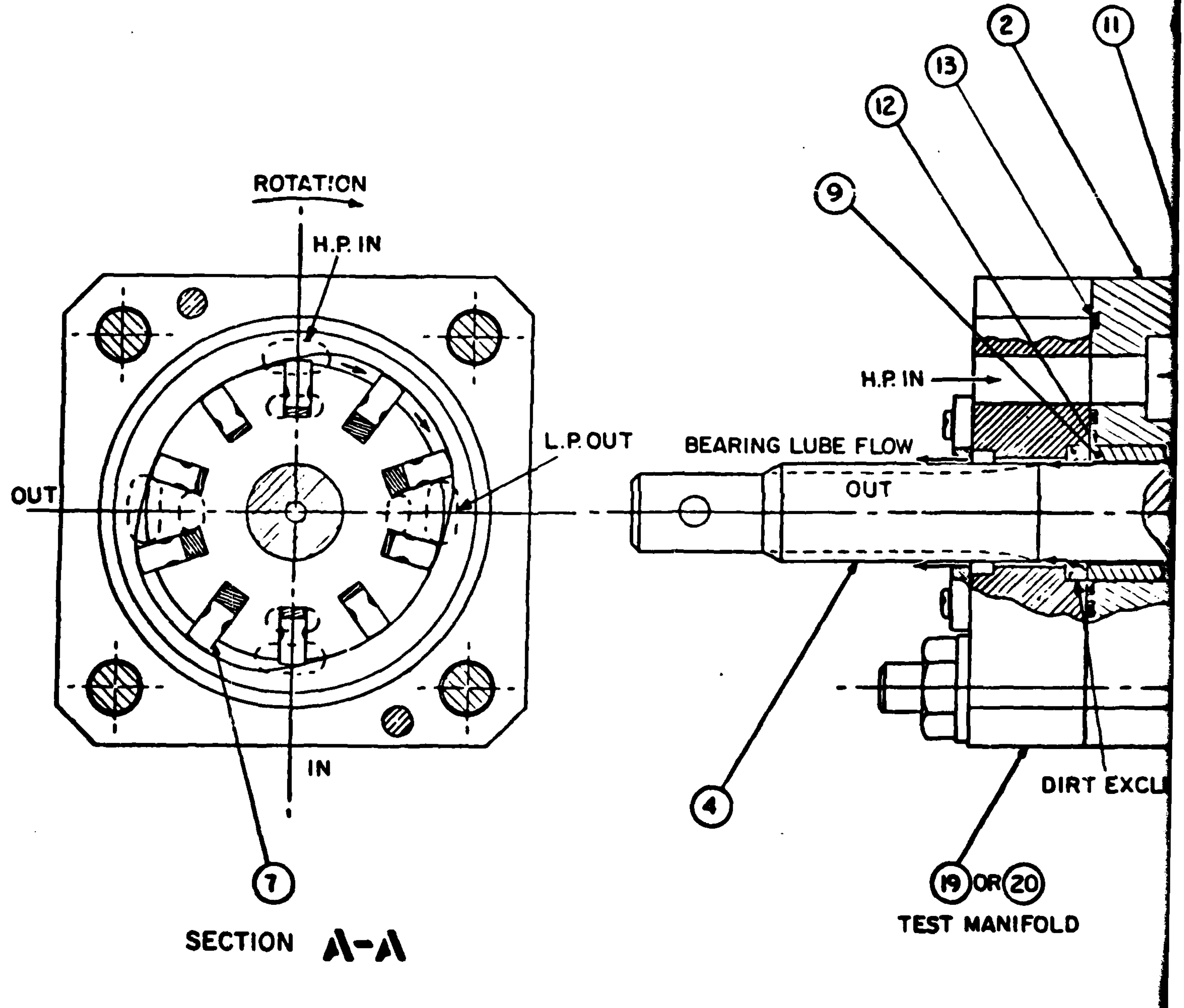


Figure 3-6. Aft Housing Pressure Pad Areas


ROTATION
H.P. IN
L.P. OUT
OUT
IN
7
SECTION A-A
2
11
13
12
9
H.P. IN
BEARING LUBE FLOW
OUT
4
DIRT EXCL
19 OR 20
TEST MANIFOLD

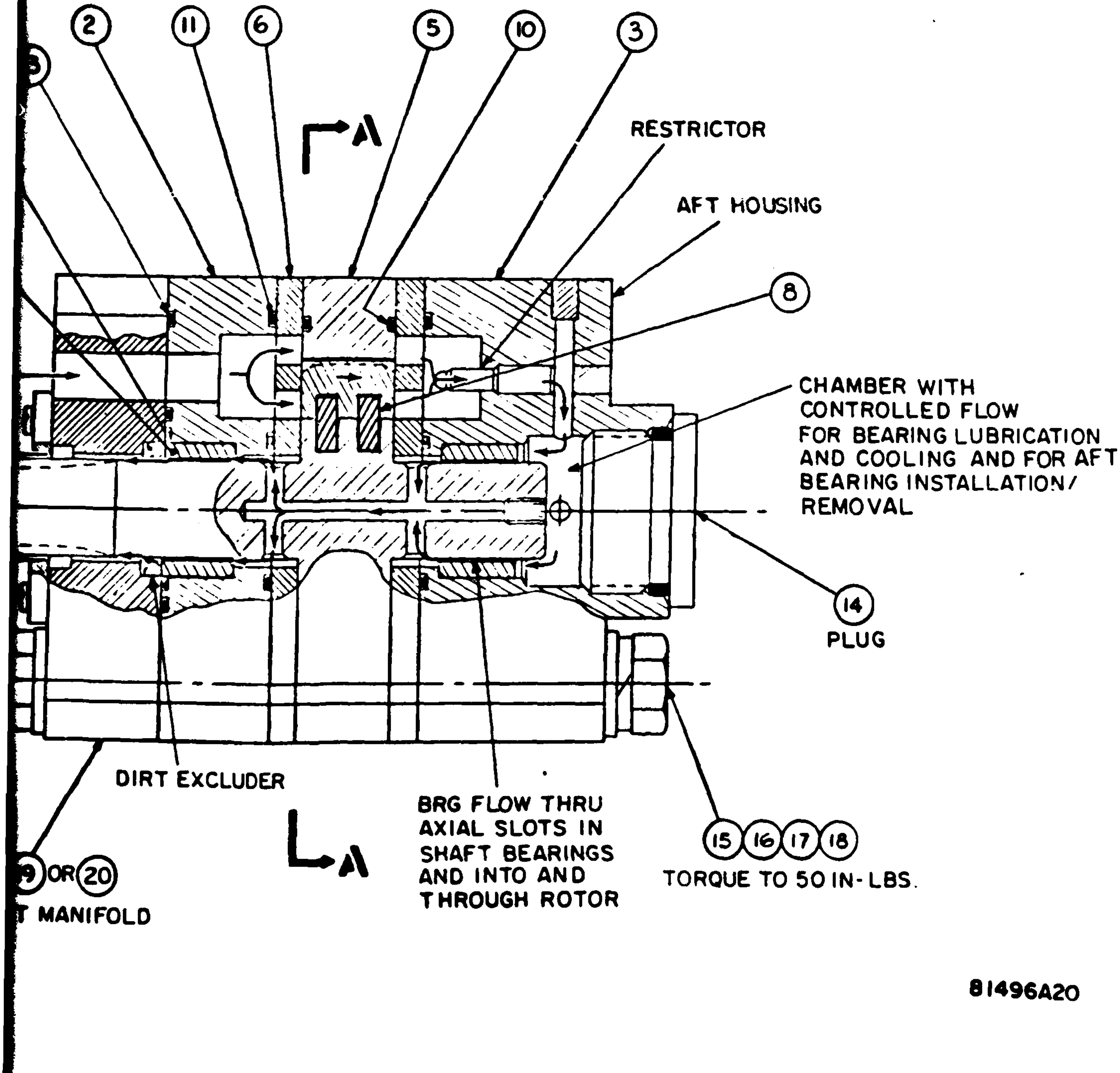


Figure 3-7. Bearing Lube Flow

3-2. DETAIL DESIGN DESCRIPTION

Besides meeting the requirements and goals, the basic design philosophy behind the fabrication of the tool motor was to design it as a production prototype. The transfer drilling and pinning of the housings and shaft bearings of the original motor design were eliminated. The pins are now part of the cam section while the bearings are press-fitted into place during assembly. Some of the tight tolerances of the original Westinghouse/NCEL seawater motor were eliminated also. See Appendix H for the complete tool motor drawing package.

Reversibility and Bearing Lubrication

To make the motor reversible required two changes: (1) machining the inlet and outlet port pressure pads equal in area, and (2) designing an alternate method to lubricate and cool the shaft bearings. For reversibility, the forward and aft housings (SKD387258H01 and G01 respectively) were easily changed as shown in Figure 3-6. However, for the bearing lubrication described earlier, more extensive modifications were required. The aft housing (SKD387258G01) was increased in length from 0.75 inches to 1.22 inches to allow for internal porting with restrictors and to allow for a center chamber with a plug. The restrictors provide a steady controlled flow of 4.3 cubic inches per minute across the bearings. The chamber provides an access to install the aft bearing. The rotor (SKD387259G01) was also modified to include central porting for flow to pass from the aft to the forward bearing. The axial slots in the shaft bearings (SKB376853H02) remained the same size as in the original Westinghouse/NCEL seawater motor. Finally, the side plate grooves for bearing lubrication in the original motor were eliminated in the new bearing plates (SKD387261H01).

Increased Operating Pressure

As described in Section 2.4, a spring (SKB376735H06) was designed to apply a 1.0 pound "mean" force to the vane (or a 2.0 pound outward radial force per vane) for a fatigue life of 250 hours at 1500 rpm. This "mean" force is applied when the vane is in the "half-stroke" position, i.e., when the vane is half-way between the major and minor diameters of the cam. See Appendix B for the spring design and analysis.

The final elgiloy spring wire size increased from a 0.018 to a 0.020 inch diameter, and the outside diameter increased from 0.125 to 0.135 inches, while the spring free length decreased from the proposed 0.875 inches to 0.614 inches. These changes in geometry and material resulted in a spring design with a margin of safety for fatigue of 14.6% and margin of safety against buckling of 24.5%.

As a result of spring size changes, the vane (SKC381517H01) width increased from 0.148 inches to 0.169 inches, and the rotor (SKD387259G01) slot increased proportionally.

Life

To extend motor life to 250 hours required a longer life spring, better wear materials, and elimination of excessive angular motion of the vanes in the rotor slots. From the previous discussion, the springs were designed for a 250-hour life at 1500 rpm barring any unforeseen wear problems. The critical motor components and their selected materials are listed below.

Components	Material
Vanes (10 each)	Torlon 4275
Springs (20 each)	Elgiloy
Rotor	Inconel 625
Cam	Inconel 625
Bearing Plates (2 each)	Torlon 4275
Housings (2 each)	Inconel 625
Shaft Bearings (2 each)	Torlon 4275
Alignment Pins (2 each)	Inconel 718

These materials were selected for their high resistance to corrosion, reasonable probability of long wear, low coefficient of friction, and availability. Finally, the rotor vane slot was redesigned for the wider vanes as shown in Figure 3-8 and as previously discussed in Section 2.4 (see Figures 2-5 and 2-6).

Impactor Drive

To mate the motor to the wrench handle and to the impactor required only the following two changes: (1) the porting was from one set of inlet and outlet ports (two sets, one set on each side were used for the original motor), and (2) the rotor shaft size was increased because of the high impact loads being directly transmitted to the rotor. The new porting arrangement is shown in Figure 3-5. The rotor shaft was changed from a 0.500 diameter pin drive to a 0.625 major diameter spline drive. A splined drive was chosen so that any external axial loads applied to the impactor during operation would not be transmitted to the rotor face and therefore to the motor bearing plates. The external spline on the rotor slides in the internal spline of the coupling. For a further discussion of the coupling, see Sections 4 and 5.

Performance Test Modifications

During the development phase of the tool motor design, the inner and outer slots in the bearing plates were increased from 26° and 40° to the calculated values of 30.6° and 45°, respectively (15.3° and 22.5° from the centerline). When this concept was tested in the original Westinghouse/NCEL motor, the motor was very hard to start (starting pressures of 330 psig versus 190 psig normally) and the motor would "knock" at lower pressures (knocking at 600 psig versus 1000 psig). Therefore, the bearing plate slots for the tool motor were changed back to 26° and 40° and were retested. The tool motor ran, but sounded "rough," as if just on the verge of "knocking" (see Appendix G for test data). During the performance tests, only the inner slots were opened to 30°, and the motor performed well. Therefore, for the final bearing plate design (SKD387261H01), the inner slot is 15.3° ± 0.5° either side of centerline or 30.6° ± 0.5° total.

Another modification incorporated into the final design during performance tests was the elimination of the spring pins that align the cam with the side plates (SKD387260G01). The spring pins "bowed" at their ends preventing assembly of bearing plates and housings. The spring pins were replaced with a solid, Inconel 718, alignment pin (SKB376859H01).

Finally, during the program development stages it was observed that the length of the Torlon 4275 vanes (SKC381517H01) would swell approximately 0.5 to 0.7 mils; and that for proper operation, the vanes should be 0.3 to 0.5 mils less than the rotor width. Therefore, the vane length was changed from $0.6240^{+0.0000}_{-0.0005}$ to $0.6237^{+0.0003}_{-0.0005}$ to allow for vane swelling.

Many design changes were made to the original seawater motor components. For a summary of these changes, see Table 3-3.

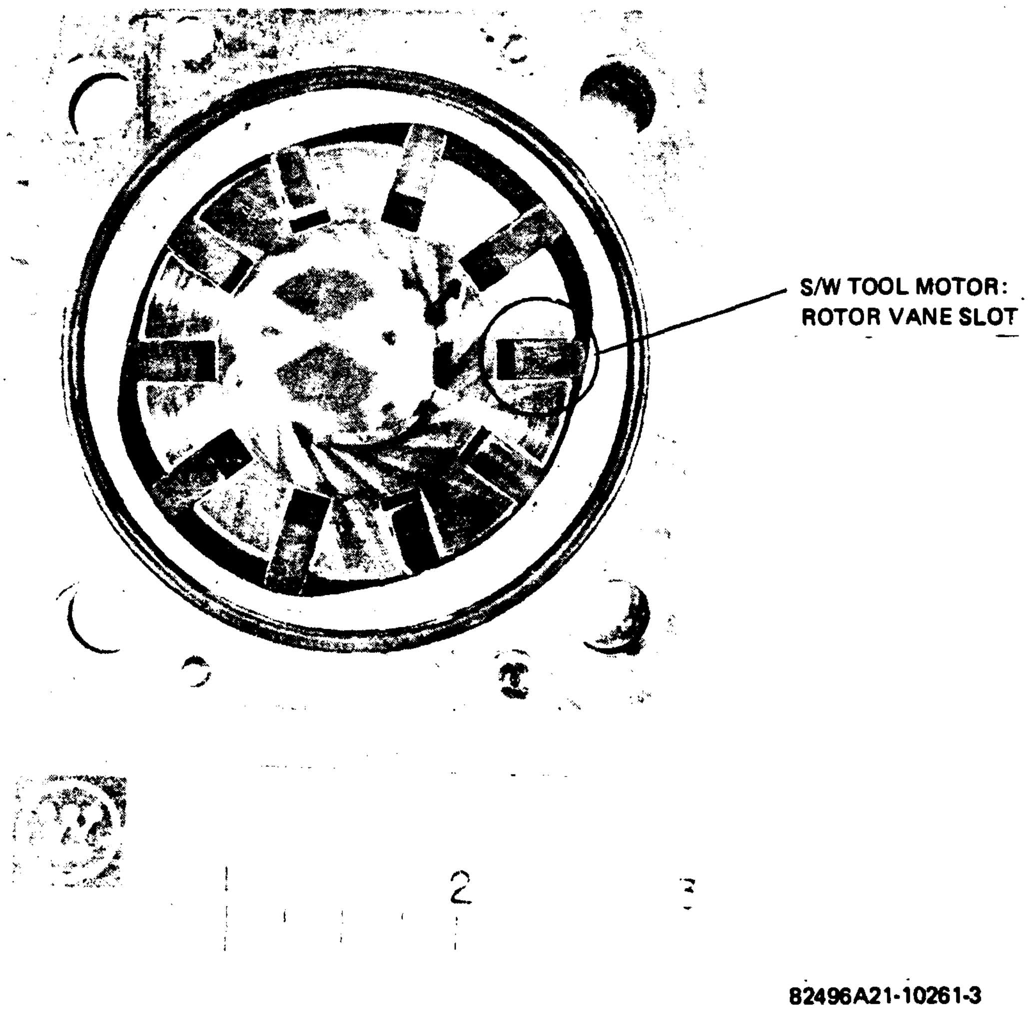


Figure 3-8. Final Rotor Vane Slot Configuration

Table 3-3. Summary Of Final Motor Component Modifications

COMPONENT	MODIFICATION	REASON FOR MODIFICATION
Springs SKB376753H06 SKB376735H07	● Eligiloy ● Larger wire diameter and outside diameter with shorter length ● Tiodized	● Excellent corrosion resistance and fatigue life ● Increased force with non-buckling and longer fatigue life ● Reduce wear effects
Vanes SKC381517H01	● Torlon 4275 ● 0.169 Width ● 0.03 tip chamfers ● 0.6237 length	● Excellent wear characteristics ● Longer O.D. springs ● Reversibility and wear-in ● Compensate for Torlon swelling
Rotor Bearings SKB376853H02	● Torlon 4275 ● 0.875 O.D. x 0.625 I.D.	● Excellent wear characteristics ● Larger rotor shaft diameter
Rotor SKD387259G01	● Vane slot configuration ● 0.625 shaft O.D. ● Splined shaft ● Internal porting	● Wider vanes and improved motor timing ● High impact loads ● Eliminate axial loads transmitted to motor ● Bearing lubrication
Cam SKD387260G01	● Pins	● Elimination of transfer drilling and matched set design
Bearing Plate SKD387261H01	● Torlon 4275 ● Elimination of bearing lube grooves ● Inner slots of 30.6°	● Excellent wear characteristics ● Improved bearing lube design ● Improved motor timing for wider vanes
Housings SKD387258H01 (Fwd) SKD387258G01 (Aft)	● Equal port pressure pads ● Porting from one side ● Restrictors ● Aft chamber	● Reversibility ● Wrench handle interface ● Bearing lube design ● Shaft bearing removal/ installation

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4.0 IMPACT WRENCH ASSEMBLY DEVELOPMENT AND DESIGN SELECTION

Westinghouse was contracted to develop a hand-held, diver operated impact wrench compatible with the seawater hydraulic motor, capable of useful and reliable work, and suitable for demonstration and evaluation of a seawater hydraulic tool system.

The impact wrench assembly combined an impactor of proven design with the improved version of the Westinghouse/NCEL seawater motor. Initially, an appropriate impact mechanism was selected and the motor/handle/impactor interface was specified allowing parallel development of the motor and wrench assembly designs. Design emphasis was placed upon developing a useful tool assembly while minimizing adverse effects of this application on seawater tool motor life.

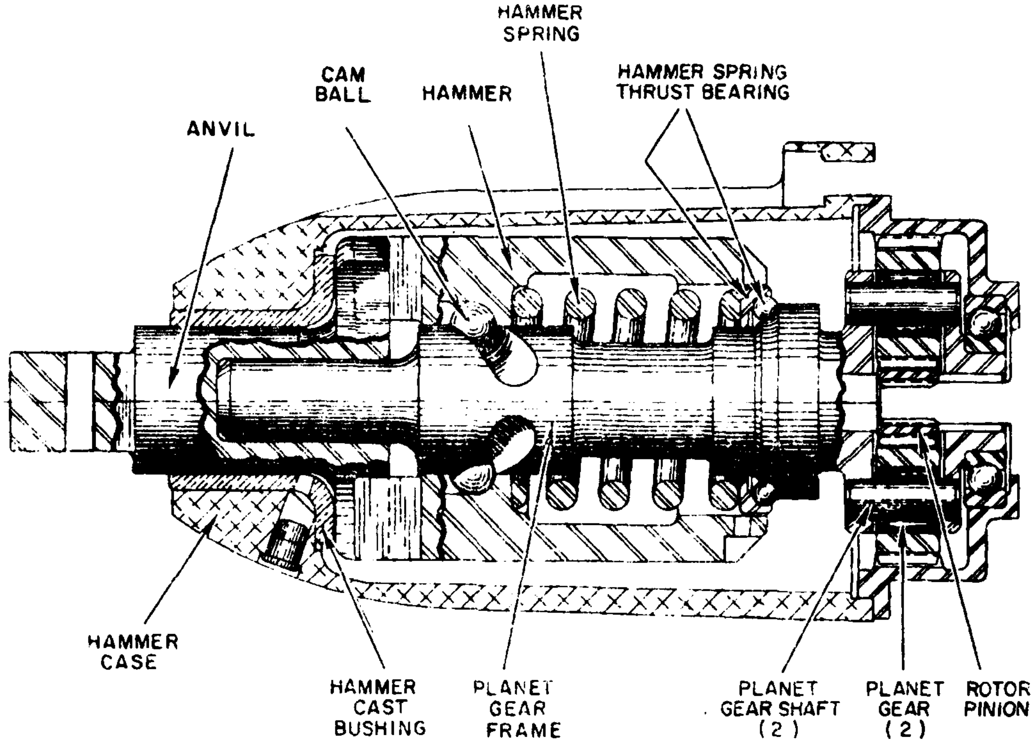
The sections that follow will discuss the development and final design selection of the major components which make up the overall impact wrench assembly. The major components are the impact mechanism, the coupling, and the handle.

Impact Mechanism

The primary concern during the wrench assembly development phase was the impact mechanism selection. The two impactors considered were the Ingersoll-Rand Model 2910 which was recommended for use in the Westinghouse proposal of 23 July 1980 and the Ingersoll-Rand Model 5100 which is used in the Stanley Model IWO6 impact wrench. Figure 4-1 illustrates the internal configuration of both impact mechanisms.

The Model 2910 incorporates two hammers which are rotated directly by the input shaft. The drive pins which rotate the hammers engage slots in the hammers so arranged to allow the hammer center to shift with respect to the input and output shaft centers. While rotating, centripetal force holds the hammers in the shifted position which allows the hammer to strike an anvil once per revolution. When the hammer engages the anvil, it imparts a majority of its kinetic energy to the output shaft and them cams off the anvil, permitting another input shaft rotation to begin. The two hammers strike their respective anvils simultaneously, generating a single impact per revolution. Note that the input shaft is directly coupled to the hammers forcing the drive motor to accelerate, then stop once per revolution. This device is ideally suited to motors having very rapid acceleration characteristics. This unit is designed to combine the motor rotor inertia with that of the hammers, increasing potential output torque of the motor/impactor assembly.

The Model 5100 varies significantly from the Model 2910 in the method of coupling motor and impactor. The motor is coupled through a planetary gearbox to a coil spring. The other end of the spring mates with the cylindrical hammer. As the motor rotates the impact shaft, the spring rotates, rotating the hammer. The hammer is in an axial position which allows engagement of the hammer dogs with the anvil dogs (which are attached to the output shaft). When these dogs engage, hammer kinetic energy is transmitted to the output shaft at high torque. As the motor continues to turn, it stores energy in the coil spring which is designed to reduce length as it is loaded. At a preset load, the hammer moves axially, away from the anvil, and the dogs disengage. Now, the spring transmits its potential energy to the hammer, accelerates and the cycle repeats, twice per revolution of the gearbox output shaft.



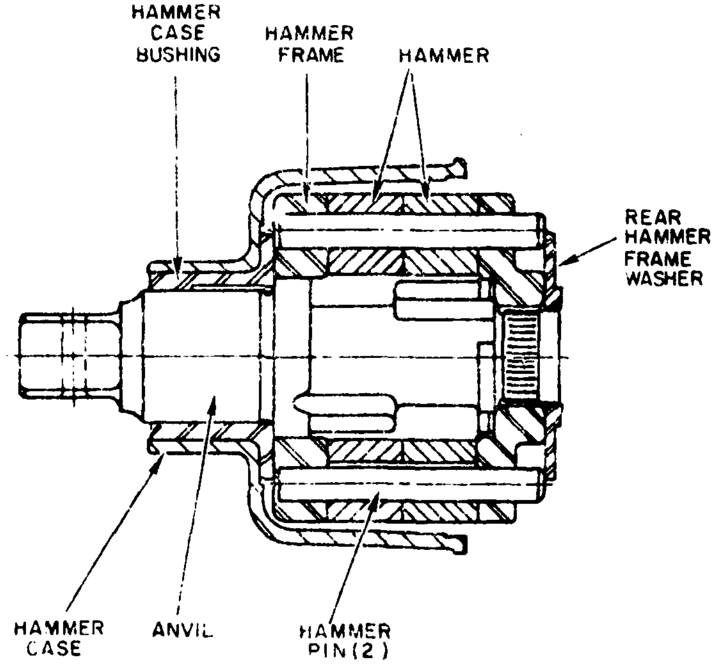


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Figure 4-1. Impact Mechanisms

The following comparison of operational parameters were considered prior to making the final impact mechanism selection:

1. It was calculated that under normal operating conditions, both the 2910 and 5100 impactors require less than 1 input horsepower to operate at rated torque. Ingersoll-Rand estimated that the maximum starting torque for the 2910 is less than 1 ft-lb. Measurements made with a 5100 impactor at Westinghouse indicated that breakaway torque for this unit should be less than 4 ft-lb. The original motor is capable of approximately 10 ft-lb shaft output. Even under flooded conditions, the tool motor should be capable of starting either unit. Although the impact input horsepower will increase for flooded operation, the seawater tool motor should have ample power margin to obtain full performance from either unit. This margin could be as high as 100%.

2. The 5100 does not use the motor rotor inertia while the 2910 does.

3. If the Model 5100 impactor were used, the seawater motor could be directly coupled to the mechanism drive spring by removing the planetary gears and substituting a suitable shaft mounting fixture. The overall performance of the impactor should be the same and the resultant tool should be quite suitable for seawater hydraulic tool demonstration.

4. The Model 5100 impactor is an obsolete design and production of this unit was stopped completely in 1980. The Model 2910 is of advanced design and offers a potential reduction in overall tool vibration levels in the axial direction (axial motion of the 5100 hammer results in oscillating axial forces).

5. Calculations have shown that the Model 5100 would require extensive modifications to overcome the effects of water flow and acceleration axially past the hammer, e.g., the impactor spring force would have to be increased approximately 25%.

6. During Impactor Tests at NCEL (see Section 8 for complete details), the Model 2910 was directly driven by the original seawater motor at rated rpm and impactor output torque with no noticeable degradation in motor performance or abnormal wear.

7. While driving the impactor directly, the motor must momentarily start and stop once per revolution with the 2910 while the motor rotates continuously with the 5100 mechanism. This results in a significant decoupling of impact shock to the motor with the 5100.

8. The shock loads delivered to the motor by the Model 2910 impactor are considered undesirable for the seawater motor which is sensitive to bearing loads.

9. The acceleration characteristics of the seawater motor may be marginal for the Model 2910 application knowing that the motor must stop and start once per revolution.

10. A flexible coupling could be designed, fabricated, and tested in further development programs for use with the Model 2910. By the addition of an intermediate coupler between the motor and impactor, the Model 2910 may be operated with a continuously turning motor shaft, significantly reducing rotary accelerations on the motor.

11. The motor may be required to start with significant initial torque load. The original motor did not operate as smoothly or efficiently at low speed and high torque as it did under normal, high speed operation. The motor has been observed to suddenly stop under low speed, high torque conditions. Use of

a compliant coupling with the Model 2910 impactor or use of the Model 5100 significantly reduces the risk involved in this new load application.

12. The spring force in the tool motor may have to be increased slightly to improve its low speed characteristics required for Model 2910 operation.

13. Driving the Model 2910 directly may cause shorter motor life because of high stresses from cavitation when motor suddenly stops.

14. A commercially available Torlon spline liner could be used between the rotor shaft and the rigid coupling to reduce shock loads from the impactor to the motor.

After reviewing the advantages and disadvantages of the above list, and after reviewing the impactor test results, the Model 2910 impact mechanism was selected to be used with a splined, rigid direct coupling. The development and design selection of the coupling will be discussed in the following section.

Two parallel design approaches were taken: (1) use of a rigid splined coupling (see Figure 4-2 a) in the impact wrench assembly and (2) use of a flexible coupling (see Figure 4-2 b). The flexible coupling was considered to reduce shock and rotary acceleration affects on the seawater tool motor. The major concern was that the seawater motor would stop completely after impact because of the motor's low speed characteristics. However, impactor tests at NCEL showed that the original motor would drive the Model 2910 impact mechanism via a rigid splined coupling. This test showed no visible signs of abnormal wear on motor components. Time did not permit running performance tests on the motor, so long term wear was not assessed.

Design calculations and analyses of the flexible coupling are contained in Appendix C. The rigid splined coupling is described in detail in Section 5.

Wrench Handle

The final development effort associated with the wrench assembly was the design or the selection of a compact tool housing. Early in the program a vendor was located that builds and markets a complete line of hydraulic hand tools for divers. Fairmont Hydraulics, a division of Fairmont Railway Motors, Inc., was very much interested in our seawater hydraulic work and worked with us to develop the final impact wrench handle. The preliminary handle design is shown in Figure 4-3. Fairmont was already in the process of switching from the Ingersoll-Rand Model 5100 Impactor (which was being discontinued) to the Model 2910 Impactor. The cast aluminum handle shown in Figure 4-3 is an integral part of their new design, and already included stainless steel on-off (no speed control) and reversing valves. The internal porting had to be modified to accommodate the two inlet and two outlet ports of the motor.

In addition to the porting modification, Westinghouse incorporated changes to improve operation in the seawater environment. To reduce the opportunity for corrosion of the aluminum area near the valves, the material of the valves was changed from stainless steel to Torlon 4301. The possibility of having the handles made of Titanium was investigated and found feasible. Titanium should be considered for future wrench assemblies. To eliminate other corrosion problems, the carbon steel mounting hardware was changed to stainless steel, and stainless, self-locking, helicoil inserts were installed in the housing. Hard anodizing was considered for handle protection, but was abandoned because of time and cost. This coating should be considered in the future.

To protect the motor, a dirt excluder was installed in the rotor shaft hole. The excluder will prevent contaminants from the ambient seawater from entering the motor via the motor bearing lube slots. The commercially available excluders have carbon steel housings so, for minimum protection, a light coating of "Lubriplate" was applied to the steel surfaces. These steel excluders can be replaced with special (long lead) 316 stainless steel or Inconel in the future.

Finally, to eliminate the transfer of thrust loads from the impactor to the motor and to add radial support to the rigid coupling, a Torlon 4275 radial thrust bearing was added to the handle extension/adapter housing. Detail drawings of the excluder and bearing are found in Appendix H and in the following section which discusses the wrench assembly detail design description and operation.

MATERIAL ALUMINUM – NICKLE – BRONZE, AMPCO 45

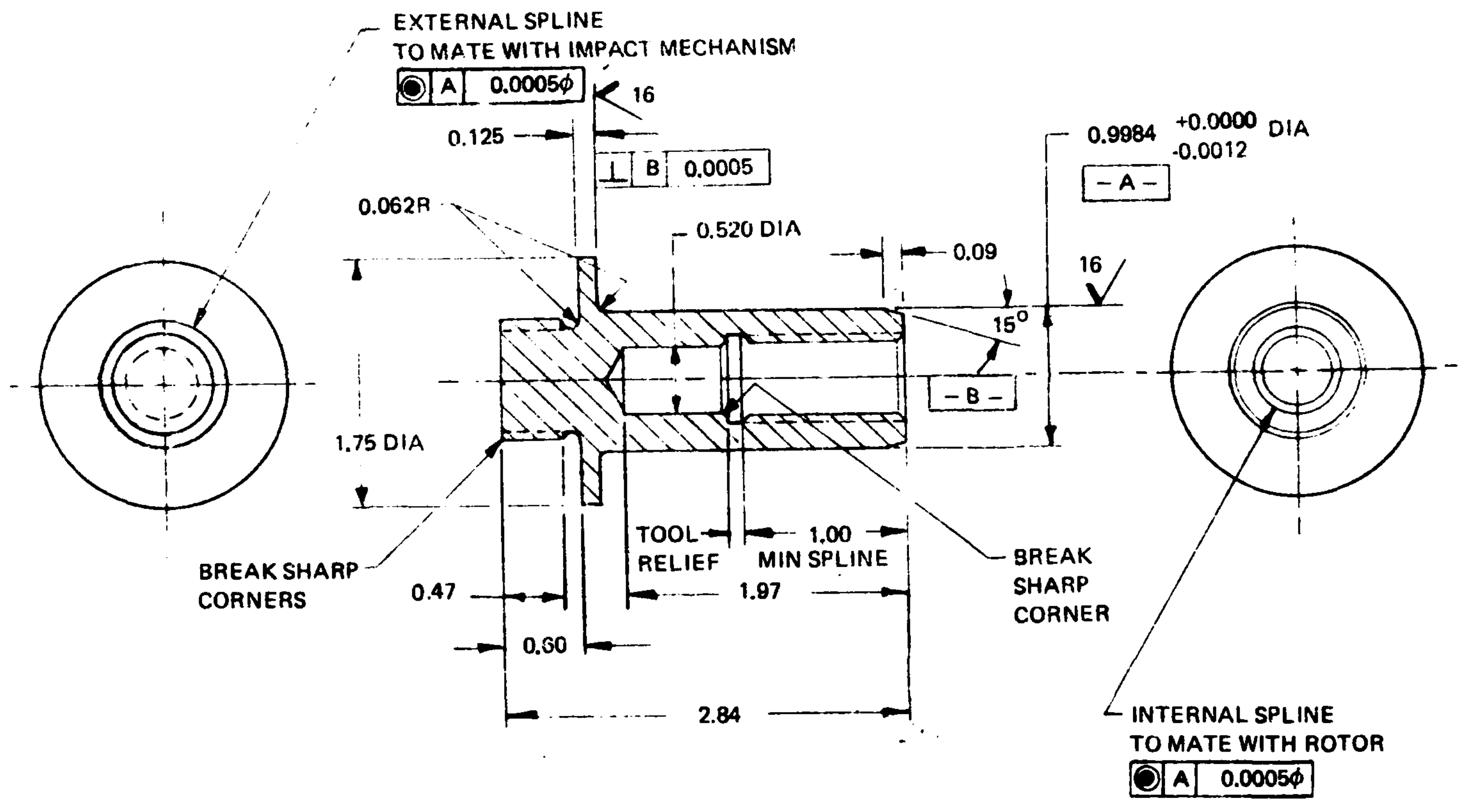

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Figure 4-2a. Rigid Coupling

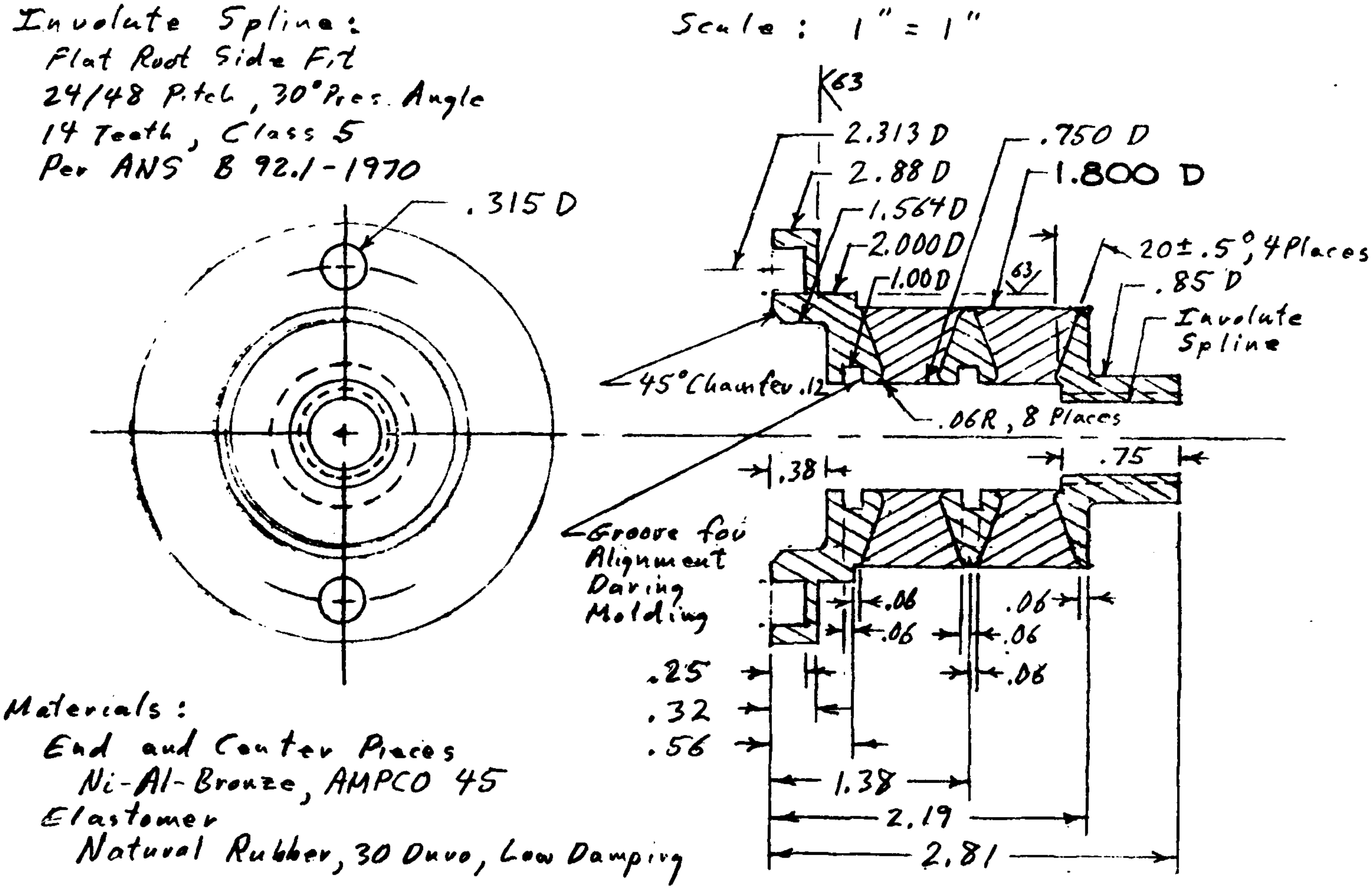

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Figure 4-2b. Elastomeric Coupling, Preliminary Design

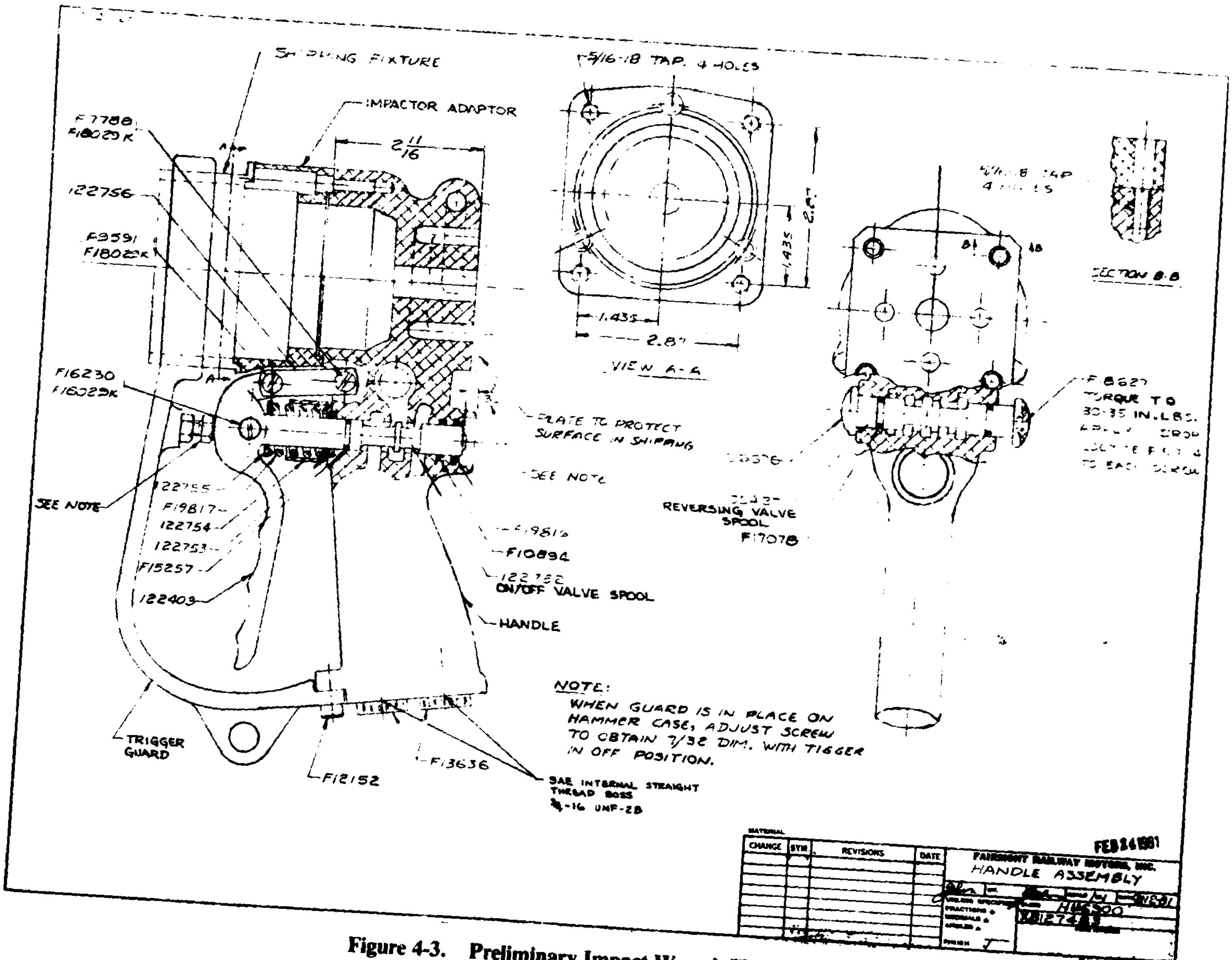


Figure 4-3. Preliminary Impact Wrench Handle

5.0 IMPACT WRENCH ASSEMBLY DESIGN DESCRIPTION AND OPERATION

The final Impact Wrench Assembly resulted from making minor improvements to the Ingersoll-Rand Model 2910 impact mechanism and the Fairmont Hydraulic handle assembly; and from designing and installing a splined rigid coupling to connect the impactor to the tool motor. This section describes the impact wrench assembly operation and final design.

5.1 Description And Operation

The diver tool is a reversible, seawater hydraulic impact wrench capable of delivering an output torque of 250 to 400 foot pounds at shaft speeds to 1000 rpm. The impact wrench is shown assembled in Figure 5-1 and disassembled in Figure 5-2. The in-air weight of the entire assembly is just 17.4 pounds.

Operation

The seawater impact wrench (see Figure 5-3) operates in the same general manner as an oil hydraulic wrench. The fluid (seawater) enters the supply port in the cast aluminum handle from the power source via a 250-foot umbilical and the 3-foot quick-disconnect hose assembly. Once the trigger is depressed, the high pressure fluid flows through internal porting in the handle, past the reversing valve, then to the seawater tool motor. In the tool motor (operation described in Section 3.0), the fluid energy is converted to mechanical energy, turning the rotor, which in turn drives the impact/mechanism via the Aluminum-Nickle-Bronze rigid coupling. Discharge flow from the tool motor enters the handle and exists via internal porting through the 3-foot pigtail. A filter assembly on the discharge side keeps ambient seawater contaminants from entering the handle and eventually the motor during nonoperating periods such as diver descent. The bearing lubricating flow from the tool motor exits the motor via the forward bearing and enters the handle through the modified dirt excluder. The dirt excluder prevents seawater contaminants from entering the tool motor also. Lubricating flow then passes through the radial/thrust coupling bearing into the impact mechanism. The bearing has lubricating grooves machined into it for the purpose of lubricating and cooling the bearing. The bearing provides radial support for the coupling and removes thrust loads the diver may place on the impact mechanism by bearing down on the work piece. The splines on the coupling and impactor also act as sliding surfaces to prevent transfer of thrust loads to the tool motor rotor. To enhance this sliding effect, the coupling was fabricated of Aluminum-Nickle-Bronze, a material with good self-lubricating and bearing properties. Finally, the flow exits the impact mechanism via the machined grooves in the forward impactor case bushing. A Torlon 4275 bushing is suggested as replacement for the "bar-stock" steel case bushing in the future. See Appendix H for a drawing of this replacement bushing.

The impactor tests performed at NCEL showed that, with an average input flow of 5.2 gpm and an average pressure of 460 psig, the impactor would produce 440 foot pounds output torque at 935 rpm. If this pressure and flow could be supplied to the wrench assembly, slightly smaller but similar output torques from the impactor can be expected because of the pressure drop in the handle porting.

Westinghouse made several improvements to the major components in the wrench assembly. These improvements are described in the following sections.

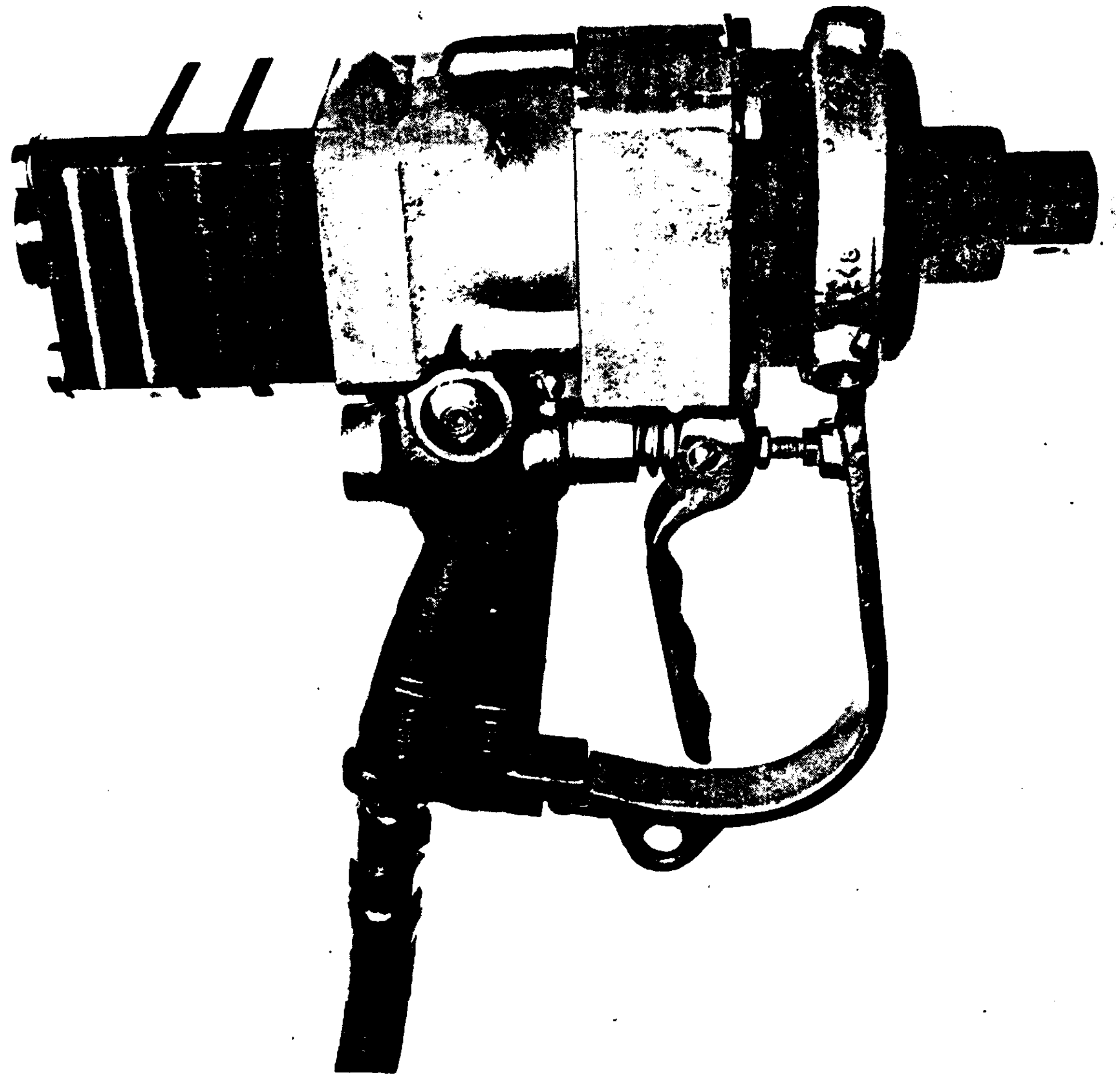


Figure 5-1. Impact Wrench Assembly

81496A32-10268-81

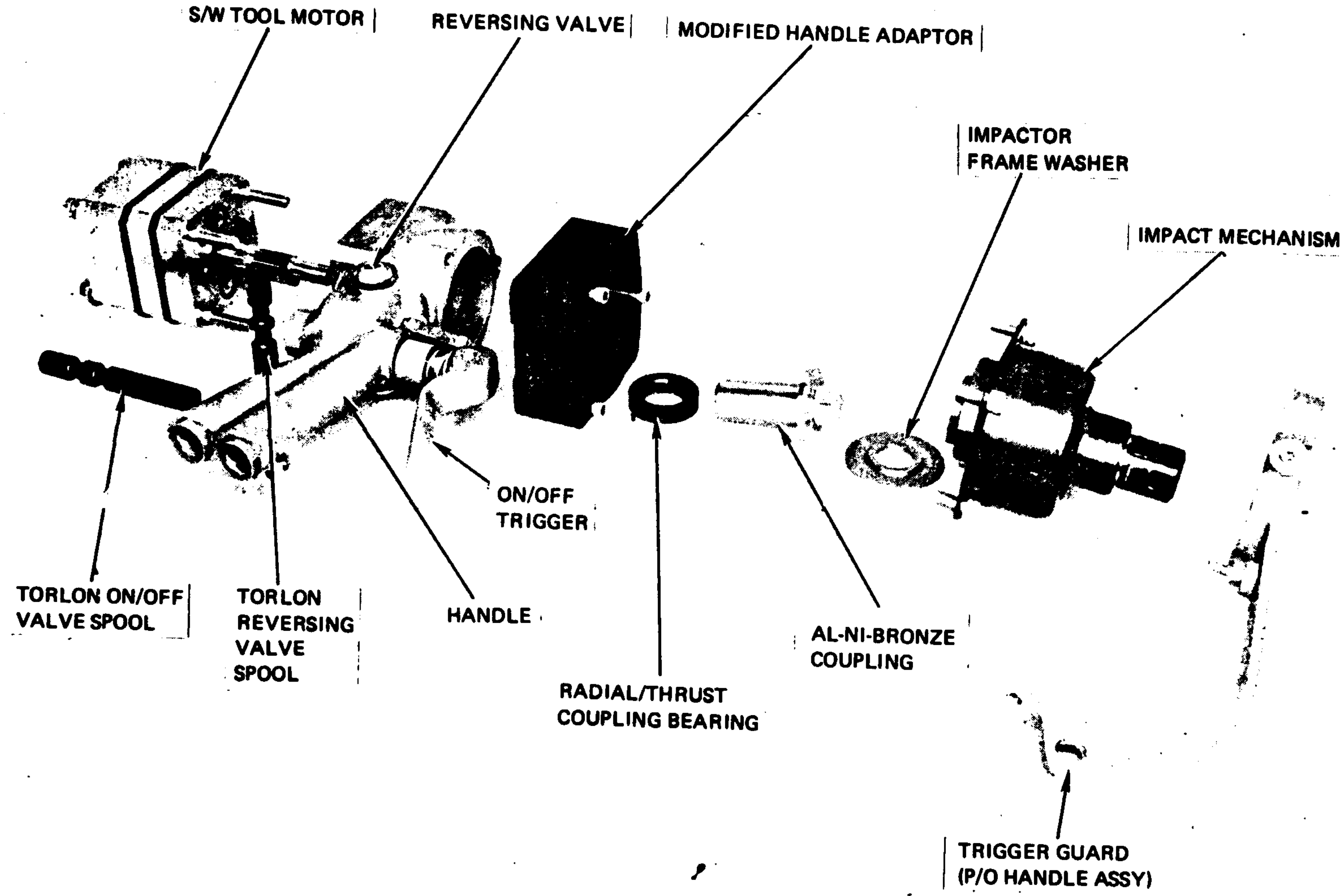


Figure 5-2. Impact Wrench Assembly (Disassembled)

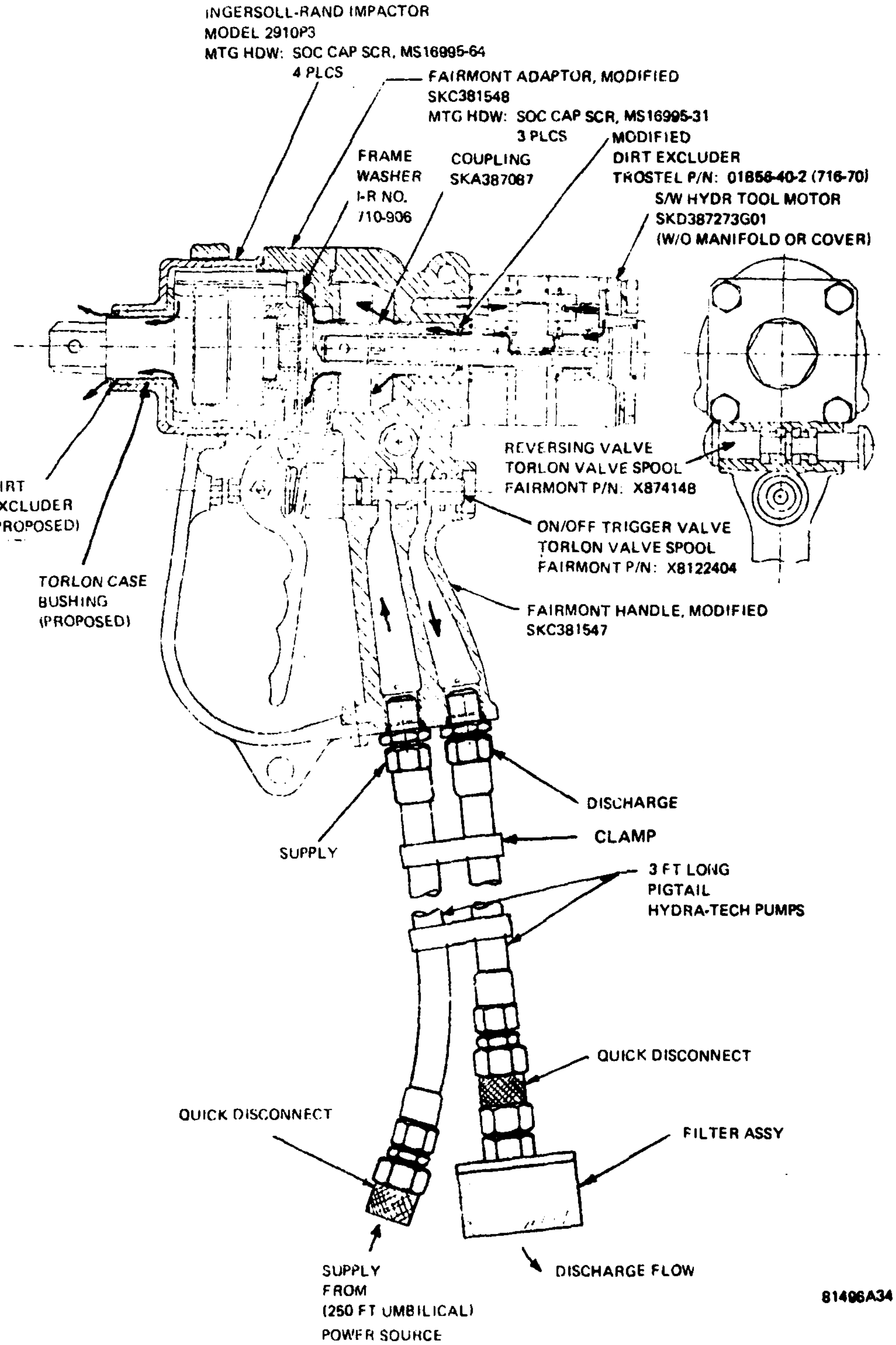


Figure 5-3. Wrench Assembly

5.2 Impact Mechanism Improvements

An off-the-shelf Ingersoll-Rand Model 2910P3 impactor (see Figure 5-4) was procured for use in the diver tool impact wrench assembly. Only minor modifications were made to the impactor:

1. For corrosion protection, all the impactor parts were coated with "Lubriplate" - an extreme pressure type lubricant composed of zinc oxide grease and lithium soap base developed by Fiske Bros. Refining Co. in Newark, N.J. Lubriplate contains an oxidation inhibitor and repels water. The parts were immersed in boiling (approximately 400°F) Lubriplate for increased penetration and corrosion protection.

2. One complete impact mechanism was coated with Aluminite "Z" - a corrosion inhibitive organic based, aluminum-filled coating developed by Tiodize Co., Inc. of Huntington Beach, California. However, during impactor tests performed at Westinghouse Oceanic Division, some of the coating chipped off. We then removed the coating by sandblasting and had the parts "Lubriplated".

3. A frame washer, Ingersoll-Rand P/N: 910-706, was not part of the impactor assembly, but is required for the wrench assembly. The purpose of the frame washer is to keep the hammer pins (two each) in place during impacting. The frame washers were also coated with Lubriplate.

4. The steel case bushing was modified by machining existing lubricating grooves the entire length of the bushing. This allows tool motor lubricating flow to exit at the impactor. A Torlon bushing (see Figure 5-5) with a dirt excluder is suggested for future development programs.

5.3 Coupling Design

The final coupling design is shown in Figure 5-6. AMPCO 45, extruded aluminium-nickle bronze, was selected as the spline material because of its good corrosion resistance properties, exceptional wear resistance properties, high strength, extra toughness, and good bearing qualities. The bearing properties of this material were required to allow the rotor spline to slide in the coupling and to allow the coupling spline to slide in the impactor spline. This sliding action minimizes transfer of thrust loads from the diver workpiece to the seawater tool motor. The high strength and extra-toughness of this material was required to take the high shock loads from the impactor. The estimated life of this coupling is approximately 7500 hours at 1000 rpm.

The purpose of the 1.75 inch diameter coupling flange is to serve as a thrust washer - taking loads imposed by the diver bearing on the workpiece and transfering those loads to the handle rather than to the tool motor.

To allow lubricating flow into spline teeth, and to decrease the torsional spring constant for better tool motor acceleration characteristics, four 0.25-inch diameter holes can be drilled into the coupling at a later date for further evaluation.

Design calculations and analysis of the rigid splined coupling are found in Appendix B.

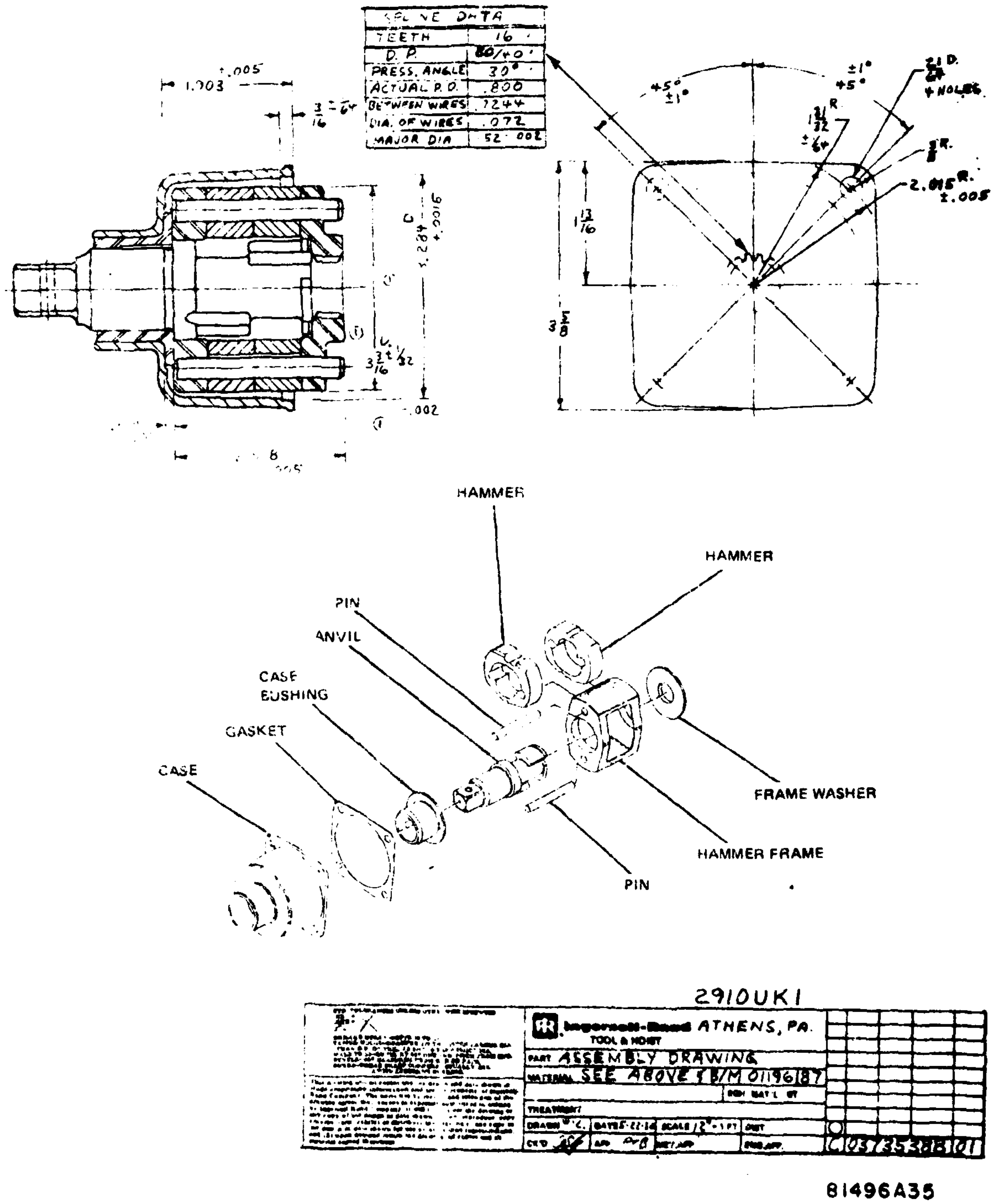


Figure 5-4. Model 2910 Impact Mechanism

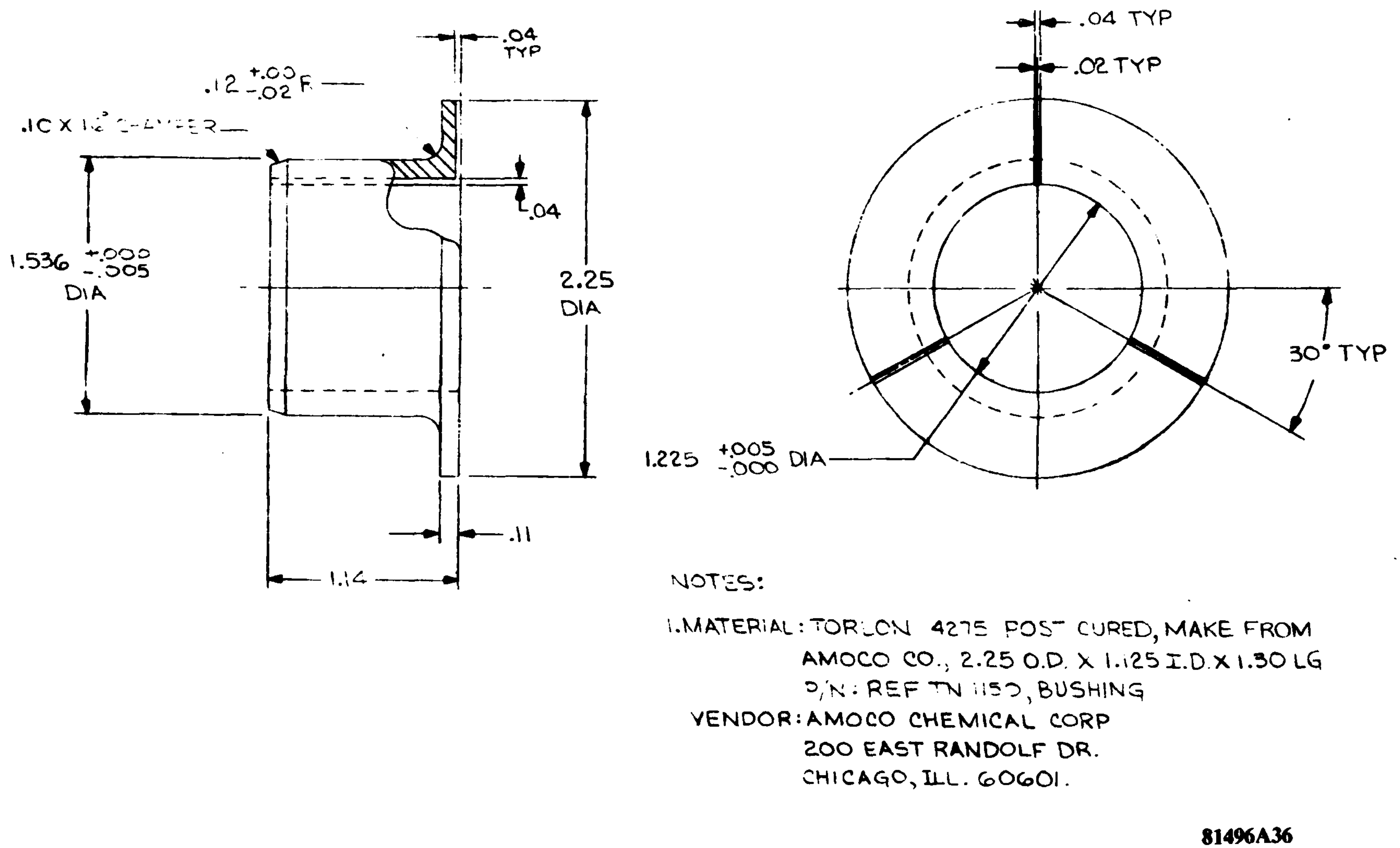


Figure 5-5. Proposed Torlon Impactor Case Bushing

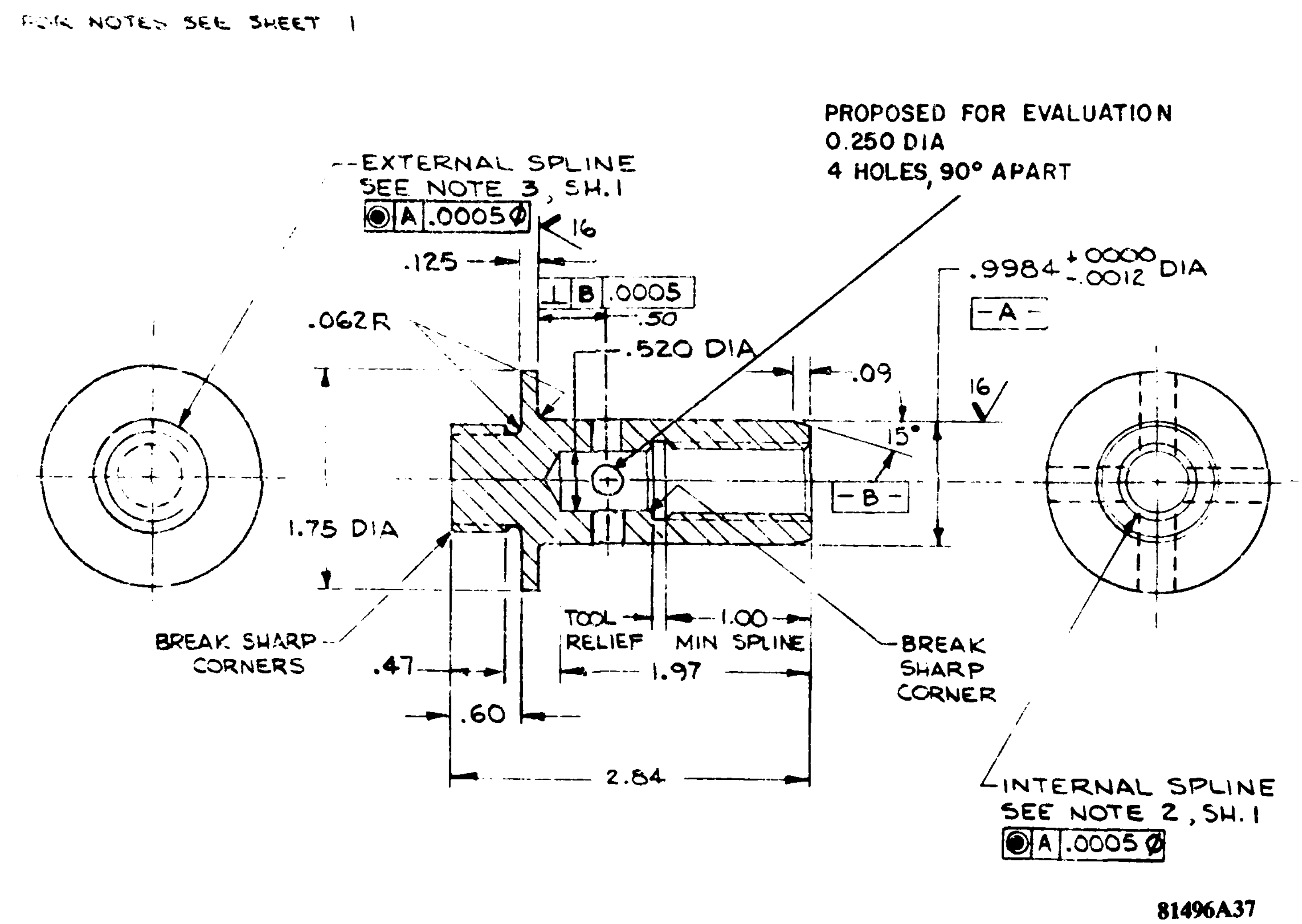


Figure 5-6. Rigid Splined Coupling

5.4 Handle Improvements

The final handle design for the impact wrench assembly, shown in Figure 5-3, resulted from incorporating the improvement modifications specified in Section 4.0. The following describes the handle and the incorporated changes:

1. The unmodified handle assembly is shown in Figure 5-7, with the aluminum alloy casting spec sheet shown in Figure 5-8. The handle was modified by enlarging the rotor shaft through-hole, machining a support land for the dirt excluder, and installing stainless steel, self-locking helicoils. These modifications are shown in Figure 5-9, with the dirt excluder shown in Figure 5-10.

2. The handle impactor-adaptor housing was modified to include a radial/thrust bearing to radially support the rigid coupling and to eliminate thrust loads to the tool motor. The modifications are shown in Figure 5-11 with the Torlon bearing shown in Figure 5-12.

3. To minimize corrosion due to dissimilar metals in seawater, the stainless steel spool valves for the on-off trigger valve and the reversing valve were replaced by Torlon 4301 spools supplied by Fairmont Hydraulics. The Torlon valves are shown in Figure 5-13 and 5-14 respectively. Further modifications to the valve spools were incorporated to produce the proper valving action. These changes included machining the two O-ring grooves in the reversing valve spool, machining the outside diameters of both spools to 0.5620 inches, and enlarging the on/off spool valve land from 3/32" to 5/16" to eliminate the constant flow through the handle while not operating the impactor.

5.5 Pigtail Hose Assembly

A special pigtail hose assembly is required to connect the power source 250-foot umbilical to the wrench assembly. The Pigtail Hose Assembly is shown in Figure 5-3 with details in Figure 5-15. For laboratory tests, the filter assembly can be removed and the discharge hose can be connected to an outlet. In the field, the filter keeps ambient seawater particles from entering the motor under nonoperating conditions such as diver descent. The 250-foot umbilical attaches to the quick-disconnect on the supply side of the pigtail assembly.

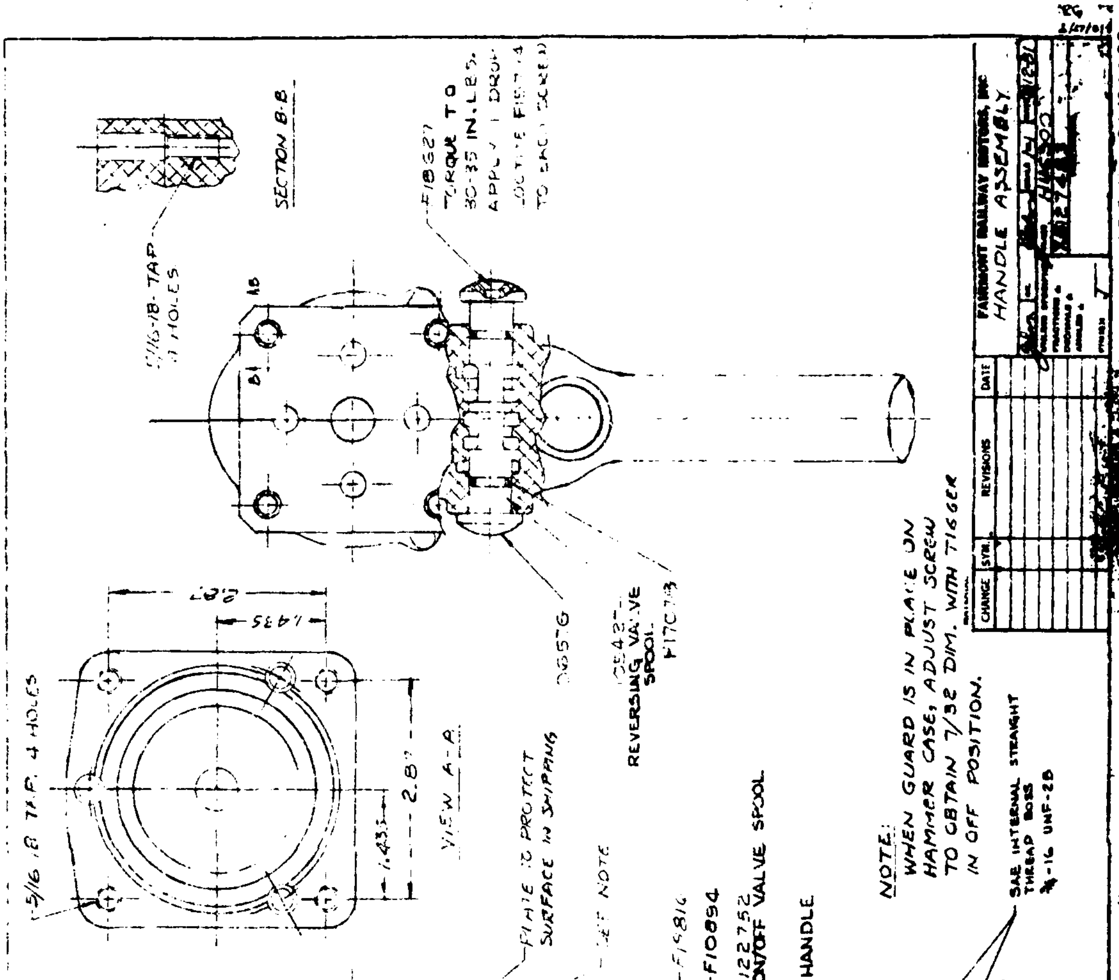


Figure 5-7. Handle Assembly

(B) CASTING ALLOY 357

COMPOSITION

Copper		0.05 Max.
Zinc		0.05 Max.
Iron		0.15 Max.
Silicon	6.5 -	7.5
(A) Magnesium	0.45 -	0.60
Manganese		0.03 Max.
Titanium		0.20 Max.
Other Elements Each		0.05 Max.
Other Elements Total		0.15 Max.
Balance	-	Aluminum

RECOMMENDED HEAT TREATMENT

12 Hrs. @ 1000° F.

Quench in Water 150° - 212° F

Age 3-5 Hrs. @ 310° F.

Typical Values
Tensile Strength 50,000 PSI
Tensile Yield 43,000 PSI
Elong. (% in 2") - 2

NOTE: AV. COEF. OF EXPANSION
68° F. - 212° F. = .0000119
68° F. - 392° F. = .0000127
68° F. - 572° F. = .000013

HEAT TREATED T6 TEMPER
BRINELL HARDNESS
500 KG LOAD 10 MM BALL-80-100

MAY 1 1981

RETYPED 3-21-78

MATERIAL

CHANGE	SYM.	REVISIONS	DATE
40839			
43982		CHANGE SPECS	1-72
44554	A	WAS 0.30-0.40	2-73
46818	B	CHANGE SPECS	3-78

FAIRMONT RAILWAY MOTORS, INC.

CAST ALUMINUM

DR JAK | CK. | APP. | SCALE | DATE 1-27-72

UNLESS SPECIFIED
FRACTIONS ±
DECIMALS ±
ANGLES ±
FINISH √

CLASS X

PART NUMBER F14043

F14043

Figure 5-8. Handle Assembly Casting Alloy Spec Sheet 81496A39

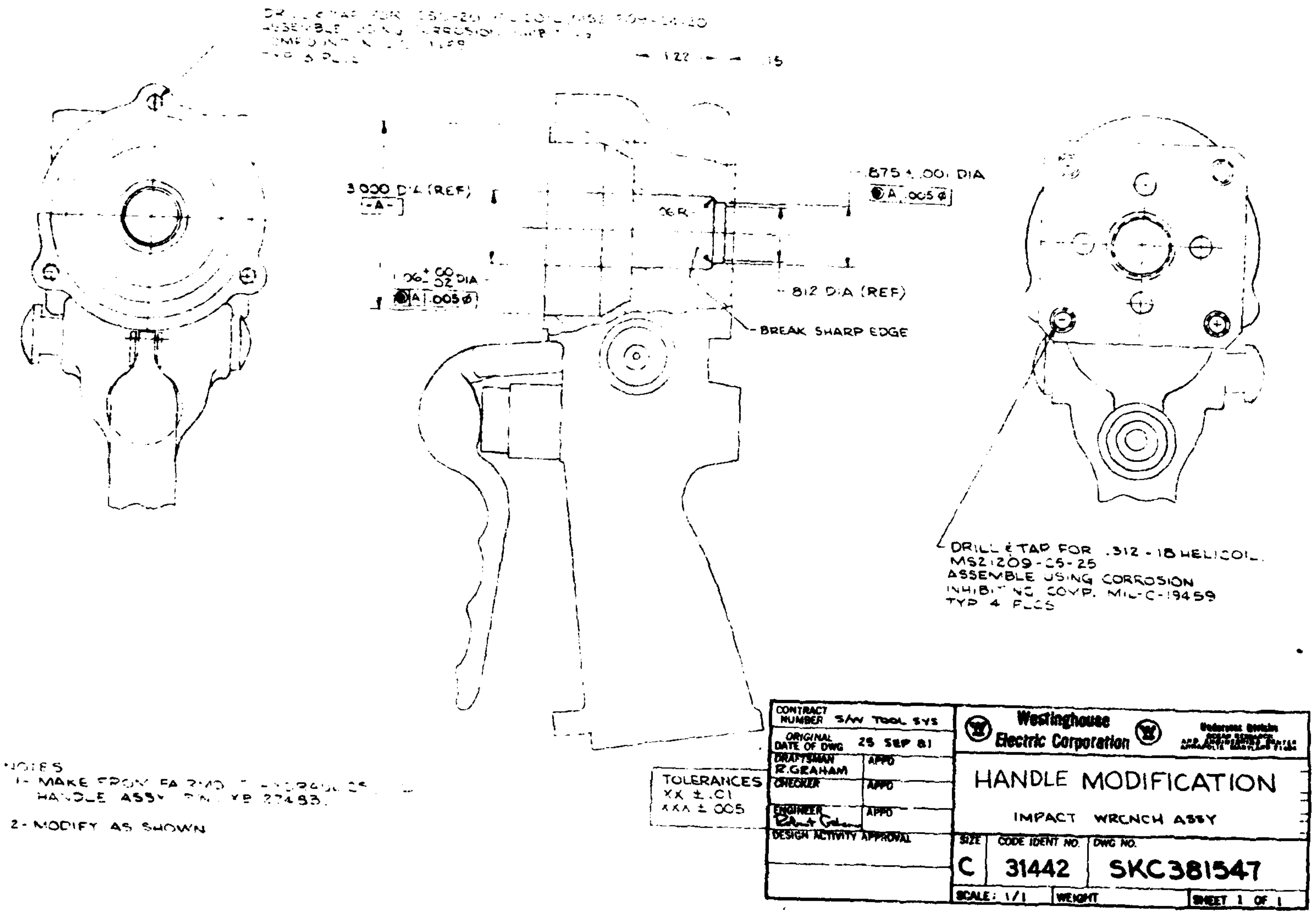
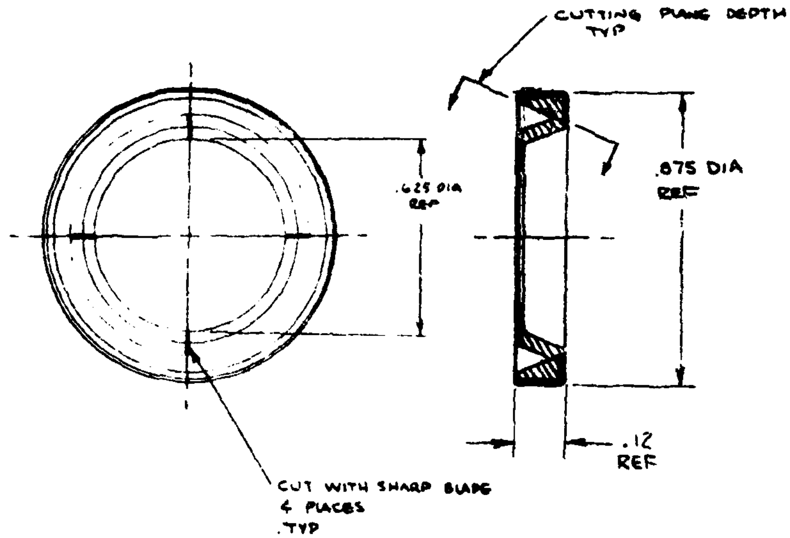


Figure 5-9. Handle Modifications

81496A40

NOTES:

1 - MODIFY AS SHOWN
2 - MAKE FROM TROSTEL TYPE B SEAL
P/N: 01856-40-2
3 - SUGGESTED SOURCE OF SUPPLY:
ALBERT TROSTEL PACKINGS, LTD.
LAKE GENEVA, WIS. 53147
(414) 248-4481



MODIFIED DIRT EXCLUDER

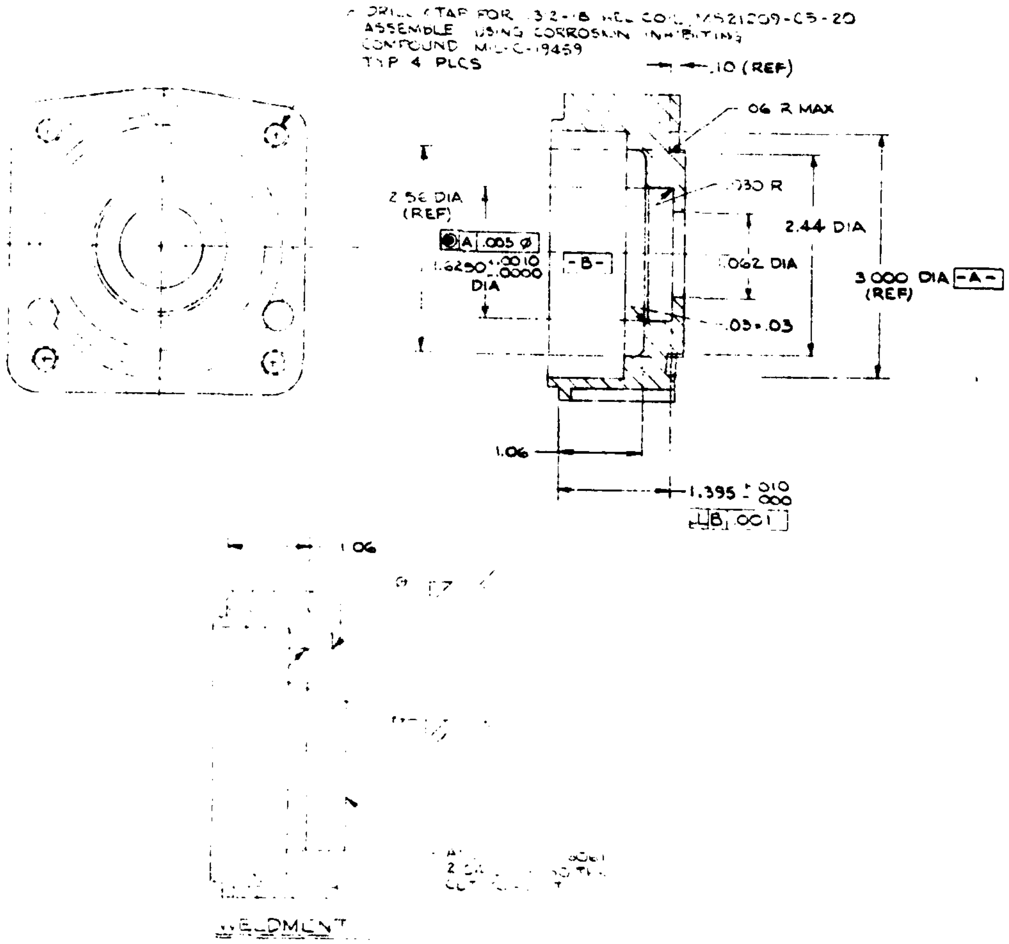
10/26/81

81496A41

Figure 5-10. Dirt Excluder

NOTES:
1. MAKE FROM FAIRMONT HYDRAULICS INC. HANDLE ASSY P/N: 7B127485

AS SHOWN



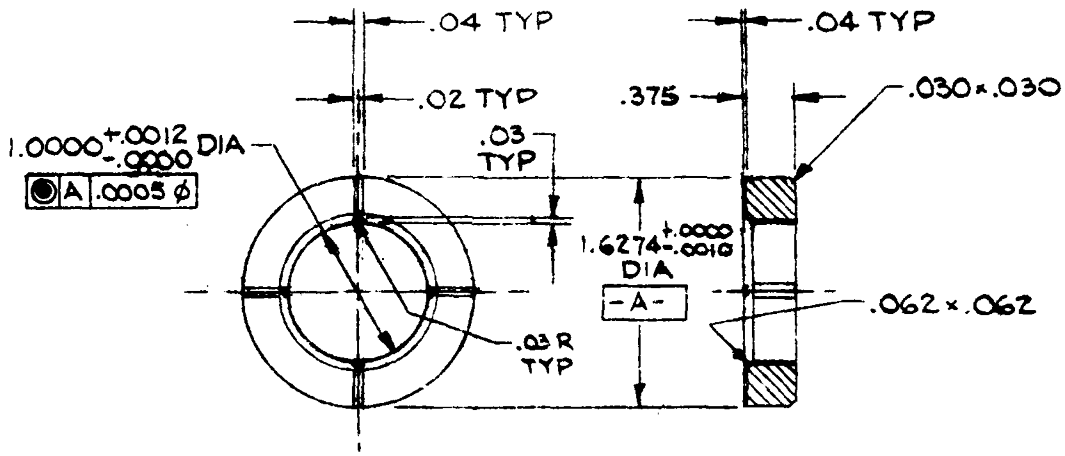
TOLERANCES	
.XX ±.01	
.XXX ±.005	

CONTRACT NUMBER	SAW TOOL SYS	Westinghouse Electric Corporation
ORIGINAL DATE OF DWG	25 SEP 81	
DRAFTSMAN R. GRAHAM	APPD	HANDLE EXTENSION MOD
CHECKER	APPD	IMPACT WRENCH ASSY
ENGINEER	APPD	SIZE C · CODE IDENT NO 31442 · DWG NO. SKC381548
DESIGN ACTIVITY APPROVAL		SCALE: 1/1 · WEIGHT · SHEET 1 OF 1

81496A42

Figure 5-11 Handle Adaptor Modification

NOTES:
1 – MAT'L: TORLON 4275, MAKE FROM ATP-8-2 DISC,
3.9" DIA × .52 THK
VENDOR: AMOCO CHEM. CORP
CHICAGO, ILL.

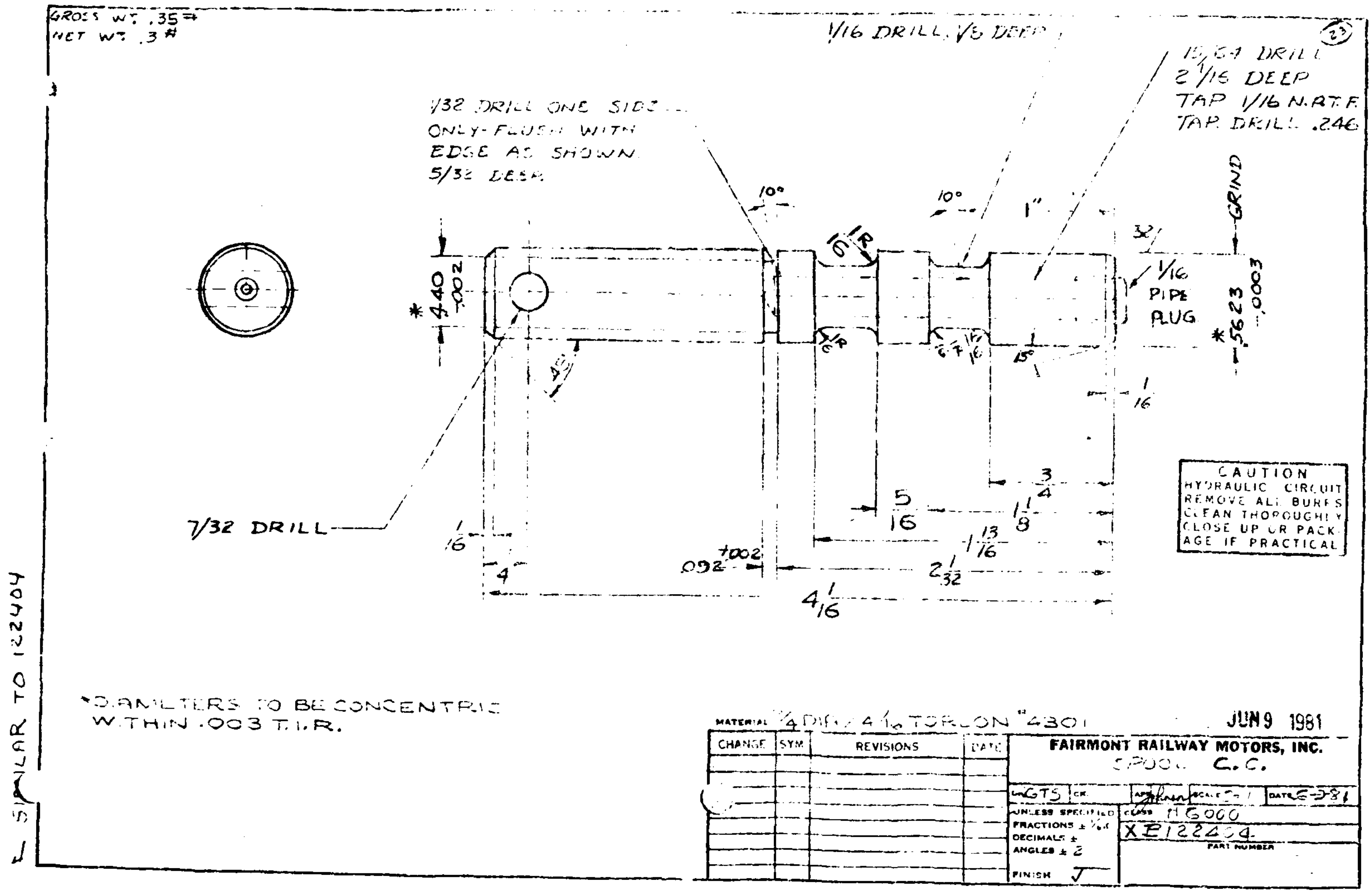


CONTRACT NUMBER S/W TOOL SYS	Westinghouse Electric Corporation — Underseas Division, Ocean Engineering and Development Center, Annapolis, Maryland 21404
ORIGINAL DATE OF DWG SEP 25 1981	
DRAFTSMAN R GRAHAM / APPD	BEARING, HANDLE IMPACT WRENCH ASSY
CHECKER / APPD	
ENGINEER 9/25/81 R. Graham / APPD	
DESIGN ACTIVITY APPROVAL	SIZE A / CODE IDENT NO. 31442 / DWG NO. SKA387088
	SCALE: 1/1 / WEIGHT / SHEET 1 OF 1

TOLERANCES
XX ±.01
XXX ±.005

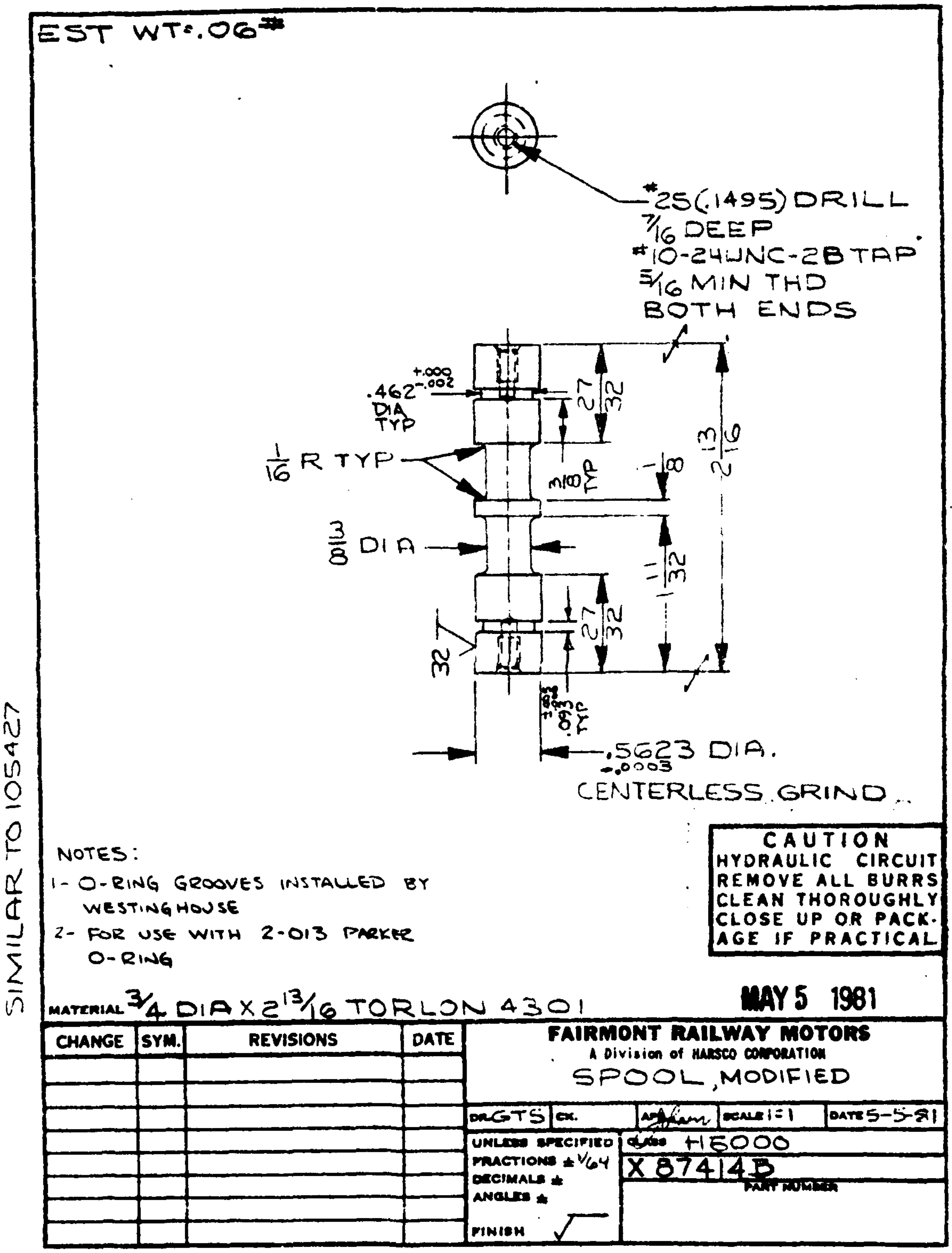
81496A43

Figure 5-12. Axial/Thrust Bearing



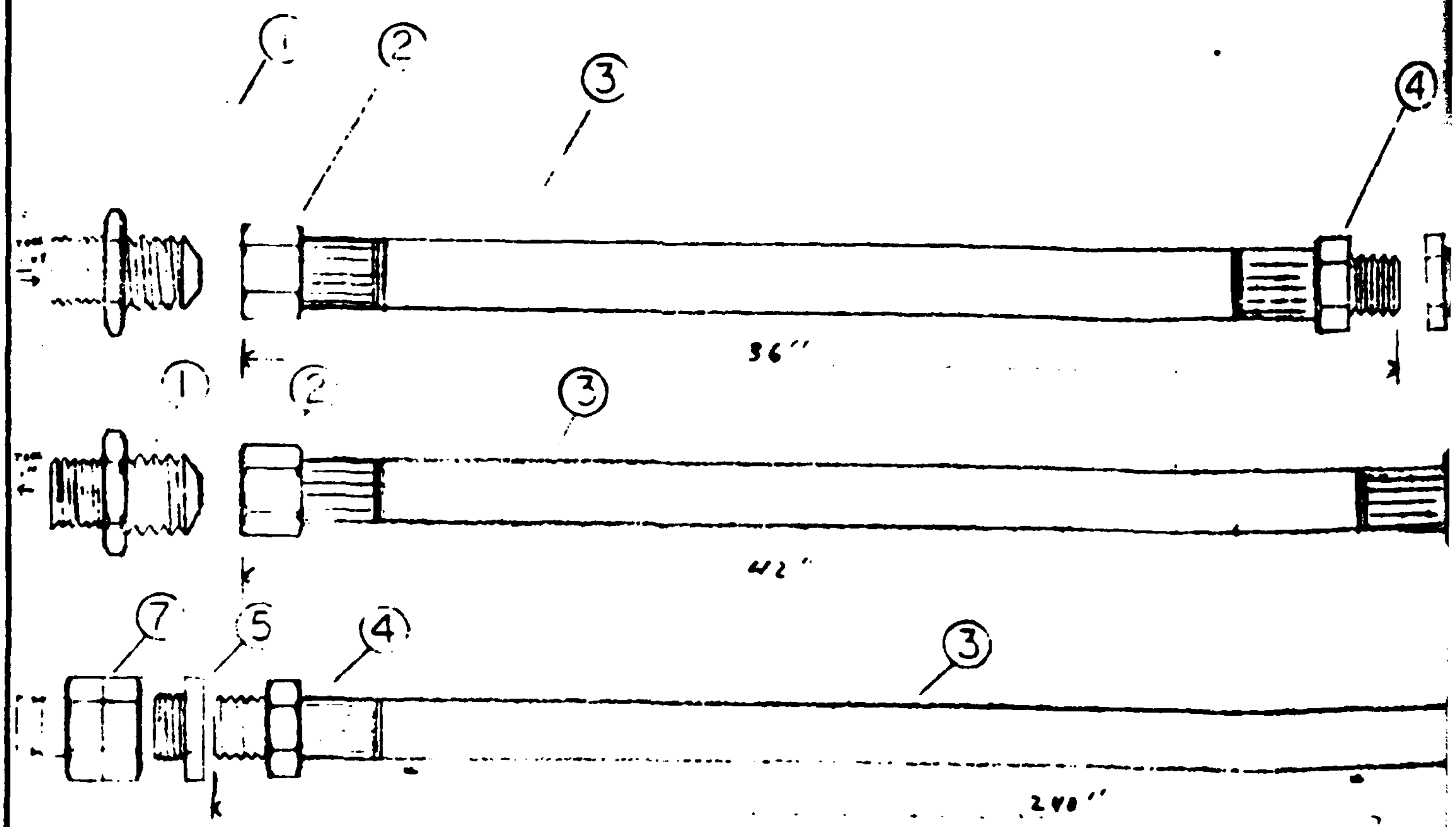
81496A44

Figure 5-13. On/Off Trigger Valve

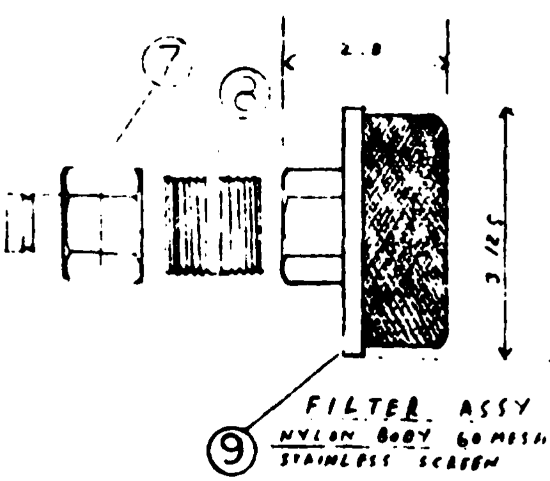


81496A45

Figure 5-14. Reversing Valve Modification



1	ADAPTOR 1/2" JIC MALE TO 3/4 -16UNF PART # CF50X-SS
2	HOSE END 1/2" JIC FEMALE SWIVEL PART # 39[illegible]L-08548
3	HOSE 1/2" SYNFLEX 3500 PSI W/P PART # 3R80-08
4	HOSE END 1/2" NPT RIGID PART # 390L-08108
5	REDUCING BUSHING 3/4" x 1/2" S.S.
6	FEMALE COUPLER 1/2" SNAPTITE PART # SVHC12-12F/SL
7	MALE COUPLER 3/4" SNAPTITE PART # SVHN12-12F
8	PIPE NIPPLE 3/4" S.S.



FILTER ASSY
NYLON BODY 60 MESH
STAINLESS SCREEN

NOTE: CLA[illegible]

FIG-R24 FILTERS

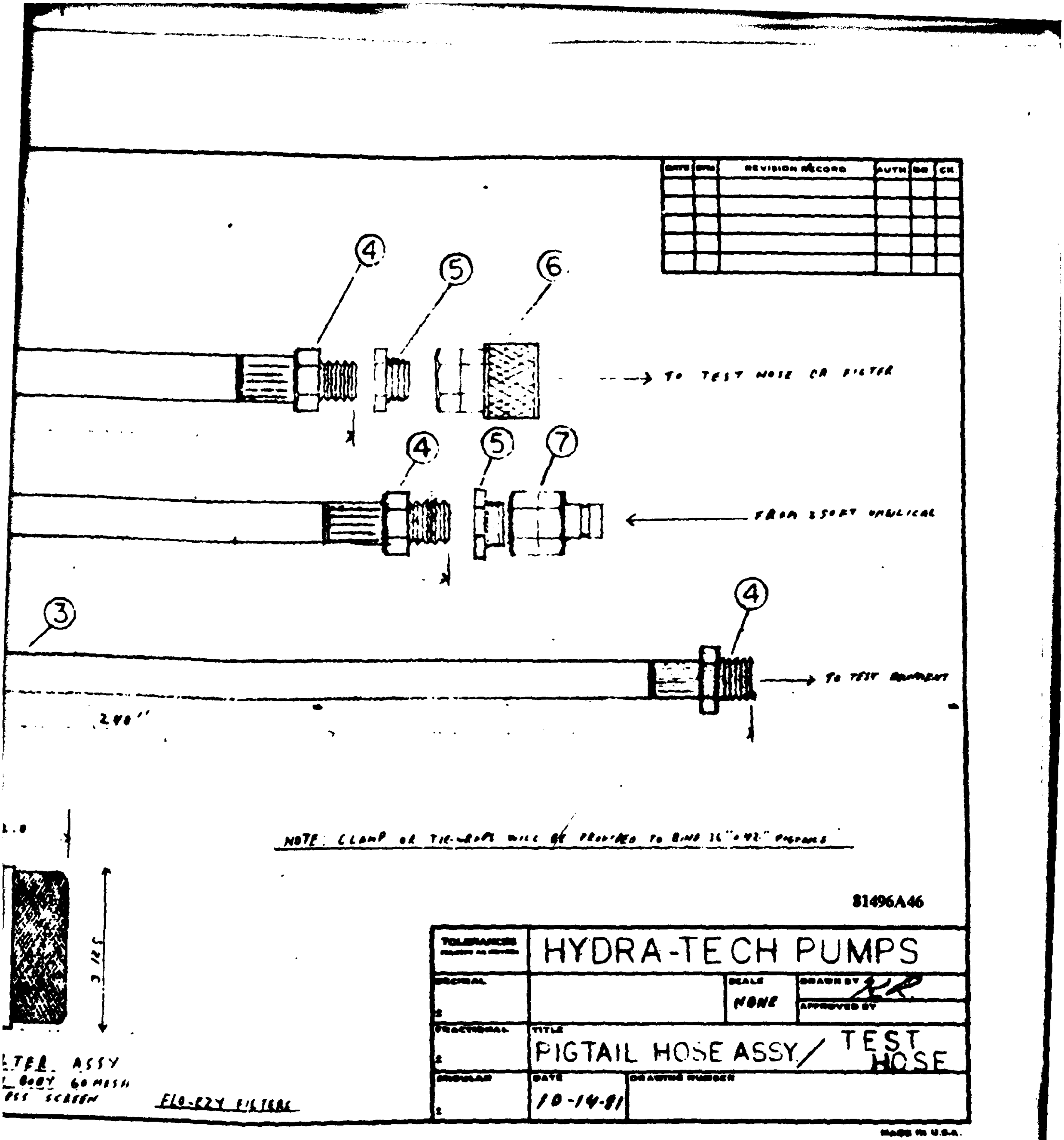


Figure 5-15. Pigtail Hose Assembly

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6.0 POWER SOURCE DESIGN APPROACH

6.1 Introduction

This section presents presents the design approach, general description, basic requirements, and design results for the seawater hydraulic power source (SWHPS). Design details follow in Section 7.0.

The objective of this design is to provide a versatile and reliable seawater power source capable of providing seawater hydraulic power to a variety of diver-operated tools.

6.2 Approach

Detailed power source physical and operational requirements based upon contract and the overall system requirements were developed. Available off the shelf subassemblies and detailed requirements for the operation of these units were identified.

The schematic presented in the RFP was used as a starting point for SWHPS design. Care was taken to ensure that the design had sufficient flexibility to power a broad spectrum of hydraulic tools.

6.3 General Description

The SWHPS uses seawater as a working fluid. It draws raw seawater up to the deck of a ship or land based site as a source of working fluid. The SWHPS then filters the seawater, pressurizes it, and delivers it as hydraulic power to a diver operated tool working up to 250 feet away. The SWHPS uses the closed-center-open-ended hydraulic circuit shown in Figure 6-1, the seawater flow schematic. The closed-center design reduces the system filter load. Power for the SWHPS is provided by an air cooled diesel engine. A photograph of the SWHPS is presented in Figure 6-2.

6.4 Performance Requirements

The primary requirements for the power source are listed below:

- Working Fluid: Seawater drawn from sea surface at 15-foot suction head.
- Discharge Pressure: to 2,500 psi
- Flow: Adjustable to 12 gpm
- Engine: Diesel
- Configuration: Trailer-mounted for California highway service with appropriate fittings for crane handling and shipboard tiedown.
- Supply umbilical length: 250 ft.
- Gauges and Meters:
  - Engine oil pressure per manufacturer recommendation
  - Engine oil temperature: 0 to 230°F.
  - Seawater supply line pressure: 0 to 3,000 psig
  - Seawater supply flow rate: 0 to 15 gpm

- Pressure and flow ripple: less than 2%
- Filtration: as determined in this design

6.5 Summary of Results

- The design should satisfy all the requirements of flow, pressure and configuration.

- The contamination analysis yielded a requirement to remove all particulate above 15 micron. A duplex filter arrangement was selected to satisfy the requirement.

- Component construction is dominated by the use of 316SS. This material is the most corrosion resistant material offered as an option by component suppliers.

- A maximum heat rejection capability of 44,000 BTU/HR is included in the design. This allows the pump to deliver full power without damage even if the tool is not attached.

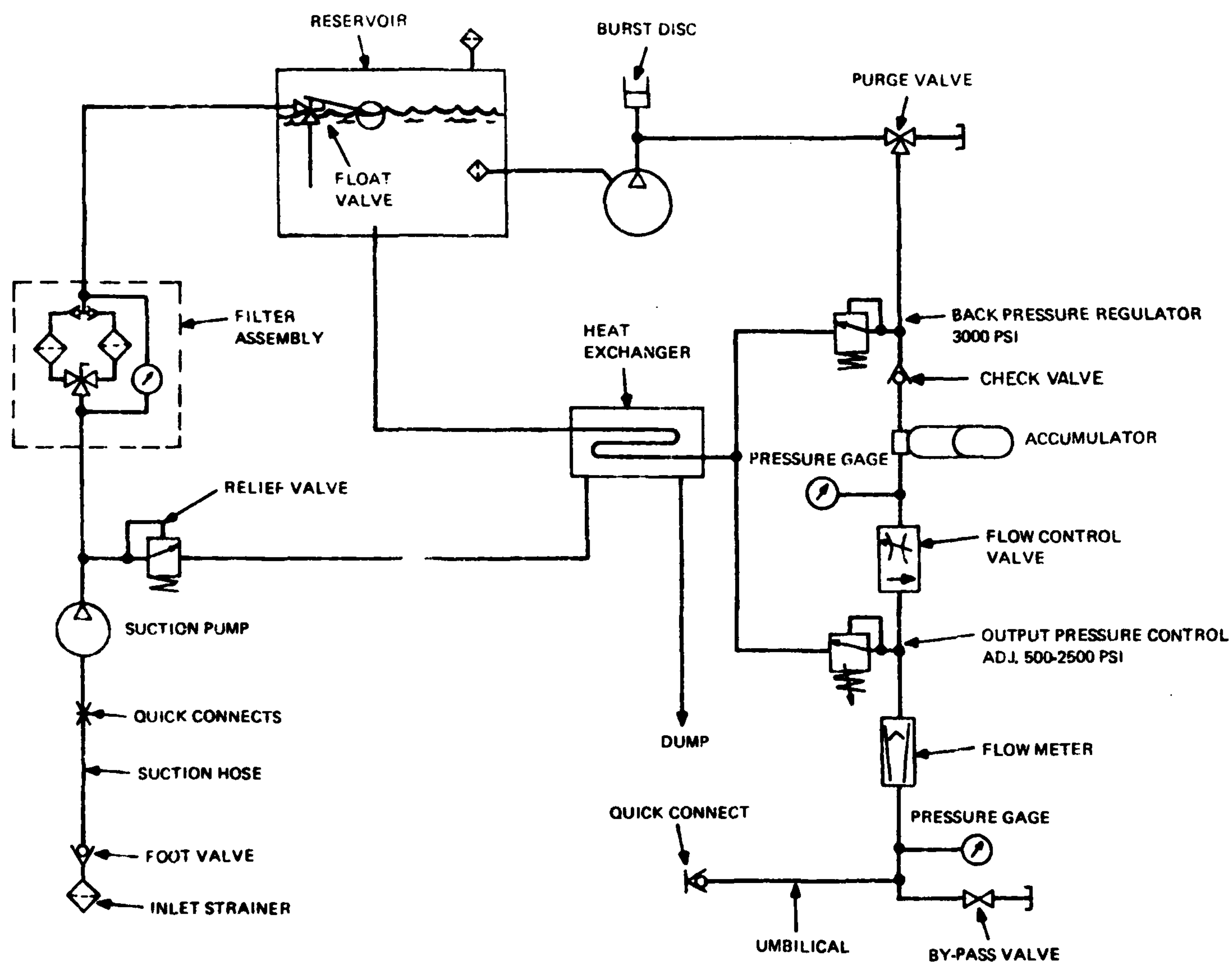

Figure 6-1. Seawater Hydraulic Power Source Seawater Flow Schematic

81496A48

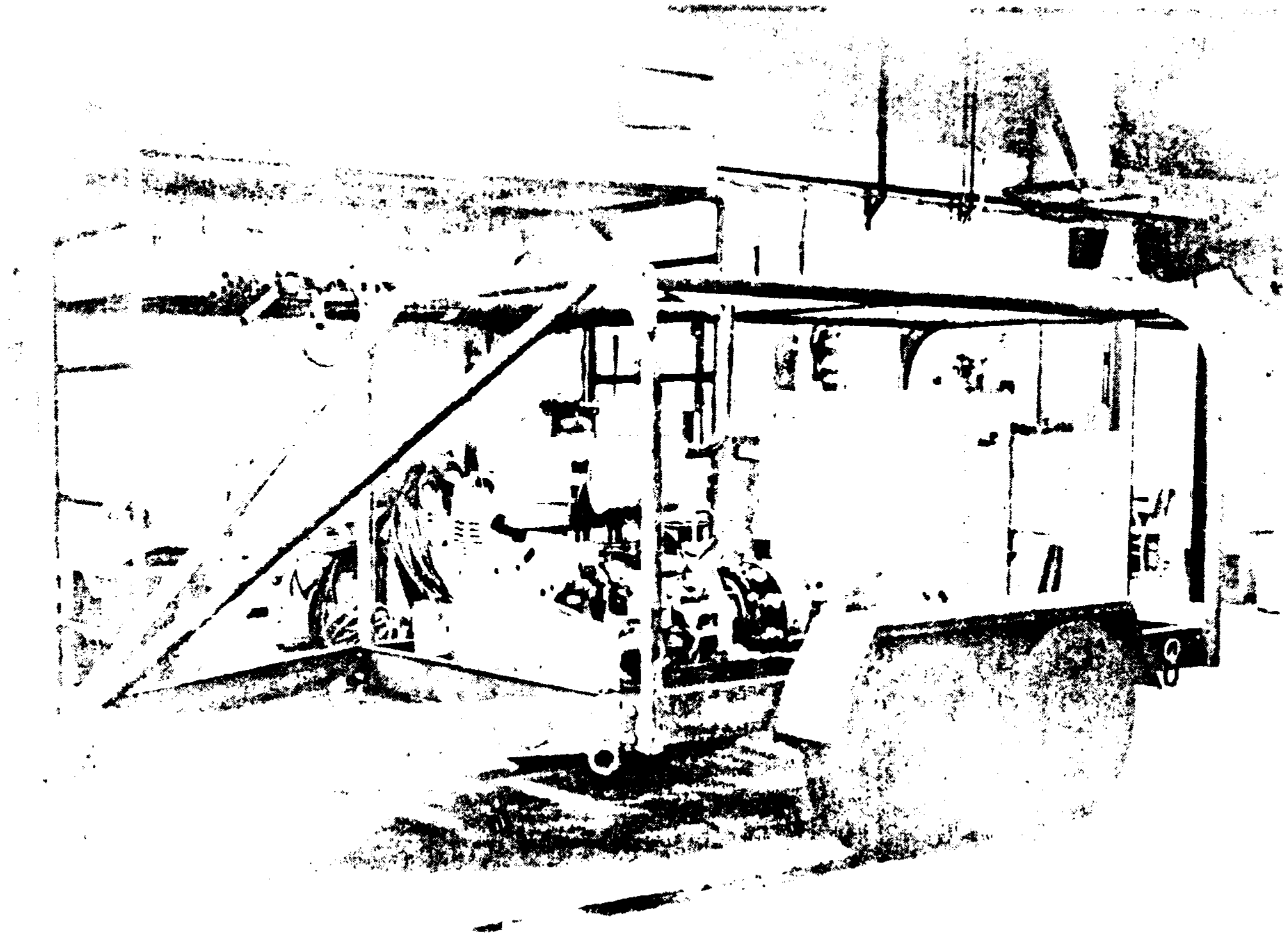

Figure 6-2. Seawater Hydraulic Power Source

81496A69

7.0 POWER SOURCE DESIGN DETAILS

7.1 Hardware Family Tree

The SWHPS can be separated into four major subassemblies: Flow circuit, power plant, umbilical, and trailer. This facilitates component recognition and description. Figure 7-1 graphically displays hardware groupings.

Flow Circuit

The flow circuit hardware includes all components actually in the flow path of the working fluid. This includes pumps, valves, hoses, pipes, fittings, reservoirs, etc., up to the delivery umbilical. The flow circuit can be divided into three major groups by function, and these are the following:

a. Suction Inlet Assembly - hardware necessary to provide a source of filtered seawater to the inlet of the main pressure pump.

b. Main Pressure Pump

c. Hydraulic Control Circuit - Valves fittings, and other hardware required to control the pressure and flow delivered to diver operated tools.

Power Plant

Hardware in the power plant subassembly includes the diesel engine, 12-volt battery, electric starter, switches, fuel tank, etc., required to provide the mechanical energy necessary to operate the SWHPS.

Delivery Umbilical

This group of hardware delivers hydraulic power to tools operated by divers at depths up to 250 ft from the power source. Major components of this group are delivery hose, storage reel, and end fittings.

Trailer

Hardware in this group includes those components which are required to provide the SWHPS with a rugged mobile base. Suspension springs, axles, wheels, lights, mountings, and frame are all examples of hardware in this group.

7.2 Contaminant Analysis And Filter Design

Introduction

Analysis of the filtration requirement for the system is based upon:

- Determination of average suspended particle size and quantities in the region in which the tool is intended to be used.

- Determination of the maximum particle size which will be acceptable to typical hydraulic devices and the seawater motor.

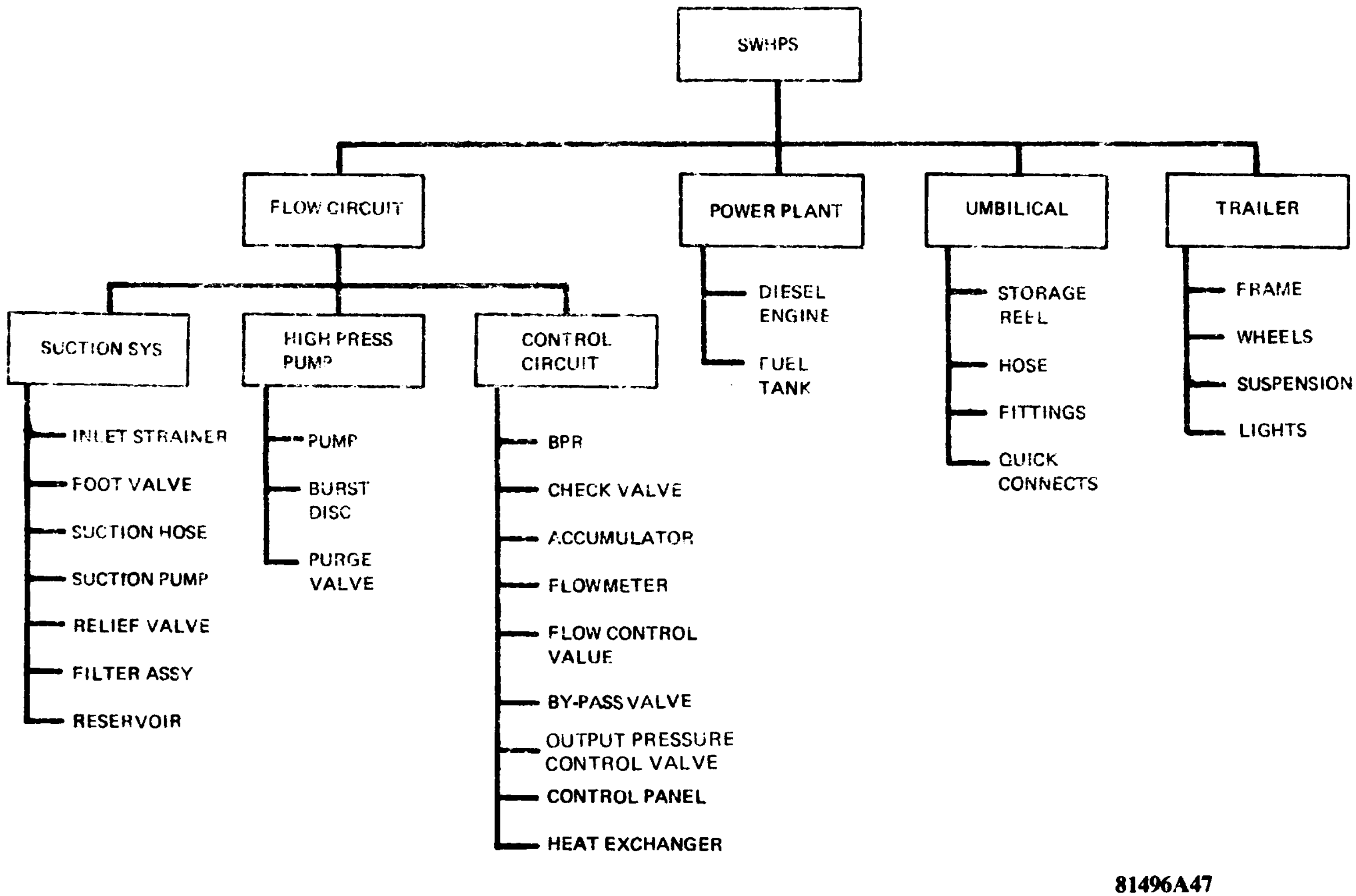

Figure 7-1. Seawater Hydraulic Power Source Hardware Family Tree

Suspended Matter In Seawater

Because of the nature of the seawater environment, it is impractical to select a single number or quantity which represents realistic levels of contaminants in a specified region. To use a number which would represent a maximum possible level would result in a filter design impractical for a mobile system.

While it is true that in the open ocean the levels are fairly constant and the particle sizes are typically on the order of 1-20 microns as indicated in Table 7-1, these levels of contaminants would result in very low filter loads, and the main requirements would be for preventing large biotic contaminants, including fish, kelp, and various other marine life, from entering the system.

The regions in which this system is intended to operate are quite different from our "ideal" open ocean situation. Large variations in filtration requirements result from storms, seasonal biological "blooms," physical disturbances caused by man, tides, and wave action. Figures 7-2, 7-3 and 7-4 represent typical variations in contaminant levels which result from tides and wave action. Such factors as storms, marine life activity, and human intervention as well as differences due to specific locality are not addressed in detail as any one of these factors can result in unpredictable changes on the order of several magnitudes.

This study determines a realistic filtration load for a prototype system possessing adequate flexibility for use in various anticipated environments without burdening the system with excessive package size. The filter network is also designed to offer adequate system protection in extreme situations, but filter life will be degraded. Operation of the system in extreme situations may require filtration schemes addressed for that specific environment.

Hydraulic Components - Particle Acceptance

Industry practice usually bases filtration requirements upon the minimum critical clearance in a device. Tables 7-2a and 7-2b present typical device clearances and manufacturer recommendations for filtration. This information serves as adequate guidance, but is not directly applicable because the filter recommendations are based on closed-loop systems.

The minimum nominal clearance in the seawater hydraulic motor is 0.00075 inch located at the vane sides (approximately 20 micron). Therefore this design suggests the next most common filtration below 20 micron which is 15 micron absolute.

This analysis does not include the effects of hard particles imbedding in the vanes and increasing wear. The most useful determination of this event will have to be based on experimental tests performed with the selected materials. This is beyond the scope of this prototype design.

Prototype Filter Design

Examination of the information presented in Figures 7-2, 7-3 and 7-4 indicates that a realistic maximum contaminant level above 16 microns is 300 milligrams/liter (this comes from Figure 7-4). This number represents tidal current effects in an estuary or bay area with relatively clean sandy bottom and moderate tidal velocities. Much lower levels can be found in areas such as the Chesapeake Bay; however, this analysis indicates that the higher level selected will yield a design more useful in its intended environments.

Table 7-1. Seawater Tool System, Open Ocean Particle Size

DEEP OCEAN SEDIMENT SUSPENSION	
REGION	**QUANTITY IN SUSPENSION**
English Channel	0.5 — 2 mg/L
East North Atlantic	0.05 — 1 mg/L
East Pacific	0.5 — 5 mg/L
Bering Sea	2 — 4 mg/L
Baltic Sea	2 — 10 mg/L
Indian Ocean	0.39 — 2.2 mg/L
Particle Size	
1-20 μ typical in Pacific Ocean 10 μ typical in Indian Ocean	

The prototype filter is designed to remove 300 mg/liter from inlet water at a rate of 15 gal./min and to have a service life of 100 hours per element.

Manufacturers' experience with these applications of filtering high quantities of biotic life has shown that it is desirable to have very low velocity flow into the element. Low flow velocity reduces the tendency for soft biotic contaminants to plug the element. This results in large physical size. The amount of biotic material will adversely affect filter life; therefore, the actual life is dependent on the specific environment.

When real operating demands are placed on the filters for the hydraulic motor - specifically 7 gallons/minute and less than 30% work cycle - great confidence is achieved with this filter system which is represented in Figure 7-5.

7.3 Control Circuit

The SWHPS design uses the hydraulic circuit shown in Figure 6-1 of Section 6. The figure is included again here for the reader's convenience.

The main pump selected for this system must be supplied by a positive head at its inlet. This, coupled with the requirement of a 15-foot suction lift capability for the system, necessitated the inclusion of a separate suction pump in the design.

A single diesel engine drives both the suction pump and the main pump at constant speed. This design controls the output of the pumps in order to interface their performance with varying system requirements.

Table 7-2a. Typical Critical Clearances

ITEM	MICROMETERS	TYPICAL CLEARANCE INCHES
Gear Pump (Pressure Loaded)		
Gear to Side Plate	0.5 — 5	0.000,02 — 0.000,2
Gear Tip to Case	0.5 — 5	0.000,02 — 0.000,2
Vane Pump or Motor		
Tip of Vane	0.5 — 1*	0.000,02 — 0.000,04
Sides of Vane	5 — 13	0.000,2 — 0.000,5
Piston Pump or Motor		
Piston to Bore (R)†	5 — 40	0.000,2 — 0.001,6
Valve Plate to Cylinder	0.5 — 5	0.000,02 — 0.000,2
Servo-Valve		
Orifice	130 — 450	0.0005 — 0.018
Flapper Wall	18 — 63	0.000,7 — 0.002,5
Spool-Sleeve (R)†	1 — 4	0.000,04 — 0.000,15
Control Valve		
Orifice	130 — 10,000	0.005 — 0.40
Spool-Sleeve (R)†	1 — 23	0.000,04 — 0.000,90
Disc Type	0.5 — 1*	0.000,02 — 0.000,04
Poppet Type	13 — 40	0.000,5 — 0.001,5
Actuators	50 — 250	0.002 — 0.010
Hydrostatic Bearings	1 — 25	0.000,04 — 0.001
Anti-Friction Bearings	0.5* —	0.000,02 —
Sleeve Bearings	0.5* —	0.000,02 —

*Estimate for thin lubricant film
†Radial clearance
Ref. Machine Design May 25, 1967

81496T28

Table 7-2b. Manufacturer Recommendations

DEVICE	SERVICE	RECOMMENDED FILTRATION*
Vane Pumps	≤1000 psi	74 Micron
Gear Pumps	≤1000 psi	74 Micron
Piston Pumps	≤750 psi	74 Micron
Hydraulic Systems	≤2000 psi	40 Micron
Hydraulic Systems	>2000 psi	25 Micron

*Absolute

From Parker Fluid Power

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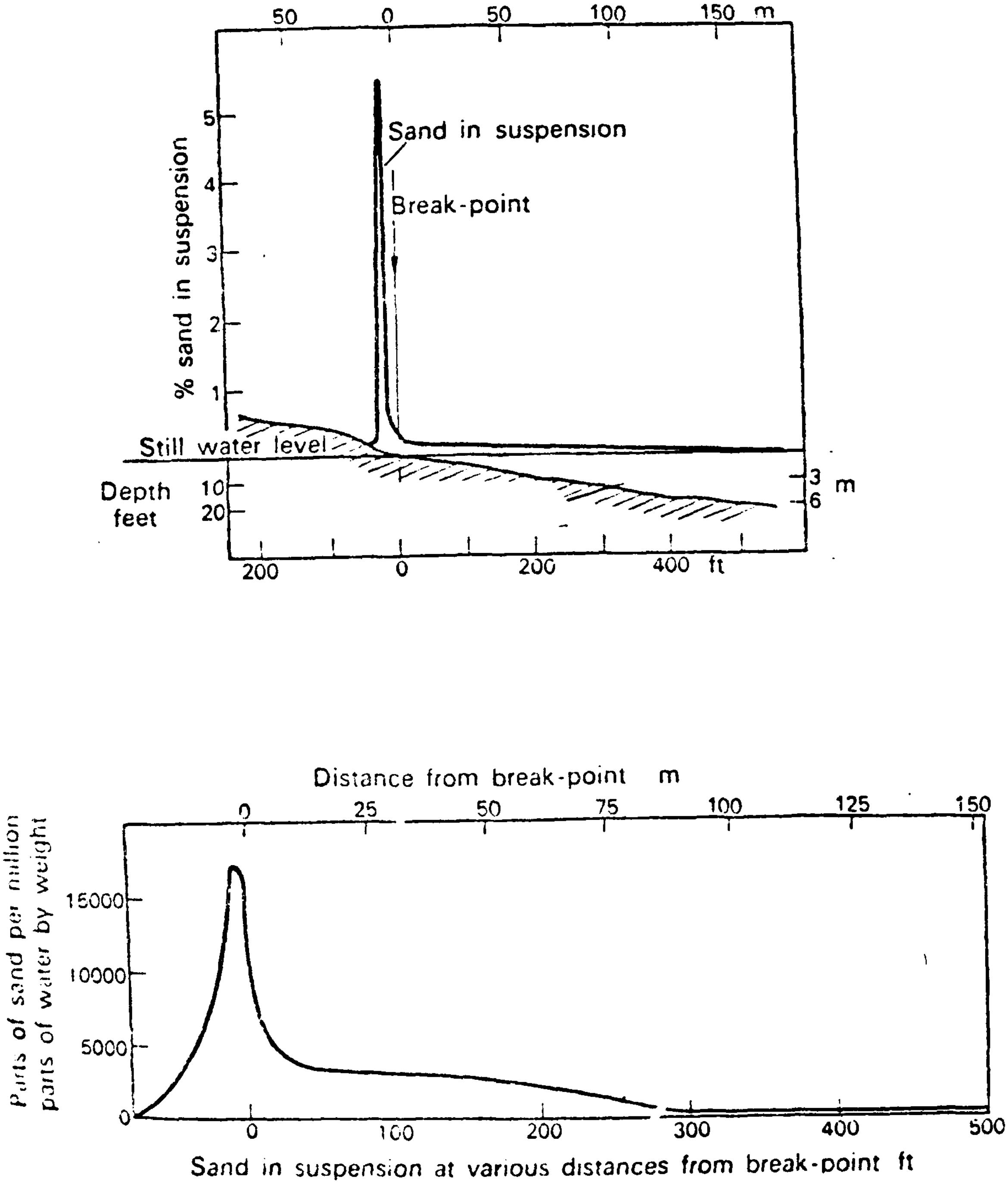

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Figure 7-2. Wave Action Effects On Suspended Particles

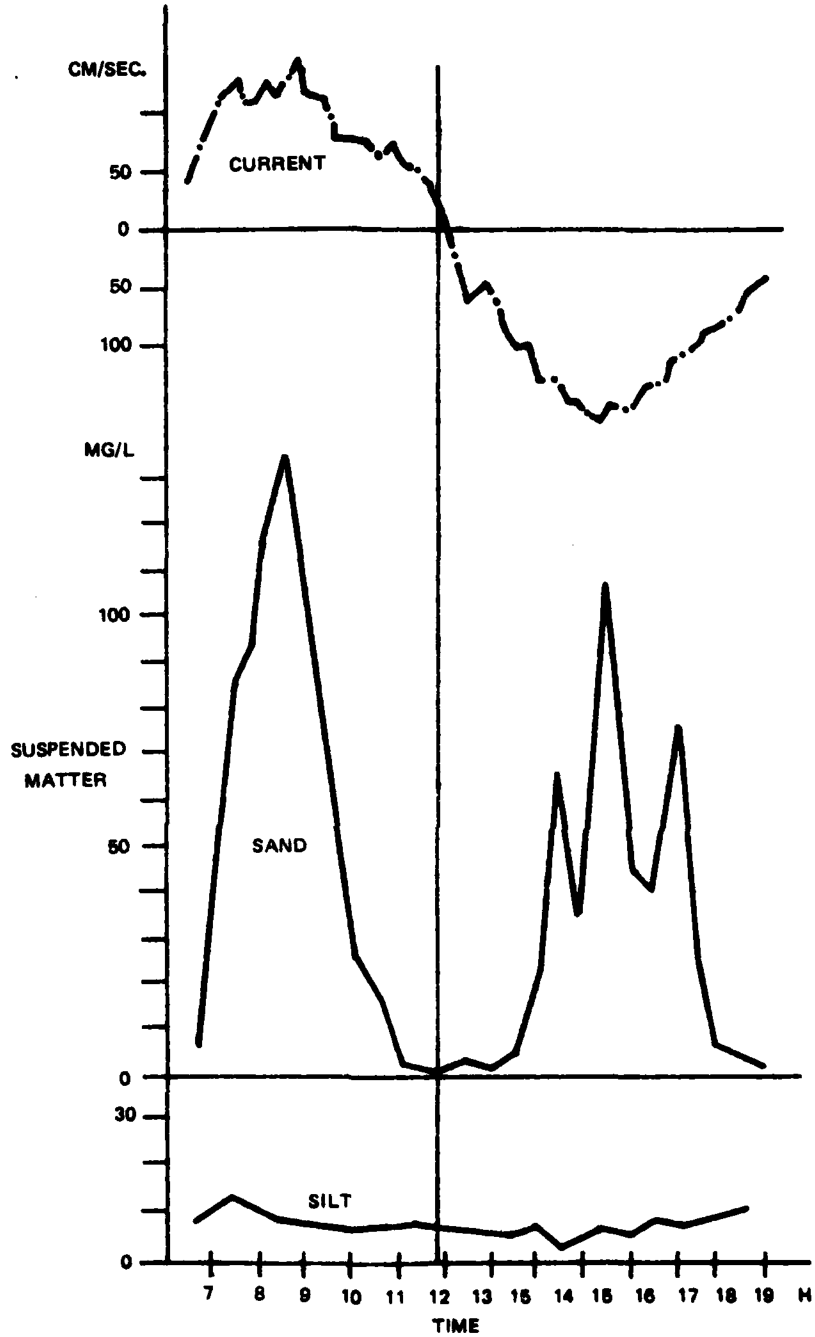

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Figure 7-3. Variations In Suspended Matter Due To Tide

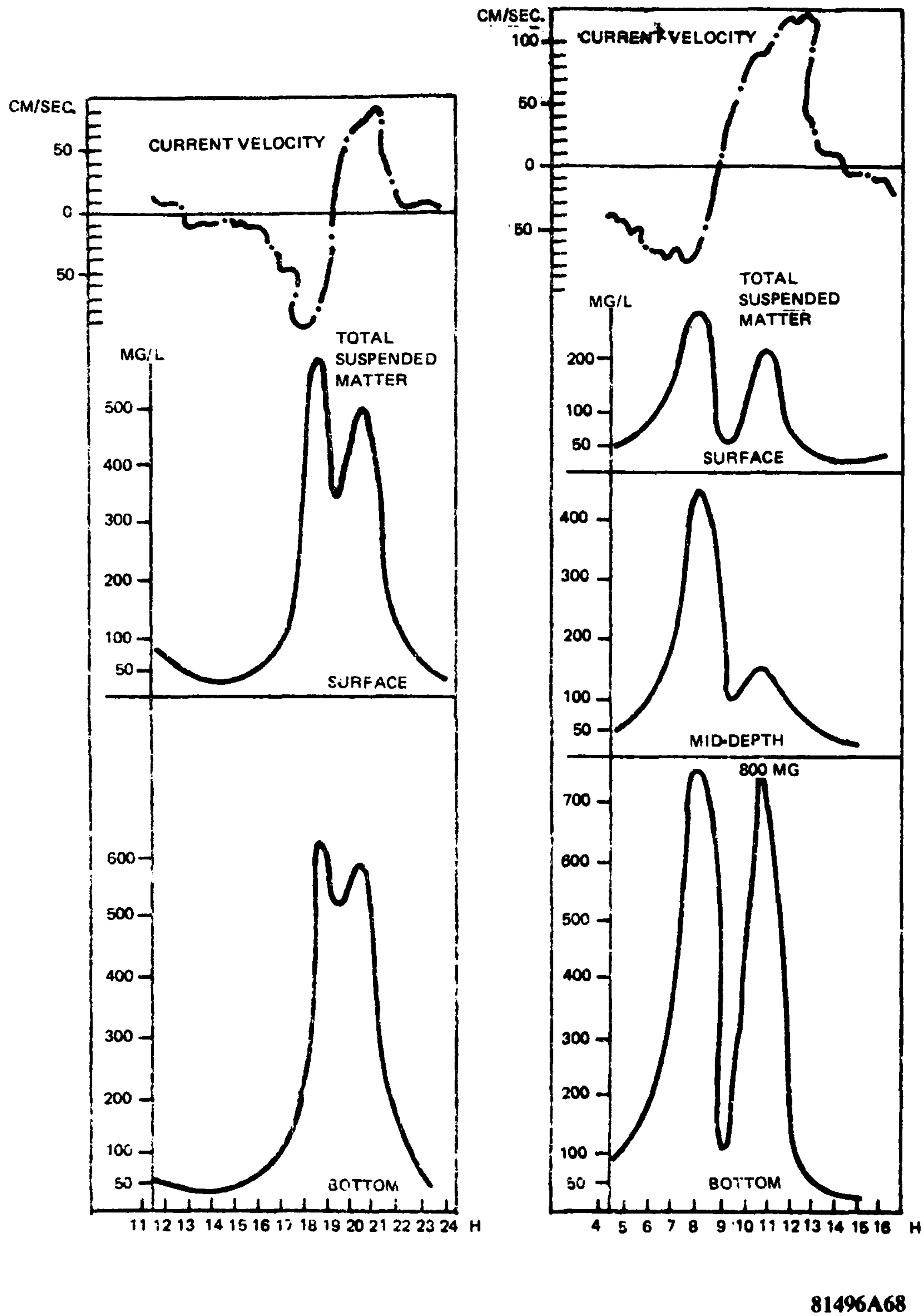

Figure 7-4. Variations In Suspended Matter Due To Current

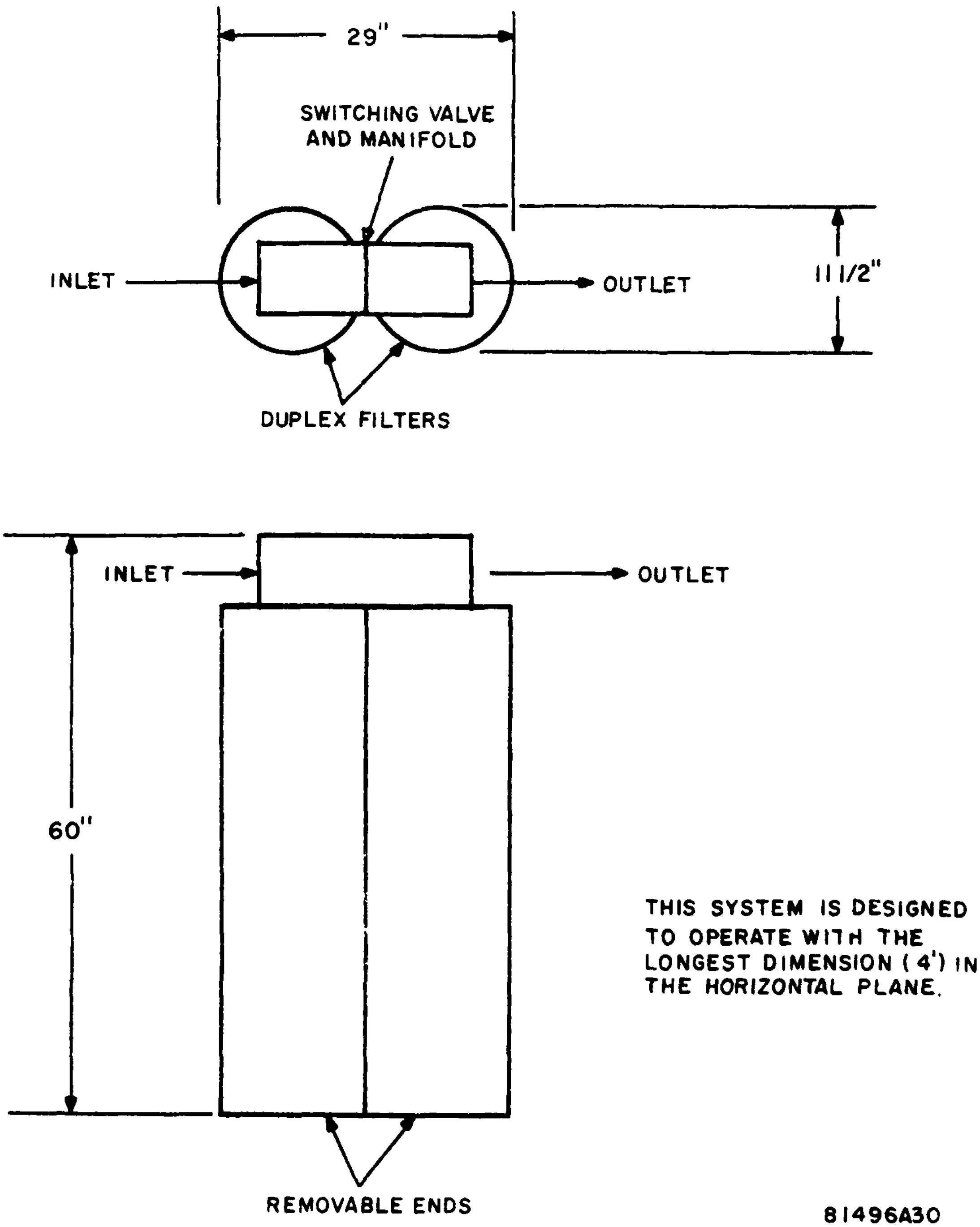

Figure 7-5. Prototype Filter Design

7.4 Component Functional Description

A description of the function of each major circuit component is presented below. The format follows the water flow through the circuit (see Figure 6-1).

The SWHPS draws seawater from the ocean through an inlet strainer. The inlet strainer provides coarse initial filtration preventing fish, kelp, and other large contaminants from being introduced into the system.

A foot valve is installed immediately downstream of the inlet strainer. The foot valve will retain water in the suction system during short periods of system inactivity.

A suction pump separate from the high pressure pump, but driven by the same diesel engine, lifts seawater from the surface of the ocean to the deck of ships and barges where the SWHPS is located. The suction pump discharges seawater at an elevated pressure sufficient to force the water through the main filter assembly.

Certain operating conditions do not require constant inlet flow. At such times the suction pump discharge will be directed to flow through a heat exchanger, providing cooling for the filtered seawater that is recycling in the closed center loop. A float valve is installed on the inlet to the reservoir. When the reservoir is filled the float valve will shut off the inlet flow causing the suction pump discharge pressure to increase. A relief valve will relieve the increased pressure. The relieved flow will be directed through the heat exchanger.

The positive displacement main pressure pump for this system provides a discharge flow rate that is relatively constant throughout the intended system operating pressure range. This system is required to be able to deliver variable flows. For that purpose a back pressure regulator (BPR) has been introduced into the circuit. The relief pressure of the BPR can be set between 2750-3000 psi. Valves installed downstream will restrict flow, increasing system pressure whenever less than full pump discharge flow is required. The increased pressure will cause the BPR to relieve flow. The closed-center circuit design directs this relieved flow back to the reservoir reducing inlet filter requirements. A heat exchanger installed in the BPR relief line removes energy from the fluid being recycled. Maintaining the high pressure setting allows optimal damping to be achieved from the accumulator. Varying the relief pressure would reduce the amount of damping provided by the accumulator.

The Flow Control Valve serves as a flow limiter for delivered seawater flows. The flow control valve is adjustable and outlet pressure compensated so that an operator may set the delivery flow rate independent of delivery pressure. The valve restricts the flow line, causing increased inlet pressure whenever delivery flow rates are above the operator settings.

When delivery flow rates are less than or equal to operator settings, there are no pressure controlling elements between the BPR and the output pressure controller. The output pressure control limits the delivery pressure to operator adjustable settings whenever delivery flow rates are less than the flow control valve setting.

Output pressure control is performed by an adjustable relief valve piped back to the reservoir.

7.5 System Production

Hydra-Tech Pumps, Inc., Mt. Holly, NJ, Produced The SWHPS.

The production of the system progressed smoothly with only minor difficulties. No significant changes were made from the system specified in Westinghouse Requirement Specification for a Seawater Hydraulic Power Source SKA 387071. The specification is included in Appendix F, SWHPS Documentation.

The only negative event during system production was the failure of a synthetic poppet in the flow control valve during testing. A stainless steel poppet was installed which satisfied test requirements.

A system test specification (SKA 387078) for the SWHPS is included in Appendix F. This test is to be performed after submission of this report.

Table 7-3 lists the major components and suppliers selected. The list shows pertinent documentation for each component. All documentation for the SWHPS is included in Appendix F.

Table 7-3. Major Component Manufacturers

COMPONENT	REQUIREMENT SPECIFICATION NO. (WHERE APPLICABLE)	PRODUCT STATUS	MANUFACTURER	ASSEMBLY DRAWING	SPARE PARTS LIST
Inlet Strainer		New Design	Aircraft Porous Media, Inc. Pinellas Park. FL	X	
Suction Pump		Standard	Gorman-Rupp Mansfield, Ohio		
Filter Assembly		New Design	Aircraft Porous Media, Inc.	X	
Main Pump		Material Modified	Harben Systems Ltd. Salisbury, England		
Diesel Engine		Standard	Deutz Atlanta, Georgia		
Back Pressure Regulator	SKA387072	Standard	Gould, Waterman Chicago, Illinois	X	X
Heat Exchanger	SKA387075	Standard	Yula Corp. Bronz, New York	X	X
Flow Control Valve	SKA387073	Material Mod.	Gould, Waterman	X	X
Flow Meter		Standard	Hedland Products Racine, Wisconsin		
Output Pressure Control Valve	SKA387074	Standard — Knob Added	Gould, Waterman	X	X

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8.0 TEST PROGRAM AND RESULTS (MOTOR AND IMPACTOR)

During the seawater tool motor development, preliminary design and final design phases, Westinghouse and NCEL worked together on various developmental and performance tests. Developmental tests were conducted to determine the feasibility of the Westinghouse design modifications and the results were used for the final design. Evaluation tests were than performed for the final design. Additional minor modifications resulted from the evaluation tests. This section discusses the testing and presents the results of the developmental and performance evaluation tests for the motor and impactor.

8.1 TOOL MOTOR DEVELOPMENTAL TESTS

The following paragraphs describe the developmental tests performed at NCEL to determine the feasibility of the Westinghouse modifications to the seawater tool motor.

Porting Modification Tests

The original motor had two inlet and two outlet pressure ports on each side. To determine if the seawater motor would operate efficiently with porting from one side only (required for tool handle mounting), the forward (long shaft end) inlet and outlet ports were blocked with plugs, and the motor was operated at 7 gpm and various pressures. As can be seen from the curves in Figures 8-1 and 8-2, there was no appreciable degradation of motor performance, and only a slight decrease in overall motor efficiency. This concept was incorporated into all of the following developmental tests as well as the final tool motor design.

Bearing Lube Flow Tests

During the development phase of the seawater tool motor design, we decided to maintain the same lubricating and cooling flow across the shaft bearings as in the original motor. To determine the amount of flow required, flow out of the forward and aft bearings was measured for motor inputs of 6 and 7 gpm at various pressures and with the aft set of parts either opened or blocked. These measurements showed the following:

1. when the aft ports were blocked, flow across the bearings more than doubled,
2. a maximum flow of 720 ml/min (43.9 in.^{3}/min) across the bearings occurred at 600 psig supply pressure and 7 gpm (aft ports opened), and
3. a minimum flow of 580 ml/min (34.5 in.^{3}/min) occurred at 1100 psig supply pressure and 7 gpm (aft ports opened).

See Appendix D for specific values and test curves.

A minimum flow of 580 ml/min (35.4 in.^{3}/min) was used as a baseline during the preliminary design. However, during analysis and evaluation testing of the final bearing lube design, a flow of 0.07 ml/min (4.3 in.^{3}/min) across the bearings was determined to be adequate. See Appendix B for heat transfer and lubricating flow calculations which support this value.

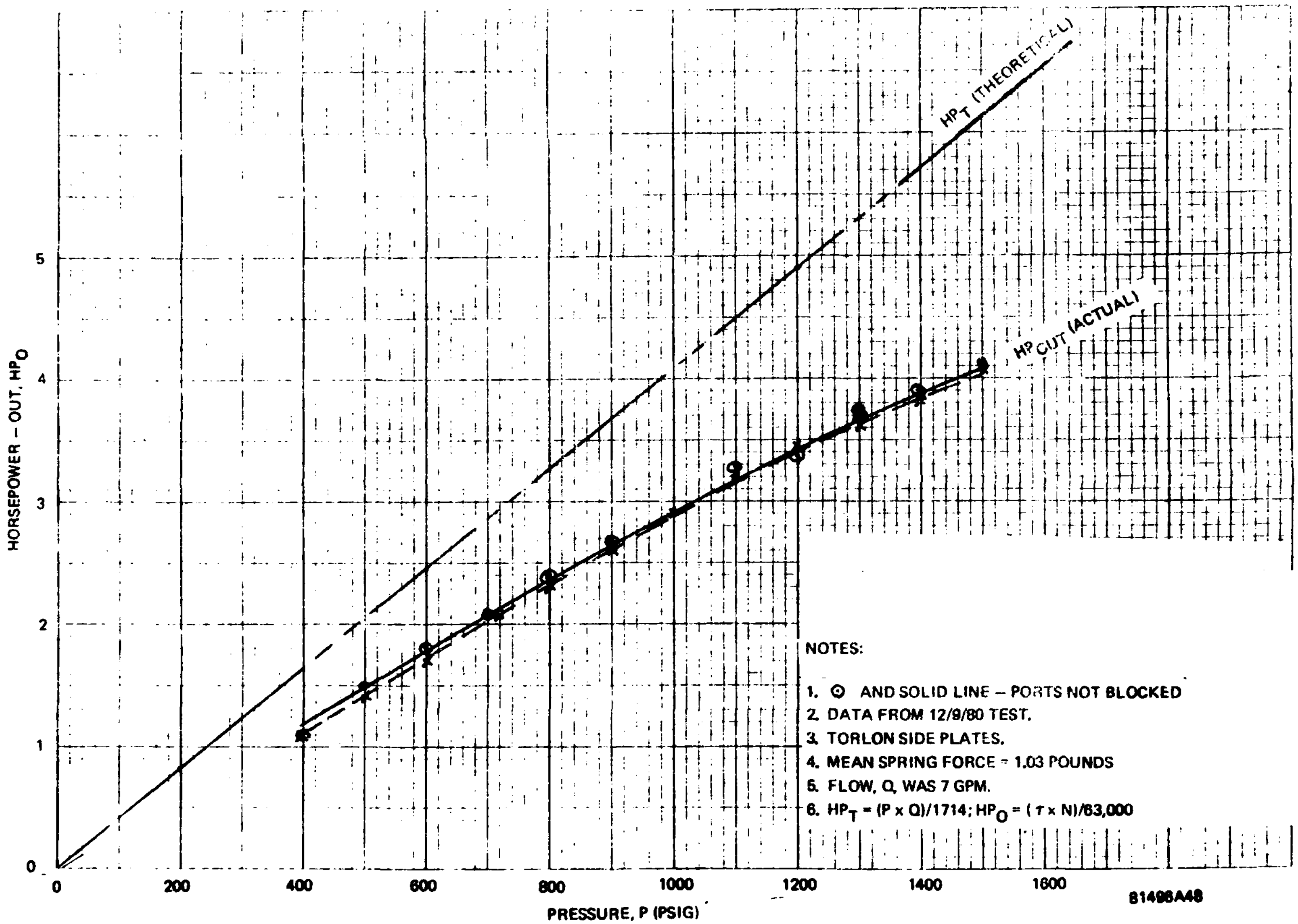

Figure 8-1. Overall Efficiency Comparison, Porting Modification Tests

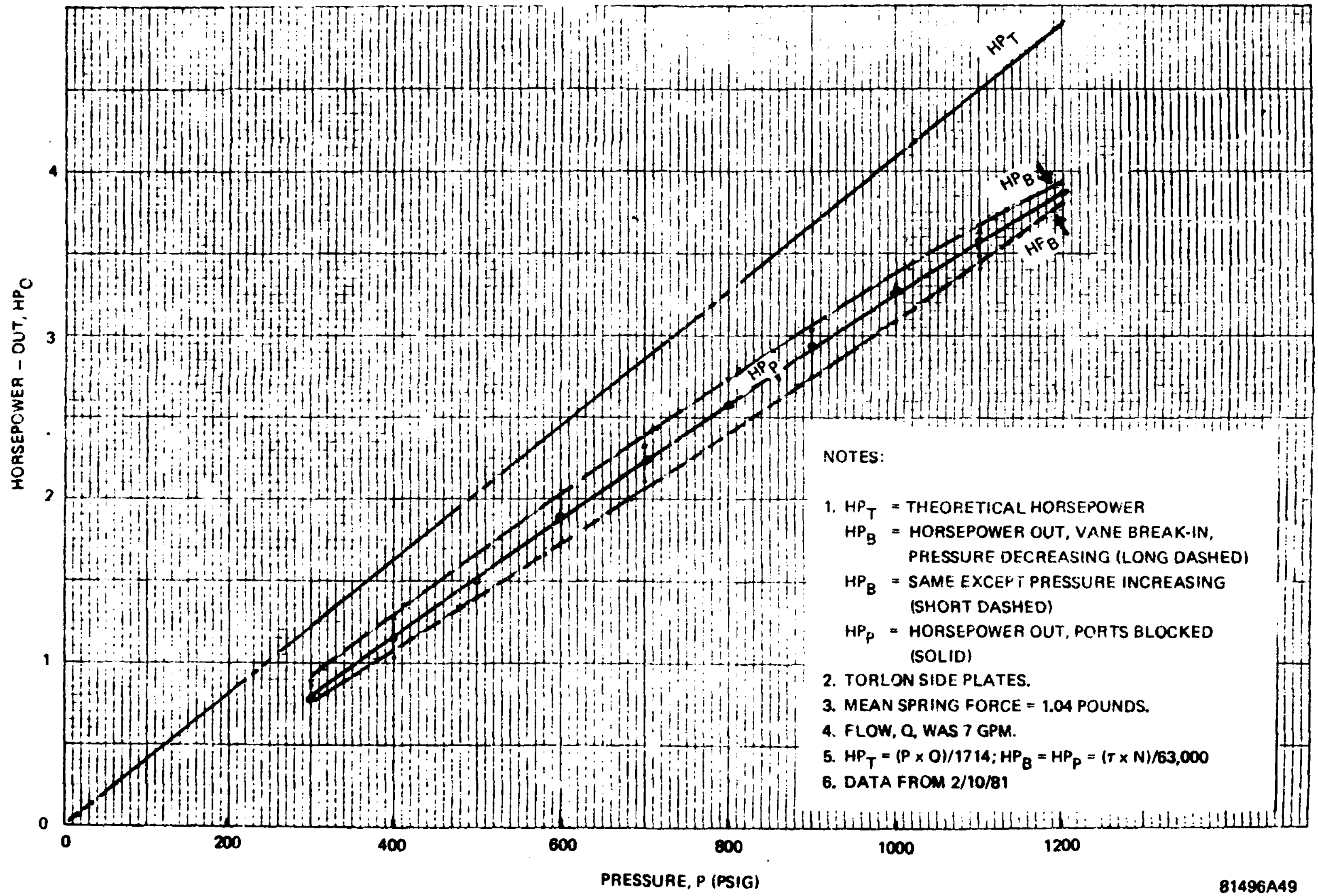

Figure 8-2. Overall Efficiency Comparison, Porting Modification Test and Vane Break-In Tests

Vane Spring Force Determination Tests

The object of the vane spring force tests was to determine the minimum vane spring force for increased pressure operation, without "knocking," for minimum vane tip wear and for longer spring life.

Using the original Westinghouse/NCEL motor as a test bed, various sets of springs with mean force values ranging from 0.25 lb_f/spring to 1.50 lb_f/spring were tested. Two series of tests were run: one series with the original cam and one series with a new cam. The curves of Figures 8-3 a through 8-3 e show the general results.

The test results showed that a spring exerting a mean force between 0.75 and 1.00 pounds would be optimum; but because of the increased vane thickness, the 1.00 lb_f was nominally selected for the final spring design. It must be noted, however, that these spring force determination tests did not take into account the reduction of spring force due to wear. Any reduction in force will allow "knocking" to occur sooner than expected. This spring wear problem will be discussed further and in more detail in the following sections.

Vane Spring Life Tests

After obtaining the optimum spring force from the previous tests, a series of motor performance tests were planned to extrapolate spring life. These tests included operating the motor at approximately 1500 rpm for a specific time period to determine spring wear and therefore spring life for the following conditions:

1. using a set of optimum force springs coated with a dry film lubricant (Tiodize);
2. using a set of uncoated optimum force springs in the rotor which had the spring holes either coated with a dry film lubricant (Tiodize) or had bearing material (Karon) inserts installed; and
3. using a set of coated springs in the rotor described in (2).

The vane spring life tests were not performed for the following reasons:

1. the Karon inserts for the rotor spring holes would have delayed testing for a minimum of 14 to 16 weeks;
2. the Tiodized 17-7PH springs were in very poor condition; and
3. the dimensional mismatch (approximately 0.012 inches due to tolerances) between the spring holes in the rotor and those in the vanes were thought to cause undue wear on the springs (coated or uncoated);
4. the assumption that minimizing the tolerance mismatch and tapering the rotor spring holes would eliminate the spring wear problem.

We found during the 50-hour tool motor performance/evaluation tests that spring wear is still a problem to be solved. Some solutions to the spring wear problem will be discussed in Section 9.

End Plate O-Ring Force Determination Test

The object of this test was to determine if the O-ring forces from the equal area pressure pads (for reversibility) or the pressure forces from the pads themselves would have any effect on motor performance.

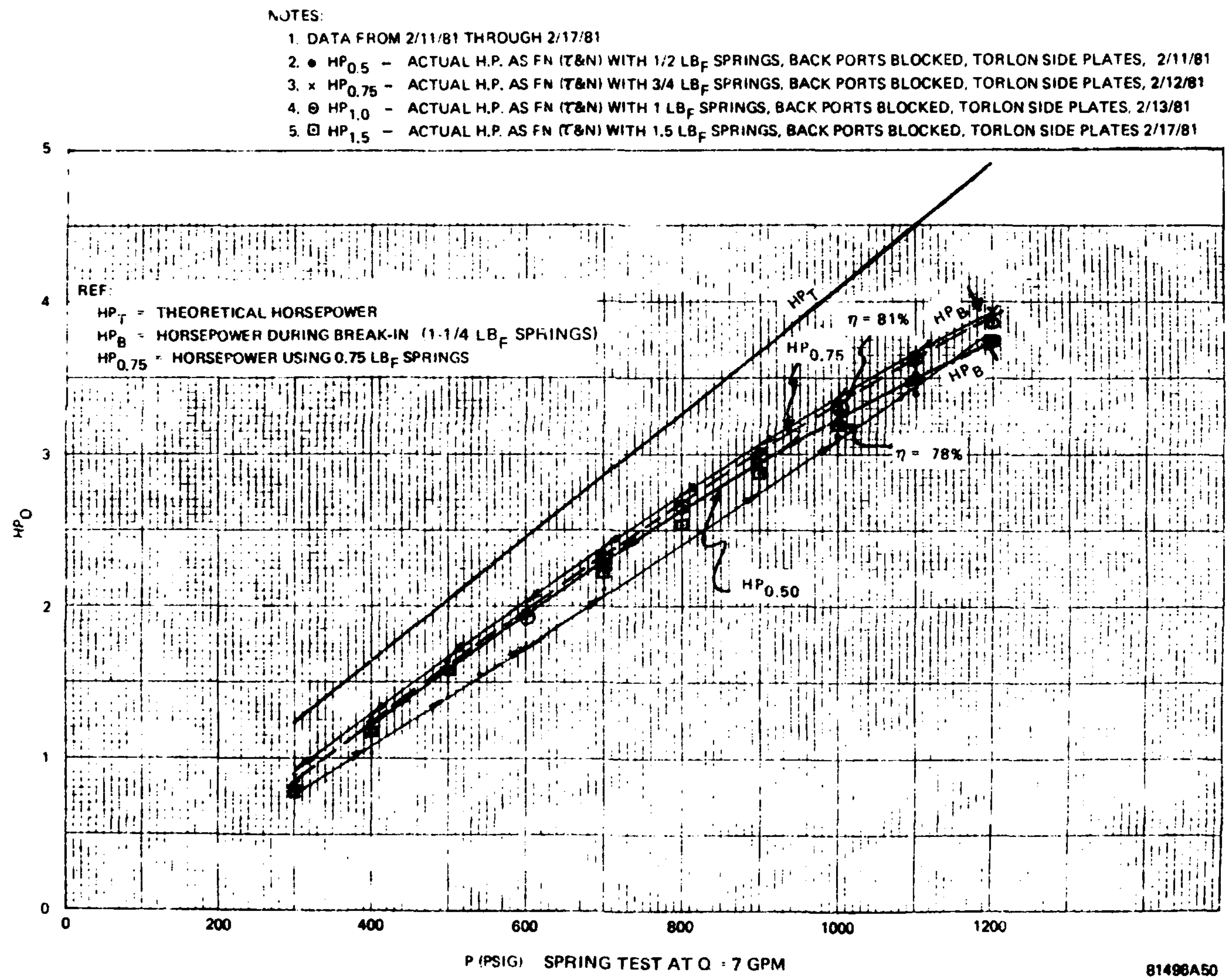

Figure 8-3 a . Spring Force Test at 7 gpm (Old Cam)

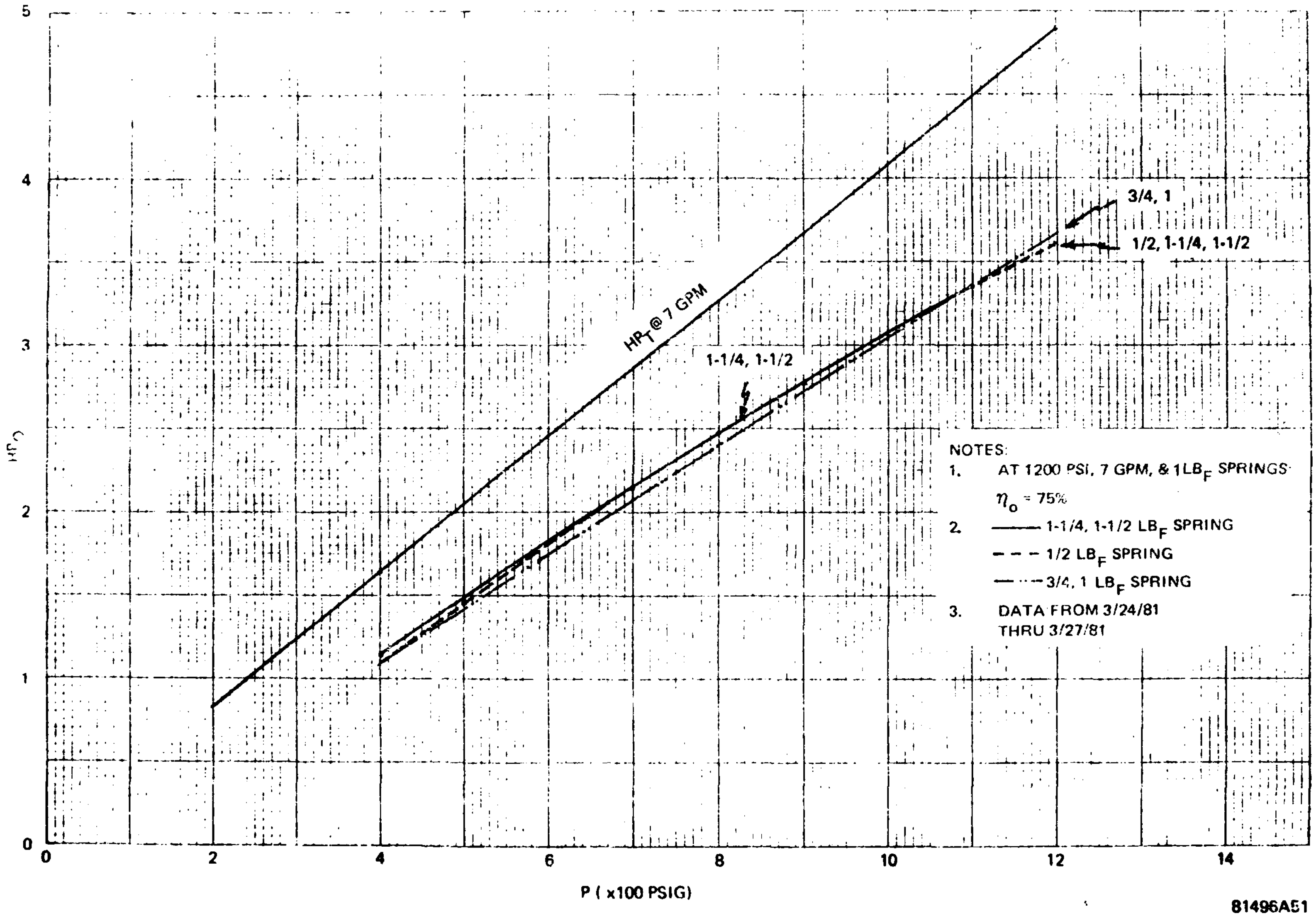

Figure 8-3 b . Spring Force Test at 7 gpm (New Cam)

NOTES:

1. ● :2/11/81: $HP_{0.25}$ - ACTUAL H.P. AS FN (τ & N) WITH 1/4 LB_F SPRINGS, EACK PORTS BLOCKED, METCAR SIDE PLATES
   × :2/11/81: $HP_{0.50}$ - ACTUAL H.P. AS FN (τ & N) WITH 1/2 LB_F SPRINGS, BACK PORTS BLOCKED, TORLON SIDE PLATES
   △ :2/12/81: $HP_{0.75}$ - ACTUAL H.P. AS FN (τ & N) WITH 3/4 LB_F SPRINGS, BACK PORTS BLOCKED, TORLON SIDE PLATES
   ⊙ :2/13/81: $HP_{1.0}$ - ACTUAL H.P. AS FN (τ & N) WITH 1 LB_F SPRINGS, BACK PORTS BLOCKED, TORLON SIDE PLATES
   ⊡ :2/17/81: $HP_{1.5}$ - ACTUAL H.P. AS FN (τ & N) WITH 1.5 LB_F SPRINGS, BACK PORTS BLOCKED, TORLON SIDE PLATES

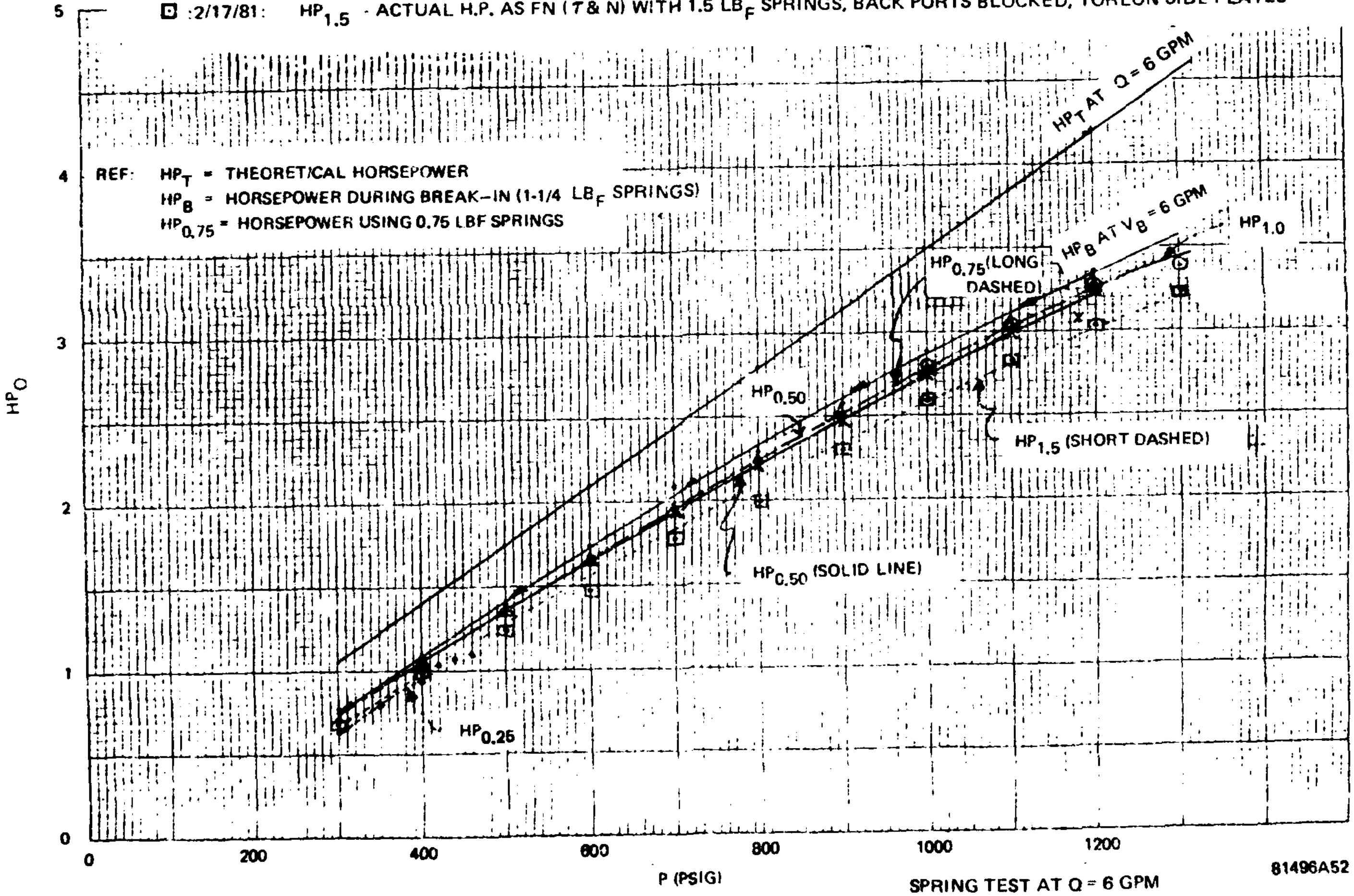

Figure 8-3 c . Spring Force Test at 6 gpm (Old Cam)

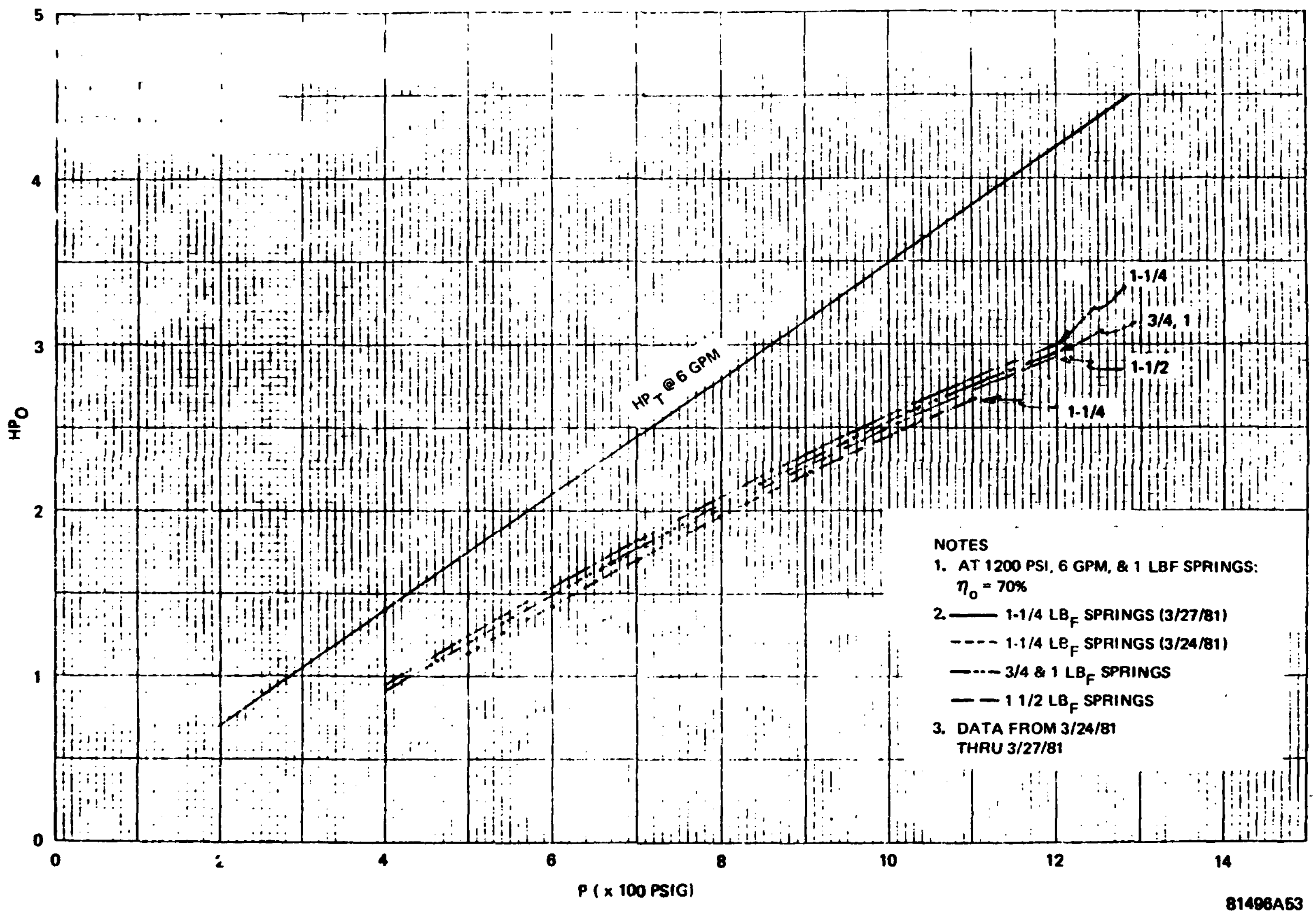

Figure 8-3 d. Spring Force Test at 6 gpm (New Cam)

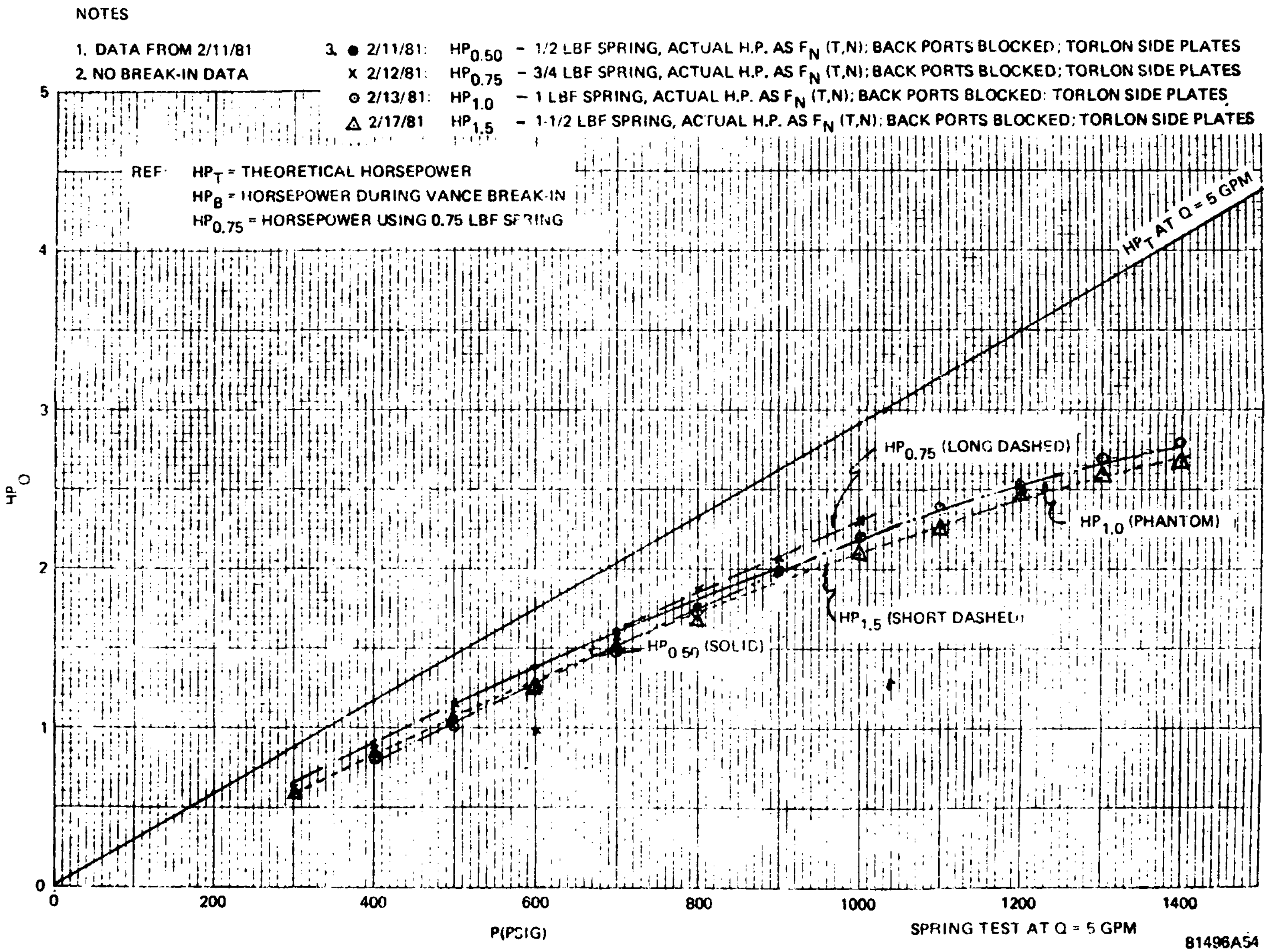

Figure 8-3 e . Spring Force Test at 5 gpm (Old Cam)

Two new aluminum end plates were machined with equal area pressure pads (drawing number SKHAH1052). The original Westinghouse/NCEL motor was tested using the new end plates. There was a 3% to 5% increase in motor performance, and the motor ran much more smoothly than in previous tests. Flow across the bearings was reduced by approximately 50% but this is still considered adequate based on the studies of Appendix B. Therefore, this concept was incorporated into the final design of the seawater tool motor.

Rotor Vane Slot Modification Tests

To reduce rocking of the vane in the rotor vane slot, the configuration of Figure 2-5 was developed. To determine what effect the new vane slot configuration would have on motor performance, a new Inconel 625 rotor was machined (SKHAH1006H01) and tested. The only change to the rotor was in moving the 0.18 inch diameter holes radially inward 0.04 inches to provide a sliding surface for the vanes. The new rotor was placed in the original motor which had a new cam section. Due to numerous problems encountered during the test and due to the elimination of the 0.18 inch diameter holes (see Figure 2-5) for the wider tool motor vanes, the test was aborted.

The new rectangular vane slot configuration was tested as part of the 50-hour tool motor evaluation/performance test described later in this section. Test results showed that the basic concept was good.

Bolt Torque Determination Test

The object of this test was to determine the bolt torque required on the four motor mounting bolts for proper operation and performance of the seawater tool motor.

Operating the original motor at constant flow and pressure, the bolt torque valves were increased from 50 in.-lb_f to 100 in.-lb_f in 25 in.-lb_f increments. The test results showed only a slight increase in overall motor efficiency (approximately 0.4%). Therefore, we concluded that bolt torque did not greatly affect motor performance.

The 50 in.-lb_f bolt torque (used throughout the development tests) was incorporated into the tool motor design as an initial torque value — to be increased if required.

Vane Material Tests

Using the original motor, different grades of Torion vanes were tested to determine the material with the best vane tip wear and overall wear characteristics. These tests were done independently by NCEL with the overall results transferred to Westinghouse for incorporation into the tool motor design. Generally, the tests showed the following:

Torlon Grade	Wear Characteristics/Motor Performance
4347	Excellent
4275	Good
4301	Fair
7130	Very Poor (not recommended for further testing)

Torlon 4275 was selected early in the development and was incorporated into the seawater tool motor design. However, the use of Torlon 4347 should be considered for any follow-up programs.

8.2 SEAWATER TOOL MOTOR EVALUATION/PERFORMANCE TEST

The main object of the seawater tool motor tests performed at NCEL was to operate the "as manufactured" motor for 50 hours, determining and incorporating any necessary modifications required to meet the specified requirements and goals. Included in the main objective was testing motor reversibility and disassembling the motor to inspect and measure component wear to estimate motor life.

The detailed test setup and procedure are presented in Appendix D. Basically, the test procedure was as follows:

1. Performance Tests — The motor was operated at constant flows of 5, 6, and 7 gpm and constant pressures of 250, 500, 750, 1000, 1250, and 1500 psig to determine speed, output torque, and efficiencies. The test data was printed out continually, and performance curves were plotted approximately every four hours. See Figure 8-4 for an example of typical performance curves and test data taken. This particular set of curves was made after 14 hours of operating and showed that motor performance had not degraded with time.
2. Life Tests — The motor was tested for 100 hours at a nominal output 2.0 horsepower. Nominally, the operating parameters of flow, pressure, speed, and output torque were 6 gpm, 825 psig, 1415 rpm and 90-lb_f-in., respectively. The tests were summarized in Table 8-1. Generally, at the operating parameters specified, the motor performance was satisfactory with an overall efficiency of 69% to 70% during the entire test period. Also, there was no knocking at the higher pressures of 1200 to 1500 psig initially. But, as the springs wore in, pressure values where knocking occurred dropped to approximately 900 to 1100 psig.

Results of Performance Tests

A comparison of the overall motor performance results for two key time periods during the first 50 hours of testing are shown in Figure 8-5. The curves are an indication of the overall efficiency and the pressure knock points obtained after 15.9 and 50 hours accumulated operating time. As can be seen from the curves, there was a slight increase in efficiency and a drop in "nonknocking" operating pressure near the end of the 50-hour test. During the tests, knocking occurred erratically throughout, i.e., the motor would knock at 1000 psig and 6 gpm one minute, but would operate properly at 1300 psig and 6 gpm a minute later. The table 8-2 shows the average knock points observed during individual tests and the mean spring force used. The slight increase of efficiency at 50 hours was expected as a result of vane tip wear-in. Table 8-3 lists a comparison of the test runs and the effects of the changes made during the 50 hour test.

Performance curves for test runs 8 through 11 (64 hours through 98 hours running time) are shown in Figures 8-6 through 8-9, respectively.

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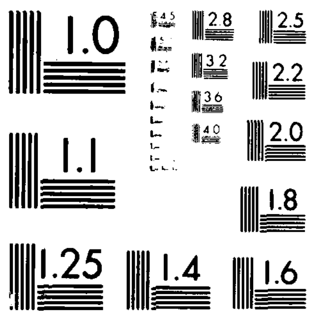

MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

Table 8-1. Summary of Tool Motor Tests

RUN NO.	DATE	TEST CONDITIONS	NOMINAL PERFORMANCE PARAMETERS								RUN TIME (HR)	COMMENTS
			FLOW GPM	PRES PSIG	TORQ in.-lbf	SPEED RPM	POWER OUT HP	EFFICIENCY, % MECH	VOL	TOTAL		
1a	8/14	As manufactured fresh water media	4	320	30	1039	0.5	—	—	66	0.7	247 psig start, knocking at 500 psig and 6 gpm
1b	8/17	Same as 1a except better alignment of motor on test stand	6	802	89	1520	2.2	89	87	77	4.1	225 psig start, periodic to excessive knocking, sounded "rough"
2	8/17	Same as 1 b except axial van length shaved to 0.6240 in. nom.	6	600	64	1457	1.5	85	83	71	6.6	184 psig start, knocking at 6 gpm and 600 psig, sounded "rough"
3	8/18	Same as 2 except with spring 0.027 in. thk spacers to increase spring force	7	1000	113	1716	3.1	90	83	75	3.8	137 psig start, knocking at 6 gpm and 1170 psig, knocking at 7 gpm and 1300, psig sounded "rough"
4	8/18	Same as 3 except test stand replumbed to run motor in reverse	7 7	1200 1000	127 106	1594 1651	3.2 2.8	84 85	78 80	65 68	0.2 1.3	140 psig start, knocking at 7 gpm and 1300 psig, sounded "rough"
5	8/19	Same as 3 except seawater media	7 7	1000 1000	107 108	1634 1671	2.8 2.9	86 86	79 81	68 70	2.4 1.5	185 psig start, sounded rough
6	8/19	Same as 5 except new bearing plates with 30° inner slots	7	1000	107	1686	2.9	85	82	70	1.3	140 psig start, "rough" sound disappeared
7	8/20	Same as 6 except spring spacers removed	6	830	89	1413	2.0	86	80	69	15.8	170 psig start; near end of test, knocking observed at 1000 psig and 5 gpm

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Table 8-1. Summary of Tool Motor Tests (Continued)

RUN NO.	DATE	TEST CONDITIONS	NOMINAL PERFORMANCE PARAMETERS								RUN TIME (HR)	COMMENTS
			FLOW GPM	PRES PSIG	TORQ in.-lbf	SPEED RPM	POWER OUT HP	EFFICIENCY, % MECH	VOL	TOTAL		
7 Contd	8/21	Continued test #7	6	825	90	1411	2.0	88	80	70	12.3	420 psig start, knock points at 1000 psig and 5 gpm 1000 psig and 6 gpm 1200 psig and 7 gpm
	8/22	Continued test #7	6	827	90	1399	2.0	87	79	69	6.0	160 psig start, knock points at 1080 psig and 5 gpm 1020 psig and 6 gpm 1044 psig and 7 gpm
8	8/26	Disassembled motor and installed new springs, seawater media	6.1	500	53	1523	1.3	84	85	71	1.5	Knock observed at 715 psig and 6.1 gpm
9	8/27	Same as 8 except with 0.027 in. thick spring spacers	6	822	89	1434	2.0	86	81	70	12.0	160 psig start, at end of test, knock observed at 1180 psig and 5 gpm 1230 psig and 6 gpm 1225 psig and 7 gpm
	8/28	Continued test #9	6	820	89	1428	2.0	87	81	70	3.5	400 psig start, slight knock observed at 1124 psig and 5 gpm 1300 psig and 6 gpm 1409 psig and 7 gpm
10	8/31	Disassemblied motor and installed new springs and 0.040 in. thick spring spacers, freshwater media	6	822	87	1450	2.0	85	82	70	6.0	225 psig start, knock at 1115 psig and 5 gpm

81496T56-2

Table 8-1. Summary of Tool Motor Tests (Continued)

RUN NO.	DATE	TEST CONDITIONS	NOMINAL PERFORMANCE PARAMETERS								RUN TIME (HR)	COMMENTS
			FLOW GPM	PRES PSIG	TORQ in.-lbf	SPEED RPM	POWER OUT HP	EFFICIENCY, % MECH	VOL	TOTAL		
11	9/1	Same as 10 except seawater media	6	822	89	1444	2.0	86	82	71	9.0	200 psig start, knock observed at
			7	1041	112	1668	3.0	86	81	70	2.2	875 psig and 5 gpm 1120 psig and 6 gpm
			5	640	70	1189	1.3	88	81	71	1.0	no knock at 1400 psig and 7 gpm
	9/2	Continued test #11	6	830	91	1447	2.1	88	82	72	10.3	Mounting bolts torqued to 200 lbf-in., knocking at
			7	1053	113	1672	3.0	86	81	70	1.7	825 psig and 5 gpm 1130 psig and 6 gpm
	9/3	Continued test #11	7	1089	118	1665	3.1	87	81	70	2.3	Knocking at 1010 psig and 5 gpm 1300 psig and 6 gpm
											99.5	Total Motor Run Time

81496T56-3

Table 8-2. Spring Force Vs. Knock Pressures

RUN NO.	AVERAGE MEAN SPRING FORCE (lbf)	RUN TIME (HOURS)	AVG KNOCK POINT PRESSURES (psig)			SPRING SPACER THICK (IN.)
			5 gpm	6 gpm	7 gpm	
1a	1.05		1000	550	500	No Spacer
1b	1.05	9.2	640	875	1100	
2	1.05		715	665	Not Observed	
3	1.29		1085	1010	1060	
4	1.29	6.7	Not Observed		1300	0.027
5 & 6	1.29		Not Observed			
7	1.05	34.1	875	1100	1200	No Spacer
8	1.08	1.5	800	720	720	
9	1.32	15.5	1150	1270	1310	
10	1.44	6.0	1060	1000	No Knock At 1400	0.040
11	1.44	26.5	890	1170		

81496T58

Table 8-3. 50-Hour Performance Test Comparison

COMPARISON BY RUN NO.	CHANGE	EFFECT OF CHANGE ON PERFORMANCE	COMMENTS
#2 vs. #1b	Shaved vane length	η_o Decreased	A decrease in η_v resulting from increased leakage past vanes
#3 vs. #2	Increased spring force	η_o Increased	Leakage not affected by shaft speed, therefore η_o increases with higher flows and higher speed; also η M is higher at higher pressures obtained via larger spring force
#4 vs. #3	Reversed operation	η_o Decreased	Vane tip wear-in effect, as vane wears in, η_o will increase
#5 vs. #3	Seawater	Slight η_o Decrease	Test #3 performed at higher speed, η_o higher expected at increased speed
#6 vs. #5	Bearing plate slots increased	No Change, ' Rough ' Sound Disappeared	Higher pressure fluid beneath the vanes allowed to escape sooner thereby allowing vanes to follow cam track transition points smoother
#7 vs. #6	Spring spacers removed	Knocking at lower pressures near end of 50-hour test	Decreased spring force from spacer removal and spring wear

81496T59

SEAWATER HYDRAULIC MOTOR

DATE: 08-19-81

TEST#5 USING SEAWATER. MOTOR DIRECTION CCW.

TIME	FLOW	PRESS	TEMP	TORQ	RPM	HPin	HPout	E-M	E-V	E-T
	4.92	253		26	1296	.73	.54	.82	.90	.74
	5.01	500		54	1200	1.46	1.03	.86	.82	.70
	5.07	754		81	1141	2.23	1.47	.86	.77	.66
	4.97	1001		109	1092	2.90	1.89	.87	.75	.65
	5.07	1245		135	1072	3.69	2.29	.87	.72	.62
	5.95	237		23	1595	.82	.58	.78	.91	.71
	5.98	504	86°F	54	1482	1.76	1.26	.85	.84	.72
	5.98	756		83	1418	2.64	1.88	.88	.81	.71
	5.98	991		109	1360	3.46	2.35	.88	.77	.68
	6.03	1252		136	1312	4.40	2.82	.87	.74	.64
	6.98	501		51	1775	2.04	1.45	.82	.87	.71
	7.01	752		80	1716	3.08	2.17	.85	.83	.71
	6.96	1001		109	1653	4.06	2.85	.87	.81	.70
	7.01	1249		135	1585	5.11	3.40	.87	.77	.67
	6.96	1501		163	1519	6.10	3.92	.87	.74	.64

SEAWATER MOTOR TEST

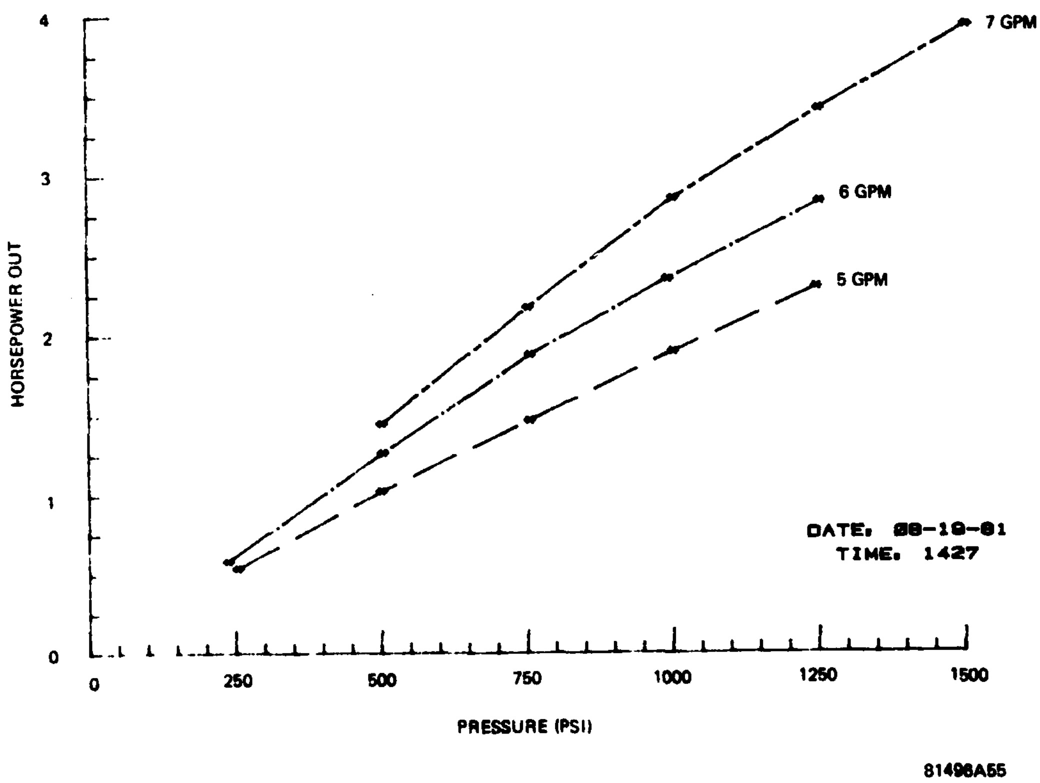

Figure 8-4. Typical Figure: Test No. 5 Using Seawater Motor Direction

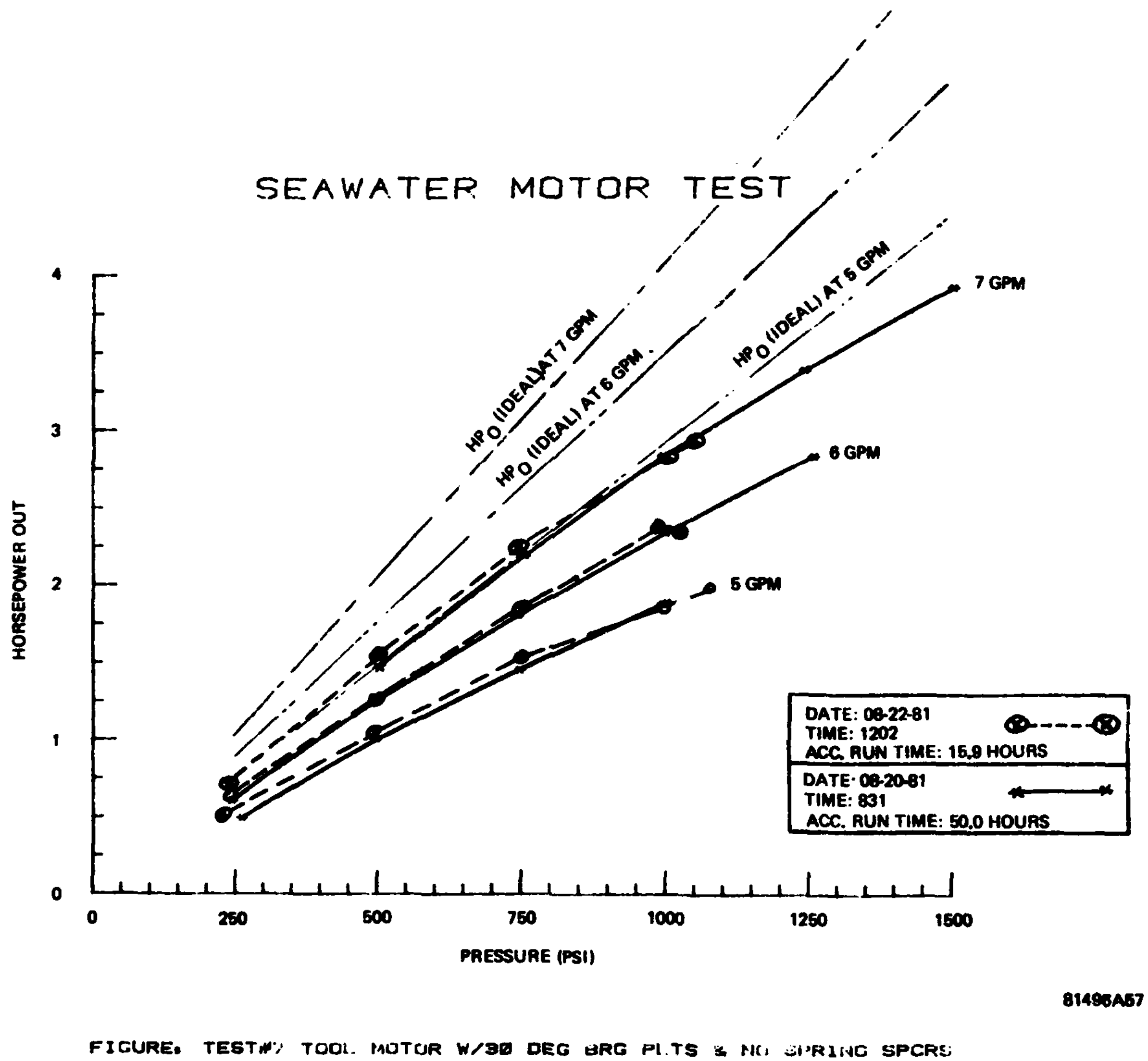

Figure 8-5. 50-Hour Performance Curves Comparison

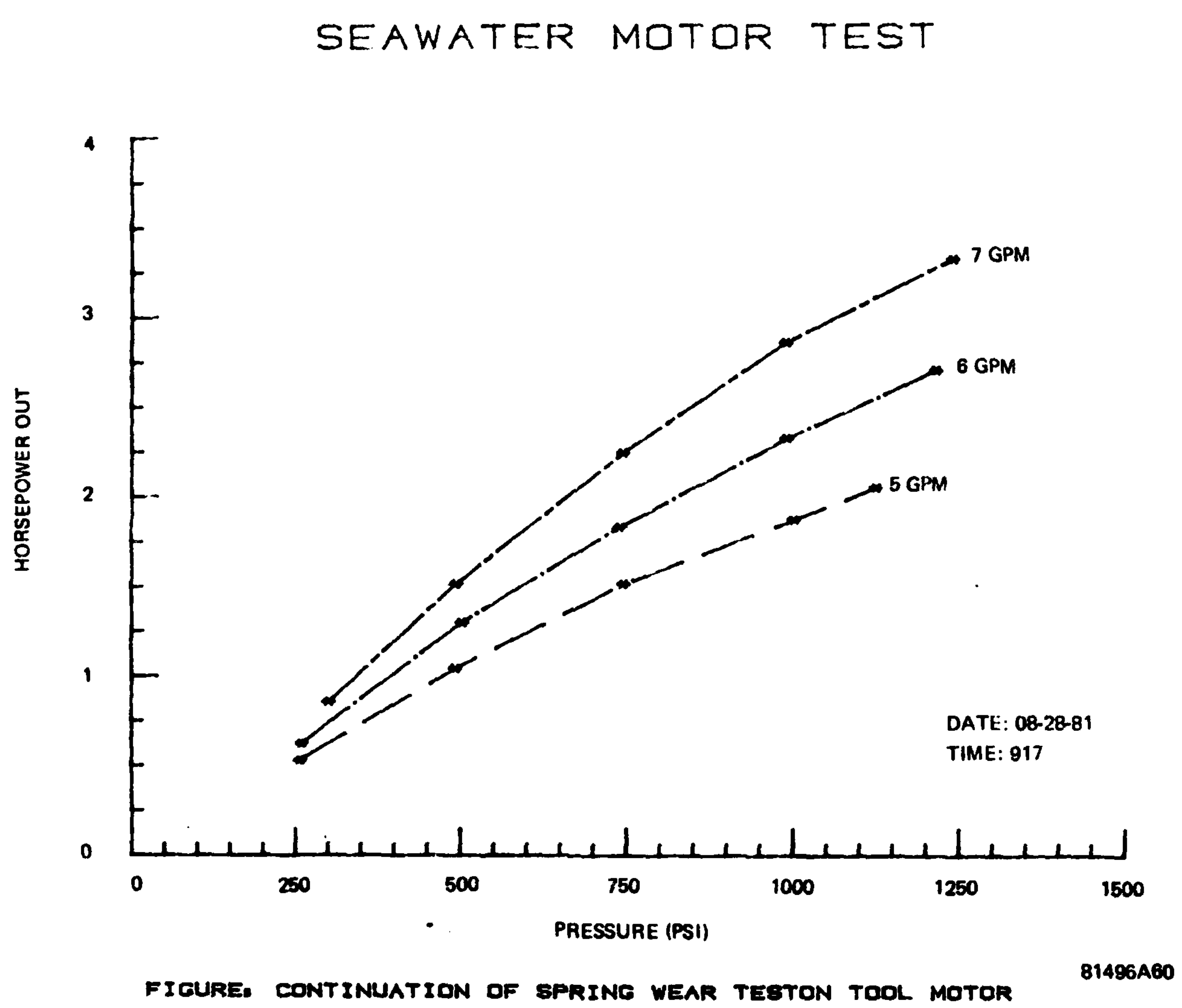

Figure 8-6. Continuation of Spring Wear Test on Tool Motor

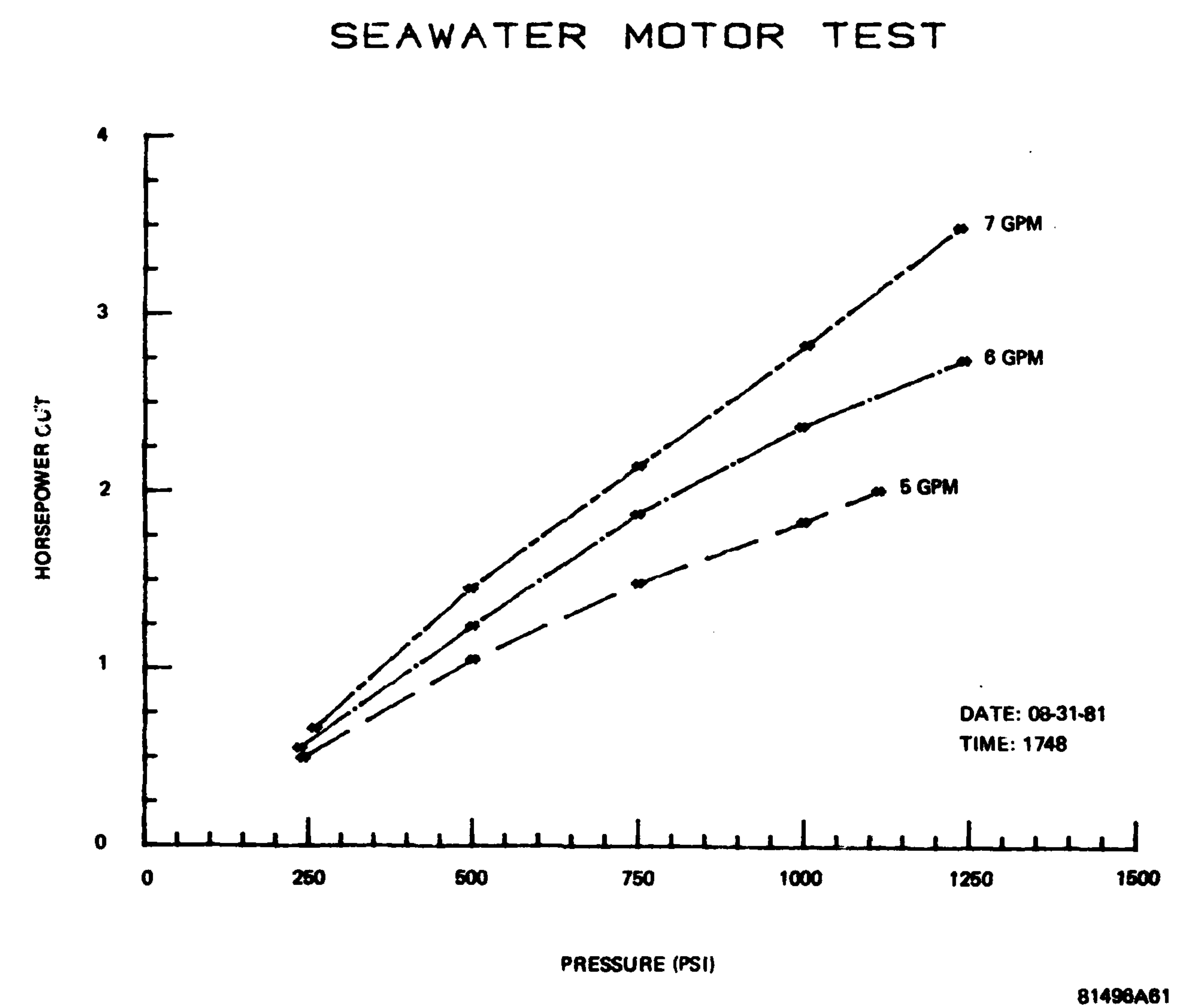

Figure 8-7. Spring Wear Test 0.035 Shim New Springs

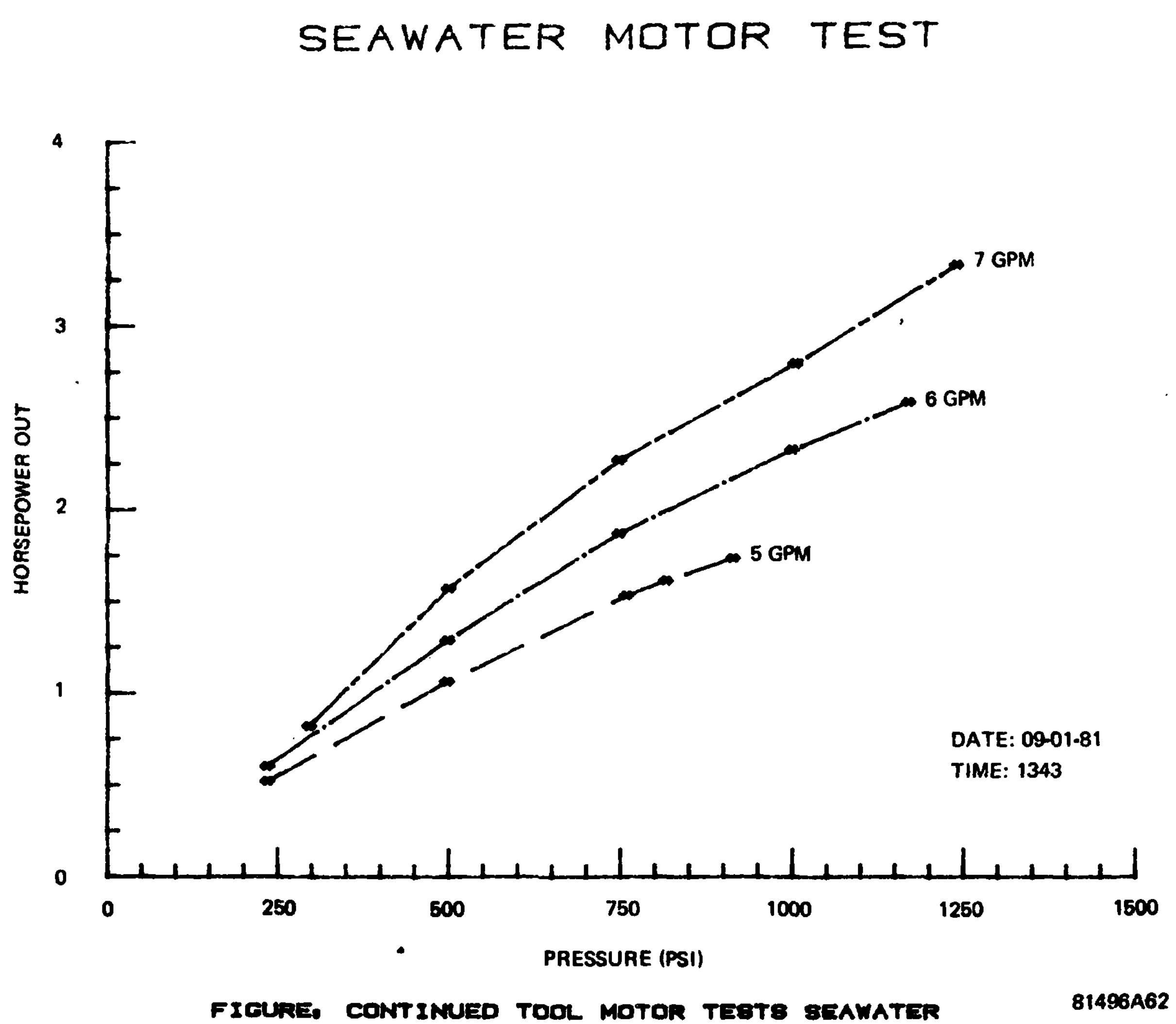

Figure 8-8. Continued Tool Motor Tests

SEAWATER MOTOR TEST

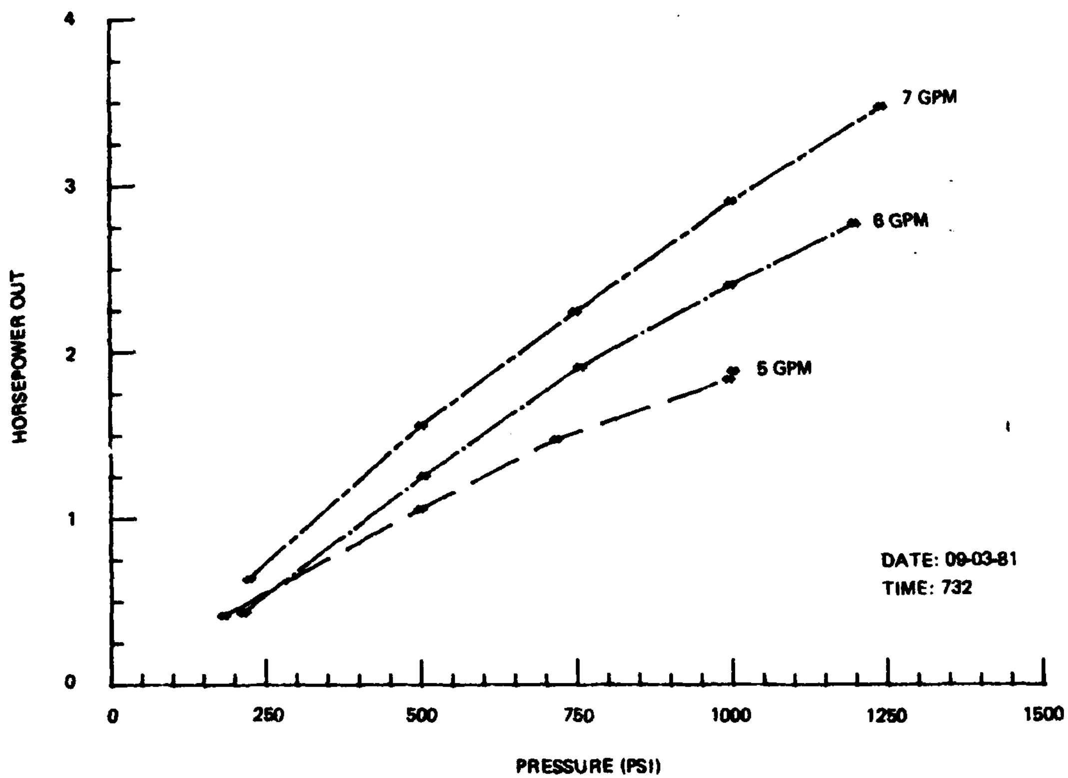

FIGURE: CONTINUED SEAWATER TOOL MOTOR

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Figure 8-9. Continued Tool Motor Tests

Results of Life Tests (Component Wear)

The life determination portion of the 50-hour tests was designed to determine the effects of component wear on the 250-hour life and increased operating pressure goals. The two most critical components in which excessive wear would shorten motor life and reduce operating pressure are the vanes and the springs.

Measurements on the vanes were taken after soaking in fresh water at room temperature (70°F) for 48 hours and again after approximately 100 hours of motor operation with the following results:

1. Vane Tip Wear:

   Mean Value = (4.2×10^{-4} in.) in./100 hr or (10.4×10^{-4} in.)/250 hr
   Max Value = (15.0×10^{-4} in.)/250 hr or 1.5 mils max vane tip wear expected in 250 hours

2. Vane Side Wear:

   Mean Value = (9.4×10^{-4} in.)/100 hr or (23.4×10^{-4} in.)/250 hr
   Max Value (50.5×10^{-4} in.)/250 hr or 5 mils max vane side wear expected in 250 hours

3. Vane side groove wear (caused from vane rubbing edge of vane slot at rotor outside diameter):

   Mean Value = (62×10^{-4} in.)/100 hr or (155×10^{-4} in.)/250 hr
   Max Value = (200×10^{-4} in.)/250 hr or 20 mils max groove depth expected after 250 hours

4. *Vane Axial Length:* Measurements showed an average increase in vane axial length of 1.6×10^{-4} in./100 hours indicating that the swelling effect from exposure to water is greater than wear.

5. Other Vane Wear/Erosion/Material Defects Observed: The tips of the 100-hour vanes were examined under a microscope. The wear marks on the vane tips looked like the highly polished surface had chipped away from rubbing or cavitation erosion. These defects could be a property of the homogeneity of the Torlon 4275 material.

During the 50-hour test, knocking was observed at higher pressures, indicating that the spring force was marginal. Spring spacers were designed and installed into the rotor spring holes (beneath the springs) to increase that force, and to allow the motor to run at the 1200 psig goal. The first set of spacers were made from stainless steel shim stock. The average thickness of the spacers was 0.027 in. with an O.D. of 0.125 in. Using these spacers increased the mean spring force to a calculated value of 1.29 pounds from 1.05 pounds without spacers. Another set of stainless steel shims were fabricated with a final thickness of 0.040 inches giving a mean force of 1.44 pounds. Table 8-2 shows the spring force versus knocking pressure points. Also, a slight difference in spring free length produces the small differences in forces indicated in Table 8-2.

Upon disassembly of the seawater tool motor after 50 hours of operation, we found that 14 of the 20 springs showed wear spots on the third to ninth coils from the end that fit into the rotor. These wear spots ranged from a slight polishing to a measurable 0.005 inches. Approximately another 50 hours of testing was performed using two new spring sets (one set for 18 hours, second set for 32 hours) to determine if spring wear was related to rotor spring hole discontinuities. The determination proved difficult to make. After motor testing was completed, the springs were sent to the spring manufacturer to determine if spring force is affected by wear. The results of manufacturer measurements on the first 50-hour springs showed a 9.7% average loss in spring force (from the calculated force), with a maximum loss of 12.9% on the badly worn spring. The 32-hour springs showed an average spring force loss of 5.2%, with a maximum of 10.7%. It should be pointed out, however, that when new unused springs were checked, the measured force was 4.9% to 8.6% less than the calculated force. Thus, actual spring force loss during test was probably lower by a corresponding amount.

8.3 IMPACTOR EVALUATION/PERFORMANCE TEST

The object of the impactor test was to drive the Ingersoll-Rand Model 2910 impact mechanism with the original seawater motor (or new seawater tool motor if time had permitted) and to determine the output under flooded and nonflooded conditions. The output is the average bolt tension produced on a Skidmore-Wilhelm Impact Wrench Tester (Model RL110) for a specific time interval. This output is referred to as the "norm" output. For a detailed description of the test setup, see Appendix E.

Basically, the test procedure was to:

1. drive the unloaded impact mechanism with the seawater motor to anticipate motor parameters required and to evaluate further test plans and goals;
2. drive the impact mechanism under load by connecting the Skidmore-Wilhelm Impact Wrench Tester and determine the "norm" output; and
3. flood the loaded impactor by using bearing lube discharge flow plumbed to the impactor housing and determine the "norm" output.

A more detailed discussion of the test procedure can be found in Appendix E.

The results of the loaded impactor evaluation tests are summarized in Table 8-4. During the tests, a 3/4"-16 test bolt was sheared and replaced by a 1"-12 test bolt assembly. Motor parameters were also measured during the tests in an effort to determine the effects on the motor and to correlate motor output to impactor output. Bolt tension measured on the Skidmore-Wilhelm Tester was converted to wrench output torque by using the Torque-Tension Relationship plot shown in Figure 8-10. This plot was made by applying known torque valves from a torque wrench and comparing these values to the wrench tester gage readings for bolt tension. The tests showed that the motor, when driven at 2 hp, operated the impact mechanism with sufficient output to meet maximum expected 1"-12 bolt torque values. Also, typical output data from the strip recorder taken during impactor tests indicated that the motor stops momentarily, but starts immediately thereafter (see Figure 8-11).

Table 8-4. Summary of Impactor Test Results

RUN NO.	DATE	CONDITION	MOTOR PARAMETERS: FLOW (gpm)	MOTOR PARAMETERS: PRESS (psig)	MOTOR PARAMETERS: SPEED (RPM)	MOTOR PARAMETERS: HORSE-POWER IN (calc)	T RUN TIME (sec)	T BOLT TENSION ($lb_f \times 10^3$)	τ IMPACTOR OUTPUT TORQUE (lb_f - ft)	τ/t (lb_f - ft/sec)	NUT ROTA-TION
1	8/11	DRY	4.5	363	2274	0.95	6.7	48.0	395	60.8	—
2	8/11	DRY	4.1	342	1540	0.82	6.7	42.5	355	53.0	—
3	8/11	DRY	3.5	419	1999	0.86	6.7	47.0	390	58.2	—
4	8/11	DRY	4.2	400	1320	0.98	81.	48.5	400	49.4	160°
5	8/11	DRY	4.2	428	—	1.05	8.1	48.0	395	48.8	160°
6	8/11	DRY	4.3	416	—	1.04	7.5	47.5	390	52.0	160°
"NORM" (1-6)			4.1	395	—	0.95	7.3	46.9	388	53.7	—
7a	8/11	DRY	5.9	600	—	2.07	5.5	56.0	455	82.7	180°
7b	8/11	DRY	5.4	620	—	1.95	6.5	55.0	448	68.9	—
8	8/11	DRY	6.0	700	—	2.45	6.5	70.0	560	86.2	270°
"NORM" (7-8)			5.8	640	—	2.2	6.2	60.3	488	79.3	—
9	8/11	DRY	2.6	—	—	—	3.9	12.0	125	32.0	—
10	8/11	DRY	3.7	355	—	0.77	5.8	49.0	400	69.0	—

FOLLOWING DATA FROM STRIP RECORDER PLOTS

RUN NO.	DATE	CONDITION	FLOW MAX	FLOW AVG	PRESS MAX	PRESS AVG	SPEED (RPM)	HP MAX	HP AVG	RUN TIME (sec)	BOLT TENSION ($lb_f \times 10^3$)	TORQUE (lb_f - ft)	τ/t (lb_f - ft/sec)	NUT ROTATION
11	8/11	DRY	7.5	5.5	522	451	934	2.3	1.4	11.2	54.5	445	39.7	180°
12	8/11	DRY	7.2	5.0	569	462	936	2.4	1.4	11.4	54.0	440	38.6	200°
"NORM"(11-12)			7.4	5.2	546	456	935	2.4	1.4	11.3	54.2	442	39.2	190°
1	8/12	DRY	7.5	5.3	546	445	922	2.4	1.4	30.0	67.0	535	17.8	--
2	8/12	DRY	7.7	5.2	569	450	926	2.6	14.	39.5	70.0	560	14.2	--
"NORM" (1-2)			7.6	5.2	558	448	924	2.5	1.4	34.8	68.5	548	16.0	—
3	8/12	DRY	8.9	6.1	759	652	1140	3.9	2.3	8.6	73.0	580	67.4	—
4	8/12	DRY	8.4	5.8	712	617	1125	3.5	2.1	7.5	75.5	600	80.0	270°
"NORM" (3-4)			8.6	6.0	736	634	1132	3.7	2.2	8.0	74.2	590	73.7	—
5	8/12	DRY	8.9	—	676	—	1084	3.5	—	Sheared 3/4 In.-16 Test Bolt Assembly				
6	8/12	DRY	8.9	—	640	—	1180	3.3	—	Used 1 In.-12 Test Bolt Assy — Sheared 3/16 In. Dia Drive Pin				
7	8/12	DRY	7.5	5.0	557	456	956	2.4	1.3	10.2	60.0	920	90.2	—
8	8/12	FLOODED	7.9	5.5	557	456	878	2.6	1.5	20.0	69.0	1055	52.8	—
9	8/12	FLOODED	8.7	6.4	783	688	1040	4.0	2.6	9.0	70.0	1070	119	—
10	8/12	DRY	9.4	6.6	735	634	1120	4.0	2.4	8.5	77.7	1175	138	—

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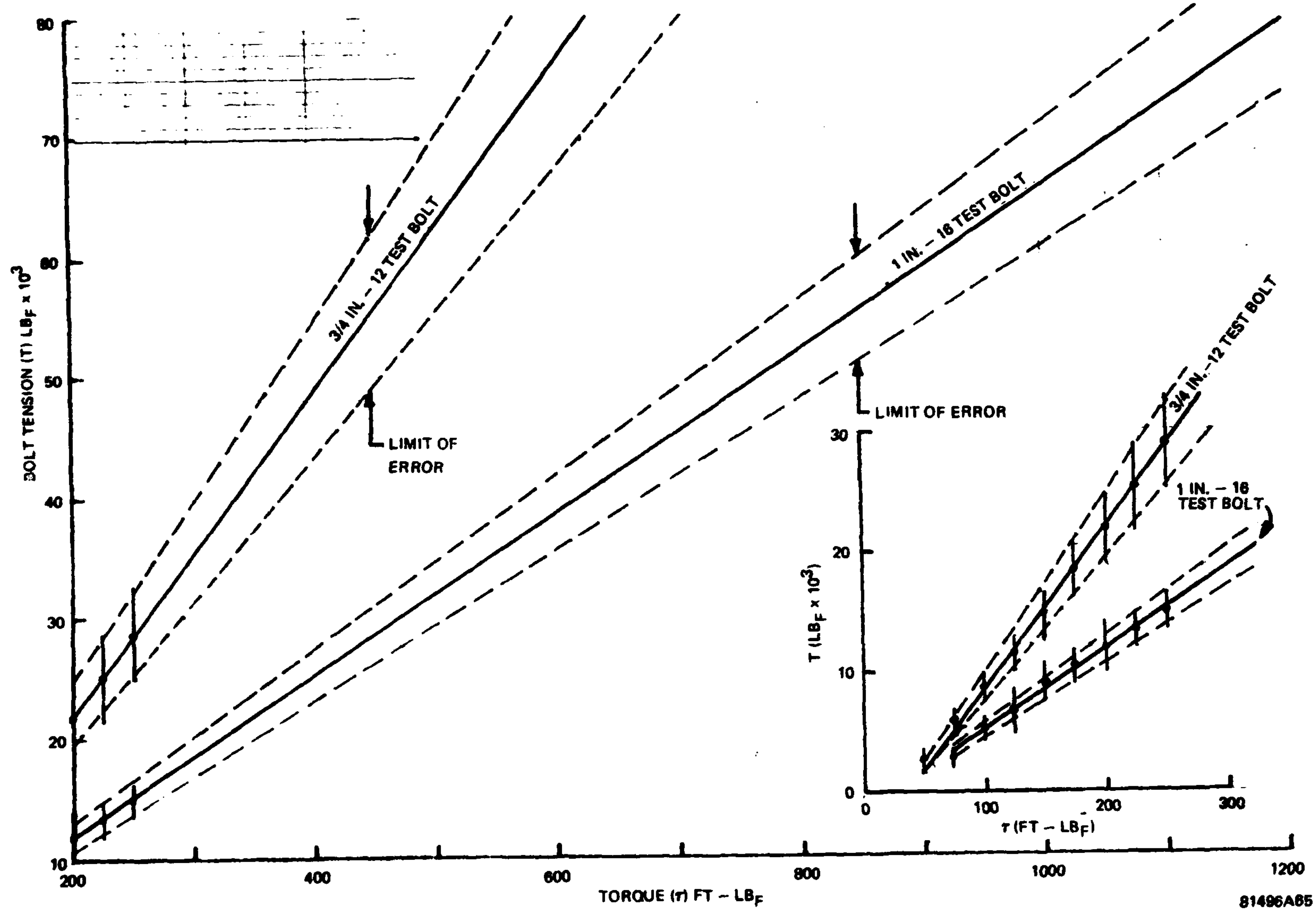

Figure 8-10. Torque Tension Relationship

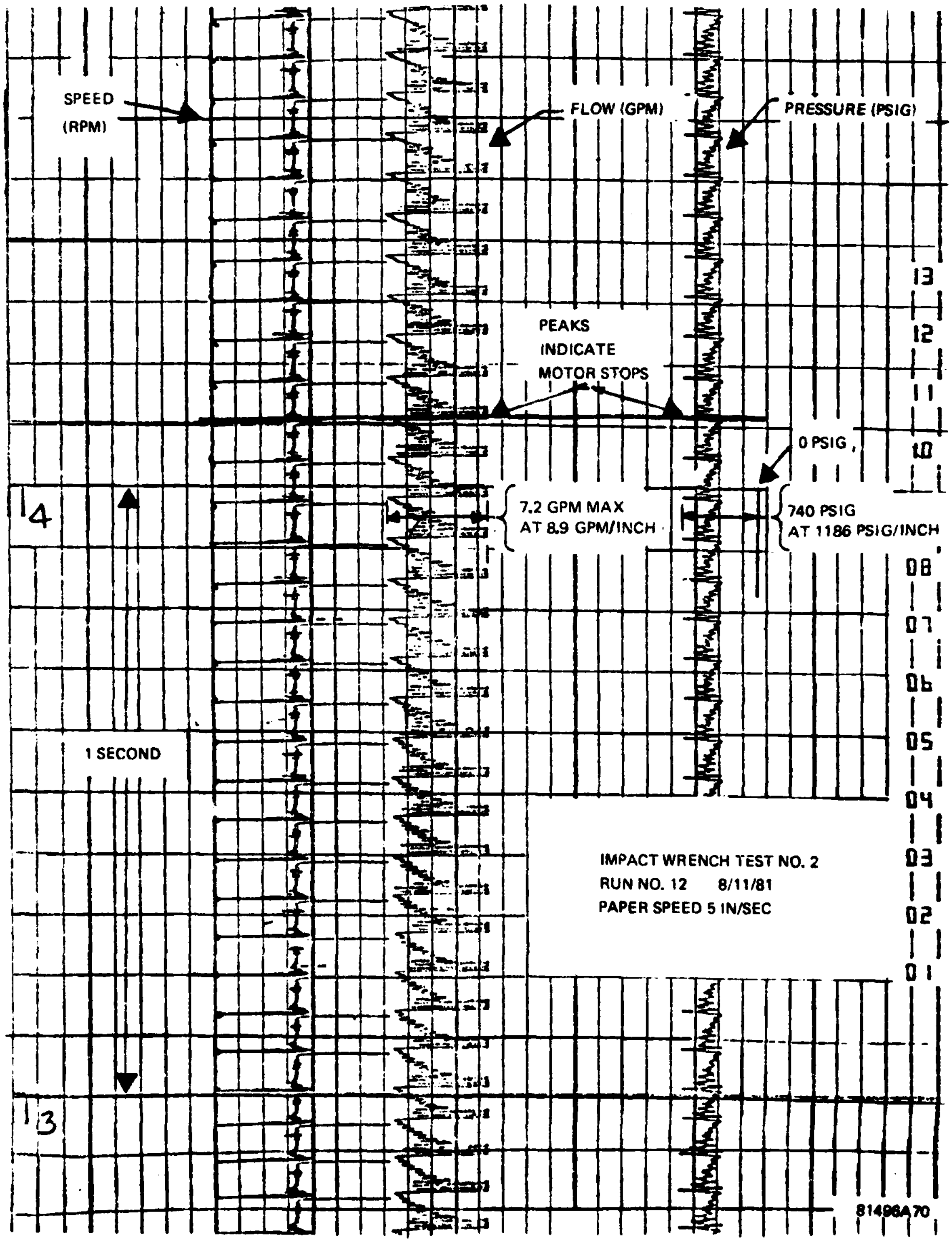

Figure 8-11. Impactor Test Data

9.0 CONCLUSIONS AND RECOMMENDATIONS

The primary object of this program was to develop and build a complete engineering model diver tool system to demonstrate that a seawater hydraulic system is capable of doing useful work over an extended period of time. To fulfill this objective, three major tasks had to be accomplished. The first was to improve the original seawater motor by extending its operating life, by increasing its operating pressure, and by making it reversible. The second task was to use the improved motor in an impact wrench assembly with capabilities similar to those of the oil-hydraulic Stanley IW06 impact wrench. Finally, the third task was to power the wrench assembly by a seawater hydraulic power source capable of providing dependable service with the impact wrench and other seawater tools to be developed in the future. The overall program successfully met the requirements of these tasks.

Based on the preceding sections of this report, the conclusions and recommendations are categorized as follows:

1. Seawater Tool Motor
2. Impact Wrench Assembly
3. Seawater Hydraulic Power Source

9.1 SEAWATER TOOL MOTOR

Conclusions

1. Spring force and spring wear still present a problem to be studied with respect to 1200 psig operation for a life of 250 hours. During the development spring force tests, the optimum spring force of 1.0 pound was determined. However, these tests were not run for an extended period of time, thereby neglecting the effects of wear and set on the spring force. After a spring is used for a period of time, it will take a set and provide a constant force if wear and in homogeneities in spring material are neglected. However, wear on springs reduces this spring force proportionally with time. As the tool motor performance tests showed, the 1.0 pound force springs worked initially, but after approximately 35 hours, "knocking" occurred at lower than desired operating pressures. Spring force tests performed on the 50-hours springs showed a decrease in force from 7% to 13%; the spring with the most visible wear had the highest decrease. The most probable cause of spring wear is shown in Figure 9-1. As can be seen from the figure, the depth of the vane side wear groove affects spring wear also.

2. The vane side wear groove is another problem to be studied to achieve extended life at higher operating pressures. Measurements made after 100 hours indicate the maximum depth of this wear groove is expected to be 20 mils after 250 hours, with the configuration shown in Figure 9-1. Even though the vane is not expected to fracture at that time, excessive vane "cocking" will occur, which in turn will cause more spring bending, higher spring wear, and reduced spring force. The smaller area of the sharp edge on the 0.03 inch chamfer causes a higher PV value and thus a higher wear rate when compared to the vane tip wear.

3. Generally, vane tip wear was excellent, with only 1.5 mils maximum wear expected after 250 hours. However, the tips did experience some pitting, probably due to cavitation from knocking or in homogeneities in the vane material.

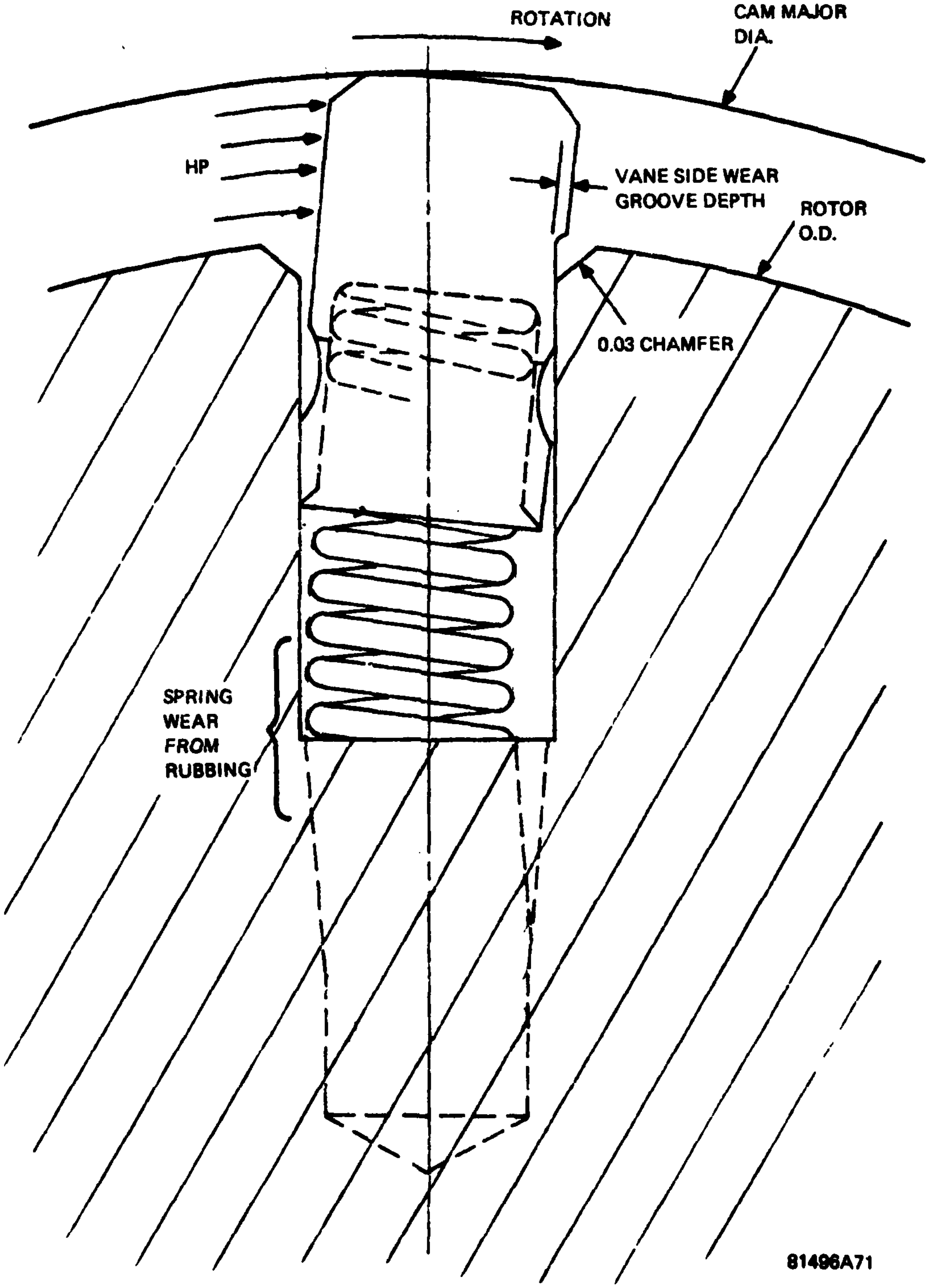

Figure 9-1. Spring Wear and Vane Side Groove Wear

4. During the entire 100-hour test period, the motor's overall efficiency averaged about 70% at 6 gpm and 825 psig input (2.0 horsepower out) with slightly lower efficiencies (approximately 68%) experienced at higher pressures and increased motor outputs (3 to 4 horsepower). As indicated by test results of Section 8, with increased spring force, the overall efficiencies increased to 72%, a little below the design goal of 75%. The most dramatic drops in efficiencies came when the axial lengths of the vanes were shaved and when the motor was reversed. This was expected, however, because of increased leakage past the vanes and because of absence of sufficient vane tip "wear-in."

5. The tool motor is reversible. The new tool motor was operated in a reverse direction, and the only difference in performance was an initially large drop in overall efficiency. As the motor was operated, the efficiency increased with time (as expected) because slight vane tip "wear-in" decreases leakage past the vane.

6. The new tool motor was more economical to manufacture and assemble because of elimination of the transfer drilling and pinning of the housings and because of the elimination of the final reaming of the bearings. More value and design engineering will be required in the future to further reduce motor production costs. Also, analysis should be performed to see which tight tolerances can be eliminated, and to determine the effects of tolerance stack-up on overall motor performance.

Recommendations

1. Perform further testing on new motor (SN001) to determine if increased spring force will improve higher operating goals and if a dry film lubricant (Tiodize) will improve spring wear. The recommended test sequence is as follows:

a. Use new vanes and Tiodized springs (SKB 376735H07).

b. Install 0.027 inch thick spacers in spring holes. These spacers will increase mean spring force to a nominal calculated value of 1.25 pounds, with margins of safety of 8.1% for fatigue and 6.4% for buckling.

c. Measure and record spring force when compressed to 0.136 inches. This value should be 1.25 pounds.

d. Install spacers and springs in rotor, noting location of springs with respect to rotor holes.

e. Run tests at a level to maintain a pressure of 1200 psig at 1500 rpm.

2. For the tool rotor (SN001), change the 0.03 chamfer at the vane slot edge to a 0.010 inch (maximum) radius. To further improve vane side groove wear, a slight taper (2°) could be machined in the rotor vane slots as shown in Figure 9-2.

3. In addition, if spring wear still presents a problem after the test recommended in (1), a Torlon insert could be installed at the point where the spring rubs against the rotor (see Figure 9-2).

4. If spring force still remains a problem for 1200 psig operation, an 0.038 inch thick spacer could be used under the springs. This would increase the mean force to a nominal 1.35 pounds with margins of safety of 6% for fatigue and 1/2% for buckling.

5. To improve vane tip wear and possibly eliminate cavitation pitting effects, Torlon 4347 could be substituted for the Torlon 4275 vane material.

6. Because spring force and spring wear are the major obstacles to the tool motor design goals, the Pressurized Vane Motor should be studied further.

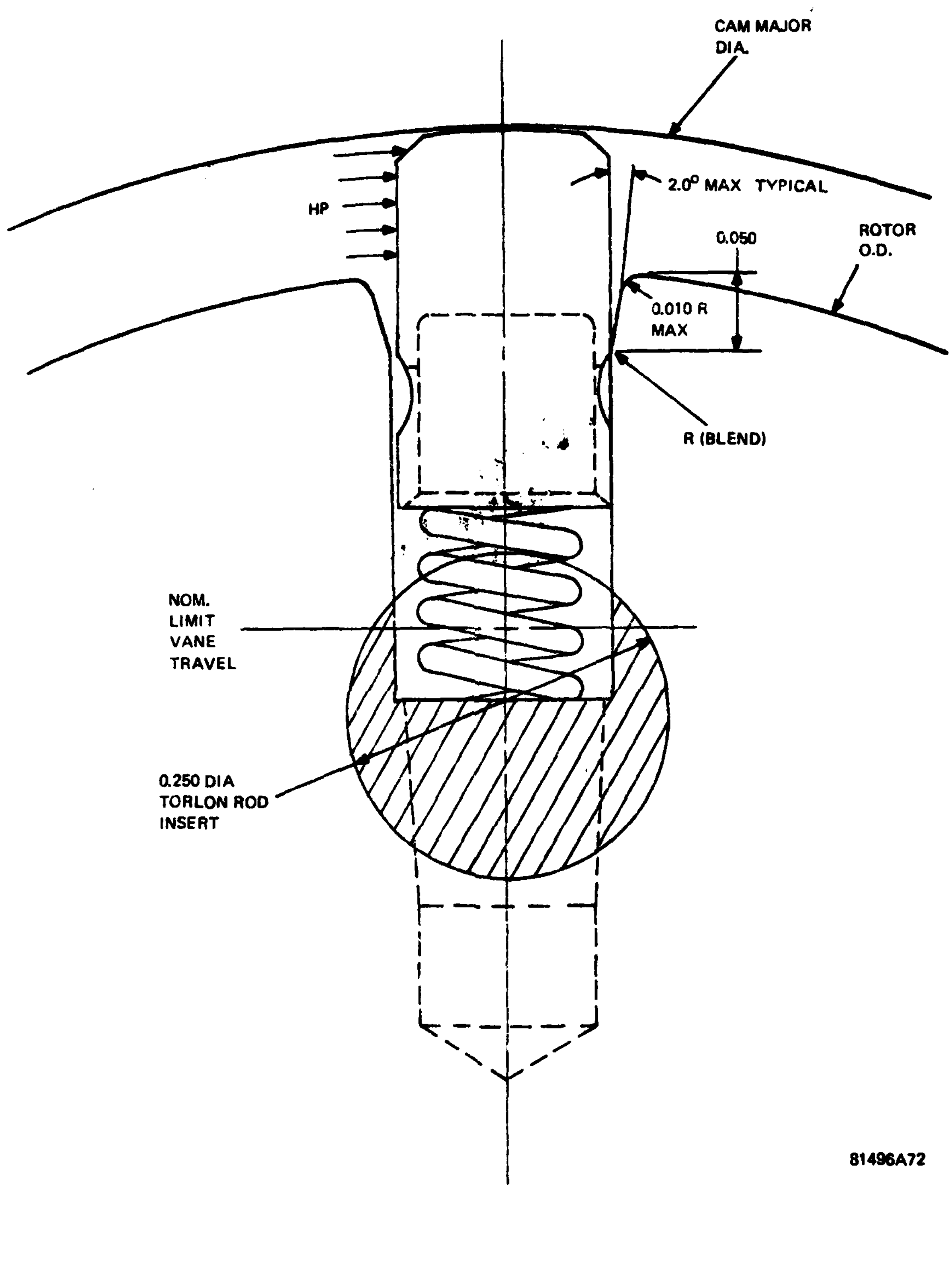

Figure 9-2. Recommended Rotor Vane Slot Improvements

9.2 IMPACT WRENCH ASSEMBLY

Conclusions

1. The impactor tests conducted at NCEL showed that the seawater motor, operating at 2.0 output horsepower, could directly drive the Model 2910 impactor with sufficient output torque values (250 to 400 foot-pounds).

2. Impactor test data indicates that the motor stops momentarily, but starts immediately thereafter.

3. Driving the impactor directly by the seawater tool motor may reduce motor life because of high stresses from the cavitation that occurs when the motor suddenly stops.

4. The spring force in the tool motor may have to be increased to improve the low speed characteristics for direct driving the impactor.

5. Although the Aluminum-Nickel-Bronze coupling was not tested, coupling life was calculated to be 7500 hours (4.5×10^6 cycles @ 1000 rpm) with the spline interfacing with the impactor failing at that time.

6. Copper ions from the Copper-Nickel heat exchanger in the SWHPS will cause a corrosion problem in the cast aluminum handle over an extended period of time.

7. The carbon steel housing of the dirt excluder will corrode after the lubriplate coating wears away.

8. The Torlon 4301 valve spools may swell after long term immersion in seawater and could cause stiff valve action.

9. With the direct drive arrangement, overall handle and motor size and weight could be reduced for better balance.

10. Without corrosion protection and with the dissimilar metals of the motor, coupling, and impactor, the cast aluminum handle and the steel impactor will eventually corrode.

Recommendations

1. With new vanes and increased force springs in the tool motor, test the wrench assembly. Inspect vane tips for cavitation erosion. Also inspect the aft bearing plate in the motor to determine if any thrust loads were transferred. One impact wrench cycle is estimated to be six seconds at 1000 rpm. For 250 hours, this gives 1.5×10^5 impact cycles or 7.5×10^4 reverse impact cycles.

2. Use of a titanium handle would help eliminate corrosion problems from dissimilar metals and from copper ions from heat exchanger. An alternative would be to hard anodize the aluminum handle and install titanium inserts in the handle at the valves. With these inserts, Inconel or stainless steel valves could be used.

3. The steel dirt excluder should be replaced with Inconel or 316 stainless steel.

4. To reduce size and weight of the wrench assembly, the handle porting could be modified and the tool motor forward housing could be eliminated.

5. To reduce corrosion, the impactor housing material could be changed from steel to aluminum (or titanium), and the impactor assembly could be sealed and packed with grease.

6. A replaceable Torlon spline liner could be used between the rotor and coupling splines to reduce shock loads to the tool motor.

7. Finally, four 0.250 inch diameter holes could be drilled into the coupling (see Figure 5-6) to allow lubricating flow into spline teeth and to decrease the coupling's torsional spring constant. Reducing the torsional spring constant would improve motor acceleration characteristics.

9.3 SEAWATER HYDRAULIC POWER SOURCE

Conclusions

1. The design of a SWHPS for use in the development of seawater hydraulic tools and devices has been completed. Standard commercial products were used for almost all major components. Many of the standard components are normally manufactured from corrosion resistant materials and the rest were easily modified to be corrosion resistant.

2. A great deal of effort was expended in analyzing standard hydraulic designs for compatibility with a seawater working fluid. Commercial oil hydraulic valves predominantly use close tolerance sliding metal to metal interfaces. These designs depend on working fluids with high lubricity. Additionally, they are susceptible to sticking or accelerated wear when subjected to fluids with contaminants. These qualities cause most commercial oil hydraulic designs to have a low probability of success in seawater.

3. Designs were sought, therefore, which possessed synthetic wear members instead of metal to metal combinations, and used seats for sealing instead of spools. Both the back pressure regulator and output pressure control valve possess such features. The flowmeter has moving parts of synthetic materials as well. The flow control valve manufacturer attempted to substitute a synthetic poppet into their design, but found the strength of the material was not sufficient without a major redesign of the valve.

4. The designs selected for all components are suitable for a development system; however, it is believed that more development and design effort must be expended to attain a system for fleet use. The effort should concentrate on material selection, mechanical operation, and safety.

Recommendations

1. Development of major system components with high acceptability to contaminants and corrosive media is needed for long lived maintenance free systems.

2. A more power efficient variable displacement pump could be developed for a more versatile system.

3. A long range plan for seawater hydraulic systems development detailing the required steps should be prepared to lay out a logical and productive program continuation.

4. Filter performance and alternate filtering schemes should be examined to ensure successful operation in possible harsh environments.

5. To improve SWHPS safety, install a relief valve in the reservoir. The relief valve will function to prevent pressure build up in event that the vent filter becomes clogged with salt (See Appendix G).

6. Install an automatic engine shutdown package on future models of the SWHPS. This package protects the power unit from damage due to the following:

a. Low engine oil pressure
b. Fanbelt breakage (engine cooling fan)
c. High engine temperature
d. Low water level in reservoir

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