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Design of a cold-flow test facility for the high-pressure fuel turbopump turbine of the space-shuttle main engine

Studevan, Colin C.

Monterey, California. Naval Postgraduate School

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Design of a Cold-Flow Test Facility for the High-Pressure Fuel Turbopump Turbine of the Space-Shuttle Main Engine

by

Colin C. Studevan
Lieutenant, United States Navy
B.S., U.S. Naval Academy, 1985

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ABSTRACT

The design and installation at the Naval Postgraduate School of a cold-flow test facility for the turbine of the high pressure fuel turbopump of the Space Shuttle Main Engine, is reported. The specific article to be tested is the "Alternate Development Model" designed and manufactured by Pratt & Whitney. The design of individual components is documented. The installation of the facility subsystems is described in detail. A preliminary estimation of turbine performance is made.

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I. INTRODUCTION

The purpose of this report is to describe the design and installation at the Naval Postgraduate School's Turbopropulsion Laboratory (TPL), of a cold-flow test facility for a high-work turbine. The test turbine, intended to power the liquid fuel pump on the Space Shuttle Main Engine (SSME), was designed and manufactured by Pratt & Whitney (P&W) as the Alternate Turbopump Development (ATD) model. The original High-Pressure Fuel Turbopump (HPFTP) for the SSME was designed and manufactured by Rocketdyne. The HPFTP consists of a high pressure hydrogen pump driven by a two-stage, axial-flow turbine. The turbine is a low-pressure ratio reaction turbine driven by a mixture of steam and gaseous hydrogen. The turbine produces approximately 73,000 hp. At its design point, the turbine operates with an inlet temperature of about 1,900°R and an inlet pressure of approximately 5,200 psi.

NASA's Marshall Space Flight Center established a "cold-flow" testing facility to experimentally determine the performance of the SSME HPFTP. The facility was a short-duration, blow-down rig, and the turbine was spun up to speed prior to activating the airflow. Hudson et al [Ref. 1] performed the first tests on the Rocketdyne HPFTP. Subsequent testing of the Pratt & Whitney ATD HPFTP was conducted by

Gaddis et al [Ref. 2] for "back-to-back" comparisons with the Rocketdyne HPFTP. The turbine performance and turbine parameters such as airfoil surface static pressure distributions, static pressure drop through the turbine, and exit swirl angle were investigated at the design point, over its operating range, and at extreme off-design points. In the present work, following a program of tests using only a single stage, two-stage performance data will be obtained and compared with those reported by Gaddis.

The purpose of the present installation was not primarily for performance measurements. Rather, it was to continue and extend a program of time-resolved flow measurements in turbomachines for the purposes of both evaluating advanced designs and validating viscous flow analysis codes. The complex turbine test rig (TTR) at TPL, last reported by Kane (1978) and Eargle (1980), was not suited for this purpose. The rig was expressly designed not to require interblade measurements, could not readily allow access for optical measurements, and the turbine stages available for the rig were of no current interest. The rig was therefore removed from the test cell, and the installation described in the present document was put in its place. Where possible, components of the old rig were refurbished and reused (air supply system, hydraulic dynamometer, test table and rotor support), however the installation was a complete redesign and required the manufacture, installation and alignment of all components between the flange from the air supply system and the drive shaft to the dynamometer.

In the present document, the design and installation of the new turbine test facility are described. Chapter III gives results of some preliminary performance predictions for the rig. Chapter IV contains conclusions and recommendations. Engineering drawings and other system documentation are contained in the Appendices. It is noted that the present documentation was completed prior to initial tests to check out the installation.

II. FACILITY DESIGN AND INSTALLATION

A. AIR SUPPLY SYSTEM

Air to drive the HPFTP turbine was supplied by an Allis-Chalmers twelve-stage axial compressor. The compressor is driven by a 1250 horsepower electric motor; at a rotational speed of 12,000 revolutions per minute, pumping a maximum volume flow rate of 10,000 cubic feet per minute, to a maximum pressure ratio of 3 to 1. The Allis-Chalmers compressor is shown in Figure 1. The air supply system arrangement for the High Speed Laboratory of the TPL and the test cell is shown in Figure 2.

During normal operation, air enters the compressor through a filtered intake open to the atmosphere. The air then passes through the 12-stage axial compressor, and enters the aftercooler where the air is cooled to about 110°F. Before leaving the compressor room, the air flows through a flow metering orifice in the facility supply line. After entering the TTR the air flows through a manually-controlled butterfly valve, before entering the TTR plenum chamber. The air supply from the plenum chamber is remotely controlled by a motor-driven butterfly valve. The air exits the plenum chamber through an eight-inch pipe, passes through an eight-to-ten

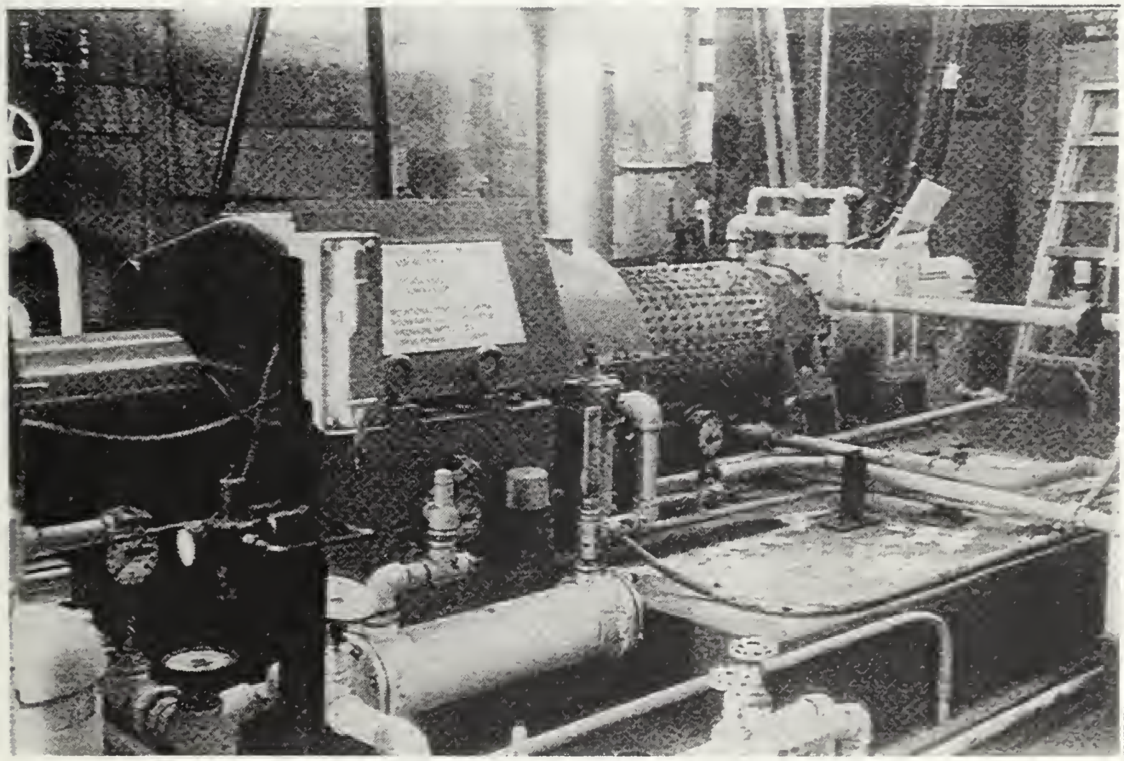


Figure 1. Allis-Chalmers Compressor

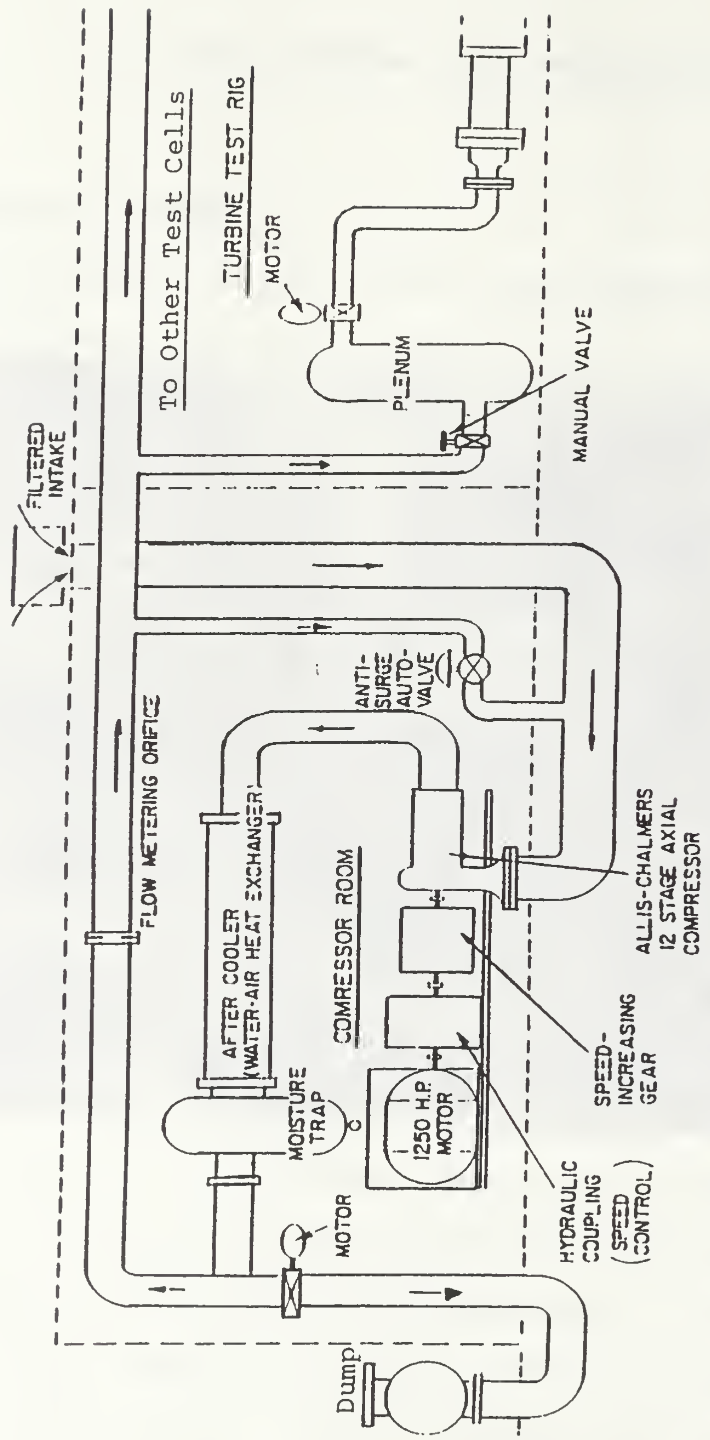


Figure 2. Air Supply for the High Speed Laboratory

inch pipe expander, and flows into the turbine inlet piping. A two-degree taper was machined on the inside surface of the inlet piping which mated flush against the inlet strut housing of the test turbine. Access holes were drilled at 120° around the inlet piping for installing performance instrumentation such as inlet total pressure and temperature probes. Engineering drawings of the inlet piping are contained in Appendix I.

B. TURBINE ASSEMBLY

High pressure air from the inlet piping, flows through the inlet strut housing into the turbine as shown in Figure 3. The air then flows axially into the first stage stator which imparts swirl to the flow. The swirling airflow is turned through the rotor, which produces shaft power, before exhausting into the test cell. The shaft power is absorbed by a water dynamometer, shown schematically in Figure 3.

The HPFTP test specimen was supplied by Pratt and Whitney. The components which were supplied included the inlet-strut housing, the test rotor, the first and second stage stator and rotor blades, the outer casing, and the retainer ring for the first stage stator.

The remaining components that were needed to complete the test facility were either designed and machined, or were purchased from commercial vendors. Drive-shaft components which were designed included the main bearing shaft, the

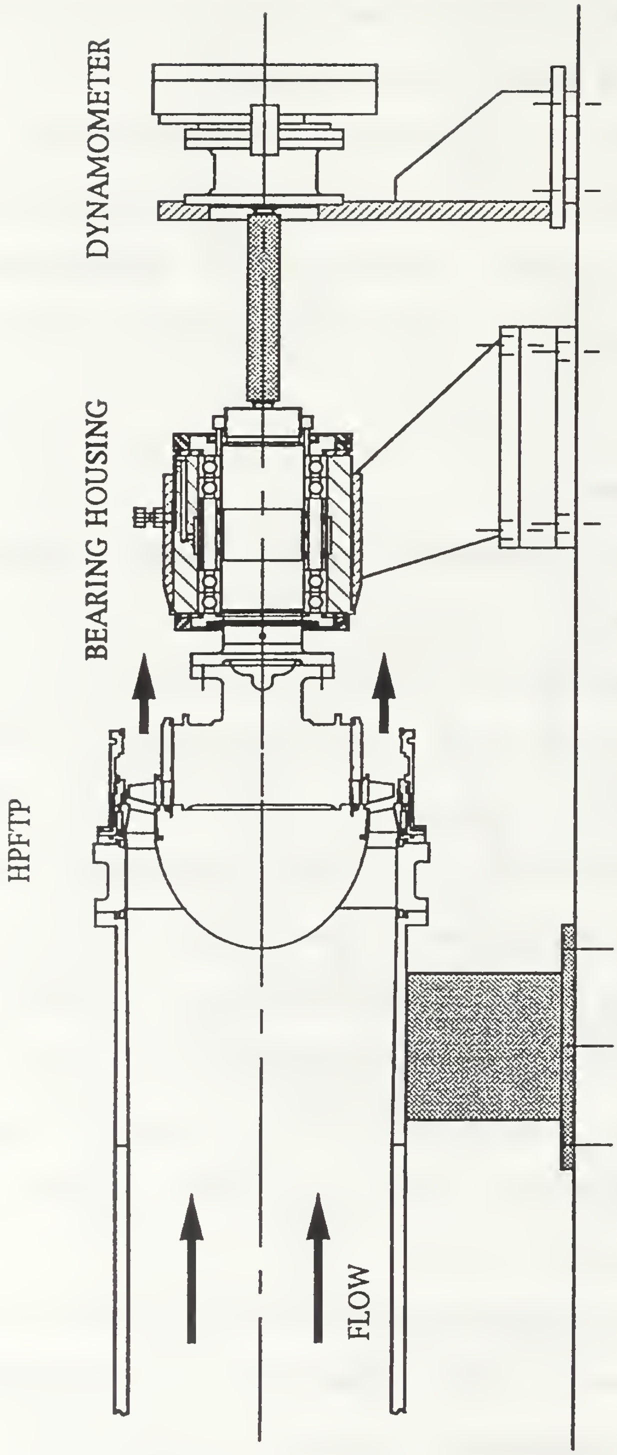


Figure 3.   TTR Layout

bearing sleeve, the bearing spacers and retainer rings, and the oil seals. Component diagrams are contained in Appendix I.

The test rotor consisted of a single stainless-steel disk with fifty "fir tree" slots broached along the entire length to accommodate an equal number of first and second stage rotor blades, tabs, and pan weights. The slots were marked and numbered clockwise from 1 to 50 when looking forward from the rear. The blades were separated by tabs and pan weights and were held in place by spring loaded retainer rings on each end of the rotor disk. The pan weights had labyrinth knife edges which formed the seal underneath the shrouded hub of the second-stage stator. A view of the rotor is shown in Figure 4.

Each of the first stage rotor blades was weighed and then mounted so that blades with equivalent mass were positioned 180 degrees apart. The pan-weight spacers were mounted in identical fashion. P&W supplied TPL with two sets of second stage blading. Initial testing of the turbine was intended to use only the first stage stator and rotor. To safely interlock the blading on the rotor disk, the extra set of second stage blades were cut off at the root and ground to be equivalent in height and weight. The blades, tabs, and pan weights fed easily and securely into the "fir tree" grooves on the rotor disk.

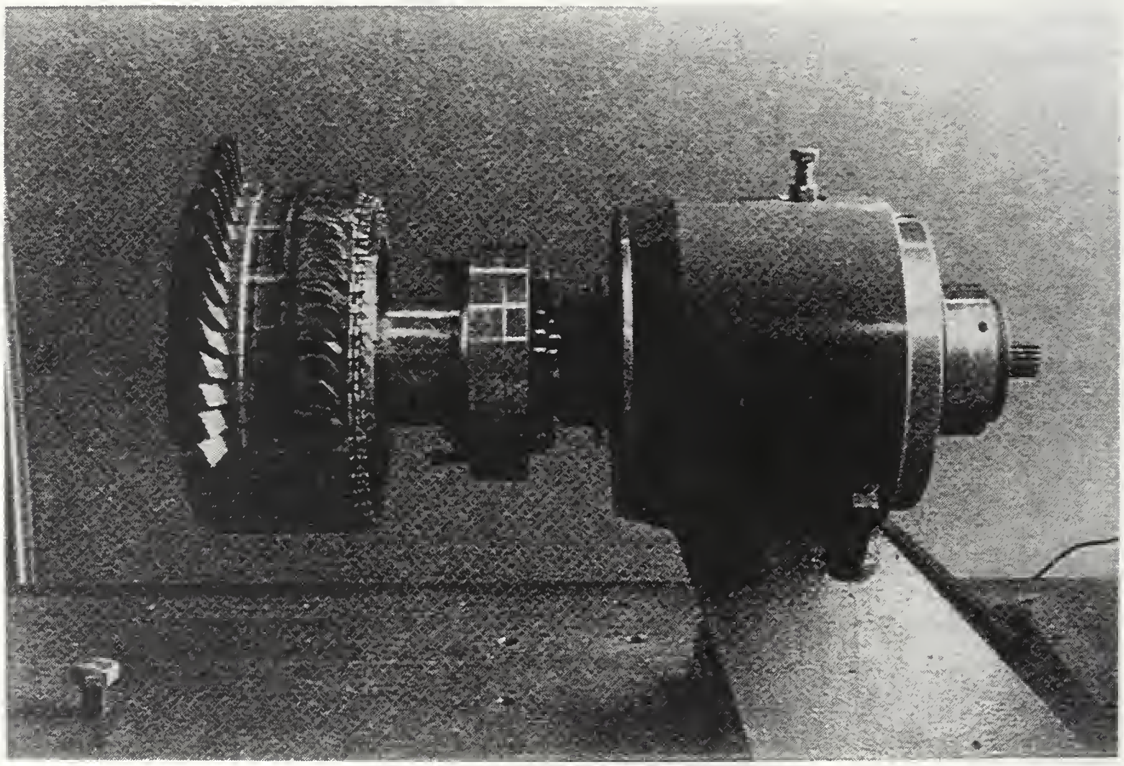


Figure 4. Test Rotor with Sample Blading

When mounting these components on the rotor disk, care had to be taken to ensure a correct fit. The components were machined to self-lock once in place. To mount correctly, components were initially slipped into place so that only 1/8" was inserted into the groove. Then, moving clockwise (looking forward from the rear), each blade, tab, or pan weight was slipped into the 1/8" position sequentially until all components of that row were similarly mounted. Once all components of that row were inserted 1/8" into the groove, each blade was slowly moved forward, in sequence, until all components of that row were in their most forward position (Fig. 4). An exact listing of individual blade positions is given in Appendix II.

The main shaft was designed to separate aft of the flange which was bolted to the test rotor (Fig. 3). This was done to facilitate the mounting of all components into the bearing housing sleeve. The shaft incorporated a 3/16" diameter hole which extended through the flange and the shaft so that once in place a lock pin could be installed. There were two bearing surfaces along the length of the shaft, one forward and one aft. The Fafnir high-precision bearings were mounted to the bearing shaft in pairs. Each bearing was marked with a thrust point on the outer and inner races. Following manufacturer's recommendations for proper installation, each pair of bearings was mounted so that the outer race thrust points were aligned and faced together and the inner race thrust points were aligned and faced away from each other as shown in Figure 5.

During assembly, the flange was separated from the main bearing shaft. The first pair of bearings was inserted onto the shaft from the aft end and moved forward to rest against the retainer flange forward of the bearing surface. The bearing inner spacer was then inserted onto the shaft so that it rested against the inner race of the first set of bearings. The bearing outer spacer was inserted over the bearing inner spacer so that its edges rested against the outer race of the bearing. The second pair of bearings was then slipped onto the aft bearing surface of the shaft until the inner race of the bearing rested against the bearing inner spacer (Fig. 5). The oil-flinger was mounted onto the shaft against the inner race of the second set of bearings. A circular locking nut was used at the end of the shaft to secure the components onto the shaft. The entire assembly was inserted into the main bearing sleeve. Bearing retainers were then secured to each end of the sleeve. Installation of the oil-seals over the bearing retainers completed the main-bearing sleeve assembly.

C. HYDRAULIC DYNAMOMETER

The energy output of the turbine was designed to be absorbed by a 250-horsepower Series 061 hydraulic dynamometer manufactured by Kahn Industries, as shown in Figure 6. A

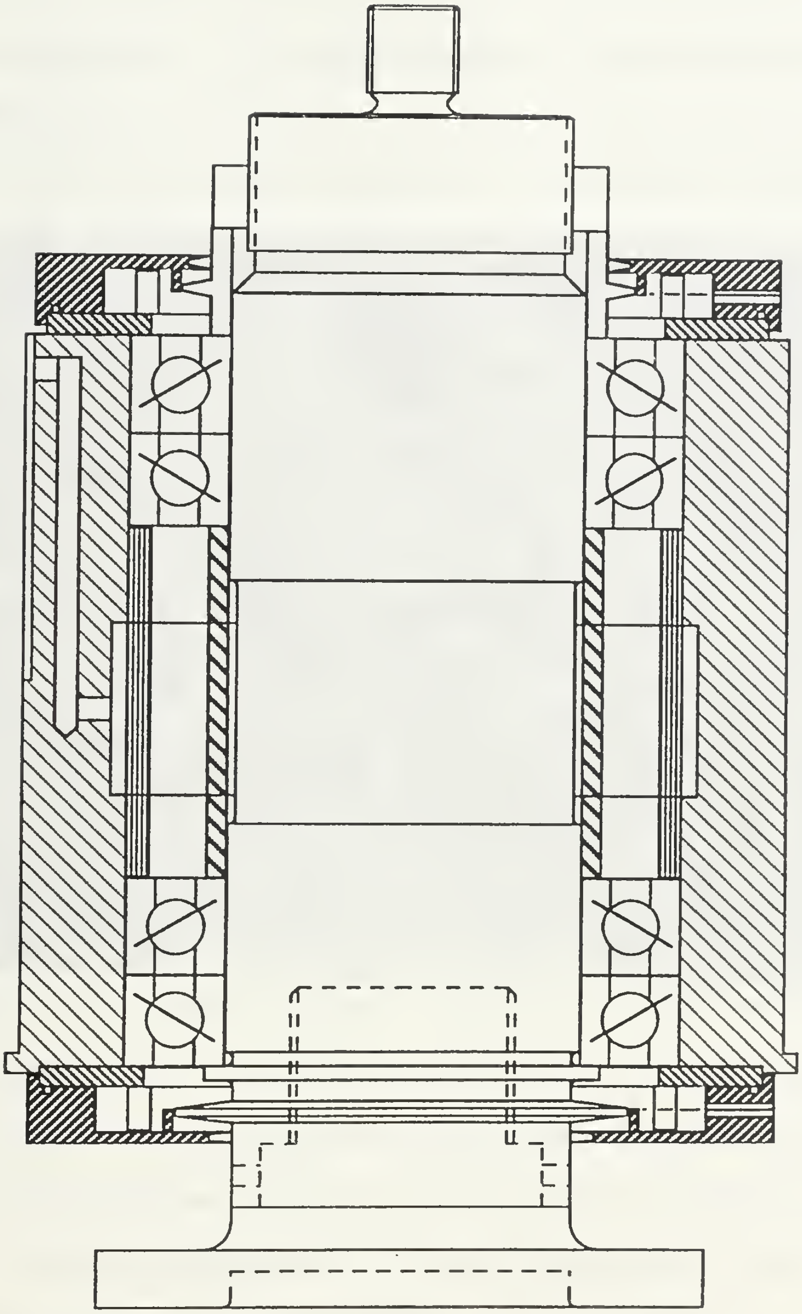


Figure 5. Bearing Sleeve Assembly

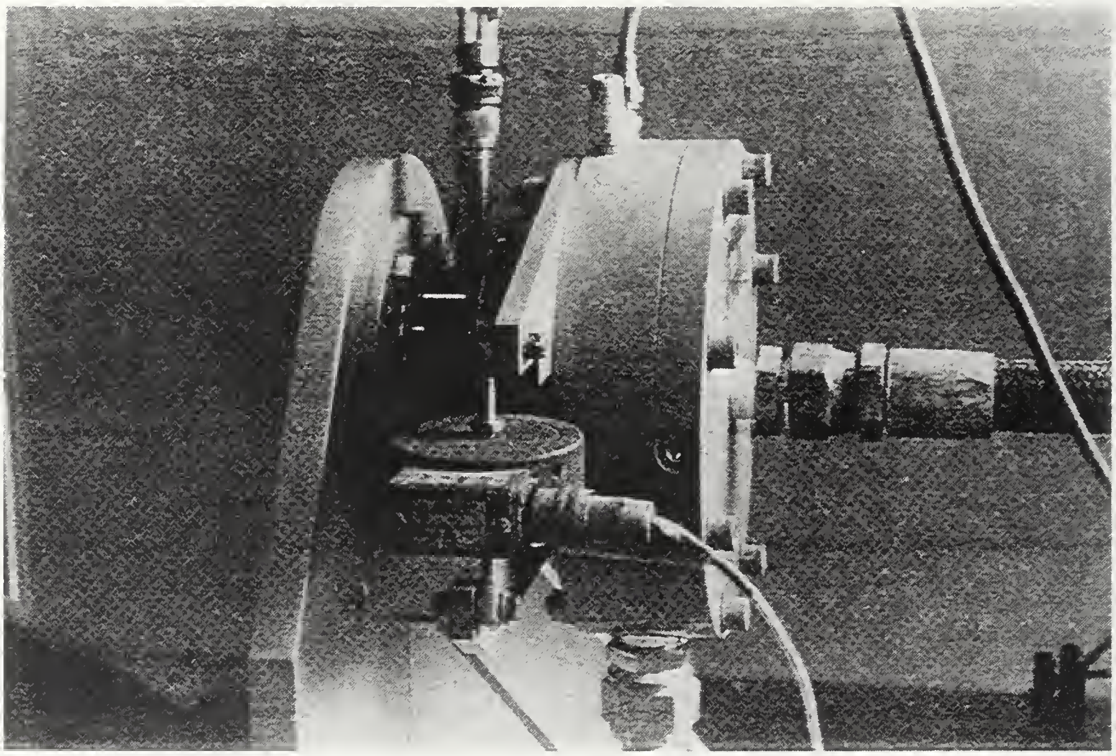


Figure 6. Hydraulic Dynamometer

schematic of the automatic load control system is shown in Figure 7. The dynamometer consisted of one perforated disk enclosed in a housing which rotated between similar sets of perforated stators. In the Kahn design, cold water enters the rotor chamber at the center. The water is accelerated by the rotating disc and thrown outward by centrifugal action. From the outer diameter of the rotor chamber inward, the water forms an annulus which rotates at approximately half of the angular disc speed. The resulting centrifugal pressure discharges the water through a radial hole at the bottom of the housing. Power is absorbed by vortices created in the rotor and stator holes. The resulting drag acts as a resistance to rotation and tends, with an equal torque, to turn the dynamometer housing. The power absorbed by the dynamometer is a function of rotor speed and of the water level in the rotor chamber.

As shown in Figure 7, the water level in the dynamometer is regulated by a set of Fischer control valves located at the water inlet and outlet lines. Opening of the inlet control valve or closing of the outlet control valve causes the water level to increase. Closing of the inlet control valve or opening of the outlet control valve causes the water level to decrease. Power absorption by the dynamometer gives rise to an increase in water temperature. In order to remove the heat, a continuous flow of water through the dynamometer is required.

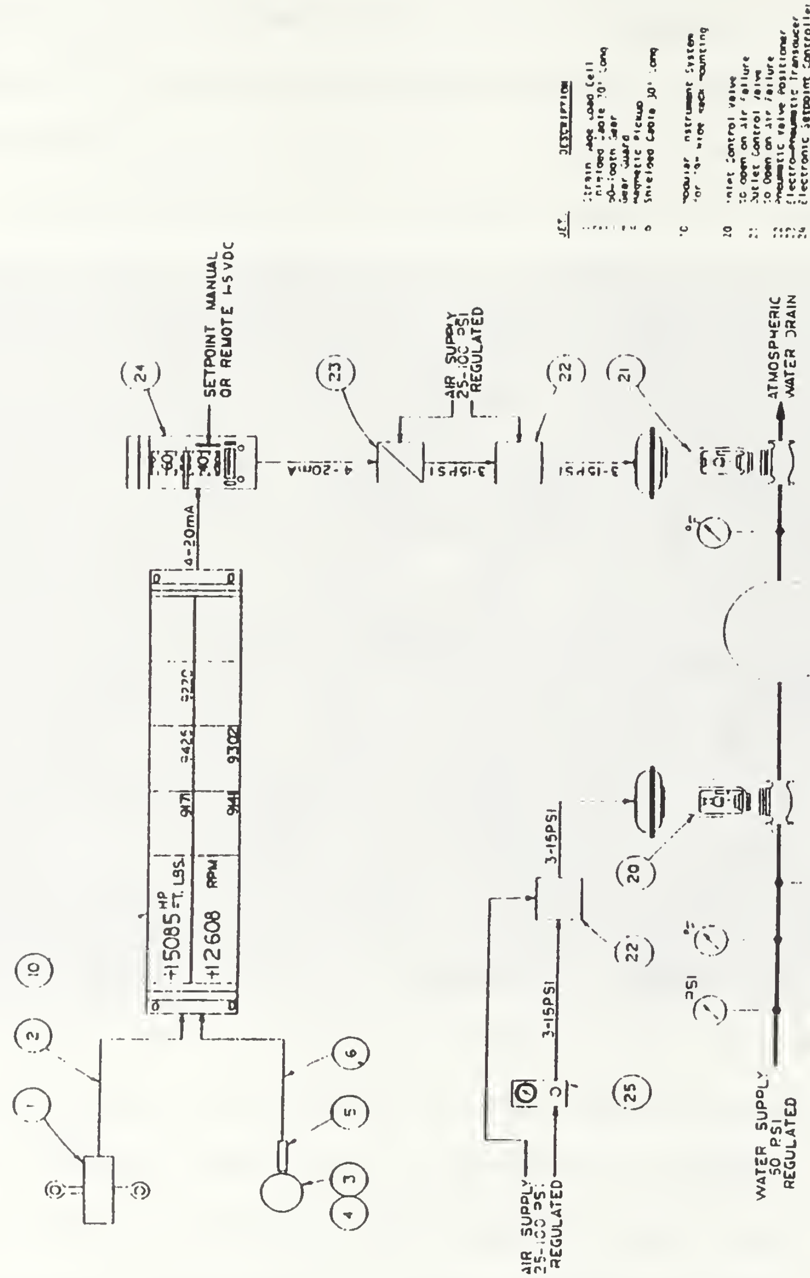


Figure 7. Schematic of the Automatic Dynamometer Load Control System

The present installation was changed from the installation described in Kane (Ref. 3) because changes had been made in the water cooling system for the Allis-Chalmers compressor. Water was supplied to the dynamometer, from the Allis-Chalmers cooling system, at a constant pressure of 50 psi. It entered the inlet control valve, flowed through the dynamometer housing, and exited through the outlet control valve and dumped into a fifty gallon holding tank at atmospheric pressure. The water was then pumped back into the pressurized, closed-loop system by a Gould nine-stage centrifugal pump located at the base of the holding tank.

The water level in the holding tank was set by the position of three electronic probes, which were inserted into the top of the holding tank, as shown in Figure 8. The probes were connected to a Warrick controller which controlled the on-off cycle of the Gould 9-stage pump. These probes were set so that the pump switched to the ON position when the water level reached the desired high point, and switched OFF when it reached the desired low point. The low point was chosen to avoid running the holding tank dry and damaging the pump. Initial testing of the system revealed that the cyclic operation of the pump had a adverse effect on the ability of the system to maintain a constant pressure. Continuous operation of the pump, by controlling the rate at which the water was returned to the closed-loop system, corrected the problem, and enabled the system to maintain a constant

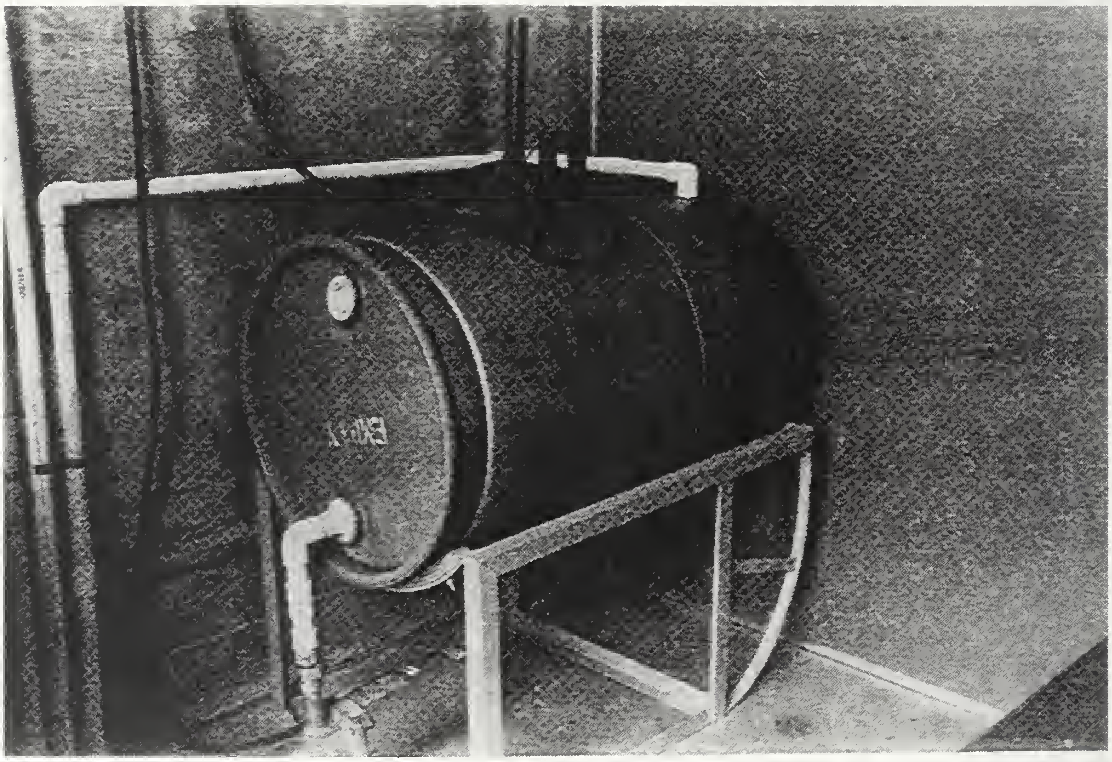


Figure 8. Fifty-Gallon Holding Tank and Pump

pressure. To control the rate of water return, a manually-controlled restriction valve was installed downstream of the Gould water pump. By maintaining a near constant level in the holding tank, the system pressure remained constant.

D. LUBRICATION SUBSYSTEM

Lubrication for the high precision bearings was provided by a Portable Turbine Lubrication Unit (PTLU). The unit is shown in Figure 9. Oil entered through an opening located atop and aft on the bearing sleeve, as shown in Figure 10. The oil then flowed through openings in the outer bearing spacer, passed through the bearings, and exited from two openings located on the bottom of the oil seals. The oil was collected and returned to the reservoir of the PTLU.

E. CONTROL AND INSTRUMENTATION

The speed control was unchanged from that described by Kane (Ref. 3). Control for the hydraulic dynamometer was provided by a Fischer-Porter electronic setpoint controller (Fig. 7). The controller received a conditioned feedback signal from the dynamometer, via a magnetic speed pickup, and compared it with the speed setpoint. When the feedback signal deviated from the setpoint, the controller produced a corrective signal which opened or closed the outlet control valve, until the deviation became zero, and the feedback signal coincided with the setpoint. The controller could be

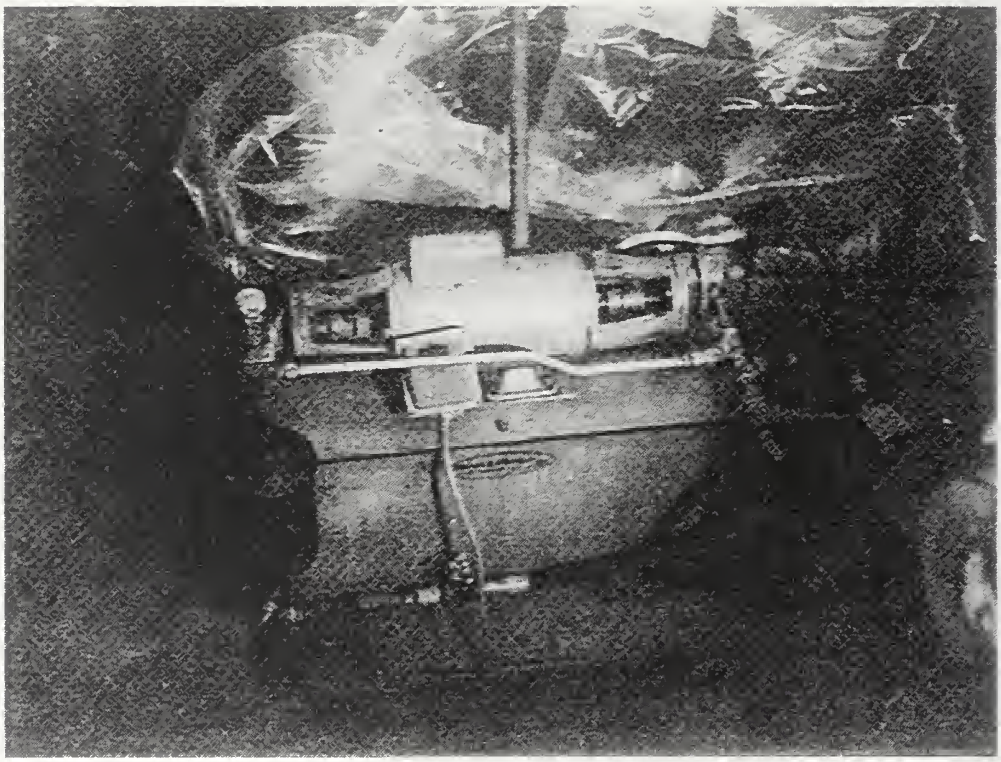


Figure 9. Portable Turbine Lubrication Unit

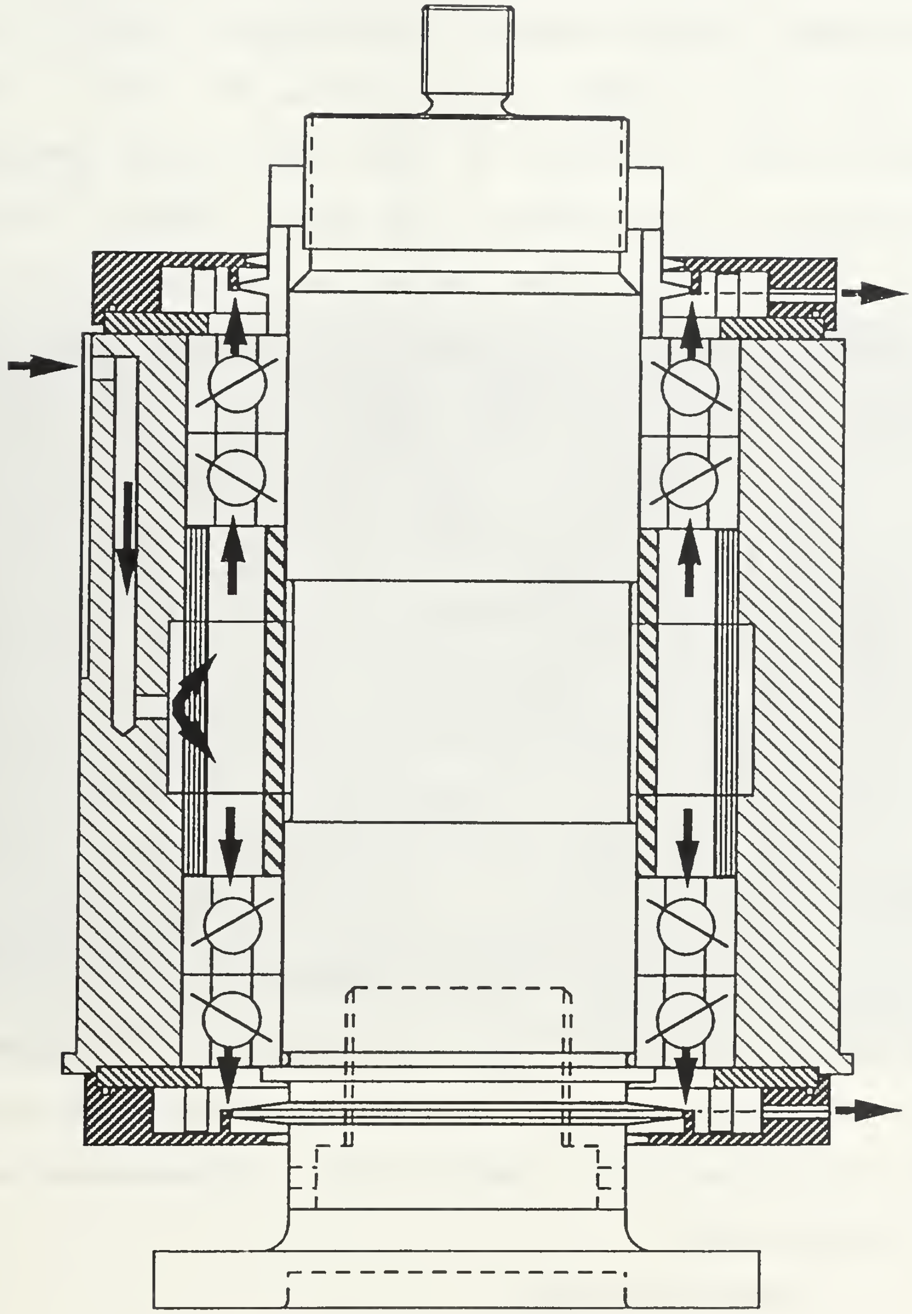


Figure 10. Bearing Lubrication Path

set to automatic or operated manually. The inlet-control valve was used to adjust the water flow rate for a particular power range, and was manually controlled. Figure 11 shows the TTR test cell looking forward from the rear. The inlet control valve is shown at the lower left position, the dynamometer in the center, the outlet control valve at the upper right. Both the inlet and outlet control valves incorporated pneumatic valve positioners attached to the valve bodies. Connections were made using 3/8" O.D. copper transmission lines. Supply pressure was set at 20 psi gauge. References 5 and 6 contain specific guidelines for setting control modes on the electronic controller.

The dynamometer was instrumented to provide output power measurements. Two thermocouples were mounted in the water lines, upstream and downstream of the dynamometer. Temperature readouts were given on the system control panel shown in Figure 12. The instrument in the upper left corner gave time histories of the dynamometer inlet and outlet temperatures. The instrument in the lower left corner allowed selection of the desired channel to monitor the dynamometer inlet or outlet water temperatures. Dynamometer parameters are listed below:

OPERATING RANGE

Max Capacity	250 HP
Max Speed	19,000 RPM
Max Torque	75 FT. LBS

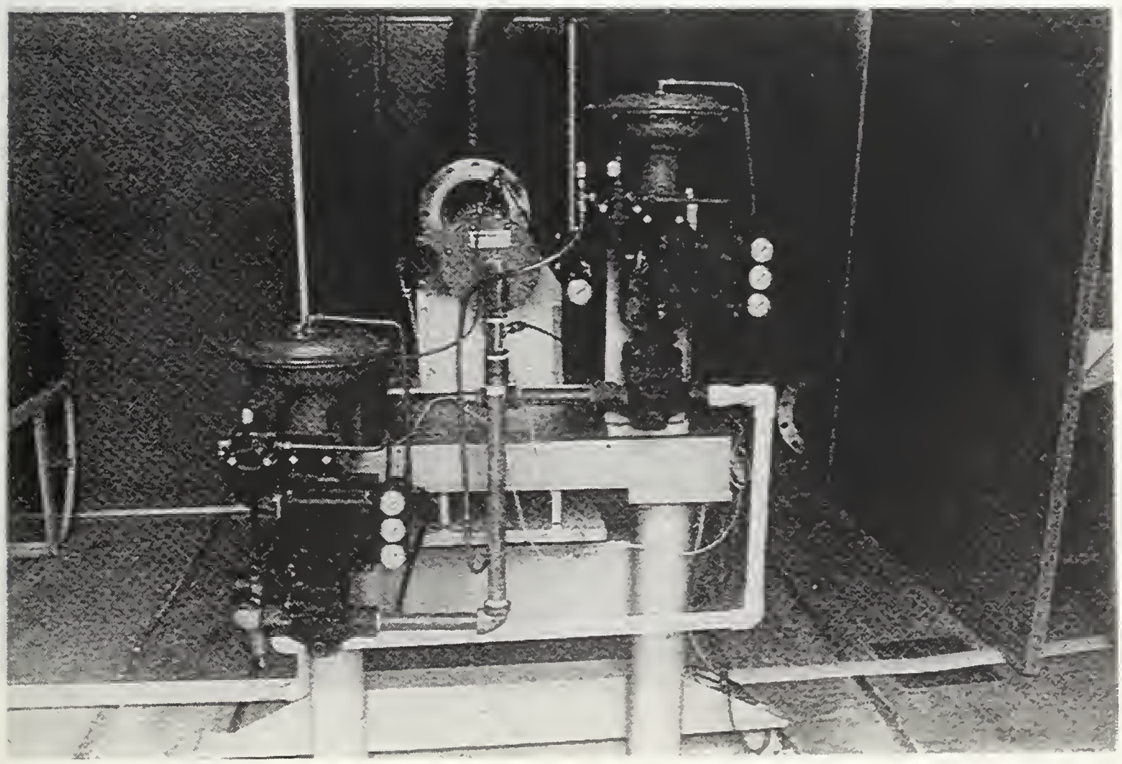


Figure 11. Inlet and Outlet Control Valves

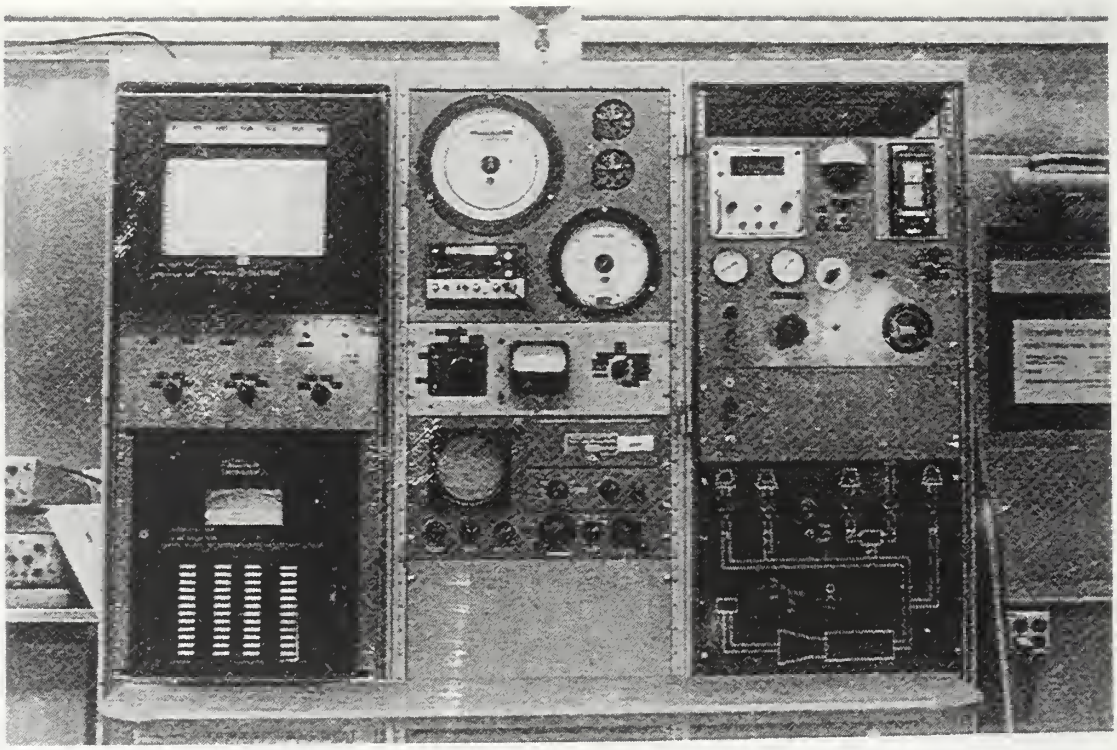


Figure 12. TTR Control Console

WATER SUPPLY REQUIREMENTS

Water Flow @ Delta T= 76°F	4 GPH/HP
Supply Pressure	50 psig
Max Inlet Temp	80° F
Max Outlet Temp	156° F

A water flowmeter was also installed upstream of the dynamometer inlet. Power measurements are normally determined from readouts taken from a Lebow load cell mounted on the dynamometer housing (which measure torque input) and RPM measurements taken from the magnetic speed pickup. The measurement of water flow rate and temperature rise was to provide a redundant check on the measurement of power.

III. PERFORMANCE ESTIMATION

The power output (P) of the turbine was estimated using the expression

$$P=\frac{m^2 R T_o 2\pi N r}{A_o p_o 60}(\tan a_1-\tan a_2) \qquad (1)$$

where p_o is the total pressure, T_o the total temperature, N the rotor speed, $\underline{m}$ the mass flow rate, r is the meanline radius, a_1 is the absolute flow-angle into rotor, a_2 is the absolute flow-angle out of rotor, and A_o is the annulus cross-sectional area.

Equation (1) was derived using the Euler turbine equation, with the assumption of constant density, as described in Reference 7. Specifying

p_o=279 kPa	T_o=288 K
N=7500 RPM	r=0.11684 m
$\underline{m}$=6.5 kg/sec	a_1=71°
A_o=0.02231 m2	a_2=-46.75°

the power for the single-stage axial turbine was estimated at 212 kW, which is equal to approximately 283 horsepower.

IV. CONCLUSIONS AND RECOMMENDATIONS

A. CONCLUSIONS

A test rig has been designed for the High Pressure Fuel Turbopump turbine of the Space Shuttle Main Engine. Inlet piping was designed, fabricated, and installed to connect the air supply to the turbine. Provisions were made for turbine inlet instrumentation.

A single stage of the two-stage Pratt & Whitney "Alternate Turbopump Development" model was assembled and installed on a steel table within the test cell. The shaft and bearing housing between the rotor and the hydraulic dynamometer were designed, manufactured, and installed. The dynamometer was refurbished and the speed control system was commissioned. The upgraded cooling water system for the laboratory's Allis-Chalmers (A/C) compressor required the addition of a pump to recirculate the coolant. The control system instrumentation was checked for integrity and functionality. An existing recirculating oil system was used for bearing lubrication. A performance estimate of the single-stage turbine was made based on the pressure and temperature supplied by the A/C compressor. The power output was within the operating envelope of the dynamometer. Problems encountered during balancing of the rotor and bearing housing assembly prevented initial test operation of the system.

B. RECOMMENDATIONS

The following are recommended:

1. Install instrumentation for bearing temperature and vibration measurements.

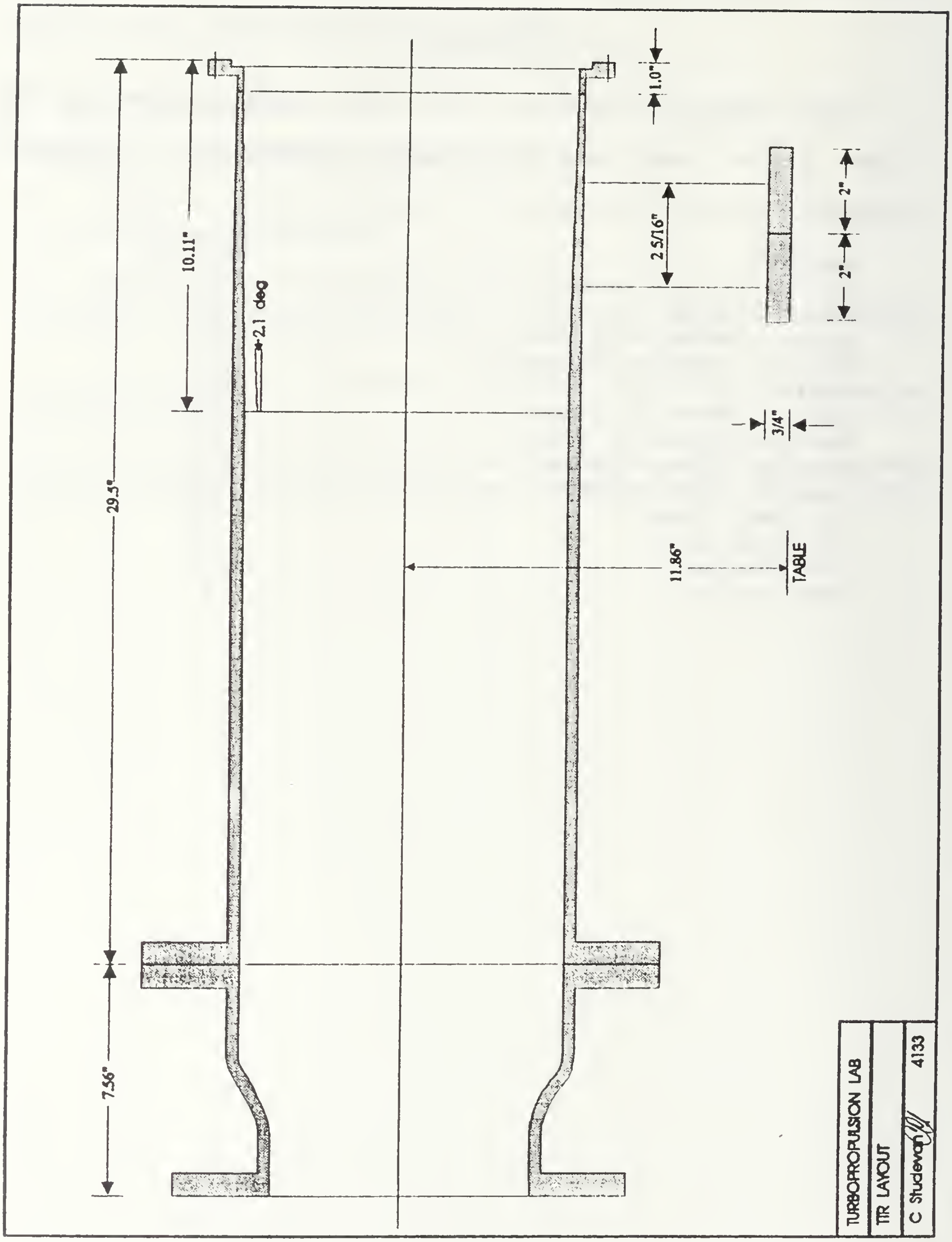
2. Incorporate a mass-flow measurement device in the turbine air supply so that air from the A/C can be bypassed as a means of setting turbine inlet pressure.

3. Include more extensive instrumnetation for performance measurements.

4. Design and manufacture a new casing for LDV measurements which includes optical access windows over the rotor blades.

APPENDIX I.

This appendix contains engineering diagrams for the TTR inlet piping and bearing housing components. Contents include: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TURBOPROPULSION LAB
TTR LAYOUT
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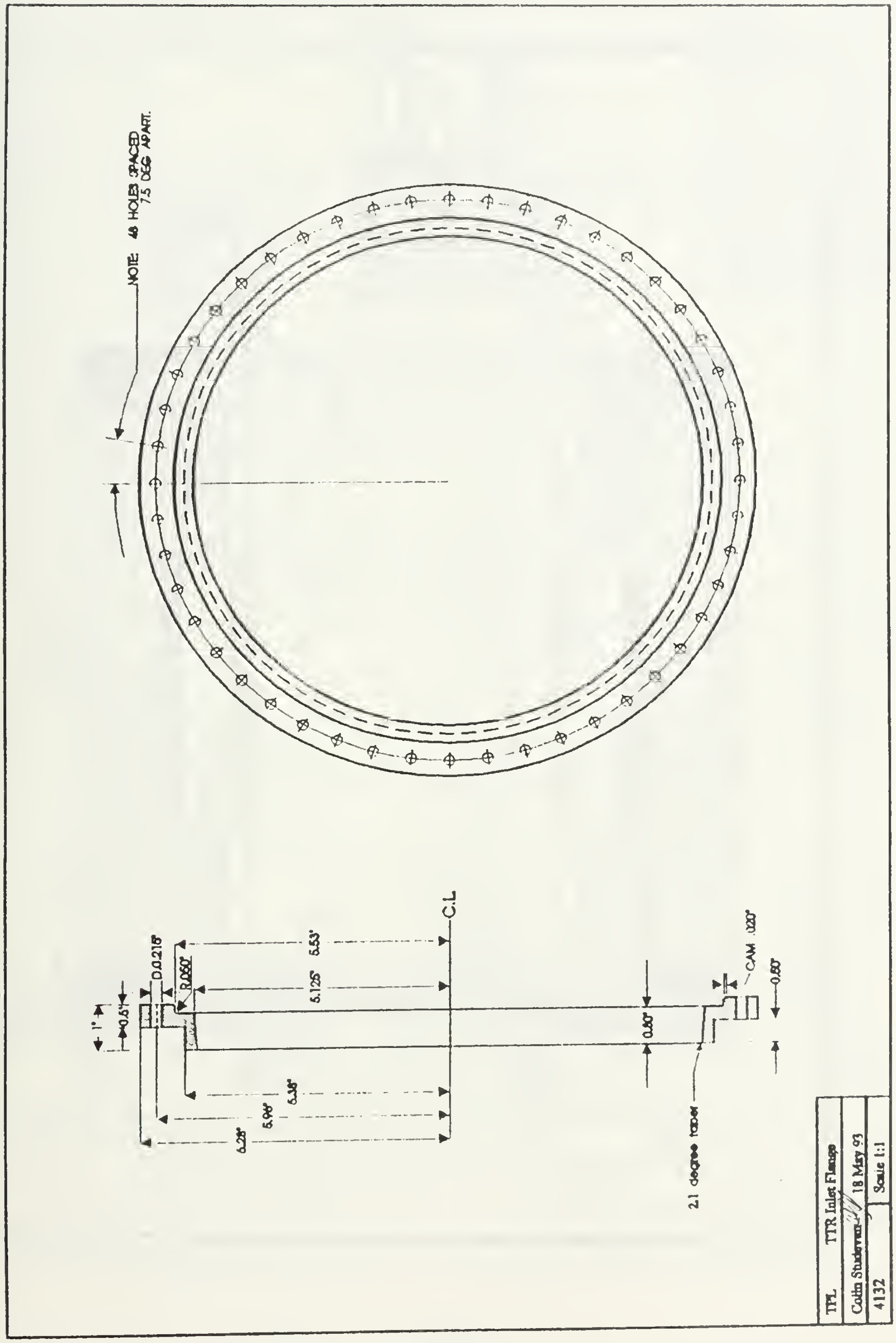
4133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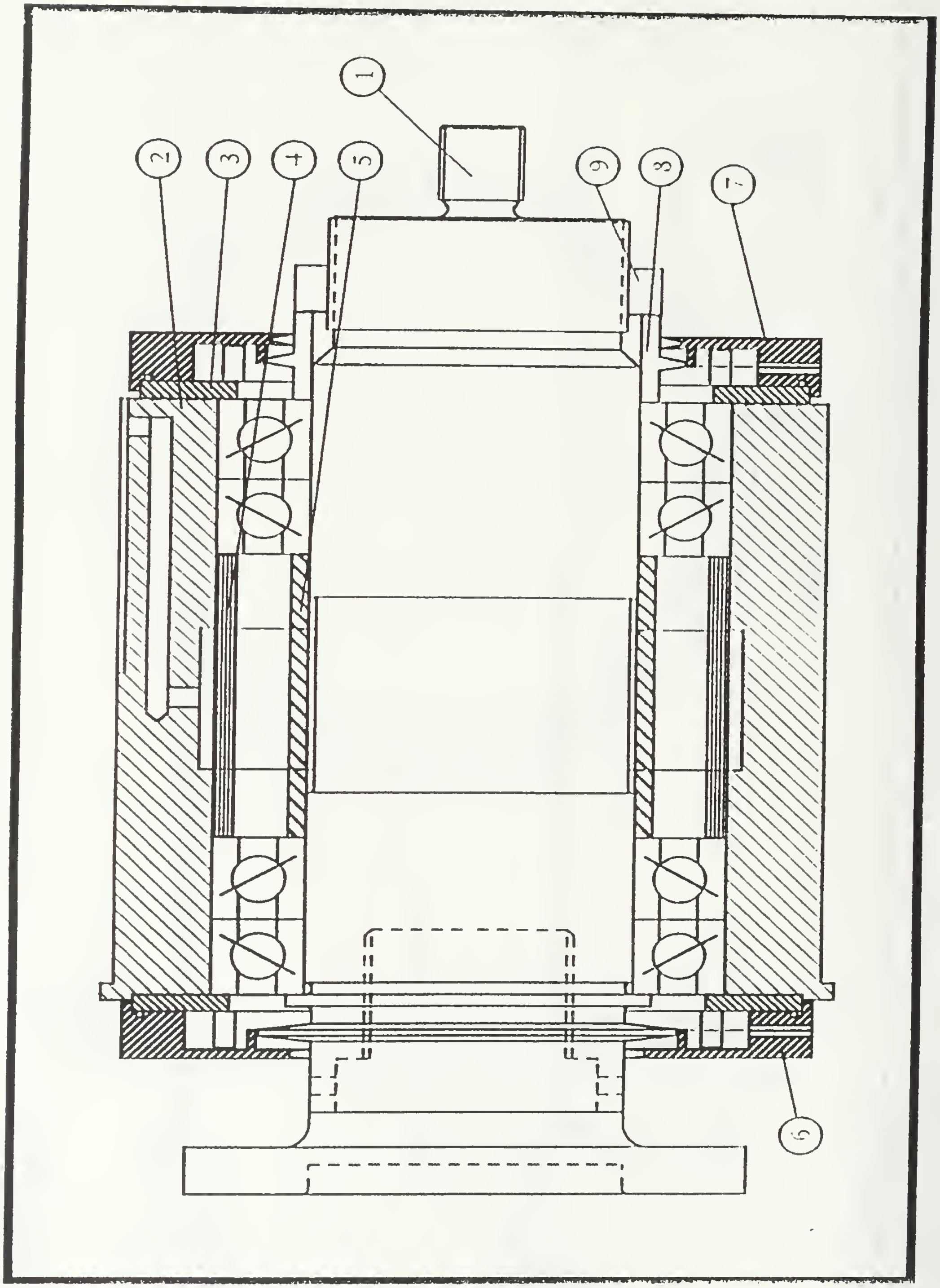
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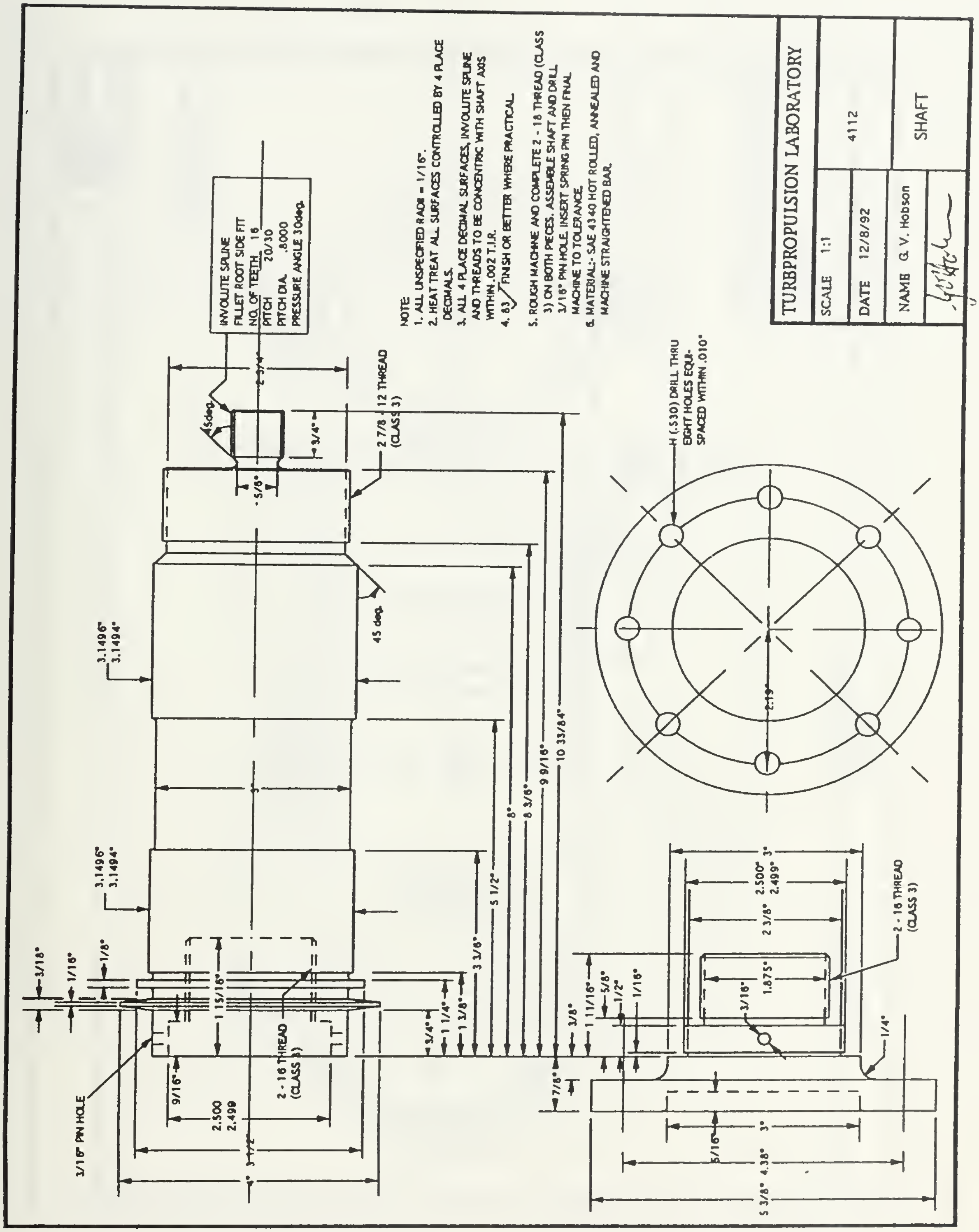
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SHAFT
NOTE:
1. ALL UNSPECIFIED RADII = 1/16".
2. HEAT TREAT ALL SURFACES CONTROLLED BY 4 PLACE DECIMALS.
3. ALL 4 PLACE DECIMAL SURFACES, INVOLUTE SPLINE AND THREADS TO BE CONCENTRIC WITH SHAFT AXIS WITHIN .002 T.I.R.
4. 83 FINISH OR BETTER WHERE PRACTICAL.
5. ROUGH MACHINE AND COMPLETE 2 - 18 THREAD (CLASS 3) ON BOTH PIECES. ASSEMBLE SHAFT AND DRILL 3/16" PIN HOLE. INSERT SPRING PIN THEN FINAL MACHINE TO TOLERANCE.
6. MATERIAL:- SAE 4340 HOT ROLLED, ANNEALED AND MACHINE STRAIGHTENED BAR.
INVOLUTE SPLINE
FILLET ROOT SIDE FIT
NO. OF TEETH 16
PITCH 20/30
PITCH DIA. .8000
PRESSURE ANGLE 30deg.
2 7/8 - 12 THREAD (CLASS 3)
H (.530) DRILL THRU EIGHT HOLES EQUI-SPACED WITHIN .010"
2 - 18 THREAD (CLASS 3)
3/16" PIN HOLE
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3.1494"
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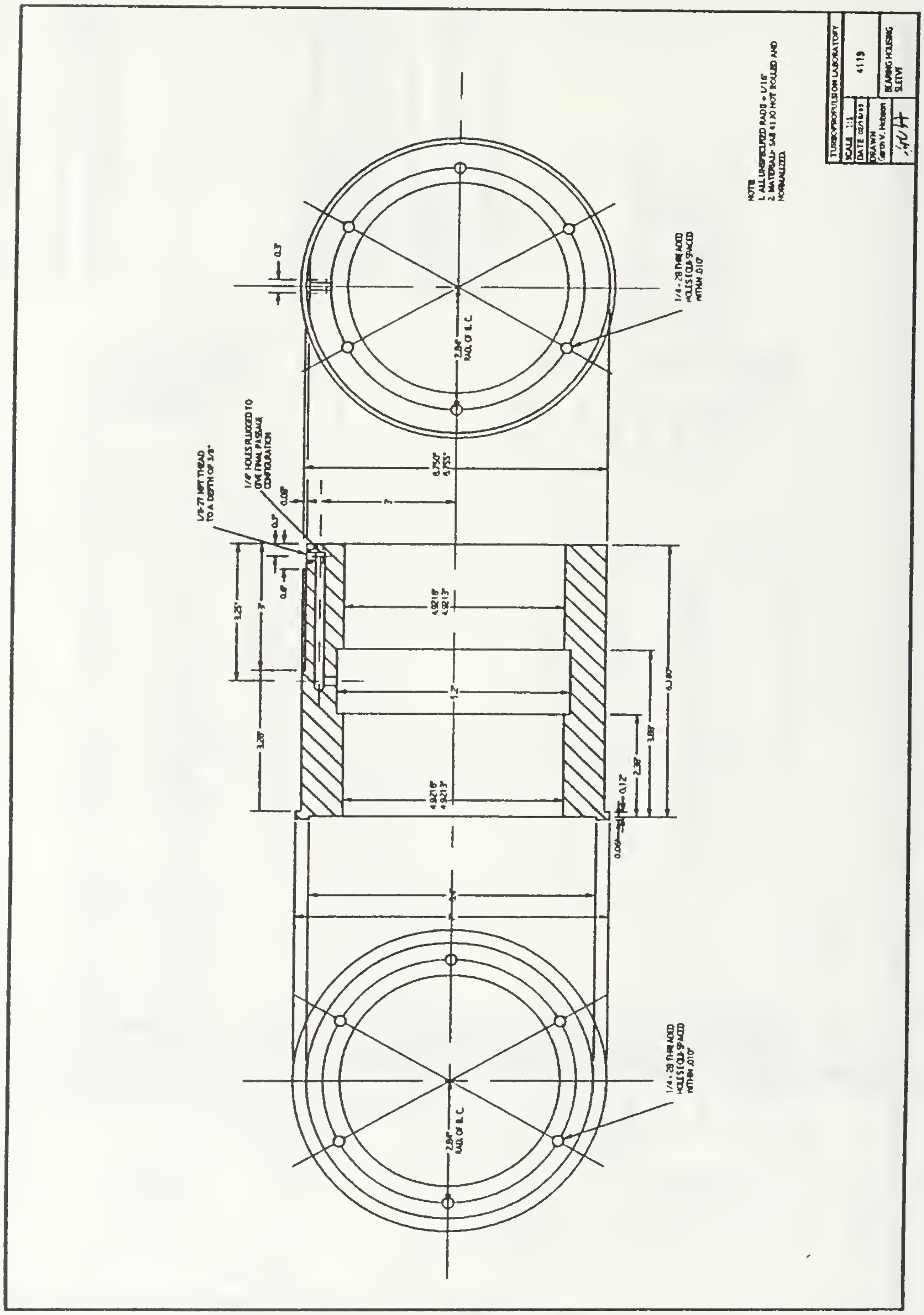

NOTE
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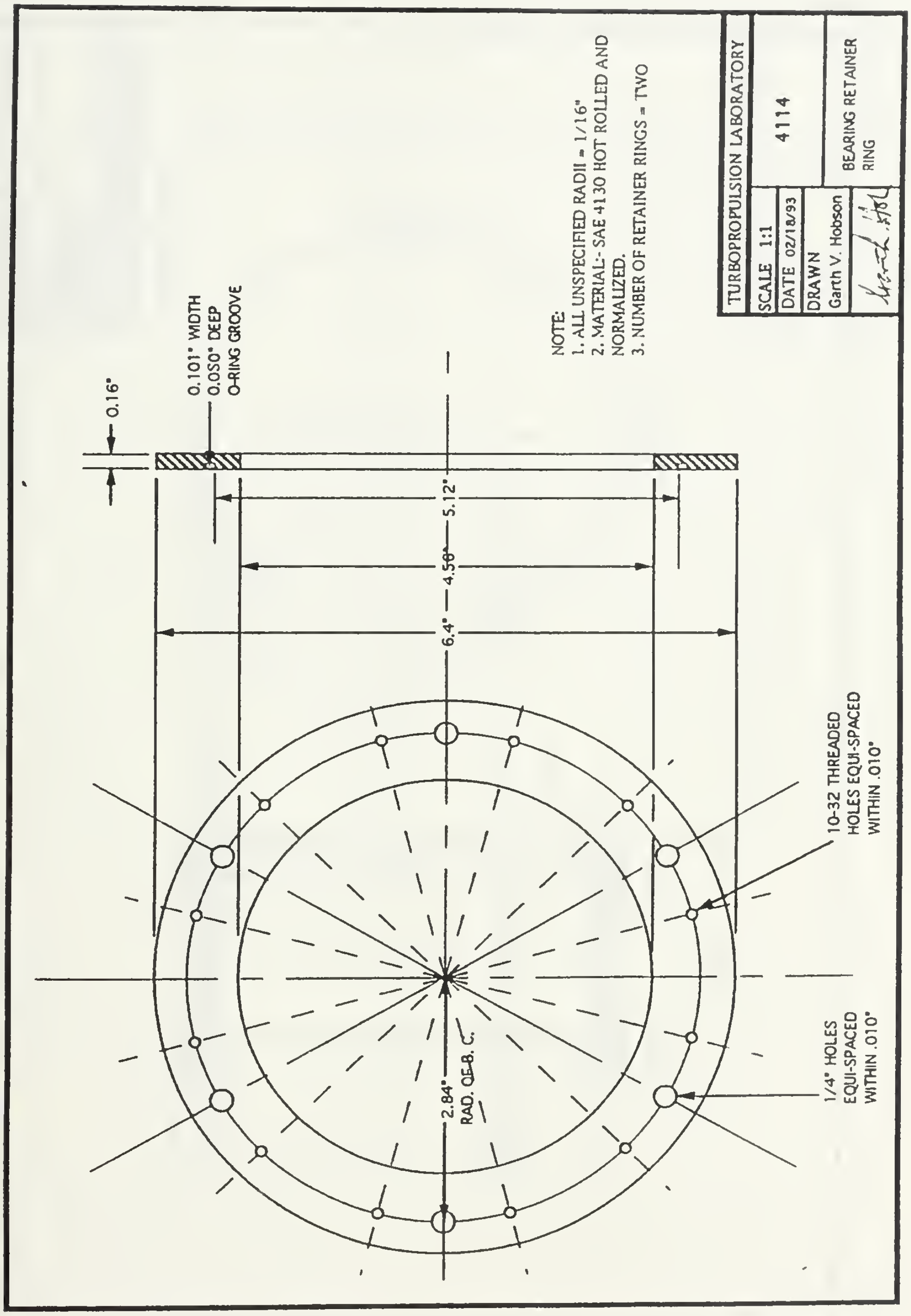
BEARING HOUSING SLEEVE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RAD. OF B. C.
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EQUI-SPACED
WITHIN .010"
NOTE:
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NORMALIZED.
3. NUMBER OF RETAINER RINGS = TWO
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Garth V. Hobson
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BEARING RETAINER
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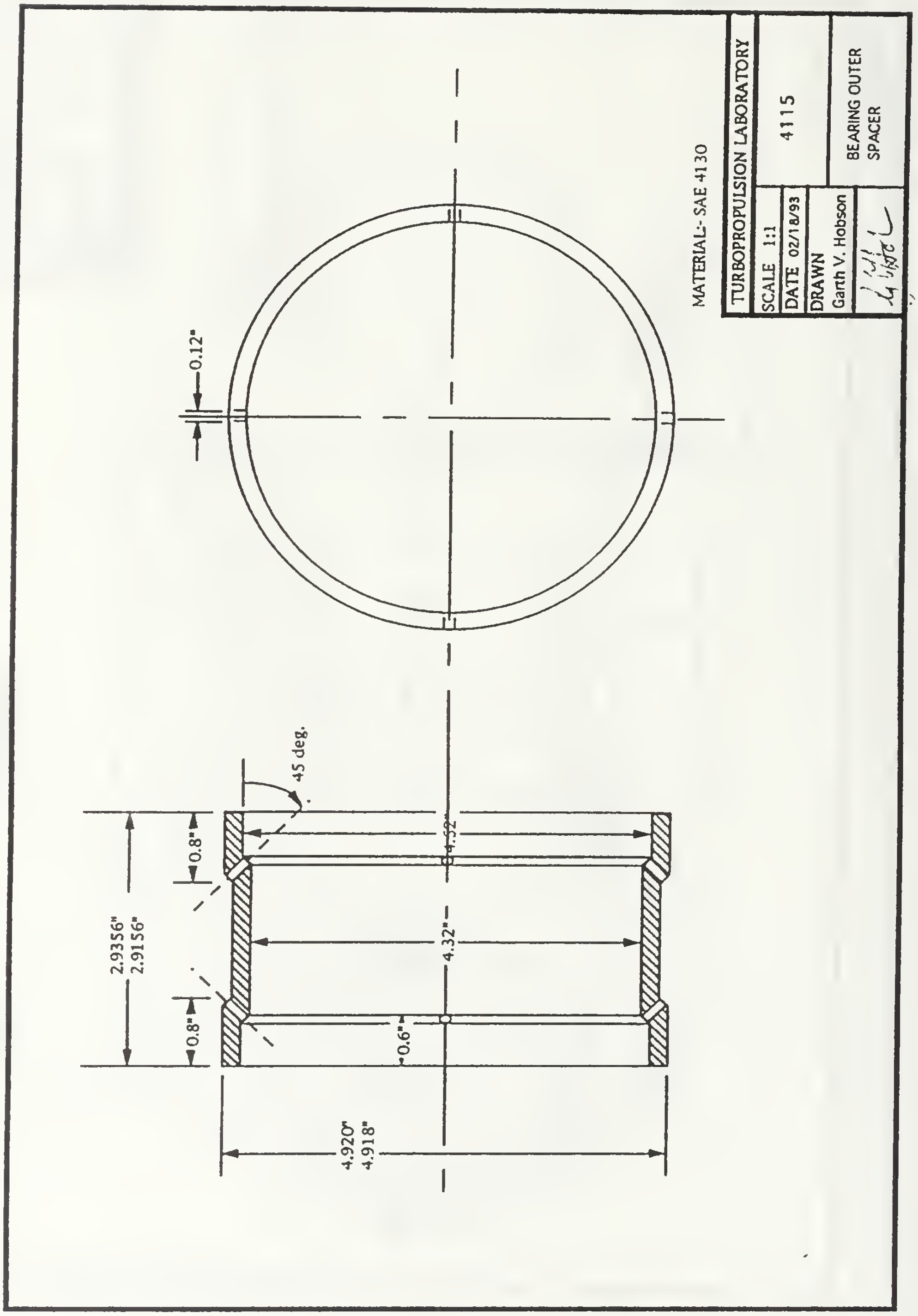
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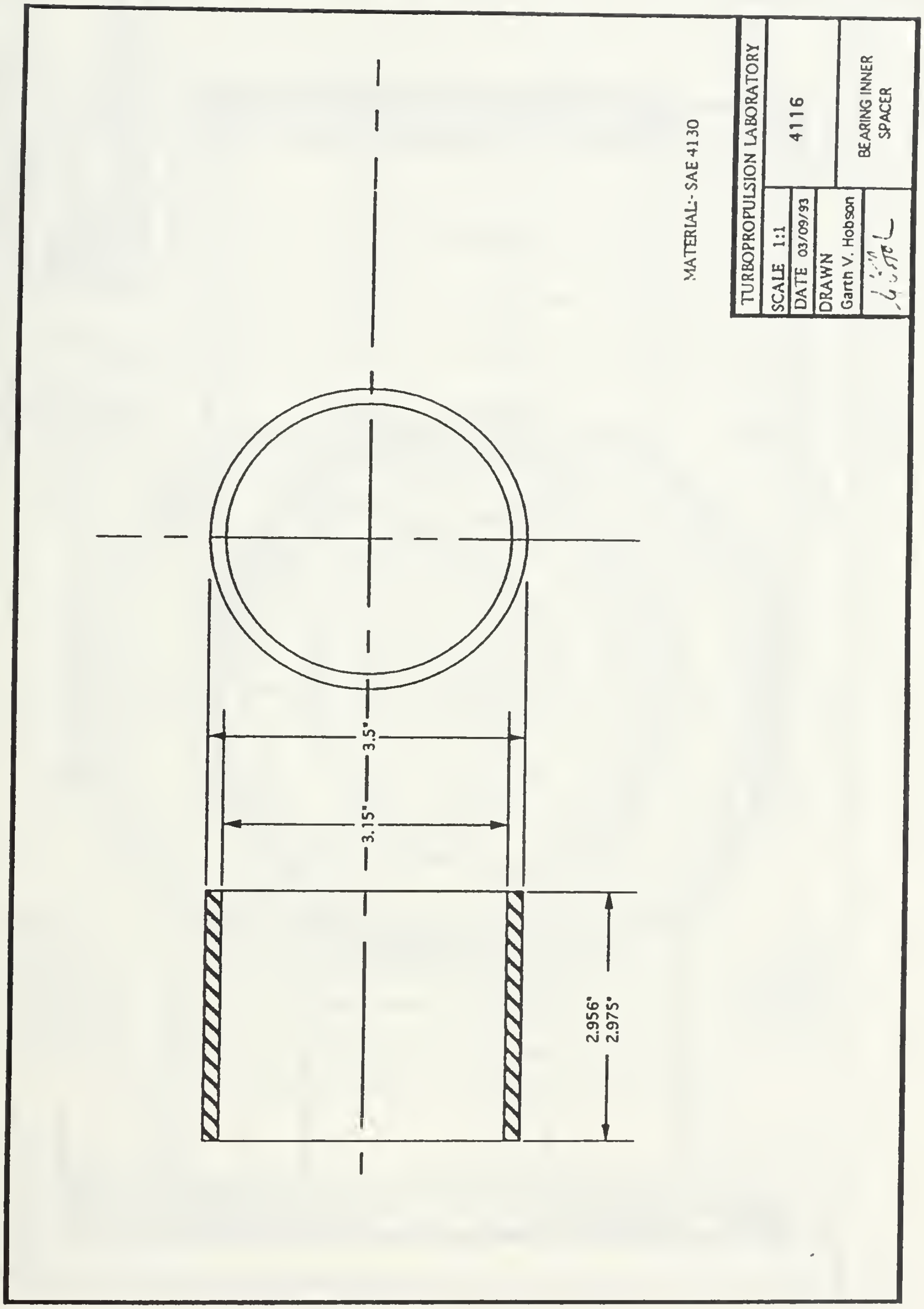
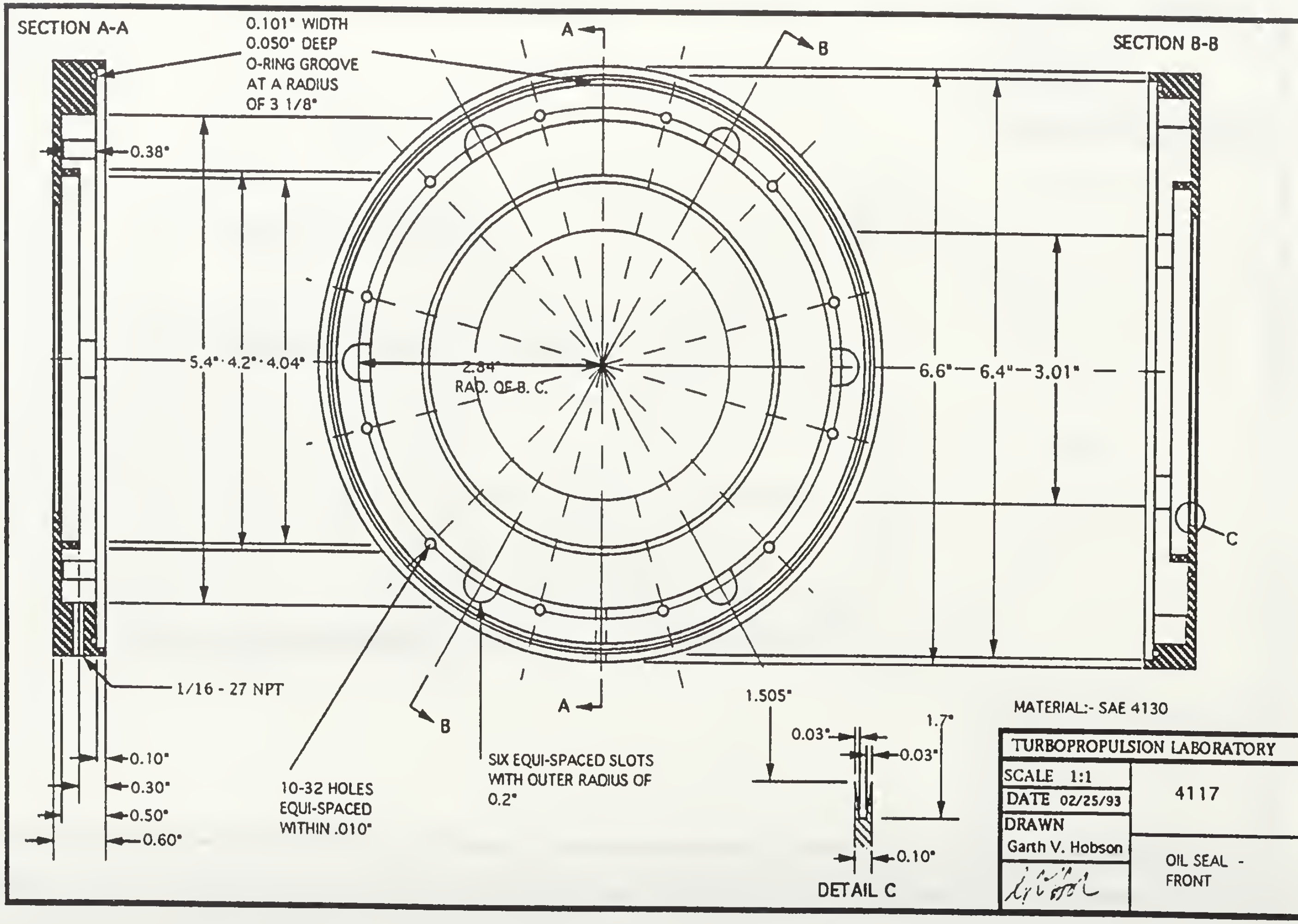
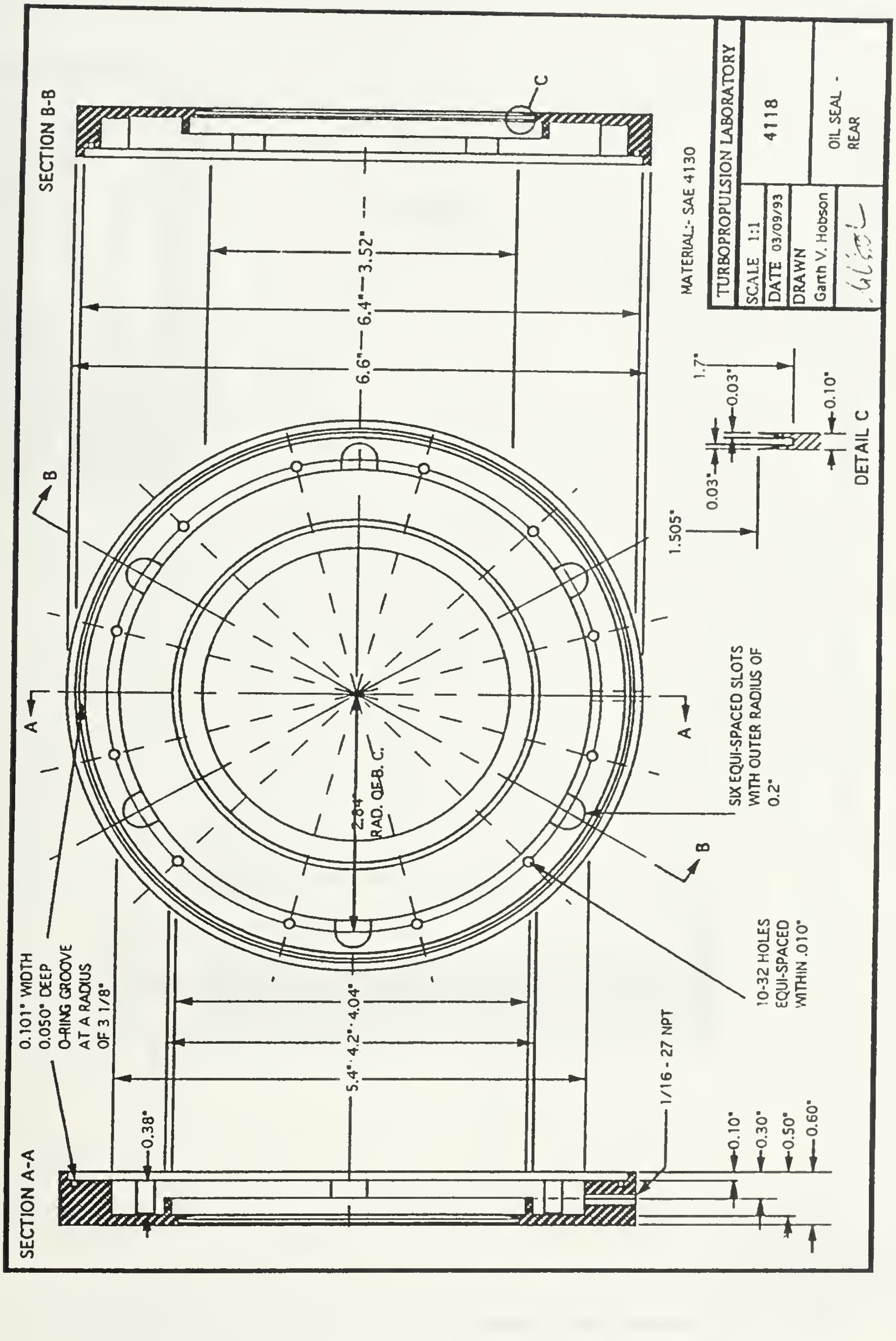
0.8"
4.52"
4.32"
0.6"
4.920"
4.918"

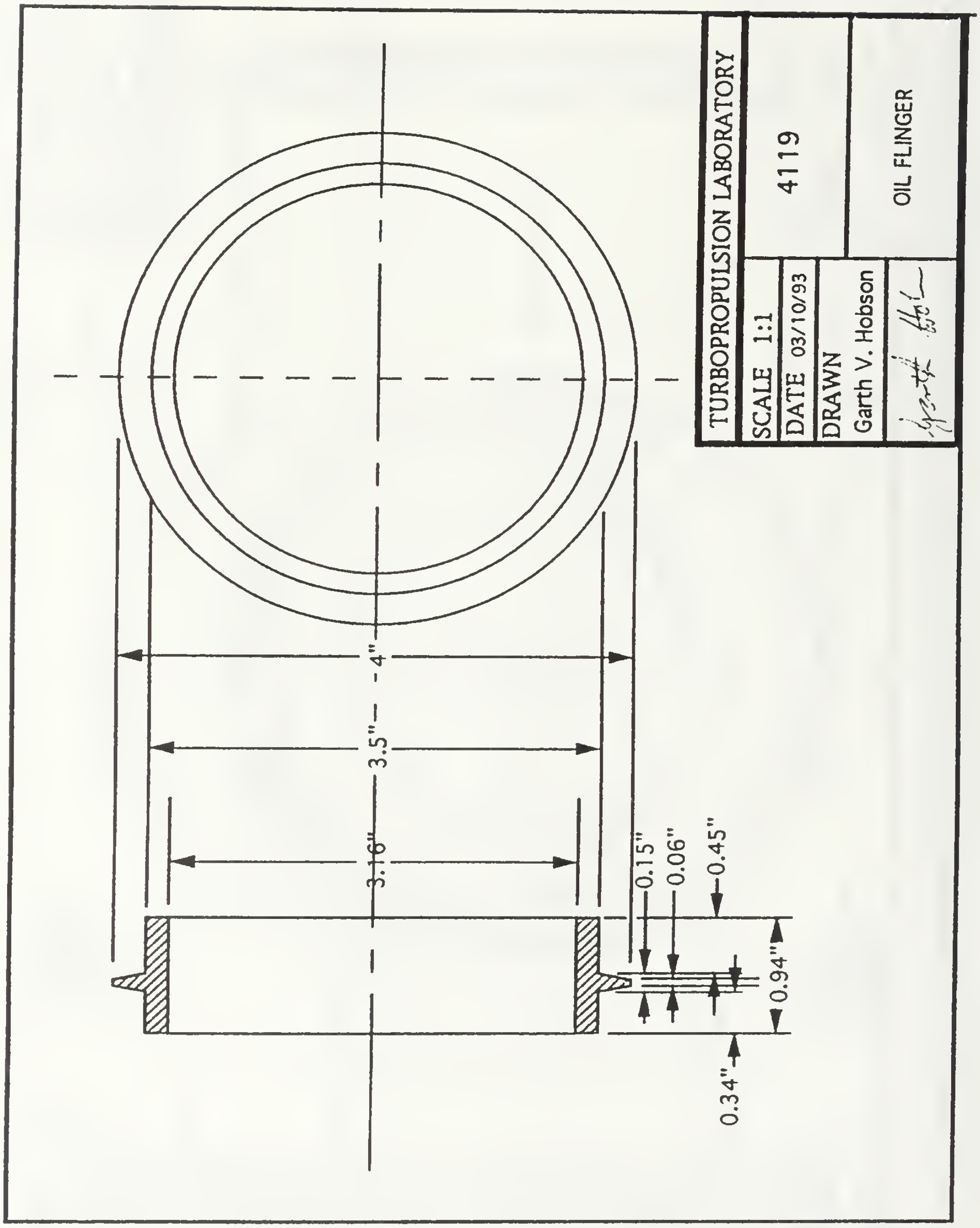

TURBOPROPULSION LABORATORY
SCALE 1:1
DATE 03/09/93
DRAWN
Garth V. Hobson
4116
BEARING INNER SPACER
MATERIAL:- SAE 4130
3.5"
3.15"
2.956"
2.975"

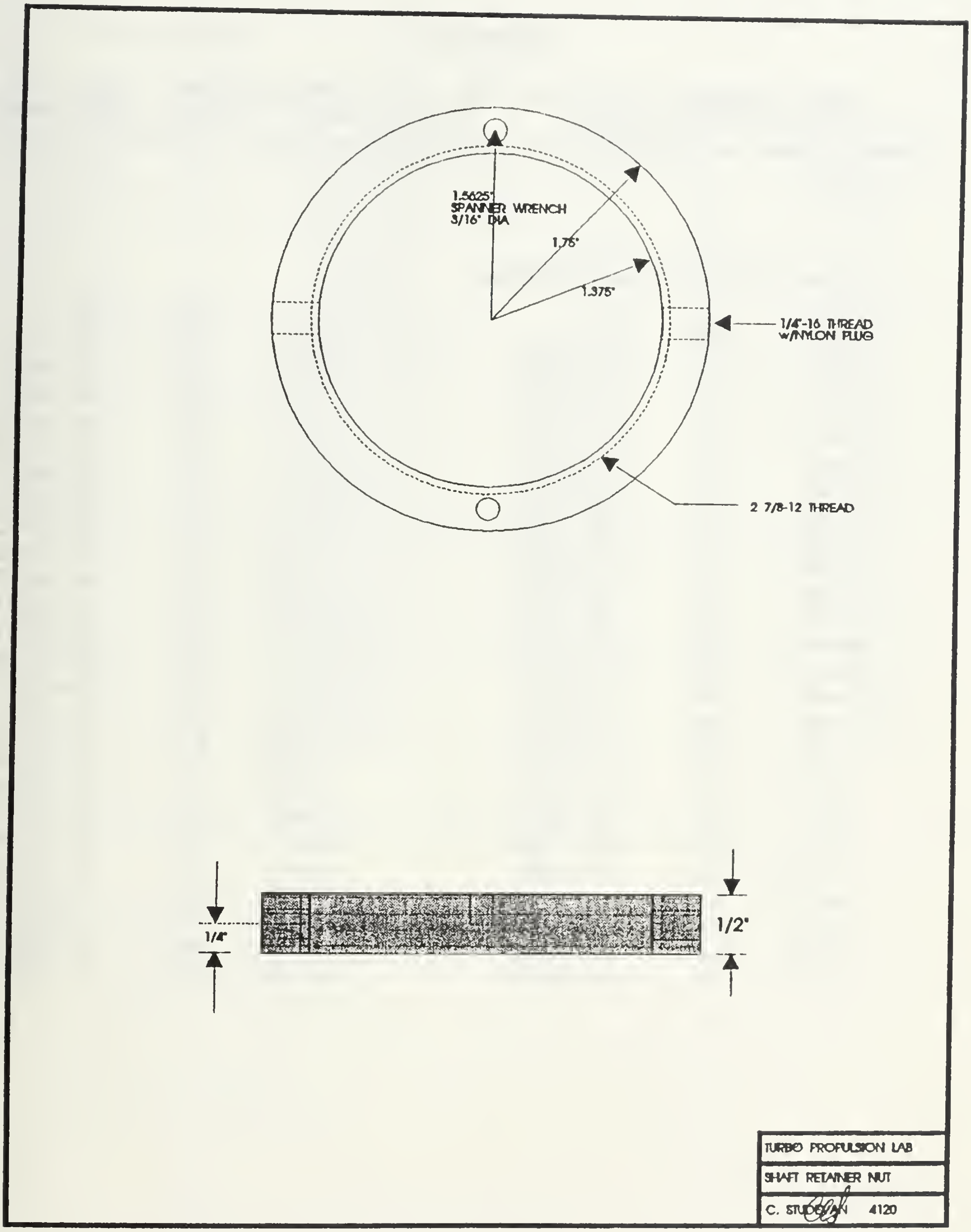


SECTION A-A
0.101" WIDTH
0.050" DEEP
O-RING GROOVE
AT A RADIUS
OF 3 1/8"
0.38"
5.4"· 4.2"· 4.04"
2.84"
RAD. OF B. C.
A
B
SECTION B-B
6.6" — 6.4" — 3.01"
C
1/16 - 27 NPT
0.10"
0.30"
0.50"
0.60"
10-32 HOLES
EQUI-SPACED
WITHIN .010"
SIX EQUI-SPACED SLOTS
WITH OUTER RADIUS OF
0.2"
1.505"
0.03"
0.03"
1.7"
0.10"
DETAIL C
MATERIAL:- SAE 4130
TURBOPROPULSION LABORATORY
SCALE 1:1
DATE 02/25/93
DRAWN
Garth V. Hobson
4117
OIL SEAL -
FRONT



SECTION B-B
C
6.6" — 6.4" — 3.52"
MATERIAL:- SAE 4130
TURBOPROPULSION LABORATORY
SCALE 1:1
DATE 03/09/93
DRAWN
Garth V. Hobson
4118
OIL SEAL - REAR
1.7"
0.03"
0.03"
0.10"
DETAIL C
1.505"
B
A
A
SIX EQUI-SPACED SLOTS WITH OUTER RADIUS OF 0.2"
2.84" RAD. OF B. C.
B
10-32 HOLES EQUI-SPACED WITHIN .010"
0.101" WIDTH 0.050" DEEP O-RING GROOVE AT A RADIUS OF 3 1/8"
5.4" 4.2" 4.04"
1/16 - 27 NPT
0.38"
0.10"
0.30"
0.50"
0.60"
SECTION A-A


TURBOPROPULSION LABORATORY
4119
OIL FLINGER
SCALE 1:1
DATE 03/10/93
DRAWN
Garth V. Hobson
4"
3.5"
3.16"
0.15"
0.06"
0.45"
0.94"
0.34"


1.5625"
SPANNER WRENCH
3/16" DIA
1.75"
1.375"
1/4"-16 THREAD
w/NYLON PLUG
2 7/8-12 THREAD
1/4"
1/2"
TURBO PROPULSION LAB
SHAFT RETAINER NUT
4120

APPENDIX II.

This appendix contains the first-stage rotor blades and pan weight positions. Items are listed by their identification number (I.D.), position (POS), and weight (WT) in grams.

First Stage Rotor Blades:

I.D.	POS	WT	I.D.	POS	WT
A-79	1	34.9489	I-86	26	34.9148
I-92	2	34.8078	L-79	27	34.7939
G-92	3	34.7565	J-76	28	34.7215
K-62	4	34.5687	C-55	29	34.5642
J-91	5	34.5337	F-61	30	34.5215
L-63	6	34.5205	B-62	31	34.4945
D-95	7	34.4888	H-57	32	34.4881
K-47	8	34.4589	B-55	33	34.4571
K-60	9	34.4519	C-98	34	34.4495
L-85	10	34.4320	G-106	35	34.4267
A-62	11	34.4137	O-80	36	34.4134
M-60	12	34.4106	G-47	37	34.4060
L-60	13	34.3963	L-77	38	34.3590
K-64	14	34.3468	O-60	39	34.3455
G-55	15	34.3377	H-91	40	34.3071
E-58	16	34.3225	K-104	41	34.3103
O-73	17	34.2797	L-55	42	34.2578
E-73	18	34.2196	K-100	43	34.1478
B-61	19	34.1111	A-81	44	34.0956
O-85	20	34.0786	A-41	45	34.0728
O-100	21	34.0527	L-61	46	34.0486
J-55	22	34.0145	O-75	47	34.0092
P-55	23	33.8166	M-81	48	33.8111
I-70	24	33.7000	K-32	49	33.6408
A-65	25	33.5672	A-106	50	33.5588

Pan Weights:

I.D.	POS	WT	I.D.	POS	WT
26	1	14.6299	1	26	14.6199
3	2	14.6079	28	27	14.6045
30	3	14.6045	5	28	14.5835
34	4	14.5833	36	29	14.5812
9	5	14.5779	32	30	14.5770
7	6	14.5715	38	31	14.5711
42	7	14.5693	13	32	14.5691
50	8	14.5638	15	33	14.5620
40	9	14.5611	24	34	14.5577
20	10	14.5548	45	35	14.5473
21	11	14.5463	48	36	14.5460
22	12	14.5455	44	37	14.5452
11	13	14.5451	49	38	14.5387
47	14	14.5375	25	39	14.5343
17	15	14.5326	23	40	14.5325
41	16	14.5260	16	41	14.5256
19	17	14.5217	18	42	14.5196
43	18	14.5184	46	43	14.5142
39	19	14.5087	37	44	14.4993
14	20	14.4990	31	45	14.4834
8	21	14.4832	10	46	14.4736
12	22	14.4693	33	47	14.4620
35	23	14.4616	4	48	14.4559
6	24	14.4517	27	49	14.4165
29	25	14.4137	2	50	14.4139

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