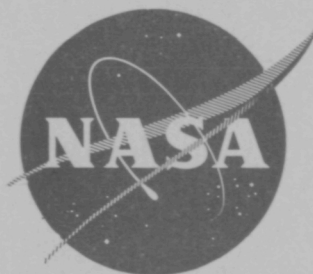


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GESP-101




**POWER TRAIN FOR THREE STAGE POTASSIUM
TEST TURBINE**

FINAL DESIGN – VOLUME 1

prepared for

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

NASA Lewis Research Center
Contract NAS 3-10606
Joseph P. Joyce, Project Manager

NUCLEAR SYSTEMS PROGRAMS
SPACE SYSTEMS
GENERAL  ELECTRIC
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FINAL DESIGN

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I. INTRODUCTION

This report represents the final design of the auxiliary drive and power absorbing system (herein referred to as the "power train") for the three-stage potassium vapor test turbine which is being designed and fabricated under NASA Contract NAS3-8520 (Three-Stage Potassium Vapor Turbine, Design and Fabrication), and which is described in detail in NASA reports NASA-CR 72249 and NASA-CR 72250 (References 1 and 2). The power train described herein is being developed as part of Task I of separate Contract NAS3-10606 (Three-Stage Potassium Vapor Turbine Test), which is for the necessary test facility modifications and performance and endurance testing of the three-stage potassium vapor turbine. Work on the latter contract was initiated on April 10, 1967, and is for a 28-month duration. The design and fabrication of the power train milestone is scheduled for completion during December 1967, to precede the assembly checkout of the three-stage turbine mentioned above. This report comprises a presentation of the mechanical design effort corresponding to the requirements as defined under Exhibit "A" of Contract NAS3-10606. These requirements are listed in detail in Section II of this report entitled "DESIGN SPECIFICATIONS".

The power train is designed for a 5000 hour life at a rotational speed of 18250 rpm (as defined under the "DESIGN SPECIFICATIONS" of Section II), and a nominal power absorbing capacity of 328 horsepower. These conditions comprise what is termed throughout this report as "design-point" operations.

The power train is similar in overall geometry to one used on the two-stage potassium test turbine which was successfully designed and developed under NASA Contract NAS5-1143 (see Item 5 in Section II). The two-stage turbine testing was completed in September 1966, and that program is described fully by References 3, 4, 5, 6 and 8. With some modifications, the same test facility will be used for testing the three-stage turbine.

Although similar in overall geometry, the power train presented here differs in detail considerably from the two-stage turbine's power train, as necessitated primarily by the present requirement for 5000 hours continuous operation. The two-stage turbine power train never anticipated more than 1000 hours continuous operation. In addition, the present test rig reflects

several modifications made for greater reliability as a result of experience gained in the two-stage turbine test. The turbine power train requires several new or significantly modified components such as new couplings, mounting structure, bearings, and improvement in the water brake and steam turbine mechanical reliability. Table I shows the significant parameters of design and operation of the three-stage turbine power trains.

The testing program identified for this test rig anticipates a pre-test mechanical checkout with speeds of from 0 to 21000 rpm, followed by a potassium vapor performance test of approximately 100 hours, with varying rotational speeds of 15000 to 20000 rpm and a power absorption of up to 342 horsepower. This will be followed by a 5000-hour endurance test at the design point conditions defined above.

This report presents the design conditions, major assumptions, and pertinent analyses involved in the power train design, along with descriptive graphs, figures, and drawings of actual test rig components and major assemblies of test equipment.

II. DESIGN SPECIFICATIONS

Following are the design specifications for the Three-Stage Turbine Power Train, as defined under Task I of Exhibit "A" of Contract NAS3-10606.

A. DESIGN SPECIFICATIONS

- 1.
 - 2.
 - 3.
 - 4.
- } These items not applicable to power train
5. The turbine start-up and power absorption and subloop hardware developed under NASA Contract NAS5-1143 shall be inspected, and the required maintenance and modifications performed. Also, all hardware shall be checked for operation, and to assure a reliable life of 5,000 hours. Final checkout shall be performed with the three-stage turbine and its new coupling installed in the test loop.
 6. Increase power output of the steam turbine start-up drive as required to assure measurement of parasitic torque values of the three-stage turbine up to 20,000 rpm.
 7. Compare the reliability of grease packed flexible couplings and non-lubricated, flexible couplings versus the required minimum life of 5,000 hours. If required to obtain the reliability and balance, design and fabricate a special flexible coupling to connect the test turbine and water brake.
 8. Update the design of the water brake power absorber and steam turbine drive bearings to assure reliable life of 5,000 hours. Apply design principles in NASA TN D-2664, incorporated herein by reference. One set plus one spare set of water brake and steam turbine bearings and seals shall be purchased to support test operation.
 9. Dynamically balance each rotor and coupling to a level equal to or less than the value of unbalance in gram-inches given by the following equation in which W is the rotor weight in pounds and N is rotational speed in revolutions per minute.

$$U \text{ (gm-in.)} = \frac{113.6 W(\text{lb})}{N(\text{rpm})}$$

10. Provide damping to the water brake housing which is cantilevered from the strain gage torque meter to reduce the resonance vibration amplitude (reference paragraph 11 below) at the front flange near the operating speed. This damping shall be applied such that torque measurement is not influenced.

11. Mount the water brake, steam turbine and required coupling in the test facility. Use the steam turbine to drive the water brake from 0 to 21,000 rpm. Measure the water brake and steam turbine horizontal and vertical vibration levels both on acceleration and deceleration. All vibration amplitude levels in the turbine operating speed range (15,000 to 20,000 rpm) shall be 0.25 mil or less and a maximum of 0.5 mil from 0 to 21,000 rpm.
12. Install the three-stage turbine with its coupling into the test facility. Use the steam turbine to drive the water brake and potassium turbine from 0 to 21,000 rpm. Measure the test turbine, water brake, and steam turbine horizontal and vertical vibration amplitudes both on an acceleration and deceleration. All vibration amplitude levels in the 15,000 to 20,000 rpm range shall be 0.25 mil or less and a maximum of 0.5 mil from 0 to 21,000 rpm. The casing of the turbine shall be heated to a minimum of 700°F prior to and during this test.
13. Vibration test in paragraphs 11 and 12 shall include inspection, checkout and operation of the following subloops:
 - a) High pressure oil lubricant.
 - b) Low pressure oil lubricant.
 - c) Steam supply.
 - d) Water supply and drainage.
 - e) Argon supply and reclamation.
 - f) Liquid potassium slinger seal loop.

All modifications proposed for the power absorption and start-up drive and subloop hardware shall be submitted to the NASA Project Manager for his review and approval.

14. From subsequent correspondence.

"A caliper type disk brake shall be designed, procured and mounted on the initial rotor assembly. The most up-to-date materials shall be used and a cooling air supply provided at the areas where the pucks contact the disk. At least two disks shall be fabricated. Each shall be balanced with the rotor to achieve part interchangeability."

III. INTEGRATED POWER TRAIN DESIGN

General

The cross-sectional drawing of the three-stage potassium test turbine and its power train assembly is shown in Figures 1 and 3. The power train and support structure system comprises all equipment aft of the flange labeled "Point A", and the "environmental chamber" enclosing and supporting the three-stage potassium test turbine. It consists of a water brake and steam turbine connected in tandem to each other and to the three-stage test turbine by flexible couplings. No clutching or gearing is employed, and therefore, all components rotate continuously and at the same speed during testing. Both the water brake and steam turbine are cantilevered off the aft flanges of two "Bytrex" torquemeters whose forward flanges are attached to conical and cylindrical support structures, and which sense the torque delivered to or from the water brake and steam turbine respectively, by means of internal strain-gaged flexure plates. Both the water brake and steam turbine are counter-weighted to eliminate the overhang moment of the torquemeters. The design of each of the individual power train system components is discussed in detail in Section IV. The water brake absorbs the output power of the potassium test turbine and simulates an electric alternator which ultimately will be incorporated in a prototype space power conversion system. The steam turbine (aft of the water brake) provides rotation of the test rig during startup and shutdown, mechanical checkout, and during system tare-testing. The steam turbine is a single stage re-entry ("Terry" type) turbine capable of motoring the entire potassium turbine - power train system to speeds exceeding the required 21,000 rpm (see DESIGN SPECIFICATIONS). The fluids to both the water brake and steam turbine are supplied through flexible hoses to minimize the residual torque produced by the flow lines coming to or from the system. (The remaining small residual torque will be calibrated out before testing).

As a safety device in case of coupling or water brake failure, which might be created by loss of water, an automatically-actuated caliper disk brake is mounted on the aft end of the potassium turbine shaft (see Figure 1), and is considered part of the power train.

The torque from the potassium turbine is transmitted to the water brake through a flexible diaphragm coupling (manufactured by Bendix Corporation) typical of the non-lubricated coupling which will be required by a turbo-alternator system in space. A quill shaft comprises the aft coupling connecting the water brake with the steam turbine. In addition, a flexible gear-toothed coupling has been designed and manufactured with special provisions for retaining lubricating oil and for injection of lubricant (actuated from the control room) periodically during operation.

The power train is supported in a structure consisting of a number of conical and cylindrical members, assembled to maintain accurate axial and radial alignment of the rotating components. The power train structure attaches to, and cantilevers from, an environment chamber surrounding the potassium test turbine (see Figures 2 and 3). Although not completely sealed, this environment chamber can be purged with a cover of argon gas, maintaining a semi-inert environment for the potassium turbine during testing, as discussed later. The environment chamber is supplied with several glass porthole observation windows and instrumentation bulkheads.

The entire test rig assembly is mounted in a cross-spindle trunnion arrangement as shown in Figure 4, which allows the turbine train to pivot freely to accommodate the vertical thermal expansion of the furnace structure and the associated movement of the 8-inch diameter inlet pipe carrying vapor into the turbine. Axial thermal expansion of the 15 foot long inlet pipe is accommodated by mounting the support trunnions of the turbine and power train system on a fore-and-aft rolling dolly. The turbine-drive train system is installed in the existing potassium vapor test facility as shown in Figures 5a, b, c.

Alignment

Two of the prime requirements for obtaining reliable, long-life operation are: (1) good dynamic balance of rotating components, (2) proper axial spacing and radial alignment of all the rotating components. Proper radial and axial alignment is particularly important for the reliability of the flexible diaphragm coupling between the test turbine and water brake. This is obtained by axial shimming at assembly. Shims are provided at the mounting flange for proper axial positioning of the water brake, and

within the end-piece of the flexible diaphragm coupling for precise axial location of its diaphragms.

Proper radial alignment is secured by maintaining:

1. Proper parallelism between the flange faces of each Bytrex torquemeter,
2. Proper parallelism and squareness of the conical mounting faces of the support structure,
3. Proper radial location of rotating parts.

The first two conditions are obtained by machining the structure cone faces parallel to each other with the structure in a partially assembled condition, and by careful internal shimming of torquemeters to get parallelism between the two faces of each torquemeter. The third condition is obtained by using four diametrically opposed adjusting bolts mounted on the aft face of the conical support structures (as shown in Figures 1 and 2), where the water brake and steam turbine torquemeter faces mount. The water brake and steam turbine are moved from side-to-side and up-and-down until their radial alignment is satisfactory, as measured by a rotating dial indicator mounted on the rotor shaft of one component and measured against the internal mounting flange of the other. Radial alignment of the rotating components and squareness of the rotor centerline and component mounting face within 0.001 inches (Full Indicator Reading) will be obtained.

External Supply and Control Systems

The major subloop systems supplying the power train comprise.

1. Bearing lubrication system
2. Water brake water supply and control system (including over-speed sensing system)
3. Actuation gas to disk brake-shoe cylinders
4. Cooling air to disk brake
5. Steam turbine power and control system
6. Air to steam turbine purge seal
7. Cooling air to Bytrex torquemeters
8. Lube supply to gear coupling (if used)

A single lubrication system supplies the oil for lubricating all the power train bearings, and is separate from that which supplies the bearings of the three-stage test turbine. A schematic of the hook up, control and lubricant flow rates of the power train lubrication system is shown in Figure 6. It is a recirculating loop which supplies oil (Socony-Mobil DTE 797 light turbine oil) in liquid jets to the water brake and steam turbine bearings. A scavenge pump (connected to the same shaft as the lube supply pump) is employed to return the oil to the reservoir, fifteen feet below the test rig (see Figures 5c and 6).

The water brake supply and load control system is shown schematically in Figure 7. Control is maintained by setting the back-pressure of the water brake exhaust by means of valves WDV-2 and WDV-2A (see Figure 7). This in turn, varies the water level on the outer periphery of the rotating disks. The water flow will be maintained at a rate corresponding to approximately a 200°F exhaust temperature. Although if the temperature rises above 212°F, boiling will occur at the inner free surface of the water in the brake, boiling at the exhaust valve is inhibited by the increased water pressure of approximately 20 psig. The flexible hose which carries water to and from the water brake is made of rubber-impregnated fabric with a layer of wire braid winding for maximum safety against a leak in either the inlet or exhaust line, either of which could produce a loss of load on the potassium test turbine with subsequent overspeed. City water flows straight through the water brake and into a drain; no recirculating is employed. (See NOTE, end of Section III).

In the event of an overspeed of the turbine rotor, one of two possible emergency procedures will proceed automatically (provided the emergency "arming" switches are closed, which is the intended condition for all testing).

1. At a predetermined overspeed (about 21,000 rpm) the primary overspeed trip (see Figure 8a) will be energized. This will cause the water brake solenoid valve WSV-2 to open, allowing an additional 10 gpm of water to enter the brake. The disk brake will not yet be energized. If the rotational speed decreases to 16,000 rpm, the above corrective action will cease, and normal operation will resume.

2. In the event of a more excessive overspeed (characteristic of a run-away condition), the above actuation will occur, and if the rotor speed reaches 24,000 rpm, the secondary (panic overspeed) trip will energize the disk brake to completely stop the turbine. Simultaneously, the disk brake cooling air valve will be opened.

As indicated above, cooling air to the disk is required only in the event of emergency actuation of the brake, and then is required in large quantities. An air line of 1½-inch diameter pipe supplies 90 psig shop air to the special coolant distribution manifold straddling the brake disk (discussed in Section IV). This air blows to atmosphere.

Figure 8a shows the overspeed sensing electrical schematic for condition #1 above, and Figures 8b and 8c show the schematic flow line hook-ups for the disk brake coolant and the water brake flooding. Overspeed is sensed by a speed pickup mounted on the potassium turbine adjacent to the brake disk (Figure 3). Figure 9a shows the higher run-away overspeed sensing electrical schematic, and Figure 9b shows the flow line schematic for actuation of the disk brake pucks (shoes). The gas pressure to drive the pucks against the rotating disk is derived from a bottle of gaseous argon, as shown.

The steam turbine is supplied with 165 psig steam which is exhausted from the turbine directly to atmosphere. Control is maintained by the remotely-operated valve in the inlet line shown in the flow schematic of Figure 10. To minimize flow-line resistance to the slight rotation of the turbine housing necessary for torque measurement, the steam is supplied and taken from the steam turbine through steel "dog-bone" tubes with O-ring gaskets at inlet and exit as indicated in Figure 1. Steam turbine driving torque is recorded both during the potassium turbine startup and during the power train tare-testing. In potassium turbine testing, after the turbine has reached self-sustaining power, the steam flow to the steam turbine is reduced but not entirely stopped, allowing a small amount of steam to flow continuously to keep the steam turbine always hot and ready to respond quickly to maintain rotational speed in case of an emergency stoppage of vapor to the potassium turbine.

Purge air is supplied to the steam turbine's purge seal (Figures 1 and 11) to prevent mixture of steam and oil. The air is supplied from a 90 psig shop air line, and vents out to atmosphere with the exhaust steam.

The cooling air flow to the Bytrex torquemeters is supplied continuously during testing to the internal strain-gaged regions through 1/4-inch tubing and simple hand-valves. This air is exhausted to atmosphere.

Figure 12 shows the lubrication supply to the gear coupling. The coupling (described in detail later) is designed to hold lube oil by internal O-ring seals and overhanging cups on its ends. A stationary tube injects heavy lubricating oil by actuation from the control room. (If the quill shaft is used instead of the gear coupling, this lube system will not be needed.)

Instrumentation

Figure 13 is the instrumentation drawing for the power train. It shows the numerous pressure, temperature, and dynamic measurements used to monitor the train during testing and to identify any major malfunction or deterioration of the critical components. A list of all power train instrumentation is given in Table II.

Assembly and Disassembly Techniques

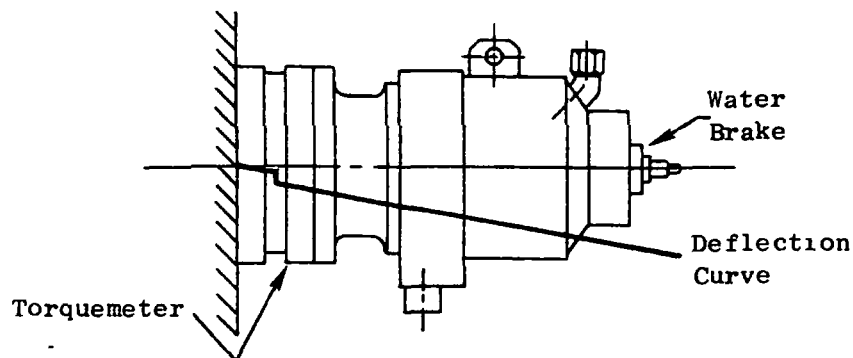
The power train can be separated as an assembled unit from the environmental chamber surrounding the potassium turbine at flange A (Figures 2 and 3). After removal of the flexible membrane coupling, it can be remounted at flange A and operated as an assembled unit without running the potassium test turbine. The initial checkout of the power train will be done with this assembly so as to allow its mechanical checkout prior to installation of the potassium test turbine. The general assembly procedures of the power train are given in Table III.

Checkout Testing

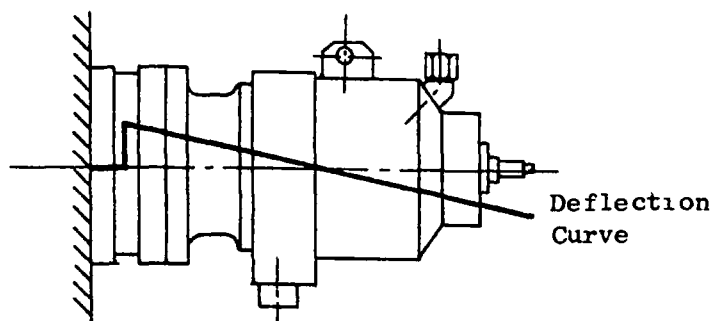
This is performed by assembling the drive train to flange A prior to installation of the potassium test turbine and running the drive train through its speed range as indicated in Item #11 of Section II. This allows measurement of rotor vibrations and critical speeds, system temperatures, pressures, loop flows, torques, speed, and observation of other instrumentation functions, to identify any operational problems of the power train before committing the three-stage potassium turbine to test. In addition, as indicated previously, tare losses of the rotor system are determined by running the power train to full speed with the steam turbine, and simultaneously observing its torque requirements. These data are necessary in evaluating the true power output of the potassium turbine during later testing.

Vibrational Analysis

Due to the geometric complexity of the power train, several natural frequencies and critical speeds are encountered on the various components at a number of operating speeds. Previous experience on the similar two-stage turbine drive train revealed a critical speed in the steam turbine at about 3500 rpm (which is sufficiently low as to constitute no problem) and in the water brake at 7000 rpm and at 17,000 rpm. The critical speed at 17,000 rpm constituted a problem because it was within the range of performance testing, and sufficiently near the design-point to be encountered many times. During a checkout period, by placing accelerometers on various specific locations along the running power train, the critical speeds and their specific mode shapes were plotted. It was found that the water brake experienced (at 7000 rpm) a first flexural frequency as a rigid body cantilevering off its flexible torquemeter as shown below:



The 17,000 rpm (second flexural frequency) resulted in a maximum deflection near the brake forward flange as shown below, and again was one in which the water brake acted as a rigid body on the rotating and translating torquemeter.



It was found also that the critical speed measured on the running power train could be changed from about 20,000 rpm down to the 17,000 rpm mentioned above by simply adding water to the brake, with a corresponding change in its mass. Because of the geometric complexity of the power train, and the difficulty of determining the amount of water in the brake at any given operating condition, an analytical prediction of the critical speeds of the power train is not presented. Rather, the natural frequencies of the various assembled and mounted components will be determined experimentally in laboratory set-ups and in the test cell, while rotating, as described above.

In anticipation, however, of future analyses of power train critical speeds and mode shapes, the spring constants of the Bytrex torquemeters (both transverse and rotational) have been experimentally determined, and are presented in Section IV.

Prior to testing, it will be possible to change the natural frequency of the water brake or steam turbine if desired by simply adding mass to their casings, and straps of lead are being procured for this purpose. In addition, a vibration damper has been designed* and fabricated for installation on the aft flange of the water brake (if needed) as shown in Figure 1, and as discussed later.

*Work performed by the Mechanical Engineering Department of the University of Cincinnati.

Rotor Balancing

An important means for avoiding high critical speed amplitudes with associated deterioration of bearings, seals, and rotors, is to obtain the best initial dynamic balance of rotating components. Item 9 in the DESIGN SPECIFICATIONS of Section II defines these requirements. The balance of parts is accomplished on a balancing machine (the manufacturer's specifications of which are given in Table IV) which is shown in Figure 14. The typical balancing sequence employed in all rotor balance is given by Table V.

A standard GE Vibration Analysis Form for rapid conversion of peak-to-peak vibration amplitude, velocity, and vibration frequency to "g"-loads (handy for use in the test cell during testing) is included as Table IX.

NOTE In June, 1968, the water brake exit hose was replaced by a flexible stainless steel hose. The water hose between the load train and facility piping was replaced by a Teflon metal braid hose.

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PAGE 14

IV. INDIVIDUAL COMPONENT DESIGN

The following discussion presents the detail design and specifications of individual components and subassemblies of the power train. A geometric description of each component is included in the set of manufacturing drawings of Figures 35 through 86. For those components purchased as completed assemblies from a single manufacturing source, an assembly drawing only, is included. Since the power train components operate at moderately low temperatures, they are in general, designed with the 0.2 percent yield strength as the limiting stress. For those materials operating under significant stresses, their 0.2 percent yield strength is given as a function of temperature in Figure 15.

Support Structure and Environmental Chamber

The power train and potassium turbine environment chamber structure assembly comprises three major subassemblies as indicated in Figures 2 and 3.

1. Forward environmental chamber
2. Aft environmental chamber
3. Power train conical and cylindrical support structure and enclosing water brake and steam turbine covers, as discussed previously.

The environmental chamber, as indicated previously, serves as a cover surrounding the potassium turbine which can be purged with argon if desired to minimize the possibility of contamination of the main potassium loop in case of a leak in the vicinity of the turbine exhaust (which is below atmospheric pressure) and contains any vapor which might escape from the turbine inlet region and directs it into the exhaust scrubber. It also provides a shield in the unlikely event of a catastrophic rotor failure, and acts as a support for the turbine and inlet vapor pipe (Figures 2, 3 and 4).

The forward environmental chamber, shown in Figure 3, is the same forward section of the glove box which was employed on the previously discussed two-stage turbine test program (see photos in Reference 3) and was, therefore, not procured as a component for this test program. It comprises a cylindrical section with visual ports and an exhaust duct

directly to the scrubber system, as indicated above. It is made of $\frac{1}{2}$ -inch thick stainless steel plate, and attaches to the inlet vapor pipe by a bellows-connection (Figure 2).

The aft environmental chamber (which is shown in Figure 3, and in detail in Figure 43) is an enclosure larger than the one required for the two-stage potassium turbine, due to the increased size of the three-stage turbine. It consists of a cylindrical section, and is located eccentrically with the centerlines of the inlet duct, forward environmental chamber (discussed above), and the centerline of the aft power train (Figure 3). It also has visual ports, and provides a bulkhead for the attachment of the various pressure and temperature probes emanating from the potassium turbine to the efflux pressure measuring system (discussed in References 4 and 6) and to the test cell control room. It also is made of $\frac{1}{2}$ -inch thick stainless steel plate rolled into a cylinder. The three-stage turbine exhaust extends downward through a port in the bottom of this environment chamber, which is attached to the turbine exhaust duct by a bellows (Figure 2). It can be seen that the test turbine is mounted to the aft flange of this environmental chamber, and cantilevers forward into the chamber. The turbine is also supported by the 8-inch diameter inlet duct which is indexed to the forward enclosure (discussed above), so that the turbine is held redundantly by the environmental chamber. The entire potassium turbine-enclosure-power train is supported by transverse trunnions in the aft enclosure (Figure 4), allowing automatic rotational freedom of the test rig resulting from the vertical thermal expansion of the boiler tube bundle (located 15 feet upstream of the turbine). As indicated previously, the entire trunnioned system is mounted on a fore-to-aft rolling dolly to compensate for axial thermal expansion of the inlet potassium vapor duct.

The power train is also cantilevered off the same aft face of the above environmental chamber, and thereby, is aligned and indexed axially with the potassium turbine. The power train support structure consists of a cylindrical support ring, and two conical sections in tandem (as shown in Figure 1, and also as shown in Figure 48 as a subassembly for

machining parallel faces for the water brake and steam turbine torque-meters). The aft portion of the power train structure is covered by a light weight cylindrical housing which serves as a bulkhead for attachment of steam, water, oil and air service lines, and pressure measuring gages. It also provides a thermal shield, and a protective cover against accidental impact of the accurately aligned and delicately mounted water brake and steam turbine. The conical support structure is made of $\frac{1}{2}$ -inch thick rolled steel (the forward cone is 316 stainless steel to match the axial thermal expansion of the flexible coupling-rotor system), and the aft cone and thin cylindrical cover is made of AISI 1020 sheet steel with painted surfaces to prevent deterioration with time.

It can be seen that all the weight load is transmitted through the heavy cylinder A (Figure 3) and thence is divided to the water brake through the conical disk B, and the steam turbine through the conical segment C. Surfaces D and E of Figure 3 must be held parallel (within 0.001 inches FIR) to minimize the angular misalignment within the rotor system. As indicated above, radial location of the rotating component centerlines is obtained during assembly by adjustment of the four transverse positioning bolts (Figure 2) located 90 degrees apart on both the water brake and steam turbine support structure mounting face.

Water Brake

The water brake (shown in the assembly of Figure 1, and enlarged in Figure 16) is purchased as a complete unit from the Industrial Engineering Corporation of Philadelphia, Pennsylvania. The brake was purchased according to the GE manufacturing specification 02-0203-00-A given in Appendix B, and the manufacturing vendor retained the responsibility for the detailed engineering design, drafting, manufacturing, and assembly. However, as a result of GE experience gained on the similar two-stage turbine water brake, and the requirement to ensure a 5000 hour continuous operating life, GE-SPPS engineers maintained close engineering surveillance and liaison with the brake manufacturer, and made acceptance of the brake contingent on GE approval of the design details.

Figure 17 shows the water brake used in the two-stage turbine program, and the design modifications incorporated in the brake for the present three-stage turbine. Although similar in geometry and operating principle, the present brake retains none of the original hardware, with the exception of the Stellite-6B rotor and stator disks which withstood the previous 5000 hours of testing (References 6 and 8) without deterioration. Following are the main design modifications and improvements incorporated in the present brake to obtain maximum reliability:

1. Stainless steel end caps for the brake replace the previously used aluminum caps which distorted and corroded during the several disassemblies and long running time during testing.
2. The length of the end-cap rabbets with the main cylindrical housing has been increased to make a tighter, more reliable and durable fit, and the brake has been provided with jack-screws for disassembly.
3. The diameters for the bearings are line-bored in the stator housings with the stator assembled to provide maximum bearing alignment after assembly.
4. The stator mass has been increased (partly by employing the above stainless steel end caps and partly by adding external weight) to reduce the critical speed of 17,000 rpm (which was encountered on the similar brake during previous testing) to a level further below the design-point operating speed.
5. The bearing lubrication and seal system has been improved (the original water brake was designed for air-oil mist lubrication, and was later converted to liquid jet lubrication with minimum redesign of that brake).
6. The inlet and outlet water hose sizes have been reduced to obtain greater flexibility with resulting reduction in interference with torque read-out (inlet hose diameter: 3/4 inch. Exit hose diameter: 1.0 inch), and the quality of the hose has been improved.

The materials employed in the brake fabrication are shown in Table VI and are as follows:

Shaft. H-11 tool steel (high strength, non-rusting alloy).
Rotor disks: Stellite-6B (same disks as used in two-stage turbine brake).
Stator disks: Stellite-6B (same disks as used in two-stage turbine brake).
End caps and cylindrical housing: 316 stainless steel.
Bearing balls and races: 52100 vacuum-remelted steel.

The general mechanical design of the water brake, and specifically the design of the disks for maximum energy absorption is a product of the brake manufacturers experience. However, several checks of rotor stress, brake rotor critical speed, ball bearing life, and seal design were performed by GE-SPPS and coordinated with the manufacturer. As a basis for rotor stress analysis, the following assumptions were made:

Operation at design point speed (18,250 rpm) for 5000 hours continuous

Brake temperature: 200°F.

Ball bearing spring constants: 10^6 lbs/in.

Continuous torque input from three-stage turbine 1130 in.-lbs.

Water brake rotor weight moment of inertia - 19 lb/in.^2
(see Table VII).

To calculate the shear stress on the water brake rotor shaft under the maximum torque conditions and at the weakest point (where the flexible coupling is attached), it was assumed that the potassium turbine would be brought from 20,000 rpm to a dead stop in two (2) seconds by some form of water brake seizure, or otherwise heavy instantaneous loading with the resulting rotating inertia of the potassium turbine rotor (see Table VII) and the continuing driving power of the turbine being transmitted through the flexible coupling and felt by the water brake shaft at the flexible coupling joint. It is assumed that under this instantaneous high braking torque from the water brake, the potassium turbine (and power train) rotor decelerates according to a cosine curve as shown in Figure 18a. The inertia of the decelerating potassium turbine rotor then produces the torque shown as the lower curve in Figure 18b. In addition, it is assumed that the torque output of the potassium turbine increases linearly from 1000 in.-lb to 2500 in.-lb

during this deceleration due to the undiminished flow of vapor to the turbine. The sum of these torque values is shown as the upper curve in Figure 18b as a function of time after onset of the emergency braking, and yields a maximum of about 3500 in.-lb at 1.2 seconds after brake initiation. It is this maximum torque value which has been used to size both the water brake shaft and the flexible diaphragm coupling (discussed later) for intermittent duty.

For the water brake shaft, at the location where the two diametrically opposed 3/16-inch square keys attached the flexible coupling to the brake, the water brake shaft torsional stress produced (concentrated at the keyways) during this maximum braking condition is 93,000 psi, which is almost exactly equal to the allowable shear stress limit for the H-11 rotor shaft material, yielding a shaft torsional factor-of-safety of

$$\text{F.S.}_{\text{shaft}} \approx 1.0$$

The stresses produced in the keys is discussed later along with the flexible diaphragm coupling and in Appendix A. Neglecting the holes in the rotor disks and any shrink-fit of the disks on the rotor shaft, the maximum rotating tangential stress at the inner bore of the disks at 18,250 rpm is 11,200 psi, which is safely within the allowable 0.2 percent yield stress limit of 88,000 psi for Stellite-6B at 200°F.

To determine whether the water brake rotor will experience critical speeds within the operating range with subsequent bearing life deterioration, the rotor was analyzed for critical speeds and it was found that the rotor-bending-mode critical speed occurs at about 32,000 rpm. This is substantially above the operating speed, and therefore, is not a cause for concern. The only critical speed that would be experienced would be that of the entire water brake acting as a mass cantilevered off the end of the flexible Bytrex torquemeter. As indicated previously, to cope with this problem, increase of the water brake mass and the design and procurement of a vibration damper (as discussed later) have been done.

A major design consideration in obtaining adequate reliability for 5000 hours operation involves the life of the water brake ball bearings. In calculating the available bearing life, the following environmental conditions were assumed, and are stated in the Design Specification 02-0203-00-A (Appendix B) issued to the brake manufacturer:

1. Each bearing has an axial load of 100 lbs.
2. The bearing radial load = 8 lbs for the forward bearing and 9 lbs for the aft bearing (corresponding to the actual rotor weight of about 17 lbs) (no dynamic unbalance load was considered).

The rotor bearings selected by the brake manufacturer and verified by GE-SPPS are:

	<u>Size</u>	<u>Type</u>	<u>Predicted Life (hrs)</u>
Forward	25 mm (305 bearing)	New Departure (N-Dur-300)	79000
Aft	20 mm (304 bearing)	New Departure (N-Dur-300)	64000

The predicted life is the B_{10} life (90 percent of all bearings tested exceed this value, and the average bearing runs over five times as long).

These calculated bearing lives are obviously more than adequate, as long as no unaccounted-for conditions such as high rotor unbalance, undefined critical speeds, or excessive thrust loads (which are not now anticipated) are imposed on the bearings. The detailed presentation of bearing design assumptions and life calculations for the various bearings considered is given in Appendix A.

If it is assumed that the water in the brake produces no axial thrust loads on the rotor (which appears reasonable) the only thrust loads that could be imposed on the rotor would be from the thermal expansion of the flexible coupling which is firmly attached to both the water brake rotor shaft and the aft end of the potassium turbine rotor. The thermal expansion of this coupling was calculated according to the assumptions of temperatures and dimensions shown in Figure 32, and as

presented in the analysis summary in Appendix A. It is assumed that the coupling is installed (room temperature of 70°F) in a relaxed position (each end of the coupling being fastened rigidly to its associated shaft). Then, after thermal expansion of the coupling due to increase of environmental temperature during testing (amounting to three(3) mils tension) the coupling exerts a total forward force of 5.8 lbs, on the water brake bearings, which can be seen to be greatly less than the assumed load of 100 lbs per bearing used in the bearing life calculations indicated above. The spring loading on the bearings to prevent "skidding" is about 80 lbs.

The manufacturer's rating of the water brake capacity is given by Figures 19a and 19b which show the maximum horsepower and torque-absorbing characteristics of the brake as a function of rotor speed. These curves assume that the brake is completely full of water. It can be seen from Figure 19b that at a torque value of about 2000 in.-lbs, the mechanical shear capacity of the brake shaft becomes limiting, and at a torque of 1250 in./lbs, the full-scale capacity of the water brake torquemeter is exceeded. Normal operation of the brake will of course, be below this torque limit, (it will be at 1130 in.-lbs). Previous experience with the similar brake on the two-stage turbine indicated rather, that the brake operates toward the lower end of its torque capacity.

Water Brake Vibration Damper

As indicated above, in addition to increasing the mass of the water brake casing (obtained mainly by employing stainless steel end caps in place of the previously used aluminum caps and by strapping lead weights to the brake exterior) which will lower the previously experienced 17,000 rpm critical speed to some value further below the design point speed, a water brake vibration damper has been designed and fabricated to facilitate safe operation when going through these critical speeds. Minimum amplitude at critical speed is obtained by tuning the vibration damper to a 180° out-of-phase relationship with the brake's vibration. This will be used only if needed to facilitate safe operation through critical speeds.

The damper assembly attaches to the aft flange of the water brake, and its mechanical details are shown in Figure 20 and in Figures 53 through 55. The damper comprises a mass "M" (Figure 20) mounted on a variable-length cantilever beam (bolt) immersed in a viscous oil with closely controlled clearance "C" between the mass "M" and the outer housing. The assembly is mounted on the water brake aft flange, and "tuned" by mounting the brake on its torquemeter in a laboratory set-up, and exciting the system by a variable-frequency external driver. The bolt length is adjustable until the proper phase relationship is established, at which time maximum energy is absorbed in the oil-filled clearance space "C". This concept for vibration damping has been successfully employed on high-speed precision machine tools now in operation. The fluid dynamics and vibration damper analysis was performed by the Mechanical Engineering Department of the University of Cincinnati, and the detailed mechanical component design was performed by GE Space Power and Propulsion Section personnel.

Steam Turbine

The steam drive turbine is mounted on the aft end of the power train (Figure 1), and is essentially the same as the one used on the two-stage turbine, with two exceptions:

1. Its power-producing capacity has been increased twenty-five percent by an increase in its flow nozzle area.
2. Critical components (rotor, bearings, and some housings) have been replaced and/or reworked, and its seals have been redesigned for improved reliability (considering 5000 hours continuous operation).

The steam turbine is made for continuous operation at speeds up to 30,000 rpm, and almost unlimited service. These turbines are sold commercially, and are highly reliable (many are used by GE as work-horse turbines for spinpit drives).

The detailed assembly of the steam turbine is shown in Figure 21. The basic machine is manufactured by Barbour Stockwell Corporation, of Cambridge, Massachusetts, and has been modified by the manufacturer for use in this power train according to the GE requirements dictated by

the Steam Turbine Rework Specification 02-0195-00-A (and modification) shown in Appendix B. Figure 22 shows the changes made to the steam turbine used in the two-stage turbine test for use in the forthcoming three-stage turbine test.

Figure 23 shows the approximate power requirements for the steam turbine to drive the entire test rig, including steam turbine, water brake, and potassium turbine to speed levels as indicated in Section II (Items 6 and 11) during the tare-testing immediately preceding the potassium turbine vapor test. This comprises the maximum anticipated power demand ever to be made on the steam turbine in this test. Although the steam turbine, as used previously in the two-stage turbine test appeared adequate, it was sufficiently marginal in power to justify an increase (as has been done on similar turbines by the manufacturer) by increasing the nozzle plate flow area from 0.375 in.² to 0.470 in.² (a 25 percent increase). This change correspondingly provides about 25 percent increase in power, making possible better acceleration and deceleration response when operating at the highest speeds and power demands. Figures 24a and 24b show the manufacturer's estimated performance of the steam turbine over its rated speed range with the previous 0.375 in.² nozzle plate area, and the anticipated 25 percent increase in power at the design-point with the 0.470 in.² plate.

In addition to this power increase, the turbine was made more reliable by replacing its rotor, seals and bearings, with either new standard parts or newly designed parts as shown in Figure 22. As can also be seen, the forward seal closing the oil sump was redesigned. A carbon face rubbing seal was incorporated to replace the previously used rubber lip seal. The seal used is a "Sealol No. 3-6B007CH-13" with the seal runner ground optically flat within 3-to-6 helium light bands (see Figures 22 and 58).

Disk Brake Design

The disk brake is provided as a safety against a potassium turbine runaway condition resulting from loss of normal water-brake load, or rupture of the coupling between turbine and water brake. It also may

serve as a supplement to the water brake in the event of the need for immediate turbine braking to zero speed. The disk brake assembly is mounted on the aft end of the potassium turbine shaft (Figure 1) and is shown in detail in Figure 25. Figure 26 shows the basic differences between the disk brake employed previously on the two-stage potassium turbine program and the one for the present three-stage turbine.

A disk brake was introduced during endurance testing of the two-stage potassium turbine before the last 3000 hour endurance test. It was tripped at different times at turbine overspeed conditions and was extremely effective in reducing turbine speed in a matter of seconds. Every braking action resulted, however, in surface damage of the Udimet 700 brake disk which in turn resulted in unbalance of the turbine rotor. Also, the heat generated at the brake disk flowed out and coked the grease used for lubrication of the gear type coupling. As a result, the brake disk was removed from the turbine shaft and was not replaced for the remaining endurance testing. This short experience showed, however, that proper design of a disk brake poses considerable problems in regard to disk stress levels, heat transfer and cooling requirements. Information available from commercially used disk brakes in airplanes and other applications indicated that our requirements by far exceeded established experience in regard to brake surface speed and heat loading. Problem No. 1 was to identify a suitable disk material of high strength but also sufficient ductility not to be prone to surface cracking. The second effort was directed towards reducing plug pressure and increasing the swept areas. The third was an attempt to remove the heat of friction generated at the disk surface almost as fast as it is generated by effective impingement cooling.

By NASA instruction, the plan of an experimental disk brake development program was abandoned in favor of providing at least two interchangeable disk brakes to be able to exchange disks in case surface damage and unbalance should still result from brake actuation.

The disk brake has been designed to operate in either of the following two manners.

1. The first and most desirable disk brake performance would be to completely stop the potassium turbine and associated power train rotor in about 5.3 seconds from an initial speed of 24,500 rpm without any assistance from the water brake, and while the potassium turbine continues to be supplied with vapor at design-point power. The disk brake should remain intact during such stoppage.
2. The second minimum requirement on the disk brake is to reduce the rotative speed of the potassium turbine and power train rotor without assistance from the water brake from 24,500 rpm to 18,000 rpm without cracking or adversely affecting the balance of the brake disk.

Under the first braking condition, the brake disk would be removed to be inspected for damage and replaced with another one of identical design and equal balance (as closely as can be obtained). Under the second braking condition, the disk would still be serviceable, and a complete shutdown of the system would not be required. The first braking condition is more severe, and is the condition which constitutes the basis for design of the disk and the associated disk cooling system discussed below.

The major design parameter for sizing an uncooled brake disk is the heat generated per unit of swept area (which is the total area of the two annular traces left by the contacting pucks on the disks radial faces). It is assumed that all of this heat is absorbed into the disk face, and has been empirically determined to be a maximum allowable value of $1200 \text{ ft-lb/in.}^2 \text{ sec}$. At energy absorption levels above this, disk surface cracking can be expected. For this turbine-power train system, the disk brake loading will be set to decelerate the turbine from 24,500 rpm to a complete stop in 5.3 seconds. The total energy dissipated, with the potassium turbine still driving, will be about one million ft-lb. On the basis of the above criterion, an uncooled disk swept area of 167 in.^2 would be required. However, rotating stresses limit the disk diameter to ten (10) inches and the largest puck diameter which can be used (providing a maximum swept area) is 2.375 inches resulting in a total swept area of only 114 in.^2 . Although this size combination would easily provide the required stopping torque,

it will handle only about one-half of the heat absorption rate before cracking. To remove the remaining half of the heat load, a disk air impingement cooler (Figure 42) has been provided. The disk cooling air is actuated simultaneously with any actuation of the brake, as discussed previously (Figure 8). The cooler employs impingement cooling concepts which have produced convection coefficients in excess of $1000 \text{ Btu/hr ft}^2 \text{ }^\circ\text{F}$. The cooler sizing analysis is given in detail in Appendix A.

Referring to the disk brake assembly (Figure 25), the ten-inch diameter disk is stopped by the two $2 \frac{3}{8}$ -inch diameter shoes (pucks) acting in opposed fashion. The disk is symmetrical about its center plane, being 0.35 inches thick at the puck contact surfaces and one (1) inch thick at the bore. The disk bore is 1.84 inches, and it delivers braking torque to the turbine shaft by two keys, integrally machined on the disk bore to avoid the disk stress concentrations inherent with removable keys. The disk bore is ground to a 0.0035 inch tight fit with the shaft. The combined rotating disk stress ("Effective" Henky v. Mises stress) is shown in Figure 27a, and is seen to be a maximum of 53,000 psi at the bore. This is within the allowable limit of about 60,000 psi, which is the 0.2 percent yield strength of the L605 disk material (see Figure 15) at room temperature (see Figure 15).

The caliper assembly is a Goodyear Industrial Brake Model PD1473-1, consisting of two half-assemblies which can be spaced for any disk thickness. The puck moves as a piston in the housing bore, and is actuated by argon gas pressure (Figure 9b). Each puck assembly comprises an aluminum piston to which is bolted the shoe. The shoe is a resin material, impregnated with brass granules, and when inactive, is held $1/16$ inch away from the rotating disk by a retracting spring.

The choice of L605 (Haynes alloy #25) for the disk material was made partly as a result of previous experience on the disk brake of the two-stage potassium turbine (References 3, 5 and 6).

For the present application, L605 is chosen because of its resistance to thermal cracking and its high-temperature strength. Thermal crack resistance is considered to be, in part, measured by the weldability of the material, since the heated surface due to puck friction is roughly comparable to the heat-affected zone of a weld. It is known that L605 can be readily welded while the previously used U-700 is essentially non-weldable, and other commonly available material candidates such as A286 and Rene'41 require careful heat-treating subsequent to welding to prevent thermal cracking.

The puck material, Goodyear BL76, is similar to the material of the previous two-stage turbine's pucks in that both are impregnated resins. However, the new material is expected to be less abrasive because of smaller particles in the resin, thereby minimizing the amount of material removal from the disk with corresponding reduction in the risk of disk unbalance resulting from such unequal metal removal from its surface. A decreased coefficient of friction will result, but it can be compensated by an increase in puck pressure. The puck clamping force will be 2890 lbs, corresponding to a braking torque at the pucks of 2200 in.-lb (assuming a friction coefficient of 0.2). The deceleration of the total power train after initiating braking action is shown in Figure 27b for various levels of braking torque.

The material for the forced air cooler and ducting is AISI 316 stainless steel, chosen for its weldability, and resistance to atmospheric corrosion.

Bytrex Torquemeters

The torquemeters, from which are cantilevered the water brake and steam turbine, sense torque as a function of slight deflection of strain-gaged flexure plates between two parallel circular mounting flanges, as indicated previously. The detailed internal designs of the two torque-meters are shown respectively in Figures 28 and 29. Torque is sensed by the slight rotation of the water brake or steam turbine casing produced by (1) tangential force transmitted through the water of the brake and by (2) the reaction torque on the casing of the steam turbine. The

torque vs. deflection curves for each torquemeter are shown in Figure 30. The maximum capacity of the water brake torquemeter is 1250 in.-lbs, with a corresponding angular rotation of the water brake casing of approximately 0.1 degrees. The maximum torque of the steam turbine torquemeter is 200 in.-lbs with an associated rotation of 0.08 degrees. The steam turbine torquemeter is more delicate than that of the water brake, necessitated by the lower full scale torque-measuring capacity of the former. Each torquemeter is accurate within a one percent error of the full scale readout. During assembly of the drive train, considerable care must be exercised in mounting the water brake and steam turbine (especially the heavier steam turbine) to prevent any damage of the torque-sensing strain gages.

As an aid in calculating the natural frequency of the power train system, the shear and bending spring constants of the two torquemeters were experimentally determined* by attaching one face to a large mass and exciting the other face.



For each torquemeter, six directional stiffnesses were obtained - four in the shearing direction, and two in bending. The results are presented in Table VIII.

Flexible Membrane Coupling.

The flexible membrane coupling connecting the potassium vapor turbine and the water brake (Figure 1) is designed and manufactured by the Bendix Corporation of Utica, N.Y., according to the GE Specification requirements

*Work performed by Department of Mechanical Engineering of the University of Cincinnati.

shown in Appendix B. The manufacturer has had considerable experience in the design and manufacture of this type of coupling, which by virtue of its having no requirement for any type of lubrication, is characteristic of couplings which will ultimately be required for space turboalternator systems. Figure 31 shows an enlarged view of the flexible membrane coupling, and Figure 33a compares it with the gear-type coupling used on the previous two-stage turbine. It comprises a rigid center spool connected to two face flanges through varying-thickness flexible membrane disks which allow for limited radial and angular misalignment of the mating shafts. Certain details regarding installation connections and balancing arbor requirements were specified by GE-SPPS personnel, (residual dynamic unbalance of less than 0.010 gm-in. required). while the design of the membranes and their attachment to mating pieces was the full responsibility of Bendix Corporation. Initial attempts to design the coupling with a floating end so as to eliminate the need for precise axial positioning of the coupling ends was ultimately abandoned as a result of the strong recommendation of Bendix personnel that each end of this type of coupling be rigidly attached to its associated shaft. A floating end would first require shaft lubrication, and also could, through a "ratcheting" action, result in axial extension of the coupling during operation and stretching of the membranes. Therefore, precise axial positioning of the coupling ends is performed through use of shims (Figure 31), which are ground to fit at assembly.

The membrane's OD and ID are 3.120 inches and 1.580 inches respectively. The thickness of the membrane tapers from 0.047 inches at its hub to 0.012 inches at its OD. The membrane is electron-beam welded to the adjoining spool and end flange retainer. The membrane has been sized to withstand the maximum torque produced by an emergency shutdown (20,000 rpm to 0 rpm in two seconds along a cosine curve) corresponding to a flooded or seized water brake, but with the potassium turbine vapor undiminished in flow and temperature.

Refer again to Figure 18 for the resulting torque versus speed transmitted by the coupling during this extreme shutdown (max. $\tau = 3500$ in.-lb). The coupling has been designed both to operate for an

infinite life under design point torque (~1100 in.-lb) and nominal shaft misalignment and for this emergency-condition torque (see Design Specification 02-0199-00-A in Appendix B).

A design summary of the flexible diaphragm coupling is given in Appendix A. It presents the operating requirements, design assumptions, and detailed physical information regarding the flexible membranes. Both the allowable and the intended misalignments are compared, and the intermittent-duty and steady-state membrane and shaft stress conditions and limits are given, along with their resultant factors-of-safety.

Following is a comparison of the conditions under which the manufacturer guarantees infinite coupling life and the intended steady-state operating conditions set as a goal by SPPS engineers:

	Manufacturer's Allowable Limits (Hot)	Assembly Goals (Hot)
Parallel offset of shaft ϵ	0.020 in.	0.001 in.
Axial extension (two membranes)	+ 0.040 in.	0.008 in.
Shaft out-of-parallelism	- 12 mn.	1 mn.
Total membrane angular misalignment	22.8 mn.	1.5 mn.

Due to the requirement to rigidly attach each end of the flexible diaphragm coupling to its respective shaft, and the necessity to minimize axial "stretch" of its membranes (with associated reduction in the coupling's operating life and reliability) the accurate prediction of axial thermal expansion affecting the coupling is necessary. Such a thermal analysis could result in the need to assemble under some initial axial deflection of the couplings so that the membranes would be moved to a relaxed condition upon heat-up of the power train and turbine system accompanying testing.

The thermal expansion model is shown in Figure 32. The temperatures, dimensions, and thermal coefficients of expansion for all components in the thermal expansion "loop" are given. It is assumed that the axial differential expansion between the outer static structure and the power-train rotor is divided equally between the two flexible membranes. The

results of the analysis (Figure 32) show that each membrane undergoes an axial extension of only 0.0015 inches upon heat-up. This is almost insignificant, compared to the allowable extension of 0.020 inches quoted by the coupling manufacturer (without sacrificing infinite coupling life), and has, therefore, been ignored in the membrane stress calculations in Appendix A. The stress produced in the membrane due to the 0.0015 inch axial extension is about 4600 psi. The ultimate tensile strength of the E-18A membrane alloy is 255,000 psi, and its 0.2 percent yield strength is about 250,000 psi at 200°F, which is obviously well above the actual membrane stress level.

As indicated in the flexible coupling Specification 02-0199-00-A of Appendix B, the flexing membrane is to be tested by the manufacturer for 10 million cycles (28 hours at 6,000 rpm) at a flexing membrane stress of 80,000 psi before its delivery and installation by General Electric Company personnel. This test will reveal any previously undetected flaws in the material of the membrane, and if the coupling survives this test, it will run for essentially an infinite life at the design-point conditions (18,250 rpm at 200°F with 1130 in.-lbs torque).

Stress calculations were performed for the forward end of the water brake shaft where the coupling adjoins it by means of two 3/16-inch square, diametrically-opposed keys. It was assumed that all of the torque experienced during the above emergency shutdown (3500 in.-lbs) would be transmitted by one of the keys. The key shearing stress and side-compressive bearing stresses were calculated, and yield the following factors-of-safety (see Appendix A for detail).

Key Shear Stress F.S. = 1.2

Key Bearing Stress F.S. = 1.1

Several alternate couplings were investigated for this application, and some of these included flexible membrane couplings with several convolutions of various thicknesses. Generally, the thinner membranes and several convolutions result in lower fatigue stress for any given misalignments or deflections. However, the single membrane-per-end coupling possesses geometrical simplicity and is of adequate capacity as indicated above.

Quill-Shaft Coupling

Figure 34a shows a quill-shaft coupling for connecting the water brake to the steam turbine. It is a spindle which fits inside the steam turbine rotor shaft and is reduced to a diameter of 0.375 inches over a length of approximately 5.38 inches. The quill shaft is attached firmly to the aft end of the water brake shaft and is allowed to move axially in the steam turbine rotor (under any differential thermal expansion) but is constrained against rotation within this rotor by two keys located at the aft end of the steam turbine.

The quill shaft is made of AISI 4340 steel and is heat-treated to a hardness of Rc 30-35. Under maximum applied torque (200 in.-lb) from the steam turbine, the quill shaft torsional stress is 14,900 psi. (The allowable 0.2 percent yield stress for this shaft is 120,000 psi.) Under emergency stopping conditions wherein the steam turbine rotor would be stopped by the quill shaft in 2.0 seconds, the maximum torsional stress would be 31750 psi. The first critical speed (bending) of the quill shaft is calculated (by computer program - VAST) to be 64,500 rpm. The torsional frequency of the quill shaft - steam turbine rotor is only 1237 rpm, and is therefore, passed-through during the first few seconds of system start-up.

Gear Coupling

An alternate coupling for connecting the water brake and steam turbine (shown in detail in Figure 34b) is a conventional gear-tooth type coupling, and is manufactured according to GE Specification 02-0206-00-A (Appendix B) by the Koppers Company of Baltimore, Md. As indicated previously, a coupling with a similar tooth form (also made by Koppers, and shown in Figure 33a) was used in the two-stage turbine test as the main potassium turbine - water brake coupling, and was employed throughout the 5000 hours of testing (Reference 6). In the previous two-stage turbine test, the water brake and steam turbine were connected by a coupling sleeve having straight spline teeth in each end. The differences between that coupling and the one presently discussed are shown in Figure 33b. The present coupling is lubricated

with SAE #140 lubrication oil instead of the molybdenum disulfide grease which was previously used, to eliminate the problem of liquid-solid separation due to centrifuging-action, typically encountered with grease lubrication. As mentioned previously, this coupling differs from conventional design in that provision has been made for retaining oil and allowing for injection of lubricant during operation as necessary (Figure 12). The tooth slots machined inside the end-bells of the coupling are straight-sided, while the teeth on the connecting spool have rounded sides and a crowned top. This provides for point-contact of the teeth, and allows for misalignment between the coupling ends with much less deleterious effects than would be encountered with straight-splined teeth on both the spool and end-caps. This coupling automatically adjusts for axial differential thermal expansion of mating rotor and stator structures by a slight axial sliding of the teeth. The total tooth length over which proper engagement can take place is about 1/4 inch. The coupling is 4.9 in. long, and it is considered feasible to keep its centerline misalignment to within 0.001 inch at assembly.

V. CONCLUSIONS

From the analysis and discussion presented herein, it is concluded that the Power Train for the Three-Stage Potassium Test Turbine is capable of continuous operation for 5000 hours, as defined under item 5 of Section II, and in fact, should actually operate reliably for a life several times this value. Where it was considered necessary, newly designed components have been installed (e.g. shaft couplings and rotor bearings) to minimize vibration and to maximize load-carrying capacity. On all components (with the exception of bearings and seals) the calculations attending the design show that they will operate for virtually an infinite life, provided the required alignment and balance discussed herein is maintained. Also, while intended only as a reliable test-facility power-absorbing and drive system, the power train also promises added operational information and experience on a typical non-lubricated coupling, characteristic of that which will ultimately be required in space.

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SECTION VII

TABLES

“PAGE MISSING FROM AVAILABLE VERSION”

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TABLE I
THREE-STAGE TURBINE POWER TRAIN - GENERAL DATA

GENERAL

Minimum Design Life	5000 hours
Performance Test Speed Range	15400 to 20000 rpm
Endurance Test Planned Duration	5000 hours
Endurance Test Speed	18250 rpm

COUPLINGS

Forward Coupling Type	Flexible Membrane (Bendix)
Aft Coupling Type	Quill Shaft (Gear Tooth Alternate)
Forward Coupling Length (Between Membranes)	6.6 in.
Aft Coupling Length (Teeth ϕ)	5.0 in.
Forward Coupling OD (Membranes)	3.3 in.
Aft Coupling OD (Teeth ϕ)	2.0 in.
Forward Coupling Lubricant	None
Aft Coupling Lubricant	SAE 140
How Lubricate Forward Coupling	Unnecessary
How Lubricate Aft Coupling	Unnecessary

WATER BRAKE

Manufacturer	Industrial Engineering
Disk Material	Stellite 6-B
Rotor Shaft Material	H-11 Tool Steel
Total Brake Weight	82 lb
Number of Rotor Disks	6
Number of Stator Disks	4
Forward Bearing Type	New Departure Q3305XR7A
Aft Bearing Type	New Departure Q3305XR7A
Inlet Hose Size	3/4 in. Standard AN Hose
Exhaust Hose Size	1 in. Standard AN Hose
Maximum Capacity at 21500 rpm	680 hp
Lube Flow Rate - Each Bearing	0.1 gpm

DISK BRAKE

Disk Material	L-605 (Haynes 25)
Disk Thickness	0.500 in.
Disk OD	11.00 in.
Caliper - Puck Assy Manufacturer	Goodyear
Caliper - Puck Designation	BL76
Forced Air Disk Cooling Employed	Yes
Forced Cooling Air Flow Rate	1.5 lb/sec
Puck Clamping Force	2890 lbs
Puck Material	Metal-Impregnated Organic Compound
Puck Actuating Gas	Bottled A or N ₂

TABLE I (Cont'd)

STEAM TURBINE

Manufacturer	Barbour-Stockwell
Turbine Type	Re-Entry (Terry Type)
Rotor OD	8 in.
Maximum Continuous Rated Speed	25000 rpm
Nozzle Plate Flow Area	0.470 in. ²
Turbine Weight	514 lb
Forward Bearing Type	Fafnir 306 K
Aft Bearing Pair Type	Fafnir 306 W1
Total Lube Flow Rate	0.7 gpm
Forward Shaft Seal Type	Carbon Face
Spindle Length	10.8 in.
Rotor Material	AISI 4340
Saturated Steam Inlet Pressure Available	165 psig
Approximate Maximum Power at 20000 rpm	91 hp
Approximate Steam Flow Rate	4000 $\frac{\text{lb}}{\text{hr}}$ @ 150 psig

TORQUEMETERS

Torquemeter Manufacturer	Bytrex
Torquemeter Type	Strain Gage
W/B Torquemeter Capacity	1250 in.-lb
W/B Torquemeter Maximum Deflection	0.1 degrees
S/T Torquemeter Capacity	200 in.-lb
S/T Torquemeter Maximum Deflection	0.085 degrees
W/B Torquemeter Internal Geometry	Flex. "Dog-Bones"
S/T Torquemeter Internal Geometry	Flex. "L"-Plate

STRUCTURE

Support Geometry	Cones and Cylinders
Means for Aligning W/B and S/T	Cross-Bolts
Means for Setting Axial Positions	Shims
Forward Environmental Chamber ID	30 in., 1/4 in. thick
Aft Environmental Chamber ID	36 in., 3/8 in. thick

VIBRATION DAMPER

Principle Employed	Increase in Casing Mass, or Resonating Cantilevered Mass
Fabrication Material	304 SS
Damping Fluid	Dow Corning 210 Silicone Fluid
Approximate Tuning Speed	15000 - 17000 rpm

**TABLE II. INSTRUMENTATION USED TO MONITOR OPERATION OF
POWER TRAIN OF THREE-STAGE POTASSIUM TEST TURBINE**

(See Figure 13 for Location and Installation Details.)

Item No.	Parameter	Location	Station	Range	Control Room Readout	Cats No.	Channel or Position
						Digital No.	
92	Water Brake Torque	Water Brake	11	0-1250 lbs	Dial, Sanborn & Digital	115*	2A
93	Steam Turbine Torque					116*	
94	Speed	Steam Turbine		0-25,000 rpm	Berkley		
96	Steam Turbine Inlet Pressure	Steam Turbine		0-100 psig	Digital	109	
126	Displacement Stm. Turb. Aft Brg. Vert.			0-5 mils	Sanborn		6A
127	Displacement Stm. Turb. Aft Brg. Horz.			0-5 mils	Sanborn		6A
130	Displacement Wtr. Brk. Forward Vert.			0-5 mils	Sanborn		7A
131	Displacement Wtr. Brk. Forward Horz.			0-5 mils	Sanborn		7A
132	Steam Turbine Bearing Fwd. Temp.			250°F	TR#3-15		
133	Steam Turbine Bearing Mid. Temp.			250°F	TR#3-16		
134	Steam Turbine Bearing Aft Temp.			250°F	TR#3-17		

TABLE II. INSTRUMENTATION USED TO MONITOR OPERATION OF
POWER TRAIN OF THREE-STAGE POTASSIUM TEST TURBINE

Item No.	Parameter	Location	Station	Range	Control Room Readout	Cats No.	Channel or Position
						Digital No.	
135	Steam Turbine Lube In Temp.			250°F	TR#3-13		
136	Steam Turbine Lube Out Temp.			250°F	TR#3-14		
137	Steam Turbine Lube Oil Press.			80 psig	Dial & Wrng. Lt.		
138	Steam Turbine & Wtr. Brk. Lube Flow			1 gpm	Warning Light		
139	Wtr. Brk. Water Inlet Temperature	Water Inlet Line		60°F	Digital	31 163	
140	Wtr. Brk. Water Inlet Temperature	Water Inlet Line		60°F	TR#3-10		
141	Wtr. Brk. Water Inlet Temperature	Water Inlet Line		60°F	Alarm		
142	Wtr. Brk. Water Outlet Temperature	Water Line		190°F	Digital	32 164	
143	Wtr. Brk. Water Outlet Temperature	Water Line		190°F	TR#3-20		
144	Wtr. Brk. Bearing Fwd. Temperature	Water Brake		180°F	TR#3-18		
145	Wtr. Brk. Bearing Aft. Temperature	Water Brake		180°F	TR#3-19		

TABLE II. INSTRUMENTATION USED TO MONITOR OPERATION OF
POWER TRAIN OF THREE-STAGE POTASSIUM TEST TURBINE

Item No.	Parameter	Location	Station	Range	Control Room Readout	Cats No.	Channel or Position
						Digital No.	
146	Water Brake Water Flow	Water Inlet Line			Digital	106	
147	Water Brake Water Flow	Water Inlet			Dial		
174	Water Brake Seal Water Flow	Water Brake		1 gpm	Digital	107	
176	W.B. Lube Oil Out Temp. Fwd. Brg.				TR#3-7		
177	Ambient Temp. Bendix Coupling				TR#3-23		

* Also Channels 123, 130, 139, 150, 159

** Also Channels 124, 131, 140, 151, 160

*** Also Channels 117, 125, 132, 141, 152, 161

TABLE III
GENERAL PROCEDURES FOR ASSEMBLY OF POWER TRAIN

The following is the general sequence of assembly of power train components. It does not comprise a detailed step-by-step procedure of the many intricate and complex details required in all its alignment, manifold and flow-line hook-ups, nor an itemizing of the comprehensive battery of stack-up measurements and checks against predicted dimensions which must be performed along with the assembly. Refer to Figures 1 through 5 for geometric clarification.

1. Mount aft environment chamber in supporting main cross-trunnions of the facility support dolly. (See Figure 5)
2. Mount forward cone support (Figure 45) to aft face of the aft environmental chamber.
3. Mount mid-cone cylindrical segment (Figure 46) to the forward cone support.
4. Mount water brake Bytrex torquemeter (and align it with the potassium turbine shaft centerline if the potassium turbine has been installed). Axially position the torquemeter by shimming to obtain proper axial positioning for the flexible diaphragm coupling when installed.
5. Mount water brake and secure it to the torquemeter. Hook-up its flow lines (water and oil).
6. Mount the aft cone support (Figure 47) and feed the flow lines through access holes and attach them to the provided supports.
7. Install the gear (Koppers) coupling center spool piece in the bore of the steam turbine Bytrex torquemeter.
8. Mount the steam turbine Bytrex torquemeter with the proper shim (for axial positioning of the steam turbine) and center it with respect to the water brake shaft by adjusting the cross-bolts which radially position the torquemeter.
9. Mount the steam turbine whose spindle has a receptacle for engaging the water brake shaft aft end.
10. Mount the aft cylindrical cover (Figures 49 and 50) and hook-up all flow and pressure gage lines. Install the aft cover plate (Figure 51).
11. Install the steam turbine "dog-bone" connectors (Figures 60 and 63) to the turbine's inlet and exhaust connectors. Attach the steam feed and exhaust hoses to the aft plate.

TABLE III (Cont'd)

12. Hook-up all remaining flow lines, thermocouples, and pressure-measuring systems, and connect with control-room.
13. Check-out test the unit by running on the steam turbine.
14. After completing all preliminary power train rotational tests, install the flexible (Bendix) coupling between the potassium turbine and water brake shafts. Shim the coupling axially, as provided for in its design, to eliminate axial-extension flexing of its membranes.

TABLE IV
BALANCING MACHINE SPECIFICATIONS

Specifications

Sensitivity. Measurable unbalance displacement with low noise level bearings	(1 micro-inch) 0.0000010 inch
Measurable maximum unbalance displacement double amplitude	0.375 inch
Dynamic, pure static, or Force-Couple Indication	As desired
Ratio of direct unbalance unit indication	10,000 to 1
Minimum meter deflection for 1 micro-inch unbalance	5/16 inch
Maximum rotor weight	130 pounds
Minimum rotor weight	1/4-ounce
Each work support has 130 pound weight capacity	
Distance between work supports, Standard Machine (Can be increased on special orders)	24 inches
Width of bearing support	1-11/32 inch
Swing diameter	20 inches
MU-6-24	24 inches
Swing diameter outboard mounting	36 inches
MU-6-24	40 inches
No limitation of bearing diameter Standard V-blocks except 3" max.	
Balancing speed, standard	900 to 3000 RPM
Drive motor, standard (On order, 1/2 HP, or 3500 RPM)	1/4 HP, 1780 RPM
Unbalance Plane Separation Ratio (Crosstalk)	50 to 1 or better
Floor vibration rejection ratio	100 to 1
Weight of balancing stand	95 lbs.
Weight of Amplifier	35 lbs.
Bench Size, recommended minimum	2 x 4 feet
Foundation and hold down requirements	None
Power requirements	115 to 125 v, 1 phase 60 Cycles

TABLE V

GENERAL PROCEDURE FOR DYNAMIC ROTOR BALANCING

(See Table IV for Balancing Machine Specifications)

The following is a listing of major steps in rotor balancing, and does not attempt to list the many detailed step-by-step procedures involved in the technique of balancing, which to a large degree still resides in the craftsmanship of the operator.

The balancing machine functions generally as follows: The rotor to be balanced is placed on the two "vee-block" mounts on top of the adjustable arms extending upward from the support cross-rails (see Figure 14 or Table IV). These mounts are lined with nylon cylindrical rods as running surfaces for the rotor being balanced. These "vee-block" assemblies float on ball bearings which allow them to move in response to rotor unbalance vibration. As the rotor is driven (within the machine's standard speed range of from 900-to-3000 rpm) electromagnetic transducers generate electrical signals proportional to the motion of the floating mounts, caused by the unbalance. All other signals are eliminated by means of a tuneable filter. A stable oscillator, slaved by the unbalance signal, triggers the flashing of a stroboscopic lamp, which illuminates the location of rotor unbalance instantaneously, once each revolution. A meter, measuring units of unbalance, displays the amount of unbalance in each balance plane. For a given set-up, the meter must be calibrated using a standard with known unbalance.

In performing the balance,

1. The rotor is assembled with all its components (if possible) including only the inner race of the rotor ball bearings (on assemblies which use ball bearings). This inner race comprises a spacer which takes its place in the stack of rotor sleeves and parts, and its ball groove provides a smooth running surface for mounting the assembly on the nylon rods in the balance machine "vee-blocks."
2. The rotor unbalance is then temporarily cancelled-out or "removed" by placing a sticky wax opposite the indicated rotor heavy spots. This continues until a minimum attainable meter reading is reached. The balance machine is then calibrated by placing a known unbalance weight (wax piece) at a known radius. Then

$$\frac{\text{known unbalance} \times \text{radius}}{\text{meter units read}} = \frac{\text{gram-inches of unbalance}}{\text{meter unit}}$$

TABLE V (Cont'd)

All wax is then removed, and the rotor unbalance in each balancing plane and their static sum is determined.

3. The balance of the rotor is achieved by then removing metal (by local grinding) from the rotor as near as possible to the indicated heavy spot on the rotor. This is continued (and strongly dependent upon the operator's skill) until an acceptably low residual unbalance is achieved (see defined requirements for this program in Section II). On a number of balancing arbors for various rotor disks, etc. for this test, residual unbalance levels as low as 0.002 gram-inches have been accomplished.
4. Replacement of the rotor bearing inner race with the actual test bearing, or other re-assembly steps for the rotor are then performed, and the build-up of the test component ensues.

(See Table III for assembly procedures of power train components.)

TABLE VI. FABRICATION MATERIALS FOR POWER TRAIN COMPONENTS

	<u>Drawing No. *</u>	<u>Fabrication Material</u>
I. DISK BRAKE	--	--
Brake Disk	941D362	L-605
Speed Counter Disk	47C141499	AISI 4150
Cooling Manifold and Support	263E159	304 Stainless Steel
Brake Caliper Assembly	PD 1473-1	Ductile Iron (Housing) BL 76 (Pucks)
II. WATER BRAKE TORQUEMETER	941D325	--
Flanges	V	AISI 1020 Steel
Internal Flexure Beams	47C141476	Ladish D6AC Alloy
III. STEAM TURBINE TORQUEMETER	941D324	--
Flanges		AISI 1020 Steel
Internal Flexure Plates	V	V
IV. FLEXIBLE MEMBRANE COUPLING	941D314	--
Turbine Attachment Member	V	E-18V
Flexible Membranes	V	E-18V
Water Brake Attachment Member	V	E-18V
Center Spool Piece	V	E-18V
Shaft Keys	142B1495	SAE 1020
V. WATER BRAKE	GE 941D311	--
	Ind. Eng. C9529-5	--
Rotor Shaft	V	H-11 Tool Steel
Rotating Seals	V	Carbon Face Seals
Rotor and Stator Disks	V	Stellite 6-B
Center and End Housings	V	316 Stainless Steel
Ball Bearings (Barden)	N.A.	52100 Steel
Inlet Hose	N.A.	Wire Wrapped Rubber Fabric
Exit Hose	N.A.	Wire Wrapped Rubber Fabric
Static Seals	N.A.	Buna-N O-Rings
Calibrating Torque Arm	941D334 & 47B116014	AISI 1050 Steel

TABLE VI. FABRICATION MATERIALS FOR POWER TRAIN COMPONENTS (Cont'd)

VI. WATER BRAKE VIBRATION DAMPER	941D379	--
Mounting Flange	941D376	347 Stainless Steel
Damper Housing	941D385	304 Stainless Steel
Damper Mass and Cantilever	47C141537	303 Stainless Steel
Damper Oil	N.A.	Dow Silicone Oil 210 or 210 H
VII. AFT GEAR COUPLING	47C141479	--
Water Brake Attachment Member	V	AISI 4140
Steam Turbine Attachment Member	V	AISI 4140
Central Spool Piece	V	AISI 4140
Flexible Seals	N.A.	Buna-N O-Rings
Lubricating Oil	N.A.	SAE 140 Oil
VIII. STEAM TURBINE	Barbour Stockwell 6037	--
Outer Housing	V	Cast Steel
Rotor Spindle	47B116005	AISI 4140
Rotor Wheel	V	V
Speed Counter Disk	941D323	AISI 4140
Labyrinth Purge Seal Plate	V	Bronze
Ball Bearings	N.A.	52100 Steel - Phenolic Retainer
Dynamic Seal	47C141480	Carbon Face - Nitrided Steel Runner
Shaft Keys	MS 20066-182	AISI 1020
Inlet and Exhaust Flexible Ducts	47C141492 & 47C141493	304 Stainless Steel
Static Seals	N.A.	Buna-N O-Rings
IX. SUPPORT STRUCTURE	47R199263	--
K Turbine Mounting Cone	47R199262	304 Stainless Steel
Cylinder and Water Brake Cone Assy	47R199260	AISI 1020 Steel
Steam Turbine Cone Assembly	263E149	AISI 1020 Steel
Outer Cylindrical Cover	47R199261	AISI 1020 Steel
Aft Support Plate	263E151	AISI 1020 Steel
Forward Environmental Chamber	4012211-127	304 Stainless Steel
Aft Environmental Chamber	47R199258	304 Stainless Steel
Adjustment Bolts	47B116021	SAE 1050

TABLE VI. FABRICATION MATERIALS FOR POWER TRAIN COMPONENTS (Cont'd)

X. MISCELLANEOUS	--	--
Lube Fittings and Manifolds	941D344	316 Stainless Steel
Water Inlet and Exhaust Fittings	N.A.	316 Stainless Steel
Assembly Bolts	N.A.	316 Stainless Steel and Carbon Steel

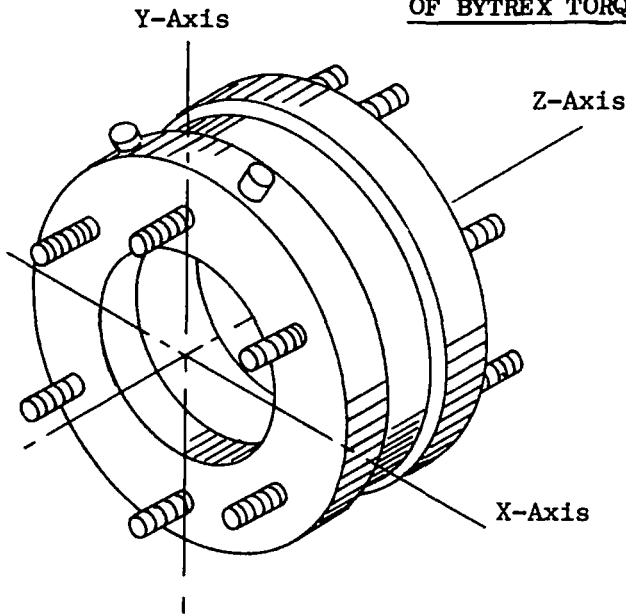
*"N.A." Denotes "Not Applicable." - All Drawings Presented in Section IX
 "V" Denotes Drawing or Information Available to Manufacturing Vendor Only.

TABLE VII
APPROXIMATE WEIGHTS AND DYNAMIC DATA FOR
POWER TRAIN COMPONENTS

<u>Component</u>	<u>Weight (lbs)</u>	<u>Rotor Weight Moment of Inertia (WR²) (lb-in.²)</u>
THREE-STAGE TURBINE		
Rotor*	94.9	393
Stator*	420.0	--
DISK BRAKE		
Brake Disk	14.4	118
Static Cooling, Braking and Mounting Structure	20.5	--
FLEXIBLE MEMBRANE COUPLING	5.4	10
WATER BRAKE TORQUEMETER	18.3	--
WATER BRAKE		
Rotor	10.8	19
Stator	71.2	--
Flow Ducts in Power Train	7.0	--
BENDIX COUPLING	3.2	5
STEAM TURBINE TORQUEMETER	18.4	--
STEAM TURBINE		
Mounting Spacer	15.0	--
Stator	160.0	--
Rotor	20.0	94
Flow Ducts in Power Train	10.0	--
POWER TRAIN SUPPORT STRUCTURE		
Conical and Cylindrical Supports	784.0	--
Cover and Aft Plate	96.0	--
ENVIRONMENTAL CHAMBER		
Forward Section	550.0	--
Aft Section	890.0	--
MISCELLANEOUS		
Lube Hoses, Fittings	10.0	--
Instrumentation (Pressure, Temperature)	20.0	--
TOTAL	3239	639

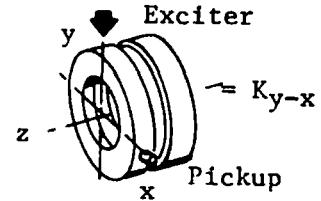
*See Reference 2 for detailed breakdown.

TABLE VIII
TRANSVERSE AND ROTATIONAL SPRING CONSTANTS
OF BYTREX TORQUEMETERS

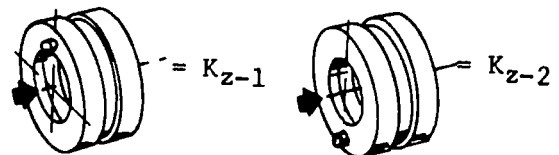


Nomenclature

For shear spring constants, the first subscript indicates axis along which the torquemeter is excited, and second is axis along which deflection is measured, e.g.,



For bending spring constants, the KZ-1 constant denotes excitation along Z-axis and measurement of deflection axially at top of torquemeter, and KZ-2 denotes the same, except measurement of deflection is axial at the bottom of the torquemeter, e.g.,



For 1250 in.-lb Torquemeter (Water Brake)

Identification	Compliance in./lb	Spring Stiffness lb/in.
K_{x-x}	2.9×10^{-6}	3.45×10^5
K_{x-y}	10^{-8}	10^{-8}
K_{y-y}	3.2×10^{-6}	3.2×10^5
K_{y-x}	3.3×10^{-6}	3×10^5
K_{z-1}	1.2×10^{-6}	8.3×10^5
K_{z-2}	2.8×10^{-6}	3.57×10^5

For 200 in.-lb Torquemeter (Steam Turbine)

Identification	Compliance in./lb	Spring Stiffness lb/in.
K_{x-x}	4.8×10^{-6}	2.08×10^5
K_{x-y}	4.5×10^{-6}	2.22×10^5
K_{y-y}	2.8×10^{-5}	3.57×10^4
K_{z-1}	1.0×10^{-6}	1.0×10^6
K_{z-2}	1.0×10^{-6}	1.0×10^6

Table IX (Cont'd)

HOW TO USE THE VIBRATION ANALYSIS FORM

THE VIBRATION QUANTITIES

A vibration of pure sinusoidal wave shape has four characteristic quantities. displacement, velocity, acceleration, and frequency

Two of these quantities sufficiently describe a vibration of pure sinusoidal wave shape, and, when two of these quantities are known, the remaining two quantities may be determined either by calculation or more conveniently by the graph on the reverse side

THE GRAPH

The graph consists of four logarithmic scales. Frequency is plotted on the horizontal axis, velocity on the vertical, displacement on the diagonal downward from left to right, and acceleration on the diagonal upward from left to right. The lines composing these scales will be referred to as coordinates

THE USE OF THE GRAPH

On the graph, follow those lines which represent the two known quantities to their point of intersection. Follow a line from this point parallel to the coordinates of the desired quantity, and read its intersection with the corresponding axis

EXAMPLE

Given—peak-to-peak displacement=20 mils, frequency=100 cps
To find—acceleration and velocity

Procedure—follow the diagonal coordinate of 20 mils displacement to its intersection with the vertical coordinate representing 100 cps frequency. 1. A line from this intersection, parallel to the diagonal acceleration coordinates, intersects the acceleration axis at a value of approximately 10.2 times gravitational acceleration. 2. A line from this intersection, parallel to the horizontal coordinates, intersects the velocity axis at 6.3 in/sec

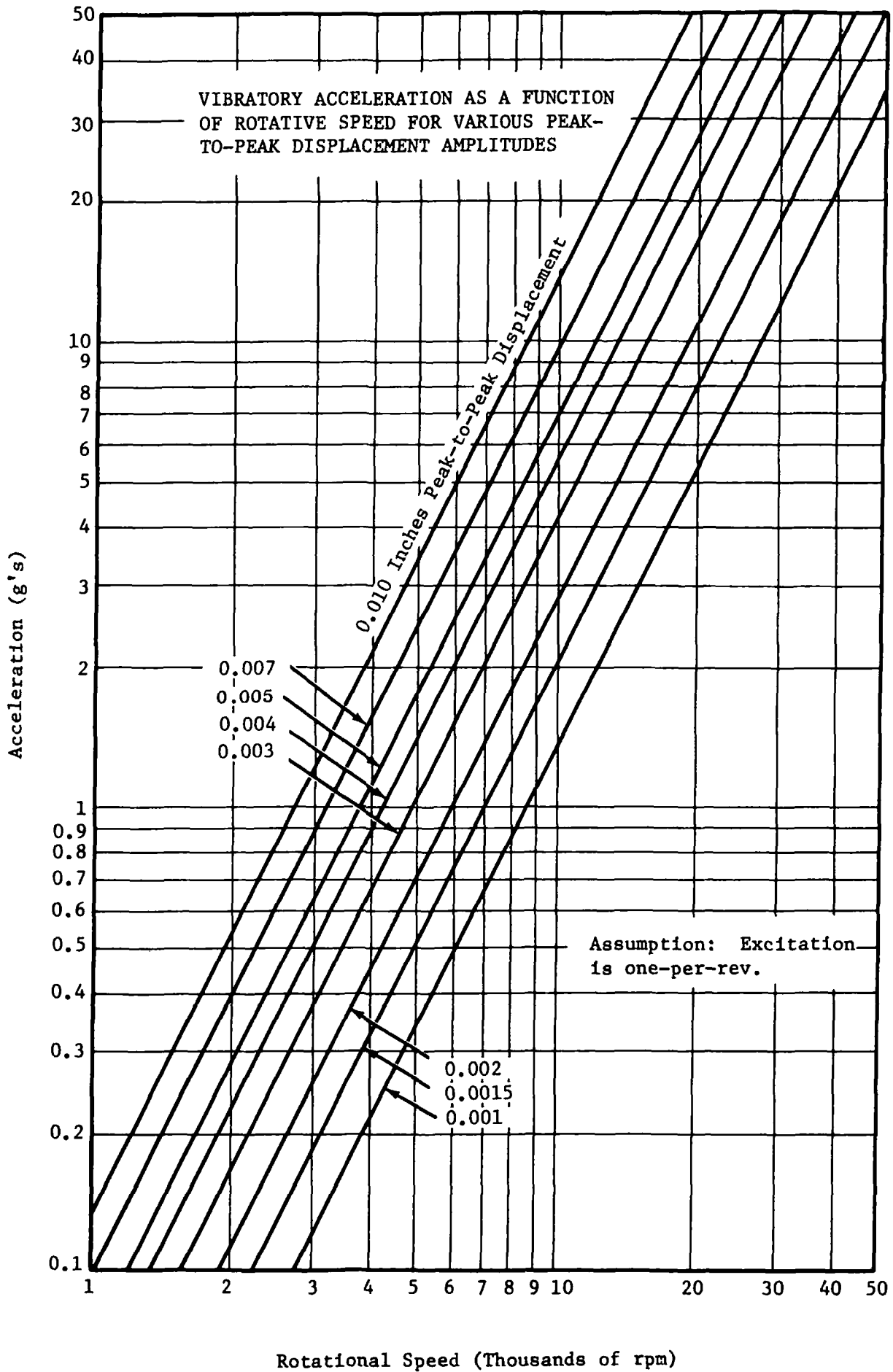
Result—1. Acceleration 10.2 x gravity
-2. Velocity 6.3 in/sec.

PEAK VALUES, RMS-VALUES, AVERAGE VALUES

The figures on the coordinates of the graph refer to peak-to-peak displacement, peak velocity and peak acceleration. When rms values and average values of displacement, velocity and acceleration are desired, use the table below to obtain the conversion factors

To Obtain ↓	Multiply Number of →	Peak-to-Peak	Peak	RMS	Average
Peak-to-peak		1 000	2 000	2 828	3 142
Peak		500	1 000	1 414	1 571
RMS		354	707	1 000	1 111
Average		318	636	900	1 000

Table IX. Cont'd.



SECTION VIII

FIGURES

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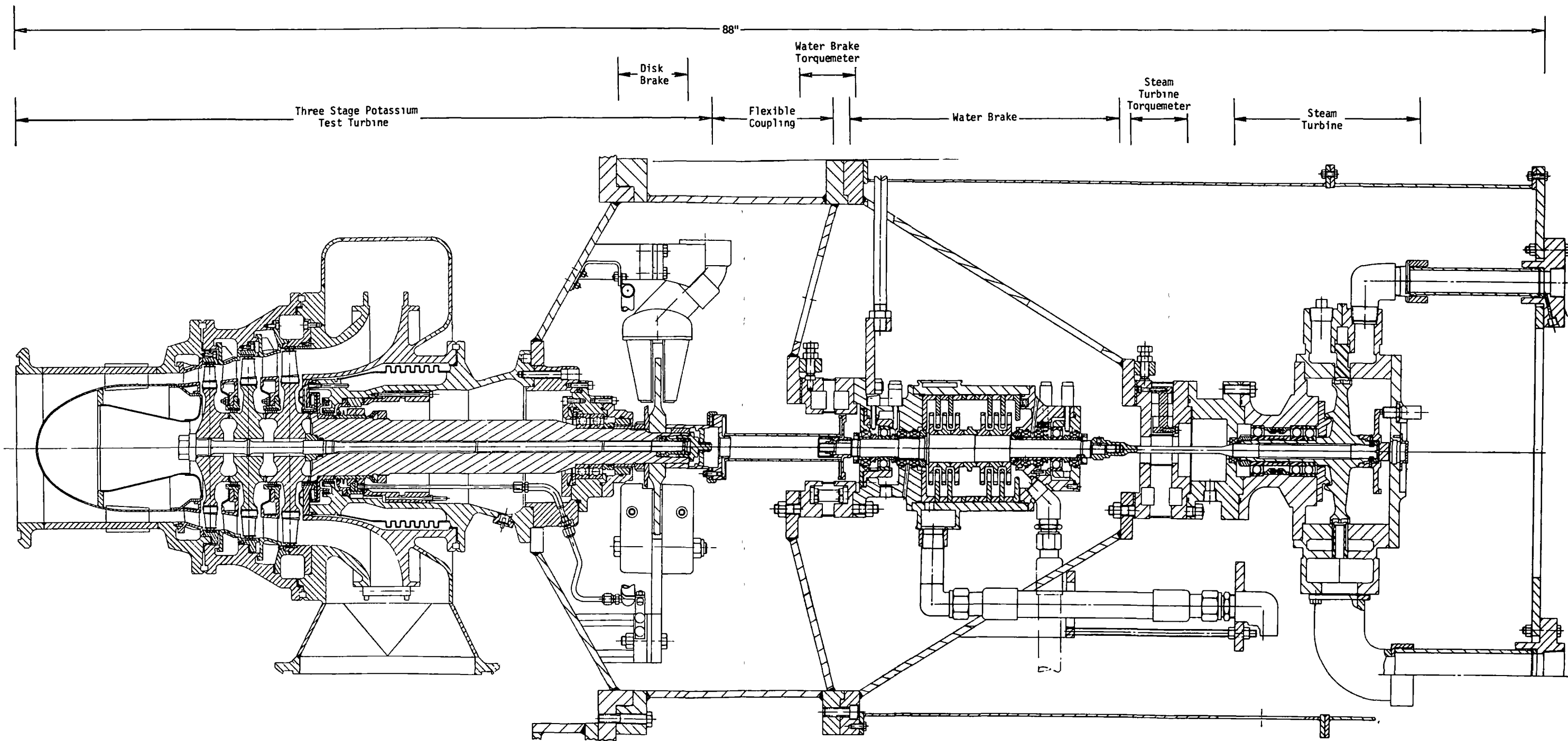


Figure 1. Test Turbine and Power Train Assembly. (47R199273)

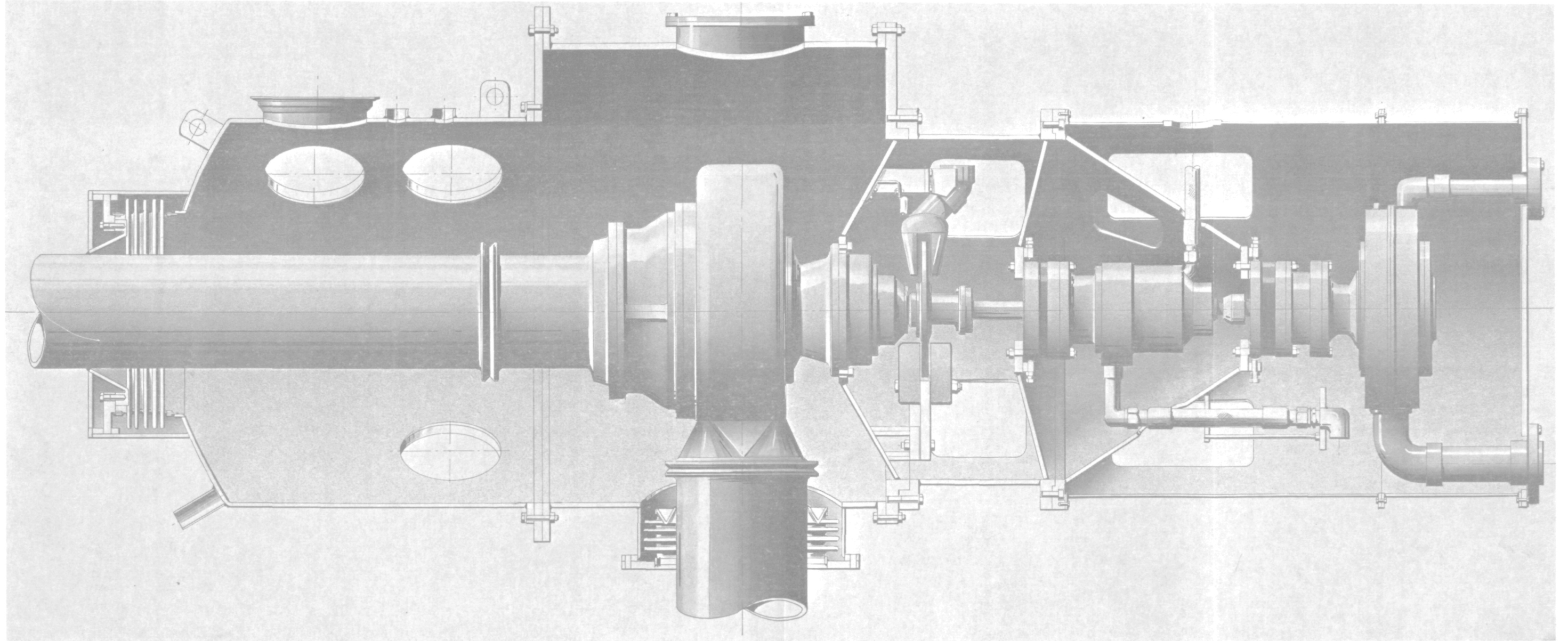


Figure 2. Test Rig Assembly

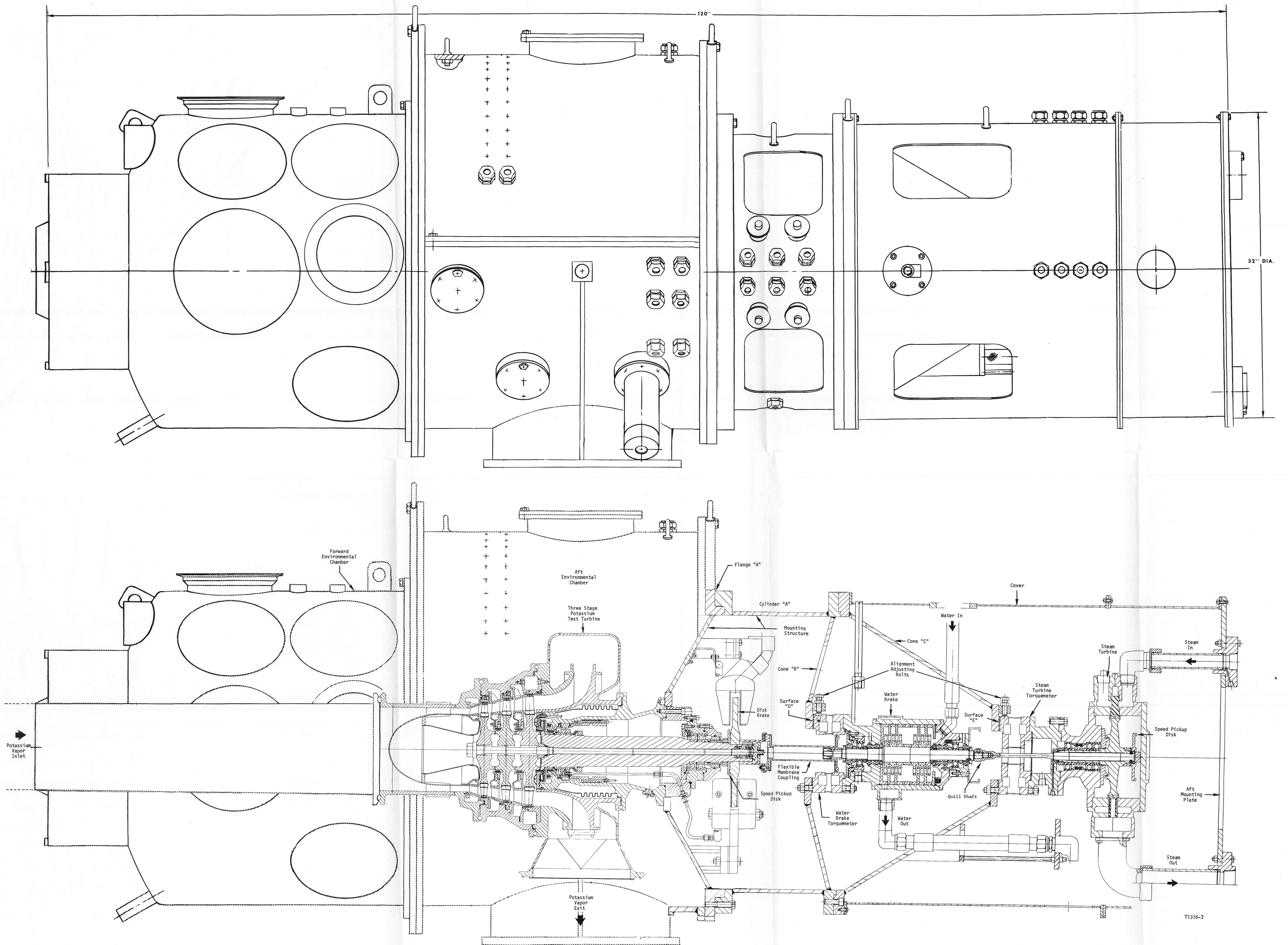


Figure 3. Test Rig Assembly. (External and Internal)

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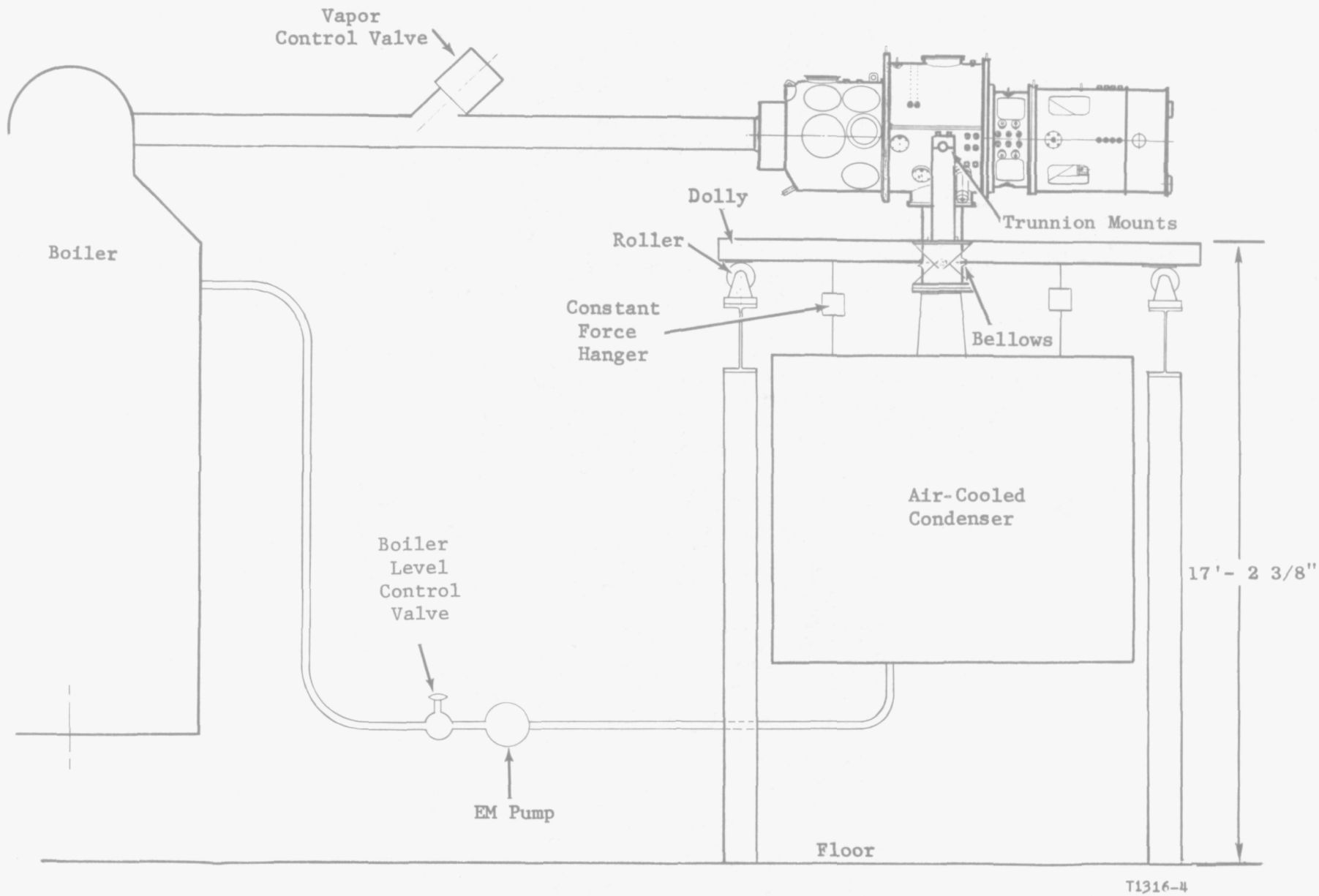


Figure 4. Test Rig Installation.

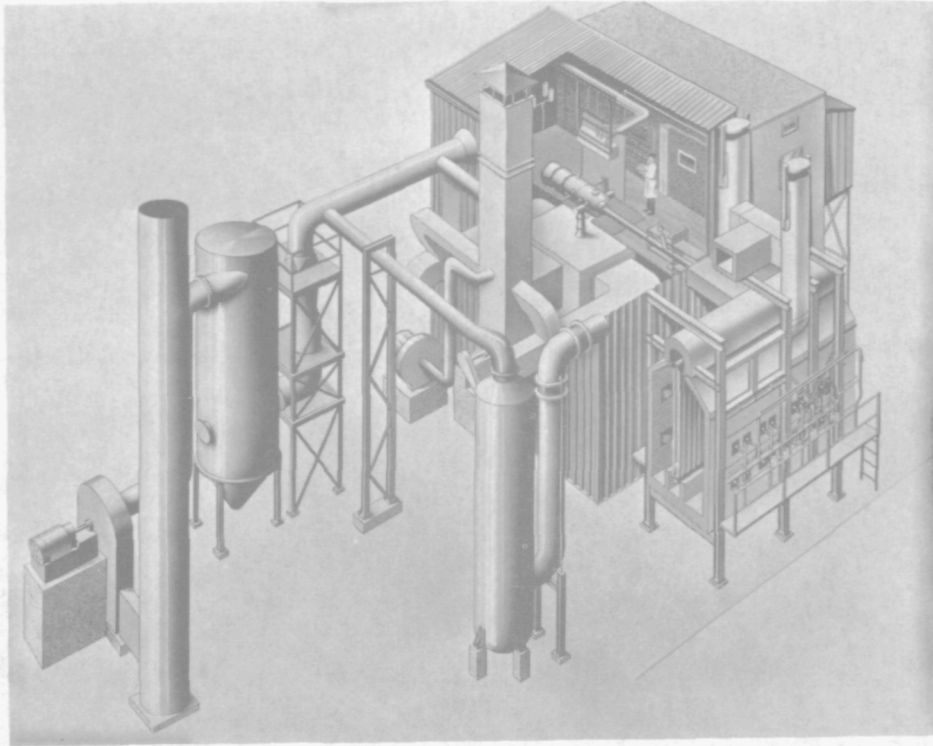


Figure 5a . Trimetric View of 3000 KW Potassium Turbine Test Facility.

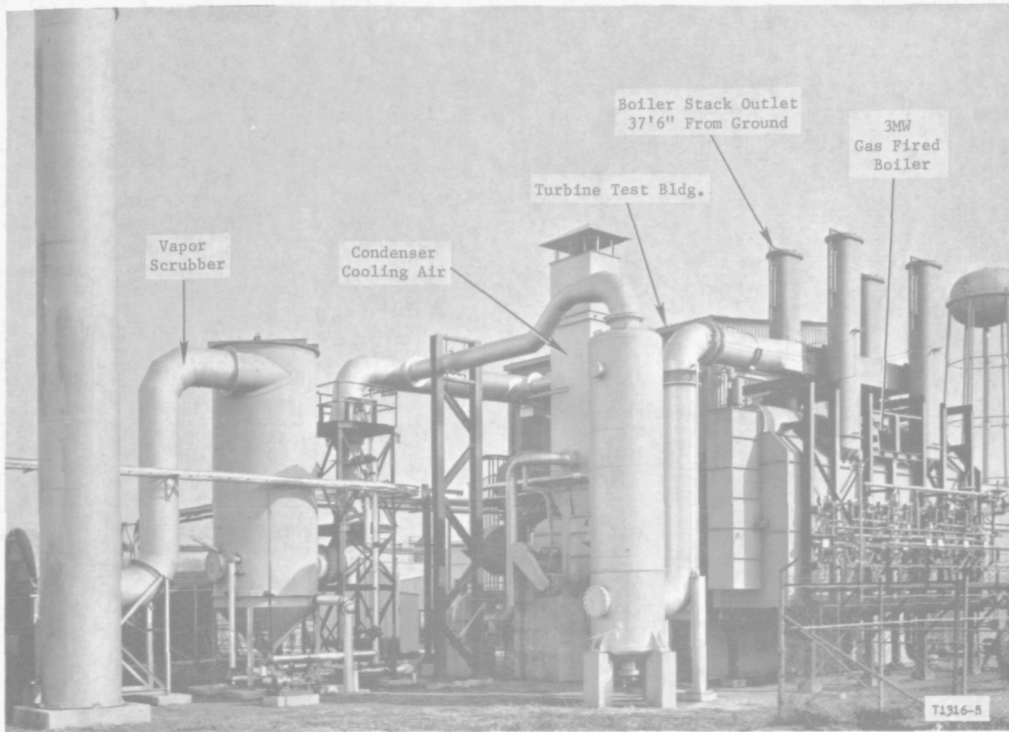
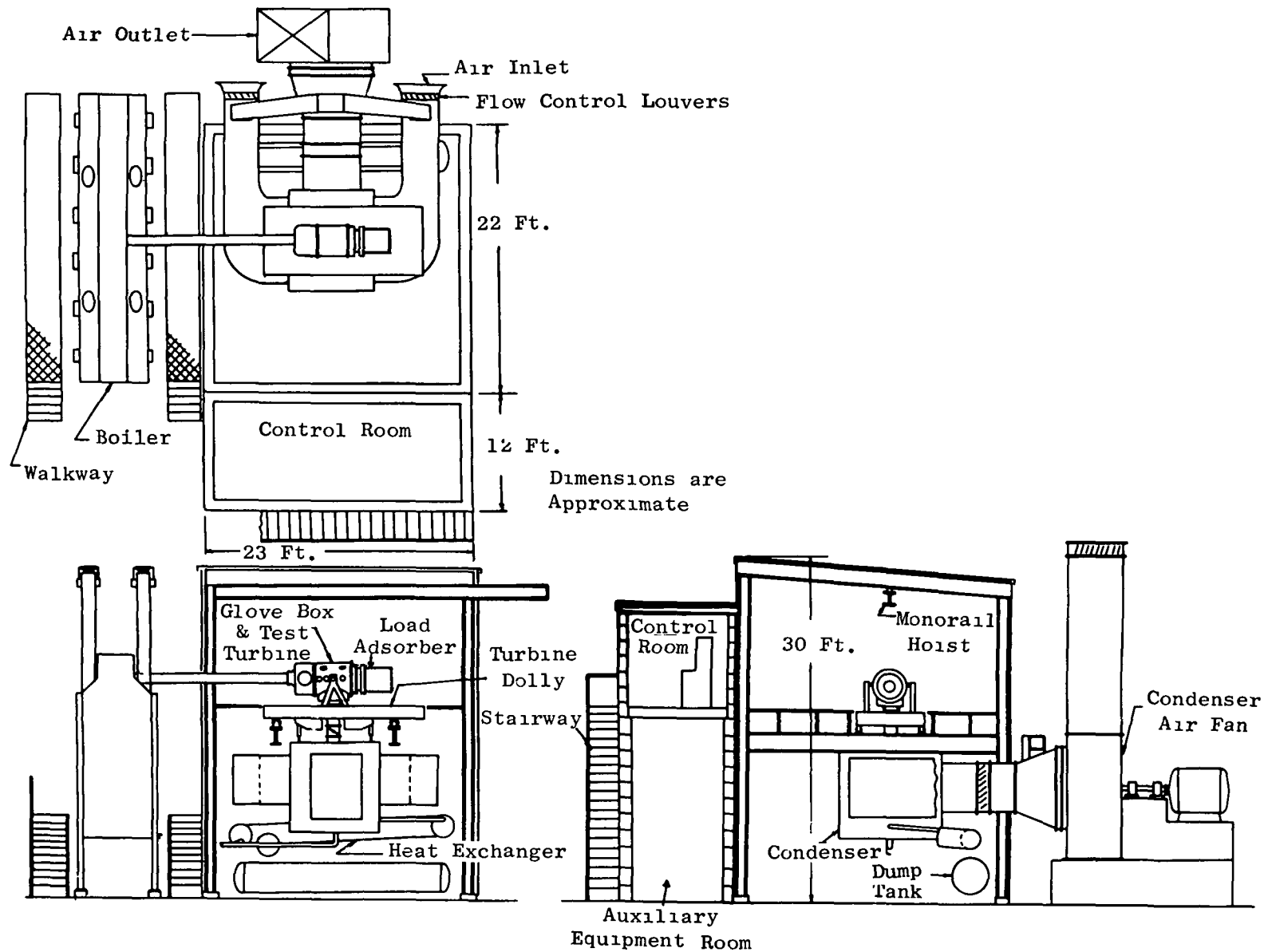


Figure 5b. External View of 3000 KW Potassium Turbine Test Facility.



T1316-6

Figure 5c. 3000 KW Potassium Turbine Test Facility. (Elevation)

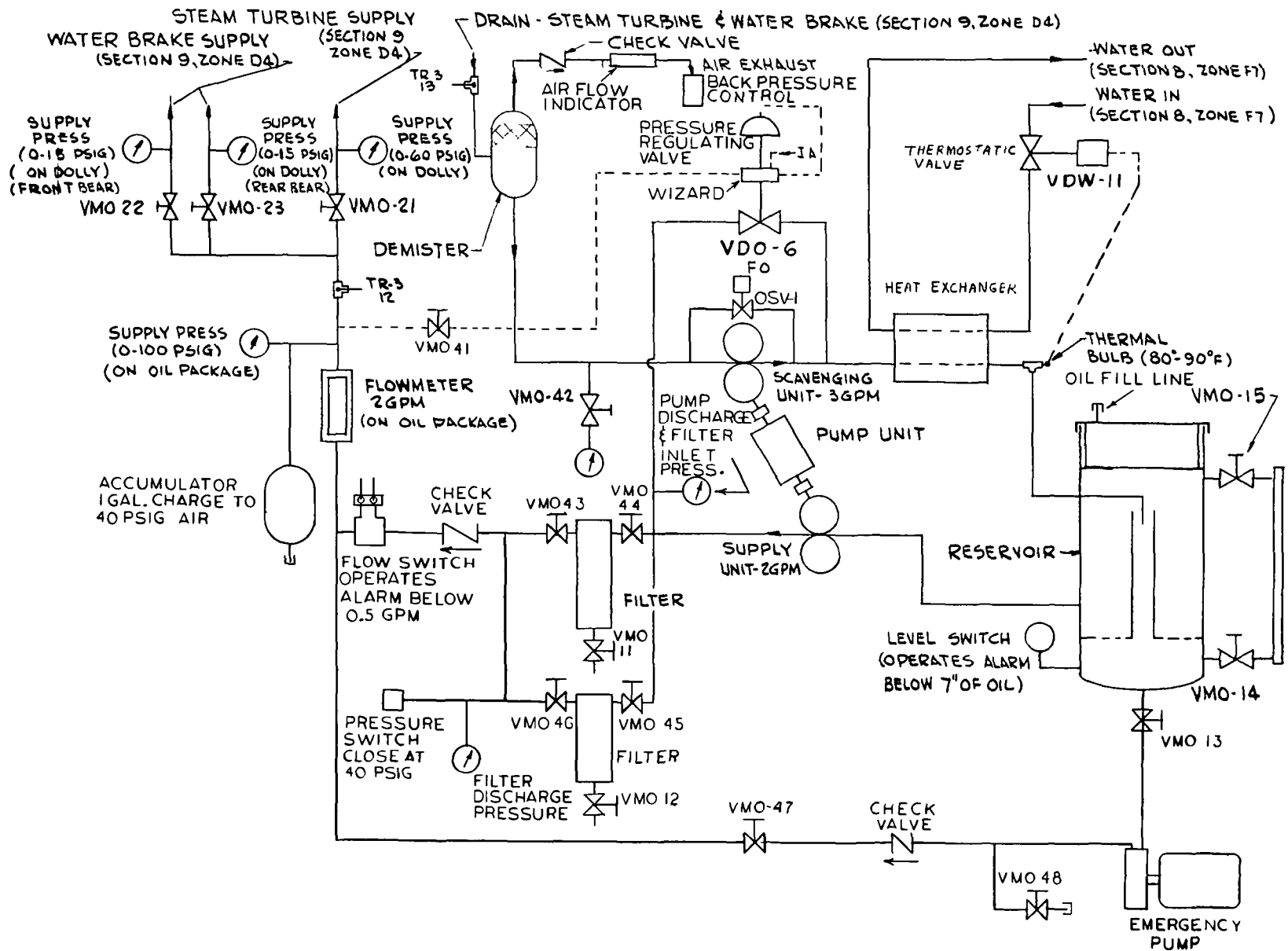


Figure 6. Water Brake and Steam Turbine Lubrication System Schematic.

T1316-7

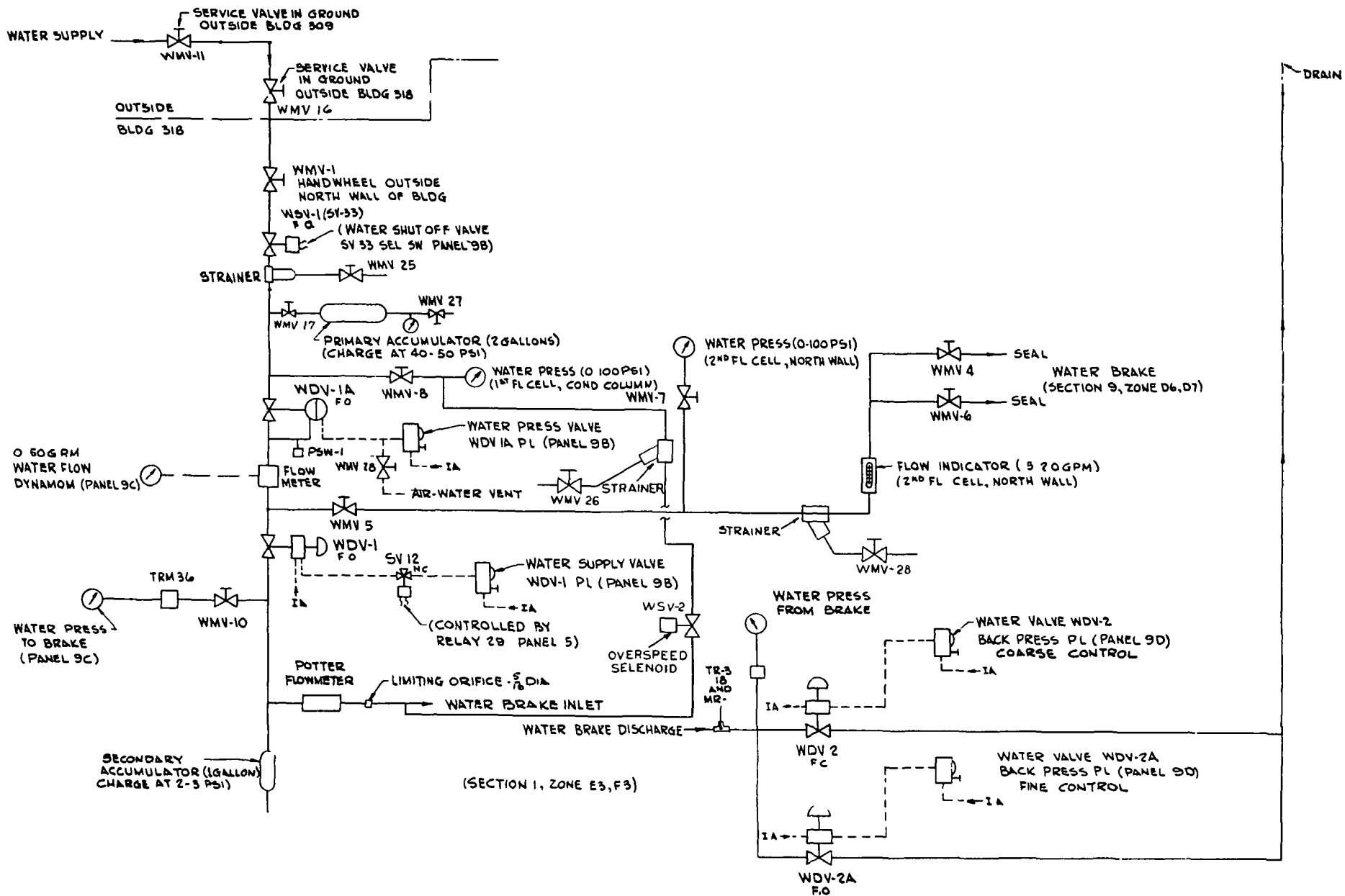


Figure 7. Water Brake Supply and Control System Schematic.

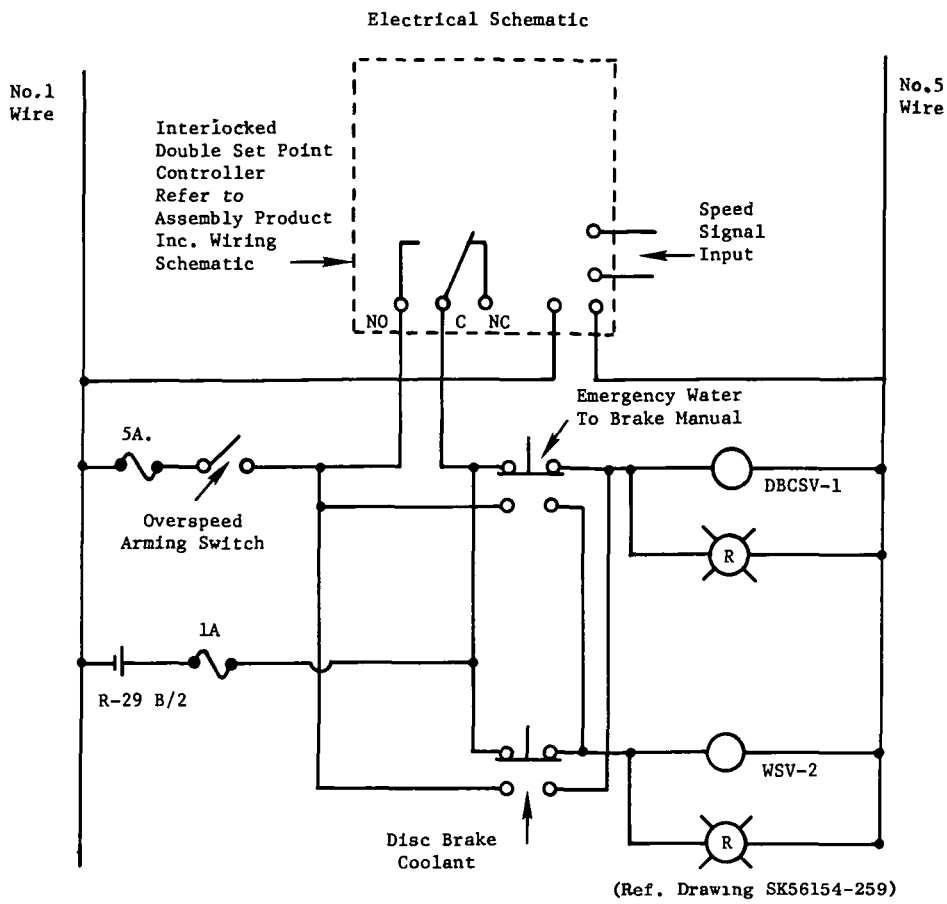


Figure 8a. Water Brake Overspeed Trip Electrical Schematic.

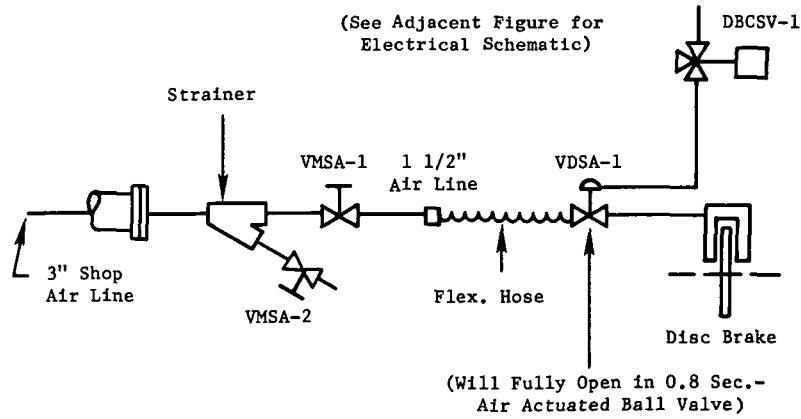


Figure 8b. Disk Brake Cooling Air Flow Hook-Up.

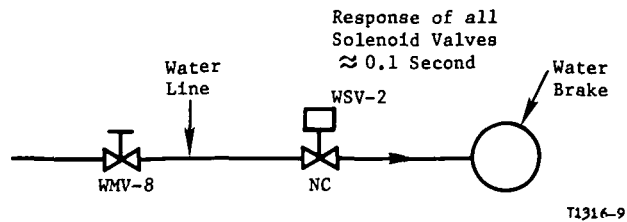
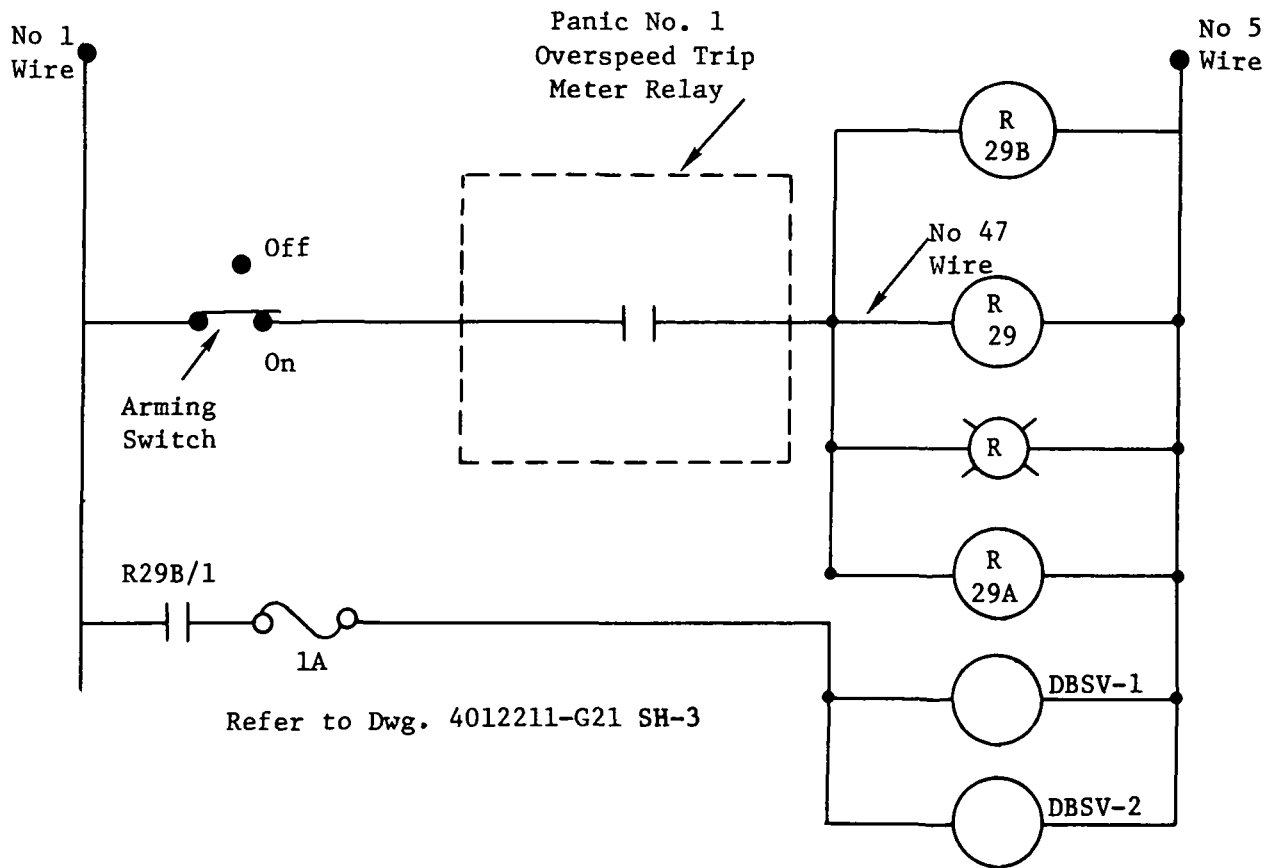


Figure 8c. Water Brake Emergency Flow Hook-Up.



Refer to Dwg. 4012211-G21 SH-3

Figure 9a. Overspeed Trip - Panic No. 1.

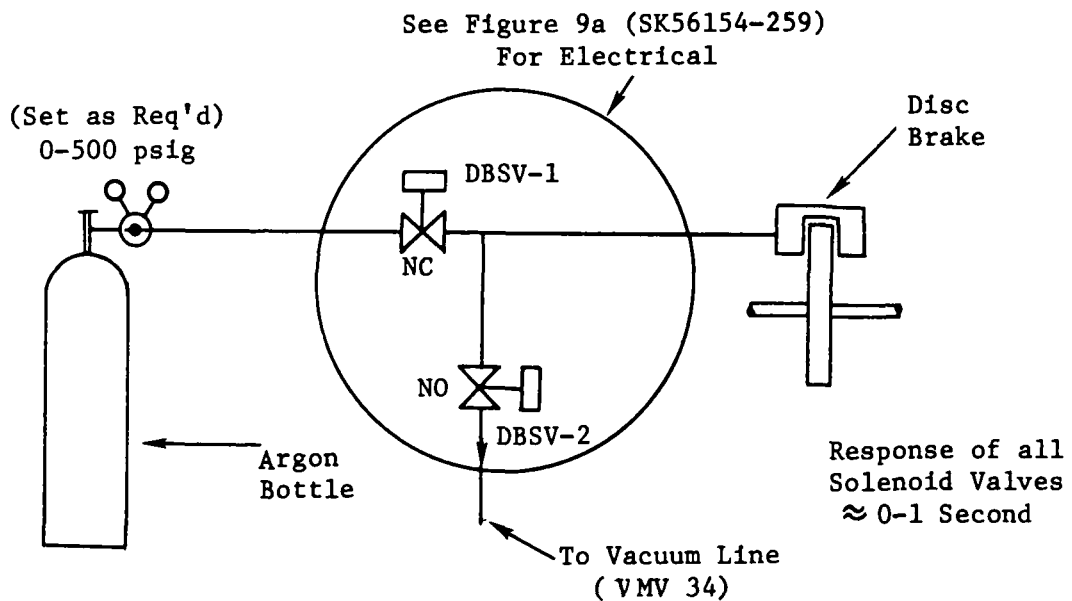


Figure 9b. Disk Brake Puck Actuation Flow Hook-up.

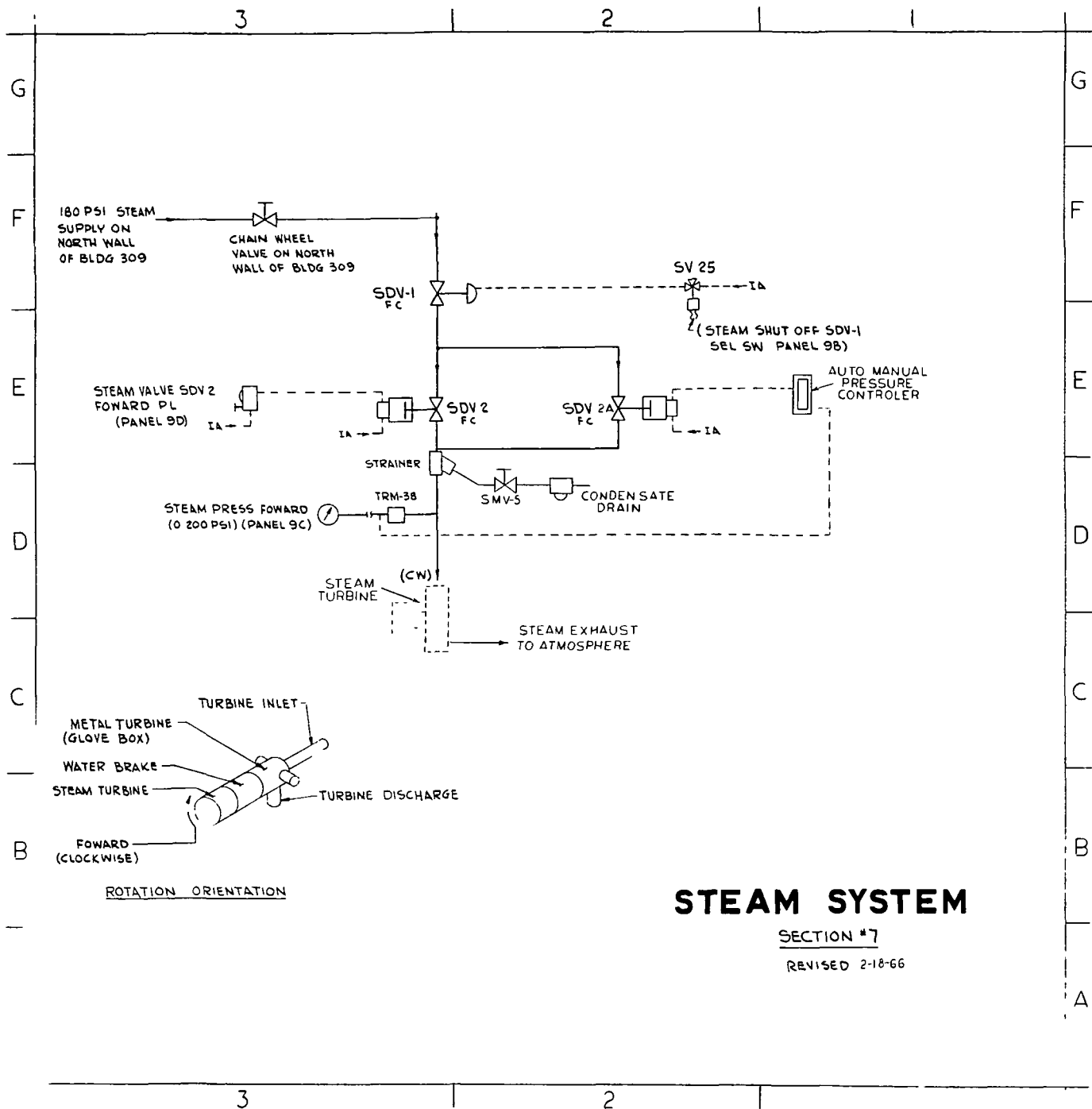
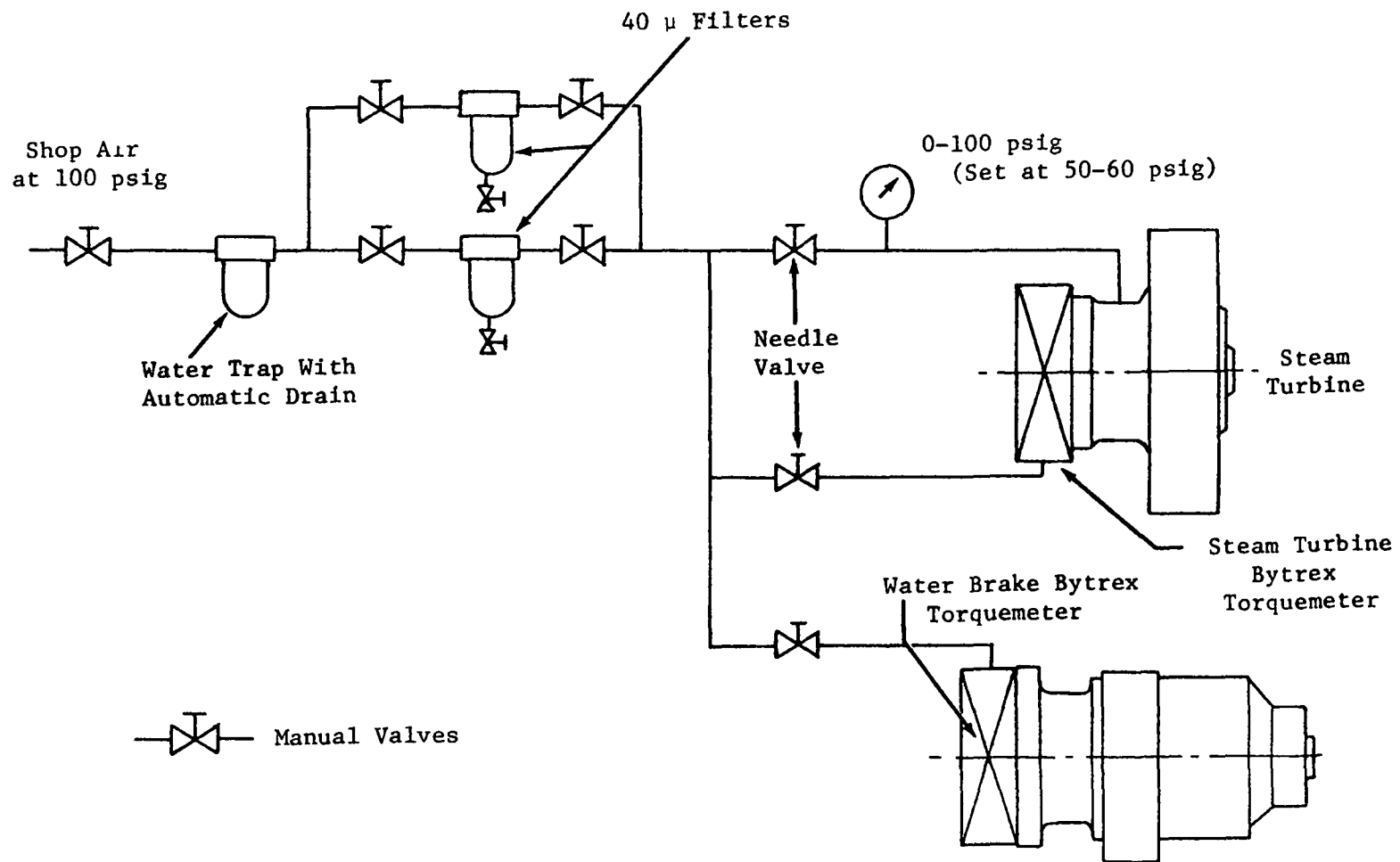
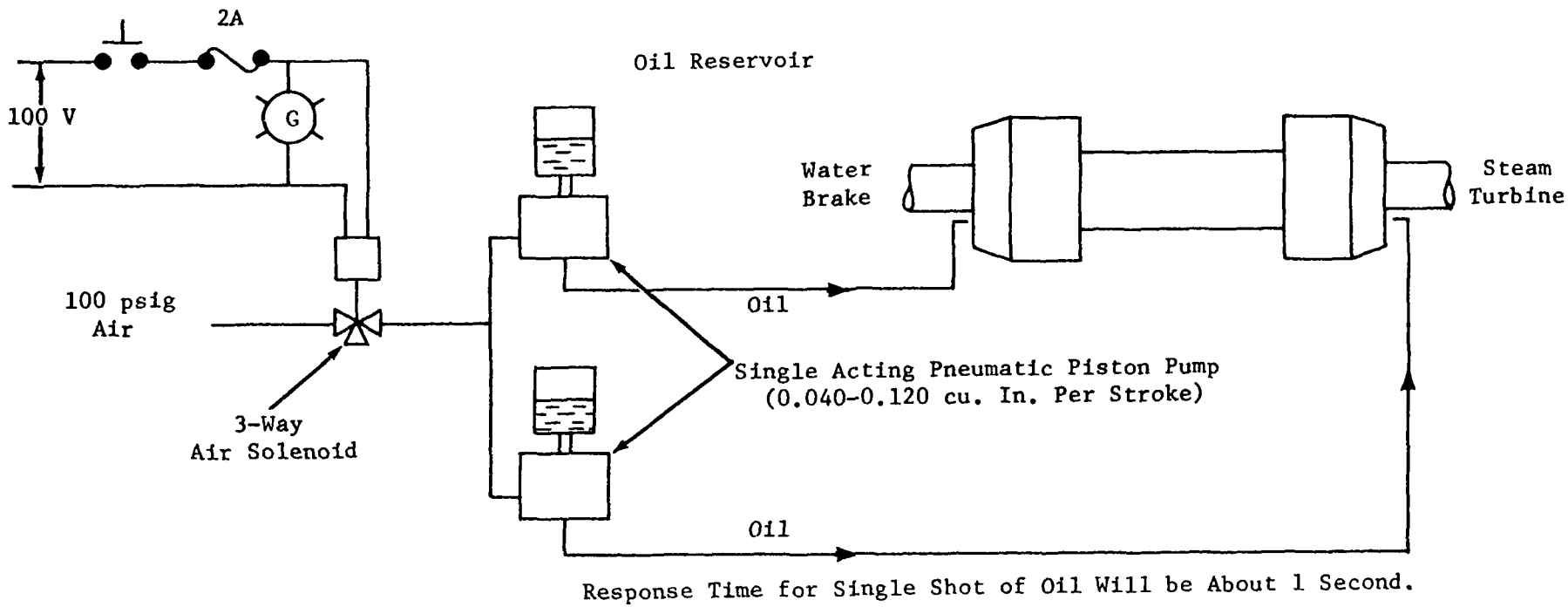


Figure 10. Steam Turbine Supply and Control System Schematic.



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Figure 11. Air to Steam Turbine Labyrinth Seal and Bytrex Torquemeters.



Lubricant: Oil, SAE 140

Figure 12. Lube System for Gear (Koppers) Coupling. (Alternate)

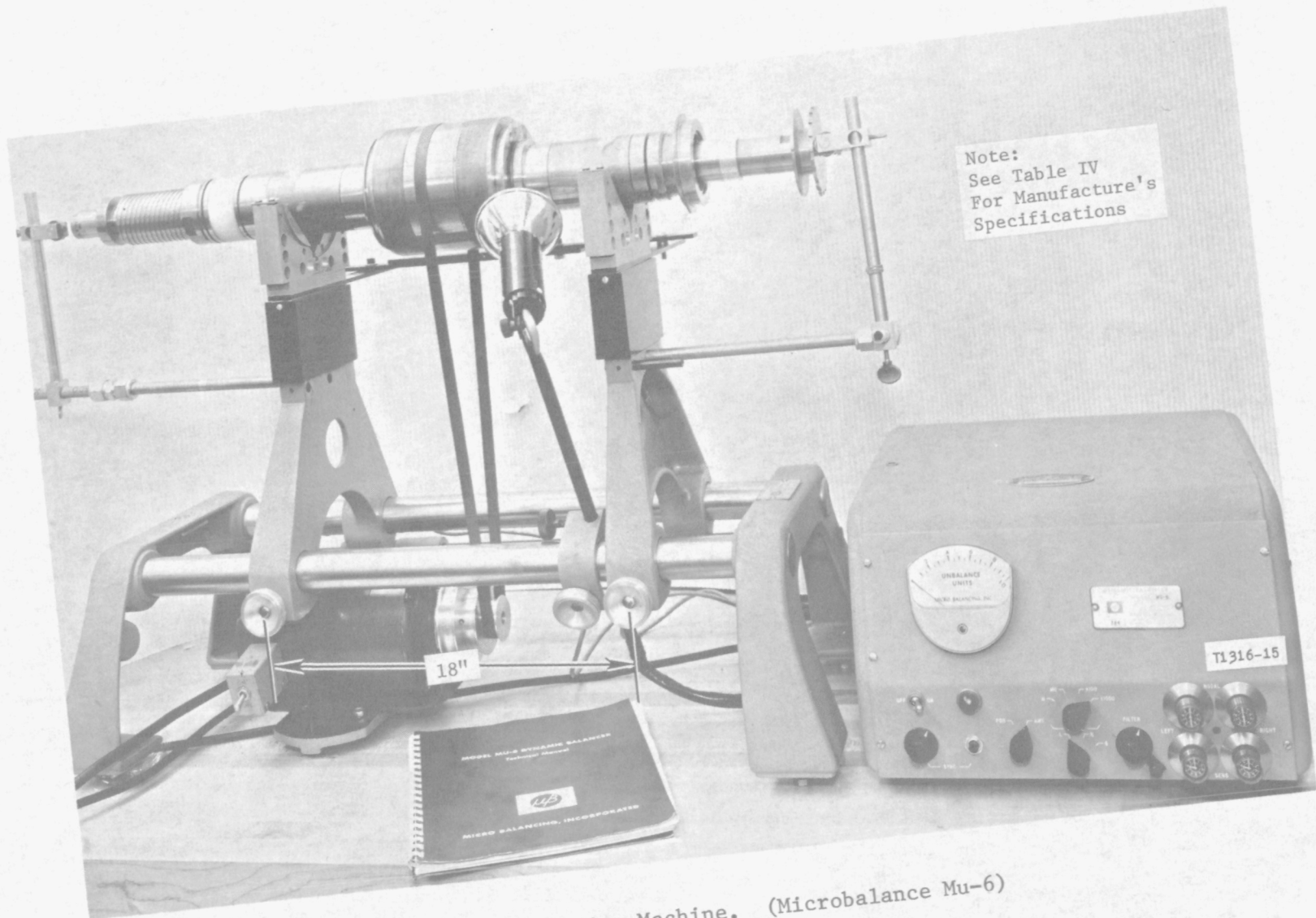


Figure 14. Rotor Balancing Machine. (Microbalance Mu-6)

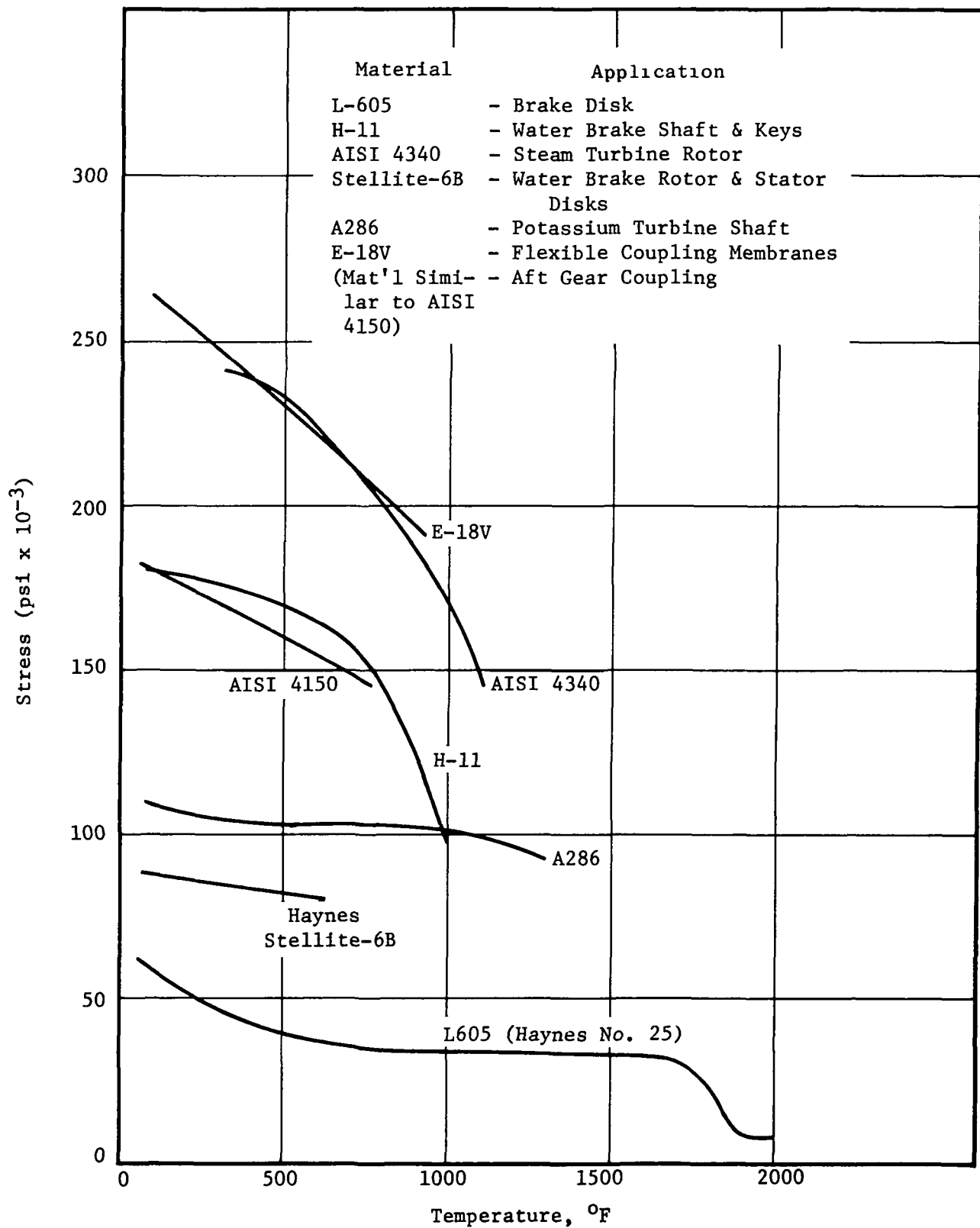


Figure 15. 0.2 Per Cent Yield Strength for the Major Alloys in the Power Train (Average Values).

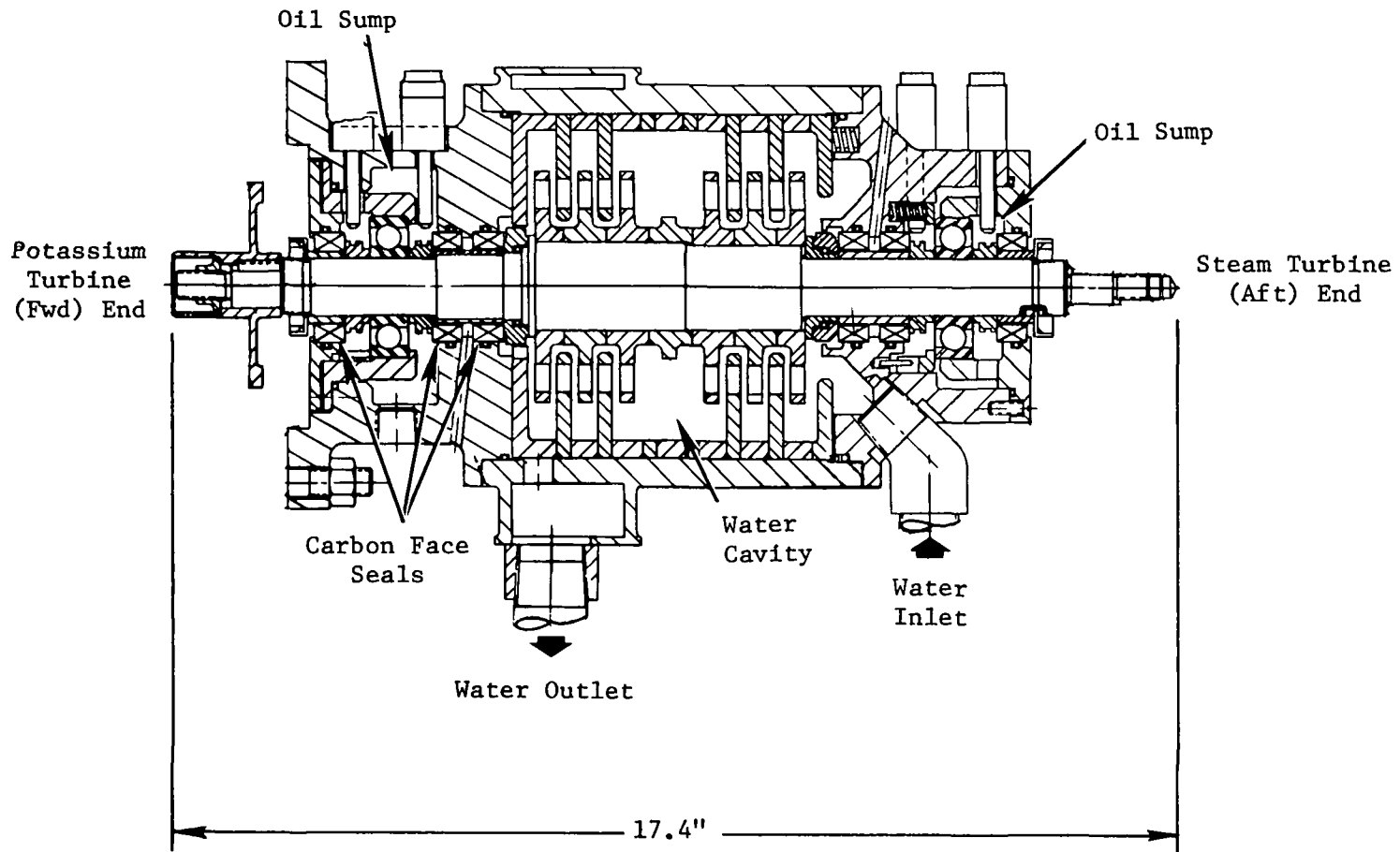
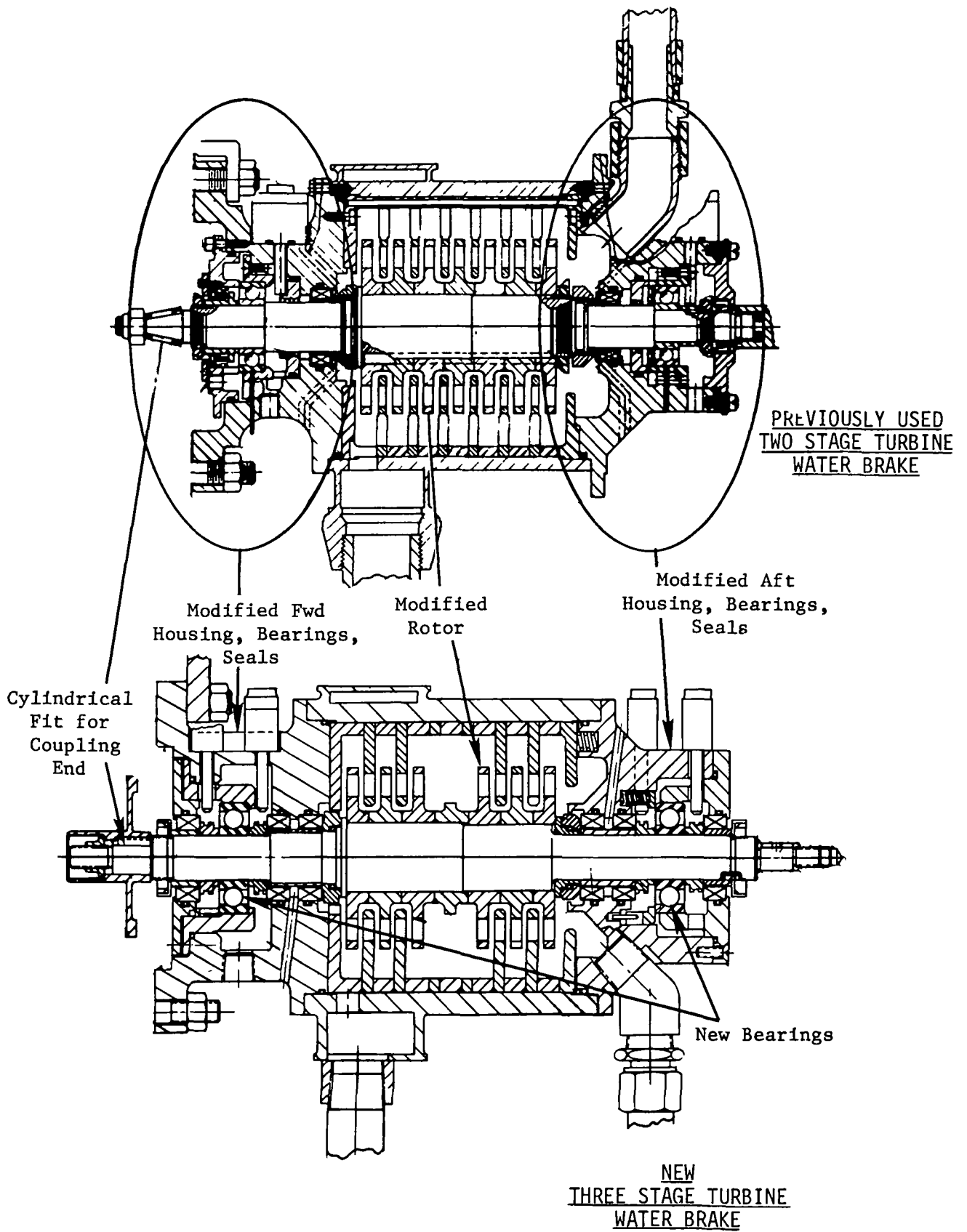


Figure 16. Water Brake Assembly.



T1316-18

Figure 17. Modifications to Water Brake Design.

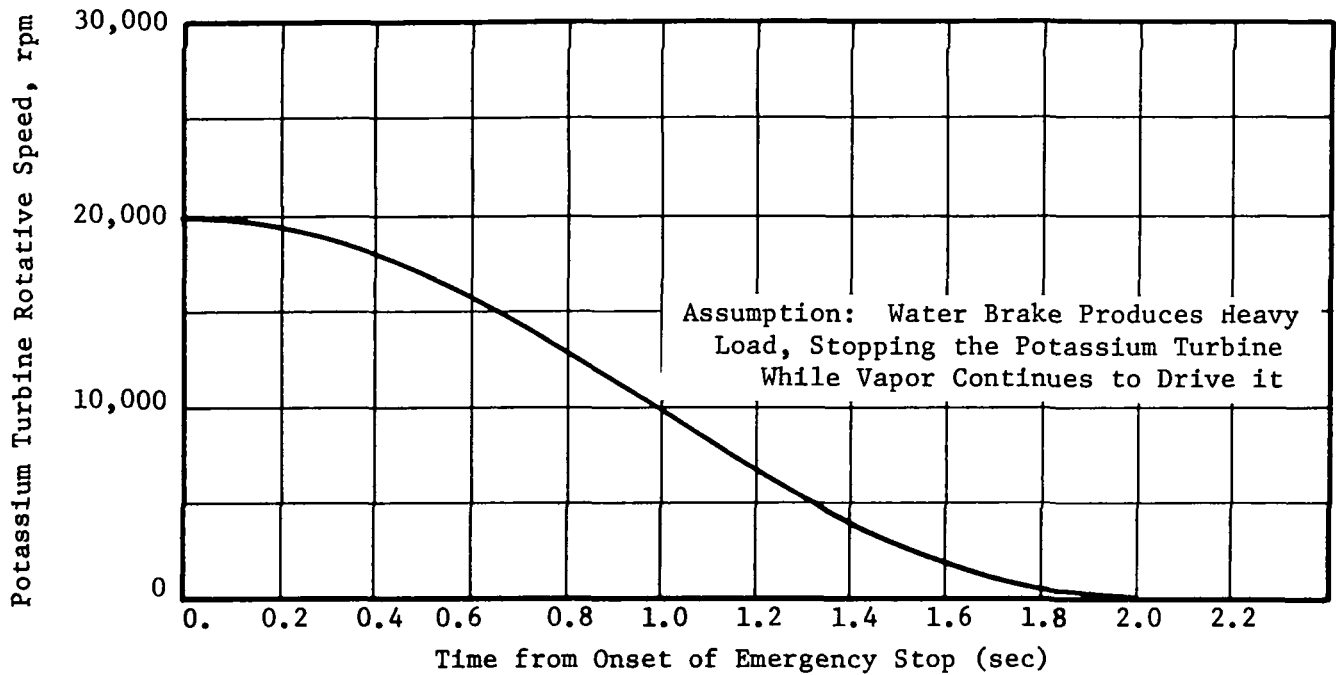


Figure 18a. Rotative Speed of Potassium Turbine and Power Train Rotors During Emergency Stop.

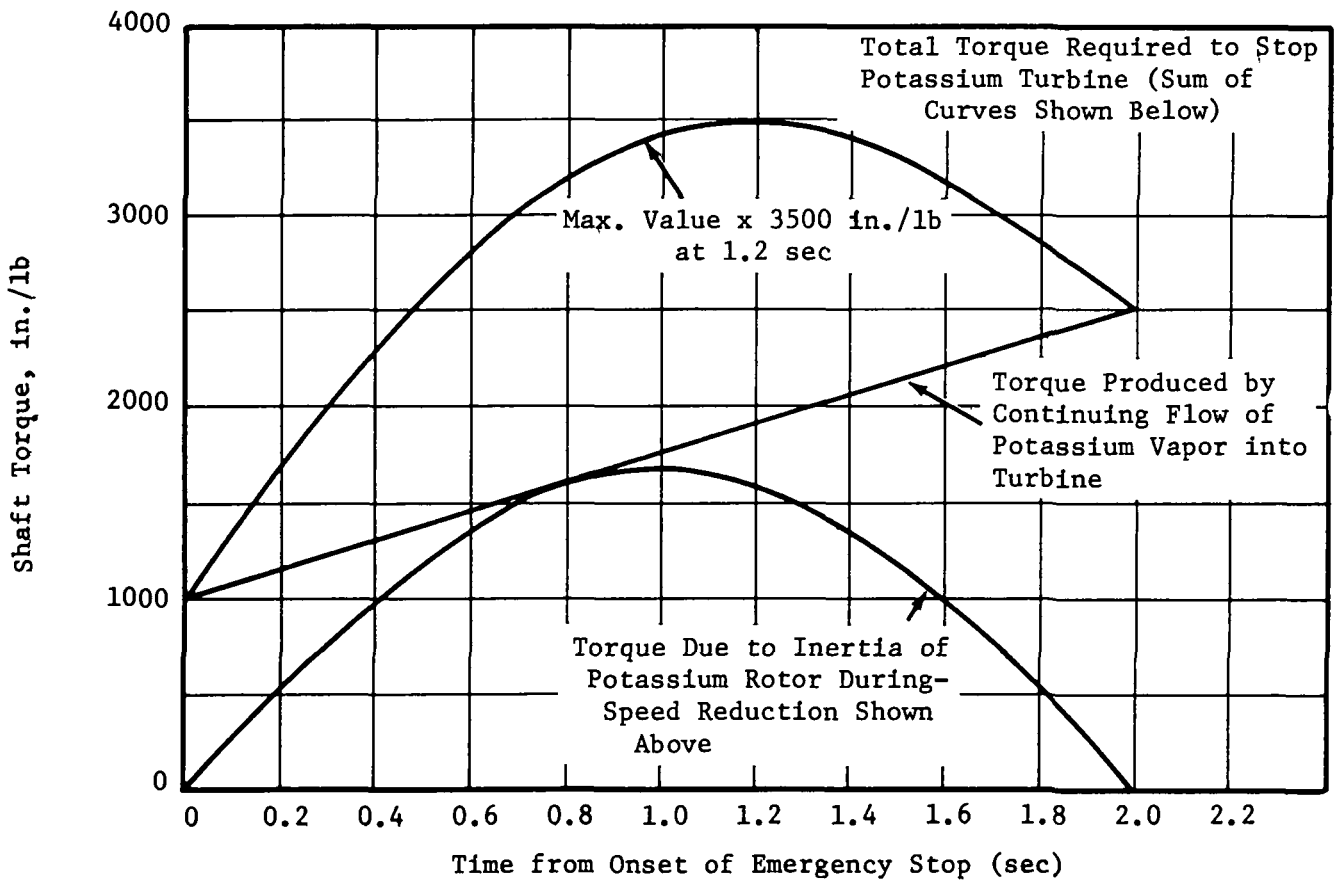


Figure 18b. Torque Produced in Coupling Between Potassium Turbine and Water Brake During Emergency Stop Shown Above.

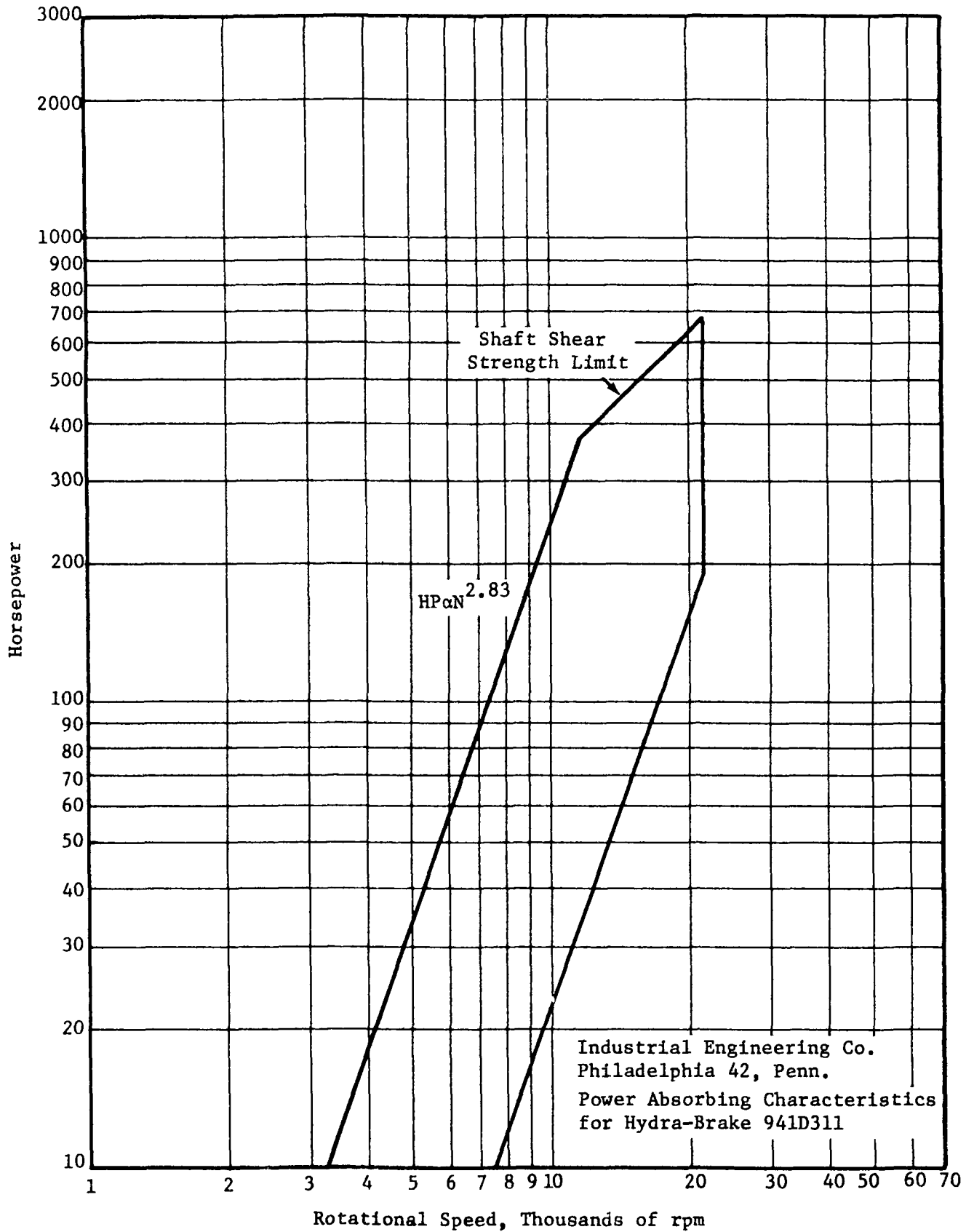


Figure 19a. Water Brake Horsepower Absorbing Capacity as a Function of Rotative Speed.

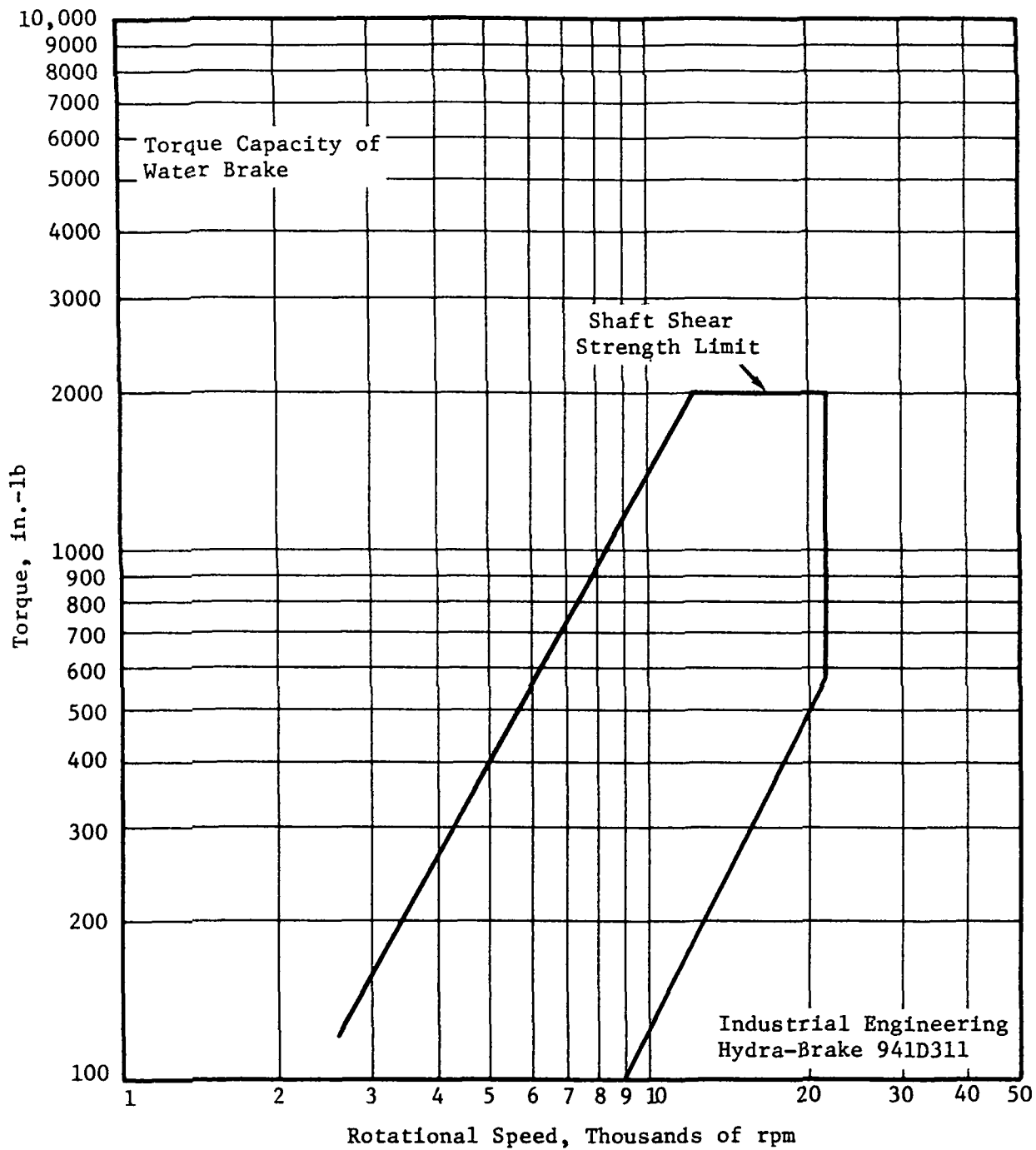


Figure 19b. Water Brake Maximum Torque Capacity as a Function of Rotative Speed.

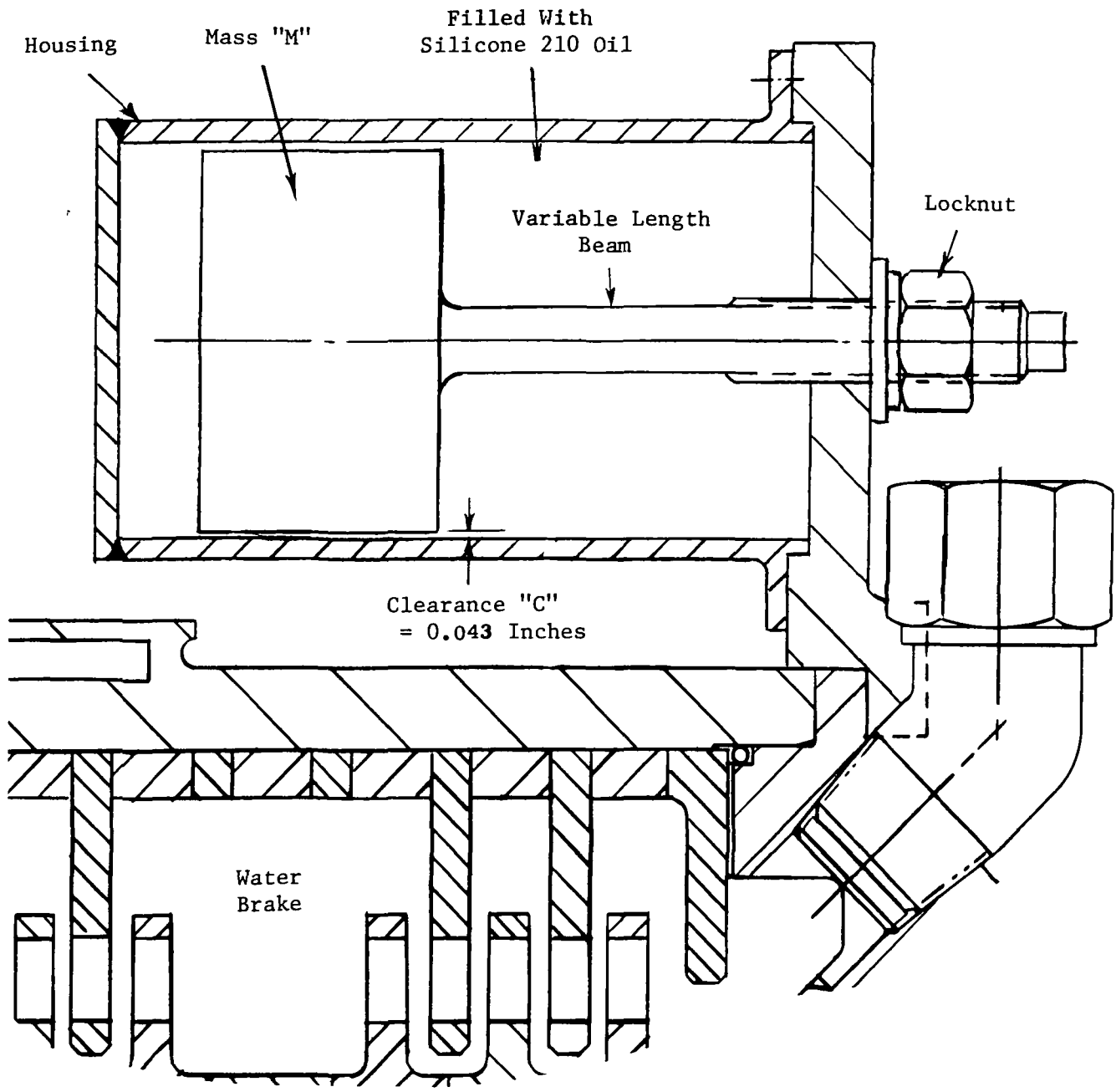


Figure 20. Water Brake Vibration Damper.

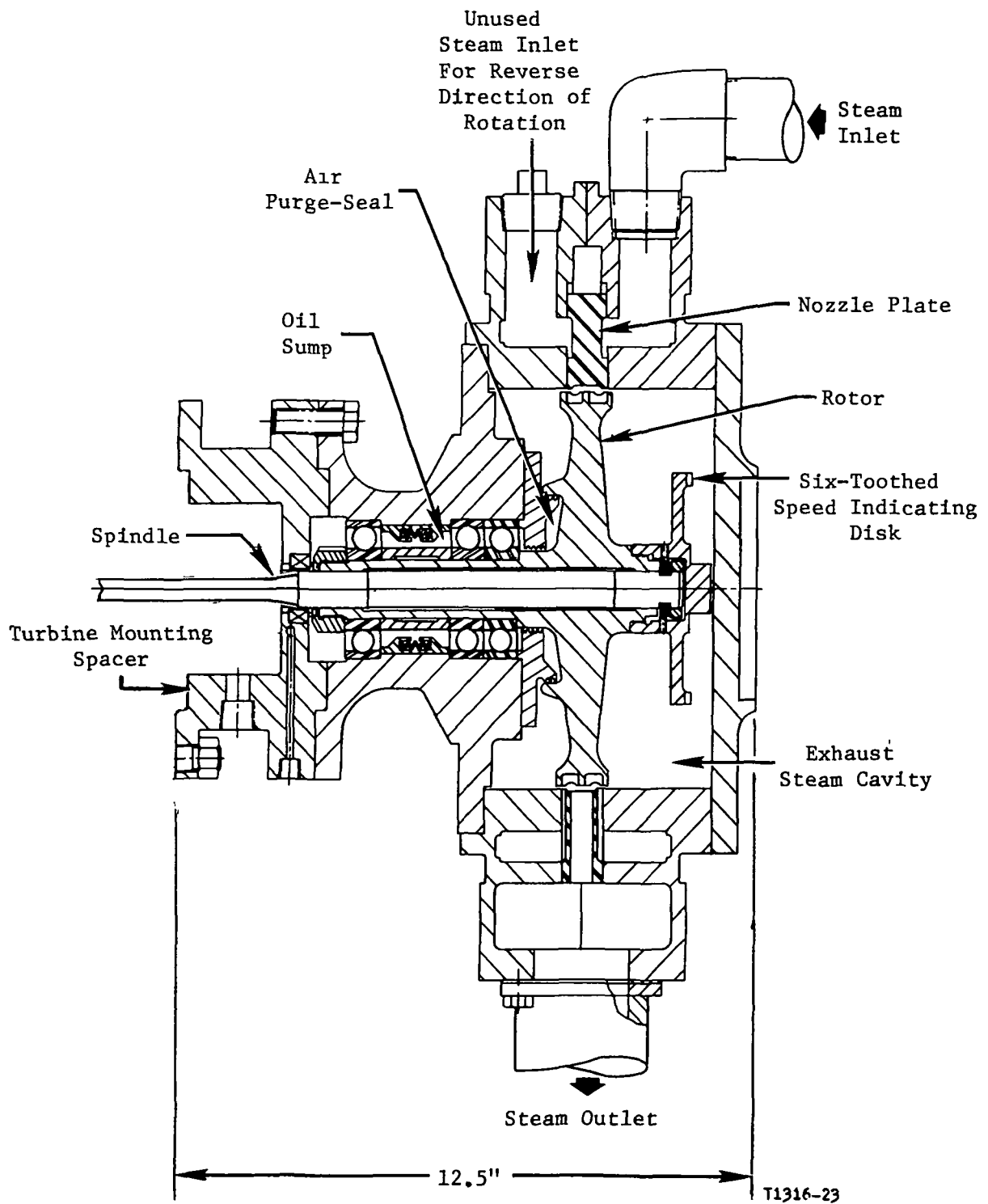


Figure 21. Steam Turbine Assembly.

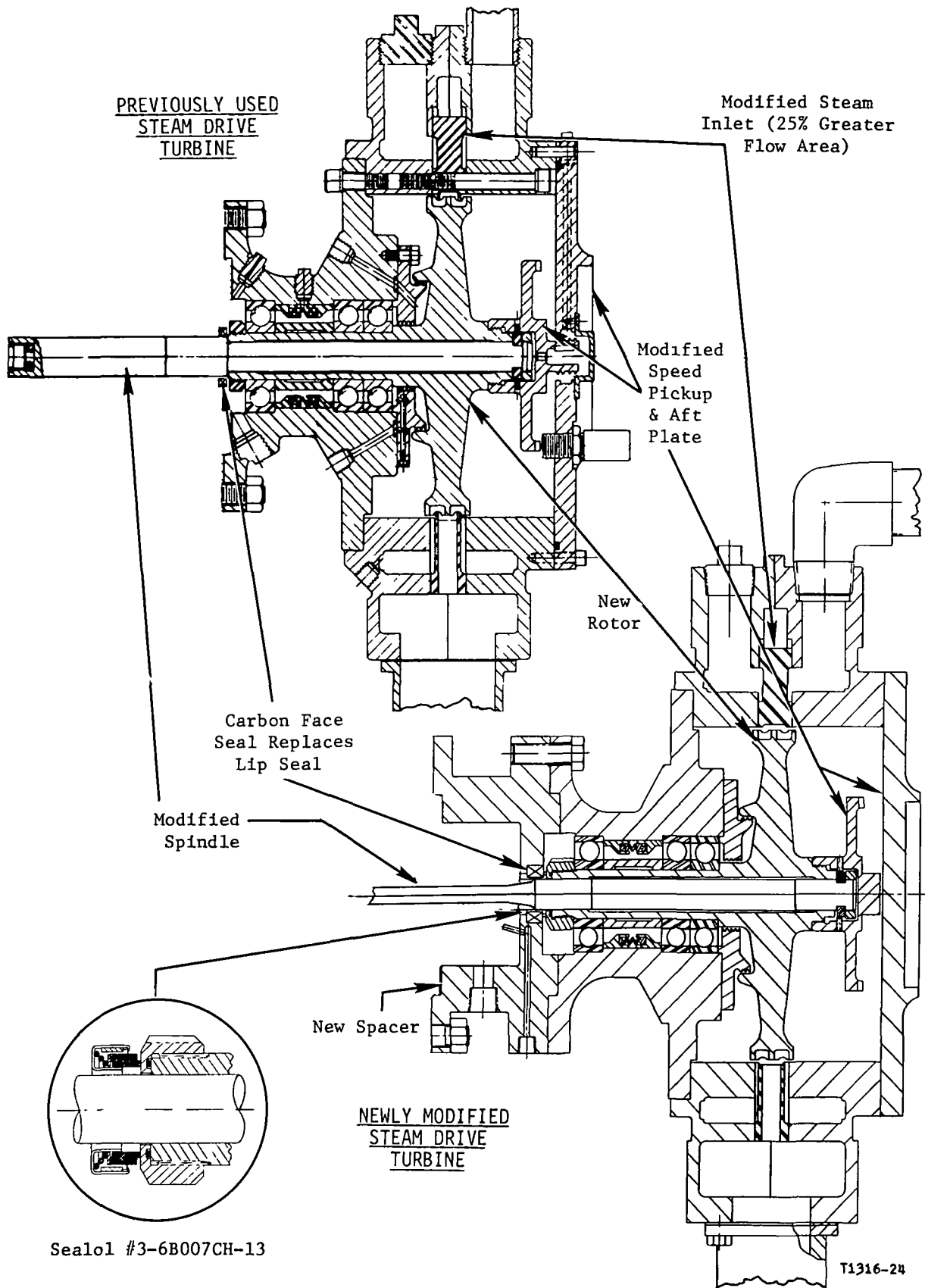


Figure 22. Modifications To Steam Drive Turbine.

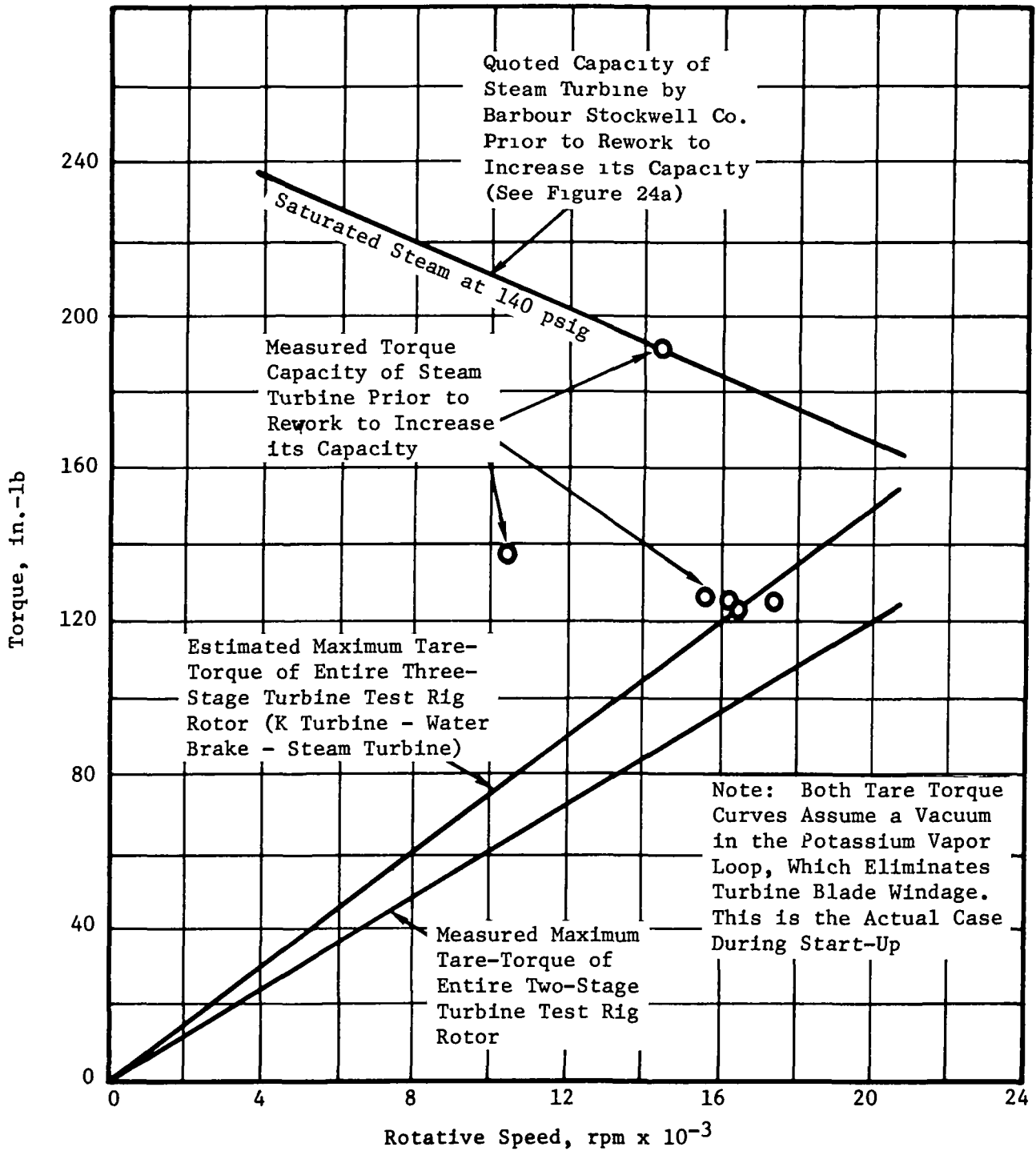


Figure 23. Steam Turbine Requirements for Tare Testing of Previous Two-Stage and Present Three-Stage Turbine and Capacity of Steam Turbine Prior to the Rework to Increase its Capacity.

150 psig Steam
3200 Lbs. Per Hr.

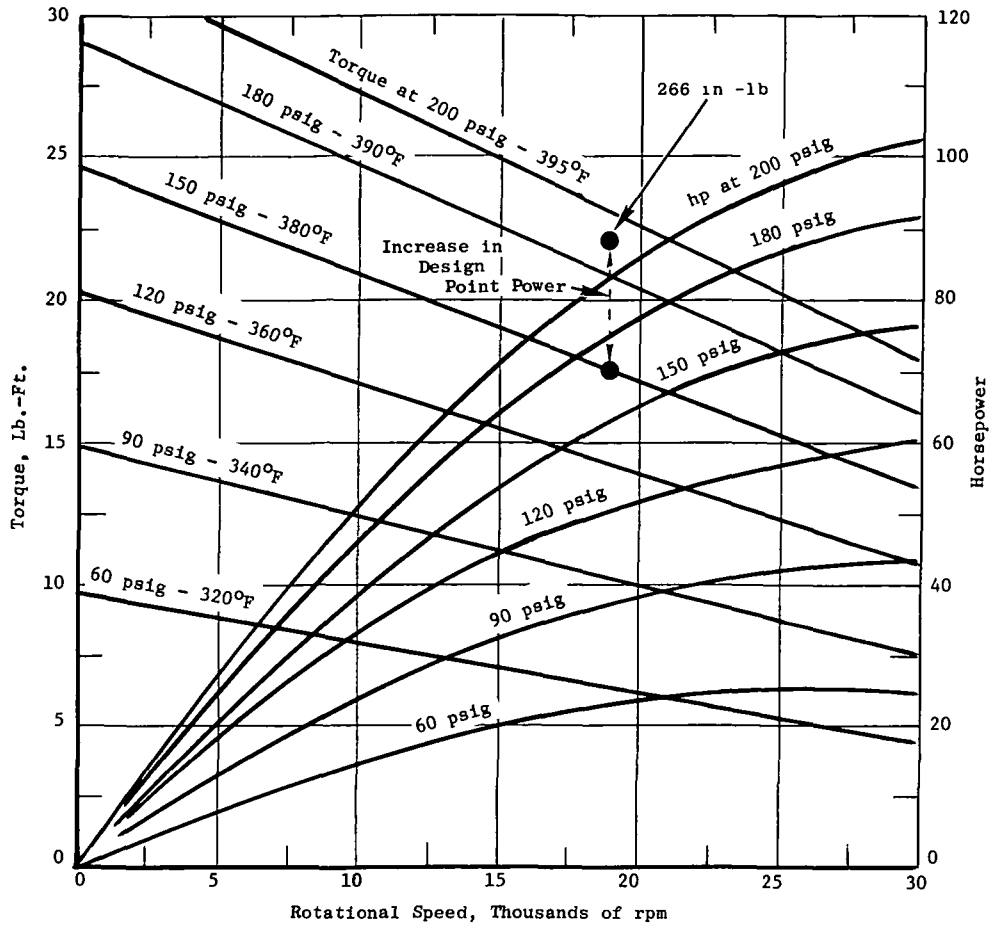


Figure 24a. Manufacturer's Performance Curve for Steam Turbine (Prior to Rework) and Estimated Increase in Design - Point Power.

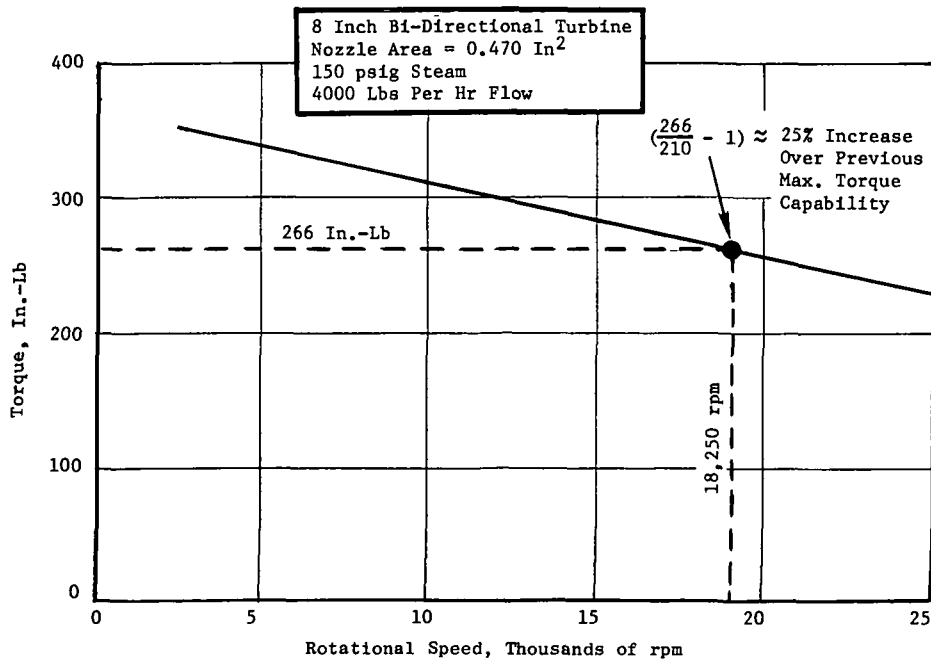


Figure 24b. Manufacturer's Estimated Maximum Torque Output of Steam Turbine Following Rework to Increase Nozzle Area by 25 Per Cent.

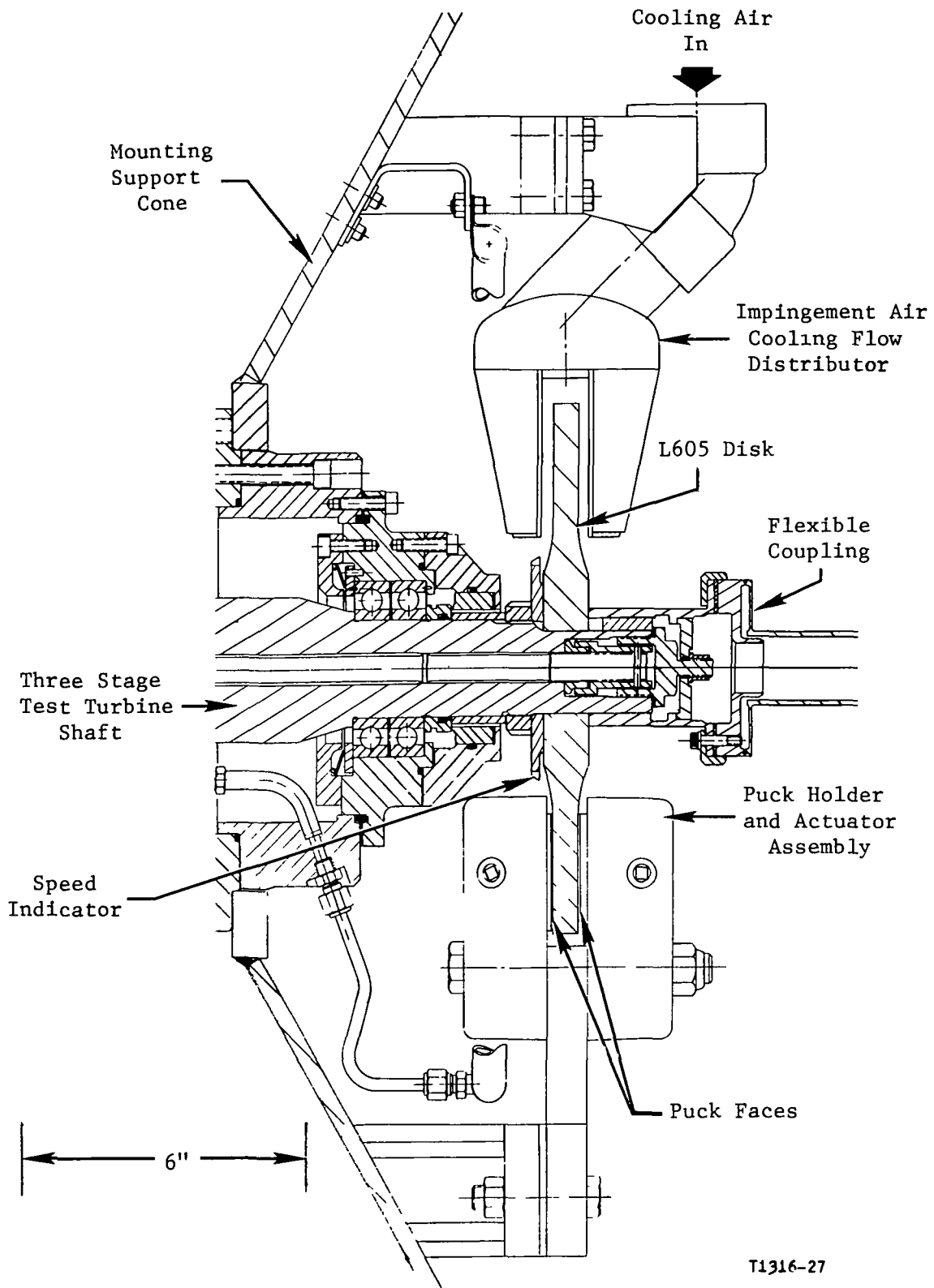


Figure 25. Disk Brake Assembly.

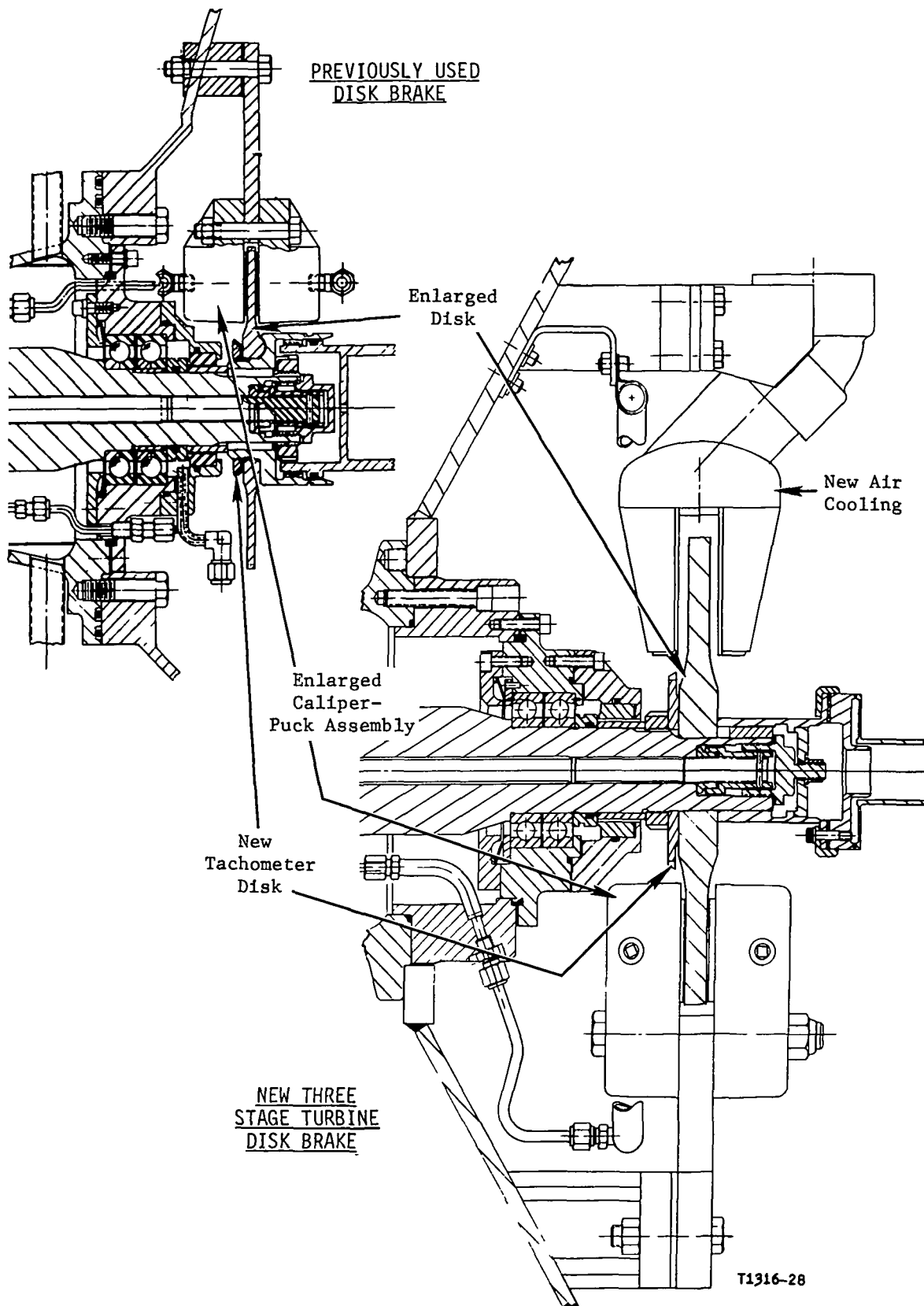


Figure 26. Modification to Disk Brake Design.

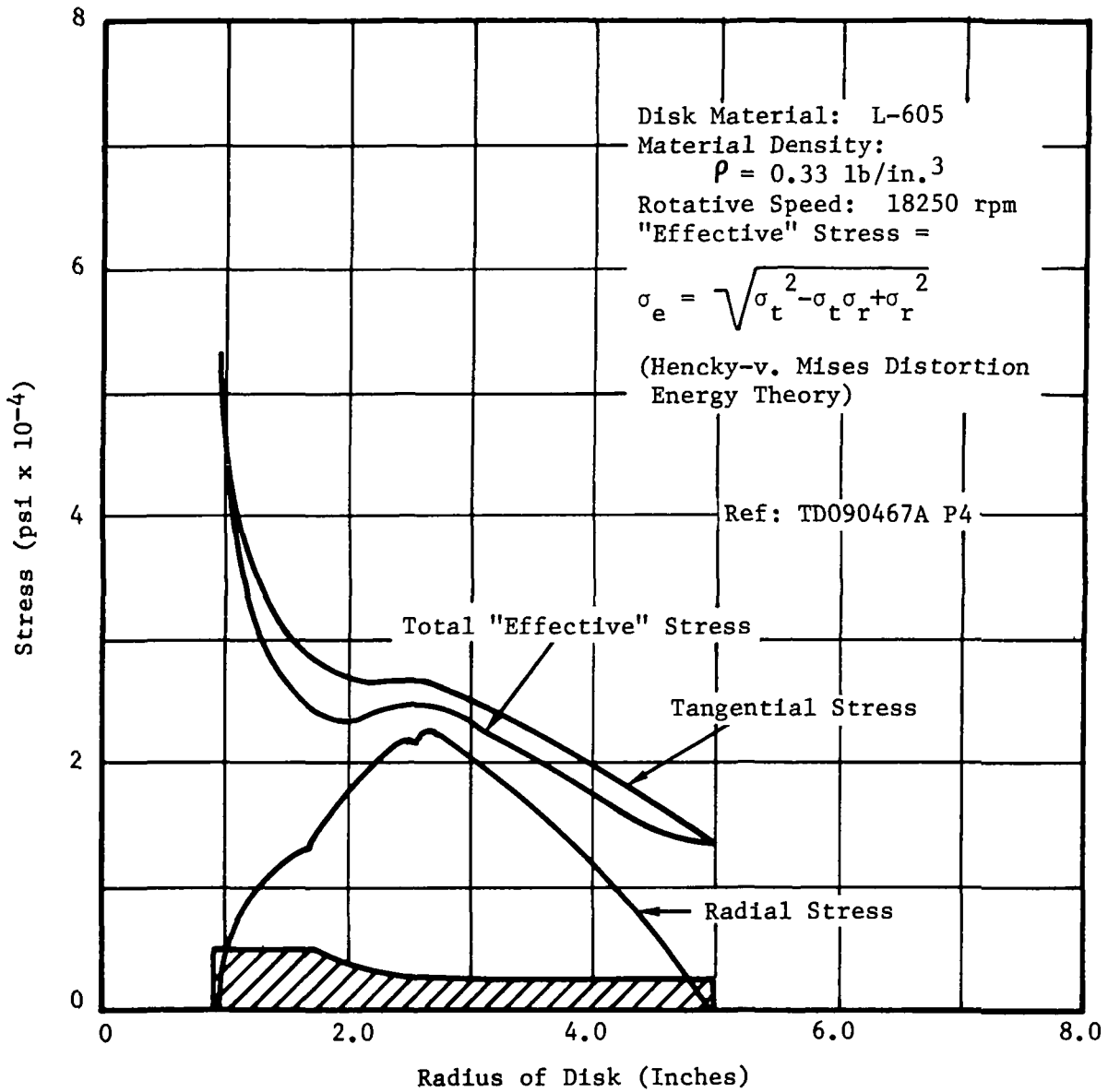


Figure 27a. Rotative Stresses in Disk Brake at Design Point Operation as a Function of Disk Radius.

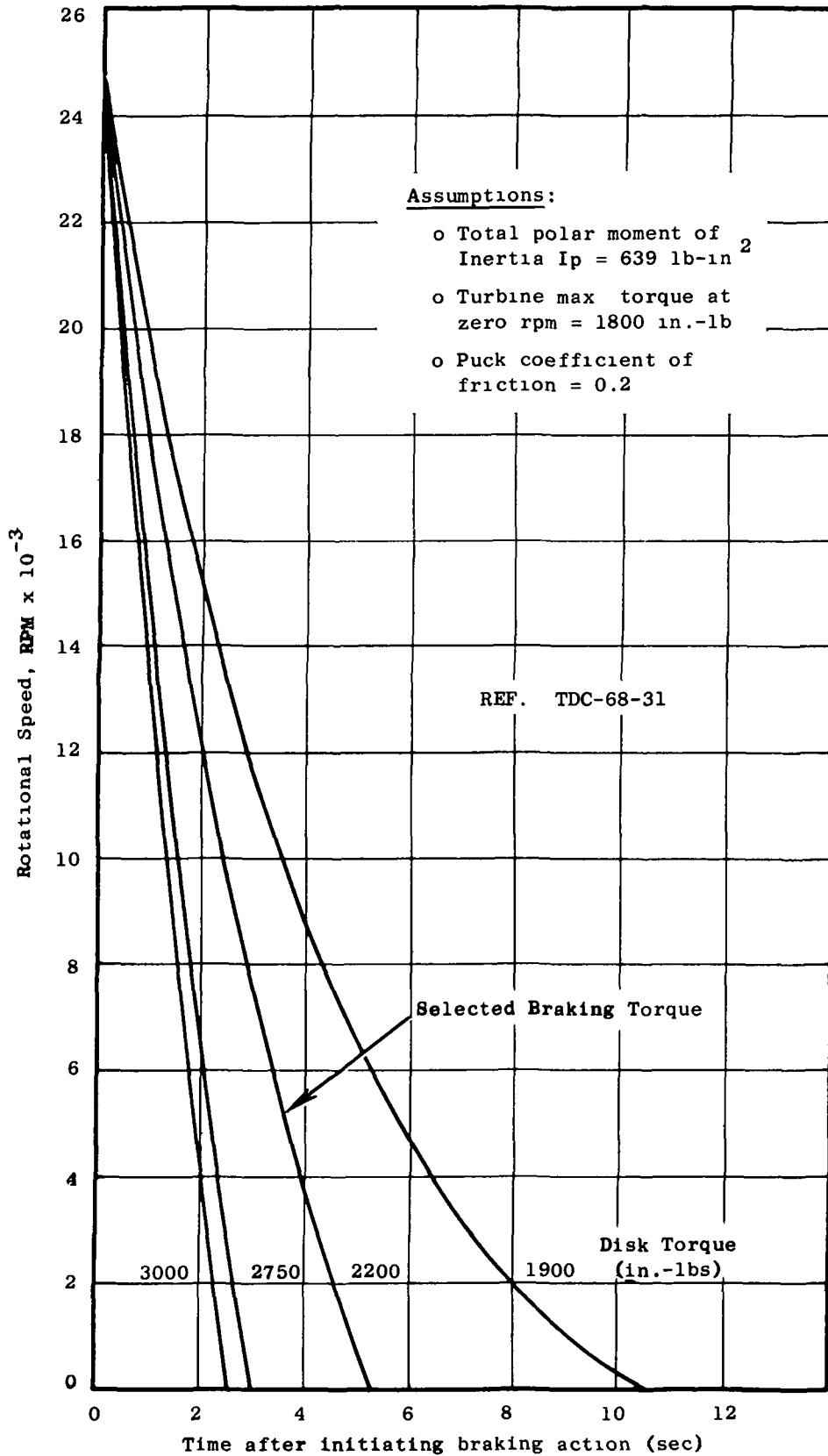


Figure 27(b) Deceleration of Turbine Power Train as a Function of Applied Braking Torque to the Disk Brake

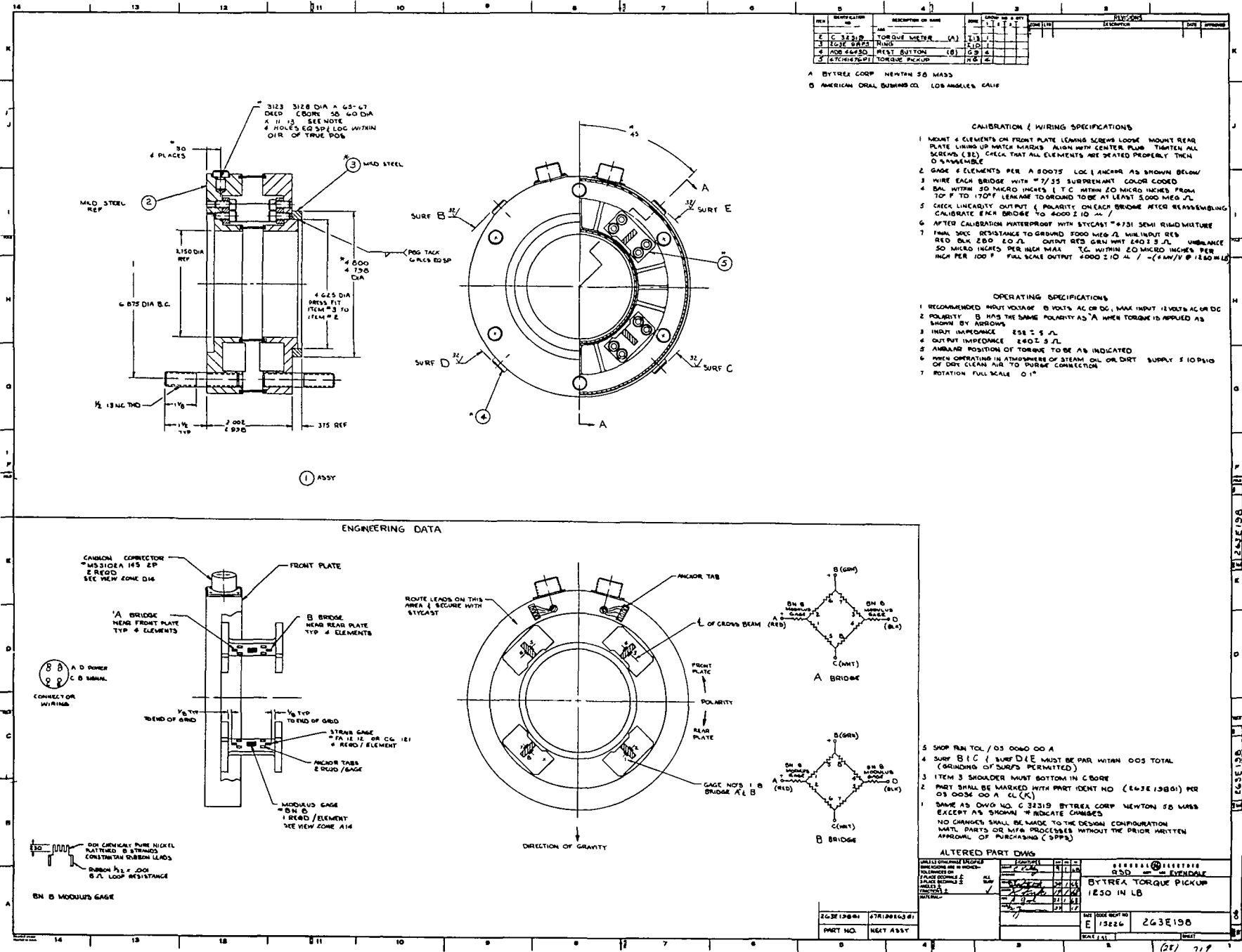


Figure 29. 2500 In.-Lb. Bytrex Torquemeter. (Water Brake)

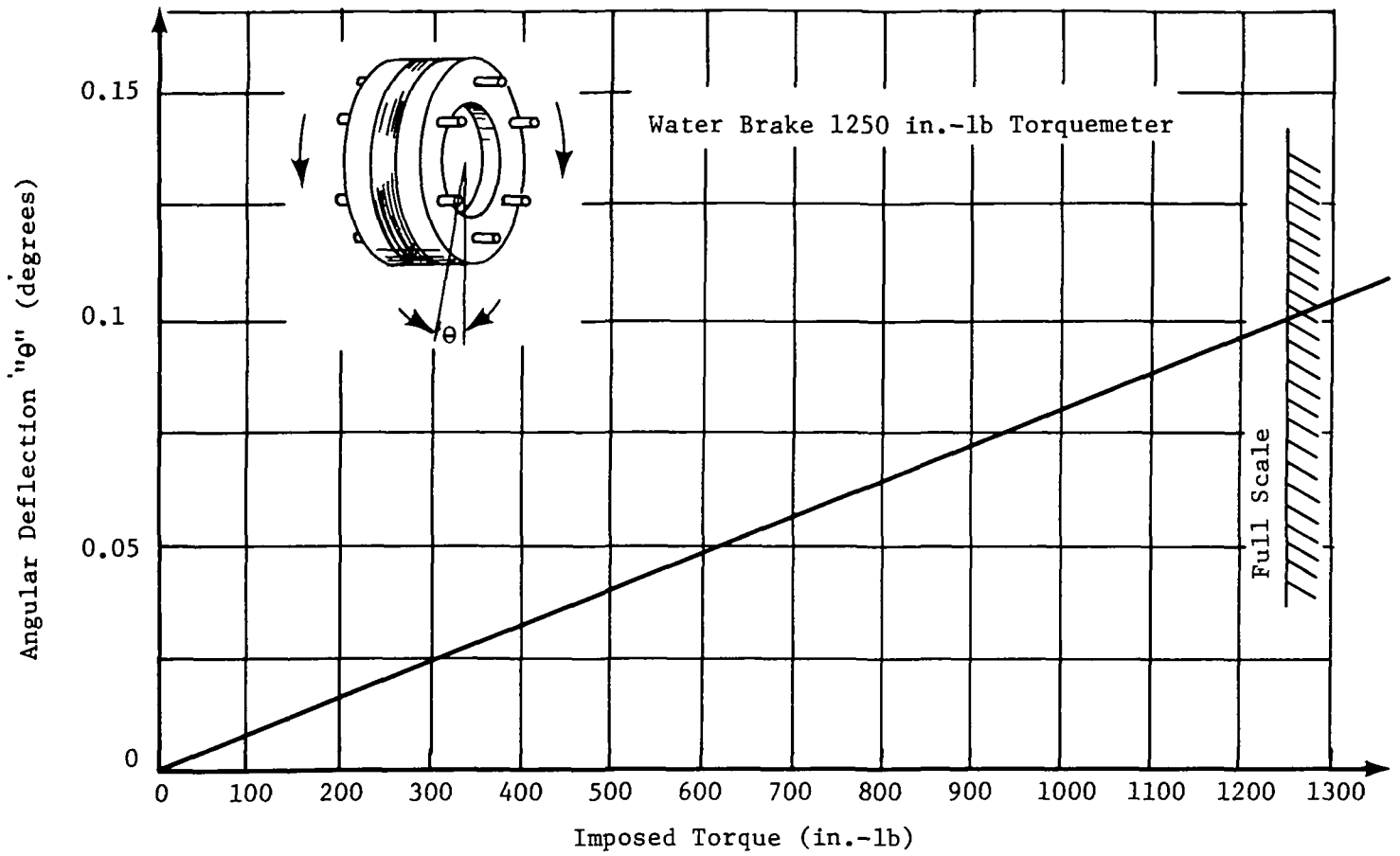


Figure 30a. Angular Deflection of Water Brake Bytrex Torquemeter as a Function of Imposed Torque.

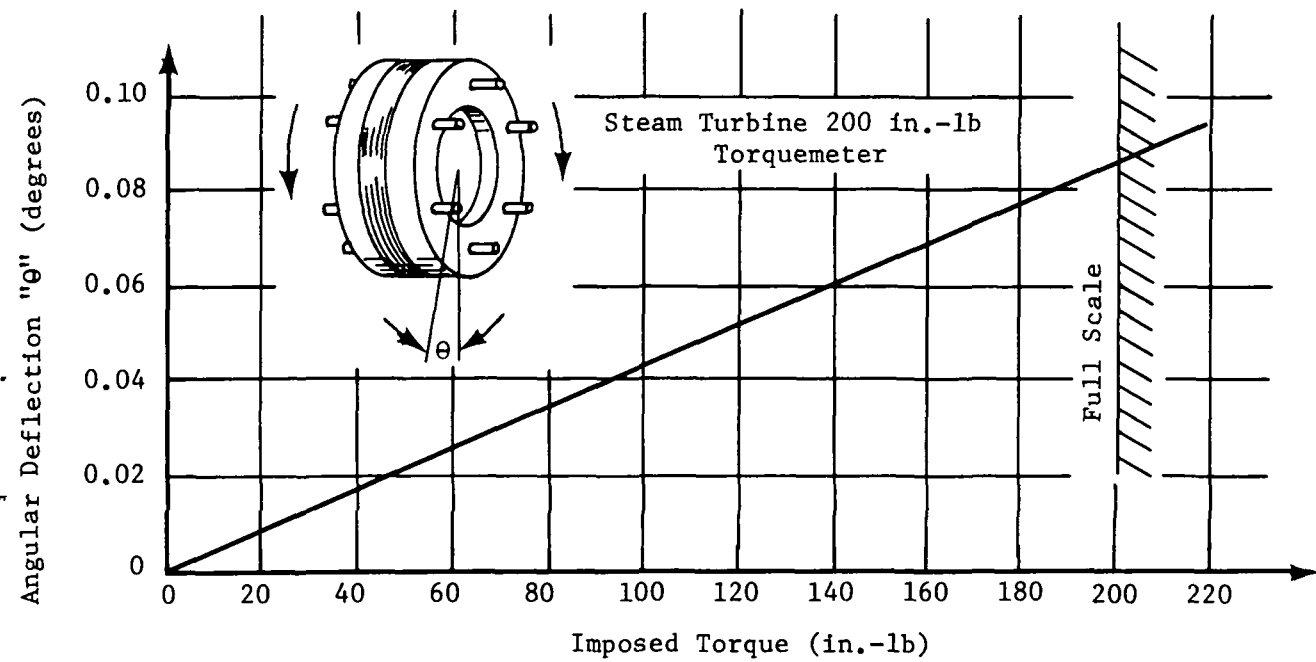
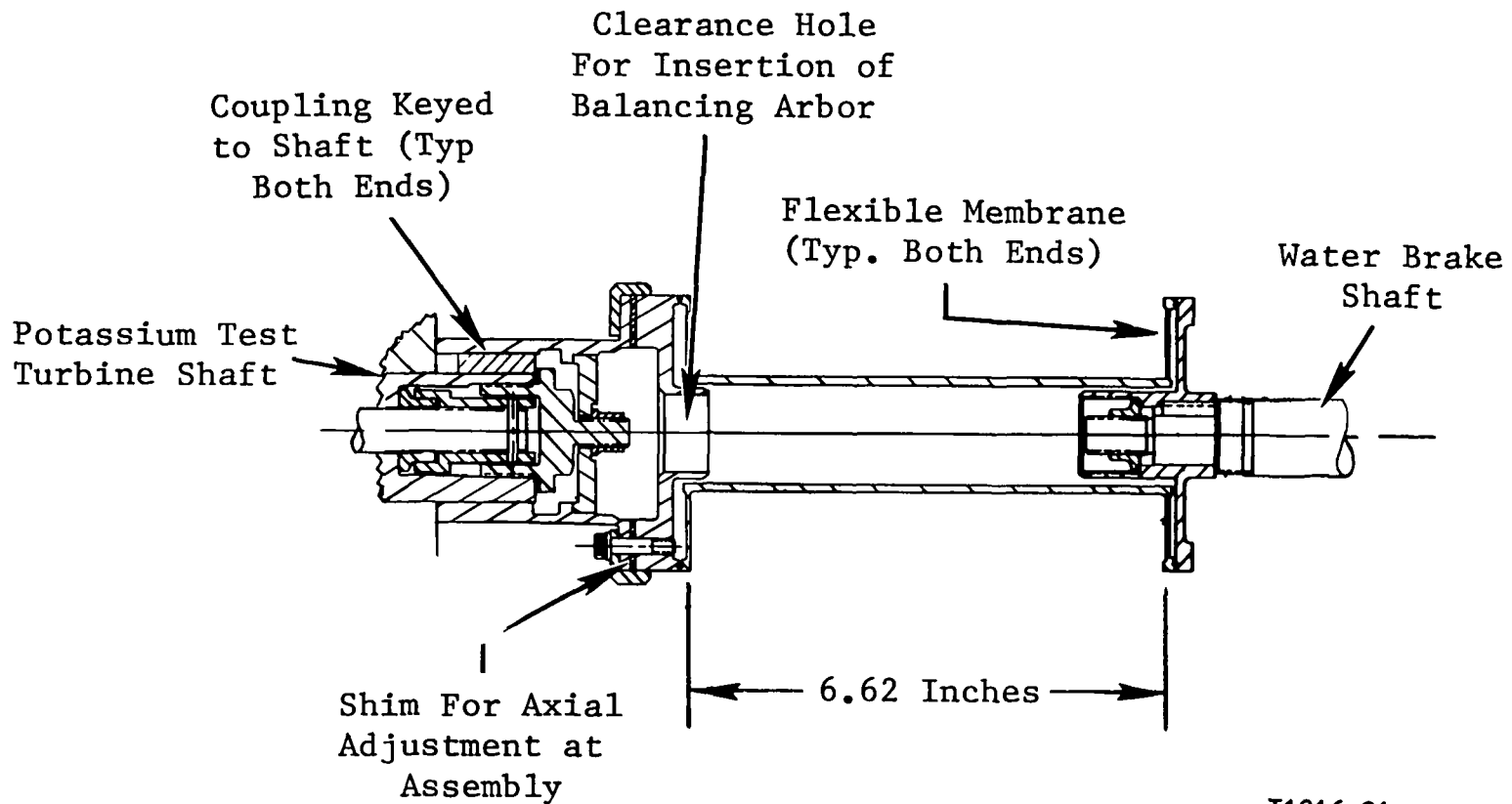


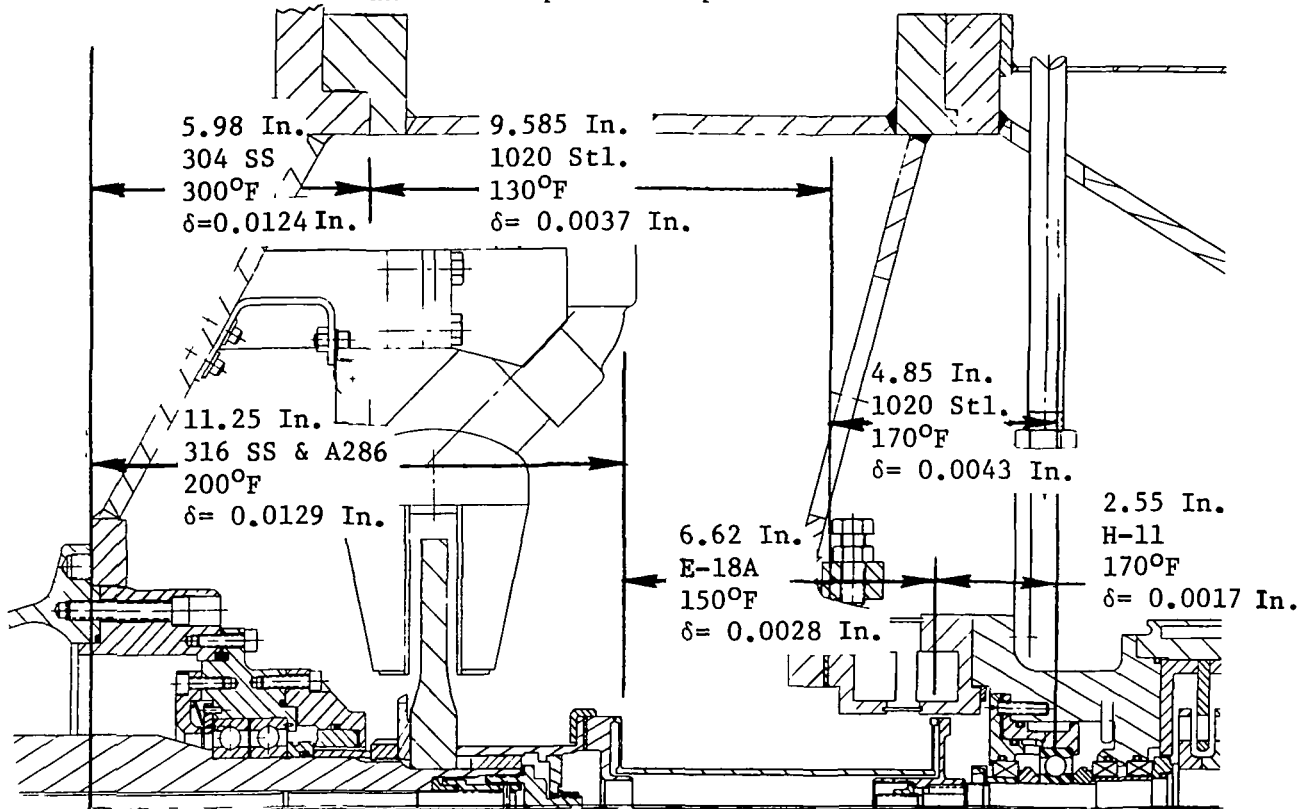
Figure 30b. Angular Deflection of Steam Turbine Bytrex Torquemeter as a Function of Imposed Torque.



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Figure 31. Flexible Membrane (Bendix) Coupling.

Initial Component Temperature = 70°F

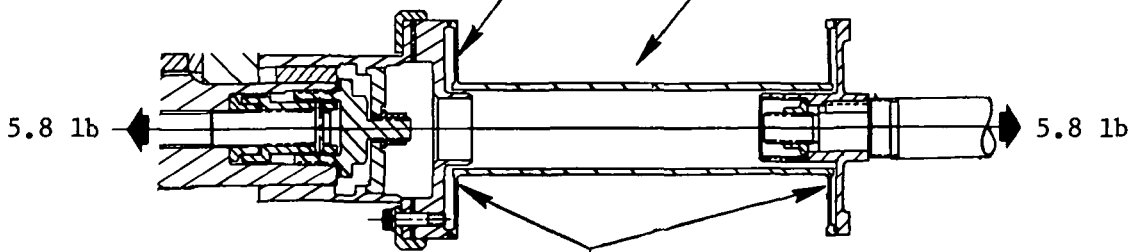


Total Stator Thermal Expansion = 0.0204 Inches

Total Rotor Thermal Expansion = 0.0174 Inches

Total Differential Thermal Expansion = 0.003 Inches, Resulting in 0.0015 Inches Tension Per Membrane in Coupling. This Produces a Tensile Force of $T = 4.0$ lbs. in the Rotor.

Axial Spring Constant of Each Membrane = 1340 lb/in.



Membrane Stress Resulting From 4.0 lb Axial Force = 4626 psi
 Allowable Total Membrane Stress for Infinite Life = 50,000 psi

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Figure 32. Axial Extension of Flexible Membrane Coupling and Resulting Membrane Stresses Due to Differential Thermal Expansion of Support Structure and Power Train Rotor in Vicinity of Coupling.

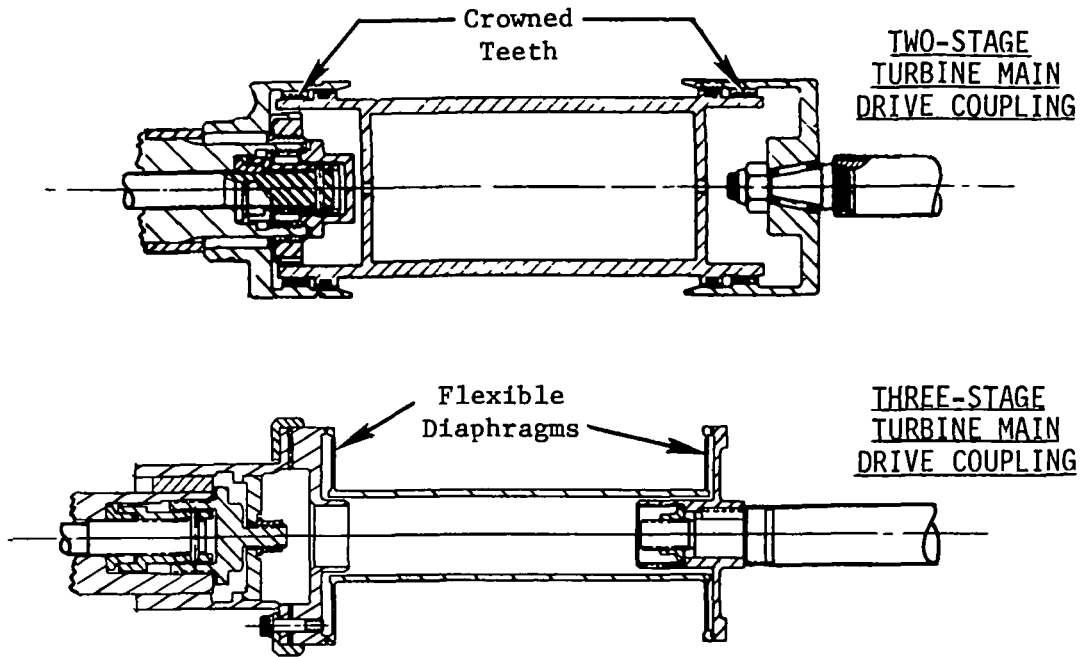


Figure 33a. Comparison of Main Potassium Turbine Drive Couplings of the Two-Stage and Three-Stage Test Turbines.

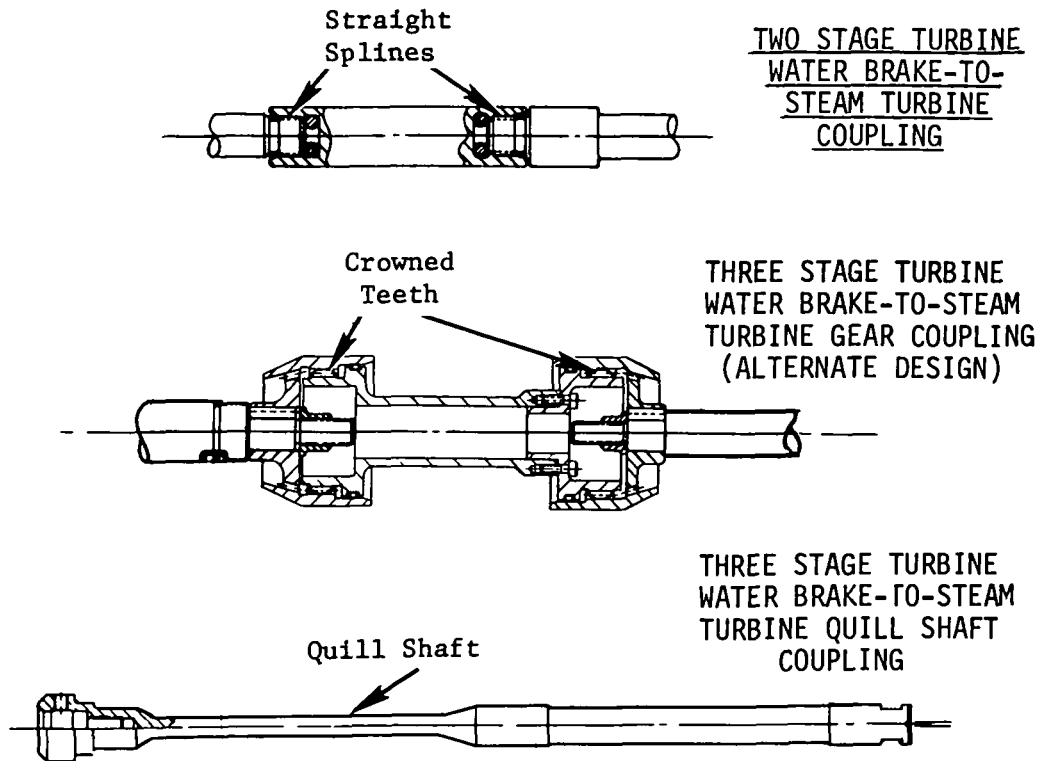


Figure 33b. Comparison of the Aft (Steam Turbine) Drive Couplings of the Two-Stage and Three-Stage Test Turbines.

T1316-33

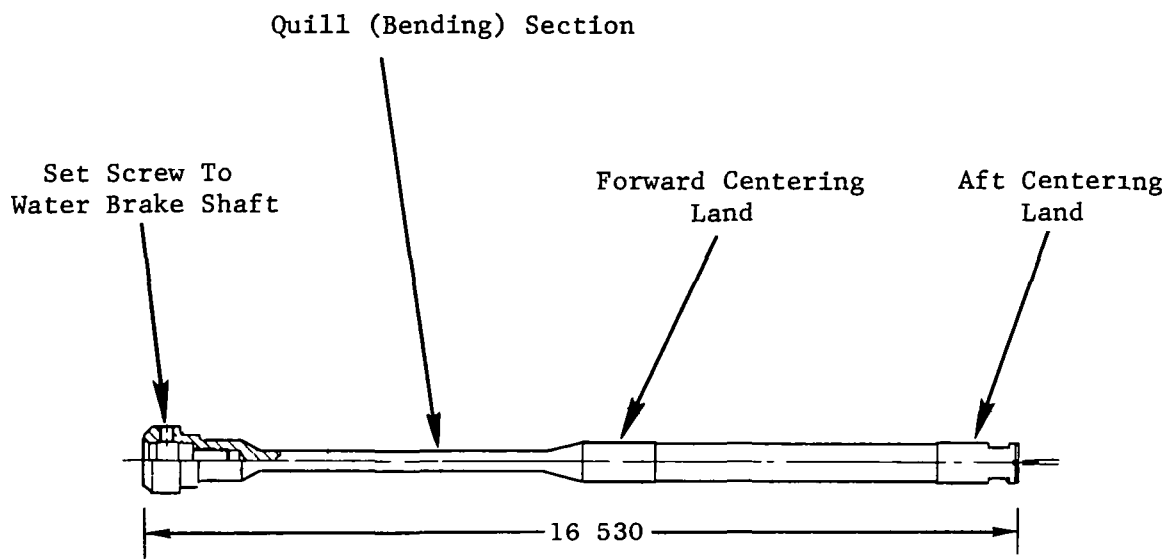


Figure 34a. Quill - Shaft Coupling.

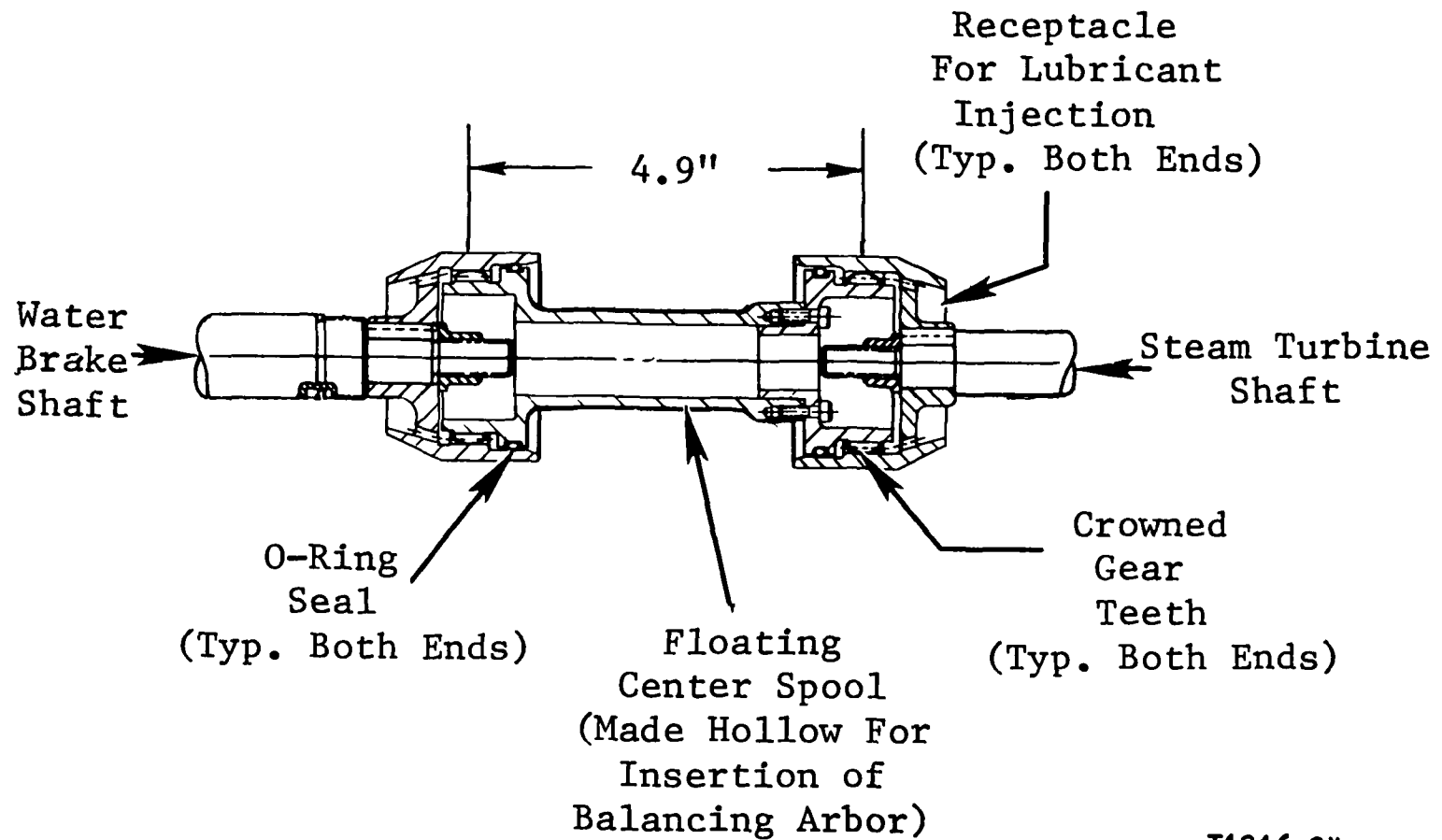


Figure 34b. Gear-Tooth (Koppers) Coupling. (Alternate Water Brake - Steam Turbine Coupling)

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SECTION IX

MANUFACTURING DRAWINGS

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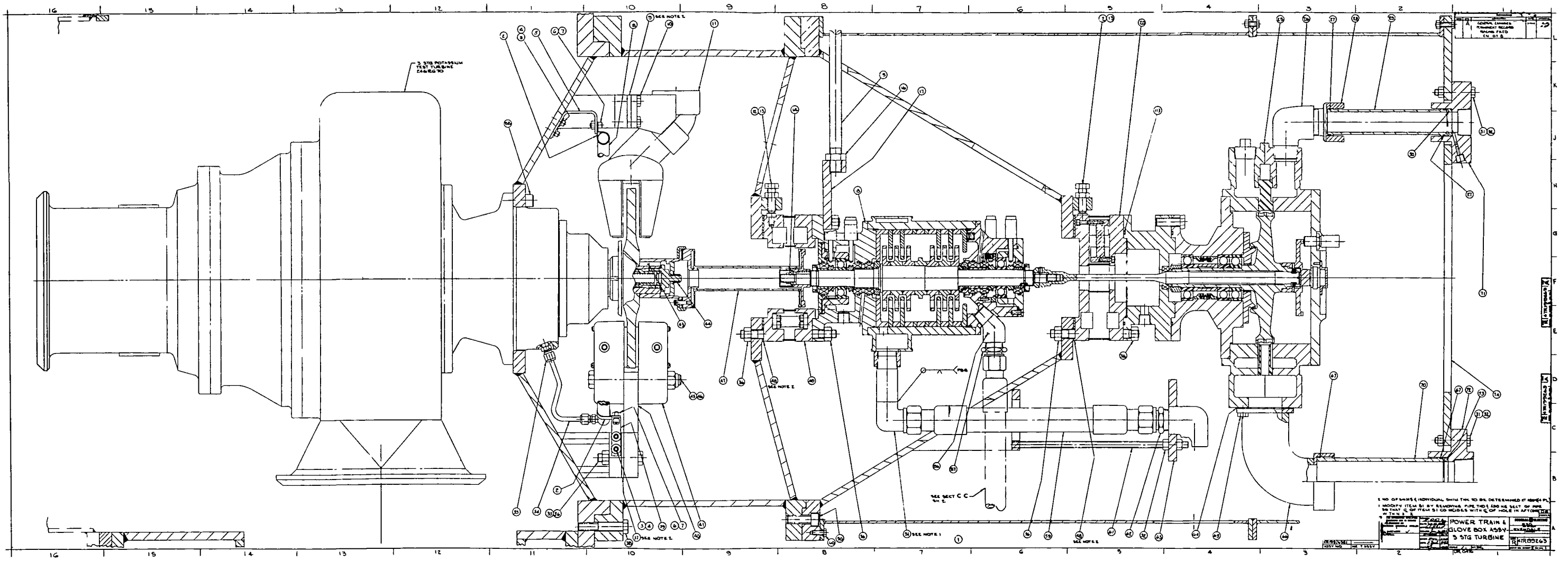


Figure 35. Main Assembly Drawing of the 3 Stage Turbine Power Train and Aft Environmental Chamber. (47R199263 - sheet 1)

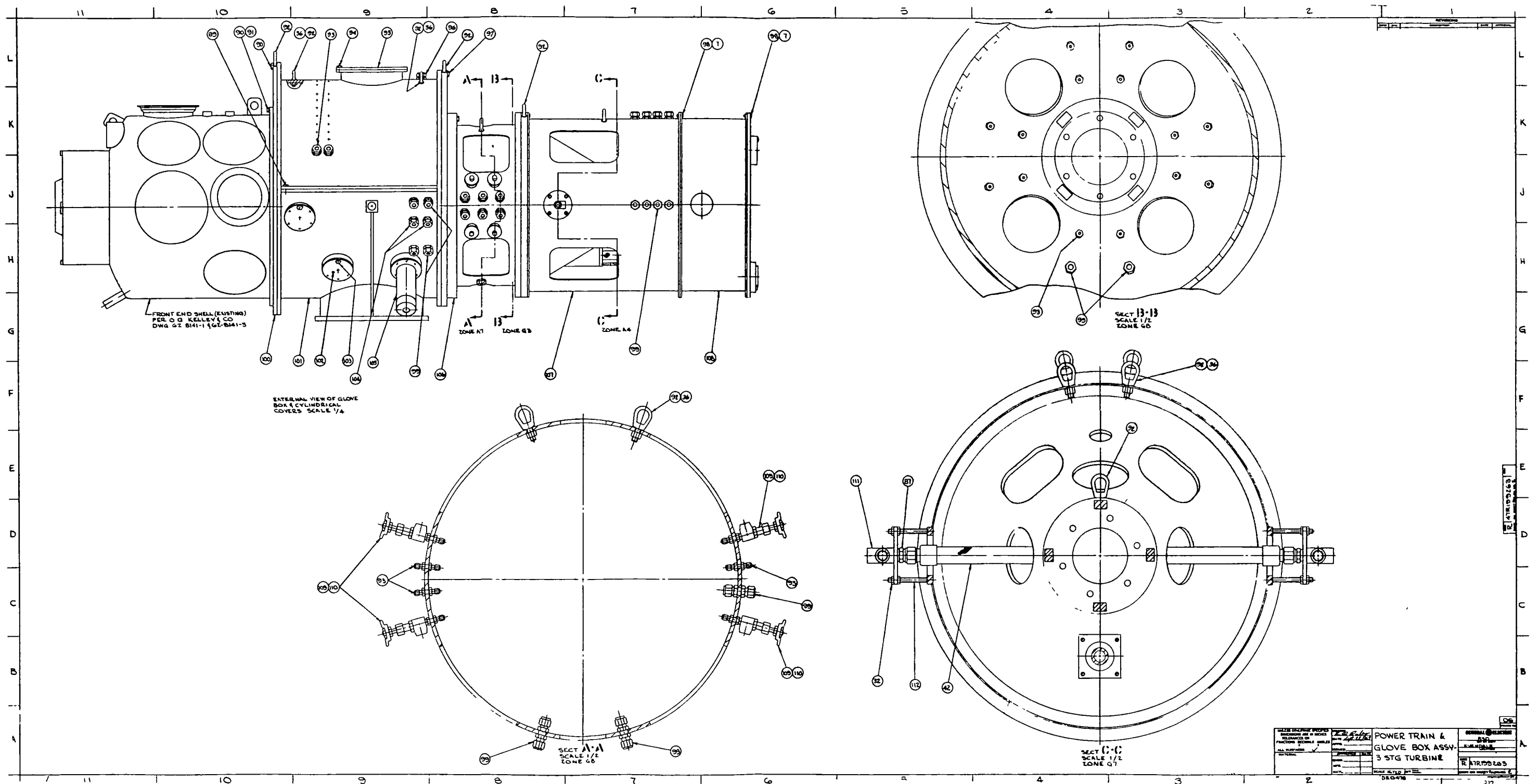


Figure 36. Assembly of Power Train and Environmental Chamber Surrounding the 3 Stage Potassium Test Turbine. (47R199263 - sheet 2)

GENERAL ELECTRIC SPACE POWER & PROPULSION SECTION CINCINNATI, OHIO 45215	PARTS LIST (SEE SPPS DRAFTING STANDARD NO. 5702-19)	(A) PARTS LIST NO. 47R199263G1
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(B) CODE IDENT. NO. 15226	(C) TITLE Power Train & Glove Box Ass'y - 3 Stg. Turbine
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(D) ITEM NO.	(E) ZONE	(F) DWG CODE	(G) IDENTIFICATION NO.	(H) DESCRIPTION/NOMENCLATURE	(I) QTY	(J) SHEET NO.	(K) REV CODE	(L) INTRO REF.
1	A-7		47R199263G1	Ass'y	X	1, 2		
2			MS9025-11 (Zone & Qty. B-11-2,	Clamp - Loop L-11-1)	3	1		
3			MS24675-11 (Zone & Qty. L-11-2,	Socket Head Cap Screw A-10-4)	6	1		
4			AN960-10 (Zone & Qty. A-10-4,	Washer - Flat L-11-2)	6	1		
5	L-11		47C141498P1	Bracket-Manifold Upper	1	1		
6			MS24675-20 (Zone & Qty. A-10-2,	Socket Head Cap Screw L-11-1)	3	1		
7			MS9356-10 (Zone & Qty. (Sh 1 - A-10-2, L-11-1)(Sh 2 - L-6-	Nut, Hex.	39	1, 2		
8	L-10		941D344G1	Manifold - Lube	1	1		
9	L-10		47B116030	Shim - Cooler Bracket	AR	1		
10	L-10		MS35308-363	Hex Head Cap Screw	4	1		
11	L-9		263E159G1	Cooler - Disk Brake	1	1		
12			47B116021P1 (Zone & Qty. L-5-4, J-9-4)	Screw - Adjusting	8	1		
13			AN924-5 (Zone & Qty. L-5-4, J-9-4)	Jam Nut	8	1		
14	K-8	5	ZEE1845-080	Nut - Double Hex (A)	1	1		
15	K-7		47B116014P1	Torque Arm	1	1		
16	K-7		AN924-8	Jam Nut	1	1		
17	K-7		941D334P1	Support - Torque Arm	1	1		
18	K-7	2	263E306G1	Water Brake	1	1	4	
19	L-7		941D385G1	Housing - Vibration Damper	2	1		
20	L-5	3	263E199G1	Bytrex Torque Pick-up Aft	1	1	4	
21	H-5		MS20066-182	Key - Sq, Mach.	2	1		
22	L-5		47B116005P1	Spindle - Steam Turbine	1	1		
23	L-4		941D322G1	Housing - Lube	1	1		

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(N) REVISION CODES 1. ITEM ADDED 2. ITEM DELETED 3. QTY CHGD 4. IDENT. NO. CHGD	(Q) ISSUED BY <i>D.A. Pritchett</i> D.A. Pritchett	(R) DATE ISSUED January 31, 1968
5. IDENT NO. & QTY CHGD 6. EDIT CORRECTION	(S) REVISION AUTHORITY CN 0718	(T) DISTRIBUTION CODE 06

SP 1198-A

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24	L-4		MS35307-413	Hex. Head Cap Screw	6	1		
25	L-3	3	263E305G1	Steam Turbine Ass'y (I)	1	1	4	
26	L-3		AN914-10	Elbow 90°, Int & Ext Pipe Thread.	1	1		
27		5	2-128 (Zone & Qty. 4-1-1, 1-3-1)	"O" Ring (B)	2	1		
28	L-3		47C141496P1	Nipple - Inlet	1	1		
29	L-2		47C141492P1	Coupling - Inlet	1	1		
30	J-2		47B116022P1	Spacer - Inlet	1	1		
31			MS35307-367 (Zone & Qty. J-1-4, C-1-4)	Hex Head Cap Screw	8	1		
32			MS51971-3 (Zone & Qty. (Sh 1 C-1-4, J-1-4, A-5-8 (Sh 2 B-5-16)	Hex - Nut	32	1, 2		
33	A-11	5	400-1-2-316	Connector - Male (C)	5	1		
34	A-11		47B116019P1	Tube - Lube	5	1		
35	A-11		MS35307-417	Hex Head Cap Screw	2	1		
36			MS51971-5 (Zone & Qty. (Sh 1 D-3-6, A-6-6, A-8-6, D-9-6, A-11-2 (Sh 2 L-9-1, I-9-2, F-7-2, F3-2	Nut - Hex	33	1, 2		
37	A-10		47B116029	Shim - Puck Holder	AR	1		
38	A-8		MS35308-418	Hex Head Cap Screw	18	1		
39	A-10		47B116026P1	Bracket - Manifold, Lower	2	1		
40	A-9		47C141504P2	Bracket - Puck Holder	1	1		
41	A-9	5	PD 1473-1	Puck Holder Ass'y (D)	1	1		
42	B-5	5	RR122FF 1 1/16-12 12 1/2" lg	Hose (E)	2	2		
43	E-9		142B1495P1	Key	2	1		
44	E-9	1	47B116025P1	Nut - Double Hex	1	1		
45	D-9		AN10-61A	Hex Head Cap Screw	2	1		
46	D-9	5	ZEB1845-108	Nut, Hex (A)	2	1		

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(N) REVISION CODES 1. ITEM ADDED 2. ITEM DELETED 3. QTY CHGD 4. IDENT. NO. CHGD 5. IDENT NO. & QTY CHGD 6. EDIT CORRECTION	(S) REVISION AUTHORITY CN 0718	(T) DISTRIBUTION CODE 06

SP 1128-A

GENERAL ELECTRIC SPACE POWER & PROPULSION SECTION CINCINNATI, OHIO 45215	PARTS LIST (SEE SPPS DRAFTING STANDARD NO. S702-19)	(A) PARTS LIST NO. 47R199263G1
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(B) CODE IDENT. NO. <p style="text-align: center; font-size: 1.2em;">1 5226</p>	(C) TITLE Power Train & Glove Box Ass'y - 3 Stg. Turbine
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(D) ITEM NO.	(E) ZONE	(F) DWG CODE	(G) IDENTIFICATION NO.	(H) DESCRIPTION/NOMENCLATURE	(I) QTY	(J) SHEET NO.	(K) REV CODE	(L) INTRO REF.
47	D-9	2	941D314P1	Flex Coupling - Turb to Water Brake	1	1		
48			47B116020 (Zone & Qty. D-8-AR, A-5-AR)	Shim - Bytrex	AR	1		
49	D-8	3	263E198G1	Bytrex Torque Pickup -Fwd	1	1		
50	A-8		MS24675-59	Socket Head Cap Screw	18	1		
51	A-7		MS20822-16	Elbow 90°, Pipe thd to FL Tube (Modified)	1	1		
52	B-6		47C141497G1	Ring - Oil	1	1		
53	B-6		AN101112	Hex Head Cap Screw	3	1		
54	D-6	5	ZEB1845-070	Nut - Double Hex (A)	1	1		
55	D-5	5	ZEB1845-064	Nut - Double Hex (A)	1	1		
56	D-5	2	47C141479P1	Gear Coupling - Water Brake to Steam Turbine	1	1		
57	D-4		47C141480P1	Locknut	1	1		
58	D-4	5	2-20	"O" Ring (B)	1	1		
59	A-6	5	RR162-FF-1 5/16-12 Length 15 1/4"	Hose (E)	1	1		
60	A-8		MS35308-305	Hex Head Cap Screw	18	1		
61	A-5	5	A9-8 (Length 11.25-11.75)	The Rod (Modified) (F)	4	1		
62	A-5		AN816-16	Nipple - FL Tube to Pipe Thread	1	1		
63	A-5		47B116015G1	Elbow Adapter	1	1		
64	A-4		47B116024P1	Gasket Elbow Steam Out	1	1		
65	A-4		AN101310	Hex Head Cap Screw	4	1		
66	A-3		941D345G1	Elbow - Steam Out	1	1		
67		5	2-150 (Zone & Qty. D-3-1, D-2-1)	"O" Ring (B)	2	1		
68	D-3		MS51023-30 Type A	Set Screw - Hex Socket	4	1		

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SP 1198-A

GENERAL ELECTRIC SPACE POWER & PROPULSION SECTION CINCINNATI, OHIO 45215	PARTS LIST (SEE SPFS DRAFTING STANDARD NO. S702-18)	(A) PARTS LIST NO. 47F199263G1
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(B) CODE IDENT. NO. 15226	(C) TITLE Power Train & Glove Box Ass'y - 3 Stg. Turbine
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69	D-2		941D323P1	Pickup Disk-Speed	1	1		
70	D-2		47C141493P1	Coupling Outlet Steam Turbine	1	1		
71	F-1		47C141489P1	Connector - Inlet	1	1		
72	D-1		941D336P1	Connector - Outlet	1	1		
73	D-1		47B116022P2	Spacer - Steam Out	1	1		
74	D-1		263E151P1	Cover Aft, Steam Turbine	1	1		
75	L-7		MS49005-2	Plug Pipe	6	1		
76	L-7		47C141537P1	Damper Vibration	2	1		
77	L-6	5	2-38	"0" Ring (B)	2	1		
78	L-6		MS9360-14	Hex Nut	2	1		
79	K-6		MS2000208	Washer - CSK	2	1		
80			MS20995-032 (Zone & Qty. G-6-AR, K-6-AR)	Lockwire	AR	1		
81	J-6	5	7500-1/2	Flredseal (B)	2	1		
82	H-6		941D376P1	Bracket- Vibration Damper	1	1		
83	H-6		MS24675-14	Socket Head Cap Screw	16	1		
84	G-6	5	52LH3324-02	Locknut (A)	16	1		
85	G-6		MS20073-03-11	Hex Head Screw	13	1		
86	B-7		AN915-6J	Elbow - Int & Ext Pipe Thread 45°	2	1		
87			AN816-12J (Zone & Qty. (Shl B-7-2 (SH 2 E-5-2)	Nipple - FL Tube & Pipe Thread	4	1, 2		
88	K-11		MS24675-57	Socket Head Cap Screw	16	1		
89	L-10		MS35308-360	Hex Head Cap Screw	28	2		
90			MS35308-413 (Zone & Qty. L-10-28, L-10-18)	Hex Head Cap Screw	46	2		
91	L-10		AN960-816	Flat Washer	28	2		

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SP 1198-A

GENERAL ELECTRIC SPACE POWER & PROPULSION SECTION CINCINNATI, OHIO 45215	PARTS LIST (SEE SPSS DRAFTING STANDARD NO. 5702-19)	(A) PARTS LIST NO. 47R199263G1
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(B) CODE IDENT. NO. 15226	(C) TITLE Power Train & Glove Box Ass'y - 3 Stg Turbine
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(D) ITEM NO.	(E) ZONE	(F) DWG CODE	(G) IDENTIFICATION NO.	(H) DESCRIPTION/NOMENCLATURE	(I) QTY	(J) SHEET NO.	(K) REV CODE	(L) INTRO REF.
92		5	30Y23 (with 1/2 - 13 UNC Thread Shank) (Zone & Qty. L-10-2, L-9-1, L-9-2, L-8-2, L-8-2, F-7-2, F-3-2, E-3-1)	Lifting Eye (G)	14	2		
93		5	400-61-316 (Zone & Qty. L-9-48, G-4-14, C-9-6, C-6-3)	Bulkhead Union (C)	71	2		
94	L-9		MS35307-306	Hex Head Cap Screw	6	2		
95	L-9		47C141486P1	Cover Plate - Split	1	2		
96	L-8	5	1610-61-316	Bulkhead Union (C)	1	2		
97	L-8		MS35308-414	Hex Head Cap Screw	18	2		
98			MS35308-308 (Zone & Qty. L-7-18, L-5-18)	Hex Head Cap Screw	36	2		
99		5	810-61-316 (Zone & Qty. G-4-2, G-7-12, F-9-4, A-8-1, A-7-1, C-6-3)	Bulkhead Union (C)	23	2		
100	G-10		941D329P1	Flange, Adapter	1	2		
101	G-9		47R199258G1	Aft Enclosure, Glove Box	1	2		
102	G-9		47R116031P1	Cover - Glove Box	3	2		
103	G-9		MS35307-407	Hex Head Cap Screw	30	2		
104	G-9	5	1010-61-316	Bulkhead Union (C)	2	2		
105	G-9		47C141508G1	Cover - Port	1	2		
106	G-8		941D343G1	Support Structure	1	2		
107	F-8		47R199261G1	Cyl Cover - Fwd	1	2		
108	G-7		941D339G1	Cyl Cover - Aft	1	2		
109		5	400-11-4-316 (Zone & Qty. C-9-4, E-6-2, B-6-2)	Bulkhead Male Connector (C)	8	2		
110		5	Y354H 340 Series (Zone & Qty. C-9-4, E-6-2, B-6-2)	Needle Valve (H)	8	2		
111	E-5		47R116013G1	Tee- Adapter	2	2		
112	B-5	5	A9-8 (Modified) 4.50-5.00 Lg	Threaded Rod (F)	8	2		
113	J4	5	2-250	O'Ring Viton A (B)	1	1		

(M) DRAWING CODES. 1. VENDOR ITEM - SPEC CONT DWG 2. VENDOR ITEM - SOURCE CONT DWG 3. ALTERED PART 4. CUSTOMER FURNISHED PART 5. CATALOG ITEM	(O) CONTRACT NO.	(P) PREPARED BY R. Conley
(N) REVISION CODES 1. ITEM ADDED 2. ITEM DELETED 3. QTY CHGD 4. IDENT. NO. CHGD.	(Q) ISSUED BY D.A. Pritchett	(R) DATE ISSUED January 31, 1968
5. IDENT NO. & QTY CHGD 6. EDIT CORRECTION	(S) REVISION AUTHORITY CN 0718	(T) DISTRIBUTION CODE 06

SP 1198-A

GENERAL ELECTRIC SPACE POWER & PROPULSION SECTION CINCINNATI, OHIO 45215	PARTS LIST (SEE SPPS DRAFTING STANDARD NO. 5702-10)	(A) PARTS LIST NO. 47E199263G1
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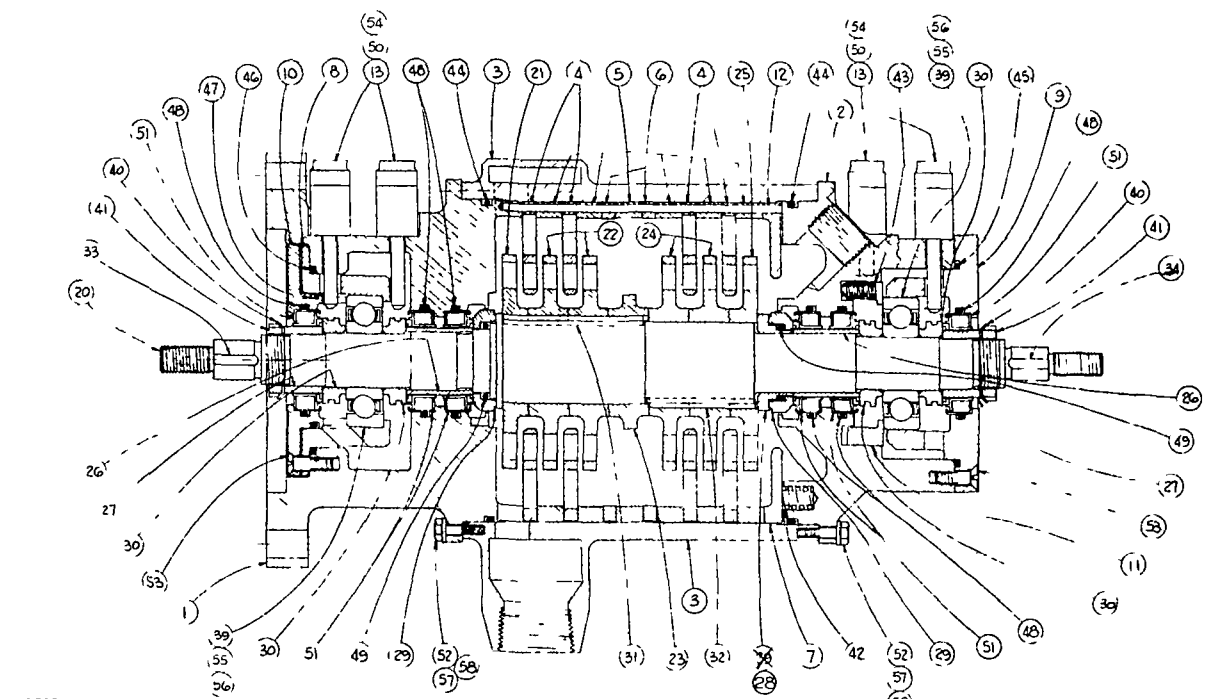
(B) CODE IDENT. NO. 15226	(C) TITLE Power Train & Glove Box Ass'y - 3 Stg. Turbine
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(D) ITEM NO.	(E) ZONE	(F) DWG CODE	(G) IDENTIFICATION NO.	(H) DESCRIPTION/NOMENCLATURE	(I) QTY	(J) SHEET NO.	(K) REV CODE	(L) INTRO REF.
			<u>VENDOR'S NAMES AND ADDRESSES</u>					
			A.	ELASTIC STOP NUT CORP OF AMERICA UNION, NEW JERSEY (or equiv)				
			B.	PARKER SEAL COMPANY CLEVELAND, OHIO (or equiv)				
			C.	CRAWFORD FITTING COMPANY CLEVELAND 10, OHIO (or equiv)				
			D.	GOODYEAR TIRE & RUBBER COMPANY AKRON, OHIO (or equiv)				
			E.	J.N. FAUVER COMPANY CINCINNATI, OHIO (or equiv)				
			F.	PIC DESIGN CORP. EAST ROCKAWAY, NEW YORK (or equiv)				
			G.	McMASTER CARR SUPPLY COMPANY CHICAGO, ILLINOIS (or equiv)				
			H.	HOKE INC. CRESSKILL, NEW JERSEY (or equiv)				
			I.	BARBOUR STOCKWELL CAMBRIDGE, MASSACHUSETTS (or equiv)				

(M) DRAWING CODES 1. VENDOR ITEM - SPEC CONT DWG 2. VENDOR ITEM - SOURCE CONT DWG 3. ALTERED PART 4. CUSTOMER FURNISHED PART 5. CATALOG ITEM (N) REVISION CODES 1. ITEM ADDED 2. ITEM DELETED 3. QTY CHGD 4. IDENT. NO. CHGD. 5. IDENT NO. & QTY CHGD 6. EDIT CORRECTION	(O) CONTRACT NO. (Q) ISSUED BY D.A. Pritchett (S) REVISION AUTHORITY CN 0718	(P) PREPARED BY R. Conley (R) DATE ISSUED January 31, 1968 (T) DISTRIBUTION CODE 06
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SP 1198-A

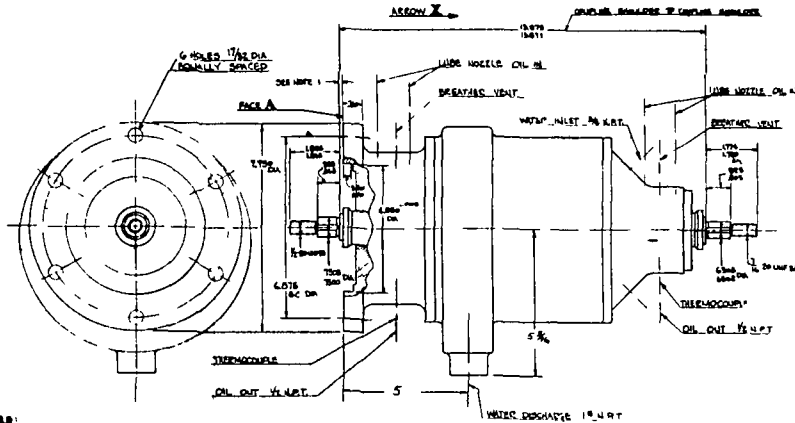
ITEM	PART NR	DESCRIPTION	QTY
1	D1936 1	MOUNTING FLANGE	1
2	C1937 1	END FLANGE	1
3	C1934 1	CASING	1
4	B1942 1	STATOR DISC	4
5	B1971 1	STATOR DISC SPACER	6
6	B1971 2	STATOR DISC SPACER	2
7	B1943 1	WEAR PLATE	1
8	D1949 1	BEARING LINER	1
9	B1951 1	BEARING RETAINER	1
10	A1952 1	RING SPRING HOLDING	1
11	P9114 20	ROD STATOR DISC HOLDING	20
12	B1954 1	LUBE NOZZLE	1
13			
14			
15			
16			
17			
18			
19			
20	D1935 1	SHAFT	1
21	B1941 1	ROTOR DISC	1
22	B1941 2	ROTOR DISC	2
23	B1941 3	ROTOR DISC	1
24	B1941 3	ROTOR DISC	2
25	B1941 4	ROTOR DISC	1
26	A1948 1	SLEEVE	2
27	A1947 1	SLEEVE	2
28	A1944 1	COLLAR	1
29	A1946 1	MATING RING	2
30	A1945 1	MATING RING	4
31	B1065 2	KEY	1
32	B1065 1	KEY	1
33	B1147 2	KEY	2
34	B1147 1	KEY	2
35			
36			
37			
38			
39	P1705	BALL BEARING NEW DEPART #Q3305XR7A	2
40	P1954	LOCK WASHCR OS NILE MS1008115	2
41	P1045 1	LOCK NUT OS SIZE MS172241	2
42	P8905	HELICAL SPRING LEE # LC04395	20
43	P8902	HELICAL SPRING LEE # LC03212	6
44	P4084	O RING 5/8" ID x 1/2" OD x 1/8" TH AN6230 34	2
45	P4016	O RING 3/4" ID x 3/4" OD x 1/8" TH AN6230 16	1
46	P4015	O RING 3/4" ID x 3/4" OD x 1/8" TH AN6230 13	1
47	P4008	O RING 2 1/8" ID x 2 1/8" OD x 1/8" TH AN6230 6	1
48	P4008	O RING 1 7/8" ID x 1 7/8" OD x 1/8" TH AN6230 3	1
49	P4121	O RING 1 1/2" ID x 1 1/2" OD x 1/8" TH AN6227 21	2
50	P4109	O RING 1 1/2" ID x 1 1/2" OD x 1/8" TH AN6227 9	4
51	P3020	CARBON FACE SEAL SEALOR #346002742	6
52	P5113	HEX HEAD BOLT #10 3/4 LG	30
53	P5335	FLAT HEAD SCREW #10 2 1/2 LG	12
54	P5407	FLAT HEAD SCREW #10-24 x 1 1/2 LG	8
55	P8951	BREATHER CAP AIR MATR #A14192	2
56	P8960	TEMPERATURE ASSY CONAL #L 360 CA	4
57	P6008	WASHER PLAIN #10 SHIE	30
58	P8001	LOCKWIRE 1/8 DIA	200
59			
60			
61			
62			
63			
64			
65			
66			
67			
68			
69			
70			
71			



NOTES
 1 USE GREASE ON ALL O RINGS WHEN INSTALLING AT ASSEMBLY
 2 BEARING #Q3305XR7CF IS ALTERNATE EQUIVALENT

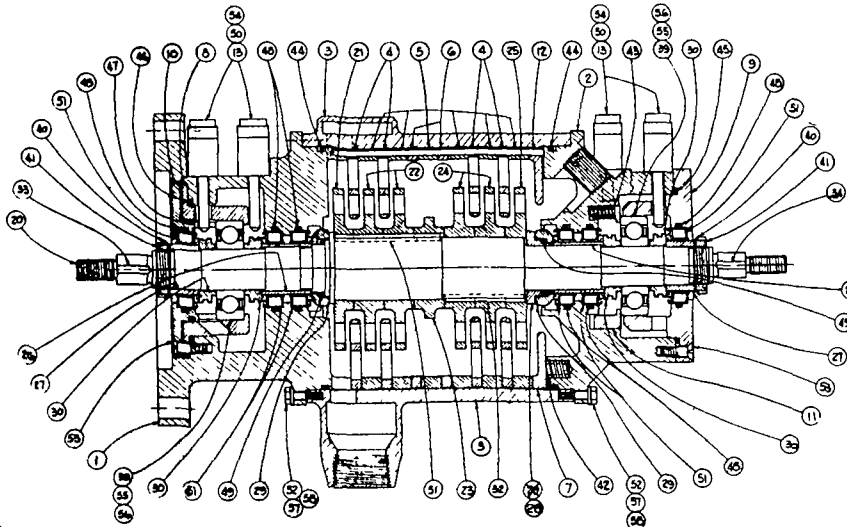
INDUSTRIAL ENGINEERING CO	
EST. BY 1918	
HYDRA-BRAKE ASSEMBLY	
STELLITE, RUBBER AND STAINLESS STEEL	C9529-5

Figure 37. Manufacturer's Assembly Drawing of Water Brake. (C9529-5)



- NOTES:
1. SHAFT SHOULD BE LOCATED IN LINE OF THE MOUNTING FLANGE FACE. THE ACTUAL POSITION IS TO BE DETERMINED AS TIME OF INSTALLATION.
 2. SEE DRAWING A 1790 FOR SERVICE CONNECTION IDENTIFICATION AND DIMENSIONS.

INSTALLATION DATA REF # SK 255



- NOTES:
1. USE GREASE ON ALL FITTING SURFACES AT ASSEMBLY.
 2. REMOVE TOOLS AND DEBRIS FROM EQUIPMENT.

ASSEMBLY REF # D9529 5

SERVICE CONNECTION DATA REF # A1790

CONNECTION	TYPE	LOCATION FROM FACE A	THREAD FOR FITTING	SEE NOTE
WATER DISCHARGE	WATER COMPART	0 00	5	1 NPT
INLET		3 00	10 7/8	1/2 NPT
TO SEAL COMPART, PRESS VENT		12 00	11 1/2	3/8 NPT
VENT		5 00	10 1/2	3/8 NPT
TO SEAL COMPART, PRESS VENT		7 00	11	1/2 NPT
DRIVE END		4 00	3	1/2 NPT
TO SEAL COMPART, PRESS VENT		3 00	6 3/4	1/2 NPT
OIL INLET	BALL BEG COMPART	11 00	7 1/2	1/2 NPT
OIL INLET		11 00	6 3/4	1/2 NPT
OIL SCAVENGE		5 00	6	1/2 NPT
OIL VENT		10 00	6	1/2 NPT
THERMOCOUPLE		10 00	7 1/2	1/2 NPT
OIL INLET		2 00	11 3/4	1/2 NPT
OIL INLET		2 00	12 1/4	1/2 NPT
OIL SCAVENGE		6 00	12	1/2 NPT
OIL VENT		8 00	12	1/2 NPT
OIL VENT		11 00	12	1/2 NPT
OIL VENT		4 00	12	1/2 NPT
THERMOCOUPLE		1 00	11 3/4	1/2 NPT
THERMOCOUPLE		3 00	11 3/4	1/2 NPT

NOTES

1. THESE TAPPED HOLES ARE RADIAL (PERP TO AXIS)
2. THESE TAPPED HOLES ARE INCLINED 45° FROM AXIS
3. STRAINER TYPE OIL FILTER IS INSTALLED
4. AIR WAZE BREATHER #1152 IS INSTALLED
5. THESE CONNECTIONS NOT USED PIPE PLUGS INSTALLED
6. CONNECT TO TURBINE RIG OIL SUPPLY

ITEM	PART #	DESCRIPTION	QTY
1	D1952-1	SHAFT	1
2	C1931-1	END FLANGE	1
3	C1931-1	CASING	1
4	D1952-1	STATOR DISC	4
5	D1952-1	STATOR DISC SPACER	4
6	D1952-1	ROTOR DISC SPACER	7
7	D1952-1	WEAR PLATE	1
8	D1952-1	BEARING LINER	1
9	D1952-1	BEARING LINER	1
10	D1952-1	BEARING RING	1
11	A1952-1	RING SPRING HOLDING	1
12	P1952-1	ROD STATOR DISC HOLDING	6
13	D1952-1	LUBE NOZZLE	1
14			
15			
16			
17			
18			
19			
20	D1952-1	SHAFT	1
21	D1952-1	ROTOR DISC	4
22	D1952-1	ROTOR DISC	4
23	D1952-1	ROTOR DISC	4
24	D1952-1	ROTOR DISC	4
25	D1952-1	ROTOR DISC	4
26	A1952-1	SLEEVE	2
27	A1952-1	SLEEVE	2
28	A1952-1	COLLAR	1
29	A1952-1	MATING RING	4
30	A1952-1	MATING RING	4
31	D1952-1	KEY	1
32	D1952-1	KEY	1
33	D1952-1	KEY	1
34	D1952-1	KEY	1
35			
36			
37			
38			
39	P1952	BALL BEARING NEW DIMPT	2
40	P1952	LOCK WASHER 3/8\"/>	

THE INFORMATION ON THIS DWG WAS REPRODUCED FROM THE FOLLOWING VENDOR DWG'S

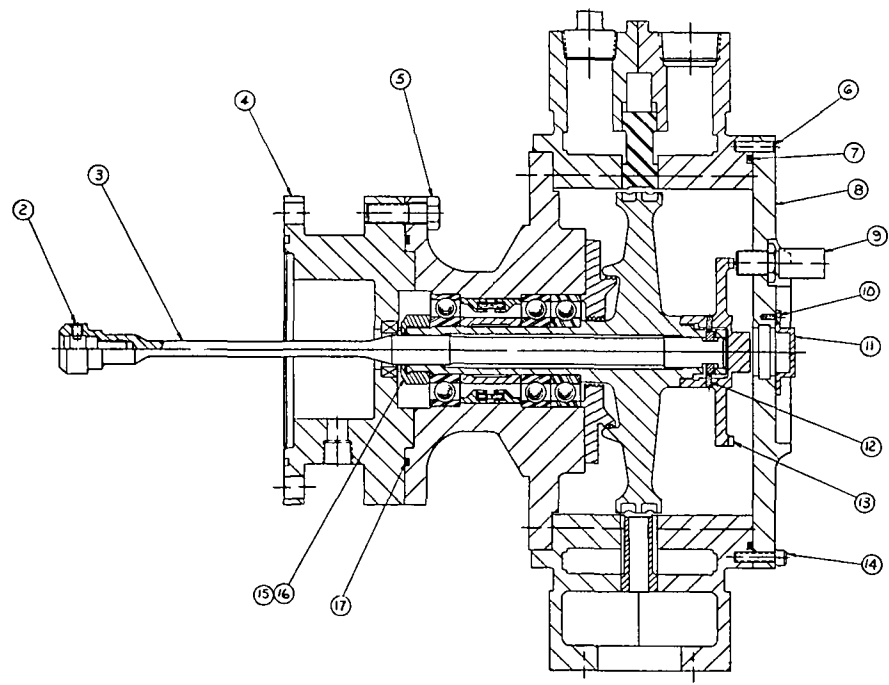
ASSEMBLY REF # D9529 5
 INSTALLATION DATA REF # SK 255
 SERVICE CONNECTION DATE REF # A 1790

VENDOR NAME & ADDRESS
 INDUSTRIAL ENGINEERS CO
 PHILADELPHIA, PENNSYLVANIA

DATE	2014	BY	[Signature]
REV	1	DATE	2014
REV	2	DATE	2014
REV	3	DATE	2014
REV	4	DATE	2014
REV	5	DATE	2014
REV	6	DATE	2014
REV	7	DATE	2014
REV	8	DATE	2014
REV	9	DATE	2014
REV	10	DATE	2014
REV	11	DATE	2014
REV	12	DATE	2014
REV	13	DATE	2014
REV	14	DATE	2014
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REV	16	DATE	2014
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REV	18	DATE	2014
REV	19	DATE	2014
REV	20	DATE	2014
REV	21	DATE	2014
REV	22	DATE	2014
REV	23	DATE	2014
REV	24	DATE	2014
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REV	41	DATE	2014
REV	42	DATE	2014
REV	43	DATE	2014
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REV	45	DATE	2014
REV	46	DATE	2014
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REV	48	DATE	2014
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REV	55	DATE	2014
REV	56	DATE	2014
REV	57	DATE	2014
REV	58	DATE	2014
REV	59	DATE	2014
REV	60	DATE	2014
REV	61	DATE	2014
REV	62	DATE	2014
REV	63	DATE	2014
REV	64	DATE	2014
REV	65	DATE	2014
REV	66	DATE	2014

Figure 38a. Water Brake Installation. (Dwg. No. 263E306)

DATE	DESCRIPTION	BY	APPROVED



NOTES

- ALTERED PART:** NO CHANGES SHALL BE MADE TO THE DESIGN CONFIGURATION UNLESS PARTS OR MANUFACTURING PROCESSES WITHOUT PRIOR WRITTEN APPROVAL OF PURCHASING (SPPS)
- 1 SHOP RUN TOOL PER OS 0000 00 A
 - 2 ALL BALLOUED ITEMS ARE SUBSTITUTIONS OR ADDITIONS TO THE ITEM AS SUPPLIED UNDER VENDOR'S IDENT. NO.
 - 3 SAME AS #4037 P BARBOUR STOCKWELL CO CAMBRIDGE MASS EXCEPT AS SHOWN

VENDOR'S NAME & ADDRESS

- A PAPER SEAL CO CLEVELAND OHIO OR EQWY
- B ELECTRO PRODUCTS LBS INC CINCINNATI OH OR EQWY

① ASSEMBLY

QTY	UNIT	DESCRIPTION	REVISION	DATE	BY	CHKD	APPROVED
1	17	ENG	2	150			
1	16	DI	2	20			
1	15	DI	1	27	CH	480	PI
1	14	DI	1	1/4	20	1/2	1/2
1	13	DI	1	1/4	20	1/2	1/2
2	12	DI	1	1/4	20	1/2	1/2
1	11	DI	1	1/4	20	1/2	1/2
3	10	DI	1	1/4	20	1/2	1/2
1	9	DI	1	1/4	20	1/2	1/2
1	8	DI	1	1/4	20	1/2	1/2
1	7	DI	1	1/4	20	1/2	1/2
2	6	DI	1	1/4	20	1/2	1/2
1	5	DI	1	1/4	20	1/2	1/2
1	4	DI	1	1/4	20	1/2	1/2
1	3	DI	1	1/4	20	1/2	1/2
2	2	DI	1	1/4	20	1/2	1/2
1	1	DI	1	1/4	20	1/2	1/2

LIST OF MATERIAL

QTY	UNIT	DESCRIPTION	REVISION	DATE	BY	CHKD	APPROVED
1	17	ENG	2	150			
1	16	DI	2	20			
1	15	DI	1	27	CH	480	PI
1	14	DI	1	1/4	20	1/2	1/2
1	13	DI	1	1/4	20	1/2	1/2
2	12	DI	1	1/4	20	1/2	1/2
1	11	DI	1	1/4	20	1/2	1/2
3	10	DI	1	1/4	20	1/2	1/2
1	9	DI	1	1/4	20	1/2	1/2
1	8	DI	1	1/4	20	1/2	1/2
1	7	DI	1	1/4	20	1/2	1/2
2	6	DI	1	1/4	20	1/2	1/2
1	5	DI	1	1/4	20	1/2	1/2
1	4	DI	1	1/4	20	1/2	1/2
1	3	DI	1	1/4	20	1/2	1/2
2	2	DI	1	1/4	20	1/2	1/2
1	1	DI	1	1/4	20	1/2	1/2

263E305G1 47R15B249 01
PART NO. HEAT ASSY

CONTROL DRAWING
 M3D
STEAM TURBINE ASSY
 CONTROL NO. HAS 1.8310
 E 15866 263E305
 SCALE

Figure 38b. Steam Turbine Assembly. (Dwg. No. 263E305)

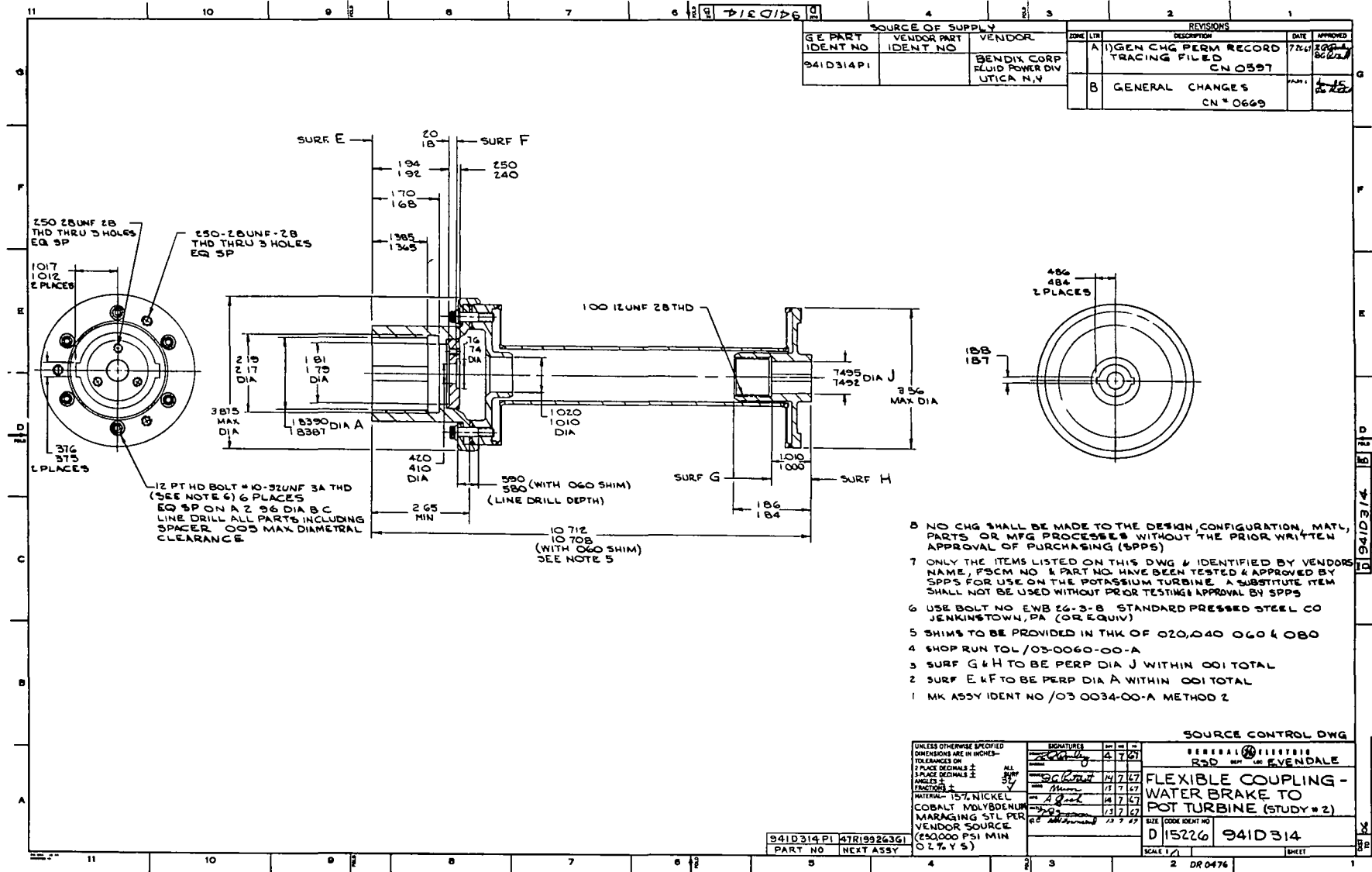


Figure 39. Flexible Membrane (Bendix) Coupling Connecting 3 Stage Test Turbine and Water-Brake. (941D314)

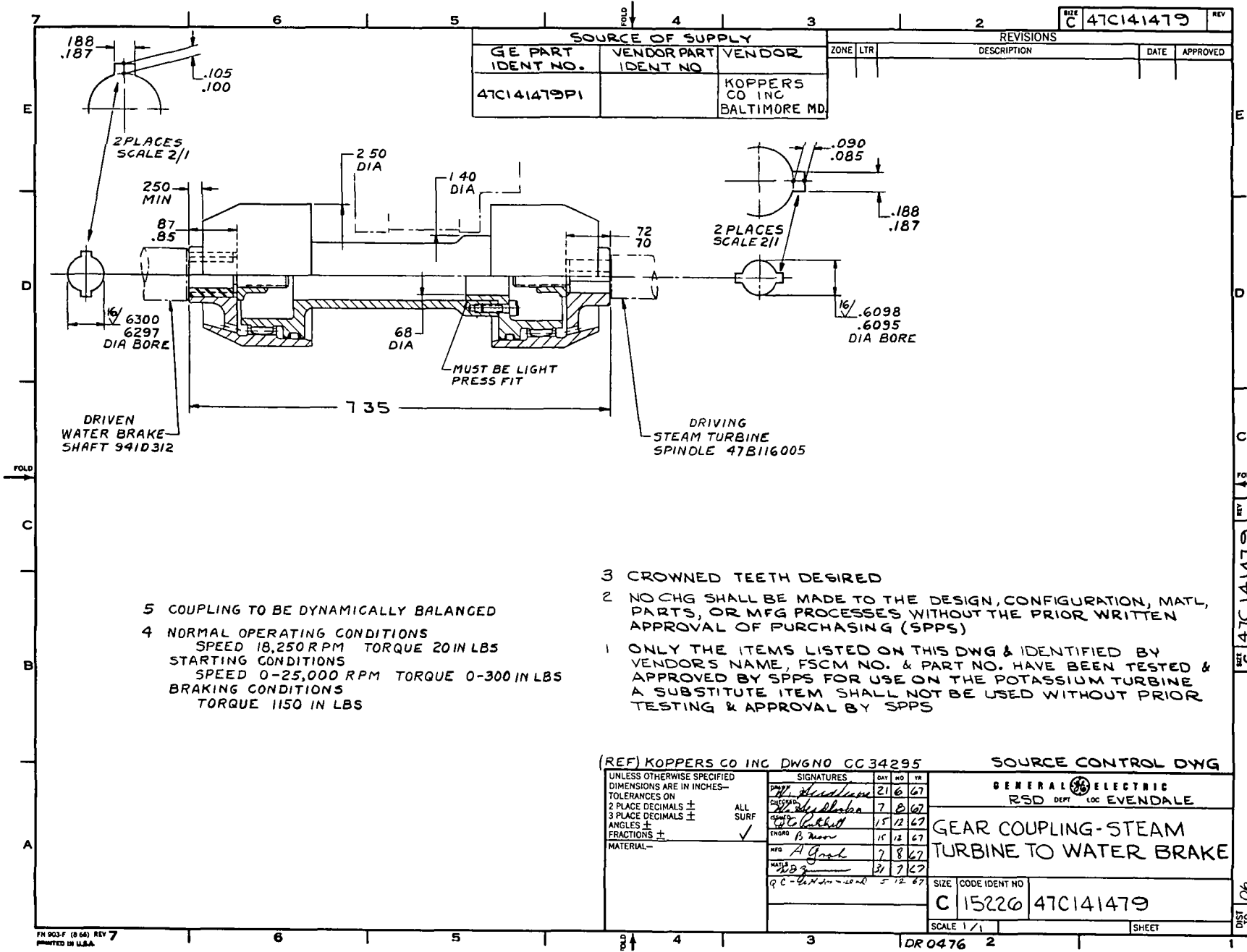


Figure 40a. Alternate Flexible Gear Tooth (Koppers) Coupling Connecting Water-Brake and Steam Turbine. (47C141479)

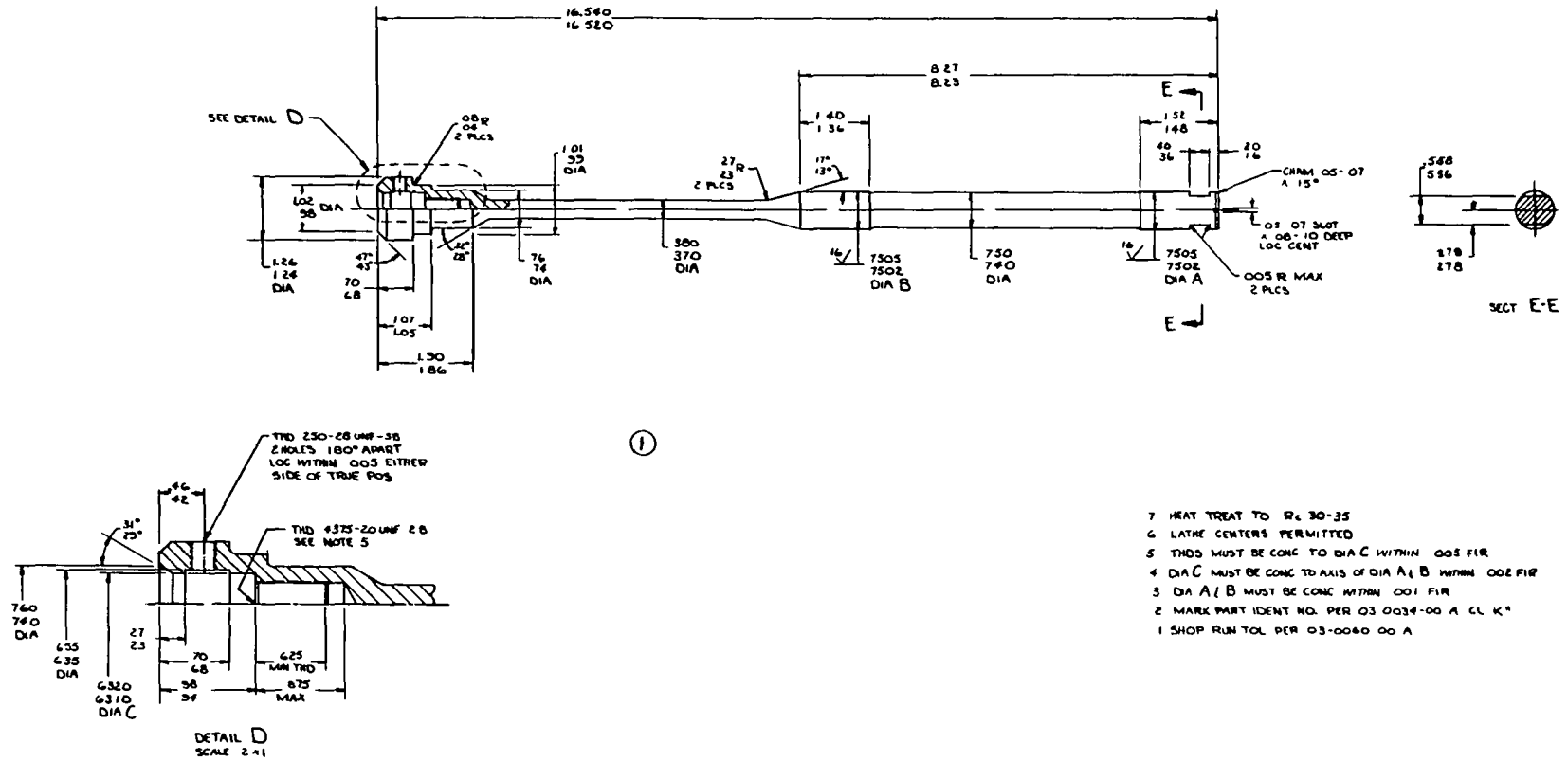


Figure 40b. Spindle, Steam Turbine. (941D394)

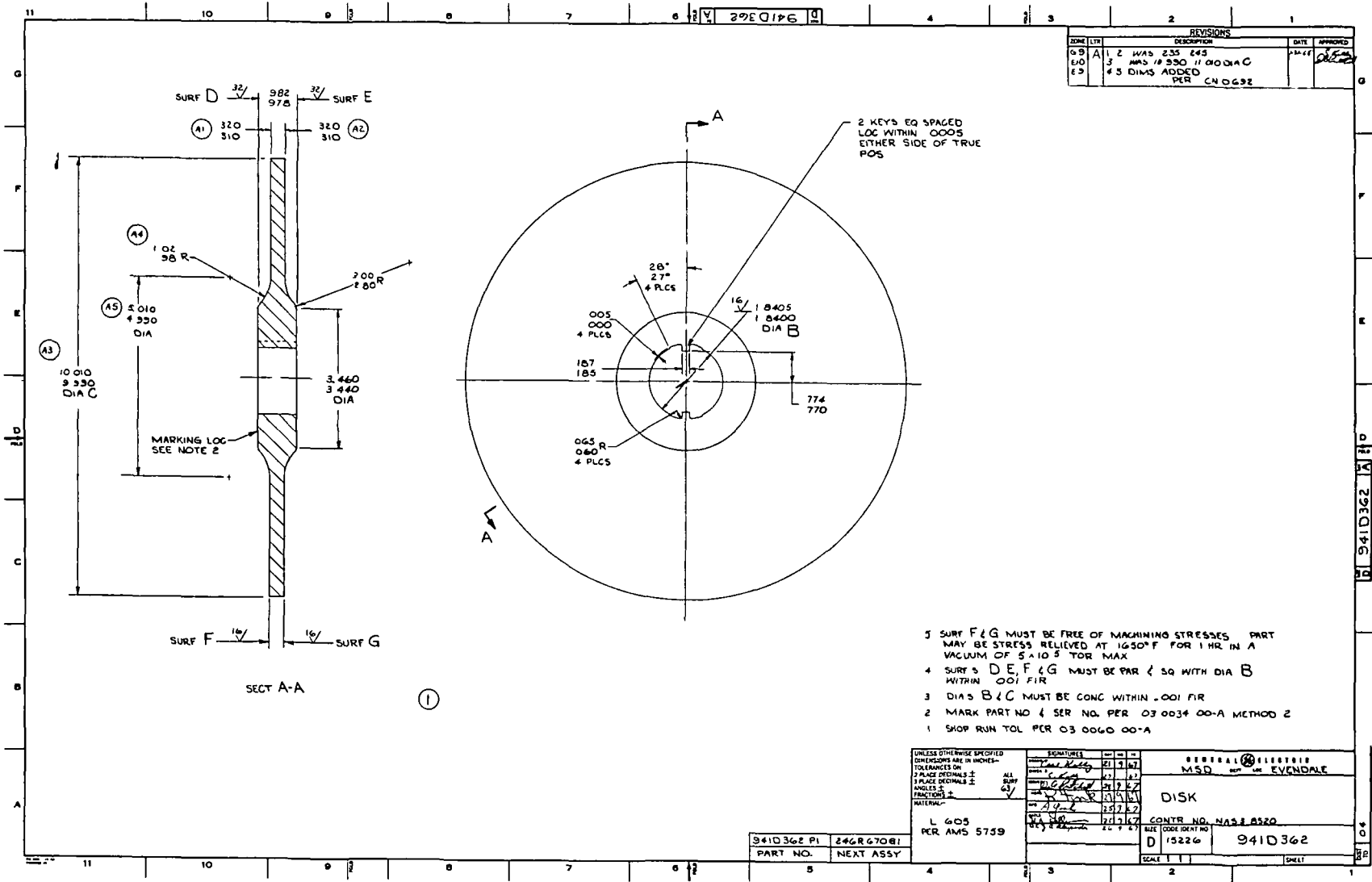


Figure 41. Disk of the Caliper Disk Brake. (Dwg. No. 941D362)

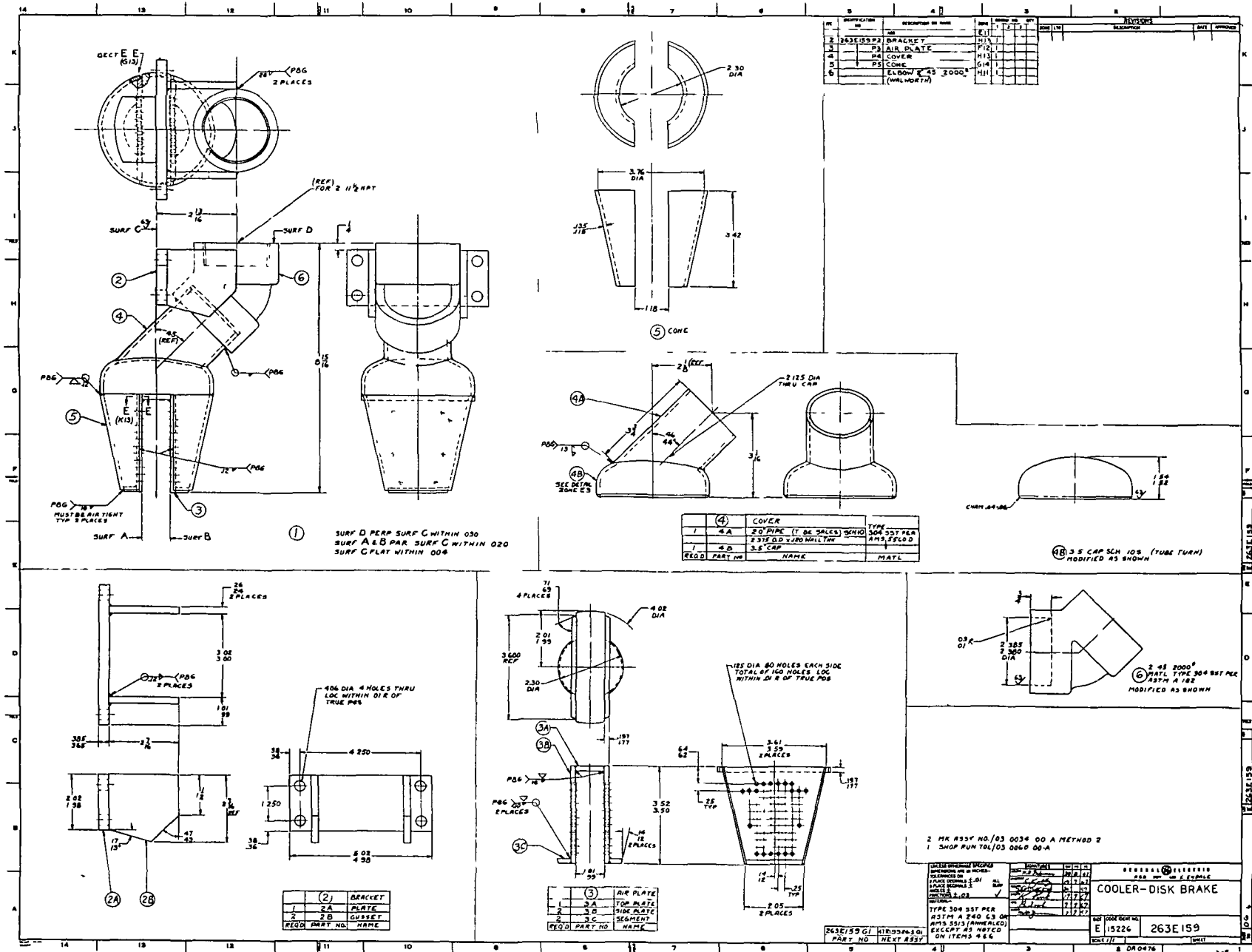


Figure 42. Cooling Air Manifold and Distributor for Face of Caliper Disk Brake. (263E159)

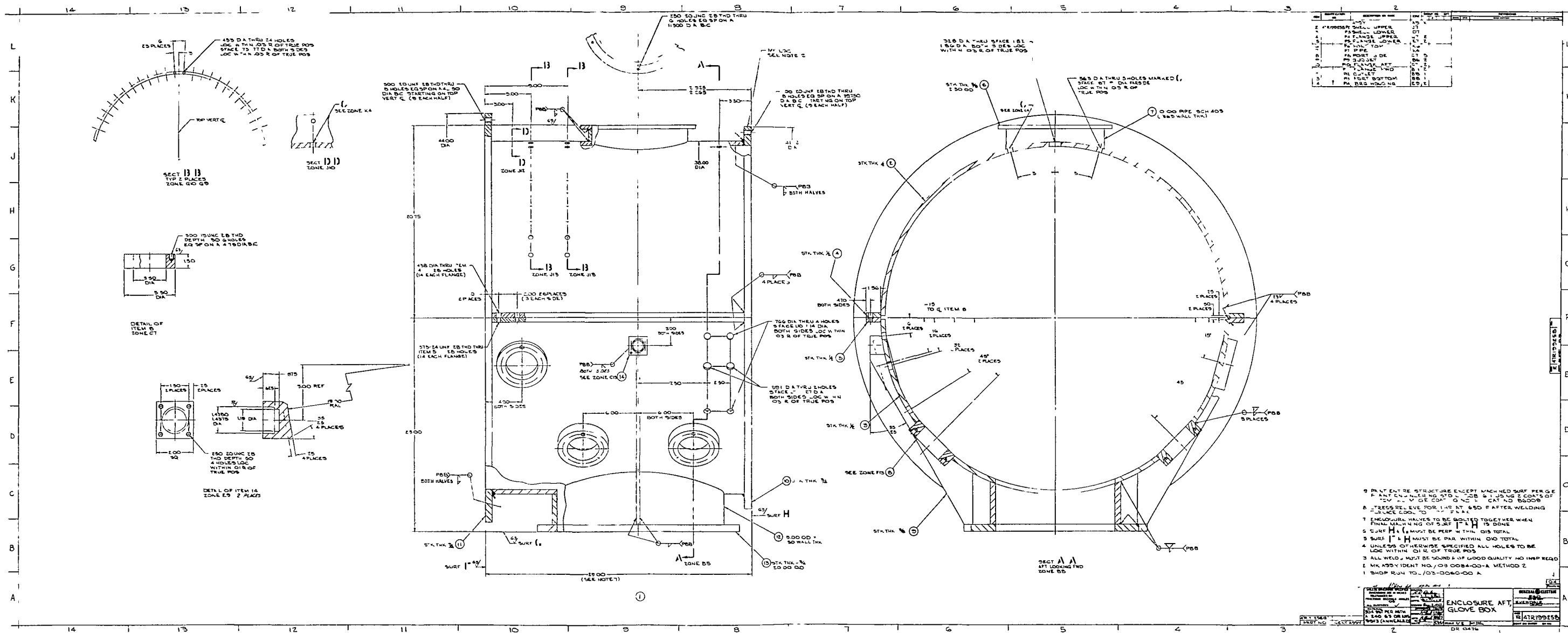


Figure 43. Aft Enclosure Surrounding the 3 Stage Potassium Test Turbine. (47R199258)

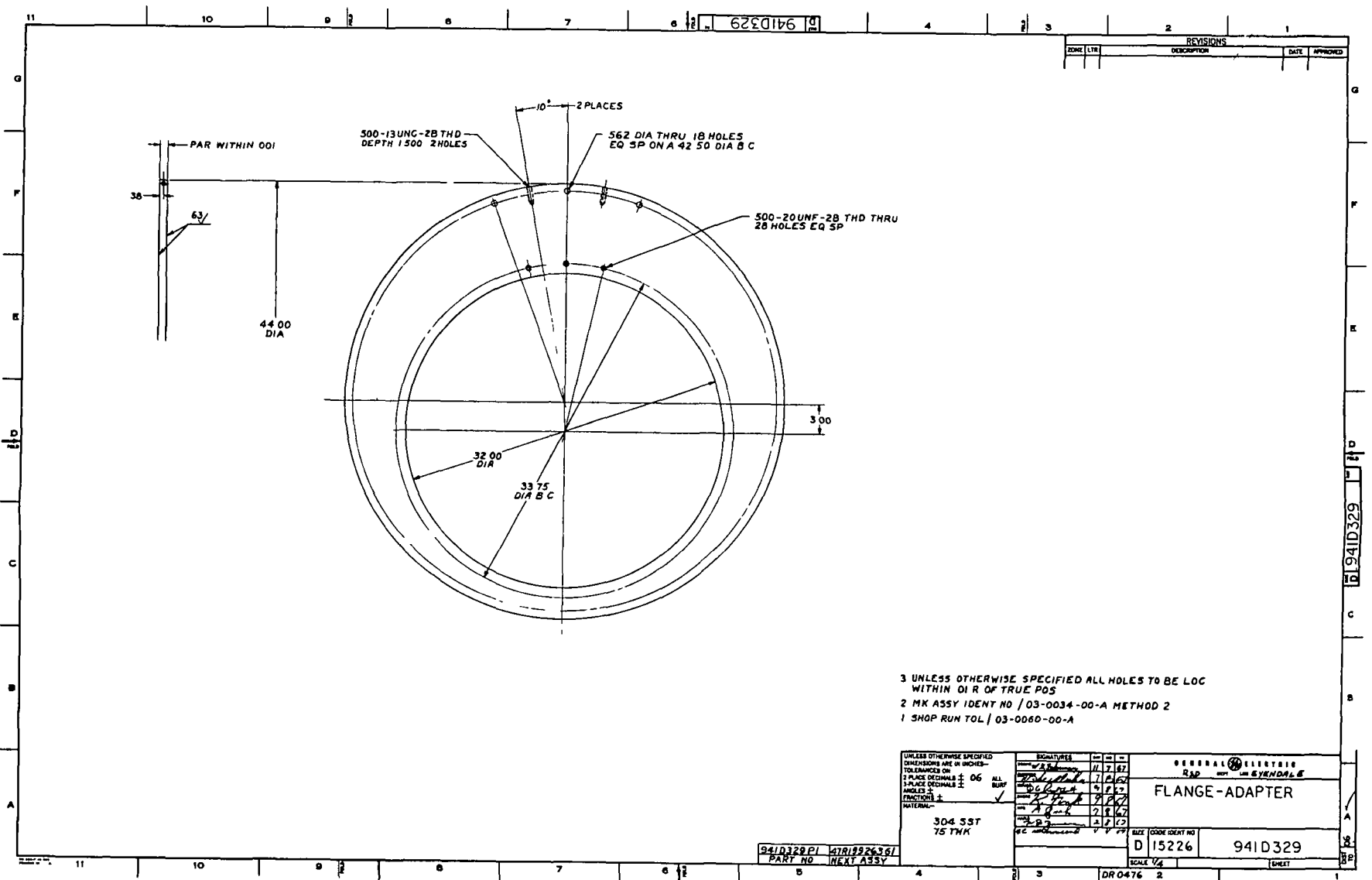


Figure 44. Eccentric Adapter Flange for Connecting Power Train Support Structure to Environmental Chamber Enclosing 3 Stage Test Turbine. (941D329)

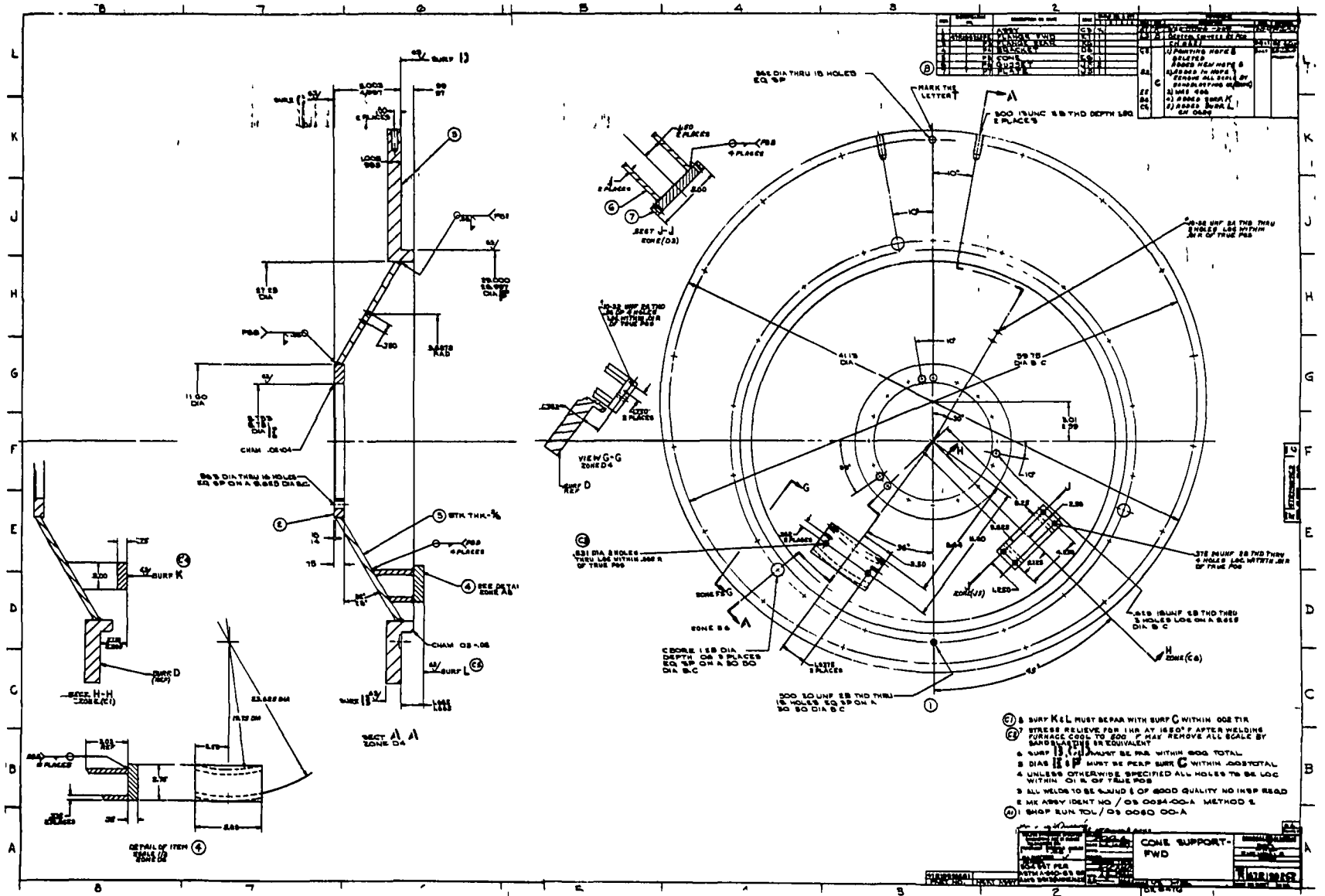


Figure 45. Forward Conical Support for Mounting Potassium Turbine to Environmental Chamber. (47R199262)

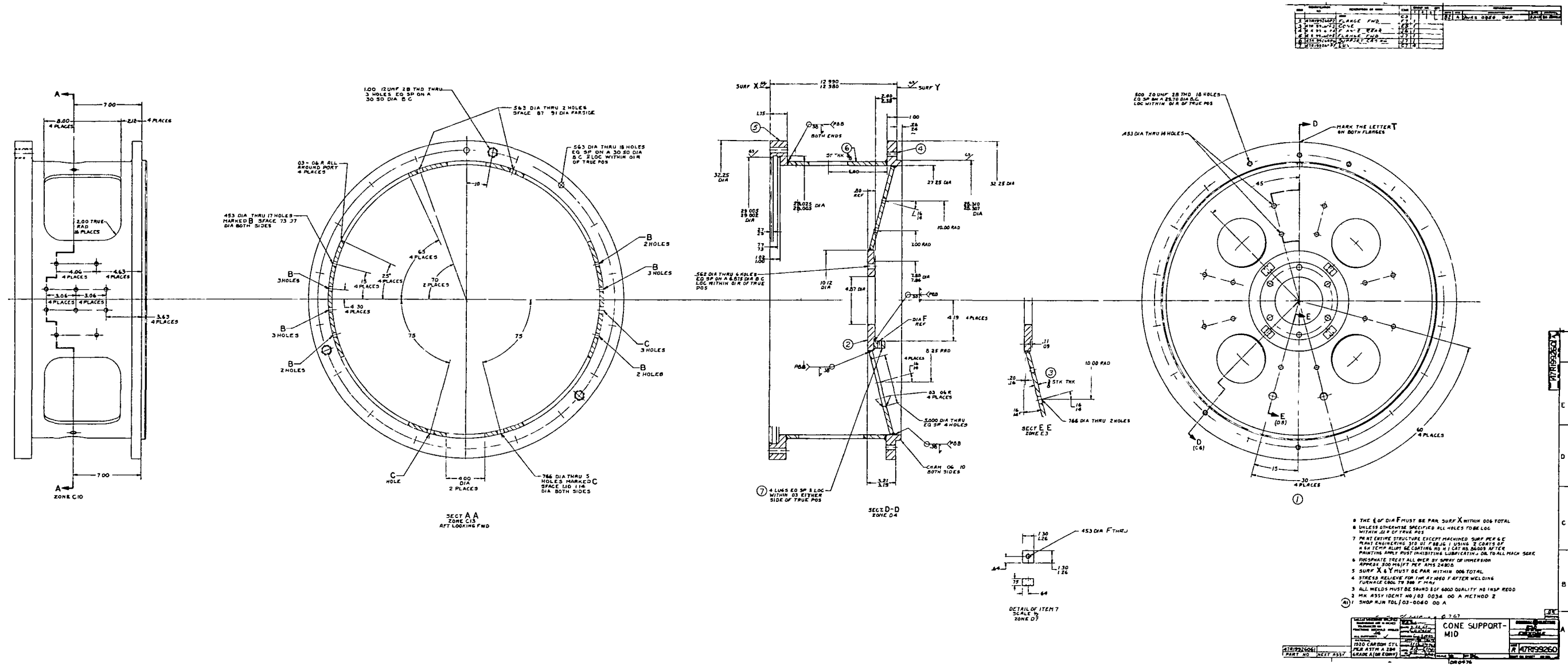


Figure 46. Conical Support Structure for Mounting Water Brake. (47R199260)

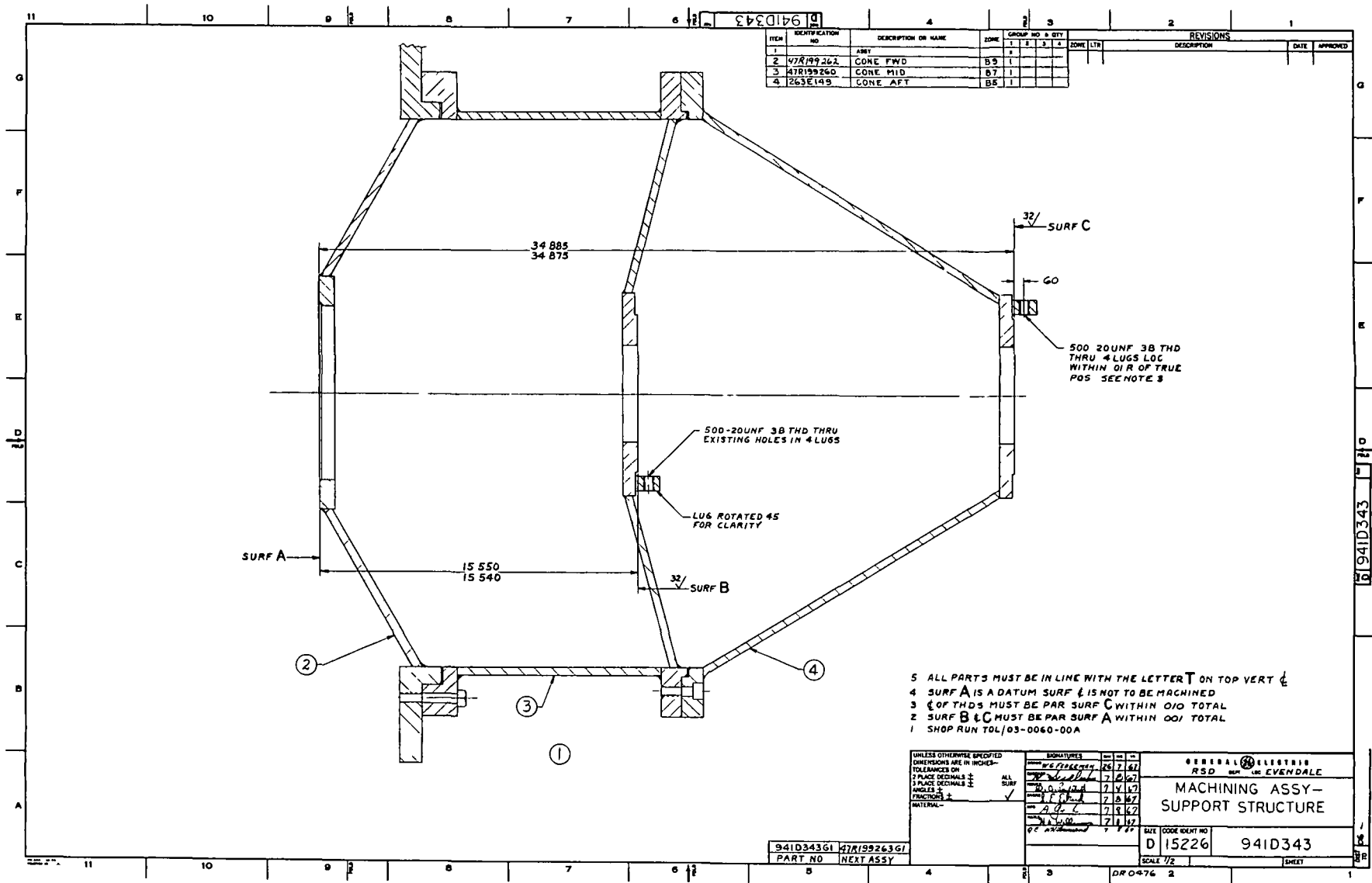


Figure 48. Subassembly of Power Train Conical Support Structure for Machining Mounting Faces Parallel to Each Other. (941D343)

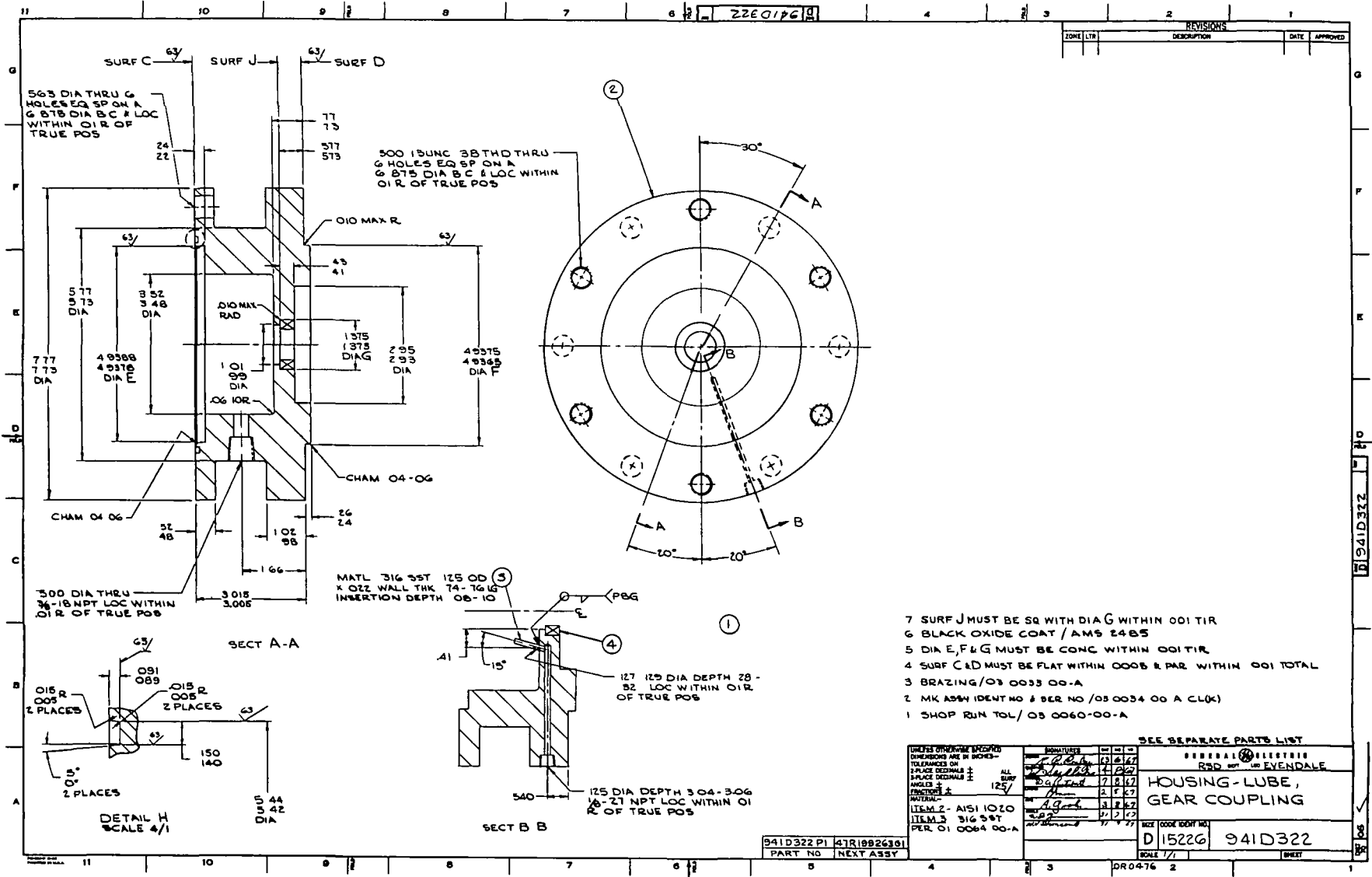


Figure 52. Housing for Supplying and Catching Overflow Lubricant to Aft End of Flexible Gear Coupling Between Water Brake and Steam Turbine. (941D322)

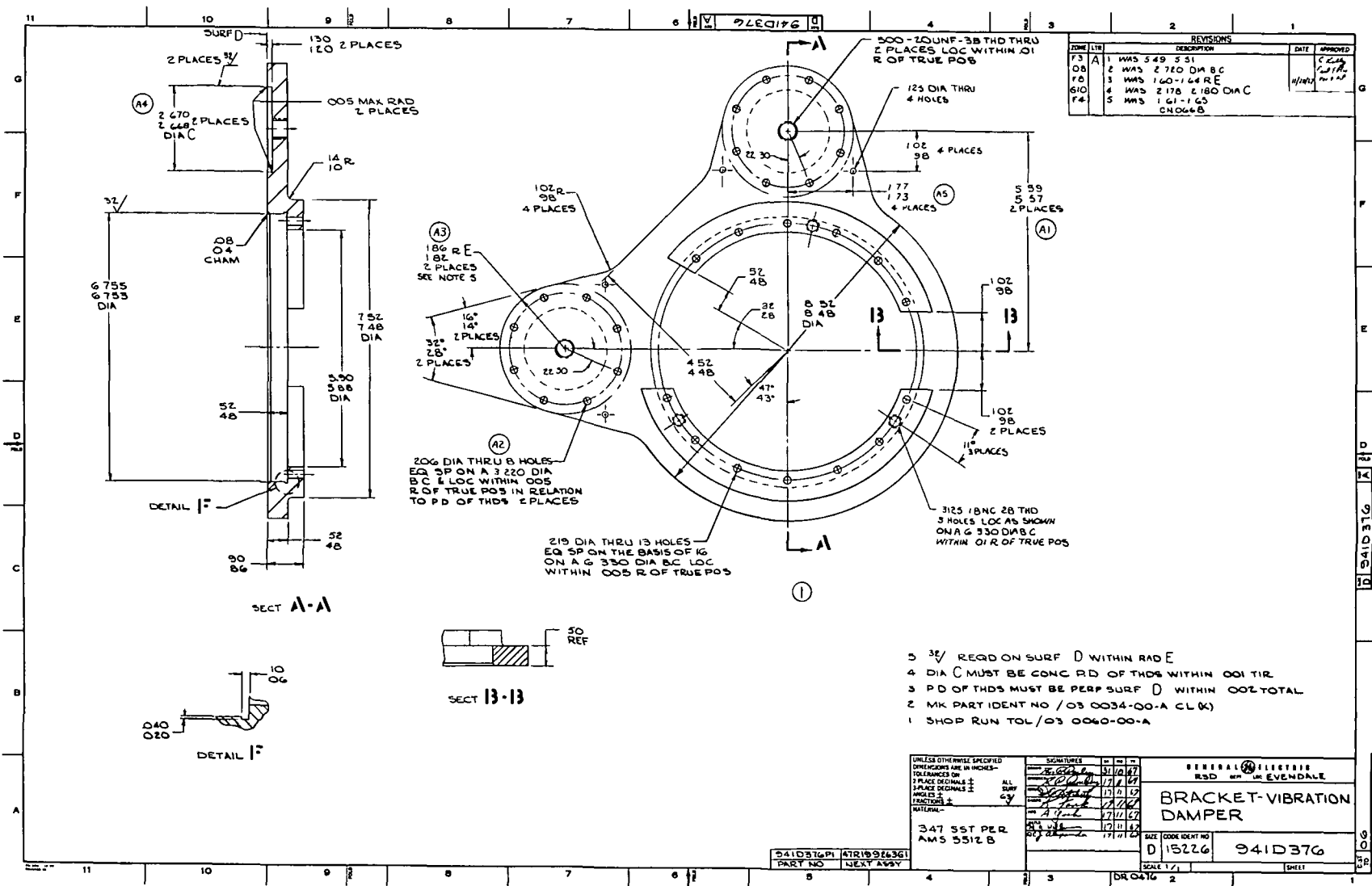


Figure 53. Bracket for Mounting Vibration Damper to Water Brake. (941D376)

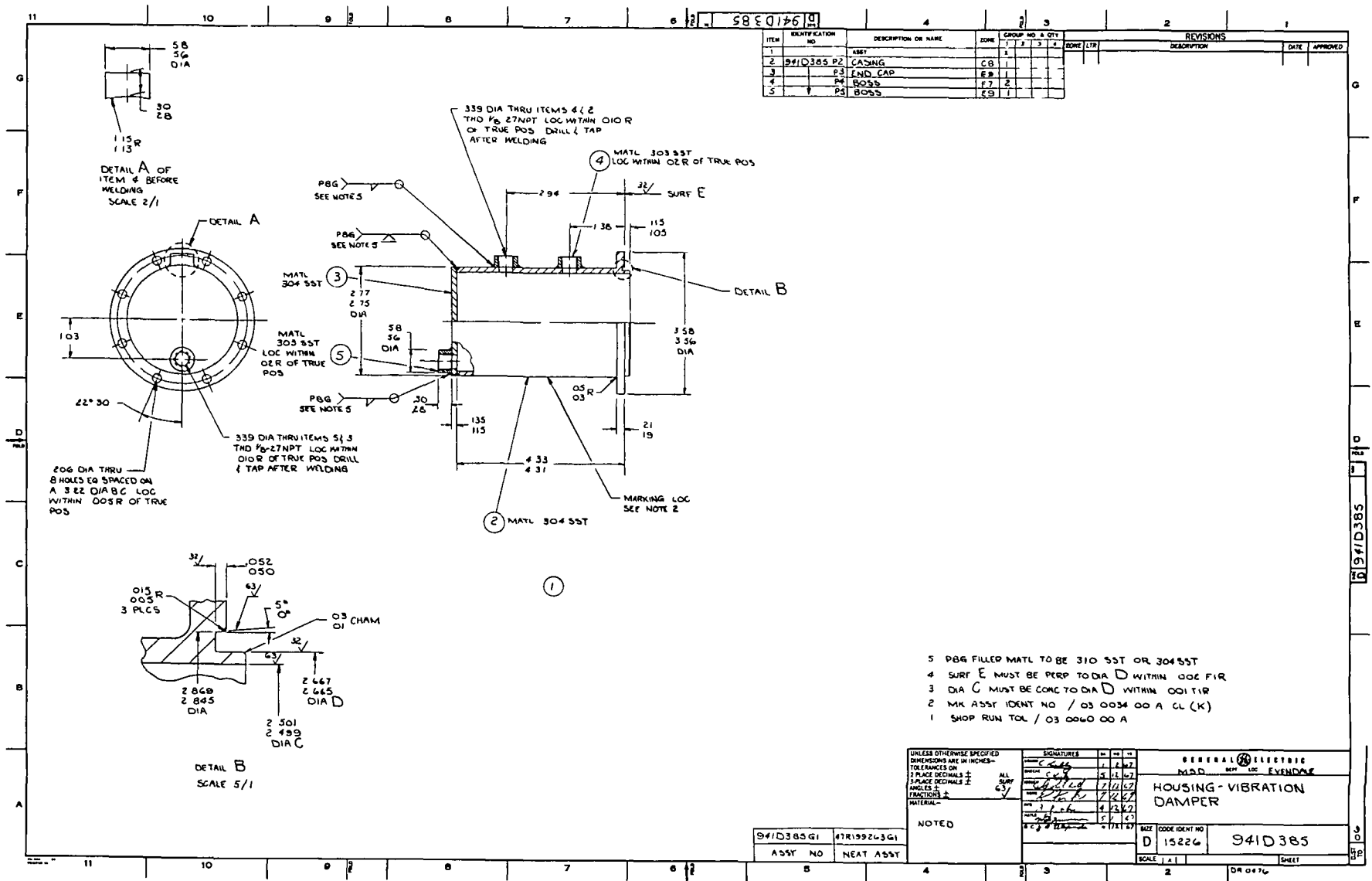


Figure 54. Housing for Water-Brake Vibration Damper. (47C141538)

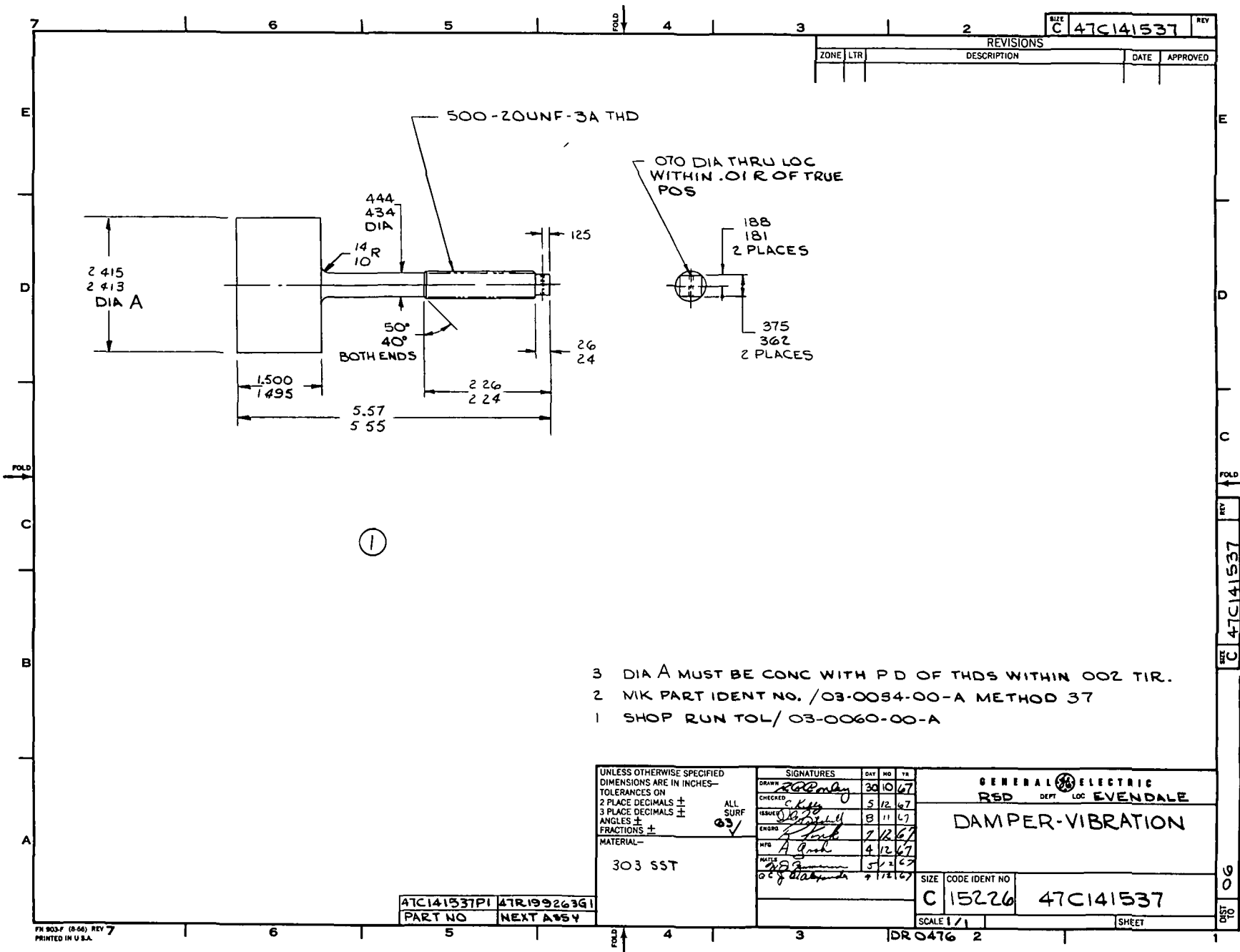


Figure 55. Cantilevered Mass - Beam for Water-Brake Vibration Damper. (47C141537)

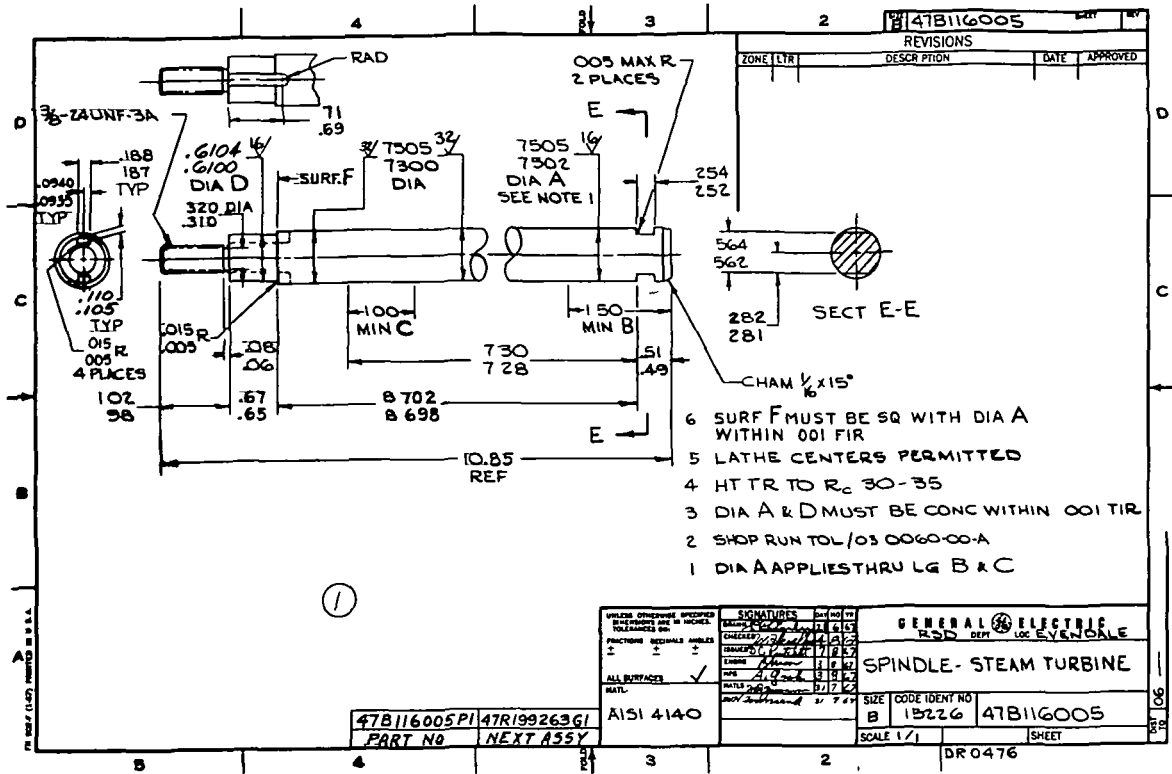


Figure 56. Shaft for Steam Drive Turbine. (47B116005)

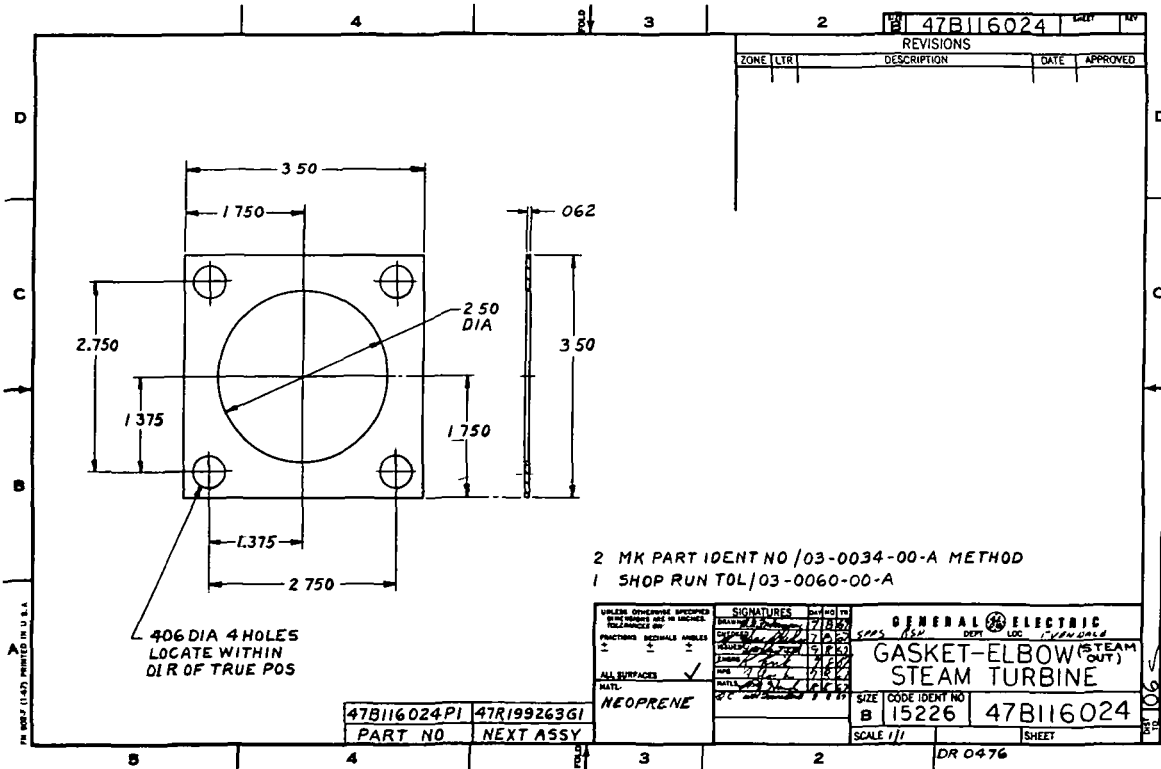
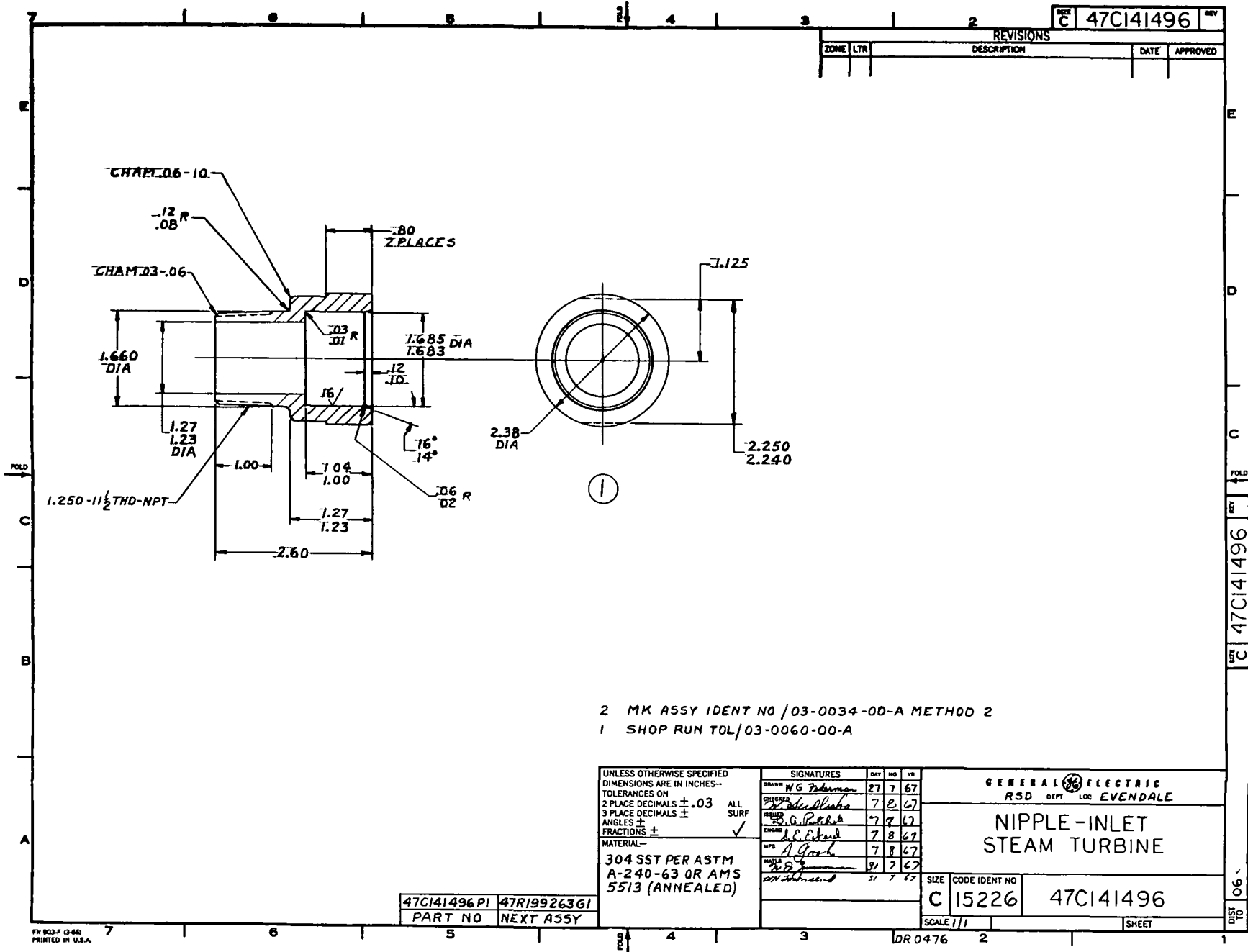


Figure 57. Gasket for Steam Outlet Line Attachment to Aft Support Structure Plate. (47B116024)



2 MK ASSY IDENT NO /03-0034-00-A METHOD 2
 1 SHOP RUN TOL/03-0060-00-A

Figure 61. Adapter Nipple to Receive Flexible Inlet Pipe Supplying Steam to the Steam Drive Turbine. (47C141496)

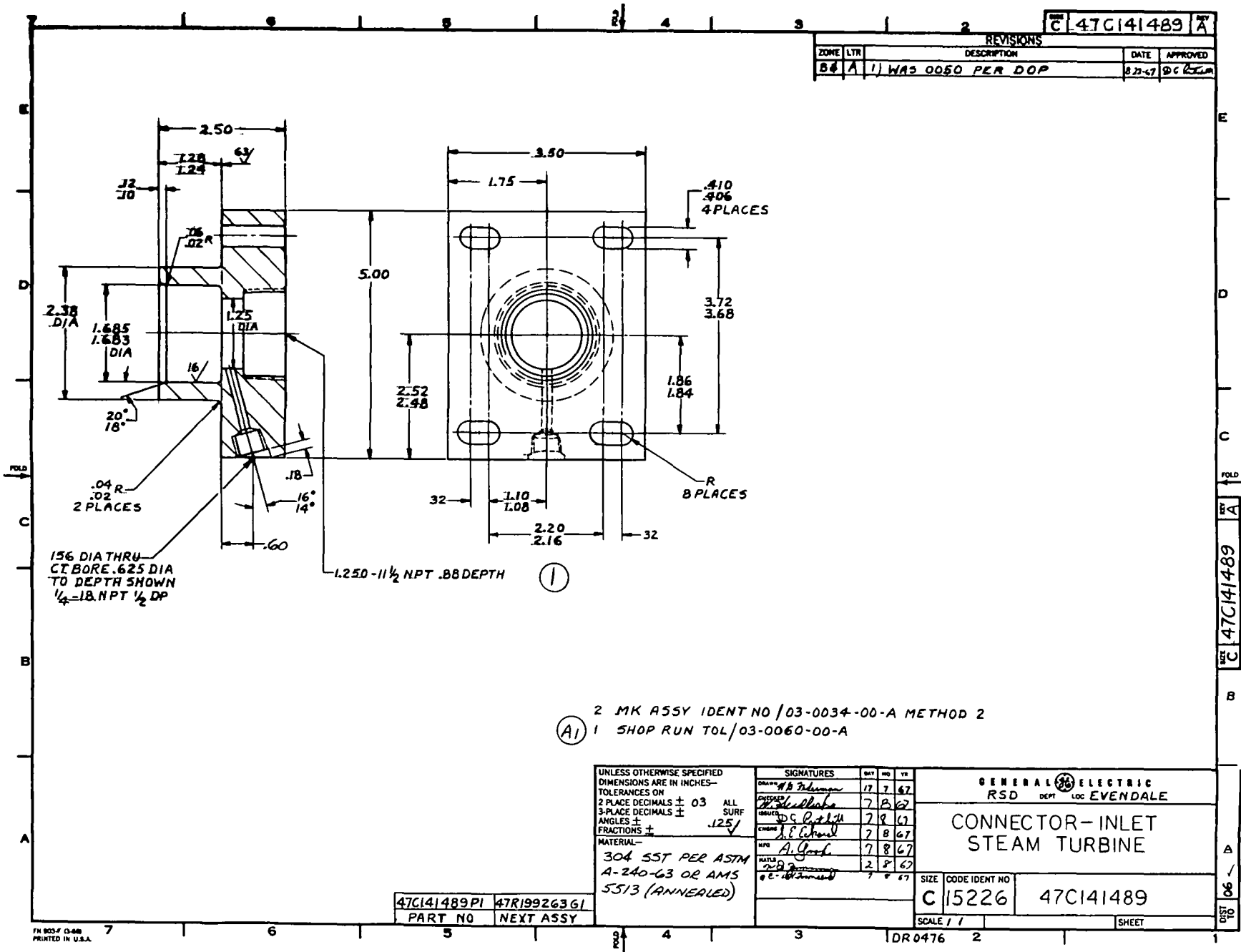


Figure 62. Connector for Steam Inlet Line at Aft Plate of Power Train Support Structure. (47C141489)

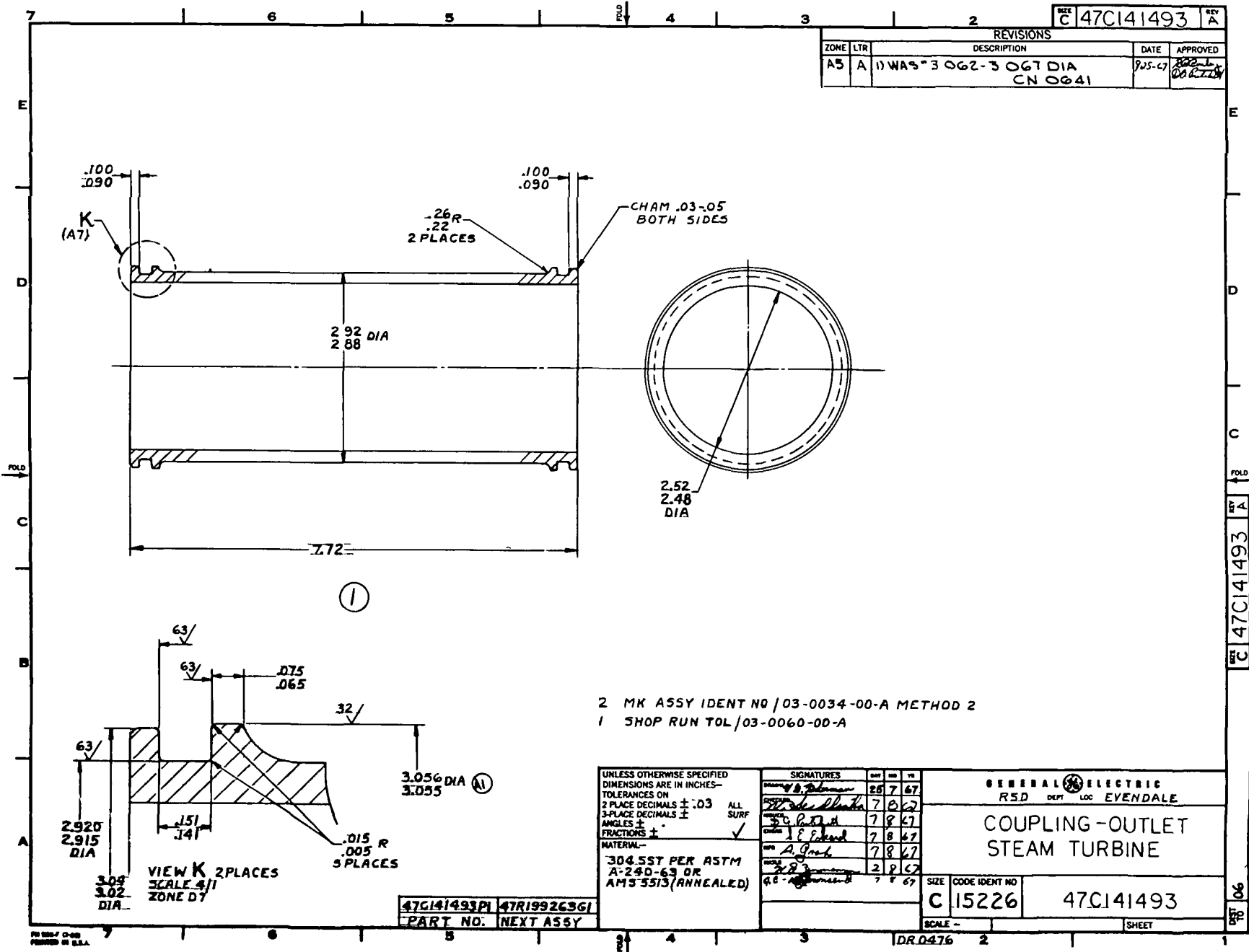


Figure 63. Flexible Coupling for Steam Exhaust from Steam Drive Turbine.
(47C141493)

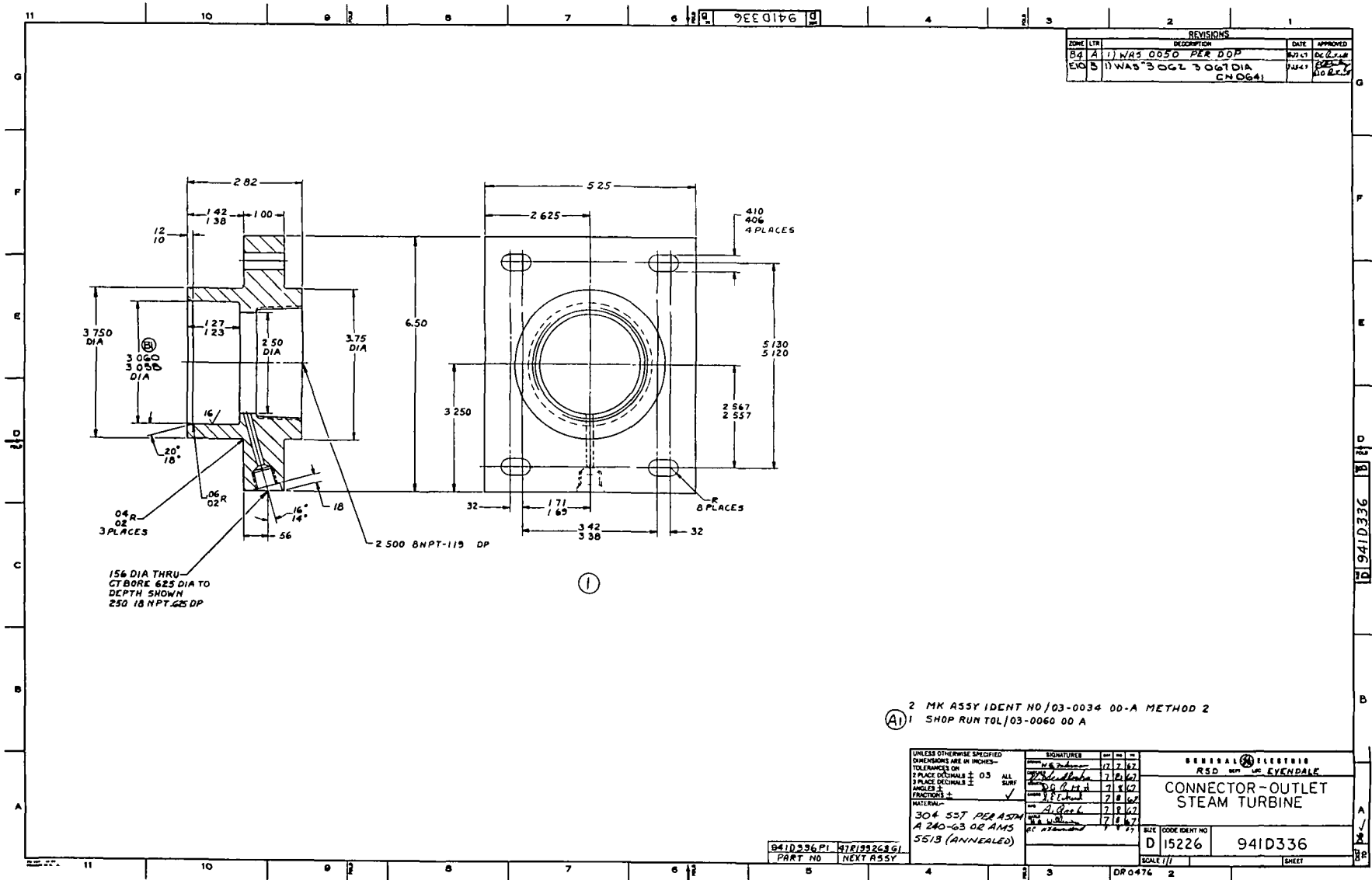


Figure 64. Connector for Steam Exhaust Line at Aft Plate of Power Train Support Structure. (941D336)

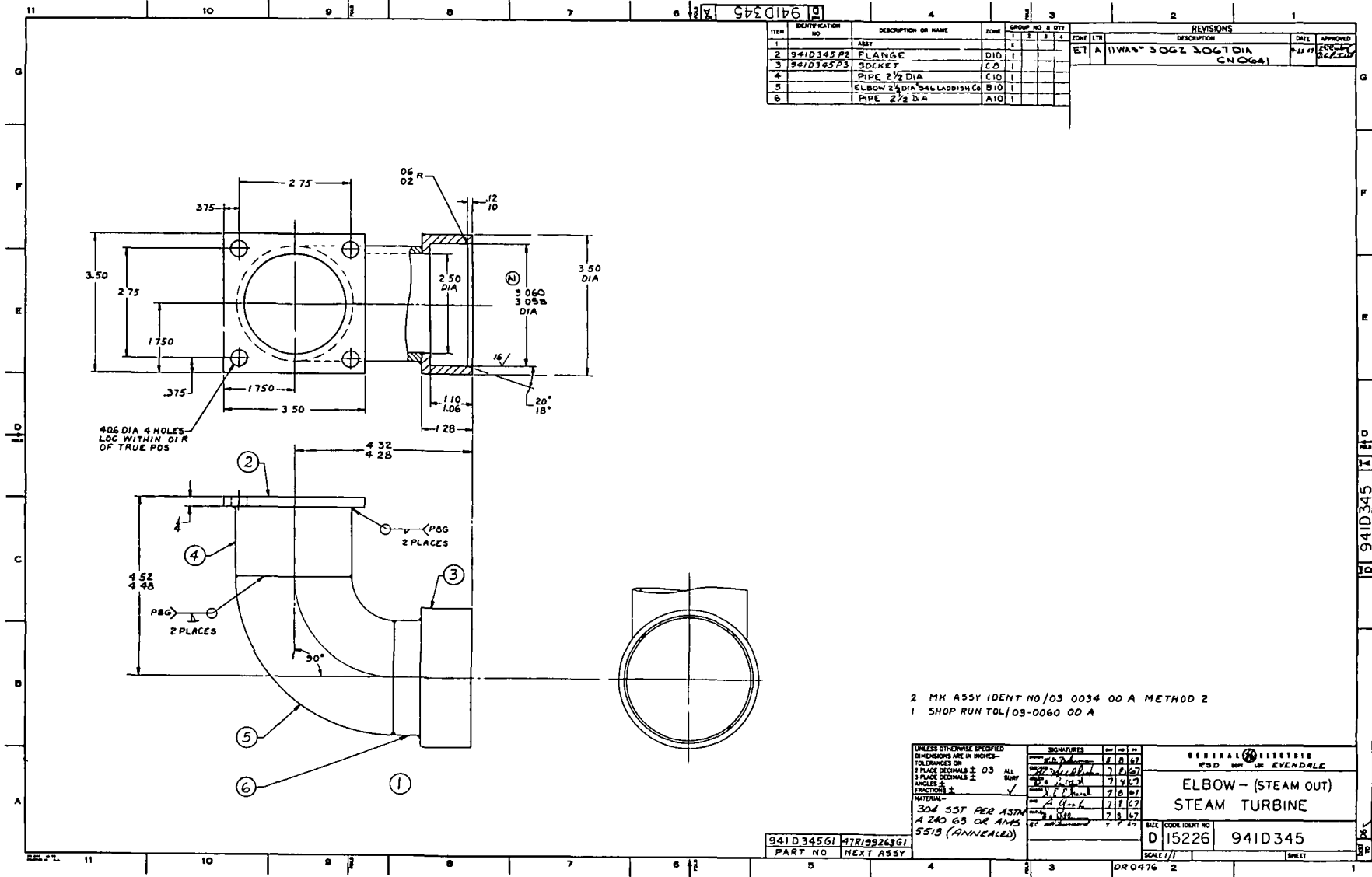


Figure 65. Connector for Steam Exit Line Attached to Outlet of Steam Turbine. (941D345)

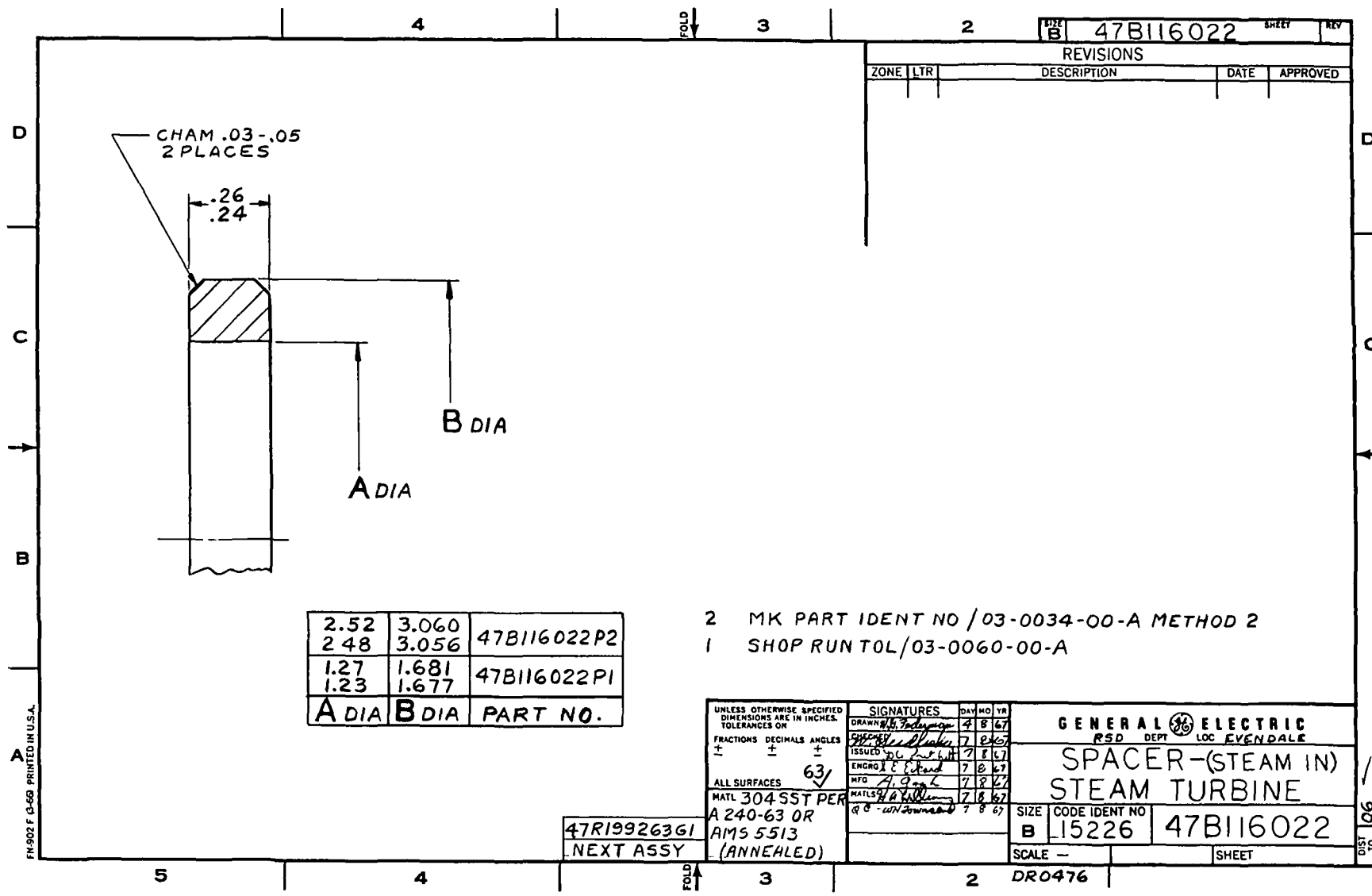


Figure 66. Steam Turbine Inlet Line Spacer. (47B116022)

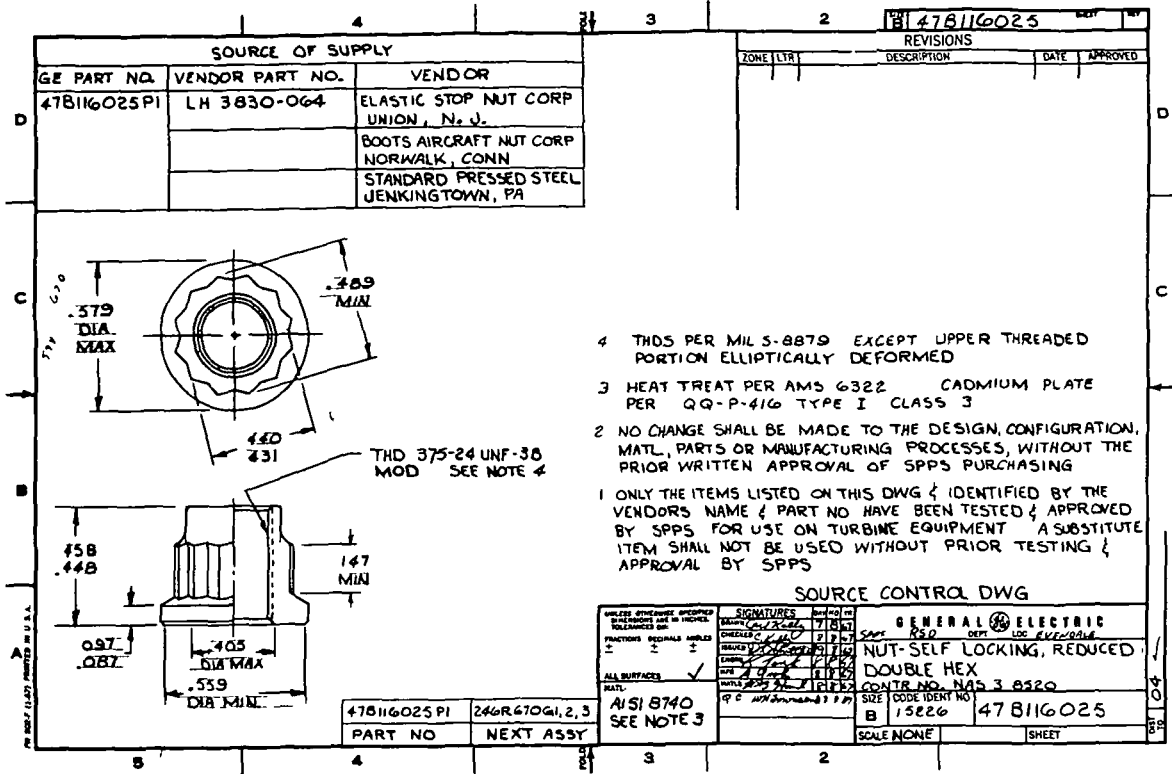


Figure 67. Self-Locking Nut Used to Retain Flexible Membrane Coupling on the 3 Stage Test Turbine Shaft. (47B116025)

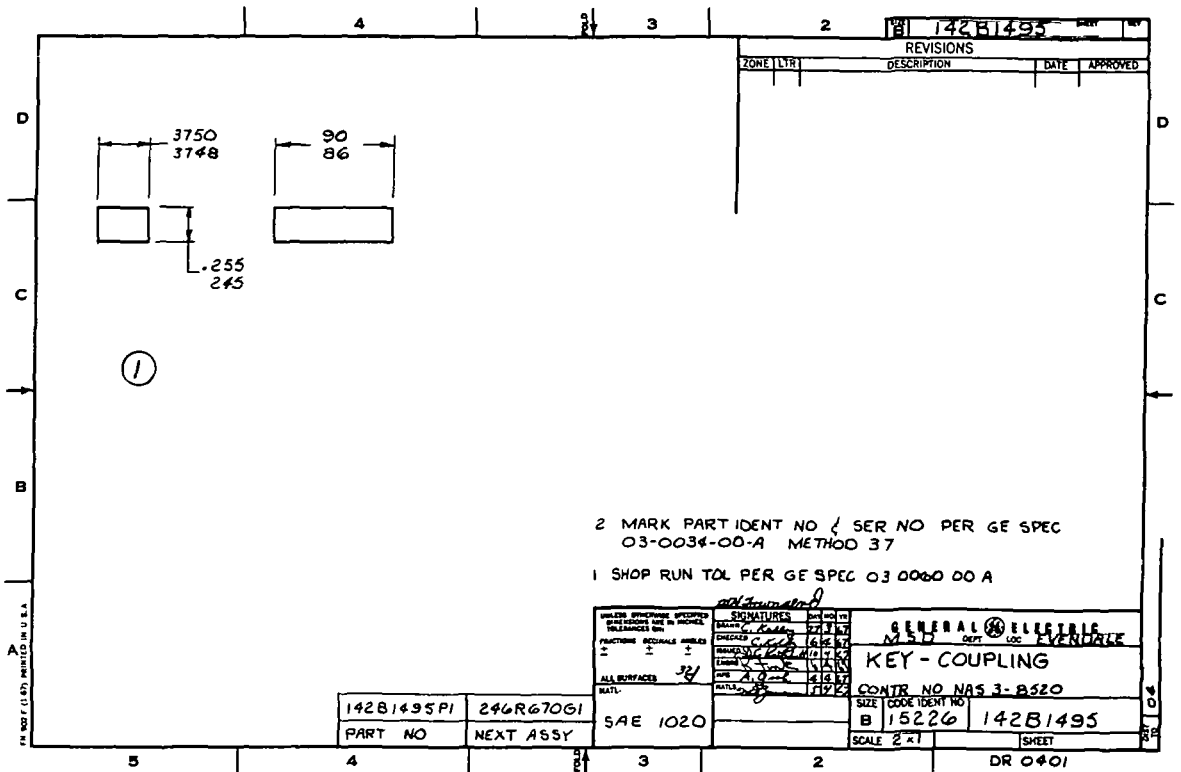


Figure 68. Key Used to Key the Flexible Membrane Coupling to 3 Stage Test Turbine Shaft. (142B1495)

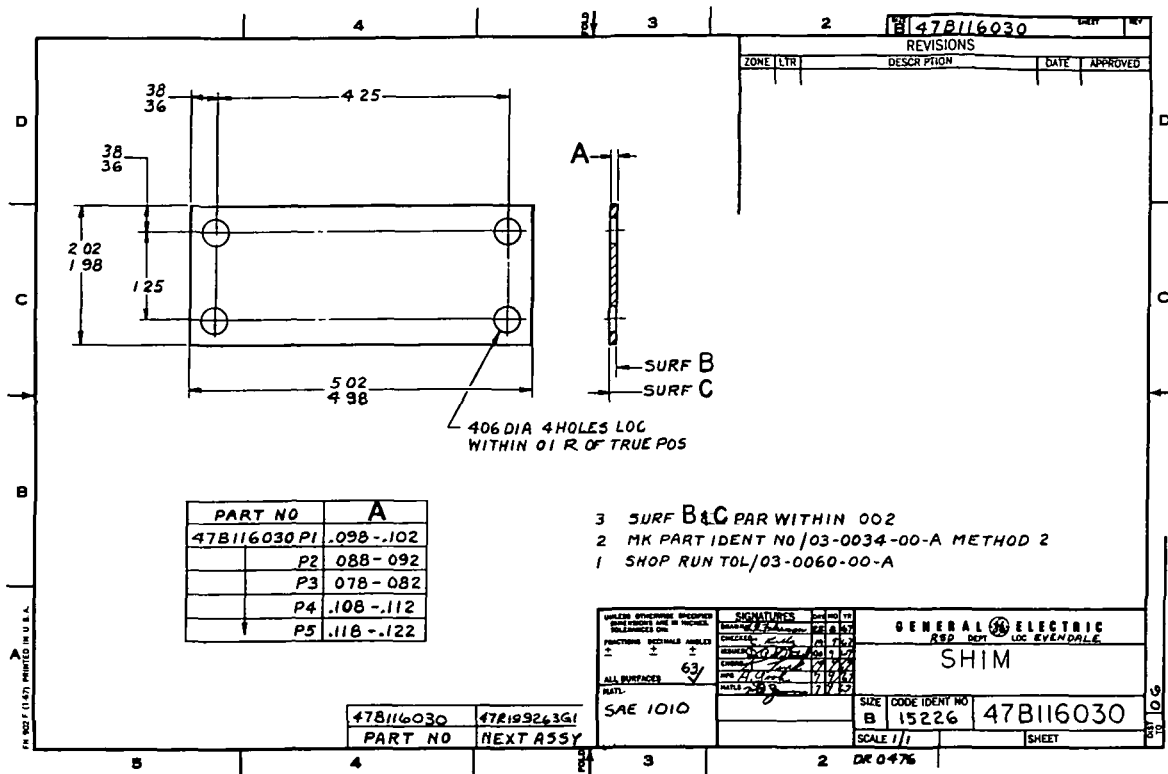


Figure 69. Shim Used to Axially Position the Disk Cooler Straddling Caliper Disk Brake. (47B116030)

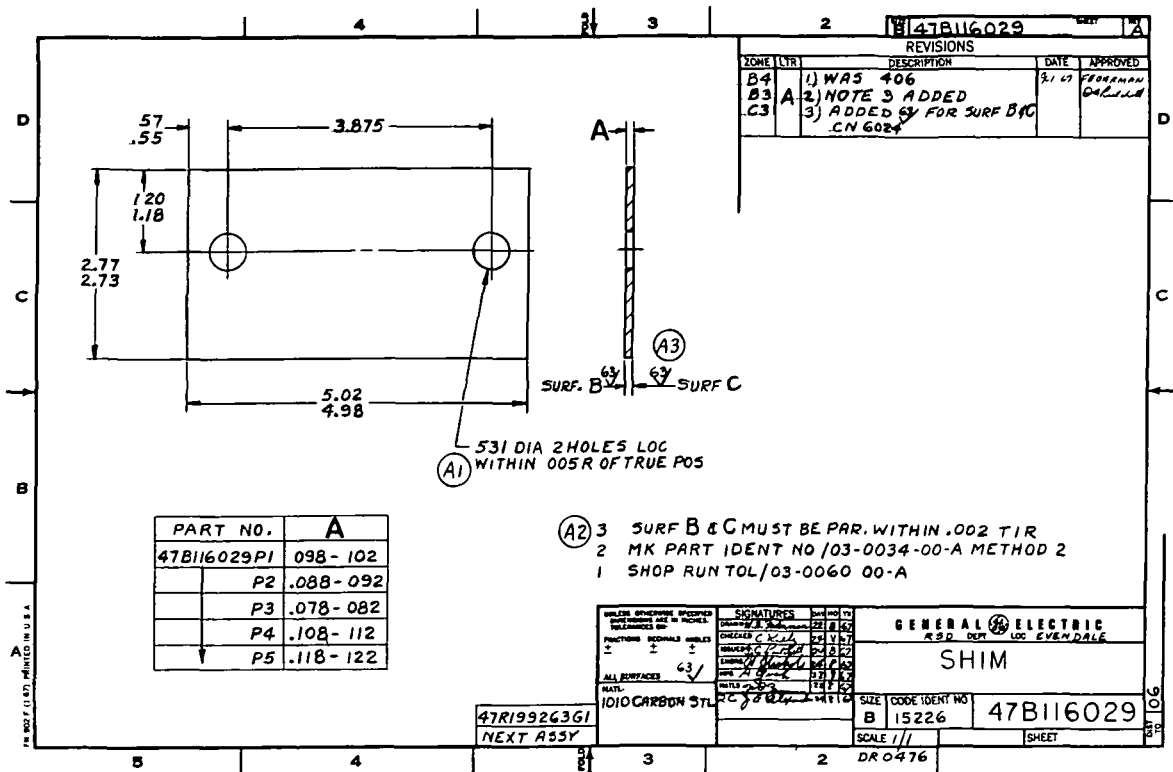


Figure 70. Shim Used to Axially Position the Puck Mounting Bracket Over the Disk of the Caliper Disk Brake. (47B116029)

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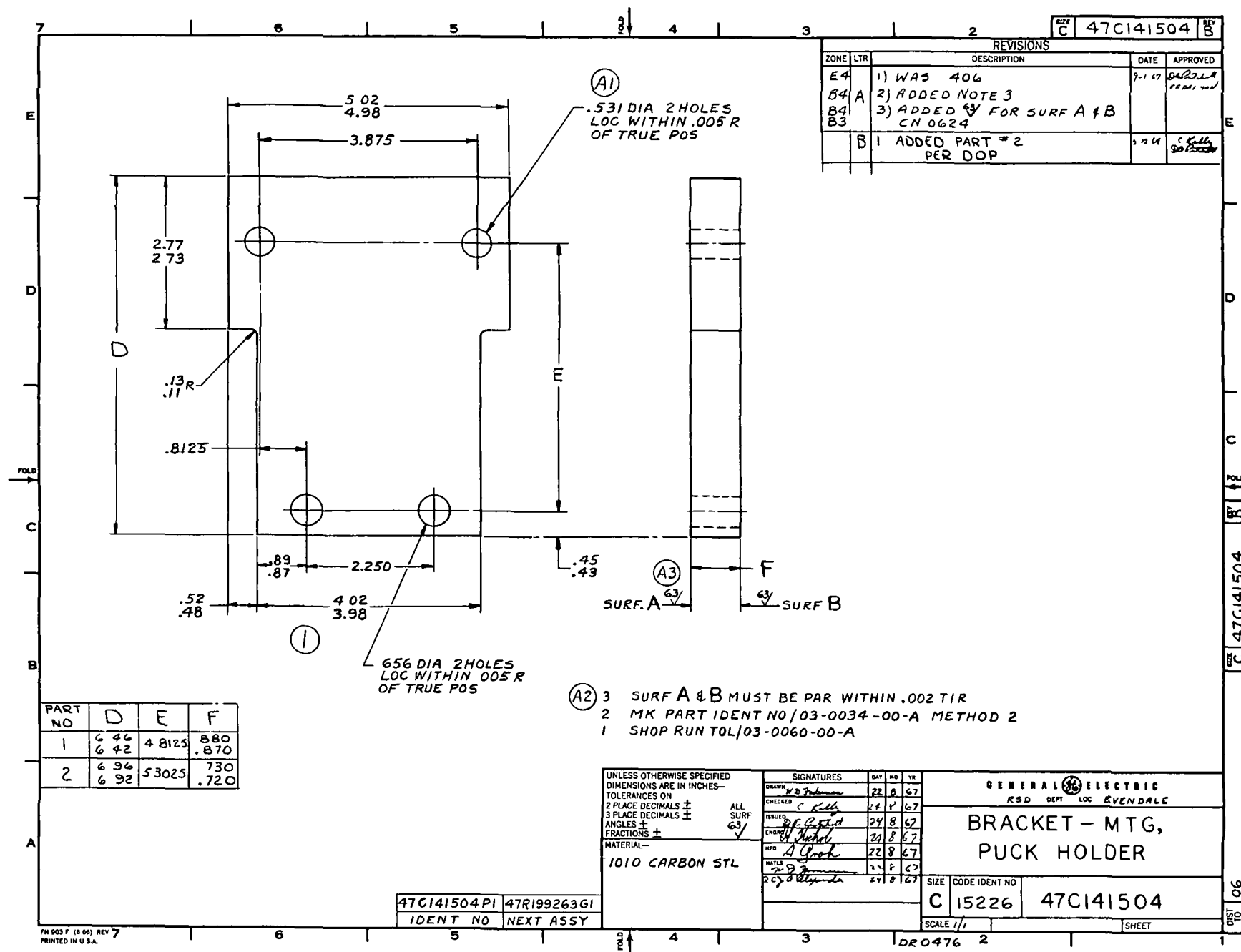


Figure 71. Bracket Supporting Puck Holders for Caliper Disk Brake. (47C141504)

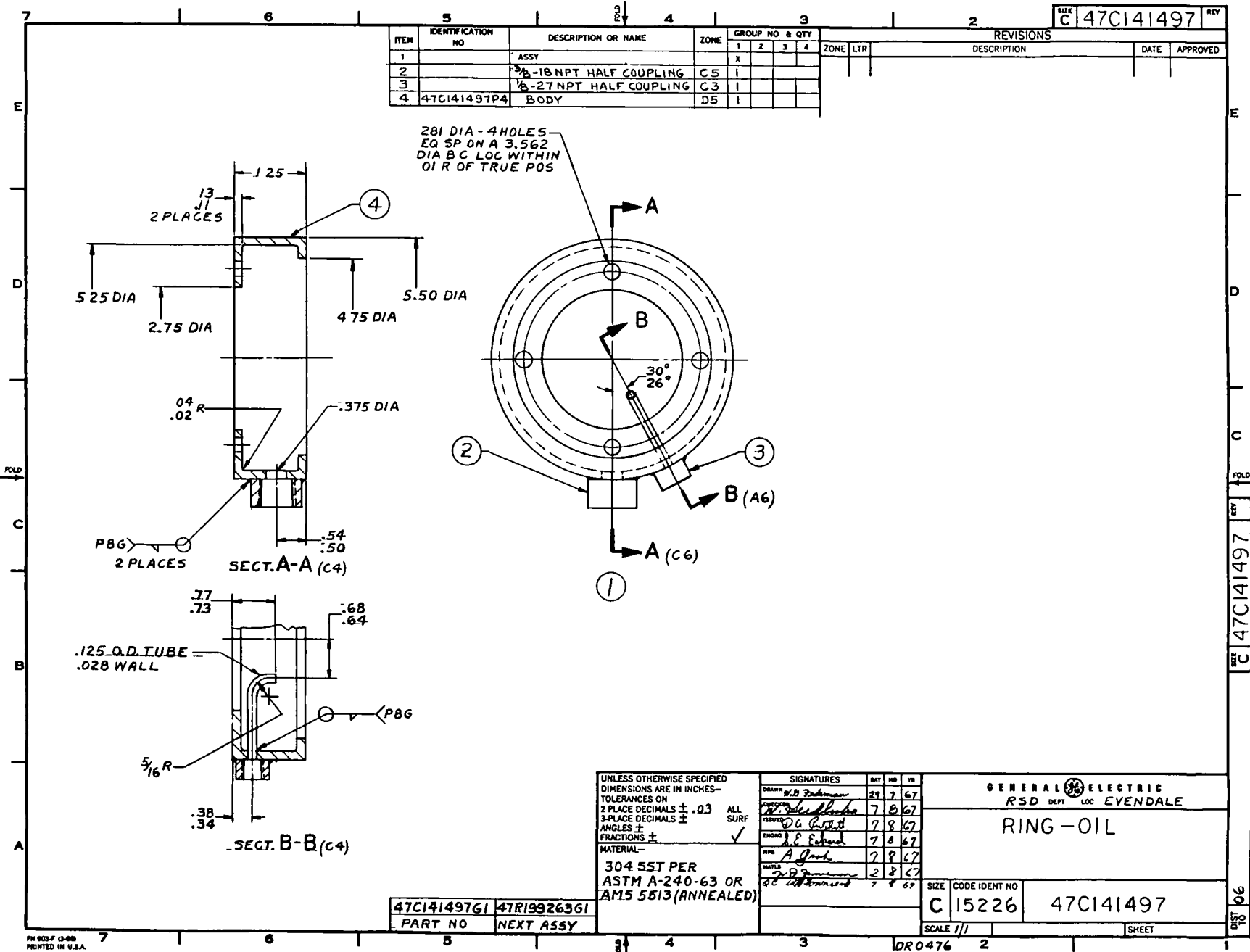


Figure 73. Ring for Supplying and Catching Overflow Lubricant to Forward End of Flexible Gear Coupling Between Water-Brake and Steam Turbine. (47C141497)

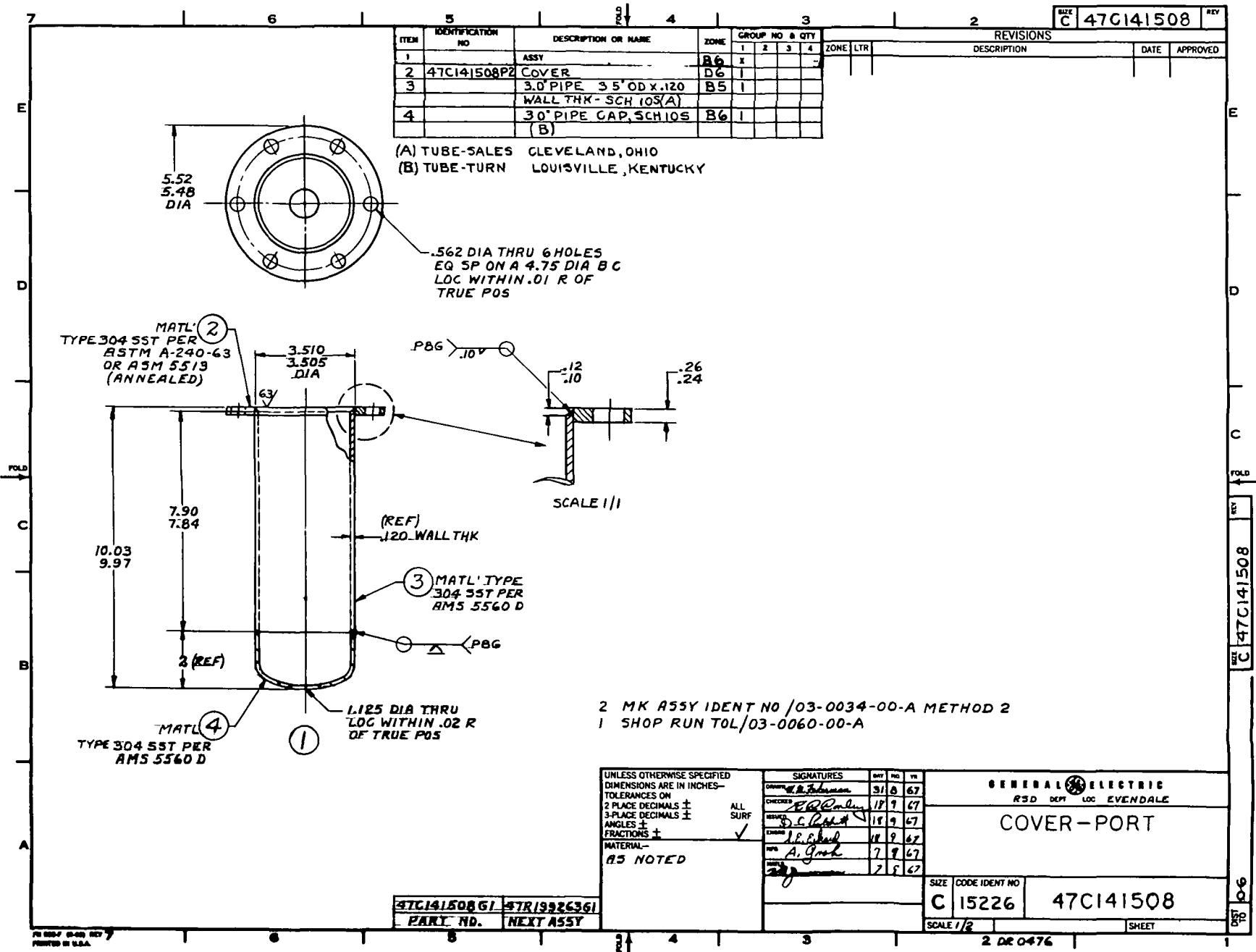


Figure 74. Cover for Porthole on 3 Stage Turbine Environmental Chamber.
(47C141508)

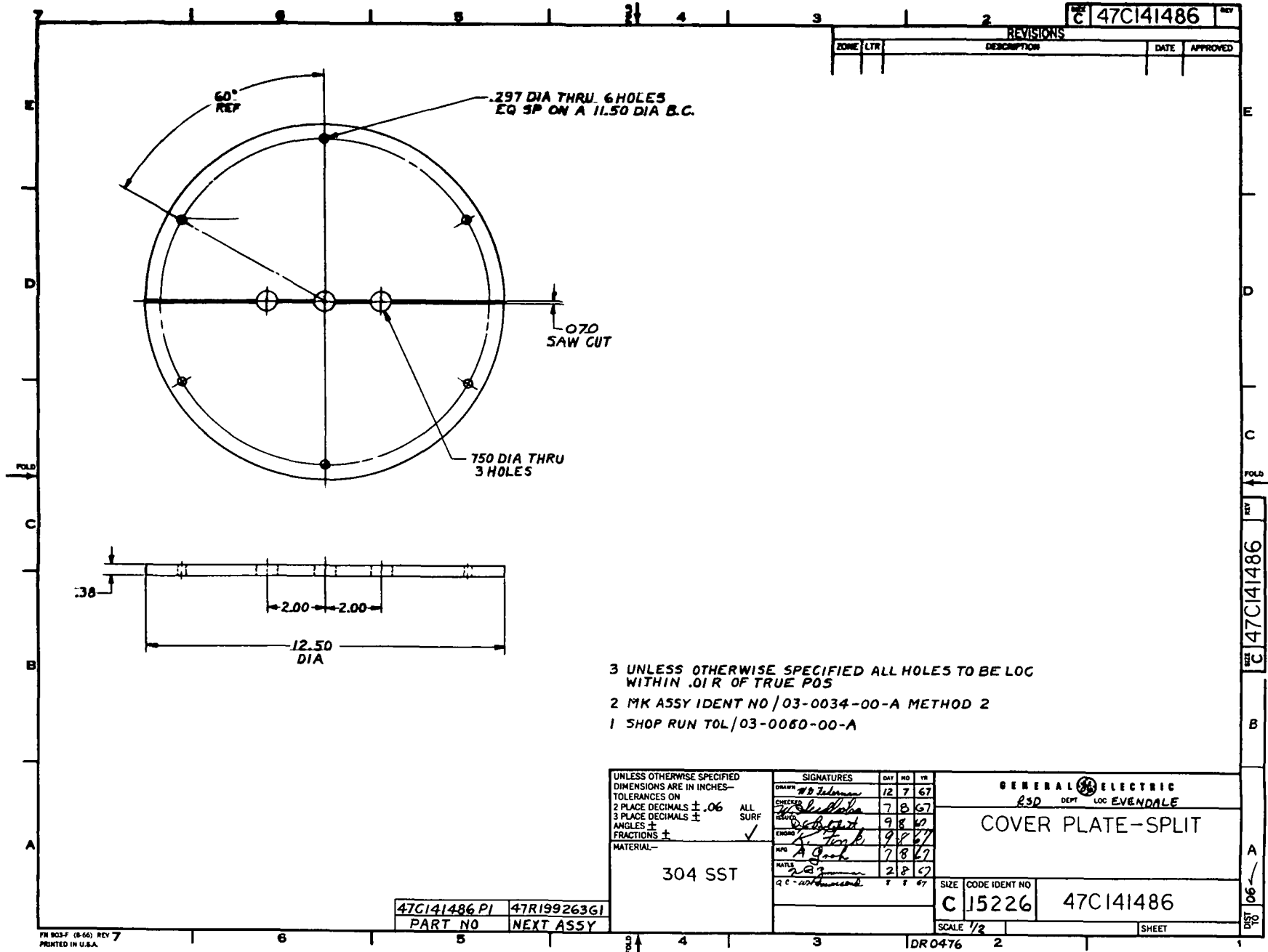


Figure 75. Split Cover Plate Used on Test Turbine Environmental Chamber Porthole for Passing Instrumentation Lines. (47C141486)

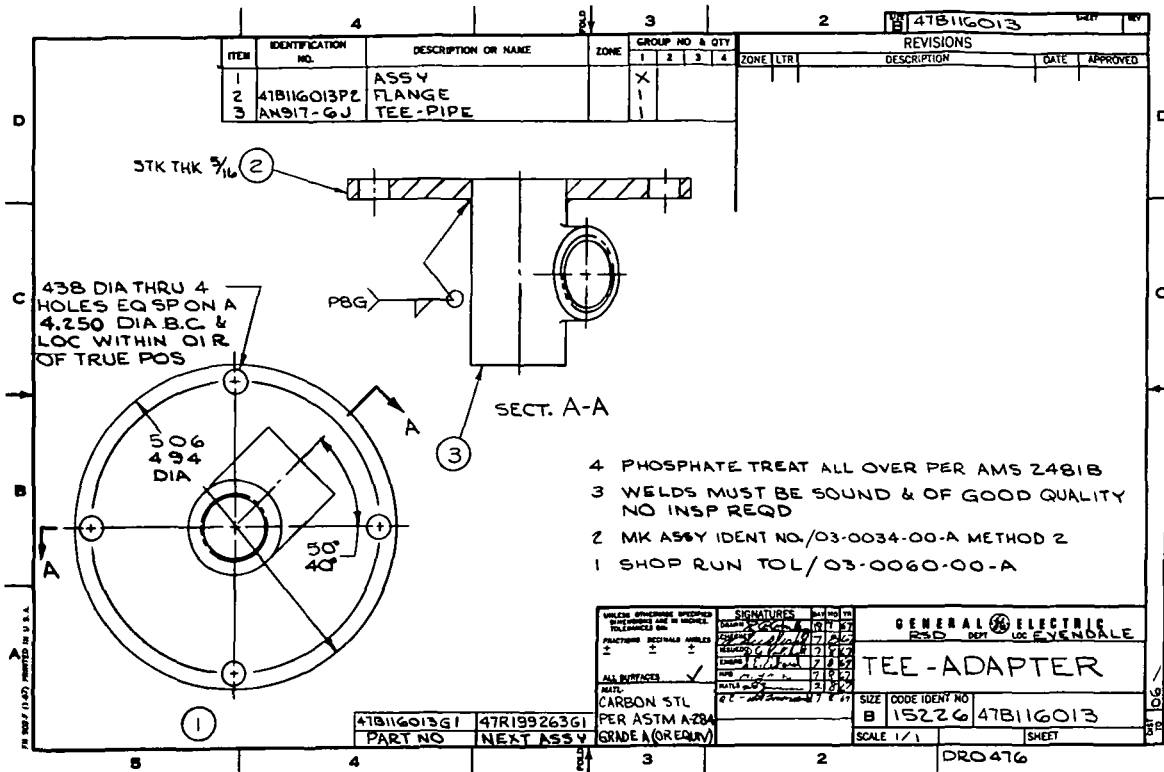


Figure 78. Adapter for Mounting Water-Brake Inlet Line on Cylindrical Cover Around Brake. (47B116013)

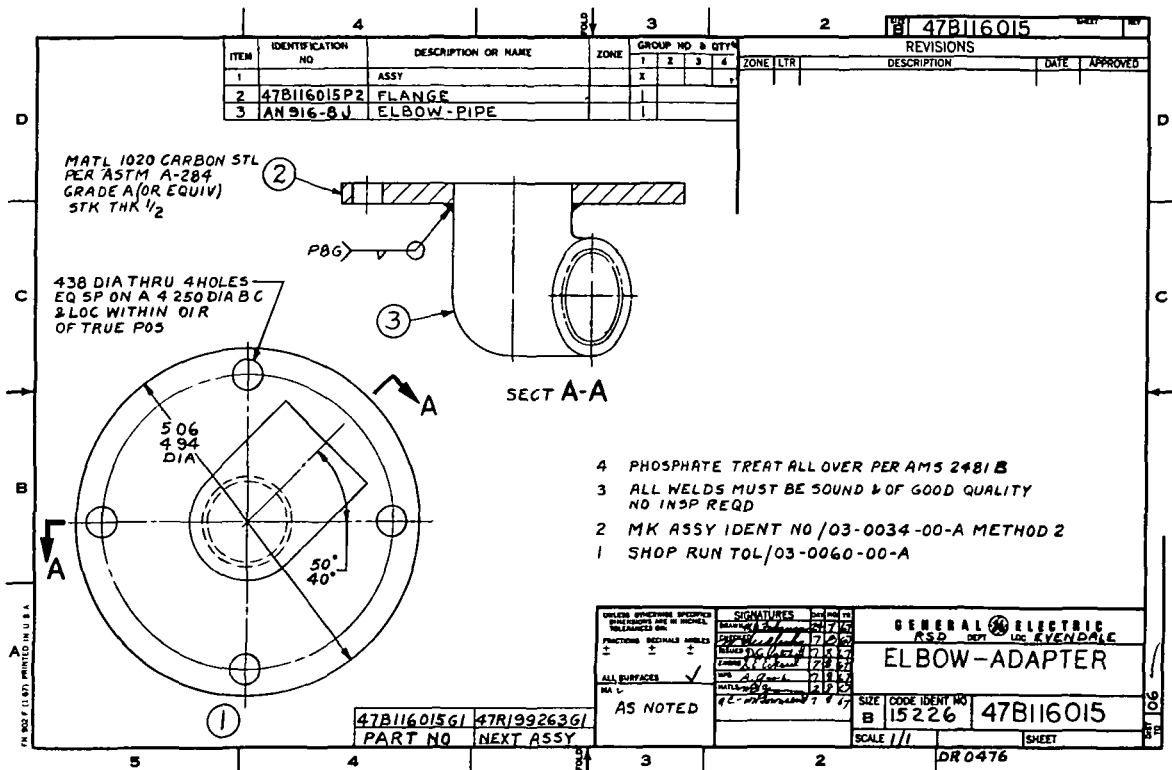


Figure 79. Elbow Adapter Used in Water-Brake Exhaust Line. (47B116015)

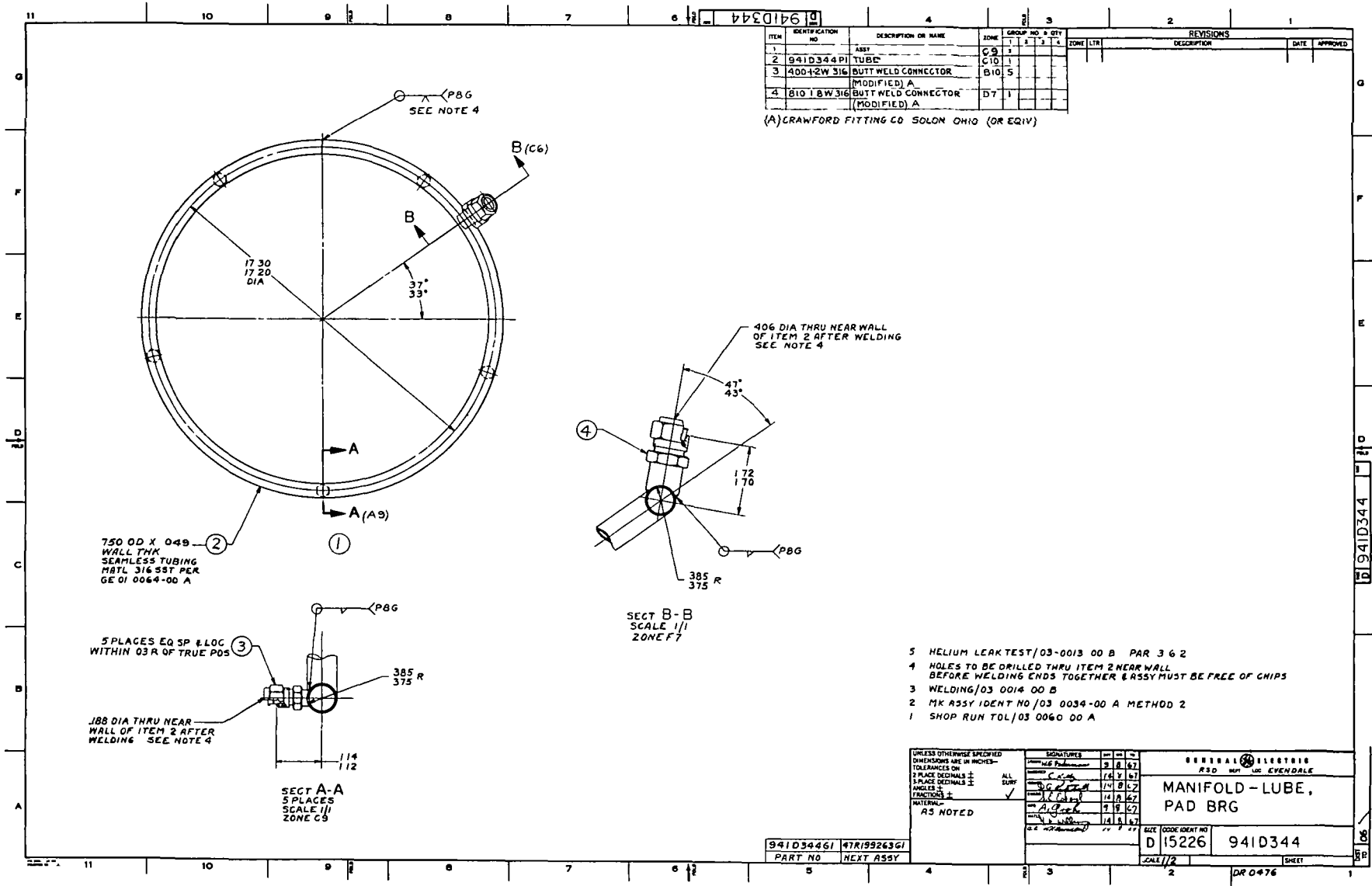
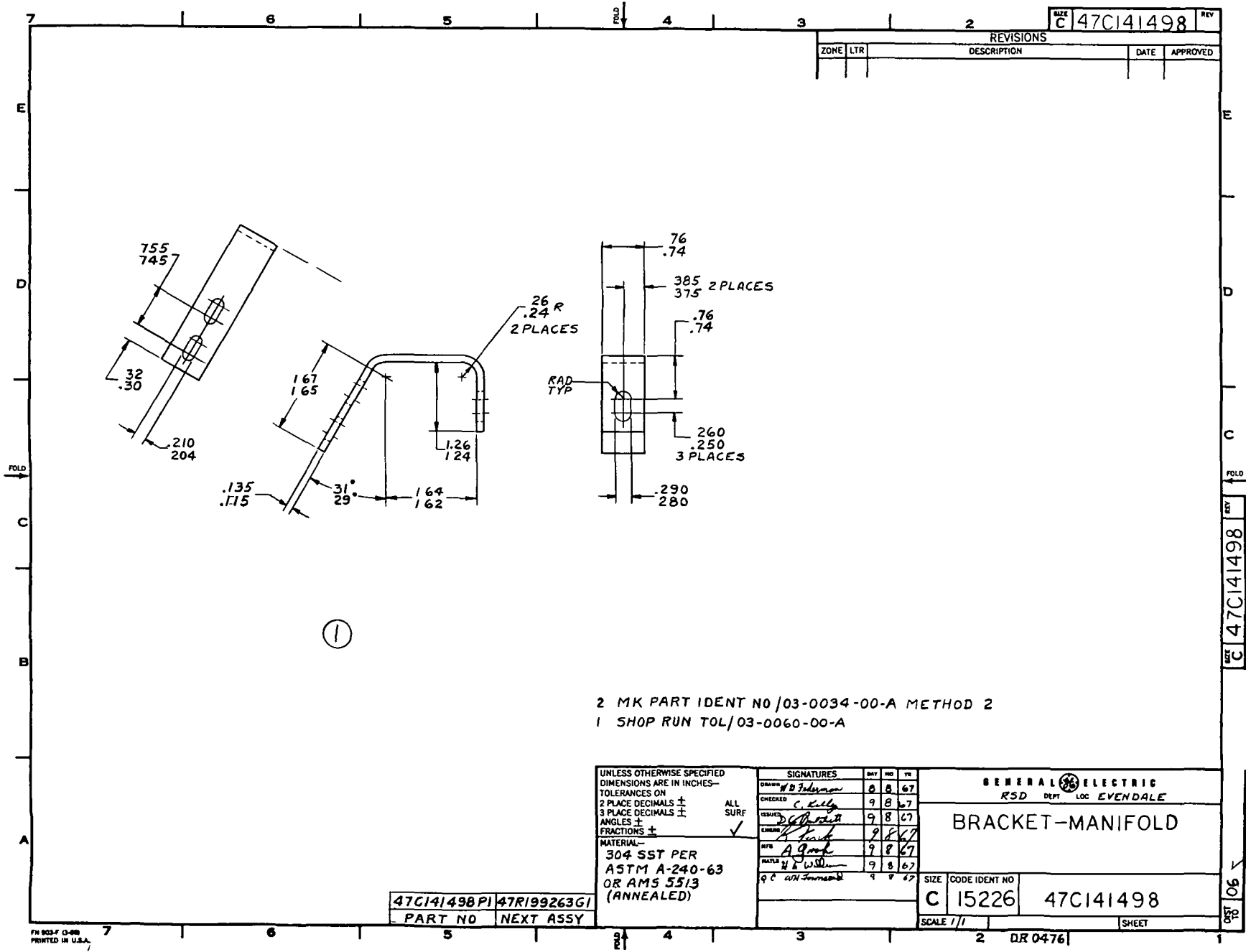


Figure 80. Lube Oil Manifold for Pivoted Pad Bearing of 3 Stage Test Turbine. (941D344)



2 MK PART IDENT NO /03-0034-00-A METHOD 2
 1 SHOP RUN TOL/03-0060-00-A

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES— TOLERANCES ON 2 PLACE DECIMALS ± 3 PLACE DECIMALS ± ANGLES ± FRACTIONS ± MATERIAL— 304 SST PER ASTM A-240-63 OR AMS 5513 (ANNEALED)	SIGNATURES			GENERAL ELECTRIC RSD DEPT LOC EVENDALE		
	DRAWN <i>W B Johnson</i> CHECKED <i>C Kelly</i> ISSUED <i>D [Signature]</i> ENGINEER <i>A [Signature]</i> MFR <i>A [Signature]</i> MTL <i>H & Wilson</i> QC <i>W H [Signature]</i>	MAY 8 67 9 8 67 9 8 67 9 8 67 9 8 67 9 8 67	MAY 8 67 9 8 67 9 8 67 9 8 67 9 8 67 9 8 67	BRACKET-MANIFOLD		
	ALL SURF ✓			SIZE C CODE IDENT NO 15226	47C141498	
	PART NO 47C141498 PI 47R199263 GI NEXT ASSY			SCALE 1/1	SHEET DR 0476	
	EST 106					

Figure 81. Support Bracket for Lube Oil Manifold Supplying Pivoted Pad Bearing of 3 Stage Test Turbine. (47C141498)

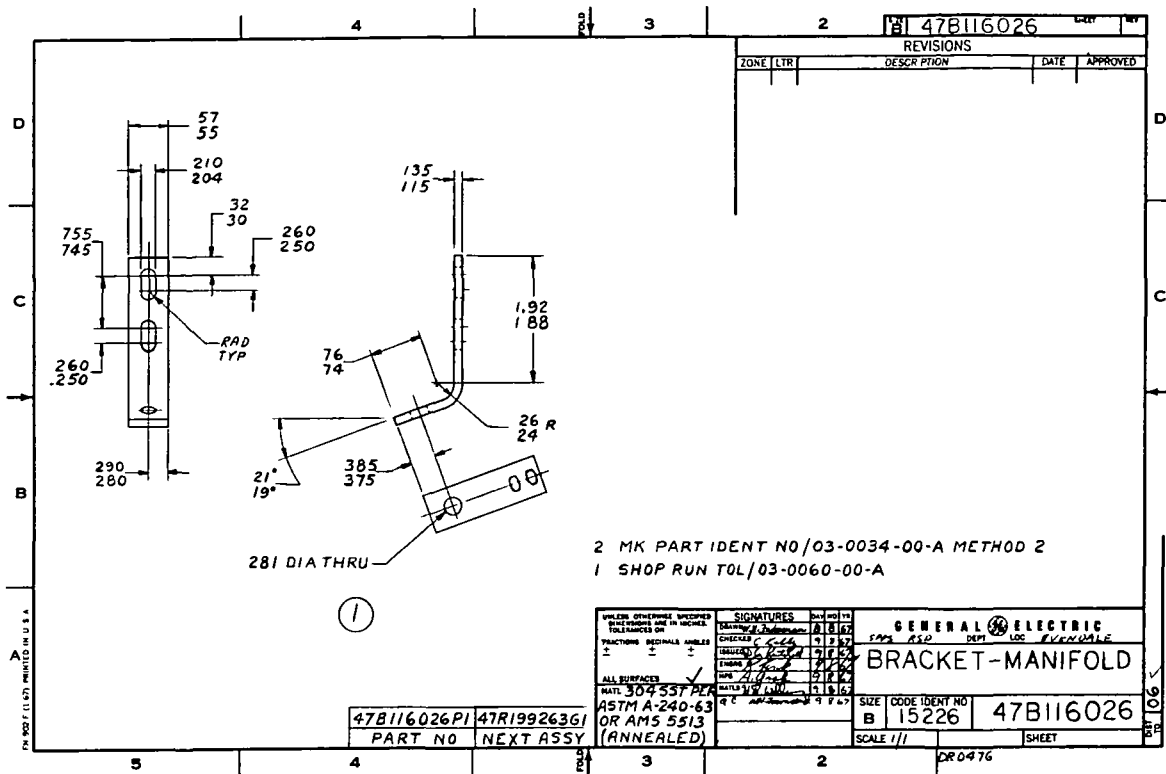


Figure 82. Bracket for Supporting Pad Bearing Lube Oil Manifold for 3 Stage Test Turbine. (47B116026)

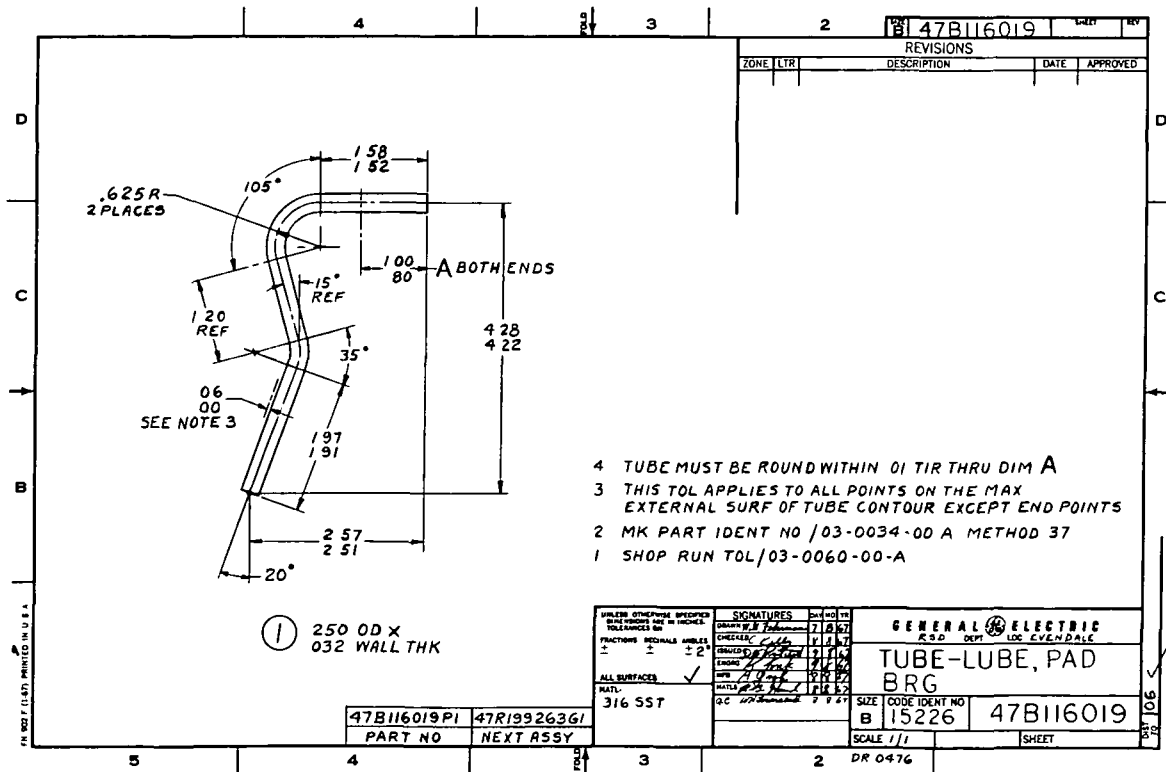


Figure 83. Lubricant Supply Tube Connecting Oil Manifold to Sump Housing of 3 Stage Turbine for Pivoted Pad Bearing Lubrication. (47B116019)

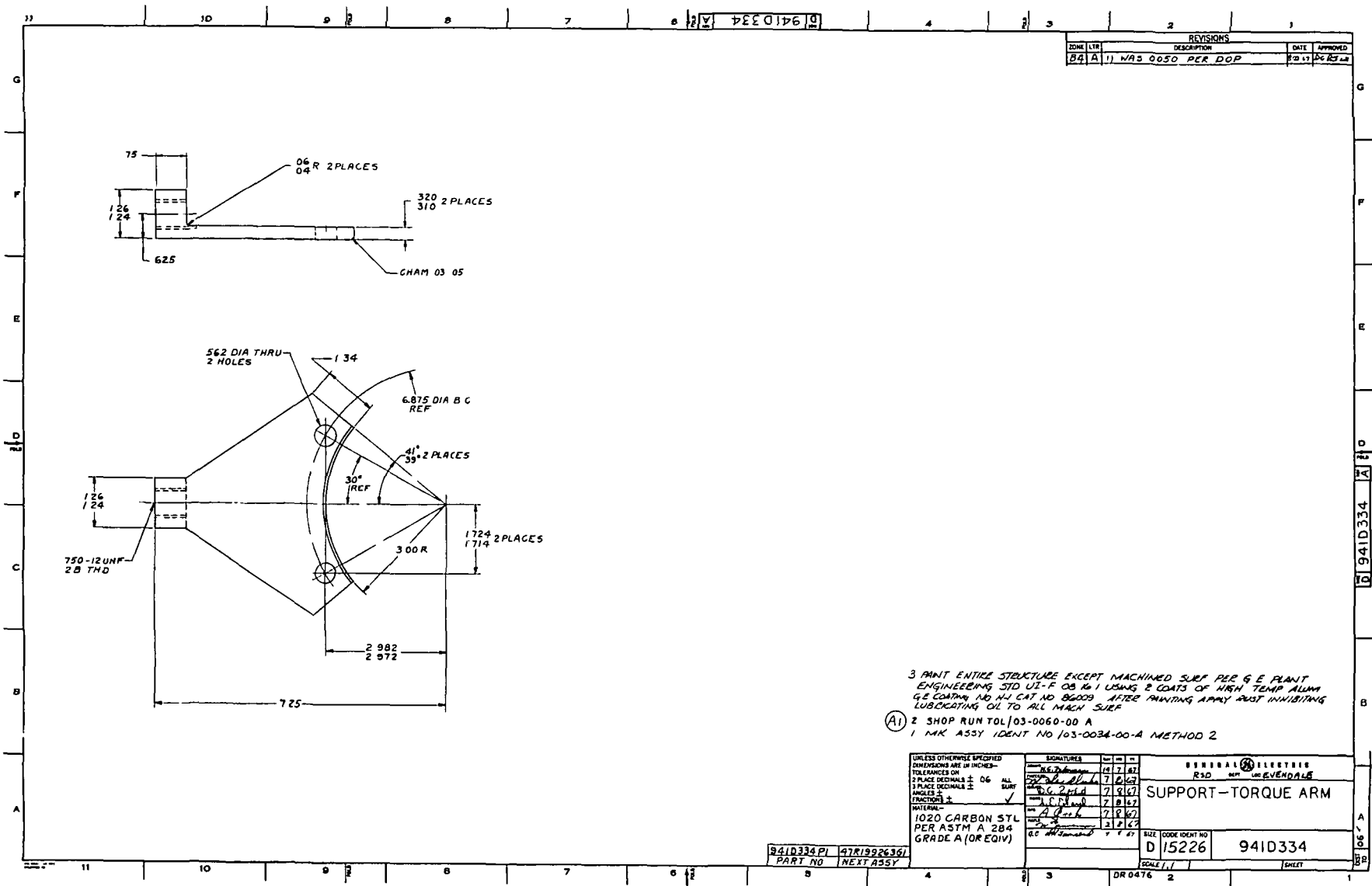


Figure 84. Support Bracket Attaching Calibration Torque Aim to Water-Brake Bytrex Torquemeter. (941D334)

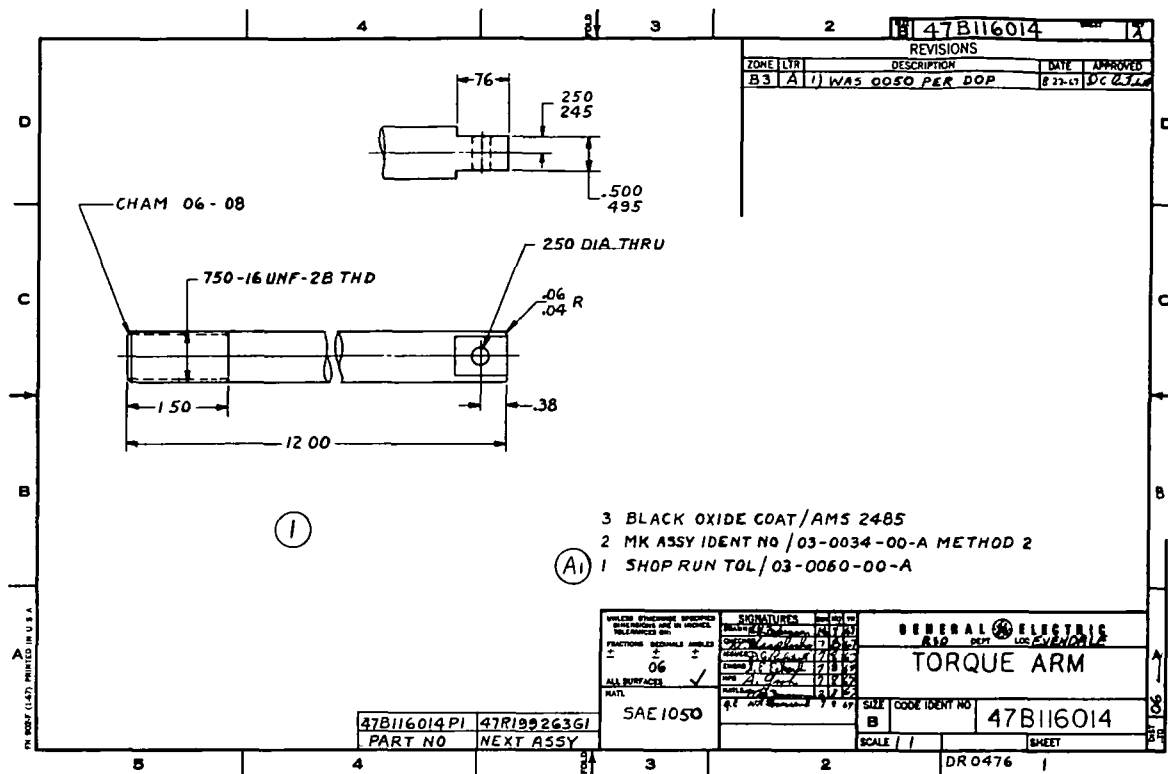


Figure 85. Calibration Torque Arm for the Water-Brake Bytrex Torquemeter. (47B116014)

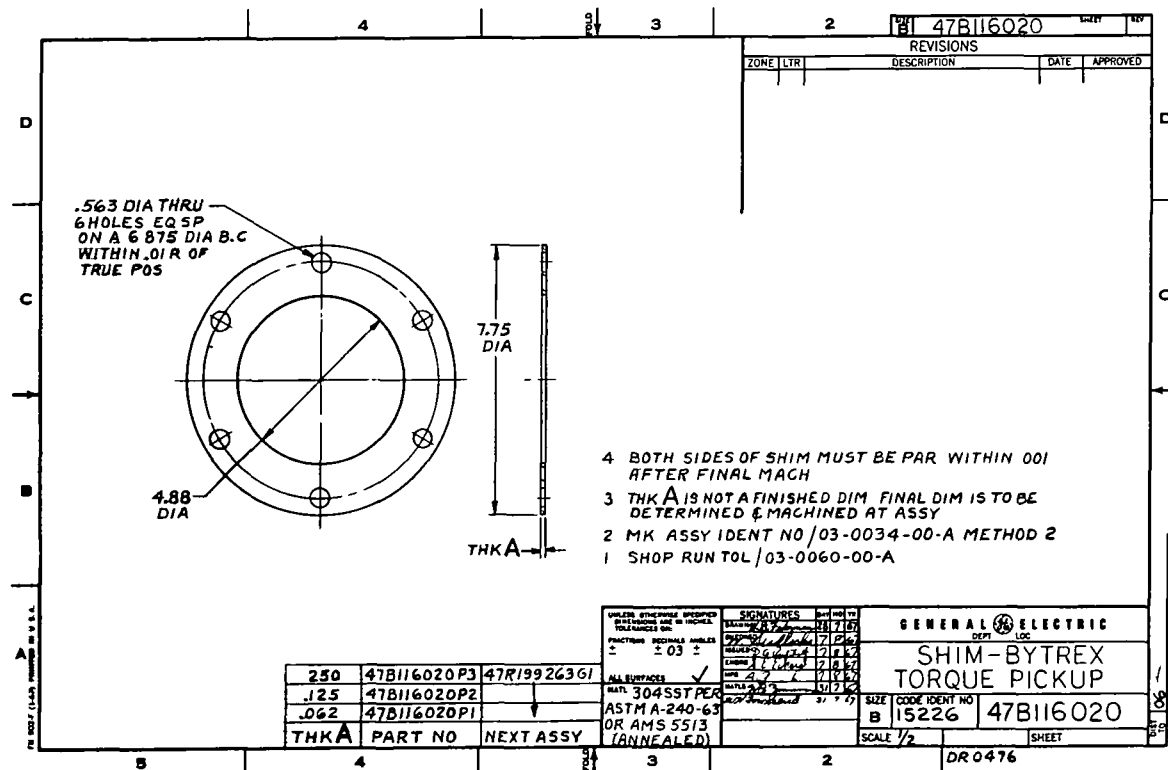


Figure 86. Shims for Axial Positioning of Both Bytrex Torquemeters. (47B116020)

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APPENDIX A

DESIGN CALCULATIONS FOR THE LIFE-LIMITING AND
MORE HIGHLY-STRESSED POWER TRAIN COMPONENTS

APPENDIX A

Contents

Flexible Diaphragm (Bendix) Coupling Design Summary

Water Brake Ball Bearing Design Assumptions and Analysis Summary

Steam Turbine Ball Bearing Design Summary

Disk Brake Cooling Design Summary

FLEXIBLE DIAPHRAGM (BENDIX) COUPLING

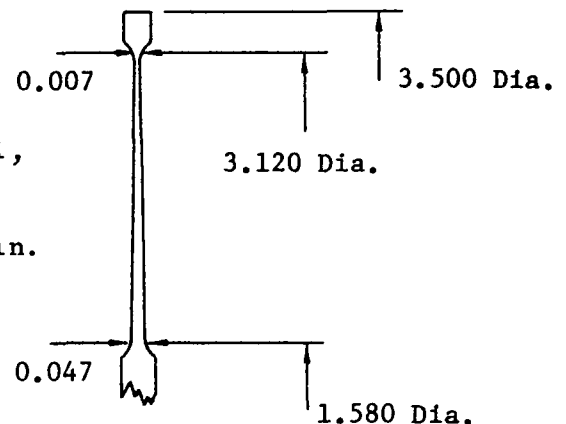
DESIGN SUMMARY

DESIGN ASSUMPTIONS

1. All detail design of coupling, including end attachments, membranes, and center spool is the responsibility of Bendix Corporation per the design and operating conditions stated in Design Specification 02-0199-00-A (included in Appendix B).
2. Operating and Emergency - Stop torque values are (as stated in Design Specification 02-0199-00-A) 1100 in.-lb during 5000 hours of testing and 3500 in.-lb during emergency stop (see below).
3. Maximum torque in the coupling is developed by a sudden seizure or excessive braking of the water brake, resulting in deceleration of the potassium turbine from 20000 rpm to 0 rpm in two seconds (through the flexible membrane coupling) while potassium vapor continues to supply driving torque to the potassium turbine. This deceleration is assumed to occur according to the cosine function shown in Figure 18. This graph also shows the approximate potassium turbine torque increase as a function of decrease in rotative speed while vapor continues to flow into the turbine. (The typical maximum torque at zero speed characteristic of turbines can be observed.) The total coupling torque resulting from the sum of the deceleration rotor inertia and continuing turbine vapor flow is shown, and results in a maximum value of 3500 in.-lb.

DESIGN ANALYSIS

1. Physical Data on membranes:
Material: E-18V (Vacuum Melted)
18 Ni, 8 Co, 5 Mo, 0.4 Ti,
0.1 Al, and others
UTS = 255,000 psi min.
0.2%Y.S. = 250,000 psi min.
Elong = 8% min.
R.A. = 48% min.
E = 27 (10⁶) psi

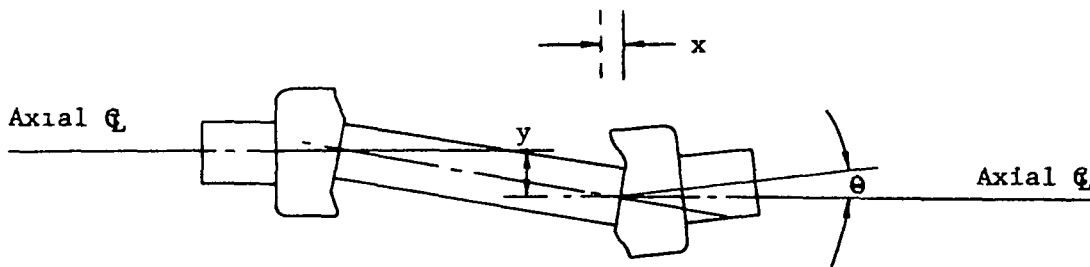


Distance between membranes in coupling 6.62 inches

2. Manufacturer Requirements

The Design Specification (02-0199-00-A) requires the manufacturer (Bendix Corporation) to guarantee their coupling for an infinite life while transmitting the 1100 in.-lb torque at 18250 rpm or under emergency-stop torque of 3500 in.-lb while under the following installation alignment conditions:

	<u>At Assy</u>	<u>Hot Operation</u>
a) Parallel offset of shaft ϵ "y" over 6.62 in. length.	0.002 in.	0.008 in.
b) Axial extension of each membrane from relaxed position "x"	0.005 in.	0.020 in.
c) Shaft out-of-parallelism (angular misalignment)	3 min.	6 min.
d) Total angular misalignment " θ " resulting from (a) and (c) - each membrane	4 min.	10 min.



3. Actual Assembly Goals

During power-train assembly, the following shaft alignment goals will be attempted:

	<u>At Assy</u>	<u>Hot Operation</u>
a) Parallel offset of shaft ϵ	0.001 in.	0.001 in.
b) Axial extension of each membrane	0.001 in.	0.004 in.*
c) Shaft out-of-parallelism	1 min.	1 min.
d) Total membrane angular misalignment	1.5 min.	1.5 min.

* See analysis of Thermal Expansion of coupling and surrounding structure (Figure 32).

4. Manufacturer's Quoted Allowable Misalignments

The following shaft misalignments may be tolerated by the coupling without jeopardizing its "infinite" life" as quoted by the manufacturer.

	<u>Hot Operation</u>
a) Parallel offset of shaft ϕ	0.020 in.
b) Axial extension of total coupling (two membranes)	<u>+0.040 in.</u>
c) Shaft out-of-parallelism	12 minutes
d) Total membrane angular misalignment	22.8 minutes

5. Combined Misalignments

For the flexible diaphragm, the allowable angular misalignment (causing alternating membrane stresses with each revolution) can be combined with the allowable axial extensions (causing steady membrane stresses) in a graphical manner similar to a standard "Goodman Diagram" which shows allowable alternating stresses as a function of steady-state stresses for a given material at a specific temperature and number of stress-reversal cycles. This curve (presenting the manufacturers data) is shown in Figure A-1. The upper curve is the absolute safe limit, and the dotted curve shows the actual design limit, providing a corresponding factor of safety in operation. Also given are the significant equations for calculation of membrane stresses. The design point for the maximum allowable misalignments of the coupling (see item 4 above) is shown.

6. Maximum Capacity Limits

In addition to withstanding "endurance test" torque for infinite life, the coupling must be adequate to:

- Withstand emergency-stop shear stresses in the membranes.
- Resist torsional buckling of membranes under emergency-stop.
- Withstand centrifugal stresses at 18250 rpm.
- Withstand shear of keys holding coupling to water brake shaft, or shear of the shaft itself.

- a) Safety against emergency-stop and normal-operation membrane shear. The allowable shear stress in the membrane to fracture is 74000 psi, corresponding to a torque of 8296 lb-in. With the actual design-point torque of 1100 lb-in., the continuous-operation factor-of-safety is:

$$F.S._{\text{continuous}} = \frac{8296}{1100} = 7.54$$

With the actual emergency-stop torque of 3500 lb-in., the resulting emergency-stop factor-of-safety is:

$$F.S._{\text{emergency}} = \frac{8296}{3500} = 2.37$$

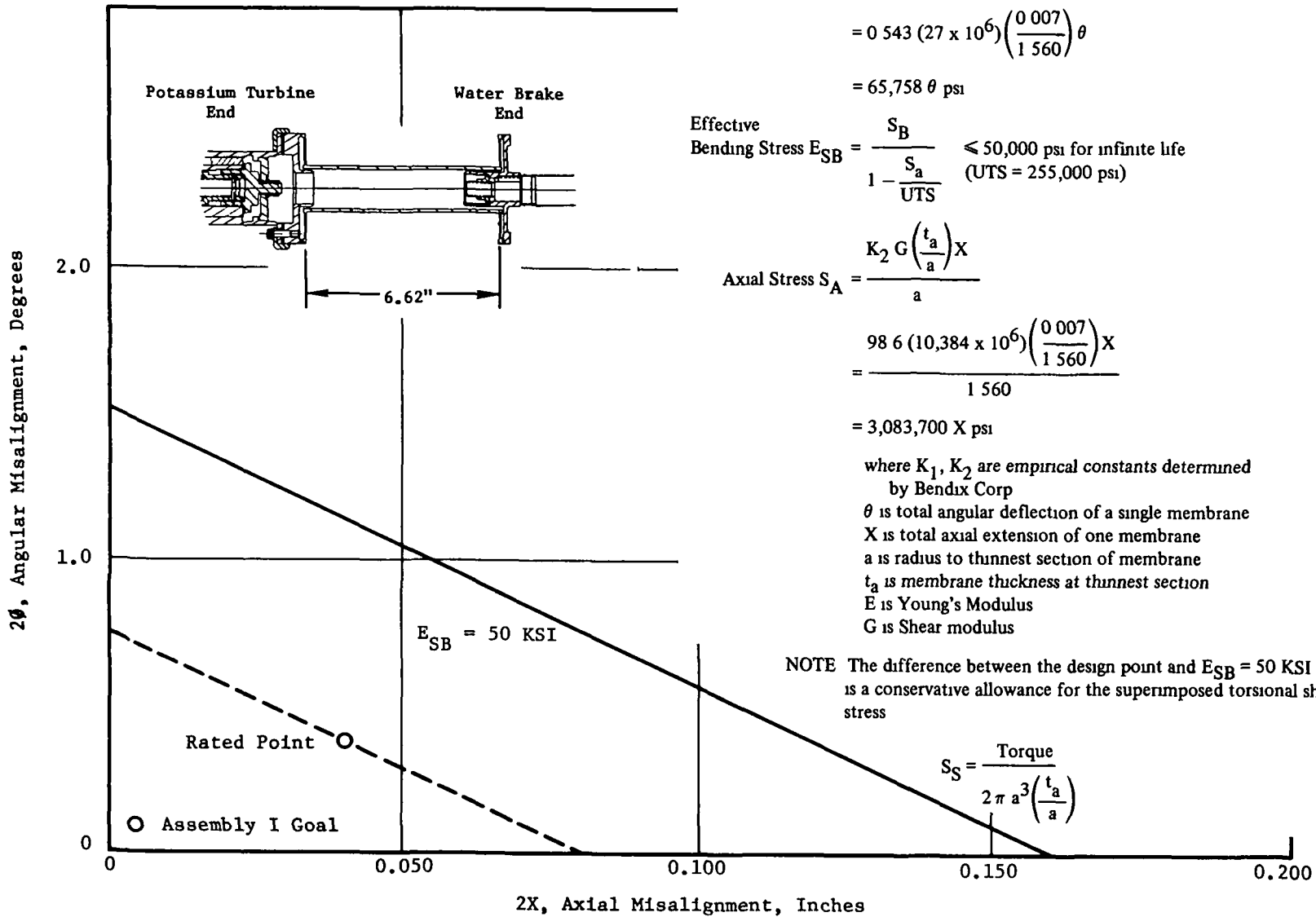


Figure A-1. Maximum Combined Angular Misalignment and Axial Extension Without Sacrificing "Infinite Life" Capability of Flexible Coupling 941D314.

b) Safety against buckling -

The allowable membrane torque before buckling is:

$$T_B = 6,123 \text{ lb-in.}$$

With the actual emergency-stop torque of 3500 lb-in., the buckling factor-of-safety is

$$F.S._B = \frac{6123}{3500} = 1.75$$

c) Safety against overspeed -

The overspeed limit for the coupling is 57000 rpm. Therefore, the overspeed factor-of safety is

$$F.S._{\text{overspeed}} = \frac{57000}{18250} = 3.12$$

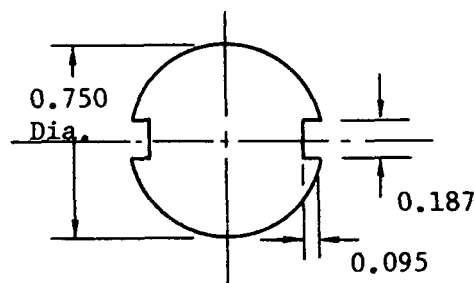
d) Safety against water brake shaft or keys' failure -

Although two diametrically-opposed 3/16-inch square keys, 3/4-inches long are used, the design was made assuming only one key is active in absorbing the shaft torque. The calculations were made for the emergency-stop torque of 3500 in.-lb.

Under this torque, the key shear and side-compressive stresses are

$$\sigma_{\text{shear}} = 66,600 \text{ psi}$$

$$\sigma_{\text{compression}} = 131,000 \text{ psi}$$



Maximum shaft torsional shear stress is:

(including the stress-concentrations around the keyways):

$$\begin{aligned} \tau_{\text{shaft}} &= 42300 (2.2) \text{ where } 2.2 = \text{stress conc. factor} \\ &= 93000 \text{ psi} \end{aligned}$$

The key and shaft material used is H-11 tool steel with a 0.2 percent yield strength of 140,000 psi min., and therefore, safe.

Assuming an allowable shear stress of 60 percent of the allowable shear strength, the shaft factor-of-safety is

$$F.S._{\text{shaft}} = \frac{150,000 (0.6)}{93,000} = 0.97$$

Which is sufficiently close to unity to be safe, considering the localized nature of the keyway stresses and the pessimistic torque assumptions in calculating the stresses. The H-11 material has sufficient toughness and ductility to tolerate local yielding if necessary.

GENERALIZED COUPLING DESIGN

Figure A-2 shows the allowable design limits of coupling membranes as a function of membrane geometry. The curves show maximum allowable shear and bending stresses (representing maximum transmitted torque and shaft misalignment, respectively) versus membrane thickness-to-diameter ratio. These curves were constructed from the generalized data presented in Reference 7. The design point of the coupling employed here is shown, assuming the conservative alignment conditions given under item 4 above. While the actual alignment is expected to be better than this, it can be seen that the coupling design is adequate, in any case, and will possess infinite life at the power-train design point conditions.

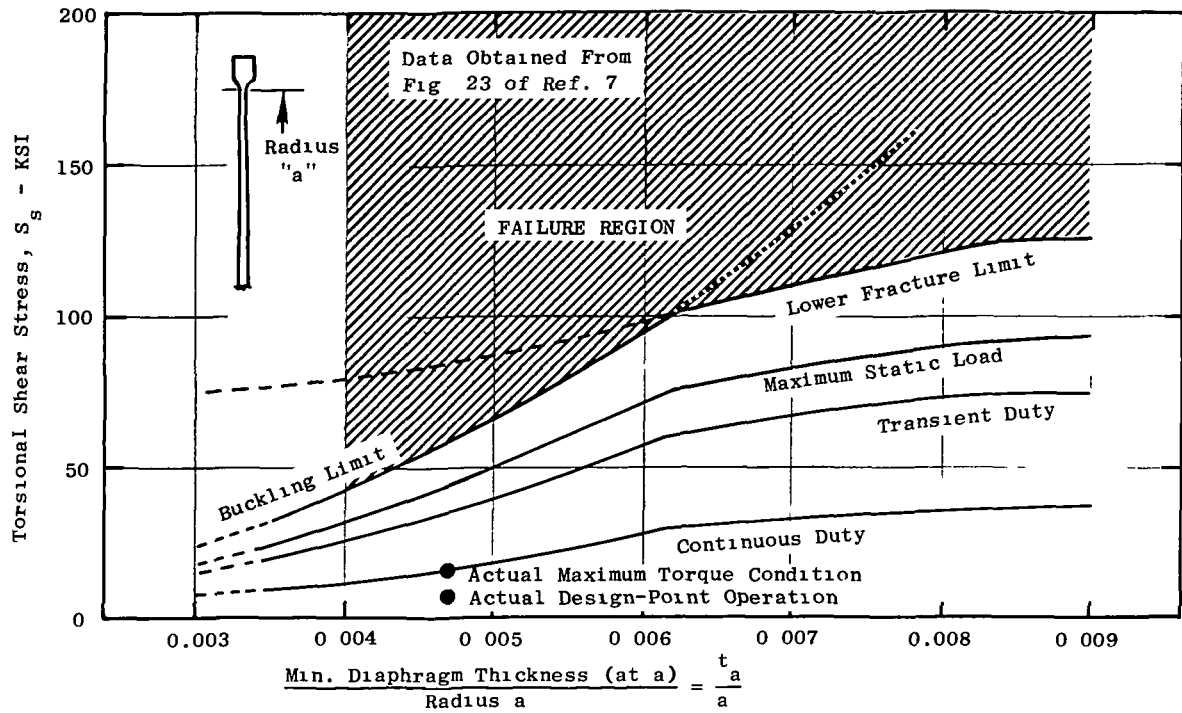


Figure A-2. Maximum Allowable Flexible Diaphragm Torsional Stress Levels as a Function of Membrane Geometry.

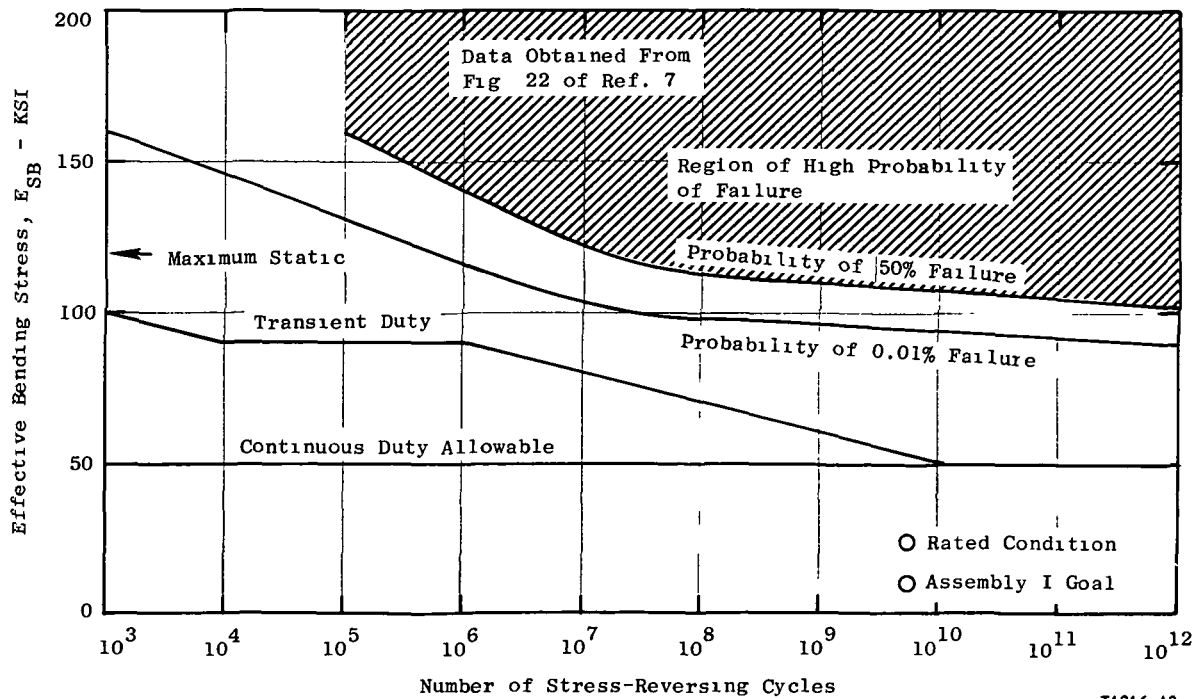


Figure A-3. Maximum Allowable Flexible Diaphragm Effective Bending Stress as a Function of Membrane Geometry.

WATER BRAKE
BALL BEARING DESIGN ASSUMPTIONS AND
ANALYSIS SUMMARY

ASSUMPTIONS

1. 52100 steel is used. Analysis employs GE "RECAP" bearing analysis computer program.
2. Bearings are isothermal and mounted in rigid structure.
3. Total radial load = rotor weight = 17 lb (47 percent on fwd bearing and 53 percent on aft bearing).
4. Total thrust load on bearings = 200 lbs (54 percent on fwd bearing and 46 percent on aft bearing). This load is thought to be quite conservative, considering that the actual bearing spring preload is 80 lbs.
5. Details of bearing geometry were obtained from manufacturer.
6. Computer-assigned bearing friction coefficient = 0.075.

CALCULATIONS (Computer results)

Analysis was performed on light series, medium series, and heavy series bearings for a 25 mm forward bearing and a 20 mm aft bearing as follows. The medium series bearings were selected for the final design. The " B_{10} " life calculated is the operating life which 90 percent of all tested bearings have exceeded. The average bearing lasts over five times as long as the B_{10} life.

	<u>Fwd Bearing</u>	<u>Aft Bearing</u>
1. Light Series	Size 205	Size 204
	9 balls, 0.3125 in. dia	8 balls, 0.3125 in. dia
	pitch dia 1.520 in.	pitch dia. 1.342 in.
	ID: 25 mm = 0.9843 in.	ID. 20 mm = 0.7874 in.
	OD: 52 mm = 2.0472 in.	OD. 47 mm = 1.8504 in.
B_{10} life for 200		
lb total thrust	6,954 hrs	7,790 hrs

	<u>Fwd Bearing</u>	<u>Aft Bearing</u>
2. Medium Series	Size 305	Size 304
(bearings	8 balls, 0.4375 in. dia	7 balls, 0.4063 in. dia
used in	pitch dia 1.7126 in.	pitch dia 1.4173 in.
water brake)	ID 25 mm = 0.9843 in.	ID: 20 mm = 0.7874 in.
	OD 62 mm = 2.4409 in.	OD: 52 mm = 2.0492 in.
B ₁₀ life for 200 lb		
total thrust	26,332 hrs	21,387 hrs

	<u>Fwd Bearing</u>	<u>Aft Bearing</u>
3. Heavy Series	Size 405	Size 404
	7 balls, 0.5625 in. dia	7 balls, 0.5313 in. dia
	pitch dia 2.067 in.	pitch dia 1.811 in.
	ID 25 mm = 0.9843 in.	ID 20 mm = 0.7874 in.
	OD 80 mm = 3.1496 in.	OD: 72 mm = 2.8546 in.
B ₁₀ life for 200 lb		
total thrust	75,661 hrs	81,918 hrs

CORRECTED BEARING LIFE

New Departure - Hyatt "NDur-300" (trademark) No. Q3305XRTCF bearings (medium series as indicated above) were selected as among the best standard bearings available. Because of good quality control, having balls of 2 points R_C harder than the races, and using vacuum re-melted 52100 steel, the estimated life is about three(3) times those values calculated above (which are for bearings made from conventional air-melted 52100 steel).

Therefore, the corrected values of expected operating life of the selected water brake bearings with an assumed total rotor thrust load of 200 lbs are:

	<u>Forward Bearing (305)</u>	<u>Aft Bearing (304)</u>
conventional life	26,332	21,387
NDur-300 life factor	<u> x 3</u>	<u> x 3</u>
Expected B ₁₀ life	78,996 hrs	64,161 hrs

For a required life of 5000 hrs, the factors of safety are
15.8
12.8

The manufacturer's estimates of bearing life even exceed these values, but are not presented since these shown are far more than adequate.

An investigation was made into the possibility of using a more advanced design wherein the bearing races are forged to shape. This is termed by New Departure as NDur-600 material, yielding an expected bearing life of six (6) times that of conventional bearings. However, since it is readily available only in bearings of larger size, and is actually not needed here, no serious attempts were made to secure such premium bearings.

STEAM TURBINE
BALL BEARING DESIGN SUMMARY

ASSUMPTIONS AND ANALYSIS

The steam turbine bearings are 306 size, having an ID of 30 mm and an OD of 72 mm. Due to the radial-flow nature of the steam, the bearings are subjected to a negligible axial force beyond that (a few pounds) which is imposed by the loading spring assembly (Figure 21). The standard bearings installed by the manufacturer are made from vacuum re-melted 52100 steel with balls of two (2) R_C points harder than the races and are considered equivalent to the quality of the bearings employed in the water brake. Exact geometry and thrust loads were not obtainable, and no computer analysis of these bearings was made. However, previous experience with the steam turbine revealed its bearings to be highly reliable, and turbines such as this one typically see years of continuous duty without teardown.

Considering the steam turbine to have rotor thrust and radial loads approximately equal to those of the water brake, the steam turbine bearings should have a life exceeding that of the brake bearings, since the steam turbine bearings are larger.

Therefore, for the steam turbine,

Expected B_{10} life > 79000 hrs.

For a required life of 5000 hrs, the bearings factor of safety is approximately

$F S > 15.8$

DISK BRAKE COOLING - DESIGN SUMMARY

The major objective of the cooling scheme employed on the disk for the three-stage turbine is to prevent (in the event of required brake activation) thermal cracking during the deceleration period. It is presumed that following such a shutdown (occurring over a 5.3 sec. interval) the brake disk could be replaced with a new one of identical design.

Assumptions

1. The disk brake must (without any assistance from the water brake), in 5.3 seconds, bring the entire test rig rotor (comprising the rotors of the potassium turbine, water brake, and steam turbine and associated couplings) to a complete stop from an initial rotating speed of 24500 rpm. (2560 rad/sec)
2. The entire rotor weight moment of inertia ($W\bar{r}^2$) is 639 lb-in.² (see Table VII).
3. During the deceleration period, the turbine continues to receive potassium vapor and drives against the brake with an average output of 164KW, or 155 Btu/sec (see Section I).
4. Shop air at 70°F and 60 psig (in the cooler head) is available as a disk cooling medium. This allows for a 30 psi pressure drop from the 90 psig shop air line while flowing to the cooler head. Upon being directed against the disk, the air heats up by a temperature increment of 165°F.
5. The flow coefficient for the air metering holes in the cooler head is 0.7, and the holes are each 1/8 inches in diameter.
6. The total energy absorbed by the brake during the 5.3 sec. rotor deceleration time is distributed evenly over the time span.
7. The face of the cooler head from which the impingement jets issue is held at a distance of 0.125 inch from the face of the rotating disk.
8. The temperature difference between the hot disk and the cooling air is 1700°F.

Analysis*

A deceleration from 24500 rpm to zero rpm is a kinetic energy change of

$$\begin{aligned}\Delta KE &= \frac{I}{2} (\omega_1^2 - \omega_2^2) \\ &= \frac{639 \left[(2560)^2 - (0)^2 \right]}{2 (32.2) (144) (778)} \text{ Btu}\end{aligned}$$

*From calculation by G. Wesling 8-24-67.

$$= 582 \text{ Btu}$$

Adding to this, the 155 Btu/sec generated by the turbine vapor during the deceleration, the total power absorbed by the disk (per second) is:

$$P = \frac{582 \text{ Btu}}{5.3 \text{ sec}} + 155 \frac{\text{Btu}}{\text{sec}}$$

$$= 265 \frac{\text{Btu}}{\text{sec}}$$

It is estimated that the disk can absorb (without cooling) about 147 Btu/sec. Then, the power which must be dissipated by the air impingement cooling system is

$$\text{Cooling System } P = 265 \frac{\text{Btu}}{\text{sec}} - 147 \frac{\text{Btu}}{\text{sec}}$$

$$= 118 \frac{\text{Btu}}{\text{sec}}$$

The cooling air head contains 160 orifices of 1/8 inch diameter (see Figure 42). With an assumed flow coefficient of 0.7 for each hole, the total flow area metering the disk cooling air is:

$$A_h = \frac{\pi}{4} (0.125)^2 (0.7)(160) = 1.37 \text{ in.}^2$$

For critical flow of air through the cooler head orifices, the total flow will be.

$$W_a = 0.532 \frac{P_t A_h}{\sqrt{T_t}} \quad \text{where} \quad \begin{array}{l} P_t = 75 \text{ psia} \\ A_h = 1.37 \text{ in.}^2 \\ T_t = 530 \text{ }^\circ\text{R} \end{array}$$

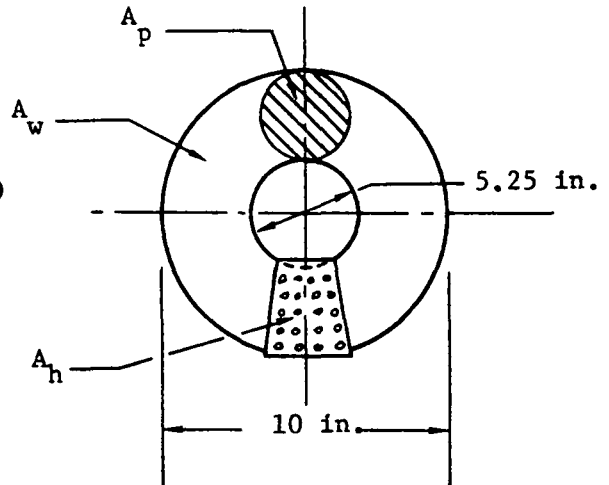
$$= 0.532 \frac{(75)(1.37)}{\sqrt{530}}$$

$$= 2.38 \text{ lb/sec}$$

With an inlet flow area to the cooler head of about 3.3 times that of the metering orifices (the air enters through a two inch pipe) the inlet mach number is about 0.2. This is sufficiently low that no significant restriction at the inlet will occur, and obtaining the above calculated airflow is, therefore, possible, since the assumed line pressure drop of from 90 psig (shop air) to 60 psig is reasonable, and probably conservative.

The total area of the disk wiped by the pucks is.

$$\begin{aligned}
 A_w &= \frac{\pi}{4} [d_o^2 - d_1^2] (2 \text{ faces}) \\
 &= \frac{\pi}{4} [(10)^2 - (5.25)^2] (2) \\
 &= 114 \text{ in.}^2
 \end{aligned}$$



The approximate area under the cooler head is:

$$A_h \approx 3 \text{ in.} (3 \text{ in.}) (2 \text{ sides}) = 18 \text{ in.}^2$$

The area under the puck is:

$$A_p = \frac{\pi}{4} (2.375)^2 (2 \text{ sides}) = 8.85 \text{ in.}^2$$

Then the area wiped by the pucks, exclusive of that under either the pucks or the cooler head, and thus, exposed to convection of the free-atmosphere air is:

$$\begin{aligned}
 A_a &= A_w - (A_h + A_p) \\
 &= 114 - (18 + 8.85) \\
 &= 87.2 \text{ in.}^2
 \end{aligned}$$

An estimate of the convective heat transfer coefficient for cooling under the cooler head is as follows:

From R. Gardon and J. Cobonpue, "Heat Transfer Between a Flat Plate and Jets of Air Impinging on It" 1961 - Int. Dev. in Heat Transfer, Boulder Conf, Part II, the recommended correlation is:

$$\text{Nu}_{\text{avg}} = 0.286 (\text{Re}_{\text{avg}})^{0.625}$$

$$\text{where } \text{Re}_{\text{avg}} = \frac{8.45 G X_n^3}{Z_n D \mu}$$

$$\text{and } h_{\text{avg}} = \frac{\text{Nu}_{\text{avg}} k}{X_n}$$

For this case, $X_n = 0.25$ in. (hole spacing)

$Z_n = 0.125$ in. (from jet to disk)

$D = 0.125$ in. (hole size)

$$G = \frac{2.38 \text{ lb}}{\text{sec} (18) \text{ in.}^2} = 0.1325 \frac{\text{lb}}{\text{sec in.}^2}$$

$$\mu = 0.6 \times 10^{-6} \frac{\text{lb sec}}{\text{ft}^2}$$

$$k = 0.025 \frac{\text{Btu}}{\text{hr ft } ^\circ\text{F}}$$

Then, substituting,

$$\text{Re}_{\text{avg}} = 8.45 (0.1325 \frac{\text{lb}}{\text{sec in.}^2}) \left[\frac{(0.25)^3 \text{ in.}^3 (10^6) (12) \text{ in.}}{(0.125) \text{ in.} (0.125) \text{ in.}} \right]$$

$$\bullet \left[\frac{\text{ft}^2 \text{ sec}^2}{(0.6) \text{ lb sec ft} (32.2) \text{ ft}} \right]$$

$$\text{Re}_{\text{avg}} = 6.95 \times 10^5$$

$$\text{and } \text{Nu}_{\text{avg}} = (0.286)(6.95 \times 10^5)^{0.625}$$

$$= 1288$$

$$h_{\text{avg cooler}} = \frac{1288 (0.025) \text{ Btu}}{(0.25) \text{ in. hr ft}^{\circ}\text{F}} \frac{(12) \text{ in.}}{\text{ft}}$$

$$= 1545 \frac{\text{Btu}}{\text{hr ft}^2 \text{ }^{\circ}\text{F}}$$

for the portion of the disk under the cooler head. For the portion of the disk exposed to the open atmosphere, a surface coefficient of

$$h_{\text{avg open}} = 100 \frac{\text{Btu}}{\text{hr ft}^2 \text{ }^{\circ}\text{F}} \text{ is assumed}$$

Then the heat removed is

$$Q = [h A \Delta T]_{\text{cooler}} + [h A \Delta T]_{\text{convection}}$$

$$Q = \frac{(1545)(18) \Delta T}{(144)(3600)} + \frac{(100)(87) \Delta T}{(144)(3600)} = 118 \text{ Btu/sec}$$

$$Q = 0.0537 \Delta T + 0.0168 \Delta T = 118 \text{ Btu/sec}$$

$$\Delta T = \frac{118}{0.0705} = 1675^{\circ}\text{F}$$

From this analysis, the disk will reach a temperature of 1775°F (1675°F + 100°F ambient) in order to reject the heat to the cooling air and the ambient atmosphere. The L605 material has a 0.2 percent yield strength of 25,000 psi at this temperature and therefore, is adequately cooled.

APPENDIX B

DESIGN AND MANUFACTURING SPECIFICATIONS
FOR PROCUREMENT OF MAJOR POWER TRAIN COMPONENTS

APPENDIX B

Contents

Title	Spec. No.	Issue Date
Flexible Diaphragm Coupling for Three-Stage Potassium Vapor Turbine Test	02-0199-00-A	5-29-67
Water Brake for Three-Stage Potassium Test Turbine	02-0203-00-A	6-7-67
Modifications to Engineering Specification 02-0203-00-A and Additional Design Requirements for Water Brake on GE P/O 037-129651		7-13-67
Vibration Damper for Water Brake on Drive Train of Three-Stage Potassium Vapor Turbine	02-0201-00-A	6-6-67
Gear Coupling for Three-Stage Potassium Vapor Turbine Test	02-0206-00-A	12-5-67
Rework of Steam Turbine for Three-Stage Potassium Vapor Test Turbine	02-0195-00-A	5-19-67
Exceptions to Steam Turbine Rework Specification 02-0195-00-A		
Letter of Recommended Steam Turbine Rework from Manufacturer		6-19-67

General

Included herein are the Manufacturing Specifications and associated modifications which were submitted to the various manufacturing vendors who fabricated the several specialized components and assemblies for the power train. In general, the modifications to original specifications resulting from engineering negotiations with the manufacturer, in which he provided sufficient justification (based upon experience and/or analytical proof) to obviate the need for certain specified requirements, or did not have adequate testing facilities or statistical experience on components sufficiently similar to ours to guarantee the required unattended life. However, in most instances, our previous successful experience on the Two-Stage Turbine power train supplied adequate assurance of reliability to allow acceptance of design without extensive manufacturer testing (e.g. cavitation of the Stellite-6B water brake disks in 5000 hours was previously found to be virtually non-existent, and comprised a more significant test than the manufacturer would have ever been able to make).

GENERAL ELECTRIC

SPACE POWER & PROPULSION SECTION
CINCINNATI, OHIO 45215

SPECIFICATION NO.

02-0199-00-A

ENGINEERING SPECIFICATION

ISSUE DATE

May 29, 1967

TITLE

FLEXIBLE DIAPHRAGM COUPLINGS FOR
THREE-STAGE POTASSIUM VAPOR TURBINE TEST

ORIGINAL CONTRACT

NAS3-10606

SECURITY CLASSIFICATION

Unclassified

PREPARED BY: B. Moon DATE 5-29-67
 APPROVED BY: H. Nichols L. E. Eckard DATE 5-29-67
 APPROVED BY: _____ DATE _____

ISSUED BY: D.A. Pritchett MAIL ZONE 3929/2438 PHONE N-21
DRAFTING

SP 1096 (11/66)

SUPERSEDES SPECIFICATION NO. DATED New	REVISION AUTHORIZATION	SEE SPPS INSTRUCTIONS SERIES 03.106
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SP 1073 A

FLEXIBLE DIAPHRAGM COUPLINGS FOR
THREE-STAGE POTASSIUM VAPOR TURBINE TEST - CONTINUED

DATE

5-29-67

NO.

02-0199-00-A

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FLEXIBLE DIAPHRAGM COUPLINGS FOR
THREE-STAGE POTASSIUM VAPOR TURBINE TEST - CONTINUED

DATE

5-29-67

NO.

02-0199-00-A

1. SCOPE.

1.1 Purpose. This document describes two special diaphragm couplings, for the purpose of obtaining vendor analysis and quotations and for purchasing.

2. APPLICABLE DOCUMENTS

2.1 Non-Government Documents.

Dwg. No. 47C141467	Cap, Shaft
Dwg. No. 47R199250	Three-Stage Turbine Aft Drive Train Layout
Dwg. No. 263E118	Shaft
Dwg. No. 263E136	Steam Turbine - Water Brake Coupling
Dwg. No. 941D312	Water Brake Shaft
Dwg. No. 941D313	Steam Turbine Rotor Modification
Dwg. No. 941D314	Water Brake - Turbine Shaft Coupling
Dwg. No. SK56131-341P1	Nut, Turbine

3. REQUIREMENTS.

3.1 Normal Operating Conditions.

- a) Speed - 18,250 RPM.
- b) Torque - 1340 in.-lb. (941D314 coupling) for 1 hr., 1100 in.-lb. for 5000 hr.
20 in.-lb. (263E136 coupling).
- c) Time - 5,000 hrs. (continuous). Treat as "flight safety" equipment.
- d) Temperature - 200°F.
- e) Environment - air, except steam at 212°F and 2 in. H₂O pressure at aft end of 263E136 coupling
- f) Unbalance - not to exceed $U = 0.03 W$
where U = unbalance, gm.-in.
 W = weight of rotating part, lb.
- g) Geometrical envelope - defined by dwg. no. 47R199250 - "Three-Stage Turbine Aft Drive Train Layout" (may be scaled) and SK56131-341P1 "Nut, Turbine".
- h) Geometrical interfaces - defined by dwgs. no. 47C141467 "Cap, Shaft" and no. 263E118 "Shaft" (geometry fixed), and dwgs. no. 941D312 "Water Brake Shaft" and no. 941D313 "Steam Turbine Rotor Modification", (interface geometry can be dictated by coupling requirements).
- i) Radial misalignment/coupling - 0.002 in. at assy., 0.008 in. hot.
- j) Axial misalignment/coupling - 0.005 in. at assy., 0.040 in. hot.
- k) Shafts' out of parallelism/coupling - 3 min. at assy., 6 min. hot.

3.2 Starting Conditions.

- a) Speed - 0 to 25,000 RPM.
- b) Torque - 0 to 300 in.-lb. (each coupling)
- c) Time - 100 hrs. (intermittent).
- d) Otherwise same as 3.1 above.

3.3 Braking Conditions.

- a) 3,270 lb.-in., to occur at least 10 times.

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- b) Temperature - 1150°F (of disk brake adjacent to 941D314 coupling).
 c) System inertias - potassium turbine, 396 lb.-in.², water brake, 11 lb.-in.² steam turbine 102 lb.-in.², total 509 lb.-in.².
 d) Otherwise same as 3.1 above.

3.4 Quality Control.

3.4.1 Material shall be per Bendix Corporation Material Specification No. E-18A, V, Rev. L, CO Nos 7A899 and 1B060, 4 Sept. 1964, "18% Nickel- Cobalt- Molybdenum-Maraging Steel". Vacuum melted steel shall be specified.

3.4.2 Material Processing shall be per Bendix Corporation Process Specification No. 413, Rev. F, CO Nos. 8A307-5B056, 27 Feb. 1967, "Processing of E-18A, V Maraging Steels".

3.4.3 Electron Beam Welding shall be utilized for all joining required and shall be per Bendix Corporation Process Specification No. 415, Rev. C, CO No. CNR-2B305, 11 May 1965, "Electron Beam Welding". All welds shall be classified as Part A - Critical Welds.

3.4.4 Manufacturing Process Quality Control shall be per Bendix Corporation "flight safety" equipment standards. A statement of the quality control plan for the couplings described by this specification shall accompany the quotation of price and delivery schedule. Government source inspection shall be used, with final approval of all deviations to reside with the purchaser.

3.4.5 Unbalance shall be ascertained by high-speed rotation of each coupling. The actual amount of unbalance at each end of each coupling shall be reported, and not simply that "unbalance is within $U = 0.03 W$ requirement".

3.4.6 Vibration of each coupling shall be done to ascertain their critical frequencies. No major critical frequency shall occur between 15,000 and 20,000 RPM. A curve of vibration amplitude vs. applied frequency shall be developed.

3.4.7 Life Testing. Each coupling shall be submitted to at least 10^7 cycles of operation at misalignment conditions which produce at least twice the stresses which should result from the normal misalignment conditions of 3.1 above. Vendor shall submit a statement of the proposed test plan for purchaser approval at least one week prior to testing. No indications which can be interpreted as cracks by magnetic particle inspection shall be allowable.

3.4.8 Weight. The weight of each component of each coupling, accurate to within one (1) gram, shall be ascertained immediately prior to joining with other components, and the finished weight of each coupling shall be reported with similar accuracy.

3.4.9 Marking of each coupling in a non-critical area by a permanent method, such as electro-chemical etching, with the marking including drawing no. and serial no. shall be done. Removable red tags or a similar method shall be used to warn assembly personnel that the diaphragms are extremely delicate. Couplings shall be suitably protected during shipment.

3.5 Information to be Submitted With Proposal.

FLEXILLE DIAPHRAGM COUPLINGS FOR THREE-STAGE POTASSIUM VAPOR TURBINE TEST - CONTINUED	DATE 5-29-67	NO. 02-0199-00-A
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3.5.1 Price of quantities of 1, 2, and 3 of each coupling design described herein, with price breakdown by design and by significant tasks, such as special testing, which could be eliminated without affecting the rest of the proposal.

3.5.2 Delivery Schedule based upon receipt of order. Purchaser requires freedom to adjust length of one coupling between diaphragms at the latest reasonable date.

3.5.3 Drawings defining the geometry of the couplings in sufficient detail to allow inspection of the parts and interface control. Proprietary information should be omitted.

3.5.4 Calculation Results in sufficient detail to prove the capability of the couplings to operate continuously under the "normal operating conditions", as drawing notes.

3.6 Information to be Submitted After Receipt of Order.

3.6.1 Drawings and Specifications. As soon as possible, but not less than thirty (30) days after receipt of Purchase Order, the vendor shall provide two (2) reproducible copies or drawings and specifications which define the geometry of the couplings to be supplied. Proprietary information should be omitted.

3.6.2 Inspection Reports. Six (6) copies or two (2) reproducible copies of all reports of inspections which were made on the couplings specified herein, identified by part drawing no. and serial no. and with actual dimensions instead of "within specified tolerance", shall be supplied to the purchaser upon or before delivery of the parts.

3.6.3 Computer Results of calculations performed on the final coupling designs, including "normal" design conditions and three "off-design" conditions to be later specified for each coupling. The results shall include:

- | | |
|---|---|
| a) Coupling identification | p) Continuous duty rated speed, torque, angle, effective bending stress, bending stress, axial stress, shear stress, and angle per diaphragm. |
| b) Coupling nominal o.d. | q) Maximum speed, torque, stresses, and angle. |
| c) Coupling length, l | |
| d) Contour major radius, a | |
| e) $\frac{\text{Contour minor radius } b}{\text{Contour major radius } a}$ | |
| f) $\frac{\text{Diaphragm minimum thickness } t_a}{\text{Contour major radius } a}$ | |
| g) $\frac{\text{Bending stress}}{\text{Angular deflection}}, S_b$ | |
| h) $\frac{\text{Shear stress}}{\text{Torque}}, S_s$ | |
| i) $\frac{\text{Axial Stress}}{\text{Axial deflection}}, S_a$ | |
| j) Bending spring rate, K_b | |
| k) Torsional spring rate, K_t | |
| l) Axial spring rate, K_a | |
| m) Buckling torque, T_B | |
| n) Minimum diaphragm thickness, t_o | |
| o) Maximum speed, RPM | |

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3.6.4 Certification that the requirements of (state each specification) have been met, the parts are balanced within (state values for each coupling), and that life testing per (state test plan) of each coupling has been successfully accomplished shall be provided on or before delivery of the parts. Two (2) reproducible copies of the vibration amplitude vs. applied frequency curve shall be provided.

GENERAL ELECTRIC SPACE POWER & PROPULSION SECTION CINCINNATI, OHIO 45215	SPECIFICATION NO. 02-0203-00-A
ENGINEERING SPECIFICATION	ISSUE DATE June 7, 1967

TITLE WATER BRAKE FOR THREE-STAGE POTASSIUM TEST TURBINE	ORIGINAL CONTRACT NAS 3-10606
	SECURITY CLASSIFICATION Unclassified

PREPARED BY: B. Mon DATE 6.1.67
 APPROVED BY: H. Nichols DATE 6.1.67
 APPROVED BY: J. E. Eckhard DATE 6-6-67

ISSUED BY: D.A. Pritchett *D.A. Pritchett* MAIL ZONE N-21 PHONE 3929/2438
DRAFTING

SP 1096 (11/66)

SUPERSEDES SPECIFICATION NO. NEW	DATED REVISION AUTHORIZATION	SEE SPPS INSTRUCTIONS SERIES 03.106
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WATER BRAKE FOR THREE-STAGE
POTASSIUM TEST TURBINE

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WATER BRAKE FOR THREE-STAGE
POTASSIUM TEST TURBINE

- CONTINUED

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6-7-67

NO.

02-0203-00-A

1. SCOPE

1.1 Purpose. This document describes a modified version of a commercial water brake, for the purpose of obtaining quotations and purchasing.

1.2 Intent. This document is intended to incorporate the experience of the Space Power and Propulsion Section in the use of a similar water brake under required operating conditions, and to allow freedom for the vendor to incorporate improvements, which will result in a design capable of at least 5000 hour life under the specified operating conditions. Since cavitation of the rotor disks is the primary limitation on operating life, these will be supplied by the General Electric Company. The vendor should concern himself primarily with incorporating such improvements as he feels can be made to other components.

2. APPLICABLE DOCUMENTS

2.1 Non-Government Documents.

Spec. No. AMS 2640 G	Magnetic Particle Inspection
Dwg. No. 941D311	Water Brake Proposal
Dwg. No. 941D312	Shaft-Waterbrake
Dwg. No. 47R199250	Three-Stage Turbine Aft Drive Train Layout

3. REQUIREMENTS

3.1 Operating Conditions.

- a) Speed - 18,250 RPM
- b) Power - 315 HP
- c) Time - 5,000 hours (continuous)
- d) Dynamometer Fluid - city water at 60°F-80°F, flow rate ~6 gpm nominal, ~20 gpm max.
- e) Temperature - 200°F max. allowable fluid outlet temperature and approximate expected housing temperature.
- f) Unbalance - 0.10 gm-in. max.
- g) Lubricant - Socony-Mobil Type DTE 797 lubricating oil
- h) Purchaser supplies all system components not shown on Dwg. 941D311 Water Brake Proposal.

3.2 Required Changes.

3.2.1 Dwg. 941D311 Water Brake Proposal. Shows the general requirements and changes needed to increase the reliability and life of the water brake, based upon operating experience at the General Electric Company. It does not show minor geometrical details, which are to be settled by the vendor, but incorporation of the major concepts shown are mandatory. This document shall not be interpreted as prohibiting the vendor from incorporating changes which will increase the operating life of the water brake.

3.3 Quality Control.

3.3.1 Rotating Parts. Shall reveal no indications which may be interpreted as cracks when subjected to magnetic particle inspection per AMS 2640 G.

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POTASSIUM TEST TURBINE

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3.4 Information to be Submitted With Proposal.3.4.1 Price of a water brake meeting the requirements of this proposal.3.4.2 Delivery Schedule based upon receipt of order.3.4.3 Drawings and Specifications sufficient to define the product offered, specifically including installation dimensions and supply lines required, and expected speed-torque characteristics of the water brake.3.4.4 Recommended Spare Parts List with prices.3.5 Information to be Submitted After Receipt of Order.3.5.1 Drawings and Specifications As soon as possible, but not later than thirty (30) days after receipt of Purchase Order, the vendor shall furnish two (2) reproducible copies of certified prints of drawings defining all items to be supplied, including referenced specifications. Sufficient detail shall be provided to allow proper installation, assembly and disassembly, and damage control inspection by the purchaser. Proprietary details should be omitted.3.5.2 Maintenance Instructions. Six (6) non-reproducible or two (2) reproducible complete sets of instructions for the care and maintenance of the equipment shall be provided with or before the delivery of the equipment.3.5.3 Certification that dynamic balancing to (state value) gm-in. and magnetic particle inspection conforming to AMS 2640 G have been successfully accomplished is required with or before delivery of the equipment. Provide written statement from the bearing vendor that bearings are vacuum degassed NDur-300⁽¹⁾ or NDur-600⁽¹⁾ bearings, and certify that bearings used and spares provided are NDur-300⁽¹⁾ or NDur-600⁽¹⁾. Two (2) reproducible copies of the expected speed vs. power map of the water brake shall be provided.4. BEARING OPERATING EXPERIENCE AND CALCULATED LIFE.4.1 Operating Experience. Several water brake bearings were lost during potassium turbine pretesting. It was not proved that the oil-mist lubrication system was at fault, but the inconvenience of the escaping mist, cooling requirements, and the susceptibility of the system to fluctuations in oil and shop air pressure caused conversion to a liquid lubrication system. The combination of about 200°F housing temperature and about 70°F bearing temperature rise would decrease bearing life if lubricant flow does not extract heat. With oil supplied to each bearing at about 0.1 gpm and 80°F, the forward bearing normally ran at about 155°F. To convert to liquid oil supply, the orifice sizes of the original oil nozzles were increased from about 0.030 in. diameter to about 0.060 in., and the oil outlet line size was increased to 1/2 in. NPT.4.2 Calculated Life of Water Brake Bearings.

(1) Trade name of products of New Departure-Hyatt Bearing Division of General Motors Corporation.

WATER BRAKE FOR THREE-STAGE
POTASSIUM TEST TURBINE

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4.2.1 Assumptions.

- a) Cage reactions not considered
- b) Fatigue constants based on 52100 steel
- c) Rigid system (all deflections in bearings)
- d) System isothermal
- e) Radial load = 17 lb
- f) Thrust load = 100 lb and 200 lb
- g) Friction coefficient = 0.075
- h) AFBMA calculation methods used in RECAP computer program.

4.2.2 Results.

Ball Bearing 205 - ABEC 7 (Fwd.).

Load Direction		axial	radial	axial	radial
Light Series	Load, lb	54.206	-8.0684	106.75	-8.0483
	L ₁₀ Life, hr		50,972		6,954
Medium Series	Load, lb			109.47	-8.0516
	L ₁₀ Life, hr				26,332
Heavy Series	Load, lb			107.07	-7.9688
	L ₁₀ Life, hr				75,661

(P30) P1604-2 Ball Bearing 204-ABEC 7 (Aft).

Load Direction		axial	radial	axial	radial
Light Series	Load, lb	45.795	-8.9289	93.250	-8.9517
	L ₁₀ Life, in.		62,676		7,790
Medium Series	Load, lb			90.531	-8.9481
	L ₁₀ Life, in.				21,387
Heavy Series	Load, lb			92.932	-9.0305
	L ₁₀ Life, in.				81,918

4.2.3 Material Change to Increase Life. Requests for quotations to provide deep-groove radial, ultra high speed, phenolic retainer, unshielded ball bearings with code 3 or 5 radial play, ABEC 7 tolerances, and balls and races of consumable electrode vacuum melt 52100 or M50 steel have yielded only one quotation. The unit cost would be \$342.00 each and delivery time would be 42 weeks. If vacuum melt material could be used, however, an increase to about 500 percent of the present life of the bearings would be expected.

At the present time it does not appear worth the cost in time and money to incorporate the suggested changes by special bearing order. However, superior quality bearings can be obtained as stock items by specifying New Departure-Hyatt type NDur-300 or NDur-600 bearings. The latter is preferred, but is generally available only on large bearings. Assurance that NDur-300 or NDur-600 bearings are obtained can be obtained by ordering directly from a New Departure-Hyatt sales engineering office (Bristol, Connecticut), requesting a written statement that the bearings supplied are NDur-300 or NDur-600. Since older types of New Departure bearings might be obtained from normal distribution channels, it is mandatory that the above ordering procedure be followed. Loading grooves are prohibited, medium series is required.

MODIFICATIONS TO ENGINEERING SPECIFICATION 02-0203-00-A AND ADDITIONAL
DESIGN REQUIREMENTS FOR WATER BRAKE ON G.E.P.O.037-129651

July 13, 1967

(Paragraph numbers refer to specification 02-0203-00-A.)

Para. 3.1 c) - Vendor exception to the requirement of guaranteeing 5,000 hours continuous operation is accepted.

Para. 3.1 f) - Vendor exception to the requirement of guaranteeing 0.10 gm-in. maximum unbalance is accepted, provided that all reasonable effort shall be made by the vendor to design, fabricate, and test the rotating parts of the assembly in an attempt to achieve less unbalance than the maximum allowable.

Para. 3.5.1) - Drawings and Specifications have not been supplied by vendor in sufficient detail to allow the determination of whether Purchaser agrees with specific design details. By acceptance of this fabrication order, vendor shall agree that the design principles stated by Drwg. 941D311 "Water Brake Assembly" as modified by these specification exceptions, additional requirements, and considerations will be incorporated, or formal agreement to do otherwise will be obtained from the Purchaser. The Purchaser shall also be allowed to approve the final design before irrevocable machining operations are reached. Purchaser shall be allowed five (5) working days after receipt of the final set of manufacturing drawings to evaluate and comment.

Spare Parts are desired but cannot be ordered until the final design is approved. Vendor should submit a formal quotation and detailed list of recommended spare parts at that time.

Installation Data per SK 255 is accepted with the following exceptions:

- a) The housing length of 13 in. max. shall be 12.79 - 12.81 in.
- b) The length of the 3/8 - 24 UNF 3A thread on each end of the shaft shall be increased from 0.5 in. to 1.0 in., and the overall shaft length shall be increased by 1.0 in.
- c) The following critical length dimensions shall have a tolerance not to exceed ± 0.002 in.:

Forward bearing shoulder to forward coupling shoulder,
forward housing face, aft bearing shoulder, and aft
coupling shoulder.

Maximum Power Absorbing Characteristics for Proposed Hydro-Brake are not acceptable, as described by connection of the following points on a log-log plot:

<u>Point</u>	<u>Horsepower</u>	<u>RPM</u>
1	10	5,500
2	400	20,000
3	45	20,000
4	10	12,000

Where the area enclosed by the above points represents the region of stable operation of the water brake. It is understood that the true power absorption characteristics will be substantially greater than the above, but cannot be calculated until the final design is reached. Approval of an estimate based upon the final configuration is required as a part of the final design approval.

Stellite Stators and Rotors are to be shipped to the Vendor within thirty (30) days after the receipt of the fabrication order for evaluation. The decision as to their reusability is required at the earliest possible date, so that new components may be ordered if required.

Keys and Locknuts will be furnished by Industrial Engineering Co. for all keyways and bearing locknut screw threads which they create. One end of the keys will be contoured to fill the keyways. Locknuts with nylon inserts are to be used instead of tab washers.

Hydrostatic Testing of the water brake assembly for one hour minimum at 80 psig minimum pressure. Total leakage shall not exceed 0.25 in.³.

Bearings shall be medium-series, 205 size in order to absorb loads generated outside the water brake. One bearing locknut with spacer shall be used instead of two locknuts.

Corrosion Resistance shall be provided by making all parts of stainless steel or providing corrosion-resistant coatings wherever possible.

GENERAL ELECTRIC SPACE POWER & PROPULSION SECTION CINCINNATI, OHIO 45215	SPECIFICATION NO. 02-0201-00-A
ENGINEERING SPECIFICATION	ISSUE DATE June 6, 1967

TITLE VIBRATION DAMPER FOR WATER BRAKE ON DRIVE TRAIN OF THREE-STAGE POTASSIUM VAPOR TURBINE	ORIGINAL CONTRACT NAS3-10606
	SECURITY CLASSIFICATION Unclassified

PREPARED BY: *H. Nichols* DATE 5-31-67
 APPROVED BY: *E. Schuetzler* DATE 6-1-67
 APPROVED BY: _____ DATE _____

ISSUED BY: D.A. Pritchett *D.A. Pritchett* MAIL ZONE N-21 PHONE 3929/2438
DRAFTING

SF 1092 (11/66)

SUPERSEDES SPECIFICATION NO. NEW	DATED	REVISION AUTHORIZATION -191-	SEE SPPS INSTRUCTIONS SERIES 03.106
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VIBRATION DAMPER FOR WATER BRAKE ON DRIVE TRAIN
OF THREE-STAGE POTASSIUM VAPOR TURBINE

- CONTINUED

DATE

6-6-67

NO.

02-0201-00-A

1. SCOPE.

1.1 Purpose. Design of a variable frequency vibration damper for installation on the water brake of the three-stage turbine drive train.

2. APPLICABLE DOCUMENTS.

2.1 Non-Government Documents. Dwg. 47R199250 - Layout of three-stage turbine drive train (may be scaled).

3. REQUIREMENTS.

3.1 Normal Operating Conditions

- a) Speed 0 - 20,000 RPM.
- b) Water brake torque - 1,100 in.-lb.
- c) Continuous operation requirement: 5,000 hrs.
- d) Total operating life. 15,000 hrs.
- e) Environmental Temperature: 200°F approximately.

3.2 Design Considerations

- a) Damping of natural frequencies of the water-brake system in the range of 15,000-to-20,000 RPM is of major interest, since those of lower speed will be of short duration and tolerable amplitudes.
- b) Final tuning of the damper may be performed in the test cell with the drive train operating at up to full speed.
- c) A layout design drawing or sketch with pertinent envelope and critical internal dimensions is required. GE will make final manufacturing drawings and will procure the damper components.
- d) Attachment to either the forward or aft flange of the water brake is desirable.
- e) No continuous flow of oil can be provided, nor any escape of oil or vapors into the test cell can be tolerated.
- f) Adjustment of the damper tuning mechanism during the 5,000 hour testing period cannot be performed due to the inherent danger of the operating potassium vapor system.
- g) The design must be such that the entire water brake is unrestrained for small rotations (of up to 5°) required for the measurement of torque.

GENERAL  ELECTRIC

**SPACE POWER & PROPULSION SECTION
CINCINNATI, OHIO 45215**

SPECIFICATION NO.

02-0206-00-A

ISSUE DATE

December 5, 1967

ENGINEERING SPECIFICATION

TITLE

GEAR COUPLING FOR THREE-STAGE POTASSIUM VAPOR
TURBINE TEST

ORIGINAL CONTRACT

NAS 3-10606

TYPE REVISION SYSTEM

 CN CID

PREPARED BY: _____ DATE _____

APPROVED BY: _____ DATE _____

APPROVED BY: _____ DATE _____

ISSUED BY: D.A. Pritchett MAIL ZONE N-21 PHONE 3929/2438
DRAFTING

SUPERSEDES SPECIFICATION NO.

DATED

REVISION AUTHORIZATION

New

SEE SPPS INSTRUCTIONS
SERIES 03.106

GEAR COUPLING FOR THREE-STAGE POTASSIUM VAPOR
TURBINE TEST -CONTINUED

DATE

12-5-67

NO

02-0206-00-A

- NS 1081
1. SCOPE
 - 1.1 Purpose
 2. APPLICABLE DOCUMENTS
 - 2.1 Non-Government Documents
 3. REQUIREMENTS
 - 3.1 Normal Operating Conditions
 - 3.2 Starting Conditions
 - 3.3 Braking Conditions
 - 3.4 Quality Control
 - 3.4.1 Material
 - 3.4.2 Unbalance
 - 3.4.3 Marking
 - 3.4.4 Manufacturing Process Quality Control
 - 3.5 Information to be Submitted With Proposal.
 - 3.5.1 Price Quotation
 - 3.5.2 Delivery Schedule
 - 3.5.3 Drawings
 - 3.6 Information to be Submitted After Receipt of Order
 - 3.6.1 Drawings and Specifications
 - 3.6.2 Inspection Reports
 4. DESIGN
 - 4.1 Vendor Analysis

GEAR COUPLING FOR THREE-STAGE POTASSIUM VAPOR
TURBINE TEST

-CONTINUED

DATE

12-5-67

NO

02-0206-00-A

1. SCOPE

1.1 Purpose. This document describes a special Fast-type gear coupling, for the purpose of obtaining vendor quotations and for purchasing.

2. APPLICABLE DOCUMENTS

2.1 Non-Government Documents

Dwg. No. 47R199253	Potassium Turbine Aft Drive Train Layout (3-stage)
Dwg. No. 47C141479	Gear Coupling - Steam Turbine to Water Brake
Dwg. No. 941D311	Water Brake Assembly
Dwg. No. 47B116005	Steam Turbine Spindle

3. REQUIREMENTS

3.1 Normal Operating Conditions

- a) Speed - 18,250 rpm
- b) Torque - 20 in.-lb
- c) Time - 5,000 hrs (continuous)
- d) Temperature - 200^oF
- e) Environment - Air, 100 percent relative humidity possible but not probable.
- f) Unbalance - Not to exceed $U = 0.006 W$
where $U =$ unbalance, gm-in.
 $W =$ weight of rotating part, lb
- g) Geometrical envelope - Defined by dwg. no. 47R199253 - "Potassium Turbine Aft Drive Train Layout" (may be scaled).
- h) Geometrical interfaces - Defined by dwgs. no. 941D311 - "Water Brake Assembly" and no. 47B116005 "Steam Turbine Spindle".
- i) Radial misalignment/coupling - 0.002 in. at assy., 0.008 in. hot.
- j) Axial misalignment/coupling - 0.005 in. at assy., 0.040 in. hot.
- k) Shafts' out of parallelism/coupling - 3 min. at assy., 6 min. hot.
- l) Coupling lubricant to be specified by vendor.

3.2 Starting Conditions

- a) Speed - 0 to 25,000 rpm
- b) Torque - 0 to 300 in.-lb
- c) Time - 100 hrs (intermittent)
- d) Otherwise same as 3.1 above

3.3 Braking Conditions

- a) 1,150 lb-in., to occur at least 10 times
- b) Otherwise same as 3.1 above

3.4 Quality Control

- 3.4.1 Material, material state, and surface treatment shall be specified by the vendor. A permanent rust-preventative treatment of all surfaces is desired.
- 3.4.2 Unbalance shall be ascertained by high-speed rotation of the coupling. The actual amount of unbalance at each end of the coupling shall be reported, and not simply that "unbalance is within $U = 0.006 W$ requirement". Parts shall be permanently matchmarked to indicate their relative positions during balancing.

GEAR COUPLING FOR THREE-STAGE POTASSIUM VAPOR
TURBINE TEST

-CONTINUED

DATE

12-5-67

NO

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3.4.3 Marking of each part by drawing no. and serial no. shall be done, using a permanent method such as electro-chemical etching.

3.4.4 Manufacturing Process Quality Control. The couplings shall be manufactured to the closest tolerances consistent with the most precise industrial practice, where accurate dynamic balance is a primary requirement of the system.

3.5 Information to be Submitted With Proposal

3.5.1 Price of quantities of 1, 2, and 3, of the coupling described herein, with price breakdown by significant tasks, such as special testing, which could be eliminated without affecting the rest of the proposal.

3.5.2 Delivery Schedule based upon receipt of order. Purchaser requires freedom to adjust the length of the floating shaft of the coupling at the latest reasonable date.

3.5.3 Drawings defining the geometry of the couplings in sufficient detail to allow length and interface control. Proprietary information should be omitted.

3.6 Information to be Submitted After Receipt of Order

3.6.1 Drawings and Specifications. As soon as possible, but not less than thirty (30) days after receipt of Purchase Order, the vendor shall provide two (2) reproducible copies of drawings and specifications which define the geometry of the couplings to be supplied, in sufficient detail to allow customer inspection. Proprietary information should be omitted.

3.6.2 Inspection Reports. Six (6) copies of two (2) reproducible copies of all reports of inspections which were made on the couplings specified herein, identified by part drawing no. and serial no. and with actual dimensions instead of "within specified tolerance", shall be supplied to the purchaser upon or before delivery of the parts.

4. DESIGN

4.1 Vendor Analysis of the proposed design is expected, since this specification and the accompanying sketches are only intended to show general concepts and geometrical limitations. Comments and evaluations of the oil collector intermittent lubrication method versus a continuous lube or constant supply system, the use of heavy oils versus grease, the effects of different misalignment conditions, and the use of straight versus curved teeth are especially solicited.

GENERAL ELECTRIC

SPACE POWER & PROPULSION SECTION
CINCINNATI, OHIO 45215

SPECIFICATION NO.

02-0145-00-A

ENGINEERING SPECIFICATION

ISSUE DATE

May 19, 1967

TITLE

REWORK OF STEAM TURBINE FOR THREE-STAGE
POTASSIUM VAPOR TURBINE TEST

ORIGINAL CONTRACT

NAS 3-10606

SECURITY CLASSIFICATION

PREPARED BY: B. L. Moon DATE 5.18.67

APPROVED BY: A. E. Eckard DATE 5-18-67

APPROVED BY: A. E. Nichols DATE 5-18-67

ISSUED BY: _____ MAIL ZONE _____ PHONE _____
DRAFTING

SUPERSEDES SPECIFICATION NO.

DATED

REVISION AUTHORIZATION

SEE SPPS INSTRUCTIONS
SERIES 03.106

REWORK OF STEAM TURBINE FOR THREE-STAGE
POTASSIUM VAPOR TURBINE TEST

- CONTINUED

DATE

May 19, 1967

NO.

02-0195-00A

1. SCOPE

1.1 Purpose

2. APPLICABLE DOCUMENTS

2.1 Non-Government Documents

3. REQUIREMENTS

3.1 Normal Operating Conditions

3.2 Starting Conditions

3.3 Overspeed Conditions

3.4 Overhaul

3.5 P/N 6042 Rotor Rework

3.6 Bearing Thermocouple Rework

3.7 Power Rework

3.8 Bearings

3.9 Lifting Eye Rework

3.10 Painting

3.11 Quality Control

3.11.1 Inspection

3.11.2 Spin Testing

3.11.3 Test Plan

3.11.4 Purchaser Monitoring of Spin Testing

3.12 Information to be Submitted With Proposal.

3.12.1 Price Quotation

3.12.2 Delivery Schedule

3.12.3 Drawings and Specifications

3.12 Information to be Submitted After Receipt of Order

3.13.1 Drawings and Specifications

3.13.2 Inspection Reports

3.13.3 Installation and Maintenance Instructions

3.13.4 Certification

REWORK OF STEAM TURBINE FOR THREE-STAGE
POTASSIUM VAPOR TURBINE TEST

- CONTINUED

DATE

May 19, 1967

NO.

02-195 00-A

1. SCOPE

1.1 Purpose. This document is a request for a quotation of price and delivery schedule for original-vendor modifications and repair to an existing Barbour Stockwell Co. Model 6037-S3, 8 in Horizontal Turbine.

2. APPLICABLE DOCUMENTS

2.1 Non-Government Documents. Dwg. no. 941D313 - Steam Turbine Rotor Modification.

3. REQUIREMENTS

3.1 Normal Operating Conditions.

- a) Speed - 18,250 RPM.
- b) Time - 5,000 hr. (continuous)
- c) Direction of rotation - clockwise when facing P/N 4276 Cover Assembly.
- d) Load - 20 in. lb. windmilling, with steam turbine acting as a standby unit to maintain 18,250 RPM if potassium test turbine loses speed.
- e) Steam supply - 150 psig. saturated steam available in quantities limited only by turbine inlet port diameter.
- f) Oil supply - 0.7 gpm of Socony-Mobil DTE 797 lubricating oil.

3.2 Starting Conditions.

- a) Speed - 0-18,250 RPM.
- b) Time - 100 hr. at 18,250 RPM.
- c) Maximum Torque - 200 in.-lb. at least, with rework intended to achieve at least 300 in.-lb. at 18,250 RPM.

3.3 Overspeed Conditions.

- a) Speed - turbine shall be capable of 5,000-hr. continuous operation at 25,000 RPM without sustaining internal damage.

3.4 Overhaul. The vendor shall thoroughly inspect the existing turbine at his plant, record significant observations, and supply purchaser with six (6) copies or two (2) reproducible copies of the statement of observations within thirty (30) days of the completion of inspection. Vendor shall provide a quotation of the price of overhauling which lists in detail the parts to be reworked or replaced. A completely new set of bearings, seals, and gaskets shall be installed, at least. The repair or replacement of other items shall be based upon the best experience and judgement of the vendor, considering the rework which is to also be accomplished. The vendor shall provide a list of recommended spare parts, including quantity and price of each.

3.5 P/N 6042 Rotor Rework. The turbine rotor shall be reworked per dwg. no 941D313 "Steam Turbine Rotor Modification", with the vendor acceptance of the order to rework

REWORK OF STEAM TURBINE FOR THREE-STAGE
POTASSIUM VAPOR TURBINE TEST

- CONTINUED

DATE

May 19, 1967

NO.

02-3195-00-A

indicating his agreement that the rework will not result in a deterioration of the reliability of the turbine. The fabrication of new locknuts similar to P/N 4195 Bearing Locknut shall be considered a part of the rotor rework. The rework quotation shall include supplying four (4) of the altered locknuts, with their costs listed as a separable sub-total. Rework shall also include rotor rebalancing to limits at least as tight as normal production specifications, which in any case shall not exceed

$$U = \frac{113.6 W}{N}$$

where. U = Maximum allowable dynamic unbalance, gr.-in.

W = Rotor weight, lb.

N = 18,250 RPM.

The mounting flange face and rabbet of P/N 6035 Bearing Housing shall be reworked, if necessary, to assure that the rotor runs concentric and square within 0.0005 in. TIR, as shown on the rework proposal drawing.

The vendor shall provide detail drawings with the quotation of price to which the rotor rework and new locknuts may be inspected, and which will allow coordination with mating hardware.

3.6 Bearing Thermocouple Rework. The vendor shall machine three thermocouple wells into P/N 6035 Bearing Housing, all located within 90° of the steam inlet ports, if possible. Each well should communicate with a different bearing and accept an oil-sealing chromel-alumel thermocouple Conax cat. no. SL-360-CA (turbine vendor specifying length). The bearing housing shall be permanently marked "fwd", "cntr", or "aft" beside the thermocouple wells which communicate with P/N 4169 Ball Bearing and P/N 5014 Ball Bearings Matched Pair, respectively.

The vendor shall install one set of thermocouples and supply two additional sets. Detail drawings showing thermocouple locations shall accompany the quotation of price.

3.7 Power Rework. The existing turbine shall be reworked according to procedures already established by the vendor for this type of turbine, which will increase its power output by approximately 50 percent. The rework shall be based upon unidirectional rotation clockwise when facing P/N 4276 Cover Assembly and an available steam supply of 150 psig saturated steam in a quantity limited only by the turbine inlet ports. The quotation of the price of power rework shall include a curve of predicted performance before and after rework, and steam flow rate requirements.

3.8 Bearings The vendor shall investigate and report the possibilities of improving bearing life. While the present bearing design is apparently quite satisfactory, it is possible that life could be substantially improved by using vacuum melted, forged races and balls 2 R_c points harder than the races. It is not the intention of this paragraph to require geometrical changes nor greatly increase the cost or procurement times of the bearings, but some delay and price increase would be acceptable to obtain the material changes.

SP 1073 A

REWORK OF STEAM TURBINE FOR THREE-STAGE
POTASSIUM VAPOR TURBINE TEST

- CONTINUED

DATE

May 19, 1967

NO.

72-0195-00-A

DELETE

3.9 Lifting Eye Rework. The vendor shall permanently attach a lifting eye which is circumferentially aligned with the steam inlet ports. With the weight vector directed vertically upward in Barbour Stockwell Co. dwg. no. 6037, the eye should be aligned to cause the weight vector to pass through the center of gravity of the assembled turbine (inlet ports are at the top of the General Electric installation). If the above requirements be impossible to meet, the vendor should propose an alternate lifting method.

3.10 Painting. All external parts of the steam turbine, excluding flange faces, shafts, threads, and other working surfaces, shall be painted with a color of the vendor's choosing which will withstand the working temperatures expected. Mask thermocouple well identification markings, if necessary.

3.11 Quality Control.

3.11.1 Inspection. Six (6) copies or two (2) reproducible copies of all inspection reports of dimensions generated by the rework required shall be delivered to the purchaser with or before delivery of the turbine. Actual dimensions measured shall be reported, instead of "within allowable tolerances".

3.11.2 Spin Testing of the turbine assembly after all rework is completed shall be conducted by the vendor for at least one (1) hour at 20,000 RPM and five (5) minutes at 25,000 RPM. The following parameters are to be recorded

- a) Temperature of each bearing.
- b) Lube oil flow rate, pressure, and type.
- c) Speed
- d) Vibration
- e) A maximum power or torque, speed, steam pressure and flow rate curve to 25,000 RPM is to be developed.

3.11.3 Test Plan. A statement of the values of the parameters to be imposed or produced during the test is to be submitted to the purchaser and approved by him in writing at least one week prior to testing. It is intended that the test be as comprehensive as possible using the vendor's existing test equipment. The testing cost shall be quoted as a separable sub-total.

3.11.4 Purchaser Monitoring of Spin Testing shall be allowed. The vendor shall notify the General Electric Company of their intention to test at least three working days in advance.

3.12 Information to be Submitted with Proposal.

3.12.1 Price Quotation. Quote the price of all work required by this proposal subdivided into the following items, each of which could be eliminated without affecting the other items

- a) Overhaul
- b) Recommended spare parts.
- c) Rotor rework: rotor
locknuts

REWORK OF STEAM TURBINE FOR THREE-STAGE
POTASSIUM VAPOR TURBINE TEST

- CONTINUED

DATE

May 19, 1967

NO.

72-0195-00-A

- d) Bearing thermocouple rework.
- e) Power rework.
- f) Bearings with improved material.
- g) Spin testing.
- h) Lifting eye rework. DELETE
- i) Painting

3.12.2 Delivery Schedule based upon receipt of order.

3.12.3 Drawings and Specifications sufficient to define the geometrical changes to be accomplished, specifically including interface dimensions, supply lines required, and expected maximum power or torque, speed, steam pressure and flow rate curve.

3.13 Information to be Submitted After Receipt of Order.

3.13.1 Drawings and Specifications. As soon as possible, but not later than thirty (30) days after receipt of Purchase Order, the vendor shall furnish two (2) reproducible copies of certified prints of drawings defining the basic steam turbine and rework to be accomplished, including referenced specifications. Sufficient detail shall be provided to allow proper installation, assembly and disassembly, and damage control inspection by the purchaser.

3.13.2 Inspection Reports. Six (6) copies or two (2) reproducible copies of all reports of inspections which were made as a result of work required by this specification shall be provided to the purchaser within one (1) week of the completion of the inspections.

3.13.3 Installation and Maintenance Instructions. Six (6) copies or two (2) reproducible complete sets of any installation or maintenance instructions which exist for Model 6037 turbine shall be provided with or before the delivery of the equipment.

3.13.4 Certification that dynamic balancing to (state value) gm.-in. and spin testing conforming to (state test plan) have been successfully accomplished, and that bearings of (state identification numbers and material) have been installed in the turbine and provided as spare parts is required with or before delivery of the equipment. Two (2) reproducible copies of the maximum power or torque, speed, steam pressure and flow rate curve developed by testing shall be provided.

EXCEPTIONS TO STEAM TURBINE REWORK SPECIFICATION 02-0195-00-A

Accomplish the following tasks concerning the existing Barbour Stockwell Model 6037 S3, 8 in. horizontal turbine (referenced paragraph numbers refer to Engineering Specification No. 02-0195-00-A "Rework of Steam Turbine for Three-Stage Potassium Vapor Turbine Test".) The full test of the specification shall apply except as noted:

Para. 3.2 c) - Delete "with rework intended to achieve at least 300 in.-lb at 18,250 rpm."

Para. 3.4 - Overhaul the turbine as outlined in 6.19-67 Barbour Stockwell letter W.B. Coffin to J.L. King re: Request for Quotation No. X-280714 -

- a) Rework rotor to put it in first-class operating condition without special machining per dwg. 941D313 Rev. O.
- b) Use nozzle box assembly with minimum overhaul rework and with power rework.
- c) Replace bearing housing and all other parts except those specifically excluded below.
- d) Do not replace the following parts of Barbour Stockwell Co. dwg. no. 6037:

Ref. #9, Part 6041 End Cap

Ref. #12, Part 2429 Screw

Ref. #13, Part 5011 Oil Seal

Ref. #14, Part 6043 Retainer Plate

Ref. #40, Part 5007 Rotor Nut

Ref. #41, Part 5009 Rotor Nut

(A request for quotation on parts replacing the above will forthcoming.)

Ref. #37, Part 2433 Magnetic Pickup and Cable

Ref. #38, Part 4176 Cover Assembly

Ref. #39, Part 4276 Cover Assembly

(Existing parts believed satisfactory - Notify Purchaser in case of disagreement.)

Para. 3.5 - Delete rotor rework, locknut, and quotation requirements only. Hold balancing until new rotor locknut and spindle are received.

Para. 3.6 - Accomplish bearing thermocouple rework.

Para. 3.7 - Accomplish power rework, using existing rotor.

Para. 3.8 - Use present stock bearings.

Para. 3.9 - Delete lifting eye rework.

Para. 3.10 - Accomplish painting.

Para. 3.11.2 - Accomplish spin testing except delete:

"e) A maximum power or torque, speed, steam pressure, and flow rate curve to 25,000 rpm is to be developed."

Para. 3.11.3 - Delete test plan requirements.

Para. 3.12 - Delete proposal information.

Para. 3.13.4 - Change to "Certification that dynamic balancing to (state value) gm.-in. and spin testing have been successfully accomplished is required with or before delivery of the equipment."

(Re-type copy of original letter, excluding detailed cost quotation and recommended spare parts.)

June 19, 1967

General Electric Company
Cincinnati, Ohio 45215

Attention Mr. J. L. King, Buyer
Space Power & Propulsion Section
Purchasing & Programming Dept.

Re: Request for Quotation No. X-280714

Gentlemen:

Per paragraph 3.4 of applicable specifications 02-0195-00-A, we advise as follows.

Your Turbine Serial No. 6037-S3 has been disassembled and all parts have been inspected and reviewed in order to ascertain the extent of parts that can be reworked and used again and those that must be replaced. The following is the result:

Ref. #44, Part 6042, Rotor will require extensive work in order to put it in first class condition not taking into consideration the special machining in order to modify it in accordance with GE Drawing 941D313 Rev. 0 or the milling to enlarge buckets for maximum output performance of a standard bi-directional Turbine.

Ref. #50, Part 6038, Nozzle Box Assembly can be used with a minimum amount of rework except that we would add additional flange bolts and, for either power increase option, changes would be made as regards inlet drive steam and the exhaust outlet.

Ref. #20, Part 6035, Bearing Housing, in our opinion, should be replaced with a new one as critical bearing bores are far out of tolerance and, for the three new thermocouples we do not think it wise to contend with the existing entrance holes in the old Bearing Housing. Furthermore, your old Bearing Housing is pretty well beaten up.

All other parts for your Special Flange Mounted Turbine would be new as existing ones are badly worn or battered and in the case of Ref. #11, Part 4181, Pressure Ring and Ref. #54, Part 5062, Preload Sleeve we have improved our design by incorporating ample oil galleries to eliminate lubricating oil build up which causes excessive internal temperatures thereby causing bearing failures.

June 19, 1967

We enclose, as requested, our list of Recommended Spare Parts with current unit prices noted thereon.

In order to accomplish torque requirements outlined under paragraph 3.2, there are two possible approaches. The most economical would be that of incorporating oversize nozzles and other related internal changes which will give you predicted output performance in accordance with the enclosed torque speed characteristics. This may be compared with our standard curves. Maximum flow rate or consumption under full load conditions are noted on each for your 150 psig operating pressure. For maximum performance for our 8" Turbine it would be necessary to go to a Special Uni-directional configuration which would essentially double the output performance characteristics shown by our standard performance curves.

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:
:

(excluded detailed cost quotation)

When you have had a chance to go over the enclosed quotation and material, we will be pleased to answer any questions you may have. In the interim, we are holding your disassembled Turbine at our plant pending your decision.

Assuring you of our cooperation, we are

Cordially yours,

BARBOUR STOCKWELL CO.

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:
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:
:

(excluded recommended list of spare parts)

DISTRIBUTION LIST

NAS3-10606

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NASA - Lewis Research Center
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Cleveland, Ohio 44135
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National Aeronautics & Space Administration
Scientific & Technical Information Facility
P.O. Box 33
College Park, Maryland 20740
ATTN: Acquisition Branch, (SQT-34054)
(2 + reproducible)

NASA - Lewis Research Center
21000 Brookpark Road
Cleveland, Ohio 44135
ATTN: W. L. Stewart (77-2)

NASA - Lewis Research Center
21000 Brookpark Road
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ATTN Dr. B. Lubarsky (500-201)

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21000 Brookpark Road
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ATTN: J. E. Dilley (500-309)

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ATTN T. A. Moss (500-201)

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Cleveland, Ohio 44135
ATTN N. Musial (501-3)

NASA - Lewis Research Center
21000 Brookpark Road
Cleveland, Ohio 44135
ATTN: Library (60-3) (2)

NAS3-10606

NASA - Lewis Research Center
21000 Brookpark Road
Cleveland, Ohio 44135
ATTN Report Control Office (5-5)

NASA - Lewis Research Center
21000 Brookpark Road
Cleveland, Ohio 44135
ATTN P. E. Foster (3-19)

NASA - Lewis Research Center
21000 Brookpark Road
Cleveland, Ohio 44135
ATTN: D. Drier (21-4)

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