NASA CR 54823
AGC 8800-18

MECHANICAL DESIGN OF THE

M-1 AXIAL FLOW LIQUID HYDROGEN FUEL PUMP

By

P. J. Regan

Prepared for

National Aeronautics and Space Administration

Contract NAS 3-2555

AEROJET GENERAL TIRE GENERAL

AEROJET-GENERAL CORPORATION

SACRAMENTO, CALIFORNIA

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TECHNOLOGY REPORT

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15 February 1966

Contract NAS3-2555

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ABSTRACT

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This report presents the mechanical design of the M-1 axial flow liquid hydrogen pump. Included are descriptions of the pump assembly and major pump components, anticipated stress and vibration levels, materials, fabrication considerations, rotor balancing and assembly procedures. Emphasis is placed upon the first development unit that was built. Author

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I. SUMMARY

The mechanical design and the assembly of the fuel turbopump are described in this report. The hydrodynamic design, turbine design, and bearing design are discussed to a lesser extent because these technological aspects are covered in depth in separate reports.

The fuel pump is an axial flow pump consisting of a first-stage inducer and stator, a second-stage inducer and stator, and eight mainstages; each stage consisting of rotor and stator blades.

The liquid hydrogen passes through the pump and is discharged to a 10-in. inside diameter line located in a plane normal to the axis of pump rotation.

The basic mechanical design considerations for the fuel turbopump are listed below.

Blading is removable to facilitate stage matching to obtain desired performance. This permits maximum variations in blading with a minimum hardware cost.

All exterior housings are one-piece (i.e., no axially split housings) to minimize the possibility of interior surfaces adjacent to rotating blades from becoming out-of-round. It also improves static seal reliability.

Component weight was considered to be of secondary importance since the major objective of the fuel turbopump assembly was to obtain satisfactory function and performance. The early stage of the development program and the program schedules were contributing factors in making this decision.

The initial fuel turbopump assembly, as delivered to the test area for installation on Test Stand E-1, weighed approximately 6800 lb. Target weight for the flight type turbopump is 3300 lb.

Additional design and component development information pertinent to the M-1 axial flow liquid hydrogen pump are contained in separate reports.(1),(2),(3)

II. INTRODUCTION

The M-1 fuel turbopump assembly, Model I, S/N 001, is described in this report. It was designed, built, and tested by the Aerojet-General Corporation,

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(1) Report No. NASA CR-54822, entitled Hydraulic Design of the M-1 Liquid Hydrogen Turbopump

(2) Report No. NASA CR-54821, entitled Mechanical Design of a Two-Stage Impulse Turbine for the Liquid Hydrogen Turbopump for the M-1 Engine

(3) Report No. NASA CR-54826, entitled Design and Development of Liquid Hydrogen Cooled 120mm Roller, 110mm Roller, and 110mm Tandem Ball Bearings for the M-1 Fuel Turbopump

Sacramento, California under Contract NAS3-2555 from the NASA Lewis Research Center, Cleveland, Ohio.

The turbopump is of axial flow design with two inducer stages and eight mainstages and is intended for liquid hydrogen service. The propellant is supplied to the pump at a minimum pressure of 30 psia and is discharged from the pump at 1820 psia. The design discharge flow rate of liquid hydrogen is 600 lb/sec.

The turbopump described in this report was the first unit to be assembled. It has a single-stage turbine developing a maximum of 62,000 horsepower, which while less than the required 90,000 horsepower, is sufficient for the initial testing phase of the development program.

All fuel turbopump assembly rotating components and turbine stationary components are close to the flightweight configuration. Other stationary components were functionally designed with no consideration given to weight.

III. GENERAL CONSTRUCTION OF ASSEMBLY

A. DESCRIPTION

The fuel turbopump assembly Model I is a 10-stage axial flow unit designed to deliver 600 lb/sec of liquid hydrogen at a total discharge pressure of 1800 psia. The 10 stages consist of a first-stage mixed flow inducer and stator, a second-stage axial inducer and stator, and eight axial mainstages. Each mainstage consists of a rotating row of blades and a stator. Power is supplied by a single-stage turbine which is coupled directly to the pump rotating assembly. A cross-sectional view of the fuel turbopump assembly, including nomenclature and material notations is shown as Figure 1.

The fuel turbopump assembly is an integrated design in that no separate pump assembly, power transmission assembly, or turbine assembly exist as completely independent items. The bearings are contained within the pump envelope. The turbine shaft, which carries the turbine-end roller bearing, is installed directly into the pump rotor by using a series of pilot diameters and a spline. This coupling is designed so that rigid joint is formed at operating temperature, which results in a single rotating assembly comprised of turbine and pump components.

There are three versions of the M-1 fuel turbopump assembly. The only differences are in the turbine area. In the Model I, there is a single-stage turbine and an exhaust cone. The Model IIA has a two-stage turbine and an exhaust cone identical to that of Model I. The Model IIB unit is identical to Model IIA except that the exhaust cone is replaced with a hemispherical exhaust collector having two outlets for ducts which route the exhaust gas to the oxidizer pump turbine inlet manifold.

Various turbopump assembly components are shown in APPENDIX A of this report.

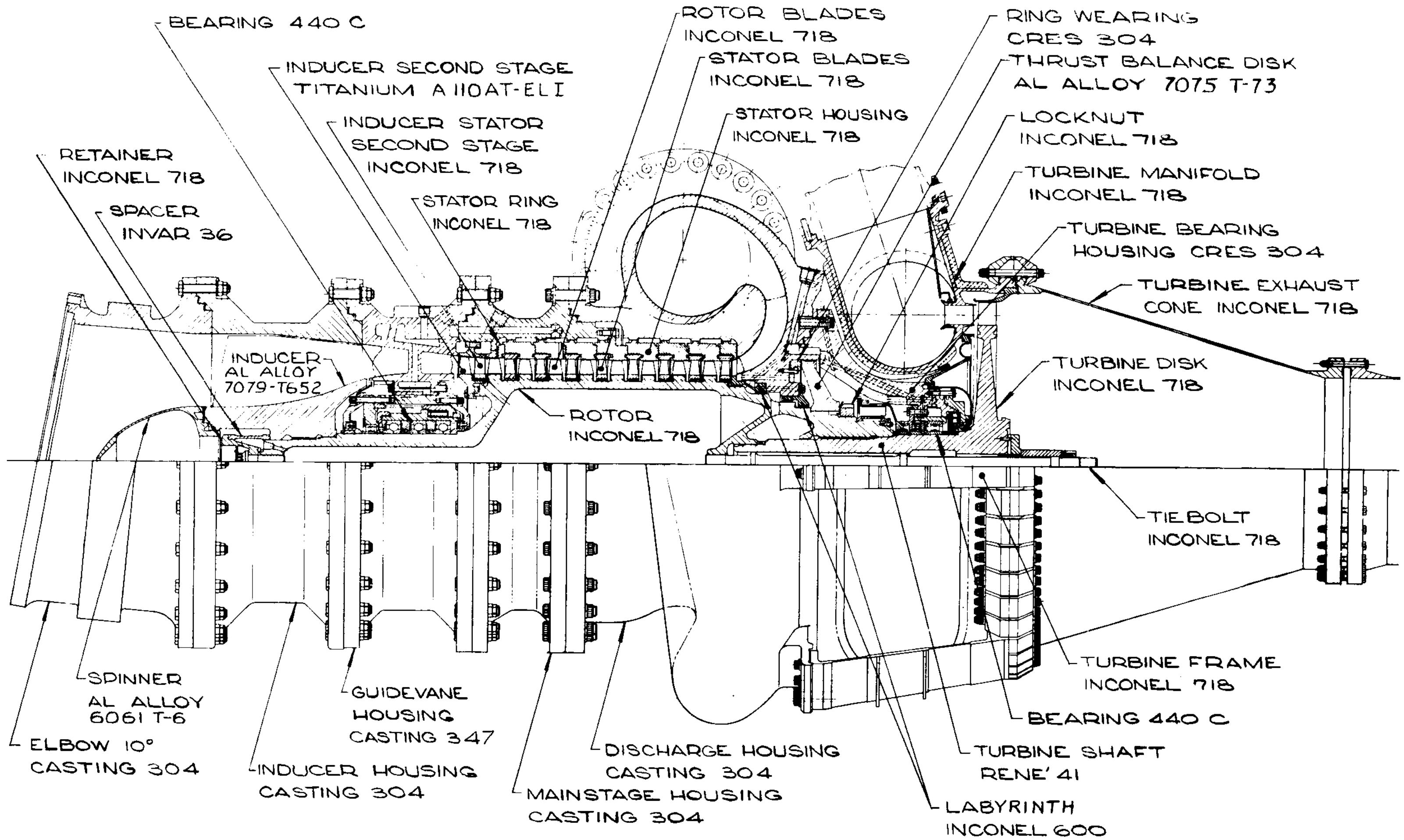


Figure 1. Nomenclature and Material

1. Rotating Components

The basic rotating component of the fuel turbopump assembly is the pump rotor, which is an integral unit fabricated by welding to eliminate any need for bolted joints. The pump rotor carries the 10 stages of rotating blades. All blading is removable to facilitate replacement should hardware become damaged as well as to permit blade variations at minimum cost to permit optimum stage matching to achieve desired pump performance.

The pump rotor also carries a thrust balance disc, which supplies a force to counteract or balance the thrust developed by the pump. Without this thrust balancing disc, bearing loads would be so large as to make design of adequate bearings impossible.

The Model I and Model IIA units are intended for component testing only. The Model I is capable of operating up to 12,000 rpm and thus cannot meet the nominal design flow and pressure which requires a pump speed of 13,225 rpm. The Model IIA is capable of full speed operation. The Model IIB version is intended for use in the engine system.

A torquemeter is installed on the turbine end of the pump rotor, where it measures the total turbine torque input to the pump section. This data is used in determining pump efficiency as well as the accuracy of the design calculations related to turbine and pump performance.

All rotating ocmponents are of flight-weight design. Weight reduction became a vital factor in obtaining an acceptable calculated critical speed for the fuel turbopump assembly.

2. Stationary Components

In contrast to the lightweight rotating components, the exterior housing, which are machined from castings, are of heavy construction. Component weight was only a minor consideration when function and schedule lead time were considered. All housings are basically cylindrical in design, which eliminated the sealing and distortion problems associated with axially split housings. Through bolts and nuts are used as fasteners where possible for the housing interfaces.

Two housings are of particular interest. The first is the guide vane housing which serves two functions. It provides a set of guide vanes that direct the fluid flow from the first-stage inducer to the second-stage inducer. It also serves as a bearing housing which supports the pump-end bearing package. All loads, radial and thrust, are transmitted through this housing, which also contains passages for bearing coolant flow, both in and out of the bearing package. This housing contains passages used for routing instrumentation wires which originate within the bearing package.

The discharge housing is the second housing of special interest because of its fluid passage. This fluid passage consists of a set of 20 diffuser vanes directing the fluid flow into a constant velocity volute. The liquid is discharged from the pump in a plane normal to the pump axis of rotation. It also contains all of the necessary passages plus a high pressure inlet and low pressure return flow which is needed to operate thrust balance piston. Passages are also provided for the turbine-end bearing coolant supply.

Sealing of all external housing flanged joints is accomplished by using double conical seals arranged concentrically. The inner seal is considered the primary seal. At the time of the initial pump design, this system of redundant seals was considered to be the most satisfactory for containing liquid hydrogen even though large heavy flanges and high seating loads were required. During fuel turbopump assembly tests of the first build-up, the primary seal leaked at all points instrumented for seal leakage, but the secondary seal performed satisfactorily with no evidence of leakage. None of these joints were disassembled for inspection.

Sealing of all the external lines and fittings required for return flow lines, bearing coolant lines, and instrumentation was accomplished using AN fittings and metallic seals, Voi-shan and/or K-seals. These seals appeared to be satisfactory in all cases.

The stationary blading is contained within the exterior housings and is removable. This design was developed for the same reasons delineated for the rotating component blading.

The static turbine components consist of the support frame inlet manifold and exhaust cone.

The support frame consists of three conical segments or support arms, fabricated from sheet stock. The three segments were required to provide suitable openings for the turbine manifold, inlet pump balance disc return flow lines, turbine bearing coolant lines, and instrumentation. The support frame can be adjusted axially for properly locating the inlet manifold nozzle in relationship to the first-stage turbine rotor.

The turbine inlet manifold is a constant area torus containing 37 nozzle vanes. The manifold is insulated to minimize the transfer of heat to pump components. Heat shields and a metal seal are provided to confine all hot gas flow to the turbine area and exhaust system. The area between the manifold and the turbine bearing housing has been provided with a purge line to keep the area free of water. Gaseous nitrogen is the purge gas used prior to chilldown and gaseous helium is used during chilldown and testing. The presence of water from condensation, rain, or the deluge system could damage the turbine bearing housing inlet manifold seal because of expansion resulting from freezing.

The exhaust cone is a simple sheet metal part that directs the turbine exhaust gases into a test stand exhaust duct.

The pump-end bearing package contains a roller bearing for radial loads and a ball bearing system for thrust loads. The roller bearing has a free inner race and a bearing cage with roller retainers. These features facilitate pump assembly.

The thrust bearing system consists of three split inner race ball bearings of gothic arch design. These bearings are ground as a set for maximum load sharing and are installed to assure load sharing when the net thrust direction is toward the turbine.

The thrust bearing system is designed so that the ball bearings can transmit a maximum radial load of 200 lb. This is accomplished by mounting the thrust bearings in a radially flexible housing, which ensures that the roller bearing carries virtually all of the radial loads.

Stacked Belleville washers are installed so that they react to thrust bearing loads when net thrust is toward the turbine. This installation permits increased axial travel of the pump rotor, thereby providing a greater variation in thrust balance disc control pressure. An axial gap on the high pressure side of the thrust disc provides variation in thrust balancing loads. The stacked Belleville washers also are a source of friction damping to suppress rotor axial vibrations. The design of the washer stacks permits variations in the net spring constant so that by varying the method of stacking washers, the spring constant can be made either linear or non-linear.

The bearing package instrumentation consists of: two platinum-resistance-type temperature sensors on each bearing; a strain gage network bonded to the flexible housing for measuring thrust; and two quartz-crystal-type accelerometers, which are spaced 90 degrees apart and are mounted on each bearing outer race. Instrumentation wires are routed through passages in the guide vane housing to suitable connectors on the exterior of the housing.

Each bearing is provided with a coolant spray ring. Coolant is routed into and out of the bearing areas via a series of passages.

All bearing coolant flow passes through 10 micron strainers prior to entering the guide vane housing passages and flowing into the bearing spray ring circuits.

Shaft riding seals are provided at each end of the bearing package to prevent foreign matter from entering the bearing areas. These seals operate with 0.001-in. to 0.003-in. radial clearance from the shaft, thus providing a controlled flow of coolant from the bearing area to the pump interior.

The turbine end of the rotating assembly is supported in a single roller bearing. This bearing is supplied with coolant which is routed from the pump discharge housing through 10 micron filters and through the bearing housing into a spray ring located on the pump side of the roller bearing.

A shaft riding seal is provided to separate the bearing cavity from the low pressure side of the thrust balancing disc.

This shaft riding seal operates with a 0.001-in. to 0.003-in. radial clearance thus providing a controlled leakage.

The bearing coolant passes through the bearing and into the turbine area. A labyrinth on the turbine side of the bearing provides control of the coolant flow. A lift-off seal, which seats against a running ring mounted on the turbine shaft, is positioned between the bearing and the labyrinth. This seal is actuated by introducing pressure into a double bellows; applying pressure lifts the seal face from the running ring allowing bearing coolant to flow. During storage, childown, and all non-running conditions, the lift-off seal remains closed, thereby retaining purge gases or liquid hydrogen within the pump. The lift-off seal is actuated immediately prior to starting pump operation to permit the bearing coolant to flow in accordance with design requirements.

Turbine bearing instrumentation consists of temperature sensors and accelerometers applied in the same manner as those used in the pump side bearing package. Instrumentation wires are routed through the turbine bearing housing to suitable connectors on the exterior.

3. Flow Paths

The liquid hydrogen flow through the fuel turbopump assembly falls into two categories: main flow and recirculating flow. The recirculating flow includes the thrust balance system, pump-end bearing coolant, part of the turbine bearing coolant, and pump rotor internal flow.

a. Main

The liquid hydrogen enters the pump through the inlet elbow and travels into the first-stage inducer, from where it moves into the guide vane housing. The guide vanes direct the flow, preparing it for entry into the second-stage inducer. After passing through the second-stage inducer, the fluid pressure is super-critical. The liquid travels through the eight row mainstage and into the discharge housing, where the flow is diffused and flow direction is changed so that it is traveling in a plane normal to the pump axis of rotation. Propellant flow is shown in Figure 2.

b. Internal Rotor

The hollow pump rotor is used as a flow path to transfer liquid passing through the discharge housing labyrinths to a low pressure area, specifically the second-stage inducer stator. A small portion of the flow travels through the vent hole in the spinner into the area immediately upstream of the first-stage inducer. Pump rotor internal flow is shown in Figure 3.

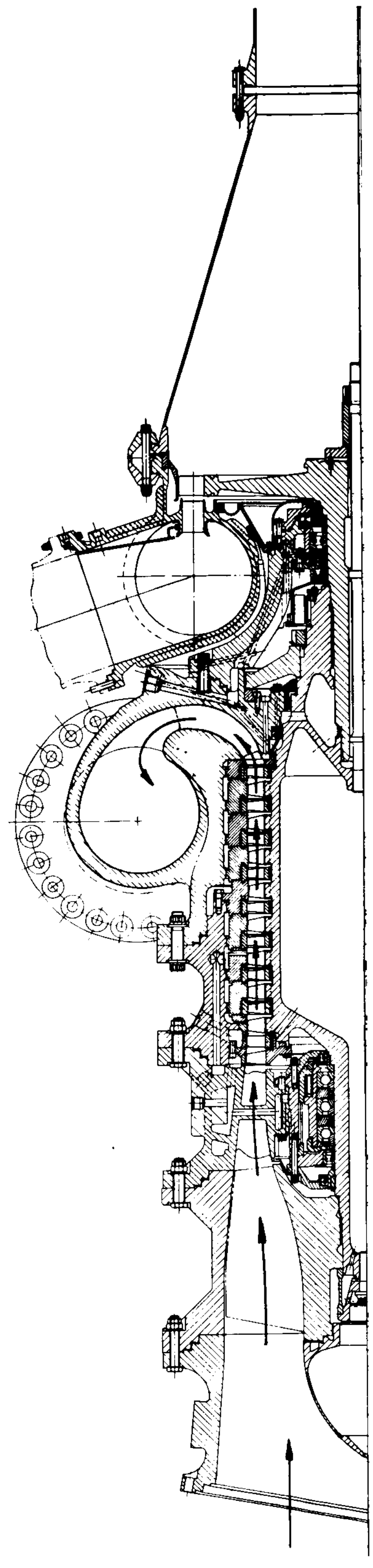


Figure 2. Propellant Flow

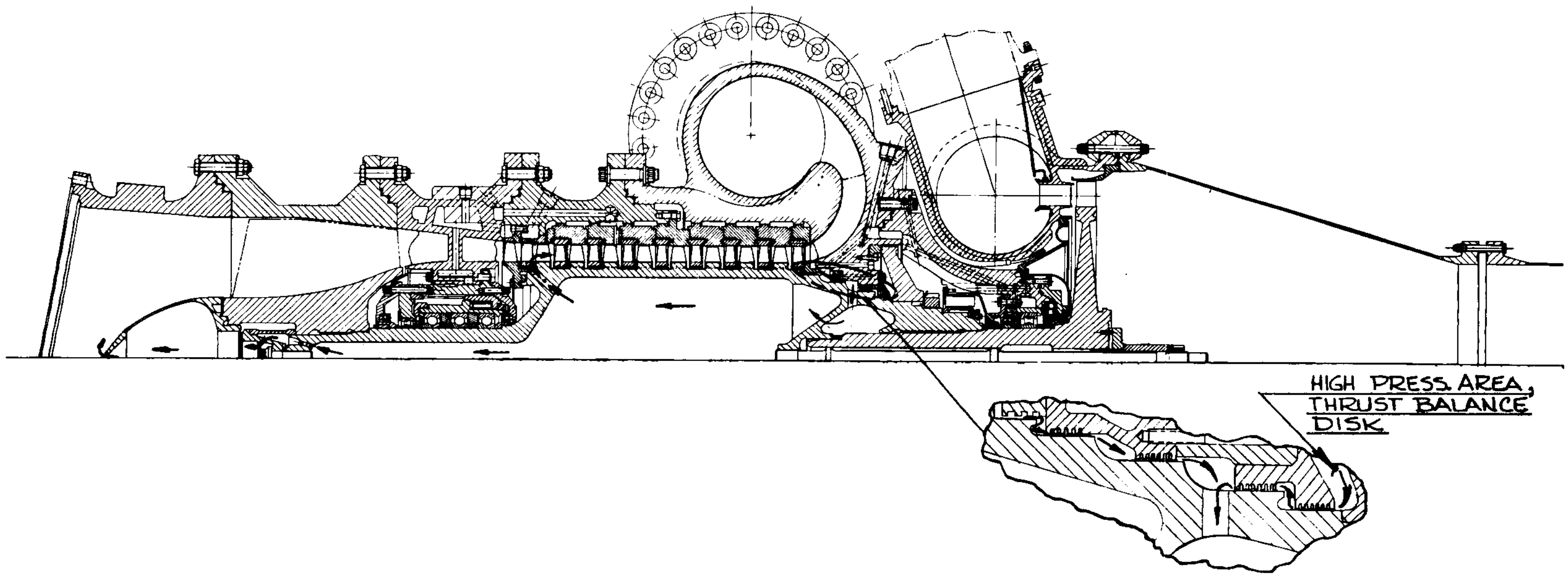


Figure 3. Pump Rotor Internal Flow

This internal flow serves two purposes. It assures the maximum pressure drop across the discharge housing labyrinths by transporting the labyrinth flow to a low pressure area and the passages permit venting and rapid chill-down of the rotor.

c. Thrust Balance System

High pressure liquid hydrogen is taken from the discharge of the last mainstage stator (see Figure 4). The liquid is carried through passages in the discharge housing to the high pressure side of the thrust balance disc. The flow then passes through two restrictions arranged in series. The first restriction is an axial gap which varies directly with thrust-caused pump rotor axial movement. The pressure decrease varies inversely with the gap width, thus supplying a variable force to the thrust balancing system. This variable force tends to cause the rotor to return to a centered position should axial displacement occur.

The second restriction is a constant radial gap; the fluid moving in the axial direction. This radial gap provides the major pressure drop across the thrust balance disc thus creating the major part of the pressure differential with it's resulting thrust balancing force. Since the radial gap remains constant at the pump operating design point, the thrust balancing force caused by the radial gap remains constant. Analysis reveals that this major force is of such magnitude that the pump can operate satisfactorily if the axial gap restriction is removed from the thrust balancing system.

The fluid, now at a low pressure of 300 psia to 600 psia static, travels from the area on the turbine side of the balance disc into an annular collector. It then moves through passages in the discharge housing to the return flow manifold. This manifold ducts the liquid hydrogen to the mainstage housing where the fluid re-enters the main pump flow by means of ports in the second-stage inducer stator.

The return flow manifold is comprised of two identical systems; each system containing an instrumented manifold. Fluid pressure, velocity, and temperature are measured. These manifolds also house a replaceable orifice plate, which permits variations in pressure differential across the thrust disc by controlling the liquid pressure on the turbine, low pressure, side of the thrust disc. These orifices can be readily changed because they are located on the exterior of the turbopump assembly and are accessible at all times.

The fuel turbopump assembly has two separate bearing coolant circuits; one for the pump-end bearings and a second for the turbine-end bearings (see Figure 5).

d. Pump-End Bearings

The pump-end bearing coolant system inlet is located between the third mainstage rotor row and the third mainstage stator row (see Figure 5).

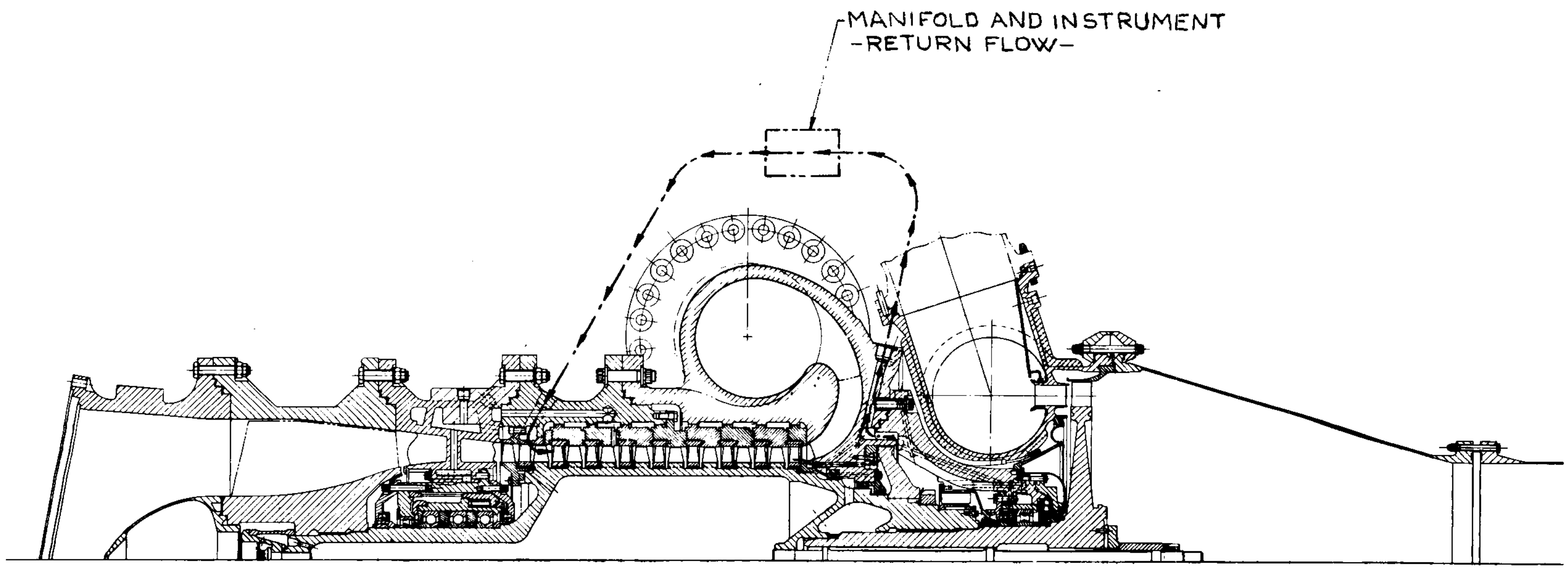


Figure 4. Thrust Balance System Flow

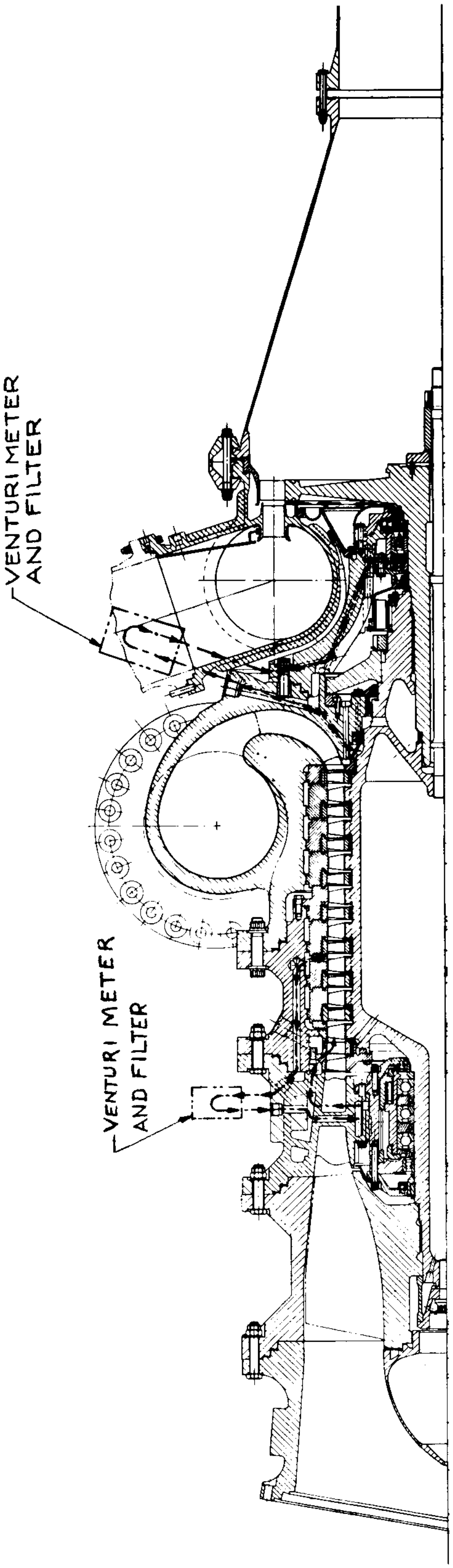


Figure 5. Bearing Coolant Flow Circuits

The coolant enters a series of holes in the stator blade retaining ring and is then carried forward through the mainstage housing into the guide vane housing. External tubes carry the fluid through flow measuring venturi, through 10 micron filters, and then back into the guide vane housing.

The coolant is carried to the bearing package where it is routed to the spray rings provided for each of the four bearings. After passing through the bearings, the coolant is collected, routed in exhaust passages to the guide vane housing, then by means of internal passages to the second-stage inducer stator.

The coolant re-enters the main pump flow by means of ports provided in the second-stage inducer stator. A small percentage of the bearing coolant leaks through the shaft riding seals installed on each side of the bearing package. This seal leakage flow re-enters the main pump flow in the areas of the leading and trailing edges of the guide vanes.

e. Turbine-End Bearings

The turbine-end bearing coolant inlet is located between the two labyrinths adjacent to the eighth mainstage stator (see Figure 5). The coolant is carried by means of drilled holes to the outside of the discharge housing. External lines containing 10 micron filters and flow measuring venturi are used to bridge the discharge housing turbine bearing housing interface. The coolant, after passing through the external lines, enters the turbine bearing housing and is routed to the roller bearing spray ring. The coolant then passes through the lift-off seal and labyrinth areas, and is finally exhausted into the area between the turbine inlet manifold and the first-stage turbine.

B. DESIGN PHILOSOPHY

1. Sealing

Seals are divided into two categories, static and dynamic. Each category has its own special requirements.

Static seals were selected based upon a zero leakage criteria and existing technological information. Repeated assembly and disassembly of components was added to this zero leakage requirement. Conical seals appeared to be the best available for this application although the seal glands required tight machining tolerances. In an effort to achieve zero leakage, double conical seals were to be used for all external housing flanged joints. A leakage monitoring port was located between the two concentric conical seals as an aid in determining the effectiveness of the seals.

Static seals for instrumentation ports and external flow lines were selected to be compatible with a standard AND 10050 port. Another consideration was that the seal selected did not damage the port surface. Such damage would necessitate rework prior to installing new seals. Conical seals were considered for this application but proved to be too cumbersome and expensive. NERVA test experience was used as a guide in selecting the port seals.

Internal static seals were generally selected upon the basis of available installation area. In general, metallic O-rings fulfilled design requirements for minimum leakage combined with a minimum installation size and ease of gland machining.

All housing interfaces having double conical seals use the metal-to-metal flange face option permitted by seal gland dimensioning specification. The metal-to-metal flange interfaces provide maximum control of housing assembly length and alignment, which has a direct effect upon bearing spacing and alignment.

The design of the dynamic shaft riding seal was based upon the need for a controlled low leakage rate. No attempt was made to achieve zero leakage. Pump design was simplified by the controlled leakage approach because the need for vent lines to draw off liquid from the areas adjacent to the seals and to return this fluid to the main flow stream was eliminated. The need for purge lines into these same areas was also eliminated because the flow through the seals under static conditions is sufficient to achieve a satisfactory purge. Existing design data were applied to the selection of seal diameters to minimize the need for development in this area. Seal surface speeds were kept within acceptable limits in accordance with existing available test data.

Tests of the shaft riding seals designed for use in the fuel turbopump assembly were conducted. These tests confirmed the materials selected; P5N carbon and LW-5 flame-plated journal surfaces. The tests also revealed that a seal radial clearance of 0.001-in. to 0.003-in. was desirable because carbon wear was greatly reduced when compared with the initial design requirement of an interference fit, carbon to shaft, at assembly. The small radial clearance did not result in an unacceptable leakage rate. The shaft riding seals installed in the fuel turbopump assembly reflect the results of these tests.

The lift-off seal located at the turbine-end of the pump falls partially into both of the above categories. This seal was to provide zero leakage under all conditions when the pump was not operating (zero rpm). Also the seating face of the seal was to lift off during all pump operating conditions and provide a passage for liquid hydrogen. During pump shutdown, the seal closed before the pump components stopped rotating. Based upon these requirements, dynamic face seal data, coupled with a suitable test and development program, were used to produce an acceptable final design.

2. Rotor Alignment

The alignment of the rotor as an individual component was achieved by making the rotor an integral unit. Thus, all bolts, joints, pilots, or similar complicating design features required by a segmented design were eliminated. Suitable low temperature tests were made to assure that the integral rotor was not subject to thermal distortion.

Alignment of the rotor to the pump housings was achieved in the following manner.

Metal-to-metal flange interfaces are specified for all external housing to control axial length. This restriction is compatible with the use of double conical seals.

A primary piloting surface is used at each external housing joint instead of relying solely upon the piloting capabilities of the conical seals.

Line bored housing subassemblies are used to eliminate tolerance accumulation caused by intermediate housing interfaces. Dowels are used to assure accurate alignment of housings after initial assembly in preparation for line boring operations.

All pump rotor concentricity tolerances are based upon the use of roller bearing journals as coincident datums.

Minimum tolerances are consistent with the size of the individual parts and machining difficulty of the materials involved.

3. <u>Thermal Symmetry</u>

Wherever possible, thermal symmetry is achieved by the use of cylindrical components. Split rings and axial split housings are avoided, if possible, because of their susceptibility to distortion caused by thermal gradients within a specific part. Ribs are considered detrimental because of the probable effect upon thermal gradients in the part which results in distortion and high local stresses. The use of ribs is restricted to absolute necessity. In general, efforts were made to maintain uniform wall thicknesses to keep variations in thermal gradients to a minimum within any component. Only one component deviated from the design philosophy; the discharge housing. This housing is asymmetric because of the discharge volute. Suitable tests were planned to determine the amount of distortion caused by both temperature variations and fluid pressures.

4. <u>Assembly</u>

Assembly was an important consideration in the design of the fuel turbopump assembly because of the following considerations.

a. The complex nature of the axial flow pump.

b. The need for frequent disassembly and reassembly of the same hardware in the development phase.

c. The capability to replace only specific items instead of a complete turbopump assembly because of the high hardware costs. This permits maximum reuse of components in the event of either failure or redesign of pumping elements.

d. Removable blading was required throughout the pump to permit changing the blades if required to produce design performance.

A design philosophy that facilitates assembly was followed. Through bolts with nuts were used for all major joints. Bolts secured in tapped holes would be used only where no option existed. This minimized the possibility of damage to internal threads, which is costly and time consuming to rework or repair during assembly. Static seals were to be replaceable. The glands or sealing surfaces were not to require rework or refurbishing prior to reuse. Wherever possible, components would be provided with positive keys. guides, and tooling holes to assure that they could be assembled correctly. Puller holes or grooves would be provided where necessary to assist disassembly. Handling holes or bosses would be provided where necessary to aid assembly, machining, and transportation.

5. Fabrication

Wherever possible, component design was based upon the use of existing technology for manufacturing, tools, and methods.

Castings were preferred because of low tooling costs for the small number of units to be made in the initial development phase of the program.

All rotating components, because of their high cost, are basically flightweight design with ample margins of safety based upon the existing design and hydraulic data predictions.

6. Ability to Instrument

Because of the integrated nature of the fuel turbopump assembly, instrumentation presented a problem at the initial conceptual stage. Instrumentation location and lead routing was a major consideration in the detailed design of each affected component. Space limitations dictated instrument locations. This required design approaches different from those generally used for turbopumps. In the case of the mainstage static pressure pickups, probes had to be designed so that they would pass through an outer and an inner housing, be in the proper location in relationship to the blades, and remain seated during operation.

In general, the philosophy was to fully instrument the fuel turbopump assembly using available sensors wherever possible and to design special equipment only where space or location rendered standard items inadequate.

7. Mounting

Mounting consisted of three points; two being located near the center of gravity and the third point, a stabilizer, located near the pump inlet flange. Mounting brackets were removable so that changes could be made simply by redesigning the bracket, should this prove necessary. Mounting points were designated to prevent any interference problems with either the thrust chamber or the gas generator.

8. Maintenance

The basic design concept of having the blading, seals, mounts, and all major components removable was dictated not only by hydraulic design but also by maintenance. Should a failure occur, the damaged items could be replaced and the pump returned to operation with a minimum of expense.

A routine maintenance schedule has not been established. Such a schedule would be based upon bearing life because this appears to be the most critical item. Bearing test data plus data obtained from turbopump tests are necessary to determine maintenance needs and schedules.

9. Weight

Weight of all static components was of secondary concern for this initial pump design. Adequate strength and function were the primary concern.

All rotating components are as near flightweight as possible, consistent with conservative margins of safety. Test data and general pump operation would serve as guides for further weight reduction in subsequent designs.

Reduction of rotating component weight became a major design factor as critical speed analysis became more refined.

10. Critical Speed

The pump operating speed range is below the first critical speed. The first critical speed is a minimum of 15% above the maximum pump operating speed.

11. Materials

Materials were selected upon the basis of part function, structural requirements, and fabrication considerations. If possible, materials were those in common use, but this factor was a secondary consideration relative to the previously noted items. Materials are corrosion resistant and suitable for cryogenic applications, specifically, a liquid hydrogen environment.

12. Structural Safety Factors

Pump integrity was a primary consideration because of predicted hardware costs and the extent of the proposed test and development program. The following are considered minimum design safety factors.

a. Design yield = 1.0 times the limit load. The limit load is defined as the maximum predicted load that the system may experience under specified operating conditions.

b. Design ultimate = 1.5 times the limit load.

c. Proof pressure is 1.2 times the maximum static pressure predicted for the component.

d. Burst pressure is 1.6 times the maximum static pressure predicted for the component.

e. Tubes, flexible lines, and fittings having a diameter of less than 1.5-in. must withstand a proof pressure that is 2.0 times the maximum operating pressure and have a burst pressure equal to 4.0 times the maximum operating pressure.

The mechanical properties of the material at the operating temperature of the part is used in all stress analyses. Proof tests conducted at room temperatures have pressures adjusted to compensate for any change in material mechanical properties from operating temperature to room temperature.

The margin of safety is defined as follows:

$$M.S. = \frac{\text{allowable stress}}{\text{calculated stress}} - 1$$

In general, the calculated stress includes an applicable safety factor. The margin of safety may be based upon either the allowable yield stress or the allowable ultimate stress depending upon the specific part being analyzed, its function, and the analytical approach (including assumptions) used.

The fatigue margin of safety is applied to components having a fluctuating stress and is determined by use of a modified Goodman diagram. For these calculations, the alternating stress is assumed to be 30% of the maximum mean stress. The fatigue margin of safety is calculated as follows:

$$M.S. = \frac{\text{allowable stress from Goodman Diagram}}{1.25 \times \text{calculated stress}} - 1$$

This calculation includes a 1.25 safety factor.

C. DESIGN REQUIREMENTS

The hydrodynamic design specifications are presented in Tables 1 and 2.

The design objective for the pump was to produce the required pump discharge pressure of 1800 psia with a minimum number of stages and acceptable blade loadings. A parametric study was conducted with diameter and flow coefficients as the variables. The results of this analysis are discussed below.

The mainstage tip diffusion parameter, D, was defined as 0.4. This value was also used for the transition rotor and all stator blading.

The final design diameter selected was a compromise based upon blade aspect ratio, flow coefficient, and weight.

The final design selected consisted of a first-stage inducer having an eye diameter of 19.5-in. with three full vanes and three partial vanes.

It should be noted that total pump weight was considered in arriving at the final hydraulic design since the number of stages, hub diameters, and tip diameters determine over-all pump size and weight. The selected design was considered to be optimum for the end product, a flightweight turbopump having a total weight of approximately 3,300 lb. Figure 6 shows the pressure profile through the pump. Table 3 shows the flightweight pump component target weights and the actual weights of the Model I fuel turbopump assembly components.

TABLE 1

M-1 FUEL PUMP DESIGN SPECIFICATION
FOR LIQUID HYDROGEN

$\dot{W}$ (Inducer Stage) (lb/sec)	600
* $\dot{W}$ (Transition and Mainstages) (lb/sec)	640
P_d (psia)	1800
N (rpm)	13,225
NPSP (Minimum) (psi)	10
T (Inlet) Minimum (°F)	-423.5
T (Inlet) Nominal (°F)	-421.0
T (Inlet) Maximum (°F)	-418.0
Off-Design Range:	
Flow Coefficient (%)	± 13
Speed (%)	± 10

*Includes bearing and balance piston recirculating flow.

TABLE 2

M-1 FUEL PUMP DESIGN POINT

Propellant	LH_2
Propellant Inlet Temperature (°F)	-421
Propellant Inlet Density (lb/ft^3)	4.33
Shaft Speed (rpm)	13,225
Total Discharge Pressure (Cavitating) (psia)	1826
Total Suction Pressure (psia)	30.6
Total Pressure Rise (Cavitating) (psi)	1795
Total Head Rise (Cavitating) (ft)	56,500
Weight Flow (lb/sec)	600
Capacity (gpm)	62,300
Efficiency	81.2
Fluid Horsepower (Cavitating)	61,700
Shaft Horsepower (Cavitating)	76,000
Net Positive Suction Head (ft)	333
Suction Specific Speed	
$\frac{\text{rpm} \times \text{gpm}^{1/2}}{\text{ft}^{1/4}}$	43,000

TABLE 3

COMPONENT WEIGHT

COMPONENT	TARGET WEIGHT PER D.I.R. * LBS.	ACTUAL WEIGHT LBS.
10° Inlet Elbow	228	NA
Inducer Housing	252	NA
Discharge Housing	949	NA
Spinner	3	NA
Inducer Spacer & Retainer	4	NA
Inducer First Stage	47	NA
Inducer Second Stage	5	NA
Pump Rotor & Blades	220	377
Thrust Balance Disk & Locknut	15	27
Second Stage Inducer Stator	13	
Mainstage Stator Retaining Rings	180	296
Mainstage Stator Blades	101	NA
Turbine Inlet Manifold	169	NA
Turbine Exhaust Housing	140	NA
Bearing Housing Seal	15	NA
Reversing Row	20	NA
Inlet Elbow	54	NA
Turbine Rotor First Stage	180	208
Turbine Rotor Second Stage	142	NA
Tiebolt, Nut, & Lock Ring	12	12
Turbine Manifold Clamps	40	NA
Insulation	12	NA
Turbine Frame, Shims, & Bolts	80	NA
Pump End Bearing Package	130	NA
Turbine End Bearing Package	255	NA
TPA Mounting Brackets	100	NA
TOTAL FTPA ASSEMBLY	3300	6800

NA = Wt. not available. Individual parts to be weighed at FTPA disassembly after completion of test series.

*D.I.R. = M-1 Engine Design Information Report

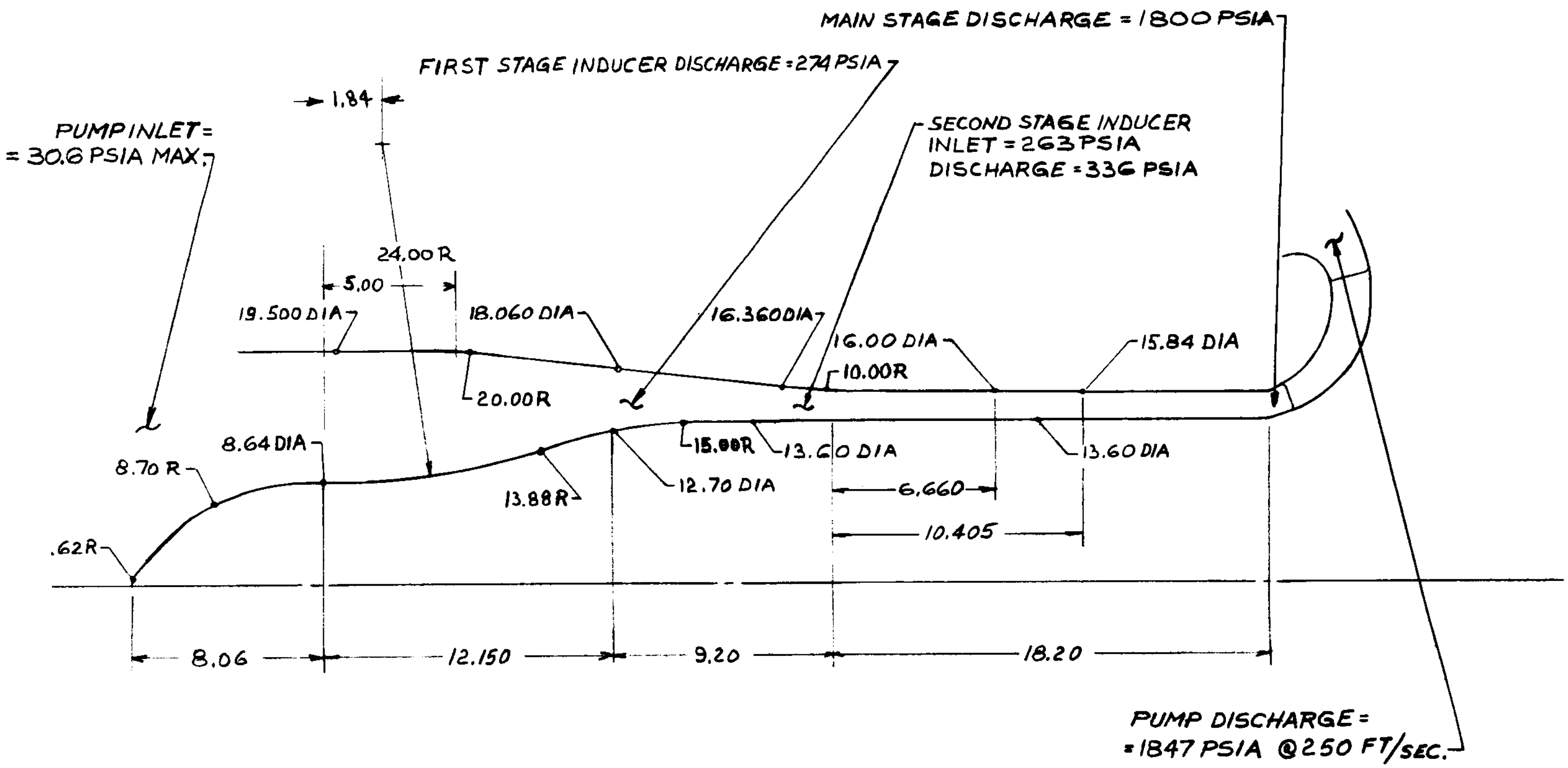


Figure 6. Hydraulic Passage Contour and Predicted Pressure Profile

From Table 1, it can be seen that the transition and mainstage are designed for a flow of 640 lb/sec instead of the required discharge flow of 600 lb/sec. The additional 40 lb/sec are for recirculating flow. This flow is required for pump-end bearing coolant, thrust balance system flow, and internal flow through the pump rotor which is related to thrust balancing.

Leakage of liquid hydrogen through all external joints was to be zero; therefore, no allowance was required for leakage in hydraulic design. The zero leakage requirement was mandatory because of the nature of the propellant and the safety hazard should leaks occur. This zero leakage requirement resulted in the selection of a redundant conical seal system for all external housing interfaces.

The final design selected also consisted of a first-stage inducer stator having eleven vanes with a C-4 airfoil. This stator serves two primary functions: it guides the hydraulic flow from the first-stage inducer to the second-stage inducer, and it is a structural support member for the pump-end radial and thrust bearings.

The remaining three components of the final design are delineated below.

The second-stage inducer and stator, or transition stage, is a lightly-loaded stage which compensates for the non-uniform head generated by the first-stage inducer. The light-loading conditions also ensure a broad operating range to minimize the effects of some mismatch that may occur with the first-stage inducer outlet condiitons.

The mainstage section consists of eight high pressure stages that are identical. These stages vary only in blade heights, which are reduced on the outside diameter only, to compensate for a 6.5% increase in fluid density.

The discharge diffuser has 20 vanes which diffuse the fluid from 460 ft/sec to 250 ft/sec. These diffuser vanes also function as a structural member. Consequently, the vane design is a compromise resulting in adequate structural performance at some sacrifice in ideal fluid angles.

D. DETAILED DESIGN

1. Major Assembly Dimensions

Significant fits, concentricities, dimensions, and materials are shown in Figures 1, 7, 8, 9, and 10 and Table 4.

2. Significant Features

The typical external flange seal consisting of a double conical seal system is shown in Figure 11. The design objective for this system was zero leakage.

TABLE 4

BLADING

Part Name	Number of Blades	Airfoil Type	Maximum Thickness (%)
Inducer First Stage	3 Full & 3 Partial	Cont. Vane	
Guide Vane Housing	11	C-4	10
Inducer Second Stage	31	C-4	10
Stator Inducer Second Stage	45	C-4	10
Rotor Mainstages	47	C-4	10
Stator Mainstages	57	C-4 Mod.	15
Discharge Housing	20	C-4	10
Turbine Nozzle Vane	37	Converging Diverging	
Turbine Rotor	80	Supersonic Converging	

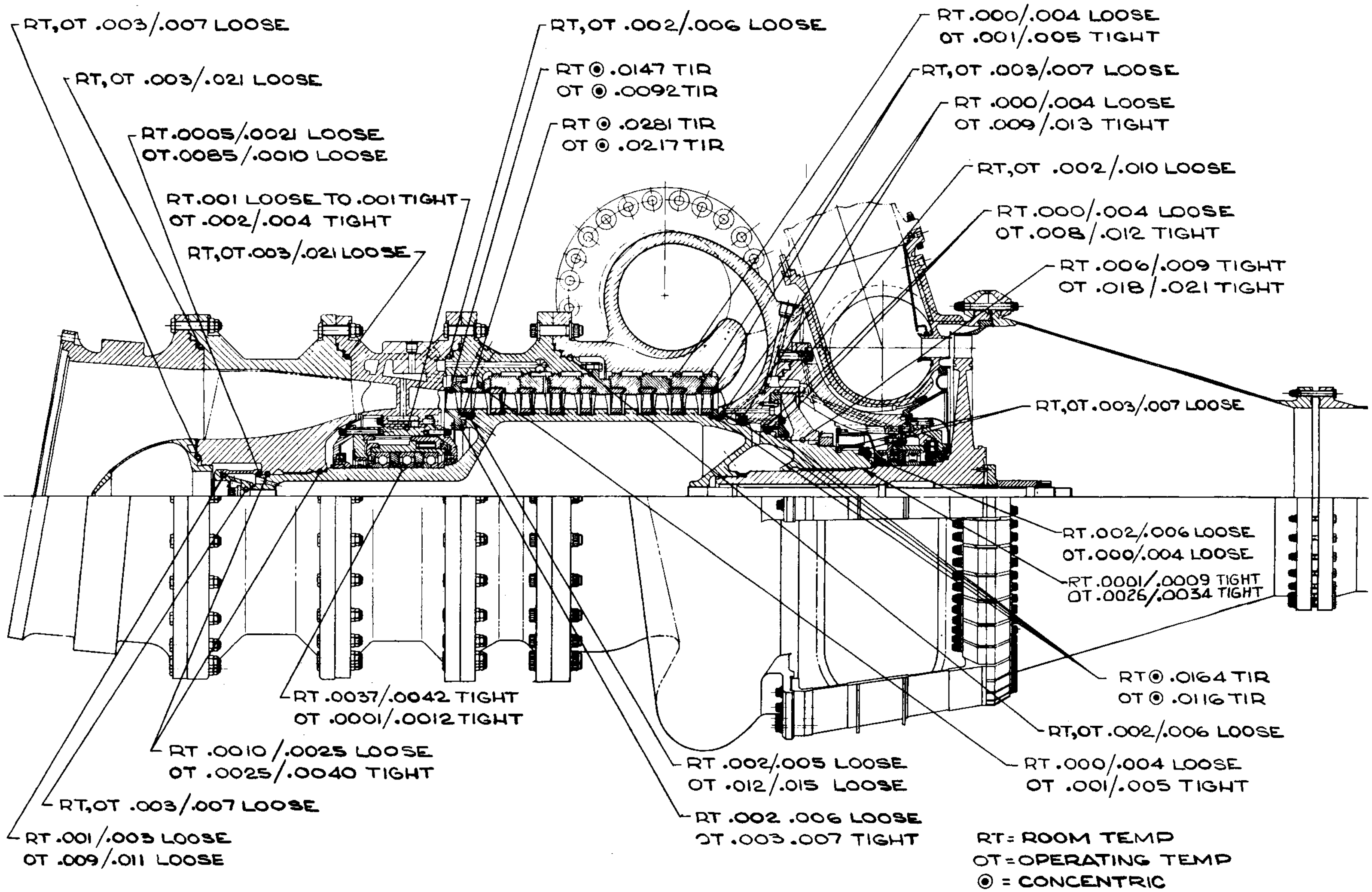


Figure 7. Fits and Concentricities

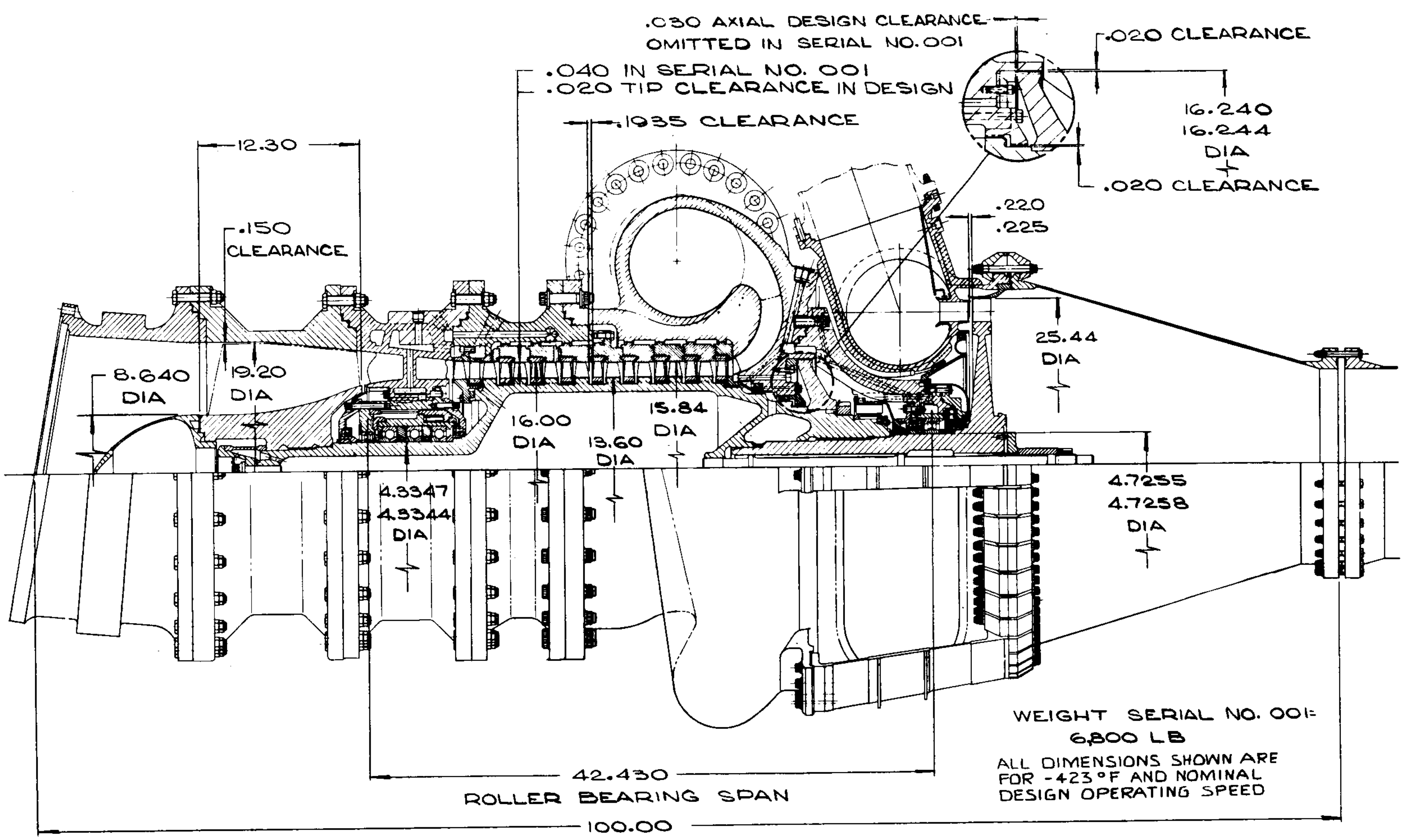
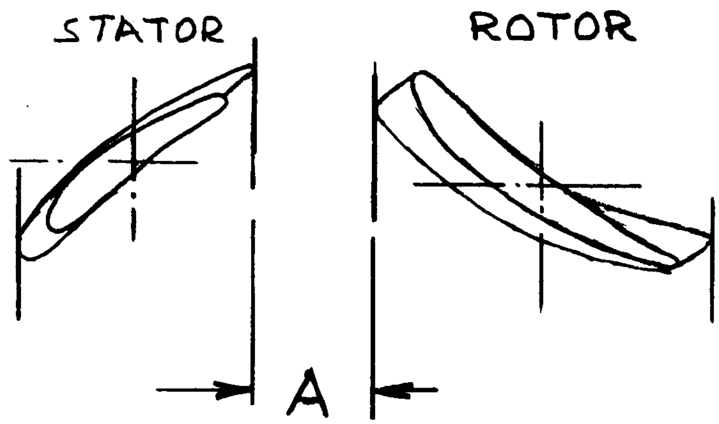
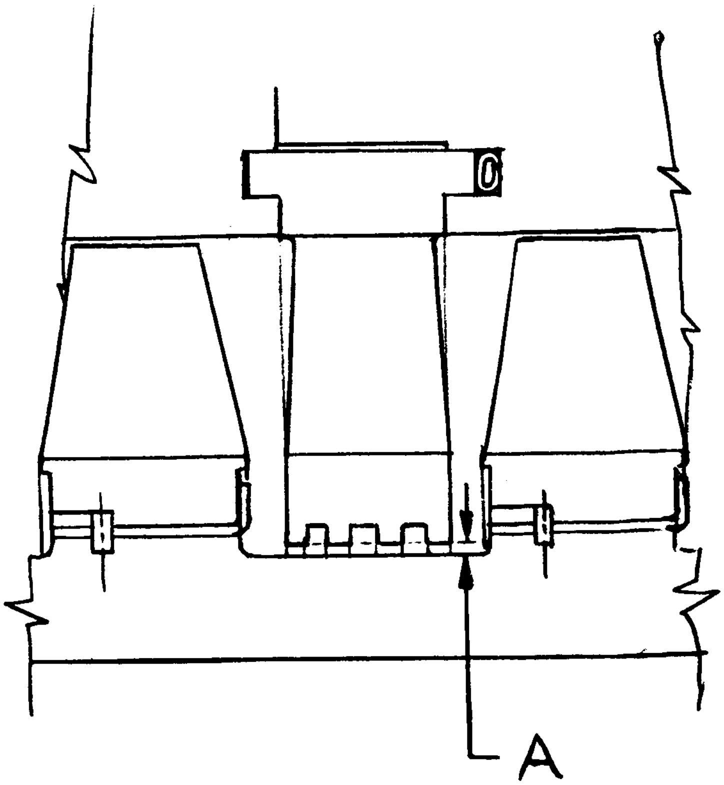


Figure 8. Significant Dimensions



DIM INCHES	DESIGN R.T. ASSEMBLY	ACTUAL R.T. ASSEMBLY	AT -423° F -20,000 LB THRUST
A	.180/.196	.192	.170

Figure 9. Rotor and Stator Blade Axial Clearance



DIM INCHES	ROOM TEMP. STATIC	-423°F STATIC	-423°F DESIGN SPEED
A	.067/.078	.067/.078	.044/.055

Figure 10. Rotor and Stator Radial Clearance

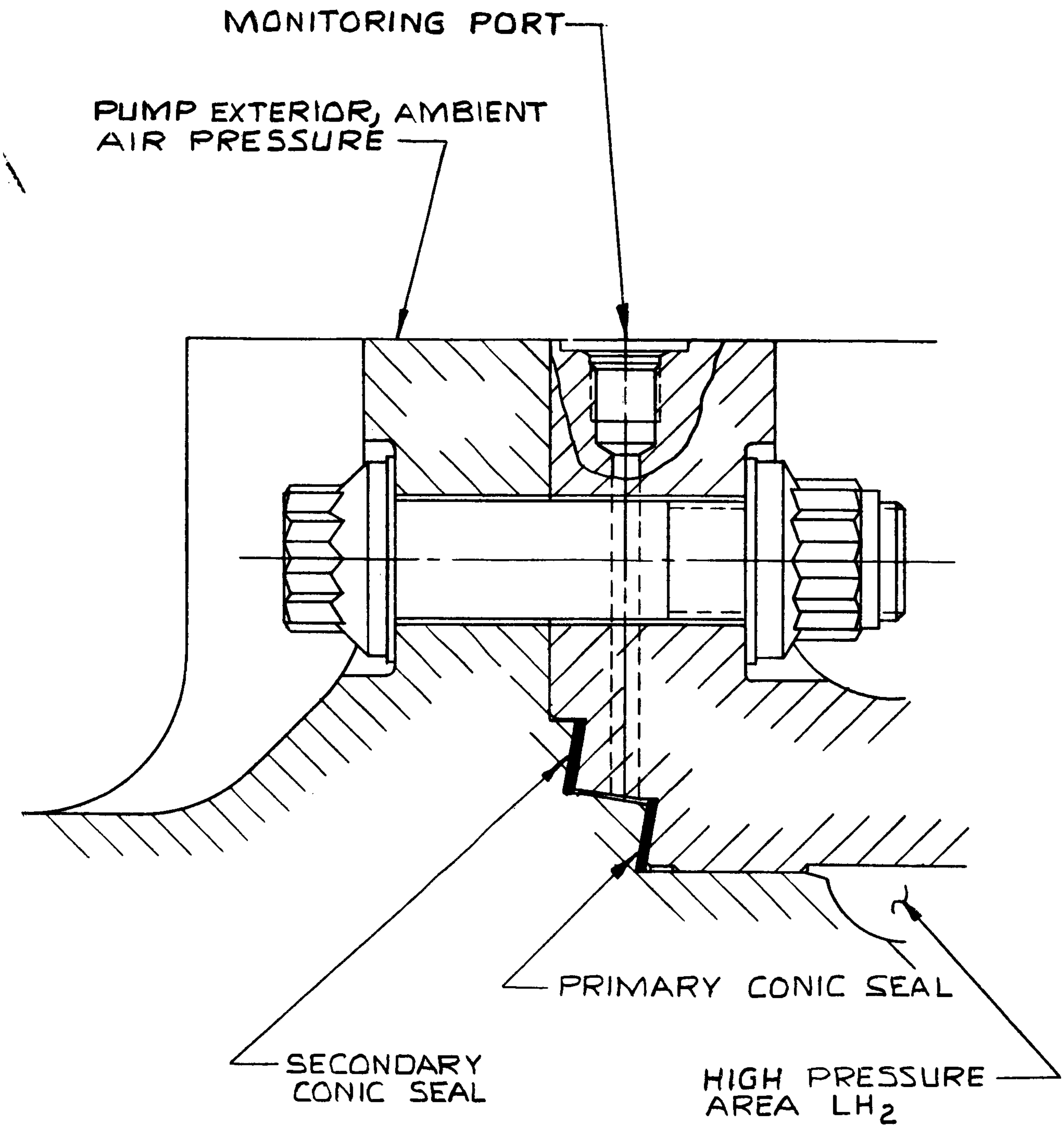


Figure 11. Flange and Seals

3. Bearing Load Summary

The fuel turbopump assembly Model I S/N 001 is equipped with the following bearings:

a. A pump-end roller bearing having a 110mm bore and 24 0.433-in. diameter x 0.433-in. long rollers. (See Table 5 for complete bearing specifications.)

b. A thrust bearing consisting of three ball bearings ground for load sharing in one direction. These bearings are of gothic arch design with a three-point contact. Three-point contact can occur when the bearing is transmitting a radial load. These bearings have split inner races, a 110mm bore, and 20 balls of 0.719-in. diameter. (See Table 6 for complete bearing specification.)

c. A turbine-end roller bearing having a 120mm bore and 26 rollers of 0.526-in. diameter x 0.654-in. long. (See Table 7 for complete bearing specification.)

The roller bearings are supported in rigid housings while the thrust bearings are supported in a radially flexible housing. This housing is designed to limit the radial load carried by the thrust bearings.

The calculated critical speed for the Model I fuel turbopump assembly is 16,000 rpm (see Figure 12). A critical speed analysis was prepared under separate cover.[4]

Calculated radial bearing loads for the Model II two-stage turbine fuel turbopump are presented in Tables 8 and 9. The calculations considered the effect of bearing looseness, dynamic unbalance, hydraulic unbalance, lateral acceleration, longitudinal acceleration, and gimbal snubbing.

Thrust bearing loads were established at 35,000 lb maximum in the load sharing direction. The limit for a thrust reversal is 15,000 lb. Maximum radial load is limited to 200 lb by the bearing housing design. The thrust balancing system was designed upon the basis of these bearing limits. Predicted thrust bearing loads at the pump design point are: nominal 20,000 lb, minimum 5,000 lb, and maximum 35,000 lb.

E. DYNAMIC BALANCING

1. Description

Dynamic balancing of the rotating components is accomplished in a series of operations starting with the balancing of individual components and then building to the final balancing of a rotating assembly. This method was used

(4) Report No. NASA CR-54825, entitled Analysis of the Liquid Hydrogen Turbopump Shaft Critical Whirling Speed and Bearing Loads

TABLE 5

ROLLER BEARING - PUMP END - 110 mm

(P/N 288260)

Inside Diameter	110 mm
Outside Diameter	150 mm
Width	20 mm
Roller Diameter	0.433-in.
Roller Number	24
Roller Length	0.433-in.
Race and Roller Material	440C
Cage Material	Armalon
Vendor	Industrial Tectonics
Diametral Play (As Built)	0.003/0.0035

TABLE 6

BALL BEARING - 110 mm THRUST BEARING

(P/N 288410)

Inside Diameter	110 mm
Outside Diameter	170 mm
Width	28 mm
Ball Diameter	23/32
Ball Number	20
Race and Ball Material	440C
Cage	Armalon
Vendor	Industrial Tectonics
Contact Angle	30°
Race Curvature:	
Inner	53%
Outer	52%
Diametral Play (As Built)	0.0068/0.0074
Axial Play (As Built)	0.014/0.020
Dynamic Contact Angle:	
Inner	35.8°
Outer	32.2°
Maximum Hertz Stress psi at -420°F	291,600

TABLE 7

ROLLER BEARING - TURBINE END - 120mm

(P/N 288340)

Inside Diameter	120 mm
Outside Diameter	180 mm
Width	28 mm
Roller Diameter	0.526-in.
Roller Number	26
Roller Length	0.645-in.
Spring Rate in.-lb	8×10^6
Race and Roller Material	440C
Cage	Armalon
Vendor	Bower
Diametral Play (As Built)	0.0031/0.0035

TABLE 8

RADIAL BEARING LOADS

Cause of Load	Pump Bearing Load (lb)	Turbine Bearing Load (lb)
Longitudinal Acceleration, 10Gs.	510	1050
Lateral Acceleration, 1G.	290	590
Gimbal Snubbing	1030	2120
TOTAL	1830	3760

TABLE 9

RADIAL BEARING LOADS CAUSED BY ROTATION

Cause of Load	Pump Bearing Load (lb) 13,255 (rpm)	Pump Bearing Load (lb) 14,548 (rpm)	Turbine Bearing Load (lb) 13,225 (rpm)	Turbine Bearing Load (lb) 14,548 (rpm)
Dynamic Unbalance	940	1030	940	1030
Static Radial Eccentricity	4980	8180	10,780	20,030
TOTAL	5920	9210	11,720	21,060
Acceleration From Table 7	1830	1830	3760	3760
Total Bearing Load	7750	11,040	15,480	24,820

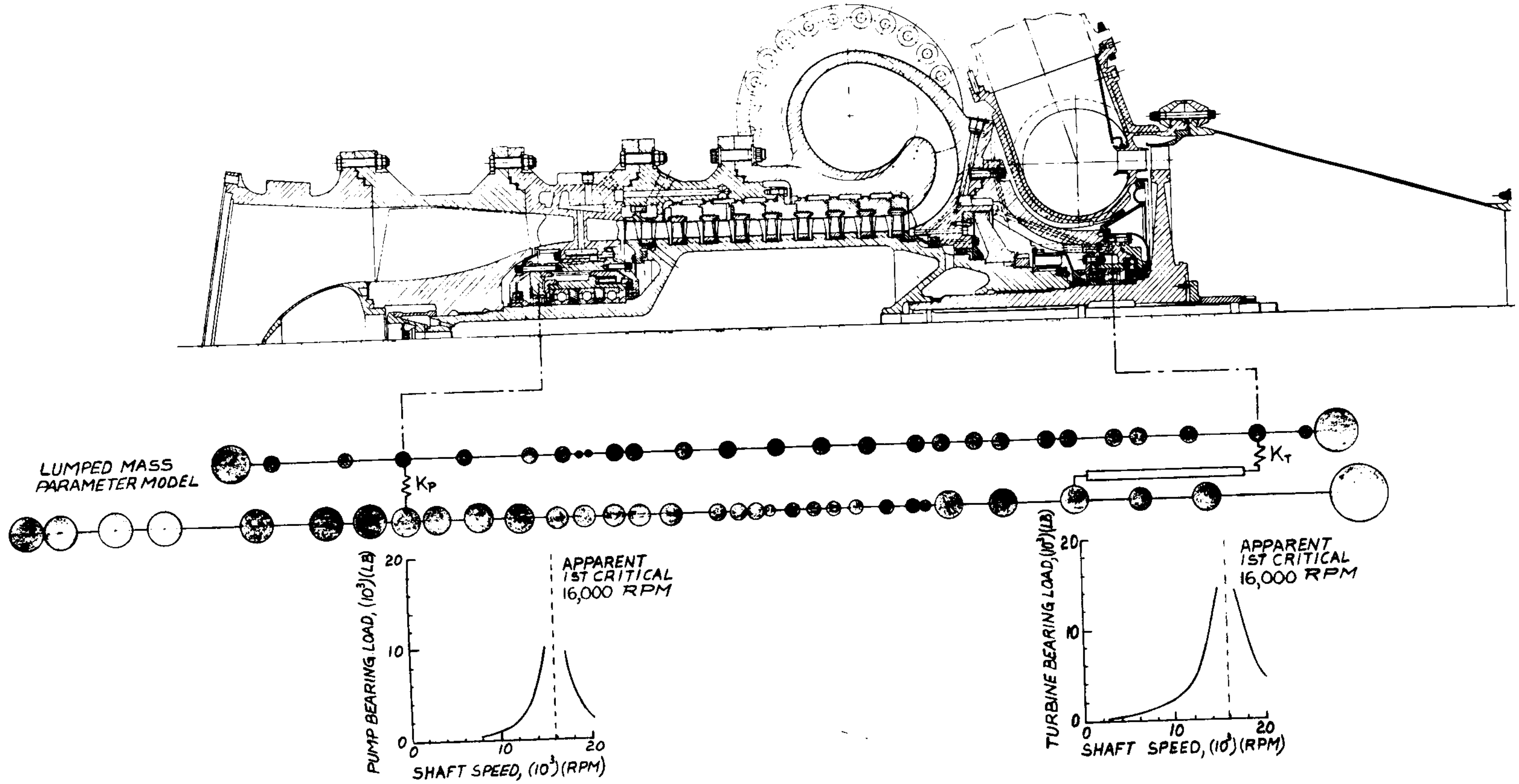


Figure 12. Rotor Assembly Critical Speed Diagram

with those parts that had virtually line-to-line or interference fits, specifically the pump rotor, thrust balancing disc, and turbine rotor. The first- and second-stage inducers were balanced separately because of their loose room temperature fit with the pump rotor.

All balancing was performed to the minimum limits obtainable with the specific balancing machine used. These limits were less than the engineering requirements for each component. The required limits are shown in Table 10.

2. First-Stage Inducer

The first-stage inducer was dynamically balanced in a Type-S Gisholt machine. Power for rotation of the part was supplied by a belt drive.

The inducer was installed on a balanced arbor prior to balancing. There was an interference fit between the inducer and the arbor to assure positive positioning and rotation of the part being balanced. The interference fit was achieved by heating the inducer to 200°F, then positioning it on the balancing arbor.

Normal balancing practices for a two-plane dynamic system were used. The amount of residual unbalance was minimized by rotating the inducer 180 degrees on the balancing arbor and then making a final balancing check.

After balancing, the first-stage inducer was stored until final assembly of the turbopump.

3. Second-Stage Inducer

The second-stage inducer was balanced in the belt drive Type-S Gisholt machine. The part was mounted on a balanced arbor and work proceeded in accordance with standard practice for single-plane static balancing. This type of balance was considered adequate because of the light weight of the part, its location relative to the pump bearings, and the slightly loose room temperature fit of the piloting diameter at assembly.

To keep the unbalance of the complete rotating system to a minimum, the 12 bolts used to mount the second-stage inducer were matched by weight. Material was removed from the threaded ends of the bolts to achieve identical weights. These bolts were identified and stored as matched sets.

4. Pump Rotor and Rotor Subassemblies

All parts discussed in this section were balanced on a Type-U Gisholt machine. The balancing technique was accomplished in accordance with standard practice consistent with the size of the part, the balance limits required, and a two-plane system.

TABLE 10

BALANCING TOLERANCES

Component	Dynamic Balance (In. Ounce) Plane I	Dynamic Balance (In. Ounce) Plane II	Static (ounce) Plane I
First-Stage Inducer	0.1	0.15	---
Second-Stage Inducer	---	---	0.05
Pump Rotor Assembly	0.2	0.3	---
Pump Rotor Assembly With Balance Disc	---	0.5	---
Turbine Rotor	0.3	---	---
Pump Rotor Assembly With Thrust Disc and Turbine Rotor	0.35	0.75	---

The pump rotor, including blades, was balanced as an individual item (see Figure 13). A balance mandrel was inserted in the turbine end of the pump rotor so that balance machine support points could duplicate the pump roller bearing span. No cover was provided for the blades to reduce any effects of the air being set in motion because of rotor rotation. The balancing speed was such that any effects of the air were considered negligible. The pump rotor was balanced easily and required the removal of only very small amounts of metal. The areas provided for material removal during balancing were more than adequate.

With the pump rotor assembly balanced and the mandrel still in place, the thrust balance disc and its retaining lock ring and nut were assembled to the rotor (see Figure 14). This unit was then balanced with corrections being made in one plane only. This plane was centered on the thrust balance disc. All balance corrections were made by removing material from the thrust balance disc. After completing this operation, the balance mandrel was removed. The pump rotor thrust disc assembly was kept intact awaiting the installation of the balanced turbine rotor.

The turbine rotor was balanced as a single component (see Figure 15). The turbine balancing used a single plane, which was centered on the turbine disc. Some difficulty was experienced in obtaining a suitable drive coupling between the balance machine and the turbine shaft. The problem was looseness of the adapter in its fit to the turbine. This problem was resolved by using an aluminum coupling that had an interference fit in the turbine shaft bore. The assembly of the aluminum coupling to the turbine shaft was accomplished by temperature differential. The balance tooling included installation of a slave bearing on the turbine shaft. This bearing duplicated the pump turbine-end bearing as to size and position on the shaft thus simplifying the support of the turbine in the balance machine. The slave bearing also protected critical surfaces on the shaft.

The final balancing operation began with the assembly of the balanced turbine rotor, with its tie bolt and retainers, to the balanced pump rotor thrust disc assembly (see Figure 16). This complete assembly was placed in the balance machine and checked for its degree of unbalance. The turbine was removed, rotated 90 degrees relative to the original position and reassembled to the pump rotor. The degree of unbalance was measured again. This turbine rotation and measure of unbalance was repeated for turbine positions of 180 degrees and 270 degrees. The data was reviewed and the turbine installed in the condition that had the least unbalance. The final two-plane dynamic balancing was then completed. Primary metal removal areas were on the thrust balance disc and its retainers. After completion of the balancing, all components of the assembly were match marked and identified with the same serial number. The unit was then disassembled as necessary for final assembly of the turbopump assembly.

F. THRUST BALANCE SYSTEM SUMMARY

Simply stated, the thrust balancing system utilizes a liquid hydrogen flow taken at the maximum pump pressure point, immediately after the eighth main-stage, to create a force opposing that developed within the pump. This opposing

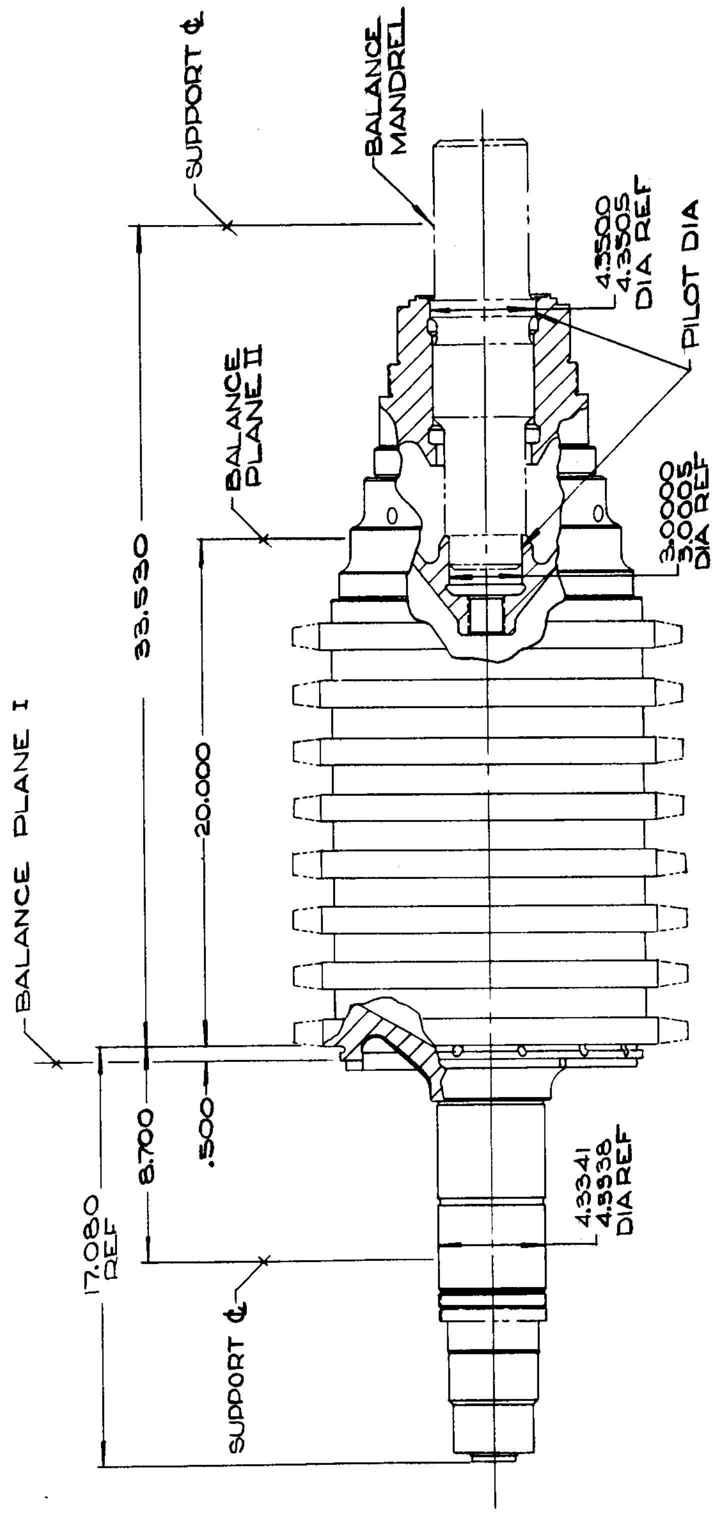


Figure 13. Pump Rotor Balance

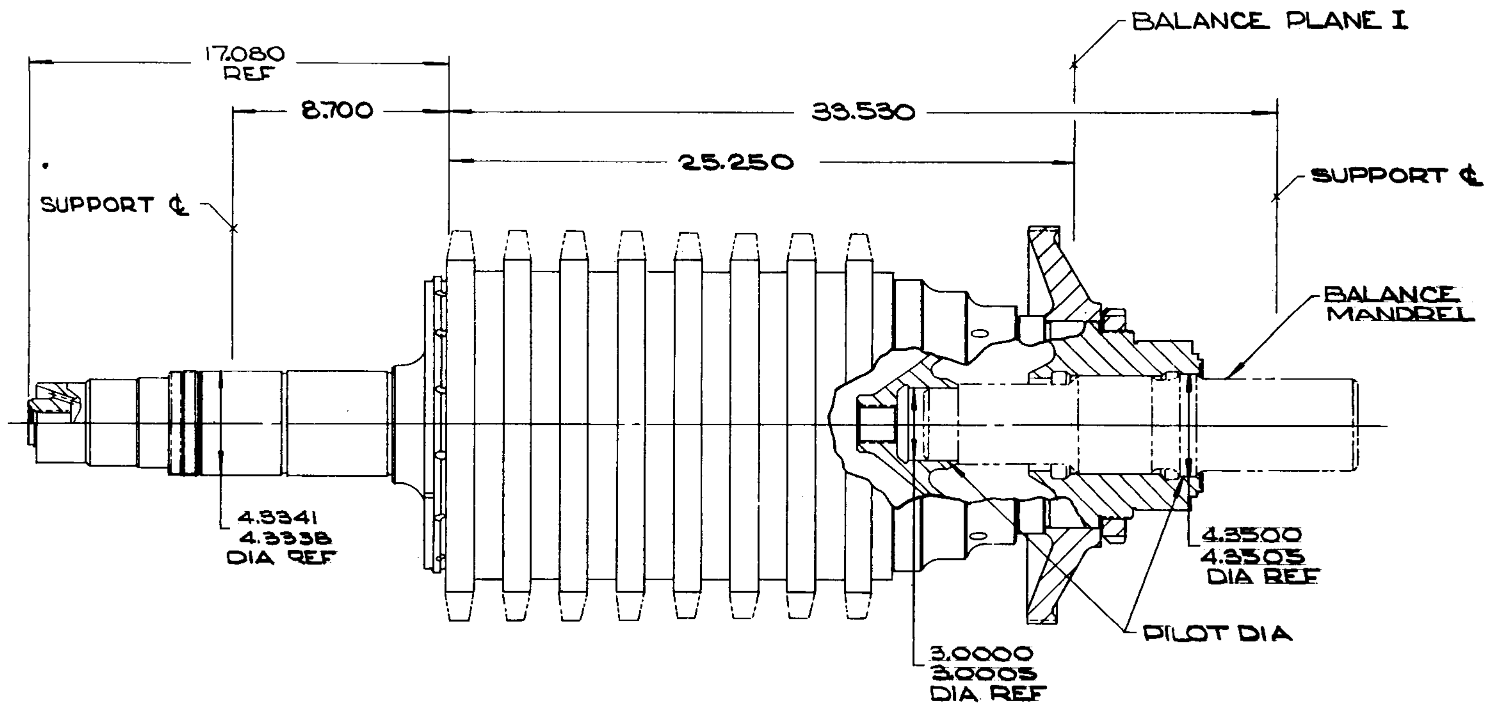


Figure 14. Rotor Subassembly Balance

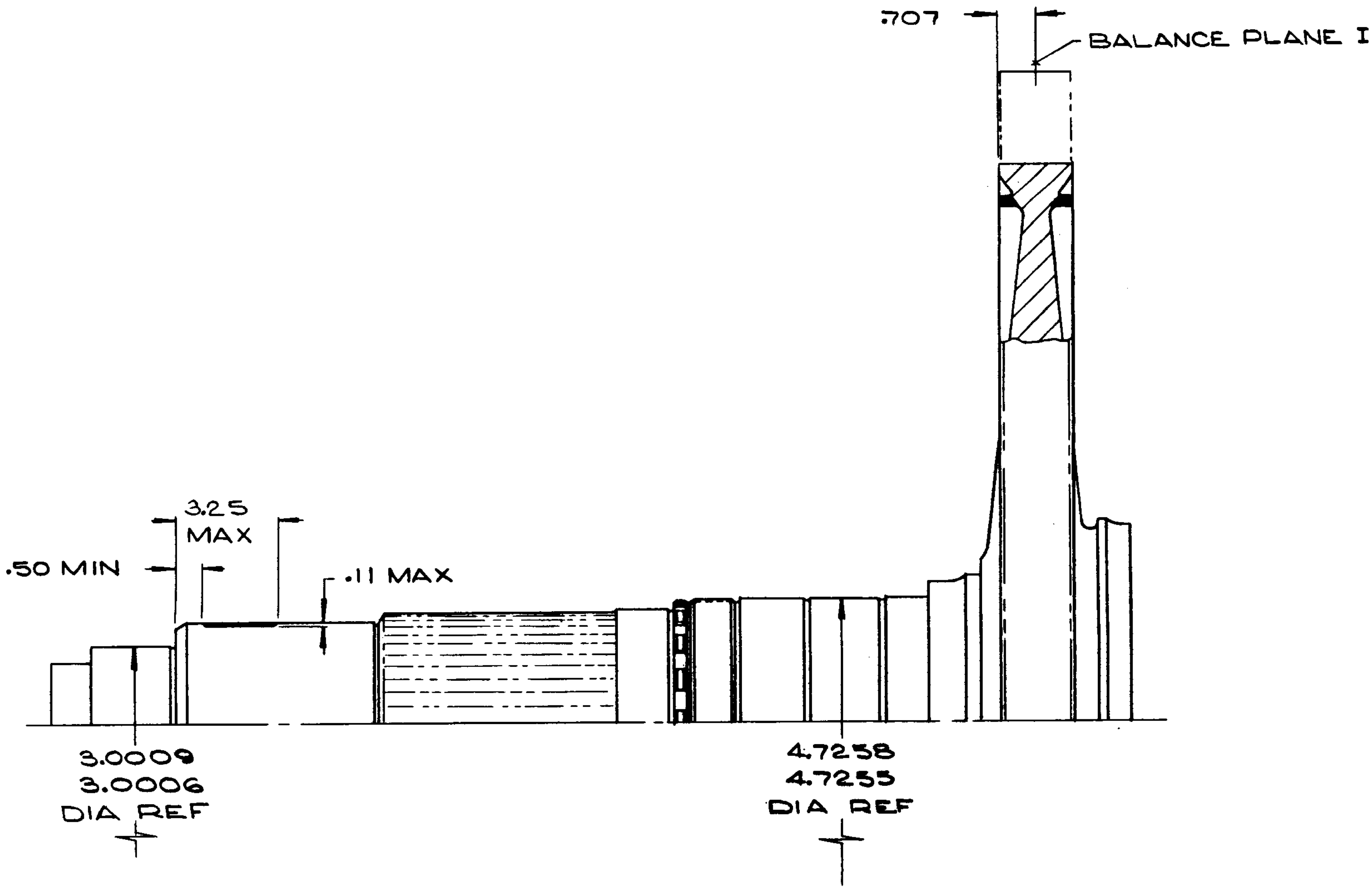


Figure 15. Turbine Rotor Balancing

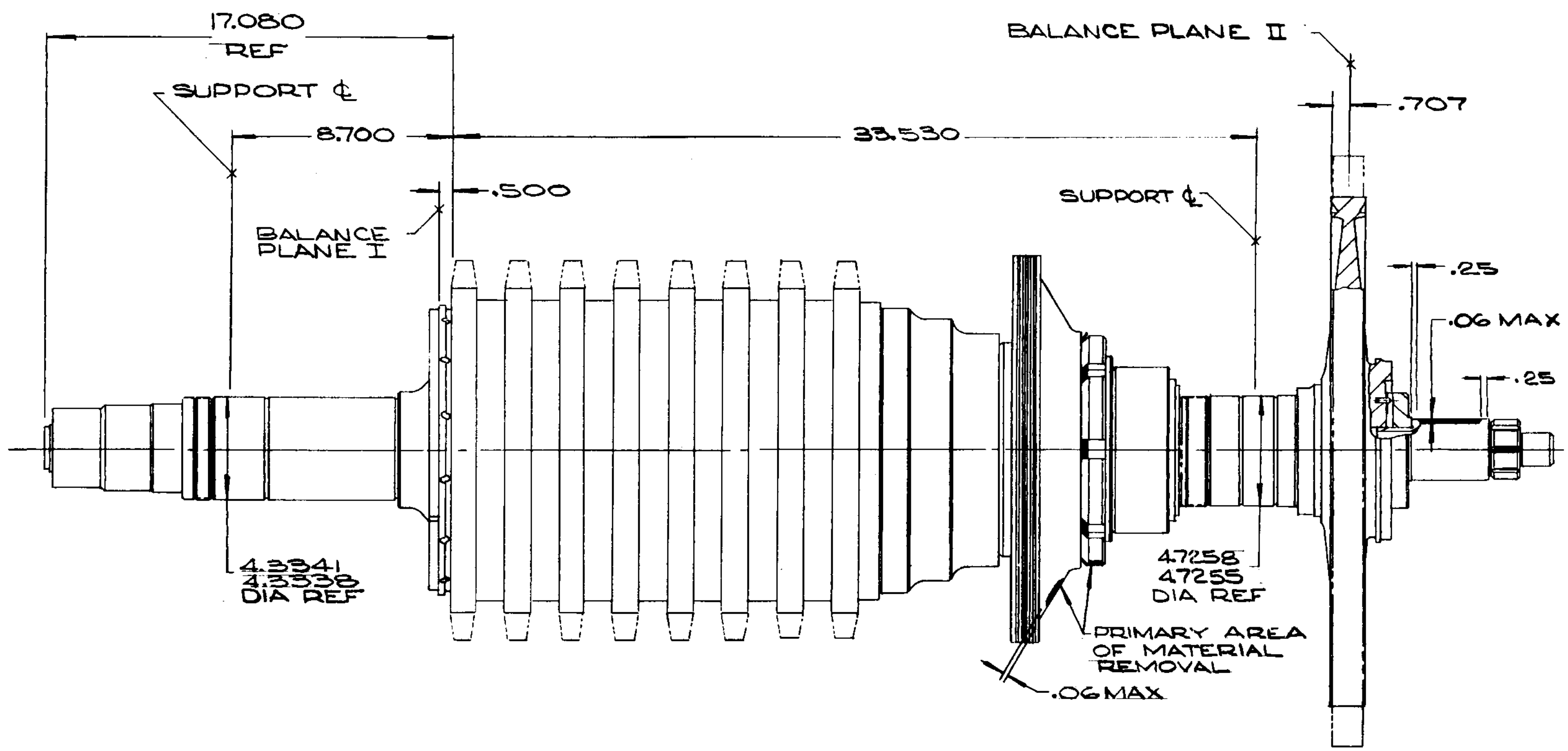


Figure 16. Rotor Assembly Balance

force is generated by creating large pressure drops across the outer edge of a disc. The analysis of the thrust balancing system is presented under separate cover.[5]

The high pressure flow is routed through passages in the discharge housing to the pump side face of the thrust balancing disc (see Figures 4 and 17). The flow passes to the outer edge of the disc, where it enters two pressure reducing areas.

The first pressure drop occurs by means of a variable axial gap. The gap width is a function of the pump rotor displacement caused by pump thrust acting upon the spring-mounted thrust bearings. As the axial gap varies, the pressure drop through the gap varies and the net thrust balancing load created on the thrust disc varies. This variable force serves to make the system self-centering. As the axial gap increases, the thrust disc load decreases and the rotor system moves to reduce the gap and thereby, establish equilibrium. As the axial gap decreases, the thrust disc load increases and the rotor moves to establish equilibrium.

The second pressure drop occurs along the periphery of the thrust disc by means of a constant radial gap. This device creates the major thrust balancing force, which is a constant for any given set of conditions.

A series of four labyrinths were provided between the eighth mainstage and the high pressure side of the thrust disc. These labyrinths create pressure drops on the pump rotor surface thereby reducing the thrust force on this surface, which acts towards the pump suction. This reduces the amount of thrust compensation required from the thrust balancing disc. Leakage through these labyrinths is routed through the pump rotor interior to re-enter the main flow stream at the second-stage inducer stator.

The thrust balancing system is intended to be a self-compensating or self-centering system.

The thrust balancing flow, after passing through the two pressure reducing areas, enters the low pressure chamber on the turbine side of the thrust balancing disc. The fluid then moves through a series of radial slots into an annular collecting area. This system was designed to maintain pressure variations across the low pressure side of the thrust disc at a minimum.

Finally, the flow is routed into a series of external tubes, through instrumentation manifolds, and into the mainstage housing. Within the mainstage housing, the balance system flow is returned to the main flow stream at the discharge side of the second-stage inducer stator.

The fuel turbopump assembly Model I S/N 001 was assembled without incorporating the axial gap. The amount of thrust balancing force created by this

(5) Report No. NASA CR 54824, entitled Analytical and Experimental Studies of End Thrust Control in The M-1 Liquid Hydrogen Turbopump

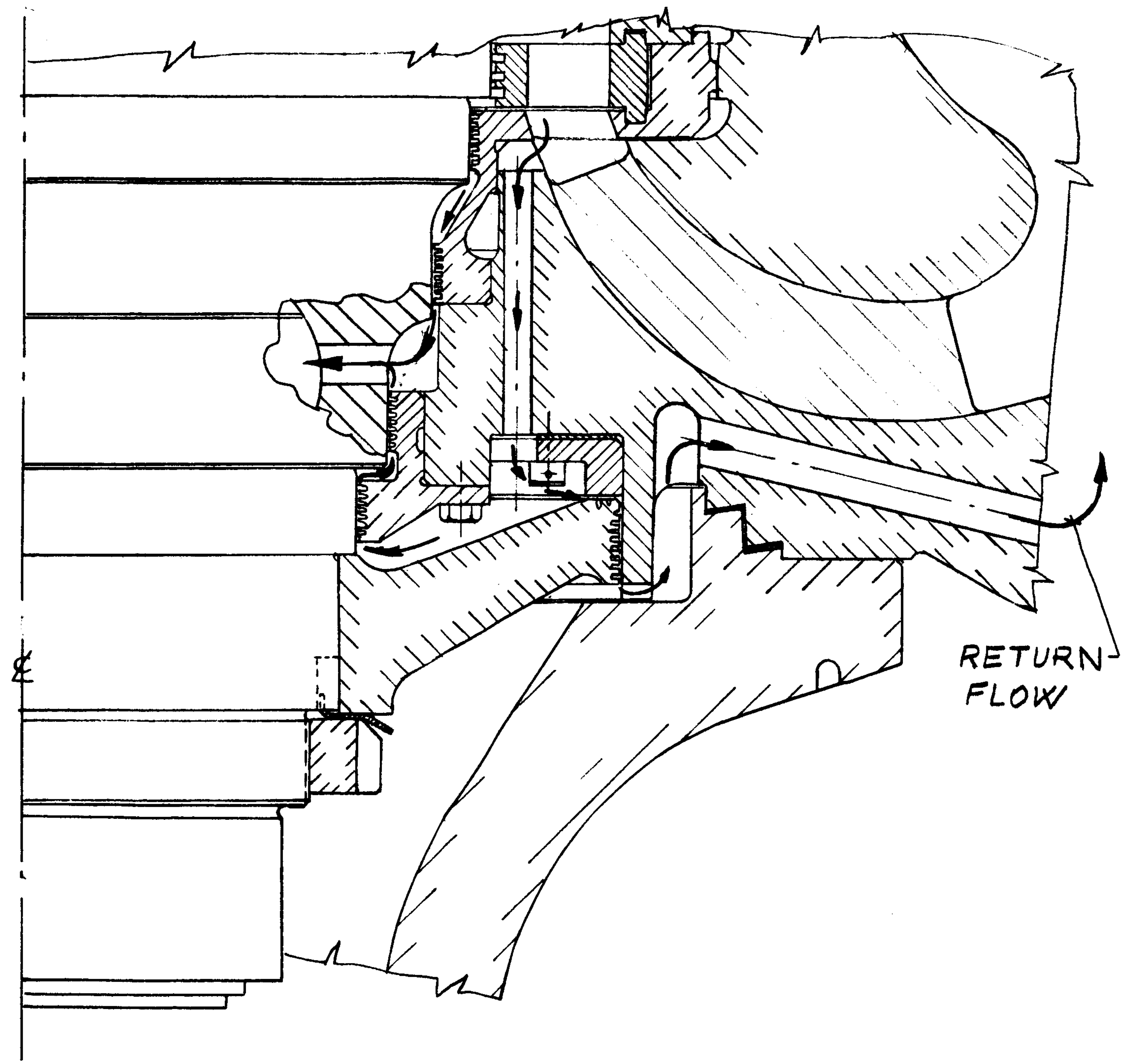


Figure 17. Thrust Balance Piston Arrangement

gap is small, and its omission would not jeopardize thrust balancing capabilities. The omission of this gap would remove one serious point of possible contact during operation between rotating and stationary components. The axial gap feature is to be installed in subsequent assemblies when more is known about the actual performance of the thrust balancing system and related pump rotor displacements.

IV. COMPONENT DESIGN

A. ROTATING ELEMENTS (See Figures 18, 19, and 20)

1. Pump Rotor

a. Design Philosophy

The pump rotor had to be a lightweight rigid component capable of transmitting the required power and being compatible with the environment in which it must operate.

Maximum rigidity coupled with minimum weight could be solved through the use of a welded unit. Welding also eliminated all leak paths that can occur when using bolted joints. This made feasible using the interior of the rotor as a return flow system.

In general, this component was to be as simple in design as its function would permit.

b. Design Requirements

Operating speed 13,225 rpm $\pm$ 10%

Operating temperature - 423°F

Power input 90,000 HP

All blading to be removable

Minimum weight consistent with a suitable margin of safety.

c. Description

The pump rotor is an integral unit fabricated from four forged and machined components TIG welded together. The rotor is the principal rotating component of the fuel turbopump assembly. The first-stage inducer is mounted on the shaft end of the rotor and driven by means of a spline. The shaft end also carries the pump-end roller and thrust bearings.

The transition from pump-end shaft to the mainstage blading diameter is conical. This provided maximum stiffness radially and minimum deflection axially, thereby increasing over-all rotor rigidity. The rigidity is necessary to produce a critical speed that is acceptable rleative to pump operation.

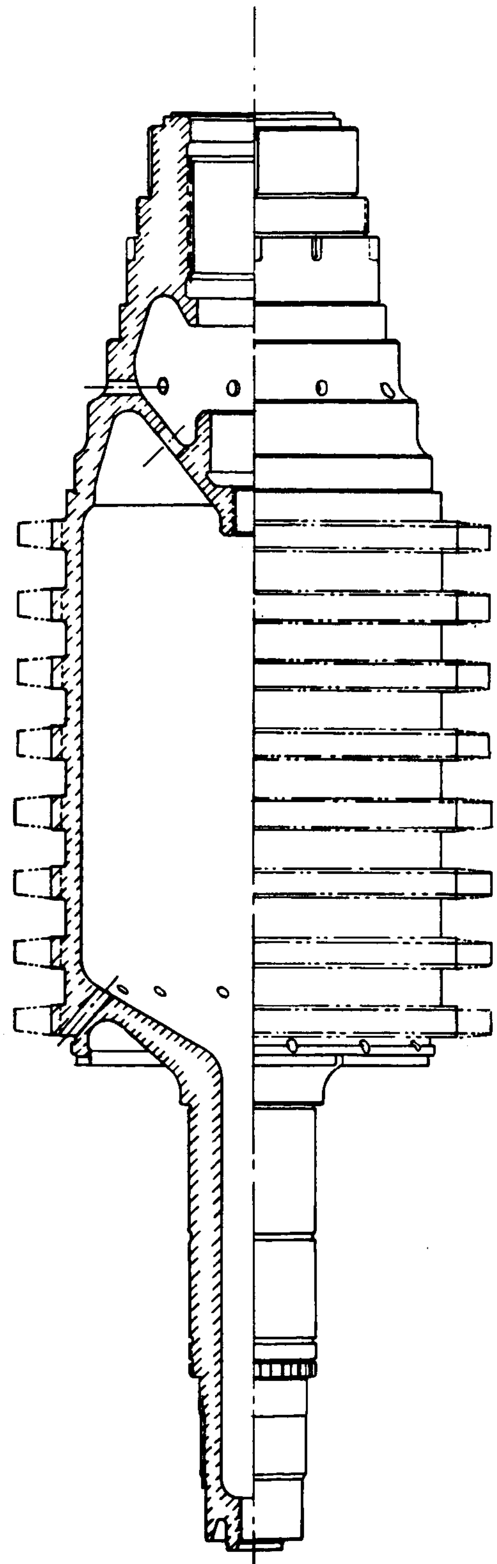


Figure 18. Pump Rotor

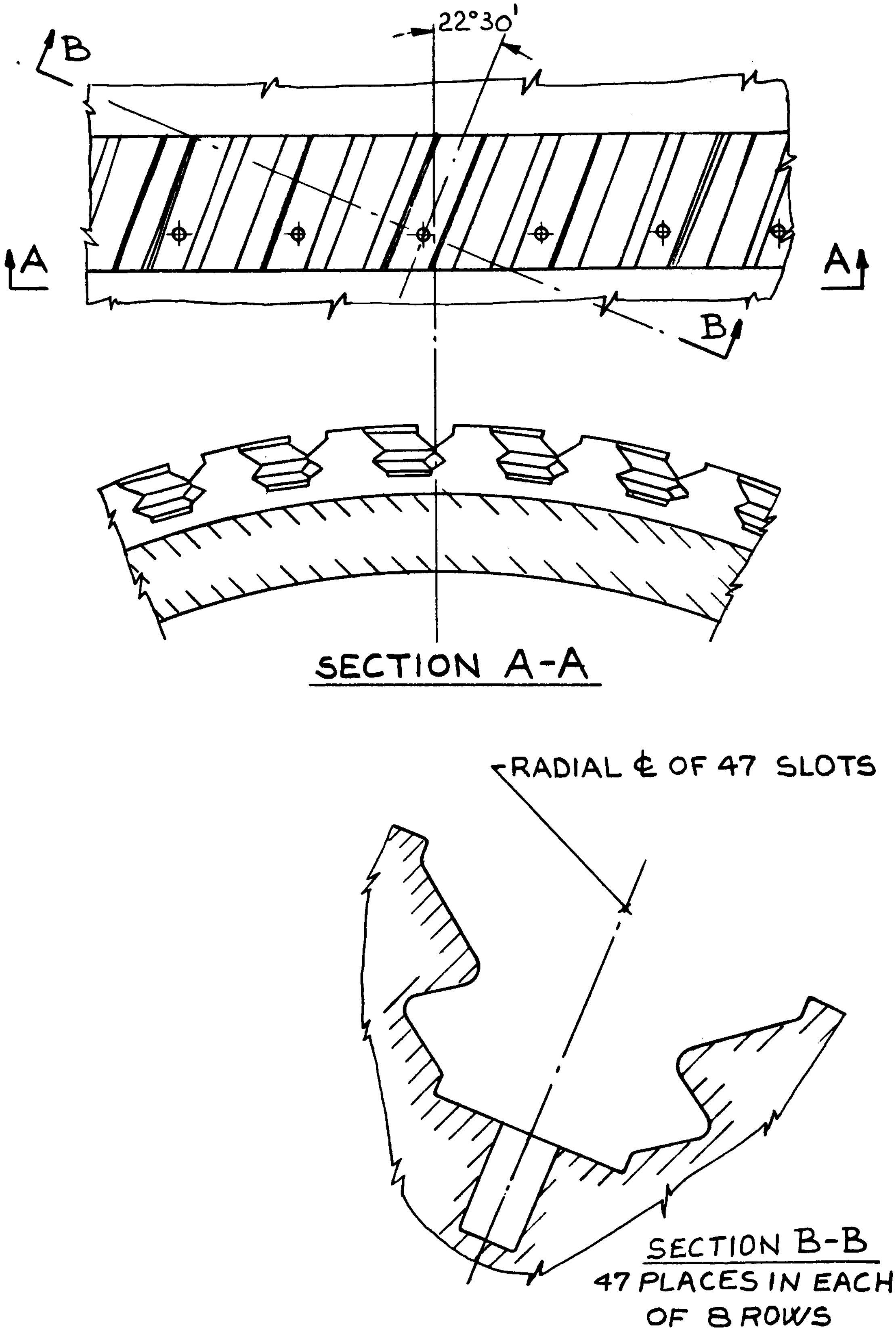


Figure 19. Dove-Tail Configuration

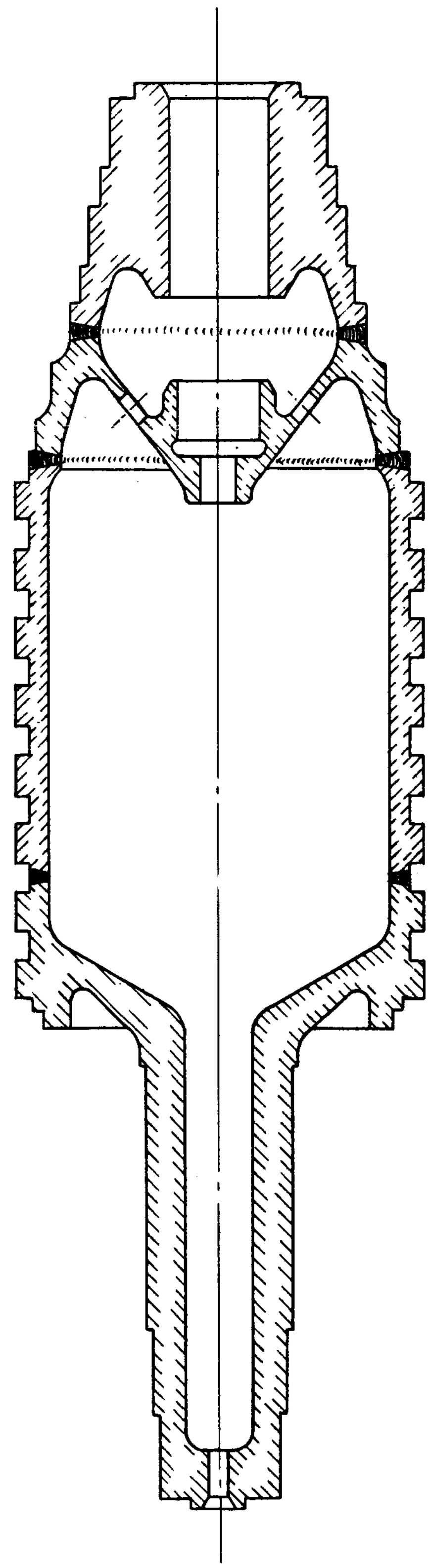


Figure 20. Rotor Weldments

Adjacent to the conical transition area, pilot diameters and driving slots are provided for mounting the second-stage inducer.

Eight rows of 47 dove-tail slots are machined on the large cylindrical surface. These slots accept the mainstage rotor blades.

Immediately past the last row of slots, the rotor becomes conical, thereby reducing its diameter to a size compatible with the internal spline and pilot diameters which accept the turbine rotor shaft. This conical reduction is accomplished in a series of steps which provide cylindrical surfaces for labyrinths, mounting the thrust disc, and mounting the torquemeter sleeve.

The interior of the rotor is generally free of all irregularities. A conical section is provided to accept the turbine tie-bolt. The cone shape was selected because it is capable of handling the high tie-bolt loads with a minimum of axial deflection.

Holes through the rotor exterior walls are provided for the return flow of liquid hydrogen from between the discharge housing labyrinths.

d. Pump Rotor Stress

Two stress analyses are applicable to the rotor installed in fuel turbopump assembly Model I S/N 001.

The first analysis concerns the pump rotor as designed and assumes sound welds. This analysis method utilized a mathematical model of the pump rotor and was based upon thin shell theory. The results of this analysis show that maximum stress values occur in the conical transition from the pump shaft to the cylindrical section, which carries the mainstage blading. The calculated maximum values are:

Hoop 120,000 psi

Tension in Meriodional Plane 40,000 psi

Maximum Radial Deflection .023-in.

The minimum margin of safety based upon room temperature material properties is + 0.71 (see Figures 21, 22, 23, 24, 25, and 26).

The second analysis contains a study of the effect of weld flaws or cracks upon rotor integrity. This analysis resulted when flaws were detected in what had been previously considered excellent welds. The finite element procedure was used in this analysis for determining stress values for the weld areas. The maximum stress in the area of the eighth mainstage blade row was found to be 55,000 psi. The analysis concluded that the flaws present in the weld should not cause fast unstable cracking, and fatigue of the weld was unlikely.

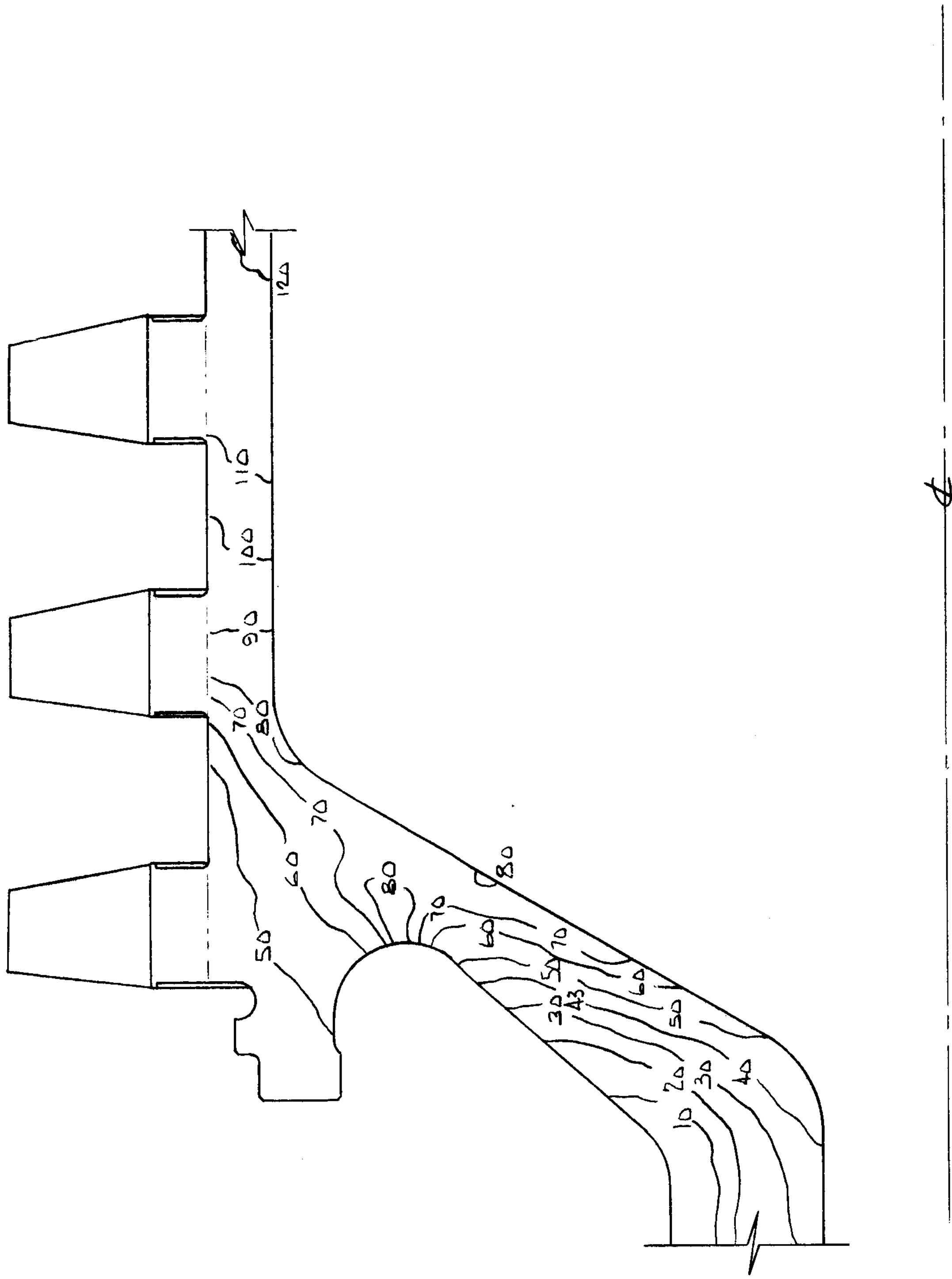


Figure 21. Hoop Stress Profile (10^3 psi)

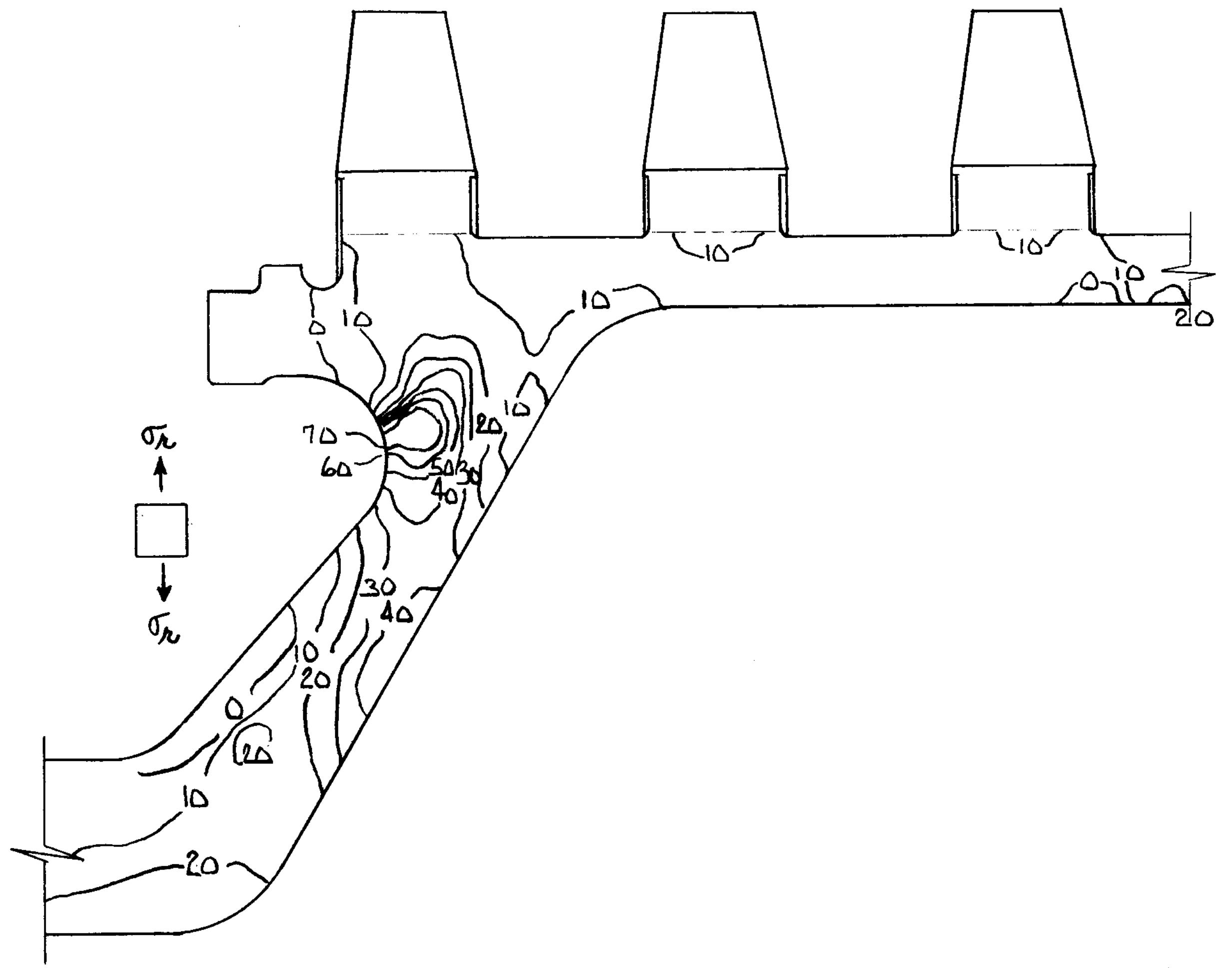


Figure 22. Radial Stress Profile (10^3 psi)

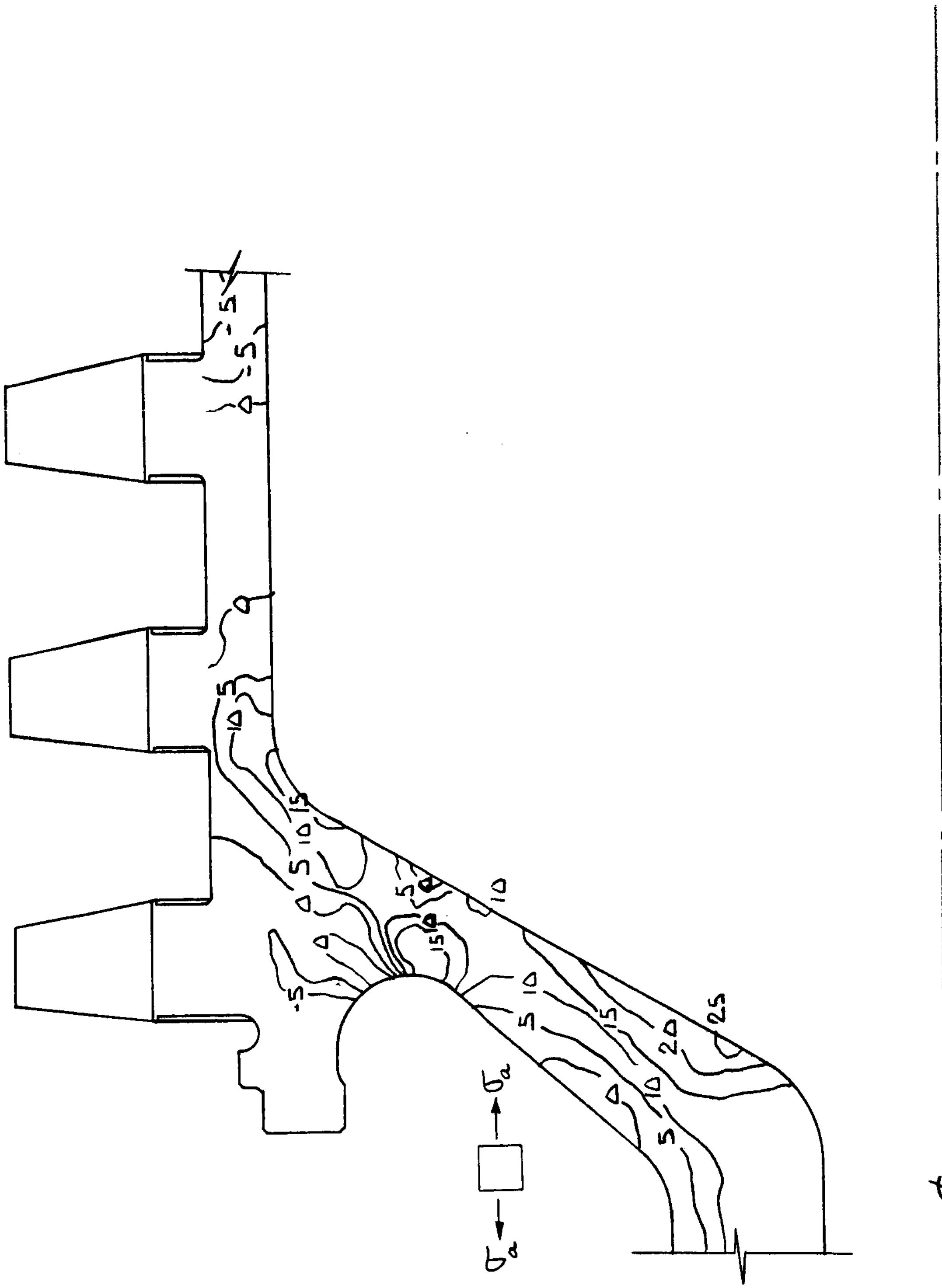


Figure 23. Axial Stress Profile (10^3 psi)

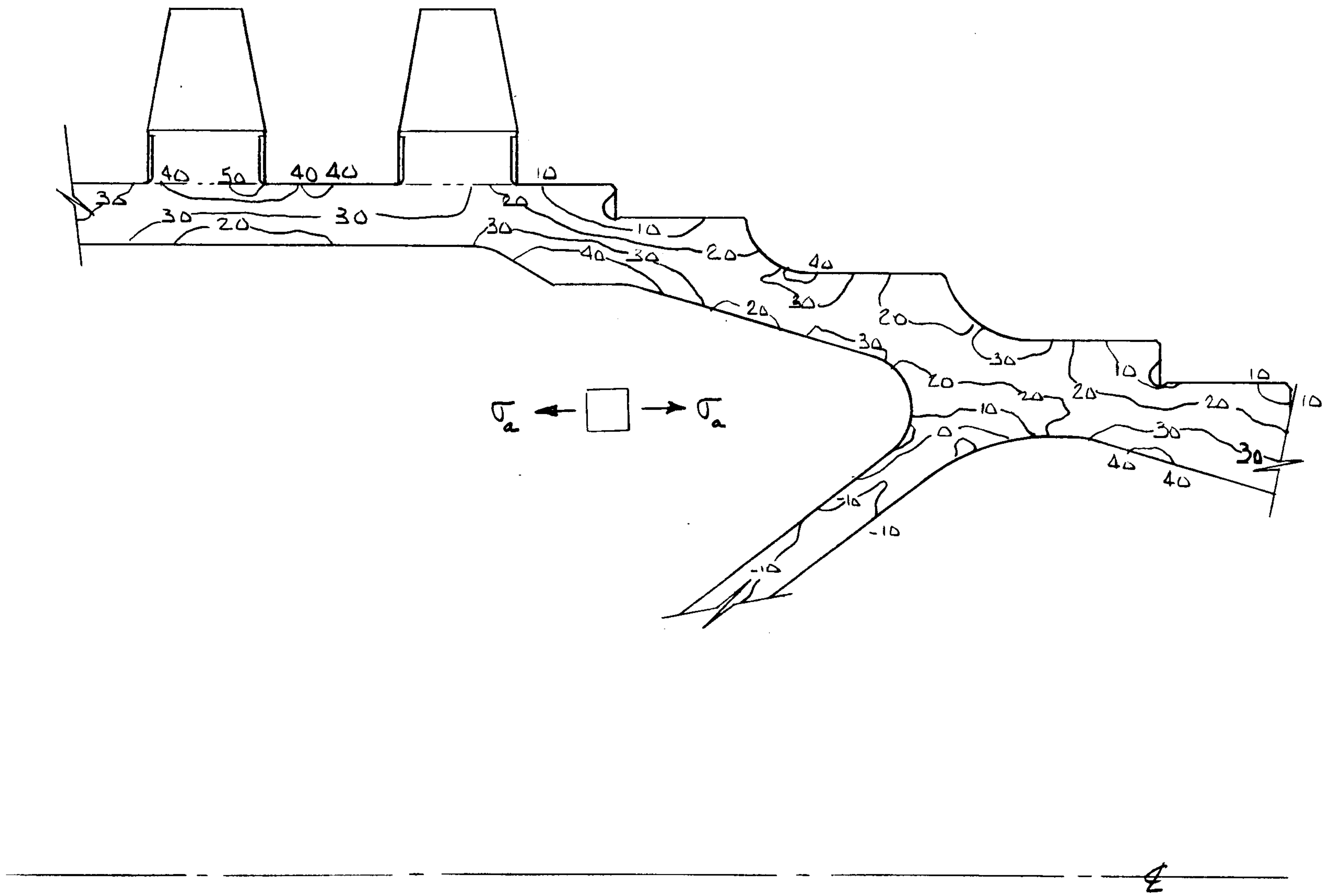


Figure 24. Axial Stress Profile (10^3 psi)

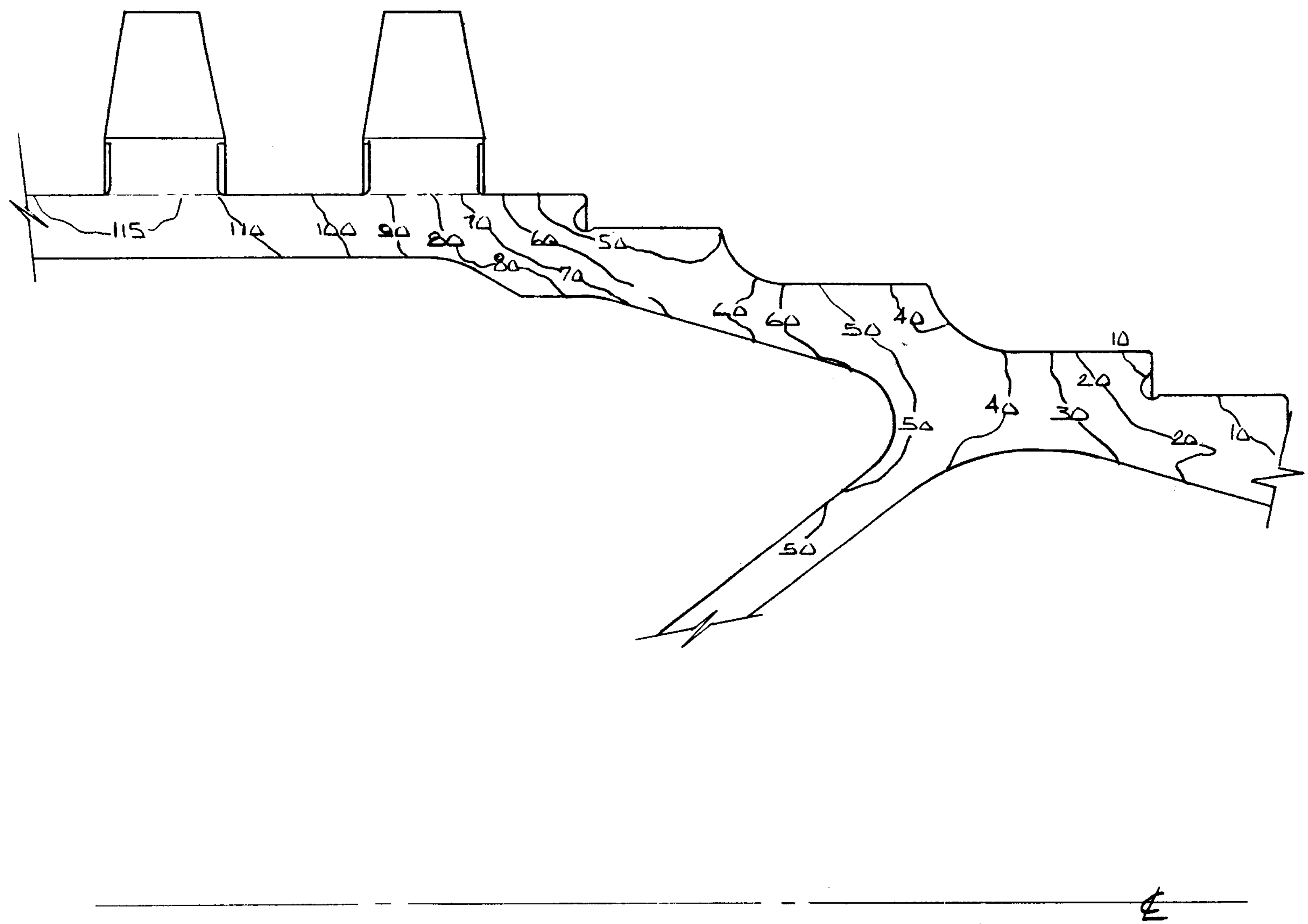


Figure 25. Hoop Stress Profile (10^3 psi)

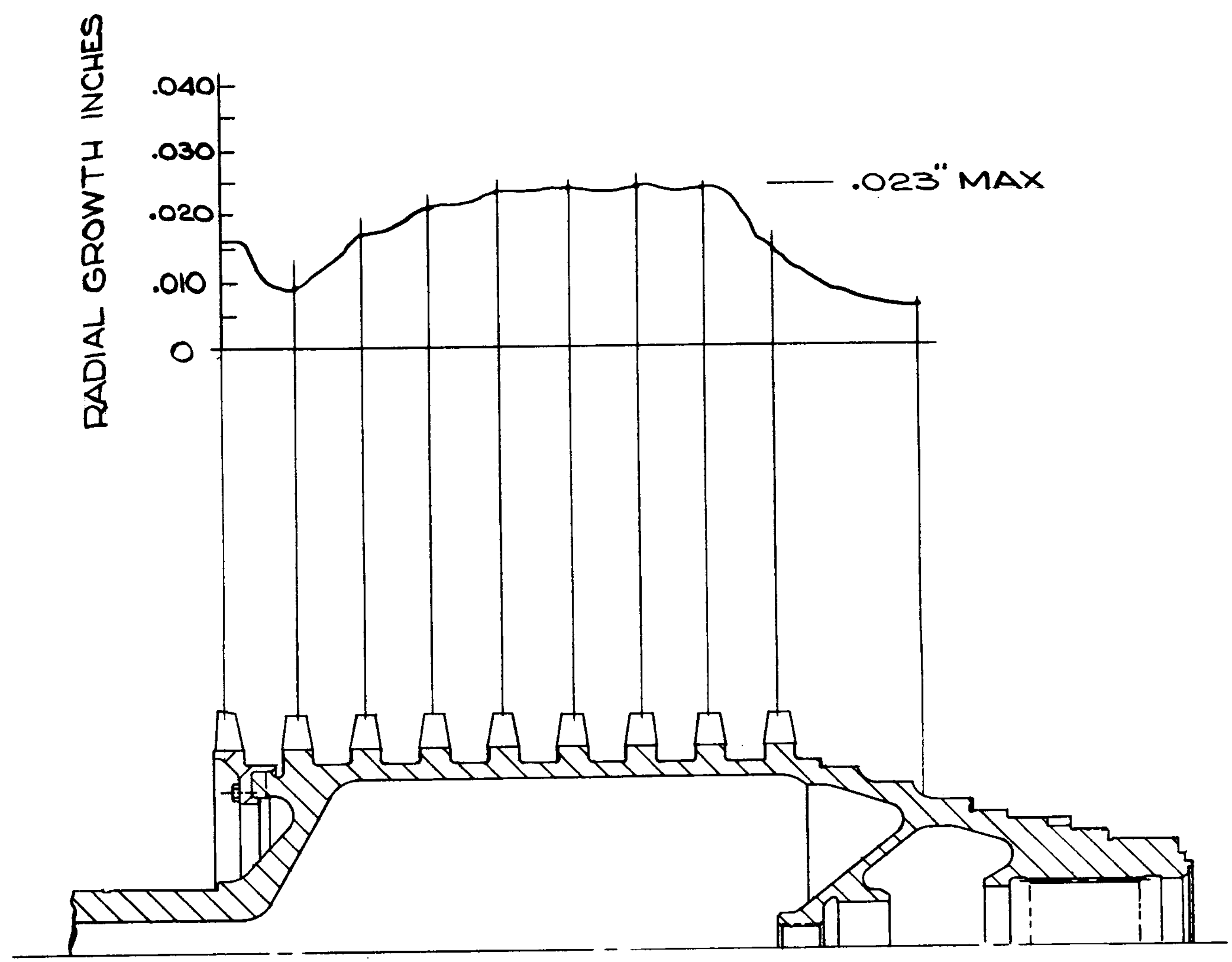


Figure 26. Rotor Assembly Centrifugal Growth, Diagram

The minimum margin of safety indicates the pump rotor could be reduced in weight for future designs. This possibility was analyzed. It is indicated that future rotor designs could be 53 lb lighter in weight and possess a margin of safety of +0.54, based upon the cryogenic mechanical properties of Inconel 718.

e. Materials

The material used for the pump rotor was Inconel 718. This material was selected for its excellent cryogenic mechanical properties and because it can be welded. All rotor weld components were machined from forgings. The weld wire and consumable inserts used for each weld joint were also Inconel 718. The end product pump rotor was solution annealed and aged.

Each forging was designed to include areas for removal of tensile test bars. It was mandatory that these test bars be cut from each forging and material mechanical properties verified as part of the forging acceptance program. This use of test bars proved extremely valuable.

In one case, test bars were taken from opposite faces of a large pancake forging. When tested, they showed a large difference in mechanical properties. It was determined from the resulting investigation that the heat treatment was inadequate to control formation of a laves phase in large Inconel 718 forgings. The heat-treatment schedule was revised and a new specification written which improved the quality of these large forgings.

f. Fabrication

(1) Consideration in Design

The pump rotor was designed with existing fabrication technology as the primary guide. Functional design requirements forces a compromise in some areas particularly those concerning the rotor blade slots. Blade stresses required these slots be set at a 22-1/2 degree angle to reduce the blade base weight. This requirement eliminated broaching of these slots because axial slot alignment is required to give the clearances required for broach entrance and exit.

TIG welding was selected, in preference to other methods, because sufficient information was available to assure success at the end of a relatively inexpensive weld development program. Not enough information was available for other welding methods, such as electron beam and Menasco Uniweld, for them to be considered as the primary weld method.

The consumable weld insert was used to assure a smooth and uniform weld bead in interior locations that were inaccessible for machining after completion of the weld. This smoother bead eliminated stress concentrations and possible crack-generating irregularities (see Figure 27).

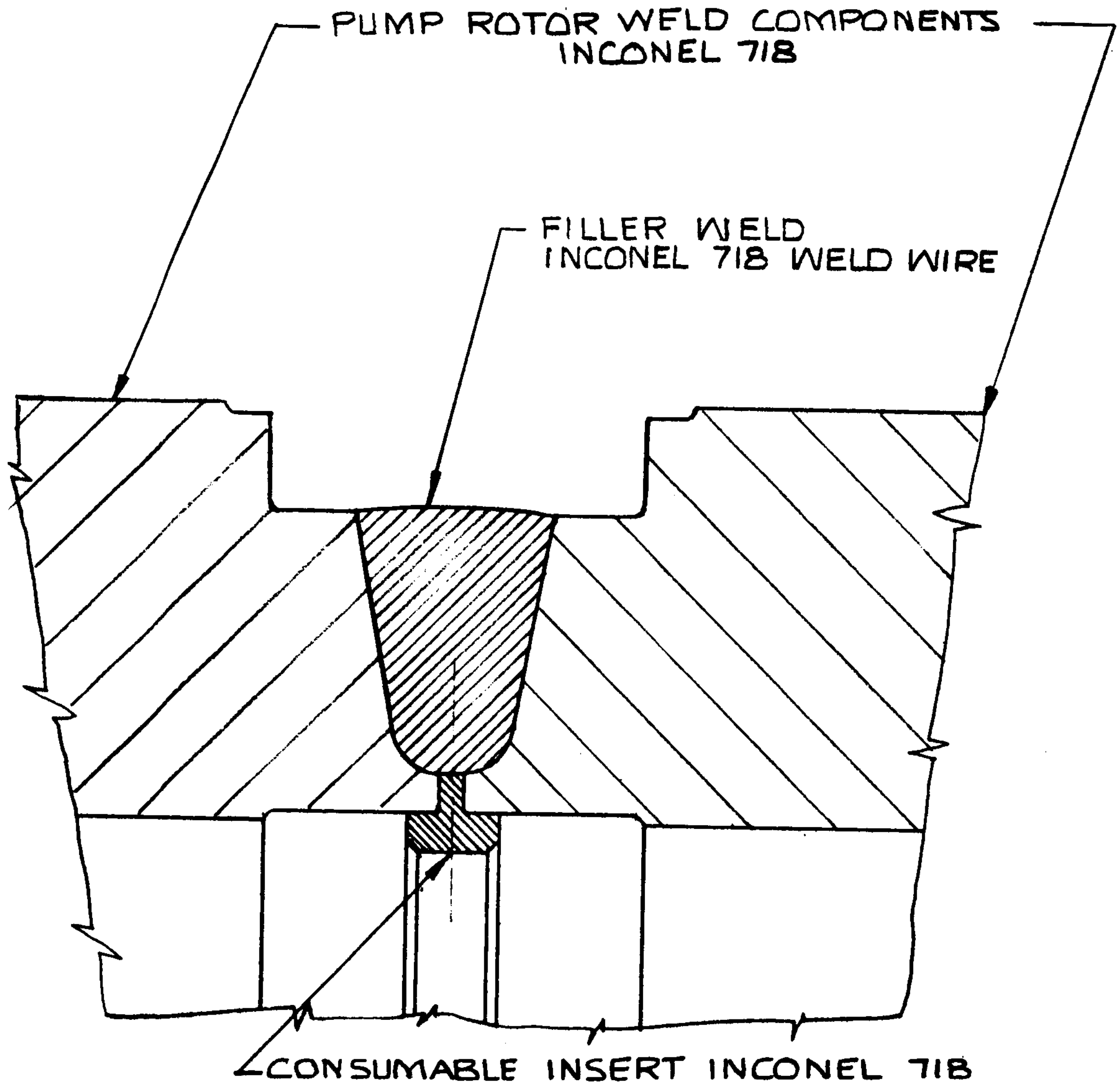


Figure 27. Weld Joint Configuration and Consumable Insert

As an aid to all machining operations, centering conical surfaces and center drills are provided at each end of the rotor. These surfaces, when once machined early in the manufacturing sequence, are never remachined or relocated; therefore, they serve as true tooling reference points at all times.

(2) Problems Encountered

The initial pieces of forged material displayed inconsistencies in mechanical properties. This problem was solved by revising the heat-treatment schedule and releasing a new heat-treatment specification.

Welding of the rotor joints presented much greater problems than indicated by the weld development program. The weld development program proceeded with a minimum of problems from the joining of flat plates through the joining of full scale rings, which duplicated the rotor weld joints. The rotor weld joint preparation, consumable insert size, and weld schedule were established based upon the welding of the full size rings. The final test welds had minimum joint efficiencies of 95%.

The pump rotor welds proved inconsistent as to quality, specifically, the presence of voids and cracks within the weld, which exceeded the Aerojet-General specification limits. The cause was attributed to weld technique, weld schedule, and inadequate nondestructive testing methods.

Those portions of the welding technique that were revised are listed below.

(a) Inert gas mixture and its application.

(b) The number of filler weld passes between intermediate stress relief heat-treatment would be limited.

(c) More frequent dye penetrant inspection of the filler weld buildup.

(d) Exercising extreme care in cleaning the weld surfaces prior to proceeding, after stress relieving, with the next filler weld passes. Weld surfaces that were free of oxide and any foreign matter were found to be a major factor in obtaining satisfactory weld quality.

Nondestructive testing was centered upon improving the use of ultrasonic inspection equipment to supplement dye penetrant and radiographic inspections. The ultrasonic methods were improved to a point where they are considered reliable. Weld flaws that were detected were confirmed by removing weld material to expose the flaw prior to weld repair.

Distortion caused by machining of the pump rotor was considered, at various times, as a possible problem. Constant inspections of concentricities during the final machining operation showed no distortion that could be attributed to machining stresses.

Thermal distortion was considered to be a problem because of the presence of the welds and possible variations in material chemistry of the various forgings. The first pump rotor was placed in an instrumented fixture and the rotor was filled with liquid nitrogen. Dial indicator readings of deflection were made during the chilldown, steady-state temperature period, and warm-up. The cold cycle test showed no appreciable centerline distortion of the pump rotor because of temperature change.

g. Special Tooling Required

Welding required special tooling to properly position and rotate the weld components under the TIG-welding head. All welding, except for any repair of flaws, was done on an Airco automatic welding machine. Special tooling was also required to provide proper back-up gas coverage for the welding operations.

The machining of the dove-tail slots in the rotor periphery required the following special tooling:

(1) Contoured milling cutters that were ground to produce both the desired shape and tolerances.

(2) An indexing fixture that is capable of holding the rotor rigidly and indexing the 47 slots accurately. This fixture was equipped with electric circuits which prevented machine operation in the event that the fixture indexing pins were not properly engaged. In addition, there was a warning light, conspicuously located, that provided visual indication of improper indexing.

2. Inducer, First Stage (See Figure 28)

a. Design Philosophy

The inducer had to meet the hydrodynamic design requirements, be lightweight, and capable of manufacture by more than one type of vane generating equipment.

b. Design Requirements

The hydrodynamic requirements were previously established, specifically the number of vanes, vane length, hub, and tip contours.

The mechanical design aspects required:

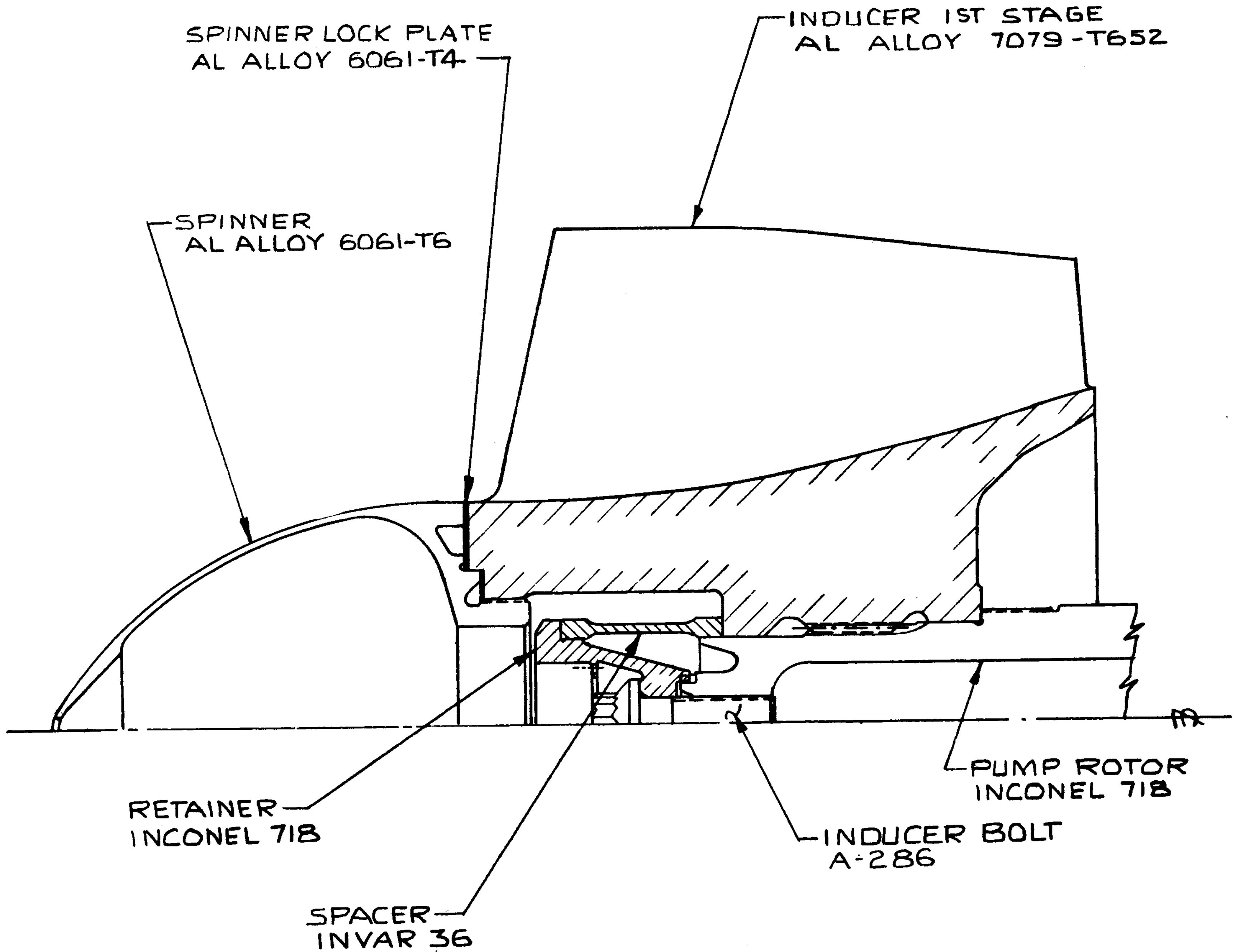


Figure 28. Inducer, First-Stage and Retainers

(1) The inducer to be lightweight to aid in achieving an acceptable critical speed value for the rotating components. Because the inducer is an overhung mass relative to the radial bearings, any excess weight affected the critical speed adversely.

(2) The vanes to have adequate strength for the loads imposed.

(3) Material to be compatible with liquid hydrogen.

c. Description

The inducer is an aluminum alloy part. There are three full vanes, each vane extending for 460 degrees of rotation. There are three partial vanes, one partial vane equally spaced in each of the passages formed by the full vanes. All vane surfaces are canted forward; the vane tip is nearer the inducer inlet than the corresponding point at the vane-hub intersection. This canting of the vanes lowers the vane stress level because of partial cancelling of hydraulic loads by vane centrifugal loads.

The inducer has two internal pilots and an internal spline that form the rotor interface surfaces. The inlet end of the hub has provisions for attaching and locking the spinner in place.

d. Stress and Vibration

The inducer installed in the fuel turbopump assembly Model I S/N 001 is of interim design. This part is the result of the following two significant items.

(1) Lower than expected material mechanical properties at -423°F.

(2) Excessive vane stresses in the original design were revealed by an upgraded stress approach. The stress analysis upgrading was based upon the test history of a "POGO" impeller having an inducer section similar to the fuel turbopump assembly first-stage inducer. Also, a flat plate inducer pressure loading theory was included in the analysis. In this analysis, the bending stresses caused by theoretical pressure loadings were modified by a factor which took into account a relationship of stress at failure to analytical stress. This relationship was determined from "POGO" impeller test history.

The thick vanes and three degree forward inclination of the vane centerline are the direct result of a stress analysis. The three degree centerline inclination served to reduce vane bending stresses by using vane rotational loads to oppose the vane hydraulic loads thereby achieving an acceptable vane stress level. The final interim design has a maximum combined stress of 38,000 psi because of the pressure and rotational loads. The inducer spline can more than adequately withstand stresses as the calculated margin of safety is +3.4.

Vibration analysis results are tabulated in Figure 29.

e. Material

The inducer is machined from an aluminum alloy 7079-T652 forging. This material was selected because of its strength-to-weight ratio and ease of machining. Such data as was available at the time of material selection indicated 7079-T652 had elongation exceeding 3% at -423°F. Tests of material samples taken from large forgings revealed the actual elongation was less than 3%. This resulted in an increase in the inducer vane thickness to lower stresses sufficiently to be compatible with such low elongation.

The forging design included areas for test bar removal. Material properties were verified by taking a test sample from the forging.

f. Fabrication

(1) Consideration in Design

Machining of the vanes presented the only difficult aspect of manufacture. Vane surface geometry was based upon cutting full vane depth with a cylindrical cutter, side milling. This selection made possible the use of the following different manufacturing methods: Omnimill, cam-controlled machines, and three-dimensional duplication from a model.

Vane surfaces were defined by a coordinate system that permitted accurate inspection without the need for special tools. Standard inspection equipment available in any machine shop can be used.

(2) Actual Problems Encountered

The inducer installed in fuel turbopump assembly Model I S/N 001 was manufactured using cam-controlled equipment. No difficulties were experienced.

3. Inducer First-Stage Retainers (See Figure 28)

a. Design Philosophy

The inducer retainers were to be as simple as possible, consistent with their function.

b. Design Requirements

The retainer system had to keep the inducer positioned on the rotor shaft. This system had to compensate for thermal contraction differences caused by the aluminum inducer and the Inconel 718 rotor shaft. It also had to be capable of withstanding the thrust loads created by the hydraulic loads on the inducer surfaces.

VIBRATION TESTS AT ROOM TEMP.			
FULL VANES		PARTIAL VANES	
MODE	FREQ (CPS)	MODE	FREQ (CPS)
1ST BENDING	1,078	1ST BENDING	1,300
2ND	1,253	2ND	1,928
3RD	1,295	3RD	2,308
4TH	1,350	4TH BENDING	2,706
5TH	1,419		
6TH	1,525		
7TH	1,723		
8TH	1,935		
9TH	2,012		
10TH	2,160		
11TH BENDING	2,272		

POSSIBLE EXCITATIONS, FTPA
MOD I AT 12,000 RPM

(1.) SHAFT SPEED - 200 CPS
2× (200) = 400 CPS
3× (200) = 600 CPS

(2.) 3 PARTIAL VANES × RPM = 600 CPS
6 TOTAL VANES × RPM = 1,200 CPS

(3.) 11 EXIT GUIDE VANES × RPM = 2,200 CPS

NOTE:
LH_2 VIRTUAL MASS EFFECTS WILL CHANGE OPERATING FREQUENCIES SLIGHTLY.

Figure 29. First-Stage Inducer

c. Description

The inducer retainer system consists of three parts; an Invar 36 spacer, an Inconel 718 retainer, and a high strength bolt.

The Invar spacer is a cylindrical part, which because of its extremely low coefficient of thermal expansion, serves as the compensating unit for differences in axial dimension between the inducer and the pump rotor. The spacer walls were kept as thin as possible to assure rapid and complete cooling of the part to -423°F.

The retainer is the connecting member between the spacer and the inducer bolt. It also serves as the locking member for the inducer bolt. There are a series of holes through the retainer that meter the liquid hydrogen flow from the rotor interior into the inducer spinner cavity.

The inducer bolt is the final member that joins the retaining system to the pump rotor.

d. Stress and Vibration

The stress analysis of the inducer attachment took into consideration the bolt pre-load of 54,000 lb plus the inducer thrust load of 70,000 lb and the effects of differential contraction of the components in chilling down to -423°F. From the analysis it can be concluded that the attachment is adequate for all loading conditions considered. The minimum margins of safety are: bolt + 0.36, retainer + 0.34, and spacer + 0.63.

These margins of safety correspond to the following maximum stress values: bolt = 132,000 psi, retainer = 141,000 psi, and spacer = 61,200 psi.

The maximum stress occurs under pump operating conditions.

e. Materials

Each material was selected for a specific characteristic.

The Invar 36 was selected for its low coefficient of thermal expansion.

The Inconel 718 was selected for its high mechanical properties. Its thermal expansion coefficient was identical to the pump rotor.

The inducer bolt is made from A-286, which has high mechanical properties and is suitable for cryogenic use.

f. Fabrication

These parts are of simple and conventional design. No manufacturing difficulty was expected or experienced.

4. <u>Inducer, Second Stage</u> (See Figure 30)

a. Design Philosophy

This part, simple in concept, is a full ring with unshrouded blades on the periphery.

b. Design Requirements

This component had to be of minimum weight consistent with its function, be removable, and have positive piloting during operation to preserve both concentricity and dynamic balance. There had to be a positive driving engagement between the inducer and the pump rotor.

c. Description

The second-stage inducer consists of a ring having 31 unshrouded blades on the outer surface. Two concentric piloting diameters are provided; one for locating the part at room temperature, the other for locating the part under operating conditions. This double piloting was necessary because of the differences in contraction rates of the titanium inducer and the Inconel 718 pump rotor. Three lugs were used for positive engagement to the pump rotor.

Twelve bolts hold the second-stage inducer to the pump rotor.

d. Stress and Vibration

The low blade weight, resulting from the use of a titanium alloy, and the relatively light hydraulic loading result in low blade stresses. The maximum stress occurs at the blade hub and has a calculated value of 20,340 psi. This gives a margin of safety of +5.3.

The attachment of this inducer to the rotor was analyzed. Thermal effects, blade weights, bolt pre-torque, and rotational loads were considered in this analysis. The inducer ring has a maximum stress of 112,200 psi with a resulting margin of safety of +0.28.

The attaching bolts show low axial loads, 28,700 psi, but high bending loads, 216,000 psi, because of ring deflection during operation. Disregarding a safety factor, it was stated in a stress analysis that the bolts are adequate and have a margin of safety of +0.28. During the analysis a safety factor of 1.5 was applied to a high strain low cycle fatigue investigation. It is indicated from this investigation that 615 loading cycles could occur before the bolt would fail. This is considered an adequate margin for the fuel turbopump assembly Model I.

Blade vibration proved to be a problem. This resulted in an airfoil redesign so the resonant frequencies would not fall within the pump design speed band. The blades used in the existing pump are satisfactory in all

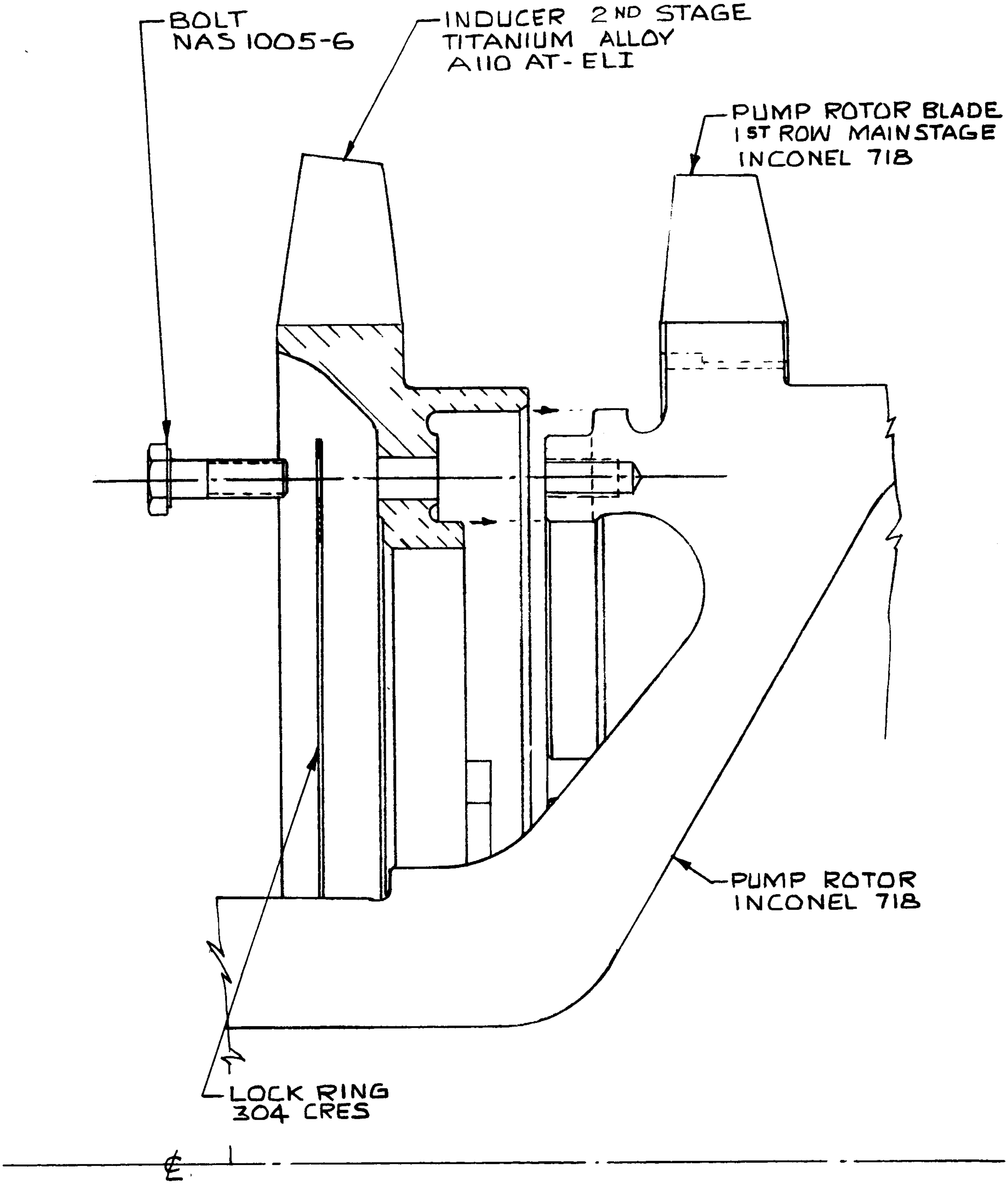


Figure 30. Inducer, Second-Stage and Mountings

respects as shown both by analysis and by vibration tests. Blade natural frequencies and the log decrements for material damping were determined by test. The resonant stress levels were determined by equating vibratory work input to damping work dissipated per cycle by using the experimentally determined log decrement. The following data were obtained based upon blade excitation resulting from eleven upstream vanes in the guide vane housing.

First Flatwise Flexural = 3050 cps

First Torsional = 6150 cps

These are equivalent to the following pump speeds:

3050 cps = 8300 rpm

6150 cps = 11,100 rpm

This analysis shows stress levels to be low at the resonant points (see Table 11).

First Flatwise Flexural = 10,000 psi and a margin of safety of +9.0

First Torsional = 2400 psi and a margin of safety of +42.0

e. Materials

This part is machined from a ring forging of titanium alloy, A110 AT-ELI (extra low interstitial). This alloy contains 5.5% tin and 2.5% aluminum. The titanium alloy was selected for its excellent strength-to-weight ratio plus its suitability for use in liquid hydrogen.

To obtain maximum material strength, the material was specified as a die-forged ring that approximates the contours of the finished machined inducer. As with other forgings, areas were provided for removal of tensile test bars to be used in assuring material quality prior to accepting each forging.

f. Fabrication

(1) Consideration in Design

The design of the second-stage inducer was based upon common manufacturing methods. The only area presenting any machining problems were the driving lugs. The design included adequate clearances and tool run-out areas to facilitate manufacture.

TABLE 11

SECOND-STAGE INDUCER

Mode	Test Frequency (cps) at 70°F	Design Frequency (cps) at -420°F
1st Flatwise Flex.	3136	3050
2nd Flatwise Flex.	12,160	12,100
1st Torsional	6410	6150

Mode	Frequency (cps)	Harmonic Of Excitation	Resonant Speed (rpm)	Calculated Resonant Stress Level (ksi)	M.S.
1st Flatwise Flex.	3050	2	8300	10.0	9.0
		3	5500	11.0	8.1
1st Torsional	6150	3	11,100	24.0	42.0

Excitation = 11 Upstream Guidevane x rpm x 1, 2, or 3.

(2) Actual Problems Encountered

No problems were encountered in the machining of this part.

Material difficulties were experienced in the form of excessive hydrogen gas content. This was discovered in materials tests of the ring forgings at cryogenic temperatures.

The high hydrogen content resulted in extremely low elongation at cryogenic temperatures. The material would be subject to failure from vibration and the anticipated loads. This problem was solved through degassing of the completed parts. The degassing was done at elevated temperature in a vacuum furnace. Subsequent materials tests showed the material to have an acceptable hydrogen content and excellent cryogenic mechanical properties.

5. Mainstage Rotor Blades (see Figure 31)

a. Design Philosophy

The rotor blade was based upon design experiences.

b. Design Requirements

The blade had to be readily removable from the pump rotor, yet be capable of withstanding all predicted loading conditions. This resulted in the dove-tail blade base set at 22 degrees 30 minutes relative to the pump axis of rotation.

The angled base was necessary to reduce blade weight and blade stresses to an acceptable level.

The blade design had to be capable of stagger angle changes of ± 2 degrees within the blade platform limits. This would permit reblading of the rotor at minimum cost should test data show stage matching unsatisfactory.

c. Description

The airfoil section varies in thickness, chord length, and stagger angle from blade base to tip with the base section being the thickest. The net result is a twisted airfoil surface typical of the type used in compressor work. The C4 airfoil was used for all sections.

The blade base has a dove-tail that mates with the pump rotor. The dove-tail centerline is angled 22 degrees 30 minutes relative to the pump centerline.

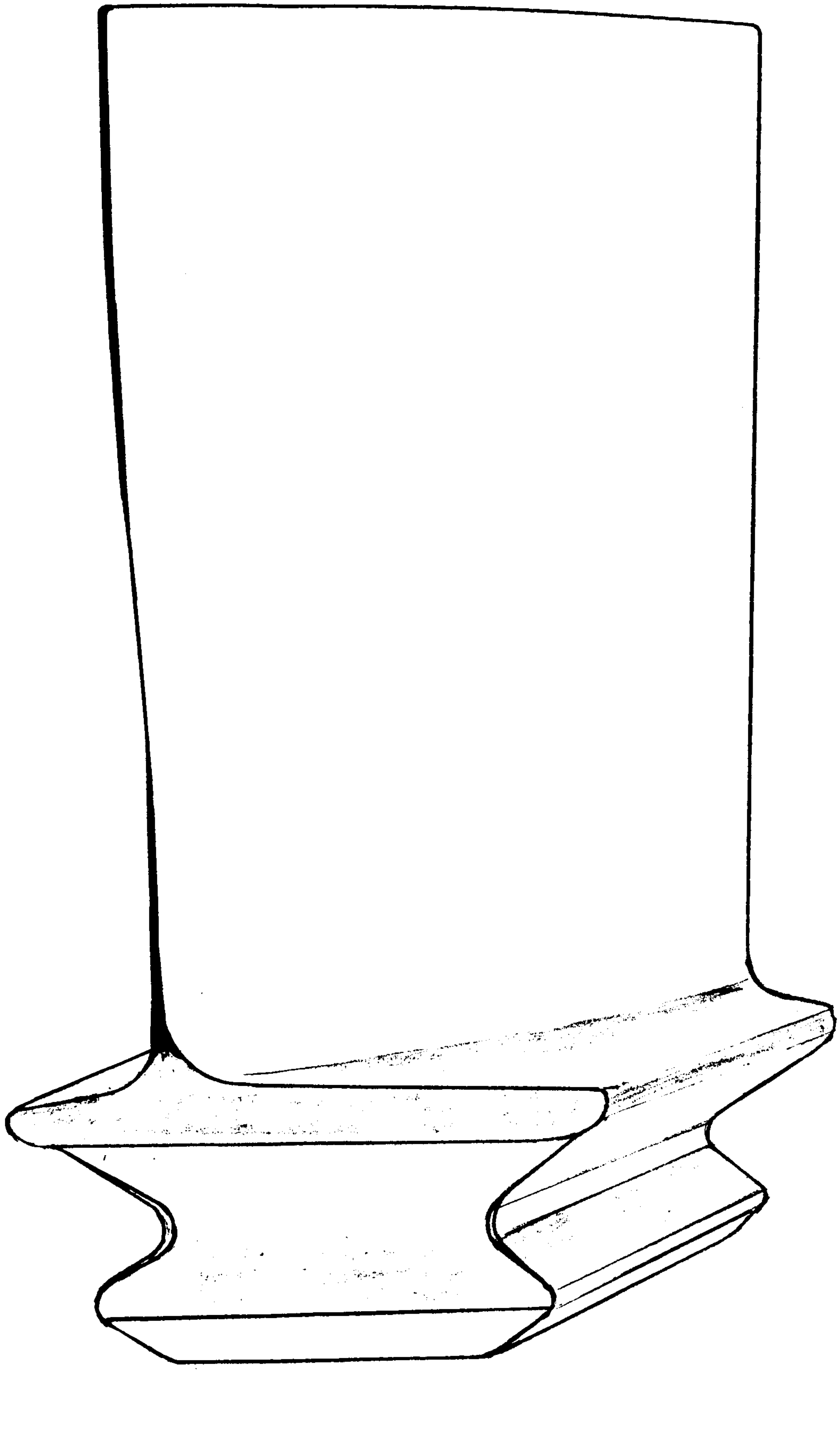


Figure 31. Rotor Blade

d. Stress and Vibration

An analysis of the mainstage rotor blade stresses indicated that the stresses were relatively low in both the dove-tail and the blade root. These maximum stresses are:

Neck of dove-tail = 68,500 psi with a margin of safety of +1.03.

The blade root = 53,000 psi with a margin of safety of +1.64.

Fatigue considerations show a minimum margin of safety of +0.37. This is based upon a mean stress of 56,000 psi and an alternating stress equal to 30% of the mean stress. These stresses were applied to a Goodman diagram giving an allowable stress of 129,000 psi. A safety factor of 1.25 was included when calculating the margin of safety.

A vibration analysis revealed only two problems. Both occur in the first flat flexural mode at 4100 rpm and 4400 rpm. The stress levels become sufficiently high to result in a negative margin of safety. Tabulations of all resonant frequencies and corresponding stress values are shown in Figure 32.

This critical vibration point should be passed through quickly during the start transient and shutdown phases of pump operation and not cause any blade failure. All other modes show positive margins of safety and appear to be satisfactory with regard to pump performance.

e. Materials

The blade is fabricated from an Inconel 718 forging that has been solution-annealed and aged. This material was selected for its high mechanical properties and suitability for use in liquid hydrogen. The use of this material also eliminates any differential contraction problems between the blade and the pump rotor to assure excellent blade-to-rotor fit under all temperature and operating conditions.

f. Fabrication

This part is typical of turbine and compressor blade design. There is nothing unusual or different from common manufacturing methods or practices.

6. <u>Pump Rotor, Rotor Blade Assembly</u> (See Figure 33)

a. Design Philosophy

This assembly was to be as uncomplicated as possible, consistent with the function of the primary parts involved. Emphasis was placed upon keeping the parts and assembly tools as simple as possible.

MODE	TEST FREQUENCIES AT 70°F (CPS)	DESIGN FREQ (CPS) AT 420°F *	
		45 BLADES EXCIT	57 BLADES EXCIT
1ST FLAT FREQ	2,970	3,300	3,300-3,900
2ND FLAT FREQ	14,210	16,000	16,000-18,500
1ST TORSION	6,530	7,850	7,850-8,500
2ND TORSION	14,710	16,700	16,700-19,200

* TEST DATA MODIFIED TO ACCOUNT FOR VARIABLE BLADE HEIGHT.

EXCITATION: 45 UP-STREAM 2ND STAGE INDUCER STATOR BLADES

MODE	FREQ (CPS)	HARMONIC	RESONANT SPEED (RPM)	RESONANT STRESS (KSI)	M. S.	RECOMM MIN CLOSENESS TO RESONANT (RPM)
1ST FLAT FLEX	3,300	1	4,400	56.5	-.08	± 80
		2	2,200	19.6	+1.65	
2ND FLAT FLEX	16,000	2	10,900	36.0	+.27	±107
		3	7,100	17.2	+1.56	±72
1ST TOR	7,850	1	10,700	23.6	+.29	±178
		2	5,200	7.0	+3.35	
2ND TOR	16,750	2	11,200	2.64	+10.5	
		3	7,400	1.65	+17.4	

EXCITATION: 57 UP-STREAM MAINSTAGE STATOR BLADES

MODE	FREQ (CPS)	HARMONIC	RESONANT SPEED (RPM)	RESONANT STRESS (KSI)	M.S.	RECOMM MIN CLOSENESS TO RESONANT (RPM)
1ST FLAT FLEX	3300-3900	1	4,100	82.0	-.35	± 35
		2	2,050	19.8	+1.7	
2ND FLAT FLEX	16,000-18,500	2	9,800	35.6	+.24	±59
		3	6,500	17.4	+1.39	±37
1ST TOR	7,850-8,500	1	8,900	17.8	+.71	± 134
		2	4,500	5.7	+4.3	
2ND TOR	16,200-19,200	2	10,200	2.50	+11.1	
		3	6,800	1.44	+20.1	

Figure 32. Mainstage Rotor Blade

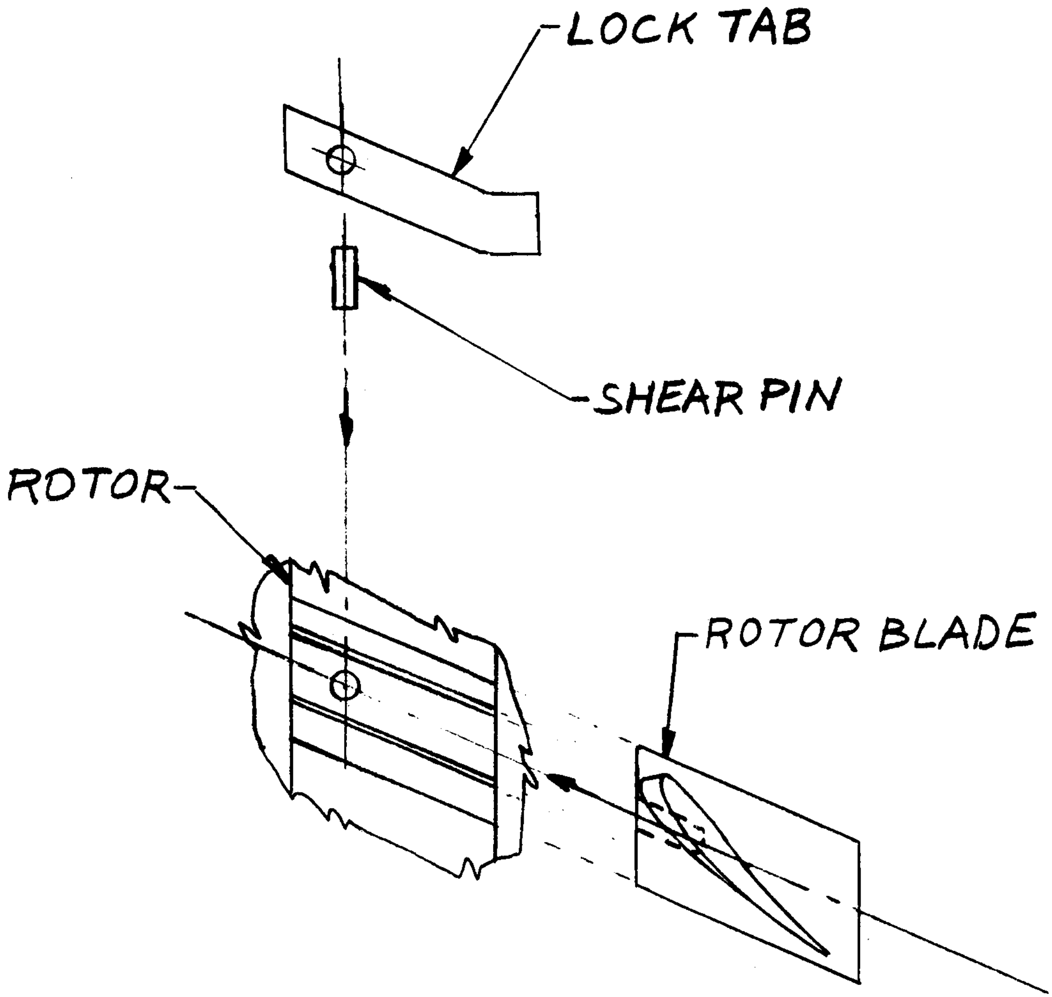


Figure 33. Rotor Blade Assembly

b. Design Requirements

The blades must meet the following requirements:

(1) Be removable

(2) Be positioned accurately to keep axial clearances to adjacent stator blades to minimum tolerance variations.

(3) Be positioned radially in the same contact situation as would be experienced during operation. This factor would aid blade tip grinding and the maintaining of the close diameter and concentricity tolerances required for satisfactory pump operation.

(4) Blade installation and removal must be as easy as possible. Necessary tooling is to be uncomplicated and easy to use.

c. Description

The blade is axially positioned in its slot by a shear pin. Accuracy of position is achieved by controlling the pin hole location in the pump rotor and the length of a groove cut in the underside of the blade dove-tail. The forces acting upon the rotor blade keep it against the shear pin under all operating conditions. Radial positioning of the blade is accomplished by means of a bend in the locking tab. The tab, being bent instead of flat, must be depressed when inserting the blade in its rotor slot so that the tab acts as a leaf spring to keep the rotor blade positioned in the same contact conditions as will rotational loads. The rotor blade is locked in position by bending the protruding section of the lock tab up and under the overhanging portion of the rotor blade base.

The following precautions are taken during pump rotor blade assembly. Each blade is weighed. The blades are positioned by weight so as to distribute weight variations evenly, thus keeping unbalance to a minimum. Each lock tab is inspected for cracks after bending. Any indication of a crack is cuase for tab removal and replacement.

The assembly operation is completed by grinding the blade tips to the required finished diameter and concentricity.

d. Stress and Vibration

The stress and vibration studies of the pump rotor and the mainstage rotor blades have been discussed above. Only the locking device needs to be reviewed.

Analysis was conducted on the blade shear pin and locking tab. The shear pin has the following stress values:

(1) Shear Stress = 26,900 psi

Margin of Safety = +3.33

(2) Bending Stress = 224,000 psi

Margin of Safety = +0.87

The locking tab has a maximum load capability, based upon material ultimate tensile strength, of 47 lb. Because no estimate can be made as to reverse blade loading, no margin of safety can be calculated. To date, it appears unlikely that the rotor blades can be subjected to any sort of load reversal. The lock tabs appear to be completely satisfactory.

e. Materials

Materials for the rotor blades and pump rotor have been previously discussed in this report.

The shear pin is made from Inconel 718, solution annealed and aged. Maximum strength is required.

The lock tabs are made from Inconel 718, which is solution annealed. Maximum ductility combined with moderate strength was required for these parts.

f. Fabrication

Fabrication and assembly were primary factors in the design of this assembly and locking method.

The shear pin holes are easily held to low tolerance limits because they were located to be easily jig-bored.

The slots in the base of the rotor blades were designed to be easily machined by conventional milling procedures to low tolerance limits.

The lock tabs were designed to be die-cut and formed in a punch press.

The final product, its ease of part manufacture, and ease of assembly supported the design philosophy and concept.

7. <u>Thrust Balance Disc and Attachment</u> (See Figures 34 and 35)

a. Design Philosophy

The design of this unit was restricted by space limitations and functional requirements.

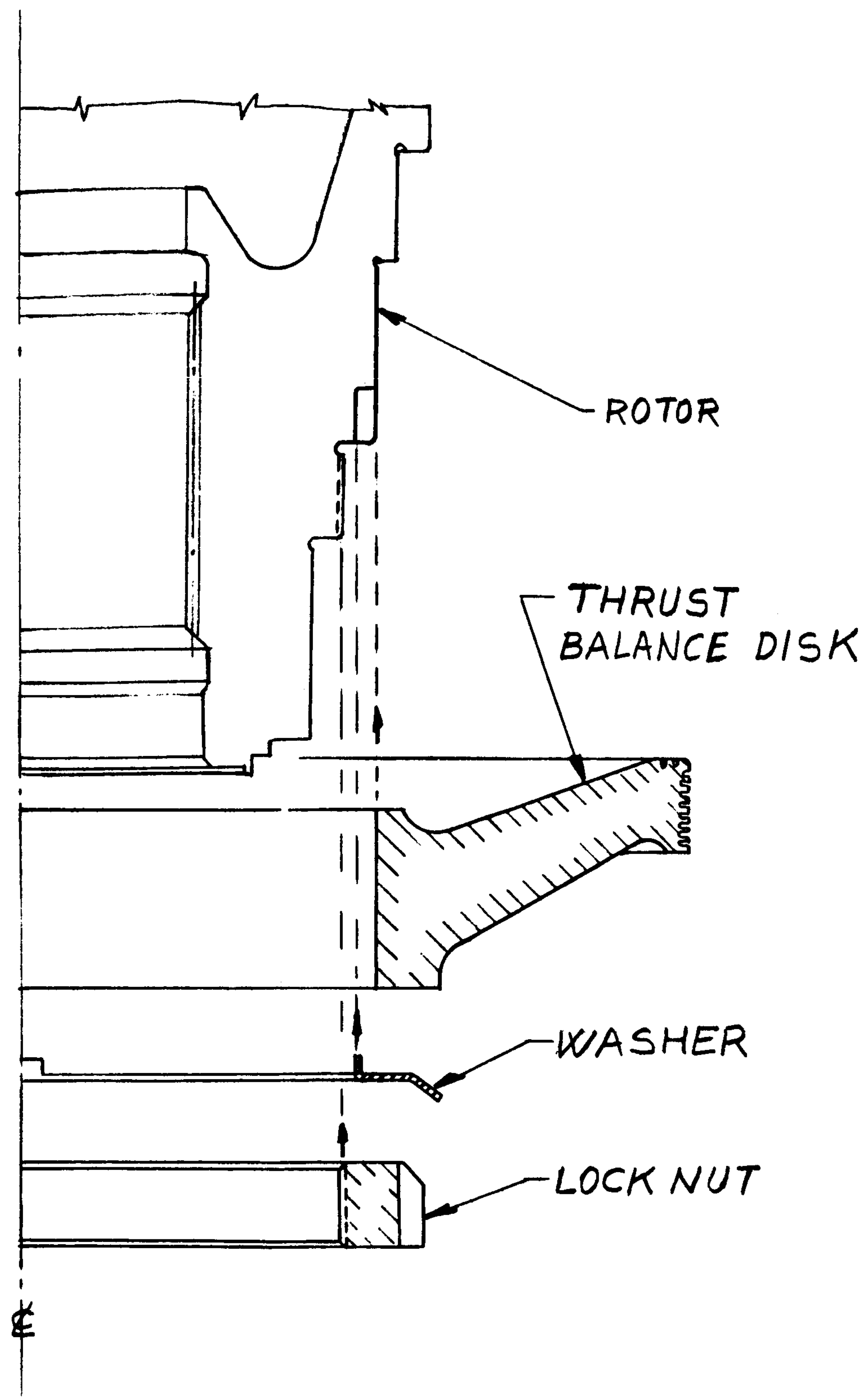


Figure 34. Thrust Balance Disc and Retainers

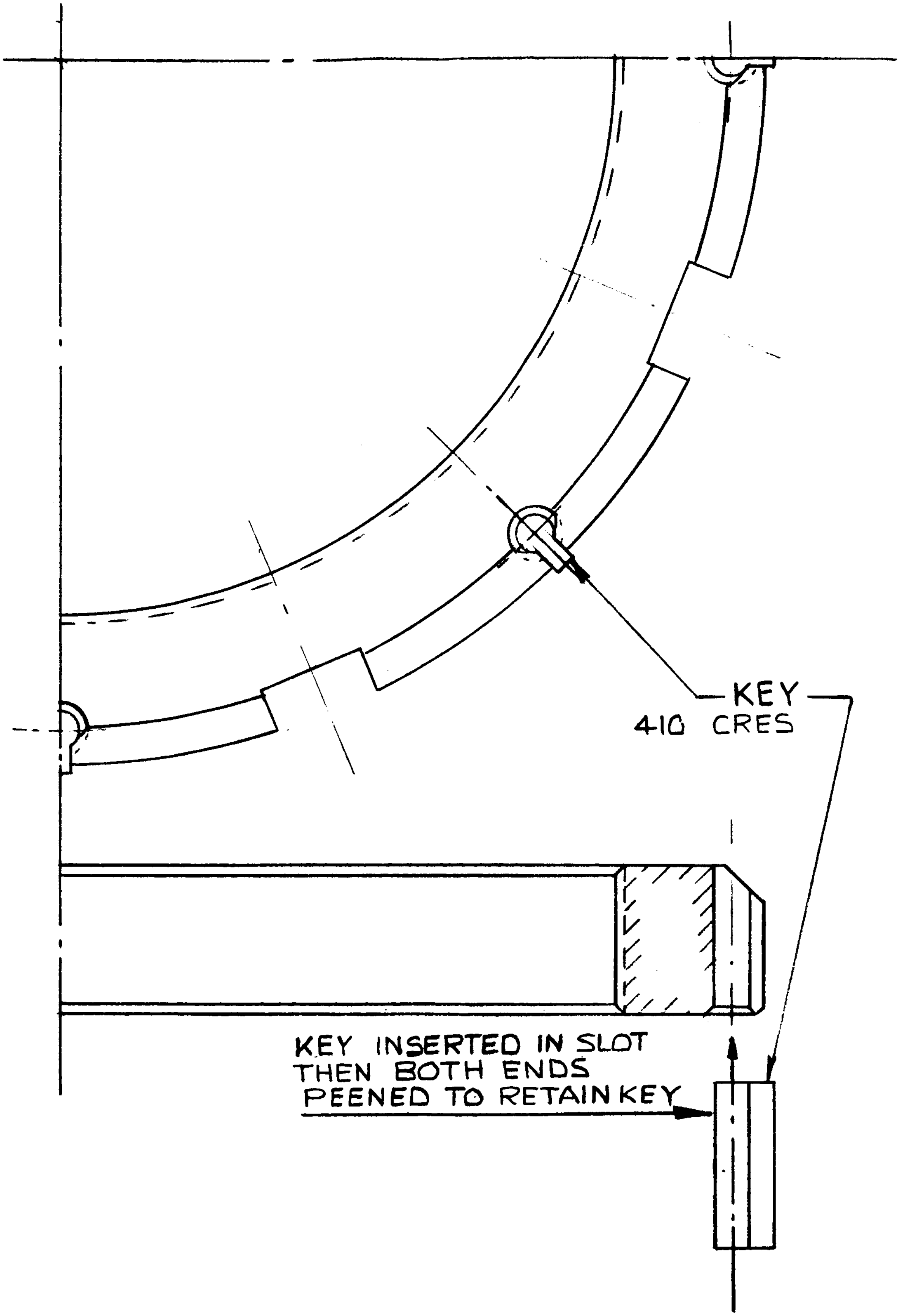


Figure 35. Lock Nut, Balance Disc

b. Design Requirements

The design requirements for the thrust disc were as follows.

(1) Minimum weight, thus producing the least effect upon critical speed.

(2) Maximum face area, thus producing adequate thrust balancing force.

(3) Fit within limits of available space.

(4) Be attached to the pump rotor so as to have positive drive and zero liquid hydrogen leakage past the interface.

c. Description

The thrust disc is a conically-shaped aluminum alloy part. The cone design was necessary to achieve maximum outside diameter and minimum inside diameter within the space available for the unit. The disc drive and seal are accomplished by an interference fit on the pump rotor. The disc is held in position by a lock ring and lock nut. These serve to provide a shaft shoulder, which transmits thrust disc forces into the pump rotor. The lock nut also serves as the toothed member for the magnetic speed probe. Because Inconel 718 is nonmagnetic, small pieces of 410 stainless steel are inserted in the periphery of the nut to actuate the speed probe.

d. Stress and Vibration

The final design of the thrust balancing disc was based upon a stress analysis. The maximum stress is 49,100 psi in the tangential direction at the pump end of the disc bore. The resulting margin of safety is +0.61.

The radial growth of the thrust disc was also calculated during this analysis. The calculations included disc variation in size caused by rotational loads and temperature effects. The change in surrounding housing size caused by temperature change was also taken into account. The final calculation resulted in a disc diameter net change necessary to achieve a 0.020-in. radial clearance to the housing during pump operation (see Figure 36). Note that the radial growth is uneven because of disc deflection from axial loads as well as rotational loads.

A vibration analysis of the thrust balancing disc indicated a critical speed of 26,100 rpm, which is nearly double the pump operating speed. No vibration difficulties should occur.

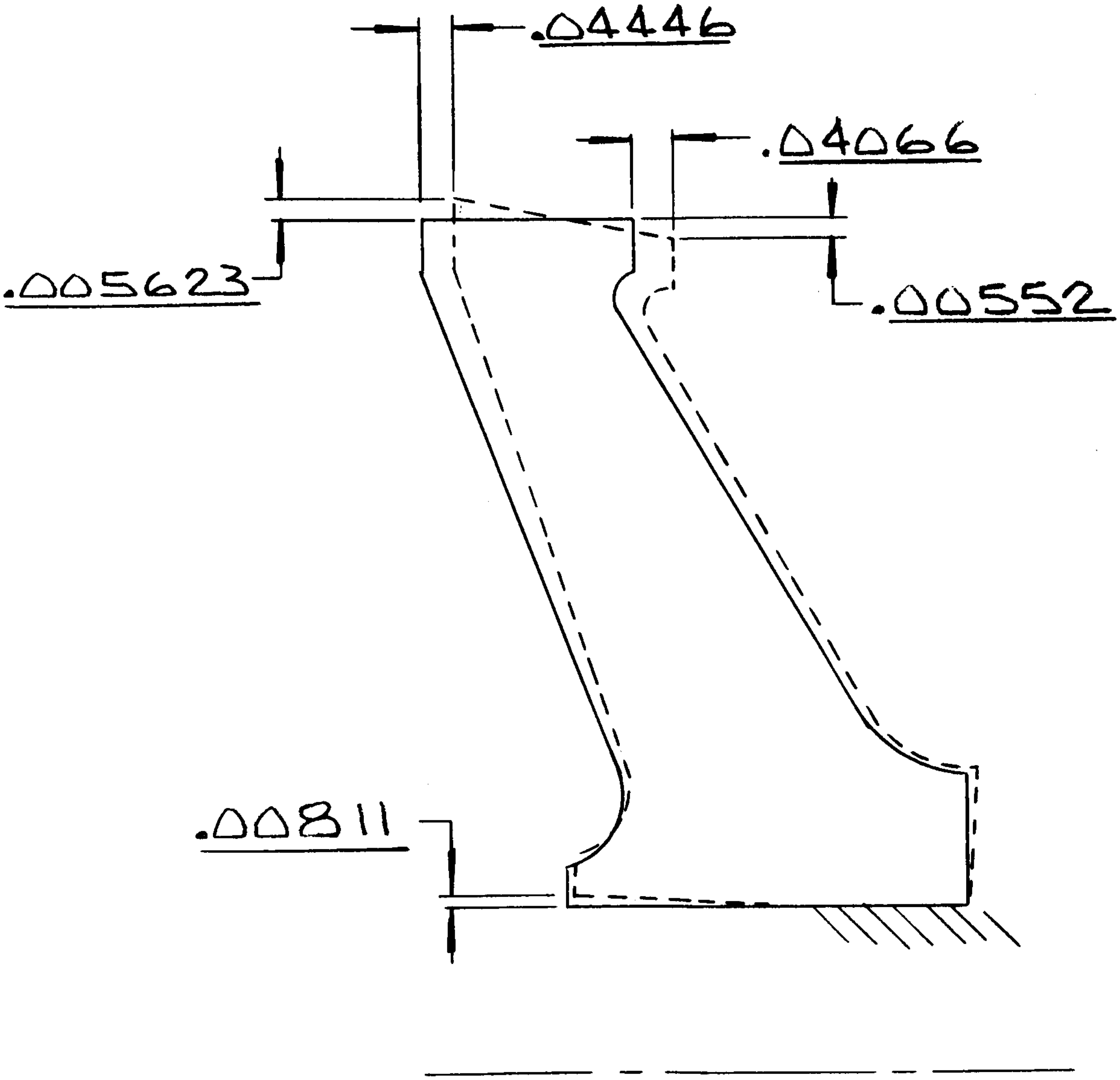


Figure 36. Thrust Balance Disc Deflection

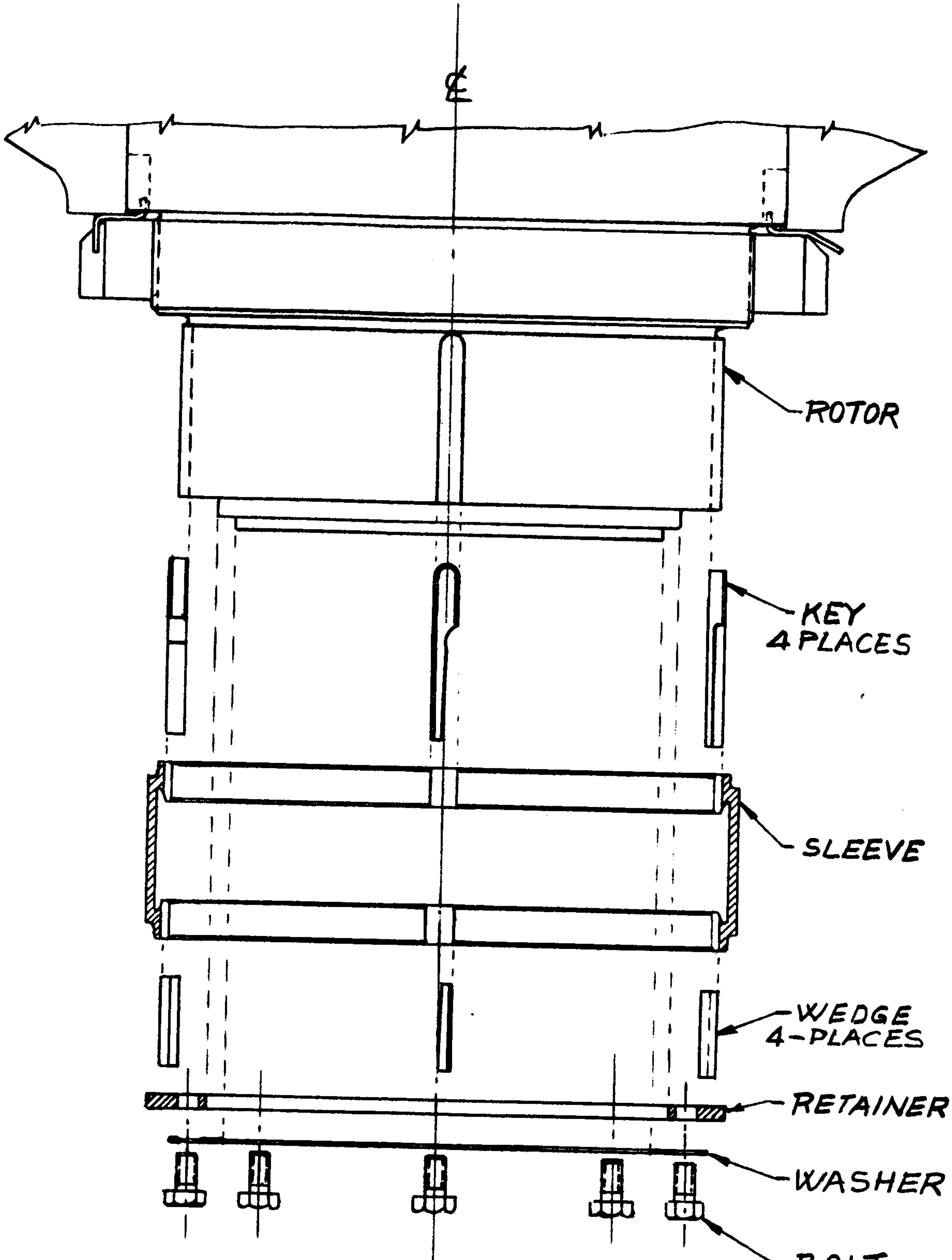


Figure 37. Assembly of Torquemeter Sleeve

e. Materials

Aluminum alloy 7075-T6 was selected for the thrust disc because of its lightweight and acceptable cryogenic mechanical properties. Adequate material elongation was achieved by designing a die forging having a maximum thickness of less than 3-in. Provision was made in the forging design for removal of test bars so that the mechanical properties of each forging can be checked prior to acceptance.

The lockwasher was made from 304 stainless steel, which is adequate for this purpose.

The locknut is made from Inconel 718, which is solution-annealed and aged. Strength plus the same thermal coefficient of expansion as the pump rotor were necessary for this part.

f. Fabrication

These parts were typical of previous designs and presented no manufacturing problems.

8. Torquemeter and Attachment (See Figure 37)

a. Design Philosophy

This part is a simple sleeve, which had to be attached in a pre-loaded condition to the pump rotor. The problem was one of attachment and sleeve pre-loading.

b. Design Requirements

The sleeve had to be rigidly attached to the pump rotor so as to respond to torsional deflections. The material had to be of high magnetic permeability to provide the desired response in the surrounding stationary coil.

c. Description

The sleeve is quite conventional in design. It is relieved on the interior so that only the ends of the sleeve are keyed to the pump rotor. The sleeve walls are as thin as possible to reduce rotating weight and provide minimum resistance to torsional deflection.

Each key is composed of two wedge-shaped units. The wedges compensate for machining tolerance accumulation, thereby yielding a rigid attachment. These keys also provide a sleeve torsional pre-load. The installation method is discussed in the assembly section.

d. Stress and Vibration

The torquemeter sleeve, including its keys, was shown by stress analysis to have a maximum stress of 21,000 psi. This value results in a margin of safety of +1.15. The analysis included not only the effect of speed but also the pre-load caused by the installation of the wedge keys.

e. Material

The torquemeter sleeve is made from Permanickel Alloy 300. This material was selected strictly for its permeability.

The wedge keys were fabricated from Inconel 718 because of its strength and to obtain the same coefficient of thermal contraction as the pump rotor.

f. Fabrication

These parts were readily made by conventional methods. No problems were experienced.

B. STATIONARY ELEMENTS

1. <u>Inlet Elbow, Inducer Housing, and Mainstage Housing</u>

These housings will be discussed as a group because they are similar in concept, material, function, and fabrication.

a. Design Philosophy

The external housings were not to be axially split under any conditions. The cylindrical housing solved all problems of sealing axial interfaces and a major problem of distortion caused by thermal gradients from the housing interior to the exterior. These thermal gradients would cause the housing to contract unevenly during chilldown, thereby making the interior surfaces become out-of-round. The housings being discussed are all basically cylindrical with uniform wall thickness and a minimum of ribs or bosses.

b. Design Requirements

These external housings were to be sufficiently strong to withstand all predicted pump loads and provide adequate protection to contain components should a failure occur in the rotating members.

Weight of the housings was of minor importance.

c. Description

Each of these housings is essentially a cast stainless steel heavy walled cylinder with a flange provided at each end. These flanges contain the conical seal seating surfaces. They also serve as the interface to the adjacent member. Through-bolts, washers, and nuts are used as the fastening devices. Ports and passages are provided as necessary for instrumentation and fluid circuits.

d. Stress and Vibration

The following considerations were included in analyzing these housings.

(1) The loads stipulated in the M-1 Design Information Report.

(2) Internal pressures and a stabilizer strut load of 80,000 lb/strut.

In all cases, the maximum stresses result in positive margins of safety when utilizing the material mechanical properties at -423°F. A tabulation of maximum stress values is shown in Table 12.

e. Materials

The housings are made from cast 304 stainless steel. All parts were cast in sand molds. The material was selected because of its suitability for cryogenic use and it is a common casting material. Material strength was of secondary importance. Because there was no weight restriction, casting walls could be as thick as necessary for the predicted loads.

f. Fabrication

These housings were all of simple design. There were no unusual features that would affect either casting or machining. No problems were experienced.

2. <u>Guide Vane Housing</u> (See Figures 38 and 39)

a. Design Philosophy

This housing was complex because of the nature of its multiple functions. The design had to include many passages for coolant flow and instrumentation wires, yet be constructed from castings to be compatible with adjacent housings. The design was guided by foundry techniques and capabilities.

TABLE 12

STRESS OF INLET ELBOW, INDUCER HOUSING, AND MAINSTAGE HOUSING

Component	Condition	M.S.	Stress (psi)
10° Elbow	I	+0.26	34,600
	II	High	2500
Inducer Housing	I	+0.48	29,400
	II	+1.10	20,700
Mainstage Housing	I	--	--
	II	+0.40	31,000

Condition I - D.I.R. Loading

Condition II - Pressure and Strut Loads

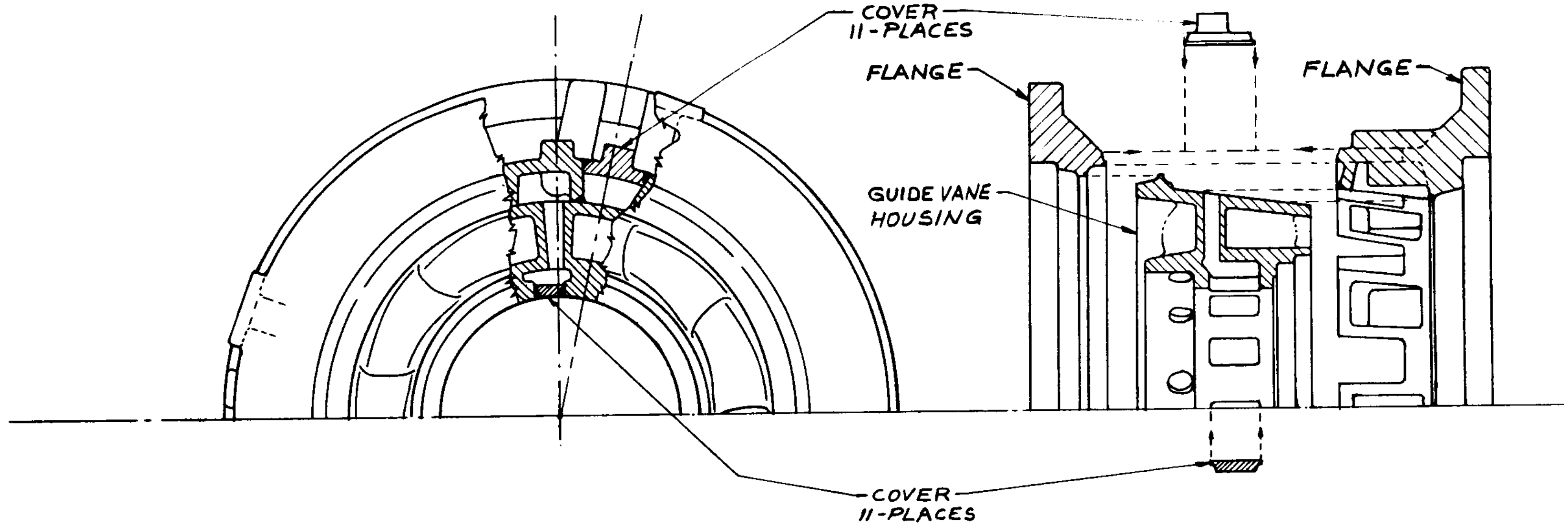


Figure 38. Guide Vane Housing Weldments

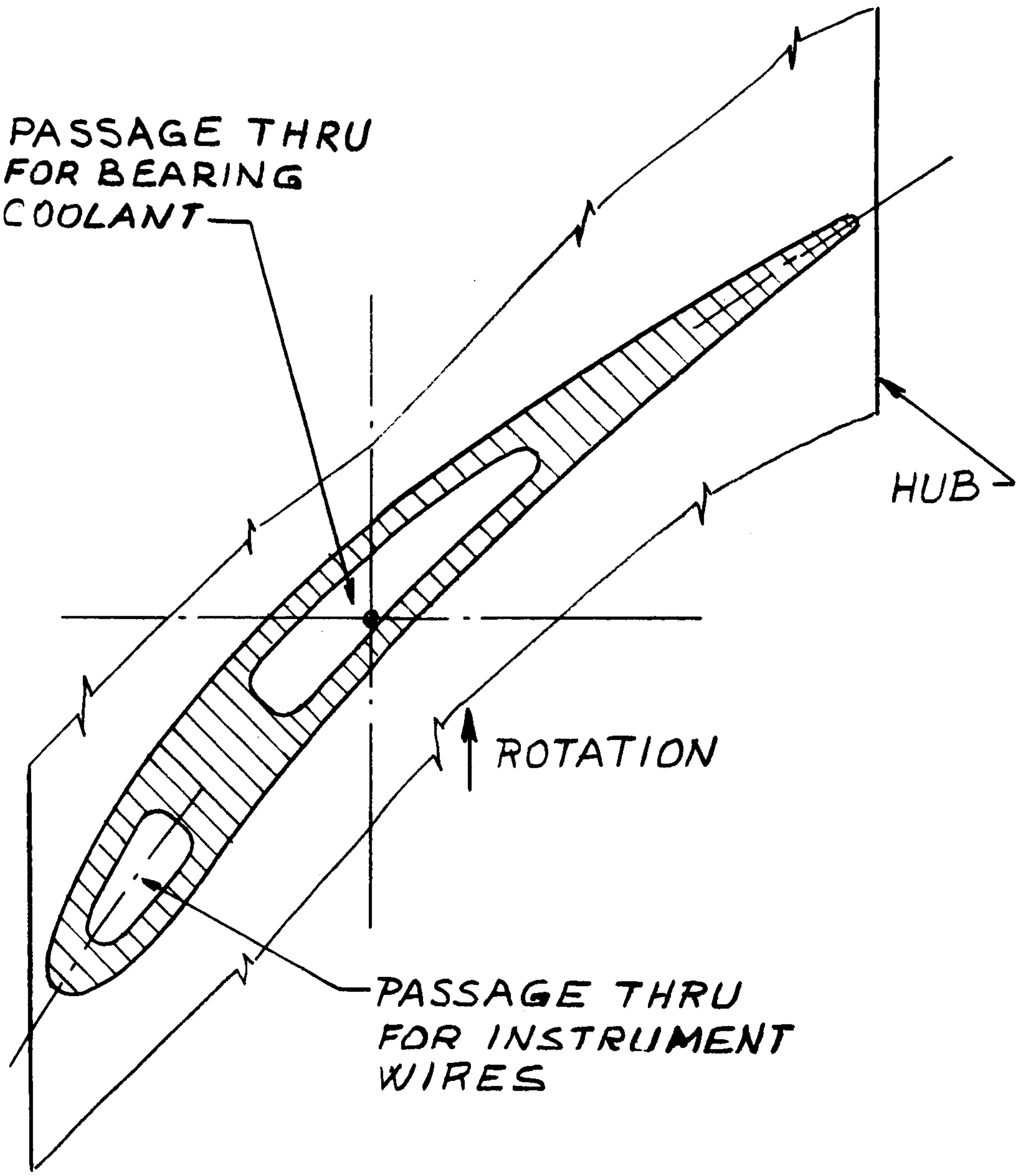


Figure 39. Typical Guide Vane Configuration and Passage Location

b. Design Requirements

The housing had to meet the following criteria.

(1) Be capable of transmitting all thrust and axial bearing loads through the vane system to the housing interfaces.

(2) Have hydraulic passages, including vanes, to properly guide the liquid hydrogen flow from the first-stage inducer to the second-stage inducer.

(3) Provide passages for bearing coolant flow both to and from the bearing package.

(4) Provide passages for the routing of instrumentation wires from the various strain gages, accelerometers, and temperature sensors located in the bearing package.

(5) Provide suitable mounting surfaces and bolt holes to accommodate the pump-end bearing package.

(6) If possible, be capable of either internal or external bearing coolant circuit routing entering from and returning to the mainstage housing.

(7) Be constructed of cast stainless steel to be consistent with all other exterior housings.

c. Description

The guide vane housing is basically a welded assembly consisting of guide vanes, front flange, rear flange, 11 exterior covers, and 11 interior covers.

This construction was necessary so the housing could function properly and the castings, with one exception, could be simple and easy to produce.

The exterior shell is a double-walled structure. This shell has the mounting flanges and all exterior ports for bearing coolant tubes and instrumentation connectors. Within the shell walls are a series of passages, sealed from each other to provide control and direction of bearing coolants. Four passages direct bearing coolant to the bearings. Seven passages conduct the coolant away from the bearings. In addition, there are eleven passages for instrumentation circuitry. One surface of the exterior shell forms the hydraulic passage contour at the tips of the guide vanes.

There are 11 guide vanes that join the exterior shell and the interior shroud. These vanes are generated by a series of C4 airfoils having varying stagger angles from station-to-station to produce a complicated blade form. Each vane has two interior passages. A small passage is located in the leading edge area for the routing of instrumentation wires. A large passage is located about the stacking axis of the vane for bearing coolant flow. Four vanes pass coolant to the bearings and seven vanes carry coolant away from the bearings.

The interior shroud is an integral part of the guide vanes, forming the hydraulic passage contour at the vane root. This shell also provides all the necessary interface surfaces, passages, and bolt holes for the pump-end bearing package.

d. Stress, Guide Vane Housing

The guide vane housing, though complicated structurally, did not present any stress problems. The 11 vanes, which transmit both radial and thrust loads from the pump-end bearings, have a maximum stress of 23,600 psi, which gives a margin of safety of +0.70.

The exterior portion of the housing, including the mounting flanges, was treated in the same manner as the 10 degree elbow, inducer housing, and mainstage housing. The maximum stress is 28,000 psi. The margin of safety is +0.25.

e. Materials

The components for the housing are cast from 347 stainless steel with all castings being made in sand molds. The 347 stainless steel was selected because of its corrosion resistance in the as-welded condition. This feature was considered of primary importance because of the large amount of welding required to complete the assembly prior to any machining. Castability of the material was compromised to obtain the as-welded corrosion resistance.

f. Fabrication

(1) Consideration in Design

Foundry practice was a primary consideration in the design of this part. One part, the guide vanes, was complicated and difficult to cast. The design of the guide vane casting was made final after obtaining advice and guidance from Aerojet-General foundry personnel.

The guide vane housing was broken down into individual castings. This was done to use good casting design practice, such as uniform wall thickness wherever possible, adequate draft angles, no sudden change in cross-section thickness, and provide adequate fillets.

(2) Actual Problems Encountered

The casting of the guide vanes was a major problem because of the complex vane shape combined with cored passages through these vanes. Specifically, it was these passages that created the problems. Because of the high pouring temperature of 347 stainless steel, the passage cores became too hot and failed. Experimentation with various core sands, binders, and core coatings produced a successful set of cores that withstood both the heat and erosion effects of the molten 347 material. The problems involved in producing a usable casting were solved in the Aerojet-General foundry, which proved to be the only source of acceptable guide vane castings.

A second problem was experienced during welding of the cast flanges to the cast guide vanes. These welds, because of their size, required large amounts of filler wire. This produced excessive distortion of the first unit as a result of weld shrinkage. The weld joint configuration and weld build-up procedures were revised. Though some weld shrinkage was experienced in the second unit, it was within acceptable limits.

g. Special Tooling Required

No special tooling was required for machining operations. The patterns for the castings are representative of common foundry practice. The patterns for the guide vane casting, though complex, are not an uncommon or significantly different design. The special tooling designation typifies the core materials, coatings, and core manufacture techniques developed to successfully produce the guide vane casting.

3. Discharge Housing (See Figures 40 and 41)

a. Design Philosophy

The design of this housing was based upon the cylindrical housing reasoning previously presented. The volute, which is wrapped around a basic cylinder, is of constant velocity design. A primary consideration was to keep the complete housing envelope to a minimum in consideration of an eventual flight-weight housing wherein minimum size would result in minimum weight.

b. Design Requirements

The discharge housing had to fulfill the following requirements.

(1) Collect the liquid hydrogen flow behind the eighth mainstage and turn the flow so that it would be discharged from the pump in a plane normal to the axis of rotation.

(2) Diffuse the flow from a velocity of 460 ft/sec to 250 ft/sec.

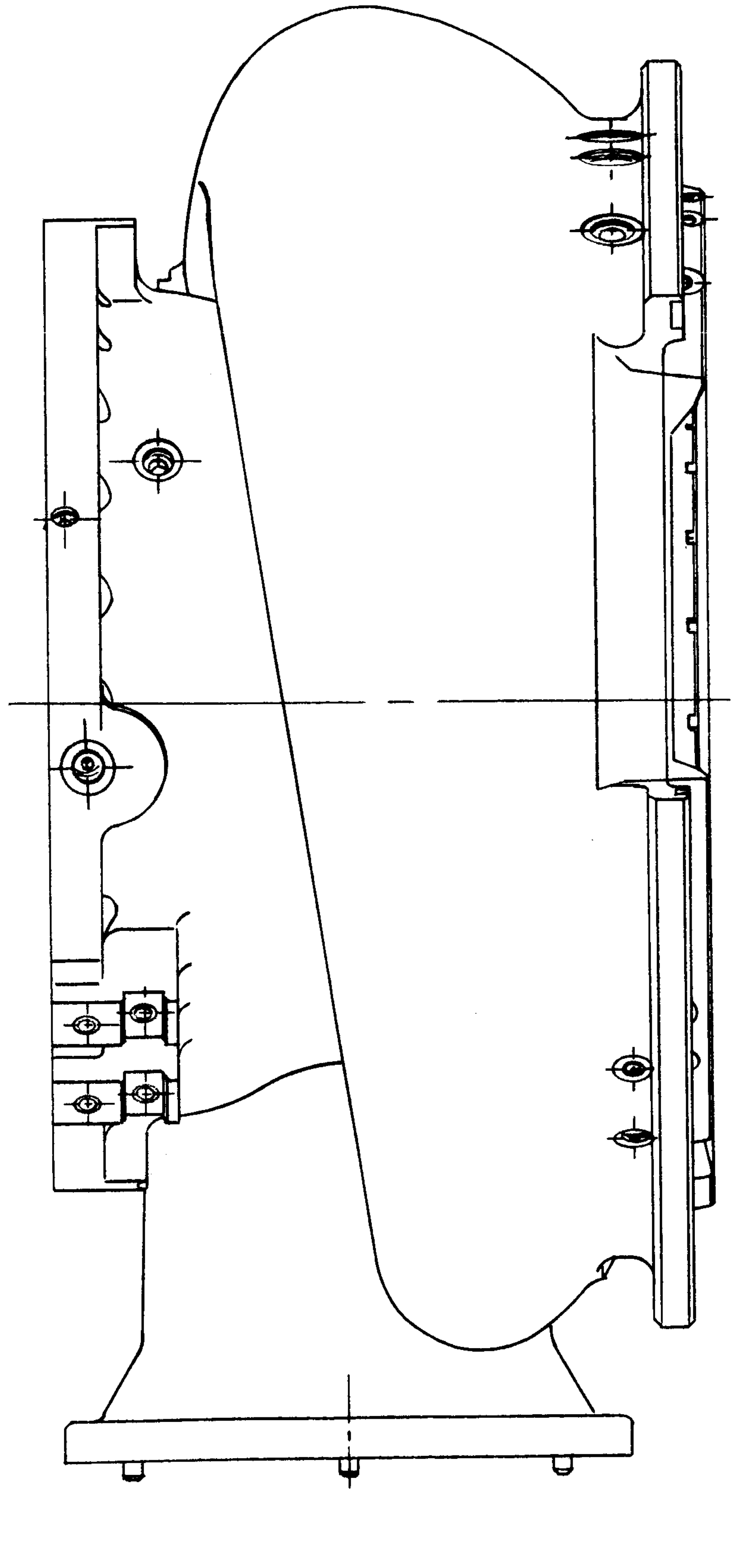


Figure 40. Discharge Housing

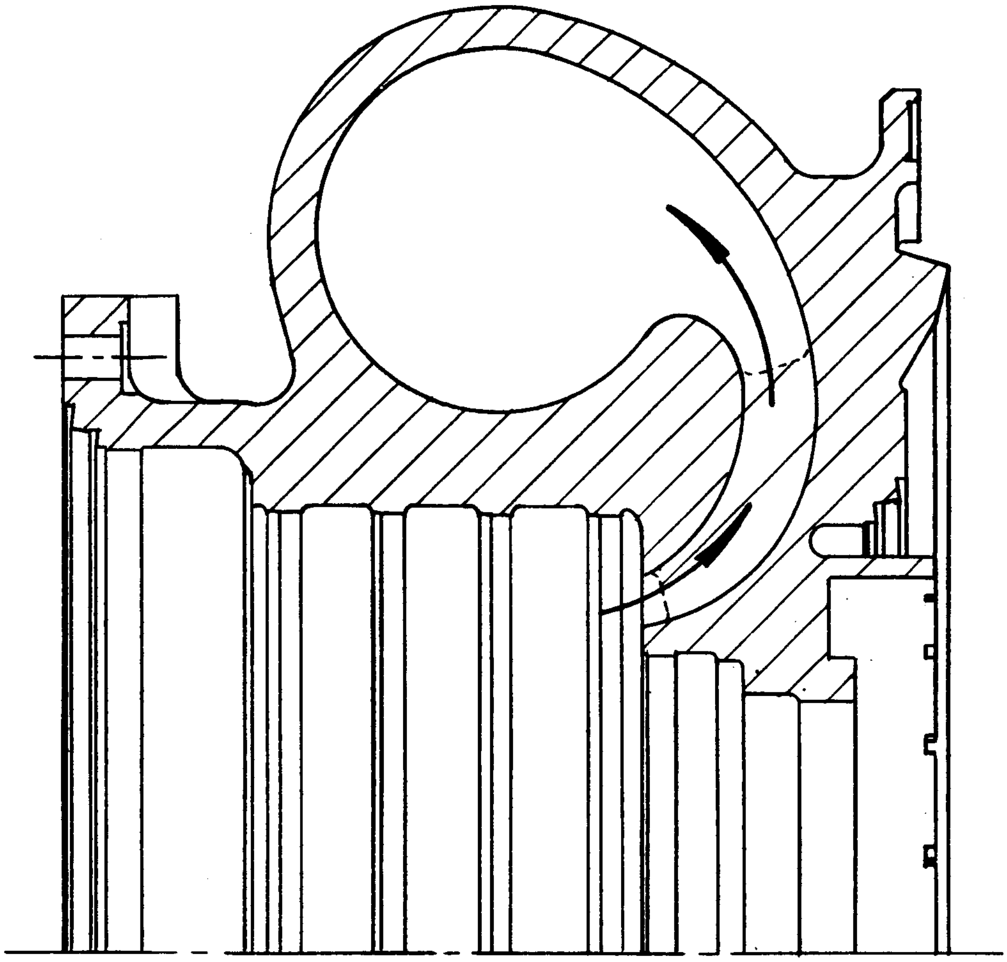


Figure 41. Discharge Housing Propellant Flow Passage

(3) The diffuser vanes must act as structural members and be capable of carrying not only those loads created by fluid pressures within the housing but also loads from external sources such as the turbine assembly, radial bearings, thrust balancing system, and dynamic loading caused by acceleration, gimbaling, etc.

(4) The housing is a casting.

c. Description

The discharge housing is a large 304 stainless steel casting. The main flow passage starts at the center of the part with a set of 20 diffuser vanes. These vanes are placed in a passage that turns the fluid from the axial direction into a plane normal to the axis of rotation. At the trailing edge of the diffuser vanes is an annular collector that serves to equalize the flow before it enters the volute. The volute collects the flow and directs it into the discharge line. This volute extends for a full 360 degrees around the housing before entering the final discharge section. This required the volute to wrap over itself. This wrap-over was handled within the smallest possible envelope size by moving the volute section forward towards the suction end of the pump.

The discharge housing provides mounting surfaces for the turbine bearing housing, turbine frame, pump mounting brackets, and the mainstage housing.

Flow passages are provided for turbine bearing coolant flow and balance disc high and low pressure flows.

Instrumentation, in the form of static pressure taps, is provided for both the main liquid hydrogen flow and the high pressure side of the balance disc.

d. Stress and Vibration

The housing sections, flanges, and volute walls were found to be satisfactory in all respects. The maximum stress in these areas is 30,000 psi. The resulting margin of safety is +0.33.

The set of 20 diffuser vanes in the discharge housing created a problem. In arriving at the final vane configuration, hydro-dynamic requirements were compromised to achieve a vane capable of withstanding the loads imposed by both hydraulic pressure and external forces caused by the turbine-end components. Analysis of these vanes was difficult because of the vane geometry. These vanes were simulated by single beams and double beams. Various influence factors were applied. In all instances, the assumptions made were aimed at producing a conservative stress analysis. The stress values vary from point-to-point along the vane. Margins of safety range from "high" to -0.23 at local points. It is indicated from the stress analysis that negative margins of safety will result in local yielding but will not jeopardize the housing integrity.

To confirm the stress analysis, the first housing to be completed was instrumented and hydrostatically proof tested. Instrumentation consisted of strain gages placed as close to critical areas as accessibility would permit. The housing was successfully subjected to full proof pressure with no evidence of distortion or cracking in any area. The maximum measured vane stress was 28,000 psi while the calculated stress for the same area was 34,400 psi. The maximum measured volute wall stress was 32,000 psi. The theoretical maximum stress was 30,000 psi. The proof test confirmed the structural integrity of the discharge housing. A second housing was subjected to a noninstrumented proof test and housing passed the test satisfactorily in all respects.

e. Materials

The discharge housing is a sand casting made from 304 stainless steel ELC (extra-low carbon). The ELC requirement facilitates the welding required for repair of defects and the closing of openings that are required in the volute areas for core support.

f. Fabrication

(1) Consideration in Design

Being a large and complicated casting, it was necessary at the outset to consider foundry practices and limitation and personnel from various foundries were contacted. The design was revised based upon the information from these discussions. Also, the use of 304 ELC resulted from these discussions as were numerous casting design details.

(2) Actual Problems Encountered

The major problem was the casting of 20 diffuser vanes. Many failures were experienced in this area because of core breakdown from heat, erosion, and fluid pressures of the liquid metal. Various core sand mixtures were tried and several castings were scrapped because of core failure before the problem was resolved.

The vendor was eventually able to cast parts that were acceptable only after a great deal of grinding was done in the vane areas. The final core materials used were able to withstand the heat and pressures, but the eroding effects of the molten metal as it filled the cavities caused unacceptable irregularities in the hydraulic passage areas.

Machining and welding of these housings presented no problems.

4. Second-Stage Inducer Stator (See Figure 42)

a. Design Philosophy

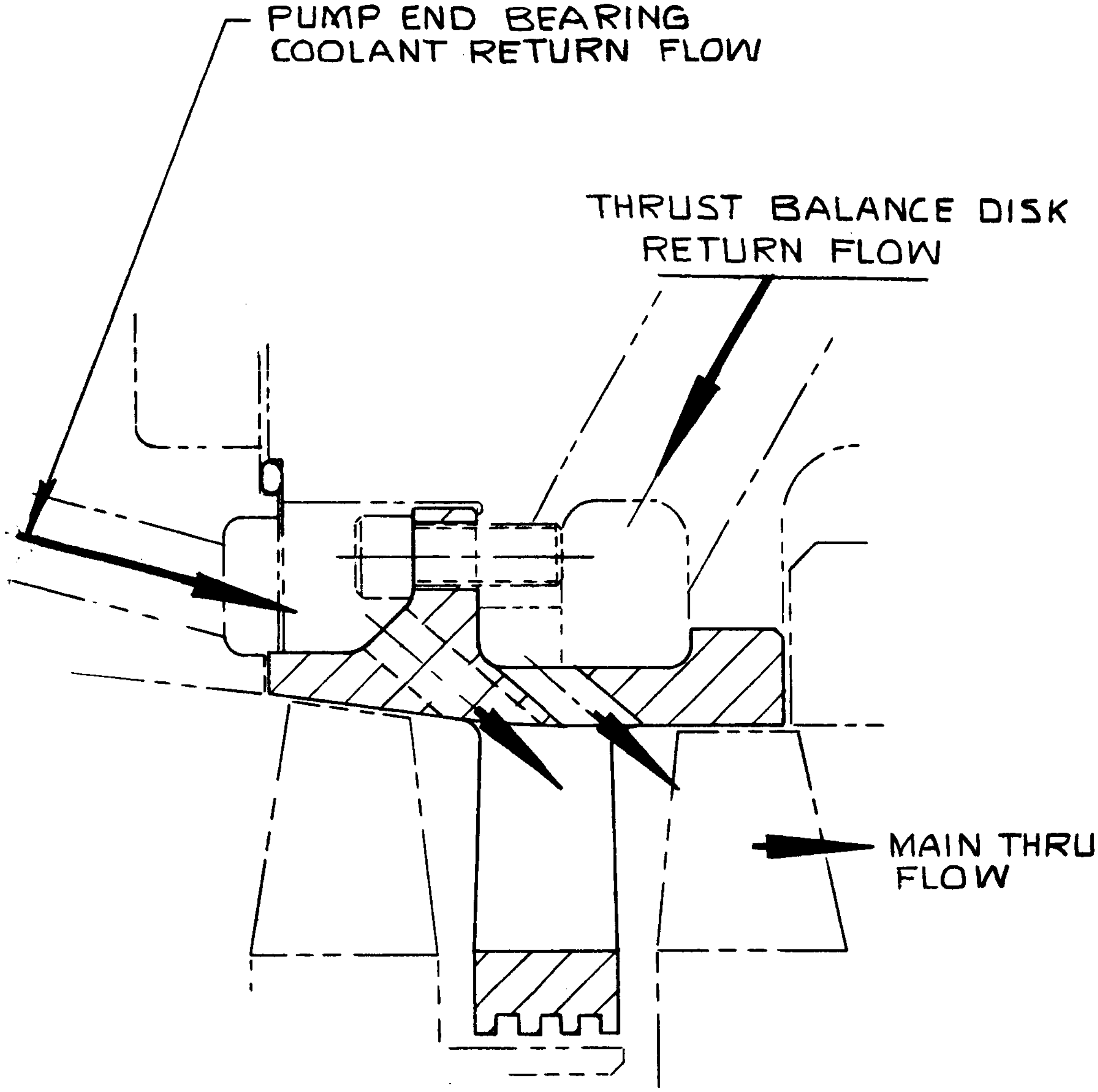


Figure 42. Second-Stage Inducer Stator Return Flow Passages

This part was to be a full ring with an integral outer shroud, vanes, and an inner shroud. The position of this stator in the pump permitted use of a ring design. The ring also simplified or eliminated all sealing problems associated with flow re-entry ports.

b. Design Requirements

This stator had to perform its hydrodynamic function and also provide passages for the re-entry of the pump-end bearing coolant and thrust balance system flow into the main flow system.

c. Description

The second-stage stator is an integral unit having the following features:

(1) An outer shroud, which serves several functions. It is the mounting device and has the necessary flange face, bolt holes, and piloting diameter. It also has two sets of ports which direct re-entering flows through the shroud and into the passages formed by the vanes.

(2) Forty-five vanes having a C-4 airfoil section.

(3) An inner shroud that not only restrains the vanes but also controls recirculating flow between stator entry and discharge.

d. Stress and Vibration

This stator, as designed, proved adequate for stress loads under maximum loading conditions. The maximum stress is 148,000 psi, including safety factors. The margin of safety is +0.63. When checked for fatigue, the margin of safety becomes +0.36 at the critical location. The point of highest stress occurs at the intersection of the vane leading edge and the outer ring.

Vibration analysis revealed the blades to be critical in the first flexural mode with the second and third harmonics of the second-stage inducer blade passing frequency as the excitation source. The pump speeds for these two points are 10,200 rpm and 6900 rpm, respectively. Both speeds are well removed from the design operating speed band and both should be passed through quickly during the start transient and the shutdown phases of pump operation. The results of the vibration analysis are tabulated in Figure 43.

e. Material

The stator is machined from an Inconel 718 ring forging. The material for the final part is solution-annealed and aged for maximum mechanical properties. This material was selected for its strength and because it has the same coefficient of thermal contraction as the adjacent rotating components.

MODE	TEST FREQ (CPS) AT 70°F	DESIGN FREQ (CPS) AT -420°F
1ST FLAT FLEX	10,750	10,750 *
2ND FLAT FLEX	31,050	31,050 *
1ST TORSIONAL	6,570	6,570 *
2ND TORSIONAL	8,660	8,660 *

* VIRTUAL MASS EFFECT OFFSETS MODULUS OF ELASTICITY INCREASE AT CRYOGENIC TEMP.

MODE	FREQ (CPS)	HARMONIC OF EXCITATION	RESONANT SPEED (RPM)	CALCULATED RESONANT STRESS KSI	M.S.	RECOMM MINIMUM CLOSENESS TO RESONANT SPEED (RPM)
1ST FLAT FLEX	10,750	2	10,200	167.0	-.69	±135
		3	6,900	63.4	-.20	±90
1ST TORS	6,570	1	12,800	20.5	+.48	±254
		2	6,300	6.2	+3.9	—
		3	4,200	3.6	+7.4	—
2ND TORS	8,660	2	8,300	1.8	+16.3	—
		3	5,500	1.0	+29.4	—

EXCITATION = 31 UPSTREAM TRANSITION ROTOR BLADES × RPM × 1, 2 OR 3

Figure 43. Second-Stage Inducer Stator

f. Fabrication

The initial layouts of the vane airfoils and the relative position of the adjacent blades revealed that the machining of the integral part was not only feasible but also economical. The narrow width of the shrouds plus easy access to all vane surfaces further confirmed the ease of machining an integral unit.

No problems were encountered in any phase of producing these stator rings.

5. <u>Mainstage Stator Blades</u> (See Figure 44)

a. Design Philosophy

The assembly method required for installation of the stator blades eliminated the use of continuous blade rings. Therefore, the stator blades had to either be grouped into segments or be individual blades. The selection was dependent upon stress and vibration studies.

b. Design Requirements

The stator blades had to perform their hydrodynamic function in the system. They also had to be capable of relatively easy installation and removal.

c. Description

Each mainstage stator blade row consists of 57 individual blades, which are held in position between continuous retaining rings. Each stator blade has a portion of the outer shroud containing the mounting lugs and a section of the inner shroud having labyrinth grooves.

The airfoils used in the mainstage stators are a highly modified C-4. The original 10% thick airfoil proved to be inadequate with respect to stress and vibration; therefore, its thickness was increased to 15% by moving the 10% airfoil high and low pressure surfaces further apart and rounding the leading and trailing edges to make a smooth contour. Hydrodynamic analysis showed only 2% to 3% loss with this airfoil modification.

d. Stress and Vibration

The stator blades installed in fuel turbopump assembly Model I S/N 001 are the direct result of stress analysis. The airfoils were originally of a C4 configuration, 10% thick. From the stress analysis these blades were shown to be completely inadequate. It was indicated from the analysis that a 15% thick airfoil would satisfy all loading conditions. The airfoil shape was compromised to prevent hardware delay. The blades are fully cantilevered to facilitate manufacture. The maximum mean stress in the stator blades is 75,000 psi. The minimum margin of

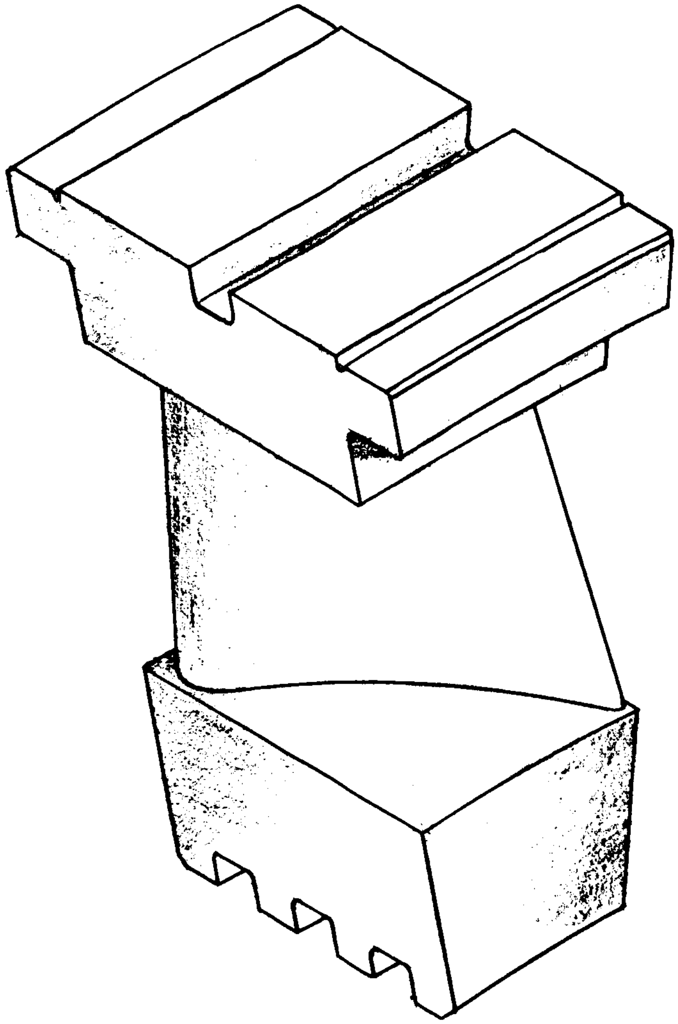


Figure 44. Stator Blade - Mainstage

safety, +0.31, occurs when considering material fatigue. Thirty percent of the maximum mean stress was considered to be alternating stress. This number was applied to a Goodman diagram to obtain the allowable stress of 123,000 psi and the margin of safety. A safety factor of 1.25 was included in the calculation of the margin of safety.

A problem that was revealed by the blade vibration analysis was the second flatwise flexural mode. The margin of safety at this point is -0.21. The pump speed is 7500 rpm, which is considerably below the design speed operating band. All other vibration modes show large positive margins of safety. A tabulation of the vibration analysis results is shown as Table 13.

e. Material

The mainstage stator blades are machined from individual Inconel 718 forgings. These forgings are solution annealed and aged for maximum material mechanical properties. The material, which was selected for its strength, is identical to adjacent rotating parts, thereby eliminating problems of differential thermal contraction.

f. Fabrication

(1) Consideration in Design

Because the blades were individual, they presented no manufacturing problem. These blades could be produced by any one of several common blade manufacturing methods.

(2) Actual Problems Encountered

The airfoil surfaces were produced by electrochemical milling (ECM). During this ECM process, a large number of the blades were found to have pits or small cavities on the surface. Materials tests disclosed the problem to be caused by material segregation within the individual forgings. This material segregation would not have produced any surface effect under conventional machining operations, but it did cause a nonuniform rate of material removal in the ECM process, which resulted in the pitted surfaces.

6. Mainstage Stator Retaining Rings (See Figure 45)

a. Design Philosophy

The mainstage stator assembly was to be composed of circular units rather than axially split housings. The circular units, or rings, were preferred because they would not be subject to the high degree of thermal distortion that could be expected with split rings or housings. Keeping the mainstage stator section as round as possible was essential to operating the pump with blade tip clearances as small as 0.020-in.

TABLE 13

MAINSTAGE STATOR BLADE

Mode	Test Frequency (cps) at 70°F	Design Frequency (cps) at -420°F
1st Flat Flex	722	720 **
2nd Flat Flex	5953	5950 **
1st Torsional	2792	2790 **
2nd Torsional	8370 *	8370 **

* Predicted from first torsional ~ 2π (2792) = 8370.
** Virtual mass effect offsets modulus of elasticity increase at cryogenic temperature.

Mode	Freq. (cps)	Harmonic of Excitation	Resonant Speed (rpm)	Calculated Resonant Stress (ksi)	M. S.	Recomm. Min. Closeness To Reson. Speed (rpm)
1st Torsional	2792	1	3500	10.4	+ 2.8	
		1	10,600	9.6	+ 2.2	
2nd Torsional	8370					
		2	5400	2.0	+14.0	
		1	7500	56.0	- 0.2	±173
2nd Flat Flex	5953					
		2	3800	15.3	+ 1.4	± 86

Excitation: 47 upstream mainstage rotor vanes x rpm x 1, 2, or 3.

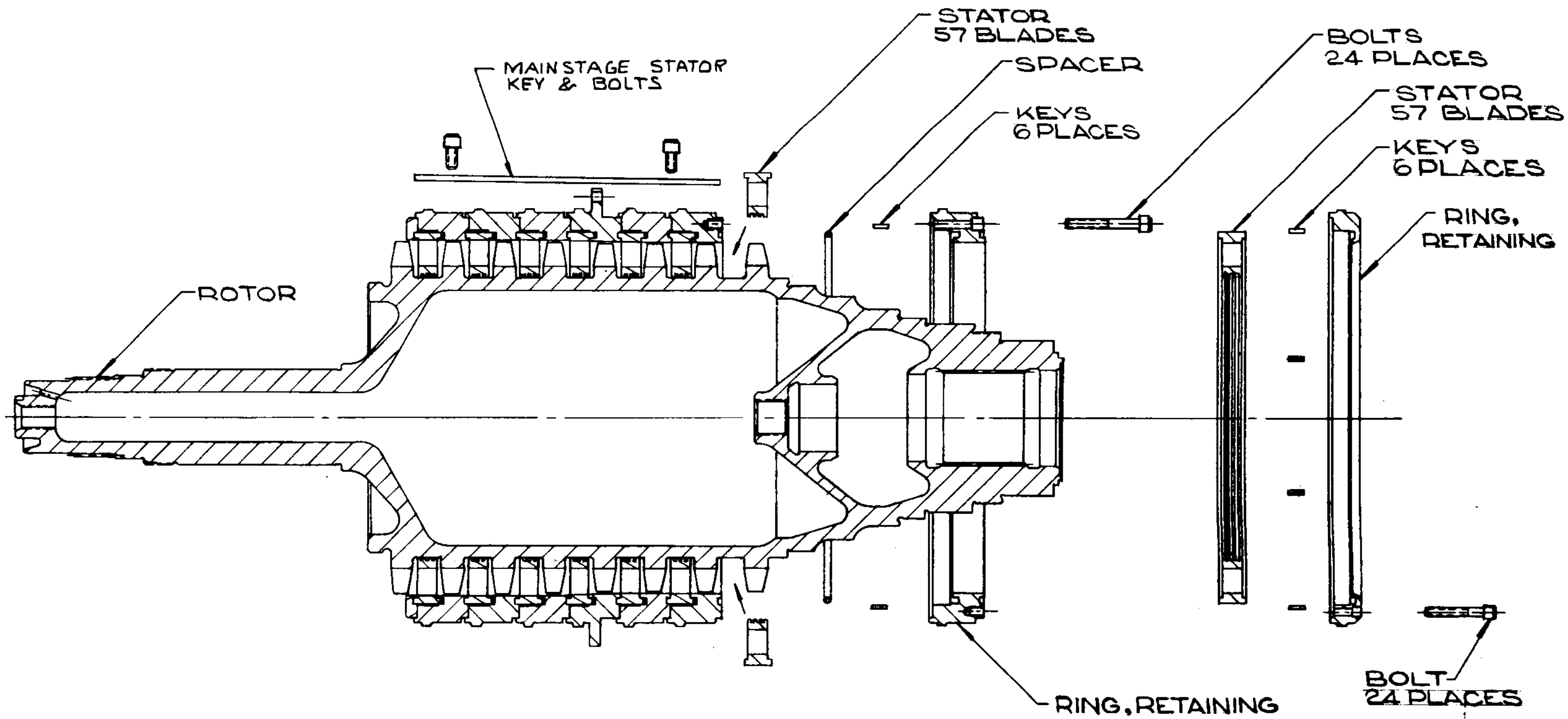


Figure 45. Mainstage Stator Assembly

b. Design Requirements

The stator retaining rings must perform the following functions:

(1) Remain round under all conditions.

(2) Position the stator blades accurately between the rotor blade rows.

(3) Hold the stator blades securely under all loading conditions including vibration.

(4) Provide access, where required, for interstage instrumentation.

c. Description

Each stator blade retaining ring is a full ring; there are no splits or joints. The ring has the necessary grooves and key slots to accept the stator blades. Where necessary, instrumentation ports extend radially through the ring. The exterior diameter has both a close tolerance band, which is the interface to the exterior housing, and a keyway, which orients the stator assembly to the external housings.

d. Stress and Vibration

The retaining rings were designed with no consideration given to minimum weight. The rings were made heavy to assure a minimum of distortion during manufacturing as well as during pump operation. The maximum stress in the rings is 26,400 psi; the margin of safety is +3.64.

e. Materials

Each stator retaining ring is machined from an Inconel 718 ring forging, which is solution-annealed and aged prior to the final machining operation. The material was selected to provide adequate strength in the blade groove areas. It is the same as that used for all rotating components, thus eliminating the problems caused by different coefficients of thermal contraction.

f. Fabrication

The success of the whole ring concept depended upon the ability to machine those dimensions that control blade positions to very close tolerances. Machining practices, methods, and readily-available tooling were primary considerations in the stator ring design. Diameters and their concentricities presented no great problem because most of the machining could be done with a single lathe set-up. The axial stacking dimension created a design problem because a configuration was required that could be accurately and easily manufactured.

This was solved by designing the part so that one of the two ring stacking surfaces was completely exposed for surface grinding. Thus, one plane was machined as part of a lathe operation and the ring was then set up in a surface grinder on this finished plane and the second plane was ground to dimension.

No difficulties were experienced in manufacturing any of the retaining rings. The accuracy of ring manufacture was reflected in the relative ease and the accuracy of the actual mainstage stator assembly. No special tools were required for the ring manufacture.

7. Thrust Balance Flow System (See Figures 46, 47, and 48)

a. Design Philosophy

The performance of the return flow system was questionable. Therefore, it was necessary to design this system so that it could be readily modified. Provision had to be made for controlling flow.

The design of the flow system was to be as simple as possible. It was also to be capable of modification while the fuel turbopump assembly was mounted in the test stand, to permit the continuation of testing programs with minimum or no delays.

b. Design Requirements

The return flow system had to meet the following conditions:

(1) Carry the predicted thrust balance flow.

(2) Have adequate instrumentation to determine the liquid hydrogen condition in the system.

(3) Have a method for varying the flow through the system.

(4) Be capable of withstanding fluid pressures up to 1000 psi.

c. Description

The thrust balance return system consists of two identical circuits. Each circuit contains the following components.

(1) Four 3/4-in. diameter tubes that carry the flow from outlet ports in the discharge housing to the instrumentation manifold.

(2) An instrumentation manifold that is composed of two sections. This unit has total pressure, static pressure, and temperature instrumentation. There is a replaceable orifice plate for flow control. This orifice plate can be removed simply by disassembling the manifold flange joint.

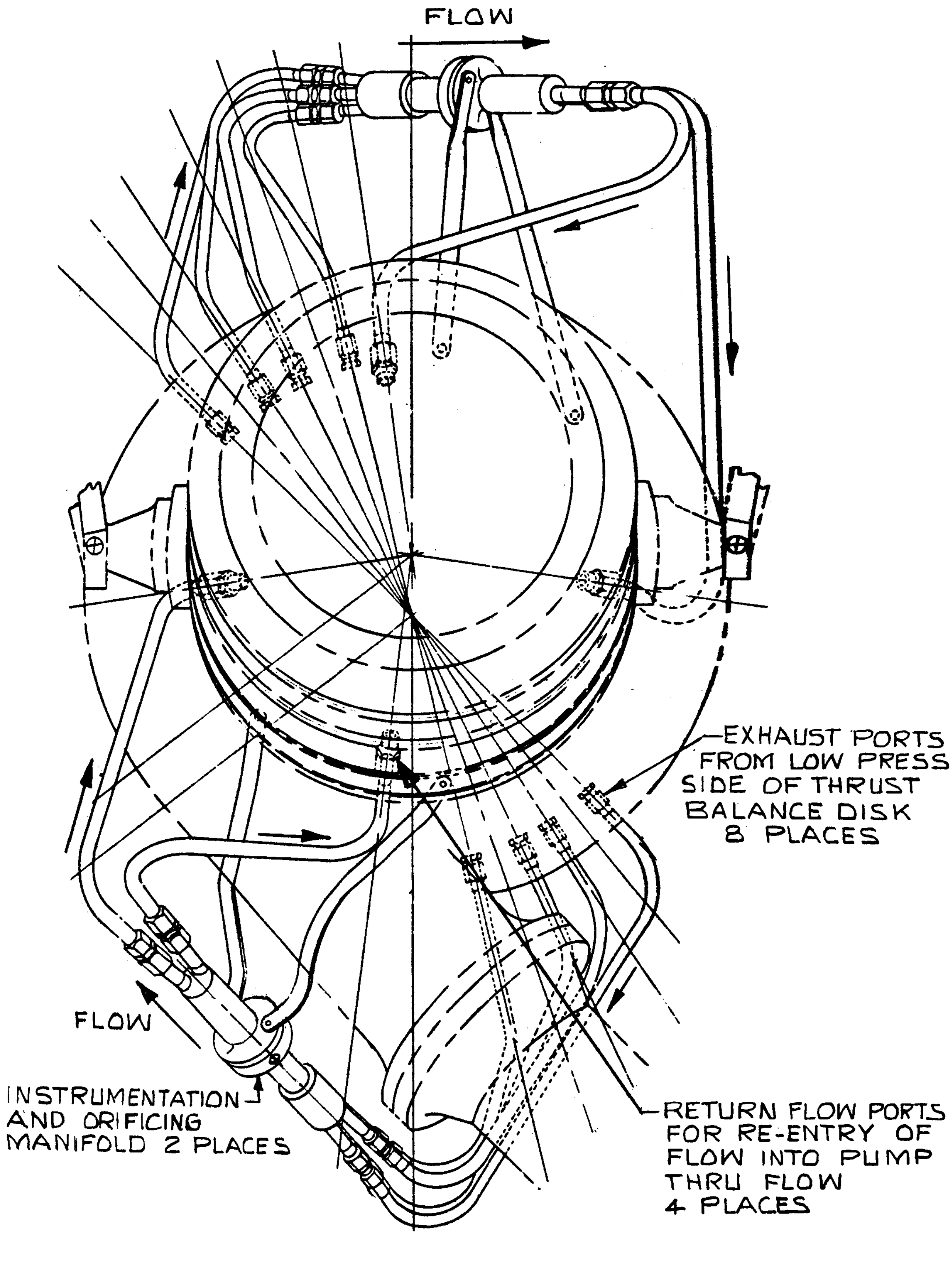


Figure 46. Manifold Thrust Balance Return Flow

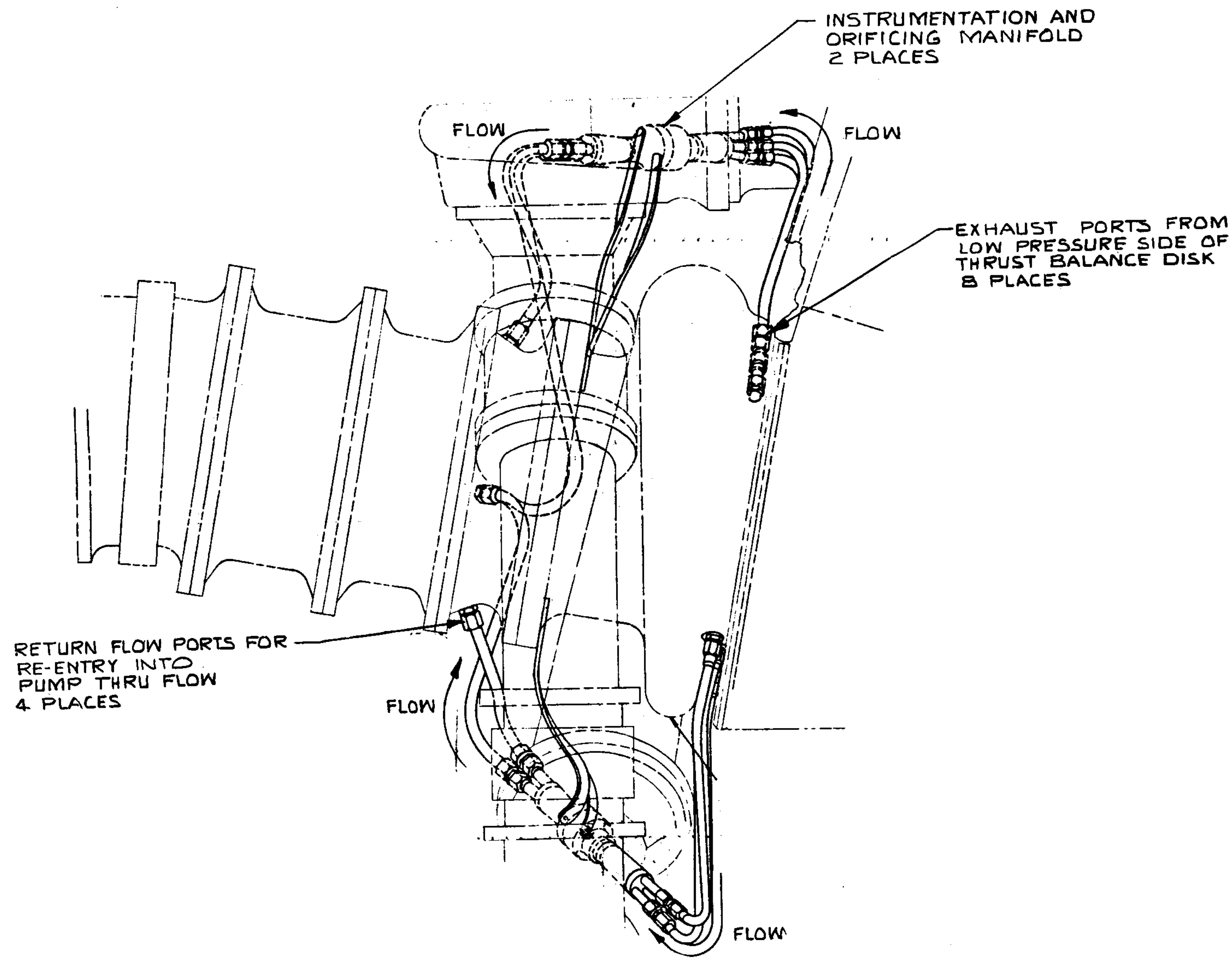


Figure 47. Manifold Thrust Balance Return Flow

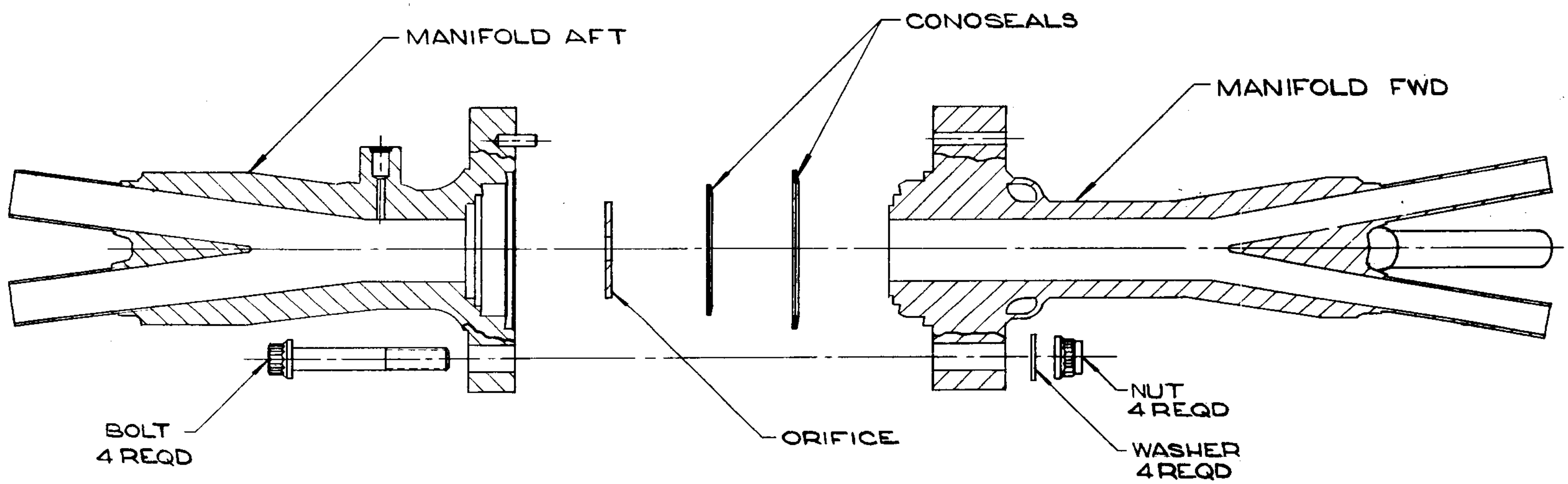


Figure 48. Manifold Thrust Balance Return Flow Orifice and Instrumentation

(3) Two 1.00-in. tubes that carry the flow from the instrumentation manifold to the inlet ports in the mainstage housing.

The double system was used to keep tube lengths to a minimum and because the double outlets from the balance piston reduced pressure variations within the low pressure area.

The return flow system is supported by several brackets. Tubes are supported at various points to dampen vibration. The entire flow return system is insulated with fiberglass blankets and an exterior wrapping of reflective tape.

d. Stress and Vibration

In this system, the tubes were subjected to the highest stresses. It was indicated from an analysis that the highest tube stress was 6750 psi, which is well below the material yield stress of 35,000 psi.

System vibration problems were removed by providing additional straps between the tube groups and the pump housings.

e. Materials

All of the return flow system components are made from 347 stainless steel. This material provided adequate strength, was weldable without weld corrosion problems, and was readily-available. The coefficient of thermal contraction is also compatible with that of the cast 304 stainless steel housings, to which the return flow system is attached.

f. Fabrication

The fabrication of this unit was not complicated. The instrumentation manifold design and its use of weldments were based upon the most economical manufacturing methods consistent with the function of the finished item.

C. BEARINGS

1. Pump End Bearings

a. Design Philosophy

The pump end bearing package was to contain both a radial bearing and a triplet tandem ball bearing. The design of the bearing mountings was to ensure that all radial loads were carried by the roller bearing thus assuring maximum thrust capacity of the ball bearings and the maintaining of bearing load paths and spring rates per those used in the critical speed analysis.

This bearing package must contain sufficient instrumentation to adequately monitor bearing temperatures and vibrations, and measure the thrust loads carried by the thrust bearings.

Axial deflections of the thrust bearing mounting system had to be compatible with those required by the thrust balancing system, specifically the axial gap flow restriction of the thrust disc.

b. Design Requirements

The bearing package had to meet the following criteria:

(1) The thrust bearings are to be mounted in a radially flexible housing, thereby reducing the radial load carrying ability to a minimum.

(2) The thrust bearing housing must be capable of transmitting a continuous load of 35,000 lb and short duration loads of 50,000 lb.

(3) The thrust bearing mounting is to be designed for a normal operating thrust load acting toward the turbine end, but the unit must be capable of a complete thrust reversal of the same magnitudes as noted in (2) above.

(4) The thrust measuring instrumentation must be capable of reading thrust loads acting in either axial direction.

c. Description

The primary member of the bearing package is the exterior housing. This housing not only mates to the pump guide vane housing but also is the mount for the radial bearing support and the radially flexible thrust bearing support.

The radial, roller, bearing is carried in a simple circular support plate. This plate bolts to the pump side of the exterior bearing housing. The radial bearing support contains necessary mounting holes for two temperature sensors and two accelerometers.

The thrust bearing consists of a matched set of three split inner race ball bearings. These bearings have been manufactured as a set, including the coolant spray rings which act as spacers between the outer bearing races, to assure load sharing. These thrust bearings are contained in an inner cylindrical housing which has mounting surfaces for the necessary instrumentation. Each bearing is instrumented with two temperature sensors and two accelerometers.

The inner cylindrical bearing housing with its instrumentation is attached to a radially flexible housing. Flexibility is achieved by slotting the area between the flanged ends of the housing so that this area consists of 72 bars, each bar having a square cross-section of 0.150-in. per side. The purpose of the flexible housing is to limit the radial load capability of the thrust bearings to a maximum of 200 lb.

The inner bearing housing is attached to a radially flexible housing. This interface is designed to permit controlled axial movement of the rotating system as required for proper operation of the thrust balancing disc. Control of axial movement is achieved by incorporating 36 stacks of belleville washers; each stack contains 40 washers.

The spring control operates only when the net unbalanced pump thrust is acting toward the turbine. When the pump thrust is acting toward the pump inlet, the spring system is inoperative and thrust loads are transmitted through this housing interface by the face-to-face contact of shoulders on both the inner bearing housing and the flexible housing.

Total spring controlled axial movement is limited by a stop to prevent rotating components from striking stationary components.

The thrust measuring instrumentation is mounted on the bars of the radially flexible housing. This instrumentation consists of a strain gage network which incorporates temperature compensation and has gages located to eliminate strains caused by bending. The net result is a system which reads strain resulting from thrust loads only. One feature of this strain gage system is that each strain gage has its wires routed to the exterior of the pump. The network circuitry is completed outside the pump. This permits circuit revision without pump disassembly in the event a strain gage or gages should become inoperative.

The radially flexible housing is bolted to the exterior bearing housing.

Shaft riding seals are installed on each end of the bearing package. These seals provide leakage control of the bearing coolant from the high pressure bearing package interior to the surrounding pump cavities.

Bearing coolant passes through the housings to the spray rings by means of a series of annular grooves and tubes.

d. Stress and Vibration

Stress analysis of the pump end bearing package components revealed all items to be more than adequate for their respective functions. The margins of safety for all bolts are high.

The flexible bearing housing analysis shows this housing to be capable of withstanding loads in excess of 150,000 lb in either tension or compression. Nominal design thrust bearing load is 35,000 lb.

e. Materials

In general, materials were selected for their strength and suitability to liquid hydrogen environment. Inconel 718 was used for all components except the inner bearing housing. This housing was made from K-Monel. This material was selected because the coefficient of thermal expansion most nearly matched that of the 440C bearing races. The matching of expansion coefficients permitted maintaining of desired bearing to housing fit throughout a broad temperature range.

f. Fabrication

All parts are of simple design and offered no unusual manufacturing problems. Tolerances, in some areas, were very small but such areas were easily accessible for grinding thus permitting the desired tolerances to be met.

2. <u>Turbine End Bearing</u>

a. Design Philosophy

The turbine end bearing assembly must provide adequate support for the roller bearing with sufficient housing stiffness to maintain the bearing spring rate at the maximum amount thus preventing a lowering of the calculated critical speed.

The bearing assembly must be compatible with the mating pump discharge housing.

b. Design Requirements

The bearing assembly must satisfy the following requirements:

(1) Maintain maximum bearing spring rate.

(2) Provide adequate instrumentation to monitor bearing operation.

(3) Include a static seal which will prevent leakage of liquid hydrogen into the turbine area during chilldown and zero speed conditions. This seal must open to provide an exhaust passage for bearing coolant during pump operation.

c. Description

The bearing assembly consists of a single 120mm roller bearing mounted in an internal housing. This housing has the necessary bearing instrumentation, two temperature sensors and two accelerometers. A coolant spray ring is located on the pump side of the bearing. Coolant flows through the bearing, through a labyrinth system, and finally into the turbine area.

A shaft riding seal and a labyrinth are installed on the pump side of the coolant spray ring to provide coolant leakage control into the low pressure area of the thrust disc and isolate the bearing cavity from the thrust disc area.

The bearing (with its housing and instruments), the spray ring, the shaft riding seal, and labyrinth are bolted into the exterior turbine bearing housing. This exterior housing is machined from a thick walled casting. The large end joins to the pump discharge housing. The small or turbine end has mounting surfaces for a lift-off seal and a heat shield.

The exterior housing has passages to route bearing coolant which enters the bearing housing after passing from the pump discharge housing through externally mounted filters and venturi flow meters, to the spray ring. Passages are also provided to route instrumentation wires to connectors mounted on the periphery of the large flange. Internal mounting surfaces are provided for two speed probes and the torque meter coil. Drilled passages are provided for the static pressure supply which actuates the lift-off seal.

The lift-off seal assembly, mounted on the turbine end of the exterior bearing housing, consists of a carbon face seal mounted on a bellows. Under ambient pressure, the bellows holds the carbon seal against the face of a rotating ring mounted on the turbine shaft. When pressurized, the bellows length increases moving the carbon seal away from the rotating ring thus providing an annular passage for the exhaust of bearing coolant. The lift-off seal provides the static closure, at the turbine end of the pump, to keep leakage of liquid hydrogen into the turbine area to a minimum.

The lift-off seal assembly is protected by a heat shield. The heat shield has a labyrinth on the inside diameter to act as control for the flow of bearing coolant.

d. Stress and Vibration

Stress analysis of the external bearing housing show a margin of safety of +1.05 under operating conditions. The margin of safety for the bolts which attach this housing to the pump discharge housing is +0.45. In general, all components are satisfactory, stresswise, for the predicted fuel turbopump assembly operating loads.

e. Materials

Materials were selected for their suitability for liquid hydrogen service. The exterior bearing housing is machined from a 304L casting; this material being the same as used for the various pump housings.

The various components such as the coolant spray ring, the bearing housing, lift-off seal assembly, and labyrinths are made from Inconel 718. This material has high mechanical properties and good ductility at liquid hydrogen temperatures.

f. Fabrication

The components of the turbine bearing housing, except the lift-off seal, presented no fabrication problems. These parts were of straight forward design and therefore could be machined by standard methods and without need for special tooling.

The lift-off seal proved to be a real manufacturing problem. The area of difficulty was the bellows. Both inner and outer bellows were to be made initially from Inconel 718. The final assembly had an Inconel 718 outer bellows and a 347 stainless steel inner bellows. This change was requested because of difficulty in welding the formed discs into a completed bellows. At least one of the bellows had to be Inconel 718 in order to act as a spring loading device to keep the carbon seal face sealed against the turbine shaft running ring under static conditions.

Two lift-off seal assemblies failed in the bellows welds during vibration tests and the initial series of fuel turbopump assembly tests. Failure analysis of these seal assemblies revealed the Inconel 718 welds to be of extremely poor quality. These welds displayed a lack of penetration and the presence of inclusions. It was determined that control of the individual bellows discs was inadequate with respect to the weld area. The weld fixturing was inadequate. These two items resulted in a poor and inconsistent joint fit prior to welding. The material was not properly cleaned prior to welding nor was an adequate backup gas purge present during the welding operation. It is apparent from the failure analysis that not only must the welding procedures be carefully controlled in all respects but also the bellows weld joint must be carefully designed to assure proper fit when using Inconel 718.

V. ASSEMBLY TECHNIQUE

The sequence of events required to assemble the fuel turbopump is discussed in this section.

A. PUMP ROTOR AND ROTOR BLADES (See Figure 33)

The rotor blades fit into dove-tail slots machined in the pump rotor. There is one blade per slot. Assembly is accomplished in the following manner.

The retaining pins are fitted by pressure into the pump rotor. The height that each pin projects from the bottom of each slot is gaged.

The blade retaining tab is installed by engaging the hole in the tab to the blade retaining pin. Care must be taken to assure that the tab is installed correctly. This can be checked visually by noting that the portion of the tab extending out of the rotor slot is perpendicular to the circumferential rings in which the rotor slots are machined when looking down on each slot.

The rotor blade is inserted, the leading edge entering the slot, and pushed toward the inducer-end of the rotor until the dove-tail is fully engaged. Both blade and rotor surfaces adjacent to the dove-tail will be flush. Note that blades can be installed only one way because of the position of the retaining pin and the matching slot machined in the blade. Some force is required to push the blade into the slot because the locking tabs are curved, thereby forcing the blades out in a radial direction. This forces the blade into the same contact situation it will experience as a result of centrifugal loads.

Blade installation is completed by bending the lock tab (which is protruding out of the dove-tail at the base of the blade) upward so that it is completely under the blade platform.

After all the blades have been installed, the last machining operation is performed. The blade tips are ground, to proper diameter and concentricity, to satisfy the drawing requirements.

B. MAINSTAGE STATOR BLADE ASSEMBLY (See Figures 45 and 49)

The mainstage stator is assembled around the pump rotor. After balancing, the pump rotor is positioned in the assembly stand by means of suitable fixtures. The pump rotor is in a vertical position with the turbine-end up. These fixtures are also used to position the mainstage stator, as it is assembled, axially and radially to the rotor. Adjusting features are incorporated into the fixtures to establish the correct set dimensions of the mainstage stator blades to the pump rotor in accordance with drawing requirements. After the first stator retaining ring is installed in the fixture, the assembly procedure delineated below is followed.

Each of the 57 stator blades is placed in position in the stator retaining ring groove.

A visual check is made to determine that each blade is installed correctly. This is done by reading the identifying numbers denoting blade position in the ring. Blades are correctly installed when all numbers are visible and are in numerical order reading clockwise.

The six anti-rotation keys are installed.

The crush tubes are installed in the next retaining ring.

The retaining ring is lowered over the rotor and positioned over the stator blades. Care must be exercised to prevent damaging the rotor blades and to assure that all stator blades are properly engaged in the retaining ring groove.

The cap screws are installed, but not tightened to full torque value. Checks are made of retaining ring concentricity and parallelism, with respect to the fixture and the previously-installed retaining ring. When these checks are within acceptable limits, the cap screws are torqued to the full drawing requirement.

A final inspection is made to verify correct concentricity, parallelism, and axial position of the retaining ring relative to the fixture. All dimensions must be within acceptable limits before proceeding with assembly.

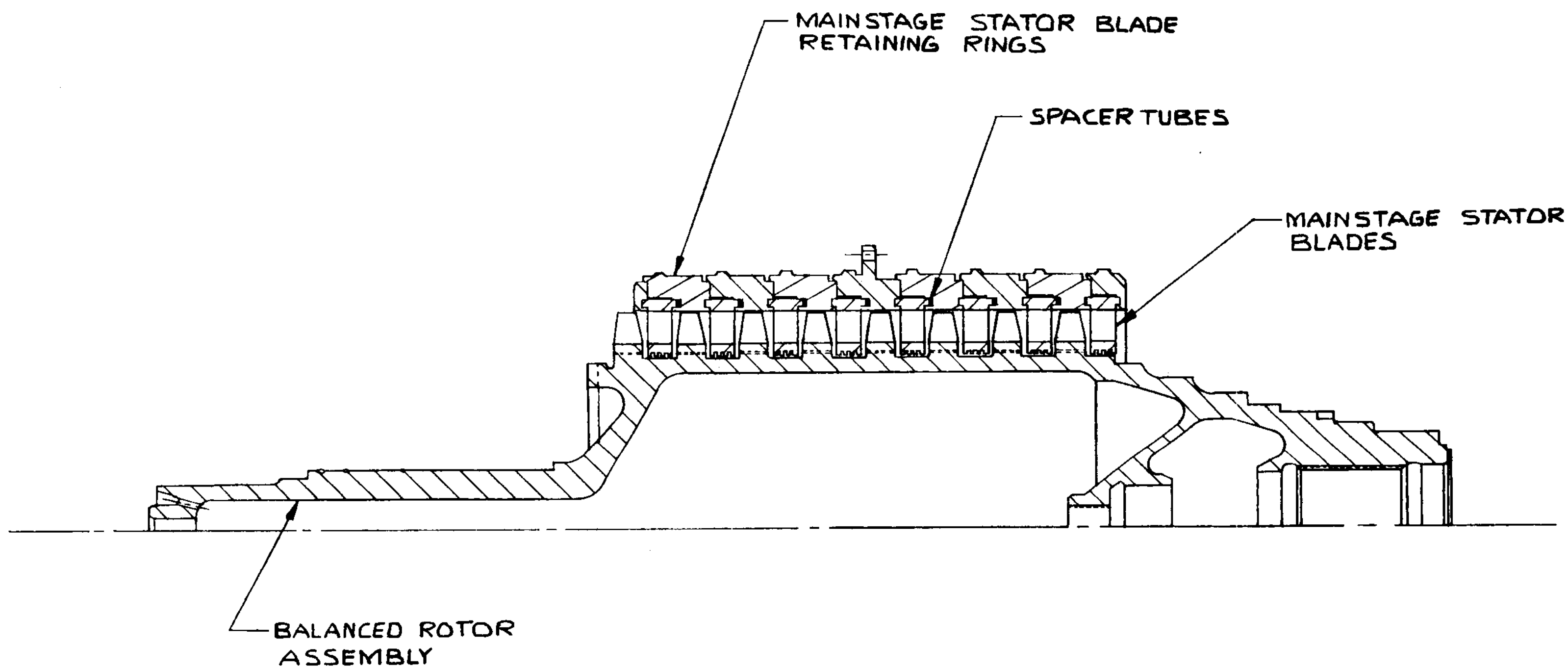


Figure 49. Assembly Sequence, Phase I

All of the above items are repeated for each of the remaining seven mainstage stator blade rows.

After the eighth stator row has been assembled, the external mainstage key is installed.

C. MAINSTAGE HOUSING (See Figure 50)

Before the housing is actually installed, the pump rotor mainstage stator assembly must be refixtured. This is accomplished by installing a fixture, which maintains the pump rotor stator assembly relationship, on the turbine end of the pump rotor. After this fixture installation is completed, the pump-end fixture used for the stator assembly is removed, thereby providing the necessary clearance for installation of the housing.

With the pump rotor position unchanged, turbine-end up, the mainstage housing installation proceeds in the following manner.

The housing is heated to 200°F and then placed in position on an elevator, which is part of the assembly stand.

The housing is raised into position, with care being exercised to correctly align the external mainstage stator assembly key.

The bolts used to attach the mainstage stator assembly to the housing are installed, torqued, and safety-wired.

The second-stage inducer stator is placed in position in the mainstage housing. The necessary bolts are installed, torqued, and safety-wired.

The second-stage inducer is placed in position on the pump rotor and the lock ring and a set of bolts, which have been matched by weight are installed.

D. GUIDE VANE HOUSING AND PUMP-END BEARING ASSEMBLY (See Figure 50)

Installation of these components begins with assembly of the thrust bearings, instrumentation, Belleville washers, flex housing, external bearing housing, one shaft riding seal, and all related components. This work is performed as a subassembly. Note that the roller bearing and inducer-end shaft riding seal are omitted at this time.

After assembly of the thrust bearing subassembly, assembly of the pump proceeds in the following manner.

The bearing subassembly is installed on the pump rotor. Because of interference fits, the bearing package must be heated to 200°F and the pump rotor cooled to liquid nitrogen temperature, thus providing adequate radial clearance to complete the assembly without the use of force.

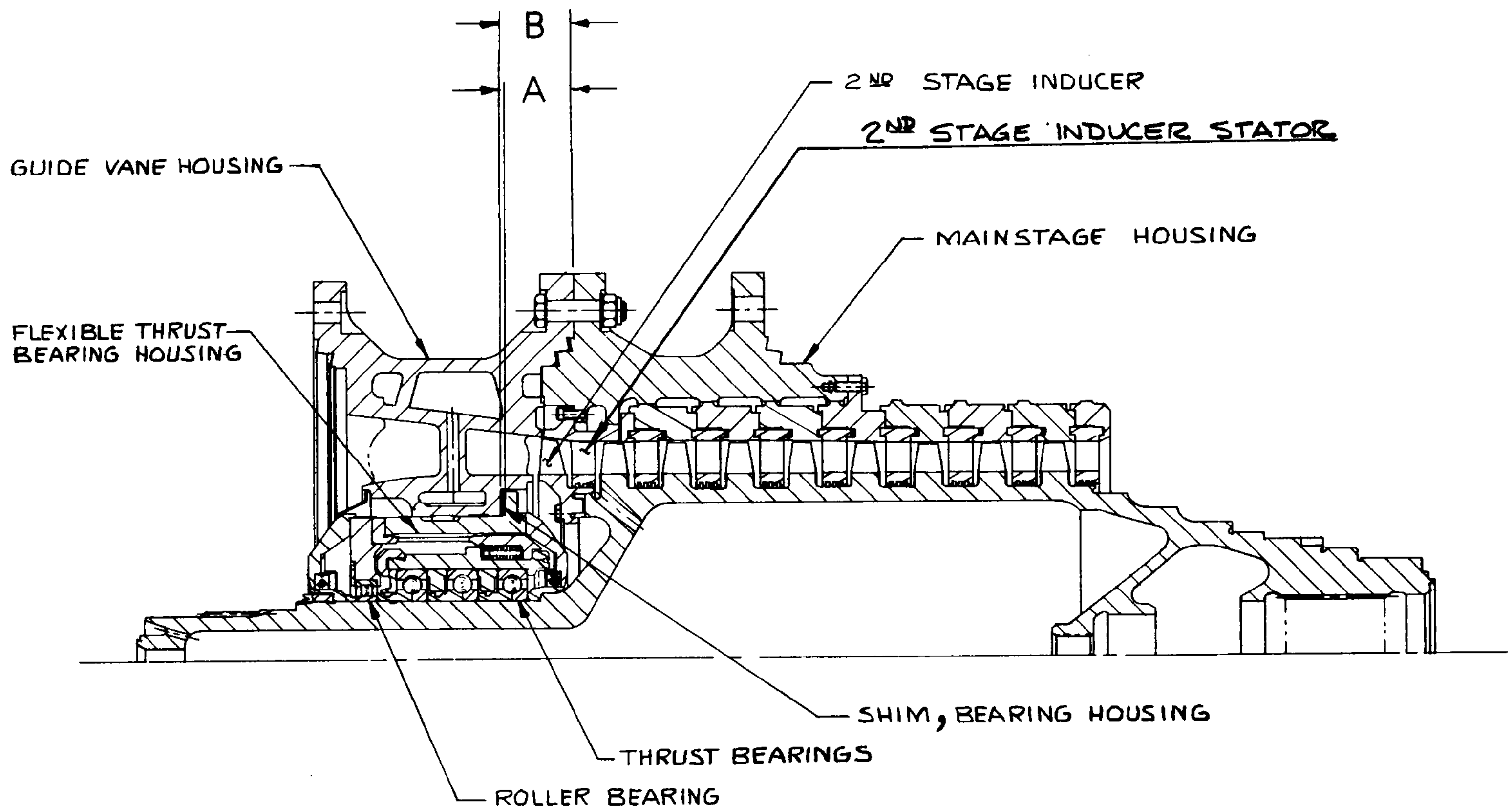


Figure 50. Assembly Sequence, Phase II

The thrust bearing pump rotor shaft assembly is allowed to return to room temperature. The following measurements are made to determine the proper spacer thickness; bearing housing flange to mainstage flange, "A" on Figure 50, and guide vane flange to bearing housing mounting surface, "B" on Figure 50.

The shim that fills the space between the bearing housing and the guide vane housing is machined to the thickness determined from the measurements taken.

The shim is positioned on the bearing housing and the guide vane housing is then moved into place. All necessary bolts are installed and torqued.

The pump-end roller bearing and its retainer are moved into place. As these parts are positioned, the wires from the thrust bearing instrumentation must be threaded through holes in the roller bearing retainer.

With the roller bearing in place, all instrumentation wires are routed through the passages provided in the guide vane housing to the connectors, which mount on the exterior surface. After all connections are made, an electrical checkout of all instruments must be completed.

The installation of the inducer side shaft riding seal and a shield completes this phase of the assembly.

E. DISCHARGE HOUSING (See Figure 51)

Before the discharge housing is installed, the pump is inverted in the assembly stand. This places the turbine-end down. The fixture used to maintain pump rotor position during the preceding assembly operations is removed. The installation of the discharge housing proceeds in the following manner.

The two labyrinth rings are installed in the discharge housing. The bolts are torqued and secured with safety wire.

The discharge housing is heated to 200°F.

The housing is positioned on the assembly stand elevator.

Conical seals are placed in the seal glands.

The housing is raised into position.

Bolts, nuts, and washers are installed and torqued in accordance with drawing requirements.

F. THRUST BALANCE PISTON (See Figure 51)

The pump rotor remains turbine-end down for this phase of assembly. This assures that the thrust bearings are in the same contact position, balls to races, as will be experienced at pump design point operation.

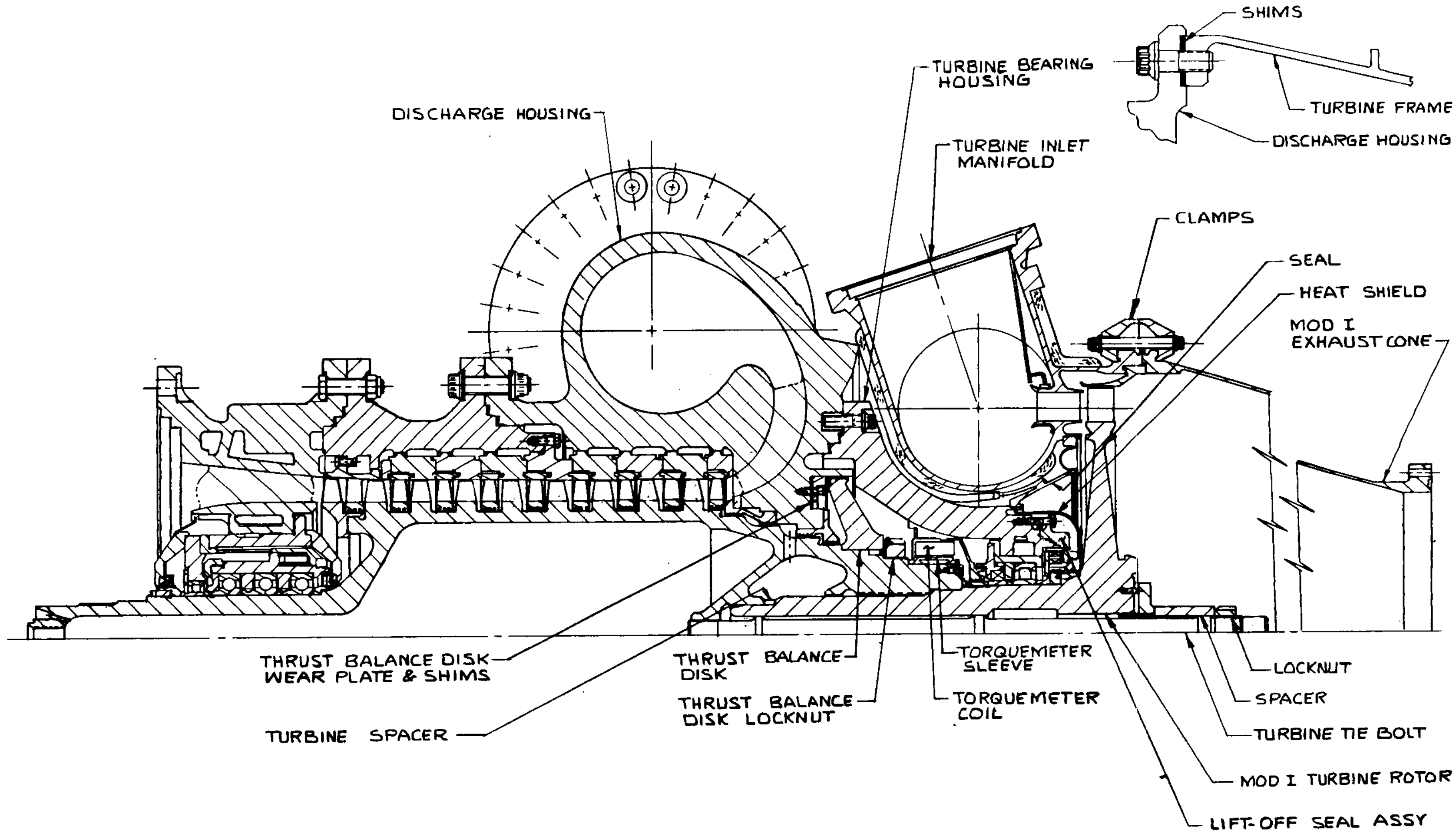


Figure 51. Assembly Sequence, Phase III

Assembly of the thrust disc then proceeds in the following manner.

Measurements must be made and recorded of all of the axial dimensions required to determine the distance between the high pressure gap face of the thrust disc and the discharge housing.

Using these measurements plus the measured thickness of the wear plate, the shim thickness of the wear plate is determined.

Shims are machined to the calculated dimensions.

The wear plate and shims are installed, bolts torqued, and safety-wired.

The thrust disc is heated to 200°F maximum and allowed to soak at this temperature. It is essential that all areas of the disc be at the same temperature.

While the disc is being heated, the rotor is cooled with liquid nitrogen.

With rotor and disc at their respective temperatures, the disc is moved into position on the rotor and held against the positioning shoulder on the pump rotor. The match marks on the thrust disc and pump rotor must then be aligned. This operation must be accomplished as rapidly as possible. The room temperature fit, disc to pump rotor, is 0.009-in. tight on the diameter. Any delay during installation of the disc can cause the disc to freeze on the rotor when it is only partially on the rotor journal.

The thrust disc lock nut and lock washer are installed. In fuel turbopump assembly Model I S/N 001, the wear plate was omitted. In this case the first four items above were not required for pump assembly.

G. TORQUEMETER SLEEVE (See Figure 37)

The torquemeter was not installed in fuel turbopump assembly Model I S/N 001 because of failure of the coil potting material when subjected to liquid nitrogen temperatures. The coil is a stationary part that is installed in the turbine bearing housing. Should the torquemeter be used, the sleeve is installed on the pump rotor in the following manner.

The sleeve is placed in position on the pump rotor. The rotor and sleeve keyways are aligned.

The large keys are inserted into the keyways; these keys must be bottomed against the ends of the keyways.

The wedge keys are inserted by hand pressure as far as they will go. Each wedge is scribed flush with the end of the torquemeter. Each wedge is marked so that it can be removed and later returned to the same keyway.

The wedges are removed and the ends cut off in such a manner that when they are reinstalled the ends of the wedges protrude 0.075-in. from the torquemeter sleeve face.

After machining, the wedges are replaced in their respective keyways. These wedges are then driven in so that the ends are flush with the sleeve face.

The retaining ring, lock ring, and bolts are installed.

H. TURBINE BEARING HOUSING (See Figure 51)

Before the turbine bearing housing is installed on the pump, a subassembly must be completed. The following items must be installed in the turbine bearing housing:

1. Roller bearing outer race and its housing.

2. Bearing coolant spray ring.

3. Pump-side shaft riding seal.

4. Pump-side labyrinth.

5. Torquemeter coil, if used.

6. Speed probes (see Figure 52).

7. Instrumentation leads, routed through the housing and connected to the exterior connectors.

With this subassembly completed, the installation proceeds in the following manner.

1. The bearing housing, including conical seals, is raised into position.

2. Bolts are installed, torqued, and secured with safety wire.

3. The external bearing coolant lines, strainers, and venturi units are installed.

I. TURBINE INSTALLATION (See Figure 51)

This phase of assembly includes the turbine rotor plus all related turbine hardware. The assembly procedure is described below.

A gaging tool, which simulates the turbine rotor shaft, is inserted into the pump rotor. Measurements are made from the gaging surface to a specific turbine bearing surface, as denoted on the applicable drawing. This measurement is recorded.

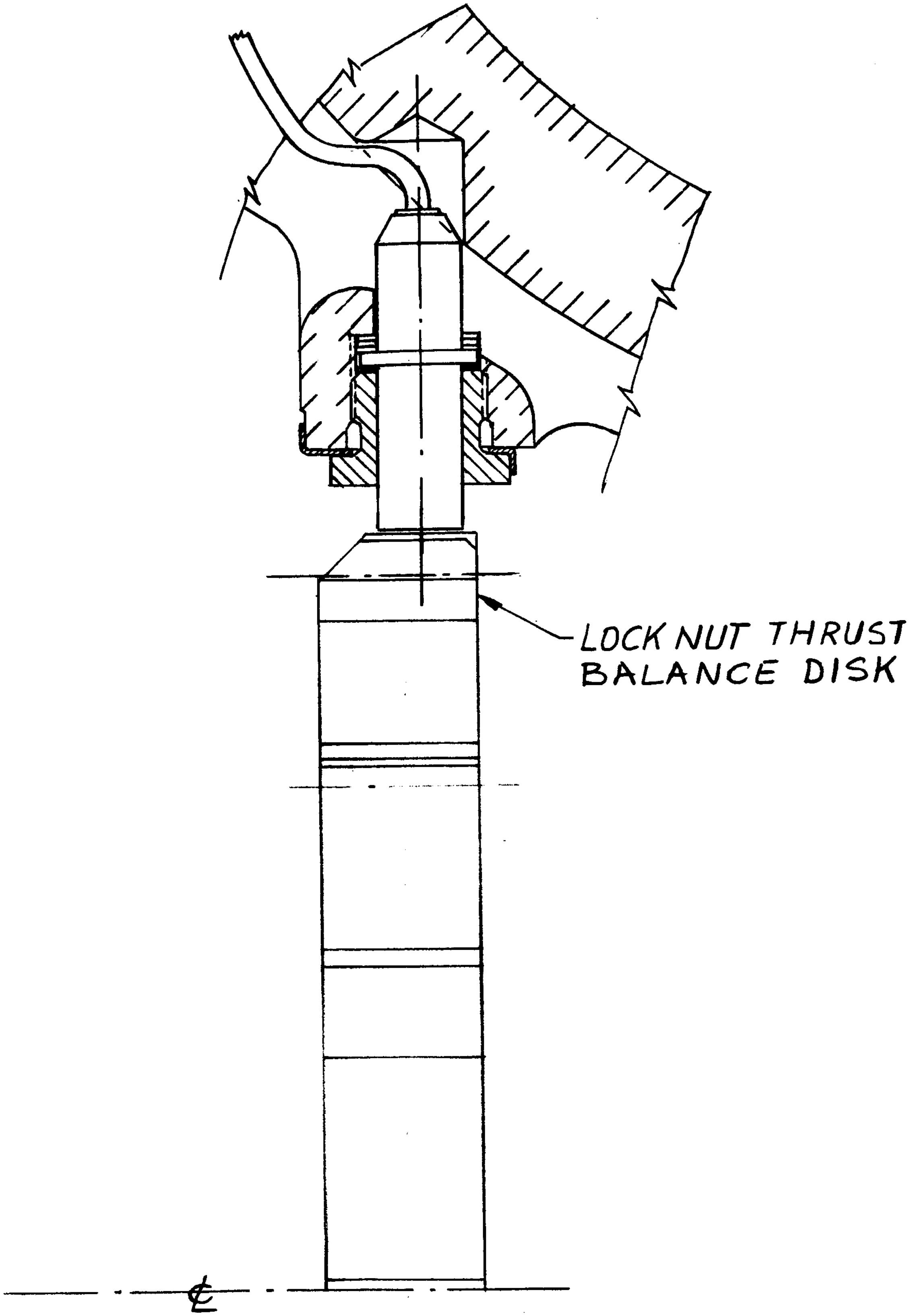


Figure 52. Speed Probe

Using the gage dimensions plus the recorded dimension taken in the above item, the following calculations can be made: the required length for the spacer that controls the axial positioning of the turbine rotor relative to the roller bearing outer race mounted within the bearing housing; and the thickness of shims required to position the turbine inlet manifold for the desired inlet nozzle to turbine blade gap.

The spacer and the required shims are machined in accordance with the calculated dimensions for the specific assembly.

The heat shield is installed on the inlet manifold.

The inlet manifold; turbine frame, consisting of three segments; and the shims are placed in position. The frame with its shims is bolted to the discharge housing. The turbine inlet manifold is positioned and clamped temporarily into the frame segments.

The seal, which extends from the turbine bearing housing to the inlet manifold is installed. Two seal welds are required to complete the seal installation.

The lift-off seal assembly with its seals is installed.

The heat shield, which protects the lift-off seal assembly, is installed and locked by bending tabs into slots that are provided in the bearing housing.

The turbine rotor subassembly is completed. This consists of installing the roller bearing inner race, spacers, running rings, the bearing lock nut, lock washer, and the turbine spacer, which were machined as explained in the second item above, on the turbine rotor shaft.

The turbine rotor shaft is placed in liquid nitrogen. When the shaft temperature has stabilized, the turbine is quickly removed from the liquid nitrogen and installed in the pump rotor and the match marks are aligned.

The turbine tie-bolt, spacer, lock washer, and nut are installed. Care must be taken to align all match marks to preserve dynamic balance of the rotating components. The breakaway torque is checked at this point to assure freedom of rotation. This torque is 120-in. lb for fuel turbopump assembly Model I S/N 001.

To facilitate assembly of the remaining turbine hardware, the pump assembly is rotated so that the turbine end is up. The turbine shroud is then installed. Centering of the shroud relative to the turbine is accomplished by set screws provided in the manifold flange. After centering the shroud, sealing plugs are welded over the set screws.

The exhaust cone is installed and clamped into position temporarily. The exhaust cone inlet manifold joint is sealed by welding.

The turbine assembly is completed by installing all clamps, keys, and bolts at the turbine frame-inlet manifold-exhaust cone interface.

J. INDUCER HOUSING (See Figure 53)

This housing, with the necessary conical seals, is raised into position. Bolts, washers, and nuts are installed.

K. FIRST-STAGE INDUCER (See Figure 28)

The inducer is placed in position on the pump rotor. The Invar 36 spacer, the retainer, and inducer bolt are installed. The bolt is torqued to the drawing requirements and locked with safety wire. The inducer spinner with its lock ring is installed. The spinner is locked by bending the lock ring tabs into slots provided in both the inducer and the spinner.

L. INLET ELBOW (See Figure 53)

The inlet elbow with its conical seals is raised into position. Care must be taken to assure correct orientation of the inlet elbow to the pump mounting bracket positions. The sway brace bracket, bolts, washers, and nuts are installed.

M. MOUNTING BRACKET (See Figure 53)

The pump assembly is rotated so that the turbine-end is down. The assembly is placed in the transport stand. The mounting brackets are positioned on their pads, which are located on the discharge housing. There is a left-hand and a right-hand bracket. They cannot be installed incorrectly because of the bolt patterns. The bolts are installed, torqued, and safety-wired.

N. THRUST BALANCE SYSTEM MANIFOLD (See Figures 46 and 47)

This installation is accomplished in the following manner.

The return flow manifold housing is assembled using the desired orifice plate.

The assembled manifolds are positioned, by means of brackets, over the discharge housing volute.

The 3/4-in. diameter and 1-in. diameter tubes are then bent and fitted between the manifolds and the various inlet and outlet ports in the pump housings. Tube bends and routing are determined as assembly progresses. Care must be taken to assure the least possible flow restriction. Each tube is completed by the installation of nuts and all other items required for flared tube fittings.

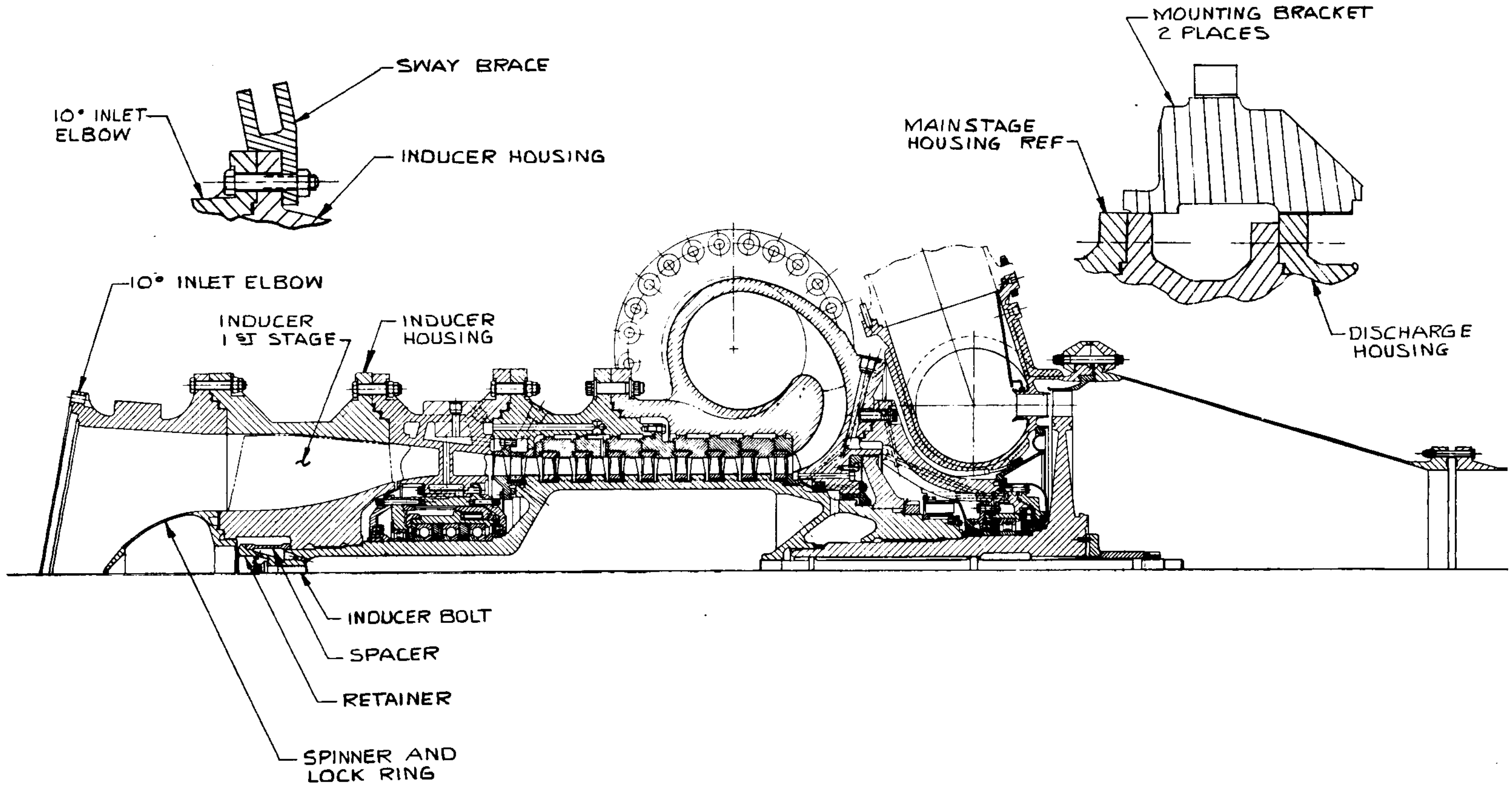


Figure 53. Assembly Sequence, Phase IV

The installation is completed by modifying the manifold support brackets as required. Bolts and safety-wire are installed.

O. MISCELLANEOUS COMPONENTS

The following items complete the assembly of the fuel turbopump assembly.

The turbine-end instrumentation panels are installed and all of the turbine bearing housing instrumentation is connected to these panels.

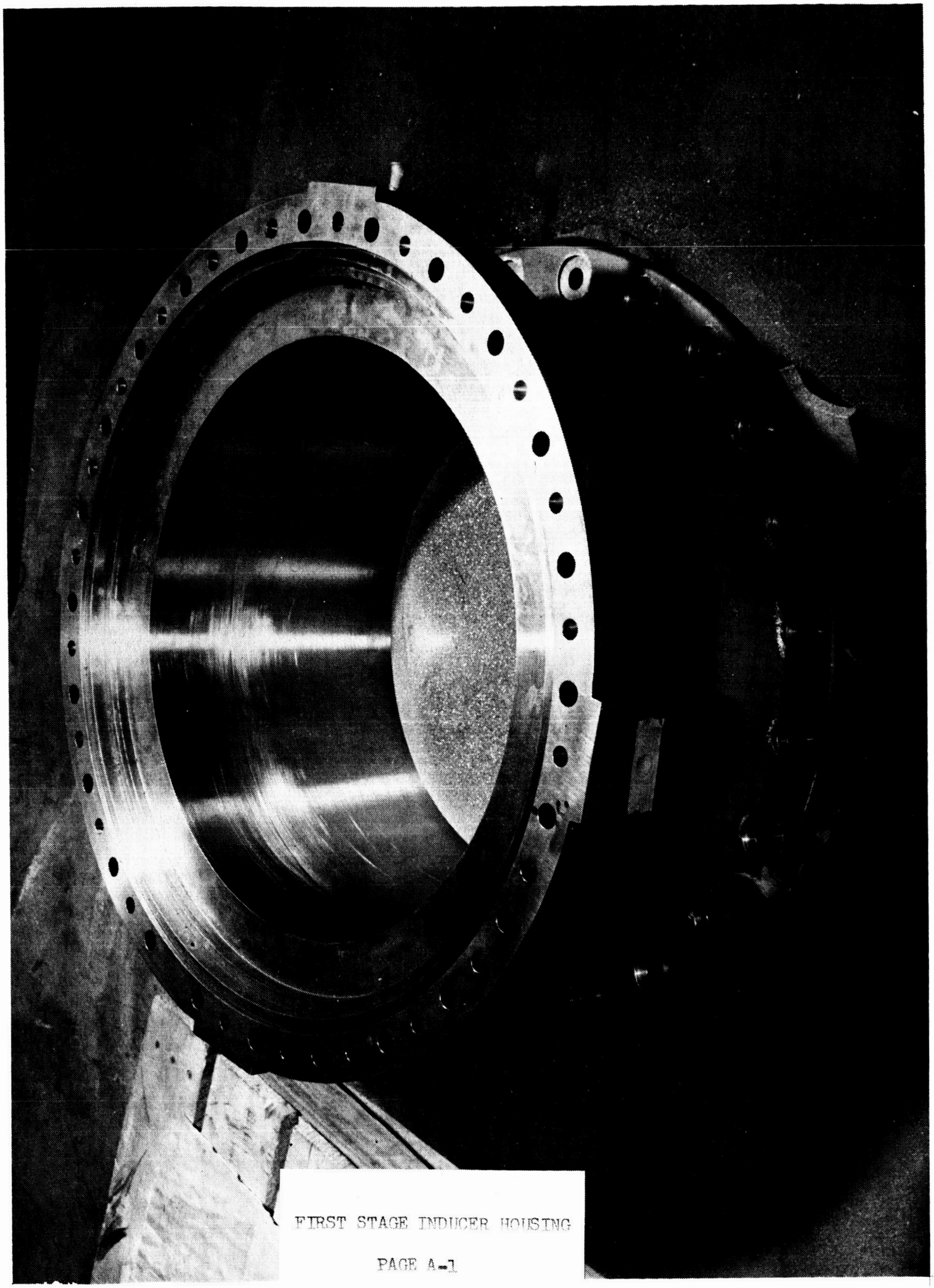
The pump-end bearing instrumentation panels are installed to the guide vane housing. All pump-end bearing instrumentation is connected to these panels.

Protective closures are installed on all open flanges and all open ports.

Purge and final leak check of fuel turbopump assembly is performed with gaseous nitrogen. After completion of the purge, a gaseous nitrogen atmosphere pressure of 2 psig is maintained within the pump at all times.

APPENDIX A

TURBOPUMP COMPONENTS



FIRST STAGE INDUCER HOUSING

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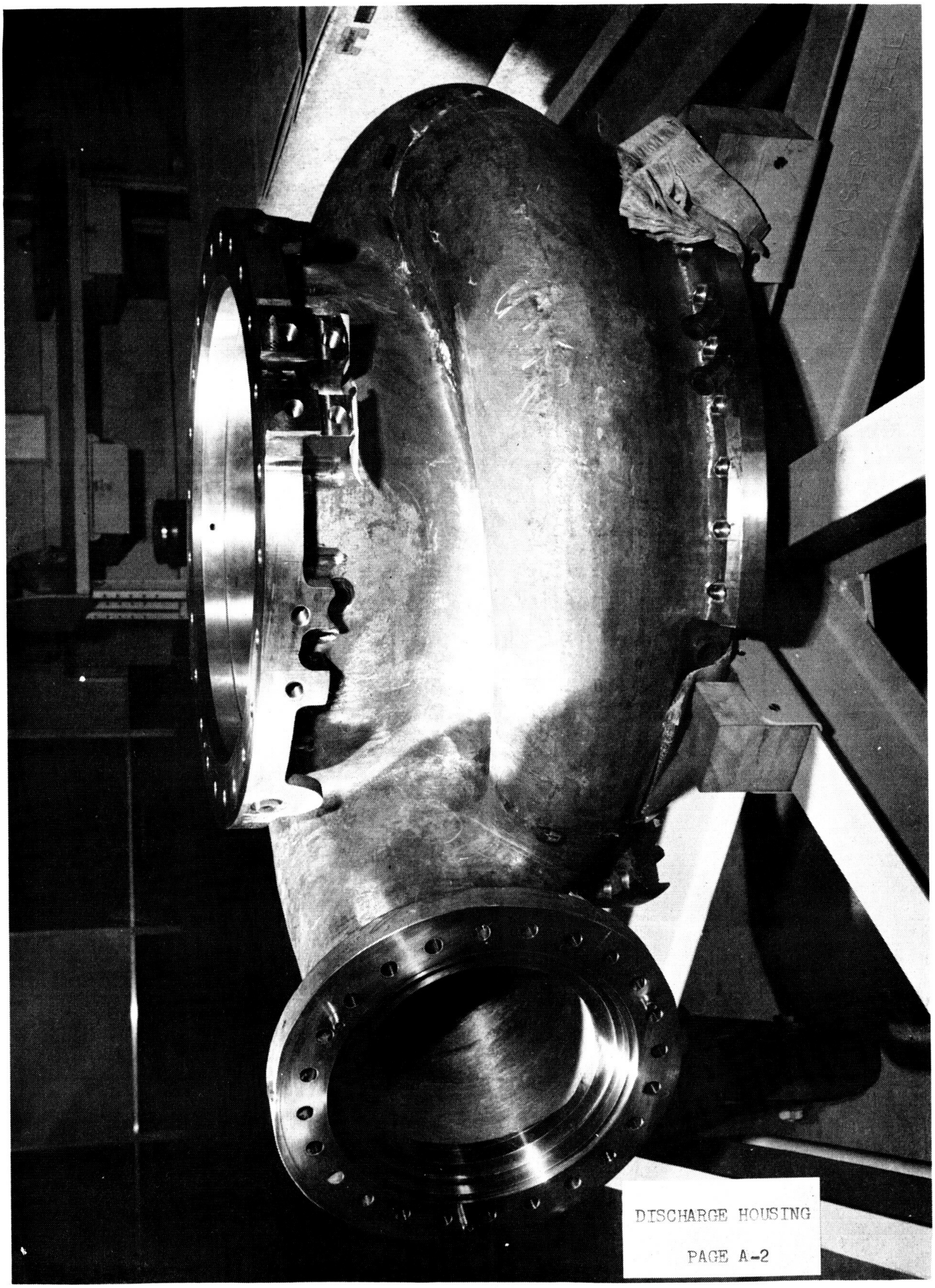
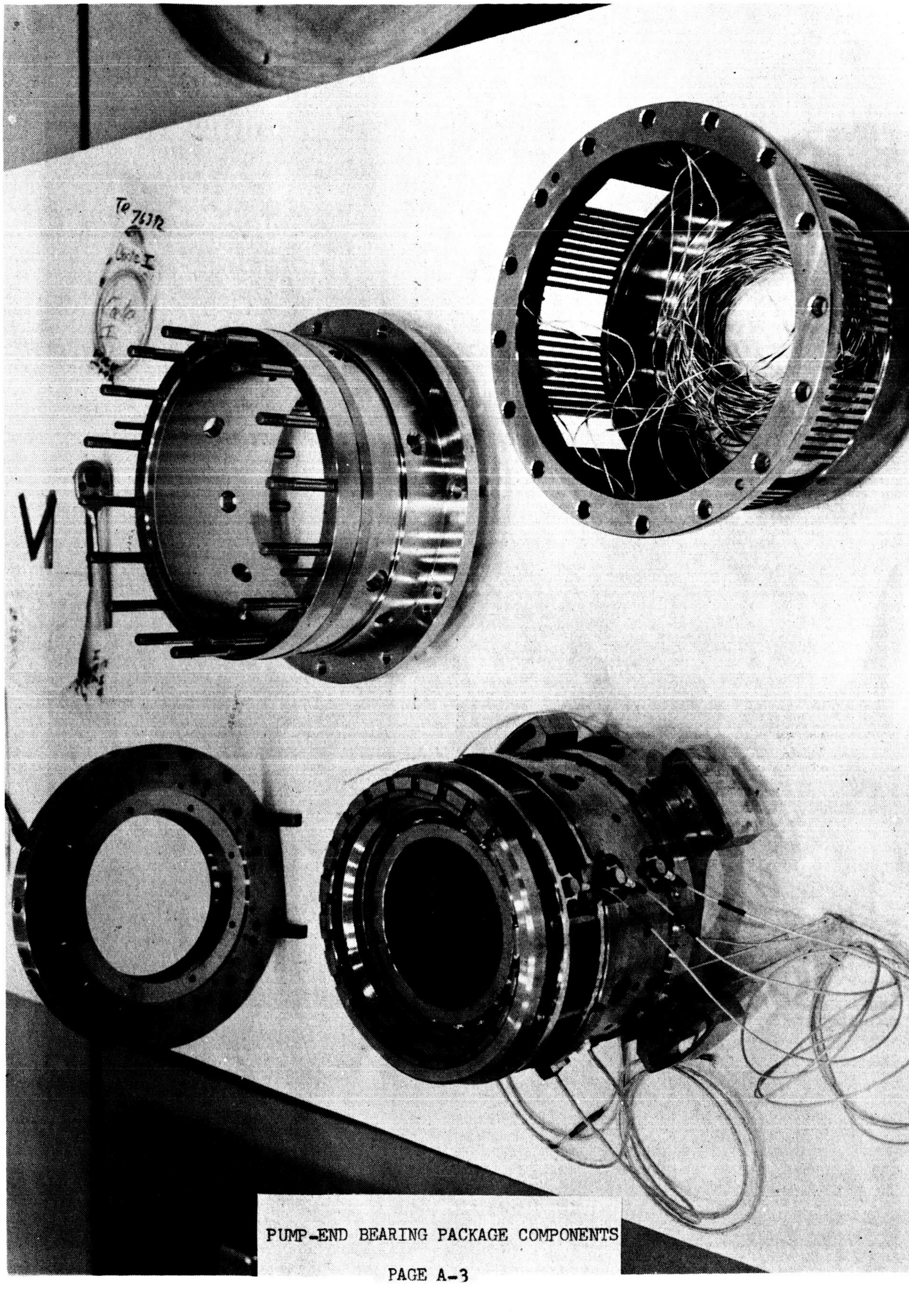


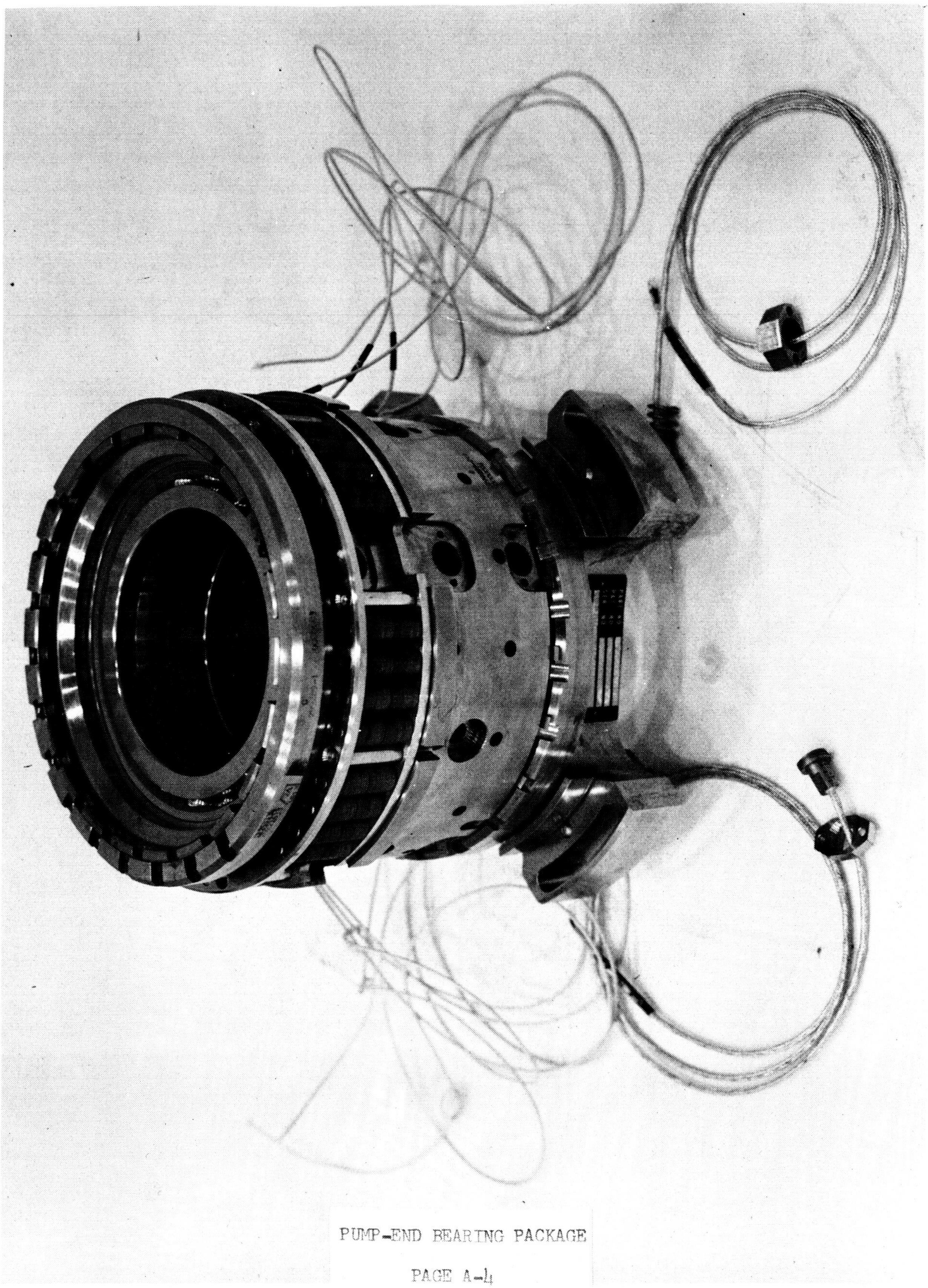
DISCHARGE HOUSING

PAGE A-2

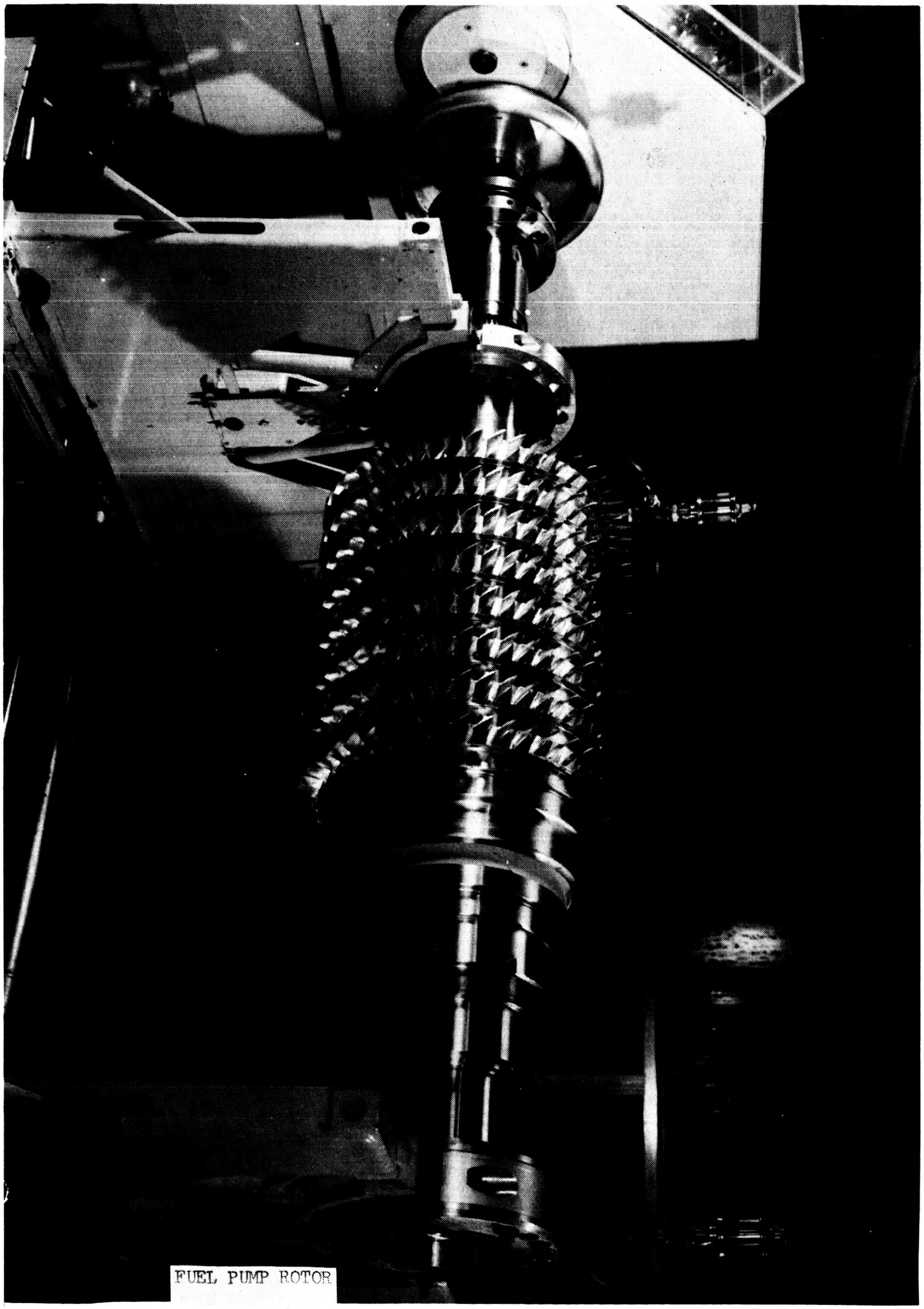


PUMP-END BEARING PACKAGE COMPONENTS

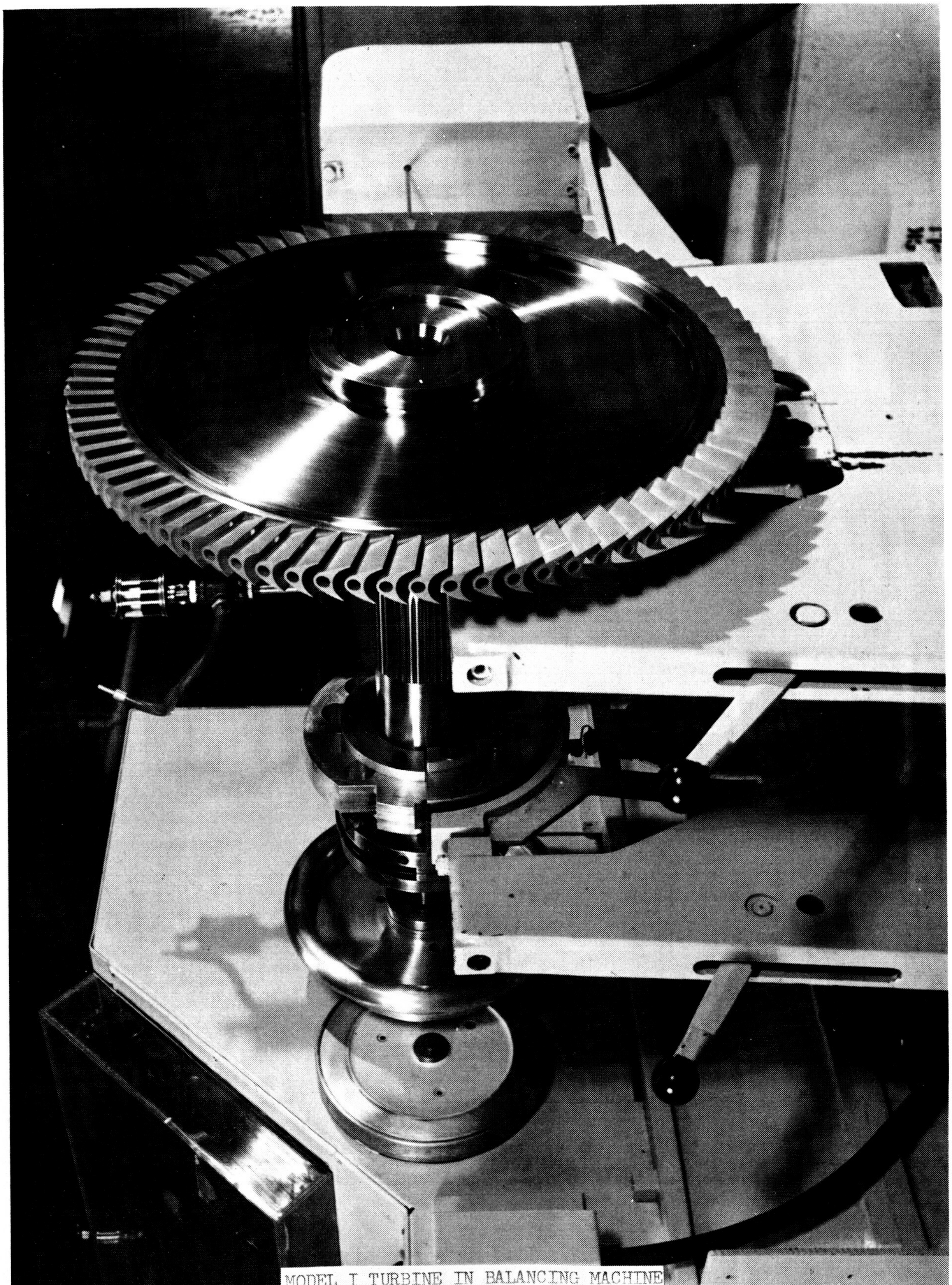
2



PUMP-END BEARING PACKAGE

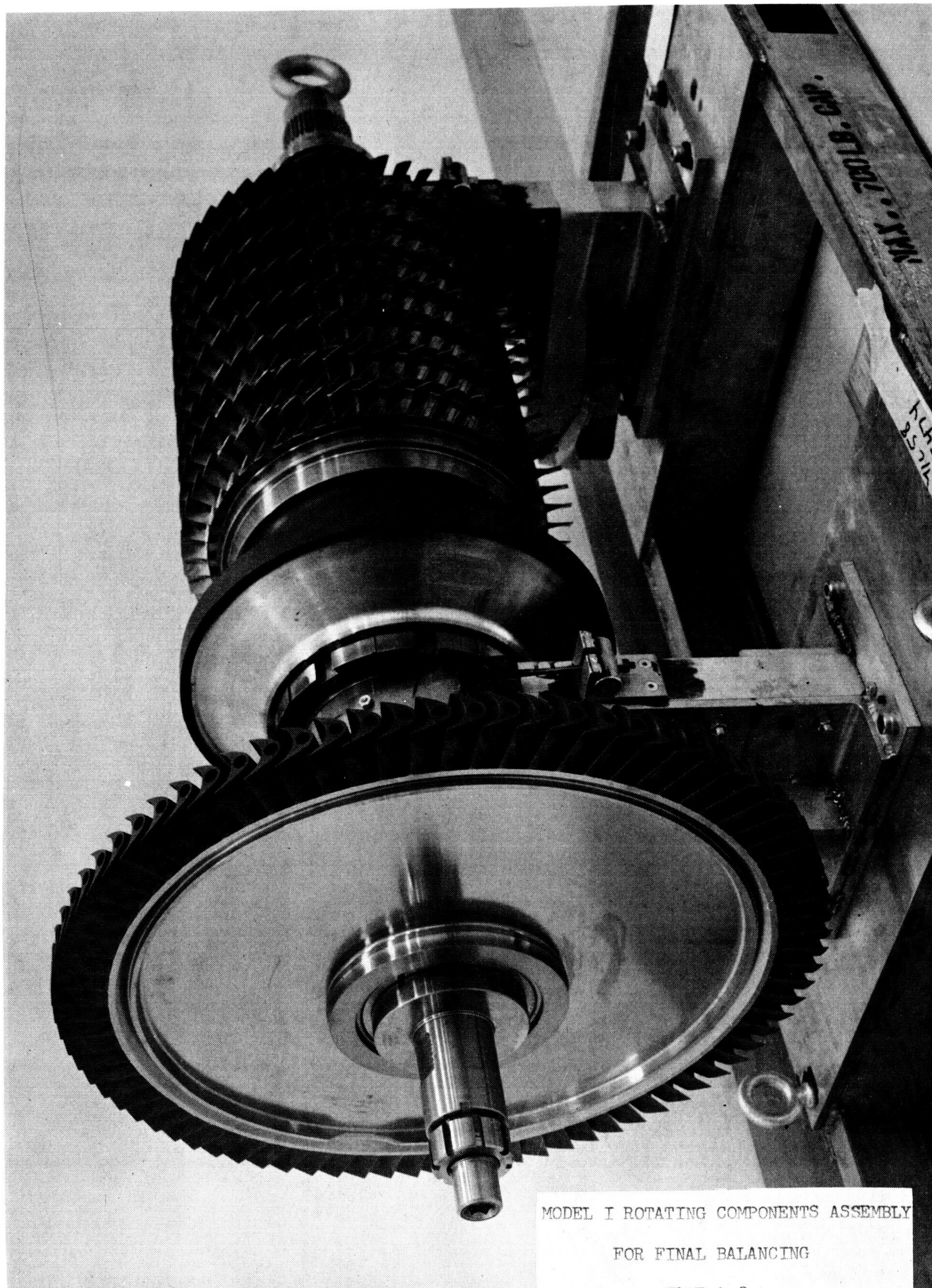


FUEL PUMP ROTOR



MODEL I TURBINE IN BALANCING MACHINE

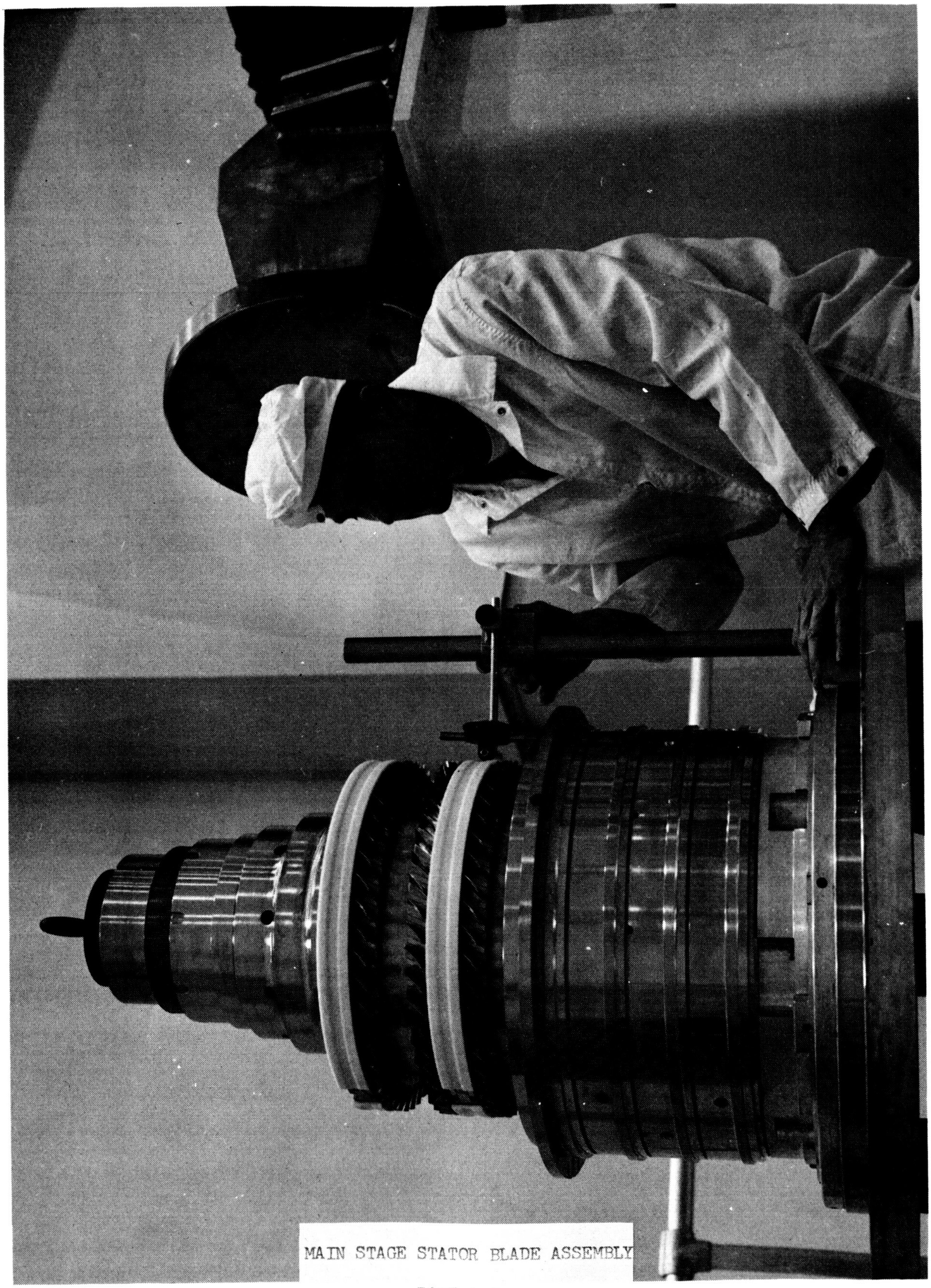
MODEL I ROTATING COMPONENTS ASSEMBLY
FOR FINAL BALANCING



MODEL I ROTATING COMPONENTS ASSEMBLY

FOR FINAL BALANCING

MAIN STAGE HOUSING INSTALLATION



MAIN STAGE STATOR BLADE ASSEMBLY

INSTALLATION OF FTPA ON DUMMY TCA

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