DAMPING SEAL VERIFICATION SETUP

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Introduction

The heart of the Space Shuttle Main Engine (SSME) is a set of turbopumps that propel cryogenic fluids at very high pressures and flow rates, at rotor speeds up to 37,000 rpm. The interaction and phase relations between the rotor's elastic and inertial forces with fluid restoring and damping forces is apt to give rise to subsynchronous whirl, a type of self-excited vibration at frequencies near the lowest critical frequency of the rotor. This potentially destructive dynamic phenomenon imposes limits on the performance of the engine.

Bushing seals that cause the flow in the fluid film to become turbulent, by means of a multiplicity of pockets, have been shown theoretically, Reference 1, not only to inhibit subsynchronous whirl, but to reduce leakage as well. However, experimental data that relate these two desirable characteristics to such parameters as pocket depth, Reynolds number (based on clearance and axial flow rate), and rotating speed are limited.

To obtain the required data, NASA's Marshall Space Flight Center (MSFC) commissioned Wyle Laboratories to design, build and operate a test rig in which the damping efficacy and leakage reduction of typical candidate seals are to be evaluated. Experimental conditions will either reproduce operating conditions, such as rotating speed and seal geometry, or will be relatable to them, such as Reynolds Number and dynamic characteristics.

Experimental Design

Rotor Dynamics

Normally, a flexible rotor turns in rigid seals and bearings. Here, the test setup order is reversed: a softly suspended bushing moves about a stiff rotor. The damping seal responds similarly in either case, by inducing, with the fluid film stiffness, damping, mass, and whirl cross coupling effects. The mass of the damping seal bushing is suspended against the rotor by the fluid film stiffness. The gap flow is established by axial pressure difference and by the Couette flow from the shaft rotation. Dynamically, the real and test systems are analogous.

Seal Test Configuration

Two principal objectives determined the design of the Damping Seal Test Rig: minimization of risk, and reduction of variables in the machinery. To attain the first of these, water was substituted for cryogenic fluids, as the substance most nearly comparable with respect to specific gravity and viscosity.

Simplification was achieved by replacing the flexible rotor and rigidly mounted seal by a rigid shaft (i.e. one with a critical speed above the highest operating speed) and a seal virtually unrestrained in a plane normal to the shaft axis, but, obviously, prevented from rotation due to circumferential viscous shear forces. The effect of shaft bending stiffness is eliminated, and the only forces acting on the seal are those due to its mass, and the fluid film. In contrast to the actual turbopump, the shaft is externally driven.

Experimental Variables

Seal: The seal to be tested is shown in Fig. 1. It has a length of 1.8 in. and fits a shaft with a diameter of 3.6 in. The inside surface is machined in a pattern consisting of right-angle triangles separated by lands .020 in. wide. Four seals are to be retested, three with pocket depths of .020, .040 and .100 in., respectively, and a smooth seal for comparison.

Flow Rates: Axial flow rates through the seal will be varied from full through 1/2, 1/4, 1/8 and 1/16, for a total of five conditions.

Reynolds Number: All twenty configurations of pocket depth and flow rates are to be tested at two maximum flow rate settings, based on Reynolds Numbers of 100,000 and 200,000. The absolute viscosity of water can be varied only through rather narrow limits, by a factor of approximately 2, if one eliminates the need for costly cooling facilities and keeps the highest inlet temperature low enough to prevent flashing of the pressurized water emerging from the seal.

Of the two remaining variables in the Reynolds Number, the axial flow velocity through the seal is specified to lie between 75% and 125% of the shaft surface velocity at 36,000 rpm, or 5089 to 8432 in./sec. All but the lowest of these values were found to be impractical because of the high pressures required to overcome fluid friction in the seal, entrance and discharge losses. Concern was felt over potential problems due to deformation of the shaft seals at pressures much above 2000 psi, as well as the high leakage flow rates through these seals at high pressures, which would increase the already high flow requirements beyond practical limits.

Reduction in the damping seal gap width, as a means of lowering flow, was precluded by the high temperatures required to achieve the specified Reynolds Numbers. These considerations narrowed the feasible gap widths to .015 to .025 in. Because of the cost of the seals, and of the assembly and disassembly, a single gap width of .020 was selected. The resulting maximum calculated entrance pressure is 2150 psi, and the flow rate, 299 gpm, excluding leakage through the shaft seals that must contain this pressure, in addition to the exit pressure. Inlet temperatures for the two Reynolds Numbers at maximum flow are 108 and 183 degrees F. All calculated values are based on the smooth seal. The seal analysis is derived from Reference 2.

Test Method

With water flowing through the seal, and the shaft rotating, the seal carrier is forced off center and released. Its subsequent motions are measured, in two orthogonal directions, by two proximity probes with respect to the shaft, and in inertial space by two accelerometers. The force required to decenter the shaft is also measured to give the static spring rate and cross coupling coefficient. Damping coefficients are derived from the amplitudes and phase relations of the seal motion in the plane normal to the shaft axis.

Test Setup

The damping seal test setup comprises three major subsystems: The test rig, test rig drive, and pressurized water supply.

Test Rig

The heart of the test rig is the seal suspension assembly, shown in Fig. 2, with the housing, shaft seals and water passages removed for clarity. The cylindrical damping seal carrier has flanges at both ends which are captured between water-lubricated hydrostatic thrust bearings that also act as seals. The recesses in the thrust bearings are supplied through orifice restrictors that maintain nearly constant film thickness in the presence of supply pressure variations. The upstream thrust bearing is supplied from the inlet pressure, with connecting passages leading through the chamber wall to the downstream bearing, thus ensuring thrust balance over a range of inlet pressures. The downstream bearing carries a greater thrust load, due to the unbalanced area of the damping seal. Bearing film thicknesses are of the order of one half mil.

The hydrostatic thrust bearings constrain the seal carrier to radial displacements in a plane normal to the shaft axis. To restrain rotation of the seal, the seal carrier is connected by two parallel links with spherical bearings to two short cantilever lugs on a torque bar which is mounted in bearings on the test rig housing. The kinematics of this mechanism allow free radial movement of the seal in any direction, through a combination of rotation of the torque bar about its axis, and inclination of the links. However, the fluid torque applies a couple to the links which is resisted by the torsional stiffness of the torque bar.

The same mechanism is used to pull the seal off center and release it, so as to initiate the oscillations mentioned above. A cam follower mounted on an arm at the end of the torque bar is engaged by a cam, driven by a small electric motor through a high gear reduction (not shown). Rotation of the cam is stopped by a limit switch the follower of which rides on a second cam shaped to allow one revolution only. A third cam cocks a release mechanism which does not permit the seal to move until the first cam follower is free of its cam surface; this prevents the seal from being slowed in its rapid return motion by the inertia of the mechanism.

The links are strain-gauged to allow measurement of the decentering force, as well as the torque due to fluid friction. The proximity probes and accelerometers mentioned above are mounted on a split circumferential band that supports the seal carrier; the padeyes for the links are welded to this band. This method of supporting the seal, in place of direct attachment to the seal carrier, is required to ensure complete axial symmetry in the seal carrier. Lack of symmetry in the seal carrier cross section is apt to lead to distortion of the flanges under thrust loading, which must be avoided owing to the very small film thicknesses of the thrust bearings.

The hollow, stepped test shaft runs in preloaded angular contact ball bearings having an internal diameter of 30 mm. Two opposed bearings are located at the downstream end. At the upstream end, a single upstream bearing is preloaded against a pair of downstream bearings to increase the thrust capacity which is required to resist the unbalanced pressure load on a narrow shaft shoulder just ahead of the test seal. The bearings are supplied by SKF.

In view of the high shaft rotating speed and the high water pressure, the shaft seals are among the most critical components of the test rig. A pair of air-pressurized, segmented carbon seals is used at each end of the shaft to isolate the shaft bearings and their lubricant from the water. The downstream bushing seal provides closure of the test chamber against presumably low pressure; it, too, makes use of carbon running on stainless steel. A leakage path out of the test rig housing is provided; however, the seal is designed to be capable of running dry without damage.

The greatest technical challenge is posed by the upstream seal which must contain the maximum pressure of about 2200 psi. This is a film-riding, balanced bushing seal that depends on leakage past the lip for lubrication. It makes use of bronze and stainless steel at the sliding interface.

Orifice compensated hydrostatic bearings are incorporated in the radially floating seal to center it on the surface-hardened shaft. The recesses are supplied from the pressurized water entering the test rig housing through three radial openings.

All seals and hydrostatic bearings were designed, and are being built by the Stein Seal Company of Philadelphia.

In order to minimize the downstream stagnation pressure, the water emerging from the test seal is led into a circumferential diffuser from which it leaves through six tangentially oriented openings. It is caught in large tubes and dumped in a pond.

Considerable leakage must be accepted for lubrication of the seals and hydrostatic bearings: about 10 gpm for the upstream seal, and 6 gpm for each of the bearings. Since only the entering flow can be measured, by means of a strain-gauged flowmeter venturi, the leakage flow must be collected in 55 gallon drums and measured after the test, to determine the net flow through test seal.

Since the test rig must be disassembled several times for installation of different damping seals, its wet interior will be exposed to the atmosphere. To preclude corrosion which may deposit particles in the very small seal and bearing gaps, all housing components are made from stainless steel.

Test Rig Drive

The prime mover for the test rig is a surplus AVCO Lycoming helicopter gas turbine with a maximum output speed of 37,000 rpm, rated at 500 HP. It was originally intended to modify the high-speed turbine shaft so as to drive the test shaft directly. However, this would have removed some of the internal balance due to gear thrust components, which would have shortened the bearing life. Thus, loading of the turbine output shaft, by such means as a hydraulic pump working against a relief valve, would be required, together with a heat exchanger.

The complexity, cost and time required for these two modifications were considered excessive. The chosen alternate approach is based on a single-stage speed increaser gear box with a 1:6.2 ratio, with the low speed input shaft connected by a splined shaft to the turbine output shaft, and the high speed output shaft driving the test shaft through another spline connection. The gearbox, with shafts running in hydrodynamic bearings, is a modification of a standard unit built by Philadelphia Gear Corporation in King of Prussia, Pennsylvania.

Pressurized Water Supply

The required flow of over 300 gpm, at pressures up to 2200 psi, will be supplied by an existing blowdown system originally built to simulate accident conditions involving safety relief valves in nuclear power plants. Its main components are a 340 cu ft pressure vessel (Figs. 3 and 4), and a pair of gaseous nitrogen storage tanks rated at 6000 psi (Fig. 5). In order to extend the run time, these tanks will be supplemented by two trailers carrying additional nitrogen bottles, supplied by NASA/MSFC. At the maximum flow rate, the required inlet pressure can be maintained for nine minutes.

The system schematic is shown in Fig. 6. The test rig supply line branches off from the existing 10 inch, carbon steel line from the pressure vessel into a set of filters provided to remove corrosion products; all lines downstream of the filter are made of stainless steel. A remotely actuated flow control valve and a flow meter are provided to meter the water flow.

The water in the pressure vessel is brought up to the required temperature by passing steam through it from an existing boiler. Thermocouples at three levels measure the water temperature.

With the high shaft speeds and small clearances, a failure of the pressurized water supply could very quickly lead to serious damage in the seals and bearings of the test rig. To protect against this, an emergency water supply, consisting of a high flow, low pressure centrifugal pump and a reservoir, is provided; it is isolated from the high pressure supply line by a check valve. An additional safety feature is the rapid response of the helicopter turbine which, in its usual condiguration, can reach full speed in 3.5 seconds, and even faster when connected to the low inertia of the test rig. It can be stopped very rapidly by shutting off the fuel supply, which will be implemented by a solenoid-actuated valve controlled by an emergency button.

Control and Instrumentation

Test rig controls have been purposely kept simple. Variables controlled in preparation for a test run include nitrogen pressure, pressure vessel water level and temperature, and flow rate set point. When all conditions are fulfilled water is admitted to the test rig by opening the main shutoff valve, and the turbine engine is started. The turbine speed is controlled manually through a valve that regulates the fuel supply. A manual switch then overrides the cam limit switch and initiates the decentering and release of the damping seal. The dynamic stability of the damping seal is determined visually by monitoring, on a CRT, the seal displacement measured by the proximity sensors. The turbine is shut down when oscillation amplitudes are seen to grow. Preset speed limits are provided for safely approaching stability limits.

Data recorded on magnetic tape include turbine rpm, water temperature, flow rate, test rig inlet and discharge pressures, and the outputs of the proximity gauges and accelerometers on the damping seal carrier.

Current Status

Construction of the test rig is about to begin. The spline shafts connecting the gearbox to the test rig and turbine are in design, as is the modification of the standard gearbox. Modification of the blowdown test facility has begun.

REFERENCES

1. Damping Seals for Turbomachinery, G. L. von Pragenau, NASA Technical Paper 1987 (1982).

2. A Discussion of the Theory of Sealing Devices, P. C. Stein, SAE Southern New England Section, 1961.

3. Heat Transmission, W. H. McAdams, McGraw-Hill, 1954 (data on water properties).

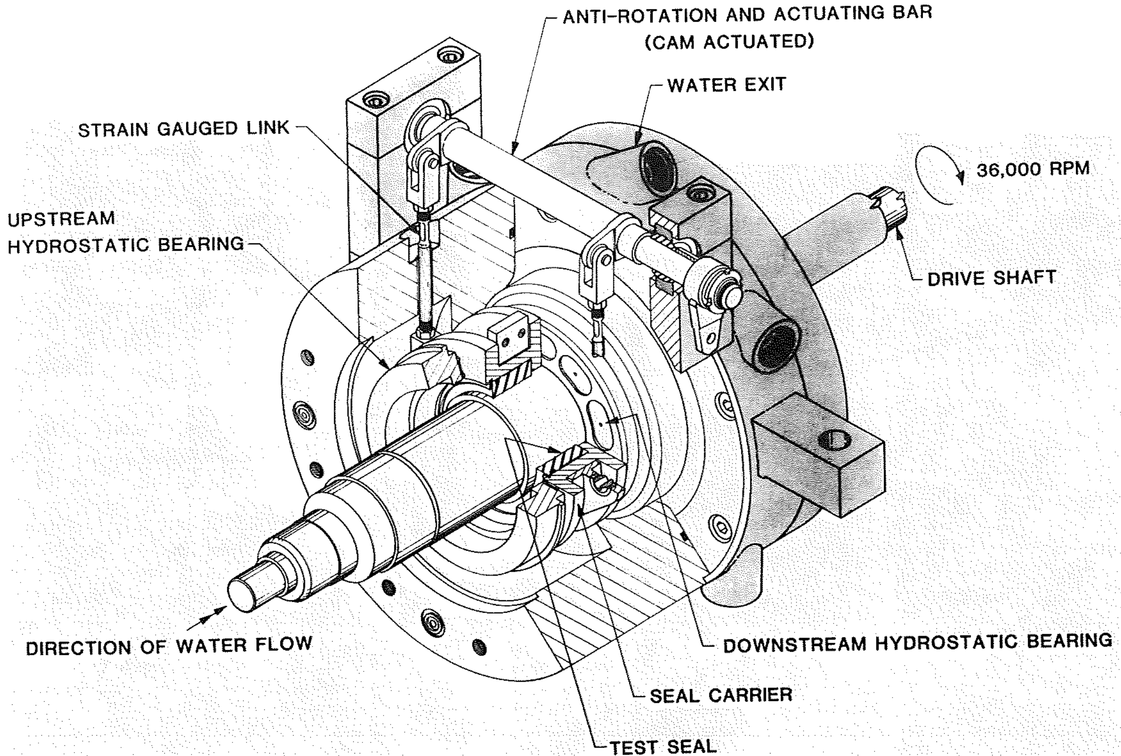
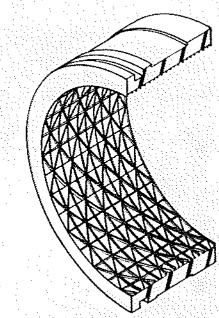


FIGURE 2.



TEST SEAL

FIGURE 1.

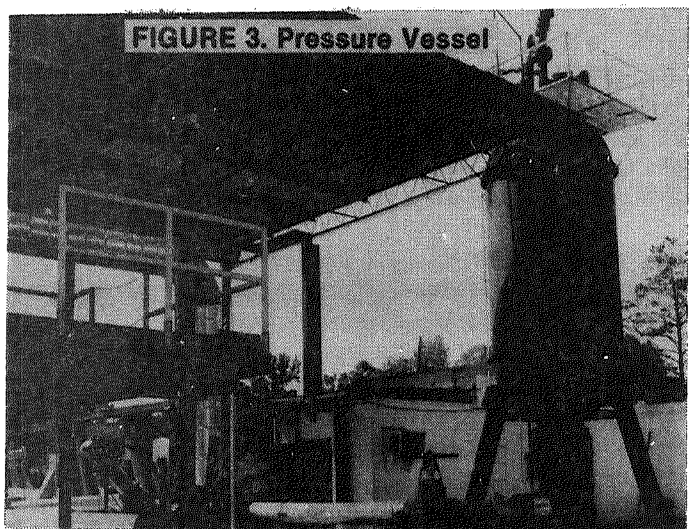


FIGURE 3. Pressure Vessel

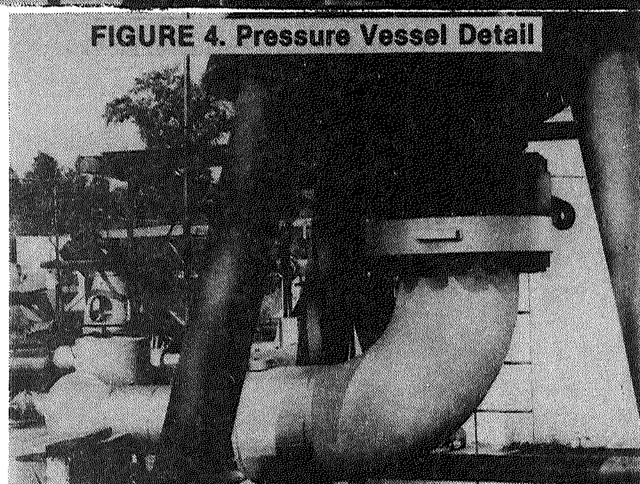


FIGURE 4. Pressure Vessel Detail

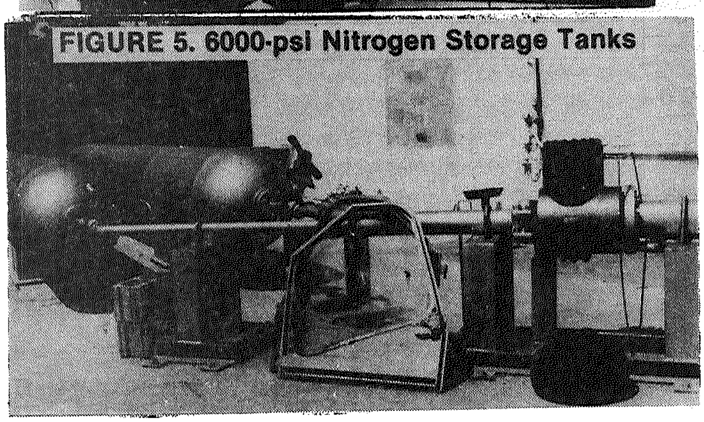


FIGURE 5. 6000-psi Nitrogen Storage Tanks

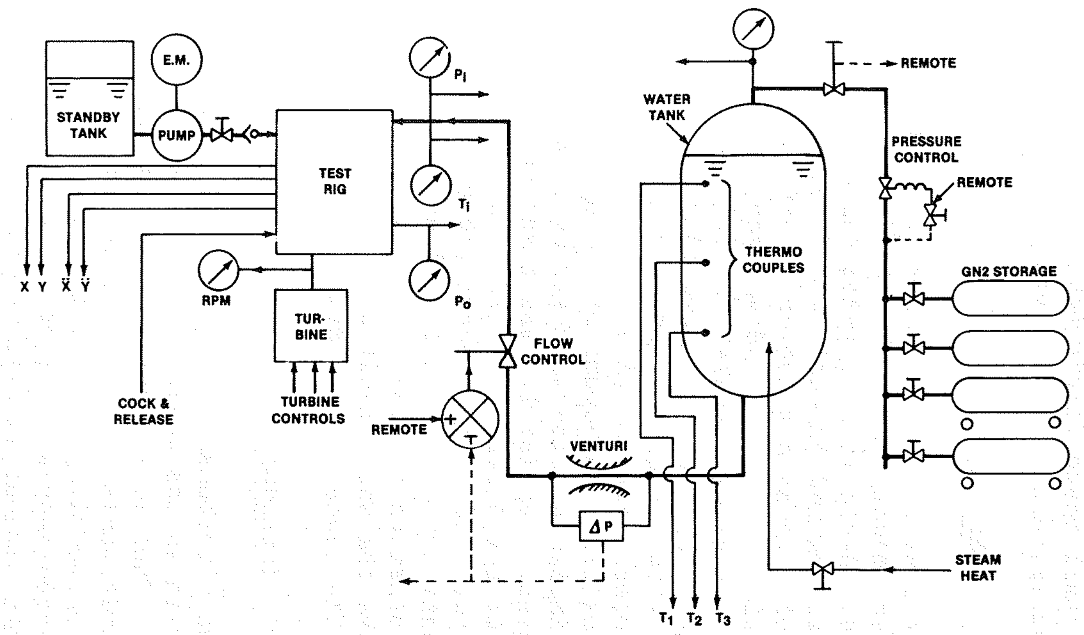


FIGURE 6. Damping Seal Test Rig Instrumentation & Control (Preliminary)