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LARGE-SCALE ADVANCED PROP-FAN (LAP) PITCH CHANGE ACTUATOR AND CONTROL DESIGN REPORT

By: Robert A. Schwartz
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HAMILTON STANDARD DIVISION
UNITED TECHNOLOGIES CORPORATION

Prepared For
National Aeronautics and Space Administration
NASA-Lewis Research Center

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1.0 Summary

The pitch control system provided for the LAP program was designed to provide a low risk development program. This was accomplished by using existing hardware where possible and conservative state of the art design and manufacturing practices. The use of an existing hydraulic control resulted in less than optimum transient response. This was not considered critical as the primary purpose of this program is to evaluate the structural integrity and aeroacoustic performance of the blades. This approach also resulted in a heavier design than would have been expected in a production configuration. The calculated weights of the pitch control system are:

Actuator: 214 Kilograms (472 pounds)

Control: 61.2 Kilograms (135 pounds)

Fluids

Control: 0.017 meter^3, 15.3 kilograms (18.0 quarts, 33.75 pounds)

Actuator: 0.011 meter^3, 10.2 kilograms (12.0 quarts, 22.5 pounds)

The pitch control system is designed for fail-safe operation. Any malfunction of the control will either pitchlock the blades at their present blade angle (less approximately one degree) or feather the blades. In addition, the pitch control system provides the following features:

- Constant speed governing
- Mechanical in-place pitchlock
- Feather via mechanical input signal
- Feather via electrical input signal
- Ground adjustable low pitch stop
- Fixed reverse angle capability
- Electrical aux feather/unfeather motor
- Single lever input
- Individual replacement of blades
- Self-contained oil management (except for heat exchanger)

2.0 Introduction

In recent years, considerable attention has been directed toward improving aircraft fuel consumption. Studies have shown that the inherent efficiency advantage that turboprop propulsion systems have demonstrated at lower cruise speeds may now be extended to the higher speed of today's turbofan and turbojetpowered aircraft. To achieve this goal, new propeller designs which feature more blades with thin airfoils and aerodynamic sweep are required.

Since 1975, Hamilton Standard has been deeply involved with the NASA Lewis Research Center in the development of the advanced turboprop or Prop-Fan. Many aircraft system studies have been accomplished for a variety of subsonic air transport applications and all these studies have shown significant fuel savings with Prop-Fan propulsion. The fuel savings potential of future Prop-Fan powered aircraft is generally 15-20% for commercial applications and 25-35% for military patrol aircraft compared to equal technology turbofan systems, depending upon the specific application, cruise speed, stage length and other requirements.

To date, several propeller models have been designed, manufactured and subjected to a number of tests. A series of small-scale 0.6223 meter (24.5 inch) diameter model tests have been conducted in both UTRC and NASA wind tunnels and on a modified NASA airplane. These tests have shown that propellers with 8-10 swept blades, high tip speeds and high power loadings can offer increased fuel efficiencies at speeds up to 0.8 Mn.

Under the NASA sponsored Large-Scale Advanced Prop-Fan (LAP) Program, Hamilton Standard has recently designed a 2.743 meter (9-foot) dia. single-rotation Prop-Fan (Figure 2.1) and is now in the process of manufacturing this system. This hardware will be tested at Wright Field and in the ONERA S1 wind tunnel in France. The hardware will then be used in a follow-on program where it will be run with an engine on a static test stand, and on a research aircraft. The major objective of this testing is to establish the structural integrity of large-scale Prop-Fans of advanced construction in addition to the evaluation of aero-acoustic performance.

Previous reports have described the design of the blades, the spinner and the hub & retention (references 1-3).

This report is limited to the operation and design description of the LAP pitch control system. The pitch control system consists of two separate assemblies. The first is the control unit which provides the hydraulic supply, the speed governing and the feather function for the system. The second unit is the hydro-mechanical pitch change actuator which directly changes blade angle (pitch) as scheduled by the control. This report describes the operation, the requirements and actual hardware that are used in the LAP pitch control system.

3.0 Pitch Change Operation

3.1 Pitch Change Actuator and Control Schematic

Figure 3.1 is a complete schematic that shows all of the operational features of the pitch change actuator and control. The following sections break the control schematic into more easily understood subsystems.

The overall system was based on using an existing 54460 control with a pitch-locking propeller. The system was carefully reviewed to ensure all control malfunctions will either cause the system to pitchlock or go to feather. The actuator itself is considered prime structure and designed to conservative loads and allowable stresses.

3.1.1 Pitch Change Actuator and Pitchlock

The hub mounted pitch change actuator assembly (See Figure 3.2) consists of an internal stationary piston, a translating outer cylinder with an integral yoke, a centrally located pitchlock screw, and a four-way metering actuator valve. The pitch change actuator is the prime mover for blade angle change. The translating motion of the cylinder is converted to rotary motion at the blades through an integral yoke on the cylinder and a trunnion attached to the base of each blade. The blade rollers are mounted on a pin which is attached to the butt of the blade (trunnion) and eccentric to the blade retention axis. Translation of the yoke moves the eccentric roller and converts the linear motion and force of the actuator to a rotary output at the blades.

To change blade pitch, a pressure signal is generated by the propeller control and converted to a rotary signal by the servo and ballscrew (Operation of the screw and ballscrew is described in Section 3.1.2). This rotary input is applied to a pitchlock screw. To increase pitch, the rotary signal drives the screw rearward a small amount relative to the cylinder assembly. This rearward motion temporarily decreases the pitchlock gap and moves the fourway valve rearward relative to the valve housing and sleeve. The valve stroke directs the supply oil to the increase pitch chamber of the actuator, and ports the decrease chamber to drain. This causes the actuator cylinder to move forward to increase the blade pitch toward feather. The motion of the actuator to increase pitch also carries the pitchlock screw forward returning the four-way valve to null and re-establishing the pitchlock gap. Thus, the pitchlock gap, in steady state operation, is independent of actual blade angle. The no load pitchlock gap is the equivalent of .6° of blade angle, approximately .762mm (.030in) and is always ready to prevent uncontrolled decrease pitch if oil pressure is lost. To decrease pitch the rotary input signal is reversed.

3.1.2 Servo Ballscrew & Delta Press. Reg. Valve

The rotary input to the pitchlock screw comes from the servo and ballscrew (Figure 3.3). The servo consists of a half area actuator with supply pressure always acting on the smaller area. The control or metered pressure is normally half supply pressure and acts on the twice area side of the servo. The servo piston is attached to the screw element of the ballscrew through a set of ball bearings. The nut element of the ball screw is grounded with the stationary element of the servo. Axial motion of the servo piston is converted to both rotary motion of the ballscrew and an axial translation of the screw. This output drives the pitchlock screw through a sliding spline thus obtaining an equivalent of a pure rotational motion of the pitchlock screw.

The metered hydraulic pressure from the pitch control to the servo is normally half of the supply pressure. For this condition the servo is in equilibrium and there is no output. Increasing or decreasing the metered pressure will cause the servo to translate and the ballscrew will convert this to the rotary signal to the pitchlock screw. The system has been designed such that an increase in metered pressure will cause the servo to call for a decrease in blade angle. Conversely, a decrease in metered pressure will call for an increase in blade angle. Thus, shutting off the metered pressure to the servo will cause the blades to feather.

The servo was designed to have the maximum area possible to improve dynamic response and accuracy. Under normal operating conditions, the metered pressure from the control is approximately half of supply pressure and the force output of the servo is only enough to overcome the friction forces in the system which are less than 1334 Newtons (300 lbs). However, when the feather valve is actuated, metered pressure can drop to drain pressure resulting in a potential force output of the servo of as much as 30700 Newtons (6900 lbs). Another case where high loads could be generated by the servo is when the actuator is driven against the adjustable low pitch stop. In this case, metered pressure could equal supply and the servo would generate an output of almost 26700 Newtons (6000 lbs). The delta P regulating valve controls these loads to less than 6230 Newtons (1400 lbs.) and results in an appreciably lighter and more efficient linkage between the servo and pitchlock screw.

The purpose of the delta P regulating valve is to limit the output force of the servo under extremes of metered pressure. The delta P regulating valve (see Figure 3.3) is a double-acting relief valve. It has the same area ratio as the servo and has metered pressure acting on the larger area and supply pressure on the small area. The valve is preloaded such that for small changes in control metered pressure nothing changes. If the ratio of metered pressure to supply increases beyond 59%, the valve opens (i.e., moves to the right). Pressure to the servo is than regulated between the valve and the fixed orifice to approximately 59% of supply pressure. Conversely, if the signal pressure changes below 42% of supply, the valve moves to the left

3.1.2 (Continued)

and maintains pressure to the servo at approximately 42%. By controlling the relationship between metered and supply pressure in this band, the output force of the servo can also be controlled at the desired level.

3.1.3 Governor & Feather Valve

The only control functions provided by the LAP pitch control system are constant speed governing via a pilot selected schedule and feathering.

The constant speed servo governor is provided which automatically controls blade angle under all aerodynamic conditions. Changing blade angle changes the power absorption of the blades and thereby matching power absorption to input torque at a fixed RPM. (See Figure 3.4)

The governor consists of a flyweight assembly, a pilot valve, and a speeder spring. The flyweights are geared to the propeller shaft so that their rotation develops centrifugal force in direct relation to the Prop-Fan speed squared. This centrifugal force is converted by the assembly to an axial load at one end of the pilot valve. The speeder spring imparts an axial force on the other end of the pilot valve. The position of the pilot valve is controlled by maintaining equilibrium between the flyweight force and speed set spring. In this condition, the pilot valve is at its null position and the metered pressure is at 1/2 supply pressure resulting in a zero force output at the ballscrew servo.

An error in governing speed alters the force balance between the flyweights and speeder spring moving the governor valve off null. This valve stroke results in a change in metered pressure which changes the force balance in the half area servo, thus driving it in the direction of the unbalanced load. The half area servo moves thus changing blade angle in a direction that corrects the speed error and returns the governor valve back to null. For example, an increase in RPM will lower metered pressure causing the blade pitch to increase, slowing the RPM to its scheduled governing speed.

Thus, the blade pitch is continually adjusting slightly about the scheduled speed condition.

The force on the speeder spring can be changed via the input linkage and can thus allow the governor to schedule different propeller operating speeds.

The feather valve enables the control to bypass the governor and feather the Prop-Fan. The feather valve can be actuated mechanically through the input linkage or hydraulically through the electric feather solenoid valve. The electric signal to the solenoid is either a result of a pilot command to feather or activated by the engine overspeed governor.

3.1.3 (Continued)

The basic function of the feather valve is to dump metered pressure to drain. Dumping metered pressure drives the servo and the beta valve and moves the actuator in the increase pitch (feather) direction. In addition, valve actuation blocks main and standby pump flow to the governor system, pressure regulating valve, and orifice pack. With the pressure regulating valve blocked, both main and standby flows are mixed, and the supply pressure is controlled by the high pressure relief valve. The feather valve is spring loaded in the unfeather position, and removal of the input signal will restore the control to normal (unfeathered) operation.

3.1.4 Hydraulic System

The hydraulic system is essentially the same as the 54460 propeller control. The schematic of the pumping system is shown on figure 3.5. The pump housing assembly contains three mechanically driven pumps (main, scavange, and standby) and an electrically driven double element auxiliary pump. The purpose of the two mechanically driven supply pumps (main & standby) is to minimize heat generation during non-transient conditions. In addition, a measure of redundancy is achieved by separating the main and standby flows with two check valves which allows the system to operate with one pump operational. The combined flow of the two pumps is only required during transient high demand conditions. To minimize the power requirements of the pumping system during low demand conditions the standby pump operates at low pressure with its output dumped to drain. Operation of the two stage flow system is controlled by the main and standby regulating valve shown previously in Figure 3.4. The regulating valve operation is illustrated on figures 3.6 and 3.7. It provides the system with a relatively constant supply pressure over a wide range of system flow conditions. The valve balances the force to supply pressure on one end of the valve against a spring and a fixed reference pressure on the other end. The fixed reference pressure is set by the orifice pack which meters a fixed percentage of P_s to the pressure regulating valve. This is accomplished by connecting two orifices in series between supply pressure and drain, and the reference pressure is taken between them.

Under conditions of low flow demand, most of the combined flow of the main and standby pumps is diverted to the pressurized sump through separate metering windows. With an increase in demand supply pressure drops and plunger motion closes the main pump metering window resulting in an increase in supply flow to the actuator. At a maximum slewing condition the plunger has displaced sufficiently to close the main metering window and meter across the standby land. Standby pressure increases and opens its check valve thereby augmenting main pump flow.

A flow switch is located downstream of the standby pump; a positive indication occurs when the pump output is below an acceptable level signifying leakage or a malfunction of the standby pump.

3.1.4 (Continued)

The auxiliary pumps (scavenge & supply) provide ground handling and feather capability in the event of a main hydraulic system malfunction. The auxiliary pumps are separated from the rest of the hydraulic system, during normal operation, via two check valves. The auxiliary pumps are driven by an electric motor on the control.

The pump housing also contains two oil sumps, a pressurized sump and an atmospheric sump. The pressurized sump is located in the rear of the pump housing, upstream of the main pump, standby pump, and auxiliary pump. The pressurized sump provides a boost to the pump inlets in high altitude conditions. The pressurized sump is fed by the scavenge pump, which draws from the atmospheric sump. The atmospheric sump is located in the front of the pump housing and its pressure is maintained with an atmospheric breather.

All output flow is protected from contamination with a filter located in the control. The hydraulic system is protected from excessive pressures with a high pressure relief. This valve also regulates supply pressure when the feather valve is actuated.

The hydraulic oil is cooled by an airframe supplied heat exchanger and circulation pump. For cold temperature starts the heat exchanger is bypassed by a check valve.

3.2 Design Constants and Accuracies

3.2.1 Design Constants

Key dimensional and operational constants were generated at the early stages of the design to coordinate consistent criteria between Design and Analysis. The values include such data as areas, strokes, time constants, and spring rates and gains. The actuator constants are shown on Figure 3.8 and the governor constants are shown on Figure 3.9.

3.2.2 Design Accuracies

The constant speed accuracy of the governor is dependent on the governor pressure gain, the servo area, and system friction as seen by the servo. (A detailed description of the governor pressure gain is covered in Section 5.4) The components of the governor loop friction hysteresis are shown on Figure 3.10. Assumptions on the coefficients of friction and the ballscrew efficiency are based on previous experience with similar hardware. A statistical summation yields a 1380 Newtons (310 lb.) maximum friction load at the servo. The following calculation results in a .205% maximum speed error due to the governor loop friction hysteresis. This error manifests itself as a .205% speed dead band. A change in propeller speed, exceeding the dead band, is required prior to the pitch change system reacting to maintain constant speed.

3.2.2 (Continued)

GOVERNOR LOOP FRICTION HYSTERESIS

$$\%e = \partial Fp(\partial P/\partial Fp)(\partial Xg/\partial P)(\partial Fg/\partial Xg)(\partial Ng/\partial Fg)(100\%/Ng)$$

∂Fp = Max. Delta Force Required To Move Servo = 310#

$\partial P/\partial Fp$ = 1/(Pm Servo Area) = 1/10.22in

$\partial Xg/\partial P$ = 1/(Pressure Gain) = 1/18,000psi/in @ 170°F

$\partial Fg/\ Xg$ = Governor System Spring Rate = 98.31#/in

$\partial Ng/\ Fg$ = 1/(Flyweight Sensitivity) = $1/4.86 \times 10^{-6}$Ng#/RPM

Ng = Governor RPM = 4072 RPM @100% Ng

$$\%e = (310)\frac{1}{10.22}\;\frac{1}{18,000}\;(98.31)\;\frac{1}{(4.86 \times 10^{-6})(4072)}\;\frac{100}{4072}$$

$\%e$ = .205% Max. Speed Error

The calculated maximum speed error is less than the original estimate. The original estimate was used in the dynamic simulation stability and the potential to syncrophase. Since the final estimate is lower than the value used in the simulation, the simulation is considered valid.

3.2.3 <u>Actuator Mechanical Hysteresis</u>

Mechanical hysteresis of the actuator is the amount of servo motion required before blade angle change occurs. The majority of this hysteresis is a function of the backlash and spring rate of the linkage that connects the servo to the beta valve. The spring rate reacts against the friction loads described in Section 3.2.2. The linkage consists of the ballscrew, pitchlock screw, and beta valve connecting rod. Using a statistical summation, the mechanical hysteresis was determined to be ±.25 degrees of blade angle. This is higher than the target hysteresis of ±.15 degrees. The increase was primarily due to higher friction loads at the servo and the relatively soft spring rate of the ballscrew and duplex bearing. Hysteresis values of ±.08 and ±.24 degrees were used for the dynamic analysis. The analysis indicates a decrease in stability margin at the ±.24 degree hysteresis; however, experience with other propeller systems has shown that the actual control system is more stable than analysis predicts. The potential destabilizing effect of the additional hysteresis is counteracted by additional servo friction. If the stabilizing effect of the servo friction is less than anticipated, the

3.2.3 (Continued)

dynamic stability can be improved by decreasing the governor gain. Analysis indicates that a 25% decrease in gain should be sufficient to insure stability during flight conditions. A decrease in governor gain can be obtained by changing the governor metering window configuration and has been provided for in the LAP program.

4.0 Design Requirements

4.1 Detail Requirements

The overall aim of the LAP program is to evaluate the structural integrity and aeroacoustic performance characteristics of the blades. The pitch change and control system must provide those features and characteristics necessary to properly control blade angle and propeller speed. A detailed design specification similar to one expected on a production installation was generated by Hamilton Standard. This specification integrated the entire Prop-Fan Propulsion requirements and was used as the preliminary interface document between the propeller, the hub and retention, and the pitch control system. The actual design specification is included in Appendix A. The following summarizes some of the key specific requirements of the pitch control system.

- Governing Speed: 75-105%, Tip Speed: 183-256 meters/sec. (600-840 feet/sec.)
- Ground Adjustable Feather Stop: 87.5 ± 2.5 Deg (ß 3/4)
- Ground Adjustable Low Pitch Stop: -10 to +40 Deg (ß 3/4)
- In-Place Pitchlock: 1.5 Deg. Max.
- Slew Rate (Max): 9 Deg/Sec
- Steady State Accuracy: .22 % Max. RPM
- Hysteresis: .15 Deg.

4.2 Design Philosophy

The basic design philosophy was to provide a fail-safe propulsion system. This is done by insuring that any malfunction of the propeller will cause the blades to either pitchlock or feather. Section 6.5.2 describes a typical pitchlock case. The basic actuator design is similar to pitchlocking actuator installations on the 14RF and 14SF commuter propellers. All changes in hardware execution or conceptual approach were reviewed to ensure that the operation was fail-safe. A detailed Failure Mode and Effect Analysis (FMEA) was conducted but is not a part of this report.

Since LAP is a demonstration test program, the design effort concentrated on providing hardware with minimum manufacturing and development risk. This was accomplished by using conservative design and construction approaches and by using service proven concepts and existing hardware wherever possible. The

4.2 (Continued)

basic concept for the actuator is used on several aircraft like the deHavilland DHC-7 (24PF). The pitchlock and actuator concept is almost identical to the EMBRAER EMB 120 (14RF) and the deHavilland DHC-8 (14SF). In addition, the ballscrew is identical to the 14SF ballscrew. The control is a modified 54460 which is used on the Grumman E2/C2.

4.3 Airframe/Engine Interface

The Prop-Fan has several interfaces with the engine and airframe. These are:

- Mounting
- Cooling
- Overspeed governor
- Input lever
- Solenoid (electrical signal)
- Aux pump (electrical signal)

Mounting

The propeller is mounted on a modified AND10152-60 propeller shaft as shown in Figure 4.1. Dimensionally, the configuration is compatible with the T56 gear box. The bulkhead will not have to be modified for the higher operating pressure of 3000 psi.

Cooling

Due to the higher normal operating speeds, pressures and flows of the LAP as compared to the 54460, an external heat exchange is required. The mechanical interface is defined on L/O L14325-20 (Appendix B). The airframe supplied heat exchanger must provide oil to the control at 77°C (170°F) maximum after removing 7.68 kilowatts (10.3 horsepower) from the control oil. Inlet and outlet pressure shall vary between 138,000 and 276,000 PA (20 and 40 psi). Normal flow shall vary between .028 and .056 cubic meters/sec (30 and 60 qts/min). During transients or actuation, the heat exchanger bypass valve may drop the heat exchanger flow to zero.

Overspeed governor

An overspeed governor is required to protect the system in case of malfunction of the normal control systems. The engine overspeed governor will

4.3 (Continued)

activate the feather solenoid causing the propeller to increase pitch. When the propeller slows sufficiently, the signal will be shut off and the Prop-Fan may modulate at the overspeed setting. At this point, pilot evaluation is required to determine if the system should be shut down.

The overspeed governor should be wired as part of the electrical feather circuit to the feather solenoid.

Input lever

The control accepts a single mechanical signal from the cockpit to control the operation of the Prop-Fan. Figure 4.2 shows the input lever interface. Section 5.6 describes the function of the linkage and Figures 5.8 and 5.10 describe the schedule of operation.

Feather Solenoid

The feather solenoid will cause the Prop-Fan to go to feather (high) pitch whenever energized. If the solenoid remains energized, the Prop-Fan will go to feather. The solenoid operates on a 28VDC signal and has a resistance of 22.5-26.5 ohms at 24°C (76°F). The electrical connector conforms to MS3102R-12S3P.

Auxiliary Motor and Pump

The auxiliary motor and pump is described in Section 5.11.

5.0 Pitch Control Design

5.1 General Description

The control is required to provide the speed governing and feathering functions required for this program. To minimize cost and development time, it was decided to use an existing propeller control. Based on the type of engine and gearbox anticipated to be used on the LAP, the control selected prior to the start of the program was a modified 54460 control. The 54460 is a hydromechanical control in use on the Grumman E2 and C2 aircraft which was modeled after the Lockheed C-130 and P3 controls. Since the first production unit in 1956, there have been over 11,000 built and they have logged over 73,000,000 hours. The control will provide the constant speed governing function in the range of 75% to 105% of design speed and the capability to either manually or electrically feather the propeller.

The use of a modified 54460 control instead of a completely new and optimized control configuration resulted in certain restrictions on the design. The 54460 control and tailshaft arrangement resulted in a nonmodular design where the pitch change system must be built up inside the hub. Because of physical restraints on the installation, no beta (i.e., direct blade angle) control is provided. However, for ground testing, modification to the system by disabling most control functions permit limited blade angle control via an electrical signal. In addition, the maximum pressures and flows are limited by the 54460 control. This means a large actuator and a low pitch change rate. This, in conjunction with the 54460 transfer bearings, will limit transient response and the ability to synchrophase under turbulent conditions.

An aircraft supplied overspeed electrical signal is required in the event of a malfunction of the on-speed governor. The control will utilize this signal through the feather solenoid to cause the blade angle to increase until the propeller speed does not exceed the overspeed setting and will govern there.

A number of modifications in the control components were necessary to accommodate the increase in RPM, the opposite hand rotation, and different type of actuator. A description of both the existing components and the design changes follows.

5.2 Main, Standby, and Scavenge Pumps

The main, standby, and scavenge pumps are modified E2/C2 pumps. The modifications were necessary to accommodate the change in direction of the pump drive (see section 5.8). This was accomplished by interchanging the drive and driven gears on the pumps and by making a new front cover for the pump housing. Table 5.1 compares the LAP and the existing E2/C2 pumps. The main and scavenge pumps are running about 12.4% higher RPM than the E2/C2. Both the LAP and the present pumps have the same maximum operating pressures.

5.2 (Continued)

The LAP pump will operate continuously at 7.6×10^6 PA (1100 psi) while the present pumps normally operate at about 4.14×10^6 PA about (600 psi). It is felt that for the shorter life required for LAP, the pumps should be acceptable. An accelerated fatigue test is planned for the pumps to justify this assumption.

5.3 Main and Standby Regulating Valve with Orifice Pack

The purpose of the main and standby regulating valve is to provide the system with a relatively constant supply pressure over a wide range of system flow conditions. The valve balances the force due to supply pressure on one end of the valve against a spring and a fixed reference pressure on the other end. Pressure regulation occurs in response to a change in actuator line flow requirements. Under conditions of low flow demand, most of the combined flow of the main and standby pumps is diverted to the pressurized sump through separate metering windows. The separate windows allow the main pump to operate at high pressure while the standby pump operates at low pressure 1.1×10^6 PA (approximately 160 PSI).

With an increase in demand the supply pressure drops slightly causing a displacement of the plunger. The plunger motion closes the main pump metering window, restricting flow to drain, and results in an increase in supply flow to the actuator. When the flow to the system approaches the flow capacity of the main pump, the plunger has displaced sufficiently to completely close the main metering window and begins to meter across the standby land. Under this condition the standby pressure increases to the same value as supply pressure and opens the check valve located upstream of the regulating valve. In this manner the standby pump augments main pump flow to meet high system flow demand while allowing it to operate at lower pressure and heat generation levels during low demand conditions.

There are only two differences between the LAP and E2/C2 regulating valves. The LAP regulating valve must be shimmed at assembly to meet system operating requirements as discussed above. The mating diameters of the plunger and sleeve of the LAP regulating valve are held closer than the existing valve by matching the spool and sleeve. This is done to minimize the effect of leakage across the standby to reference pressure land, which would affect the reference pressure.

Figure 5.1 and Table 5.2 show the valve and summarize its materials and dimensional characteristics. Figure 5.2 shows a typical pressure vs flow characteristic expected for this valve.

The purpose of the orifice pack is to provide the desired reference pressure across the regulating valve under all conditions of system operation. This is accomplished by connecting two restrictors in series with the reference

5.3 (Continued)

pressure being taken between the two orifices. The orifice pack is designed to occupy the space normally reserved for the selector valve of the E2/C2 propeller control which is not required for this design. The orifice pack consists of new hardware with the exception of the restrictors and seals, which are off-the-shelf items. Figure 5.3 and Table 5.3 summarize the orifice pack design details.

5.4 Governor

The purpose of the governor is to maintain a desired propeller speed under all aerodynamic conditions. This is accomplished by maintaining equilibrium between the flyweight force and speed set spring. The flyweight assembly, driven by the transfer bearing gear, rotates in direct proportion to the propeller speed (Figure 5.4). This rotation of the flyweights develops a force as a function of the rpm squared. An error in governing speed alters the force balance between the flyweights and speeder spring moving the governor valve from null. This valve stroke results in a change in metered pressure which changes the force balance in the half area servo, thus driving it in the direction of the unbalanced load. The half area servo indirectly controls blade pitch via the ballscrew, pitchlock screw, beta valve, and actuator (ref. section 3.1). The blade angle change is made in the direction that corrects the speed error returning the governor valve back to null. For example, an increase in rpm will lower metered pressure causing the blade pitch to increase, slowing the rpm to its correct governing speed. The performance data of the governor is listed in Table 5.4.

The governor's ability to accurately govern at a constant speed is dependent on the effective governor pressure gain at the servo. Before motion occurs at the servo to correct speed errors the pressure level must be increased to overcome system friction. The pressure gain of the valve is the increase in metered pressure as a function of valve stroke and is metered pressure leakage.

The pressure gain of the valve without any external leakage is 1.56×10^8 PA/MM (576,000 psi/in); however, the transfer bearing leakage reduces the effective pressure gain to 4.89×10^6 PA/MM @ 77°C (18,000 psi/in @ 170°F). This effect is illustrated on Figure 5.5. An attempt was made to minimize this reduction in pressure gain by reducing the E2/C2 transfer bearing clearances (ref. section 5.7).

With system friction levels based on experience and the above pressure gains the calculated governing accuracy was determined to be .205% (ref. Figure 3.10).

5.4 (Continued)

The flow gain of the valve is the volume of metered flow supplied by the governor to the servo valve as a function of valve stroke (ref figure 5.6). There are two requirements that have to be met by the flow gain. Based on experience and the dynamic analysis of section 3.2 the flow gain should result in a blade angle rate of 2.5 deg/sec/%rpm around the null position. In addition, with the valve wide open the flow of the valve should provide for a slewing rate of the servo equal to or slightly higher than the actuator slewing rate of 9 deg/sec. To limit the magnitude of the dynamic overspeed the system should reach slewing rate as soon as possible. Based on experience, this is a valve stroke corresponding to a 3% speed error. These two requirements result in the stepped window design shown on figure 5.4. Each window is diametrically opposed to eliminate side loading due to fluid momentum forces.

The governor is designed to incorporate as much of the E2/C2 hardware as possible and occupies the same cavity of the valve housing. There are a number of differences between the two governors. The sleeve and servo of the former governor have been removed and replaced with a single valve housing having screened ports. The spool was also redesigned, incorporating a lubrication path to the ball bearing at the speed set end of the governor. Spool lands were also repositioned as required. The flyweights have been machined from existing E2/C2 flyweights, and generate approximately the same force and spring rate as the original flyweights.

Table 5.5 lists the materials used for the governor.

5.5 Feather Valve

The feather valve enables the hydraulic system to bypass normal control signals, from the governor, to feather the propeller (reference Figure 5.7). The feather valve can be actuated mechanically through the input linkage or hydraulically through the electric feather solenoid valve. The electric signal to the solenoid is either a result of a pilot command to feather or activated by the engine overspeed governor.

The basic function of the feather valve is to dump metered pressure to drain. Dumping metered pressure drives the servo and beta valve which moves the actuator in the increase pitch (feather) direction.

In addition, valve actuation blocks standby and main pump flow to the governor system, pressure regulating valve, and orifice pack. With the pressure regulating valve blocked both main and standby flows are mixed and controlled by the high pressure relief valve. The valve is returned to normal operation via the spring loaded spool.

The feather valve hardware is identical to the E2/C2 except for one seal groove on the sleeve that was removed to accommodate plumbing changes necessary for the LAP control design (see Table 5.6).

5.6 Governor and Feather Linkage

Figure 5.8 is a schematic representation of the control linkage, which is identical to that of the E2/C2 with the exception of the governor speed-setting cam.

The governing range is 1273 to 1783 RPM, (75 to 105%), as defined in section 5.4.3, (Appendix A) of the Design Requirements for the SR-7L Propeller. Figures 5.9 and 5.10 give a comparison of the operating schedules for the E2/C2 and the LAP feather and speed-setting cams. The total range, reverse to feather, remained the same as the 54460. The governing range, however, was extended into the beta range of the E2/C2 schedule in order to allow for more degrees of input rotation per percent change in RPM. There are no provisions for beta control.

The aircraft interface is the control input lever. The input schedule was designed to be linear over the required 30% speed variation, with 2.5356° of input lever rotation equivalent to a 1% change in RPM. The input lever rotation is transmitted to the speed-setting cam by a set of spur gears such that 1° of input rotation results in 2.8824° of cam rotation. The speed-setting cam rise is translated to the speeder spring by the same series of levers utilized in the E2/C2 control, where 2.54mm (.1 inch) of cam rise yields 3.238mm (.12746 inches) of speeder spring displacement. In order to obtain the speeder spring loads that would keep the governor at null, for an operating range of 75-105% RPM, the speed-setting cam had to be designed to incorporate a total rise of 3.5738mm (.1407 in). Beyond the governing range the cam radius remains constant.

Mechanical feathering of the system is accomplished through the same feather cam and linkage presently utilized in the E2/C2 valve housing. Rotating the control input lever to the feather position causes the feather cam to activate the feather linkage and mechanically displace the feather valve spool. Reverse is obtained by setting the speed-setting cam for 105% RPM and limiting the fuel flow to the engine. The reduction in fuel flow prevents the engine from supplying enough power to the system to allow it to rotate at 105% RPM. In this condition, the governor runs underspeed and causes the servo to call for a lower blade angle. The system will continue to decrease pitch until it reaches the low pitch stop and will remain in reverse until the governor runs overspeed and signals for increase pitch. This overspeed condition can be accomplished by increasing engine power until the propeller speed is above 105% RPM or by lowering the governor setting to a value below the propeller RPM in reverse.

5.7 Transfer Bearing and Seals

5.7 1 Transfer Bearing

The transfer bearing supplies hydraulic fluid to the rotating propeller from the stationary pump housing. The transfer bearing has three passage across

5.7.1 (Continued)

the rotating interface, supply pressure, metered pressure, and drain pressure. The transfer bearing configuration is identical to the E2/C2 design except for the diametral clearance at the metered pressure land which was reduced, per Figure 5.11, to minimize the leakage effect on pressure gain (ref. section 5.4). The E2/C2 has a history of seizing at land A. This condition is created by the combined effect of minimal lubrication, from using this land to transfer drain pressure, and radial barrel deflections changing the diametral clearance of the bearing. The use of this land for metered pressure increases the lubrication flow and the barrel support ring (see section 5.7.2) stabilizes the diametral bearing fit. These two changes from the E2/C2 design will minimize the possibility of a bearing seizure. A test was conducted to determine the effects of the clearance change and rpm increase (1130 rpm vs. 1698 rpm) on the transfer bearing. Preliminary analysis of the test indicates that this configuration will be successful.

The bearing inner sleeve incorporates a pump drive gear. The gear was modified from the E2/C2 design to accommodate the change in speed and direction of the propeller while still maintaining approximately the same pump output (ref. section 5.8).

5.7.2 Rear Barrel Support

The rear barrel support was stiffened in order to prevent the radial deflections of the barrel from reducing the transfer bearing clearances. Using deflections for shells and vessels as a means of calculating stiffness, the stiffness of the rear barrel support was found to have increased by a factor of 3.59 over the present E2/C2 support. Stiffening the rear barrel support also enabled the use of the same lip seal in the rear as in the front of the control.

5.7.3 Lip Seal

The lip seals (Figure 5.12 and Table 5.7) provide a dynamic seal between the pump housing cavities and the atmosphere. The seals incorporate helical flute located on the sealing face of the lip which prevent oil leakage by providing a pumping action back into the sump. The direction of these flutes was originally designed to seal with a right hand rotation propeller. With left hand rotation the front seal will be operating in a direction that will permit some leakage. To avoid the cost of new lip seal dies, tests were run to determine if the extent of the leakage problem is significant. This test also determined the effect of operating at an increase in speed. Initial tests have shown that the front seal runs hot and requires additional cooling and lubrication. Features have been added to the final design to correct this situation.

The stiffened rear barrel support (ref. Section 5.7.2) enabled the use of the front lip seal at the rear of the control. In this new location the lip seal will have the proper orientation for left hand rotation and provide pumping action back to the sump.

5.8 Gearing

5.8.1 Description

The configuration of the LAP geartrain is similar to the E2/C2 geartrain. The system was designed with the intention of keeping the speeds of the pumps and governor as close as possible to the present speeds of the E2/C2 components. The system was also designed to accommodate left hand rotation, which is opposite of the E2/C2, with minimum control modifications.

Plumbing changes to the control were minimized by maintaining the same flow direction in the LAP pumps as in the E2/C2 pumps. This was accomplished by reversing the input gear on the pumps. Figures 5.13 and 5.14 illustrate the changes that were made to the LAP pumps in comparison to the E2/C2 pumps.

The transfer bearing gear, used to drive the pumps and idler, was enlarged in order to achieve the desired gear ratios. This was accomplished by machining off the old gear and riveting on a new gear, as indicated in Figure 5.15. Twelve 1/8 in. rivets were used and the resulting stress on each rivet, based on a transfer bearing gear torque of 89NM (789 in.-lbs), is 10^7PA (1450 psi).

The differential geartrain, used to run the governor and for beta feedback in the E2/C2 control, was replaced by a compound idler gear. To minimize modifications to the governor assembly, the governor gear utilized in the E2/C2 control is incorporated into the system and the resulting increase in RPM is only 15.78%.

The idler gear is held in place by an eccentric support shaft, located in the same position on the control as was the differential gear shaft. Anti-torque for the idler is achieved by a dowel pin pressed into the gear support. The bearings used on the support are identical to the ones found on the 14RF/14SF control and are preloaded by a wave spring with a working load of 337-365N (75-82 lbs). Lubrication for the bearings is provided by plumbing in the support shaft, (Figure 5.16) and the expected life for these bearings is 19500 hours.

5.8.2 Stress Calculations

The control gears have been sized per allowables established by Hamilton Standard for gears in this type of application based on our experience in both large aircraft gearbox design and for control hardware, such as the 54460 control. The stress levels established represent a X - 3.0 sigma deviation from data, or 1 failure in 1000.

The gear material for the transfer bearing, pumps, and idler is AMS 6260, all carburized with the exception of the idler gear that meshes with the governor gear (Table 5.8). This is because the governor gear is not presently carburized. The gears are splash lubricated with MIL-H-5606.

5.8.2 (Continued)

The dynamic hertz and dynamic being stresses in the gear teeth were determined using the following formulas;

$S_{cs}=(.70(W_t)(1/PD_1+1/PD_2))/(L(\sin PA)(\cos PA)(1/E_1+1/E_2))$

where S_{cs} = static hertz stress
W_t = tangential tooth load
PD_1 and PD_2 = pitch diameters of mating teeth
PA = pressure angle
L = min. axial engagement
E_1 and E_2 = modulus of elasticity

$S_{CD} = W_d/W_t\ (S_{cs})$

where S_{CD} = dynamic hertz stress
W_d = dynamic load

$S_{BD} = 1.5(W_t)(K_t)/FW(x)$

where S_{BD} = dynamic bending stress
K_t = Heywood stress concentration factor
FW = min. face width
x = Lewis geometry factor

Table 5.9 lists the gear data for the gears used in the control and the calculated stress levels.

5.9 Filters

The main and standby flows are protected by a double element filter located in the valve housing. The filter is the same as the one used on the E2/C2 control. The filter consists of a phosphor bronze screen mesh (120 X 120) with a .081mm (.0032 inch) nominal screen opening.

5.10 High Pressure Relief Valve

A high pressure relief valve is provided as a backup to the pressure regulating valve. It only operates during start-up and shut-down or if the pressure regulating valve malfunctions. The high pressure relief valve is set to operate between 7.93×10^6 and 9.48×10^6 PA (1150 and 1375 PSI). The valve is identical to the present E2/C2 hardware. The valve design details are summarized on Figure 5.17.

5.11 Auxiliary Pump and Motor

The auxiliary pump and motor are the same hardware used on the E2/C2 control. The pump output is used for ground handling and to feather the propeller in case of main and standby pump failure. The auxiliary pump supplies hydraulics to the actuator during static operation. The auxiliary pump is a two element pump. The main element supplies .0132 M^3/MM @ 9.48 x 10^6 PA (14 qpm at 1375 psi) to the supply system and the second element acts as a scavenge to return drain oil from the atmospheric sump to the pressurized sump. The auxiliary pump is driven, through a geartrain, by a 200 volt 400 Hz 3 phase electric motor. The motor limits the duty cycle to 20 seconds on at 3350 watts (4.5 horsepower), and 15 minutes off.

5.12 Pump and Valve Housings

The pump housing is the same as the E2/C2 housing except for the hydraulic plumbing in the transfer bearing area. The housing is made of cast magnesium.

The valve housing is cast aluminum and is the same as the E2/C2 housing except for clearance cuts made to accommodate the modified gear support.

6.0 Pitch Change Actuator Design

6.1 General Description

The pitch change actuator is the prime mover for blade angle change. The pitch change actuator consists of a stationary piston and a translating cylinder. The translating motion of the cylinder is converted to rotary motion at the blades with an integral yoke on the cylinder and a trunnion attached to the base of each blade. The trunnion design includes a pin mounted roller offset from the center of the blade retention. The stationary piston is grounded axially to the hub with a retaining nut.

The force exerted by the trunnion roller combined with its rolling motion during pitch change creates a tangential friction force on the wear plates. This force at each actuator ear results in a twisting moment on the cylinder. This moment is reacted by an anti-torque quill which grounds the cylinder rotationally to the dome. The quill also acts as an indexing device to ensure alignment of the actuator ears with the trunnion rollers.

The main components of the actuator are shown on Figure 6.1 and the materials selected for the main structural elements are shown on Figure 6.2.

This type of pitch change actuator and material components are typical of several Hamilton Standard production actuator designs.

The cylinder and bulkhead assembly pilots on Torlon wear strips at both ends of the actuator to prevent metal to metal contact at the sealing surfaces. The actuator was designed so that under the worst tolerance condition, including deflections due to pressure loads, there will be a minimum of .0127mm (.0005 in.) radial clearance between the main piston seal lands and the cylinder ID. (ref Figure 6.3)

The dynamic seals for this design were chosen for their successful performance in similar applications. The ID seals have a "D" cross section to prevent rolling in the seal grooves. Teflon backup rings are located on the atmospheric side of the seals to control the extrusion gap. This combination is used successfully on the 14RF and 14SF commuter propeller actuators.

The main piston seal consists of a double-delta seal loaded by a standard O-ring. This seal was designed for the maximum operating radial extrusion gap of .762mm (.030 in.) and at an operating pressure of 9.48×10^6 PA (1375 psi). The seal material, Turcon/Moly, was chosen for its resistance to extrusion, low friction, good wear characteristics, and its ability to seal under the operating range of the actuator. This type of configuration has been tested on the 54H60 dome 1.0mm (.040 in.) rad. extrusion gap with good results after 30,000 cycles.

6.2 Servo and Ballscrew

The servo and ballscrew convert a hydraulic signal from the control to a rotary input to the pitchlock screw (see Figure 6.4). For example, for an underspeed condition the governor would increase metered pressure which in turn changes the force balance in the servo driving it forward. The forward motion of the servo, which is axially attached to the ballscrew through a duplex ball bearing, rotates the ballscrew. The ballscrew is directly connected to the pitchlock screw through a splined quill and the rotary motion of the ballscrew is imparted to the pitchlock screw. The pitchlock screw translates forward displacing the beta valve which routes hydraulics to the decrease side of the actuator. Speed is increased to the set speed of the governor nulling the governor valve and returning the force balance in the servo. For an overspeed condition this process is reversed. The governor at null regulates metered pressure at one-half of the supply pressure, therefore to maintain a force balance at the servo in an on-speed condition the area of the metered pressure side of the servo was designed to be twice that of the supply pressure side.

The servo areas were sized to minimize the loss in governing accuracy due to the low effective pressure gain of the governor caused by transfer bearing leakage. As a result, the servo areas are oversized for load capacity requirements. Under normal operating conditions, the metered pressure from the control is approximately half of supply pressure and the force output of the servo is only enough to overcome the friction forces in the system which are about 1335 Newtons (300 lbs). However, when the feather valve is actuated, metered pressure can drop to drain pressure resulting in a potential force output of the servo of as much as 30,700 Newtons (6900 pounds). Another case where high loads could be generated by the servo is when the actuator is driven against the adjustable low pitch stop. In this case, metered pressure could equal supply and the servo would generate an output of almost 26,700 Newtons (6000 pounds). In order to incorporate an existing ballscrew that is currently used on the 14RF and 14SF propeller controls, a delta-P regulating valve was added to the system. This valve prevents overloading of the ballscrew by limiting the difference in supply pressure from metered pressure. The delta P regulating valve controls the ballscrew loads to less than 6230 Newtons (1400 pounds).

The servo is similar to the 14SF and 14RF designs except for the double seal configuration separating supply pressure from metered pressure. (Figure 6.5). This additional seal will prevent supply pressure leakage from increasing metered pressure which would drive the actuator toward low pitch creating a possible overspeed condition. The sleeve and piston are heat treated steel (AISI 4340)(See Figure 6.6). Running surfaces are shot peened and chrome plated. Rulon glide rings are used to pilot the piston in the sleeve. Glass filled teflon slipper seals are used in all dynamic sealing locations to minimize friction. The servo is retained in the valve housing with six bolts. Backlash in the system is controlled by preloading the duplex bearing and ballscrew. The ballscrew is mounted on a spherical washer to provide self-centering and to prevent binding due to misalignment between the ballscrew support and duplex bearing centers.

6.3 Pitchlock Screw

The pitchlock screw assembly consists of the rotating pitchlock screw, a pitchlock support (nut), an adjustable low pitch stop, a drive quill from the ballscrew, and an adjustable connecting rod to drive the beta valve. Figure 6.7 shows the pitchlock screw assembly. The pitchlock assembly converts the rotational signal from the ballscrew to axial motion of the beta valve. The screw maintains a constant gap with reference to actuator ground independent of blade angle. This gap is set to .660 - .762mm (.026 - .030 inch) with the beta valve at hydraulic null (i.e., metered pressure equal to half supply). Loss of hydraulics will cause the actuator to shift thru this gap locking the actuator and transferring the blade twisting moment load directly to ground.

For normal operation the pitchlock screw is driven thru a quill shaft from the ballscrew. The quill shaft accommodates any angular or radial misalignment due to manufacturing tolerances between the ballscrew and the pitchlock screw. The quill shaft has a sliding spline at the pitchlock screw to accommodate the relative axial stroke of the ballscrew. The sliding spline on the pitchlock screw quill is nitrided and coated with a dry film lubricant. The mating internal spline is chrome-plated. The fixed spline on the quill is also nitrided. This combination for the quill shaft was selected for its compatibility with the existing ballscrew spline and is based on experience with similar applications.

The pitchlock screw and nut threads are 30° modified stub acme threads with a 55.626mm (2.19 in.) P.D. and a 12.7mm (.5 inch) pitch (Table 6.1). This lead was selected to match the axial stroke of the actuator to the rotational output of the ballscrew. The screw is designed to prevent the actuator from backdriving under positive load with a coefficient of friction of .03. A similar screw on the 14RF and 14SF propellers was designed at a coefficient of .05. These screws were tested under various conditions including a full vibration spectrum. For light loads, under 200 lbs., the 14RF and 14SF screw showed some slippage when shaken at their natural frequency. Once the loads were increased above this value, all slippage stopped. The minimum load expected in flight is 164,600 Newtons (37,000 lbs.) at 25° ß 3/4, Sea Level Standard Day, 75% RPM. In addition, the screw on the LAP has been designed to be less susceptible to vibrational loads than those used on the 14RF and 14SF propellers.

The screw and nut are carburized steel to prevent wear and to maintain good surface quality under all operating conditions. The screw and nut are a matched set, lapped together so as to limit the maximum backlash to .0508mm (.002 inches) and the maximum driving torque to .565NM (5 inch lbs.) under a 66.7N (15 lb.) axial load. The pitchlock screw and support were analyzed using a finite element body of revolution computer program (H727). (This program is similar to H561, see Section 6.2) The sizing criteria was a proof load of 140% overspeed including a 1.2 contingency factor. The stress levels for this condition were compared to the allowable yield strength of the material.

6.3 (Continued)

The pitchlock screw is attached to the beta valve through the connecting rod. The effective length between the screw and beta valve is adjusted by a screw mechanism on the rod assembly at the screw end. This adjustment allows the pitchlock gap calibration previously described. The rod is supported at either end by spherical bearings which accommodate any radial or angular tolerance between the screw and valve. The first critical speed of the rod is slightly less than twice the operating speed (1700 RPM).

The pitchlock screw assembly also incorporates the adjustable low pitch operating stop. This stop limits the minimum allowable blade angle by limiting the rotation of the screw with respect to the nut. This prevents further motion of the servo, screw, and beta valve in the decrease pitch direction. The stop is ground adjustable from -10 degree blade angle to +40 degree blade angle. For flight conditions, the stop is a safety device which limits the minimum flight angle to a specific level independent of any other signal including a control failure. For reverse thrust testing, the stop will be the reverse blade angle.

The stop is matched to the screw and nut to ensure proper timing and engagement at the stop surface. The stop is designed to react both the inertial load of the system when hitting the stop plus the stall load of the servo. The control the inertia loads, one element of the stop is machined as a cantilever off the pitchlock support (nut) and its spring rate will soften the impact loads.

6.4 Beta Valve

The beta valve is a four way, force compensated, spool valve with direct mechanical feedback with the actuator (Figure 6.8). The steel sleeve and spool are manufactured as a flow matched set. Their running surfaces are carburized to prevent erosion wear at the metering lands (Figure 6.9).

Motion of the valve aft of the null position meters oil to the increase pitch side of the actuator cylinder, and allows flow out of the decrease pitch side of the cylinder. This causes blade motion to increase pitch through the yoke and blade trunnion rollers. Since the valve's inner spool is attached to the actuator cylinder via the connecting rod, motion of the actuator brings the beta valve back to the null position. Conversely, forward motion of the valve moves the blades toward decrease pitch. When the valve lands are near the null position, the proper blade angle is attained, and the bulk of the supply flow is bypassed by the pressure regulating valve as described in section 5.3.

Pressure gain and flow gain are the two important hydraulic characteristics of the valve (Figures 6.10 and 6.11). The pressure gain of the beta valve determines the dead band hysteresis of the actuator. To minimize this hysteresis the pressure gain was made as large as possible within the constraints of the desired flow gain and envelope restrictions. Experience with

6.4 (Continued)

this type of valve combined with a detailed analysis of its structure restricts the maximum allowable window to 60% of the sleeve internal circumference. The resulting peripheral length of the window, combined with tight diametral clearances and an underlapped spool, yields a pressure gain of 1.25 x 10^8 PA/MM (460 psi/.001 in). Flow gain is the change in supply flow to the actuator from the beta valve as a function of valve stroke (Figure 6.11). A flow gain of $3.5 \times 10^6 mm^3$/sec/mm (5.433 in^3/sec/.001in.) was required to obtain the desired time constant of .020 sec. Experience with propeller control systems have shown that this time constant is required for stable dynamic response. A detailed analysis of the dynamic response can be found in section 3.2.

Force compensation has been provided by contouring the spool. The contours will reduce the closing force, inherent in four-way valves, to 8.9 Newtons (2 lbs.). The contour geometry required is based on both theoretical analysis and empirical data derived from similar hardware. The force thus obtained is appreciably less than the 53 Newtons (12 lb.) expected from the valve seal friction and will have a negligible effect on hysteresis. These seals are required at both ends of the spool to prevent valve leakage from filling the dome.

The beta valve, ballscrew servo, and delta-P regulating valve are packaged in an aluminum valve housing which is mounted in the stationary piston I.D.. The valve housing acts as a manifold that routes hydraulics from the hub to the beta valve and then to the actuator. In addition, there are plumbing lines that connect the servo and delta-P regulating valve hydraulics to the hub.

The actuator dome incorporates high and low pitch pressure sensors that will be used to analyze the control and determine actual blade loads during flight. Low pitch and high pitch pressure are routed to the pressure transducers at the front of the dome through tubes that are connected to the valve housing. Locating the pressure transducers in the front of the dome provides better accessibility for maintenance.

6.5 Structural Analysis

The actuator was structurally analyzed with a finite element shell of revolution computer program (H561). This program performs linear elastic analysis of axisymmetric structures which can be modeled as collections of thin shells of revolution. Basically, the finite element displacement method is used to convert the mathematical model of the physical problem from one of solving the simultaneous partial differential equations of thin shell theory to one of forming and solving a large set of simultaneous linear algebraic equations. This algebraic set represents equilibrium and compatibility equations at a number of discrete points in the structure, called nodes, written in terms of unknown displacement and rotation values at these same points.

6.5 (Continued)

Output consists of printed listings of displacement, load and stress values at a large number of points throughout the structure. Plots of input geometry, deflected shape, and stress distributions are also available to assist in interpretation of the results.

There were four major considerations in determining the wall thicknesses of the actuator.

1. allowable stress levels for the materials selected.

2. radial extrusion gaps at dynamic seal surfaces

3. axial deflection in the pitchlock condition

4. radial thread engagement of the retaining nuts

6.5.1 Typical Structural Analysis

1. pressure spectrum on the increase side of the actuator with the cylinder at mid-stroke.

2. pressure spectrum on the decrease side of the actuator with the cylinder at mid-stroke.

3. pressure on increase side of actuator with cylinder against the feather stop.

4. pressure on decrease side of actuator with cylinder against the reverse stop.

The results of a sample calculation for the actuator piston is shown in Figures 6.12 thru 6.16. This is typical of the analysis performed on each structural item of the actuator assembly. The loading condition of case 3 (above) for the stationary piston is illustrated on Figure 6.12. Figure 6.13 is a plot of the input geometry for the H561 model (ref. section 6.2). Figure 6.14 is an exaggerated plot of the deflected piston shape showing the areas with the largest radial deflections. These deflections were used to determine the actual operating piston clearances. The high stress areas in the piston and their respective stress concentration factors are shown on Figure 6.15. The high stress areas are below the allowable S/P values for the piston (ref. Figure 6.16). The S/P allowables are based on the load spectrum and the ma- terial allowables. A typical derivation of S/P allowables can be found in Section 6.11.

6.5.2 Axial Deflection of Actuator

In designing the actuator, axial deflections were of prime concern. The axial deflections are a result of pressure induced deflections in the cylinder and load dependent deflections of the blade, trunnion, and actuator ears. The axial deflections affect operation in three ways.

1. There is a shift in actual blade angle with respect to null of the beta valve. During governing this has no effect on operation as the actual blade angle is not required to maintain constant rpm.

2. The adjustable low pitch stop blade angle setting changes with actuator deflection. A typical flight load case (ref. fig.6.17) of 318,000 Newtons (71,500 Pounds) would yield .762mm (.03 in.) axial deflection at the pitchlock screw and results in a .56 degree increase in the low pitch stop blade angle.

3. Actuator deflections add to the loss of blade angle in a pitchlock condition. Figure 6.18 tabulates the total blade angle loss, including the pitchlock gap, for various loading conditions. Figure 6.19 shows the effective drag or loss in thrust vs. loss in blade angle for a pitchlocked propeller operating in a typical cruise condition. This drag on the aircraft is not considered to be significant.

6.6 Trunnion Assembly

The trunnion assembly consists of an eccentric arm (trunnion) bolted and doweled to the bottom of the blade shank. On the end of the trunnion is a crowned roller bearing that engages the yoke ears of the actuator. The linear motion of the actuator working against the roller on the trunnion arm causes the blade to change pitch in the barrel retention bearing. (figure 6.20)

The blade trunnion rollers and wear plates are similar to the design found on the 14SF and 14RF propellers. The angular misalignment between the trunnion shaft and yoke, under load, requires a crowned roller to control the location of the hertz contact pattern on the wear plates. This misalignment was minimized by matching the stiffness of the yoke with the combined stiffness of the trunnion and blade retention. The crowned roller is thru hardened 52100 bearing steel. A teflon filled polyimide journal bearing, which interfaces with the chrome plated trunnion shaft, is pressed into the roller. The roller assembly is retained with a pin and sleeve combination (ref. figure 6.21 & figure 6.22)

6.6 (Continued)

The loads used for sizing the trunnion are summarized in Table 6.2. Figure 6.23 lists the critical stress areas of the trunnion roller assembly. The wear plate subsurface shear stress is comparable to similar pitch change roller designs tabulated in Table 6.3. The comparison requested by NASA between the LAP and Quiet Clean Short-Haul Experimental Engine (QCSEE) roller designs is illustrated on figure 6.24. The stressing on the LAP is shown to be greater than the QCSEE but the higher hardness of the wear plates on the LAP compared to the hardness of the cam tracks on the QCSEE yield higher subsurface shear stress allowables. In addition to the higher allowables, the LAP roller is also immersed in oil to reduce wear while the QCSEE roller ran dry.

The actuator ears were stressed using the pressure spectrum developed to analyze the cylinder and piston. This is conservative since the actuator ears are not subjected to the start-stop pressure relief valve load. The resulting S/P values are summarized on figure 6.25. The S/P allowables are based on the load spectrum and material allowables. A typical derivation of the S/P allowables is illustrated in Section 6.11.

6.7 Delta P Regulating Valve

The purpose of the delta P regulating valve is to limit the output force of the servo under extremes of metered pressure. The servo was designed to have the maximum area possible to improve dynamic response (see Section 6.4). Under normal operating conditions, the metered pressure from the control is approximately half of supply pressure and the force output of the servo is only enough to overcome the friction forces in the system which are about 133 PA (300 lbs). However, when the feather valve is open, metered pressure can drop to drain pressure resulting in a potential force output of the servo of as much as 30,700 Newtons (6900 lbs). Another case where high loads could be generated by the servo is when the actuator is driven against the adjustable low pitch stop. In this case, metered pressure could equal supply and the servo would generate an output of almost 26,700 Newtons (6000 lbs). The valve controls these loads to less than 6230 Newtons (1400 lbs) and results in an appreciably lighter and more efficient linkage between the servo and pitchlock screw. Lower loads also allow the use of existing ballscrew hardware presently used on the 14SF and 14RF propeller controls.

The delta P regulating valve (see Figure 6.26) is a double-acting relief valve. It has the same area ratio as the servo and has control metered pressure acting on the larger area and supply pressure on the small area. The valve is preloaded such that for small changes in control metered pressure nothing changes. If the ratio of metered pressure to supply increases beyond 59%, the valve opens (i.e., moves to the right). Pressure to the servo is then regulated between the valve and the fixed orifice to approximately 59% of supply pressure. Conversely, if the signal pressure changes below 42% of supply, the valve moves to the left and maintains pressure to the servo at approximately 42%. By controlling the relationship between metered and supply pressure in this band, the output force of the servo can also be controlled at the desired level.

6.7 (Continued)

Figure 6.27 shows the characteristics of this valve and the force output of the servo for various conditions. The minimum output is approximately three times the maximum expected servo friction forces. The maximum output is within the structural capability of the linkage.

The valve is installed in the actuator on the center line of rotation to eliminate centrifugal loading on the valve. The preload and areas of the valve are sized to permit the valve from stroking at the normal pressure variations and to provide enough force to overcome friction or break small chips. The valve is carburized to prevent wear and is further protected by screen meshes on both metered and supply flow. Both screens have .1676mm (.0066 in) diameter holes and 28% open area. The supply pressure screen also acts to prevent chips from getting to the beta valve. The valve is internally damped by an orifice between the metered pressure signal and the metered pressure load cavity. The materials used for the delta P valve are shown in Figure 6.28.

6.8 Actuator Load Definition

6.8.1 Total Twisting Moment (TTM)

The forces that are reacted by the pitch change system consist of a Centrifugal Twisting Moment (CTM) and an Aerodynamic Twisting Moment (ATM). In addition, when changing blade angle, the actuators must overcome blade retention friction (FTM).

The CTM is the moment produced by the centrifugal forces acting on the blade which tends to align the blade parallel to the plane of rotation. The CTM is the dominant loading component. Figure 6.29 is a plot of CTM at 100% RPM. CTM is proportional to RPM squared. A positive CTM is a load tending to twist the blade toward a lower blade angle.

The ATM is the result of the summation of unbalanced aerodynamic forces about the pitch change axis. The ATM is a function of the effective air speed, shaft horsepower, air density, blade angle and RPM. A positive ATM is a load toward increase blade angle. Figure 6.30 shows typical ATM characteristics for the type of blade used on LAP (SR-7). The ATM for an SR-7 type blade is usually less than 15% of the CTM and can either add to or subtract from it. Increasing power or decreasing speed decreases the summation of the CTM and ATM.

The FTM is a constant friction torque vs. RPM due to blade bearing friction and blade seal friction. FTM is a function of the centrifugal load, blade pitch radius and coefficient of friction (assumed to be 0.005 based on previous experience). This relationship has been determined from past experience with similar blade retentions. The FTM is the same for all blade angles and always adds to the load required to move the blades. The FTM at 100% speed is 1256 in-lbs/blade (1.419×10^5 MM-Newton/blade) and is proportional to RPM squared.

6.8.2 Actuator Efficiency

The piston area output force must be combined with the actuator efficiency to determine the effective actuator capacity. The efficiency is dependent on the actuator seals friction, antitorque spline friction, trunnion journal bearing friction, and the effective trunnion arm radius which changes with blade angle. Figure 6.31 illustrates their combined effects in a plot of efficiency vs. blade angle.

6.8.3 Actuator Margin

Figure 6.32 demonstrates that the effective actuator capacity of the LAP meets or exceeds the three design case requirements. This capacity was obtained with a 6.96×10^{-2} M^2 (108 in.^{2}) area on the increase pitch side of the actuator combined with 76.2mm (3.00 inch) moment arm at the trunnion. The requirements of the decrease pitch side of the actuator are minimal, but to simplify the design its area is approximately the same as the increase side of the actuator 7.35×10^{-2} M^2 (114 in^2).

The stall output is calculated at 7.03×10^6 PA (1020 psi), or 7.24×10^6 PA (1050 psi) at the increase pitch side of the actuator and $.20 \times 10^6$ PA (30 psi) drain pressure on the decrease pitch side of the actuator. The actuator output for a pitch rate of 9 deg./sec. is shown at 4.83×10^6 PA (700 psi) to reflect the following full flow pressure drops through the system.

1.	B-valve windows (supply & drain) -	1.21×10^6 PA	(175 psi)
2.	Check valves and filter -	$.517 \times 10^6$ PA	(75 psi)
3.	Plumbing to B-valve -	$.345 \times 10^6$ PA	(50 psi)
4.	Drain line from B-valve to sump -	$.345 \times 10^6$ PA	(50 psi)
	Total Pressure Drop:	2.413×10^6 PA	(350 psi)

6.9 Sizing Criteria

The first step in sizing the actuator is to determine which operating conditions require blade angle control. The LAP pitch control consists of a single hydraulic actuator with in-place pitch lock. The in-place pitch lock allows loading conditions to exceed actuator capacity without loss of blade angle and the corresponding propeller speed increase. Based on Hamilton Standard's experience with propeller systems, the following conditions were chosen for actuator sizing criteria.

The first criteria is a sea level dive, zero horsepower, maximum velocity, with a 10% overspeed. At this condition the actuator shall be capable of changing pitch. This is an extreme condition and could only happen in an emergency or test condition. In addition, since maximum velocity at zero power could not be maintained, slowing the aircraft would reduce actuator loads. A second overspeed condition looked at is somewhat more realistic but is also conservative. It assumes a more typical flight condition of Vmax. at 1520 meters (5000 ft.) and 2237 kilowatts (3000 horsepower). An extreme

6.9 (Continued)

transient or failure causes the propeller to then overspeed to 125% RPM. The actuator should still be able to hold blade angle using only hydraulics at this condition. Slowing the aircraft would allow the actuator to increase blade angle and bring RPM down to 100% without pitchlocking. The third sizing criteria was actuator slewing capability at a maximum velocity and 100% RPM condition. Since this is an experimental aircraft and a new blade configuration, a 20% margin was applied to all calculated blade loads.

The following table summarizes the actuator design sizing criteria.

Pitch Change Actuator Sizing Criteria

Case	V-Kts EAS	% RPM	Alt.	Kw (SHP)	Criteria
1. Overspeed	333	110%	SLSD	0	Actuator margin
2. V_{max}	333	100%	SLSD	3580 Kw (4800)	Pitch rate = 9°/sec
3. Overspeed	333	125%	1520 meters (5,000 Ft)	2237 Kw (3,000)	Actuator margin

6.10 Stressing Criteria

To structurally analyze the actuator, a life/load spectrum must be determined. Since this is only a test program, Hamilton Standard made an approximation of a typical flight and the resulting loads generated on the actuator. It was conservatively assumed that there would be 10,000 flights or start/stop cycles.

Figure 6.33 represents a typical flight. At startup, there is no load or pressure acting upon the actuator. When the Prop-Fan is started, pressure reaches the high pressure relief valve (HPRV) setting. this pressure load is seen by all internal parts even though there is no external load. The actuator is then unfeathered and a typical flight begins. It was assumed that the operating loads varied from a maximum RPM and effective airspeed condition to a minimum TTM flight condition. This cycle is repeated 50 times each flight varying actuator metered pressure between 60 and 100% of the regulated supply pressure (PRV). The system also operates at the PRV setting with a plus or minus 15% variation in pressure. The condition simulates either pump pulsations on the pressure side or cyclid bending moments on the blade. In addition, the actuator must be capable of meeting proof and burst test requirements. The life/load spectrum is summarized below.

6.10 (Continued)

Pitch Change Actuator
Life/Load Spectrum

Start-Stop Cycles	10,000	$0 - 9.48\times10^6$ PA	(0 - 1375 PSI)
Flight Cycle	500,000	$6.55\times10^6 \pm 1.65\times10^6$ PA	(950 $\pm$ 240 PSI)
Pressure Pulsations	Life	$7.93\times10^6 \pm 1.21\times10^6$ PA	(1150 $\pm$ 175 PSI)
Proof		11.72×10^6 PA	(1700 PSI)
Burst		17.58×10^6 PA	(2550 PSI)

Use of this type of pressure spectrum on all elements of the actuator has been found to be a satisfactory procedure to analyze pitch control systems for the type of conditions actually encountered in operation.

6.11 Typical Material Allowables

Hamilton Standard analysis of low cycle fatigue, where there are a number of loading conditions, is based on Miner's Rule of cumulative fatigue damage. Each condition uses part of the total life of the part in proportion to the applied cycles divided by the allowable number of cycles. For example (see Figure 6.34), both the steady and the cyclic stress (including stress concentration) is calculated for one condition. The effective fully reversed stress is then calculated and the allowable number of cycles is determined from the S/N diagram. The ratio of actual cycles (example = 10,000) over the allowable cycles (example 70,000) is then calculated (example 1/7 = .1428). This is the proportion of fatigue life (14.3%) used by the operating condition. When the summation of all the conditions equals one, all the life is used up. Expressed mathematically:

CUMMULATIVE FATIGUE DAMAGE

$$n/N = n_1/N_1 + n_2/N_2 + n_3/N_3$$

n = NUMBER OF APPLIED CYCLES

N = NUMBER OF ALLOWABLE CYCLES

When using this rule, Hamilton Standard only works to a summation of .5, or 50% of the life. The design material allowables are based on $x - 3\sigma$ fatigue allowables which represent a statistical stress envelope based on test data that results in less than one failure in a thousand test cases for similar type hardware in a similar environment.

6.11 (Continued)

Calculating each individual point for each condition, including any stress concentration, can be very time consuming and lead to possible errors. To simplify the process, Hamilton Standard transforms all of the sizing conditions to a single curve called an S/P curve as shown in Figure 6.35. Since all our loading is pressure related, all stresses are normalized for a single pressure 6.9×10^6 PA (1000 psi). The stress is then calculated for 6.9×10^6 PA (1000 psi) applied load and the S/P (stress/1000 psi) determined. The curve is only good for a particular material and a particular loading condition. For example, Figure 6.35 shows that the actuator cylinder is burst limited for low stress concentration factors (Allowable = 496×10^6 PA (72,000 psi). As the stress concentration gets large, such as in a thread root ($K_t = 4.0$), the actuator is fatigue limited (allowable = 345×10^6 PA (50,000 psi)). Use of curves like this for each component allows almost immediate review of the geometry of a component to determine its structural integrity.

7.0 Dynamic Analysis

7.1 Introduction

A dynamic analysis of the large-scale advanced Prop-Fan (LAP) system was performed to determine the pitch change governor gain needed for acceptable stability and also to investigate the overall dynamic performance of the control system. In addition to small signal stability and larger power excursion transient performance, the expected synchrophaser performance was also investigated. Results show stable transient performance at various flight conditions with the pitch change governor gain chosen and also that acceptable synchrophasing accuracy can be achieved. This analysis was specifically geared toward the investigation of this particular test hardware and was not intended as indicative of future Prop-Fan system performance.

The LAP system dynamic analysis was accomplished with the development and use of a non-linear computer simulation model. This simulation model was needed to help understand the dynamic inter-relationships among sub-systems within the overall LAP system and to relate these dynamic inter-relationships to system stability and overall transient performance. The model was used to determine the appropriate value of pitch change governor gain and to analyze dynamic performance characteristics of this Prop-Fan system. Previous prop-fan and propeller control studies have shown the need to thoroughly analyze and understand system stability, transient overshoot, rate limitations and general transient performance in order to assist in the design of the control hardware and to define the capabilities and limitations of each specific control application.

7.2 Model Simulation

A simplified LAP system block diagram consisting of the Allison Gas Turbine 501-M78 engine and controls, a modified 54460 prop control, a 54460 type synchrophaser, and the Prop-Fan model is shown in Figure 7.1. This functional signal flow diagram shows the input/output relationships of each turboprop sub-system.

The dynamic engine and fuel control models were created by Hamilton Standard with the use of a customer supplied steady state engine cycle deck and fuel control system block diagram. The dynamic engine model block diagram is shown in Figure 7.2. This dynamic engine model format is successfully used by Hamilton Standard to simulate free turbine turboprop engine. Block diagrams of the gas generator fuel control and the electronic engine control are shown in figures 7.3 and 7.4 respectively. The gas generator fuel control provides transient fuel scheduling and isochronous gas generator speed governing and the electronic engine control provides power turbine overspeed limiting, power turbine inlet temperature limiting, and gas generator overspeed limiting.

7.2 (Continued)

The LAP Prop-Fan control is a hybrid control which combines 54460 hardware and commuter prop technology to achieve satisfactory Prop-Fan governing. The resulting control block diagram is shown in Figure 7.5. The Prop-Fan speed governor strokes a pilot valve and flow is sent through a transfer bearing to the servo. The servo stroke rotates a ball screw which commands the blade actuator. The 54460 transfer bearing concept is combined with the servo, ballscrew, and blade actuator technology of HSD commuter propeller controls.

The Prop-Fan model, shown in Figure 7.6 uses power and thrust coefficient maps which are functions of blade angle and advance ratio. The power turbine torque output, accessory torque, and Prop-Fan torque absorbed are summed to determine the torque available to accelerate the Prop-Fan, power turbine, and gearbox inertias to determine Prop-Fan speed.

The synchrophaser model used for this study is a production C-130 electro-mechanical synchrophaser and is shown in block diagram form in Figure 7.7. The model consists of a synchrophaser portion with proportional plus integral control on phase error, an electronic trim portion which provides prop speed derivative compensation and power lever anticipation, and a motor loop which is the electro-mechanical interface to the Prop-Fan control.

Table 7.1 lists and defines the major variables used in the LAP system model.

7.3 Prop-Fan Control Performance

To determine an acceptable pitch change governor gain, the system response to small and large changes in engine speed reference, which is indicative of engine power level, was investigated at various sea level and altitude flight conditions. Figures 7.8 and 7.9 show acceptably stable response to 1.5% step changes in engine speed reference at sea level, static and 10700M (35,000 Ft), 108 M/S (390 KTS) respectively. These conditions are believed to be the least stable of those conditions at which the Prop-Fan is to be tested. Figure 7.10 shows the system response to a 2.0 sec engine speed reference increase from 88% to 100% at sea level, static conditions. Figure 7.11 shows a 2.0 sec engine speed reference decrease from 100% to 88% at the same condition. These large power excursion transients exhibit stable performance at both power levels; however, the power advance of Figure 7.10 shows a Prop-Fan peak transient speed of 108.8% which is caused by the combination of a 10.0 DEG/SEC maximum blade angle rate and a large blade angle excursion needed to absorb the power increase. The maximum Prop-Fan transient speed is 123.5% and occurs when the speed reference change from 88% to 100% is made in 0.5 sec or less. The peak transient Prop-Fan speed is limited by the electronic engine control power turbine overspeed limiter set speed of 120%. If this power turbine overspeed limiter setting is lowered to 109%, the maximum Prop-Fan speed is reduced to 112%. Figure 7.12 shows the simulation responses to 0.5 sec engine speed reference advances from 88% to 100% NG at a sea level, static condition with the power turbine overspeed limit set at

7.3 (Continued)

120% and 109%. These same types of transients were run at 10,700M (35,000 Ft), 201 M/S (390 KTS). A 2.0 sec speed reference increase from 78% to 100% NG is shown in Figure 7.13 and the corresponding 2.0 sec chop from 100% to 78% NG is shown in Figure 7.14. No excessive transient Prop-Fan speed error occurs at these conditions. The adjustable low pitch stop setting was 20.0 DEG Beta 3/4 for all the transients studied for the determination of the pitch change governor gain. Note that 88% NG was chosen for the above sea level, static transients because this is approximately the minimum power required to assure Prop-Fan governing above 20.0 DEG Beta 3/4. The 78% NG setting was chosen for the above altitude transients because this dictates minimum fuel flow.

7.4 Prop-Fan Control Limitations

All engine power setting changes should be input slowly as to avoid large Prop-Fan speed overshoot. Figure 7.12 shows the Prop-Fan response to a 0.5 sec power setting advance from 88% to 100% NG done at a sea level, static condition. The 123.5% Prop-Fan speed peak shown can be reduced to 112.% if the fuel control power turbine overspeed setting is reduced from 120.% to 109.%. Therefore, Prop-Fan overspeed can be limited either by avoiding quick power setting advances or by reducing the fuel control power turbine overspeed limiter setting.

It should be noted at this point that the results of this study are valid only for this Prop-Fan control test hardware and must not be viewed as indicative of Prop-Fan control performance potential. The low blade angle rate, large control deadband, and constant minimum blade angle schedule cause limitations in this specific test hardware application that cannot be generalized for all Prop-Fan controls.

Operation of the control in the reverse regime is severely limited in that the transition into reverse must be done near minimum fuel flow, static conditions to avoid Prop-Fan overspeed and the maximum attainable power at full reverse (-6.0 DEG Beta 3/4) without exceeding 100% NPROP is approximately 130500 watts (175.0 HP) at sea level, static conditions.

Simulation of reversing transients including simple reversing from flight idle and aborted takeoffs were not run because the low pitch stop will not allow the control to attempt these transients. The low pitch stop acts as protection against excessive transient overspeed which may occur if the stop were not there to limit blade angle travel.

7.5 Synchrophaser Performance

Use of a production 54460 synchrophaser in conjunction with the LAP pitch change system is impractical because the Prop-Fan control deadband of this test hardware causes phase error limit cycle of approximately ± 50. DEG. Several possible options to achieve adequate synchrophasing were studied. One option was to increase the synchrophaser gain to decrease the limit cycle amplitude. The limit cycle amplitude can be decreased to ± 26. DEG, as shown in Figure 7.15, if the synchrophaser proportional and integral gains are increased by a factor of eight. This does not yield acceptable synchrophaser performance. An option studied that does yield acceptable synchrophaser performance is the addition of an electronic dither signal to the synchrophaser voltage output, as depicted in Figure 7.7. This dither signal moves the control back-and-forth through the deadband so that, on the average, the control acts as if no deadband exists. If a 2.0 HZ square wave dither voltage signal of ± 2.5V is added which generates an NTRIM signal of ± 8.5 RPM, the limit cycle amplitude is decreased to ± 3.4 DEG. This result is seen in Figure 7.16. The 2.0 HZ NTRIM signal, which is the input to the hydromechanical Prop-Fan control, is the waveform which results from a square wave voltage of ± 2.5V being added to the synchrophaser voltage output, as shown in Figure 7.7. If the dither amplitude is increased to ± 3.5V, the resulting ± 10.75 RPM NTRIM signal reduces phase error to less than ± 0.5 DEG. Figure 7.17 shows that the increased dither amplitude moves both the servo and the blade actuator back-and-forth through their respective deadbands, thus minimizing the effects of both non-linearities. Dither has been successfully used in commuter propeller applications to counteract the negative effects on synchrophasing of prop control non-linearities. The synchrophaser accuracies defined in this report must be viewed as analytical estimates since random atmospheric condition changes and engine power variations which add to synchrophasing inaccuracy were not included in this study. All synchrophaser accuracy investigation was done at 10700M, 201 M/S (35,000. FT, 390 KTS) flight conditions.

7.6 Summary

The results of this study are valid only for this LAP test hardware and cannot be generalized for future Prop-Fan control applications.

The pitch change governor gain recommended as a result of this analysis is 2.5 DEG/SEC/% RPM. Simulation studies show acceptable stability at those operating conditions at which the control system was designed to govern.

Since no minimum blade angle schedule exists in this control, an adjustable low pitch stop is needed to assure that the system will not be forced to govern at low blade angles. Stability margin decreases considerably at low blade angles.

7.6 (Continued)

Changes in power setting must be input slowly to avoid excessive transient Prop-Fan speed variation. Since larger power excursions can occur at sea level than at altitude, the system is much more susceptible to Prop-Fan overspeed at sea level operation.

This control must not be used for gross reversing transients because severe Prop-Fan overspeed could result. All transitions to reverse must be done under static conditions at or near minimum power. Operation in the reverse regime with a minimum Beta 3/4 of -6.0 DEG will only permit approximately 130,500 watts (175 HP) before exceeding 100% NPROP at sea level, static conditions.

The modified 54460 hybrid Prop-Fan control used in the LAP system has a ± .205% RPM governor deadband and a ± .25 DEG Beta 3/4 actuator hysteresis band. These control non-linearities have been found to help Prop-Fan control stability and adversely affect synchrophaser performance. A production 54460 synchrophaser will not accomplish satisfactory synchrophasing accuracy due to these Prop-Fan control non-linearities. The combination of these Prop-Fan control non-linearities and synchrophaser integral control causes a phase error limit cycle of ± 50 DEG.

An effective method of reducing the effect of Prop-Fan control hysteresis and deadband on synchrophaser accuracy is the addition of an electronic square wave dither signal to the synchrophaser voltage output. The phase error variation is reduced to ± 3.5 DEG if a 2.0 HZ square wave dither of ± 2.5V (± 0.5% RPM) is added to the synchrophaser output. This variation in phase error can be further reduced to less than ± 0.5 DEG if the dither amplitude is increased to ± 3.5V (± 0.63% RPM). Random atmospheric disturbances and engine power variations have not been included in this synchrophaser accuracy analysis. The C-130 synchrophaser was arbitrarily chosen for use in this analysis as a typical 54460 type synchrophaser.

8.0 Beta Control

The SR-7L will be tested on the static test rig at Wright Field to explore the stall flutter boundary and in the Modane wind tunnel to explore the high speed flutter boundary. For these tests it is desirable to operate the Prop-Fan with direct blade angle control and not with speed control. These boundaries are established by varying speed over a range while holding blade angle constant. This procedure is repeated for many different blade angles.

To incorporate blade angle control on the LAP, the rotary input to the pitchlock screw from the servo is replaced by a D.C. motor and gearhead. This arrangement replaces the governing mode configuration and is shown in Figure 8.1.

The motor is powered by a remote open loop electronic controller. The controller is designed to pulse power in discrete time increments to obtain an incremental change in blade angle. The actual change in blade angle is dependent on the length of the pulse and the level of friction at the pitchlock screw. Power is transmitted across the rotating interface through two slip rings located on the spinner bulkhead. The motor, which is mounted on the front of the dome, is a D.C. permanent magnet planetary gear motor. The motor operates on 24 volts $\pm$ 2.5 volts at 6500 RPM (no load speed). The planetary gearhead provides a $\bar{7}80$ to 1 speed reduction ratio. A flexible shaft is mounted on the output shaft of the motor and drives the pitchlock screw. The output speed of the motor is a function of pitchlock screw friction. This relationship, expressed in deg/sec of blade angle change vs. screw friction torque, is shown on figure 8.2. The maximum expected torque requirement due to the pitchlock screw friction is 565 mm-N (5 in-lbs). This results in an actuator slewing capacity of 1.36 deg/sec in the beta control mode. This rate is sufficient since there are no dynamic response requirements in the beta mode. The main pump flow could provide 3 deg/sec, therefore the standby pump flow is not required and could be removed from the system during beta control. However, to minimize the modifications necessary to convert from the governing mode to the beta mode, the standby pump will not be removed.

Blade angle control in the governing mode is accomplished by motion of the servo which changes blade angle via the ballscrew, pitchlock screw, and beta valve. This sequence of events is detailed in sections 3.1.1 and 3.1.2. In order to give the motor full control of blade angle the servo and ballscrew must be disconnected from the pitchlock screw by removing the quill shaft. This disables the control signal and allows the motor to work directly on the pitchlock screw without having to fight the servo output.

The servo is loaded against the low pitch stop by setting the governor at 105% speed via the condition lever. This prevents uncontrolled cycling of the servo during beta control operation. Metered pressure is regulated by the delta-P regulating valve at slightly higher than one half of supply pressure. This condition provides the proper amount of lubrication and cooling flow to the transfer bearing.

9.0 TRUNNION DESIGN

The trunnion assembly provides the link between the blade and the blade pitch actuator. It transmits the torque generated by the blade about the retention radial axis to the actuator and translates actuator axial motion to blade rotation about the retention radial axis. Figure 6.20 is a sketch of the trunnion assembly. The trunnion is bolted to the blade shank at four places. The trunnion roller bearing mates with "ears" on the actuator yoke. Figure 9.1 shows the actuator/bearing relationship at the extremes of travel and at mid travel.

The major component of the assembly is the trunnion which is bolted directly to the blade shank. This part is sized to withstand the blade/actuator loads imposed on it. Since the blade shank material is aluminum, threaded inserts were incorporated to take advantage of the increased shear and tensile stress areas of the oversized threads of the inserts. The threads in the blade shank will be rolled to increase the fatigue life. The bolts are designed to react the moment caused by the offset of the trunnion bearing reaction plane and the blade shank face. Locking cups provide a means of locking the bolt heads and also function as a means of securing the dowel pins.

The dowel pins are used for torque restraint. They are also used as a pilot during assembly to obtain the proper angular relationship between the blade airfoil and the trunnion bearing centerline. This results in the blade and trunnion being a matched set.

Fatigue life is a concern whenever components are designed to withstand cyclic loading. One means of increasing the life of the blade shank and trunnion, is to shot peen highly loaded areas of each component. To increase fatigue strength, both the blade shank and trunnion mounting face are shot peened, also, the trunnion face has been undercut to increase the interface pressure and thus reduce the possibilities for fretting. The remaining shot peened area includes the trunnion pin radius, and was incorporated primarily for the higher allowable strength accompanying shot peening.

The trunnion roller bearing is through hardened steel and mates with surface hardened steel wear plates on the actuator. A detailed description of the roller bearing and wear plate can be found in Paragraph 6.6.

9.1 TRUNNION LOADING

As stated above, the trunnion reacts the blade forces and transmits them to the actuator system. The trunnion was positioned at a maximum pin radius of 7.62 cm (3.00 in.) from the propeller pitch change axis with the actuator at midstroke and the blade delivering its maximum forces. Therefore, the summation of the blade loads, including CTM (Centrifugal Twisting Moment), ATM (Aerodynamic Twisting Moment) and FTM (Frictional Twisting Moment), and reacted at the maximum moment arm of the trunnion. This summation of blade

9.1 (Continued)

loads, TTM (Total Twisting Moment), is applied to the trunnion as a steady force. In addition, the ATM component of the TTM is applied as a cyclic load. This combined load was used to determine the physical size of the trunnion for high cycle fatigue (HCF) (10^8 cycles).

Another loading case, the number of start - stop cycles was considered for low cycle fatigue (LCF) (10^4 cycles). These two loading cases will size the trunnion. Figure 9.2 shows the breakdown of trunnion loading.

The calculation of the bolt loads from the blade TTM was determined by straight beam theory. Considering the trunnion as a straight beam fixed at both ends with an intermediate moment (the moment caused by the offset from the trunnion bearing midplane to the blade shank face), the applied bolt loads were determined to be 21305.9 ± 2473.1 N (4790 ± 556 lb.). However, this applied load must be modified by the effects of preload and temperature difference. The minimum required preload force was calculated using the maximum applied load plus the additional bolt load due to a temperature change of 43.3 degree C. (110 degree F.). Bolt friction factors and torque tolerance requirements were used in determining the maximum resulting preload. The final bolt load used in the stress computations is 34249.6 - 52264 N (7700 - 11750 lb.) preload force and 21305.9 ± 2473.1 N (4790 ± 556 lb.) applied load caused by external trunnion loadings. The bolts were also analyzed for resistance to shear loading caused by dowel failure of 20967.9 ± 5239.7 N (4714 ± 1178 lb.) for the outer bolts and 17084.8 ± 4283.4 N (3841 ± 963 lb.) for the bolts straddling the trunnion bearing. The tensile bolt loads are transmitted to the blade shank through threaded inserts, and computerized analyses H466 and H088 were used to calculate thread stress. The results of these analyses will be presented in a later section.

Dowel pin shear loadings were resolved in the same manner as the bolt shear loads. In the analysis, both dowels were considered to carry the same load. This was determined by applying the direct shear force, then calculating the dowel pattern c.g. and applying the moment load. The resulting shear load is then the square root of the sum of the squares of the direct load and moment load and was calculated to be 27795.5 ± 3229.2 N (6249 ± 726 lb.). From this load, dowel pin shear and bearing stresses can be determined.

9.2 TRUNNION STRESS ANALYSIS

The externally applied design loads, for reference, are as follows:

HCF = 46704 ± 5426.6 N (10500 ± 1220 lb.)

LCF = 23352 ± 23352 N (5250 ± 5250 lb.)

Figure 9.3 is an illustration of the trunnion assembly showing the components and the sections for which the stresses were calculated. The trunnion sections were analyzed using beam theory, where both ends are fixed and the beam

9.2 (Continued)

has a length equal to the arc length between the outboard bolts with a moment applied midway. This loading analysis conservatively neglects the restraint of the two inner bolts, and also the support provided by the blade shank. Trunnion bending stresses were then determined using the resultant bending moment at each of the four cross-sections of Figure 9.3. Bending stress was calculated for the trunnion pin base and was modified by a stress concentration factor to account for the fillet radius between the trunnion pin and the main load carrying section of the trunnion. The trunnion section bending stresses are as follows:

SECT A-A	HCF	=	$2.111E8 \pm 2.312E7$ N/M^2	(30582 ± 3353 psi)
	LCF	=	$1.054E8 \pm 1.054E8$ N/M^2	(15289 ± 15289 psi)
SECT B-B	HCF	=	$2.742E8 \pm 3.185E7$ N/M^2	(39765 ± 4620 psi)
	LCF	=	$1.371E8 \pm 1.371E8$ N/M^2	(19883 ± 19883 psi)
SECT C-C	HCF	=	$3.983E8 \pm 7.637E7$ N/M^2	(57774 ± 11077 psi)
	LCF	=	$1.992E8 \pm 3.286E8$ N/M^2	(28887 ± 47663 psi)
SECT D-D	HCF	=	$3.226E8 \pm 5.435E7$ N/M^2	(46790 ± 7883 psi)
	LCF	=	$1.613E8 \pm 2.339E8$ N/M^2	(23395 ± 33923 psi)

TRUNNION PIN RADIUS HCF = $4.628E8 \pm 8.066E7$ N/M^2
(67124 ± 11698 psi)

LCF = $2.314E8 \pm 8.066E7$ N/M^2
(33562 ± 50343 psi)

Sections C-C and D-D include a bolt hole and dowel hole, therefore a stress concentration factor has been applied to the cyclic stresses calculated at these sections. Figures 9-4a and 9-4b are Goodman Diagrams for the trunnion material, and are used to illustrate the stress margins available at HCF and LCF respectively for each of the trunnion sections.

Figure 9.5 presents the bolt loads and equations used in calculating the bolt shank and bolt head fillet stress. The shank stress is a P/A type stress, however, the externally applied loads travel through the entire bolted joint and are modified by the "joint efficiency". The preload is a direct load applied to the bolt and therefore is not effected by "joint efficiency." This "joint efficiency" is a measure of the relative stiffness of the bolt compared to the stiffness of the joint and is used to modify the external loads applied to the bolt.

9.2 (Continued)

The dowel pin load was calculated at maximum TTM of the blade and referenced as follows:

DOWEL LOAD = 27795.5 ± 3229.2 N (6249 ± 726 lb.)

Dowel shear was calculated using the equation T/A and determined to be 2.194E8 ± 2.549E7 N/M^2 (31826 ± 3697 psi) which has a margin of safety of 360%. The dowel pins, under side loading, bear against both the trunnion and the blade shank. The bearing stress is obtained by dividing the load on the pin bearing against the edge of the hole by the bearing area, where the area is the product of the pin diameter and shank thickness. Substituting the pin side load in the equation, results in a bearing stress of 1.149E8 ± 1.335E7 N/M^2 (16664 ± 1936 psi). This stress has a margin of safety of 365% in the blade shank, > 100% in the trunnion and > 1000% in the dowel pin.

In order to properly transmit the shear loads, the dowel pins were pressed into both the trunnion and blade shank with a diametral interference fit of .0381 - .0635 cm (.0015 - .0025 in.). Using the standard thick walled pressure vessel equations, the values for radial, shear and hoop stress for both the trunnion and blade shank were computed and listed as follows along with the corresponding margin of safety:

SHANK STRESS DUE TO PRESS FIT

RADIAL =	1.31E8 N/M^2 (18937 psi)	MS = 164%
SHEAR =	1.74E8 N/M^2 (25250 psi)	MS = 18.8%
HOOP =	2.18E8 N/M^2 (31563 psi)	MS = 58.4%

TRUNNION STRESS DUE TO PRESS FIT

RADIAL =	3.88E8N/M^2 (56250 psi)	MS = 202%
SHEAR =	5.17E8 N/M^2 (75000 psi)	MS = 36%
HOOP =	6.46E8 N/M^2 (93750 psi)	MS = 81%

9.3 BLADE SHANK THREADS

The blade shank threads carry the trunnion loads as supplied to them through the bolts. The threads are manufactured by a rolling process and they have the advantage over cut threads as the thread strength and fatigue life are increased. One factor contributing to higher fatigue strength is that rolled threads have a minimum foot radius of .0762 mm (.003 in.) while the minimum radius of cut threads is .0254 mm (.001 in.). The root radius is incorporated in the determination of the tooth stress Kt, therefore, the smaller the radius, the larger the Kt.

9.3 (Continued)

From published reports and tests conducted at Hamilton Standard, it was determined that thread loads from the preload and applied load effect the bolted joint through different load paths. As was previously explained, the preload force is constant and not effected by joint efficiency, whereas the applied load is modified by joint efficiency. The following equation was used for thread stress calculations and can be summarized into the basic form:

THREAD STRESS = TOTAL STEADY STRESS $\pm$ TOTAL CYCLIC STRESS

where the total steady stress is the summation of the preload stress and the applied load steady stress, and the total cyclic stress is the applied load cyclic stress. Independent of preload and applied load, the thread stress has two components, one is the member stress and the other is referred to as tooth stress. Both the member stress and tooth stress are determined by computerized analyses H088 and H466. H088 calculates the member stress of each thread considering it as a shell of revolution, and the tooth stress is determined through H466 which basically uses the equation P/A + MC/I for each thread. For both preload stress and applied load stress, the member and tooth stress are combined through the use of a proximity factor, A, which accounts for the fact that the member stress and tooth stress occur at different locations on a full radius thread root. The proximity factor, A, is calculated from the equation:

$$A = 1/[1 + .47(\text{MEMBER } \sigma/\text{TOOTH } \sigma)]$$

Where .47 is a function of the thread form. Therefore, the preload stress is determined using the following equation:

TOTAL PRELOAD
STRESS = PRELOAD MEMBER STRESS (H088) + A * PRELOAD TOOTH STRESS (H466)

and the applied load mean stress from the equation:

APPLIED LOAD STEADY STRESS	=	APPLIED LOAD MEMBER STRESS (H088) * KfM * Nf + APPLIED LOAD TOOTH STRESS (H466) * KfT * Nf

where Nf is the joint efficiency, KfM is the stress concentration factor in fatigue for the member stress and KfT is the stress concentration factor in fatigue for the tooth stress. It should be noted that no stress concentration factors are applied to the steady stresses resulting from the preload and applied load. KfM is determined from the case of an internal hole with an internal notch subjected to tension loading, and is modified three ways, once for the angle of the thread form, once by a multiple notch factor, finally because the material displays a different behavior in the presence of a notch in fatigue than under static loading conditions. Kt for the tooth stress is determined from an empirical equation developed by Heywood (Reference 4):

$$KtT = 1 + .26\ (e/R)^{.7}$$

9.3 (Continued)

where e is 7/16 times the thread pitch and R is the thread root fillet radius. KfT is determined in the same manner as KfM, as related to notch sensitivity of the material.

The total cyclic stress only contains the cyclic component of the applied load and is determined using the same equation for applied load steady stress except the force is the cyclic load:

TOTAL
CYCLIC = CYCLIC APPLIED LOAD MEMBER STRESS (H088) *KfM * Nf + A *
STRESS CYCLIC APPLIED LOAD TOOTH STRESS (H466) * KfT * Nf

Figures 9.6 and 9.7 illustrate the thread stress distribution in the threaded hole and also the resulting stress margins.

The trunnion assembly weight and the resulting centrifugal load is:

TRUNNION	=	7.47 N (1.68 LB)
TRUNNION BEARING & WASHERS	=	3.75 N (0.85 LB)
SHANK ATTACHING HARDWARE	=	2.85 N (0.64 LB)
TOTAL	=	14.10 N (3.18 LB)
RESULTING C.F.	=	6863 N (1543 LB)

10.0 CONCLUSIONS

Since the primary purpose of the LAP program is to evaluate the structural integrity and aeroacoustic performance of the blades, the requirements for the pitch control system were neither critical nor stringent. As shown in this report the design approach used both existing hardware, conservative analysis and state of the art manufacturing practices the original assumptions for control gains and accuracy were confirmed by analytical modeling. The calculated values of these system parameters were either the same or better than originally assumed. Therefore, the Prop-Fan pitch control system as designed is suitable for use in the LAP program.

11.0 References

1. Large Scale Advanced Prop-Fan (LAP) Blade Design Report, NASA CR174790

2. Large Scale Advanced Prop-Fan (LAP) Hub and Blade Retention Design Report, NASA CR174786

3. Large Scale Advanced Prop-Fan (LAP) Spinner Design Report, NASA CR174785

4. R.J. Roark, "Formulas for Stress and Strain", 4th Edition, 1965

TABLES

TABLE 5.1. SPEEDS AND CAPACITIES FOR THE LAP AND E2/C2

	L.A.P. 1698 PROP RPM LEFT HAND ROTATION		E2/C2 1106 PROP RPM RIGHT HAND ROTATION	
	R.P.M.	CAPACITY	R.P.M.	CAPACITY
MAIN PUMP	5294	22.47 Q.P.M. AT 1375 PSI	4712	20 Q.P.M. AT 1375 PSI
SCAVENGE PUMP	5294	32.92 Q.P.M.	4712	29.3 Q.P.M.
STANDBY PUMP	4500	39.86 Q.P.M. AT 1375 PSI	4516	40 Q.P.M. AT 1375 PSI
GOVERNOR	4072	NA	3517	NA

IN ORDER TO ACHIEVE THE SAME FLOW DIRECTION IN THE L.A.P. PUMPS AS IN THE E2/C2 PUMPS, THE PUMP DRIVE GEARS HAD TO BE REVERSED, AS INDICATED IN FIGURE 5.14

TABLE 5.2. MAIN & STANDBY REGULATING VALVE DATA

THIS VALVE IS SIMILAR TO THE MAIN AND STANDBY VALVE USED ON E2/C2 PROPELLER CONTROLS. DEVIATIONS ARE AS FOLLOWS:

-SLEEVE AND PLUNGER ARE MATCHED AT THEIR MATING DIAMETER.

-USE OF SHIMS BETWEEN THE SPRING AND PLUNGER.

MATERIALS

SLEEVE:	AMS 6415, 40-44 HRC
PLUNGER:	AMS 6270 OR AMS 6272 26-44 HRC
SPRING:	AMS 5112

DIMENSIONS

SLEEVE & PLUNGER DIA CLR:	0.0002-.0005
SLEEVE & HOUSING DIA CLR:	0.003-.007

PERFORMANCE

PISTON AREA:	0.3705 IN^2
SPRING RATE:	83 LB/IN
LOAD AT WORKING HEIGHT	56 ± 2.8 LB
OPERATING STROKE:	0.067 IN

TABLE 5.3. ORIFICE PACK DATA

ORIFICE PRESSURE RATIO: (PS-PB)/ (PS-PD)=	0.15
NET FLOW AT : PS =1140 PSI & 70 DEG F.	.48 QT/MIN
NET ORIFICE RATING: (PS → PD)	4662 1 OHMS (D =0.0127 IN)
RATING OF PS → PB ORIFICE:	1800 1 OHMS (D = 0.0205 IN)
RATING OF PB → PD ORIFICE:	4300 1 OHMS (D = 0.0133 IN)

TABLE 5.4 GOVERNOR PERFORMANCE DATA

GOVERNOR RPM (N_G) @ 100% SPEED	4072 RPM
GOVERNOR-PROP SPEED RATIO (N_G/N_P):	2.3981
FLYWEIGHT FORCE EQ:	$F_{FW} = 2.43 \times 10 \, (N_G)^2$
@ 100% RPM:	F_{FW} = 40.29 LB
FLYWEIGHT RATE EQUATION:	$K_{FW} = 1.3 \times 10 \, (N_G)^2$
@ 100% RPM:	K_{FW} = 21.56 LB/IN
SPEED SPRING RATE (K_S):	80 LB/IN
SPEED SET LEVER RATE (K_1):	1042 LB/IN
SPEED BIAS RATE (K_{BG}):	6.99 LB/IN
SYSTEM SPRING RATE:	$K_{SYS} = 1/(1/(K_S+K_{FW}=K_{BG})+1/K_1)$
@ 100% RPM:	K_{SYS} = 98.31 LB/IN
SENSITIVITY @ 100% RPM:	0.0082 IN/% RPM
AXIAL ACCELERATION SPEED SHIFT, 0.3565 % RPM/G	

TABLE 5.5. GOVERNOR MATERIAL LIST

MATERIALS	
FLYWEIGHT:	MIL-T-21014 TYPE II CL 1
SPOOL:	AMS 6260, 30–43 HRC
SLEEVE:	AMS 6260, 30–40 HRC
SCREEN:	AMS 5510
SPRING:	HS 261

DIMENSIONS	
DIA. CLR. SLEEVE & SPOOL:	0.0003–0.0005 IN.
SURFACE FINISH, SLEEVE/SPOOL MATING DIA:	8
MAX SCREEN HOLE DIA.:	0.007 IN.
SCREEN OPEN AREA:	28%

TABLE 5.6. FEATHERING VALVE DATA

FEATHERING VALVE

THE FEATHER VALVE IS SIMILAR TO THE ONE USED ON THE E2/C2 PROPELLER CONTROL. THE DIFFERENCE BEING THAT ONE SEAL GROOVE HAS BEEN MACHINED OFF THE SLEEVE.

DIMENSIONS

DIA CLR, SLEEVE & SPOOL:	.0004 - .0008 IN
DIA CLR, SLEEVE & HOUSING:	.004- .006IN
SPRING FREE LENGTH	2.091 ± .04 IN

MATERIALS

SPOOL:	AMS 4150, 45HRB MIN
SLEEVE:	AMS 4122, 88 HRB MIN
SPRING:	AMS 5673
SPRING SEAT:	AMS 5504

PERFORMANCE

PISTON AREA:	.30498 IN^2
SPRING RATE:	14.6 LB/IN
SPRING LOAD:	
. VALVE OPEN	24LB
. VALVE CLOSED	10LB
SENSITIVITY :	47.87 LB/IN

TABLE 5.7. FRONT AND REAR LIP SEAL DATA

FRONT	
LIP SEAL	
MATERIAL:	VITON
DIA CL/INTF:	
SEAL/RETAINING RING:	0.0015-0.0165 INTF.
SEAL/OIL SHIELD:	0.019-0.097 INTF.
SEALANT:	MIL-S-8802 CLB-2
CONTROL OIL SHIELD	
MATERIAL:	AMS 6415
HARDNESS:	26-33RC
BOND:	HS 812 GRADE 3
PLATING:	NICKEL PER AMS 2404
PLATING THICKNESS:	0.0008-0.0012
REAR	
LIP SEAL	
MATERIAL:	VITON (SAME AS FRONT)
DIA CL/INTF.	
SEAL/RETAINER:	0.0015-0.0165 INTF.
SEAL/BARREL SUPPORT:	0.019-0.097 INTF.
SEALANT:	MIL-S-8802 CLB-2
LIP SEAL RETAINER	
MATERIAL:	AMS 5616
HARDNESS:	32-38 RC
DIA CL/INTF.	
RETAINER/PUMP HSG	0.001-0.009 CL
RETAINER/HEAT EXCHANGER	0.001 MIN CL

ALL DIMENTIONS ARE GIVEN IN INCHES

TABLE 5.8. IDLER GEAR SUPPORT DATA

SUPPORT SHAFT	
MAT'L:	AMS 6415
HARDNESS:	26-33 RC
DIA. CL/ INTF:	
SHAFT/HSG. BORE:	0.0002-0.0015 CL
SHAFT/GEAR SUPPORT:	0.0002-0.0015 CL
SHAFT/SPACER:	0.0005- 0.008 CL
SHAFT/GEAR SUPPORT PLATE:	0.035 MIN. RAD. CL.
SHAFT/SHIM:	0.072-0.078 CL.
WAVE WASHER	
MAT'L:	AMS 5121
HARDNESS:	45-50 RC
WORKING HGT:	090-.094
LOAD AT WORKING HGT:	82-75 LBS.
FREE HGT:	.137 ± .005
THICKNESS:	.068 ± .003
BALL BEARINGS	
MAT'L/HARDNESS:	
RING:	SAE 52100/60 RC MIN
BALLS:	SAE 52100/60 RC MIN
SPEED:	3529 RPM
NO. OF BALLS:	36
BALL DIA:	.1875
RADIAL PLAY:	.0005-.0015
LUBRICATION:	MIL-H-5606
THRUST LOAD:	100 LBS
EXPECTED LIFE:	19500 HRS
DIA. CL./INTF.	
BRG./GEAR SUPPORT:	.0001-0.0015 CL.
BRG./IDLER GEAR:	.0005-0.0013 CL.

ALL DIMENTIONS ARE GIVEN IN INCHES.

TABLE 5.9. LAP GEAR DATA

		MAIN PUMP	STANDBY PUMP	SCAV.PUMP	TRANS. BRG.	LARGE IDLER	SMALL IDLER	GOV.
NO. TEETH		34	40	34	106	51	60	52
DIA. PITCH		12	12	12	12	12	16	16
PRESS. ANG.		22.50°	22.50°	22.50°	22.50°	22.50°	20°	20°
FACE WIDTH (IN)		0.20	0.20	0.20	0.22	0.20	0.15	0.20
RPM		5294	4500	5294	1698	3529	3529	4072
TORQUE (IN-LBS)		76	161	2.1	427 *	57	11.54	10
MAT'L		AMS 6260	AMS 6260	AMS 6260	AMS 6260	AMS 6260	AMS 6260	HS 256 PER HS 178
H D N S	CASE	685VHN	685VHN	685VHN	685VHN	685VHN	~	~
	CORE	30-42HRC	30-42 HRC	30-42 HRC	30-42 HRC	30-42 HRC	30-42 HRC	32-35 HRC
TOOTH LOAD	WT	53.7#	96.6#	1.5#	96.6 #	26.84#	6.2 #	6.2#
	WD	333#	373#	277#	373#	290#	180#	180#
DYNAMIC HERTZ		214,705 PSI	213,950 PSI	195,840 PSI	213,950 PSI	207,085 PSI	183,780 PSI	183,780 PSI
DYNAMIC BENDING		68,115 PSI	64,555 PSI	56,660 PSI	63,580 PSI	50,190 PSI	51,925 PSI	40,500 PSI

* SIZED BY STANDBY PUMP MESH

TABLE 6.1. PITCHLOCK SCREW ASSEMBLY - MATERIAL LIST

PITCHLOCK SCREW	
MATERIAL:	AMS 6265
HARDNESS:	35-43 (CORE), 90HR15N (CASE)
THREAD:	2.19 P.D.
BACKLASH:	0.002 IN.
FRICTION TORQUE	5 IN.-LB
PITCHLOCK SUPPORT	
MATERIAL:	AMS 6265
HARDNESS:	35-43 (CORE), 90HR15N (CASE)
BALLSCREW QUILL	
MATERIAL:	AMS 5616
HARDNESS:	32-38 RC
BETA VALVE CONNECTING ROD	
MATERIAL:	AMS 6415
HARDNESS:	36-40 RC
LOW PITCH STOP	
MATERIAL:	AMS 6415
HARDNESS:	36-40 RC
ADJUSTMENT	$40° \beta_{3/4}$ TO $-10° \beta_{3/4}$

TABLE 6.2. TRUNNION LOADS

MAX DESIGN LOAD –

CONDITION: (100% SPEED; SLSD; 0 POWER; 333 KEAS; $44° \beta_{3/4}$)

CTM:	22,130 IN# X 1.2*	=	26,556 IN#/BLADE
ATM:	3,052 IN# X 1.2*	=	3,662 IN#/BLADE
FTM:	CL X PD/2 X 0.005	=	1,256 IN#/BLADE
TTM:			31,474 IN#/BLADE

*(CONTINGENCY FACTOR)

TRUNNION LOADS (@ 3.00 MA):

(LCF)	START-STOP	10^4~	0 TO 10,500#
(HCF)	VIBRATORY	10^7~	10,500# ± 2,625#
NO FTM	YIELD CASE (125% RPM)		16,000#
NO FTM	ULT. CASE (140% RPM)		20,000#

TABLE 6.3. ACTUATOR EAR WEAR PLATE SUBSURFACE SHEAR STRESS SUMMARY

	X22A EXPERIMENTAL DUCTED V/STOL PROPELLER FOR BELL AIRCRAFT	VC825 EXPERIMENTAL VARIABLE CAMBER PROPELLER	VC400 EXPERIMENTAL V/STOL PROPELLER FOR VEREIGTE FLUGTEDINISCHE WERKE	14SF COMMUTER PROPELLER	LAP
MATERIAL CASE: CORE: CASE DEPTH:	AMS 6260 90R15N 30-40 R_C 0.055-0.075	AMS 6270 90R15N 30-40 R_C 0.060-0.075	AMS 6265 90R15N 35-43 R_C 0.130-0.170	AMS 6260 90R15N 35-43 R_C 0.035-0.045	AMS 6265 59-64 R_C 35-43 R_C 0.080-0.090
GEOMETRY PLATE THICKNESS: ROLLER O.D.: ROLLER CROWN RADIUS:	0.195 1.400 6.0	0.217 1.800 14.0	0.375 2.500 24.0	0.156 1.375 30.0	0.200 2.000 20.0
MAX DESIGN LOADS	5,000	9,150	25,000	3,900	10,500
SHEAR STRESS (@ MIN CASE DEPTH)	93,300	86,400	80,100	94,300	75,000

ALL DIMENTIONS ARE GIVEN IN INCHES
ALL LOADS ARE GIVEN IN POUNDS
ALL STRESSES ARE GIVEN IN POUNDS PER SQUARE INCH

TABLE 7.1. LAP SYSTEM MODEL VARIABLES

DESCRIPTION OF VARIABLES	
N1, NG	- GAS PRODUCER SPEED (100% - 14300. RPM)
N2, NPT	- POWER TURBINE SPEED (100% - 11500. RPM)
NPROP	- PROP-FAN SPEED (100% - 1698. RPM))
WF	- METERED FUEL FLOW
P3, CDP	- COMPRESSOR DISCHARGE PRESSURE
QPT	- POWER TURBINE TORQUE
T45, TIT	- TURBINE INLET TEMPERATURE
NREF	- PROP-FAN SPEED REFERENCE
NMASTER	- PROP-FAN SPEED REFERENCE FOR SYNCHROPHASER
NTRIM	- PROP-FAN SPEED TRIM FROM ELECTRONICS
XACT	- BLADE ACTUATOR STROKE
XS	- SERVO STROKE
THETE	- SYNCHROPHASER PHASE ERROR
ACCHP	- ACCESSORY HORSEPOWER
PLA	- POWER LEVER ANGLE (REPRESENTATIVE)

FIGURES

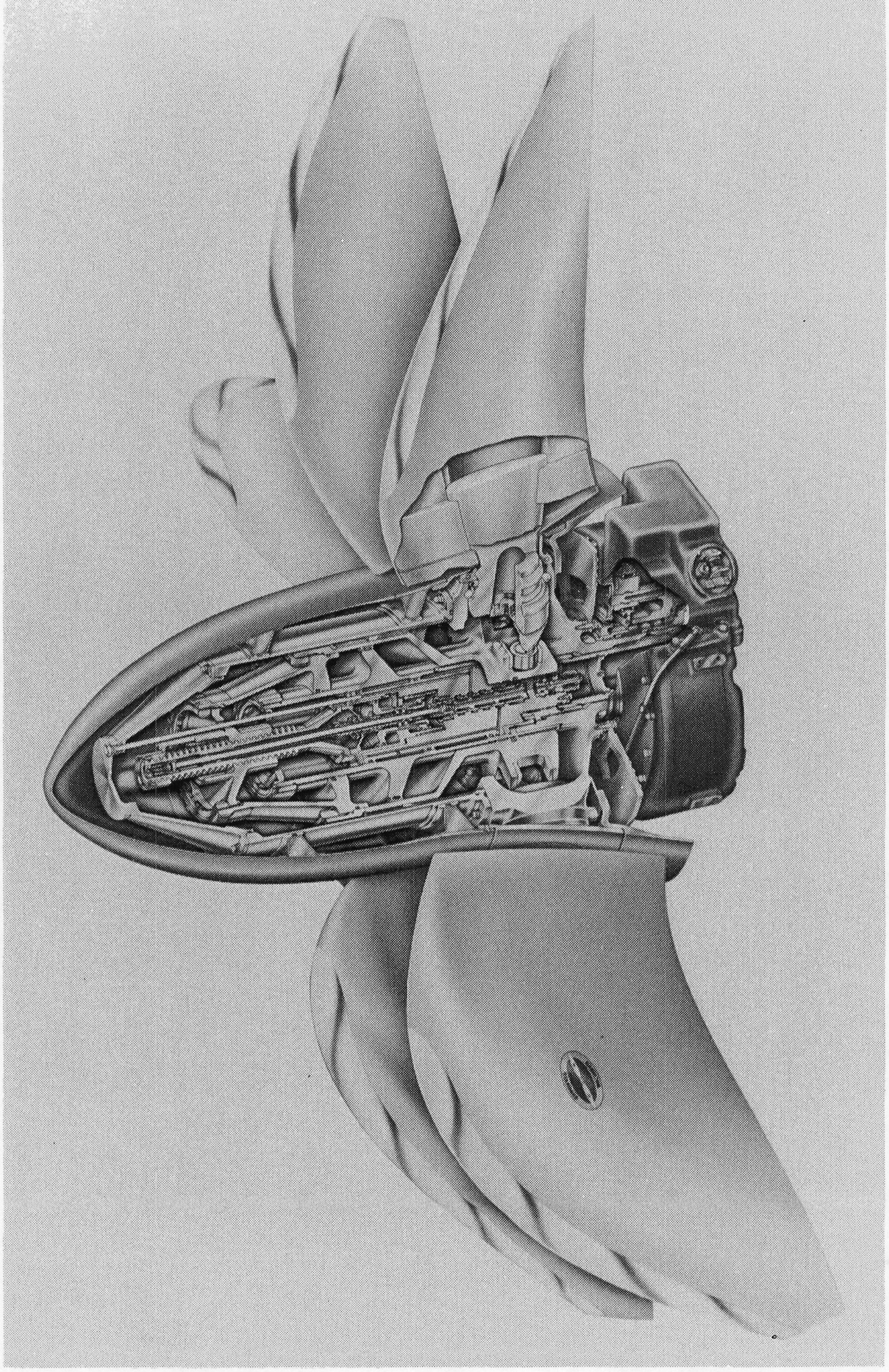


FIGURE 2.1

LARGE SCALE ADVANCED PROP-FAN (LAP)

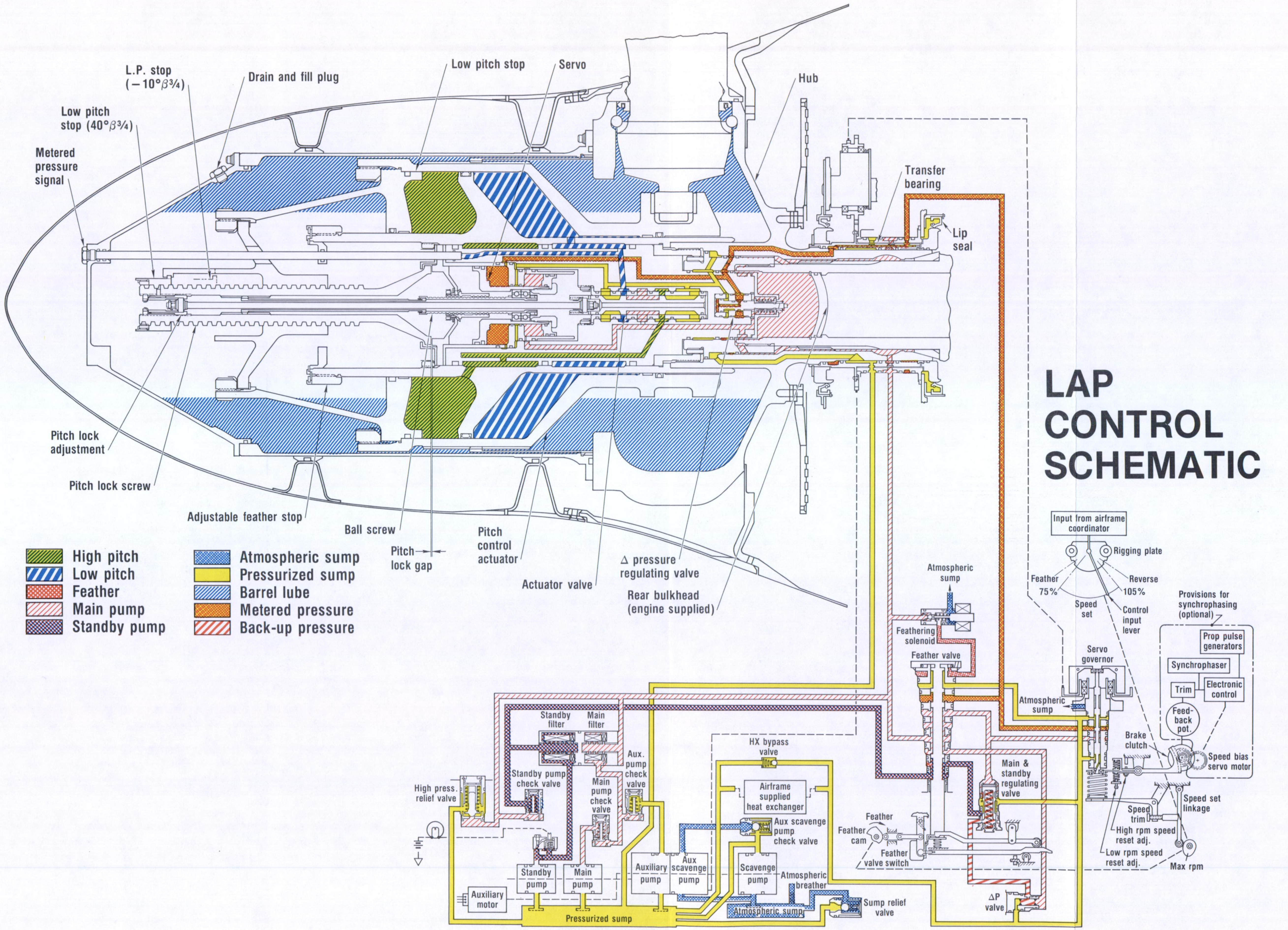


FIGURE 3.1 LAP CONTROL SCHEMATIC

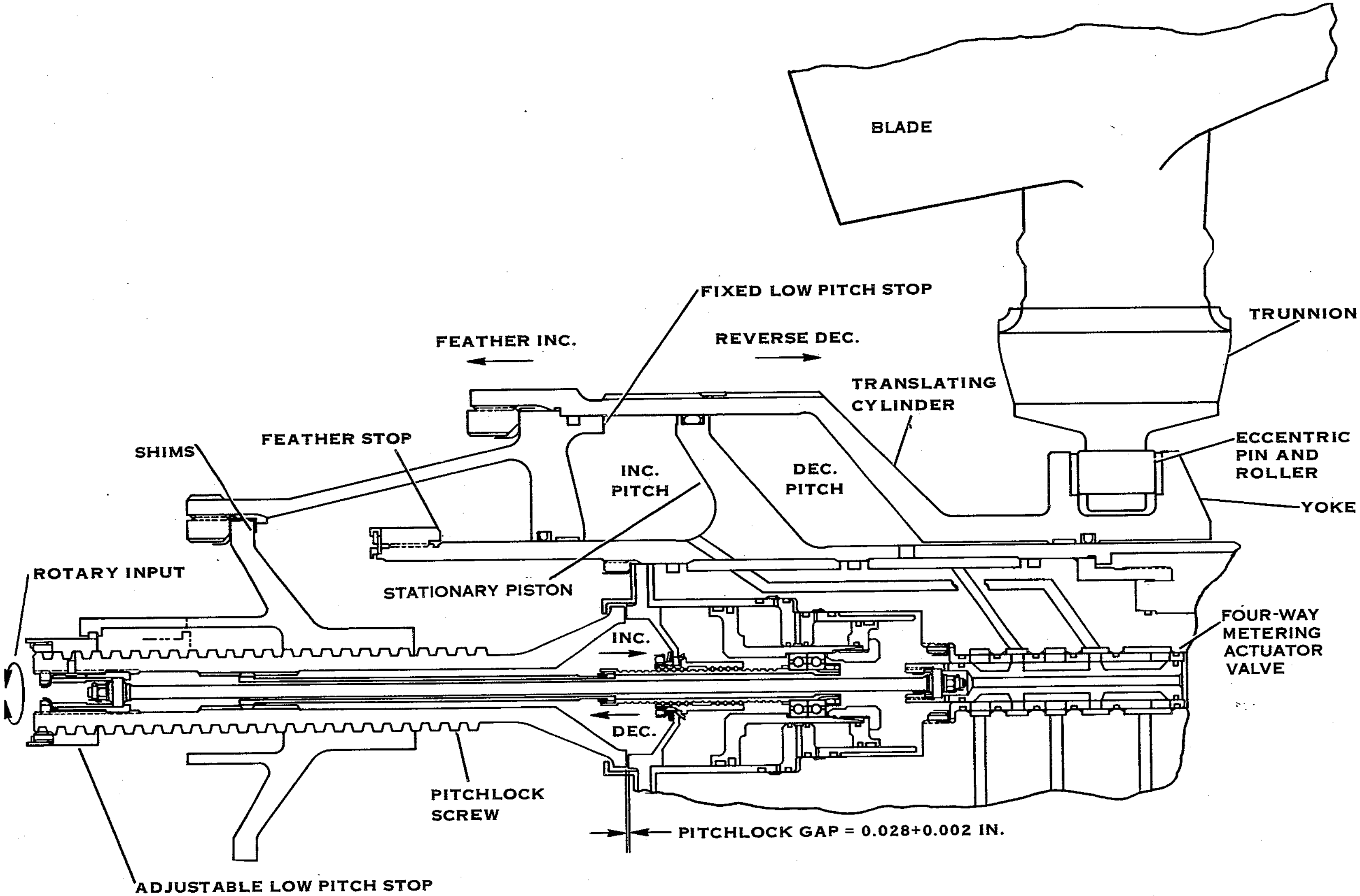


FIGURE 3.2. PITCH CHANGE ACTUATOR & PITCHLOCK

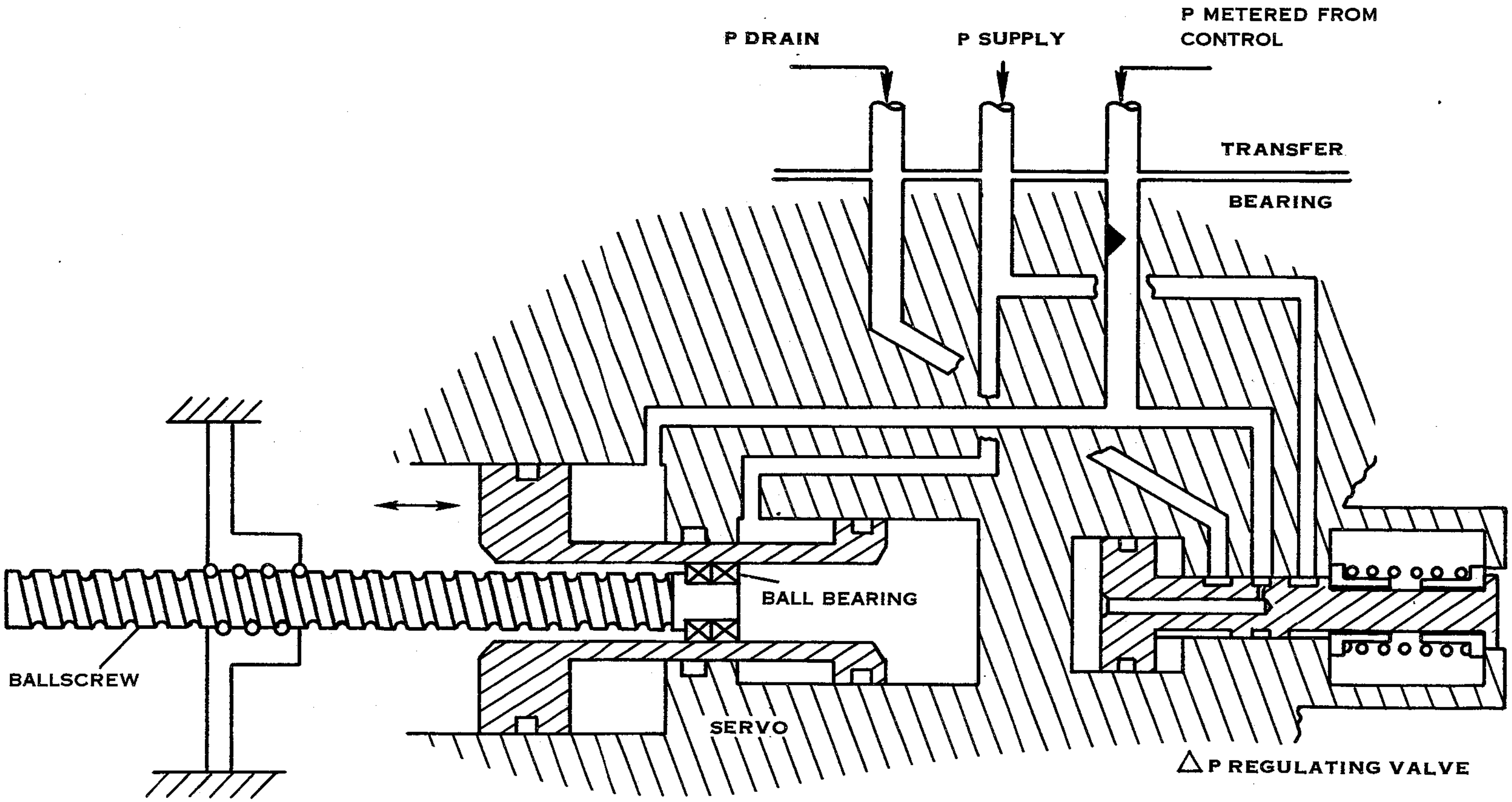


FIGURE 3.3. SCHEMATIC OF HALF-AREA SERVO VALVE AND ΔP REGULATING VALVE

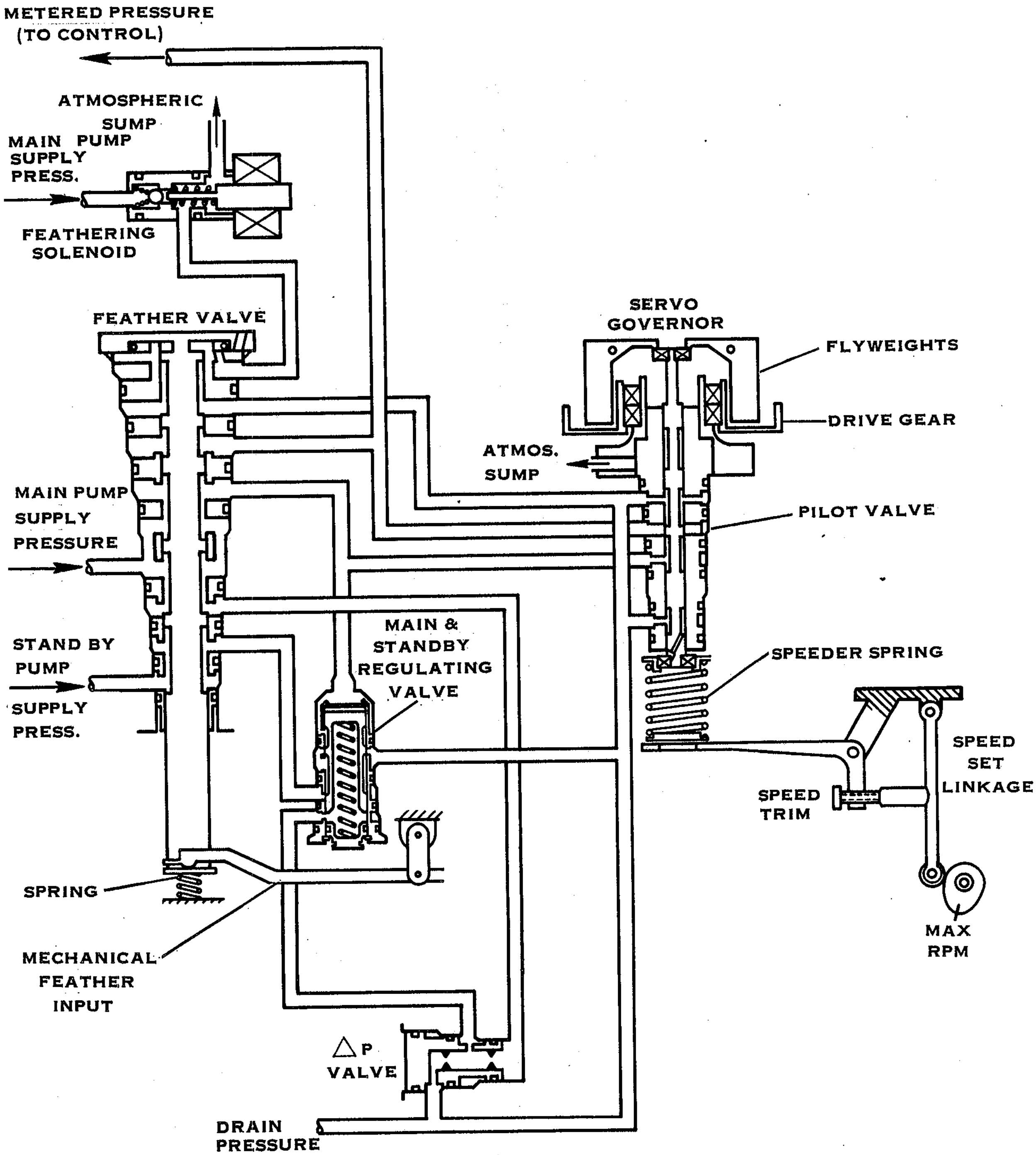


FIGURE 3.4. SERVO GOVERNOR AND FEATHER VALVE

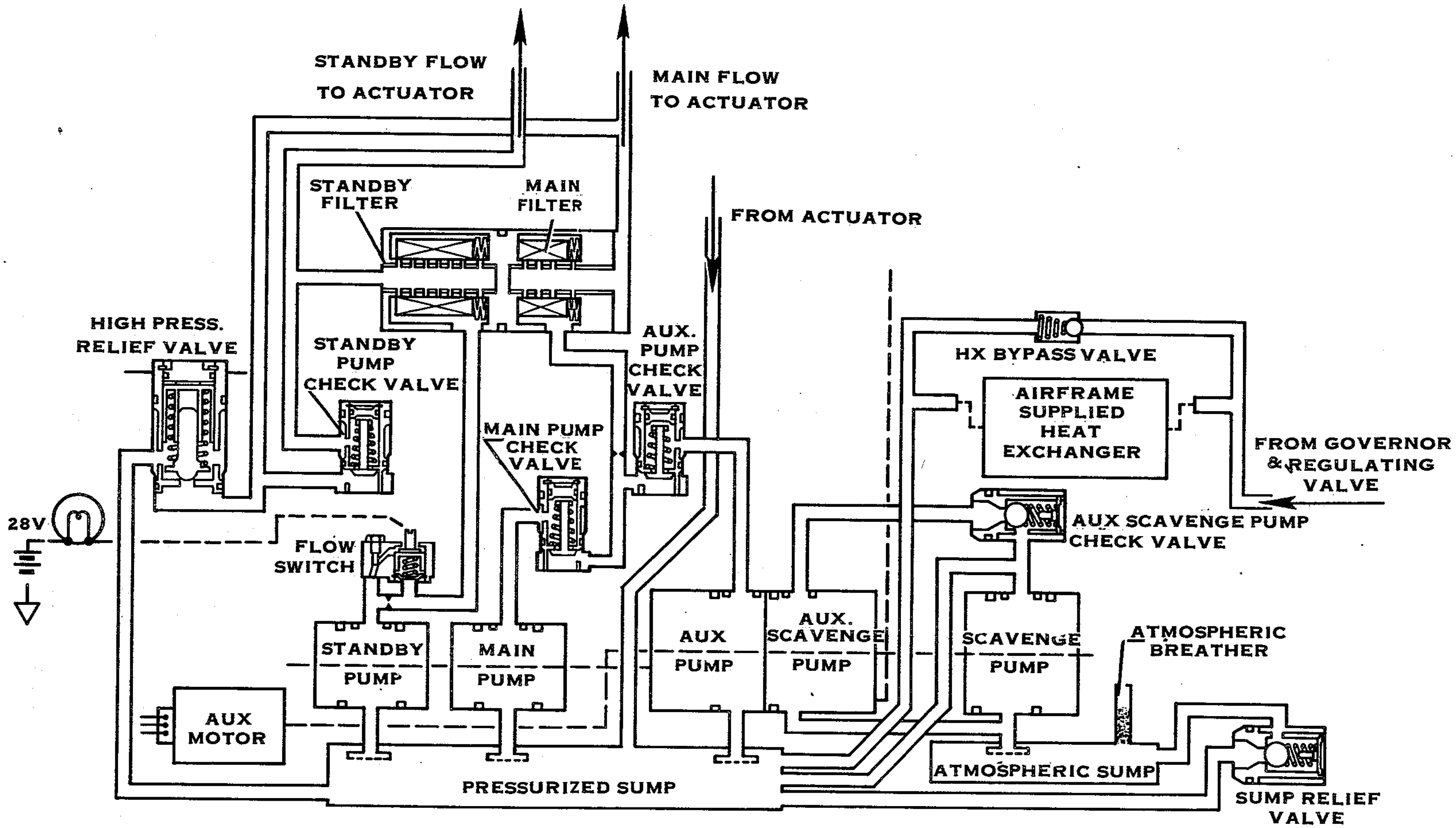


FIGURE 3.5 PITCH CONTROL PUMPING SYSTEM

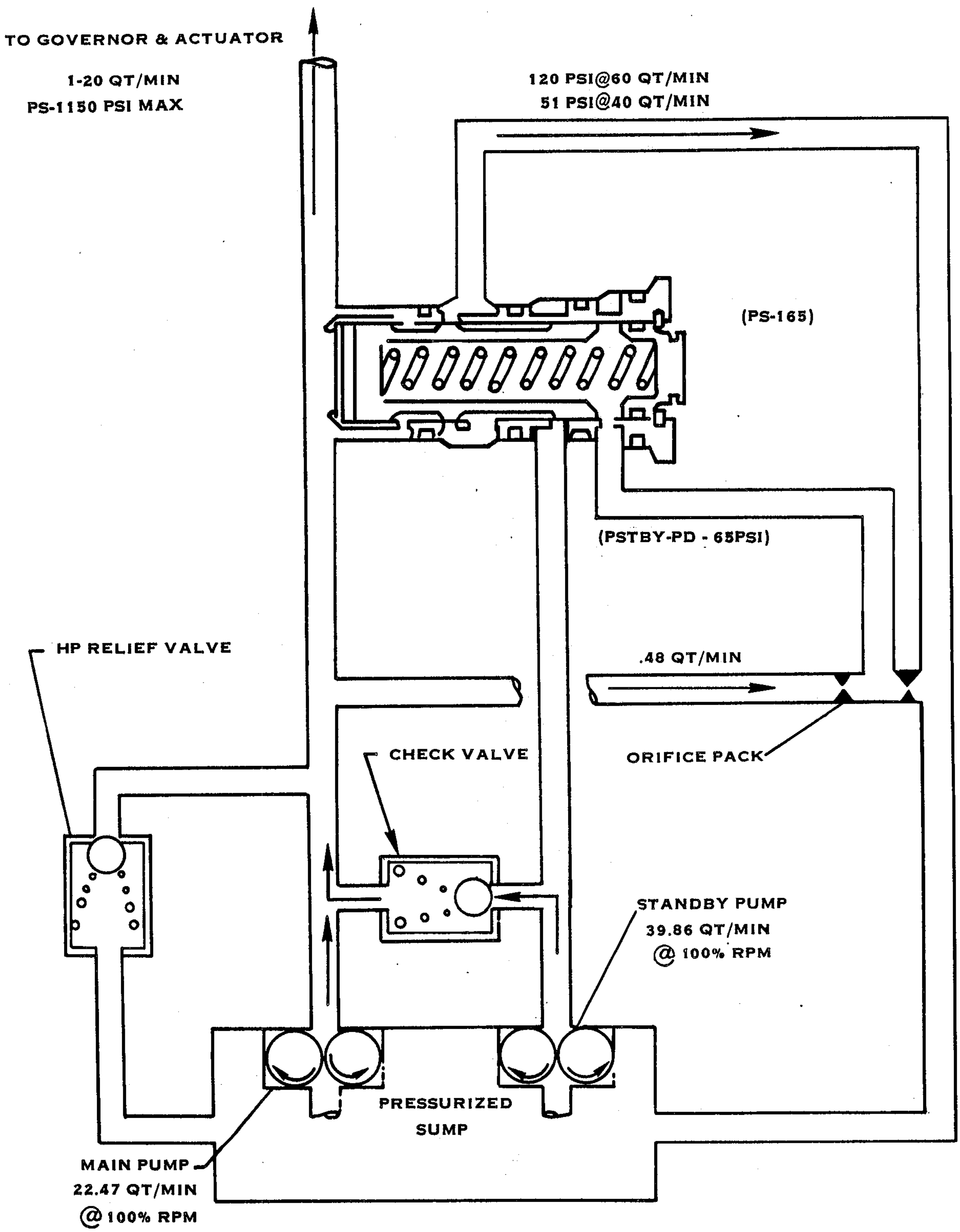


FIGURE 3.6. MAIN & STANDBY REGULATING VALVE LOW DEMAND CONDITION

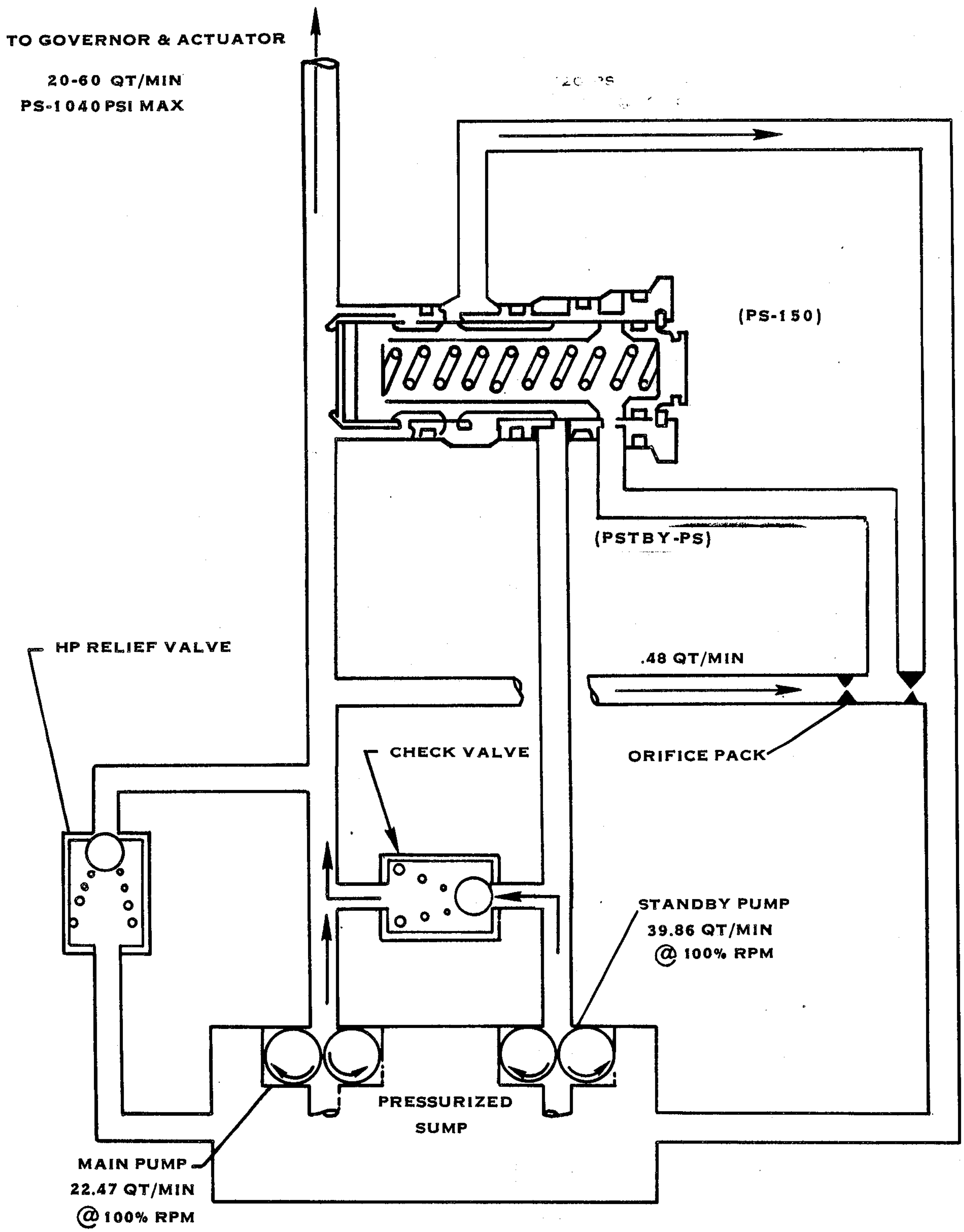


FIGURE 3.7. MAIN & STANDBY REGULATING VALVE HIGH DEMAND CONDITION

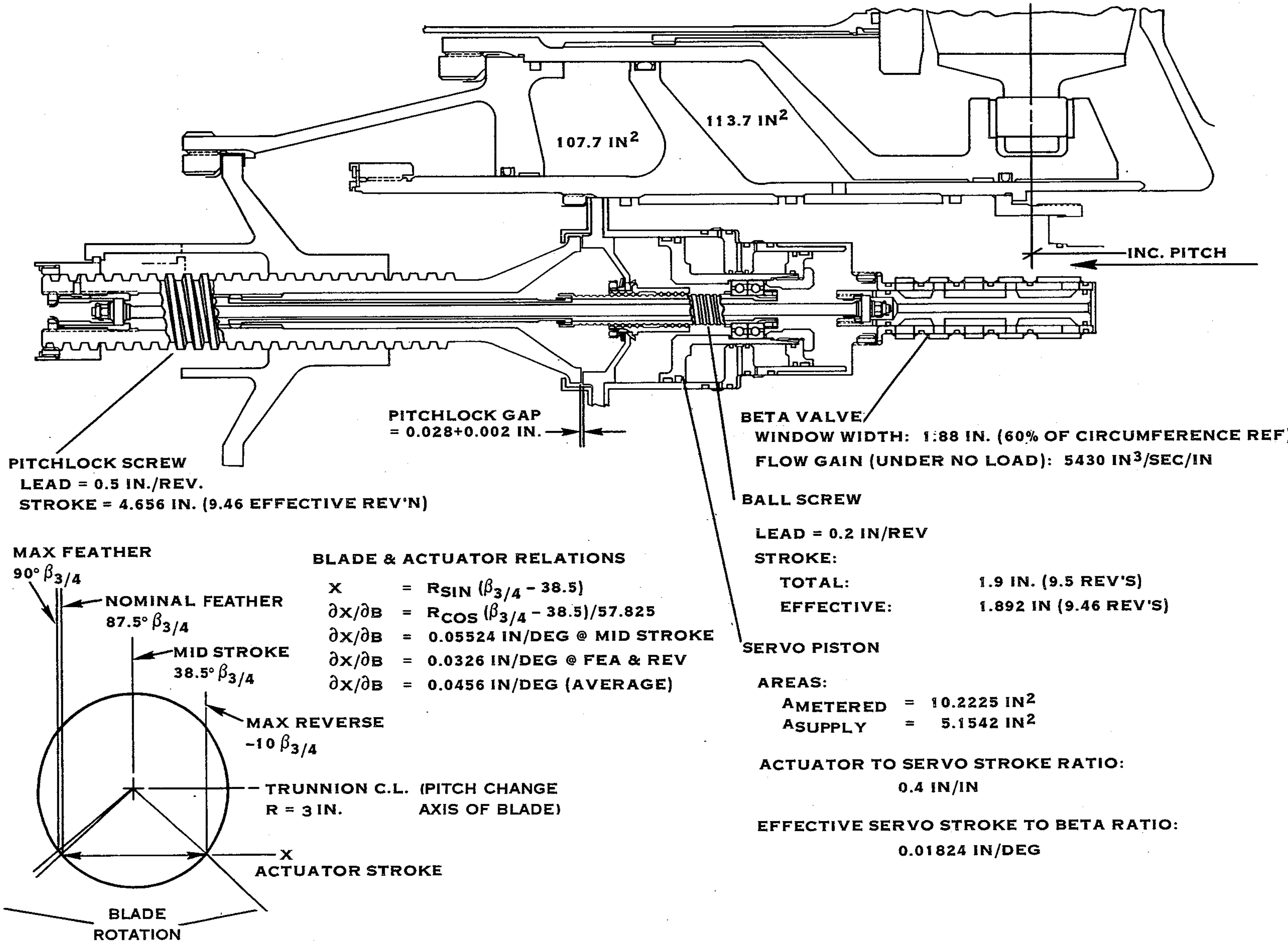
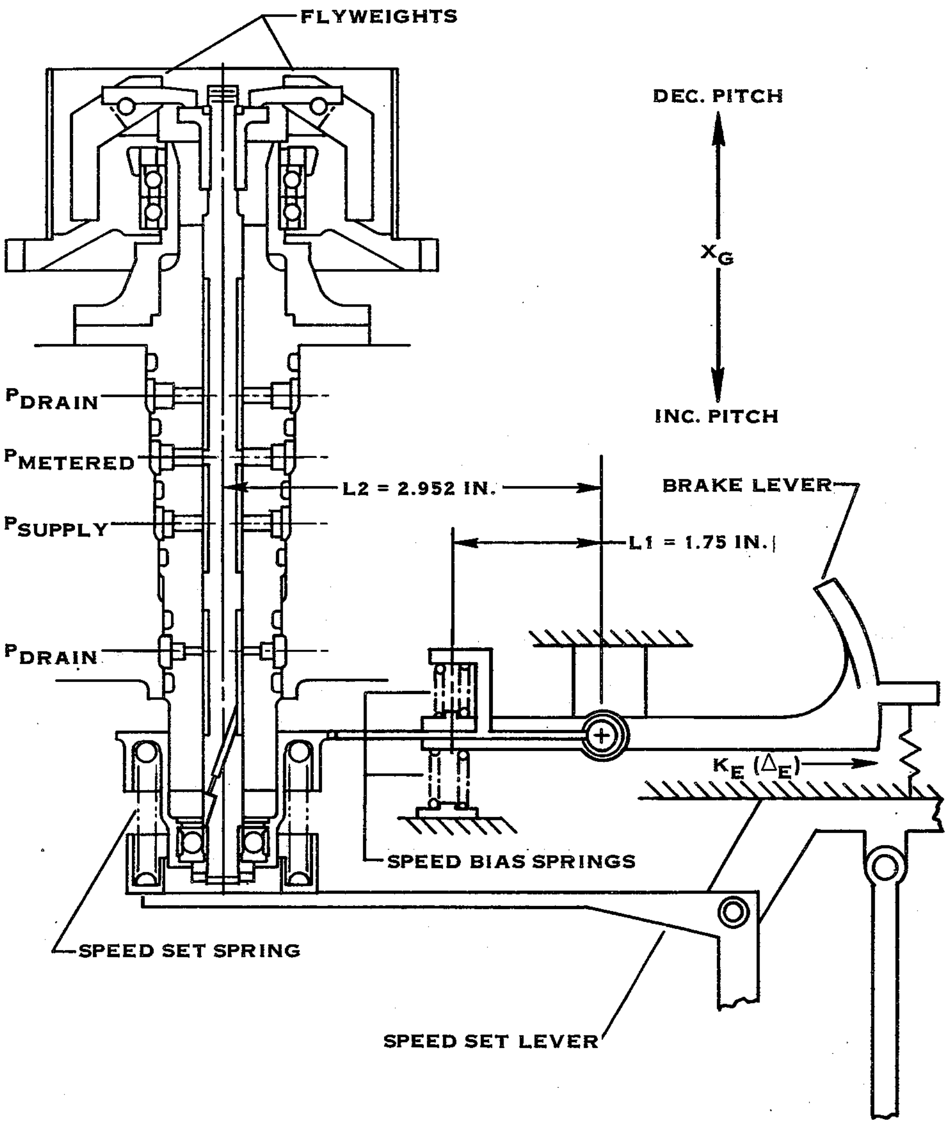


FIGURE 3.8. PITCH CHANGE ACTUATOR CONSTANTS



ELECTRONICS OFF ($K_E = 0$)
ELECTRONICS ON ($K_E = \infty$)

GOVERNOR RPM (N_G) @ 100% SPEED: 4072 RPM

GOVERNOR-PROP SPEED RATIO (N_G/N_P): 2.3981

FLYWEIGHT FORCE EQ: $F_{FW} = 0.004029\ (\%N_G)^2$
@ 100% RPM: $F_{FW} = 40.29$ LB

FLYWEIGHT RATE EQUATION: $K_{FW} = 0.002156\ (\%\ N_G)^2$
@ 100% RPM: $K_{FW} = 21.56$ LB/IN

SPEED SPRING RATE (K_S): 80 LB/IN

SPEED SET LEVER RATE (K_1): 1042 LB/IN

SPEED BIAS RATE (K_{BG}):

$K_{BG} = (L1/L2)^2\ (1/(1/K1+1/K2))$
LEVER TO GOVERNOR RATE (K_1) 65.8 LB/IN
LEVER TO GROUND RATE (K_2) 28.5 LB/IN
WHEN $K_E = 0$: $K_{BG} = 6.99$ LB/IN
K_E = : $K_{BG} = 23.12$ LB/IN

SYSTEM SPRING RATE:
$K_{SYS} = 1/(1/(K_S+K_{FW}+K_{BG})+1/K1)$
@ 100% RPM:
$K_E = 0$ $K_{SYS} = 98.31$ LB/IN
$K_E = \infty$ $K_{SYS} = 111.36$ LB/IN

SENSITIVITY $= \partial X_G/\partial N\% = (\partial F_G/\partial N\%)\ (\partial X_G/\partial F_G) = S$
@ 100% RPM:
$K_E = 0$ S = 0.0082 IN/% RPM
$K_E = \infty$ S = 0.0072 IN/% RPM

AXIAL ACCELERATION SPEED SHIFT, 0.3565 % RPM/G

FIGURE 3.9. SERVO GOVERNOR CONSTANTS

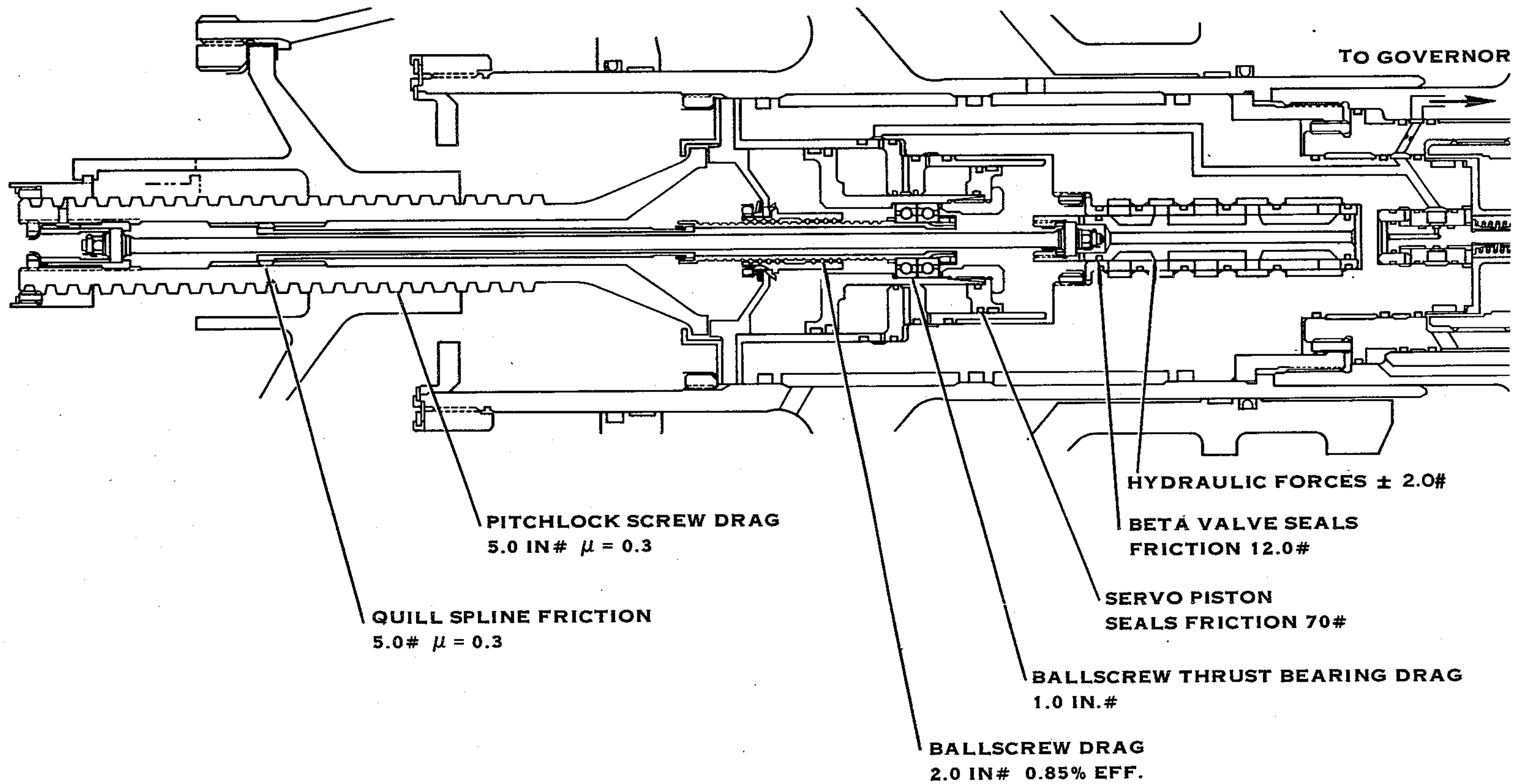
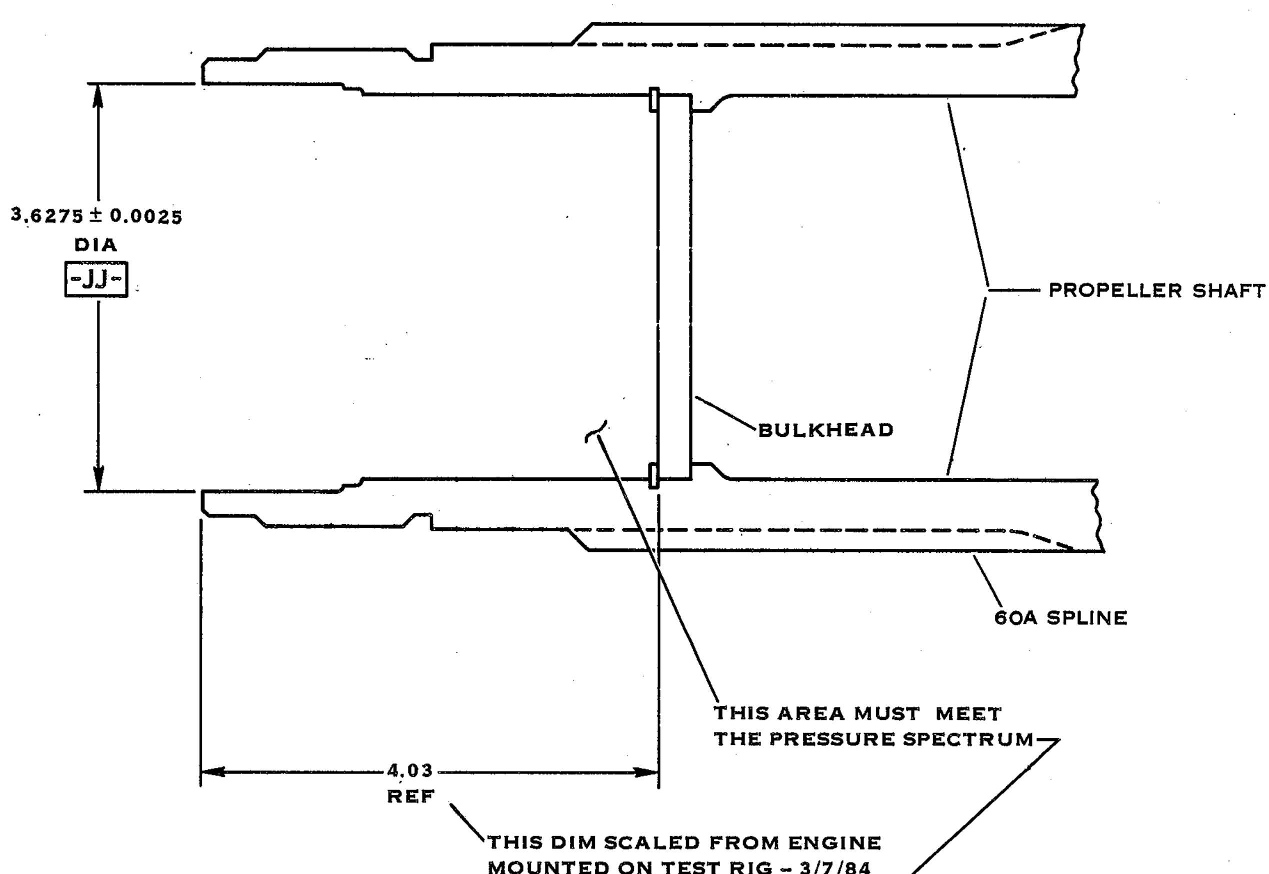


FIGURE 3.10. GOVERNOR LOOP FRICTION HYSTERESIS



PRESSURE SPECTRUM	
0-1375 PSI	10000 CYCLES
1140 ± 160	∞
2550	BURST
1700	PROOF

FIGURE 4.1. MODIFICATION TO AND10152-60 PROPELLER SHAFT

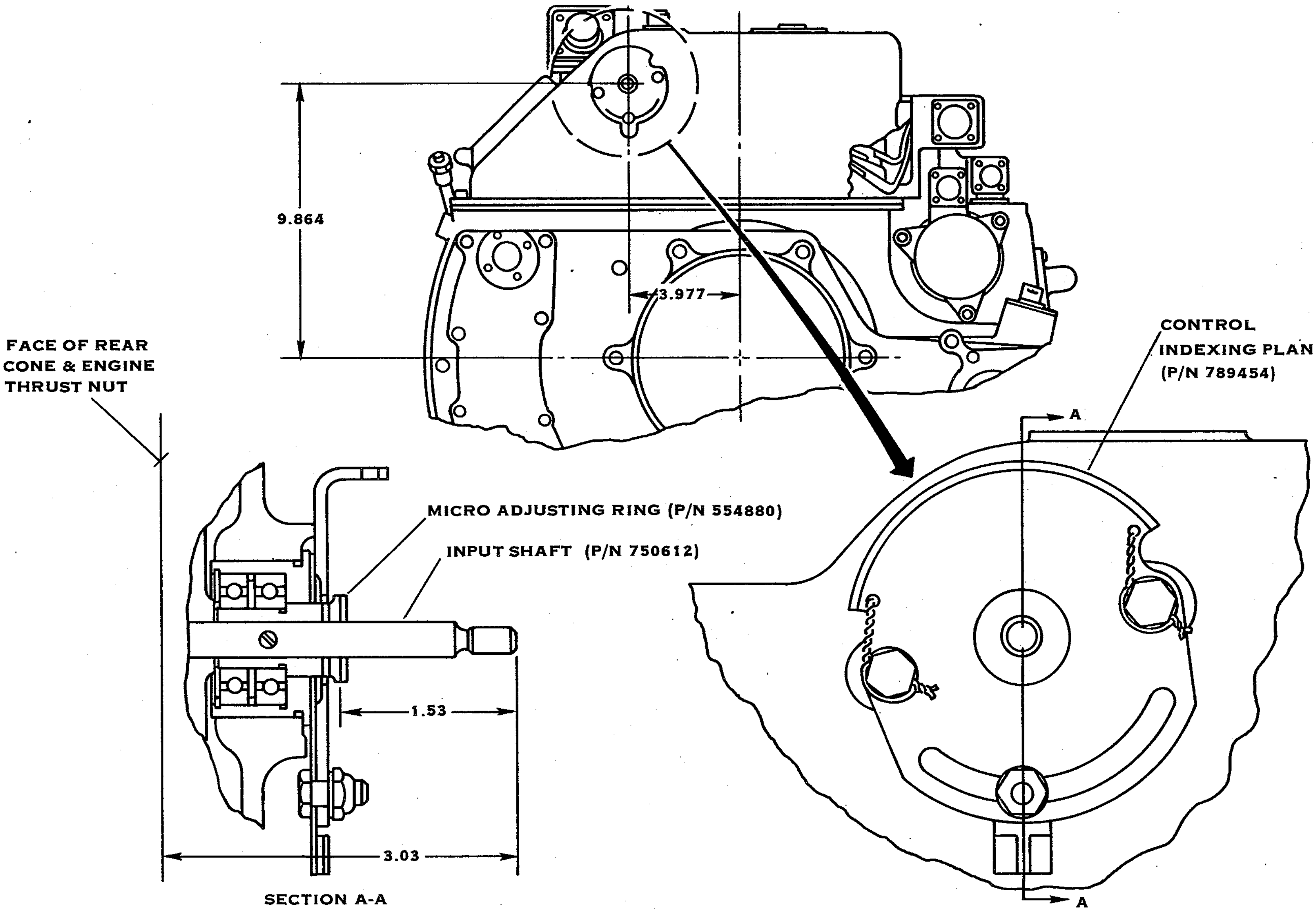


FIGURE 4.2. INPUT LEVER INTERFACE FOR PITCH CHANGE CONTROL

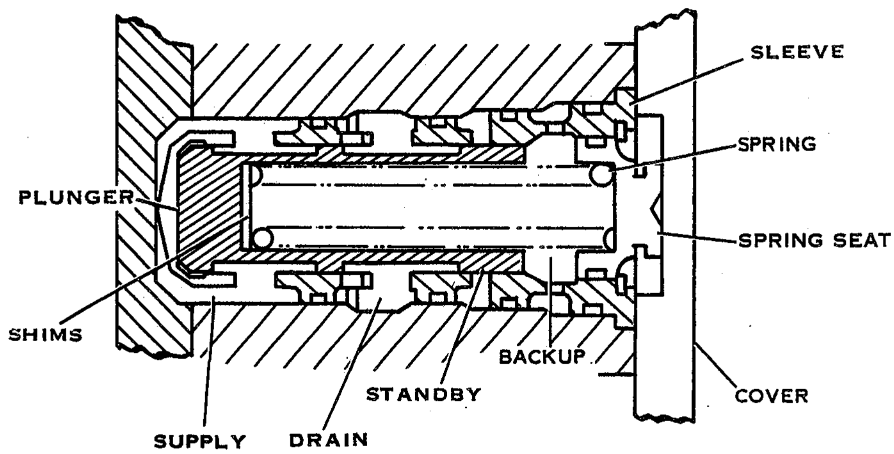


FIGURE 5.1 MAIN & STANDBY REGULATING VALVE

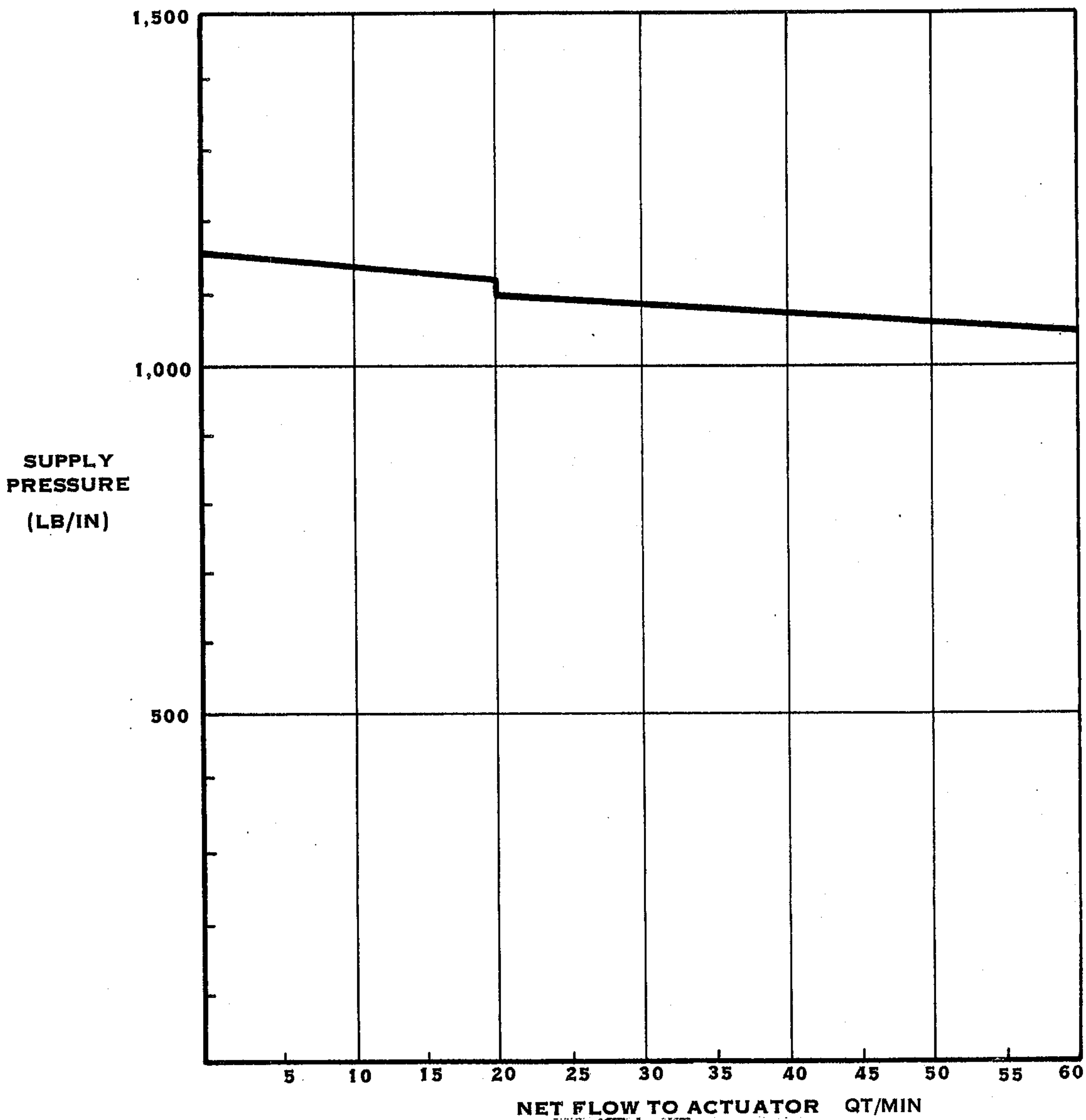
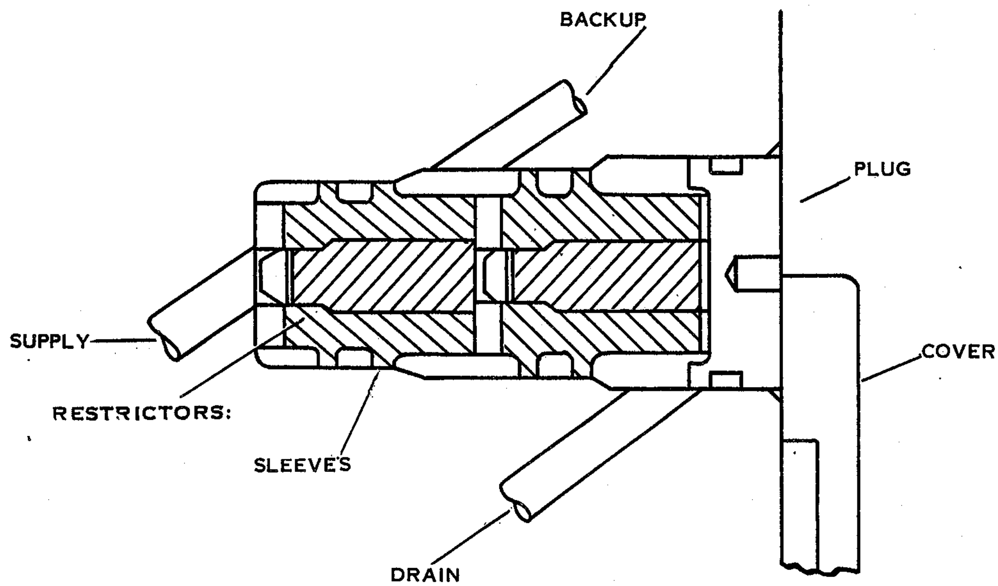


FIGURE 5.2. MAIN & STANDBY REGULATING VALVE SUPPLY PRESSURE AS A FUNCTION OF NET FLOW TO ACTUATOR



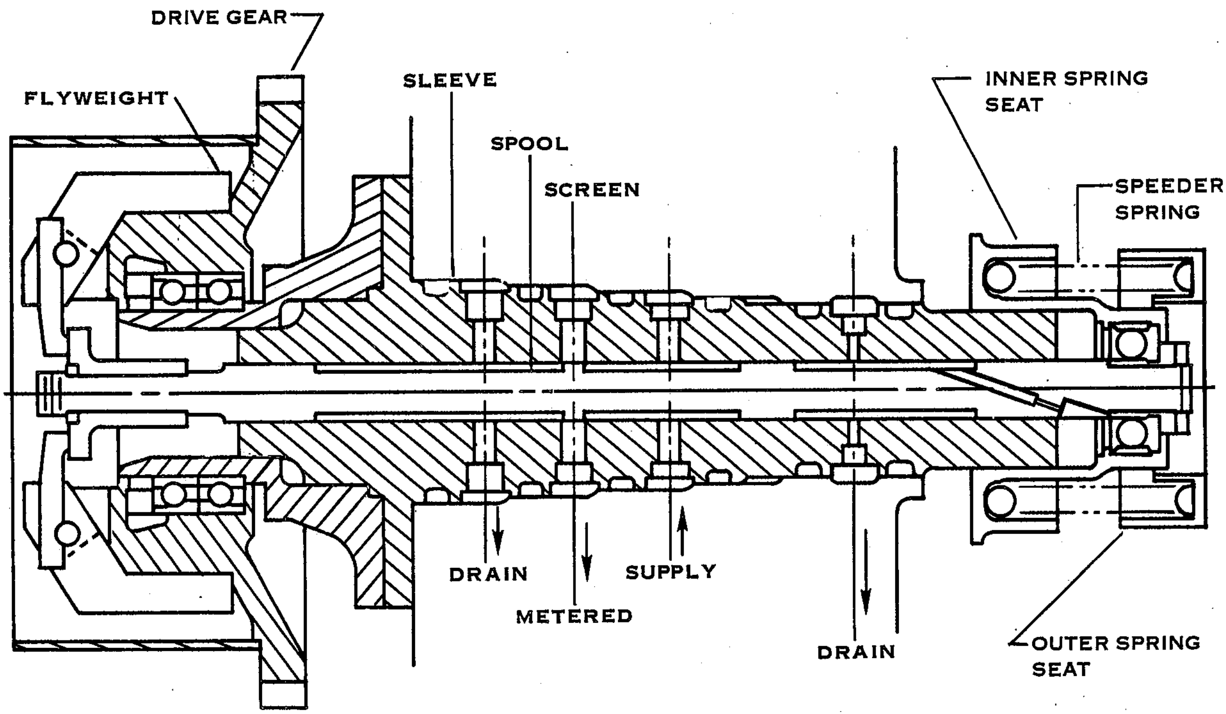
MATERIALS

RESTRICTORS:	
-SCREENS	**AISI 304**
-REMAINDER	**AISI 308**

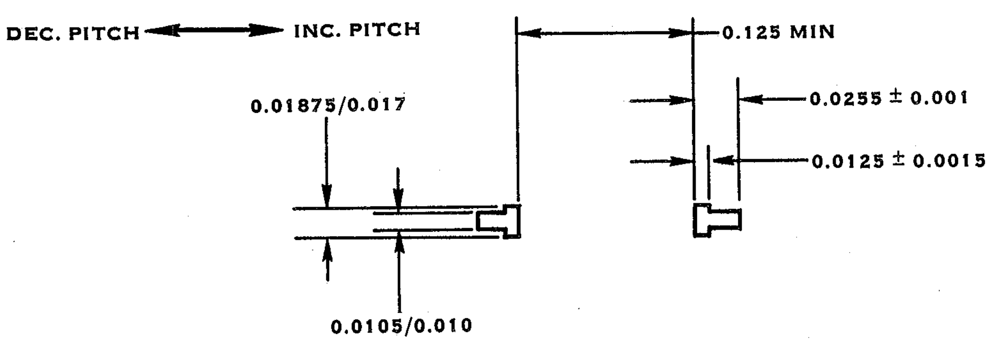
DIMENSIONS

RESTRICTOR SCREENS:	
-HOLE SIZE:	**0.007- 0.010 IN. DIA.**
-OPEN AREA	**38.7 %**

FIGURE 5.3 ORIFICE PACK



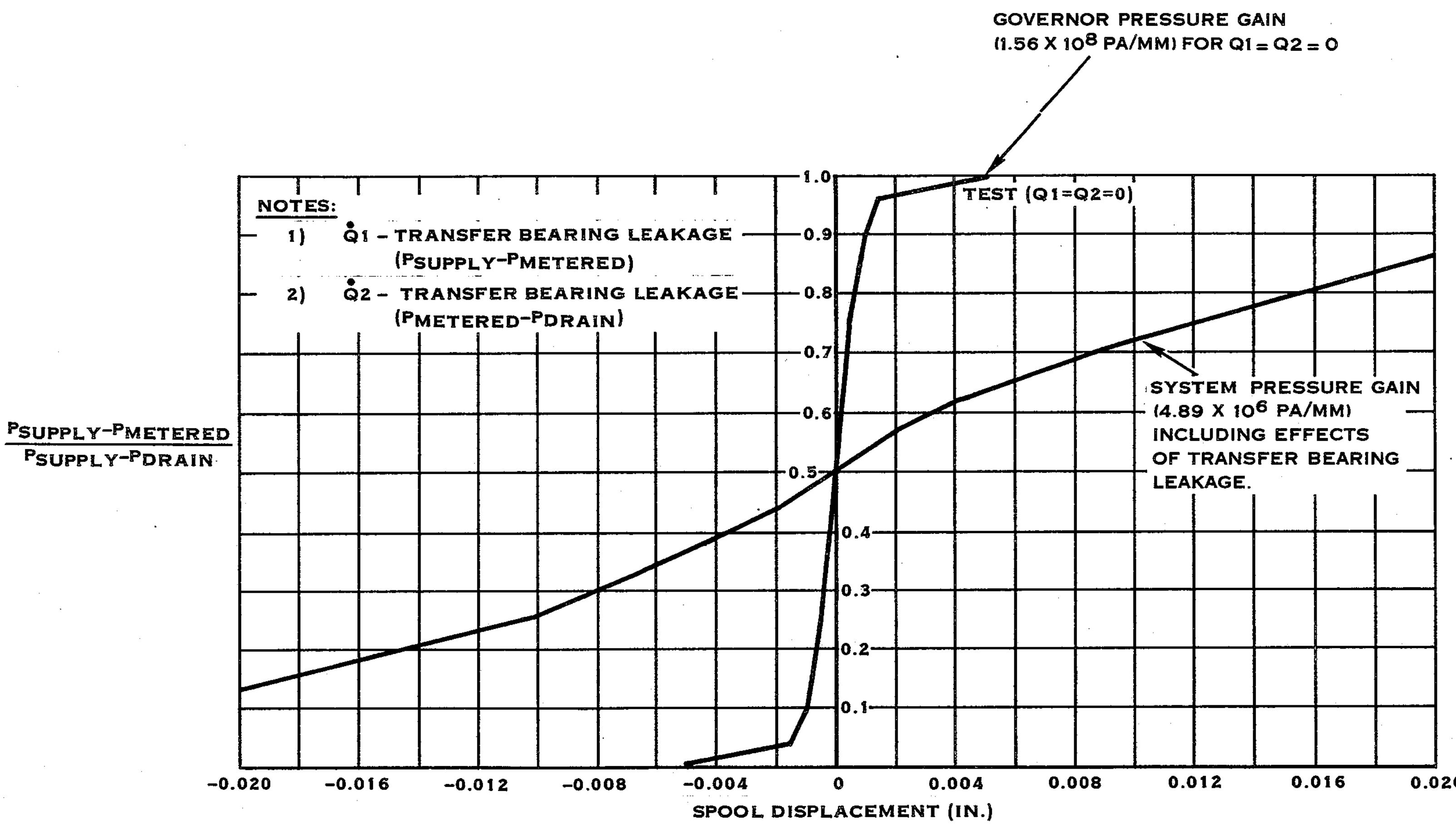
B. GOVERNOR ASSEMBLY



A. GOVERNOR METERING WINDOWS

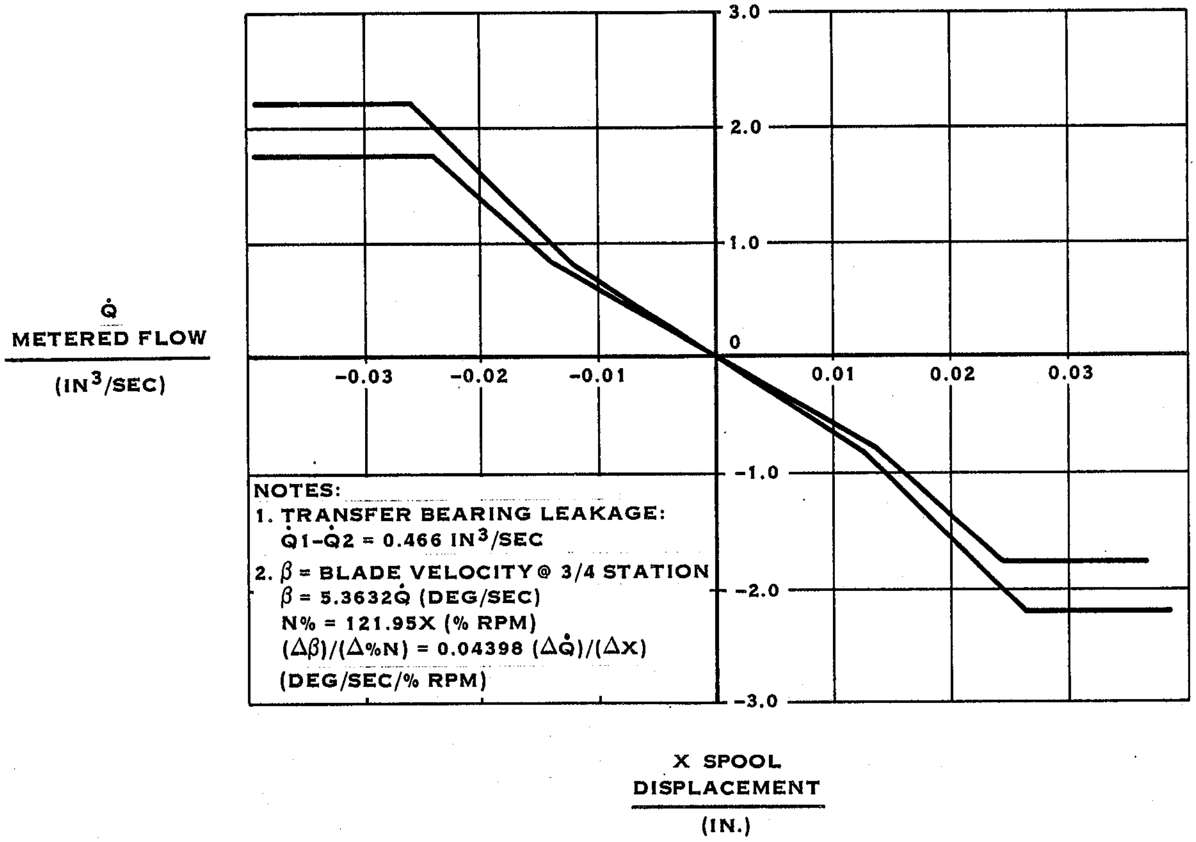
ALL DIMENSIONS ARE GIVEN IN INCHES

FIGURE 5.4. GOVERNOR

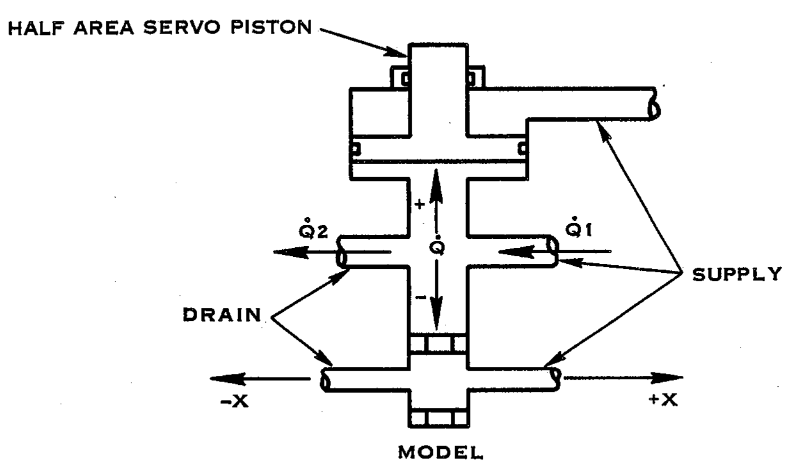


PRESSURE GAIN CHARACTERISTICS

FIGURE 5.5 GOVERNOR PRESSURE GAIN TEST AND SYSTEM REQUIREMENTS



B. FLOW GAIN CHARACTERISTICS



A. METERED FLOW PATH

FIGURE 5.6 GOVERNOR FLOW GAIN REQUIREMENTS

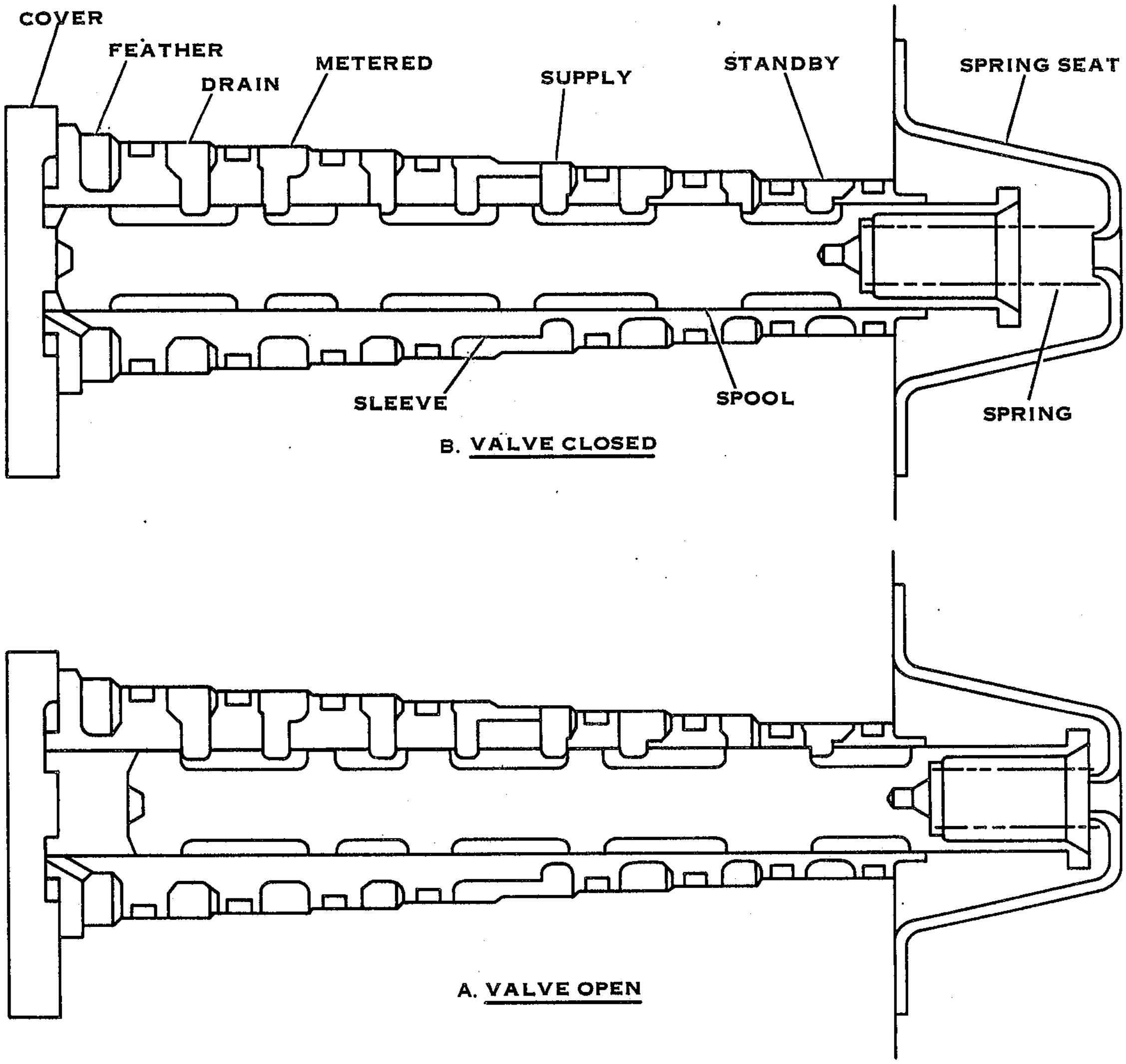


FIGURE 5.7. FEATHERING VALVE

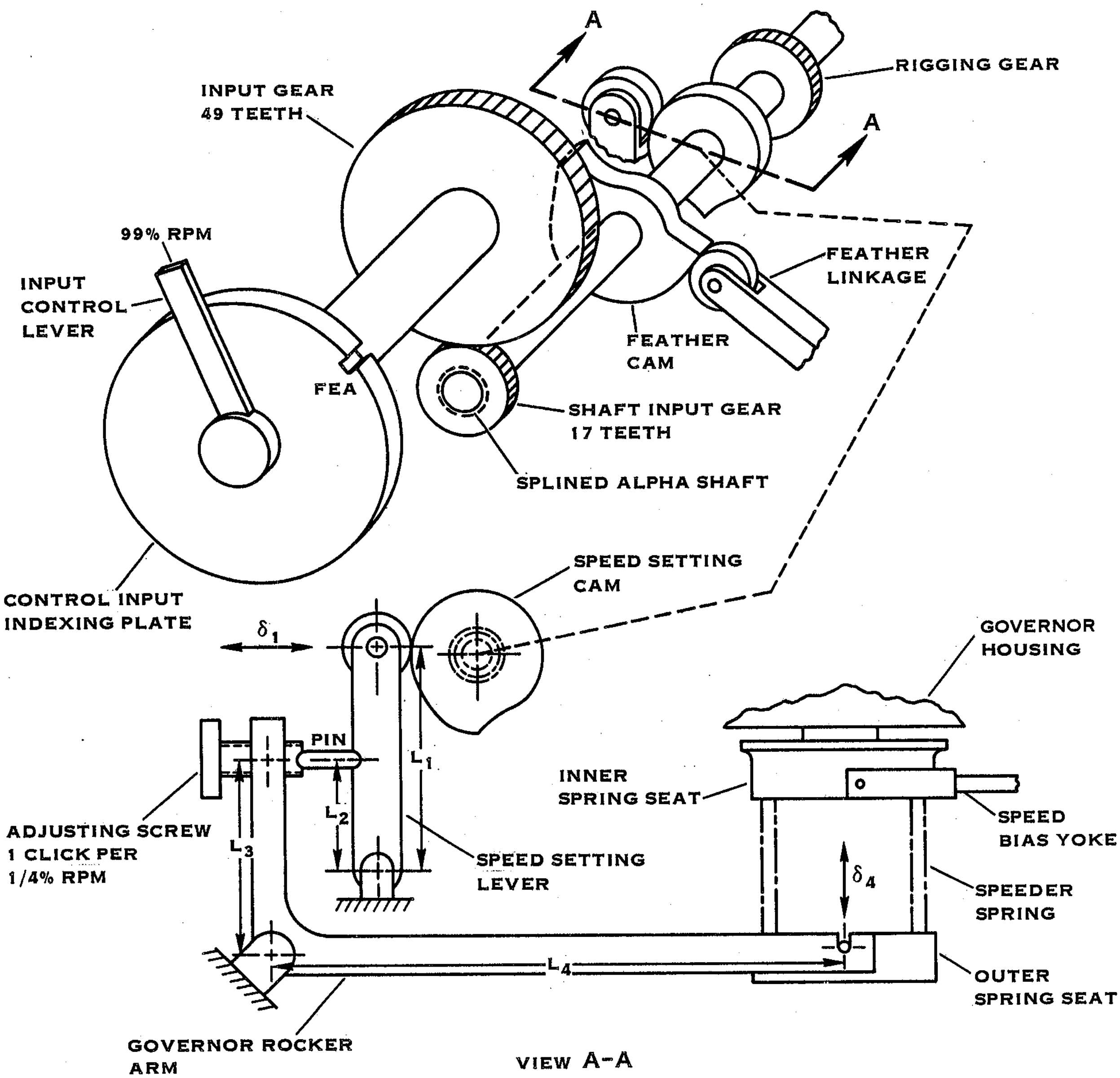


TABLE OF LENGTHS AND RATIOS
L_1 = 2.2037 IN.
L_2 = 1.00 IN.
L_3 = 1.38 IN.
L_4 = 4.18 IN.
δ_4 = 1.3746 δ_1
GEAR RATIO = 2.8824 (° CAM/° INPUT)

FIGURE 5.8 GOVERNOR LINKAGE SCHEMATIC

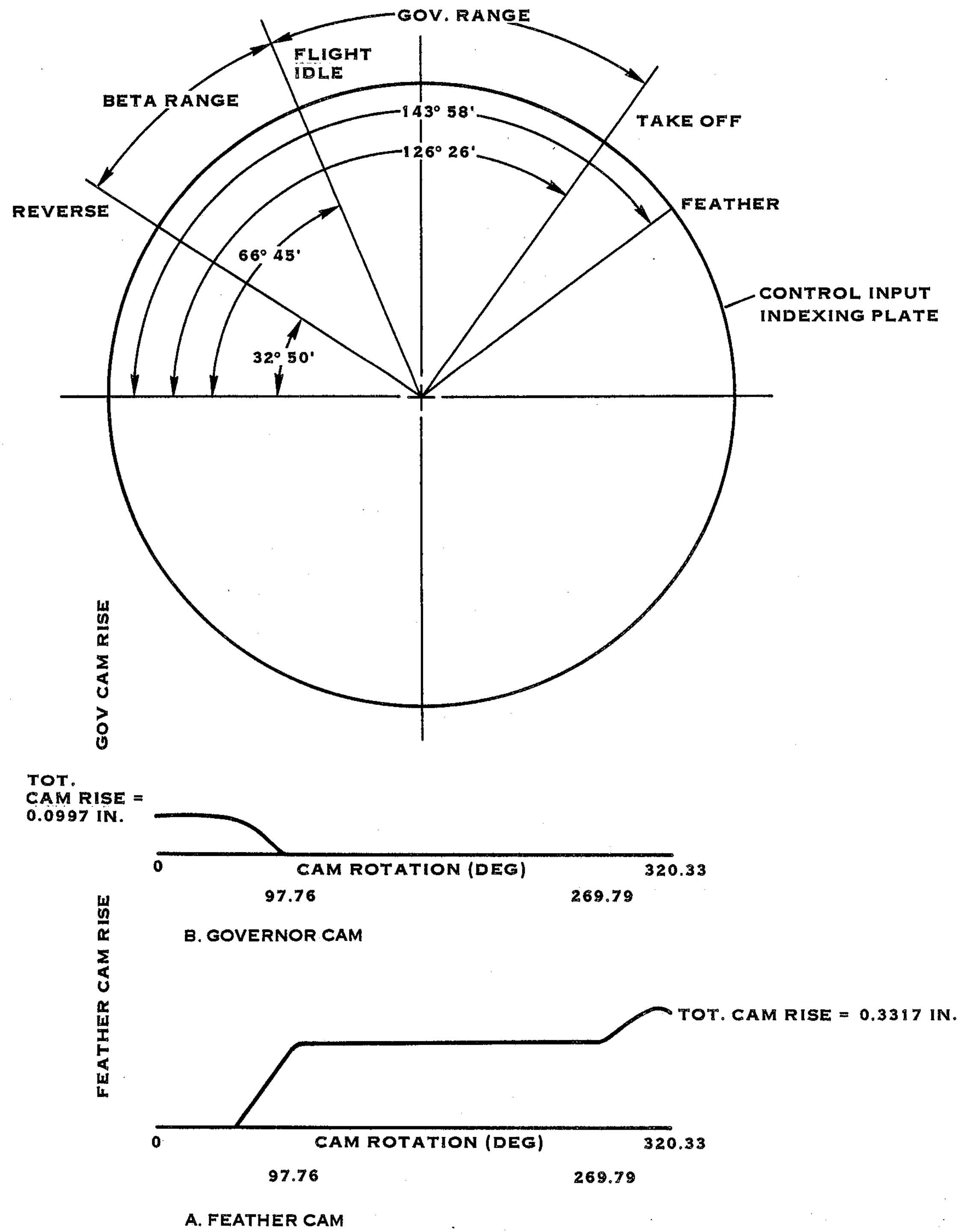
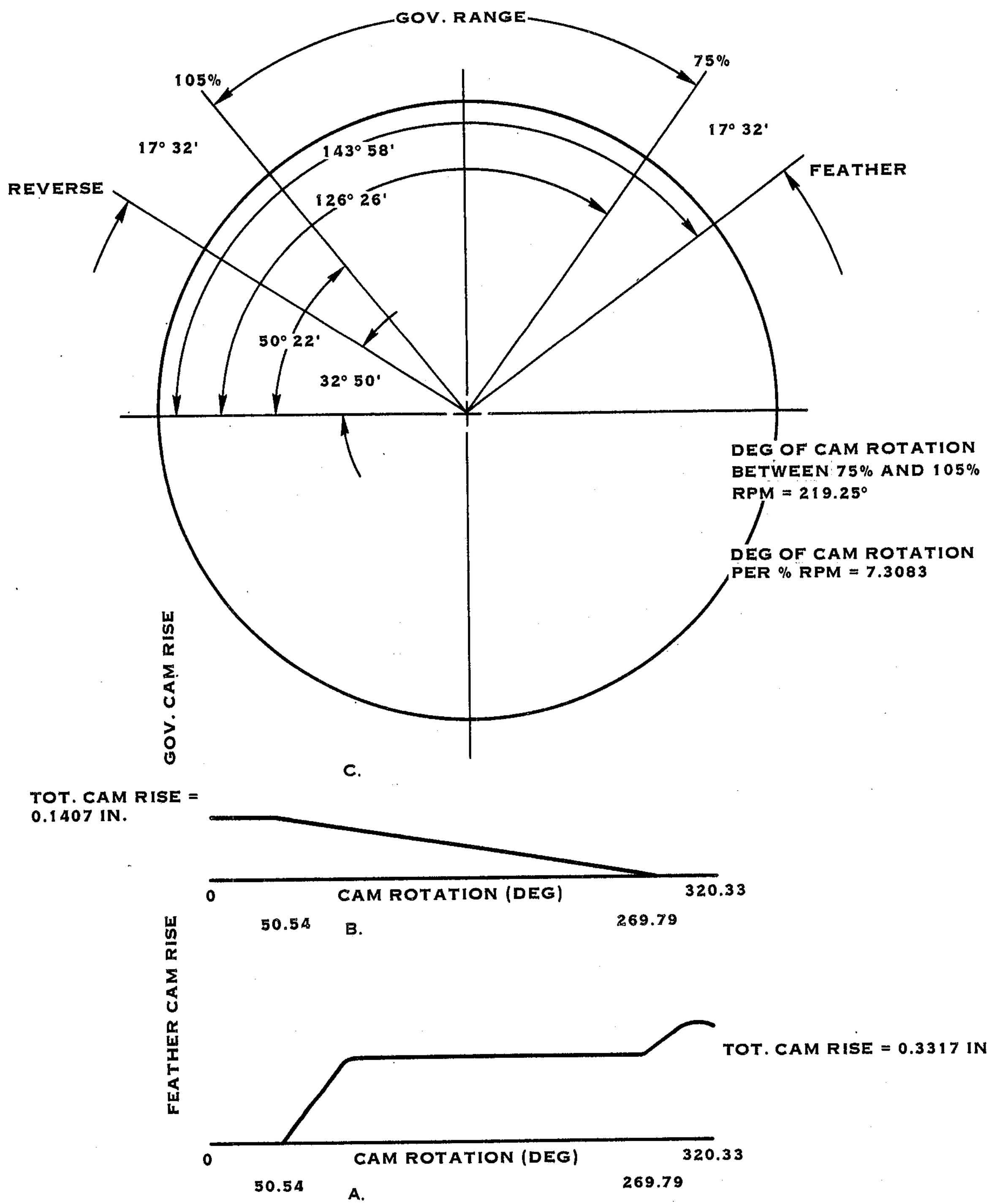
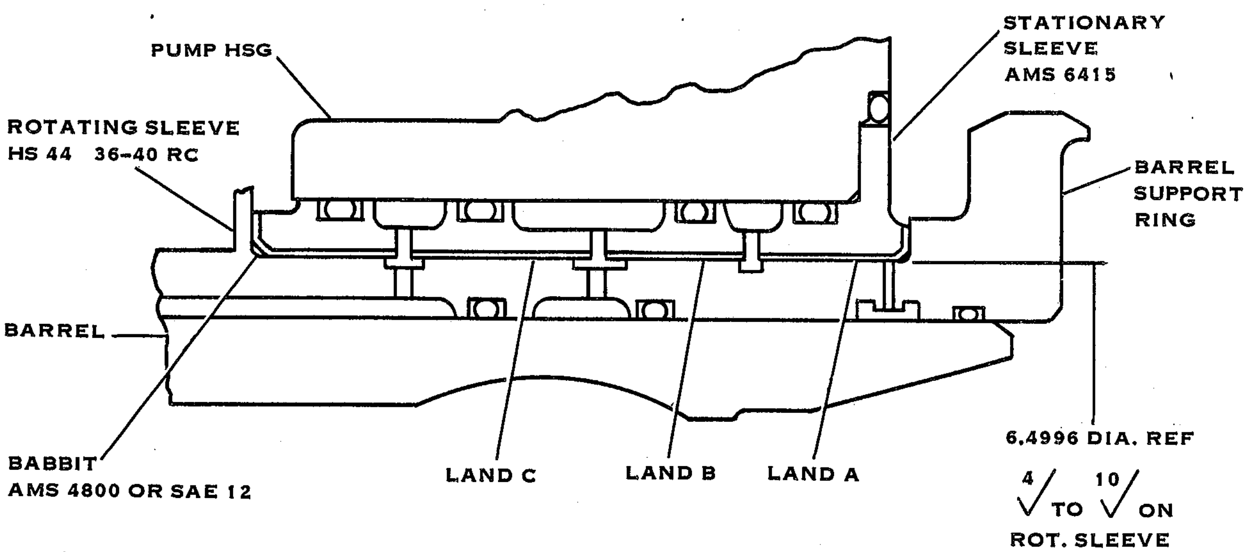


FIGURE 5.9. E2/C2 GOVERNOR AND FEATHER CAM SCHEDULE



ACTUAL CAM PROFILE EXTENDS 10° BEYOND REVERSE AND 10° BEYOND FEATHER (105% AND 75% RPM REF) AT A CONSTANT CENTER DISTANCE BETWEEN THE CAM AND ROLLER

FIGURE 5.10. LAP GOVERNOR AND FEATHER CAM SCHEDULE



DIAMETER CLEARANCE (IN)		
LAND	LAP	E2/C2
A	0.00155-0.00175	0.0035-0.0040
B	0.00155-0.00175	0.00155-0.00175
C	0.00155-0.00175	0.00165-0.00185

FIGURE 5.11. TRANSFER BEARING MODIFICATIONS

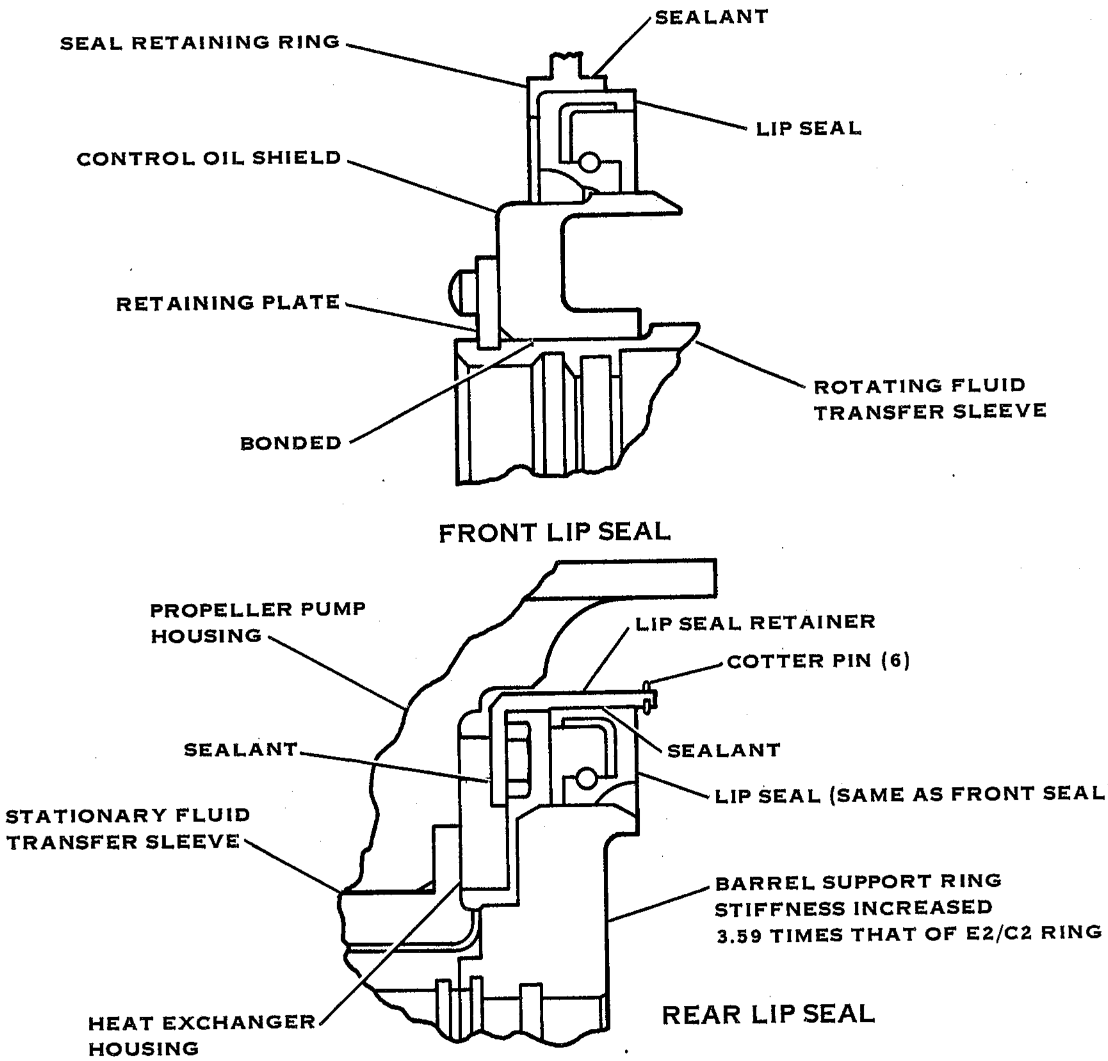


FIGURE 5.12. LIP SEALS

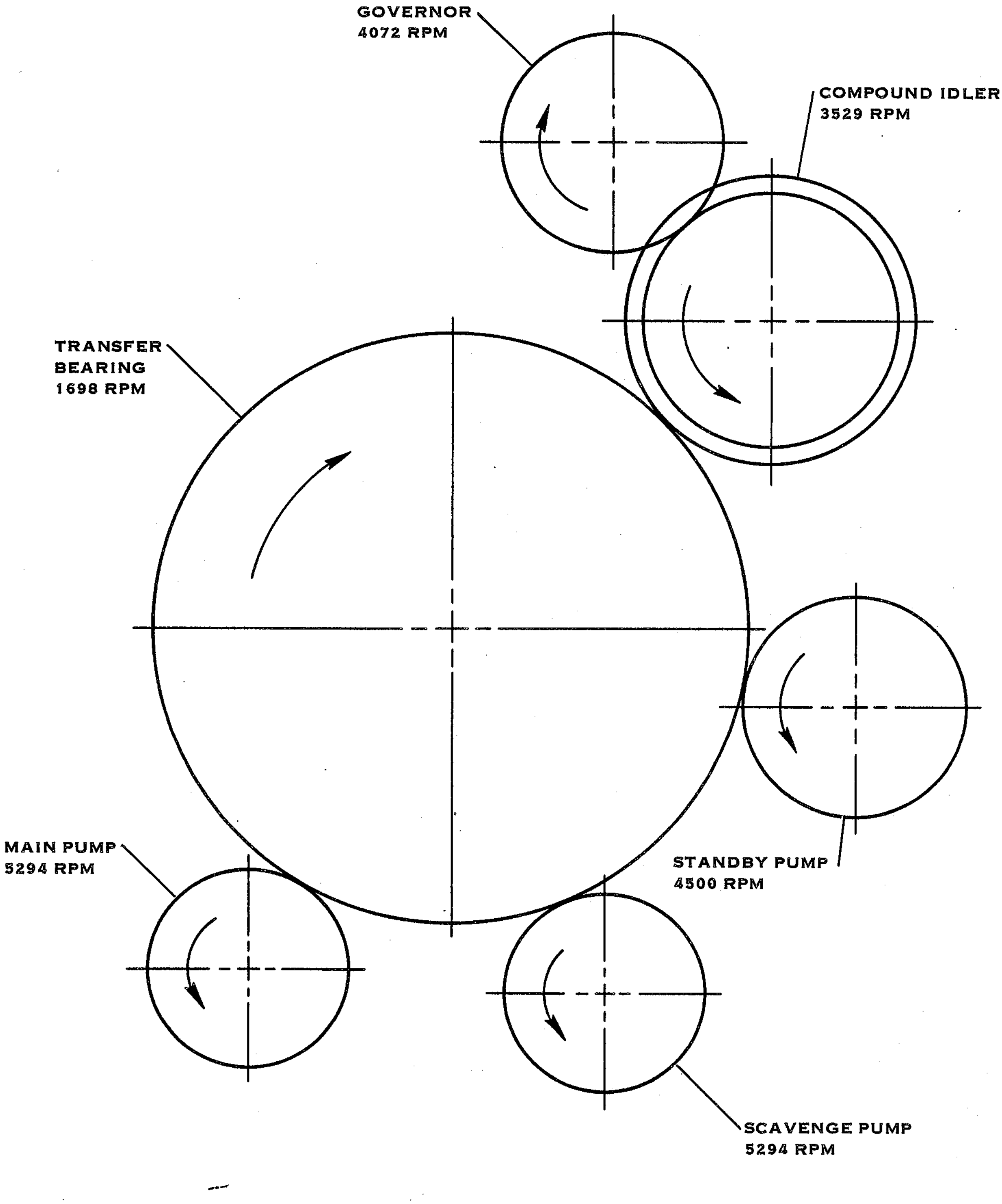


FIGURE 5.13. GEAR TRAIN SCHEMATIC - VIEW LOOKING AFT

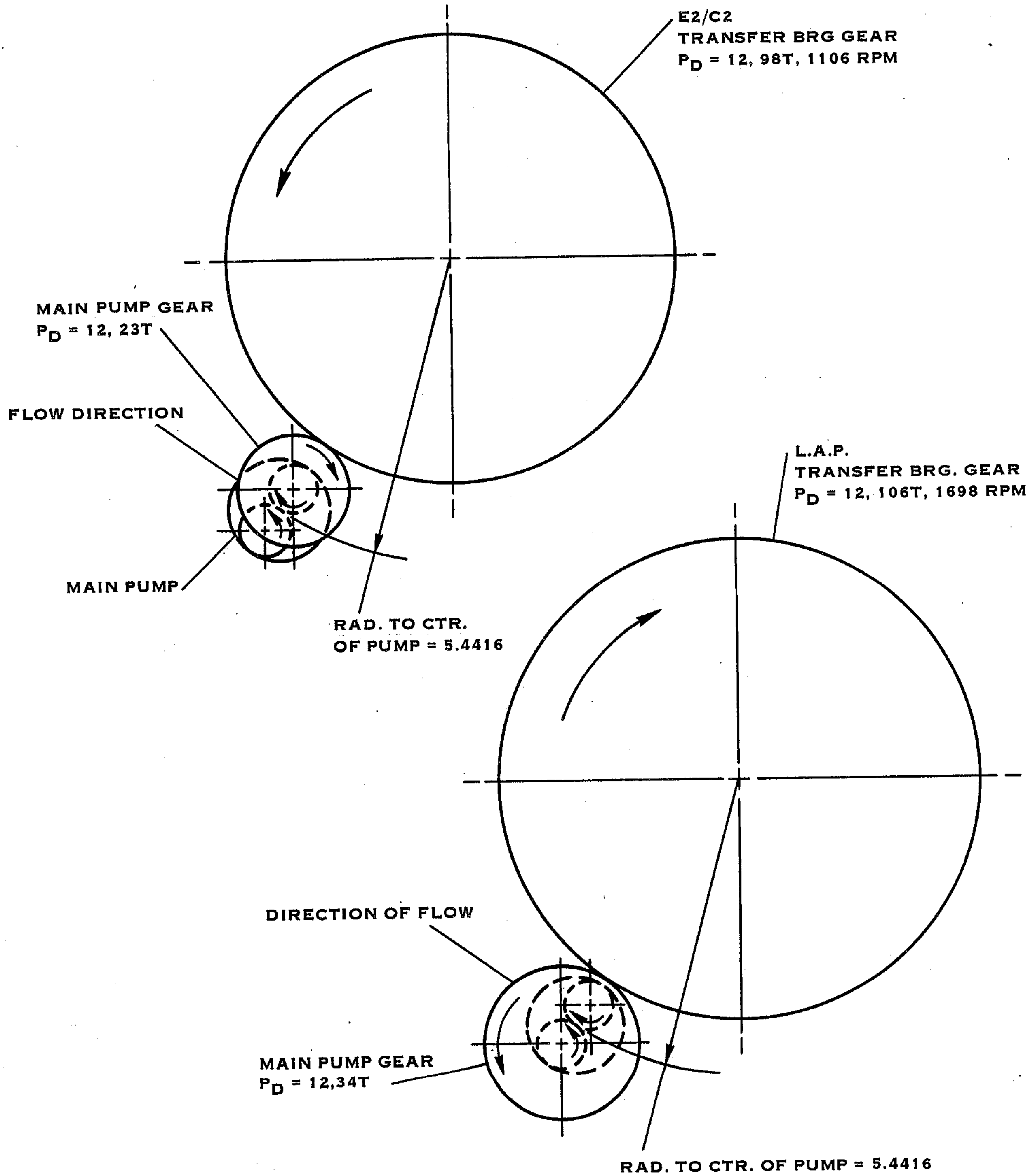
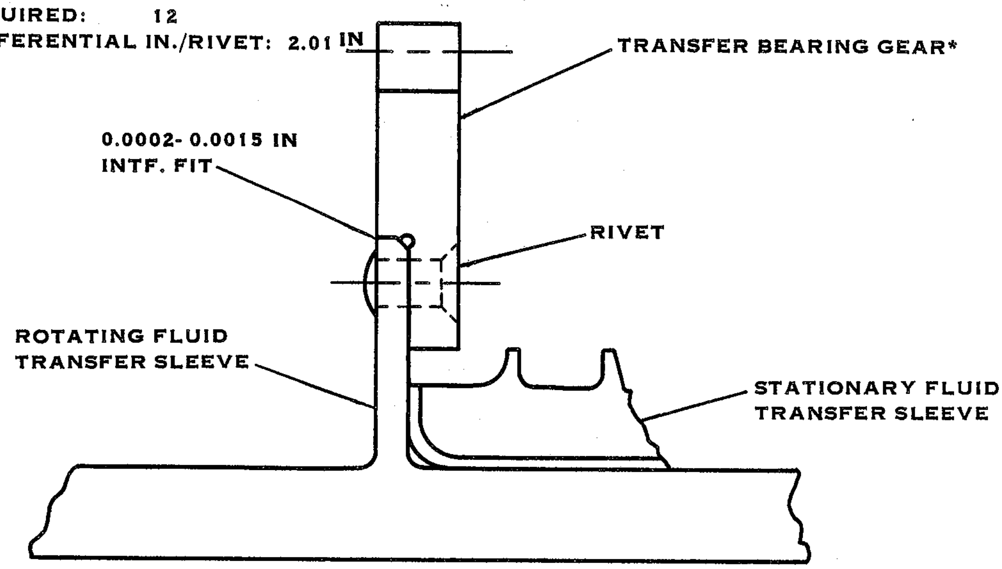


FIGURE 5.14. E2/C2 AND LAP MAIN PUMP GEARS - VIEW LOOKING AFT

TRANSFER BRG. GEAR
MATERIAL: AMS 6260
HARDNESS: 685 VH MIN (CASE), 30-42 RC (CORE)

RIVETS
MATERIAL: CORROSION RESISTANT STEEL
NO. REQUIRED: 12
CIRCUMFERENTIAL IN./RIVET: 2.01 IN



*MODIFICATIONS TO THE PRESENT TRANSFER BRG. GEAR (P/N 750621) WERE REQUIRED IN ORDER TO OBTAIN A LARGER DIAMETER DRIVE GEAR

FIGURE 5.15. MODIFICATIONS TO E2/C2 ROT. TRANSFER SLEEVE GEAR

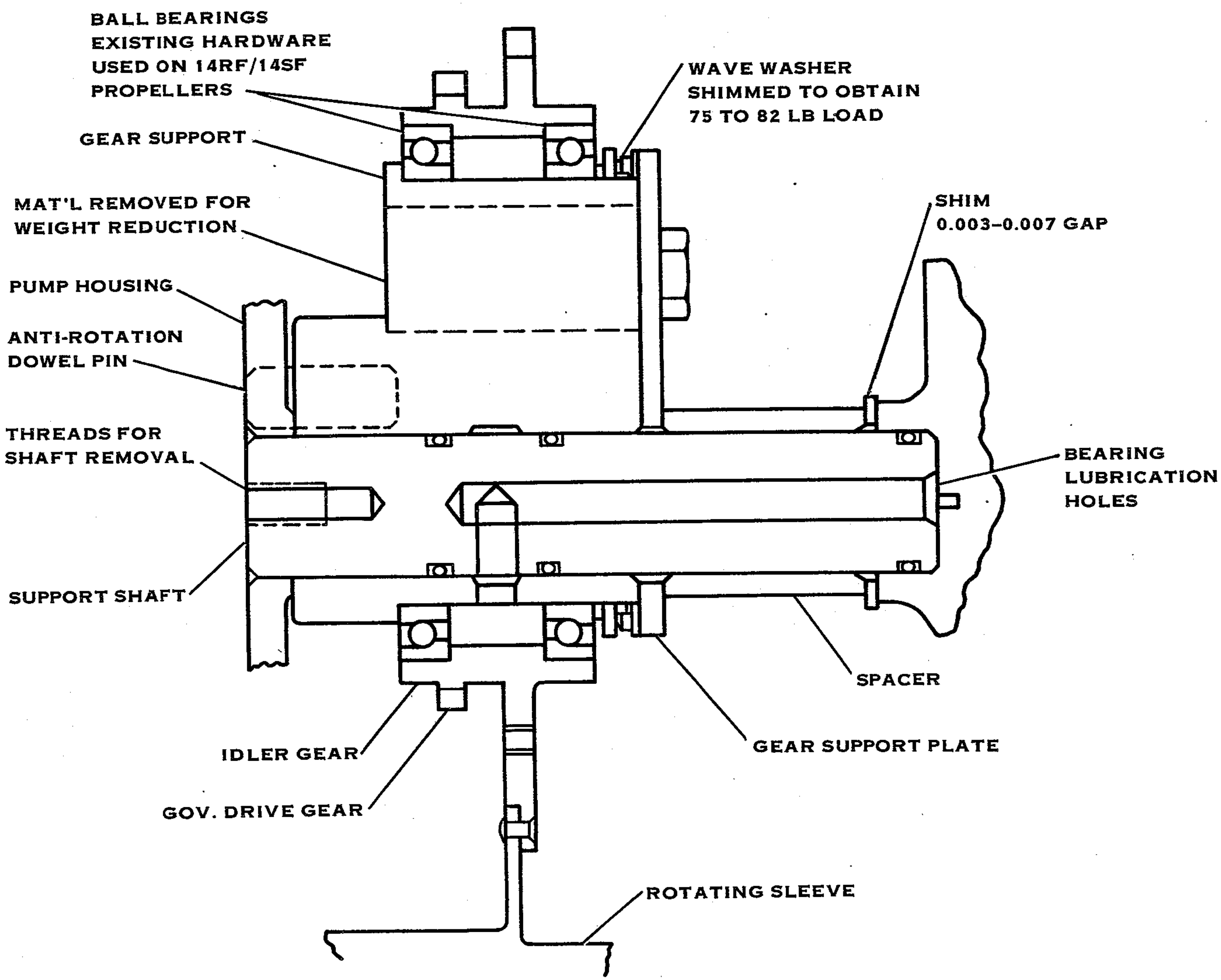
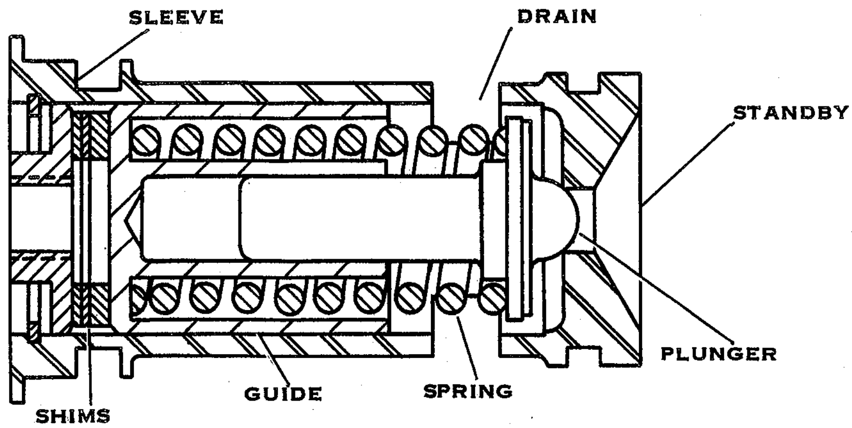


FIGURE 5.16. IDLER GEAR SUPPORT



MATERIALS

SLEEVE:	AMS 5612, 30-38 HRC
GUIDE:	AMS 5612, 30-38 HRC
PLUNGER:	AMS 5610, 30-38 HRC
SHIMS:	QQ-A-250/4 COND T3

DIMENSIONS

SLEEVE & GUIDE DIA. CLR.	0.00095 ± 0.00055 IN
GUIDE & PLUNGER DIA. CLR.	0.002 ± 0.001 IN

PERFORMANCE

SPRING RATE:	130.6 LB/IN
NORMAL OPERATING RANGE	1150- 1875 PSI

FIGURE 5.17. HIGH PRESSURE RELIEF VALVE DATA

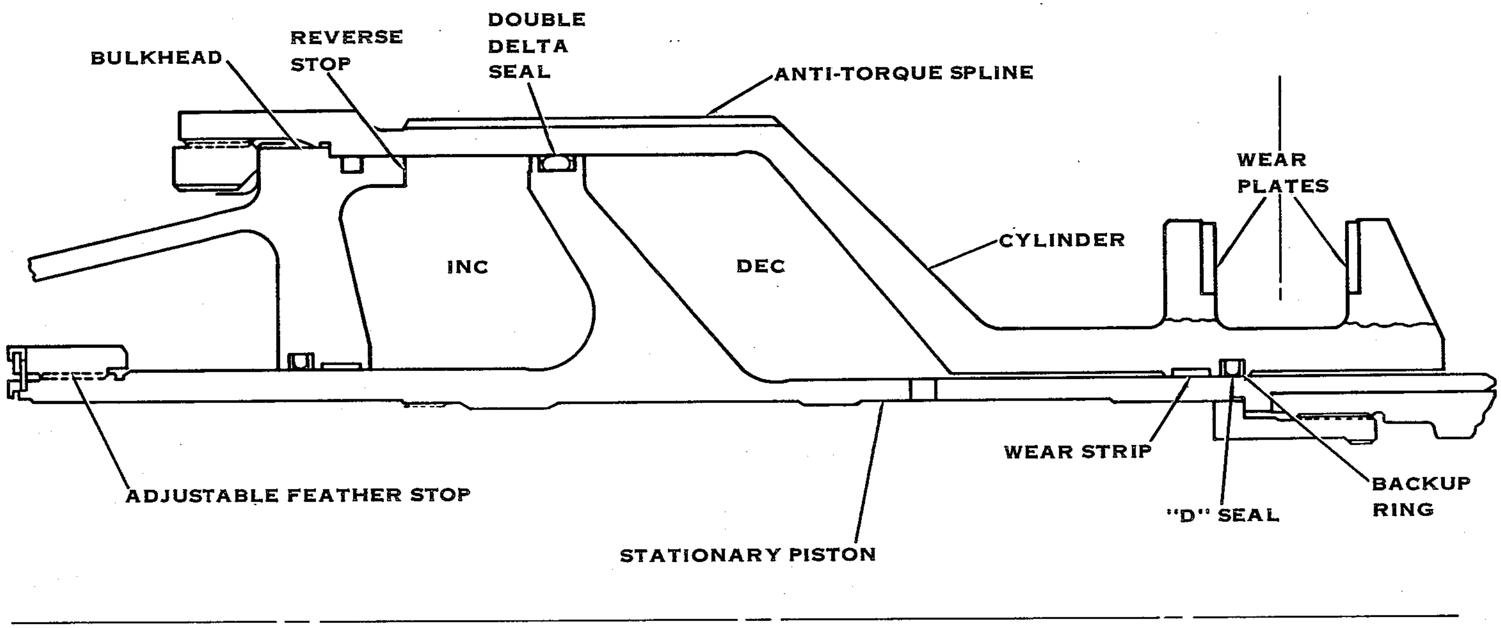


FIGURE 6.1 PITCH CHANGE ACTUATOR

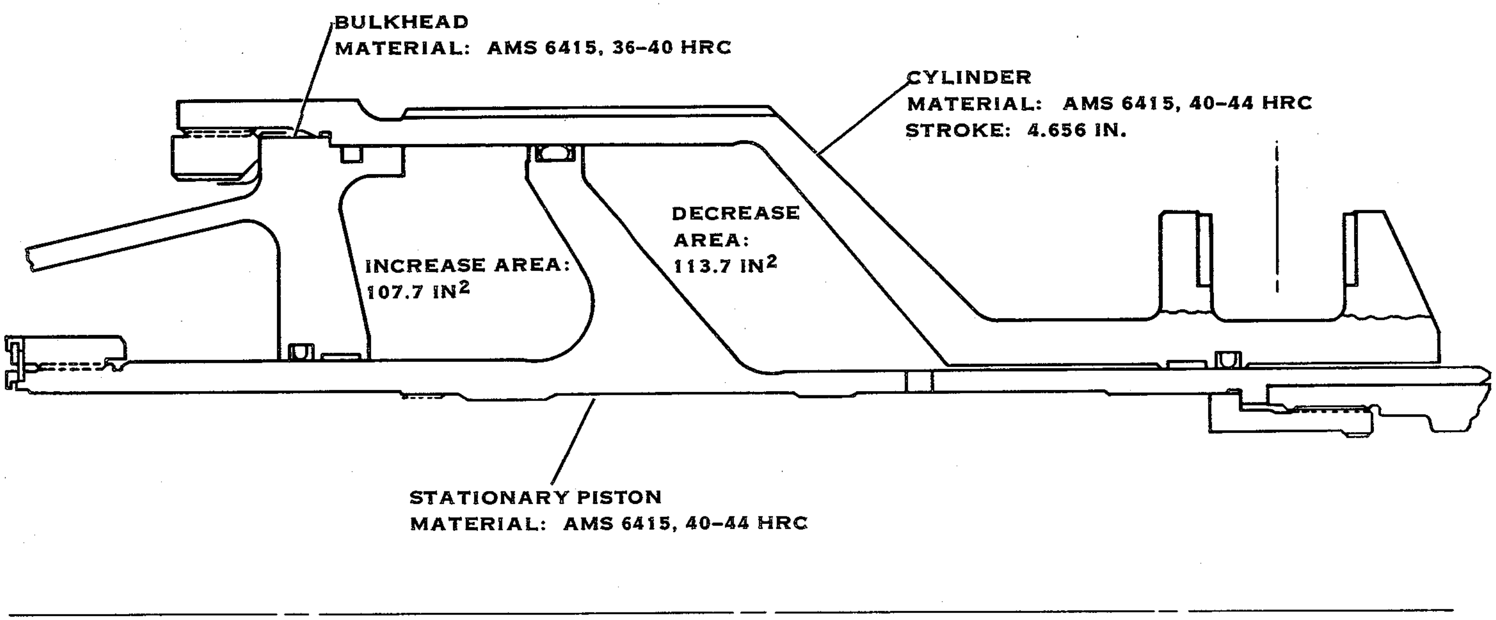


FIGURE 6.2. PITCH CHANGE ACTUATOR MATERIALS

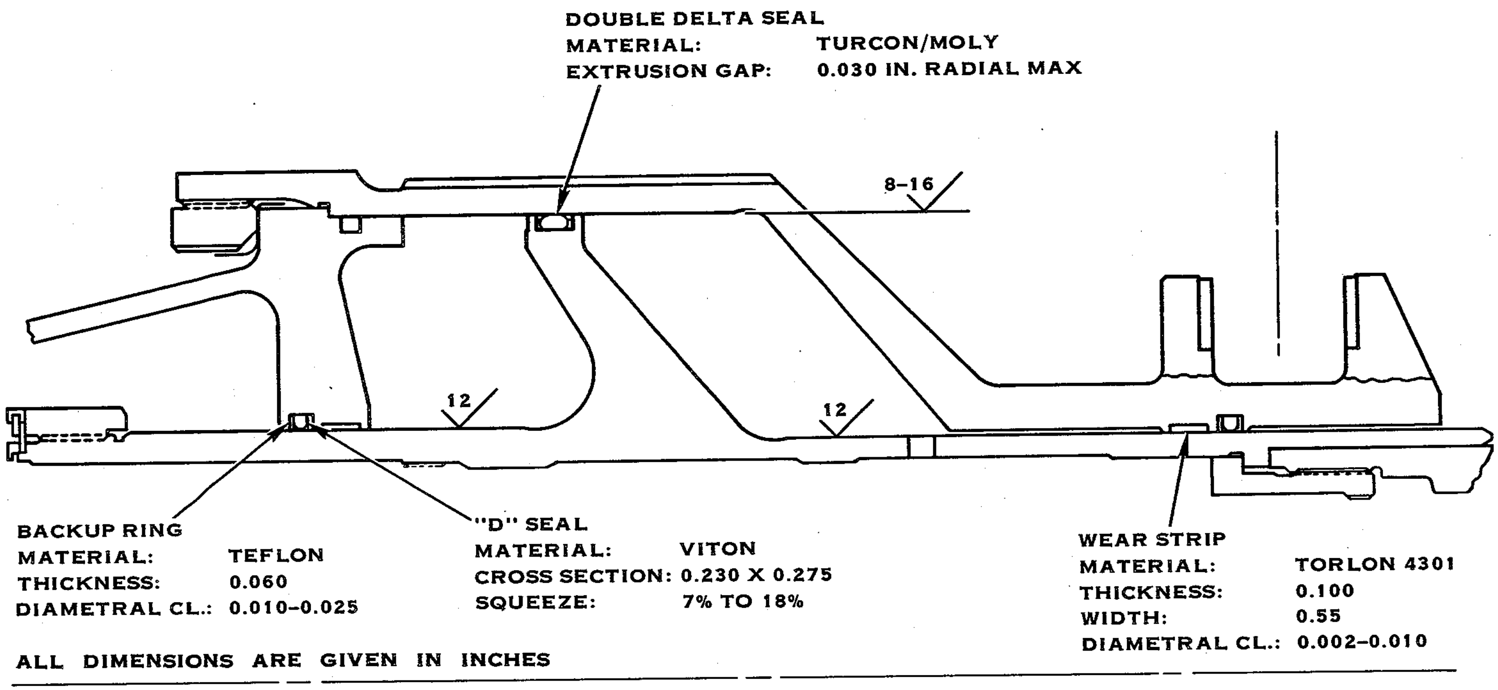


FIGURE 6.3. PITCH CHANGE ACTUATOR CLEARANCE

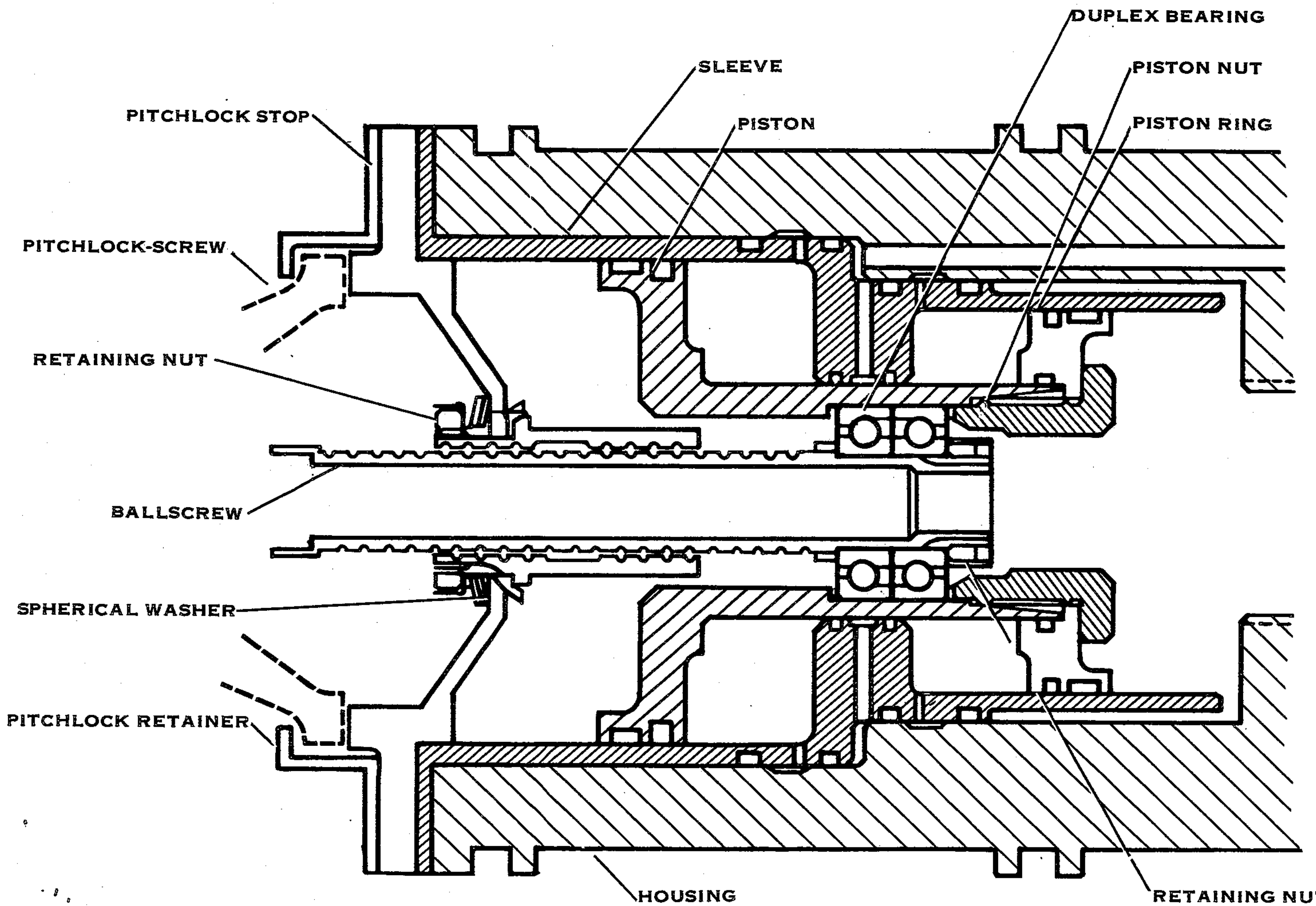


FIGURE 6.4. HALF AREA SERVO

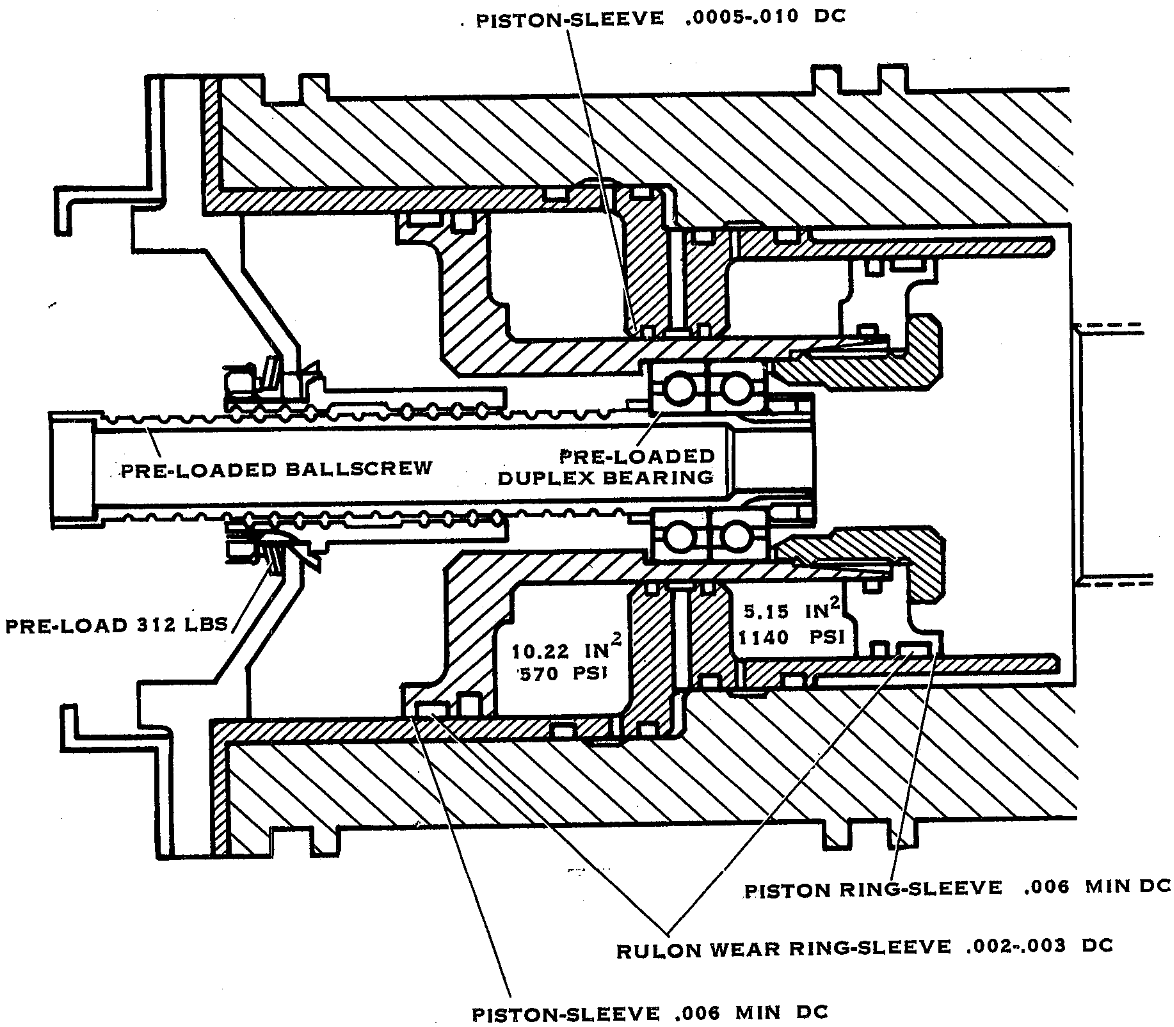


FIGURE 6.5. SERVO DOUBLE SEAL CONFIGURATION

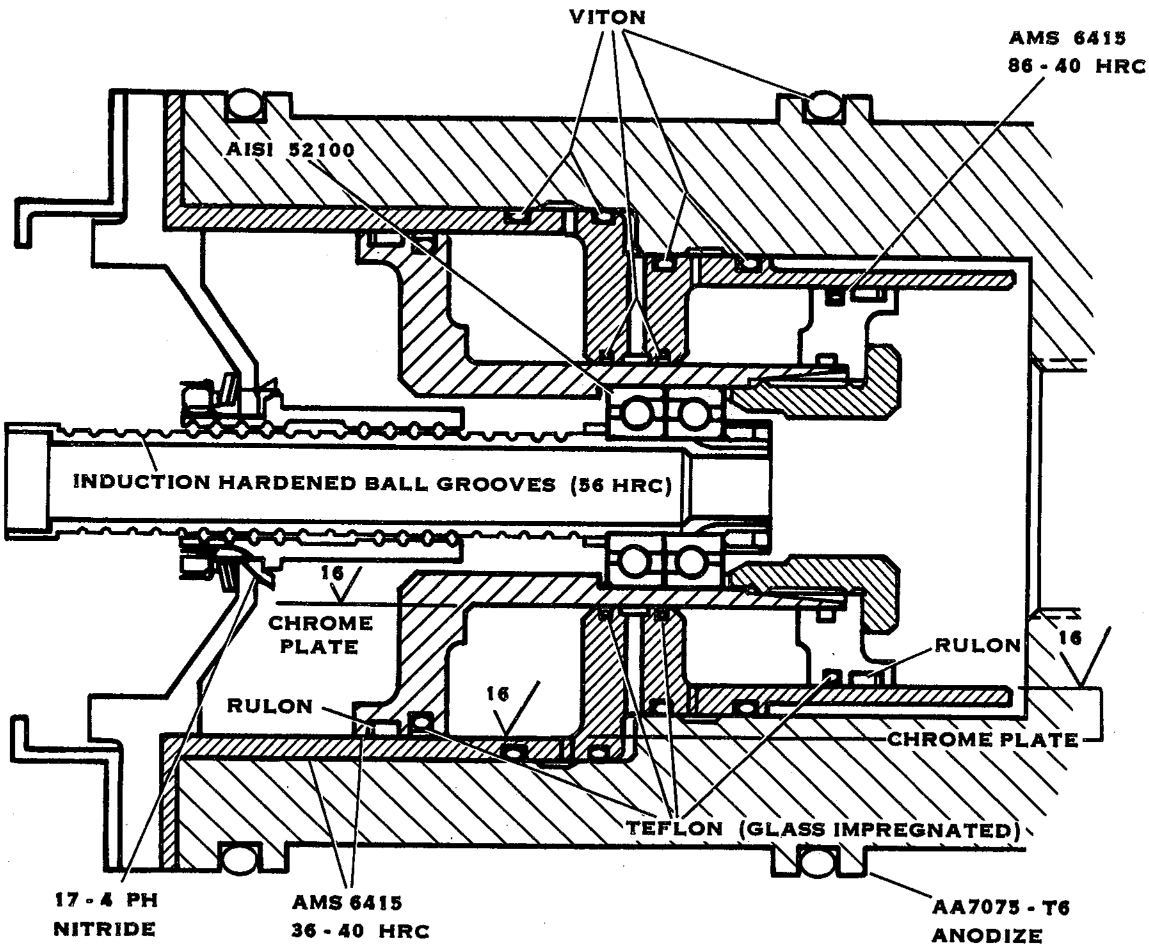


FIGURE 6.6. HALF AREA SERVO MATERIALS

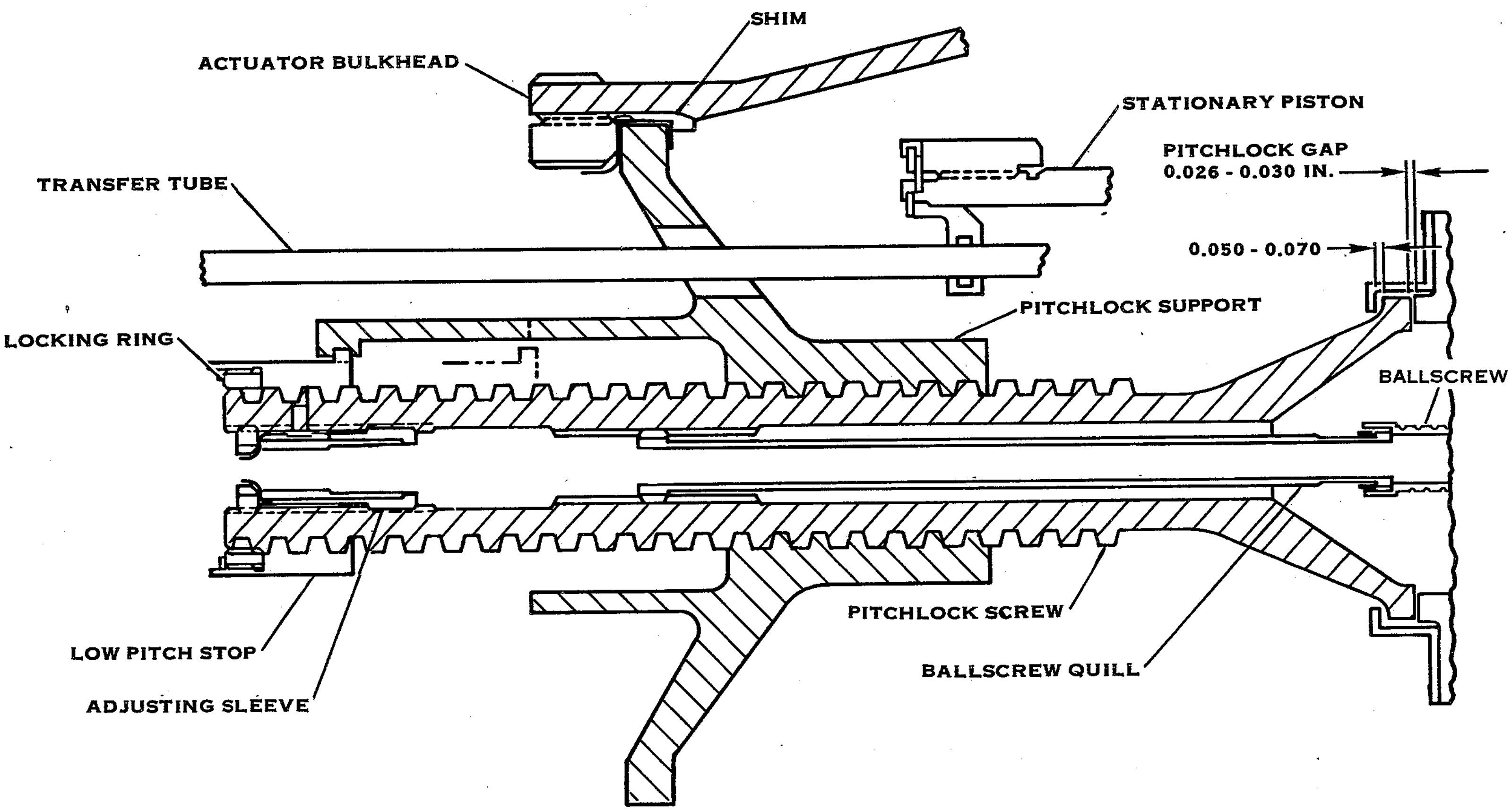


FIGURE 6.7. PITCHLOCK SCREW

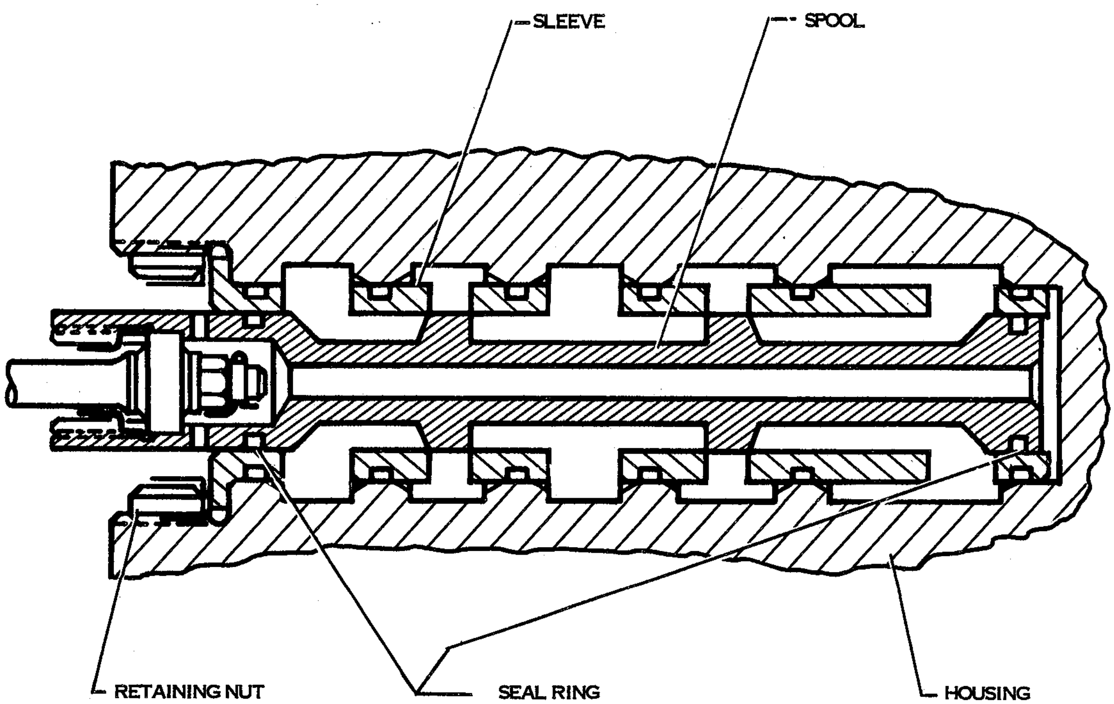
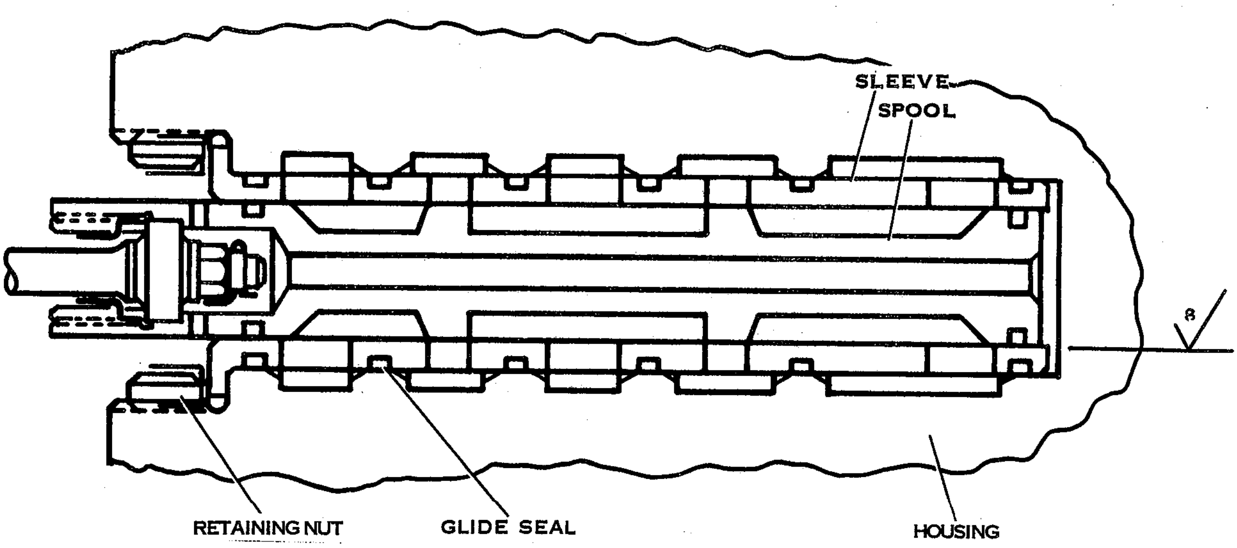


FIGURE 6.8. BETA VALVE



MATERIALS

SPOOL:	AMS 6415, 36-40 HRC CARBURIZE
SLEEVE:	AMS 6415, 36-40 HRC CARBURIZE
RETAINING NUT:	AA6061-T6 ANODIZE
GLIDE SEAL:	TEFLON (GRAPHITE IMPREGNATED)
HOUSING:	AA7075-T6 ANODIZE

DIMENSIONS

SPOOL & SLEEVE, DIA. CLR	.0003-.0006 IN.
SLEEVE & HOUSING, DIA CLR	.002-.008 IN.

FIGURE 6.9. BETA VALVE MATERIALS

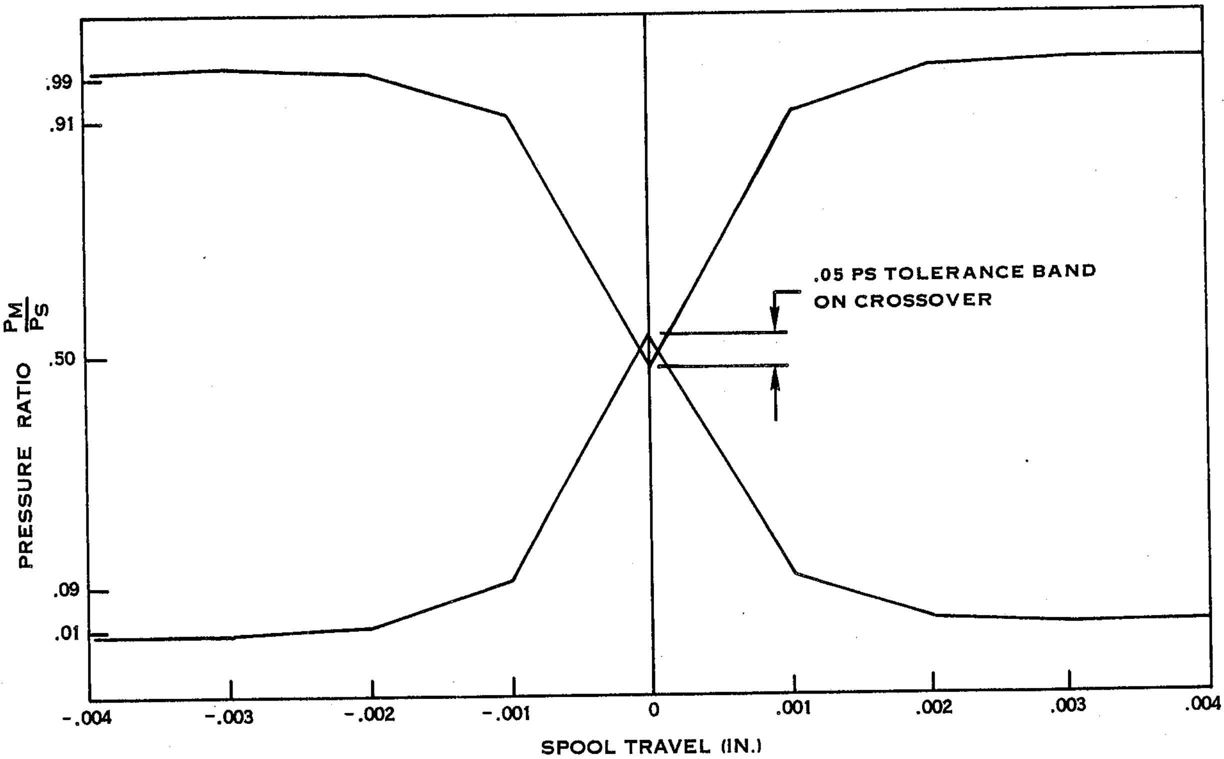


FIGURE 6.10 PRESSURE GAIN CHARACTERISTICS OF BETA VALVE

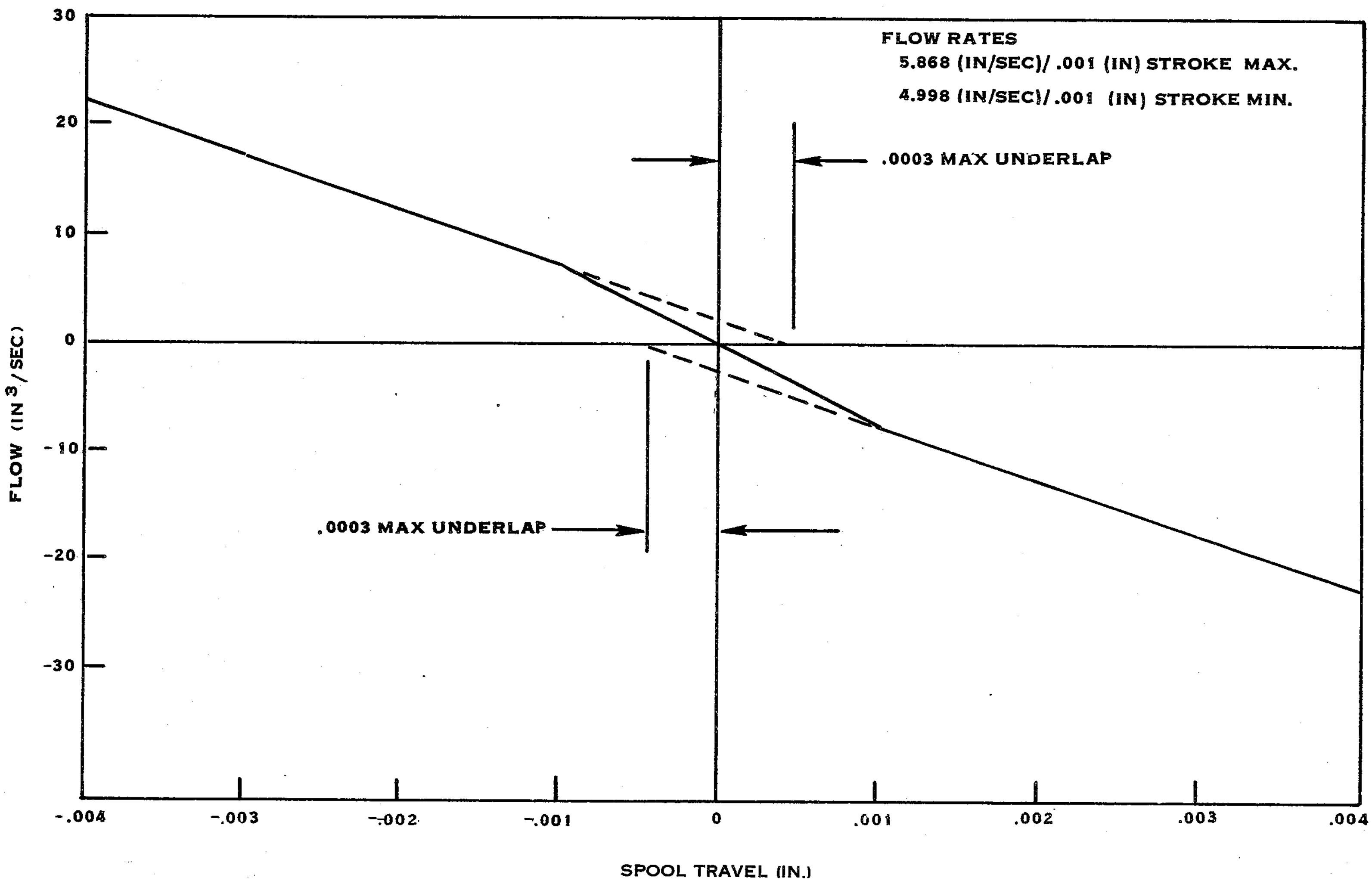


FIGURE 6.11 FLOW GAIN CHARACTERISTICS OF BETA VALVE

CASE: INCREASE PRESSURE WITH ACTUATOR AGAINST FEATHER STOP

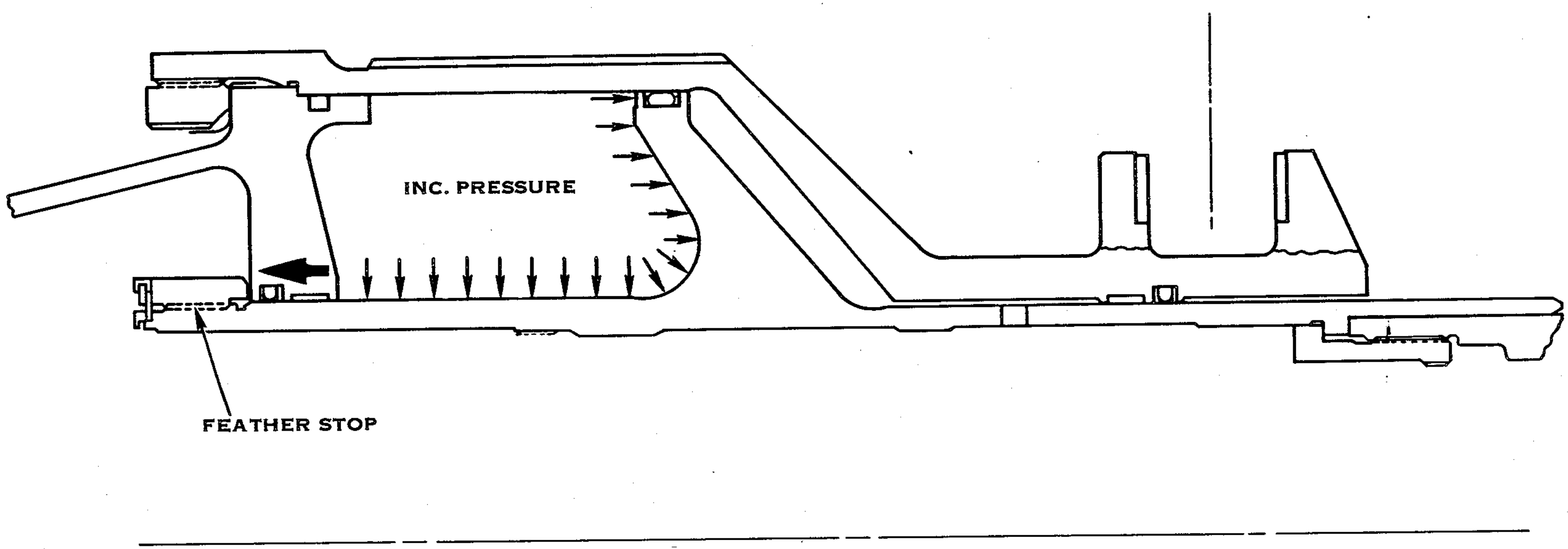


FIGURE 6.12 STATIONARY PISTON LOADS

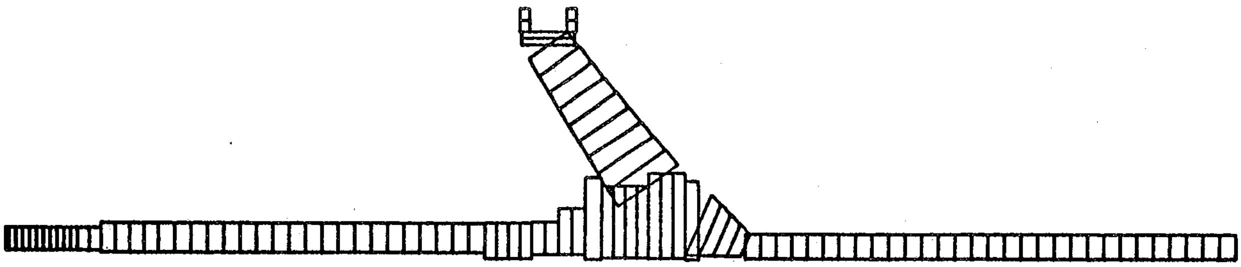


FIGURE 6.13 STATIONARY PISTON INPUT GEOMETRY

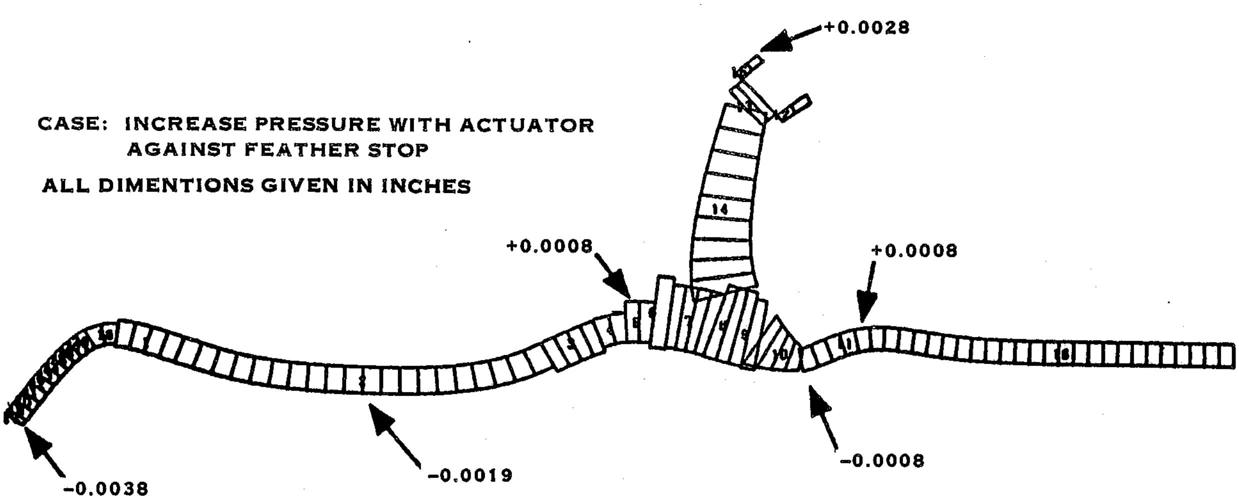


FIGURE 6.14 STATIONARY PISTON RADIAL DEFLECTIONS @ 1375 PSI

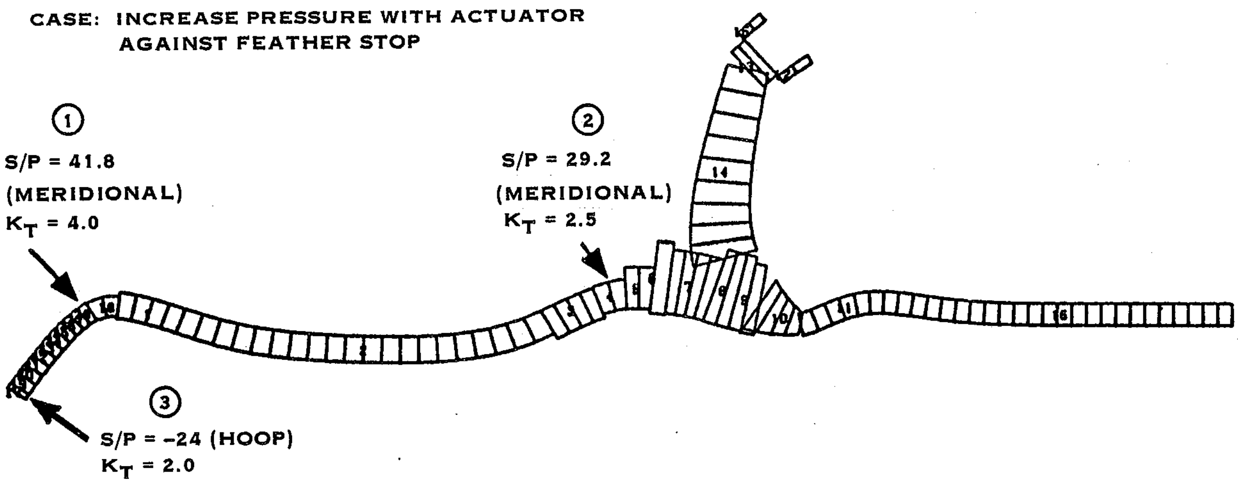


FIGURE 6.15 STATIONARY PISTON STRESSES

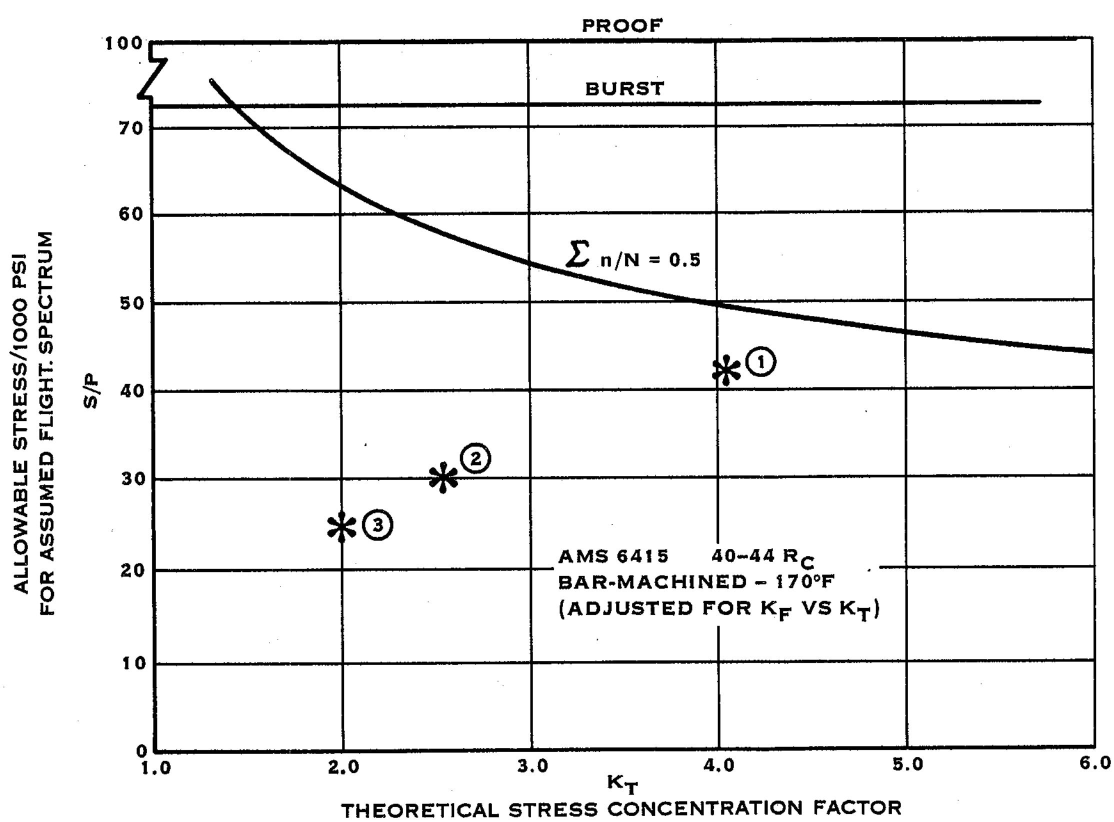


FIGURE 6.16 STATIONARY PISTON S/P CURVE

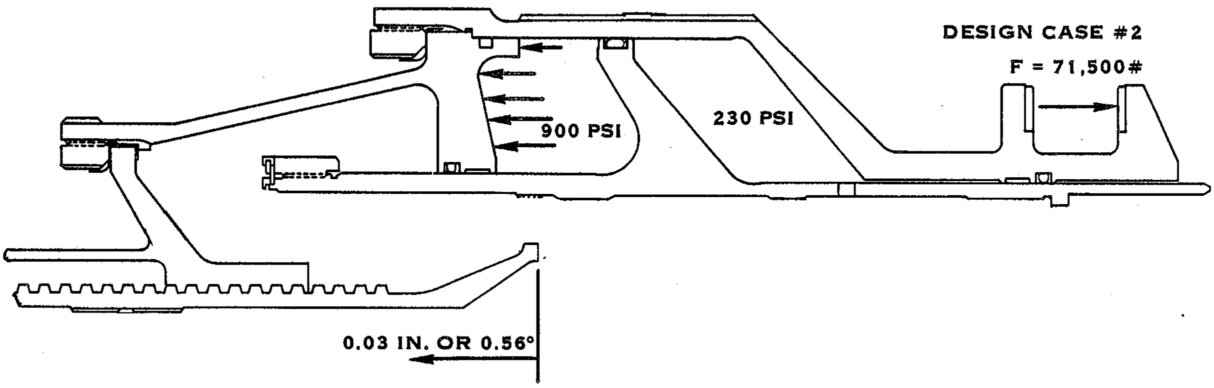
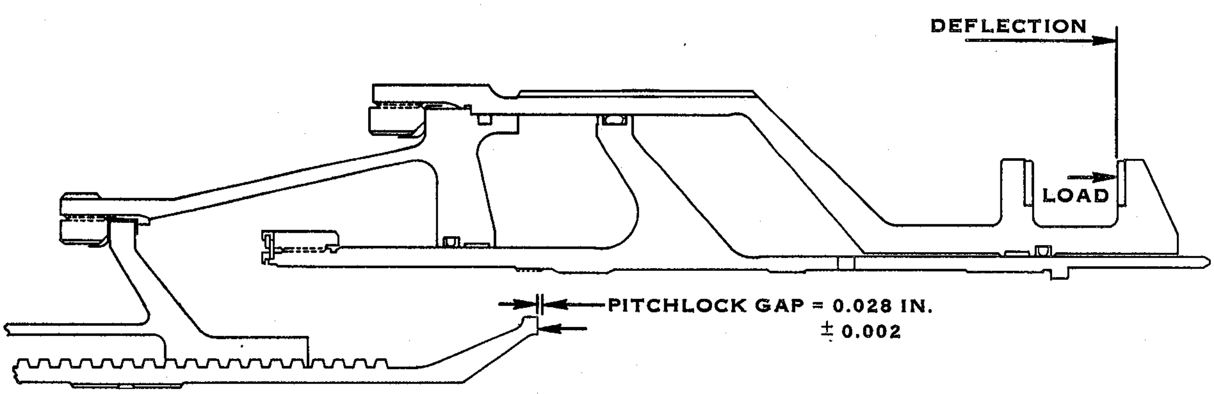


FIGURE 6.17 AXIAL DEFLECTIONS OF ACTUATOR - NORMAL OPERATING CONDITION



CASE	LOAD	DEFL. + 0.030	BLADE ANGLE CHANGE
110% O.S.	97,500#	0.082	1.58°
100% V_{MAX}	71,500#	0.068	1.30°
CRUISE & CLIMB	66,000#	0.065	1.24°

FIGURE 6.18 AXIAL DEFLECTIONS OF ACTUATOR PITCHLOCK CONDITION

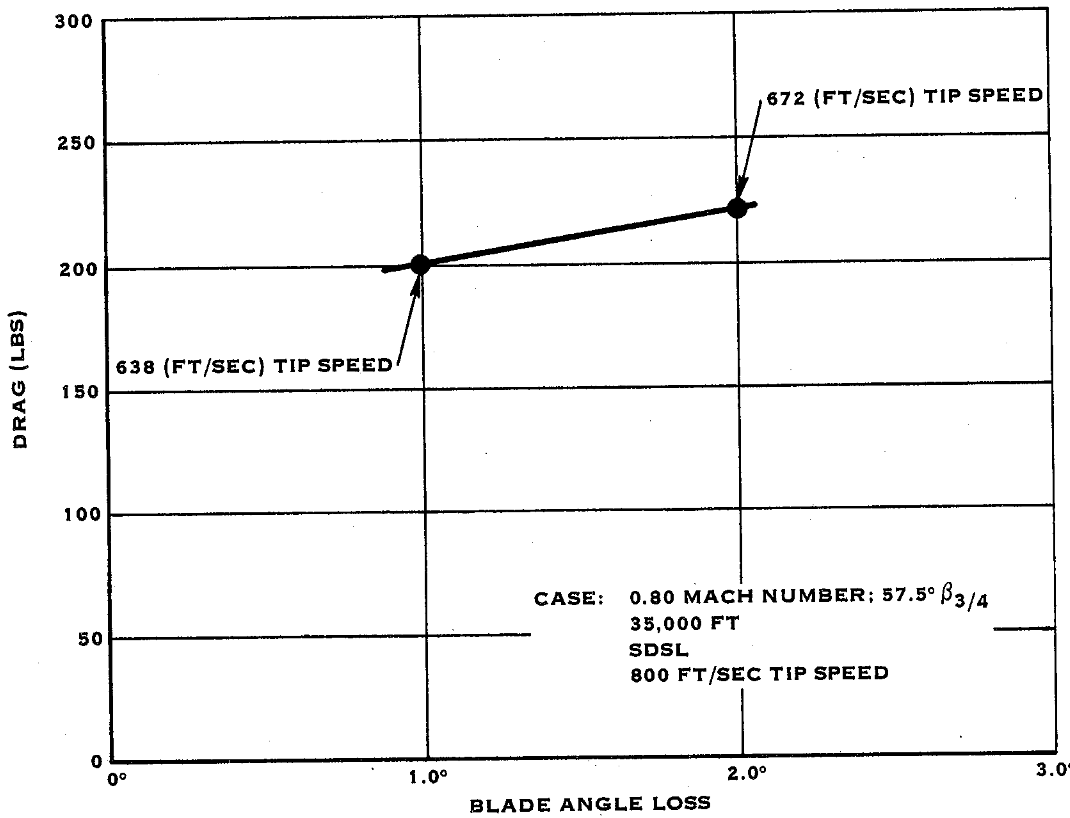


FIGURE 6.19 DRAG AND PROPELLER SPEED FOR PITCHLOCK CONDITION

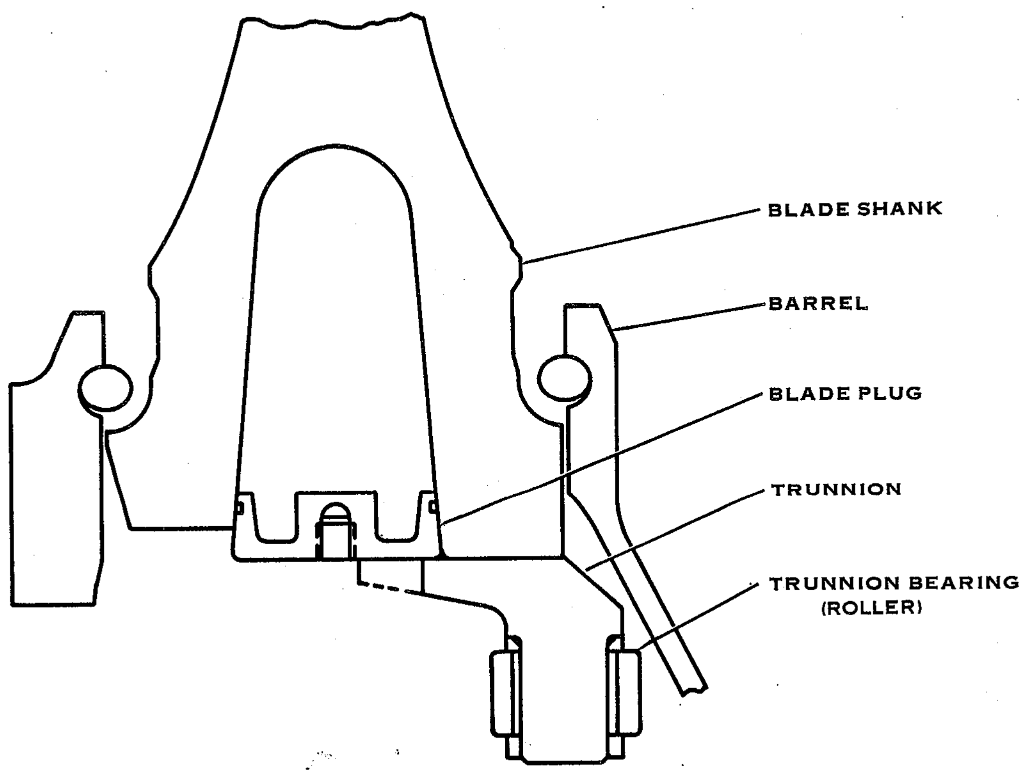


FIGURE 6-20. BLADE TRUNNION ASSEMBLY

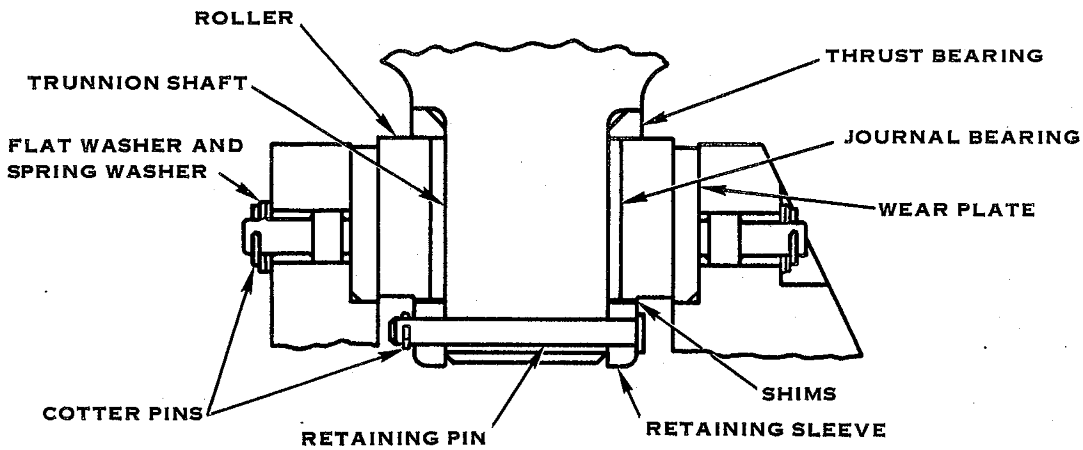


FIGURE 6.21 TRUNNION ROLLER AND WEAR PLATES

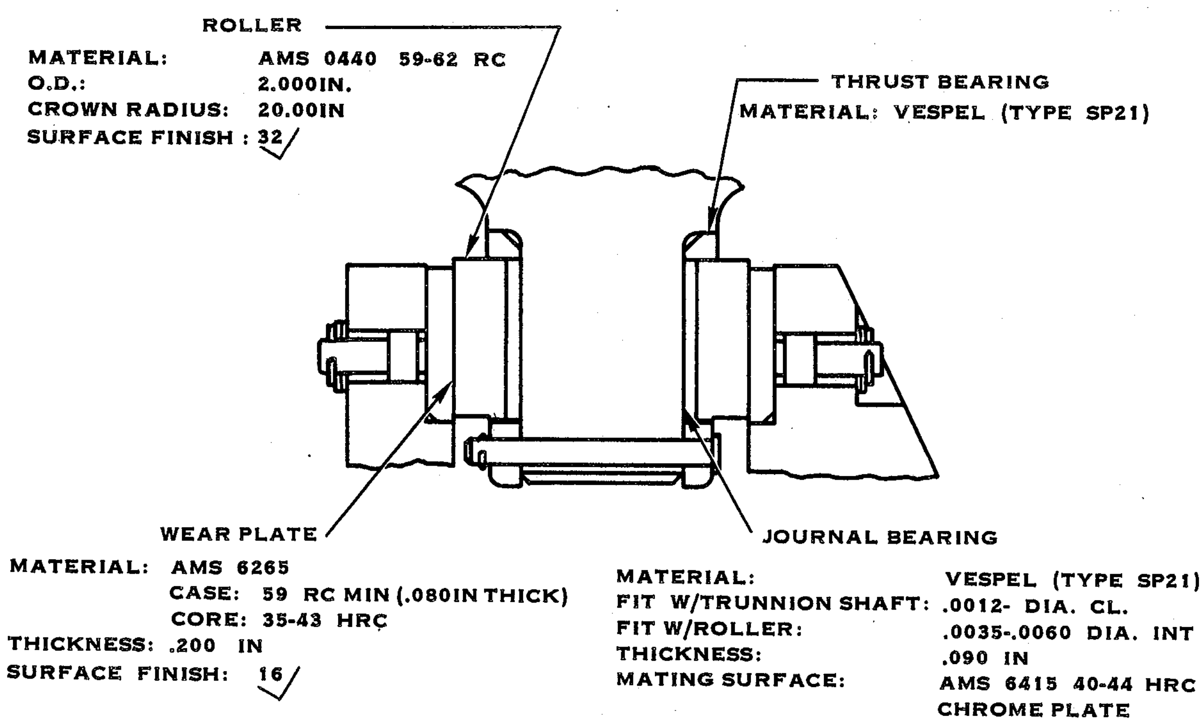


FIGURE 6.22 TRUNNION ROLLER AND WEAR PLATE MATERIALS

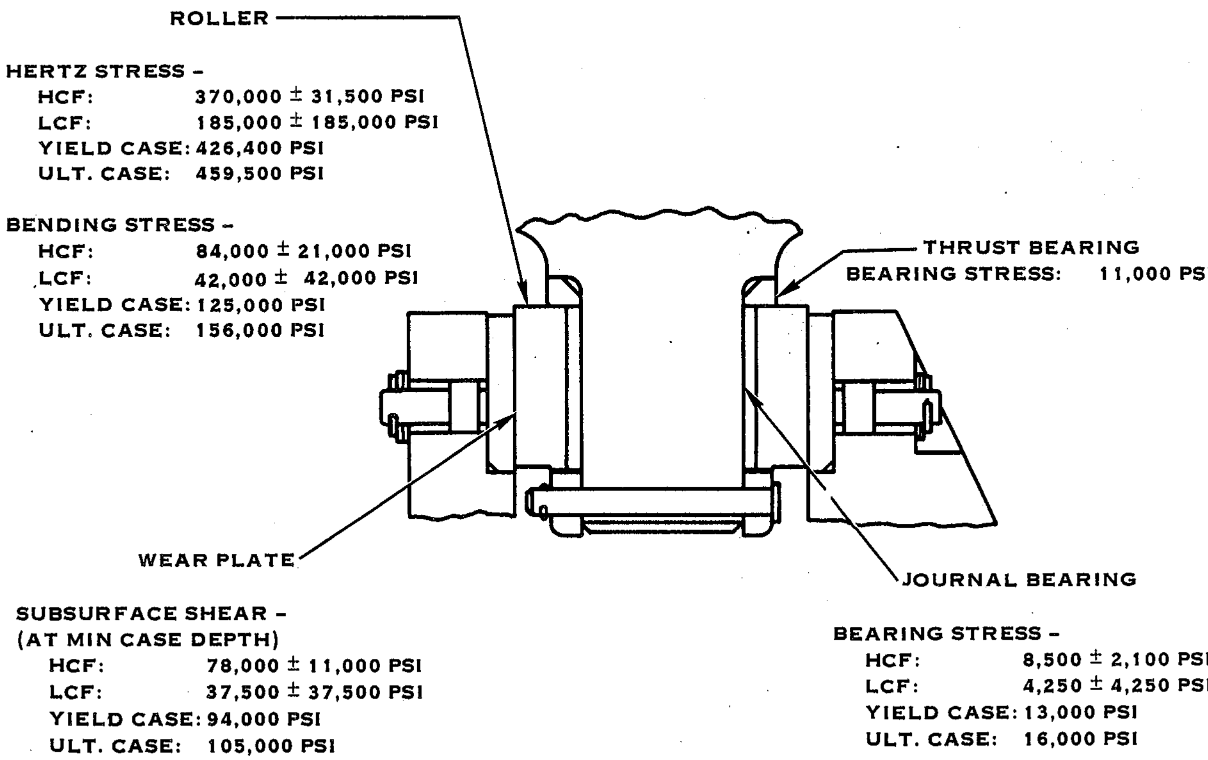


FIGURE 6.23 TRUNNION ROLLER AND WEAR PLATE STRESSES

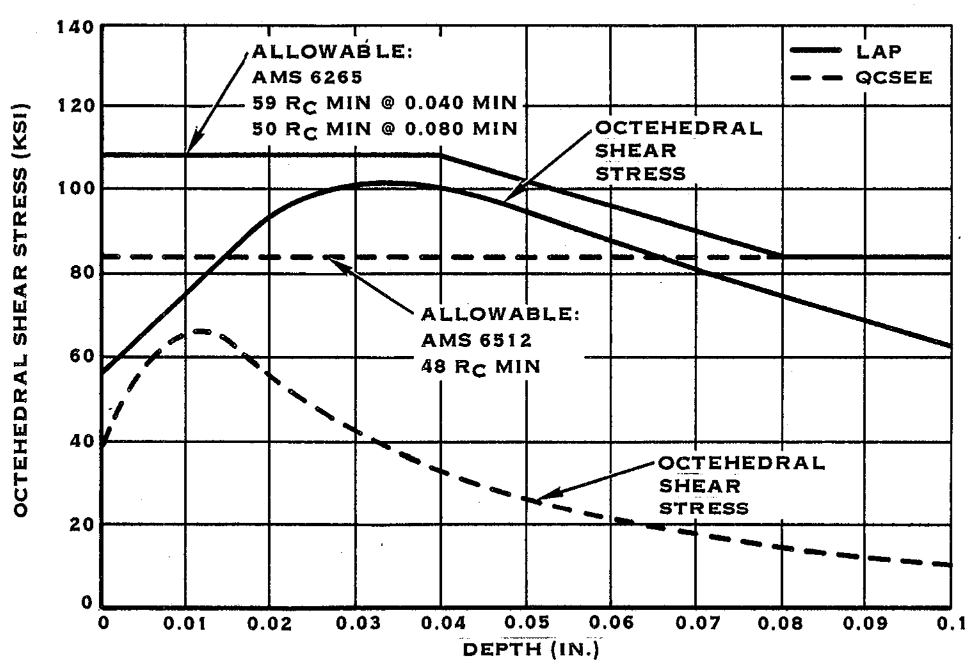


FIGURE 6.24 LAP VS. QCSEE STRESS @ 10^4~

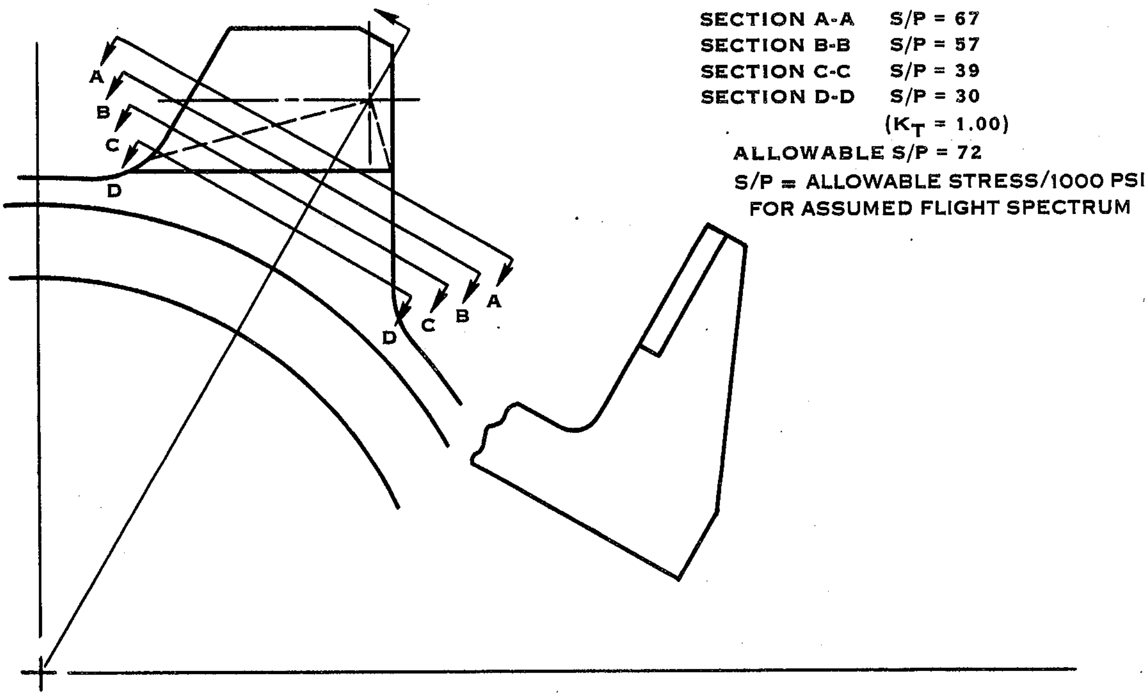


FIGURE 6.25 ACTUATOR EAR STRESS

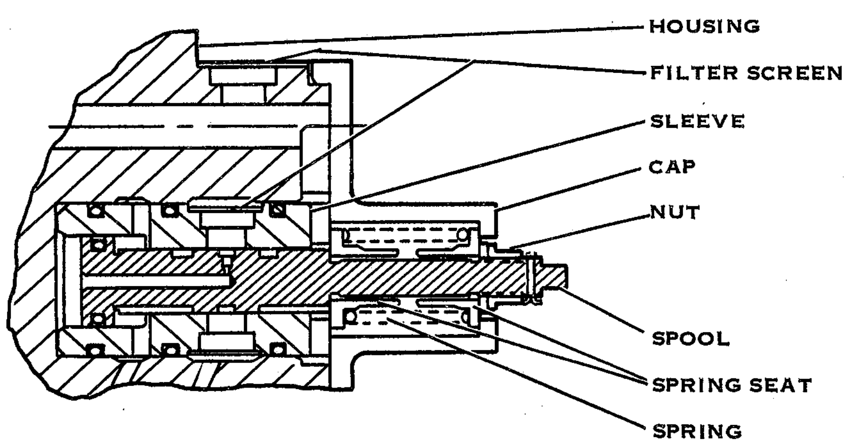


FIGURE 6-26. ΔP REGULATING VALVE

MAX SERVO LOAD		FRICTION LOAD AND SPRING
1	1400#	15# FRICTION MAX SPRING
2	1100#	5# FRICTION MAX SPRING
3	800#	0# FRICTION MIN SPRING

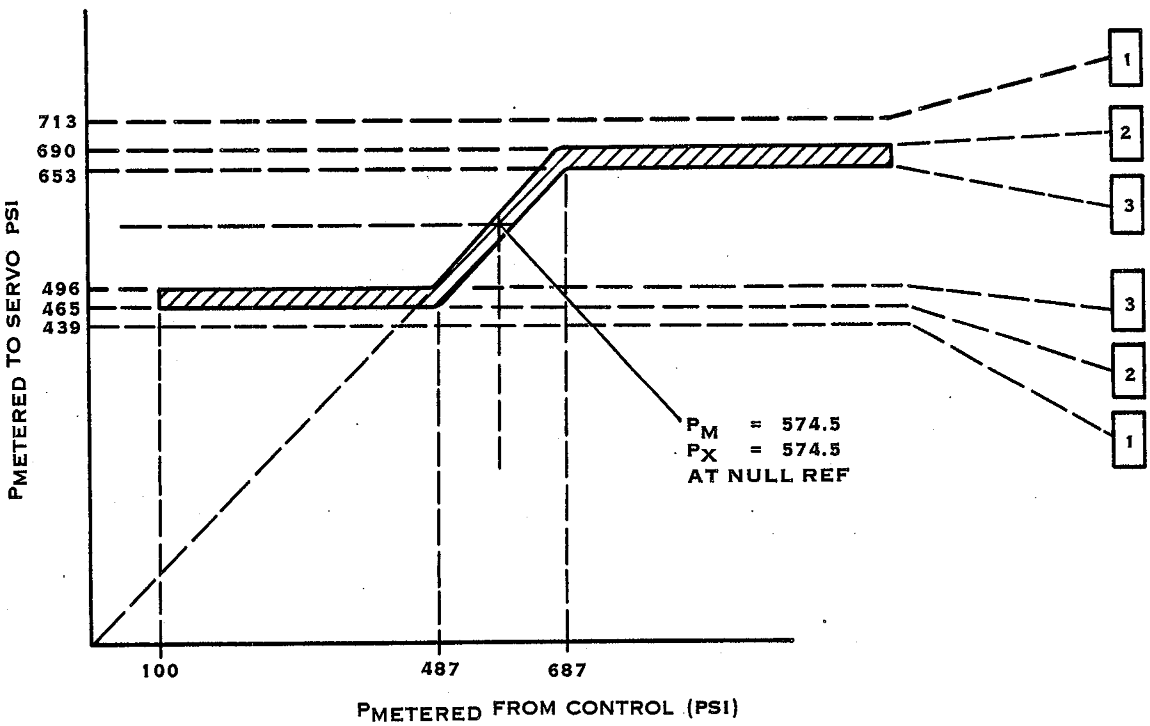
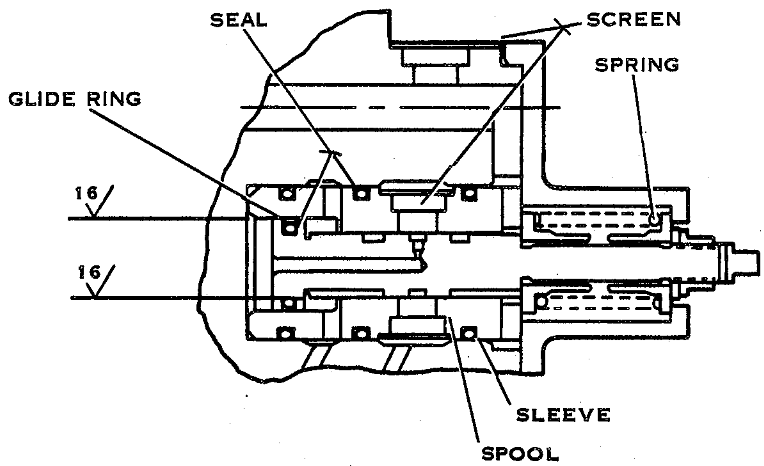


FIGURE 6.27 ΔP REGULATOR CHARACTERISTICS (REF: FIG. 3.3)



MATERIALS

SLEEVE:	AMS 6260, 35-43 HRC, CARBURIZE
SPOOL:	AMS 6260, 35-43 HRC CARBURIZE
SCREEN:	AMS 5513
SEAL:	VITON
GLIDE RING:	TEFLON (GLASS IMPREGNATED)
SPRING:	AMS 5112

DIMENSIONS

SPOOL & SLEEVE, DIA. CLR (SMALL DIA.)	.0004-.0006 IN.
SPOOL & SLEEVE, DIA. CLR (LARGE DIA.)	.0005- 0007 IN.
SLEEVE & HOUSING	.001-.005 IN.

SPRING PRELOAD 5 LBS.

FIGURE 6.28 ΔP REGULATING VALVE MATERIALS

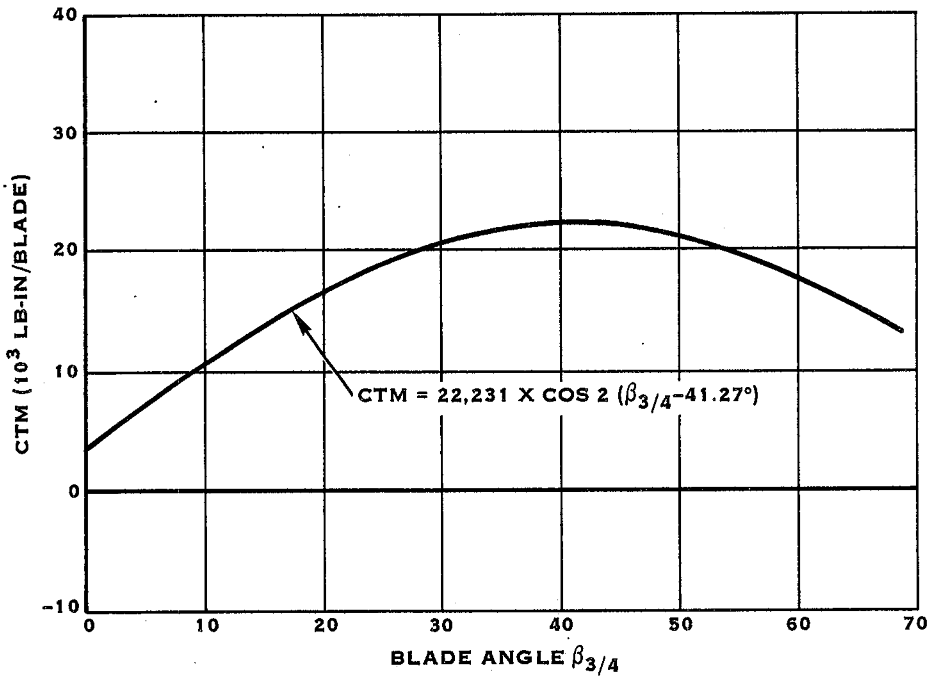


FIGURE 6-29. CENTRIFUGAL TWISTING MOMENT (CTM)-100% SPEED

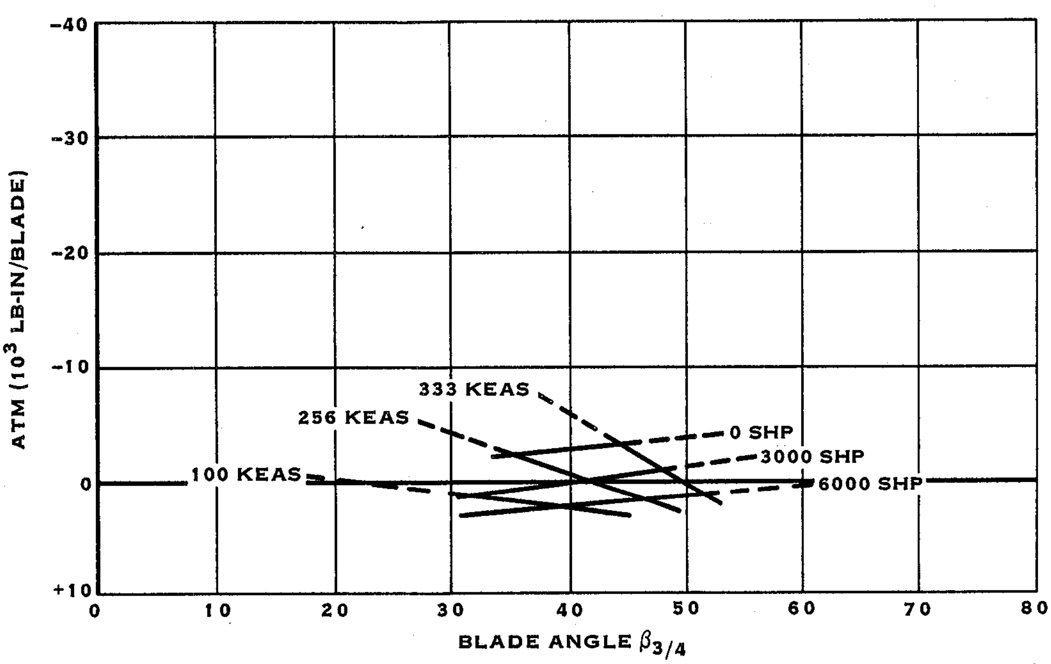


FIGURE 6-30. AERODYNAMIC TWISTING MOMENT (ATM) - 100% SPEED (5,000 FT)
(+ TOWARD HIGH PITCH)

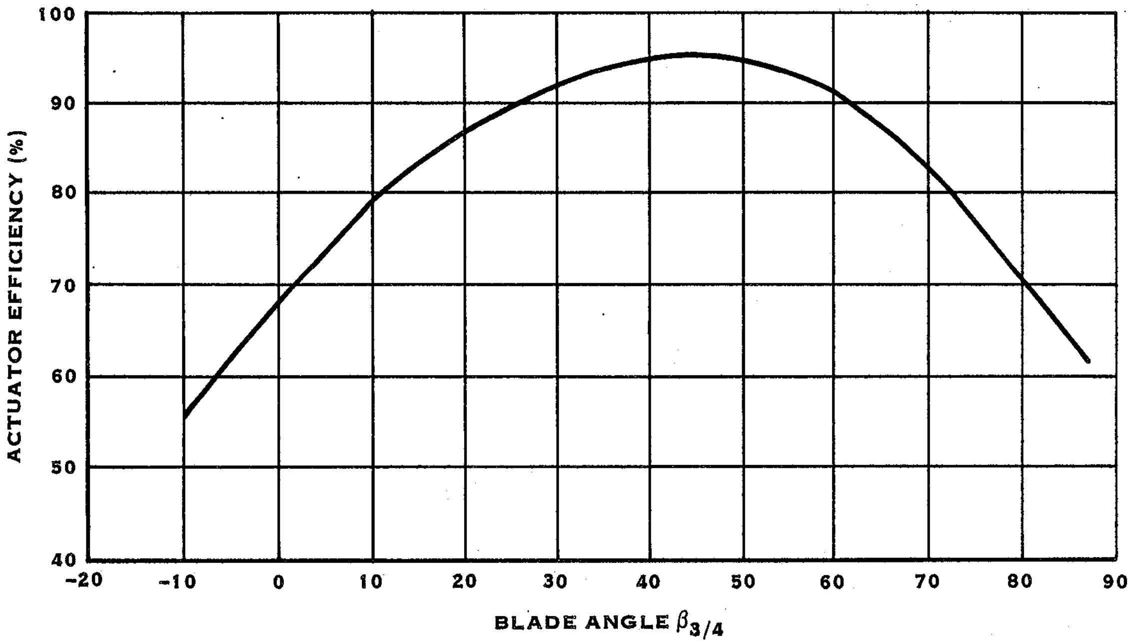


FIGURE 6-31. ACTUATOR EFFICIENCY (TOWARD HIGH PITCH)

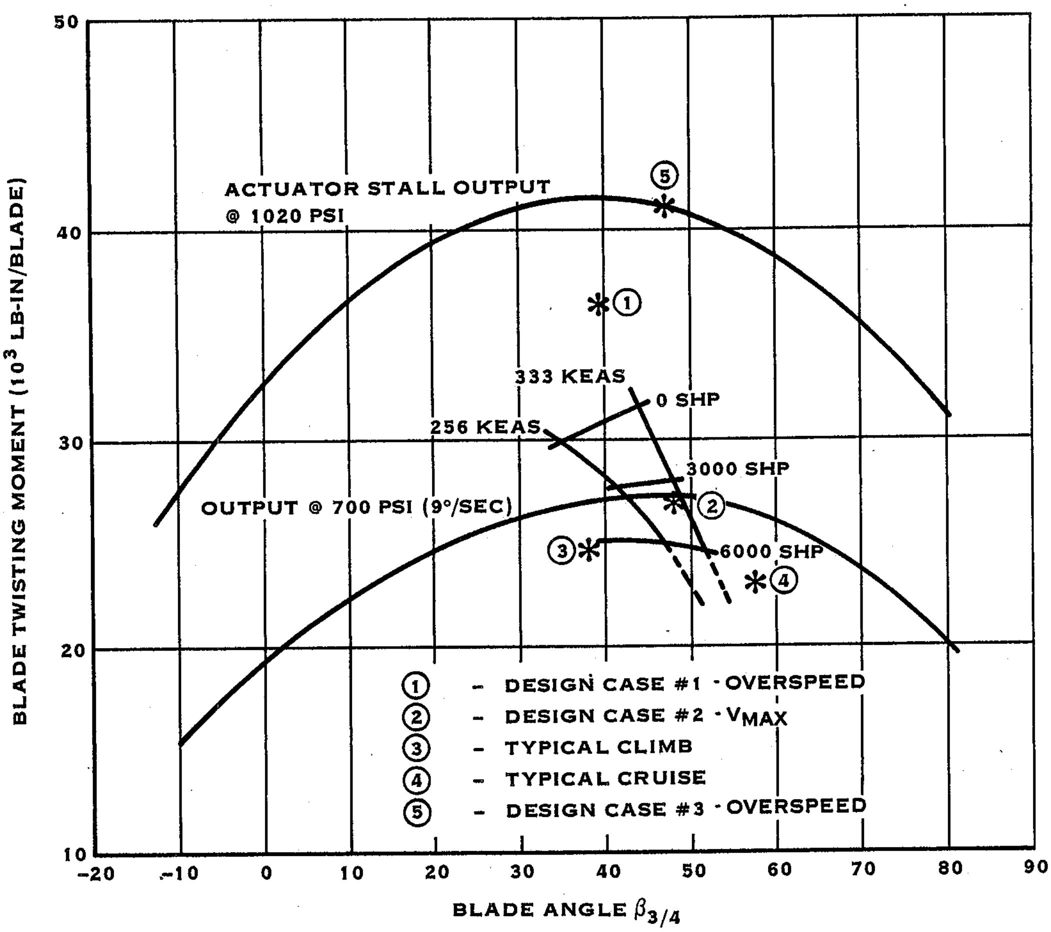


FIGURE 6-32. ACTUATOR CAPACITY

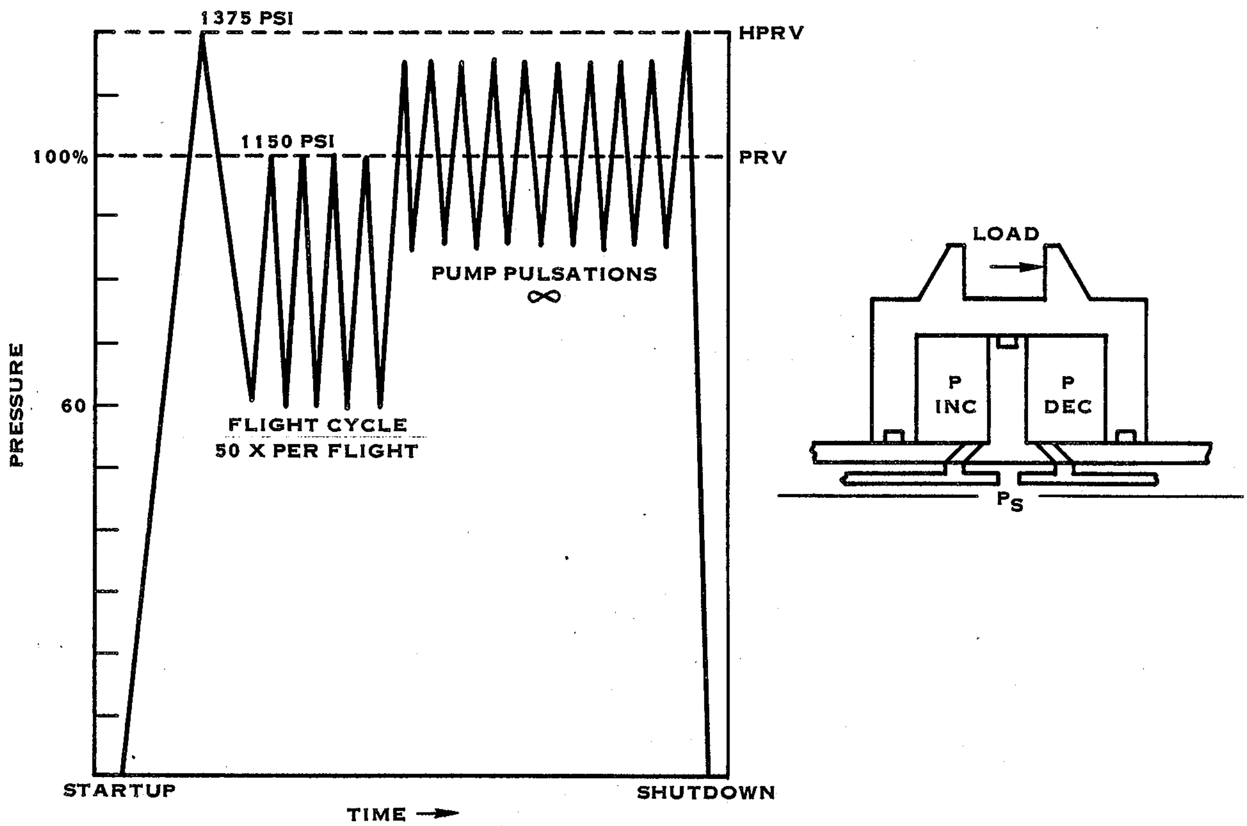
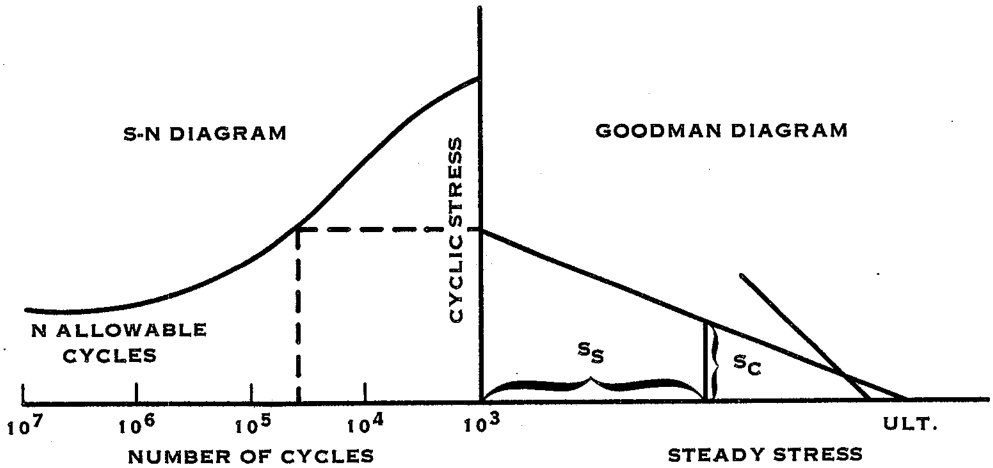


FIGURE 6-33. ACTUATOR PRESSURE SPECTRUM FOR A TYPICAL FLIGHT



$$S_S = P_S \times S/P$$
$$S_C = P_C \times S/P \times K_F$$

P_S = STEADY PRESSURE FOR LOADING CONDITION
P_C = CYCLIC PRESSURE FOR LOADING CONDITION

FIGURE 6-34. CALCULATION OF ALLOWABLE CYCLES

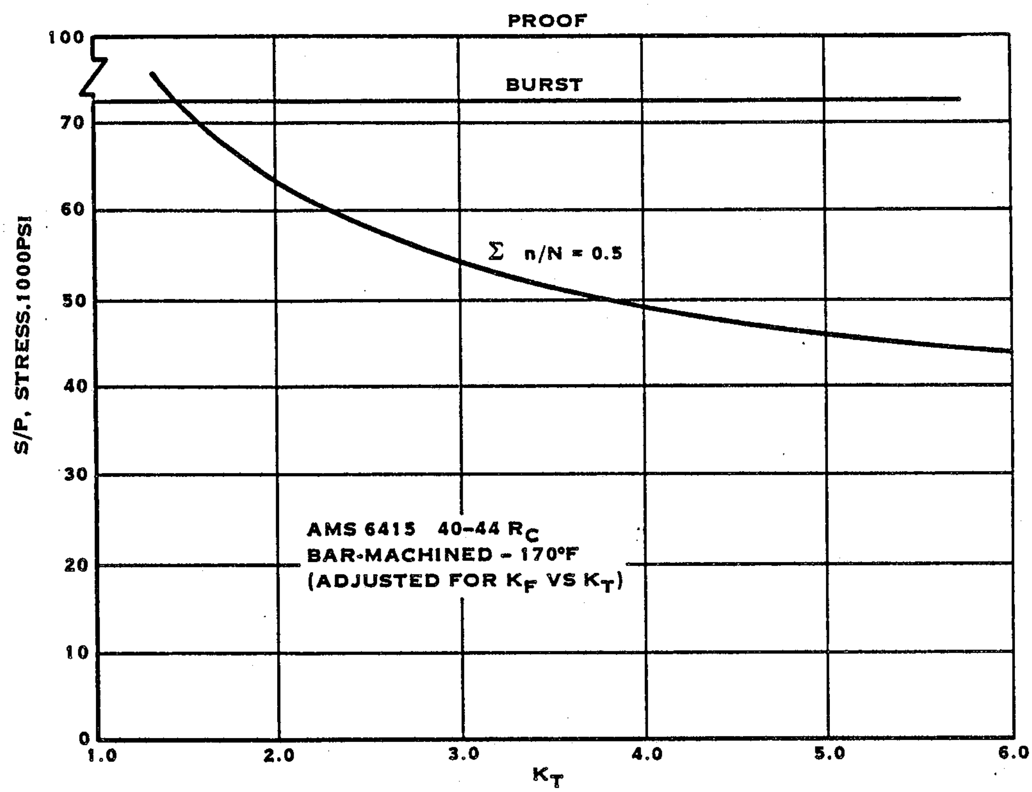


FIGURE 6.35 STRESS ALLOWABLES AS A FUNCTION OF STRESS CONCENTRATION FACTOR (KT)

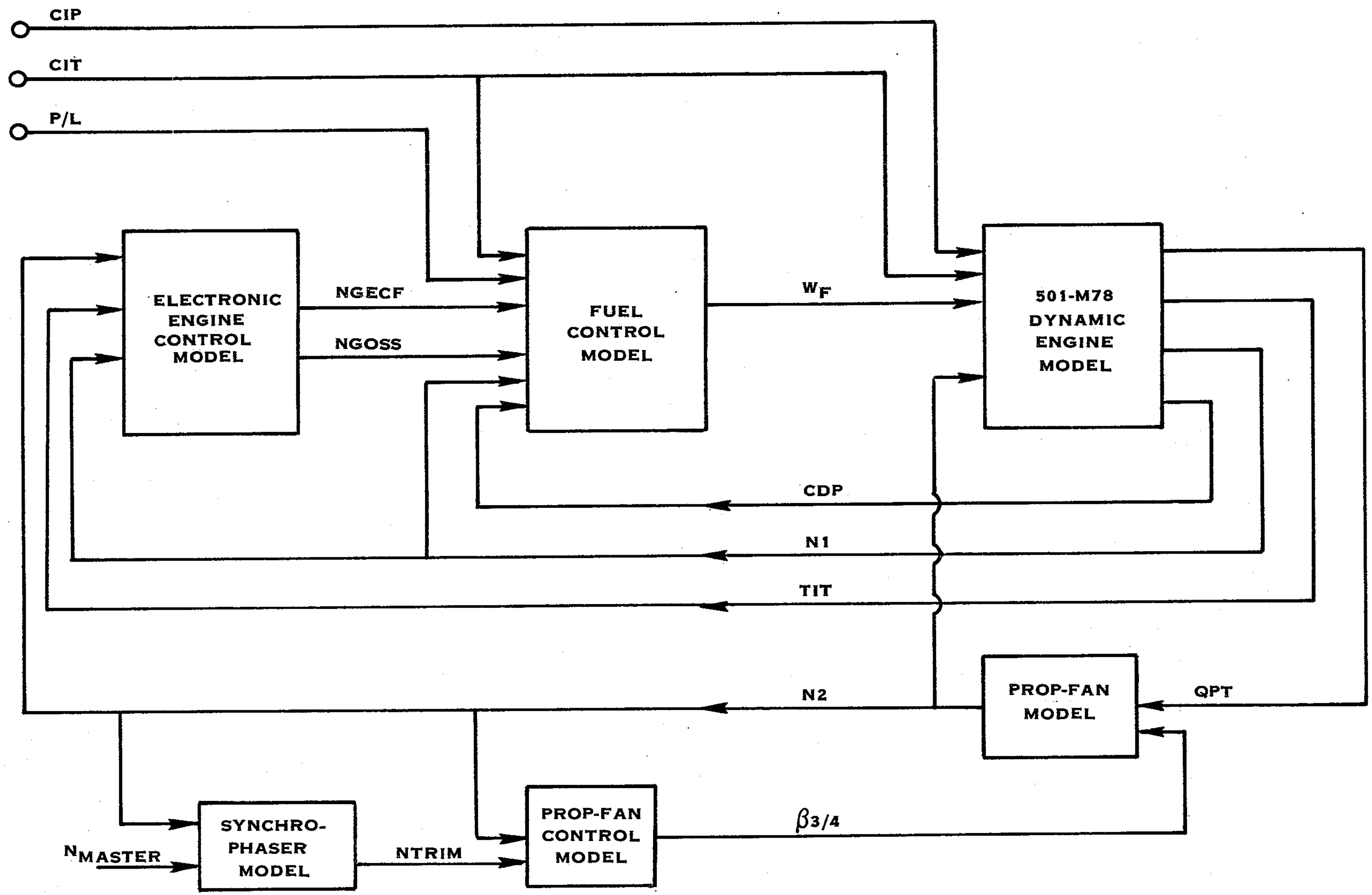


FIGURE 7.1. LAP SYSTEM FUNCTIONAL BLOCK DIAGRAM

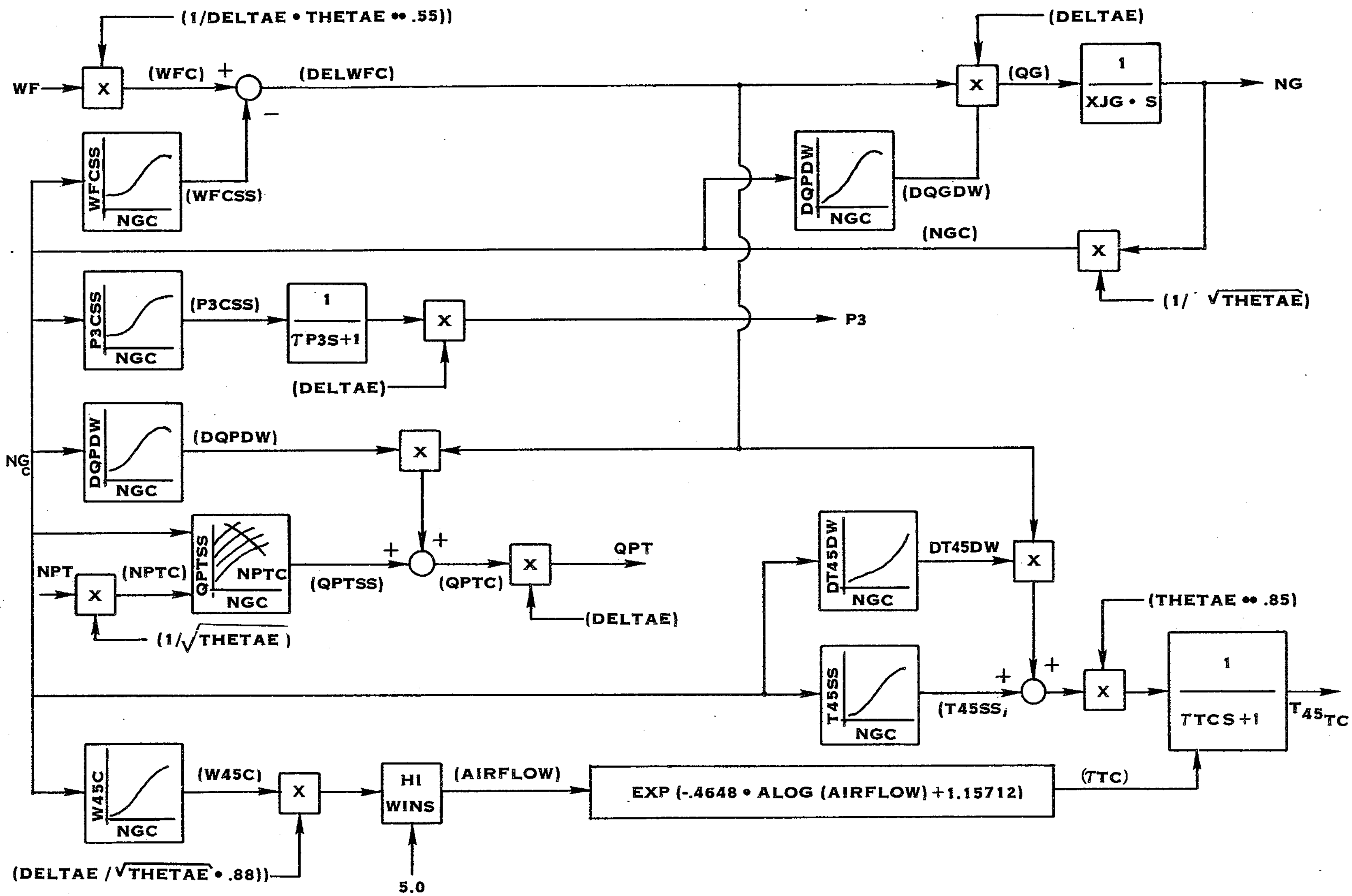


FIGURE 7.2. DYNAMIC ENGINE MODEL OF ALLISON GAS TURBINE 501-M78

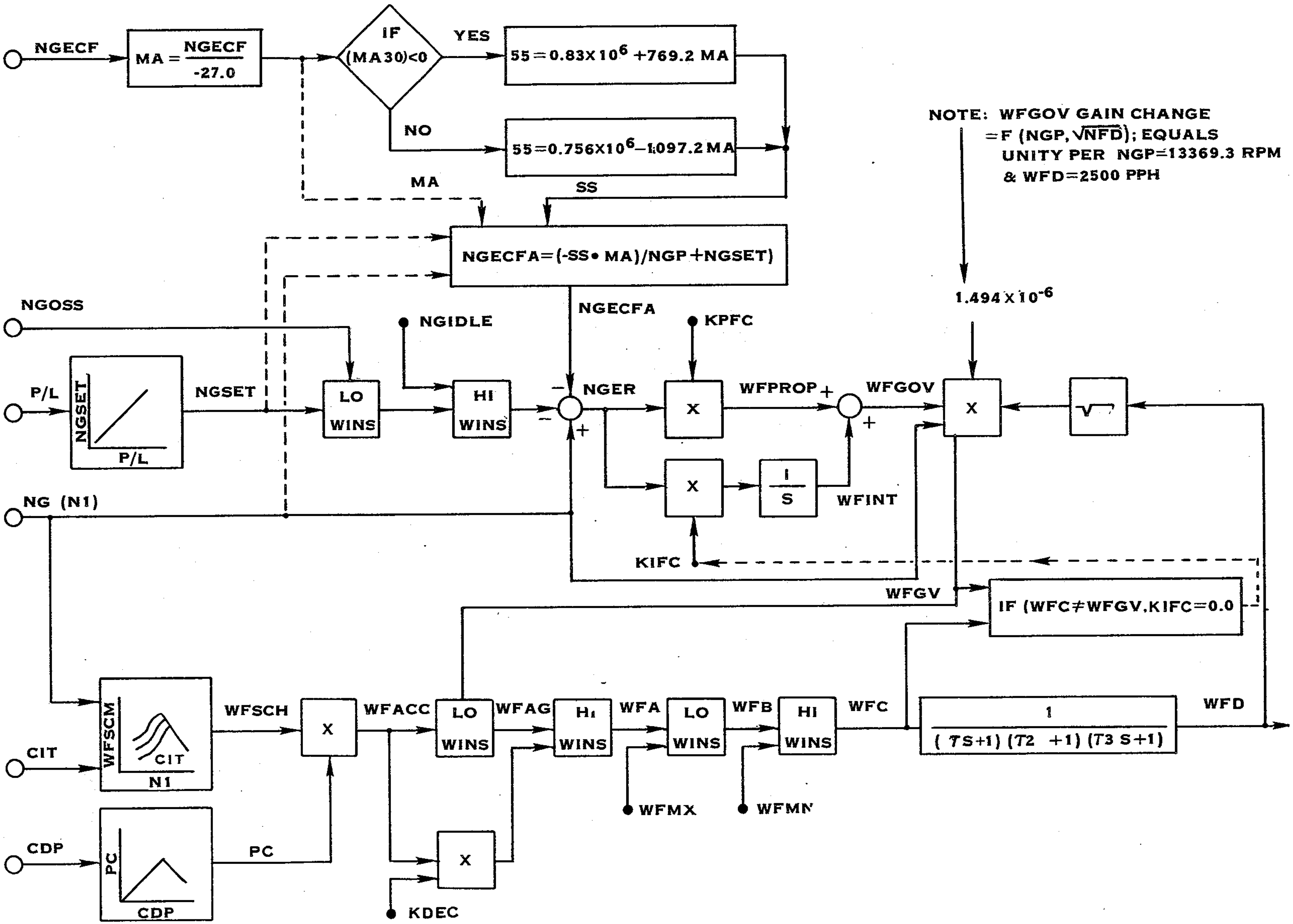


FIGURE 7.3. 501-M78 GAS GENERATOR FUEL CONTROL

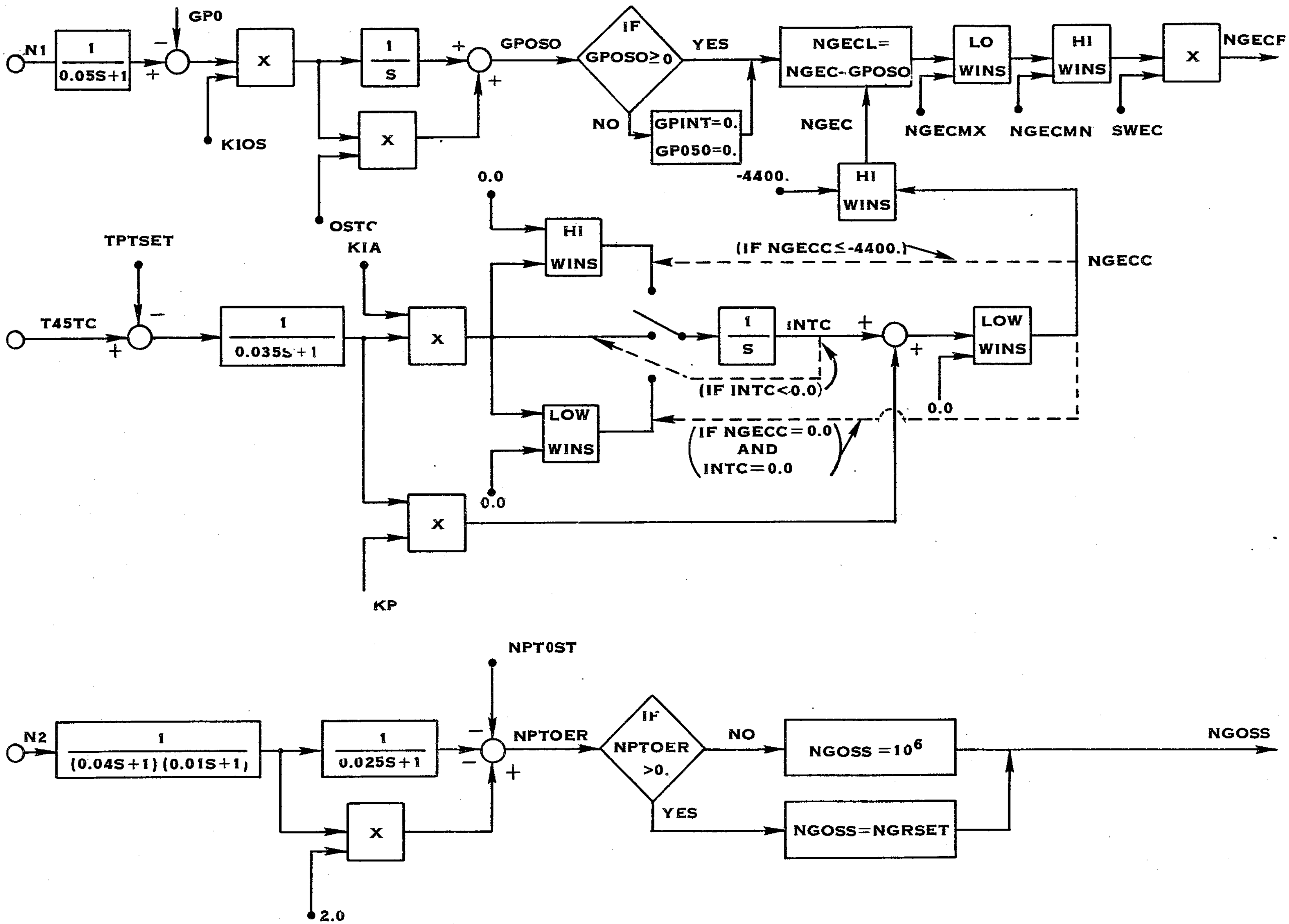
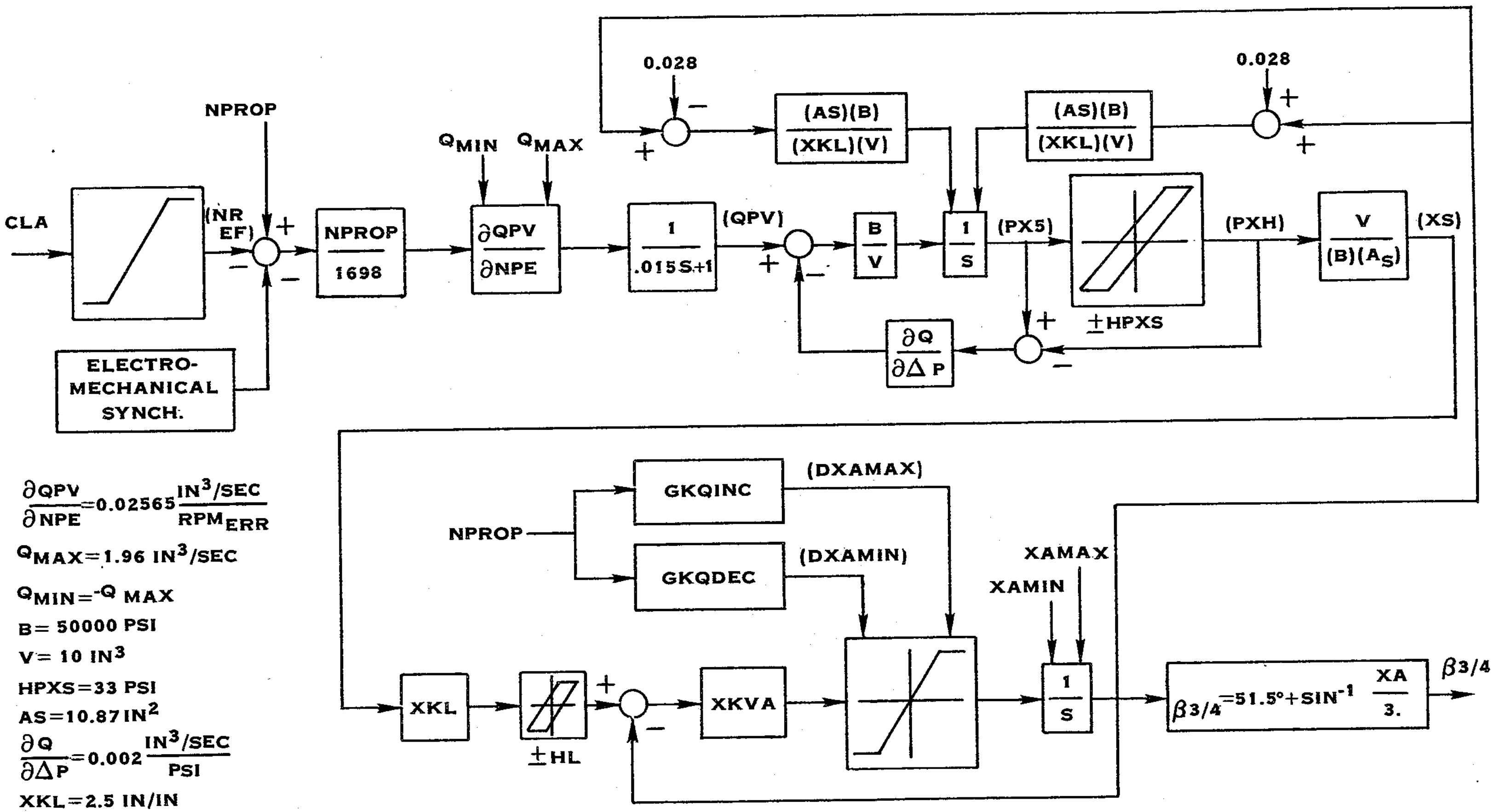


FIGURE 7.4. 501-M78 ELECTRONIC CONTROL BLOCK DIAGRAM (N1 OVERSPEED N2 OVERSPEED, AND TIT LIMITING FUNCTIONS)



$\dfrac{\partial QPV}{\partial NPE} = 0.02565 \dfrac{IN^3/SEC}{RPM_{ERR}}$

$Q_{MAX} = 1.96\ IN^3/SEC$

$Q_{MIN} = -Q\ MAX$

$B = 50000$ PSI

$V = 10\ IN^3$

HPXS = 33 PSI

$AS = 10.87\ IN^2$

$\dfrac{\partial Q}{\partial \Delta P} = 0.002 \dfrac{IN^3/SEC}{PSI}$

XKL = 2.5 IN/IN

HL = 0.003 IN

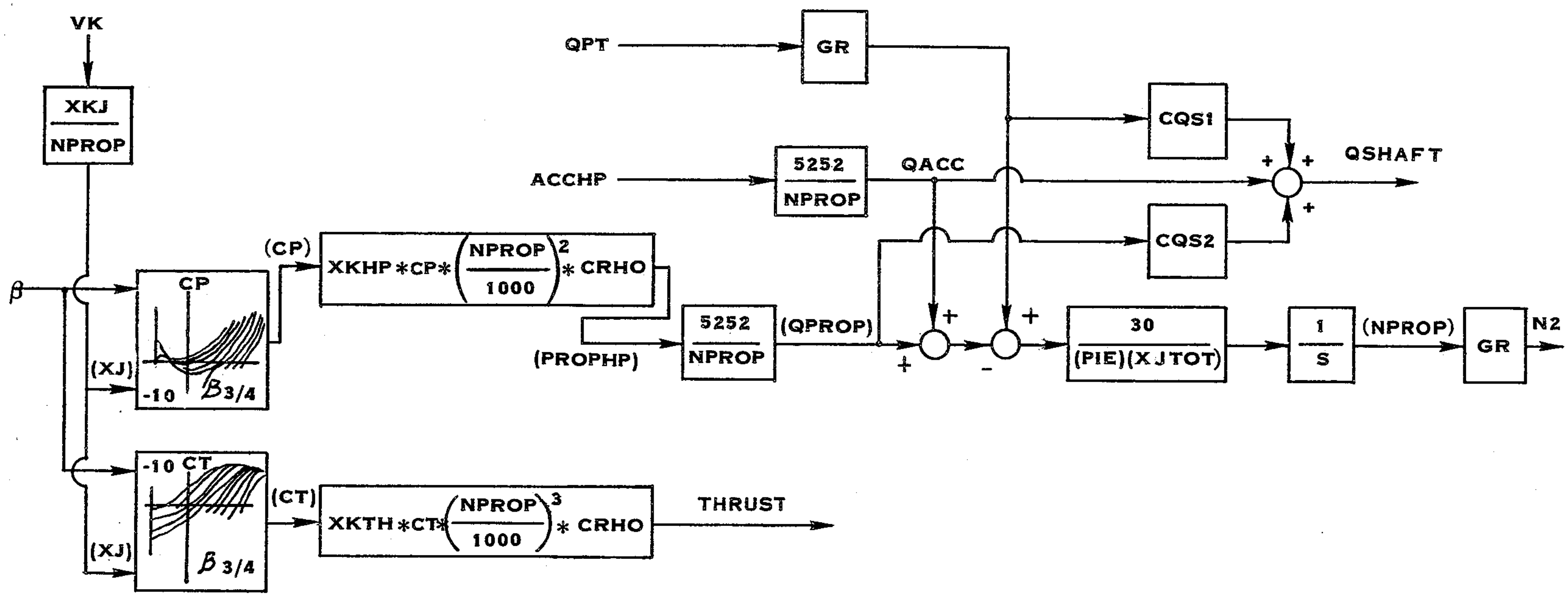
XKVA = 40

XAMAX = 1.87 IN

XAMIN = -2.7 IN

$GKQINC = -\,GKQDEC = 0.000192 \dfrac{IN/SEC}{RPM}$

FIGURE 7.5. "LAP" PROP-FAN CONTROL BLOCK DIAGRAM



XKJ = 11.26

GR = 6.77

XJTOT = 89.64

XKHP = 1181.6

XKTH = 4334.2

CQS1 = 0.5984

CQS2 = 0.4016

CRHO = (PAMB/14.7)+ ∗ (TAMB/518.7)

FIGURE 7.6. "LAP" PROP-FAN MODEL BLOCK DIAGRAM

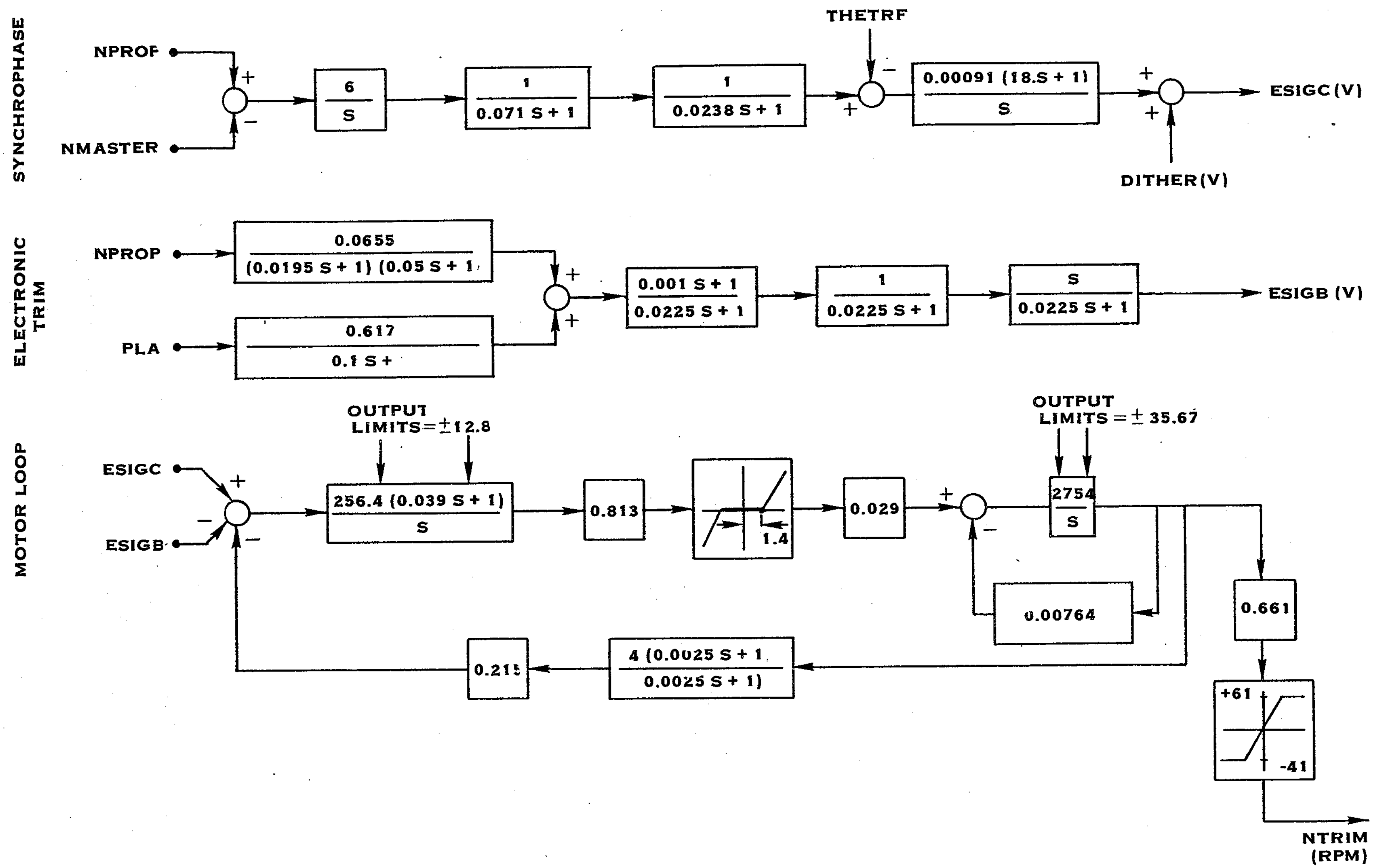


FIGURE 7.7. 54H60 (C-130) SYNCHROPHASER BLOCK DIAGRAM

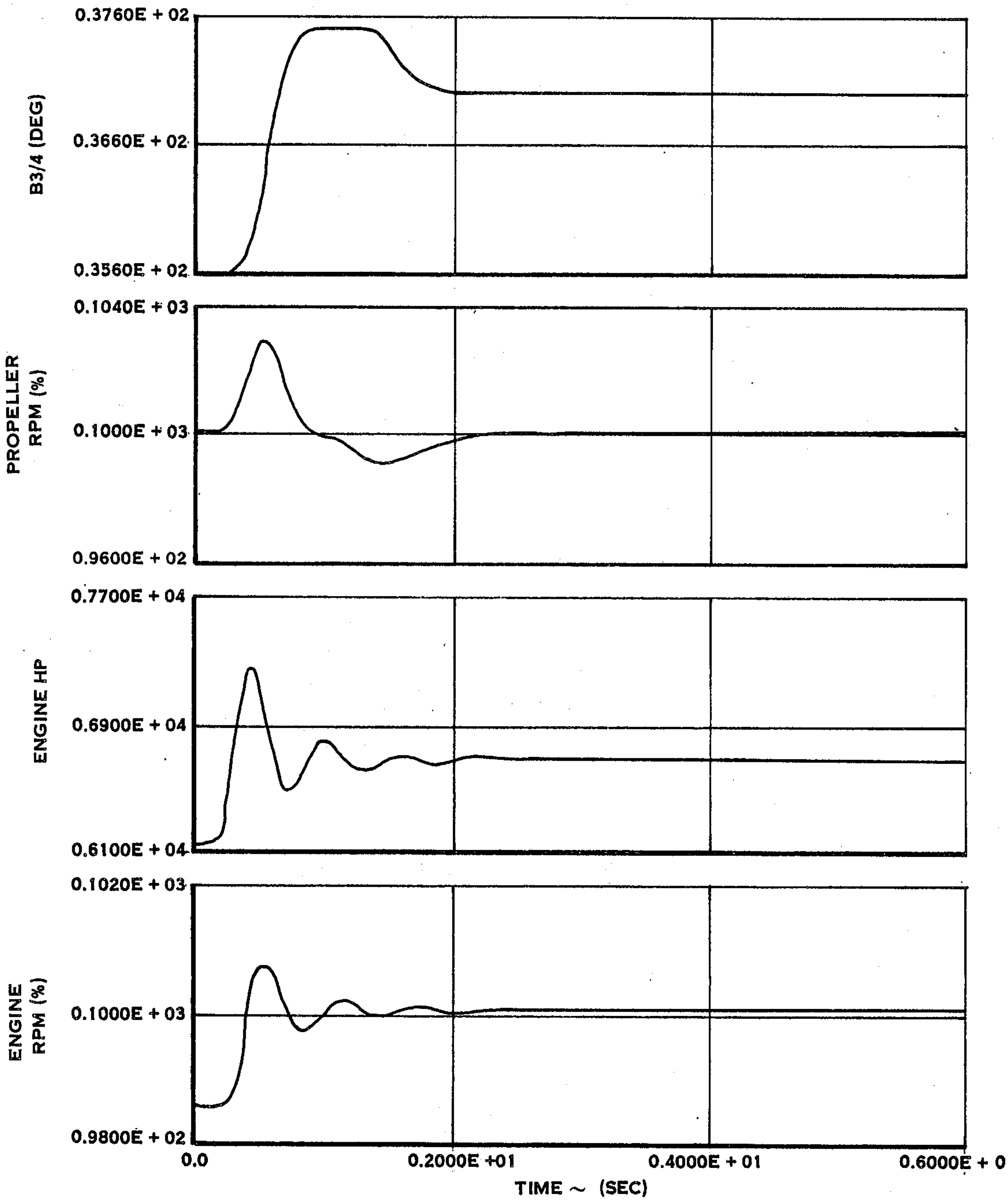


FIGURE 7.8. SIMULATION RESPONSE TO A 1.5% NG REF STEP INCREASE
(S.L. STATIC, 100 % NP)

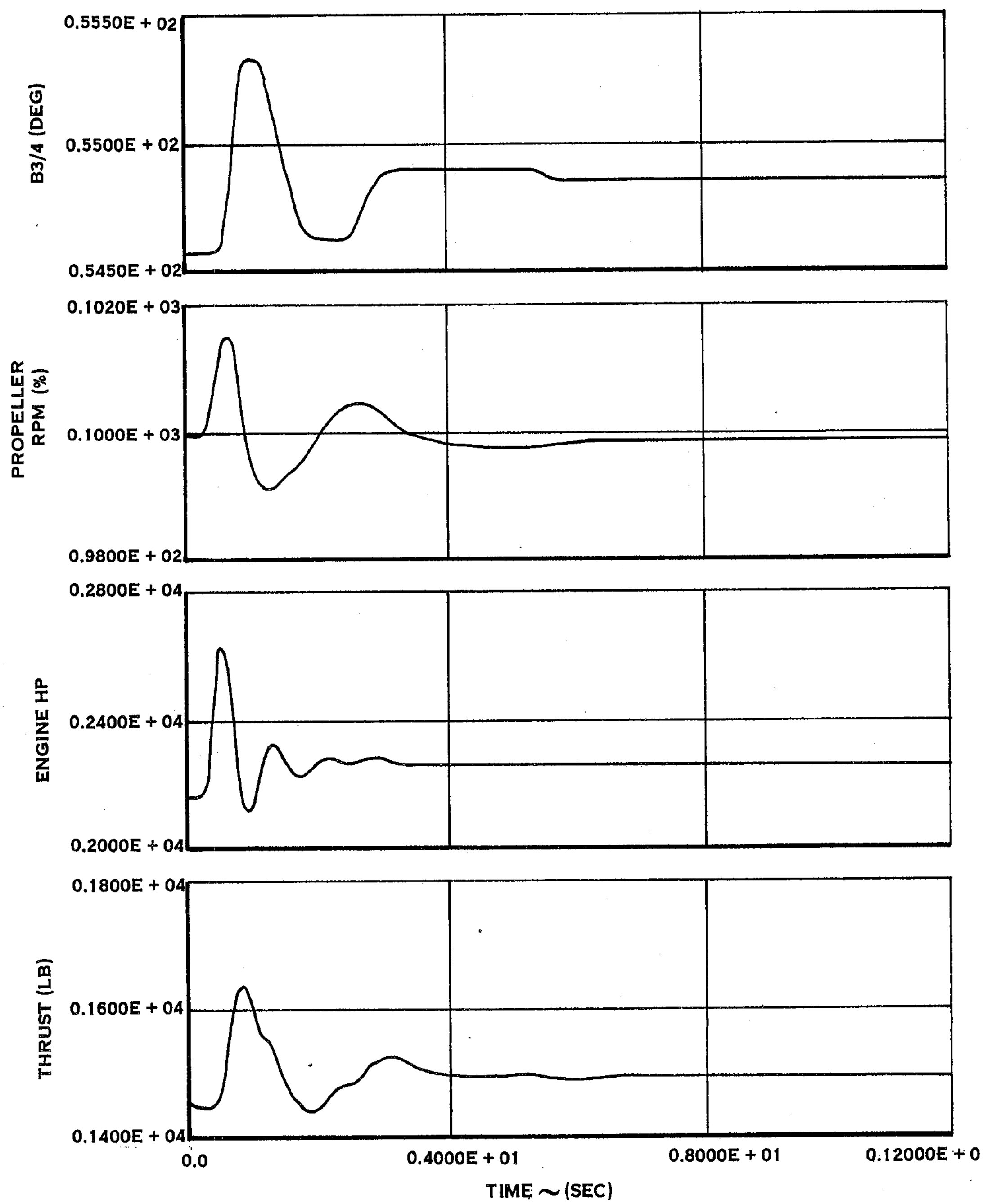


FIGURE 7.9. SIMULATION RESPONSE TO A 1.5% NG REF STEP INCREASE (35,000 FT, 390 KTS, 100% NP)

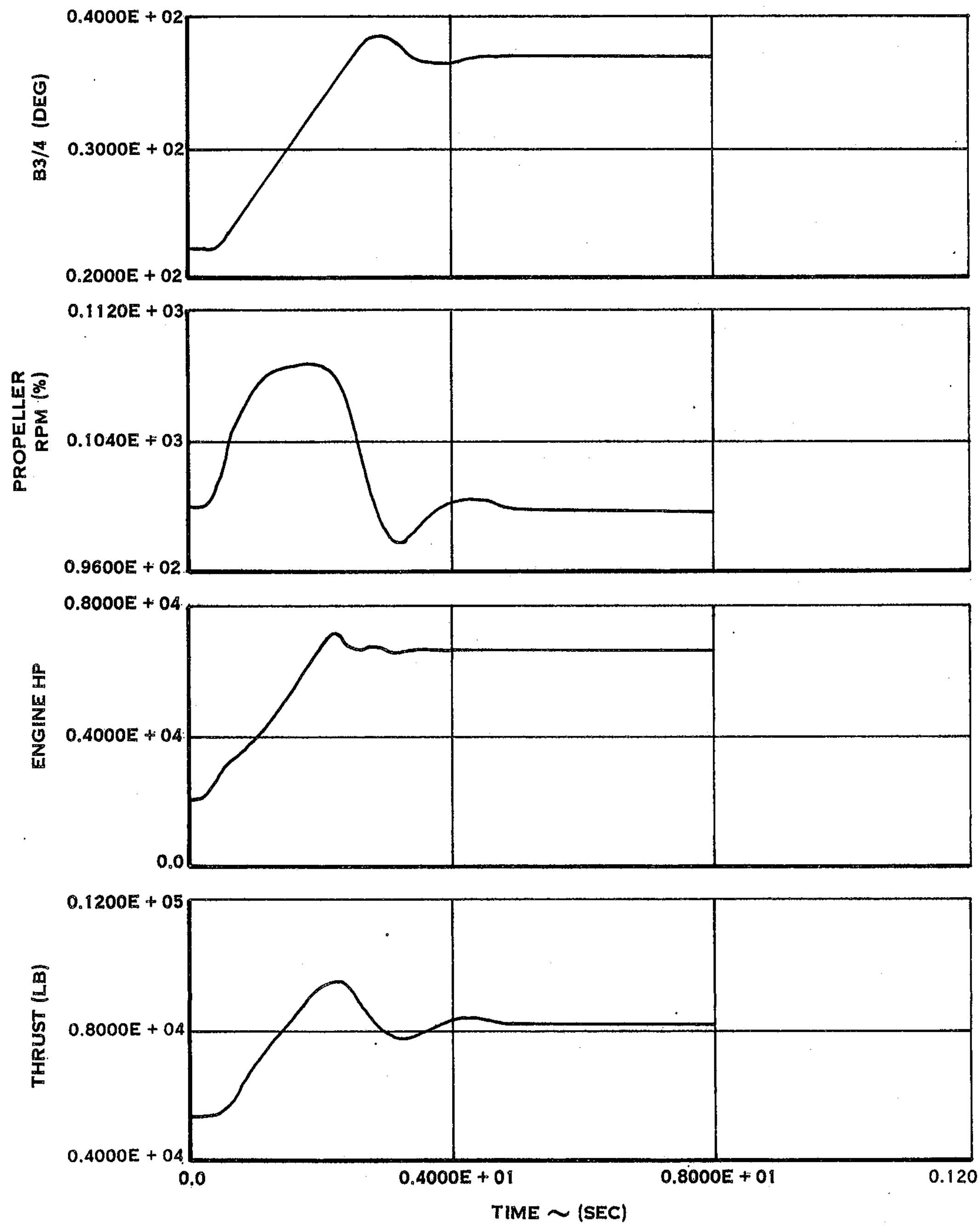


FIGURE 7.10. SIMULATION RESPONSE TO A 2.0 SEC RAMP OF NG REF (S.L., STATIC, 100% NP, 88% -100% NG)

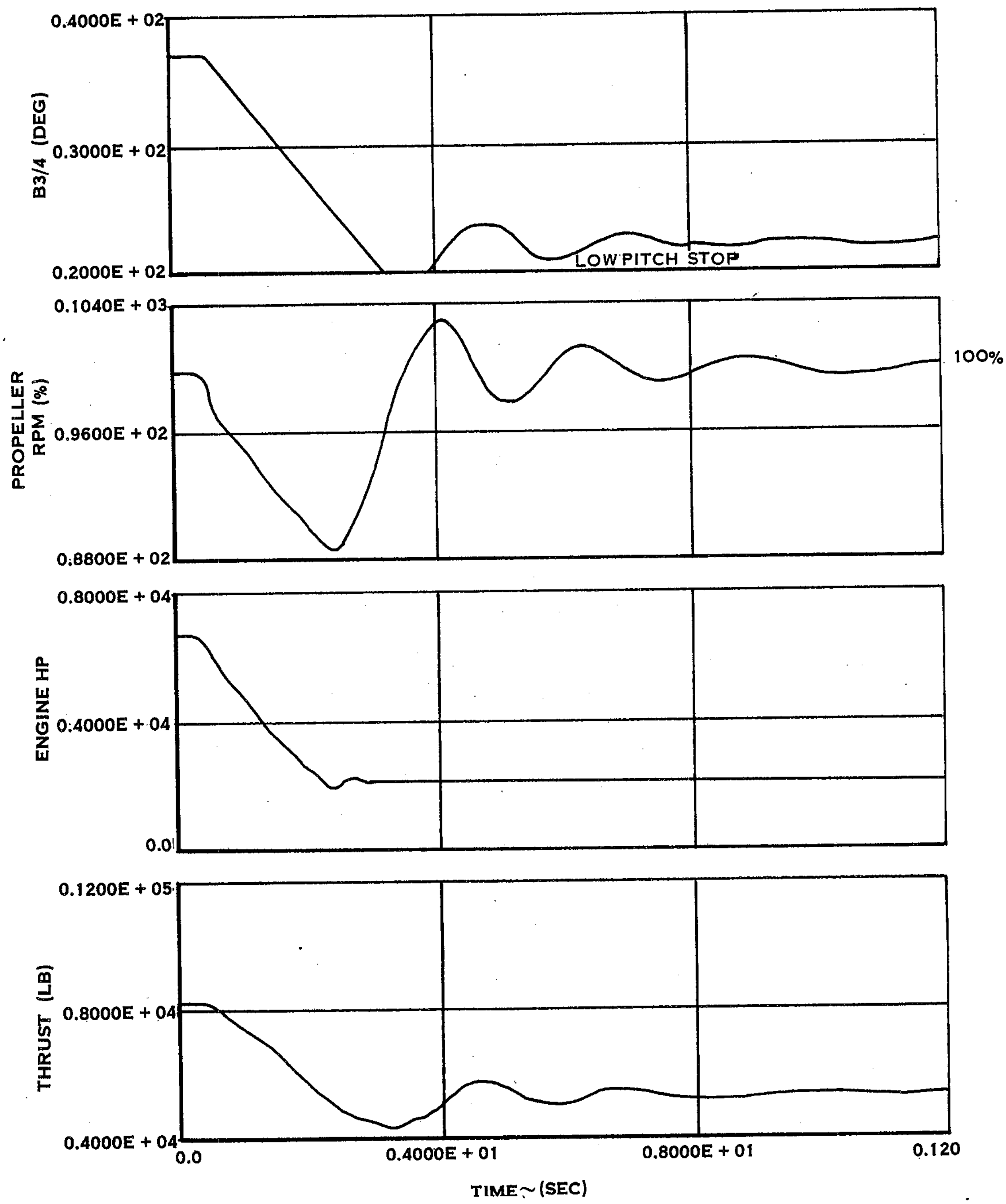


FIGURE 7.11. SIMULATION RESPONSE TO A 2.0 SEC RAMP OF NG REF (S.L. STATIC, 100% NP, 100%-88% NG)

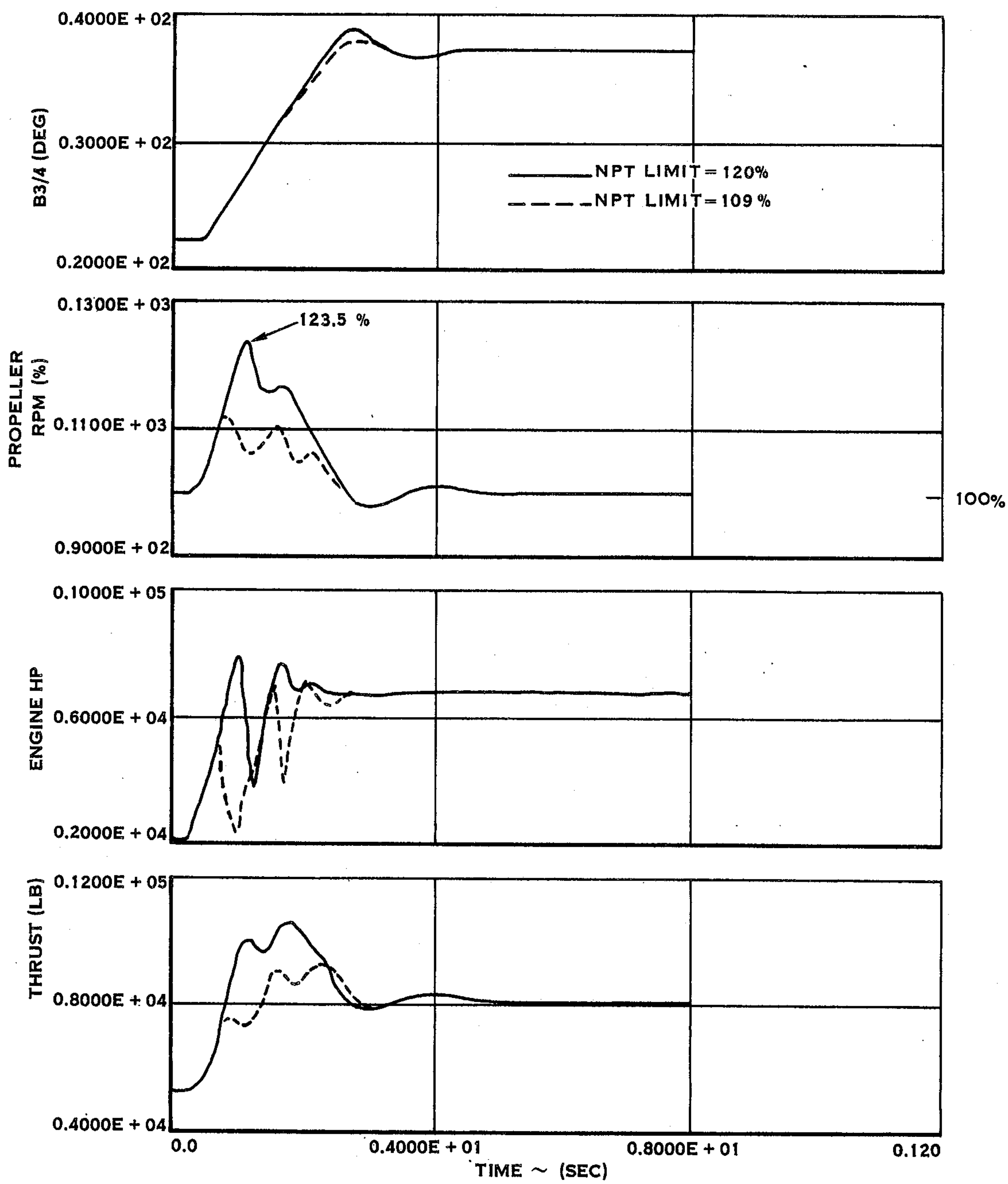


FIGURE 7.12. SIMULATION RESPONSE TO A 0.5 SEC RAMP OF NG REF (S.L. STATIC, 100% NP, 88-100% NG)

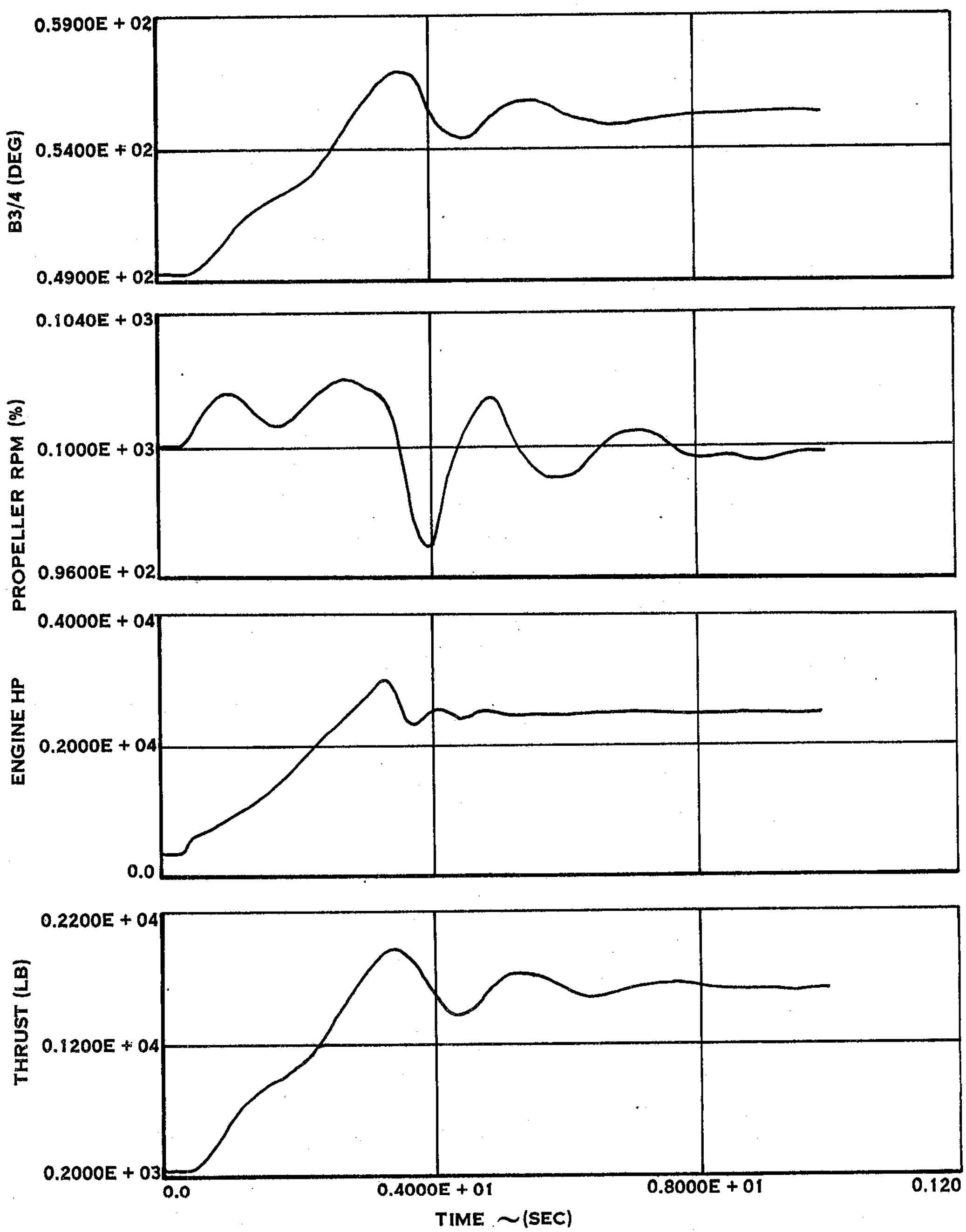


FIGURE 7.13. SIMULATON RESPONSE TO A 2.0 SEC RAMP OF NG REF
(35,000 FT, 390 KTS, 100% NP, 78%-100% NG)

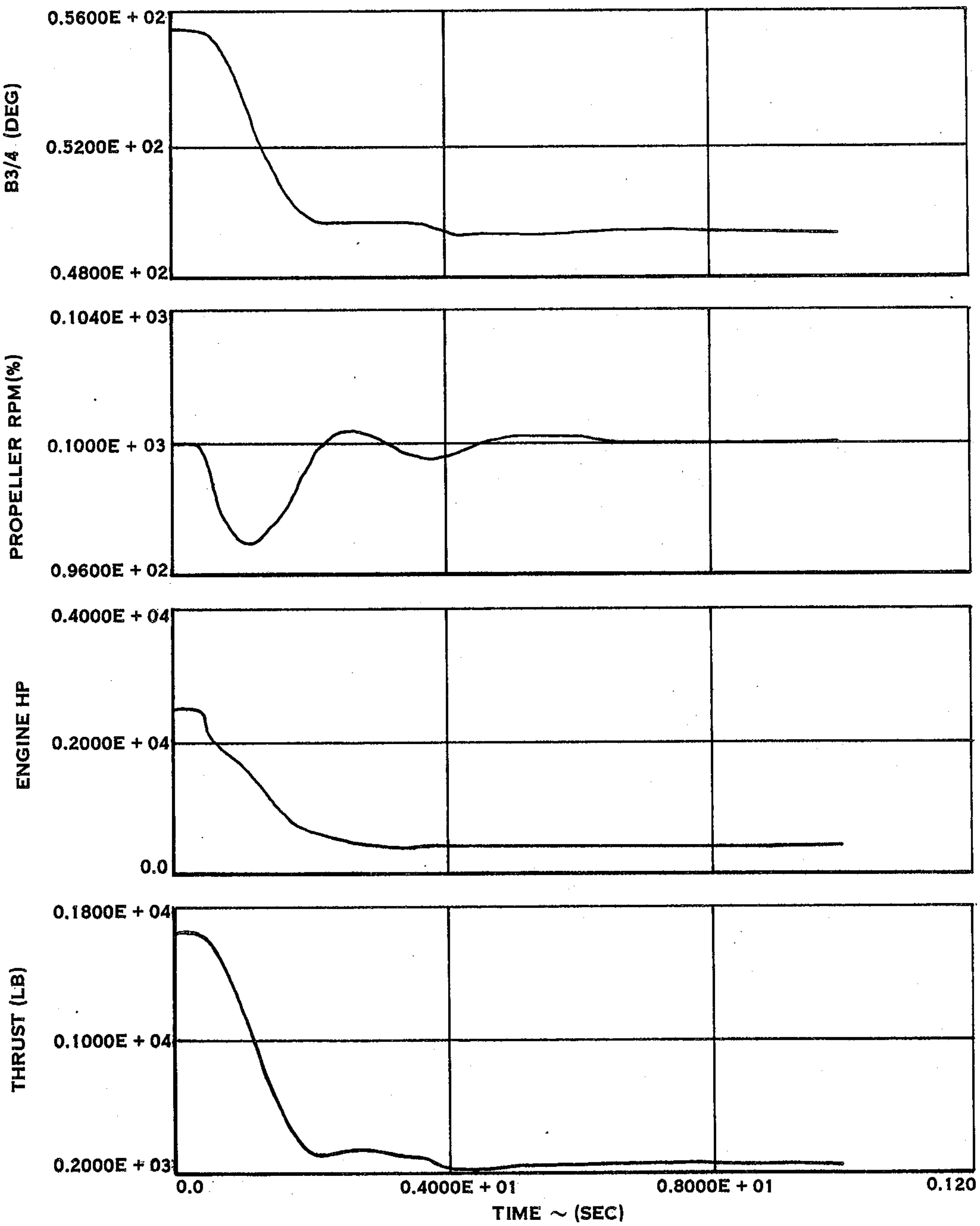


FIGURE 7.14. SIMULATION RESPONSE TO A 2.0 SEC RAMP OF NG REF
(35,000 FT, 390 KTS, 100% NP, 100%-78% NG)

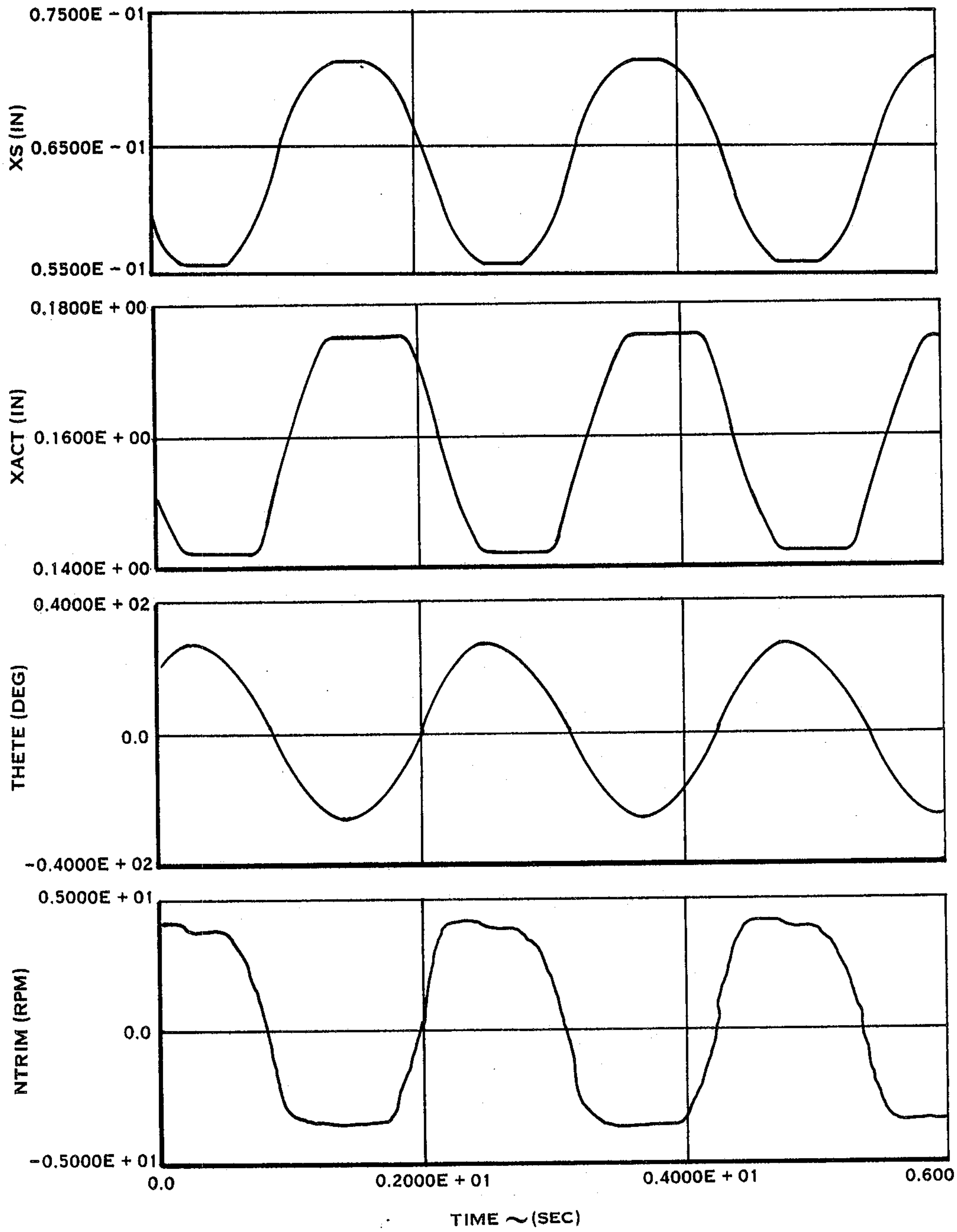


FIGURE 7.15. 54460 SYNCHROPHASER LIMIT CYCLE IN /8 X GAIN
(35,000FT, 390KTS, 100% NP, 100% NG)

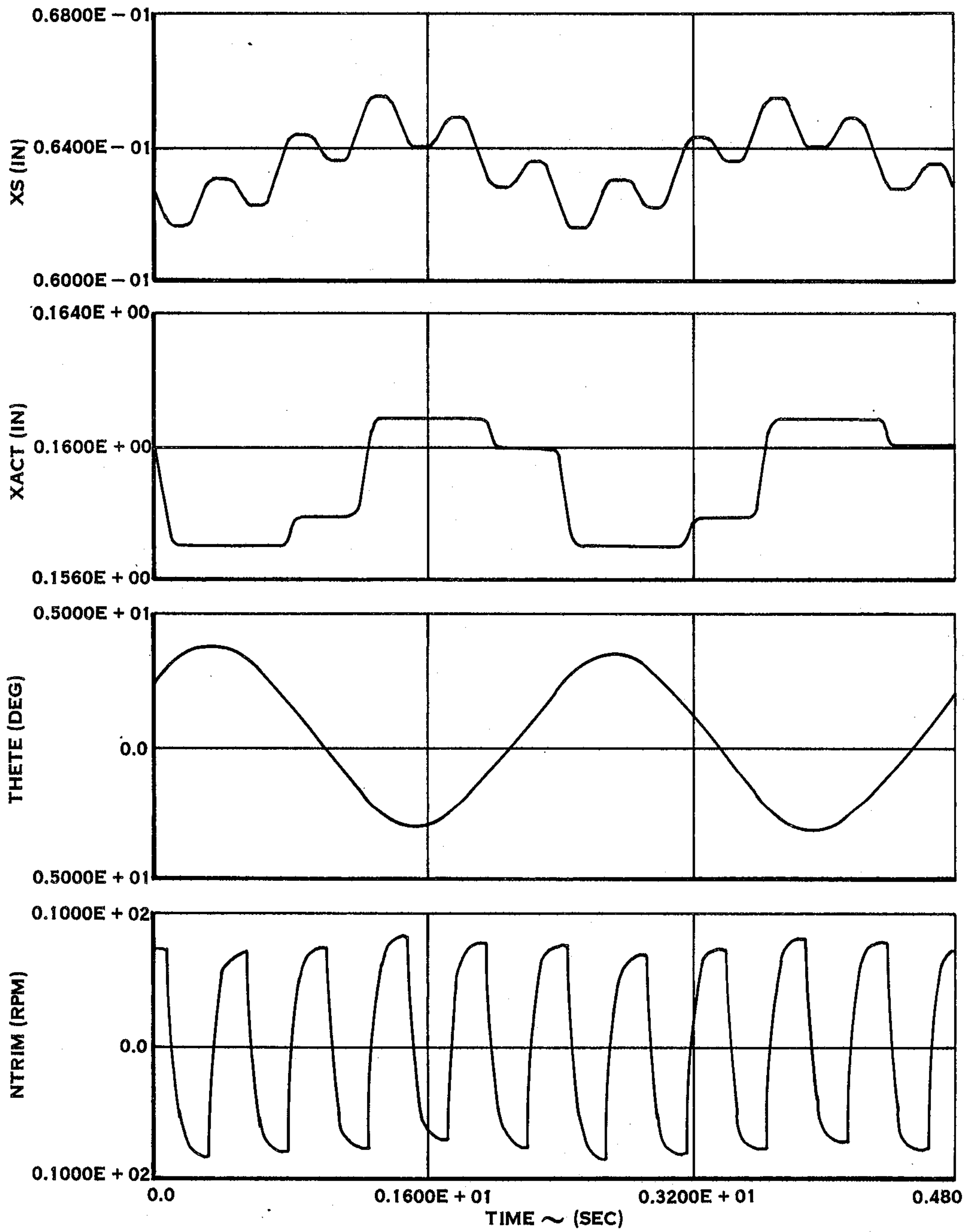


FIGURE 7.16 54460 SYNCROPHASER PERFORMANCE 2.0 HZ, + 2.5 V DITHER (35,000 FT, 390 KTS, 100%NP, 100% NG)

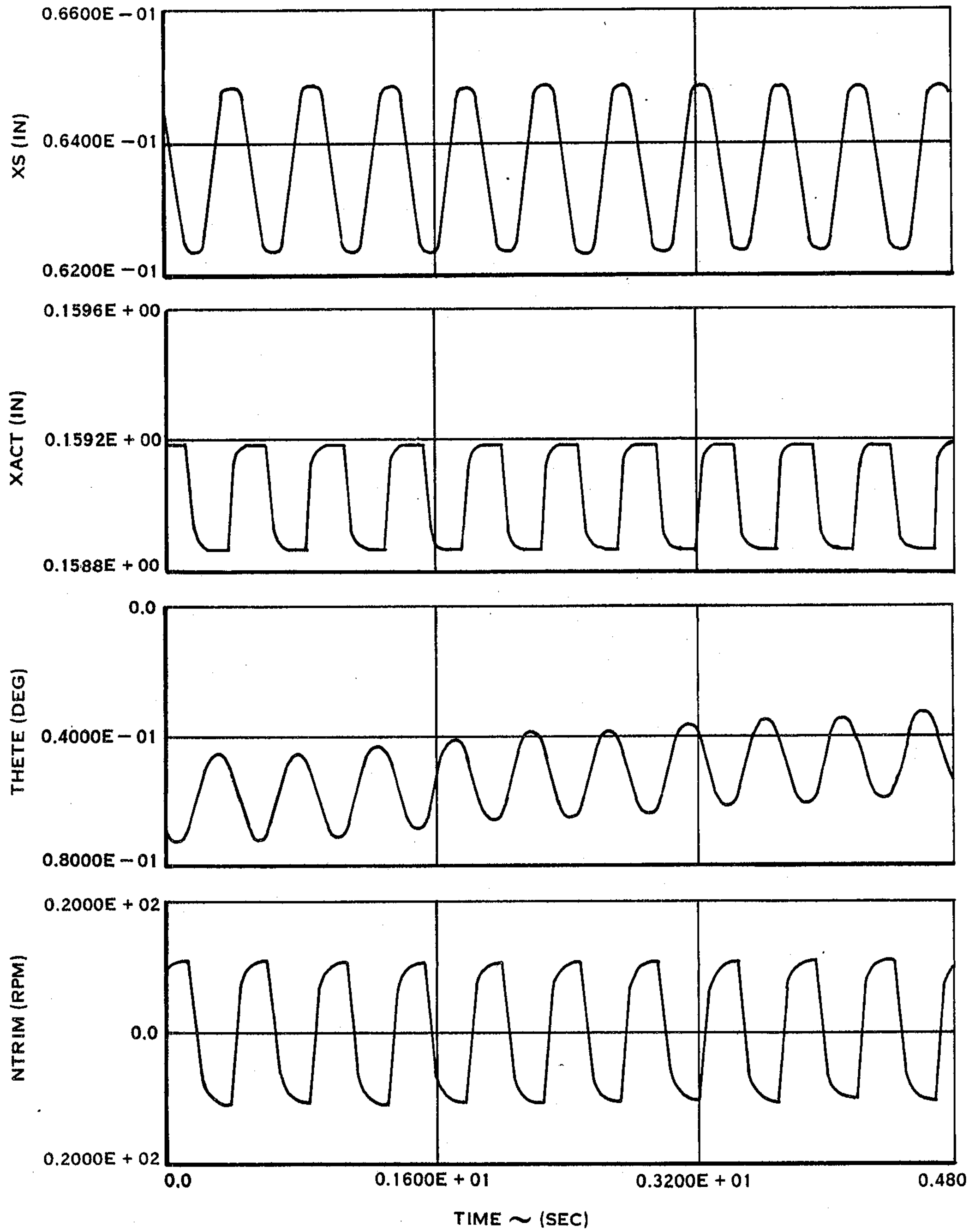


FIGURE 7.17. 54460 SYNCROPHASER PERFORMANCE WITH 2.0 HZ, + 3.5 V DITHER (35,000 FT, 390 KTS, 100%NP, 100% NG)

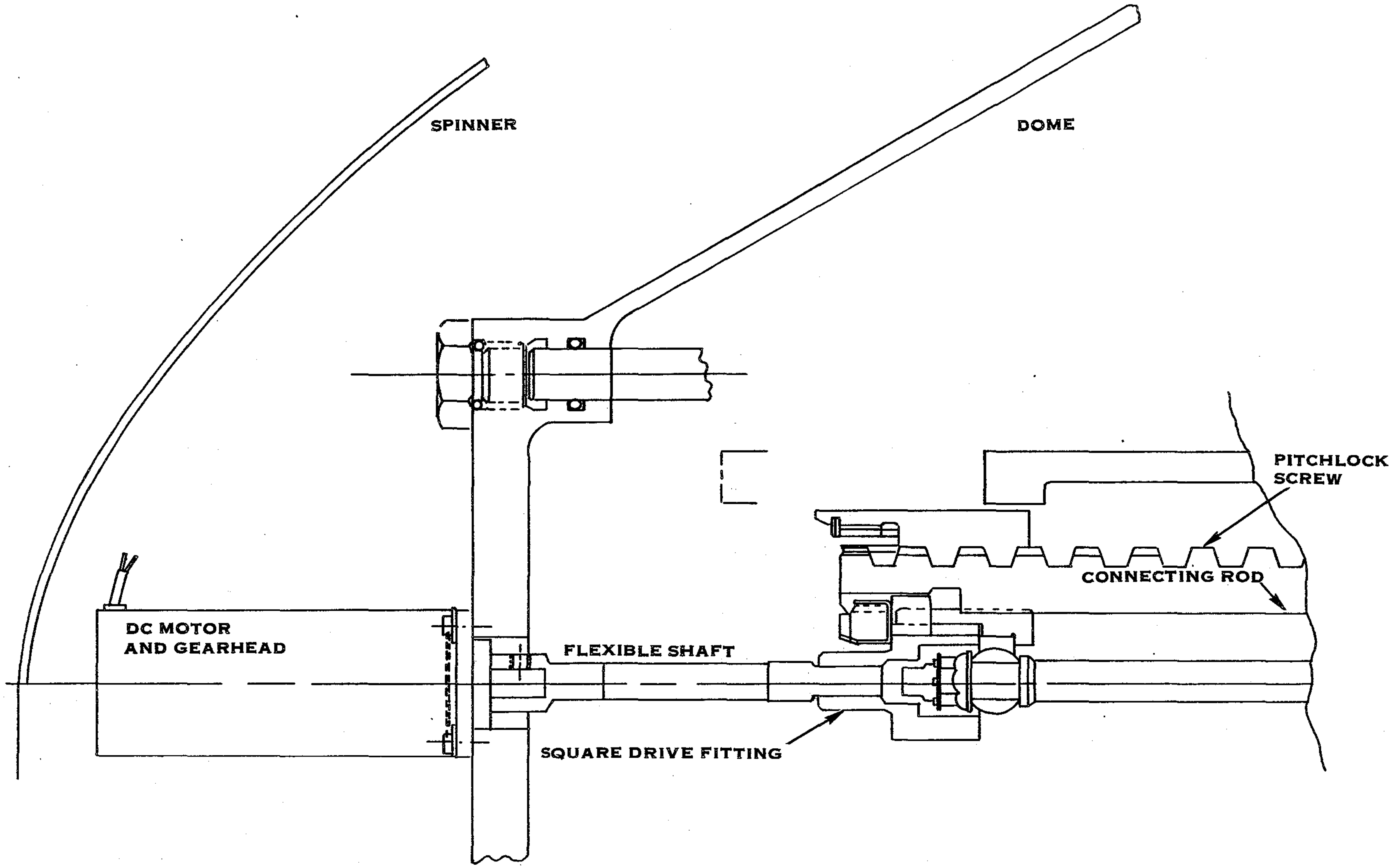


FIGURE 8.1 MOTOR DRIVEN PITCH CONTROL MECHANISM

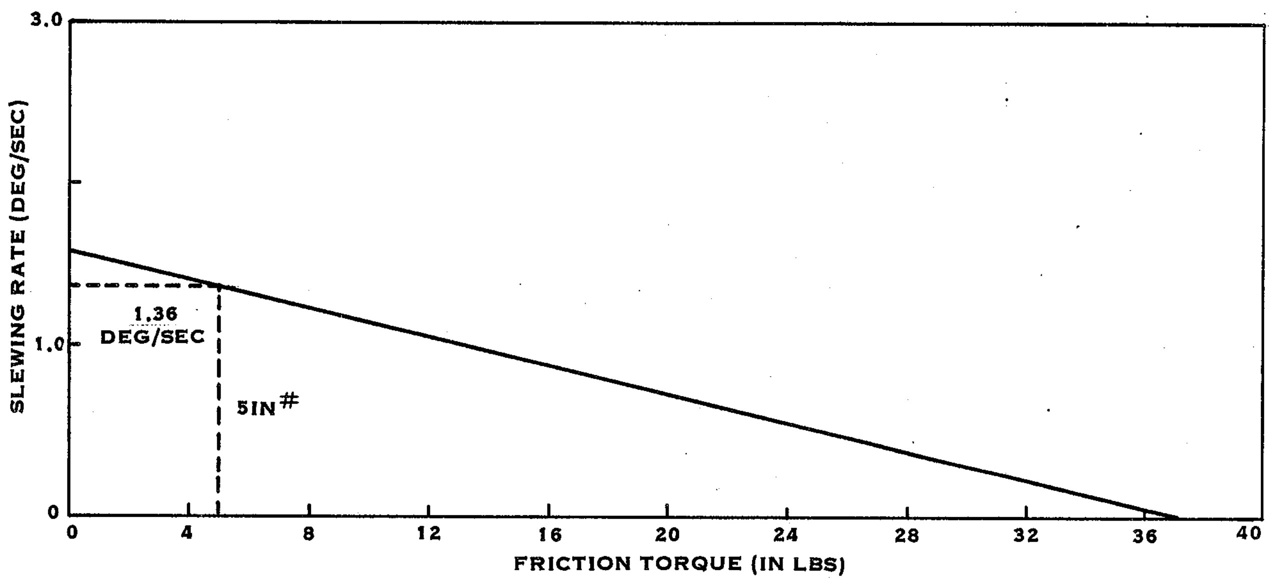
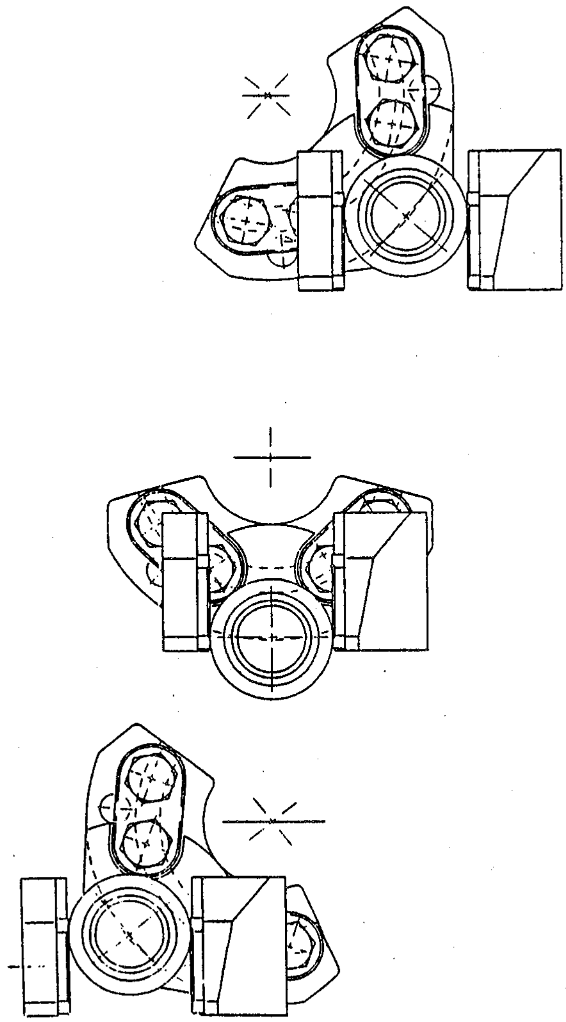


FIGURE 8.2 BLADE ANGLE SLEWING CAPACITY IN BETA MODE VS PITCHLOCK SCREW FRICTION



REVERSE (−6 DEG @ β¾)

ACTUATOR MIDSTROKE

FEATHER (90 DEG @ β¾)

FIGURE 9.1 ACTUATOR/BEARING RELATIONSHIP

CTM: 22,130 IN-LB X 1.2 (ALLOWANCE)

ATM: 3,052 IN-LB X 1.2 (ALLOWANCE)

FTM: 1,256 IN-LB

TTM: 31,474 IN-LB

TRUNNION LOAD @ 3.00 IN PIN RADIUS:

*CYCLIC STRESS OF ATM ONLY

$$\frac{31,474}{3.00} = 10,500 \text{ LB} \pm 1,220 \text{ LB}$$

HCF = 10,500 ± 1,220 LB VIBRATORY
LCF = 5,250 ± 5,250 LB START-STOP

FIGURE 9.2 TRUNNION LOADS @ 38.5 DEG $\beta 3/4$ (100%, SEA LEVEL)
REF. "SYSTEM DESIGN AND INTEGRATION OF THE LARGE-SCALE ADVANCED PROP-FAN" SECTION 3.3.1. PG. 12.

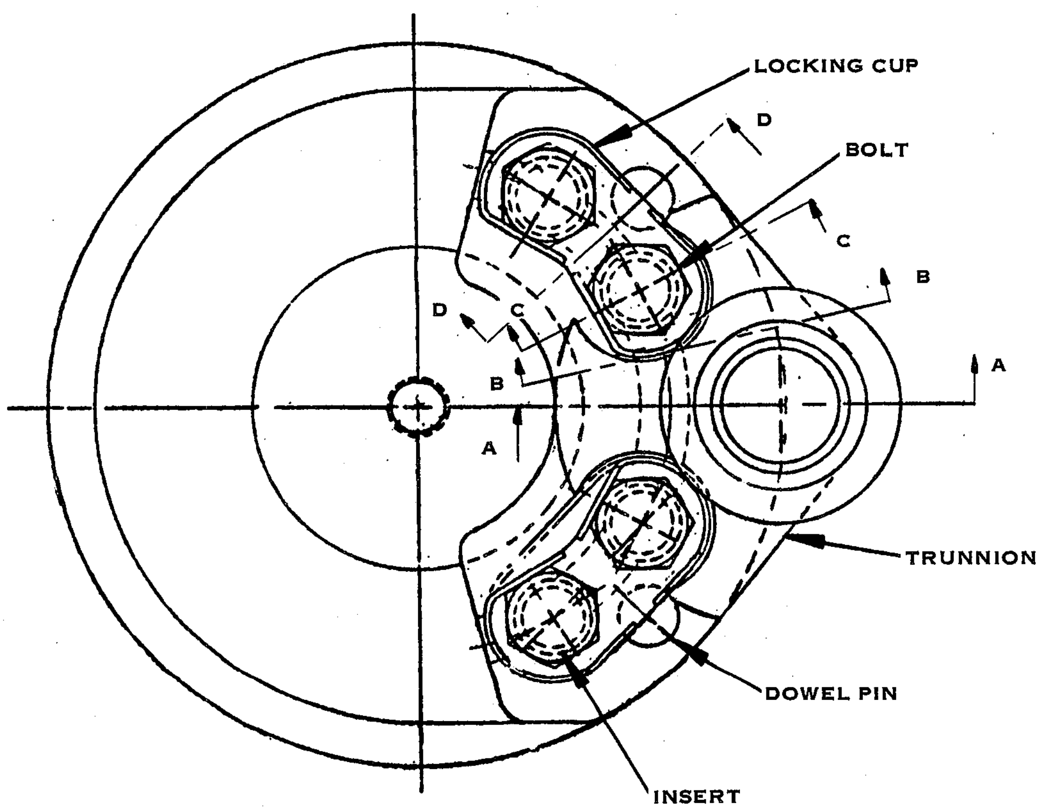


FIGURE 9.3 BLADE SHANK BOTTOM VIEW

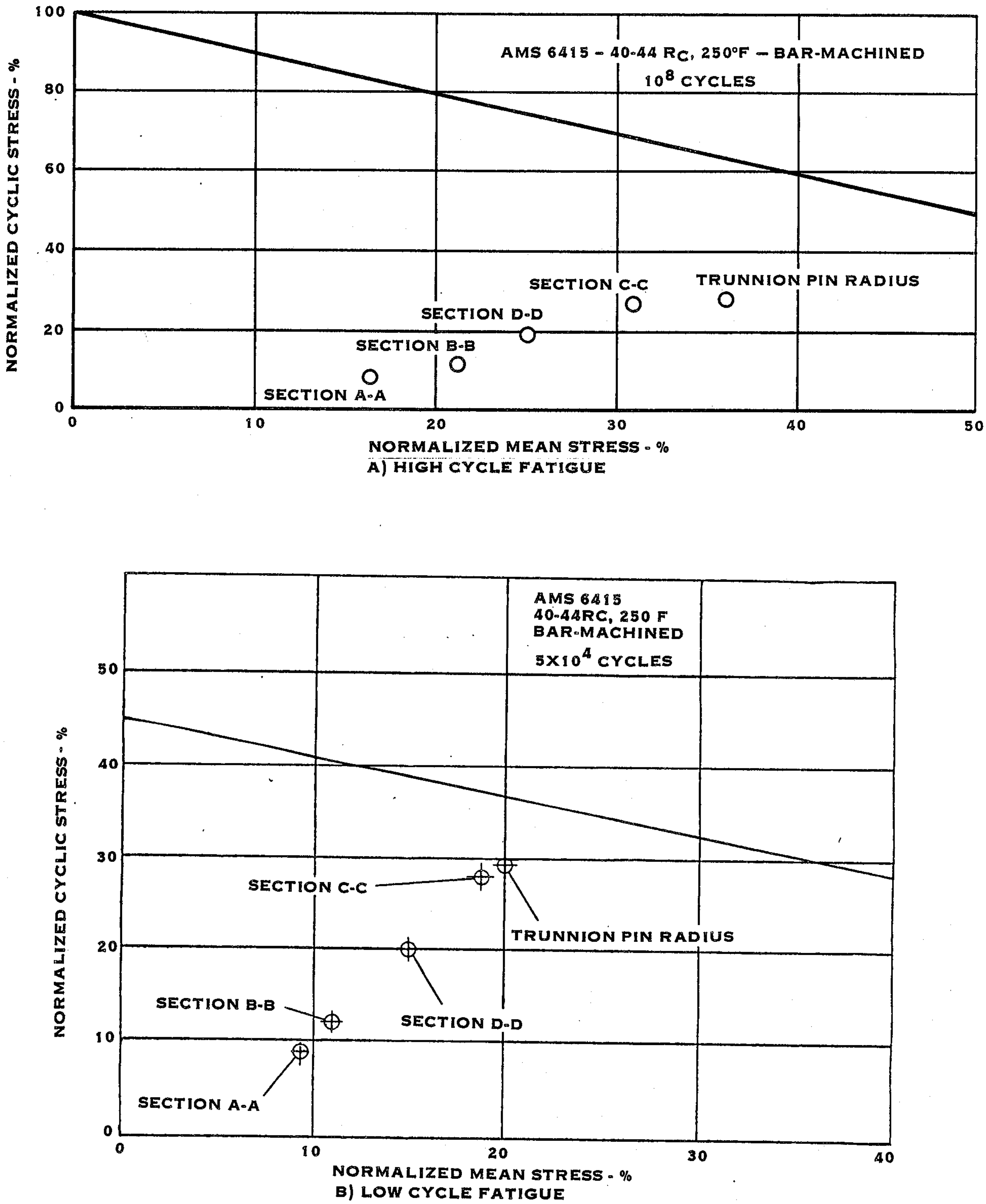


FIGURE 9.4 GOODMAN DIAGRAMS FOR TRUNNION SECTIONS

BOLT LOADS

BOLT TORQUE = 600 - 660 IN-LB
BOLT PRELOAD = 7700 -11750 LB
APPLIED BOLT LOAD = 4790 ± 556 LB

BOLT STRESS

$$S_{SHANK} = \left[\frac{\text{PRELOAD}}{\text{AREA}} + \frac{\text{MEAN APPLIED LOAD}}{\text{AREA}} * N_F\right] \pm \frac{\text{APPLIED CYCLIC LOAD}}{\text{AREA}} * N_F$$

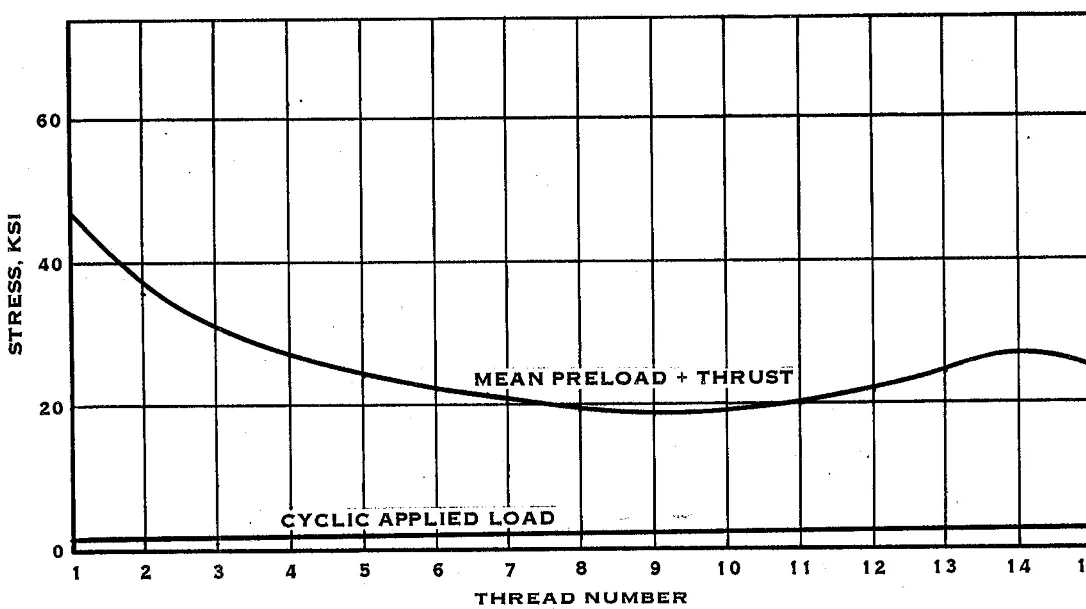
$$\text{WHERE } N_F = \text{JOINT EFFICIENCY} = \frac{K_{BOLT}}{K_{BOLT} + K_{JOINT}}$$

S_{SHANK} = 72668 ± 412 PSI MS = 2084%

$$S_{FILLET} = \text{MEAN SHANK STRESS} \pm \frac{\text{APPLIED CYCLIC LOAD}}{\text{AREA}} * N_F * K_F$$

S_{FILLET} = 72668 ± 824 PSI MS = 992%

FIGURE 9.5 SUMMARY OF BOLT LOADS AND STRESSES



TOTAL BOLT LOAD = 11750 + 4790 + 556 LB
JOINT EFFICIENCY = 0.126, UNEQUAL HOLE SPACING = 1.00
TOTAL MEAN STRESS = MEMBER STRESS + A * THREAD STRESS
TOTAL CYCLIC STRESS = K_{FM} * MEMBER STRESS + A * K_{FT} * THREAD STRESS
K_{FM} = 2.45, K_{FT} = 2.05

FIGURE 9.6 TAPPED HOLE LOAD DISTRIBUTION

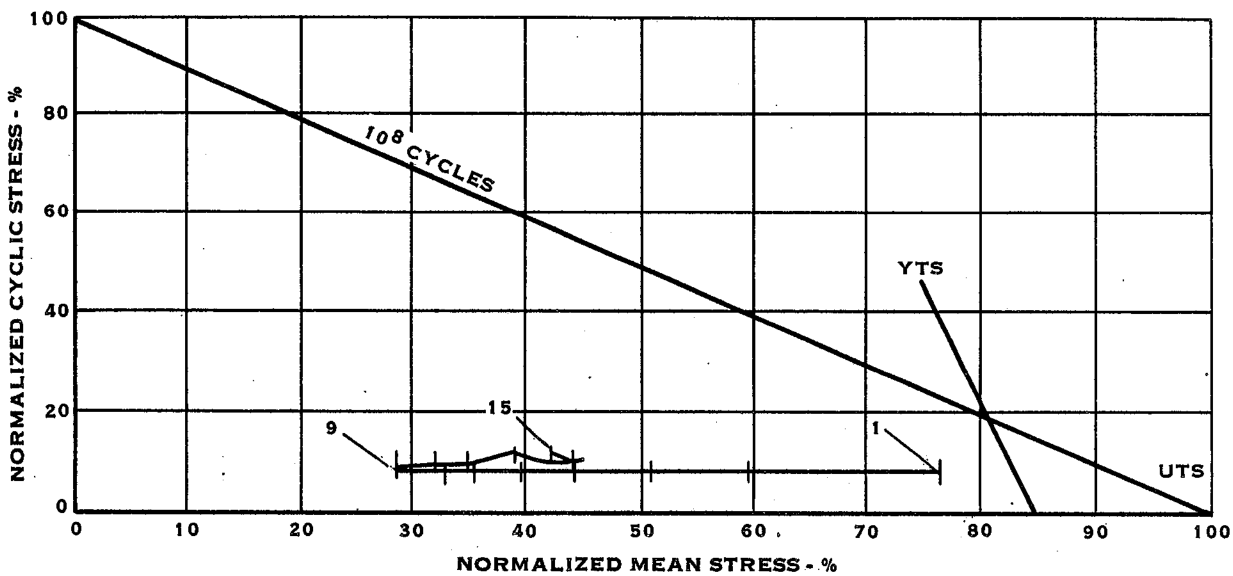


FIGURE 9.7 TAPPED HOLE STRESSES

APPENDIX A

DESIGN REQUIREMENTS

DESIGN REQUIREMENTS

FOR

SR-7L PROPELLER

267X-1

May 6, 1983

Revised 12-1-83

Revised 4-11-84

Prepared by: D W Leishman

Approved by: Dale E Smith

C. E. Baroz

Edward Rothman

LIST OF FIGURES

1.0 INTRODUCTION

In recent years, considerable attention has been directed toward reducing aircraft fuel consumption. Studies have shown that the inherent efficiency advantage that turboprop propulsion systems have demonstrated at 0.65 Mn may now be extended to the higher cruise speeds of today's turbofan powered aircraft. To achieve this goal, propeller designs require advancements such as thin high-speed airfoils and aerodynamic sweep.

A series of small-scale (24.5 inch diameter) model tests have been conducted in both UTRC and NASA wind tunnels and on a modified NASA airplane. These tests have shown that propellers with 8-10 swept blades, high tip speeds and high power loadings can offer increased fuel efficiencies at speeds up to 0.8 Mn. The next logical step is to test this advanced propeller concept (Prop-Fan) in a larger scale.

NASA Contract NAS3-22394 (Propeller Blade Structure Design Study) covers the detail design (thru layout) of a large-scale blade (SR-7L) and a preliminary design of a Prop-Fan suitable for flight test in a future program. NASA Contract NAS3-23051 (Large-Scale Advanced Prop-Fan Program) covers the detailing of the large-scale blade, completion of the design of the Prop-Fan, fabrication of hardware and testing of the isolated assembly at Wright Field and in the Modane wind tunnel in France. The hardware will then be used in a follow-on program where it will be run with an engine on a static test stand, in a low speed wind tunnel and on a research aircraft.

This document sets forth the requirements for the design of the SR-7L Prop-Fan.

1.1 Scope

This specification defines the requirements for the testbed Prop-Fan system (SR-7L) for the Large-Scale Advanced Prop-Fan program (LAP).

1.2 Classification

The testbed Prop-Fan shall consist of blades, disc/retention, actuator, trunnion, constant speed control, and spinner. The assembly shall mount on a 60A spline shaft. It is intended for testing at Wright Field and in a high speed wind tunnel and on a testbed airplane with an engine.

1.3 Features

- Constant speed governing
- Mechanical in-place pitchlock (14SF type)

- Feather via mechanical input signal
- Feather via electrical input signal
- Ground adjustable low pitch stop
- Fixed angle reverse capability
- Electrical aux feather/unfeather motor
- Instrumentation slipring
- Single lever input
- Individual replacement of blades
- Aerodynamic spinner

2.0 APPLICABLE DOCUMENTS

2.1 HS Proposal 81A12

2.2 LAP Program Plan

2.3 LAP Work Plan

2.4 MIL-STD-810C

2.5 MIL-H-5606

2.6 L-14325-1 and -2 Prop-Fan concept

2.7 Design Requirements for Advanced Turboprop Blades (SR-7) dated February 1983

2.8 MIL-P-26366 (vibration environment)

2.9 FAR-25 (deicing)

2.10 FAR Advisory Circular 33-1B dated 4/22/70 (FOD)

3.0 PROPELLER DESCRIPTION

3.1 General - This section defines the general requirements for the SR-7L Prop-Fan.

3.2 Description

Type:	Tractor
No. of blades:	8
Blade:	SR-7-21 (aero version 100)
Diameter:	9.00 ft (108 inches)
Direction of rotation	Conterclockwise (left-hand blade)

Note: Unless otherwise specified, all blade angles are specified at the 3/4 station (40.5 inch radius), per aerodynamic conic.

4.0 Velocity vs. altitude: See Figure 1.

4.2 Power: See Figure 2.

4.3 Tip Speed & RPM:

The normal max tip speed is 800 fps (1698 RPM) and is defined as 100% RPM. Provision shall be made to test up to 105% or 840 fps (1783 RPM). The min RPM to be tested is 600 fps (1273 RPM). As a design goal this range shall be provided without hardware changes.

4.4 Operating Condition

The Prop-Fan shall be designed to operate satisfactorily within the temperature versus altitude limits defined in Figure 3. The system shall be capable of rotating after soaking for a period of three hours in ambient air temperatures of -65°F to +130°F. Operation is permitted as soon as engine oil temperature reaches normal engine operating limits. Maximum

4.4 (cont'd)

temperature of oil provided shall be 170°F.

4.5 Environment Conditions -- The Prop-Fan shall be designed to satisfy the requirements of MIL-STD-810C for humidity, fungus, salt, spray, sand and dust.

4.6 Operating Fluid: MIL-H-5606

4.7 Oil Management -- The pitch change actuator and control shall have a self-contained oil system.

The hub shall have a separate oil supply to lubricate the retention bearing races, the actuator anti-torque spline and the trunnion bearings.

The capacity of the oil tank in the control is 18 quarts.

The actuator requires 12 quarts while operating. When the propeller shuts down (at feather), 6 quarts will drain back to the control.

4.8 Deicing: Deicing heaters shall be incorporated in the blades. There shall be no provisions for deicing ring or brush block and connections at this time.

4.9 Max Loading:

a) Overspeed Limit -- All elements of the rotating propeller will be designed to withstand 125% overspeed or 150% centrifugal load with no inelastic deformation.

All elements of the rotating propeller will be designed to withstand 140% overspeed or 200% centrifugal load. This includes the blade, retention, disc, and blade angle control mechanism. Local inelastic deformation will be permitted in all of these elements at this overspeed but the propeller shall be capable of feathering after exposure to 140% overspeed, but may not be operational.

4.9 b) Proof Pressure: 1.5 times normal pressure (1.5 x 1130 = 1700 psi)

c) Burst Pressure: 2.25 times normal pressure (2.25 x 1130 = 2550 psi)

4.10 Weight -- The blades and blade retention shall be designed to have weight characteristics that are representative of anticipated Prop-Fan systems for future aircraft applications.

For the rest of the system there is no weight target. This design shall be start-of-the-art and aimed at low development risk.

5.0 SPECIFIC REQUIREMENTS

5.1 Blade

5.1.1 Configuration -- Based on SR-7L version 21 (aero version 100). (Reference PDR conducted 2/25/83.)

5.1.2 Deicing -- Heater incorporated in inboard leading edge (ref. FAR-25). The heater will not be wired for use in the Prop-Fan but may be subjected to icing tests by NASA.

5.1.3 FOD Protection -- Metal sheath on outboard leading edge.

5.1.4 Flutter Margin

5.1.4.1 Stall -- Free of flutter at 100% of take-off power at 100% design speed (800 fps) at Mn = 0 to 0.2 for forward thrust and at 20% of takeoff power at 100% design speed at Mn = 0 to 0.2 for reverse thrust.

5.1.4.2 Hi-speed -- Free of flutter over the flight envelope (reference Figure 1) and range of power loadings up to 105% of deaign rotational speed (840 fps) and at 14,000 ft. altitude, the calculated flutter Mach number must be greater than 0.8 at a test rig horsepower and RPM equivalent to the design power coefficient and advance ratio to allow testing in the Modane wind tunnel.

5.1.5 Critical Speed Margins -- No 1P critical speeds shall be permitted in the operating speed range and the minimum margin shall be 40% of maximum operating speed. For 2P excitation, the ground operation critical speed margin at 800 fps shall be a minimum of 20% of Prop-Fan speed and 2P integer crossover. The flight margin shall be a minimum of 10% Prop-Fan speed and 2P integer crossover. This margin shall be reduced inversely as the exciting order is increased from 3P up to 5P. In determining these margins, the effect of blade angle on frequencies shall be determined.

5.1.6 Stress Margins -- The combined steady and cyclic stresses shall be plotted on modified Goodman Diagrams for the materials of construction. The strength boundaries shall represent a high probability of survival derived from experimental data on specimens and full-scale structures. As a minimum, the boundaries shall represent $\overline{X} - 3.5\,\sigma$ lines. The start-stop stress range shall be reflected against a boundary for a life of $50X10^3$ cycles. The high cycle combined stresses shall be reflected against a boundary for $100X10^6$ cycles or infinite life.

The maximum elastic (nominal x k_T) stressing due to 125% overspeed and the nominal stressing due to a 140% overspeed shall be kept below the 0.2% offset yield strength for homogenous metal materials. The change in elastic module shall be kept below 5% for fiber reinforced resin material regarding these same overspeed requirements.

5.1.7 Aerodynamic Excitations -- The equivalent design Excitation Factor (EF) shall be 4.5. The basic EF due to 1P only is 3.3.

5.1.8 FOD Criteria -- The foreign objects are classified into three categories as follows: minor, moderate, and major impacts. Major and moderate impacts correlate with Group I and II definitions in FAR Advisory Circular 33-1B

5.1.8 (cont'd)

dated April 22, 1970. Minor impacts include sand, small stones, and birds up to about 4 ounces. Moderate impacts include two-inch hailstones and birds up to two pounds. Major impacts include a single bird up to four pounds. The damage criteria are as follows:

Minor Impacts	- No damage allowed to basic blade structure. Operation will continue without impediment.
Moderate Impact	- Damage can include loss of material or airfoil distortion. Operation shall continue at 75% power minimum for 5 minutes. No metal fragments shall be lost which can penetrate the aircraft fuselage pressure shell. Roughness shall be tolerable and as a guide, rotor unbalance force shall be kept below 5,000 pounds.
Major Impacts	- Damage can include loss of material or airfoil distortion. Ability to feather the propeller must be maintained. A Shutdown must be accomplished without catastrophic effects on the airframe structure. As a guide, the rotor unbalance force shall be kept below 25,000 lbs. No metal fragments shall be lost which can penetrate the aircraft fuselage pressure shell.

5.1.9 Life and Reliability Goals -- The blade shall be designed for the following goals:

Maximum Continuous Stress Level	Infinite life
Replacement Life	35,000 hours
Mean Time Between Unscheduled Blade Removals (8 blade set) (Inherent)	50,000 hours

5.1.10 Lightning Protection -- Lightning protection shall be incorporated in the blade.

5.1.11 Design Cases -- See Figure 4

5.1.12 Analysis Cases -- See Figure 5

5.2 Disc/Retention

5.2.1 Mounting -- Shaft type, same as 54460 propeller.

5.2.2 Slip Ring -- Provision for mounting an instrumentation slip ring (provided by the Instrumentations Group) on the back of the disc.

5.2.3 Spinner -- Provision for attaching a spinner (reference Figure 6).

5.2.4 Control -- Provision for mounting a control on the tail shaft (reference 54460).

5.2.5 Stress Margin -- The maximum elastic (nominal X stress concentration factor) stressing due to a 125% overspeed and the nominal stressing due to a 140% overspeed shall be kept below the 0.2% yield strength for homogeneous metal material.

5.2.6 Life -- The disc shall have a life of 50,000 start/stop cycles. The retention bearing shall have a life of 50,000 cycles of low cycle fatigue and unlimited life under maximum centrifugal, maximum steady bending and maximum vibratory bending anywhere in the flight envelope.

5.2.7 Retention Bearing -- The retention bearing shall be capable of replacement or the disc shall include provisions for up to three (3) regrinds with a maximum total stock removal of 0.04.

5.2.8 Speed Pickup -- Provisions for the rotating portion of a speed pickup shall be made. This signal shall also be conpatible with synchrophasing. The device shall be the same or similar to 54460 hardware.

5.2.9 1P Shaft Moment -- The maximum 1P shaft moment is predicted to be 76,500 in.lbs. at the following condition:

1698 rpm
6000 SHP
$M_\eta = 0.2$ (132 KEAS S.L. S.D.)
EF = 3.3

5.3 Pitch Change Actuator/Trunnions

5.3.1 General Requirements -- Any malfunction of the pitch change actuator will cause the system to pitchlock or feather. The actuator shall be modular to the extent that is practical for the test program.

5.3.2 In-place Pitchlock -- An in-place pitchlock shall be provided to limit the max loss of blade angle to approximately 1 degree below the governor scheduled blade angle. The minimum pitchlock gap shall allow the pitch change system to slew at a rate of approximately 9 degrees/sec.

5.3.3 Ground Adjustable Low Pitch Stop -- A ground adjustable low pitch stop is provided for reverse blade angle ground testing. The stop shall be set to limit in-flight blade angle from going below $+40^{\circ}\beta$. The stop shall be adjustable between $-10^{\circ}\beta$ and $+40^{\circ}\beta$ for reverse testing. The low pitch stop shall be readily adjustable without dismantling the pitch change system.

5.3.4 Feather -- The pitch change system shall be capable of feathering the propeller to an angle of 87.5 degrees $\pm$ 2.5° with an accuracy of 0.1°.

5.3.5 Reverse -- The actuator shall provide the travel necessary to give a reverse blade angle of -10° set by the adjustable low pitch stop. The actual minimum blade angle will be established by blade-to-blade interference. There is no beta control to the reverse angle.

5.3.6 Actuator Sizing

5.3.6.1 Aerodynamic Sizing Criteria

CTM (Centrifugal twisting moment):	See Fig. 7
ATM (Aerodynamic twisting moment):	See Fig. 8
FTM (Frictional twisting moment):	$CL \times \frac{PD}{2} \times .005$
CL (Centrifugal Load):	82,900 lbs. at 1698 rpm
TTM (total twisting moment):	CTM + ATM + FTM

A contingency allowance of 20% shall be added to the ATM and CTM of Figs. 7 and 8.

5.3.6.2 The pitch change system shall be capable of increasing blade angle with a 10 psi margin over the operating range of Figure 1 at 110% rpm (1868 rpm) and 0 SHP.

5.3.6.3 The pitch change system shall be capable of changing blade angle at 9 deg/sec over the operating range of Figure 1 at 100% rpm (1698 rpm) at 80% (4800) SHP.

5.3.7 Life -- The minimum design fatigue life shall be 10,000 start/stop cycles with a flight cycle time of 1.5 hours.

5.3.8 Instrumentation -- Provision to measure both high and low pitch pressure during operation shall be provided. (Pressure sensors to be provided by the Instrumentation Group.) Provision to measure blade angle shall be provided.

5.3.9 Time Constant -- The actuator loop time constant shall be 0.25 sec at no load.

5.4 Control

5.4.1 Configuration -- Based on 739000-1 (54460/E2).

5.4.2 Control Schematic

a) Present 54460 Control (SK90621)

b) Modified schematic for LAP (L-14325-3)

5.4.3 Governing Range -- 1273 to 1783 rpm (600 to 840 fps) (75 to 105%)

5.4.4 Operating Pressure

Supply - 1050 to 1130 psig
HPRV setting - 1150-1375 psi
Pump Pulsation - 1150 ± 175 psi

5.4.5 Pump Flow

62.2 qts/min at 100% rpm, 1125 psi, 170°F

46.6 qts/min at 75% rpm, 1125 psi, 170°F

5.4.6 Leakage

5.4.6.1 Control - 1.5 qts/min at 100% rpm, 170°F

5.4.6.2 Transfer Bearing - 3.7 qts/min at 100% rpm, 170°F

5.4.6.3 Pitch Change - Null 13 qts/min at 100% rpm, 170°F
- slewing 1.1 qts/min at 100% rpm, 170°F

5.4.7 Net Flow - 55.9 qts/min slewing at 100% rpm, 170°F

5.4.8 Cooling Requirements - Cooling for the control shall be provided by an airframe supplied heat exchanger. The heat exchanger must remove 550 btu/min from the control hydraulic oil. Circulation of the control oil through the heat exchanger shall be provided by an airframe supplied pump. Fittings for hydraulic lines to and from the heat exchanger shall be located at the rear of the control per L-14325-20.

Oil outlet temp of heat exchanger:	170°F max
Pump outlet pressure:	30 psi max (@ 170°F)

5.4.9 Input Lever - The control shall have a single lever input for feather and speed set. The shaft can be driven by either a mechanical input or electric motor. The drive is aircraft supplied. The input lever torque is 65 in.lbs. max.

5.4.9.1 Schedule -- Figure 9 shows the condition lever schedule.

5.4.10 Feather -

Electrical -- A solenoid actuated by the engine overspeed governor and/or a feather button shall be provided.

Mechanical -- The input lever shall be the mechanical input for feather.

5.4.11 Overspeed protection -- An aircraft-supplied overspeed system shall activate the feather solenoid whenever the propeller speed exceeds 110% of 800 FPS.

5.4.12 Beta control -- There are no provisions for Beta control.

5.4.13 Unfeather -- The auxiliary motor shall be used to unfeather the Prop-Fan.

5.4.14 Reverse -- Provision to run in reverse with the adjustable low pitch stop set to the desired reverse angle shall be provided.

5.4.15 Synchrophasing -- Provisions to incorporate synchrophasing shall be provided within the constraints of ease of assembly and low development risk the hysteresis and backlash between the governor and the actuator shall be minimized.

5.4.16 Dynamic Response -- Governor gain. 2.4°/sec/% RPM at null.

5.4.17 Aux Motor & Pump -- An electrical motor-driven pump is provided to assist in feathering, airstart, and for ground handling.

5.4.17 (cont'd)

- o Pressure: 1375 PSI
- o Flow: 14 qts/min
- o Voltage: 208 VAC, 3 phase, 400 cycles
- o Amperage: 15 normal, 30 at relief valve
- o Duty cycle: 5.5 HP for 10 sec
4.5 HP for 10 sec
15 min off

5.4.18 Life -- The minimum design fatigue life shall be 10,000 start/stop cycles with a flight cycle time of 1.5 hours.

5.4.19 Instrumentation -- Provision shall be made to measure supply, governor metered pressure, and stationary transfer bearing and sump temperature. (The sensors will be provided by the Instrumentation Group.)

5.4.20 Brush Block -- Provision shall be made to mount an instrumentation brush block (provided by the Instrumentation Group) on the existing brush block mounting pad.

5.4.21 Speed Pickup -- Provision for the stationary portion of a speed pickup shall be made. This signal shall also be compatible with synchrophasing. The device shall be the same or similar to 54460 hardware.

5.5 Spinner

5.5.1 Contour - Per Figure 6.

5.5.2 Material -

5.5.2.1 Forward - Fiberglass

5.5.2.2 Aft - Fiberglass

5.5.2.3 Bulkheads - Fiberglass

5.5.3 Deicing - Not required

5.5.4 Length - 44.5 in.

5.5.5 Blade Plug - A spinner to blade ring is required.

5.5.6 Removal/Installation -- The spinner shall be designed for each of installation and removal of the forward spinner to facilitate system testing.

6.0 MAINTAINABILITY/RELIABILITY

6.1 M & R

There are no specific design requirements for maintainability and reliability. The hardware should be designed following good design practice and using state-of-the-art technique.

7.0 OTHER CONSIDERATIONS

7.1 There is the potential for a contract modification to add opposite rotation (clockwise) hardware to the program. During the design effort, ways to minimize the effect of incorporating opposite rotation should be considered. However, opposite rotation shall not be incorporated at this time.

7.2 The SR-7L will be tested on the static thrust rig at Wright Field to verify its static performance. For this test it is desirable to operate the Prop-Fan as a fixed pitch/ground adjustable unit. (The control will not be used.) Therefore, during the design of the SR-7L, Design should consider ways to incorporate a fixed pitch/ground adjustable actuator in the Prop-Fan, but this actuator shall not be designed at this time.

7.3 The SR-7L will be tested on the static thrust rig at Wright Field to explore the stall flutter boundary and in the Modane wind tunnel to explore the high speed flutter boundary. For these tests it is desirable to operate the Prop-Fan with direct blade angle control and not with speed control. To accomplish this it is envisioned that the blade angle could be varied using an external hydraulic supply directed to the high or low pitch chambers of the actuator through a commercial transfer bearing mounted on the front of the Prop-Fan. Design should consider ways to incorporate this capability in the hardware, but this hardware shall not be designed at this time.

7.4 The hardware designed in this program will be used in a future NASA Propeller Test Assembly (PTA) contract. The PTA contract will include static engine testing. At this time it is not known whether this test will be conducted on an indoor or outdoor test stand. The hardware will also be subjected to testing with an engine and wing in a low speed wind tunnel and with an engine on a testbed airplane.

8.0 PROPELLER INTERFACES

8.1 Propeller control installation drawing (Preliminary) (Ref. L14325-1 & L14325-2). This drawing defines the concept and interface between the propeller, the pitch change system, the control, and the gearbox.

8.2 Hub/Flange/Control Mounting -- See Figure 10 (T56).

8.3 Slip Ring -- A slip ring will be provided for instrumentation only. The brush block will mount on the control in place of the deicing brush block. The slip ring and brush block shall be supplied by the Instrumentation Group.

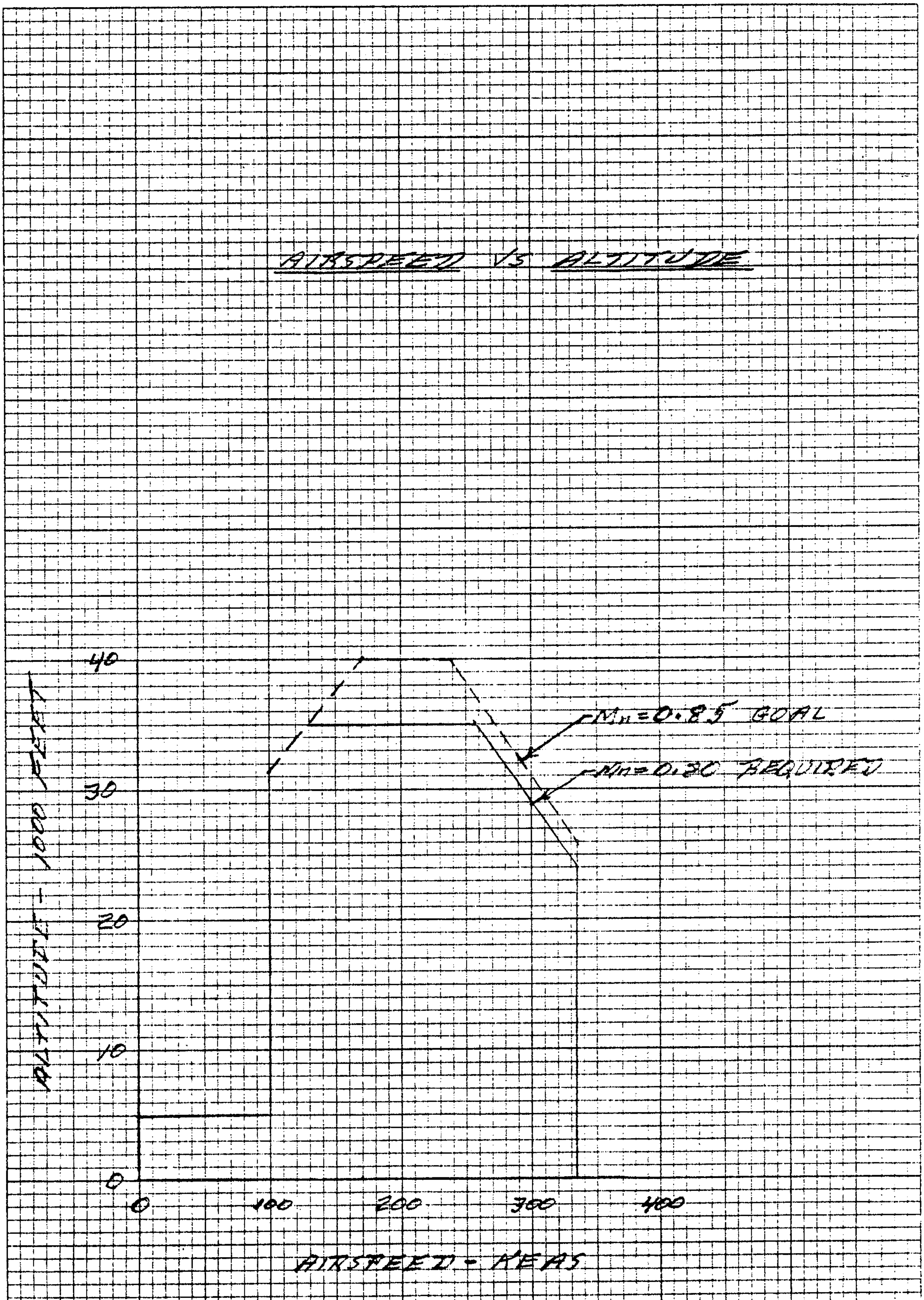

AIRSPEED VS ALTITUDE
Mn=0.85 GOAL
Mn=0.80 REQUIRED
ALTITUDE - 1000 FEET
40
30
20
10
0
0
100
200
300
400
AIRSPEED - KEAS

FIGURE 1

ENGINE POWER VS. ALTITUDE

SHP	Mn	Altitude	SHP/D^2	Tip Speed
6000	0-0.2	Sea Level	74.1	800 ft/sec
5400	0.5	10,000 ft.	66.7	800
4500	0.6	20,000	55.6	800
3820	0.8	30,000	47.2	800
3240	0.8	35,000	40.0	800
2025	0.8	35,000	25.0	600

FIGURE 2

TEMPERATURE VS ALTITUDE
ALTITUDE - 1000 FEET
40
30
20
10
0
+60
+30
0
-30
-60
-90
TEMP - °F

FIGURE 3

DESIGN CASES FOR SR-7L BLADE

CASE	CONDITION	Mn	ALTITUDE	HP	SHP/D^2	TIP SPEED	RPM	EF
1	Cruise	0.8	35,000 ft.	2592	32	800 fps	1698	4.5
2	Min Climb	0.2	Sea Level	6000	74.1	800 fps	1698	4.5
3	25% O'speed	0.8	35,000 ft.	0	0	1000 fps	2122	0
4	40% O'speed	0.8	35,000 ft.	0	0	1120 fps	2377	0

ANALYSES TO BE CONDUCTED

CASE	FLUTTER STALL	FLUTTER HI SPEED	CRITICAL SPEED MARGINS	STRESS MARGINS	FOD
1	No	Yes	Yes	Yes	No
2	Yes	No	Yes	Yes	Yes
3	No	No	No	Yes (steady)	No
4	No	No	No	Yes (steady)	No

FIGURE 4

Revised 9/15/83

SR-7L BLADE - CONDITIONS TO BE ANALYZED

1.	Wind Tunnel*	0.8 Mn	14,000 ft.	1050 HP	13 SHP/D^2	800 fps	1698 rpm	4.5 EF
2.	Static	0	Sea level	6000	74.1	800	1698	0
3.	Reverse	0	Sea level	1200	14.8	800	1698	0
4.	Cruise-lo rpm	0.8	35,000	2025	25	600	1273	4.5
5.	Cruise-hi rpm & Mn	0.85	35,000	3280	40.5	840	1783	4.5
6.	Climb	0.5	10,000	5400	66.7	800	1698	4.5
7.	Dive	0.6	20,000	0	0	800	1698	0
8.	Dive	0.8	35,000	0	0	800	1698	0

These conditions will be analyzed--however, the blade design will not be modified based upon the results.

* Conditions shown are for 8 blades. Will be done for 2 and 4 blades also.

Figure 5

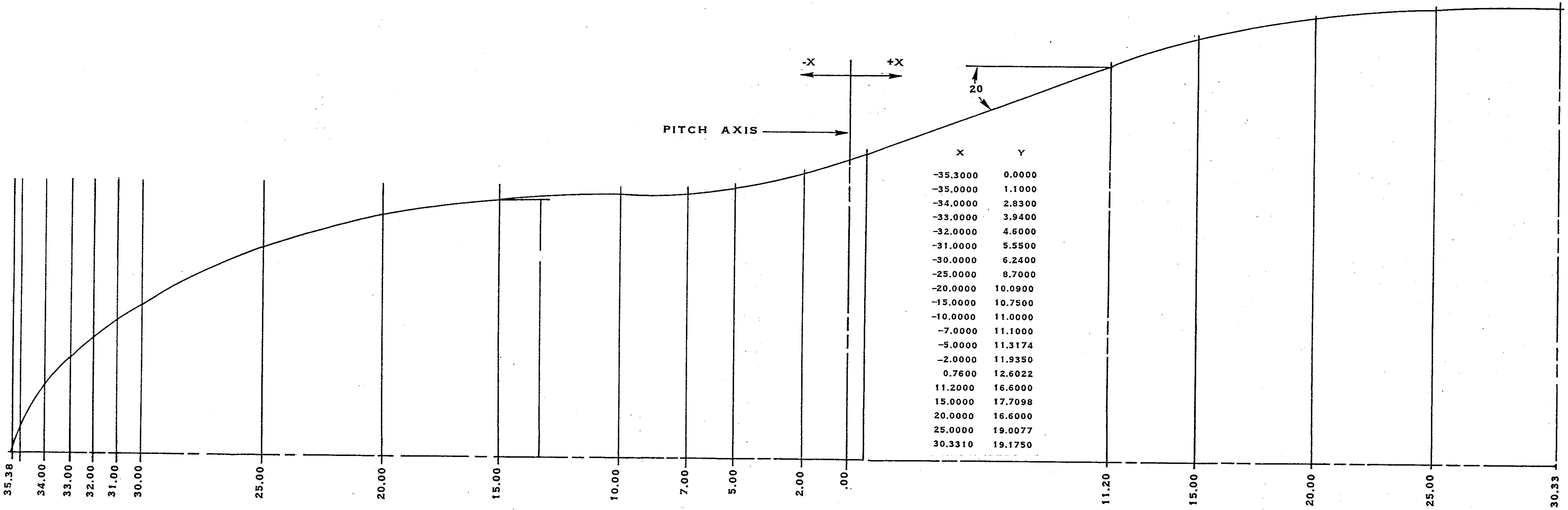
Revised 9/15/83

ANALYSES TO BE CONDUCTED

Case	Flutter: Stall	Flutter: Hi-Speed	Critical Speed Margins	Stress Margins	FOD
1	No	Yes	Yes	Yes	No
2	Yes	No	Yes	Yes (Steady)	No
3	Yes	No	Yes	Yes (Steady)	No
4	No	Yes	Yes	Yes	No
5	No	Yes	Yes	Yes	No
6	No	Yes	Yes	Yes	No
7	No	Yes	No	Yes (Steady)	No
8	No	Yes	No	No	No

Figure 5 (continued)

Revised 9/15/83



X	Y
-35.3000	0.0000
-35.0000	1.1000
-34.0000	2.8300
-33.0000	3.9400
-32.0000	4.6000
-31.0000	5.5500
-30.0000	6.2400
-25.0000	8.7000
-20.0000	10.0900
-15.0000	10.7500
-10.0000	11.0000
-7.0000	11.1000
-5.0000	11.3174
-2.0000	11.9350
0.7600	12.6022
11.2000	16.6000
15.0000	17.7098
20.0000	16.6000
25.0000	19.0077
30.3310	19.1750

FIGURE 6. PRELIMINARY SPINNER/
NACELLE CONTOUR FOR SR-7 PROP FAN

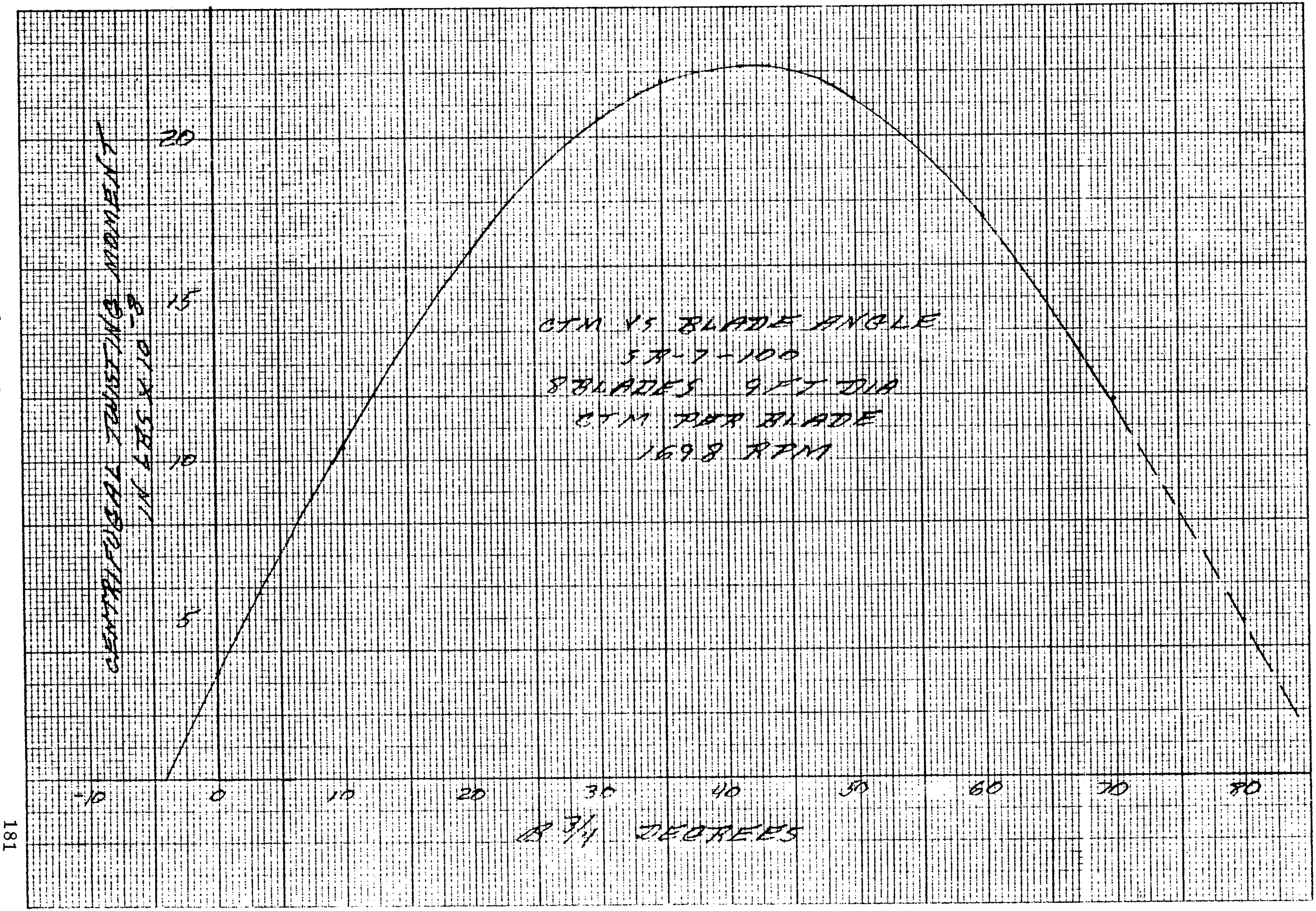

CTM VS BLADE ANGLE
3R-7-100
8 BLADES 9 FT DIA
CTM PER BLADE
1698 RPM
CENTRIFUGAL TWISTING MOMENT
IN LBS X 10-3
20
15
10
5
-10
0
10
20
30
40
50
60
70
80
β 3/4 DEGREES

FIGURE 7

NO. 340-20 DIETZGEN GRAPH PAPER
20 X 20 PER INCH

DIETZGEN CORPORATION
MADE IN U.S.A.

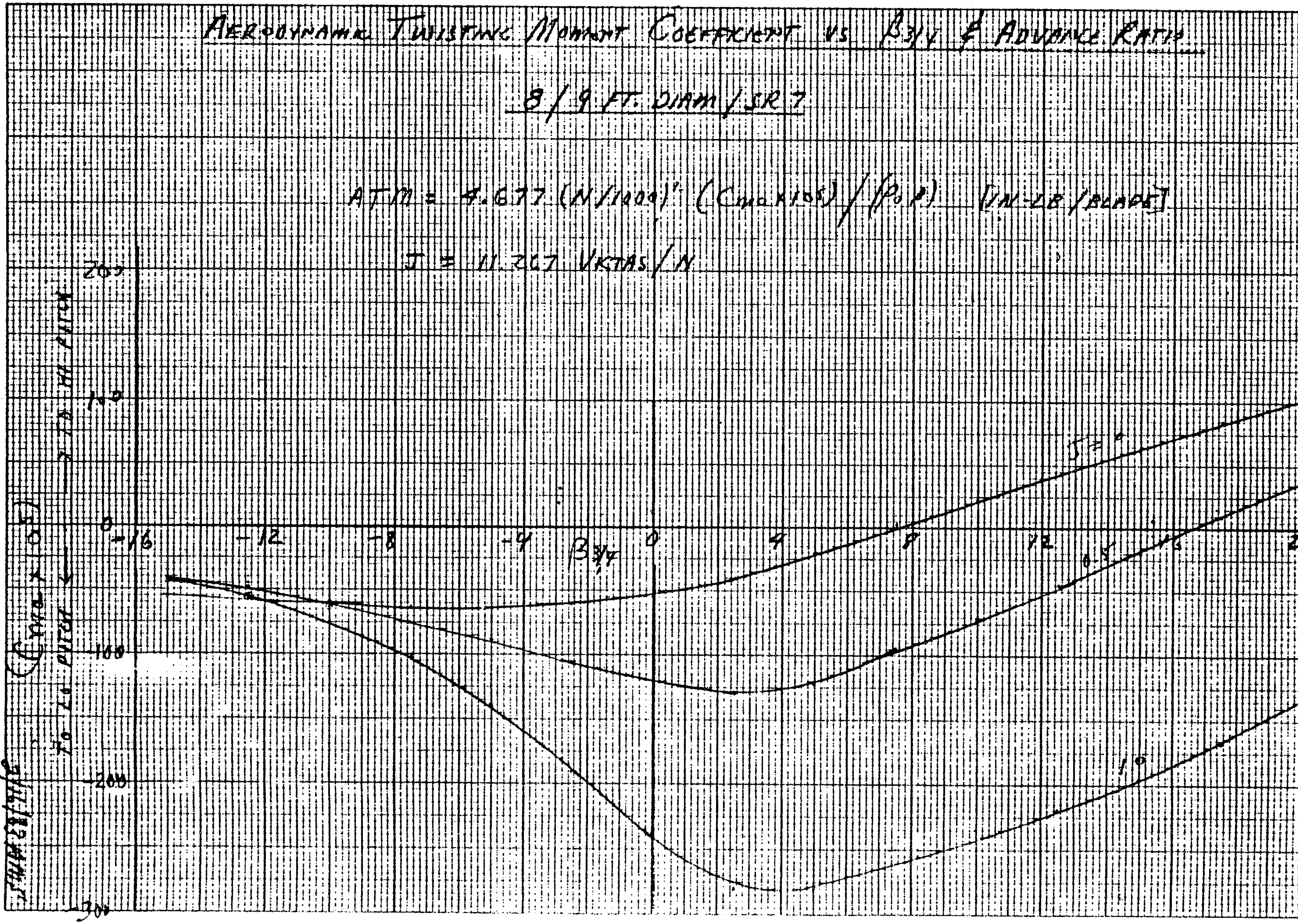
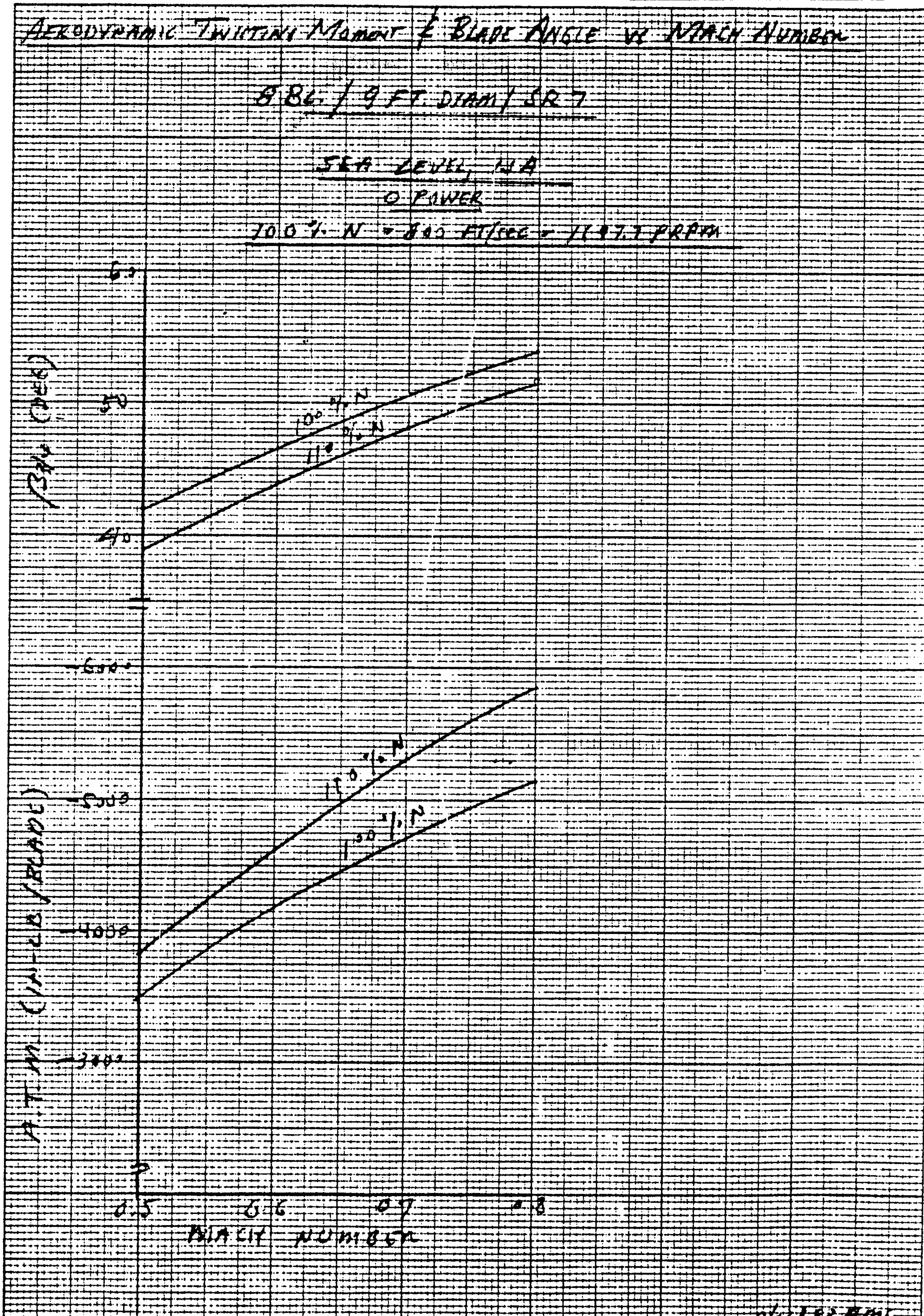


FIGURE 8

182

CALL No. 2956C(1)

CALC. No 2956 C(1)



NO. 340-20 DIETZGEN GRAPH PAPER 20 X 20 PER INCH

DIETZGEN CORPORATION MADE IN U.S.A.

FIGURE 8 (cont'd)

INPUT LEVER SCHEDULE

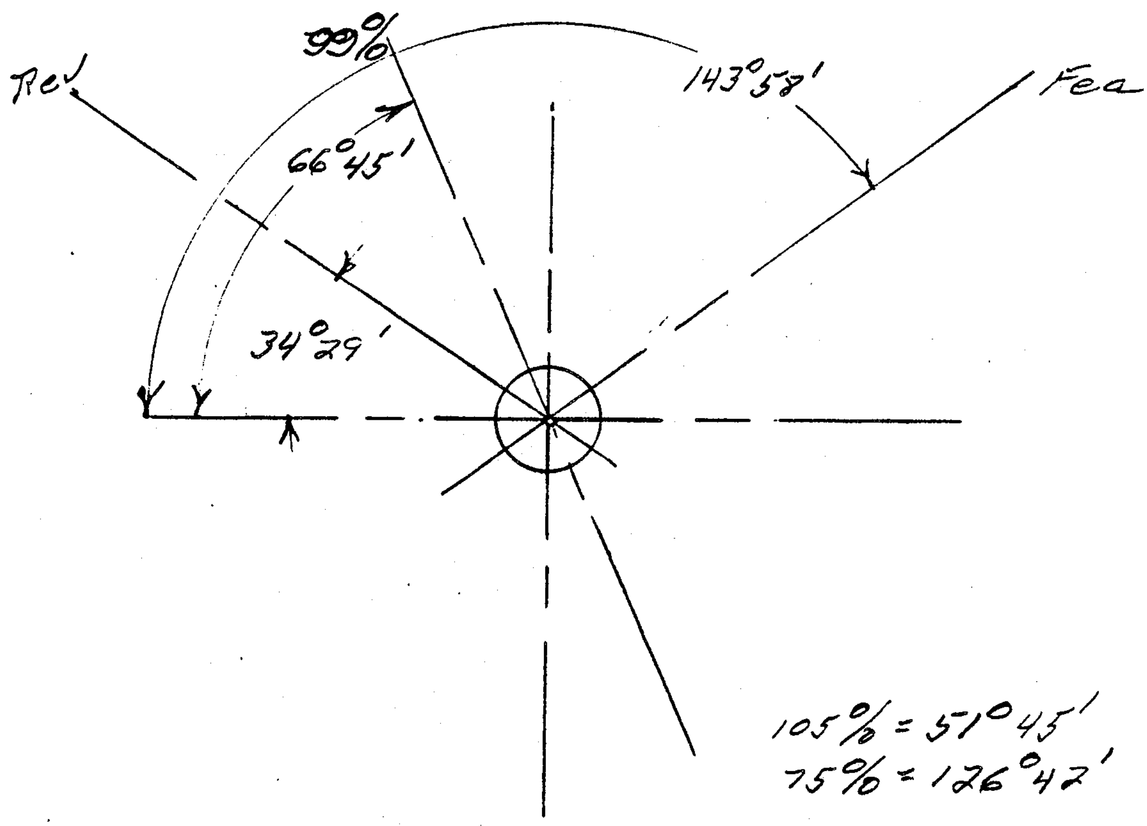
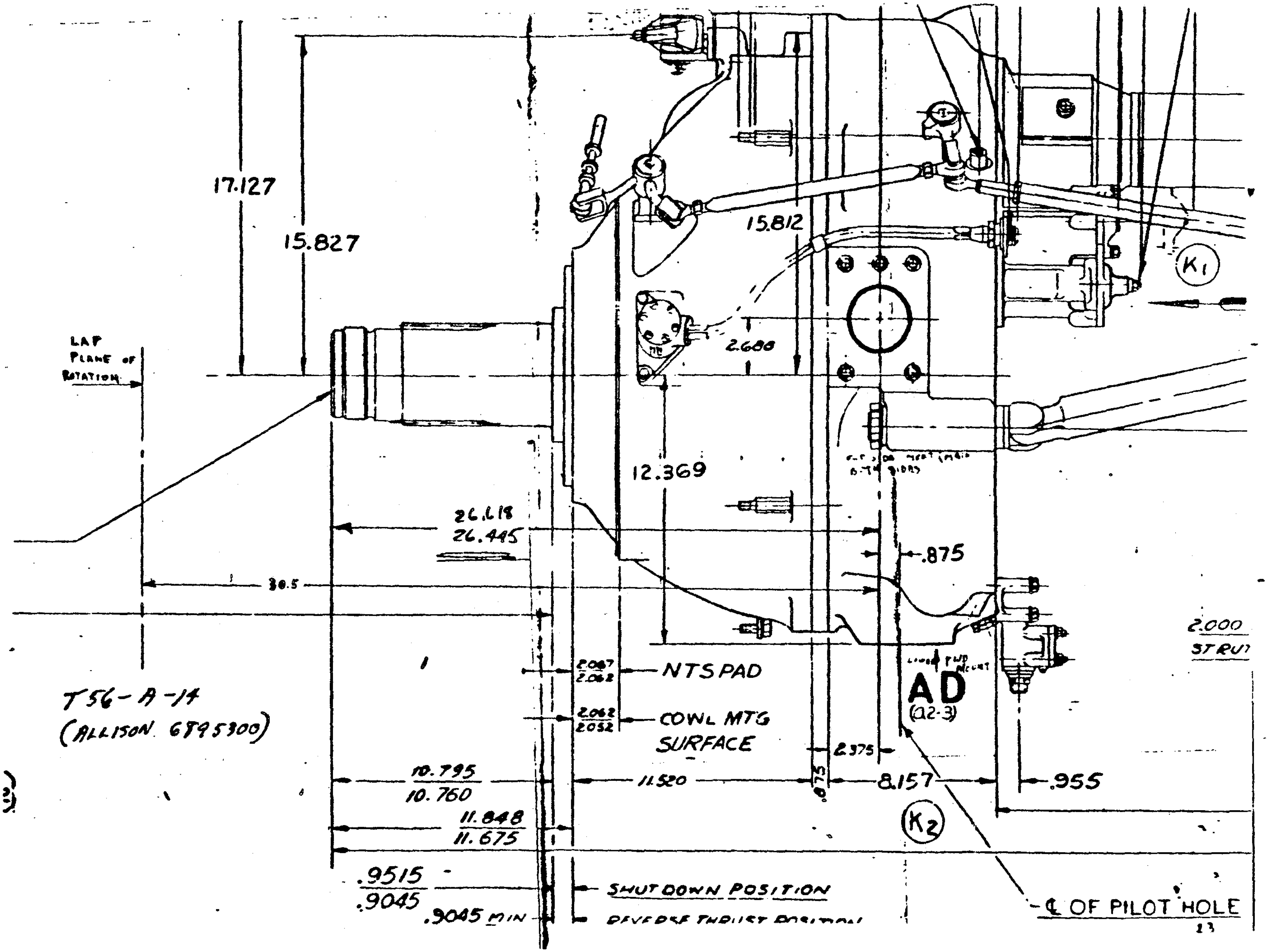


FIGURE 9



17.127
15.827
15.812
2.688
12.369
LAP PLANE OF ROTATION
26.618
26.445
30.5
.875
2.087
2.062
NTS PAD
2.062
2.052
COWL MTG SURFACE
T56-A-14
(ALLISON 6895300)
AD
(12-3)
2.375
10.795
10.760
11.520
.875
8.157
.955
11.848
11.675
.9515
.9045
.9045 MIN
SHUTDOWN POSITION
REVERSE THRUST POSITION
2.000
STRUT
℄ OF PILOT HOLE

FIGURE 10

APPENDIX B

DESIGN LAYOUTS

MATCH DWG. TO REVIEW DETAIL DWG.

UOS SCALE RADII

29. AMS 4127; ROUGH MACH. STARTING MAT'L TO 2 IN. MAX WALL AND SOLUTION HEAT TREAT & AGE TO 6061-T6 COND PER HS854; 15 MIN MIN

31. BACKLASH ALLOWS FOR .006 MIN MISAL'T & .004 RAD GROWTH (ENG. REF)

30. SPANNER INTERFACE CONFIG. & LOC. OF ITEMS 52 & 53 TO BE PER L-14325-7

27. ITEM 17 & 18 INSTALLED GAP AT SCARF CUT TO BE .080±.050

28. THIS DIA TO BE .060 MIN LARGER THAN O.D OF ITEM 46

23. ASS'D BEARING SURFACE OF NUT TO BE PARALLEL TO BEARING SURF. OF MATING PART WITHIN .005

26. CLEARANCE BETWEEN BLADE BEARING ASSY AND ITEM 2 SHALL BE SUFF. TO ALLOW BLADE REMOVAL AT -6°

25. ITEM 50 FIT WITH ITEM 5 SHALL PROVIDE CONC. OF DIA P TO DIA N

24. ANAL CL SHALL NOT LIMIT MISALIGNMENT CAPABILITY OF ITEM 12

23. THIS MAT'L ON ITEM 3 TO BE MACH. AT ASSY TO PROVIDE TBD BLADE TO BLADE CLEARANCE PER PROJECT ENG. WITH ITEMS 3 & 1 IN CONTACT.

22. THIS DIM. SHALL ALLOW ASSY. OF ITEM 34 IN POS. SHOWN ONLY

21. ⌖ .003 TOTAL Ⓜ

20. ITEMS 52 & 53 IN STRIP FORM; BOND PER HS1908 CODE I B5, OPTIONAL R.T. CURE, 48 HRS. MINIMUM. SURF. PREP: ITEMS 52 & 53 PER HS1265 METHOD P. SOLVENT CLEAN ITEM 5 PER HS1265 METHOD (AFTER ANOD) NOTE: USE LEND WEAVE FABRIC (TAPE) FOR BONDLINE CONTROL.

19. .003 MIN RAD CL. .024 MAX RAD EXTRUSION GAP (.030 MAX EXTRUSION GAP OPERATING REF.)

18. SOLID FILM LUBE PER HS 218 TYPE A2

17. WEAR PATTERN DEFINITION SHOWN MUST BE ON WEAR PLATE CARBURIZED REGION UATC; INC. RADIAL BLADE DEFL. OF .025 IN (@ 36,000# CL. REF); INCLUDE ROTARY MOTION OF NOTE #5

16. ENG. REF. — SEAL COMP: 7% - 18%; GROOVE SWELL CAP: 15% - 35% SEAL OD FIT WITH GROOVE ID: 10% - 15% INTERFERANCE

15. MAT'L: TORLON 4301 PURCHASE FROM AMOCO CHEMICAL CORPS.; CHICAGO. ILL.

14. ITEM 9 SHALL HAVE AN ADJUSTMENT RANGE TO PROVIDE A FEATHER STOP FROM 85° TO 90°

13. ANODIZE PER HS 334 TYPE I

12. INSTALL PER HS1552 CODE C

11. BOLT SHALL HAVE 1.5 DIA. ENGMT.

10. INDICATED FIT REQ'D AFTER ASSY OF BEARING INTO OUTER RACE. PERMISSIBLE TO MACH. IF NECESSARY.

8. BUTTRESS THD. DETAIL PER L-14325-9 SHT 2

7. 8 SLOTS IN ITEM 9 AND 7 SLOTS IN ITEM 1 (REF. ANAL ADJ FOR 1 LOCKING INCREMENT = .002)

6. DEFORM KEY WASHER INTO SPLINE (3 PLCS EQ SPACED)

5. NO FLAWS TO EXCEED SPECIFIED SURF ROUGHNESS

4. 1 WIDE TOOTH EQ SPACED EVERY 90°

3. 1 WIDE SPACE EQ SPACED EVERY 90°

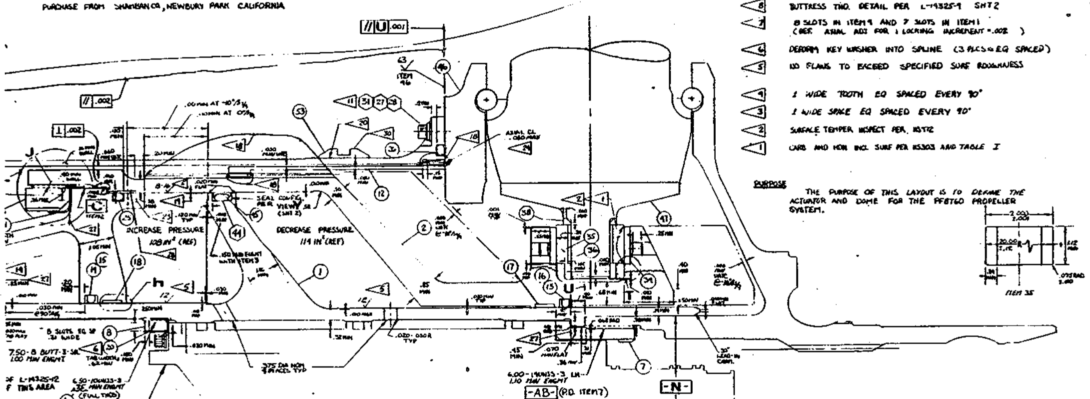
2. SURFACE TEMPER INSPECT PER HS 772

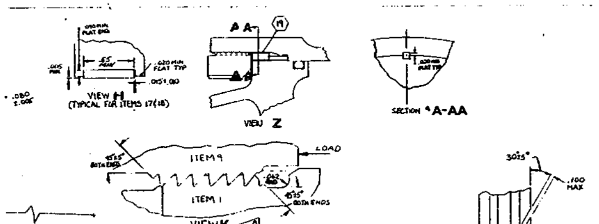
1. CARB AND HDN INCL SURF PER HS303 AND TABLE I

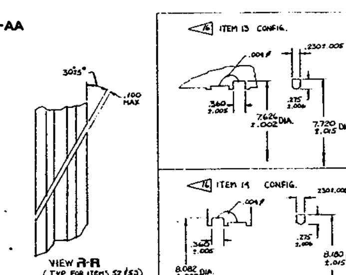
PURPOSE

THE PURPOSE OF THIS LAYOUT IS TO DEFINE THE ACTUATOR AND DOME FOR THE PF8760 PROPELLER SYSTEM.

PURCHASE FROM SHAMBAN CO., NEWBURY PARK CALIFORNIA







TORQUE TABLE	(PER HS2212)		
ITEM	TORQUE	CLASS	PRELOAD (REF)
28	135-165 IN-LB	III	2900#-4200#
7	3500-4000 FT#	III	16,000#-22,000#
8	120-150 FT#	III	500#-1,000#
10	2000-2500 FT#	III	20,000#-36,000#
11	3500-4000 FT#	III	21,000#-36,000#
33	40-60 IN-LB	III	~

DIAMETER FITS		
ITEMS	FIT	
37 to 47	.0007-.0023 DIA CL	
39 to 47	.0016-.0036 DIA CL	
39 to 37	.0006-.0022 DIA CL	
50 to 6	SECTION W-W	
50 to 5	.0009-.0016 DIA INT	@ ITEM 6
39 to 1	.0001-.0011 DIA INT	
1 to 2	VIEW H	
1 to 3	VIEW H	
1 to 2	19	
6 to 32	.009-.008 DIA CL	
5 to 6	.003-.011 DIA CL	
3 to 17	.002-.014 DIA INT.	
2 to 17	.002-.014 DIA INT	
2 to 13	.009-.013 DIA CL.	@ SEAL DIA.
1 to 17	.002-.010 DIA CL	
50 to 5	25	@ ITEM 46
1 to 18	.002-.010 DIA CL	
1 to 46	.002-.007 DIA CL	
2 to 3	.001-.005 DIA CL	@ DIA. C
37 to 47	.0007-.0023 DIA CL	
36 to 47	.0012-.0026 DIA CL 10	
38 to 47	.0096-.0110 DIA CL	
35 to 36	.0035-.0060 DIA INT	
50 to 46	.0009-.0016 DIA INT	
5 to 50	SECTION S-S & SECTION V-V	
58 to 1	.004-.010 DIA CL	
52 to 58	.004-.012 DIA CL	@ 6.75 DIA
34 to 35	.0015-.0085 TOTAL CL Ⓜ	

ITEM NO	NO REQD	PART NAME	PART NUMBER	EST WT	MATERIAL AND HEAT TREAT			REMARKS
58	1	SUPPORT, TUBE		.15	AMS 4117;		HS447	15
57	2	PACKING, PREFORMED			VITON (REF)	~	~	
56	1	RING, RETAINING		.155	AISI 302 (REF)	~	~	
55	2	PACKING, PREFORMED			VITON (REF)	~	~	
54	3	PACKING, PREFORMED			NITRILE (REF)	~	~	
53	1	SUPPORT, O-RING, MID		.74	HS 1401 GR (REF)		~	MAT'L TO BE PER P/N 733863
52	1	SUPPORT, O-RING, FRONT		.73	HS 1401 GR D (REF)		~	MAT'L TO BE PER P/N 733863
50	3	PIN			HS 2776 (REF)	~	~	
48		SHIM		~	AMS 6519 (REF)	~	~	
47	1	TRUNNION		~	AMS 6415 (REF)	~	~	
46	1	HUB		~	AMS 6265 (REF)	~	~	
45	1	SEAL RING		.074	TURCON / MOLY (REF)			37 VENDOR P/N 530662-300
44	1	PACKING, PREFORMED			MIL-P-83461 (REF)		~	
40	8	PIN, COTTER			CRES. (REF)	~	~	
39	8	PIN, HEADED			CRES. (REF)	~	~	DRY FILM COATED (REF)
38	8	WASHER, THRUST			HS 8441 CLB (TYPE SP 21)		~	REF P/N 772536-1
37	8	SLEEVE, RETAINING			AMS 4122 HS454		HS447	REF P/N 786359 15
36	8	BEARING, JOURNAL			HS 8441 CLB TYPE SP21		~	REF P/N 738187-1 (VESPEL)
35	8	BEARING, RACE, OUTER			AMS 6490		SEE SHT 2	REF P/N 59036-10
34	16	PLATE, WEAR			AMS 6265 34 1			REF (16 SP)
33	3	PLUG			AL ALLOY (REF)	~	~	
32	2	TUBE, PRESSURE SENSOR	L-14325-10	~		~	~	
31		LOCKWIRE		~	CRES (REF)	~	~	INSTALL PER AS567
30	16	INSERT, HELICOIL			AMS 7245 (REF)	~	~	1.5 DIA 12
29	1	PACKING, PREFORMED	69249-35	.071	NITRILE (REF)	~	~	.210 DIA CROSS SECTION
28	32	BOLT, MACHINE - 12 POINT			A286 (REF)	~	~	
27	32	WASHER			AMS 5513, 5516, or 5515 (REF)	~	~	
26	1	PACKING, PREFORMED	69249-	.140	NITRILE (REF)	~	~	.275 DIA CROSS SECTION
25	1	PACKING, PREFORMED		.118	VITON (REF)	~	~	.275 DIA CROSS SECTION
24	1	RING, RETAINING		.180	AISI 302 (REF)	~	~	MEDIUM DUTY SPIRALOX (REF)
23	1	WASHER, TAB, FEATHER STOP		.16	AMS 6350 (REF)	~	~	
22	1	WASHER, KEY, P.L. SUPPORT NUT		.38	AMS 5513 75 SR/SN MAX		HS447	REF
21	1	WASHER, KEY, BULKHEAD NUT		.63	AMS 5513 75 SR/SN MAX		HS447	REF
20	1	WASHER, KEY, VALVE HSG. NUT		.22	AMS 5513 75 SR/SN MAX		HS447	REF
19	1	KEY			ALLOY STEEL	~	~	
18	1	RING, GLIDE		.085	15		~	REF 29
17	1	RING, GLIDE		.08	15		~	REF 28
16	1	RING, BACKUP		.025	HS 1205 GRA		~	(TEFLON) REF P/N 739138 THK: .062 ±.004
15	1	RING, BACKUP		.025	HS 1205 GRA		—	(TEFLON) REF P/N 739138
14	1	PACKING, PREFORMED (D SEAL)		.065	MIL-R-83248 CL I		~	SAME AS 718091-10 EXCEPT AS SHOWN
13	1	PACKING, PREFORMED (D SEAL)		.068	MIL-R-83248 CL I		~	SAME AS 718091-10 EXCEPT AS SHOWN
12	1	QUILL, ANTI-TORQUE		7.98	AMS 6415 36-40 HRC HS43			
11	1	NUT, BULKHEAD			AMS 6415 36-40 HRC HS43		SEE SHT 2	REF TS-164
10	1	NUT, PITCHLOCK SUPPORT		4.22	AMS 6415 36-40 HRC HS43			REF TS-164
9	1	NUT, FEATHER STOP		5.80	AMS 6415 36-40 HRC HS43			REF P/N 775770
8	1	NUT, VALVE HOUSING			AMS 6415 HS43			REF TS-164
7	1	NUT, PISTON RETAINING		4.7	AMS 6415			
6	1	CAP, DOME		9.8	32		HS447	13
5	1	DOME		22.5	32		HS447	13
4	1	SUPPORT, PITCHLOCK	ITEM 2 L-14325-12		AMS 6265 (REF)	~	—	
3	1	BULKHEAD		7.9	AMS 6415 36-40 HRC HS43		SEE SHT 2	
2	1	CYLINDER			AMS 6415 40-44 HRC HS43			
1	1	PISTON, STATIONARY			AMS 6415 40-44 HRC HS43			

UNLESS OTHERWISE SPECIFIED

UNITED TECHNOLOGIES

HAMILTON STANDARD DIVISION

LAP ACTUATOR (PF8760 REF)

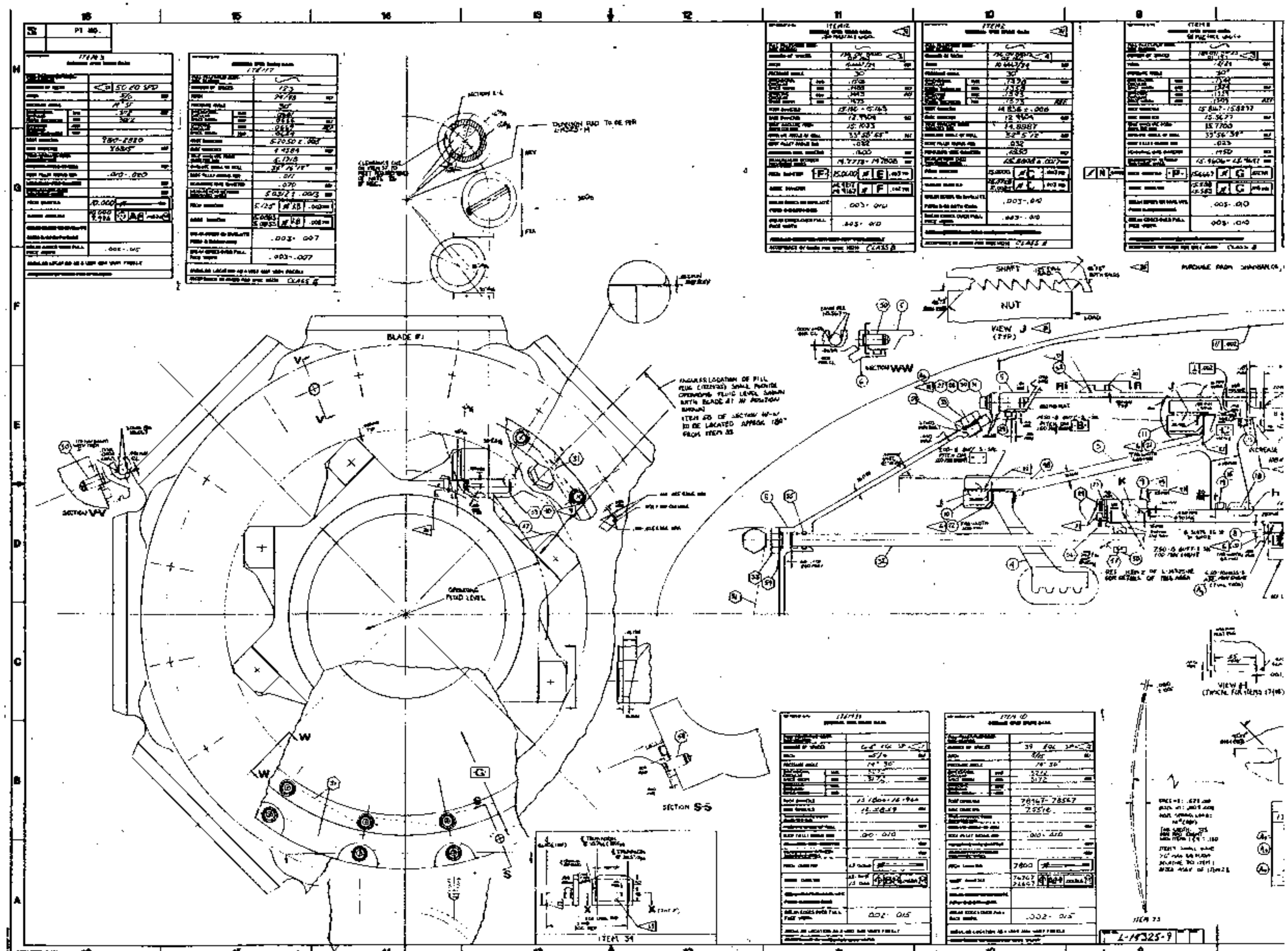
SIZE J | FSCM NO. 73030 | DRAWING NO. L-14325-9

SHEET 1 OF 2

L-14325-9

REV A

1-20-84


BLADE #1
SHAFT
NUT
SECTION S-S
ITEM 34

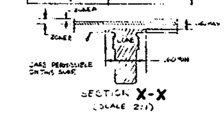


TABLE I (FOR ITEM 34)		
HARDNESS PATTERN (FINISHED MACHINED)		
AREA	RC	HARDENING DEPTH
ZONE A	59-64	.040 MIN
B	50 MIN	.050-.040
CORE	35-43	-----

L-14325-9

(A)

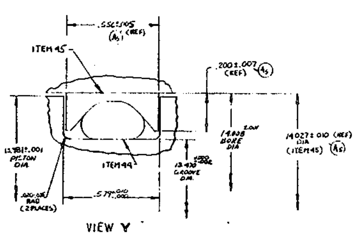
MPI TABLE FOR HS31 CL I		
ITEM 1	CENT. COND COIL	: 5000 AMPS : 1500 AMP TURNS
ITEM 2	CENT COND COIL	: 4100 AMPS : 4000 AMP TURNS
ITEM 3	CENT COND COIL	: 2200 AMPS : 1800 AMP TURNS
ITEM 7	CENT. COND DIRECT	: 7200 AMPS : 1900 AMPS
ITEM 8	CENT COND DIRECT	: 7200 AMPS : 450 AMPS
ITEM 9	CENT. COND DIRECT	: 8000 AMPS : 1750 AMPS
ITEM 10	CENT COND DIRECT	: 9500 AMPS : 1250 AMPS
ITEM 11	CENT COND DIRECT	: 8500 AMPS : 1800 AMPS
ITEM 12	CENT COND COIL	: 800 AMPS : 1400 AMP TURNS
ITEM 34	DIRECT COIL	: 1000 AMPS : 9000 AMP TURNS
ITEM 35	CENT COND DIRECT	: 1200 AMPS : 850 AMPS

L-14325-9 (A)

* 1.00 MIN THD ENGAGEMENT AT PITCH LINE	
ITEM #11	ITEM #2
EXTERNAL BUTTRESS THREAD DATA	INTERNAL BUTTRESS THREAD DATA
BASIC MAJOR DIA = 14.500	BASIC MAJOR DIA = 14.500
PITCH-8	PITCH-8
MAJOR DIA MAX 14.500-.0064 = 14.4936	MINOR DIA MAX 14.500-.1050 = 14.3950
MAJOR DIA MIN 14.500-.0084 = 14.4916	MINOR DIA MIN 14.500-.1080 = 14.3920
PITCH DIA * MAX 14.500-.0590 = 14.4410	PITCH DIA * MAX 14.500-.0462 = 14.4538
PITCH DIA * MIN 14.500-.0654 = 14.4346	PITCH DIA * MIN 14.500-.0526 = 14.4474
MIN MINOR DIA (REF) 14.500-.1650 = 14.335	MAX MAJOR DIA (REF) 14.500+.0498 = 14.5498
ROOT RADIUS .006-.013	ROOT RADIUS .0075-.0137
TRAILING FLANK ANGLE 45° +0°0' -0°30'	TRAILING FLANK ANGLE 45° +0°0' -0°30'
LEAD FLANK ANGLE 7° +0°0' -0°30'	LEAD FLANK ANGLE 7° +0°0' -0°30'
.003 MAX BREAK AT LEAD FLANK ANGLE INTERSECTION WITH MAJOR DIA	.003 MAX BREAK AT LEAD FLANK ANGLE INTERSECTION WITH MINOR DIA
63 ALL OVER EXCEPT MAJOR DIA 125	63 ALL OVER EXCEPT MINOR DIA 125
P.D. ⫽ MAJOR DIA .003	P.D. ⫽ MINOR DIA .003

* 1.00 MIN THD ENGAGEMENT AT PITCH LINE	
ITEM #10	ITEM #3
EXTERNAL BUTTRESS THREAD DATA	INTERNAL BUTTRESS THREAD DATA
BASIC MAJOR DIA 9.000	BASIC MAJOR DIA 9.000
PITCH-8	PITCH-8
MAJOR DIA MAX 9.000-.0064 = 8.9936	MINOR DIA MAX 9.000-.1050 = 8.8950
MAJOR DIA MIN 9.000-.0084 = 8.9916	MINOR DIA MIN 9.000-.1080 = 8.8920
PITCH DIA * MAX 9.000-.0590 = 8.9410	PITCH DIA * MAX 9.000-.0462 = 8.9538
PITCH DIA * MIN 9.000-.0654 = 8.9346	PITCH DIA * MIN 9.000-.0526 = 8.9474
MIN MINOR DIA (REF) 9.000-.1650 = 8.8350	MAX MAJOR DIA (REF) 9.000+.0498 = 9.0498
ROOT RADIUS .006-.013	ROOT RADIUS .0075-.0137
TRAILING FLANK ANGLE 45° +0°0' -0°30'	TRAILING FLANK ANGLE 45° +0°0' -0°30'
LEAD FLANK ANGLE 7° +0°0' -0°30'	LEAD FLANK ANGLE 7° +0°0' -0°30'
.003 MAX BREAK AT LEAD FLANK ANGLE INTERSECTION WITH MAJOR DIA	.003 MAX BREAK AT LEAD FLANK ANGLE INTERSECTION WITH MINOR DIA
63 ALL OVER EXCEPT MAJOR DIA 125	63 ALL OVER EXCEPT MINOR DIA 125
P.D. ⫽ MAJOR DIA .003	P.D. ⫽ MINOR DIA .003

* 1.00 MIN THD ENGAGEMENT AT PITCH LINE	
ITEM #1	ITEM #9
EXTERNAL BUTTRESS THREAD DATA	INTERNAL BUTTRESS THREAD DATA
BASIC MAJOR DIA 7.500	BASIC MAJOR DIA 7.500
PITCH-8	PITCH-8
MAJOR DIA MAX 7.500-.0064 = 7.4936	MINOR DIA MAX 7.500-.1050 = 7.3950
MAJOR DIA MIN 7.500-.0084 = 7.4916	MINOR DIA MIN 7.500-.1080 = 7.3920
PITCH DIA * MAX 7.500-.0590 = 7.4410	PITCH DIA * MAX 7.500-.0462 = 7.4538
PITCH DIA * MIN 7.500-.0654 = 7.4346	PITCH DIA * MIN 7.500-.0526 = 7.4474
MIN MINOR DIA (REF) 7.500-.1650 = 7.3350	MAX MAJOR DIA (REF) 7.500+.0498 = 7.5498
ROOT RADIUS .006-.013	ROOT RADIUS .0075-.0137
TRAILING FLANK ANGLE 45° +0°0' -0°30'	TRAILING FLANK ANGLE 45° +0°0' -0°30'
LEAD FLANK ANGLE 7° +0°0' -0°30'	LEAD FLANK ANGLE 7° +0°0' -0°30'
.003 MAX BREAK AT LEAD FLANK ANGLE INTERSECTION WITH MAJOR DIA	.003 MAX BREAK AT LEAD FLANK ANGLE INTERSECTION WITH MINOR DIA
63 ALL OVER EXCEPT MAJOR DIA 125	63 ALL OVER EXCEPT MINOR DIA 125
P.D. ⫽ MAJOR DIA .003	P.D. ⫽ MINOR DIA .003



HAMILTON STANDARD

LAP ACTUATOR (PF8T60 REF)

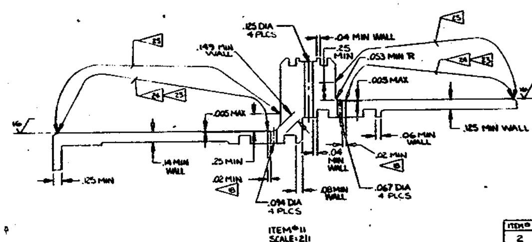
E 73030 L-14325-9

TABLE OF FITS	
ITEM #S	FIT
1 & 37	.005-.009 DC
11 & 18 (FWD)	.002-.008 DC
18 & 11 (AFT)	.002-.008 DC
L14325-9 #1 & 37 (AFT)	.001-.005 DC
L14325-9 #1 & 37 (FWD)	.0005-.005 RAD CLR
L14325-9 #1 & 1	.014-.018 DC
L14325-9 #1 & 12	.001 MIN RAD CLR
L14325-9 #1 & 1	.005 MIN RAD CLR
17 & 20 & 11	.001-.005
9 & 31 & 11	.001-.004
9 & 11	.010 MAX DC <11
9 & 19	.001-.004 DC
2 & 9	.001 MIN RAD CLR
9 & 20	.002-.008 DC
6 & 9	.0002-.0009 DC
11 & 12	.001 -.007 DC
2 & 3	.0187-.0277 DC
12 & 32	.005 MIN RAD CLR
1 & 32	.001-.005 DC
11 & 32	.005 MIN RAD CLR
12 & 20	.01 MIN RAD CLR
L14325-9 #1 & #11	.001 MIN RAD CLR
L14325-9 #1 & #21	.001 MIN RAD CLR

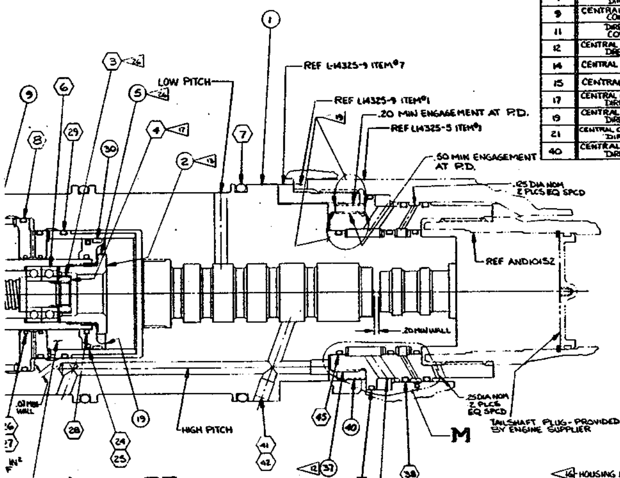
-WHEN ASSEMBLED IN ITEM #1

}WHEN ASSEMBLED IN ITEM #1

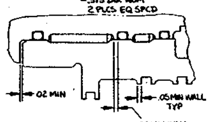
ITEM #	MPI TABLE PER HS31	CL
2	CENTRAL CONDUCTOR 11800 AMPS COIL :40000 AMP TURNS	I
4	CENTRAL CONDUCTOR : 800 AMPS DIRECT: 100 AMPS	I
9	CENTRAL CONDUCTOR : 2000 AMPS COIL : 22000 AMP TURNS	I
11	DIRECT :1000 AMPS COIL:23000 AMP TURNS	I
12	CENTRAL CONDUCTOR: 3600 AMPS DIRECT : 1100 AMPS	I
14	CENTRAL CONDUCTOR : 500 AMPS	I
15	CENTRAL CONDUCTOR :500 AMPS	II
17	CENTRAL CONDUCTOR :1800 AMPS DIRECT : 100 AMPS	I
19	CENTRAL CONDUCTOR :3600 AMPS DIRECT : 600 AMPS	I
21	CENTRAL CONDUCTOR : 6800 AMPS DIRECT: 1300 AMPS	II
40	CENTRAL CONDUCTOR :1800 AMPS DIRECT: 550 AMPS	II



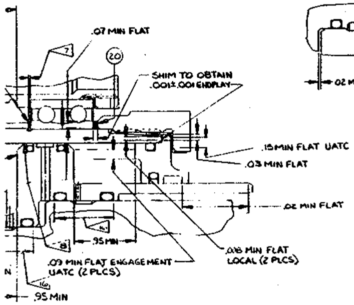
ITEM #11
SCALE: 2/1



SECTION E-E (D2-1)



VIEW M
SCALE: NONE



SECTION F-F (C12-1)
2:2/1

27. TORQUE RETAINING NUT UNTIL BELLEVILLE SPRINGS ARE COMPRESSED FLAT, THEN BACK OFF NUT 1/8 TURN
26. DEFORM CUP INTO SLOTS 2 PLCS EQ SPCD.
25. SHOT PEEN PER HS67, GRADE I SHOT PER HS293A, .012-.014A INTENSITY. .001 MAX STOCK REMOVAL AFTER PEENING
24. AFTER MEETING REQMTS OF <25 CHROME PLATE INDICATED SURF ONLY PER HS246 THICKNESS .0004±.0001.
23. CHROME PLATE MUST LIE ENTIRELY WITHIN PEENED AREA
22. WHEN PITCHLOCK SCREW (L14325-12 ITEM #1) IS UNSUPPORTED IN ACTUATOR THE MISALLIGNMENT OF BEARINGS (L14325-12 ITEM #8) FORWARD AND AFT TO BE .2 MAX.
21. INSTALL PER HS1652 CODE B
20. INSTALL PER HS1434
19. COAT INDICATED AREAS PER HS248 CL21 TYPE AI, OTHER AREAS OPTIONAL
18. SEAL MUST NOT RUN OFF CHROME PLATE
17. TORQUE TO 200-220 IN-LBS PER HS2212 CL III
16. HOUSING MUST WITHSTAND 1475 PSI IN DESIGNATED REGIONS AND ALL HYDRAULIC LINES FOR A PERIOD OF 15 MIN WITH NO VISIBLE LEAKAGE. TEST FLUID TO BE MIL-H-5606
15. DIMENSIONS APPLY WHEN BELLEVILLE SPRING IS COMPRESSED FLAT
14. THIS GAP TO BE .016-.028 WHEN PITCHLOCK GAP IS .028±.002
13. TORQUE TO 1550-1700 IN LBS PER HS2212 CL III
12. TORQUE TO 2300-2500 FT LBS PER HS2212 CL III
11. NITRIDE INDICATED SURFACE PER HS923 TYPE A2 SURF HARDNESS 1000 HV MIN, CORE HARDNESS 31-39 HRC, OTHER SURF OPTIONAL
10. ANODIZE PER HS334 TYPE I
9. TORQUE TO 65-75 IN LBS PER HS2212 CL III
8. ENGINEERING REQUIREMENTS: PISTON (ITEM 9) AND PISTON RING (ITEM 19) WHEN ASSEMBLED IN SLEEVE (ITEM 11) PER L/O MUST NEVER CONTACT SLEEVE (.0019 MIN RAD CLR) AT INDICATED AREA UNDER ALL TOLERANCE, TIPPING AND STROKING CONDITIONS
7. SHIM TO OBTAIN DIMENSION K WITH SURFACES H/J LINE ON LINE
6. 104±10 LBS LOAD AT COMPRESSED HEIGHT OF .045 IN.
5. MACHINE ITEMS 30 & 31 O.D. AS PART OF ITEM 9 & 19 TO OBTAIN INDICATED FIT WITH ITEM #11
4. NO BOND SEPARATION PERMISSIBLE AFTER MACHINING
3. BOND PER HS1265 CODE IC2 HT WITH SURFACE PREP ON ITEM #30 & 31 TO BE METHOD P, AND ITEM #9/19 TO BE METHOD V. BONDING MUST TAKE PLACE WITHIN 4 HOURS OF ABRASION
2. NO FLAWS PERMISSIBLE UNDER VISUAL EXAMINATION
1. THE PURPOSE OF THIS LAYOUT IS TO DEFINE THE SERVO PISTON AND THE BALLSCREW RETENTION FOR THE LAP PROPELLER

NOTES

ITEM NO	NO. REQ	PART NAME	PART NUMBER		MATERIAL AND HEAT TREAT	MPI	NON DEST TESTING	REMARKS
47	2	PIN, TAPERED-LONG	63641-4A		2024-T4 ALUM (REF)			<30
46	AR	SHIM-SOLID			AMS5519 PER HS178			REF 732081, 372-423 VH
46	1	KEY			HS279 (REF)	~	~	
45	3	PACKING, PREFORMED			VITON (REF)	~	~	.139 NOM CROSS SECTION
44	2	PLUG, EXPANDABLE-LONG	63662-3A		2024-T4 ALUM (REF)			<30
43	2	PIN, TAPERED-LONG	63661-3A		2024-T4 ALUM (REF)			<30
42	9	PLUG, EXPANDABLE-LONG	63662-5A		2024-T4 ALUM (REF)			<30
41	9	PIN, TAPERED-LONG	63661-5A		2024-T4 ALUM (REF)			<30
40	1	RING, ANTI-ROTATION			AMS6415 HT HS43 32-38 HRC	CR	SEE TABLE	REF 558264
39	1	PACKING, PREFORMED			VITON (REF)	~	~	.139 NOM CROSS SECTION
38	2	PACKING, PREFORMED			VITON (REF)	~	~	.139 NOM CROSS SECTION
37	1	NUT, PROPELLER HUB			MAKE FROM C730794 (REF)	~	~	REF 750794
36	2	PLUG, EXPANDABLE-LONG	63662-4A		2024-T4 ALUM (REF)			<30
35	6	WASHER, FLAT			AMS5525	~	~	REF .25 ID
34	6	INSERT, HELICAL COIL			AMS7245 (REF)			1.5D, SELF LOCKING <21
33	2	PACKING, PREFORMED			VITON (REF)	~	~	.070 NOM CROSS SECTION
32	1	TUBE, TRANSFER			AMS4083 HS454	CR		<10
31	1	RING, GLIDE			HS1401 GR D (RULON LD)	CR		REF 772532
30	1	RING, GLIDE			HS1401 GR D (RULON LD)	CR		REF 772532
29	2	PACKING, PREFORMED			VITON (REF)	~	~	.139 NOM CROSS SECTION
28	1	PACKING, PREFORMED			VITON (REF)	~	~	.103 NOM CROSS SECTION
27	2	RING, SEAL-INNER			HS1401 GR A (GLASS)	CR		
26	2	PACKING, PREFORMED			VITON (REF)	~	~	.103 NOM CROSS SECTION
25	1	RING, SEAL-OUTER			HS1401 GR A (GLASS)	CR		
24	1	PACKING, PREFORMED			VITON (REF)	~	~	.139 NOM CROSS SECTION
23	1	RING, SEAL-OUTER			HS1401 GR A (GLASS)	CR		
22	1	PACKING, PREFORMED			VITON (REF)	~	~	.139 NOM CROSS SECTION
21	1	RETAINER, SCREW-PITCHLOCK			AMS6415 HT HS43 32-38 HRC	CR	SEE TABLE	
20	AR	SHIM-SOLID			AMS5519 PER HS178			REF 732081, 372-423 VH
19	1	RING, PISTON			AMS6415 HT HS43 36-40 HRC	CR	SEE TABLE	REF 778755
18	1	BALLSCREW-SERVO	782748-1		~			USE 775715-1 AS AN ALTERNATE
17	1	NUT, RETAINING			AMS6415 (REF)	~	~	
16	1	LOCKING CUP, INT TANG			AMS6350 (REF)	~	~	REF TS1164
15	1	SPACER, RING			AMS6440 58-62 HRC	~	SEE TABLE	SAME AS 732199-1 EXCEPT AS SHOWN
14	3	SPRING, BELLEVILLE			AMS5121 HT HS43 (REF)	CR	SEE TABLE	REF 778792
13	6	BOLT, MACHINE			AMS5731 (A286) REF	~	~	.25-28 UNJF-3A
12	1	STOP, PITCHLOCK			HS279 <11	CR	SEE TABLE	REF 778807
11	1	SLEEVE, PISTON			AMS6415 HT HS43 36-40 HRC	CR	SEE TABLE	REF 778754
10	1	WASHER, SPHERICAL	778756-1		HS279 (REF)	~	~	
9	1	PISTON, SERVO			AMS6415 HT HS43 36-40 HRC	CR	SEE TABLE	REF 778754
8	2	PACKING, PREFORMED			VITON (REF)	~	~	.139 NOM CROSS SECTION
7	3	PACKING, PREFORMED			VITON (REF)	~	~	.215 NOM CROSS SECTION
6	1	BEARING, BALL, DUPLEX	775718-2		52100 (REF)	~	~	
5	1	LOCKING CUP, INT TANG			AMS6350 HT HS43	CR	SEE TABLE	
4	1	NUT, RETAINING	778797-1		AMS6415 (REF)	~	~	
3	1	LOCKING CUP, INT TANG	788380		AMS6350 (REF)	~	~	PEF TS1164
2	1	NUT, PISTON			AMS6415 HT HS43 40-44 HRC	CR	SEE TABLE	REF 782502
1	1	HOUSING, PITCHLOCK			AMS4122 HS454	CR	HS447	<10

L-14325-10

REVISIONS

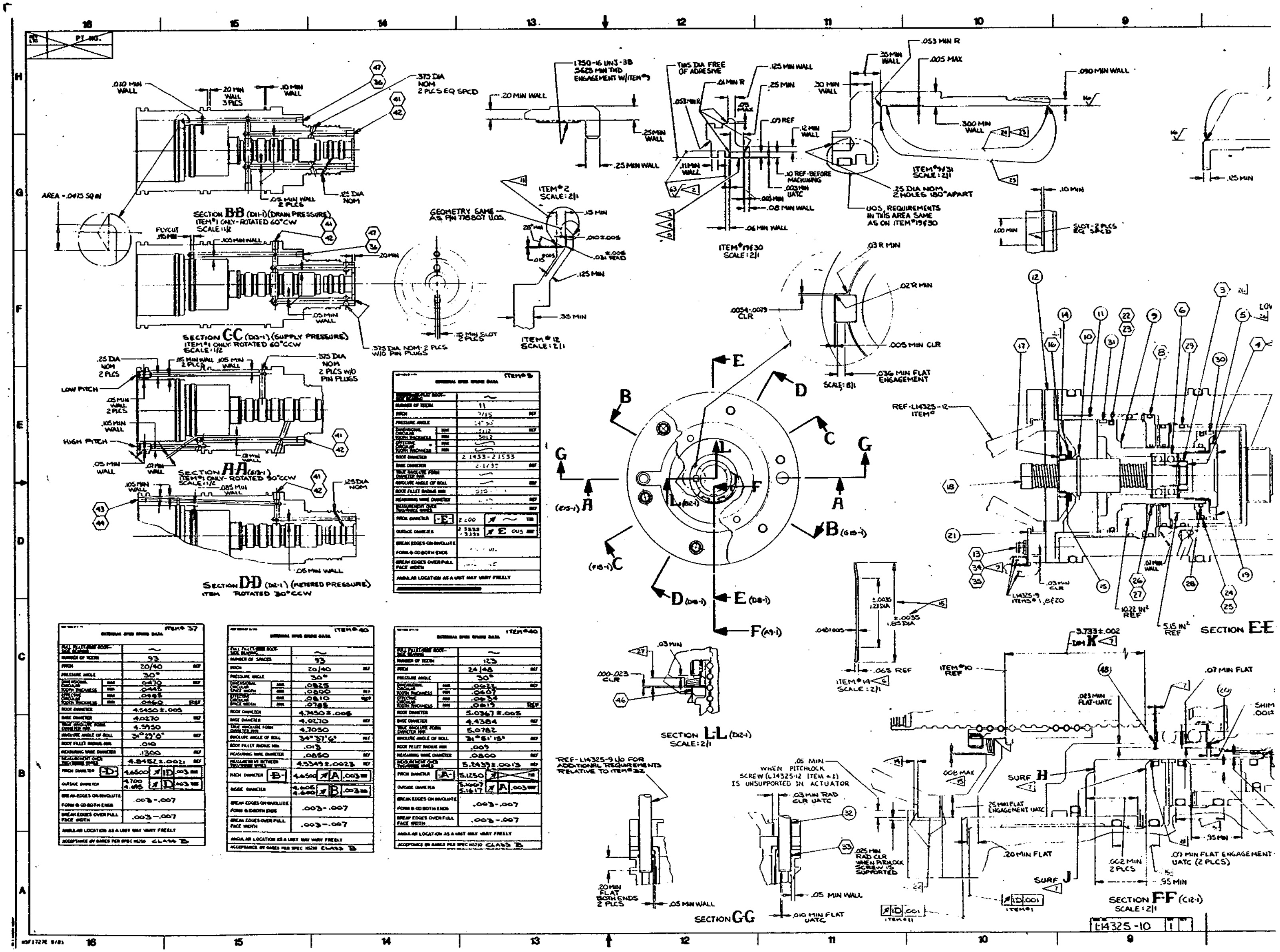
UNITED TECHNOLOGIES

HSG & PISTON, PITCHLOCK-LAP

SIZE J | FSCM NO. 73030 | DRAWING NO. L-14325-10

SCALE 1/1 | SHEET 1 OF 2

PROGRAM LAP


SECTION B-B (D11-1) (DRAIN PRESSURE)
ITEM#1 ONLY- ROTATED 60°CCW
SCALE: 1/1
SECTION C-C (D13-1) (SUPPLY PRESSURE)
ITEM#1 ONLY- ROTATED 60°CCW
SCALE: 1/2
SECTION A-A (E13-1)
ITEM#1 ONLY- ROTATED 30°CCW
SCALE: 1/1
SECTION D-D (D12-1) (METERED PRESSURE)
ITEM ROTATED 30°CCW
AREA = .0415 SQ IN
FLYCUT
LOW PITCH
HIGH PITCH
.05 MIN WALL
.375 DIA NOM 2 PLCS EQ SPCD
.125 DIA NOM
.375 DIA NOM-2 PLCS W/O PIN PLUGS
.1750-16 UNJ-3B .5625 MIN THD ENGAGEMENT W/ITEM#9
.20 MIN WALL
.25 MIN WALL
GEOMETRY SAME AS PN 778807 U.O.S.
ITEM#2 SCALE: 2/1
ITEM#12 SCALE: 2/1
THIS DIA FREE OF ADHESIVE
.125 MIN WALL
.10 REF - BEFORE MACHINING
ITEM#19630 SCALE: 2/1
.25 DIA NOM 2 HOLES 180° APART
U.O.S. REQUIREMENTS IN THIS AREA SAME AS ON ITEM#19630
ITEM#19131 SCALE: 2/1
.053 MIN R
.005 MAX
.090 MIN WALL
.300 MIN WALL
SLOT - 2 PLCS EQ SPCD
.0054-.0073 CLR
.03 R MIN
.02 R MIN
.005 MIN CLR
.036 MIN FLAT ENGAGEMENT
SCALE: 8/1
REF-L14325-12 ITEM#
SECTION E-E
L14325-9 ITEMS#1, 8 & 20
10.22 IN² REF
5.15 IN² REF
ITEM#14 SCALE: 2/1
.065 REF
SECTION L-L (D12-1) SCALE: 2/1
.000-.023 CLR
REF-L14325-9,10 FOR ADDITIONAL REQUIREMENTS RELATIVE TO ITEM#32
.05 MIN WHEN PITCHLOCK SCREW (L14325-12 ITEM#1) IS UNSUPPORTED IN ACTUATOR
.03 MIN RAD CLR UATC
.025 MIN RAD CLR WHEN PITCHLOCK SCREW IS SUPPORTED
.20 MIN FLAT BOTH ENDS 2 PLCS
.05 MIN WALL
.010 MIN FLAT UATC
SECTION G-G
ITEM#10 REF
3.733±.002
.07 MIN FLAT
.023 MIN FLAT-UATC
SURF H
.25 MIN FLAT ENGAGEMENT UATC
.20 MIN FLAT
.062 MIN 2 PLCS
.95 MIN
.03 MIN FLAT ENGAGEMENT UATC (2 PLCS)
SURF J
SECTION F-F (C12-1) SCALE: 2/1
ITEM#5
ITEM#37
INTERNAL SPUR SPLINE DATA
NUMBER OF TEETH 93
PITCH 20/40
PRESSURE ANGLE 30°
.0470 .0445 .0485 .0460
ROOT DIAMETER 4.5450±.005
BASE DIAMETER 4.0270
TRUE INVOLUTE FORM DIAMETER 4.5950
INVOLUTE ANGLE OF ROLL 31°23'8"
ROOT FILLET RADIUS MIN .010
MEASURING WIRE DIAMETER .1300
4.8452±.0021
PITCH DIAMETER -D- 4.6500 .003
OUTSIDE DIAMETER 4.700 4.695 .003
BREAK EDGES ON INVOLUTE FORM & OD BOTH ENDS .003-.007
BREAK EDGES OVER FULL FACE WIDTH .003-.007
ANGULAR LOCATION AS A UNIT MAY VARY FREELY
ACCEPTANCE BY GAGES PER SPEC HS210 CLASS B
ITEM#40
NUMBER OF SPACES 93
PITCH 20/40
PRESSURE ANGLE 30°
.0825 .0800 .0810 .0785
ROOT DIAMETER 4.7450±.005
BASE DIAMETER 4.0270
TRUE INVOLUTE FORM DIAMETER 4.7050
INVOLUTE ANGLE OF ROLL 34°37'6"
ROOT FILLET RADIUS MIN .013
MEASURING WIRE DIAMETER .0850
MEASUREMENT BETWEEN TWO WIRES 4.5349±.0023
PITCH DIAMETER -B- 4.6500 A .003
INSIDE DIAMETER 4.606 4.600 B .003
ITEM#40
NUMBER OF TEETH 123
PITCH 24/48
PRESSURE ANGLE 30°
.0623 .0607 .0634 .0619
ROOT DIAMETER 5.0367±.005
BASE DIAMETER 4.4384
TRUE INVOLUTE FORM DIAMETER 5.0782
INVOLUTE ANGLE OF ROLL 31°51'15"
ROOT FILLET RADIUS MIN .009
MEASURING WIRE DIAMETER .0800
MEASUREMENT OVER TWO WIRES 5.2439±.0013
PITCH DIAMETER -A- 5.1250
OUTSIDE DIAMETER 5.1667 5.1617 A .003
L14325-10

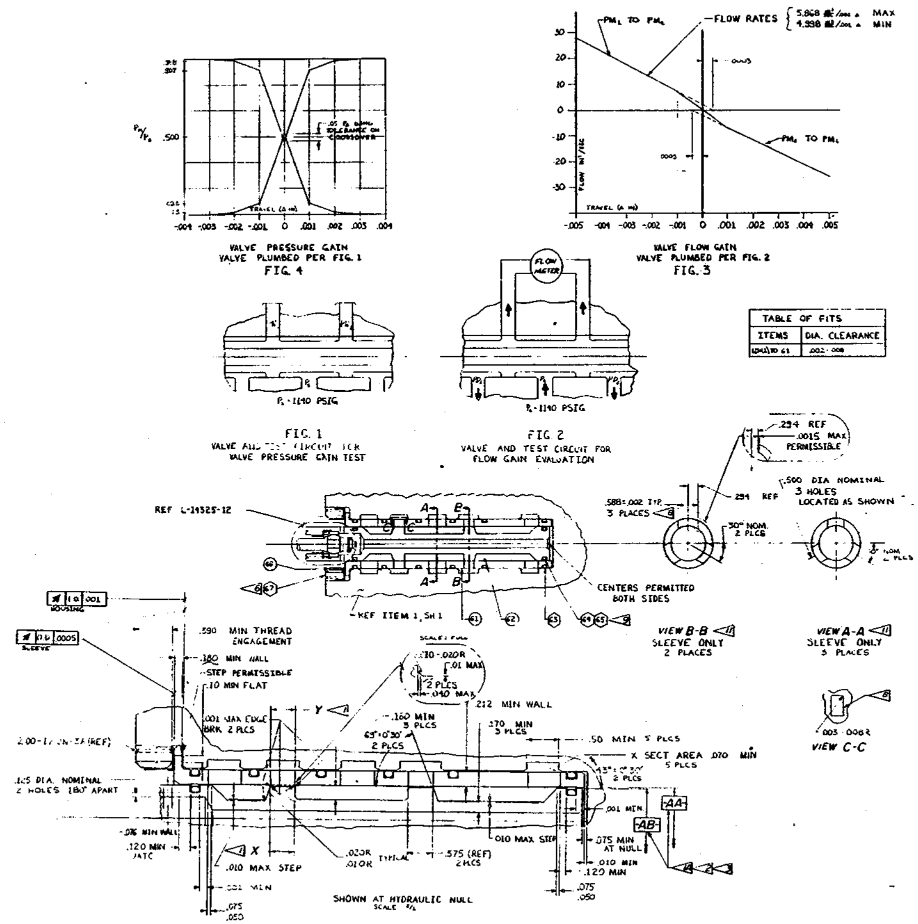


TABLE OF FITS	
ITEMS	DIA. CLEARANCE
62 TO 61	.002-.008

11. ANGULAR LOCATION AS A GROUP (5 GROUPS) TO BE WITHIN ±2°
10. MPI PER HS31, CL II, CURRENT APPLICATION: DIRECT 800 AMPS COIL 8000 AMP TURNS
9. SEALS MUST BE CHILLED FOR 10 MINUTES MINIMUM, IN DRY ICE, WITH SEAL COMPRESSORS IN PLACE, PRIOR TO INSTALLATION IN BORES
8. EDM PERMITTED IN THIS AREA
7. MPI PER HS31, CL II, CURRENT APPLICATION: CENTRAL CONDUCTOR 1000 AMP, COIL 8000 AMP TURNS
6. TORQUE TO 50-60 IN-LBS PER HS2212 CL II
5. ROUNDNESS MUST BE WITHIN .0001 RADIAL TOLERANCE BAND, STRAIGHTNESS AND TAPER MUST BE WITHIN .0001 RADIAL TOLERANCE BAND AND MUST ALSO SATISFY MATCHED SET REQUIREMENTS.
4. MARK SAME SERIAL NUMBER AND MATCHED SET
3. INSPECT PER HS 972
2. CARBURIZE AND HARDEN INCLUDED SURFACES PER HS 103, GROUP 'C'; SURFACE HARDNESS: CASE 90 HR 15N MIN; CORE HARDNESS: 35-43 HR_c, DEPTH OF FINISH CASE: .010-.020
1. VALVE ASSEMBLY TO MATCH LAND WIDTHS X TO PORT WIDTHS Y AND MATING DIAMETERS TO MEET THE FOLLOWING REQUIREMENTS:
   A — SURFACE ROUGHNESS ON DIAS AA AND AB TO BE 5√
   B — MIN DIAMETRAL CLEARANCE TO BE .0003, MAX TO BE .0006
   C — VALVE, WET WITH MIL-H-5606 SHALL SLIDE THROUGH BORE UNDER ITS OWN WEIGHT REGARDLESS OF ROTARY POSITION
   D — CONNECT VALVE IN TEST CIRCUIT AS SHOWN IN FIG. 2. WHEN VALVE IS TRAVERSED FROM LEFT TO RIGHT, THE FLOW VERSUS TRAVEL CURVE SHALL BE WITHIN THE TOLERANCE SPECIFIED IN FIG 3. ALSO, WITH VALVE PLUMBED AS IN FIG 1, THE PRESSURE VERSUS TRAVEL CURVE SHALL BE WITHIN THE SHADED AREA OF FIG 4 REGARDLESS OF ROTATIONAL POSITION. USE MIL-H-5606 FLUID AT 80° ± 10° F.
   E — FLOW MAPPING SHALL BE DONE AT P_s = 1140 PSIG.

L14325-10

THE PURPOSE OF THIS LAYOUT IS TO DEFINE THE BETA VALVE ON THE LAP PROPELLER SYSTEM

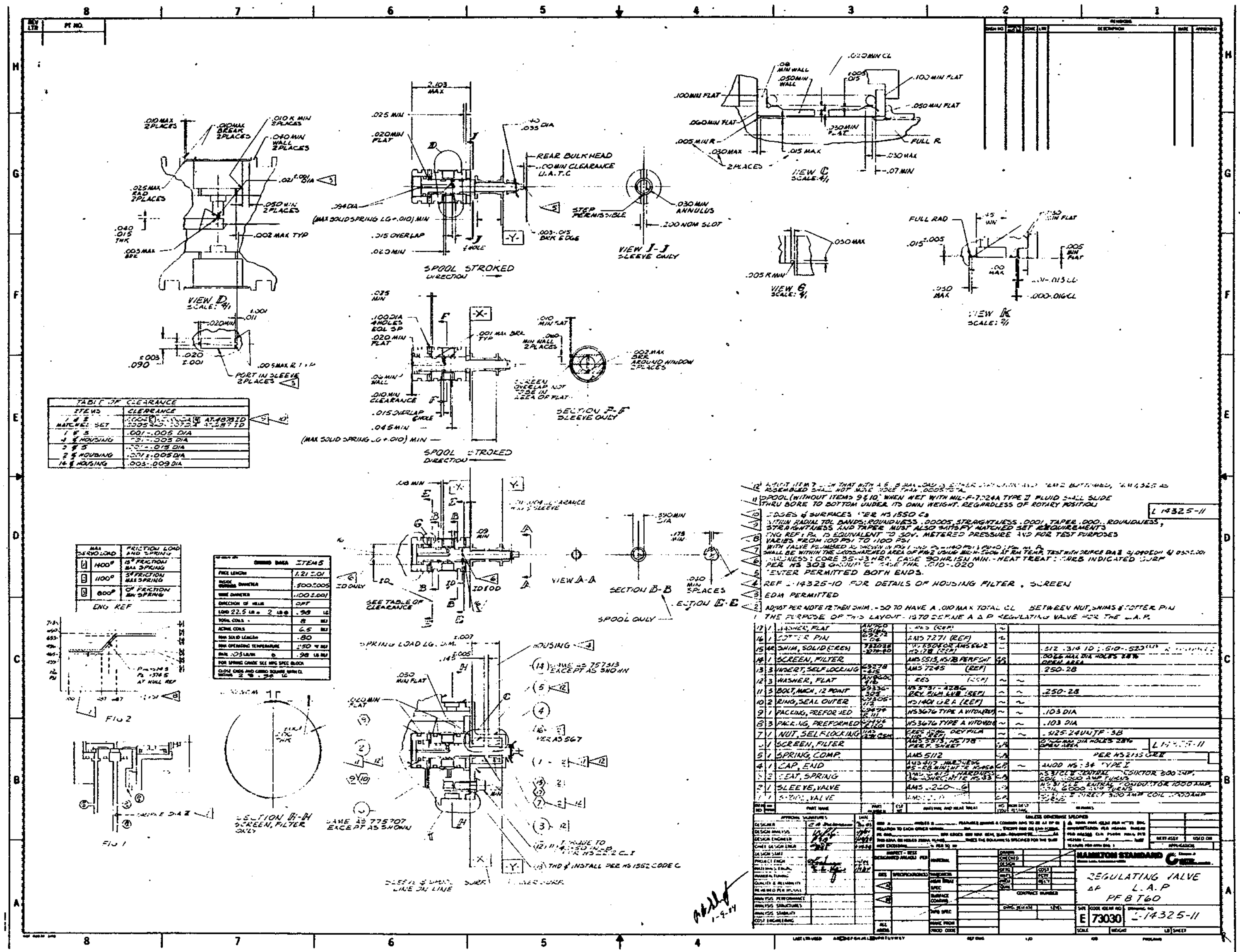
ITEM NO	QTY	PART NAME	PART NUMBER		MATERIAL AND HEAT TREAT			REMARKS
67	1	NUT, RETAINING	749808-5		(REF) AMS 4117	CIPE		
66	1	WASHER, KEY			AMS 5513 (REF)			REF 714811 L14325-10
65	2	RING, SEAL, OUTER	61296-117		HS 1401 GRADE B (REF)	CIPE		[illegible]
64	2	PACKING, PREFORMED	699R136		HS 5676 TYPE A VITON (REF)			.103 DIA CROSS SECTION
63	6	PACKING, PREFORMED	699R125		HS 5676 TYPE A VITON (REF)			.103 DIA CROSS SECTION
62	1	SPOOL, VALVE			AMS 6260	CIPE		REF 765956, 762261
61	1	SLEEVE, VALVE			AMS 6260	CIPE		REF 765956, 762261

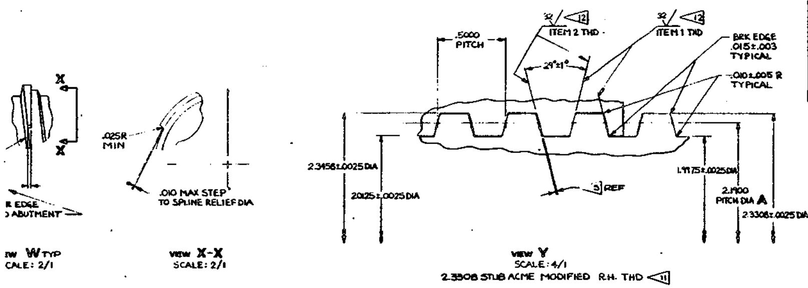
HAMILTON STANDARD

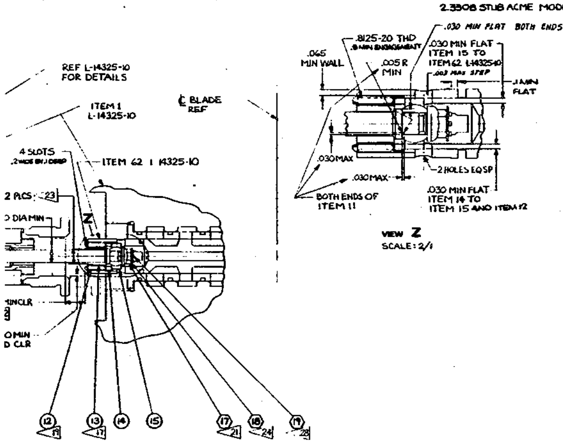
LAYOUT
LAP BETA VALVE
(PF8760)

SIZE E | CODE IDENT NO 73030 | DRAWING NO L 14325-10

SCALE 1/1 | SHEET 2/2


VIEW D
SCALE: 4/1
SPOOL STROKED DIRECTION
VIEW J-J
SLEEVE ONLY
VIEW C
SCALE: 4/1
VIEW G
SCALE: 4/1
VIEW K
SCALE: 4/1
SECTION F-F
SLEEVE ONLY
REAR BULKHEAD
TABLE OF CLEARANCE
ENG REF
FIG 2
FIG 1
SECTION H-H
SCREEN, FILTER ONLY
SAME AS 775707 EXCEPT AS SHOWN
VIEW A-A
SECTION B-B
SPOOL ONLY
SECTION E-E
HOUSING
SLEEVE & SPOOL LINE ON LINE
L 14325-11
11 SPOOL (WITHOUT ITEMS 9 & 10) WHEN WET WITH MIL-F-7024A TYPE II FLUID SHALL SLIDE THRU BORE TO BOTTOM UNDER ITS OWN WEIGHT, REGARDLESS OF ROTARY POSITION
10 EDGES & SURFACES PER HS 1550 Cs
9 WITHIN RADIAL TOL BANDS: ROUNDNESS .00005, STRAIGHTNESS .0001, TAPER .000. ROUNDNESS, STRAIGHTNESS AND TAPER MUST ALSO SATISFY MATCHED SET REQUIREMENTS
8 ENG REF: Pa IS EQUIVALENT TO SOV. METERED PRESSURE AND FOR TEST PURPOSES VARIES FROM 100 PSI TO 1100 PSI
6 HARDNESS: CORE 35-43 HRC. CASE 90 HR15N MIN. - HEAT TREAT PER HS 303 GROUP "C" CASE THK .010-.020
5 CENTER PERMITTED BOTH ENDS.
4 REF L-14325-10 FOR DETAILS OF HOUSING FILTER, SCREEN
3 EDM PERMITTED
2 ADJUST PER NOTE 12 THEN SHIM .50 TO HAVE A .010 MAX TOTAL CL BETWEEN NUT, SHIMS & COTTER PIN
1 THE PURPOSE OF THIS LAYOUT IS TO DEFINE A Δ P REGULATING VALVE FOR THE L.A.P.
17 1 WASHER, FLAT
16 1 COTTER PIN AMS 7271 (REF)
15 AR SHIM, SOLID (CRES)
14 1 SCREEN, FILTER AMS 5513, HS178 PERF SHT .0026 MAX DIA HOLES 28% OPEN AREA
13 3 INSERT, SELF LOCKING AMS 7245 (REF) .250-28
12 3 WASHER, FLAT
11 3 BOLT, MACH, 12 POINT DRY FILM LUB (REF) .250-28
10 2 RING, SEAL OUTER HS1401 GR A (REF)
9 1 PACKING, PREFORMED HS3676 TYPE A VITON (REF) .103 DIA
8 3 PACKING, PREFORMED HS3676 TYPE A VITON (REF) .103 DIA
7 1 NUT, SELF LOCKING .4125-24UNJF-3B
6 1 SCREEN, FILTER AMS 5513, HS178 PERF. SHEET
5 1 SPRING, COMP. AMS 5112 PER HS2135 GR II
4 1 CAP, END ANOD HS 134 TYPE I
3 2 SEAT, SPRING
2 1 SLEEVE, VALVE
1 1 SPOOL, VALVE
PART NAME
MATERIAL AND HEAT TREAT
REMARKS
APPROVAL SIGNATURES
DESIGNER
DESIGN ANALYSIS
DESIGN ENGINEER
UNLESS OTHERWISE SPECIFIED
HAMILTON STANDARD
REGULATING VALVE
ΔP L.A.P
PF 8 T60
E 73030 L-14325-11





MPI PER HS31

ITEM NO	HS31 CLASS	CURRENT APPLICATION: CENTRAL CONDUCTOR AMPS	DIRECT AMPS	COIL AMP TURNS
1	III	~	3000	9000
2	III	4000	~	11000
3	II	9500	~	11000
6	II	1200	~	28000
7	II	900	~	23000
9	II	1000	300	~
11	II	~	150	8000
12	II	900	200	~
20	I	300	~	8000
4	II	1300	250	~

TABLE OF FITS

ITEMS	FIT
1 AND 2	5
1 AND 6	.002 MIN RADIAL CLR
1 AND 10	.0005-.004 WIDTH CLR
1 AND 3	.004 MAX AXIAL BACKLASH
2 & (3) L-14325-9	.001-.011 DIA CLR
2 & (2) L-14325-10	.50-.59 DIA CLR
6 AND 7	.002 MIN DIA CLR
6 AND 10	.0005-.004 WIDTH CLR
6 AND 15	.0001-.0026 DIA CLR
11 AND 15	.0001-.0026 DIA CLR
23 & (3) L-14325-9	.010 MIN DIA CLR
14 & (15) L-14325-10	.002 MIN DIA CLR
15 & (15) L-14325-10	.0001-.0026 DIA CLR

REVISIONS

32 INDICATED LIMITS APPLY UATC WHEN ITEM 6 IS ADJUSTED TO POSITION ITEM 62 OF L-14325-10 FOR 0 FLOW $P_{in} \cong P_{out}$ WITH PITCHLOCK GAP SET AT .028 ± .002

31 AT MAX STROKE OF ITEM 18 L-14325-10

30 .4 MIN WALL OVER INDICATED LENGTH

29 .080 MIN RAD CLR AT MAX OPERATING ALIGNMENT

28 INSTALL PER AS567

27 PITCH DIA OF THREAD IN ITEM 3 IS DATUM B

26 BEARING REQUIREMENTS: (ENG REF)
NORMAL OPERATING LOAD - ± 15 LB
MAX AXIAL ENDPLAY (TOTAL) AT ± 50 LB AXIAL LOAD
MAX AXIAL MISALIGNMENT ± 0°15'
OPERATING ENVIRONMENT -65°F TO +250°F, AIR/OIL (MIL-H-5606) MIST
MAX FRICTION TORQUE 1.5 IN-LB AT 0 AXIAL LOAD
LIFE: WEAR SHALL NOT EXCEED .001 AXIAL IN 1000 HRS WHEN THE BEARING IS MISALIGNED 0°13' (± 0°2' PER SEC) AND SUBJECTED TO ONE CYCLE PER MINUTE AS FOLLOWS; ± 15 LB AXIAL LOAD WHILE OSCILLATING ± 1.5 REVOLUTIONS

25 TORQUE 40-50 IN-LB PER HS2212 CL III

24 SHIM FOR .005 MAX AXIAL GAP BETWEEN ITEM 17 & ITEM 18, 2 PLCS, .010 MIN SHIM REQ'D

23 THIS DIA MUST CLEAR INSIDE DIA OF ITEM 20 AND INSIDE DIA OF ITEM 18 ON L-14325-10 FOR ASSEMBLY ONLY

22 THIS CLR WITH BETA VALVE AT NULL AND ACTUATOR AT MAX FEATHER (90° β 3/4)

21 TORQUE 40-50 IN-LB ABOVE SELF LOCKING TORQUE PER HS2212 CL III

20 MAX ALLOWABLE MISALIGNMENT ±0°21' FOR SPLINE ENGAGEMENT

19 TORQUE 20-30 IN-LB PER HS2212 CL III

18 MIN CLR AT MAX OPERATING ALIGNMENT

17 DEFORM CUP INTO 2 SLOTS EQ SPACED, LOCK CUPS PER TS1164

16 CHROME PLATE PER HS246 .0002-.0004 THK

15 DRY FILM LUBE PER HS248 CL40 TYPE A7 VAPOR BLAST PRE TREATMENT TO BE PER HS1492 CL I

14 NITRIDE INDICATED SURFACES PER HS923 TYPE A2. SURF HARDNESS: 1000 VH MIN, CORE HARDNESS: 32-38 HRC *OTHER SURF OPT.*

13 MIN CLR WITH ITEM 2 STROKED TO 90° β 3/4 (FEATHER POSITION) AND ITEM 3 SET AT -10° β 3/4 (LOW PITCH STOP)

12 NO FLAWS EXCEEDING DEPTH OF SPECIFIED ROUGHNESS

11 REMOVE START OF THREAD, BOTH ENDS, TO GIVE 1/4 TO 1/2 THD ABUTMENT (THK) (REF 782280)

10 THIS REQUIREMENT IS WHEN LOW PITCH STOP IS ENGAGED ONE REVOLUTION AFTER CONDITION IN NOTE 9

9 WHEN SURF "N" ON ITEM 2 AND SURF "L" ON ITEM 3 ARE IN THE SAME RADIAL PLANE AND ONE REVOLUTION BEFORE CONTACTING EACH OTHER, THE AXIAL GAP BETWEEN SURF "J" ON ITEM 2 AND SURF "K" ON ITEM 3 SHALL BE .200 ± .050 SEE VIEW P (H5)

8 ITEMS 2 & 3 AND ITEM 3 OF L-14325-9 ARE SHOWN AT 38.6° β 3/4

7 6 SLOTS IN ITEM 1 AND 5 SLOTS IN ITEM 6, FOR .25 WIDE KEY, REF: MAX PITCHLOCK GAP ADJUSTMENT .001

6 CARBURIZE AND HARDEN INCLUDED SURFACES ONLY PER HS303, GROUP C: SURFACE HARDNESS CASE 90 HR15N MIN, CORE HARDNESS 35-43 HRC, DEPTH OF FINISHED CASE .025 ± .005

5 LAP FIT SCREW (ITEM 1) INTO SUPPORT (ITEM 2). WITH THD LUBRICATED WITH MIL-H-5606 FLUID AT R.T. SET MUST MEET THE FOLLOWING REQUIREMENTS OVER INDICATED LENGTH OF SCREW
M A) MUST NOT EXCEED 5 IN-LB TORQUE WITH A ±15 LB AXIAL LOAD
M B) IN FREE STATE, AXIAL BACKLASH SHALL NOT EXCEED .002

4 ITEMS 1, 2 & 3 ARE A MATCHED SET

3 SEE MPI DATA CHART ZONE (H4)

2 SURFACE TEMP INSPECTION PER HS172

1 - THE PURPOSE OF THIS LAYOUT IS TO DEFINE THE PITCHLOCK MECHANISM AND LOW PITCH STOP FOR THE LAP PF8T60.

NOTES

ITEM NO	QTY	PART NAME	PART NUMBER		MATERIAL AND HEAT TREAT			REMARKS
23	AR	SHIM - SOLID			AMS5519 79.5-82 R15N	~	~	732110 REF
22	1	SCREW, PITCHLOCK-SET		--		CIPS	~	ASSEMBLY OF ITEMS 1, 2 & 3
21	1	RING, RETAINING	69835-75		AISI 302 REF	~	~	
20	1	QUILL, BALLSCREW			AMS 5646 HS178	CIPS	3	
19	2	PIN, COTTER	69272-04		AMS 7211 REF	~	~	
18	AR	SHIM - SOLID	732025		AMS5504 or AMS5512, 5518 REF	~	~	.314-.312 I.D. .530-.510 O.D
17	2	NUT, SELF-LOCKING	NAS1291C5M		CRES A-286 DRY FILM LUBE REF	~	~	.3125-24 UNJF-3B THD
16								
15	2	BEARING, CONNECTING ROD				~	~	VENDOR ITEM 26
14	1	RING, SPACER			AMS6415 36-40 HRC PER HS43	CIPS	~	
13	1	LOCKING CUP, EXT. TANG			AMS5513 75.5 HR15	CIPS	HS467	REF 748839 .017-.028 THK
12	1	NUT, RETAINING			AMS6415 36-40 HRC PER HS43	CIPS	3	
11	1	ROD, CONNECTING - β VALVE			AMS6415 36-40 HRC PER HS43	CIPS	3	
10	1	KEY, MACHINE - SQUARE				~	~	
9	1	NUT, RETAINING			AMS6415 36-40 HRC PER HS43	CIPS	3	1.25-16 THD
8	1	LOCKING CUP, EXT. TANG			AMS5513 75.5 HR15N MAX	CIPS	HS467	REF 748839 .017-.028 THK
7	1	RETAINER, BEARING			AMS6415 36-40 HRC PER HS43	CIPS	3	
6	1	SLEEVE, ADJUSTING			AMS6415 36-40 HRC PER HS43	CIPS	3	
5	1	RING, RETAINER - INT.	69835-275		AISI 302 REF	~	~	
4	1	RING, LOCKING			AMS6415 36-40 HRC PER HS43	CIPS	~	
3	1	STOP, LOW PITCH			AMS6415 36-40 HRC PER HS43	CIPS	3	4, 11
2	1	SUPPORT, PITCHLOCK			AMS6265	CIPS	2	3, 4, 6, 6, 14
1	1	SCREW, PITCHLOCK			AMS6265	CIPS	2	3, 4, 5, 6, 11 REF 782280

L-14325-12

UNLESS OTHERWISE SPECIFIED

APPROVAL SIGNATURES

ANALYSIS PERFORMANCE

ANALYSIS STRUCTURES

ANALYSIS STABILITY

COST ENGINEERING

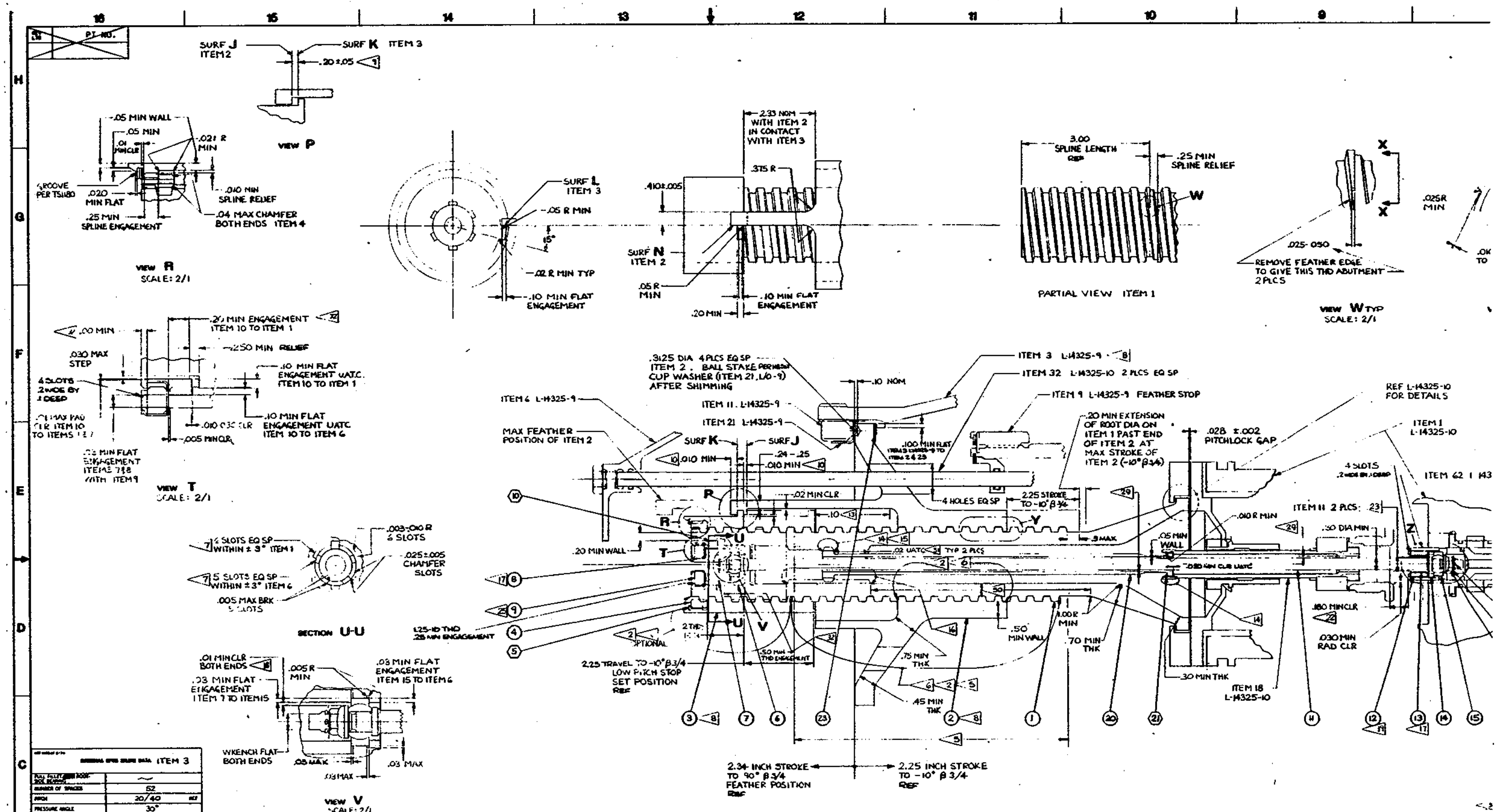
UNITED TECHNOLOGIES

PITCHLOCK SCREW
LAP PF8T60

SIZE J | FSCM NO. 73030 | DRAWING NO. L-14325-12

SCALE 1/1

LAP



[illegible] SPUR SPLINE DATA ITEM 3

FULL FILLET ROOT-SIDE BEARING	~
NUMBER OF SPACES	52
PITCH	20/40 REF
PRESSURE ANGLE	30°
DIMENSIONAL CIRCULAR SPACE WIDTH MAX	.0812
DIMENSIONAL CIRCULAR SPACE WIDTH MIN	.0797 REF
EFFECTIVE CIRCULAR SPACE WIDTH MAX	.0800 REF
EFFECTIVE CIRCULAR SPACE WIDTH MIN	.0785
ROOT DIAMETER	2.6900-2.7000
BASE DIAMETER	2.2517 REF
TRUE INVOLUTE FORM DIAMETER MIN	2.6552
INVOLUTE ANGLE OF ROLL	35° 48' 23" REF
ROOT FILLET RADIUS MIN	.013
MEASURING WIRE DIAMETER	.0850 REF
MEASUREMENT BETWEEN TWO WIRES	2.4825-2.4852 REF
PITCH DIAMETER D	2.6000 ↗ B .002
MINOR DIAMETER	2.555 / 2.550 ↗ D .003
BREAK EDGES ON INVOLUTE FORM & ID BOTH ENDS	.003-.010
BREAK EDGES OVER FULL FACE WIDTH	.003-.010
ANGULAR LOCATION AS A UNIT MAY VARY FREELY	
ACCEPTANCE BY GAGES PER SPEC HS210	CLASS B

.5 MIN FACE WIDTH — [illegible] SPUR SPLINE DATA ITEM 4

FULL FILLET ROOT-SIDE BEARING	~
NUMBER OF SPACES	55
PITCH	24/48 REF
PRESSURE ANGLE	30°
DIMENSIONAL CIRCULAR SPACE WIDTH MAX	.0681
DIMENSIONAL CIRCULAR SPACE WIDTH MIN	.0666 REF
EFFECTIVE CIRCULAR SPACE WIDTH MAX	.0669 REF
EFFECTIVE CIRCULAR SPACE WIDTH MIN	.0654
ROOT DIAMETER	2.3667-2.3767
BASE DIAMETER	1.9846 REF
TRUE INVOLUTE FORM DIAMETER MIN	2.2379
INVOLUTE ANGLE OF ROLL	35° 40' 28" REF
ROOT FILLET RADIUS MIN	.011
MEASURING WIRE DIAMETER	.0700 REF
MEASUREMENT BETWEEN TWO WIRES	2.1958-2.1984 REF
PITCH DIAMETER F	2.2917 ↗ E .002
MINOR DIAMETER	2.255 / 2.250 ↗ F .003
BREAK EDGES ON INVOLUTE FORM & ID BOTH ENDS	.003-.010
BREAK EDGES OVER FULL FACE WIDTH	.003-.010
ANGULAR LOCATION AS A UNIT MAY VARY FREELY	
ACCEPTANCE BY GAGES PER SPEC HS210	CLASS B

.5 MIN FACE WIDTH — [illegible] SPUR SPLINE DATA ITEM 4

FULL FILLET ROOT-SIDE BEARING	~
NUMBER OF TEETH	52
PITCH	20/40 REF
PRESSURE ANGLE	30°
DIMENSIONAL CIRCULAR TOOTH THICKNESS MAX	.0743 REF
DIMENSIONAL CIRCULAR TOOTH THICKNESS MIN	.0728
EFFECTIVE CIRCULAR TOOTH THICKNESS MAX	.0755
EFFECTIVE CIRCULAR TOOTH THICKNESS MIN	.0740 REF
ROOT DIAMETER	2.4900-2.5000
BASE DIAMETER	2.2517 REF
TRUE INVOLUTE FORM DIAMETER MAX	2.5448
INVOLUTE ANGLE OF ROLL	30° 10' 21" REF
ROOT FILLET RADIUS MIN	.011
MEASURING WIRE DIAMETER	.0950 REF
MEASUREMENT OVER TWO WIRES	2.7381-2.7405 REF
PITCH DIAMETER E	2.600 ↗ ~
OUTSIDE DIAMETER	2.650 / 2.645 ↗ E .003
BREAK EDGES ON INVOLUTE FORM & OD BOTH ENDS	.003-.010
BREAK EDGES OVER FULL FACE WIDTH	.003-.010
ANGULAR LOCATION AS A UNIT MAY VARY FREELY	
ACCEPTANCE BY GAGES PER SPEC HS210	CLASS B

.23-.25 FACE WIDTH — [illegible] SPUR SPLINE DATA ITEM 20

FULL FILLET ROOT-SIDE BEARING	~
NUMBER OF TEETH	12
PITCH	12/24 REF
PRESSURE ANGLE	30°
DIMENSIONAL CIRCULAR TOOTH THICKNESS MAX	.1282 REF
DIMENSIONAL CIRCULAR TOOTH THICKNESS MIN	.1267
EFFECTIVE CIRCULAR TOOTH THICKNESS MAX	.1294
EFFECTIVE CIRCULAR TOOTH THICKNESS MIN	.1279 REF
ROOT DIAMETER	.8400-.8500
BASE DIAMETER	.8660 REF
TRUE INVOLUTE FORM DIAMETER MAX	.9127
INVOLUTE ANGLE OF ROLL	19° 3' 46" REF
ROOT FILLET RADIUS MIN	.029
MEASURING WIRE DIAMETER	.1600 REF
MEASUREMENT OVER TWO WIRES	1.2380-1.2402 REF
PITCH DIAMETER G	1.000 ↗ ~
OUTSIDE DIAMETER	1.0833 / 1.0733 ↗ G .003
BREAK EDGES ON INVOLUTE FORM & OD BOTH ENDS	.003-.010
BREAK EDGES OVER FULL FACE WIDTH	.003-.010
ANGULAR LOCATION AS A UNIT MAY VARY FREELY	
ACCEPTANCE BY GAGES PER SPEC HS210	CLASS B

.085-.105 ENGAGEMENT — [illegible] SPUR SPLINE DATA ITEM 20

FULL FILLET ROOT-SIDE BEARING	~
NUMBER OF TEETH	36
PITCH	48/96 REF
PRESSURE ANGLE	30°
DIMENSIONAL CIRCULAR TOOTH THICKNESS MAX	.0300 REF
DIMENSIONAL CIRCULAR TOOTH THICKNESS MIN	.0285
EFFECTIVE CIRCULAR TOOTH THICKNESS MAX	.0312
EFFECTIVE CIRCULAR TOOTH THICKNESS MIN	.0297 REF
ROOT DIAMETER	.7003-.7083
BASE DIAMETER	.6495 REF
TRUE INVOLUTE FORM DIAMETER MAX	.7252
INVOLUTE ANGLE OF ROLL	28° 27' 11" REF
ROOT FILLET RADIUS MIN	.003
MEASURING WIRE DIAMETER	.0400 REF
MEASUREMENT OVER TWO WIRES	.8056-.8080 REF
PITCH DIAMETER H	.7500 ↗ G .002
OUTSIDE DIAMETER	.7708 / .7678 ↗ H .003
BREAK EDGES ON INVOLUTE FORM & OD BOTH ENDS	.003 MAX
BREAK EDGES OVER FULL FACE WIDTH	.003 MAX
ANGULAR LOCATION AS A UNIT MAY VARY FREELY	
ACCEPTANCE BY GAGES PER SPEC HS210	CLASS B

[illegible] SPUR SPLINE DATA ITEM 1

FULL FILLET ROOT-SIDE BEARING	~
NUMBER OF SPACES	12
PITCH	12/24 REF
PRESSURE ANGLE	30°
DIMENSIONAL CIRCULAR SPACE WIDTH MAX	.1336
DIMENSIONAL CIRCULAR SPACE WIDTH MIN	.1321 REF
EFFECTIVE CIRCULAR SPACE WIDTH MAX	.1324 REF
EFFECTIVE CIRCULAR SPACE WIDTH MIN	.1309
ROOT DIAMETER	1.1500-1.1600
BASE DIAMETER	.8660 REF
TRUE INVOLUTE FORM DIAMETER MIN	1.0873
INVOLUTE ANGLE OF ROLL	43° 29' 44" REF
ROOT FILLET RADIUS MIN	.017
MEASURING WIRE DIAMETER	.1460 REF
MEASUREMENT BETWEEN TWO WIRES	.7852-.7887 REF
PITCH DIAMETER	1.000 ↗ A .002
MINOR DIAMETER	.9217 / .9167 ↗ A .003
BREAK EDGES ON INVOLUTE FORM & ID BOTH ENDS	.003-.010
BREAK EDGES OVER FULL FACE WIDTH	.003-.010
ANGULAR LOCATION AS A UNIT MAY VARY FREELY	
ACCEPTANCE BY GAGES PER SPEC HS210	CLASS B

[illegible] SPUR SPLINE DATA ITEM 1

FULL FILLET ROOT-SIDE BEARING	~
NUMBER OF TEETH	65
PITCH	24/48 REF
PRESSURE ANGLE	30°
DIMENSIONAL CIRCULAR TOOTH THICKNESS MAX	.0612 REF
DIMENSIONAL CIRCULAR TOOTH THICKNESS MIN	.0597
EFFECTIVE CIRCULAR TOOTH THICKNESS MAX	.0624
EFFECTIVE CIRCULAR TOOTH THICKNESS MIN	.0609 REF
ROOT DIAMETER	2.1983-2.2083
BASE DIAMETER	1.9846 REF
TRUE INVOLUTE FORM DIAMETER MAX	2.2404
INVOLUTE ANGLE OF ROLL	30° 19' 13" REF
ROOT FILLET RADIUS MIN	.009
MEASURING WIRE DIAMETER	.0800 REF
MEASUREMENT OVER TWO WIRES	2.4067-2.4091 REF
PITCH DIAMETER	2.2917 ↗ A .002
OUTSIDE DIAMETER	2.3333 / 2.3283 ↗ A .003
BREAK EDGES ON INVOLUTE FORM & OD BOTH ENDS	.003-.010
BREAK EDGES OVER FULL FACE WIDTH	.003-.010
ANGULAR LOCATION AS A UNIT MAY VARY FREELY	
ACCEPTANCE BY GAGES PER SPEC HS210	CLASS B

L-14325-12

HSF 1727E 8/81

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15. Supplementary Notes

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16. Abstract

In recent years, considerable attention has been directed toward improving aircraft fuel consumption. Studies have shown that the high inherent efficiency previously demonstrated by low speed turboprop propulsion systems may now be extended to today's higher speed aircraft if advanced high-speed propeller blades having thin airfoils and aerodynamic sweep are utilized.

Hamilton Standard has designed a 9-foot diameter single-rotation Large-Scale Advanced Prop-Fan (LAP) which will be tested on a static test stand, in a high speed wind tunnel and on a research aircraft. The major objective of this testing is to establish the structural integrity of large-scale Prop-Fans of advanced construction in addition to the evaluation of aerodynamic performance and aeroacoustic design.

This report describes the operation, design features and actual hardware of the (LAP) Prop-Fan pitch control system. The pitch control system which controls blade angle and propeller speed consists of two separate assemblies. The first is the control unit which provides the hydraulic supply, speed governing and feather function for the system. The second unit is the hydro-mechanical pitch change actuator which directly changes blade angle(pitch) as scheduled by the control.

17. Key Words (Suggested by Author(s)) Prop-Fan Pitch Change Mechanism Structural/Hydraulic Design	18. Distribution Statement General Release

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