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National Aeronautics and
Space Administration

ENERGY EFFICIENT ENGINE

CORE ENGINE BEARINGS, DRIVES, AND CONFIGURATION

DETAILED DESIGN REPORT

by

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GENERAL ELECTRIC COMPANY
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Prepared for

National Aeronautics and Space Administration

NASA LEWIS RESEARCH CENTER
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Foreword

This report documents technical analysis and design performed by the General Electric Company for the National Aeronautics and Space Administration, Lewis Research Center, under Contract NAS3-20643. The work was performed as part of the Aircraft Energy Efficiency (ACEE) Program, Energy Efficient Engine (E^3) Project. Mr. Carl C. Ciepluch is the NASA Project Manager. Formerly, Mr. W. Hady was the NASA Project Engineer responsible for managing the effort associated with the Bearings, Drives, and Configuration design presented in this report. Presently, Mr. T. Strom is the responsible NASA Project Engineer.

Mr. R.W. Bucy is the Acting Manager of the Energy Efficient Engine Project for the General Electric Company. Mr. J.C. Clark is the Manager responsible for the technical effort relating to Bearings, Drives, and Configuration for the General Electric, Energy Efficient Engine.

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SUMMARY

As part of the contract requirements of the Energy Efficient Engine, a detailed design review was held on the Bearing Systems, Drives, and Configuration for the Core engine. This review was held at the NASA-Lewis Research Center on October 23, 1980.

The review included analysis of the bearing loads and lives, gear stresses, and structural design of the forward and aft sumps, and the accessory drive system. The schematic of the lubrication system was presented along with a discussion of the secondary air systems surrounding the sumps. The design of the major configuration piping was also addressed.

As a result of this design review, approval was granted by the NASA-Lewis, Energy Efficient Engine Project Manager to proceed with the procurement of the respective hardware which was discussed during the review.

INTRODUCTION

A major milestone in the development of the General Electric-Energy Efficient Engine (E^3) is the Core test scheduled for the first quarter of 1982. This report describes the design of the Bearing Systems, Drives, and Configuration required for the Core test. Since many of the drive system components to be used in the Core test are common to other General Electric engines, the design goals for these components have been established based on the requirements of the test cell engine. Table I shows these program goals compared to the design requirements of the Flight Propulsion System (FPS) which represents the fully developed engine installed on an aircraft. A detailed description of the Energy Efficient Engine Flight Propulsion System is given in References 1, 2, and 3.

A cross section of the Core engine is shown in Figure 1. Identified in this figure are the Forward Sump, Aft Sump and Accessory Gearbox. The Lube System, Secondary Air Flow System surrounding the sumps, the Engine Rotor Thrust and its control and the Configuration Piping were also included in this design review.

The forward sump contains the compressor rotor forward thrust bearing, the Power Takeoff (PTO) gearbox and the instrumentation slip ring assembly.

The PTO provides a drive which connects the engine rotor system to the Accessory Gearbox (AGB) mounted on the outer engine casing. Engine required accessories and provisions for two starters are provided on the AGB. The

TABLE I. PROGRAM DESIGN GOALS.

	FPS REQUIREMENTS	TEST CELL ENGINE
• GEARS & G/B BEARINGS	36000 HR	2000 HR
• NUMBER OF STARTS	$\geq$ 40000	$\geq$ 2000
• MISSION CYCLES	36000 CYCLES	10000
• BLADE OUT		
- CONTINUED OPERATION	-1-1/2 HP OR LP TURBINE BLADEOUT	--
	-2 HP COMPRESSOR BLADEOUT	--
	-1 HP OR LP DOVETAIL & ADJACENT BLADEOUT	--
• FLIGHT ENVIRONMENT	-MACH. NO/ALT. PROFILE -ATTITUDES	TEST CELL CONDITION
• SPEEDS	-MAX ROTOR SPEEDS	CORE TEST SPEEDS
	-MAX LP TORQUE	--

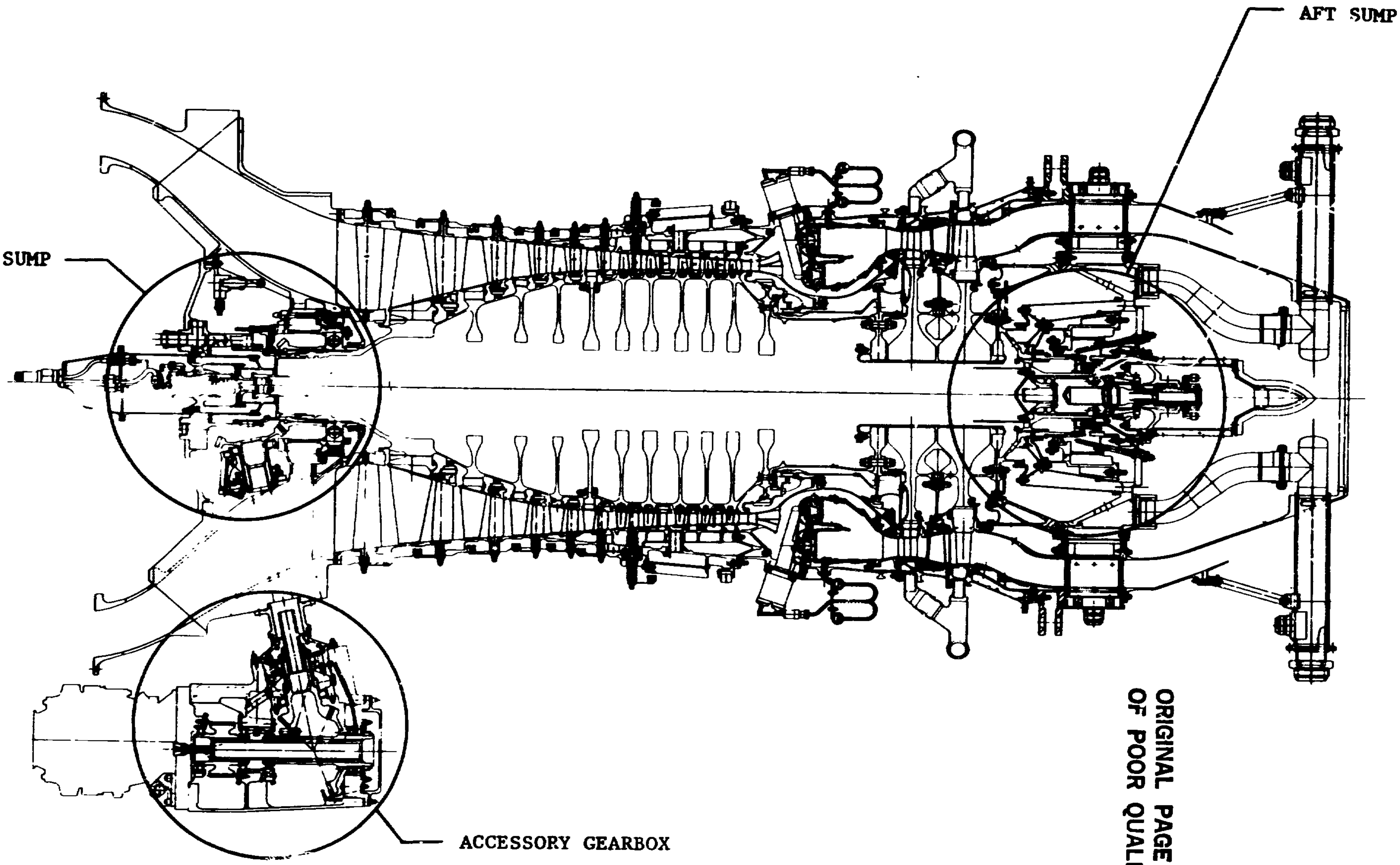

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Figure 1. Core Engine Cross Section.

AGB is designed to be used on both the Core and the Integrated Core Low Spool (ICLS) engine. The ICLS engine is a complete turbo-fan configuration. It represents the Core engine with the Fan and Low Pressure Turbine systems added. A separate Detailed Design Review will be held for the ICLS Bearings, Drives, and Configuration.

The aft sump contains a roller bearing which supports the aft end of the rotor system. A means of exhausting the compressor rotor cooling air and manifolding to control the engine rotor thrust along with an instrumentation slip ring assembly is also provided in the aft sump.

The Core engine inlet conditions will vary from test cell ambient to inlet pressures simulating the ICLS fan discharge pressure and an exhaust nozzle will be used in place of the ICLS low pressure turbine. To operate the sumps in a safe manner, sump pressures will be kept lower than the engine inlet pressure to prevent oil loss. The rotor thrust control cavity which is provided just aft of the high pressure turbine will keep the core thrust bearing loads in line with those expected in the ICLS engine.

The lube system design is critical to the successful operation of the Core engine. Many proven components will be used in the lube system but design analysis has been made to insure that these components will meet the requirements of the Energy Efficient Engine program.

The configuration piping, although unique to this engine, has been designed using proven design techniques established in other engine programs.

I. FORWARD SUMP

The design of the forward sump to be used in the Core engine for the Energy Efficient Engine is shown in Figure 2. The core thrust bearing, which is spring-mounted and fluid damped, is included in this sump. The Power Tak off (PTO) gearbox is driven directly by the compressor stub shaft. The drive shaft from the PTO gearbox to the Accessory Gearbox exits radially from the forward sump through the bottom strut in the frame.

Just forward of the horizontal PTO ge... is a manifold housing which performs the following functions:

- Supplies seal pressurization air to the forward sump labyrinth seal.
- Provides a passage for the compressor rotor cooling air, which is obtained from a shop air source for the Core engine.
- Incorporates a purge cavity for both the slip ring cooling air and the compressor rotor cooling air.

The forward manifold housing also supports the instrumentation slip ring assembly which provides for instrumentation leadout from the compressor rotor area. A cover supports the manifold housing and acts as a closure for the forward end of the sump.

The forward sump will incorporate instrumentation to determine the fol-

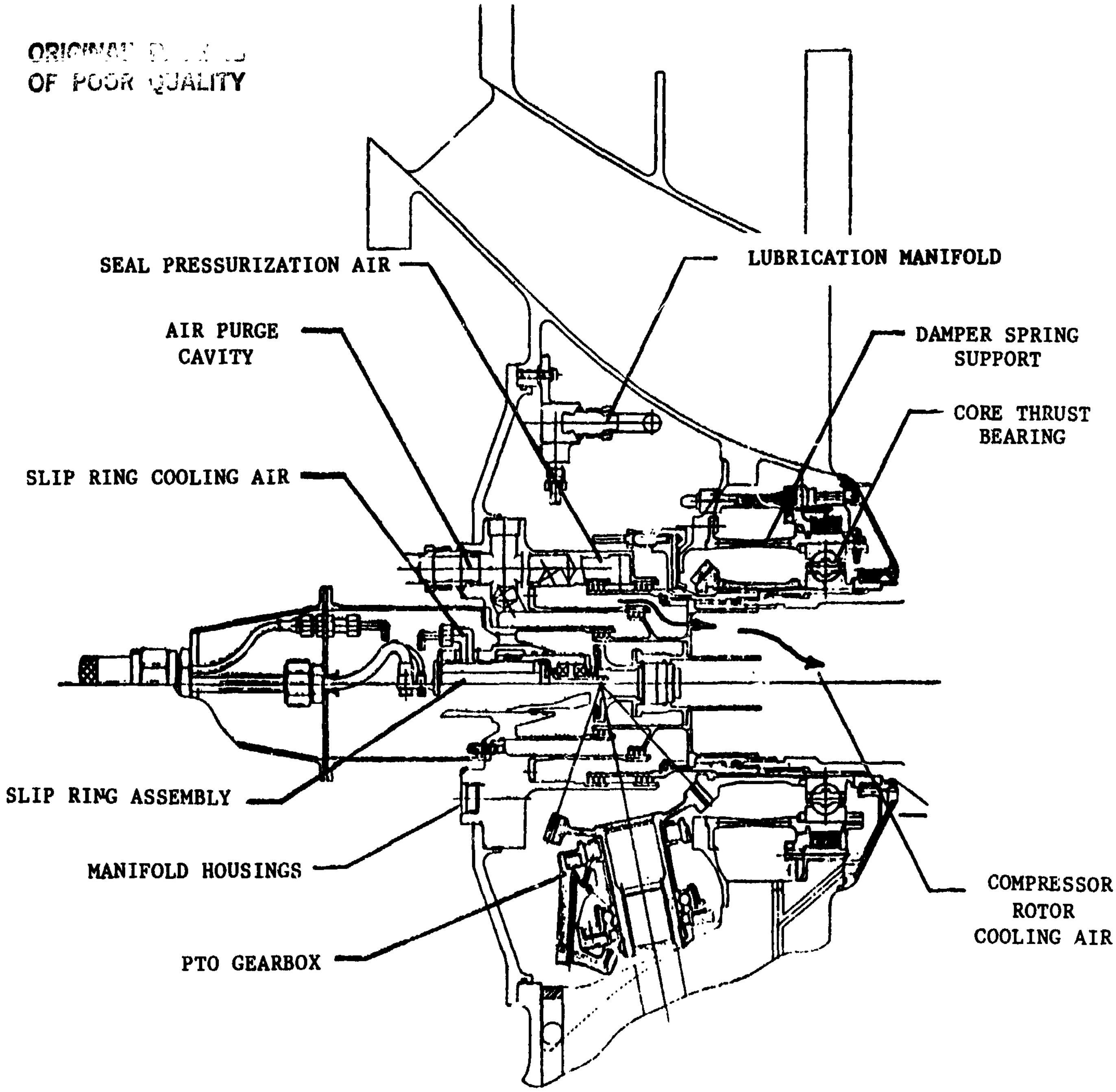

Figure 2. Core Engine Forward Sump.

lowing:

- Thrust bearing outer and inner race temperatures
- Sump and various air cavity pressures and temperatures
- Stresses and deflections in the damper spring support
- The direction of thrust for the engine rotor system
- Vibration levels at the thrust bearing support
- Radial movement of the spring housing

Lubrication is supplied to the core thrust bearing by three lube jets mounted just forward of the horizontal bevel gear which supply oil through the inner ring of the thrust bearing. A total of 9.5 liter/min. (2.5 gpm) of oil is supplied to the bearing. The PTO is supplied with 5.7 liter/min. (1.5 gpm) of oil. The fluid damper, supplied by a separate lube manifold is provided with a continuous flow of approximately .8 liter/min. (.2 gpm).

Conventional materials (See Figure 3) are used in the forward sump with the exception of the damper spring support which is manufactured from Marage 250 a high-strength vacuum-melted 18Ni maraged steel. This material is used for the spring support because of its high endurance strength.

Corrosion protection is provided where needed by using phosphate coatings or black oxide treatment.

A. Thrust Bearing Damper Design

The area around the core thrust bearings is shown in more detail in

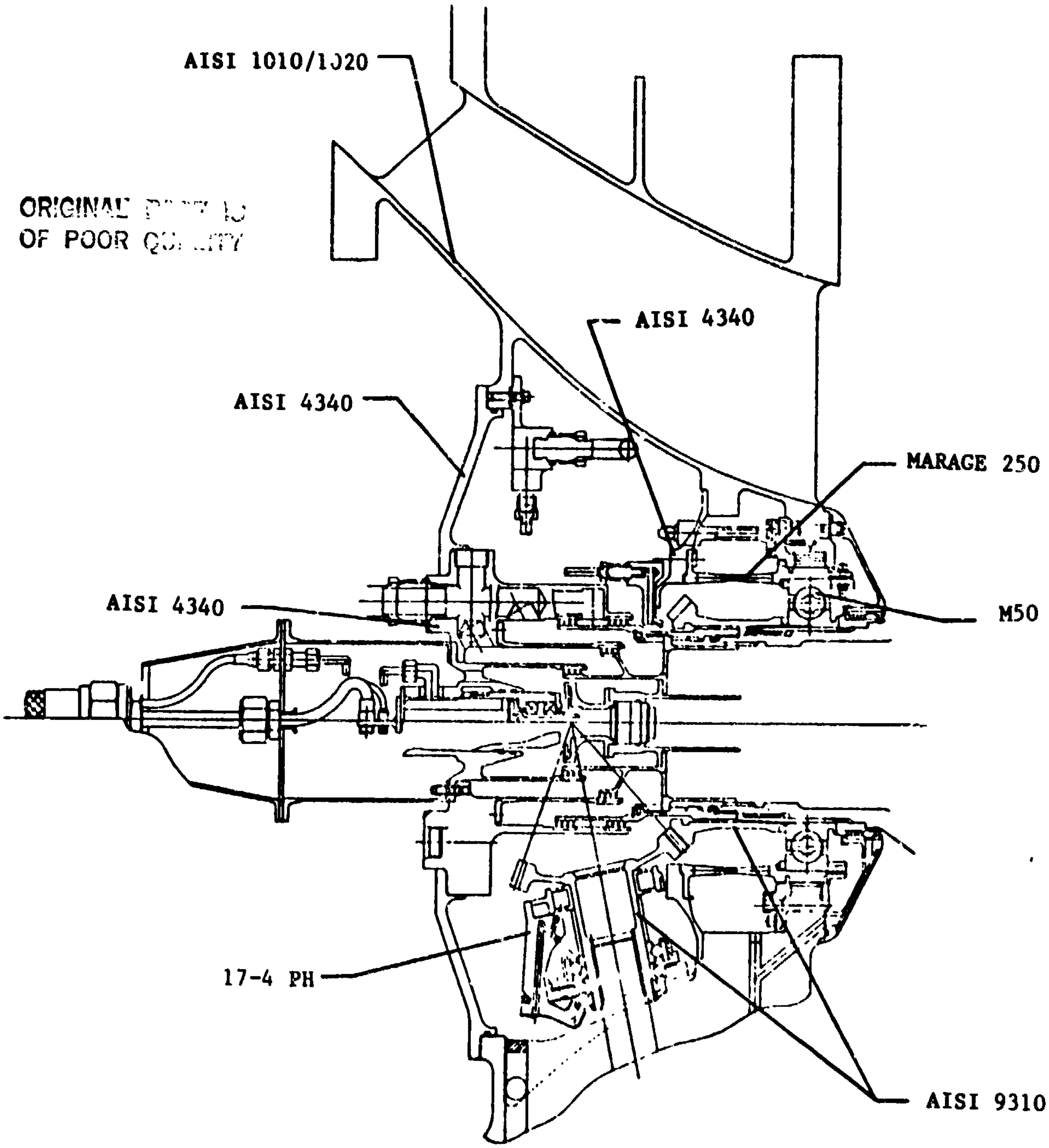

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Figure 3. Forward Sump Materials.

Figure 4. Support for the thrust bearing is through the damper spring supports which mount to a housing forward of the thrust bearing. The damper spring housing features 34 machined beams which have an overall spring constant of 525.4 kN/cm. (300,000 lb/in.). These beams are tapered to move the high stressed area out away from the ends where stress concentration factors have to be applied. The maximum stress in the beams is below the endurance limit of Marage 250 when deflected $\pm$ 0.508 mm (0.02 inch).

The damper, located radially outward from the thrust bearing, is a five shim end sealed damper. Sealing is accomplished by piston rings. Oil is supplied to the damper from a separate manifold and is distributed internally through six equally spaced holes at the outside diameter of the damper.

Six check valves are provided to prevent back flow of high pressure oil.

Meehanite, a gray cast iron, which has been shown to have excellent wear characteristics, is being used for the piston rings. The piston ring cross-section is 3.81 mm (0.150 inch) wide and 3.302 mm (0.13 inch) thick. These proportions have been selected to obtain a moment balanced design when the damper assembly has been deflected 0.249 mm (0.0098 inch) which is the calculated deflection due to 38100 gr-mm (1500 gr-in.) of high pressure turbine rotor unbalance. Expected damper oil pressure is 3447-8273 kPa (500 psi - 1200 psi) depending upon internal operating temperature.

The total damper shim clearance is 1.173-1.326 mm (0.0462 - 0.0522 inch) but a mechanical stop is provided at 0.508 mm (0.020 inch) to limit the overall radial excursions of the compressor rotor.

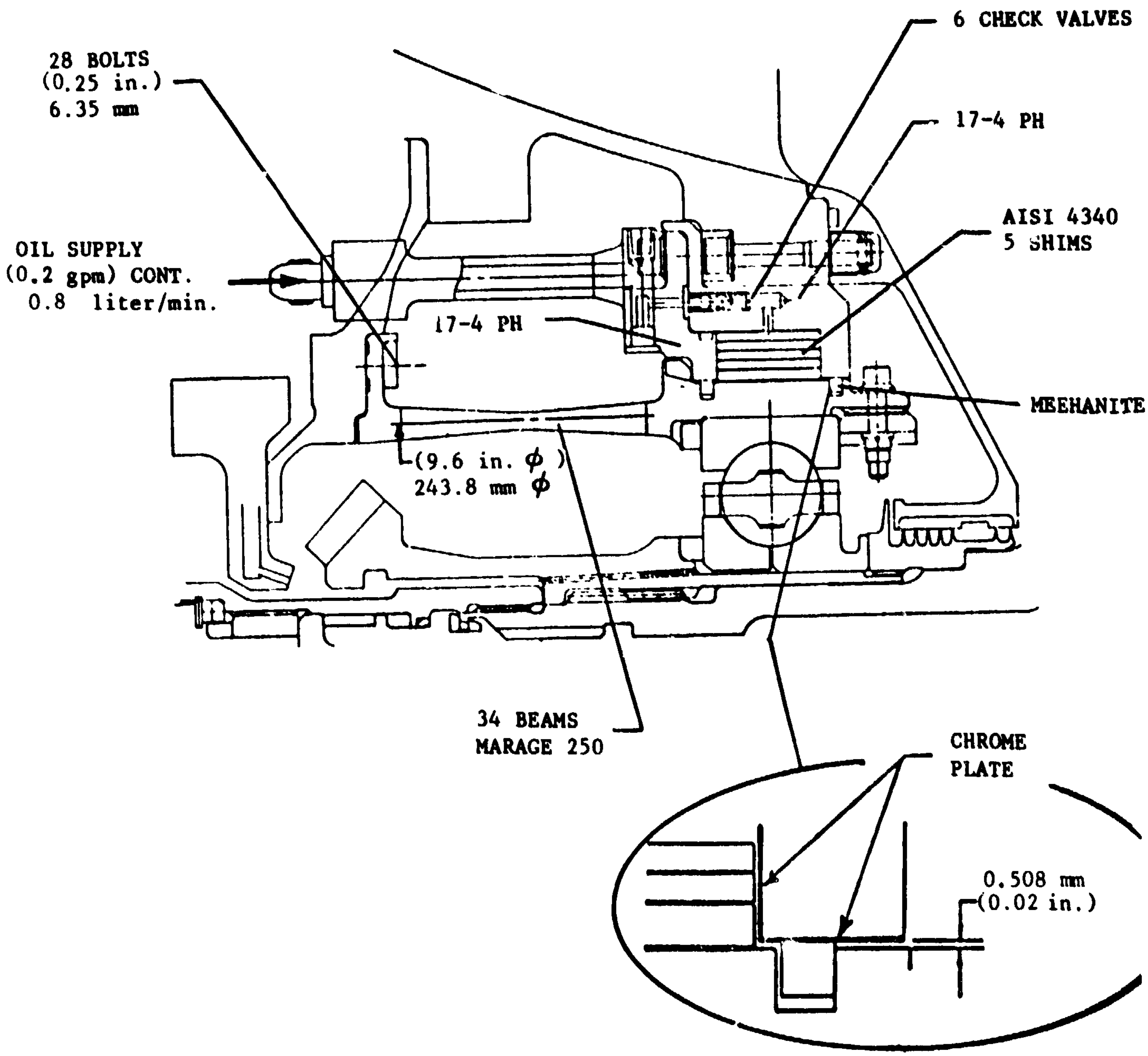

Figure 4. Core Thrust Bearing Area.

B. Thrust Bearing Design

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The core thrust bearing is shown in Figure 5 along with some information pertinent to its design. For the Core engine the maximum DN is 2.16 x 10^6 which is about 8% higher than current commercial engine experience. This, however, is not expected to be a problem. To enhance the cooling of this bearing, oil is fed through 24 slots in the forward inner ring. The bearing also incorporates anti-rotational tangs to prevent outer race rotation.

The Cubic Mean Load (CML) for the bearing has been determined by a rotor thrust analysis of a typical commercial flight mission cycle. The calculated L_{10} life utilizing a multiplying factor established from experimental data on similar bearings is 36,000 hrs. Figure 6 shows the relationship of L_{10} life versus thrust bearing load. The maximum expected rotor thrust for the Core engine is 26,686N (6,000 lbs.) which means the bearing life should be in excess of 9,000 hrs.

Figure 7 shows the bearing fits and clearances. At all operating conditions the fitup of the bearing inner ring remains tight on the shaft to prevent shaft wear. At maximum speed the minimum fit is 0.0076 mm (0.0003 in.) tight. The contact angle will vary between 27.2° and 22.5° which is consistent with the design goal of 24.3° based on GE design practices.

A thrust bearing of similar size (TF39-4B) has been component tested at a DN of 2.23 x 10^6 in support of the 1-6 and 1-10 stage compressor rig tests. An oil delivery scheme similar to that used in the Core engine was used in the test. No problems were encountered and the oil rate to the core bearing was determined from this test. A component test of the core thrust bearing will be run to confirm oil delivery rates and determine other operating characteristics.

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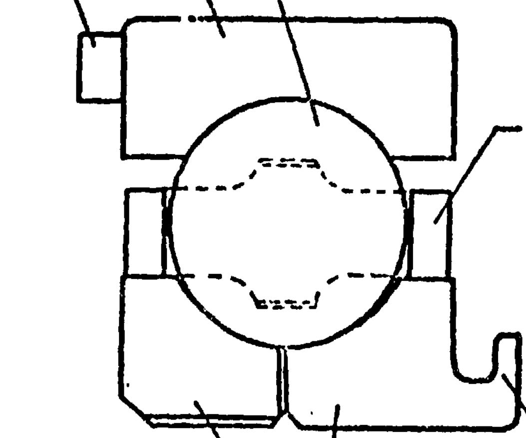

BORE DIAMETER	(6.4002 in.)	162.57 mm
MEAN DIAMETER	(8.0000 in.)	203.20 mm
OUTSIDE DIAMETER	(9.5877 in.)	243.63 mm
ELEMENT SIZE	(1.0625 in.)	26.99 mm
NUMBER OF ELEMENTS	20	
MAX SPEED	13300 rpm	
DN x 10^6	2.16	
LOAD (CML)	(3110 lb.)	13.83 kN
L_{10} LIFE	36000 hrs	

Figure 5. Core Thrust Bearing Design.

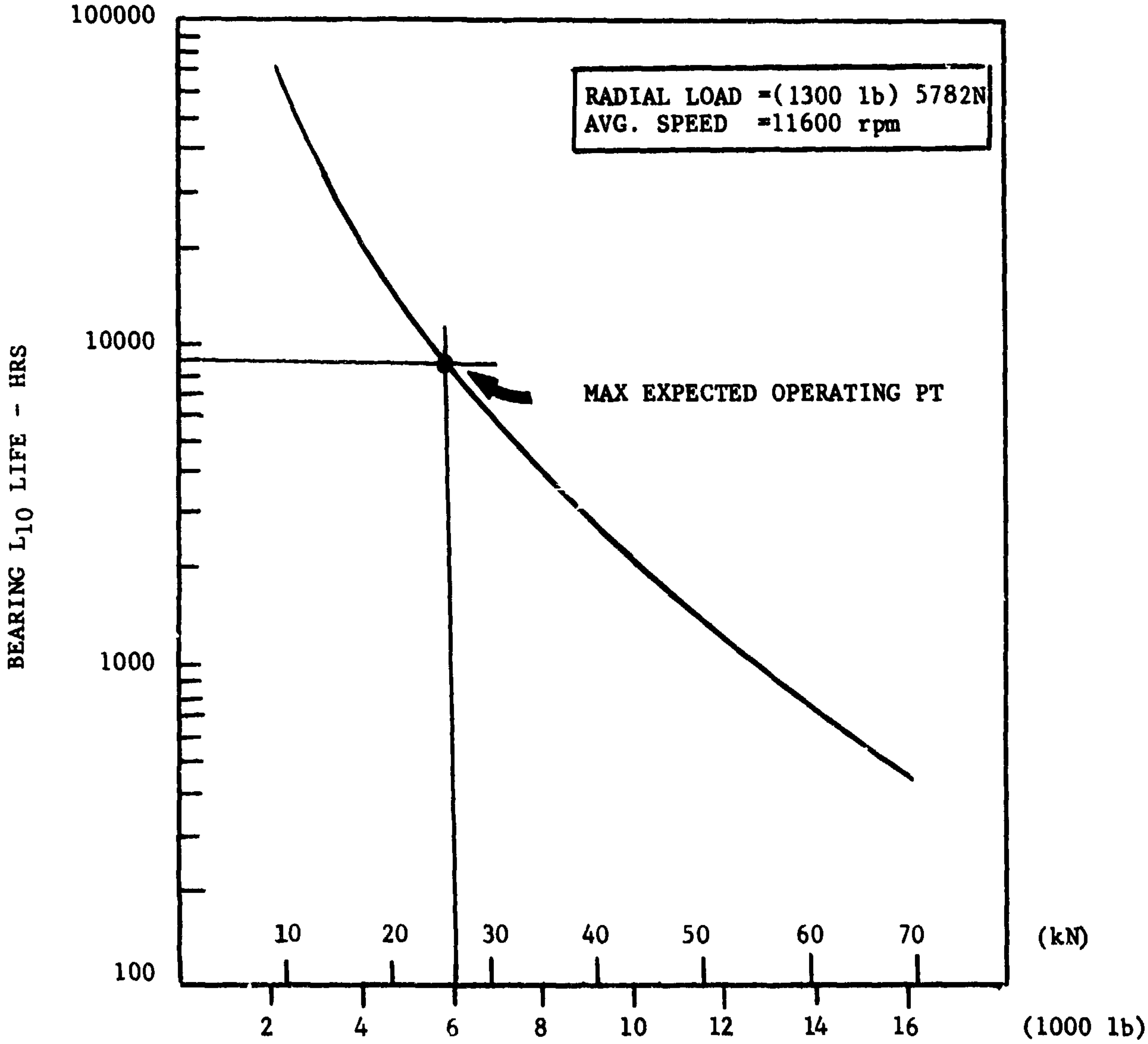

Figure 6. Thrust Bearing Life Vs. Thrust Load.

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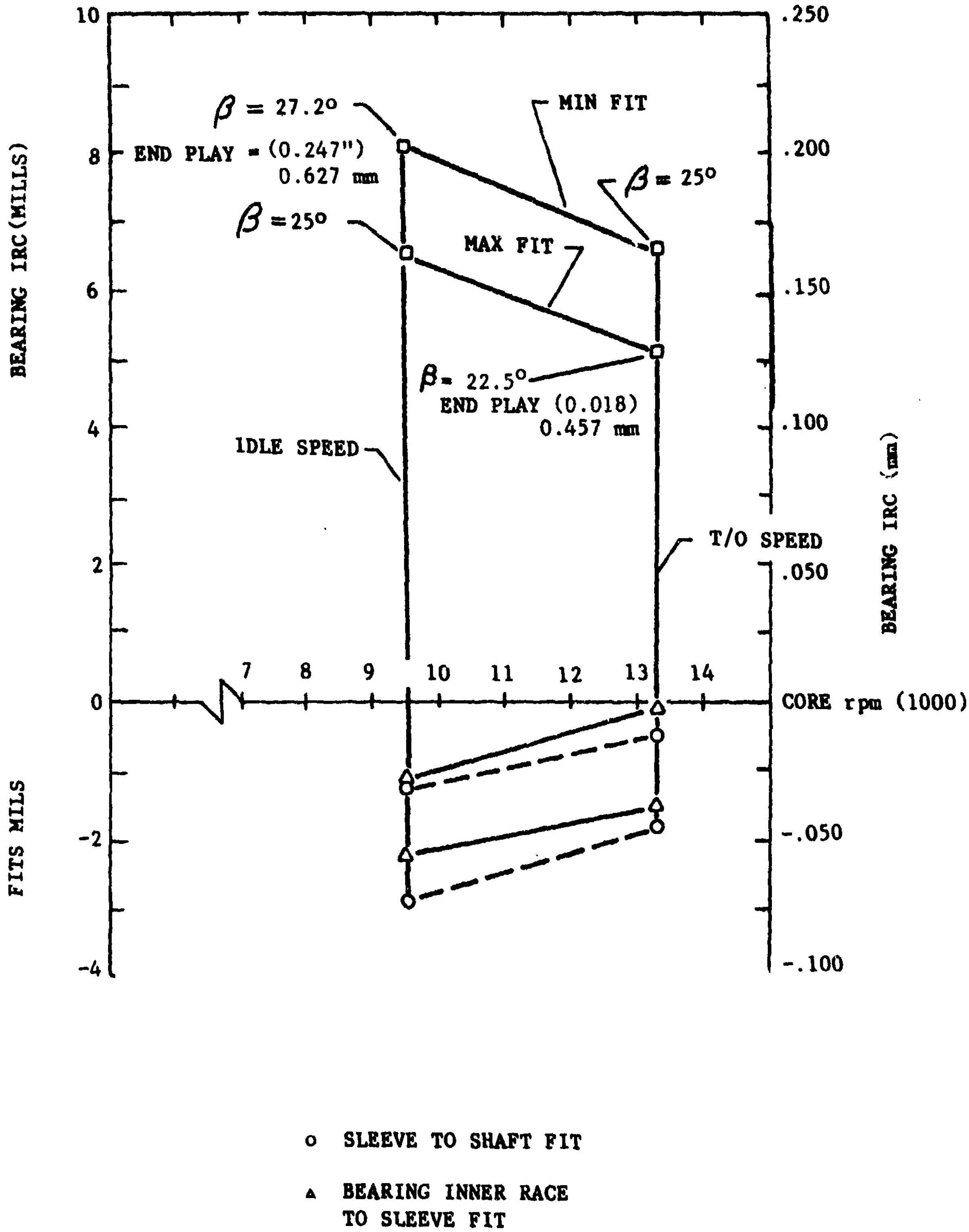


Figure 7. Core Thrust Bearing Fits and Operating Clearance.

C. Sump Sealing

The Core engine will utilize labyrinth seals throughout the sumps. Figure 8 shows the forward sump sealing arrangement. A nickel graphite coating consistent with GE experience is being used for the rub material. No hard coating is used on the sump labyrinth teeth to prevent the danger of "flakes" entering gears or bearings. The aft seal is provided with a seal drain to prevent incipient oil leakage from entering the compressor inlet.

D. Support Housing Stresses

Stress of the structural hardware of the forward sump are very low, well below the yield strength of the material used. Figure 9 shows a summation of the analysis of the bearing support housing. Stresses are only 17% of the allowable yield strength.

II. AFT SUMP

The design of the aft sump to be used in the Core engine is shown in Figure 10. This sump includes the aft support roller bearing which is mounted in a spring housing similar to that used in the forward sump. A fluid film damper is not required in the aft sump based on the engine system rotor dynamic analysis.

As in the forward sump, labyrinth seals are used for sealing the forward and aft end of the sumps. These seals are pressurized by shop air supplied through the support housing. Using shop air for seal pressurization will surround the forward end of the sump with cool air at approximately 93°C(200°F).

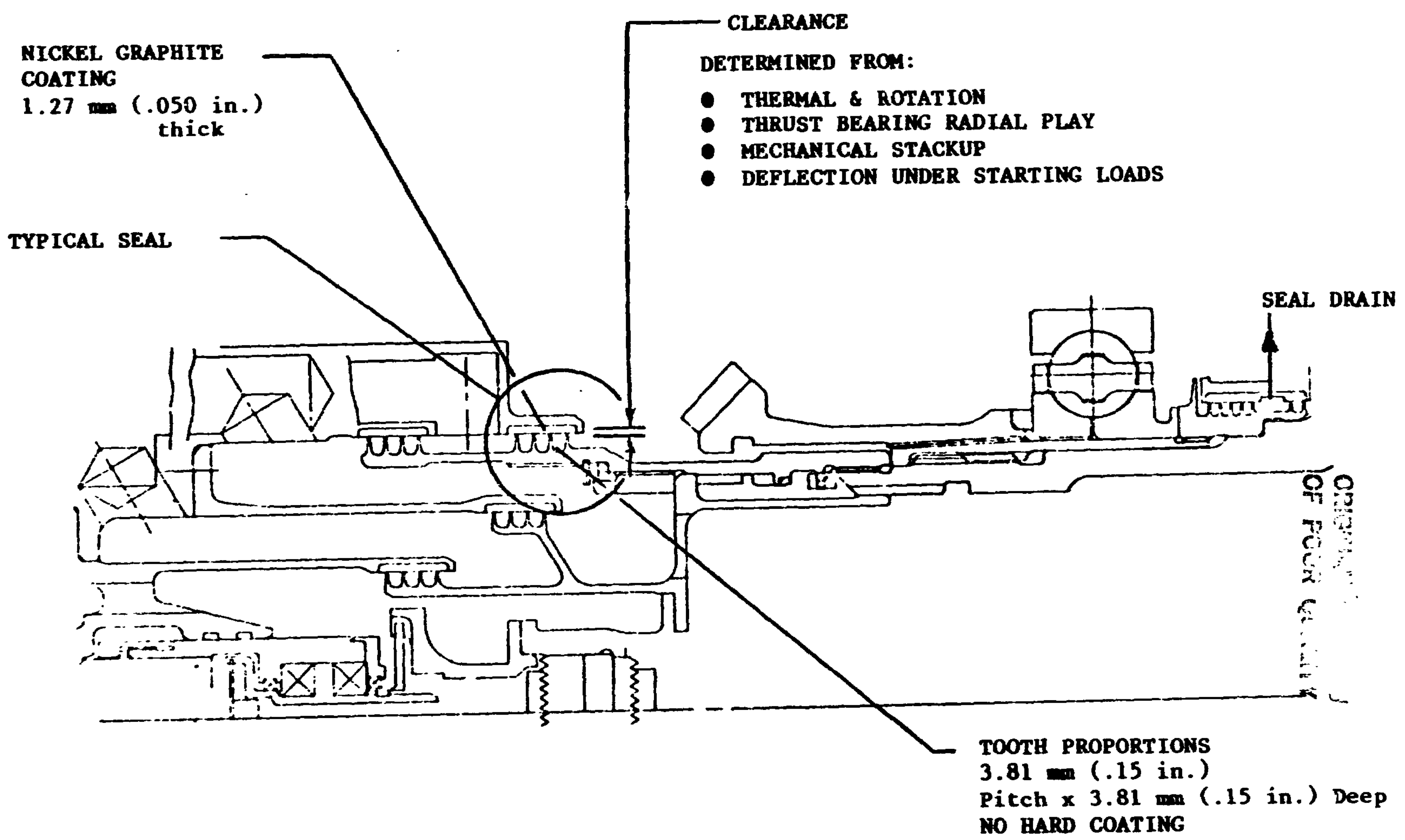
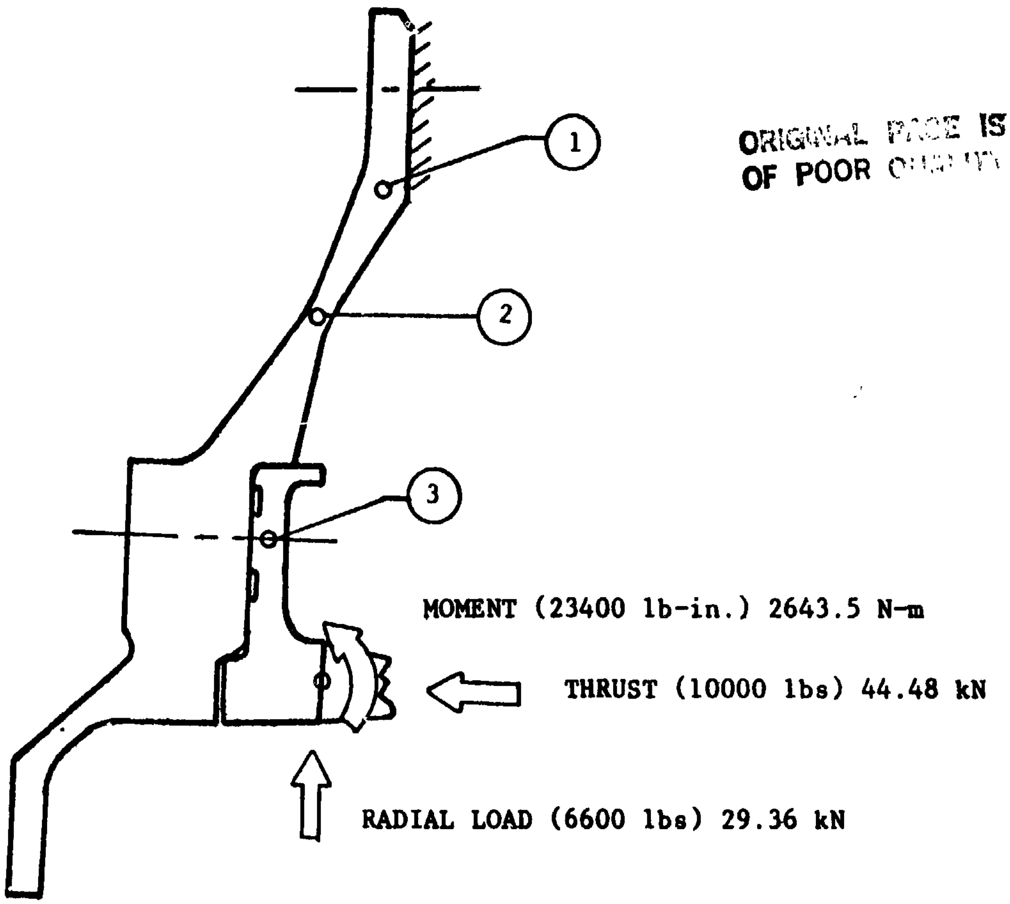


Figure 8. Core Forward Sump Labyrinth Seals.



	LOADING	
LOCATION	THRUST	MOMENT + RADIAL
1	(15 ksi) 103.4 MPa	(11 ksi) 75.8 MPa
2	(21 ksi) 144.8 MPa	(10 ksi) 68.9 MPa
3	(21 ksi) 144.8 MPa	(51 ksi) 351.6 MPa

MATL AISI 4340
Y.S. (.02%) =(125 ksi @ 200°F) 861.8 MPa @ 93.3°C

Figure 9. Core Thrust Bearing Support Housing Stresses.

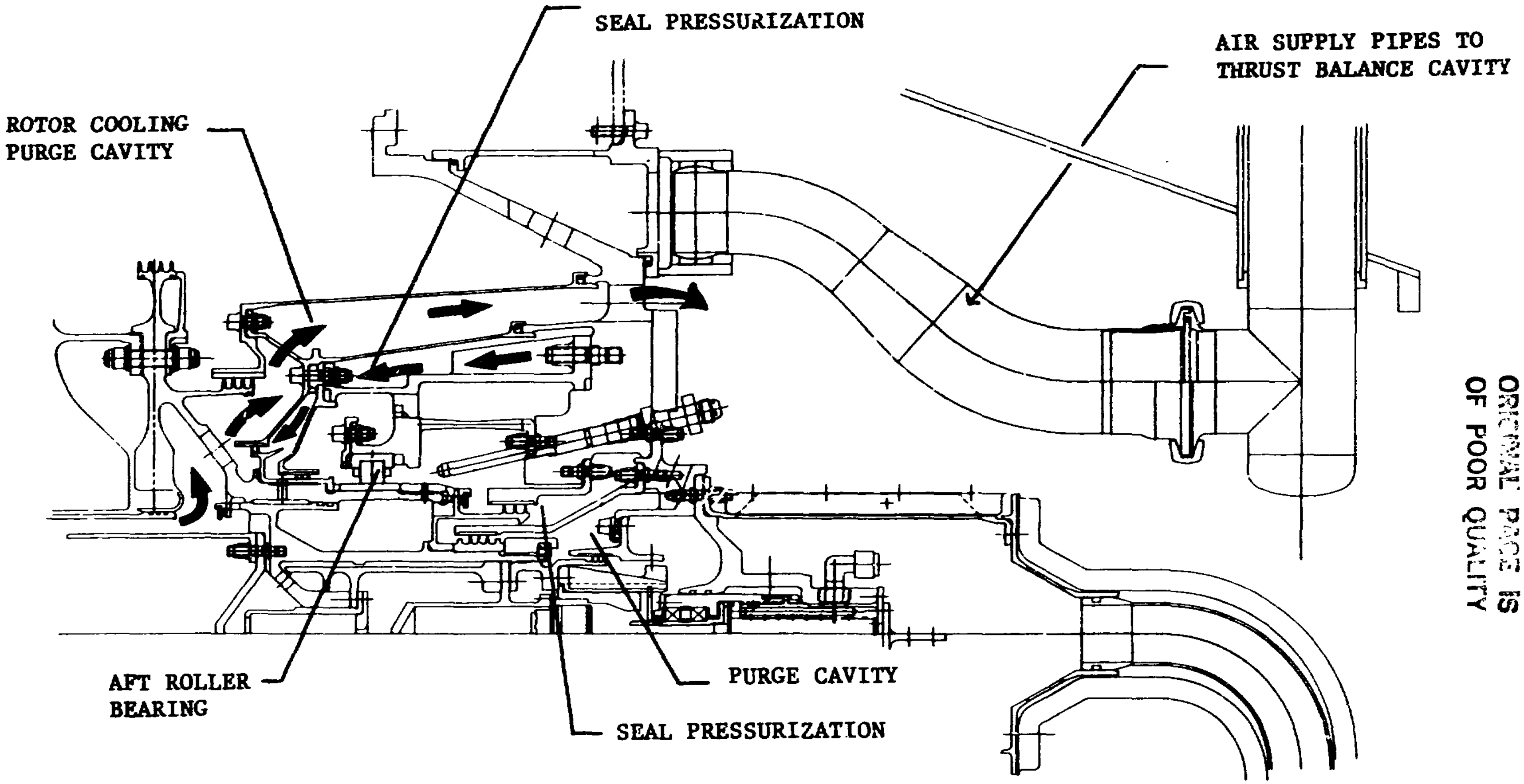


Figure 10. Core Engine Aft Sump.

The cavity outboard of the seal pressurization cavity purges the air that cools the compressor rotor. This air comes from the forward sump through the aft housing. It is expected that this air will be approximately 371°C (700°F) max. The physical flow area of this cavity is as large as possible to minimize the pressure drop.

A thrust balance cavity is provided between the aft balance piston seal and the stationary structure. This cavity is supplied pressurized air through two 50.8 mm (2 inch) diameter pipes. It is planned to maintain this pressure at 344.7 kPa absolute (50 psia) during most of the testing. This thrust balance cavity is required to keep the thrust load on the core thrust bearing to levels consistant with long life.

The aft sump also includes an instrumentation slip ring assembly. Lead-out for the slip ring instrumentation is through the aft service struts. Slip ring cooling air is purged through the aft housing exhausting to atmospheric pressure.

The materials to be used in the aft sump are shown in Figure 11. The materials used in the aft sump are capable of the higher temperatures that will be experienced during operation and during "soak back" after shutdown. Marage 250 is being used for the spring housing because of its high endurance strength.

The aft sump will incorporate instrumentation to determine the following:

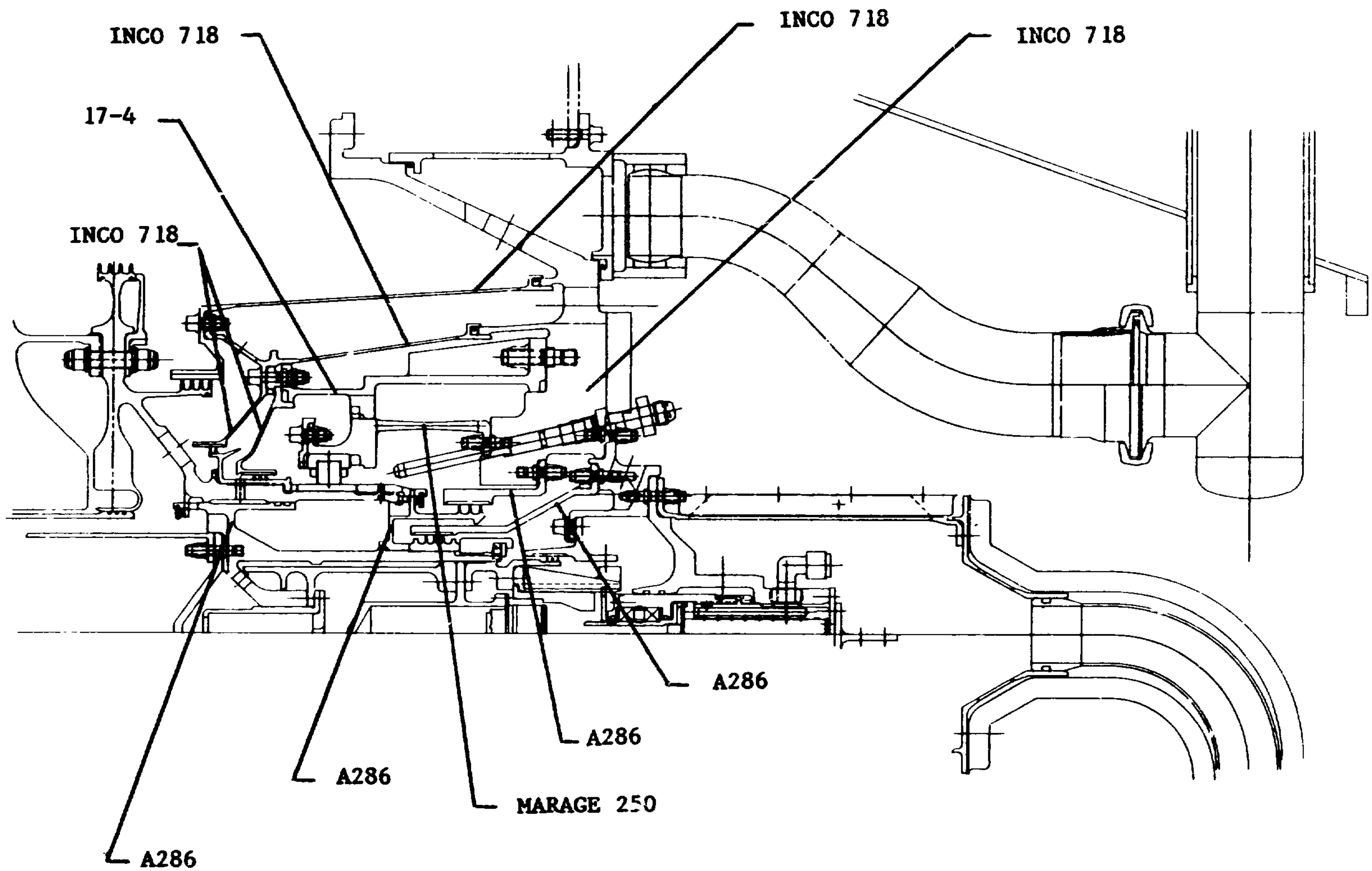


Figure 11. Core Engine Aft Sump Materials.

- The outer race and inner race temperature of the roller bearing
- Sump and various air cavity pressures and temperatures
- Stresses and deflections in bearing spring support
- Vibration levels at the bearing spring support

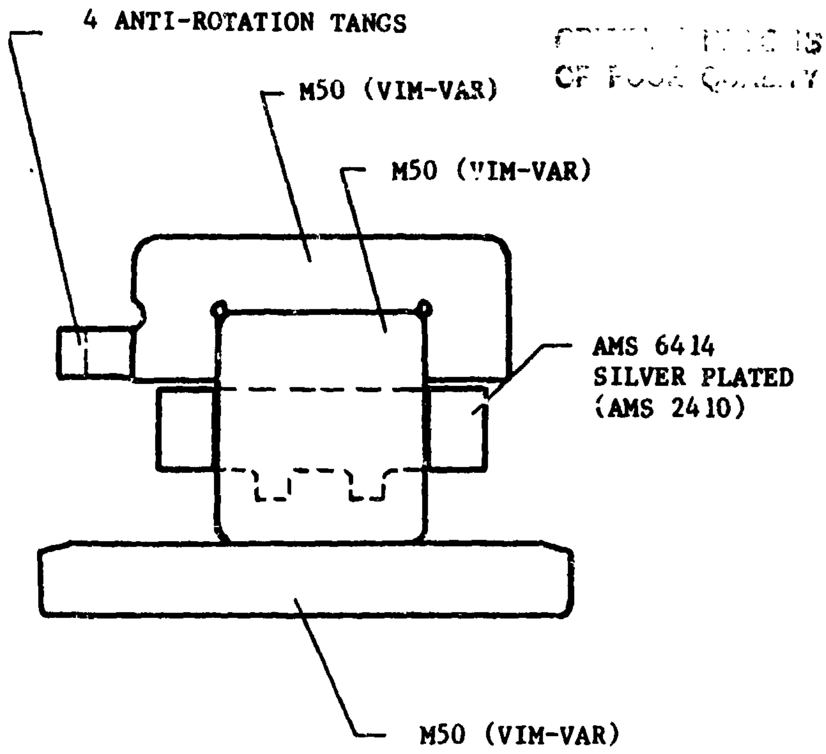
A. Aft Roller Bearing

The Core engine aft roller bearing design information is shown in Figure 12. The maximum bearing DN is 2.3×10^6 which is about 7% higher than current experience for commercial engine roller bearings. To minimize roller-to-race skidding the bearing has been designed to run with a slight negative clearance. Negative clearance results in an internal load which will prevent skidding but also reduces the life of the bearing. With the tightest bearing fitup the minimum bearing life would be 200 hrs. of engine operation at takeoff conditions. A 200-hrs. operation at takeoff conditions is equivalent to more than 1000 hrs. of normal engine testing.

The bearing is provided with four anti-rotation lugs to prevent race rotation in its housing.

Figure 13 shows the relationship of bearing L_{10} life versus internal clearance. Bearing life, which is very sensitive to negative clearance, has been calculated to be between 200 and 900 hrs. when considering the tolerance spread on the bearing fitup.

Figure 14 shows how the bearing clearance will vary as a function of Core engine speed. The maximum fitup conditions is shown at these condi-



BORE DIAMETER	(6.856 in.) 174.14 mm
MEAN DIAMETER	(7.751 in.) 196.88 mm
OUTSIDE DIAMETER	(8.660 in.) 219.96 mm
ELEMENT SIZE	(.5512 in. x .5512 in.) 14 mm x 14 mm
NUMBER OF ELEMENTS	28
MAX SPEED	13300 rpm
DN x 10^6	2.31
LOAD (CML) (LB)	N/A
L_{10} LIFE (HRS)	200 HR @ T/O (>1000 ENG. HR)

Figure 12. Core Engine Aft Roller Bearing.

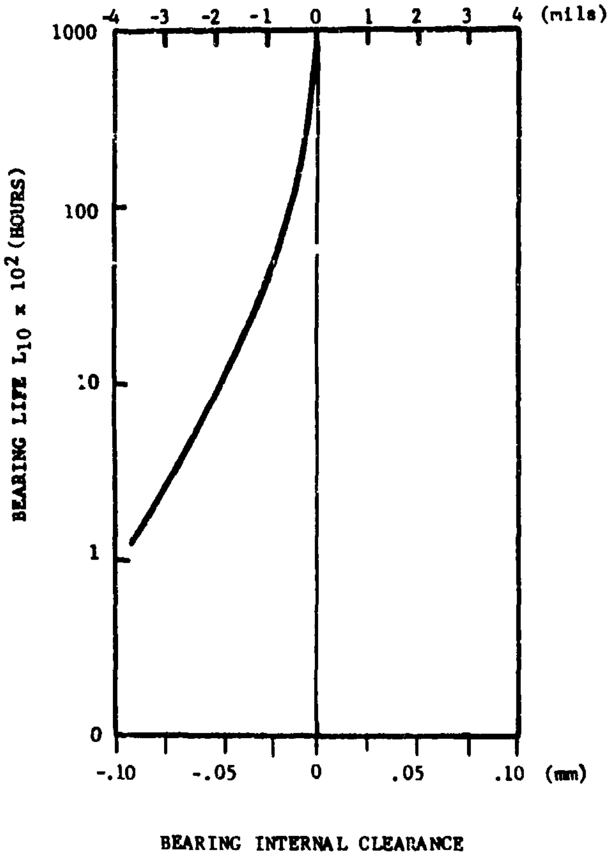


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Figure 13. Aft Bearing Life vs. Operating Clearance.

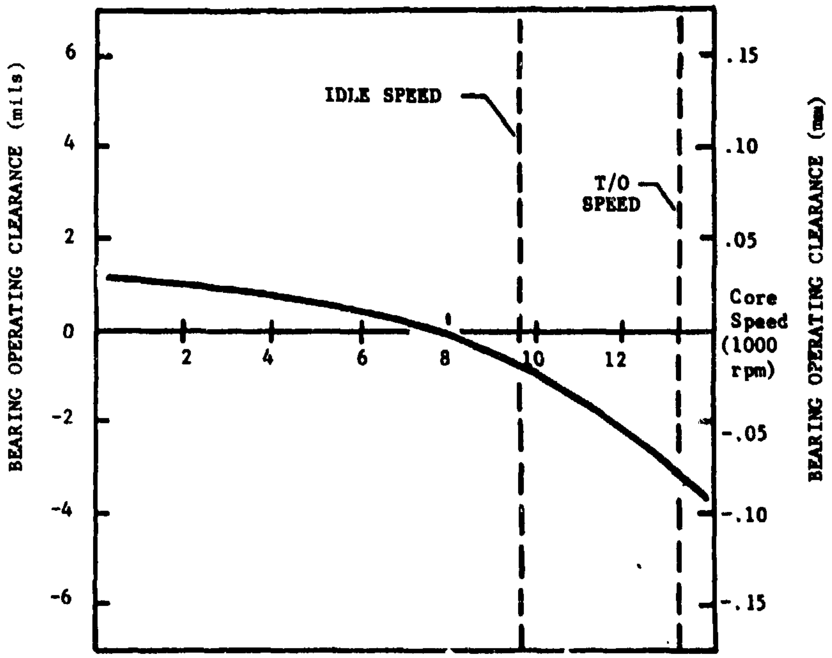


Figure 14. Aft Bearing Operating Clearance vs. Core Speed.

tions the fit will vary from -0.0254 mm (-0.001 in.) at idle to -0.0762 mm (-0.003 in.) at max. speed. At minimum fitup conditions the internal clearance will vary from zero to -0.0508 mm (-0.002 in.).

The bearing assembled into its housing is shown in Figure 15. The housing contains 34 machined beams identical to those used in the forward thrust bearing housing discussed in Section IA. The total allowable radial movement is 0.508 mm (0.02 in.) which is limited by a mechanical stop. Four anti-rotational lugs are provided in the spring housing to prevent windup. Twenty bolts secure the spring housing to the main structure.

B. Aft Sump Labyrinth Seals

The aft sump oil seals are labyrinth seals and they are designed using the same criteria as described in Section IC. Typical aft sump air seals are shown in Figure 16. Here the base metal (Inco 718) is coated with Aluminum Oxide to reduce heat generation as the teeth cut into the rub material. The rub material is Hastelloy X honeycomb brazed to a base material of Inco 718. The radial thickness of the honeycomb is 3.43 mm (0.135 in.) to insure that there will be no possible wearing of seal teeth into the base material which would generate excessive heat.

C. Aft Sump Structure Stresses

The aft sump housing and the rotor thrust air manifold are aft sump components which exhibit high stress levels. These components are shown in Figure 17. These structures are life-limited in low cycle fatigue (LCF). The air manifold has an LCF life of 10,000 cycles (A ratio = 1); the aft bearing

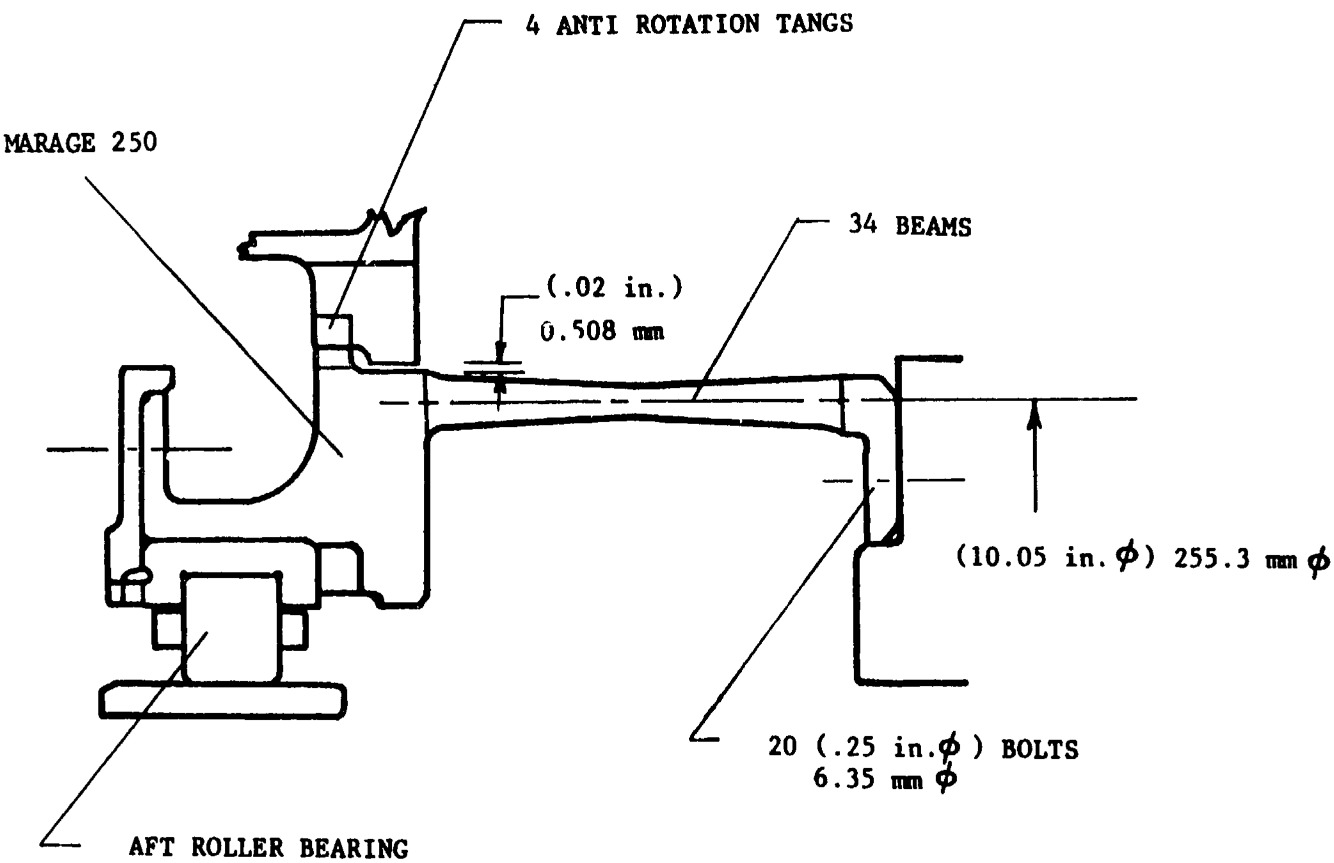
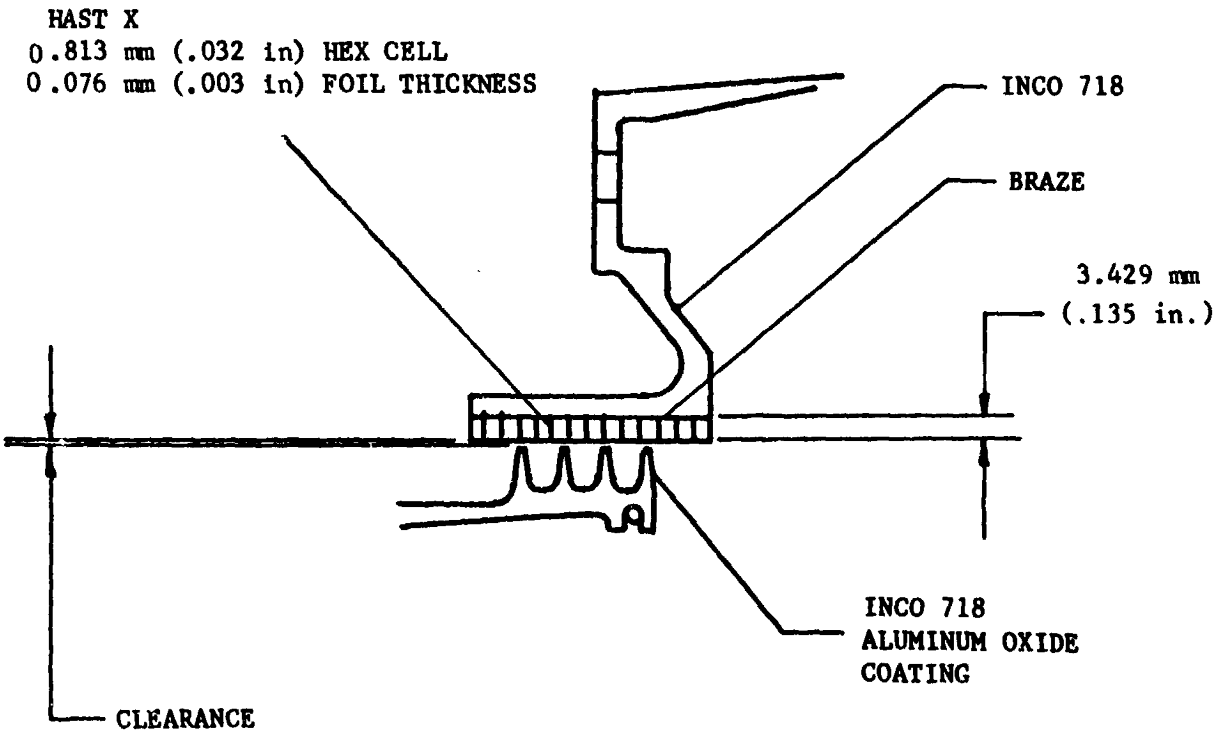


Figure 15. Aft Bearing Spring Housing.



DETERMINED FROM:

- THERMAL & ROTATION
- MAINTAINING MIN. AIR LEAKAGE
- MECHANICAL STACKUP

Figure 16. Core Engine Aft Sump Air Seal.

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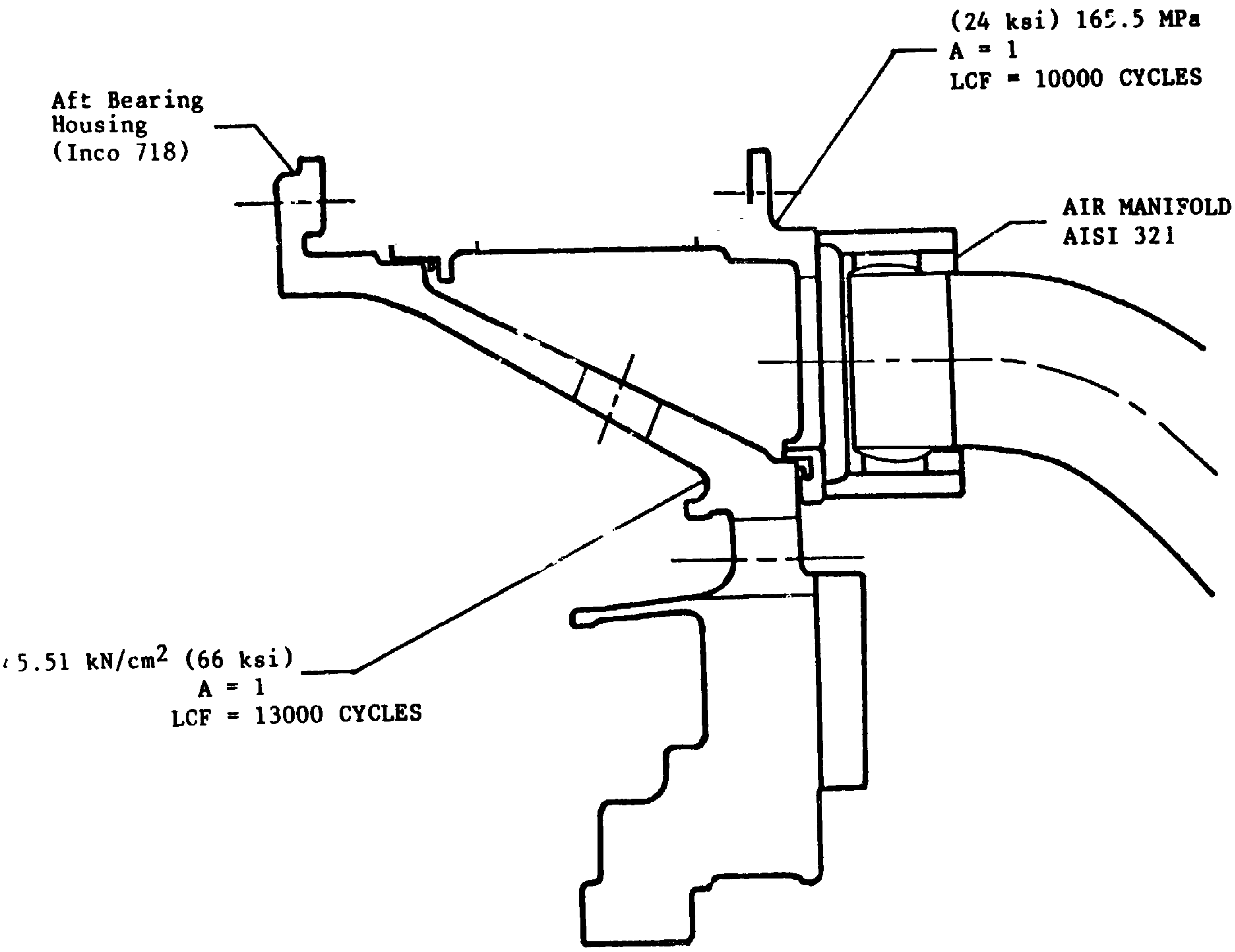

Figure 17. Aft Sump Structure Stresses.

housing has an LCF life of 13,000 cycles (A ratio = 1). Both of these meet the minimum life requirement of 10,000 test cycles.

III. DRIVE SYSTEM

The drive system schematic is shown in Figure 18. The PTO gearbox mounted in the forward sump drives the Accessory Gearbox (AGB) mounted on the engine outer casing. The AGB provides drive pads for the following accessories:

- Lube & Scavenge Pump
- Two Air Starters
- Control Alternator
- Fuel Pump/Control

The gearbox is designed for a maximum combined torque from the starters of 1084.8 N-m (800 lb-ft). The expected maximum accessory hp requirements for the Core engine is 53.7 kW (72 hp). This is much lower than a typical commercial engine which may be as high as 335.7 kW (450 hp) due to aircraft required generators.

Below is a comparison of the design requirements of the Flight Propulsion System (FPS) and the Core/ICLS engine.

TABLE II. DRIVE SYSTEM DESIGN LIFE REQUIREMENTS.

	CORE/ICLS Engine	FPS Engine
Total No. of Starts	2000	40000
Total Service Life	2000 hr.	36000 hr.

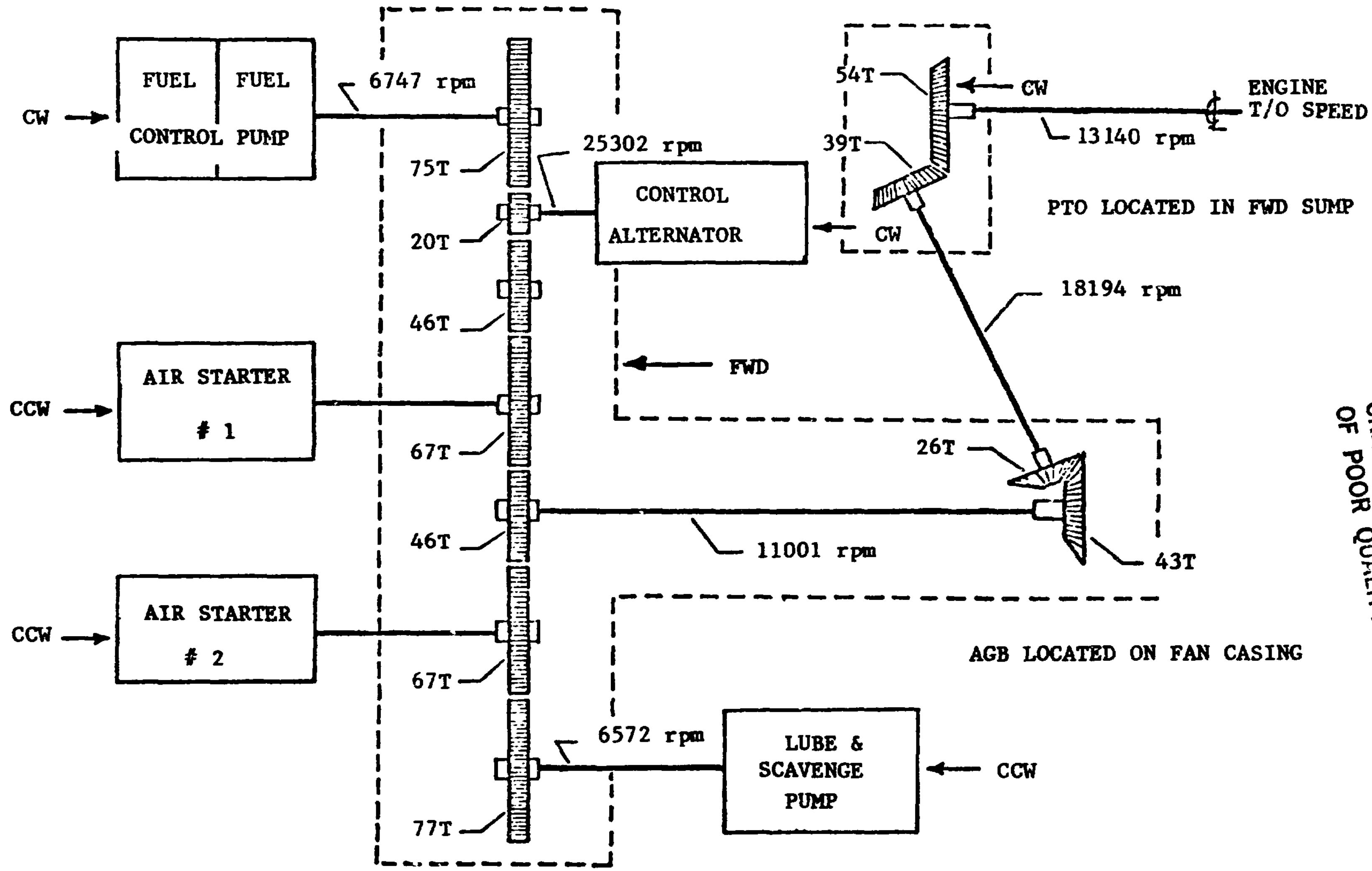

Figure 18. ICLS Accessory Drive Schematic.

A. Power Takeoff Design

The PTO is shown in Figure 19. The PTO gear is supported by three bearings. Two roller bearings react the tangential loads and the loads caused by the overturning moments. A thrust bearing reacts the gear thrust. The outer rings, inner rings, and rolling elements are M50 material; the cages are silver-plated AMS 6414 (AISI 4340). The bearings for the PTO are sized to expected FPS loads, and the calculated lives shown in Figure 20 meet the design goals of 36,000 hrs.

A summary of the bevel gear design is shown in Table III. These gears are also designed for FPS starting and accessory load conditions. The bending and compressive stresses are below the allowable stresses for the material. The shaft angle of 78° 03' was chosen to facilitate mounting the FPS accessory gearbox in the core compartment area.

Although the gear pitch line speed is 7% higher than current practice, it should not be a problem. Special attention is being given to the location of the gear mesh lubricating and cooling jets. Two are being used in close proximity to the gear mesh on both the incoming and outgoing side of the mesh.

The bevel gear contact patterns and backlash are being developed and specified to allow for slight radial movements of the horizontal gear due to the spring-mounted rotor system.

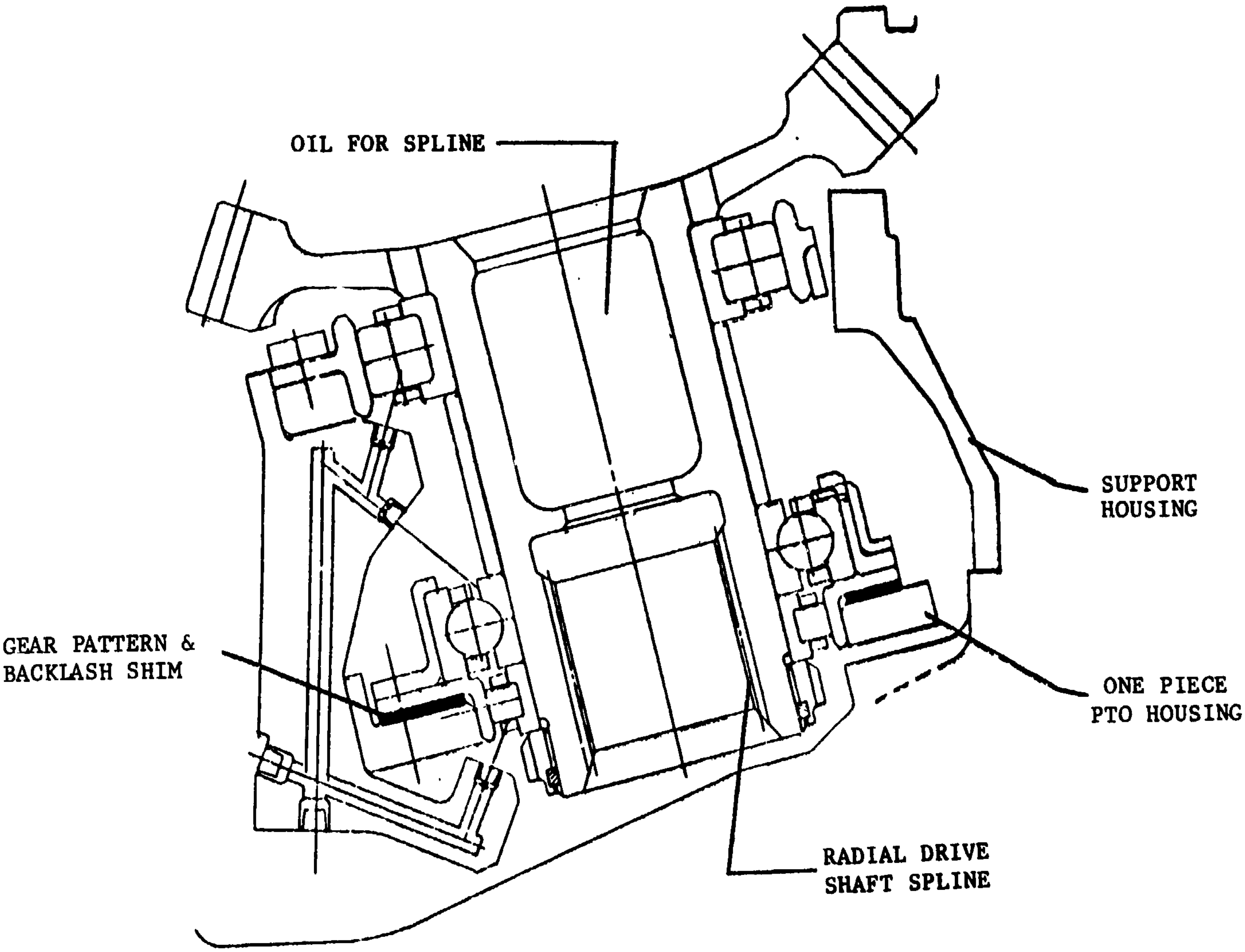

Figure 19. ICLS/Core PTO Gearbox

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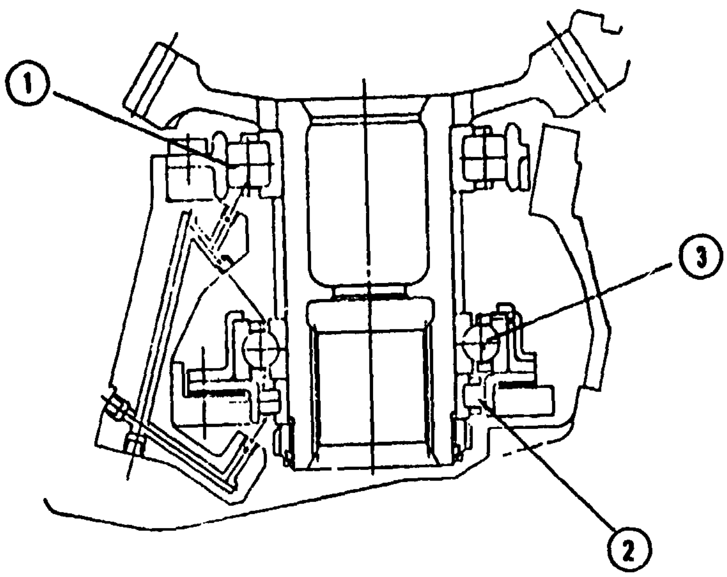


12282 ROTOR rpm

	POSITION		
	1	2	3
SIZE	211	1911	111
AVG. SPEED (rpm)	17007	17007	17007
LOAD (FPS LOADS) N(lbs)	(665) 2958	(370) 1646	(374) 1664
LIFE (hrs)	127200	97376	76419

- SYSTEM LIFE - 45970 HOURS
- FPS DESIGN REQUIREMENTS - 36000 HOURS

Figure 20. PTO Bearing Design Summary.

TABLE III. PTO BEVEL GEAR DESIGN SUMMARY.

● GEAR TOOTH NUMBER		
- PINION	39	
- GEAR	54	
● FACE WIDTH	(.92 in.) 23.4 mm	
● DIAMETRAL PITCH	6.488	
● PRESSURE ANGLE	22.5°	
● SPIRAL ANGLE	35.0°	
● SHAFT ANGLE	78° 03'	
● MAX SURFACE SPEED	(29000 FPM) 8839.2 m/min.	
● BENDING STRESS		
- START	255.1 MPa	(258.6 MPa Allowable)
	37000 psi	(37500 psi Allowable)
- MAX ACCESSORY LOAD (FPS LOAD)	10000 psi	(37500 psi Allowable)
	68.9 MPa	(258.6 MPa Allowable)
● COMPRESSIVE STRESS		
- START	1365 MPa	(1724 MPa Allowable)
	198,000 psi	(250,000 psi Allowable)
- MAX ACCESSORY LOAD (FPS LOAD)	100,200 psi	(250,000 Allowable)
	690.8 MPa	(1724 MPa Allowable)
● SCORING	ΔT IS LOW (NO PROBLEM)	

B. Accessory Gearbox Design

The AGB design is shown in Figures 21 and 22. The AGB is being designed to meet the requirements of both the Core and ICLS engine; the only difference is in the mounting of the gearbox to the engine casing.

The gearbox features a one-piece main housing with four adapters to which gears and bearings are preassembled before final assembly into the gearbox housing. Three of the seven spur gears have been obtained from other engine programs and three gears have been designed specifically for the Energy Efficient Engine gearbox. Only four new bearings out of 18 bearings will be required for the gearbox and many of the miscellaneous smaller parts have also been obtained from other programs.

The material for the housing and adapters will be 356-T6 Aluminum, the gears will be AISI 9310 and the bearings are made of M50 with silver-plated AISI 4340 cages.

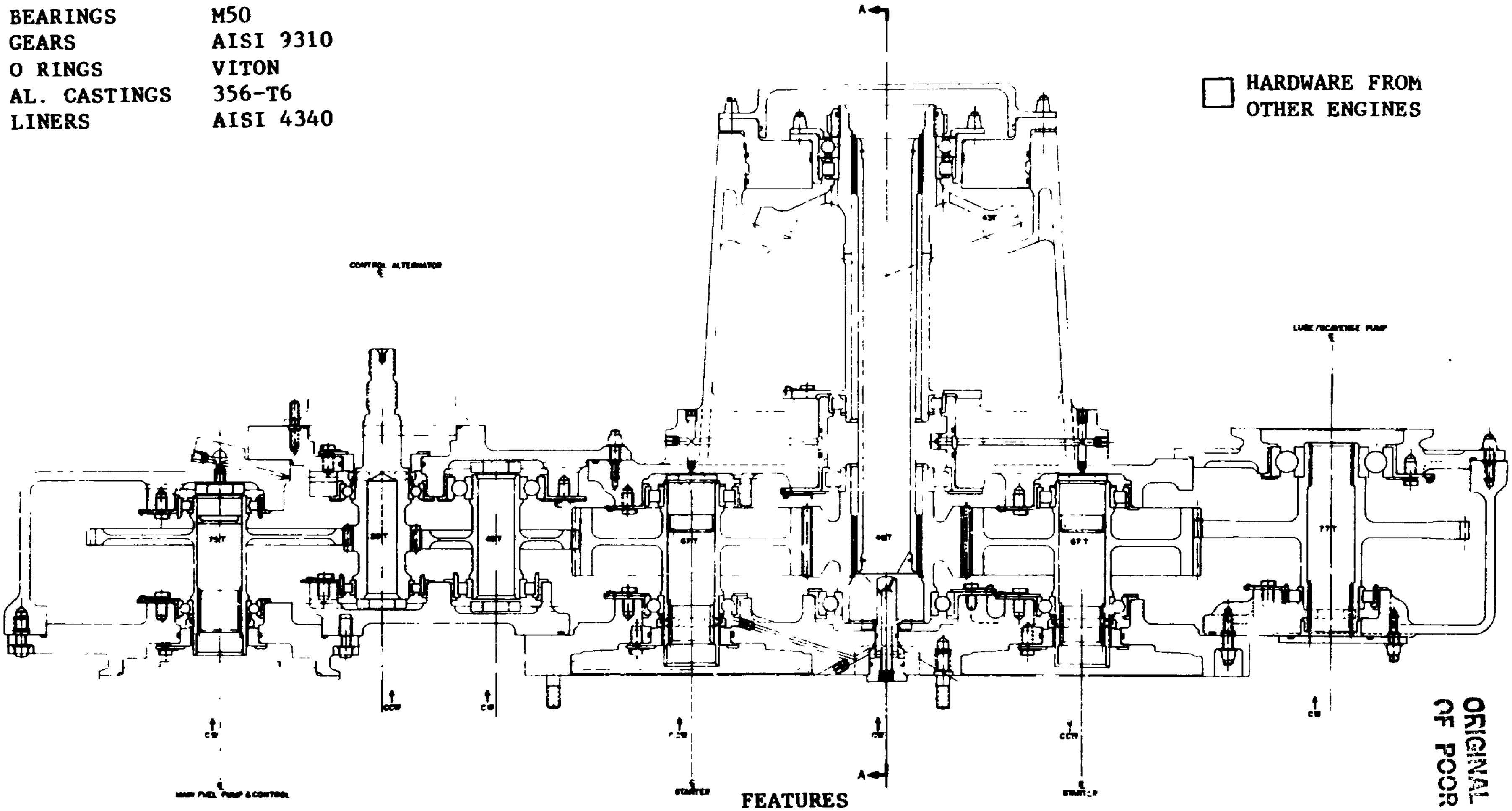
A summary of the bearing loads, speeds and lives for both the spur and bevel gears is shown in Figures 23 and 24. The lives of the gearbox bearings are more than adequate for the Core and ICLS engine with the minimum bearing life being 400,000 hrs.

The spur and bevel gear design information is shown in Figure 25 and Table IV. The bevel gears are sized for 2,000 starts assuming that each start utilizes the max torque of the starters, which is conservative. The bending strength of the bevel gears is enhanced by utilizing a 22.5° pres-

MATERIALS

BEARINGS	M50
GEARS	AISI 9310
O RINGS	VITON
AL. CASTINGS	356-T6
LINERS	AISI 4340

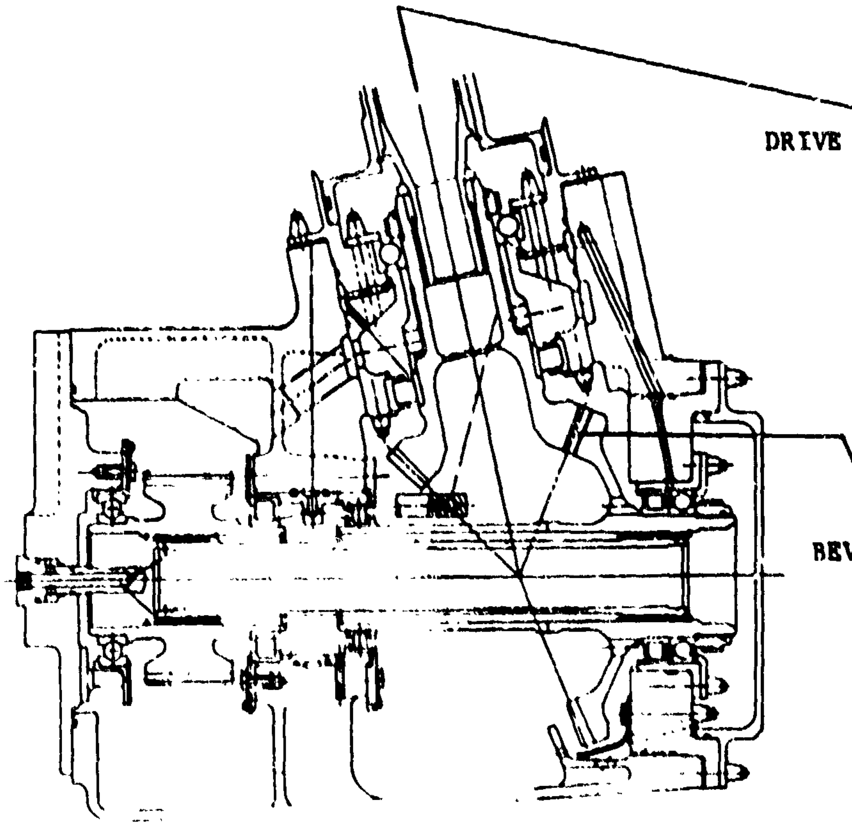
☐ HARDWARE FROM OTHER ENGINES



FEATURES

- ONE PIECE MAIN CASING
- HARDWARE COMMON WITH OTHER GE ENGINES
- CARBON SEALS - MATING RINGS OIL COOLED
- THREE NEW SIMPLE ADAPTORS

Figure 21. Core/ICLS Accessory Gearbox.



FEATURES

- BEVEL GEAR JET LUBED IN & OUT OF MESH
- BEARING JET LUBRICATED
- ALL SPLINES LUBRICATED

Figure 22. Accessory Gearbox Bevel Gear Cross Section

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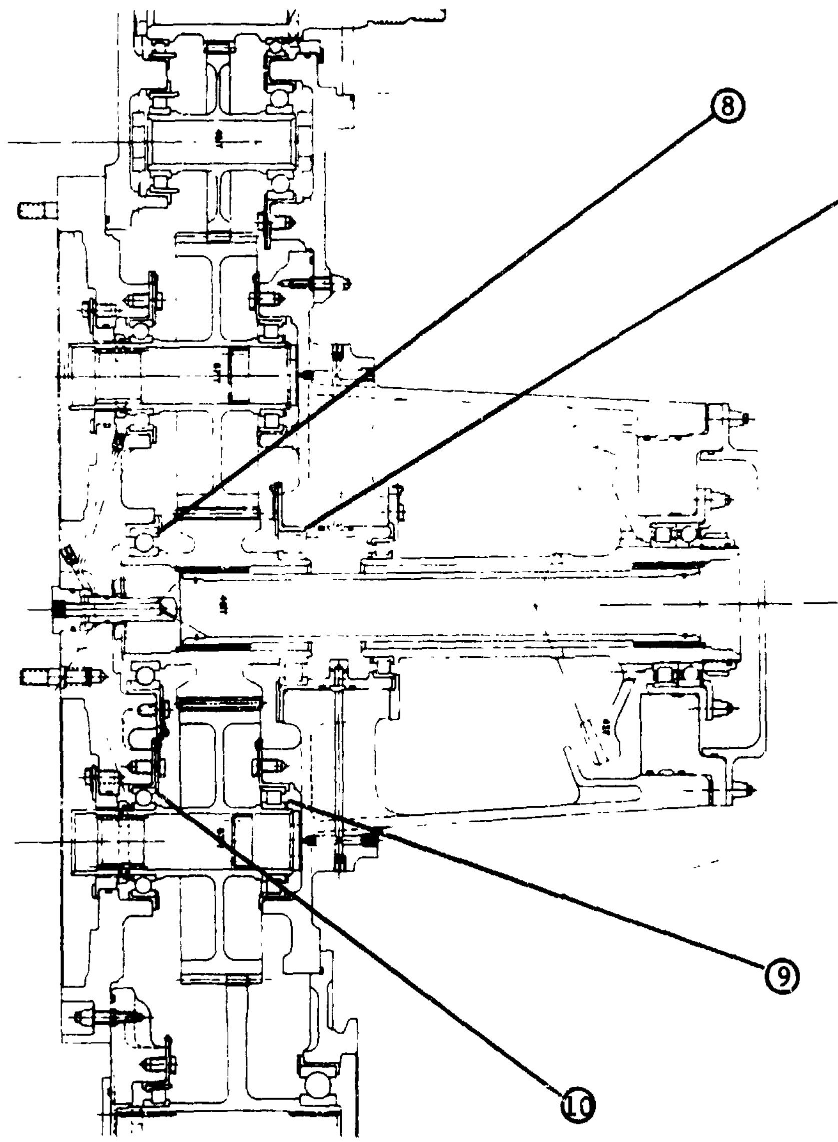
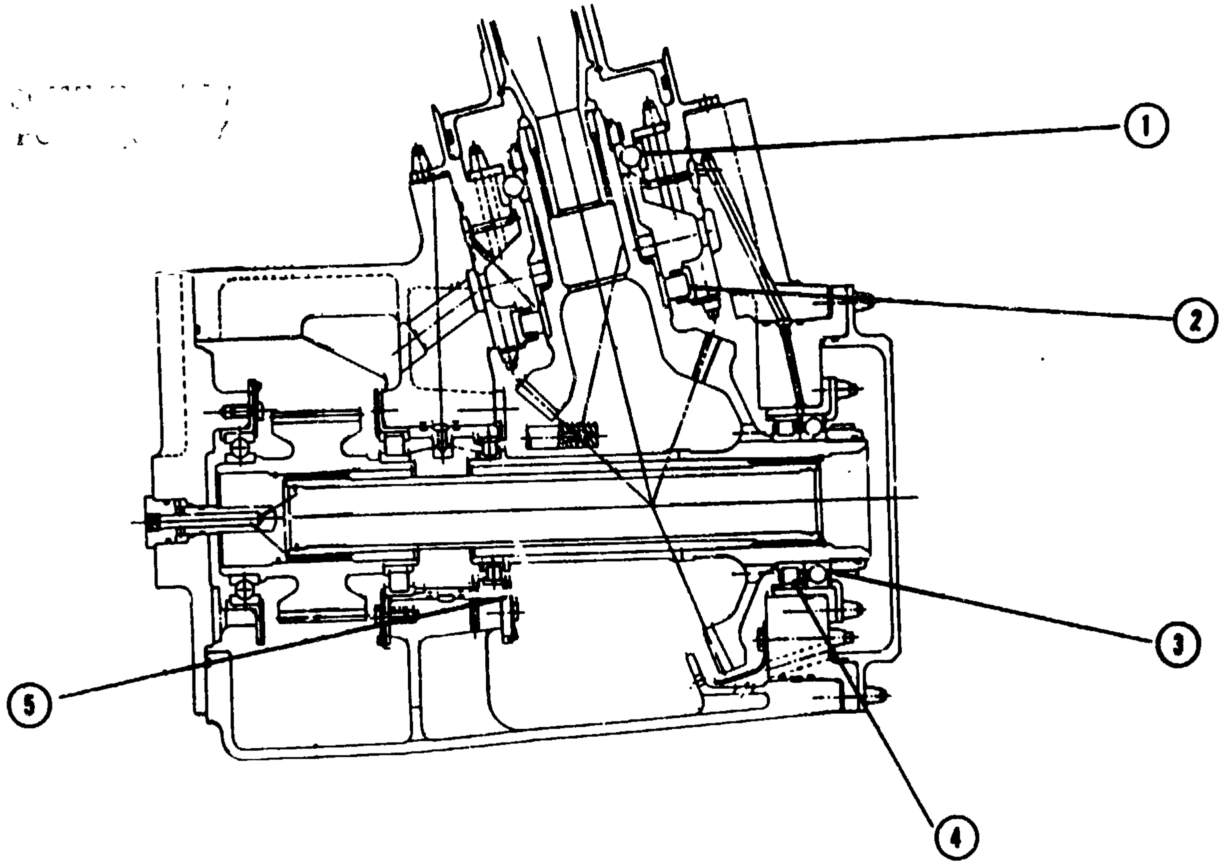


POSITION	7	8	9	10
T/O rpm	11000	11000	7550	7550
CML-N (lb)	423(95)	423(95)	391(88)	200(45)
LIFE-hr	$43x10^6$	$.6x10^6$	$1.9x10^6$	$.4x10^6$

Figure 23. Core/ICLS AGB Spur Gear Bearing Design Summary.

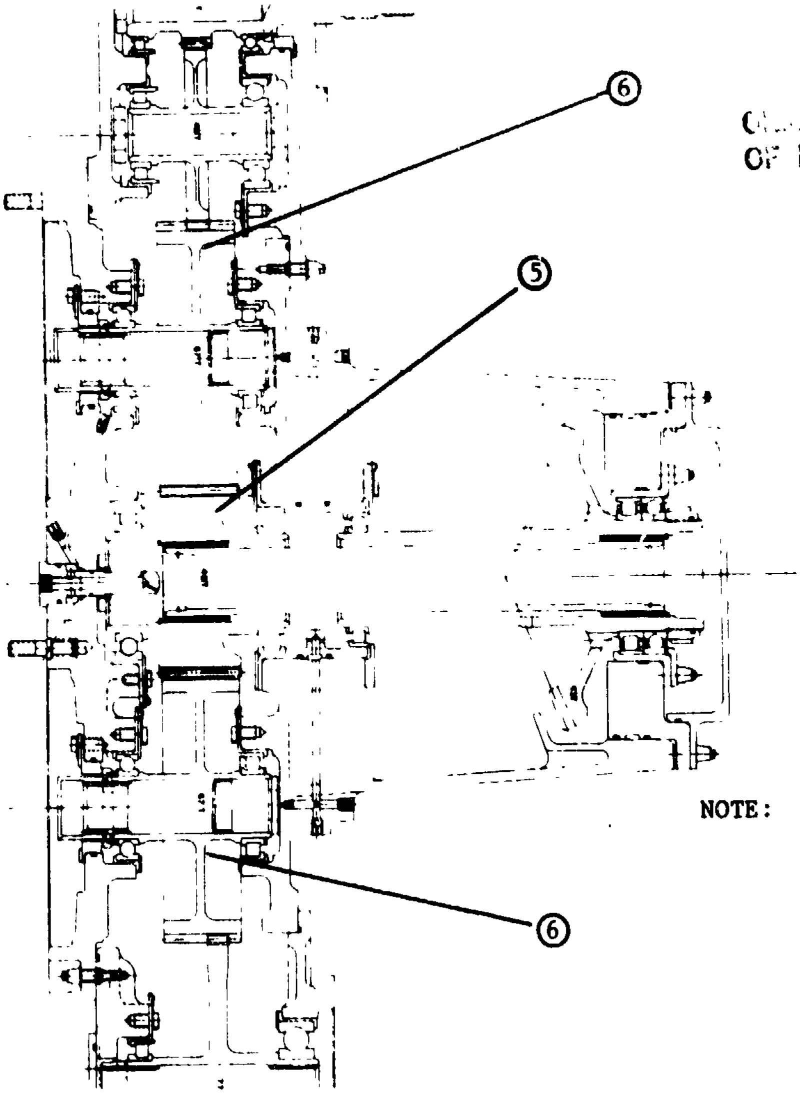


POSITION	①	②	③	④	⑤
SIZE	111	114	014	014	012
T/O rpm	18194	18194	11000	11000	11000
T/O RAD/AX (LBS)N	(152/-51) 676/.227	(372) 1655	(0/126) 0/560	(124) 552	(93) 414
B_{10} LIFE (hrs)	$.28x10^6$	$.28x10^6$	$2.1x10^6$	$9.0x10^6$	$3.8x10^6$

Figure 24. Core/ICLS AGB Bevel Gear Bearing Design Summary.

TABLE IV. CORE/ICLS AGB BEVEL GEAR DESIGN SUMMARY

• GEAR TOOTH NUMBERS		
– PINION	26	
– GEAR	43	
• FACE WIDTH	(1.10 in) 27.9 mm	
• DIAMETRAL PITCH	5.60	
• PRESSURE ANGLE	22.50°	
• SPIRAL ANGLE	35.0°	
• SHAFT ANGLE	101° 57'	
• MAX SURFACE SPEED	(22,100 fpm) 6736.1 m/min	
• BENDING STRESS		
– START (2000 STARTS)	(38,270 psi)	263.8 MPa
– MAX. ACCESSORY LOAD	(8450 psi	37500 psi ALLOWABLE)
(FPS LOADS)	58.3 MPa	258.5 MPa ALLOWABLE
• COMPRESSIVE STRESS		
– START	(227,300 psi	250,000 psi ALLOWABLE)
	1567 MPa	1723 MPa ALLOWABLE
– MAX ACCESSORY LOAD	(106,800 psi	250,000 psi ALLOWABLE)
(FPS LOADS)	736.3 MPa	1723 MPa ALLOWABLE
• SCORING	Δ T IS LOW (NO PROBLEM)	



NOTE: ACCESSORY SPUR GEAR STRESSES OUTSIDE THE STARTER DRIVE TRAIN ARE LOW (MOSTLY EXISTING HDWE. FROM CF6)

- **GEAR TOOTH NUMBERS**

5	46	
6	67	
FACE WIDTH	(2.0 in.) 50.8 mm	
DIAMETRAL PITCH	10	
PRESSURE ANGLE	20°	
BENDING STRESS		
START (2000 STARTS)	(71400 psi)	492.2 MPa
MAX ACCESSORY LOAD	(11100 psi)	76.5 MPa
COMPRESSIVE STRESS		
START	(190000 psi)	1309.9 MPa
MAX ACCESSORY LOAD	(74800 psi)	515.7 MPa
SCORING	ΔT IS LOW	

Figure 25. Core ICLS AGB Spur Gear Design Summary.

sure angle and a 35° spiral angle which gives a relatively high face contact ratio.

The new spur gear's diametral pitch must be 10 to mesh with the existing gears. The face width of the starter gears was determined from the starting requirements. 2000 starts are attainable without exceeding the allowable bending stress. The face width of the starter gears is 50.8 mm (2.0 in.) and the 46T input gear is provided with crowning of .01225-.02940 mm (.0005-.0012 in.).

Only a minimum amount of instrumentation will be required for the drive system. Internal cavity pressure and scavenge temperatures will be measured. Gearbox surface temperatures will also be read.

IV. LUBE SYSTEM

The lube system schematic for the Core engine is shown in Figure 26 and is typical of systems used in other GE engines.

Lubrication and cooling is supplied to each sump and to the gearbox by a single supply element and is scavenged utilizing separate elements. The flow distribution to each component and the capacity of the scavenge elements are shown in Table V.

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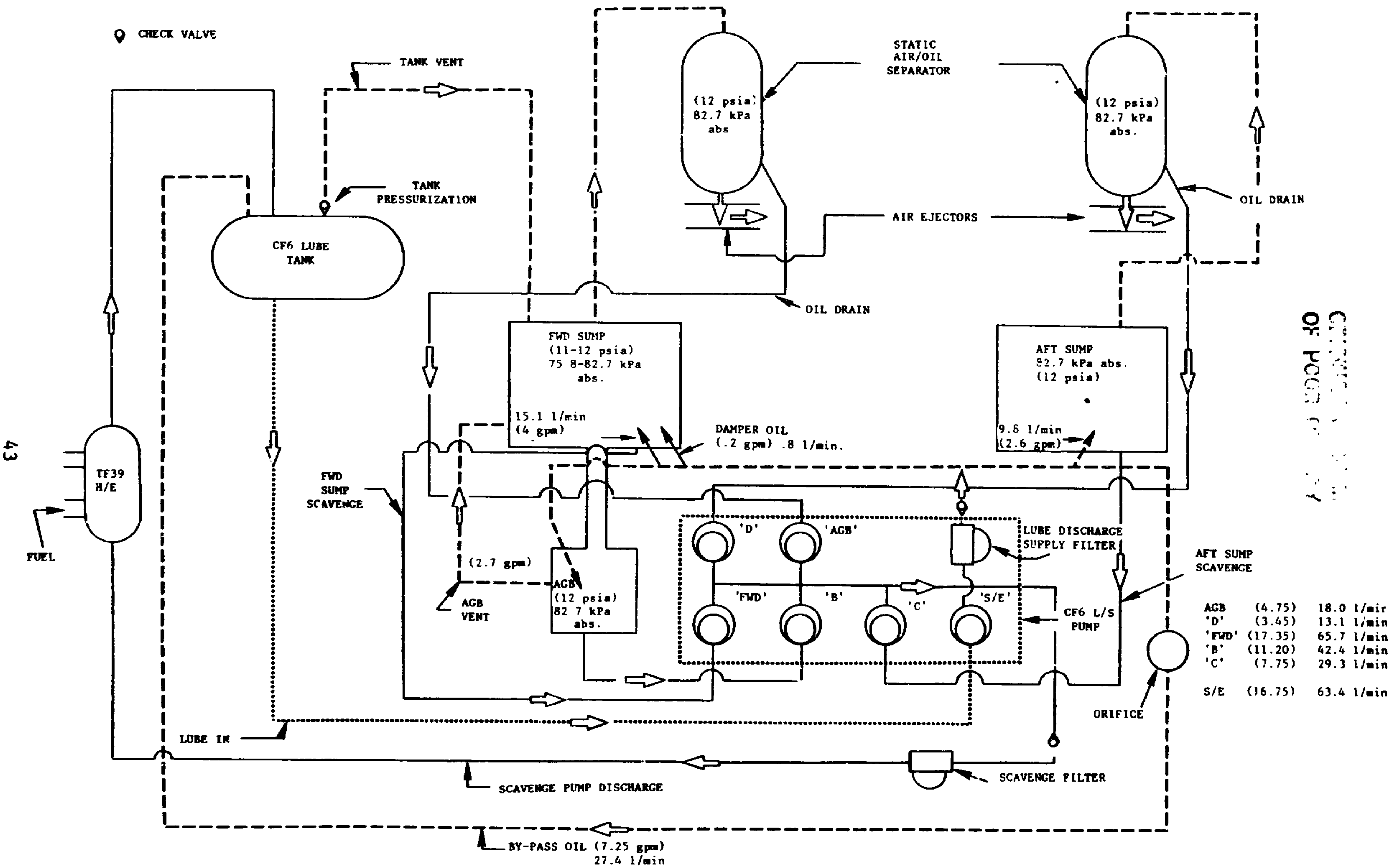


Figure 26. Core Lube System Schematic.

TABLE V. LUBE SUPPLY VS. SCAVENGE PUMP CAPACITY.

Component	Lube Supplied	Scavenge Capacity
Fwd Sump	15.9 liter/min (4.2 gpm)	$65.5 \frac{\text{liter}}{\text{min.}}$ = (17.3 gpm)
Aft Sump	9.8 liter/min (2.6 gpm)	$29.1 \frac{\text{liter}}{\text{min.}}$ = (7.7 gpm)
AGB Gearbox	10.2 liter/min (2.7 gpm)	$43.4 \frac{\text{liter}}{\text{min.}}$ = (11.2 gpm)
Bypassed Oil	27.6 liter/min (7.3 gpm)	N/A
	63.5 liter/min (16.8 gpm)	

From the table above it can be seen that the capacity of the existing supply pump being used is greater than that required by the engine. The "extra" oil will be bypassed back to the lube tank as is shown in the lube system schematic. The above table also shows that the minimum scavenge ratio is 2.98 which should be more than adequate according to GE design practices.

Oil filters are used on both the supply and scavenge side of the lube system. The supply filter protects the sumps and gearbox from contamination and the scavenge filter protects the heat exchanger and the lube tank. Each scavenge element also has an inlet screen to protect the pumping elements from larger debris.

Both the lube tank and AGB are vented to the Forward Sump. The AGB vent is used to balance the pressure between the AGB and the Forward; the vent line from the lube tank relieves the tank pressure. There is a tank pressurization valve to maintain the tank pressure at about 68.95 kPa (10 psi) above ambient pressure.

Check valves are located in the lube and supply side to prevent back-flow of oil into the engine sumps, causing flooding, at engine shutdown.

The forward and aft sumps are vented to externally mounted static air/oil separators. Any oil collected in these separators is pumped back into the system through pumping elements in the scavenge pump. These elements have the capability of pumping 13.1 liter/min (3.45 gpm) and 18.0 liter/min (4.75 gpm) respectively. At the outlets to the air/oil separator air ejectors will be used to provide the capability of lowering the sump pressure below the engine inlet pressure. For engine inlet pressures of sea level static, the sumps will be maintained at approximately 68.9 kPa absolute (10 psia).

The lube system will be monitored with appropriate instrumentation which will include:

- Lube supply temperature and pressure
- Scavenge temperature from forward and aft sump and the AGB
- Combined scavenge temperature and pressure
- Supply & scavenge filter P's

V. SECONDARY AIR SYSTEMS AND ROTOR THRUST

A computer analysis has been made of the secondary air system and the engine rotor thrust reacted by the core thrust bearing. This analysis considers the seal diameters and the blade and drum loads for the compressor and turbine rotors at the various test conditions for the engine. Figures 27 and 28 show pressures and flows in the area of the forward and the aft sump respectively for the Core engine operating at ambient inlet conditions with A8 set at nominal area.

The sumps pressure will be lowered below atmospheric to keep the sump seal flow in the proper direction. Sump pressure at ambient inlet conditions will be 68.9-75.8 kPa absolute (10-11 psia). With the sumps maintained at this pressure, the external air/oil separators will flow at .02 kg/sec (.05 lb/sec) and .03 kg/sec (.07 lb/sec) for the forward and aft sump respectively.

For the instrumentation slip ring cooling, .086-.099 kg/sec (.19-.22 lb/sec) of shop air at 172 kPa (25 psia) will be supplied and vented overboard to ambient pressure. The flow of compressor rotor cooling air has been established at .22 kg/sec (.49 lb/sec) for this test point but the system can handle up to .41 kg/sec (.90 lb/sec) of cooling air. Compressor rotor cooling air is also vented overboard through the aft sump.

Figure 29 shows the relationship of the core thrust bearing load versus core percent corrected speed. The curve shows rotor thrust calculations for both a 101 kPa absolute (14.7 psia) inlet and a 158 kPa absolute (23 psia) inlet at nominal A8 with the balance piston cavity at 345 kPa absolute (50 psia). The balance piston cavity pressure can be adjusted but for these operating conditions the core thrust bearing load will vary between 11.1 kN (2500 lbs.) and 26.7 kN (6000 lbs.)

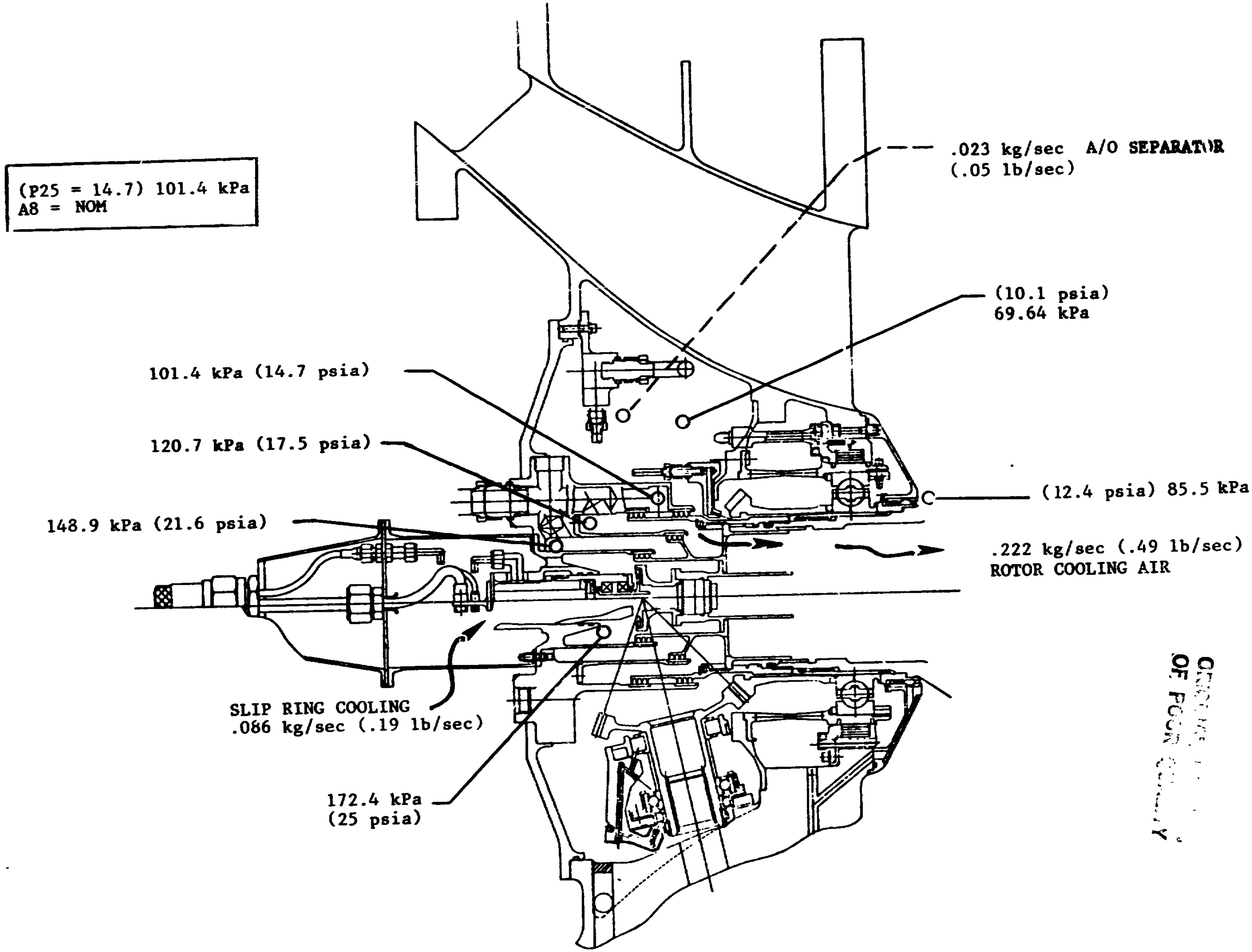


Figure 27. Core Forward Sump Pressures and Airflows.

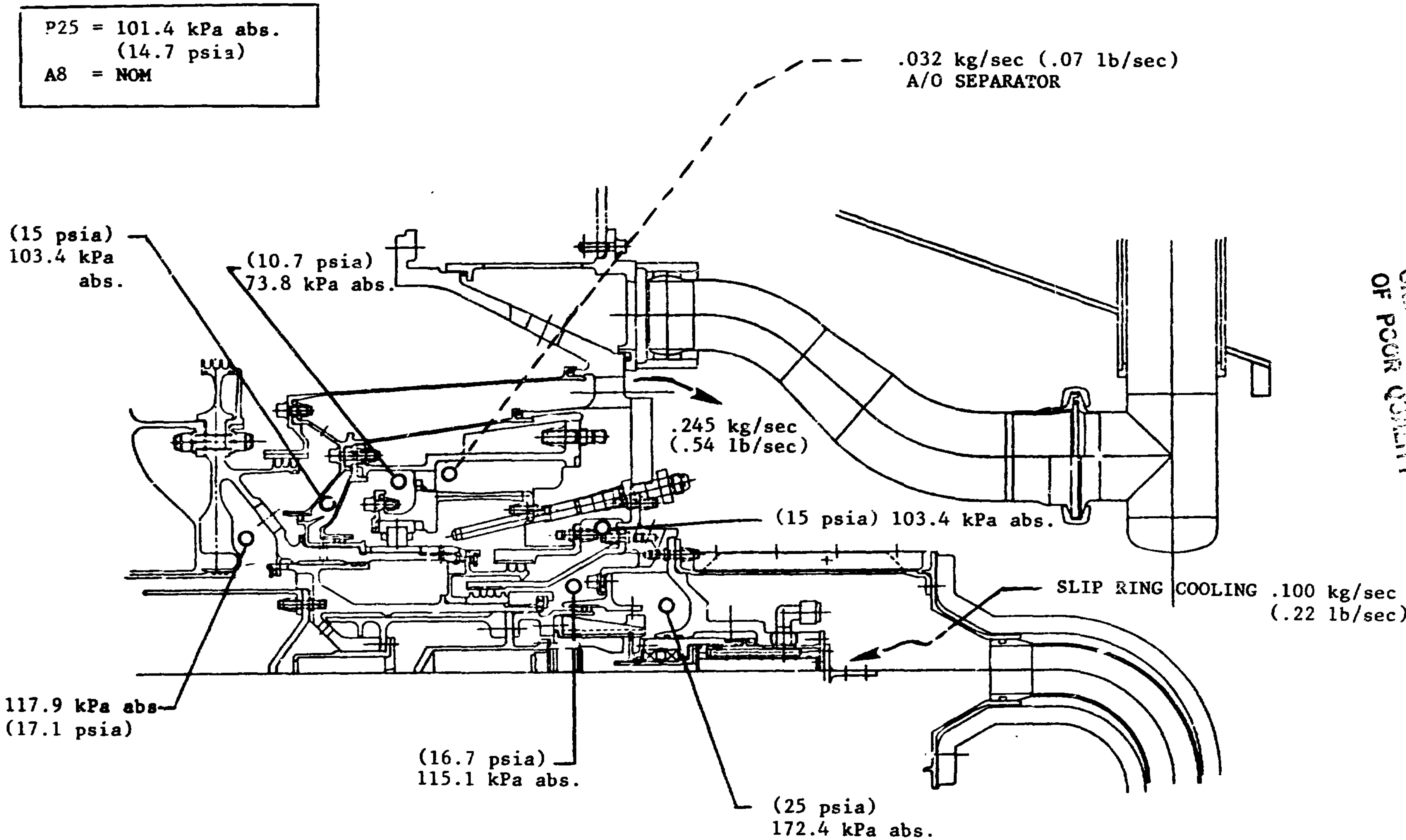
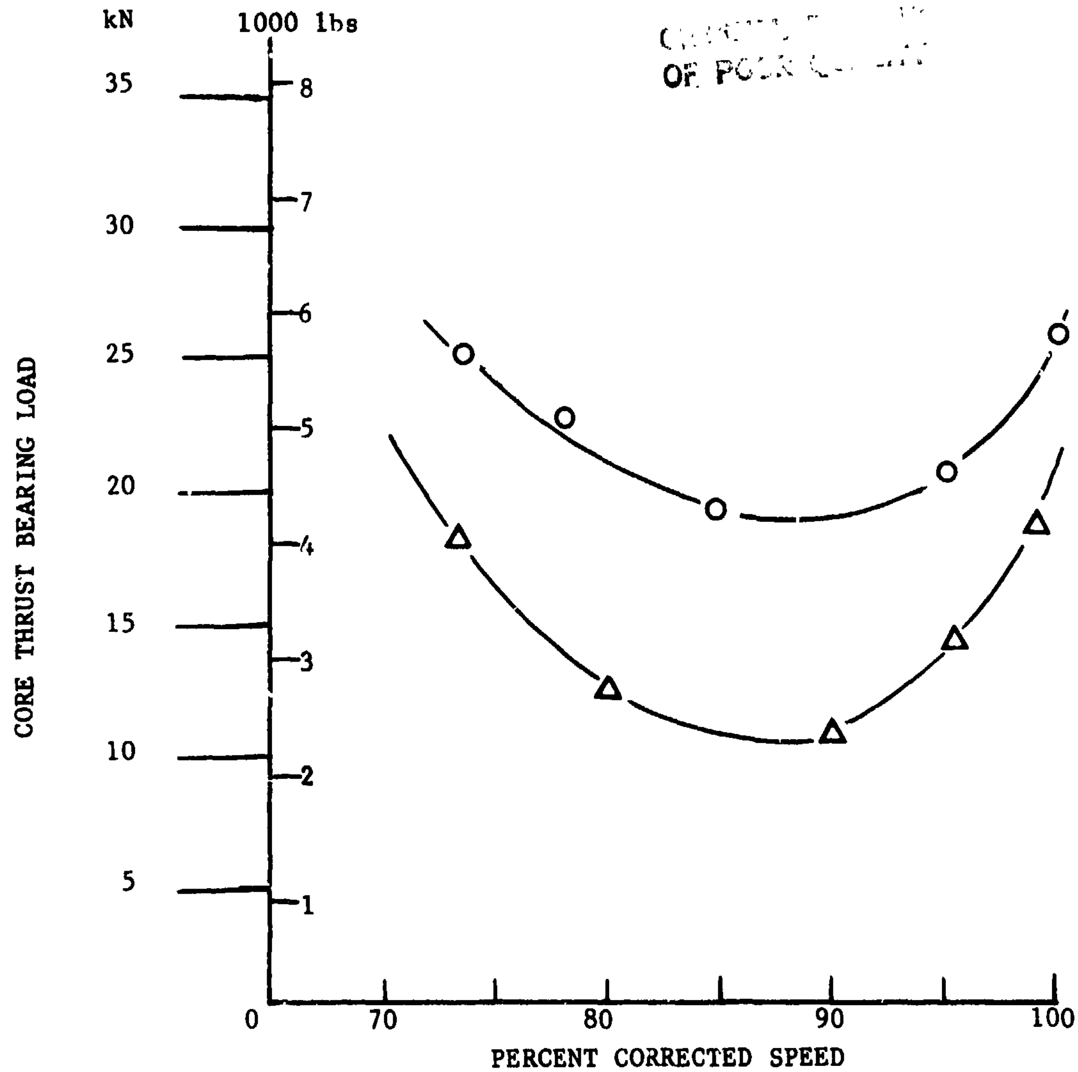


Figure 28. Core Aft Sump Pressures and Airflows.



A8 = NOMINAL
BALANCE PISTON PRESSURE = (50 psia)
344.7 kPa absolute

Figure 29. Core Thrust Bearing Load vs. Percent Corrected Speed

VI. CONFIGURATION DESIGN

The configuration design effort encompasses the following areas:

1. Large pneumatic piping required for compressor and turbine clearance control.
2. Manifolds and valving to facilitate engine starting.
3. Manifolds and piping for customer bleed air extraction.
4. All external lube and fuel lines.
5. All electrical harnesses required.

Items 1 through 3 represent the major design effort because of the physical size and required location of the piping. Figure 30 shows in schematic form the pneumatic piping layout and in Figure 31 this piping is shown configured on the Core engine. Much of the configuration hardware for the Core engine will also be utilized on the ICLS engine.

Customer air can be extracted at the 5th stage and at compressor discharge. To model the effects of removing this air from the engine cycle in the core test, piping is provided to a facility valve which will control the amount of the air bled. Check valves are provided to prevent recirculation of compressor discharge pressure back to the 5th stage manifold.

Air from the 5th stage is also used for clearance control and can either be directed over the aft compressor case for cooling to reduce clearance or through piping outside of the compressor which would allow clearances to increase. This 5th stage air then flows aft and is used for cooling the

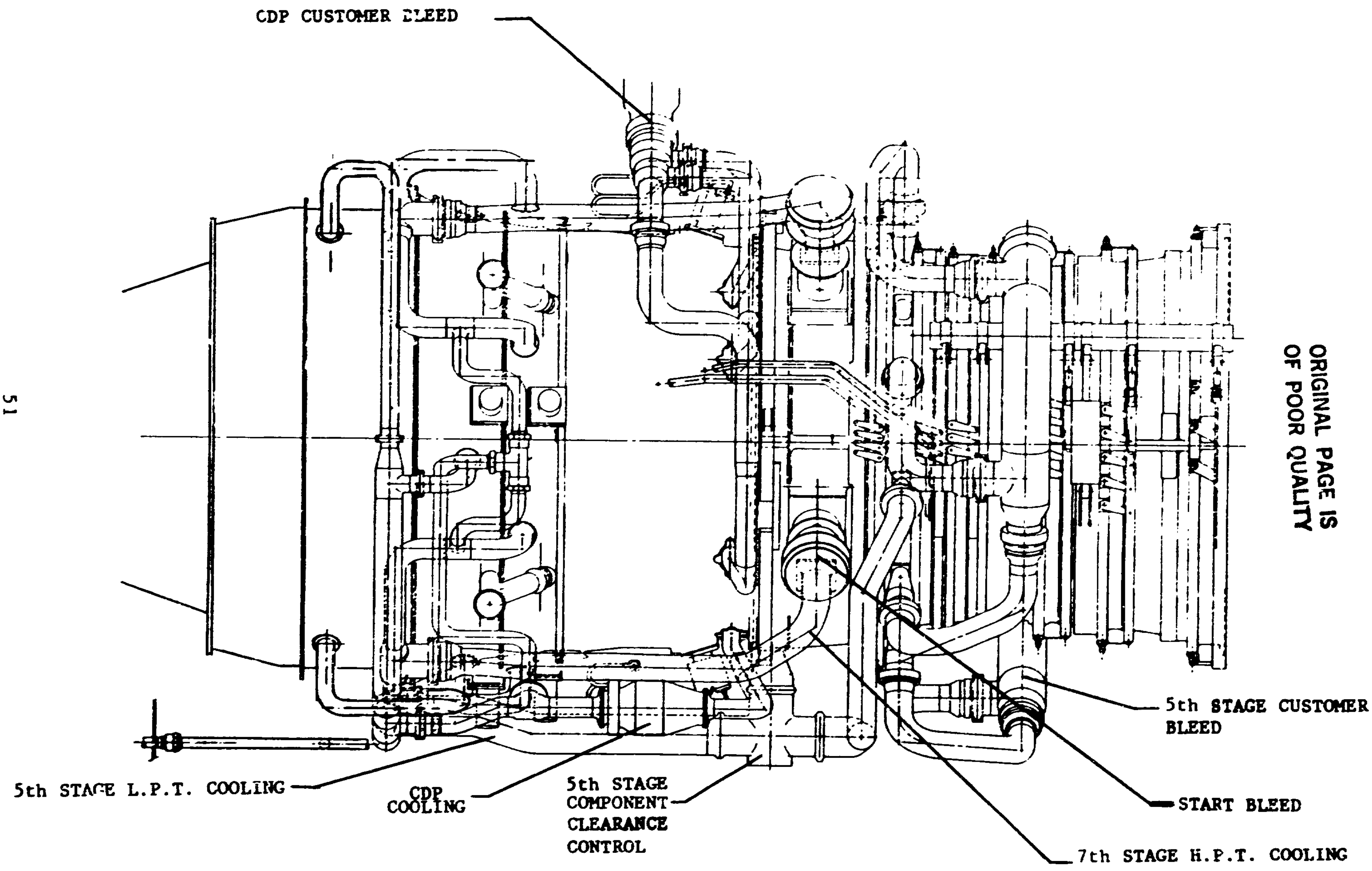


Figure 30. Core Pneumatic System.

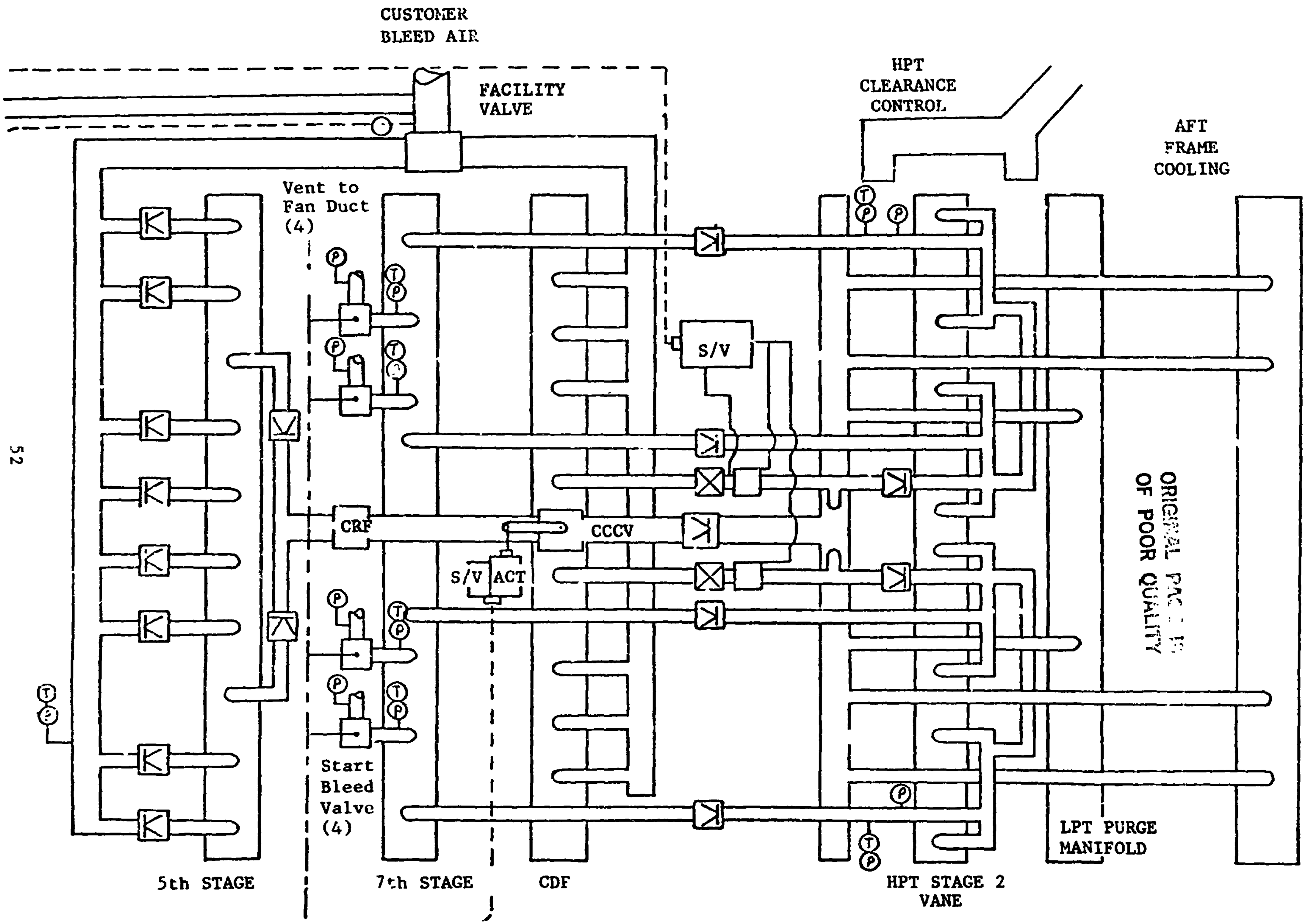


Figure 31. Core Electrical, Pneumatic, and Fuel System Schematic.

low pressure turbine (LPT) vanes in the ICLS engine. In the Core engine this air is used for rear frame cooling.

During starting the 7th stage air is bled through four valves which are vented to atmosphere in the Core engine and into the fan duct in the ICLS engine. Air is also piped from the 7th stage to the high pressure turbine (HPT) second stage for cooling.

If required during starting, additional air can be piped from the CDP manifold through on-off valves and check valves to the HPT second stage to supplement the 7th stage cooling air.

The same design approach, computer techniques, and analysis used on other GE engines have been applied to this piping. A thermal analysis has been completed to identify areas where pipes can be clamped without inducing severe thermal stresses.

At assembly, the vibration characteristics for the major piping will be established. The need for additional restraint will be identified before engine testing begins.

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2. Johnston, R.P., et al., "Energy Efficient Engine-Flight Propulsion System Preliminary Analysis and Design," NASA CR-159583, November 1979.

3. Patt, R.F., "Energy Efficient Engine-Flight Propulsion System Aircraft/ Engine Integration Evaluation," NASA CR-159584, June 1980.

APPENDIX

LIST OF SYMBOLS AND NOMENCLATURE

AGB	- Accessory Gearbox
CDP	- Compressor Discharge Pressure
CML	- Cubic Mean Load
DN	- Bearing Design Parameter (Bore in mm x rpm)
E^3	- Energy Efficient Engine
FPS	- Flight Propulsion System
HPT	- High Pressure Turbine
ICLS	- Integrated Core Low Spool (Turbo Fan Test Engine)
IRC	- Internal Radial Clearance
PTO	- Power Take Off
T	- Teeth
TF39 - 4B	- General Electric Turbo Fan Engine (Approx. 187 kN (42,000 lb.) thrust)
T/O	- Takeoff
β	- Contact Angle
ϕ	- Diameter