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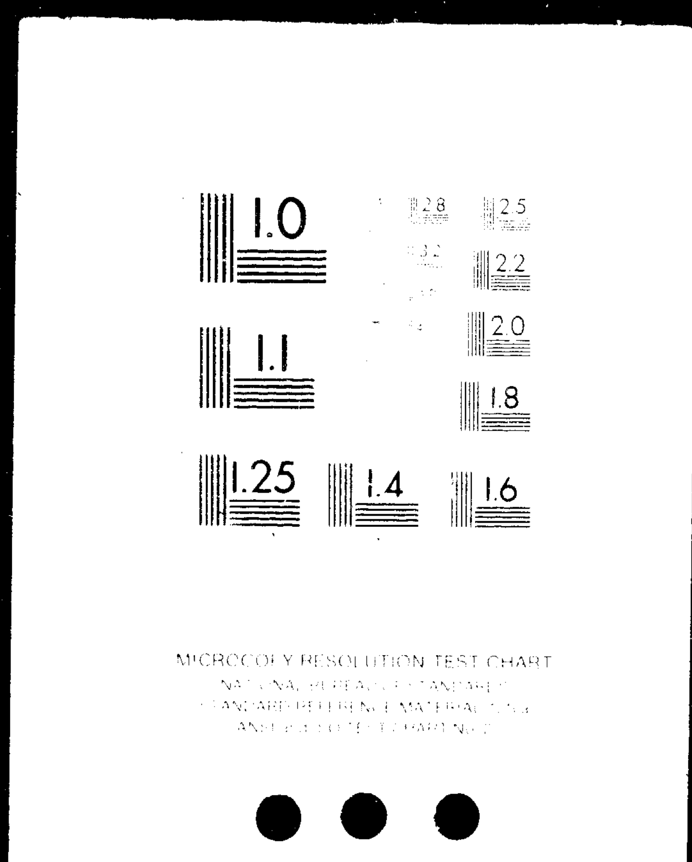

(NASA-CR-174812) STRATIFIED CHARGE ROTARY AIRCRAFT ENGINE TECHNOLOGY ENABLEMENT PROGRAM Final Report (John Deere Technologies International, Inc.) 110 p HC A06/MF A01 CSCL 21E G3/07

N85-21163

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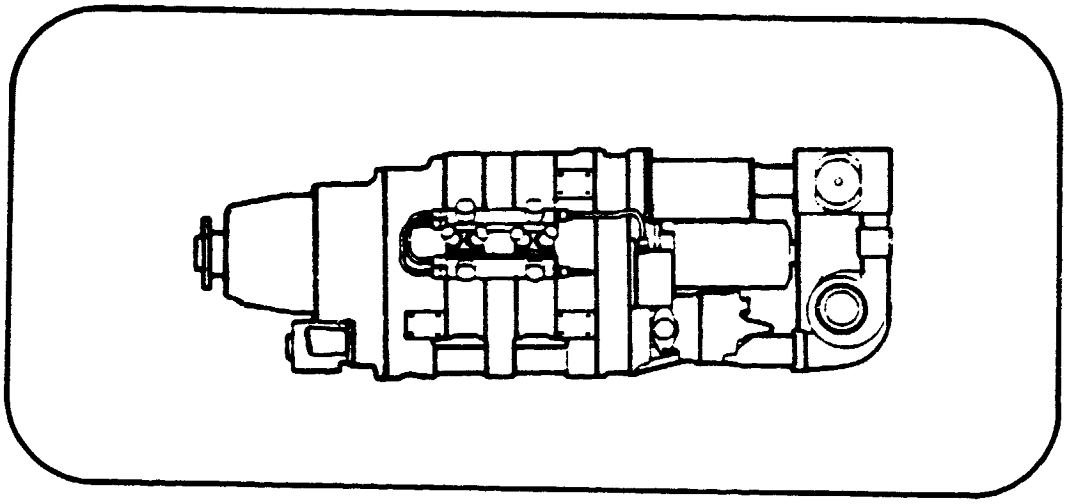
NASA CR-174812
JDTI RED-85-1

STRATIFIED CHARGE ROTARY AIRCRAFT ENGINE TECHNOLOGY ENABLEMENT PROGRAM

FINAL REPORT

By

ROTARY ENGINE DIVISION
JOHN DEERE TECHNOLOGIES INTERNATIONAL, INC.
P. Badgley, C. E. Irion and D. Myers



Prepared for

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
NASA Lewis Research Center
Contract NAS3-23056

January 31, 1985

John Deere Technologies International, Inc. • Rotary Engine Division • Wood-Ridge, N.J. 07075

APRIL 11,1985

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YOU RECENTLY RECEIVED A COPY OF THE "STRATIFIED CHARGE ROTARY AIRCRAFT ENGINE TECHNOLOGY ENABLEMENT PROGRAM" MANUAL.

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ENCLOSED ARE THE MISSING PAGES; PLEASE INSERT THEM TO COMPLETE YOUR MANUAL.

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.
NAS CR-174812		

4. Title and Subtitle

Stratified Charge Rotary Aircraft Engine Technology Enablement Program

5. Report Date

January 1985

6. Performing Organization Code

505-40-68

7. Author(s)

P. R. Badgley, C. E. Irion and D. M. Myers

8. Performing Organization Report No.

JDTI RED-85-1

9. Performing Organization Name and Address

Rotary Engine Division
John Deere Technologies International, Inc.
P. O. Box 128
Wood-Ridge, NJ 07075-0128

10. Work Unit No.

11. Contract or Grant No.

NAS3-23056

12. Sponsoring Agency Name and Address

National Aeronautics and Space Administration
Washington, D.C. 20546

13. Type of Report and Period Covered

Contractor Report

14. Sponsoring Agency Code

15. Supplementary Notes

Project Manager, W. Hady, NASA Lewis Research Center, Cleveland, Ohio 44135

16. Abstract

The multi-fuel stratified charge rotary engine has become a prime candidate for small general and military aviation powerplants. A single rotor, 0.7ℓ/40 in^3 displacement, research rig engine was designed, fabricated and acceptance tested. The research rig engine was designed for operation at high speeds and pressures, i.e., 9,600 RPM, 1,462 KPA (212 psi) BMEP, and 9,653 KPA (1,400 psi) combustion chamber peak pressure providing margin for speed and load excursions above the design requirement for a highly advanced aircraft engine. Preliminary acceptance test data and engine operating characteristics indicate that the single rotor research rig engine is capable of meeting the established design requirements of 120 kW (160 BHP), 8,000 RPM, 1,379 KPA (200 psi) BMEP. The research rig engine, when fully developed, will be a valuable tool for investigating, advanced and highly advanced technology components, and provide an understanding of the stratified charge rotary engine combustion process.

17. Key Words (Suggested by Author(s))

Adv. Engines, Internal Combustion, Aircraft Engines, Stratified Charge Rotary Engine, Fuel Injection, Turbocharging

18. Distribution Statement

Unclassified-Unlimited

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price*
Unclassified	Unclassified	TBD	

* For sale by the National Technical Information Service, Springfield, Virginia 22161

TABLE OF CONTENTS

TABLE OF CONTENTS (Cont.)

TABLE OF CONTENTS (Cont.)

LIST OF FIGURES

LIST OF FIGURES (Cont.)

LIST OF DRAWINGS

LIST OF TABLES

SUMMARY

A Final Report is provided for NASA Contract No. NAS3-23056, Stratified Charge Rotary Aircraft Engine Technology Enablement program.

The contractual effort involved the design, fabrication, assembly and acceptance testing of a single-rotor, research rig engine, complete with appropriate documentation and spare parts. The research rig engine provides a single-rotor stratified charge test engine to support advanced component technology research at contractor and NASA facilities.

The research rig engine is designed for operation at speeds and pressure loadings equivalent to the highly advanced stratified charge rotary aircraft engine defined in earlier NASA programs (NAS3-21285 and NAS3-22140). The actual design permits short-term excursions to conditions higher than those required for a highly advanced rating to permit the investigation of trends through that point.

The contractual work reported on herein was initiated by Curtiss-Wright Corporation under Contract No. NAS3-23056 on December 29, 1982. During the course of that contract, all rights to the Rotary Engine were purchased from Curtiss-Wright by Deere & Company. As of February 1, 1984, the Rotary Engine Division of John Deere Technologies International, Inc., performed the contractual work under subcontract to Curtiss-Wright. On October 18, 1984, with completion of a Novation proceeding, the Rotary Engine Division of John Deere Technologies Int'l, Inc., assumed resonsibility for completion of the contract.

The contractual effort consisted of four discrete tasks:

Task I	Design
Task II	Fabrication
Task III	Assembly and Acceptance Test
Task IV	Reporting

The design effort was completed during September 1983 with approval of the design and authorization to initiate procurement. Fabrication was completed during August 1984 with the assembly occurring during September 1984. Testing was conducted during October 1984.

The initial "run-in" acceptance testing was completed after establishing a spark plug/injection nozzle combination resulting in consistent firing. Four different spark plugs and two main injection nozzles were run to accomplish this. Checkout and acceptance testing were accomplished from 2000 to 8000 rpm. The maximum power level achieved was 79 kW (106 hp) at 7500 rpm. A total test time of 72 hours, including motoring, was accumulated. All running was done with one tubocharger configuration. No engine hardware durability problems were encountered during the initial run-in period.

The research rig engine when fully developed will be a durable vehicle for the investigation of advanced and highly advanced technology components. Further, the research rig engine can lead to improved understanding of the stratified charge rotary engine combustion process and definition of design parameters for future advanced and highly advanced aircraft engines.

The Deere & Co. designation for the research rig is 1007R, with the first digit indicating a single rotor and the last digit indicating 0.7-liter displacement. The original NASA contract designation was RC1-XT prior to selection of the displacement at 0.7 liter (40 $in.^3$).

INTRODUCTION

Background

Future requirements for changing environmental regulations, increased fuel economy, and multi-fuel capability have created a need for completely new engines for future lightweight general aviation and military aircraft. The Stratified Charge Omnivorous Rotary Engine (SCORE) has become a prime power plant candidate for these applications because of predicted advantages in fuel economy, exhaust emissions, compactness, high specific power, improved fuel tolerance, low vibration and noise levels, and producibility. NASA/C-W Contract NAS3-21285 and subsequent contractual studies by Cessna and Beech concluded that advanced and highly advanced Stratified Charge Rotary Engines can meet the future power requirements.

The objective and scope of this contract effort was the design, fabrication, assembly, and acceptance testing of a single-rotor research engine test rig, complete with appropriate documentation and spare parts. The rig will permit the mapping of key variables over a broad range of speeds and loads and the investigation of and definition of the best parameters for the advanced and highly advanced aircraft engines.

Goals and Approach

The Stratified Charge Omnivorous Rotary Engine is an attractive alternative engine candidate for future General Aviation aircraft since it can burn a broad range of diesel, kerosene, and gasoline-type fuels very efficiently. Its well-known advantages of broad fuel tolerance and good fuel ecomony, coupled with the weight and package-volume reductions achievable via advanced design and technology, indicate that a Stratified Charge Omnivorous Rotary Engine can be a competitive power plant for General Aviation aircraft.

The primary goal in this contractual effort is the provision of a research rig engine suitable for exploration and evaluation of advanced and highly advanced technologies applied to Stratified Charge Rotary Aircraft Engines.

The approach has been to design, procure, and conduct initial run-in testing on a single-rotor research rig based upon conceptual designs from NAS3-21285. The overall design emphasizes strength, structural durability, and maximum likelihood of trouble-free mechanical operation under prolonged high-output running and of surviving brief excursions of overspeed and loan. The design has the essential geometric, physical, and operating pressure (200 psi BMEP) and thermal loading characteristics of the 320 hp take-off, 250 hp cruise two-rotor highly advanced Stratified Charge Rotary Aircraft Engine concept from NAS3-21285 (Figure 2.1).

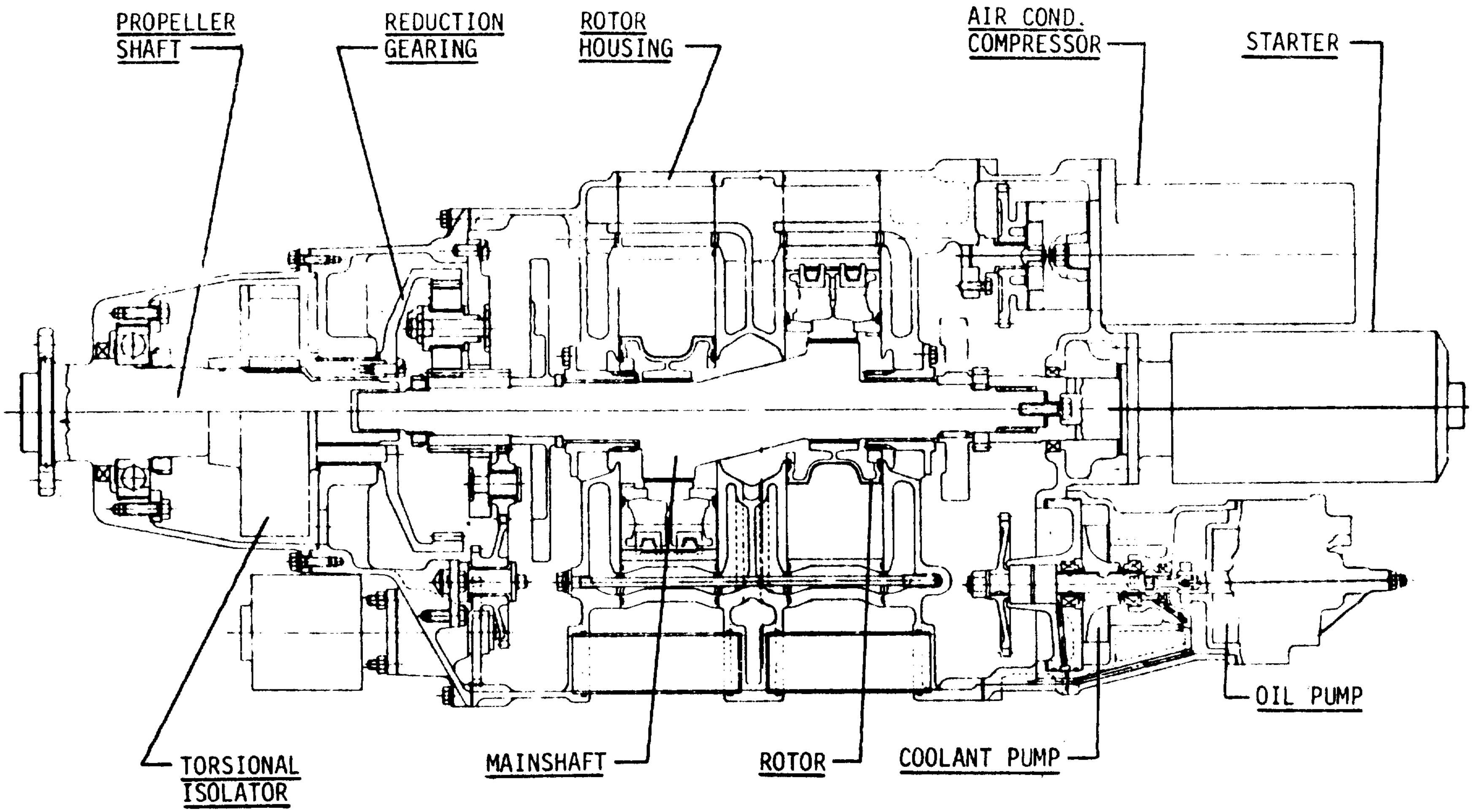

Figure 2.1. Longitudinal Section Two-Rotor Highly Advanced 320-hp Rotary Combustion Aircraft Engine

Engine Characteristics

The direct-injected stratified charge rotary engine is a competitive alternate to the advanced diesel and the shaft gas turbine as a future aircraft power plant. Relative to the diesel the rotary has inherently lower NOx emissions and operates on a wide range of fuels, without regard for self-ignition qualities or volatility. A reasonable probability of lower unregulated particulate emissions complements the basic smoothness, power density, and multi-fuel advantages. The gas turbine's loss of power density when regenerated to improve full-range BSFC and its down-scaling penalty with the smaller sizes become significant in this power range. Additionally, turboprop engine cost and performance hinge on breakthroughs in ceramic technology which will have to include significant structural gains to insure aircraft-level reliability. Briefly stated, stratified charge engines that burn lean can achieve automotive diesel fuel efficiency levels. The direct-injected unthrottled rotary engine is the only stratified charge engine configuration which can operate as lean as a diesel. To do this throughout the complete operating range a varying air velocity field must be induced, allowing the injected fuel to be effectively "layered" (or stratified). An ignitable mixture of fuel and air is consistently developed at the spark plug where the "triggering" combustion is initiated, with a significantly leaner mixture ratio at all other points in the combustion chamber.

The Stratified Charge Omnivorous Rotary Engine offers high power density because of its geometry and related kinematics which are uniquely compatible to direct-injected stratified charge combustion. The moving rotor in a rotary engine, regardless of the type of combustion employed, always moves the charge air past the stationary location of the spark plug and fuel-injection nozzles, an inherent function of its geometry. This develops the necessary flow distribution for stratification without the added friction and pumping losses associated with a reciprocating engine in which this flow pattern must be generated. Multi-fuel capability is obtained by fuel injection at the approximate combustion rate facilitated by the geometry of the rotary engine wherein the combustion chamber form varies with shaft rotation. Detailed descriptions of these approaches can be found in a prior NASA Final Report, NASA CR-165398. Figure 2.2 shows a typical stratified charge rotary engine fuel injection and ignition system similar to that incorporated in the 1007R research rig engine configuration.

Utilization

The single-rotor research rig engine provides a versatile tool for technology efforts toward highly advanced aircraft engines. Its design permits operation to high speeds and pressures as noted below:

Maximum Speed - 9600 rpm

Brake Mean Effective Pressure (BMEP) - 1462 kPa (212 psi)

Combustion Chamber Peak Pressure - 9653 kPa (1400 psi)

Engine testing at contractor and NASA facilities will permit detailed investigation of the component technologies identified by NASA that are required for obtaining high efficiency, high output, multi-fuel, and altitude capability and durability.

- High Speed, High Pressure Stratified Charge Combustion Technology
  - Multi-fuel
  - Efficiency
  - Output
- Thermal/Mechanical Technology (minimum heat rejection operation)
  - Efficiency
  - Output
  - Multi-fuel
  - Durability

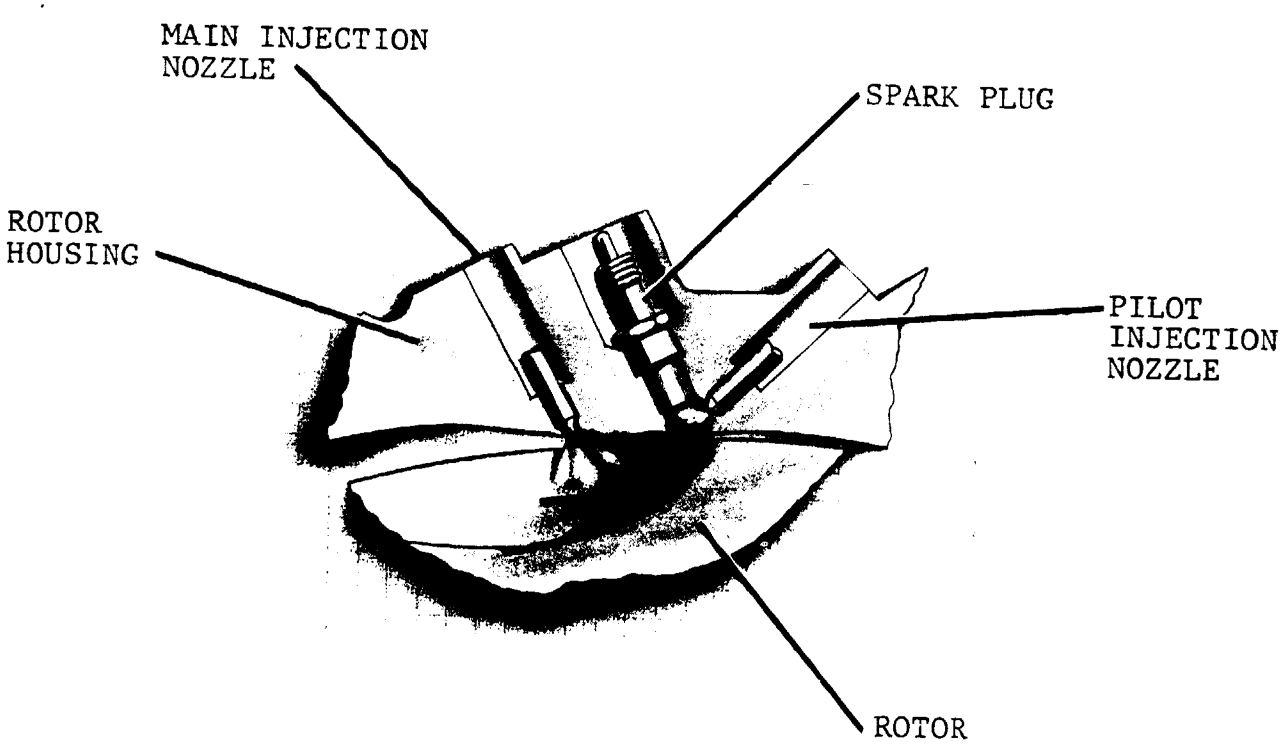

Figure 2.2. Typical Stratified Charge Rotary Engine Fuel Injection/Ignition System

- <u>Sealing and Tribology</u>
  - Durability
  - Efficiency
- <u>Turbocharging/Turbocompounding Technology</u>
  - Output
  - Efficiency
  - Durability
  - Altitude capability

DESIGN

A report summarizing the design of the NASA Stratified Charge Rotary Aircraft Engine Technology Enablement Research Engine Test Rig, dated December 16, 1983, was submitted to and has been accepted by NASA. An abridgement of that report is included herein.

Criteria

Design criteria, including an estimated life requirement, duty cycle, eventual TBO level for a flight engine, and safety factor for major components (end housings, rotor, mainshaft, and rotor housings) were generated. These provided guidance and defined requirements to the design function. The design criteria are shown below.

a. Life

The rig will be used for an estimated 700 test hours to complete acceptance testing in the proposal dated August 16, 1982, and follow-on advanced technology investigations.

b. Duty Cycle

It is estimated that 80% of the tests will be at a high power level and 20% will vary over the full range of engine power capability.

c. Reference Flight Engine TBO Target

2000 hours.

d. Rig Safety Factor Objectives (Goodman Diagrams)

End Housings:	1.5
Rotor:	1.5
Mainshaft:	2.5
Rotor Housing:	1.5

In addition, a minimum of 300 predicted thermal load cycles. Each cycle consists of changing engine temperatures from ambient to operating design point conditions and back to stabilized temperatures at ambient.

Engine Sizing/Performance

a. Basic Geometric Data

Model Designation	1007R		
Actual Displacement cm^3 ($in.^3$)		662.427	(40.424)
Eccentricity, e . . . mm (in.)		15.418	(0.607)
k factor		6.9	
W/E, width/eccentricity . . .		5.0	

Oversize, a . . . . . . mm (in.)		0.813	(0.032)
Width of Rotor Housing mm (in.)		77.114	(3.036)
Major Axis . . . . . mm (in.)		245.3005	(9.6575)
Minor Axis . . . . . . mm (in.)		183.6166	(7.2290)
Generating Radius, R mm (in.)		106.401	(4.189)

b. Reserve Capacity

To accommodate speed and load excursions and to provide reserve capacity (beyond the desired nominal levels of pressure, speed, and power associated with a 40 in.^{3} power section in the midrange of advanced to highly advanced technology) necessary to permit mapping of key variables over a wide range, the following limiting design objectives vs. contract reference levels were considered:

Parameter	Design Objective	Contract Reference Levels
Displacement cm^3 (in.^{3})	662 (40.4)	662 (40.4)
Speed, crankshaft RPM	9600	8000
BMEP kPa (psi)	1461.69 (212)	1378.95 (200)
Power kw (bhp)	149.14 (200)	119.31 (160)
Peak Pressure MPa (psi)	9.65 (1400)	-
IMEP kPa (psi)	1716.79 (249)	-

c. Basic Performance and Indicator Cards

Figure 3.1 presents estimated performance for the 1007R research rig engine. Figure 3.2 presents friction horsepower estimates. Figures 3.3 and 3.4 present indicator card data for 7.5:1 and 6.5:1 compression ratios, respectively. It should be noted that these curves are projected to technology enablement completion while the data presented in the TEST RESULTS section represent the initial performance of the engine obtained during its first build.

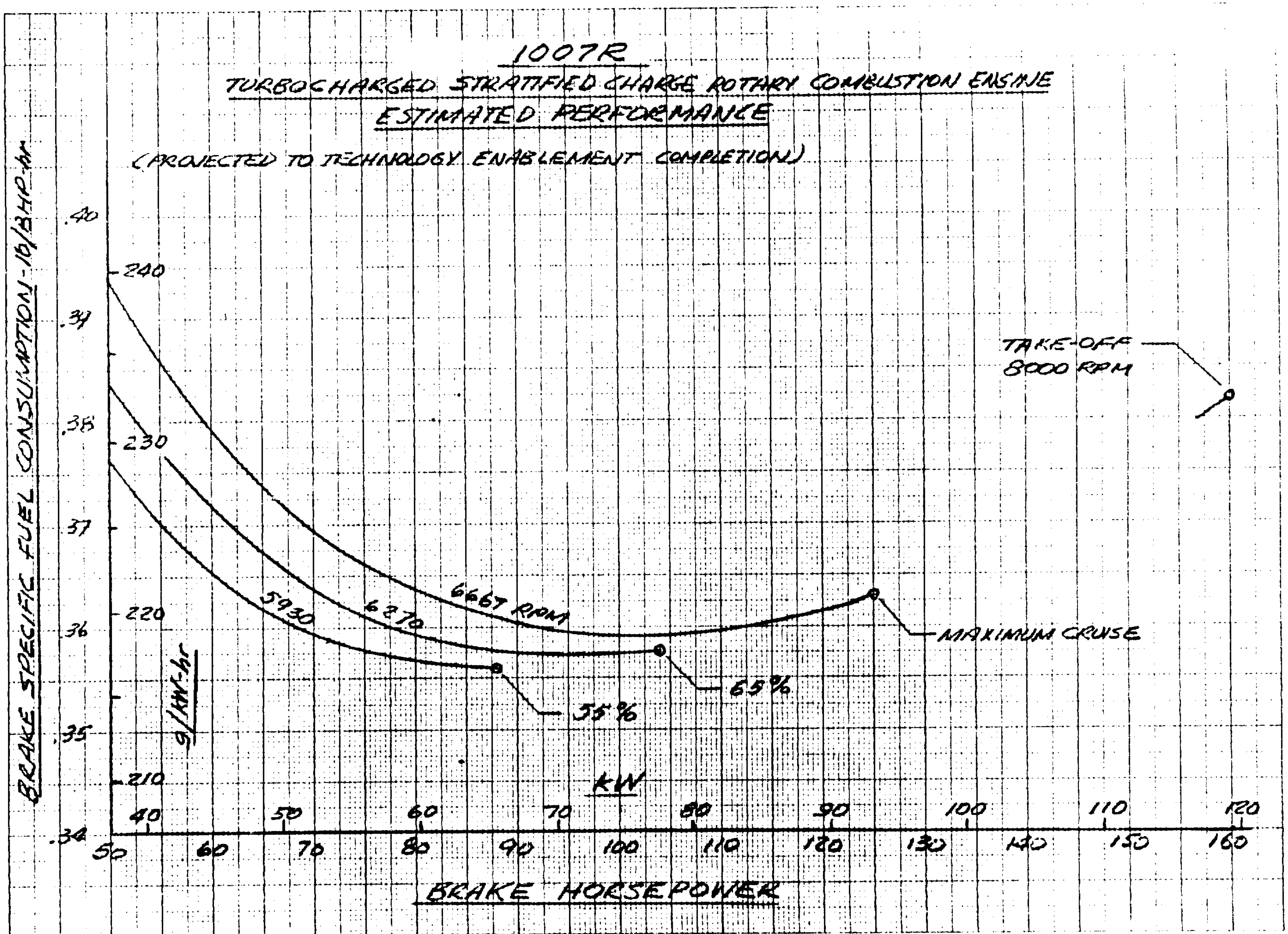

Figure 3.1. 1007R Turbocharged Stratified Charge Rotary Combustion Engine Estimated Performance

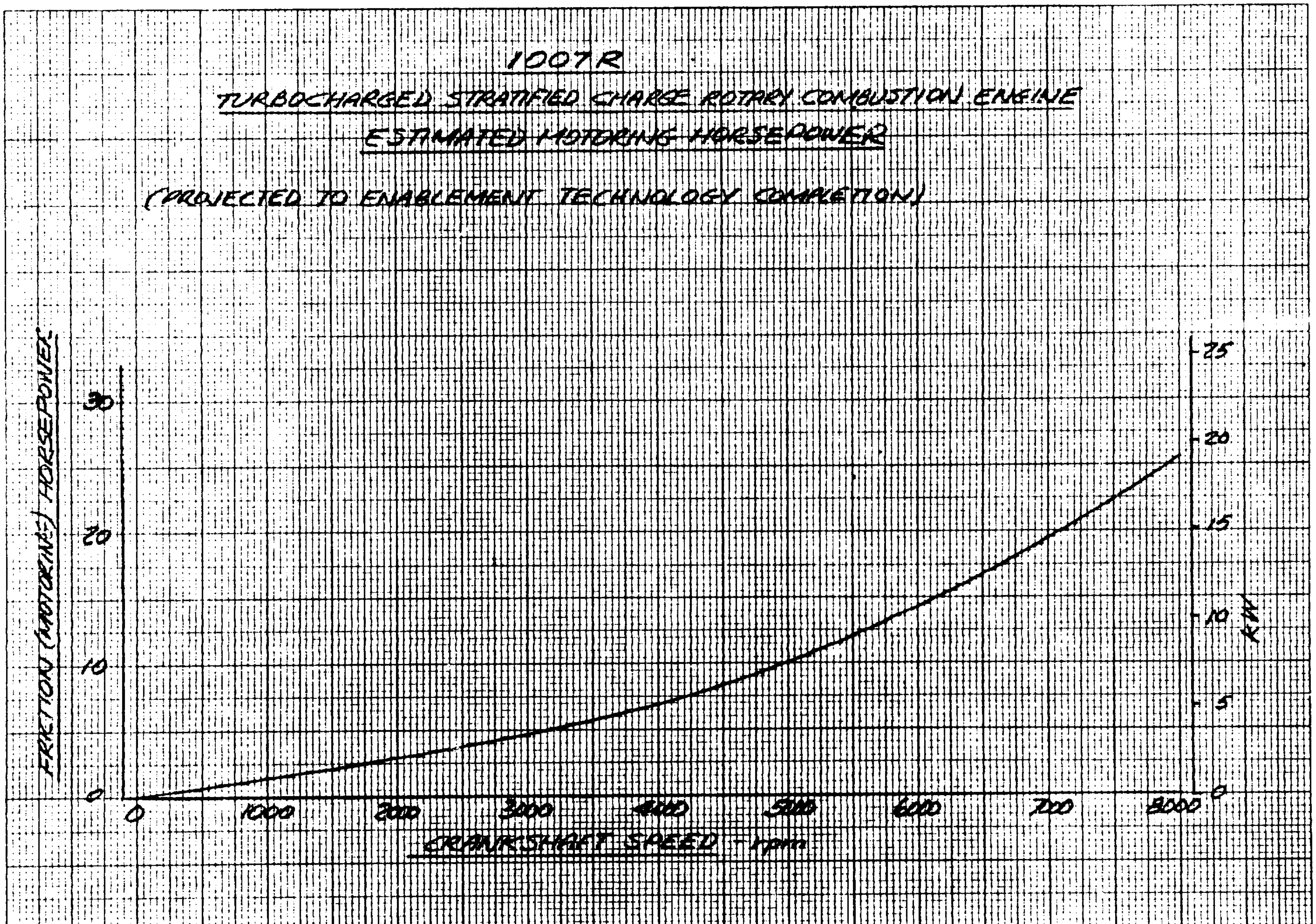

Figure 3.2. 1007R Turbocharged Stratified Charge Rotary Combustion Engine Estimated Motoring Horsepower

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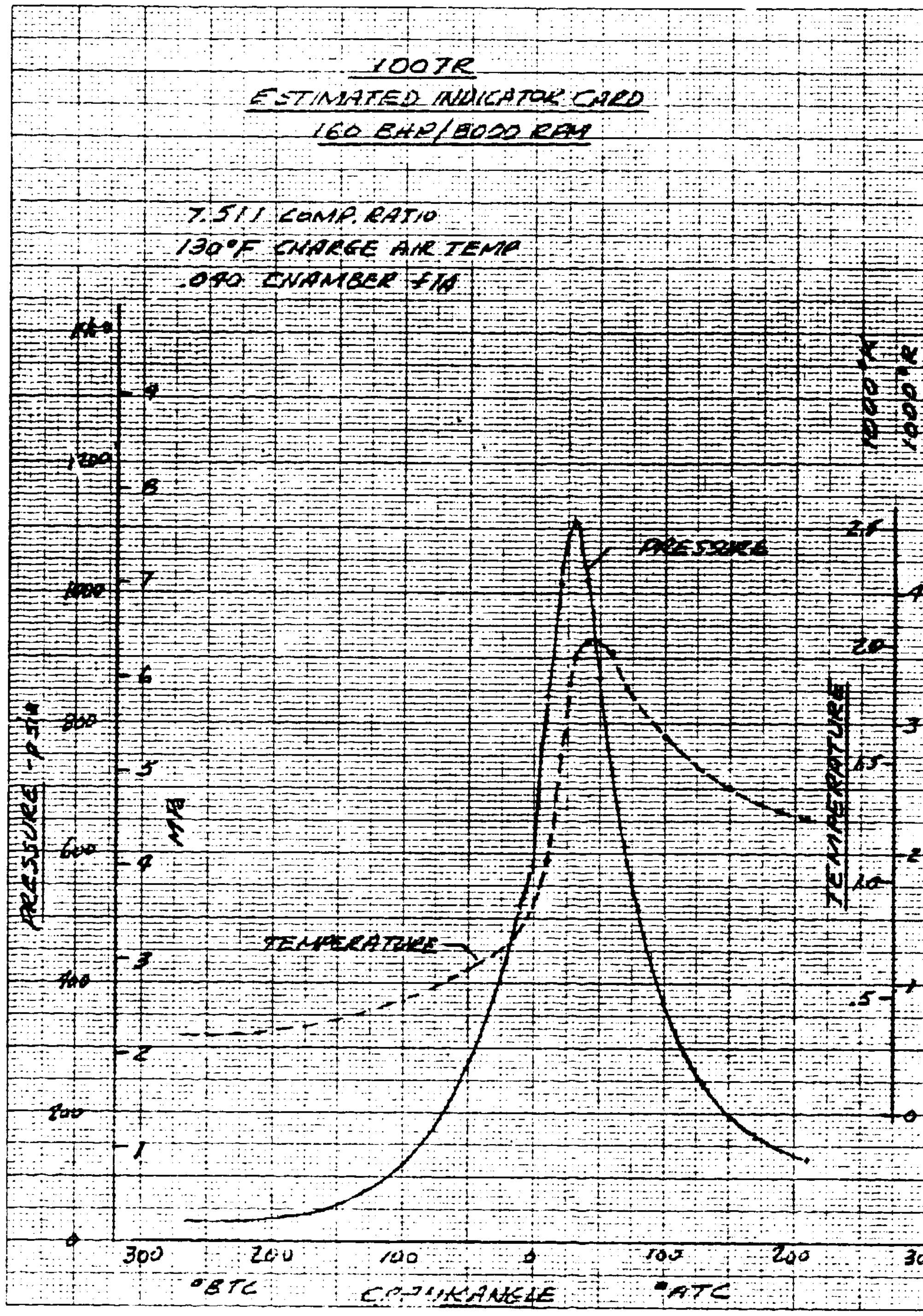


Figure 3.3. 1007R Estimated Indicator Card 160 bhp/8000 rpm

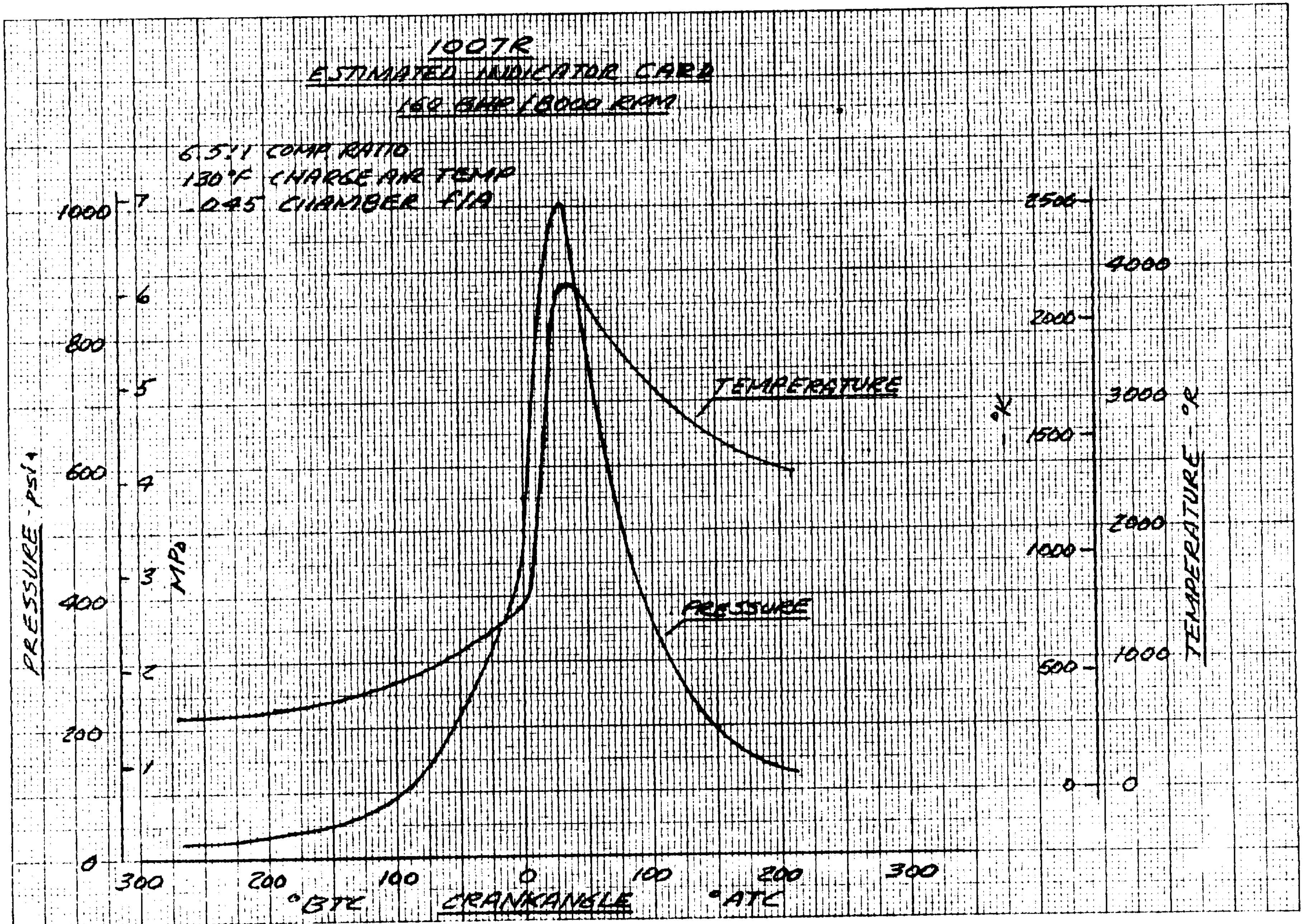


Figure 3.4. 1007R Estimated Indicator Card 160 bhp/8000 rpm

Engine Description

a. Engine Basic Assembly and Installation Drawing

Drawing No. 617000, Sheets 1 and 2, entitled "Basic & Installation 1007R" presents significant internal and external configuration details for the single-rotor research rig.

b. Functional Description - 1007R

The 1007R engine is a Stratified Charge Rotary Combustion Turbocharged engine shown in Drawing 617000. It is a four-stroke cycle, stratified charge, spark ignition, multi-fuel, direct-injected, rotary combustion, single-rotor, liquid-cooled, dry sump, turbocharged internal combustion engine without accessories but capable of complete test-stand testing. The engine consists of the following subsystems:

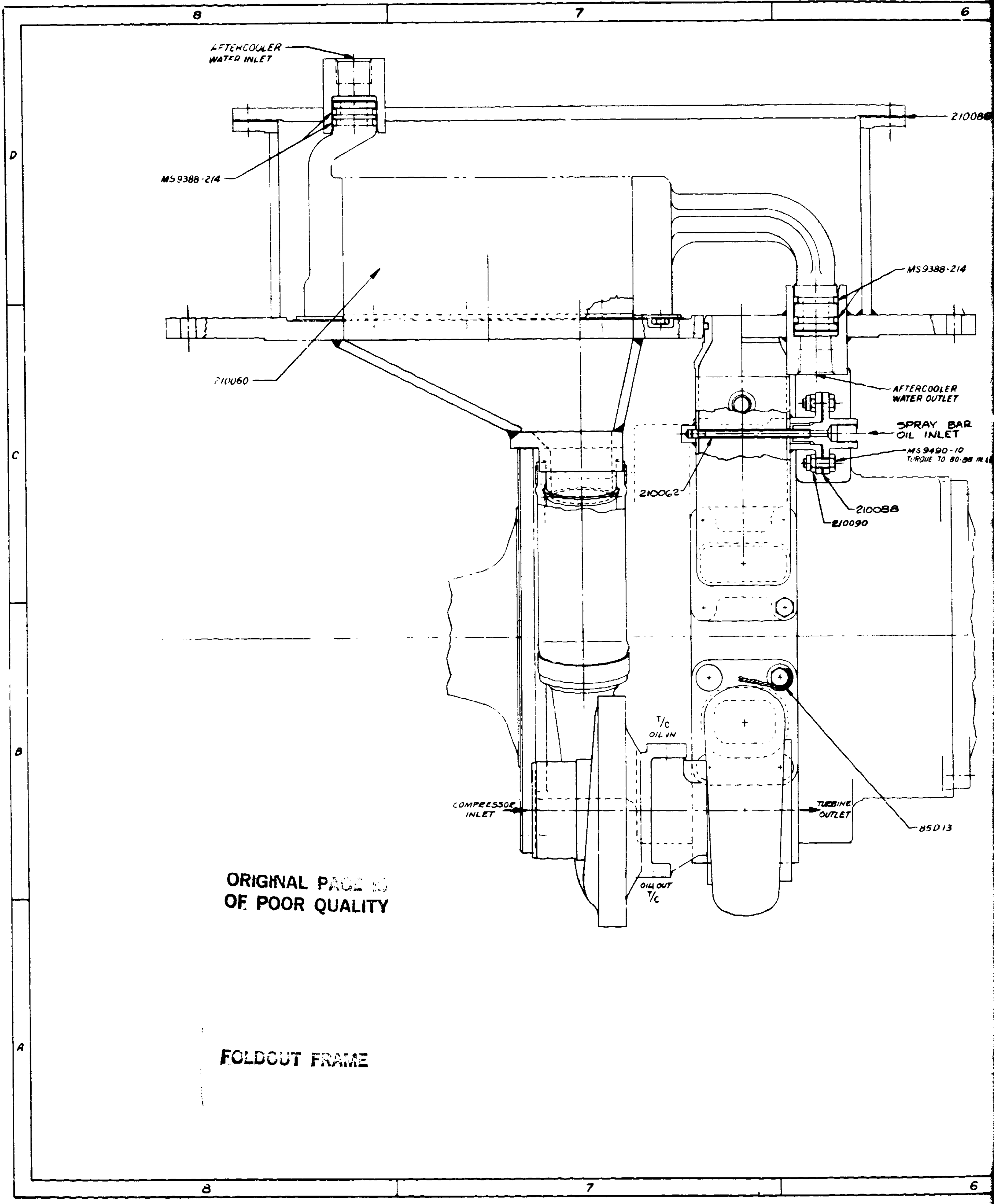
- Power System
- Ignition System
- Cooling System
- Lubrication System
- Turbocharger System
- Fuel Injection System

bl. Power Section

The power section generates the torque to overcome the internal friction, provides the engine output torque, and drives the fuel injection drive and oil metering pump drive system.

The power section consists of a single power unit (bank) which utilizes a rotor assembly running on an eccentric crankshaft contained within an assembly of a rotor housing and two side housings. Circumferentially located axial throughbolts clamp the three housings together, forming the single power section.

The operating cycle follows the sequence of intake, compression, combustion, expansion, and exhaust during one revolution of the crankshaft and is accomplished by the sinusoidal variation of volume formed by the periphery of the triangular-shaped rotor, the inner epitrochoidal-shaped rotor housing, and bounded axially by the two end (side) housings. The rotor assembly is mounted on the crankshaft eccentric and is timed by synchronizing gears - an internal gear in the rotor and a stationary external gear in one of the end housings - resulting in the rotor assembly rotating at one-third of the crankshaft speed. The crankshaft is supported by four (4) sleeve bearings - one in each end housing and one in each cover which is bolted to the outside position (away from the rotor assembly) of each end housing. A flywheel is provided at the torque output end of the crankshaft.



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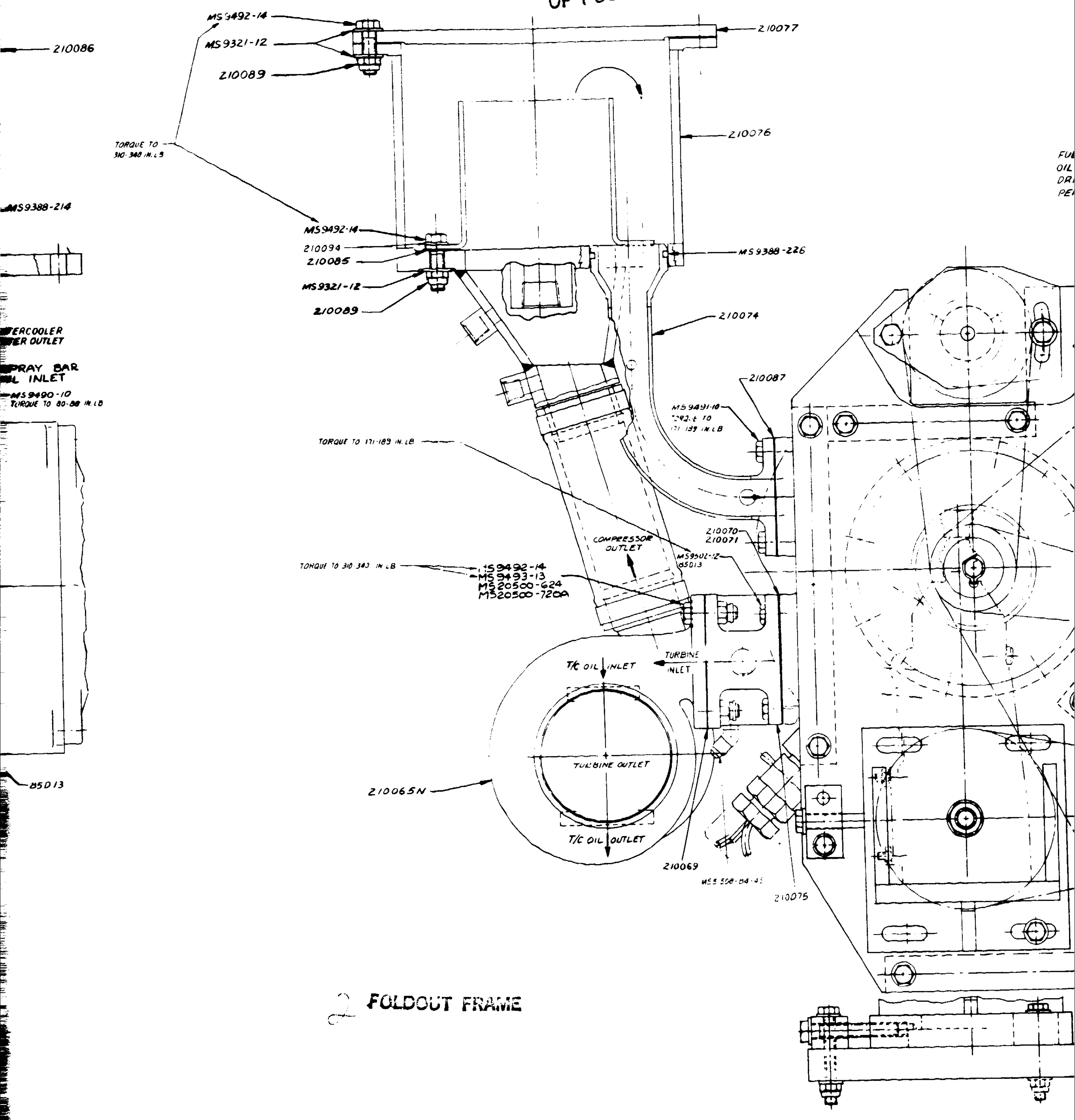

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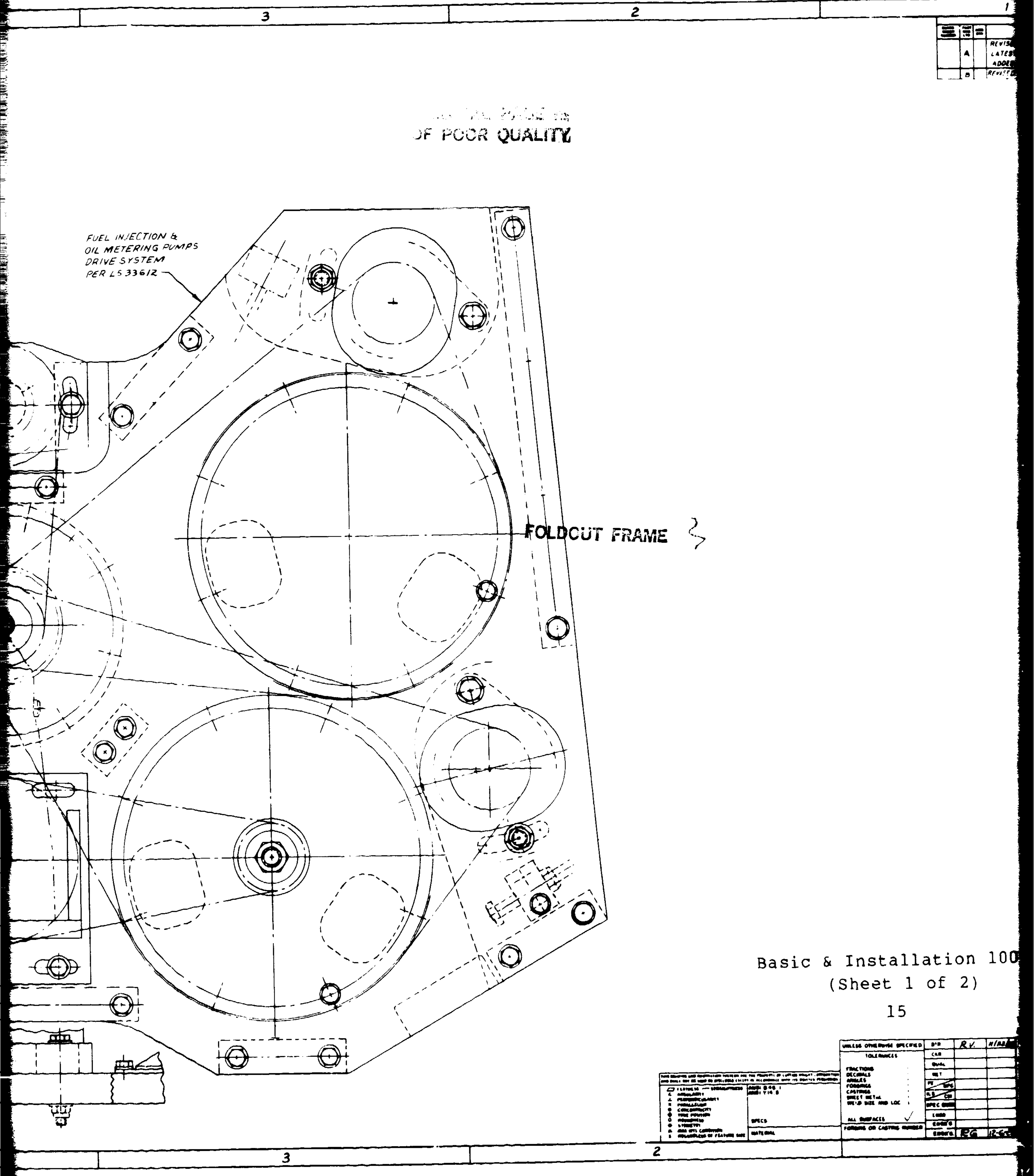

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Basic & Installation 100
(Sheet 1 of 2)

1	SHEET	617000 B

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	B		REVISED PART NO & PICTURE			

D

C

4 FOLDOUT FRAME

B

617000

B

Basic & Installation 1007R

(Sheet 1 of 2)

15

A

THIRD ANGLE PROJECTION

ALL NOTES APPLY UNLESS OTHERWISE SPECIFIED

UNLESS OTHERWISE SPECIFIED	DFTR	R.V.		CURTISS-WRIGHT CORP ROTARY COMBUSTION ENGINES
TOLERANCES: FRACTIONS, DECIMALS, ANGLES, FORGINGS, CASTINGS, SHEET METAL, WELD SIZE AND LOC	CKR			
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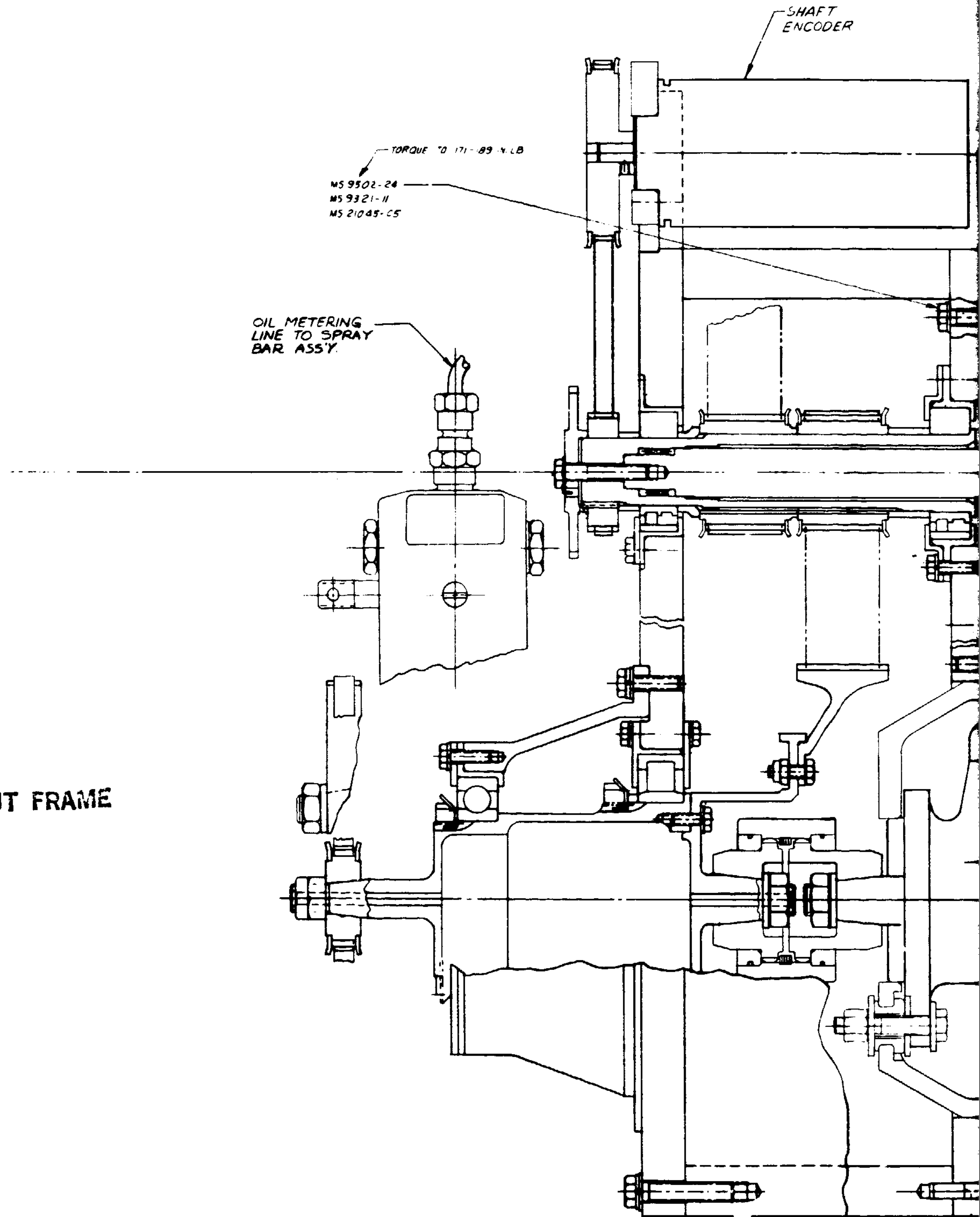

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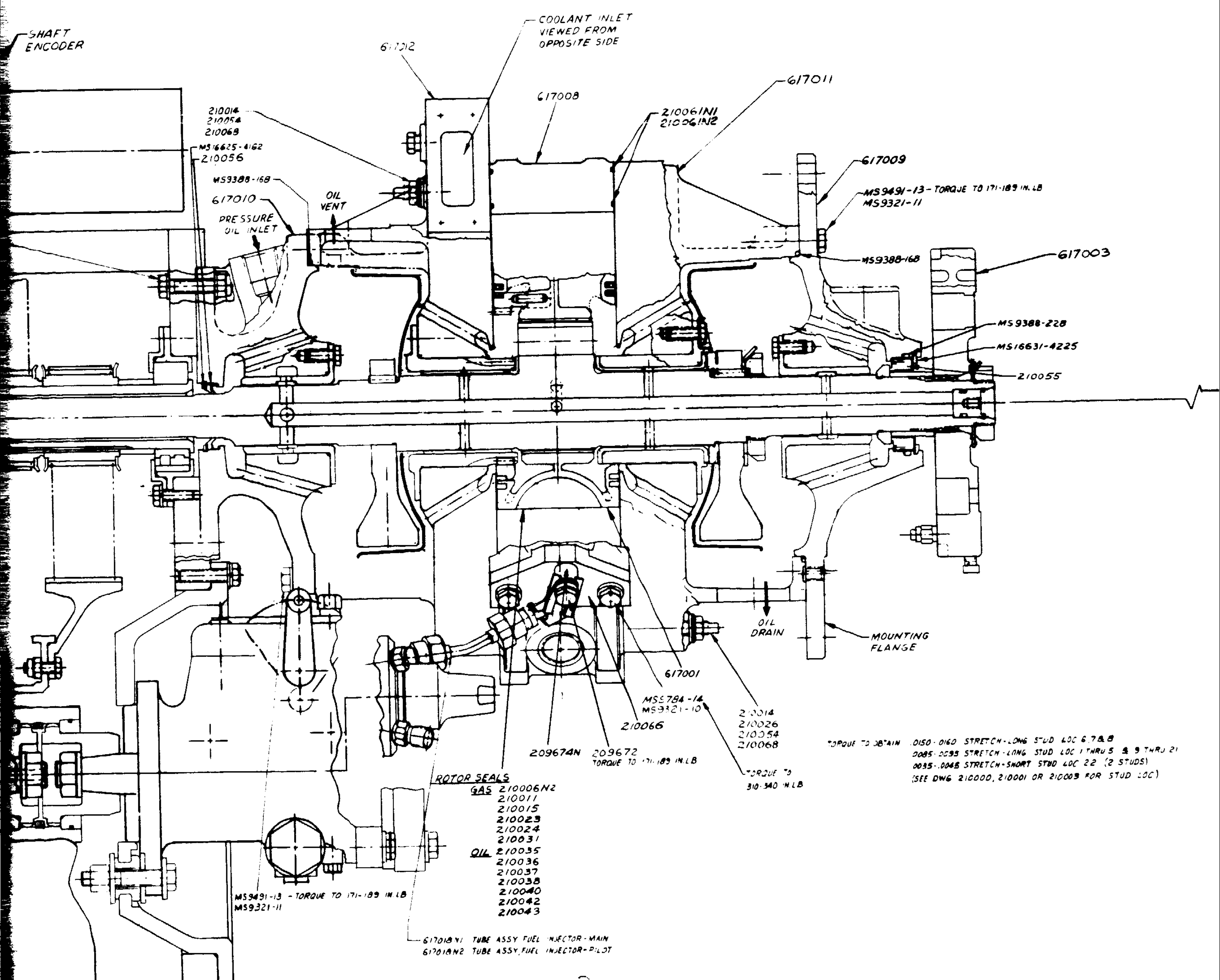

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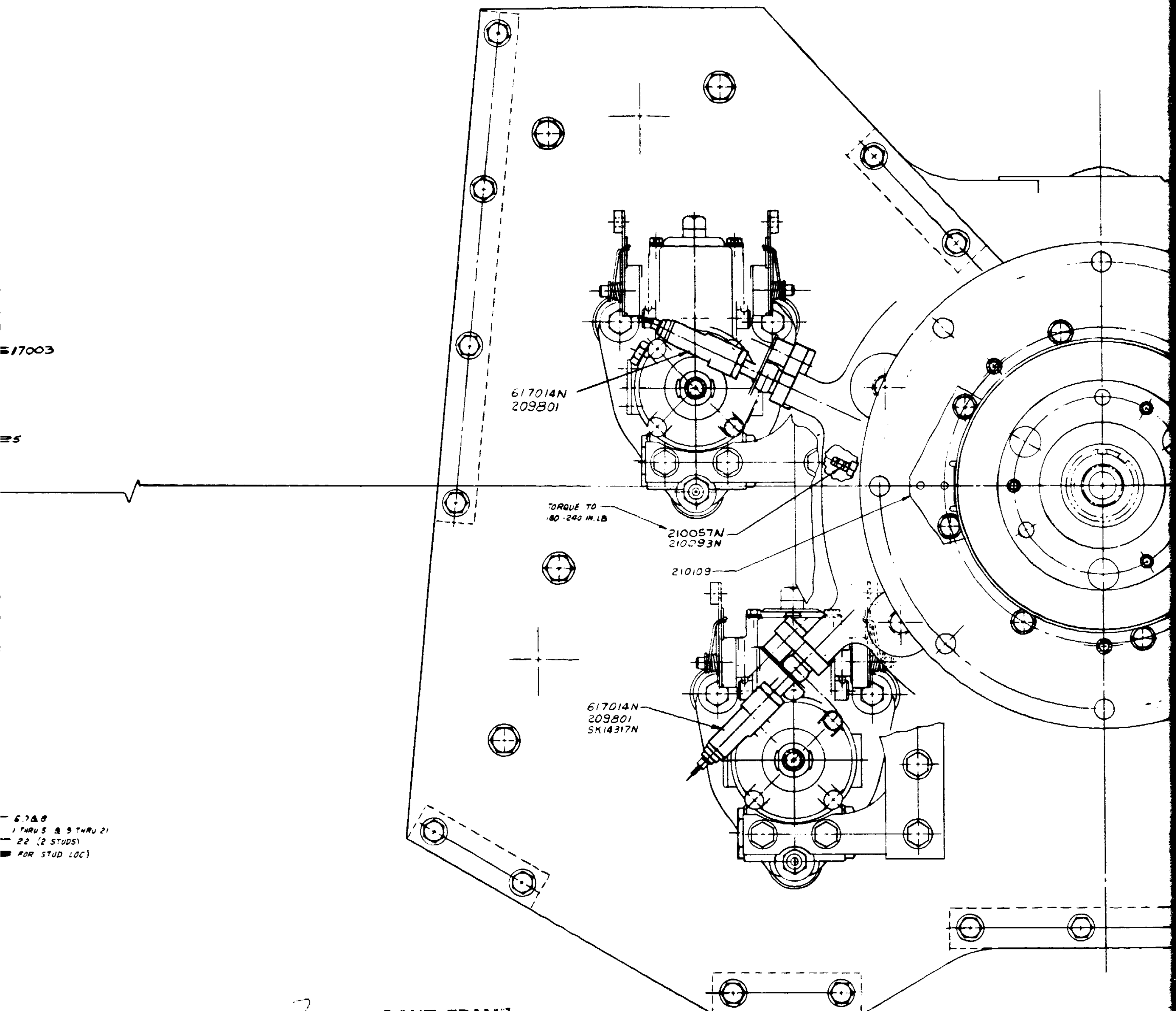

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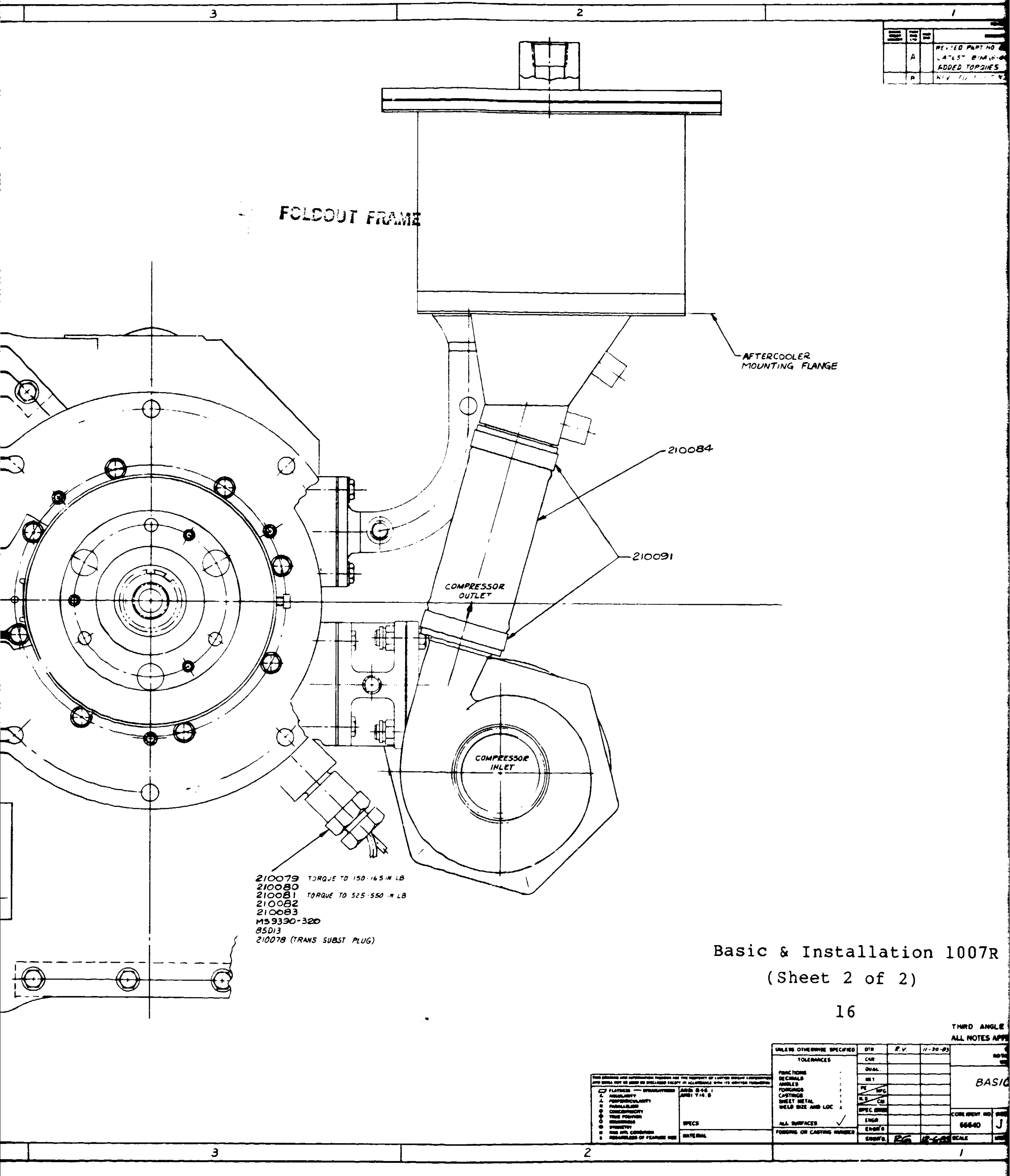

Basic & Installation 1007R
(Sheet 2 of 2)

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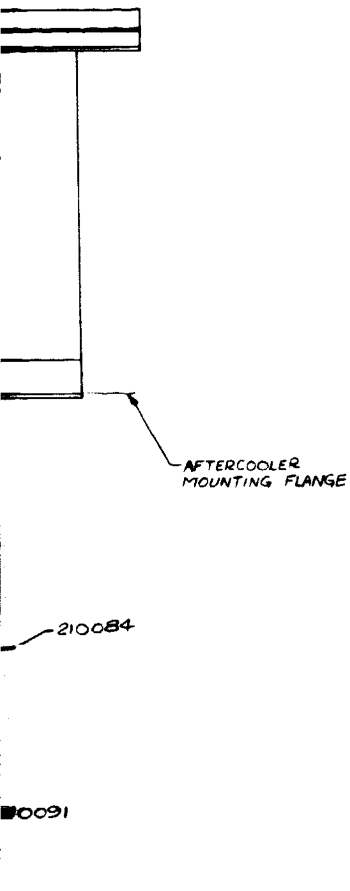

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Basic & Installation 1007R
(Sheet 2 of 2)

THIRD ANGLE PROJECTION
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UNLESS OTHERWISE SPECIFIED	DTR	R.V.	11-30-83
TOLERANCES	CKR		
FRACTIONS	QUAL		
DECIMALS	MET		
ANGLES	PE / MFG		
FORGINGS	R.S. / CM		
CASTINGS	SPEC ENGR		
SHEET METAL	ENGR		
WELD SIZE AND LOC	ENGR'G		
ALL SURFACES ✓	ENGR'G	RG	12-6-83
FORGING OR CASTING NUMBER			

CURTISS-WRIGHT CORP.
ROTARY COMBUSTION ENGINES
WOOD-RIDGE, NEW JERSEY, U.S.A.

BASIC & INSTALLATION
1007R

CODE IDENT NO.	SIZE	
66640	J	61700C
SCALE	UNIT WT	SHEET 2 of 2

SPECS
MATERIAL
ANSI B46.1
ANSI Y14.5

The power unit housing assemblies consisting of the rotor housing and two (2) end housings are assembled by the circumferentially located axial throughbolts and three (3) main dowels, and are sealed against gas and coolant leakage by "O" rings compressed between the end and rotor housings in grooves in the rotor housing side face adjacent to the inner and outer edge of the coolant passages. An intake port for introduction combustion air is located radially in the rotor housing with a similar radial port for expelling exhaust gases. Also provided in the rotor housing are locations for installing high-pressure fuel injection nozzles for the timed introduction for fuel during the combustion cycle and a spark plug for the timed initiation of combustion. Integrally cast coolant passanges arranged in a multi-pass system with the end housings utilized as headers provide the housing assembly cooling. Cool coolant enters from and hot coolant is discharged to an external heat exchanger via porting in the anti-drive end housing. Other cast-in passages in the rotor and end housings permit scavenge oil to be returned to the oil sump outside the engine via drain holes in both end covers. Internal passages within the anti-drive end cover transmit pressure oil to the mainshaft for rotor gear and bearing lubrication, rotor cooling, and main bearings lubrication.

The drive end cover assembly provides a mounting flange with provisions for the engine mounting.

The triangular-shaped rotor with radial seals at the three apexes and arc-shaped axial seals in the sides near the rotor periphery - the former operating against the trochoidal inner surface of the rotor housing and the latter against the flat surfaces of the end housings - form three gas-tight separate chambers. Successively, these chambers provide suction on the intake stroke, compression on the compression stroke and transmit the energy of the expanding gases during the combustion and expansion strokes to the eccentric of the crankshaft and produce torque. Circular type axially spring loaded seals mounted in the rotor side faces radially inward of the gas seals operate against the end housings and prevent oil (for rotor lubrication and cooling) from entering the combustion chambers.

The crankshaft assembly consists of the following basic parts: a crankshaft, two counterweights, an output flange on the flywheel, and the ADE crankshaft extension for driving the fuel injection system. The assembled crankshaft with its eccentric absorbs the power generated in the combustion chambers and transmits this power in equally spaced impulses to the shaft where power can be delivered from the flywheel. Drilled passages within the journaled crankshaft receive high-pressure oil from the test-stand lubrication system through the ADE cover drilled inlet passage. This supplies lubrication to the rotor bearings and the gears and provides cooling oil to the interior of the rotor. The counterweights are located between each ends pair of journal bearings and provide balancing since the center of gravity of the rotor is always a fixed distance from the crankshaft centerline. The small-diameter crankshaft extension at the ADE serves as a quill to provide torque to drive the fuel injection system, the oil-metering pump, and the NASA encoder unit.

b2. Ignition System

The ignition system for this rig engine consists of a timing signal from the engine crankshaft angular position, a delay unit which permits the start of ignition to be delayed in crank degrees relative to the timing signal, an ignition control module which generates the coil primary voltage and controls the duration of the spark, a coil unit which converts the primary voltage put out by the control unit to a very high voltage as required by the spark plug, a high-tension cable to conduct these high voltages to the spark plug, a high-tension cable to conduct these high voltages to the spark plug, and finally the spark plug. A brief description of each of these items is as follows:

Timing signal - The approach being used for this engine is derived from our experience with other engines, notably the 5.8 (350 in.^{3}) family. Various approaches including conventional breaker points and variable reluctance magnetic pickups have been used with limited success, which led us to the use of the Hall effect sensor type of pickup. For this engine we are mounting a magnet on the engine side of the flywheel at 90 degrees before top dead center and have located a commercial Hall effect pickup of the digital type (on-off only) to sense the passage of the magnet. We have found that this type of system is immune to dirt and electrical interference and is completely insensitive to engine speed.

Delay unit - A new electronics module is being procured for this engine which will permit the timing of the ignition relative to the trigger pulse to be accomplished electronically (rather than the mechanical system which we have used in the past). This delay unit contains the power supply and signal conditioning for the Hall effect pickup (as well as capability of using optical or breaker point triggering inputs), an externally mounted calibrated potentiometer to adjust the delay, and the actual delay unit which delays a set number of crank-angle degrees independent of engine speed.

Control unit - The ignition used is the Autotronics Controls Corporation's Low Tension Ignition System, hereafter referred to as "LTIS." This ignition system was developed for the unique requirements of the stratified charge rotary combustion engine. The LTIS ignition is a very high frequency multiple sparking system with a controllable duration. The duration control is a calibrated potentiometer which we mount in the control room which permits infinite adjustability up to a maximum duration of 100 crank-angle degrees. This system has been used for all of the 5.8 testing with excellent results and should provide the flexibility required of this new engine.

Ignition coil - A special coil matched to the characteristics of the LTIS ignition is being used. This same coil was used on the 1058R (single rotor, 5.8, 350 in.^{3}) engine.

High-tension lead - A standard automotive 9-mm high-tension lead is used between the coil and spark plug to facilitate instrumentation of the spark plug, including the occasional use of thermocouple spark plugs.

Spark plug - For the initial engine testing virtualy standard automotive spark plugs are being used. The spark plugs are 3/4" reach types with 12-mm-diameter threads. Many different types of electrode configurations are being procured based on our test experience. This will be our first use of the 12-mm plugs which permit improved cooling of the entire spark plug - pilot injector area.

b3. Cooling System

The coolant system removes the heat generated in the power section of the engine in order to maintain safe metal temperatures. The temperature of the coolant out of the engine is controlled between 170°F and 180°F. The system incorporates an external heat exchanger, coolant pump, thermostats, and the passages within the rotor and end housings of the power section.

b4. Lubrication System

The lubrication system not only supplies all engine power section bearings with filtered oil, but also supplies indirectly filtered oil for cooling the rotor assembly. In addition, a metered quantity of filtered oil is supplied to the rotor housing trochoid surface to lubricate the rotor assembly apex seals. All oil so supplied, except that which lubricates the apex seals or escapes past the oil control rings, is collected and either returned directly to the sump (external oil tank) or to an external heat exchanger and then to the sump as required to maintain the oil temperature within specified limits. Oil fumes generated in the engine are vented via holes in each end housing to the supply tank.

b5. Turbocharger System

The turbocharger system being used is very conventional and consists of an exhaust gas-driven turbo supercharger and a water-cooled charge air cooler between the compressor outlet and the engine inlet. The details of this system are being coordinated with the AiResearch Division of the Garrett Corporation. The initial hardware will be standard components which we adapt for use on our engine.

Previous turbocharger testing on the 1058R engine has revealed that standard turbine components do an excellent job of recovering the pulse energy in the exhaust which is available from the blow-down process. Since this energy recovery is dependent upon maintaining the pulse energy from the engine's exhaust port to the turbine extrance, an attempt is being made to keep the flow area constant. The initial turbochargers use available turbine housings which have large area reductions from inlet to the nozzle "tongue." Therefore, an insert was made which fills in the majority of the area reduction to aproximate constant area. Once the proper turbine sizing has been determined by engine testing and optimization, new turbine housings will have to be made using a new pattern for the inlet which is designed for contant area.

The charge air cooler is a standard high-volume diesel engine model for which a housing has been designed. Water flow through this heat exchanger will be controlled on the engine test stand to control the temperature of the air at the engine inlet to duplicate the performance of different types of charge air coolers.

b6. Fuel Injection System

The fuel injection system for this rig engine was initially specified to be an "electronic fuel injection system" with separate systems for pilot and main injection. The initial intent was to use the Bendix electronically controlled high-pressure common rail system which has independent control of injection pressure, timing, and quantity. During the first months of this program, it became obvious that we should delay the procurement of this advanced technology system until the next phase of the program and concentrate instead on using a lower capability but more readily available system using off-the-shelf commercial components.

Our first attempt at doing this job was to use a single plunger-type jerk pump with an eccentric camshaft to meet our requirements of 8000-injection-per-minute injection frequency and flows up to 70 mm^3 per stroke. This goal was accomplished along with pilot operation at 3 to 5 mm^3 per injection. Unfortunately, this testing also brought to light several durability problems with the jerk pump which we were using. Several attempts to solve these problems were undertaken with the pump's original manufacturer assisting; however, we concluded that the pump's durability would not be adequate.

Our second attempt was to use a distributor type of pump, specifically the Stanadyne DM model. Stanadyne has done extensive modeling and they are confident that the hardware which they have selected will meet our minimum objectives. We will be using their 4-injection-per-revolution cam ring piped to a single outlet. The pump will be belt driven at one-fourth engine speed. Separate and independent pumps will be used for pilot and main injections. An internal variable timing system which rotates the cam ring will be utilized which will provide for a dynamic timing range of 20 crank degrees at any given speed.

The injection nozzles which we have specified are the Stanadyne "slim tip" nozzles of conventional construction. We will instrument the nozzles for needle lift in order to obtain timing and duration information. Spray-hole geometries have been selected based on our previous experience to obtain proper fuel air mixing at acceptable injection pressures and durations.

Component Design

Detailed descriptions of the various 1007R engine components and the data supporting their design are presented in the Task I Design Report.

Design Analyses

The heat transfer and stress analyses conducted on the 1007R engine are summarized in Table 3.1. Analysis of each of the major components is discussed below:

a. Rotor

Experimental temperature measurements from the 1058R engine were the basis for an estimate of rotor temperatures for the 1007R engine, taking into account the higher relative speed and the higher BMEP of the 1007R engine. These temperatures were used in a two-dimensional ANSYS Finite Element Rotor model to calculate thermal stresses. Pressure-induced stresses were calculated by plate and shell theory and/or ratioed from the prior finite element analysis of the 1058 engine rotor. These stresses were combined to determine steady and alternating stresses, and factors of safety were determined from minimum property data plotted on a Goodman diagram. The Safety Factor for the rotor was adequate for both 17-4PH and Nodular Iron material.

b. Rotor Housing

Heat transfer analysis was conducted based on prior experimental and analytical data on the 1058 engine and with due consideration to the more stringent operating conditions for the 1007R engine, i.e., high combustion pressure levels and higher relative speed, both of which increase convection coefficients to the watercooled rotor housing.

The results of the thermal analysis were used together with combustion pressure data for stress and deflection analysis of the rotor housing, using a closed frame strain energy computer program. This program had been used by Curtiss-Wright on other rotary engines with good results. Results of the analysis were that the 1007R rotor housing met the design criteria.

TABLE 3.1

ANALYSIS SUMMARY

Part	Load Sources	Factor of Safety	Thermal Cycles
End Housings	Firing + Press Fit + Centrifugal	1.5	-
Crankshaft	Centrifugal + Firing		
	8000 rpm, 1400 psi	3.4	-
	9600 rpm, 1100 psi	2.9	-
Rotor	Thermal + Firing + Centrifugal	1.9	100,000 +
Rotor Housing	Thermal + Firing		
	External Ribs	3.5	
	TDC Region, away from bosses	-	1,000 - 10,000
	TDC Region at bosses	-	40* - 400

*40 cycles does not meet the previously identified design criteria. Prior experience is that cracks at bosses propagate very slowly. Development changes are required to improve the LCF life.

Additional analysis was done for the combined spark plug/pilot injector boss. A two-dimensional finite element heat transfer and thermal stress analysis was conducted which indicated that the high thermal stresses in this area would cause crack initiation in 40 start/stop cycles. These cracks tend to be self-relieving since the highest stress is compressive, and thus do not propagate rapidly. It is nevertheless recommended that the developmental changes outlined in the Design Report be pursued to increase the cycles to crack initiation.

c. End Housings

The end housings were analyzed using data from the 2116R engine program where failures had occurred. Essentially the 2116R configuration was strengthened to be adequate, then scaled down to the 1007R size, then strengthened an additional amount for the much higher design pressure.

d. Crankshaft

Crankshaft analysis was undertaken at contract inception to determine shaft slopes for a two main bearing single-rotor engine at the high-speed, high-firing pressure conditions. Excessive shaft slopes were calculated approximately 2 mm/m (0.002 in./in.). Consideration was then given to incorporating outboard bearings to reduce shaft slopes. This configuration reduced the slope below 1.49 mm/m (0.00149 in./in.) at the 9600 rpm (120% of design speed) and was selected for final design, procurement, and test.

Crankshaft stresses were acceptable for all speed and load conditions considered.

e. Bearing Analysis

Two (2) bearing analyses were performed: one, evaluation of SAE 30 oil in place of the Design-recommended SAE 50 and two, evaluation of the as-built bearing clearances versus the Design-recommended bearing clearances. Although both oils provided satisfactory bearing performances, the SAE 50 grade oil provided increased operating film thicknesses and therefore was preferred. In the latter study, although some main-shaft bearing clearances were a little under the low limit, bearing analysis data suggested satisfactory bearing performance would be maintained at speeds up to 8000 rpm, the initial build maximum speed objective.

Bill of Material

A complete Bill of Material was included in the Task I design report.

Design Revisions

a. Design Jobs Completed Since Release of 12/16/83 Design Report

a1. Major subassembly and assembly drawings.

a2. Bill of Material.

a3. Design layouts in support of engine assembly for test stand 20-6 installation. Some layouts provided base engine part modification for mounting of desired instrumentation/ measurement equipment.

a4. Crankshaft main bearings and rotor bearing performance evaluation using SAE 30 oil rather than the SAE 50 oil originally recommended.

a5. Low compression ratio rotor (CR=6.7:1.0) detail and assembly drawings.

a6. Drawings for engine oil drain and venting adapters, fuel injection pump supports/brackets and rotor housing coolant pressure and thermocouple instrumentation.

a7. Evaluation of Build 1 as-built main-shaft bearing clearances.

a8. Based on apex seal wear rig testing, incorporated rotor apex slot Nibron plating (0.005-0.0010 THK per JD Specification 5755) to enhance rotor slot/apex seal compatibility, reference 617001N, Revision B.

a9. Incorporated Lord isolation bushings on three (3) bolting locations of the fuel injection pump flange to provide pump isolation from accessory gearbox, reference LS-33612, Revisions D and G.

b. Layouts and Pertinent Sketches

LS-33600	-	Crankshaft & Bearing Assembly
LS-33601	-	Rotor Assembly
LS-33602	-	Housing, Rotor
LS-33603	-	Housing, Drive End
LS-33604	-	Housing, Anti-Drive End
LS-33605	-	Turbocharger After-Cooled System
LS-33606	-	Counterweight Installation & Removal
LS-33608	-	Housing, Rotor (FI & S/P Assembly)
LS-33611	-	1007R Driveline - Test Stand 20-6 Arrangement
LS-33612	-	Fuel Injection & Oil Metering Pump Drive System
LS-33613	-	Oil Drain & Vent Fittings, T/C in-out, Oil Fittings (Sheet 1); Coolant Extension, Inlet & Outlet (Sheet 2)

LS-33615	-	Plate Assembly, F.I. Nozzle (revised)
LS-33619	-	Instrumentation, 1007R Driveline Test Stand 20-6 Arrangement
SK-12838	-	Ring, Mount
SK-12846	-	Tube Assembly, Oil Drain, Turbocharger
SK-12847	-	Adapter Assembly, Oil Inlet, Turbocharger
SK-12848	-	Tube Assembly, Oil Drain, End Housing
SK-12849	-	Tube Assembly, Breather, End Housing
SK-12850N	-	Gasket, Oil Inlet Connector Flange, Turbocharger
SK-12851N	-	Gasket, Oil Drain Tube Assembly, Turbocharger
SK-12852	-	Housing Assembly, Rotor (rework-coolant pressure)
SK-12853	-	Rotor Housing Thermocouple Installation-Hot Zone
SK-12868	-	Bracket, Instrumentation
SK-12869	-	Pick-up Assembly, Ignition
SK-12970	-	Support, Speed Sensor
SK-12871	-	Support, Top Dead Center Sensor
SK-12872	-	Plate, Slider
SK-12838	-	Ring Assembly, Mount
SK-12879	-	Support Assembly, Fuel Injection and Oil Metering Pump Drive System
SK-12880	-	Extension, Coolant (weldment)
SK-12881	-	Extension, Coolant Inlet

FABRICATION

Fabrication of the major hardware for the NASA Technology Enablement Rig Engine was initiated second quarter 1983. All work was controlled in accordance with the Product Assurance provisions of Appendix B of the contract requirements.

Following is a review of major components fabrication.

Crankshafts

The crankshafts were fabricated from forged AMS6260 steel which was ultrasonic and magnetic particle inspected to assure material integrity. The forgings were machined to semi-finished size and then carburized on the journals and eccentric surface. After hardening and tempering, the shaft was finish-machined. Final NDT inspection consisted of magnetic particle test on all surfaces and etching of the carburized areas to assure absence of grinding burns.

Gears (Rotor and Stationary)

Gears were fabricated from AMS6260 pancake forgings. After rough machining the forgings were magnetic particle tested to assure a crack-free condition. The pieces were semi-finish-machined and then carburized, on the gear tooth surfaces, and hardened and finish-machined. Final NDT inspection consisted of 100% magnetic particle inspection and grinding burn etch of carburized areas.

Housings (Covers, Side Housings, Rotor Housings)

All housings were sand cast in aluminum. The rotor housing was cast from 201 material while the other housings were 357 (side housings) and 356 (covers). Selected areas of some housings were designated as high-stress areas by design and analysis. In these areas special chilling was placed so as to promote fine dendritic grain size and low gas and micro shrinkage porosity, thus resulting in higher properties after heat treatment.

All castings were radiographic and fluorescent penetrant inspected after heat treatment prior to release for machining. In a few instances small, cosmetic repair welds were permitted in areas of low stress. No major repairs were required. Following final machining, pressure checks were performed to assure part integrity.

Molybdenum was applied to the combustion chamber face of the side housings. This material was applied by flame spray and was then ground and lapped.

A coating of tungsten carbide in a cobalt matrix was applied to the rotor housing trochoid surface by the Linde D-gun (detonation gun). This coating was ground and lapped to produce the required fine finish.

Rotor

The rotor was investment cast from 17-4 PH steel. This material was chosen because of its excellent castability and its good mechanical properties. However, like all stainless steels, 17-4 PH has a tendency to gall when in sliding contact with certain other materials. It was demonstrated by compatibility tests that the apex seal might hang up against the rotor; therefore, it was necessary to coat the apex seal slots to prevent contact between the seal and the rotor material. A nickel, thallium, boron electroless nickel plate was chosen for this coating. This plate develops very high hardness to resist wear. Also, the fine intermetallic precipitate, which yields the high hardness, acts as a friction barrier to assure compatibility between the plate and adjacent seal materials.

Several problem areas were encountered in investment casting the rotor. The investment casting process is capable of reproducing details very accurately and indeed this was the case with all dimensions except the center web and the pocket. It was noted by the casting supplier that in all cases the center web dimension was undersize by approximately .03" even though the tooling was correctly sized. Several variations in casting procedure were attempted, however, this problem was not corrected. The problem in the pocket was due to an error in the tooling design which was easily corrected. In order not to delay the program, it was decided to accept two rotors with these conditions while tooling modifications were made to correct castings remaining to be poured. These rotors were made subject to operational restrictions in allowable peak pressure. The remaining rotors being procured will have proper center webs and pocket dimensions after machining.

All rotors completed and currently on order are 7.5:1 compression ratio. Procurement of the low ratio (6.0:1) rotor has been deferred with NASA concurrence.

Major Parts Procured

Part Number	Description	Quantity
210000	Housing, Drive End	2
210000Cast	Housing, Drive End-Casting	6
210001	Rotor Housing	2
210001Cast	Rotor Housing-Casting	8
210001Coat	Rotor Housing-Coating	2
210002N2	Rotor	2
210002N2Cast	Rotor - Casting	6
210002N2SP	Rotor - Shot Peening	2
210003	Housing, Anti-Drive End	2
210003Cast	Housing, Anti-Drive End Casting	6
210004	Cover, Drive End Housing	2
210004Cast	Cover, Drive End Housing	4
210005	Cover, Anti-Drive End Housing	2
210005Cast	Cover, Anti-Drive End Housing	4
210006N2	Seal, Rotor Side	24
210007	Crankshaft	2
210007HF	Crankshaft (Forging)	6
210008N	Bearing, Rotor	6
210009	Bearing, Crankshaft Main	3
210010N2	Bearing, Crankshaft Main	3
210011	Spring, Rotor Apex Seal	12
210012N	Bearing, Crankshaft Main (Thrust)	3
210013N	Bearing, Crankshaft Main	3
210015	Spring, Rotor Button	24
210017	Gear, Stationary	6
210018	Balance Weight	4
210018HF	Balance Weight, Crankshaft (Forging)	4
210019	Gear, Rotor	6
210019HF	Gear, Rotor (Forging)	10
210020	Flywheel	2
210020HF	Flywheel, Mainshaft (Forging)	5
210021	Support, Bearing	4
210022N1	Baffle Plate Assembly	3
210022N2	Baffle Plate Assembly	3
210023	Seal, Rotor Apex (Short)	8
210024	Seal, Rotor Apex (Long)	8
210025	Stud (Short)	6
210026	Stud, Rotor Housing (Long)	50
210027	Pin, Rotor Apex Seal	15
210028N	Nozzle Assembly, Partial (Main)	13
210028N	Nozzle Assembly, Partial (Pilot)	6
210031	Spring, Rotor Side Seal	25
210035	Oil Seal Set (Inn & Outer 2 Sets/Rotor)	8
210038	Spring, Front Inner	4
210039	Spring, Front Outer Oil Seal	4
210040	Spring, Rear Inner	4
210041	Spring, Rear Outer Oil Seal	4
210042	Spring, Front Outer (Racing Type)	4
210043	Spring, Rear Outer (Racing Type)	4

Part Number	Description	Quantity
210046	Nut, Flywheel Attaching	4
210047	Nut, Balance Weight	4
210049	Lock, Flywheel Attaching Nut	20
210050	Cone, Flywheel Centering	4
210057N	Spark plug (12 mm)	100
210060	Aftercooler Core	1
210062	Spray Bar Assembly, Oil Metering	4
210063	Timing Control, Ignition	1
210065N1	Turbocharger Assembly	1
210067	Dowel	9
210068	Nut, Rotor Housing Tie Bolt	200
210072	Fuel Injection Pump, Main	2
210073	Fuel Injection Pump, Pilot	2
210074	Air Intake Manifold Assembly	2
210075	Extension, Exhaust	2
210076	Aftercooler Housing Assembly	1
210077	Cover, Aftercooler Housing Assembly	1
210095	Mainshaft Master Balance Weight	1
210096	Stud, Mainshaft Master Balance Weight	3
210097N	Washer, Mainshaft Master Balance Weight Stud	20
LS-33612	Drive System-Fuel Injection & Oil Metering	1

TEST PLAN

(1) Test Objective

Run-in and acceptance testing of the 1007R Technology Enablement Rig engine to establish mechanical integrity and baseline performance.

(2) Tests

The test program consists of dynamometer tests with the first 1007R rig engine in a modified existing facility leased by John Deere Technologies Int'l, Inc., from Curtiss-Wright Corporation, Dynamometer Test Cell WX20-6.

A. Overall Test Program

The test program is anticipated to be one continuous program in which engine operation is investigated and documented. The order in which testing is planned to be conducted is in the general format outlined herein:

1. Engine/Test Cell Preparations

Basic test cell and engine systems checkout and calibration. Engine run-in.

2. Baseline Performance Documentation

Variable speed and load with injection nozzles of different spray geometry, turbocharger turbine housings with a range of A/R's and engine inlet temperatures.

B. Installation

1. Install and align the engine on test stand WX20-6 using the Rexnord T83-1784 coupling and the Cotta 1.70:1 reduction gearbox.
2. Connect injection pump (pilot and main) timing controls.
3. Connect ignition timing controls.
4. Connect test-stand oil system, including oil metering system and turbocharger, using Texaco Premium AD-100 SAE 50 oil.
5. Connect test-stand coolant system using 50% by volume of ethylene glycol antifreeze and water plus inhibitor.
6. Connect air measurement system including air filter.
7. Connect intercooler coolant system.

b. 8. Connect exhaust system.

9. Connect fuel system using Flotron measuring systems to indicate pilot and total fuel flows using Jet A, turbine engine fuel.

10. Connect Top Dead Center pickup (verify with rotor position).

11. Install and connect translational vibration pickups.

12. Connect RPM pickup.

13. Connect injection nozzle proximators.

14. Install and connect the torsiograph and accelerometers.

15. Connect the BTC location combustion pressure pickup. (Other three locations have P/N 210078 substituting plugs)

16. Connect chip detectors (3)

17. Connect turbo RPM pickup.

18. Connect ignition trigger.

19. <u>Provide temperature measurements at:</u>
(a) 2 SK-12853 rotor housing hot zone
(b) air bottle
(c) compressor inlet
(d) compressor outlet
(e) engine inlet
(f) engine outlet
(g) turbine outlet (multiple)
(h) coolant inlet (RTD)
(i) coolant outlet (RTD)
(j) intercooler coolant inlet
(k) intercooler coolant outlet
(l) gearbox coolant inlet (engine to dynamometer)
(m) gearbox coolant outlet (engine to dynamometer)
(n) pilot flotron fuel inlet
(o) main flotron fuel inlet
(p) pilot injection pump fuel inlet
(q) main injection pump fuel inlet
(r) oil inlet (RTD and thermo)
(s) oil outlet (RTD and thermo)
(t) oil outlet (anti-drive end)
(u) oil outlet (drive end)
(v) turbocharger oil outlet

20. <u>Provide pressure measurement at:</u>
(a) SK-12852 rotor housing coolant
(b) compressor inlet (total)

20. (c) compressor outlet (total)
(d) engine inlet (total)
(e) engine outlet (total)
(f) engine outlet (dynamic)
(g) turbine outlet (static)
(h) engine coolant inlet
(i) engine coolant outlet
(j) oil inlet
(k) pilot flotron fuel inlet
(l) main flotron fuel inlet
(m) pilot injection pump fuel inlet
(n) main injection pump fuel inlet
(o) turbocharger oil inlet
(p) pilot fuel pump timing pressure
(q) main fuel pump timing pressure
(r) pilot pump fuel transfer pressure
(s) main fuel pump transfer pressure

21. RPM measurements are to be engine crankshaft speed.

22. Provide for other parameter measurement/reading as required per Item H., Measurements and Precision.

C. Operational Limits/Basic Data

1.	Coolant inlet temperature	175 ± 5°F
2.	Oil inlet temperature	165 ± 5°F
3.	Oil outlet temperature	240°F max.
4.	Oil inlet pressure	65 ± 5 psig
5.	Coolant flow	2000 lb/h per 1000 rpm
6.	Minimum firing speed	2000 crankshaft rpm
7.	Minimum oil pressure to turbocharger	30 psig
8.	Coolant pressure	25 psig @ 16,000 lb/h flow
9.	Flotron flowmeter inlet pressure	15 psig
10.	Ignition system battery voltage	22-25 volts
11.	Fuel injection pumps inlet temperature	90°F max.
12.	Engine maximum speed	8000 rpm
13.	Combustion pressure	1100 psi max.

D. Run-in

1. Prior to running, check function of the following:

(a) Fuel system
(b) Coolant system
(c) Oil system - oil to turbocharger
- oil to oil metering pump

D. (d) Dynamometer load
(e) Ignition system
(f) Instrumentation
(g) Intercooler coolant system

2. Conduct static oil flow check with coolant and lubricating oil temperatures per inlet limits.

3. Conduct static air leak check with 0.075-inch orifice and 65-psig inlet pressure.

4. Since a driveline torsional exists below 2000 crankshaft rpm, the engine is to be motored to 2000 rpm before activating ignition or starting fuel injection.

5. The torsiograph and accelerometers will be monitored during the run-in.

6. Running will be initiated with the following possible component variables:

   (a) Turbocharger P/N 210065N1 utilizing 1.3 A/R turbine housing P/N 210064N3
   (b) Pilot Injection Nozzle P/N 210028N25 (0.007-inch orifice)
   (c) Main Injection Nozzle P/N 210028N8 (6 x 0.010-inch)

Run-in Schedule

1. During run-in the engine inlet temperature (intercooler outlet temperature) will be maintained at 85-95°F by varying intercooler coolant flow.

2. Best power operation will be obtained by varying ignition, pilot, and main injection timings. Since timing variation of the pumps is 32 crankshaft degrees, retiming of the pump drives may be required.

3. For run-in only, the oil metering pump will be set to meter oil equal to 2 percent of rated power (160 hp) fuel flow at rated speed (8000 rpm). All other running will utilize a 1% setting of the metering pump.

4. Run-in and Initial Performance Work Schedule
30 Minutes per point

Pt.	rpm	BMEP	Pt.	rpm	BMEP
1	2000	20	8	6000	70
2	2000	40	9	6000	100
3	3000	70	10	7000	70
4	4000	50	11	7000	100
5	4000	70	12	7000	130
6	4000	90	13	2000	40
7	5000	40	14	Hot air leak check	

Nozzle changes may be made during this period as results indicate.

5. If a turbo change is required at completion of above run-in, the turbocharger will be removed from the engine. An internal engine inspection will be made at this time.

E. Motoring Friction

Motoring friction data will be obtained over the engine speed range with coolant and lubricating oil preheated to operating unit temperature limits.

F. Engine Checkout and Acceptance Testing

Note: Based on data obtained during the run-in, the turbocharger turbine housing may be replaced for a change of A/R.

G. Performance

At selected speeds between 4000 and 8000 rpm, obtain variable load curves to define engine characteristics with optimized fuel injection and ignition timings.

(a) Variations of available main and pilot injection nozzles will be tested as dictated by previous results.

(b) At selected operating conditions, the effect of intercooler outlet temperature will be evaluated.

(c) If data dictate the turbocharger turbine housing will be replaced to vary the A/R and data obtained at selected operating conditions.

Note: Items (a), (b), and (c) may be obtained in combination.

H. Measurements and Precision

Test data recorded will be in accordance with the following list:

(a)	Barometric pressure, true	in. Hg.
(b)	Wet bulb temperature	°F
(c)	Dry bulb temperature	°F
(d)	Vapor pressure	in. Hg.
(e)	Date	Mo./Date/Year
(f)	Time of day	0-2400
(g)	Total time	hours
(h)	Engine speed	Crankshaft rpm

(i)	Dynamometer load	lb
(j)	Brake horsepower	bhp
(k)	Airflow	lb/h
(l)	Fuel flow, pilot	lb/h
(m)	Fuel flow, total	lb/h
(n)	Fuel air ratio	
(o)	Brake specific fuel consumption	lb/bhp-h
(p)	Brake specific air consumption	lb/bhp-h
(q)	Coolant flow	lb/h
(r)	Coolant inlet temperature	°F
(s)	Coolant outlet temperature	°F
(t)	Coolant inlet temperature (RTD)	°F
(u)	Coolant outlet temperature (RTD)	°F
(v)	Coolant temperature rise (RTD)	°F
(w)	Intercooler coolant inlet temperature	°F
(x)	Intercooler coolant outlet temperature	°F
(y)	Coolant inlet pressure	psig
(z)	Coolant outlet pressure	psig
(aa)	Rotor housing coolant pressure	psig
(ab)	Oil flow	lb/min
(ac)	Oil inlet temperature	°F
(ad)	Oil outlet temperature (ADE)	°F
(ae)	Oil outlet temperature (DE)	°F
(af)	Oil outlet temperature turbo-charger	°F
(ag)	Oil inlet temperature (RTD)	°F
(ah)	Oil outlet temperature (RTD)	°F
(ai)	Oil temperature rise (RTD)	°F
(aj)	Engine oil pressure	psig
(ak)	Turbocharger oil pressure	psig
(al)	Air bottle temperature	°F
(am)	Compressor inlet temperature	°F
(an)	Engine inlet temperature	°F

(ao) Turbine inlet temperature	°F
(ap) Turbine outlet temperature	°F
(aq) Compressor inlet pressure	in. H_2O/in. Hg.
(ar) Compressor outlet pressure	in. Hg.
(as) Engine inlet pressure	in. Hg.
(at) Turbine inlet pressure	in. Hg.
(au) Turbine outlet pressure	in. Hg.
(av) Ignition timing - start	deg., BTC
(aw) Ignition duration	deg.
(ax) Pilot injection - start	deg., BTC
(ay) Pilot injection - duration	deg.
(az) Main injection - start	deg., BTC
(ba) Main injection - duration	deg.
(bb) Fuel inlet temperature - pilot pump	°F
(bc) Fuel inlet temperature - main pump	°F
(bd) Fuel transfer pressure - pilot pump	psig
(be) Fuel transfer pressure - main pump	psig
(bf) Fuel inlet pressure - pilot pump	psig
(bg) Fuel inlet pressure - main pump	psig
(bh) Timing pressure - pilot pump	psig
(bi) Timing pressure - main pump	psig
(bj) Peak combustion pressure	psig
(bk) Rotor housing temperatures	°F

Note: Test equipment and instrumentation will be calibrated and maintained in accordance with MIL-C-45662.

TEST RESULTS AND DISCUSSION

Testing of the 1007R rig engine No. 0701 Build No. 1 was initiated with the following configuration:

1 - P/N 210002N2, 7.5:1 compression ratio rotor

2 - P/N 210001, ATC pilot rotor housing

3 - P/N 210028N8, main injection nozzle with six 0.254-mm (0.010-in.) orifices

4 - P/N 210028N25, pilot injection nozzle with one 0.178-mm (0.007-in.) orifice

5 - P/N 210057N4, single ground electrode spark plug

6 - P/N 210082, pressure transducer (AVL) located at the BTC location in the rotor housing for monitoring peak combustion pressures

7 - P/N 210065N1, turbocharger with P/N 210064N3 turbine housing having a 1.3 A/R

Prior to conducting the Test Plan run-in, the pilot injection nozzle and spark plug penetration was varied to produce consistent combustion when supplying fuel to only the pilot nozzle. Table 6.1 presents the variations evaluated together with a record of all nozzle and spark plug changes made throughout the complete test. It should be noted that spark plug changes were made only for location or configuration variation as no spark plug fouling or mishap occurred during the approximate 70 hours of testing. Early running with the main injector produced erratic combustion. As a diagnostic, two different hot long reach 12-mm spark plugs, SK-12164 and SK-12165, were tried and a major improvement was achieved. The run-in of the engine was completed with the SK-12164 plug.

At completion of the run-in a static air leak test was performed with the following results compared to the post assembly test:

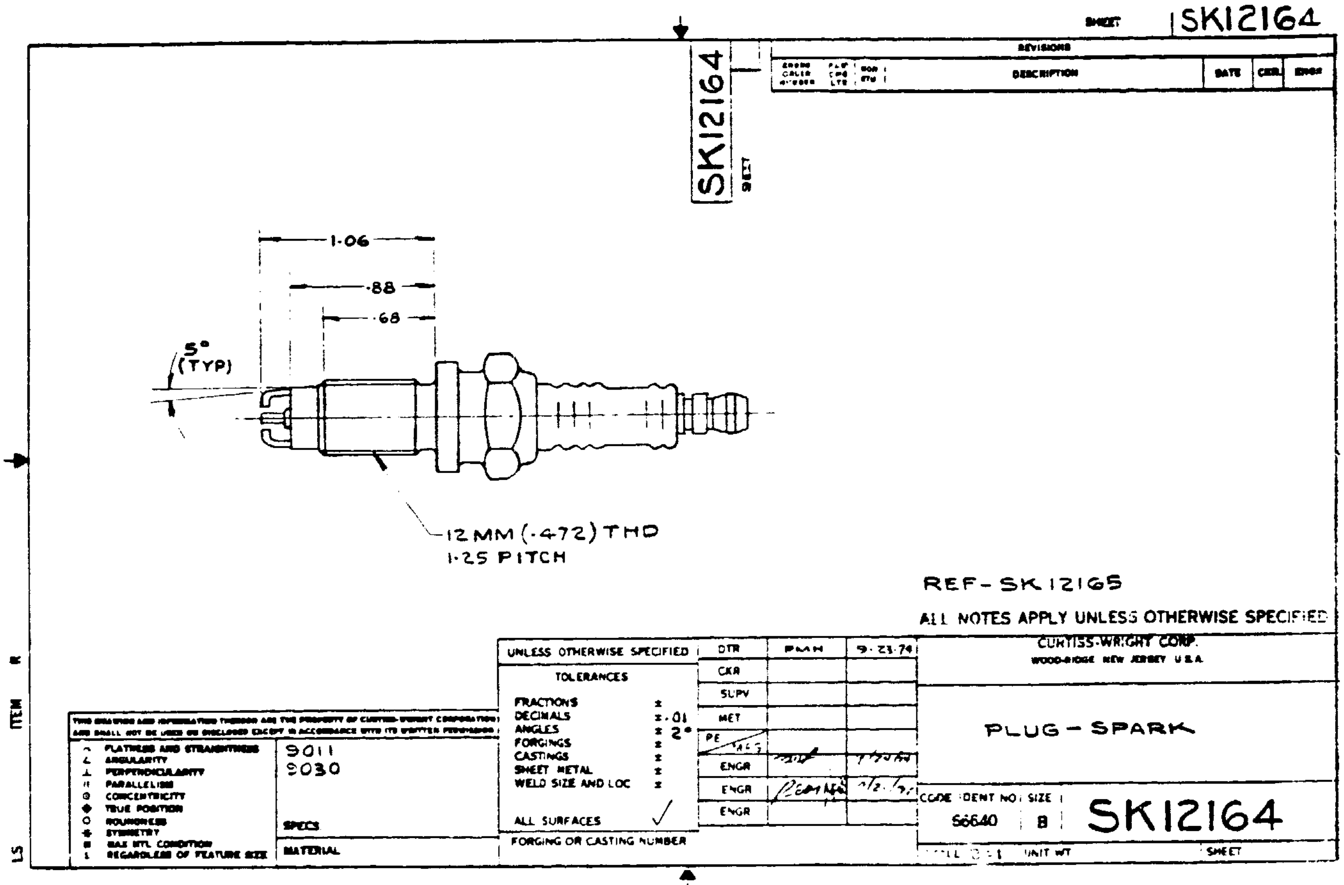
Status	Total Time-(h)	Pressure (psi)	Engine Cavity Pressure (psi)
Assembly	0	65	41, 42, 34 (cold eng.)
End of run-in	17:00	65	54, 52, 51 (hot eng.)
	46:00	65	60, 50, 55 (hot eng.)

The beneficial effects of the run-in on sealing are noted. The leakage tester consists of a gage, a 0.075-in.-diameter orifice, and a second gage in series.

TABLE 6.1

SPARK PLUGS & INJECTORS

			Spark plug			Pilot			Main		
Date	Time Total	Pt. No.	P/N	S/N	Shim	P/N	S/N	Shim	P/N	S/N	Shim
9/21	-	-	270057N4	1	0.021	210028N25	16	0.124	210028N8	1	0.080
9/24								0.149			
9/25	1:35				0.062						
	1:45							0.071			
	2:05				0.005						
	3:25			2	0.051						
	3:35				0.077						
	3:45				0.051			0.033			
	4:30		SK-12165	16	0.050			0.150			
	5:10				0.075						
	5:40		SK-12164	25	0.075						
	17:00	13		22	0.103						
	19:55	20			0.123						
	22:20	27	SK-12882N12	94	0.005						
	25:25	35			0.025						
	28:20	45			0.050						
	31:30	50			0.060						
	34:15	55			0.075						
	36:10	58		95	0.051				210028N15	4	0.008
	42:15	67							210028N8	1	0.080
	46:05	76	210057N12		0.000	210028N25	17	0.155			
	49:40	82	SK-12882N12	95	0.039						
	55:40	94	210057N12	89	0.005						
	56:20			91	0.005						
	65:50	103		97	0.010						
	66:05				0.000						
		110	END OF CURRENT TEST			69:20 Total Time					

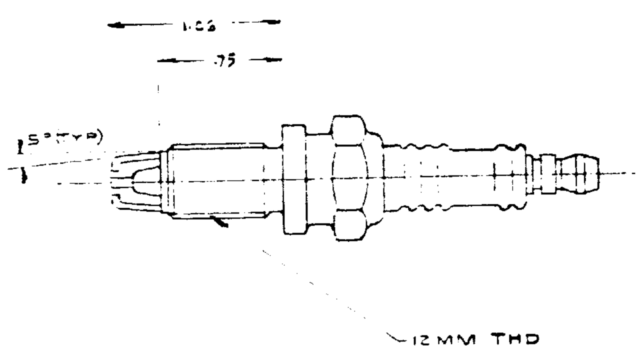

SK12164
REVISIONS
DESCRIPTION
DATE
1.06
.88
.68
5°
(TYP)
12 MM (.472) THD
1.25 PITCH
REF - SK 12165
ALL NOTES APPLY UNLESS OTHERWISE SPECIFIED
UNLESS OTHERWISE SPECIFIED
TOLERANCES
FRACTIONS ±
DECIMALS ± .01
ANGLES ± 2°
FORGINGS ±
CASTINGS ±
SHEET METAL ±
WELD SIZE AND LOC ±
ALL SURFACES √
FORGING OR CASTING NUMBER
DTR
CKR
SUPV
MET
PE
ENGR
ENGR
ENGR
9-23-74
CURTISS-WRIGHT CORP.
PLUG - SPARK
CODE IDENT NO
SIZE
56640
B
SK12164
UNIT WT
SHEET
9011
9030
SPECS
MATERIAL
FLATNESS AND STRAIGHTNESS
ANGULARITY
PERPENDICULARITY
PARALLELISM
CONCENTRICITY
TRUE POSITION
ROUNDNESS
SYMMETRY
MAX MTL CONDITION
REGARDLESS OF FEATURE SIZE
ITEM

SHEET SK12165

SK12165

REVISIONS

ENGRG ORDER NUMBER	PART CHG LTR	MOD SYM	DESCRIPTION	DATE	CART	ENGR



REF-CHAMPION SPARK PLUG CO.
NO. UP-77V & 304-107

ALL NOTES APPLY UNLESS OTHERWISE SPECIFIED

THIRD ANGLE PROJECTION

SPECS: 9030, 9011

MATERIAL —

UNLESS OTHERWISE SPECIFIED

TOLERANCES

FRACTIONS	±
DECIMALS	±.01
ANGLES	±
FORGINGS	±
CASTINGS	±
SHEET METAL	±
WELD SIZE AND LOC	±
ALL SURFACES	✓

FORGING OR CASTING NUMBER

DTR.	PMH	9-13-74
CKR		
SUPV		
MET		
PE / MFG		
ENGR		9-17-74
ENGR		9/17/74
ENGR		

CURTISS-WRIGHT CORP
ROTATING COMBUSTION ENGINES
WOOD-RIDGE, NEW JERSEY, U.S.A.

PLUG-SPARK

CODE IDENT NO.	SIZE	
66640	B	SK12165
SCALE 2:1	UNIT WT	SHEET

Engine Checkout and Acceptance Testing

This phase of the testing was directed at establishing the mechanical integrity of the engine and baseline performance as a vehicle for evaluating technology enablement features of a "highly advanced" aircraft engine.

A peak combustion pressure limit of 6.5 MPa (950 psi) was imposed due to discovery of a thin wall in a portion of the combustion pocket of a rotor of the same part number. Since the rotors are precision castings, the rotor in the engine would have the same discrepant wall thickness. Post-test calibration of the pressure tranducer, amplifier, and oscilloscope indicated that pressure readings during the test were approximately 35 percent too high. This restricted the maximum BMEP to approximately 950 kpa (138 psi).

In the interest of conservatism with a newly designed engine, it was decided to obtain all required data at 6000 rpm and below before higher speed investigations. No problems occurred at the higher speeds up to 8000 rpm.

Performance Results

a. Motoring Power

Figure 6.1 is a comparison of the estimated motoring power of the rig engine as compared with the motored rig engine with a reduction gearbox of 98% efficiency (per the manufacturer), the belt drive accessory box for driving the injection and oil metering pumps and shaft encoder, and a 1-lb dynamometer tare load. The dynamometer tare is equivalent to 0.75 kW (1 hp) at 6000 engine rpm. The major discrepancy between projected engine and "as motored" rig power is believed to be the effect of the turbocharger turbine acting as a brake and thus increasing the pumping work. Subsequent tests are planned for motoring without the belt-box and turbocharger. Evaluations will also be conducted using the Norland Digital Analyzer with four rotor housing pressure transducers to separate pumping work and mechanical friction. At some speeds Willans line projections have shown correlation with the projected motoring power.

b. Fuel Consumption

Figure 6.2 presents fuel consumption versus power characteristics of the rig engine as tested at various speeds. Data have been adjusted for the dynamometer tare and reduction gearbox efficiency. The maximum power developed was 79 kW (106 hp) at 7500 rpm as restricted by peak combustion pressure. Minimum fuel consumption was 282 g/kWh (0.464 lb/bhp-h) at 6000 rpm. Maximum power at 8000 rpm was restricted to 72 kW due to injection pump characteristics and test-stand linkage restrictions, which have since been removed. At 5000 rpm and particularly 4000 rpm, the fuel consumption characteristic versus power is undesirable and is a result of the turbocharger turbine matching.

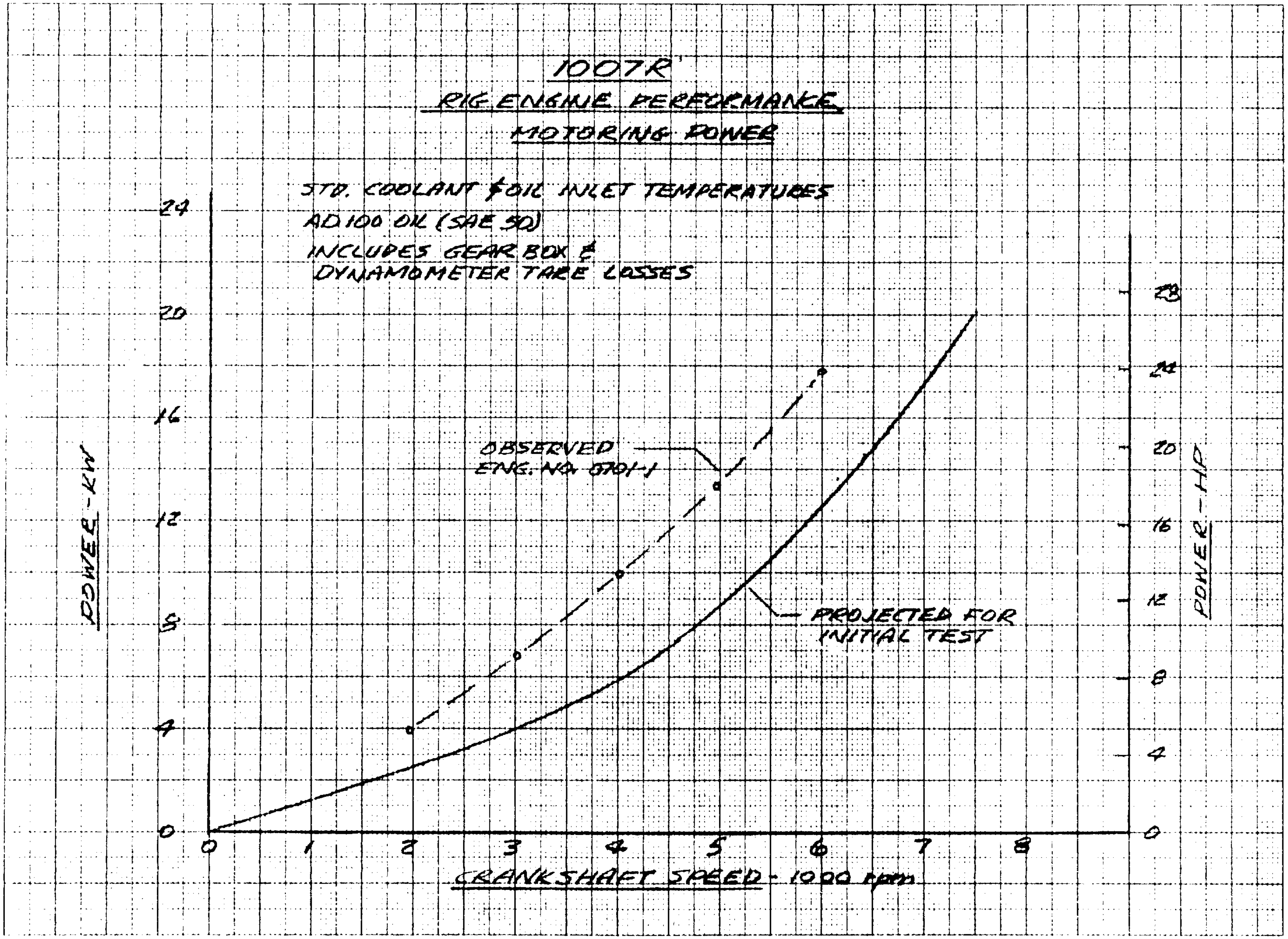


Figure 6.1. 1007R Rig Engine Performance Motoring Curve

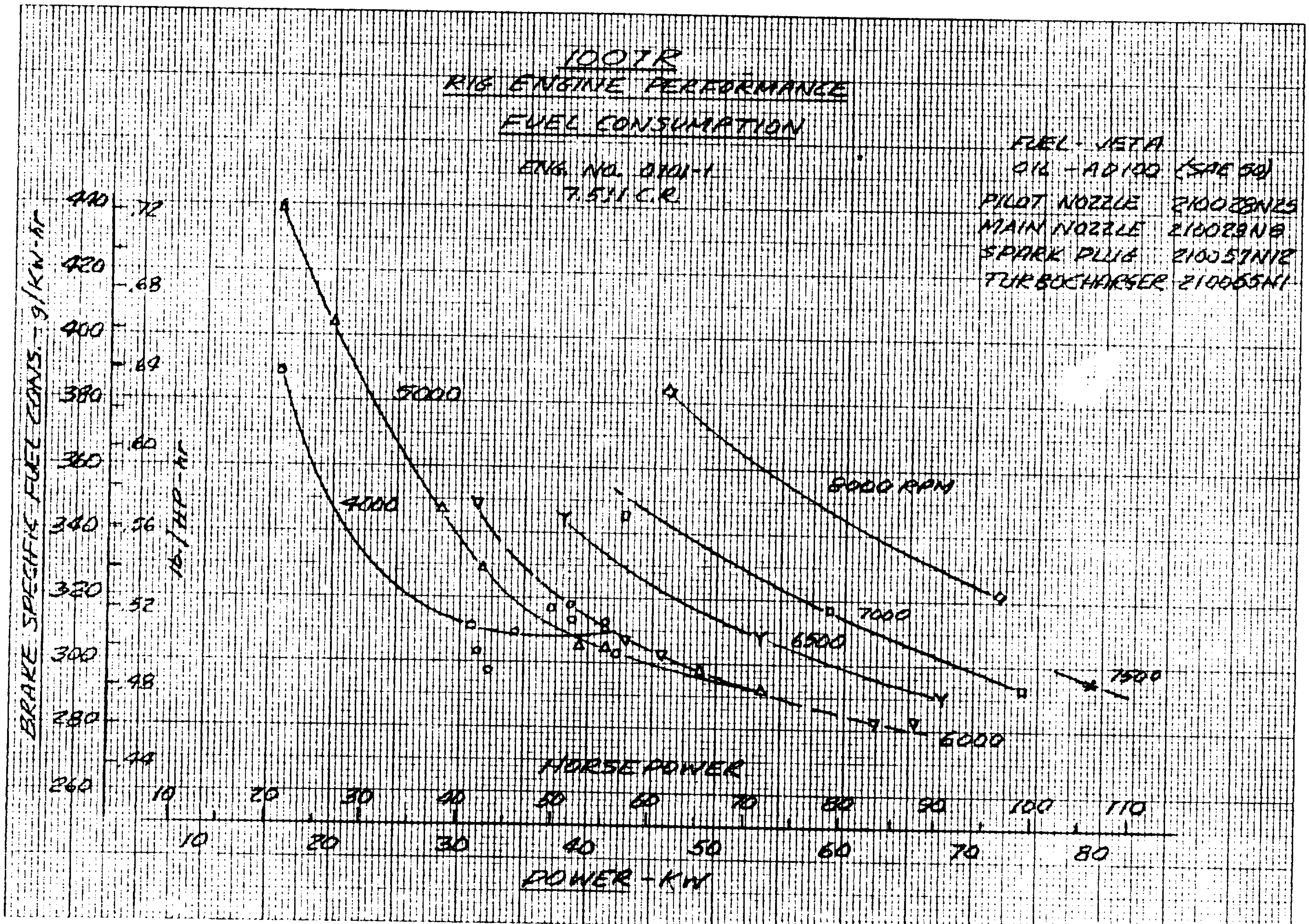


Figure 6.2. 1007R Rig Engine Performance Fuel Consumption

Figure 6.3 presents the mixture strengths associated with the data of Figure 6.2 and indicates richer than desired fuel air ratios at 4000 and 5000 rpm for optimum combustion efficiency. Extrapolation of mixture strengths at the higher speeds to higher powers results in richer mixture and lower efficiency, which suggests operation with a lower A/R turbine housing. All data shown were accumulated with the smallest total orifice area main injection nozzle. Testing of a larger orifice area nozzle producing shorter duration of injection and increased spray penetration indicated slightly reduced performance and combustion stability.

Figures 6.4 and 6.5 for 5000 and 6000 rpm, respectively, illustrate the characteristics of several engine performance and operating parameters plotted against BMEP. Noticeable are the effects of intercooling the combustion air for controlling peak combustion pressures on most parameters. Early testing had shown the desirability of high engine inlet temperature for its effect on combustion efficiency and stability. As a result intercooling was minimized.

Figure 6.6 is the map of the AiResearch compressor of the turbocharger installed on the engine. Figure 6.7 presents the engine operating lines on the excerpted lower half of the compressor map for engine speeds from 4000 through 8000 rpm. Inspection shows that an excellent match has been achieved with maximum compressor efficiency in the high engine RPM range required for the aircraft engine.

c. Heat Rejection to Coolant and Lubricant

Engine heat rejection to the coolant and lubricating oil for the test conditions are shown on Figures 6.8 and 6.9, respectively. The characteristics are basically as anticipated based on fuel-air ratios obtained at the various speeds. The heat rejection to the oil is artificially high at 7500 and 8000 rpm due to decreased below normal oil inlet temperature as a precautionary measure for the main bearings' protection.

Fuel Injection System

As described in the Task I Design Report, the fuel injection system for the 1007R engine consists of separate pilot and main systems using Stanadyne Model DM fuel pumps and Stanadyne slim tip nozzles for both systems. These systems were designed by Stanadyne using their computer simulation of the injection system dynamics. Finalized configurations and the results of their computer simulation are shown in Table 6.2 and Figure 6.10 for the main system and in Table 6.3 and Figure 6.11 for the pilot system. The fuel pumps are driven at one-quarter engine speed by a cogged belt drive system and, therefore, all the abscissa values in Figures 6.10 and 6.11 must be multiplied by a factor of four to convert from pump degrees to engine degrees.

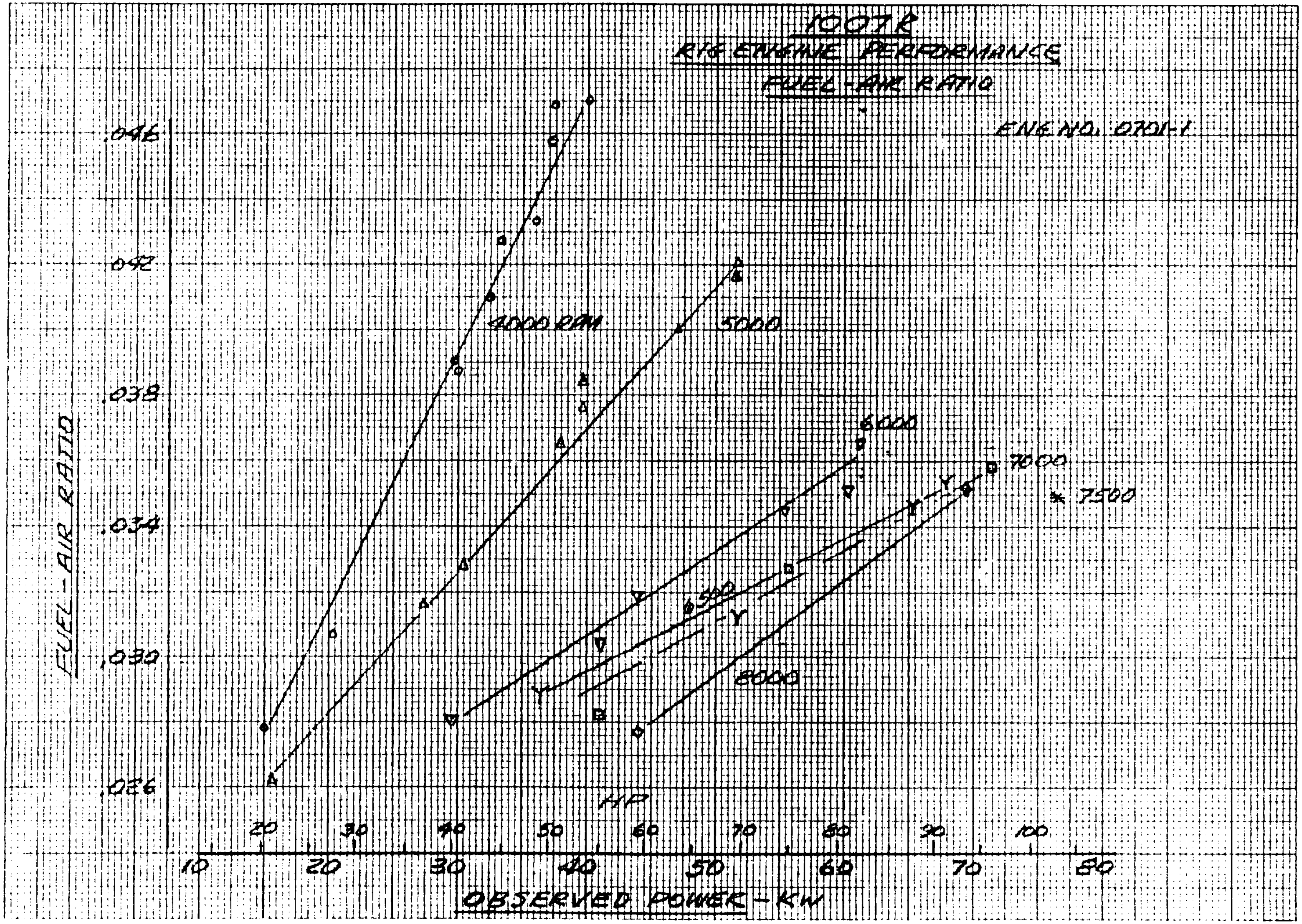


Figure 6.3. 1007R Rig Engine Performance Fuel-Air Ratio

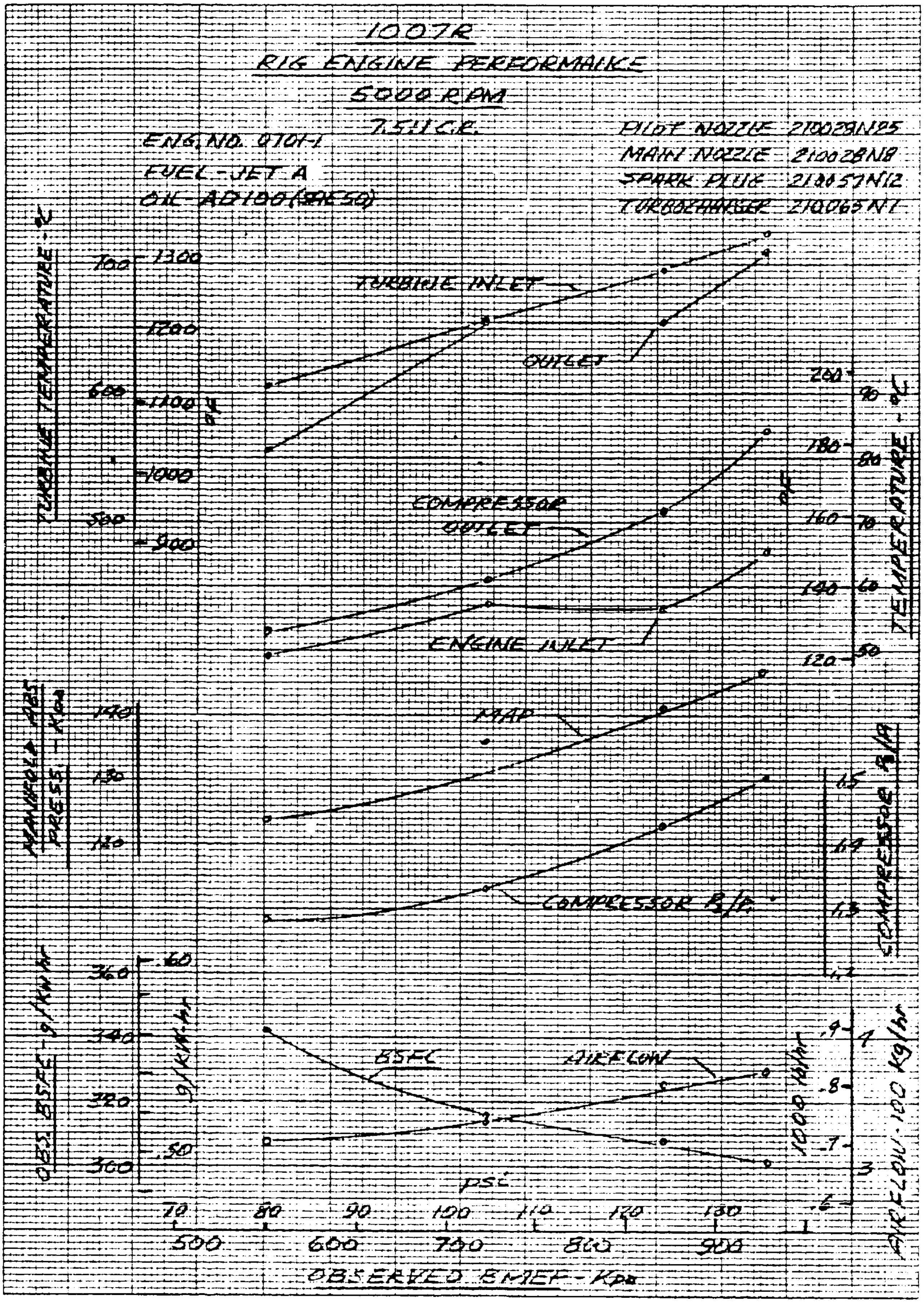


Figure 6.4. 1007R Rig Engine Performance 5000 RPM 7.5:1CR

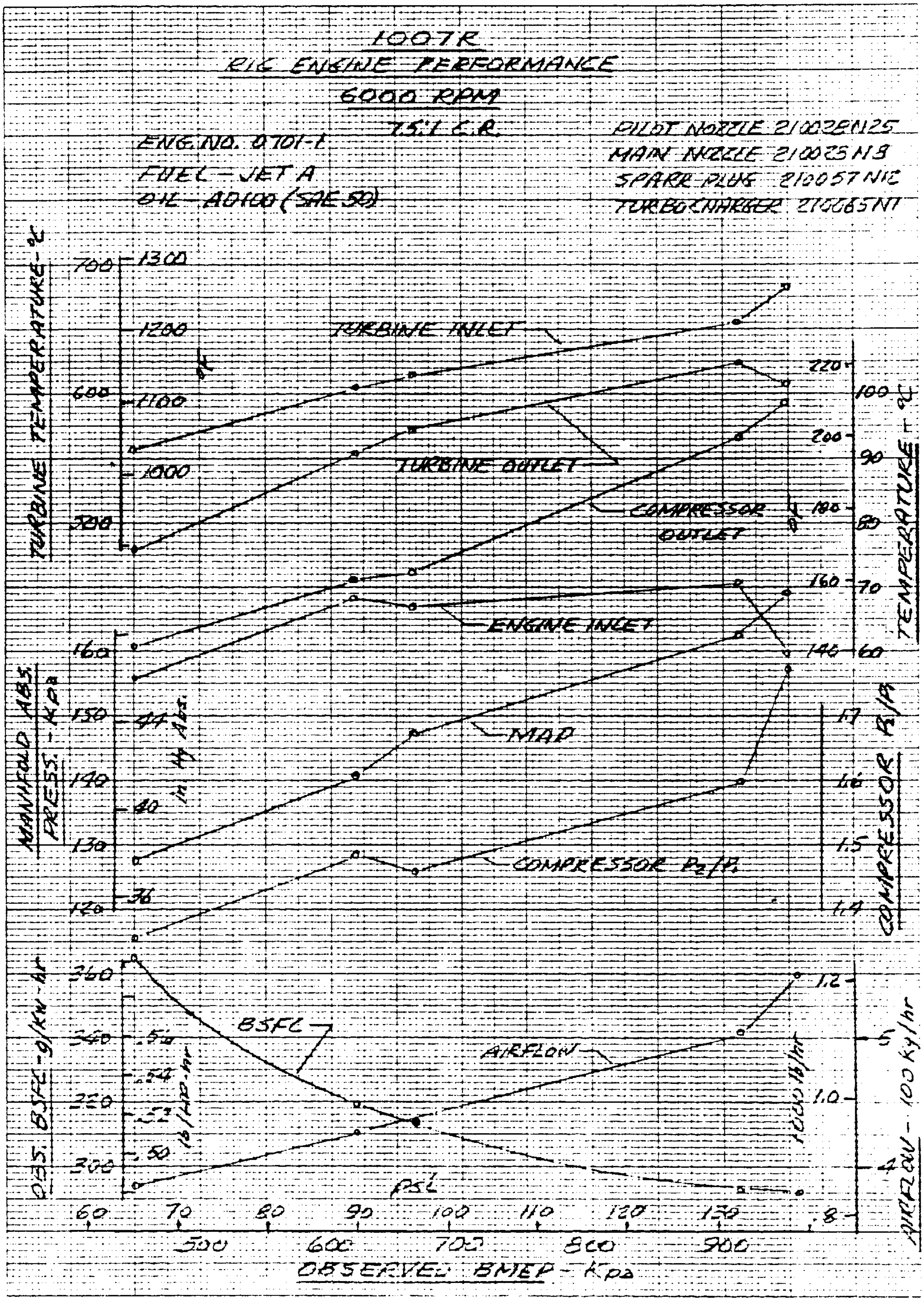


Figure 6.5. 1007R Rig Engine Performance 6000 RPM 7.5:1CR

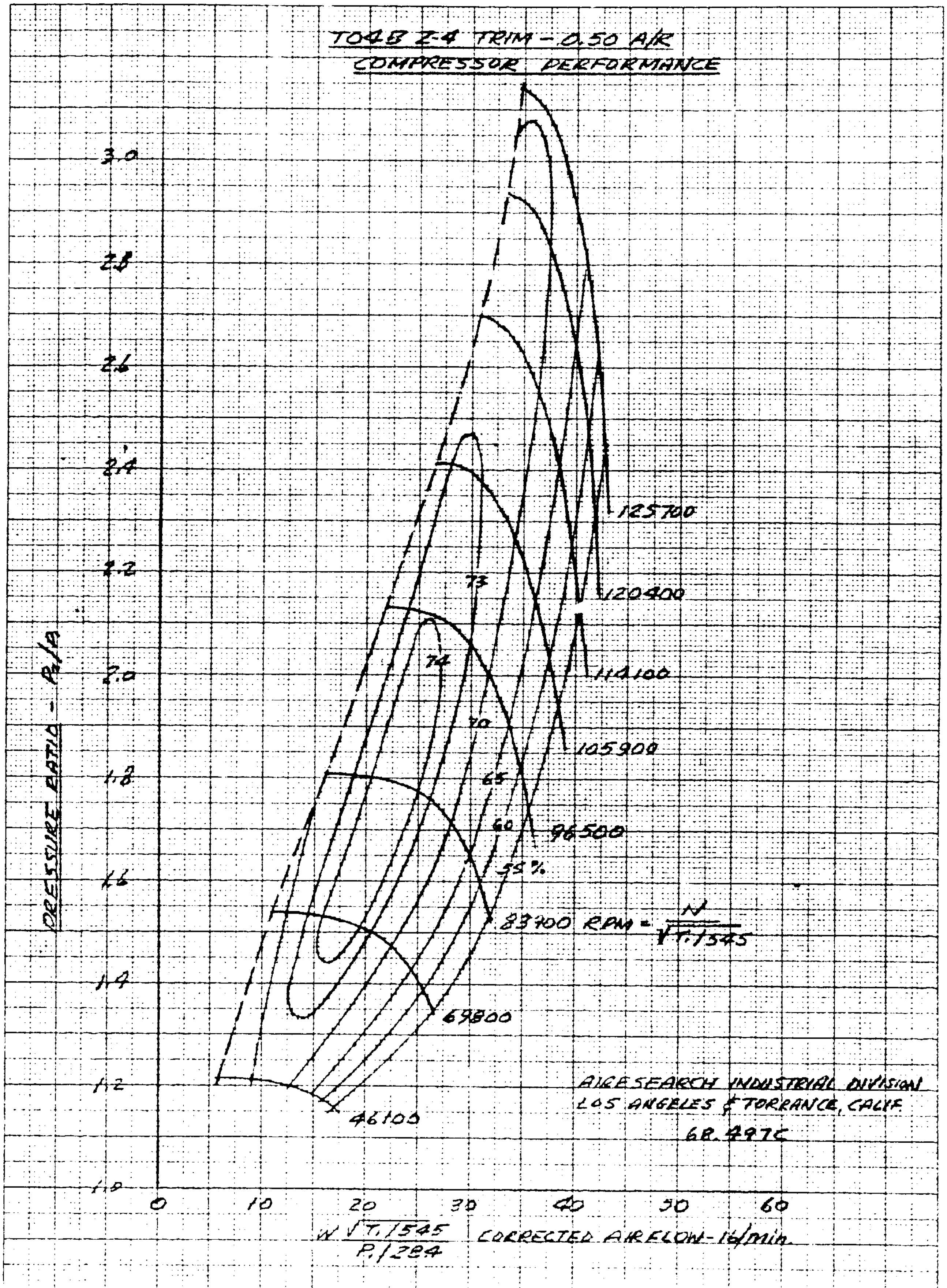


Figure 6.6. TO4B Z-4 TRIM - 0.50 A/R Compressor Performance

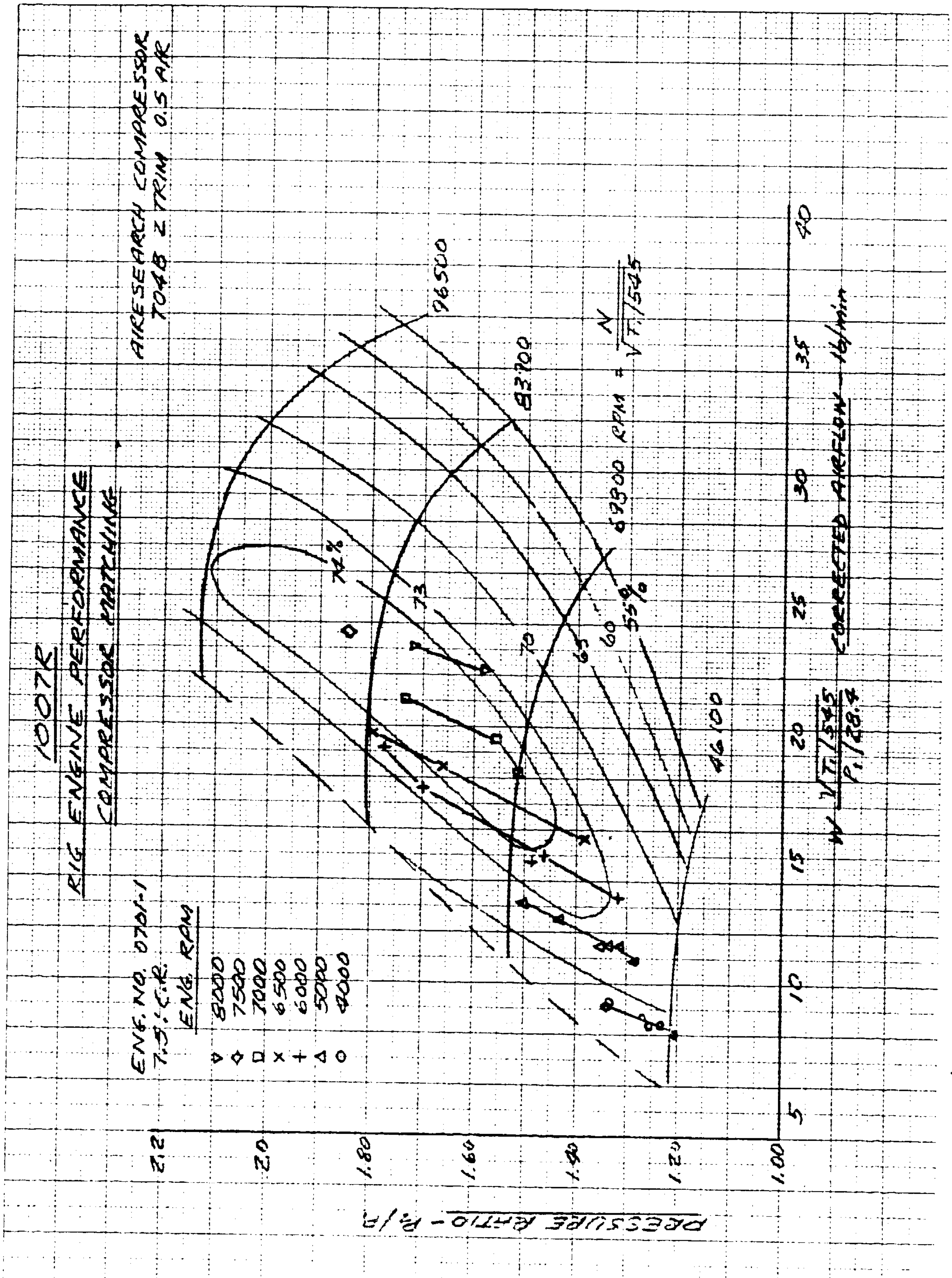


Figure 6.7. 1007R Rig Engine Performance Compressor Matching

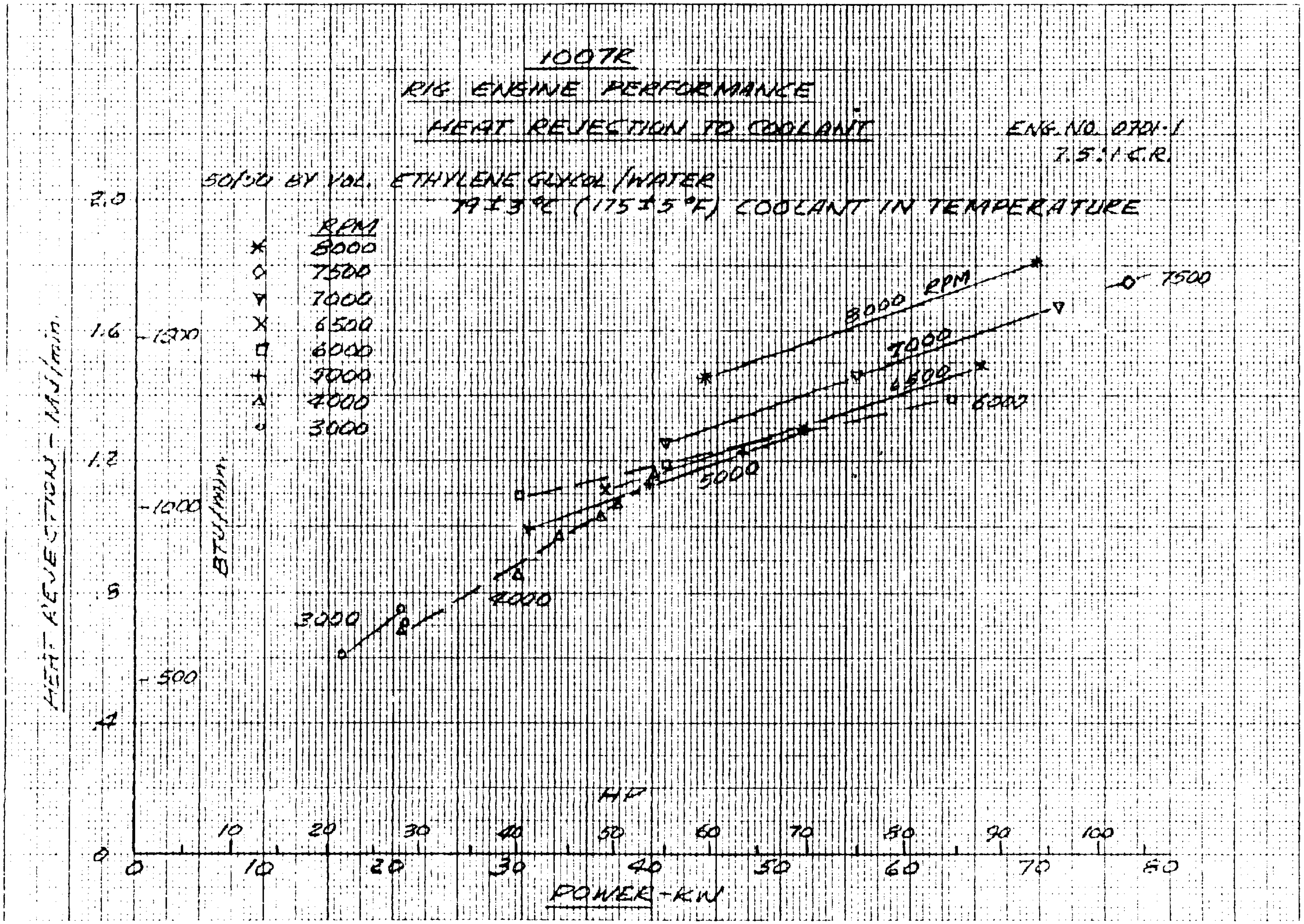


Figure 6.8. 1007R Rig Engine Performance Heat Rejection to Coolant

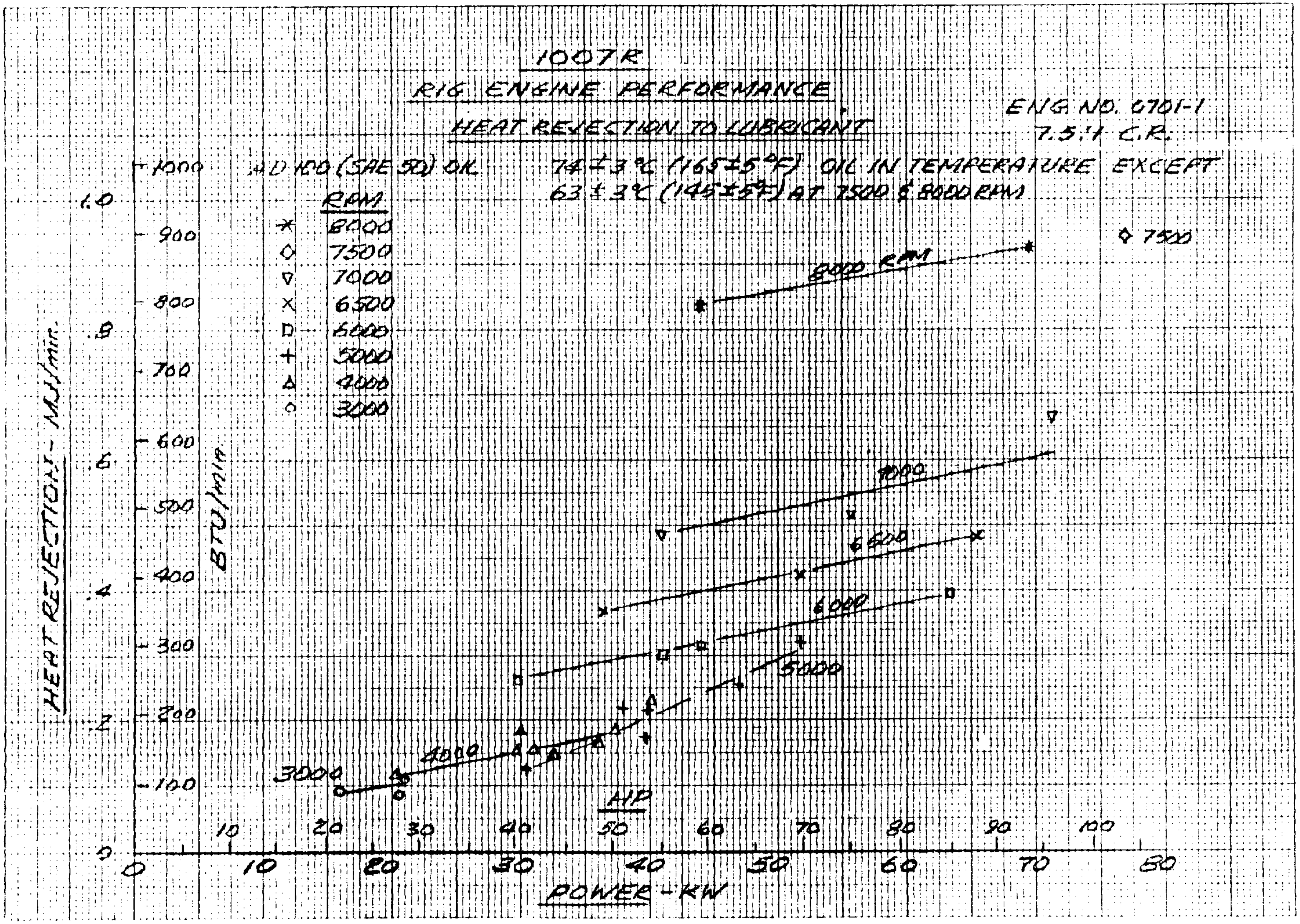


Figure 6.9. 1007R Rig Engine Performance Heat Rejection to Lubricant

TABLE 6.2

SIMULATION INPUT FOR THE MAIN INJECTION SYSTEM

1. Pump Configurations

Type	DM Pump
Number of Plungers	4
Diameter of Plungers	6.350-mm (0.250-in.)
Cam	18.288-mm (0.720-in.) radius
Delivery Valve Retraction	15 mm^3 (9.15×10^{-4} $in.^3$)
Snubber Valve Orifice	0.457-mm (0.018-in.) dia.
Fuel Delivery	75.8 mm^3 (4.63×10^{-3} $in.^3$) /stroke at 2000 pump rpm

2. Line Configuration

1.397-mm (0.055-in.) I.D. x 50.8-mm (2-in.) length

3. Nozzle Configurations

Type	Pencil Nozzle
"A" Dimension	12.7 cm (5 in.)
Length of Stud	8 cm (1.5 in.)
Opening Pressure	,442 kPa (3400 psi)
Orifice	6 x 0.279-mm (0.011-in.) dia.
Maximum Needle Travel	0.457 mm (0.018 in.)

MAIN INJECTION SYSTEMS

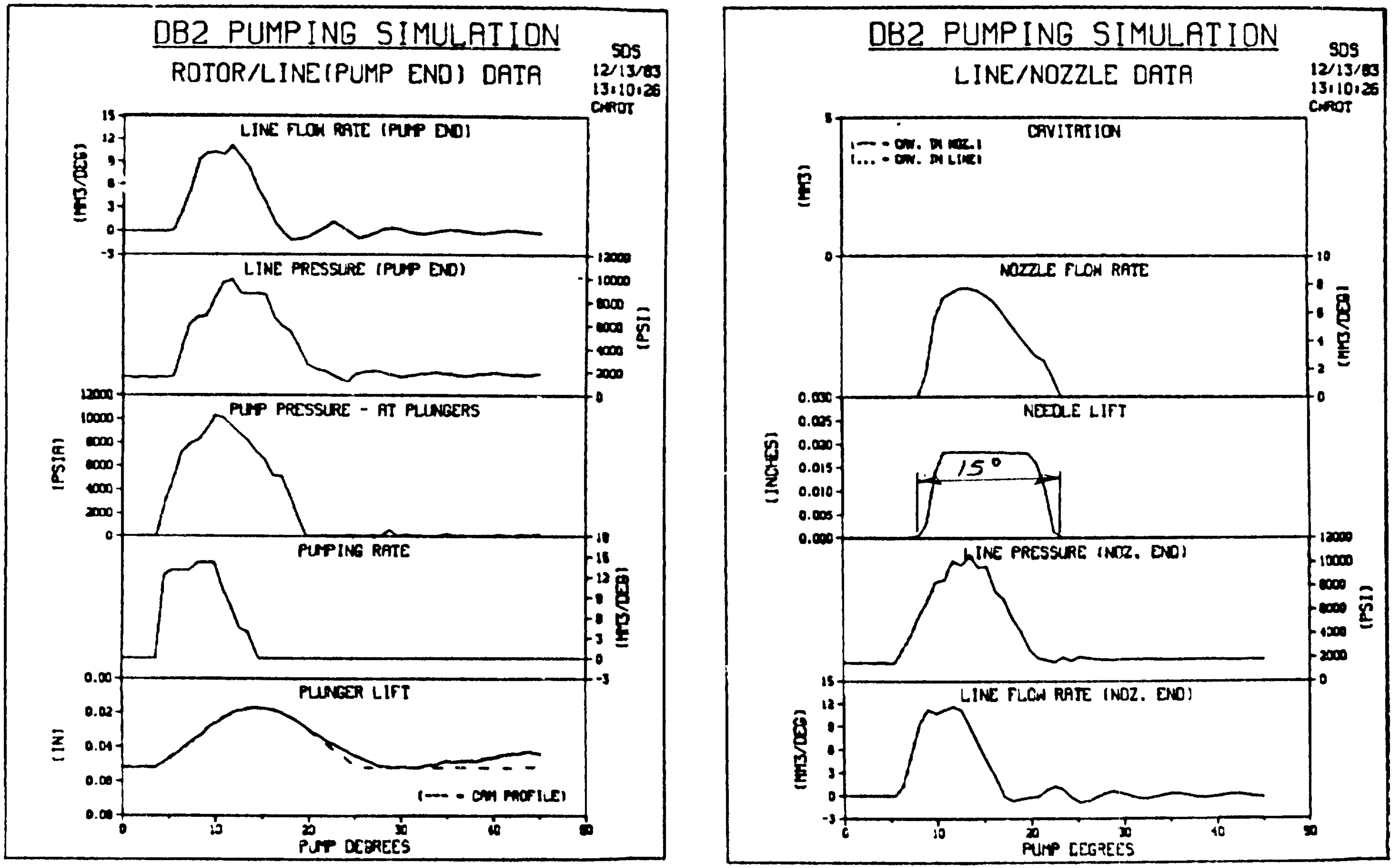


Figure 6.10. Main Injection Systems

TABLE 6.3

SIMULATION INPUT FOR THE PILOT INJECTION SYSTEMS

1. Pump Configurations

Type	DM Pump
Number of Plungers	4
Diameter of Plungers	6.858 mm (0.270 in.)
Cam	0.186 rad (10 degrees 40 min)
Delivery Valve Retraction	20 mm^3 (12.2 x 10^{-4} $in.^3$)
Snubber Valve Orifice	0.457-mm (0.018-in.) dia.
Fuel Delivery	4.65 mm^3 (2.84 x 10^{-4} $in.^3$) /stroke at 2000 pump rpm

2. Line Configuration

1.397-mm (0.055-in.) I.D. x 50.8-mm (2-in.) length

3. Nozzle Configurations

Type	Pencil Nozzle
"A" Dimension	12.7 cm (5 in.)
Length of Stud	3.8 cm (1.5 in.)
Opening Pressure	23,442 kPa (3400 psi)
Orifice	1 x 0.178-mm (0.007-in.) dia.
Maximum Needle Travel	0.3429 mm (0.135 in.)

PILOT INJECTION SYSTEMS

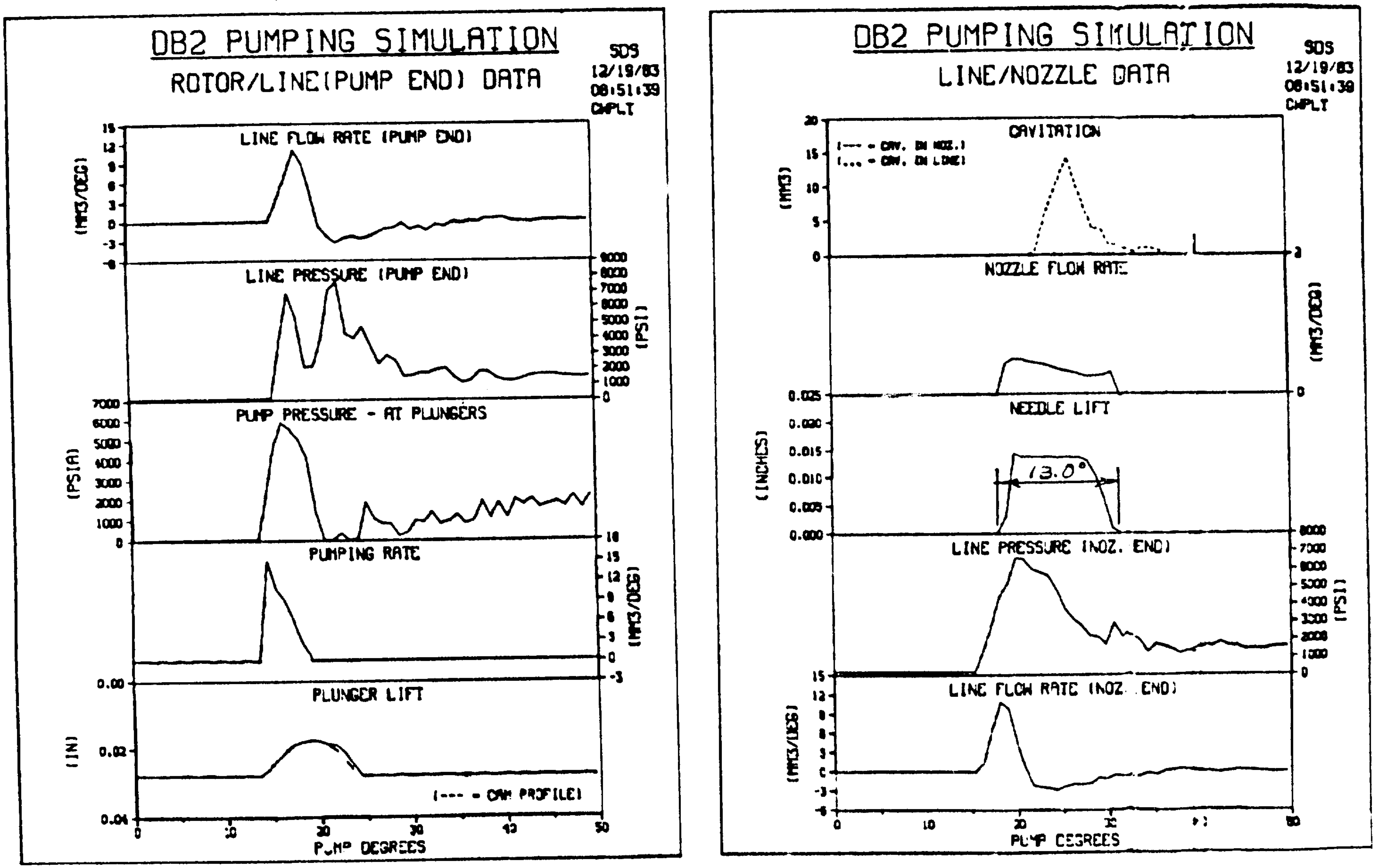


Figure 6.11. Pilot Injection Systems

a. Main Injection System

The design point for the main system was a delivery of 75 cubic millimeters of fuel at 8000 injections per minute with a peak pressure of 69 mPa (10,000 psi) and an injection duration of 60 degrees. As can be seen from the simulation results Stanadyne degrees. As can be seen from the simulation results, Stanadyne predicted that they could meet these objectives but that they were pushing each component to its established limit and that they would not be able to push the pump to either raise the speed, increase the delivery, raise the pressure, or shorten the injection duration. Based on this predicted performance two fuel pumps were built and rig-tested to verify their performance. The performance of the pumps on the test rig was documented in a test report. The report shows that they were able to achieve a delivery of 73 to 74 mm^3 per stroke at 2000 rpm pump speed (8000 rpm engine speed) with a duration of 15 pump degrees (60 engine degrees) at a peak line pressure of 55 mPa (8000 psi). A series of photographs were made of the injector needle lift and the injection line pressure at maximum delivery from 100 to 2000 rpm pump speed. As can be determined from examination of the photographs, the injection system dynamics were acceptable above 2000 rpm engine speed.

The fuel pump was equipped with a timing device which permitted the timing to be varied, using fuel pressure as a hydraulic power source, by 8 pump degrees (32 engine degrees). This device was also rig-tested as part of the fuel pump testing and the above-mentioned test report documents that it performs as expected.

The performance of the main injection system on the engine has been satisfactory, and there have been no mechanical problems with the pump or the injector. Figure 6.12 is a plot of the engine's brake mean effective pressure versus total fuel flow per injection. This plot shows each data point taken during the test program. Examination of these data reveals that there is a virtually a linear relationship between power output and fuel flow at each of the speeds and powers which have been run. This is a good indicator that there were no test points with different injection characteristics (such as eight stroking or multiple injections). Observation of the needle lift traces during engine operation also did not reveal any poor behavior.

Direct comparison of the performance of the main injection system on the engine and test rig cannot be made at this time as the engine has not been run to the maximum fuel flow available from the pumps (which is where the pumps were documented on the test stand). An analysis of the engine data to determine the average injection pressure (as measured in the injection line) was made, using the main fuel flow, measured main injection duration, and a discharge coefficient for the spray holes (including sac restrictions and needle seat restrictions) of 0.5. In examining a curve of average

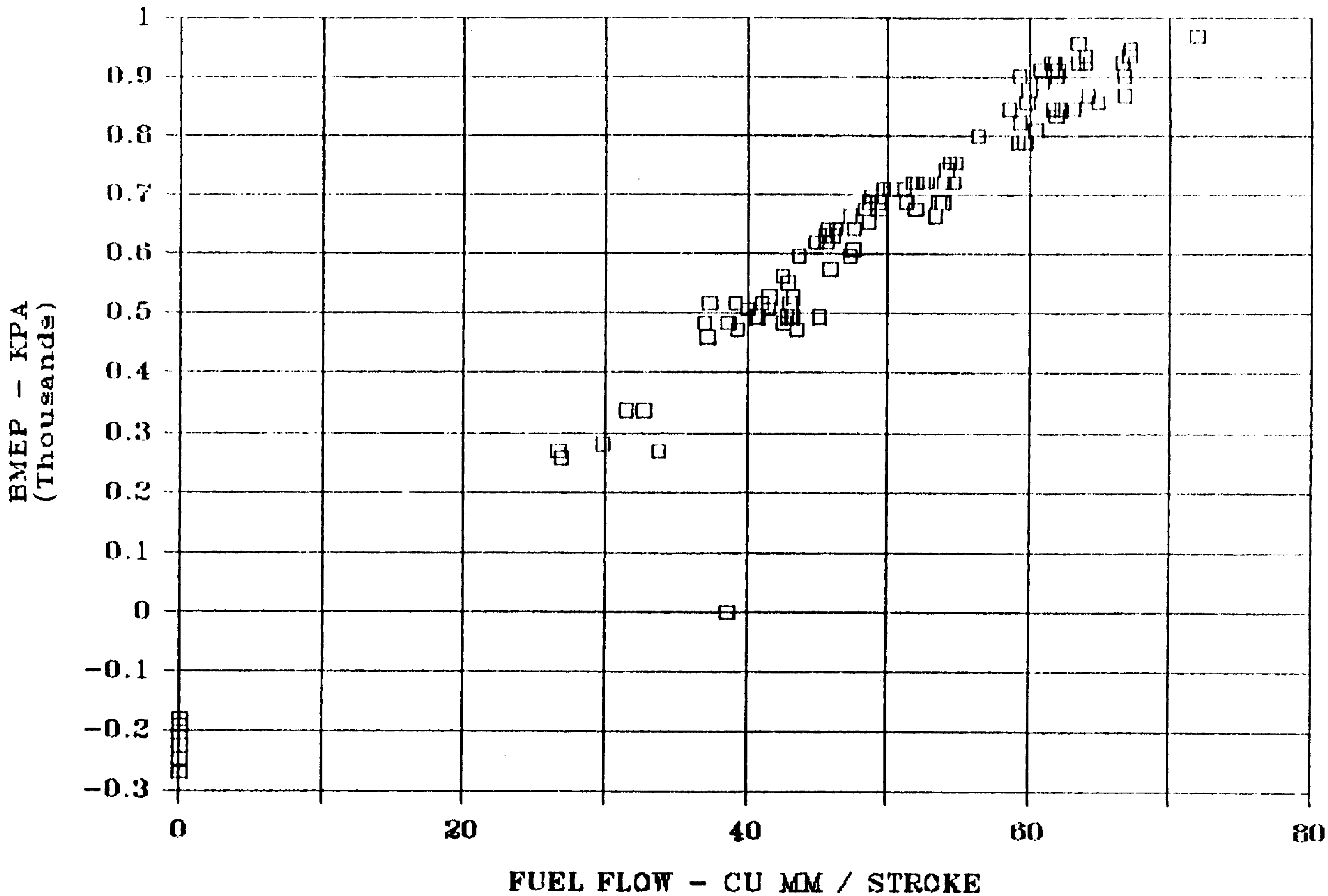


Figure 6.12. BMEP versus Total Fuel Flow for 1007R

injection pressure versus speed, Figure 6.13, it is useful to think of system compliance. To facilitate this, a system with zero compliance is defined as an infinitely stiff system with the average injection pressure being proportional to the square of the speed. At the other end would be the high-pressure common rail systems which are very soft and which for constant common rail pressure have no change in injection pressure with speed. Examination of Figure 6.13 reveals that the main injection system has a linear proportional relationship between average injection pressure and speed and would therefore be defined as a fifty-percent compliancy system.

b. Pilot Injection System

As mentioned earlier, the pilot injection system is very similar to the main injection system and was evolved in an identically parallel manner, starting with simulation and followed by rig testing, the report of which is provided in Appendix D.

The requirements of the pilot system are quite different from the main in that the fuel quantity is much lower. Past experience with other engines has revealed that the pilot fuel flow (for optimum performance) is about five percent of the maximum total flow. Thus if the engine has a rated total flow of 100 mm^3 per stroke it would be expected that the pilot flow requirement would be for 5 mm^3 per stroke. For this reason the needs for this rig engine were bracketed to a flow range of between 2 and 8 mm^3 per stroke. Since this rig engine is turbocharged to higher power levels than the naturally aspirated engine can attain, it was predicted that the pilot fuel quantity would probably not be the same as required by the naturally aspirated version of the engine which is in the 2 to 3 mm^3 per stroke range. Figure 6.14 is a plot of pilot fuel flow versus engine speed for the last fourteen points of the engine test, which reflect as nearly as is currently known the optimum pilot fuel flows for this engine. All of the pilot fuel flows are between 2 and 4 mm^3 per stroke with a mean of about 3 mm^3 per stroke. Figure 6.15 is a plot of average injection pressure versus speed with the same assumptions and means of calculation as used for the main; and looking at the 4000 and 8000 rpm points which are at virtually the same flow in cubic millimeters per stroke reveals that the pilot injection system has a compliancy index of virtually zero (meaning that it is very stiff). This type of performance can have adverse effects on engine operation as it leads to a wide variation in spray characteristics in the critical spark plug area, particularly in mean droplet size.

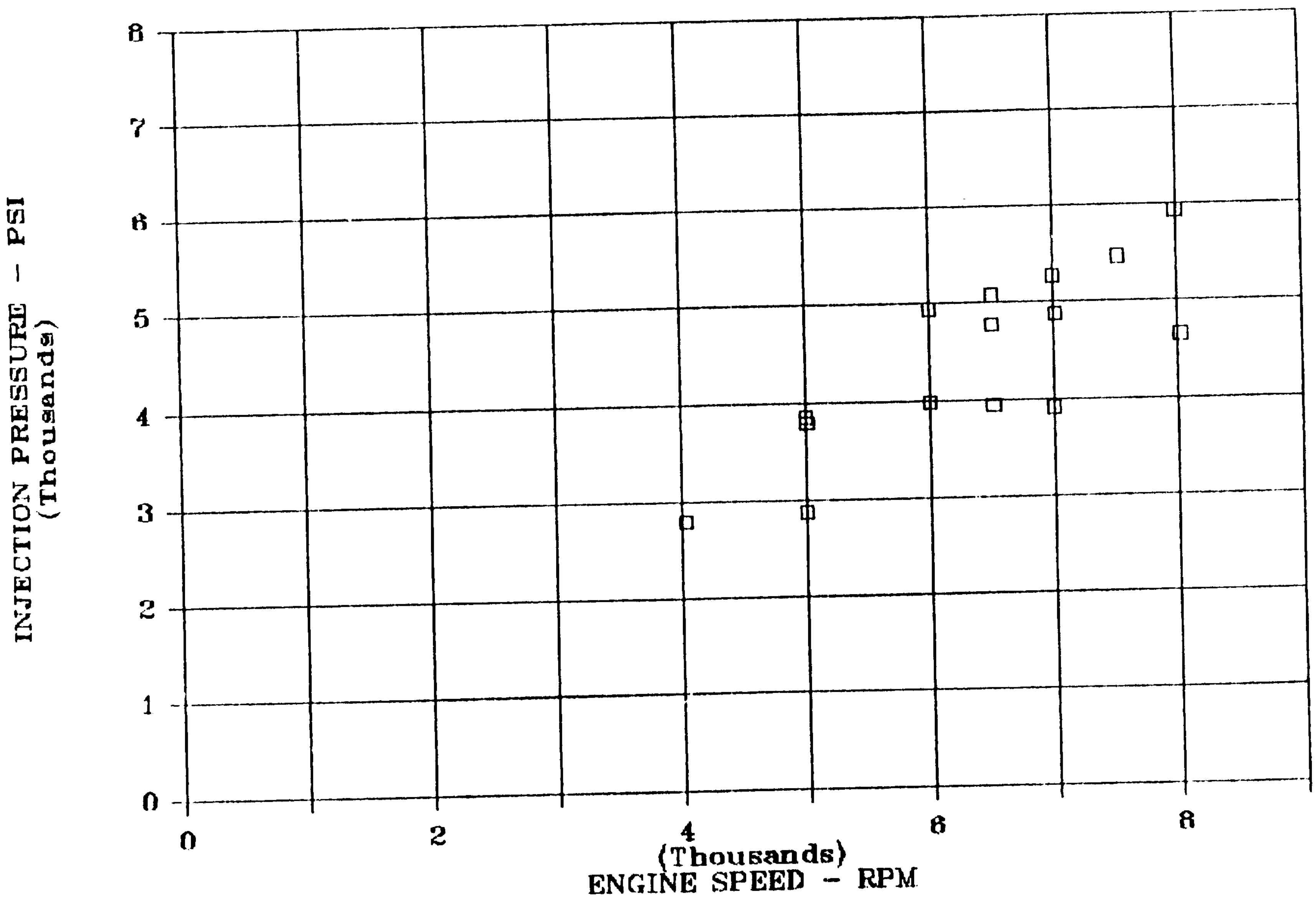


Figure 6.13. Main Injection Pressure versus Speed for 1007R

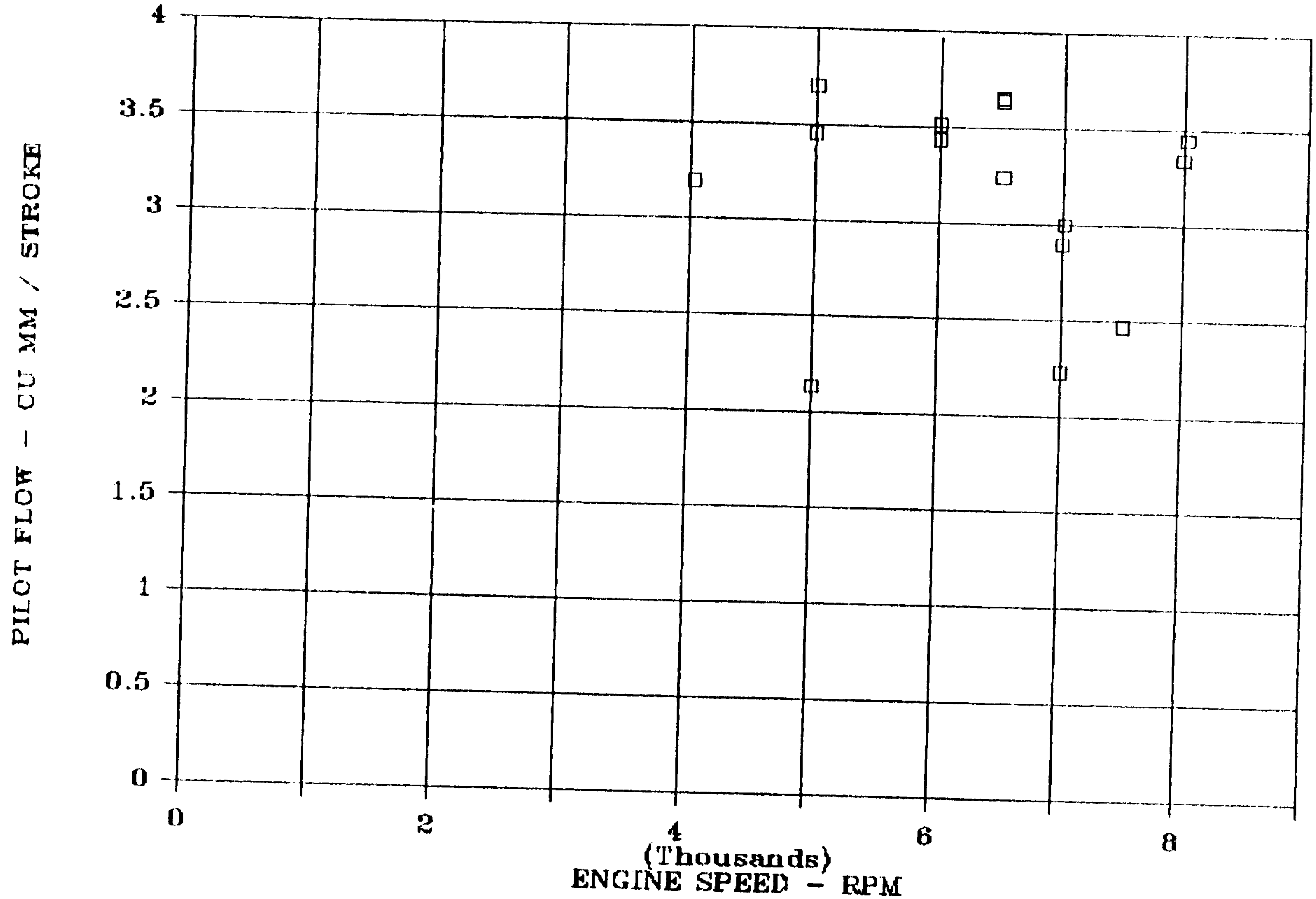


Figure 6.14. Pilot Fuel Flow versus Engine Speed for 1007R

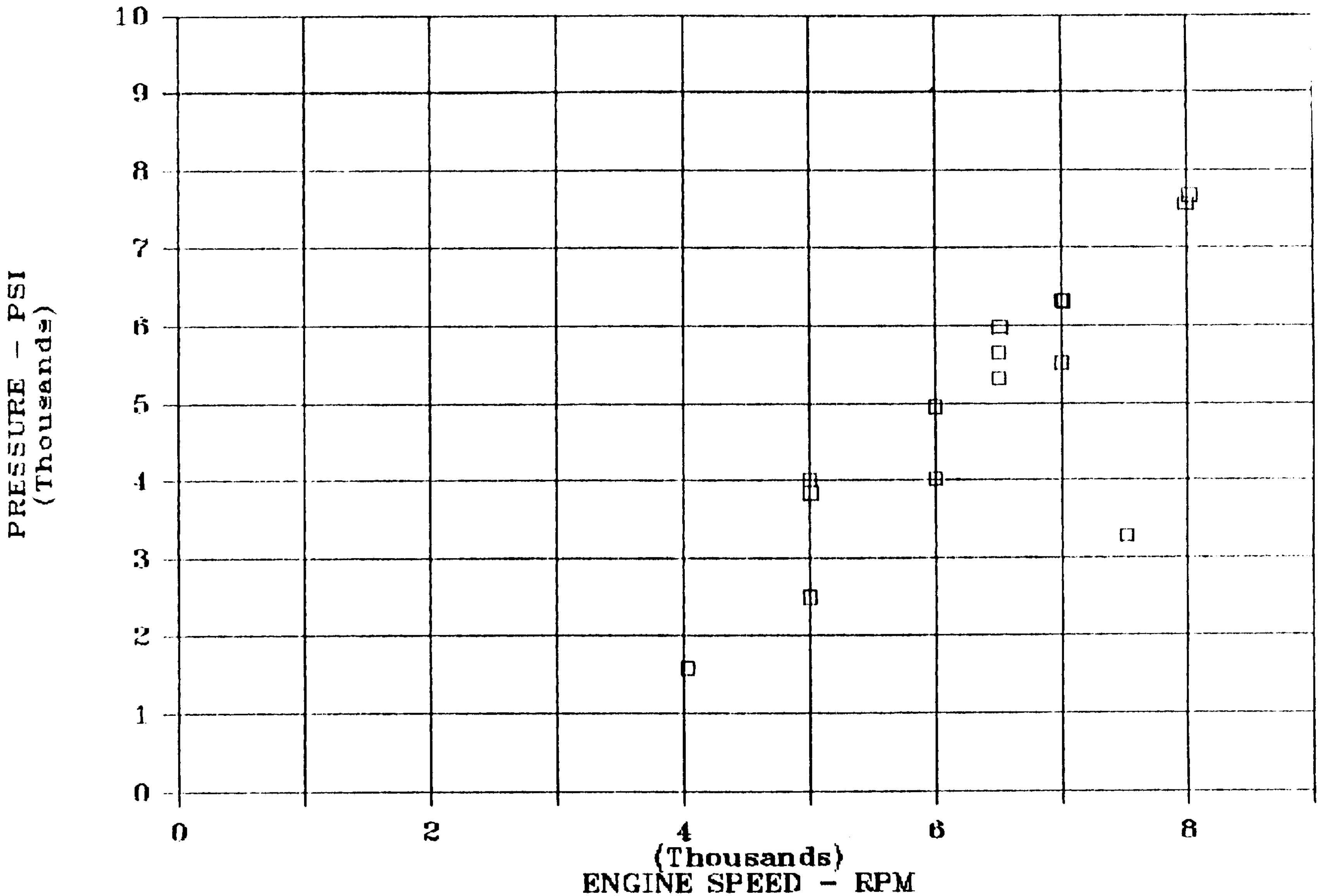


Figure 6.15. Pilot Injection Pressure versus Speed for 1007R

Vibratory Characteristics

a. Torsional Vibration

The torsiograph system used consisted of a toothed disk attached to the accessory end of the crankshaft, a magnetic inductive transducer and an FM discriminator circuit.

Based on previous rotary engine designs, an allowable torsional vibratory limit of 6360 rad/sec^2 was established. Operating below this limit will prevent damage to the engine stationary gear. The torsional analysis of the rotating components for this installation is covered by IOM, M.R. Kulina to For The Record, November 21, 1983, 1007R - WX20-6 Driveline Vibration Anaysis, included in Appendix A.

The system prime torsional resonance consists of the engine inertia, which includes the two counterweights, rotor and flywheel, together with the inertia of the test-stand gearbox oscillating against the inertia of the dynamometer. The two inertias are connected by a Koppers 'soft' coupling. The coupling's rubber elements were selected such that this resonance would occur about 1100 rpm for the first engine order excitation.

Test data (Figure 6.16) show that first mode resonance occurs during starting at 1300 cpm with a maximum torsional amplitude of ± 0.166 rad ($\pm 9.5°$). This amplitude corresponds to an acceleration of 3073 rad/sec^2 or about 50% of the allowable limit.

Nonlinearity of the rubber is noted by the shift of this resonance to 1050 rpm during engine rundown.

The actual vibratory torque on the Koppers coupling is ± 546.5 N·m (± 4.836 in·lb). Peak torque capacity is 1615.9 N.m (14,300 in·lb). Since the resonance occurs at 50% of the proposed idle speed, only a limited number of resonant cycles occur, less than 10 for any start.

For this mode of vibration there is no magnification of the vibratory torque through the Thomas coupling between the engine and gearbox. Only ± 63.7 N·m (± 564 in·lb) of vibratory torque occurs. Allowable peak torque is ± 2135.7 N·m (18,900 in·lb.).

Torsional amplitudes fall below ± 0.0175 rad ($\pm 1°$) at about 3000 rpm and continue decreasing with speed.

During an acceleration from 6000 to 7000 rpm, the torsiograph signal suddenly became erratic. First, at 6300 rpm, a sudden increase in the torsional signal would occur 2 to 3 times a second and only for a short duration. At 7000 rpm, the signal was still erratic but more persistent. Torsional amplitudes of ± 0.070 rad ($\pm 4°$) to ± 0.087 rad ($\pm 5°$) were recorded.

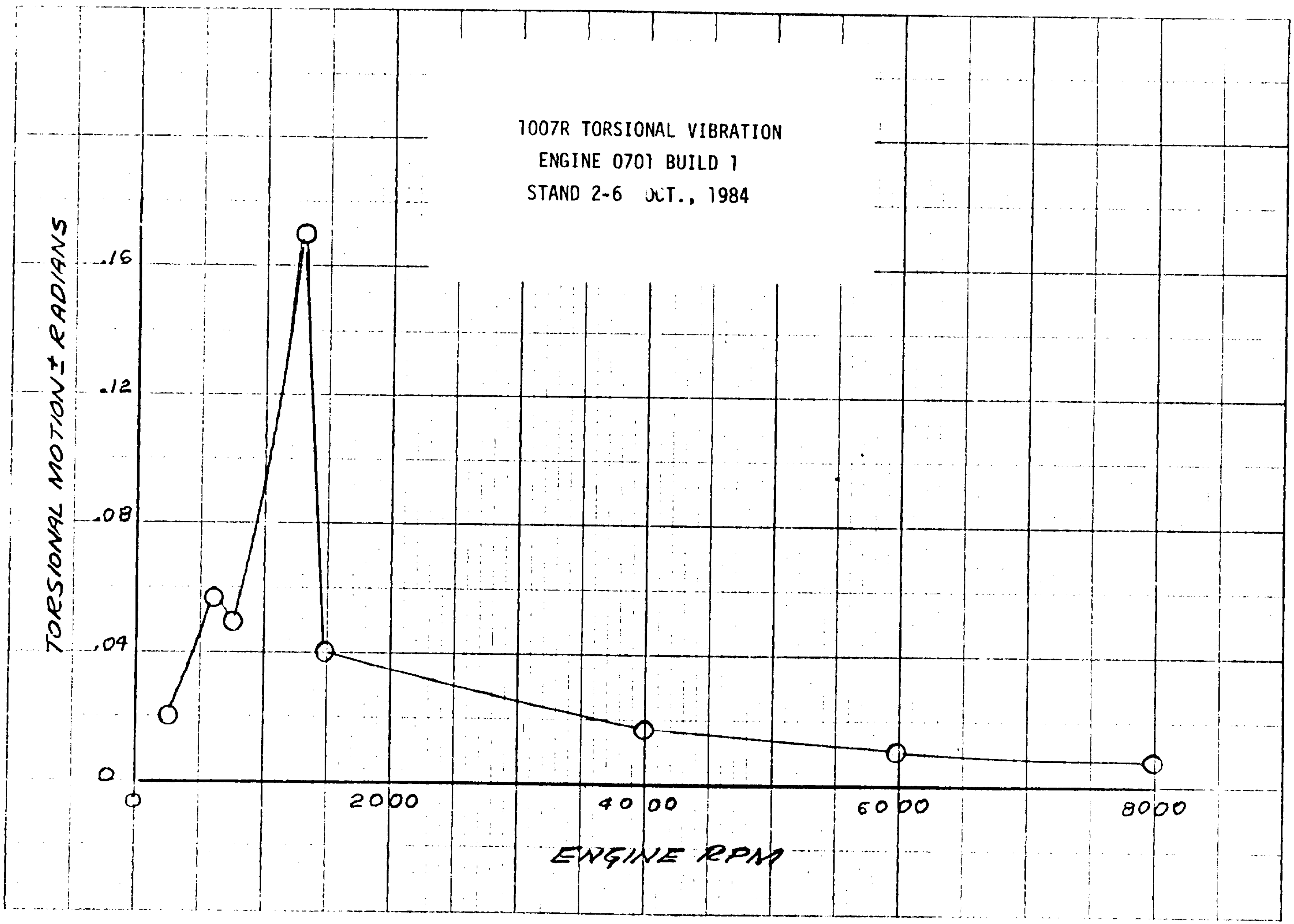


Figure 6.16. 1007R Torsional Vibration

The apparent "erratic" signal was being generated by a 120-tooth disk at the accessory end of the crankshaft. At the drive end 60 teeth were cut into the flywheel to serve as an engine tach generator. Since both locations can be used to identify torsional response of the driveline, the torsiograph was transferred to the flywheel.

The torsional response as measured at the flywheel was identical to that recorded at the accessory end of the crankshaft from 0 speed up through 6300 rpm; above that speed the flywheel torsional signal showed no erratic signal. The response was first-engine order and the amplitude at 8000 rpm was ± 0.0035 rad ($\pm 0.2°$).

The following table shows the relative torsional amplitude for the first three modes of the driveline.

Component	Mode		
	1	2	3
Torsiograph Acc. End	+1.00	+1.00	+1.00
Rotor	+1.00	+0.95	+0.88
Torsiograph Flywheel	+1.00	+0.40	-0.43
Dynamometer	-0.04	0	0

The torsiograph at the flywheel can be used as a satisfactory means of measuring rotor torsional motion.

At the present time there is no explanation of the "erratic" response of the accessory end torsiograph. Although the prevailing evidence points to an instrumentation problem, further investigations will be undertaken to confirm the cause.

b. Translational Vibration

Two translational pickups were mounted at the anti-drive end of the engine. C.E.C. type 4-131 transducers were used.

The translational motions were primarily first order with maximum amplitudes of ± 0.033 mm (± 1.3 mil). The maximum horizontal occurred at 8000 rpm while the maximum in the vertical direction occurred at 6000 rpm. There were no indications of major engine vibration resonances throughout the speed range.

Bearings

In an attempt to monitor lubricant temperature rises due to the bearings, the engine was instrumented with thermocouples to measure lubricant outlet temperatures. These temperatures were measured at two outlet points, corresponding to the drive and anti-drive ends of the engine. A sketch of these locations appears in the Task I Design Report. The flows at each location are made up of contributions from both the inner and outer main bearing leakage, as well as the rotor bearing leakage. A further complication in the analysis of these temperatures is the fact that the rotor bearing leakage fluid is also used for rotor cooling. Hence, the raw data for lubricant outlet temperatures are of limited use in the analysis of bearing performance. A more in-depth analysis of these data along with other data does, however, give some insight into actual bearing temperatures.

Figure 6.17 shows a plot of lubricant temperature rise multiplied by the volume flow rate versus engine speed. Although not an energy parameter, this quantity is proportional to the energy transferred by the fluid. (An even distribution of flow is assumed.) This quantity is significantly greater for the drive end than the anti-drive end, especially at the lower speeds. This is believed to be a result of poor scavenging at the anti-drive end due to the presence of the rotor gear. Since the energy from the bearings should be the same for both ends, this difference is attributed to convective heat transfer from the rotor. As seen in the figure, this difference becomes less as speed is increased (at constant power). This can be attributed to the fact that the component, because of bearing dissipation, increases linearly with speed[1], while rotor temperature (probably) decreases with speed (Figure 6.20). The lack of linearity in the bearing dominated curve (nongear end) is probably due (at least in part) to the character of the flow curve seen in Figure 6.18. Figure 6.17 implies that the bearing dissipation effects component of lubricant temperature rise will be dominant at both ends of the engine for speeds above 8000 rpm.

The above assertions are supported by Figure 6.19. Here, temperature rise in the lubricant is normalized by dividing by both total flow <u>and</u> engine power and plotted versus speed. Here the gear end temperature is seen to follow somewhat the trend in observed exhaust temperatures (and perhaps rotor temperatures), as seen in Figure 6.20. The other end displays similar trends at the low rpm's, but reverses course and begins a fairly linear rise at 5000 rpm. This is taken to be the transition point between rotor-dominated behavior and bearing-dominated behavior. It could be speculated that this transition point for the gear end occurs at approximately 8000 rpm and that for higher speeds both curves will be closely parallel.

Figure 6.21 indicates the dependence of the flow-temperature product on power for a constant speed. As both contributing effects (rotor temperature and bearing dissipation) are roughly linear with power, the resultant temperature rise could also be expected to be somewhat linear. It would appear that this is the case, at least for those speeds and powers plotted.

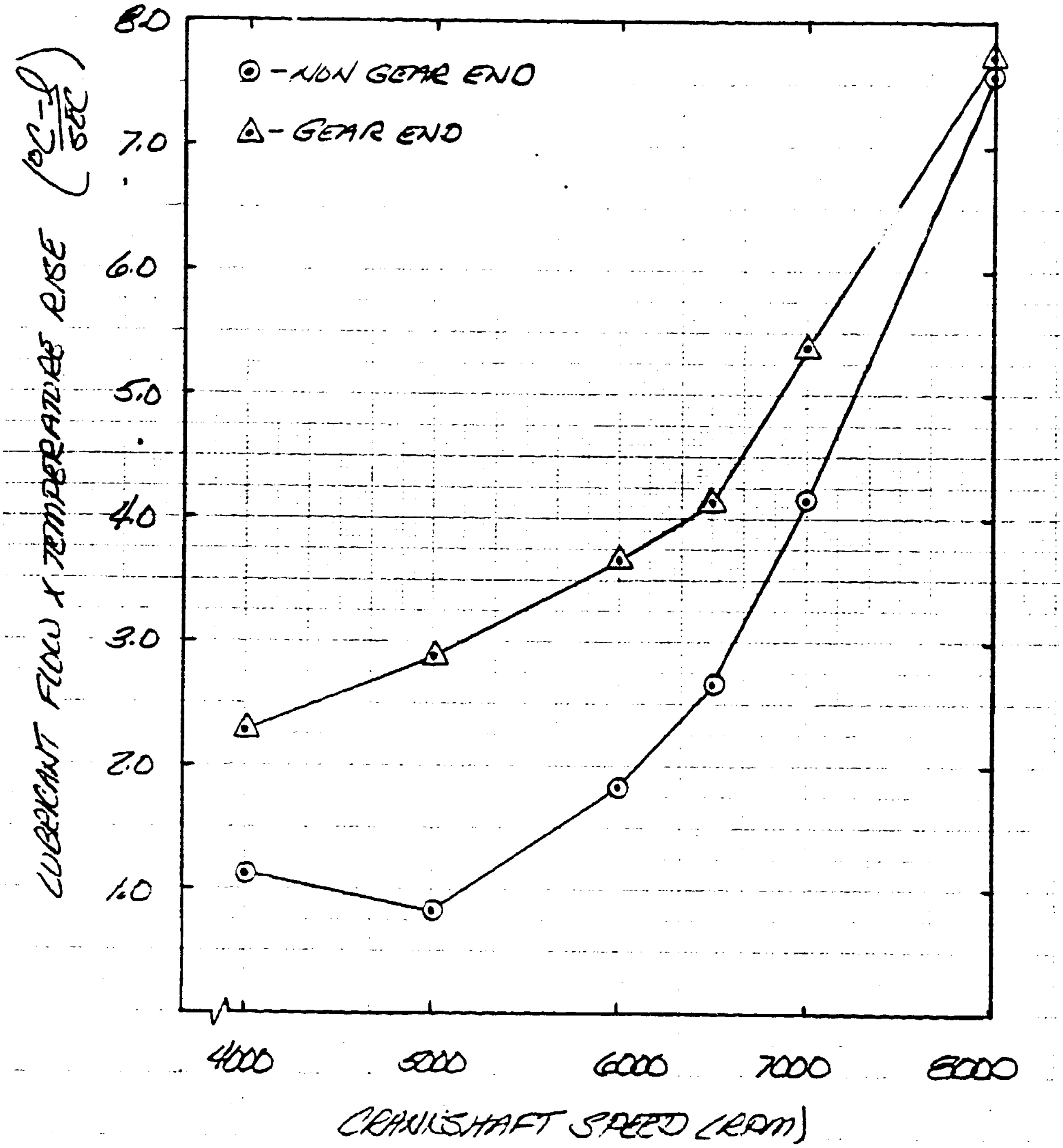


Figure 6.17. Product of Lubricant Flow and Temperature as a Function of Speed

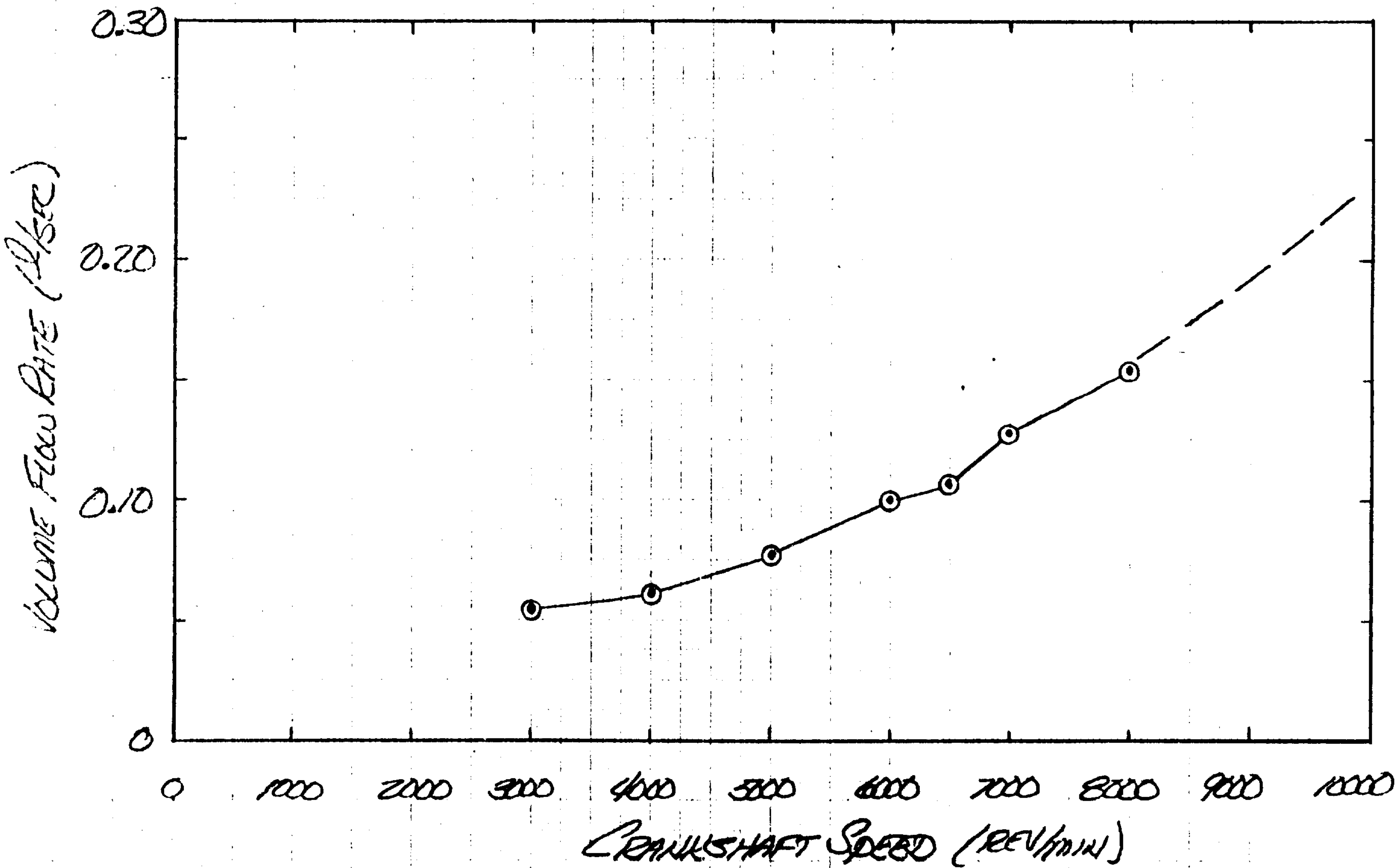


Figure 6.18. Total Lubricant Flow as a Function of Speed

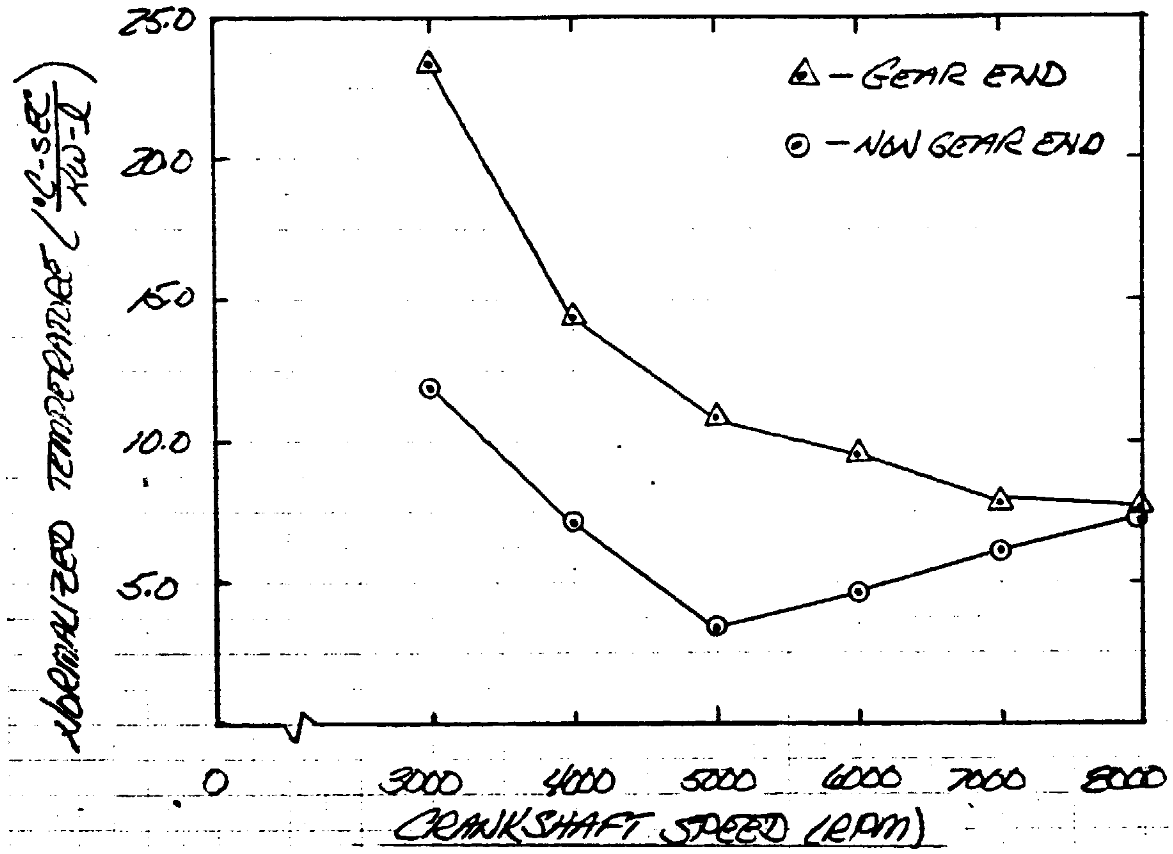


Figure 6.19. Lubricant Temperature Rise

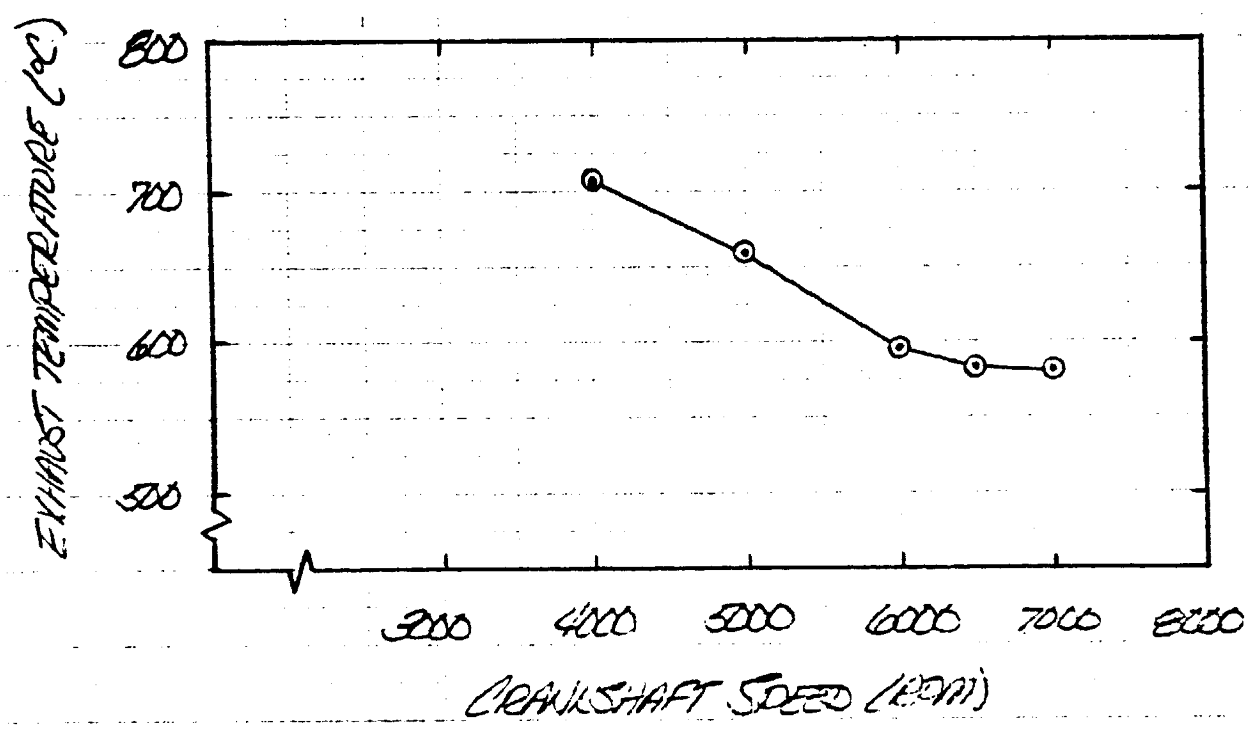


Figure 6.20. Exhaust Temperature versus Speed

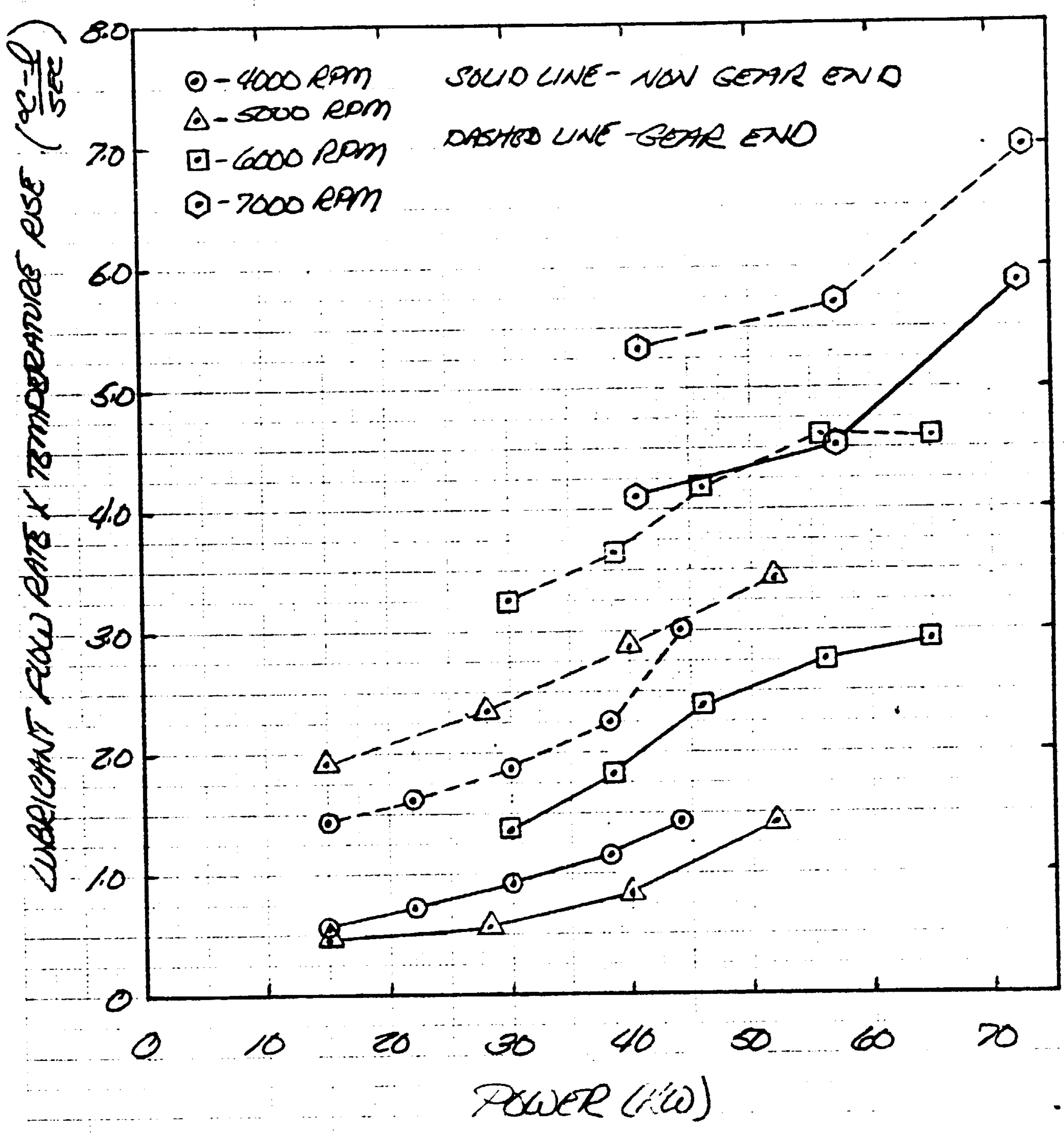


Figure 6.21. Product of Lubricant Flow Rate and Temperature Rise as a Function of Power

In summary, it would appear from these data that the lubricant outlet temperatures as measured here can be indicative of several engine operating parameters. Which parameter depends on which of three speed ranges the engine is operating in. These ranges are approximately defined as:

0 → 5000 rpm -	Both gear end and nongear end temperatures dominated by rotor temperature.
5000 → 8000 rpm -	Gear end lubricant temperatures dominated by rotor temperature. Nongear end lubricant temperature affected significantly by bearing friction.
8000 + rpm	(Speculated) both temperatures mostly governed by bearing friction.

It should be noted that in all cases for both ends, rotor heat transfer does affect the temperatures to some extent. Therefore it would seem that the measured outlet temperatures probably represent some upper limit on actual bearing outlet temperatures. Also due to the mixing of several different bearing flows, it is possible that some high (or low) temperatures are being masked. Based on the presented data and conclusions, however, it would appear that the highest lubricant temperature rise attributable mainly to bearings is approximately 40-45°C, occurring at 8000 rpm and 70 kW (93 hp).

Special Instrumentation

In order to assist in the operation of the rig engine, a special monitoring system was set up. It consisted of a 4-channel Philips scope with the following signals from the engine:

1. Top Dead Center
2. Pilot Fuel Flow
3. Main Fuel Flow
4. Combustion Pressure

1. A magnetic pickup triggered by a stud projecting from the flywheel is used to generate the Top Dead Center signal.

2&3. Proximity pickups are used to record the lift of the fuel injector needle valves. The signal identifies when each of the valves opens (and closes) with respect to the Top Dead Center position.

4. The peak pressure inside the combustion chamber is measured by an AVL water-cooled pressure pickup installed in the rotor housing at the "Before Top Dead Center" location.

In addition, two accelerometers were mounted on the accessory gearbox. The primary reason for them was to have a continuous recording of a dynamic signal from the engine as a means of identifying sudden changes in response or sequencing events in the case of an emergency. The response of the accelerometers was monitored throughout the test program. There was no evidence of any sudden or odd response from these pickups.

TEST FACILITIES

Engine testing was conducted at the Rotary Engine Division facility test cell number 20-6. Test cell 20-6 contains the driveline consisting of the engine mount, gearbox, and electric dynamometer; the lube oil, fuel, and coolant engine auxiliary systems; the inlet air system, exhaust system, and all the associated instrumentation. A control room is adjacent to the cell and contains the engine and test-stand operating controls, instruments, and instrumentation.

Drive Train - Drawing LS-33611

The test engine is connected to the dynamometer through a speed-reducer gearbox so that the engine can be operated at speeds higher than the dynamometer limit of 6000 rpm. The dynamometer is a cradle-mounted 200-hp direct current electric motoring dynamometer. Engine torque is measured via a calibrated load cell/torque arm and is read out on a digital indicator.

Fuel System - Drawing SK-12835

The test-stand fuel system supplies fuel to the pilot and main fuel injection pumps located on the engine gearbox. The fuel is pumped from an outside 500-gallon fuel carboy into the test cell to the fuel flow measuring system and then through individual lines and final filters to the two fuel injection pumps. The flow measuring system utilizes Flo-tron linear mass flow-meters to measure the total fuel flow and the pilot fuel flow. Fuel flow is read out on digital indicators located in the control room.

Auxiliary Lube Oil System - Drawing SK-12854

The auxiliary lube oil system in the test cell is designed to supply the "dry sump" test engine with properly conditioned oil. In addition to the necessary pumps, heat exchangers, filter, regulators and controls, there is a load cell mounted weighing tank so that the total oil flow and consumption can be determined. Oil is supplied through separate lines to the engine, turbocharger, and the oil metering pump. A turbine flow meter is installed in the main engine supply line for engine oil flow measurement. An oil return tank is located beneath the engine to collect the oil from all of the oil outlet ports. Each oil return line is fitted with a thermocouple and a chip detector.

Coolant System

The test-stand engine coolant system supplies coolant to the engine at a controlled rate of flow and temperature. The system includes a pump, reservoir tank, cooler, pre-heater, flow and temperature controls, and all the necessary piping, gauges, valves, etc. The system was filled with a 50/50 water glycol mixture for the test.

Inlet Air System

Inlet air for the engine is drawn into a large air box through a calibrated orifice which is selected from several available sizes for the correct flow range. This air box is located in an area adjacent to the test cell. The air flows through a stainless steel pipe into the test cell to a large air bottle located just above and aft of the engine. The air then passes via hoses through a 10-micron filter to the compressor inlet. An adapter located in the line, just prior to the compressor inlet, is fited with pressure and temperature instrumentation.

The air flow is read out with an inclinometer which is calibrated to match the orifice.

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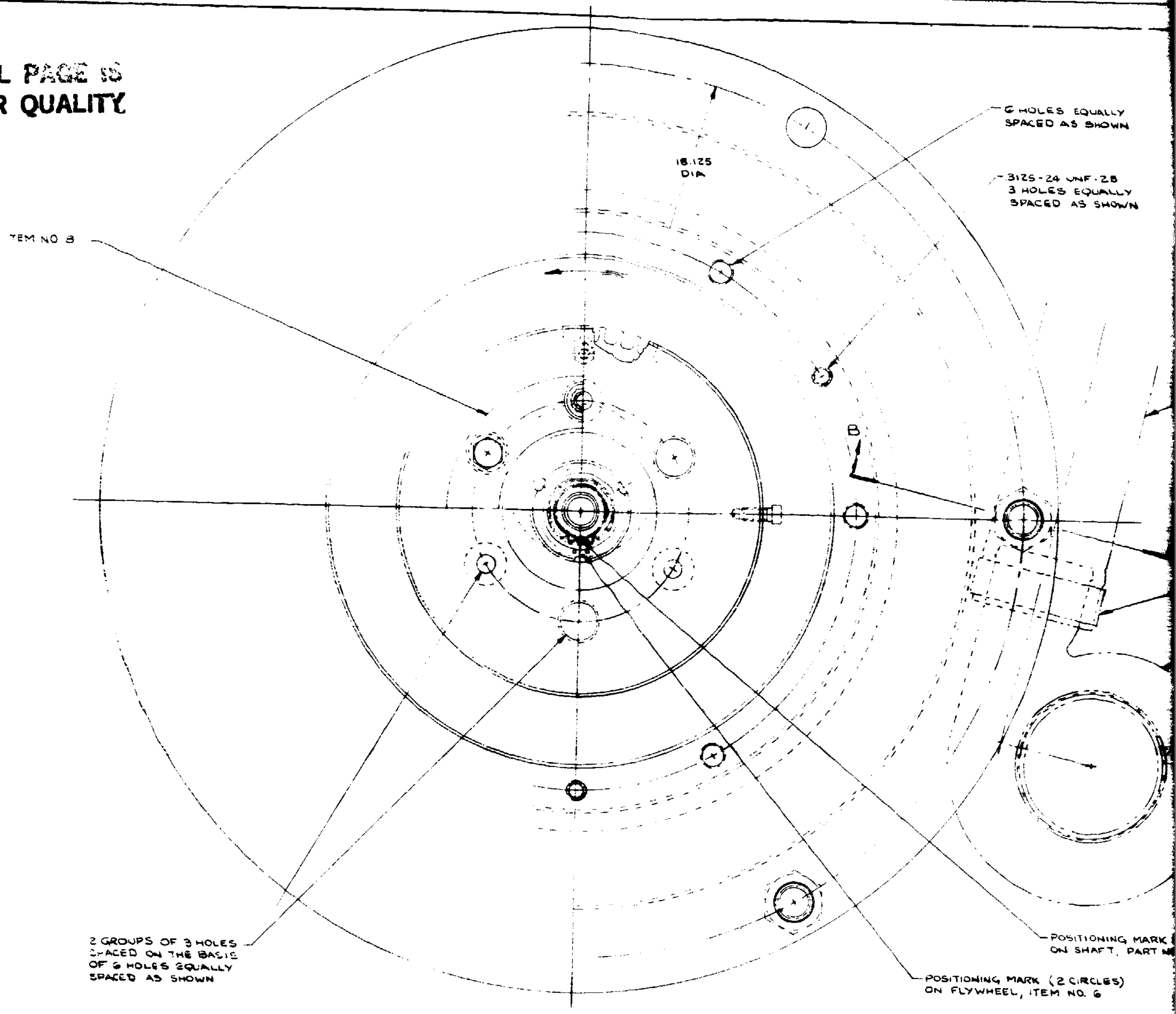

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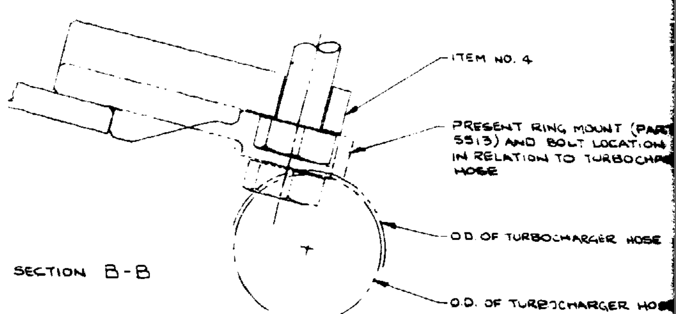
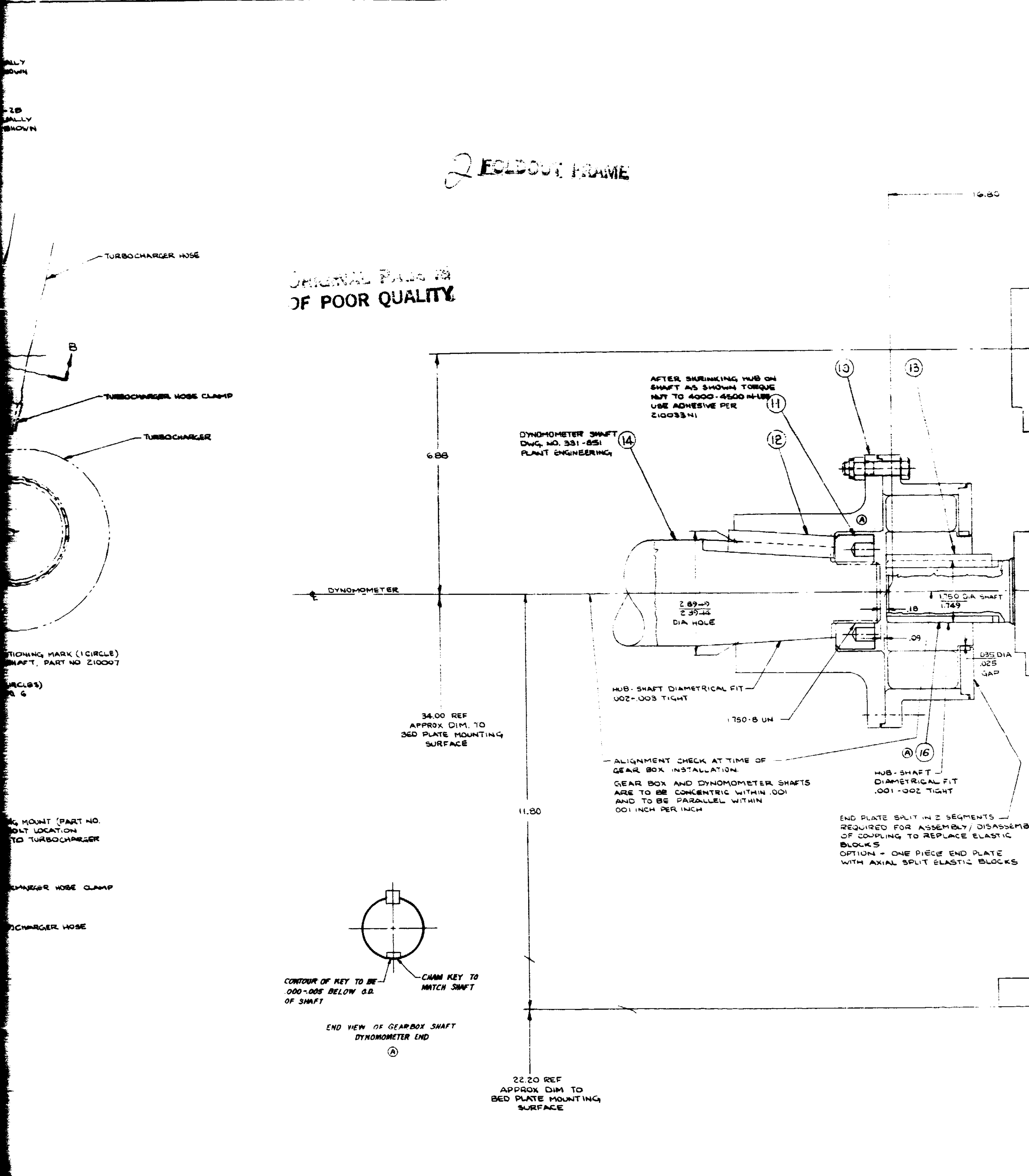
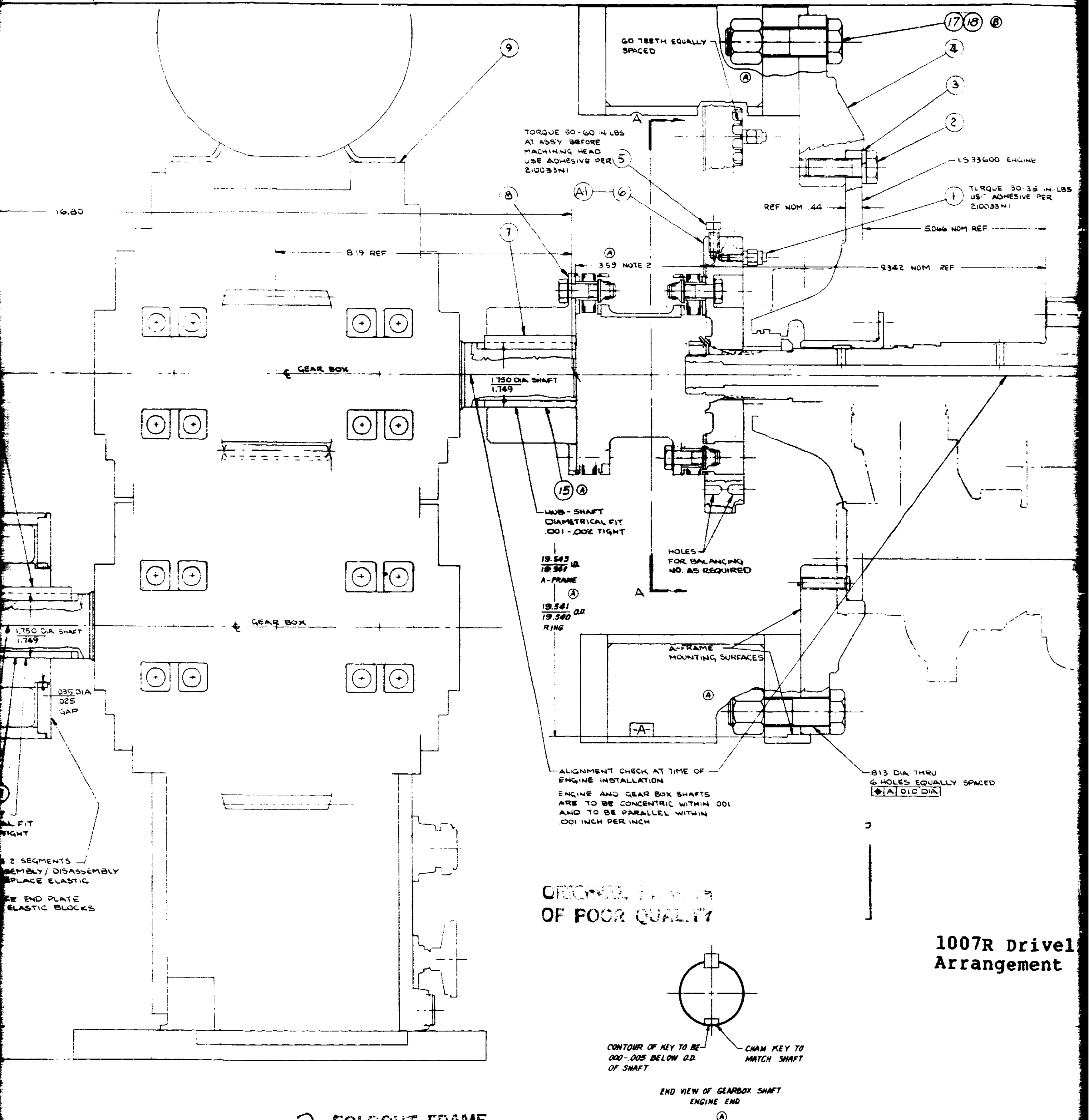


SECTION B-B

FOLDOUT FRAME


ALLY
SHOWN
-28
ALLY
SHOWN
16.80
TURBOCHARGER HOSE
B
TURBOCHARGER HOSE CLAMP
TURBOCHARGER
AFTER SHRINKING HUB ON SHAFT AS SHOWN TORQUE NUT TO 4000-4500 IN-LBS USE ADHESIVE PER 210033N1
DYNOMOMETER SHAFT DWG. NO. 331-851 PLANT ENGINEERING
6.88
DYNOMOMETER
2.89-9
2.39-14
DIA HOLE
1.750 DIA SHAFT
1.749
.18
.09
.035 DIA
.025
GAP
TIONING MARK (1 CIRCLE)
SHAFT, PART NO 210007
RCLES)
6
HUB-SHAFT DIAMETRICAL FIT .002-.003 TIGHT
1.750-8 UN
34.00 REF
APPROX DIM. TO
BED PLATE MOUNTING
SURFACE
ALIGNMENT CHECK AT TIME OF GEAR BOX INSTALLATION.
GEAR BOX AND DYNOMOMETER SHAFTS ARE TO BE CONCENTRIC WITHIN .001 AND TO BE PARALLEL WITHIN .001 INCH PER INCH
HUB-SHAFT DIAMETRICAL FIT .001-.002 TIGHT
END PLATE SPLIT IN 2 SEGMENTS REQUIRED FOR ASSEMBLY/DISASSEMB OF COUPLING TO REPLACE ELASTIC BLOCKS
OPTION - ONE PIECE END PLATE WITH AXIAL SPLIT ELASTIC BLOCKS
11.80
G MOUNT (PART NO.
OLT LOCATION
TO TURBOCHARGER
CHARGER HOSE CLAMP
CHARGER HOSE
CONTOUR OF KEY TO BE .000-.005 BELOW O.D. OF SHAFT
CHAM KEY TO MATCH SHAFT
END VIEW OF GEARBOX SHAFT
DYNOMOMETER END
A
22.20 REF
APPROX DIM TO
BED PLATE MOUNTING
SURFACE

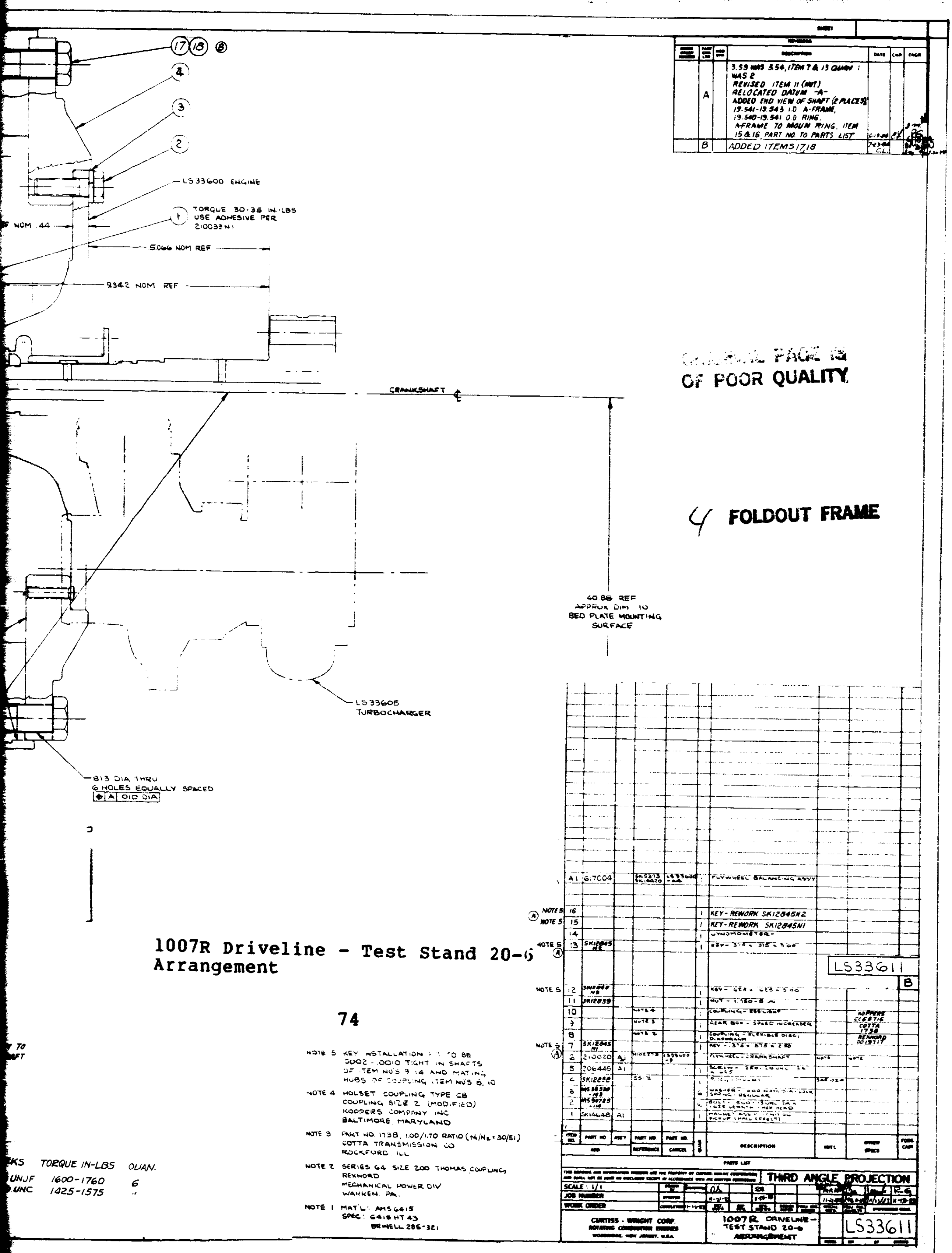


1007R Drivel
Arrangement

3 FOLDOUT FRAME

Ⓑ

	ITEM	BOLT	NUT	REMARKS	TORQUE IN-LBS	QUAN.
	17, 18	MS 9497-24	MS21045-C12	.750-16 UNJF	1600-1760	6
ALTERNATE	" "	MS 90725-190	MS51922-57 (EQIV. ESNA P/N 10094)	.750-10 UNC	1425-1575	"



1007R Driveline - Test Stand 20-6 Arrangement

INFORMATION FOR INITIAL DWG. RELEASE

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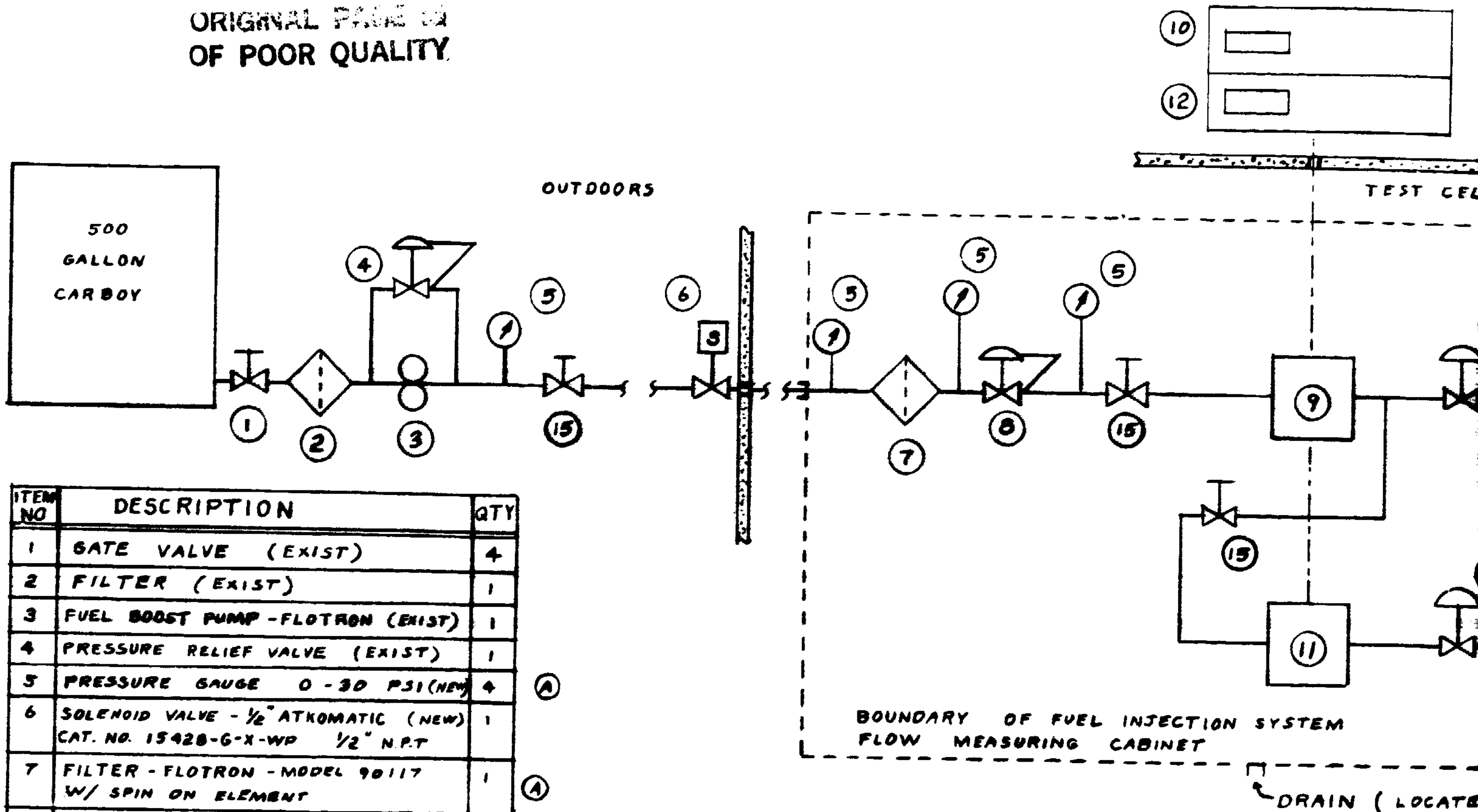

ITEM NO	DESCRIPTION	QTY	
1	GATE VALVE (EXIST)	4	
2	FILTER (EXIST)	1	
3	FUEL BOOST PUMP - FLOTRON (EXIST)	1	
4	PRESSURE RELIEF VALVE (EXIST)	1	
5	PRESSURE GAUGE 0 - 30 PSI (NEW)	4	A
6	SOLENOID VALVE - ½" ATKOMATIC (NEW) CAT. NO. 15428-G-X-WP ½" N.P.T	1	
7	FILTER - FLOTRON - MODEL 90117 W/ SPIN ON ELEMENT	1	A
8	PRESSURE REGULATING VALVE SCREWED END 1.5 GPM (NEW)	1	A
9	FLOWMETER - FLOTRON MODEL 10E48 S/N 27-87, 0 TO 150 PPH (EXIST)	1	
10	FLOW INDICATOR - FLOTRON 28-12 (EXIST) MATCH TO ITEM 9	1	
11	FLOWMETER - FLOTRON MODEL 10E61 S/N 26-55, 0 - 25 PPH (EXIST)	1	
12	FLOW INDICATOR - FLOTRON 28C01-5 (EXIST) MATCH TO ITEM 11	1	
13	DAY TANK MODEL 45-1 RECIRCULAT'G TANK ZINC PLATED - FLOTRON (NEW)	2	
14	PRESSURE REDUCING VALVE FLO-TRON INC. ½" NPT (NEW)	2	
15	GATE VALVE ½" NPT (NEW)	3	

MODEL

ITEM R

LS

FOLDOUT FRAME

ENGRG ORDER NUMBER	PART CHG. LTR.	MOD SYM.	DATE

PREVIOUS REVISIONS

FLATNESS — STRAIGHTNESS
ANGULARITY
PERPENDICULARITY
PARALLELISM
CONCENTRICITY
TRUE POSITION
ROUNDNESS
SYMMETRY
M MAX MTL CONDITION
S REGARDLESS OF FEATURE SIZE

ANSI B46.1
ANSI Y14.5

SPECS.

MATERIAL

UNLESS OTHERWISE SPECIFIED

TOLERANCES

FRACTIONS ±
DECIMALS ±
ANGLES ±
FORGINGS ±
CASTINGS ±
SHEET METAL ±
WELD SIZE AND LOC ±

ALL SURFACES ✓

FORGING OR CASTING NUMBER

DTR. PRS
CKR.
QUAL.
MET.
PE / MFG
ILS / CM
SPEC. ENGR
ENGR.
ENGR'G.
ENGR'G.

SHEET

REVISIONS

ENGRG ORDER NUMBER	PART CHG LTR	MOD SYM	DESCRIPTION	DATE	CKR	ENGR
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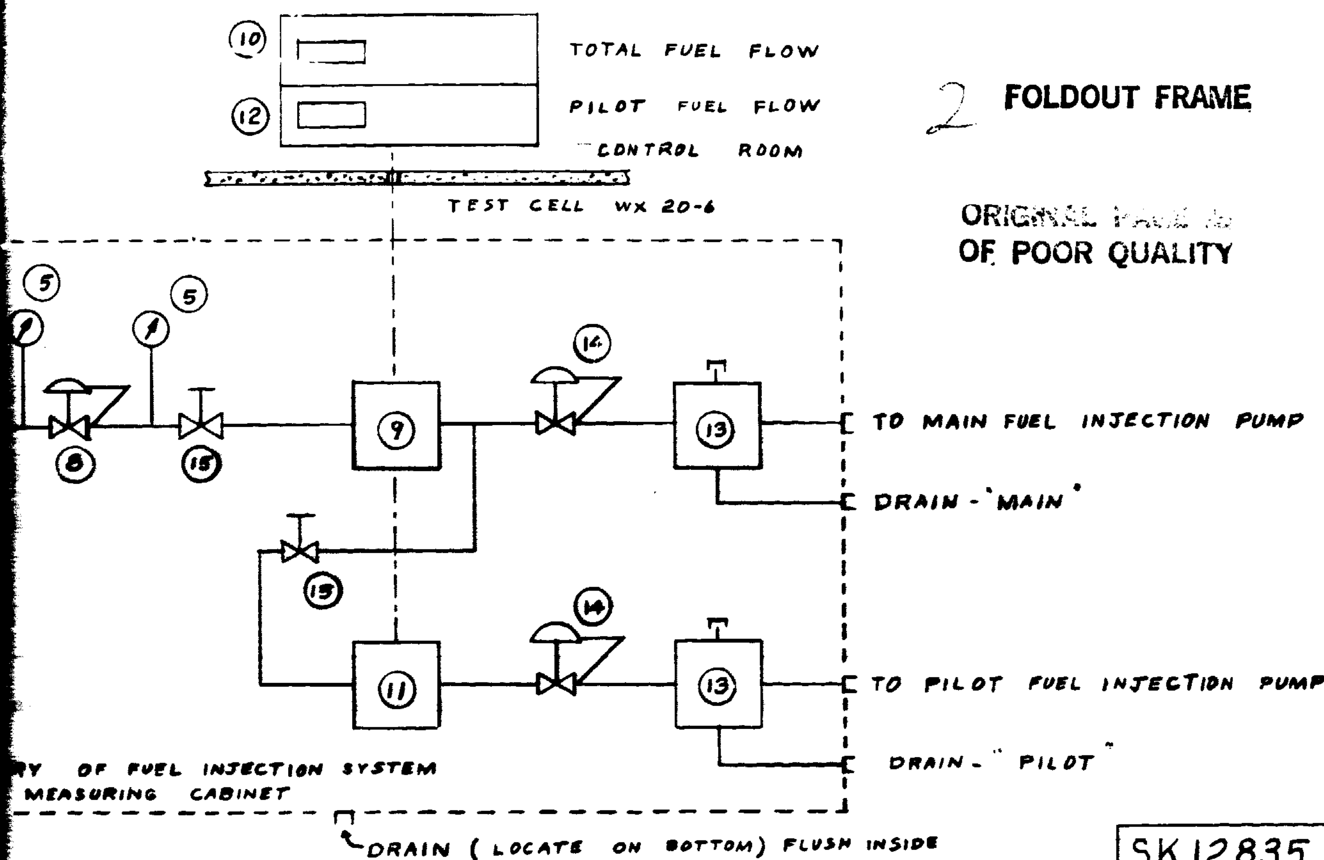

SK 12835

SHEET A

ALL EQUIP. TO BE 110 V. AC
CABINET TO BE 36"H × 48"W × 12"D, 2 DOORS, VENTILATED NEMA 12, PEDESTAL MOUNTED (FREE STANDING)
ALL BULKHEAD FITTINGS MARKED [TO BE 1/2" NPT
ALL PIPING TO BE 1/2" ϕ
THIRD ANGLE PROJECTION
ALL NOTES APPLY UNLESS OTHERWISE SPECIFIED

UNLESS OTHERWISE SPECIFIED			
TOLERANCES	DTR.	PRS	2-10-84
FRACTIONS ±	CKR.		
DECIMALS ±	QUAL.		
ANGLES ±	MET.		
FORGINGS ±	PE / MFG		
CASTINGS ±	ILS / CM		
SHEET METAL ±	SPEC. ENGR		
WELD SIZE AND LOC ±	ENGR.	J.M.S.	2-10-84
ALL SURFACES ✓	ENGR'G.	[illegible]	4/10/84
FORGING OR CASTING NUMBER	ENGR'G.		

...IGHT CORPORATION ...ITTEN PERMISSION.

JOHN DEERE TECHNOLOGIES INT'L INC.
ROTARY ENGINE DIVISION
WOOD-RIDGE, NEW JERSEY, U.S.A.

Fuel Supply System - WX20-6 1007R
Plumbing Schematic

75

CODE IDENT. NO.	SIZE	
66640	C	SK 12835
SCALE	UNIT WT.	SHEET

INFORMATION FOR INITIAL DWG. RELEASE

SK12854

SHEET

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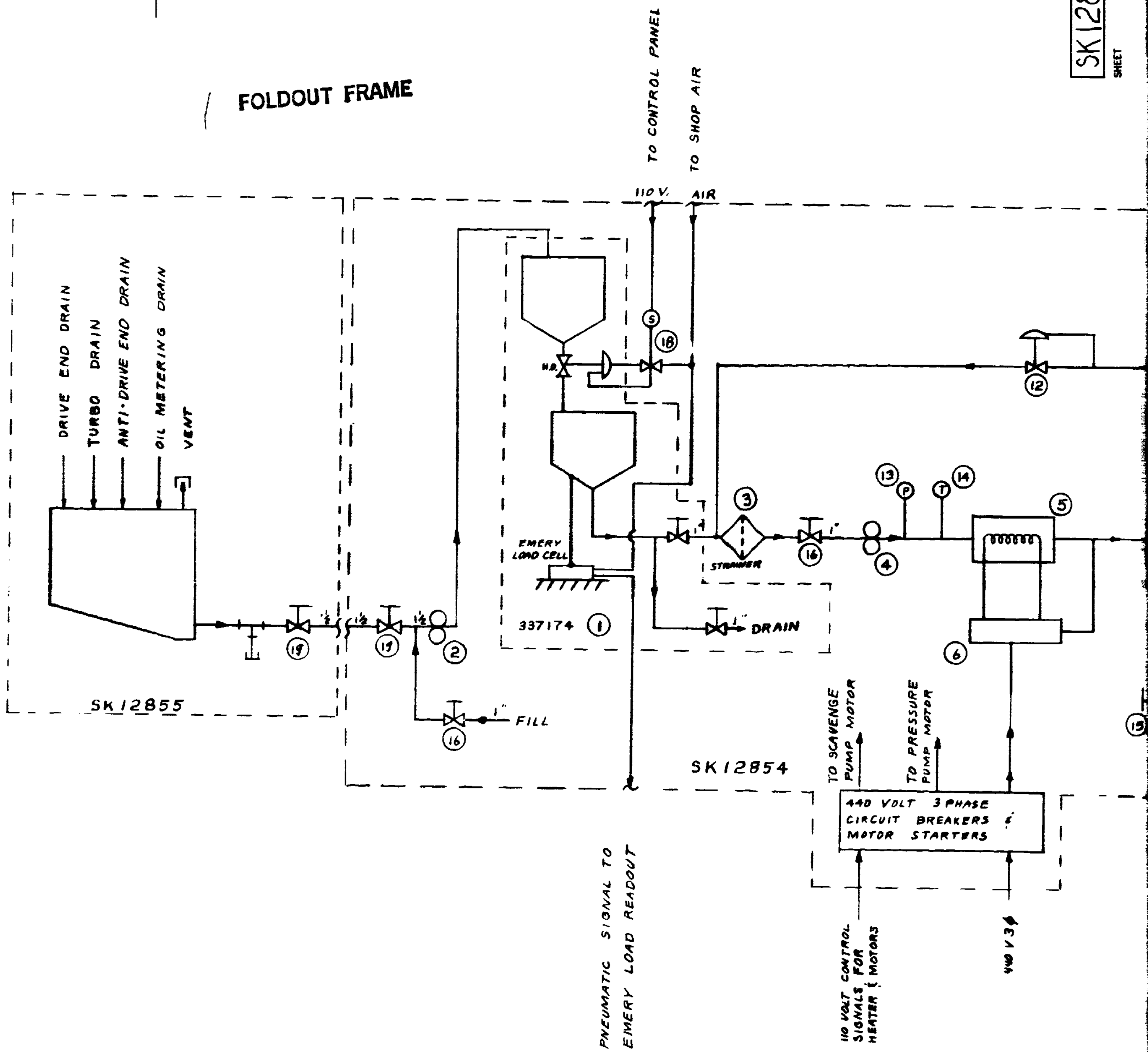

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PREVIOUS REVISIONS

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- PERPENDICULARITY
- PARALLELISM
- CONCENTRICITY
- TRUE POSITION
- ROUNDNESS
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- MAX MTL COND...
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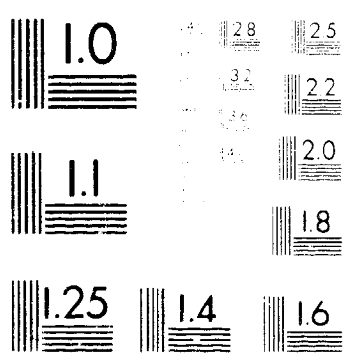

SK12854

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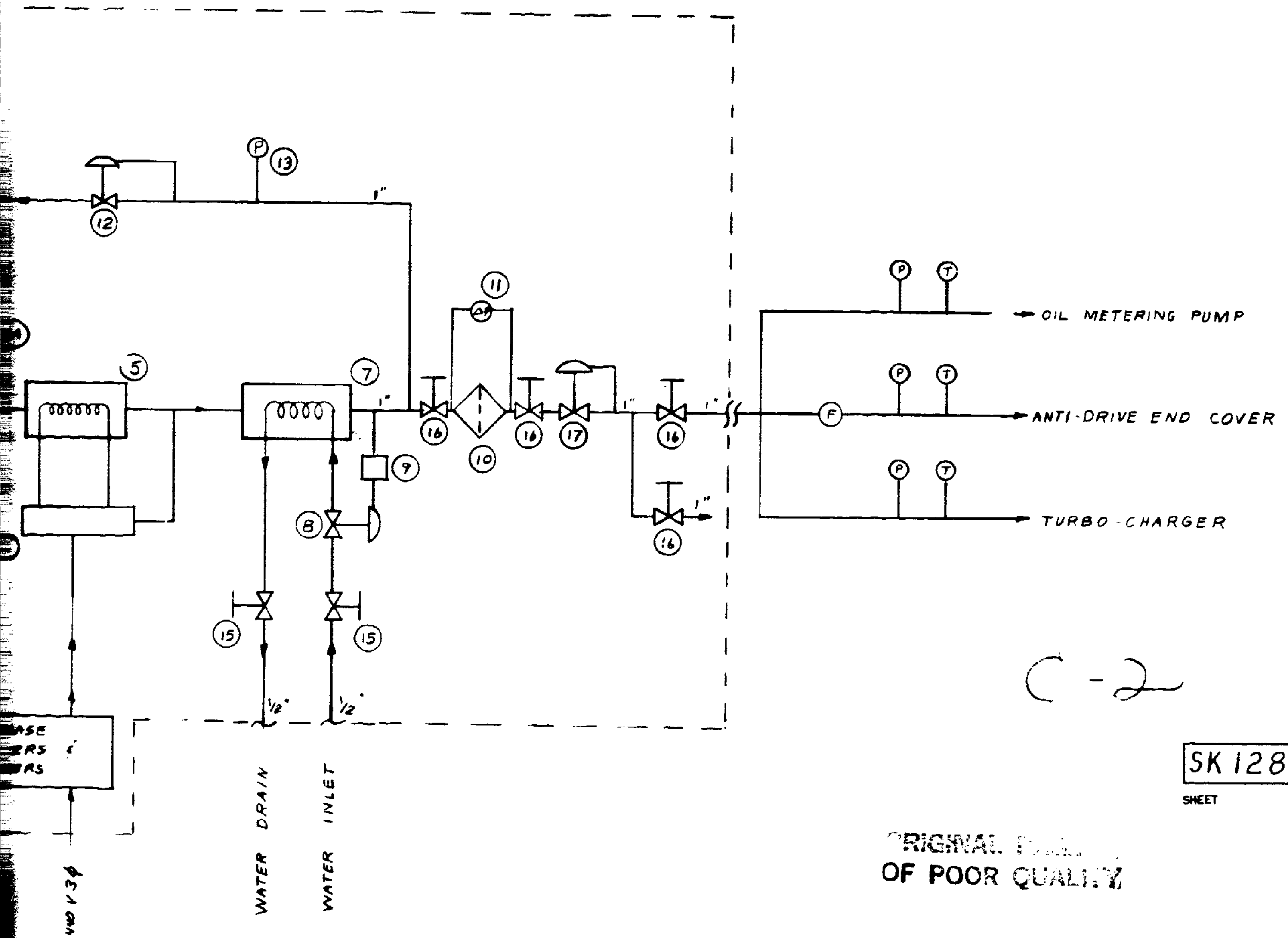

C-2

SK 12854

SHEET

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USE OF TEFLON TAPE IS PROHIBITED

ALL FLUID CONNECTIONS TO BE N.P.T.

ALL NOTES APPLY UNLESS OTHERWISE SPECIFIED

- FLATNESS AND STRAIGHTNESS
- ANGULARITY
- PERPENDICULARITY
- PARALLELISM
- CONCENTRICITY
- TRUE POSITION
- ROUNDNESS
- SYMMETRY
- M MAX MTL CONDITION
- S REGARDLESS OF FEATURE SIZE

SPECS

MATERIAL

UNLESS OTHERWISE SPECIFIED

TOLERANCES

- FRACTIONS ±
- DECIMALS ±
- ANGLES ±
- FORGINGS ±
- CASTINGS ±
- SHEET METAL ±
- WELD SIZE AND LOC ±

ALL SURFACES √ = √

FORGING OR CASTING NUMBER

DTR.	P. R. B.	4-23-84
CKR.		
SUPV.		
MET.		
PE / MFG		
ENGR.	E.W.	4-26-84
ENGR.		
ENGR.		4-23-84

JDTII ROTARY ENGINE DIVISION

WOOD-RIDGE, NEW JERSEY, U.S.A.

Lube Oil Supply System WX20-6

76

CODE IDENT. NO.	SIZE	
66640	D	SK 12854
SCALE	UNIT WT.	SHEET

CONCLUSIONS

1. The 1007R rig engine acceptance testing data and the engine operating characteristics indicate that the engine is a suitable vehicle for evaluation of the technology enablement features required for the "advanced" and "highly advanced" General Aviation aircraft engines.

2. The vibratory characteristics of the engine as exhibited during run-in and acceptance testing are satisfactory as anticipated.

3. Data obtained indicate the potential for significant improvements in combustion characteristics.

4. Effects of different turbine matches on mixture strengths and resultant effects on combustion efficiency must be evaluated.

5. Power required to drive the belt-driven injection pumps and oil metering pump must be determined.

6. Without the peak pressure restriction applied to the first two rotors because of thin areas in castings, the 120 kW (160 hp) at 8000 rpm goal is achievable. Tooling has been corrected so that all subsequent rotors currently on order will have no performance restrictions.

RECOMMENDATION

Continue technology enablement efforts toward the highly advanced stratified charge rotary aircraft engine technolgies as defined in NAS Contracts No. NAS3-21285 and NAS3-22140 including:

- Advanced Fuel Injection System
- Baseline Performance and Durability Testing
- Instrumented Engine Preparation and Testing
    - Dynamic Pressures and Heat Release
    - Heat Flux Determination
    - Friction and Heat Rejection
- Porting Modifications
- Compression Ratios

APPENDIX A

To: For the Record

Date: November 21, 1983

Place: Engineering

Subject: RC1-40-WX20-6
Driveline Vibration Analysis

Reference:

INTRODUCTION

The RC1-40 engine is to be tested on test stand WX20-6. The arrangement will consist of the engine driving a General Electric Dynamometer frame TCC-2464H through a Cotta gearbox, 1.70 to 1 speed reducer. This report discusses the results of the vibration analyses conducted on the driveline.

CONCLUSIONS

1. The driveline will have no major torsional resonances within the operating range of the engine.

   (a) Use of the Koppers flexible coupling between the gearbox and the dynamometer will result in a first-mode torsional resonance in the 1000 to 1200 rpm range. The damping of the rubber coupling will be sufficient to control the torsional response of the system during starting and stopping of the engine. Engine idle will be at 2000 rpm, far enough above the resonant rpm to eliminate any possible torsional response.

   (b) The Thomas coupling used between the engine and the gearbox is stiff enough to place the second-mode torsional resonance well above (24%) the maximum engine rpm.

2. The critical speed (shaft lateral vibration) of the high-speed driveline is well above the maximum engine rpm. The special radial stops incorporated in the Koppers flexible coupling will control any potential low-speed shaft motion.

DISCUSSION OF RESULTS

TORSIONAL ANALYSIS

The major driveline components are shown on Figure 1 along with the associated torsional stiffness values and inertias. The results of a torsional vibration analysis of this system show the following:

Natural Frequencies

Mode	Natural Frequency - cpm
1	1,108
2	24,170
3	38,548
4	71,010
5	78,222

The first mode is the inertia of the dynamometer oscillating against the inertia of the rest of the system on the torsional spring of the coupling between the gearbox and dynamometer. Since this mode, which is considered the most significant, cannot be placed above the operating range, it was placed below engine idle.

Either the Koppers No. 2 type CB Holset coupling or the Vulkan No. 43S coupling have a torsional stiffness in the desired range. The Koppers was selected because different torsional flexibility could be achieved by simply interchanging the rubber elements, using a different durometer hardness.

Since the prime resonance for the second mode cannot be placed below idie, it was placed as high as possible. This was accomplished by making the coupling between the engine and gearbox as stiff as possible and reducing the inertia of the engine flywheel as much as possible. Adding a torque measuring device (LeBow) between the engine and the gearbox reduces the torsional stiffness and places a critical resonance at 8200 rpm, which is considered unacceptable. Use of a torque meter is not recommended.

The third mode is primarily influenced by the flexibility of the engine shaft between the flywheel and counterweight.

The vibratory torque excitation of the engine will be:

Excitation Order	Magnitude
1.0	2.10 x mean torque
2.0	0.90 x mean torque
3.0	0.40 x mean torque
4.0	0.20 x mean torque
5.0	0.09 x mean torque
6.0	0.04 x mean torque

Potential resonances would be at the following rpm's:

Mode	cpm	Excitation Order 1	2	3	4	5	6
1	1108	1108	554				
2	24170		12085	8057	6043	4834	4028
3	38548			12849	9637	7710	6425
4	71010						

The engine will probably idle at 2000 rpm. The design speed is 8000 rpm and the maximum overspeed is 9600 rpm.

The following table shows the relative torsional amplitude for the first poor modes of vibrations.

Component	Relative Torsional Amplitude Mode 1	2	3	4
Engine Cwt	+1.00	+1.00	+1.00	0.0
Rotor	+1.00	+0.95	+0.88	0.0
Engine Cwt	+1.00	+0.88	+0.68	0.0
Flywheel	+1.00	+0.40	-0.43	0.0
Input Gear	+1.00	-0.25	+0.07	0.0
Output Gear(x)	+0.62	-0.17	+0.06	-0.16
Dynamometer(x)	-0.04	0.0	0.0	+1.00

(x) Actual torsional amplitudes, taking into consideration the gear ratio.

The prime resonances to avoid in the operating range of the engine are those due to first and second engine order excitations. The speed ranges to avoid are from 0-1600 rpm (motor to above 1600 before firing).

The driveline is considered satisfactory since there are no prime resonances.

1. The first torsional mode is low enough to have its principle resonance (first-order excitation) at 1108 rpm, well below engine idle.

2. The second torsional mode is high enough to place the second-order resonance at an rpm 26% above maximum engine rpm.

Secondary Resonances

Since it is impossible to place all possible resonances out of the engine operating range, several secondary resonances will occur. Only the second and third mode will be discussed, since the first mode is low enough to have none and the fourth and higher modes are above any significant resonances:

The second mode will have the following secondary resonances:

Excitation Order	% Maximum Engine rpm
3	84
4	63
5	50
6	42

There should be sufficient damping in the system (particularly in the Koppers coupling) to control the limited amount of third-order excitation.

The third mode of torsional vibration will have the following secondary resonances:

Excitation Order	% Maximum Engine rpm
4	100
5	80
6	67

The very nature of the higher modes of vibration is to inherently have considerable system damping (although the Koppers coupling will have only a limited contribution). Because the magnitude of the torsional excitation is only 20% of the mean torque, there should not be any significant torsional response at resonance. Excitation orders above 4th order should result in little, if any, torsional response.

Translational Vibration

The drive line was divided into two separate shafts for the translational (shaft bending) vibration. The high speed shaft consists of the engine main shaft, supported on its four bearings, the Thomas coupling and the high-speed Cotta gearbox shaftings, supported by its bearings. The low speed shaft consists of the General Electric Dynamometer Rotor, supported on two bearings, the Koppers flexible coupling and the low speed gearbox shaft on its bearings.

High Speed Shaft

The high speed shaft was analyzed as a single shaft divided into 35 increments supported, on six bearings and two moment springs. The gearbox shaft portion was modeled as a weight (gear), supported on each side by rolling element bearings. Spring rates were estimated to be 2×10^6 pounds per inch.

The engine shafting was modeled as a distributed mass system, supported on four journal bearings. The main bearing spring rate was set at 5×10^6 pounds per inch, while the outer bearings were given values of only 1×10^6. The Thomas coupling, No. 64-200, connects the gearbox and engine shaft through "moment" springs. Since the moment springs are quite soft (48 in.-lb per degree), they were each modeled as a pair of radial springs to ground. Two radial springs of 1.6×10^4 pounds per inch, 0.83 inch apart, were used at each end of the center portion of the coupling.

The lowest critical speed for the shaft system is 29, 441 rpm, over three times the maximum engine rpm.

Low Speed Shaft

The low speed shaft, gearbox to dynamometer, was first modeled as two separate shafts. The output shaft of the gearbox was modeled as a shaft with a concentrated weight (driven gear), supported on two bearings. A portion of the Koppers coupling was attached to the end of the shaft. With the 36.9-pound gear and bearing spring rates of 2×10^6 pounds per inch, the lowest critical speed is 22,564 rpm, over 3.7 times the maximum shaft speed.

The General Electric Dynamometer shaft was modeled as a simple shaft supported on two bearings. The core weight was 475 pounds. Half the Koppers coupling (Holset type CB Number 2), 17.78 pounds, was placed on the end of the shaft. Using a bearing spring rate of 2×10^6 pounds per inch, the following critical speeds were calculated:

Mode	Critical Speed
1	6120
2	8128
3	12186

Since this calculation did not result in a satisfactory minimum critical speed valve, it was necessary to develop a more realistic shaft model. This was accomplished by setting up a two-shaft interconnected shaft system. In essence it took the two individual shafts described above and connected them together with a radial spring. The radial spring is the rubber elements in the Koppers coupling.

The computer runs were made, one with a spring of 7100 pounds per inch (an estimate of the rubber's flexibility) and the second with 1 x 10^7 pounds per inch simulating that there was metal-to-metal contact at the coupling.

	K, pound per inch	
Mode	7,100	1,000,000
1	5,974	7,682 rpm
	8,043	10,000

The standard Koppers coupling normally carries a radial stop that permits a 0.1-in. motion (compression of rubber) before metal-to-metal contact is made. For our driveline, operation at 6000 rpm would cause sufficient relative shaft motion to "bottom" the rubber. Once the two halves of the coupling touch (metal-to-metal contact), the critical speed is raised significantly and there would be no tendency for the shaft amplitudes to increase.

Koppers does make a modified coupling in which the radial motion is limited to a much smaller value. This is done by installing a teflon-bronze bushing between the two coupling halves. The bushing clearance will be set at 0.030 in. (nominal diametrical) for our coupling. When radial contact is made, the critical speed will be significantly raised such that no further shaft motion will occur.

M. Kulina

ENGINE NO.	RECORD OF ANALYSIS	WRITTEN BY MRK
JOB NO.	SHEET ____ OF SHEETS ____	APPROVED BY ____
SUBJECT: RC1-40 / WX 20-6 TORSIONAL ANALYSIS		DATE 11-21-83

RESULTS, SKETCHES & FORMULAS

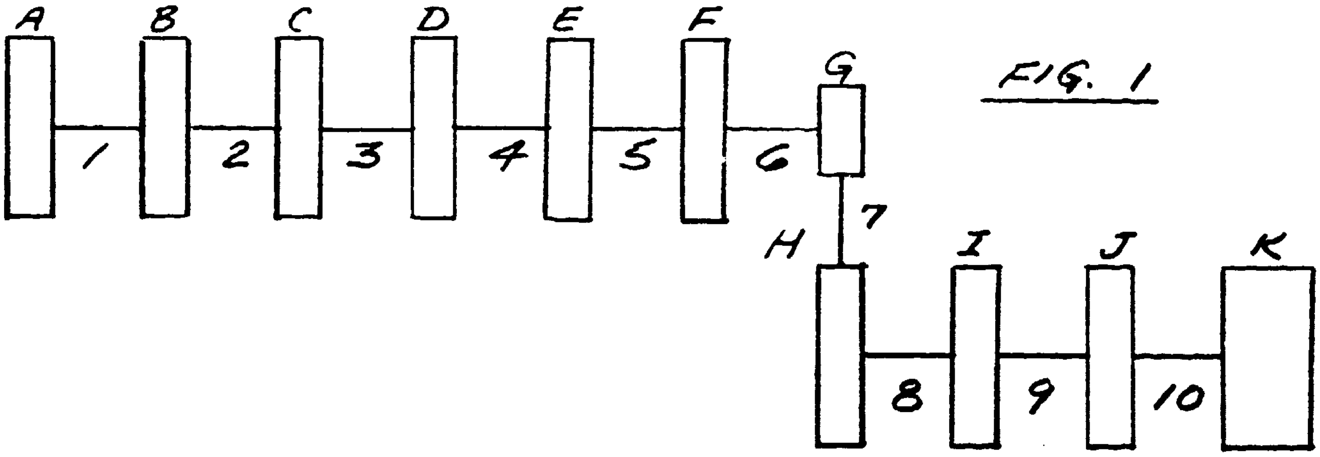

FIG. 1

J	IN-LB-SEC²	K	IN-LB/RAD	
A	ENG. COUNTERWEIGHT		0.027	
		1		3.52E6
B	ENG. ROTOR		0.015	
		2		3.40E6
C	ENG. COUNTERWEIGHT		0.027	
		3		8.52E5
D	ENG. FLYWHEEL		0.220	
		4		1.00E7
E	1/2 THOMAS COUPLING		0.010	
		5		3.93E6
F	1/2 THOMAS COUPLING		0.070	
		6		3.23E6
G	HIGH SPEED GEAR		0.073	
		7		2.62E7
H	LOW SPEED GEAR		1.252	
		8		2.80E6
I	1/2 KOPPERS COUPLING		0.041	
		9		3.36E4
J	1/2 KOPPERS COUPLING		0.440	
		10		2.39E7
K	DYNAMOMETER		33.600	

ACTUAL VALUES

REFERENCES:

APPENDIX B

TEST ENGINE ASSEMBLY

Assembly of the first 1007R rig engine was accomplished with no difficulty. Although a number of the basic operations and procedures developed for larger rotating combustion engines are common to this engine, its small size and low weight of components make it much simpler to handle. The engine power section was completely built and transferred from assembly stand to transport truck manually.

The LS-33612 Fuel Injection Pump and Oil Metering Pump Drive System requires a small hoist to lift, but this unit is a workhorse system and not part of the basic engine.

The steps followed in assembling the first rig engine are summarized in the Assembly Instructions included in this section. The special tooling required for assembly is separately listed. It is minimal and no complex devices are involved.

1007R Engine Assembly Instructions

Reference: 617000 Engine Basic and Installation Drawing

LS-33612 Fuel Injection Pump and Oil Metering Pump Drive System

Bill of Material Model 1007R

1. Read and comply with all Development Engine Instructions (DEI) issued for the current build.

2. Measure parts and ensure that they comply with limits shown on measurement sheets or special fits as called for by DEI. On new parts ensure that critical dimensions have been recorded with serial number in accordance with drawing requirements.

3. Thoroughly clean all parts before each step in final assembly sequence. Particular attention must be given to fluid passages, blind cavities, and any area where chips or other foreign material may be lodged.

4. Assemble and balance the 617001N rotor assembly as follows:

   (a) Install 617002 rotor gear assembly in rotor.

   (b) Install 210029 balance plugs dry without adhesive.

   (c) Rotor bearing 210008N is not to be installed at this time.

4. (d) Balance the rotor assembly in accordance with instructions on drawing 617001N using an arbor sized to fit the bore of the rotor to preclude possibility of damage to finished bearing surface.

   (e) Assemble pressure checking fixture ST-1072 to rotor assembly and pressure check with air to 20 psi. No leakage shall occur around the six balance plugs.

   (f) Heat the rotor to 200°F and cool the 210008N bearing in dry ice and alcohol. Press bearing into rotor from anti-gear side using tool ST-1068-1 and -2. Be sure that optional weld in bearing is located as required by drawing 617001N.

   (g) Have bearing bore machined to finished size and thoroughly clean rotor assembly being sure it is completely free of chips, particularly the gear side cavity.

   (h) Weigh rotor assembly in accordance with instructions on drawing 617001N. Determine and record axial location of rotor assembly center of gravity.

5. Assemble crankshaft complete balancing assembly and have it balanced in accordance with drawing 617003. When balance is complete, disassemble components and hold for engine build.

6. Assemble the anti-drive end housing, gear and bearing assembly 617012N as follows:

   (a) Measure detail bearing 210009N, gear 210017 and bore of housing 210003, machined per 617013N2, and record on measurement sheets.

   (b) Press dowel pin MS9390-420 into housing.

   (c) Heat gear to 200°F and cool bearing in dry ice and alcohol. Press bearing in gear from the flanged side using tool ST-1069-1 and -2. Be careful to locate tapered end of bearing and optional weld as shown on stationary gear and bearing assembly drawing 617007N.

   (d) Heat the anti-drive end housing to 250°F and cool the gear and bearing assembly in dry ice and alcohol. Press gear and bearing assembly 617007N into housing 210003 using tool ST-1069-2, -3 and three -4 guide pins.

   (e) Install baffle plate assembly 210022N1, bolts, plugs and helical inserts to complete the 617012N assembly.

7. Assemble the drive end housing, support and bearing assembly 617011N as follows:

7. (a) Measure detail bearing 210012N, support 210021 and bore of housing 210000, machined per 617013N1, and record on measure sheets.

(b) Heat support to 250°F and cool tapered half of thrust bearing in dry ice and alcohol. Press tapered half of thrust bearing in flanged end of support using tool ST-1071-1 and -2. Be careful that optional weld in bearing is located as shown on support and thrust bearing assembly drawing 617006N.

(c) Heat support with half bearing installed to 250°F, and cool remaining half bearing in dry ice and alcohol. Press half bearing in support using tool ST-1071-2, -3, and -5. Be sure that optional weld in bearing is located per drawing 617006N.

(d) Heat the drive end housing to 250°F, and cool the support and thrust bearing assembly in dry ice and alcohol. Press the support and thrust bearing assembly 617006N into housing 210000 using tool ST-1071-2 and -4 and three ST-1069-4 guide pins.

(e) Install baffle plate assembly 210022N2, bolts, dowel and helical inserts to complete the 617011N assembly.

8. Assemble the drive end cover and bearing assembly 617009N as follows:

(a) Measure detail bearing 210013N and bore of cover 210004, machined per 617013N, and record on measurements sheet.

(b) Heat cover to 250°F and cool the bearing in dry ice and alcohol. Press the bearing into cover using tools ST-1070-1 and -2 and two ST-1069-4 guide pins. Be sure that bearing is oriented in cover to locate weld in shell as shown on drawing 617009N.

(c) Install bolts, dowel, helical inserts and vibration transducer support 617016 to complete the 617009N assembly.

9. Assemble the anti-drive end cover and bearing assembly 617010N as follows:

(a) Measure detail bearing 210010N and bore of cover 210005, machined per 617013N2, and record on measurement sheets.

(b) Heat the cover to 250°F and cool the bearing in dry ice and alcohol. Press the bearing into cover using two ST-1069-4 guide pins only. Be sure that bearing is oriented in cover to locate weld in shell as shown in drawing 617010N.

9. (c) Install bolts, helical inserts, and vibration transducer bracket 617017 to complete the 617009N assembly.

10. Bolt adapter plate ST-1063 to turnover-type engine assembly stand and secure in horizontal position. Place 617012N anti-drive end housing gear and bearing assembly on adapter plate, cover side down, and position to align bolt holes. Place the flanged half of ST-1067 crankshaft immobilizing tool against underside of adapter plate and align bolt holes. Bolt anti-drive housing and immobilizer to adapter plate.

11. Place "O" ring seals in grooves on anti-drive side of rotor housing assembly 617008 using silicone grease to retain them. Lubricate rotor housing dowels and dowel holes in anti-drive housing with engine oil. Press rotor housing onto anti-drive housing using three clamps. Use protective material between clamp pads and engine parts. Advance all clamps in equal increments to avoid cocking rotor housing. Be certain "O" ring seals are still in grooves in rotor housing when they contact end housing. Tighten clamps until housings are in metal-to-metal contact and any excess lubricant has been expelled. Remove clamps.

12. Lubricate the face of the anti-drive housing, the stationary gear bearing bore, and mating crankshaft journal with engine oil. Carefully lower crankshaft into place, allowing it to rest on end of gear.

13. Lubricate crankshaft eccentric and bore of rotor bearing with engine oil. Rotate eccentric to approximate TDC position. Carefully lower rotor assembly, less all sealing components, over eccentric, rocking crankshaft back and forth as required to engage rotor and stationary gears. Carefully rotate crankshaft by hand three revolutions to be certain rotor does not contact trochoid.

14. Check stationary gear positioning, rotor gear backlash, and rotor-to-rotor housing clearance as follows:

    (a) Position magnetic block on rotor and measure backlash between rotor and stationary gear as shown on housing measurement sheet #17.

    (b) Lower crankshaft stabilizing tool ST-1057 over crankshaft and seat on rotor housing.

    (c) Measure and record minimum clearance between flanks of rotor and trochoid surface at locations shown on assembly measurement sheet #3.

14. (d) Install master apex seals ST-1075 in rotor slots and measure and record clearance between apex seals and trochoid surface at locations shown on assembly measurement sheet #4. Remove master apex seals.

(e) Remove crankshaft stabilizing tool ST-1057 and rotor.

15. Assemble all gas and oil sealing components in the anti-drive or gear side of the rotor including master apex seals. Apply silicone grease to components to hold them in place and retain apex seals with thread or an elastic. Be sure that all seals and springs are properly installed and engaged so that anti-rotation features will function.

16. Rotate assembly stand to place crankshaft in horizontal position with top of engine up. Position crankshaft eccentric toward the floor. Lubricate contact surfaces of end housing and crankshaft journals with engine oil.

17. Slide rotor assembly over crankshaft until rotor seals contact end housing, being careful that one apex is toward the floor and the other two are equidistant above the housing minor axis.

18. Rotate assembly stand to place shaft back in vertical position. Remove thread or elastic band holding master apex seals.

19. Carefully rotate crankshaft by hand several revolutions to ensure there is no interference between parts.

20. With master apex seals still in place, install all other gas seals and oil seal elements in drive side of rotor.

21. Remove master apex seals one at a time, and slide standard long apex seal and spring into position. Install short, small, triangular sections of apex seals, making sure they are properly positioned with respect to springs.

22. Position "O" ring seals in rotor housing grooves using silicone grease to hold them in place.

23. At this point in the assembly sequence, long rotor studs 210026 <u>must be</u> loosely installed at locations number 7, 17 and 18 as shown on assembly measurement sheet #3 or rotor housing drawing 210001. Studs cannot be installed at these locations after both end housings are in place on rotor housing.

24. Lubricate bearing surfaces, contact surface of drive end housing, alignment dowel holes and dowels with engine oil. Lower drive end housing assembly over crankshaft until it starts on alignment dowels. Using three clamps with protective material between clamp pads and engine parts, press drive end

24. housing assembly onto dowels until it contacts rotor housing. Tighten clamps in equal increments to avoid cocking drive end housing. Ensure that all three housings are seated properly.

25. With clamps still in place install remaining long studs 210026, sealing washers 210004, flat washers 210054, and nuts 210068. Snug nuts on five studs spaced at approximately equal intervals around the engine, and ensure that crankshaft can be rotated. Remove clamps and torque all nuts to stretch studs in the sequence and to the limits specified on assembly measurement sheet #5. Note that there are two separate but coaxial short studs at location number 22. They must be stretched independently. Measure the free length across the two short studs. Torque nut on one stud until measurement increases by amount of stretch required for one stud. Then torque nut on the opposite stud until measurement further increases by the amount of stretch required for the second short stud.

26. Referring to drawing LS-33607 install balance weight spacer 210051, balance weight 210018, spring pin MS-16562-237, balance weight lockwasher 210048, and balance weight nut 210047.

27. Install remaining part of crankshaft immobilizer ST-1067. Snug balance weight nut to seat spacer, weight, and lockwasher. Torque balance weight bolt. Using spanner socket ST-1059 torque balance weight nut 210047 to required value and bend lockwasher.

28. Check and record crankshaft end float.

29. Install transducer substituting plugs and coolant pressure fitting in rotor housing and cap pressure fitting. Install cover plate with gaskets at rotor housing inlet and exhaust ports. Temporarily install fuel injection nozzles and perform combustion chamber air leak check and coolant system water leak check.

30. Place housing cover "O" ring seal MS-9388-168 in groove in drive end housing using silicone grease to retain it. Lubricate bearing surfaces with engine oil and install drive end cover and bearing assembly 617009N. Torque attaching bolts to required value.

31. Install front oil seal spacer on crankshaft, and be sure it is seated against shoulder on shaft. Recheck crankshaft end float and fix shaft in mid-position. This can be done by wedging or clamping immobilizer at opposite end of shaft. Measure distance between oil seal spacer and shoulder in cover. Drawing LS-33600 specifies a nominal value of 0.028 in.

32. Install oil seal "O" ring MS-9388-228, face-type oil seal 210055, and oil seal retaining ring MS-16631-4225 in drive end cover.

33. Install flywheel spacer 210052, flywheel assembly 617004, flywheel centering cone 210050, flywheel attaching nut lock and flywheel attaching nut 210046. Using spanner socket ST-1058, torque flywheel attaching nut to required value and bend lock. Install oil plug 210016 and its retaining ring MS-16625-4068 in drive end of crankshaft.

34. Remove crankshaft immobilizer and remaining bolts holding engine to adapter plate. Remove the engine from assembly stand, and bolt drive end cover mounting flange to SK-12838 mount ring assembly on transport truck. The engine is now in horizontal position.

35. Referring to drawing LS-33607, install balance weight 210013 with spring pin MS-16552-235 on anti-drive end of crankshaft. Torque balance weight bolt to required value.

36. Place housing cover "O" ring seal MS-9388-168 on lip of anti-drive end cover and bearing assembly 617010N, using silicone grease to retain it. Install cover assembly and torque attaching bolts to required values.

37. Using installation mandrel ST-1073 and thimble ST-1074, press lip-type oil seal into anti-drive end cover and install oil seal retaining ring MS-16625-4162.

38. Install pressure oil inlet fitting in anti-drive end cover.

39. Attach to drive end cover the ignition timing indicator 210109 and ignition pickup assembly SK-12869 and TDC sensor SK-12870 with their support parts as called out on drawing LS-33619.

40. Attach the LS-33612 fuel injection and oil metering pump drive system to the engine as follows:

    (a) With fuel injection pumps installed on belt drive box, hang the entire assembly from two hoists using lifting eye plate on belt box and a fabric sling (weight of drive assembly is approximately 250 lbs). Level the assembly at a height that exactly matches centerline of accessory drive shaft in belt box with centerline of crankshaft.

    (b) Lubricate anti-drive end of crankshaft, spline and pilot diameter on each side of spline with engine oil. Very carefully insert crankshaft into accessory drive shaft by moving engine truck. Rock flywheel back and forth until splines engage.

40. (c) Manually push the engine and drive assembly together as far as possible. A 5/16-24 "tooling bolt" with washers as required may be threaded into crankshaft to assist in drawing the two units together. Do not exceed 89 in-lb torque limit. Be sure flange on anti-drive cover is engaging dowels in recess on drive assembly. Install attaching bolts, washers, and nuts as called out on drawing LS-33612 and torque as required. Be certain anti-drive cover flange is seated against drive assembly before full torque is applied to attaching bolts nuts.

(d) If interference is detected when attaching drive system to engine, do not attempt to force the two units together. Locate the problem area and correct before proceeding.

(e) Install pulley spacer, encoder drive pulley, torsional readout wheel, locking washer, and bolt -- items 58, 59, 60, 55, and 54 on drawing LS-33612. Torque bolt to 89 in-lb and bend lock.

41. Install remaining items identified on fuel injection and oil metering pump drive system layout drawing LS-33612 as follows:

(a) Oil metering pump.

(b) Fuel injection pump support brackets between engine and pumps.

(c) Encoder.

(d) Magnetic sensor for torsional readout.

(e) Vertical and horizontal accelerometers.

42. Install an oil drain tube assembly SK-12848 and an oil vent tube assembly SK-12849 on the drive end and anti-drive end housings.

43. Install coolant inlet extension SK-12881 and coolant outlet extension SK-12880 on rotor housing.

44. Install the following named items which are specifically identified by DEI for each engine build:

(a) Fuel injection nozzles and spacer shims.

(b) Fuel injection lines - pumps to nozzles (fit, make up, and install).

(c) Spark plug and spacer shims.

44. (d) Turbocharger and exhaust extension with required gaskets - remainder of turbocharger system including intercooler is installed after engine is mounted on test facility.

(e) Instrumentation to be installed during engine assembly.

Assembly Tooling

The special tooling fabricated to build the first 1007R rig engine is listed below.

Tool/Dwg. Number	Description
ST-1057	Crankshaft Stabilizing Tool
ST-1058	Wrench-Spanner Nut-Crankshaft/Flywheel
ST-1059	Wrench-Spanner Nut-Balance Weight Attaching Nut
ST-1060	Pilot Geometry Tool
ST-1061	Ring, Side Housing Back-up
ST-1062	Rotor Side Seal Grinding Fixture
ST-1063	Adapter Plate for Engine Build Stand
ST-1064	Spanner, Dynamometer Shaft Nut
ST-1065	Master Apex Seal
ST-1066	Rtor Gear Puller Leg
ST-1067	Crankshaft Immobilizer
ST-1068	Rotor Bearing Installing Pilot
ST-1069	Stationary Gear and Bearing Assembly-Bearing Installing Pilot
ST-1070	Drive-End Cover Bearing Installing Pilot
ST-1071	Thrust Bearing Installing Pilot
ST-1072	Pressure Checking Fixture - Rotor Assembly
ST-1073	ADE Oil Seal Installation Mandrel
ST-1074	Thimble-ADE Oil Seal Installation
ST-1075	Master Apex Seal (undersize)
ST-1078	Test Fixture - Pressure Transducer P/N 210082 (AVL P/N AVL-80P500Ca)
SK-12866	"O" Ring - Rotor Pressure Check Fixture Nut
SK-12867	"O" Ring - Rotor Pressure Check Fixture
Diaphragm	
LS-33607	Counterweight Installation and Removal

REFERENCES

1. "An Analysis of Flow and Friction in Diesel Engine Bearings," Das, P. K., and Dancer, S. B., ASME Paper 82-DGEP-7.

END

DATE

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MAY 20 1985