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CRYOGENIC GEAR TECHNOLOGY FOR AN ORBITAL TRANSFER VEHICLE ENGINE AND TESTER DESIGN

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16. Abstract

This report presents technology available for gears used in advanced Orbital Transfer Vehicle rocket engines and the design of a cryogenic adapted tester used for evaluating advanced gears. The only high-speed, unlubricated gears currently in cryogenic service are used in the RL10 rocket engine turbomachinery.

Advanced rocket engine gear systems experience operational load conditions and rotational speeds that are beyond current experience levels. The work under this task consisted of a technology assessment and requirements definition followed by design of a self-contained portable cryogenic adapted gear test rig system.

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FOREWORD

This document presents the assessment of technology available for gears for use in an advanced Orbital Transfer Vehicle (OTV) rocket engine and the design of a cryogenic adapted tester for use in evaluating advanced gears. The work was conducted by Pratt & Whitney/Government Products Division (P&W/GPD) of the United Technologies Corporation (UTC) for the National Aeronautics and Space Administration — Lewis Research Center (NASA-LeRC) under Contract NAS3-23858-B.1 with Mr. J. M. Kazaroff as Task Order Manager.

Mr. J. R. Brown was the Program Manager for this effort and Mr. G. H. Duncan was the principal investigator and author of this report.

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SECTION I
INTRODUCTION

Very little material is available regarding high speed, high load gears for cryogenic service, which have no permanent, conventional lubrication system. The information that does exist concludes that the materials and lubricants currently used in the RL10 turbomachinery are the optimum of those evaluated.

The RL10 engine is the only system with applicable hardware available for review. The advanced engine requirements were developed for a 15K thrust Orbit Transfer Vehicle (OTV) engine and refined for the current 7.5K thrust OTV engine concept. This engine is shown schematically in Figure 1.

Advanced rocket engine gear systems require operational load conditions and rotational speeds that are beyond current experience levels. Therefore, the successful design of gears to meet advanced requirements necessitates a high speed cryogenic adapted rig tester system which will allow for experimental evaluation of gear performance.

The work to advance gear technology under this task consisted of a technology assessment and requirements definition followed by the design of a self-contained portable cryogenic adapted gear test rig system. The rig may be used to test spur or helical gears, in various cryogenic media, at pitchline velocities up to 28,275 feet per minute.

Based on review of the RL10 gear system, tooth wear appears to be the primary parameter determining gear life. A number of computer programs and methods which would relate the mechanisms of gear deterioration have been reviewed. It is recommended that the acquisition of several thermal and dynamic response models be further investigated and that these models be used in conjunction with the rig evaluation of candidate designs.

The two primary means of providing more durable gears are increased hardness and optimized tooth geometry. Several hardening processes which achieve hardness values well in excess of those obtained by carburizing or nitriding are being evaluated. The gear test rig will be used to evaluate this and other processes along with geometry modifications. Processes such as coating and plating should be considered as secondary solutions, because of the inherent problems of spalling and other surface deterioration. Tests from the rig may indicate a need for gear tooth cooling or in-service lubrication. The rig has been designed to accommodate such add on features.

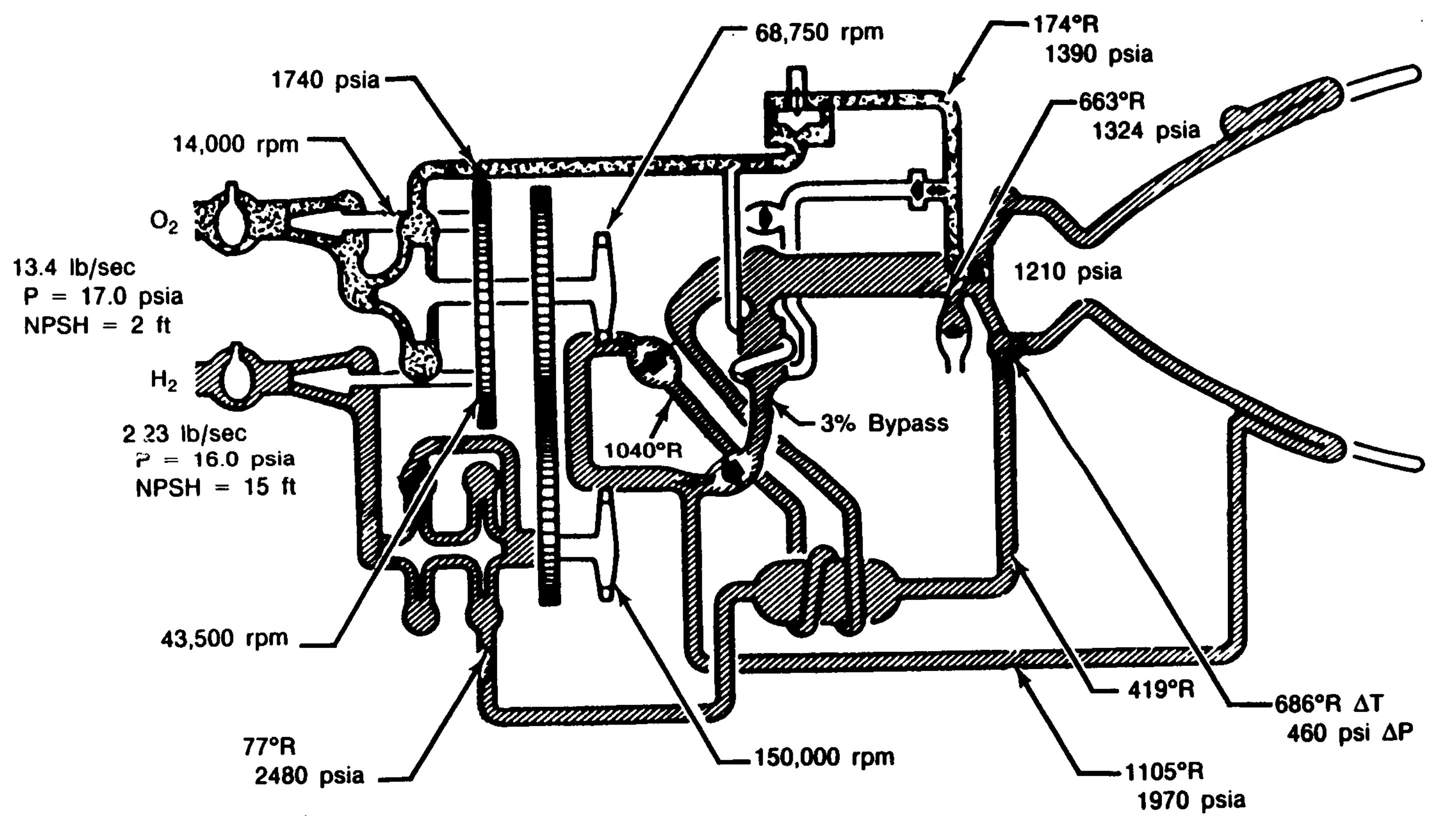

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Figure 1. Advanced Expander Engine Propellant Flow Schematic at Full Thrust (O/F = 6.0)

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SECTION II
TECHNOLOGY ASSESSMENT AND REQUIREMENTS DEFINITION

This study is a part of an exploratory technology program to evaluate materials and designs for advanced hydrogen cooled gears to be used in the turbomachinery of the Orbit Transfer Vehicle (OTV). The study addresses a gear technology assessment and a requirements definition as are applicable to the OTV program. The technology assessment effort consists of a literature survey, an engine systems review, and an advanced engine requirements review. The requirements definition effort consists of energy mechanism definition, computer modeling to relate energy mechanisms, evaluation of the models, and definition of specific requirements for gears.

A. LITERATURE SURVEY

The literature survey was conducted to gain a better understanding of the parameters affecting gear life, operation, performance, modes of failure and deterioration. The results of the survey have been compiled into a technical database. The contents of this database, along with a summary of each document contained is defined in Appendix A.

Very little information was located that addressed gears operating without conventional lubrication in a cryogenic environment. One paper was located that included information from tests performed on several bearing and gear materials in media including LH_2, LN_2, and LO_2. SAE 9310 (AMS 6265) was found to give the best results when tested in LH_2. Although the loads were considerable, the test speeds were relatively low.

The majority of the information contained in the database relates to gear design, performance and failure, and to studies of wear and friction in cryogenic environments using various "dry" lubricants on disk/rider test pieces.

Although most of this material does not relate directly to high speed, unlubricated hydrogen cooled gears, it has been useful in the understanding of basic gear performance and the wear and friction of materials in the absence of conventional lubrication. The dry lubricants used were laminar solids and polymers. These types of materials do not appear to be suitable for high load applications, because of low elastic moduli and bond strength to the base material.

Several papers which were presented at a recent ASME power transmission and gearing conference have been obtained. These relate to calculation of gear tooth flash temperatures and evaluation of the dynamic response of gears. This material may prove to be useful in providing a more sophisticated analysis of advanced gear systems.

Since the amount of related material located was very small, the literature survey was extended beyond the normal computerized search. Numerous people were contacted both in and outside of United Technologies Corporation in the hope that some additional leads could be established. These contacts included gear experts, gear manufacturers, authors of gear textbooks, and university professors. This endeavor again confirmed that very little is known about cryogenic gears.

A continuing effort is underway to review technical periodical publications and technical society journals for any information which might be related to this gear technology program. This effort includes a search for information related to new materials and processes that would produce improved gear durability, as well as computer software that would enhance the gear design and evaluation process.

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B. ENGINE SYSTEMS REVIEW

The engine systems review was directed primarily toward the RL10 since this is the only system available with hardware operating at conditions related to the study.

The first step in this review was to study the wear patterns on some of the RL10 turbopump gears. Figure 2 shows an RL10 idler gear as finished and before coating. Figure 3 shows the same gear after it has been run in the engine. The amount of run time on the gear is unknown but it is expected to be less than one hour.

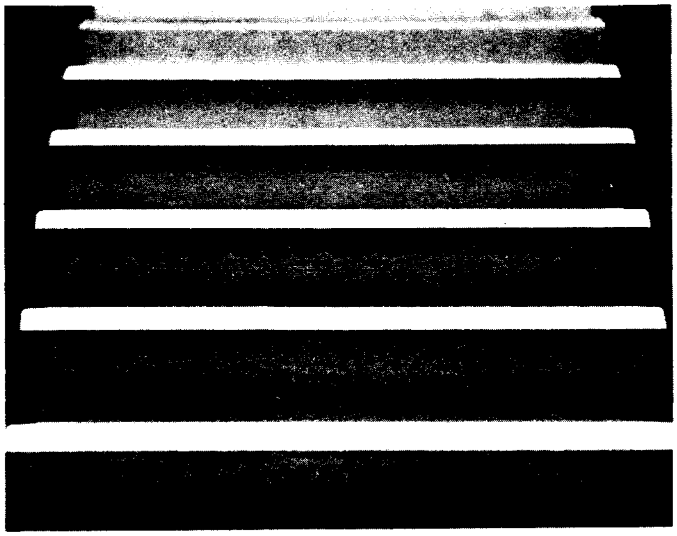

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Figure 2. RL10 Idler Gear Before Coating

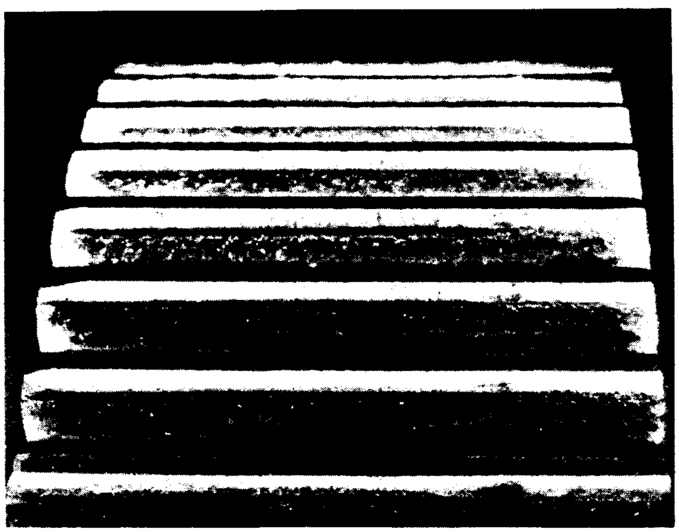

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Figure 3. RL10 Idler Gear Wear Patterns

The typical gear wear pattern was found to be scoring at the tip, scoring and fine pitting between the pitch line and tip area, and deep, smooth wear below the pitch line. This wear pattern is illustrated in Figure 3.

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Figures 4 and 5 show a more detailed description of the wear patterns for a set of RL10 turbopump gears. It is not known if the gears observed were run as a set in the engine or what the accumulated run time was for the gears. However, this type of wear is typical of that seen in various sets of RL10 gears.

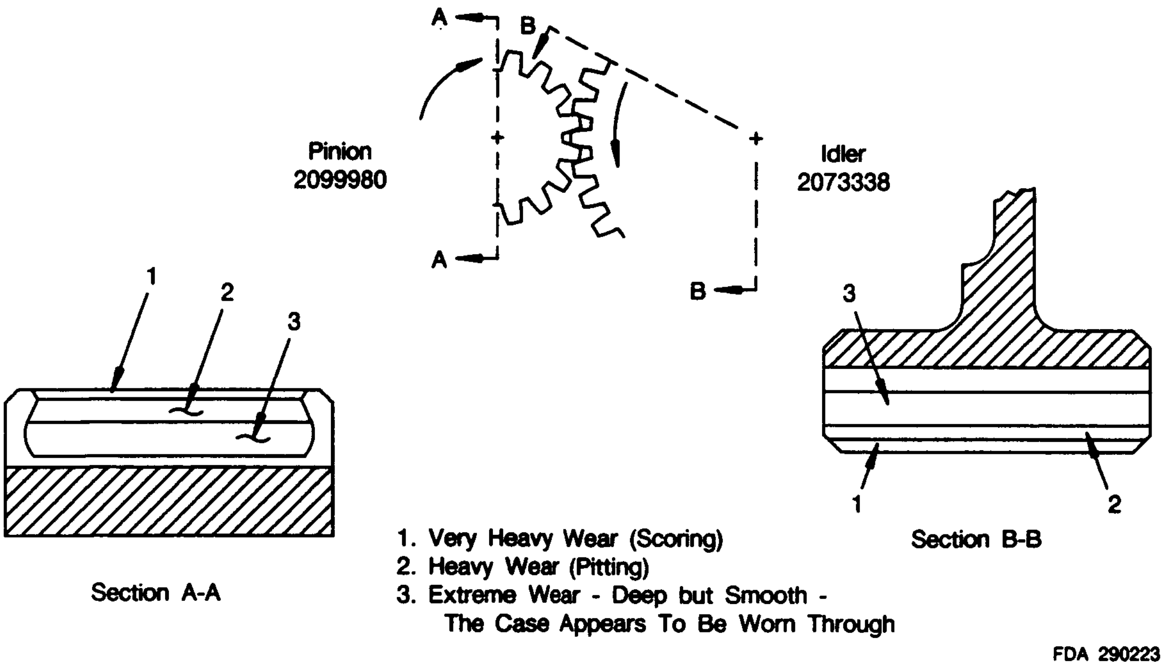

Figure 4. Detailed Description of RL10 Turbopump Gear Wear Patterns (View 1)

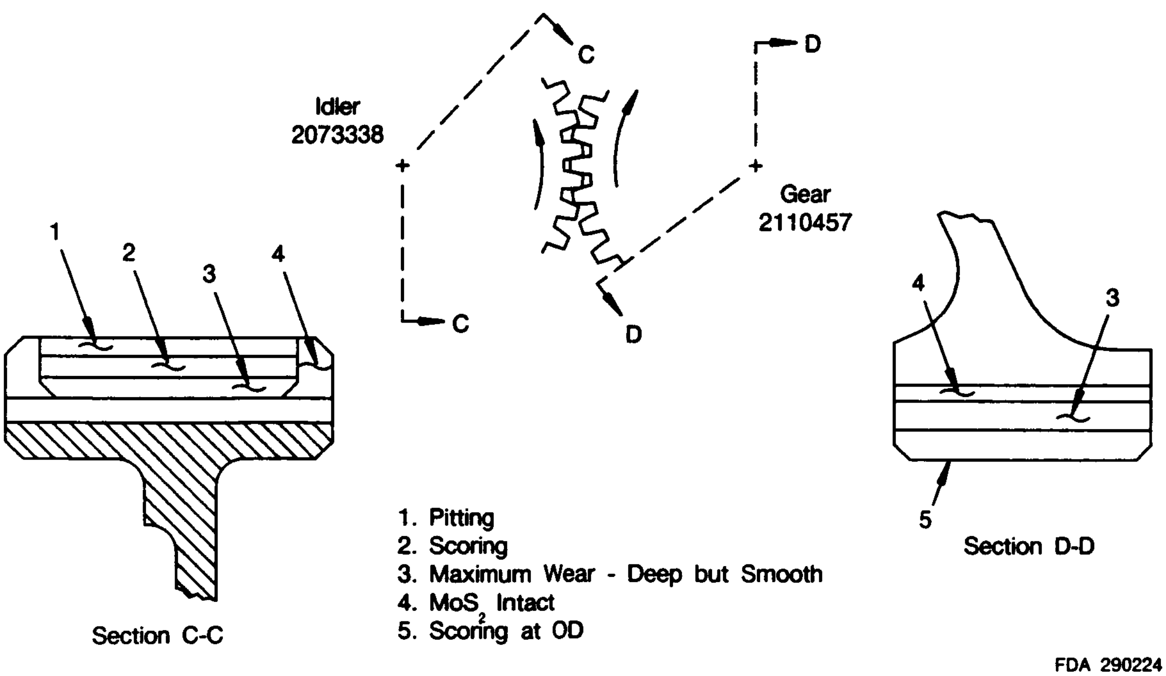

Figure 5. Detailed Description of RL10 Turbopump Gear Wear Patterns (View 2)

Figure 6 shows the wear pattern for an idler gear that had an accumulated run time of 69.13 minutes.

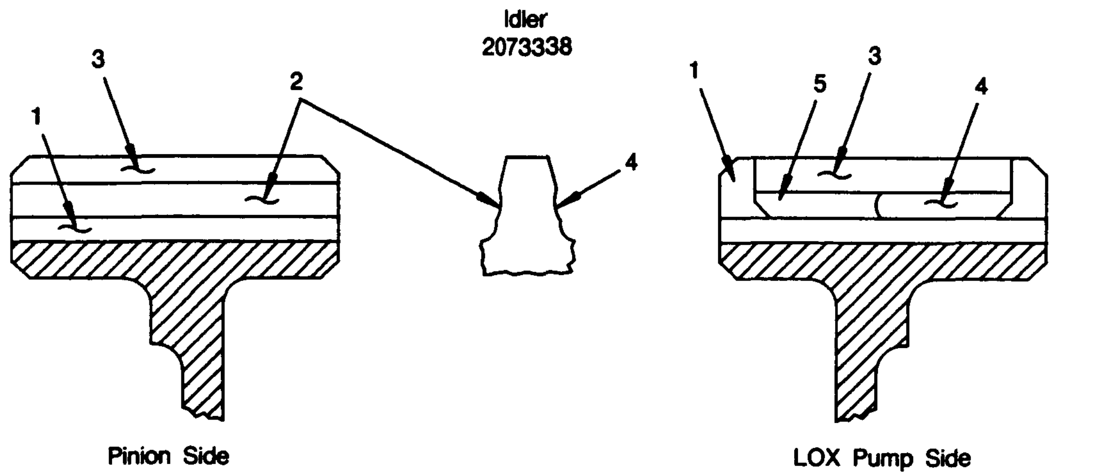

Figure 6. Idler Gear Wear Patterns

The RL10 experience contributed to identifying modes of wear and deterioration, such as initial deterioration of dry film lubricant, scoring, scuffing, pitting, and the very deep wear due to the rapid relative sliding of the gear teeth.

A spur gear tooth design and analysis computer program (Reference 1) was used to calculate performance parameters of the RL10 turbomachinery gears. These values are summarized in Table 1.

Vibratory stresses have not been addressed in much detail at this time. The gear stress program (Reference 1) calculates a dynamic bending stress, but this is based on an assumed tooth profile error factor. One method located during the literature search describes a simple model for dynamic stress calculation (Reference 2). The method could be computerized easily, but it has not been evaluated as of this writing. Another paper (Reference 3) describes an extensive computerized method for determining the dynamic response of gears, which could be useful in evaluating high speed gears. It is planned that more effort be directed toward vibrations and vibratory stresses when the specific requirements of the advanced gears are addressed.

A hardness check was run on a used gear to determine if excessive heat buildup during operation had caused a loss in hardness. The findings were that hardness was to print and there was no noticeable evidence of excessive operating temperature. The sample did reveal the heavy wear area below the pitch line and its depth was measured at about 0.008 inch, or about half way through the case.

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Table 1. *RL10 Turbomachinery Gear Performance Parameters*

Parameter	*RL10 (Baseline)* *Pinion/Idler*	*Idler/Lox*
Gear Type	Involute Spur	Involute Spur
Gear Material	AMS 6265	AMS 6265
Diametral Pitch	13.9592	13.9592
Pressure Angle	22.5	22.5
Center Distance (at assembly)	3.4744	4.9788
Pinion Pitch Diameter	2.0058	4.9430
Gear Pitch Diameter	4.943	5.0146
Pitch Line Velocity, ft/min	17,193	17,193
Contact Ratio	1.337	1.331
Environment	210 to 250°R Gaseous H_2	210 to 250°R Gaseous H_2
Lubricant	MoS_2	MoS_2
Maximum rpm	32,741	13,097
Max Sliding Velocity, ft/min	3,372	1,908
Tangential Tooth Load, lb/in.	206.4	320.7
Dynamic Tooth Load, lb/in.	1,260.3	1,490.9
Hertz Stress (Steady), psi	65,541	61,849
Bending Stress (Steady), psi	8,819	11,398
Bending Stress (Dynamic), psi	53,850	52,997

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Computer technology was also used to evaluate the action of the gear teeth during the meshing cycle (Figure 7). A CAD model (Reference 10) was generated and plotted 10X. This model was used to view gear teeth in various stages of approach and recess.

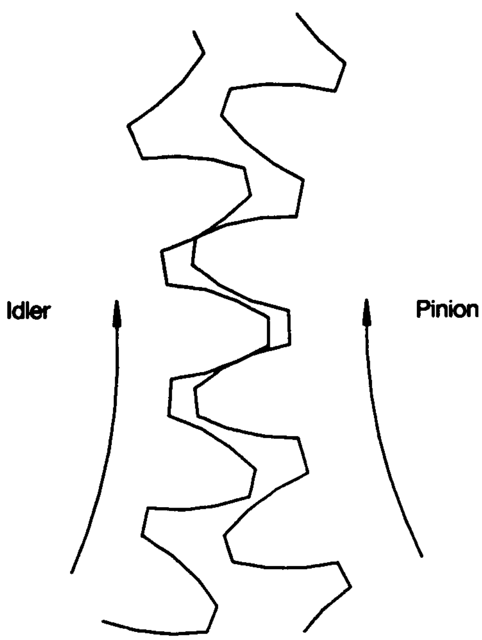

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Figure 7. Gear Tooth Mesh Plot

C. ADVANCED ENGINE REQUIREMENTS REVIEW

Current OTV gear requirements are based on an earlier 15K OTV design study and a study for a scaled down 7.5K design, which is currently underway. The requirements and operating conditions as presently defined for the 7.5K OTV fuel pump pinion and main idler gear are listed in Table 2.

Table 2. *Fuel Pump Pinion and Main Idler Gear Requirements*

Parameter	*OTV Engine Pinion/Idler*
Gear Type	Involute Spur or Helical
Gear Material	AMS 6265
Diametral Pitch	24.000
Pressure Angle	22.5
Center Distance	3.271
Pinion Pitch Diameter	1.000
Gear Pitch Diameter	5.542
Pitch Line Velocity, ft/min	39,000
Contact Ratio	1.423
Environment	50 — 150°R Gaseous H_2
Lubricant	TBD
Maximum rpm	150,000 (Pinion)
Max Sliding Velocity, ft/min	8,311
Tangential Tooth Load, lb/in.	60
Dynamic Tooth Load, lb/in.	1,009
Hertz Stress (Steady), psi	45,831
Bending Stress (Steady), psi	3,039
Bending Stress (Dynamic), psi	51,185

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State-of-the-art literature concludes that the operational limit for spur gears is a pitch line velocity of 20,000 feet/minute (Reference 4). Above these speeds, helical gears are recommended. Since the pitch line velocity of the OTV gears is double the recommended value, an alternate design using helical gears has been considered. A design scheme has been established and some preliminary calculations have been made as part of a 7.5K OTV redesign review (Reference 5), but a substantial effort is still required to develop a preliminary configuration.

The advantage of helical gears is that a larger contact ratio can be achieved than with spur gears and this promotes smoother operation with less susceptibility to dynamic stresses. On the other hand, helical gears have a resultant axial thrust load which can complicate the shaft bearing design. This load is not felt to be excessive for the OTV application.

The decision of which type of gear to use must be based on information obtained from the gear test rig. There are a number of options being investigated which may serve to improve the operation and life of spur gears. These will be discussed in another section of this report.

The requirements definition effort consists of energy mechanism definition, computer modeling to relate energy mechanisms, evaluation of the models, and definition of specific requirements for gears.

1. Energy Mechanisms

One of the primary energy producing mechanisms of spur gears is the motion of one tooth on another. Observation of the wear pattern of the baseline gears suggests that wear rate could be related to sliding velocity of one tooth on the other. Figure 6 illustrates that the maximum wear occurs at the location of maximum sliding velocity and minimum contact area. It is suspected that this wear is the result of abrasion and adhesion.

The mechanism of surface wear is rather complicated since it can be dependent on load, speed, tooth geometry, and operating environment. Wear occurs by surface abrasion and by adhesion due to cold welding. Surface stresses are dependent on direction of sliding and coefficient of friction in addition to loading. Also heat buildup due to rubbing could be a significant factor.

Since the power related to tooth motion is a function of load, coefficient of friction and sliding/rolling velocity, it is a potential quantity for relating to gear wear.

The power related to the rolling/sliding of contacting teeth may be expressed by the following equation:

$P = F\mu V = F\mu_1 V_1 + F\mu_2 V_2$ where

F = Load
μ_1 = Sliding coefficient of friction
μ_2 = Rolling coefficient of friction
V_1 = Sliding velocity
V_2 = Rolling velocity.

The mechanisms of sliding and rolling velocity may be better understood by observing Figures 8 through 10. The velocity of a given point on an involute spur gear tooth is related to the tooth radius of curvature and rotational speed by $V = 2\pi\rho N/60$. The velocity then varies with the radial location on the gear tooth.

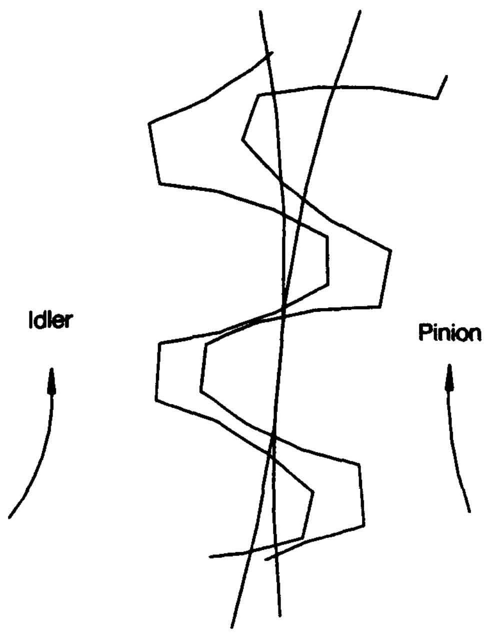

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Figure 8. Gear Teeth Relation at First Point of Contact in the Approach Cycle

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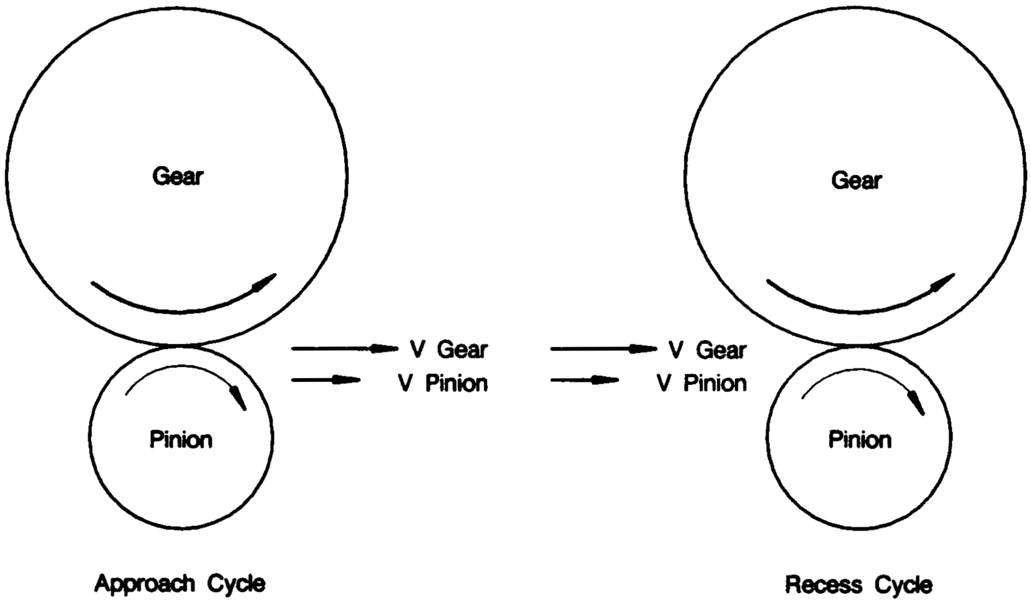

Figure 9. Approach Cycle Velocities of the Gear Tooth and Pinion Tooth

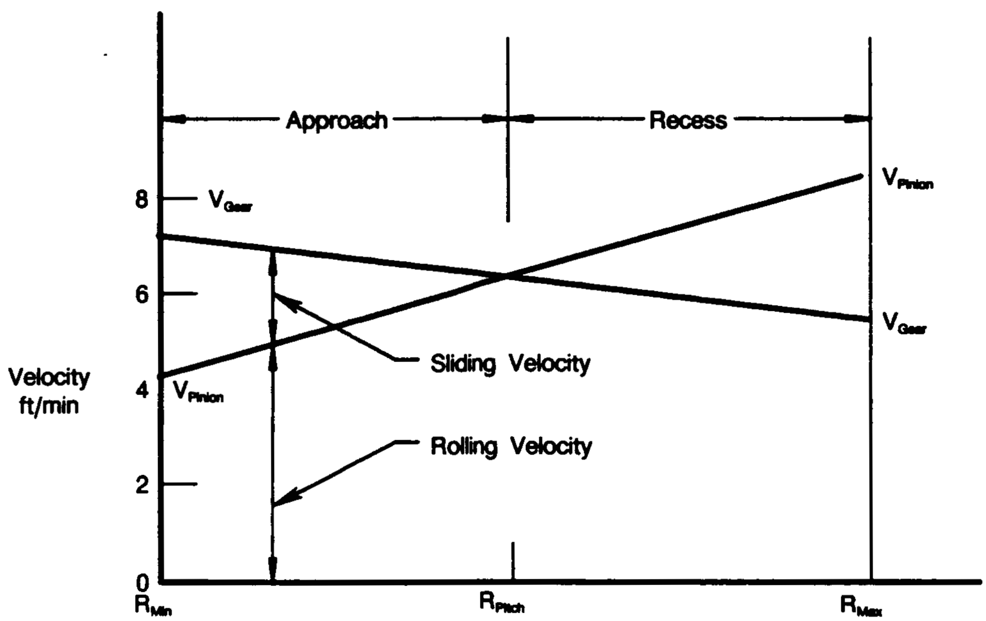

Figure 10. Relative Tooth Surface Velocities of RL10 Pinion and Idler Gears

Figure 8 shows the relation of a set of gear teeth at the first point of contact in the approach cycle. At any given time the gear and pinion teeth are rolling on each other at the velocity of the slower member. The difference in velocities is made up by sliding.

Figure 9 shows that, during the approach cycle, the velocity of the gear tooth is greater than the pinion tooth and, thus, the gear slides on the pinion. In the recess cycle the reverse is true, and the pinion slides on the gear. The relative magnitudes of this for the baseline pinion/idler set is shown in Figure 10.

The preceding relationships were used to arrive at the equation for power dissipated in sliding. Whether or not power dissipated during tooth contact is a good indicator of degree of gear tooth wear must be determined by the rig testing.

The heat generated as a result of this power dissipation may be determined from the physical size and heat transfer properties of the gear tooth. The ability of the gear tooth to dissipate this heat depends also on the heat transfer properties and the elapsed time between rubs. One of the biggest unknowns in this process is coefficient of friction which could vary over a wide range and for both sliding and rolling.

Methods for calculating flash temperatures for gear teeth, both with and without coatings, are described in two recent ASME papers (References 6 and 7). However, no attempt is made to relate these quantities to gear tooth wear. Since the hardness evaluation of an RL10 gear did not indicate the presence of excessive heat, it is suggested that efforts be directed toward more urgent areas until rig testing indicates a need for technology generation in this area. If it is found that gear durability is related to heat generation at the tooth interface, then References 6 and 7 could provide some insight into tooth cooling.

Three of the most important design variables for spur gears are diametral pitch, pressure angle, and tooth length. Figures 11 through 13 show the relationship of these design variables to various performance parameters.

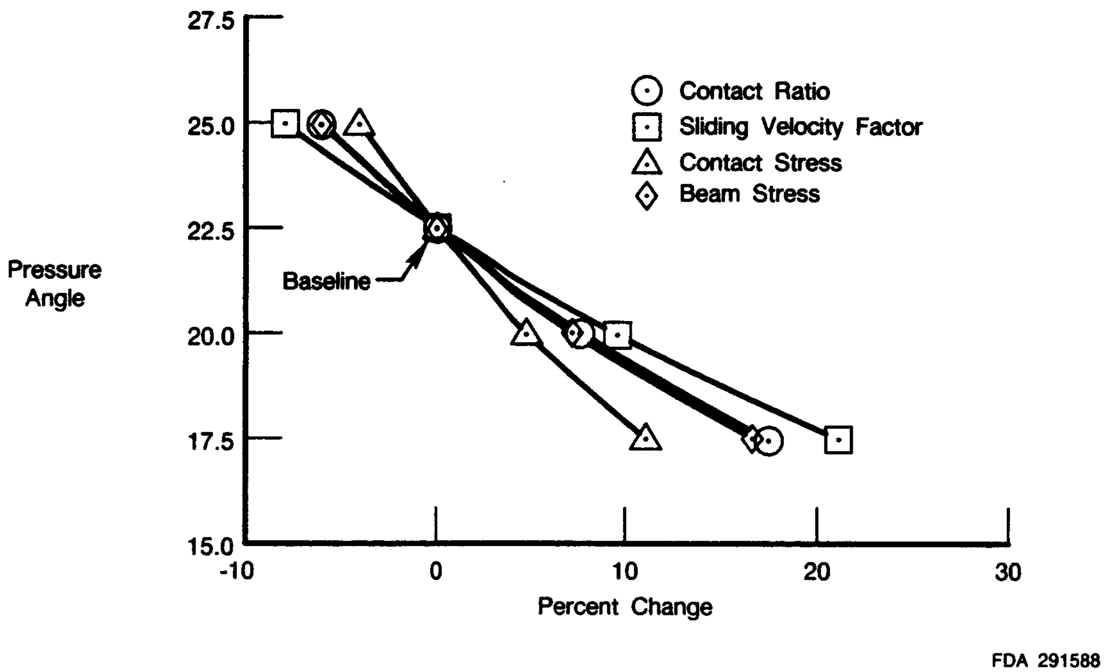

Figure 11. Pressure Angle Variations

Figure 11 shows the effect of variation in pressure angle on contact stress, beam stress, sliding velocity, and contact ratio while diametral pitch and tooth length are held constant. In Figure 12 the diametral pitch is the variable, and in Figure 13 the tooth length is varied. Pitch line velocity is a constant in all three cases.

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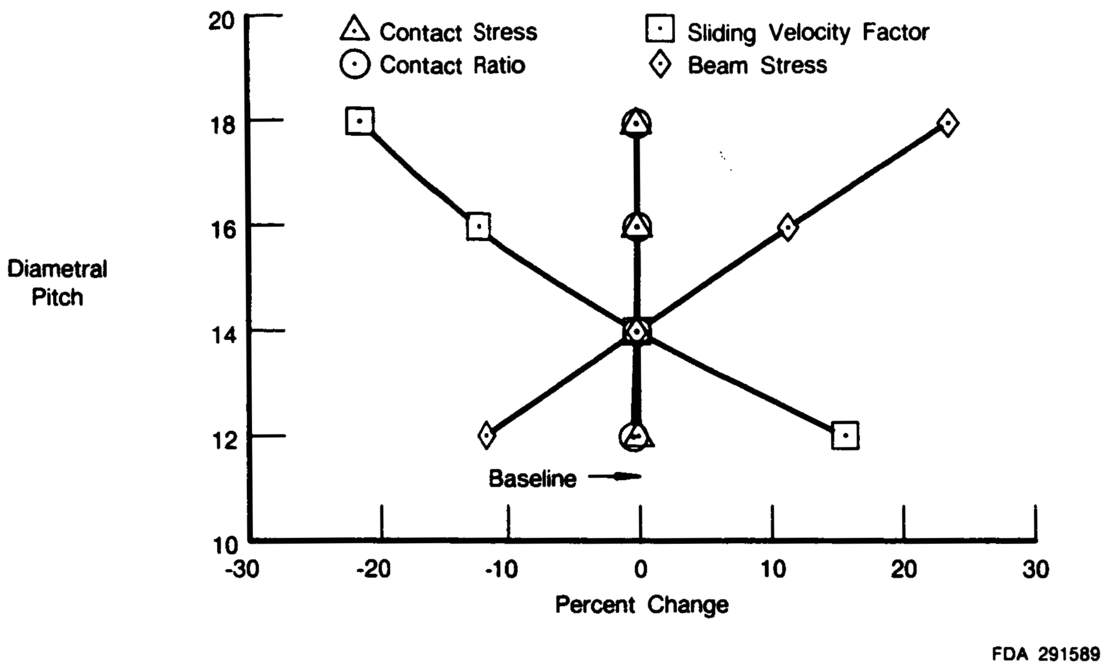

Figure 12. Diametral Pitch Variations

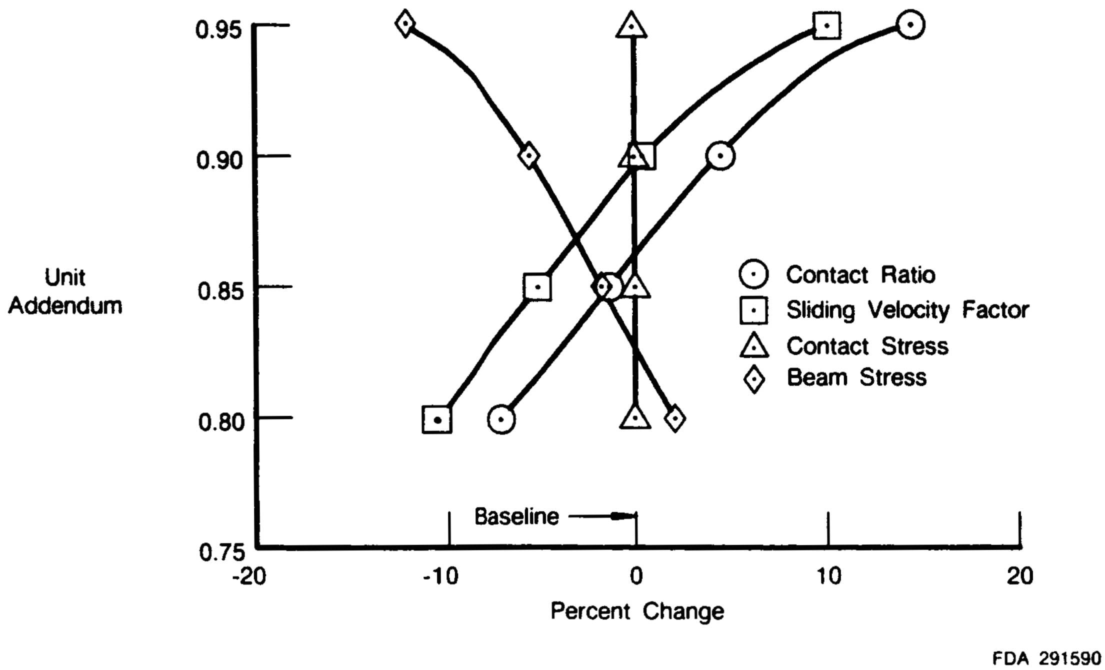

Figure 13. Tooth Length Variations

Several conclusions can be drawn from Figures 11 through 13. Increasing the pressure angle reduces relative sliding and both contact and beam stress, which is desirable. Contact ratio, however, is also reduced by increasing the pressure angle. This could be detrimental at higher speeds because the increased time of single tooth contact would lead to rougher operation and higher dynamic stresses.

Increasing the diametral pitch, which results in a finer tooth, has even more effect on sliding velocity than pressure angle. Contact stress is not affected and the effect on contact ratio is negligible, but beam stress increases due to reduction in tooth thickness.

Decreasing the length of the tooth reduces sliding velocity, but also reduces contact ratio. Contact stress remains constant but beam stress is increased.

If sliding velocity proves to be a dominant factor, it is apparent from these figures that it may be reduced by several means. It is possible that offsetting factors could occur, however, in the form of increased beam stress and reduced contact ratio. Both of these factors could be quite detrimental at high speeds where vibrations and fatigue become more of a concern. Again, the final validity of such an approach will have to be determined by rig testing.

2. Computer Models

A portion of the gear technology requirements definition is to develop or acquire computer programs which will relate the various energy mechanisms. Computer technology has been used in this program to evaluate the RL10 gears, to design the baseline test gears, and to design the cryogenic adapted gear test rig (References 8 and 9). The current effort, along with some areas for future development, may be summarized as follows.

- A CAD gear mesh model was created for the RL10 gear train and was used to study the action of the gear teeth during the meshing cycle (Reference 10). A portion of this model is shown in Figure 7.

- Computer Program P59704, "Spur Gear and Involute Spline Tooth Design and Analysis," (Reference 1) has been used extensively in the evaluation of gear tooth stress and sliding velocity and other performance parameters on the RL10 and proposed OTV designs. The program calculates bending and contact stress and dynamic stress due to profile error. Also calculated are performance parameters, such as contact ratio, sliding velocity, and tooth flash temperature. The program will generate mesh plots and information for detail drawings including tooth profile modification values.

  The program does not calculate detailed tooth deflection or stresses due to rotation, thermals, or vibration. This program will continue to be a valuable tool, and it is possible that the program could be expanded to calculate gear life based on equations developed during the gear technology effort.

- FTDM 427, "Torsional Vibration of Complex Gear Systems," is a computerized method of analyzing gear systems for torsional natural frequencies. The utility of this program is yet to be established.

- A two-dimensional Nastran model for the analysis of spur gears has been built (Reference 11). This program will calculate stresses, loads, and deflections due to rotation, thermals, and external loading. The model has been checked out for rotation and thermals and found acceptable. A version to evaluate various tooth loading conditions has been started but is not complete. Additional checkout runs would be necessary to determine how to accurately model tooth loads in order to obtain true contact stresses and deflections.

  A program of this type could prove quite useful if developed for the analysis of high speed, highly stressed gear systems. However, Nastran is not a tool that the casual user can apply without a significant amount of advice and assistance from an expert. Also it must be noted that this type of program is best used for detailed analysis and is rather cumbersome for preliminary design applications. A great deal of effort could be consumed in perfecting such a program. It is felt that at the current time efforts would be better spent on other phases of the program.

- A method is available which will provide a more accurate analysis of dynamic stresses due to tooth profile error and inertials (Reference 2). This method compares tooth inaccuracies during the mesh to the sliding together/apart of two wedges. This model has not been computerized, but could be with a relatively small effort.

- The relationships which relate sliding to power dissipation could be further developed to calculate the transient heat transfer response of the gear system. Since the power dissipated is a function of both sliding and rolling coefficients of friction, there is the possibility for much variation over the range of life and operating conditions of a given set of gears. An extensive effort could be expended while yielding only limited value results. However, no other efforts have been noted which relate sliding and resultant heat rise to gear life, so such an endeavor would certainly be in the category of advanced technology.

- Two references (6 and 7) have been cited which present methods for calculation of gear tooth flash temperatures. One ASME paper (Reference 6) investigates the thermal behavior of gear teeth coated with MoS_2 and the effect of the coating thickness on temperature rise. Another ASME paper (Reference 7) describes a method for the calculation of gear tooth transient temperatures. The model has been computerized and a printout of the program has been obtained. Also, the computer tapes will be copied and the program evaluated to determine the feasibility of conversion for in-house use.

- An extensive model for the evaluation of gear dynamic factors is presented in the ASME paper, "An Analytical Evaluation of Gear Dynamic Factors Based on Rigid Body Dynamics" (Reference 3). The method takes into account the specific tooth profile variations and is also capable of evaluating the effects of dynamic transmission errors. This method would be a useful tool for the evaluation of existing gears, particularly those which would operate at extremely high speeds. The model has been computerized, but it is not known whether it is available for acquisition.

- A general analysis program for spur and helical gears is commercially available and a preliminary review indicates that it could be a valuable tool. A request has been submitted to the computing group to purchase the program on a trial basis for further evaluation.

The current recommendation is that the acquisition of the thermal and dynamic response models be further investigated. The information provided by these models cannot be used efficiently until the gear test rig is in operation.

3. Evaluation of Computerized Models

An objective evaluation of the computerized models cannot be made until the gear test rig is in operation and has generated data over a variety of operating ranges. There is not enough information available for the RL10 gears to provide a basis for evaluation.

4. Specific Requirements for Rig Test Gears and Advanced Gears

Requirements for rig test gears for the RL10 baseline tests have been established (Reference 12). The test rig is designed to use two equal sized gears on 4.000 inch centers (4.000 inch P.D. gears). The test gears were designed to have the same tooth geometry as the RL10 pinion and idler gears, except that slight modifications were made to tooth thickness and length in order to duplicate the backlash and contact ratio of the RL10 gears during operation. The idler gear has a modified tooth profile so one of the test gears was modified to be consistent.

Since the gears must be the same size, the exact tooth curvature pattern of the pinion/idler set cannot be duplicated. It is possible, however, to vary torque and speed so that the same levels of contact stress and relative sliding that are experienced by the engine hardware can be duplicated. The material for the test gears is AMS 6265, which is the same as for the RL10 gears.

It is not possible to be specific about individual design variables for the advanced gears until the results of the baseline tests are available. The modes of failure and deterioration must be well understood before it is practical to investigate the effects of modifications to geometry, new coatings/treatments, etc.

A number of candidate improvements have been reviewed as a part of the analytical portion of this program. Some can be readily eliminated as unsuitable, while others show potential for improvement and will probably be tested once the rig test program is underway and the advanced gear requirements are better understood.

Improvements can be classified into the categories of materials, treatments, antifriction coatings, and geometry optimization. Treatments include diffusion processes, such as carburizing, nitriding, sulfurizing, borofuse, and ion implantation. Antifriction coatings largely consist of laminar solids and polymers.

The baseline material, SAE 9310 (AMS 6265), will be hard to improve on because of the high ductility at cryogenic temperatures. Previously, it was stated that 9310 proved to be the best choice of several gear materials tested in LH_2 (Reference 13). The new gear materials, such as M-50, CB5 600, and VASCO 350 which were developed for high temperature applications, are either very brittle at cryogenic temperatures or no cryogenic data is available. The same is true for most other steels which might otherwise be potential candidates.

Of the processes which involve treatment of the gear teeth to improve wear resistance and fatigue, a process known as borofuse appears to have the most merit at this time. The borofuse process involves a series of thermal diffusion treatments which result in the formation of boride compounds in the metal surfaces. The treated material then exhibits the properties of low surface coefficient of friction and high hardness, the hardness values obtained being well above those achievable by nitriding or carburizing. The merits of this process are now under investigation.

Several nitriding processes were reviewed. Although these might provide better wear resistance than the current carburizing process, the gain is not of the magnitude expected from the borofuse process. Also the case depth of the nitrided area is very small.

The process of ion implantation, which can change the surface properties of a material by embedding foreign ions, continues to be under investigation. Ion implantation can increase surface hardness and resistance to wear, and also produce a lowered coefficient of friction. Additional research will determine whether this process can be applied to gear teeth.

Quite a few processes related to coating or plating, which would lower the gear tooth coefficient of friction, were reviewed. Two of these were processes available from General Magnaplate, consisted of a lubricant which was infused into an electroplated base material. These are NEDOX which is PTFE infused into a nickel alloy and HI-T-LUBE which is MoS_2 bonded by a nickel/silver plating. HI-T-LUBE has shown some good results at cryogenic temperatures, but conditions of high sliding velocity and surface stress are conducive to spalling or other surface deterioration for any coated or plated material.

MoS_2 has shown good results in a number of tests (References 6 and 14) as far as providing a low coefficient of friction. However, as with the other candidates in this category, there is a problem of adherence. Thick coatings, though providing the most wear resistance, tend to spall from the surface and, according to Reference 2, also contribute to the temperature rise at the tooth surface as a result of their thermal characteristics. It is possible that the optimum thickness of MoS_2 might provide additional protection for the gear teeth when combined with a wear resistant base.

Rig testing may establish that it is essential to replenish the dry film lubricant on the gear teeth. The test rig is being designed accordingly to permit access to the gear for mounting a sacrificial lubricating gear or other means of lubricant transfer.

A number of reports were reviewed which evaluated polymers as lubricants. PTFE and PTFCE provided effective lubrication but were not well matched with the base metal in thermal contraction. Also the elastic moduli of the polymers are very low, so they are subjected to cold flow, and thus not very suitable for high loads.

A primary means of providing a more durable gear is to optimize the geometry design variables. As has been shown in Figures 10 through 12, adjusting the tooth geometry can have a substantial effect on performance parameters, such as sliding velocity, contact stress, beam stress and contact ratio. However, not all these factors work to a positive advantage so some experimentation in the rig will be necessary in order to determine the optimum variation. Gear operating clearance, which may not be the same as gear assembly clearance due to thermals and dynamics, could have a significant effect on gear performance parameters. Clearances, which are controlled by gear operating center distances, should be given careful consideration.

Another geometry related option, helical gears, could be advantageous at high speeds. This subject was addressed earlier, but it should be noted again that helical gears provide a smoother mesh because of their higher contact ratio. In a spur gear design the number of teeth in contact is constantly changing from one to two as the gear set progresses through the mesh cycle. The result is a vibrational excitation due to the uneven tooth deflections caused by the change in stiffness of the mesh. The situation is further aggravated by the individual tooth error profile. Furthermore the deflections of the teeth would contribute to additional scoring from increased tooth tip contact.

SECTION III
CRYOGENIC ADAPTED RIG TESTER

A. SUMMARY

Advanced rocket engine gear system requirements call for operational load conditions and/or rotational speeds that are above the current experience levels. Therefore, the successful design of gears required in advanced rocket engines necessitates a high speed cryogenic adapted rig tester. Testing is essential to evaluate the effects of high sliding velocities and contact stresses on advanced materials, surface treatments, and coatings. Also, rig testing will determine the effect of various cooling schemes and any potential dry lubrication schemes that might increase gear life and performance.

The work accomplished under this task creates a design for a self-contained portable cryogenic adapted gear test rig system for spur or helical gears. The rig system may be used to test gears in various cryogenic media, in this case gaseous hydrogen. The rig is self-contained in that it may be operated safely indoors and in the proximity of other equipment and facilities when using gaseous hydrogen or other test media which present environmental hazards. To accomplish this, the portion of the rig housing the gears is mounted in a containment compartment which is vented to the outside atmosphere. Plumbing for the rig is routed through a vent stack so that in the event of test media leakage, the gas will escape to the outside atmosphere. The rig is entirely portable and it may be driven by any conventional four square test rig having the same center distance as the cryogenic rig. The cryogenic rig is supported on a pedestal with an adjustable level sliding track which permits adjustment and alignment with the drive system.

The rig is capable of operating at up to 27,000 rpm. (This limit is a result of the current drive system.) Since the gears are on a 4.000 inch center distance, the maximum pitchline velocity is 28,275 feet per minute. At the maximum speed the B1 life of the roller bearings is 135 hours at 500 inch-pounds torque and 19 hours at 1000 inch-pounds, for gears with a 25 degree pressure angle. The roller bearings which have lower life are replaceable without disassembly of the rig. The rig coolant system is currently designed to provide gaseous hydrogen at 70 psia inside the rig with a temperature of 210°R at the exit side. Lower temperatures may be required for future programs.

B. DESIGN RATIONALE

This system is somewhat different and more extensive than the system described in the original gear technology program work plan (FR-17802-2). The original plan was to acquire an Erdco Universal Tester and a WADD high speed test head which is similar to the Ryder Gear Machine. After investigating the current acquisition terms and technical merit of the system, it was determined that the original acquisition plan was no longer feasible technically or financially. This can be attributed to several factors.

The source from which the Erdco gear tester system was to be purchased was no longer interested in selling due to increased replacement cost and current usage of the equipment. It was stated that the current replacement cost of the WADD head was almost double the price that had been previously quoted. The replacement cost of a set of drive shafts alone was quite substantial.

A review of the technical merit of the WADD test head, based on the Annual Book of ASTM Standards (pp. 110-131, 1982) and the log books of the original RL10 gear test program done in 1963, was conducted. The review revealed several limitations and potential problem areas. Although the ASTM Annual quoted a speed capability of the WADD head of up to

30,000 rpm, it could not be established that the equipment under consideration could actually meet this requirement.

Torque loading in the WADD head is achieved by hydraulic pressure exerted against one of the shafts, causing it to act like a piston. The pressure causes a relative displacement of the helical slave gears and thus "winds in" torque into the four square system. The history of the early RL10 gear test program indicated some problems maintaining repeatable torque values. This conclusion was a result of strain gaged torque measurements. Certain cases failed to indicate torque when pressure was applied. Also difficulty was encountered with high breakaway torques and with the ability of the motor to drive the rig.

Although some of these problems could have been attributed to the cryogenic adapter, they would be expected to persist if a similar system were used with a new cryogenic adapter. The combination of these undesirable factors supported the feasibility of using a different system.

A gear vendor was contacted concerning a design for a cryogenic adapted gear rig. Initially an interest was indicated, but, after reviewing the requirements, the vendor believed the design to be a more difficult and lengthy project than they were willing to pursue.

At this point an initiative was begun to locate a gear test facility within the corporation that might provide support to the cryogenic rig project. It was determined that Sikorsky had a 5.000 inch gear rig and that Pratt & Whitney Canada (P&WC) had a 4.000 inch gear rig. Not much information was obtained about the Sikorsky rig, but P&WC was very cooperative and provided some drawings and reports on their rig. The P&WC rig is a modern design capable of operating at speeds up to 27,000 rpm.

After some detailed discussion, it was agreed that the cryogenic adapted rig could be tested at the P&WC facility, using their rig as the drive system, provided that proper safety precautions were addressed. Since gaseous hydrogen can be hazardous and the cryogenic gears are unlubricated, the safety of the physical plant, personnel, and rig equipment was a major factor in the cryogenic rig system design.

Since the P&WC test facility is located indoors, a self-contained portable rig system was designed to ensure that test media leakage, if any, would be contained until it was ducted to the outside atmosphere. The portability of the rig was necessary, because the test program will be run in at least two segments which are spaced a year apart.

C. TECHNICAL DESCRIPTION

The test rig system, shown in Figures 14 through 16 consists of the cryogenic gear housing (1), the containment housing (2), the drive coupling (3), the support pedestal (4), the transition housing (23), and the drive system (5). The circled numbers in the text correspond to the numbers in the illustrations for easy identification of the cryogenic test rig parts.

The test gears (6) are mounted on the shaft in a cantilevered manner which permits easy visual inspection and replacement. The gear drive shafts (7) are supported by two bearings. The bearing (8) closest to the gear is a roller bearing; the other bearing (9) is a duplex ball bearing, which is located away from the cryogenic media in an oil lubricated environment.

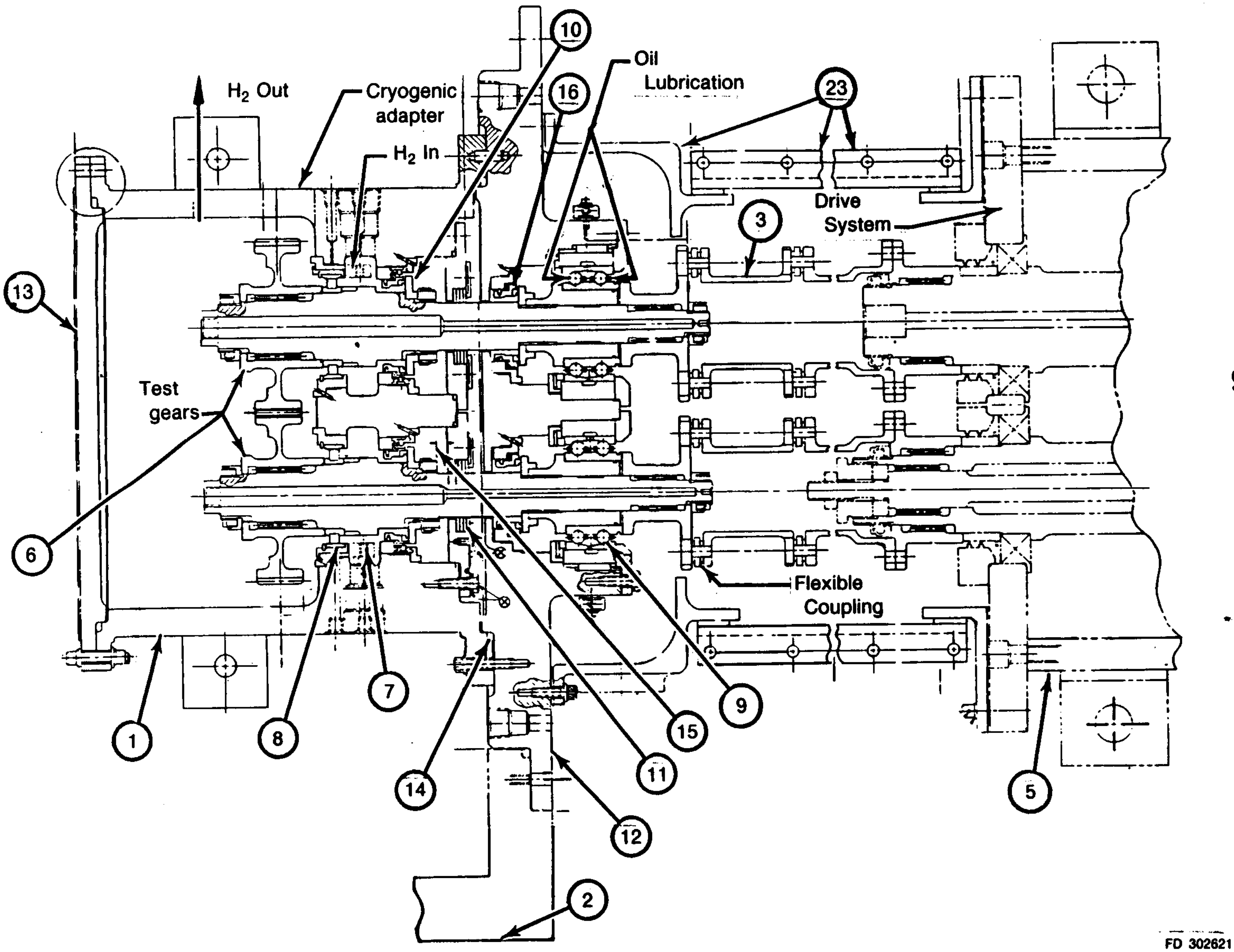

Figure 14. Cryogenic Gear Test Rig

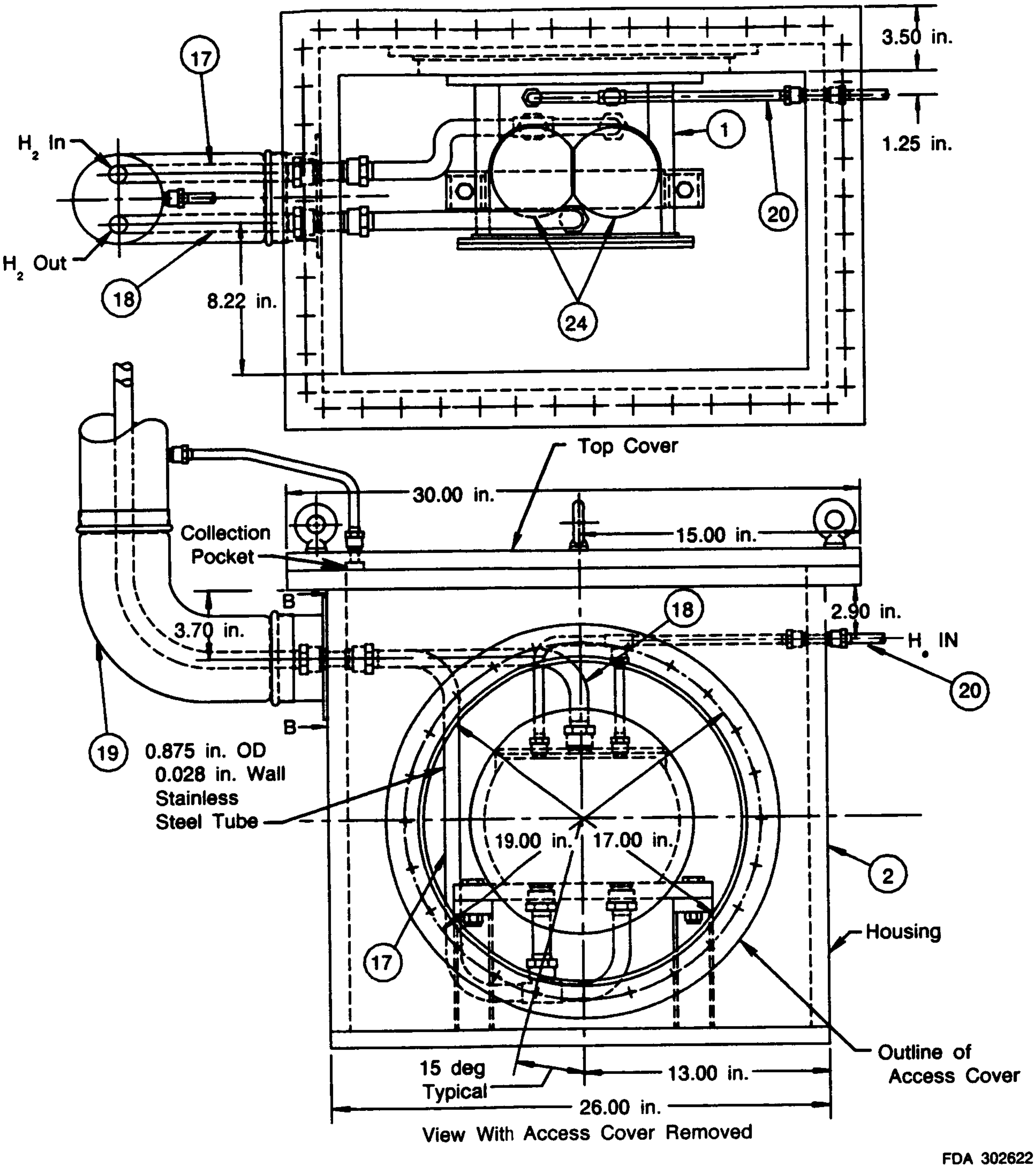

Figure 15. Containment Housing

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The cryogenic media is separated from the remaining sections of the rig by a carbon face seal (10). A series of Teflon™ washer seals (11) is located forward of the carbon seal and the cavity in between acts as a buffer zone. This cavity is pressurized with gaseous helium at a higher pressure than the test media. This ensures that any leakage at the carbon seal will be into the cryogenic housing, thus preventing leakage of the test media, which might be harmful, to other areas of the rig.

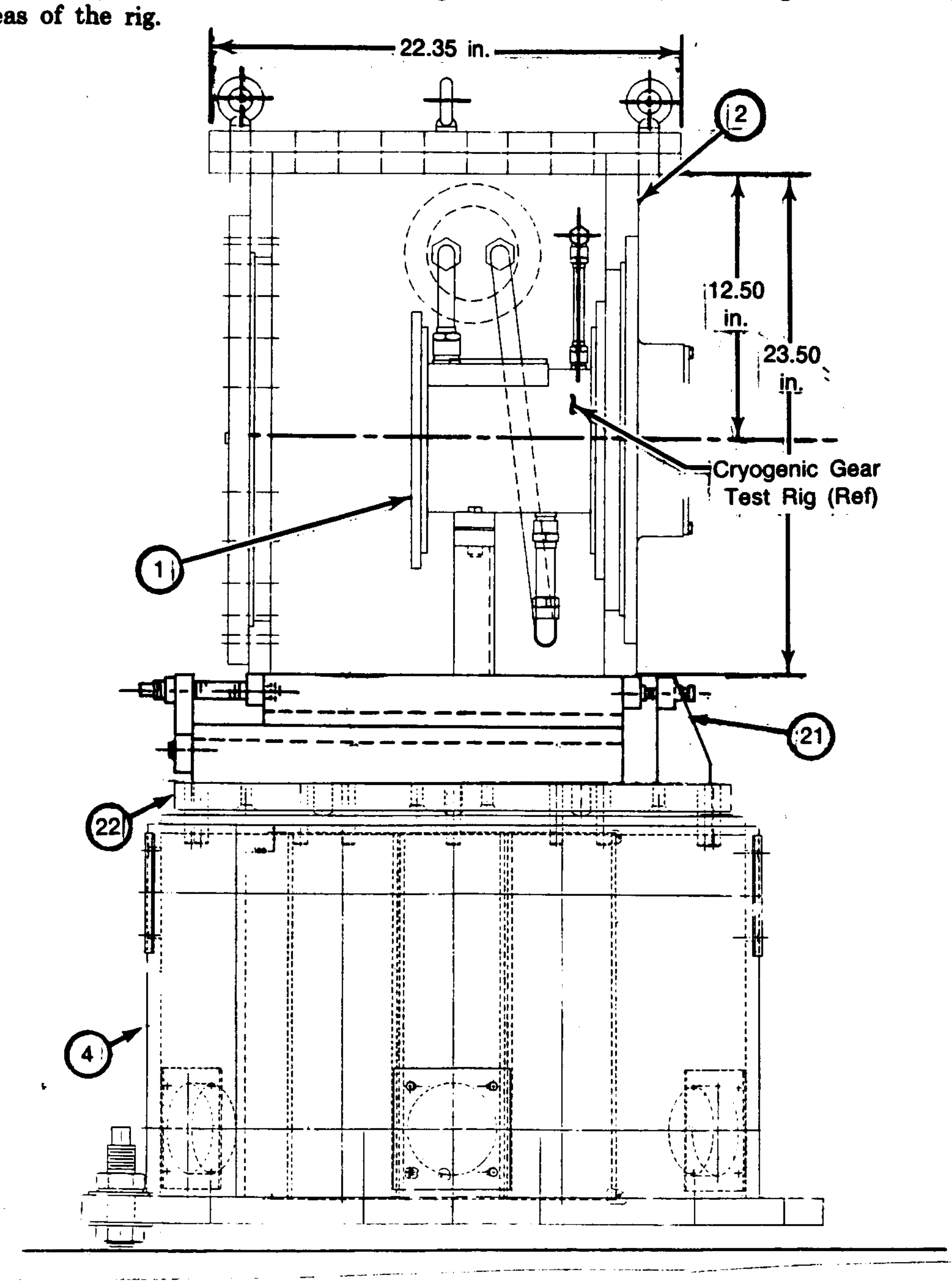

Figure 16. Containment Housing and Support Pedestal

The front face of the gear housing (12) also serves as a wall of the containment housing (2). This arrangement eliminates the thermal mismatch and associated problems which would occur if the gear housing was mounted totally within the containment housing. This design also permits the duplex ball bearings, which are not serviceable without removing the drive couplings and associated protective covers, to be oil lubricated. This substantially increases the life of the bearings. Without this feature, three bearings would be necessary to incorporate the oil lubricated axial support bearing scheme, because two bearings would be required to support the shaft in the gear housing. The roller bearings are serviceable through access cover 13 when the test gears are removed. The roller bearings are spaced slightly further apart in the horizontal plane than the ball bearings at the ambient condition, and this results in the shafts being skewed. This feature ensures that the shafts will be parallel at the cryogenic condition and thus dramatically improves bearing life.

The dual function faceplate design also simplifies the sealing of the gear housing/containment housing junction. Sealing is accomplished by a metal O-ring seal 14 which seals cavity 15 from the inside of the containment housing, and a carbon face seal 16 which seals the external portion of the rig from cavity 15 and also seals the ball bearing lubricating oil from cavity 15. Cavity 15 is vented to the vent stack and, thus, to the outside, so that any helium leakage would be routed to the outside atmosphere. Any test media leakage past the Teflon washer seals (due to a failure of both seal systems) would be routed to the outside atmosphere in a similar manner.

There are two coverplates 24 on top of the gear housing just above the test gears, as depicted in Figure 15. These covers allow access to the gears for an add on lubricating system. This might consist of sacrificial lubricating gears or an injection device for a dry lubricant. The test gears are not actively lubricated.

The cryogenic media, in this case gaseous hydrogen, enters the rig through line 17 , flows through the bearings, then cools the gears, and exits the gearbox through line 18 . Both lines are routed through the wall of the containment housing by means of bulkhead fittings. Once outside of the containment housing, the lines are routed through a vent stack 19 which goes through the roof of the building. The containment housing is designed with a collection pocket at the top which is connected to the vent stack. Thus, in the unlikely event of media leakage from the plumbing lines or the rig, the media would be contained within the containment housing/vent stack system and routed to the outside atmosphere. Because of this system, the media supply tank can be located outside and as far from the test room as necessary.

Helium is supplied to cavity 15 through line 20 . The quantity of helium necessary is only that which leaks past the seals, which is expected to be small. Since the helium is not a fire hazard and the requirements are small, the supply tank may be located in the test room with proper ventilation.

A gas analyzer is located inside of the containment housing. Should any leakage occur from the plumbing or rig, the analyzer will detect it and alert the operator to shut off the media supply.

The containment housing is mounted on a support pedestal 4 . The base of the pedestal is bolted to a concrete plinth in the floor. The containment housing is joined to the support pedestal by a dovetail slide 21 on a leveling plate 22 . These features provide for alignment of the test rig and drive rig prior to the attachment of the flexible couplings. It also allows for disengagement of the rig for servicing without repeating the realignment process.

A transition housing 23 is located between the containment housing and the drive rig. The rear section of the transition housing serves as a collection sump for the ball bearing lubrication oil. The remainder of the housing consists of a split case which acts as a protective covering for the flexible coupling.

D. TECHNICAL DISCUSSION

The design of the cryogenic adapted gear test rig system can be divided into two distinct efforts. These are the design of the cryogenic gear housing, and the containment housing and vent system.

1. Cryogenic Gear Housing — Gears, Bearings, Shafts, and Seals

The basis for loading the test gears is the Four Square Loop principle, which is incorporated into the drive rig hardware. The loop is formed by two 1 to 1 ratio test gears which are connected to two 1 to 1 ratio slave gears by shafts. The cryogenic housing is simply an extension of the normal test gear compartment.

One of the drive shafts is rotated relative to the other by means of the axial displacement of a sliding helically splined collar. Since the test gears are in mesh at the other end of this loop, a torque results. The torque is only within the loop, however, so the torque necessary to drive the shafts is only that required to overcome the friction of the gear meshes and support bearings.

The baseline test gears were designed first and then the shaft was designed to accommodate the test gears. The design rationale was to make the test gears similar to the RL10 Bill-of-Material gears, so as not to introduce any erroneous factors.

The shaft center distance in the gear rig is 4.000 inches and the test gears must be of equal pitch diameter to operate with the rig slave gears. Therefore, the test gear P.D. is set at 4.000 inches. The tooth geometry of the test gears was chosen to be as much like the Bill-of-Material turbopump gears as possible. Since the gears must be the same P.D., the tooth curvature of the Bill-of-Material pinion cannot be duplicated. However, a similar range of sliding velocities can be obtained. The Bill-of-Material idler tooth has a modified tooth profile so one of the test gears is modified to the same extent.

It is necessary to offset the nominal centers of the gear rig aft bearings so that the shafts will grow into alignment when the rig cools down. This will affect the gear operating center distance and, therefore, backlash and contact ratio. The test gear tooth thickness and length were adjusted so that the operating backlash and contact ratio will be the same as those of the RL10.

The test gear face width, rim thickness and web thickness are the same as the Bill-of-Material idler gear. Also provisions for balancing are the same as for the Bill-of-Material parts, as are the balance limits. Spherical identification marks are provided both at the teeth and mount splines to allow for duplicating the gear position after removal. The test gear material for the baseline test is AMS 6265.

AMS 5671 (INCO X-750) was chosen as the material for the drive shafts. The shafts are sized to have an infinite life at a torque load of 2500 inch-pounds, which is the limit of the drive rig. Under these conditions the shaft has a safety factor of 1.37. The shaft critical speed analysis is described in Reference 15 (Refer to Appendix B).

The test gears are mounted on the shafts by means of splines and are piloted at both ends. The end of the shaft adjacent to the gear has an immobilizing feature, which prevents rotation of the shaft while the gear retainer nut is removed. The gears are mounted on the cantilevered portion of the shaft to facilitate removal.

The shafts are supported by a 40 mm roller bearing adjacent to the gear and a 40 mm duplex ball bearing at the other end. The roller bearing is currently used on the RL10 LOX pump shaft,

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and the duplex ball bearing has been used in the F100. The maximum radial load is carried by the roller bearing; the coolant pressure thrust load is carried by the duplex ball bearing. In the event it became desirable to test helical gears in the rig, the resulting thrust load would also be carried by the duplex ball bearing. The duplex ball bearing, which is not easily replaceable, is mounted outside of the cryogenic compartment, so that it may be oil lubricated. Figure 17 shows the mounting of the bearings.

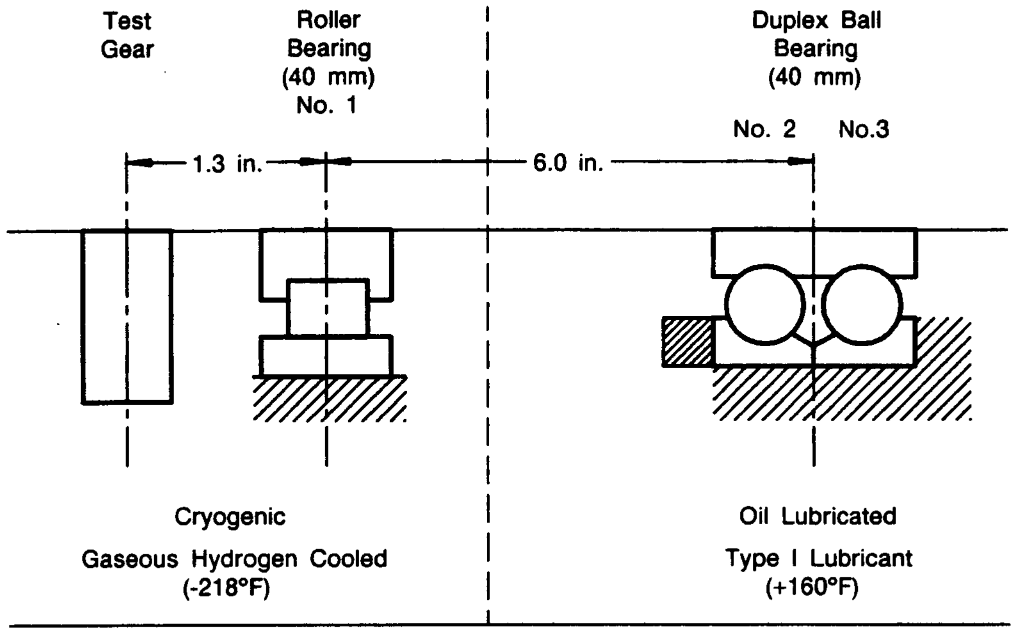

Figure 17. Mounting of Roller and Duplex Ball Bearings

The roller bearing is the more life-limited of the two, since it is unlubricated. Under average clearance conditions, the B1 bearing life is 135 hours at a torque load of 500 inch-pounds. It is not expected that the torque will exceed 500 inch-pounds for most of the testing. However, the B1 life at 1000 inch-pounds would be 19 hours. The roller bearings are cooled by gaseous hydrogen which enters the cavity just forward of the bearings and circulates aft through the bearing rollers. Figures 18 and 19 show the B1 life versus torque for the respective bearings. The bearing life and fit analysis is documented in Reference 16 (Refer to Appendix B).

Access to the roller bearings is achieved by removing the test gears. The bearing outer diameter (OD) race has a loose fit with the housing bushing, and it is removed by a set of puller keys which are incorporated into the bushing. This also permits the insertion of puller fingers, at up to six circumferential locations, which grasp the bearing ID race on the far side, thus allowing removal of the inner diameter (ID) race. The clearance between the shaft shoulder and the bearing bushing is sized to prevent loose rollers from falling into the housing inner cavity should bearing failure occur.

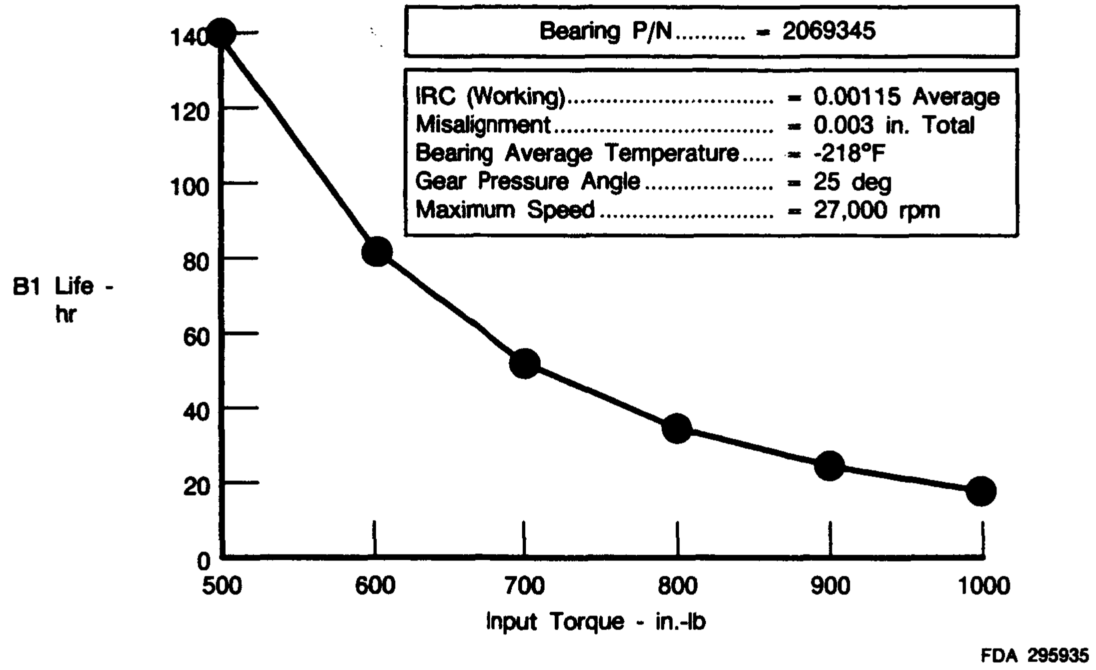

Figure 18. RL10 Cryogenic Gear Test Rig Roller Bearing Life Versus Input Torque

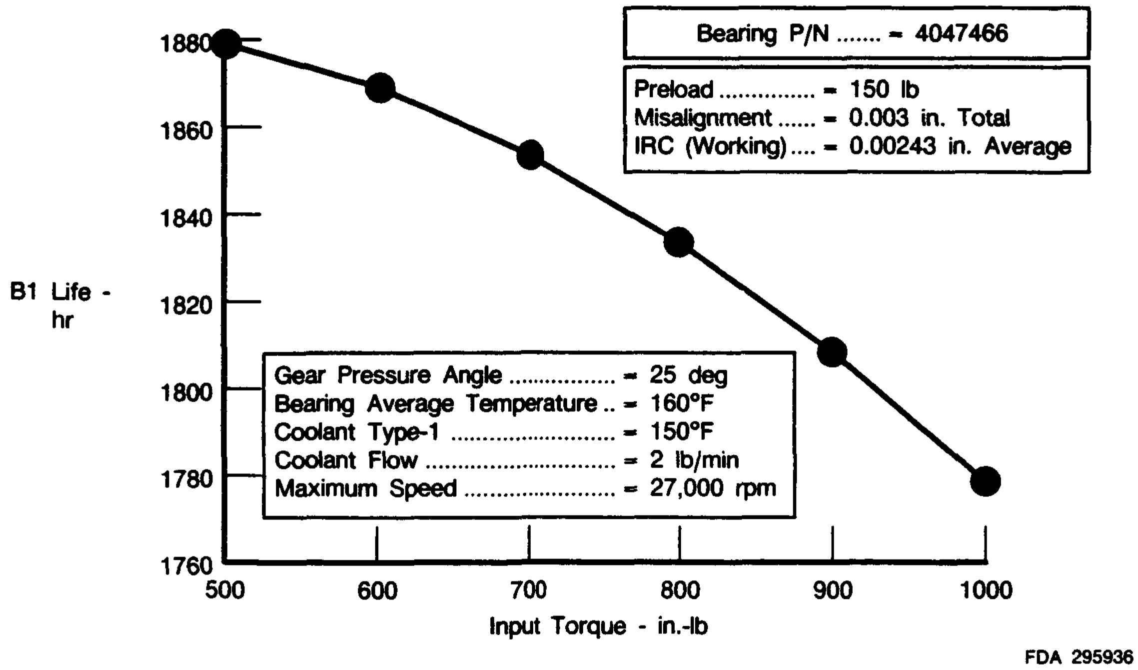

Figure 19. RL10 Cryogenic Gear Test Rig Duplex Ball Bearing Life Versus Input Torque

The duplex ball bearing supports and locates the shafts axially. In addition to the thrust loads on the bearing, which are the result of the axial pressure of the coolant and any resulting loads from helical gears, a bearing preload is provided by a wave washer. The preload prevents skidding of the rollers and maintains an accurate axial location on the shaft. Additional axial loading would result if helical test gears were used rather than spur gears.

It is necessary to incorporate an antirotation scheme at both the roller and ball bearings. The bearing OD races must be initially loose to ensure an adequate internal radial clearance at operating conditions. The ball bearings are antirotated by a rectangular pin at the interface between the two bearings. The pin must be rectangular to prevent a tendency for separation if the bearings rotate. The roller bearings are antirotated by a round pin that fits into a slot in the bearing OD race.

Each of the duplex ball bearings is oil lubricated by two nozzles, one forward and the other aft of the bearings. The used oil is collected at the base of the bearings and is drained into a sump, where it is collected and routed back to the supply system. The lubricant type and condition are described in Figure 19. Figure 20 depicts the lubricant spray nozzles. The roller bearings are unlubricated except for the lead plating contained in the cage.

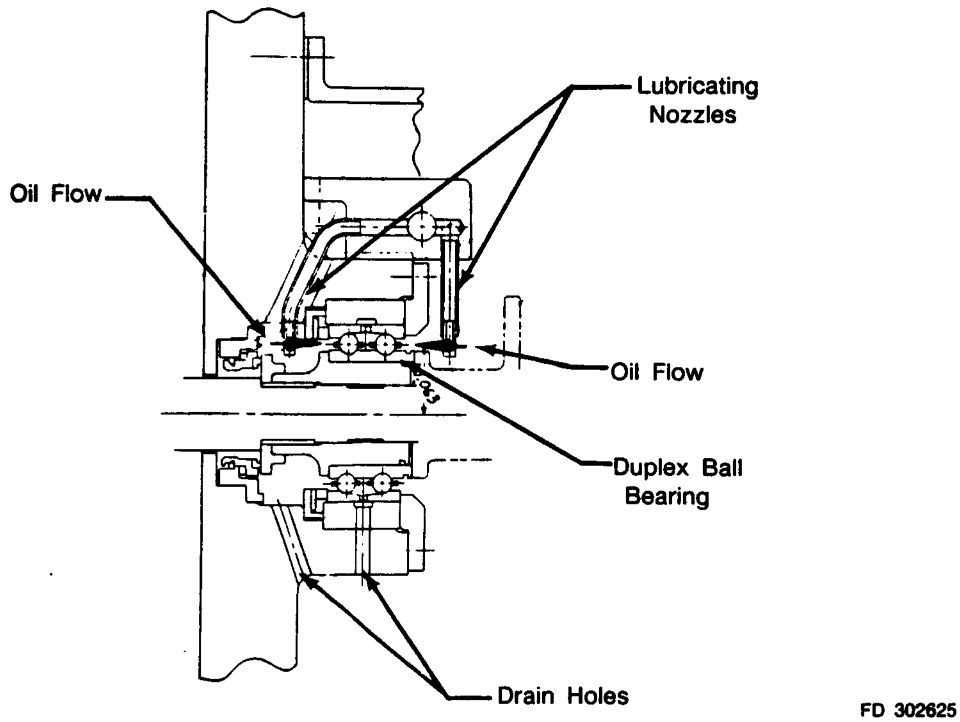

Figure 20. Lubricant Spray Nozzles and Duplex Ball Bearings

The duplex ball bearings may be removed from the front end of the rig. A puller groove is provided for this purpose. However, the flexible couplings must be removed before the bearings can be reached. It was necessary to provide a spacer sleeve on the drive shaft to use the existing duplex ball bearing.

Radial fits for parts mating with the drive shafts were determined through a computerized shell model, with the exception of the bearings, which were analyzed with a special bearing fit program (Reference 16, Appendix B). Fits for the gears, seal plates, and couplings were sized to be slightly tight under worst conditions. Axial fits were also analyzed with the shell model, and retainer nut torques were based on the fit analysis.

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A hole was placed through the center of the shaft to facilitate instrumentation. It may be desirable to strain gage the gears, and in that event the leads would be routed through the shaft. Initially, the hole will not break through the front end of the shaft, since this creates a leakage path for the hydrogen. The hole will be drilled through, once it is determined that strain gages are to be used. The area around the leads will have to be sealed.

Another reason for the hole is to reduce the axial spring rate of the shaft, so more stretch can be obtained when torquing the retainer nuts. The hole size does not appreciably affect the bending strength of the shaft or the critical speed.

Accurate balancing is essential to the successful and continued operation of the rig. Since the test gear shafts are skewed at ambient temperatures and the face seals are spring loaded, it is not possible to balance the rig as an assembled unit. An attempt to do so could result in damage to the bearings and overheating of the seals. Therefore, special consideration has been given to balancing of the detail parts. The test gears are single plane balanced to within 0.001 ounce-inch. The test gear drive shafts are balanced as a set, which includes the aft seal plate and nut, the forward seal plate, and the ball bearing ID sleeve. This set is dynamically balanced to within 0.001 ounce-inch. The flexible coupling is balanced by the vendor, but features are included for the addition of balance weights should additional balancing be necessary.

Leakage of the gaseous hydrogen to the forward wall of the containment housing is prevented by a triple seal arrangement. The rig housing cavity containing the gaseous hydrogen is sealed from the forward part of the rig by a carbon face seal. Forward of the carbon face seal is a series of four Teflon packings. The cavity between the carbon and Teflon seals is pressurized with helium to provide a protective dam against the hydrogen. Forward of the Teflon seals is another carbon face seal, which seals the roller bearing lubricating oil from the cryogenic housing. The cavity between the forward carbon face seal and Teflon seals forms a small void between the cryogenic gear housing and the faceplate, which is also a wall of the containment housing. The mating flanges of the cryogenic housing and faceplate are sealed by a metal O-ring seal. Therefore, any leakage past the Teflon seals is contained in the void. A drain plug is provided in the faceplate at the base of this cavity, so that any oil that might leak past the carbon seal can be drained from the cavity. A vent is provided for any leakage past the Teflon packing.

The rear coverplate of the cryogenic housing is also sealed by a metal O-ring seal. In the event of a failure of this seal, the leaked media would still be held in the containment housing.

The material of the cryogenic housing is AMS 5665 (INCO 600). This choice was made to reduce the thermal mismatch between the cryogenic housing and faceplate, and thus minimize the ambient misalignment of the drive shafts. The material of the housing coverplate is also AMS 5665. The coverplate is sized to withstand the bolt loads induced from seating of the flanges and O-ring seals. The cryogenic housing was evaluated to determine its containment capability in the event of a gear failure. It was assumed that the worst case would occur if the gear split in half. The kinetic energy of this mass, rotating at 28,000 rpm, was compared to energy absorption curves for airfoil containment of steel casings. These curves indicated a required thickness of 0.07 inch.

The above value seemed rather thin for the application in question, so an alternate approach was investigated. The alternate approach assumed that a section of the gear rim would separate from the web, and that this section would be relatively straight, such that it would become a thin projectile. It was assumed that a section equivalent to one-sixth of the circumference of the rim would meet this requirement.

The kinetic energy of this section was calculated, and based on the projectile's behavior as a punch, a thickness was determined through which a hole having the cross section of the projectile could be punched. This thickness was found to be 0.266 inch. Based on this value, a margin of safety of 2.45 results.

a. *Cryogenic Gear Housing — Thermal Analysis*

The thermal analysis for the cryogenic housing, shafts, bearings, and seals is documented in References 17, 18, and 19 (refer to Appendix B). Figure 21 shows the thermal model and breakup. The temperatures of the shafts and surrounding portions of the housing are determined by the heat generation from the bearings and seals.

The carbon face seals are RL10 fuel pump Bill-of-Material seals. In the RL10 these seals are spring loaded to 11.0 pounds for the aft seal and 6.5 pounds for the forward seal. The thermal analysis in Reference 18 (Appendix B) is based on seal spring loads of approximately 6 pounds. If the aft seal spring assembly height were adjusted to obtain a 6 pound load, then the seal would not function properly due to insufficient axial land for the radial seal ring. This ring, shown in Figure 22, prevents axial leakage between the outer seal housing and inner carbon seal housing.

To provide adequate axial land for the radial seal ring, a new inner carbon seal housing would be required, thus necessitating the procurement of a new seal. Seal load may be controlled by properly adjusting the helium pressure on the forward (high pressure) side of the seal.

The maximum temperature of the shaft system is 690°R and occurs at the point of contact of the seal plate and carbon seal, location 165 on Figure 21. The minimum shaft temperature is 220°R and occurs at the aft end of the shaft at location 1.

Since the cryogenic portion of the gear housing has an oval cross section, an additional heat transfer analysis was conducted to determine if a significant circumferential thermal gradient existed. The results of this analysis are documented in Reference 19 (Appendix B). The analysis shows that the maximum gradient of 61 degrees occurs at a location just outboard of the roller bearing sleeve (Reference Figure 23).

b. *Cryogenic Gear Housing — External Features*

The front faceplate of the cryogenic housing also serves as a wall of the containment housing. This has been discussed in the technical description and illustrated in Figures 14 and 16. The faceplate also serves as the housing for the forward carbon seals and duplex ball bearings. The lubrication system for the ball bearings is mounted on the faceplate.

The cryogenic housing and the faceplate are final machined as a set to minimize the effect of tolerances. Bearing and seal mount sleeves are either pinned or keyed to the housings, and the two housings are pinned together so that the proper orientation will be maintained in the event of disassembly.

There are two access covers at the top of the cryogenic housing, just above the gears. These covers permit a quick visual inspection of the gears without removing the main access cover. Another purpose for the covers is to allow the installation of a lubrication system, such as a sacrificial lubricating gear or dry lubricant injection device, for the gears or a cooling system for the gear teeth.

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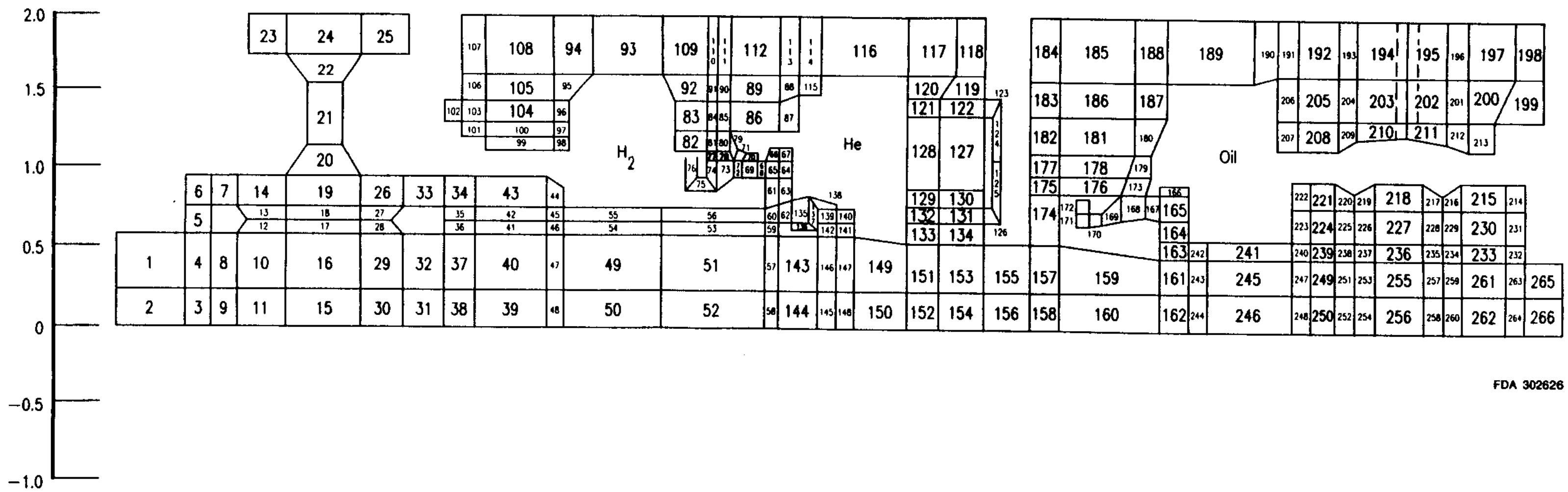

Figure 21. Cryogenic Geur Housing — Thermal Model and Breakup

FOLDOUT FRAME

FOLDOUT FRAME

Plumbing for the cryogenic housing consists of the gaseous hydrogen lines for cooling the rig and the gaseous helium lines for counteracting hydrogen leakage to the forward portions of the rig. Hydrogen enters the cryogenic housing through two lines at the bottom of the housing and exits through one line at the top after circulating through the bearings. The helium enters the housing through the top. There is no exit line since the helium is intended to pressurize the cavity forward of the aft carbon seal, thus preventing hydrogen leakage. Any flow of helium will thus be in the form of leakage in the aft direction, past the aft carbon seal.

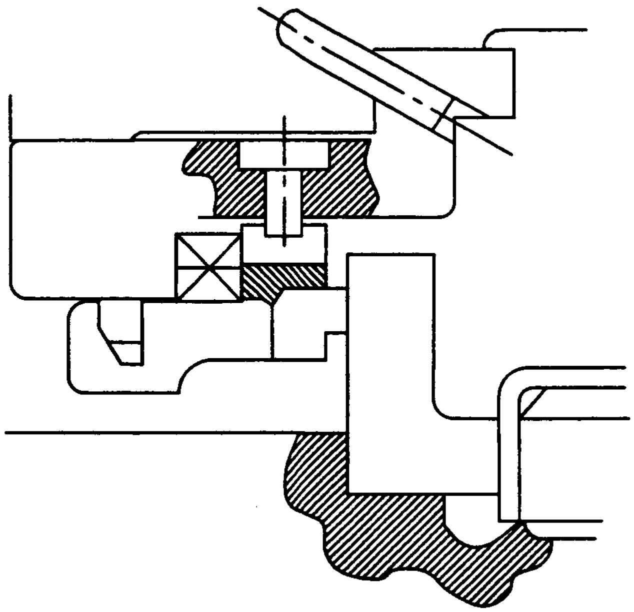

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Figure 22. Radial Seal Ring

The cryogenic rig is connected to the drive rig by a set of flexible couplings. These couplings are the wafer type and the design and coupling are vendor furnished. The coupling is designed for operation at a maximum of 27,000 rpm under a maximum torque load of 1500 inch-pounds.

The couplings must be disassembled while the cryogenic rig is being connected to the drive rig. The forward and aft flanges of the coupling are attached to the drive rig and cryogenic rig, respectively. After the two rigs have been aligned, the spool pieces of the coupling are positioned, and the cryogenic rig is moved forward on its track until both flanges contact the spool pieces. The spool pieces and wafers are then bolted into place.

c. *Cryogenic Housing — Instrumentation*

The cryogenic housing is instrumented for temperatures, pressures, vibrations, and shaft position. Each roller bearing outer race has two thermocouples approximately 90 degrees apart, and each bearing housing bushing has two thermocouples approximately 90 degrees apart. The duplex ball bearings also have two thermocouples approximately 90 degrees apart on the bearing OD race. Also, the inlet and exit coolant temperatures are monitored, as well as the temperature of the case between the roller bearings.

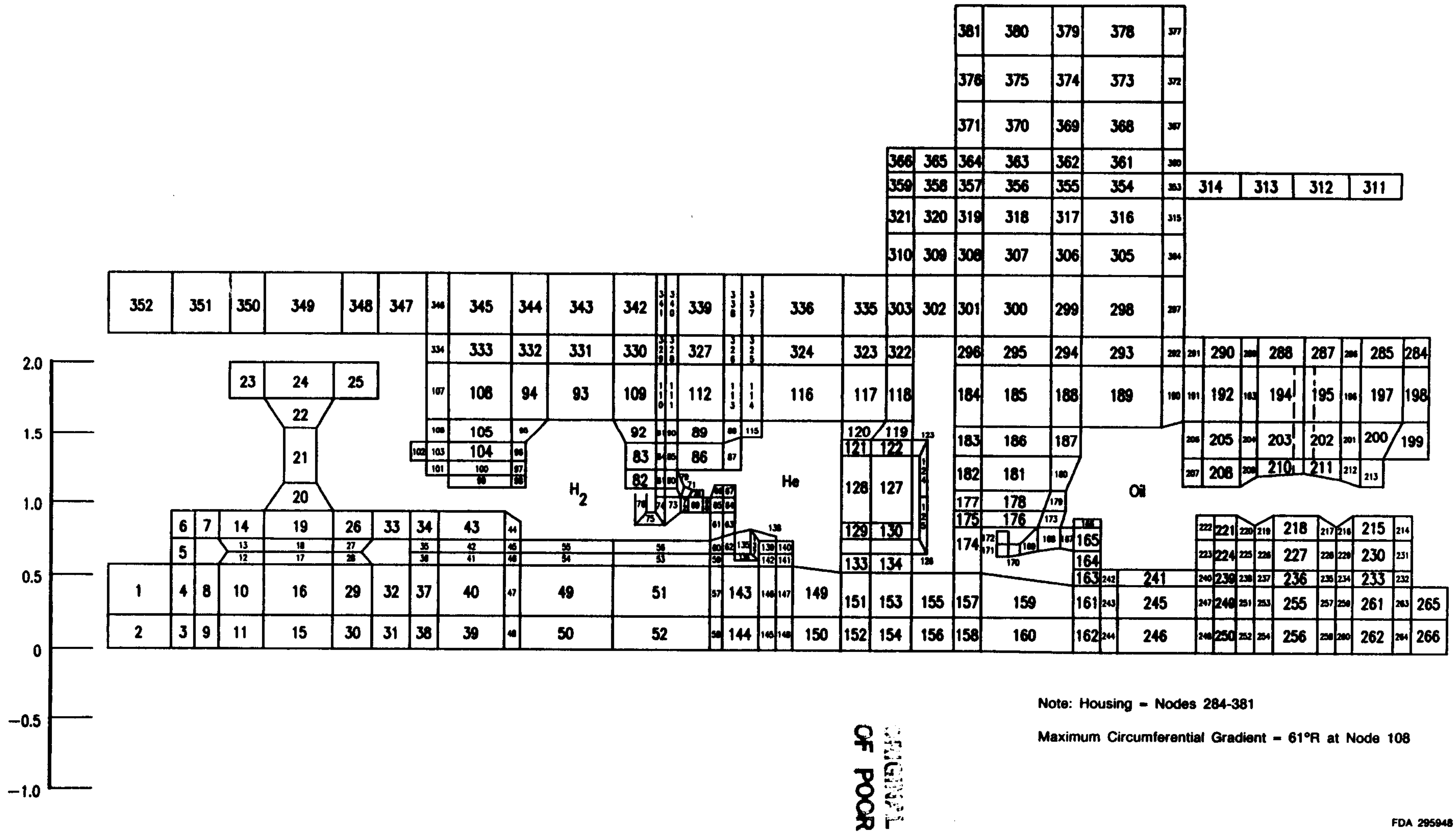

Figure 23. Total Nodal Network With Housing

Coolant inlet and exit pressures are measured at two locations. Each bearing housing bushing has two accelerometers, one horizontal and one vertical, for the measurement of bearing and gear vibrations.

Shaft position is monitored at the forward end of the shaft by means of a horizontal and vertical proximity probe on each shaft. The probes are mounted adjacent to the aft flange of the flexible couplings, which are just forward of the duplex ball bearings. The portion of the coupling adjacent to the proximity probe is nickel plated to reduce electrical runout.

2. Containment Housing — Basic Structure

The containment housing is a box, approximately cubic in shape, of the proper dimensions to enclose the cryogenic gear housing, while leaving space for maintenance of plumbing and removal of gears. The function of the containment housing is to serve as a secondary enclosure in the event of leakage of gas from the cryogenic rig or failure of the rig with an accompanying rupture of the cryogenic housing. Although the chance of occurrence of either event is very remote, the added safety is necessary because the rig is being run indoors.

Aluminum was chosen as the material for the containment housing to optimize the strength-to-weight ratio. A low weight is necessary to prevent low resonant frequencies in the housing support pedestal. The walls of the housing are 1.000 inch thick except for the front wall, which mounts the cryogenic rig faceplate.

The faceplate forms a part of the front wall, and the interface between the faceplate and the containment housing is sealed with a rubber O ring. The rear of the cryogenic housing is supported by two columns which are attached to the bottom of the containment housing. Shims are used to ensure that no load is transferred into the cryogenic housing when the housing is bolted to the columns.

The containment housing has two access panels. One is located at the rear of the housing and is intended primarily to permit access to the cryogenic housing for the removal of gears and for replacement of the roller bearings. The other panel is at the top of the housing and is intended for access to the rig plumbing and the small access covers on the cryogenic housing. Rubber O-ring seals are used with all cover panels.

a. Containment Housing — Support Pedestal and Mount System

The containment housing is mounted on a support pedestal, which is anchored to the floor of the rig test facility. The outer wall of the pedestal is circular and is formed by welding together two 0.750-inch thick members of semicircular cross section. Additional axial stiffness is obtained by three circular columns that are located inside of the pedestal. The pedestal was sized to have a natural frequency above the maximum operating range of the rig.

Access panels are provided around the outer wall of the pedestal, so that it can be filled with a damping medium, such as sand. The panels also facilitate maintenance of the pedestal by permitting access to the inside. The base of the pedestal is a circular plate 1.500 inches thick. This plate is welded to the pedestal outer wall. All pedestal materials are carbon steel.

The pedestal is attached to the concrete floor of the rig test facility by 12 anchor bolts. Since it is necessary to remove and replace the rig system periodically, internally threaded anchor assemblies are mounted in the concrete floor, flush with its surface. The anchor bolts are installed in the anchor assemblies and the pedestal baseplate is secured by nuts on both sides. This feature provides an initial leveling of the pedestal and rig. Once the pedestal has been leveled and the nuts secured, the space between the pedestal baseplate and concrete floor is filled with grout.

The rig containment housing is mounted to the support pedestal by means of a Gilman slide assembly. Once the containment housing and cryogenic rig have been properly aligned, the Gilman slide permits the housing and rig to be disengaged from the drive rig and the two moved apart. This is necessary for servicing of the rig. The rig can then be reconnected to the drive rig without additional alignment and adjustment. A screw is provided to aid in the backward and forward movement of the housing and rig.

Adjustment of the rig in the horizontal plane is accomplished by leveling screws in the baseplate of the Gilman slide. The tracks of the Gilman slide are bolted to the baseplate, and the baseplate is leveled relative to the top plate of the pedestal. When proper adjustment in the horizontal plane has been achieved, the Gilman slide baseplate is bolted securely to the top plate of the pedestal.

Vertical adjustment is accomplished by screws which position the Gilman slide baseplate prior to drilling holes for the hold-down bolts and again prior to the tightening of the hold-down bolts.

b. Containment Housing — External Features

Plumbing lines to the containment housing are required for the coolant, purge gases, and pressurization gases. The coolant, in this case gaseous hydrogen, requires an inlet and exit line. The coolant lines are routed from the cryogenic housing to bulkhead fittings in the wall of the containment housing. Fittings on the outside wall of the housing attach the remainder of the lines. The two coolant lines are enclosed in a vent pipe, which extends up through the roof of the test facility. The exit line may be vented to the atmosphere or connected to a burn stack. The inlet line is routed to the source of the cooling media once it is outside of the test facility. The vent stack serves to contain any leaked media in the event of a plumbing failure. There is a collection pocket in the top plate of the containment housing. Any gases leaking from the cryogenic housing would rise to this high point. The collection pocket is connected to the vent stack by a tube which contains a valve for controlling flow to the vent stack. The valve provides for purging or pressurization of the containment housing as needed.

The helium line enters the containment housing through a bulkhead fitting in the side. It is not necessary to route the helium line through the vent stack, since it does not present a potential environmental hazard with adequate ventilation. Only an inlet line is needed, since the helium is used to pressurize against hydrogen leakage, and the only flow will be that which leaks past the seals.

c. Containment Housing — Instrumentation

The containment housing will be fitted with a gas analyzer, which will detect any leakage of hydrogen from the cryogenic housing. No other instrumentation has been specified for the containment housing, but any requirements identified in the future coud be easily accommodated. In the event that instrumentation lines from the cryogenic housing must pass through the containment housing, care must be exercised to ensure the joints are leakproof.

REFERENCES

1. Computer Program P59704, "Spur Gear and Involute Spline Tooth Design and Analysis," R. F. Zogbaum, 1983.

2. Spotts, M. F., "Estimating Dynamic Loads on Gear Teeth," *Machine Design,* July 12, 1984.

3. Wang, C. C., "An Analytical Evaluation of Gear Dynamic Factors Based on Rigid Body Dynamics," ASME Paper 84-DET-53, 1984.

4. Dudley, Darle W., *Handbook of Practical Gear Design,* McGraw Hill Book Co., New York, 1984.

5. Job 84-711-70592A, "7.5K Orbit Transfer Vehicle (OTV) Redesign Review," K. C. Knight, 1985.

6. Terauchi, Y., Nadano, H., and Kohno, M., "Effect of MoS_2 Films on Scoring Resistance of Gears," ASME Paper 84-DET-59, 1984.

7. El-Bayoumy, L. E., Akin, L. S., and Townsend, D. P., "An Investigation of the Transient Thermal Analysis of Spur Gears," ASME Paper 84-DET-92, 1984.

8. Memo, G. H. Duncan to J. R. Brown, "Computerized Mathematical Models for Gear Technology Program," November 20, 1984.

9. Job 84-711-70613A, "Cryogenic Gear Test Rig Adapter," G. H. Duncan, 1985.

10. CAD CAM Dataset, DA-DWH-3 Gears, RL10 Gear Geometry.

11. Nastran Dataset, E097347 RL10FOUT Gear1A, Nastran Spur Gear Model.

12. Job 85-711-A0600L, "RL10 Baseline Test Gears for Cryogenic Rig," G. H. Duncan, February 27, 1985.

13. Butner, M. F., "Propellant Lubrication Properties Investigation," Defense Technical Information Center, Report AD 259143, 1961.

14. Wisander, D. W. and R. L. Johnson, "Wear and Friction in Liquid Nitrogen With Austenitic Stainless Steel Having Various Surface Coatings," Advances in Cryogenic Engineering, Vol. 4, Plenum Press, New York, 1960, p. 71.

15. Memo, F.E. Senesac to G.H. Duncan, "Critical Speed Analysis — Cryogenic Gear Test Rig," June 12, 1985.

16. Memo, Y.E. Eren to G. Duncan, "RL10 Gear Test Rig Bearings," June 3, 1985.

17. Memo, T.R. Swartwout to Yul Eren, "RL10 Gear Rig Shaft Temperatures Are Dependent on Bearing And Seal Heat Generation," April 23, 1985.

18. Memo, T.R. Swartwout to Yul Eren, "RL10 Gear Test Rig Shaft Temperatures Revised Using F100 Test Data," May 17, 1985.

19. Memo, J.T. Elliot to Dick Duncan, "RL10 Gear Test Rig Housing Can have a 61°F Circumferential Temperature Gradient," July 17, 1985.

APPENDIX A
TECHNICAL DATABASE FOR CRYOGENIC GEARS

1. Job 82-711-70441A, "Gear Technology Improvement Study for Advanced Rocket Engines," R. L. Carlson, August 2, 1982.

2. Jobs 80-711-70708A, 70709A and 70710A, "Orbit Transfer Vehicle (OTV) Advanced Expander Cycle Engine Point Design Study," A. M. Palgon, December 4, 1980.

3. Shoemaker, Robert H., "Advances in Low Temperature Nitriding," SAE Paper No. 750195, 1975.

4. Seagler, D. L., "Separation of Gear Teeth in Approach and Recess, and the Likelihood of Corner Contact," ASLE Transactions, Vol. 19, No. 2, April 1976.

5. Wilde, R. A., "Failure in Gears and Related Machine Components," NBS Special Publication 423, Mechanical Failure: Definition of the Problem, USGPO, 1976.

6. Ku, P. M., "Gear Failure Modes — Importance of Lubrication and Mechanics," ASLE Reprint No. 75AM-SA-1.

7. Grosberg, J., "A Critical Review of Gear Scoring Criteria," Wear, Vol. 43, No. 1, May 1977.

8. Bartz, W. J. and Krueger, V., "Pitting Fatigue of Gears — Some Ideas on Appearance, Mechanism and Lubricant Influence," Tribology International, October, 1973.

9. Bartz, W. J. and Krueger, V., "Influence of Lubricants on the Pitting Fatigue of Gears," Wear, Vol. 35, No. 2, December, 1973.

10. Drago, R. J., *Fundamentals of Gear Design,* Drive Systems Technology, Glen Mills, Pennsylvania, 1982.

11. Hartman, M. A., "Gears for Outer Space," Machinery, August, 1959.

12. *Source Book on Gear Design, Technology and Performance,* ASM, 1980.

13. Dudley, Darle W., *Handbook of Practical Gear Design,* McGraw Hill Book Co., New York, 1984.

14. Dudley, Darle W., *Handbook of Practical Gear Design,* McGraw Hill Book Co., New York, 1954.

15. Computer Program P59704, "Spur Gear and Involute Spline Tooth Design and Analysis," R. F. Zogbaum, 1983.

16. Spotts, M. F., "Estimating Dynamic Loads on Gear Teeth," Machine Design, July 22, 1984.

17. Tobler, R. L., "A Review of Antifriction Materials and Design for Cryogenic Environments," Proceedings of Cryogenic Engineering Conference, August, 1979.

18. Wisander, D. W. and Johnson, R. L., "Wear and Friction in Liquid Nitrogen With Austenitic Stainless Steel Having Various Surface Coatings," Advances in Cryogenic Engineering, Vol. 4, Plenum Press, New York, 1960, p. 71.

19. Wisander, D. W. and Johnson, R. L., "Friction and Wear of Nine Selected Polymers With Various Fillers in Liquid Hydrogen," National Aeronautics and Space Administration, Lewis Research Center, Cleveland, Ohio, 1969.

20. Sliney, H. E., "Plasma-Sprayed, Self-Lubricated Coatings for Use from Cryogenic Temperatures to 870°C," NASA-TM-X-71198, National Aeronautics and Space Administration, Lewis Research Center, Cleveland, Ohio, 1975.

21. Breen, D. H., "Fundamental Aspects of Gear Strength Requirements," AGMA Publication 229.17, November, 1974.

22. *AGMA Standard Design Procedure for Aircraft Engine and Power Take-Off Spur and Helical Gears,* American Gear Manufacturers Association, Arlington, Virginia.

23. Coffin, Jr., L. F., "A Study of the Sliding of Metals With Particular Reference to Atmosphere," *Lubrication Engineering,* p 50, January-February, 1956.

24. Butner, M. F., "Propellant Lubrication Properties Investigation," Defense Technical Information Center, Report AD 259143, 1961.

25. Afanasyev, V. G. and Karpinos, D. M., "Antifriction Properties of Boron Nitride During Dry Friction in Gaseous Media and at Low Temperatures," FTD HT 67 348, AD 673 802, Foreign Technology Division, Wright-Patterson AFB, Ohio, 1967.

26. Wisander, D. W. and Johnson, R. L., "A Solid Film Lubricant Composition for Use at High Sliding Velocities in Liquid Nitrogen," ASLE Annual Meeting, Cincinnati, Ohio, April, 1960.

27. Terauchi, Y., Nadano, H., and Kohno, M., "Effect of MoS_2 Films on Scoring Resistance of Gears," ASME Paper 84-DET-59, 1984

28. El-Bayoumy, L. E., Akin, L. S., and Townsend, D. P. "An Investigation of the Transient Thermal Analysis of Spur Gears," ASME Paper 84-DET-92, 1984.

29. Wang, C. C., "On Analytical Evaluation of Gear Dynamic Factors Based on Rigid Body Dynamics," ASME Paper 84-DET-53, 1984.

30. Bowen, P. H., "Lubricant of Bearings and Gears in Aerospace Environmental Facilities," Defense Technical Information Center, Report AD 411430, 1973.

1. *Job 82-711-70441A, "Gear Technology Improvement Study for Advanced Rocket Engines," R. L. Carlson, August 2, 1982.*

Summary — The original gear technology study revealed some test candidates with potential merit, but made no specific recommendations. The study deals primarily with test candidates, failure modes, and also:

- Identified current gear operating conditions and gear parameters
- Discussed various failure modes and factors affecting gear life
- Discussed some potential fixes and operating goals for advanced engines, i.e., materials, coatings, treatments, geometries.

Comments — This study, which has high value for cryogenic gear program, provides background information for the current study.

2. *Jobs 82-711-70708A, 70709A and 70710A, "Orbit Transfer Vehicle (OTV) Advanced Expander Cycle Engine Point Design Study," A. M. Palgon, December 4, 1980.*

Summary — The original OTV study job defines operating conditions and life requirements for turbopump gears and defines the gear packaging arrangement.

Comments — This study has high value for cryogenic gear program and provides background information for current OTV gears.

3. *Shoemaker, Robert H., "Advance in Low Temperature Nitriding," SAE Paper No. 750195, 1975.*

Summary — This paper discusses low temperature (1077°F) liquid nitriding as a means of increasing wear and endurance properties. The paper claims liquid nitriding has high endurance values, wear resistance, and corrosion resistance, and is superior to caburizing and hardening, gas nitriding, and chrome plating. The average case depth is only 0.0005 in.

Comments — Moderate value for cryogenic gears. Although nitriding may provide more wear resistant surfaces than carburizing, sulfurizing, etc., the increase is not of large magnitude, and the resulting case depth is only 0.0005 in. There are processes that offer more promising results.

4. *Seagler, D. L., "Separation of Gear Teeth in Approach and Recess, and the Likelihood of Corner Contact," ASLE Transaction, Vol. 19, No. 2, April, 1976.*

Summary — This publication discusses tooth separation behavior, how it can relate to scoring, etc. through tip loading and how changing various parameters affects amount of separation.

Comments — Moderate value for cryogenic gears. This is more valuable for fine tuning geometry, since it points out that tooth separation can still be very small even after relatively large increments of gear rotation, resulting in increased potential for scoring.

5. *Wilde, R. A., "Failure in Gears and Related Machine Components," NBS Special Publication 423, Mechanical Failure: Definition of the Problem, USGPO, 1976.*

Summary — This publication discusses various failure modes in oil lubricated gears, such as tooth breakage, wear, scoring, and pitting; defines failure mechanism and discusses which variables are involved; and discusses effects of surface roughness on pitting.

Comments — High value for cryogenic gears. Although failure modes discussed are related to oil lubricated gears, the mechanics of deterioration and failure are similar to those of cryogenic gears, after the lubrication has failed.

6. *K[illegible], P. M., "Gear Failure Modes — Importance of Lubrication and Mechanics," ASLE Reprint No. 75AM-SA-1.*

Summary — This publication is a broad discussion of gear lubrication and failure modes. It defines and discusses major modes of gear-tooth failure, such as scoring, pitting, plastic flow, and breakage. It discusses EHD and boundary lubrication of gear teeth and effect of gear displacement and motion on lubrication, and lists extensive references related to gear lubrication and durability.

Comments — High value for cryogenic gears. This is a good basic discussion of gear mechanics. Discussion is also related to oil lubricated gears but the general discussion of wear modes and friction also applies to nonlubricated gears.

7. *Grosberg, J., "A Critical Review of Gear Scoring Criteria," Wear, Vol. 43, No. 1, May, 1977.*

Summary — Examines criteria for scoring probability. Several criteria relate load and sliding velocity while others discuss scoring criteria as related to temperature and power dissipated by friction. Scoring factors calculated by these criteria cannot be extrapolated outside of verified range.

Comments — Moderate value for cryogenic gears. There is not much in the way of experimental results to support the scoring criterion which attempt to relate scoring to a function of load and velocity.

8. *Bartz, W. J. and Krueger, V., "Pitting Fatigue of Gears — Some Ideas on Appearance, Mechanism and Lubricant Influence," Tribology International, October, 1973.*

Summary — This publication discusses pitting, appearance of pitted areas, mechanisms of pitting, factors which might influence pitting and influence of lubricants in reducing Hertzian pressure and pitting fatigue failures.

Comments — High value for cryogenic gears. The intent of this paper is to evaluate the influence of lubricants on pitting, but the value for cryogenic gears is that the mechanism of pitting are discussed in detail and the appearance of pitting is described and supported by photographs.

9. *Bartz, W. J. and Krueger, V., "Influence of Lubricants on the Pitting Fatigue of Gears," Wear, Vol. 35, No. 2, December, 1973.*

Summary — This paper discusses pitting, appearance of pitted areas, and influence of lubricant on pitting fatigue. Very similar to reference 8.

Comments — High value for cryogenic gears.

10. *Drago, R. J., Fundamentals of Gear Design, Drive Systems Technology, Glen Mills, Pennsylvania, 1982.*

Summary — Textbook and gear design gives thorough development of equations for rolling and sliding motion of spur gear teeth.

Comments — High value for cryogenic gears. This is a general gear textbook that develops the equations for spur gear sliding velocity, flash temperature, and other parameters.

11. *Hartman, M. A., "Gears for Outer Space," Machinery, August, 1959.*

Summary — This publication provides a general description of geometry, operating conditions, and requirements of lubricated rocket engine gears, and rudimentary discussion of primary gear stresses and evaluation procedures. Optimum values for spur gear design variables, such as tooth geometry and modification, hardness, bearing geometry, and lubrication are discussed.

Comments — Moderate value for cryogenic gears. The discussion does not address nonlubricated gears nor does it go very deeply into gear mechanics. Material discussed is treated in more depth in the gear textbooks.

12. *Source Book on Gear Design, Technology and Performance, ASM, 1980.*

Summary — Contains a collection of articles related to the various aspects of gear design, analysis, lubrication, manufacturing, etc. Several of the included articles are listed as separate references in this database.

Comments — High value for cryogenic gears. This collection of papers does not address cryogenic gears but contains useful information on the general mechanics of gears. Some of the papers contained are used as other references in the technical database.

13. *Dudley, Darle W., Handbook of Practical Gear Design, McGraw Hill Book Co., New York, 1984.*

Summary — Textbook on gear design manufacture and operation is a complete gear handbook which covers all aspects of gear design and fabrication and includes up to date developments in gear technology.

Comments — High value for cryogenic gears. This complete gear textbook does not address cryogenic gears, but is valuable as a general reference on gears. This is the latest edition of the original publication.

14. *Dudley, Darle W., Handbook of Practical Gear Design, McGraw Hill Book Co., New York, 1954.*

Summary — Textbook covering the same material as reference (13) but does not include new developments of the past 30 years.

Comments — High value for cryogenic gears. This is the original publication.

15. *Computer Program P59704, Spur Gear and Involute Spline Tooth Design and Analysis, R. F. Zogbaum, 1983.*

Summary — Computer program for mesh design and analysis of spur gears. Calculates gear performance parameters, such as beam and contact stress, sliding velocity, tooth loads, and flash temperature. Also calculates tooth profile coordinates and plots tooth mesh profile, writes detail drawing data, and calculates tooth profile modification requirements.

Comments — High value for cryogenic gears. Provides an easy means for calculating various gear performance parameters. A good tool for fine tuning geometry.

16. *Spotts, M. F., "Estimating Dynamic Loads on Gear Teeth," Machine Design, July, 1984.*

Summary — Presents a method for estimating dynamic loads and gear teeth. Loads due to profile errors are estimated from a mass/spring model that considers operating speed, tooth stiffness, and rotating mass of the meshing gears.

Comments — Moderate value for cryogenic gears. Model does not take into consideration individual tooth error or dynamic deflections of teeth.

17. *Tobler, R. L., "A Review of Antifriction Materials and Design for Cryogenic Environments," Proceedings of Cryogenic Engineering Conference, August, 1979.*

Summary — This publication is a review of existing research. It includes a brief discussion of preliminary design considerations for reduced friction in cryogenic applications. Frictional characteristics of bare metals, diffusion coatings, platings, laminar solids, and polymers are discussed.

Comments — High value for cryogenic gears. A good summary of the available lubricants for cryogenic application. It does not contain a very wide range of experimental data.

18. *Wisander, D. W. and Johnson, R. L., "Wear and Friction in Liquid Nitrogen With Austenitic Stainless Steel Having Various Surface Coatings," Advances in Cryogenic Engineering, Vol. 4, Plenum Press, New York, 1960, p. 71.*

Summary — This publication reviews wear and friction properties of several surface treatments for type-304 stainless steel in liquid nitrogen. It includes diffusion coatings with nitrogen, sulfur, chromium and chromium electroplating, and also MoS_2, PTFE, and PTFCE. Surface speed was 2300 ft/min. LN_2 improved friction and wear properties by cooling the slider surfaces.

Sulfurizing was the only one of the diffused coatings to produce a significant reduction in friction or wear. Chromized and chromium plated surfaces gave poor results. Resin bonded MoS_2 was effective for only about 1 minute. It was suggested that MoS_2, thickness was not optimum for cyrogenic use. Fused PTFE and PTFCE gave effective lubrication for the entire one hour run. Coating thickness (0.010 was maximum studied) was a primary factor in PFTE/PTFCE coatings. It should be noted that the rider specimen (3/16 in. radius hemisphere) load was only 1000g.

Comments — High value for cryogenic gears. Gives experimental results of various antifriction materials in cryogenic environments. This is good background information, although the materials tested either showed no significant promise or were not suitable at gear load magnitudes.

19. *Wisander, D. W. and Johnson, R. L., "Friction and Wear of Nine Selected Polymers With Various Fillers in Liquid Hydrogen," National Aeronautics and Space Administration, Lewis Research Center, Cleveland, Ohio, 1969.*

Summary — This publication reviews wear and friction properties of various polymers on 17-4 PH stainless steel in liquid hydrogen. Data was obtained at a sliding velocity of 2300 ft/min with a 1000g load on a 3/16 in. radius hemispherical rider specimen.

Materials were evaluated which might have less tendency toward cold flow and thermal contraction than PTFE. Tables of mechanical properties of materials tested are included in report. A comparison of polymers without fillers and polymers with fillers was made. Results showed that addition of PTFE could reduce polymer wear.

Comments — High value for cryogenic gears. It contains extensive test data on polymers. Good background information, but as in other tests these materials will probably not withstand gear type loads. Mechanical properties of test materials are included.

20. *Sliney, H. E., "Plasma-Sprayed, Self-Lubricated Coatings for Use from Cryogenic Temperatures to 870°C," NASA-TM-X-71198, National Aeronautics and Space Administration, Lewis Research Center, Cleveland, Ohio, 1975.*

Summary — This publication reviews NASA Lube PS101 on self-aligning, plain cyclindrical bearings. PS101 is a plasma sprayed lubricant containing silver, nichrome, calcium fluoride, and an oxidation protective glass. Tests were conducted at loads up to 5000 psi in nitrogen gas at −160°F, in vacuum at room temperature and in air at temperatures up to 1600°F. The oscillation rate of the bearing was 1 cycle/sec through a 15 degree arc. Frictional coefficients are stated to be less than 0.25 with low corresponding wear rates.

Comments — Moderate value for cryogenic gears. Application appears to be more suited for relatively low load and speed conditions.

21. *Breen, D. H., "Fundamental Aspects of Gear Strength Requirements," AGMA Publication 229.17, November, 1974.*

Summary — Brief discussion of fundamentals of durability in carburized gears.

Comments — Moderate value for cryogenic gears. Discusses fatigue of carburized gears.

22. *AGMA Standard Design Procedure for Aircraft Engine and Power Take-Off Spur and Helical Gears, American Gear Manufactures Association, Arlington, Virginia.*

Summary — A design manual for spur and helical gears, which contains formulas to calculate stress, load, performance indexes, and performance parameters. It also addresses tolerances, allowances, materials, and treatments.

Comments — High value for cryogenic gears. It is useful on the basis of basic design information.

23. *Coffin, Jr., L. F., "A Study of the Sliding of Metals With Particular Reference to Atmosphere," Lubrication Engineering, January-February 1956, p. 50.*

Summary — This publication provides a detailed discussion on the wear mechanisms and sliding characteristics of a number of elemental couples. Frictional force and surface damage are investigated relative to the alloying tendency of the metals. The effect of gaseous atmosphere on local seizure and sliding characteristics was investigated and determined to be profound.

It was determined that sliding without local welding occurred for couples which did not alloy, when tested in an inert environment.

Comments — High value for cryogenic gears. This report provides the basics of wear mechanisms and supporting experimental data for a number of material combinations. It is very helpful in understanding the mechanics of wear of sliding materials.

24. *Butner, M. F., "Propellant Lubrication Properties Investigation," Defense Technical Information Center, Report AD 259143, 1961.*

Summary — This report reviews friction and wear characteristics of ball bearings and gears in various propellents including LH_2, LO_2, LN_2, Hydrazine and RP-1. Gear materials tested included AISI 9310, Nitralloy and AISI 440-C, among others.

An investigation was performed to determine the coefficients of friction of several rider materials on a 440-C test plate. Tests concluded that 9310 gave best results, in LH_2, at loads up to 1000 lb/in. for durations of one hour or less.

Comments — Very high value for cryogenic gears. This is the only material which actually addresses the operation of gears in cryogenic H_2. Test loads were high but duration was short and speeds were relatively slow.

25. *Afanasyev, V. F. and Karpinos, D. M., "Antifriction Properties of Boron Nitride During Dry Friction in Gaseous Media and at Low Temperatures," FTD HT 67, 348, AD 673 802, Foreign Technology Division, Wright-Patterson AFB, Ohio, 1967.*

Summary — Friction and wear characteristics of boron nitride at low temperatures in air, AR, N_2, and He are reviewed. The lubricant was studied at temperatures ranging from 300°K to 73°K. Data was obtained for boron nitride specimens running against a stainless steel disk nitrided to a hardness of Rc 62. Tests were run at pressures up to 5.0 MN/M^2 (725 psi) and at friction velocities up to 10.5 m/sec (2067.0 ft./min).

The study concludes that wear and friction were considerably higher in the N_2 and inert gases than in air. This is thought to be due to the formation of film on the specimen surface when exposed to air. Wear began to increase rapidly beyond a pressure of 3.0 to 4.0 MN/M^2 and was also found to increase as a temperature was reduced. Specimen temperature was found to increase substantially for specimens operating in gas at velocities above 3.0 m/sec.

Comments — Limited value for cryogenic gears. The study shows that the coefficient of friction increases as temperature decreases and is rather high at 73K. The study does not specify at what conditions these values were obtained. The pressures applied to the specimens are also very low compared to those encountered in gear design. The report does not suggest how boron nitride might be applied for hardware applications.

26. *Wisander, D. W. and Johnson, R. L., "A Solid Film Lubricant Composition for Use at High Sliding Velocities in Liquid Nitrogen," ASLE Annual Meeting, Cincinnati, Ohio, April, 1960.*

Summary — This publication reviews friction and wear characteristics of an experimental solid film lubricant containing PTFE, an epoxy resin, and lithium alumina - silicate. The lubricant was studied in liquid nitrogen at −320°F. Data was obtained at sliding velocities up 16,000 ft/min. Tests were made with 3/16 in. radius hemispherical riders on flat plates with a load of 1000g.

The experimental lubricant was formulated to have a coefficient of expansion approximating that of austenitic stainless steel. PTFE particles tend to spall from metal surfaces because of differences in thermal expansion. The lubricant was applied as a paste, cured at 200°F and finished machined to a thickness of 0.005 in.

The endurance of the material was said to be superior to other coatings containing PTFE except for 0.010 in. thick fused PTFE. It was found that the best film adherence was obtained with the coatings of least thickness, but best endurance in sliding was with coatings having greatest thickness for PTFE coatings other than the experimental composition.

The experimental coating failed at 9000 ft/min with a 304 stainless rider. However, with a copper rider, specimens were run up to 16,000 ft/min, suggesting that heat dissipation capability was a major factor.

It was determined that any hydrodynamic component due to the liquid nitrogen was insufficient to be considered as important.

Comments — High value for cryogenic gears. The materials were tested at high sliding velocities, but loads were low compared to those of gear applications. Usage in gear applications may also be limited because the coating must be finish machined.

27. *Terauchi, Y., Nadano, H., and Kohno, M., "Effect of MoS_2 Films on Scoring Resistance of Gears," ASME Paper 84-DET-59, 1984*

Summary — The effect of MoS_2 on balls and gears are reviewed. The paper presents equations for calculating flash temperature on gear teeth which have different heat transfer properties at the surface than at the core.

Gears were coated with MoS_2 approximately 0.0004 in. thick and tested at various speeds and loads. The test gears were also lubricated with oil during the test.

It was found that the flash temperature was very dependent on the thickness of the MoS_2 film and that increasing the film thickness resulted in a considerable rise in the flash temperature. This was found to be particularly true when the film thickness was smaller than the band length of the Hertzian contact zone.

Results of tests showed that gears coated with MoS_2 had considerably improved resistance to scoring.

Comments — High value for cryogenic gears. Results show that properly applied MoS_2 can reduce scoring of gears.

28. *El-Bayoumy, L. E., Akin, L. S., and Townsend, D. P., "An Investigation of the Transient Thermal Analysis of Spur Gears," ASME Paper 84-DET-92.*

Summary — A method for calculating transient surface temperatures using a finite element computer model is presented. The program models one tooth of a gear and can be used with both steady-state and transient heat transfer coefficients. Temperatures are calculated at various points along the contact surfaces of the pinion tooth. Calculations are made with consideration given to cooling several locations on the gear tooth.

Results of the model showed that the temperatures obtained using transient heat transfer coefficients were nearly the same as those obtained when using steady-state coefficients when only the loaded side of the tooth was cooled. Cooling the unloaded side of the tooth produced a dramatic reduction in temperature but cooling the top land was found to have little effect. The temperature rise along the recess portion of the tooth was found to be considerably less than that along the approach portion which would indicate an advantage in using recess action type gears to preclude scoring.

Comments — Very high value for cryogenic gears. This method would permit calculation of gear tooth surface temperatures, and thus would be a valuable design tool. However, the computer program may not yet be available.

29. *Wang, C. C., "On Analytical Evaluation of Gear Dynamic Factors Based on Rigid Body Dynamics," ASME Paper 84-DET-53, 1984.*

Summary — A proposal is made to improve the calculations for gear dynamic factor to include variations resulting from specific tooth error pattern, loading effects, inertials and other system-dependent characteristics. Most gear dynamic factors currently in use are dependent only on gear pitch line velocity and level of tooth accuracy. The method presented is based on the laws of mechanics for rigid bodies.

The model is set up to accept dynamic transmission errors although a good approximation can be made by using static transmission errors. The transmission error functions are input for the entire gear and are extended to include a complete revolution of the lowest speed shaft. It is assumed that gear tooth contact is always maintained.

The model has been computerized and the results of a number of cases are listed.

Comments — Very high value for cryogenic gears. The model would be very useful for determining the vibratory characteristics of high speed gears. The availability of the computer program is not known.

30. *Bowen, P. H., "Lubricant of Bearings and Gears in Aerospace Environmental Facilities," Defense Technical Information Center Report AD 411430, 1963.*

Summary — Study of lubricants for bearings, gimbal mounts and gears used in handling facilities for testing space vehicles. Various lubricants including MoS_2-PTFE mixtures were evaluated as gear lubricants. Gears were lubricated using a sacrificial idler gear made of lubricant.

A total of seven gear tests were run with one bearing totally unlubricated. Rotational speeds of the gears during the test were very low although loads and resultant stresses were high. Test temperatures ranged from −175°F to 300°F. The unlubricated gear was considered failed after five minutes.

Comments — Moderate value for cyrogenic gears. Most of the tests were run at temperatures well above of those of interest for cryogenic gears. Also gears speeds were not representative of turbomachinery. The study would be useful for slow moving, highly loaded systems such as actuators and gimbals.

APPENDIX B
CRYOGENIC GEAR TEST RIG DATA

REFERENCE 15

To:	C. H. Duncan
From:	F. E. Senesac
Subject:	Critical Speed Analysis - SSME Cryogenic Gear Test Rig
Date:	June 12, 1985
Copy To:	D. A. Lewis, J. Marshall, Group File

Reference:
1) PWA Canada Report XOI #12 "Experimental Procedure - Gear Fatigue Rig" by D. S. Robinson, dated October 1, 1984.
2) PWA Canada Report - "Spur Gear Fatigue Test Rig Critical Speed Analysis" by P. V. Sedran, dated March 15, 1982. Copy attached as Appendix I.
3) PWA Canada Drawing EFD 51290 - Top Assembly for Gear Fatigue Rig Installation.
4) EII 85-712-5589

INTRODUCTION:

The system analyzed is shown in Figure 1; it consists of the PW-EDS gear test rig adapter, layout shown in Figure 2, coupled to the PWC gear fatigue test apparatus layout and schematic shown in Figures 3 and 4.

The PWC gear fatigue test apparatus is driven by a hydraulic motor through a PT6-A34 gearbox, Ref. (1); pertinent critical speed analytical data are given in Ref. (2); copy attached as Appendix I.

For completeness an overall schematic of the PWC gear test rig is shown in Figure 5.

SUMMARY

Per Ref. (2), copy attached as Appendix I, the PWC gear fatigue test apparatus has lateral and torsional critical speeds within the 0-27,000 RPM speed range. The addition of the PW-EDS Adapter Rig will lower the speed of some of these criticals and introduce one new torsional critical speed.

Critical speed data are summarized in Table I.

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'Wafer' flexible couplings have been assumed between the PWC gear fatigue rig and the PW-EDS adapter rig to isolate lateral vibration of the two rigs.

This analysis indicates the addition of the gear box adapter rig does not detrimentally affect the dynamics of the PWC gear fatigue test rig and the conclusions of Ref. (1), see Appendix I, apply to the combined rig shown in Figure 1.

DISCUSSION

The use of 'wafer' flexible couplings between the two rigs will tend to isolate lateral vibrations; therefore, the lateral critical speeds of the PWC gear fatigue test apparatus, as cited in Ref. (2), were assumed to be applicable to the combined system shown in Figure 1. In addition, the PW-EDS adapter has lateral critical speeds as shown in Figure 6; for anticipated spring rates, the first critical is above 120% of the maximum desired operation speed (30,000 RPM).

Torsional critical speeds were calculated using a simplified model, details are contained in the calculation, see Ref. (4). In general, the addition of the gear rig adapter lowers the speed of some torsional natural frequencies, due to the added inertia, and introduces one new torsional mode, within the 0-120% speed range, at 23K RPM. The practical significance of this critical will require testing - the worst foreseeable consequence would be avoiding testing for a narrow speed range at this critical speed.

Gear mesh excitation will, in general, not be a problem since their interferences occur at low shaft speeds, under 10,000 RPM - see Figure 7.

Additional information is given in Appendix I.

F.E.C by DKM
F. E. Senesac, Ext. 4497
Structures

Approved: James K Miller 6-24-85
D. K. Mills, Ext. 4383
Structures

FES:rls

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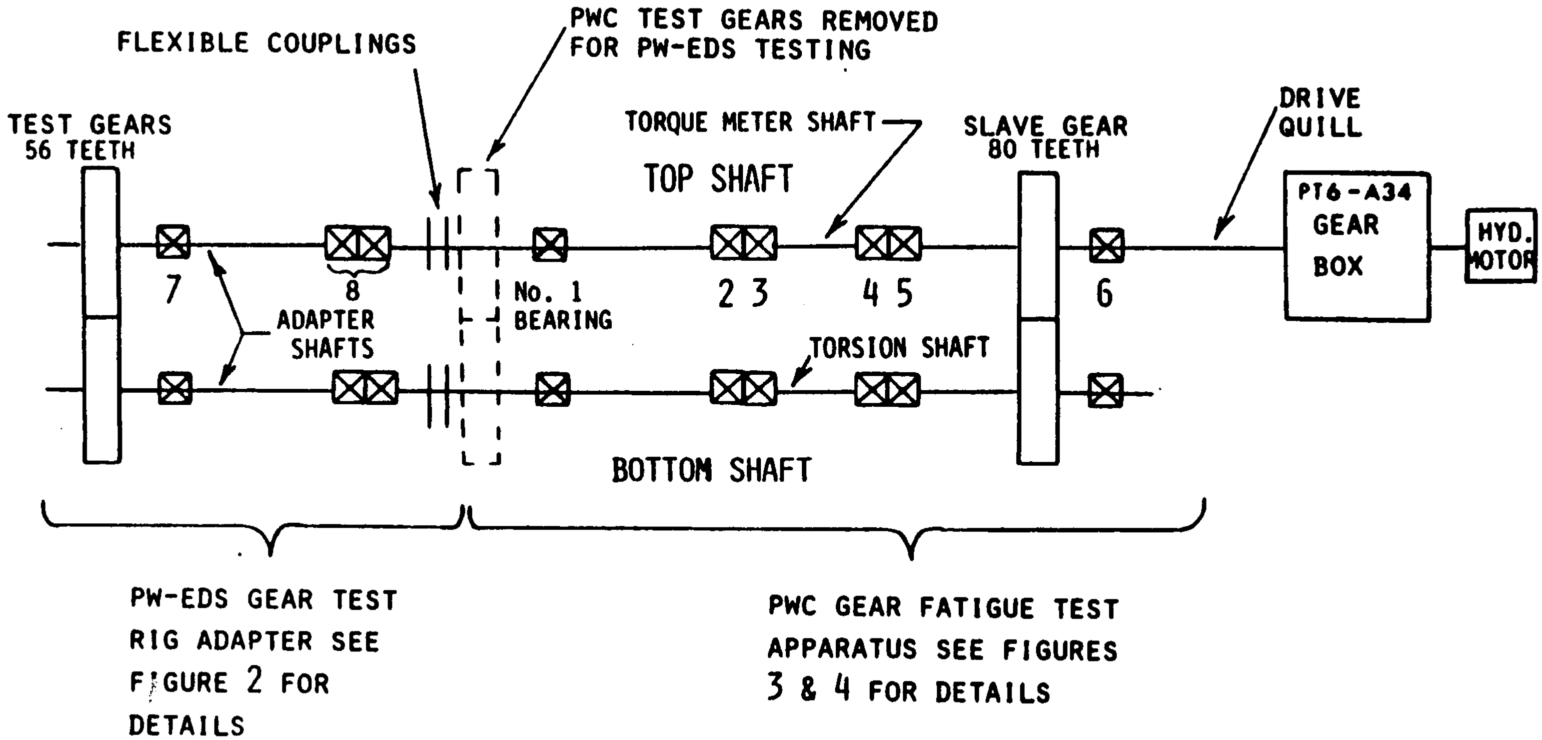

NOTES : (1) SEE FIGURES 2 & 3 FOR DETAILS
(2) K1 THRU K8 = 5×10^6 LB/IN

FIGURE 1.

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TABLE 1
CRYOGENIC TEST RIG
CRITICAL SPEED SUMMARY

LATERAL CRITICAL SPEEDS

CRTICAL SPEED RPM	ROTOR S.E. PERCENT	ROTOR LOCATION REFER TO FIGURE 1
15,750 (1)	61	PWC GEAR RIG-BOTTOM ROTOR
24,600 (1)	52	"
25,200 (2)	31	PWC GEAR RIG- TOP ROTOR
31,200 (2)	32	"
37,700 (3)	45	PW-EDS ADAPTER RIG - BOTH ROTORS
40,250 (2)	21	PWC GEAR RIG - TOP ROTOR
42,200 (1)	52	PWC GEAR RIG - BOTTOM ROTOR
49,900 (3)	24	PW-EDS ADAPTER RIG - BOTH ROTORS

TORSIONAL CRITICAL SPEEDS

MODE NUMBER	CRITICAL SPEED RPM	MODE DESCRIPTION	HIGHEST NODE LOCATION REFER TO FIGURE 1
1	4,900	BOTTOM SHAFT UNCOUPLED	TORSION SHAFT
2	5,160	SYSTEM FUNDAMENTAL (4)	DRIVE QUILL
3	19,400	SYSTEM 2nd MODE (4)	TORQUE METER SHAFT
4	23,000	SYSTEM 3rd MODE (4)	TOP ADAPTER SHAFT
5	59,000	SYSTEM 4th MODE (4)	BOTTOM ADAPTER SHAFT

(1) REFERENCE (1), TABLE 2 - SEE APPENDIX I

(2) REFERENCE (1), TABLE 1 - SEE APPENDIX I

(3) REFERENCE (4), Page 20

(4) ASSUMING THE GEARBOX INERTIA >> TEST RIG INERTIA

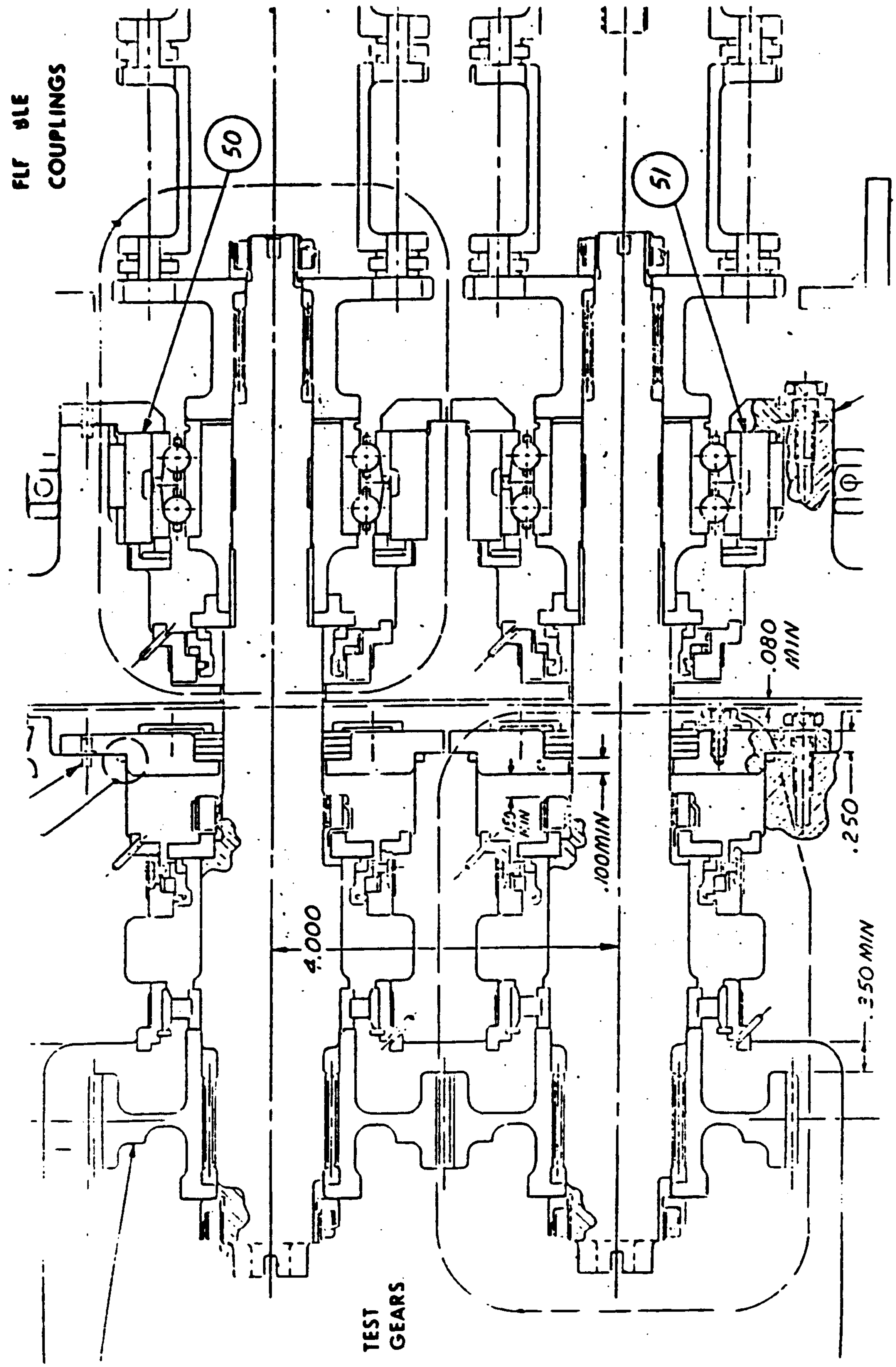
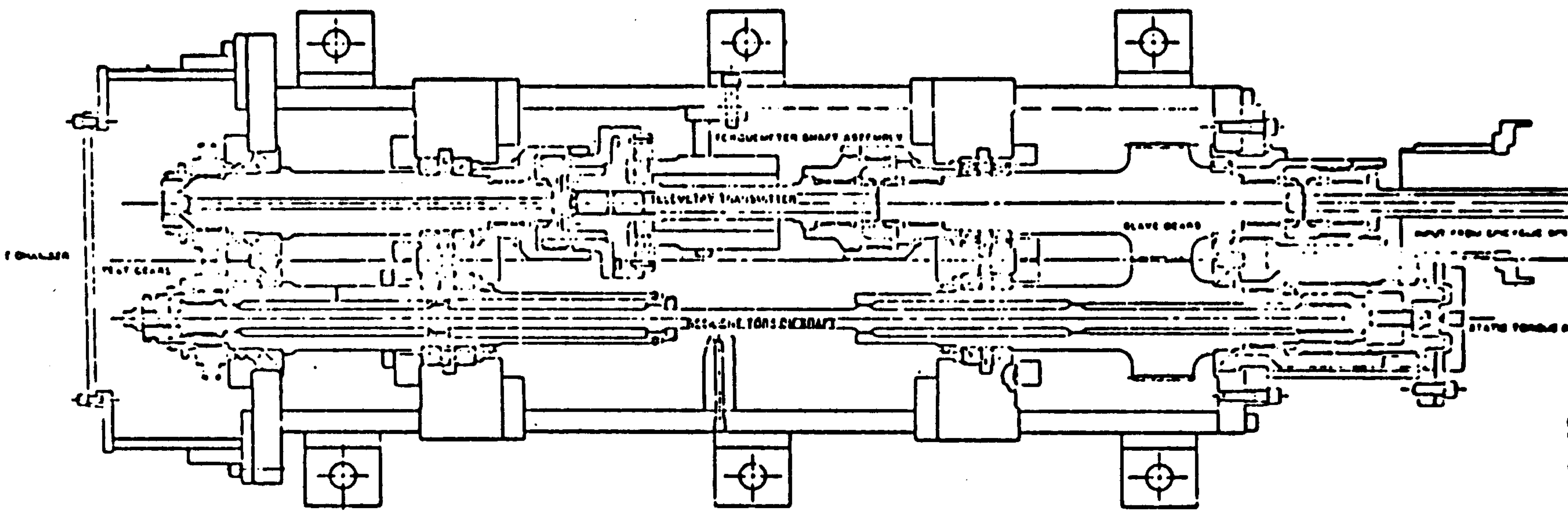


FIGURE 2
L240062 5/24/85
GEAR TEST RIG ADAPTER
LAYOUT



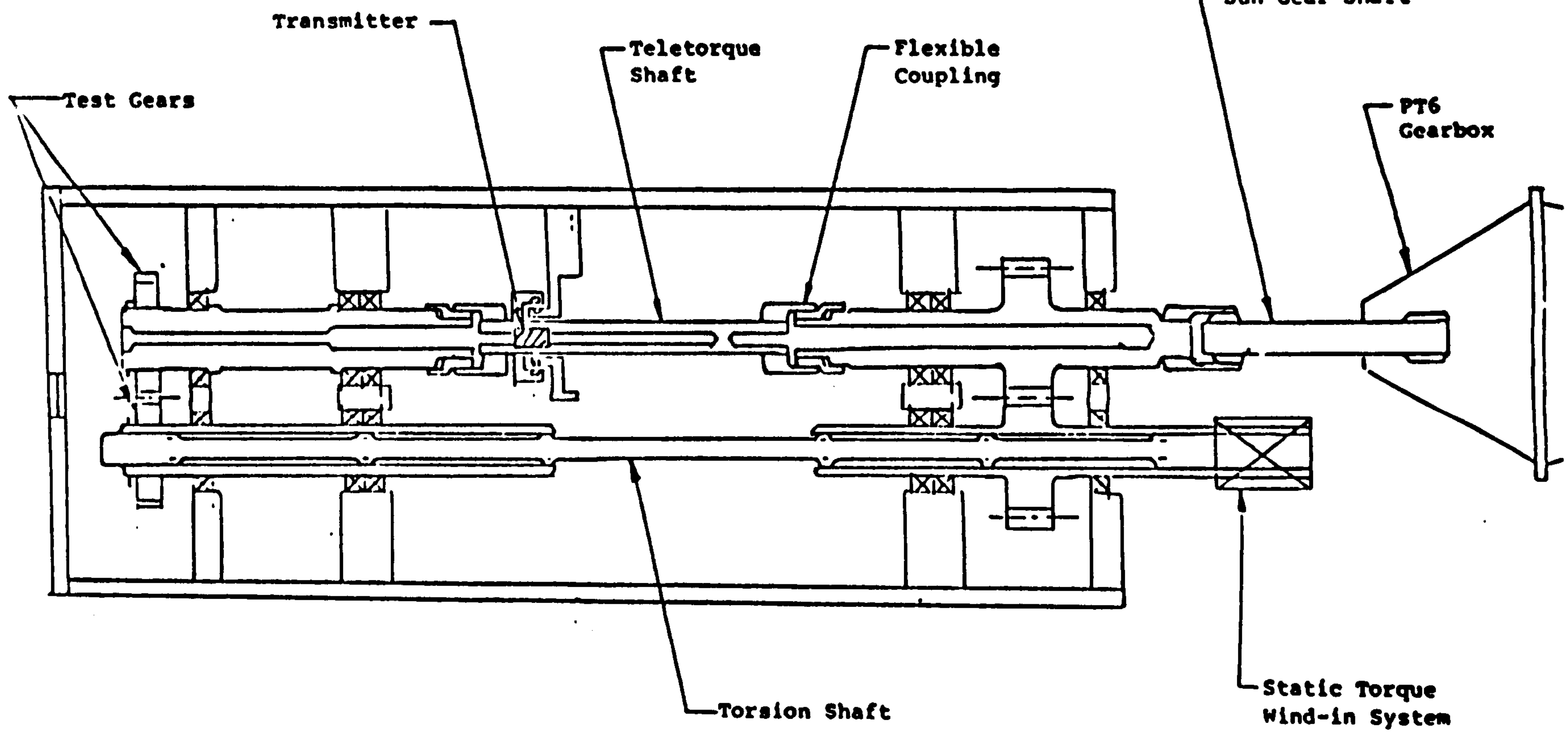
PWA CANADA GEAR FATIGUE TEST APPARATUS

NOTES: (1) REFERENCE PWA CANADA DRAWING EFD51110

(2) SEE FIGURE 4 FOR SCHEMATIC

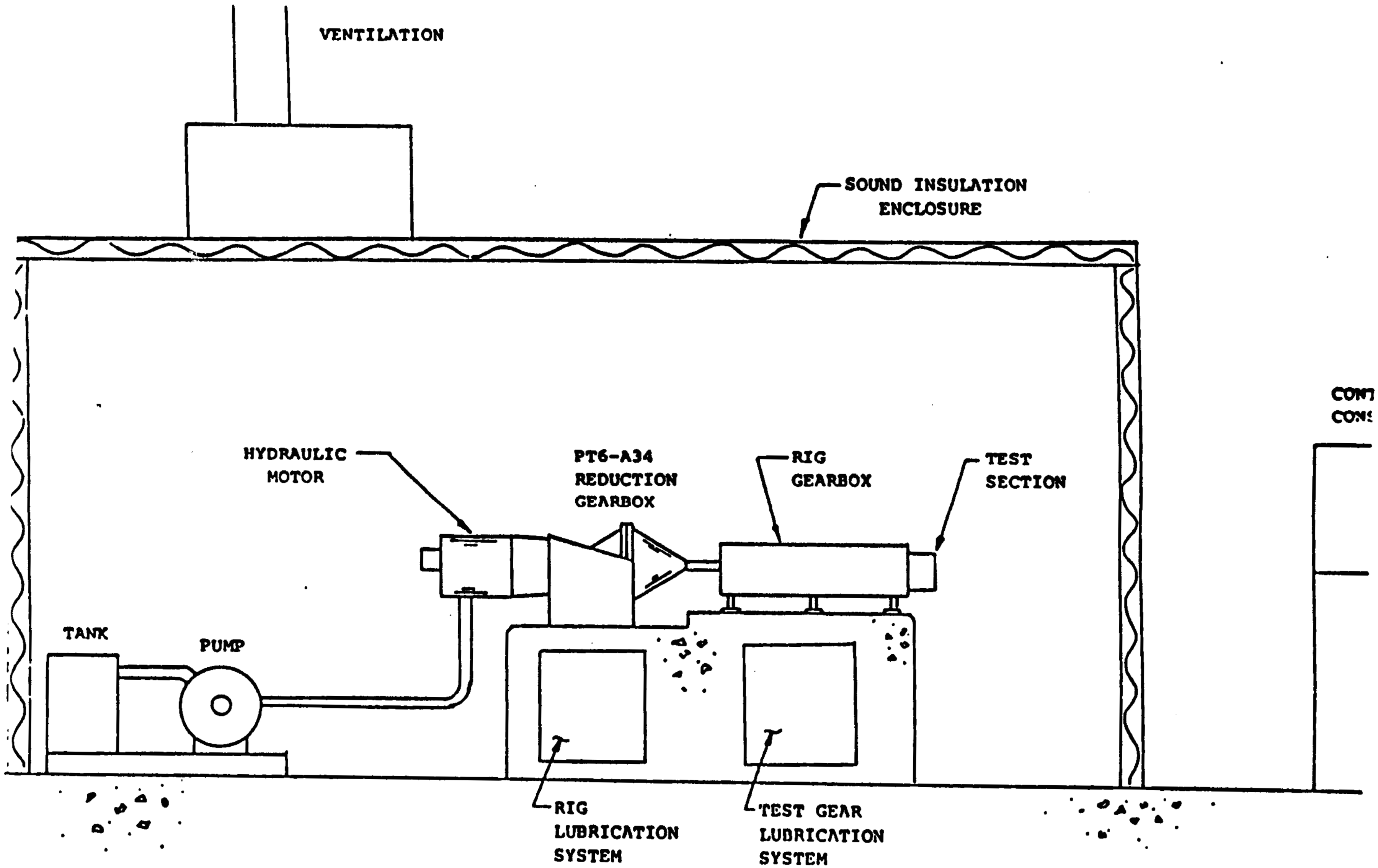
FIGURE 3

SCHEMATIC - PWA CANADA GEAR FATIGUE TEST APPARATUS



NOTE: (1) REFERENCE PWA CANADA DRAWING EFD51110

FIGURE 4



PWC
GEAR FATIGUE RIG

NOTE: (1) SEE REFERENCE (3)

FIGURE 5

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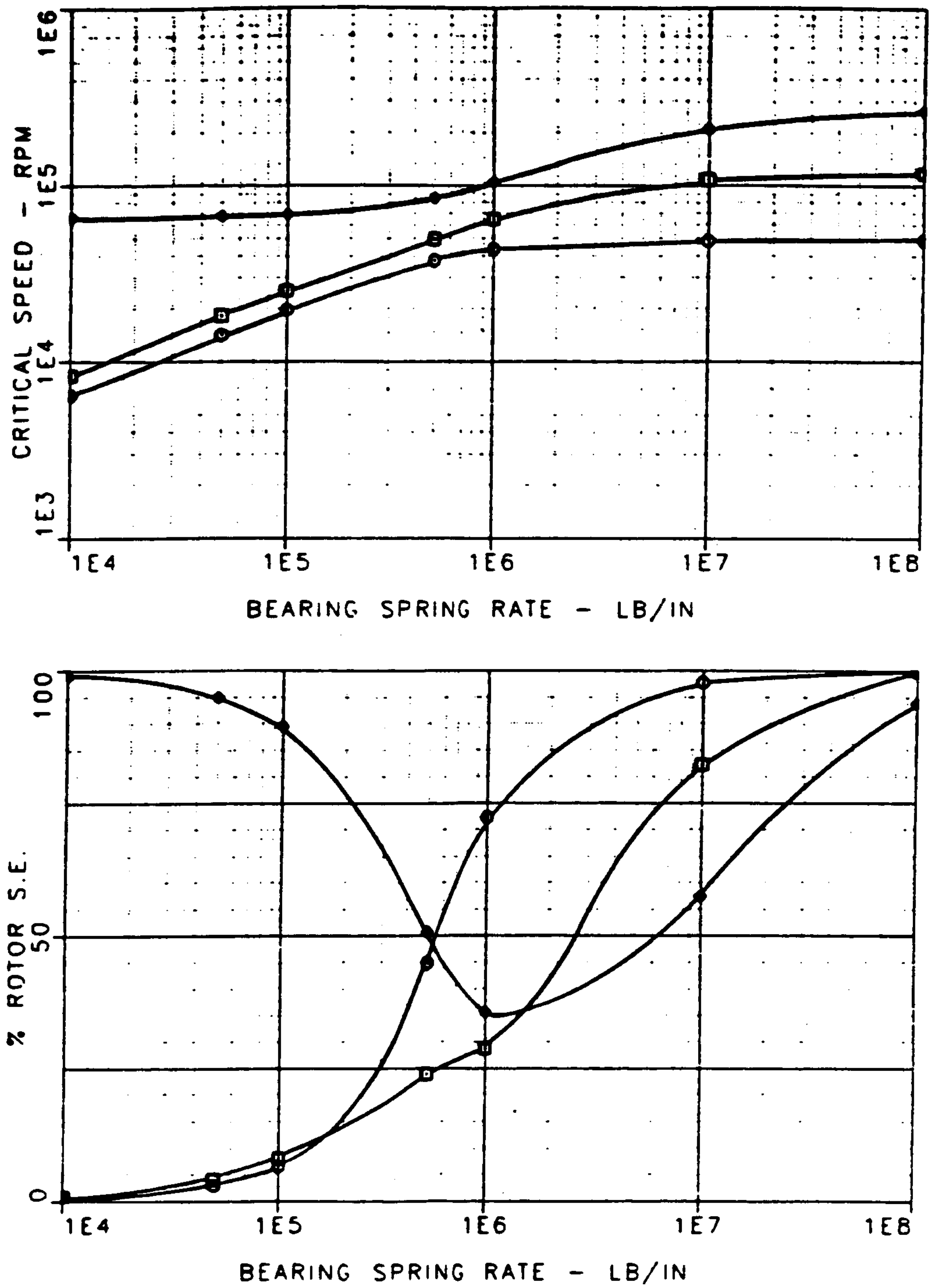
CRYOGENIC GEAR TEST RIG ADAPTER



NOTES: (1) L240062 PER FIGURE 2
(2) KROLLER = KDUPLEX

FIGURE 6

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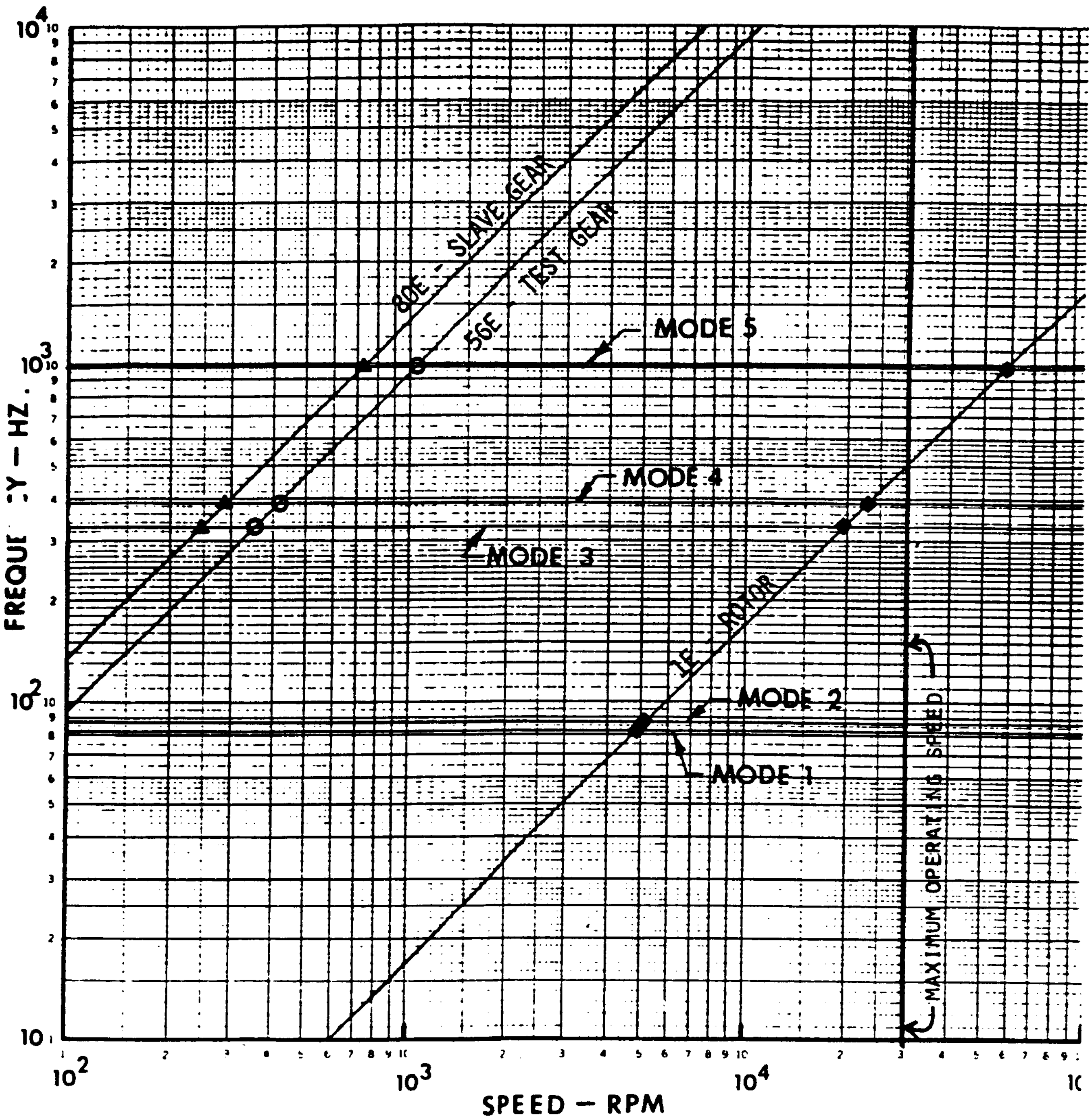
CRYOGENIC GEAR RIG

CAMPBELL DIAGRAM

TORSIONAL CRITICALS



NOTES: (1) SYSTEM SCHEMATIC - FIGURE 1

(2) MODE DEFINITION - TABLE 1

FIGURE 7

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APPENDIX I

PRATT & WHITNEY AIRCRAFT OF CANADA LTD.

2780-82-85-00200
2780-82-85-00203T
2780-82-85-31060

To: Mr. M. Botman — 15 March 1982

From: P.J. Sedran, Dept. 3320, loc. 324 9AG1

cc: S. Anderson
C. Brownridge
G.J. Brown
F.C. Desrochers
W.K. Crawford
J.F. Harrop
R. Marks
~~J.I. Nikolajsen~~
C.F. #54120

Re: Spur Gear Fatigue Test Rig
Critical Speed Analysis

1. Introduction

This memo summarizes the results of a lateral and torsional critical speed analysis of the spur gear fatigue test rig, performed as was requested by Static Structures. The rig, designed by Plant 7 Facilities, is P&WC's first gear fatigue test rig.

2. Conclusions

The analyses have shown the existence of several lateral and torsional critical speeds within the running range. It is recommended that vibration levels be monitored to avoid running the rig at resonance. If excessive lateral vibrations occur during testing at particular speeds, oil film dampers could be introduced at the #1 and #6 bearings, or fatigue testing could be done at other selected speeds.

Torsional resonances will likely occur at low speeds which must and can be easily avoided. These speeds are dependent upon the number of gear teeth on the slave or test gears and must be recalculated (see sec. 4) whenever the rig is to be run with different gears.

3. Lateral Critical Speed Analysis

Fig. #1 illustrates the rig's basic configuration. It consists of two shafts, each carrying a test gear at one end and a slave gear at the other meshing so as to form a closed four-square torque loop. The top shaft is to be driven through a PT6A-34 gearbox, acting as a speed increaser, by an hydraulic motor. The bottom shaft features helical splines which allow it to act as a torsional spring applying a torque preload to the test gears.

All the data presented in this section was obtained by running program 571 with geometrical models of the top and bottom shafts as input.

3.1 Bearing support and gearbox stiffness parametric studies

Figs. 2 and 3 present parametric studies showing the effect of bearing support stiffness upon the top and bottom shaft lateral critical speeds. Fig. 4 shows the effect of gearbox stiffness upon the top shaft critical speeds. The bearing support stiffnesses are roughly 5×10^5 $lb_f/in.$ and the gearbox stiffness is of the order of 1×10^5 $lb_f/in.$ Therefore, a total of three criticals are expected within the running range. A fourth critical exists close to the proposed maximum speed of 30,000 rpm so it may be necessary to limit the maximum speed to 28,000 rpm.

contd:

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3.2 Lateral critical speeds

Tables #1 and #2 show the lateral critical speeds and their mode shapes given the bearing support and gearbox stiffnesses indicated above. Bearing strain energies are highest in the #1 and #6 bearings for the criticals within the running range. If vibration levels are found to be unacceptably high during testing, the response of the top shaft can be reduced by oil film damping of the #6 bearing. Oil film damping of the #6 bearing of the bottom shaft will serve to reduce the vibrations experienced by the bottom shaft at the first critical. The response of the bottom shaft at the second critical speed can be reduced by oil film damping at the #1 bearing.

4. Torsional Critical Speed Analysis

The lowest torsional critical speeds of the rig were roughly estimated as follows. The lowest criticals with nodal locations at the gears were estimated by calculating the fixed-fixed criticals of the bottom shaft. This was done by solving the general d.e. for the continuous shaft. The lowest critical with a mid-span node, assuming sufficient backlash to uncouple the shafts, was estimated for each shaft, as well as the lowest critical with a mid-span node assuming torsional coupling of the two shafts. For the top shaft, this was done by lumping the polar moments of inertia in three locations and constructing a Haltzer table for the system. For the bottom shaft, the calculation was performed by lumping polar moments of inertia in two locations. A summary of the analyses is presented overleaf.

Gear mesh excitation should be avoided at these frequencies, and at half these frequencies. The gear mesh excitation frequency is simply the product of the gear rotational speed and the number of gear teeth. Thus, knowing the number of teeth, the speed corresponding to a resonant gear mesh frequency to be avoided can be found. For example, in order to avoid resonance at 583 Hz given 80 gear teeth, a speed of

$$\frac{583}{80} = 7.28 \text{ rps}$$
$$= 437.25 \text{ rpm}$$

must be avoided. There are eight of these critical frequencies and so this calculation should be performed to give eight critical rig speeds for each different number of gear teeth with which the rig is run.

In general, since these speeds are relatively low, the resonances can be easily avoided.

Paul Sedray

snvb

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TABLE #1 Top Shaft

(WHIRLING SPEED, MODE SHAPE AND ENERGY DISTRIBUTION

BEARING STIFFNESS (LB/IN)							
NO #1	NO #2	NO #3	NO #4	NO #5	NO #6	NO gearbox	NO
4.76×10^5	4.76×10^5	4.76×10^5	4.76×10^5	4.76×10^5	4.76×10^5	1.0×10^5	

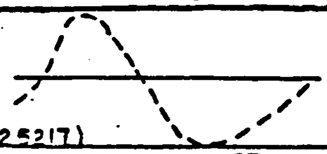
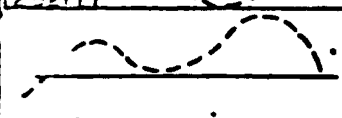
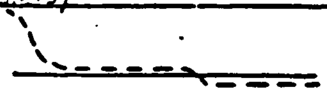
MODE SHAPE AND WHIRLING SPEED (RPM)	STRAIN ENERGY DISTRIBUTION ~ % TOTAL					
	L.P. SHAFT	H.P. SHAFT	Top Shaft	THREE MOST HIGHLY LOADED BEARINGS		
 (25217)			30.6%	NO 6: 31.0%	NO 3: 10.5%	NO 1: 10.4%
 (31213)			32.0%	NO 6: 44.0%	NO 1: 6.5%	NO 4: 5.6%
 (40250)			20.5%	NO 1: 64.4%	NO 2: 6.8%	NO 3: 5.7%
				NO	NO	NO
				NO	NO	NO
				NO	NO	NO
				NO	NO	NO

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TABLE #2 Bottom shaft

(WHIRLING SPEED, MODE SHAPE AND ENERGY DISTRIBUTION

BEARING STIFFNESS (LB/IN)							
NO 1	NO 2	NO 3	NO 4	NO 5	NO 6	NO	NO
4.76×10^5	4.76×10^5	4.76×10^5	4.76×10^5	4.76×10^5	4.76×10^5		

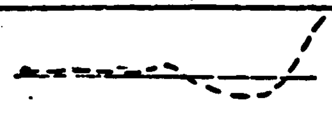
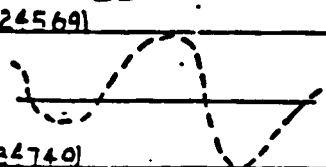
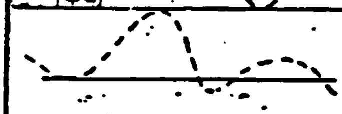
MODE SHAPE AND WHIRLING SPEED (RPM)	STRAIN ENERGY DISTRIBUTION ~ % TOTAL					
	L.P. SHAFT	H.P. SHAFT	Bottom Shaft	THREE MOST HIGHLY LOADED BEARINGS		
 (15752)			61.2	NO 6: 35.2	NO 5: 2.4	NO 1: 0.5
 (24569)			52.3	NO 1: 42.2	NO 4: 3.5	NO 3: 0.7
 (34740)			57.4	NO 6: 29.0	NO 4: 6.8	NO 3: 3.4
 (42189)			51.7	NO 6: 16.8	NO 3: 14.5	NO 2: 7.7
				NO	NO	NO
				NO	NO	NO
				NO	NO	NO

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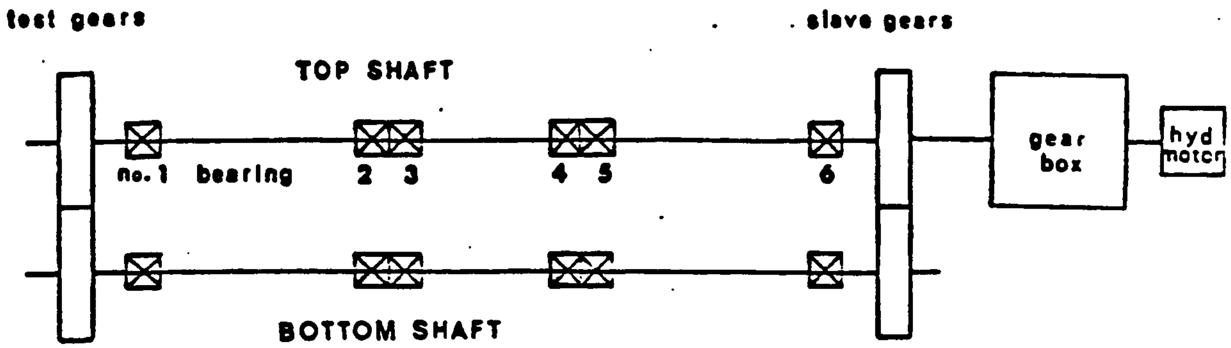
FIG.1: RIG SCHEMATIC



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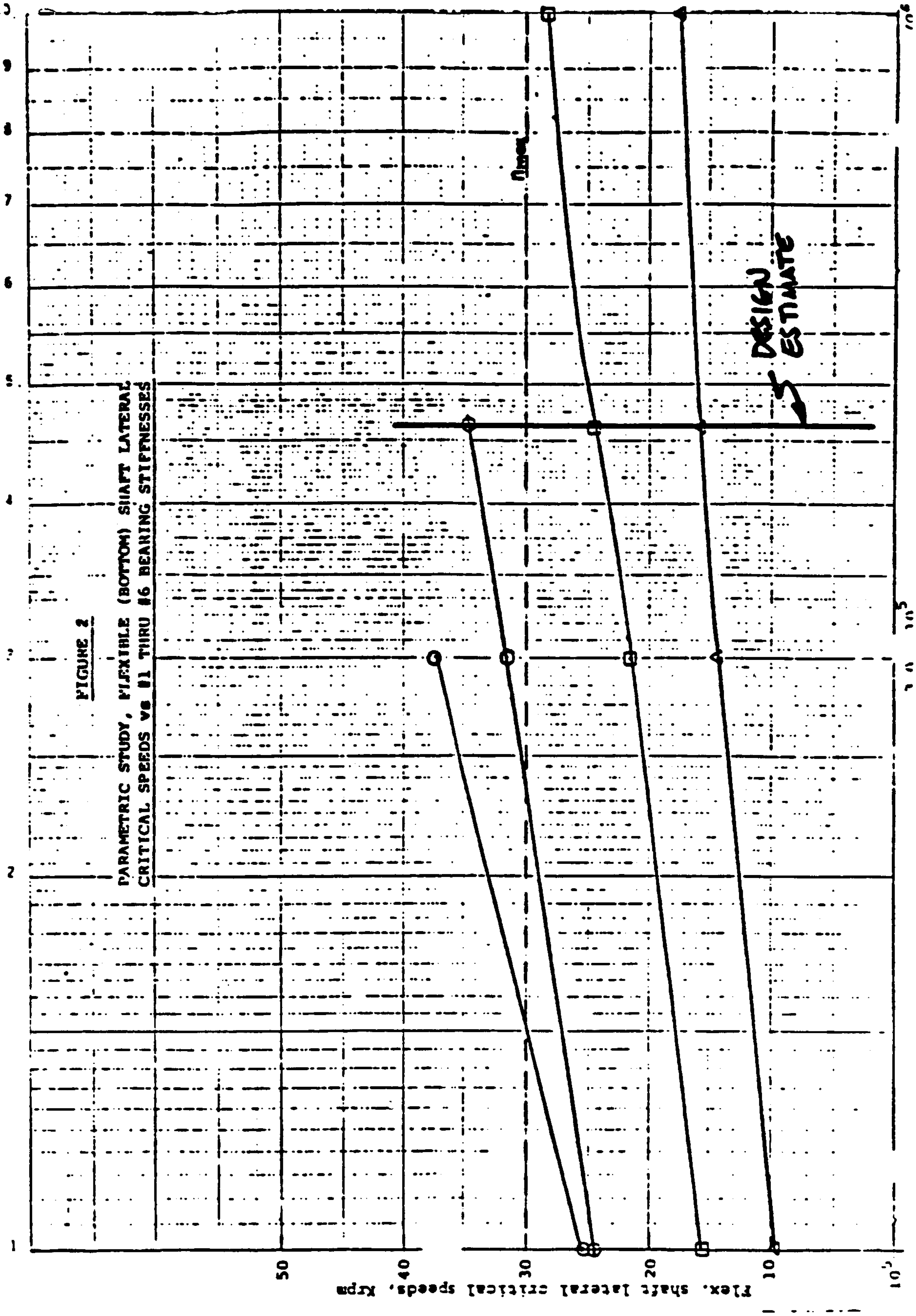

FIGURE 2

PARAMETRIC STUDY, FLEXIBLE (BOTTOM) SHAFT LATERAL CRITICAL SPEEDS vs #1 THRU #6 BEARING STIFFNESSES

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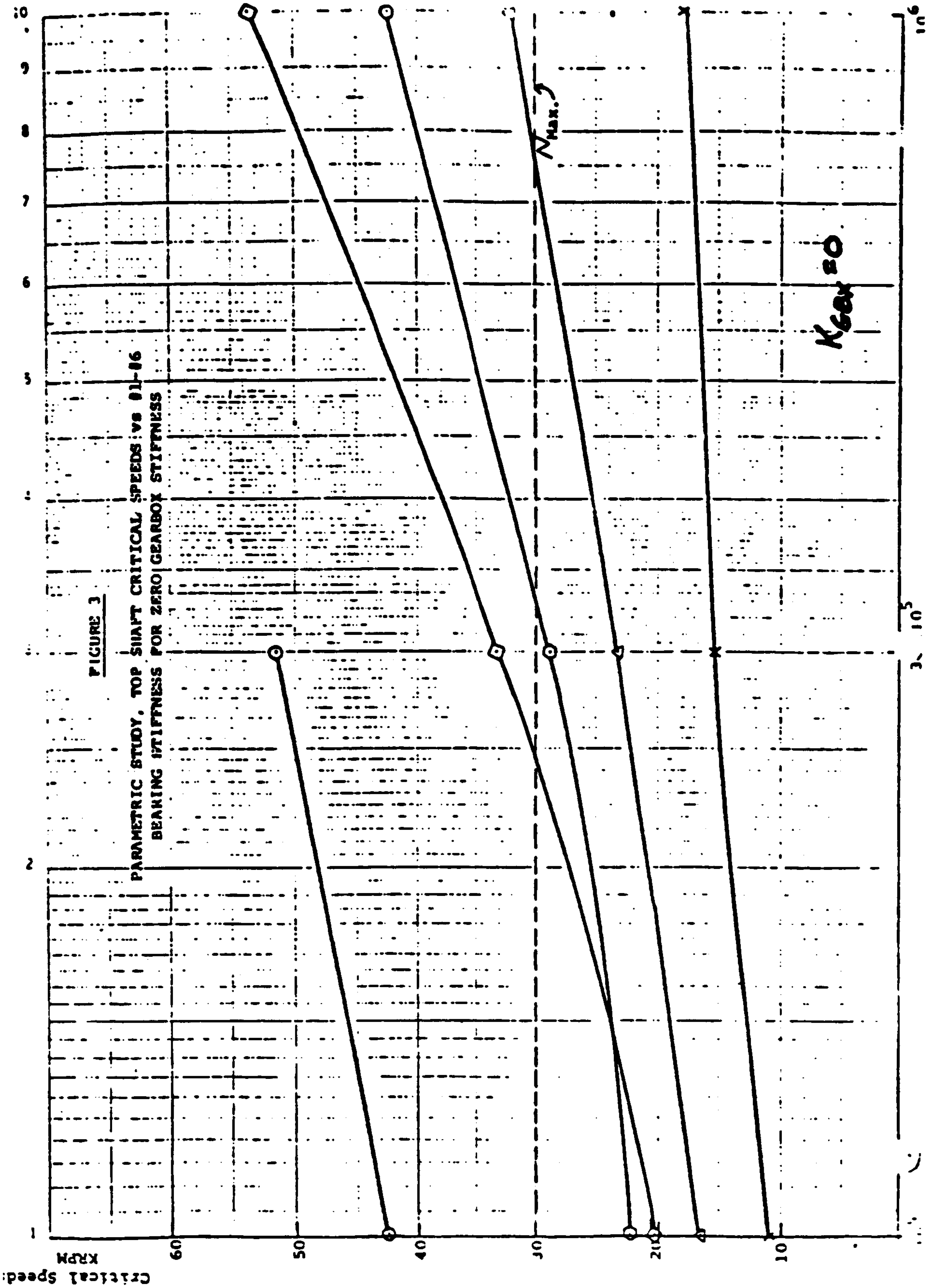

FIGURE 3

PARAMETRIC STUDY, TOP SHAFT CRITICAL SPEEDS vs B1-B6 BEARING STIFFNESS FOR ZERO GEARBOX STIFFNESS

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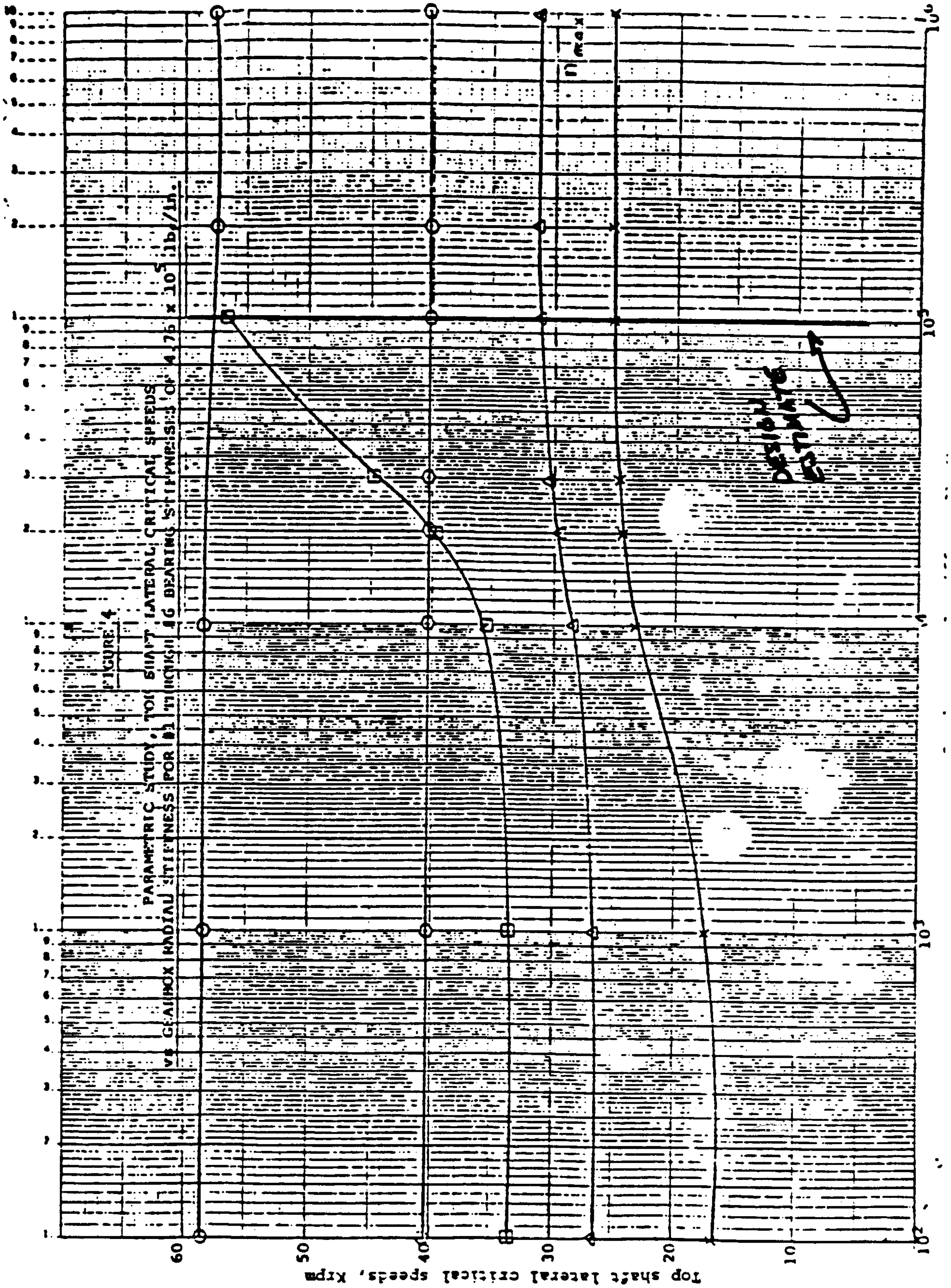

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REFERENCE 16

PRATT & WHITNEY AIRCRAFT
ENGINEERING DIVISION SOUTH
INTERNAL CORRESPONDANCE

TO : G. DUNCAN — **DATE: JUNE 3, 1985**

FROM: Y. E. EREN — **EXT : 2936**

SUBJECT: RL-10 GEAR TEST RIG BEARINGS

CC: E.M. BEVERLY
J. MARSHALL
T. KMIEC
J. BROWN

The bearing lives and the fits for the RL-10 CRYOGENIC GEAR TEST RIG based on the final material selection and layout dimensions are given with the attached tables and graphs.

1. BEARING LIVES FOR THE MIN. AND MAX. IRC.
2. BEARING COMPONENT FITS FOR OPERATING TEMPERATURES.
3. ROLLER BEARING LIFE v.s. INPUT TORQUE FOR AVG. IRC.
4. DUPLEX BEARING LIFE v.s. INPUT TORQUE FOR AVG. IRC.
5. ROLLER BEARING LIFE v.s. GEAR PRESSURE ANGLE FOR AVG. IRC.
6. BEARING RADIAL SPRING RATES FOR MIN., AVG., MAX. IRC.
7. FIT RADIAL AND TANGENTIAL STRESSES.
8. DUPLEX BEARING BALL EXCURSION v.s. PRELOAD.
9. TEST RIG LAYOUT WITH BEARING NUMBERS.
10. BASIC BEARING DIMENSIONS.

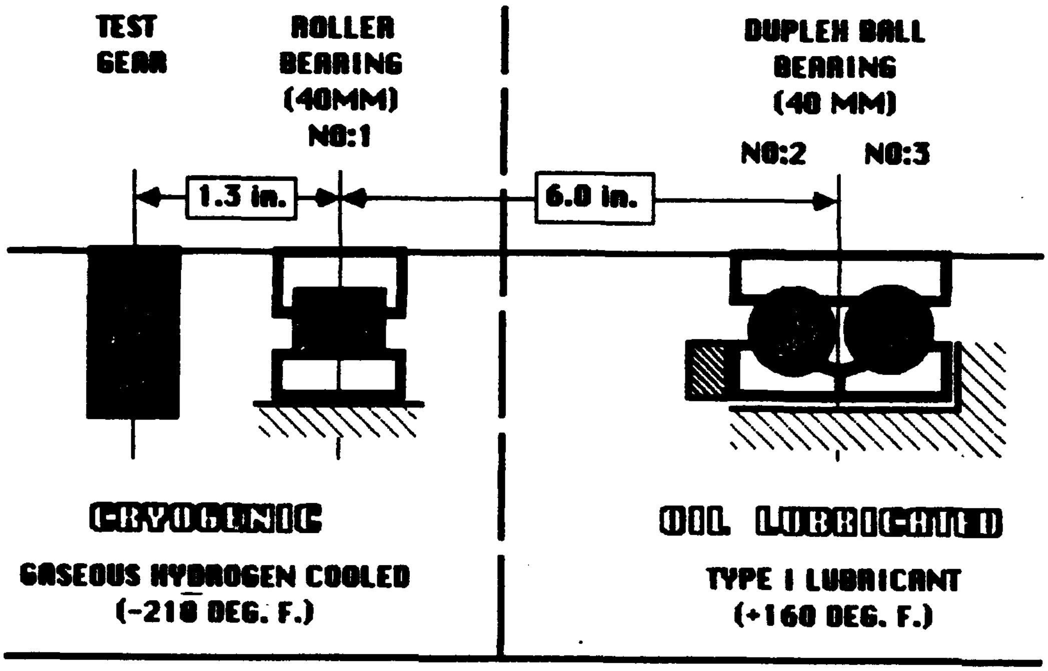
BY: Yul Eren
YUL EREN

APPROVED BY:
LARRY HESS

ROTOR SYSTEM

OVERALL CONCEPT : OVERHUNG GEAR MOUNTED SHAFT SYSTEM TO FACILITATE CHANGE OF TEST GEARS.

MAXIMUM RADIAL LOAD CARRIED BY ROLLER BEARING, COOLANT PRESSURE THRUST LOAD CARRIED BY DUPLEX BALL BEARING.



Y. EREN
JUNE 14, 1985

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By V I Tran
June 3, 1985

RL-10 CRYOGENIC GEAR TEST RIG
BEARING LIFE v.s. IRC

	ROLLER BRG BEARING#1	BALL BRG. BEARING#2	BALL BRG. BEARING#3
MINIMUM IRC	0.00068	0.00177	0.00177
BEARING LIFE	171.2	3,445.8	1,408.2
MAXIMUM IRC	0.00161	0.00309	0.00309
BEARING LIFE	107.8	4,090.1	2,353.6

INTERNAL RADIAL CLEARANCE......- IRC (INCHES)

BEARING LIFE...............................- (B1) HOURS

INPUT TORQUE...............................- 500 IN. LBS. MAX.

INPUT SPEED.................................- 27,000 RPM MAX.

EXPECTED LIFE OF TEST RIG = 30 HOURS

PRESENT LIFE OF GEARS IS LESS THAN 1-HOUR
AT INPUT TORQUE -500 IN.-LBS.
INPUT SPEED - 27,000 RPM

By: Y. E. Eren
June 3, 1985

RL-10 CRYOGENIC GEAR TEST RIG

BEARING COMPONENT FITS

ROLLER BEARING: P/N 2069345 (*)

PARTS	MATERIAL	TEMP-oF	FITS -	INCHES
HOUSING	AMS5665	-216.9		
O.R.-LINER	AMS5665	-219.6	+0.0005	+0.0020
OUTER RACE	AMS5630	-220.0	-0.0020	-0.0010
INNER RACE	AMS5630	-215.4	+0.0011	+0.0017
SHAFT	AMS5671	-212.0		

BALL BEARING: P/N 4047466 (*)

PARTS	MATERIAL	TEMP-oF	FITS -	INCHES
HOUSING	AMS5646	+160.2		
O.R.-LINER	AMS5665	+160.7	+0.0012	+0.0027
OUTER RACE	PWA725	+160.4	-0.0009	-0.0000
INNER RACE	PWA725	+162.5	+0.00025	+0.00085
I.R.-LINER	AMS5665	+163.6	+0.0001	+0.0010
SHAFT	AMS5671	+165.6		

(*) THE OUTER RACE LINER I.D. IS TO BE MACHINED AFTER IT IS ASSEMBLED.

"-" Loose Fit

"+" Tight Fit

JUNE 14, 1985

RL-10 CRYOGENIC GEAR TEST RIG

BEARING DESIGN LOADS & STRESSES

INPUT TORQUE........................ = 500 IN-LBS.
GEAR PRESSURE ANGLE......... = 25 DEGREES.
COOLANT PRESSURE.............. = 80 PSIA Max.
TOTAL MISALIGNMENT......... = 0.003 IN. Max.
SHAFT SPEED......................... = 27,000 RPM Max.

LOADS (LBS.)

	GEAR INPUT	ROLLER BEARING NO:1	DUPLEX BALL BEARING NO:2	NO:3
RADIAL	275.9	330.8	47.2	7.7
AXIAL-PRELOAD			150.0	150.0
AXIAL-COOLANT	67.8			67.8

CONTACT STRESSES (PSI)

ROLLER BEARING NO:1	DUPLEX BALL BEARING NO:2	BEARING NO:3
154,100	145,300	149,000

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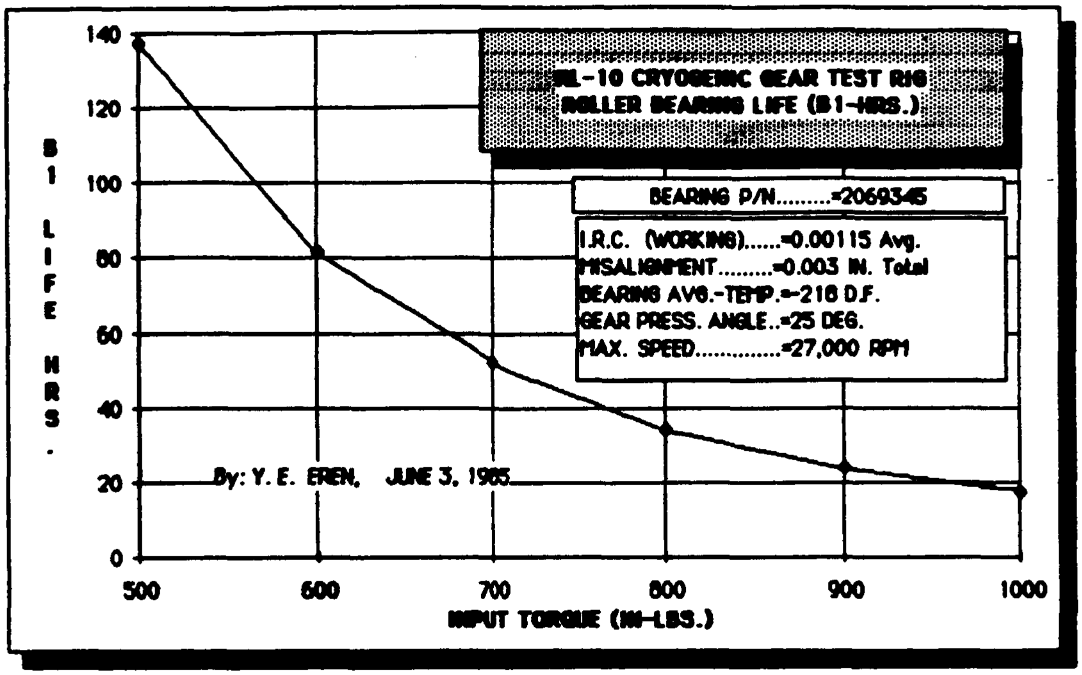

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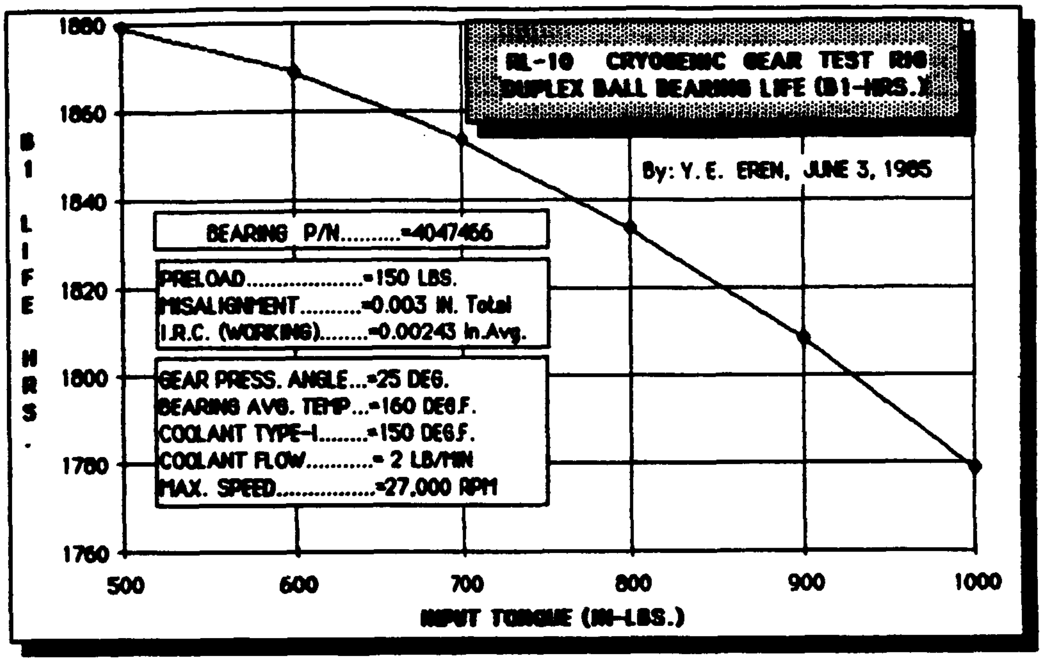

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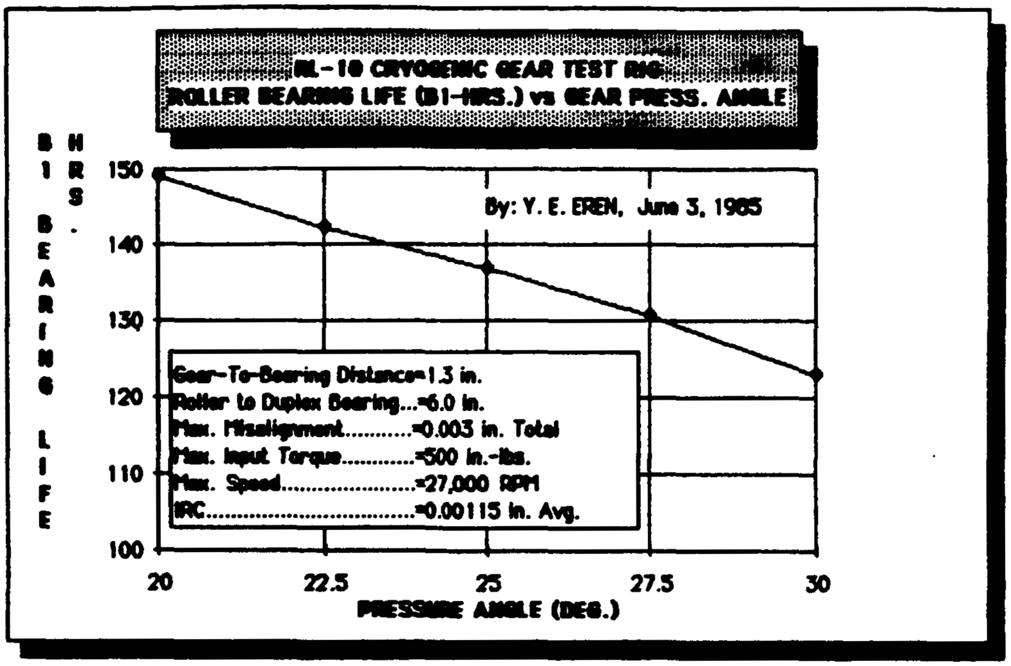
RL-10 CRYOGENIC GEAR TEST RIG
ROLLER BEARING LIFE (B1-HRS.) vs GEAR PRESS. ANGLE
B1 BEARING LIFE
HRS.
By: Y. E. EREN, June 3, 1985
150
140
130
120
110
100
Gear-To-Bearing Distance=1.3 in.
Roller to Duplex Bearing...=6.0 in.
Max. Misalignment..........=0.003 in. Total
Max. Input Torque...........=500 in.-lbs.
Max. Speed.....................=27,000 RPM
IRC...............................=0.00115 in. Avg.
20
22.5
25
27.5
30
PRESSURE ANGLE (DEG.)

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By: Y. E. Eren
June 3, 1985

RL-10 CRYOGENIC GEAR TEST RIG
BEARING RADIAL SPRING RATES

	ROLLER BRG BEARING#1	BALL BRG. BEARING#2	BALL BRG. BEARING#3
MINIMUM IRC	0.00068	0.00177	0.00177
SPRING RATE	794,200	424,900	709,390
AVG. IRC	0.00115	0.00243	0.00243
SPRING RATE	722,110	283,420	512,220
MAXIMUM IRC	0.00161	0.00309	0.00309
SPRING RATE	518,900	197,400	370,820

INTERNAL RADIAL CLEARANCE......= IRC (INCHES)

SPRING RATES..............................= LBS./IN.

INPUT TORQUE..............................= 500 IN.-LBS. MAX.

INPUT SPEED................................= 27,000 RPM MAX.

5130C

By: Y. E. Eren
June 3, 1985

RL-10 CRYOGENIC GEAR TEST RIG
BEARING COMPONENT FIT
RADIAL AND TANGENTIAL STRESSES

ROLLER BEARING: P/N 2069345 (*)

PARTS	RADIAL STRESS MAX. / MIN.	TANGENTIAL STRESS MAX. / MIN.
HOUSING		
O.R.-LINER	-2,395.0 / -558.2	-22,566.0 / -5,259.9
OUTER RACE	0.0 / 0.0	0.0 / 0.0
INNER RACE	-2,023.2 / -559.3	19,045.0 / 8,558.5
SHAFT		

BALL BEARING: P/N 4047466 (*)

PARTS	RADIAL STRESS MAX. / MIN.	TANGENTIAL STRESS MAX. / MIN.
HOUSING		
O.R.-LINER	-2,486.5 /-673.0	-12,499.0 /-3,833.2
OUTER RACE	0.0 / 0.0	0.0 / 0.0
INNER RACE	-4,027.9 / -611.6	25,348.0 / 8,210.4
I.R.-LINER	-14,360.0 / -527.3	19,776.0 / 2,670.1
SHAFT		

(*) THE OUTER RACE LINER I.D. IS TO BE MACHINED AFTER IT IS ASSEMBLED.

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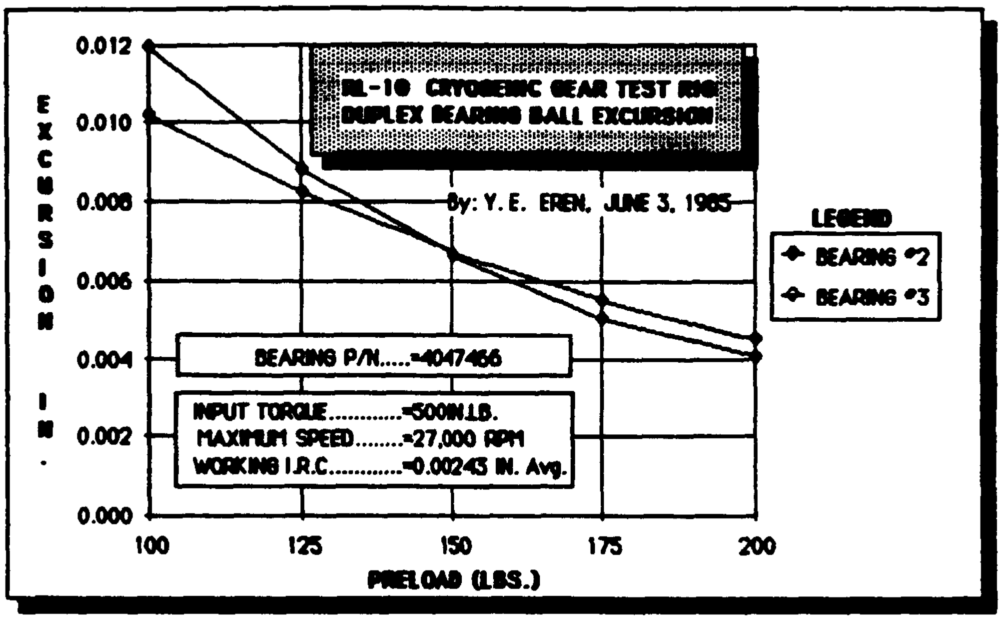
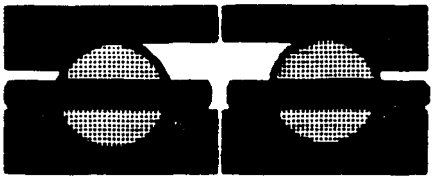


5130C

Y. E. EREN
June 3, 1985

RL-10 GEAR TEST RIG

BALL BEARING



40 x 68 x 30 mm

P/N 4047466

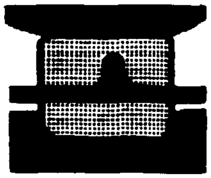
BEARING O.D.	2.6771 in.
BEARING I.D.	1.5747 in.
BEARING WIDTH	0.5906 x 2 in.
NUMBER OF ELEMENTS	16.0
BALL DIAMETER	0.3125 in.
INNER RACE CURVATURE	0.53
OUTER RACE CURVATURE	0.55
INTERNAL RADIAL CLEARANCE	0.0033 in. Avg. @11 lbs.
PITCH DIAMETER	2.1260 in.
INNER RACE LAND DIA	1.9000 in.
OUTER RACE LAND DIA	2.3170 in.
DESIGN CONTACT ANGLE	21.0 Deg.
MATERIAL: RACES, BALLS	PWA725
CAGE	AMS6414 Steel

5130C

Y. E. EREN
June 3, 1985

RL-10 GEAR TEST RIG

ROLLER BEARING



40 x 62 x 12 mm

P/N 2069345

BEARING O.D.	2.44075 in.
BEARING I.D.	1.57470 in.
BEARING WIDTH	0.46990 in.
NUMBER OF ELEMENTS	10.00
ROLLER DIAMETER.........................	0.250 in.
ROLLER LENGTH.............................	0.250 in.
ROLLER FLAT LENGTH.....................	0.1195 in.
INTERNAL RADIAL CLEARANCE	0.0010 in. AVG.
PITCH DIAMETER...........................	2.0000 in.
CROWN RADIUS.............................	27.00 in.
CORNER RADIUS............................	0.0175 in.
MATERIALS: RACES, ROLLERS........	AMS 5630 (440C)
CAGE.........................	AMS 4616 Forged Silicon Bronze

5130C

By: Y. E. Eren
June 3, 1985

RL-10 CRYOGENIC GEAR TEST RIG
BEARING DESIGN CRITERIA
CHECK

INPUT TORQUE (IN.-LBS.)	BALL-BEARING #2 GYRO SLIP COEFF.	BALL-BEARING #2 SV FACTOR ($X10^6$)	BALL-BEARING #2 SPIN ROLL RATIO	BALL-BEARING #3 GYRO SLIP COEFF.	BALL-BEARING #3 SV FACTOR ($X10^6$)	BALL-BEARING #3 SPIN ROLL RATIO
500.	0.043	0.59	0.39	0.044	0.64	0.32
600.	0.043	0.59	0.40	0.044	0.64	0.33
700.	0.043	0.59	0.41	0.045	0.64	0.34
800.	0.043	0.59	0.41	0.045	0.63	0.34
900.	0.042	0.59	0.42	0.046	0.63	0.35
1000.	0.042	0.60	0.43	0.046	0.63	0.36
MAX.LIMIT	0.060	2.00	0.50	0.060	2.00	0.50

PRELOAD.................... = 150 LBS. - Nominal
IRC.......................... = 0.00243 In. - Avg.
MISALIGNMENT......... = 0.003 Inches - Total
MAX. SPEED............... = 27,000 RPM - Max.
PRESSURE ANGLE........ = 25 DEG. - Nominal
COOLANT AXIAL LOAD..= 80 PSIA. (67.8 LBS.) - Max.

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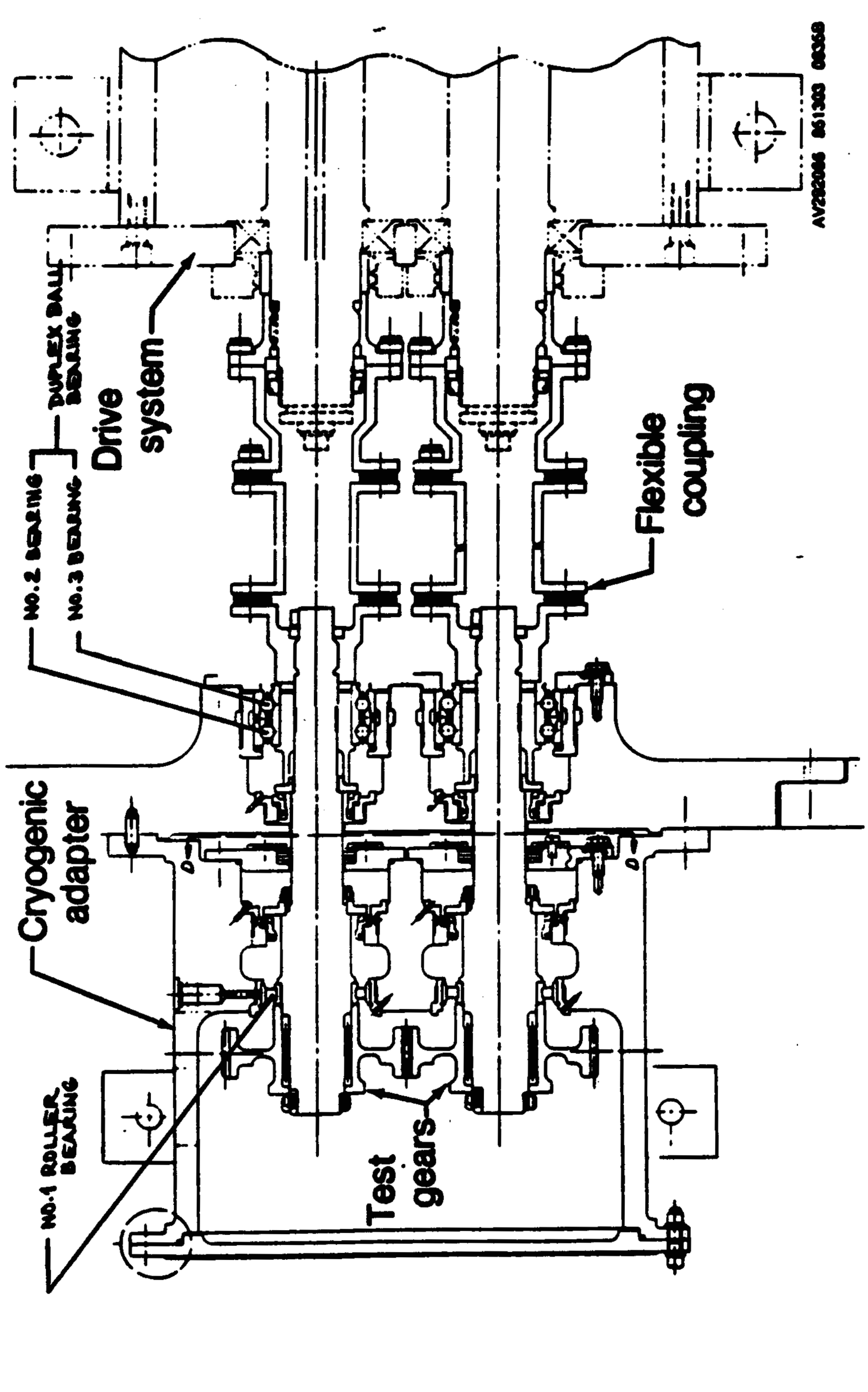
CRYOGENIC GEAR TEST RIG



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REFERENCE 17

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PRATT & WHITNEY

DON'T GIVE VERBAL ORDERS

MEMO. TO YUL EREN DATE 4/23/85

SUBJECT: RL 10 GEAR RIG SHAFT TEMPERATURES ARE DEPENDENT ON BEARING & SEAL HEAT GENERATION

CC: R. DUCAN, H. L. HESS, J. D. DOERENBACH, E. M. BEVERLY

REF: DGVO; Y. EREN TO T. SWARTWOUT, "RL10 GEAR TEST RIG BEARING AND SEAL HEAT GENERATION" 4/9/85

SUMMARY: TEMPERATURES OF THE SHAFT AND HOUSING OF THE RL10 GEAR TEST RIG HAVE BEEN CALCULATED FOR FOUR COMBINATIONS OF BEARING AND SEAL HEAT GENERATION. THE DIFFERENCE IN BEARING HEAT GENERATION IS AN EXPECTED VARIATION AND MUST BE CONSIDERED IN THE DESIGN. THE SEAL HEAT GENERATION IS A FUNCTION OF THE SPRING LOADS. MECHANICAL DESIGN SHOULD REDUCE THE SPRING FORCE TO THE LOWEST POSSIBLE VALUES TO MINIMIZE SEAL PLATE, WHICH MAY REQUIRE ADDITIONAL RUNS.

T. W. Swartwout
MC&S, CDT
X 2937

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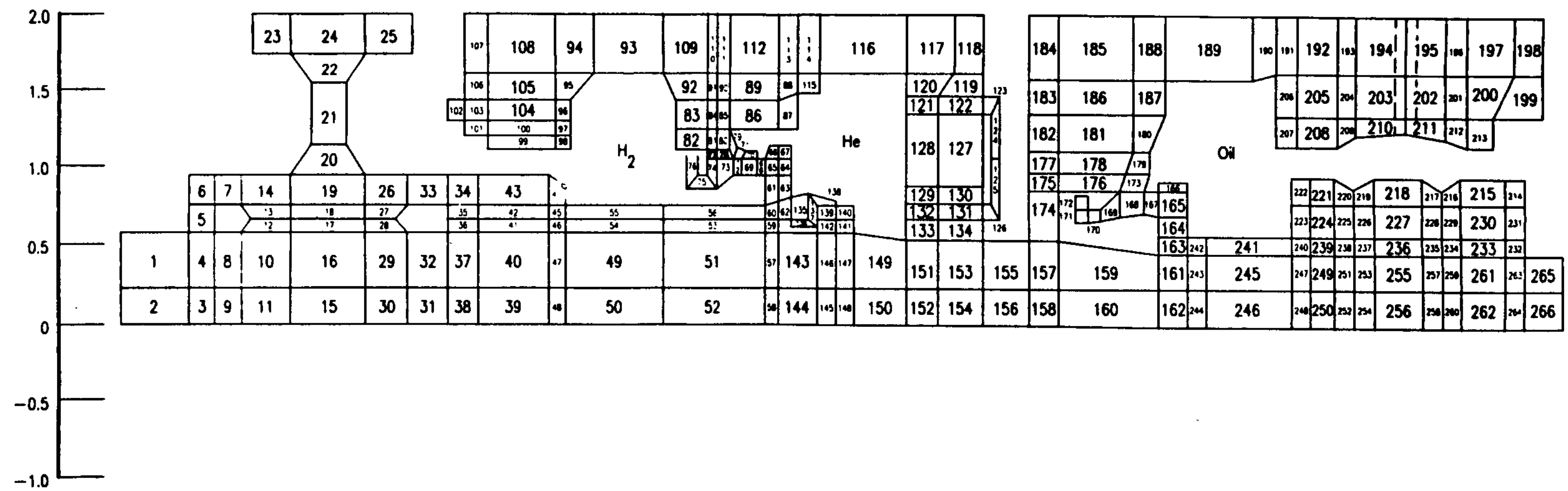
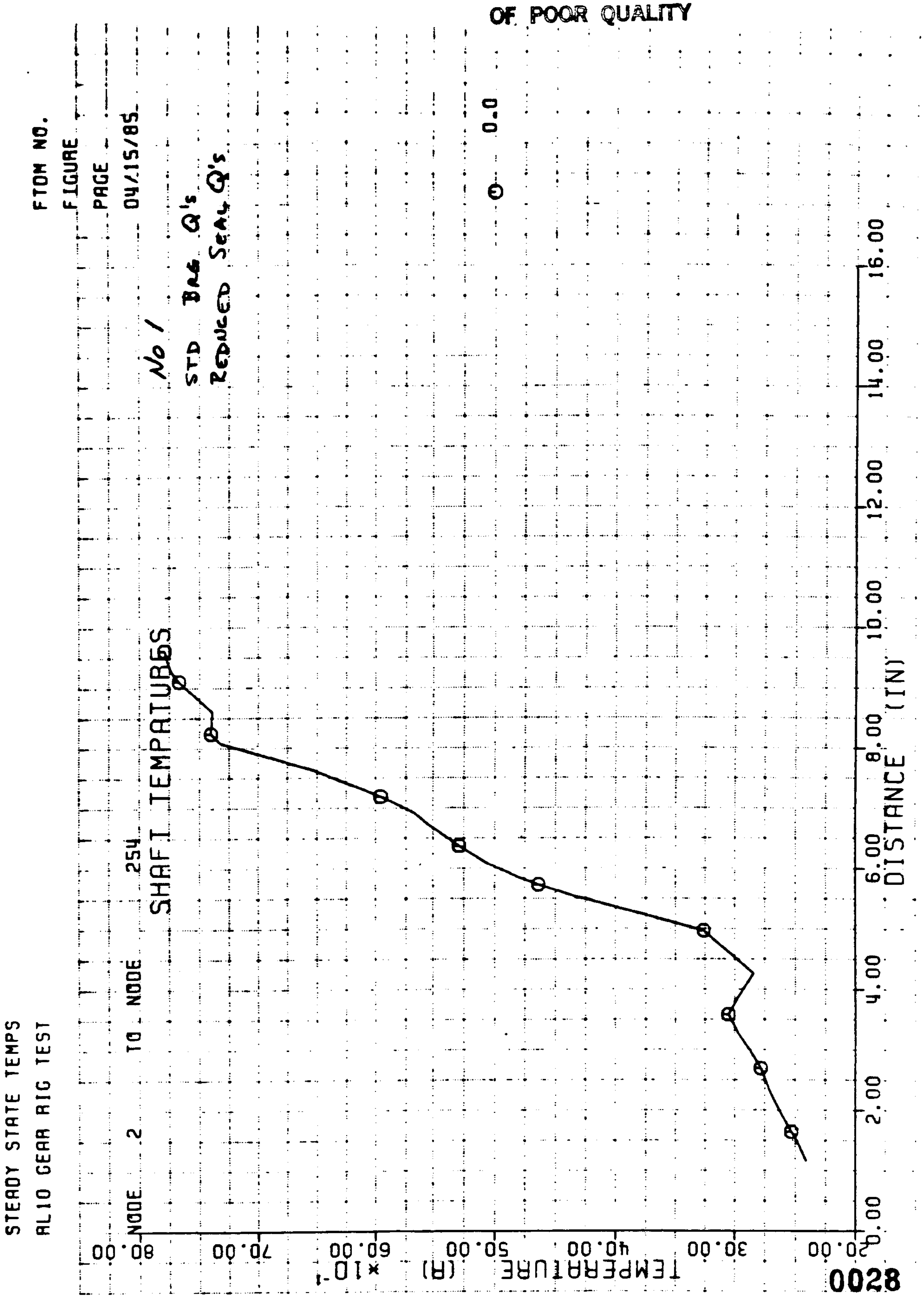


2 FOLDOUT FRAME

FOLDOUT FRAME


STEADY STATE TEMPS
AL10 GEAR RIG TEST
NODE 2 TO NODE 254
FTOM NO.
FIGURE
PAGE
04/15/85
SHAFT TEMPATURES
No 1
STD BRG Q's
REDUCED SEAL Q's
0.0
TEMPERATURE (R) ×10⁻¹
80.00
70.00
60.00
50.00
40.00
30.00
20.00
DISTANCE (IN)
0.00
2.00
4.00
6.00
8.00
10.00
12.00
14.00
16.00
0028

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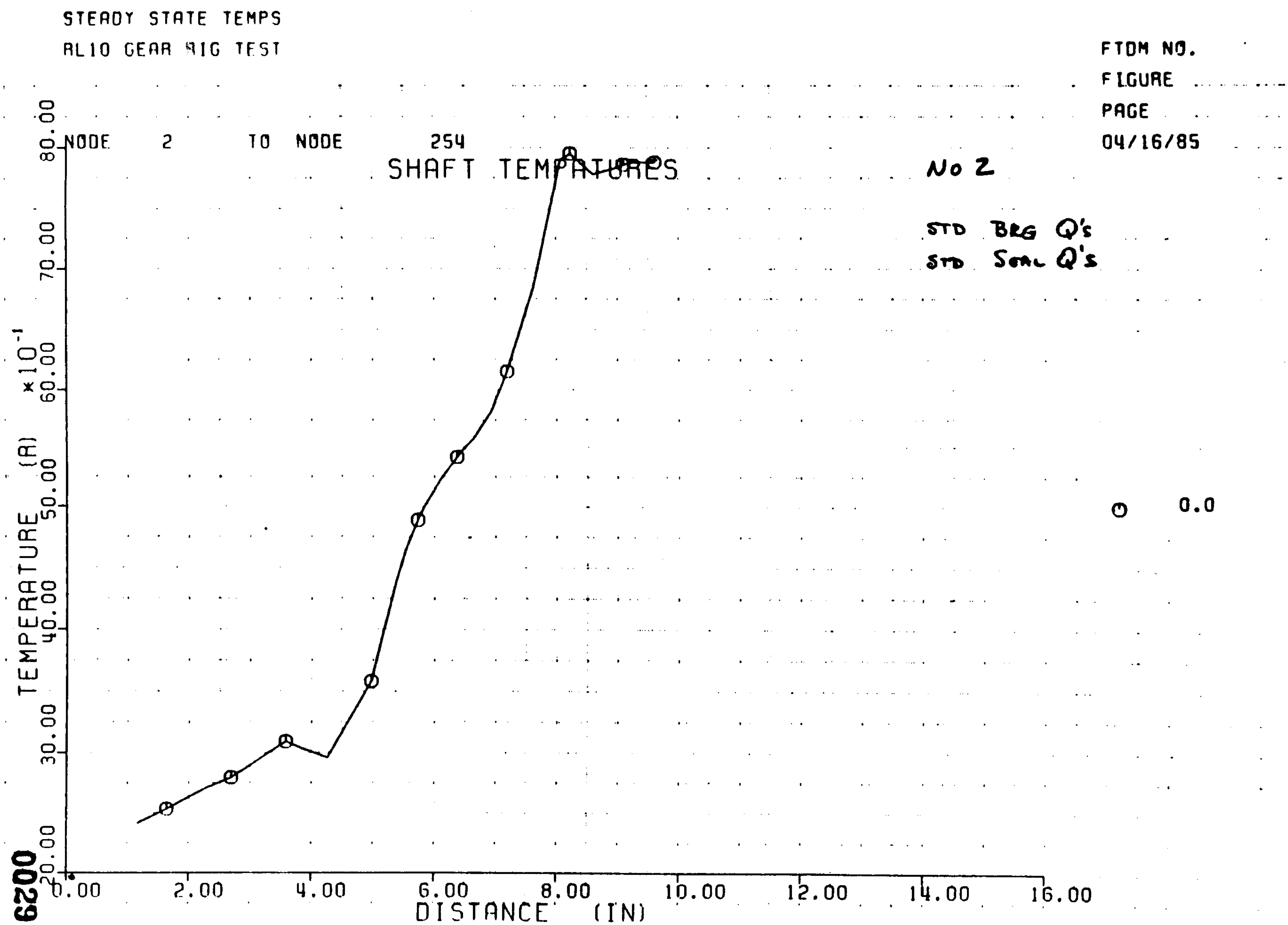
STEADY STATE TEMPS
AL10 GEAR RIG TEST
FTDM NO.
FIGURE
PAGE
04/16/85
NODE 2 TO NODE 254
SHAFT TEMPATURES
No 2
STD BRG Q's
STD SEAL Q's
0.0
TEMPERATURE (R) ×10⁻¹
80.00
70.00
60.00
50.00
40.00
30.00
20.00
0.00
2.00
4.00
6.00
8.00
10.00
12.00
14.00
16.00
DISTANCE (IN)
0029

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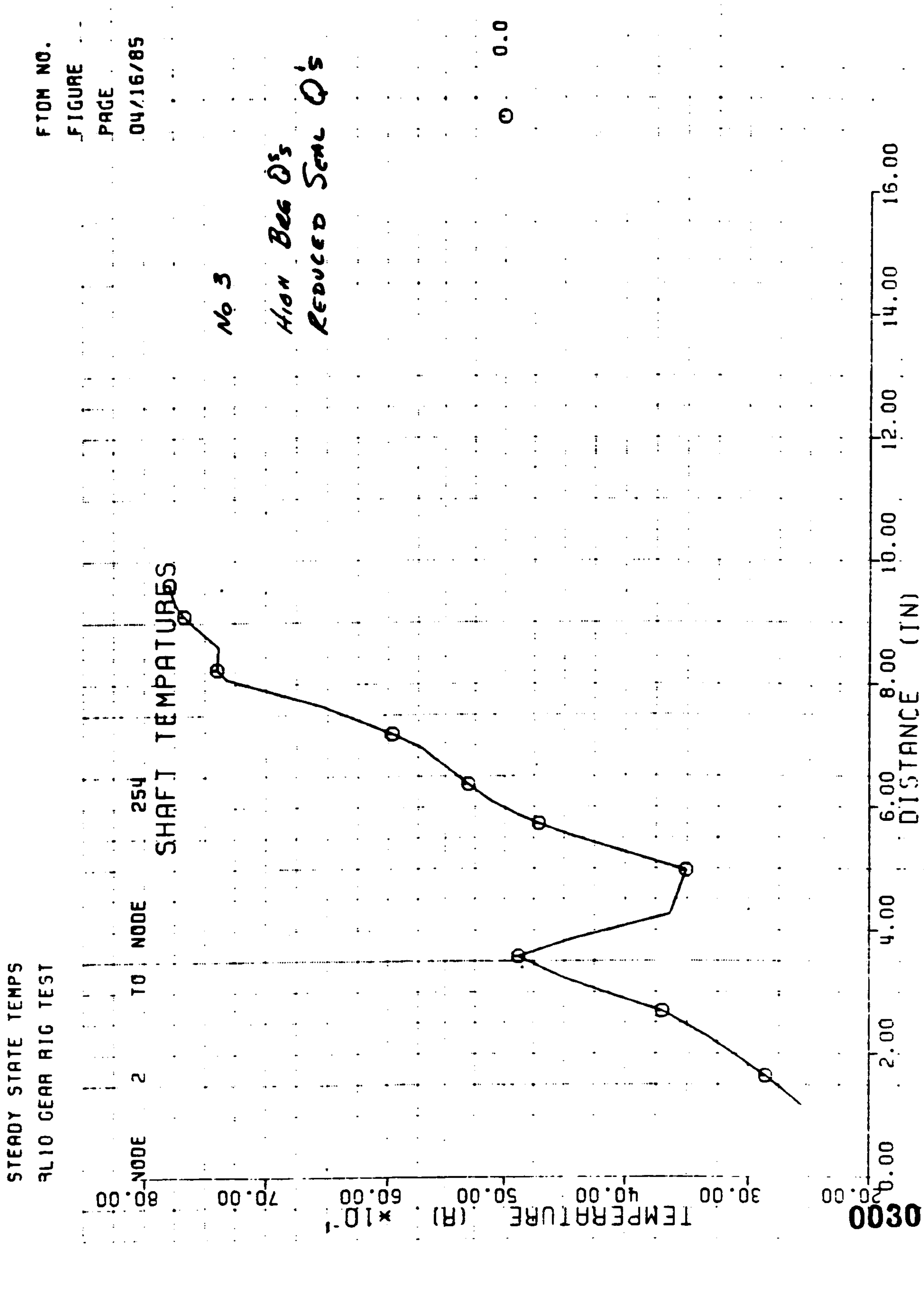

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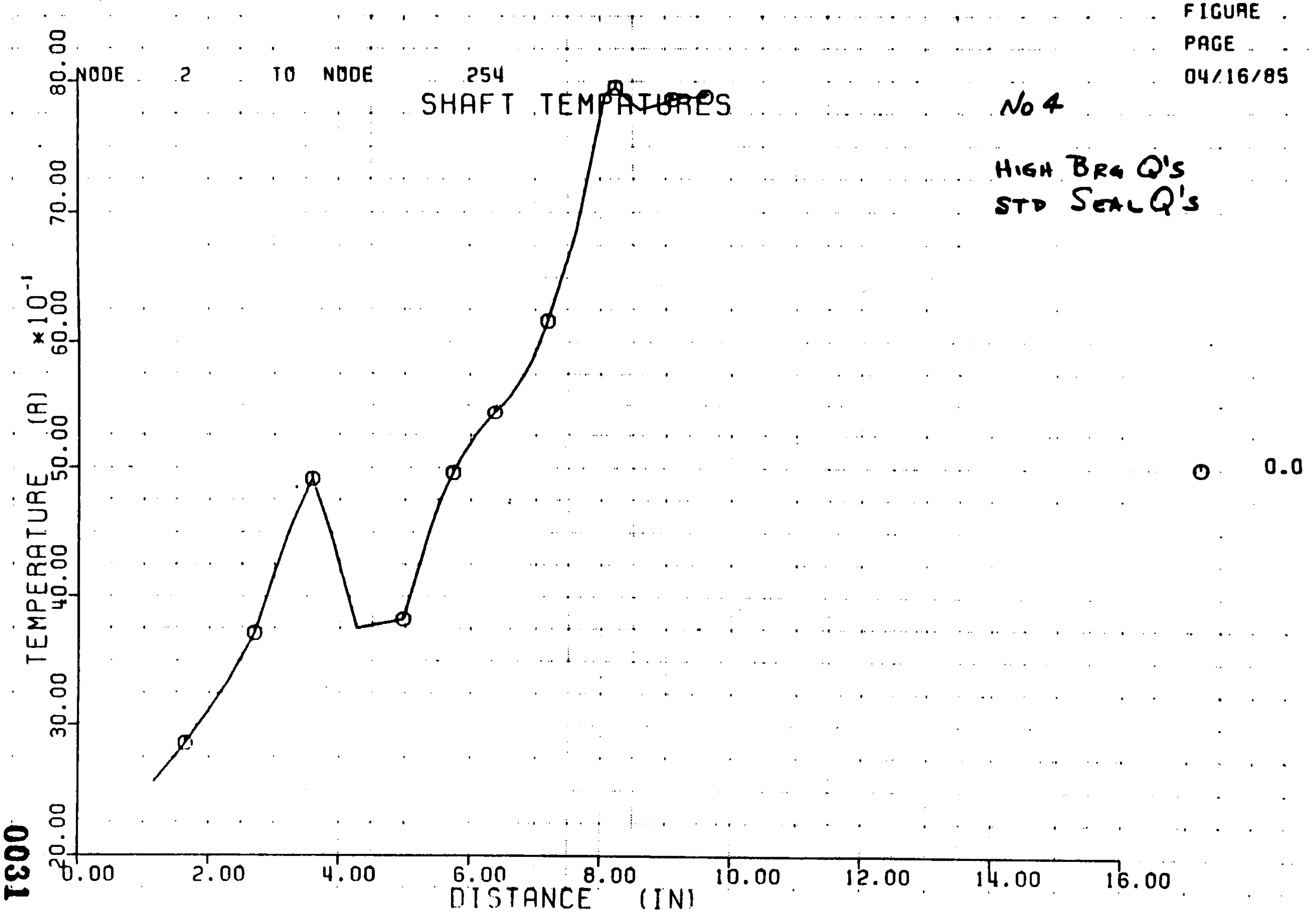
STEADY STATE TEMPS
RL10 GEAR RIG TEST

FTOM NO.
FIGURE
PAGE
04/16/85



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5130C

RL10 GEAR RIG TEST

C. H. A. O. S. DECK 8272 - (COMPUTERIZED HEAT ANALYSIS OF SYSTEMS)
NODE GEOMETRY SUMMARY --- 04/16/85

NODE	MAT ID	RADIUS	Z	WEIGHT	NODE	MAT ID	RADIUS	Z	WEIGHT
1	1	0.420	1.154	0.129	51	1	0.425	4.967	0.200
2	1	0.126	1.157	0.026	52	1	0.125	4.972	0.042
3	1	0.124	1.470	0.009	53	1	0.651	4.965	0.087
4	1	0.420	1.467	0.045	54	1	0.651	4.253	0.087
5	5	0.686	1.468	0.035	55	1	0.748	4.257	0.092
6	5	0.872	1.468	0.047	56	1	0.750	4.966	0.097
7	5	0.869	1.641	0.053	57	1	0.427	5.369	0.025
8	1	0.420	1.636	0.049	58	1	0.126	5.377	0.005
9	1	0.124	1.639	0.010	59	1	0.651	5.366	0.011
10	1	0.418	1.886	0.088	60	1	0.757	5.363	0.013
11	1	0.124	1.888	0.018	61	1	0.894	5.366	0.025
12	1	0.641	1.896	0.036	62	1	0.769	5.446	0.014
13	5	0.737	1.901	0.035	63	1	0.904	5.453	0.021
14	5	0.874	1.894	0.092	64	1	1.040	5.456	0.017
15	1	0.124	2.289	0.027	65	1	1.037	5.371	0.023
16	1	0.420	2.285	0.131	66	1	1.137	5.376	0.015
17	1	0.644	2.280	0.054	67	1	1.137	5.458	0.011
18	5	0.739	2.285	0.059	68	3	1.030	5.296	0.002
19	5	0.879	2.292	0.138	69	3	1.027	5.215	0.005
20	5	1.077	2.290	0.141	70	3	1.110	5.225	0.002
21	5	1.383	2.280	0.227	71	8	1.112	5.138	0.008
22	5	1.683	2.275	0.234	72	8	1.027	5.123	0.015
23	5	1.904	1.879	0.204	73	8	1.009	5.028	0.027
24	5	1.906	2.280	0.458	74	8	1.009	4.935	0.019
25	5	1.909	2.698	0.233	75	8	0.939	4.865	0.013
26	5	0.882	2.680	0.085	76	8	1.009	4.805	0.019
27	5	0.744	2.665	0.036	77	8	1.100	4.942	0.007
28	1	0.651	2.663	0.033	78	8	1.102	5.025	0.009
29	1	0.425	2.680	0.091	79	8	1.162	5.078	0.016
30	1	0.124	2.686	0.018	80	8	1.202	5.012	0.025
31	1	0.125	2.993	0.017	81	8	1.197	4.937	0.024
32	1	0.425	2.988	0.084	82	8	1.197	4.782	0.076
33	5	0.879	2.986	0.085	83	8	1.348	4.782	0.086
34	5	0.877	3.247	0.053	84	8	1.348	4.937	0.027
35	1	0.744	3.244	0.030	85	8	1.348	5.012	0.026
36	1	0.641	3.241	0.024	86	8	1.350	5.213	0.118
37	1	0.420	3.239	0.055	87	8	1.368	5.448	0.072
38	1	0.125	3.243	0.012	88	8	1.568	5.443	0.083
39	1	0.124	3.582	0.027	89	2	1.546	5.210	0.233
40	1	0.421	3.579	0.131	90	2	1.543	5.012	0.056
41	1	0.644	3.580	0.055	91	2	1.548	4.932	0.063
42	1	0.744	3.582	0.069	92	2	1.548	4.750	0.209
43	6	0.872	3.585	0.107	93	2	1.846	4.309	0.606
44	1	0.854	3.863	0.018	94	2	1.844	3.945	0.313
45	1	0.744	3.860	0.011	95	2	1.571	3.913	0.094
46	1	0.649	3.858	0.009	96	2	1.413	3.875	0.030
47	1	0.424	3.858	0.023	97	2	1.295	3.878	0.024
48	1	0.123	3.860	0.005	98	2	1.200	3.878	0.018
49	1	0.424	4.255	0.200	99	6	1.195	3.615	0.078
50	1	0.123	4.258	0.040	100	2	1.293	3.617	0.118

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C. H. A. O. S. D E C K 8 2 7 2 - (COMPUTERIZED HEAT ANALYSIS OF SYSTEMS)
NODE GEOMETRY SUMMARY --- 04/16/85

NODE	MAT ID	RADIUS	Z	WEIGHT	NODE	MAT ID	RADIUS	Z	WEIGHT
101	2	1.290	3.317	0.044	151	1	0.399	6.366	0.049
102	2	1.405	3.174	0.042	152	1	0.126	6.374	0.013
103	2	1.408	3.317	0.060	153	1	0.404	6.627	0.069
104	2	1.410	3.617	0.146	154	1	0.126	6.633	0.018
105	2	1.568	3.610	0.249	155	1	0.404	6.936	0.073
106	2	1.566	3.307	0.095	156	1	0.126	6.940	0.019
107	2	1.841	3.304	0.213	157	1	0.401	7.191	0.043
108	2	1.844	3.605	0.556	158	1	0.126	7.194	0.011
109	2	1.849	4.714	0.411	159	1	0.381	7.631	0.127
110	2	1.851	4.925	0.114	160	1	0.126	7.633	0.041
111	2	1.851	5.012	0.111	161	1	0.359	8.069	0.027
112	2	1.849	5.215	0.410	162	1	0.126	8.077	0.012
113	2	1.846	5.441	0.164	163	1	0.534	8.069	0.025
114	2	1.843	5.559	0.154	164	1	0.654	8.074	0.024
115	8	1.583	5.561	0.041	165	1	0.782	8.074	0.041
116	2	1.846	5.947	0.845	166	1	0.889	8.071	0.020
117	2	1.846	6.408	0.315	167	3	0.782	7.916	0.006
118	2	1.851	6.658	0.316	168	8	0.782	7.778	0.033
119	2	1.576	6.646	0.167	169	8	0.739	7.640	0.022
120	2	1.573	6.390	0.130	170	8	0.699	7.530	0.009
121	2	1.420	6.378	0.063	171	8	0.694	7.435	0.008
122	2	1.420	6.638	0.101	172	8	0.771	7.442	0.012
123	1	1.423	6.844	0.026	173	8	0.917	7.793	0.030
124	1	1.249	6.881	0.043	174	7	0.704	7.197	0.075
125	1	0.901	6.866	0.039	175	7	0.917	7.204	0.040
126	1	0.721	6.819	0.010	176	7	0.914	7.507	0.077
127	2	1.120	6.631	0.329	177	7	1.045	7.204	0.051
128	2	1.122	6.373	0.208	178	7	1.042	7.517	0.111
129	2	0.832	6.368	0.032	179	7	1.047	7.813	0.047
130	4	0.824	6.621	0.013	180	7	1.238	7.838	0.095
131	4	0.721	6.626	0.011	181	7	1.233	7.528	0.231
132	2	0.729	6.370	0.030	182	7	1.233	7.207	0.109
133	2	0.611	6.365	0.033	183	7	1.473	7.212	0.137
134	4	0.614	6.626	0.010	184	7	1.814	7.214	0.316
135	4	0.781	5.544	0.006	185	7	1.811	7.533	0.588
136	2	0.691	5.549	0.009	186	7	1.473	7.533	0.270
137	2	0.757	5.629	0.010	187	7	1.473	7.856	0.140
138	2	0.802	5.701	0.012	188	7	1.809	7.853	0.315
139	1	0.726	5.714	0.014	189	7	1.806	8.279	0.883
140	1	0.724	5.844	0.017	190	7	1.819	8.675	0.221
141	1	0.639	5.842	0.014	191	7	1.839	8.828	0.176
142	1	0.639	5.709	0.013	192	7	1.844	9.033	0.331
143	1	0.426	5.532	0.067	193	7	1.841	9.226	0.139
144	1	0.126	5.542	0.014	194	7	1.834	9.454	0.397
145	1	0.126	5.728	0.008	195	7	1.836	9.759	0.311
146	1	0.424	5.716	0.035	196	7	1.844	9.975	0.191
147	1	0.424	5.847	0.037	197	7	1.846	10.200	0.342
148	1	0.126	5.858	0.008	198	7	1.841	10.446	0.229
149	1	0.411	6.085	0.083	199	1	1.513	10.416	0.222
150	1	0.126	6.093	0.020	200	1	1.516	10.175	0.199

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RL10 GEAR RIG TEST

C. H. A. O. S. D E C K 8 2 7 2 - (COMPUTERIZED HEAT ANALYSIS OF SYSTEMS)
NODE GEOMETRY SUMMARY --- 04/16/85

NODE	MAT ID	RADIUS	Z	WEIGHT	NODE	MAT ID	RADIUS	Z	WEIGHT
201	1	1.513	9.983	0.132	251	1	0.356	9.254	0.018
202	1	1.508	9.764	0.233	252	1	0.125	9.259	0.008
203	1	1.503	9.456	0.279	253	1	0.358	9.384	0.019
204	1	1.510	9.231	0.096	254	1	0.125	9.392	0.008
205	1	1.513	9.041	0.219	255	1	0.358	9.610	[illegible]
206	1	1.505	8.835	0.117	256	1	0.126	9.621	0.017
207	1	1.262	8.833	0.066	257	1	0.358	9.840	0.020
208	9	1.265	9.043	0.130	258	1	0.126	9.854	0.008
209	9	1.280	9.238	0.041	259	1	0.361	9.980	0.021
210	9	1.300	9.459	0.079	260	1	0.125	9.990	0.009
211	9	1.300	9.769	0.075	261	1	0.356	10.200	0.042
212	9	1.273	9.993	0.063	262	1	0.125	10.205	0.018
213	9	1.257	10.153	0.078	263	1	0.356	10.408	0.016
214	1	0.884	10.418	0.037	264	1	0.125	10.418	0.007
215	9	0.884	10.205	0.095	265	1	0.361	10.652	0.056
216	9	0.867	9.988	0.029	266	1	0.126	10.663	0.022
217	9	0.867	9.857	0.032					
218	9	0.889	9.622	0.098					
219	9	0.874	9.391	0.030					
220	9	0.877	9.261	0.034					
221	9	0.894	9.108	0.050					
222	1	0.887	8.960	0.038					
223	1	0.701	8.958	0.032					
224	1	0.709	9.106	0.041					
225	1	0.706	9.259	0.033					
226	1	0.706	9.389	0.029					
227	1	0.709	9.617	0.078					
228	1	0.709	9.852	0.030					
229	1	0.711	9.985	0.028					
230	1	0.709	10.205	0.064					
231	1	0.711	10.416	0.022					
232	1	0.549	10.411	0.020					
233	1	0.546	10.203	0.053					
234	1	0.551	9.982	0.022					
235	1	0.546	9.845	0.023					
236	1	0.544	9.612	0.052					
237	1	0.541	9.386	0.019					
238	1	0.539	9.256	0.021					
239	1	0.539	9.103	0.027					
240	1	0.534	8.955	0.018					
241	1	0.534	8.592	0.081					
242	1	0.534	8.226	0.018					
243	1	0.358	8.232	0.020					
244	1	0.126	8.245	0.008					
245	1	0.358	8.599	0.086					
246	1	0.125	8.607	0.035					
247	1	0.356	8.959	0.017					
248	1	0.125	8.968	0.008					
249	1	0.356	9.104	0.024					
250	1	0.125	9.114	0.009					

0034

5130C

RL10 GEAR RIG TEST

TEMPERATURE OUTPUT SUMMARY --- 04/15/85
OUTPUT TEMPERATURES IN DEGREES R

TIME	1.0		2.0		3.0		4.0								
RPM	27000.														
NODE	TEMP S.S.	TEMP	TEMP S.S.	TEMP	TEMP S.S.	TEMP	TEMP S.S.	TEMP	TEMP	TEMP	TEMP	TEMP	TEMP	TEMP	TEMP
1	239.12		239.17		252.13		252.19								
2	240.52		240.58		255.55		255.61								
3	248.44		248.56		274.76		274.88								
4	247.90		248.00		273.08		273.19								
5	246.00		246.06		266.65		266.71								
6	247.21		247.27		269.27		269.34								
7	250.82		250.90		277.46		277.54								
8	252.90		253.04		285.28		285.42								
9	253.09		253.24		286.18		286.33								
10	260.39		260.58		303.26		303.45								
11	260.11		260.32		303.79		304.01								
12	261.20		261.35		302.70		302.85								
13	260.42		260.54		299.56		299.69								
14	259.01		259.12		296.07		296.18								
15	270.29		270.68		333.52		333.91								
16	270.75		271.08		331.98		332.32								
17	272.32		272.54		329.81		330.04								
18	273.63		273.80		329.85		330.03								
19	275.08		275.23		331.58		331.73								
20	281.14		281.25		340.33		340.44								
21	309.56		309.60		392.30		392.35								
22	374.16		374.18		521.10		521.12								
23	401.61		401.62		575.83		575.85								
24	415.29		415.31		605.05		605.07								
25	392.45		392.46		557.12		557.13								
26	268.52		268.81		332.16		332.46								
27	269.42		269.73		333.59		333.90								
28	271.60		272.03		342.09		342.52								
29	276.47		277.22		363.61		364.35								
30	277.89		278.76		370.32		371.18								
31	287.47		289.02		415.19		416.68								
32	287.22		288.67		414.51		415.91								
33	269.76		270.21		353.40		353.85								
34	280.66		281.36		403.47		404.19								
35	292.10		293.30		448.76		449.95								
36	295.87		297.37		461.50		462.95								
37	296.34		298.36		456.73		458.65								
38	295.54		297.81		451.11		453.24								
39	304.99		308.68		487.77		491.12								
40	309.34		312.59		507.83		510.78								
41	321.32		323.65		559.24		561.38								
42	328.33		330.28		587.39		589.21								
43	345.60		346.95		660.26		661.55								
44	297.84		299.70		480.81		482.59								
45	299.95		303.12		481.96		484.82								
46	299.25		303.37		473.32		476.98								
47	297.47		303.30		452.55		457.77								
48	296.78		303.29		443.40		449.27								
49	279.47		289.90		355.21		365.30								
50	284.17		295.50		364.45		375.36								

TIME	BEARING Q	SEAL Q
1	STD	REDUCED *
2	STD	STD
3	HIGH	REDUCED *
4	HIGH	STD

* 6 lb SPRING FORCE

0035

5130C

RL10 GEAR RIG TEST

TEMPERATURE OUTPUT SUMMARY --- 04/15/85
OUTPUT TEMPERATURES IN DEGREES R

TIME	1.0		2.0		3.0		4.0							
RPM	27000.													
NODE	TEMP S.S.	TEMP	TEMP S.S.	TEMP	TEMP S.S.	TEMP	TEMP S.S.	TEMP	TEMP	TEMP	TEMP	TEMP	TEMP	TEMP
51	318.99		351.73		341.35		373.80							
52	326.19		357.92		350.98		382.36							
53	302.21		335.32		319.27		352.25							
54	268.56		276.75		331.20		339.28							
55	263.39		270.50		318.84		325.94							
56	293.64		324.76		308.31		339.40							
57	404.24		448.88		416.66		461.00							
58	404.35		438.88		418.03		452.35							
59	407.53		517.21		416.47		525.37							
60	434.98		582.25		440.85		587.53							
61	470.16		661.67		472.69		664.02							
62	454.76		615.16		459.42		619.39							
63	488.43		679.28		491.06		681.72							
64	567.98		815.71		569.30		816.97							
65	598.23		907.23		599.44		908.36							
66	574.09		782.01		574.94		782.87							
67	566.16		756.80		567.03		757.68							
68	543.33		898.14		543.35		898.17							
69	460.04		649.36		460.08		649.40							
70	466.19		589.90		466.23		589.94							
71	426.50		500.85		426.58		500.93							
72	397.92		499.51		397.98		499.57							
73	345.00		398.99		345.09		399.07							
74	312.07		346.69		312.15		346.77							
75	258.26		270.63		258.29		270.66							
76	238.36		242.06		238.37		242.07							
77	330.83		360.77		330.97		360.91							
78	362.21		402.73		362.34		402.86							
79	404.41		444.29		404.54		444.42							
80	371.27		401.14		371.47		401.34							
81	344.72		367.94		344.95		368.16							
82	291.26		300.40		291.51		300.64							
83	306.19		313.58		306.63		314.01							
84	358.48		370.14		358.90		370.56							
85	384.94		397.01		385.34		397.40							
86	464.54		468.55		464.86		468.87							
87	499.54		501.31		499.74		501.51							
88	481.43		483.52		481.77		483.86							
89	442.43		446.11		442.96		446.64							
90	388.65		394.02		389.37		394.74							
91	366.16		371.64		366.95		372.43							
92	313.43		317.73		314.33		318.63							
93	262.86		263.83		268.52		269.48							
94	250.69		251.17		267.88		268.34							
95	242.28		242.47		262.54		262.77							
96	242.80		242.89		274.21		274.30							
97	248.32		248.36		301.46		301.50							
98	249.90		249.92		307.45		307.47							
99	289.41		289.44		459.55		459.59							
100	271.89		271.94		390.05		390.11							

0036

RL10 GEAR RIG TEST

TEMPERATURE OUTPUT SUMMARY --- 04/15/85
OUTPUT TEMPERATURES IN DEGREES R

TIME	1.0		2.0		3.0		4.0	
RPM	27000.							
NODE	TEMP S.S.	TEMP	TEMP S.S.	TEMP	TEMP S.S.	TEMP	TEMP S.S.	TEMP TEMP TEMP TEMP TEMP TEMP TEMP
101	238.80		238.82		264.83		264.85	
102	232.57		232.58		239.86		239.87	
103	237.75		237.78		259.43		259.46	
104	255.43		255.53		322.59		322.69	
105	248.77		248.91		292.59		292.73	
106	237 34		237.40		254.70		254.76	
107	236.03		236.11		247.15		247.23	
108	245.23		245.44		271.06		271.27	
109	330.76		333.79		333.08		336.10	
110	373.97		377.27		375.50		378.79	
111	391.29		394.55		392.58		395.84	
112	429.66		432.59		430.50		433.42	
113	465.08		467.17		465.61		467.70	
114	480.16		481.85		480.58		482.26	
115	498.03		499.45		498.27		499.70	
116	521.15		521.68		521.28		521.81	
117	531.95		532.17		532.00		532.23	
118	535.19		535.32		535.22		535.35	
119	536.94		537.03		536.96		537.05	
120	536.27		536.38		536.30		536.40	
121	537.50		537.57		537.52		537.59	
122	537.60		537.66		537.61		537.68	
123	538.76		538.80		538.77		538.80	
124	539.52		539.53		539.52		539.53	
125	539.86		539.87		539.86		539.87	
126	539.88		539.88		539.88		539.88	
127	538.36		538.40		538.37		538.42	
128	538.82		538.86		538.83		538.86	
129	539.36		539.38		539.36		539.38	
130	538.67		538.71		538.68		538.72	
131	539.22		539.24		539.23		539.28	
132	539.50		539.52		539.51		539.52	
133	539.61		539.62		539.62		539.63	
134	539.65		539.66		539.66		539.67	
135	497.70		554.88		500.56		557.61	
136	507.33		527.89		510.11		530.65	
137	508.44		524.83		511.21		527.61	
138	511.89		525.02		514.39		527.53	
139	507.12		521.80		510.07		524.77	
140	509.49		522.07		512.26		524.87	
141	504.53		518.70		507.78		521.98	
142	499.60		515.94		503.28		519.63	
143	435.39		470.00		445.04		479.50	
144	433.53		465.20		443.82		475.35	
145	463.65		488.70		470.98		495.99	
146	470.29		494.69		476.77		501.14	
147	487.60		507.07		492.58		512.05	
148	481.21		502.13		486.95		507.86	
149	510.33		523.77		513.51		526.97	
150	507.04		521.98		510.64		528.59	

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RL10 GEAR RIG TEST

TEMPERATURE OUTPUT SUMMARY --- 04/15/85
OUTPUT TEMPERATURES IN DEGREES R

TIME	1.0		2.0		3.0		4.0								
RPM	27000.														
NODE	TEMP	TEMP	TEMP	TEMP	TEMP	TEMP	TEMP	TEMP	TEMP	TEMP	TEMP	TEMP	TEMP	TEMP	TEMP
	S.S.		S.S.		S.S.		S.S.								
151	530.09		540.62		532.03		542.58								
152	528.93		540.76		531.13		542.97								
153	546.17		556.04		547.36		557.25								
154	546.89		558.02		548.24		559.39								
155	566.17		578.13		566.86		578.84								
156	569.69		583.25		570.47		584.04								
157	590.48		607.55		590.96		608.04								
158	596.04		615.09		596.58		615.63								
159	650.37		681.27		650.61		681.52								
160	652.91		685.22		653.17		685.48								
161	752.32		820.93		752.43		821.03								
162	732.06		789.59		732.18		789.71								
163	801.86		895.40		801.92		895.46								
164	851.18		970.36		851.23		970.41								
165	906.54		1051.68		906.58		1051.72								
166	881.76		1017.36		881.80		1017.40								
167	921.61		1088.21		921.61		1088.21								
168	727.06		793.95		727.06		793.95								
169	699.97		752.46		699.97		752.46								
170	677.58		717.74		677.58		717.74								
171	661.81		693.04		661.81		693.04								
172	656.04		683.96		656.04		683.96								
173	678.80		723.67		678.80		723.67								
174	563.87		565.50		563.87		565.50								
175	567.42		569.94		567.42		569.94								
176	614.36		628.88		614.36		628.88								
177	569.86		573.09		569.86		573.09								
178	610.99		624.71		610.99		624.71								
179	631.54		654.14		631.54		654.14								
180	613.19		627.17		613.19		627.17								
181	594.37		603.10		594.37		603.10								
182	574.82		579.56		574.82		579.56								
183	574.37		578.78		574.37		578.78								
184	572.83		576.44		572.83		576.44								
185	586.43		591.23		586.43		591.23								
186	589.33		595.86		589.33		595.86								
187	598.55		605.53		598.55		605.53								
188	597.87		602.43		597.87		602.43								
189	612.47		618.33		612.47		618.33								
190	626.49		628.40		626.49		628.40								
191	632.91		634.47		632.91		634.47								
192	640.01		641.21		640.01		641.21								
193	644.31		645.26		644.31		645.26								
194	648.09		648.78		648.09		648.78								
195	648.28		648.74		648.28		648.74								
196	645.75		646.13		645.75		646.13								
197	641.95		642.25		641.95		642.25								
198	630.84		631.01		630.84		631.01								
199	632.68		632.83		632.68		632.83								
200	647.28		647.49		647.28		647.49								

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RL10 GEAR RIG TEST

TEMPERATURE OUTPUT SUMMARY --- 04/15/85
OUTPUT TEMPERATURES IN DEGREES R

TIME	1.0		2.0		3.0		4.0							
RPM	27000.													
NODE	TEMP S.S.	TEMP	TEMP S.S.	TEMP	TEMP S.S.	TEMP	TEMP S.S.	TEMP	TEMP	TEMP	TEMP	TEMP	TEMP	TEMP
201	655.04		655.29		655.04		655.29							
202	658.62		658.93		658.62		658.93							
203	659.68		660.11		659.68		660.11							
204	658.38		658.92		658.38		658.92							
205	654.40		655.02		654.40		655.02							
206	648.54		649.17		648.54		649.17							
207	651.56		651.96		651.56		651.96							
208	667.16		667.52		667.16		667.52							
209	681.43		681.77		681.43		681.77							
210	670.63		67[illegible].92		670.63		670.92							
211	670.45		670.65		670.45		670.65							
212	678.87		679.05		678.87		679.05							
213	668.50		668.66		668.50		668.66							
214	750.44		754.35		750.44		754.36							
215	769.38		773.68		769.38		773.68							
216	789.57		794.22		789.56		794.22							
217	814.77		819.81		814.77		819.81							
218	807.09		812.65		807.09		812.66							
219	818.19		824.34		818.19		824.34							
220	797.39		803.91		797.39		803.92							
221	785.28		792.06		785.28		792.07							
222	766.47		773.73		766.47		773.74							
223	769.81		779.36		769.82		779.37							
224	779.88		788.94		779.89		788.95							
225	789.39		797.62		789.39		797.63							
226	796.34		803.87		796.34		803.88							
227	798.08		804.50		798.08		804.51							
228	794.02		799.62		794.02		799.63							
229	784.49		789.66		784.48		789.66							
230	768.23		772.83		768.23		772.84							
231	754.30		758.51		754.30		758.52							
232	753.82		758.24		753.82		758.24							
233	766.10		770.93		766.10		770.93							
234	779.66		785.17		779.66		785.18							
235	786.58		792.60		786.58		792.61							
236	792.31		799.37		792.31		799.38							
237	789.10		797.78		789.11		797.79							
238	784.38		794.17		784.38		794.19							
239	776.90		788.36		776.90		788.37							
240	768.65		782.56		768.66		782.57							
241	729.21		765.86		729.25		765.90							
242	765.41		835.53		765.47		835.59							
243	751.92		813.89		752.00		813.97							
244	740.03		794.90		740.12		794.99							
245	736.18		774.06		736.22		774.11							
246	739.34		777.50		739.39		777.55							
247	761.19		782.73		761.21		782.76							
248	759.79		783.11		759.81		783.14							
249	769.47		786.22		769.48		786.24							
250	767.11		785.57		767.12		785.59							

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RL10 GEAR RIG TEST

TEMPERATURE OUTPUT SUMMARY --- 04/15/85
OUTPUT TEMPERATURES IN DEGREES R

TIME	1.0		2.0		3.0		4.0							
RPM	27000.													
NODE	TEMP S.S.	TEMP	TEMP S.S.	TEMP	TEMP S.S.	TEMP	TEMP S.S.	TEMP	TEMP	TEMP	TEMP	TEMP	TEMP	TEMP
251	775.67		789.15		775.68		789.16							
252	772.60		787.52		772.62		787.54							
253	779.31		790.66		779.32		790.67							
254	775.99		788.43		776.00		788.45							
255	782.14		790.76		782.14		790.77							
256	778.64		787.87		778.64		787.88							
257	776.49		783.50		776.49		783.51							
258	773.17		780.56		773.17		780.57							
259	770.99		777.25		770.99		777.25							
260	767.98		774.53		767.98		774.54							
261	759.23		764.51		759.23		764.52							
262	756.97		762.43		756.97		762.43							
263	742.20		746.80		742.20		746.81							
264	741.46		746.20		741.46		746.20							
265	719.08		722.95		719.09		722.96							
266	722.69		726.71		722.69		726.72							

0040

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5130C

REFERENCE 18

RL10 / HEAT TRANSFER

PRATT & WHITNEY AIRCRAFT

DON'T GIVE VERBAL ORDERS

MEMO. TO YUL EREN DATE 5/17/85

SUBJECT: RL10 GEAR TEST RIG SHAFT TEMPERATURES REVISED USING F100 TEST DATA.

CC: R. DUCAN, H.L. HESS, J.D. DOERNBACH, & E.M. BEVERLY

REF: FTDM #1900, "THERMAL ANALYSIS OF THE F100(3) AND F100 DERIVATIVE II No 3 BEARING COMPARTMENT IS COMPLETE", R.J. PECKHAM 1/5/81

THE TEMPERATURES OF THE SHAFT AND HOUSING OF THE RL10 GEAR TEST RIG HAVE BEEN REVISED BASED ON THE TEST DATA COMPARISON USED IN THE REFERENCE FTDM. ALL SEAL SPRING LOADS HAVE BEEN SET AT APPROXIMATELY 6 lb. THE VARIATION IN THE ROLLER BEARING HEAT GENERATION CAUSES LESS THAN A 3 °F VARIATION IN TEMPERATURES, HENCE, ONLY ONE SET OF TEMPERATURES.

MC&S

CDT

X2937

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5130C

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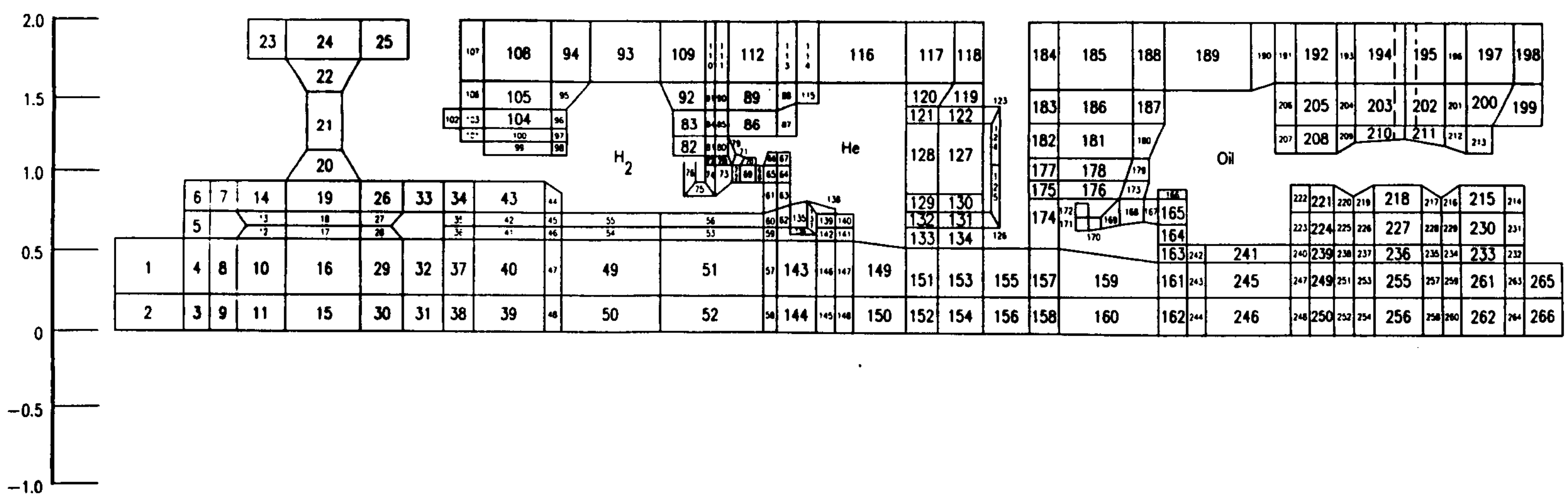
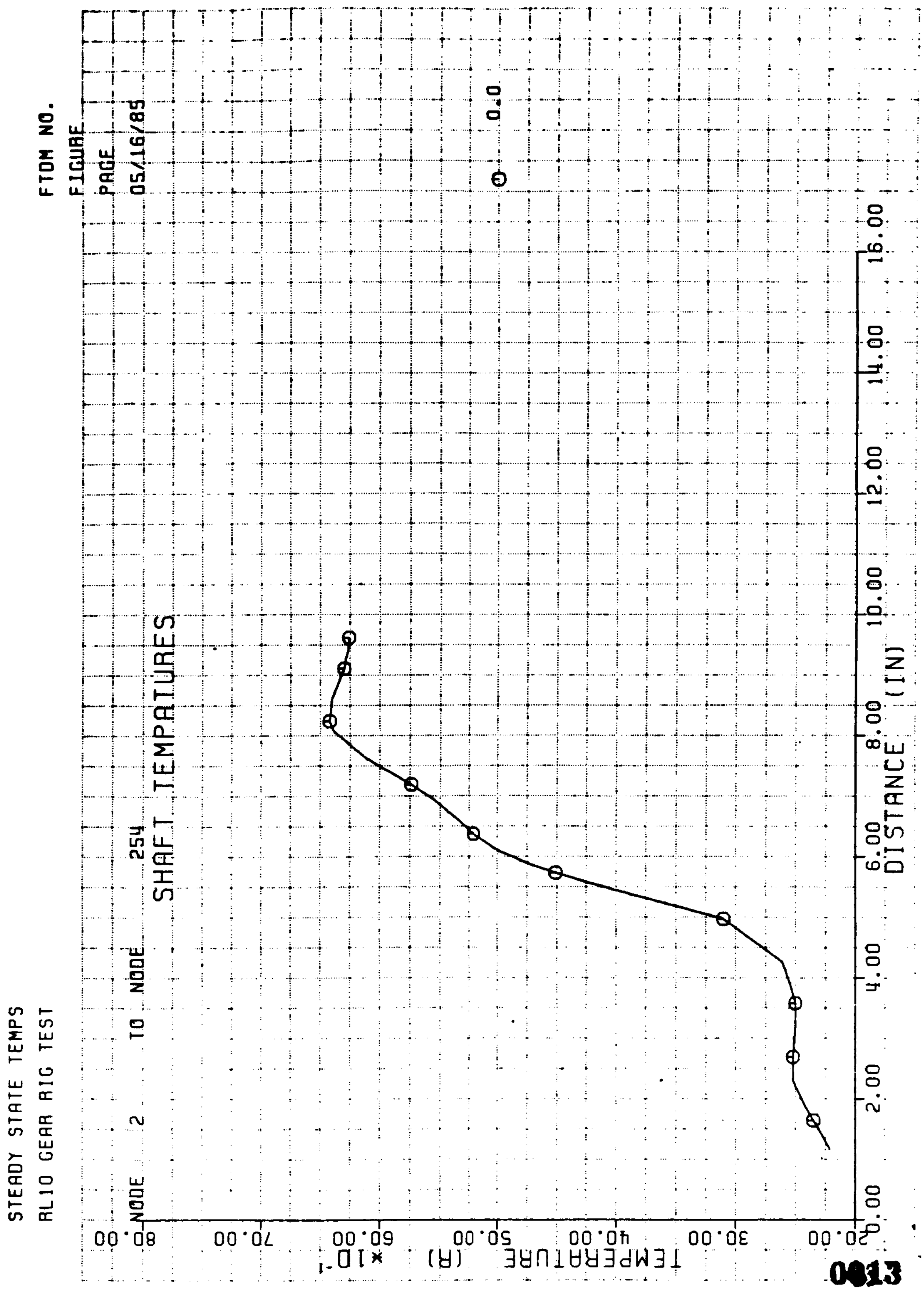


FOLDOUT FRAME

2 FOLDOUT FRAME


STEADY STATE TEMPS
RL10 GEAR RIG TEST
NODE 2 TO NODE 254
SHAFT TEMPATURES
FTDM NO.
FIGURE
PAGE
05/16/85
0.0
TEMPERATURE (R) *10⁻¹
DISTANCE (IN)
0.00
2.00
4.00
6.00
8.00
10.00
12.00
14.00
16.00
20.00
30.00
40.00
50.00
60.00
70.00
80.00

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5130C

RL10 GEAR RIG TEST

C. H. A. O. S. D E C K 8 2 7 2 - (COMPUTERIZED HEAT ANALYSIS OF SYSTEMS)
NODE GEOMETRY SUMMARY --- 05/16/85

NODE	MAT ID	RADIUS	Z	WEIGHT	NODE	MAT ID	RADIUS	Z	WEIGHT
1	1	0.420	1.154	0.129	51	1	0.425	4.967	0.200
2	1	0.126	1.157	0.026	52	1	0.125	4.972	0.042
3	1	0.124	1.470	0.009	53	1	0.651	4.965	0.087
4	1	0.420	1.467	0.045	54	1	0.651	4.253	0.087
5	5	0.686	1.468	0.035	55	1	0.748	4.257	0.092
6	5	0.872	1.468	0.047	56	1	0.750	4.966	0.097
7	5	0.869	1.641	0.053	57	1	0.427	5.369	0.025
8	1	0.420	1.636	0.049	58	1	0.126	5.377	0.005
9	1	0.124	1.639	0.010	59	1	0.651	5.366	0.011
10	1	0.418	1.886	0.088	60	1	0.757	5.363	0.013
11	1	0.124	1.888	0.018	61	1	0.894	5.366	0.025
12	1	0.641	1.896	0.036	62	1	0.769	5.446	0.014
13	5	0.737	1.901	0.035	63	1	0.904	5.453	0.021
14	5	0.874	1.894	0.092	64	1	1.040	5.456	0.017
15	1	0.124	2.289	0.027	65	1	1.037	5.371	0.023
16	1	0.420	2.285	0.131	66	1	1.137	5.376	0.015
17	1	0.644	2.280	0.054	67	1	1.137	5.458	0.011
18	5	0.739	2.285	0.059	68	3	1.030	5.296	0.002
19	5	0.879	2.292	0.138	69	3	1.027	5.215	0.005
20	5	1.077	2.290	0.141	70	3	1.110	5.225	0.002
21	5	1.383	2.280	0.227	71	8	1.112	5.138	0.008
22	5	1.683	2.275	0.234	72	8	1.027	5.123	0.015
23	5	1.904	1.879	0.204	73	8	1.009	5.028	0.027
24	5	1.906	2.280	0.458	74	8	1.009	4.935	0.019
25	5	1.909	2.698	0.233	75	8	0.939	4.865	0.013
26	5	0.882	2.680	0.085	76	8	1.009	4.805	0.019
27	5	0.744	2.665	0.036	77	8	1.100	4.942	0.007
28	1	0.651	2.663	0.033	78	8	1.102	5.025	0.009
29	1	0.425	2.680	0.091	79	8	1.162	5.078	0.016
30	1	0.124	2.686	0.018	80	8	1.202	5.012	0.025
31	1	0.125	2.993	0.017	81	8	1.197	4.937	0.024
32	1	0.425	2.988	0.084	82	8	1.197	4.782	0.076
33	5	0.879	2.986	0.085	83	8	1.348	4.782	0.086
34	5	0.877	3.247	0.053	84	8	1.348	4.937	0.027
35	1	0.744	3.244	0.030	85	8	1.348	5.012	0.026
36	1	0.641	3.241	0.024	86	8	1.350	5.213	0.115
37	1	0.420	3.239	0.055	87	8	1.368	5.448	0.072
38	1	0.125	3.243	0.012	88	8	1.568	5.443	0.083
39	1	0.124	3.582	0.027	89	2	1.546	5.210	0.233
40	1	0.421	3.579	0.131	90	2	1.543	5.012	0.056
41	1	0.844	3.580	0.055	91	2	1.548	4.932	0.063
42	1	0.744	3.582	0.069	92	2	1.548	4.750	0.209
43	6	0.872	3.585	0.107	93	2	1.846	4.309	0.606
44	1	0.854	3.863	0.018	94	2	1.844	3.945	0.313
45	1	0.744	3.860	0.011	95	2	1.371	3.913	0.094
46	1	0.649	3.858	0.009	96	2	1.413	3.875	0.030
47	1	0.424	3.858	0.023	97	2	1.295	3.878	0.024
48	1	0.123	3.860	0.005	98	2	1.200	3.878	0.018
49	1	0.424	4.255	0.200	99	8	1.193	3.815	0.078
50	1	0.123	4.258	0.040	100	2	1.293	3.617	0.118

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RL10 GEAR RIG TEST

C. H. &. O. S. DECK 8272 - (COMPUTERIZED HEAT ANALYSIS OF SYSTEMS)
NODE GEOMETRY SUMMARY --- 05/16/85

NODE	MAT ID	RADIUS	Z	WEIGHT	NODE	MAT ID	RADIUS	Z	WEIGHT
101	2	1.290	3.317	0.044	151	1	0.399	6.366	0.049
102	2	1.405	3.174	0.042	152	1	0.126	6.374	0.013
103	2	1.408	3.317	0.060	153	1	0.404	6.627	0.069
104	2	1.410	3.617	0.146	154	1	0.126	6.633	0.018
105	2	1.568	3.610	0.249	155	1	0.404	6.936	0.073
106	2	1.566	3.307	0.095	156	1	0.126	6.940	0.019
107	2	1.841	3.304	0.213	157	1	0.401	7.191	0.043
108	2	1.844	3.605	0.556	158	1	0.126	7.194	0.011
109	2	1.849	4.714	0.411	159	1	0.381	7.631	0.127
110	2	1.851	4.925	0.114	160	1	0.126	7.633	0.041
111	2	1.851	5.012	0.111	161	1	0.359	8.069	0.027
112	2	1.849	5.215	0.410	162	1	0.126	8.077	0.012
113	2	1.846	5.441	0.164	163	1	0.534	8.069	0.025
114	2	1.843	5.559	0.134	164	1	0.654	8.074	0.024
115	8	1.583	5.561	0.041	165	1	0.782	8.074	0.041
116	2	1.846	5.947	0.845	166	1	0.889	8.071	0.020
117	2	1.846	6.408	0.315	167	5	0.782	7.916	0.006
118	2	1.851	6.658	0.316	168	8	0.782	7.778	0.033
119	2	1.576	6.646	0.167	169	8	0.739	7.640	0.022
120	2	1.573	6.390	0.130	170	8	0.699	7.530	0.009
121	2	1.420	6.378	0.063	171	8	0.694	7.435	0.008
122	2	1.420	6.638	0.101	172	8	0.771	7.442	0.012
123	1	1.423	6.844	0.026	173	8	0.917	7.793	0.030
124	1	1.249	6.881	0.043	174	7	0.704	7.197	0.075
125	1	0.901	6.866	0.039	175	7	0.917	7.204	0.040
126	1	0.721	6.819	0.010	176	7	0.914	7.507	0.077
127	2	1.120	6.631	0.329	177	7	1.045	7.204	0.051
128	2	1.122	6.373	0.208	178	7	1.042	7.517	0.111
129	2	0.832	6.368	0.032	179	7	1.047	7.813	0.047
130	4	0.824	6.621	0.013	180	7	1.238	7.838	0.095
131	4	0.721	6.626	0.011	181	7	1.233	7.528	0.231
132	2	0.729	6.370	0.030	182	7	1.233	7.207	0.109
133	2	0.611	6.365	0.033	183	7	1.473	7.212	0.137
134	4	0.614	6.626	0.010	184	7	1.816	7.214	0.316
135	4	0.781	5.544	0.006	185	7	1.811	7.533	0.588
136	2	0.691	5.549	0.009	186	7	1.473	7.533	0.270
137	2	0.757	5.629	0.010	187	7	1.473	7.856	0.140
138	2	0.802	5.701	0.012	188	7	1.809	7.853	0.315
139	1	0.726	5.714	0.014	189	7	1.806	8.279	0.883
140	1	0.724	5.844	0.017	190	7	1.819	8.675	0.221
141	1	0.839	5.842	0.014	191	7	1.859	8.828	0.176
142	1	0.639	5.709	0.013	192	7	1.844	9.033	0.331
143	1	0.426	5.532	0.067	193	7	1.841	9.226	0.139
144	1	0.126	5.542	0.014	194	7	1.834	9.454	0.397
145	1	0.126	5.728	0.008	195	7	1.856	9.759	0.311
146	1	0.424	5.716	0.035	196	7	1.844	9.975	0.191
147	1	0.424	5.847	0.037	197	7	1.846	10.200	0.342
148	1	0.126	5.858	0.008	198	7	1.241	10.446	0.229
149	1	0.411	6.085	0.085	199	1	1.515	10.416	0.222
150	1	0.126	6.093	0.020	200	1	1.516	10.175	0.199

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RL10 GEAR RIG TEST

C. H. A. O. S.   D E C K   8 2 7 2 - (COMPUTERIZED HEAT ANALYSIS OF SYSTEMS)
NODE GEOMETRY SUMMARY --- 05/16/85

NODE	MAT ID	RADIUS	Z	WEIGHT	NODE	MAT ID	RADIUS	Z	WEIGHT
201	1	1.513	9.983	0.132	251	1	0.356	9.254	0.018
202	1	1.508	9.764	0.233	252	1	0.125	9.259	0.008
203	1	1.503	9.456	0.279	253	1	0.358	9.384	0.019
204	1	1.510	9.231	0.096	254	1	0.125	9.392	0.008
205	1	1.513	9.041	0.219	255	1	0.358	9.610	0.046
206	1	1.505	8.835	0.117	256	1	0.126	9.621	0.019
207	1	1.262	8.833	0.066	257	1	0.358	9.840	0.020
208	9	1.265	9.043	0.130	258	1	0.126	9.854	0.008
209	9	1.280	9.238	0.041	259	1	0.361	9.980	0.021
210	9	1.300	9.459	0.079	260	1	0.125	9.990	0.008
211	9	1.300	9.769	0.075	261	1	0.356	10.200	0.042
212	9	1.273	9.993	0.063	262	1	0.125	10.205	0.018
213	9	1.257	10.153	0.078	263	1	0.356	10.408	0.016
214	1	0.884	10.418	0.037	264	1	0.125	10.418	0.007
215	9	0.884	10.205	0.095	265	1	0.361	10.652	0.056
216	9	0.867	9.988	0.029	266	1	0.126	10.663	0.022
217	9	0.867	9.857	0.032	267	9	1.197	9.238	0.0
218	9	0.889	9.622	0.098	268	9	0.942	9.857	0.0
219	9	0.874	9.391	0.030	269	9	0.952	9.391	0.0
220	9	0.877	9.261	0.034	270	1	0.782	8.074	0.0
221	9	0.894	9.108	0.050	271	9	1.182	9.993	0.0
222	1	0.887	8.960	0.038	272	3	0.782	7.916	0.0
223	1	0.701	8.958	0.032	273	6	1.152	3.615	0.0
224	1	0.709	9.106	0.041	274	5	2.029	2.280	0.0
225	1	0.706	9.259	0.033	275	6	0.947	3.585	0.0
226	1	.0.706	9.389	0.029	276	5	2.029	2.698	0.0
227	1	0.709	9.617	0.078	277	5	2.024	1.879	0.0
228	1	0.709	9.852	0.030	278	1	1.037	5.371	0.0
229	1	0.711	9.985	0.028	279	5	1.030	5.296	0.0
230	1	0.709	10.205	0.064	280	9	0.957	9.264	0.0
231	1	0.711	10.416	0.022	281	9	1.248	9.459	0.0
232	1	0.549	10.411	0.020	282	9	1.243	9.774	0.0
233	1	0.546	10.203	0.053	283	9	0.942	9.990	0.0
234	1	0.551	9.982	0.022					
235	1	0.546	9.845	0.023					
236	1	0.544	9.612	0.052					
237	1	0.541	9.386	0.019					
238	1	0.539	9.256	0.021					
239	1	0.539	9.103	0.027					
240	1	0.534	8.955	0.018					
241	1	0.534	8.592	0.081					
242	1	0.534	8.226	0.018					
243	1	0.358	8.232	0.020					
244	1	0.126	8.245	0.008					
245	1	0.358	8.599	0.086					
246	1	0.125	8.607	0.035					
247	1	0.356	8.959	0.017					
248	1	0.125	8.968	0.008					
249	1	0.356	9.104	0.024					
250	1	0.125	9.114	0.009					

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RL10 GEAR RIG TEST

**************** NODE TEMPERATURES ****************05/16/85

NODE NO.	TIME (SEC)	TEMP (R)	NODE NO.	TIME (SEC)	TEMP (R)	NODE NO.	TIME (SEC)	TEMP (R)	NODE NO.	TIME (SEC)	TEMP (R)	NODE NO.	TIME (SEC)	TEMP (R)	NODE NO.	TIME (SEC)	TEMP (R)
1	0.0	219.7	41	0.0	246.9	81	0.0	339.3	121	0.0	537.5	161	0.0	646.1	201	0.0	622.8
2	0.0	221.2	42	0.0	246.1	82	0.0	294.1	122	0.0	537.6	162	0.0	639.0	202	0.0	620.9
3	0.0	230.0	43	0.0	244.6	83	0.0	309.3	123	0.0	538.8	163	0.0	662.2	203	0.0	619.5
4	0.0	229.5	44	0.0	243.9	84	0.0	357.9	124	0.0	539.5	164	0.0	675.9	204	0.0	619.7
5	0.0	227.7	45	0.0	246.6	85	0.0	383.4	125	0.0	539.9	165	0.0	689.9	205	0.0	620.0
6	0.0	229.1	46	0.0	248.5	86	0.0	464.6	126	0.0	539.9	166	0.0	685.7	206	0.0	621.4
7	0.0	233.3	47	0.0	252.3	87	0.0	499.7	127	0.0	538.4	167	0.0	691.6	207	0.0	621.5
8	0.0	235.0	48	0.0	254.1	88	0.0	481.9	128	0.0	538.8	168	0.0	648.8	208	0.0	619.8
9	0.0	235.0	49	0.0	258.5	89	0.0	443.1	129	0.0	539.4	169	0.0	647.4	209	0.0	619.7
10	0.0	243.2	50	0.0	261.2	90	0.0	390.0	130	0.0	538.7	170	0.0	646.3	210	0.0	619.4
11	0.0	242.5	51	0.0	303.4	91	0.0	368.1	131	0.0	539.2	171	0.0	645.5	211	0.0	620.6
12	0.0	244.9	52	0.0	310.0	92	0.0	317.5	132	0.0	539.5	172	0.0	645.2	212	0.0	621.9
13	0.0	244.4	53	0.0	288.6	93	0.0	268.1	133	0.0	539.6	173	0.0	632.9	213	0.0	622.5
14	0.0	242.8	54	0.0	253.2	94	0.0	252.9	134	0.0	539.7	174	0.0	569.6	214	0.0	623.3
15	0.0	251.7	55	0.0	250.9	95	0.0	243.8	135	0.0	470.4	175	0.0	571.8	215	0.0	622.3
16	0.0	253.4	56	0.0	281.9	96	0.0	241.0	136	0.0	497.4	176	0.0	609.9	216	0.0	622.2
17	0.0	257.4	57	0.0	383.4	97	0.0	239.4	137	0.0	500.5	177	0.0	573.0	217	0.0	622.3
18	0.0	259.9	58	0.0	387.7	98	0.0	239.0	138	0.0	505.6	178	0.0	606.4	218	0.0	622.3
19	0.0	261.9	59	0.0	357.2	99	0.0	240.0	139	0.0	500.0	179	0.0	615.8	219	0.0	622.7
20	0.0	269.4	60	0.0	365.5	100	0.0	240.4	140	0.0	503.4	180	0.0	607.7	220	0.0	623.0
21	0.0	299.9	61	0.0	376.9	101	0.0	238.8	141	0.0	497.6	181	0.0	592.1	221	0.0	623.2
22	0.0	364.9	62	0.0	378.1	102	0.0	238.4	142	0.0	491.7	182	0.0	575.1	222	0.0	626.4
23	0.0	385.0	63	0.0	395.1	103	0.0	239.1	143	0.0	419.0	183	0.0	573.9	223	0.0	626.1
24	0.0	406.1	64	0.0	440.9	104	0.0	241.7	144	0.0	418.3	184	0.0	572.0	224	0.0	625.2
25	0.0	391.1	65	0.0	432.6	105	0.0	242.7	145	0.0	451.3	185	0.0	585.2	225	0.0	624.4
26	0.0	253.6	66	0.0	471.1	106	0.0	239.9	146	0.0	458.5	186	0.0	588.0	226	0.0	623.9
27	0.0	254.2	67	0.0	473.1	107	0.0	240.5	147	0.0	477.9	187	0.0	598.2	227	0.0	623.2
28	0.0	254.0	68	0.0	372.1	108	0.0	244.9	148	0.0	470.6	188	0.0	596.3	228	0.0	623.1
29	0.0	252.8	69	0.0	370.2	109	0.0	334.2	149	0.0	502.8	189	0.0	609.8	229	0.0	623.1
30	0.0	252.2	70	0.0	407.9	110	0.0	376.3	150	0.0	498.6	190	0.0	614.8	230	0.0	623.4
31	0.0	251.1	71	0.0	393.7	111	0.0	393.3	151	0.0	522.7	191	0.0	616.1	231	0.0	624.4
32	0.0	251.0	72	0.0	352.3	112	0.0	430.9	152	0.0	520.6	192	0.0	617.4	232	0.0	625.3
33	0.0	248.7	73	0.0	324.9	113	0.0	465.8	153	0.0	537.5	193	0.0	618.2	233	0.0	624.2
34	0.0	247.0	74	0.0	302.3	114	0.0	480.8	154	0.0	537.1	194	0.0	619.1	234	0.0	623.8
35	0.0	247.3	75	0.0	260.2	115	0.0	498.3	155	0.0	553.6	195	0.0	620.9	235	0.0	623.7
36	0.0	247.8	76	0.0	244.7	116	0.0	521.3	156	0.0	555.3	196	0.0	622.5	236	0.0	623.9
37	0.0	249.5	77	0.0	322.8	117	0.0	532.0	157	0.0	571.1	197	0.0	624.2	237	0.0	624.8
38	0.0	250.1	78	0.0	348.1	118	0.0	535.2	158	0.0	575.9	198	0.0	627.8	238	0.0	625.5
39	0.0	250.1	79	0.0	389.3	119	0.0	537.0	159	0.0	613.4	199	0.0	628.1	239	0.0	626.6
40	0.0	249.0	80	0.0	361.9	120	0.0	536.3	160	0.0	612.1	200	0.0	624.8	240	0.0	628.3

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REFERENCE 19

United Technologies
Pratt & Whitney
Engineering Division-South

Internal Correspondence

Job No. 85-752-A3105I
RL10/Heat Transfer

To: Dick Duncan
From: J.T. Elliott
Subject: RL10 Gear Test Rig Housing Can Have a 61 Degree F Circumferential Temperature Gradient
Date: 17 July 1985
CC: E.M. Beverly, J. D. Doernbach, Y. Eren, R.J. Ryberg

References:

1. RL10 Gear Rig Design Review - 6/14/85. Recommendation of mounting the containment housing and to check the housing's temperature effects on gearbox alignment.

2. D.G. V.O., T. Swartwout to Y. Eren, 5/17/85. Subject: RL10 Gear Test Rig Shaft Temperatures Revised Using F100 Test Data.

Summary:

The range of circumferential housing temperatures has been calculated as requested by Reference 1. The housing was given a nodal network, Figure 1. The nodal temperatures were calculated and recorded in Table 1. The results of this study were compared to the results of Reference 2. Table 2 shows the comparison. The critical area, according to Table 2, is at the H2 bearing.

Results:

1. Figure 1 shows the total node network for this study.

2. Figure 2 shows the fluids outside the housing.

3. Table 1 shows the nodal temperatures at steady state for this study.

4. Table 2 shows the temperature differences of this study and Reference 2. These temperature differences are the circumferential temperature gradients.

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Conclusions:

1. A 61 degree F circumferential temperature gradient is along the H2 bearing.

2. A 10 degree F circumferential temperature gradient is along the oil bearing.

J. T. Elliott

J.T. Elliott x-2937
Mechanical Components & Systems
Component Design Technology

Approved By: T.R. Swartwout

JTE/nfk

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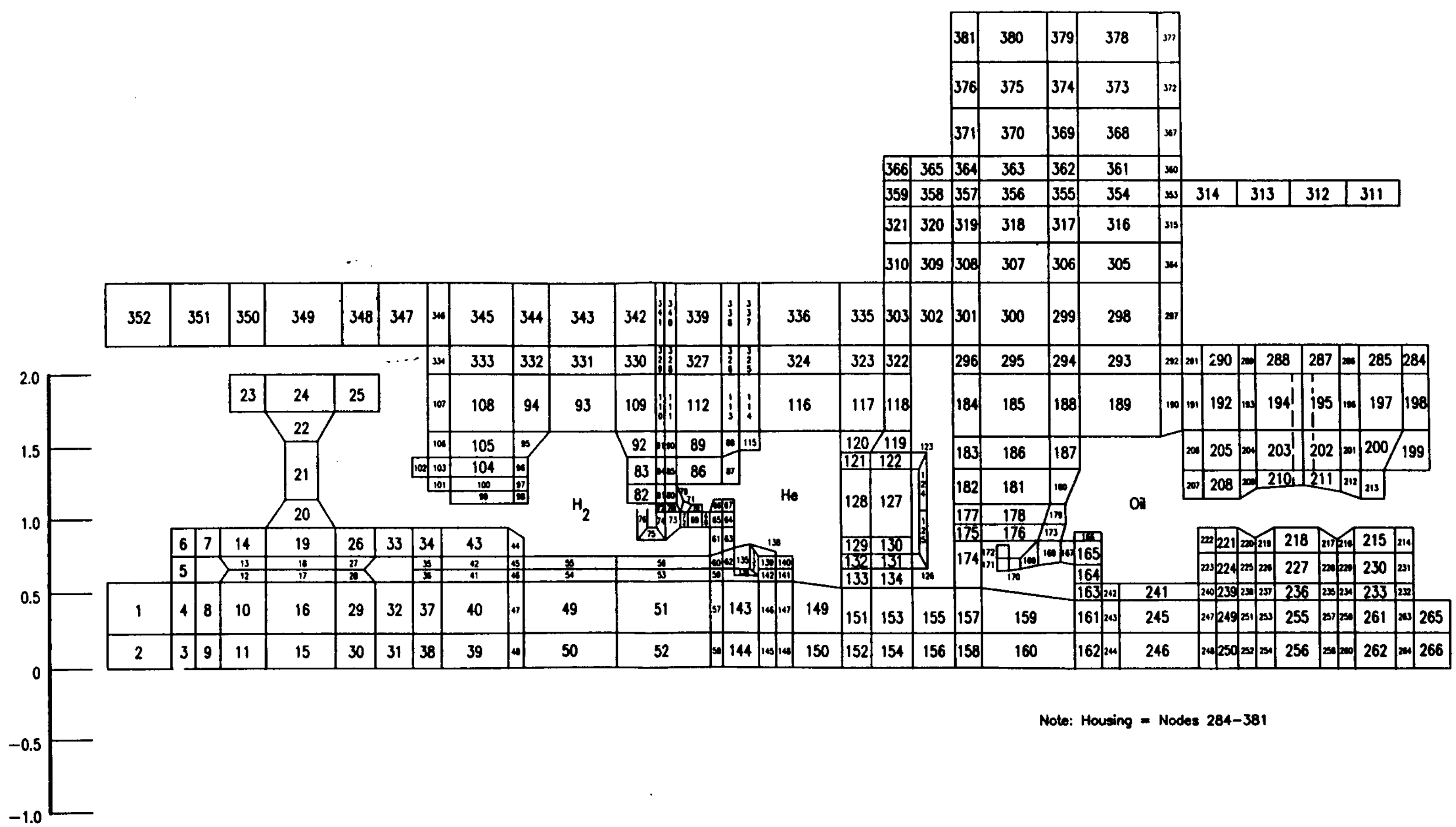

FOLDOUT FRAME

2 FOLDOUT FRAME

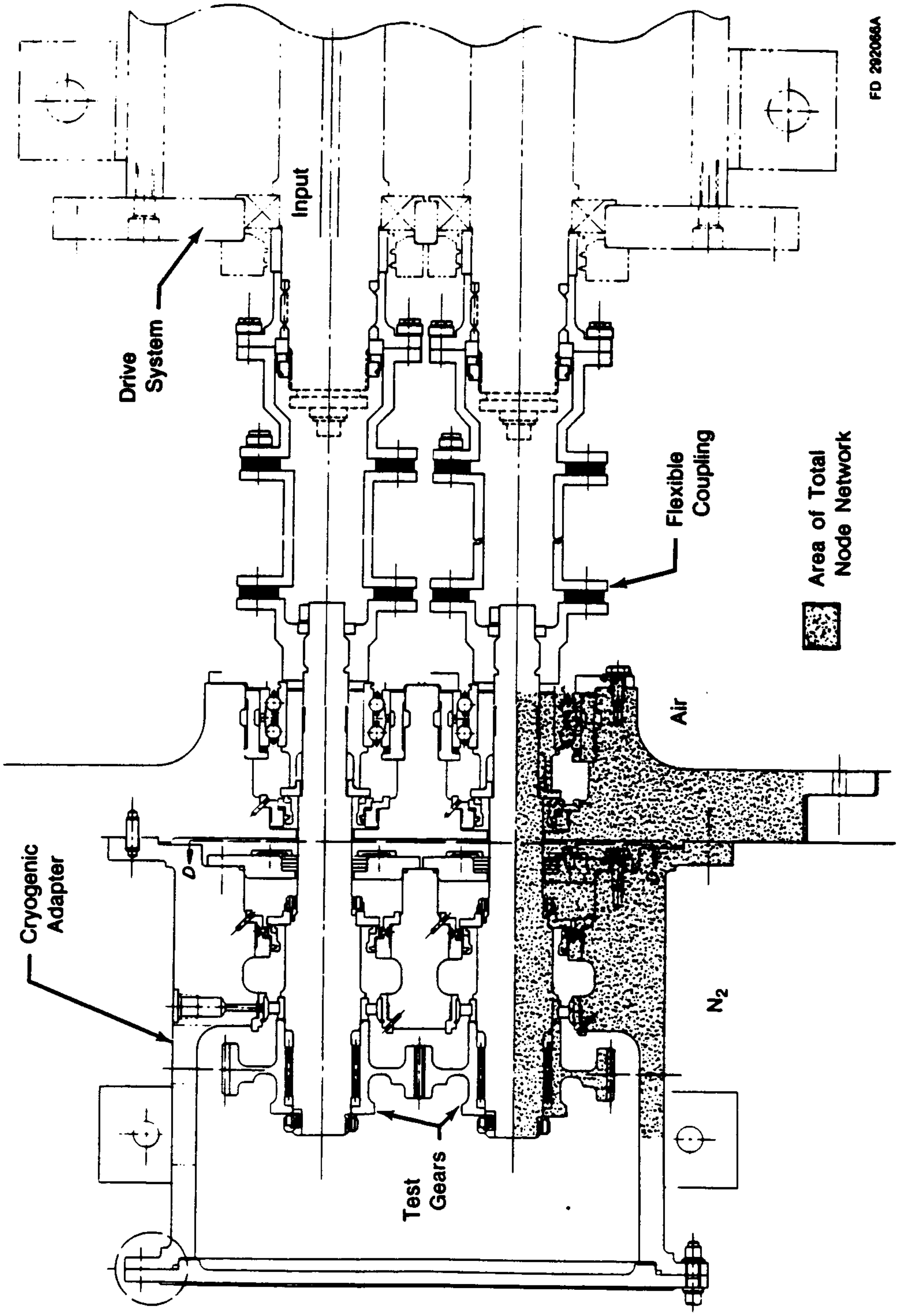
Input
Drive System
Flexible Coupling
Area of Total Node Network
Air
N_2
Cryogenic Adapter
Test Gears
FD 292068A

Table 1

NL10 GEAR RIG TEST

################ MAXIMUM NODE TEMPERATURES ##################07/10/85

NODE NO.	TIME (SEC)	TEMP (R)	NODE NO.	TIME (SEC)	TEMP (R)	NODE NO.	TIME (SEC)	TEMP (R)
1	0.0	219.7	41	0.0	246.9	81	0.0	346.9
2	0.0	221.2	42	0.0	246.1	82	0.0	302.0
3	0.0	230.0	43	0.0	244.6	83	0.0	323.0
4	0.0	229.5	.44	0.0	243.9	84	0.0	372.1
5	0.0	227.7	45	0.0	246.6	85	0.0	397.1
6	0.0	229.1	46	0.0	248.5	86	0.0	476.6
7	0.0	233.3	47	0.0	252.3	87	0.0	507.5
8	0.0	235.0	48	0.0	254.1	88	C.0	495.3
9	0.0	235.0	49	0.0	258.5	89	0.0	463.6
10	0.0	243.2	50	0.0	261.2	90	0.0	414.9
11	0.0	242.5	51	0.0	303.4	91	0.0	394.3
12	0.0	244.9	52	0.0	310.0	92	0.0	344.4
13	0.0	244.4	53	0.0	288.6	93	0.0	340.4
14	0.0	242.8	54	0.0.	253.2	94	0.0	326.4
15	0.0	251.7	55	0.0	250.9	95	0.0	277.0
16	0.0	253.4	56	0.0	281.9	96	0.0	259.4
17	0.0	257.4	57	0.0	383.4	97	0.0	247.3
18	0.0	259.9	58	0.0	387.7	98	0.0	244.4
19	0.0	261.9	59	0.0	357.2	99	0.0	247.8
20	0.0	269.4	60	0.0	365.5	100	0.0	254.5
21	0.0	299.9	61	0.0	376.9	101	0.0	244.5
22	0.0	364.9	62	0.0	378.1	102	0.0	241.2
23	0.0	385.0	63	0.0	395.1	103	0.0	247.4
24	0.0	406.1	64	0.0	440.9	104	0.0	268.6
25	0.0	391.1	65	0.0	432.6	105	0.0	280.0
26	0.0	253.6	66	0.0	471.1	106	0.0	257.1
27	0.0	254.2	67	0.0	473.1	107	0.0	267.7
28	0.0	254.0	68	0.0	373.1	108	0.0	306.0
29	0.0	252.8	69	0.0	371.6	109	0.0	398.0
30	0.0	252.2	70	0.0	409.4	110	0.0	428.1
31	0.0	251.1	71	0.0	396.5	111	0.0	440.1
32	0.0	251.0	72	0.0	354.5	112	0.0	466.2
33	0.0	248.7	73	0.0	327.8	113	0.0	488.5
34	0.0	247.0	74	0.0	305.0	114	0.0	497.9
35	0.0	247.3	75	0.0	261.2	115	0.0	508.4
36	0.0	247.8	76	0.0	245.0	116	0.0	523.4
37	0.0	249.5	77	0.0	327.6	117	0.0	531.7
38	0.0	250.1	78	0.0	352.6	118	0.0	534.7
39	0.0	250.1	79	0.0	393.7	119	0.0	536.7
40	0.0	249.0	80	0.0	368.8	120	0.0	536.1

################ MAXIMUM NODE TEMPERATURES ##################07/10/85

NODE NO.	TIME (SEC)	TEMP (R)	NODE NO.	TIME (SEC)	TEMP (R)	NODE NO.	TIME (SEC)	TEMP (R)
121	0.0	537.4	161	0.0	646.1	201	0.0	614.3
122	0.0	537.4	162	0.0	638.9	202	0.0	612.4
123	0.0	538.7	163	0.0	662.2	203	0.0	610.2
124	0.0	539.5	164	0.0	675.9	204	0.0	609.7
125	0.0	539.9	165	0.0	689.9	205	0.0	609.6
126	0.0	539.9	166	0.0	685.7	206	0.0	611.3
127	0.0	538.2	167	0.0	689.0	207	0.0	613.8
128	0.0	538.8	168	0.0	645.0	208	0.0	611.6
129	0.0	539.3	169	0.0	644.5	209	0.0	611.7
130	0.0	538.6	170	0.0	644.2	210	0.0	611.8
131	0.0	539.2	171	0.0	643.9	211	0.0	613.6
132	0.0	539.5	172	0.0	643.8	212	0.0	615.0
133	0.0	539.6	173	0.0	627.7	213	0.0	615.7
134	0.0	539.6	174	0.0	567.3	214	0.0	623.2
135	0.0	470.4	175	0.0	568.2	215	0.0	622.2
136	0.0	497.4	176	0.0	602.9	216	0.0	622.2
137	0.0	500.5	177	0.0	568.3	217	0.0	622.2
138	0.0	505.6	178	0.0	598.7	218	0.0	622.2
139	0.0	500.0	179	0.0	608.5	219	0.0	622.7
140	0.0	503.4	180	0.0	598.8	220	0.0	622.9
141	0.0	497.6	181	0.0	582.4	221	0.0	623.2
142	0.0	491.7	182	0.0	568.2	222	0.0	624.3
143	0.0	419.0	183	0.0	564.9	223	0.0	626.1
144	0.0	418.3	184	0.0	560.4	224	0.0	625.1
145	0.0	451.3	185	0.0	569.2	225	0.0	624.4
146	0.0	458.5	186	0.0	575.9	226	0.0	623.9
147	0.0	477.9	187	0.0	585.0	227	0.0	623.2
148	0.0	470.6	188	0.0	577.3	228	0.0	623.0
149	0.0	502.8	189	0.0	587.4	229	0.0	623.1
150	0.0	498.6	190	0.0	595.0	230	0.0	623.3
151	0.0	522.7	191	0.0	597.8	231	0.0	624.3
152	0.0	520.6	192	0.0	601.0	232	0.0	625.2
153	0.0	537.5	193	0.0	603.5	233	0.0	624.1
154	0.0	537.1	194	0.0	606.2	234	0.0	623.7
155	0.0	553.6	195	0.0	609.4	235	0.0	623.6
156	0.0	555.3	196	0.0	611.2	236	0.0	623.8
157	0.0	571.1	197	0.0	613.1	237	0.0	624.7
158	0.0	573.9	198	0.0	617.3	238	0.0	625.4
159	0.0	613.4	199	0.0	620.0	239	0.0	626.6
160	0.0	612.1	200	0.0	616.3	240	0.0	628.2

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Table 1 (continued)

GEAR RIG TEST

################ MAXIMUM NODE TEMPERATURES ################07/10/85

NODE NO.	TIME (SEC)	TEMP (R)	NODE NO.	TIME (SEC)	TEMP (R)	NODE NO.	TIME (SEC)	TEMP (R)
241	0.0	641.5	281	0.0	611.8	321	0.0	541.6
242	0.0	652.5	282	0.0	613.6	322	0.0	533.6
243	0.0	646.5	283	0.0	622.1	323	0.0	529.7
244	0.0	641.6	284	0.0	612.8	324	0.0	519.4
245	0.0	640.8	285	0.0	611.6	325	0.0	500.1
246	0.0	639.9	286	0.0	610.0	326	0.0	493.6
247	0.0	632.1	287	0.0	608.2	327	0.0	479.6
248	0.0	632.8	288	0.0	605.1	328	0.0	464.4
249	0.0	629.4	289	0.0	601.9	329	0.0	457.5
250	0.0	630.3	290	0.0	598.7	330	0.0	440.3
251	0.0	627.6	291	0.0	594.0	331	0.0	406.6
252	0.0	628.5	292	0.0	589.6	332	0.0	380.1
253	0.0	626.5	293	0.0	579.9	333	0.0	348.3
254	0.0	627.2	294	0.0	570.4	334	0.0	295.2
255	0.0	625.2	295	0.0	563.2	335	0.0	534.0
256	0.0	625.7	296	0.0	555.8	336	0.0	525.1
257	0.0	625.0	297	0.0	576.9	337	0.0	514.4
258	0.0	625.5	298	0.0	569.9	338	0.0	510.9
259	0.0	625.1	299	0.0	562.3	339	0.0	503.9
260	0.0	625.6	300	0.0	556.6	340	0.0	496.6
261	0.0	625.7	301	0.0	550.6	341	0.0	493.3
262	0.0	626.2	302	0.0	544.7	342	0.0	485.3
263	0.0	627.6	303	0.0	538.7	343	0.0	468.4
264	0.0	627.9	304	0.0	571.5	344	0.0	451.9
265	0.0	630.4	305	0.0	562.6	345	0.0	434.7
266	0.0	630.0	306	0.0	556.6	346	0.0	416.5
267	0.0	611.9	307	0.0	552.5	347	0.0	400.7
268	0.0	622.2	308	0.0	548.8	348	0.0	390.4
269	0.0	622.6	309	0.0	545.8	349	0.0	382.5
270	0.0	707.7	310	0.0	541.9	350	0.0	379.7
271	0.0	615.1	311	0.0	549.3	351	0.0	378.0
272	0.0	704.3	312	0.0	549.3	352	0.0	377.3
273	0.0	247.3	313	0.0	549.3	353	0.0	551.5
274	0.0	428.6	314	0.0	549.8	354	0.0	551.2
275	0.0	244.3	315	0.0	563.2	355	0.0	549.0
276	0.0	411.3	316	0.0	556.4	356	0.0	547.2
277	0.0	408.8	317	0.0	552.3	357	0.0	545.4
278	0.0	425.2	318	0.0	549.5	358	0.0	543.8
279	0.0	373.9	319	0.0	547.0	359	0.0	541.6
280	0.0	622.8	320	0.0	544.8	360	0.0	545.1

################ MAXIMUM NODE TEMPERATURES ################07/10/85

NODE NO.	TIME (SEC)	TEMP (R)	NODE NO.	TIME (SEC)	TEMP (R)	NODE NO.	TIME (SEC)	TEMP (R)
361	0.0	547.8	368	0.0	544.2	375	0.0	542.0
362	0.0	546.8	369	0.0	544.1	376	0.0	541.4
363	0.0	545.6	370	0.0	543.5	377	0.0	540.6
364	0.0	544.2	371	0.0	542.5	378	0.0	541.4
365	0.0	542.7	372	0.0	540.9	379	0.0	541.8
366	0.0	541.4	373	0.0	542.2	380	0.0	541.4
367	0.0	542.1	374	0.0	542.3	381	0.0	541.0

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Table 2: Temperature differences of This study and Reference 2

NODE	Temperature Gradient (°R)	NODE	Temperature Gradient (°R)	NODE	Temperature Gradient (°R)	NODE	Temperature Gradient (°R)
68	1.0	95	33.2	167	2.6	194	12.9
69	1.4	96	18.4	168	3.8	195	11.5
70	1.5	97	7.9	169	2.9	196	11.3
71	2.8	98	5.4	170	2.1	197	11.1
72	2.2	99	7.8	171	1.6	198	10.5
73	2.9	100	14.1	172	1.4	199	8.1
74	2.7	101	5.7	173	5.2	200	8.5
75	1.0	102	2.8	174	2.3	201	8.5
76	0.3	103	8.3	175	3.6	202	8.5
77	4.8	104	26.9	176	7.0	203	9.3
78	4.5	105	37.3	177	4.7	204	10.0
79	4.4	106	17.2	178	7.7	205	10.4
80	6.9	107	27.2	179	7.3	206	10.1
81	7.6	108	61.1	180	8.9	207	8.0
82	7.9	109	63.8	181	9.7	208	8.2
83	13.7	110	51.8	182	6.9	209	8.0
84	14.2	111	46.8	183	9.0	210	7.6
85	13.7	112	35.3	184	11.6	211	7.0
86	12.0	113	22.7	185	16.0	212	6.9
87	7.8	114	17.1	186	12.1	213	6.8
88	13.4	115	10.1	187	13.2	214	0.1
89	20.5	116	2.1	188	19.0	215	0.1
90	24.9	117	.3	189	22.4		
91	26.2	118	.5	190	19.8		
92	26.9			191	18.3		
93	72.3			192	16.4		
94	73.5			193	14.7		

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