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TI-696-22-003

COVER SHEET

DOCUMENT TITLE

Control Drum Drive Train and Actuator Design

AUTHOR

K. Bilibin
D. Harwood

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Airmics International
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SUPPORTING DOCUMENT

NUMBER		TI-696-22-003		REV	LTR
DOCUMENT TYPE Technical Information					
PROGRAM TITLE ZrH Reactor Development Program			KEY NOUNS Drive Train Analysis		
DOCUMENT TITLE Control Drum Drive Train and Actuator Design			ORIGINAL ISSUE DATE April 10, 1970		
			GO NO.	S/A NO.	PAGE 1 OF TOTAL PAGES 137
			07759	41620	REL DATE 4-15-70
ABSTRACT The design analysis of the control drum drive train including the mockup actuator assemblies are documented.			PREPARED BY <i>K.B.</i> DEPT. MAIL ADDR K. Bilibin/D. Harwood <i>D. Harwood</i> DATE 4-13-70 737-753 PA02		
			APPROVALS W. J. Kurzeka <i>W. Kurzeka</i> 4/13/70 T. S. Nakae <i>T. Nakae</i> 4/16/70 O. P. Steele <i>O. P. Steele</i> 4/16/70		
			SECURITY CLASSIFICATION		
			(CHECK ONE BOX ONLY) (CHECK ONE BOX ONLY) AEC DOD RESTRICTED UNCL. ☒ ☐ DATA ☐ CONF. ☐ ☐ DEFENSE ☐ SECRET ☐ ☐ INFO. ☐		
AUTHORIZED CLASSIFIER <i>T. Nakae</i> DATE 4/16/70					

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I. INTRODUCTION

The design and analysis for the coupling, gear, locking mechanism, actuator rotor-shaft and actuator bearings have been completed.

Drawings reflecting the current configuration of these components have been released and are listed below. The listed assemblies are being fabricated.

1. EX-N69626142 Drive Train Assembly S&V Test Mockup
2. EX-N69626146 Coupling Assembly - Upper
3. EX-N69626147 Coupling Assembly -Lower
4. EX-N69626164 Actuator Assembly - ZrH Reactor Drum Assy
5. R-N69626144 Control Drum Drive Train Layout
6. R-N69626145 Coupling Assembly Layout

Upon completion of fabrication, these components will be assembled and will be subjected to shock and vibration tests (Ref 1 and 2) designed to simulate the conditions expected during Saturn V launch (Ref 3).

Following the shock and vibration tests, these components will be checked for evidence of any damage and will subsequently undergo performance tests at simulated operational temperature and vacuum environment.

II. DRIVE TRAIN COUPLING

In response to the action of reference ZrH reactor design review (items 30 and 31 of Ref 4) a design study was performed to determine the coupling concept which best satisfies the overall launch and operating requirements with high performance reliability.

The coupling concepts chosen for the comparative study and evaluation were: a) torque arm or pin and slot, b) universal joint, and c) bellows. In addition to the above, a coupling utilizing a bellows in conjunction with a yoke, ("hybrid"), applicable to the 4 Pi shield configuration, was studied. The details of these concepts are shown in Dwg R-N69626143.

All of the coupling concepts except for the bellows, depend upon sliding surfaces with close fits to provide the required flexibility. There is an uncertainty as to the effect of long term high temperature on these friction surfaces, particularly loss of dry lubricants by sublimation and the possibility of self-welding. The design analysis performed for this series are compiled in Appendix B. Tables I and II illustrate the comparative grading of these couplings. Listed in the tables are the most important characteristics required. Each characteristic is given an "importance value" as it affects the overall system. Each coupling configuration is "ranked" in accordance with its comparative superiority over competitive couplings. The product of the "importance value" and "ranking" is tabulated under the column "Grade". The summation of the "grade" points provides a basis for overall comparison of the coupling concepts.

For the shadow shield configuration the bellows coupling is distinctly preferable to the other two. For the 4 Pi shield configuration the advantage of the bellows concept over the hybrid and the universal couplings is less distinct. In both cases the deciding factors in favor of the bellows concept were the absence of rubbing friction joints and minimal backlash. The bellows concept has been selected as the reference design for both the shadow shield and 4 Pi shield configuration. The designs of the bellows coupling are defined on Dwg EX-N69626146 and EX-N69626147.

The material selected for the bellows is Inconel 718, because of its favorable properties at both room and operating temperatures and state of art in bellows fabrication.

III. DRUM LOCKING MECHANISM AND ACTUATOR ASSEMBLY

The centroid of the poison backed control drum does not coincide with the control drum bearing centerline. When the drum is subjected to lateral forces, such as experienced during the space vehicle launch, the drum can rotate unless restrained. To prevent control drum rotation prior to reactor startup, a method of positively locking the drum in the shutdown position is required.

The unique feature of the current control drum drive train design is the drum locking mechanism, which is integral with the drive train and requires no separate signals or forces to unlock the drum. The details of the locking mechanism is illustrated on Dwg R-N69626144. The pin and slotted cam mechanism directs the reaction force of the drum rotation through the centerline of the actuator output shaft effectively preventing drum rotation. Drum rotation can only be initiated when the torque is from the actuator and is accomplished by a cam surface sliding against a pin to rotate the drum and mesh the gear teeth with the pinion. Further drum rotation is accomplished by the gearing while the locking mechanism becomes disengaged. In the drum locked position the drum drive gear is not meshed with the actuator pinion so that no launch torque loads are being reacted by the gear teeth.

The mechanism fulfills its primary function of holding the drums during launch and unlocking the drums for space operation. However, this feature imposes loads on the drive train and actuator components which are at least one hundred times greater than imposed during space operation. The design analysis for these components to accommodate the high loads are included in Appendix C.

The drive train coupling, discussed in the previous section, was designed for these high loads to lock the drum during the launch. High stresses on the locking cam surface and the locking pin on the gear created problems in finding the proper material for these components. Rene 41 was finally chosen because, in addition to its high strength, high fatigue strength, hardness and good high temperature properties, its thermal coefficient of expansion falls between that of Inconel 718 (coupling) and the materials considered for the actuator shaft and end bells. This provides a smooth transition and minimizes the thermal expansion problems.

Due to the congested area, the lock-pin size cannot be increased without impairing the action of the cam and therefore high localized fatigue stresses result. However, all margins of safety are positive with the minimum being 5% based on the fatigue strength of the material.

To avoid the problems associated with differential thermal expansion the S8DR actuator design utilized "hiperco 27" material for all its components. Because of the high load imposed on the actuator shaft by the drum locking mechanism, the homogeneous feature of the S8DR actuator design could not be retained in the advanced ZrH reactor design. The requirement for a high strength shaft having a coefficient of thermal expansion near that of "hiperco 27" resulted in the selection of Hastelloy "B" material for the actuator shaft. The design of the composite rotor-shaft, which evolved from the analyses, is shown on Dwg EX-N69626163. To further minimize the effect of differential thermal expansion and to ensure clearances between the rotor-stator, shaft-bearing and pinion-gear under all tolerance and temperature conditions, both the front and rear end bells of the actuator will also be fabricated from Hastelloy "B" material.

For the shock and vibration test, only the front end bell will be manufactured from Hastelloy "B". The rear end bell will be reworked from an S8DR actuator for economic reasons. This is justifiable since the loads on the rear end bell and its bearing are quite low.

An area of great concern is the graphite bearings for the rotor shaft and gear shaft. Because of the 1200°F service temperature, the shrink fit stresses on all the bearings are very high and near the ultimate compressive strength of the graphite. To reduce these shrink fit stresses a selective fit is proposed so that component tolerances can be kept reasonable. The launch loads imposed to the bearings through the locking device are also quite high. However, the stress due to these loads tends to reduce the combined stress to a value less than the shrink fit stresses.

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Shock and vibration tests, which will be performed on these components, should establish those areas which may be marginal. Any design modification to increase design margins as indicated from the vibration test may be difficult. To provide a common design for both shadow and 4 Pi shielded application, every part has been designed for maximum capability within the available space. Therefore, very little is to be gained by small localized changes. Thus, catastrophic failure of any component during testing may require complete drum drive system design study, including evaluation of reducing the drum unbalance and other drum locking mechanisms. The limiting space was defined by the shield cone angle of the shadow shield (Ref 5).

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REFERENCES

1. TP-696-26-002 Vibration, Backlash, and Friction Tests of Drive Train for 4 Pi Shielded ZrH Reactor
2. DTP-696-26-022 Vibration, Backlash and Friction Tests of Drive Train for 4 Pi Shielded ZrH Reactor
3. SS696N17001 Environmental Requirements for Nuclear Power Systems for Space Applications
4. DRR-696-13-002 Design Review Minutes for Reference ZrH Reactor
5. N69626101 Control Drum to Drywell Interface-ZrH Reactor with Shadow Shield

TABLE I

SHADOW SHIELD								
		PIN & SLOT		UNIVERSAL		BELLWS		
	CHARACTERISTIC	Wt.	RANK	GRADE	RANK	GRADE	RANK	GRADE
1.	BACKLASH	8	3	24	2	16	1	8
2.	WEIGHT	1	1	1	3	3	2	2
3.	COST	3	1	3	2	6	3	9
4.	RELIABILITY	10	3	30	2	20	1	10
5.	ACCESS TO DRUM	3	1	3	2	6	3	9
6.	EASE OF ASSEMBLY TO REACTOR	3	1	3	1	3	3	9
7.	MATERIAL COMPATIBILITY	6	1	6	1	6	3	18
8.	EST. FRICTION	8	2	16	3	24	1	8
9.								
10.								
TOTALS			86		84		73	

The "Grade" is determined by multiplying the "Rank" (1st, 2nd, or 3rd best) times "Wt.". The weighting factor - is a scale of importance from 1 to 10 (least to most important respectively).

TABLE II

RATING FOR CONTROL DRUM DRIVE COUPLINGS FOR 4 π SHIELD									
		PIN & SLOT		UNIVERSAL		HYBRID		BELLOWS	
CHARACTERISTIC	Wt.	RANK	GRADE	RANK	GRADE	RANK	GRADE	RANK	GRADE
1. BACKLASH	8	4	32	3	24	2	16	1	8
2. WEIGHT	1	1	1	4	4	2	2	2	2
3. COST	3	1	3	2	6	3	9	4	12
4. RELIABILITY	10	4	40	1	10	2	20	2	20
5. ACCESS TO DRUM	3	1	3	2	6	2	6	2	6
6. EASE OF ASS'Y TO REACTOR	3	4	12	1	3	1	3	3	9
7. MAT'L COMPATIBILITY	6	4	24	1	6	2	12	2	12
8. EST. FRICTION	8	4	32	3	24	2	16	1	8
9.									
10.									
TOTALS		147		83		84		77	
The "Grade is determined by multiplying the "Rank" (1st, 2nd, 3rd, or 4th best) times "Wt.". The weighting factor - is a scale from 1 to 10 (least to most important, respectively).									

APPENDIX A

PERTINENT CORRESPONDENCE

INTERNAL LETTER

ADVANCED INTERNATIONAL

17759 of North American Aviation, Inc.

DATE August 15, 1969

TO D. Ellis
ADDRESS 737-073 BEO2

FROM E. Blalock
ADDRESS 737-075 BEO4

PHONE 2171

SUBJECT Advanced EAF Reactor Material Selection Support.

Please provide support for approving and/or selecting materials for the reactor control drum actuator and drive train components. The effort to the extent required should be charged to the following: #7759 contract, A1620 S/A and #202000 activity for the drive train components and to #7759 contract, A1210 S/A and #202000 activity for the actuator components.

Listed below are the materials which I am presently using for these components and the properties which dictated my choice. Your approval or substitution is required, so that purchasing of materials can commence. In addition to the material application recommendations, please advise the processes required to achieve the optimum physical properties.

A. Actuator Shaft and End Bell

The coefficient of thermal expansion shall be within 6.25 to 6.75×10^{-6} in/in°F at 1200°F. It must possess a fatigue strength, for 250,000 cycles, in excess of 40,000 psi at room temperature. The minimum operating temperature is 1200°F at 10^{-5} torr to 10^{-12} torr for 20,000 hours. The yield strength shall be in excess of 30,000 psi at temperature and the sublimation rate shall be low. In reviewing various materials, Hastelloy "B" satisfies the above property requirements and is used for these components.

B. Gear and Cam Locking Mechanism

The coefficient of expansion must closely match the coefficient of expansion of the actuator and bell and shaft. The room temperature endurance strength shall be in excess of 85,000 psi at room temperature for the 250,000 cycles. The yield and ultimate strength at room temperature shall be at least 85,000 and 130,000 psi respectively. The minimum operating temperature is 1200°F for 20,000 hours at 10^{-5} to 10^{-12} torr. The minimum yield strength at operating temperature shall be in excess of 30,000 psi and the sublimation rate shall be low. René 41 satisfies most of the above requirements and therefore it is being considered as the prime candidate for these components.

C. Drive Train Coupling

The coefficient of expansion must match that of the gear as close

INTERNAL LETTER

GENERAL ELECTRIC COMPANY
AIRCRAFT AND SYSTEMS GROUP

		Date	October 7, 1969
TO	E. Bilibin	FROM	R. G. Berryman
Address	737-75	Address	737-73
	PA04		FE02
		Page	2878

Subject: ADVANCED ZrE REACTOR MATERIALS SELECTION SUPPORT

Reference: IL, E. Bilibin to D. E. Ellis, Same Subject, 8/15/69

A review of the candidate materials, proposed in the reference IL, was made in light of the selection criteria also therein presented. The conclusions made from this review are that the candidate materials for the actuator shaft and end bell (Hastelloy "B"), the gear and cam locking mechanism (Rene 41), the drive train coupling assembly (Inconel 718), and the drum clevis (Ta-10W) should adequately meet the design requirements presented in the above IL.

The data compiled in support of this analysis is shown in Table I. The recommended heat treatments of the material for use in the above application are shown in Table II.

The approval of Inconel 718 is made with some reservations in that the operating temperature of 1400°F is above the aging temperature of this alloy. Rene 41, having a higher aging temperature, would be the best alternative were it not for high fabrication costs of Rene 41, which is not readily amenable to bellows fabrication processes.

A review of the available literature on Inconel 718 revealed short term tensile properties after exposure up to only 1000 hours at 1400°F, and creep rupture data up to only 1300°F. Analysis of this data and extrapolations to 20,000 hours and 1400°F did not reveal any real indications that application of pertinent loading would lead to premature failure of the components (Figure 1).

R. G. Berryman to K. Bilibin
Advanced ZrH Reactor Materials Selection Support

October 7, 1969
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During operation, irradiation damage and over-aging will tend to decrease ductility of Inconel 718 while the high temperature will tend to offset this decrease. The literature revealed that the total elongation of this material at 1200°F did not drop below ~5-7% after exposure at 1364°F to a total fluence level of $\sim 4 \times 10^{20}$ nvt/cm² (E > 1 Mev). At 1400°F this value is expected to be even higher (perhaps 8-10%). These values are to be compared with the elongation of ~22% in the as-heat-treated (per AMS 5596B) condition. Therefore, while ductility is expected to decrease substantially, the final value is expected to remain high enough to prevent premature component failure.

A review of the properties of the elements contained in these alloys was also undertaken to determine the effect of high temperature operation in a high vacuum. The "binder" or matrix element, nickel, was deemed to be the controlling element with regard to sublimation rate. At 1200°F the vapor pressure of nickel is 10^{-13} torr; sublimation in a vacuum of 10^{-12} torr should not occur. At 1400°F the vapor pressure of nickel is $\sim 10^{-11}$ torr; some sublimation of the lower end drive train coupling may occur - to a very limited extent. The indefinite term "may" is used because information on sublimation rates of alloys is not available.

R. G. Berryman

R. G. Berryman
Materials and Processes Unit
Materials Engineering Group

gc

dc:	C. T. Arkenoff	PE02
	R. L. Detterman	LE03
	D. E. Ellis	PE02
	W. J. Kurzeka	PA03
	L. M. Maki	LE03
	E. Matlin	PAC2
	T. S. Mahne	PA04
	J. P. Page	PE02
	O. P. Steele	PA02
	C. C. Woolsey	PE02
	W. D. Whitaker	PA03

Berryman to Bilibin

October 7, 1959
Page 3

Table I

Material Properties and Design Requirements

A. Hastelloy "B" - Room Temperature Fatigue and 1200°F Tensile Loads

R.T. Fatigue Strength Required - 40 ksi at 2.5×10^5 cycles
Actual - 64 ksi at 2.5×10^8 cycles and 1200°F

1200°F Yield Strength Required - 30 ksi
Actual - 50.4 ksi

Coefficient of Thermal

Expansion Required - $6.25-6.75 \times 10^{-6}$ in./in. (70-1200°F)
Actual - 6.73×10^{-6} in./in.

B. René 41 - Room Temperature Fatigue, Room Temperature and 1200°F Tensile Loads

R.T. Fatigue Strength Required - 85 ksi at 2.5×10^5 cycles
Actual - 85 ksi* at 1.08×10^7 cycles
* Mean with Alternating Load of ± 45 ksi

R.T. Ultimate Tensile Required - 130 ksi
Actual - 203 ksi

R.T. Yield Strength Required - 85 ksi
Actual - 155 ksi

1200°F Yield Strength Required - 30 ksi
Actual - 150 ksi

Coefficient of Thermal Expansion Required - Close to that of Hastelloy "B"
Actual - 7.82×10^{-6} in./in. (70-1200°F)

C. Inconel 718 - Room Temperature Fatigue and 1400°F Tensile Loads

R.T. Fatigue Strength Required - 90 ksi at 2.5×10^5 cycles
Actual - 101 ksi at 10^6 cycles

1400°F Yield Strength Required - 5 ksi
Actual - 110 ksi

Coefficient of Thermal Expansion Required - Close to that of René 41
Actual - 9.1×10^{-6} in./in. (70-1400°F)

Berryman to Bilibin

October 7, 1959

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Table II

Heat Treatments Required to Produce "Actual"
Properties Shown in Table I

- A. Hastelloy "B" - $2000^{\circ}\text{F}/1 \text{ hr/W.Q.} + 1200^{\circ}\text{F}/4 \text{ hr/A.C.}$ (1,2)
- B. René 41 - $1950^{\circ}\text{F}/4 \text{ hr/A.C.} + 1400^{\circ}\text{F}/16 \text{ hr/A.C.}$ (1,2)
- C. Inconel 718 - $1800^{\circ}\text{F}/1 \text{ hr/A.C.} + 1325^{\circ}\text{F}/8 \text{ hr} + \text{F.C. to } 1150^{\circ}\text{F}$
 $+ 1150^{\circ}\text{F}/\text{Remainder of } 18 \text{ hr aging time/A.C.}$ (1,2,3)
(AMS 5596B Condition)
- D. Ta-10W - $2600^{\circ}\text{F}/1 \text{ hr/Inert Gas Cool} + 2200^{\circ}\text{F}/1\frac{1}{2} \text{ hr/Inert}$
Gas Cool (4,5)

-
- (1) Heat treat per NAA Specification 1A0111-015
- (2) Clean per AI Specification ST0610MA0001
- (3) Purchase per AMS 5562 Solution Annealed Condition
- (4) Heat Treat per AI Specification ST0611MA0006
- (5) Clean per AI Specification ST0610MA0005.

FIGURE 1 - ELEVATED TEMPERATURE PROPERTIES OF INCONEL 718

MATERIALS AND PROCESSES DRAWING REVIEW SHEET

Component/Assembly

ZrE Reactor Drive Train

Drawing NumberTitle

TX-N59526142	Coupling Assembly - Design Layout
146	Coupling Assembly - Upper Bellows
147	Coupling Assembly - Lower Bellows
159	Bearing Details
160	Actuator Assembly
161	Details - Drive Gear and Actuator Shaft Assembly
162	Details - Cam and Yoke
163	Actuator Rotor Assembly

Program/Subaccount

77 59-41210

Engineer/Designer

K. Bilibin/D. Harwood

Date Received:

10/8/69

Date Returned:

10/9/69

M&P Reviewer(s)

C. C. Woolsey/R. G. Berryman

Type Approval

Approved as is

Approved with Reservations

Approved with Recommended Changes

Not Approved

x (a)

(a) Changes were made before drawings were signed.

M&P Engineer

Signature

Date

Distribution:

K. Bilibin

R. E. Gordon

T. S. Rabae

O. P. Steele

PA04

LA00

PA04

PA02

D. E. Bilibin

W. J. Murzeka

J. P. Page

M&P File

PB02

PA03

PB02

Design Requirements:

Reference II, K. Bilibin to D. E. Ellis, dated August 15, 1969, copy attached. In addition,

effects of irradiation to a fluence of 1.9×10^{21} n/cm² (E>1 Mev) were considered.

Also reference II, R. G. Barryman to K. Bilibin, dated October 7, 1969, copy attached.

<u>Materials</u>	Yes	No	N/A	See Remarks
1. Are proper specifications called out?	x			(1)
2. Do the specifications reflect the requirements of the components (i.e., heat treatment, surface condition, shape, etc.)?	x			
3. Is further information required on drawing?		x		
4. Is more than one type or grade of construction material specified?	x			
If so, is there a dissimilar material compatibility problem?		x		
If so, is there a thermal expansion problem?		x		
5. Is the material of construction compatible with service environments?	x			
Radiation?	x			
Temperature?	x			(2)
Environment?	x			

(1) F-5 graphite is a proprietary product of Pure Carbon Company. It has been tested extensively at AI in high temperature bearing applications.

(2) See II, Barryman to Bilibin regarding reservations on use of Inconel 718 at 1400°F.

Processes:	Yes	No	N/A	See Details
1. Are forming processes acceptable? Is post-forming heat treatment, if any, acceptable?			X X	
2. Are machining processes acceptable?	X			
3. Are joining processes acceptable? Are Joint designs acceptable? Are weld symbols correct? Is an acceptable joining specification listed? Are braze/solder/weld metals acceptable for application?			X	
4. Is heat treatment specified? Do heat treat procedures reflect requirements of the component (i.e., stress relief, properties)?	X X			
5. Is cleaning specified? Does cleaning specification or procedure reflect requirements of the component (consider surface, corrosion, etc.)?	X X			
6. Will material be sensitized or otherwise affected by these processes?			X	
7. Will material require pre-operational environmental control (protection coatings, inside storage, temperature, etc.) for protection from corrosion, low temperature impact damage?	X			(1)
8. Is coating specified? Is there a potential compatibility problem between coating and substrate under operating or processing conditions?	X	X		(2)

(1) Finished parts should be stored to maintain cleanliness prior to assembly.

(2) Al_2O_3 coating to be applied to form one bearing surface. AI tests have shown Al_2O_3 vs P-5 graphite to act as a good high temperature bearing.

INTERNAL LETTER
 UNITED STATES ATOMIC ENERGY COMMISSION
 Washington, D. C.

DATE September 8, 1969

TO: T. S. Fakas
 ADDRESS: 737-75 EA02

FROM: P. H. Horton
 ADDRESS: 737-75 EA03

PHONE: 2126

SUBJECT: Drum Position Error of the Advanced ZrH Reactor Drive System, 4 T Shield Case

- REF:
- 1) IL, T. S. Fakas to L. H. Maki, "Drawings of ZrH Reactor Control Drum Drive System for your Review," Sept. 2, 1969
 - 2) IL, L. H. Maki/H. Rood to W. J. Kurzeka, "Control Drum Drive Train Tolerances," August 14, 1969

The control drum position error of the reactor drive system for the 4 T shield case has been calculated based on layout drawings of the actuator, drive train and control drum bearings listed in Reference 1. Reference 2 establishes the maximum step error as $2/3$ of the nominal step size or 0.8° (arc) at the drum (nominal step = 1.2°). This is equivalent to $43'$ (arc) of position error.

The position error is composed of backlash due to bearing and coupling clearances and the tooth-to-tooth machining tolerances of the actuator brake and of the drive train gear. The tooth-to-tooth tolerances of the brake are such that step accuracy is $7.2^\circ \pm 0.1^\circ$ at the actuator. Through the 6:1 gear reduction of the drive train, the deviation is reduced from $6'$ (0.1°) to $1'$ at the control drum. The tooth-to-tooth accuracy of the drive gear is $8'$ and is included in the actuator bearing and gear position error values. The drive coupling and control drum bearing position error is due to necessary bearing and assembly clearances.

With the brake engaged, the total position error at the drum has been calculated to be a maximum of $38'$ at ambient and $33' 12''$ at operating temperatures (1400°F bearing temperature). Nominal accuracy is $26' 31''$ cold and $24' 53''$ hot. Figure 1 shows nominal and maximum error as a function of temperature for, a) the actuator bearings and gear, b) the drive coupling, c) the drum bearings, and d) for the sum of the above. Note that for the nominal case, the drive train coupling clearance goes to zero at 680°F . This is due to the different thermal expansion rates of the coupling pins and yoke. The drum bearing clearances of 0.0072 in. maximum (cold) and 0.0057 in. maximum (1400°F) result in the equivalent backlashes of $15'$ and $17'$, respectively.

IL to: T. S. Malone
 From: P. H. Horton
 Subject: Drive Position Error of the Advanced Zirc Reactor Drive System, 47 Shield Case

Page 2
 September 3, 1969

Table I below summarizes the maximum position error of Figure 1.

TABLE I
 CALCULATED MAXIMUM POSITION ERROR

Item	TEMPERATURE	
	Atmospheric	Operating
Actuator Brake	1' 0"	1' 0" (1200°F)
Actuator Bearing & Conn	16' 9"	14' 48" (1200°F)
Drive Coupling	5' 48"	0' 29" (1400°F)
Drive Bearings	15' 02"	16' 55" (1400°F)
Total	38'	33' 12"

This compares to the 48' maximum allowable position error and results in a reactivity change of 0.7% for the 33' error.

P. Horton
 P. H. Horton
 Control & Shielding
 Engineering Unit

PHM:23

cc: O. P. Steele, IRI
 W. J. Hensley
 X. Bilibin
 G. P. Gussner
 D. B. Harwood

DRUM POSITION ERROR - AT SHIELD CASE

INCLUDES: ACTUATOR GEARS, BRAKE,
DRIVE COUPLING, DRUM BEARINGS

BACKLASH - (MIL)

APPENDIX B

DRIVE TRAIN COUPLING DESIGN ANALYSIS

PREPARED BY: <i>K.B.</i>	ATOMICS INTERNATIONAL A DIVISION OF NORTH AMERICAN AVIATION, INC.	PAGE NO. 25 OF
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TORQUE ARM OR PIN TYPE COUPLING.
(ALSO SEE R-N69626143)

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SINGLE PIN IN SLOT:

I. HERTZ CONTACT STRESSES AT ROTATING PIN:

CYLINDER ON PLATE:

$$\text{MAX } S_c = 0.591 \sqrt{\frac{4E}{D}}$$

$$\text{MAX } S_c = 161,000 \text{ PSI}$$

$$E = 30 \times 10^6 \text{ PSI}$$

$$D = 1 \text{ INCH}$$

$$P = \frac{(2154)}{(\frac{1}{2})(1\frac{3}{4})} = 2460 \text{ LB/IN}$$

II. BENDING STRESSES ON PIN:

$$S_T = \frac{M_c}{I} = \frac{32M}{\pi d^3}$$

$$d = \frac{1}{2}, M = \frac{2154}{(1.75)} \left(\frac{9}{16} \right) = 692 \text{ IN-LB}$$

IN THE WORST CASE

$$S_T = 56,400 \text{ PSI}$$

STRESS CONR. FACTOR FOR SQUARE SHOULDER WITH FILLET IN CIRCULAR SHAFT

$$K = 1 + .375 \sqrt{\frac{d}{r}}, r = \frac{1}{16}$$

$$K = 2.06$$

$$S_T, S_T = 116,000 \text{ PSI}$$

III. BACKLASH

A. WHEN DESIGNED TO TAKE AN ANGULAR MISALIGNMENT OF ϕ THE ANGULAR BACKLASH CAN BE:

$$\phi = 0^\circ 16' \text{ FOR SHADOW SHIELD } \left[0^\circ 18' 19'' \text{ FOR } 1\frac{3}{4}'' \text{ LEVER ARM} \right]$$

B. FOR 4TH SHEED NO SWIVEL IS REQ'D, SO

$$\phi = 0^\circ 27' 30'' \text{ (TOTAL FOR BOTH COUPLINES)}$$

$$\left[0^\circ 31' 24'' \text{ FOR } 1\frac{3}{4}'' \text{ LEVER ARM} \right]$$

IV. MATERIALS

A. DRUM SHAFT - TA-10W

B. SERRATIONS CLAMP - TA-10W

C. GEAR - RENE 41

D. PINS & ARMS - RENE 41

D. HIRWOOD PREPARED BY:	ATOMICS INTERNATIONAL A DIVISION OF NORTH AMERICAN AVIATION, INC.	PAGE NO. 27 OF TI-696-22-003 REPORT NO.
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CALCULATIONS ON HERTZ CONTACT STRESSES IN TWO POSSIBLE CONFIGURATIONS OF TORQUE-ARM TYPE DRUM DRIVE COUPLING.

CASE I - BALL IN SLOT

THIS IS A BALL ON A PLATE TYPE PROBLEM.
THE APPLICABLE EQUATION IS:

$$q_0 = 0.388 \sqrt[3]{\frac{PE^2}{R_2^2}} \quad (1)$$

WHERE

q_0 = MAX. COMPRESSIVE STRESS (PSI)

P = LOAD ON BALL (LBS.)

E = MOD. OF ELASTICITY (PSI)

R_2 = RADIUS OF BALL (IN)

IN TERMS OF R_2 ,

$$R_2 = \sqrt{(0.388)^3 \frac{PE^2}{q_0^3}}$$

DURING LAUNCH THE ULTIMATE TORQUE IS 2154 IN-LB (2). AT A RADIUS OF 2 INCHES, THIS MAKES "P" EQUAL TO 1077 LB. WE ALSO CHOSE THE FOLLOWING:

$$E = 30 \times 10^6 \text{ PSI}$$

$$q_0 = 200,000 \text{ PSI}$$

INSERTING INTO THE EQUATION:

$$R_2 = \sqrt{\frac{(0.388)^3 (1077) (30 \times 10^6)^2}{(200,000)^3}}$$

$$= \sqrt{\frac{(0.0584) (1.077) (9) (10^3)}{(27)}}$$

$$= \sqrt{2.097}$$

$$R_2 = 1.45 \text{ IN.}$$

THIS MEANS THAT A BALL OF AT LEAST A 2.9 INCH DIAMETER IS REQ'D.
THEREFORE IMPRACTICAL

CASE II - CYLINDER ON PLATE

IF A SQUARE PIN IS APPLIED TO A ROUNDED SLOT, A CYLINDER ON A PLATE PROBLEM RESULTS. THE APPLICABLE EQUATION IS:

$$\text{MAX } S_c = 0.591 \sqrt{\frac{PE}{D}} \quad (3) \quad \text{IF BOTH ARE OF THE SAME MAT'L. AND POISSON'S RATIO = 0.3}$$

WHERE S_c = COMPRESSIVE STRESS (PSI)

P = LOAD PER LINEAR INCH (LB/IN)

D = DIAMETER OF CYLINDER (IN)

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WE ASSUME THE SIDE OF THE SQUARE PIN TO BE $\frac{1}{4}$ INCH LONG, MAKING $p = 4308 \text{ LBS/IN.}$ ALSO, WE AGAIN ASSUME THE FOLLOWING:

$$E = 30 \times 10^6 \text{ PSI}$$

$$S_c = 300,000 \text{ PSI}$$

SOLVING THE EQUATION FOR "D",

$$D = \frac{(0.591)^2 p E}{(S_c)^2}$$

$$= \frac{(0.591)^2 (4308) (30 \times 10^6)}{(300,000)^2}$$

$$D = .502 \text{ in}$$

A RADIUS OF .251 IN. MINIMUM IS REQ'D. FOR THE SLOT.

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BENDING STRESSES ON PIN:

$$P = \frac{2154}{1.75} = 1230 \text{ LB.}$$

$$M = 1230 \left(\frac{3}{8} \right) = 461 \text{ IN-LB.}$$

if $l = \frac{3}{16}$, THE WORST CASE

$$M = 1230 \left(\frac{3}{16} \right) = 692 \text{ IN-LB.}$$

$$S_T = \frac{Mc}{I}, \quad c = \frac{1}{2}d = r, \quad I = \frac{\pi}{64} d^4$$

$$S_T = \frac{M \left(\frac{1}{2} \right) d}{\frac{\pi}{64} d^4} = \frac{32 M}{\pi d^3} = \frac{(512)(32) M}{\pi (27)} = 193 M$$

$$\text{IF } M = 461 \text{ IN-LB}$$

$$S_T = 88,700 \text{ PSI}$$

$$\text{IF } M = 692 \text{ IN-LB}$$

$$S_T = 133,000 \text{ PSI}$$

Using $d = \frac{1}{2}$, WITH NO UNDERCUT

$$S_T = \frac{32 M}{\pi d^3} = \frac{32 (692)}{\pi \left(\frac{1}{2} \right)^3}$$

$$S_T = 56,400 \text{ psi}$$

CONCENTRATION FACTOR FOR SQUARE SHOULDER WITH FILLET IN CIRCULAR SHAFT (PERKINS)

$$K = 1 + .375 \sqrt{\frac{d}{r}}$$

$$\text{IF } d = \frac{1}{2}, \quad r = \frac{1}{16}$$

$$K = 2.06$$

$$S_T = 116,000 \text{ psi}$$

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MISALIGNMENT OF CENTERS:

$$\sin \gamma = e/l$$

$$q = d \sin \gamma$$

$$C = q + d \cos \gamma$$

$$G = C - d$$

$$\text{BACKLASH}(\phi) = \tan^{-1}\left(\frac{G}{l}\right)$$

$$\text{LET } e = .0625 \text{ in, } l = 2 \text{ in, } d = 1/4 \text{ in}$$

$$\sin \gamma = .03125$$

$$\cos \gamma = .99952$$

$$C = d(\sin \gamma + \cos \gamma)$$

$$C = .2576925$$

$$G = .0076925$$

$$\phi = \tan^{-1}(.00384625)$$

$$\phi = 0^\circ 13'$$

COMBINATION MISALIGNMENT OF CENTERS AND ANGULAR MISALIGNMENT:

$$b = \frac{c}{\cos \theta} + \frac{2r}{\cos \theta} - 2r$$

$$b = C \sec \theta + 2r(\sec \theta - 1)$$

$$C = .2576925, \text{ FROM ABOVE}$$

$$\text{CHOOSE: } r = .5 \text{ in}$$

$$\text{AND } \theta = 4^\circ, \sec(4^\circ) = 1.0024$$

$$b = (1.0024)(.2576925) + 0.0024$$

$$b = .260710962$$

$$G' = b - d = .010710962 \text{ in}$$

$$\phi = \tan^{-1}\left(\frac{G'}{l}\right) = \tan^{-1}(.00535548)$$

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$\phi = 0^{\circ}18'$, AND THIS IS WITHOUT ALLOWANCE FOR DIMENSIONAL TOLERANCE.
ASSUMING THAT THE DIMENSIONS ARE AS FOLLOWS:

PIN ACROSS FLATS $.250 \begin{smallmatrix} +.010 \\ -.011 \end{smallmatrix}$

"b" DIMENSION OF SLOT $.261 \begin{smallmatrix} +.003 \\ -.000 \end{smallmatrix}$

$$G' = .264 - .249 = .015$$

$$\phi = \tan^{-1}\left(\frac{.015}{2}\right) = \tan^{-1}(.0075)$$

$\phi = 0^{\circ}26'$, FAR BEYOND THE TOTAL ALLOWABLE SYSTEM
BACKLASH OF $0^{\circ}18'$

IF A PIN IS USED WHICH ROTATES ON AN AXIS PARALLEL TO
THE SYSTEM AXIS OF ROTATION, THE BACKLASH RESULTING FROM MISALIGNMENT
OF CENTERS CAN BE ELIMINATED. THE PIN COULD LOOK LIKE THIS:

IN THIS CASE, THE BACKLASH WOULD RESULT FROM ANGULAR MISALIGNMENT
AND TOLERANCE DIFFERENCES.

$$b = C \sec \theta + 2r (\sec \theta - 1)$$

CHOOSING $C = 0.5$, $\theta = 4^{\circ}$, $r = 0.5$, $l = 2$

$$b = (0.5)(1.0024) + 0.0024$$

$$b = .5036$$

$$G' = .0036$$

$$\phi = \tan^{-1}(.0018) = 0^{\circ}6'$$
, WITHOUT ALLOWANCE FOR DIMENSIONS

CHOOSING THE FOLLOWING DIMENSIONS:

SIDE OF SQUARE $.500 \begin{smallmatrix} +.000 \\ -.001 \end{smallmatrix}$

"b" DIA. OF SLOT $.504 \begin{smallmatrix} +.003 \\ -.000 \end{smallmatrix}$

DIFFERENCE = .008 MAX.

DIA. OF HOLE IN SQUARE $.2500 \begin{smallmatrix} +.0005 \\ -.0000 \end{smallmatrix}$

DIA. OF .25 PIN $.2498 \begin{smallmatrix} +.0000 \\ -.0004 \end{smallmatrix}$

DIFF. = .0011 MAX.

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TOTAL MAX. DIFFERENCE = $G' = .0021$

$$\phi = \tan^{-1}(.00455)$$

$$\phi = 0^{\circ}16'$$

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UNIVERSAL JOINT:

I. SHEAR ACROSS PINS

$$S_s = \frac{T r}{2 \left(\frac{\pi d^3}{4} \right)} = 13,400 \text{ PSI, WHERE } r = 1.375, d = 0.375$$

II. PIN AS CANTILEVERED BEAM - BENDING STRESS

$$P = \frac{T L}{2} = \frac{(2154)(1.375)}{2}$$

$$S_T = \frac{4 P l}{\pi R^3}, l = \frac{1}{4} \text{ IN}, R = \frac{3}{16} \left(\frac{3}{8} \text{ DIA PINS} \right)$$

$$S_T = 37,800 \text{ PSI}$$

IF. $R = \frac{1}{4}$ (HALF-INCH PINS)

$$S_T = 15,995 \text{ PSI}$$

III. BACKLASH THRO JOINT -

A. SHADOW SHIELD

$$\phi = 0^\circ 6' 20''$$

B. 4TH SHIELD - ONLY ONE JOINT AT DRUM END

$$\phi = 0^\circ 9' 30''$$

IV. EXTENDED GEAR FREE - BENDING STRESS

$$I = \frac{b l^3}{12} = \frac{(\frac{1}{4})(1)^3}{12} = \frac{1}{48}$$

$$P = \frac{(2154)(2)}{4.5} = 956 \text{ LB}$$

$$M = P l = 956 \left(\frac{3}{4} \right) = 716 \text{ IN-LB}$$

$$S_T = \frac{M c}{I} = 17,200 \text{ PSI}$$

V. HERTZ STRESSES OF PIN IN SLOT (CYLINDER ON PLATE)

$$\text{MAX } S_c = 0.591 \sqrt{\frac{P E}{D}}$$

$$E = 30 \times 10^6$$

$$P = \frac{2154}{3} = 718 \text{ LB}$$

$$P = 718 / (\frac{1}{4}) = 2872 \text{ LB/IN}$$

$$S_c = 293,050 \text{ PSI}$$

$$D = \frac{3}{8}$$

VI. MATERIALS

A. GEAR - RENE 41

B. SPLINE CLAMP - TA-10W

C. DRUM SHAFT - TA-10W

D. UNIVERSAL JOINT - RENE 41

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SHEAR ON PINS:

$$P = Tr$$

$$A = 2\left(\frac{\pi d^2}{4}\right)$$

$$d = \frac{3}{8} \text{ in}$$

$$r = 1\frac{3}{8} \text{ in}$$

$$T = 2154 \text{ IN-LB}$$

$$S_s = \frac{P}{A}$$

$$= \frac{Tr}{2\left(\frac{\pi d^2}{4}\right)} = \frac{(2154)\left(1\frac{1}{8}\right)}{(\pi)\left(\frac{3}{8}\right)\left(\frac{1}{4}\right)\frac{2}{64}}$$

$$= \frac{(2154)(11)(16)}{9(\pi)}$$

$$S_s = 13,400 \text{ PSI}$$

PIN AS CANTILEVERED BEAM:

$$M_{\text{BY}} M = Pl$$

$$I = \frac{1}{4} \pi R^4$$

$$R = \frac{3}{16}$$

$$S = \frac{MR}{I}$$

$$S_r = \frac{4Pl}{\pi R^3}$$

$$S_r = 37,800 \text{ psi}$$

$$\text{IF } R = \frac{1}{4}$$

$$S_r = 15,995 \text{ psi}$$

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BACKLASH THRU UNIVERSAL JOINT:

ASSUME THAT THE PINS ARE PRESS FIT INTO THE RING

$$\tan \alpha = \frac{e}{r} \quad r = 1\frac{3}{16}''$$

CHOOSING DIMENSIONS & TOLERANCES:

$$\text{FOR HOLE DIA.} \quad 0.3750 \quad \begin{matrix} +.0005 \\ -.0000 \end{matrix}$$

$$\text{FOR PIN DIA.} \quad 0.3748 \quad \begin{matrix} +.0000 \\ -.0004 \end{matrix}$$

$$C(\text{CLEARANCE}) = 0.3755 - 0.3744$$

$$C = 0.0011$$

$$\tan \alpha = \frac{0.00055}{(1\frac{3}{16})} = .00046$$

$$\alpha = 0^{\circ}1'35''$$

$$\text{BACKLASH IN THIS ELEMENT} = 2\alpha = 0^{\circ}3'10''$$

FOR THE ENTIRE UNIVERSAL JOINT, CONSISTING OF TWO ELEMENTS SUCH AS THE ONE DESCRIBED ABOVE, THE BACKLASH IS EQUAL TO 4α

$$\text{TOTAL BACKLASH THRU UNIVERSAL} = 0^{\circ}6'20''$$

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COMPRESSION IN CLEVIS:

$$A = d(D_o - D_i)$$

D_o = OUTER DIA.

$$P_{HL} = 2154 \left[\frac{D_o + D_i}{2} \right]$$

D_i = INNER DIA

d = PIN DIAMETER = $3/8$ in

$$S_c = \frac{P}{A} = \frac{1077(D_o + D_i)}{d(D_o - D_i)}$$

For $D_o = 2\frac{5}{8}$

$D_i = 1\frac{5}{8}$

$$S_c = \frac{1077 \left(\frac{34}{8} \right)}{\frac{3}{8} (1)} = 12,200 \text{ psi}$$

For $D_o = 2\frac{7}{8}$

$D_i = 2\frac{1}{8}$

$$S_c = \frac{1077 \left(\frac{39}{8} \right)}{\left(\frac{3}{8} \right) \left(\frac{1}{2} \right)} = 29,800 \text{ psi}$$

BEAM STRESS ON CLEVIS:

$$P = \frac{2154(2)}{(D_o + D_i)}$$

$D_o = 2\frac{5}{8}$

$D_i = 2\frac{1}{8}$

$$P = \frac{2154(2)}{(3\frac{3}{8})}$$

$$M = \frac{3}{4}P = \frac{2154(6)}{(19)}$$

$$S_t = \frac{Mc}{I} = \frac{(2154)(6)(48)}{19(2)}$$

$$S_t = 17,200 \text{ psi}$$

ASSUME X-SECTION

TENSILE SHEAR:

$$S_s = \frac{P}{A}, P = 1077(D_o + D_i), A = (D_o - D_i)t$$

$D_o = 2\frac{5}{8}$, $D_i = 2\frac{1}{8}$, $t = \frac{1}{2}$

$$S_s = \frac{1077 \left(\frac{34}{8} \right)}{\left(\frac{1}{2} \right) \left(\frac{1}{2} \right)}$$

$$S_s = 20,500 \text{ psi}$$

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D_o = HOLE DIA.

D_i = PIN DIA.

L = HOLE DEPTH

$$\tan \beta = \frac{D_o - D_i}{L}, \text{ ASSUMING CURVATURE} = 0$$

PIN LOOKS LIKE THIS:

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BACKLASH DUE TO TOLERANCES BETWEEN PW AND SLOT:

ASSUME:

- (1) NO ROCKING EFFECT IS PRESENT
- (2) CURVATURE = 0
- (3)

$$\tan \gamma = \frac{\left(\frac{W-C}{2}\right)}{R_2} = \frac{W-C}{2R_2}$$

EFFECT OF ROUND PIN IN HOLE ON BACKLASH

REFERRING TO OVERALL EFFECT SKETCH:

$$q = \left(\frac{1}{2}L + A\right) \tan \beta = \left(\frac{1}{2}L + A\right) \frac{(D_o - D_i)}{L} = \frac{(L + 2A)(D_o - D_i)}{2L}$$

ϕ = ANGULAR BACKLASH CAUSED BY PIN ROCKING IN HOLE

$$\tan \phi = \frac{q}{R_2}$$

TOTAL ANGULAR BACKLASH (θ):

$$\theta = 2\alpha = 2(\phi + \gamma)$$

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AN EXAMPLE, USING DIMENSIONS RELATED TO OTHER "UNIVERSAL":

$$\text{ROUND PIN DIA. } (D_o) \quad .3748 \begin{matrix} +.0000 \\ -.0004 \end{matrix}$$

$$\text{ROUND HOLE DIA. } (D_i) \quad .3750 \begin{matrix} +.0005 \\ -.0000 \end{matrix}$$

$$\text{WORST CASE } (D_o - D_i) = .3755 - .3744 = .0011$$

$$\text{ASSUME: } L = 0.250, A = .125$$

$$q = \frac{(0.250 + 2(0.125))(.0011)}{2(.250)}$$

$$q = .0011$$

$$\text{IF } R_2 = 1\frac{7}{16}''$$

$$\tan \phi = \frac{.0011}{1.4375} = .0007652$$

$$\phi = 0^\circ 2' 39''$$

$$\text{SLOT WIDTH } (W) \quad .5000 \begin{matrix} +.0005 \\ -.0000 \end{matrix}$$

$$\text{SQUARE DIM. } (C) \quad .4998 \begin{matrix} +.0000 \\ -.0004 \end{matrix}$$

$$\text{WORST CASE } (W - C) = .5005 - .4994 = .0011$$

$$\tan \gamma = \frac{.0011}{2(1.4375)} = .0003826$$

$$\gamma = 0^\circ 1' 19''$$

$$\alpha = \phi + \gamma = 0^\circ 3' 58''$$

$$\theta = 2\alpha = 0^\circ 7' 56'' \approx 0^\circ 8'$$

ADD THE BACKLASH THRU ANOTHER, BUT STANDARD, UNIVERSAL ELEMENT

$$\theta_T = 0^\circ 7' 56'' + 0^\circ 3' 10'' = 0^\circ 11' 6''$$

COMPARE THIS TO THE VALUE FOR A STANDARD UNIVERSAL, CONSISTING OF TWO STANDARD ELEMENTS AND A SLIDING SCRAPPED SHAFT.

$$\theta_{\text{SHAFT}} = 0^\circ 7'$$

$$\theta_{\text{UNIVERSAL}} = 0^\circ 6' 20''$$

$$\theta_T = 0^\circ 13' 20''$$

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HERTZ STRESSER IN SLOT:

CYLINDER ON A FLAT PLATE SITUATION

$$\text{MAX } S_c = 0.591 \sqrt{\frac{P E}{D}}$$

IF BOTH ARE SAME
MATER. & POISSON'S RATIO
IS 0.3

$$E = 26.4 \times 10^6 \text{ PSI}$$

$$S_c = 300,000 \text{ PSI}$$

$$P = \frac{2154 \text{ in-lb}}{3 \text{ in}} = 718 \text{ lb}$$

MINIMUM CONTACT LENGTH IS $\frac{1}{4}$ "

$$p = \frac{718}{\frac{1}{4}} = 2872 \text{ LB/IN}$$

$$D = \frac{(0.591)^2 P E}{(S_c)^2}$$

$$= \frac{(0.591)^2 (2872) (26.4 \times 10^6)}{(3 \times 10^5)^2}$$

$$D = 0.234 \text{ IN}$$

CHISEL USE $\frac{3}{8}$ " PIN

D. H. H. H. H. PREPARED BY:	ATOMICS INTERNATIONAL A DIVISION OF NORTH AMERICAN AVIATION, INC.	PAGE NO. 44 OF TI-696-22-003 REPORT NO.
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DATE:		

USING THE TEETH OF THE GEAR AS SPLINES BY EXTENDING THEM:

$$\left. \begin{array}{l} \text{PITCH DIA.} = 2.875 \\ \text{ROOT DIA} = 2.83 \end{array} \right\} 40^\circ \text{ OF GEAR HAS 10 TEETH}$$

$$\text{SHEAR AREA} = [(2.83)\pi] \left(1 - \frac{40}{360}\right) \left(\frac{1}{2}\right) = (2.83)\pi \left(\frac{8}{9}\right) \left(\frac{1}{2}\right) \times 1$$

$$\text{SHEAR FORCE} = \frac{2.875 (2154)}{2}$$

$l = \text{length of fit}$

$$S_s = \frac{(2.875) (2154) \left(\frac{1}{2}\right) l}{(2.83)(\pi) \left(\frac{8}{9}\right) \left(\frac{1}{2}\right)} = \frac{9 (2.875) (2154) l}{(2.83)(\pi) (8)}$$

$$S_s = 784 l$$

$$\frac{S_s}{l} = 784 \text{ PSI/in.}$$

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BELLOWS COUPLING - YOKE (HYBRID)

I. BACKLASH

A. SHIELD SHIELD

$$\phi = 0^{\circ} 0' 0''$$

B. 4 π SHIELD - SLOTTED COUPLING AT BOTTOM

$$\phi = 0^{\circ} 3' 10''$$

II. CONNECTING PINS - SHEAR

AT A 1.5" RAD FROM THE CNTR

$$P = \frac{2154}{1.5} = 1436 \text{ LB}$$

$$A = m \frac{\pi}{4} (d^2), \quad m=2, \quad d = \frac{3}{16}$$

$$S_s = 25,800 \text{ psi}$$

IF $d = \frac{1}{4}$ "

$$S_s = 13,650 \text{ psi}$$

III. SHEAR IN BELLOWS SUPPORTING MEMBER

$$S_s = \frac{2 T r_i}{\pi (r_i^4 - r_o^4)}, \quad r_i = \sqrt{\frac{23}{32}}, \quad r_o = \sqrt{\frac{22}{32}}$$

$$S_s = 3490 \text{ PSI}$$

IV. MATERIALS

A. GEAR - RENE 41

B. DRUM SHAFT - TA-10W

C. SPLINE CLAMP (LOWER) - TA-10W

D. BELLOWS - 304 SST OR INCONEL 718

E. TUBE, LOWER CLEVIS, PINS, UPPER SPLINE CLAMP, ETC. -

SAME AS BELLOWS

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TWO BELLOWS COUPLING:

I. BACKLASH

A. SHADOW SHIELD

$$\phi = 0^{\circ} 0' 0''$$

B. 4 π SHIELD

$$\phi = 0^{\circ} 4' 8'' \quad \text{AT LAUNCH (5'48")}$$

$$\phi = 0^{\circ} 0' 0'' \quad \text{AT TEMP.}$$

II. DIFFERENTIAL THERMAL EXPANSION AT LOWER PINS

@ 1400°F

$$\text{TA-10W} \quad \alpha = 3.75 \times 10^{-6} \text{ IN/IN-}^{\circ}\text{F}$$

$$\text{INCONEL 718} \quad \alpha = 9.12 \times 10^{-6} \text{ IN/IN-}^{\circ}\text{F}$$

$$\Delta\alpha = 5.37 \times 10^{-6} \text{ IN/IN-}^{\circ}\text{F}$$

$$\Delta T = 1330^{\circ}\text{F}$$

$$\frac{\Delta L}{L} = 7.142 \times 10^{-3} \text{ IN/IN}$$

$$\text{IF } L = .375$$

$$\Delta L = 2.68 \times 10^{-3} \text{ IN} = .0027 \text{ IN}$$

$$\text{MAX. CLEARANCE} = .0024 \text{ IN}$$

$$\text{IF } L = .4375$$

$$\Delta L = 3.12 \times 10^{-3} \text{ IN} = .0031 \text{ IN}$$

$$\text{MAX. CLEARANCE} = .0019 \text{ IN}$$

III. BENDING LOADS ON LOWER PINS (WORST CASE)

$$T = 1436 \text{ IN-LB} \quad (\text{FOR FATIGUE})$$

$$D = 3.562 \text{ IN}$$

$$P, \text{ APPLIED LOAD} = 413.1 \text{ LB.}$$

$$S = \frac{Mc}{I}, \quad M = Pl$$

$$\frac{I}{c} = \frac{\pi d^3}{32}$$

$$\text{IF } d = 3/8"$$

$$\frac{I}{c} = .00518 \text{ IN}^3$$

$$\text{IF } l = 5/8"$$

$$S = \frac{(5)(413.1)}{(8)(.00518)} = 48,843 \text{ PSI}$$

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TWO BELLOWS COUPLINGS (CONT'D.)

$$\text{STRESS CONC. FACTOR } K = 1 + .375 \sqrt{\frac{d}{r}}$$

$$d = \frac{7}{8}, r = \frac{1}{8}$$

$$K = 1 + .375 \sqrt{3} = 1.6495$$

$$S = 148,843 (1.6495) = 80,567 \text{ PSI}$$

III. DIFF. THERMAL EXPANSION ACROSS LOWER CLEVIS

$$\frac{\Delta L}{L} = 7.142 \times 10^{-3} \text{ in/in}$$

$$L = 3.562$$

$$\Delta L = 25.44 \times 10^{-3} \text{ in} = .0254 \text{ in}$$

MAX. DIM. TO EDGE OF PINS:

$$3.572 + \frac{.4271 + .3746}{2} + \frac{.0027 + .0019}{2} = 3.9806$$

MIN DIM. TO EDGE OF HOLES:

$$3.552 + \frac{.4375 + .430}{2} = 4.0158$$

CLEARANCE AT TEMP:

$$4.0158 - 3.9806 = .0352 \text{ MIN.}$$

V. BELLOWS

A. UPPER BELLOWS

10 CONVOLUTIONS. (3.50 O.D. X 2.688 I.D.)

.005 THK. DISKS

SOLID HT = .160

FREE HT = .375

MOVEMENT OF .215 ALLOWABLE

TOLERABLE = 5220, OR A M.S. OF 42% OVER TULT

$$K = 24.9 \text{ #/in @ } 1200^\circ\text{F}$$

$$K = 30.6 \text{ #/in @ } 70^\circ\text{F}$$

B. LOWER BELLOWS

8 CONVOLUTIONS (4.000 O.D. X 3.125 I.D.)

.005 THK. DISKS

SOLID HT = .128

FREE HT = .375

MOVEMENT OF .247 ALLOWABLE

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TWO BELLOWS COUPLING (CONT'D)

TALLOWABLE = 6400 IN-LB OR AN M.S. OF 197% OVER TULT
 EST. NAT. FREQ. = 14.1 Hz

RANDOM LOAD = 2.6g

ACCELERATION LOAD (MAX) = 5g = A_{MAX}

$F = KX = (34) \left(\frac{3}{16} \right) = 6.375 \text{ LB (RESTRAINED TO } \frac{3}{16} \text{ INCH)}$

$$a = \frac{F}{m} = \frac{6.375}{.888} = 7.17g$$

$$A = 7.17 - 2.6 = 4.57g$$

IF $A > 4.57$, BANGING OCCURS

$A > 4.57$ FOR 5-10 SECONDS BEFORE
 FIRST STAGE SEPARATION

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UPPER BELLOWS: (MAT'L. - INCONEL 718)

3.50 O.D. x 2.688 I.D.

$D_m = 3.094$ in

$L = .40$ in

$T = 2154$ in-# @ 70°F FOR 30,000 CYCLES

$F_{ult} = 169,000$ @ 1200°F $E = 23.6 \times 10^6$

$= 205,000$ @ 70°F $E = 29 \times 10^6$

$F_y = 142,000$ @ 1200°F

$= 173,000$ @ 70°F

1. Cycle Life

$F_{P 30,000} F_{Ver} = 110,000$ TENSILE (FROM TI-095-22-043)

$f_a = 0$

$f_r = f_s = \frac{f_{max} - f_{min}}{2}$; $f_{ves} = .65 f_{Ver}$

$f_{ves} = .65(110,000) = 1.1 f_s$

$f_s = 65,000$ PSI

$T = g(\pi D)$

$f_s = \frac{g}{t} = \frac{T}{\pi D t}$

$f_s = 65,000 = \frac{2154}{\pi(2.688) t}$

$t_{min} = \frac{2154}{\pi(2.688)(65,000)} = .00393$ "

USE $t = .005$

SEE HT. = $[3.2t + .01] N_c = 0.016 N_c$

DETERMINE N_{max} :

$T_{ALLOW} = \frac{24.9 E t^{2.25} D_m}{\sqrt{N_D L}}$
(CRISTENCO)

$N_D = 2 N_c$

$L = \frac{OD-ID}{2}$

$\sqrt{N_D L} = \frac{24.9 (29 \times 10^6) (.005)^{2.25} (3.094)}{2154} = \frac{2.49 (29) (6.6) (3.094)}{21.54}$

$= 6.74$

$N_D = \frac{(6.74)^2}{2} = 116$ OR $N_c = 59$ CONVOLUTIONS

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N_c = TOTAL BELLOWS CONVOLUTIONS, M_c = AFFECTED CONVOLUTIONS

$M_c = \frac{1}{3} N_c$ FOR $\frac{1}{3}$ OF BELLOWS (CONSERVATIVE CASE)

$$M_c \delta = R\theta = \frac{D_m}{2} (2^\circ) \left(\frac{R\alpha}{57.3} \right) = \text{TOTAL DEFLECTION}$$

$$\delta = \frac{3.094(2^\circ)}{2 M_c (57.3)} = \frac{.0556}{M_c}$$

at 1200°F

$$f_\delta = \frac{3 E \delta t}{2 N_c L^2} = \frac{3 (23.6) \delta (.005) (10^6)}{2 (3 M_c) (.4)^2} = \frac{1}{2} F_y = 70,000 \text{ psi}$$

SOLVE FOR M_c

$$M_c = \frac{(23.6) \delta (.005) (10^6)^{3/2}}{(70,000) 2 (.16)} = \frac{(2.36) (5) \delta}{14 (.16)}$$

$$\text{SINCE } \delta = \frac{.0556}{M_c}$$

$$(M_c)^2 = \frac{(2.36) (5) (.0556)}{14 (.16)} = .293$$

$$M_c = .541 = \frac{1}{3} N_c$$

$$\text{So } N_c = 3 (.541) = 1.623 \text{ OR 2 CONV. REQ'D.}$$

SPRING RATE

$$K = \frac{(1.1) \pi E D_m t^3}{N_D L^3}$$

$$N_D = 2 N_c$$

$$\text{Sol. HT} = 0.016 N_c$$

$$\text{OR } K = \frac{(.55) \pi D_m t^3 E}{N_c L^3}$$

@ 1200°F

$$K N_c = \frac{(.55) \pi (3.094) (.005)^3 (2.36) (10^7)}{(.4)^3}$$

$$K N_c = 249$$

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CHOOSE 10 CONVOLUTION BELLOWS, $\begin{cases} K = 24.9 \text{ #/in @ } 1200^\circ\text{F} \\ K = 32.6 \text{ #/in @ } 75^\circ\text{F} \end{cases}$

$$\text{Solid Ht.} = 10(.016) = .160$$

MAKE FREE HEIGHT = .375"

ALLOWANCE FOR MOVEMENT = .215"

$$T_{allow} = \frac{(24.9)(29)(.005)^{2.25}(3.054)(10^6)}{\sqrt{20(.4)}}$$

$$T_{allow} = 5220, \text{ OR A M.S. OF } 142\%$$

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LOWER BEZELONS: (MAT'L. - INCONEL 718)

4 INCH O.D. X 3.125 I.D.

$$D_m = 3.5625$$

$$L = .44$$

$T = 2154 \text{ in}^{-\frac{1}{2}}$ @ 70°F FOR 30,000 cycles

$f_s = 65,000 \text{ psi}$ (SEE P.10)

$$f_s = \frac{T}{\pi D t} = \frac{2154}{\pi (3.125) t}$$

$$t_{min} = \frac{2154}{\pi (3.125) (65,000)} = .00338"$$

use $t = .005$

$$\text{Sol. Hr} = (3.2t + 1.0) N_c = 0.016 N_c$$

DETERMINE N_{max} :

$$T_{allow} = \frac{24.9 E t^{2.25} D_m}{\sqrt{N_D L}}$$

$$\sqrt{N_D L} = \frac{(24.9)(29 \times 10^6)(.005)^{2.25}(3.5625)}{2154}$$

$$= \frac{(24.9)(2.9)(6.6)(3.5625)}{2154} = 7.94$$

$$N_D = \frac{(7.94)^2}{.44} = 143 \text{ or } N_c = 71 \text{ CONVOLUTIONS}$$

N_c = TOTAL BEZELONS CONVOLUTIONS, m_c = AFFECTED CONVOLUTIONS

$$m_c \delta = R \theta = \frac{D_m (2^\circ / \text{Rad})}{2 (57.3)} = \text{TOTAL DEFLECTION}$$

$$\delta = \frac{3.56}{m_c (57.3)} = \frac{.0622}{m_c}$$

@ 1200°F

$$f_s = \frac{3 E \delta t}{2 N_c L^2} = \frac{3 (23.6) \delta (.005) (10^6)}{2 (3 m_c) (.44^2)} = \frac{1}{2} = 7,000$$

Slipform

$$m_c = \frac{(23.6) \delta (.005) (10^6)}{2 (.44^2) (70,000)}$$

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$$\delta = \frac{.0622}{m_c}$$

$$(m_c)^2 = \frac{(23.6)(.005)(10^6)(.0622)}{2(.44)^2(70,000)} = \frac{(2.36)(.311)(5)}{(70)(.1936)}$$

$$(m_c)^2 = .271$$

$$m_c = .521 = \frac{1}{3}N_c$$

$N_c = 1.563$ OR 2 CONVOLUTIONS MINIMUM REQ'D.

SPRING RATE

$$K = \frac{(.55) \pi (D_m) t^3 E}{N_c L^3}$$

@ Room Temp

$$KN_c = \frac{(.55) \pi (3.56) (.005)^3 (29)(10^6)}{(.44)^3}$$

$$= \frac{(.55) \pi (3.56) (12.5)(10^6)(29)(10^6)}{8.5184 \times 10^{-2}}$$

$$= \frac{(.55) (\pi) (3.56) (12.5) (29)}{8.5184}$$

$$KN_c = 262$$

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CHOOSING A 10 CONVULSION BELLOWS $\begin{cases} K = 21.4 \text{ #/in @ } 1200^\circ\text{F} \\ K = 26.2 \text{ #/in @ } 70^\circ\text{F} \end{cases}$
 SOLID HEIGHT = $10(.016) = .160"$
 MAKE FREE HEIGHT = $.375"$
 ALLOWANCE FOR MOVEMENT = $.215$

$$T_{allow} = \frac{(24.2)(2)(3.56)(.005)^{2.5}(100)}{\sqrt{20(44)}}$$

 $T_{allow} = 5730 \text{ in-LB, GIVING A M.S. OF } 178\%$

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LOWER BELLOWS

CITRUSE 14 CONVOLUTIONS, $K = 18.7 \text{ \#/IN.}$

$$O.D. = 4.0$$

$$I.D. = 3.125$$

$$t = .005$$

$$P = .296 \text{ \#/IN}^2$$

$$W = \frac{\pi}{4} P t (2N_c) (O.D.^2 - I.D.^2)$$

$$W = \frac{\pi}{2} P t N_c (16 - 9.73)$$

$$= \frac{\pi (.296) (.005) (7) (6.27)}{2}$$

$$= .24$$

$$W_c = \frac{1}{2} W = .08 \text{ LB}$$

WEIGHT OF LOWER RING & PINS

4 $\frac{1}{16}$ OD x $\frac{7}{32}$ WALLY $\frac{7}{8}$ LG

$$W_1 = (.296) \pi \left(\frac{7}{32}\right) \left(4\frac{1}{16} - \frac{7}{32}\right) \left(\frac{7}{8}\right)$$

$$= (.296) \pi (.2187) (3.8437) (.375)$$

$$W_1 = .293 \text{ \#}$$

$$W_2 = \frac{\pi}{4} (4)^2 (.296) (.125) = .465 \text{ \#}$$

PINS

$$W_3 = (1.125) (.296) \frac{\pi}{4} \left[\left(\frac{7}{8}\right)^2 + \left(\frac{7}{16}\right)^2 \right]$$

$$= (1.125) (.296) \frac{\pi}{4} (.332) = .084 \text{ \#}$$

$$W_{TOT} = .084 + .465 + .293 + .09 = .918 \text{ \#}$$

$$\omega = \frac{1}{2\pi} \sqrt{\frac{(18.7)(.396)}{(.918)}} = \frac{1}{2\pi} \sqrt{7900}$$

$$\omega = \frac{88.9}{2\pi} = 14.1 \text{ Hz}$$

Handwritten note: 14.1 Hz

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$$5.0 - 2.2 = 2.8g$$

$$P = (2.8)(.918) = 2.57\#$$

$$\delta = \frac{(2.57)}{(19.7)} = .1375"$$

$$\text{Solid Ht} = (14)(.016) = .224$$

$$\text{CHOOSE FREE HT} = .375$$

$$\text{ALLOW. FOR MOVEMENT} = .375 - .224 = .151"$$

CHOOSE NOW A BELLOWS WHERE $N_c = 8$

$$(K = 34 \text{ LB/IN } (@ 70^\circ\text{F}))$$

$$W = .888 \text{ LB } K = 24.6 \#/\text{IN } @ 1200^\circ\text{F}$$

$$\omega = \frac{1}{2\pi} \sqrt{\frac{(34)(396)}{.888}} = \frac{121.5}{2\pi} = 19.3 \text{ Hz}$$

$$\text{RANDOM VIBRATION} = 2.6g$$

$$\text{Solid Ht.} = 8(.016) = .128$$

$$\text{FREE HT} = .375$$

$$\text{MOVEMENT} = (.375 - .128) = .247$$

IF MOVEMENT IS RESTRAINED TO $\pm 3/16"$ BY STOPS

$$F = Kx = (34)\left(\frac{3}{16}\right) = 6.375 \text{ LB}$$

$$F = ma$$

$$a = \frac{F}{m} = \frac{6.375}{.888} = 7.17g$$

$$A = 7.17 - 2.6 = 4.57g$$

IF ACCELERATION OF BOOSTER $> 4.57g$, BANGING OCCURS

$$T_{\text{ALLOW}} = \frac{(29.5)(29)(16.0)(3.5625)}{\sqrt{16(44)}}$$

$$T_{\text{ALLOW}} = 6400 \text{ IN-LB}$$

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SUPPORTING CALCULATIONS FOR ERH FLEXIBLE COUPLING:

DRUM UPPER SHAFT:

USING A $\frac{5}{16}$ " (NOMINAL) SERRATION AT THE DRUM SHAF

TOP:

$$\text{MINOR DIA.} = .5625 \begin{matrix} +.0000 \\ -.0030 \end{matrix}$$

$$\text{PITCH DIA.} = .5938$$

$$\text{MAJOR DIA.} = .625 \begin{matrix} +.000 \\ -.003 \end{matrix}$$

$$C = \text{CIRCUMFERENCE} = \pi(\text{ROOT DIA}) = 1.767 \text{ in.}$$

L, ASSUME UNIT LENGTH

$$F = \text{SHEAR FORCE} = \frac{T}{L} = T / (.5625/2)$$

$$A = \text{SHEAR AREA} = (\frac{1}{2})(C)(L)$$

$$S_s = \text{SHEAR STRESS} = F/A$$

$$S_s = \frac{4T}{\pi D^2 L}$$

$$\text{FOR ULT. } T = 2154 \text{ IN-LB}$$

$$\text{FOR LIMIT } T = 1436 \text{ IN-LB}$$

$$S_s = 8665 \text{ PSI IN } 1" \text{ LENGTH}$$

IN $\frac{1}{4}"$ LENGTH

$$S_s = 34,660 \text{ PSI}$$

IN $\frac{5}{16}"$ LENGTH

$$S_s = 27,700$$

$$S_s = 5777 \text{ PSI IN } 1" \text{ LENGTH}$$

IN $\frac{1}{4}"$ LENGTH

$$S_s = 23,100 \text{ PSI}$$

IN $\frac{5}{16}"$ LENGTH

$$S_s = 17,800 \text{ PSI}$$

FROM MATLS IN DES. ENGR. MID-OCTOBER, 1964, VOL. 60, NO. 5

FOR TA-10W @ ROOM TEMP.

$$\sigma_y = 158,000 \text{ PSI}$$

$$\sigma_u = 160,000 \text{ PSI}$$

ASSUME SHEAR PROPERTIES AS .65 OF TENSILE PROPERTIES

$$\tau_y = 102,700 \text{ PSI}$$

$$\tau_u = 104,000 \text{ PSI}$$

ASSUME ENDURANCE LIMIT IS .25 TIMES ULT. STRENGTH (10^6 CYCLES)

$$\tau_e = 26,000 \text{ PSI}$$

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$$M.S._{torque} = \frac{104,000}{34,660} - 1.0 = 2.0$$

$$M.S._{yield} = \frac{102,700}{23,100} - 1.0 = 3.44$$

M.S. = 2.0 or 200% IN A 1/4" LENGTH
SPLINE ENGAGEMENT

FOR THE SPLINED CONNECTION AT THE UPPER BELLLOWS:

MAT'L. = INCONEL 718

PREP @ ROOM TEMP.

$$\sigma_y = 170,000 \text{ psi} ; \quad \gamma_y = 110,500 \text{ psi}$$

$$\sigma_u = 212,000 \text{ psi} , \quad \gamma_u = 138,000 \text{ psi}$$

$$M.S._{torque} = \frac{139,000}{34,660} - 1.0 = 2.97$$

$$M.S._{yield} = \frac{110,500}{23,100} - 1.0 = 3.78$$

M.S. = 2.97 or 297%

STRESS AT UNDERCUT

FROM PARK, P. 355, FOR SQUARE STUB

$$K = 1 + 0.188 \sqrt{\frac{d}{r}} , \quad d = \text{MINOR DIA} \\ r = \text{FILLET RAD}$$

$$K = 1 + 0.188 \sqrt{\frac{.531}{.062}}$$

$$K = 1.55$$

$$S_s = \frac{KT_c}{J}$$

$$J/c = \frac{\pi d^3}{16} = \frac{\pi (.531)^3}{16}$$

$$S_s = \frac{1.55}{2.935 \times 10^{-2}} T = 52.8 T$$

$$@ T = 2154 \text{ IN-LB (LT.)}$$

$$S_s = 114,000 \text{ PSI}$$

$$M.S._{TENSILE} = \frac{139,000}{114,000} - 1 = 0.21$$

$$@ T = 1436 \text{ IN-LB (LIMIT)}$$

$$S_s = 76,000 \text{ PSI}$$

$$M.S._{TENSILE} = \frac{110,500}{76,000} - 1 = 0.45$$

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UNDERCUT ON DRUM SHAFT BELOW SPLINES:

$$D = .5625$$

$$d = .550$$

$$r = .05$$

Use 260,000 CYCLE FATIGUE
STRENGTH = 71,500 PSI

$$D/d = \frac{.5625}{.550} = 1.02$$

$$r/d = \frac{.05}{.550} = .091$$

$$K = 1.3$$

SHIGLEY, P.620

$$J/c = \frac{\pi d^3}{16} = \frac{\pi (.550)^3}{16} = .0328$$

$$\gamma_s = \frac{KT}{(J/c)} = \frac{1.3}{.0328} T = 39.6 T$$

$$@ \text{ULT. } \gamma_s = 85,500 \text{ PSI} \quad @ \text{LIMIT } \gamma_s = 56,800 \text{ PSI}$$

$$M.S. = \frac{104,000}{85,500} - 1.0 = 0.22$$

$$M.S. = \frac{71,500}{56,800} - 1.0 = 0.25$$

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LOWER BELLOWS TO DRUM YOKE CONNECTION.

$$\sin \theta = \frac{(0.0012 + 0.00095)}{3.562} = \frac{0.00215}{3.562}$$

$$\sin \theta = 0.0006034$$

$$\theta = 0^{\circ} 2' 4''$$

$$\text{BACKLASH} = 2\theta = 0^{\circ} 4' 8''$$

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<u>BACKLASH - FULL BELLOWS</u>		

CHANGE THE DIMENSIONS TO LOOSEN UP THE HOLES:

$$\left\{ \begin{array}{l} \text{PIN} \quad .3740 \quad \begin{array}{l} +.0030 \\ -.0010 \end{array} \\ \text{SLOT} \quad .3756 \quad \begin{array}{l} +.0010 \\ -.0005 \end{array} \end{array} \right\}$$

$$\begin{array}{l} \text{MAX. CLEARANCE} = .0030 \\ \text{MIN.} \quad \quad \quad = .0010 \end{array}$$

$$\sin \theta = \frac{(.0030)}{(3.562)} = .000842$$

$$\theta = 0^{\circ} 2' 54''$$

$$\text{BACKLASH} = 2\theta = 0^{\circ} 5' 48''$$

BACKLASH AT TEMPERATURE (1400°F)

INCONEL

$$\alpha = 5.12 \times 10^{-6} \text{ IN/IN-}^{\circ}\text{F}$$

$$\frac{\Delta L}{L} = 12.123 \times 10^{-3} \text{ IN/IN}$$

TA-16W

$$\alpha = 3.75 \times 10^{-6} \text{ IN/IN-}^{\circ}\text{F}$$

$$\frac{\Delta L}{L} = 4.997 \times 10^{-3} \text{ IN/IN}$$

WORST CASE

$$L = .373 \text{ IN}$$

$$\Delta L = 4.524 \times 10^{-3} \text{ IN}$$

$$D' = .37736 \quad \text{LOSS BY .00052''}$$

$$L = .4350$$

$$\Delta L = 5.276 \times 10^{-3} \text{ IN}$$

$$D' = .44028$$

$$L = .376 \text{ IN}$$

$$\Delta L = 1.875 \times 10^{-3} \text{ IN (PIN SLOT)}$$

$$D' = .37788$$

$$L = .4380$$

$$\Delta L = 2.184 \times 10^{-3} \text{ IN (PIN-HOLE)}$$

$$D' = .44018$$

INTERFERE BY .0001''

$$\sin \theta = \frac{.00026}{3.562} = .000073$$

$$\theta = 0^{\circ} 0' 14.5''$$

$$2\theta = 0^{\circ} 0' 29''$$

BEST CASE:

$$L = .374 \text{ IN}$$

$$\Delta L = 4.526 \times 10^{-3} \text{ IN}$$

$$D' = .37854 \quad \text{INTERFERE BY .00067''}$$

$$L = .436 \text{ IN}$$

$$\Delta L = 5.298 \times 10^{-3} \text{ IN}$$

$$D' = .44129 \text{ IN}$$

INTERFERE BY .00211''

$$L = .375 \text{ IN}$$

$$\Delta L = 1.870 \times 10^{-3} \text{ IN}$$

$$D' = .37687$$

$$L = .437 \text{ IN}$$

$$\Delta L = 2.173 \times 10^{-3} \text{ IN}$$

$$D' = .43918 \text{ IN}$$

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DATE: 8-29-69		

LOWER BELLOWS PIN TO DRUM YOKE
 HERTZ STRESSES FOR CYLINDER ON PLATE

$$\text{MAX } S_c = 0.798 \sqrt{\frac{p}{D \left[\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2} \right]}}$$

$$D = .374"$$

$$p = \frac{1436(4)}{3.562} = 1612 \text{ LB/IN.}$$

$$\nu_1 = \nu_2 = 0.3$$

$$E_1 = 21 \times 10^6 \text{ psi}$$

$$E_2 = 30 \times 10^6 \text{ psi}$$

$$\text{MAX } S_c = 0.798 \sqrt{\frac{1612}{(.374) \left[\frac{.91}{21 \times 10^6} + \frac{.91}{30 \times 10^6} \right]}}$$

$$= 0.798 \sqrt{\frac{1612}{(.374)(.91)(10^7) \left(\frac{1}{21} + \frac{1}{30} \right)}}$$

$$= 0.798 \sqrt{\frac{(1612)(21)(10^7)}{(.374)(.91)(17)}}$$

$$= 0.798 \sqrt{5.85 \times 10^{10}}$$

$$= 0.798 (2.42) (10^5)$$

$$\text{MAX } S_c = 1.23 \times 10^5 = 123,000 \text{ psi}$$

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BENDING IN CONNECTING PINS: (WORST CASE) (LWR BELLOWS PINS)

$$P = T/D \quad T = 2154 \text{ IN-LB (ULT.) or } 1436 \text{ IN-LB (LIMIT)}$$

$$M = Pl \quad D = 3.562"$$

$$l = 5/8"$$

USING THE SMALLER (3/8" DIA.) PIN

$$I_c = \frac{\pi d^3}{32} = .00518 \text{ IN}^3$$

$$S = \frac{M}{I/c} = \frac{Tl}{D(I/c)} = 33.9 T$$

$$S_{\text{LIMIT}} = 48,700 \text{ psi}$$

$$S_{\text{ULT}} = 73,000 \text{ psi}$$

$$K = 1 + .375 \sqrt{\frac{d}{r}} \quad \text{FROM ROARK P.355}$$

$$\left. \begin{array}{l} d = 3/8 \\ r = 1/8 \end{array} \right\} d/r = \frac{3/8}{1/8} = 3$$

$$K = 1.65$$

$$S_{\text{ULT}} = 120,500 \text{ psi} \quad M.S._{\text{ULT}} = \frac{212,000}{120,500} - 1 = 0.76$$

$$S_{\text{LIMIT}} = 80,500 \text{ psi} \quad M.S._{\text{LIMIT}} = \frac{170,000}{80,500} - 1 = 1.11$$

$$M.S. = 0.76 \text{ or } 76\%$$

SHAFT IN TORQUE TRANSMISSION TUBE (HALLOWED SECTION)

$$S_s = \frac{2Tr_1}{\pi(r_1^4 - r_0^4)} \quad , r_1 = 0.5, R_2 = 13/32, T = 2154 \text{ or } 1436 \text{ IN-LB}$$

$$S_s(\text{ULT.}) = 21,540 \text{ psi}$$

$$S_s(\text{LIMIT}) = 14,360 \text{ psi}$$

$$M.S._{\text{ULT}} = \frac{133,000}{61,540} - 1.0 = 5.4$$

$$M.S._{\text{LIMIT}} = \frac{110,500}{14,360} - 1.0 = 6.8$$

$$M.S. = 5.4 \text{ or } 540\%$$

SHAFT IS SOLID TO REDUCE STREAMING

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$$\theta = \frac{360 T l}{\pi^2 G (r_i^4 - r_o^4)} \quad , \text{ in degrees (ROTAK)}$$

$$\text{Max } S_s = \frac{2 T r_i}{\pi (r_i^4 - r_o^4)} \quad , \text{ in psi (ROTAK)}$$

USING THESE PROPERTIES

$$G = \frac{E}{2(1+\mu)} \quad , \mu = 0.3$$

TEMP. (°F)	E (psi)	G (psi)	S _r (psi)	S _s (psi)
Room	26.4 x 10 ⁶	10.2 x 10 ⁶	121,000	60,500
1000	29.6 x 10 ⁶	11.4 x 10 ⁶	101,000	50,500
1500	21.8 x 10 ⁶	8.4 x 10 ⁶	—	—
1600	—	—	59,000	28,500
1800	—	—	38,000	19,000

At Room Temp, $T = 2154 \text{ IN-LB}$, $r_o = 0$, $S_s = 60,500 \text{ psi}$, $l = 10 \text{ in}$

$$r_i^3 = \frac{(2)(2154)}{(60,500) \pi} = .0227$$

$$r_i = 0.283 \quad , \quad d_i = 0.566 \text{ in}$$

$$\theta = \frac{360 (2154) (10)}{(\pi)^2 (10.2 \times 10^6) (.283)^4} = 12^\circ$$

Using $r_i = 0.5''$, $r_2 = 13/32$ AS USED ON DWG# EX-N6X626115

$$\theta = \frac{360 (2154) (10)}{\pi^2 (10.2 \times 10^6) (.5)^4 - (.4062)^4} = \frac{(3.60) (2.154) (10^6)}{\pi^2 (1.02) (3.53) (10^5)}$$

$$\theta = 2.18^\circ$$

$$S_s = \frac{2 (2154) (.5)}{\pi (.0353)} = 21,540 \text{ psi}$$

IF THIS SHAFT WERE SUBJECTED TO 200 IN-OZ @ 1200°F

$$G (\text{INTERPOLATED}) = 10.2 \times 10^6 \text{ psi}$$

$$\theta = \frac{(360) \left(\frac{200}{16}\right) 10}{\pi^2 (10.2) (10^6) (.0353)} = \frac{4.50}{(\pi^2) (10.2) (3.53)} = .01265^\circ$$

$$\theta = 0^\circ 0' 46''$$

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At A 1.5" RAD FROM THE CENTER, THE SHEAR FORCE IS:

$$P = \frac{2154}{1.5} = 1436 \text{ LB.}$$

THE AREA OF A 1/4 DIA. PIN IS:

$$A = \frac{\pi}{4} \left(\frac{1}{4}\right)^2 = .049$$

THE SHEAR STRESS ON ONE PIN IS:

$$S_s = \frac{P}{A} = \frac{1436}{.049} = 27,300 \text{ psi}$$

BY USING 2 PINS,

$$S_{s4} = 13,650 \text{ psi}$$

USING 2 3/16 DIA. PINS

$$S_s = 25,800 \text{ psi}$$

SHEAR IN BELLOWS SUPPORTING MEMBERS

(SHADOW SHELL)
.030 THICK

$$\text{Max } S_s = \frac{2T r_i}{\pi (r_i^4 - r_o^4)}$$

$$r_i = 1 \frac{23}{32} = \frac{55}{32}$$

$$r_o = 1 \frac{22}{32} = \frac{54}{32}$$

$$S_s = \frac{2(2154) \left(\frac{55}{32}\right)}{\pi \left[\frac{55^4 - 54^4}{(32)^4} \right]} = 3490 \text{ PSI}$$

A
↑

(DOWEL PINS TRANSMITTING TORQUE FROM BELLOWS TO GEAR.)

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SHEAR ON DOWEL PINS:

$$S_s = \frac{T}{D \left(\frac{\pi d^2}{4} \right)} = \frac{4T}{(3.062) \pi \left(\frac{3}{16} \right)^2} = 11.8 T$$

$$S_s \text{ ULT} = 25,500 \text{ psi}$$

$$S_s \text{ LIMIT} = 16,950 \text{ psi}$$

MAT'L. OF DOWELS = A296

$$\left. \begin{array}{l} \sigma_u = 146,000 \text{ psi} ; \gamma_u = 95,000 \text{ psi} \\ \sigma_y = 100,000 \text{ psi} ; \gamma_y = 65,000 \text{ psi} \end{array} \right\} \text{Room TEMP.}$$

$$M.S. (\text{LIMIT}) = \frac{65,000}{16,950} - 1.0 = 2.83$$

$$M.S. (\text{ULT}) = \frac{95,000}{25,500} - 1.0 = 2.73$$

$$M.S. = 2.73 \text{ or } 273\%$$

DIFFERENTIAL THERMAL EXPANSION AT GEAR INTERFACE:

CHOOSE A TEMP OF 1000°F

$$\text{REJEX 41 : } \alpha = 7.5 \times 10^{-6} \text{ IN/IN-}^\circ\text{F}$$

$$\text{INCONEL 718 : } \alpha = 8.2 \times 10^{-6} \text{ IN/IN-}^\circ\text{F}$$

$$\Delta T = 930^\circ\text{F}$$

$$\Delta \alpha = 0.7 \times 10^{-6} \text{ IN/IN-}^\circ\text{F}$$

$$\frac{\Delta L}{L} = 6.51 \times 10^{-6} \text{ IN/IN}$$

$$L = 3.062$$

$$\Delta L = .0021 \text{ IN ACROSS PINS}$$

$$E = 23.5 \times 10^6 \text{ psi @ } 1000^\circ\text{F}$$

ASSUME $\mu = 0.3$

$$G = \frac{E}{2(1+\mu)} = \frac{23.5 \times 10^6}{2.6} = 9.03 \times 10^6 \text{ psi}$$

$$\gamma = G \epsilon \quad \epsilon = \frac{.00105}{.1875} = .0056 \text{ IN/IN}$$

$$\gamma = (9.03)(5.6)(10^{-3})(10^5) = 50,568 \text{ PSI}$$

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@ 1000°F A-286 (INTERPOLATED) STRENGTHS ARE:

$$\sigma_u = 112,000 \text{ psi} \quad ; \quad \gamma_u = 72,800 \text{ psi}$$

$$\sigma_y = 90,000 \text{ psi} \quad ; \quad \gamma_y = 58,500 \text{ psi}$$

$$M.S. \text{ TENSILE} = \frac{72,800}{50,568} - 1.0 = 0.44 = 44\%$$

$$M.S. \text{ YIELD} = \frac{58,500}{50,568} - 1.0 = 0.116 = 11.6\%$$

THIS ASSUMES THAT ALL DEFORMATION IS TAKEN AS SHEAR BY THE DOWEL PINS.

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SQUEEZE IN .003" ON SERRATIONS

$$\begin{array}{l}
 \text{PITCH DIA } .5938 \\
 \text{ROOT DIA } .6375
 \end{array}
 \left. \begin{array}{l}
 \text{USE: AVG. FOR I.D.} = .6156 \\
 \text{O.D.} = .750
 \end{array} \right\}
 \begin{array}{l}
 R = 3/8 \\
 t = .0672
 \end{array}$$

$$A = 2\alpha R t$$

$$I_1 = R^3 t \left(\alpha + \sin \alpha \cos \alpha - 2 \frac{\sin^3 \alpha}{\alpha} \right)$$

$$y_1 = R \left(1 - \frac{\sin \alpha}{\alpha} \right)$$

$$y_2 = R \left(\frac{\sin \alpha}{\alpha} - \cos \alpha \right)$$

$$\alpha = 45^\circ = \pi/4 \text{ RADIANS}$$

$$A = 2 \left(\pi/4 \right) \left(3/8 \right) \left(.0672 \right) = .03958 \text{ in}^2$$

$$I_1 = \frac{27(.0672)}{512} \left(\pi/4 + \frac{1}{\sqrt{2}} \frac{1}{\sqrt{2}} - \frac{2 \cdot \frac{1}{2}}{\pi/4} \right)$$

$$= \frac{27(.0672)(.0121597)}{512}$$

$$= .00004309 \text{ in}^4$$

$$y_2 = \left(\frac{3}{8} \right) \left(\frac{1/\sqrt{2}}{\pi/4} - 1/\sqrt{2} \right) = \frac{3}{8} \left(\frac{1}{\sqrt{2}} \right) \left(\frac{4}{\pi} - 1 \right) = 0.07244$$

$$y_1 = \left(\frac{3}{8} \right) \left(1 - \frac{1/\sqrt{2}}{\pi/4} \right) = \frac{3}{8} \left(1 - \left[\frac{1}{\sqrt{2}} \right] \left[\frac{4}{\pi} \right] \right) =$$

$$y_1 = 0.03743"$$

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$$MAX M = -W a @ C$$

$$y = -\frac{1}{6} \frac{W}{EI} (2a^3) @ B$$

$$y_{MAX} = -\frac{1}{6} \frac{W}{EI} (3a^2 l - a^3)$$

l = LENGTH OF SAW CUTS

a = DIST. - BOTT. OF SAW CUTS TO BOTT. OF CLAMP.

$$y_{MAX} = .003", l = 17\frac{1}{32}", a = 7\frac{1}{32}", E = 30 \times 10^6 \text{ PSI}$$

$$.003 = -\frac{1}{6} \frac{W}{(30)(106)(4.309)(10^{-5})} [3(7\frac{1}{32})^2 17\frac{1}{32} - (7\frac{1}{32})^3]$$

$$.003 = -\frac{W [2156]}{(6)(30)(106)(4.309)(10^{-5})(32)^3}$$

$$W = -\frac{(.003)(6)(10)(4.309)(30)(32)^3}{2156}$$

$$= -\frac{(.18)(4.309)(32)^3(30)}{2156}$$

$$W = -354 \text{ LBS.}$$

$$y_B = -\frac{1}{6} \frac{(354)(2)(7\frac{1}{32})^3}{(30)(106)(4.309 \times 10^{-5})}$$

$$= -\frac{(354)(343)}{(3)(30)(10)(4.309)(32)^3}$$

$$= -.00093 \text{ in.}$$

$$S_c = \frac{My_c}{I} = -\frac{59(7\frac{1}{32})(.07244)}{(4.309)(10^{-5})}$$

$$S_c = 130000 \text{ psi}$$

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$$\begin{aligned}
 P &= Q \left(\frac{l}{L} \right) \\
 Q &= W \sqrt{2} \\
 &= (11.78)(\sqrt{2}) \\
 &= 16.65 \# \\
 l &= 7/16", L = 1" \\
 P &= 7/16 Q \\
 P &= 7.28 \#
 \end{aligned}$$

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FROM MARK'S P. 5-59

$$\delta_x = + \frac{FR^3}{4EI}$$

$$I = \frac{bh^3}{12} = \frac{(\frac{1}{4})(\frac{1}{16})^3}{12}$$

$$E = 21 \times 10^6$$

$$R = \frac{27}{64}$$

$$b = \frac{1}{4}$$

$$h = \frac{1}{16}$$

$$R = \frac{25}{64} + \frac{1}{32} = \frac{27}{64}$$

$$\begin{aligned} \frac{\delta}{F} &= \frac{(\frac{27}{64})^3 12}{4(21)(10^6)(\frac{1}{4})(\frac{1}{16})^3} \\ &= \frac{(27)^3 (12) (10)^3}{(21)(10^6) (64)^3} \\ &= \frac{(27)^3 (12) (\frac{1}{4})^3 2^2}{7(10^6)} = \frac{(27)^3}{112(10^6)} \end{aligned}$$

$$\frac{1}{112} = \frac{(2.7)^3 (10^3)}{1.12 (10^9)} = 1.625 \times 10^{-4}$$

$$F = \frac{\delta (10^4)}{1.625} = \frac{(0.03) (10^4)}{1.625} = 0.00185 \times 10^4$$

$$F = 18.5 \#$$

$$S = \frac{Mc}{I} = \frac{(6)(18.5)(\frac{27}{64})}{\frac{1}{4}(\frac{1}{16})^2} = \frac{(4)(6)(18.5)(27)2^2}{64}$$

$$S = (96)(18.5)(27) = 47,952 \text{ PSI}$$

(68,500 FOR INCONEL)

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REFERRING TO P.1

$$\left. \begin{array}{l} \text{I.D. (AUG)} = .616 \\ \text{O.D.} = .781 \end{array} \right\} \begin{array}{l} R = \frac{25}{64} \\ t = .082 \end{array}$$

$$\begin{aligned} I_1 &= R^3 t \left(\frac{\pi}{4} + \frac{1}{2} - \frac{4}{\pi} \right) \\ &= \left(\frac{25}{64} \right)^3 (.082) (.0121587) \\ &= \frac{(1.5625) (.082) (.0121587)}{262,144} \end{aligned}$$

$$I_1 = 5.942 \times 10^{-5} \quad \text{IN.}^4$$

$$\begin{aligned} y_2 &= \left(\frac{25}{64} \right) \left(\frac{1}{\sqrt{2}} \right) (.2732325) \\ &= \frac{(25) (.2732325)}{(64) (1.414214)} \end{aligned}$$

$$y_2 = .07545 \quad \text{IN.}$$

$$y_{\max} = -\frac{1}{6} \frac{W}{EI} (3a^2l - a^3)$$

UPPER CLAMP

$$y_{\max} = .003; E = 30 \times 10^6 \text{ PSI}, a = \frac{1}{4}, l = \frac{3}{4}$$

$$.003 = -\frac{(1/4) W [3(\frac{1}{4})^2 \frac{3}{4} - (\frac{1}{4})^3]}{6 (30 \times 10^6) (5.942) (10^{-5})}$$

$$.003 = -\frac{W [8]}{(6) (300) (5.942) (4)^{3/2}}$$

$$.003 = -\frac{W}{3(300)(5.942) 16}$$

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$$W = (.003) (3) (3.3) (5.942) (16) = 180 \#$$

$$W = 256 \#, Q = 363 \#$$

$$\text{AREA OF CLAMP SPRING SECTION} = \frac{1}{16} \times \frac{1}{4} = \frac{1}{64}$$

$$\text{LOWER CLAMP: } a = \frac{7}{32}, l = \frac{9}{16}, y_{\max} = .003, E = 21 \times 10^6$$

$$.003 = \frac{W [3(\frac{7}{32})^2 (\frac{9}{16}) - (\frac{7}{32})^3]}{(6)(21)(10^6)(5.942)(10^{-5})}$$

$$.003 = \frac{W [(6)(7)^2 (9) - (7)^3]}{(6)(21)(5.942)(10) (32)^3}$$

$$.003 = \frac{W (2303)}{(6)(21)(5.942)(10)(32)^3}$$

$$W = \frac{(.003) (6) (21) (5.942) (10) (32,768)}{2303}$$

$$W = \frac{(19)(2.1)(5.942)(32,768)}{2303}$$

$$W = 320 \#, Q = 452 \#$$

FORCES IN CLAMP:

$$F + P = Q$$

$$FR = dP$$

$$P = \frac{R}{d} F$$

$$(1 + \frac{R}{d}) F = Q$$

$$F = \frac{Q}{1 + \frac{R}{d}}$$

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$$\left. \begin{array}{l} R = \frac{27}{64} \\ d = \frac{19}{32} \end{array} \right\} \frac{R}{d} = \frac{\frac{27}{64}}{\frac{19}{32}} = \frac{27}{38}$$

$$\left(1 + \frac{R}{d}\right) = 1.71$$

$$F = \frac{Q}{1.71}$$

FOR UPPER CLAMPING, $Q = 363\#$

$$F = 212\#$$

FOR LOWER CLAMPING, $Q = 452\#$

$$F = 264\#$$

AREA OF THIN SECTION = $\frac{1}{16} \times \frac{1}{4} = \frac{1}{64}$

STRESS AT THIN AREA DUE TO TENSION

FOR UPPER CLAMPING

$$S_T = \frac{F}{A} = (212)(64) = 13,600 \text{ PSI}$$

FOR LOWER CLAMPING

$$S_T = \frac{F}{A} = (264)(64) = 16,900 \text{ PSI}$$

TOTAL TENSILE STRESS IN SECTION OF CLAMP

UPPER CLAMP

$$S_T = 68,500 + 13,600 = 82,100$$

LOWER CLAMP

$$S_T = 47,250 + 16,900 = 64,150$$

STRESS IN SCREWS AT EDGE OF SPACERS

$$S = \frac{M y_2}{I} = \frac{11.25 \text{ IN}^2}{1.25 \text{ IN}} = 11.25 \times 10^3$$

$$S = 900(1.25 \text{ IN})$$

UPPER

$$S = (1/4)(900)(1.25 \text{ IN}) = 281 \text{ PSI}$$

LOWER

$$S = (7/8)(900)(1.25 \text{ IN}) = 806 \text{ PSI}$$

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FORCE ON BOLT:

$$P = \frac{R}{d} F = 1.71 F$$

UPPER:

$$P = \frac{27}{38} (212) = 151 \#$$

LOWER:

$$P = \frac{27}{38} (264) = 188 \#$$

FORCE FROM CLAMP BENDING

$$\text{UPPER} = 27.45 \#$$

$$\text{LOWER} = 18.5 \#$$

TOTAL AXIAL FORCE ON BOLT

$$\text{UPPER} = 178 \#$$

$$\text{LOWER} = 206 \#$$

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DATE:	WEIGHT OF LOWER BELLows ASSY DRIVE TRAIN COUPLING	REPORT NO. MODEL NO.

SHAFT : 1" O.D. SOLID. ~11.5 LONG. 1 PIECE
 PINS : ~.4 O.D SOLID 1.5 LONG 2 PIECES
 END PLATE : 4. O.D x .125 THICK 3 "

$$DENSITY = .296 \text{ LB/IN}^3$$

NOTE: NEGLECTED CONVOLUTIONS WEIGHT BUT
 COMPENSATED @ THE END PLATES & SHAFT.

$$W = .296 \left(\frac{\pi}{4} 1^2 \times 11.5 + 2 \frac{\pi}{4} .4^2 \times 1.5 + \frac{\pi}{4} 4^2 \times .125 \times 3 \right)$$

$$W = .296 \frac{\pi}{4} \left(11.5 + 3 \times .16 + 16 \frac{3}{8} \right)$$

$$W = .296 \frac{\pi}{4} (11.5 + .48 + 6)$$

$$W = .296 \frac{\pi}{4} \times 18$$

$W = 4.19 \text{ LB}$ WEIGHT OF DRIVE COUPLING
 BELLows SUSPENDED OF
 THE GRAPHITE BALL.

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ITEM #1 - BACKLASH

FROM FIGURES DEVELOPED IN DESIGN

ITEM #2 - WEIGHT

PIN IN SLOT - SINGLE ARM

BELLOWS - THIN MEMBERS

UNIVERSAL - HEAVY RING AND DUAL ARMS

ITEM #3 - COST

PIN IN SLOT - FEW PIECES

UNIVERSAL - ALL MACHINED

BELLOWS - COST OF BELLOWS ASS'Y. + MACHINING

ITEM #4 - RELIABILITY

BELLOWS - NO. LUBED SURFACES (SHADOW)

BALL JOINT (4π)

UNIVERSAL - LUBED SURFACES ENCLOSED MORE
NO BALL JOINTS

MANY PARTS

PIN IN SLOT - EXPOSED LUBRICATED SURFACES
DUAL BALL JOINTS (4π)

ITEM #5 - ACCESS TO DRUM

PIN IN SLOT - SINGLE ARM

UNIVERSAL - TWO ARMS

BELLOWS - TWO ARMS IN 4π

BELLOWS COVERS DRUM IN SHADOW SHIELD

ITEM #6 - EASE OF ASS'Y. TO REACTOR

PIN IN SLOT - DROP-IN ASS'Y (SHADOW)

WON'T CONFORM TO BOTH CONFIGURATIONS

UNIVERSAL - DROP-IN ASS'Y.

BELLOWS - SCREWED-IN ASS'Y (SHADOW)

DROP-IN ASS'Y. (4π)

ITEM #7 - MATERIAL COMPATABILITY

UNIVERSAL - MAT'L. SAME AS GEAR

PIN IN SLOT - MAT'L. SAME AS GEAR (SHADOW)

DIFFERENTIAL EXPANSION BETWEEN CARBON & METAL
IN TWO PLACES (4π)

BELLOWS - BELLOWS MAT'L. DIFFERENT FROM GEAR

ITEM #8 - EST. FRICTION

BELLOWS - SIDE LOADS & THRUST LOADS ON BAGS.

PIN IN SLOT - 1 CONTACT POINT (SHADOW)

SLIDING CONTACT (4π)

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ITEM #8 - CONT'D.

UNIVERSAL - SLIDING ON 4 SURFACES (SHADOW)
 ROTATING CONTACT ONLY (4T)

APPENDIX C

GEAR, LOCKING MECHANISM, ROTOR SHAFT

BEARING ANALYSIS DESIGN

MEAN COEFFICIENT OF THERMAL EXPANSION FOR VARIOUS CANDIDATE MATERIALS

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GEAR BEARING & LOCK. PIN. LOAD.

$$T = R_p a$$

$$R_{ig} = R_p$$

$$R_p = \frac{T}{a} = \frac{2154}{1.52}$$

ULTIMATE

$$R_p = 1417 \text{ LB} \quad \text{LOAD ON THE PIN}$$

$$R_{ig} = 1417 \text{ LB} \quad \text{LOAD ON THE BEARING}$$

LIMIT

$$R_p = \frac{1435}{1.52} = 944 \text{ LB @ THE PIN}$$

$$R_{ig} = 944 \text{ LB @ THE BEG.}$$

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FREE BODY - LOCKING MECH.

$$a = 1.520$$

$$b = .50$$

$$c = 3.375$$

R_G = REACTION AT THE GEAR BEG
 R_R = REACTION @ THE REAR LWR BEG
 R_T = REACTION @ THE TOP UP. BEG.
 R_P = REACTION @ THE PIN

11.11.11

$$\text{TORQUE } T_U = 85 \text{ LB} \times .704'' \times 365's = 2154 \text{ IN-LB ULTIMATE}$$

$$T_L = 85 \text{ LB} \times .704 \text{ IN} \times 249's = 1435 \text{ IN-LB LIMIT}$$

BENDING STRESS IN THE SHAFT. (GEAR)

$$r = .020 \text{ MAX } .010 \text{ MIN}$$

$$\frac{f}{d} = \frac{.020}{.5} = .04 \quad \text{therefore}$$

$$\frac{D}{d} = \frac{.3}{.5} = 6 \quad K = 2.6$$

$$\sigma = \frac{MCK}{I} \quad M = R_{IG} \times X = .18R' + .56R'$$

$$C = \frac{d}{2} \quad X = \frac{.74R'}{R_{IG}}$$

$$I = \frac{\pi}{64} (d^4 - h^4) \quad R_{IG} = 2R'$$

$$I = \frac{\pi}{64} \times .0609 \quad X = \frac{.74}{2}$$

$$\sigma = \frac{R_{IG} \times .74 \times .25 \times 64}{2 \times \pi \times .0609} K = 31 \times R_{IG} \times K$$

$$\sigma = 31 \times 1417 \times 2.6 = 114,000 \text{ PSI (ULT)}$$

$$\sigma = 31 \times 944 \times 2.6 = 76,000 \text{ PSI (LIMIT)}$$

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TORQUE = 240 IN-LB = 15 IN-LB

DIAMETRAL PITCH	32	36	40	48
NO. TEETH PINION	15	17	19	23 (15.652174)
NO. TEETH (GEAR)	90	102	114	138 (2.608615)
PITCH DIA (PINION)	.46875	.4722	.475	.4792 (d)
PITCH DIA (GEAR)	2.8125	2.8333	2.850	2.8750
φ T & DIST.	1.6406	1.6527	1.6625	1.6771
PINION O.D.	.5311	.529	.525	.521
GEAR O.D.	2.875	2.890	2.900	2.917
"Y" FACTOR (14½°)	.078	.084	.088	.094 (Y)
"Y" FACTOR (20°)	.092	.096	.100	.106 (Y)
CIRC. PITCH	.0982	.0873	.0785	.0654 (P)
σ × W (14½°)	8340	8670	9130	10200
σ × W (20°)	7060	7580	8040	9030

$$\sigma = \frac{2T}{d \cdot y \cdot P \cdot W}$$

$$\sigma = \frac{2 \times 240 \text{ IN-LB}}{d \cdot y \cdot P \cdot W} \cdot \frac{1 \text{ LB}}{1692} = \frac{30}{d \cdot y \cdot P \cdot W}$$

$$\sigma = \frac{30}{d \cdot y \cdot P \cdot W}$$

W = GEAR FACE WIDTH

GEAR FACE WIDTH Vs STRESS

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FOR VARIOUS PITCHES

22.5 - 25.0 - 27.5 - 30.0 - 32.5 - 35.0 - 37.5 - 40.0 - 42.5 - 45.0 - 47.5 - 50.0 - 52.5 - 55.0 - 57.5 - 60.0 - 62.5 - 65.0 - 67.5 - 70.0 - 72.5 - 75.0 - 77.5 - 80.0 - 82.5 - 85.0 - 87.5 - 90.0 - 92.5 - 95.0 - 97.5 - 100.0

S - STRESS - PSI

W - GEAR FACE WIDTH - IN

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PIN SUPPORTS GEAR & COUPLING ASSY. THE MAX LOADS OCCUR DURING LAUNCH IN THE AXIAL DIRECTION OF THE PIN.

ASSUME WEIGHT OF GEAR, COUPLING & OTHERS = $W = 10 \text{ LB}$ (Actually will be $\sim 8 \text{ LB}$)

ALSO ASSUME NATURAL FREQUENCY OF SUSPENDED ASSY IS WHERE PLAK MAX. G'S WILL BE EXPERIENCED

$$R = W \times G_{\text{MAX}}$$

$G_{\text{MAX}} = 40.5 \text{ ULTIMATE}$
& 27 LIMIT.

$$\sigma_t = \frac{W \times G_{\text{MAX}}}{\frac{\pi d_r^2}{4}} \times K$$

K - FOR ULTIMATE LOAD

$$r = .10$$

$$d = .19$$

$$D = .66$$

$$\frac{h}{r} = \frac{D-d}{2r} = \frac{.66-.19}{.2} = 2.35$$

$$\frac{h}{d} = \frac{.1}{.19} = .528$$

$$K = \sim 1.4$$

K - FOR LIMIT LOAD & FATIGUE COMPARISON

$$K \approx 2 \text{ (THIS IS AN ESTIMATE)}$$

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PIN MATERIAL IS RENE 41
 note Root dia $d_r = .147$

$$\sigma_t = \frac{10 \times 40.5}{\frac{\pi \times .147^2}{4}} \times 1.4$$

$$\sigma_t = 33400 \text{ PSI (ULTIMATE LOAD)}$$

LIMIT LOAD.

$$\sigma_t = \frac{10 \times 36}{\frac{\pi \times .147^2}{4}} \times 2$$

$$\sigma_t = 42400 \text{ PSI (LIMIT LOAD)}$$

MARGIN OF SAFETY.

ALLOWABLE STRESS

a. ULTIMATE 160000 PSI

b. FATIGUE 85000 PSI

c. YIELD 120000 PSI

$$\text{ULTIMATE M.S.} = \frac{160000}{33400} - 1 = +3.8$$

$$\text{FATIGUE M.S.} = \frac{85000}{42400} - 1 = +1.0$$

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COMPRESSIVE STRESS ON THE GEAR BEARING
DUE TO THRUST LOAD.

THRUST Load $L_T = WG$

AREA = $\pi(D_m)t$.

$D_m = .500 + \frac{.656 - .500}{2}$

$t = \frac{.656 - .500}{2} - 2(.020)$

AREA = $\frac{\pi}{2} [(1 + .156)(.156 - .08)]$

AREA = $\frac{\pi}{2} 1.156 \times .076$

$S_C = \frac{L_T}{A_{RE}} = \frac{WG}{A_{RE}}$

$S_C = \frac{10 \times 40.5}{\frac{\pi}{2} \times 1.156 \times .076}$

$S_C = 2900 \text{ PSI OK.}$

I.D. = .500

O.D. = .656

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BENDING STRESS ON THE LOCK PIN

FROM MECHANICAL ENG. DESIGN
(SHIGLEY.)

$R_p = 1417 \text{ LB}$ ULTIM.
944 LB LIMIT

$$r/h = \frac{.06}{.28} = .214$$

$d = .22 \text{ IN.}$

$$D/h = > 3$$

$h = ?$

let $b = L$

$r = .06$

$$\sigma = \frac{Mc}{I} = R_p \times d \times \frac{L}{2h^4} \quad \left. \begin{array}{l} \text{THEREFORE:} \\ K \approx 1.5 \end{array} \right\}$$

$$\sigma = \frac{6 R_p d}{h^3} \times K$$

$$h = \sqrt[3]{\frac{6 R_p d K}{\sigma}}$$

MAT'L HASTELLOY B.

σ_u ULTIM

σ_y YIELD σ_f

127,000

56,000

60,000

LENE 41

160,000

120,000

85,000

INCONEL 718

180,000

150,000

90,000

- h - IF HASTELLOY B IS USED

$$\textcircled{1} \quad h \geq \sqrt[3]{\frac{6 \times 1417 \times .22}{127,000} \times 1.5} = .280 \text{ IN}$$

FOR ULTIMATE LOAD
& ALLOWABLE ULTIM

$$\textcircled{2} \quad h \geq \sqrt[3]{\frac{6 \times 944 \times .22}{56,000} \times 1.5} = .322 \text{ IN}$$

FOR LIMIT LOAD
& YIELD ALLOWABLE

$$\textcircled{3} \quad h = \sqrt[3]{\frac{6 \times 944 \times .22}{60,000} \times 1.5} = .314 \text{ IN}$$

FOR LIMIT LOAD
VS FATIGUE STRENGTH

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h - IF RENE 41 IS USED

① $h \geq \sqrt[3]{\frac{6 \times 1,417 \times .22}{160,000}} \times 1.5 \geq .259 \text{ IN}$ FOR ULTIMATE LOAD AND ULTIMATE ALLOW.

② $h \geq \sqrt[3]{\frac{6 \times 7,44 \times .22}{120,000}} \times 1.5 \geq .250 \text{ IN}$ FOR LIMIT LOAD AND YIELD ALLOW.

③ $h \geq \sqrt[3]{\frac{6 \times 940 \times .22}{85,000}} \times 1.5 \geq .280 \text{ IN}$ FOR LIMIT LOAD VS. FATIGUE STRENGTH.

INCONEL 718 IS NOT CONSIDERED BECAUSE OF ITS UNDESIRABLE HIGH COEFFICIENT OF THERMAL EXPANSION (RELATIVE TO RENE 41).

TO LIMIT THE THERMAL EXPANSION PROBLEMS IT IS DESIRABLE TO USE A LIKE MATERIAL WHEREVER POSSIBLE. SINCE THE ACTUATOR SHAFT AND END BELL IS BEST BALANCED IF HASTELLOY B IS USED, THEN THE GEAR, PIN AND CAM SHOULD ALSO USE HASTELLOY B.

THE PIN h DIM'S $\geq .322 \text{ IN}$ FOR HAST. B
 $\geq .280 \text{ IN}$ FOR RENE 41

THE SPACE AVAILABLE IS VERY LIMITED AND ~~THE~~ THE SMALLER PIN DIMENSION IS MORE ATTRACTIVE. THEREFORE DESPITE THE ADDITIONAL PROBLEM WITH THERMAL EXPANSION THE PIN & THEREFORE THE GEAR SHOULD BE MFD OUT OF RENE 41. ALSO RENE 41 HAS GREATER HARDNESS TO RESIST DAMAGE AT THE CONTACT SURFACES.
MAKE PIN $\therefore h = .286$ $b = .282$

THEN:

$\sigma_1 = 122,000 \text{ PSI}$ ULTIMATE LOAD STRESS

$\sigma_2 = 81,000 \text{ PSI}$ LIMIT LOAD STRESS

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CHECK CONTACT STRESSES @ CAM SURFACE

$$\text{MAX } \sigma_c = .591 \sqrt{\frac{PE}{D}}$$

$$P = \frac{R_p}{W} = \frac{R_p}{.344} \quad (W = \text{THICKNESS OF THINNER MEMBER})$$

$$E = 30 \times 10^6 \text{ psi}$$

$$D = 2r = 2 \times .566$$

$$\sigma_c = .591 \sqrt{\frac{1 R_p \times 30 \times 10^6}{.344 \times 2 \times .566}} = 5.2 \times 10^3 P^{1/2}$$

$$\begin{aligned} \text{ULTIMATE } \sigma_c &= 5.2 \times 10^3 (1417)^{1/2} = 195000 \text{ psi} \\ \text{LIMIT } \sigma_c &= 5.2 \times 10^3 (944)^{1/2} = 160000 \text{ psi} \end{aligned} \quad \left. \vphantom{\begin{aligned} \text{ULTIMATE } \sigma_c &= 5.2 \times 10^3 (1417)^{1/2} = 195000 \text{ psi} \\ \text{LIMIT } \sigma_c &= 5.2 \times 10^3 (944)^{1/2} = 160000 \text{ psi} \end{aligned}} \right\} \text{CONTACT STRESS}$$

BENDING STRESS ON THE CAM

$$\sigma = \frac{MC}{I}$$

$$I = \frac{.344 (1.188 - .854)^3}{12}$$

$$C = \frac{1.188 - .854}{2}$$

$$\sigma = \frac{R_p \times .4 \times .334 \times 12}{.344 \times .334^3 \times 2}$$

$$\sigma = 1417 \times 62.5 = 88500 \text{ psi ULTIM. STRESS}$$

$$\sigma = 944 \times 62.5 = 59000 \text{ psi LIMIT & FATIGUE STRESS}$$

Ratio 1.3
Therefore multiply by 1.3

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$R_p = 1417 \text{ LB ULTIMATE}$

$R_p = 944 \text{ LB LIMIT}$

deflection $y = \frac{1}{2} (D_{SHAFT} - D_{HOLE})$

$D_{HOLE} = .5625 \pm .0005$

$D_{SHAFT} = .562 \pm .001$

then $y = \frac{1}{2} (.002) = .001$

(NOTE $D_{PINION} = .521 \pm .002$)

$y = \frac{F d^3}{3 E t}$

$\sigma = \frac{M c}{I}$

$M = F d$

$F = \frac{\sigma I}{c d}$

$y = \frac{\sigma I d^3}{3 E t c d} = \frac{\sigma d^2}{3 E c}$

$c = \frac{t}{2}$

$y = \frac{2 \sigma d^2}{3 E t}$

$\sigma = \frac{3 E t y}{2 d^2}$

$\sigma = \frac{3 \times 30 \times 10^6 \times .188 \times .001}{2 \times .52}$

$d = .50 \text{ IN}$

$E = 30 \times 10^6$

$t = .125 \text{ TO } .188$

$w = .344 \text{ IN}$

$\sigma_{max} = \frac{6 \times 30 \times 3}{16} \times 10^3$

$\sigma_{MAX} = 33800 \text{ PSI @ MIN SECTION DUE BENDING TABS}$

FORCE (F) TO BEND TABS

$F = \frac{33800 \times .344 \times .188^2 \times 2}{1 \times .188 \times .52 \times 3} = \frac{33800 \times .344 \times 9^3}{1.6^2 \times 3}$

$F_1 = 136 \text{ LB}$

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$$\sum F_H = 0$$

$$R_p = 2 R_1$$

$$R_1 = \frac{R_p}{2}$$

$$\sum F_V = 0$$

$$F_2 = R_2$$

$$R_1 = R \cos \theta$$

$$R_2 = R \sin \theta$$

$$\theta = 45^\circ$$

$$\therefore R_1 = R_2$$

$$R_1 = .707 R = \frac{R_p}{2}$$

$$R = \frac{R_p}{2 \times .707}$$

$$\therefore F_2 = \frac{R_p}{2 \times .707}$$

$$F_2 = \frac{1417}{1.414} = 1000 \text{ LB ULTIMATE LOAD ON BOLT}$$

$$F_2 = \frac{944}{1.414} = 666 \text{ LB LIMIT LOAD ON BOLT}$$

TOTAL MAX. LOAD ON CAN RETAIN. BOLT.

$$F_T = F_1 + F_2$$

$$F_T = 136 + 1000 = 1136 \text{ LB ULTIMATE}$$

$$F_T = 136 + 666 = 802 \text{ LB LIMIT (FATIGUE LOAD)}$$

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THE TOTAL STRESS @ t due to CLAMPING AND DRUM VIBR. LOADS.

MAX TENSILE STRESS (INNER FIBER @ .5625 DIA)

$$\sigma_t = \sigma_B + \sigma_v$$

$$\sigma_v = \frac{R_1}{A} = \frac{R_1}{wt} = \frac{R_p}{2wt}$$

$$\sigma_t = \sigma_B + \frac{R_p}{2wt}$$

$$\sigma_t = 33800 + \frac{1417}{2 \times .344 \times .125} = 33800 + 16500$$

$$\sigma_t = 50100 \text{ PSI ULTIMATE STRESS}$$

$$\sigma_t = 44800 \text{ PSI LIMIT STRESS}$$

RENC 41 MORE THEN ADEQUATE

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MARGIN OF SAFETIES (GEAR)

A. PIN

1. $M.S. = \frac{160000}{122,000} - 1 = +.3$ ULTIMATE LOADING
2. $M.S. = \frac{120000}{81,000} - 1 = +.48$ LIMIT LOADING
3. $M.S. = \frac{85000}{81000} - 1 = +.05$ FATIGUE CONSIDERATION

B. SHAFT

- 1 $MS = \frac{160000}{114000} - 1 = +.4$ ULTIMATE LOADING
- 2 $M.S. = \frac{120000}{76000} - 1 = +.58$ LIMIT LOADING
- 3 $M.S. = \frac{85000}{76000} - 1 = +.12$ FATIGUE CONSIDERATION

C. GEAR TEETH - @ 1200°F M.S.

$$M.S. = \frac{116000}{30000} - 1 = +2.86 \text{ GEAR TEETH}$$

$$M = \frac{38000}{30000} - 1 = +.27 \text{ PINION TEETH.}$$

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MAXIMUM CAPABILITY OF THE LOCKING PIN

$$\sigma_{ult} = \frac{M C}{I} = \frac{F d c}{I}$$

$$F = \frac{\sigma_{ult} \cdot I}{d c}$$

$$\sigma_{ult} = 206,000 \text{ psi}$$

$$F = \frac{206,000 \times .282 \times .286^2 \times 2}{.22 \times .286 \times 12}$$

$$F = \underline{3600 \text{ LB}}$$

$$T = F r = 3600 \times 1.52 \text{ in}$$

$$T = \underline{5460 \text{ IN-LB}}$$

$$F \cdot r = \underline{5460}$$

$$W \times G \times r = 5460$$

$$G = \frac{5460}{W r} = \frac{5460}{.7 \times 85}$$

$$G = 92 \text{ g's} \quad \text{ULTIMATE IMPACT CAPABILITY OF THE DRIVE TRAIN.}$$

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POSSIBLE TO CALCULATE AT THE TIME WITH AN
IN-ACCURACY AN EQUIVALENT REACTOR
DIT AT WHICH THE DRIVE TRAIN
PIN SNAPS AND THE DRUMS ARE FREE
FALL.

THE OVERALL SYSTEM HAS NOT BEEN
DESIGNED AND DUE TO THE LIMITED FUNDING
AT THIS TIME, FOR THIS EFFORT, THE FOLLOWING
EVALUATION IS MADE TO PROVIDE
FEEL OF WHAT THE FREE FALL HEIGHT
WOULD BE IN TERMS OF ENERGY ABSORBING
TIME. A PLOT OF THIS RELATIONSHIP IS
ALSO INCLUDED.

IT MAY BE THAT THE SIMPLEST APPROACH
FOR PROVIDING LOCKING CAPABILITY TO THE DRUMS,
IF SUCH AN ACCIDENT OCCURS, WILL BE THE
INCORPORATION OF A LOCKING PIN, WHICH
DIRECTLY LOCKS THE DRUMS TO THE SUPPORT
PLATE.

THIS PIN CAN BE MADE STRONG ENOUGH TO
PROVIDE LOCKING CAPABILITY FOR ANY CREDIBLE
ACCIDENT.

ALL THE PINS CAN REMAIN IN ASSEMBLY JUST
PRIOR TO LAUNCH AT WHICH TIME THE DRIVE
TRAIN ACTUATOR ASSY CAN BE INSTALLED.
WITH SUCH A DEVICE IT WILL BE IMPOSSIBLE TO
INADVERTENDLY FORGET THE LOCKING PIN, BECAUSE
THE INSTALLATION OF THE DRIVE TRAIN AND
ACTUATOR WILL NOT BE POSSIBLE.

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DETERMINE THE HEIGHT FROM WHICH THE DRUM ASSY CAN BE DROPPED BEFORE FAILING THE LOCKING PIN.

$$l = 1.52 \text{ IN}$$

$$d = .7 \text{ IN}$$

$$W = 85 \text{ LB}$$

FORCE AT IMPACT

$$F_i = Ma$$

$$a = \frac{2h}{t^2} \quad \& \quad M = \frac{W}{g}$$

TORQUE GENERATED AROUND DRUM &

$$T_D = F_i \times d$$

RESISTING TORQUE AT THE PIN

$$T_{PIN} = F_{PIN(MAX)} \times l$$

$$\text{but } T_D = T_{PIN}$$

$$\therefore F_i \times d = F_{PIN(MAX)} \times l$$

$$\frac{2W \cdot h}{g \cdot t^2} \times d = F_{PIN(MAX)} \times l$$

USING THE ULTIMATE STRENGTH OF THE PIN MATERIAL

$$F_{PIN MAX} = 3600 \text{ LB } \& \text{ FAIL.}$$

$$\text{THEN } h = \frac{F_{PIN MAX} \cdot l \cdot g \cdot t^2}{2Wd}$$

$$\text{AND } h = \frac{3600 \times 1.52 \times 32.2 \cdot t^2}{2 \times 85 \times .7}$$

$$h = 1.48 \times 10^3 t^2$$

SEE PLOT IN FIGURE

TIME ELAPSED IN PROCESSING THE FALLING
FROM CURRENT HEIGHT TO LOCKING RPK FAILURE
VS HEIGHT OF FALL

$$h = \frac{g t^2}{2}$$

$N = 12$ (to give 144 divisions on 12 ft.)
WILL MEASURE THE TIME

FALLING DISTANCE - FT

(C)

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BENDING LOADS & SHAFT BENDING STRESSES BASED ON REVISED GEOMETRY.

GIVEN:

$W_R = \sim 2.5 \text{ LB}$ ROTOR

$a = .5 \text{ IN}$

$W_B = .36 \text{ LB}$ DISC

$b = 2.12 \text{ IN}$

$W_S = 15 \text{ LB}$ SPRING ARM

$c = 1.25 \text{ IN}$

$P_{yu} = 1417 \text{ LB}$

$d = .62 \text{ IN}$

$P_{yL} = 944 \text{ LB}$

ACCELERATION

RADIAL = $36g$ ULTIM.
(GR) $24g$'s LIMIT

AXIAL $40.5g$'s ULTIM.
(GA) $27g$'s LIMIT

$$\sum F(+Y) = 0 \quad (P_y \text{ IN } +Y \text{ DIRECTION})$$

$$P_y - W_R - W_B + R_{2y} = R_{1y}$$

$$\sum M(R_1) = 0$$

$$a P_y = b W_R + (b+c) R_{2y} + (b+c+d) W_B$$

$$R_{2y} = \left\{ a P_y - GR [b W_R + (b+c+d) W_B] \right\} \frac{1}{b+c}$$

$$R_{2y} = \left\{ .5 P_y - GR (2.12 \times 2.5 + 4 \times .36) \right\} \frac{1}{3.37}$$

$$R_{2y} = .143 P_y - 2 GR$$

$$R_{2y} = .143 \times 1417 - 2 \times 36 = 138 \text{ LB ULTIMATE}$$

$$(.143 \times 944 - 2 \times 24) = 92 \text{ LB LIMIT}$$

$P_y = 0$

$R_{2y} = -72$

$R_{2y} = +8$

$$R_{1y} = 1417 + 2.5 \times 36 + .36 \times 36 + 138 = 1658 \text{ LB ULTIMATE}$$

$$(944 + 2.5 \times 24 + .36 \times 24 + 92) = 1105 \text{ LB LIMIT}$$

$R_{1y} = 31$

$R_{1y} = 2$

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$$\Sigma P_y = 0 \quad (P_y \text{ IN } +Y \text{ DIRECTION})$$

$$R_{1y} + W_R G_R + W_B G_R = P_y + R_{2y}$$

$$\Sigma M_R = 0$$

$$a P_y + b W_R G_R + (b+c+d) W_B G_R = (b+c) R_{1y}$$

$$R_{1y} = .148 P_y + 2 G_R$$

$$R_{1y} = .148 \times 1417 + 2 \times 36 = \underline{282} \text{ LB ULTIMATE}$$

$$(.148 \times 944 + 2 \times 24) = \underline{188} \text{ LB LIMIT}$$

$$R_{1y} = 1417 + 282 - 36(2.5 + 36) = 1599 \text{ LB}$$

$$[944 + 188 - 24(2.56)] = 1052 \text{ LB.}$$

THE MAXIMUM LOADS:

$$1. \text{ RADIAL BEARING: } = 1658 \text{ LB ULTIMATE} \left. \vphantom{\begin{array}{l} 1658 \\ 1105 \end{array}} \right\} \text{ RADIAL}$$

$$= 1105 \text{ LB LIMIT}$$

2. THRUST BEARING:

$$a. \text{ RADIAL } = 282 \text{ LB ULTIMATE}$$

$$= 188 \text{ LB LIMIT}$$

$$b. \text{ AXIAL } = 131 \text{ LB ULTIMATE}$$

$$= 192 \text{ LB LIMIT.}$$

BENDING STRESS ON THE SHAFT

$$\sigma = \frac{M c}{I}$$

$$c = \frac{d}{2}$$

$$I = \frac{\pi d^4}{64}$$

$$\sigma = \frac{32 P_y a}{\pi d^3}$$

$$M = P_y \times a$$

$$d = \sqrt[3]{\frac{32 P_y a}{\sigma}}$$

$$a = .5 \text{ IN}$$

$$b = 2.12 \text{ IN}$$

$$c = 1.25 \text{ IN}$$

$$d = .62 \text{ IN}$$

$$W_R = 2.5 \times 36 \text{ g's} = 90 \text{ LB}$$

$$W_d = .36 \times 36 \text{ g's} = 13 \text{ LB}$$

$$P = 1417 \text{ LB ULTIM.}$$

$$R_1 = 1658 \text{ LB ULTIM.}$$

$$R_2 = 138 \text{ LB}$$

ACT'R SHAFT LOADS DURING LAUNCH
(ULTIMATE)

STRESS IN ROTARY SHAFT
(FOR BENDING MOMENT LOADS)

$\frac{1}{32}$ P.C.K.
N.T.

P. 1417 W. DISTANCE

2.5 IN

5.5 STRESS CONCENTRATION FACTOR

W. DISTANCE W. DISTANCE

2.5

1.8

1.6

1.4

1.2

1.0

STRESS - PSI

(1) ACTUATOR SHAFT DIA - 1.0

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ERN ACTUATOR SHAFT RADIAL BEARING.

1. BEARINGS SHRINK FITTED TO PROVIDE A MIN. OF .0002 INTERFERENCE @ 1200°F.

2. BEARING TO ACTUATOR SHAFT CLEARANCE = .0005 $\pm$.0009 @ ROOM TEMPER.

d_i = BRG INSIDE DIA = $.6250 \pm .0005$ ($.6245 \pm .0000$ SHAFT DIA)
(MACHINED AFTER PRESS FIT.)

D_o = BRG OUTSIDE DIA =
PRIOR TO SHRINK FIT

d_c = DIA OF CONTACT SURFACE OF COMPOUND CYLINDERS (BRG HOUSING).

d_o = BRG HOUSING OUTSIDE DIA.

BRG HOUSING PRIOR TO PRESS

d_H = HOUSING I. DIA.

BRG PRIOR TO PRESS FIT

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D_0 - ER6 O.DIA TO PROVIDE .0002 IN INTERFERENCE @ 1200°F

$$D_0 (1 + \alpha_B \Delta T) - d_H (1 + \alpha_H \Delta T) = .0002 \text{ MIN}$$

$$\textcircled{1} \quad D_0 = \left[.0002 + d_H (1 + \alpha_H \Delta T) \right] \frac{1}{(1 + \alpha_B \Delta T)}$$

THE BEARING STRESSES ARE AS FOLLOWS:

(IN A SHRINK FIT, PRESSURE P IS DEVELOPED @ THE CONTACT DIA DUE THE INTERFERENCE. THIS PRESSURE CAUSES THE HOLE (HOUSING) DIA TO INCREASE AND THE BEARING DIA TO DECREASE.

THE EQUATION FOR THE PRESSURE @ THE CONTACT SURFACE FOR SHRINK FITTED CYLINDERS IS:
(SEE NAA-SP-TDR-NO-10667)

$$\textcircled{2} \quad 2\Delta r_b = P d_c \left[\frac{d_c^2 + d_i^2}{E(d_c^2 - d_i^2)} + \frac{\mu_B}{E_B} + \frac{d_o^2 + d_c^2}{E_H(d_o^2 - d_c^2)} + \frac{\mu_H}{E_H} \right]$$

TANGENTIAL STRESS AT d_i :

$$\textcircled{3} \quad \sigma_{t,d_i} = \frac{2P d_c^2}{d_o^2 - d_i^2}$$

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$$\text{LET } d_H = \underline{\underline{.8750 \pm .0005}}_{-.0000}$$

Then FROM ①

$$D_o = \left[.0002 + .8755 (1 + \alpha_H \Delta T) \right] \frac{1}{(1 + \alpha_B \Delta T)}$$

$$\text{HASTELLOY "B" } \alpha_H = 6.75 \times 10^{-6} \text{ @ } 1200^\circ\text{F}$$

$$\text{PB GRAPHITE } \alpha_B = -2.5 \times 10^{-6}$$

$$\Delta T = (1200 - 70) = 1130^\circ\text{F}$$

$$D_o = \left[.0002 + .8755 (1 + 6.75 \times 1130 \times 10^{-6}) \right] \frac{1}{(1 + 2.5 \times 1130 \times 10^{-6})}$$

$$D_{o_{\text{MIN}}} = \left[.0002 + .8755 \times 1.0076275 \right] \frac{1}{1.002825}$$

$$D_{o_{\text{MIN}}} = \frac{.0002 + .882171876}{1.002825}$$

$$D_{o_{\text{MIN}}} = \frac{.882377876}{1.002825} = \underline{\underline{.8799}}$$

GRAPHITE O.D.

$$D_o = \underline{\underline{.8803 \pm .0000}}_{+.0004}$$

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SINCE MOST OF THE SHRINK WILL BE IN THE BLARING THE ASSUMPTION CAN BE MADE THAT $d_c = d_H$ @ R. TEMPERATURE.

THEN FROM (3)

$$2\Delta r_b = \frac{P d_c}{E_B} \left[\frac{d_c^2 + d_i^2}{d_c^2 - d_i^2} - \mu_B \right] + \frac{P d_c}{E_H} \left[\frac{d_o^2 + d_c^2}{d_o^2 - d_c^2} + \mu_H \right]$$

$$2\Delta r_{b_{max}} = D_{O_{max}} - d_{H_{min}}$$

$$2\Delta r_{b_{max}} = .8803 - .8750$$

$$2\Delta r_{b_{max}} = .0053 \text{ IN}$$

$$d_c = .875 \text{ IN}$$

$$d_c^2 = .7652$$

$$d_i = .625 \text{ IN}$$

$$d_i^2 = .3906$$

$$d_o = 1.2 \text{ IN}$$

$$d_o^2 = 1.44 \text{ IN}^2$$

$$\mu_B = .08$$

$$\mu_H = .28$$

$$E_B = 2.6 \times 10^6 \text{ PSI}$$

$$E_H = 27 \times 10^6 \text{ PSI}$$

$$P = \frac{.0053}{\frac{d_c}{E_B} \left[\frac{d_c^2 + d_i^2}{d_c^2 - d_i^2} - \mu_B \right] + \frac{d_c}{E_H} \left[\frac{d_o^2 + d_c^2}{d_o^2 - d_c^2} + \mu_H \right]}$$

$$P = \frac{.0053}{.875 \left[\frac{1}{2.6 \times 10^6} \left(\frac{.7652 + .3906}{.7652 - .3906} - .08 \right) + \frac{1}{27 \times 10^6} \left(\frac{1.44 + .7652}{1.44 - .7652} + .28 \right) \right]}$$

$$P = \frac{5300}{.875 \left[\frac{1}{2.6} \left(\frac{1.1553}{.3746} - .08 \right) + \frac{1}{27} \left(\frac{2.2052}{.6743} + .28 \right) \right]}$$

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$$P = \frac{5300}{-.875(1.15 + .131)}$$

$$P = \underline{4,730 \text{ PSI}} \quad (\text{PRESSURE})$$

STRESS @ INNER SURFACE OF BRC DIA.

$$S_i = -\frac{2Pd_o^2}{d_o^2 - d_i^2}$$

$$S_i = -\frac{2 \times 4,730 \times .7652}{.7652^2 - .3906}$$

$$S_i = -\frac{9460 \times .7652}{.3746} \quad 24$$

$$S_i = \underline{19,200 \text{ PSI}} \quad \text{COMPRESSIVE STRESS}$$

(COMPR. STRESS IS LARGEST
IN SHRINK FIT CASES)

THE MARGIN OF SAFETY

$$\sigma_c = 25000 \text{ psi allowable from "piston" data}$$

$$MS = \frac{25000}{19200} = 1.3$$

$$\underline{MS = 1.3}$$

DURING LAUNCH THE LOADS ON THE BEARINGS
WILL CREATE ADDITIONAL STRESSES WHICH MAY
EXCEED THE ALLOWABLES. THEREFORE REDUCE
BY STRESS DURING LAUNCH

1.2) TRY TO REDUCE SHRINK FIT STRESSES.

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2nd TRY
INCREASING THE HSG HOLE DIA. d_H
TO 1.000 ± 0.005

FROM ①

$$D_{\text{MIN}} = \frac{.0002 + 1.008131375}{1.002825}$$

$$D_{\text{MIN}} = \frac{1.008331375}{1.002825} = 1.0055$$

GRAPHITE $D_0 = 1.0059 \pm 0.0004$
O.D.

DID COMBINATION ALLOWS FOR
INTERFERENCE @ 7200°F OF .00021 TO .00111

SHRINK FIT PRESSURE @ R.TEMP.

$$P = \frac{2\Delta Y_B}{d_c \left[\frac{1}{E_B} \left(\frac{d_c^2 + d_i^2}{d_o^2 - d_i^2} - \mu_B \right) + \frac{1}{E_H} \left(\frac{d_o^2 + d_c^2}{d_o^2 - d_c^2} + \mu_H \right) \right]}$$

μ_B, μ_H, E_H & E_B SAME
AS PREVIOUSLY USED

$$2\Delta Y_{B_{\text{max}}} = 1.0059 - 1.000 = .0059$$

$$d_c^2 + d_i^2 = 1.3906$$

$$d_c = 1.000 \text{ IN} \quad d_c^2 = 1.000 \text{ IN}^2$$

$$d_o^2 - d_i^2 = .6094$$

$$d_i = .625 \text{ IN} \quad d_i^2 = .3906 \text{ IN}^2$$

$$d_o^2 + d_c^2 = 2.74$$

$$d_o = 1.32 \text{ IN} \quad d_o^2 = 1.74 \text{ IN}^2$$

$$d_o^2 - d_c^2 = .74$$

$$P = \frac{.0059 \times 10^6}{1.0 \left[\frac{1}{2.6 \times 10^6} \left(\frac{1.3906}{.6094} - .05 \right) + \frac{1}{27 \times 10^6} \left(\frac{2.74}{.74} + .28 \right) \right]}$$

$$P = \frac{5900}{.909 + .147} = \frac{5900}{1.056}$$

$P = 5580 \text{ PSI (PRESS.) @ MAX PRESS FIT}$

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THE STRESS AT THE INNER SURFACE OF THE GRAPHITE.

$$S_t = - \frac{2P d_o^2}{d_o^2 - d_i^2}$$

$$S_t = - \frac{2 \times 5580 \times 1.0}{.6994}$$

$$S_t = 18,300 \text{ PSI} \quad (\text{a gain of } \sim 1000 \text{ PSI})$$

@ max press fit

MARGIN OF SAFETY

$$M.S. = \frac{25000}{18350} - 1$$

$$M.S. = +.36$$

A GREATER B&B DIAMETER TO FURTHER REDUCE THE STRESSES IS IMPRACTICAL WITHIN THE REF ACTUATOR ENVELOPE

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BRC INSTALLATION TEMPERATURE:

PROVIDE .001 CLEARANCE FOR ASSEMBLY

HOUSING DIA HOT - BRC DIA COLD = .001 CLEARANCE

$$d_{H, \text{HOT}} = .001 + 1.0059$$

$$d_{H, \text{HOT}} = d_{H, \text{COLD}} (1 + \alpha_H \Delta T)$$

$$d_{H, \text{COLD}} = 1.000 \text{ IN MIN}$$

$$\alpha_H = 6.8 \times 10^{-6}$$

$$\Delta T = \left[\frac{d_{H, \text{HOT}}}{d_{H, \text{COLD}}} - 1 \right] \frac{1}{\alpha_H}$$

$$\Delta T = \left(\frac{.001 + 1.0059}{1.000} - 1 \right) \frac{1}{6.8 \times 10^{-6}}$$

$$\Delta T = (1.0069 - 1)$$

$$\Delta T = \frac{6900}{6.8}$$

$$\Delta T = 1015^\circ \text{F}$$

MIN TEMPERATURE TO PROVIDE .001 CLEARANCE AT ASSY:

$$\text{HOUSING TEMP} = 1015 - 70 = \underline{\underline{1085^\circ \text{F}}}$$

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GEAR BEARING.

1. MIN DIAMETRAL INTERFERENCE @ 1200°F = .0002 IN

2. DIMENSIONS:

$$d_i = .4375 \pm \begin{smallmatrix} .0005 \\ .0000 \end{smallmatrix} \text{ IN (BRC I.DIA)}$$

$$d_s = .4373 \pm \begin{smallmatrix} .0000 \\ .0004 \end{smallmatrix} \text{ IN (SHAFT DIA)}$$

$$d_H = .6250 \pm \begin{smallmatrix} .0005 \\ .0000 \end{smallmatrix} \text{ IN (BRC HOUSING I.DIA)}$$

$$d_o = 1.000 \text{ IN BRC HOUSING O.D.}$$

BEARING OUTSIDE DIA - D_o

$$D_o = [.0002 + d_H (1 + \alpha_H \Delta T)] \frac{1}{1 + \alpha_B \Delta T}$$

$$\Delta T = 1200 - 70 = 1130^\circ \text{F}$$

$$\alpha_H = 6.75 \times 10^{-6}$$

INSTEELLOY B

$$\alpha_B = 2.5 \times 10^{-6} \text{ GRAPHITE P5}$$

$$D_o = \frac{.0002 + .6255 \times 1.0076275}{1.002825} = \frac{.0002 + .630271}{1.002825}$$

MIN

$$D_{o \text{ MIN}} = .6287$$

THEREFORE

$$D_o = .6291 \pm \begin{smallmatrix} .0000 \\ .0004 \end{smallmatrix}$$

DIA COMBINATION ALLOWS FOR
INTERFERENCE @ 1200°F .000205 MIN
TO .001105 MAX

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THE SHRINK FIT PRESSURE - P. - @ R. TEMP.

$$P = \frac{2 \Delta r_B}{d_c \left[\frac{1}{E_B} \left(\frac{d_c^2 + d_i^2}{d_c^2 - d_i^2} - \mu_B \right) + \frac{1}{E_H} \left(\frac{d_o^2 + d_c^2}{d_o^2 - d_c^2} + \mu_H \right) \right]}$$

$$2 \Delta r_{B_{MAX}} = .6291 - .6250 = .0041$$

$$d_c^2 + d_i^2 = .5820$$

$$d_c = d_H = .6250 \text{ IN} \quad d_c^2 = .3906 \text{ IN}^2$$

$$d_c^2 - d_i^2 = .1992$$

$$d_i = .4375$$

$$d_i^2 = .1914 \text{ IN}^2$$

$$d_o^2 + d_c^2 = 1.3906$$

$$d_o = 1.000$$

$$d_o^2 = 1.000$$

$$d_o^2 - d_c^2 = .6094$$

$$\mu_H = .28$$

$$E_H = 27 \times 10^{+6}$$

$$\mu_B = .08$$

$$E_B = 2.6 \times 10^{+6}$$

$$P = \frac{.0041 \times 10^6}{.625 \left[\frac{1}{2.6} \left(\frac{.582}{.1992} - .08 \right) + \frac{1}{27} \left(\frac{1.3906}{.6094} + .28 \right) \right]}$$

$$P = \frac{4100}{.625 \left(\frac{2.84}{2.6} + \frac{2.562}{27} \right)}$$

$$P = \frac{4100}{.625 \times 1.189}$$

$$P = \frac{5520 \text{ PSI (PRESSURE)}}{(4920) \text{ @ AVERAGE PRESS FIT}} \text{ @ MAX PRESS. FIT}$$

STRESS @ THE INNER SURFACE OF THE BEARING

$$S = - \frac{2 P d_c^2}{d_c^2 - d_i^2} = \frac{2 \times 5520 \times .3906}{.1992} = 21700 \text{ PSI @ MAX PRESSURE}$$

THIS IS TOO HIGH AND ALLOWS VERY LITTLE OF ADDITIONAL LOAD CARRYING CAPACITY

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$$\text{Let } d_c, d_u = .6875 \pm_{-0.000}^{+0.005}$$

$$d_i = .5000 \pm_{-0.0005}^{+0.0005}$$

$$d_o = 1.000$$

BEARING OUTSIDE DIA. (.0002 MIN SHRINK FIT @ 120°F)

$$D_{o \text{ MIN}} = \frac{.0002 + d_u(1 + \alpha_u \Delta T)}{(1 + \alpha_B \Delta T)}$$

$$1 + \alpha_u \Delta T = 1 + 6.75 \times 10^{-6} \times 1130 = 1.0076275$$

$$1 + \alpha_B \Delta T = 1 + 2.5 \times 10^{-6} \times 1130 = 1.002825$$

$$D_{o \text{ MIN}} = \frac{.0002 + .688 \times 1.0076275}{1.002825} = \frac{.0002 + .693248}{1.002825}$$

$$D_{o \text{ MIN}} = .6915 \text{ MIN}$$

$$D_o = \underline{\underline{.6919 \pm_{-0.0004}^{+0.0000}}}$$

SHRINK FIT PRESSURE

$$2\Delta F_{B \text{ MAX}} = .6919 - .6875 = .0044$$

$$d_c^2 = .472$$

$$d_i^2 = .250$$

$$d_c^2 + d_i^2 = .722$$

$$d_c^2 - d_i^2 = .222$$

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$$P = \frac{.0044 \times 10^{-6}}{.6875 \left[\frac{1}{2.6} \left(\frac{.722}{.222} - .08 \right) + \frac{1}{27} \left(\frac{1.3906}{.6094} + .28 \right) \right]}$$

[SEE PAGE 13]

$$P = \frac{4400}{.6875 [1.22 + .095]}$$

$$P = \frac{4400}{.6875 \times 1.315}$$

$$P = 4880 \text{ PSI (PRESSURE)}$$

STRESS @ THE INNER SURFACE

$$S = - \frac{2Pd_c^2}{d_c^2 - d_i^2}$$

$$S = - \frac{2 \times 4880 \times .472^2}{.222^2}$$

$$S = 20700 \text{ PSI}$$

$$MS = \frac{25000}{20700} - 1$$

MS = +.21 excluding launch load stresses

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IF THE SHAFT DIAMETER IS INCREASED TO d ;
 $.5000^{+0.005}_{-0.000}$ IN. THE SHRINK FIT STRESSES ARE:

$$d_c = .625 \quad d_c^2 = .3906$$

$$d_i = .500 \quad d_i^2 = .250$$

$$d_c^2 + d_i^2 = .6406$$

$$d_c^2 - d_i^2 = .1406$$

THEREFORE:

$$P = \frac{.0041 \times 10^6}{.625 \left[\frac{1}{2.6} \left(\frac{.6406}{.1406} - .08 \right) + \frac{1}{27} \left(\frac{.3906}{.6094} + .26 \right) \right]}$$

$$P = \frac{4100}{.625 \left(\frac{4.27}{2.6} + \frac{2.562}{27} \right)}$$

$$P = \frac{4100}{.625 \times 1.81}$$

$$P = 3620 \text{ psi (pressure) @ MAX PRESS FIT}$$

(3220) @ average press fit

STRESS

$$S = \frac{2 \times 3620 \times .3906}{.1406}$$

$$S = 20,000 \text{ PSI @ MAXIMUM PRESS FIT TOLER.}$$

(17500 @ average press fit)

MARGIN OF SAFETY EXCLUDING ADDITIONAL STRESS DUE TO LAUNCH LOADS

$$M.S. = \frac{25000}{20000} - 1$$

$$M.S. = +.25$$

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THE SHRINK FIT STRESSES FOR BOTH THE ROTOR SHAFT BEARING AND THE GEAR SHAFT BEARING IS VERY HIGH. THEREFORE IF A SELECTIVE FIT METHOD IS USED, THE BRC SHRINK FIT STRESSES CAN BE REDUCED TO THE FOLLOWING: (GOAL IS LESS THAN 15000 PSI)

SEE PAGE 1 FOR NOMENCLATURE:

RADIAL BEARING - ROTOR SHAFT DIM'S.

$$d_i = .6250 \pm \frac{.0005}{.0000}$$

$$d_u = .8125 \pm \frac{.0005}{.0000} = dc$$

$$d_o = \sim 1.21 \text{ IN.}$$

D_o = BEARING O.D. PRIOR TO SHRINK FIT

$$D_o = \frac{.0001 \text{ INTER. @ } 1200^\circ + d_u (1 + \alpha_u \Delta T)}{1 + \alpha_B \Delta T}$$

$$\text{Housing } 1 + \alpha_u \Delta T = 1.0076275$$

$$\text{Bearing } 1 + \alpha_B \Delta T = 1.002825$$

$$D_o = \frac{.0001 + d_u (1.0076275)}{1.002825}$$

$$D_o = .8172 \pm \frac{.0000}{.0007}$$

SELECTIVE FIT TABLE

FOR Housing DIA = .8125	.8126	.8127	.8128	.8129	.8130
SELECT BEARING } D_o = .8165	.8166	.8167	.8168	.8169	.8170
O.D } D_o = .8167	.8168	.8169	.8170	.8171	.8172
ΔK_{max}	.0042	.0042	.0042	.0042	.0042

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$$d_c^2 = .660$$

$$\mu_H = .28$$

$$d_i^2 = .391$$

$$\mu_s = .08$$

$$d_c^2 + d_i^2 = 1.051$$

$$E_H = 27 \times 10^6$$

$$d_c^2 - d_i^2 = .269$$

$$E_s = 2.6 \times 10^6$$

$$d_o^2 + d_c^2 = 2.10$$

$$d_o^2 - d_c^2 = .78$$

SHRINK FIT PRESSURE

$$P = \frac{2418}{d_c \left[\frac{1}{E_H} \left(\frac{d_o^2 + d_i^2}{d_o^2 - d_i^2} - \mu_H \right) + \frac{1}{E_s} \left(\frac{d_o^2 + d_c^2}{d_o^2 - d_c^2} + \mu_s \right) \right]}$$

$$P = \frac{.0042 \times 10^6}{.812 \left[\frac{1}{2.6} \left(\frac{1.051}{.269} - .08 \right) + \frac{1}{27} \left(\frac{2.10}{.78} + .28 \right) \right]}$$

$$P = \frac{4200}{.812(1.47 + .11)} = \frac{4200}{.812(1.58)}$$

$$P = 3280 \text{ PSI PRESSURE}$$

BEARING STRESS

$$S_s = - \frac{2 P d_c^2}{d_c^2 - d_i^2} = - \frac{2 \times 3280 \times .660}{.269}$$

$$S_s = 16,100 \text{ PSI}$$

SELECTIVE FIT MAY BE ABANDONED IF
BEARING O.D & TOLERANCE IS REVISED
TO THE FOLLOWING:

$$D_o = .8169 \begin{matrix} +.0000 \\ -.0004 \end{matrix}$$

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THE STRESS ON THE BEARING DUE TO LAUNCH LOADS.

$$Load = 1658 \text{ LB ULTIMATE}$$

USING THE BEARING PROJECTED AREA AS BEING THE EFFECTIVE SUPPORT FOR THE LOAD

$$S_b = \frac{Load}{Area} = \frac{1658}{.625 \times .500} \leftarrow \text{Bolt I.D.} = .625 \text{ EFFECT. A LENGTH} = .500$$

$$S_b = 5300 \text{ PSI COMPRESSIVE}$$

THIS STRESS ADDED TO THE SHANK FIT STRESS COMBINES TO GIVE:

$$S_c = S_s + S_b$$

$$S_c = 16100 + 5300$$

$$S_c = 21400 \text{ PSI COMPRESSION}$$

$$M.S. = \frac{25000}{21400} - 1$$

$$\text{ULTM. } M.S. = +.17$$

THIS IS QUITE CLOSE TO THE MATERIAL ULTIMATE & FAILURE MAY OCCUR.

$$\text{LIMIT } M.S. = \frac{25000}{17640} - 1$$

$$M.S. = +.27$$

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... MARGIN OF SAFETY USING THE COMBINED STRESS ON THE BEARINGS.

COMBINED STRESS

$$S_1 = (S_B \sin^2 \theta + S_S \cos^2 \theta)$$

$$S_2 = -\frac{1}{2} (S_B - S_S) \sin 2\theta$$

$$\tan 2\theta = \frac{S_S}{S_B}$$

ACTR. BEARING

$S_S = 16100$ PSI SHRINK FIT STRESS

THE DIRECT BEARING STRESS S_B CALCULATED EQUALS 5300 PSI AND IS THE AVERAGE BEARING STRESS.

THE ACTUAL BEARING STRESS DISTRIBUTION IS PROBABLY SINUSOIDAL & THEREFORE A CONSERVATIVE APPROACH IS TO ASSUME THAT THE MAX STRESS (BEARINGS) IS TWICE THE AVERAGE OCCURRING UNDER THE LOAD.

$$\therefore S_{B \text{ MAX}} = 2 \times 5300 = 10600 \text{ PSI}$$

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the combined stress then is.

$$\tan \theta = \frac{S_s}{S_b} = \frac{16100}{10600} = 1.52$$

$$\theta = 56^\circ 40'$$

$$\sin \theta = .835$$

$$\sin^2 \theta = .697$$

$$\sin 2\theta = \sin 113^\circ 20' = \cos 23^\circ 20' = .918$$

$$\cos \theta = .549$$

$$\cos^2 \theta = .301$$

$$\text{Then } S_n = -[.697(10600) + .301(16100)]$$

$$S_n = -(7400 + 4850)$$

$$S_n = -12250 \text{ PSI}$$

$$\sigma \quad S_p = -\frac{1}{2}(S_b - S_s) \sin 2\theta$$

$$S_p = -\frac{1}{2}(10600 - 16100) \times .918$$

$$S_p = \frac{918}{2} \times 5500$$

$$S_p = 2530 \text{ PSI (SHEAR)}$$

MARGIN OF SAFETIES ACTUATOR BEARING

$$1. \text{ SHRINK PRESS. ONLY } M.S. = \frac{25000}{16100} - 1 = +.55$$

$$2. \text{ SHRINK PRESS + BEARING STRESS COMBINED. } M.S. = \frac{25000}{12250} - 1 = +1.04$$

$$3. \text{ SHRINK PRESS + BEARING STRESS COMBINED WITH } M.S. = \frac{25000}{21400} - 1 = +.17$$

$$4. \text{ SHRINK PRESS. ONLY NON SELECTIVE FIT } M.S. = \frac{25000}{19800} - 1 = +.26$$

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USE SELECTIVE FIT METHOD FOR THE GEAR BEARINGS

$$d_i = .5000 \pm .0005$$

$$d_u = .6562 \pm .0005 = d_c$$

$$d_o = 1.00$$

D_o BEARING O.D. PRIOR TO SHRINK FIT

$$D_o = \frac{(.0001 \text{ MIN. INTERF}) + d_u (1 + \alpha \Delta T)}{(1 + \alpha_o \Delta T)}$$

$$D_o = .6601 \pm .0006$$

$$\text{HOUSING} - 1 + \alpha \Delta T = 1.0076275$$

$$\text{BRG} - 1 + \alpha \Delta T = 1.002825$$

SELECTIVE FIT TABLE:

FOR						
Housing	.6562	.6563	.6564	.6565	.6566	.6567
dia						
SELECT	.6594	.6595	.6596	.6597	.6598	.6599
D_o	.6596	.6597	.6598	.6599	.6600	.6601
$2\Delta F_{B \max}$	.0034					.0034

$$d_c^2 = .431$$

$$d_i^2 = .250$$

$$d_o^2 = 1.0$$

$$d_c^2 + d_i^2 = .681$$

$$d_c^2 - d_i^2 = .181$$

$$\left\{ \begin{array}{l} d_o^2 + d_c^2 = 1.431 \\ d_o^2 - d_i^2 = .659 \end{array} \right.$$

SHRINK PRESS.

$$P = \frac{.0034 \times 10^6}{.659 \left[\frac{1}{2.0} \left(\frac{.681}{.181} - .08 \right) + \frac{1}{27} \left(\frac{1.431}{.659} + .78 \right) \right]}$$

$$P = \frac{3400}{.656 (1.42 + .09)} = \frac{3400}{.655 \times 1.51}$$

$$P = 3440 \text{ PSI pressure.}$$

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STRESS $S = - \frac{2 P d_o^2}{d_o^2 - d_i^2}$

$$S = - \frac{2 \times 3440 \times .431}{.181}$$

$$S = 16300 \text{ PSI COMPRESSION}$$

THE STRESS ON THE BRG DUE TO LAUNCH LOADS:

$$\text{LOAD} = 1417 \text{ LB. ULTIMATE.}$$

$$A = d_i \times \text{BRG LENGTH} = .48 \times .48 = .237 \text{ in}^2$$

$$S_b = \frac{1417}{.237}$$

$$S_b = 6000 \text{ PSI}$$

THE LAUNCH LOAD STRESSES ADDED TO THE SHRINK FIT STRESSES PROVIDES THE FOLLOWING MARGIN OF SAFETY.

$$M.S. = \frac{25000}{16300 + 6000} - 1$$

ULTIM. M.S. = +.12 THIS IS QUITE LOW AND FAILURE OF THE BRG MAY OCCUR.

$$\text{LIMIT M.S.} = \frac{25000}{20300} - 1$$

$$M.S. = +.23 \text{ (LIMIT LOAD)}$$

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Combining stresses for the gear shaft bearing

$$S_s = 16300 \text{ psi}$$

$$S_{B \text{ MAX}} = 2 S_B = 2 \times 6000 = 12000 \text{ psi}$$

combined stress.

$$S_n = - (S_B \sin^2 \theta + S_s \cos^2 \theta) \quad \text{normal stress to plane of max shear stress}$$

$$S_p = -\frac{1}{2} (S_B - S_s) \sin 2\theta$$

$$\theta = \tan^{-1} \frac{S_s}{S_{B \text{ max}}}$$

$$\theta = \tan^{-1} \frac{16300}{12000} = \tan^{-1} 1.36$$

$$\theta = 53^\circ 40'$$

$$2\theta = 107^\circ 20'$$

$$\sin \theta = .805$$

$$\sin^2 \theta = .648$$

$$\cos \theta = .592$$

$$\cos^2 \theta = .35$$

$$\sin 2\theta = .954$$

$$S_n = - (12000 \times .648 + 16300 \times .35) = 13470 \text{ psi compress. stress}$$

$$S_p = -\frac{1}{2} (12000 - 16300) \times .954 = 2050 \text{ psi shear}$$

MARGIN OF SAFETY:

1. SHRINK PRESS. ONLY
SELECTIVE FIT

$$M.S. = \frac{25000}{16300} - 1 = +.53$$

2. SHRINK PRESS + BEARING
STRESS COMB. ABOVE

$$M.S. = \frac{25000}{13470} - 1 = +.85$$

3. SHRINK + BEARING STRESS
COMBINED DIRECTLY

$$M.S. = \frac{25000}{22300} - 1 = +.12$$

4. SHRINK PRESS ONLY
NON SELECTIVE FIT

$$M.S. = \frac{25000}{20000} - 1 = +.25$$

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ROOM TEMPERATURE AND OPERATING TEMP ACT'R
SHAFT TO BEARING CLEARANCES

ROOM TEMP. DIM'S & CLEARANCES. (SHAFT - BRG)

MAX. CLEARANCE = $.6255 - .6241 = .0014$

MIN. CLEARANCE = $.6250 - .6245 = .0005$ MIN & NOM

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1200° TEMP. ACT'G SHAFT & BRG DIM'S
AND CLEARANCES.

$$G_1 = .6245 \pm \begin{smallmatrix} .0000 \\ -.0004 \end{smallmatrix} (1 + \alpha \Delta T)$$

$$\alpha = 6.75 \times 10^{-6}$$

$$\Delta T = 1130^\circ F$$

$$\alpha \Delta T = .0076275$$

$$G_1 = .6245 \pm \begin{smallmatrix} .0000 \\ -.0004 \end{smallmatrix} (1.0076275)$$

$$G_1 = \underline{\underline{.629263 \pm \begin{smallmatrix} .000000 \\ -.000403 \end{smallmatrix}}} @ 1200^\circ F \text{ SHAFT DIA.}$$

$$d_{C 1200} = .8125 \pm \begin{smallmatrix} .0005 \\ -.0000 \end{smallmatrix} (1.0076275)$$

$$d_{C 1200} = \underline{\underline{.818697 \pm \begin{smallmatrix} .000504 \\ -.000000 \end{smallmatrix}}}$$

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THE INSIDE DIA OF THE BEARING
INCREASES IN THE FOLLOWING MANNER

$$B_1 = d_{C1200} - 2t_{1200}$$

t = BEARING SLEEVE THICKNESS.
assembled.

$$t = (.8125 \pm .0005 - .6250 \pm .0005) \frac{1}{2}$$

$$2t = .1875 \pm .0005 @ R.TEMP.$$

$$2t_{1200} = .1875 \pm .0005 (1 + \alpha \Delta T)$$

$$\alpha = 2.5 \times 10^{-6}$$

$$\Delta T = 1130$$

$$\alpha \Delta T = .002825$$

$$2t_{1200} = .1875 \pm .0005 (1 + .002825)$$

$$2t_{1200} = \underline{\underline{.18803 \pm .0005014}}$$

$$B_1 = .818697 \pm .000504 - .18303 \pm .0005014$$

$$\underline{\underline{B_1 = .630667 \pm .0010054 @ 1200^\circ F}}$$

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CLEARANCES BETWEEN SHAFT & BRG @ 1200°F

$$\text{MAX. CLEARANCE} = \overbrace{.630667}^{B_{\text{MAX}}} + \overbrace{.0010054}^{G_{\text{MIN}}} - (.629263 - .000403)$$

$$\text{MAX CLEAR} = \underline{\underline{.0028124 \text{ IN}}}$$

$$\text{MIN CLEARANCE} = .630667 - .0005014 - .629263$$

$$\text{MIN CLEARANCE} = \underline{\underline{.000903 \text{ IN}}}$$

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ESTABLISH GEAR SHAFT TO BRG CLEARANCES FOR
PROPER MESHING OF GEARS @ 1200°F.

$$\text{GEAR SHAFT DIA (1200°F) MAX} = 2L_{\text{MIN (1200)}} + B_{1,1200} + B_{2,1200} - 2\text{ecc.} - P.D(\text{GEAR})_{1200}^{\text{MAX}} - P.D. \text{PINION MAX (1200)} - \text{DIA SHAFT MIN (1200)}$$

$$2L_{\text{MIN 1200}} = 2 \times 1.679 \times 1.0076275 = 3.383613$$

$$B_{1, \text{MIN } 1200^\circ\text{F}} = .630166$$

$$B_{2, \text{MIN } 1200^\circ\text{F}} = .504063$$

$$2\text{ecc.} = .002$$

$$P.D. \text{GEAR}_{\text{MAX } 1200} = 2.875 \times 1.008814 = 2.90034025$$

$$P.D. \text{PINION}_{\text{MAX } 1200^\circ\text{F}} = .4792 \times 1.0076275 = .4828551$$

$$\text{ACTR SHAFT DIA}_{\text{MAX } 1200^\circ\text{F}} = .6245 \times 1.0076275 = .6292634$$

$$4.517842$$

$$4.01445875$$

$$\text{GEAR SHAFT DIA MAX @ } 1200^\circ\text{F} = .503383$$

GEAR SHAFT DIA @ 2.1TEMP.

$$G.S.D._{\text{MAX}} = \frac{.503383}{1.008814}$$

$$G.S.D. = .4990 \text{ MAX}$$

$$\text{MAKE: } G.S.D. = .4989 \pm 0009 (G_2)$$

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ROOM TEMP. & 1200°F GEAR SHAFT TO BEARING CLEARANCE

$$DIM. HOT = DIM. COLD (1 + \alpha \Delta T)$$

$$BRG (1 + \alpha \Delta T) = 1.002825$$

$$HSG (1 + \alpha \Delta T) = 1.0076275$$

$$SHAFT (1 + \alpha \Delta T) = 1.008814$$

$$B_2 @ 1200^\circ F = d_{HSG} - 2t_{1200}$$

$$t = \left(.6562 \pm \frac{.0005}{.0000} - .5000 \pm \frac{.0005}{.0000} \right) \frac{1}{2}$$

$$2t = .1562 \pm .0005 \text{ (COLD.)}$$

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$$B_{1200} = (.6562 \pm .0005)(1.0076275) - (.1562 \pm .0005)(1.002825)$$

$$B_{1200} = .6612052 \pm .000504 - .1566413 \pm .0005014$$

$$B_{1200} = \underline{.504564 \pm .0010054} \quad 1200^\circ\text{F}$$

SHAFT DIA @ 1200°F

$$G_2' = .4989 \pm .0004 \times 1.008814$$

$$G_2 = \underline{.5032973 \pm .000403} \quad 1200^\circ\text{F}$$

CLEARANCES

1. ROOM TEMP.

$$a. \text{ MAX. } = .5005 - .4985 = .002 \text{ MAX}$$

$$b. \text{ MIN. } = .5000 - .4989 = .0011 \text{ MIN \& Nom}$$

2. 1200°F.

$$a. \text{ MAX. } = .5055693 - .5028948 = .002675 \text{ MAX}$$

$$b. \text{ MIN. } = .504063 - .5032973 = .000766 \text{ MIN}$$

$$c. \text{ Nom } = .504564 - .503297 = .001267 \text{ Nom}$$

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BACKLASH BEFORE COATING AT Room Temp.

$$1) \text{ ACTUATOR SHAFT OD. - BEARING I.D. } = .6250_{-0.0000}^{+0.0005} - .6245_{-0.0002}^{+0.0000} \\ = \underline{.0005 \pm .0009}$$

$$2) \text{ GEAR SHAFT OD. - GEAR BEARING I.D. } \\ .5000_{-0.0000}^{+0.0005} - .4989_{-0.0004}^{+0.0000} = \underline{.0011 \pm .0009}$$

$$3) \text{ CENTERS DEVIATION FROM STD CENTERS } \\ 1.679_{-0.000}^{+0.001} - 1.6771 = \underline{.0019 \pm .0010}$$

4.) CONCENTRICITY BETWEEN DIAMETERS = .002 TOTAL

DEVIATION FROM STD CENTERS

$$\Delta_C = (.0005 \pm .0009 + .0011 \pm .0009 + .002) \frac{1}{2} + .0019 \pm .0010$$

$$a) \Delta_{C_{MAX}} = .0056 \text{ IN}$$

$$b) \Delta_{C_{MIN}} = .0027 \text{ IN}$$

FOR 20° P. ANGLE GEARS THE BACKLASH IN INCHES IS:

$$B_{N.T.} = \Delta_C \times 2 \tan(P.A.) = \Delta_C \times 2 \tan 20^\circ$$

$$B_{N.T.} = .72794 \Delta_C$$

$$a.) B_{MAX} = .72794 \times .0056 = .00407$$

$$b.) B_{MIN} = .72794 \times .0027 = .00196$$

} BEARING
RUNOUT
TOLERANCES

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THE MATING GEARS WILL BE MANUFACTURED TO AGMA QUALITY 10B, WHICH ALLOWS A BACKLASH IN THE MATING GEARS OF .002 TO .004 IN. (B_G)

TOTAL BACKLASH AT ROOM TEMPERATURE IS:

$$B_{\text{TOTAL}} = B_{\text{RT}} + B_G$$

$$\begin{aligned} \text{a) } B_{\text{TOTAL}}(\text{MAX}) &= .00407 + .004 = \underline{\underline{.00807 \text{ IN. MAX}}} \\ \text{b) } B_{\text{TOTAL}}(\text{MIN}) &= .00196 + .002 = \underline{\underline{.00396 \text{ IN. MIN}}} \end{aligned} \quad \left. \vphantom{\begin{aligned} \text{a) } B_{\text{TOTAL}}(\text{MAX}) &= .00407 + .004 = \underline{\underline{.00807 \text{ IN. MAX}}} \\ \text{b) } B_{\text{TOTAL}}(\text{MIN}) &= .00196 + .002 = \underline{\underline{.00396 \text{ IN. MIN}}} \end{aligned}} \right\} \text{UNCOATED}$$

THE MoS_2 COATING IN EFFECT WILL REDUCE THE BACKLASH IN THE MATING GEARS EQUIVALENT TO TWICE THE COATING THICKNESS.

LET MoS_2 COATING THICKNESS BE .0004 TO .001

THEN:

$$\begin{aligned} \text{a) } B_{\text{TOTAL}}(\text{COATED})_{\text{MAX}} &= .00807 - 2 \times .0004 = \underline{\underline{.00727 \text{ IN MAX}}} \\ \text{b) } B_{\text{TOTAL}}(\text{COATED})_{\text{MIN}} &= .00396 - 2(.001) = \underline{\underline{.00196 \text{ IN MIN}}} \end{aligned}$$

EXPRESSING THESE IN TERMS OF ANGULAR BACKLASH OF THE GEAR (PITCH DIA 2.575 IN)

$$\theta = \frac{B}{P}$$

$$\begin{aligned} \text{a) } \theta_{\text{MAX}} &= \frac{.00807}{1.4375} \times 57.3^\circ \times \frac{60}{\pi} = 19'18'' \\ \text{b) } \theta_{\text{MIN}} &= \frac{.00396}{1.4375} \times \text{''} \times \text{''} = 9'28'' \\ \text{c) } \theta_{\text{MAX}} &= \frac{.00727}{1.4375} \times \text{''} \times \text{''} = 17'24'' \\ \text{d) } \theta_{\text{MIN}} &= \frac{.00196}{1.4375} \times \text{''} \times \text{''} = 4'41'' \end{aligned} \quad \left. \vphantom{\begin{aligned} \text{a) } \theta_{\text{MAX}} &= \frac{.00807}{1.4375} \times 57.3^\circ \times \frac{60}{\pi} = 19'18'' \\ \text{b) } \theta_{\text{MIN}} &= \frac{.00396}{1.4375} \times \text{''} \times \text{''} = 9'28'' \\ \text{c) } \theta_{\text{MAX}} &= \frac{.00727}{1.4375} \times \text{''} \times \text{''} = 17'24'' \\ \text{d) } \theta_{\text{MIN}} &= \frac{.00196}{1.4375} \times \text{''} \times \text{''} = 4'41'' \end{aligned}} \right\} \begin{array}{l} \text{UNCOATED} \\ \text{COATED.} \end{array}$$

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I. BACKLASH ASSUMING MoS₂ COATING PRESENT ON GEAR.

II. BACKLASH ASSUMING MoS₂ COATING COMPLETELY LOST.

1. ACTR SHAFT TO BEARING CLEAR: .002812 IN MAX
.000903 IN MIN

2. GEAR SHAFT TO BEARING CLEAR: .002675 IN MAX
.000766 IN MIN

3. TOTAL CONCENTRICITY: .002 MAX
.000 MIN,

4. CHANGE FROM STD CENTER LOCATING ϕ OF BEARINGS

$$\epsilon_c = (1.679 \pm_{-0.000}^{+0.001}) 1.0076275 - 1.6771 \cdot 1.0076275 - 1.4375 \quad (\Delta T \times \Delta \alpha)$$

GEAR FITTED RAD

$$\Delta T = 1130$$

$$\Delta \alpha = \alpha_{\text{Rem 41}} - \alpha_{\text{H-ALLOY B}}$$

$$\Delta \alpha = 7.8 - 6.75 = 1.05 \times 10^{-6} \frac{1}{\text{IN}^\circ\text{F}}$$

$$\epsilon_c = .0019 \pm_{-0.0000}^{+0.0012} \times 1.0076275 - 1.4375 (1130 \times 1.05 \times 10^{-6})$$

$$\epsilon_c = .001945 \pm_{-0.000000}^{+0.001008} - .001706 \quad \leftarrow \text{DIFF. EXPANS. TERM}$$

$$\epsilon_c = .000185 \pm_{-0.000000}^{+0.001008}$$

DEVIATION FROM STD CENTERS @ 1200°F DUE TO TOL OTHER THAN GEAR BACKLASH

$$\Delta_{\text{MAX}} = (.002812 + .002675 + .002) \frac{1}{2} + .001193$$

a) $\Delta_{\text{MAX}} = .004936$

b) $\Delta_{\text{MIN}} = (.000903 + .000766) \frac{1}{2} + .000185 = .00102$

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		MODEL NO.

TOTAL BACKLASH DUE TO DEVIATION FROM STD CENTERS PLUS BACKLASH OF THE MATING GEARS. (.002 TO .004)

I

$$B_{1200 \text{ MAX}} = \underbrace{.72794 \times .004936}_{\text{STD CENTER DEVIATION BACKLASH}} + \underbrace{.004 - 2 \times .0004}_{\text{MAX BACKLASH PAIR OF GEARS}} \underbrace{MOS_2 \text{ MIN COAT THICKNESS.}}$$

$$B_{1200 \text{ MAX}} = .003593 + .0032 = \underline{.006793} \text{ IN MAX}$$

$$B_{1200 \text{ MIN}} = \underbrace{(.72794 \times .0162)}_{\text{MIN GEAR BACKLASH}} + \underbrace{.002}_{\text{MIN BACKLASH DUE TO STD CENTER DEVIATION}} - 2 \underbrace{(.001)}_{\text{MOS}_2 \text{ MAX COAT. THICK.}}$$

$$B_{1200 \text{ MIN}} = \underline{.00074} \text{ IN MIN}$$

II BACKLASH IF MO_2 COATING IS LOST:

$$B_{1200 \text{ MAX}} = \underline{.007593} \text{ IN MAX}$$

$$B_{1200 \text{ MIN}} = \underline{.00274} \text{ IN MIN}$$

EXPRESSED IN TERMS OF ANGULAR BACKLASH OF THE GEAR (PITCH DIA 2.575)

$$a) \theta = \frac{.006793}{1.4375} \times 57.3 \times 60 \frac{1}{2} = 16' 12''$$

$$b) \theta = \frac{.00074}{1.4375} \times 11 \times 11 = 1' 48''$$

$$c) \theta = \frac{.007593}{1.4375} \times 11 \times 11 = 18' 9''$$

$$d) \theta = \frac{.00274}{1.4375} \times 11 \times 11 = 6' 30''$$

COATING ON GEAR

NO COATING ON GEAR

PAVIL. CASE AT ACTIVATION
TO GEAR INTERFACE
GEAR MO2 COATED & UNCOATED

(X)
2000 PSI - 1000 PSI

10 X 10 TO THE 1/4 INCH 353-11
KLEUFEL & ESSEN CO. MADE IN U.S.A.