NASA CR-107244
TR-1044

VACUUM JACKETED UMBILICAL LINES TECHNOLOGY ADVANCEMENT STUDY

ROTARY JOINT

AMETEK/Straza
790 Greenfield Drive
El Cajon, California 92021

November 28, 1969

Final Technical Report, Task IV
Contract Number NAS 10-6098

Prepared for

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
John F. Kennedy Space Center
Design Engineering - Mechanical Systems Division
J. B. Downs, Project Manager
Kennedy Space Center, Florida 32899

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This report is one of seven similar tasks related to vacuum jacketed components used on the service arms of the launch umbilical tower at Launch Complex 39, Kennedy Space Center, Florida.

16. Abstract

The purpose of this study was to develop a low heat leak cryogenic rotary coupling capable of 360° rotation. The study was divided into two phases. The first phase covered a period of 6 months and consisted of completing the design tasks. The second phase covered a period of 9 months and consisted of fabricating and testing one 8-inch inside diameter vacuum jacketed rotary joint. One 6-inch inside diameter jacketed rotary was also fabricated. The vacuum jacketed rotary joint concept developed under this contract proved entirely practical for transfer of liquid oxygen or liquid hydrogen across rotating interfaces. The heat leak was not greater than that experienced with vacuum jacketed having bayonet joints. The space envelope required did not exceed that for conventional ring-joint flanges.

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Cryogenic Rotary Joint, Vacuum Jacketed Rotary Joint, Rotary Joint, Insulated Rotary Joint	Government and Contractors

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price
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AMETEK/Straza
790 Greenfield Drive
El Cajon, California 92021

FINAL REPORT

For

Vacuum Jacketed Umbilical Lines
Technology Advancement Study
Task IV - Rotary Joint
Contract Number NAS 10-6098
November 28, 1969

Prepared by: ______
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ABSTRACT

The purpose of this study was to develop a low heat leak cryogenic rotary coupling capable of 360° rotation. The need for this type hardware was understood and investigated during the design phase of the Saturn V launch facility. A rotating joint on the propellant lines located at the swing arm pivot point would preclude using a flex hose in a position where torsional loads can seriously jeopardize the useful life of the line. The torsional loads at this location are due to the slope of the line or out-of-plane motion of the flexing member center line. There are also excessive torsional loads at the missile umbilical interface flex. These are caused by the swing of the umbilical plate just after disconnect.

Several vendors submitted proposals on cryogenic rotaries at the beginning of the Saturn V design programs, but they were considered inadequate at the time. The main disadvantage was that they were not jacketed.

Two basic concepts for jacketed single-plane-assembly cryogenic rotaries were developed during this study. They use the same coupling configuration developed under Task VII of this program for the bayonet joint. Both represent a minimum envelope compatible with existing ring-joint flanges and heat leak values similar to existing bayonet joints.

A cross section of these joints is shown in Drawing 8-100089 and 8-100098. These units use the existing ring joint flange as well as inner and outer pipe diameter envelope currently used on the Launch Complex 39 propellant lines.

This program was divided into two phases. The first phase, covering a period of six months, consisted of completing the design tasks. After receiving approval of the Phase II proposal and test plan, the second phase was started. This covered a period of nine months and consisted of fabricating and testing one (1) eight inch diameter vacuum jacketed rotary joint. One (1) six inch diameter jacketed rotary was also fabricated. However, due to lack of funds, this unit was not tested.

Conclusions

The vacuum jacketed rotary joint concept developed under this contract proved entirely practical. The design goals which were met are as follows:

A. Maintain a low heat leak which is equal to or less than existing cryogenic couplings.

B. Maintain a minimum envelope compatible with the new single-plane entry flanged coupling developed under Task VII of NASA Contract Number NAS10-6098.

C. Use anti-convection barriers developed under Task VII which make this joint compatible with both LO_2 and LH_2 systems.

D. Maintain minimum rotational torque loads and low gas leakage while rotating and non-rotating under high bending moment loads.

Recommendations

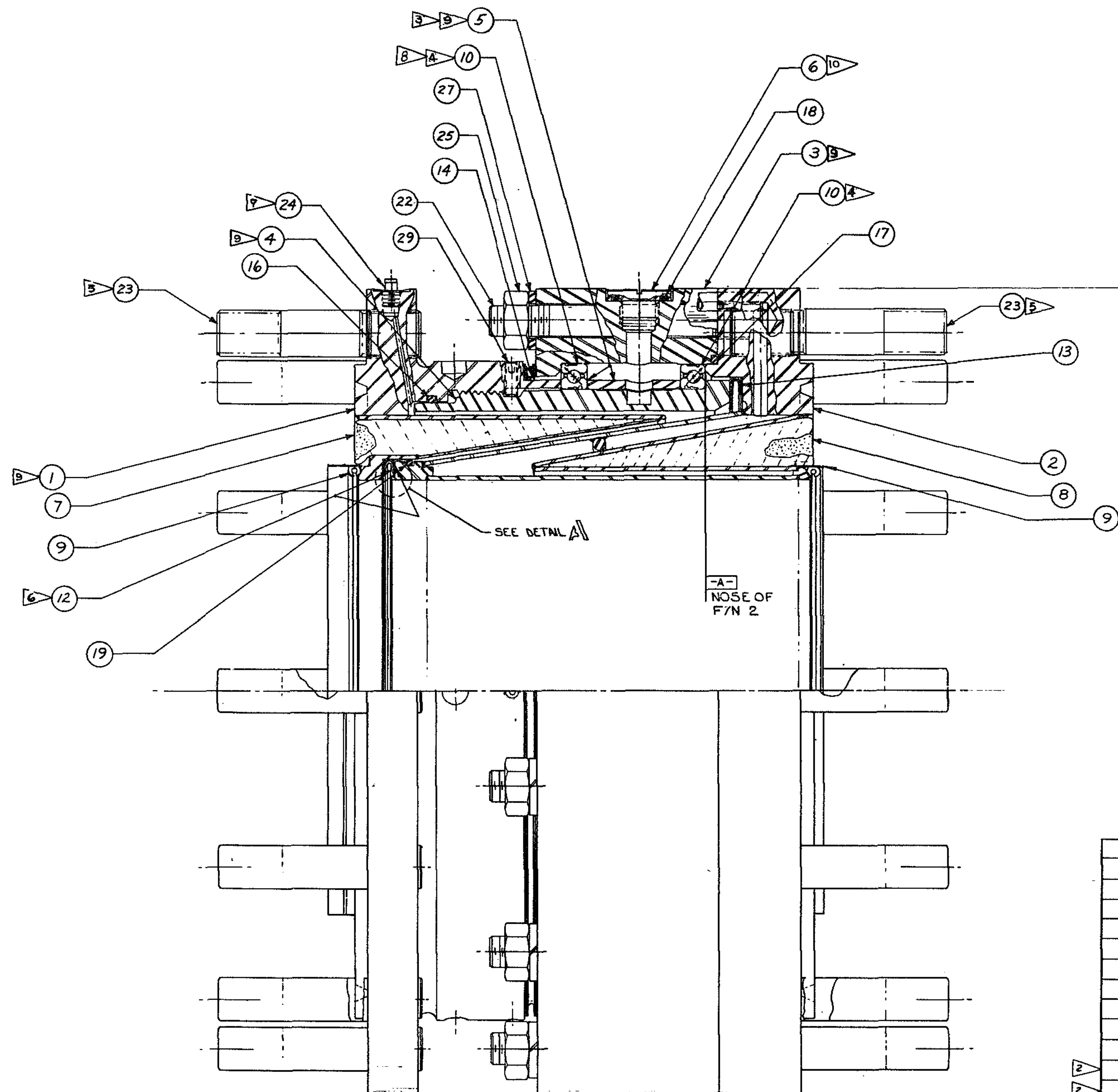
The six inch Chiksan unit should be tested to verify heat leak values, load carrying capacity, leakage and torque values under bending moment load conditions.

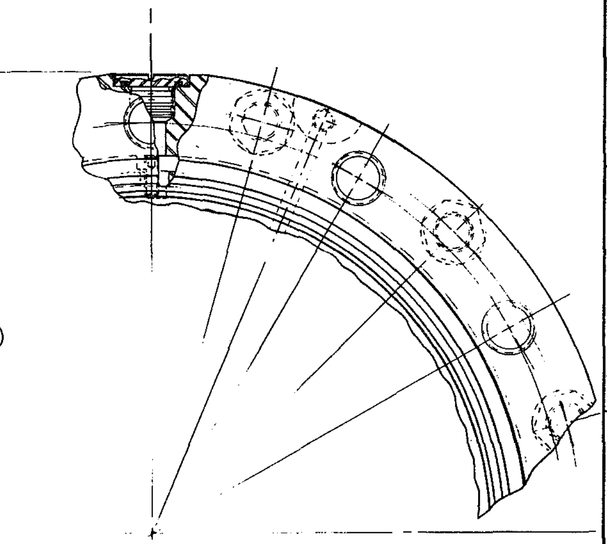
The life cycle test should be completed on the 8 inch unit.

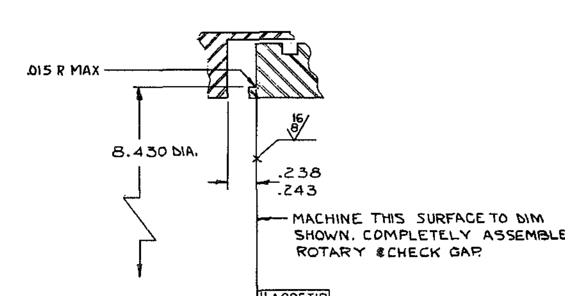
Additional heat leak tests for LH_2 systems should be run in liquid hydrogen.

A full size swing arm fitted with a rotary joint similar to SK 4057-13, Page 22 should be tested to confirm the bending load carrying capability which is a function of the arm pivotal-center to rotary pivotal-center variations.

The compliant bearing should be further investigated to determine design changes required to make it compatible with thrust as well as radial load.







DETAIL A

SCALE 2/1

Note			QTY REQD PER DASH NO. −1	ZONE	FIND NO.	CODE IDENT	PART NUMBER	NOMENCLATURE	SIZE	MATERIAL	SPECIFICATION
					30						
			4		29		MS51021-61	SET SCR, CUP PT	1/4-20UNC X 5/8		
					28						
			12		27		MS35338-145	WASHER, LK, SPLIT	5/8		
					26						
			12		25		MS35690-1010	NUT, HEX, PLAIN	5/8-11UNC		
			1		24		MS20913-1CRK	PLUG, SQHD,	1/8-27NPT		
			24		23		8-040335-5	STUD, LUBRICATED	3/4-9UNC-2AX4.36		
			12		22		8-040335-6	STUD, LUBRICATED	5/8-11UNC-2AX5.25		
					21						
					20						
2			1		19		PRP211B-270	PACKING, TEFLON	8.984 I.D.		
2			1		18		PRP568-117	PACKING, PREFORM	0.815 I.D.		
2			1		17		PRP568-279	PACKING, PREFORM	13.000 I.D.		
2			1		16		PRP568-277	PACKING, PREFORM	11.500 I.D.		
					15						
1			1		14		10025-12,375A	SEAL, AXIAL, EXT,	1/4		
1			1		13		10025-12,50	SEAL, AXIAL, INT,	1/4		
1			1		12		10025-9,25	SEAL, AXIAL, INT,	1/4		
					11						
			2		10		8-040334	BEARING, SLIM, RELIEVED			
			2		9		8-070034-1	NOSE SEAL, BANDED			
			1		8		8-070033-1	INSULATOR, OUTSIDE TAPER			
			1		7		8-070032-1	INSULATOR, INSIDE TAPER			
			1		6		8-040337-1	PLUG, ALIGNMENT HOLE			
			1		5		8-040340-1	SPACER, BEARING			
			1		4		8-040339-1	SLEEVE, BEARING			
			1		3		8-040338-1	COLLAR, BEARING			
			1		2		8-040343-1	COUPLING, ROTARY, JACKETED			
			1		1		8-040341-1	COUPLING, ROTARY, THREADED, JACKETED			

PARTS LIST

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES
DEC TOL: X = ±.1; XX = ±.03; XXX = ±.010. ANGULAR TOLERANCE 1/2°

MATERIAL

MACH SURFACE FINISH: 125 RMS — MACH RAD: .030

CONTRACT NO. NAS-10-6098

PROJECT R. Wussin(?) 4-2-69

DSGN ENGR

STRESS

CHECK [signature] 4-3-69

DRAWN H. Wilber 3-4-69

AMETEK / Straza
790 GREENFIELD DRIVE, EL CAJON, CALIFORNIA 92022

TITLE: COUPLING, ROTARY, CRYOGENIC, 8IN. JACKETED

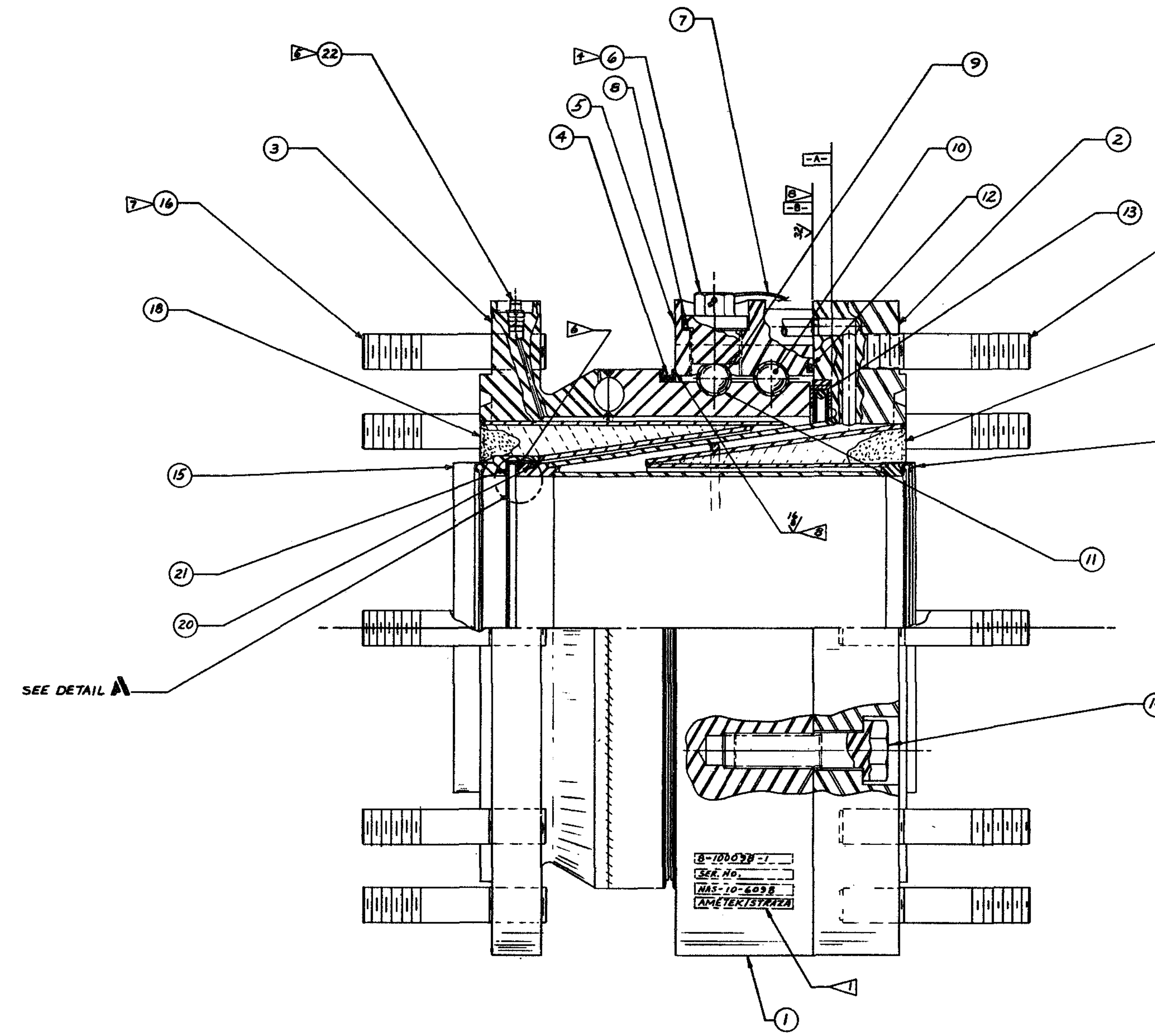
SIZE E — CODE IDENT NO. 03024 — DRAWING NO. 8-100089 — REV

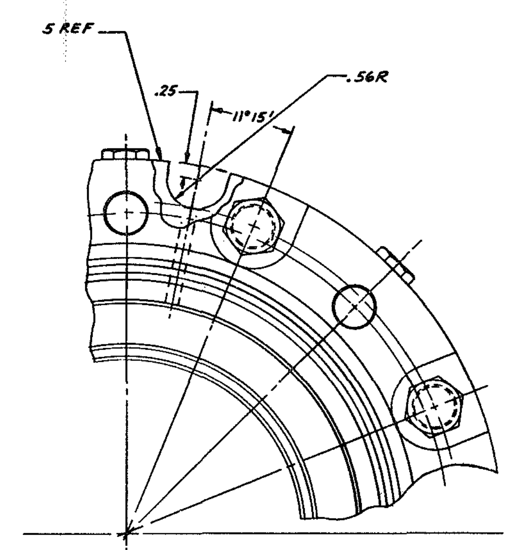
NEXT ASSY

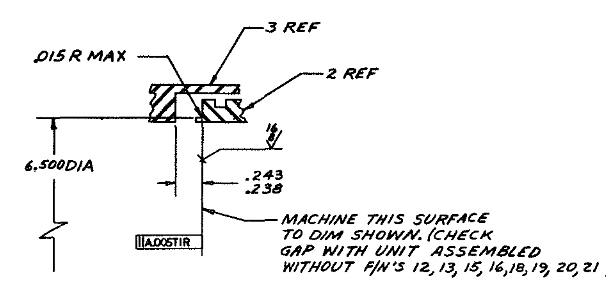
REVISIONS: ZONE | LTR | DESCRIPTION | DATE | APPROVED

10. WRAP WITH TEFLON TAPE PRIOR TO ASSY.
9. TOOLS REQUIRED FOR ASSEMBLY: 8-090153 REF & SPANNER WRENCH #5479C. WRENCH MAY BE PURCHASED FROM: MC MASTER-CARR, P.O. BOX 54960, LOS ANGELES, CALIF. 90054.
8. SEE TEST PROCEEDURE 8-480086 FOR BEARING CHECK.
7. INSTRUMENT PORT - DO NOT CONTAMINATE.
6. SCORE INNER SURFACE OF F/N 12 WITH .03 X 45° GROOVES (12) EQSP RADIALLY.
5. PROTECT ALL THREADS.
4. INSTALL F/N 10's WITH "LOAD FACES" AS INDICATED BY ARROWS.
3. GRIND END OF F/N 5 INDICATED TO ACCOMPLISH PROPER CLOSURE OF THIS ASSEMBLY.
2. AVAILABLE FROM: PLASTIC AND RUBBER PRODUCTS CO, 2100 HYDE PARK BLVD, LOS ANGELES 47, CALIF.
1. AVAILABLE FROM: RAYCO ENGINEERING, 1650-21st. ST., SANTA MONICA, CALIF., 90404.

AMETEK/Straza Rotary, Drawing No. 8-100089







DETAIL A
(SCALE 2/1)

8. FACE THESE SURFACES A/R TO OBTAIN PROPER CLOSURE OF ASSEMBLY.
7. PROTECT ALL THREADS.
6. SCORE SURFACE OF F/N 21 INDICATED WITH GROOVES .03X45°, RADIALLY CUT, 12 PLACES EQUALLY SPACED WITHIN .1.
5. INSTRUMENT PORT – DO NOT CONTAMINATE
4. WRAP THREADS WITH TEFLON THREAD TAPE BEFORE ASSEMBLY.
3. AVAILABLE FROM: RAYCO ENGINEERING CO., 1650-21ST ST., SANTA MONICA, CALIF. 90404.
2. AVAILABLE FROM: PLASTIC AND RUBBER PRODUCTS CO., 2100 HYDE PARK BLVD, LOS ANGELES 47, CALIF.
1. MARK PART NO. & SER. NO. & CONTRACT NO., PER STRAZA WAS 1752.

NOTES:

Note			QTY REQD PER DASH NO. -1	ZONE	FIND NO.	CODE IDENT	PART NUMBER	NOMENCLATURE	SIZE	MATERIAL	SPECIFICATION
			1		22		MS20913-KRK	PLUG, SQ HEAD	1/8-27 NPT		
3			1		21		10025-7,25	SEAL, AXIAL, INTERNAL	1/4		
2			1		20		PRP2118-262	PACKING, TEFLON	7.000 I.D.		
			1		19		8-070033-2	INSULATOR, ANTI-CONVECTION			
			1		18		8-070032-2	INSULATOR, ANTI-CONVECTION			
					17						
			16		16		8-040335-2	STUD BOLT, LUBRICATED	3/4-10 UNC-2A X 3.87		
			2		15		8-070034-2	NOSE SEAL, BANDED			
			8		14		8-040352-13	CAP SCREW HEX HEAD	3/4-10 NC X 2.75		
			1		13		-11	SEAL			
			1		12		-10	PACKING, PREFORMED			
			2		11		-9	BALL RACE, MALE			
			104		10		-8	BALL	.625 DIA		
			2		9		-7	BALL RACE, FEMALE			
			2		8		-6	WASHER, BALL PLUG			
			1		7		-5	WIRE, LOCK			
			2		6		8-040352-4	PLUG, BALL RETAINER			
			1		5		-5	COUPLING	MAKE FROM 8-04352-3		
			1		4		8-04352-2	SEAL			
			1		3		8-040354-1	FLANGE			
			1		2		8-040353-1	FLANGE			
			X		1		-1	ASSEMBLY			

PARTS LIST

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES DEC TOL ANGULAR TOLERANCE X = ± .1 XX = ± .03 1/2° XXX = ± .010	CONTRACT NO. NAS-10-6098	AMETEK/Straza 790 GREENFIELD DRIVE, EL CAJON, CALIFORNIA 92022	
MATERIAL	PROJECT DSGN ENGR STRESS CHECK DRAWN [illegible] 3-19-69	TITLE COUPLING, ROTARY, CRYOGENIC, 6 IN. PIPE	
MACH SURFACE FINISH: 125 RMS MACH RAD: .030 DO NOT SCALE PRINT PARTS TO BE FREE OF BURRS & SHARP EDGES	4057	SIZE E CODE IDENT NO. 03024 DRAWING NO. 8-100098 REV	SCALE 1/1 WEIGHT SHEET 1 OF 1

NEXT ASSY
APPLICATION

Chiksan Rotary, Drawing No. 8-100098

4.0 FINAL REPORT

4.1 PHASE I TECHNICAL REPORT

The following report presents the results of all the tasks required to accomplish the program objectives. These objectives are summarized as follows:

A. "Perform an on-site review to evaluate system operating conditions including: Motion data for calculation of torsional stresses and deflections; inspection of existing flex hose installations for establishing rotary joint locations and types of rotary joints."

B. "Inspect missile umbilical interface and arm hinge for establishing requirements at these points: Test data and failure conditions."

C. "Conduct a search of existing military and commercial rotary joints. Review recent domestic and foreign patents on cryogenic vacuum jacketed joints and latest liquified natural gas cryogenic equipment for applicability to this system."

D. "Review all data and perform design evaluation including: Evaluation of present rotary joints and torsional bellows; evaluation of low torsional deflection and high rotary angulation joints."

E. "Perform an analysis of motions and establish true induced torsional stresses and deflections on the flex hose. Concurrently, perform an evaluation of flange design for the missile umbilical interface."

F. "Prepare a technical report of all Phase I findings."

4.1.1 Hardware Evaluation

4.1.1.1 Functional Review

The week of October 7 to October 11, 1968, was spent reviewing data and talking with personnel at Huntsville and Kennedy Space Center. At Huntsville, Arms 4 and 7 were examined to determine application of rotary joints on similar systems.

Motion pictures of arm testing were reviewed. These showed areas where rotaries could have been used to lessen the torsion loads in flex hoses at the missile umbilical.

The replacement of arm hinge-point flex hose with rotary joints would require a linear compensation device since it would be difficult to maintain arm hinge center to rotary center alignment. The existing arm design would put the rotary out-of-center by about one to one and one-half inches. The linear compensation could be left out only if the rigid piping system attached to the rotaries had sufficient springiness. This would limit the radial and bending moment loads on the bearings to their rated capacity.

While investigating line installations on the LUT at Kennedy Space Center, it was noted that, in some cases, the use of rotaries at arm hinge-points could be seriously restricted due to lack of room between swing arms.

Since there are no rotaries used on the existing system, a direct functional analysis could not be made.

4.1.1.2 Component Operating Requirements

The design data which will be used as a guide for the development of a rotary joint is taken from NASA Specification 75M09783, Rev. D, and 75M06519, Rev. E. These are as follows:

Vent Lines 75M09783

This covers Arms 5 and 7.

Material:	316L	Exposed
	321	Internal
Temperature:	Ambient	+125°F
	Outer Duct	-250°F
	Inner Duct	-423°F
Pressure:	Operating	50 psig
	Proof	100 psig
	Burst	200 psig
Flow:	35,000 scfm on 8-inch	
	70,000 scfm on 12-inch	
Cycling:	10,000 pressure cyc 0 to 50 at -423°F	
Flex Cycling:	500 cycles	
Leak:	1 x 10^{-6} atm-cc/sec Helium at 50 psig	

Propellant Lines 75M06519 ($LH_2 + LO_2$ 8 x 10 V.J.)

Torsional Moment:	600 ft/lbs at 2.3 rpm	
Pressure Cycles:	10,000	0 to w.p. at operating temperature

Pressure: The worst case line pressure will be used to design the rotary:

Operating	190 psi
Proof	285 psi
Burst	760 psi

Rotational Cycles: 500 through 360°

Flow Rate: 15,000 gpm LO_2 or 79,000 scfm GO_2

Temperature +125°F ambient to -423°F LH_2 internal

Heat Gain: The existing flex line heat gain requirements for the 8-inch size runs from 180 to 300 Btu/hr. Most of this heat gain comes from the two bayonet joints and, since a rotary consists of two joints, the higher value will be used as a design goal for the rotary. This will also allow shorter conducting members to minimize the rotary envelope.

Flex Moment: The bending moment design value is taken from the existing line specification for a 90° bend as follows:

100 in.-lbs./degree of unpressurized flexing x 90° =

9000 in.-lbs. ÷ 12 = 750 ft.-lbs., use 900 ft.-lbs

Leakage: Since there are no rotaries to use as a guide for establishing leakage rates, the following will be used as representing a reasonable goal for this study.

Static: Bubble tight He at -423°F and 190 psi.

Dynamic: 3 scim He at -423°F and 30 psig.

Vibration: The vibration g loads will be selected from Procedure II of KSC-STD-164 and are equal to 20 g rms.

4.1.1.3 Evaluation of Equipment

With no equipment of this type presently in use on the S-V propellant loading system, this hardware will be evaluated on the following basis.

A. Allowable envelope where rotary joint might replace flex hose.

B. Load carrying capacity of rotary as function of blast damage, flow, pressure, vibration, heat, and noise.

C. Type of rotary.

(1) Axial
(2) Angular
(3) Tee

These are shown in SK 4057-0.

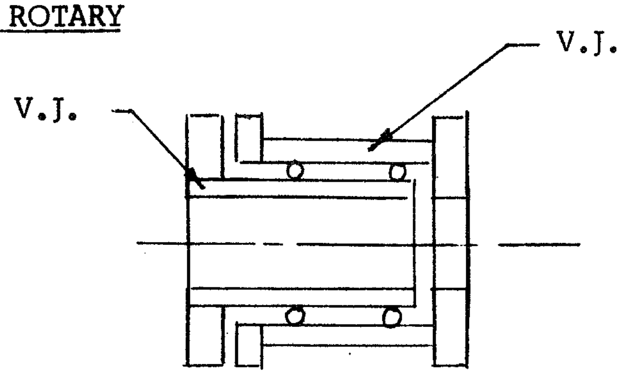
The configuration chosen for this study is the Type A axial. This type is more versatile, less expensive and represents the smallest envelope. It also allows for simpler test fixturing. SK 4057-1 represents a preliminary design concept based on a modified OPW or Barco type rotary with vacuum jacketing.

4.1.1.4 Review of Conditions and Failures

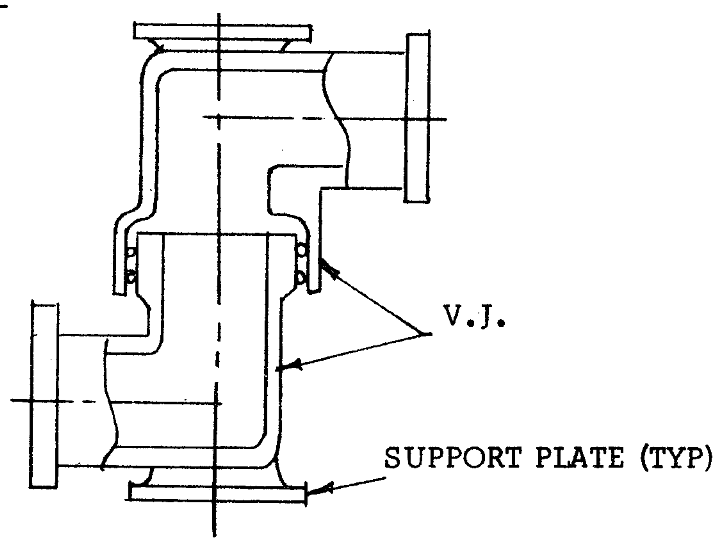
The following conditions are listed as representing those that could cause usage problems if not taken into consideration during the design phase of this type hardware.

A. Assembly
B. Handling abuse
C. G loads — vibrational
D. Heat due to blast
E. Noise — blast
F. Leak allowables (gas)
G. Heat leak
H. Flow loads (fluid or gas)
I. Pressure loads
J. Misalignment loads
K. Lubrication
L. Torque
M. Bearing design
N. Contamination
O. Seals — Static and Dynamic

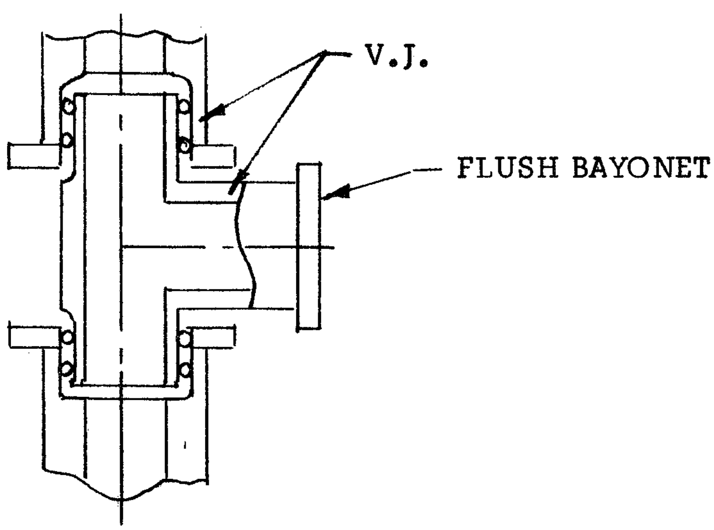
A. AXIAL ROTARY



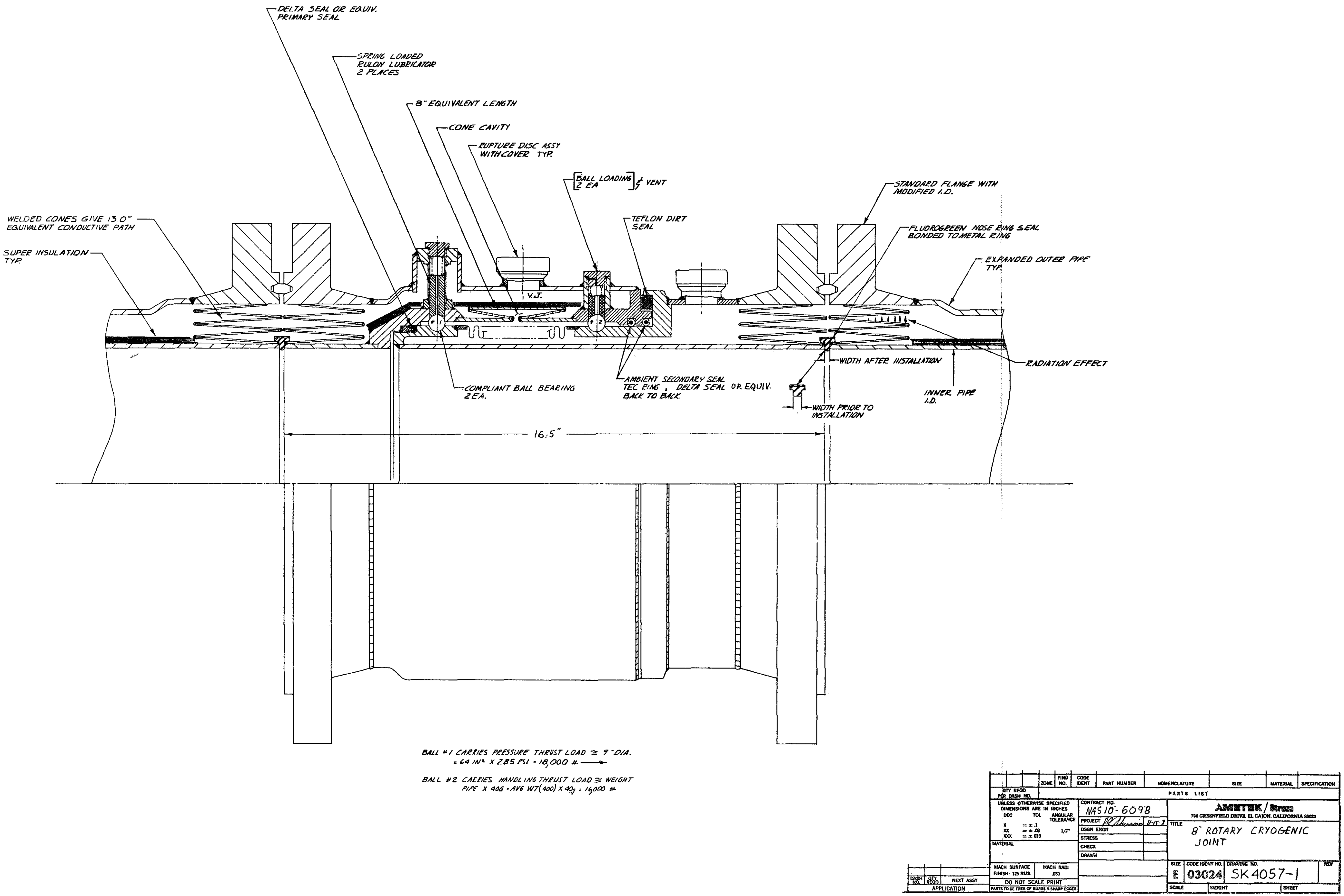
B. ANGULAR



C. TEE



SK 4057-0 Rotary Types



SK 4057-1 (8 inch Rotary)

4.1.1.5 Review of Test Data

With no post test data to work with, the preliminary reliability goal will be based on the preliminary design shown on SK 4057-1.

The first assumption is to establish quantity of units per launch tower. The quantities shown below per arm are not intended to relate to the existing system, but rather to represent a theoretical quantity for establishing a reliability number.

2	at hinge	Arm 1	
3	each	Arm 4	LH_2
2	each	Arm 4	LO_2
2	Umbilical		
1	Hinge		
3	Total	Arm 5	
6	each	Arm 6	
2	each	Arm 7	
18	per launch tower		

The second assumption is that three launch towers are in operation.

54 units + 20% spares = 65 total manufactured

Failure Potential

Assumed failure modes: these will be based on the design shown in SK 4057-1.

		Failures
A.	Primary or nose seal (dynamic) failure at installation.	1
B.	Moisture contamination at ball race No. 1 and No. 2, causing joint to hang up in rotation.	2
C.	Secondary seals failure causing cryo-pumping or external leakage.	1
D.	Bellows damage at installation.	0.25
E.	Contamination in cone cavity causing bearing hang up.	0.50
F.	Failure due to environmental conditions such as shock, vibration, g loads, mis-alignment.	1
	TOTAL	5.75

Preliminary Reliability Number

$$\frac{65 - 5.75}{65.0} \approx 0.9115$$

4.1.2 Product Review

4.1.2.1 State-of-the-Art Investigation

In order to determine vendors who may have developed a jacketed rotary or have an existing rotary suitable for jacketing, the following sources were reviewed.

A. Patent files back to 1950 indexed by Class 285-396, 285-5, and 285-135.

B. Cryogenic Engineering News, Buyers Guide, 1968, 1969.

C. Visual Search Microfilm Files (VSMF)

D. IDEP (Interagency Data Exchange Program).

E. D.D.C. (Defense Documentation Center).

F. Prince File (NASA).

The best source was the VSMF and, from this source, the following vendors were contacted to determine their interest in supplying an existing rotary modified by cryogenic jacketing.

A. Continental-Emsco Company, Garland, Texas

B. Consolidated Controls Corporation, El Segundo, California

C. Langley Corporation, San Diego, California

D. Barco, Division of Aeroquip, Barrington, Illinois

E. Dumont Aviation, Long Beach, California

F. Deublin Company, Northbrook, Illinois

G. Chiksan Division, FMC, Brea, California

H. Johnson Corporation, Three Rivers, Michigan

I. Sealol, Incorporated, Canoga Park, California

J. OPW, Division of Dover Corporation, Cincinnati, Ohio

K. Cryolab, Division of Statics-Dynamics, Incorporated, Los Osos, California

From all the above vendors, only Langley, Barco and Chiksan have worked directly in unjacketed cryogenic rotaries.

Barco submitted an extensive proposal on their LO_2 ball joint showing leakage rates, pressure ratings, torque, etc.; however, the unit which they recommended did not lend itself to vacuum jacketing, and torque values on the 10-inch unit ranged from 1400 ft-lbs at 25 psig to 4750 ft-lbs at 100 psig. The price of this unit was higher than the one selected for the study.

The ball bearing type rotary which Langley manufactured for the ATLAS program appeared to be a reasonable design for jacketing. A tour of their plant was made to determine their present capabilities. Langley is still manufacturing the cryogenic NAFLEX seals for North American; however, their aerospace engineering capabilities are no longer in existence. Straza would have to supply all technical effort required to modify their original 3-inch rotary design to a larger size. The envelope required for their type dynamic seal was larger than could be tolerated for the jacketed unit and the basic single row ball bearing would have very limited moment load capacity.

Chiksan appeared to have the best off-the-shelf design which would lend itself to jacketing. A specification control drawing was prepared and submitted for pricing. When jacketed, it will be as shown in Drawing 8-100098. Chiksan's basic unit shown is presently being used in liquid ethylene and LNG service in the unjacketed condition. It should be noted that in jacketing this unit it becomes a 6-inch line size rotary. In the interest of schedule and budget, we did not press Chiksan into designing a larger 8-inch line size unit.

It was at this point that we decided to develop the Straza rotary shown in SK 4057-7, along with the Chiksan unit. This is being done for the following reasons:

A. Design was not limited in size. Since future propellant lines will be larger, it was felt that an 8-inch unit would be more representative regarding leakage allowables, torque, heat leak, etc.

B. The government would not be limited to a single source procurement as is the case with the patented Chiksan unit.

C. Proved an opportunity to develop the Autonetics Compliant Bearing concept. This allows for a smaller standard bearing envelope. The Compliant Bearing is discussed in the design phase of this report.

The seals which were investigated for use in the AMETEK/Straz-designed rotary are as follows:

A. Bal-Seals Engineering, La Habra, California

B. Raco, Santa Monica, California

C. Skinner Seals, North Hollywood, California

D. Rubly Engineering (Delta Seal), Los Angeles, California

E. Tec Seal Corporation, Wilmington, California

F. Naflex, Langley, California

G. Omniseal-Aeroquip, Burbank, California

The one chosen for this unit is the Raco seal. This seal has the best cross-sectional dimensions compatible with its end use. The Bal-Seals, for instance, have too small a cross-section to work within the tolerances of this design. The Skinner and Delta Seals are too complicated for our envelope. The Tec Seal and Naflex Seals appear to have too limited a deflection. The Omni-Seal would be a close second choice, but it has a limited off-the-shelf size selection.

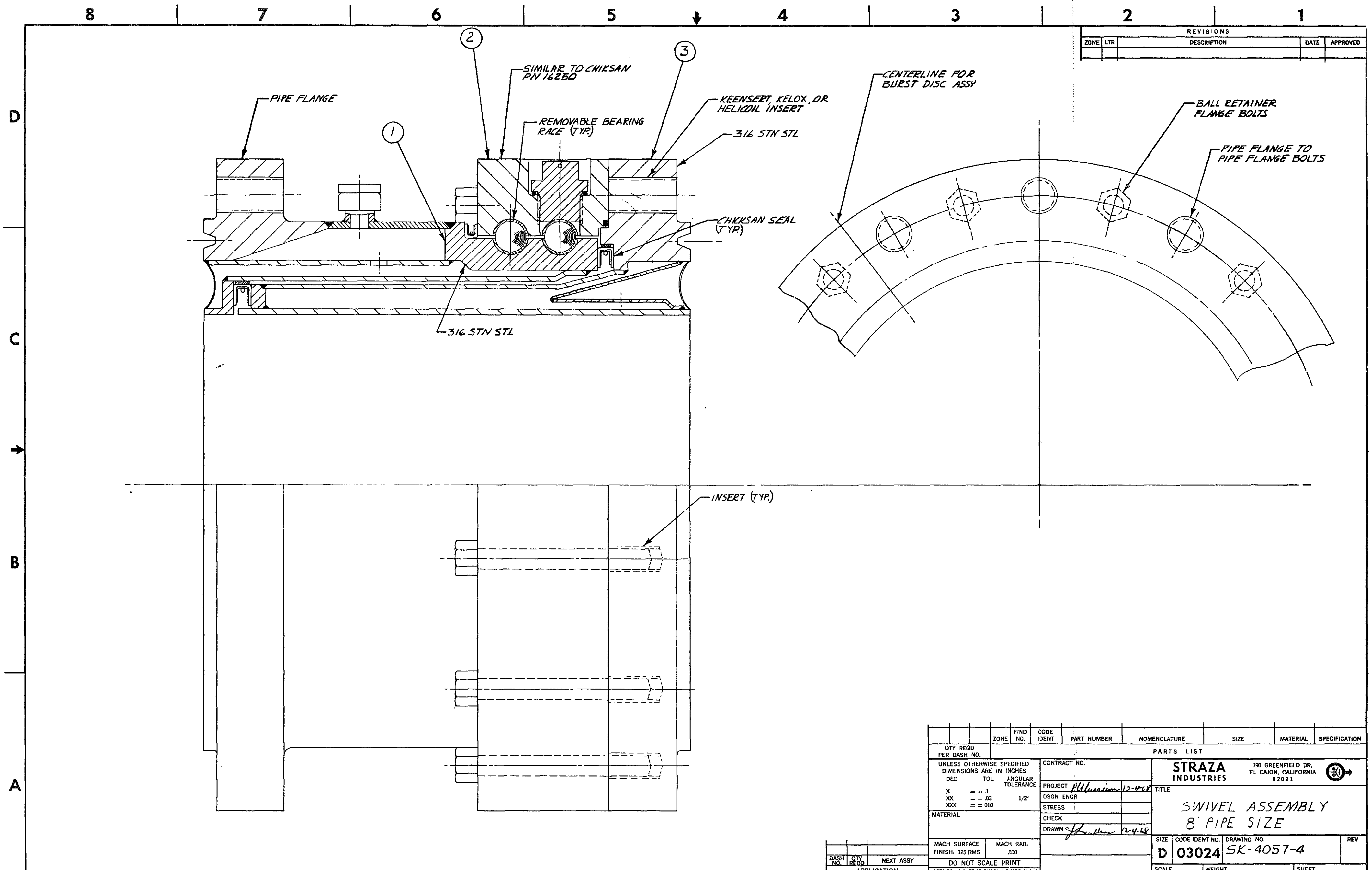
4.1.2.2 Conclusions

The apparently wide selections of rotary designs provided only a limited number to choose from for jacketing. This was due to either disinterest on the part of some vendors, or inadequate cryogenic design capabilities and expensive or impractical envelopes to modify for jacketing. The best design is the Chiksan unit since it provides the smallest envelope when coupled with the standard ring joint flanges that are presently being used on the swing arms.

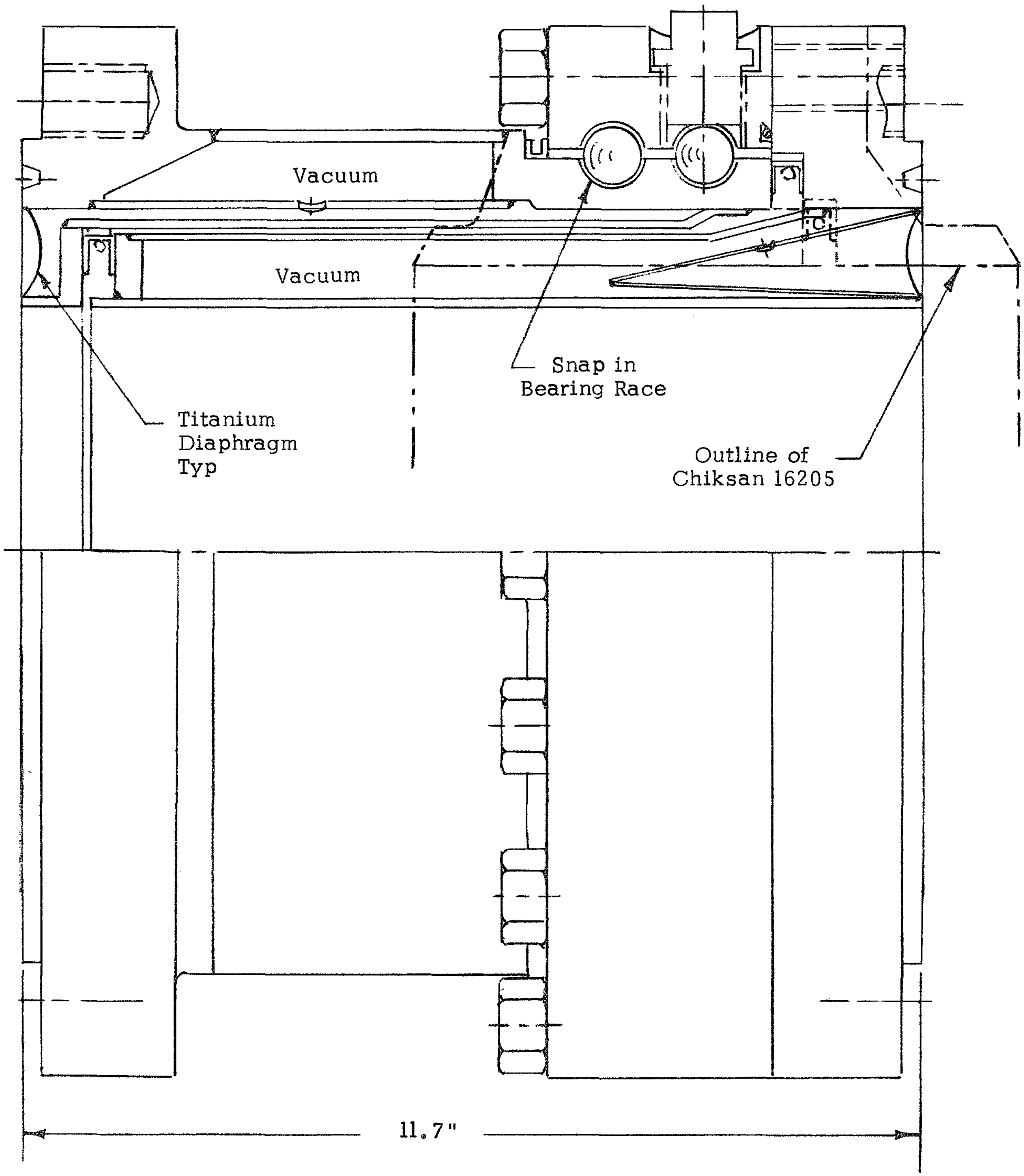
4.1.3 Design Phase

4.1.3.1 Design Evaluation

The final design of the two rotaries selected for analysis evolved as a result of envelope studies similar to those shown in SK 4057-1 and -4. The bayonet portion of the rotary shown in the SK 4057-1 sketch reflects our original thinking prior to evaluating the patented A.D. Little Cryogenic Coupling. The lengths of these units were reduced considerably by employing the joint developed by A.D. Little. The details of this joint with a copy of the patent are discussed in the Bayonet Report (Ref. 1).

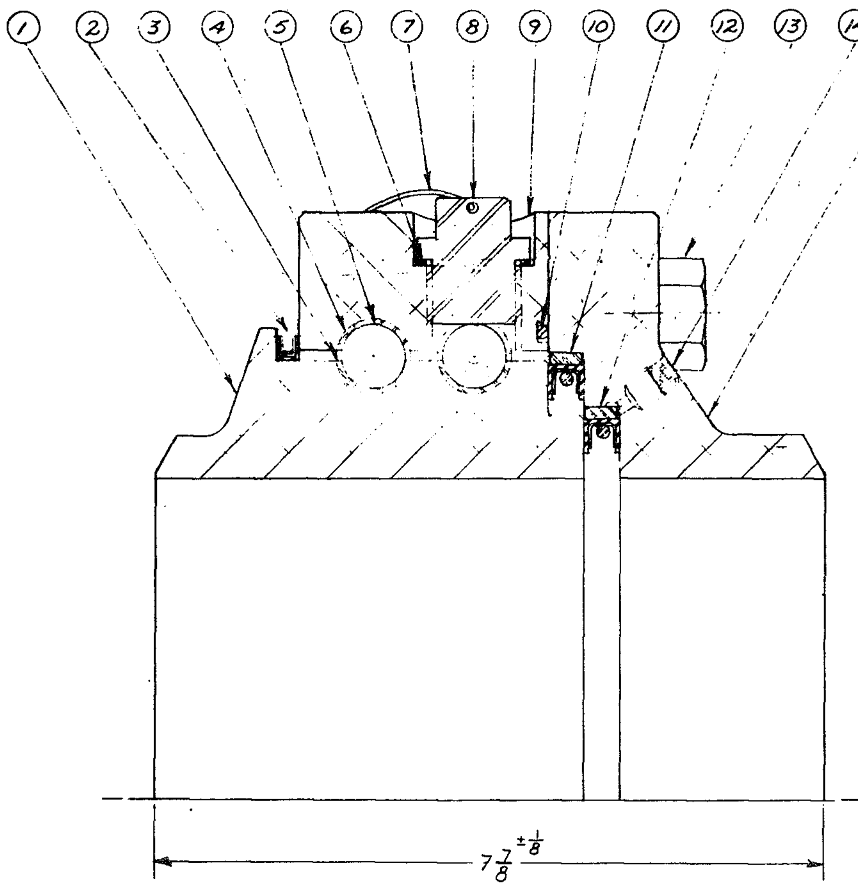


SK 4057-4 (8 inch Rotary)



6" PIPE SIZE
M/F MODIFIED CHIKSAN 16205
SK 4057-5

NOTES:
1. OPERATING PRESSURE - 150 PSI
2. TEST PRESSURE - 300 PSI.
3. OPERATING TEMP. RANGE: -155°F. TO +150°F
4. TEST MEDIA - #LT-99 LACQUER THINNER OR TOLUENE.
5. BOLT THREADS ARE TO BE DRY FILM LUBRICATED (LUBECO PROCESS N350A).
6. BALLS & BALL RACES ARE TO BE COATED WITH TEFLON DISPERSION (SEE ENG. INSTRUCTION DATED MAY 9, 1967).



ITEM	REQUIRED	PART NO.	DESCRIPTION, MATERIAL AND SPECIFICATION (ALL PURCHASED PARTS AS SPECIFIED OR EQUAL)	EST. WT.
15	1	16205-256	COUPLING NIPPLE - ALUMINUM	
14	2		$\frac{1}{8}$" TAPERED PIPE PLUG - STAINLESS STEEL	
13	8		$\frac{3}{4}$" - 10NC - 2 X $2\frac{3}{4}$" HEX HD. BOLT - TYPE 304	
12	1	3102310	PRIMARY SEAL - KEL-F & S.S.	
11	1	3102315	SECONDARY SEAL - KEL-F & S.S.	
10	1	3102302	'O' RING - #6W SYNTHETIC	
9	1	16205-255A	COUPLING - ALUMINUM & S.S. PINS	
8	2	3188961	BALL RETAINER PLUG - TYPE 304 S.S.	
7	1		LOCKWIRE - STAINLESS STEEL	
6	2	3188962	BALL PLUG WASHER - 5052-0 ALUMINUM	
5	104		.625 DIA BALLS - TYPE 440C S.S.	
4	2	3103071	FEMALE SNAPIN BALL RACE - TYPE 301	
3	2	3102299	MALE SNAPIN BALL RACE - TYPE 301	
2	1	3102330	WATER SEAL - TEFLON & S.S.	
1	1	16205-253	SWING NIPPLE - ALUMINUM	
				57

NO	DATE	BY	CHK'D	APP'D

REVISIONS (SEE ENGINEERING CHANGE NOTICE)

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ARTICLES MANUFACTURED IN ACCORDANCE WITH THESE DRAWINGS SHALL BE CONSIDERED CHIKSAN'S DESIGN AND MANUFACTURER AGREES NOT TO MANUFACTURE FOR THE USE OR SALE BY ANY OTHER PERSON OR CONCERN OR FOR MANUFACTURER'S USE OR SALE IDENTICAL ARTICLES OR PARTS THEREOF WITHOUT CHIKSAN'S WRITTEN PERMISSION.

DIMENSION TOLERANCES EXCEPT AS SPECIFIED
FRACTIONAL ±
ANGULAR ±
DEC. X ± XX ± XXX ±
MACH. SURFACES
BREAK ALL SHARP CORNERS
FINISH

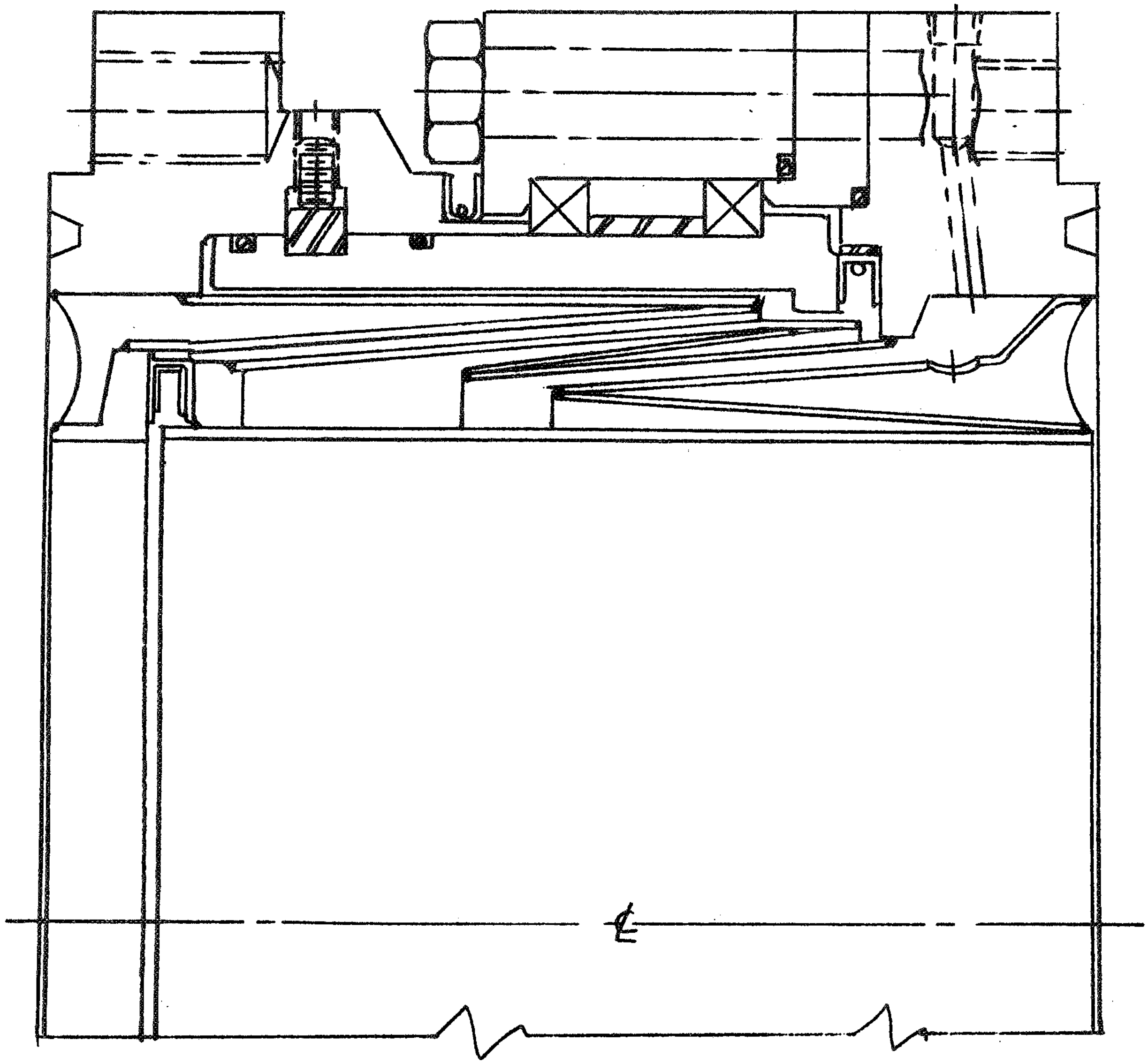
fmc
CHIKSAN
CHIKSAN DIVISION
FMC CORPORATION
BREA, CALIFORNIA

8" WROUGHT AL SWIVEL JOINT FOR LIQUID ETHYLENE SERVICE STYLE 20 ASSEMBLY FOR STYLE 50 JOINT

REF. GB7-23017
NEXT ASSY.
DRAWN Jackson
SCALE FULL SIZE
CHK'D
DATE 9-12-67
APP'D
16205
SHEET 220

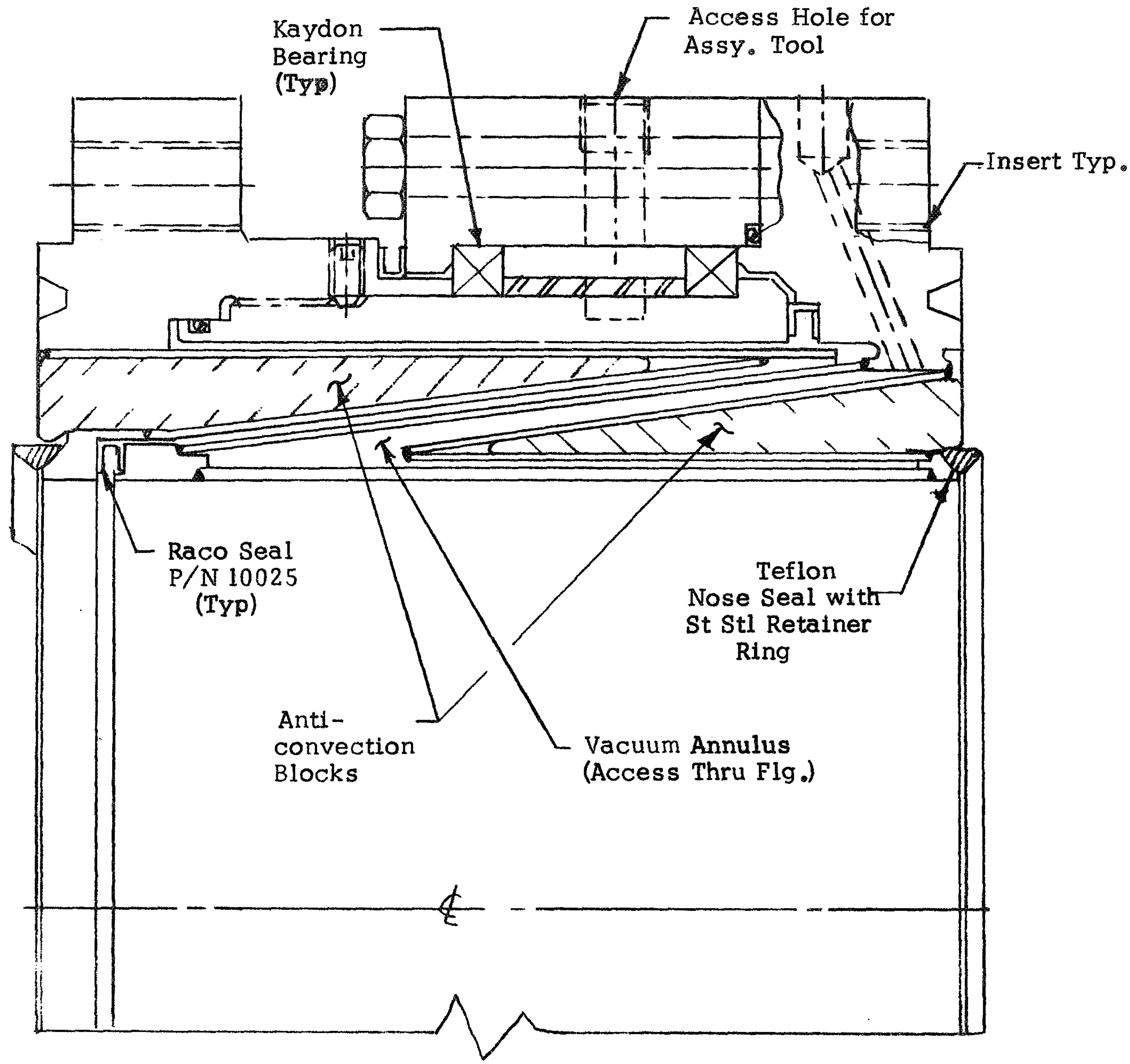
16205 SHEET 220

Chiksan Style 20 Rotary



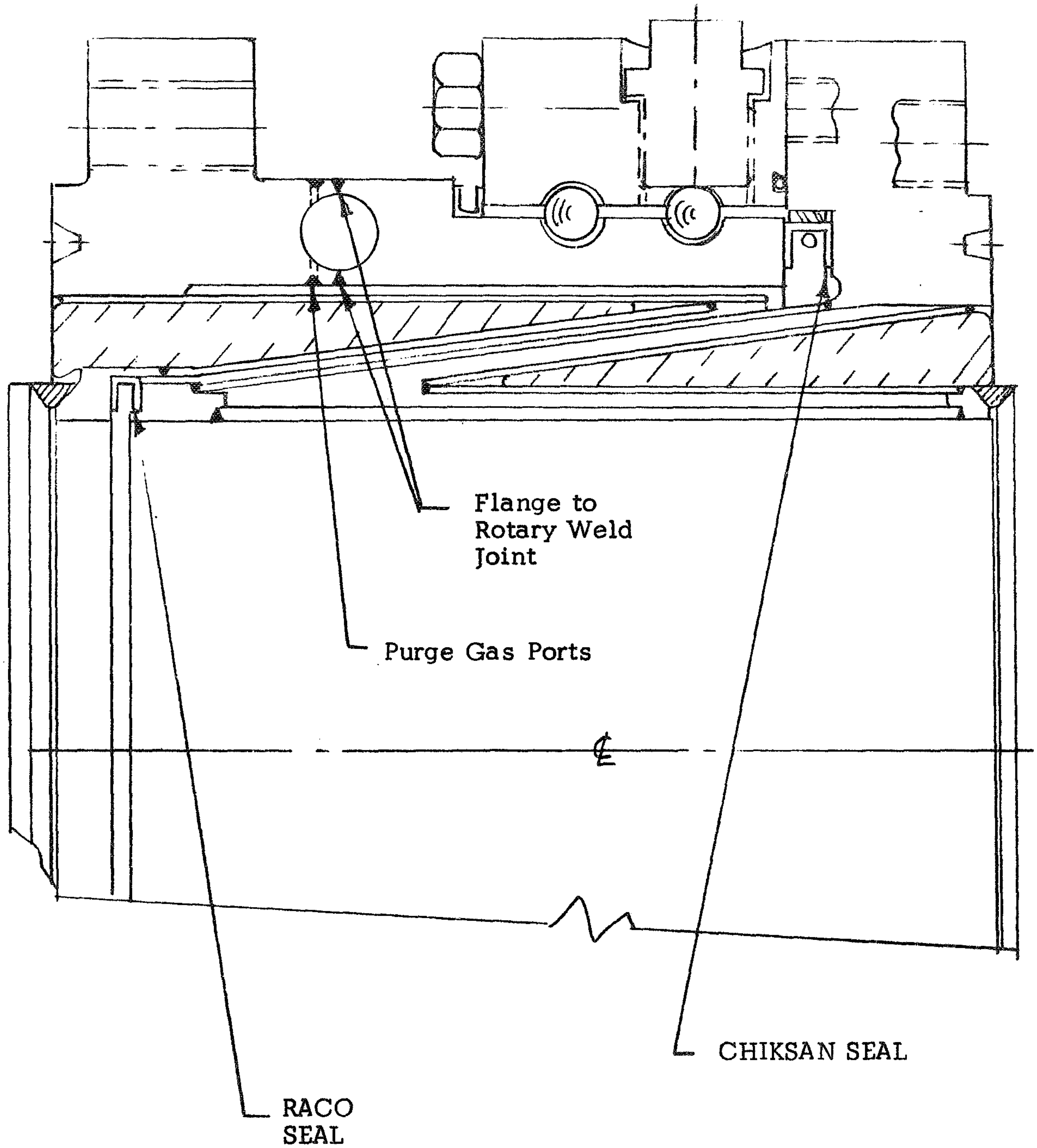
8" Straza Rotary Joint with
a toroidal annular seal (either
titanium or equal low thermal
conductive material).

SK 4057-7 (8 inch Rotary)



8" STRAZA ROTARY

SK 4057-7a (8 inch Rotary)



CHIKSAN 6" ROTARY

SK 4057-8 (6 inch Rotary)

The use of the A.D. Little design jacketing coupled with the Ra[illegible]o Dynamic seal in lieu of the larger Chiksan seal brought the design through the SK 4057-7 stage to the SK 4057-7a unit (8-inch duct size) and the SK 4058-8 unit (6-inch duct size).

One of the main reasons that the Chiksan design was chosen over other ball-loaded type joints similar to Langley or OPW is that it can be welded to any line or flange material, such as stainless steel or aluminum. Their patented snap-in ball races provide this versatility (see SK 4057-5 and SK 4057-8). The load-carrying capacity of this design is still limited to the nominal Hertz stresses whereas the compliant concept used on the AMETEK/Straza unit allows for two to three times more load than that normally shown as "rated" static loads in the standard bearing catalogs.

SK 4057-10 and SK 4057-11 represent several methods where this type rotary could be used in conjunction with hard lines or flex lines. The SK 4057-10 sketch is a modification of the existing 75M07823-9 Arm 6 propellant flex line interface at the missile umbilical. The SK 4057-11 sketch is a modification to the 75M09667-5 Arm 5 vent line flex.

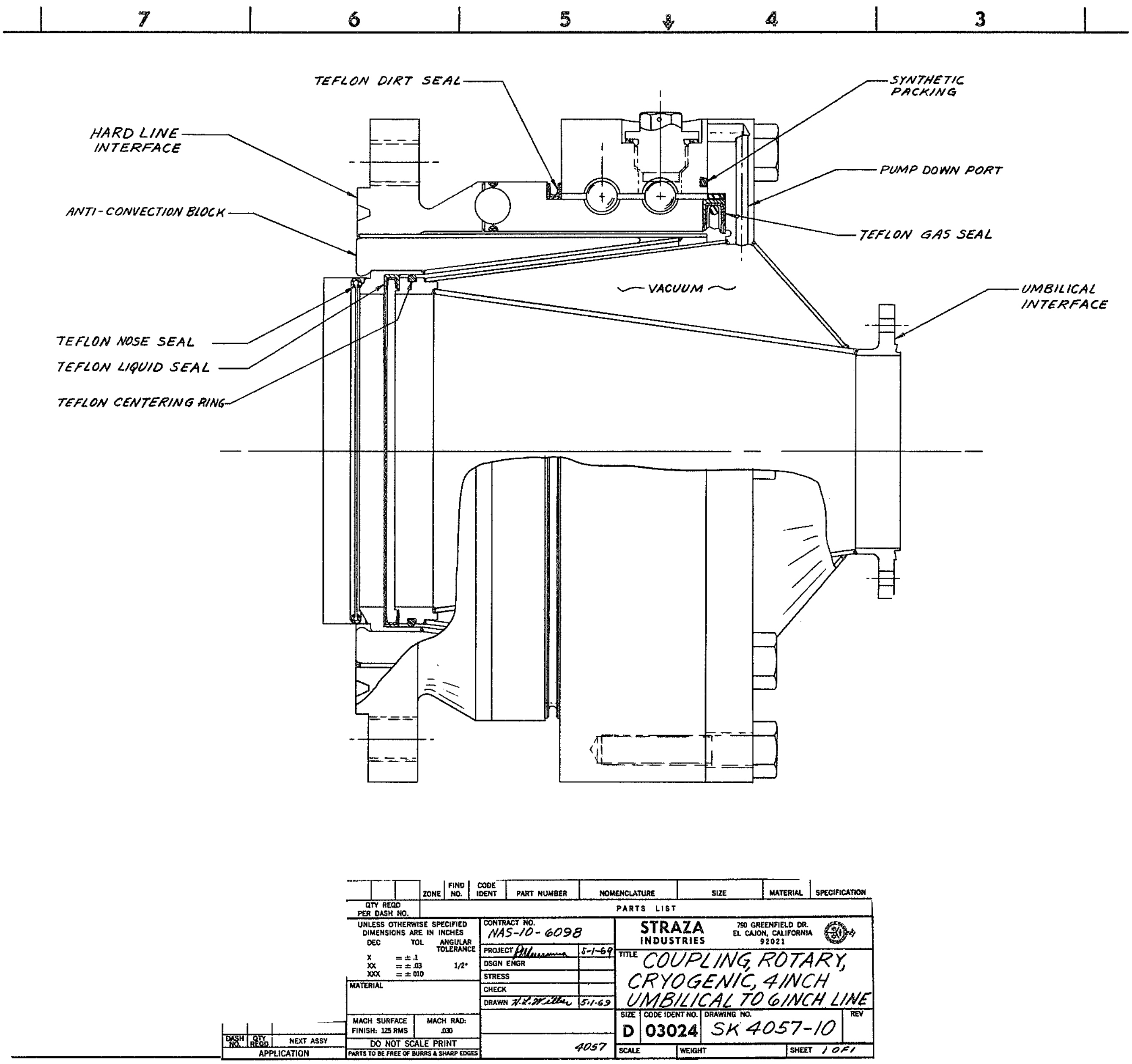
The SK 4057-12 sketch employs a slight venturi action to create a light-weight conoseal type, fully vacuum jacketed rotary. The total pressure drop of this basic 6-inch line size unit has a pressure drop equivalent to a single additional 90° duct bend. This can be seen by comparing the K values on the graphs taken from Product Engineering.

$$\text{The } \frac{d_2}{d_1} \text{ value} = \frac{6.4}{4.8} = 1.32$$

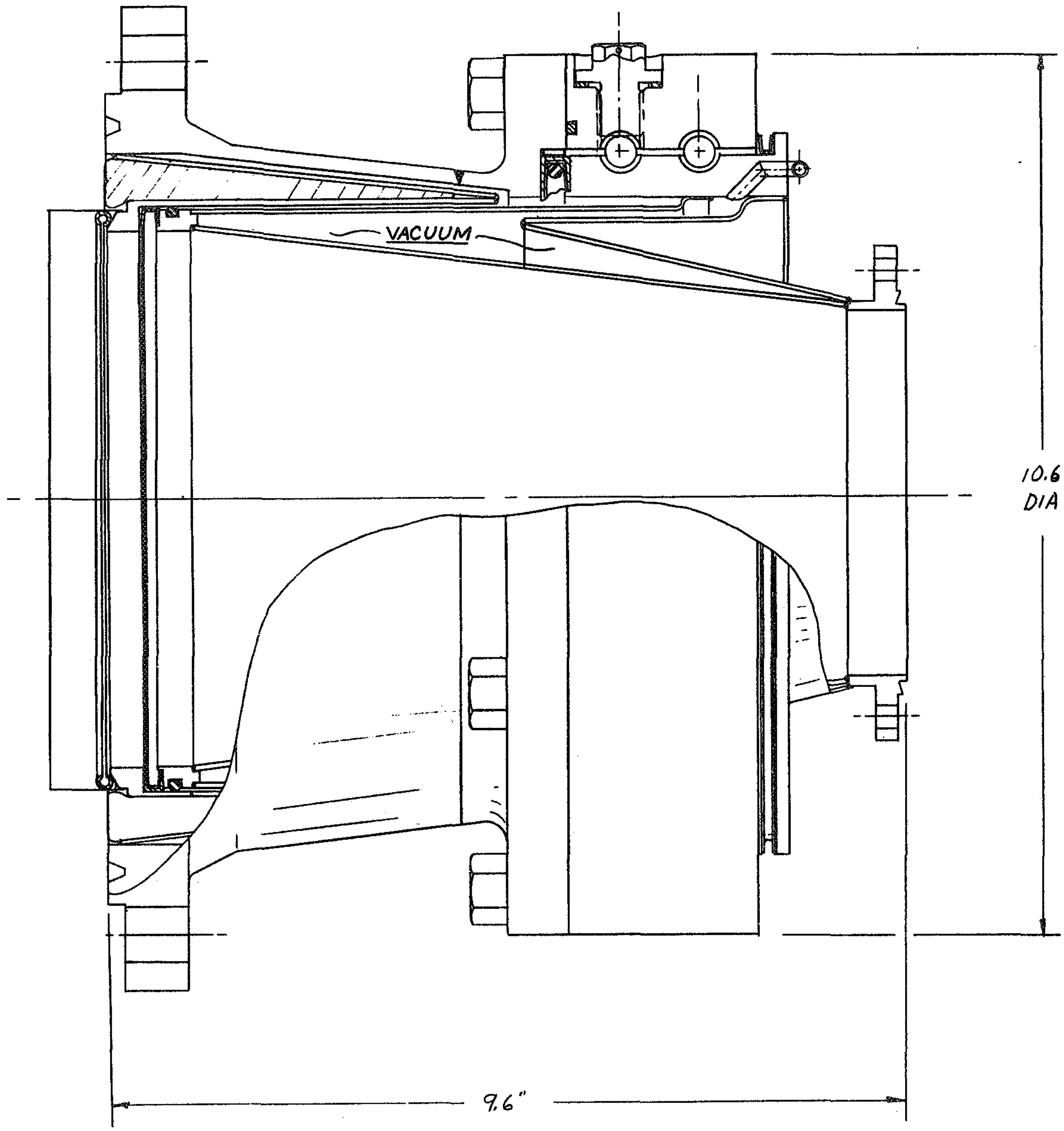
This gives a "gradual changing" K factor of approximately 0.05 for the 10° angle. The entrance K factor is 0.06 for an r/d (0.77/4.8) = 0.16. The sum of these K factors (0.05 + 0.16 = 0.21) is equal to 1d band radius.

The outside diameter of this type unit is 1.4 inches less than the unit shown in SK 4057-10. This produces a 44-pound weight-saving, not including the weight difference between the conoseal flange and the 150-pound ring joint flange. The heat leak will be considerably improved due to the full vacuum jacketing plus the 2.5-inch additional length of the ring-stiffened cone.

The SK 4057-13 sketch shows a typical hinge point rotary installation. The horizontal line must be free to move in its arm supports so that the eccentric hinge-to-rotary centerline motion can be transferred to the flex line at the umbilical plate.



SK 4057-10 (6 inch Rotary)



ZONE | FIND NO. | CODE IDENT | PART NUMBER | NOMENCLATURE | SIZE | MATERIAL | SPECIFICATION

QTY REQD PER DASH NO.

PARTS LIST

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES
DEC TOL ANGULAR TOLERANCE
X = ± .1
XX = ± .03 1/2°
XXX = ± 010

MATERIAL

MACH SURFACE FINISH: 125 RMS
MACH RAD: .030

DO NOT SCALE PRINT

PARTS TO BE FREE OF BURRS & SHARP EDGES

CONTRACT NO. NAS-10-6098

PROJECT Pillumin 5-1-69

DSGN ENGR

STRESS

CHECK

DRAWN H.L. Wilber 5.1.69

4057

STRAZA INDUSTRIES
790 GREENFIELD DR. EL CAJON, CALIFORNIA 92021

TITLE COUPLING, ROTARY, CRYO-GENIC, 4 IN. UMBILICAL TO 6 IN. LINE ('SLIM; LIGHTWEIGHT)

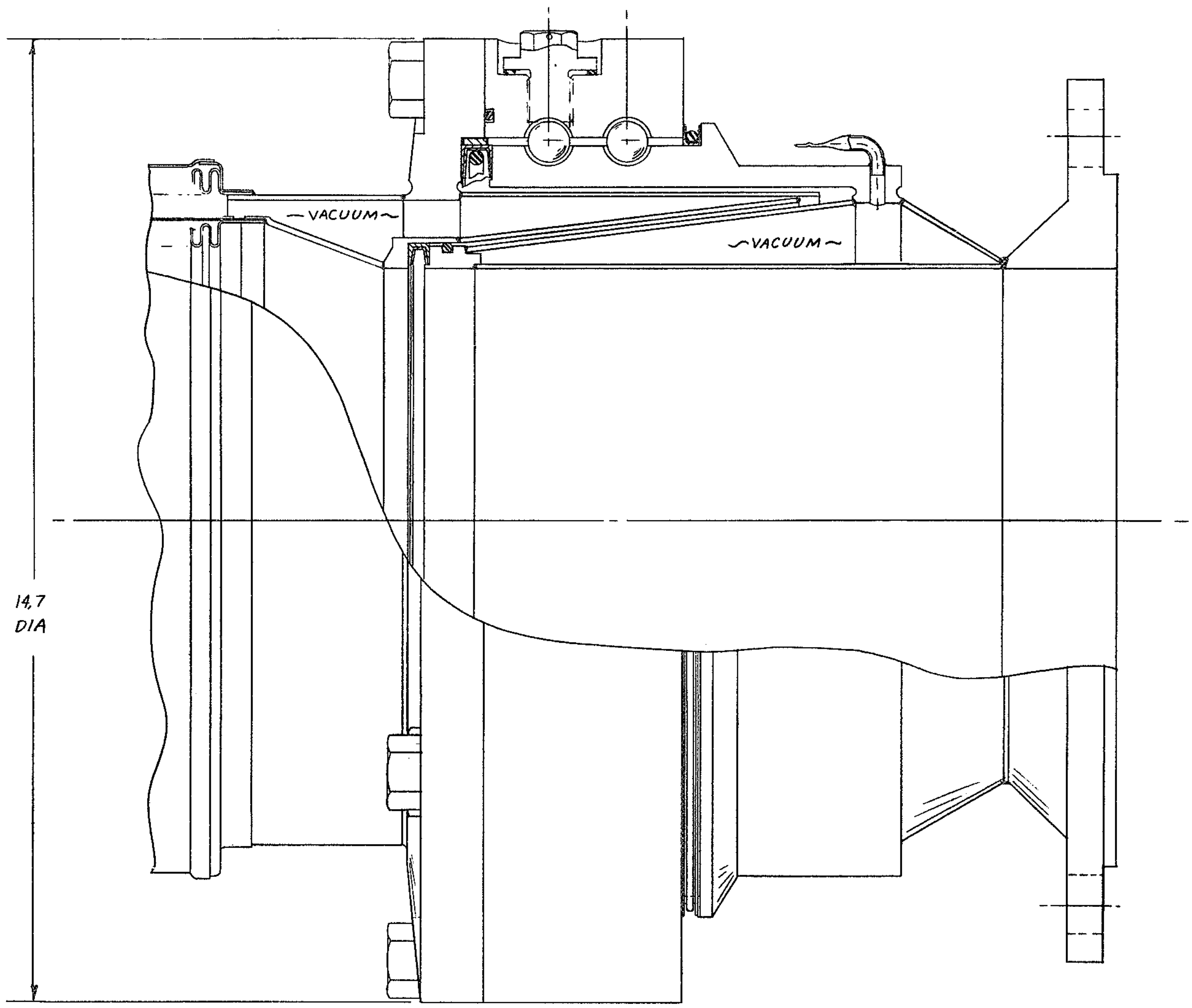
SIZE D | CODE IDENT NO. 03024 | DRAWING NO. SK 4057-10A | REV

SCALE — | WEIGHT | SHEET 1 OF 1

DASH NO. | QTY REQD | NEXT ASSY

APPLICATION

SK 4057-10a (6 inch Rotary) Lightweight

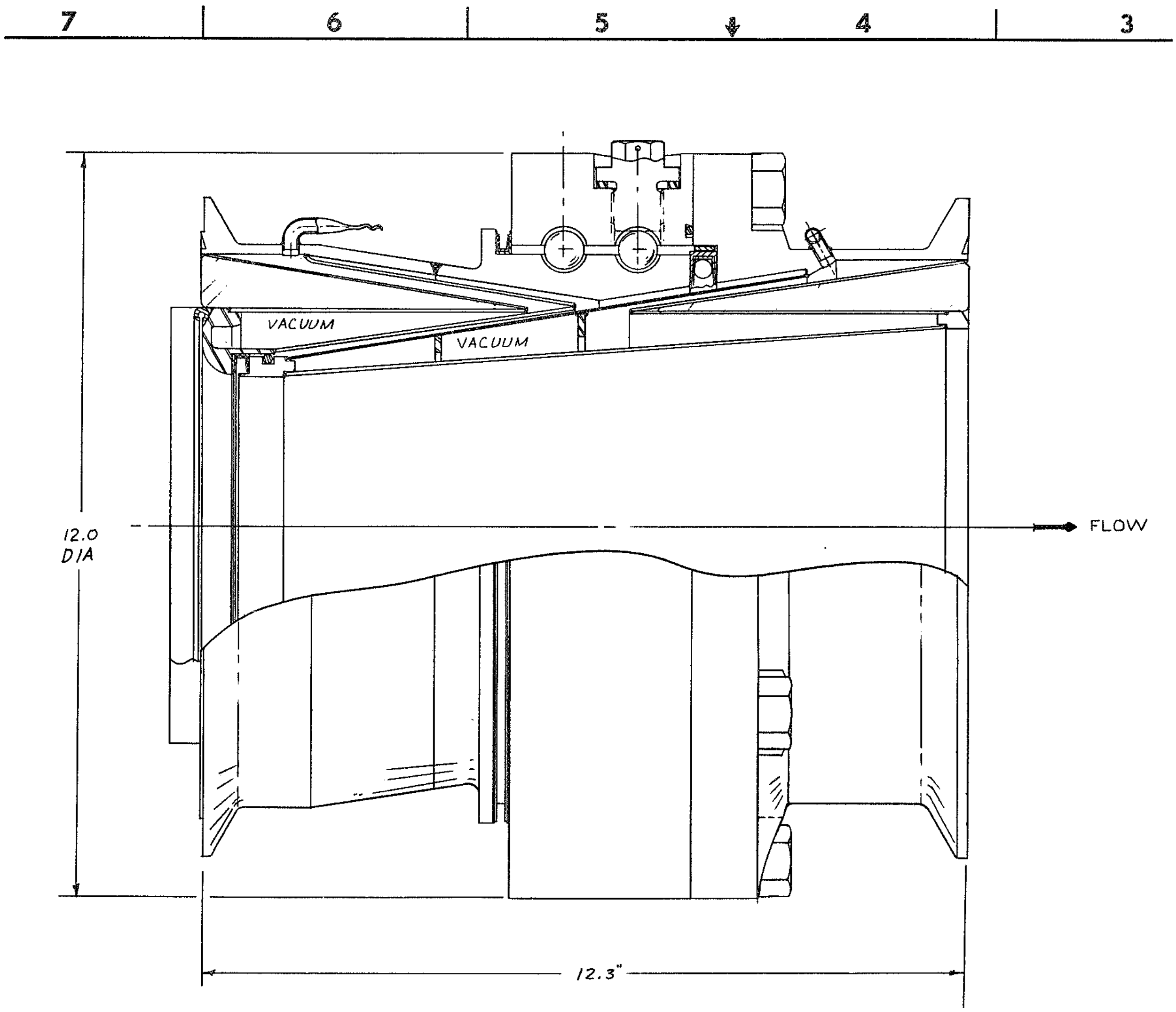


			ZONE	FIND NO.	CODE IDENT	PART NUMBER	NOMENCLATURE	SIZE	MATERIAL	SPECIFICATION
QTY REQD PER DASH NO.			PARTS LIST							

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES DEC TOL ANGULAR TOLERANCE X = ± .1 XX = ± .03 1/2° XXX = ± .010	CONTRACT NO. NAS-10-6098	STRAZA INDUSTRIES 790 GREENFIELD DR. EL CAJON, CALIFORNIA 92021
MATERIAL	PROJECT [signature] 5-1-69 DSGN ENGR STRESS CHECK DRAWN H. L. Wilber 5-1-69	TITLE FLEX VENT LINE, ROTARY UMBILICAL, FLANGE INTERFACE
MACH SURFACE FINISH: 125 RMS / MACH RAD: .030 DO NOT SCALE PRINT PARTS TO BE FREE OF BURRS & SHARP EDGES	4057	SIZE D / CODE IDENT NO. 03024 / DRAWING NO. SK4057-11 / REV SCALE — / WEIGHT / SHEET 1 OF 1

DASH NO.	QTY REQD	NEXT ASSY
APPLICATION		

SK 4057-11 Flex Line Rotary



			ZONE	FIND NO.	CODE IDENT	PART NUMBER	NOMENCLATURE	SIZE	MATERIAL	SPECIFICATION
QTY REQD PER DASH NO.			PARTS LIST							

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES

DEC	TOL	ANGULAR TOLERANCE
X	= ± .1	
XX	= ± .03	1/2°
XXX	= ± 010	

MATERIAL

MACH SURFACE FINISH: 125 RMS — MACH RAD: .030

DO NOT SCALE PRINT

PARTS TO BE FREE OF BURRS & SHARP EDGES

CONTRACT NO. NAS-10-6098

PROJECT	[signature]	5-1-69
DSGN ENGR		
STRESS		
CHECK		
DRAWN	H. L. Wilber	5.1.69

4057

STRAZA INDUSTRIES 790 GREENFIELD DR. EL CAJON, CALIFORNIA 92021

TITLE COUPLING, ROTARY, CRYOGENIC, 6" CONOSEAL

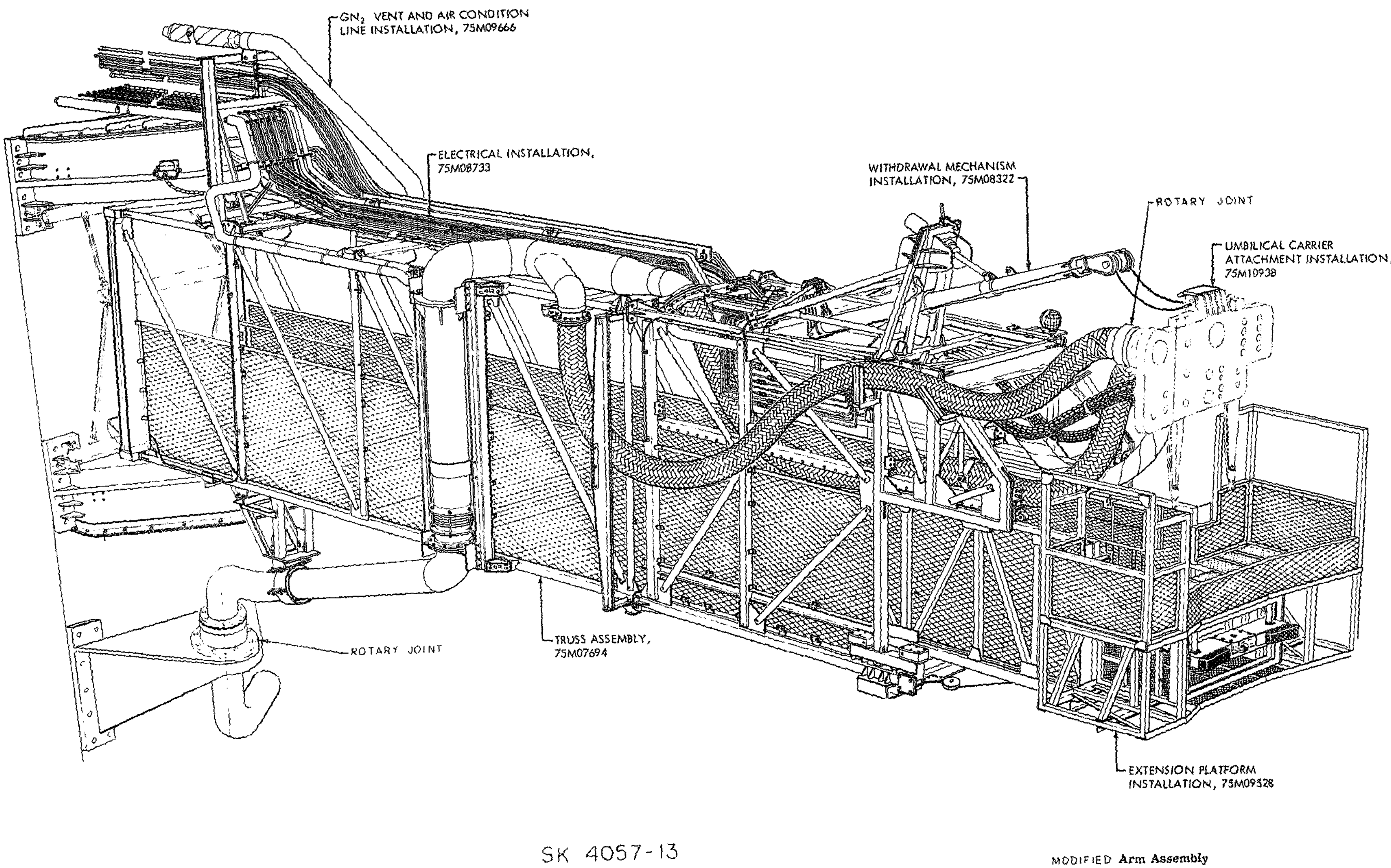
SIZE	CODE IDENT NO.	DRAWING NO.	REV
D	03024	SK 4057-12	

SCALE ~ WEIGHT SHEET 1 OF 1

A

DASH NO.	QTY REQD	NEXT ASSY
APPLICATION		

SK 4057-12 Conoseal Rotary



SK 4057-13 Arm Assembly

The rotary in SK 4057-13 is provided with an additional tower mounting flange. This flange can be added as an extension to either the upper or lower rotary flange, or as shown on the bearing ring. This flange is not provided on the test model used in this study since the optimum location would be a function of the rotary position in the actual installation.

It should be noted that the optimum mounting condition for the rotary is with the bearing housing up. Tests on standard bayonets (Ref. 2) have shown that if the nose seal (SK 4057-8) leaks liquid, it can fall by gravity into the higher temperature cavity of the bearing, creating a high heat leak. If the nose seal is down, this liquid will remain at the cold end of the bayonet section of the rotary.

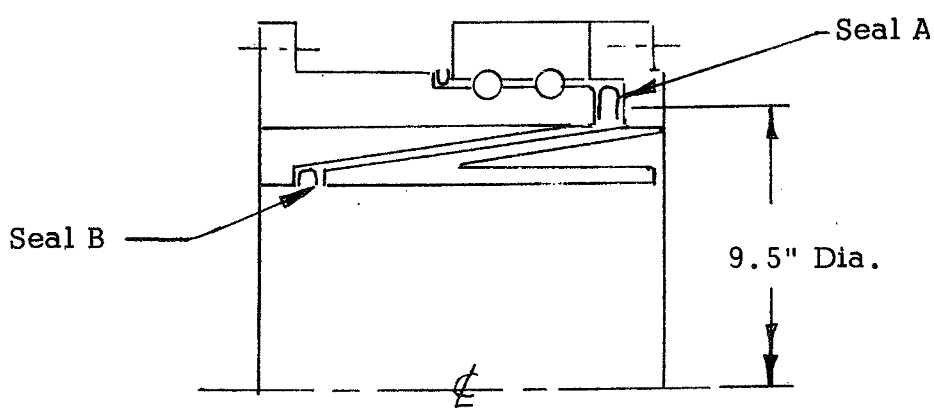
4.1.3.2 Analysis

The design criteria used to develop the Chiksan and the AMETEK/Straza jacketed rotaries is as outlined in Component Operating Requirements section. The first part of this analysis establishes the structural requirements for these units. The second part covers the heat leak analysis plus the cone structural analysis.

A. Structural

Ball bearing loads are based on carrying proof pressure of 285 psi in axial thrust and 30 g radial bending moment loads.

6-inch unit thrust loads:



Assume line proof pressure to exist at Seal A.

Proof Thrust Load = Proof Pressure p in psi x area as encompassed by seal.

$= p\ As$

$= 285 \frac{\pi}{4} (9.50)^2$

= 20,300 pounds proof thrust load or force

8-inch unit thrust load: (Seal A = 11.2 in. diameter)

$$0.785 \times (11.2)^2 = 98.4 \text{ sq. in.}$$

Thrust load = 98.4 sq. in. x 285 psi = 28,000 pounds

These pressure thrust loads provide a vibration or shock g load capability as follows: Assume 30 g and check mass.

G loading capability is based on the assumption that the proof force is equivalent to a mass W/g undergoing an impact or vibratory shock load of 30 g.

Force $\quad P = \frac{W}{g} \times 30\ g$

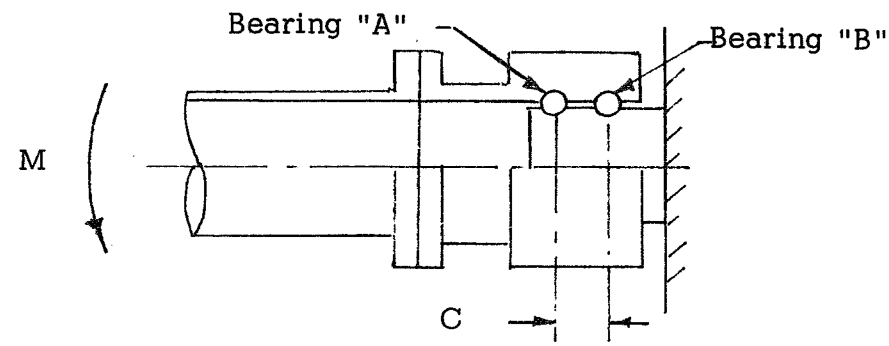
$$W = \frac{P}{30}$$

$$W = \frac{20,300}{30}$$

= 680 pounds equivalent weight

$$\frac{20,300 \text{ lbs g}}{30 \text{ g}} = 680 \text{ lbs (6-inch)}; \quad \frac{28,300}{30} = 935 \text{ lbs (8-inch)}$$

A 680-pound and 935 pound unsupported mass is a reasonable condition to expect in this system since the average weight of a typical hard line in the existing system is 354 pounds.



$M = 900$ ft-lbs $= 10,800$ in.-lbs

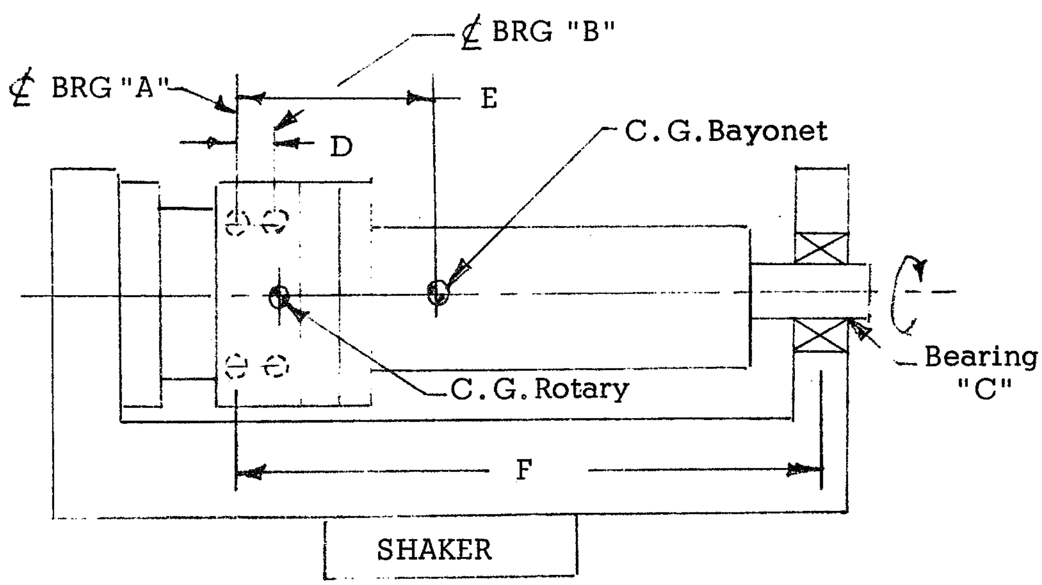
$C = 1.2$ (6-inch unit)

$= 2.25$ (8-inch unit)

Brg Couple Load $R_A = \frac{10,800 \text{ in.-lbs}}{1.2} = 9000$ lbs (6-inch unit)

$\frac{10,800 \text{ in.-lbs}}{2.25} = 4800$ lbs (8-inch unit)

These loads will be compared with those imposed on the unit during vibration tests. The test fixture will subject both the rotary and on-half of a bayonet to 30 g as shown below.



Approximate weight at bearing C.G.	=	112 (6-inch)
	=	140 (8-inch)
Approximate weight at bayonet C.G.	=	150 (6-inch)
	=	188 (8-inch)
Dimensions E and F are same for each unit	E =	20.0 in.
	F =	56.0 in.
Dimension D	=	1.2 (6-inch)
	=	2.25 (8-inch)

For 6-inch unit assume 1/2 bayonet is supported at Bearing C.

The radial reaction loads at Bearings A and B are resolved by summing moments about either Bearing A or B. Assuming Bearing A is the moment center and equating the algebraic sum of forces times the distance to Bearing A:

M_A about A = 0

$M_B = 1.2R_B$ where R_B is the radial reaction loat at Bearing B.

$1.2\ R_B$ - Weight "$W_{C.G}$" at bearing center of gravity x moment arm to "A" - 1/2 weight W_B at bayonet C.G. x moment arm to "A" = 0

$$1.2R_B - 1.2W_{C.G.} - 20 \times \frac{W_B}{2} = 0$$

$$\Sigma M_A = 112 \times 1.2 + \frac{150}{2} \times 20 = 1634 \text{ in.-lbs}$$

$$R_B = \frac{1}{1.2}\left(1.2W_{C.G.} + \frac{20}{2}W_B\right) = \frac{1}{1.2}\left(1.2 \times 112 + \frac{20}{2} \times 150\right)$$

$$= 1800 \text{ lbs}$$

Therefore, at 30 g the radial load on Bearing B $= \frac{1634}{1.2 \times g} \times 30g$

$$= 40{,}900 \text{ lbs}$$

For 8-inch Unit:

$$\Sigma M_A = 140 \times 2.25 + \frac{188}{2} \times 20 = 2194 \text{ in.-lbs}$$

$$\text{Bearing B Load} = \frac{2194}{2.25} = 975 \text{ lbs}$$

$$\text{At 30 g: } \frac{975}{g} \times 30\,g = 29,200 \text{ lbs}$$

The next step is to compare the actual loads to the bearing capabilities:

<u>6-inch Unit</u>: Thrust load component 20,300 lbs

Try 0.50 in. diameter balls on 10.00 pitch circle or 63 balls total balls assembled through outer race hole. Try race depth

$$h = 0.160d = 0.16 \times 0.50 = 0.080 \text{ in.}$$

for either inner or outer races.

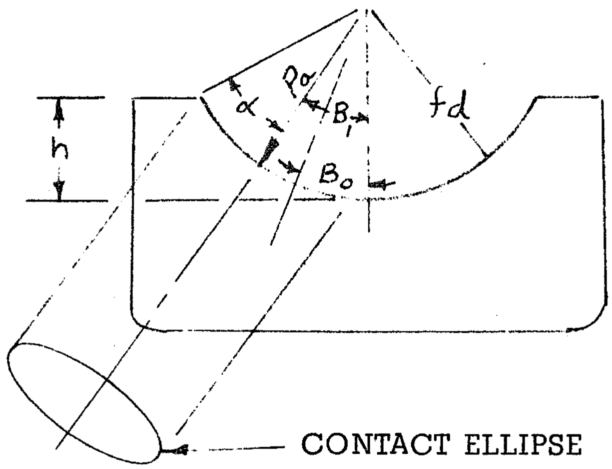
$$\frac{h}{d} = f \text{ vers} \left[\sin^{-1} \frac{0.0045944\, K_a\, K^{1/3} \left(\frac{\cos B_o}{\cos B_1} \right)^{1/2}}{f} + B_1 \right]$$

Reference: New Departure. Analysis of stresses and deflections Volume One, Pages 176 through 180.

Assume $B_o = 10°$ $\cos B_o = 0.98481$

Assume $B_1 = 23°$ $\cos B_1 = 0.92050$

$\sin B_1 = 0.39073$



Assume Curvatures

r_o = 0.57 in. outer race

r_i = 0.54 in. inner race

$B = 0.57 + 0.54 - 1$

$= 0.11$ Total Curvature

K = 369,000 from Chart 57, Volume II, for B = 0.11

$$\frac{d \text{ Cos } B_1}{E} = \frac{0.50 \times 0.90631}{9.0} = 0.050$$

$\therefore K_a$ = 2.20 for inner race from Chart 47 for f_i = 54%

And

$$\frac{d \text{ Cos } B_1}{E} = 0.050$$

$$\frac{h}{d} = 0.54 \text{ vers}\left[\text{Sin}^{-1} \frac{0.0045944 \times 2.20\,(0.369 \times 10^6)^{1/3}\left(\frac{0.98481}{0.92050} - 1\right)^{1/2}}{0.54} + 23°\right]$$

$= 0.1501$

This is less than 0.160, the assumed value for h/d. ∴ We will use $B_1 = 23°$ for the limiting value.

$$T_L = \eta d^2 \; K \sin B_1 \left(\frac{\cos B_o}{\cos B_1} - 1 \right)^{3/2} = 63\,(0.50)^2 \times 369{,}000$$

$$\times \; 0.39073 \left(\frac{0.98481}{0.92050} - 1 \right)^{3/2} = 41{,}900 \text{ lbs}$$

allowable static load in thrust for 0.50 in. diameter balls (63) on 10.0 diameter pitch circle.

Since the actual load is 20,300 pounds, this is a reasonable size for this bearing. The actual unit as proposed by Chiksan contains 304 cres body and type 440C balls. Their unit is shown on Specification Control Drawing 8-040352 on page 30.

8-inch Unit:

Thrust load component = 28,000 pounds
Radial load component = 29,200 pounds

The rated thrust load on the KD 120 x P Kaydon Bearing is 26,700 pounds. The low rpm at which this unit is running allows us to use this static load rating as shown on Table I. By making this bearing compliant as shown in SK 4058-9, we can expect a 2.5 to 3 x increase in the radial load carrying capacity

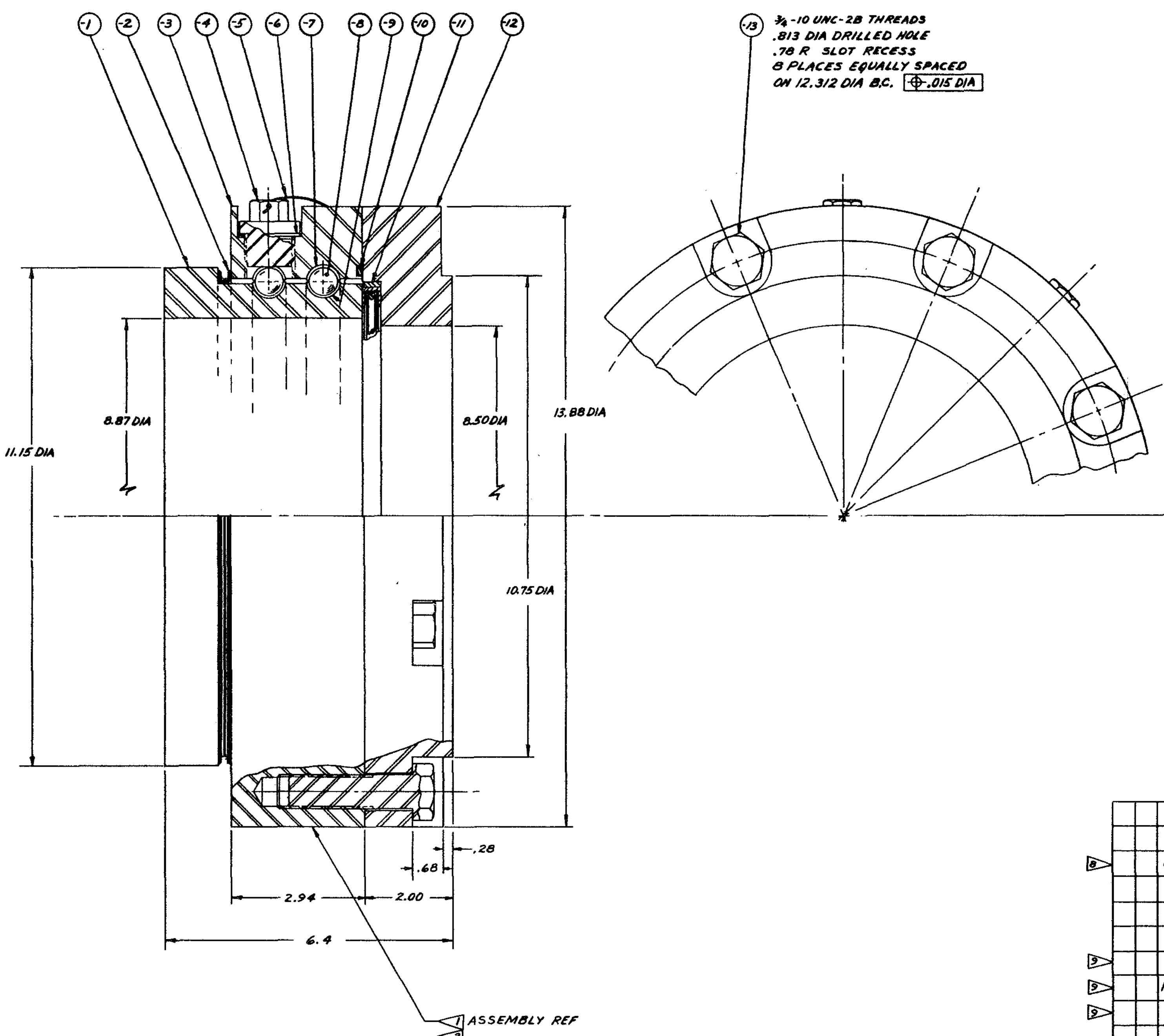
This would give us a minimum capacity of:

Radial	10,600 x 2.5 = 26,500 pounds
Thrust	26,700 x 2.5 = 66,500 pounds

These loads are close enough to the applied loads to make this a reasonable size bearing for the application. Data on the compliant bearing design was taken from Ref. 4.

The radial capability of the 6-inch unit is based on the ball load as calculated from the following equation taken from Page 79 of Ref. 3.

$$V_o = \frac{4.452 \times 40{,}000}{63} = 2800 \text{ pounds}$$



SPECIFICATION CONTROL DRAWING

1 SINGLE SOURCE ITEM

8	8		13	71454	-13	CAP SCREW, HEX HD	¾-10 NC2 X 2 ¾	SST, 18-8	3199333-13
	1		12		-12	FLANGE, COUPLING		SST, 316	-12
	1		11		-11	SEAL, PRESSURE	(SST & KEL-F)	312315)	-11
	1		10		-10	PACKING, PREFORMED	(SYNTHETIC, COMP NO. 6W)	3102302)	-10
9	2		9		-9	BALL RACE, SNAP-IN, MALE	(SST, 301)	3102099)	-9
9	104		8		-8	BALL	.625 DIA	SST, 440C	-8
9	2		7		-7	BALL RACE, SNAP-IN, FEMALE	(SST, 301)	3103071)	-7
	2		6		-6	WASHER, BALL PLUG		SST, 18-8	-6
	1		5		-5	WIRE, LOCK		SST, 18-8	-5
	2		4		-4	PLUG, BALL RETAINER	(SST, 304)	3188961)	-4
	1		3		-3	COUPLING		SST, 316	-3
	1		2		-2	SEAL, WATER	(SST & TEFLON	3102330)	-2
	1		1	71454	-1	NIPPLE, SWING		SST, 316	3199333-1
	QTY REQD PER ASSY	ZONE	FIND NO.	CODE IDENT	PART NUMBER	NOMENCLATURE	DESCRIPTION, SIZE	REF MATERIAL	SPECIFICATION 1

PARTS LIST

10. ALL STAINLESS STEEL PARTS TO BE PASSIVATED.

9 BALL RACES & BALLS ARE TO BE COATED WITH TEFLON DISPERSION PER CHIKSAN DIV. ENG. INSTRUCTION (MAY 9, 1967) REF.

8 THREADS OF CAPSCREWS TO BE DRY FILM LUBRICATED (LUBECO PROCESS N 350 A REF).

7. ASSEMBLY TO BE CAPABLE OF WITHSTANDING A MAX BENDING MOMENT OF 50,000 IN. LBS.

6. OPERATING TEMPERATURE RANGE: 0°F TO +150°F.

5. PROOF PRESSURE: 285 PSI.

4. OPERATING PRESSURE: 190 PSI.

3. OPERATION SERVICE MEDIUM: LIQUID HYDROGEN AND OXYGEN.

2 ASSEMBLY TO BE DIS-ASSEMBLED BY AMETEK/STRAZA FOR MACHINING & JACKETING & NEXT-ASSEMBLY INTEGRATION.

1 SINGLE SOURCE ITEM. AVAILABLE ONLY FROM CHIKSAN DIV, FMC CORP. (CODE 71454) BREA, CALIF, AS PART NO. 3199333 (& APPLICABLE DASH NO.)

NOTES:

DASH NO.	QTY REQD	NEXT ASSY
-2 THRU -11 & -13	ALL	8-100098
-12	1	8-040353
-1	1	8-040354
APPLICATION		

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES
DEC TOL: X = ±.1; XX = ±.03; XXX = ±.010. ANGULAR TOLERANCE 1/2°

MATERIAL 1

MACH SURFACE FINISH: 125 RMS — MACH RAD: .03

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4057

AMETEK / Straza
790 GREENFIELD DRIVE, EL CAJON, CALIFORNIA

TITLE: COUPLING, ROTARY, CRYOGENIC, FLANGED, SPECIAL

SIZE E — CODE IDENT NO. 03024 — DRAWING NO. 8-040352 — REV

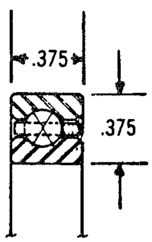
SCALE 1/1 — WEIGHT — SHEET 1 OF 1

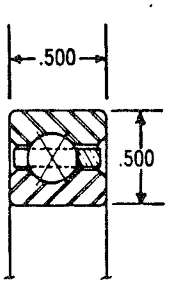
Chiksan Specification Drawing No. 8-040352

TABLE I

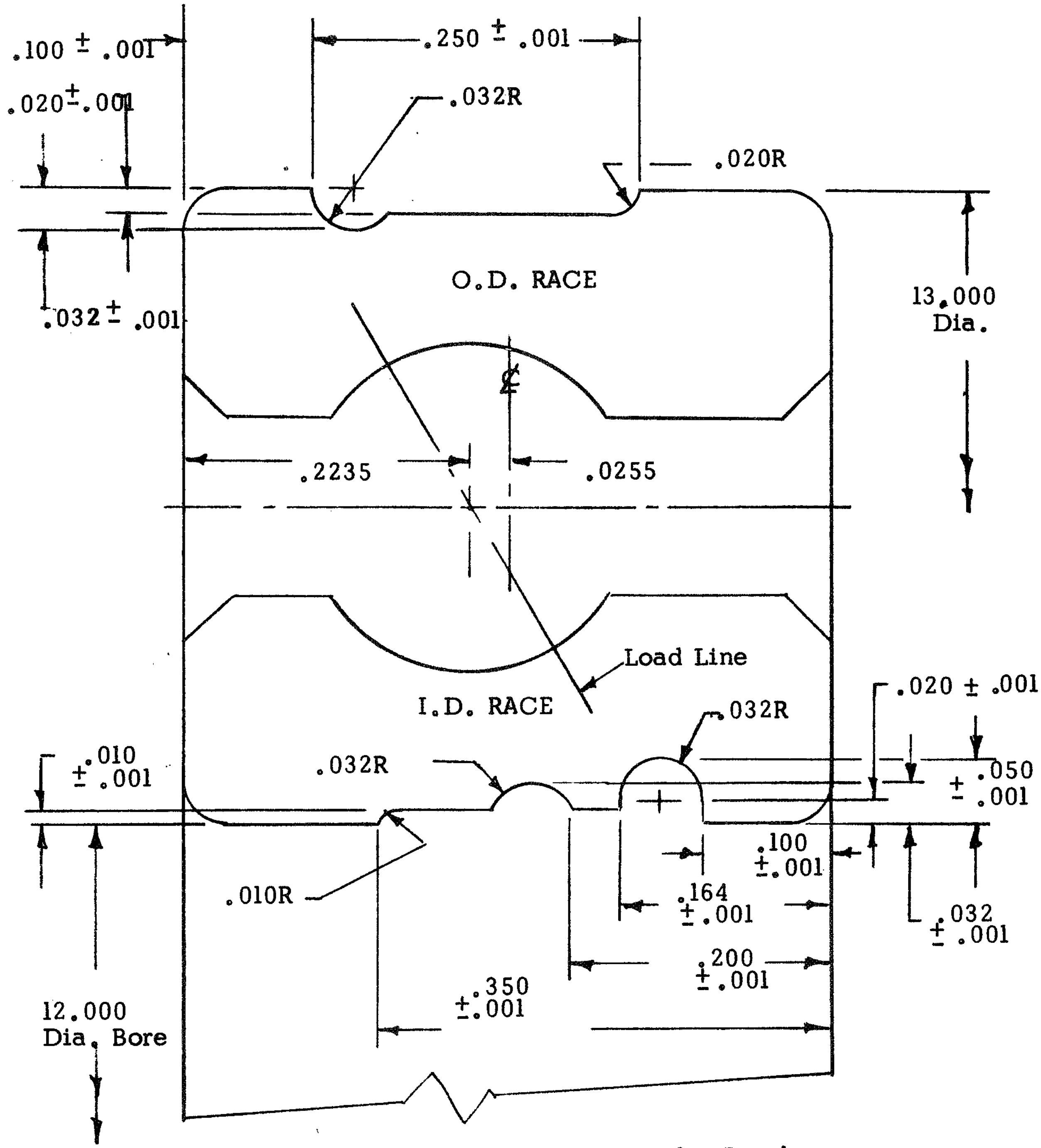
"XP" Kaydon Reali-Slim four-point contact ball radial bearings (Continued)

(Shaded sizes are available from stock.)

KC SERIES	KAYDON Bearing Number	Dimensions in Inches		Radial Capacity in Pounds		Thrust Capacity in Pounds		Weight in Pounds
		Bore	Outside Diameter	Static	100 rpm*	Static	100 rpm*	
⅜" width • 3/16" balls	KC40XP	4.000	4.750	2,800	750	7,000	1,850	.45
	KC42XP	4.250	5.000	2,950	780	7,400	1,950	.47
	KC45XP	4.500	5.250	3,100	800	7,800	2,000	.48
	KC47XP	4.750	5.500	3,250	830	8,200	2,050	.50
	KC50XP	5.000	5.750	3,450	860	8,600	2,150	.58
	KC55XP	5.500	6.250	3,750	910	9,400	2,250	.59
	KC60XP	6.000	6.750	4,050	960	10,200	2,400	.63
	KC65XP	6.500	7.250	4,400	1,000	11,000	2,500	.68
	KC70XP	7.000	7.750	4,700	1,050	11,800	2,650	.73
	KC75XP	7.500	8.250	5,050	1,100	12,600	2,750	.78
	KC80XP	8.000	8.750	5,350	1,150	13,400	2,900	.84
	KC90XP	9.000	9.750	6,000	1,250	15,000	3,100	.94
	KC100XP	10.000	10.750	6,650	1,300	16,600	3,300	1.06
	KC110XP	11.000	11.750	7,300	1,400	13,200	3,550	1.16
	KC120XP	12.000	12.750	7,900	1,500	19,800	3,750	1.25
	KC140XP	14.000	14.750	9,200	1,650	23,000	4,150	1.52
	KC160XP	16.000	16.750	10,500	1,800	26,200	4,500	1.73
	KC180XP	18.000	18.750	11,700	1,950	29,400	4,900	1.94
	KC200XP	20.000	20.750	13,000	2,100	32,600	5,250	2.16
	KC250XP	25.000	25.750	16,200	2,400	40,600	6,050	2.69
	KC300XP	30.000	30.750	19,500	2,700	48,600	6,850	3.21

KD SERIES	KAYDON Bearing Number	Dimensions in Inches		Radial Capacity in Pounds		Thrust Capacity in Pounds		Weight in Pounds
		Bore	Outside Diameter	Static	100 rpm*	Static	100 rpm*	
½" width • ¼" balls	KD40XP	4.000	5.000	3,800	1,100	9,600	2,800	.78
	KD42XP	4.250	5.250	3,950	1,150	9,950	2,850	.83
	KD45XP	4.500	5.500	4,250	1,200	10,600	3,000	.88
	KD47XP	4.750	5.750	4,400	1,200	11,000	3,050	.94
	KD50XP	5.000	6.000	4,700	1,250	11,700	3,200	1.00
	KD55XP	5.500	6.500	5,100	1,350	12,800	3,400	1.06
	KD60XP	6.000	7.000	5,550	1,400	13,800	3,550	1.16
	KD65XP	6.500	7.500	5,950	1,500	14,900	3,750	1.22
	KD70XP	7.000	8.000	6,400	1,550	16,000	3,950	1.31
	KD75XP	7.500	8.500	6,800	1,650	17,100	4,100	1.41
	KD80XP	8.000	9.000	7,250	1,700	18,100	4,250	1.53
	KD90XP	9.000	10.000	8,100	1,850	20,300	4,600	1.72
	KD100XP	10.000	11.000	8,950	1,950	22,400	4,900	1.88
	KD110XP	11.000	12.000	9,800	2,100	24,500	5,250	2.06
	KD120XP	12.000	13.000	10,600	2,200	26,700	5,550	2.25
	KD140XP	14.000	15.000	12,300	2,450	30,900	6,100	2.73
	KD160XP	16.000	17.000	14,100	2,650	35,200	6,650	3.10
	KD180XP	18.000	19.000	15,800	2,850	39,500	7,200	3.48
	KD200XP	20.000	21.000	17,500	3,050	43,800	7,700	3.85
	KD250XP	25.000	26.000	21,800	3,550	54,500	8,900	4.79
	KD300XP	30.000	31.000	26,000	4,000	65,100	10,000	5.73

*Dynamic capacities are based on 600 hours minimum life.



SK 4057-9 Compliant Kaydon Bearing

Sketch of a typical compliant bearing, showing relieved inside and outside diameters, to increase ball contact areas under load.

SK 4057-9

The maximum allowable load: (Page 175, Ref. 3)

$$\Sigma Vm = 5631\ hd^2 = 5631 \times 63 \times 0.5^2 = 89{,}000 \text{ pounds}$$

Thread stress check in 8-inch unit:

To prevent failure of the thread at proof load, the stress will be checked at 760 psi.

Pressure area: $0.785 \times (11.25)^2 \cong 100$ sq. in.

Thrust load: $F = 100 \times 760 = 76{,}000$ lbs.

Stress area: $A = 12\pi \times 1 \div 2 = 18.9$ sq. in.

Stress: $S_s = \frac{F}{A} = \frac{76{,}000}{18.9} = 4{,}000$ psi

Flange bolt sizing for 8-inch unit:

The maximum load on the bearing flange bolts is at the burst pressure of 760 psi.

Thrust load $= 100 \times 760 = 76{,}000$ pounds

$$\frac{76{,}000}{12 \text{ bolts}} = 6{,}500 \text{ pounds/bolt}$$

From MIL-B-857A use 5/8-11 UNC studs which for 316 stailness steel is good for 6,770 pounds.

In order to establish an approximate torsional load, the following analysis is performed. The conditions are assumed to be under normal loading during arm retract while purging through an 8-inch unit.

Moment = 900 ft-lbs = 10,800 in.-lbs

Pressure = 30 psig

Vibration g loads small

Bearing radial load = 4,800 pounds

Friction torque for radial load is

$$F_R = 2 \text{ brgs} \times 0.003 \times \frac{6.2}{12} \times 4{,}800 = 14.8 \text{ ft-lbs}$$

Where: Coefficient of friction = 0.003

Ball race radius = 6.2 in.

Friction torque for thrust load is

$$F_t = 0.003 \times \frac{6.2}{12} \times 100 \times 30 = 4.6 \text{ ft-lbs}$$

Seal friction:

$$F_s = 0.08 \times \frac{6.2}{12} \times 100 \times 30 = 124.0 \text{ ft-lbs}$$

Assume 10 ft-lb bearing preload = 10.0 ft-lbs

Total torsional load = 153.4 ft-lbs

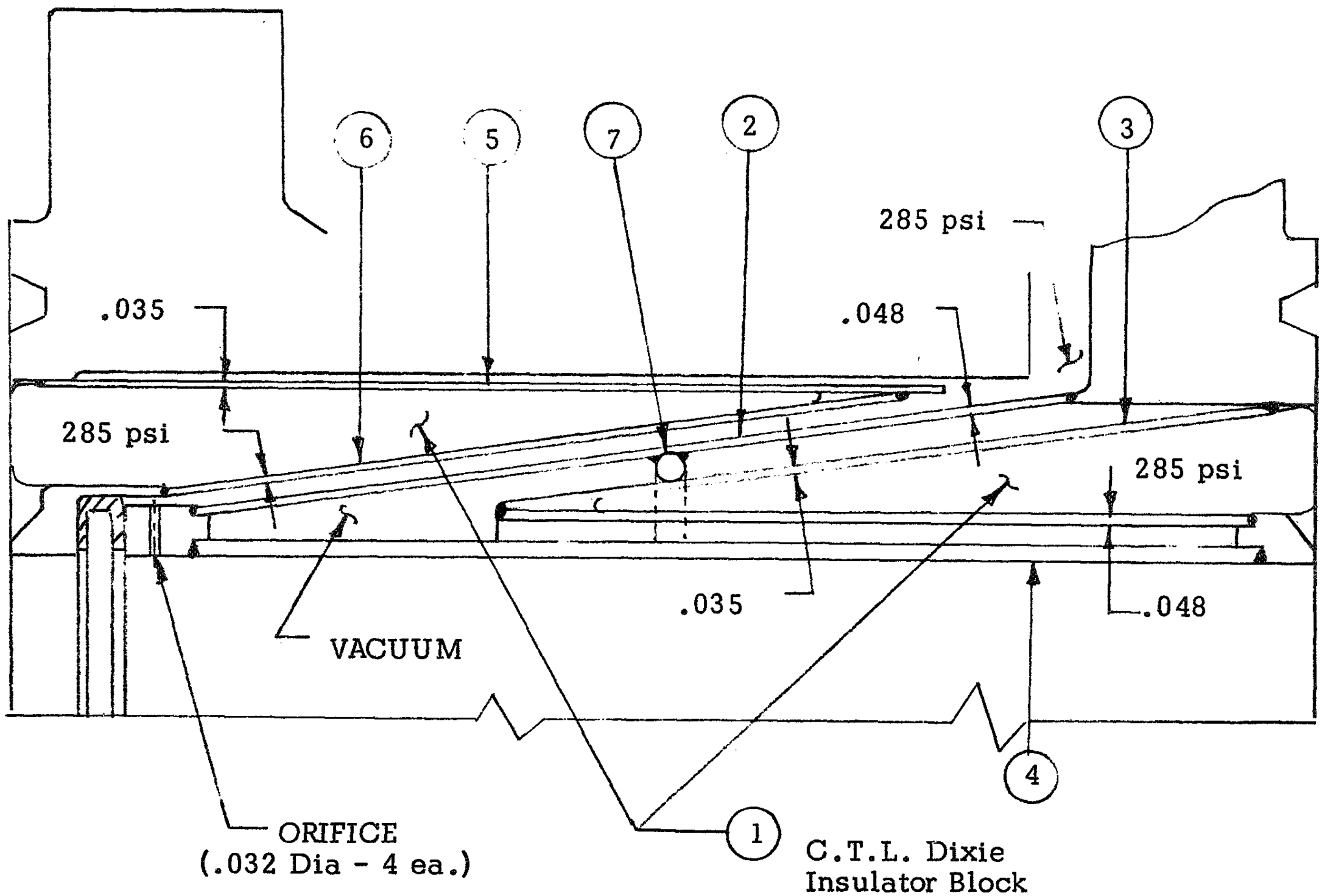
This is within our 600 ft-lb design goal.

B. Heat Leak (LO_2):

The method used to maintain a low heat leak for the rotary joints employs the design concept developed for the cone type cryogenic coupling. This allows for direct mate-up of a typical cryogenic coupling to a rotary.

The heat leak for the LO_2 system is based on using a fiberglass layup anti-convection block similar to that developed by CTL Dixie for Masoneilan. This material has an apparent thermal conductivity of 0.144 Btu - in/hr - ft^2 - °F. Since this material is allowed to breathe, the block conductivity includes that of the GO_2. This material was compounded to be compatible with LO_2. Table II presents the heat leak comparison for the 6-inch and 8-inch rotary plus the design using the evacuated titanium diaphragm concept. Since the titanium diaphragm heat leak is almost the same as for the block, this concept is not justified, mainly due to the dissimilar metal welding or brazing problems.

The following sketch shows the optimum material thickness as established for the 6-inch diameter rotary joint. The 8-inch diameter unit will differ only in that item 4 wall thickness will be increased to 0.060 inches. The stiffner ring (item 7) is welded to the I.D. of cone 2 for collapse pressure support. It is used on both the 6-inch and 8-inch diameter rotaries.



SK 4057-14 Heat Leak Item Numbers

HEAT LEAKS OF 6" & 8" ROTARY
JOINT CONE & DIAPHRAGM CONCEPTS

TABLE II

Tabulated heat leak comparisons - optimized inner cone and outer cyl. thickness.

Cone Concept

Find No.	1	2	3	4		5	6	Total B/Hr LO_2
Line Size	A.D. Little Block (2)ea	Center Cone .048" Th	Inner Cone .0355" Th	Inner Cyl. .048 Th.	.060 Th	Outer Cyl. .0355" Th	Outer Cone .0355" Th	
6"	159.8	42.5	19.0	23.0		19.0	18.4	281.7
8"	196.0	53.8	23.0		39.0	22.7	20.6	355.1

Diaphragm Concept

Find No.	1	2	3	4	5	Total Btu/Hr LO_2
Line Size	A.D. Little Block (2)ea	Center Cone	Outer Cone	Cuter Cyl	Diaph	
6"	141.6	42.5	18.4	19.0	38.6	260.1

See Sketch SK4057-14 for Find Number Identification.

Analysis Symbols

D = Diameter

E = Modulus of Elasticity PS1

F_{TY} = Allowable yield stress of material

K = Thermal Conductivity $\frac{Btu - ft}{hr - ft^2 - °F}$

L = Length

P_{EXT} = Critical Pressure on External Surface

P_{INT} = Critical Pressure on Internal Surface

q = Heat Leak $\frac{Btu}{hr}$

Q = Total Heat Leak $\frac{Btu}{hr}$

r_a = Outer Radius

r_b = Inner Radius

t = Thickness

T_a = Temperature of Warm Side (°F)

T_b = Temperature of Cold Side (°F)

$T_a - T_b$ = Differential Temperature of Subscript

6 - 2 indicates the 6-inch diameter rotary cone No. 2. See SK 4057-14

A.D.L. = A. D. Little Company

Center Cone ② Sizing for 285 psi external proof pressure

Try 21-6-9 CRES with modulus of elasticity $E = 29 \times 10^6$ psi.

For 6-inch IPS Line

$$Pcr = \frac{2.60E\left(\frac{t}{D}\right)^{5/2}}{\frac{L}{D} - 0.45\left(\frac{t}{D}\right)^{1/2}}$$

Try t = 0.048

D = 7.95 in. mean diameter of cone

$E = 29 \times 10^6$

L = 3.12 in. lt. to rib

$$Pcr = \frac{2.60 \times 29 \times 10^6 \left(\frac{0.048}{7.95}\right)^{5/2}}{\frac{3.12}{7.95} - 0.45\left(\frac{0.048}{7.95}\right)^{1/2}} = 596 \text{ psi, OK for center cone.}$$

Ref: ASME — Pressure Vessel & Piping Design Collected Papers
1927 to 1959, Page 614

Inner Cylinder ④

L = 5.30 in.

D = 7.00 in.

t = 0.048

$$Pcr = \frac{2.60 \times 29 \times 10^6 \left(\frac{0.048}{7.00}\right)^{5/2}}{\frac{5.30}{7.00} - 0.45\left(\frac{0.048}{7.00}\right)^{1/2}} = 408 \text{ psi OK}$$

Cone ② Heat Leaks — Cryogen LO_2 with ambient temperature assumed to be 100°F

$$q_2 = 2\pi K t\left[1 + \left(\frac{r_a - r_b}{L}\right)^2\right]^{1/2} \times \left(\frac{r_a + r_b}{2L}\right)\left(T_a - T_b\right)$$

K = 7.0 Btu hr-ft-°F for 304 CRES at mean temperature of -161°F

Ref: Cryogenic Materials Data Handbook, Page B4V

$r_a = 4.35$ in.

$r_b = 3.50$ in.

$T_a = 100°F$

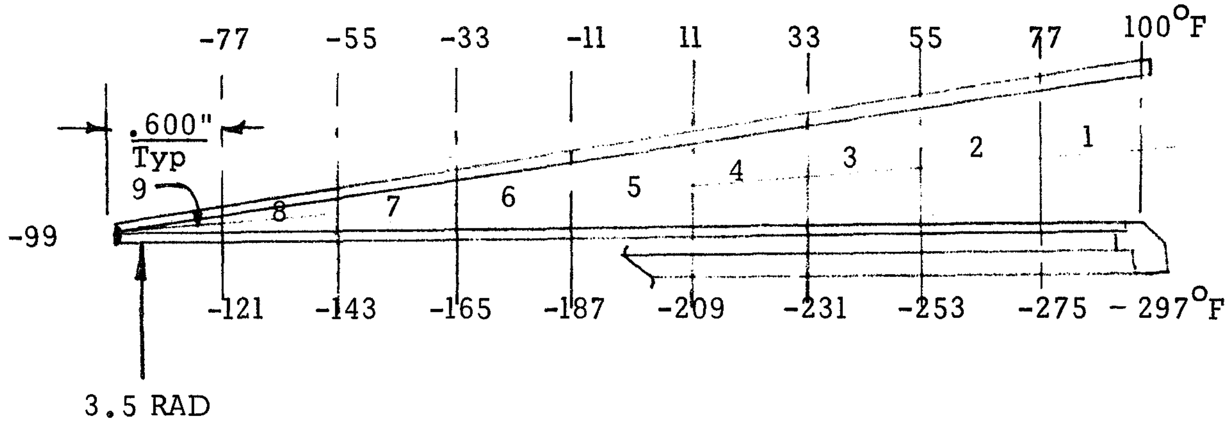
$T_b = 297°F$

$$q_2 = 2\pi \times 7.0 \times \frac{0.048}{12}\left[1 + \left(\frac{4.35 - 3.50}{6.5}\right)^2\right]^{1/2} \times \left(\frac{4.35 + 3.50}{2 \times 6.5}\right)\left[100 - (-297)\right]$$

= 42.5 Btu/hr at an ambient temperature of 100°F middle cone 2 = cone 6

<u>6-inch Rotary joint heat leak inner cones filled with A.D.L. Block</u>.

Assume flange temperature T_a = 100°F at cone & flange junction



<u>Heat leak thru A.D.L. Block</u> — assuming the thermal conductivity of the material to be 0.144 Btu/ft²-hr-°F/in.

$$k_F = \frac{0.144}{12}$$

$$= \frac{\text{Btu-ft}}{\text{ft}^2\text{-hr-°F}}$$ This value taken from bayonet report as quoted by A. D. Little

$$\Sigma Q = k_F \, \Sigma \frac{A}{L} \, \Delta T$$

6-Inch Rotary joint inner cone ③ 0.0355 in. and Inner cylinder ④ 0.048 in. Heat Leaks

Inner ③ Cone (0.0355 in. with internal pressure)

$r_a = 4.30$ in.

$r_b = 3.56$ in.

$L = 5.4$

$$s = \frac{Pr}{t} = \frac{285 \times 3.95}{0.035} = 32,000 \text{ psi}$$

$$q = -2\pi \times 7.0 \times \frac{0.035}{12}\left[1 + \left(\frac{4.30 - 3.56}{5.4}\right)^2\right]^{1/2} \left(\frac{4.30 + 3.56}{2 \times 5.4}\right)\left[-99 - (+100)\right]$$

$$= 19.0 \frac{\text{Btu}}{\text{hr}}$$

Inner ④ Cylinder (5.3 in. long)

$$q = 7.0\left(\frac{\pi \times 7.0 \times 0.048}{12 \times 12}\right) \times \frac{12}{5.30} \times \left[-99 - (-297)\right] = 23.0 \frac{\text{Btu}}{\text{hr}}$$

Mean Values

Section	Radial Area (sq. ft.)	Radial Length (ft.)	Temperature (° F)	Mean Heat Leak Q Btu/hr $= k \frac{A}{L} \Delta T$
1	.101	.058	397	8.30
2	.099	.050	352	8.35
3	.098	.046	308	7.88
4	.097	.038	264	8.06
5	.096	.032	220	7.92
6	.095	.025	176	8.03
7	.097	.018	132	8.27
8	.093	.010	98	10.94
9	.092	.004	44	12.14

Heat leak through A.D.L. Block $q = 79.9 \frac{\text{Btu}}{\text{Hr}}$

Assume outer block to be approximately equal inner block heat leak

6-inch Rotary joint outer cone 6 0.060 in. & outer cylinder 5 0.0355 in.

$r_a = 4.30$

$r_b = 3.68$

$L = 5.6$ in.

$$q = -2\pi \times 7.0 \times \frac{0.0355}{12}\left[1 + \left(\frac{4.30 - 3.68}{5.6}\right)^2\right]^{1/2}\left(\frac{4.30 + 3.68}{2 \times 5.6}\right)\left[-297 - (-99)\right]$$

$$= 18.4 \ \frac{\text{Btu}}{\text{hr}} \quad \text{cone heat leak, outer cone 6}$$

$$q = 7.0\left(\frac{\pi \times 8.7 \times 0.0355}{12}\right) \times \frac{1}{5.9}\left[-99 - (+100)\right] = 19.0 \ \frac{\text{Btu}}{\text{hr}} \quad \text{cylinder heat leak, outer cylinder 5}$$

Noting that both inner and outer cones have approximately the same heat leaks, it can be assumed that the A.D.L. Block has similar heat leaks.

$$Q = \underset{2}{42.5} + 2 \times \underset{1}{79.9} + \underset{3}{19.0} + \underset{4}{23.0} + \underset{6}{18.4} + \underset{5}{19.0} = 281.7 \ \frac{\text{Btu}}{\text{hr}}$$

<u>8-inch Rotary</u>

Size center cone 2 for 285 psi exterior proof pressure. Try

$t = 0.048$ in. 21 - 6 - 9 CRES

$D = 9.80$ in. mean cone diameter

$L = 6.40$ in. axial length

$$Pcr = \frac{2.60 \times 29 \times 10^6 \left(\frac{0.048}{9.80}\right)^{5/2}}{\frac{3.2}{9.8} - 0.45\left(\frac{0.048}{9.80}\right)^{1/2}} = 429 \text{ psi} \quad > 285 \text{ psi proof pressure}$$

Size inner cylinder

$D = 9.0$ in.

$L = 5.10$ in.

$t = 0.048$ in.

$$Pcr = \frac{2.60 \times 29 \times 10^6 \left(\frac{0.048}{9.00}\right)^{5/2}}{\frac{5.1}{9.00} - 0.45 \left(\frac{0.048}{9.00}\right)^{1/2}} = 293 \text{ psi}$$

Center cone (2) heat leak

r_a = 5.35 in. major radius of cone

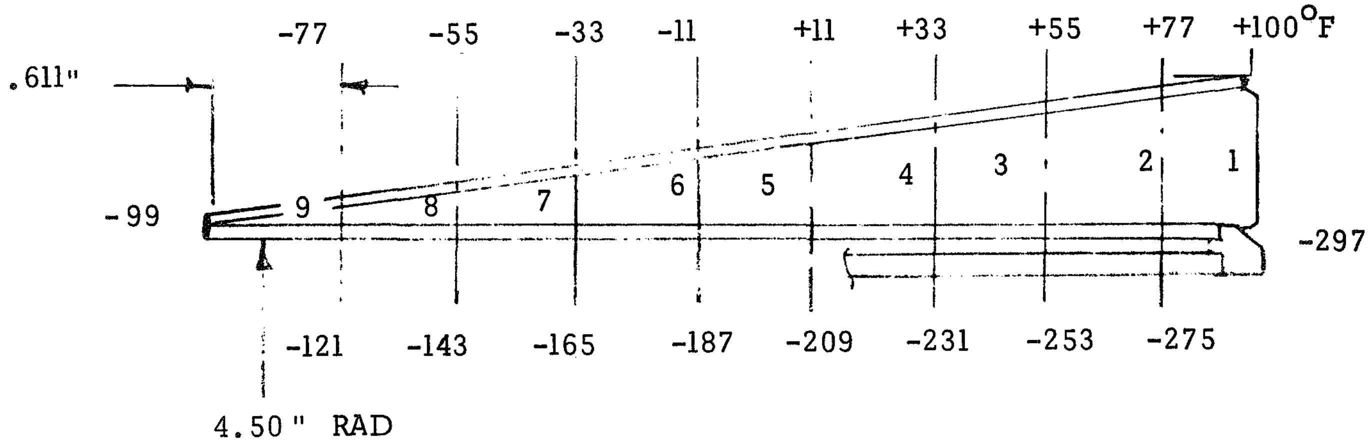
r_b = 4.50 in. minor radius of cone

L = 6.4 in.

$$q = 2\pi \times 7.0 \times \frac{0.048}{12} \left[1 + \left(\frac{5.35 - 4.50}{6.4}\right)^2\right]^{1/2} \left(\frac{5.35 + 4.50}{2 \times 6.4}\right) \left[100 - (-297)\right]$$

$$= 53.8 \frac{\text{Btu}}{\text{hr}}$$

A.D.L. Block heat leak, inner block (1)



Radial Area $A = \frac{0.611}{12} \times \frac{2\pi r}{12} = 0.02665r$

Mean Values

Section	Radial Area (sq. ft.)	Radial Length (ft.)	Temperature (°F)	Mean Heat Leak Q Btu/hr $= 0.012 \frac{A}{L} \Delta T$
1	.129	.068	397	9.0
2	.128	.052	352	10.4
3	.127	.044	308	10.7
4	.126	.038	264	10.5
5	.125	.032	220	10.3
6	.124	.025	176	10.5
7	.123	.018	132	10.8
8	.122	.011	98	13.0
9	.121	.005	44	12.8

TOTAL 98.0

A.D.L. Block Heat Leak

$$q = 98.0 \frac{\text{Btu}}{\text{hr}}$$

Outer Block $\cong$ Inner Block Heat Leak

8-inch Rotary joint inner cone (3) 0.0355 in. & Inner Cylinder (4) 0.060 in. Heat Leak

Inner Cone (3)

$$r_a = 5.32 \text{ in.}$$

$$r_b = 4.55 \text{ in.}$$

$$L = 5.50 \text{ in.}$$

$$s = \frac{pr}{t} = \frac{300 \times 5.0}{0.0355} = 42{,}000 \text{ psi} \leq 60{,}000 \text{ psi for 21-6-9 steel}$$

$$q = -2\pi \times 7.0 \times \frac{0.0355}{12} \left[1 + \left(\frac{5.32 - 4.55}{5.50}\right)^2\right]^{1/2} \left(\frac{5.32 + 4.55}{2 \times 5.50}\right) \left[-99 - (+100)\right]$$

$$= 23.0 \frac{\text{Btu}}{\text{hr}}$$

Inner Cylinder (4) 5.1 in. long x 0.060 in. thick material

$$q = 7.0 \left(\frac{9 \times 0.06 \times \pi}{12 \times 12}\right) \times \frac{12}{5.1} \left[-99 - (-297)\right] = 39.0 \frac{\text{Btu}}{\text{hr}}$$

8-inch Diameter Outer Cone (6) 0.0355 in. & Outer Cylinder 0.0355 in. (5)

Outer Cone

$r_a = 5.37$ in.

$r_b = 4.65$ in.

$L = 6.30$ in.

$$q = 2\pi \times 7.0 \times \frac{0.0355}{12}\left[1 + \left(\frac{5.37 - 4.65}{6.30}\right)^2\right]^{1/2}\left(\frac{5.37 + 4.65}{2 \times 6.30}\right)\left[-99 - (-297)\right]$$

$$= 20.6\ \frac{\text{Btu}}{\text{hr}}$$

Outer (5) Cylinder 5.70 in. length x 0.0355 in.

$$q = \frac{7.0\pi}{12 \times 12}(11.0 \times 0.0355) \times \frac{12}{6.30}\left[-99 - (+100)\right] = 22.7\ \frac{\text{Btu}}{\text{hr}}$$

6-inch Diameter Rotary Joint Titanium Diaphragm Heat Leak with LO_2

$$q = k\frac{A}{L}\Delta T$$

$$k = 3.50\ \frac{\text{Btu - ft}}{\text{hr - ft}^2 - {}^\circ\text{F}}$$

$$A = \frac{0.012 \times 3.90 \times 2\pi \times 2}{144} = 4.08 \times 10^{-3}\ \text{ft}^2$$ mean area of 2 diaphragms

$$L = \frac{56}{12} = 0.147\ \text{ft}$$

$$\Delta T = 397^\circ\text{F}$$

$$q = 3.50 \times \frac{4.08 \times 10^{-3}}{0.147} = 38.6\ \frac{\text{Btu}}{\text{hr}}$$ heat leak of 2 diaphragms

Total Heat Leak

Q = q center cone + q outer cone & cylinder + q teflon foam

+ q diaphragm = 42.5 + (18.4 + 19.0) + (70.8 + 70.8)

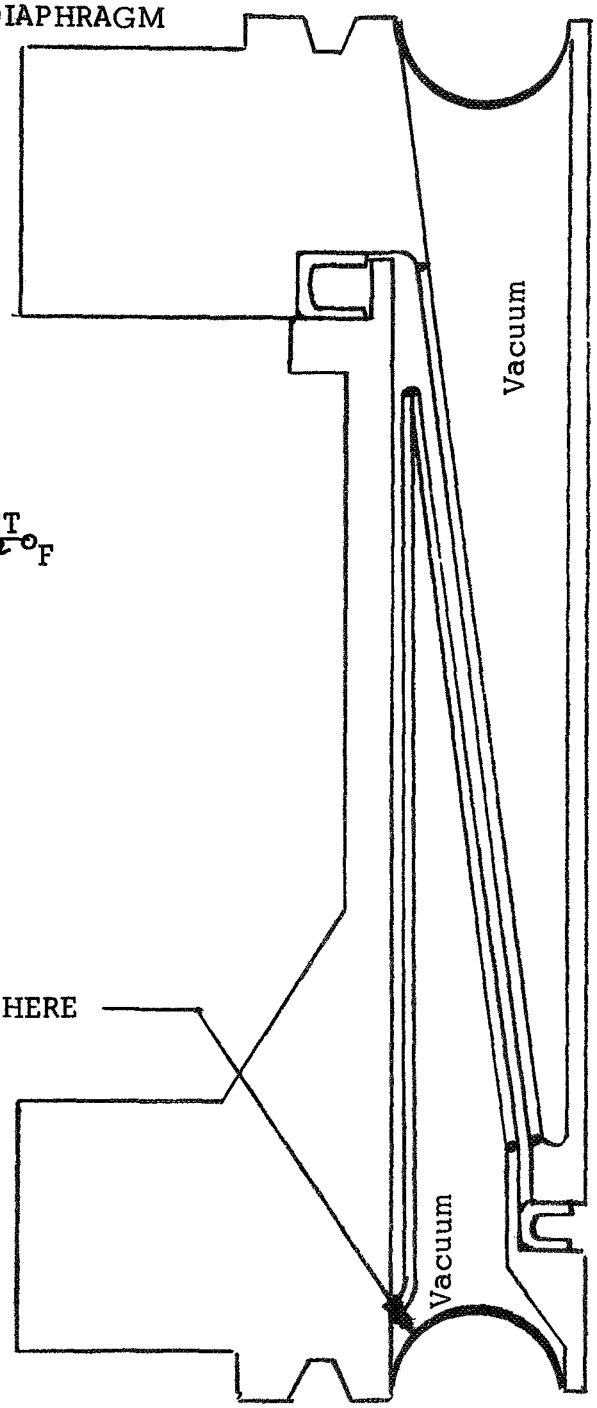
$+ 38.6 = 260.1\ \frac{\text{Btu}}{\text{hr}}$

Total heat leak assuming A.D.L. Block on both sides have same heat leak. This total heat leak assumes gas conduction heat leak block.

(1) block		(2)		(3)		(4)		(5)		(6)
Q = 2 x 98.1	+	53.8	+	23.0	+	39.0	+	22.7	+	20.6

=355.1 B/HR Heat Leak.

TITANIUM DIAPHRAGM



ASSUME CENTER CONE #2 HEAT LEAK OF THIS DESIGN IS THE SAME AS PREVIOUSLY CALCULATED FOR THE 6" ϕ ROTARY JOINT.

$Q = 42.5 \frac{B}{HR}$

TITANIUM DIAPHRAGM SIZING

$t = \frac{Pr}{Fty}$

F_{ty} = 130 KSI FOR CRUCIBLE C-130AM ALLOY, WITH A k = 3.50 $\frac{B - FT}{HR - Ft^2\,{}^\circ F}$

P = 285 PSI PROOF PRESS

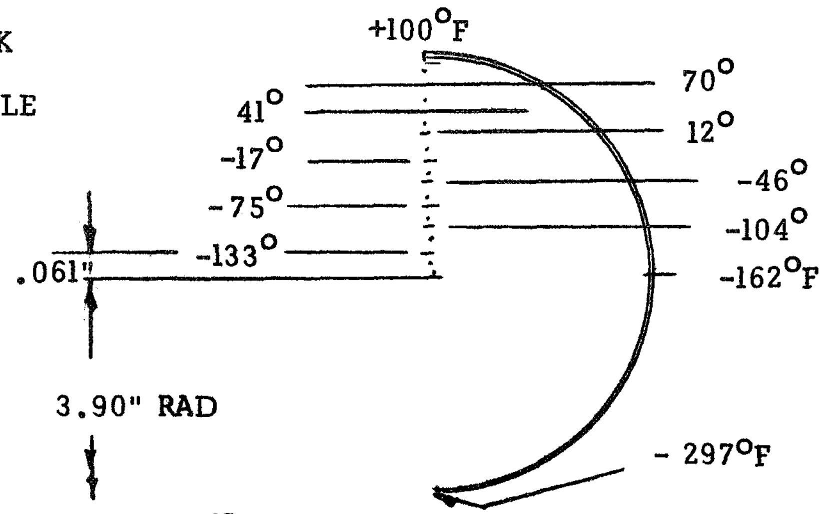
r = .56 IN

$t = \frac{285 \times .56}{130{,}000}$

= .0012 In. WALL TH. REQ'D, TOO LOW

TRY .012 INCH TH. TITANIUM HEMISPHERE DIAPHRAGM

ANTI-CONVECTION BLOCK
HEAT LEAK.
MADE SAME AS A.D. LITTLE
TYPE CONST.



SECTION	CIRCUM-FERENTIAL AREA SQ.FT.	INCREMENTS OF RADIAL LENGTH FEET	ΔT TEMP °F	MEAN HEAT LEAK $Q = k \frac{A}{L} \Delta T$ $= .012 \frac{A}{L} \Delta T$
1	.038	.0050	22	2.00
2	.061	.0050	22	3.22
3	.073	.0050	22	3.84
4	.081	.0050	22	4.28
5	.088	.0050	22	4.65
6	.091	.0050	22	4.82
7	.093	.0050	22	4.92
8	.094	.0050	22	4.97
9	.092	.0050	22	4.86
10	.090	.0050	22	4.75
11	.089	.0050	22	4.70
12	.087	.0050	22	4.60
13	.084	.0050	22	4.44
14	.079	.0050	22	4.18
15	.070	.0050	22	3.70
16	.057	.0050	22	3.00
17	.047	.0050	22	2.48
18	.027	.0050	22	1.43
			ΣQ	70.8 B/HR

These calculations show that the heat leak for the 6-inch rotary is within the heat leak target of 300 Btu/hr. The 8-inch unit is over. It should be point out that during the second phase of this study, a thermal conductivity test will be made of the foamed teflon. This information was unavailable during the first phase. If this material has a conductivity equal to or less than that of GO_2 (0.010 Btu/hr-ft^2-°F/ft) the heat loss will be decreased to:

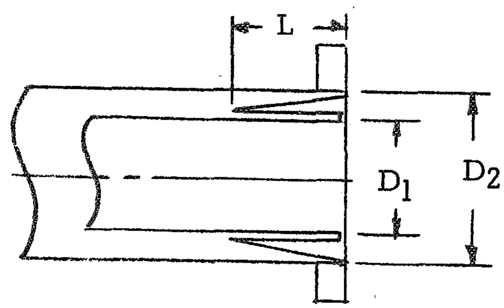
$$159.8 - 159.8 \times 0.010/0.012 = 26.6 \text{ or } 281.3 - 26.6 = 254.7$$

and

$$196.0 - 196.0 \times 0.010/0.012 = 32.7 \text{ or } 355.1 - 32.7 = 322.4$$

These same units, when used in liquid hydrogen, will have a heat leak increase of 523/397 or 1.35 over the LO_2 units. A 10 lb/cu ft density polyurethane foam will be used to hold against collapse of the foam at 300 psi. The conductivity of this material is slightly higher than the A.D. Little block. If the block is allowed to breathe as in the LO_2 units, the block heat leak will be higher by the GO_2/GH_2 conductivity ratio. If a hydrogen gas seal can be developed to seal the foam, then this heat loss can be reduced.

The above heat leaks, of course, can be reduced if it becomes more important than envelope in the final analysis. This can be accomplished as follows:



Increase D_2/D_1 ratio

Increase L/D_1 ratio

In the interest of small envelope and so as not to cause frost to form on these units, an L/D, ratio of ≈ 0.5 was used.

Another area that must be taken into account when designing a hydrogen vent or propellant system is the operating and proof pressure associated with it. The previous calculations were based on the existing LO_2 system pressure of 285 psi proof.

If these designs were optimized to actual system requirements, the vent line bayonets would be sized for 100 psi proof pressure. These lower pressures will reduce the density of foam needed for anti-convector blocks and, therefore, reduce heat leak. The cone and cylinder thickness will be reduced proportionately.

A copy of the compliant ball bearing report (Ref. 4) was acquired through Mr. J. F. Robinson of Autonetics. This report covers the work done on this new design concept. During a visit with Mr. Robinson earlier in this program, he suggested that we do not try to build in compliance on the Chiksan type bearing assembly since they had no experience in this area. A sketch (SK 4057-9) of the plain Kaydon bearing was sent to Autonetics. They submitted the SK 4057-9 drawing modified as shown. It was their suggestion to test two standard Kaydon bearings followed by two compliant bearings. This would establish the load increase capacity. The bearings will be loaded until brinelling occurs.

<u>8-inch Rotary Joint & Bayonet Weights Analysis</u>

<u>8-inch Rotary Joint</u>		Weight (lbs)
8-inch Line Sch 10		8.10
8-inch Line Adapter Ring Bayonet Side		.50
8-inch Line Adapter Seal Ring Bayonet Side		2.00
8-inch Line Adapter Seal Line Side		1.50
Inner Cylinder 0.060 in. Wall		2.56
Inner Cone 0.0355 in. Wall		2.96
Center Cone 0.048 in. Wall		2.68
Outer Cone 0.0355 in. Wall		2.02
Outer Cylinder 0.0355 in. Wall		2.55
Bearing Shaft		30.38
Bearing Housing		72.95
Bearing Spacer		2.68
Bearing (2)		4.29
Flange		63.66
Bolts (Included in Flange Holes)		77.44
Foam Weight		1.43
	TOTAL	278.00

8-inch actual weight = 240 lbs
6-inch actual weight = 177 lbs

Bayonet		Weight (lbs)
8-inch Line 48 in. Lt Sch 10		53.60
Inner Cylinder		2.56
Inner Cone		2.96
Flange 150 Weld Neck		65.48
Vacuum Jacket 10 in. Sch 10		68.60
Foam		.50
	TOTAL	193.70

4.1.3.3 Reliability Program

The preliminary reliability goal has been established in Section title "Review of Test Data," based on the original design. In this section, the state-of-the-art reliability and final reliability goal is established. These last two numbers are based on the rotary designs of SK 4057-7a and SK 4057-8.

For the state-of-the-art reliability, the assumed population of parts is the same as used in the preliminary reliability goal prediction (65 units).

The inherent failure modes are:

		Failures
A. Seal failure (O-ring, Raco, Chiksan type)	Probability -	1
B. Contamination (Anti-convection block particles)	Probability -	2
C. Structural failure in cone area	Probability -	1
D. Insulation material reaction with LO_2	Probability -	0.5
E. Bowed annular seal surface causing liquid leadage	Probability -	0.5
F. Structural failure in flange area	Probability -	1
	TOTAL	6

$$R = \frac{T - F}{T} = \frac{65 - 6}{65} = 0.9077$$

Final — Reliability Goal

The assumed population is as previously calculated (65 units).

The inherent failure modes are the same as above.

			Failures
A.	Seal failure	Probability -	1
B.	Contamination	Probability -	1
C.	Structural failure in cone area to be eliminated during test		0
D.	Insulation material reaction with fluid must be eliminated by design and test		0
E.	Bowed annular seal surface		0
F.	Structural failure in flange area eliminated by design		0
		TOTAL	2

$$R = \frac{T - F}{T} = \frac{65 - 2}{65} = 0.9692$$

4.1.3.4 Quality Assurance Program

In order to ensure against inferior or out-of-tolerance parts, all parts of both rotaries will be subjected to 100% inspection. All detail parts will be 100% traceable since they are being released through Straza's standard release system. A complete engineering planning paper will be written against all detail in-house fabricated parts to ensure for adequate tooling and inspection during the fabrication process. All special processes and welding will be covered by appropriate Straza control specifications. These specifications are called out on each detail drawing as applicable.

All inspection reports will be reviewed by Straza's Quality Assurance group for proper corrective action. The Material Review Board, which establishes the corrective action, consists of one member each from Quality Control, Reliability and the appropriate project engineer.

4.1.3.5 Conclusion

The tabulated heat leak of 282 Btu/hr for the 6-inch rotary and 355 Btu/hr for the 8-inch rotary indicate the feasibility of a low heat leak rotary coupling. Even though the heat leak of the larger unit is higher than the original goal of 300 Btu/hr, the design approach shows where modifications to these basic units can lower this heat leak. One must, however, be willing to accept the larger envelope, in diameter or length, to accomplish this end.

The basic design lends itself to a variety of configurations, depending on weight, heat leak, or envelope restrictions existing in the area where the unit will be installed.

4.1.4 Phase II Proposal

4.1.4.1 Test Plan

One 6-inch and one 8-inch rotary will be tested in conjunction with the bayonet joint as shown in SK 4057.

The heat leak test will be performed using the setup as shown in SK 4057-15, except the rotary will be mounted between the bayonet sections within the controlled temperature box.

The tests to be run on these assemblies were chosen from and will be run in accordance with the requirements of KSC-STD-164D, "Environmental Test Methods for Ground Support Equipment." It is felt that the tests chosen represent the most severe environments that we can subject the units to and still remain within the test budget.

The following developmental functional tests will be performed on units noted:

8-inch AMETEK/Straza Rotary

A. Compliant bearing capacity increase over standard bearing.

(1) Torque measurement on two standard bearings at various pressures up to brinelling as determined by torque measurement.

(2) Repeat (1) above on two compliant bearings.

B. <u>Pressure Test</u>

(1) Operating pressure — 190 psi

(2) Proof pressure — 285 psi, hold each for five minutes

C. <u>Leakage</u>

(1) Static — Bubble tight at -320°F and 190 psi

(2) Dynamic — 3 scim helium at -320°F and 30 psi

D. <u>Heat Leak</u>

300 Btu/hr maximum tested as discussed above

E. <u>Torque</u>

Measure torque with test setup shown in SK 4057-17, record bending load, pressure, leakage, torque (at 2.3 rpm) at 70°F and -320°F from 0 psi to 190 psi in 10 psi increments. Bending moment increments will be 50 ft-lbs to 900 ft-lbs maximum.

<u>6-inch Chiksan Rotary</u>

Repeat "B" through "E" above.

The following functional test shall be run once before and after environmental test on both units.

A. Pressure Test

B. Leakage (a and b)

C. Torque

With unit set up per SK 4057-17, apply 300 ft-lbs bending moment with unit pressurized to 30 psi. Measure leakage and torque at 2.3 rpm at -320°F using GN_2.

The following environmental test will be performed on each rotary unit in the sequence shown below:

A. <u>Vibration</u> (Procedure II of KSC-STD-164D)

(1) Sinusoidal vibration to Level C (20 g's) using test setup shown in SK 4057-16.

(2) Random vibration to Level C (0.05 G^2/cps).

B. Sand and Dust

Per KSC-STD-164D.

C. Salt Fog

Per KSC-STD-164D.

D. Life Cycle

A minimum of 250 life cycles shall be run at a pressure of 30 +0/-5 psig and a temperature of -300 +00/-20°F using GN_2. Two hundred-fifty cycles shall be run at 0 psig and ambient temperature. The above 500 cycles shall be run with a fixed moment of 300 ft-lbs on the rotary. One cyle shall consist of rotating 360° at 2.3 rpm.

E. Burst

At the completion of all evaluation and environmental testing, the unit shall be subjected to 760 psig.

4.1.4.2 Procurement Plan

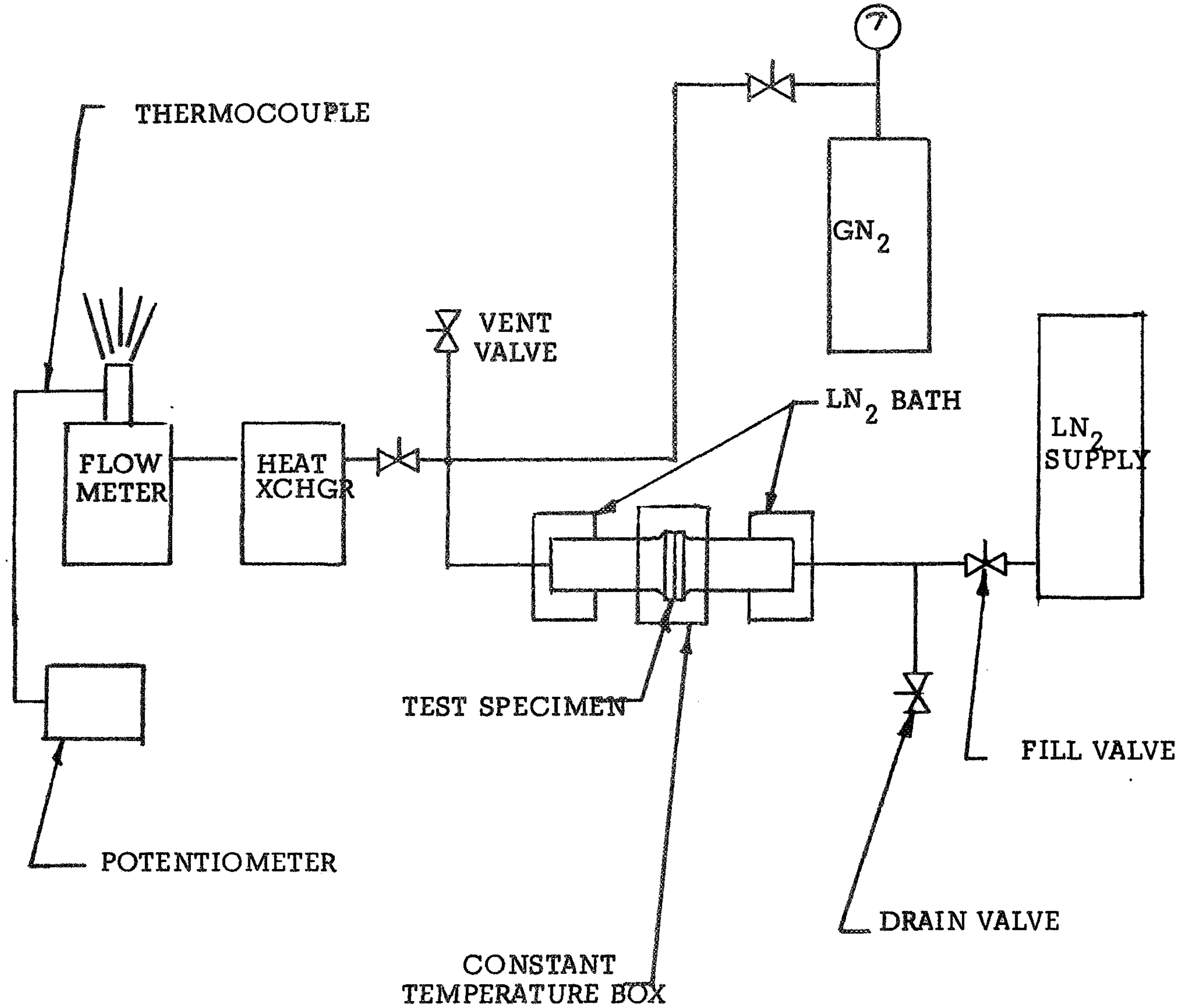
The design phase is now at a point where part procurement can proceed in order to make Phase II test schedules.

The parts which will be procured outside are as follows:

Item	Size	Quantity
Chiksan style rotary & seals	8-in. (Jackets to 6 in.)	1 ea
Raco Seals	Assorted	4 ea
Kaydon Bearing	12 in. I.D.	6 ea*
Foamed Teflon anti-convection blocks	6 in.	2 ea
Forged 316 Stainless Steel rings (for end flanges)	A/R	A/R

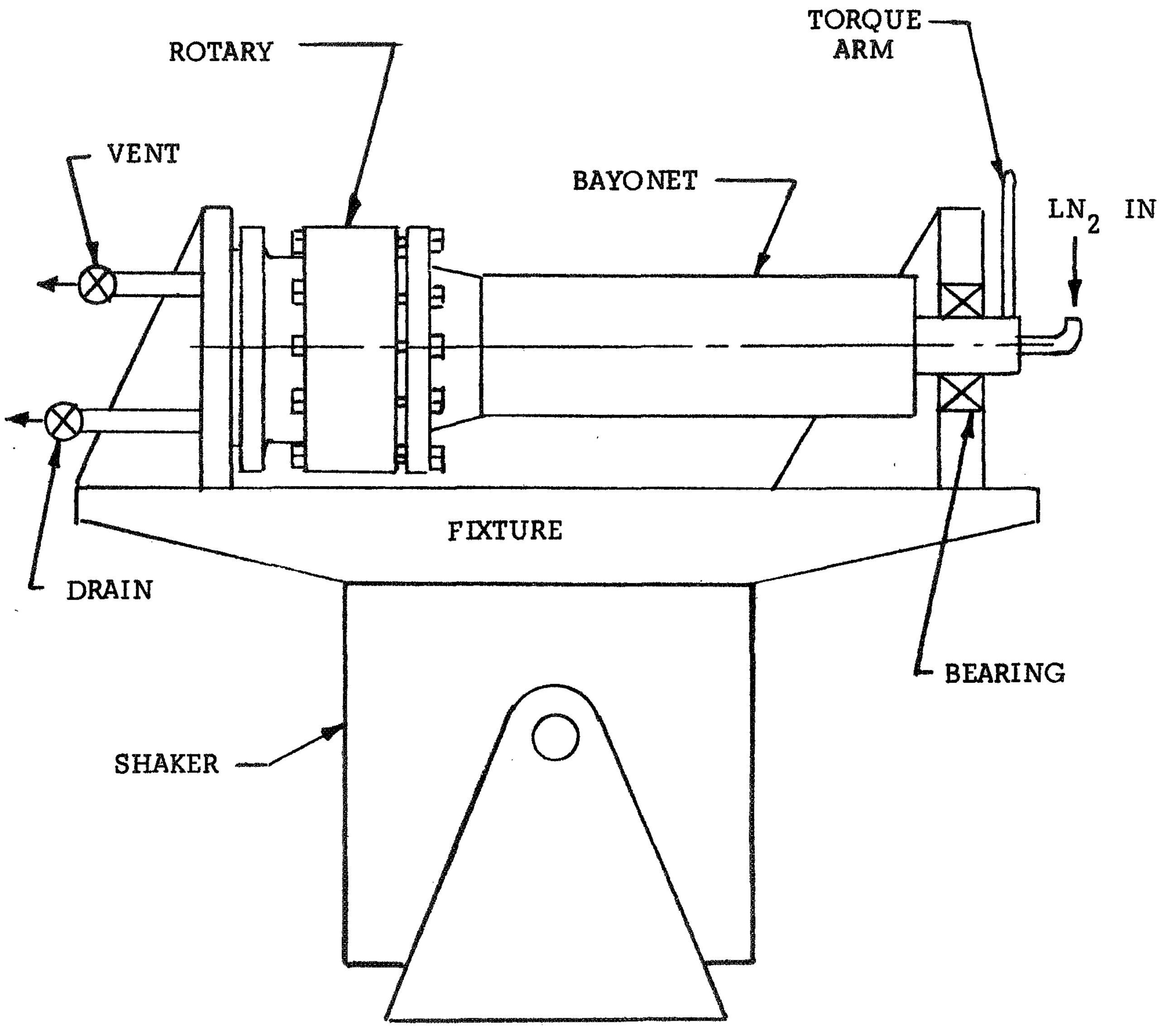
The above hardware will make up one complete 8-inch joint as shown in SK 4057-7A and one complete 6-inch joint as shown in SK 4057-8.

*Two extra bearings are required to make an accurate determination of optimum compliance.



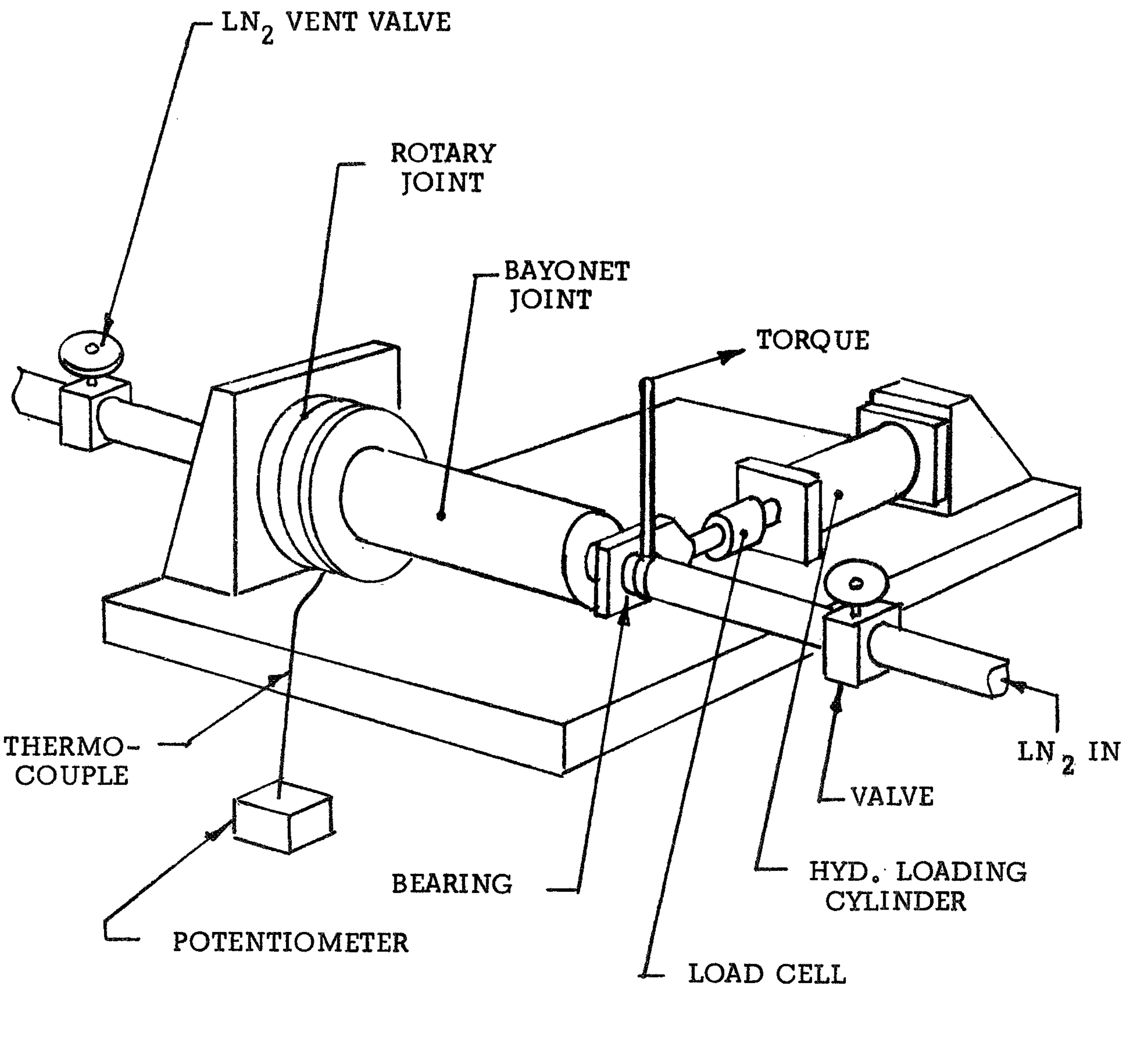
HEAT LEAK TEST SET-UP BAYONET JOINT

SK 4057-15 Heat Leak Test Set-up



VIBRATION TEST

SK 4057-16 Vibration Test



BENDING MOMENT & TORQUE TEST SET-UP

SK 4057-17 Torque Test Set-up

4.1.4.3 Procurement Specifications

The only procurement specification required for the rotaries is the Specification Control Drawing that establishes the envelope which Chiksan must hold in order to complete the jacketing fabrication of this unit at AMETEK/Straza. (Shown on Specification Control Drawing 8-040352.)

All other parts of both rotaries are off-the-shelf standards and are procured by vendor part number.

4.2 PHASE II PROGRAM

Fabrication and Assembly

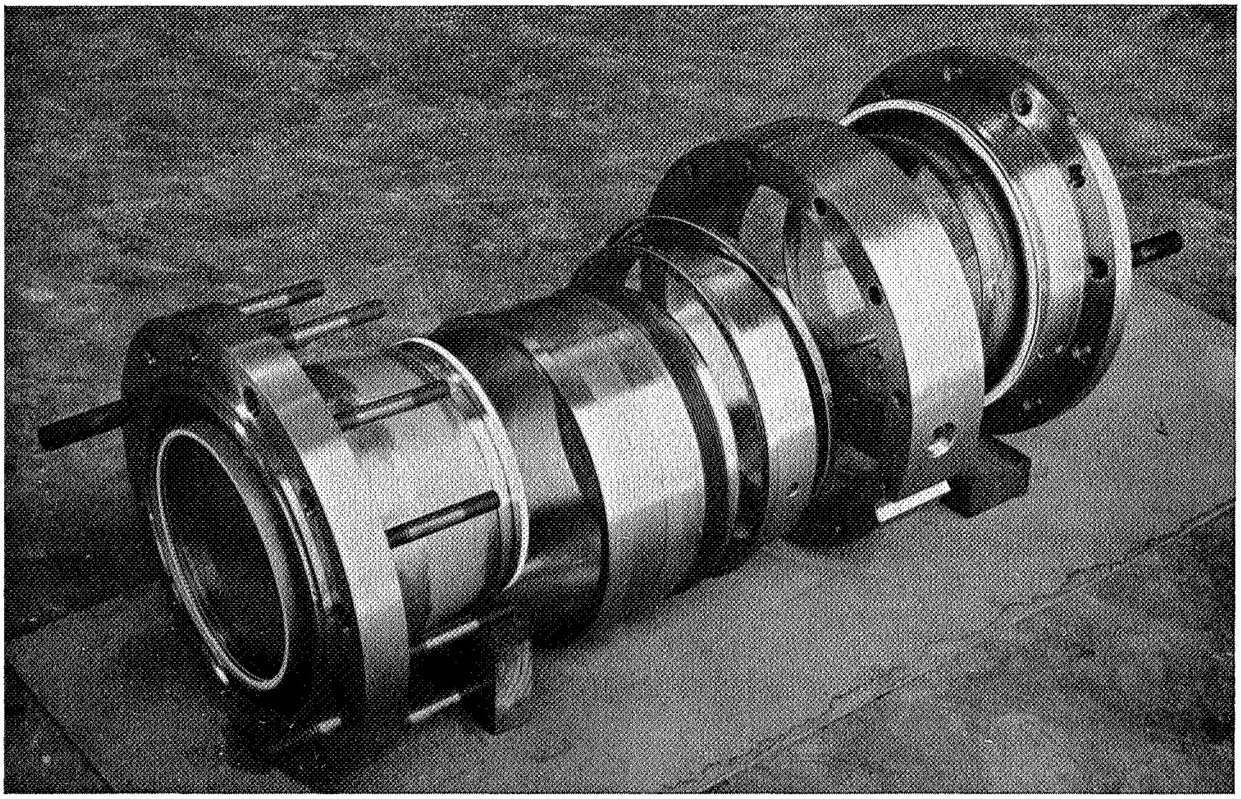
The second phase of this program consisted of fabricating and testing one 8 inch rotary and one 6 inch rotary. The fabrication of the 8 inch unit progressed with only a few assembly problems. The fabrication method used on the Saturn V propellent line bayonet development worked well for the rotary. This consisted of final machining of the metal cone nose seal gland after welding and concurrent with the machining of the ring-joint groove in the attaching flanges.

The assembly procedure consisting of inserting a locking tool through the hole in the bearing collar and into the bearing sleeve, worked very well. This provides a good hold for assembly of the threaded joint which was held at the left end (Dwg. 8-100089) with an adjustable spanner wrench. In order to provide for the proper bearing preload, a .006 inch thick shim was placed between Item 2 and Item 3 (Dwg. 8-100089). This amount of shim was enough to remove all axial play without overloading the bearing in thrust.

Test Philosophy

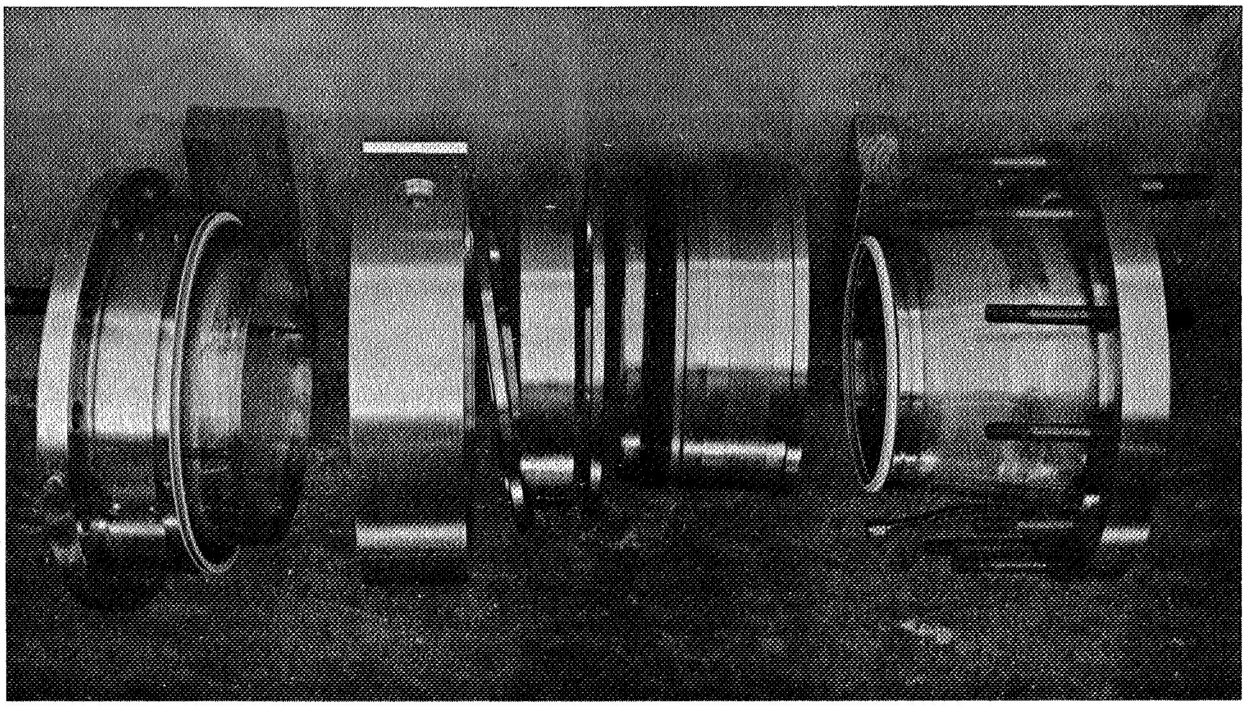
Both the 6 inch and the 8 inch rotaries were heat leak tested with the CTL DIXIE insulators for LO_2 systems. A photograph of these insulators is included along with an exploded view of the 8 inch rotary. The rotary insulators were not heat leak tested for LH_2 system application due to the added expense and time. It was determined that sufficient data could be obtained from the Task IV bayonet study to preclude duplication in the rotary study. The LH_2 insulator, which would be used in the rotary, is identical in construction to the unit used in the bayonet. A photograph of that insulator is included.

A bending moment and rotational torque measurement test fixture was constructed to simulate installation loading. The design of this fixture is shown in Dwg. 8-100096. A photograph of this fixture with the 8 inch rotary/bayonet assembly installed, is included in the test report. The fixture was designed for continuous automatic rotational cycling through 360° clockwise and counterclockwise. A separate cylinder (Item 17 in Dwg. 8-100096) provided a fixed bending load at the end of the 4 foot long bayonet tube. A position potentiometer provided a continuous 2.3 RPM cycle rate readout on the Sanborn recorder. The bayonet assembly provided the necessary jacketing and end closures for the cryogenic portion of the rotational tests.



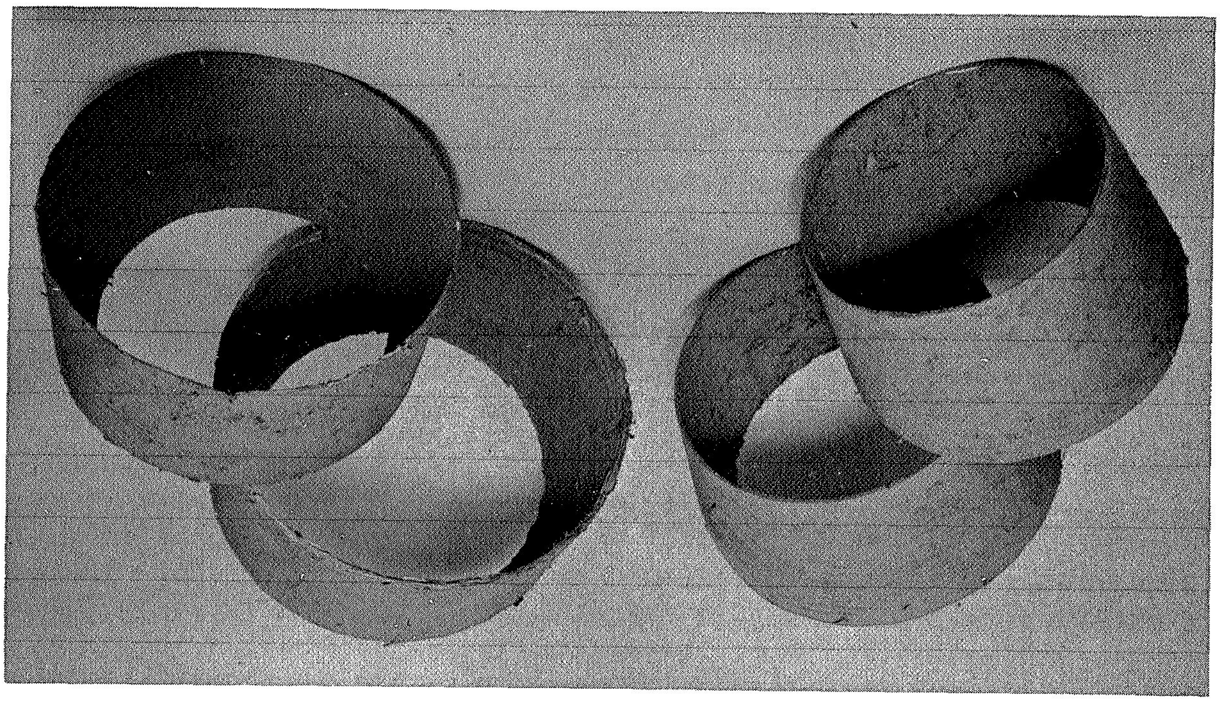
AMETEK/Straza 8-inch Rotary Joint — Disassembled
Oblique Views from Both Ends

Exploded Photograph of 8 inch Unit



AMETEK/Straza 8-inch Rotary Joint — Disassembled
Side View

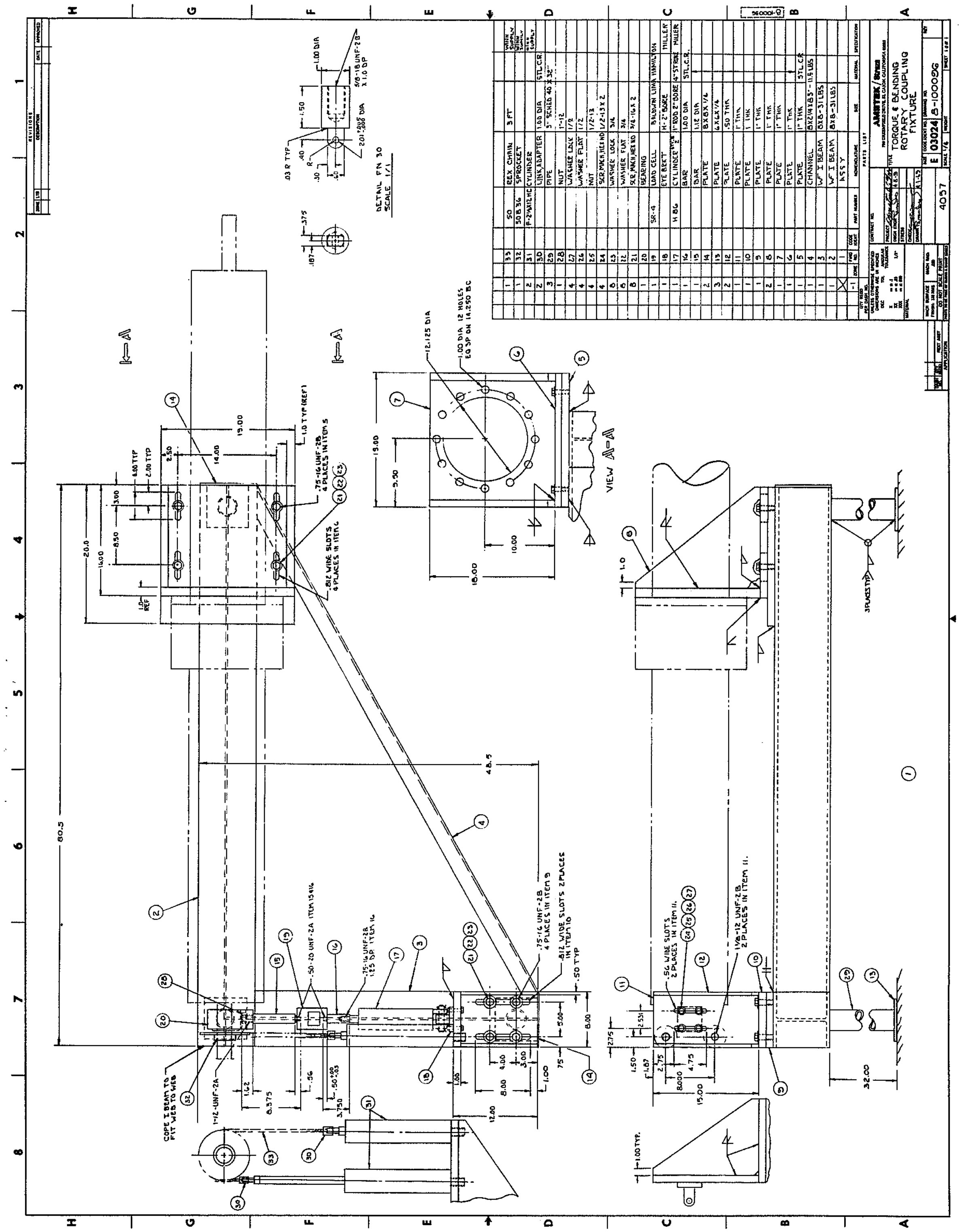
Exploded Photograph of 8 inch Unit



LO_2 Insulator Blocks

LH_2 System Anti-Convection Barrier Cone After
285 psi Proof Pressure

LH_2 **System Anti-convection Barrier**



Bending Fixture Drawing No. 8-100096

4.2.1 Test Report

4.2.1.1 Introduction

The following report presents the results of the design verification tests performed on the rotary joint (Task IV) and includes the data sheets of the environmental tests for the bayonet joint (Task VII) of the Technology Advancement Study. These tests were performed per the requirements established in the Design Verification Test Procedure, Number 8-480086.

One 8 inch rotary joint was tested. The 8 inch unit performed as anticipated, meeting the heat leak and torque test goals established in the early part of the program. The compliant Kaydon bearings did not perform as predicted in their thrust load carrying ability. This was attributed to the insufficient radially tight fit and a bearing cross-section approximately one size too small.

The leakage test in the 8 inch unit indicated that the teflon primary seal should have been Kel-F to prevent excessive scuffing. Leakage values ran from 0 SCIM to 10 SCIM at cryogenic temperatures while rotating.

The 6 inch Chiksan unit performed satisfactorily as far as it was tested. Due to funding limitations this unit was subjected only to the Receiving and Inspection Tests.

4.2.1.2 Scope

The purpose of this document is to set forth the tests and methods of testing to verify the performance and reliability of Rotary Joints under environmental conditions to satisfy the service requirements at the John F. Kennedy Space Center. The test items were subjected to the following tests in the sequence listed:

TYPE OF TEST

8 inch Unit	6 inch Unit
Bearing Evaluation	Receiving Inspection
Functional	
Pressure-Proof	
Heat Leak	
Torque	
Sand and Dust	
Salt Fog	
Vibration	

The burst test was proposed in the Test Plan but it was deleted since it would damage the units to the point of precluding any additional testing or data confirmation which NASA might desire after the completion of this particular program.

4.2.1.3 Item Description

The test items are two rotary joints; one fabricated completely by AMETEK/Straza as shown in Dwg. No. 8-100089, and one supplied by the Chiksan Division, FMC. Chiksan supplied the rotary unit only and it was jacketed by AMETEK/Straza. This item is shown in Dwg. 8-100098. Both units consist of a basic rolling element, a double dynamic seal and overlapping cone jacket. The bayonet test section of Task VII was used during this test as end closures and torque fixtures for the rotaries during the majority of these tests.

In lieu of a pinch-off tube at the vacuum annulus port, a portable vacuum pump was used to allow for inspection of this cavity during the tests.

4.2.1.4 Applicable Documents

The following documents or the revision noted form a part of this test requirement to the extent specified herein:

KSC-STD-164D, Environmental Test Methods for Ground Support Equipment Installations at Cape Kennedy.

Straza Report No. 8-480086, Design Verification Test Procedure-Rotary Joint.

Final Technical Report for Vacuum Jacketed Umbilical Lines-Technology Advancement Study Task VII - Bayonet Joint.

4.2.1.5 Test Performed

4.2.1.5.1 Receiving Inspection (6 inch unit only)

Test Requirement

This inspection was made to determine conformance with applicable drawings and specifications to the extent possible without disassembly of the test item.

Test Procedure

The Receiving Inspection included the following:

Identification of test items by marking to establish manufacturer's part number.

Visual inspection to establish the "as received condition" and verify that the items' configuration and external dimensions are in conformance with the applicable drawings and specifications.

Test Results

The unit was identified by part number and met the dimensional and visual requirements. Chiksan also submitted a certification with the unit.

4.2.1.5.2 Bearing Evaluation Test (8 inch unit only)

Test Requirements

The test requirements stated that a bearing load test be performed to determine increase in capacity of the compliant Kaydon bearing over the standard Kaydon bearing used in the 8 inch rotary.

Test Procedure

A set of standard bearings was installed in the 8-100089 test item which was installed in the test fixture shown in Test Report Figure 1. The test item was pressurized with nitrogen gas over oil, using the test setup as shown in Test Report Figure 2, while being cycled at approximately 2.3 RPM clockwise and counterclockwise through 90° min. from a nominal position. Pressurization was in increments of approximately 50 psi from 0 to Brinelling pressure. A separate pressure line was connected to the vacuum jacket cavity of the rotary to maintain equal pressure and prevent cone collapse in the event that Brinelling pressures were higher than proof.

At each pressure increment, the torque versus angle of rotation and Brinell point was recorded. The standard bearing was removed from the test item and a second standard bearing installed in the failed bearing position and the above procedure repeated.

After two standard bearings have been tested, the above test procedure was repeated using two compliant bearings.

Test Results

A series of five (5) tests were run for the bearing evaluation. The first test was carried to a pressure of 300 psi where it was stopped due to bearing noise which was noticed when the pressure was reduced to zero. The unit was disassembled and contamination had deposited on the loaded bearing race. The unit was cleaned, reassembled, and subjected to the second test. It was found, after completing the torque test to 425 psi, that in order to get a good Brinelling reading the pressure must be reduced to 30 psi or less between each pressure cycle. Since this was not done on this test, the second standard bearing was installed and Test #3 was conducted. Brinelling was observed at approximately 250 psi.

The unit was disassembled and the bearing examined. Brinelling was observed at the upper edge of the bearing and Brinelling occurred at 175 psi on Test #4.

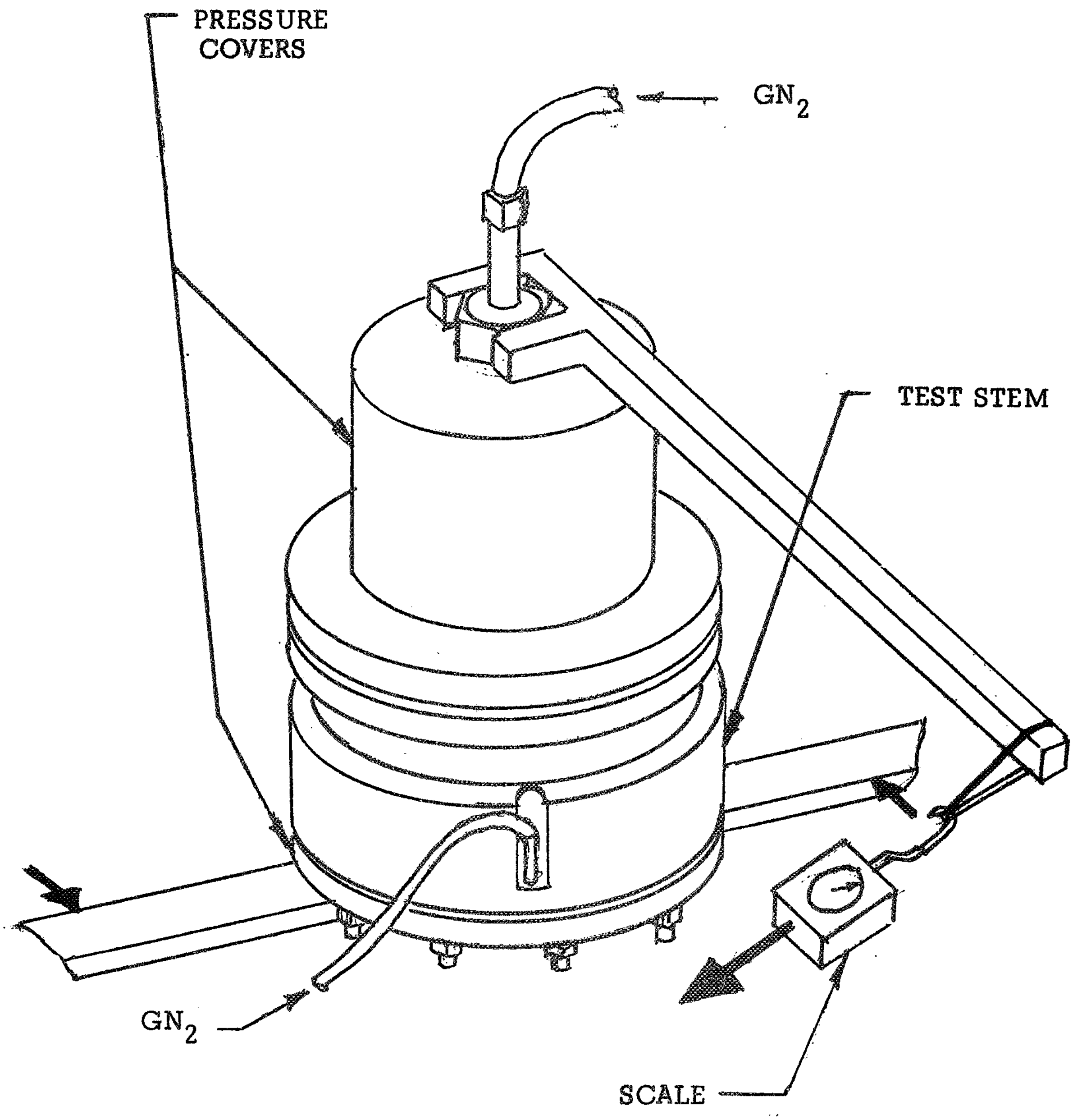
The heavy thrust load on these angular contact bearings was spreading the races in a radial direction, allowing the load line through the ball to go well beyond the 30° angle. In order to reduce this tendency, shims were placed between the I. D. race and sleeve. Test #5 was conducted but Brinelling pressure was still at the 175 psi level.

It was decided to downgrade the operating pressure for this 8 inch unit to 140 psi so that the remaining tests could be completed without damaging the bearings.

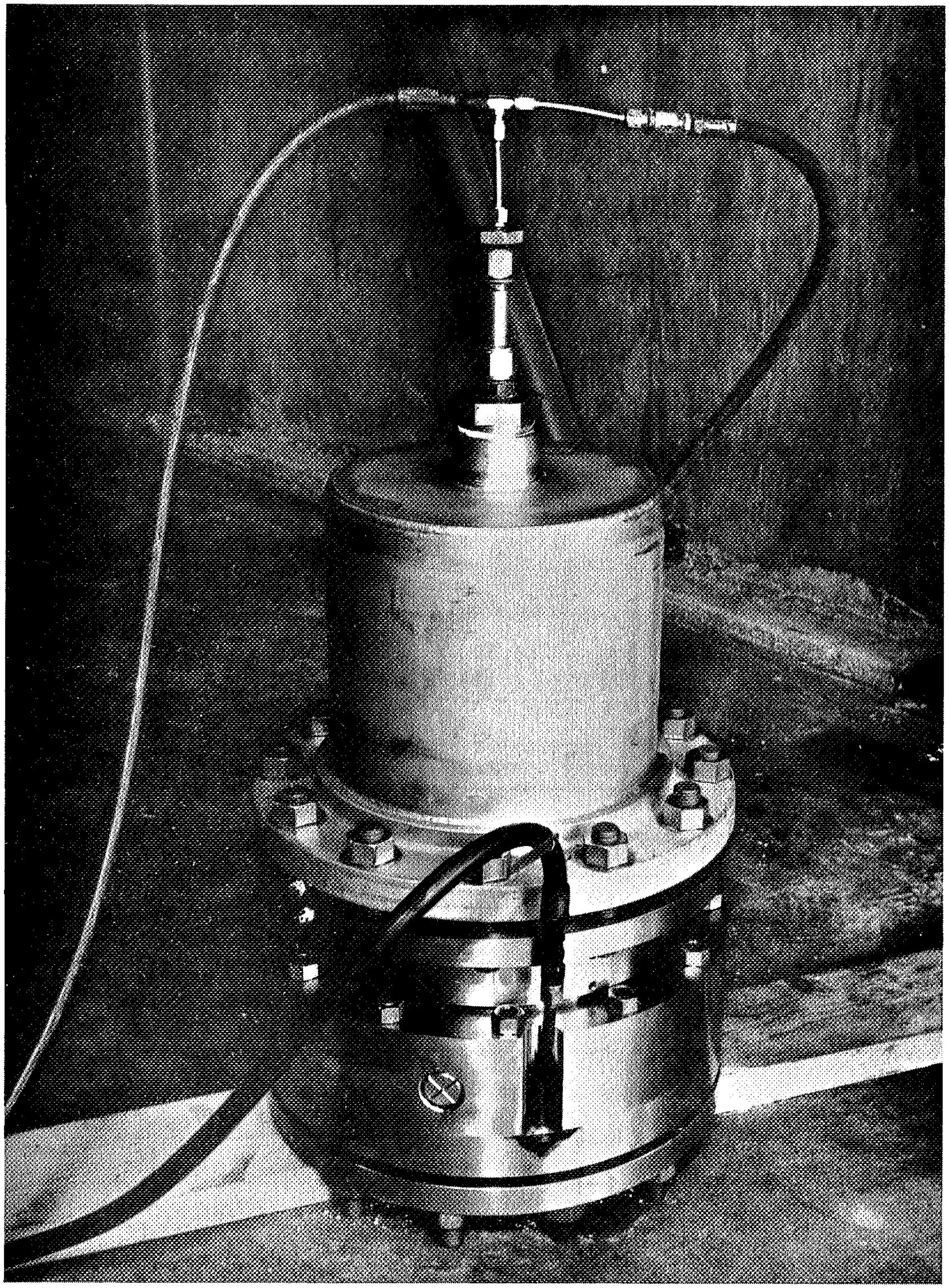
It was concluded that to make these bearings work properly, a press or shrink fit would be required. Since this would make the assembly and disassembly extremely difficult the better modification would be to go to the next size bearing. The outer bearing housing could have been modified to do this but there was not enough time or money in this program. It was also concluded that since the Chiksan unit had larger bearings, this fix would be confirmed when the 6 inch unit was tested.

Test Data

A photograph of the test set-up is included, along with the test data sheets.



TEST REPORT
FIGURE I



AMETEK/Straza 8-inch Rotary Joint

Bearing Evaluation Fixture, 8 inch Unit

DESIGN VERIFICATION TEST

TEST DATA SHEET

Type of Test Bearing Evaluation #1 Date of Test 11 July 1969

Part Name S.I. Rotary Joint 8" Part Number 8-100089-1

Test Procedure 8-480086 (Para 5.2) Part Serial # 1

Type of Bearing Std Kaydon Cycling Rate 2 RPM

Pressure Media GN_2 over oil Weight 240# with studs

Pressure (in psig)	Torque (in in-lb)	Angulation (in degrees)	Remarks
0	1260	180	
30	1080	90	
50	1120	90	
100	1420	90	
150	1770	90	
175	2000	90	
200	2160	90	
225	2700	90	
250	2970	90	
275	3510	90	
285	3510	90	
300	3780	90	
0	1128	90	Test stopped because of
			bearing noise at no load.
			Examination disclosed metal

particle in loaded bearing.

Test Technician /s/ S. Moore Test Engineer /s/ R. C. Mursinna

DESIGN VERIFICATION TEST

TEST DATA SHEET

Type of Test Bearing Evaluation #2 Date of Test 14 July 1969

Part Name S.I. Rotary Joint 8" Part Number 8-100089-1

Test Procedure 8-480086 (Para 5.2) Part Serial # 1

Type of Bearing Standard Kaydon Cycling Rate 2 RPM

Pressure Media GN_2 over oil

Pressure (in psig)	Torque (in in-lb)	Angulation (in degrees)	Remarks
0	600	90	
250	1620	90	
275	1900	90	
300	2150	90	
325	2550	90	Breakaway 2550
350	2700	90	Breakaway 2700
375	2920	90	
400	3300	90	Breakaway 3500
425	3940	90	Breakaway 4325
			Rotation during load applica-
			tion does not give accurate
			failure point reading.

Test Technician /s/ S. Moore Test Engineer /s/ R. C. Mursinna

DESIGN VERIFICATION TEST

TEST DATA SHEET

Type of Test Bearing Evaluation #3 Date of Test 15 July 1969

Part Name S.I. Rotary Joint 8" Part Number 8-100089-1

Test Procedure 8-480086 (Para 5.2) Part Serial # 1

Type of Bearing Standard Kaydon Cycling Rate 2 RPM

Pressure Media GN_2 over oil

Pressure (in psig)	Torque (in in-lb)	Angulation (in degrees)	Remarks
0	740 (0 psig)	90	
50	740 (25 psig)	90	
100	770 "	90	Slight Brinell Indication
150	700 "	90	"
175	700 "	90	"
200	700 "	90	"
225	680 "	90	1100 "# Breakaway
250	680 "	90	
275	680 "	90	
285			

Increased press held for 30 sec, released to 25 and torqued.

Excessive clearance between Bearing bore and Housing O.D. caused swagging in of Bearing race and Brinelling of race at lip.

Test Technician /s/ S. Moore Test Engineer /s/ R. C. Mursinna

DESIGN VERIFICATION TEST

TEST DATA SHEET

Type of Test Bearing Evaluation #3 Date of Test 15 July 1969

Part Name S. I. Rotary Joint 8" ⌀ Part Number 8-100089-1

Test Procedure 8-480086 (Para 5.2) Part Serial # 1

Type of Bearing Standard Kaydon Cycling Rate 2 RPM

Pressure Media GN_2 over oil

Pressure (in psig)	Torque (in in-lb)	Angulation (in degrees)	Remarks
0	575 (0 psig)	90	
50	600 (25 psig)	90	
100	700 "	90	
150	700 "	90	No indication of brinell
175	700 "	90	"
200	700 "	90	"
225	700 "	90	"
250	700 "	90	Slight indication of brinell
275	700 "	90	Brinell indicated*
			Load applied, held for 1/2
			min., released to 25 psi
			and torqued.

*Footprint approximately .040 X .020

Test Technician /s/ S. Moore Test Engineer /s/ R. C. Mursinna

DESIGN VERIFICATION TEST

TEST DATA SHEET

Type of Test Bearing Evaluation #5 Date of Test 19 July 1969

Part Name S.I. Rotary Joint 8" Part Number 8-100089-1

Test Procedure 8-480086 (Para 5.2) Part Serial #

Type of Bearing Compliant Kaydon Cycling Rate 2 RPM

Pressure Media GN_2 over oil

Pressure (in psig)	Torque (in in-lb)	Angulation (in degrees)	Remarks
0	575	90	
50			
100	560 (25 psi)	90	
150	560 "	90	
175	550 "	90	Indication of Brinell
200	550 "	90	Indication of Brinell
225	550 "	90	Indication of Brinell
250	620 "	90	Indication of Brinell
275	620 "	90	Indication of Brinell
300	680 "	90	Indication of Brinell
			.003 SHIM Under BRG (lower)
			.002 SHIM Under BRG (upper)

Test Technician /s/ L. Mc Knight Test Engineer /s/ R. C. Mursinna

4.2.1.5.3 Functional Test

Test Requirements

The test requirements specified that prior to environmental testing, each test item be subjected to the following tests.

A. Pressure - Operating

The test items shall not deform when subjected to 140 psig (8 inch unit) or 190 psig (6 inch unit).

B. Leakage - Dynamic/Static

The test items shall not leak GN_2 at $320^{O}F$ in excess of 3 SCIM at 30 psig while rotating at 2.3 RPM. While non-rotating, the items shall be bubble tight to 140 psig at $-320^{O}F$.

C. Torque

The test items shall not require more than 600 ft-lbs. rotational torque when pressurized with GN_2 at $-320^{O}F$ and 30 psig. At the same time, the unit shall be subjected to a bending moment of 300 ft-lbs. and shall be rotating at 2.3 RPM.

Test Procedure

A. Pressure - Operating

The test items were installed in a test set-up as shown in Figure 2 and pressurized to 140 ±10 psig (8 inch unit) and 190 psig (6 inch unit) and that pressure maintained for five (5) minutes. Test media was gaseous nitrogen at room ambient temperature.

The test pressure, test media, and duration of the test was recorded. Any deformation or defects as a result of this test were recorded and photographed.

B. Leakage - Dynamic/Static

The test item was installed in a test set-up as shown in Figure 3 and filled with liquid nitrogen and allowed to stabilize. Stabilized temperature was -300^{+0}_{-20} ^{O}F as determined by a thermocouple. After stabilization, the liquid flow was shut off and the vent valve (Figure 3) was adjusted and monitored to maintain 140 $^{+0}_{-10}$psig (8 inch unit) and 190 psig (6 inch unit). Pressurization was to be obtained by using GHE over the LN_2 and maintained for five (5) minutes. The test item pressure was then reduced to 30^{+0}_{-5} psig and cycled at 2.3 RPM for five (5) minutes. One cycle consisted of rotating through 360° clockwise and counterclockwise from a nominal position. Leakage was measured at the assembly tool port in the bearing housing by the water displacement method.

C. Torque

The test item was installed in a test set-up as shown in Figure 3 and filled with liquid nitrogen allowed to stabilize. Stabilized temperature was -300^{+0}_{-20} °F as determined by a thermocouple. After stabilization, the liquid flow was shut off and the vent valve (Figure 3) was adjusted and monitored to maintain 30^{+0}_{-5} psig; 300 ft-lbs. bending moment was applied to the rotary joint. The test item was cycled at 2.3 RPM for five (5) minutes. The leakage and torque was measured and recorded. A photograph of the bending moment versus rotational torque test set-up is included.

In order to obtain additional radial load carrying capacity data on the compliant bearing, the bending moment was increased to 2100 ft-lbs.

Test Results

A. Pressure - Operating

The 8 inch unit was pressurized to operating pressure with zero leakage and no visible damage or distortion.

B. Leakage - Dynamic/Static

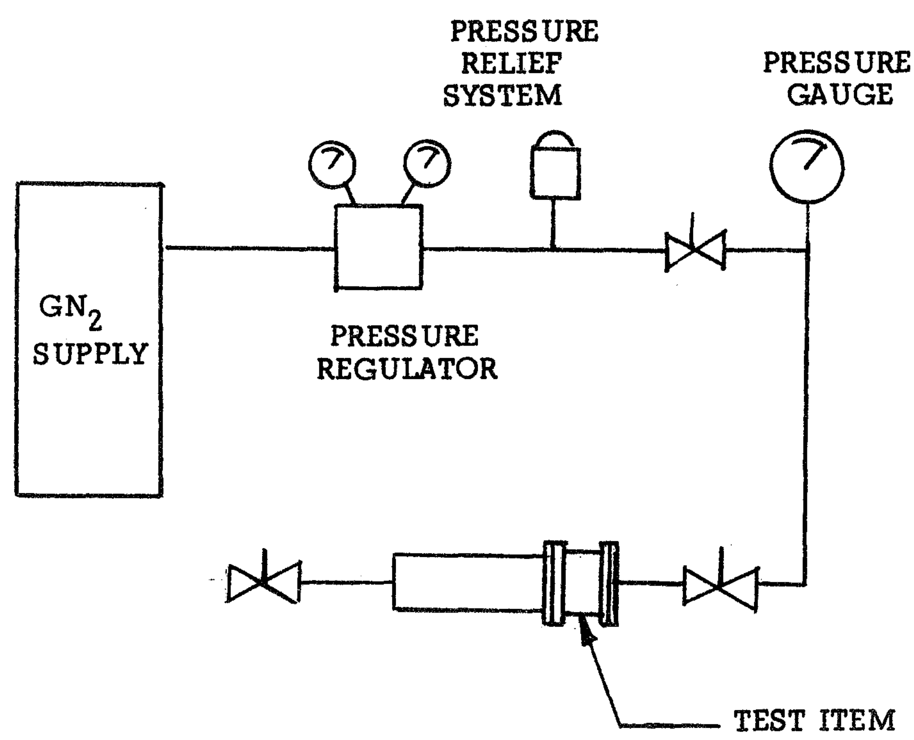
The 8 inch unit was pressure tested and complied with the leakage criteria. The unit had zero leakage with the rotary static and 1.4 SCIM leakage with the rotary rotating at 2.3 RPM.

C. Torque

The 8 inch unit was torque tested with an average torque of 242 ft-lbs. over the bending moment load range of 0 to 2100 ft-lbs. The bending moment load carrying capability of the 8 inch unit using compliant bearings, should be not less than 4000 ft-lbs. The leakage under the maximum bending load of 2100 ft-lbs. was 5.0 SCIM, slightly over the specification goal of 3.0 SCIM.

Test Data

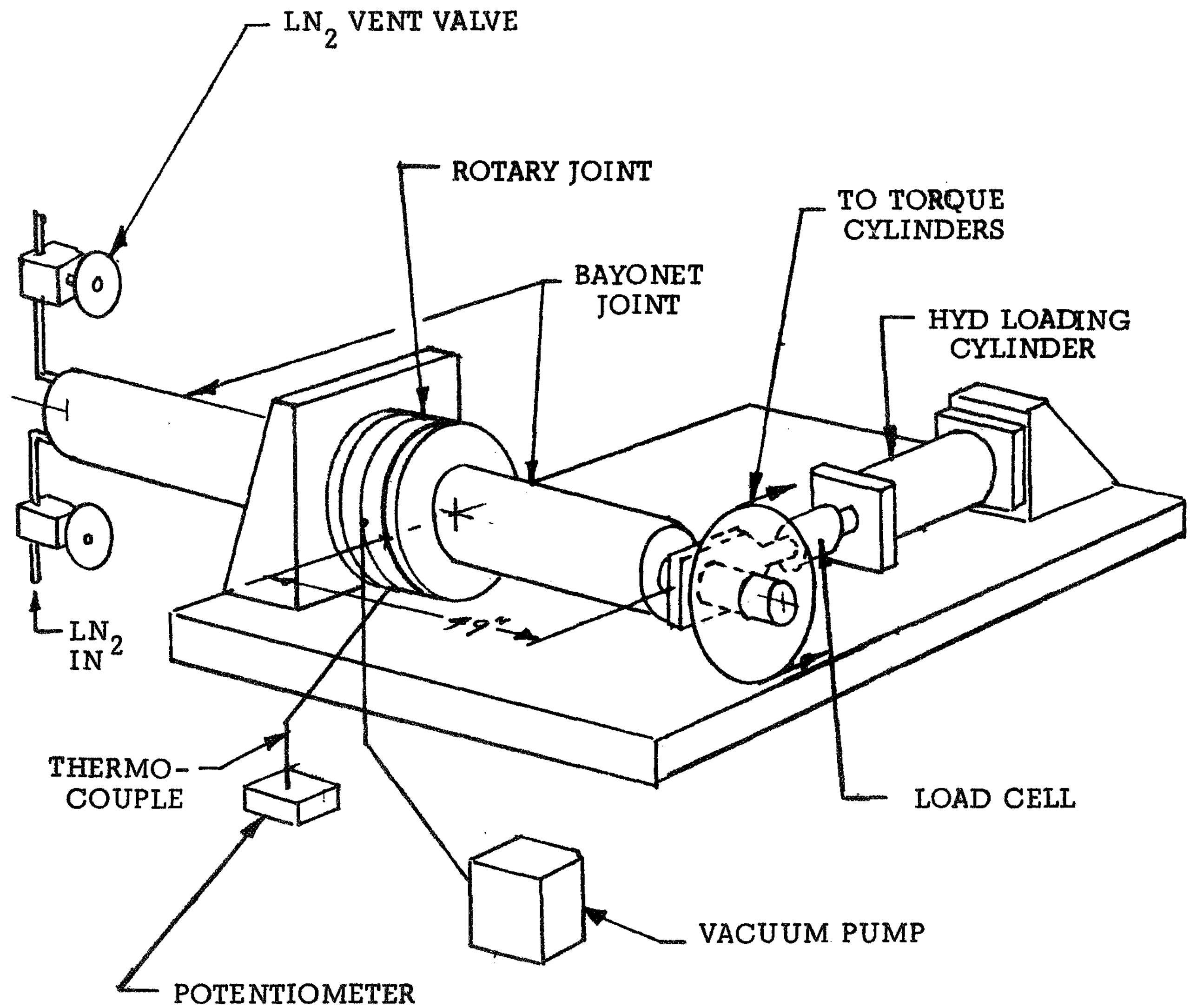
Data Sheets and copies of Sanborn tapes are included.



PRESSURE TEST SET-UP

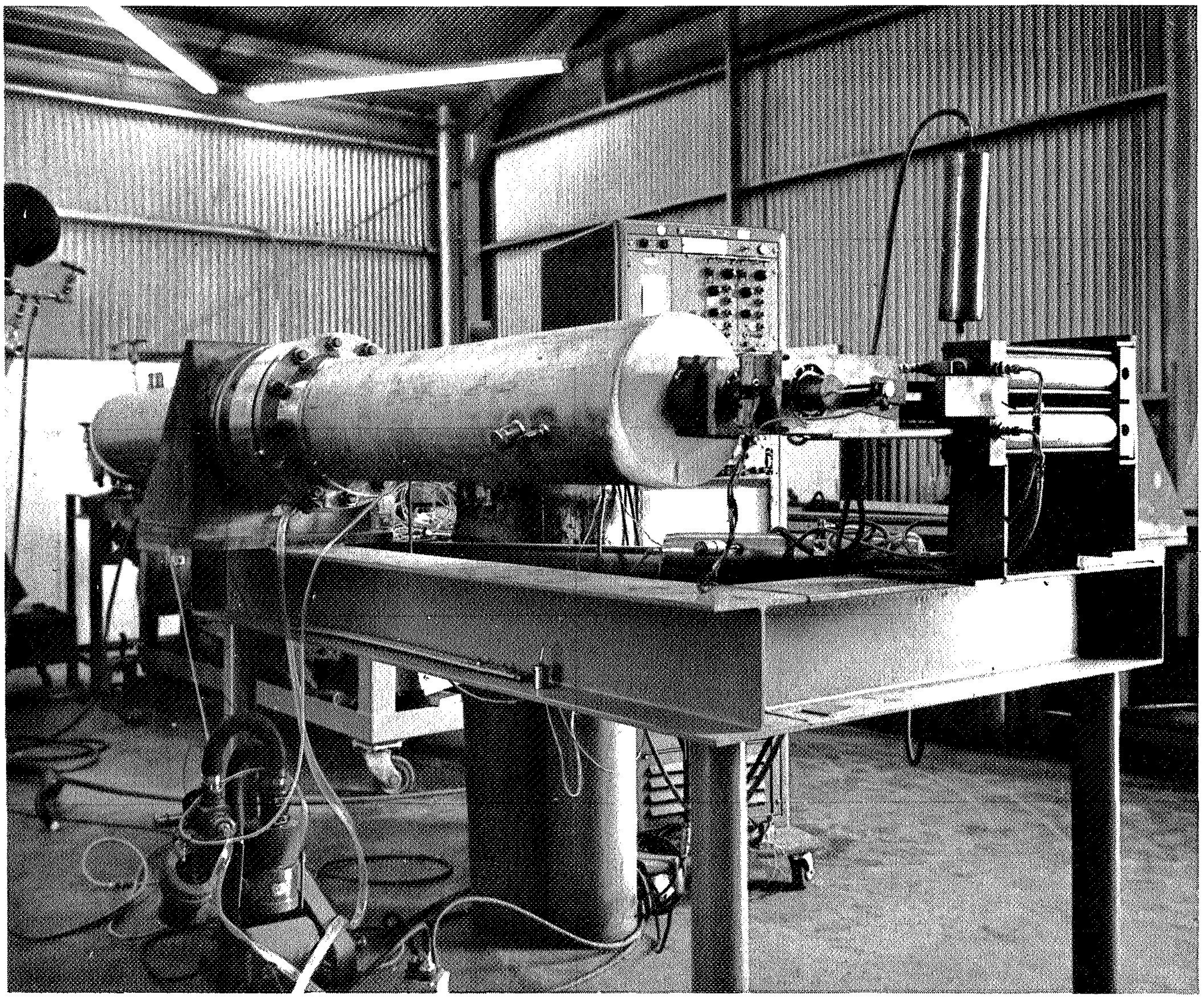
TEST REPORT

FIGURE 2



BENDING MOMENT & TORQUE TEST SET-UP

TEST REPORT
FIGURE 3



Bending Moment/Torque Fixture Photograph

Bending Moment vs Rotational Torque Test Setup for
8-Inch Rotary/Bayonet Joint Assembly

DESIGN VERIFICATION TEST

TEST DATA SHEET

Type of Test: Functional #1 — Date of Test: 12 September 1969

Part Name: Rotary Joint 8" — Part Number: 8-100089

Test Procedure: 8-480086 (Para. 5.3) — Part Serial Number: --

NOTE: Enter in REMARKS column test prior to this Functional Test

		REMARKS
A.	**Pressure**	
	Test Pressure 140	"O" Leak
	Test Media GN_2	
	Duration of Test 5 Min.	
B.	**Leakage - Static**	
	Test Pressure 140 psig	O SCIM Leakage
	Test Media GHe over LN_2	Stabilized Temp at Point "A" on
	Test Temperature -320°F	rotary = 42°F
	Duration of Test 1 Min	
C.	**Leakage - Dynamic**	
	Test Pressure 30 psig	1. 1.4 SCIM Leakage
	Test Media GHe over LN_2	220 psi on Cyl to rotate @ 2.3 RPM
	Test Temperature -320	= 250 Ft.# Torque
	Cycle Rate 2.3 RPM	
	Duration of Test 1 Min	
D.	**Torque**	
	Test Pressure 30 psig	1. 3 SCIM@ 980 Lb B.M.L. and 2.3RPM
	Test Media LN_2	2. 1.6 SCIM @ 1400 Lb. B.M.L. &
	Test Temperature -320	2.3 RPM
	Bending Moment ---	3. 5 SCIM @2100 Lb. B.M.L. @
	Torque Average = 242 Ft. #	2.3 RPM
	Cycle Rate	TOTAL CYCLES - 50

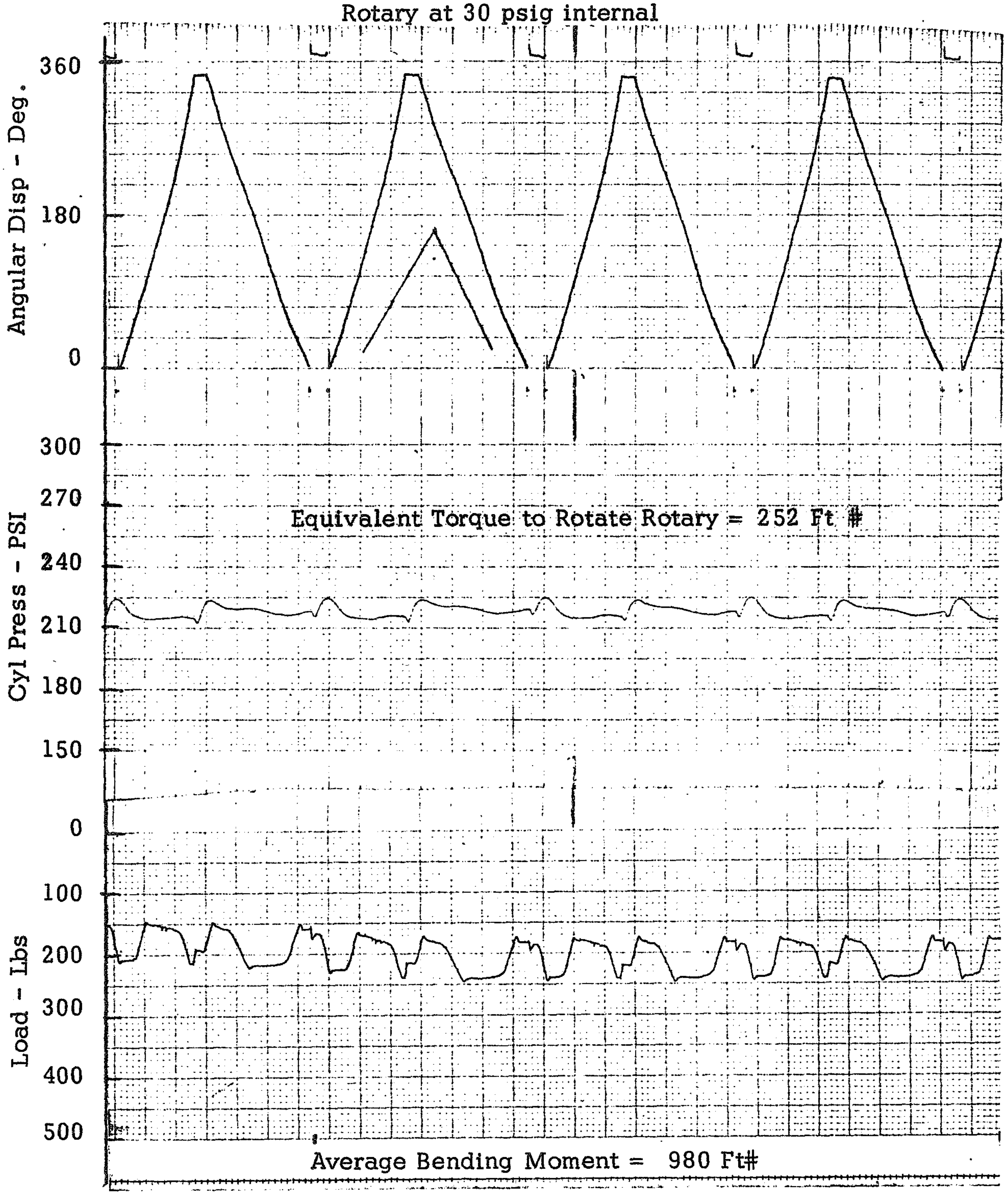
Test Technician /s/ L. Mc Knight — Test Engineer /s/ R. C. Mursinna

Date of Test
12 September 1969

Functional Test #1

Paragraph D.1

Rotary at 30 psig internal



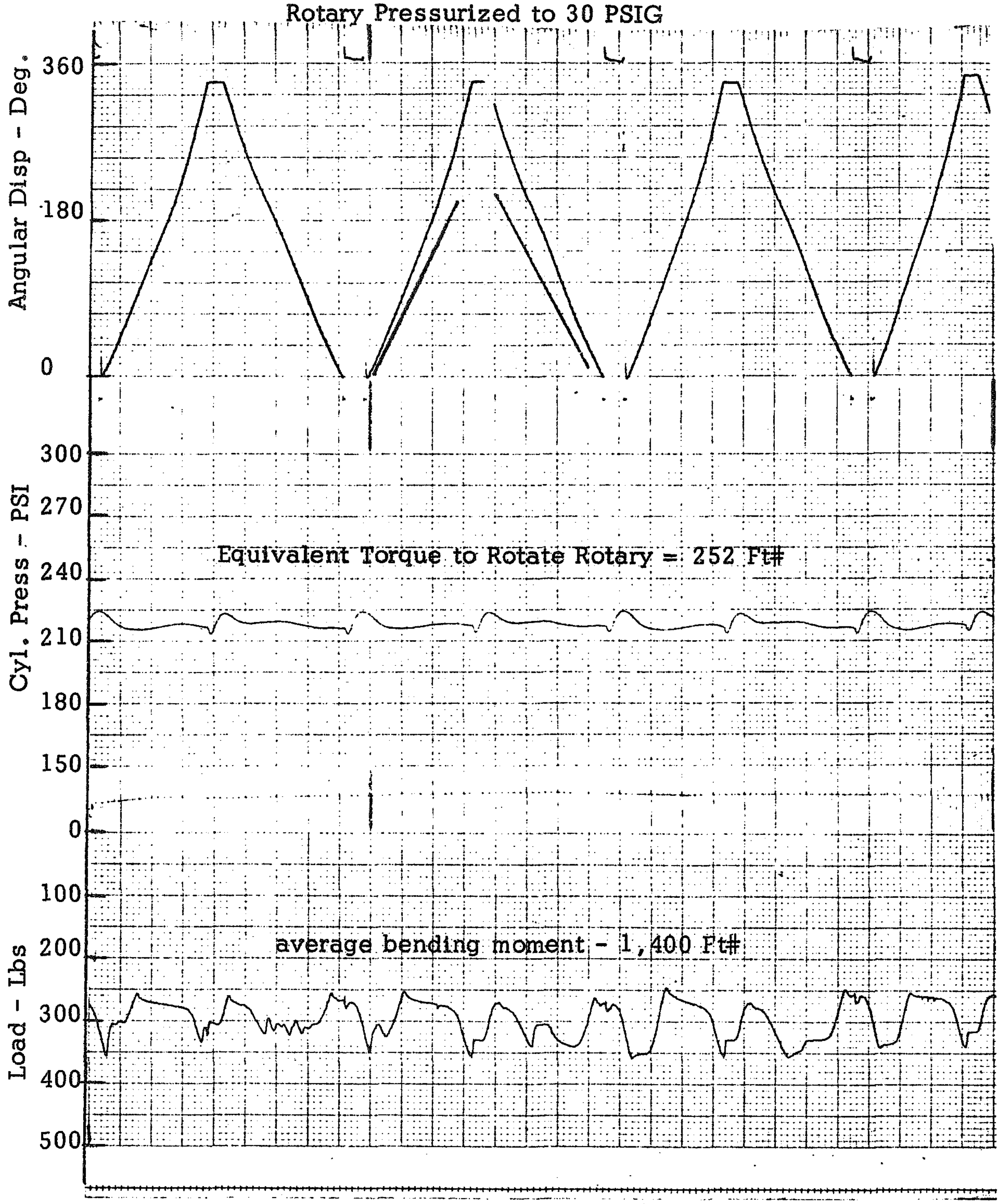
Sanborn Tapes, 8 inch Unit

Date of Test
12 September 1969

Functional Test #1

Paragraph D.2

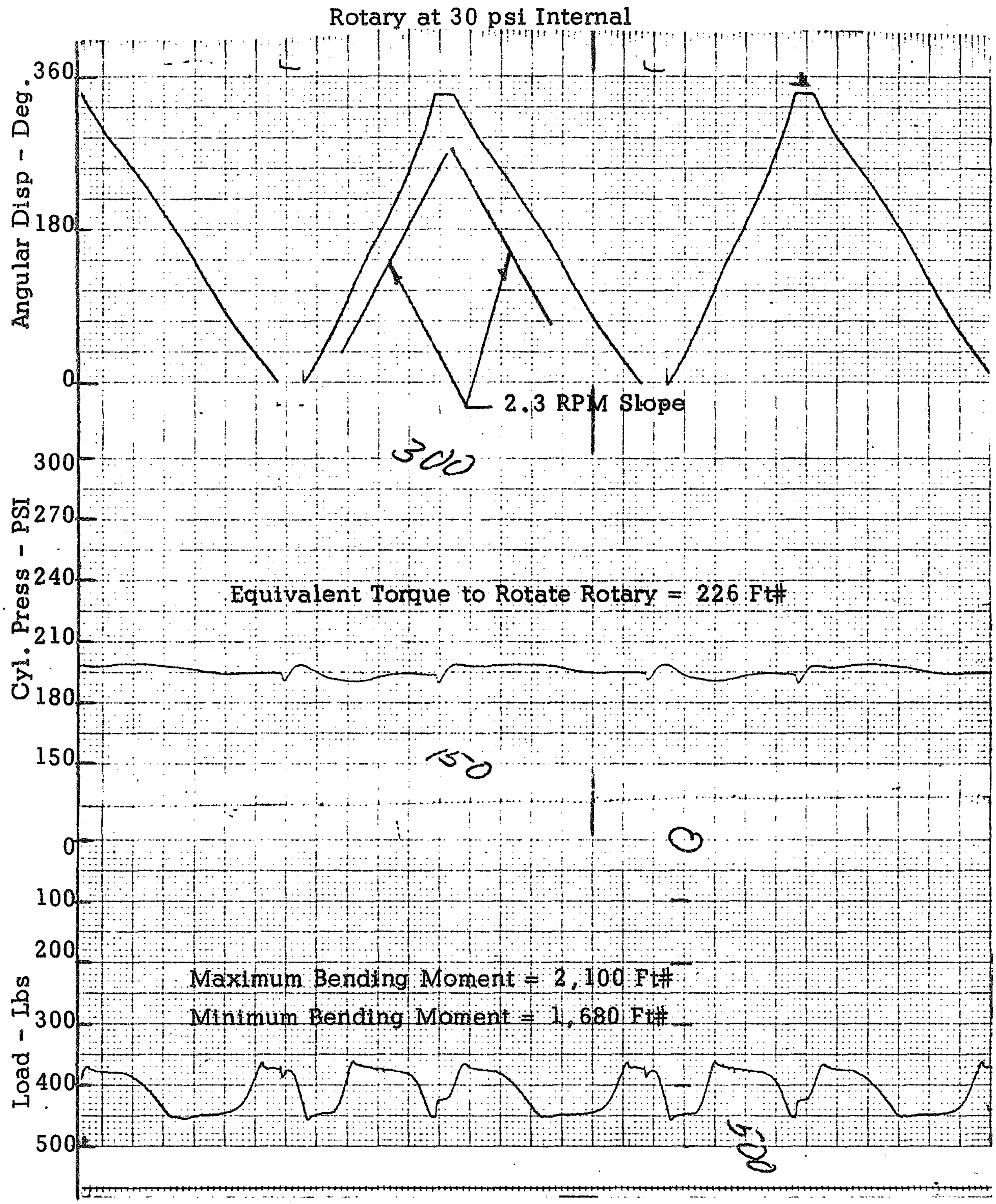
Rotary Pressurized to 30 PSIG



Sanborn Tapes, 8 inch Unit

Date of Test
12 September 1969

Functional Test #1
Paragraph D.3



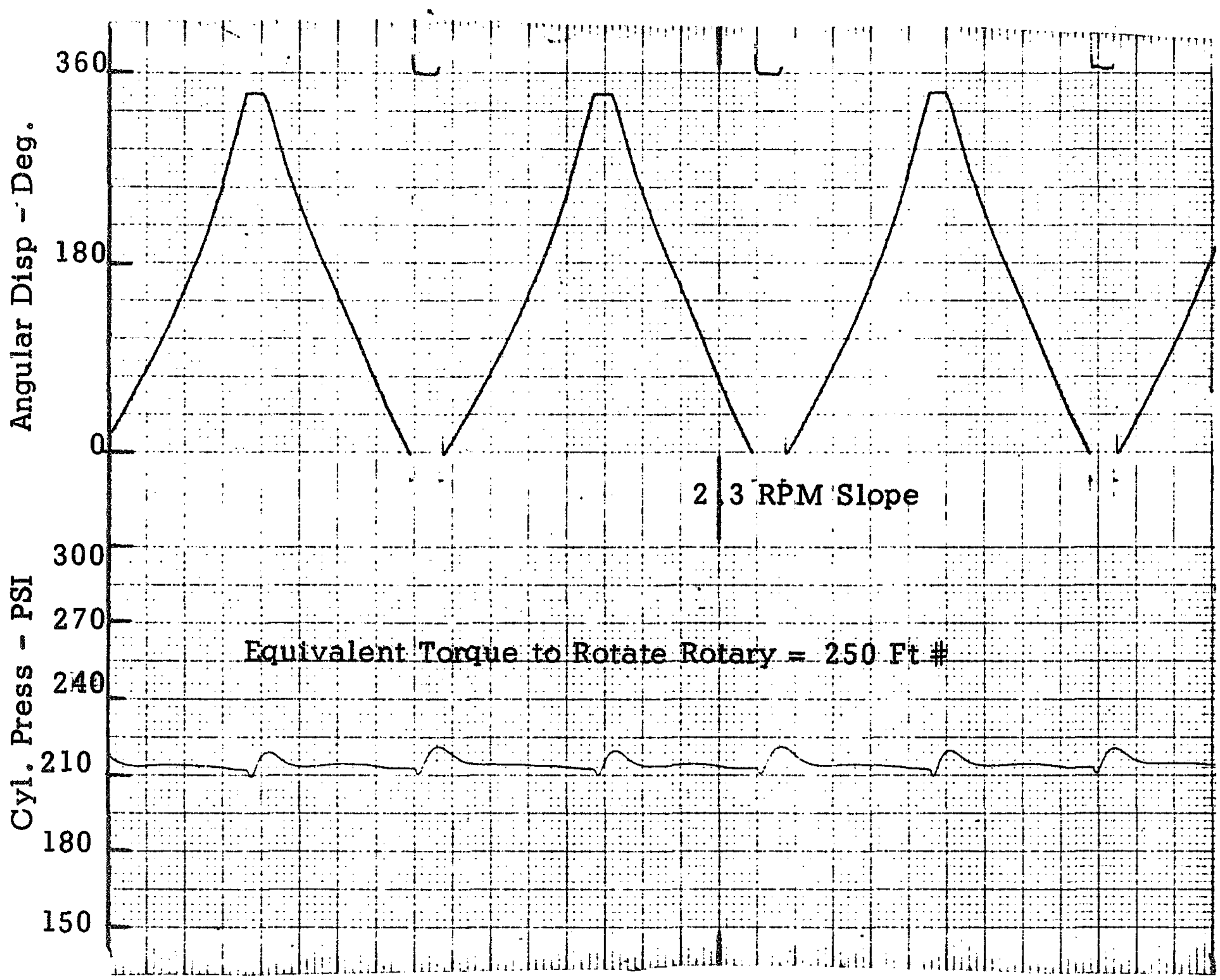
Sanborn Tapes, 8 inch Unit

Date of Test
12 September 1969

Functional Test #1

Paragraph C.1

Rotary at 30 psig Internal



Sanborn Tapes, 8 inch Unit

4.2.1.5.4 Pressure - Proof

Test Requirements

The test requirements stated that the test specimens not deform or malfunction when subjected to 285 psi proof-pressure.

Test Procedure

The 8 inch unit was not subjected to proof pressure until the end of the test program so as not to Brinell the last set of compliant bearings. Due to a lack of funding it was not subjected to this test.

Test Results

None.

Test Data

None.

4.2.1.5.5 Heat Leak Tests

Test Requirements

It is required that the test specimens be subjected to a heat leak test to determine whether or not they fall under the LO_2 system heat leak of 282 Btu/hr. for the 6 inch unit and 355 Btu/hr. for the 8 inch unit.

Test Procedure

The test item was installed in a test fixture as shown in Figure 4. The test item was filled with LN_2 until it has stabilized. The heat leak was determined by LN_2 boil-off over a four (4) hour period. A totalizing flow meter was used to determine the GN_2 loss. A thermocouple was used to determine GN_2 temperature as it entered the flow meter. Another was used to measure specimen temperature. One (1) hour after the test item has stabilized and each hour thereafter for the duration of the test, the GN_2 loss, GN_2 temperature, and specimen temperature, box temperature barometric pressure were recorded. The gas loss was corrected to standard pressure and temperature. Applicable cable conversion calculations were made to determine the heat leak in terms of Btu/hr. for LO_2 temperatures. The tare heat leak for this test was obtained during the heat leak test of the bayonet (Task VII).

A functional test following the heat leak was not performed since the torque test does the same thing.

The heat leak test set-up was photographed and is included along with the bayonet tare leak set-up.

Test Results

In order to compare the actual heat leak with the calculated heat leak, the recorded values were corrected as follows:

For the LO_2 System: [Test item ambient temperature (+100°F) - Liquid oxygen temperature (-297)] ÷ [Average test item temperature as measured on top and on bottom - Liquid nitrogen temperature (-320)] X Heat leak.

Example: For 8 inch unit + bayonet

$$\frac{100 - (-297)}{68 - (-320)} \text{ X } 528 = \frac{397}{387} \text{ X } 528 = 540 \text{ Btu/hr.}$$

The corrected heat leak of the bayonet assembly from Task VII Test Report = 227 + 83 = 310 Btu/hr.

Total 8 inch rotary heat leak = 540 - 310 = 230 Btu/hr.

Test Data

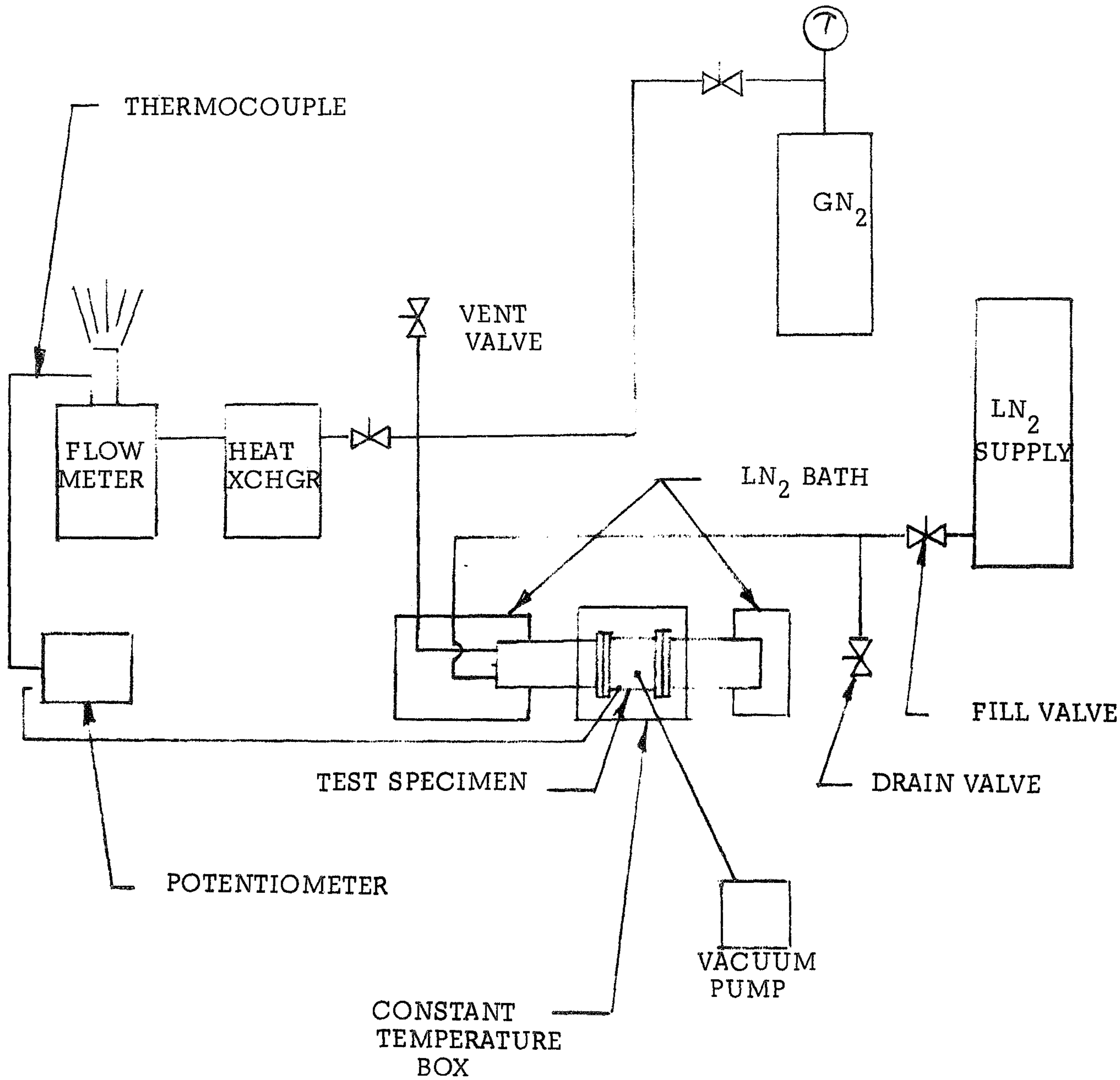
The data sheets for the 8 inch rotary/bayonet assembly are presented on the following pages.

Table I summarizes the corrected measured heat leak values and compares them with the calculated data.

TEST REPORT - TABLE I

Insulator	Line Size	Calculated	Actual
CTL Dixie	6 Inch	282	---
CTL Dixie	8 Inch	355	230

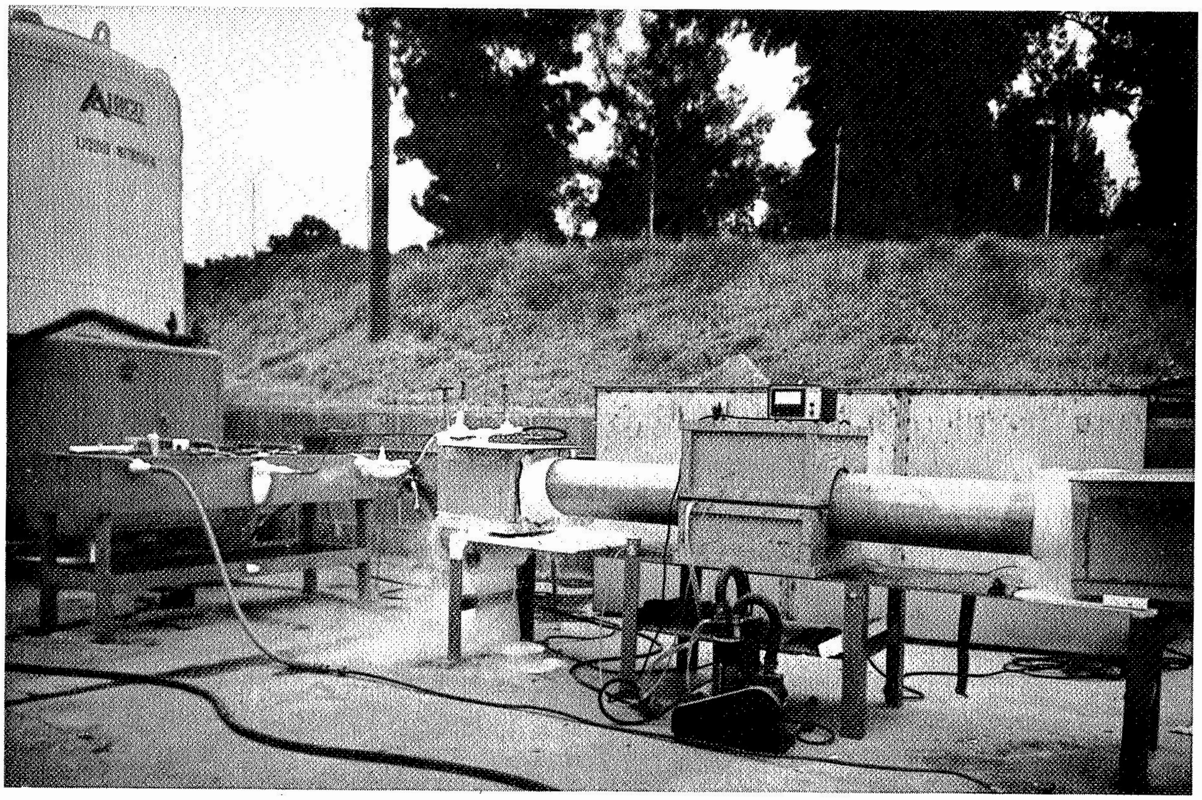
Since the LO_2 heat leak values for the rotary are so close to the LO_2 heat leak for the bayonet joint, the LH_2 system heat leaks can be proportioned from the bayonet joint values.



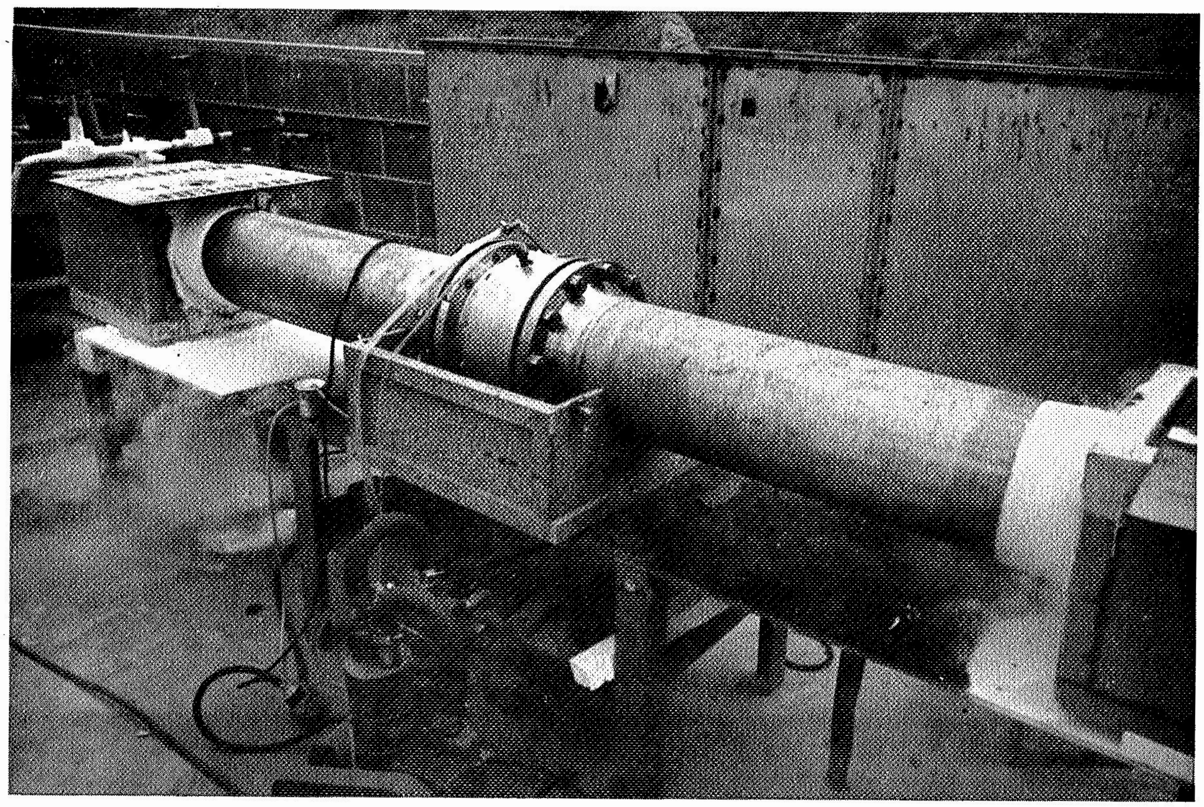
HEAT LEAK TEST SET-UP ROTARY JOINT

TEST REPORT
FIGURE 4

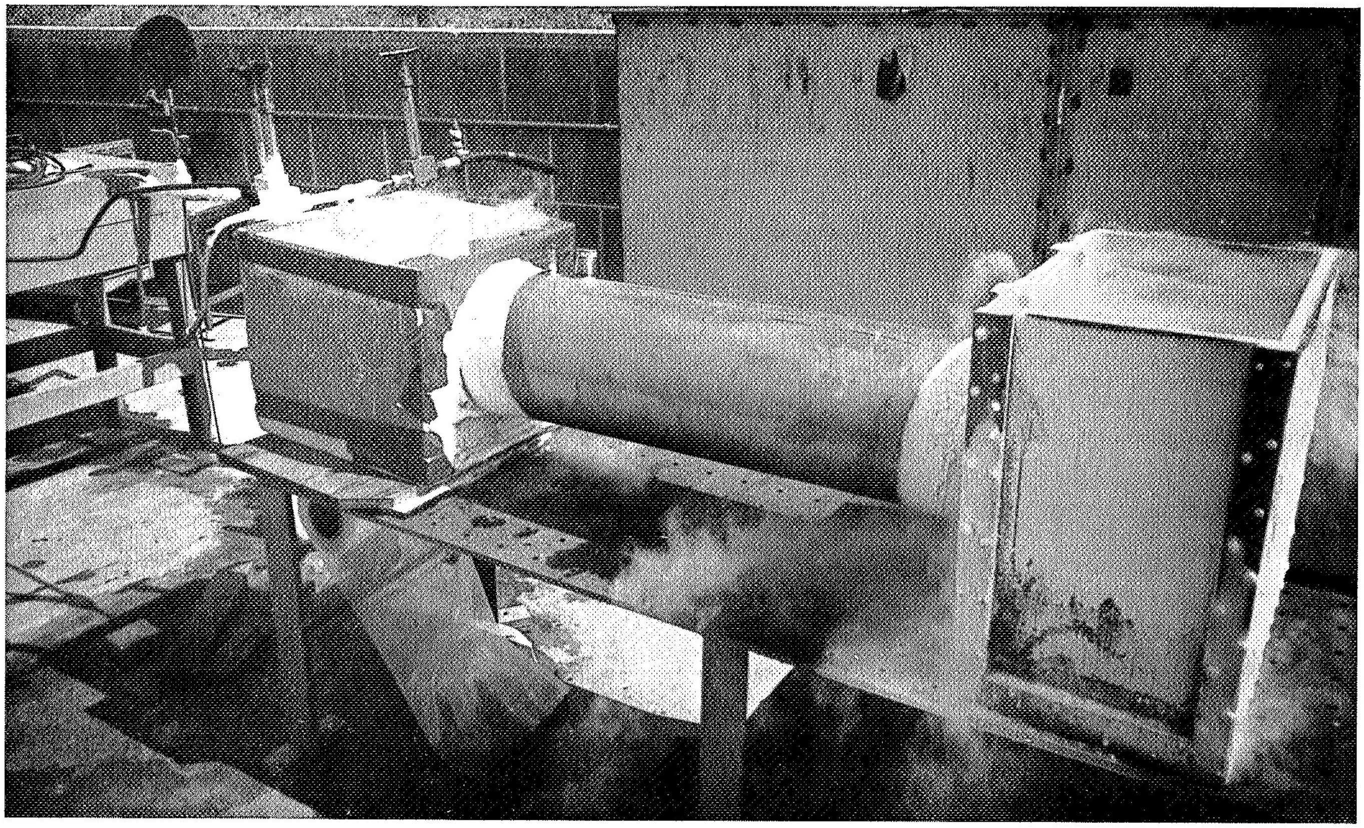
Test Report, Figure 4



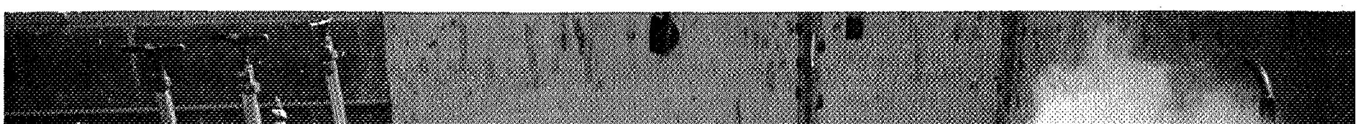
Heat Leak Test Setup For 8-Inch Bayonet/Rotary Assembly



Heat Leak Test Set-up



Tare Heat Leakage — 8-Inch Bayonet



Tare Heat Leakage Test — 8-Inch Bayonet

Tare Heat Leak Test

DESIGN VERIFICATION TEST

TEST DATA SHEET

Type of Test Heat Leak (CTL Dixie Insul) Date of Test 17 September 1969

Part Name Bayonet & Rotary Joints 8" Part Number 8-030928 / 8-100089

Test Procedure 8-480086, Para. 5.5 Part Serial Number ---

FLUID LN_2; HEAT OF VAPORIZATION = 85.7 Btu/#

Time	GN_2 Loss	GN_2 Temp	Barometric Pressure	Remarks
11:00	1.40 CuFt/Min	$\underline{38^{O}F}$ $V = 12.975$ $\frac{CuFt}{\#}$	29.96	"O" SCIM Seal Leak
	Boil-off = .108 #/Min			Unit Temp = $69^{O}F$
				Box Avg Temp = 67.2
12:00	1.3 CuFt/Min	$\underline{58^{O}F}$ $V = 13.498$ $\frac{CuFt}{\#}$	29.96	Unit Temp = $68^{O}F$
	Boil-off = .097 #/Min			Box Avg Temp = 68.0
1:00	1.5 CuFt/Min	$60^{O}F$ $\overline{V} = 13.550$	29.94	Top Unit Temp = $71^{O}F$
	Boil-off = .11 #/Min.			Box Avg Temp $71^{O}F$
1:45	Temp on Bottom of	Temp on Top of	Temp on Line	
	Joint $59^{O}F$	Joint $80^{O}F$	in sun $99^{O}F$	
2:00	1.3 CuFt/Min	$61^{O}F$ $\overline{V} = 13.557$	29.88	Bottom of Unit Temp $62^{O}F$
	Boil-off = .096 #/Min			Top of Unit Temp $73^{O}F$
				Box Avg Temp $67^{O}F$
3:00	1.4 CuFt/Min	$63^{O}F$ $V = 13.625$	29.84	Unit Temp $76^{O}F$
	Boil-off = .103 #/Min			Box Avg Temp $69^{O}F$

Average #/Min = .103

Total Heat Leak = .103 X 85.7 X 60 = 528 Btu/Hr.

Test Technician ________ Test Engineer /s/ R. C. Mursinna

4.2.1.5.6 Torque Test

Test Requirements

It was required to determine how the rotary joint torque is affected by pressure, bending loads, and temperature conditions. This test also establishes the radial load carrying capability of the compliant bearings on the 8 inch unit.

Test Procedure

The test item was installed in a test set-up as shown in Figure 3. This test was conducted in two parts: (1) at room ambient temperature, and (2) at cryogenic temperature.

A. Part I

The test item was pressurized using the test set-up shown in Figure 2 to 30^{+0}_{-5} psig with gaseous nitrogen. The rotary joint was cycled at 2.3 RPM during the test. The bending moment was increased in 200 ft-lb increments from 0 - 2500 ft-lbs. At each 200 ft-lb increment, the torque and leakage was measured and recorded.

B. Part II

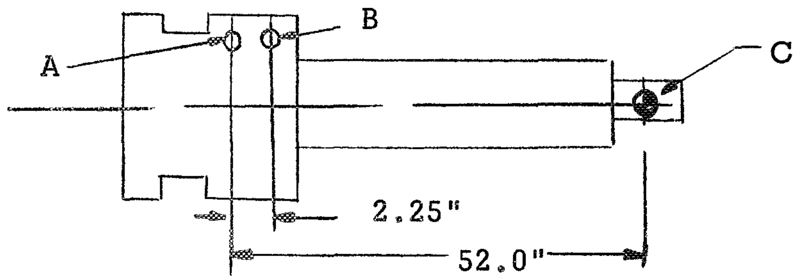
The test item was filled with liquid nitrogen and allowed to stabilize. The stabilized temperature was determined by a thermocouple. After stabilization, the liquid flow was shut off. The rotary joint was cycled at 2.3 RPM during the test. The vent valve (Figure 3) was continuously monitored while pressurizing the test item in 10 psi increments from 30 to 140 ±10 psig (8 inch unit), 0 to 190 ±10 psig (6 inch unit). Each 10 psi increment, the torque, and leakage were measured and recorded. At the conclusion of the test, the test item was drained and returned to ambient conditions.

Test Results

The 8 inch unit performed satisfactorily during the test except that the leakage was running above the 3 SCIM allowable during the cold test. The rotation torques were well within the 600 ft-lb. design goal. The maximum value recorded was 260 ft-lbs.

The cyclic nature of the bending load and torque pressure shown on the data sheets was caused by a crooked support shaft on the bayonet. The data sheet values were the maximum of the maximum and minimum values per cycle.

The recorded moments are referenced from Bearing "A" shown in the sketch below.



The moments were calculated as follows:

Bending load at C X 52.0" = Moment (ft-lb).

The maximum radial load on the bearings are:

Bending load at C X 52 = Reaction
load at bearing B X 2.25

$$\frac{540 \text{ lb. X } 52}{2.25} = \text{radial load on B} = 12{,}500 \text{ lb.}$$

Since this bearing did not indicate any Brinelling, the compliance was adding to its load carrying capability. This bearing, with the compliance working properly, should be capable of carrying a 22,000 lb. radial load minimum.

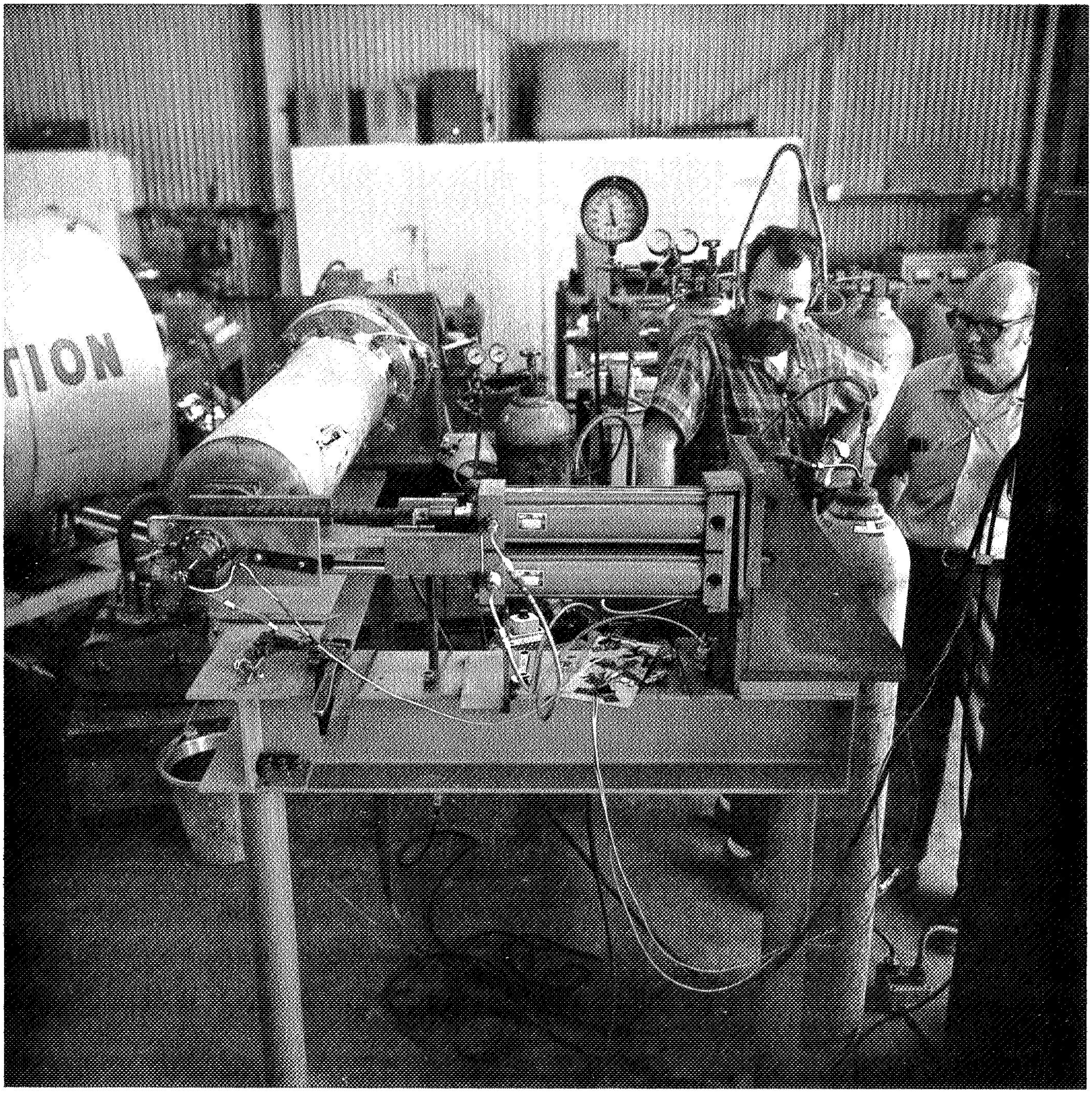
Taking moments about Bearing "B", the radial load on Bearing "A" is calculated as follows:

$$\frac{540 \text{ X } 49.75}{2.25} = \text{radial load on A} = 11{,}900 \text{ lbs}$$

The rotational torque varied only slightly during the LN_2 pressure test with a maximum of 232 ft-lbs. at 140 psig. The maximum leakage was 10.4 SCIM rotating and 11.7 SCIM static.

Test Data

Data sheets and a photograph of the test set-up are included, along with portions of the Sanborn tapes.



Torque versus Bending Moment Test Set-up

Torque vs Bending Moment Test Setup After
Heat Leak Test

DESIGN VERIFICATION TEST

TEST DATA SHEET

Type of Test Torque Date of Test 30 September 196[illegible]

Part Name Rotary Joint Part Number 8-100089

Test Procedure 8-480086 Para 5.6 Part Serial Number ---

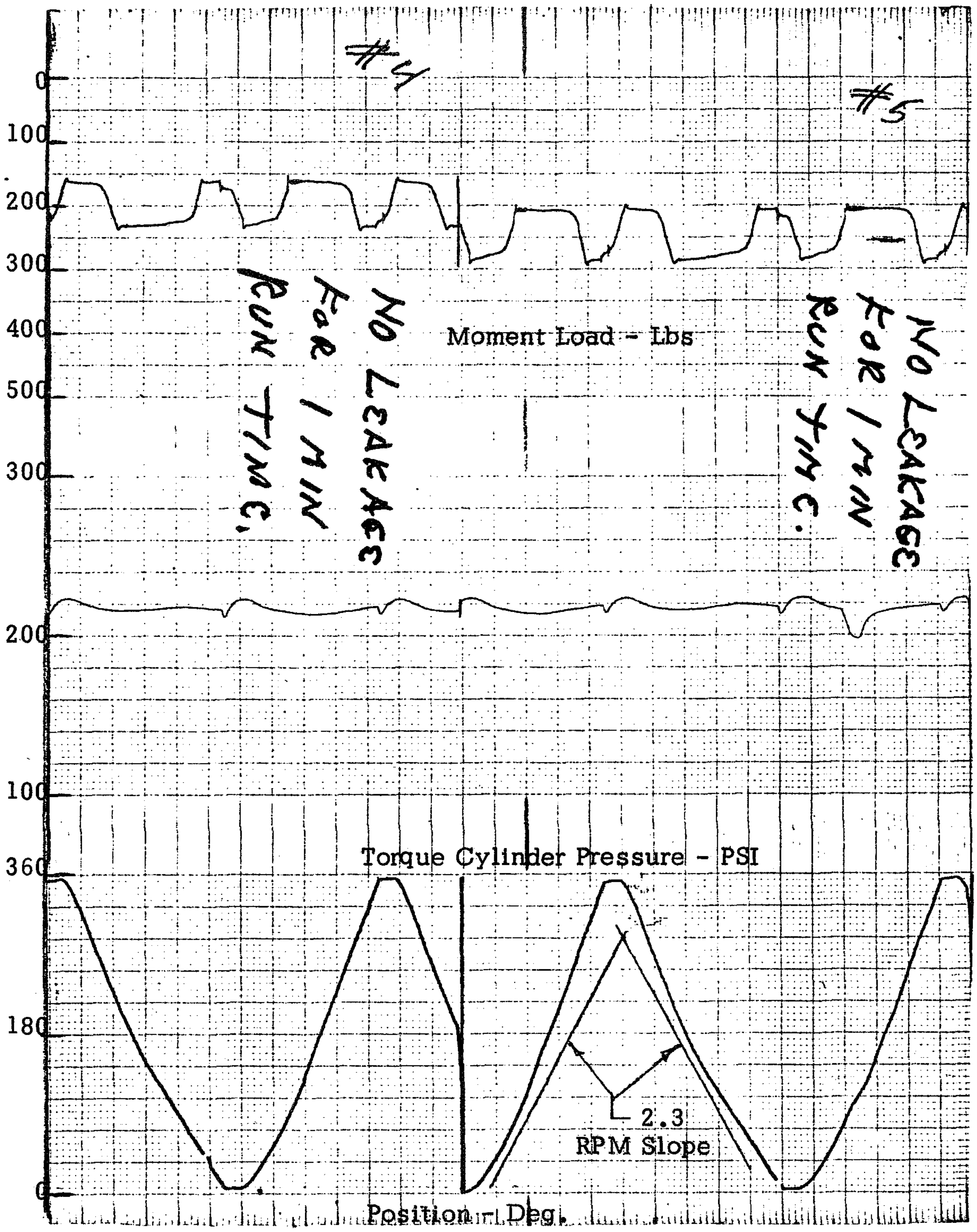
A. At Ambient Temperature

Test Pressure 30 PSIG Cycle Rate 2.3 RPM

Test Media GN_2 @ 70°F Duration of Test [illegible]0 Min (52 Cycles)

Ft # Bending Moment	Avg Ft # Torque	Leakage (SCIM)	Remarks
233	254	11.6	See sample of Sanborn Test Data
467	246	3.0	
700	246	2.1	
933	254	0.0	
1167	254	0.0	
1400	261	0.0	
1633	254	0.0	
1867	248	0.0	
2100	258	0.0	
2333	260	0.0	Unit performed satisfactorily.
~~550~~			
~~600~~			
~~650~~			
~~700~~			
~~750~~			
~~800~~			
~~850~~	*Bending mo	ment loads were	increased based on data taken
~~900~~	during first	Functional test.	

Torque Test @ 70°F



Sanborn Tape

B. At Cryogenic Temperature

Specimen Temperature $70^{\circ}F$ Cycle Rate 2.3 RPM

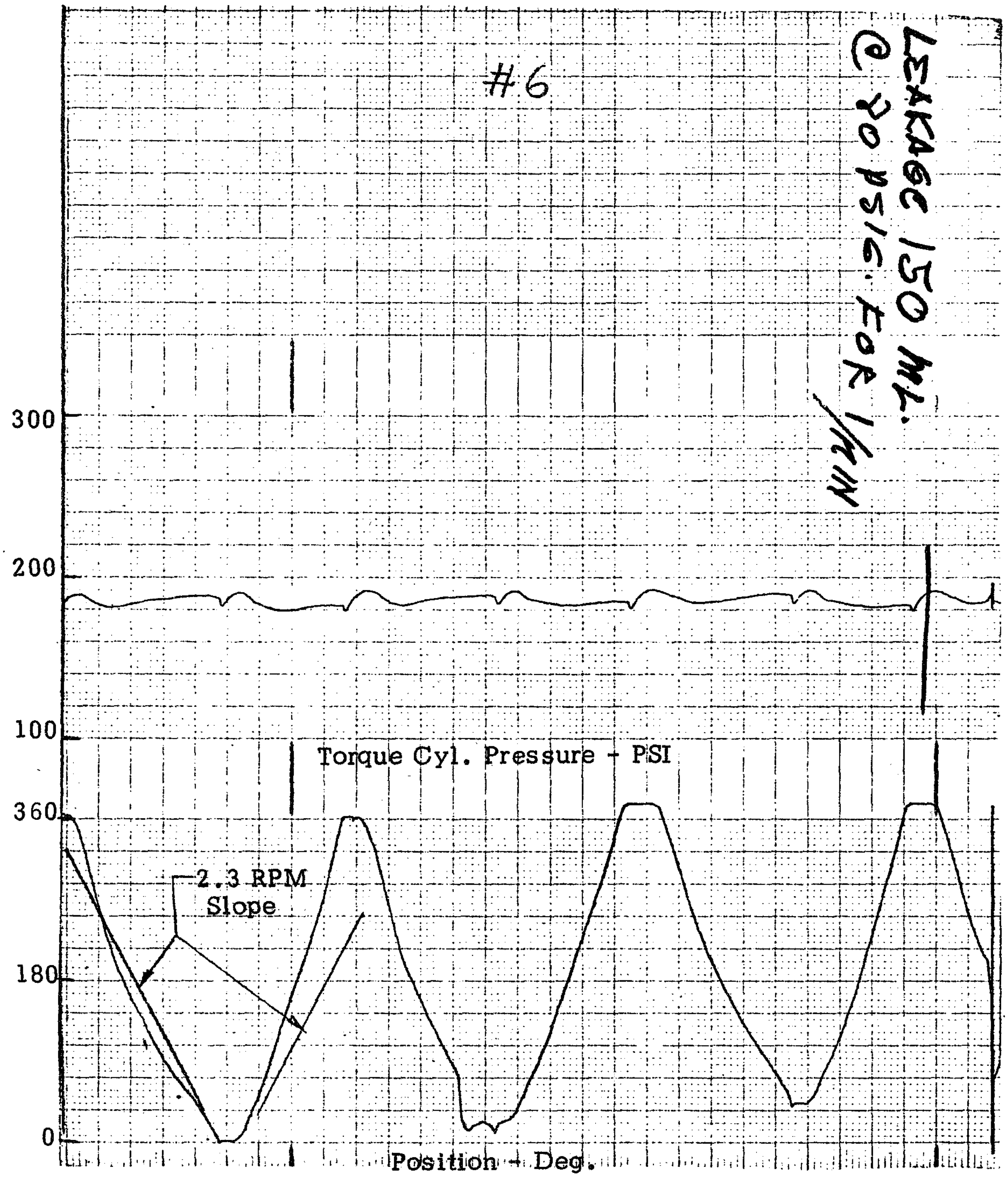
Test Media LN_2 Duration of Test 25 Min (37 Cycles)

Pressure (psig)	Avg Ft. # Torque	Leakage (SCIM)	Remarks
10			Minimum pressure was 30 psi due
20			to high boil-off
30	195	7.2	(See Sample Sanborn Trace)
40	197	6.7	
50	216	8.5	Unit performed satisfactorily
60	216	8.5	except for leakage
70	216	9.0	Allowable is 3 SCIM. Higher
80	209	9.2	leakage is attributed to Teflon
90	216	8.0	seal sluffing and getting under
100	220	9.2	seal face.
110	225	10.4	
120	230	9.0	
130	223	9.8	
140	232	10.4	

~~150~~	ROTATIONAL POSITION LEAK CHECK (STATIC)			
~~160~~	0°	90°	180°	270°
~~170~~	11.7 SCIM	10.0 SCIM	8.5SCIM	8.5 SCIM
~~180~~				
~~190~~				

Test Technician ________ Test Engineer /s/ R. C. Mursinna

Torque Test - LN_2



Sanborn Tape

4.2.1.5.7 Sand and Dust Test

Test Requirement

Reference: AMETEK/Straza Specification No. 8-480086, Paragraph 5.8.

The sand and dust test was performed to determine the resistance of the test item to blowing fine sand and dust particles. This test was run only on the 8 inch unit. This test, along with the salt fog and vibration, was conducted at Ogden Technology Laboratories.

Test Procedure

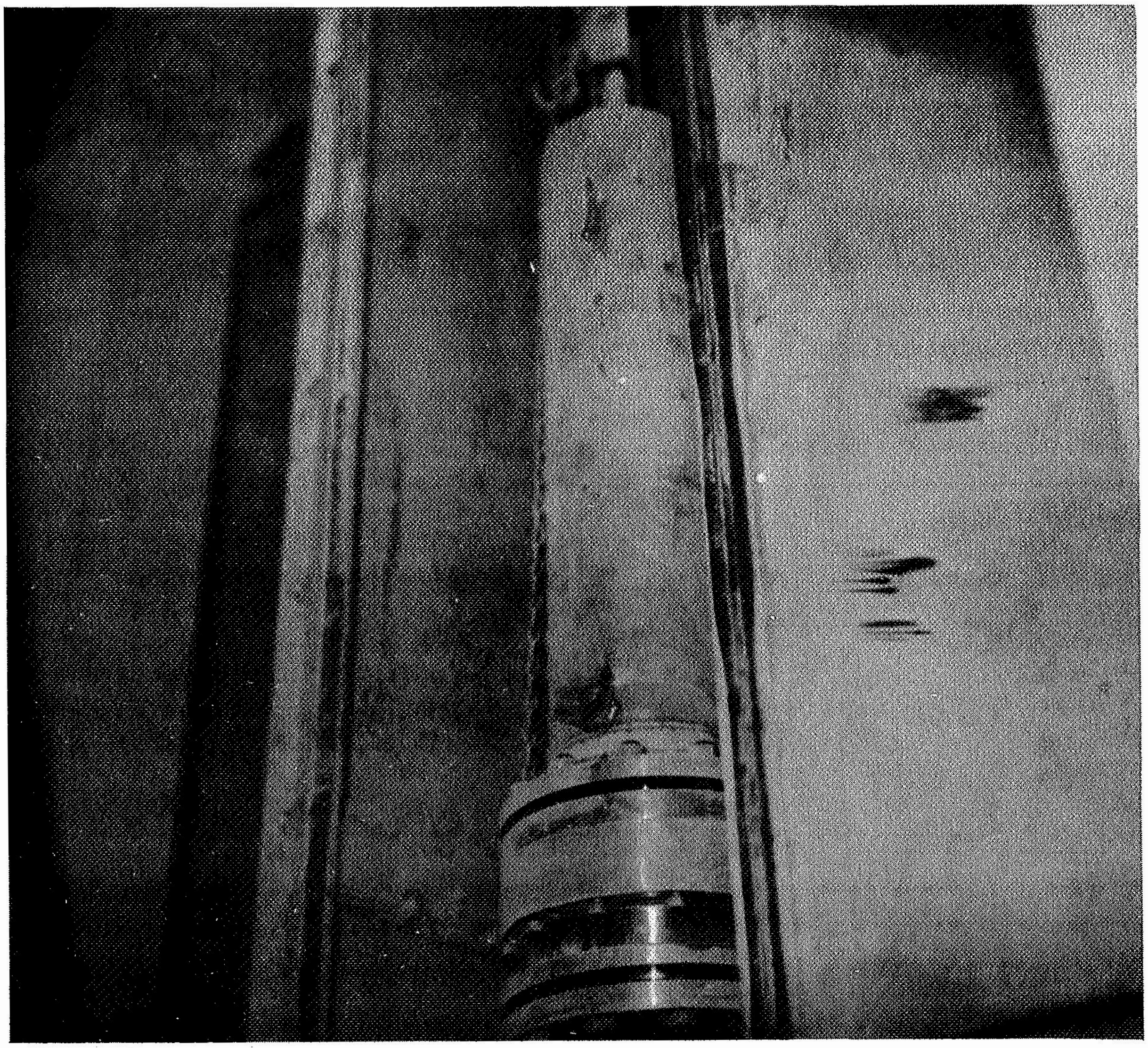
The test specimen was placed in a test chamber as shown in Photograph No. 5. The chamber was programmed to provide a sand and dust environment with a sand to air ratio of 0.1 to 0.25 grams per cubic foot, with an air velocity of 100 to 500 feet per minute. The test specimen was exposed to this environment at a temperature of $+77 \pm 2^{\circ}F$ for two (2) hours, then the temperature was raised to $+160 \pm 2^{\circ}F$ and the specimens subjected to a further two (2) hours under these test conditions. At the conclusion of the four (4) hour exposure to sand and dust particles, the specimen was returned to room ambient and a functional test was performed.

Test Results

Visual inspection on completion of the sand and dust test did not reveal any damage or permanent deformation. During the performance of the functional test, no leakage was observed with the specimen pressurized to 140 psig using gaseous Nitrogen at ambient temperature. However, excessive leakage was observed at 140 psig at a temperature of $-320^{\circ}F$. The functional test was rerun at 30 psig at liquid Nitrogen temperature ($-320^{\circ}F$). Leakage was still encountered and the results are to be found on data sheets. The unit was disassembled and it was found that excessive sluffing of the seal caused the leakage. Upon further investigation it was found that the seal gland was crushing the seal beyond the recommended dimension. The gland was machined to allow for proper seal loading and the unit was placed back into test.

Test Data

The test data sheets were extracted from Ogden's test report No. B-69478 and are presented on the following page.



Test Report Photograph

Test Report
Photograph No. 5

OGDEN TECHNOLOGY LABORATORIES, INC.

Date 10-15-69 SAND AND DUST DATA SHEET Job Number 26942

Customer AMETEC / STRAZA Page Number ①

Specimen ROTARY JOINT 8 inch Part No. – Serial No. –

Specification No. S-480086 Para.No. 58

Preparation of Specimen(s) —

Protective Covering on Non-Tested Parts —

Vents, Ports, Connectors, etc. Capped: Yes ✓ No ____ Remarks —

Support Method SUSPENDED

Orientation of Specimen(s) VERTICAL

Chamber Controls: Sand and Dust Density 0.1 TO 0.25 grams/cubic foot
Wind Velocity 100 - 500 feet/minute
Relative Humidity 0 - 28 percent
Temperature AMB. TO +165 °F

Elapsed Time (hours)	Sand And Dust Density (grams/cu.ft.)	Air Velocity (ft/minute)	Temperature (°F)	Relative Humidity (%)
0	0.2	350	78	21
1	0.2	350	77	20
2	0.2	350	77	18
0	0.2	350	158	15
1	0.2	350	160	15
2	0.2	350	161	14

Remarks: NONE

Interruptions during test (Explain): NONE

Results: Damage or Deformation: Yes ____ No ✓ (Explain above)
Photograph taken: Yes ✓ No ____
Test Technician MANGINO Test Engineer H. Pegueder
Inspector (Customer or USGov't) none

Date

A. Donco
Q.A. Mgr. Signature

DESIGN VERIFICATION TEST

TEST DATA SHEET

Type of Test Functional Date of Test 18 October 1969

Part Name Rotary Joint Part Number

Test Procedure 8-480086 (Para. 5.3) Part Serial Number 8"

NOTE: Enter in REMARKS column test prior to this Functional Test.

		REMARKS
A.	Pressure	
	Test Pressure 140 ± 10 psig	Sand and Dust Test
	Test Media GN_2	
	Duration of Test 5 Min	Compl WOL
B.	Leakage - Static	
	Test Pressure 140± 10 psig	
	Test Media LN_2	
	Test Temperature $-320^{\circ}F$	
	Duration of Test 5 Min	Extreme leakage encountered, unable to notify Straza rep. Stop testing
C.	Leakage - Dynamic	
	Test Pressure	
	Test Media	
	Test Temperature	
	Cycle Rate	
	Duration of Test	
D.	Torque	
	Test Pressure	
	Test Media	
	Test Temperature	
	Bending Moment	
	Torque	
	Cycle Rate	

Test Technician ________ Test Engineer /s/ Ronald G. Powell

DESIGN VERIFICATION TEST

TEST DATA SHEET

Type of Test: Functional   Date of Test: 30 October 1969

Part Name: Rotary Joint   Part Number: ____

Test Procedure: 8-480086 (Para. 5.3)   Part Serial Number: ____

NOTE: Enter in REMARKS column test prior to this Functional Test.

		REMARKS
A.	Pressure	
	Test Pressure 140 ±10 psig	Sand and Dust
	Test Media GN_2	(Rerun per Straza #1)
	Duration of Test 5 Min	
B.	Leakage - Static	Start rotating 1 RPM
	Test Pressure 82 psig	> 3 SCIM leakage
	Test Media LN_2	
	Test Temperature $-320^{\circ}F$	
	Duration of Test 1 Min	
C.	Leakage - Dynamic	
	Test Pressure 30 psig	> 3 SCIM leakage
	Test Media LN_2	
	Test Temperature $-320^{\circ}F$	
	Cycle Rate 2.3 RPM	
	Duration of Test 5 Min	
D.	Torque	
	Test Pressure 5 psig	Specimen stopped rotating
	Test Media	Frost on exterior or rotating joint
	Test Temperature	
	Bending Moment 300 ' lbs	
	Torque	
	Cycle Rate	

Test Technician /s/ W. O. Lebus   Test Engineer /s/ Ronald G. Powell

DESIGN VERIFICATION TEST

TEST DATA SHEET

Type of Test Functional Date of Test 31 October 1969

Part Name Rotary Joint Part Number

Test Procedure 8-480086 (Para. 5.3) Part Serial Number

NOTE: Enter in REMARKS column test prior to this Functional Test.

			REMARKS
A.	Pressure		
	Test Pressure		Sand and Dust
	Test Media		(Rerun per Straza #2)
	Duration of Test		
B.	Leakage - Static		
	Test Pressure		
	Test Media		
	Test Temperature		
	Duration of Test		
C.	Leakage - Dynamic		Start Vacuum Pump Start Childown
	Test Pressure	30 psig	Rotate specimen 1 RPM
	Test Media	LN_2	
	Test Temperature	$-320^{\circ}F$	
	Cycle Rate	2.3 RPM	
	Duration of Test	5 Min	
D.	Torque		
	Test Pressure	30 psig	
	Test Media	LN_2	
	Test Temperature	-320	
	Bending Moment	300 Ft Lbs	
	Torque	230 foot pounds	
	Cycle Rate	2.3 RPM	

Test Technician /s/ W. O Lebus Test Engineer /s/ Ronald G. Powell

4.2.1.5.8 Salt Fog Test

Test Requirement

Reference: AMETEK/Straza Specification No. 8-480086, Paragraph 5.9.

This test shall be performed to determine the specimens' resistance to a salt fog atmosphere. Only the 8 inch unit was subjected to this test.

Test Procedure

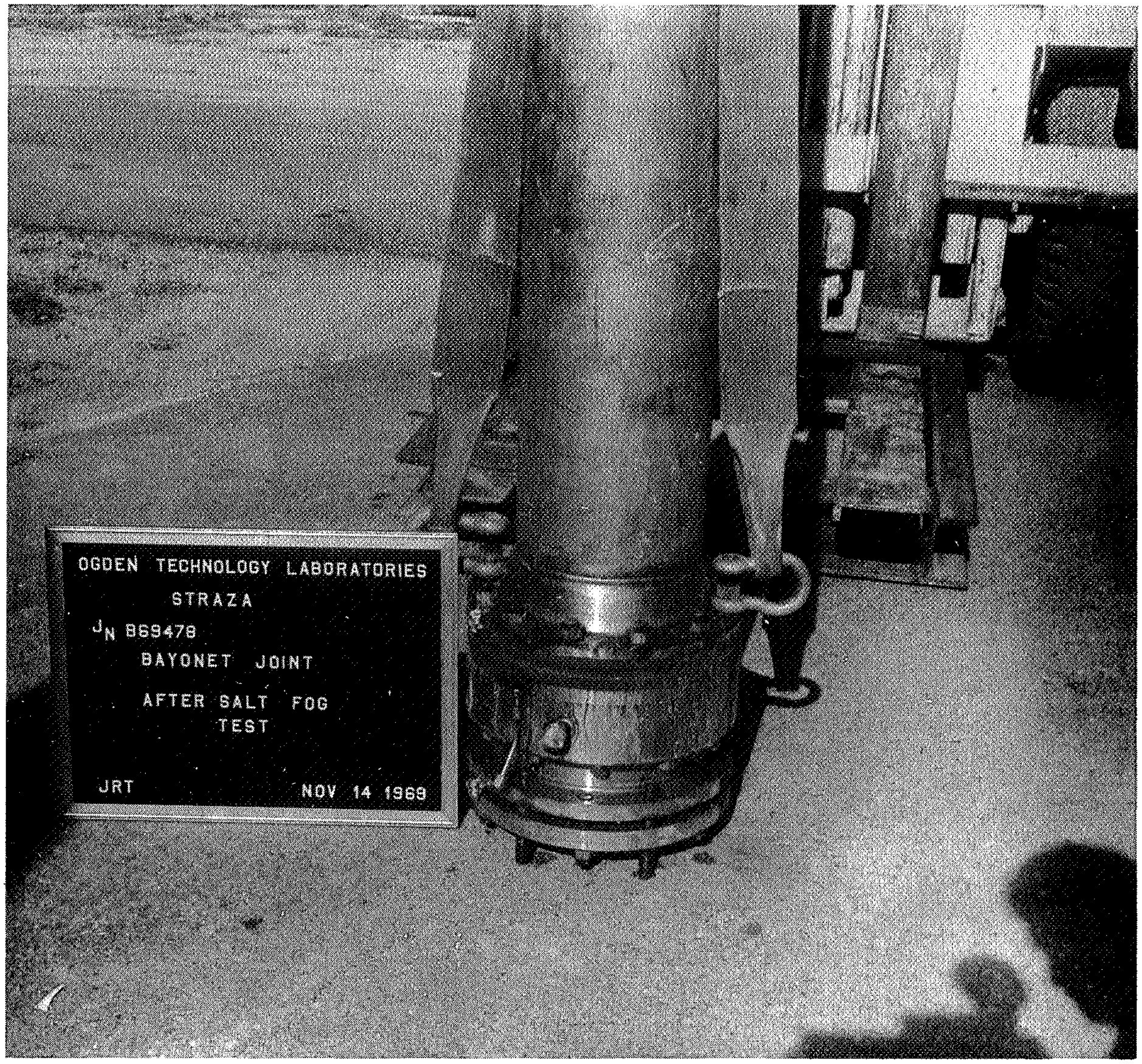
The test specimen was installed in a salt fog chamber in a vertical position as shown in Photograph Number 7. The specimen was then subjected to 240 hours exposure in a salt fog atmosphere of 5% salt and 95% water, at a temperature of 95 $^{+2}_{-4}$°F. On completion of the 240 hour exposure, the specimen was allowed to dry, a visual inspection made, and a functional test performed.

Test Results

Visual examination, post salt fog test, revealed severe oxidation of the flange bolts. Extreme leakage was observed during the functional test at -320°F. The extreme sluffing of the primary Teflon seal continued to cause excessive leakage. This, in turn caused frosting of the joint. Since a new seal fabricated from KEL-F could not be installed in the unit within the remaining program cost and time schedule, the testing was continued.

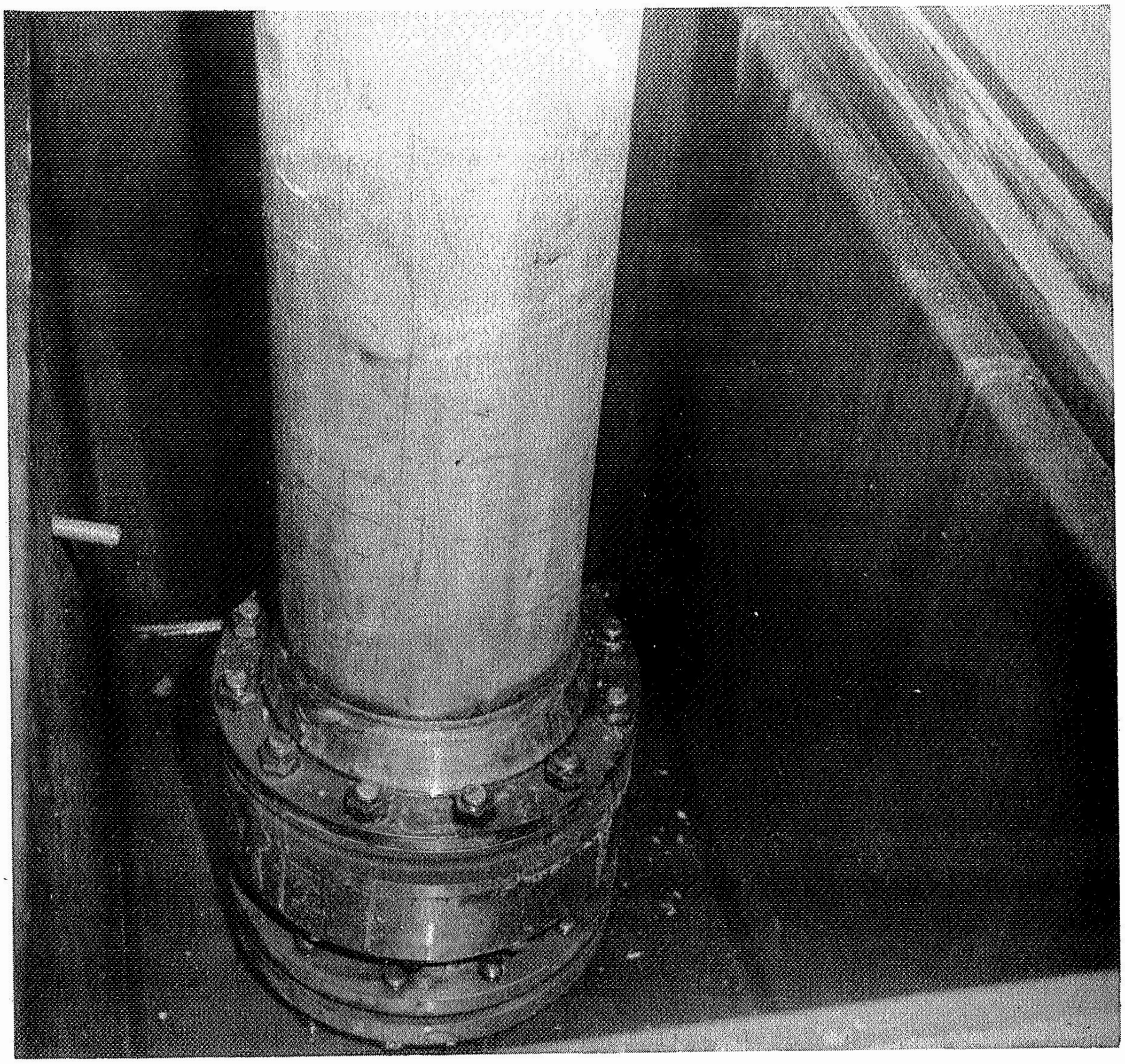
Test Data

The test data sheets taken from Ogden Report No. B-69478 are presented on the following pages.



Test Report Photograph

Test Report
Photograph No. 6



Test Report Photograph

Test Report
Photograph No. 7

UNITED AEROTEST LABORATORIES

Date Nov 4, 1969 SALT SPRAY DATA SHEET Job Number: B69478

Customer AmeTeck / Straza Page Number ________

Specimen Rotary Joint Assy Part No. 8-030930 Serial No. 8"

Specification No. 8-480086 RwB Para. No. 5.9

Preparation of Specimen(s) None

Protective Coating or Covering for Non-Tested Parts None

Vents, Ports, Connectors etc. Capped: Yes X No ____ Remarks ________

Support Method Supports on chamber floor

Orientation of Specimen(s) Vertical

Solution: Salt 5% % H_2O 95% % (by weight)

pH of Solution 6.8 at 95 °F

Specific Gravity of Solution 1.148 at 95 °F

Start date and time Nov 4 1969 0900hrs Nozzle Pressure 12 PSIG

Chamber Temperature 95 °F Water Column Temperature 110 °F

TEST RECORD
(Each 24 Hours)

Elapsed Time (Hours)	Collected Solution (Volume) per 80 square centimeters of Horizontal Surface Area (milliliters per hour)	Collected Solution pH Value	Collected Solution Specific Gravity	Chamber Temperature (°F)
48	0.80	6.6	1.140	95°F

Stop date and time Nov 14 0930 hrs. Test Duration 48 hours

Interruptions (explain) None

Results of Test extreme oxidation of flange bolts but no other apparent damage or degradation

Photograph taken: Yes ✓ No ____

Test Technician WO Zebus Test Engineer Ronald G Powell

Inspector (Customer or USGov't) ________

DESIGN VERIFICATION TEST

TEST DATA SHEET

Type of Test: Functional   Date of Test: 11 December 1969

Part Name: Rotary Joint   Part Number:

Test Procedure: 8-480086 (Para. 5.3)   Part Serial Number: 8 "

NOTE: Enter in REMARKS column test prior to this Functional Test.

		REMARKS
A.	Pressure	
	Test Pressure 140 psig	After Salt Spray Test
	Test Media GN_2	
	Duration of Test 5 Min	Compl WOL
B.	Leakage - Static	
	Test Pressure 30 psig	During chilldown w/specimen rotating
	Test Media LN_2	and vacuum pump conn. heavy leakage.
	Test Temperature $-320^{o}F$	LN_2 press 28 psig. Compl WOL.
	Duration of Test 5 Min	Notified Straza rep.
C.	Leakage - Dynamic	
	Test Pressure 30 psig	Extreme leakage
	Test Media LN_2	Torque - 200 Ft #
	Test Temperature $-320^{o}F$	
	Cycle Rate 2.3 RPM	
	Duration of Test 5 Min	Compl WOL
D.	Torque	
	Test Pressure 30 psig	Continued extreme leakage
	Test Media LN_2	
	Test Temperature $-320^{o}F$	
	Bending Moment (31.8 psig) 300FtLbs	
	Torque 230 Foot Lbs	Extreme frosting on unit
	Cycle Rate 2.3 RPM	Compl WOLebus

Test Technician: /s/ W.O.Lebus   Test Engineer: /s/ Ronald G. Powell

4.2.1.5.9 Vibration Test

Test Requirement

Reference: AMETEK/Straza Specification No. 8-480086, Paragraph 5.7.

The purpose of this test is to demonstrate the structural integrity of the specimen when subjected to vibration. This integrity shall be measured by: (a) visual observation during vibration and (b) functional test on completion of vibration in all three (3) axes.

The random vibration was not performed due to the short test schedule. It was also felt that it would not add significantly to the equipment evaluation on this type of development program.

Test Procedure

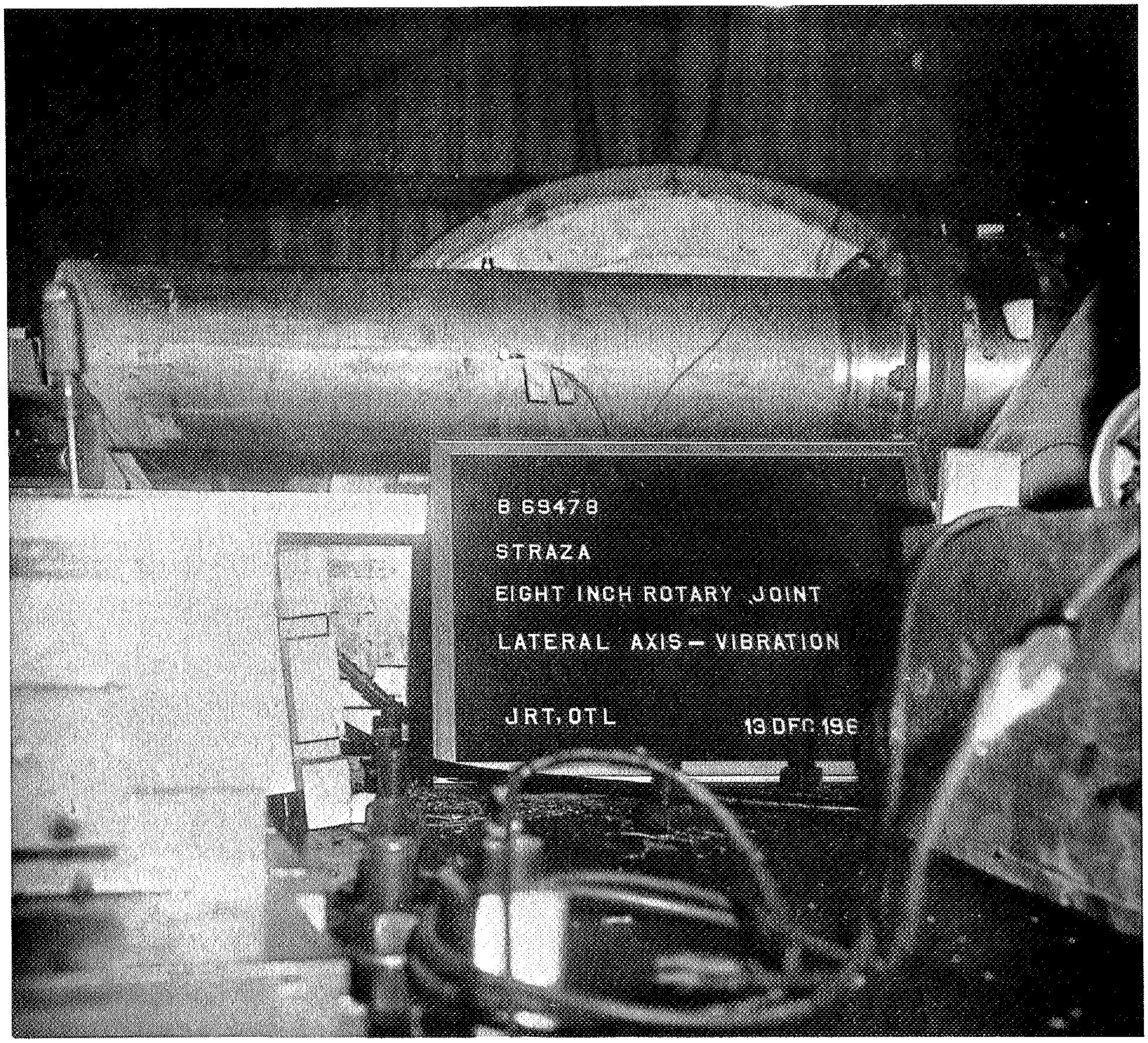
A Ling A-249 Vibration System, with six (6) Accelerometers, was calibration in accordance with Standard Procedures. The specimen was installed in a test fixture which in turn was mounted on team tables. A liquid Nitrogen supply and vent system was connected to the specimen, and accelerometers mounted to monitor cross talk and vibration levels. Beginning in the lateral axis, the specimen was subjected to a sinusoidal resonant frequency search at an acceleration level of 1 g for a frequency range of 5 to 3000 Hz transversed logarithmically for a period of fifteen (15) minutes per axis. All accelerometer outputs were continuously recorded on the oscillograph. The specimen was filled and then emptied of liquid Nitrogen prior to applying any vibration excitation. On completion of vibration in the lateral axis, the specimen was re-orientated in the transverse and longitudinal axes and the foregoing procedure repeated.

Test Results

No functional failure was visually evident during vibration in any one of the three (3) major orthogonal axes. The maximum transmissibilities occurred at the center of the bayonet support tube. Values as high as 40 did not impair the operation of the bayonet or bearing load carrying capability. Leakage continued to be excessive.

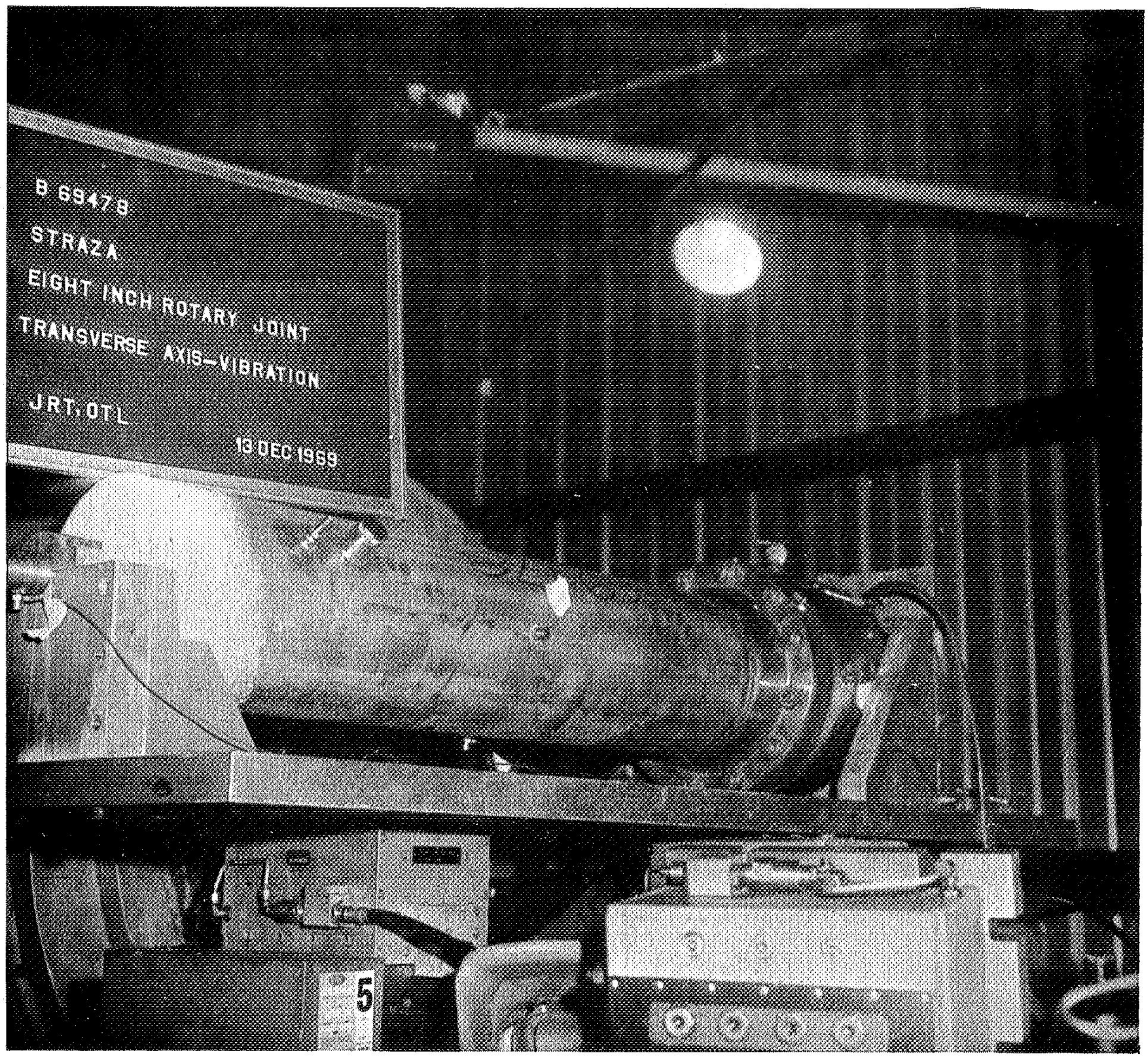
Test Data

Data sheets and photographs 8, 9 and 10 are included on the following pages.



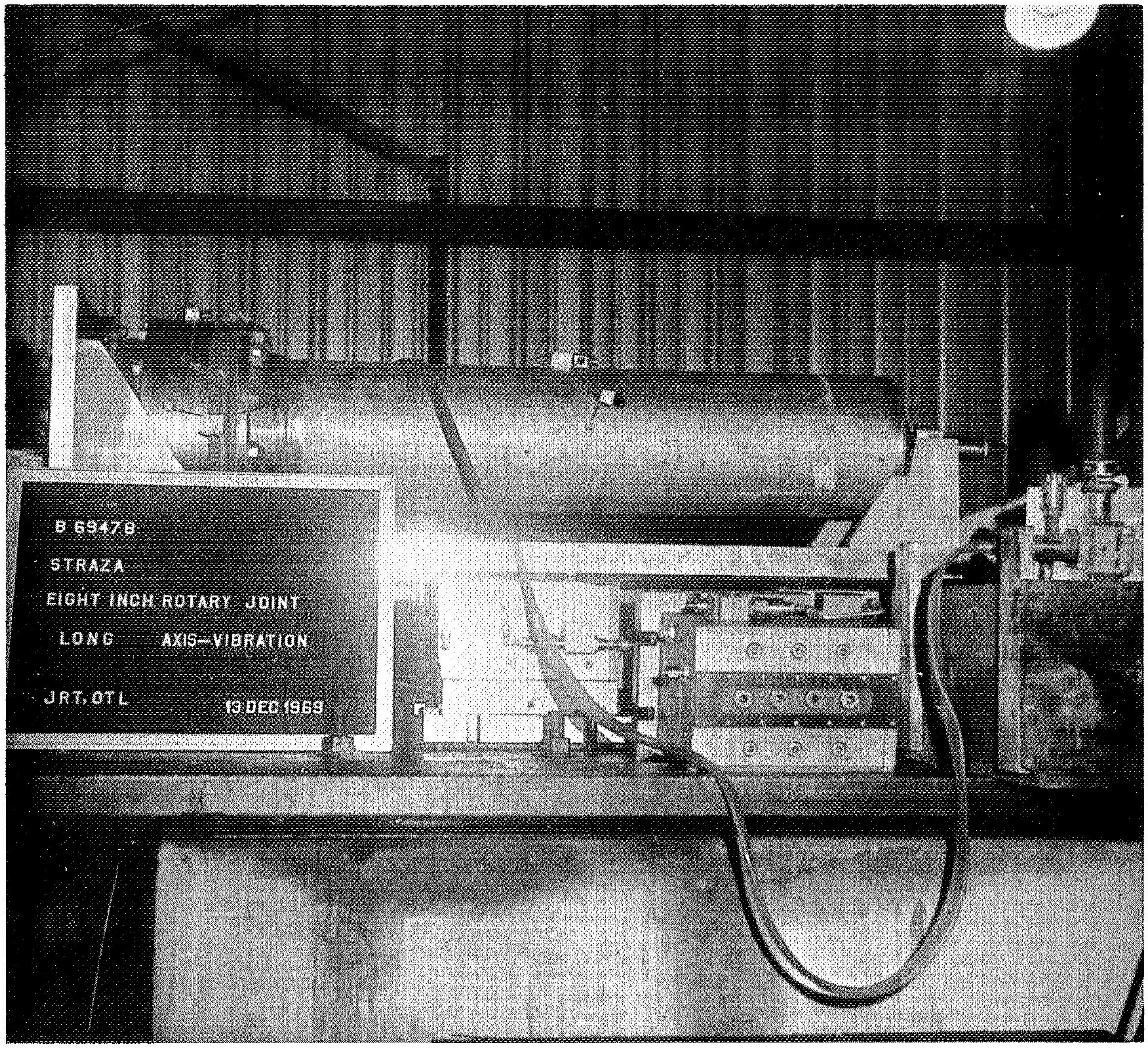
Test Report Photograph

Test Report
Photograph No. 8



Test Report Photograph

Test Report
Photograph No. 9



Test Report Photograph

Test Report
Photograph No. 10

PARTS TEST DATA

Date Started: 12-13-69		Test Engineer: C. H. / G. Adams / K.L.
Date Completed: 12-13-69	Specimen Description: 8" Rotary Joint	Test: Vibration, Sinusoidal	Performed by: G. Stull / H.M.B
Temp: Amb Humidity: Amb.	Vendor: STRAZA	Part No: 8-030930	Spec. No: 8-480086	Para: 5.7.2.1
		Gov't. Inspector	Job No: B-69478.

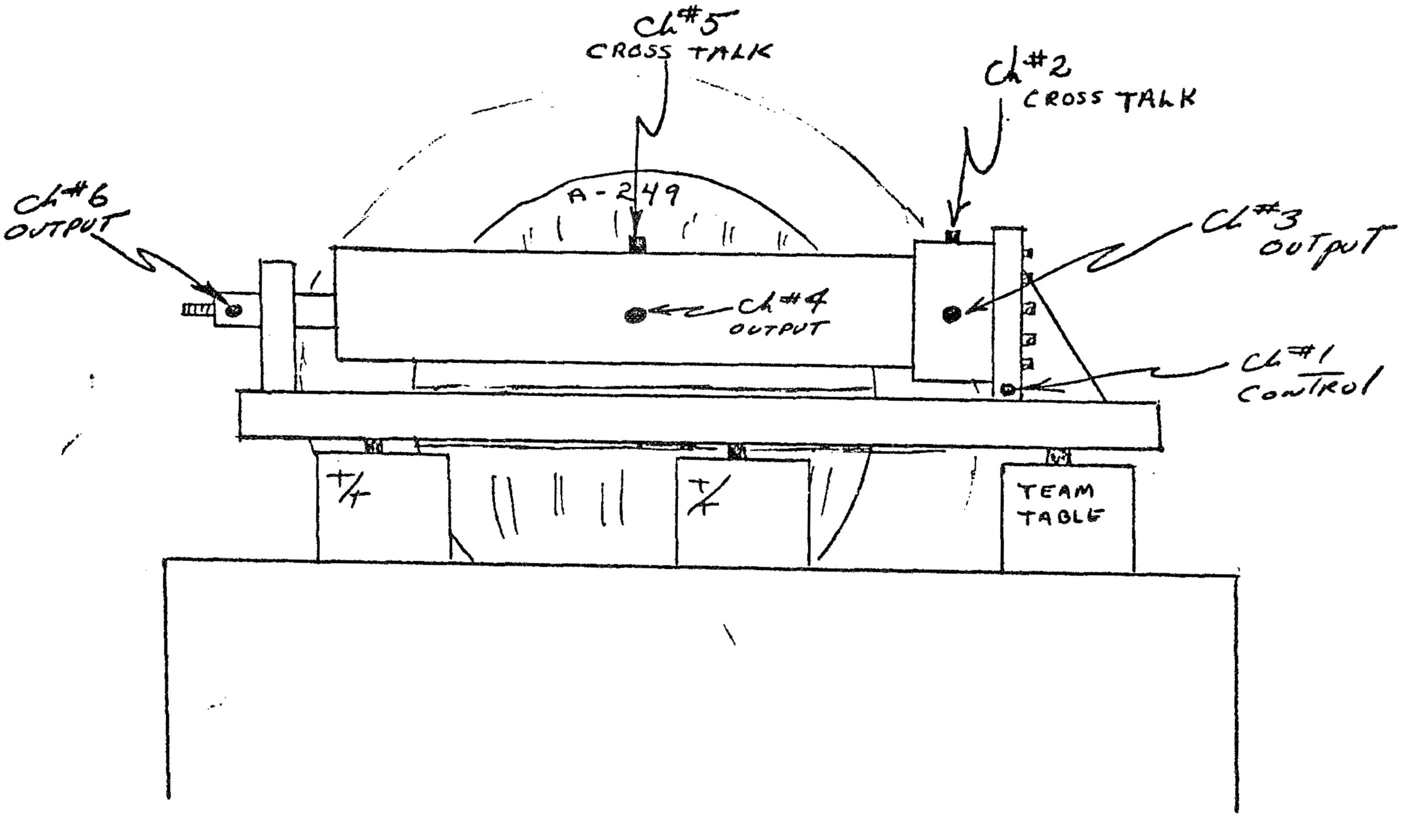
Date	Time	Temperature	Specimen S/N	Axis	Frequency	Displacement	Acceleration Input	Acceleration Output	Acceleration Location	Comment
Mo/day	Hour	°F			Hz	in.DA	±G			Specified
12-13	1714	70	01	Lateral	5-3000		1.0			Conducted Sine cycle
					3000-5		1.0			5-3000-5 HZ
										Recorded outputs on
										recorder.
12-17	1229		-	- -		—	-			Completed O.K.
										Drawing attached
										& Reduced data

JOB NO. B-69478 TEST DATA PAGE NO. 1a

Type of Test Sine Cycle

Lateral Axis

Accelerometer Location



TEST DATA Page 16

TYPE TEST: Resonant Scan	TEMP. Amb	JOB B-69428
SPECIMEN: 8" Rotary Joint	S/N 01	AXIS LATERAL

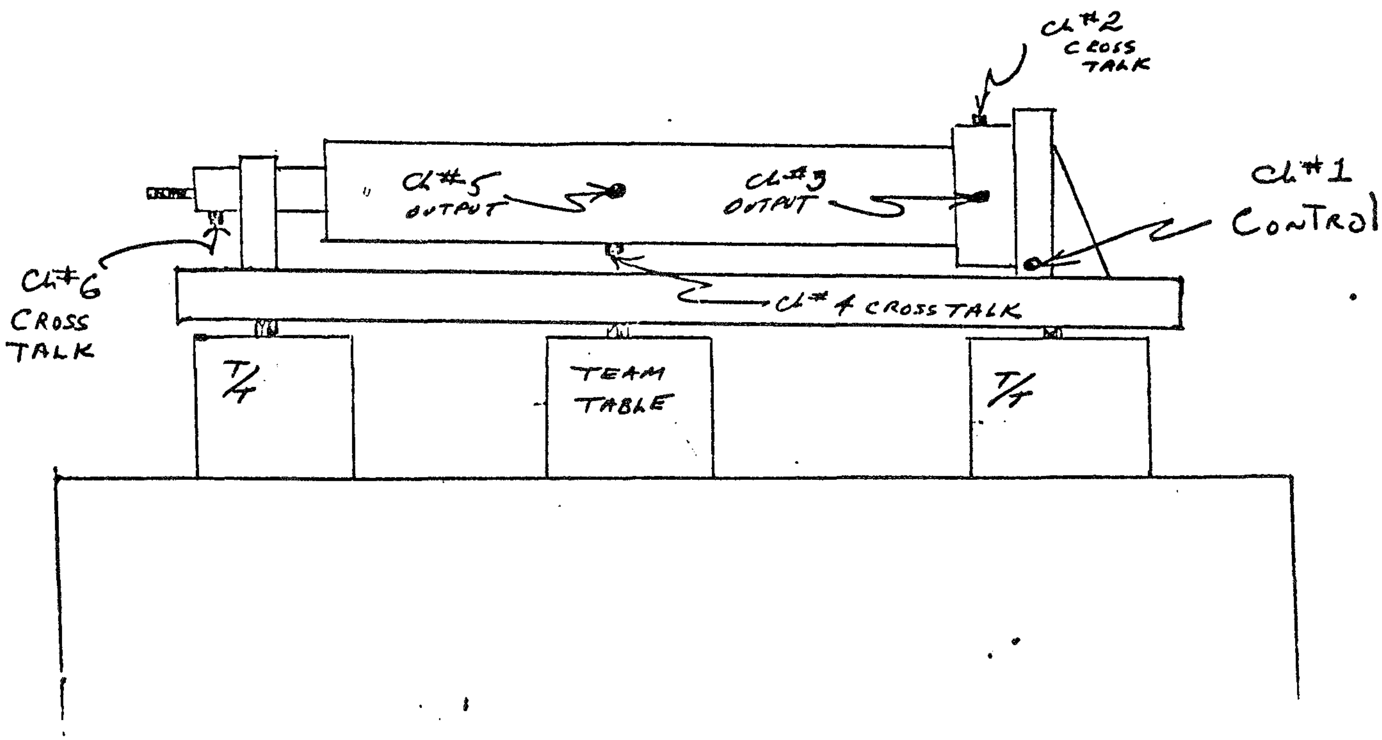
DATE	TIME	CH#1 gPK	CH#2 gPK	CH#3 gPK	CH#4 gPK	CH#5 gPK	CH#6 gPK	FREQ.						TECHNICIAN	COMMENTS
12-13	1714	1.0	3.6	1.0	7.0	1.0	8.0	62							
		1.0	2.0	.6	3.8	.6	2.5	87							
		1.0	1.5	1.0	3.0	.7	3.5	150							
		1.0	1.0	1.7	5.0	1.5	1.5	230							
		1.0	.5	.8	3.0	.9	3.8	310							
		1.0	.5	1.2	3.2	.8	5.0	360							
		1.0	1.1	1.0	4.0	.8	6.5	514							
		1.0	16	.8	1.9	.5	9.8	700							
		1.0	2.3	2.5	15.0	3.2	16.0	860							
		1.0	8.0	9.0	9.0	7.0	20.0	1000							
		1.0	5.2	7.0	40.0	5.0	7.0	1800							
		1.0	2.5	1.5	260	14.0	6.0	1300							
		1.0	1.0	1.0	22.0	7.0	2.0	1675							
		1.0	2.5	2.0	200	2.0	1.0	1800							
		~~1.0~~	~~2.5~~	~~2.0~~	~~20.0~~	~~2.0~~	~~1.0~~	~~1800~~							
		1.0	2.0	1.2	130	5.0	3.0	2800							

DATE STARTED: 12-13-69 DATE COMPLETED: 12-13-69	PARTS TEST DATA SPECIMEN DESCRIPTION: 8" Rotory Joint	TEST: Vibration, Sinusoidal	TEST ENGINEER: Ron Powell / C.H. Performed by: C. Stull / H.M.B
TEMP: Amb Humidity: Amb.	VENDOR: STRAZA	Part No: 8-030930 Spec. No: 8-480086 Para: 5.7.2.1 Gov't. Inspector	Job No: B-69478

DATE	TIME	Temperature	Specimen S/N	Axis	Frequency	Displacement	Acceleration Input	Acceleration Output	Acceleration Location	COMMENT
Mo/day	Hour	°F			Hz	in.DA	±G			SPECIFIED
12-13	1744	70	01	TRANS.	5-3000		1.0			Conducted sine cycle
					3000-5		1.0			5-3000-5 HZ
										Recorded data
										on visicorder.
12-13	1759									Completed all
										Drawing attached
										& reduced data

Type of Test Sine Cycling Vibration

Transverse Axis
accelerometer Location



TEST DATA

TYPE TEST: Sine Resonant Scan	TEMP. Amb.	JOB # B-69178
SPECIMEN: 8" Rotory Joint	S/N 01	AXIS Transverse

DATE	TIME	Freq.	Ch#1	Ch#2	Ch#3	Ch#4	Ch#5	Ch#6						Technician	Comments
12-13	1744	62	1.0	1.0	3.5	3.0		2.5						OB	
		150	1.0	.8	2.5	3.0		3.5							
		475	1.0	1.0	1.0	3.0		7.0							
		535	1.0	1.5	1.3	8.0		2.0							
		700	1.0	1.0	2.0	11.0		2.8							
		993	1.0	5.0	4.5	27.0		30.0							
		1100	1.0	5.0	2.8	27.0		7.5							
		1180	1.0	5.5	4.0	24.0		9.0							
		1400	1.0	1.5	3.0	33.0		7.0							
		1620	1.0	1.0	2.5	10.0		3.0							
		1720	1.0	1.0	2.5	15.0	—	3.0							
		1900	1.0	1.0	2.5	10.0		1.5							
		2400	1.0	1.0	2.0	17.0	—	1.5							

PARTS TEST DATA

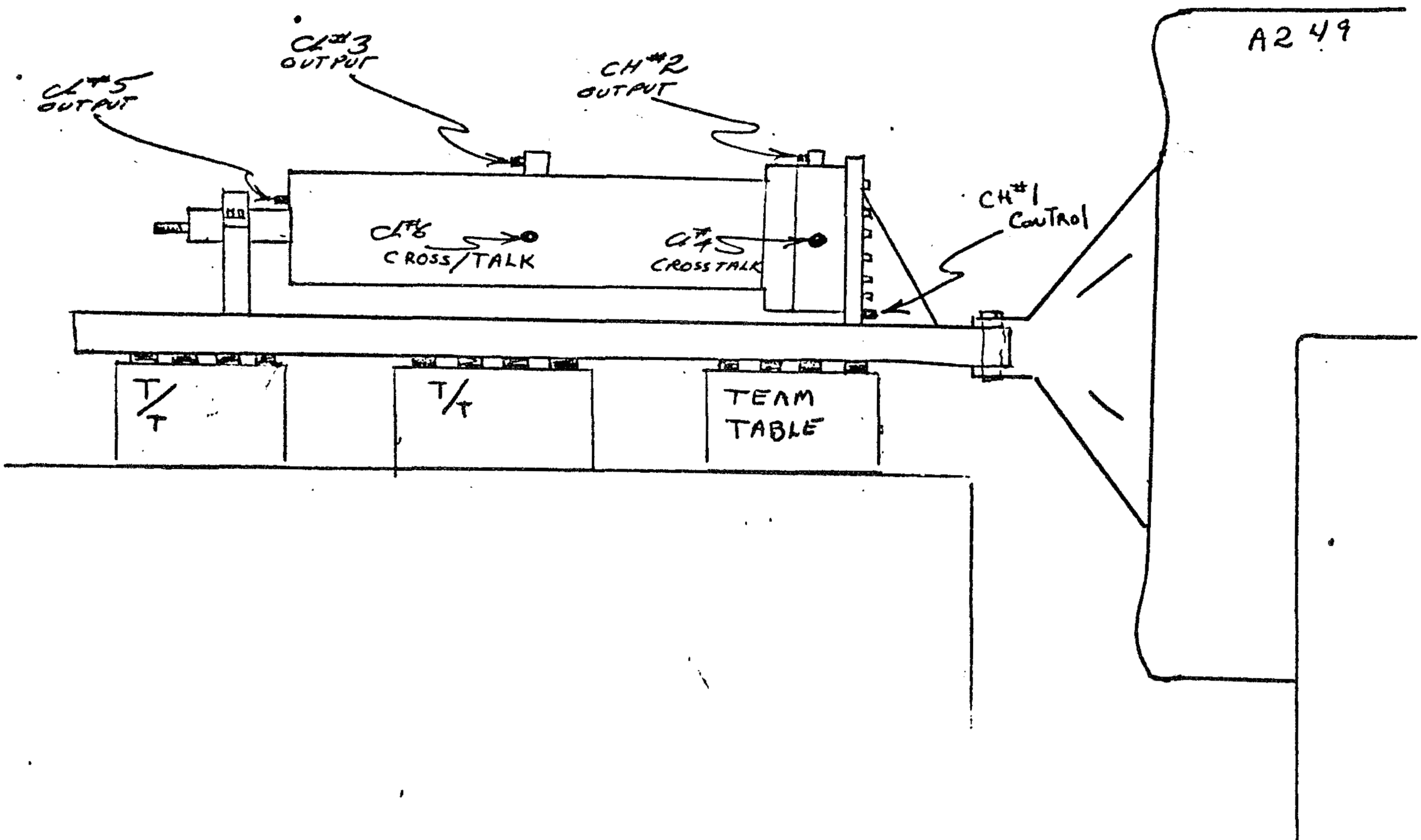
Date Started: 12-13-69 Date Completed: 12-13-69	Specimen Description: 8" Rotary Joint	Test: Vibration, Sinusoidal	Test Engineer: R. Powell C. H. / G. Adams Performed by: G. Stull / H.M.B.
Temp: Ambient Humidity: Ambient	Vendor: STRAZA	Part No: 8-030930	Spec. No: 8-480086 — Para: 5.7.2.1
		Gov't. Inspector	Job No: B-69478

Date	Time	Temperature	Specimen S/N	Axis	Frequency	Displacement	Acceleration, Input	Acceleration Output	Acceleration Location	Comment
Mo/day	Hour	°F			Hz	in.DA	±G			Specified
12-13	2100	+65	01	Longit.	5-3000		1.0			Conducted Sine cycle
	2115	+65			3000-5		1.0			5-3000-5 Hz in 15 Minutes.
										Recorded outputs on vis-
										corder.
										Completed: O.S.
										Drawing attached
										& Reduced data sheet.

JOB NO. B-69478 TEST DATA PAGE NO. 3a

Type of Test Sine Scan

9-13-69



Longitudinal Axis
accelerometer Set Up.

TEST DATA

Page 36

TYPE TEST: Resonant Scan | TEMP. | JOB # B-64478

SPECIMEN: 8" Rotary Joint | S/N 01 | AXIS Longitudinal

DATE	TIME	FREQ.	Ch #1 "G" PK	Ch #2 "G" PK	Ch #3 "G" PK	Ch #4 "G" PK	Ch #5 "G" PK	Ch #6 "G" PK						Technician	COMMENTS
12-13	2100	91	1.0	5.3	4.5	3.0	Lost Accelerometer Signal	.5						[illegible]	The following data was reduced from visicorder recorded records during sine scan
		128	1.0	7.0	7.0	5.0		.8							[illegible]
		200	1.0	1.5	2.0	3.5		.5							
		275	1.0	2.3	5.3	4.0		.6							
		300	1.0	1.0	2.5	19.0		.4							
		600	1.0	2.5	2.2	1.8		.5							
		625	1.0	5.0	2.0	1.5		.7							
		690	1.2	4.5	2.6	3.0		.5							
		730	1.0	1.8	3.0	10.0		.2							
		790	1.0	1.8	7.0	1.8		.5							
		800	1.0	1.0	4.0	4.5		.3							
		960	1.0	5.0	1.5	3.0		2.0							
		1160	1.0	1.0	2.5	10.0		.7							
		1250	1.0	3.0	1.8	11.0		.6							
		1275	1.0	4.0	2.0	1.0		.6							
		1610	1.0	1.2	3.0	7.0		.5							
		1930	1.0	1.0	4.0	9.0		.5							
		2800	1.0	3.5	13.0	5.0		.8							
		2950	1.0	1.0	10.0	2.0		.8							
		3000	1.0	2.0	9.0	5.5		4.0						[illegible]	

DESIGN VERIFICATION TEST

TEST DATA SHEET

Type of Test: Functional Date of Test: 15 December 1969

Part Name: Rotary Joint Part Number: ____

Test Procedure: 8-480086 (Para. 5.3) Part Serial Number: 8"

NOTE: Enter in REMARKS column test prior to this Functional Test.

		REMARKS
A.	Pressure	
	Test Pressure 30 psig	Vibration (Resonance Freq Search)
	Test Media GN_2	Extreme leakage
	Duration of Test 1 Min	
B.	Leakage - Static	Start Vacuum Pump
	Test Pressure 30 psig	Start Chilldown
	Test Media LN_2	
	Test Temperature $-320^{\circ}F$	Extreme leakage
	Duration of Test 5 Min	Compl WOL
C.	Leakage - Dynamic	
	Test Pressure 30 psig	
	Test Media LN_2	
	Test Temperature $-320^{\circ}F$	Torque 175 Ft Lbs
	Cycle Rate 2.5 RPM	Extreme leakage
	Duration of Test 5 Min	Compl WOL
D.	Torque	
	Test Pressure 30 psig	Continued extreme leakage
	Test Media LN_2	
	Test Temperature $-320^{\circ}F$	
	Bending Moment (31.8 psig) 300 Ft Lbs	
	Torque 195 Ft #	Extreme frosting on unit
	Cycle Rate 2.5 RPM	Compl WOLebus

Test Technician /s/ W.O.Lebus Test Engineer /s/ Ronald G. Powell

4.2.1.5.10 Life Cycle Test

This test was not performed due to schedule and economic impact.

4.3 DESIGN SPECIFICATION

A procurement specification for the rotary joint titled "Rotary Joint, Cryogenic-Vacuum Jacketed Cryogenic Transfer and Storage Systems" and assigned number 79K00113 was submitted to NASA under Contract Number NAS 10-6098. The document was prepared on "B" size KSC format.

4.4 CONCLUSIONS

The test data indicates the primary problem which must be solved before the 8 inch joint design can be considered truly operational is the external leakage. Additional testing on this unit equipped with KEL-F seals in lieu of Teflon, should bring the leakage down to a reasonable value. Any excessive external gas leakage aggravates the heat leak as was proven during the final test phase when the unit iced up. This also increases the bearing friction which affects the rotational torque load.

The liquid nose seal "B" (page 23) must be non-sluffing also. Liquid leakage into the intra cone area which is controlled by the four small bleed holes located behind the nose seal is limited by percolation. If liquid leakage becomes excessive this process cannot control the liquid flow and the heat leak will rise sharply.

A short cryogenic leakage and torque test which was performed on the Chiksan 6 inch rotary indicated that the KEL-F primary dynamic seal was performing well. The average leakage thru one rotational cycle was less than 10 SCIM. The torque required to rotate was 240 Ft. Lbs. The heat leak thru the unit was not measured. However, during the above mentioned test, the bottom of the rotary in the horizontal position felt cold to the touch. This is an indication that the heat leak is higher than specification limits and is probably a result of the air leakage into the evacuated cone area. This leakage was detected at the beginning of the test.

4.5 RECOMMENDATIONS

All remaining tests on both units should be completed to confirm this design. This includes life cycle, proof and burst pressure on the 8 inch unit and all tests discussed in Paragraph 4.2.1.5 on the 6 inch unit.

During the previously mentioned tests of the 6 inch unit, GN_2 was leaking into the vacuum chamber of the sealed portion of the unit. This should be fixed prior to any additional tests. This leak is probably caused by a weld crack in the cone assembly. During this same test, the three (3) .032 diameter holes drilled on the O.D. of the nose seal to allow pressurization of the CTL Dixie anti-convection barriers were placed incorrectly. This permitted the hole to reseal under pressure against the aluminum retaining ring. These holes should be placed at the outer periphery of either face where the aluminum band and the tapered nose ring are ineffective as a seal.

The 8 inch unit should be remachined to accept 3/4 inch cross section compliant Kaydon bearing prior to any retest. This larger size should handle both thrust and radial loads as required in the original specification. The only rework required to use the 3/4 inch bearing is to bore out Item 3 on drawing 8-100089 and relocate "O" ring, Item 17. Item 5 would have to be shortened to accept the new bearings.

REFERENCES

1. Design Study for Cryogenic Coupling, W.O. 4053 (AMETEK/Straza 1969), Contract NAS 10-6098.

2. NBS Report 6084, "An Investigation of Some Problems Connected with the Handling of Liquid Hydrogen", by R. J. Richards, W. G. Steward, December 2, 1959

3. New Departure Bearing Design Book, Volume I (1946)

4. "The Compliant Ball Bearing", by J. F. Robinson, Navigation Systems Division, Autonetics Division of North Am erican Rockwell Corporation, September 1968

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