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CONTRACT: NAS 10-1266
SOLAR REF: 6-1501-7

FINAL REPORT



HIGH PRESSURE LARGE FLOW RATE FLEXIBLE CONNECTOR SYSTEM

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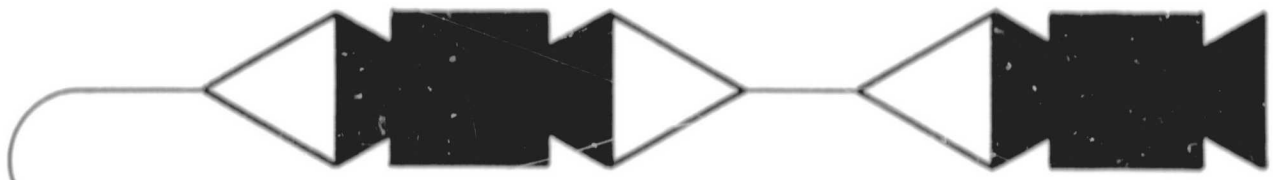


SOLAR San Diego California

A DIVISION OF INTERNATIONAL HARVESTER COMPANY

CONTRACT: NAS 10-1266
SOLAR REF: 6-1501-7

FINAL REPORT



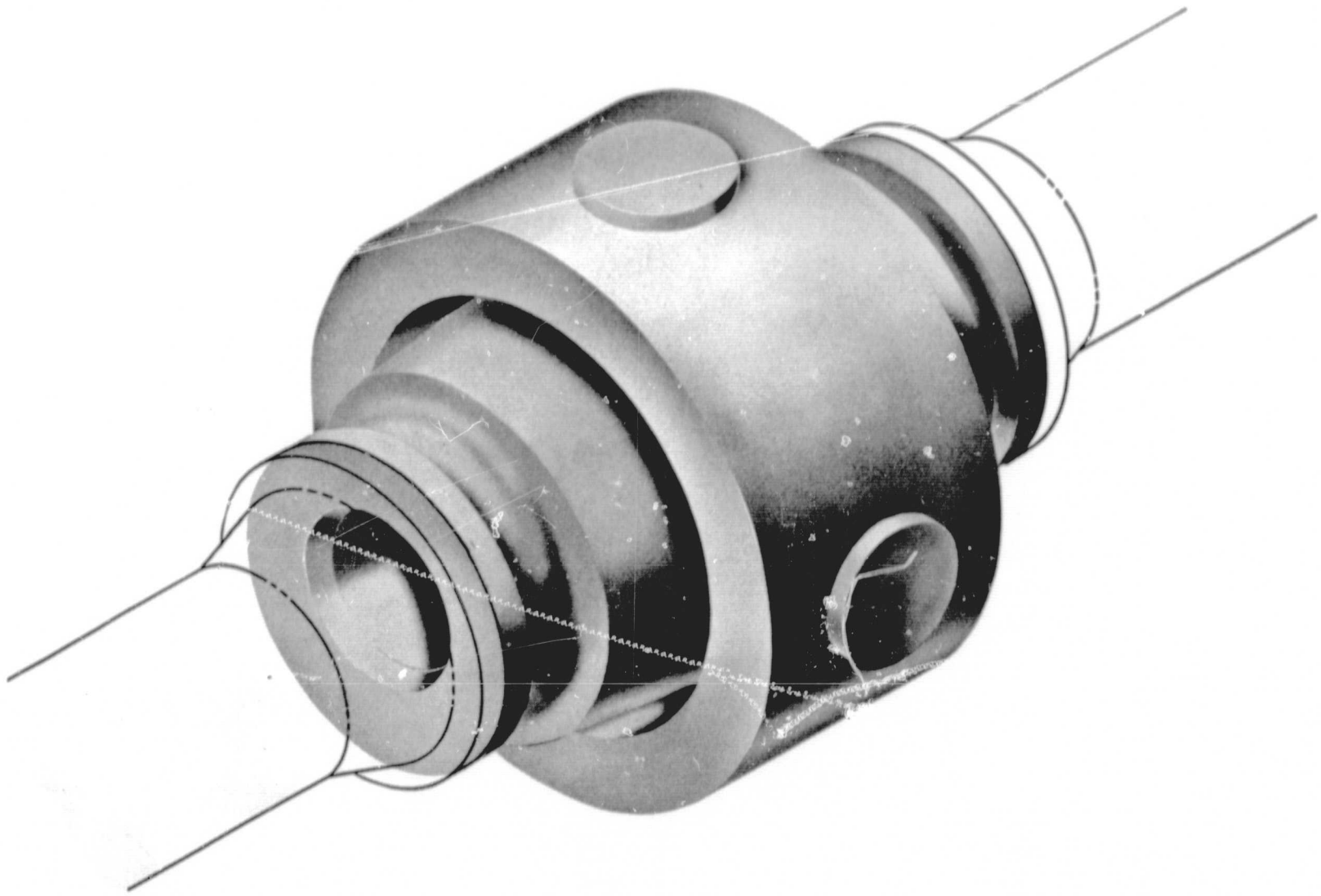
HIGH PRESSURE LARGE FLOW RATE FLEXIBLE CONNECTOR SYSTEM

Submitted to:

National Aeronautics and Space Administration
John F. Kennedy Space Center
Huntsville, Alabama



SOLAR San Diego California
A DIVISION OF INTERNATIONAL HARVESTER COMPANY



ACKNOWLEDGMENT

This report was prepared by the Aerospace Department of Solar, a Division of International Harvester Company. The work was accomplished under the direction of H.T. Mischel, Project Engineer.

Personnel contributing to the program were: Design, N.G. Fritz, D.H. Betts, L.R. Brown, M.J. Hughes; Stress Analysis, G.W. Bowers, Jr.; Reliability, R.L. McKinnon; Electron Beam Welding, F.C. Nelson; Manufacturing, P.H. Person, N.D. Crego, W.H. Fallon; Purchasing, D.G. Dedmon.

Gratitude is expressed for the information furnished by the Gray Tool Company, Houston, Texas, relating to the Graylock hubs.

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I. INTRODUCTION

This final report summarizes the work performed by Solar, a Division of International Harvester Company, on the design study on a high-pressure, large flow rate flexible connector system. The work was performed under Contract NAS 10-1266 issued by the National Aeronautic and Space Administration, Kennedy Space Center, Huntsville, Alabama. Mr. Joe Morris was the Technical Supervisor for NASA on this program.

The purpose of the program was to perform a design study and materials selection resulting in a detailed design for fabrication of a connector system which will meet the following requirements:

Diameter (ID)	3.50 inches
Pressure	
Working	10,000 psig
Proof	15,000 psig
Burst	40,000 psig
Working Flow Rates	
GN_2	104,000 cfm at 10,000 psig
GH_e	204,000 cfm at 10,000 psig
Pressure Drop	Not to exceed an equivalent pipe length of 20 diameters in each Flexible Component.
Misalignment Tolerances	Capable of compensating for a maximum misalignment and operational movement delineated by a 12 inch cube.
Seal Requirements	Leakage limited to one bubble per 15 minutes when pressurized at 15,000 psig. underwater.
Life Requirements	100,000 cycles minimum of pressurizing from zero to 10,000 psig through a temperature range of -65 F to +150 F.

A complete set of drawings developed during the program is included in Appendix A.

II. STUDY METHOD

This study program to develop a feasible design of a high-pressure large flow rate flexible connector system was conducted in three phases:

Phase IA	-	Design Study
Phase IB	-	Specific Design
Phase IC	-	Review and Analysis

2.1 PHASE IA - DESIGN STUDY

Phase IA began with an orientation conference at Huntsville, Alabama, on 24 March 1964, and was culminated by a report to NASA personnel at the mid-term presentation at Huntsville on 5 May 1964. The study phase was devoted to surveys of various candidate materials and to the development of conceptual designs of the flexible connector system. Conceptual line routings and arrangements of components were developed and investigated. Concepts of various types of components were developed and studied. Preliminary analytical methods were outlined and recommendations were prepared. The results of this phase were presented to NASA at the mid-term presentation, at which time the NASA Technical Supervisor, Mr. Joe Morris, indicated the direction Solar was to follow in Phase IB of the program.

2.2 PHASE IB - SPECIFIC DESIGN

Phase IB was devoted to developing for fabrication those detail designs of the system concept selected by NASA during the mid-term presentation. During this phase, the Solar Engineering groups, in consultation with the various other groups which would be involved in the fabrication of the system, prepared detail design drawings of each of the system components. Industry surveys were initiated to obtain pricing, availability, and delivery data of the raw materials and operations to be subcontracted. Conferences were held to determine the most fabricable approach to each design. Various departments at Solar were asked to prepare presentations on the aspects of the fabrication task in which they would participate. Estimating groups and manufacturing engineers prepared detail pricing for one unit of the system. The re-

sults, less the pricing, were submitted to NASA at the final presentation on 7 July 1964. At this time, the NASA attendees issued instructions that the final pricing was to be based on one complete flexible connector system with the system modified to delete a venturi which was originally included.

III. PHASE IA - DESIGN STUDY

The design study phase was devoted to material survey and selection and to the development of connector system concepts. Prior to initiation of this study phase, an orientation conference was held at Huntsville, Alabama, on 24 March 1964. At this time, various clarifications to the Specifications of Design and Test contained in Exhibit I of Contract NAS 10-1266 were presented to Solar. The misalignment tolerances to be accepted were clarified. These clarifications consisted of the fact that the major portion of motion within the 12-inch cube would be imposed upon the flexible connector system during the unpressurized make-and-break operation. Only vibratory and launch umbilical tower (LUT) settling motions, of much smaller magnitudes, would have to be accepted while under working pressure. The desired routing of the system was also described. Further clarification of these motions was received by Solar in a letter from Mr. Joe Morris, NASA Technical Supervisor, dated 3 April 1964. This letter stated that the displacement which the present LUT platform experiences under vibration is within a 0.15-inch cube at 1 to 3 cycles per second.

Solar's objective was to develop system designs which would least affect the design work already done on the support towers as depicted on NASA drawing 7510178, sheets 1 and 3 of 7. This drawing was received as an enclosure with the letter of 3 April 1964.

3.1 MATERIAL SURVEY

A material survey was conducted on a series of candidate materials which Solar felt would offer the best possibility of producing a feasible approach to the design problem. The candidate materials are listed in Table I with a comparison of the minimum wall thickness for 40,000 psig minimum burst pressure based on the ultimate tensile strength of each material. These calculated wall thicknesses were determined by heavy wall cylinder formulas.(1)

Type 316L corrosion resistant steel (CRES 316L) was chosen as a candidate material in the survey in order to provide a comparison with a material that is presently

1. Roark, R.J., Formulas for Stress and Strain. New York: McGraw-Hill Book Company, Inc., Third Edition (1954), formulas 27 and 29, p 276.

being used in systems of this type. Tables II through VIII contain the chemical compositions, mechanical properties, fabrication considerations, and relative costs for each of the candidate materials listed in Table I.

At the beginning of the program, Solar determined that system pressures and operating requirements would require the use of materials of much higher strength-to-density ratio than are presently being used. Such high ratios would be needed to scale down the sizes of the components, thereby easing the fabrication and handling problems.

Since it was originally envisioned that the LUT interface end of the flexible connector system under consideration would be manually connected and disconnected, Solar recommended at the mid-term presentation that the material of construction be selected with a consideration for achieving light weight. The fabricability problems would be greatly reduced with the lighter construction, and the cost and reliability of weld joints would be improved. While the cost per pound of the high-strength alloys is generally higher than the more common low-strength materials such as CRES 316L, the weight of the lighter system makes possible predictions of equivalent reduced material cost. Studies have also indicated that the increased weight in the pipe and flexible joints using the low strength-to-density material creates structural problems which, in turn, necessitate a further increase in weight.

Consideration was given to the extremely corrosive environment at Cape Kennedy. While the outside of the line assembly may be protected by coatings, it is conceivable that moisture could condense on the interior of the system during those periods when the connector is not in use. Solar recommended rejection of those materials which do not have inherently good corrosion resistance properties.

Recognizing that the total energy stored in the fluid at these pressures is extremely high, Solar recommended rejection of those materials which exhibit relatively low impact strength. Also, materials which do not exhibit excellent weld joint efficiencies and weld ductility were rejected.

As a result of applying the above guidelines, Solar recommended Inconel 718 as the material for fabrication of the flexible joints and adjacent pipe. This choice is particularly aligned to Solar's extensive experience with the material. The Inconel 718 raw material for the pipe and the flexible joint forgings can be obtained directly from the producers, or they can be formed from billets supplied to vendors in southern California.

During this survey, the Gray Tool Company in Houston, Texas, was contacted following the recommendation of NASA. The original concepts presented at the mid-term presentation showed flexible joints integrally welded to the pipe to create a single unit construction. For that purpose, Grayiock connector P/N 45692 of Cres 316L was selected to be used as the end interfaces of the connector system presented.

3.2 CONNECTOR SYSTEM CONCEPTS

This phase also was directed toward development of concepts for the flexible connector system. Four basic system approaches were used in the development of the most feasible concept:

1. Flexible hose systems
2. Swivel or rotary joint systems
3. Angulation or three gimbal joint systems
4. Angulation and axial systems utilizing gimbal joints and compensators.

After an initial study effort, the flexible hose system and the swivel or rotary joint system were disgarded because of their inflexibility and low reliability. In flexible hose systems, the braid required to support a bellows at these operating pressures would be extremely stiff. The braid would have had to have been applied in several layers and would have resulted in an exceedingly stiff system. The method for braid attachment with the number of layers required would be extremely unreliable since the equal distribution of load is determined by the almost impossible condition of equalized layer tension during braid installation.

The swivel or rotary joint systems depend upon the use of a component that relies on a mechanical type of seal. Such contact seals have been used for low pressures with certain fluids, but their inherent tendency to leak at high pressures makes them unattractive for this application. Though there have been high-pressure rotary joints developed, none has approached the required operating pressures. The inherent leakage in the design also produces operational problems, as the leakage usually occurs at the most inopportune time. Therefore, because of the limitations of the above two systems, efforts in the study phase were concentrated on the remaining two systems which employed metallic bellows seals in gimbal or compensator designs to handle the required motions.

Motion studies were conducted on the two systems since both systems have been used in space vehicles where they have exhibited high degrees of reliability. The selected system was a three-gimbal system, since the axial travel required of a compensator was considered to be too great to be practical. The three-gimbal system also presented the interesting characteristic of permitting a routing so that the line could counterbalance itself over the bottom gimbal joint and therefore relax the forces required to effect an interface connection. Motion study layout 39071 was prepared to investigate arrangements of components for this system.

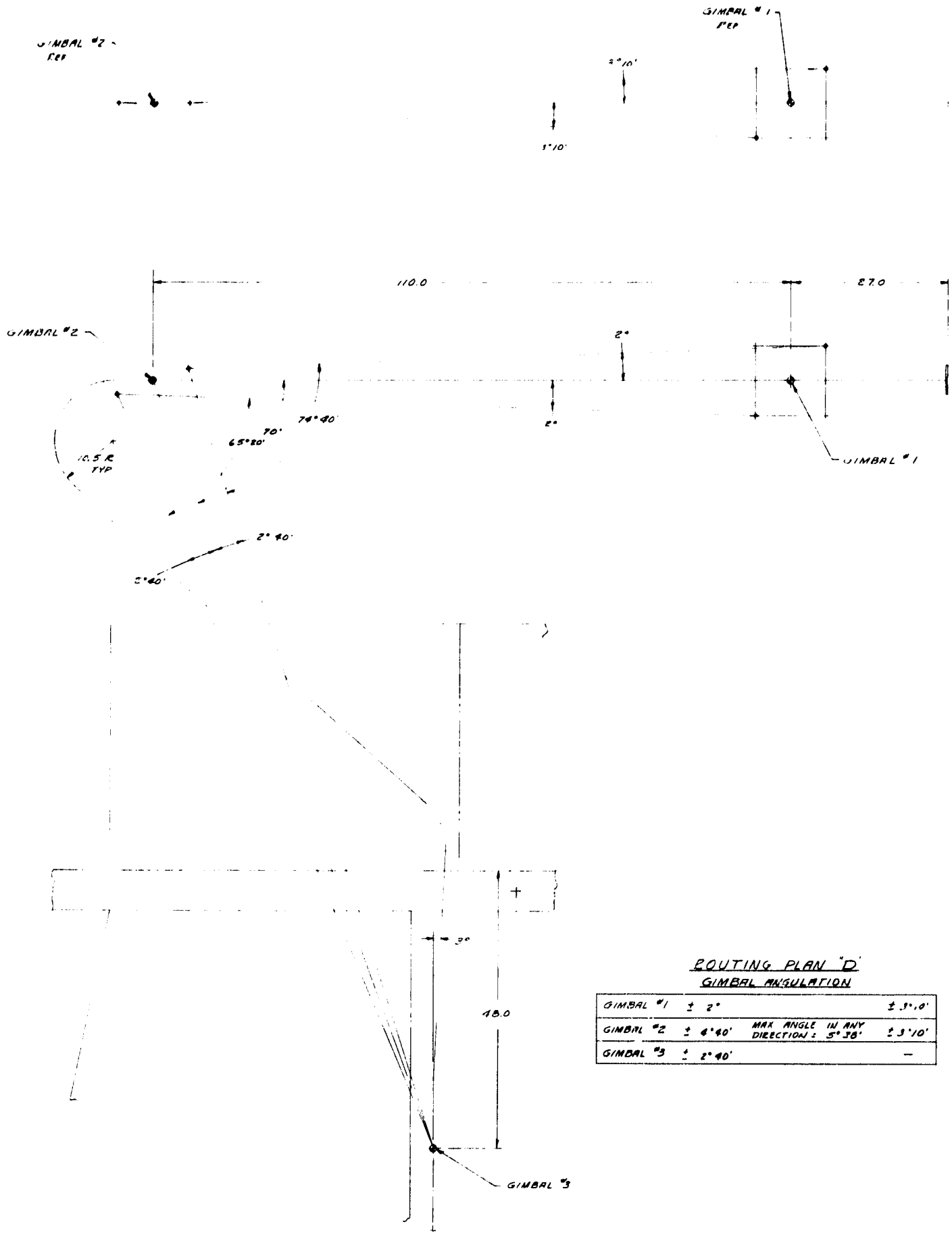
It was recommended at the mid-term presentation that routing plan D of layout 39071 was optimum. This system is a three-gimbal system arranged in such a manner that the maximum deflection of the gimbal joint is less than 6 degrees. This low deflection level was determined to be practical and feasible for bellows designs subjected to pressures of 10,000 psi. This arrangment requires only minor modification of the present service tower design. As the lower joint is located below the platform working level, the excursion of the pipe requires larger openings at that level and also requires relocation of the attaching brackets.

At this time also, the requirements for the flexible connector system to withstand a sudden venting to atmosphere from the 15,000 psig proof pressure prompted Solar to investigate a unique approach which would throttle the gases down to velocities which would ensure no adverse effects. A venturi design, depicted on Solar drawing 39705 was developed for use adjacent to the outlet flange to inhibit the rate of pressure decay.

3.2.1 Flexible Joint Concepts

After the development of the optimum layout and arrangement of components, Solar's effort was directed toward the development of the flexible joints.

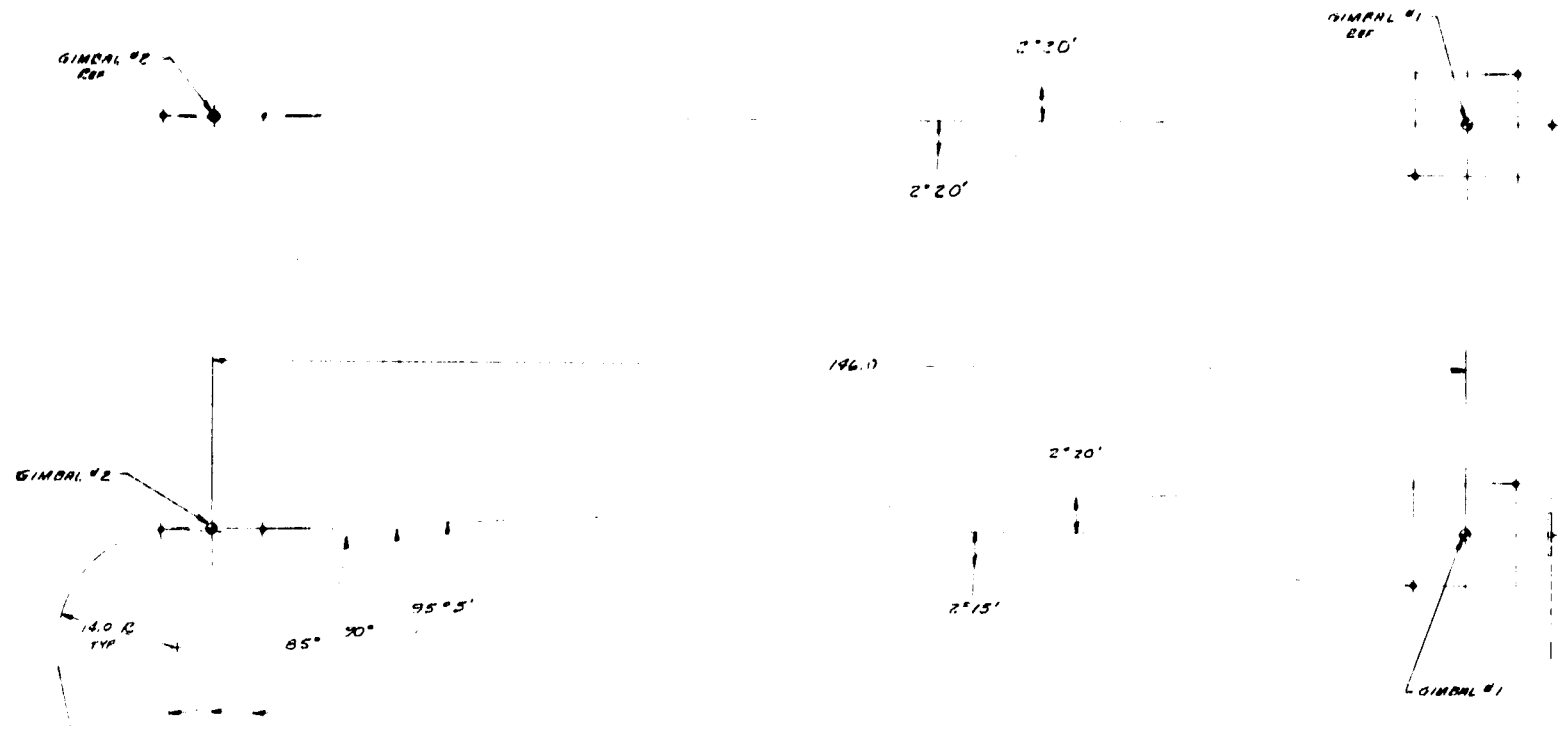
The design study revealed that the optimum restraining device for this system was a gimbal type universal joint. Other types of restraining devices were reviewed and were rejected because of inherent problems. One such type was the ball-type, in which its characteristic of line or surface contact results in exceedingly high breakaway friction force and relatively low cycle life.



ROUTING PLAN "D"
GIMBAL ANGULATION

GIMBAL #1	± 2°		± 2°10'
GIMBAL #2	± 4°40'	MAX ANGLE IN ANY DIRECTION ± 5°38'	± 2°10'
GIMBAL #3	± 2°40'		—

Solar Drawing No. 39701 (Sheet 1 of 4)

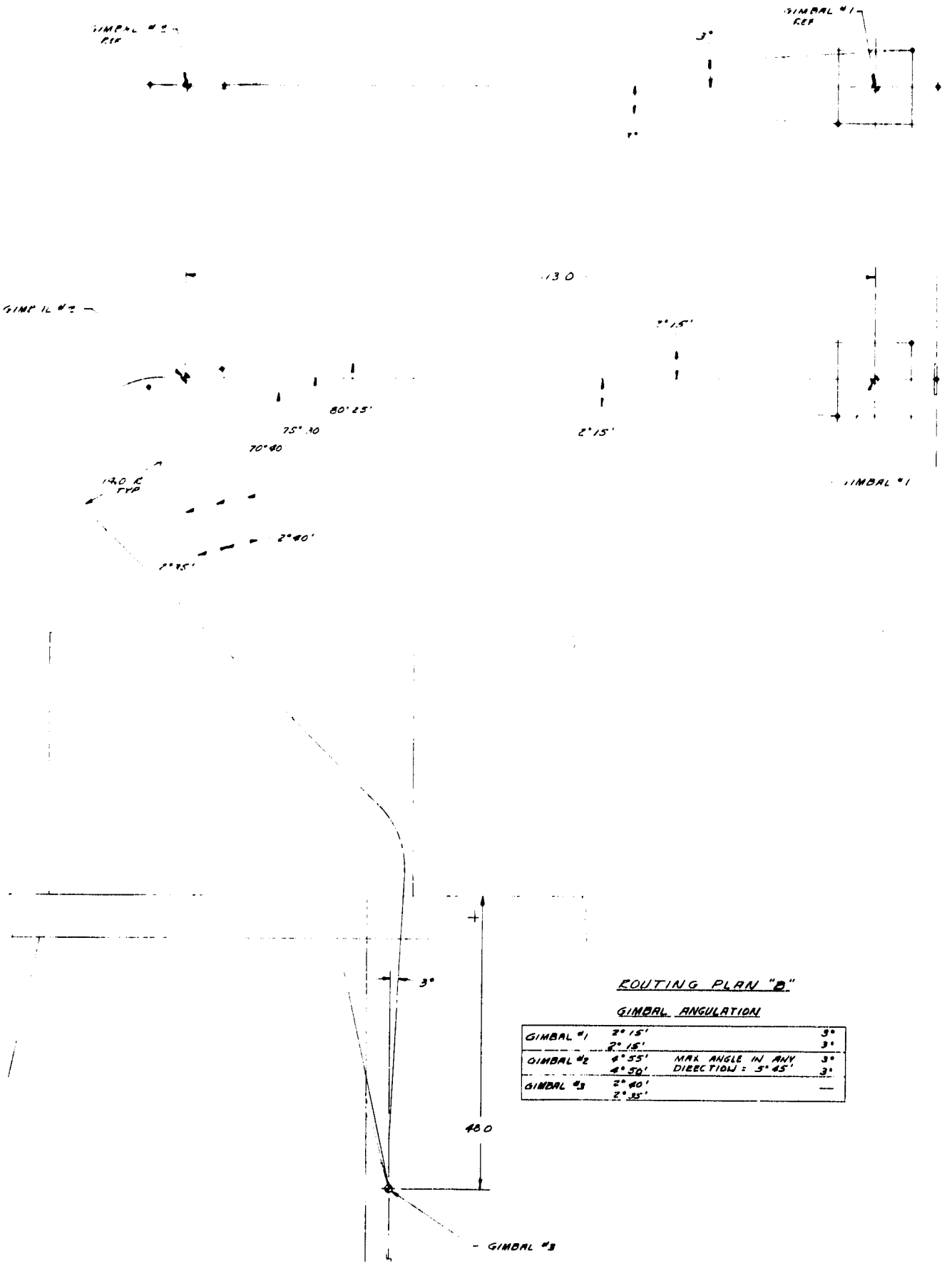


ROUTING PLAN "C"

GIMBAL ANGULATION

GIMBAL #1	2°20' 2°15'		2°20' 2°20'
GIMBAL #2	5°5' 5°	MAX ANGLE IN ANY DIRECTION = 5°35'	2°20' 2°20'
GIMBAL #3	2°45' 2°45'		—

Solar Drawing No. 39701 (Sheet 2 of 4)



ROUTING PLAN "B"

GIMBAL ANGULATION

GIMBAL #1	2° 15'		3°
	2° 15'		3°
GIMBAL #2	4° 55'	MAX ANGLE IN ANY DIRECTION = 5° 45'	3°
	4° 50'		3°
GIMBAL #3	2° 40'		—
	2° 35'		

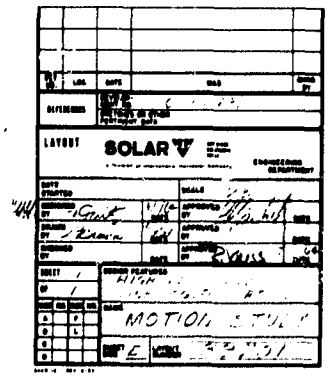
Solar Drawing No. 39701 (Sheet 3 of 4)



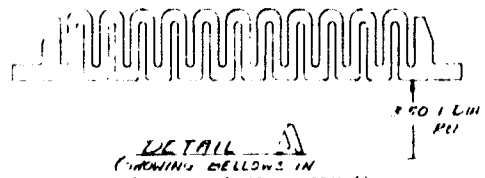
ROUTING PLAN "A"

GIMBAL ANGULATION

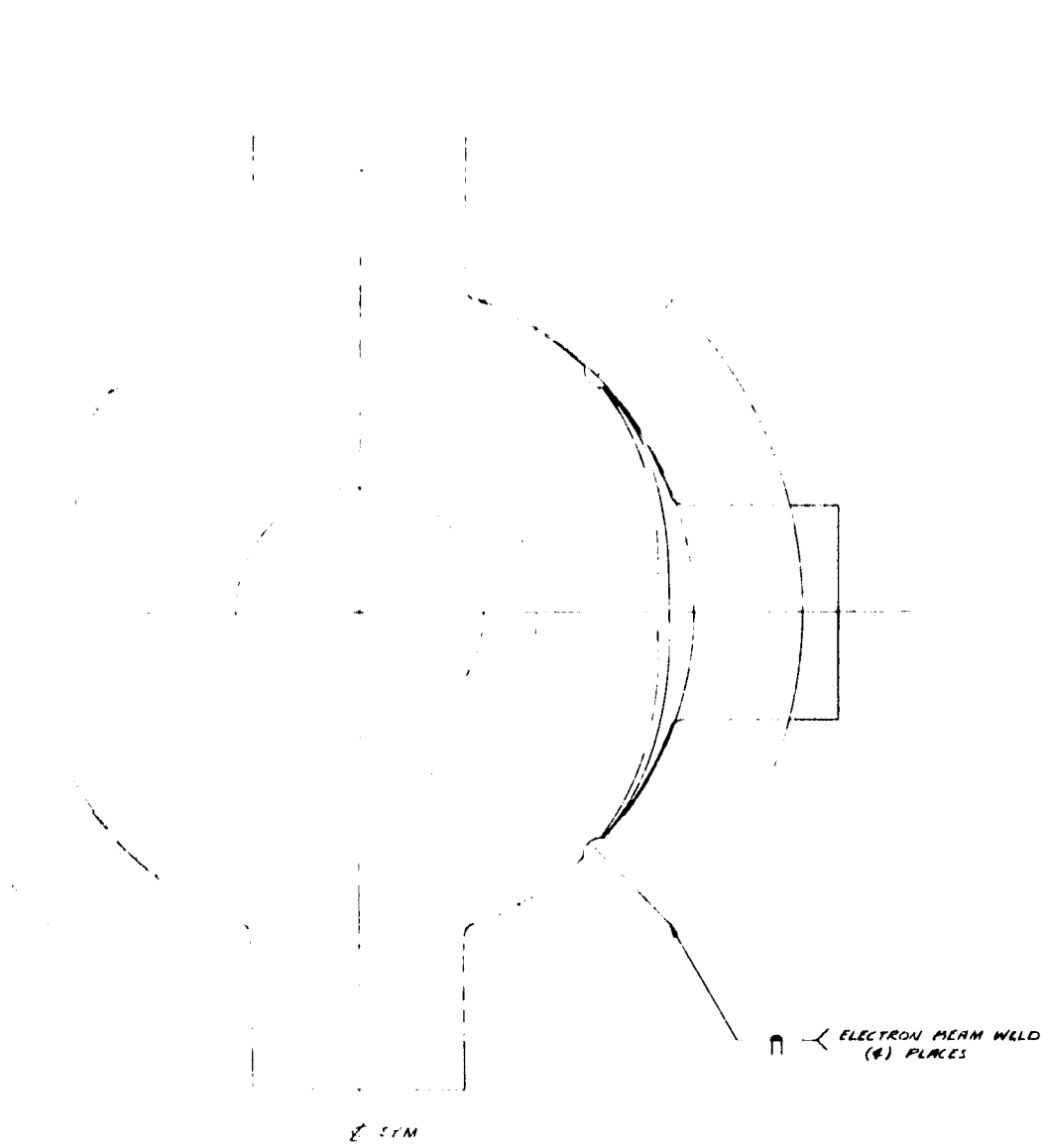
GIMBAL #1	6°15' 5°50'		5°15' 5°10'
GIMBAL #2	10°55' 10°	MAX ANGLE IN ANY DIRECTION = 12°7'	5°15' 5°10'
GIMBAL #3	4°40' 4°10'		—



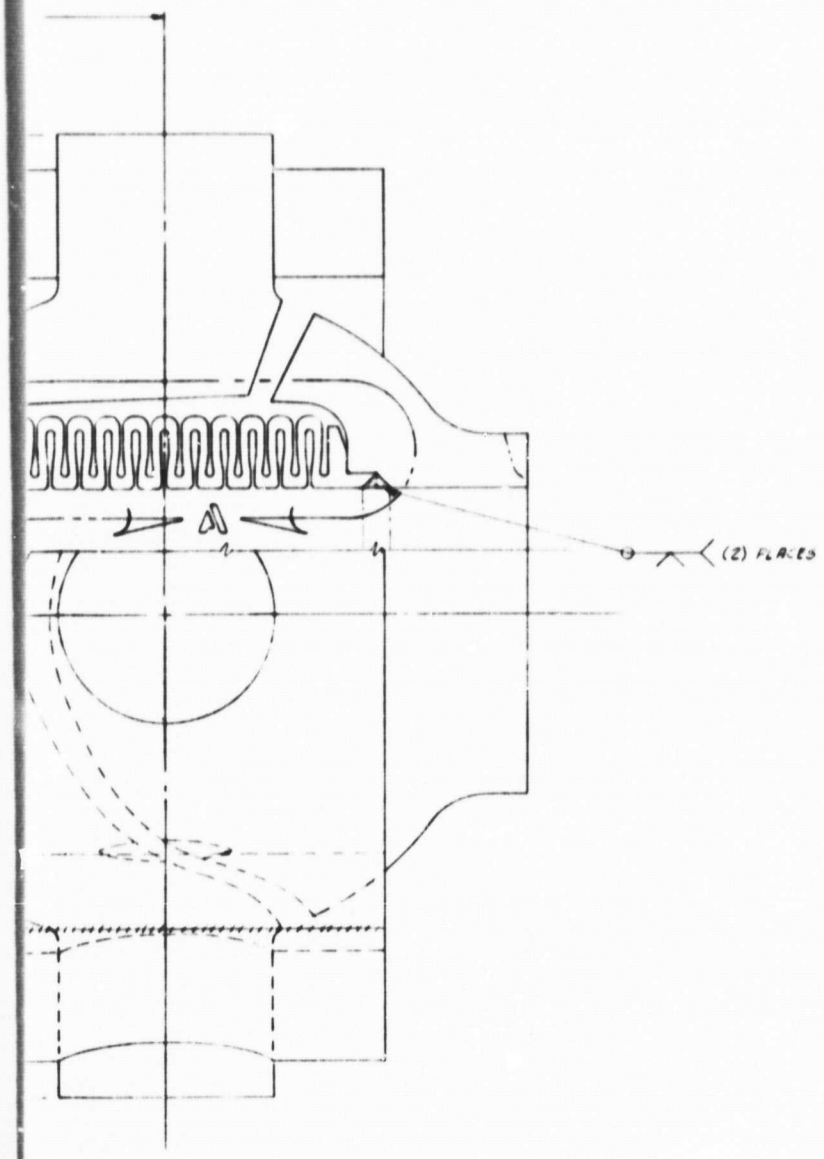
Solar Drawing No. 39701 (Sheet 4 of 4)



LC TAIL
(SHOWING BELLOWS IN MACHINED CONFIGURATION)



12-1



NOTES: UNLESS OTHERWISE SPECIFIED

1. DIMENSIONS LOCATING THE TRUE POSITION ARE BASIC
2. DO NOT SCALE DRAWING. WORK TO DIMENSIONS GIVEN
3. DRAWING INTERPRETATION PER MIL-D-70327 AND SOLAR SPEC 9-4, SECT 6.0
4. ALL MATERIAL IS INCONEL 718, HEAT TREATED PER SOLAR SPEC 9-222 EXCEPT WELDS MAY REMAIN IN AS-WELDED CONDITION
5. DESIGN CONDITIONS:
   OPERATING PRESSURE : 10,000 PSIG
   PROOF PRESSURE : 15,000 PSIG
   BURST PRESSURE : 40,000 PSIG
6. ANGULATION LIMITED TO APPROX ± 6° IN ANY PLANE THRU LONGITUDINAL AXIS
7. CALCULATED WEIGHT = 174 LBS

QTY REQD	REV LTR	PART NUMBER	DESCRIPTION	MATERIAL	SPECIFICATION	UNIT WT	ASSY WT

LIST OF MATERIAL (PARTS LIST)

APPROVED BY
STAFF
STAFF
PROJ. ENG
DESIGN
CHECKED
DRAWN 4-5-66

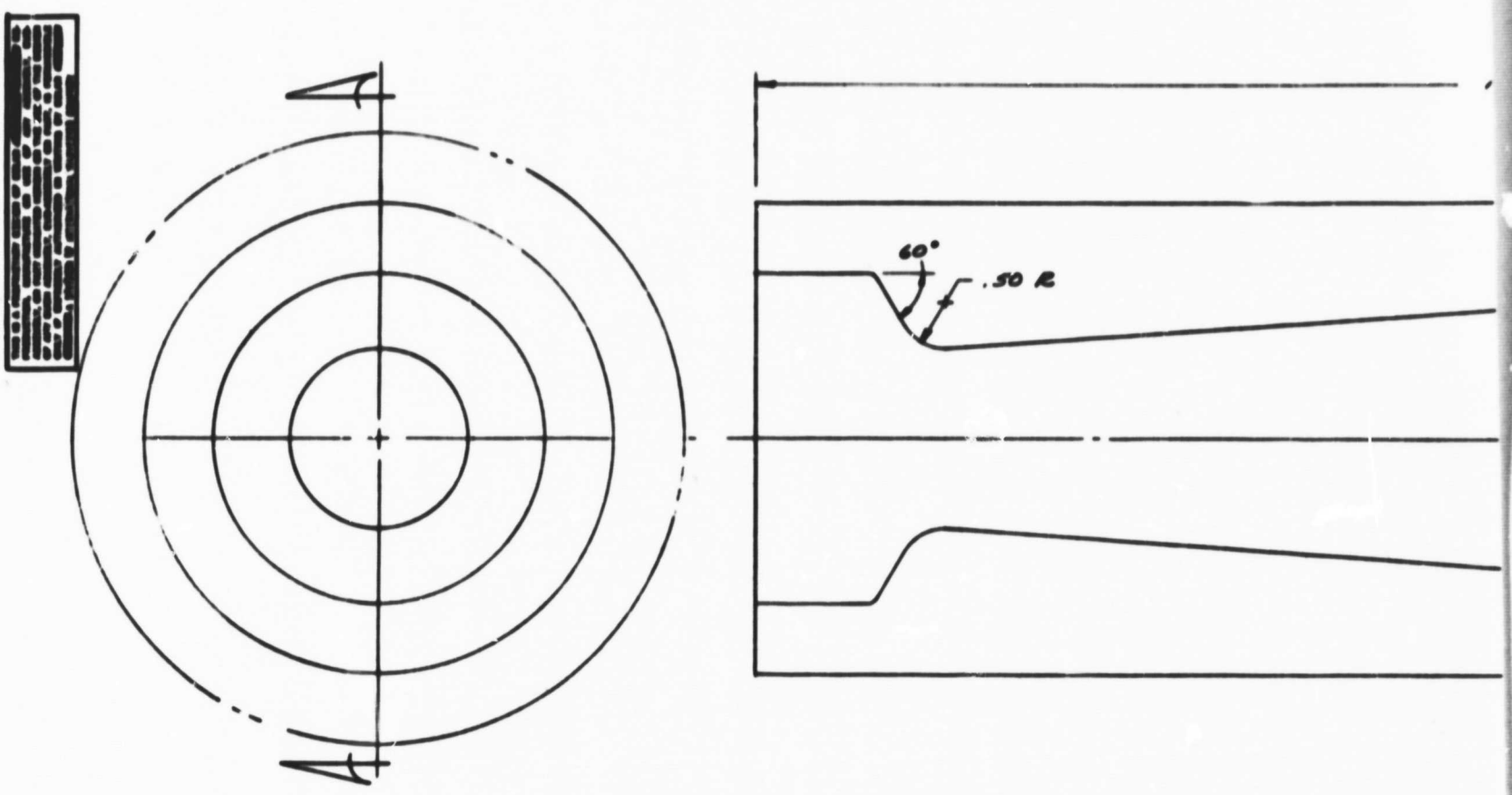
SOLAR

DRAWING TITLE
SIMEAL
MACHINED LOW HEIGHT BELLOWS

CODE IDENT NO.	SIZE	DRAWING NO.
55820	E	39704

SCALE FULL SHEET 1 OF

12-2



VENTURI DATA

THROAT DIA = 1.92

PIPE I. DIA = 3.50

$\frac{\text{THROAT AREA}}{\text{PIPE AREA}} = .3$

TOTAL PRESSURE DROP THRU VENTURI $\cong (.07\rho)\, g_{(ID=1.92)}^{\cong .73}\, g_{(ID=3.50)}$

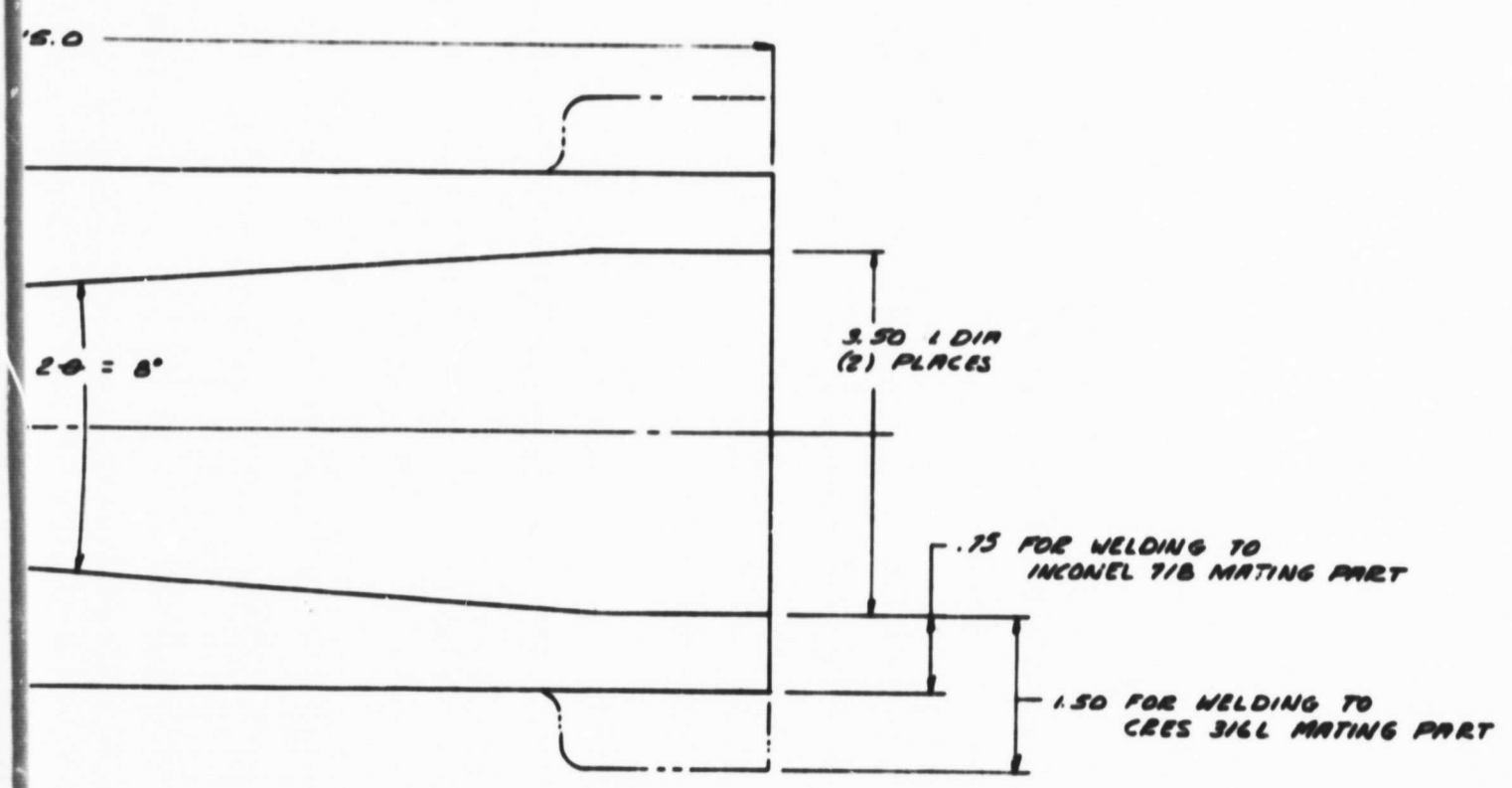
NOTES: UNLESS OTHERWISE SP

1 DIMENSIONS LOCATING T

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8

QTY REQD	QTY REQD	PART NUMBER	DESCRIPTION	MATERIAL	SPECIFICATION	UNIT WT	ASSY WT
		LIST OF MATERIAL (PARTS LIST)					

APPROVED BY
STAFF
STAFF
PROJ. ENG
DESIGN
CHECKED
DRAWN 4-7-64

SOLAR
A Division of International Harvester Company

DRAWING TITLE

VENTURI SECTION

CODE IDENT NO.	SIZE	DRAWING NO.
55820	D	39705

SCALE FULL — SHEET 1 OF 1

13-2

Three flexible joint concepts were evolved:

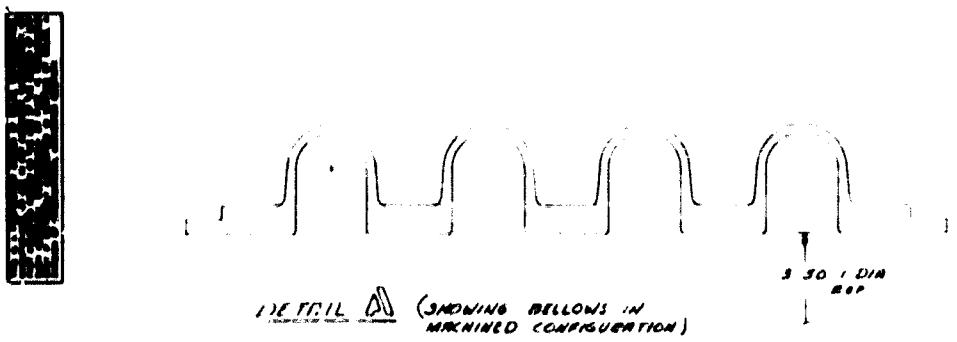
- A gimbal joint with machined, low height bellows (drawing 39704)
- A gimbal joint utilizing a machined bellows with integral root rings (drawing 39703)
- A gimbal joint utilizing a multi-ply bellows with machined root rings (drawing 39702)

It was noted that the size of the structural members of the individual gimbal joints was influenced greatly by the type of bellows being used; the higher the convolution, the greater the end thrust to be resisted in the gimbal joint structure with resulting heavier and bulkier members.

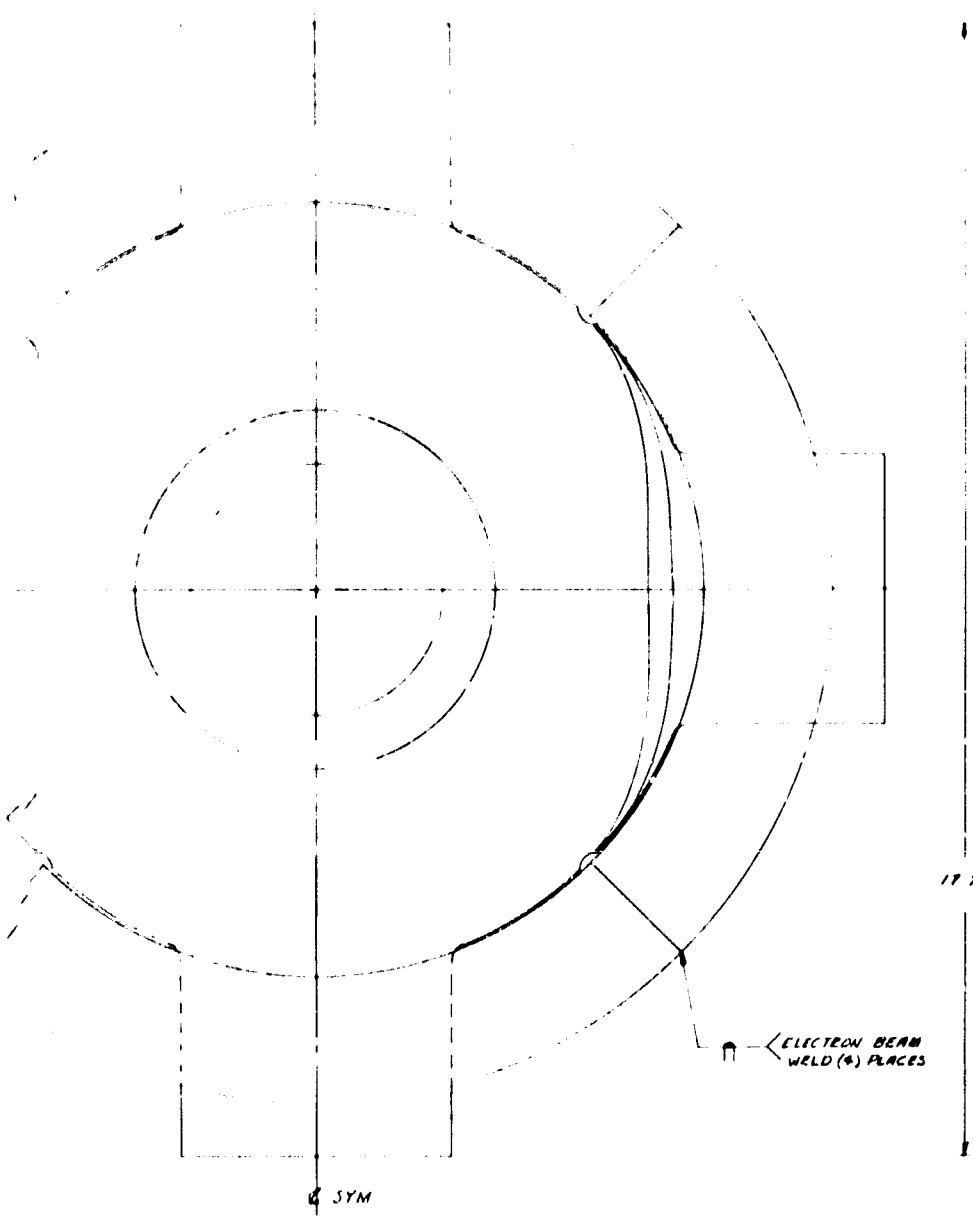
The machined bellows with integral root rings (drawing 39703) was a concept utilizing toroidal pressure shells. The relatively thin-wall torodial shell would adequately withstand the pressure but required integral root rings to provide hoop strength and stability. The principal advantage of this all machined approach is that the inner surface of the bellows is relatively smooth from a fluid flow standpoint, thereby requiring no inner liners. However, the maintenance of the tolerances necessary to guarantee the predicted characteristics would be extremely difficult.

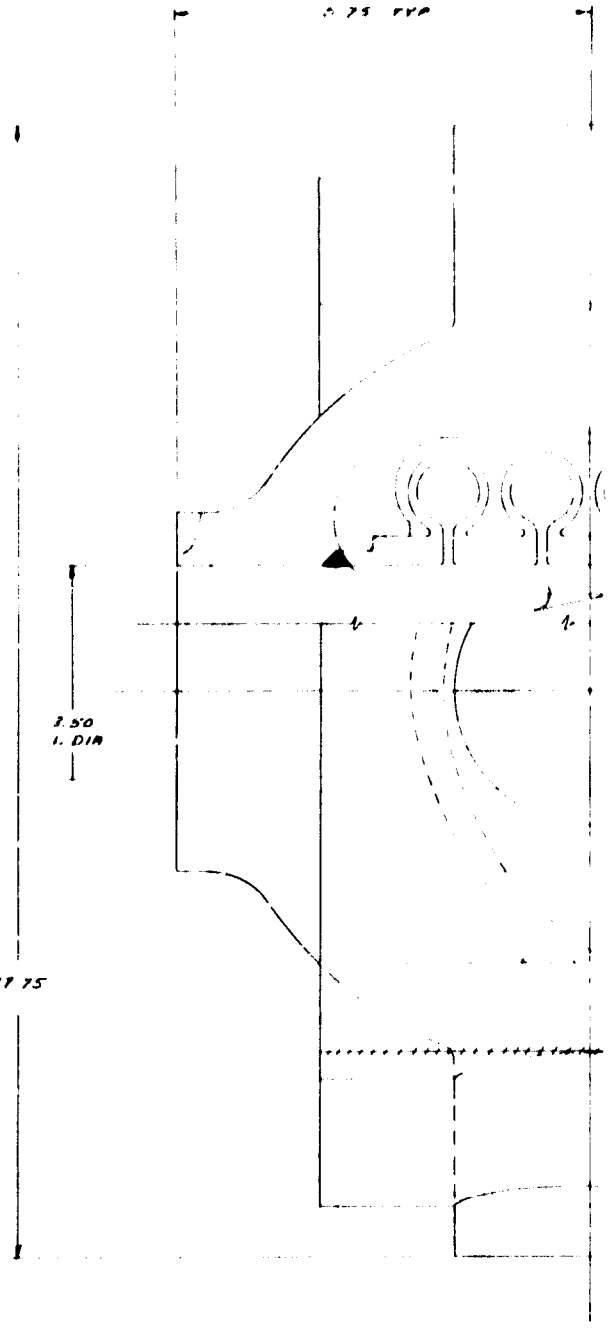
The formed multi-ply bellows with machined root rings was the development of a concept presented in Solar's original proposal. This approach is similar to the above concept in that the convolutions are toroidal shells. However, the shells are formed from tubes by hydraulic or mechanical methods in the same manner that conventional bellows are fabricated. This bellows has a very low spring rate, therefore, low angulation forces. However, it is incapable of withstanding hoop loads and requires the use of machined root rings.

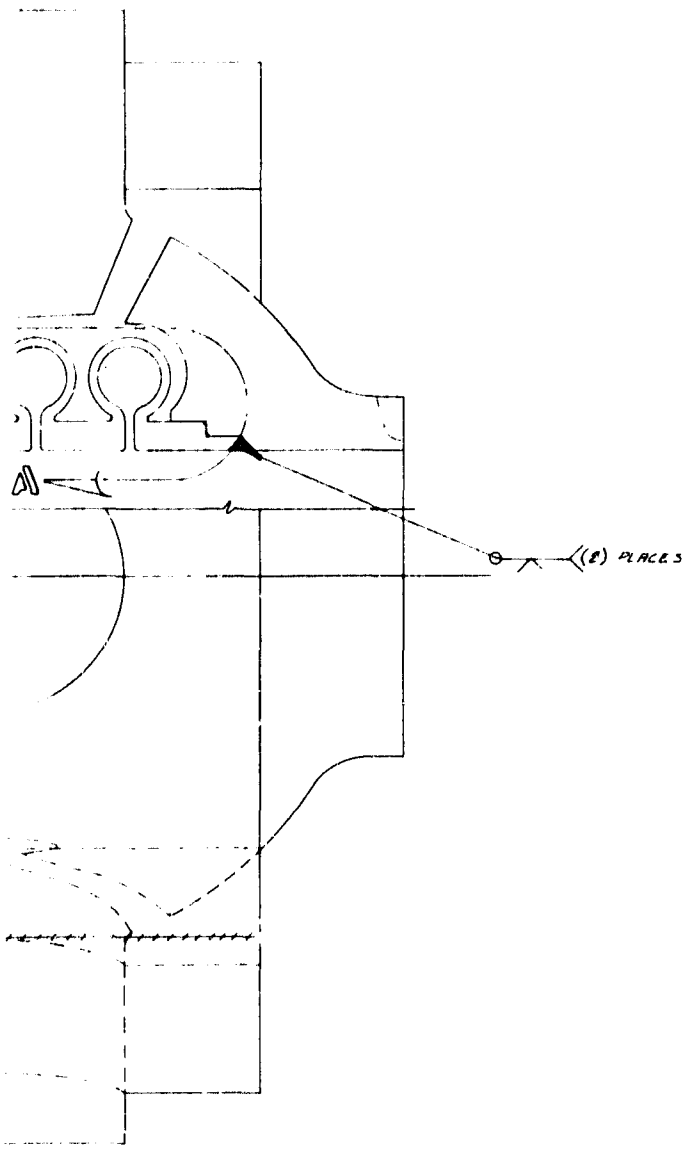
The machined, low-height bellows was the most compact design evolved and was the one recommended by Solar for future work. This machined approach permitted greater efficiency of metal usage in that the material in the bellows is placed where it performs the most useful function. The heavier root and crest of the convolution withstands the hoop pressure loads while the thin side walls permit higher cycle life and lower angulation forces. The compactness of the design and its characteristic of presenting a relatively smooth inner surface to the fluid flow also permits



DETAIL A (SHOWING BELLOWS IN MACHINED CONFIGURATION)







NOTES: UNLESS OTHERWISE SPECIFIED

1. DIMENSIONS LOCATING THE TRUE POSITION ARE BASIC
2. DO NOT SCALE DRAWING. WORK TO DIMENSIONS GIVEN
3. DRAWING INTERPRETATION PER MIL-D-70327 AND SOLAR SPEC 9-4, SECT 6.0
4. ALL MATERIAL IS INCONEL 718, HEAT TREATED PER SOLAR SPEC 9-282 EXCEPT WELDS MAY REMAIN IN AS-WELDED CONDITION
5. DESIGN CONDITIONS:
   OPERATING PRESSURE : 10,000 PSIG
   PROOF PRESSURE : 15,000 PSIG
   BURST PRESSURE : 20,000 PSIG
6. ANGULATION LIMITED TO APPROX ± 6° IN ANY PLANE THRU LONGITUDINAL AXIS
7. CALCULATED WEIGHT = 291 LBS

QTY REQ'D	QTY REQ'D	PART NUMBER	DESCRIPTION	MATERIAL	SPECIFICATION	UNIT WT

LIST OF MATERIAL (PARTS LIST)

APPROVED BY
STAFF
STAFF
PROJ. ENG.
DESIGN 4-26-67
CHECKED
DRAWN 4-26-67

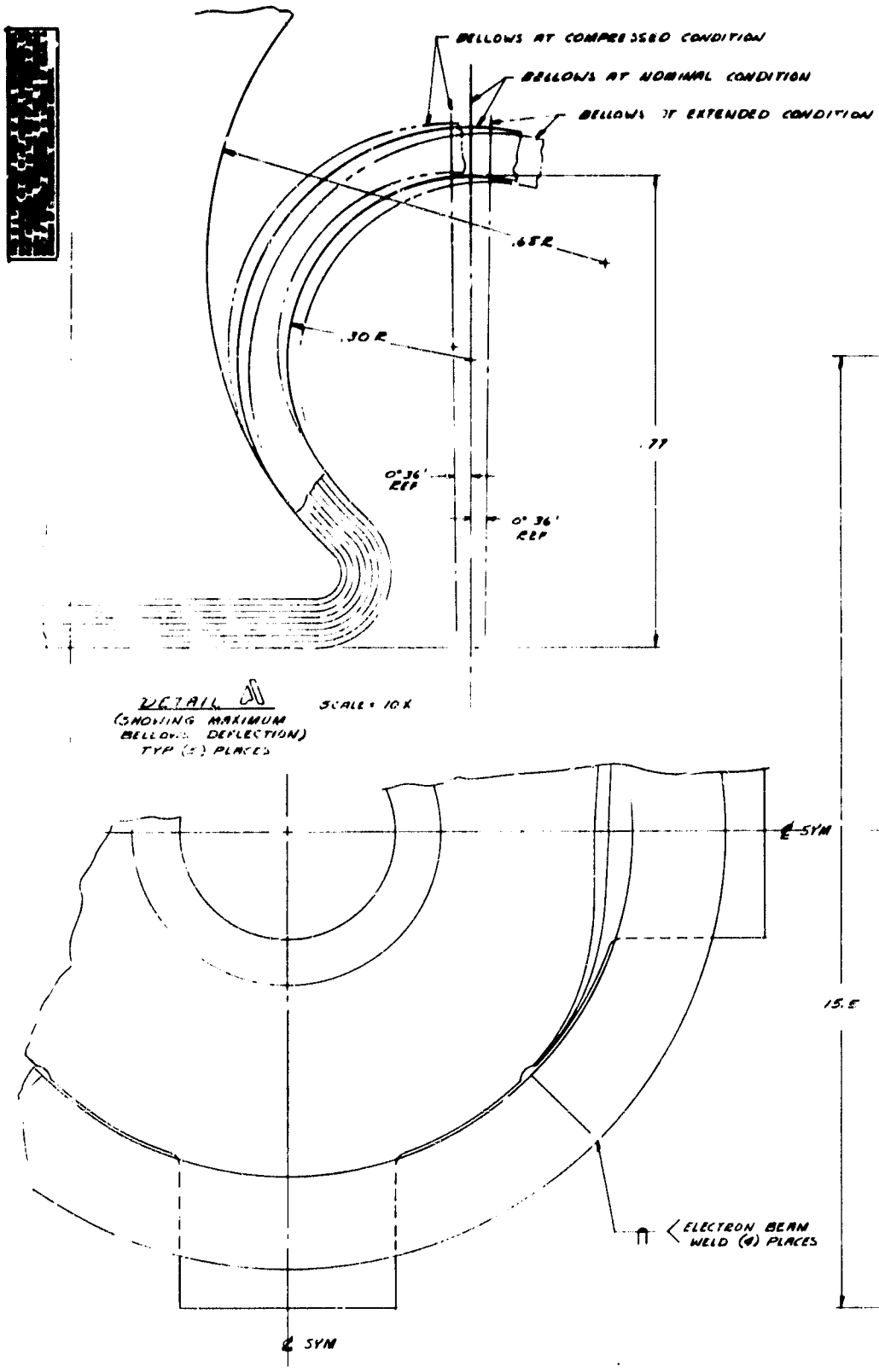
SOLAR

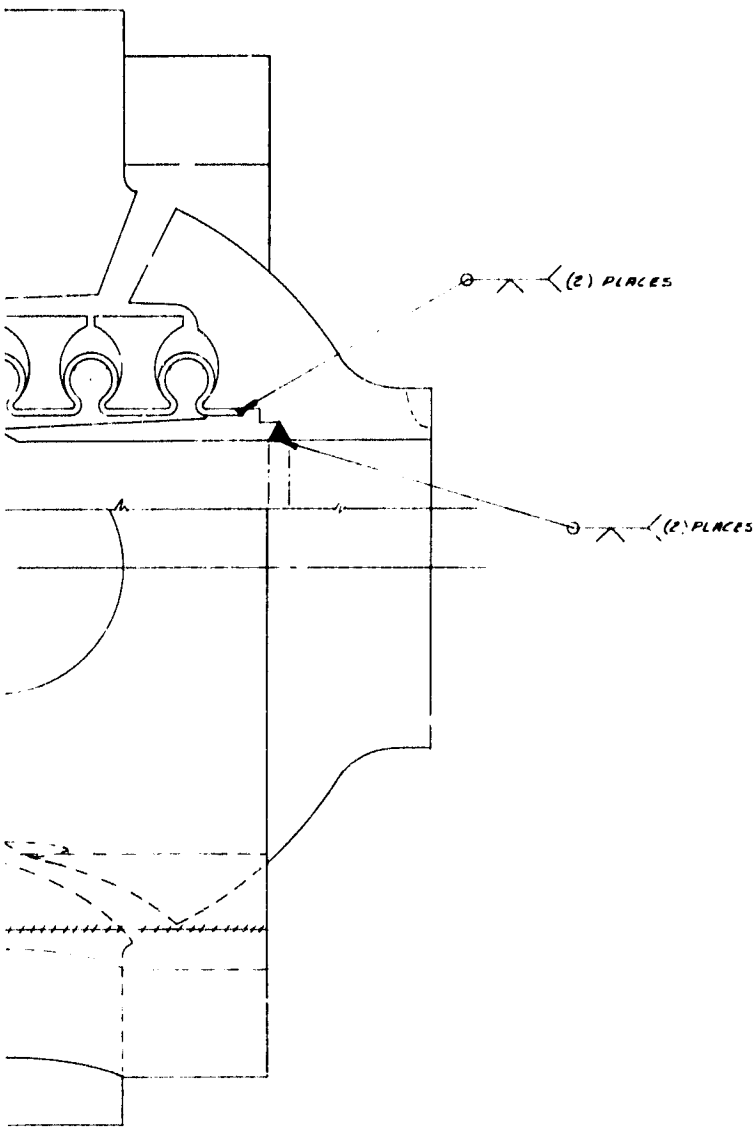
DRAWING TITLE
GIMBAL,
MACHINED BELLOWS WITH
INTEGRAL ROOT RINGS

CODE IDENT NO.	SIZE	DRAWING NO.
55820	E	39703

SCALE FULL SHEET 1 OF 1

15-2


BELLOWS AT COMPRESSED CONDITION
BELLOWS AT NOMINAL CONDITION
BELLOWS AT EXTENDED CONDITION
.65 R
.30 R
.77
0° 36' REF
0° 36' REF
DETAIL A
(SHOWING MAXIMUM BELLOWS DEFLECTION)
TYP (2) PLACES
SCALE = 10 X
℄ SYM
℄ SYM
ELECTRON BEAM WELD (4) PLACES
6.0 TYP
3.50 DIA
15.5



NOTES: UNLESS OTHERWISE SPECIFIED

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   BURST PRESSURE : 20,000 PSIG
6. ANGULATION LIMITED TO APPROX ± 6° IN ANY PLANE THRU LONGITUDINAL AXIS
7. CALCULATED WEIGHT = 290 LBS

QTY REQD	QTY ALSO	PART NUMBER	DESCRIPTION	MATERIAL	SPECIFICATION	UNIT WT	ASSY WT.

LIST OF MATERIAL (PARTS LIST)

APPROVED BY

STAFF

STAFF

PROJ. ENG. 5/1/64

DESIGN 5/1/64

CHECKED

DRAWN 4-23-64

SOLAR

DRAWING TITLE
GIMBAL, FORMED BELLOWS WITH MACHINED ROOT RINGS

CODE IDENT NO.	SIZE	DRAWING NO.
55820	E	39702

SCALE FULL SHEET 1 OF 1

16-2

its use without flow smoothing liners. These features also demonstrated, by calculation, its ability to remain stable when subjected to pressures above the 15,000 psig proof pressure requirement.

It was therefore recommended that the most practical, feasible, and efficient design was a gimbal joint utilizing the machined, low-height bellows concept (drawing 39704). The basic advantage of this gimbal type structure is that the rotational pin is an integral part of the flange and requires no welds for carrying primary loads. The gimbal ring was designed to be made in four individual segments, with the segments joined by electron beam welding. These segmental weld joints are longitudinal to the centerline, and are located at 45 degrees to the axis of the rotational pins where the bending stress is zero and the shear stress is a maximum. This shear load is much lower than the strength of the section at this point. An alternate method is to bolt the individual segments into a circular ring.

3.2.2 Cycle Life

The cycle life of the bellows was determined by an estimate of the performance life of the unit. Three operating regimes were established which were later used to determine the design of the bellows. These regimes were:

- An angulation of ± 6 degrees for 2500 cycles while unpressurized to simulate the installation of the end of the connector through the 12-inch cube
- Angulations corresponding to vibratory displacement of the end of the connector through the 0.15-inch cube for 50,000 cycles while pressurized to 10,000 psig
- Angulations corresponding to 3000 cycles of the end of the connector moving from zero to 3 inches down and return to zero within the 12-inch cube while at 10,000 psig to allow for settling and other motions.

The last motion regime was later determined by NASA at the mid-term presentation to be nonexistent. The angulations resulting from vibratory displacements, as described for the second regime, were considered to be insignificant and the bellows life in this regime was predicted to be infinite. Calculations determine that the first motion described would be the limiting regime from a stress standpoint.

The greatest confidence to meet or exceed the cycle life was developed in the machined low-height bellows. That bellows design has the highest spring rate and therefore the highest angulation force required to deflect through the 12-inch cube.

This feature is felt to be an advantage from the standpoint of vibration dampening of the entire connector system. The line has a large mass and requires the use of heavy forces to maintain high resonant frequencies. This bellows-induced spring force is felt to be sufficiently high so that no dampening is necessary in the pin-in-ring connection. For this reason, the pins are coated with dry molybdenum disulfide for lubrication, anti-galling, and to decrease the installation forces. A criterion for design of the gimbal component was that the gimbal joint should, under all loadings, exceed the strength of the pipe. Therefore, the design of the gimbal joint is such that when physically bottomed at the 6-degree maximum angulation, the gimbal joint can force the pipe to bend plastically without damage to the joint. The gimbal was also designed to accept separating loads which are greater than the bursting strength of the pipe.

In the area of pressure stability, study revealed that instability or squirm would not occur below 20,000 psi internal pressure. As the pressure approaches 40,000 psi, the flat sides of the convolutions will bulge until they come into contact, thus preventing any further instability until burst occurs. The average hoop stress at 40,000 psi internal pressure is 111,000 psi. The material chosen, Inconel 718, has an ultimate tensile strength of 180,000 psi.

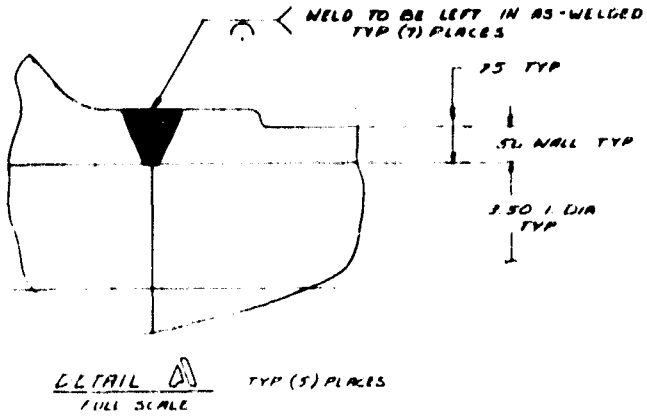
3.2.3. Mid-Term Presentation

A summary of the Phase IA effort and the following drawings were included at the mid-term presentation:

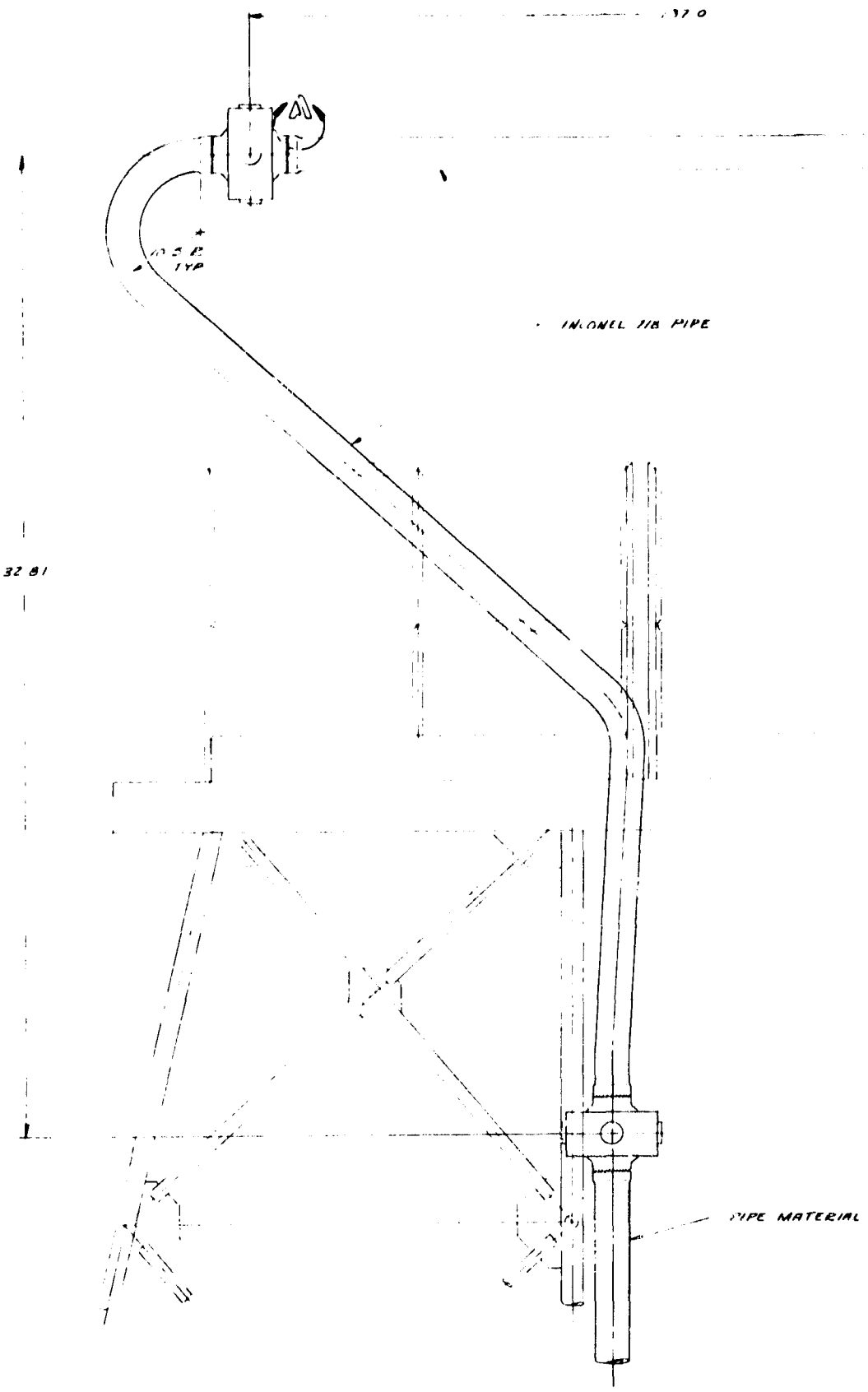
39704	Gimbal	-	Machined Low Height Bellows
39704	Gimbal	-	Machined Bellows with Integral Root Rings
39702	Gimbal	-	Formed Bellows with Machined Root Rings
39701	Motion Studies	-	High Pressure Line Configurations
39705	Venturi Section	-	
39706	Line Assembly	-	High Pressure Connector

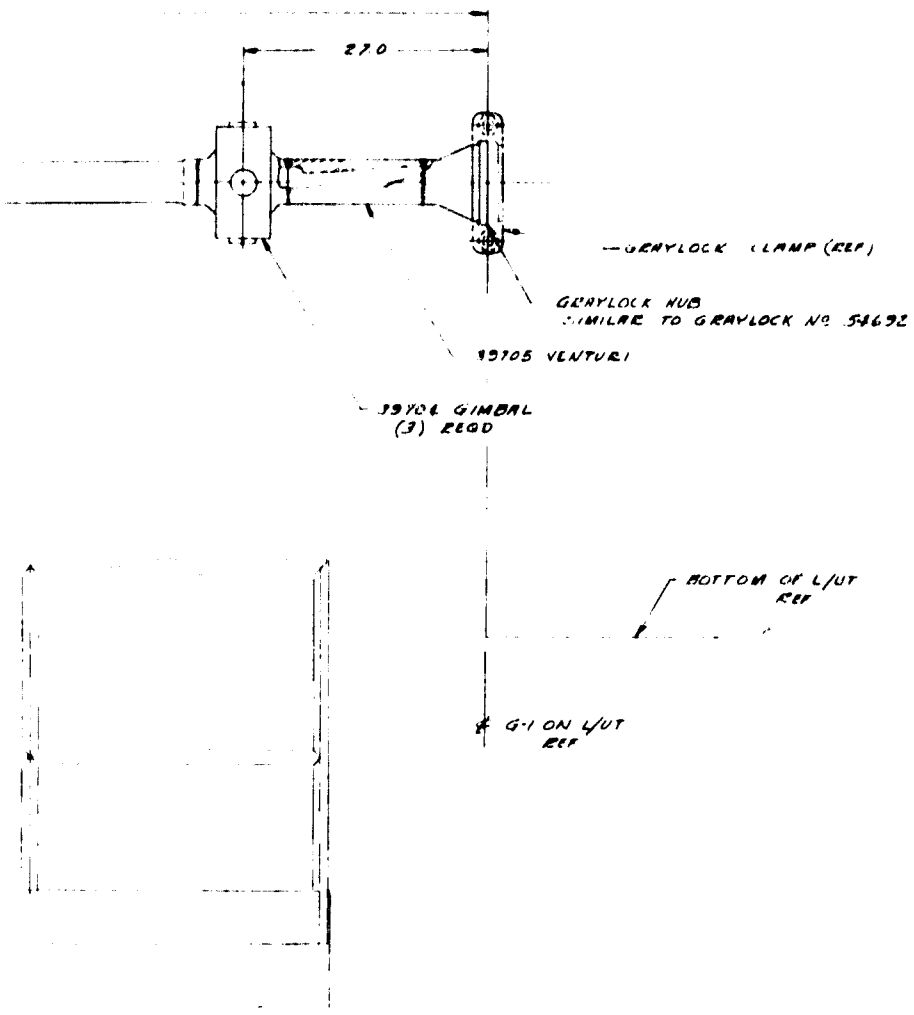
These drawings are included in Appendix A of this report.

Solar's instructions to continue were contained in a letter from the NASA Technical Supervisor, dated 25 May 1964. These instructions accepted Solar's recommendations, and permitted the Phase IB effort to concentrate on the design of the machined low-height gimbal joint and the system depicted on Solar drawing 39706.



C.L TAIL Δ1 TYP (5) PLACES
FULL SCALE





TO BE DETERMINED LATER

NOTES: UNLESS OTHERWISE SPECIFIED
1. DIMENSIONS LOCATING THE TRUE POSITION ARE BASIC
2. DO NOT SCALE DRAWING. WORK TO DIMENSIONS GIVEN
3. DRAWING INTERPRETATION PER MIL-D-70327 AND SOLAR SPEC 9-4, SECT

19-2

SET REQD	QTY REQD	PART NUMBER	DESCRIPTION	MATERIAL	SPECIFICATION	UNIT WT
		LIST OF MATERIAL (PARTS LIST)				REF

APPROVED BY	
STAFF	
STAFF	
PROJ. ENG.	5/14/64
DESIGN	5/14/64
CHECKED	
DRAWN	5/12/64

SOLAR

DRAWING TITLE
LINE ASSY,
HIGH PRESSURE CONNECTOR

CODE IDENT NO.	SIZE	DRAWING NO.
55820	E	39706

SCALE 1/8 SHEET 1 OF 1

Solar was also directed to develop the individual components of the flexible connector system as individually complete assemblies with Graylock hubs interfacing each of the gimbals to each of the pipe segments.

3.3 COMPENSATOR DESIGN STUDY

During the design study phase, attention was also paid to compensator designs. While the use of a compensator did not appear to be practical for the flexible connector system, the compensator has merit in the replacement of the expansion loops in the vertical tower pipe runs. A compensator concept was demonstrated at the mid-term presentation. This concept was characterized by the advantage of having no internal faying edges or entrapments and therefore would ensure a high level of cleanability and cleanliness during operation. This compensator for 10,000 psi would utilize a development of the same bellows designed for the gimbal joints. As a result, NASA indicated to Solar in the letter of 24 May that Solar was to continue its study effort in the area of the compensators for various sizes and pressure ranges which were presently being used in the LUT piping system

Information was received from Mr. Jack Kunkel of KDE-3 regarding the sizes of the lines and the operating pressures for which compensators would be desirable. They were:

Size	Medium	Operating Pressure
4-inch schedule 40	Hydraulic	500 psi
3-inch double extra strong	Helium	6000 psi
1-1/2-inch schedule 40	Air	125 psi
2-inch schedule 40	GN_2	50 psig
3-inch double extra strong	GN_2	6000 psig
2-inch double extra strong	GH_2	6000 psig

An effort at Solar, parallel to the flexible connector system design, was initiated to develop these designs. The results of the compensator study are contained in Section V of this report.

IV. PHASE IB - SPECIFIC DESIGN

During the specific design phase, work was performed in the following areas:

- Design considerations
- Stress Analysis
- Reliability
- Fabricability
- Cost

It must be mentioned at this time that in all instances no great strides in increasing the state of the art have been made; each of the designs proposed and each of the methods to be utilized to fabricate the flexible connector system are extensions of Solar's experience in the flexible connector field. Solar's long-standing experience in the manufacture and design of gimbal joints similar to that proposed for this system is the basis for the design techniques utilized on this program. Solar's experience in developing flexible connector systems and determining the individual component requirements, based upon the overall system requirements, has been used in the development of this system. The electron beam welding, the bellows design, and the gimbal joint design are all approaches which have been previously utilized to manufacture similar products for other uses. Solar feels confident that this design will meet the desired requirements in their first usage.[1]

4.1 DESIGN CONSIDERATIONS

On 7 May 1964, the detail design effort was initiated on the line assembly (Solar drawing 39706) and the gimbal joint (Solar drawing 39704) as directed by NASA. Preparation of individual detail drawings of each component of the gimbal joints and the line assembly was started. Concurrently, the Gray Tool Company of Houston, Texas, was requested to prepare designs and submit prices for a Graylock hub design using Inconel 718 as the material. Upon completion of the preliminary design drawings of the various gimbal joint components, inquiries were sent out to obtain preliminary pricing information and fabricability comments from various sources in the forging

1. Solar's capabilities in research, design, and manufacture of similar components are described in "Solar Space Components," Solar document SD 603, submitted as an enclosure to this report.

field. These initial inquiries were based on an assumed flange configuration. Prior to receipt of these evaluations, Solar received, from the Gray Tool Company, the configuration of the Inconel 718 hub. This hub design indicated that Solar had overestimated the size of the hub. Consequently, the forging became less complicated and would probably result in a lower cost in the final configuration.

At the beginning of this phase of the study, it was determined that the controlling material specifications should be those covering the types most readily available in the industry. Solar maintains its own specifications for Inconel 718, and these specifications result in extremely high-quality materials with strength levels exceeding those of most other specifications. However, it was felt that this line assembly, being ground support equipment, could be designed with slightly heavier wall thicknesses and slightly lower stress levels. In this manner, it would be possible to take advantage of the more readily available Inconel 718 material, such as described in Rocketdyne Division of North American Aviation, Inc., Specification RB01070-038. The use of this Rocketdyne specification would ensure the availability of the necessary raw material with less lead time.

Inquiry was also made of the various pipe manufacturers for Inconel 718 pipe in the following configurations:

- 3-1/2-inch ID by 1/2-inch average wall
- 4-inch schedule 160, a standard ASA pipe size which closely approximates the size of the line assembly being studied. This size was felt to be more available.
- 3-1/2-inch schedule 80 for 6000 psi for use as a comparison
- 3-1/2-inch ID by 0.30-inch average wall

Prices and recommendations were requested for both cold drawn and extruded pipe. Interestingly enough, 3-1/2-inch ID and 1/2-inch average wall thickness pipe was found to be less expensive than the corresponding standard ASA pipe size and could be obtained as extruded seamless tubing which would ensure a higher inherent reliability. The M.L. Sheldon Co. was also contacted for information regarding the Cu-Ni lined, Type 4320 alloy steel pipe which they are now providing to Complex 39. It was determined that this composition type pipe would present handling and welding problems. As a comparison, inquiries were also made for a CRES 316L pipe with a 3-1/2-inch

ID and a 1.6-inch wall. It was found, as previously predicted, that the Inconel 718 in the sizes mentioned was cheaper and more readily available than the extremely heavy-wall CRES 316L configuration. Only one response was received on the CRES 316L configuration, and this response was qualified to a single quantity of 100 feet with no assurance of success.

Inquiries were also made to the forging industry concerning fabrication of the venturis previously mentioned. On 25 June 1964, Mr. Jack Kunkel and Mr. Geoffry Bottomly, representing NASA, made a survey of Solar's facility in San Diego. At this time it was offered by Mr. Jack Kunkel that the work on the venturi be stopped as it would not be required by NASA. This was confirmed at the final presentation on 7 July. The design was modified to eliminate the venturi and to move the gimbal joint, which was adjacent to the venturi, to the end of the connector. The decision was made to increase the bend radius of the bent pipe instead of increasing the length of the straight section. This change maintained the same relative angulation of each of the gimbal joints and resulted in a more easily fabricable bent pipe assembly.

The final design of each component was based upon the following manufacturing procedures:

- Weld the pipe to the Graylock hubs prior to heat treat, forming a complete assembly.
- Age the complete assembly after welding.
- Age all the gimbal joint details as details.
- Weld the bellows to the gimbal flange after aging and leave in the as-welded condition.
- Electron beam weld the gimbal ring quadrants as the final assembly procedure. These welds remain in the as-welded condition.

The bellows-to-gimbal flange weld can remain in the as-welded condition since this weld is primarily a seal weld and carries, as its only structural load, that couple necessary to overcome the angular spring rate of the bellows. Since an Inconel 718 weld in the as-welded condition has a relatively high strength, this weld has an adequate safety factor to allow it to remain in the as-welded condition.

The basis for the original design of this gimbal joint was that the electron beam weld for joining the ring quadrants may also remain in the as-welded condition

since the weld joint is in the plane of zero-bending and maximum shear and the area in shear is a maximum. The heat-affected zone of an electron beam weld is highly confined and would not affect the strength of the ring in those areas where bending stress is a maximum. The limiting stress for the design of the ring was the bending stress, which is a maximum at the planes of the pins. This stress is a function of the moment of inertia of the section at the pins. Since the pin holes do not exist in the plane of the weld, the area is much greater and the margin of safety in shear is high.

The gimbal joint will be provided with a removable external non-metallic flexible boot to prevent the entrance of foreign particles into the gimbal structure. The boot design will be worked out with an acceptable vendor in this field.

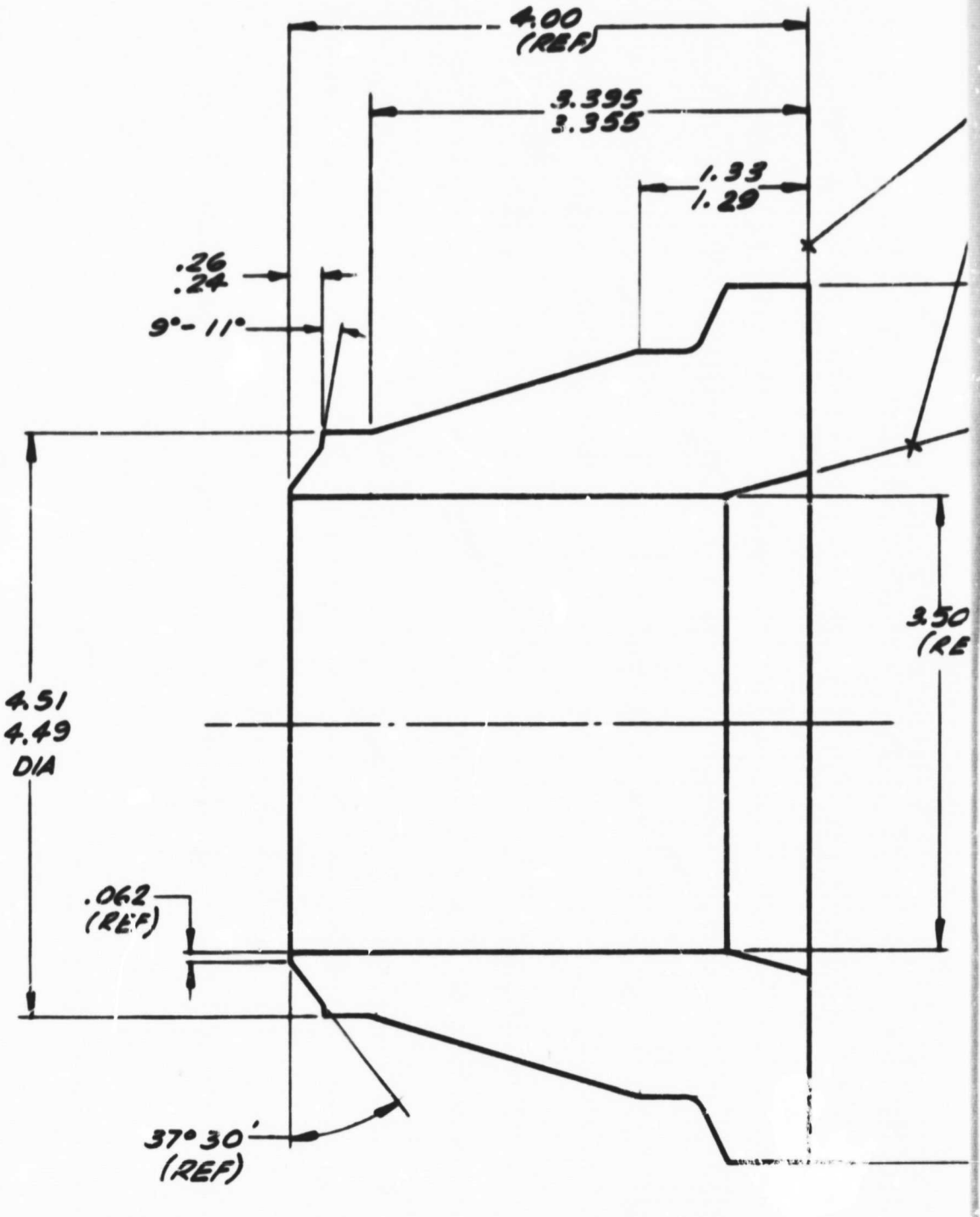
Each component will be cleaned in accordance with Marshall Space Flight Center Specification 164 and packaged in clean polyethylene bags prior to delivery. The gimbal joints will be provided with Graylock blank hubs as closures, and the pipe assemblies will be provided with standard plastic pipe closures to preclude the contamination of the line.

4.2 FABRICABILITY

Sources for pipe bending have indicated that 3-1/2-inch ID by 1/2-inch wall Inconel 718 pipe would present no problem in the configuration indicated on Solar drawing 41313.

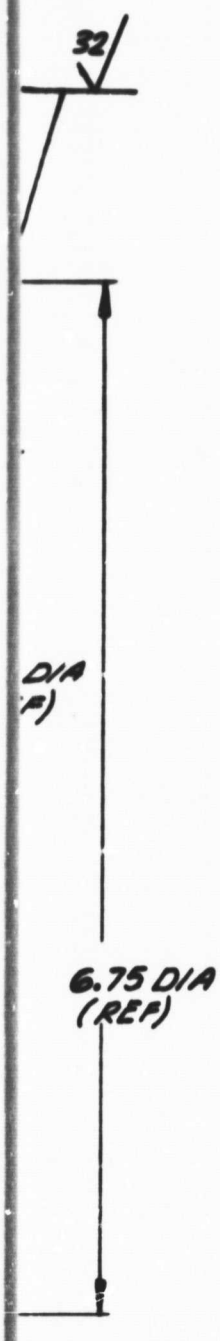
The gimbal ring, Solar drawing 41306 will be machined from a ring forging. The ring forging will be machined on the outside diameter only and split into four segments. The split faces of the segment will be ground, rematched into a complete ring, and the inside diameter bored to the required dimension. In this manner, the inside diameter will be established and the minimum wall thickness will have been maintained.

The bellows will be machined from a pipe segment or ring forging. The initial operation will be rough machining, followed by grinding to the desired shape and finish requirements. In accordance with Solar drawing 41309, a maximum of 32 rms finish is allowed at the roots and crowns of the convolution. An optional finishing method is shown that permits electro-chemical metal removal methods. With this option, a maximum of 64 rms is allowed since the remaining surface marks would have rounded edges.



NOTES: UNLESS OTHERWISE SPECIFIED

1	DIMENSIONS LOCATING THE TRUE POSITION ARE BASIC
2	DO NOT SCALE DRAWING. WORK TO DIMENSIONS GIVEN
3	DRAWING INTERPRETATION PER MIL-D-70327 AND SOLAR SPEC 9-4, SE
4	ONLY THE ITEM ON THIS DWG, IDENTIFIED E VENDORS NAME, ADDRESS & PART NO. HAS TESTED & APPROVED BY SOLAR. A SUBSTITUTE ITEM SHALL NOT BE USED WITHOUT PRIOR TESTING AND APPROVA BY COGNIZANT SOLAR CHIEF PROJECT ENGI.
5	ITEM SHALL BE PURCHASED FROM GR TOOL CO., HOUSTON 1, TEXAS, DWG N I-0341
6	VENDOR TO PERMANENTLY IDENTIFY PA NO PER MIL-STD-130; BRAND SOLAR A NO IN PARENTHESIS WITH PERMANENT INK



SOURCE CONTROL DWG

QTY REQD	QTY REQD	PART NUMBER	DESCRIPTION	MATERIAL	SPECIFICATION	UNIT WT
		-1			INCONEL 718	16.93

← ASSY NO. LIST OF MATERIAL (PARTS LIST)

THIS IS A PROPRIETARY DESIGN OF SOLAR. REPRODUCTION, MANUFACTURE, OR USE OF ANY ASSEMBLY, SUBASSEMBLY, OR PART INDICATED HEREIN OR THE USE OF THE DESIGN OF ANY SUCH ASSEMBLY, SUBASSEMBLY OR PART, IF PERMISSIBLE ONLY IF EXPRESSLY AUTHORIZED IN WRITING BY SOLAR, A DIVISION OF INTERNATIONAL HARVESTER COMPANY.

APPROVED BY

GOVT CONTRACT NO.

PROJECT

DESIGN

CHECK

DRAFT

DESIGN ACTIVITY

CUSTOMER

SOLAR — A Division of International Harvester Company

DRAWING TITLE

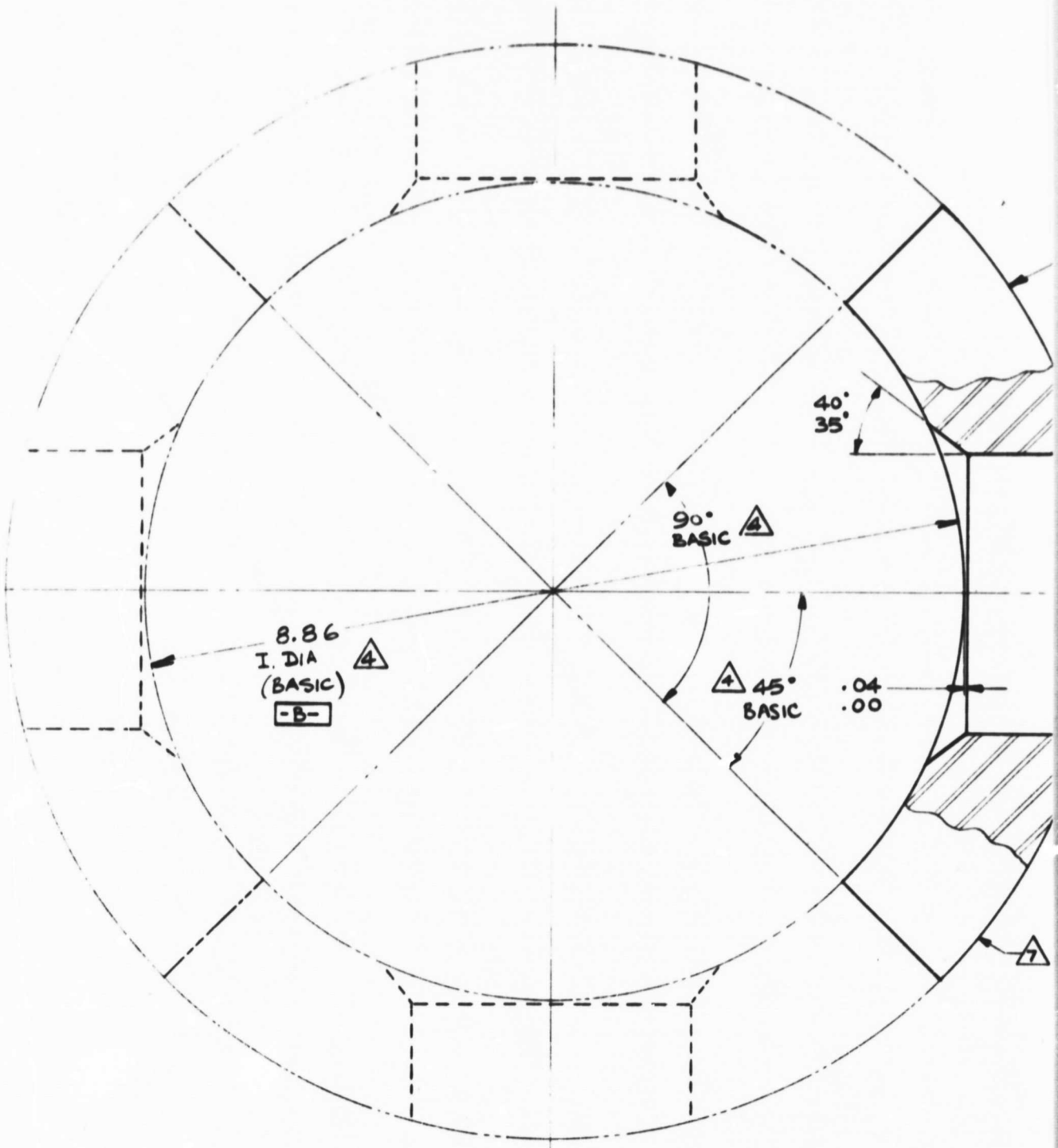
UNION

3.50 I.D. PIPE

SIZE	CODE IDENT NO.	DRAWING NO.
C	55820	41313

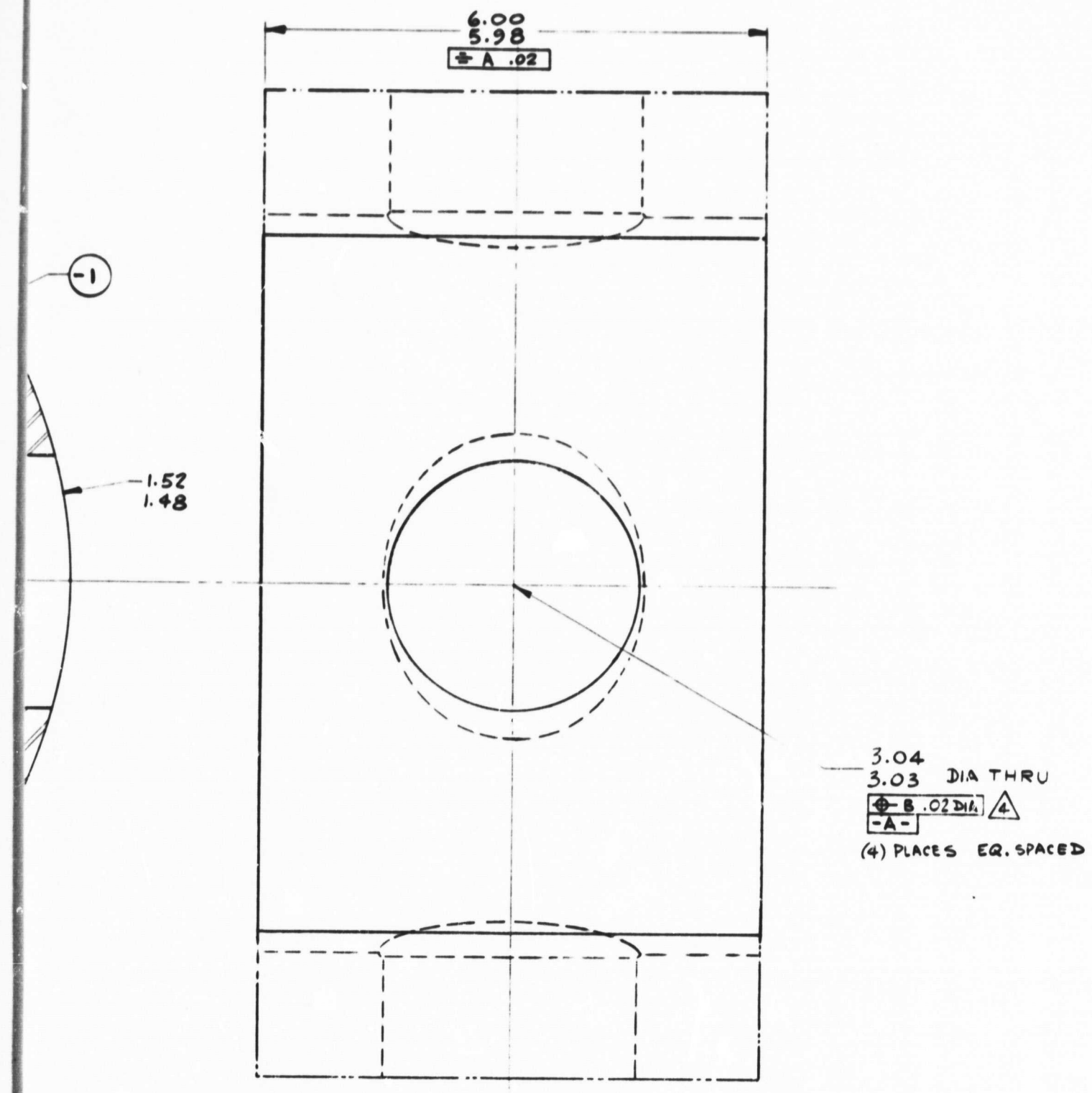
SCALE FULL | TOTAL WT. 16.93 | SHEET 1 OF 1

25-2



NOTES: UNLESS OTHERWISE SPECIFIED

1. DIMENSIONS LOCATING THE TRUE POSITION ARE
2. DO NOT SCALE DRAWING. WORK TO DIMENSIONS
3. DRAWING INTERPRETATION PER MIL-D-70327 AND SOLAR SPEC 9-4. SECT 6.0
4. DIMENSIONS OF RING QUADRANT WITH (3) OTHER RING QUADRAN 8.84 - 8.88 I.D. RING AT NEX 41308-0 GIMBAL ASSY. TOLERAN QUADRANT THAT BUTT AGAINST A TO BE MATCHED TO PROVIDE CLEAR ELECTRON BEAM WELDING SHOW
5. INCONEL 718 ALLOY BAR & FO ROCKETDYNE SPEC RB 0170-038
6. HEAT TREAT -1 (INCONEL 718) ULTIMATE TENSILE STRENGTH & I
7. IDENTIFY WITH SOLAR PART Nº E ETCH METHOD PER MIL-STD-13C SOLAR SPEC MPS 7-815
8. IDENTIFICATION & TRACEABILITY TO SPEC 60B00010 PARA 14.3



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COMPLY WITH NSFC

QTY REQD	QTY REQD	PART NUMBER	DESCRIPTION	MATERIAL	SPECIFICATION	UNIT WT	NOTE
		-1	QUADRANT	INCONEL 718	5	74.15	

LIST OF MATERIAL (PARTS LIST) ASSY WT

APPROVED BY
STAFF
STAFF
PROJ. ENG. 7.21.64
DESIGN 7/9/64
CHECKED 7/10/64
DRAWN 6-22-64

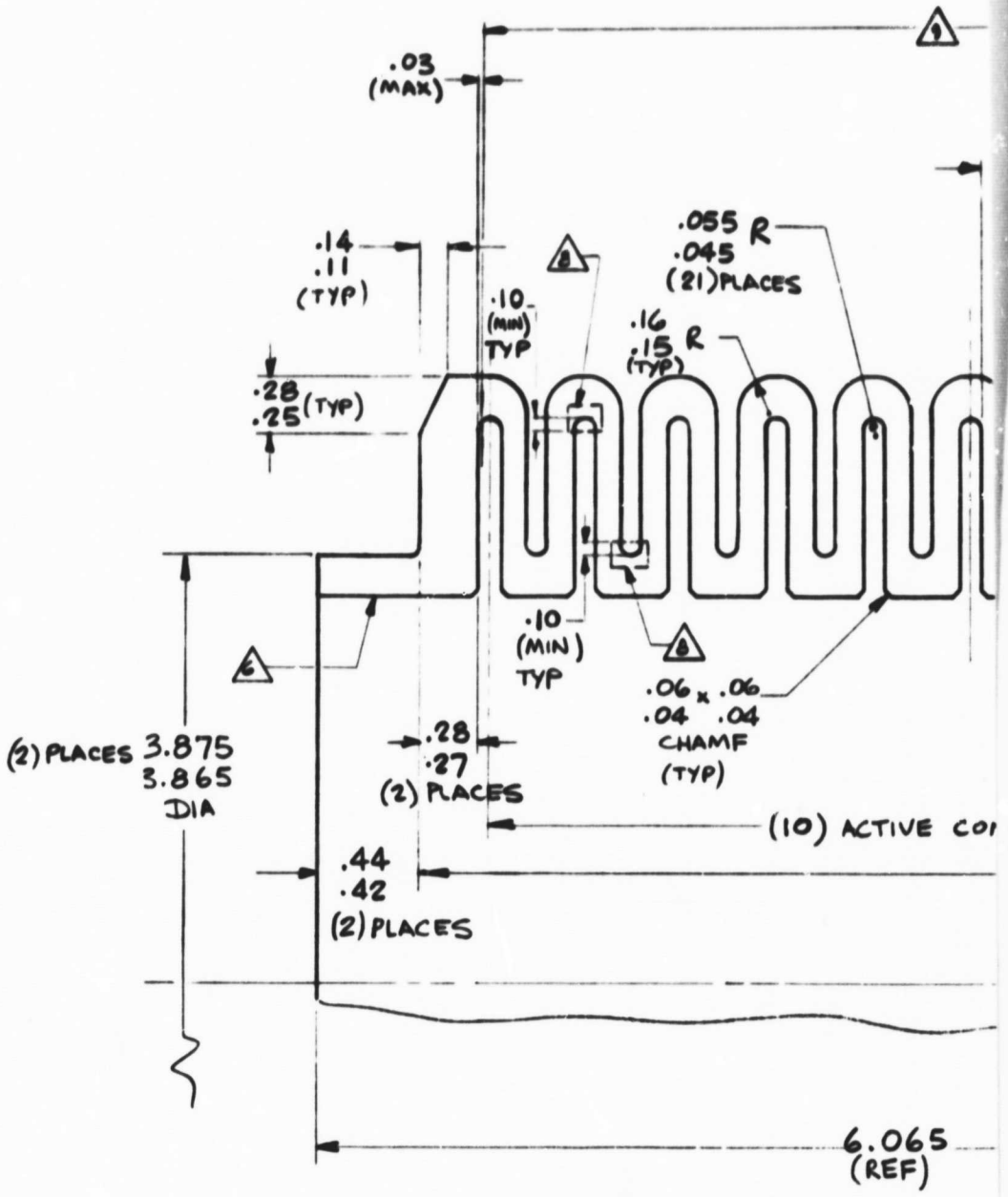
SOLAR SAN DIEGO 12, CALIFORNIA
A Subsidiary of International Harvester Company

DRAWING TITLE
RING, QUADRANT, GIMBAL
±6° 10,000 PSI (MATCHED SET)

CODE IDENT NO.	SIZE	DRAWING NO.
55820	D	41306

SCALE FULL SHEET 1 OF 1

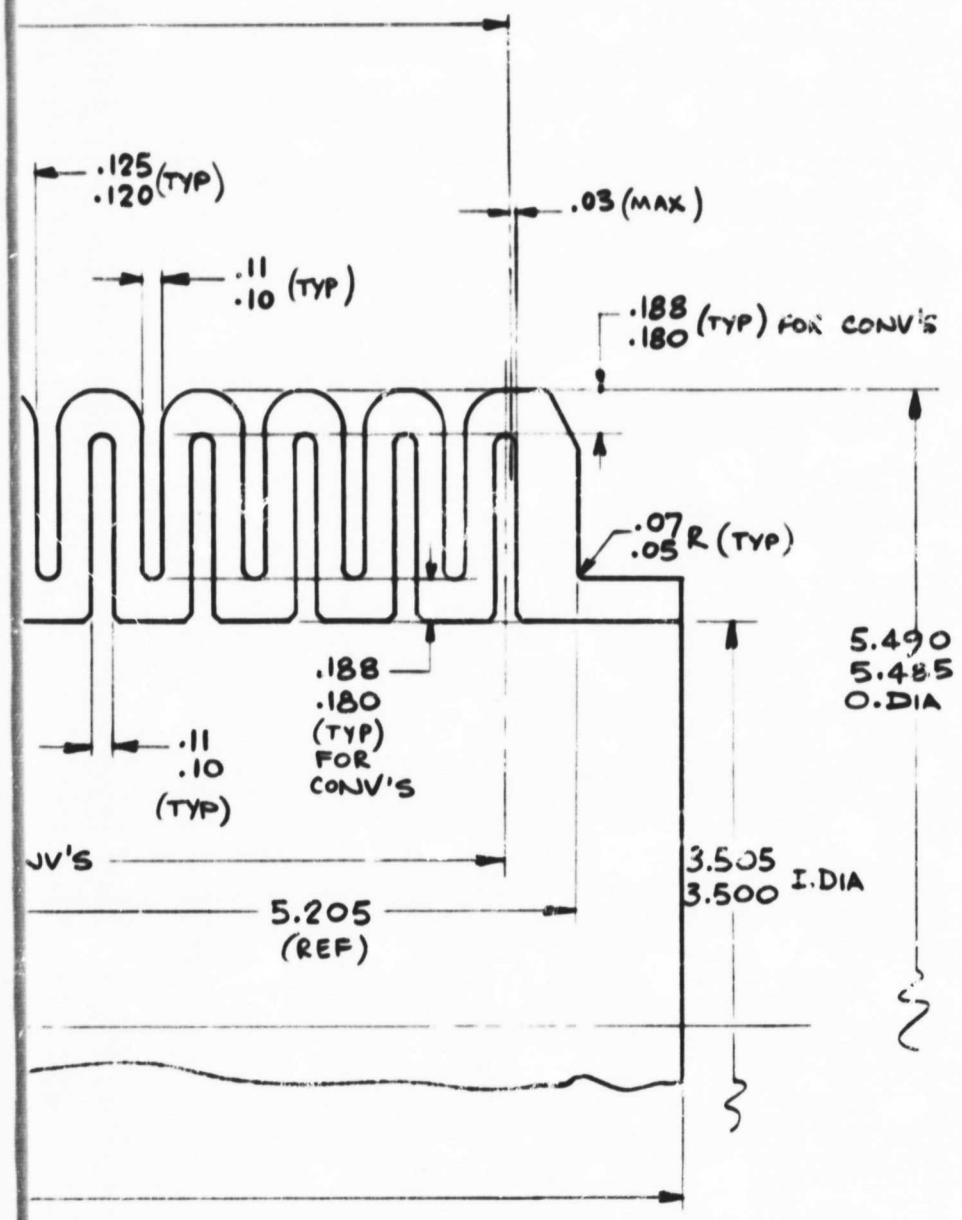
26-2



NOTES: UNLESS OTHERWISE SPECIFIED

1. DIMENSIONS LOCATING THE TRUE POSITION ARE
2. DO NOT SCALE DRAWING. WORK TO DIMENSIONS
3. DRAWING INTERPRETATION PER MIL-D-70327 AND SOLAR SPEC 9-4, SECT 6.0
4. INCONEL 718 ALLOY BARS & FORGIN PER ROCKETDYNE SPEC RB017C-03
5. HEAT TREAT INCONEL 718 ITEMS TO 18C ULTIMATE TENSILE STRENGTH & 15%
6. IDENTIFY WITH SOLAR PART Nº 413C HIGH NUMERALS & NON-PERMANEN SOLAR SPEC 9-6 CLASS "E".
7. IDENTIFICATION & TRACEABILITY TO MSFC SPEC 60B00010 PARA 14.3
8. SURFACE INSIDE THIS AREA TO BE 32 NO SCRATCHES DEEPER THAN .001 (T CRESTS & 10 PLACES FOR ROOTS OF IF FINAL FINISH IS ACCOMPLISHED ELECTRO-CHEMICAL METHOD, SU 64 R.M.S.(MAX) WITH SCRATCHES
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27-1



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RMS.
CONDITION

QTY REQD	QTY REQD	PART NUMBER	DESCRIPTION	MATERIAL	SPECIFICATION	UNIT WT	NOTE
		-1		INCONEL 718	4		

LIST OF MATERIAL (PARTS LIST) — ASSY WT 14.06 lb

APPROVED BY	
STAFF	
STAFF	
PROJ. ENG	R. Kuss 7.21.64
DESIGN	M. Smith 6/23/64
CHECKED	Conlon 7/10/64
DRAWN	W. Wells 6-17-64

SOLAR, a division of International Harvester Company

DRAWING TITLE

BELLOWS, MACHINED - ±6°, 10,000 PSI

CODE IDENT NO.	SIZE	DRAWING NO.
55820	D	41309

SCALE TWICE SIZE — SHEET 1 OF 1

27-2

A second operation in the formation of the finished bellows is required. This operation is a compression set using internal pressure in accordance with Solar drawing 41310. The finished machined bellows will be placed in a press and pressurized to approximately 35,000 psig while being axially compressed. This operation will cause the side walls to bulge slightly and take a permanent set which will lower the operating stresses and increase the stability.

The bellows will then be stress relieved and aged.

4.2.1 Electron Beam Welding Procedure

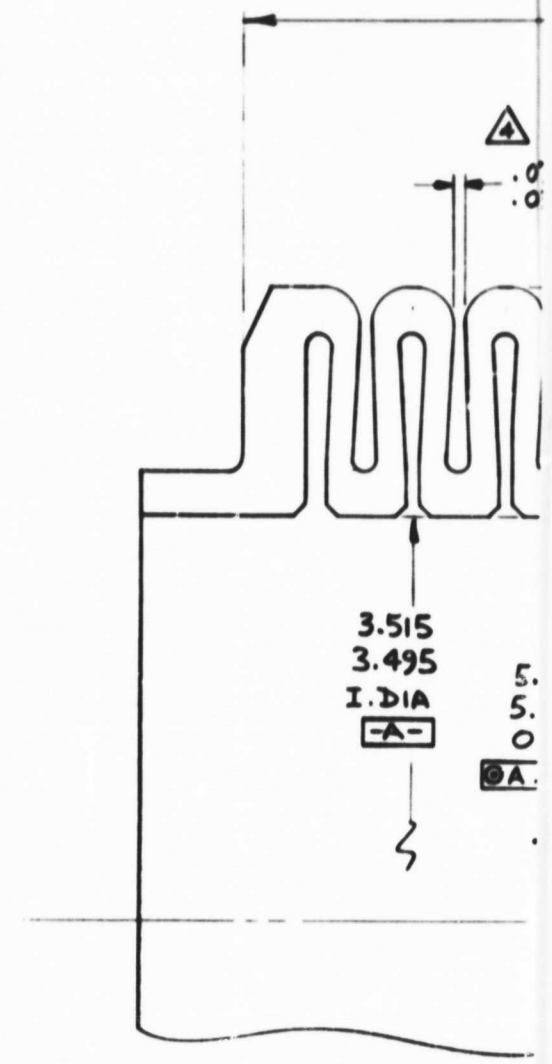
The Welding Research Section of Solar's Research Laboratories performed a thorough analysis of the electron beam welds required to join the ring quadrants. The weld configuration presented is basically similar to many other electron beam welds performed by Solar[1].

The following procedure will be followed in developing the electron beam welding technique. One full-size ring forging will be fabricated for use by the Welding Research Section in the development of weld schedules, evaluation of the shrinkage factors across the weld joint, and for metallurgical examination of the weld. This unit will also be used for a verification of the mechanical properties of the weld joint. The ring forging will be machined to the required dimensions of the actual gimbal ring in accordance with Solar drawing 41306 and aged. This forging will be sectioned as indicated in Figure 1 and will be scribed on the end faces as indicated in Figure 2. These scribe lines will be used as references for determining weld shrinkage. Weld strips will be provided on the inside of the ring to prevent excess penetration of the weld bead. These weld strips will be removed after welding. These weld strips will also protect the assembly from the expulsion material from the weld due to excess penetration. Weld strips at the faces of the ring will also be provided to prevent cratering at the start and stop of the weld joint. Tooling will be designed to hold the four segments of the gimbal ring in their proper location, and will be rugged enough

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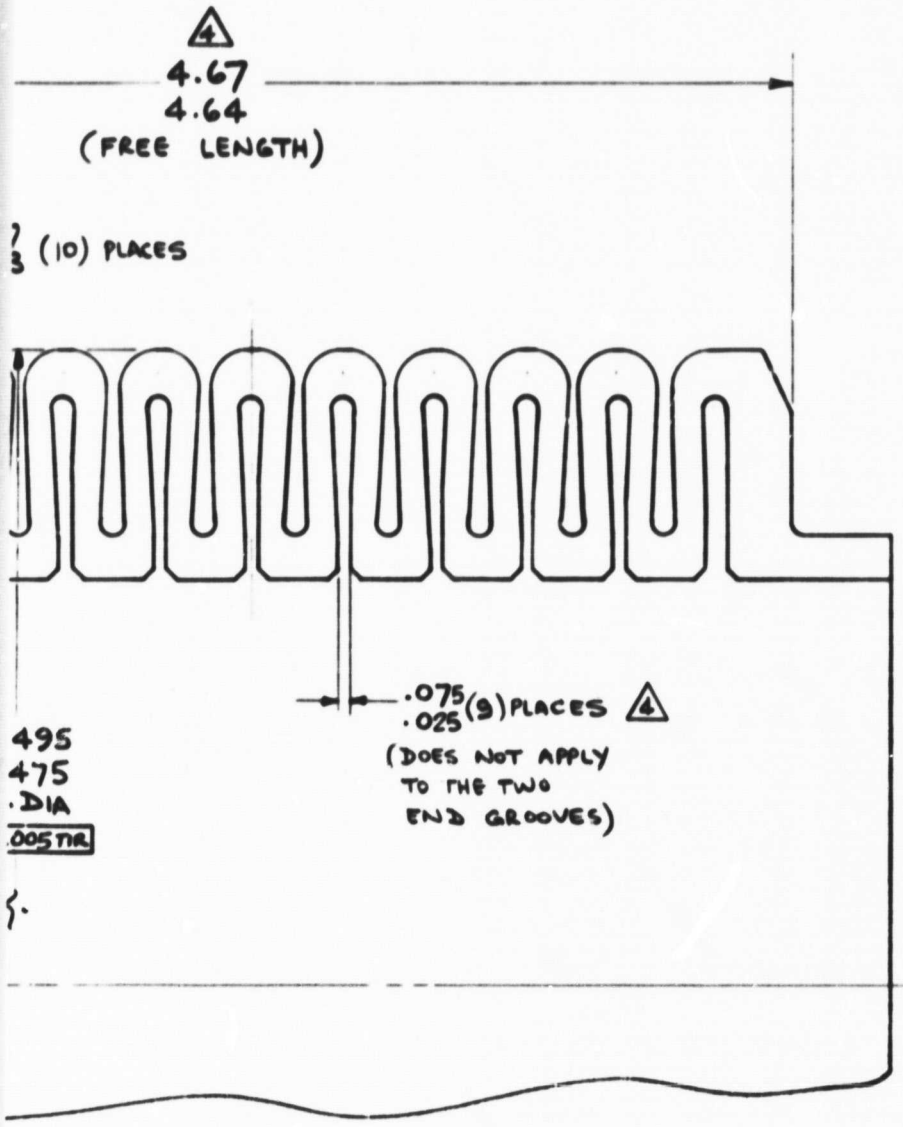
1. Solar's extensive prior experience in, and facilities for, electron beam welding are described in Solar Report R-6425, "Electron Beam Welding," submitted as an enclosure to this report.





NOTES: UNLESS OTHERWISE SPECIFIED

1. DIMENSIONS LOCATING THE TRUE POSITION ARE
2. DO NOT SCALE DRAWING. WORK TO DIMENSIONS
3. DRAWING INTERPRETATION PER MIL-D-70327 ~~AND SOLAR SPEC 8-1, SECT~~
4. BELLOWS TO BE COMPRESSED, AN PRESSURIZED TO 35,000PSI ±5(THE PRESSURE & THE COMPRESS REMOVED, THE FREE LENGTH A SPACING DIMENSIONS ARE ME
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   - OPERATING PRESSURE : 1(
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   - BURST PRESSURE : 4(
   - OPERATING MEDIA : G B
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BASIC
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65°F TO +160°F

QTY REQD	QTY REQD	PART NUMBER	DESCRIPTION	MATERIAL	SPECIFICATION	UNIT WT	
		-1			MAKE FROM 41309-1		

LIST OF MATERIAL (PARTS LIST) ASSY WT

APPROVED BY	
STAFF	
STAFF	
	R. Kelly 7.3.64
PROJ. ENG	
DESIGN	6/23/64
CHECKED	Comment 7/10/64
DRAWN	6-18-64

SOLAR
A Subsidiary of International Harvester Company

DRAWING TITLE

BELLOWS, COMPRESSION SET - ± 6°, 10,000 PSI

CODE IDENT NO.	SIZE	DRAWING NO.
55820	D	41310

SCALE TWICE SIZE SHEET 1 OF 1

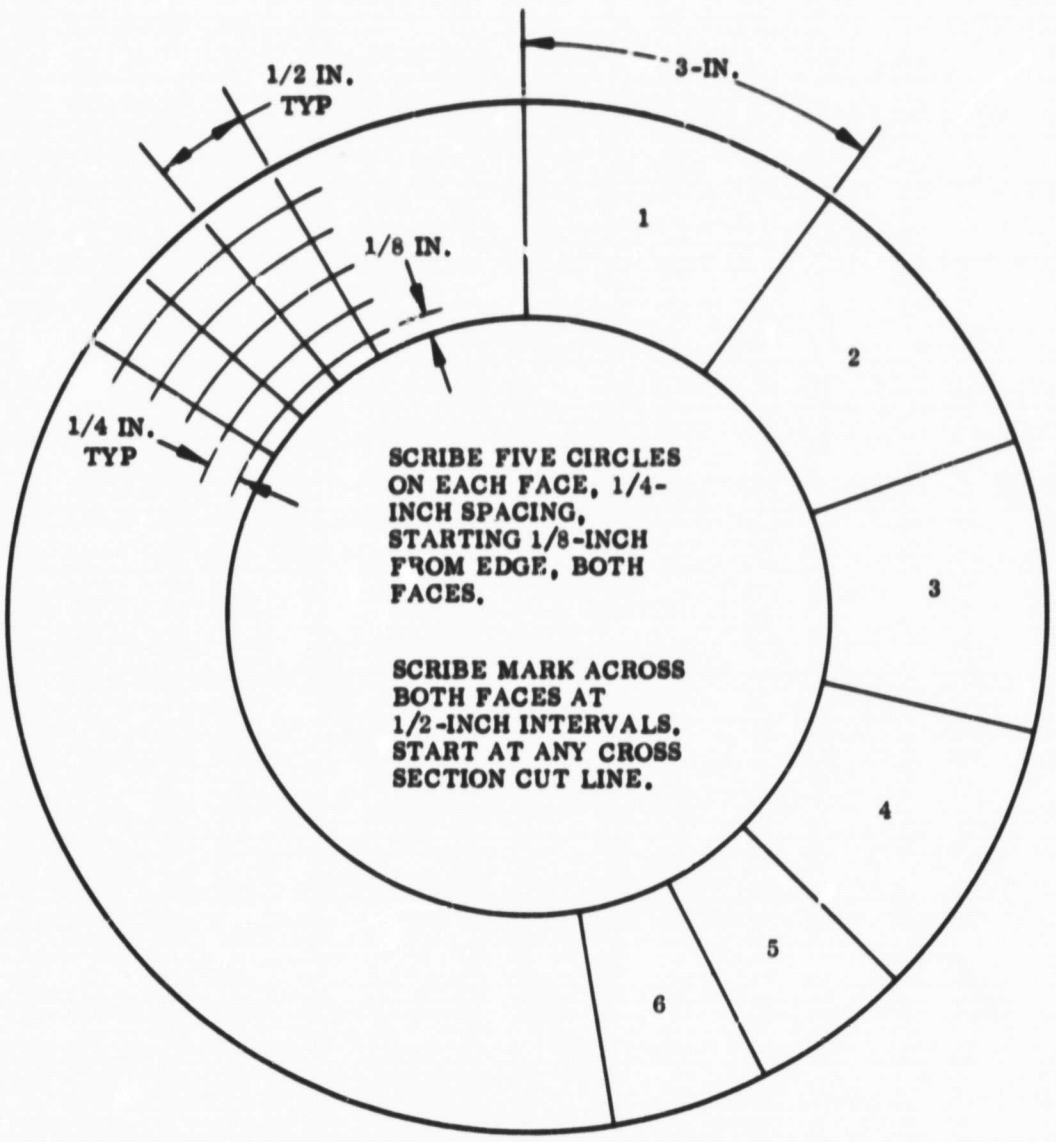


FIGURE 1. GIMBAL RING SECTIONING PLAN

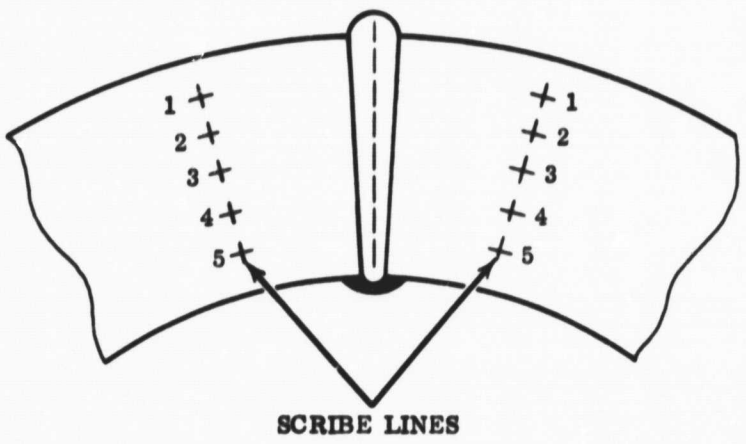


FIGURE 2. SCRIBING PATTERN FOR DETERMINING ELECTRON BEAM WELD SHRINKAGE OF GIMBAL RING SEGMENT

to prevent movement during the welding operation. This tooling will provide for the exposure of all four weld joints to the machine and will be designed to properly position and hold the weld strips. The tooling will be capable of remotely rotating the assembly from outside the electron beam vacuum chamber, thereby precluding the necessity of opening the chamber and breaking the vacuum for each weld.

4.2.2 General Welding Procedure

All detail parts to be welded will be cleaned, packaged, and sealed in polyethylene bags. Clean white gloves will be worn during the handling of the detail parts to be welded. The gimbal ring segments will be placed in position on the basic gimbal assembly mounted in the tooling fixture. The gimbal assembly will be provided with its own tooling in order to maintain the axial orientation of the gimbal flanges and a slight bit of precompression of the bellows. The segments will be secured in place by a holding fixture which will leave each of the four weld joints exposed. The weld strips will be installed at the ends and beneath each weld joint. The welding fixture will then be secured to the traverse in the vacuum chamber after lining up the weld joints with a viewing scope. The parameters of the established welding schedule will be checked out in a setup block mounted conveniently in the chamber at the same height as the weld joints.

All segments will be tack welded together and all joints secured with comparatively shallow depth welds. At this point, the chamber will be opened, the gimbal ring checked for concentricity, and the holding fixture checked to be sure that all segments are properly restrained. The chamber will again be evacuated and the full penetration welds completed using the established welding procedure.

4.3 QUALITY CONTROL

Solar recommends that X-ray procedures in accordance with Specification MIL-I-6865B and MIL-R-11468 (ORD) for determining the quality of welds be applied only to the fusion buttwelds. It has been Solar's experience that X-rays of fillet type welds in heavy structures are subject to misinterpretation. Solar, therefore, proposes that by the utilization of process controls and other inspection methods, justified by the establishment of adequate safety factors in the original design, these weld joints can be repeatably made with the desired level of reliability.

In the weld joining the bellows to the gimbal flange, X-ray is not considered necessary since this weld is primarily a seal weld. This weld will be inspected by the dye penetrant method in accordance with Specification MIL-I-6866, Type II, and by helium mass spectrometer test.

Since the margin of safety on the electron beam weld of the gimbal ring segments is extremely high, this weld will not be X-rayed. Also, owing to the fact that film cannot be inserted directly beneath the weld, but must be inserted directly inside the gimbal joint, the integrity of the interpretation would be greatly reduced due to all the gimbal structure and bellows between the weld and the film. The nature of electron beam welding is strict process control since the weld is performed automatically and in a vacuum.

4.4 STRESS ANALYSIS

A stress analysis on the complete flexible connector system is contained in this report as Appendix B. The gimbal joint bellows was subjected to a computer analysis utilizing Solar program 204 on an IBM 7090 computer and the results are contained in the stress analysis. This stress analysis verified the design approaches in the original concept. The stress analysis also presents those forces required to physically move the flexible connector system through the envelope required. At the final presentation in Huntsville on 7 July, it was determined that Solar should not consider methods for augmenting the forces required to deflect the flexible connector system through the envelope required as that mechanism will be provided by NASA as an integral part of the support tower. These loads are being provided as additional information to aid NASA in the design of the movement system.

4.5 RELIABILITY

A reliability analysis has been prepared and is contained in this report as Appendix C. This analysis presents a predicted reliability level since no reliability goal was presented in the original scope of work. A primary goal in the design of the flexible connector system was the development of high levels of inherent reliability. Each of the manufacturing methods described has been demonstrated in the past to have a high level of reliability. While the scope of work stated that Solar will do no testing on the flexible connector system fabricated during the Phase II program, Solar intends to hydrostatically proof pressure test each of the components prior to delivery

to NASA. This in-process procedure will guarantee the structural integrity of each of the members and assure the delivery of hardware which will meet the basic requirements of the system. It will also verify the design concepts employed.

V. PRESSURE BALANCE COMPENSATORS

Preliminary design concepts for pressure balance compensators to operate at 10,000 psig were prepared during the study phase (Phase IA) of the program. These design concepts were prepared with the criteria that the unit must:

- operate at 10,000 psig,
- be easily cleaned and provide a minimum of dirt entrapment areas,
- lend itself to heavy-wall construction.

This design concept was presented to NASA at the mid-term presentation on 5 May. The compensators as presented could replace the expansion loops that would be designed into the 10,000 psig piping runs on the tower. At the direction of NASA in a letter of 25 May, Solar proceeded with developing designs for pressure balanced expansion compensators for the various line sizes on the present launch umbilical tower vertical runs. Analyses were conducted to determine the growth and shrinkage values of the piping runs when subjected to a temperature changing environment of -65 to +165 F. A comparison of the limits of deflection, for which the units would have to compensate, determined the number of compensators being recommended for each of the piping runs. For the lower-pressure systems, Solar proposes two compensators in each vertical run to handle the change in length as a result of the above temperature changes and the swaying of the tower.

A design for the 6,000 psi system was prepared which had sufficient capability to accept half the total change in length of that tower run. However, this unit was very heavy and very long and would not eliminate the necessity of providing flexibility to the tee off from the vertical run to the consoles mounted on each swing arm level. After discussions with Mr. Jack Kunkel in San Diego on 25 June, Solar proceeded to develop a smaller version of the 6,000 psi compensator which would accept a half inch of axial travel and be installed at each of the swing arm levels in the vertical runs. This unit

established, as the fixed point of the line with respect to thermal growth, that point at which the tee off to the console occurs. This, therefore, would preclude the necessity of utilizing any flexible members in the console run, and would make the overall system more reliable and less costly.

The compensator design proposed was acceptable to NASA except for the usage of Inconel 718 material throughout. As a result Solar is proposing, at the direction of NASA, a compensator with a CRES 316L structure and utilizing the original Inconel 718 bellows. It is recommended that Inconel 718, in the bellows, permits compactness of the design since the bellows are capable of accepting more deflection with a higher cycle life than bellows of a CRES 300 series material. This combination of materials is not new to Solar and is presently being used on various space flight vehicle programs.

5.1 PRINCIPLES OF OPERATION

The principle of the pressure balance compensator is one of utilizing pressure-induced forces, in balance, and constant volume. These devices for permitting axial deflections in duct or piping systems, without external restraint such as ground or vehicle structure, have been used for many years in most industries engaged in the transfer of fluids. The following is a brief description of the function of the compensator and a discussion of the existing technology to impart the feeling that this design is not extending the state of the art of this field.

The compensator is one of the only fluid carrying flexible joints which permits axial deflections without requiring an external restraining force equal to the piston load which is equal to the duct cross sectional area times the duct pressure. The compensator consists of three bellows and a crossover structure. This basic construction is shown in Figure 3. The compensator operates by creating a force equal and opposite to the piston load. This force is created in the compensating chamber by the duct pressure acting against an annular area equal to the duct cross sectional area. The ratio of the diameter of the compensating chamber to the duct diameter is the square root of two times the original duct diameter resulting in a compensating chamber area to duct area ratio of 2.

By the arrangement of the crossover structure and the three bellows, the system is basically three bellows in parallel and not in series. Therefore, if the compensator is being compressed due to an elongation of an adjacent duct, the primary

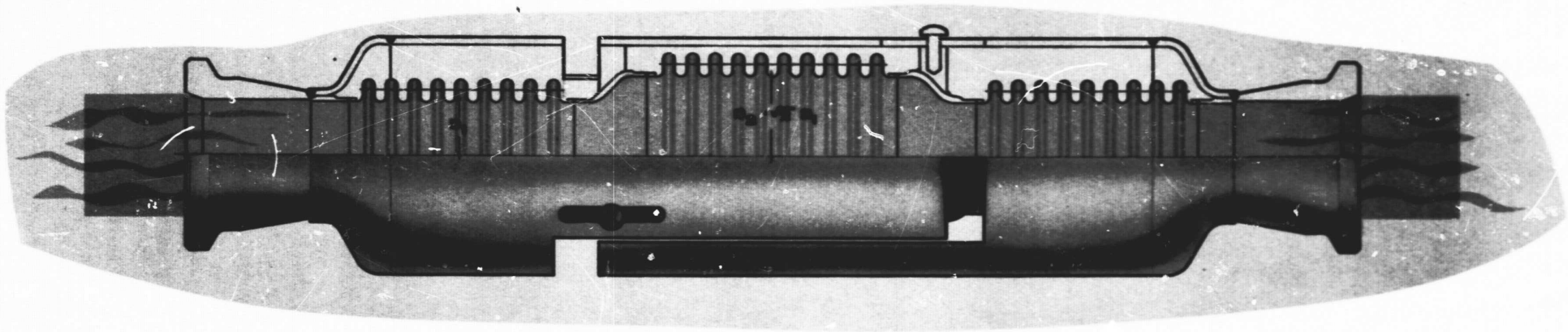


FIGURE 3. BASIC CONSTRUCTION OF COMPENSATOR

or smaller bellows on either side of the compensating or larger bellows compresses also. The compensating bellows elongates an equal amount. By geometry, the compensating bellows is expanding its volume at a rate equal to the decrease in volume of both primary bellows. Therefore, the system maintains a constant volume since an equal area is being maintained at all times. Since no volumetric change occurs during a change in length of the compensator, no change in pressure takes place and therefore no force is required other than that to overcome the bellows spring rate.

This theory is identical in all compensators. The compensator being presented utilizes this theory, however, the arrangement of the bellows and the crossover structure is the single feature which guarantees the added advantages of being cleanable and having a relatively slim overall shape. For example, the outside diameter of the compensator is consistently smaller than the envelope of the adjacent flanges. The small diameter precludes the necessity for re-routing lines to accommodate the compensator and permits pipe bundling with only the flanges as the size limiting components of the duct.

5.2 DESIGN

The compensator would be mounted in the vertical tower with the lowest side of the unit mounted to the structure and the upper portion free to deflect. The upper portion would accept the changes in length of the verticle run suspended above it. The lower portion of the compensator, being the fixed tie, would suspend and support the weight of the line hanging below the compensator. The tower structure would feel only the weight of this line and the compensator.

The individual bellows in the compensator have been designed for a minimum of 1500 cycles of maximum deflection while at operating pressure. Since the cycle life of a bellows is a function of the pressure induced stress and the bending stress due to deflection, calculations have shown that the day-to-day non-pressurized deflections due to the ambient temperature changes will not affect the cycle life of the bellows. The bellows have an infinite cycle life when being subjected to axial deflection only.

In order to reduce the number of different designs required by NASA for the five vertical pipe runs, Solar has been concentrating on designing three basic compensators with various size hubs to match those lines which have inside diameters

different than the compensators. Specifically, there will be a design for 6,000 psig to match the inside diameter of a 3-inch double extra strong pipe which will be used also for a 2-inch double extra strong 6,000 psig system. There will be a compensator to match a 4-inch schedule 40 pipe for 500 psig operating pressures and a compensator to match a 2-inch schedule 40 pipe for 125 psig service. This latter compensator will be used in the 50 psi system and will also be used in the 1-1/2-inch schedule 40 pipe system, by simply changing the size of the hubs. The advantages of multiple usage compensators in these various systems is in the areas of cost saving and reliability. The pressure drop increase as a result of using larger diameter compensators for smaller diameter lines is considered to be minimal since the velocities of flow in each of these systems are very low.

The basic arrangement of bellows and the crossover structure in these compensators has been proven by actual construction. Solar has participated in compensator designs and construction programs for North American S&ID on the Saturn S-II Battleship Stand. This compensator has met all the requirements for which it was designed.

APPENDIX A

DRAWINGS

CONCEPTUAL DESIGN DRAWINGS

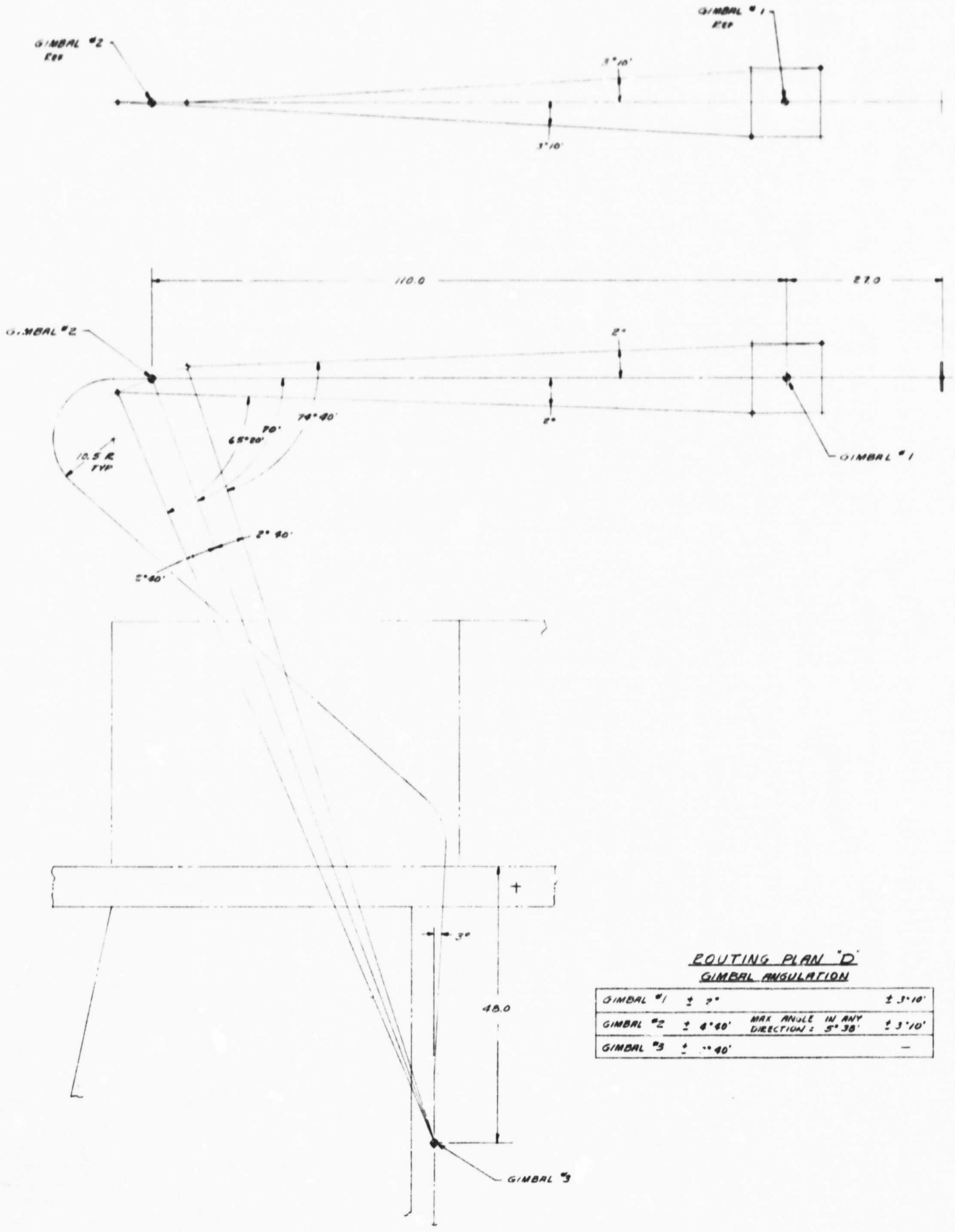
39701	Motion Study; High Pressure Line Configuration
39702	Gimbal; Formed Bellows with Machined Root Rings
39703	Gimbal; Machined Bellows with Integral Root Rings
39704	Gimbal; Machined Low Height Bellows
39705	Venturi Section
39706	Line Assembly; High Pressure Connector
39745	Gimbal; Machined Low Height Bellows (Alternate Design for 39704)
39746	Gimbal; Machined Low Height Bellows (Bolted Ring Design for 39704)

MANUFACTURING DRAWINGS

41306	Gimbal Ring Quadrant, ± 6°, 10,000 psi (Matched Set)
41307	Gimbal Flange; ± 6°, 10,000 psi (two sheets)
41308	Gimbal Assembly, ± 6°, 10,000 psi
41309	Machined Bellows, ± 6°, 10,000 psi
41310	Compression Set Bellows, ± 6°, 10,000 psi
41311	Assembly of Flanged Pipe, 10,000 psi
41312	Bent Flanged Pipe Assembly, 10,000 psi
41313	Pipe Union, 3.50 ID
41314	Connector Line Assembly, 10,000 psi
41389	Motion Study; High Pressure Line Configuration

COMPENSATOR PRELIMINARY DESIGN DRAWINGS

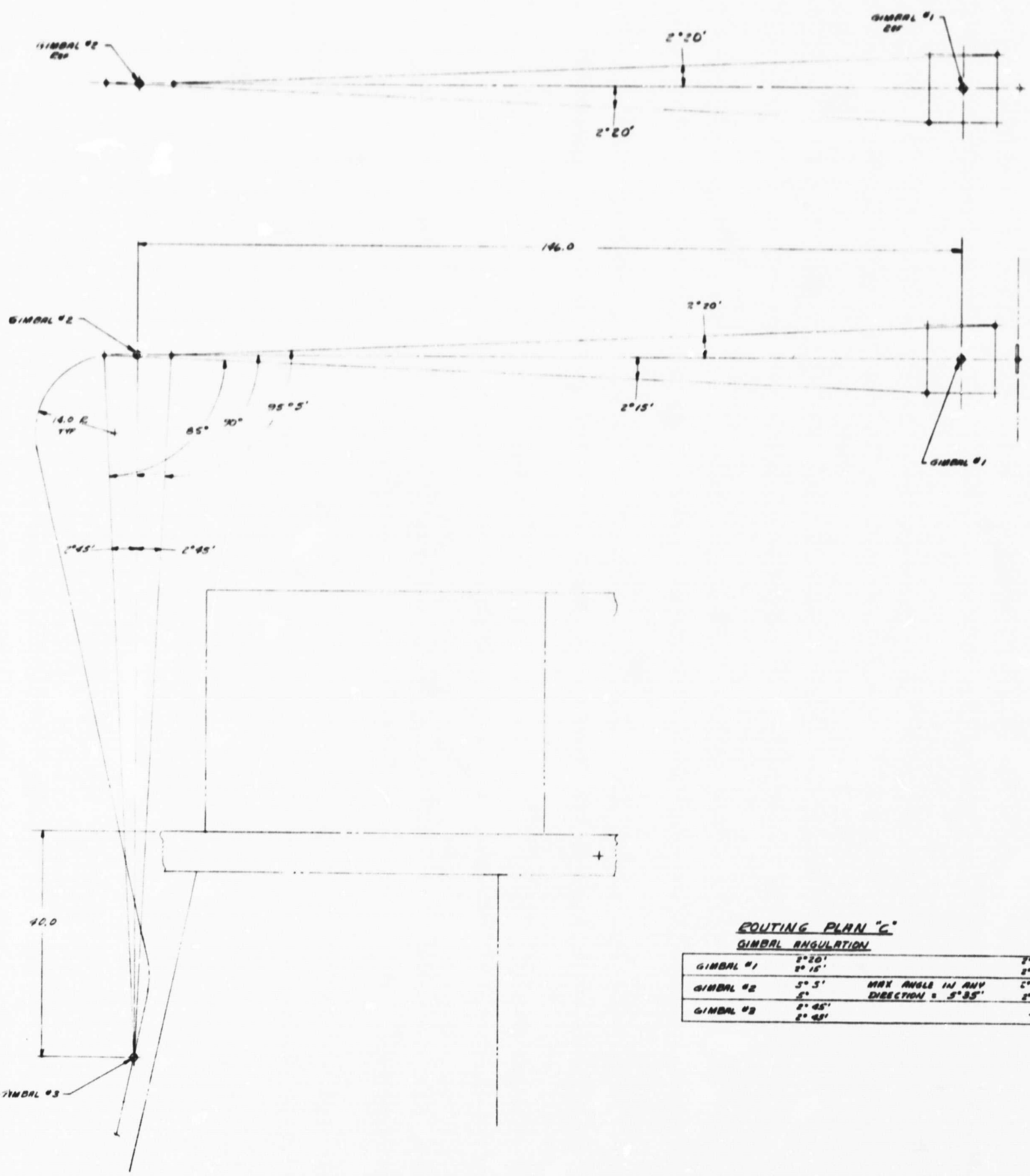
41135	Compensator, 4.00 Dia
41136	Compensator, 2.00 Dia
41137	Compensator, 2.00 Dia for 1.50 Dia Line
41195	Compensator for 3.50 Dia Line
41333	Compensator, 6000 psi, 2.30 Dia Line
41658	Compensator, 6000 psi, 2.30 Dia Line (Alternate to 41133)



ROUTING PLAN "D"
GIMBAL ANGULATION

GIMBAL #1	± 7°		± 3°10'
GIMBAL #2	± 6°40'	MAX ANGLE IN ANY DIRECTION = 5°38'	± 3°10'
GIMBAL #3	± 2°40'		—

Solar Drawing No. 39701 (Sheet 1 of 4)

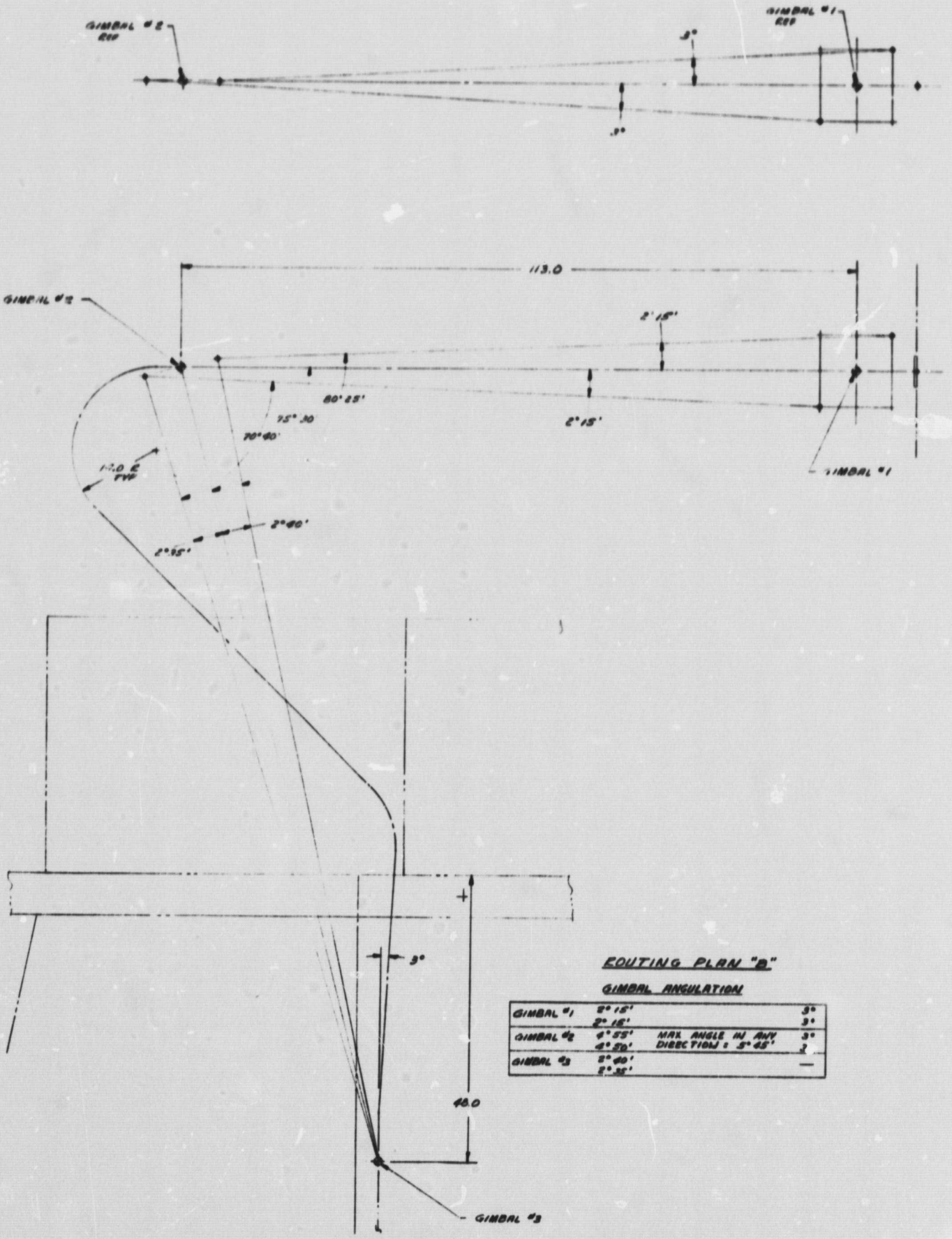


ROUTING PLAN "C"

GIMBAL ANGULATION

GIMBAL #1	2°20' 2°15'		2°20' 2°20'
GIMBAL #2	5°5' 5°	MAX ANGLE IN ANY DIRECTION = 5°35'	2°20' 2°20'
GIMBAL #3	2°45' 2°45'		—

Solar Drawing No. 39701 (Sheet 2 of 4)



ROUTING PLAN "B"

GIMBAL ANGULATION

GIMBAL #1	2° 15'		3°
	2° 15'		3°
GIMBAL #2	4° 55'	MAX ANGLE IN ANY DIRECTION = 5° 45'	3°
	4° 50'		3
GIMBAL #3	2° 40'		—
	2° 35'		

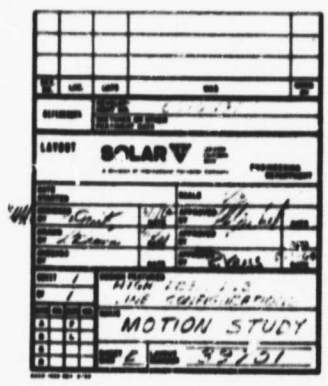
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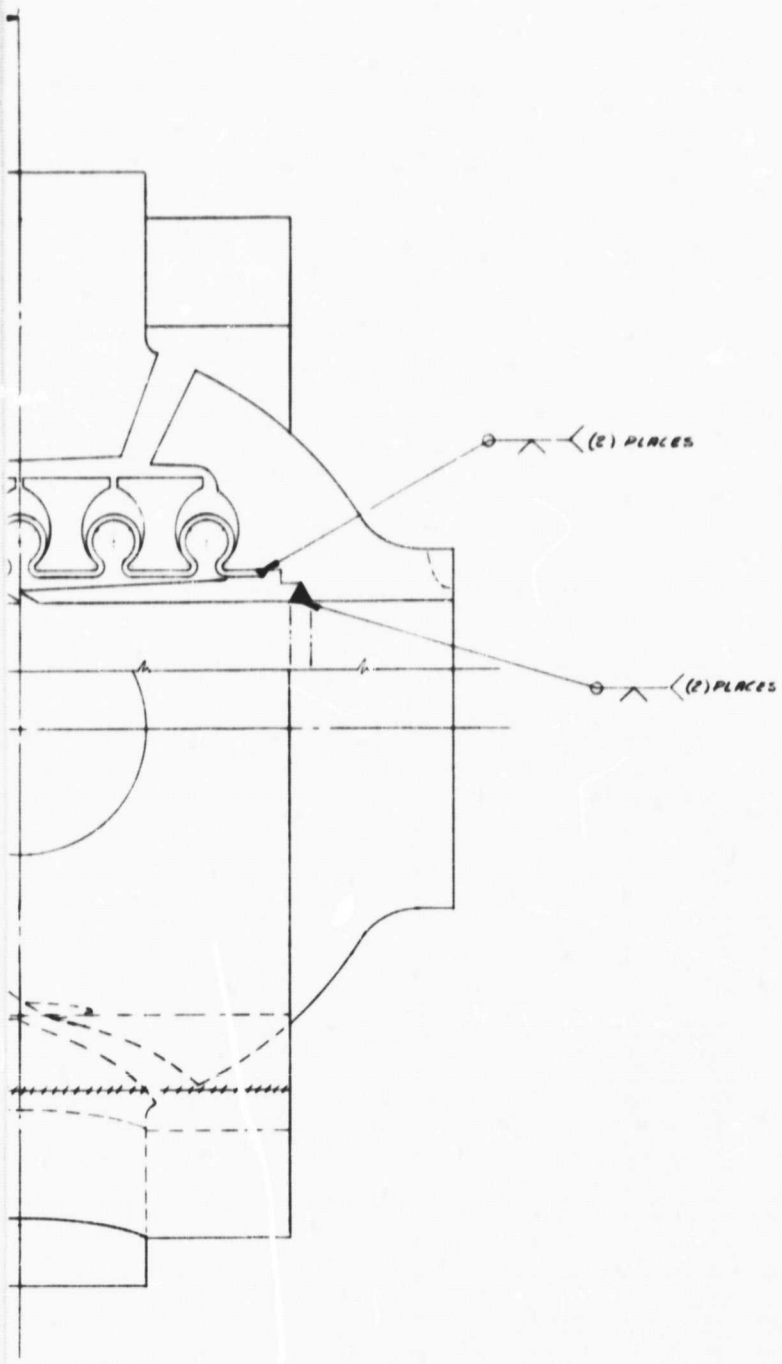
ROUTING PLAN "A"

GIMBAL ANGULATION

GIMBAL #1	6°15'		5°15'
	5°50'		5°10'
GIMBAL #2	10°55'	MAX ANGLE IN A IV	5°15'
	10°	DIRECTION : 12° 7'	5°10'
GIMBAL #3	5°40'		—
	6°10'		



Solar Drawing No. 39701 (Sheet 4 of 4)



NOTES: UNLESS OTHERWISE SPECIFIED

1. DIMENSIONS LOCATING THE TRUE POSITION ARE BASIC
2. DO NOT SCALE DRAWING. WORK TO DIMENSIONS GIVEN
3. DRAWING INTERPRETATION PER MIL-D-70327 AND SOLAR SPEC 9-4, SECT 6.0
4. ALL MATERIAL IS INCONEL 718, HEAT TREATED PER SOLAR SPEC 9-222. ALL WELDS SHOWN MAY REMAIN IN AS-WELDED CONDITION
5. DESIGN CONDITIONS:
   - OPERATING PRESSURE : 10,000 PSIG
   - PROOF PRESSURE : 15,000 PSIG
   - BURST PRESSURE : 40,000 PSIG
6. ANGULATION LIMITED TO APPROX ± 6° IN ANY PLANE THRU LONGITUDINAL AXIS
7. CALCULATED WEIGHT = 290 LBS

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LIST OF MATERIAL (PARTS LIST) ASSY WT.

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STAFF		
PROJ. ENG.		
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CHECKED		
DRAWN		4-27-64

SOLAR

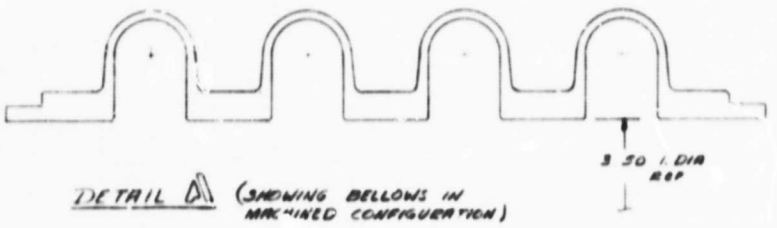
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GIMBAL, FORMED BELLOWS WITH MACHINED ROOT RINGS

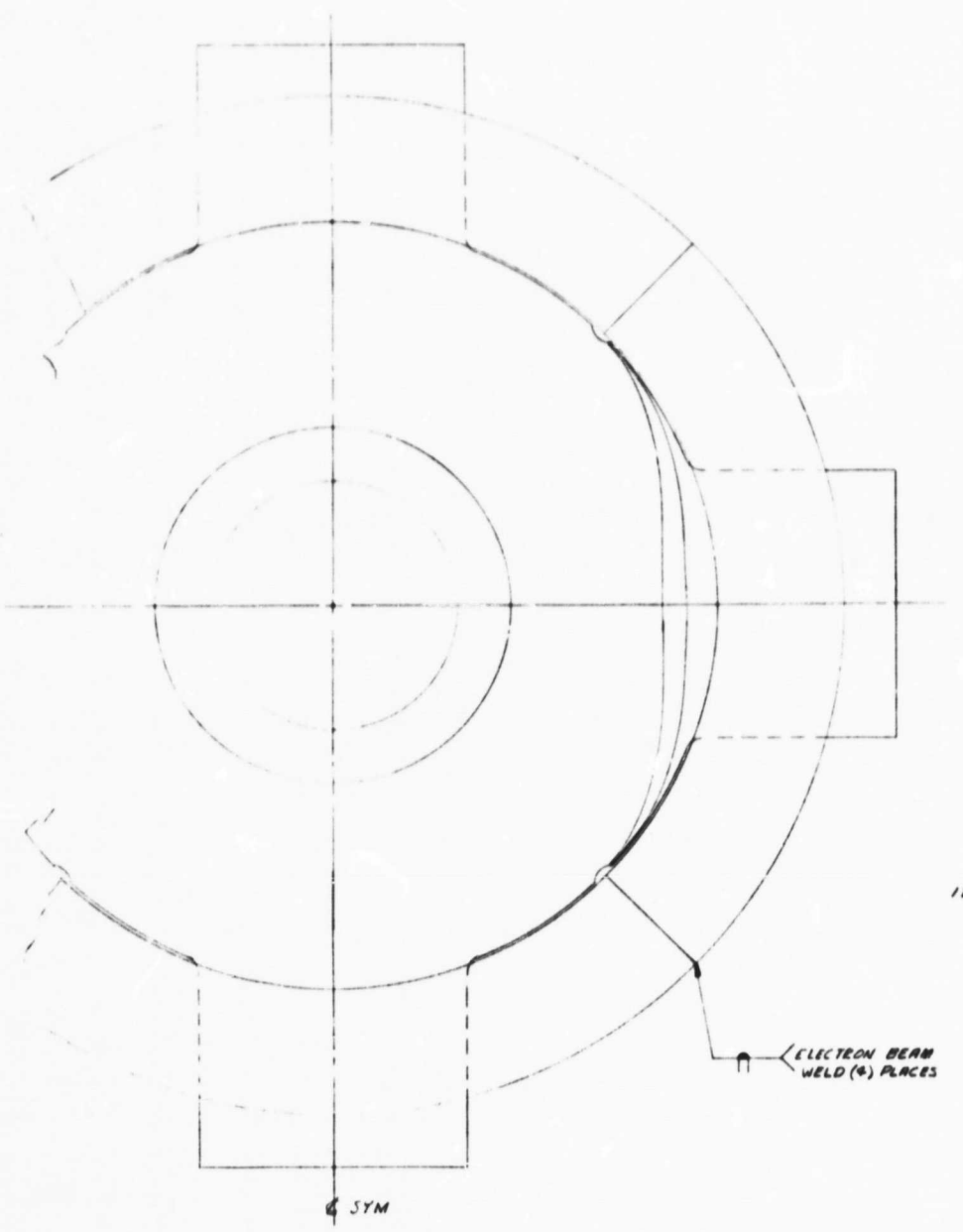
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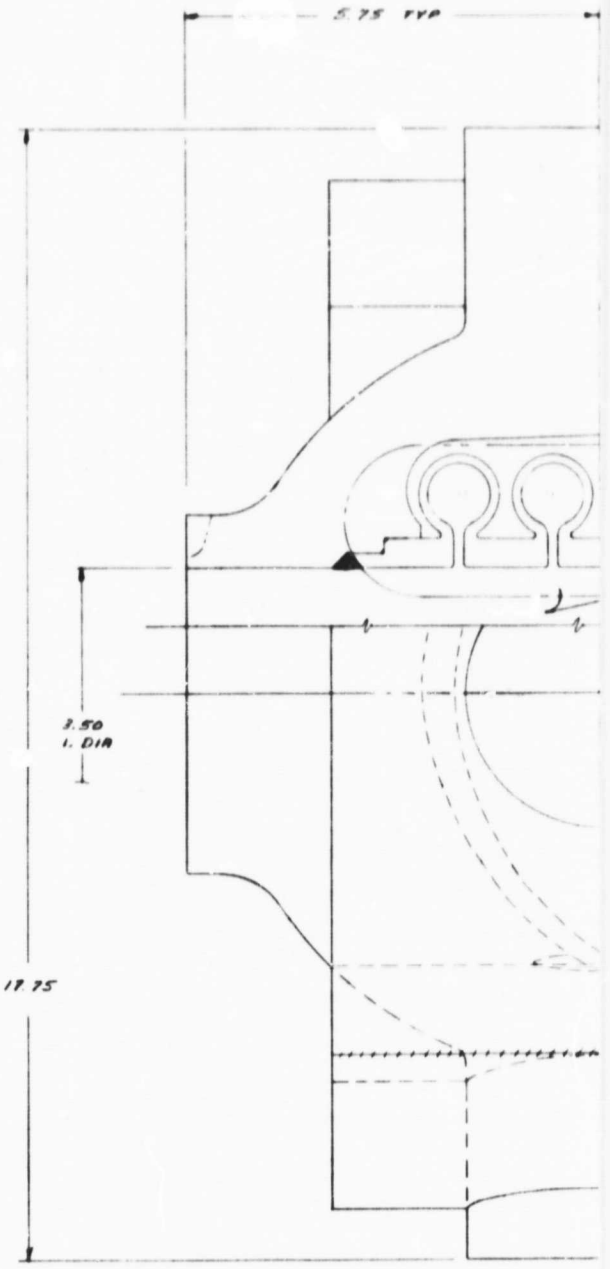
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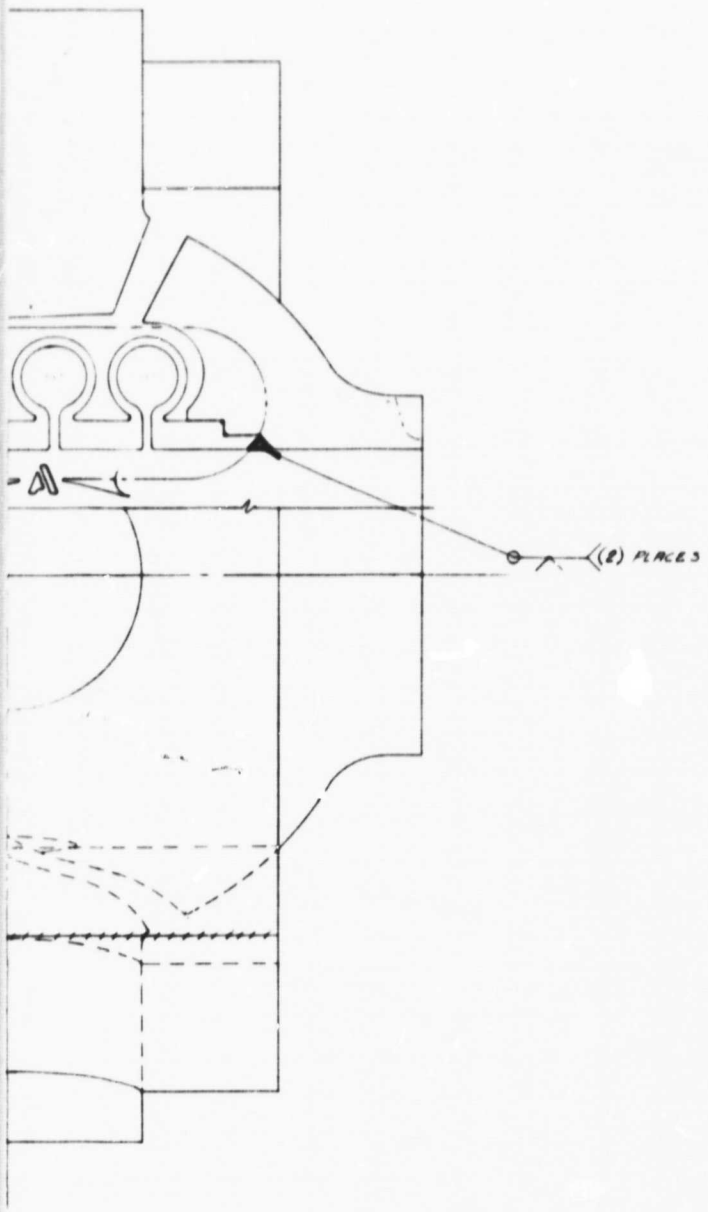
A-2-2



DETAIL A (SHOWING BELLOWS IN MACHINED CONFIGURATION)







NOTES: UNLESS OTHERWISE SPECIFIED

1. DIMENSIONS LOCATING THE TRUE POSITION ARE BASIC
2. DO NOT SCALE DRAWING. WORK TO DIMENSIONS GIVEN
3. DRAWING INTERPRETATION PER MIL-D-70327 AND SOLAR SPEC 9-4, SECT 6.0
4. ALL MATERIAL IS INCONEL 718, HEAT TREATED PER SOLAR SPEC 9-222 EXCEPT WELDS MAY REMAIN IN AS-WELDED CONDITION
5. DESIGN CONDITIONS:
   OPERATING PRESSURE : 10,000 PSIG
   PROOF PRESSURE : 15,000 PSIG
   BURST PRESSURE : 20,000 PSIG
6. ANGULATION LIMITED TO APPROX ± 6° IN ANY PLANE THRU LONGITUDINAL AXIS
7. CALCULATED WEIGHT = 291 LBS

QTY REQ'D	QTY REQ'D	PART NUMBER	DESCRIPTION	MATERIAL	SPECIFICATION	UNIT QTY

LIST OF MATERIAL (PARTS LIST)

APPROVED BY

STAFF

STAFF

PROJ ENG

DESIGN

CHECKED

DRAWN 4-11-69

SOLAR

DRAWING TITLE

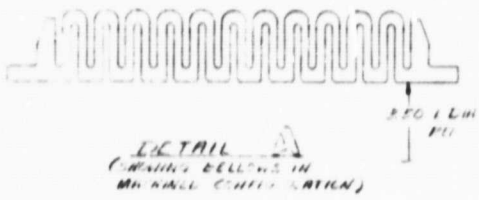
GIMBAL,
MACHINED BELLOWS WITH
INTEGRAL ROOT RINGS

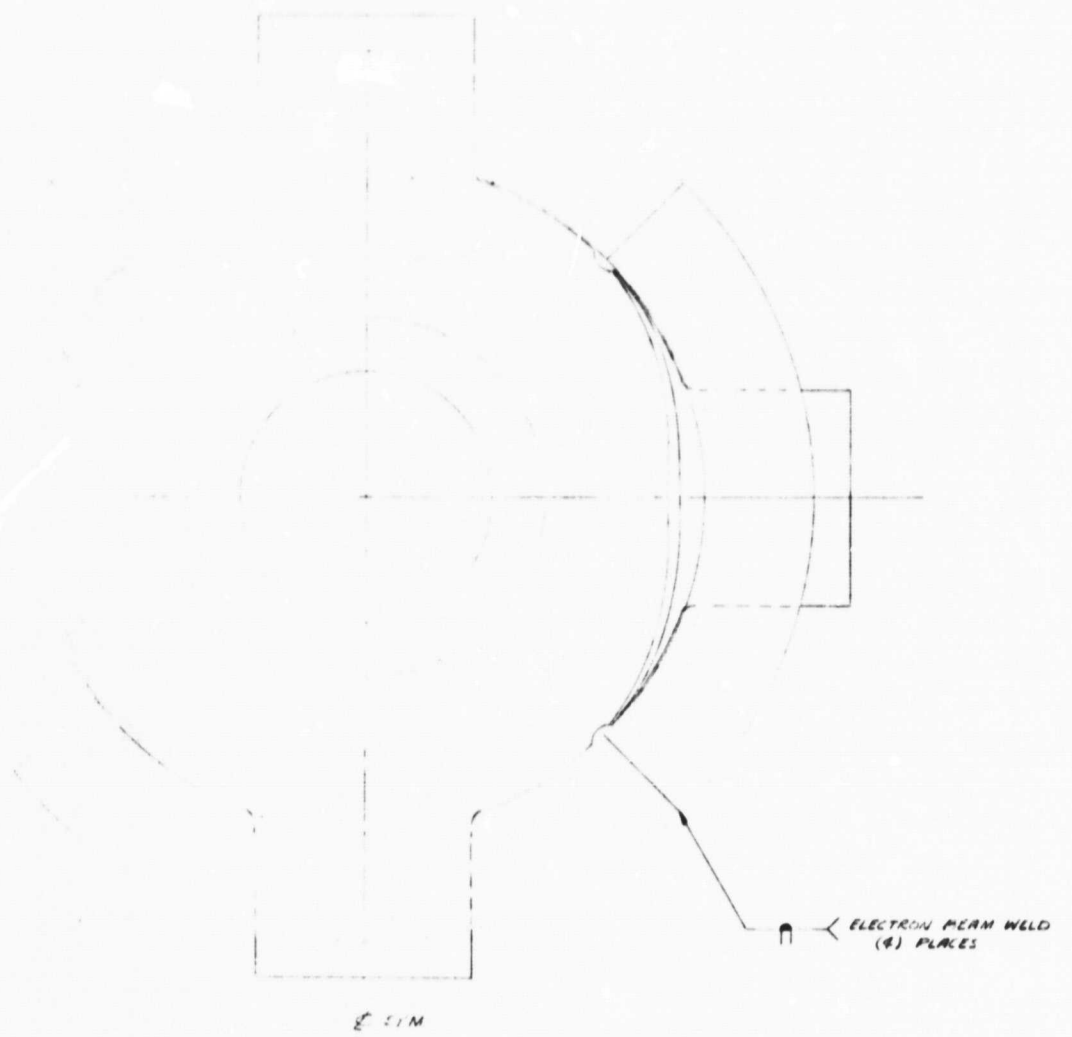
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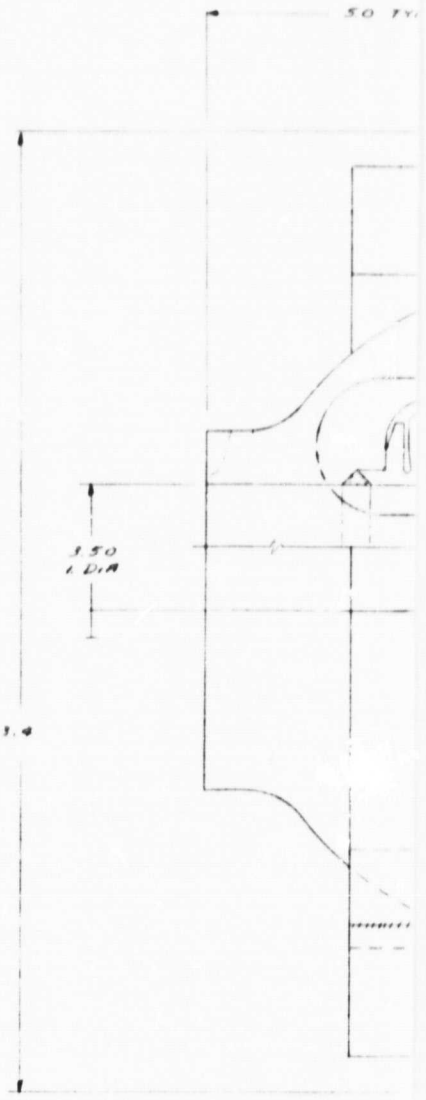
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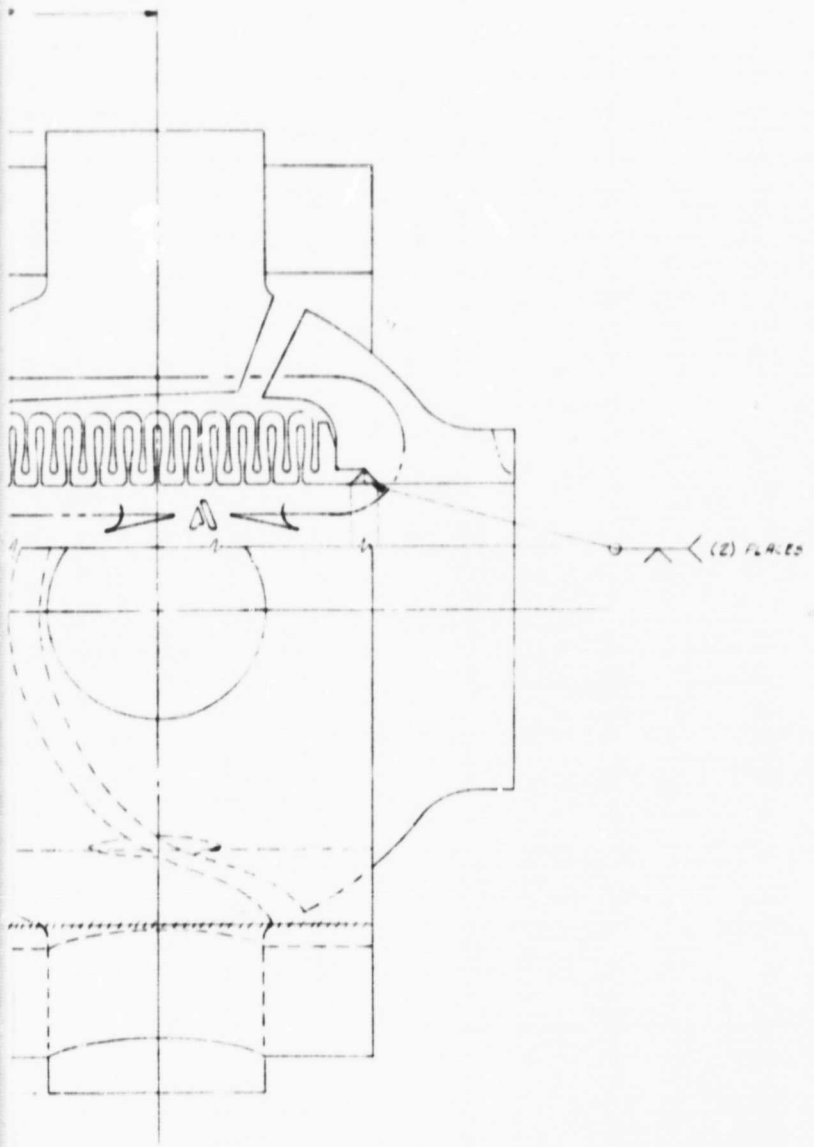
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PLY
DETAIL A
(SHOWING BELLOWS IN
MACHINED CONFIGURATION)


ELECTRON BEAM WELD
(4) PLACES
℄ SYM


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NOTES: UNLESS OTHERWISE SPECIFIED

1. DIMENSIONS LOCATING THE TRUE POSITION ARE BASIC
2. DO NOT SCALE DRAWING. WORK TO DIMENSIONS GIVEN
3. DRAWING INTERPRETATION PER MIL-D-70327 AND SOLAR SPEC 9-4, SECT 6.0
4. ALL MATERIAL IS INCONEL 718, HEAT TREATED PER SOLAR SPEC 9-222 EXCEPT WELDS MAY REMAIN IN AS-WELDED CONDITION
5. DESIGN CONDITIONS:
   - OPERATING PRESSURE : 10,000 PSIG
   - PROOF PRESSURE : 15,000 PSIG
   - BURST PRESSURE : 20,000 PSIG
6. ANGULATION LIMITED TO APPROX ± 6° IN ANY PLANE THRU LONGITUDINAL AXIS
7. CALCULATED WEIGHT = 174 LBS

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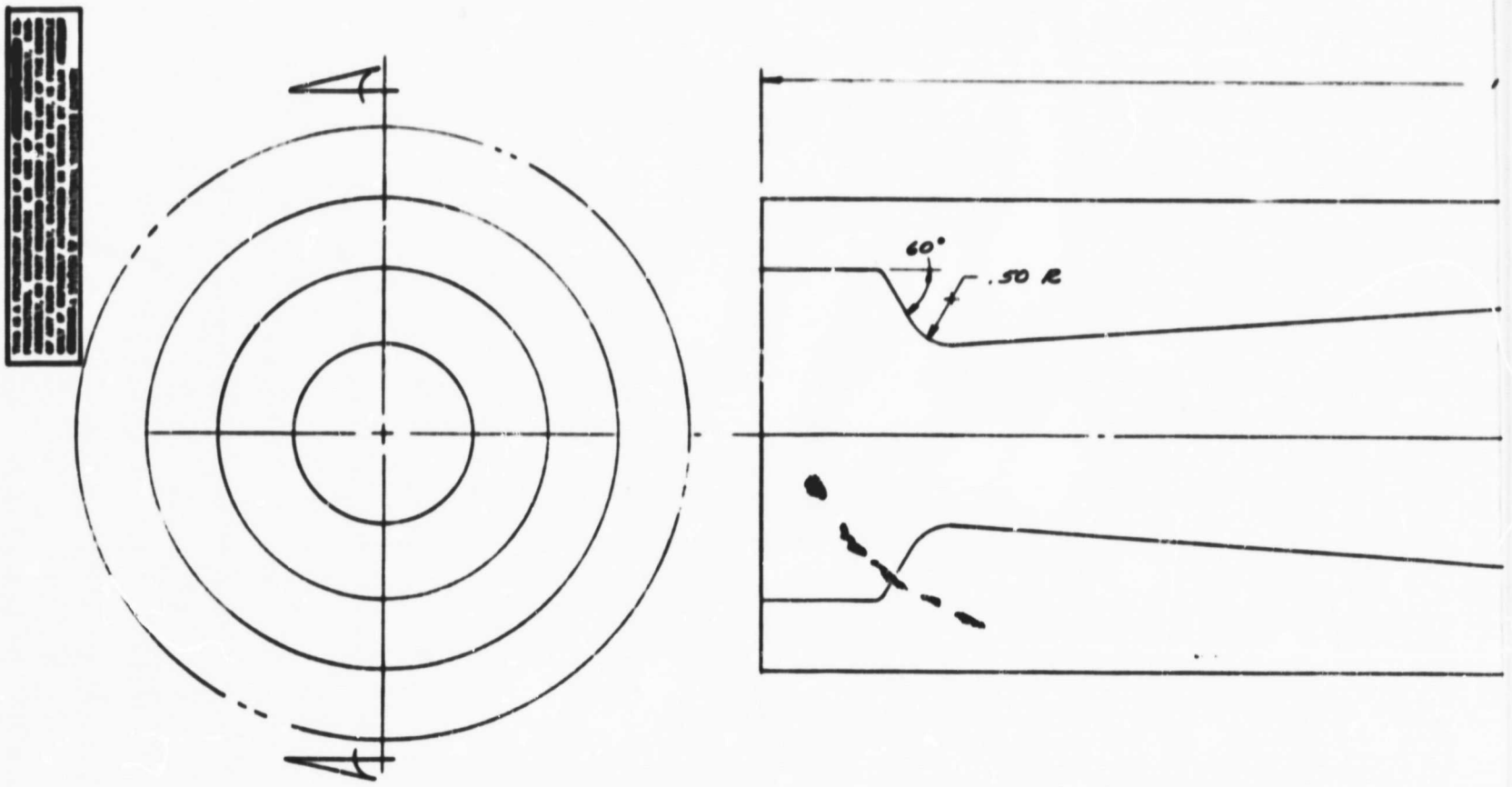
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SCALE FULL SHEET 1 OF

A-6-2



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PIPE 1. DIA = 3.50

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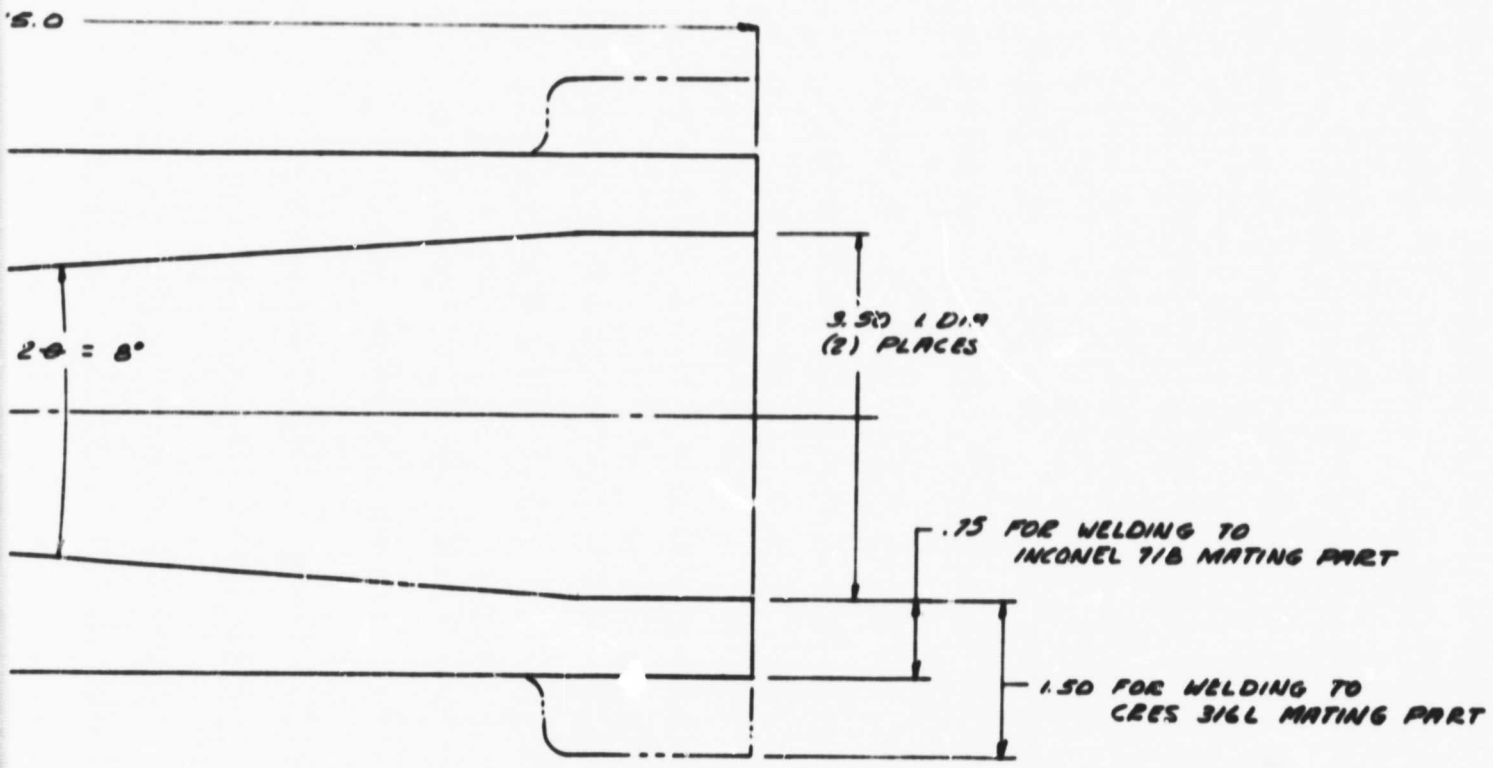
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2 DO NOT SCALE DRAWING.

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4 MATERIAL : S OR ALLOY INCONEL 71



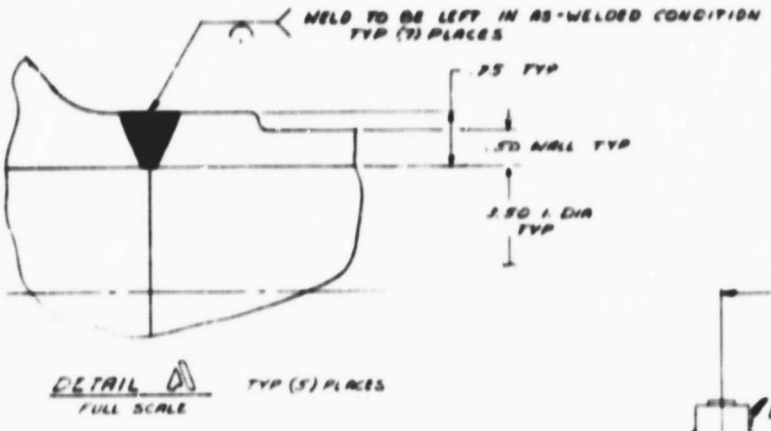
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SHAPE SHOWN IS FOR INCONEL 718
SIMILAR TO ANNEALED STRENGTH OF
8

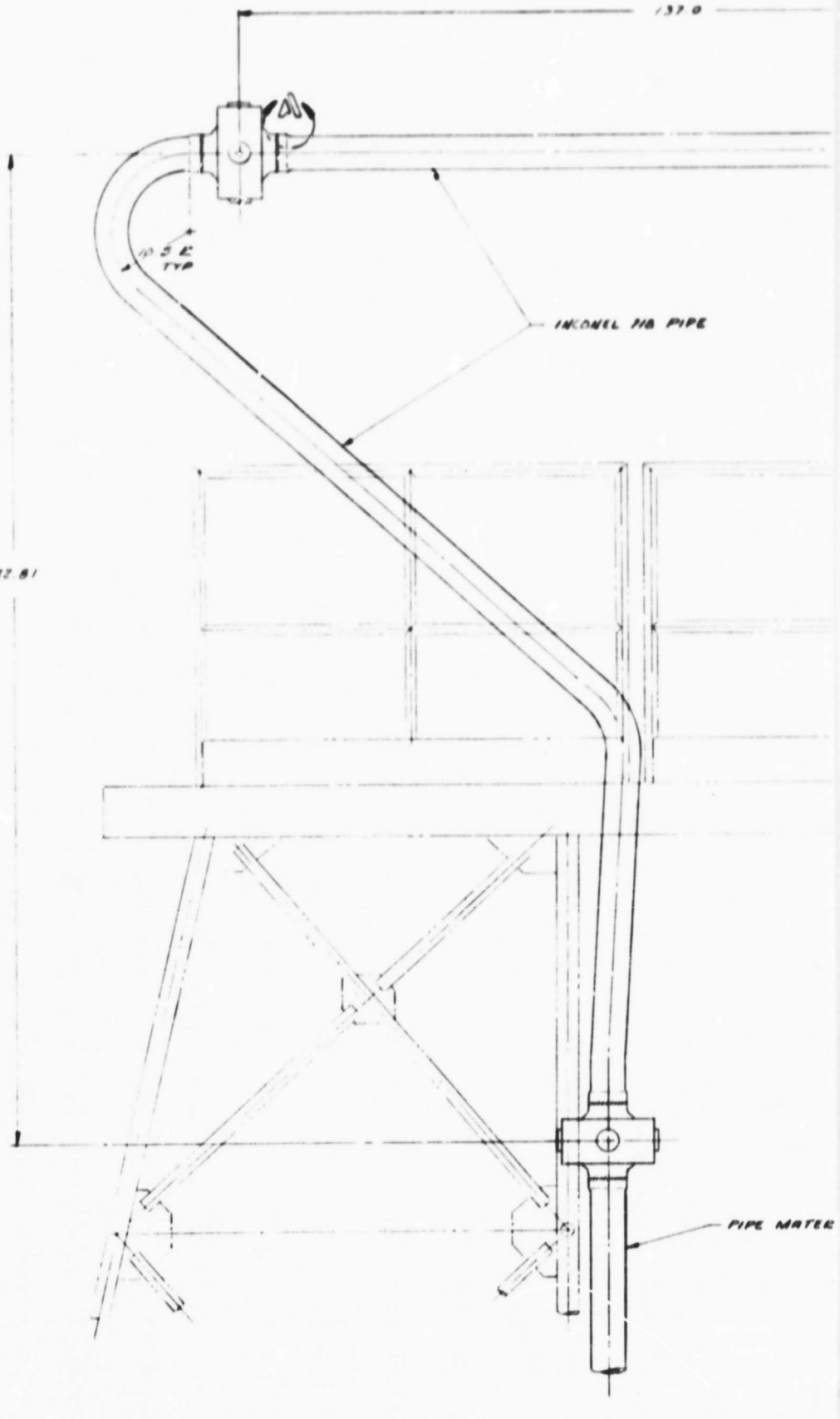
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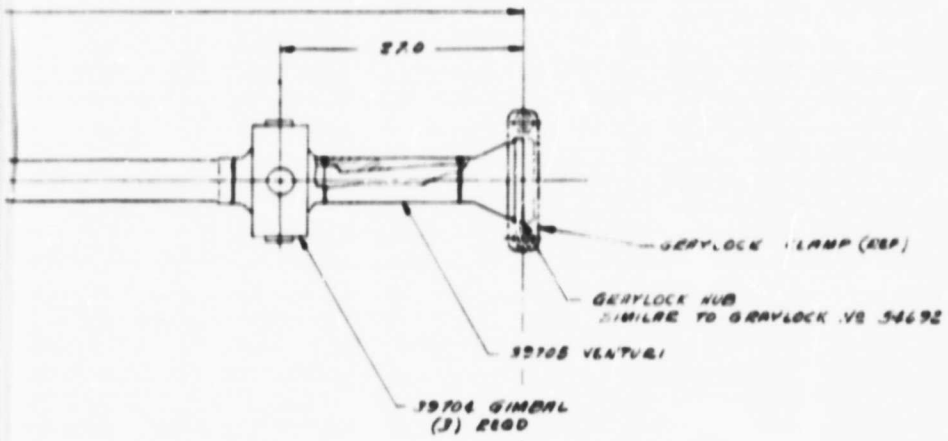
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	SCALE FULL · SHEET 1 OF 1

A-8-2

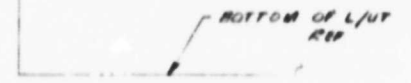


DETAIL Δ1
FULL SCALE
TYP (5) PLACES









6-1 ON L/UT
ROD

IAL TO BE DETERMINED LATER.

NOTES: UNLESS OTHERWISE SPECIFIED
1. DIMENSIONS LOCATING THE TRUE POSITION ARE BASIC
2. DO NOT SCALE DRAWING. WORK TO DIMENSIONS GIVEN
3. DRAWING INTERPRETATION PER MIL-D-70327 AND SOLAR SPEC 9-4, SECT

ITEM NO.	QTY. REQD.	PART NUMBER	DESCRIPTION	MATERIAL	SPECIFICATION	WEIGHT LB.

LIST OF MATERIAL (PARTS LIST)

APPROVED BY
STAFF
STAFF
PROJ. ENGR.
DESIGN
CHECKED
DRAWN

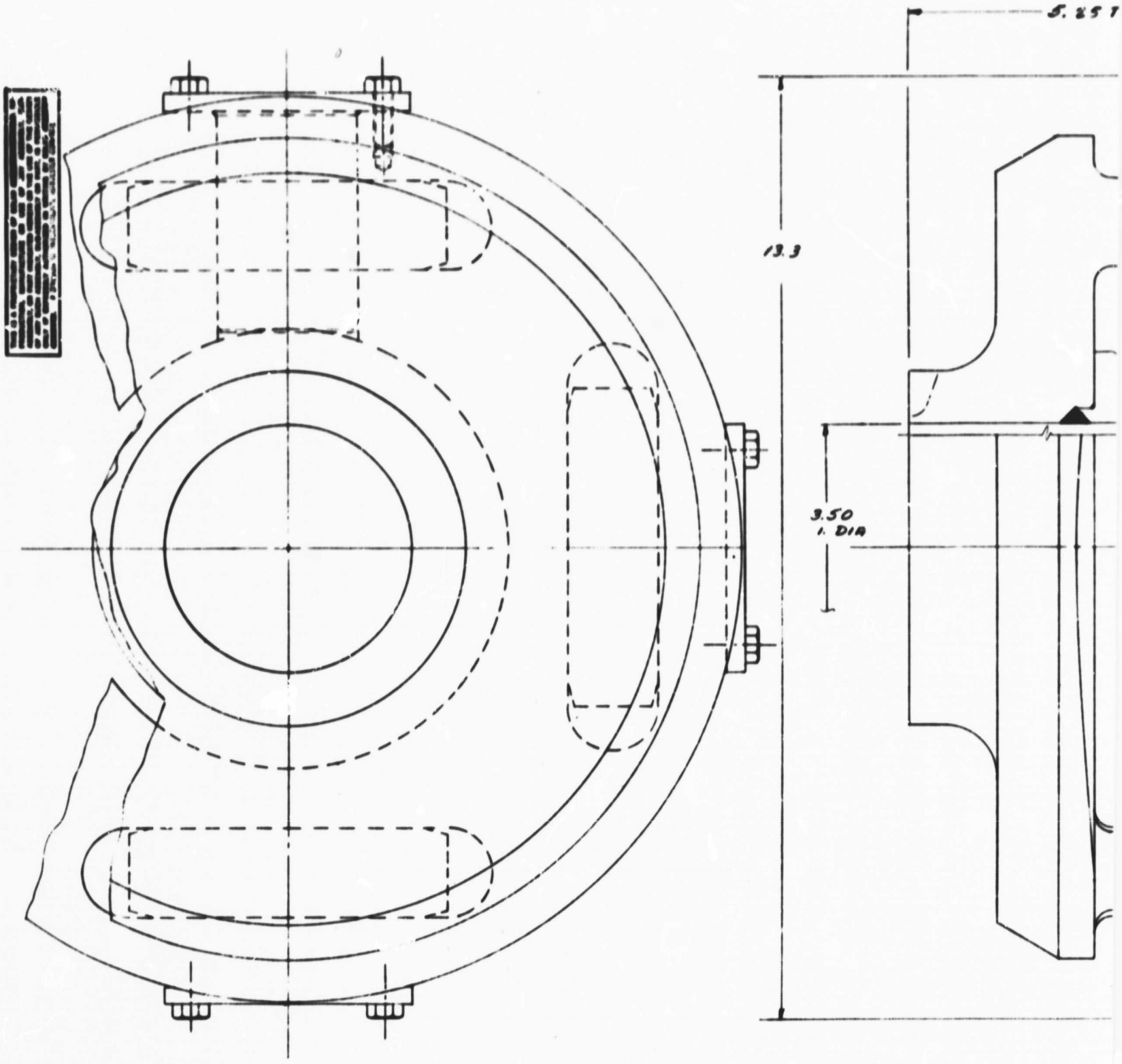
SOLAR
A DIVISION OF INTERNATIONAL HARVESTER COMPANY

DRAWING TITLE
LINE ASSY,
HIGH PRESSURE CONNECTOR

CODE IDENT NO. 55820 | SIZE E | DRAWING NO. 39706

SCALE 1/8 | SHEET 1 OF 1

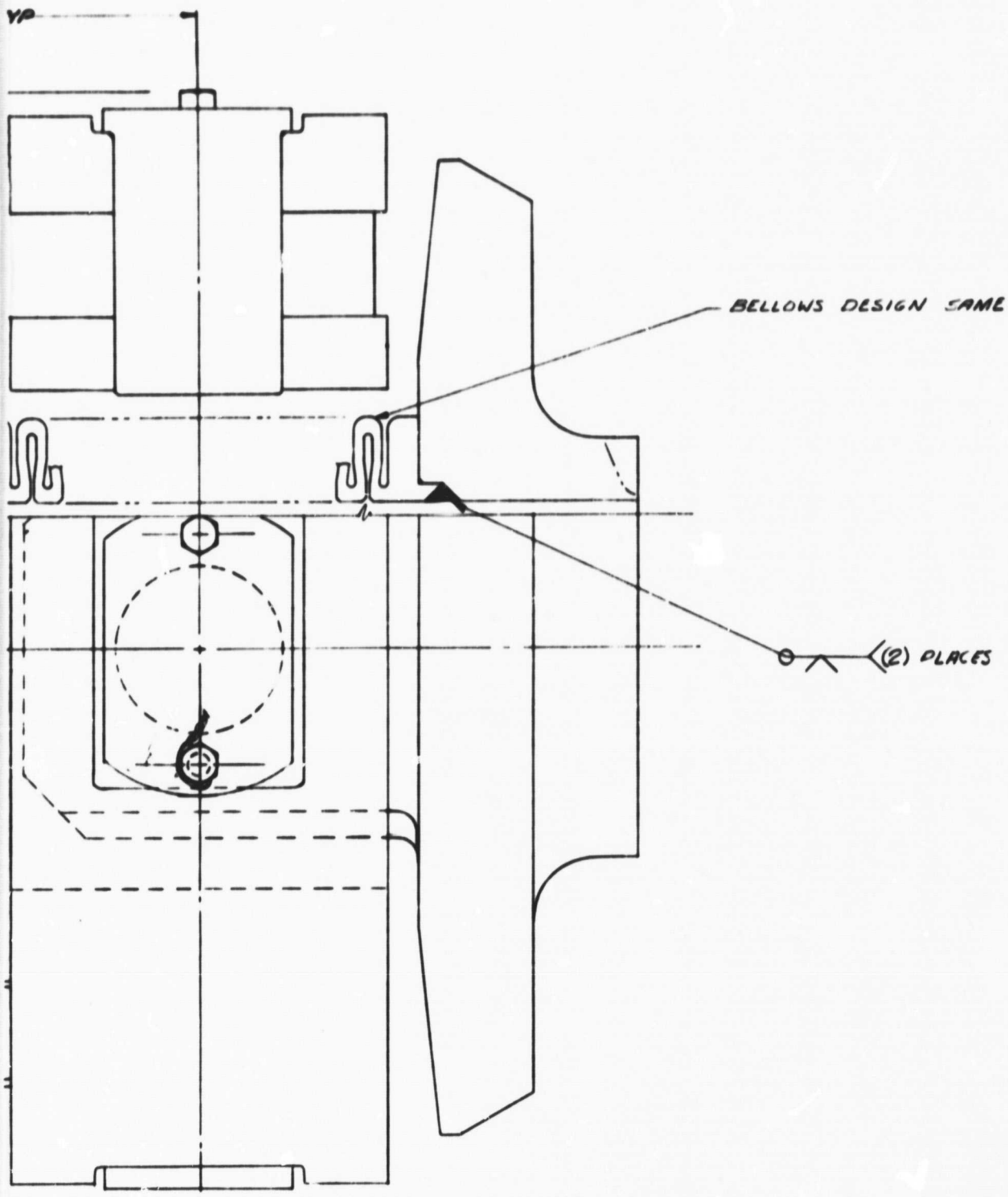
A-10 -2



NOTES: UNLESS OTHERWISE SPECIFIED

1. DIMENSIONS LOCATING THE TRUE POSITION ARE BASIC
2. DO NOT SCALE DRAWING. WORK TO DIMENSIONS GIVEN
3. DRAWING INTERPRETATION PER MIL-D-70327 AND SOLAR SPEC 9-4, SECT 6.0
4. ALL MATERIAL IS INCONEL 718, HEAT TREATED PE. SPEC 9-222 EXCEPT WELDS MAY REMAIN IN AS- CONDITION
5. DESIGN CONDITIONS:
   - OPERATING PRESSURE : 10,000 PSIG
   - PROOF PRESSURE : 15,000 PSIG
   - BURST PRESSURE : 40,000 PSIG
6. ANGULATION LIMITED TO APPROX ± 6° IN ANY PL. LONGITUDINAL AXIS
7. CALCULATED WEIGHT: 231 LBS

A-11-1



QTY REQD	QTY REQD	PART NUMBER	DESCRIPTION	MATERIAL	SPEC ICATION	UNIT WT	
		LIST OF MATERIAL (PARTS LIST)				ASSY WT	

APPROVED BY	
STAFF	
STAFF	
PROJ. ENG	Robt Davis 5-1-64
DESIGN	
CHECKED	
DRAWN	R Brown 4-30-64

SOLAR
A Division of International Harvester Company

DRAWING TITLE
GIMBAL,
MACHINED LOW HEIGHT BELLOWS,
ALTERNATE DESIGN FOR 39704

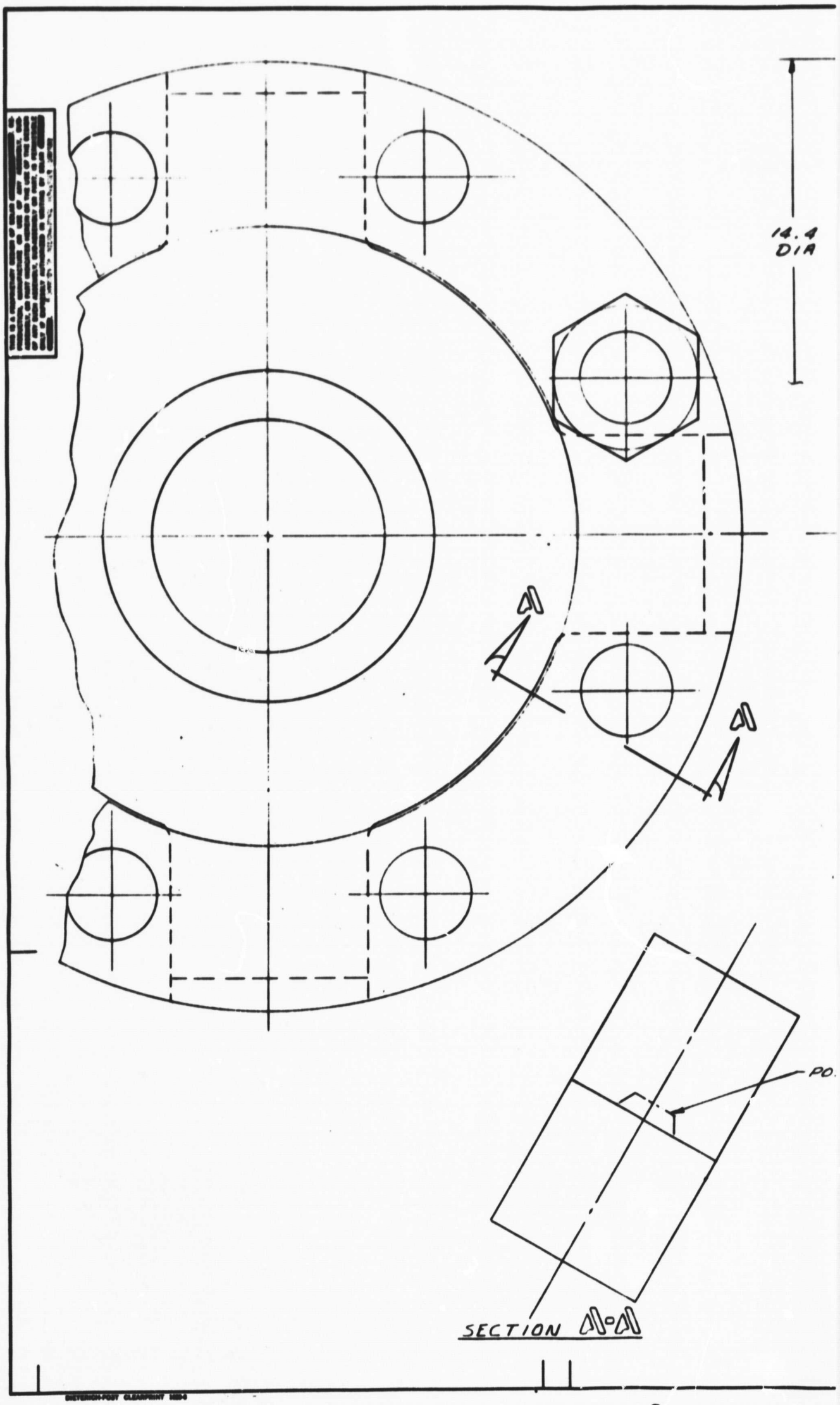
CODE IDENT NO.	SIZE	DRAWING NO.
55820	D	39745

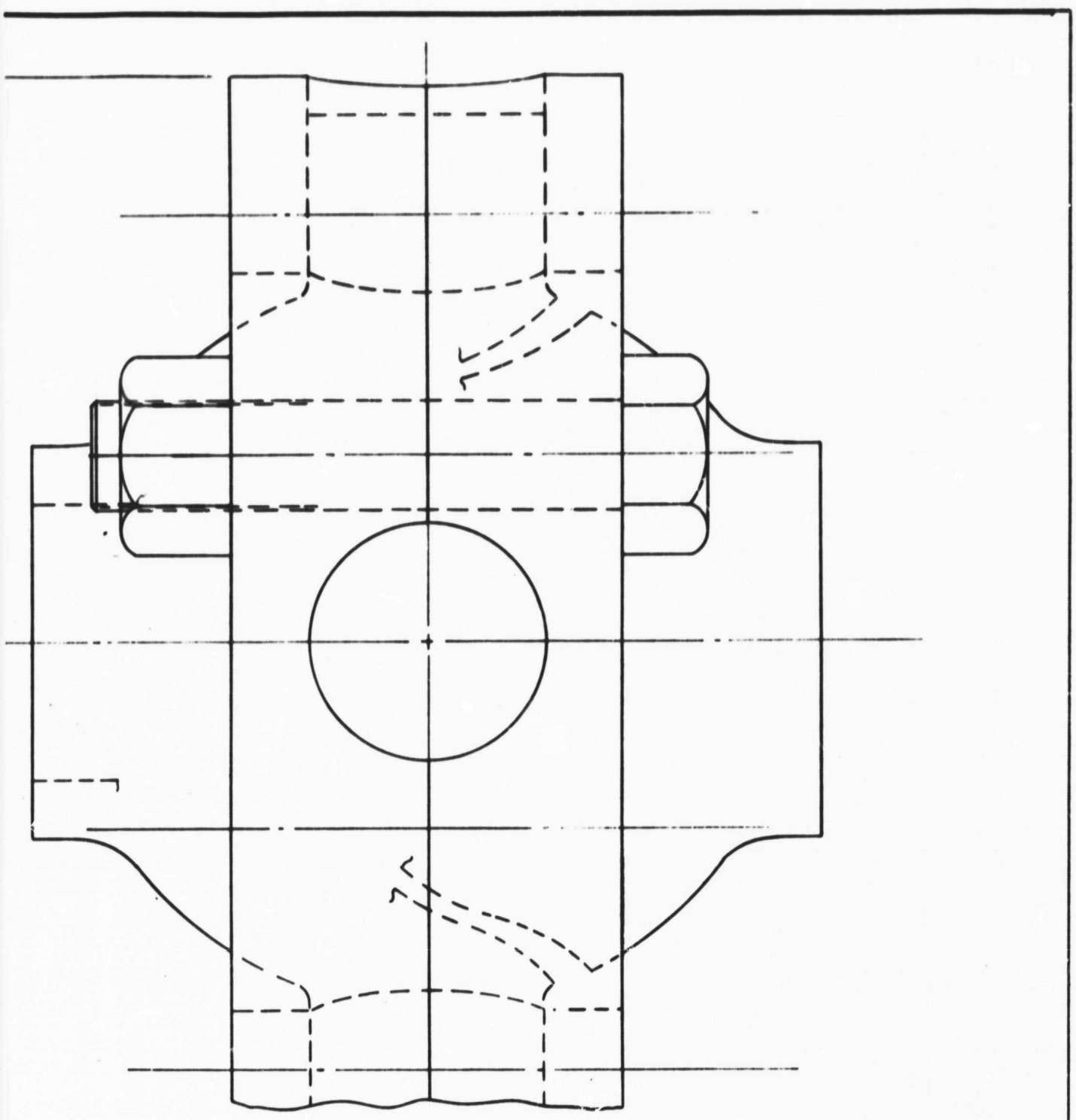
SCALE FULL | SHEET 1 OF 1

e SOLAR
WELDED

INC THRU

A-12-2


14.4
DIA
SECTION A-A



NOTES: UNLESS OTHERWISE SPECIFIED

1. DIMENSIONS LOCATING THE TRUE POSITION ARE BASIC
2. DO NOT SCALE DRAWING. WORK TO DIMENSIONS GIVEN
3. DRAWING INTERPRETATION PER MIL-D-70327 AND SOLAR SPEC 9-4, SECT
4. RING MATERIAL IS INCONEL 718

SSIBLE SHEAR SHOULDER

QTY REQD	QTY REQD	PART NUMBER	DESCRIPTION	MATERIAL	SPECIFICATION	UNIT WT	
			LIST OF MATERIAL (PARTS LIST)			ASSY WT	

APPROVED BY	
STAFF	
STAFF	
R. Wills	5-1-64
PROJ. ENG	5-1-64
DESIGN	
CHECKED	
DRAWN R. Brown	5-1-64

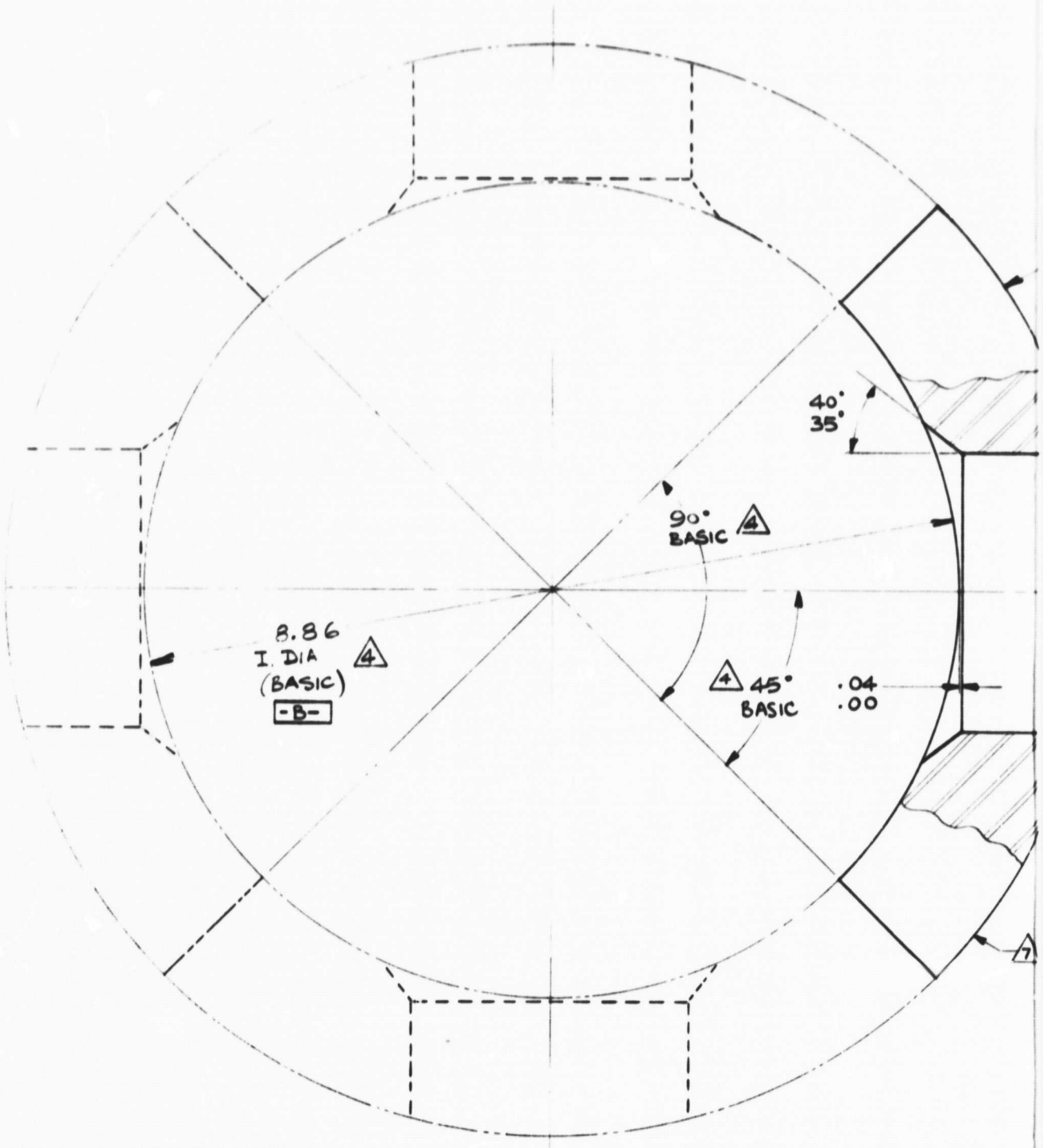
SOLAR
A Division of International Harvester Company

DRAWING TITLE
GIMBAL
MACHINED LOW HEIGHT BELLOWS
BOLTED RING DESIGN FOR 39704

CODE IDENT NO.	SIZE	DRAWING NO.
55820	D	39746

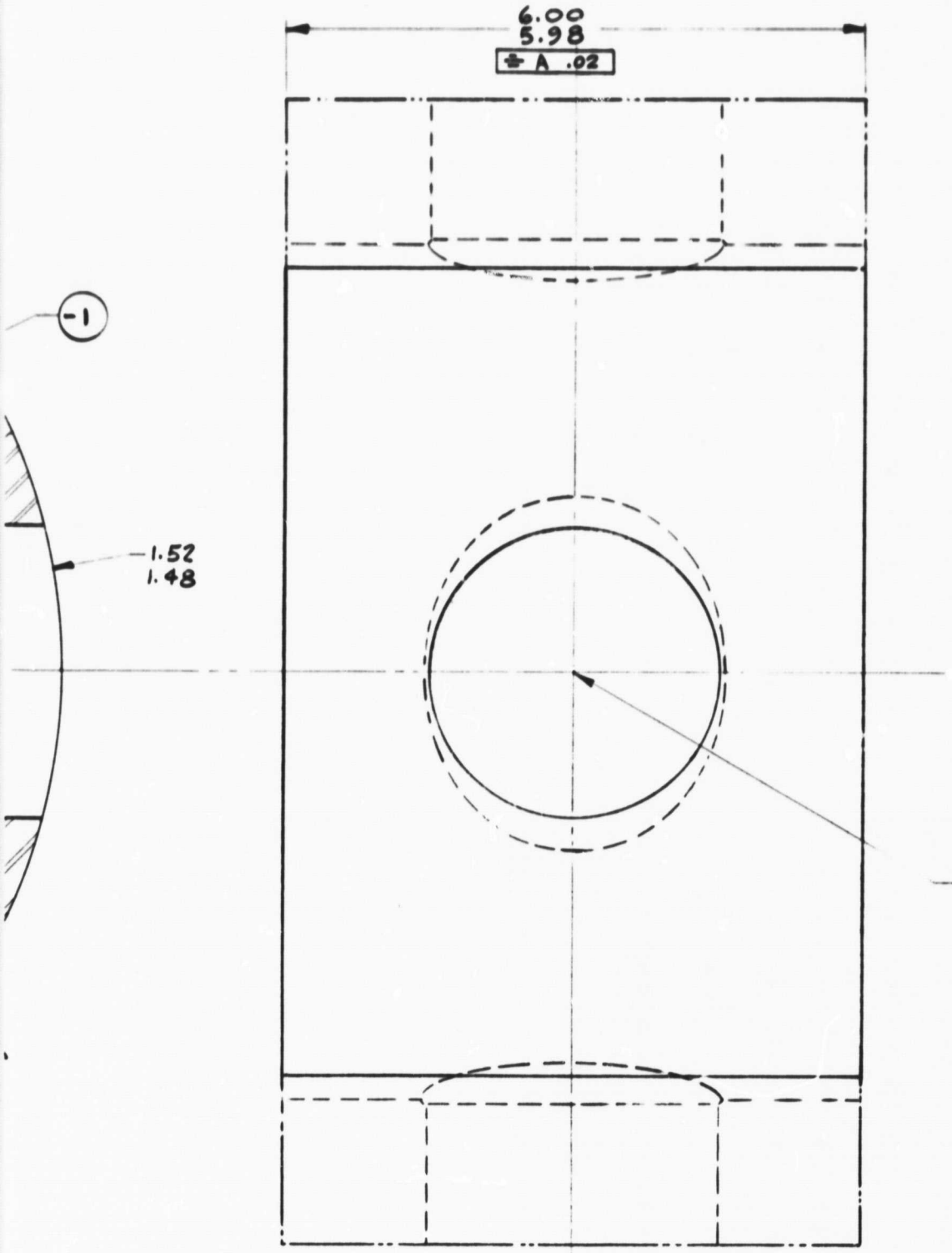
SCALE FULL | SHEET 1 OF 1

A-14-2



NOTES: UNLESS OTHERWISE SPECIFIED

1	DIMENSIONS LOCATING THE TRUE POSITION AR
2	DO NOT SCALE DRAWING. WORK TO DIMENSION
3	DRAWING INTERPRETATION PER MIL-D-70327 AND SOLAR SPEC 9-4, SECT 6.0
4	DIMENSIONS OF RING QUADRANT WITH (3) OTHER RING QUADRA 8.84 - 8.88 I.D. RING AT NEX 41308-0 GIMBAL ASSY. TOLERAN QUADRANT THAT BUTT AGAINST TO BE MATCHED TO PROVIDE CLEAR ELECTRON BEAM WELDING SHO
5	INCONEL 718 ALLOY BAR & FO ROCKETDYNE SPEC RB 0170-038
6	HEAT TREAT -1 (INCONEL 718) ULTIMATE TENSILE STRENGTH &
7	IDENTIFY WITH SOLAR PART Nº ETCH METHOD PER MIL-STD-13 SOLAR SPEC MPS 7-815
8	IDENTIFICATION & TRACEABILITY TO SPEC 60B00010 PARA 14.3



: BASIC
: GIVEN

TO BE MATCHED
STS TO OBTAIN
T ASSY,
CES ON ENDS OF
DJACENT QUADRANTS
ANCE REQUIRED FOR
JN IN NEXT ASSY.
RGING TO BE PER

TO 180,000 PSI MIN.
5% MIN. ELONGATION.
SY ELECTRO-CHEMICAL
), PARA 5.3.1 &

COMPLY WITH MSFC

QTY REQD	QTY REQD	PART NUMBER	DESCRIPTION	MATERIAL	SPECIFICATION	UNIT WT	NOTE
		-1	QUADRANT	INCONEL 718	5	74.15	

LIST OF MATERIAL (PARTS LIST) — ASSY WT

APPROVED BY	
STAFF	
STAFF	
PROJ. ENG	R. Kress 7.21.64
DESIGN	[illegible]
CHECKED	[illegible] 7/10/64
DRAWN	W. Wolf 6-22-64

SOLAR — A Subsidiary of International Harvester Company

DRAWING TITLE

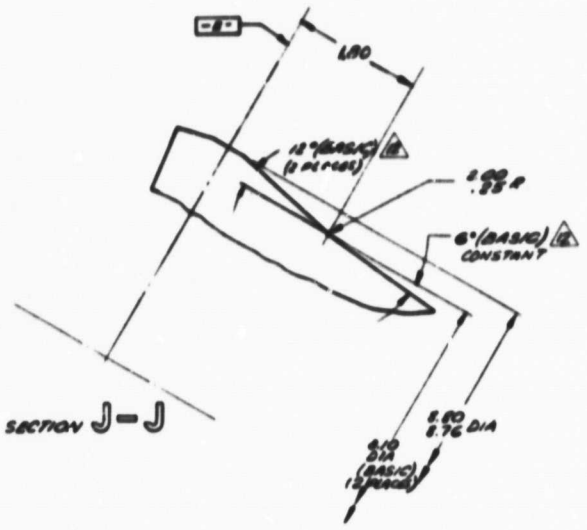
RING, QUADRANT, GIMBAL

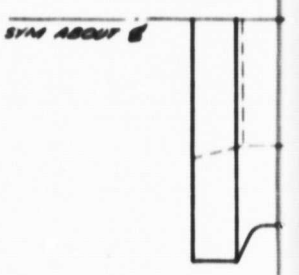
±6° 10,000 PSI (MATCHED SET)

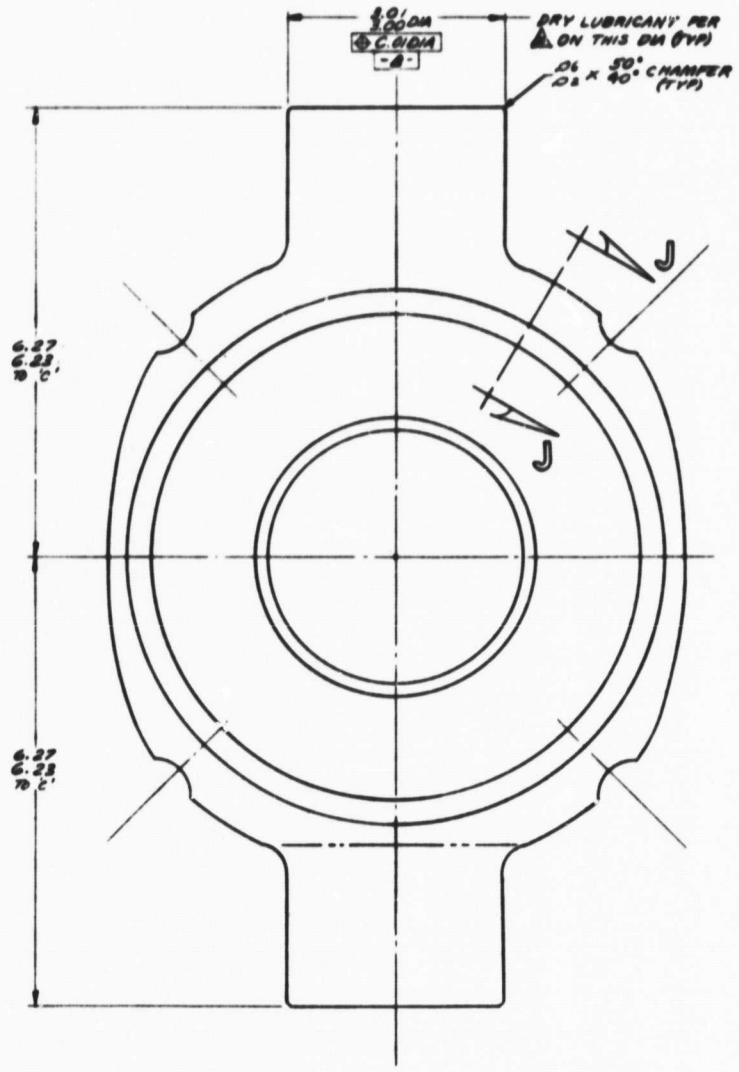
CODE IDENT NO.	SIZE	DRAWING NO.
55820	D	41306

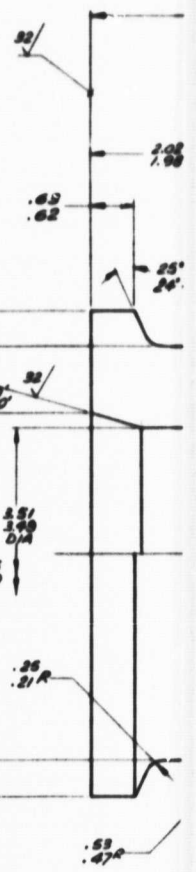
SCALE FULL — SHEET 1 OF 1

A-16-2







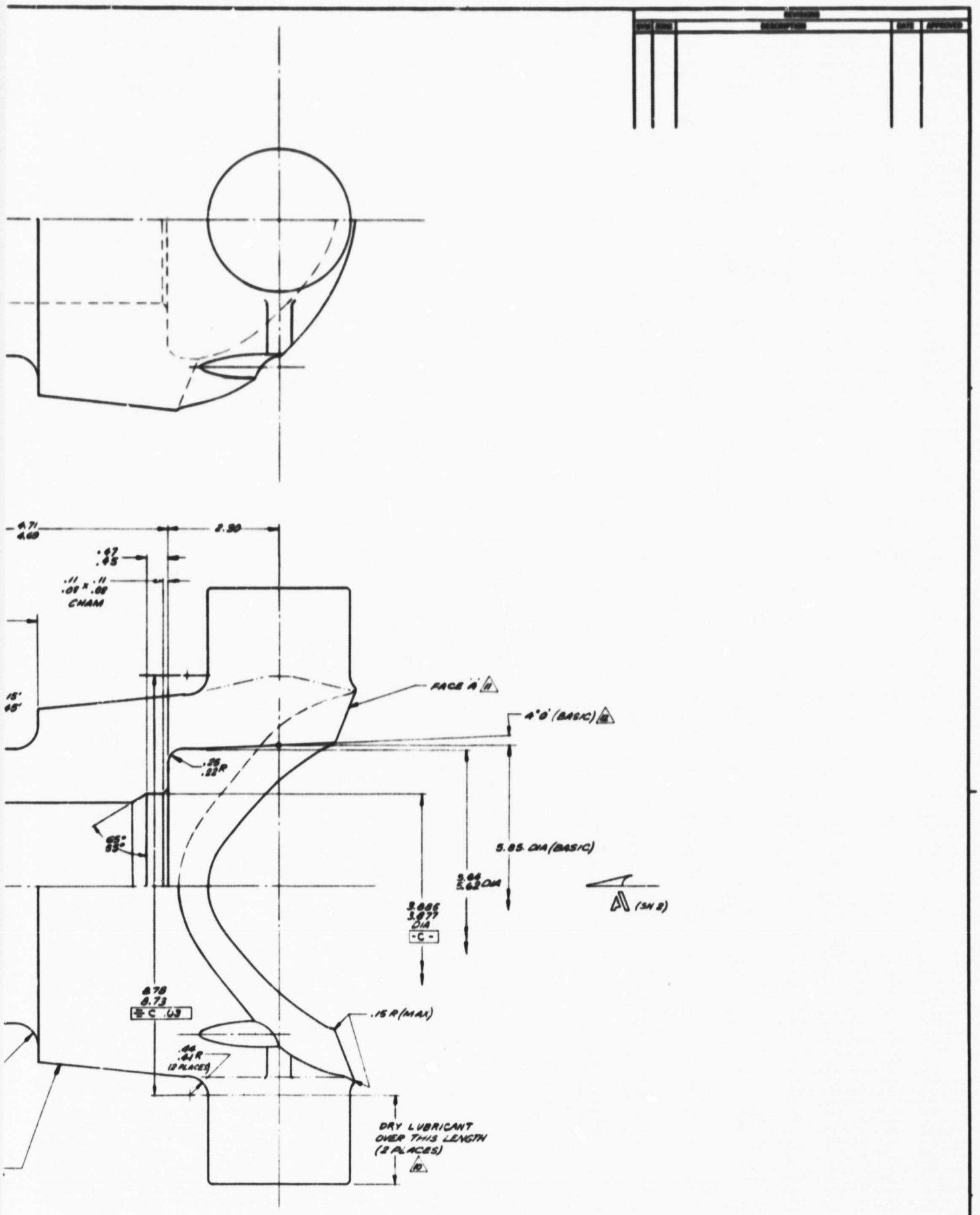


11 SURFACE DEFINED AS FACE 'A' TO BE WITHIN .0
OF BASIC LOCATION.

12 SURFACES DEFINED BY THESE ANGLES (6°, 6°)
TO BE WITHIN .06 TIR OF BASIC LOCATION

13 ALL SURFACES TO BE 125 UNLESS OTHERWISE

A-17-1



NOTES: UNLESS OTHERWISE SPECIFIED

1. DIMENSIONS LOCATING THE TRUE POSITION ARE BASIC
2. DO NOT SCALE DRAWING. WORK TO DIMENSIONS GIVEN
3. DRAWING INTERPRETATION PER MIL-D-70327 AND SOLAR SPEC 94, SECT 6.0, & 7.0
4. INCONEL 718 ALLOY FORGING TO BE PER ROCKETDYNE SPEC RB0170-038
5. PASSIVATE PER MIL-S-5002
6. HEAT TREAT INCONEL 718 TO 180,000 PSI MIN ULTIMATE TENSILE STRENGTH & 15% MIN ELONGATION
7. IDENTIFY WITH SOLAR NO BY ELECTRO-CHEMICAL ETCH METHOD PER MIL-STD 130 PARA 5.3.1 & SOLAR SPEC MPS 7-815.
8. IDENTIFICATION & TRACEABILITY TO COMPLY WITH MSFC SPEC 60B00010 PARA 4.3
9. THIS ASSY HAS BEEN DESIGNED FOR NASA CONTRACT NAS10-1266
   - OPERATING PRESS — 10,000 PSIG
   - PROOF PRESS — 15,000 PSIG
   - BURST PRESS — 40,000 PSIG
   - OPERATING MEDIA — GASEOUS NITROGEN & GASEOUS HELIUM
   - OPERATING TEMP — -65°F TO +160°F
10. DRY LUBRICANT PER MIL-N-25027

ITEM	QTY REQ'D	PART NUMBER	DESCRIPTION	MATERIAL	SPECIFICATION	UNIT WT
	-1			INCONEL 718 4		

LIST OF MATERIAL (PARTS LIST)

APPROVED BY

STAFF

STAFF

PROJ ENG

DESIGN

CHECKED

STRESS

SOLAR

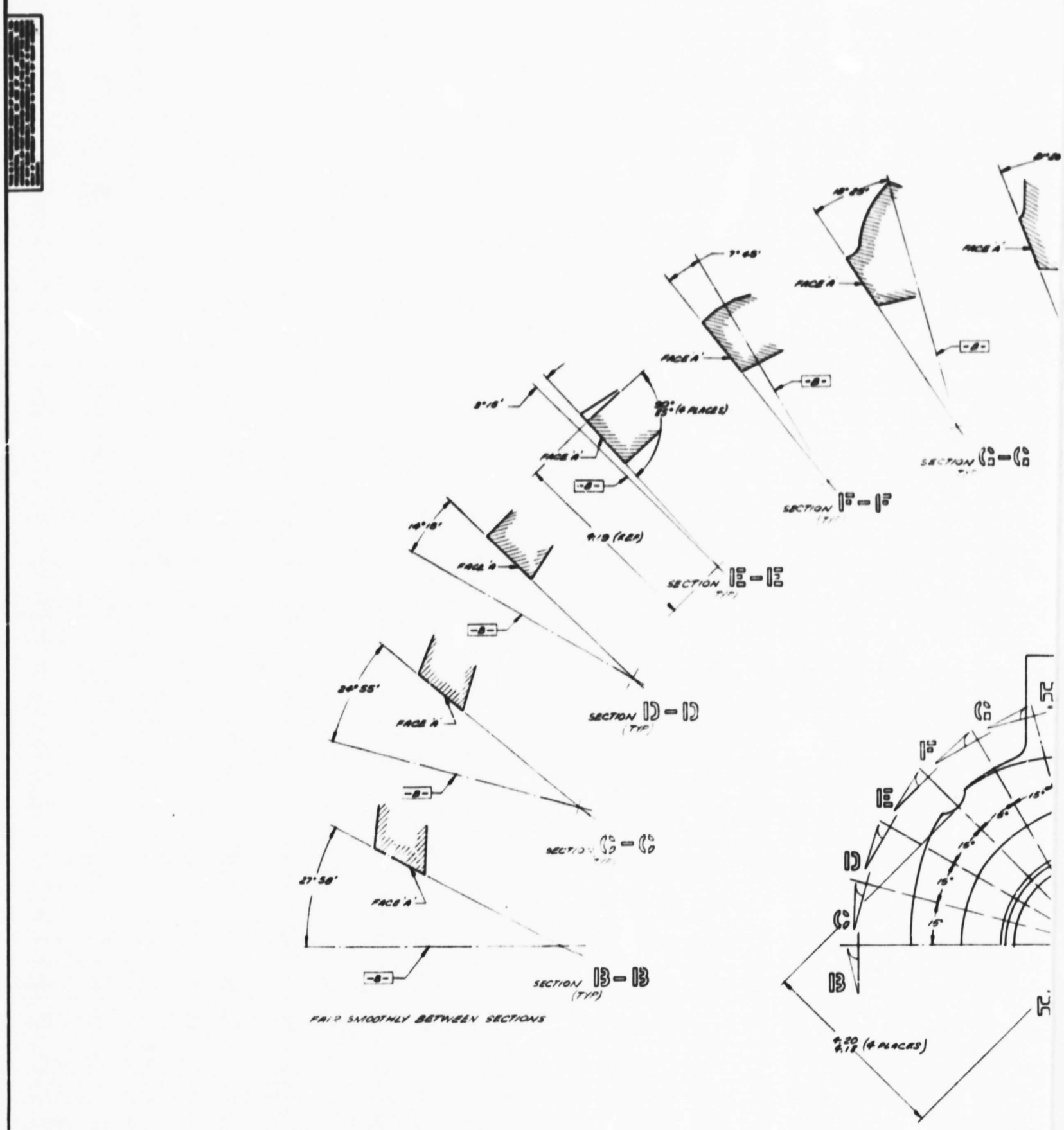
DRAWING TITLE

FLANGE, GIMBAL - ±6°, 10,000 P.S.I.

CODE IDENT NO. 55820 | SIZE E | DRAWING NO. 41307

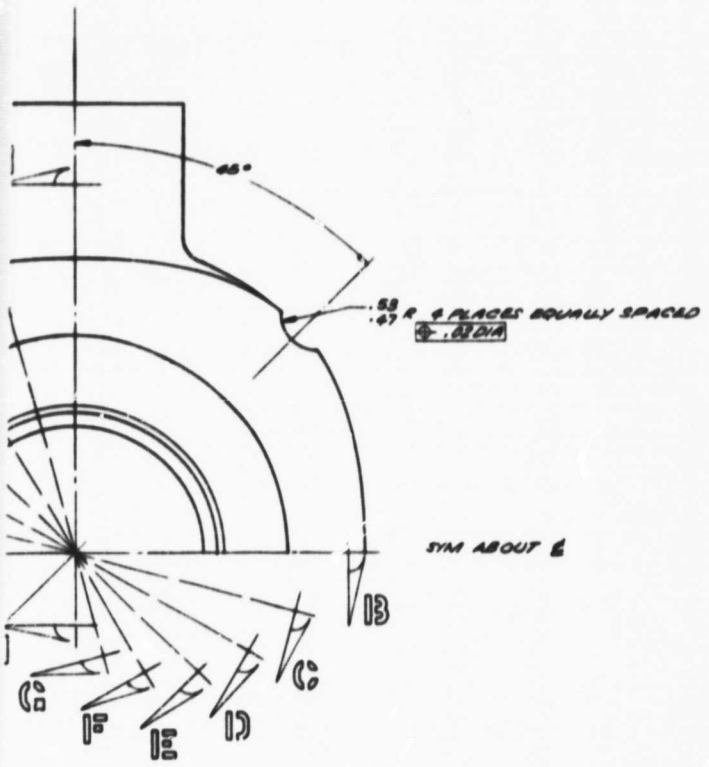
SCALE FULL | SHEET 1 OF 2

4-18-2


FACE 'A'
SECTION 15 - 15
SECTION 13 - 13
FAIR SMOOTHLY BETWEEN SECTIONS
7.20
4.12 (4 PLACES)

A-19-3

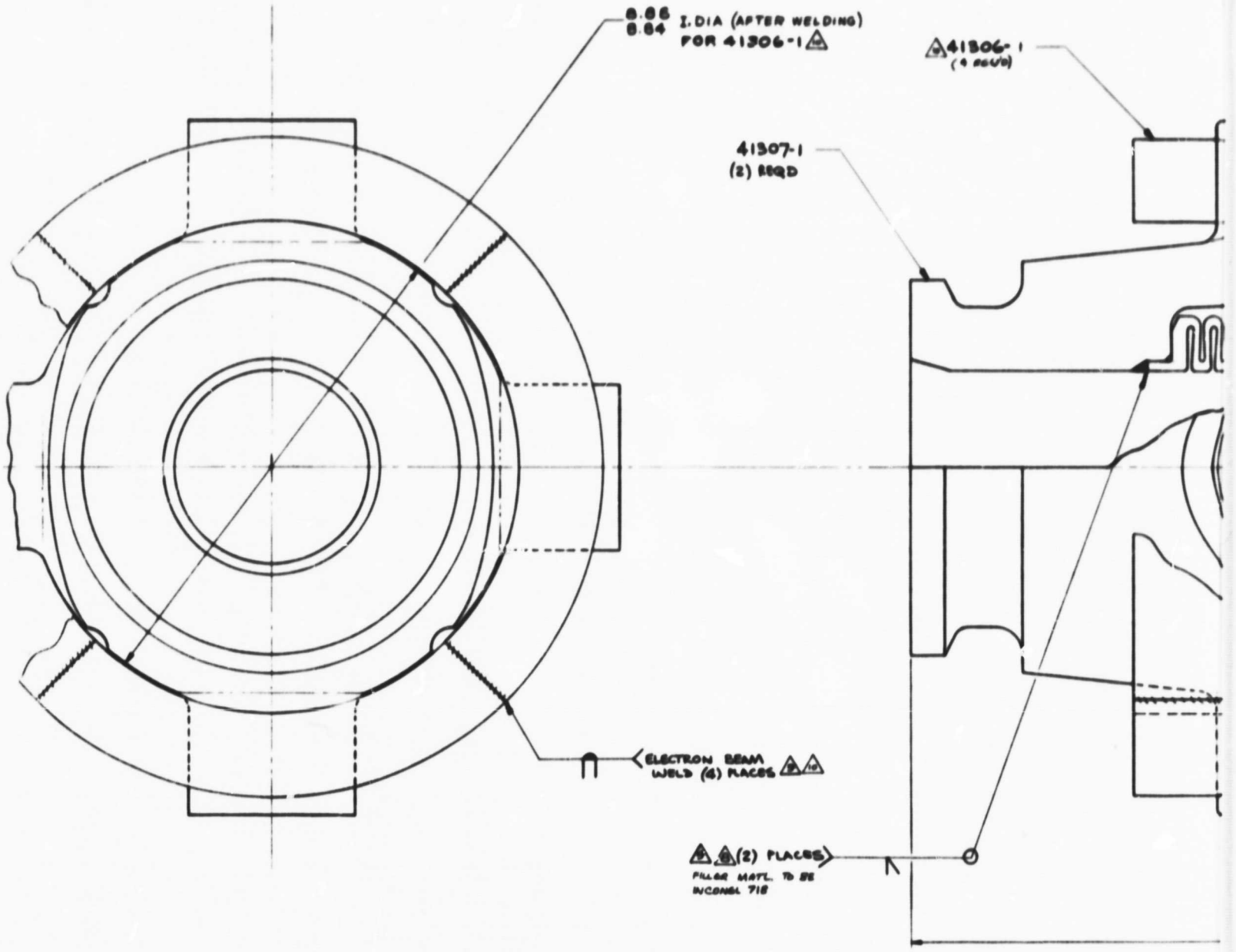




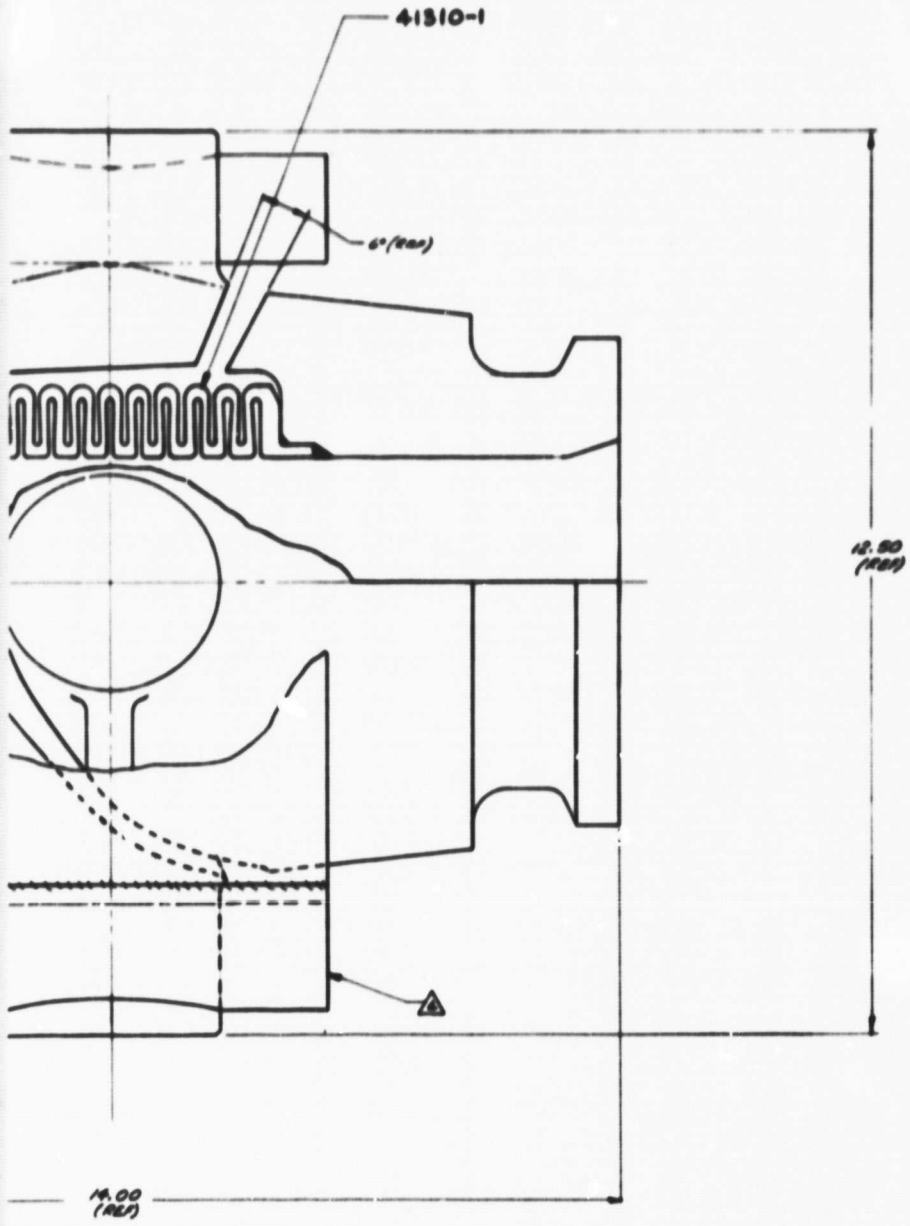
VIEW A (SHT 1)

SOLAR		
55820	E	41307
SCALE FULL		SHEET 2 OF 2

A-20-4



11 GIMBAL ASSY DESIGNED FOR ± 5°46' (MIN) ANGUL
IN ANY PLANE



ATION

NOTES: UNLESS OTHERWISE SPECIFIED

1. DIMENSIONS LOCATING THE TRUE POSITION ARE BASIC
2. DO NOT SCALE DRAWING. WORK TO DIMENSIONS GIVEN
3. DRAWING INTERPRETATION PER SSL-D-70327 AND SOLAR SPEC 94 SECT 6 7.0
4. IDENTIFY WITH SOLAR NO BY ELECTRO-CHEMICAL ETCH METHOD PER MIL-STD-130, PARA 5.3.1.4 SOLAR SPEC MPS 7-815
5. IDENTIFICATION & TRACEABILITY TO COMPLY WITH MSFC SPEC 50B00010 PARA 14.3.
6. GIMBAL JOINT SHALL ANGULATE STOP TO STOP IN ANY PLANE THROUGH ± 6° 45' WITHOUT DAMAGE.
7. THIS GIMBAL HAS BEEN DESIGNED FOR NASA CONTRACT NAS 10-1266 (REP)
   - OPERATING PRESSURE : 10,000 PSIG
   - PROOF PRESSURE : 15,000 PSIG
   - BURST PRESSURE : 40,000 PSIG
   - OPERATING MEDIA : GASEOUS NITROGEN & GASEOUS HELIUM
   - OPERATING TEMP. : -65°F TO +160°F
4. WELD PER MIL-W-8611 (CRITICAL) EXCEPTION NO RADIOGRAPHIC INSPECTION.
5. HEAT TREATMENT OF THESE WELDS TO BE IN AS-WELDED CONDITION.
6. AFTER WELDING, THE I.D. OF THE GIMBAL RING ASSY AND THE POSITION OF THE HOLES FOR THE PINS SHALL CONFORM TO THE REQUIREMENTS OF 41306

ITEM	PART NUMBER	DESCRIPTION	MATERIAL	SPECIFICATION	QTY
1	41310-1	BELLOWS			4
2	41307-1	FLANGE, GIMBAL			
4	41306-0	RING, QUADRANT			
-0		LIST OF MATERIAL (PARTS LIST)		22C.SF.1	

APPROVED BY:

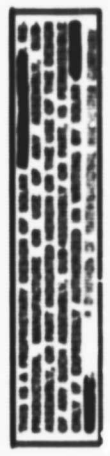
SOLAR

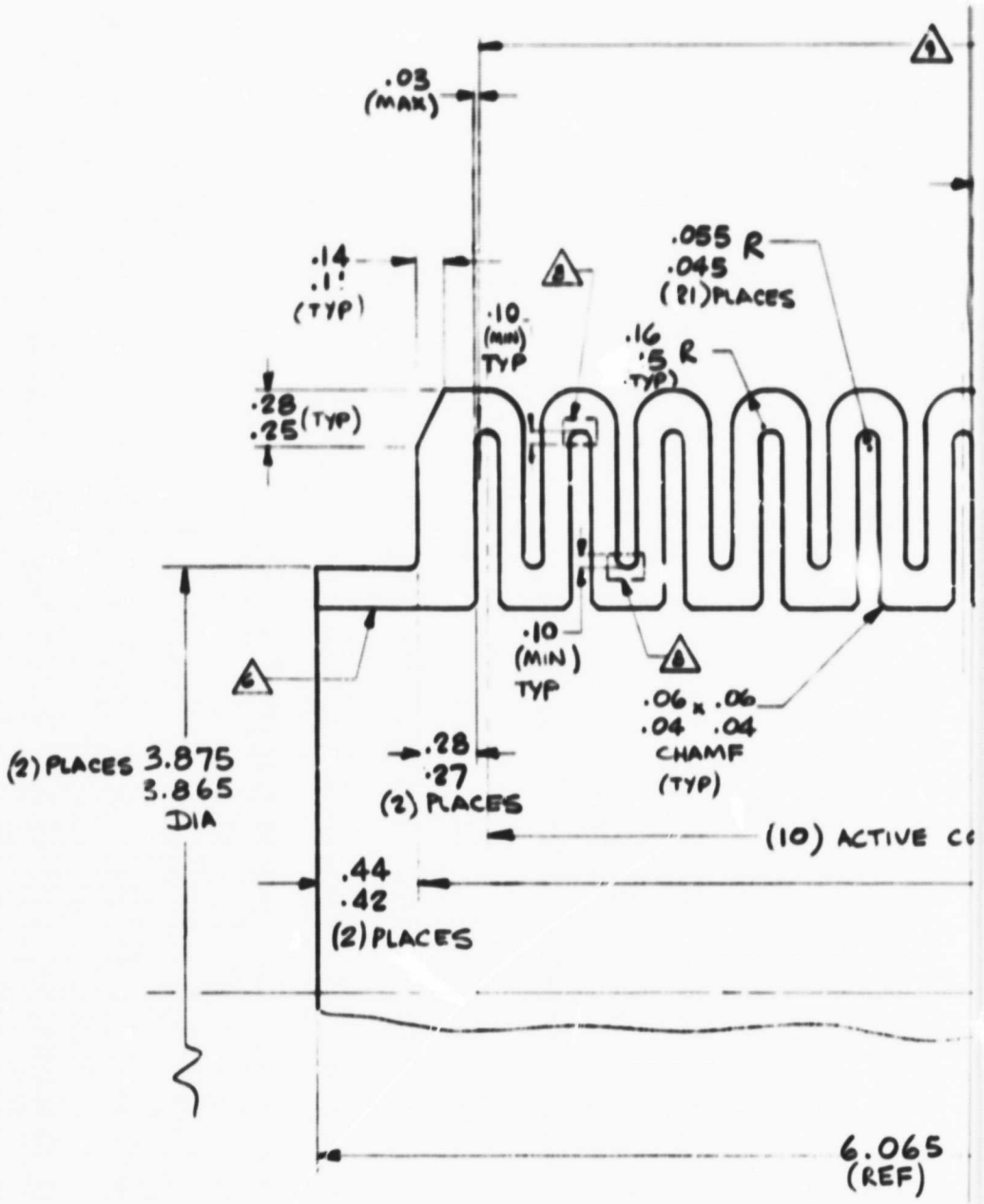
GIMBAL ASSY - ±6°, 10,000 PSI

55820 | E | 41308

SIZE FSCM NO. | DWG NO. | REV | SHEET 1

11-22-2

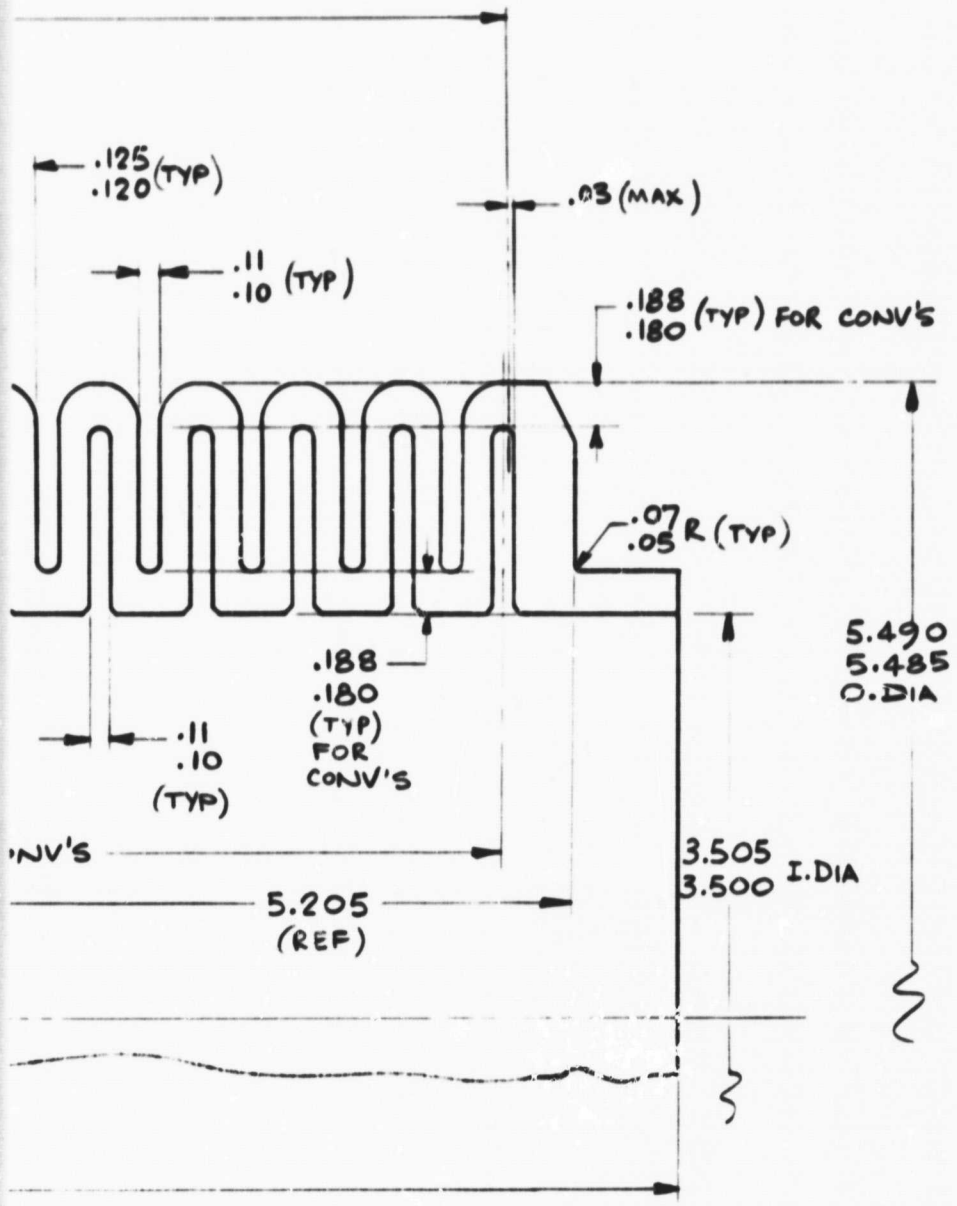




NOTES: UNLESS OTHERWISE SPECIFIED

1. DIMENSIONS LOCATING THE TRUE POSITION AR
2. DO NOT SCALE DRAWING. WORK TO DIMENSION
3. DRAWING INTERPRETATION PER MIL-D-70327 AND SOLAR SPEC 9-4, SECT 6.0
4. INCONEL 718 ALLOY BARS & FORG PER ROCKETDYNE SPEC RB0170-
5. HEAT TREAT INCONEL 718 ITEMS TO I ULTIMATE TENSILE STRENGTH & 15
6. IDENTIFY WITH SOLAR PART Nº 4 HIGH NUMERALS & NON-PERMANE SOLAR SPEC 9-6 CLASS "E".
7. IDENTIFICATION & TRACEABILITY MSFC SPEC 60B00010 PARA 14
8. SURFACE INSIDE THIS AREA TO BE NO SCRATCHES DEEPER THAN .001 CRESTS & 10 PLACES FOR ROOTS O IF FINAL FINISH IS ACCOMPLISHE ELECTRO-CHEMICAL METHOD, BE 64 R.M.S. (MAX) WITH SCRATCH
9. EXCEPT FOR 8 ALL SURFACES OF C TO BE 64 RMS (MAX) WITH NO SCRA FINISH OUTSIDE OF THIS LENGTH TO BE SEE 4-1310-1 FOR COMPRESSION SE

A-23-1



E BASIC
S GIVEN

INGS TO BE
238
30,000 PSI MIN.
% MIN. ELONGATION
309-1 USING .12-.24
NT INK PER

TO COMPLY WITH
.3
32 RMS (MAX) WITH
(TYP 11 PLACES FOR
CONVOLUTIONS).
D BY A CHEMICAL OR
SURFACE IN THIS AREA MAY
NO DEEPER THAN .002.
ONVOLUTIONS IN THIS LENGTH
TCHES DEEPER THAN .002
25 RMS
T CONDITION

QTY REQD	QTY REQD	PART NUMBER	DESCRIPTION	MATERIAL	SPECIFICATION	UNIT WT	ITEM
		-1		INCONEL 718			

LIST OF MATERIAL (PARTS LIST) ASSY WT 14.06lb

APPROVED BY	
STAFF	
STAFF	
PROJ. ENG	R. Kiess 7.21.64
DESIGN	M. Swift 6/23/64
CHECKED	Contents 7/10/64
DRAWN	W. Wells 6-17-64

SOLAR a Division of International Harvester Company

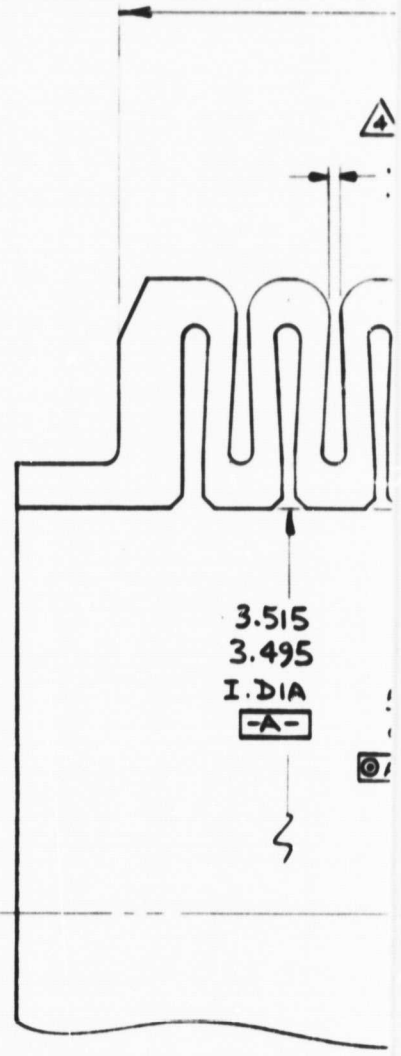
DRAWING TITLE

BELLOWS, MACHINED - ±6°, 10,000 PSI

CODE IDENT NO.	SIZE	DRAWING NO.
55820	D	41309

SCALE TWICE SIZE SHEET 1 OF 1

A-24-2

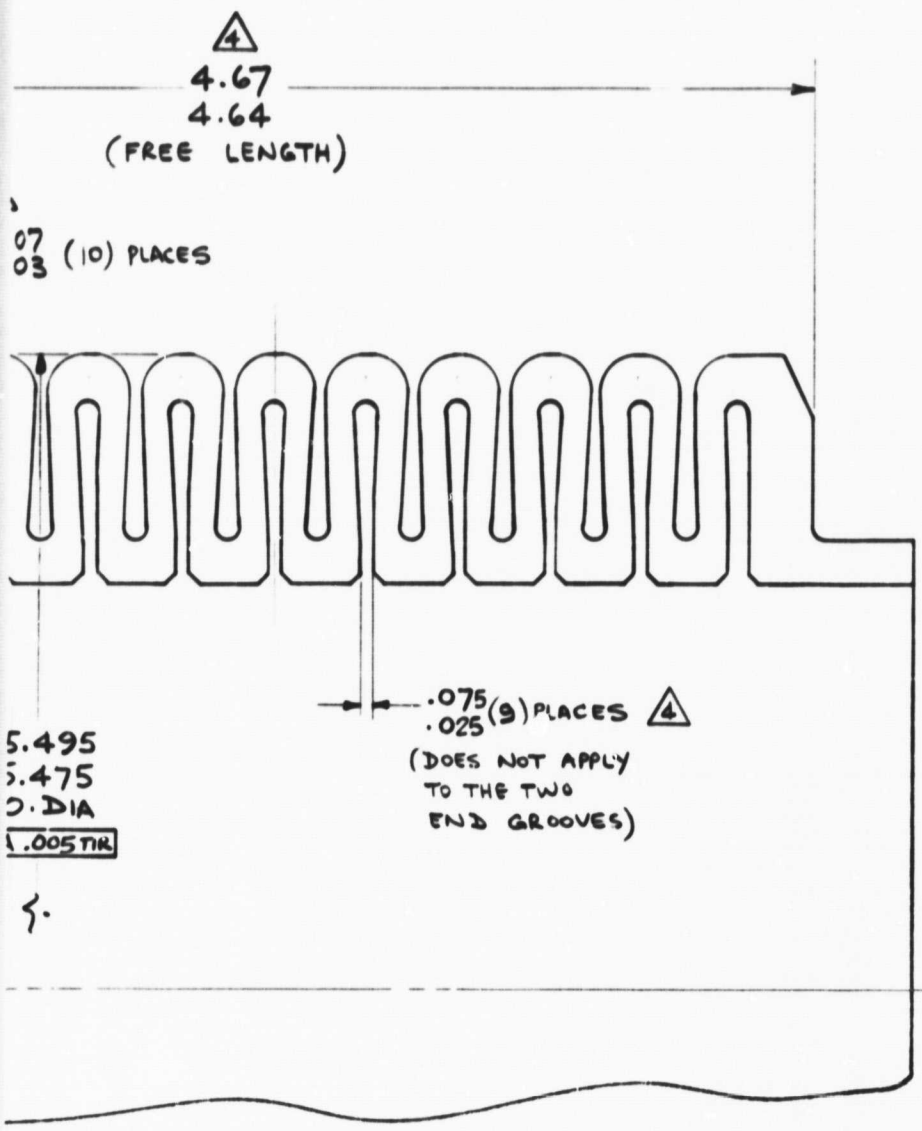


NOTES: UNLESS OTHERWISE SPECIFIED

1. DIMENSIONS LOCATING THE TRUE POSITION A
2. DO NOT SCALE DRAWING. WORK TO DIMENSIO
3. DRAWING INTERPRETATION PER MIL-D-70327 ~~AND SOLAR SPEC S-4, SECT~~
4. BELLOWS TO BE COMPRESSED, A PRESSURIZED TO 35,000 PSI ± THE PRESSURE & THE COMPRES REMOVED, THE FREE LENGTH SPACING DIMENSIONS ARE M
5. PASSIVATE PER MIL-S-5002
6. THIS BELLOWS HAS BEEN D CONTRACT NAS 10-1266 (

OPERATING PRESSURE	:
PROOF PRESSURE	:
BURST PRESSURE	:
OPERATING MEDIA	:
OPERATING TEMP.	:

A-25-1



RE BASIC
IS GIVEN

JD WHILE COMPRESSED,
500 PSI, SO THAT WHEN
SING FORCE ARE
AND CONVOLUTION
ET.

ESIGNED FOR NASA
REF)
10,000 PSIG
15,000 PSIG
40,000 PSIG
GASEOUS NITROGEN
& GASEOUS HELIUM
-65°F TO +160°F

QTY REQD	QTY REQD	PART NUMBER	DESCRIPTION	MATERIAL	SPECIFICATION	UNIT WT	ZONE
		-1			MAKE FROM 41309-1		

LIST OF MATERIAL (PARTS LIST) ASSY WT

APPROVED BY	
STAFF	
STAFF	
PROJ. ENG	R. Kiley 7-2-64
DESIGN	M. Sims 6/23/64
CHECKED	Conrad 7/10/64
DRAWN	J. Wolcott 6-18-64

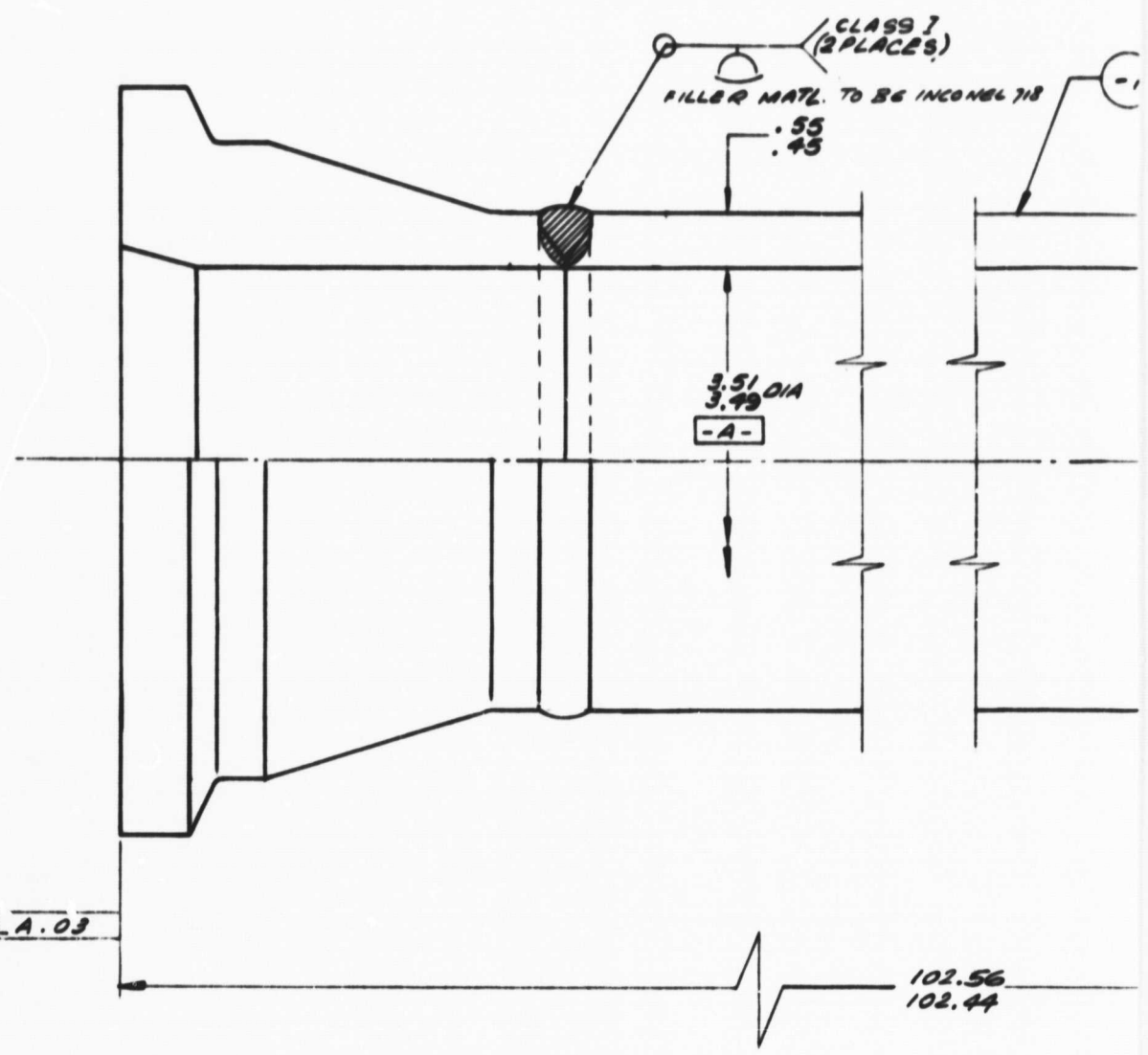
SOLAR — SAN DIEGO 11, CALIFORNIA
A Subsidiary of International Harvester Company

DRAWING TITLE
BELLOWS, COMPRESSION SET - ± 6°, 10,000 PSI

CODE IDENT NO.	SIZE	DRAWING NO.
55820	D	41310

SCALE TWICE SIZE | SHEET 1 OF 1

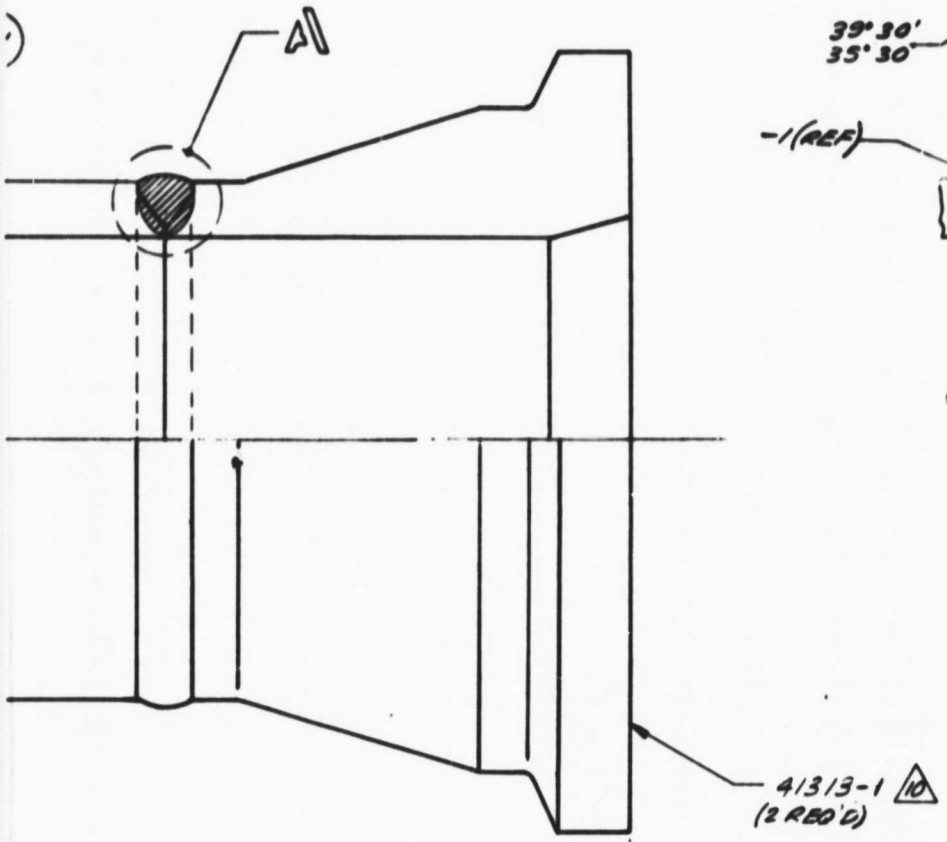
A-26-2

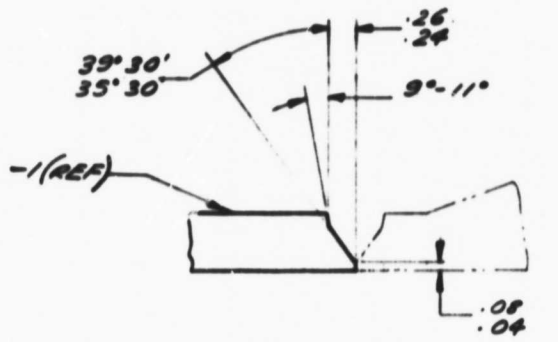


11	INCONEL 718 ALLOY BARS & FORGINGS TO BE PER ROCKETDYNE SPEC RB0170-038
12	MASS SPECTROMETER LEAKAGE TEST TO 1×10^{-6} mm Hg

NOTES: UNLESS OTHERWISE SPECIFIED

1	DIMENSIONS LOCATING THE TRUE POSITION ARE
2	DO NOT SCALE DRAWING. WORK TO DIMENSIONS
3	DRAWING INTERPRETATION PER MIL-D-70327 AND SOLAR SPEC 9-4. SECT 6 0 & 7.0
4	PASSIVATE PER MIL-S-5002
5	WELD PER MIL-W-8611 (CRITICAL,
6	HEAT TREAT AFTER WELDING TO TENSILE STRENGTH
7	IDENTIFY WITH SOLAR NO BY METHOD PER MIL-STD-130 PAR.
8	IDENTIFICATION & TRACEABILI SPEC 60B00010 PARA 14.3
9	THIS ASSY HAS BEEN DESIGNE. 10-1266 OPERATING PRESS. —— PROOF PRESS —— BURST PRESS —— OPERATING MEDIA —— —— OPERATING TEMP ——
10	FOR PROCUREMENT OR PART DWG





DETAIL A
(BEFORE WELDING) (TYP)

BASIC
GIVEN

80,000 PSI MIN ULTIMATE

ELECTRO-CHEMICAL ETCH
A 5.3.1. & SOLAR SPEC MPS 7-815

TY TO COMPLY WITH MSFC

D FOR NASA CONTRACT NAS

——10,000 PSIG
——15,000 PSIG
——40,000 PSIG
——GASEOUS NITROGEN &
——GASEOUS HELIUM.
—— -65°F TO + 160°F

NO SEE SOURCE CONTROL

QTY REQD	QTY REQD	PART NUMBER	DESCRIPTION	MATERIAL	SPECIFICATION	UNIT WT	NOTE
	2	41312-1	UNION (10)			169?	
	1	-1	TUBE		INCONEL 718 (14)	176.34	
	-0	LIST OF MATERIAL (PARTS LIST)				ASSY WT 210.2	

APPROVED BY	
STAFF	
STAFF	
	R. Glass 7/21/64
PROJ. ENG	
DESIGN	7/20/64
CHECKED	Comments 7/13/64
DRAWN	7/16/64

SOLAR
A Subsidiary of International Harvester Company

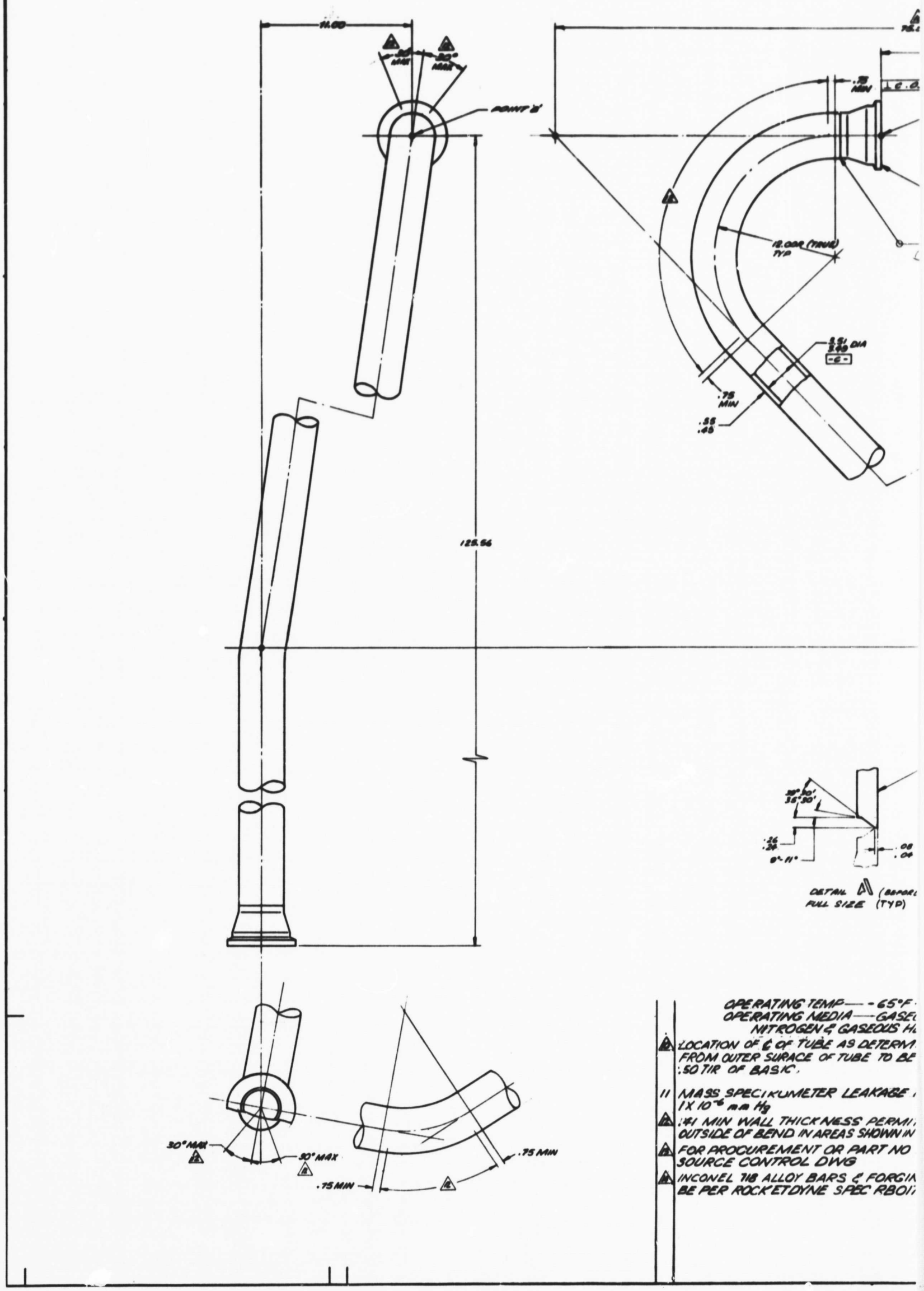
DRAWING TITLE

PIPE, ASSY OF, FLANGED
10,000 PSI

CODE IDENT NO.	SIZE	DRAWING NO.
55820	D	41311

SCALE FULL — SHEET 1 OF 1

A-28-2



OPERATING TEMP —— -65°F
OPERATING MEDIA —— GASE
NITROGEN & GASEOUS H

A LOCATION OF ℄ OF TUBE AS DETERM
FROM OUTER SURACE OF TUBE TO BE
:50 TIR OF BASIC.

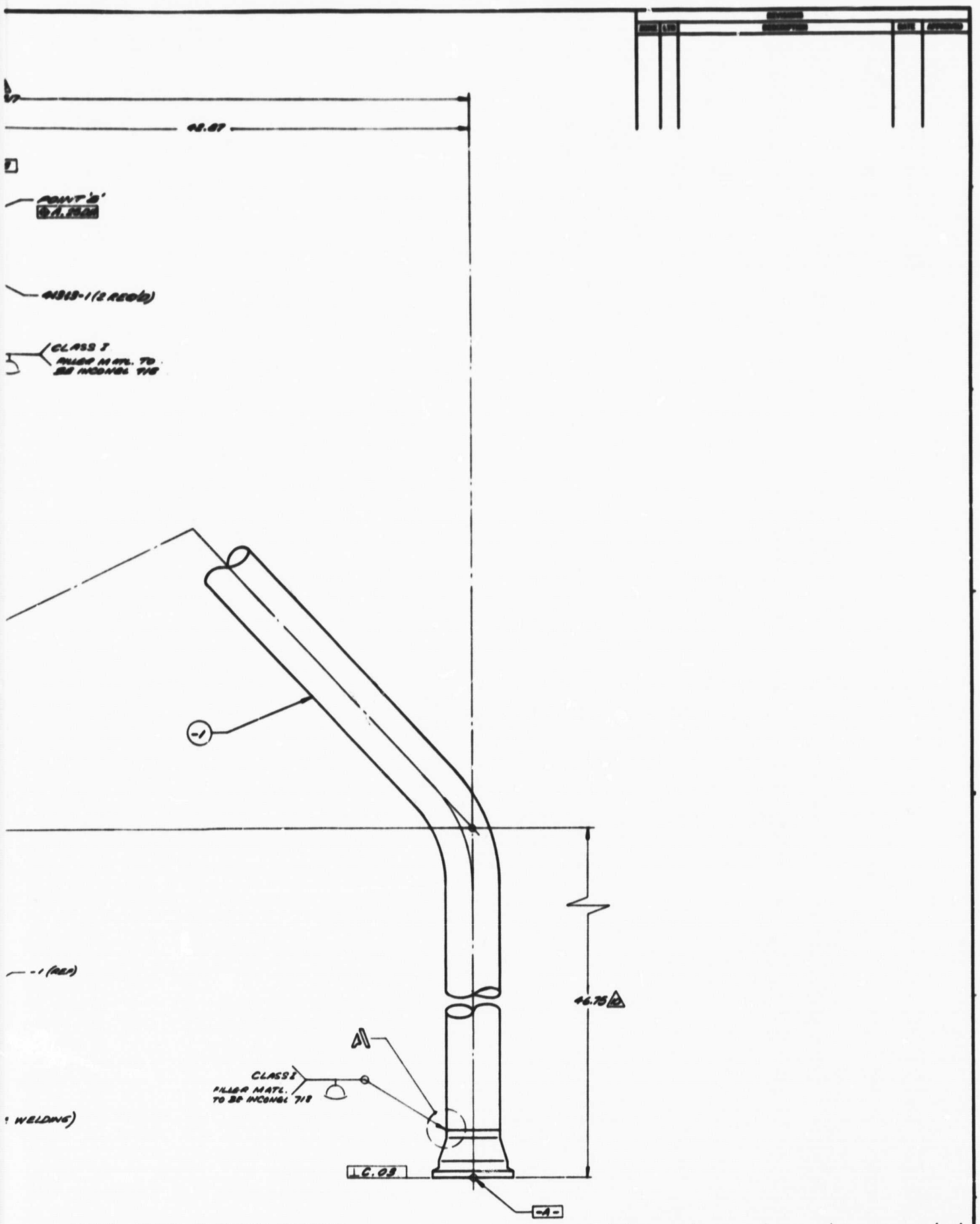
11 MASS SPECTROMETER LEAKAGE
1 X 10⁻⁶ mm Hg

A :41 MIN WALL THICKNESS PERMI
OUTSIDE OF BEND IN AREAS SHOWN IN

A FOR PROCUREMENT OR PART NO
SOURCE CONTROL DWG

A INCONEL 718 ALLOY BARS & FORGI
BE PER ROCKETDYNE SPEC RBOI

A-29-1



NOTES: UNLESS OTHERWISE SPECIFIED

1. DIMENSIONS LOCATING THE TRUE POSITION ARE BASIC
2. DO NOT SCALE DRAWING. WORK TO DIMENSIONS GIVEN
3. DRAWING INTERPRETATION PER MIL-D-70327 AND SOLAR SPEC 9-4, SECT 6.0 & 7.0
4. PASSIVATE PER MIL-S-5002
5. WELD PER MIL-W-8611 (CRITICAL)
6. HEAT TREAT AFTER WELDING TO 180,000 PSI MIN ULTIMATE TENSILE STRENGTH.
7. IDENTIFY WITH SOLAR NO BY ELECTRO-CHEMICAL ETCH METHOD PER MIL-STD-130 PARA 6.3.1 & SOLAR SPEC MPS 7-815
8. IDENTIFICATION & TRACEABILITY TO COMPLY WITH MSFC SPEC 60B00010 PARA 14.3
9. THIS ASSY HAS BEEN DESIGNED FOR NASA CONTRACT NAS10-1266
   OPERATING PRESS — 10,000 PSIG
   PROOF PRESS — 15,000 PSIG
   BURST PRESS — 20,000 PSIG

QTY	ITEM	PART NUMBER	DESCRIPTION	MATERIAL	SPECIFICATION	UNIT WT
2		41313-1	UNION		INCONEL 718	
1		-1	TUBE			
	-0	<-- ASSY NO.	LIST OF MATERIAL (PARTS LIST)			

APPROVED BY GOVT CONTRACT NO.

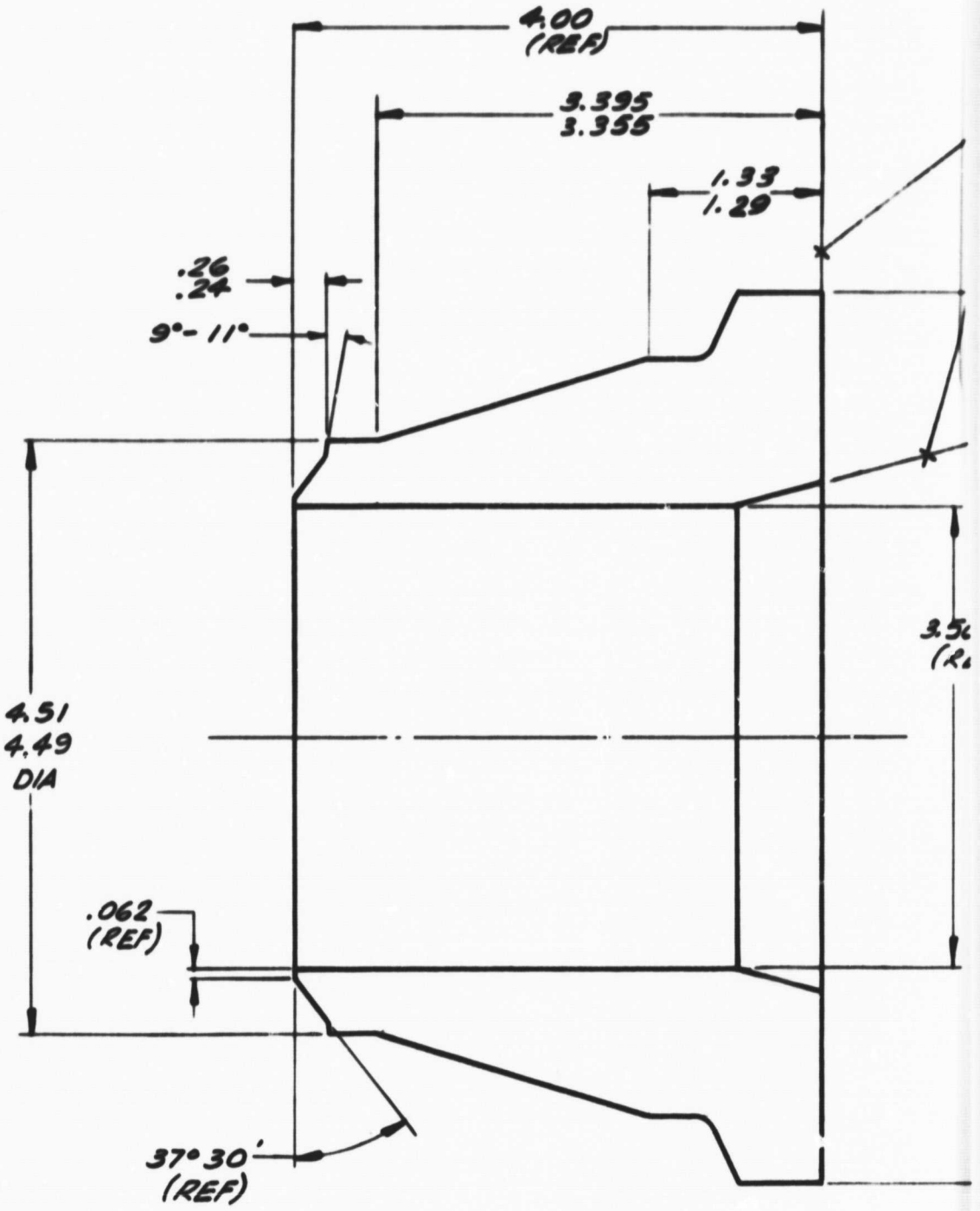
SOLAR
A Division of International Harvester Company

DRAWING TITLE
PIPE ASSY, FLANGED, BENT - 10,000 PSI

SIZE	CODE IDENT NO.	DRAWING NO.
E	55820	41312

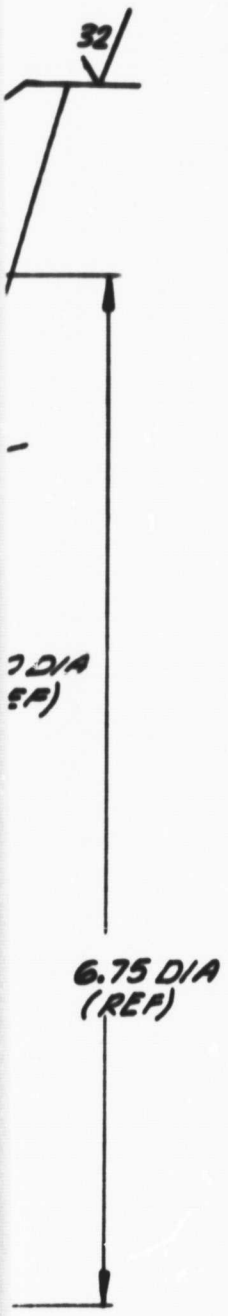
SCALE 1/2 TOTAL WT. 222.55 LB SHEET 1 OF 1

A-30-2



NOTES: UNLESS OTHERWISE SPECIFIED

1. DIMENSIONS LOCATING THE TRUE POSITION ARE BASIC
2. DO NOT SCALE DRAWING. WORK TO DIMENSIONS GIVEN
3. DRAWING INTERPRETATION PER MIL-D-70327 AND SOLAR SPEC 9-4, SI
4. ONLY THE ITEM ON THIS DWG, IDENTIFIED
   VENDORS NAME, ADDRESS & PART NO. HAS
   TESTED & APPROVED BY SOLAR.
   A SUBSTITUTE ITEM SHALL NOT BE USED
   WITHOUT PRIOR TESTING AND APPROVA
   BY COGNIZANT SOLAR CHIEF PROJECT ENGI
5. ITEM SHALL BE PURCHASED FROM GR
   TOOL CO., HOUSTON 1, TEXAS, DWG N
   I-0341
6. VENDOR TO PERMANENTLY IDENTIFY PA
   NO PER MIL-STD-130; BRAND SOLAR A
   NO IN PARENTHESIS WITH PERMANENT
   INK



SOURCE CONTROL DWG

QTY REQD	QTY REQD	PART NUMBER	DESCRIPTION	MATERIAL	SPECIFICATION	UNIT WT
		-1			INCONEL 718	16.93

← ASSY NO. LIST OF MATERIAL (PARTS LIST)

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APPROVED BY

GOVT CONTRACT NO.

PROJECT

DESIGN

CHECK

DRAFT

DESIGN ACTIVITY

CUSTOMER

SOLAR
A Division of International Harvester Company

DRAWING TITLE

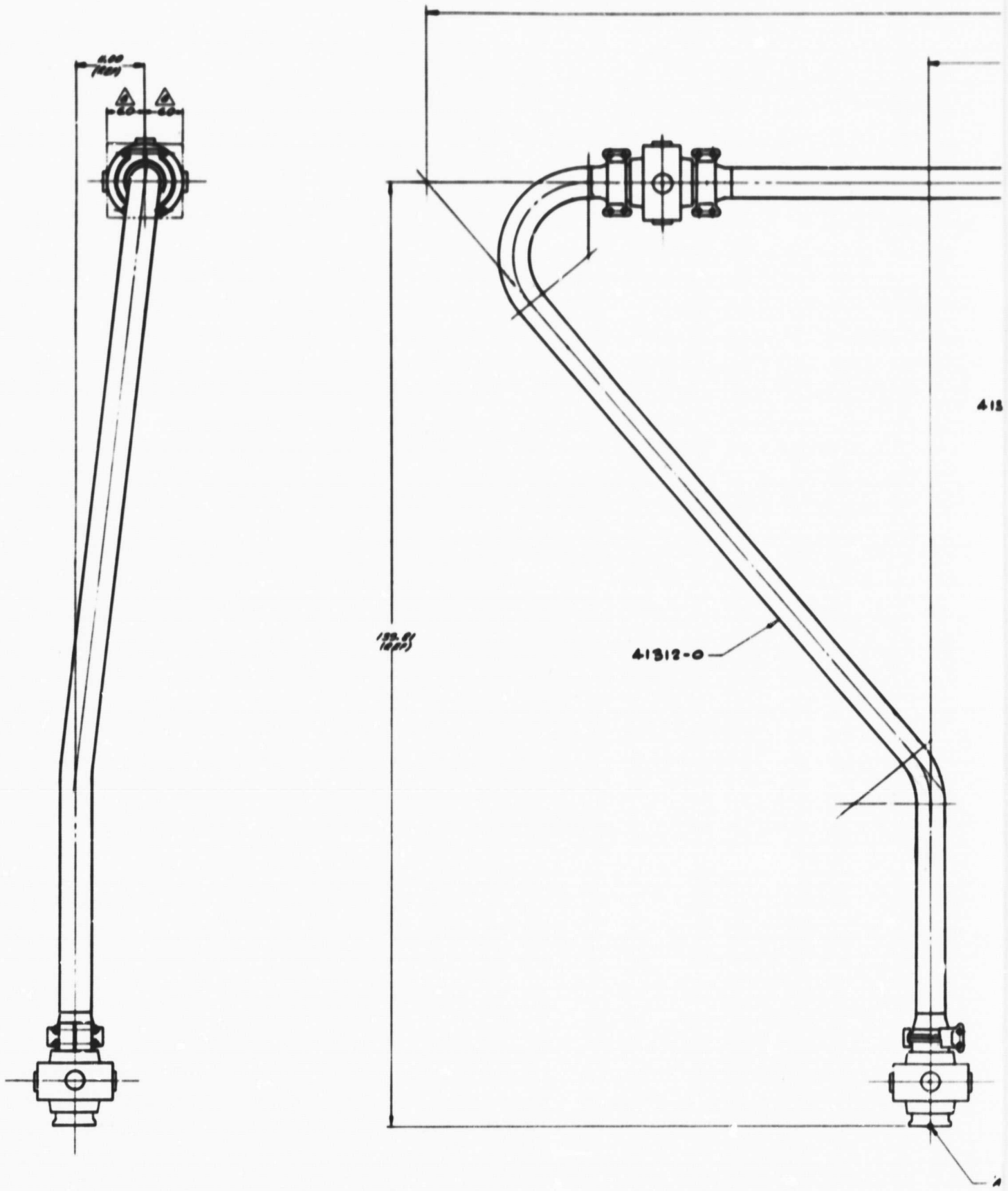
UNION

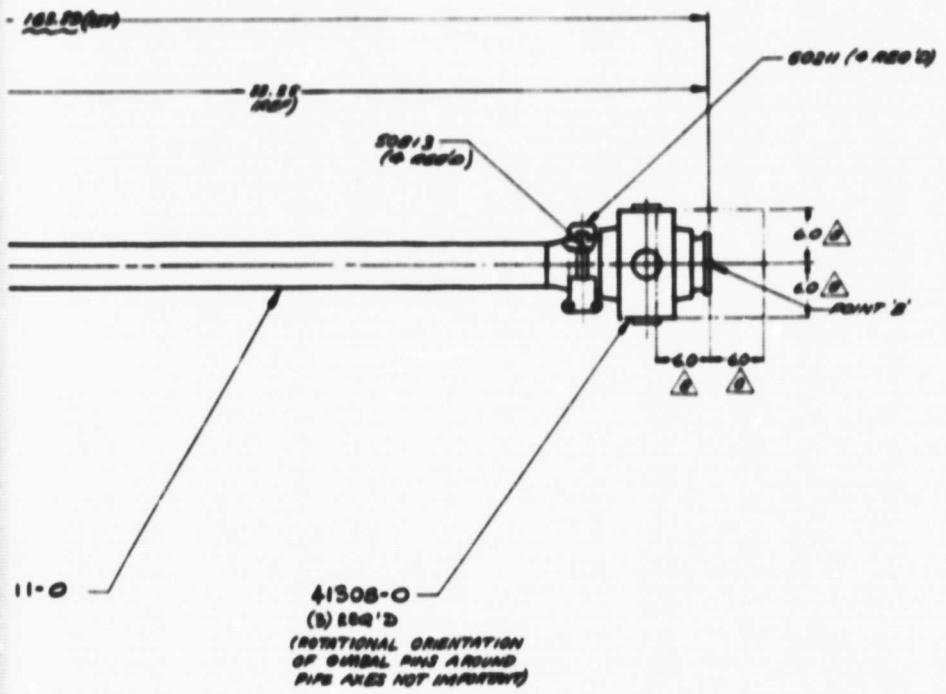
3.50 I.D. PIPE

SIZE	CODE IDENT NO.	DRAWING NO.
C	55820	41313

SCALE FULL | TOTAL WT. 16.93 | SHEET 1 OF 1

A-32-2


41312-0



NOTES: UNLESS OTHERWISE SPECIFIED

1. DIMENSIONS LOCATING THE TRUE POSITION ARE BASIC
2. DO NOT SCALE DRAWING. WORK TO DIMENSIONS GIVEN
3. DRAWING INTERPRETATION PER MIL-D-70327 AND SOLAR SPEC 9-4, SECT
4. PURCHASE FROM GRAY TOOL CO, HOUSTON, TEXAS
5. FOR MOTION STUDY SEE SOLAR DWG NO 41389
6. THIS ASSY HAS BEEN DESIGNED FOR FOR NASA CONTRACT NAS10-266
   - OPERATING PRESS ——— 10,000 PSIG
   - PROOF PRESS ——— 15,000 PSIG
   - BURST PRESS ——— 20,000 PSIG
   - OPERATING TEMP ——— -65°F TO +160°F
   - OPERATING MEDIA ——— GASEOUS NITROGEN & GASEOUS HELIUM
7. THIS ASSY HAS BEEN DESIGNED TO ACCOMMODATE A FLOW RATE OF 20,000 SCFM OF HELIUM OR 100,000 SCFM OF GASEOUS NITROGEN AT OPERATING CONDITIONS
8. THIS ASSY HAS BEEN DESIGNED SO THAT WITH POINT 'A' FIXED, POINT 'B' CAN HAVE ANYWHERE IN THE 12.0 INCH CUBE SHOWN
9. CLEANLINESS TO BE PER MSFC SPEC 164

ITEM	PART NUMBER	DESCRIPTION	MATERIAL	SPECIFICATION	QUANTITY PER
2	50211	CLAMP, TWO PIECE			
4	50813	SEAL, RING 1"-2.44			
1	41312-0	PIPE ASSY, BENT			AS REQ
1	41311-0	PIPE ASSY			AS REQ
2	41308-0	GIMBAL ASSY			AS REQ
-0		LIST OF MATERIAL (PARTS LIST)			

APPROVED BY

STAFF

ENGR

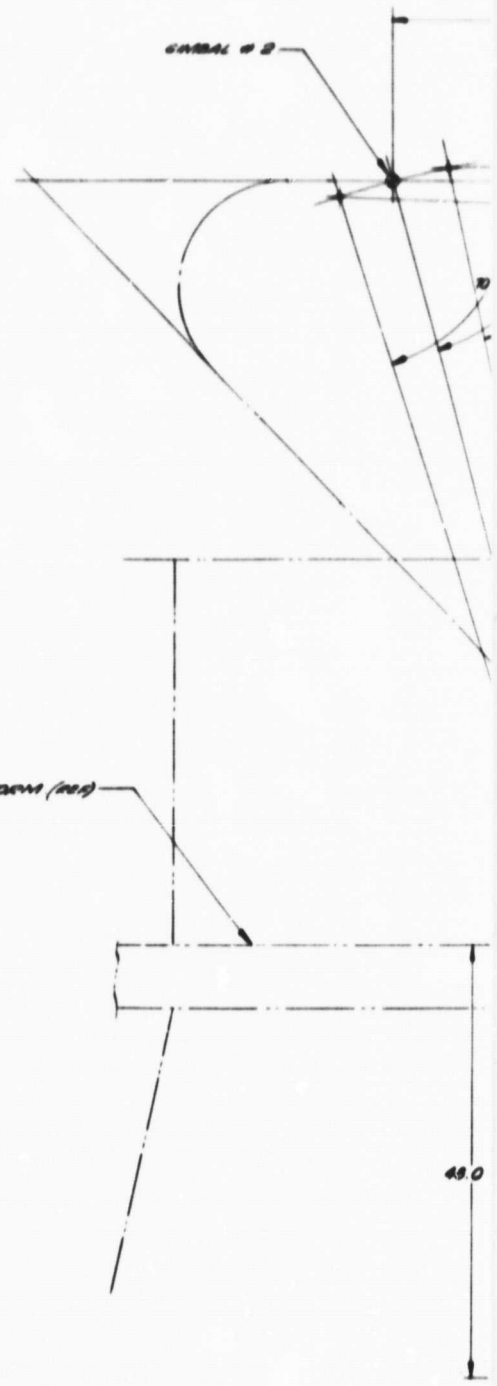
SOLAR

LINE ASSY, CONNECTOR, 10,000 PSI

55820 | E | 41314

SHEET 1 OF 1

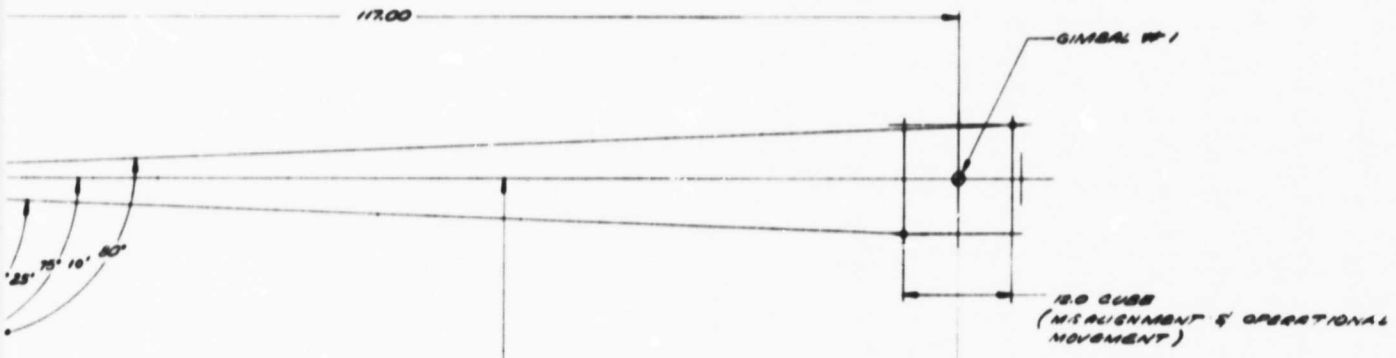
A-34-2

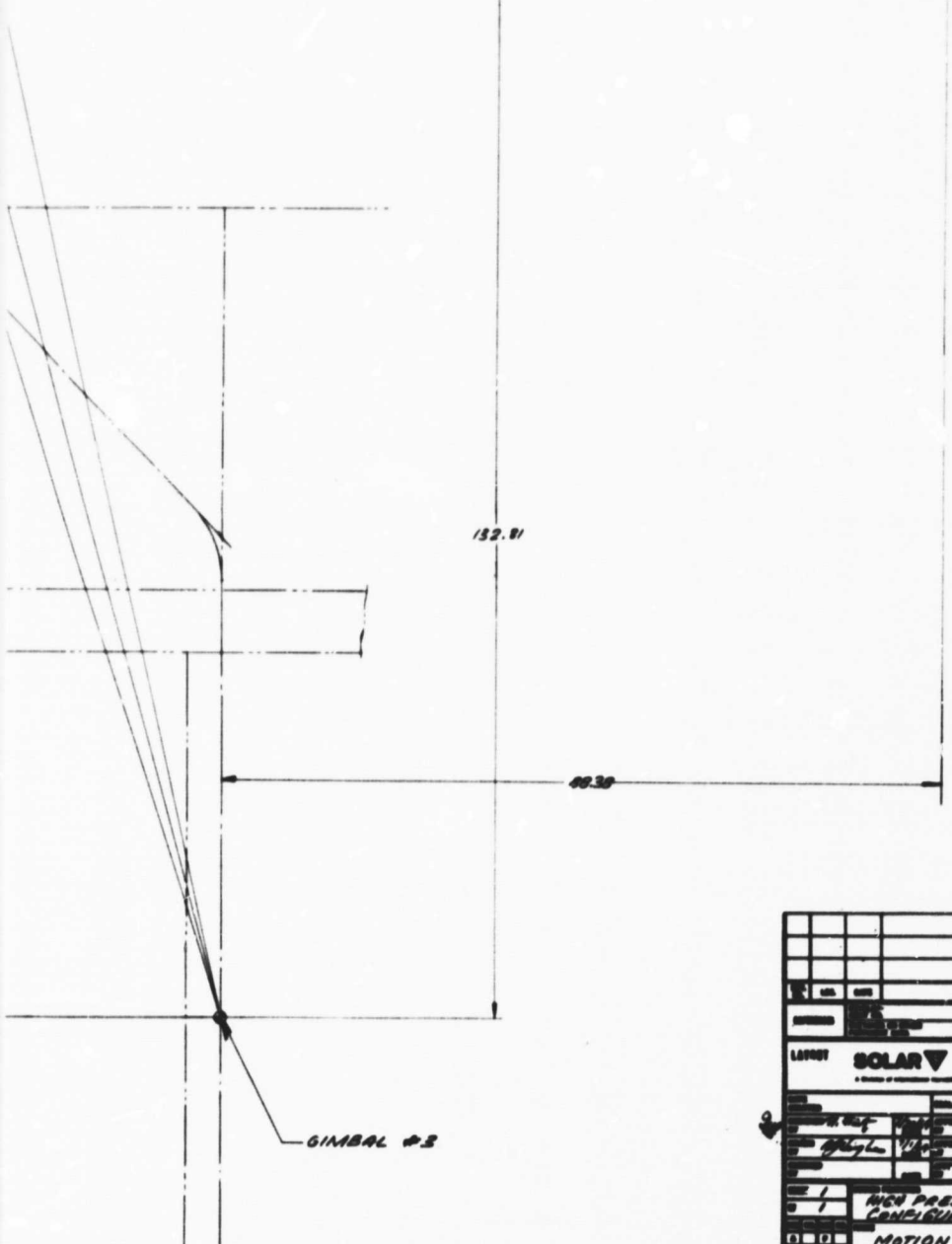


GIMBAL ANGULATION

	IN SIDE VIEW		IN PLAN VIEW
GIMBAL #1	+2°10' -2°10'		+3°0' -3°0'
GIMBAL #2	+4°30' -4°45'	MAX ANGLE IN ANY PLANE: +4°45' -3°30'	+3°0' -3°0'
GIMBAL #3	+2°35' -2°35'		0°



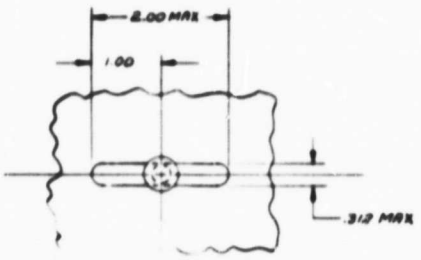




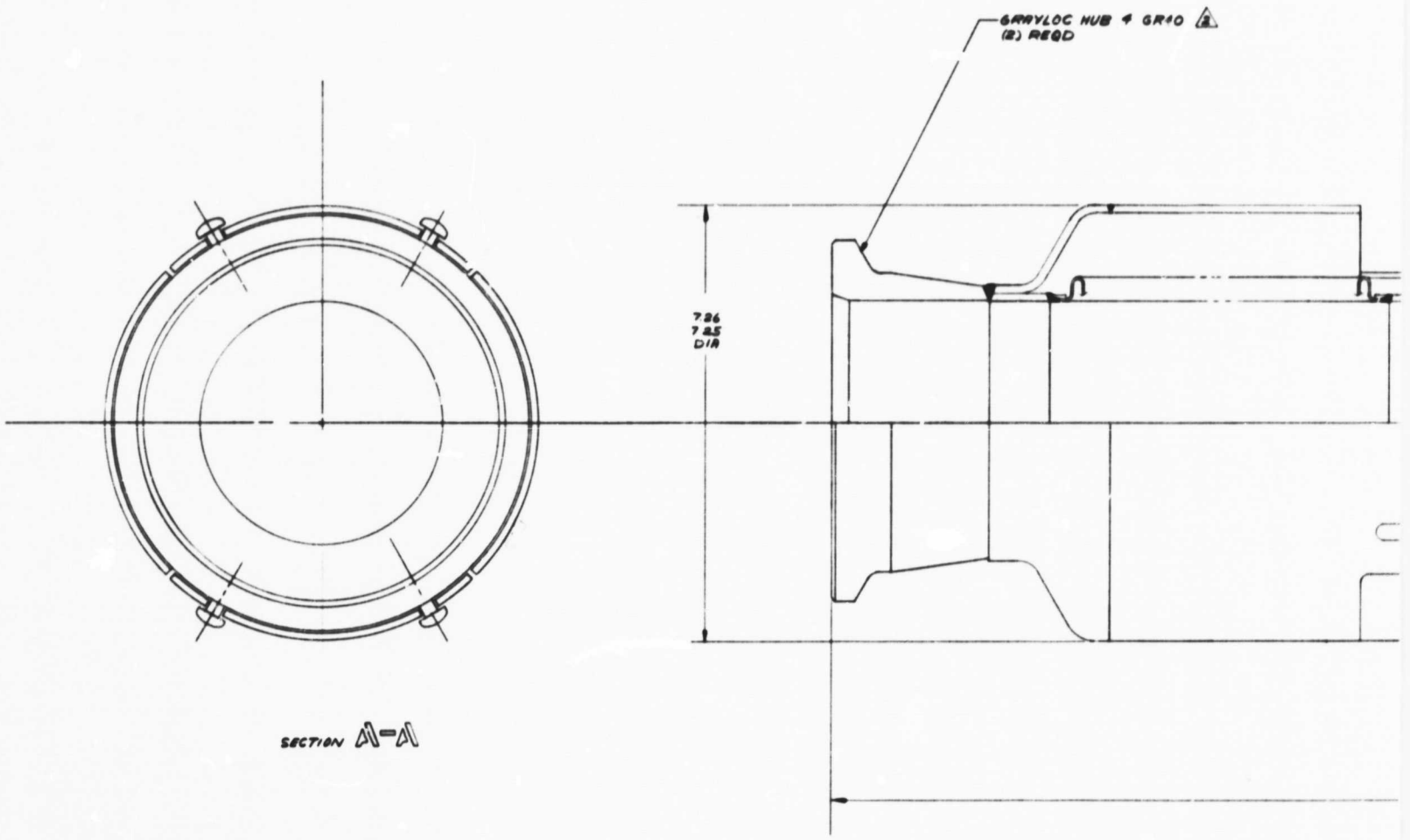
NOTE: 1. THIS MOTION STUDY IS FOR 4134-0 LINE ASSY.

LAYOUT	SOLAR	
	HIGH PRESSURE LINE CONFIGURATION	
	MOTION STUDY	
	41379	

A-36-2

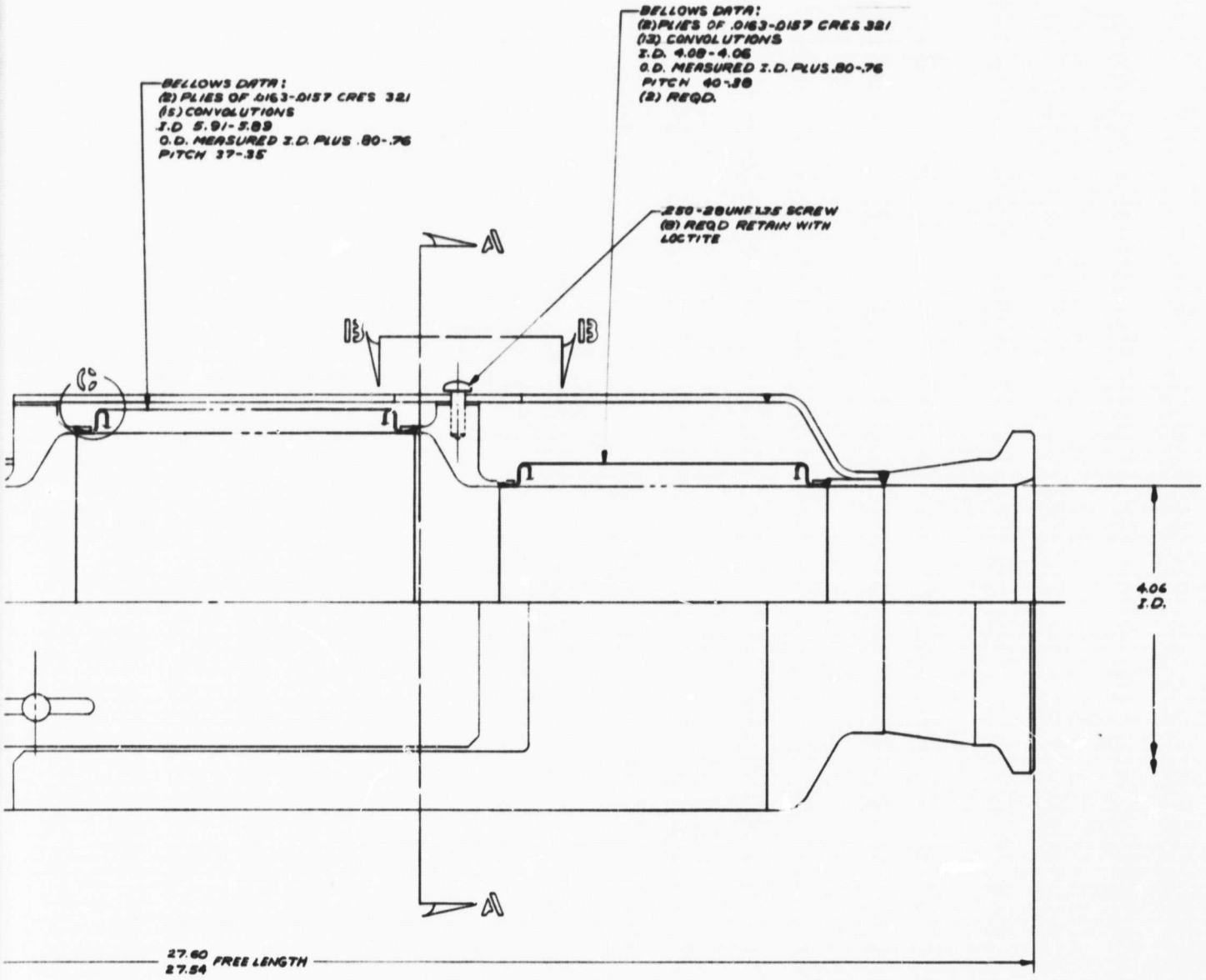


SECTION B-B
SLOT TYP (8) PLACES



SECTION A-A



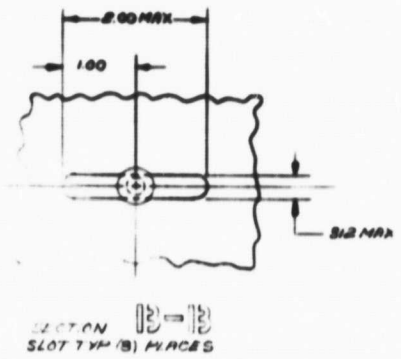


NOTES:
1. MATERIAL - 316L CRES PER AMS 5507 UNLESS OTHERWISE SPECIFIED
2. MAY BE PURCHASED FROM GRAY TOOL CO. P.O. BOX 2291 HOUSTON 1, TEXAS
3. MAXIMUM TRAVEL ±.88
4. OPERATING DATA

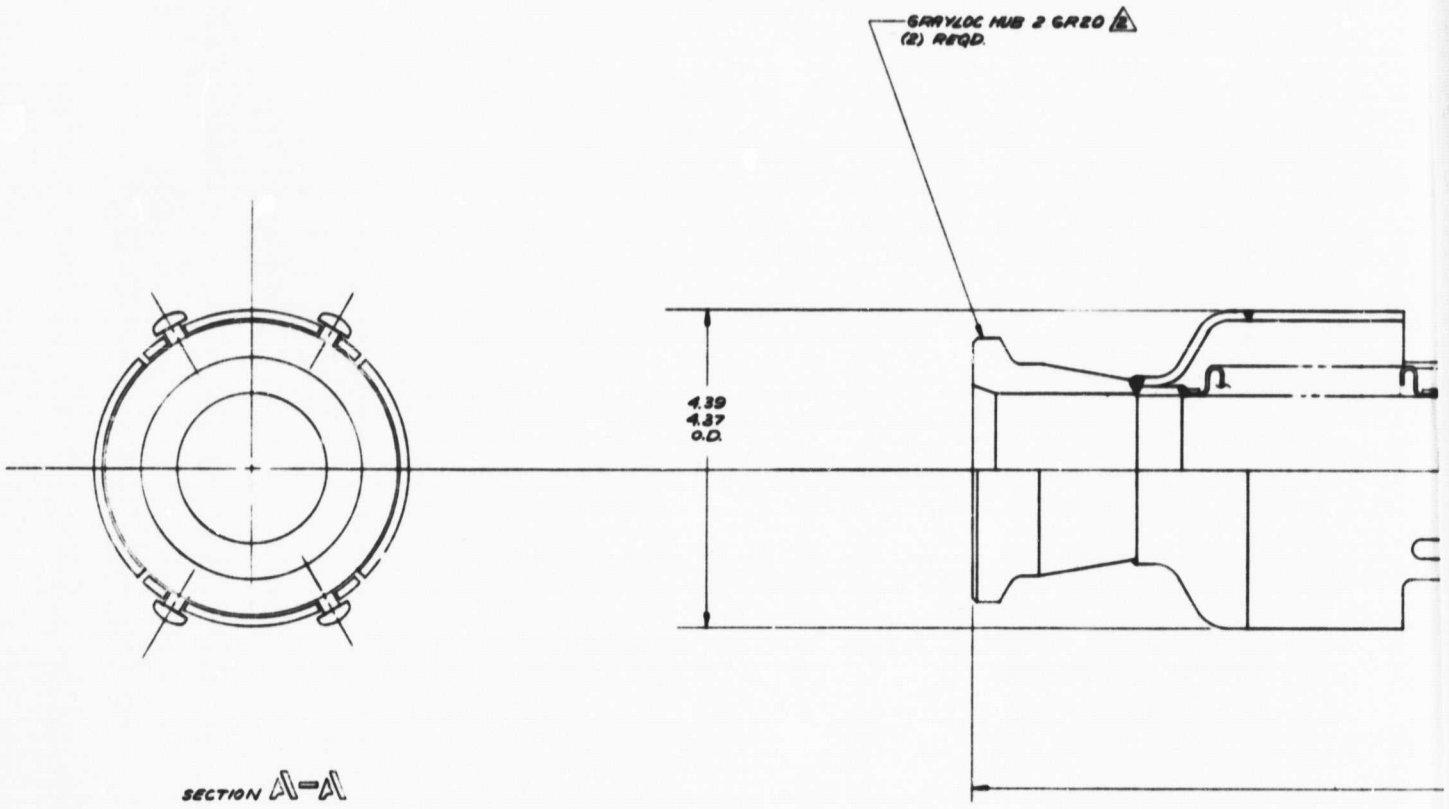
OPERATING PRESSURE	500 PSIG
PROOF PRESSURE	1000 PSIG
BURST PRESSURE	2000 PSIG
OPERATING MEDIA	HYDRAULIC FLUID

DRAWING NO.	6010473
LAYOUT	SOLAR
SCALE	FULL
DESIGN FEATURES	PRELIMINARY DESIGN
	COMPENSATOR 4.00 DIA
SIZE	A
	4135

9-38-1



SECTION B-B
SLOT TYP (8) PLACES

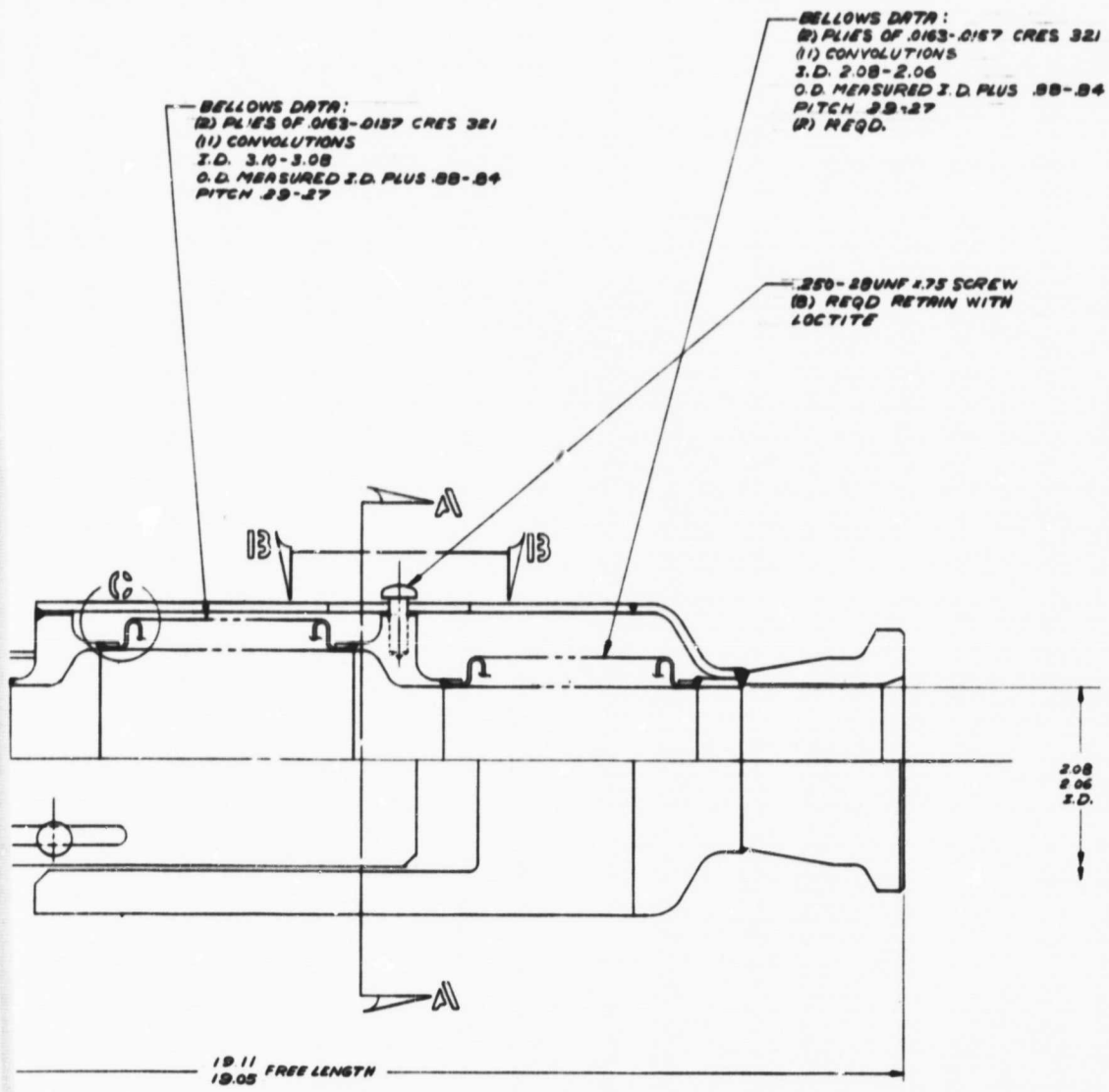


SECTION A-A



AFTER SEAM WELDING TRIM BACK INTO EDGE OF FUSED ZONE TO PERMIT SUBSEQUENT FUSION WELDING

VIEW C

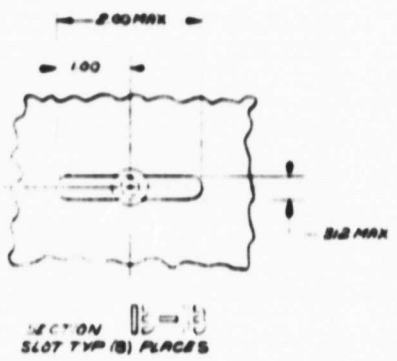


NOTES:
1. MATERIAL - 316 L CRES PER AMS 5507 UNLESS OTHERWISE SPECIFIED
2. MAY BE PURCHASED FROM GRAY TOOL CO. P.O. BOX 2291 HOUSTON 1, TEXAS
3. MAXIMUM TRAVEL ±.88
4. OPERATING DATA:

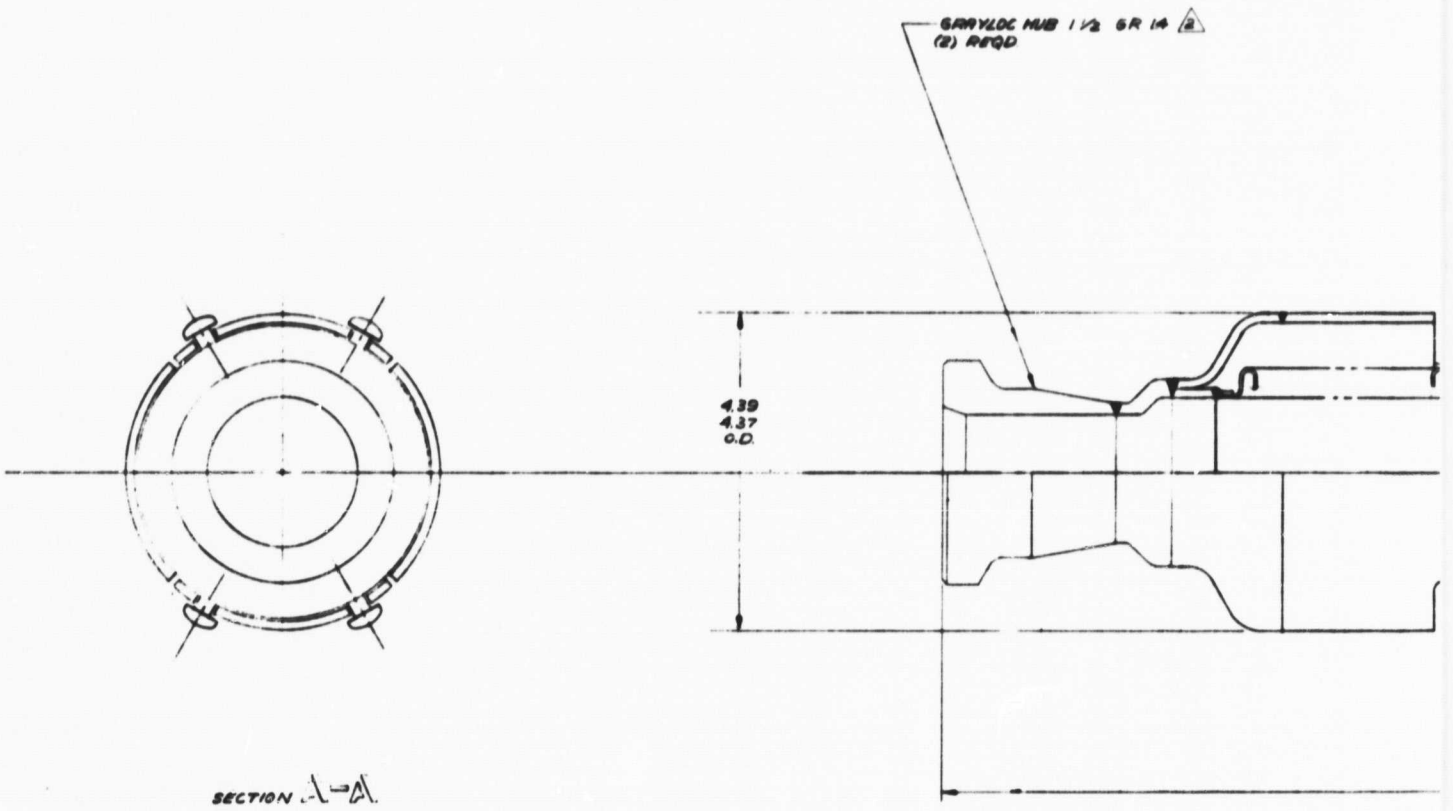
OPERATING PRESSURE	125 PSIG
PROOF PRESSURE	250 PSIG
BURST PRESSURE	500 PSIG
OPERATING MEDIA	GN_2 AND AIR

	6010478
LAYOUT	SOLAR
	SCALE FULL
	PRELIMINARY DESIGN
	COMPENSATOR 2.00 DIA
	SIZE R 41136

A-40-L



SECTION B-B
SLOT TYP (3) PLACES



SECTION A-A



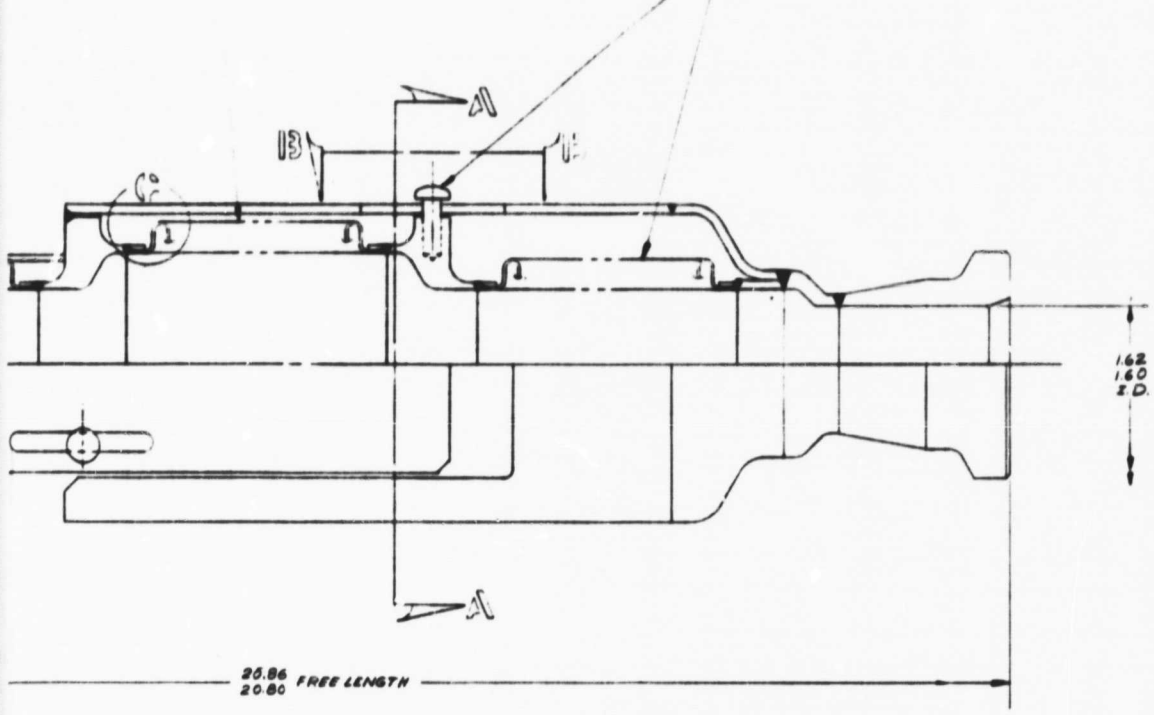
AFTER SEAM WELDING TRIM BACK INTO EDGE OF FUSED ZONE TO PERMIT SUBSEQUENT FUSION WELDING

VIEW C

BELLOWS DATA:
(2) PLIES OF .0163-.0157 CRES 321
(11) CONVOLUTIONS
I.D. 3.10-3.08
O.D. MEASURED I.D. PLUS .88-.84
PITCH .29-.27

BELLOWS DATA:
(2) PLIES OF .0163-.0157 CRES 321
(11) CONVOLUTIONS
I.D. 2.08-2.06
O.D. MEASURED I.D. PLUS .88-.84
PITCH .29-.27
(2) REQD.

.250-28UNF-.25 SCREW
(8) REQD RETAIN WITH LOCTITE

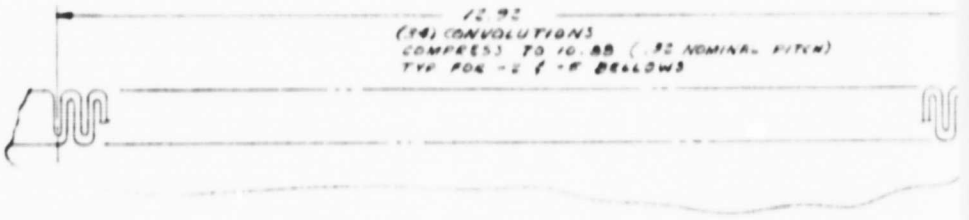


NOTES:
1. MATERIAL - 316 L CRES PER AMS 5507 UNLESS OTHERWISE SPECIFIED
2. MAY BE PURCHASED FROM GRAY TOOL CO. P.O. BOX 2291 HOUSTON 1, TEXAS
3. MAXIMUM TRAVEL ± .88
4. OPERATING DATA:

OPERATING PRESSURE	125 PSIG
PROOF PRESSURE	250 PSIG
BURST PRESSURE	500 PSIG
OPERATING MEDIA	GN_2 AND AIR

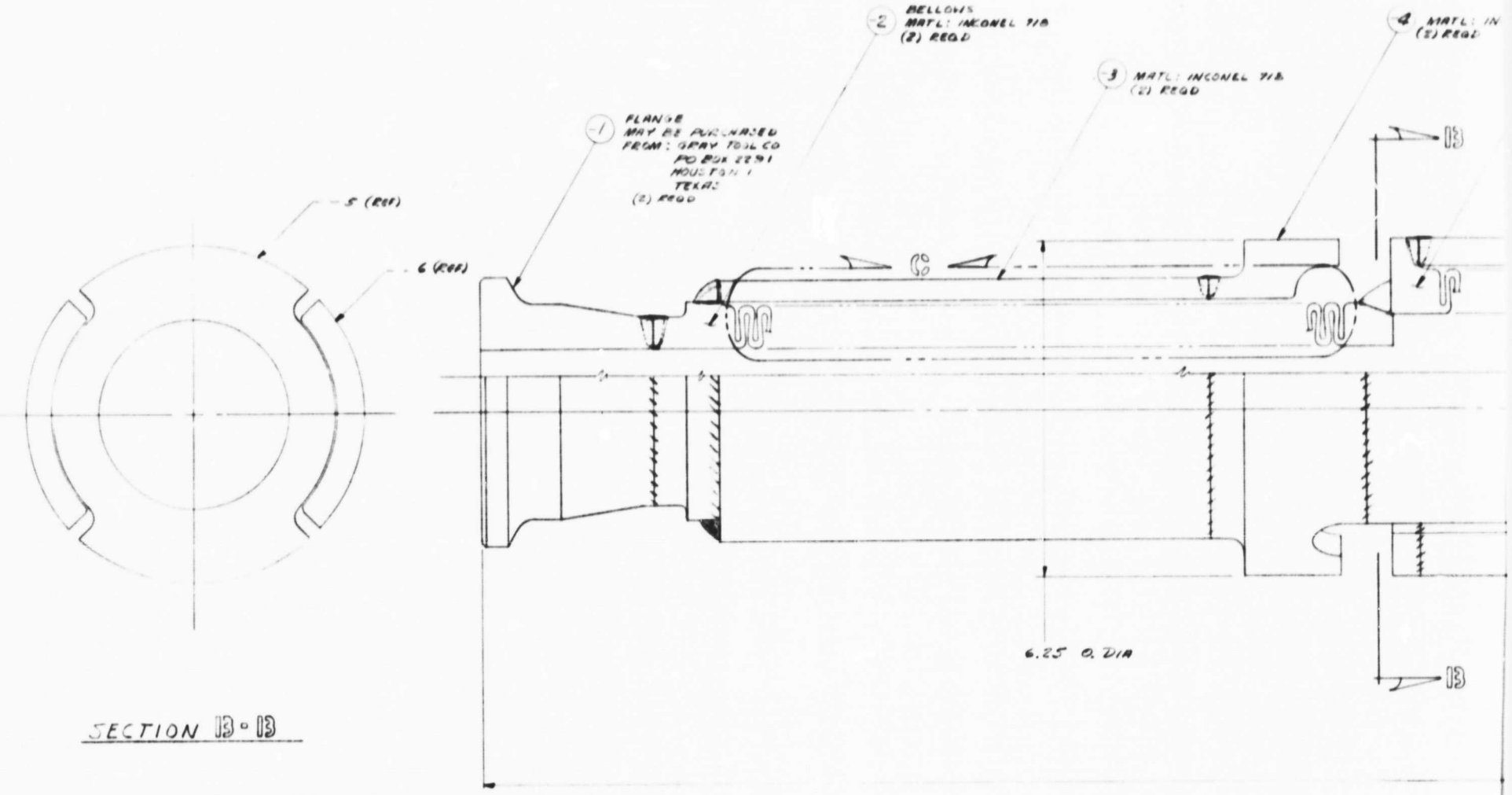
	SC10478
LAYOUT	SOLAR
	SCALE FULL
	PRELIMINARY DESIGN
	COMPENSATOR 2.00 L/A FOR 1.50 DIA LINE
	R 4137

A42-2



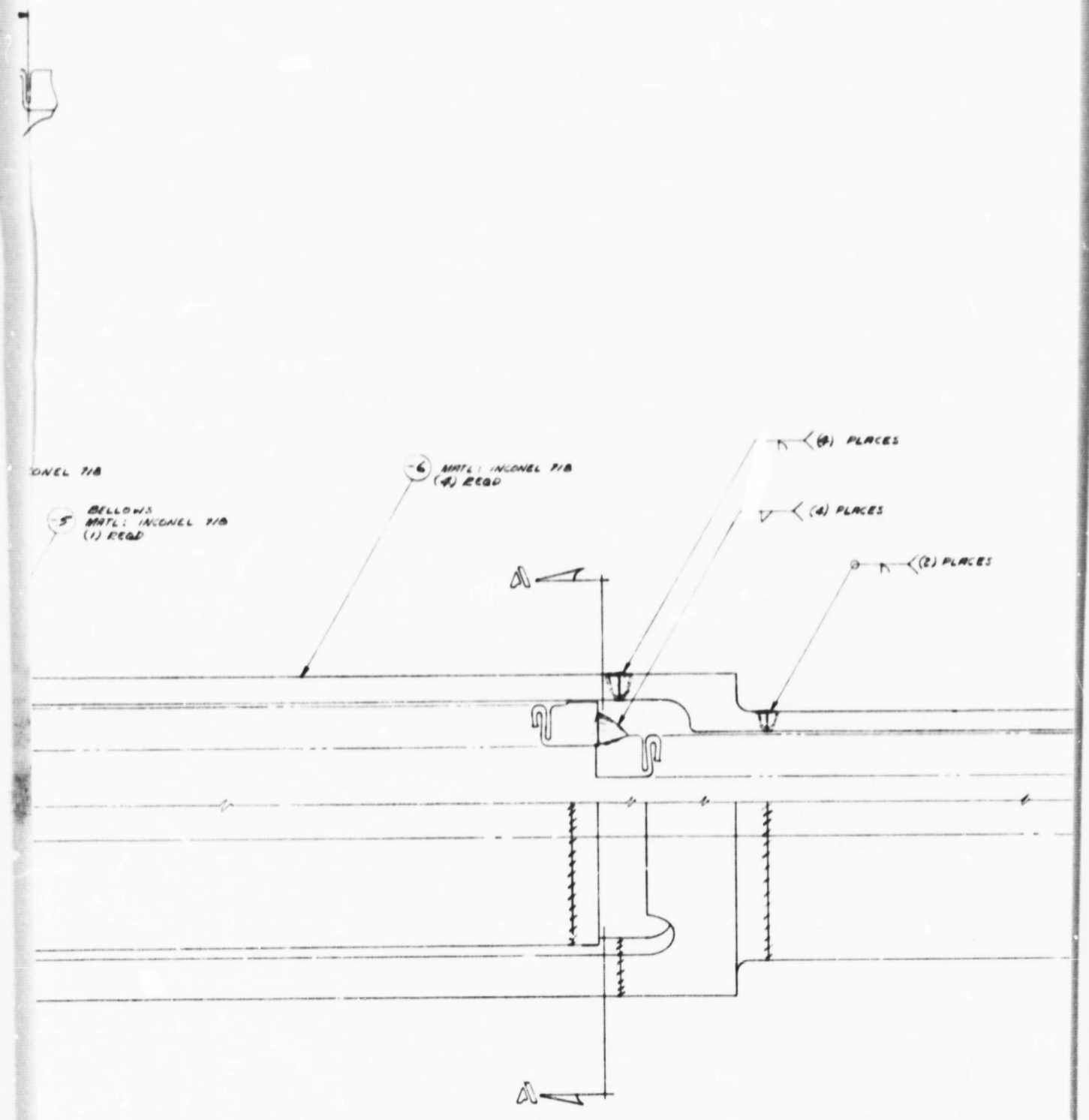
DETAIL C6

SHOWING MACHINED CONFIGURATION OF -2 & -5 BELLOWS



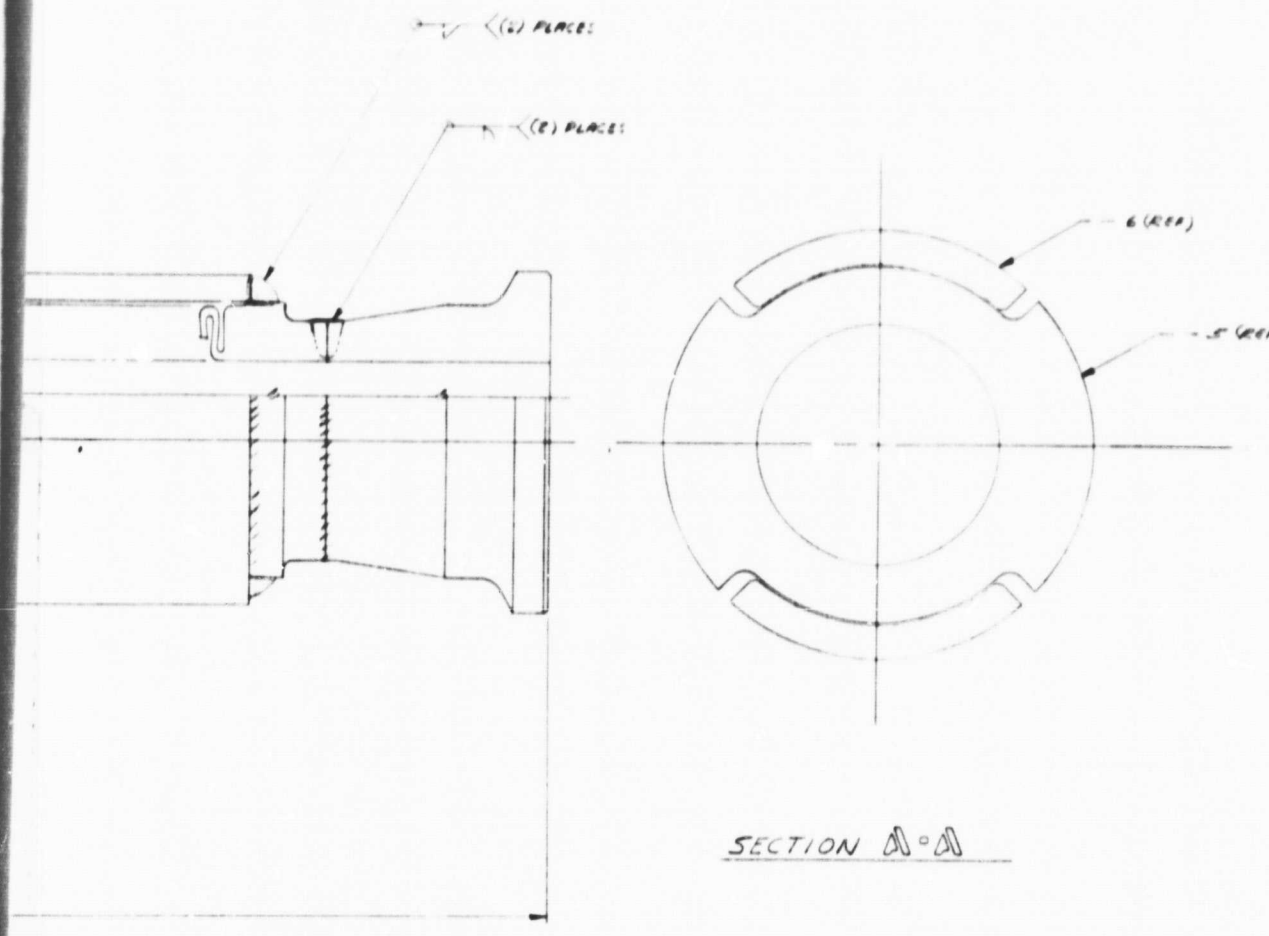
SECTION 13-13

A-43-1


ONEL 718
5 BELLOWS
MATL: INCONEL 718
(1) REQD
6 MATL: INCONEL 718
(4) REQD
(4) PLACES
(4) PLACES
(2) PLACES
A
A

70.0

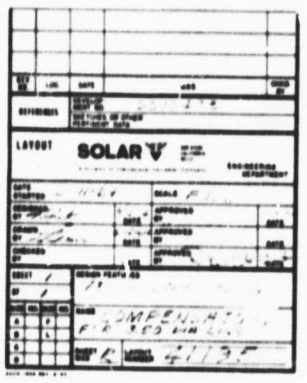
A-44-2

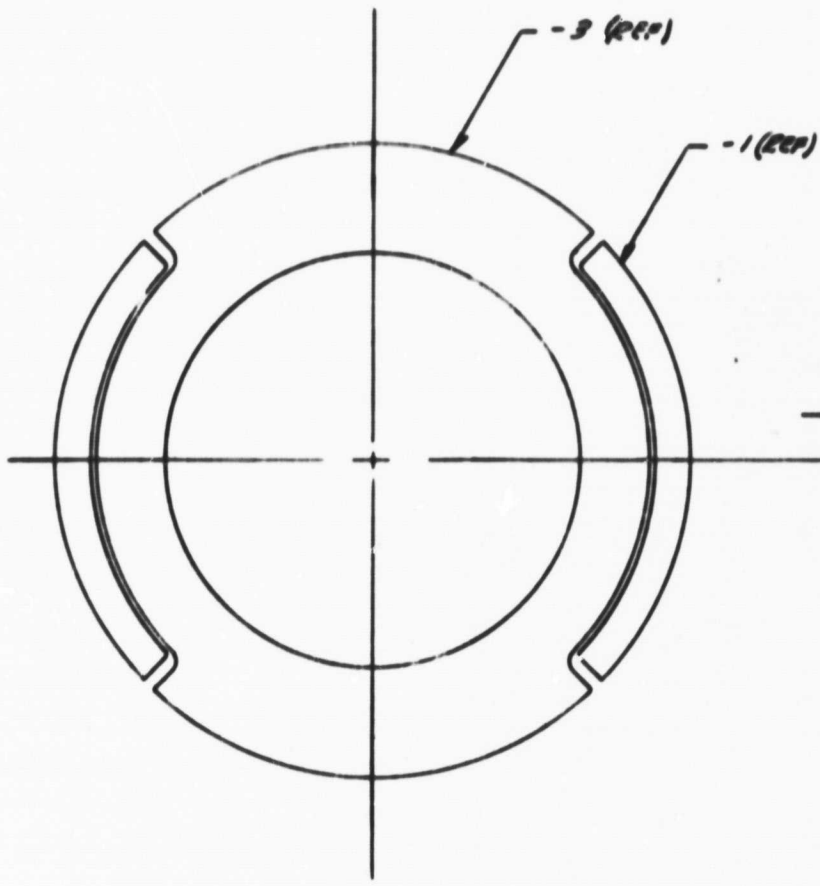


SECTION A-A

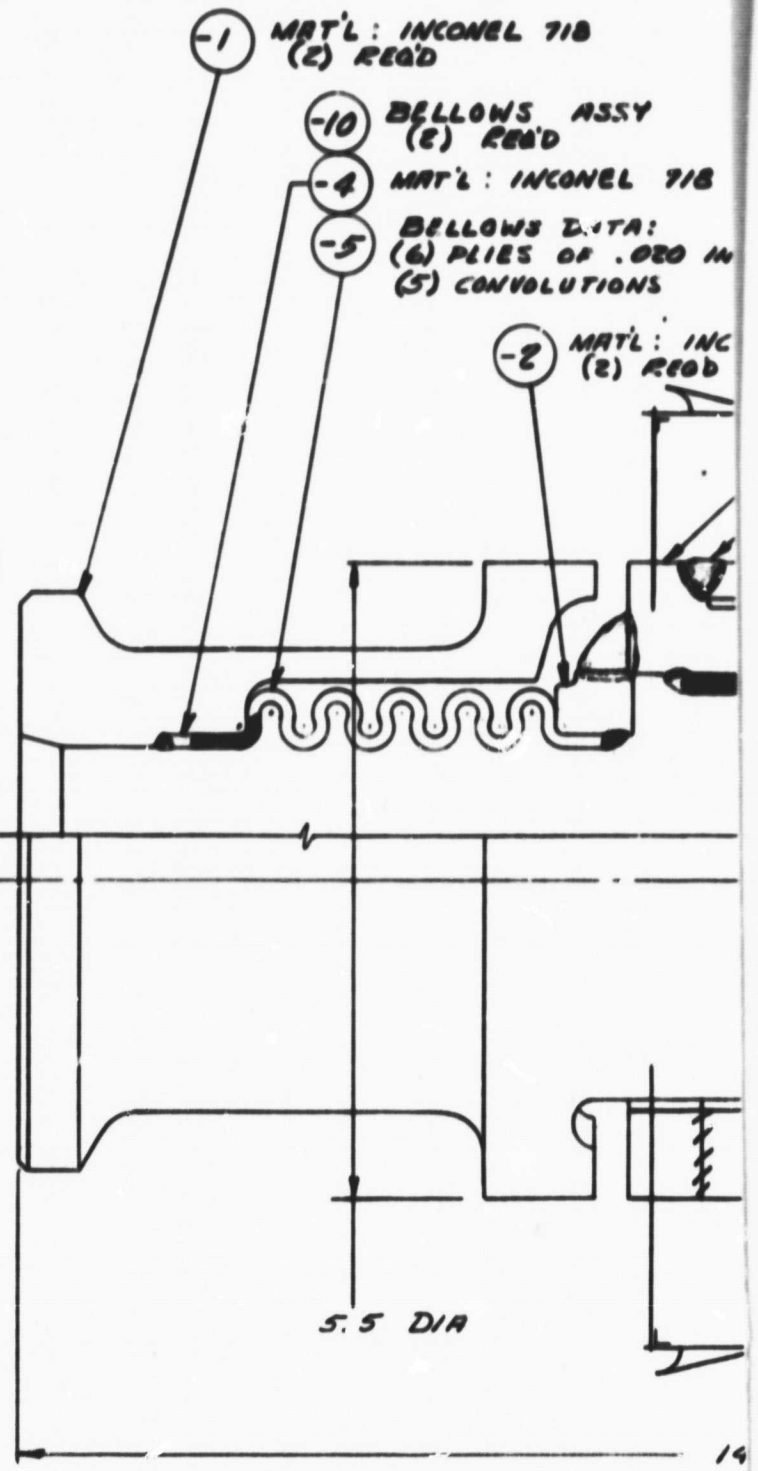
NOTES:

1. AXIAL DEFLECTION DESIGNED FOR ± .90 MIN TRAVEL
2. OPERATING DATA:
   - OPERATING PRESSURE : 6000 PSIG
   - PROOF PRESSURE : 12,000 PSIG
   - BURST PRESSURE : 24,000 PSIG
   - OPERATING TEMPERATURE : + 100° F
   - OPERATING MEDIA : GN_2, GH_2 AND N_2O





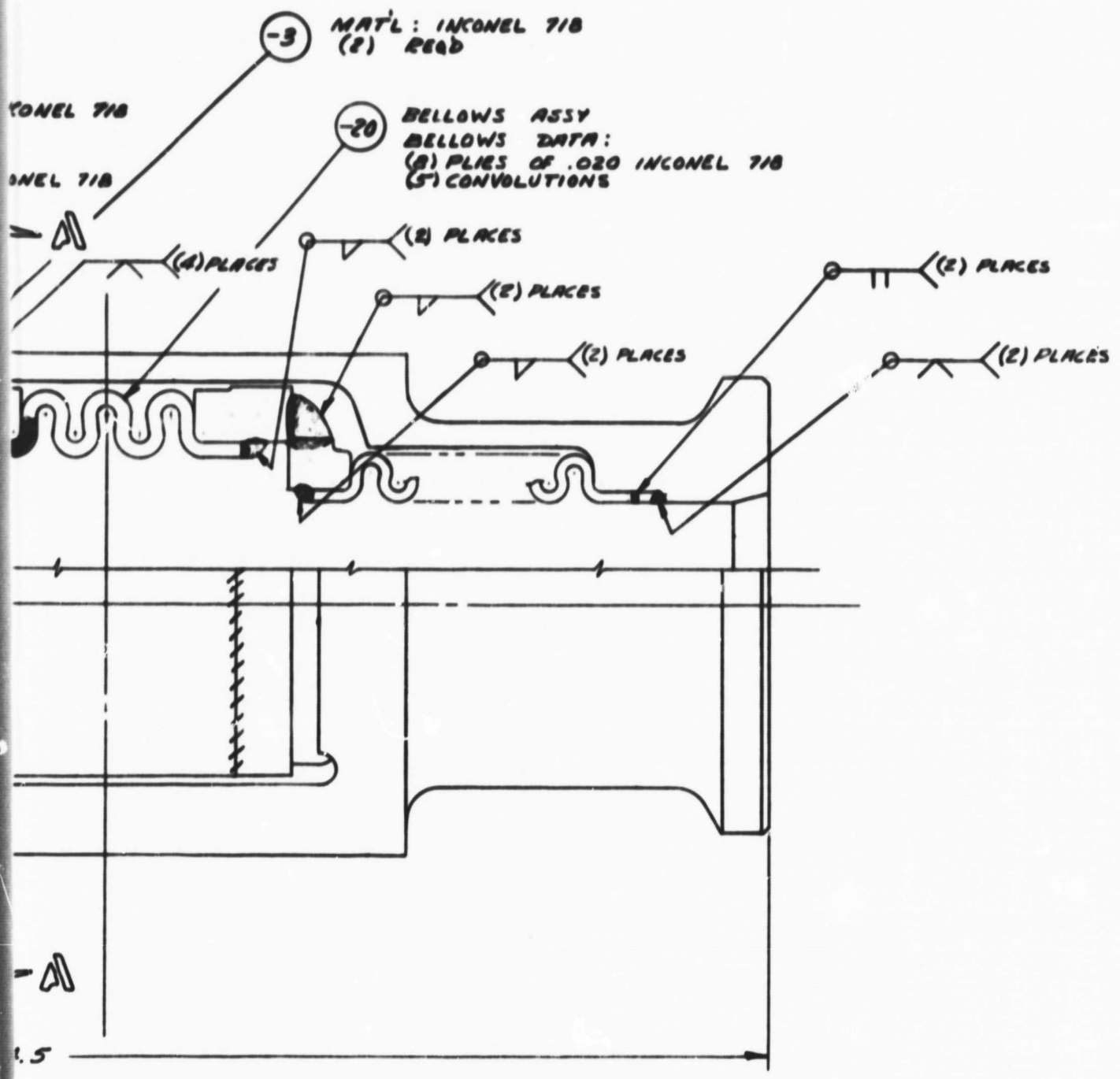
SECTION A-A



NOTES: UNLESS OTHERWISE

1	DIMENSIONS LOCATING T ARE BASIC
2	DO NOT SCALE DRAWING GIVEN
3	DRAWING INTERPRETATIO AND SOLAR SPEC 9-4, SE
4	AXIAL DEFLECTION DESIG TRAVEL
5	THIS ASSEMBLY HAS BEEN DE OPERATING PRESSURE : PROOF PRESSURE : BURST PRESSURE : OPERATING TEMP : OPERATING MEDIA : G H

A-46-1



SPECIFIED
THE TRUE POSITION

. WORK TO DIMENSIONS

N PER MIL-D-70327
CT

VED FOR ±.25 MIN

SIGNED FOR THE FOLLOWING:
6,000 PSIG
12,000 PSIG
24,000 PSIG
-65°F TO +160°F
GASEOUS NITROGEN, GASEOUS
HYDROGEN & HELIUM

QTY REQD	QTY REQD	PART NUMBER	DESCRIPTION	MATERIAL	SPECIFICATION	UNIT WT	
		← ASSY NO.	LIST OF MATERIAL (PARTS LIST)				

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APPROVED BY
GOVT CONTRACT NO.
PROJECT
DESIGN R. Ruiz 7/21/64
CHECK
DRAFT L. Brown 6/26/64
DESIGN ACTIVITY
CUSTOMER

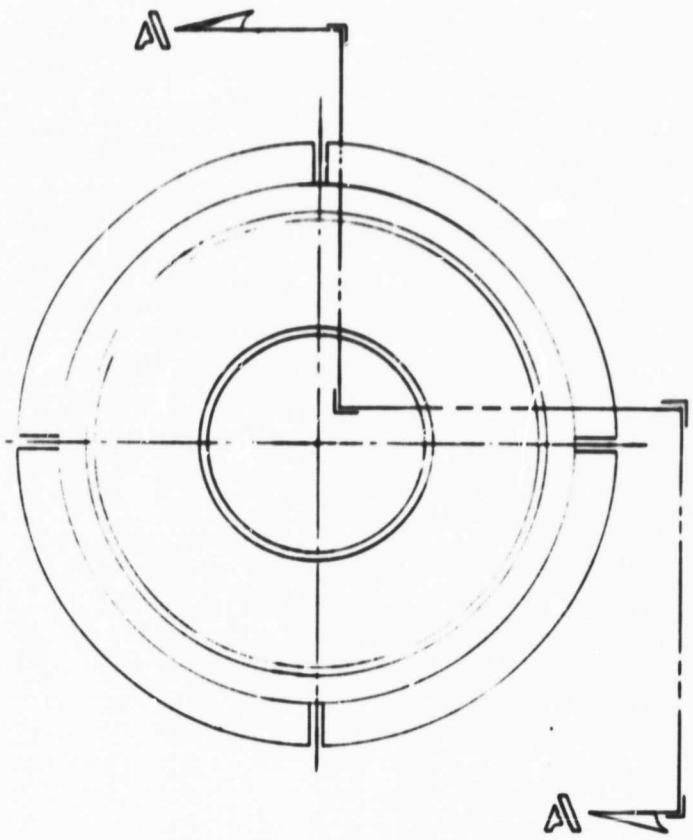
SOLAR
A Division of International Harvester Company

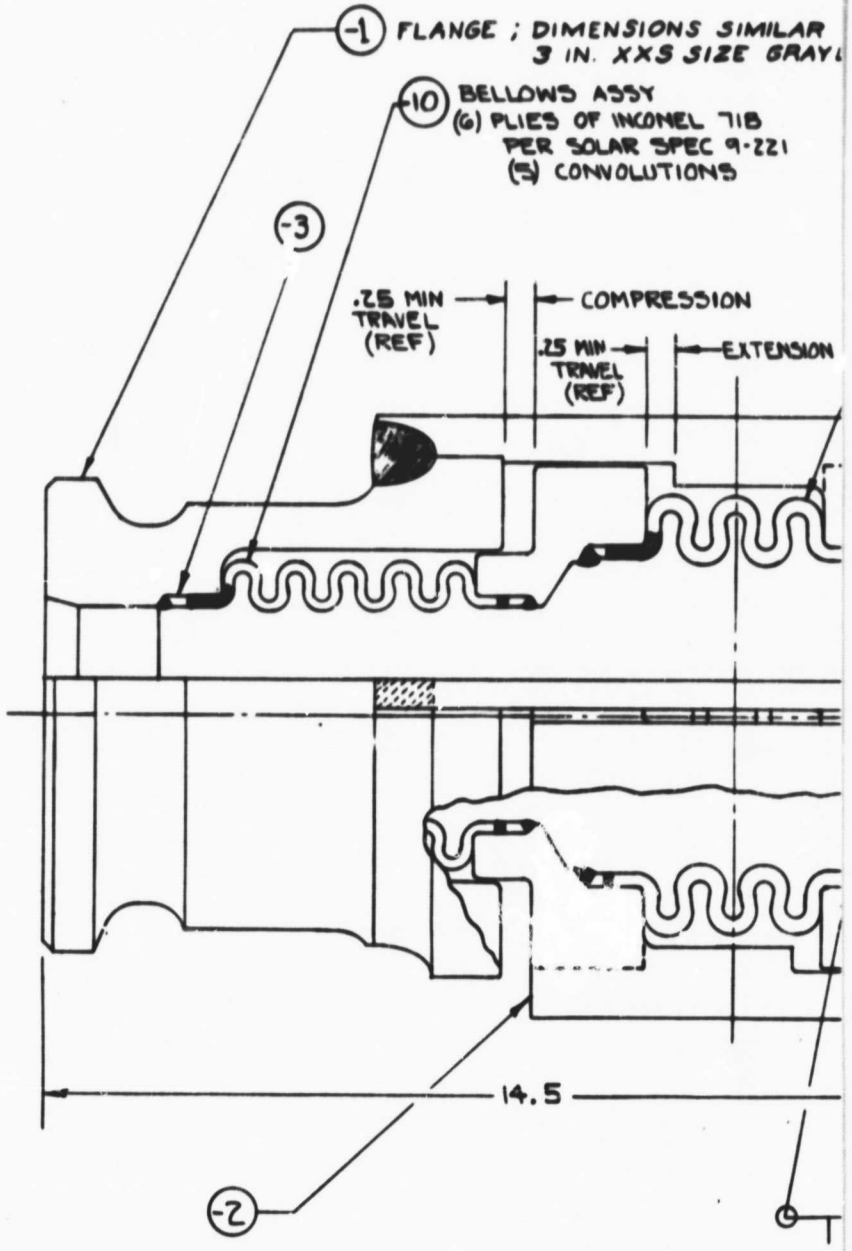
DRAWING TITLE
PRELIMINARY DESIGN
COMPENSATOR
6000 PSI, 2.30 I. DIA LINE

SIZE	CODE IDENT NO.	DRAWING NO.
D	55820	41333

SCALE FULL | TOTAL WT. | SHEET 1 OF 1

A-47-2



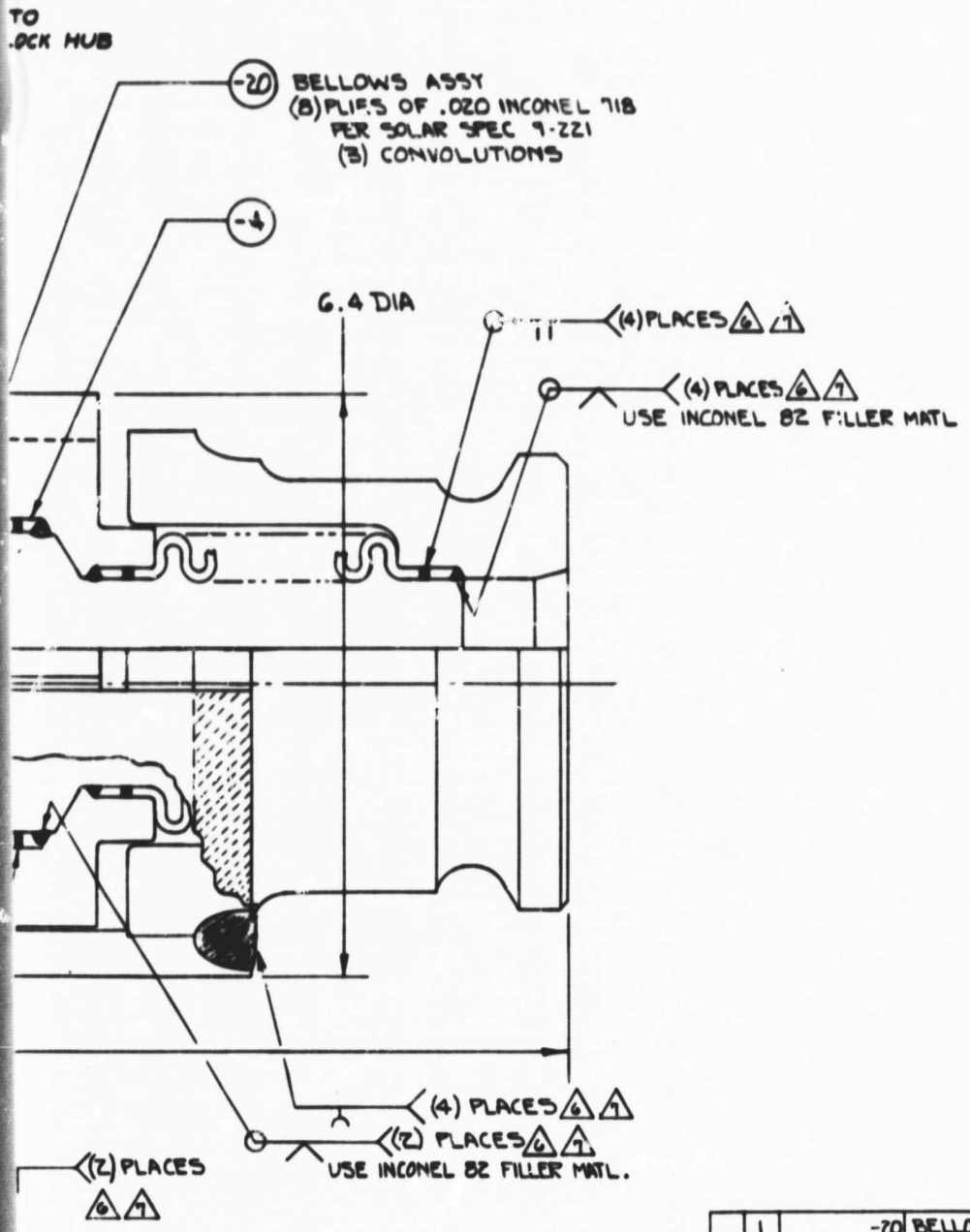


SECT. A A

FUSION WELDING PER ABMA 10509308 CLASS II
DYE PENETRANT INSPECT PER MIL-I-6866

NOTE
1 D
A
2 D
G
3 D
A
4 A
5 TH

A-48-1



S: UNLESS OTHERWISE SPECIFIED

IMENSIONS LOCATING THE TRUE POSITION
RE BASIC

O NOT SCALE DRAWING. WORK TO DIMENSIONS
IVEN

RAWING INTERPRETATION PER MIL-D-70327
ND SOLAR SPEC 9-4, SECT

IAL DEFLECTION DESIGNED FOR ± .25 MIN
TRAVEL

ASSEMBLY HAS BEEN DESIGNED FOR THE FOLLOWING:

OPERATING PRESSURE	: 6000 PSIG
PROOF PRESSURE	: 12 000 PSIG
BURST PRESSURE	: 24 000 PSIG
OPERATING TEMP	: -65°F TO +160°F
OPERATING MEDIA	: GASEOUS NITROGEN GASEOUS HYDROGEN & HELIUM

QTY REQD	QTY REQD	PART NUMBER	DESCRIPTION	MATERIAL	SPECIFICATION	UNIT WT	
	1	-20	BELLOWS	ASSY OF			
	2	-10	BELLOWS	ASSY OF			
	2	-4	WELD END	INCONEL 718 BAR	SOLAR SPEC 9-221		
	4	-3	WELD END	INCONEL 718 BAR	SOLAR SPEC 9-221		
	2	-2	SPIDER	TYPE 316L BAR	AMS-5653		
	2	-1	FLANGE	TYPE 316L BAR	AMS-5653		
	-0	← ASSY NO.	LIST OF MATERIAL (PARTS LIST)				

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APPROVED BY
GOVT CONTRACT NO.
PROJECT
DESIGN
CHECK
DRAFT M. Young 7-22-64
DESIGN ACTIVITY
CUSTOMER

SOLAR
A Division of International Harvester Company

DRAWING TITLE
PRELIMINARY DESIGN
COMPENSATOR
6000 PSI 2.30 I.DIA LINE

SIZE	CODE IDENT NO.	DRAWING NO.
D	55820	41658

SCALE FULL | TOTAL WT. | SHEET 1 OF 1

A-49-2

APPENDIX B

STRESS ANALYSIS

APPENDIX B

SUMMARY OF STRESS ANALYSIS FOR NASA HIGH-PRESSURE CONNECTOR

A preliminary analysis was performed to determine the capability of the high-pressure connector (P/N 41314) to meet loading conditions specified in Exhibit I, Section D, of Contract NAS 10-1266. The major requirements specified were:

- 10,000 psig working pressure
- 15,000 psig proof pressure
- 40,000 psig burst pressure
- Maximum misalignment and operational movement delineated by 12-inch cube.
- Minimum of 100,000 cycles to maximum working pressure at -65 to +150 F.

METHODS OF ANALYSIS

Bellows

The analysis of bellows was largely based on empirically derived or verified formulas. Solar has established working equations for the evaluation of bellows designs, which have proven reliable in many diverse applications. All calculated spring rates, stresses, distortion (squirm) pressures, etc. are approximate and individual bellows, when tested, will exhibit different values depending on how well they match the mathematical model.

Gimbal

The gimbal pins were analyzed as pure cantilever beams uniformly loaded by the ring. This analysis should be conservative since slight deformation will result in peaking of the load and consequent reduction of maximum bending moment.

The gimbal ring was treated as a ring loaded equally by the four pins in the transverse direction through the centroid of the ring cross section. This treatment is not strictly true due to slight deformations of pins, etc., which tend to twist the ring about the centroid. This effect is considered negligible in this design, however.

The gimbal flange was considered a simple beam with a 90-degree section effectively resisting the bending moment transferred from the pin. This approach has been found acceptable in previous gimbal designs which require a redistribution of local moments into the full circular cross section.

Pipe Wall

Thick wall cylinder formulas were used to determine maximum elastic stresses in the pipe. An equivalent uni-axial stress was determined using the strain energy of distortion criterion of yielding. At the burst pressure condition, the pipe material was permitted to yield, resulting in average stresses which were combined to give an equivalent stress using the same failure criterion.

Vibration Analysis

A rough estimate was made of the lowest natural flexural frequency of the connector assembly in the transverse direction. The Kellogg method of effective masses was used. The frequency found should be on the low side due to the assumtions made.

Flexibility Analysis

The end moments and reactions of the connector assembly were crudely estimated on the following basis:

1. The weight of the pipe itself probably causes the maximum vertical reactions to be encountered. These reactions were found by a simple static analysis at 1 g.

2. The maximum end moments are probably caused by the misalignment motion. These moments were calculated using the angular spring rate of the bellows in both the pressurized and unpressurized conditions.

3. The horizontal reactions were assumed equal to zero to simplify the analysis.

4. Reactions due to flow (change in momentum) were not calculated since the fluid is GN_2 or GH_e.

RESULTS OF ANALYSES

Bellows

Motion of ±6 degrees results in an estimated average life of 2500 cycles using room temperature fatigue data for Inconel 718 bellows formed from sheet. There is no reason to expect inferior fatigue strength in the bellows machined from bar.

Pressurization to 10,000 psi results in an estimated cycle life of 150,000 cycles which is 50 percent in excess of the required life.

The estimated critical (squirm) pressure is 105,650 psi based on an average spring rate.

Ring

Large margins of safety exist in bending. For torsion, the margins at proof and burst pressure are 1.10 and 0.05, respectively. These values are based on an assumed 85 percent weld efficiency. It is expected that the electron beam welds will be virtually 100 percent efficient and therefore will greatly increase the margins.

The margin of safety (MS) for the ring due to 10^5 cycles to 10,000 psi is estimated to be 2.84 in bending and 1.35 in torsion. The latter value is based on the same 85 percent weld efficiency and using test data from axial tension fatigue tests (R = 0.05) multiplied by a factor of 0.6 to obtain torsion allowable.

Pin

The pin MS in bending is 2.09 at proof and 0.55 at burst. Shear stresses and deflections are small. MS at 10^5 cycles of 10,000 psi is 2.48.

Flange

The flange is shown to require at least 102 degrees effective at the burst condition to transmit the pin bending. MS at 10^5 cycles of 10,000 psi is 0.74.

Pipe

The margins at proof and burst are:

	away from weld		at weld	
straight	1.06	0.03	0.75	-0.02
bend	0.785	-0.015	-	-
	proof	burst	proof	burst

MS at 10^5 cycles of 10,000 psi is 1.32 in the tube away from the weld. Assuming the same 85 percent weld efficiency in fatigue gives an MS at the weld of 0.97. At the bend, the MS at 10^5 cycles is 0.92.

Vibration

Since the estimated natural frequency is 5.22 cps, which is well above the exciting frequency of 3 cps, no resonance should occur and the maximum vibratory stresses are due to less than 0.1 g loading and can therefore be ignored.

End Moment and Reactions

The end moments are as follows:

ρ = 0	3,000 in.-lb/deg
ρ = 10,000 psig	5,800 in.-lb/deg

The end reactions due to static 1 g loading are assumed vertical and are :

lower	1150 lb
upper	170 lb
Total weight	1320 lb

CONCLUSIONS

The most marginal area is the pipe wall itself in the burst pressure test condition where the margin is virtually zero. It is recommended, however, that bellows tests be undertaken as soon as possible to verify the pressure cycle life and motion cycle life predictions since they are based on test data obtained with thin sheet metal formed bellows.

R.T. CURVE IS BASED ON TEST DATA
OTHER CURVES ARE OBTAINED BY PROPORTION
FROM METHOD DEVELOPED BY S.S. MANSON &
M.H. HIRSCHBERG. 10TH AFMAT MATERIALS
RESEARCH CONF., SAGAMORE. AUG 13-16, 1963.

260
220
180
140
100
60

CALCULATED BENDING STRESS KPSI

R.T.
-320°F
-423°F

BELLOWS
REVERSED BENDING STRESS - CYCLE LIFE
ALLOWABLES FOR INC 718 BELLOWS.

-423°F
-320°F
R.T.

CYCLES TO FAILURE

10 10^2 10^3 10^4 10^5 10^6

Ref. 68

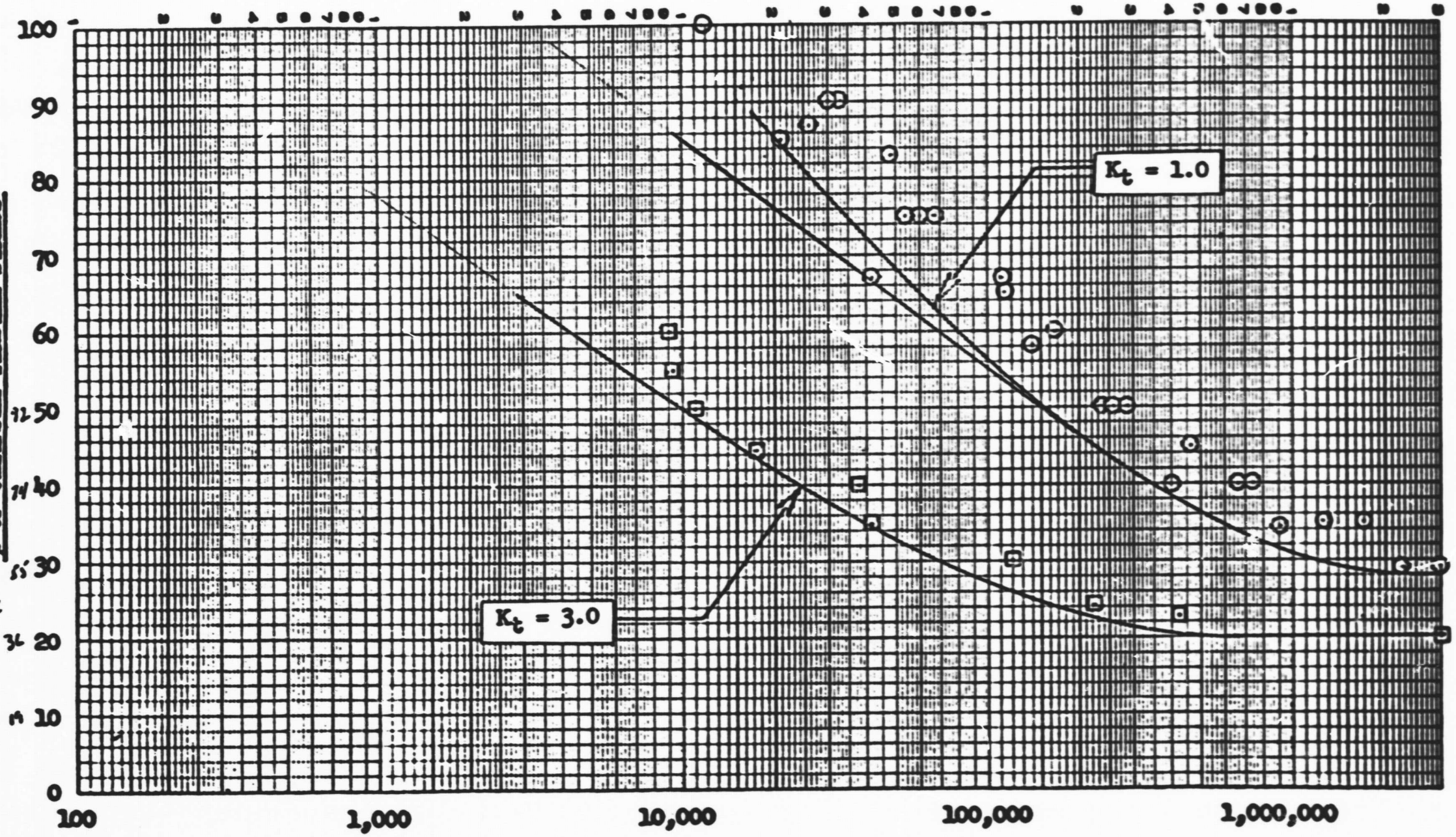
S-N CURVE

MATERIAL Inconel 718 F_{tu} = 184,700 PSI

TYPE OF LOADING Axial Tension

R FACTOR .05

TYPE OF CONCENTRATION Notched



PERCENT OF ULTIMATE TENSILE STRESS

NUMBER OF CYCLES TO FAILURE

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JOB NO. 6010438

SUBJECT: NASA HI-PRESSURE BELLOWS DEVELOPMENT

BY: O. EDWARDS

STRESS ANALYSIS

1. BELLOWS (P/N 41310)

REQMNTS: EXHIBIT I CONTRACT NAS10-1266

PRESSURE: 10,000 psig WORKING
15,000 psig PROOF
40,000 psig BURST

MOTION: 12 IN. CUBE SHOWN ON 41389 (MOTION ZERO)

CYCLIC: 100,000 CYCLES, ZERO TO 10,000 psig (-65°F MIN.)

TESTS: 1) CYCLE FROM 7,500 psig TO 15,000 psig 500 TIMES. ONE MIN. EA. MINIMUM

2) VENT FROM 15,000 psig TO AMBIENT IN LESS THAN 0.1 SEC. 50 TIMES.

3) SURGE 50 TIMES TO 95% MIN. BURST OF CONNECTOR MAT'L. ATMOS. TO PEAK PRESS. IN LESS THAN 0.1 SEC.

4) 4 CYCLES, ΔT +140°F TO −100°F IN LESS THAN 2 MIN. THEN TO 15,000 psig IN 5 MIN. OR LESS.

BELLOWS DATA (P/N 41309-1)

N = 10 EFFECTIVE CONVOLUTIONS

n = 1 PLY (MACHINED)

$S = \frac{5.415 - 3.505}{2} = .990$ MAX. SPAN

h = .495 IN. MAX. ½ SPAN

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BY: GWB

JOB NO. 6010438

t = .125 IN. NOMINAL (SIDES)

L = 4.655 IN. − 2(.275 + .03)

= 4.045 IN. FREE LENGTH

$a_o = \frac{.050 + .155}{2}$ = .1025 IN. RADIUS OF TORUS

$a_i \approx a_o$ = .1025

t @ OD = .184 IN. = t @ ID

$S_{MIN.}$ = .990 − 2(.184) = .622 IN. MIN. SPAN

$a_{min.}$ = .050 IN. @ INSIDE OF TORUS.

ABOVE DATA ENTERED IN IBM 7090 PROGRAM #204
SEE p.3 FOR RESULTS

---

2. GIMBAL (P/N 41308) ASSY.

RING (P/N 41306)

ID = 8.86 IN.

OD = 8.86 + 2(1.50) = 11.86 IN.

W = 5.99 IN. (WIDTH)

D_H = 3.035 IN. (HOLE DIA.)

$F_S = p\,\pi R_m^2 = p\,\pi (2.247)^2 = 15.9(p)$ LBS. SEPAR.

t = 1.50 IN. RADIAL (THICKNESS)

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JOB NO. 6010438

SUBJECT: _____

BY: G. BOWERS

BELLOWS RESULTS FROM PROG. #204 *

1ST MAX SPAN, MIN. t RUN (LOW SPRING RATE)

SR·S 14180

L/h .207 } S = .445 Km = 1.0

h/H 1.21 }

$$SR = \frac{14180}{.445} = 31{,}800 \text{ LB/IN.} \quad \text{(AXIAL SPRING RATE)}$$

$$\sigma_{b\phi}\left(\frac{S}{K_m}\right) = 7773\left(\frac{1.0}{.445}\right) = 17{,}400 \text{ psi per DEG. BEND.}$$

$\pm 6°$ $\sigma_{b\phi} = 6(17{,}400) = 104{,}500$ psi

$$M_\phi \cdot S = \frac{623.2}{.445} = 1400 \text{ IN-LB/DEG} = \text{ANG. SPRING RATE UN-PRESSURIZED.}$$

$M_p = 2792$ IN-LB/DEG DUE TO 10,000 psig = MOMENT DUE TO INT. PRESS

$$P_{cr} = \frac{2\pi SR}{L} = \frac{2\pi(31{,}800)}{4.045} = 49{,}300 \text{ psi} \quad \text{MIN. CRITICAL PRESSURE.}$$

σ_H · 33,520 psi Hoop STRESS DUE TO PRESS.

WGHT = 7.93 LBS MIN. WEIGHT.

* METHODS OF ANALYSIS DESCRIBED IN INTRODUCTION.

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JOB NO. 6010438

SUBJECT: _____

BY: G. BOWERS

2ND RUN - MAX. SPAN - MAX. t (CREST & ROOT ONLY)

SR·S = 48050

a/h = .207 } S = .445 Km = 1.0 } FOR INFO. ONLY

h/2 = .988 }

$SR = \frac{48050}{.445} = 108{,}000$ LB/IN.

3RD RUN - MIN. SPAN MIN. t (MAX. SPRING RATE)

SR·S = 52,490

a/h = .33 } S = .505 Km = 1.0

h/2 = .794 }

$SR = \frac{52{,}490}{.505} = 104{,}000$ LB/IN., $\sigma_{bf} = \frac{(6)18080}{.505} = 215{,}000$ psi BENDING.

$P_{CR} = 104{,}000 \frac{2\pi}{4.045} = 162{,}000$ psi MAX.

$\sigma_{bf} = 128{,}900$ psi (RADIAL BEND. DUE TO P.) $\sigma_H = 46{,}100$ psi Hoop.

4TH RUN - MIN. SPAN, MIN. t, MIN. a

a/h = .161 } S = .42 Km = 1.0 } REF. ONLY.

h/H = .794 }

$SR = \frac{52{,}490}{.42} = 125{,}000$ LB/IN.

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JOB NO. 6010438

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BY: GWB

BELLOWS MARGINS OF SAFETY

$\sigma_{b\phi}$ = STRESS DUE TO ANGULATION OF ±6°

$$= \frac{104{,}500 + 215{,}000}{2} = 160{,}000 \text{ psi}$$

AN AVERAGE VALUE TAKEN MIN. TO MAX. SPAN.

FOR INCO 718 at R.T. USING BELLOWS ALLOWABLES (Fig. 1)

LIFE AT 160,000 psi = 2500 CYCLES AVERAGE

σ_{bp} = RADIAL BEND. STRESS DUE TO PRESS.

= 128,900 psi @ 10,000 psi (WORKING)

= 193,500 psi @ 15,000 psi (PROOF)

= 516,000 psi @ 40,000 psi (BURST)

FOR INCO 718 @ R.T. USING S-N CURVE (Fig. 1)

LIFE AT $\pm \frac{128{,}900}{2} = \pm 64{,}450$ psi

150,000 cycles

LIFE AT $\pm \frac{193{,}500}{2} = 96{,}750$ psi

40,000 cycles.

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BY: GWB

JOB NO. 6010438

$$M.S._{PROOF} = \frac{135{,}000}{193{,}500} - 1 = \underline{\underline{-.3}}$$

THIS IS A FICTICIOUS M.S. SINCE BELLOWS IS ACTUALLY FORMED & PERMANENTLY SET DURING MANUFACTURE AT 35,000 PSI (NOTE 4 ON DW. 413.10)

σ_H = HOOP STRESS DUE TO PRESS. (USING AVERAGE VALUE)

$$= \frac{33{,}520 + 46{,}100}{2} = 39{,}810 \text{ psi @ WORKING STRESS}$$

$\sigma_H = 1.5\,(39{,}810) = 59{,}800$ psi

$= 4.0\,(39{,}810) = 159{,}000$ psi

ALL M.S. ARE OBVIOUSLY LARGE.

CRITICAL PRESSURE (SQUIRM)

$$P_{CR} = \frac{2\pi}{L}(SR) = \frac{2\pi}{4.045}(SR)$$

$$Avg.\ P_{CR} = \frac{2\pi}{4.045}\left(\frac{31{,}800 + 104{,}000}{2}\right) = 105{,}650 \text{ psi}$$

AT BURST $\dfrac{P_{CR}}{P_{BURST}} = \dfrac{105{,}650}{40{,}000} = \underline{\underline{2.64}}$

ALL BELLOWS STRESSES ARE CONSIDERED ACCEPTABLE

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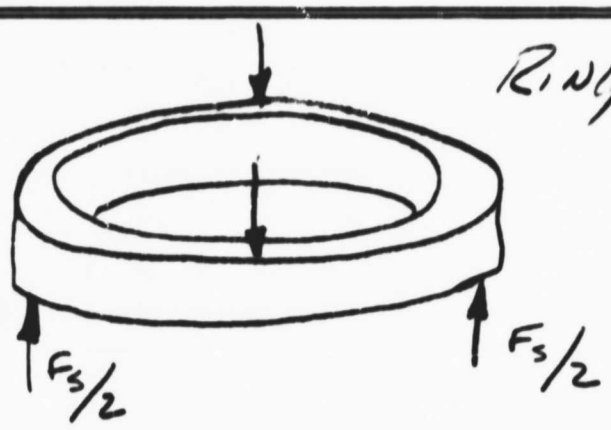
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RING ANALYSIS PER JPL TECHNICAL REPORT No. 32-178, 1961 "SOLUTION OF A CIRCULAR RING STRUCTURAL PROBLEM" H.D. McGINNESS.

M = MAX. MOM. = $F_s R (.250)$ @ PIN HOLES

T = MAX. TORQUE = $F_s R (.1035)$ @ 45° TO PIN HOLE ℄

Y = MAX. DEFL. = $\frac{FR^3}{16}\left[\frac{4.566}{2EI} + \frac{1.133}{2GK}\right]$ @ 45° TO PIN HOLE ℄

$R = \frac{8.86 + 11.86}{4} = 5.18$ IN. (MEAN RADIUS)

@ WORKING PRESS. (10,000 PSIG)

$M = \frac{15.9 (10^4)(5.18)}{4} = 206{,}000$ IN-LB.

$\sigma_b = \frac{Mc}{I} = \frac{206{,}000 \left(\frac{5.99}{2}\right)}{\frac{1}{12}(1.50)\left(\overline{5.99}^3 - \overline{3.035}^3\right)}$

$\sigma_b = \frac{6 (206{,}000)(5.99)}{1.5 (187.2)} = 26{,}350$ PSI WORKING

$\sigma_b = 26{,}350 (1.5) = 39{,}400$ PSI PROOF

$\sigma_b = 26{,}350 (4.0) = 105{,}000$ PSI BURST

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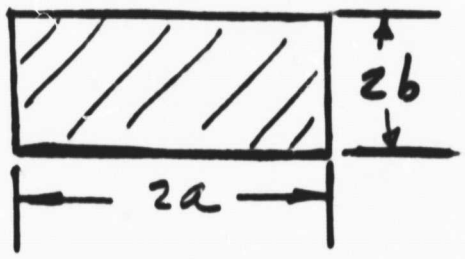
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BY: GWB

$T = 15.9(10^4)(5.18)(.1035) =$ 85,300 IN-LB. TORSION

$\tau = T\left(\frac{3a+1.86}{8a^2b^2}\right)$ +



$\tau = \frac{85,300\left(8.97+1.35\right)}{8\left(2.99\right)^2\left(.750\right)^2}$

$a = 2.99$ IN.

$b = .750$ IN.

$\tau = 21,900$ psi WORK.

$\tau = 32,800$ psi PROOF

$\tau = 87,700$ psi BURST

$y_{max} = \frac{15.9(10^4)(5.18)^3}{16}\left[\frac{4.57}{2(29)10^6 I} + \frac{1.133(2.6)}{2(29)10^6 K}\right]$

$I = \frac{1}{12}(1.50)(5.99)^3 = 26.85$

$K = ab^3\left[\frac{16}{3} - 3.36\frac{b}{a}\left(1-\frac{b^4}{12a^4}\right)\right]$

$= 2.99(.75)^3\left[5.33 - 3.36\left(\frac{.75}{2.99}\right)\left(1-\frac{(.75)^4}{12(2.99)^4}\right)\right]$

$K = 2.99(.422)(4.49) = 5.65$

* ROARK, R. "FORMULAS FOR STRESS AND STRAIN" 3RD EDITION McGRAW-HILL BOOK CO. NY. NY.

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BY: LEWIS

$$y_{max} = 137 \times \frac{10^4}{10^6} [.0293 + .009]$$

$$= 1.37 [.0119] = .0163 \text{ IN. WORKING}$$

$y_{max} = 1.5(.0163) = .0245$ IN. PROOF

$= 4.0(.0163) = .0654$ IN. BURST OK.

MATL. ALLOWABLES

RING MATL IS INCONEL 718 @ $F_{tu} = 180{,}000$ PSI *

CORRESPONDING $F_{ty} = 135{,}000$ PSI

$F_{su} = .6 F_{tu} = 108{,}000$ PSI

$F_{sy} = .6 F_{ty} = 81{,}000$ PSI

MARGINS OF SAFETY

BENDING AT PIN HOLES

$$M.S._{work} = \frac{135{,}000}{26{,}350} - 1 = 4.13$$

$$proof = \frac{135{,}000}{39{,}400} - 1 = 2.43$$

$$BURST = \frac{180{,}000}{105{,}000} - 1 = .715$$

* INTL. NICKEL CO. DATA

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BY: GWB

TORSION @ 45° FROM pin ℄ *

$$M.S.\ WORK = \frac{(.85)81,000}{21,900} - 1 = 2.15$$

$$PROOF = \frac{(.85)81,000}{32,800} - 1 = 1.10$$

$$BURST = \frac{(.85)108,000}{87,700} - 1 = .05$$

* THE ELECTRON BEAM WELD JOINING THE 4 RING QUADRANTS OCCURS HERE, BUT WELD JOINT EFFICIENCY HAS NOT BEEN ESTABLISHED. A VALUE OF 85% IS ASSUMED.

CYCLE LIFE MARGINS

THE LOW CYCLE FATIGUE STRENGTH OF INCO 718 IS GIVEN BY DATA OBTAINED FROM NAA & REPRODUCED ON FIG. 2.

FOR $K_T = 1.0$, AXIAL TENSION

$F_e(10^5) = 101,000$ psi

$F_e(10^4) = 156,000$ psi

$F_e(3\times10^3) = 184,700$ psi = F_{TU}

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BY: GWB

BENDING -

M.S. @ 10^5 CYCLES (WORKING PRESS) = $\frac{101,000}{26,350} - 1 = 2.84$

TORSION

M.S. @ 10^5 CYCLES (W.P.) = $\frac{(.6)(.65)(101,000)}{21,900} - 1 = 1.35$

OTHER M.S. VALUES FOR CYCLIC TESTING ARE OBVIOUSLY LARGER BY INSPECTION.

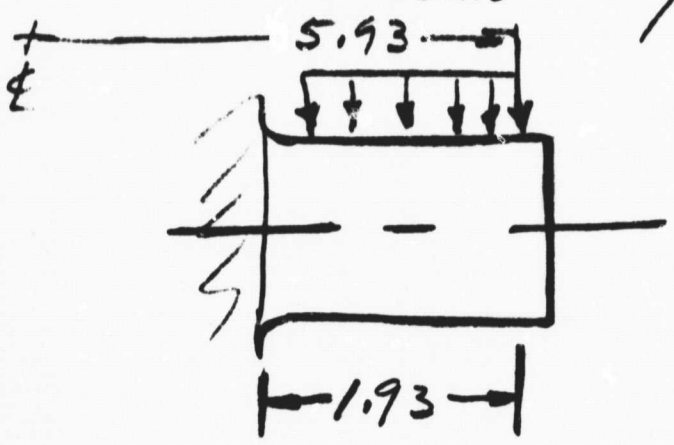
PIN (P/N 41307) ASSUME CANTILEVER

DIA. = 3.005 IN.

LOADED TO = 5.93 IN. FROM ℄ (RING OD)

FIXED = 4.00 IN. "

DIST. LOAD = $F_s/2 = \frac{15.9 \times 10^4}{2} = 8000$ LBS./PIN



$M_{max} = 8 \times 10^4 \frac{(1.93)}{2}$

$= \frac{1535}{2} \times 10^4$ IN-LB

$I = \frac{\pi}{4}(1.50)^4 = 3.96$ IN^4

$\sigma_b = \frac{Mc}{I} = \frac{15.35 \times 10^4 (1.5)}{(2)3.96} = 29,100$ PSI WORKING

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JOB NO. 6010438

SUBJECT: _____

BY: GWB

σ_b = 1.5 (29,100) = 43,700 psi @ proof

= 4.0 (29,100) = 116,200 psi @ burst

$$M.S._{proof} = \frac{135,000}{43,700} - 1 = 2.09$$

$$M.S._{BURST} = \frac{180,000}{116,200} - 1 = .55$$

BEND.

ALL MARGINS ARE ADEQUATE

$$Avg.\ SHEAR\ ON\ PIN = \frac{8000\ LBS}{\pi(1.50)^2} = 1,130\ psi \quad (NEGLIGIBLE)$$

PIN DEFLECTION (BENDING)

$$y = \frac{WL^4}{8EI} = \frac{8000(1.93)^3}{8(29)10^6(3.96)} = .0000622\ IN.\ (NEGLIG.)$$

CYCLE LIFE MARGIN FOR 10^5 CYCLES @ 10,000 PSI IS

$$M.S. = \frac{101,000}{29,100} - 1 = \underline{\underline{2.48}}$$

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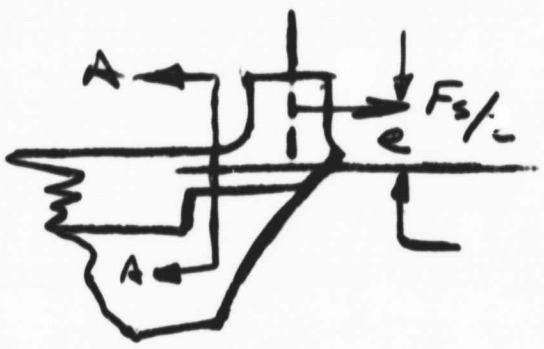
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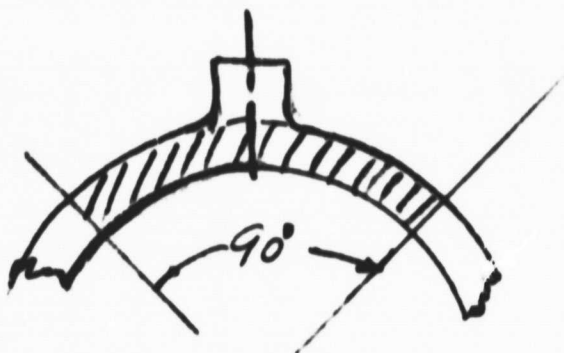
JOB NO. 6010438

SUBJECT: _____

BY: GWR

FLANGE ANALYSIS (P/N 41307)





— - — ℄ +

CHECK BENDING ACROSS A-A

ASSUME 90° SECTION EFFECTIVE

$$\text{MOMENT} = F_s/_2 (e) = 8000 (1.0) = 8000 \text{ IN-LB}$$

$$I = .012 R^3 t = .012 (3.4)^3 (1.0) = .468 \text{ IN}^4$$

$$c = 0.1 R = 0.1 (3.4) = 3.4 \text{ IN.}$$

$$\sigma_b = \frac{Mc}{I} = \frac{8000(3.4)}{.468} = 58{,}000 \text{ psi (WORKING)}$$

$$= 1.5 (58{,}000) = 87{,}000 \text{ psi (PROOF)}$$

$$= 4.0 (58{,}000) = 232{,}000 \text{ psi (BURST)}$$

$$M.S._{PROOF} = \frac{135{,}000}{87{,}000} - 1 = .555$$

$$M.S._{BURST} = \frac{180{,}000}{232{,}000} - 1 = -.223$$

$$M.S._{10^5} = \frac{101{,}000}{58{,}000} - 1$$

$$M.S._{10^5} = 0.74$$

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SUBJECT: _____

BY: GWB

THIS NEGATIVE M.S. MEANS THAT MORE THAN A 90° SECTION IS REQUIRED TO CARRY BENDING LOAD.

FOR A M.S. = 0.0, σ_b = 180,000 psi

but, $\sigma_b = 4(8000)\left(\frac{c}{I}\right)$ $\therefore$ $\frac{I}{c} = \frac{32000}{180,000} = .178$

USING CURVES - $\alpha = \underline{102^\circ}$

THIS IS ONLY 13% GREATER THAN ORIGINALLY ASSUMED $\therefore$ FLANGE IS CONSIDERED ADEQUATE TO HANDLE PIN LOAD.

PIPE STRESSES DUE TO PRESS.

AWAY FROM DISCONTINUITY

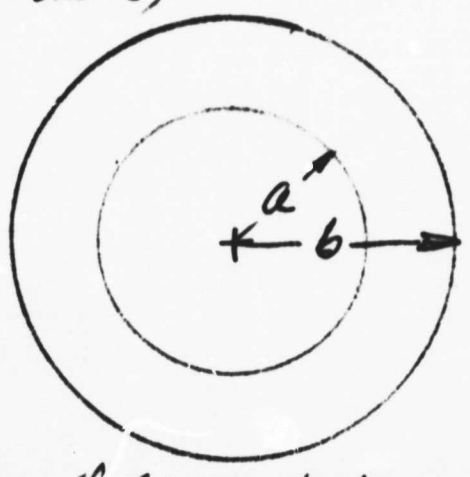
ID = 3.50 IN. NOM., t = .50 NOM. $\frac{R}{t} = \frac{1.75}{.50} = 3.5$

THICK WALL FORMULAS ARE REQ'D. +

$\sigma_m = p\frac{a^2}{b^2-a^2}$ THROUGHOUT (LONGITUDINAL)

MAX. $\sigma_h = p\frac{b^2+a^2}{b^2-a^2}$ @ INSIDE (HOOP)

MAX. $\sigma_r = -p$ @ INSIDE (RADIAL)



+ ROARK, R. "FORMULAS FOR STRESS AND STRAIN" 3RD ED. McGRAW-HILL.

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SUBJECT: _____

BY: GWB

$a = 1.75$ IN., $b = 2.25$ IN.

$a^2 = 3.05$ $\quad b^2 = 5.05$

$a^2+b^2 = 8.10$ $\quad b^2-a^2 = 2.0$

$$\sigma_2 = \sigma_M = p\left(\frac{3.05}{2.0}\right) = 1.525\,p$$

$$\sigma_1 = \sigma_h = p\left(\frac{8.10}{2.0}\right) = 4.05\,p.$$

$$\sigma_3 = \sigma_r = -p$$

$$\sigma_{eq} = \frac{1}{\sqrt{2}}\sqrt{(\sigma_1-\sigma_2)^2+(\sigma_2-\sigma_3)^2+(\sigma_1-\sigma_3)^2}$$

AT INSIDE SURFACE

MAX. STRESSES

	WORK 10,000	PROOF 15,000	BURST 40,000
σ_2	15,250	22,900	61,000
σ_1	40,500	60,800	162,000
σ_3	-10,000	-15,000	-40,000
σ_{eq}	43,600	65,700	175,000

$$M.S._{PROOF} = \frac{135,000}{65,700} - 1 = 1.06$$

$$M.S._{BURST} = \frac{180,000}{175,000} - 1 = .03$$

} PARENT METAL

ASSUMING 85% EFFICIENCY IN THE WELDS GIVES

$M.S._{PROOF} = .75$

$M.S._{BURST} = -.125$

} WELD

THE SLIGHT NEGATIVE MARGIN AT BURST IS ACCEPTABLE SINCE YIELDING OF THE MATERIAL WILL REDISTRIBUTE THE PEAK HOOP STRESS TO AN AVERAGE VALUE OF $\frac{pR_m}{t}$

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& THE PEAK RADIAL STRESS TO A VALUE OF $-\frac{P}{.5}$

THUS

$\sigma_h = \sigma_1 = \frac{P(2)}{.5} = 4P$

$\sigma_m = 1.525P = \sigma_2$

$\sigma_r = -.5P = \sigma_3$

$$\sigma_{eq} = .707P\sqrt{(4-1.525)^2+(4+.5)^2+(1.525+.5)^2}$$

$$= .707P\sqrt{6.1+20.2+4.1} = 3.9P$$

$$= 3.5(40{,}000) = 156{,}000 \text{ psi}$$

$$M.S. = \frac{.85(180{,}000)}{156{,}000} - 1 = \underline{-.02}$$

THIS IS CONSIDERED CLOSE ENOUGH.

$$M.S. @ 10^5 \text{ CYCLES} = \frac{101{,}000}{43{,}600} - 1 = \underline{1.32}$$

@ the weld $M.S. = \frac{(.85)(101{,}000)}{43{,}600} - 1 = \underline{0.97}$

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JOB NO. _____

SUBJECT: _____

BY: GWB

IN THE BENT TUBE (41312) A WALL THICKNESS OF .410 IN. IS PERMITTED AROUND THE BEND ± 30° FROM OUTERMOST PT. ASSUMING THE THINNING OCCURS ON OUTSIDE OF WALL, THE MAX. HOOP STRESS IS FOUND AS FOLLOWS:

$$\sigma_H = P\left(\frac{\overline{1.75}^2 + \overline{2.16}^2}{\overline{2.16}^2 - \overline{1.75}^2}\right) = P\left(\frac{3.05 + 4.65}{4.65 - 3.05}\right) = P\left(\frac{7.70}{1.60}\right)$$

$\sigma_1 = \sigma_H = 4.81\,p = 48{,}100$ psi @ work. press

$= 1.5(4.81p) = 72{,}100$ @ proof

$= 4.0(4.81p) = 192{,}500$ @ burst

$$\sigma_2 = \sigma_M = P\left(\frac{\overline{1.75}^2}{\overline{2.16}^2 - \overline{1.75}^2}\right) = P\left(\frac{3.05}{1.60}\right) = 1.91\,p$$

$\sigma_M = 19{,}100$ psi @ work.

$= 28{,}600$ psi @ proof

$= 76{,}300$ psi @ burst

$\sigma_3 = \sigma_2 = -p = -10{,}000$ psi @ work

$-15{,}000$ psi proof

$-40{,}000$ psi burst

$$\sigma_{eq} = .707\sqrt{(\sigma_1 - \sigma_2)^2 + (\sigma_1 - \sigma_3)^2 + (\sigma_2 - \sigma_3)^2}$$

SEE TABLE p. 17

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SUBJECT: _____

BY: GWB

σ \ P	10,000	15,000	40,000
σ_1	48,100	72,100	192,500
σ_2	19,100	28,600	76,300
σ_3	-10,000	-15,000	-40,000
σ_{eq}	59,250	75,600	201,500

@ INSIDE OF PIPE

$$M.S._{PROOF} = \frac{135,000}{75,600} - 1 = .785$$

$$M.S._{BURST} = \frac{180,000}{201,500} - 1 = -.105$$

AGAIN, YIELDING WILL REDUCE STRESSES TO AVG. VALUES.

$$\sigma_1 = \frac{P R_m}{t} = P\left(\frac{1.955}{.41}\right) = 4.77P$$

$$\sigma_2 = 1.91P$$

$$\sigma_3 = -\frac{P}{2}$$

$$@\ burst,\ \sigma_{eq} = .707P\sqrt{(4.77-1.91)^2 + (4.77+.5)^2 + (1.91+.5)^2}$$

$$= .707P\sqrt{8.15 + 27.7 + 5.8} = .707P\sqrt{41.65}$$

$$= 4.575P = 4.575(40,000) = 183,000\ psi$$

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JOB NO. 6010438

SUBJECT: _____

BY: GWB

USING THE AVERAGE STRESSES GIVES A BURST M.S. OF

$$M.S. = \frac{180,000}{183,000} - 1 = -.015$$

THIS IS CONSIDERED CLOSE ENOUGH.

AT 10^5 CYCLES $M.S. = \frac{101,000}{52,500} - 1 = .92$

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JOB NO. 6010438

SUBJECT: _____

BY: GWB

VIBRATION ANALYSIS

.15 IN. D.A. @ 1-3 cps IN ALL 3 DIRECTIONS

THIS RESULTS IN < 0.1 $\ddot{g}$ ACCELERATION PROVIDED SYSTEM IS NOT IN RESONANCE.

APPROXIMATE NATURAL FREQUENCIES ARE FOUND BY THE METHOD DESCRIBED IN REF. 1

$$f = \alpha \sqrt{\frac{EI}{WL^3}} \qquad \left\{ \begin{matrix} E \text{ psi} & I \text{ in}^4 \\ W \text{ LBS} & L \text{ FT.} \end{matrix} \right\}$$

FOR 2 LEGS @ $\theta = 50\text{-}60°$ $\alpha = 1.27$ (Fig. 9.13)

$$\frac{L_2}{L_1} \approx \frac{3}{4}$$

$E = 29 \times 10^6$ psi

$$I = \frac{\pi}{4}\left(R_o^4 - R_i^4\right) = \frac{\pi}{4}\left(\overline{2.25}^4 - \overline{1.75}^4\right) = 12.7 \text{ in}^4$$

W = TOTAL WEIGHT OF PIPE

$$\text{AREA} = \pi\left(R_o^2 - R_i^2\right) = \pi\left(\overline{2.25}^2 - \overline{1.75}^2\right) = 2\pi \text{ in}^2$$

REF. 1 - "DESIGN OF PIPING SYSTEMS" THE M.W. KELLOGG Co. 2ND EDITION, JOHN WILEY & SONS NY, NY. CHAPT. 9 - VIBRATION: PREVENTION & CONTROL.

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BY: GWB

JOB NO. 6010438

LENGTH = 115.6 + 49.15 + 76.41 (min 50°) + 3(14) = 306 IN.

$$WGHT = .297^{\#/IN^3}(306)^{IN}(6.28)^{IN^2} = 570 \text{ LBS.}$$

THIS IS $\frac{570}{306} = 1.86$ LBS/IN.

GIMBALS ARE SOMEWHAT HEAVIER

GIMBAL WT. = BELLOWS + 2 FLANGES + RING + 2 COUPLINGS

BELLOWS = 10.5 LBS (EST.)

$$RING = .297(6.0)\pi(5.93^2 - 4.43^2)$$

= 86.4 LBS INCLUDING PINS

$$EA.\ FLANGE = .297\pi\left[4.7\left(2.815^2 - 1.75^2\right) + 5.0\left(4^2 - 2.815^2\right)\right]$$

$$= .935\left[22.8 + 40.5\right] = 59.1 \text{ LBS.}$$

EA. COUPLINGS = 30.0 LBS (EST.)

GIMBAL WT. = 10.5 + 86.4 + 2(59.1) + 30.0(2)

= 275 LBS

ASSUMED WT. = 14.0(1.86) = 26 LBS.

INCREASED WT. = 275 − 26 ≈ 250 LBS.

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SUBJECT: _____

BY: GWB

TOTAL WT. = 570 + 3(250) = 1320 LBS.

$$f = 1.27\sqrt{\frac{29\times10^6 \times 12.7}{1320\left(\frac{306}{12}\right)^3}} = 1.27\sqrt{16.93}$$

$f = 5.22$ cps

THIS IS FAIRLY CLOSE TO EXC. FREQ., ALTHOUGH CONSERVATIVE, SINCE TWO OF THE GIMBALS (500 LBS) ARE ADJACENT TO ENDS & A SUPPORT WILL BE USED SOMEWHERE BETWEEN THE ENDS.

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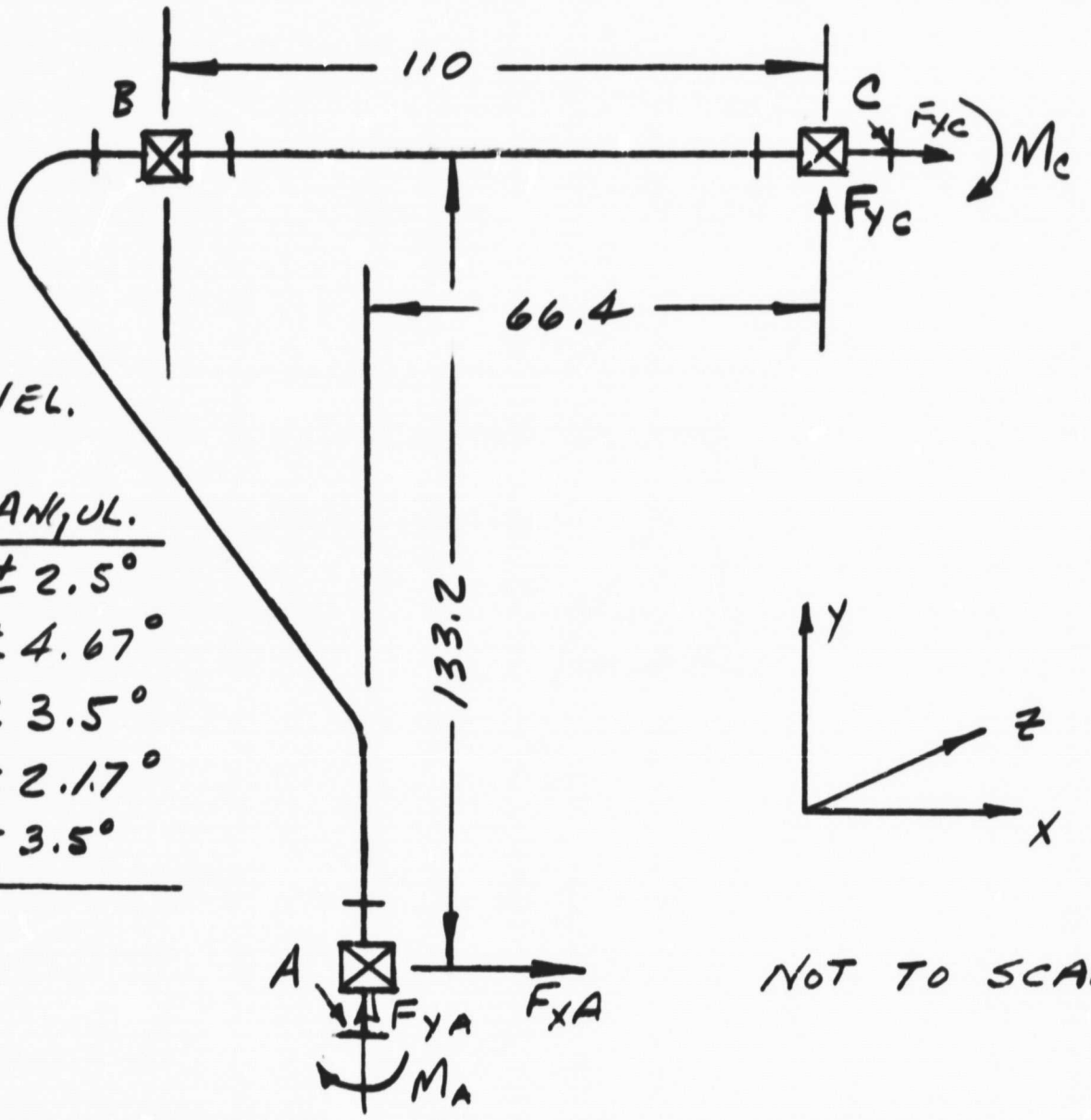
JOB NO. 6010438

SUBJECT: _____

BY: GWB

FLEXIBILITY ANALYSIS (END MOMENTS & REACTIONS)

MOTION STUDY IS SHOWN ON 41389



GIMB.	AXIS	ANGUL.
A	Z	±2.5°
B	Z	±4.67°
B	Y	±3.5°
C	Z	±2.17°
C	Y	±3.5°

12 IN. CUBIC ENVEL. AT PT. C

NOT TO SCALE

FROM BELLOWS ANALYSIS THE SPRING RATE OF GIMBALS IS FOUND TO BE

$M_{\phi T}$ = 1400 IN-LB/DEG. MIN. UN-PRESSURIZED
4575 IN-LB/DEG. MAX. UN-PRESS.

$$\text{AVG. } M_{\phi T} = \frac{1400 + 4575}{2} = 2988 \text{ IN-LBS/DEG}$$

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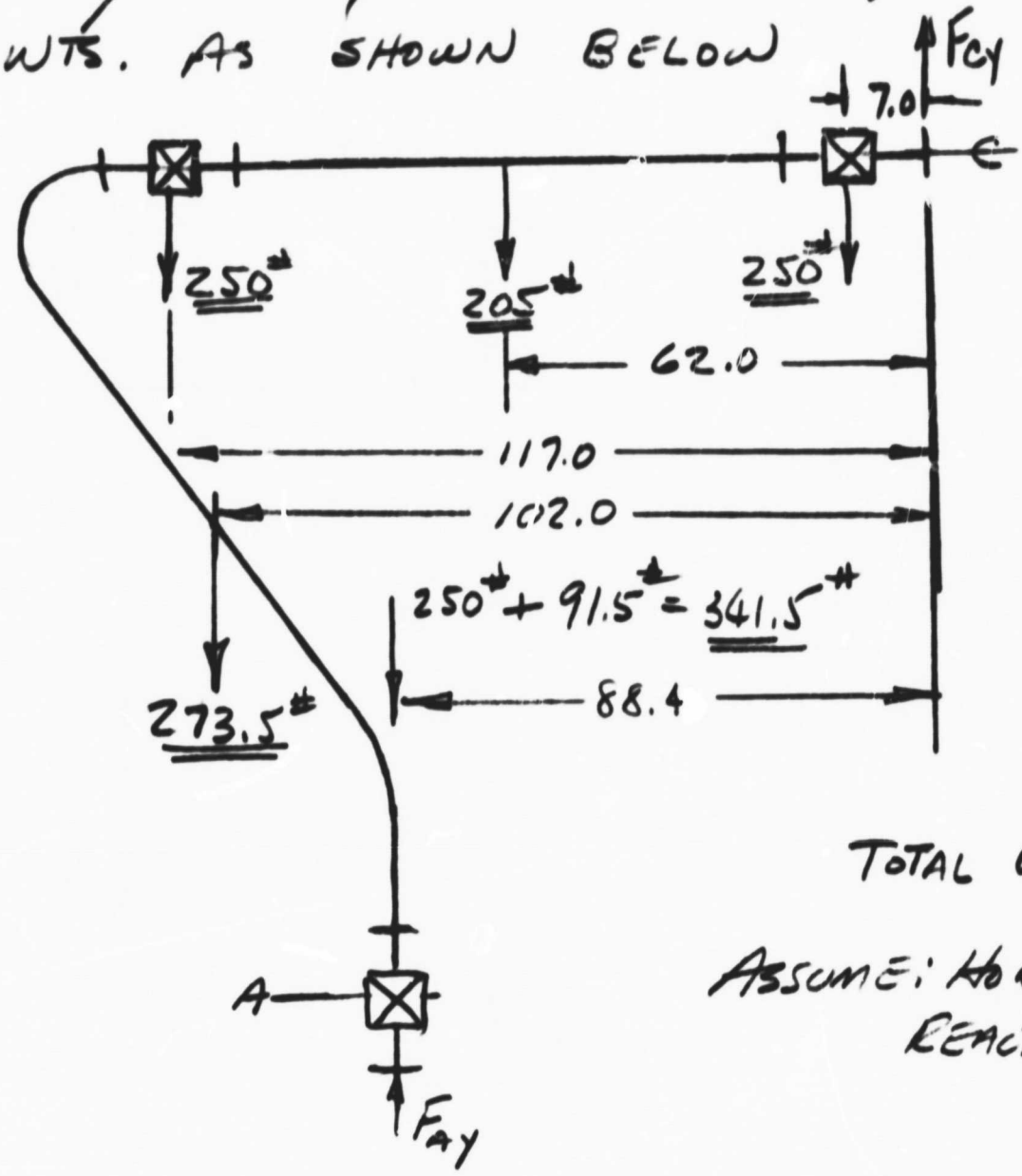
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SUBJECT: _____

BY: GWB

ASSUME 1G LOADING IN VERTICAL "Y" DIRECTION ONLY
LUMP WTS. AS SHOWN BELOW



TOTAL WT. = 1320 LBS.

ASSUME: HORIZONTAL REACTIONS = ZERO

$\Sigma F_y = 0, \quad F_{Ay} + F_{cy} = 1320 \text{ LBS.}$

$\Sigma M_C = 0, \quad F_{Ay}(88.4) = 250(7) + 205(62) + 250(117) + 273.5(102) + 341.5(88.4)$

$F_{Ay} = \frac{101,800}{88.4} = \underline{1150 \text{ LBS}}$ MAX.

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BY: CWB

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F_{CY} = 1320 − 1150 = 170 LBS MAX.

MAX. MOMENTS AT GIMBALS AT ±6 DEG.,

IS 6(2988) ≈ 18,000 IN-LBS UNPRESSURIZED

ANGULATION MOMENT DUE TO PRESSURIZATION OF BELLOWS

@ 12000psi IS 2792 IN-LBS/DEG OR 6(2792) = 16,750 IN-LBS

∴ TOTAL MAXIMUM END MOMENT AT ±6 DEG. @

10,000 psi IS 18,000 + 16,750 = 34,750 IN-LBS

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BELLOWS

The analysis of bellows is largely based on empirically derived or verified formulae. Solar has established working equations for the evaluation of bellows designs, which have proven reliable in many diverse applications. The specific analysis performed for this contract is described in this section.

It should be emphasized that all calculated spring rates, stresses, distortion (squirm) pressures, etc. are approximate and that individual bellows when tested will exhibit different values depending on how well they match the mathematical model.

The stress calculation methods outlined below do not necessarily apply at the end convolution where particular conditions of attachment may affect the convolution deflection in a unique way. In any case, the calculated stresses should be considered as only approximate.

The notation used throughout the analysis is given below.

BELLOWS NOTATION

a	Convolution radius, in.
B	Convolution shape factor
c	Convolution diameter, in.
E	Modulus of elasticity in tension, psi
F	Force, lb.
h	Half-span, inches
I	Area moment of inertia, in^4
K_M	Internal bending moment factor for stress calculation
k	Ratio of convolution outside to inside radius
ℓ	Active length of bellows, inches
M	Internal bending moment in bellows wall, in.-lb/in.-circum.
M_ϕ	Applied moment required to angulate unpressurized bellows, inch-lb
M_p	Angulation moment due to internal or external pressure, inch-lb

M_t	Total applied angulation moment = M + M_p, inch-lb
N	Number of active convolutions
η	Number of plies
P	Force, lb
p	Internal pressure, psi
P_{cr}	Critical stability pressure for a convoluted bellows, psi
R	Convolution outside radius, inches
r	Convolution inside radius, inches
R_m	Convolution mean radius, inches
S	Convolution span, inches
(SR)	Bellows axial spring rate, lb/inch
$(SR)_{1/4}$	Axial spring rate of a quarter - convolution, lb/inch
t	Average convolution wall thickness of a single ply, inches
t_{eff}	Effective thickness for hoop stress calculation, inches
t_{eq}	Equivalent single ply thickness of a multiple ply convolution, inches
t_{tot}	Total ply thickness of multiple ply convolution. inches
δ_A	Axial deflection, inches
$\delta_{1/4}$	Axial deflection of a quarter-convolution, inches
Φ	Bellows angle of rotation about midpoint, radians
ζ	Geometrical axial spring rate parameter, inches
$\varkappa$	Geometrical spring rate parameter, inches
ν	Poisson's ratio
ρ	Ratio of convolution inside to outside radius
$\beta, \xi, \varepsilon, \chi$	Geometrical stability parameters for convoluted bellows

σ_{bp}	Bending stress, due to pressure, psi
$\sigma_{b\delta}$	Bending stress due to deflection, psi
$\sigma_{b\phi}$	Bending stress due to angulation, psi
σ_{bt}	Total bending stress, psi
σ_{p}	Membrane pressure stress, psi

Cycle Life

The cycle life of gimbal bellows is based on the computed maximum radial bending stress in the convolutions due to the angulation of the gimbal. The maximum angulation for each gimbal is given on the motion study layout drawing for the duct assemblies.

The computed stress is compared with an S-N curve for the bellows material which was determined by cyclic axial deflection tests on individual bellows.

For the stress due to axial or angular movement it is first necessary to calculate the axial spring rate per quarter convolution. This calculation is outlined in "Bellows Spring Rate for Seven Typical Convolution Shapes", by James D. Matheny, Machine Design, January 4, 1962, p. 137.

For a known deflection of a quarter convolution, $\delta_{1/4}$, the shear force, P, at the point of inflection can be found from

$$P = (SR)_{1/4}\ \delta_{1/4}$$

For a total axial deflection of δ_A, the deflection per quarter convolution is

$$\delta_{1/4} = \frac{\delta_A}{4N}$$

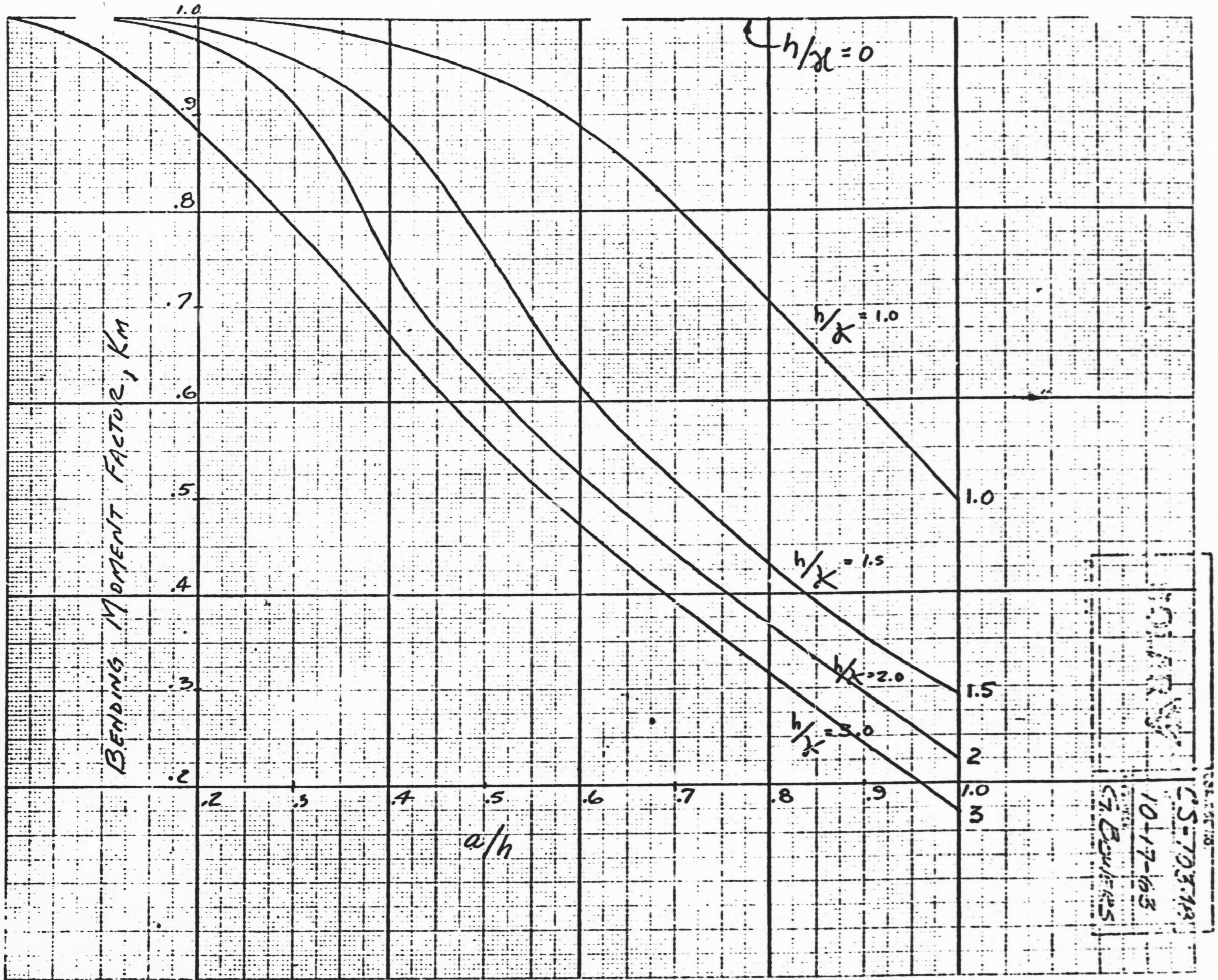
For an angular deflection of ϕ radians about the bellows center, the maximum deflection per quarter convolution is

$$\delta_{1/4} = \frac{R_m \phi}{4N}$$

Having determined the value of shear force, the maximum bending moment per inch of circumference is

$$M = \frac{K_m\ P_h}{2\pi R_m}$$

where K_m is a factor dependent on the convolution geometry and determines the maximum stress. For a flat-disk type bellows, $K_m = 1$, but for convoluted bellows with large values of a/λ, i.e., > 0.5, the toroidal section of the bellows accounts for only a small portion of the total deflection because of the constraining effect of circumferential hoop stresses. The calculation K_m as a function of h/λ is a rather lengthy operation, and has been done in Reference 4 for a constant value of $a/h = 0.5$. A value of $K_m = 0.8$ is used normally in calculations, however recent bellows designs show this to be too conservative; consequently the value is taken from Figure (CS-10348).


1.0
.9
.8
.7
.6
.5
.4
.3
.2
BENDING MOMENT FACTOR, KM
h/ℓ = 0
h/ℓ = 1.0
h/ℓ = 1.5
h/ℓ = 2.0
h/ℓ = 3.0
1.0
1.5
2
3
.2
.3
.4
.5
.6
.7
.8
.9
a/h

The radial bending stress due to quarter-convolution deflection, $\delta_{1/4}$, is

$$\sigma_{b\delta} = \frac{6\,M}{t^2}$$

where M is the bending moment per inch of circumference.

Since shear force P is the total value for the bellows it must be divided by the circumference at the mean diameter.

The radial bending stress for either axial movement or angulation can be expressed as

$$\sigma_{b\delta} = \frac{3K_m\,Ph}{\pi R_m t^2}$$

This reduces to the following equations for $K_m = 0.8$

$$\sigma_{b\delta} = \frac{.764\,\delta\,(SR)\,h}{R_m\,t^2} \quad \text{for axial deflection}$$

$$\sigma_{b\phi} = \frac{.764\,\phi\,(SR)\,h}{t^2} \quad \text{for angular rotation } \phi \text{ (radians)}$$

In case of multiple-ply bellows, the radial bending stress in each ply due to deflection is calculated as though the ply were an individual bellows.

The following rules can be used to determine whether tension or compression stresses exist on the outside surface:

1. Axial compression = tension at crest and compression at root
2. Axial extension = compression at crest and tension at root
3. Internal pressure = compression at crest and compression at root

Pressure Stresses

From Feely, F. J., Jr. and Goryl, W. M., 'Stress Studies on Piping Expansion Bellows", Journal of Applied Mechanics, Vol. 17, June 1950, pp. 135, the radial bending stress due to pressure, sometimes called bulging stress, is given by

$$\sigma_{bp} = \frac{p\,S^2}{2\eta t^2}$$

This equation is derived from flat plate theory, and as a result has restricted accuracy where applied to convoluted shapes. As a general rule it will give more accurate values for bellows with the geometrical parameter a/h between 0 and 0.5. For a/h greater than 0.5, the constraining effect of circumferential hoop stresses resulting from convolution deflection will give conservative values.

For bellows in which the quarter pitch (a) is nearly equal to the half span (h), the above formula is much too conservative. In these cases, the following equation given by Clark (Reference 5) is applicable:

$$\sigma_{bp} = \frac{0.955\ p}{\sqrt[3]{1-\nu^2}} \sqrt[3]{\left(\frac{Rm\ a}{t^2}\right)^2}$$

ν = Poisson's Ratio

Radial bending stresses due to pressure are largely self-relieving, therefore, if the radial bending stresses are in excess of the allowable yield, the convolution will permanently deform to a shape where radial membrane and bending stresses are near the yield value. This usually involves some local cold-working which shifts the yield point to a higher stress level. Without root rings, only a small amount of this deformation can be tolerated before the convolution root has "failed" through deformation. It should also be remembered that this pressure stress represents a mean stress upon which the alternating or fluctuating stresses due to movement are superimposed. As this mean stress approaches the yield value, the cycle life is more seriously affected.

Since the number of surge pressure cycles is generally on the order of 100 or less, no cycle life calculations were performed. Computed linear elastic stresses to 150% of the ultimate strength of the material will not result in failure.

The bellows hoop stress is calculated using an "effective" thickness at the mean radius of the bellows which is based on the total cross section of material per unit length. Thus,

$$\sigma_H = \frac{p\,Rm}{t_{eff}}$$

and for a convoluted bellows,

$$t_{eff} = \frac{t\,\eta}{pitch} \left[\frac{2\pi\ pitch}{4} + 2\ span - pitch\right]$$

therefore by substitution,

$$\sigma_H = \frac{p\ Rm\ pitch}{t\left[.571\ pitch + 2\ span\right]\eta}$$

Hoop stress alone which is very seldome the cause of bellows failure, however, is routinely calculated to demonstrate the large margin of safety.

Spring Rate and Angulation Moment The axial spring rate (SR) is computed by the method cited in "Bellows Spring Rate for Seven Typical Convolution Shapes", Machine Design, January 4, 1962. For multiple ply bellows, the spring rate is the sum of the individual spring rates. Using the nomenclature of this report, the spring rate is given by

$$SR = \frac{2\pi R_m E t^3}{\zeta 12 (1-\nu^2) h^3 4N}$$

where ζ is shown on Figure of the following page.

ANGULATION MOMENT

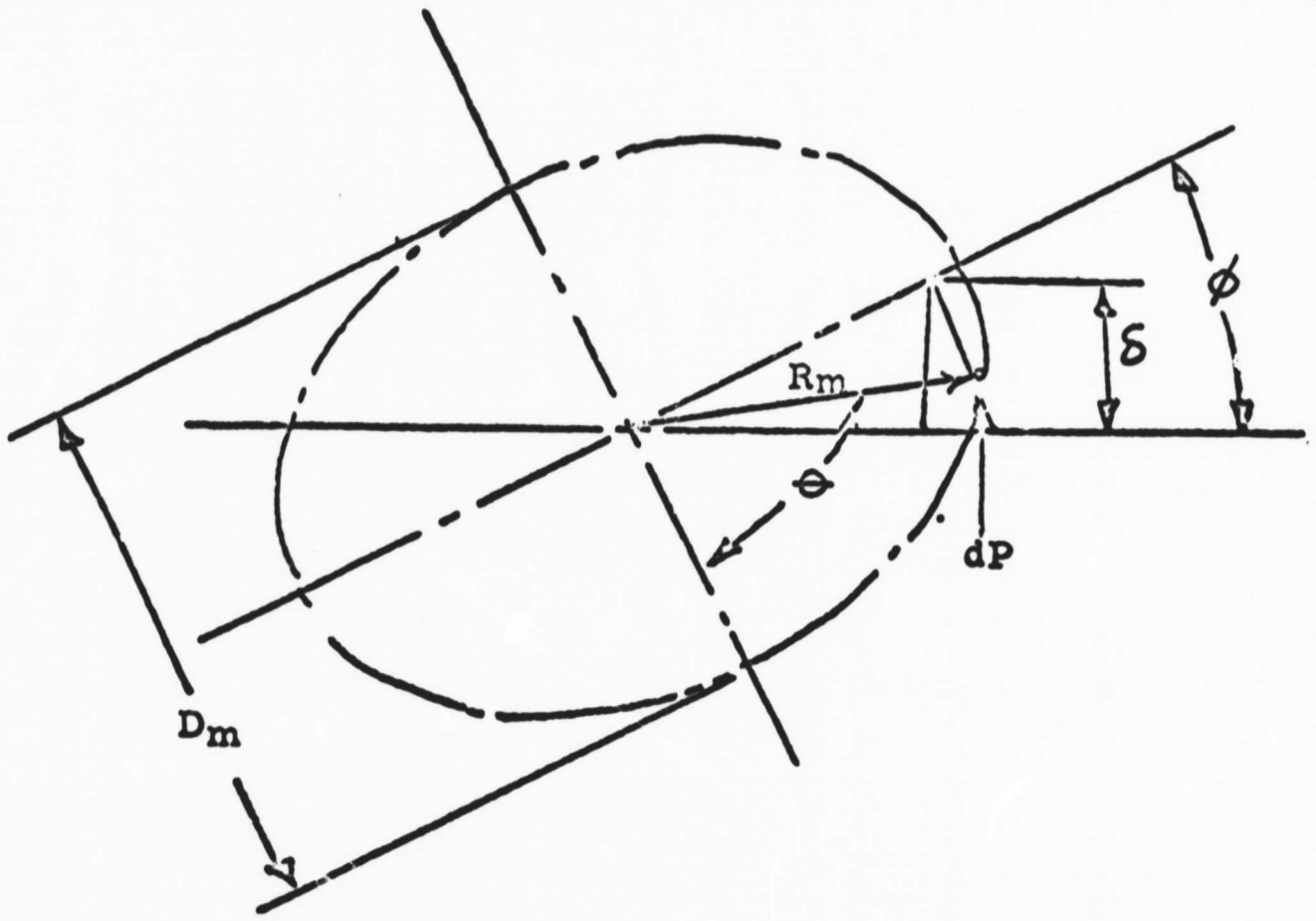
Neglecting the effect of pressure, the applied moment required to angulate a convoluted bellows about a pivot point at the bellows center is given by

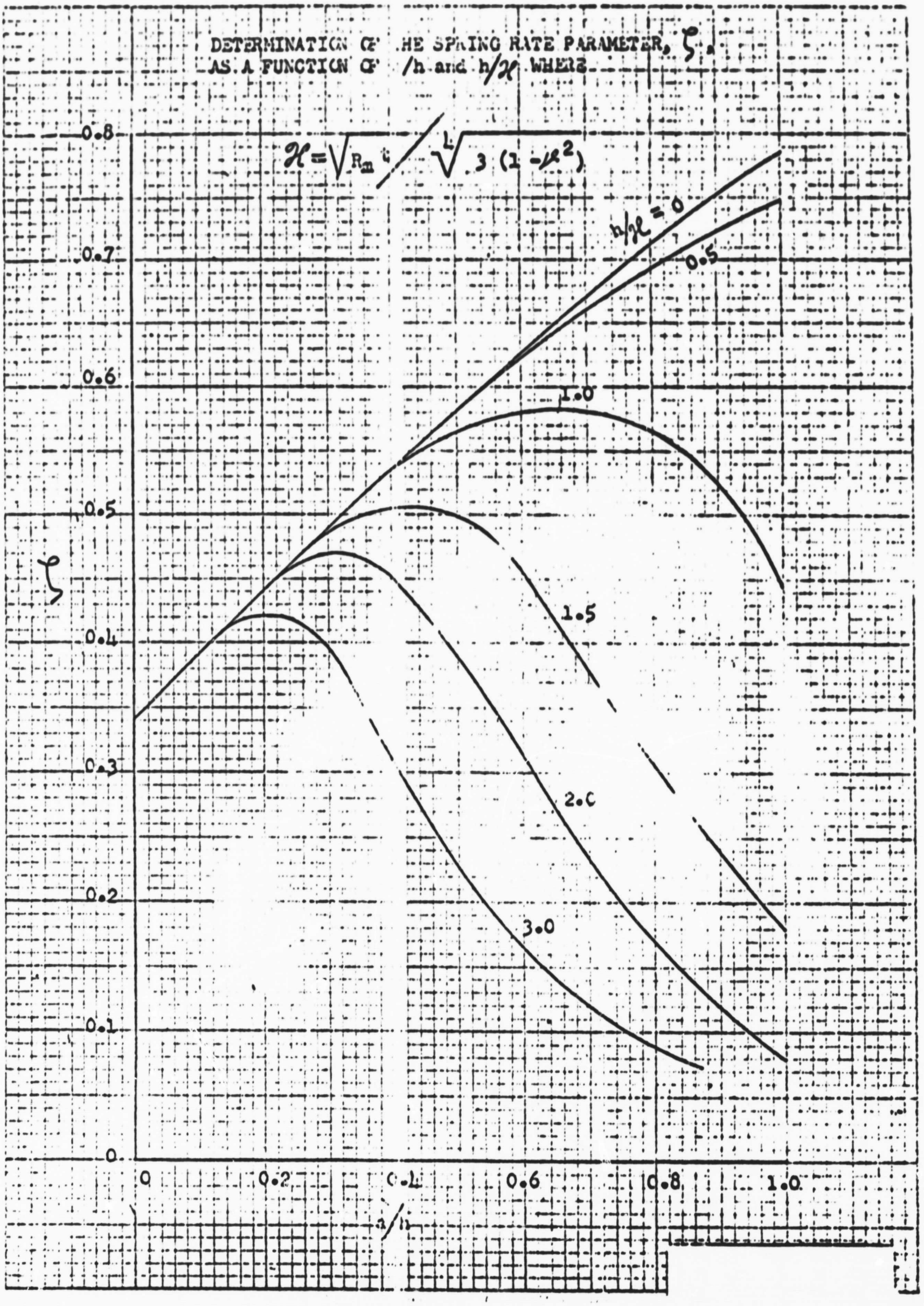
$$M_\phi = \frac{(SR) R_m^2 \phi}{2}$$

where ϕ is expressed in radians for small angles.

The derivation of this equation follows.

The circumferential force P causing the convolution deflection acts at the mean radius, R_m. Referring to the following figure and noting that symmetrical tension and compression of the bellows element is assumed about the mean diameter,




DETERMINATION OF THE SPRING RATE PARAMETER, ζ,
AS A FUNCTION OF /h and h/ WHERE
$\mathcal{H} = \sqrt{R_m t} \Big/ \sqrt[4]{3(1-\mu^2)}$
h/ = 0
0.5
1.0
1.5
2.0
3.0
ζ
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0
0.2
0.4
0.6
0.8
1.0
a/

the deflection forces which vary around the mean circumference must be multiplied by the corresponding moment arms and integrated to obtain the total force. From Reference /6, we can write

$$\delta_{1/4} = \frac{P h^3 (1-\mu^2)}{E I} \zeta$$

or

$$P = \frac{\delta_{1/4} E I}{h^3 (1-\mu^2) \zeta}$$

Therefore the elemental force at any location will be

$$dP = \frac{\delta_{1/4} E I}{h^3 (1-\mu^2) \zeta}$$

and since

$$I = \frac{R_m t^3 d\theta}{12}$$

we can write

$$dP = \frac{\delta_{1/4} E R_m t^3 d\theta}{12 h^3 (1-\mu^2) \zeta}$$

From the above figures it can be seen that

$$\delta_{1/4} = R_m \sin\theta \sin\phi$$

and by substitution,

$$dP = \frac{R_m \operatorname{Sin}\theta \sin\phi E R_m t^3 d\theta}{12 h^3 (1-\mu^2) \zeta}$$

now $dm = dP R_m \operatorname{Sin}\theta$

therefore:

$$M_\phi = \int dm \int R_m \operatorname{Sin}\theta \, dP$$

substituting the expression for dP, and taking the total moment as four times the integral over a quadrant

$$M_\phi = \frac{R_m^3 \sin\phi \; E \; t^3}{12\,(1-\nu^2)\,\rho\, h^3} \; 4 \int_0^{\pi/2} \sin^2\theta \; d\theta$$

$$= \frac{R_m^3 \sin\phi \; E \; t^3 \pi}{12\,(1-\mu^2)\,\rho\, h^3}$$

This represents the moment required to deflect a quarter convolution, therefore for N active convolutions the above equation must be divided by 4N to get the total moment M_ϕ . The expression for axial spring rate was given as

$$(SR) = \frac{2\pi R_m \; E \; t^3}{\rho\, 12\,(1-\nu^2)\, h^3 \; 4N}$$

We can therefore write an expression for the total moment in terms of axial spring rate, and since for small angles $\sin\phi \cong \phi$ (in radians)

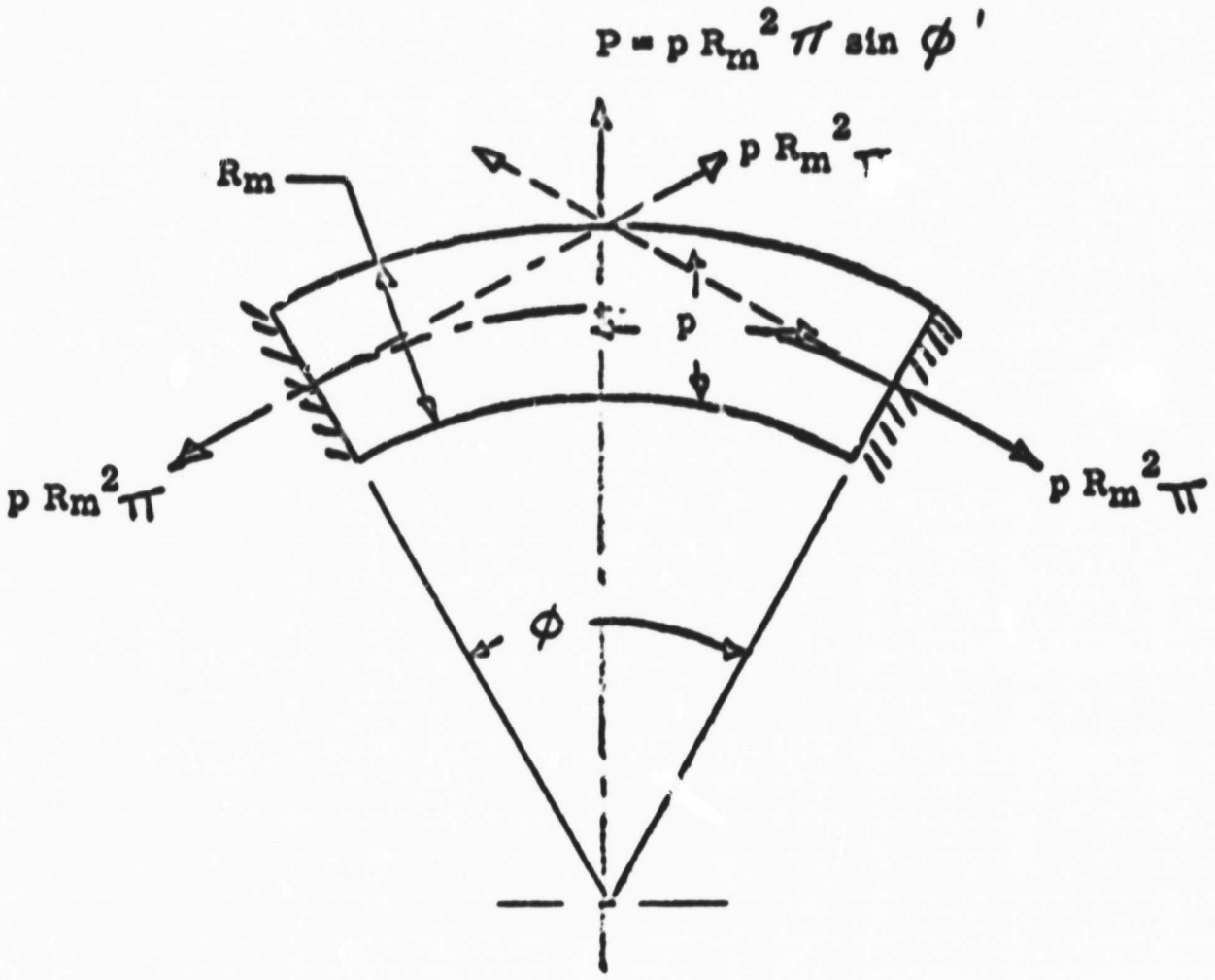
$$M_\phi = \frac{(SR)\;\; R_m^2\, \phi}{2}$$

The prediction of pressure effect on applied angulation moment is a theoretically formidable problem. An approximate expression for the applied moment due to pressure alone is given by

$$M_p \approx \frac{R_m^2 \pi p \, \phi \ell}{4}$$

where ϕ is expressed in radians for small angles. A derivation of this equation follows.

Consider the following schematic of an angulated bellows:



The bellows curvature results in a resultant pressure reaction

$$P = p R_m^2 \pi \sin \phi$$

which for bellows with uniform curvature acts at the bellows center. The bellows ends are normal to the curvature. The bellows is thus considered as a simply supported beam with concentrated load at the center. The necessary applied moment for equilibrium can be written as

$$M = \frac{P \ell}{4} = \frac{p R_m^2 \pi \sin \phi}{4}$$

This expression is restricted to small angles therefore

$$M = \frac{p R_m^2 \pi \phi \ell}{4}$$

where ϕ is expressed in radians.

It will be noticed that Mp is essentially linear for small angles. Based on testing at Solar the equation for Mp will give an approximate value of Mp up to 4 degrees angulation and 300 psig.

The total angulation moment of an internally pressurized bellows can be expressed as

$$M_t = M_\phi + M_p$$

Distortion Pressure or Squirm

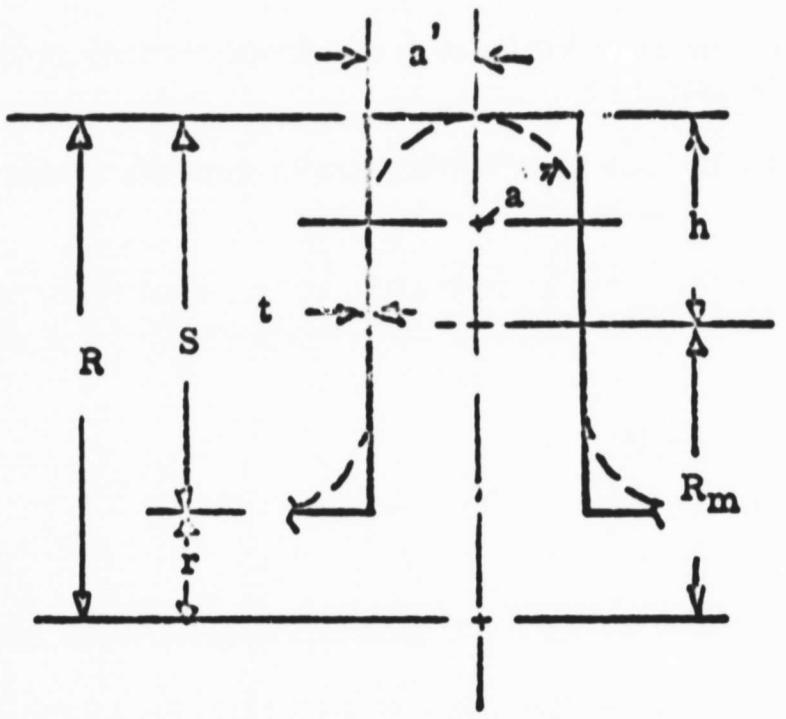
Theoretical methods for determining the critical distortion pressure at which convoluted bellows become unstable have been established for a bellows loaded by either internal or external pressure, and either in the neutral or angulated position. (See References 13, 14, 15.) Limited experimental verification has been obtained from tests conducted at Solar for the case of a convoluted bellows in the neutral position with fixed ends and internal pressure. More experimental data are required to validate or improve the above methods for a range of bellows geometry paramcters and conditions of end fixity.

It is assumed that the pressure end load on the bellows is reacted by the bellows support structure. Also, it should be pointed out that, in line with stability calculations in general, a calculated value of bellows stability cannot be expected to be exact. Therefore, in design decisions, judgment must be used based on an order of magnitude prediction. Furthermore, it is assumed that the analysis presented below pertains to a convoluted bellows. More investigation is required to determine the applicability to other shapes. Since stability is governed by the bending stiffness, it can be reasonably expected that the methods are essentially general.

Internal pressure causes each convolution to try to increase in length, resulting in a column type of instability. Therefore the critical internal pressure is a function of the length to diameter ratio, other things being equal.

The method used in this case is derived in Reference 13. In order to be mathematically tractable, the assumption of a rectangularly shaped convolution has been made. However, since stability is a function of the bending stiffness which in turn is a function of the axial spring rate, a shape correction factor has been calculated based on theoretical data from Reference 16. This factor is shown graphically in Figure 1. A schematic showing the corrugated shape approximation is illustrated below.

NOTE: See p. 1-38 for Figure 1.



The effect of bellows geometry on this approximation is self-explanatory.

The following expression gives the critical internal pressure for a bellows maintained in a neutral axial position.

$$P_{cr} = B\frac{\beta' E' t_{eq}^3 R}{S^3 N' \ell' (1-f/f0)}$$

where:

$$E' = \frac{E}{(1-\mu^2)}$$

B = Convolution shape correction factor

$\beta'' = \beta/\chi' = \frac{\pi^2 \xi}{2 \xi \chi'}$ (see Figure 2) p 1-39

$\chi \approx 1 + F(\varrho, c/\sqrt{Rt_{eq}})\, c/R$ (see Figure 3) p 1-40

f_o = Bellows axial deflection due to pressure, p, assuming the end is unconstrained

f = Bellows axial deflection due to applied compressive force where always $f < f_o$

ℓ', N' = Equivalent buckling length and equivalent number of active convolutions, as a function of end restraint, defined for gimbals as follows

$$\ell' = \ell/2$$
$$N' = N/2$$

For multiple ply bellows, the equivalent thickness (t_{eq}) must be used.

Theoretically this is given for plies of equal thickness by the following expression:

$$t_{eq} = t_{tot} \frac{\sqrt[3]{\eta}}{\eta}$$

Recently, another method for determining pressure and motion stresses and spring rates has been presented by A. Laupa and N. A. Weil, "Analysis of U-Shaped Expansion Joints", Journal of Applied Mechanics, March 1962, p. 115. Solar is programming this method for an IBM 7090 Digital Computer and will include these results in the report, if possible.

APPENDIX C

RELIABILITY PROGRAM

This Appendix is a description of the intended reliability program for, and analysis of the high pressure, high flow rate ducting system, which Solar proposes to develop for ground pressurant supply for the LUT. Documentation in support of the program and analysis are included.

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APPENDIX C

C.1 RELIABILITY PROGRAM

C.1.1 Objectives

The reliability program for the LUT high pressure, high flow rate ducting system will be similar to existing programs at Solar. The objectives are thorough planning and effective management of the reliability effort, including careful definition of tasks and problems and analytical, monitoring, and reporting techniques consistent with the best correct practice.

C.1.2 Experience

Solar has produced aerospace products for many years and has consistently adhered to conservative practice and meticulous attention to detail to obtain reliability in the product. During the past several years, Solar has been applying formal reliability techniques to ducting programs for Saturn S-1C and S-II.

C.1.3 Organization

Reliability at Solar is administered by a corporate officer who coordinates departmental efforts, as described in SPI 6.930. A copy of this document is included at the end of this Appendix. Each project has a reliability team specifically assigned to analyze, plan, and follow up throughout the life of the program. Figure C-1 shows the interrelationships which are maintained during a program. The reliability director leads the design reliability effort and is chairman of the reliability committee. The reliability committee consists of delegates from design, analytical, production, quality control, and project engineering. It is a working level group with excellent visibility which finds and illuminates problems. The committee recommends action to the project manager upon definition of the problem and location of avenues of solution. Personnel training for reliability is carried out by Industrial Relations as a matter of reliability policy.

C.1.4 Design Review

Design review is a project management function with participation by all affected groups. Prints are distributed, meetings are held, and recommendations are made to the project manager.

C.1.5 Failure Reporting

Solar's failure reporting system for data collection and constructive feedback is described by SPI 2.490. It includes forms and procedures by which Solar, Solar's vendors, and Solar's customers can operate in an orderly organized manner, to improve product reliability for all.

C.2 RELIABILITY ANALYSIS

A system of reliability analysis is described in Solar ER 1404-1. This method has been followed in the current effort, as well as several parallel programs involving ducting for the Saturn vehicle.

C.2.1 Description

The analysis of a portion of a ducting system, such as the high pressure duct assembly, is based on a detailed analysis of separable components. The modes of failure which are possible are listed as well as the effect of such failure (on two levels) and generalized methods of preventing failure. At the time of the present analysis, specific means were not available for study, but preparation of the system of specifications, instructions, and standards required to prevent failure must follow as part of the design and production engineering effort. The perfection of this network of specifications will be evaluated by the reliability committee. The failure mode and effect analysis prediction for this duct may be found in Appendices C and D.

C.2.2 Prediction

The probability of the duct assembly adequately fulfilling its function during the 36 hours of pre-launch is estimated to be 0.999814. No confidence level applies to this estimate.

41314 Line Assembly - Connector

The subject duct assembly is used to supply pressurant to the L.U.T. system from ground storage. The mode of operation is long periods of high pressure punctuated by surges and sudden vent conditions. At launch, a short period of vibration exists. The function which must be predicted is maintenance of flow and pressure during pre-launch and launch activities. It has been assumed that operation time is 36 hours per launch, during which time the function must be provided. Reliability analysis is based on the assumption that inspection, repair, and replacement is possible between launches.

The environment which surrounds the duct is marine air @ -65F to /150F. This environment is not likely to degrade the strength of the unit, except by corrosion. Also, human damage is possible.

Media are GN_2 & GHe @ 104000 SCFM & 204000 SCFM flow rates respectively. Operating pressure is 10,000 psig. Temperatures are -65F to /150F, as noted above.

H1 PRESSURE, H1 FLOW DUCT FAILURE MODE & EFFECT ANALYSIS
SOLAR ASSEMBLY NUMBER 41314

ELEMENT DESCRIPTION	FAILURE MODE, CAUSE & FREQUENCY RATIO	FAILURE EFFECT UPON ASSEMBLY & FUNCTION	MEANS OF REDUCING OR ELIMINATING FAILURE
Tubes 41311-1 41312-1 INCO 718 Tubing 3.50 ID X .50 Wall (2) Required Total	Minor Leakage (r = 0.80) 1) Pores & cracks in weld region 2) Fatigue Cracks 3) Corrosion 4) Cracks due to forming 5) Pores & cracks in parent matl.	Leakage flow may induce vibration then fatigue damage, leading to major leakage and rupture. Reduced pressurant flow.	1) Material selection + procurement control welding procedure and inspection standards. Process control and acceptance test. 2) MS + PC Support Design & Damping Provisions 3) MS + PC; Process Control 4) Same as 3) + Fab. Standards 5) MS + PC
	Major Leakage (r = 0.19) 1) thru 5) same as minor leakage.	Vibration Fatigue Inadequate pressurant flow.	1) to 5) same as minor leakage.
	Rupture (r = 0.01) 1) thru 5) same as minor leakage. 6) deficient ductility	Secondary structural failure due to loss of support. Loss of pressurant flow; personnel hazard	1) to 5) same as minor leakage. 6) weld procedure and inspection standards MS + PC.
Flanges No Number "Grayloc" Flange 3.50 Dia	Minor Leakage (r = 0.80) 1) thru 3) same as tubes 4) cracks in parent material. 5) seal surface damage 6) misalignment	Vibration fatigue Reduced pressurant flow.	1) to 3) same as tubes. 4) MS + PC 5) handling procedure 6) assembly procedure
	Major Leakage (r = 0.19) 1) thru 6) same as minor leakage	Vibration fatigue Inadequate pressurant flow.	1) to 6) same as minor leakage
	Fails to Maintain Support (r = 0.01) 1) thru 4) same as minor leakage 5) deficient ductility	Secondary structural failure. Loss of pressurant flow, secondary structural damage to associated components	1) thru 4) same as minor leakage 5) MS + PC, WP + I.S., acceptance standards

H1 PRESSURE, H1 FLOW DUCT FAILURE MODE & EFFECT ANALYSIS
SOLAR ASSEMBLY NUMBER 41314

ELEMENT DESCRIPTION	FAILURE MODE, CAUSE & FREQUENCY RATIO	FAILURE EFFECT UPON ASSEMBLY & FUNCTION	MEANS OF REDUCING OR ELIMINATING FAILURE
Seals "Grayloc" Type 3.50 Dia	Minor Leakage (r = 0.60) 1) Pores & cracks in material 2) Fatigue cracks 3) Corrosion 4) Handling damage 5) Faulty assembly	Vibration fatigue Reduced pressurant flow.	1) Material selection and procurement control acceptance standards. 2) MS + PC, Support design + damping provisions 3) MS + PC Process control, maintenance procedures. 4) Handling procedures, maintenance procedures 5) Same as 4)
	Major Leakage (r = 0.19) 1) to 5) same as minor leakage	Vibration fatigue Inadequate pressurant flow.	1) to 5) same as minor leakage
	Rupture (r = 0.01) 1) thru 5) same as minor leakage 6) deficient ductility	Secondary structural failure due to loss of support. Loss of pressurant flow; contamination of downstream components by rubble	1) thru 5) same as minor leakage 6) MS + PC, acceptance standards
Gimbal Assemblies 41308 (3 Required)	← SEE DETAIL ANALYSIS BELOW		→
Gimbal Flanges 41307-1 8.78 Dia X 9.0 Long INCO 718 Forgings (2 Required per assembly)	Minor Leakage (r = 0.75) 1) thru 6) same as flanges	Vibration fatigue Reduced pressurant flow	1) thru 6) same as flanges.
	Major Leakage (r = 0.14) 1) thru 6) same as minor leakage	Vibration fatigue Inadequate pressurant flow	1) thru 6) same as flanges.
	Fails to Maintain Support (r = 0.06) 1) thru 5) same as flanges.	Secondary structural damage Secondary structural dange to associated components.	1) thru 5) same as flanges.
	Impedes Gimballing (r = 0.04) 1) corrosion 2) seizure 3) handling damage 4) contamination	Increased loads in assembly. Increased loads in associated components.	1) & 2) MS + PC 3) & 4) Handling - Maintenance procedure
	Fails to Gimbal (r = 0.01) 1) to 3) same as "impedes gimballing"	Structural damage in assembly. Structural damage in associated components.	Same as "impedes gimballing"

H1 PRESSURE, H1 FLOW DUCT FAILURE MODE & EFFECT ANALYSIS

SOLAR ASSEMBLY NUMBER 41314

ELEMENT DESCRIPTION	FAILURE MODE, CAUSE & FREQUENCY RATIO	FAILURE EFFECTS UPON ASSEMBLY & FUNCTION	MEANS OF REDUCING OR ELIMINATING FAILURE
Gimbal Ring 41306-0 12^{00} OD X 1^{50} Thk X 6^{00} Long INCO 718 Bar (4 Quadrants) (1) Required per assembly	Fails to Maintain Support (r = 0.20) 1) thru 3) same as tube leakage. 4) wear 5) deficient ductility	Secondary structural failure due to loss of support. Secondary failure of associated components.	1) thru 3) same as tubes. 4) fit control; MS + FC 5) WP + IS; MS + FS
	Impedes Gimballing (r = 0.79) 1) thru 4) same as gimbal flanges	Increased loads in assembly Increased loads in associated components.	1) & 2) MS + FC 3) & 4) handling and Maintenance Procedure.
	Fails to Gimbal (r = 0.71) 1) thru 4) same as "impedes"	Structural damage in assembly Structural damage in associated components	1) thru 4) same as "impedes"
Gimbal Bellows 41310 (Made from 41309) ID = 3^{50} S = 1^{00} P = .75 N = 10 t = .20 INCO 718 Bar (1) Required per assembly	Minor Leakage (r = 0.80) 1) thru 5) same as tubes	Vibration fatigue Reduced pressurant flow	1) thru 5) same as tubes
	Major Leakage (r = 0.18) 1) thru 5) same as minor leakage	Vibration fatigue Inadequate pressurant flow	1) thru 5) same as minor leakage.
	Rupture (r = 0.01) 1) thru 6) same as tube rupture	None Loss of pressurant flow	1) thru 6) same as tube rupture
	Impedes Gimballing (r = 0.01) 1) reshaping or bulging of convolutions	Increased loads in assembly Increased loads in associated components	1) MS + FC; fab inspection standards; Development test.

$$K_A = \frac{1}{\text{Factor of Safety}} + .01$$

TUBES Leakage & Rupture

$$K_A = \frac{1}{2.83} + .01 = \underline{.36}$$

Based on Hoop Stress in Weld

FLANGES LEAKAGE - Same as Tubes $K_A = \underline{.36}$

Fails to Support (Assumed per ER 1404-1) $K_A = \underline{.04}$

SEALS LEAKAGE & RUPTURE - Same as Tube Parent Mat'l.

$$K_A = \frac{.85}{2.83} + .01 = \underline{.31}$$

Based on Hoop Stress

GIMBAL FLANGES Leakage Same as Tubes $K_A = \underline{.36}$

Fails to Support $K_A = \frac{1}{1.74} + .01 = \underline{.59}$

Based on Pin Bending Fatigue

Fails to Gimbal $K_A = \frac{17540}{30000} + .01 = \underline{.59}$

Based on Galling Limit

GIMBAL RING Fails to Support $K_A = \frac{1}{2.35} + .01 = \underline{.44}$

Based on Torsional Fatigue

Fails to Gimbal Same as Gimbal Flange $K_A = \underline{.59}$

Based on Galling Limit

GIMBAL BELLOWS Rupture $K_A = \frac{1}{2.21} + .01 = \underline{.46}$ - Based on Hoop Stress

Leakage $K_A = \frac{1}{1.123} + .01 = 0.90$ - Based on Deflection Bending Fatigue

SOLAR ASSEMBLY NO. 41314 H1 PRESSURE H1 FLOW DUCT

RELIABILITY PREDICTION

ELEMENT DESCRIPTION AND NOTE REFERENCE	QUANT. N	FAILURE MODE	FAIL. MODE FREQ. RATE	DATA SOURCE ER1404-1 APP A Page/Line	GENERIC FAIL. RATE F Fail/10^6 H	DERATING FACTOR K_a	PREDICTED FAIL. RATE $Fp = \pi F_g K_e K_a$ $K_e = 8$ Fail/10^6 H	PREDICTED UNRELIAB. FOR t = 36 H $10^6 A = Fp\ t$ Fail/10^6 Trials
Tubes - 41311-1, 41312-1 3^{50} Dia X .50 Wall	2	Major Leakage Rupture	.19 .01	14/2	.224	0.36	.1225 .0065	4.410 0.023
Flanges - No Number 3^{50} Dia "Grayloc" Type	4	Major Leakage Fails to Maintain Support	.19 .01	5/19 12/25	1.200 .331	0.36 0.04	.6560 .0011	23.616 0.040
Seals - No Number 3^{50} Dia "Grayloc" Type	4	Major Leakage Rupture	.19 .01	12/1	.50	0.31	.2360 .0124	8.496 .447
Gimbal Flanges 41307-1 8^{78} Dia X 9.0 Long	6	Major Leakage Fails to Maintain Support Fails to Gimbal	.14 .06 .01	14/2 12/25 2/17	.224 .331 .046	0.36 0.59 0.59	.0902 .0930 .0022	3.247 3.350 .079
Gimbal Ring 41306-0 12^{00} Dia X 1^{50} Thk X 6^{00} Long	3	Fails to Maintain Support Fails to Gimbal	.20 .01	12/25 2/17	.331 .046	0.44 0.59	.2330 .0022	8.388 .079
Gimbal Bellows 41310	3	Major Leakage Rupture	.18 .01	2/5	2.80	0.90 0.46	3.6400 .1030 5.198	131.040 3.708 187.130

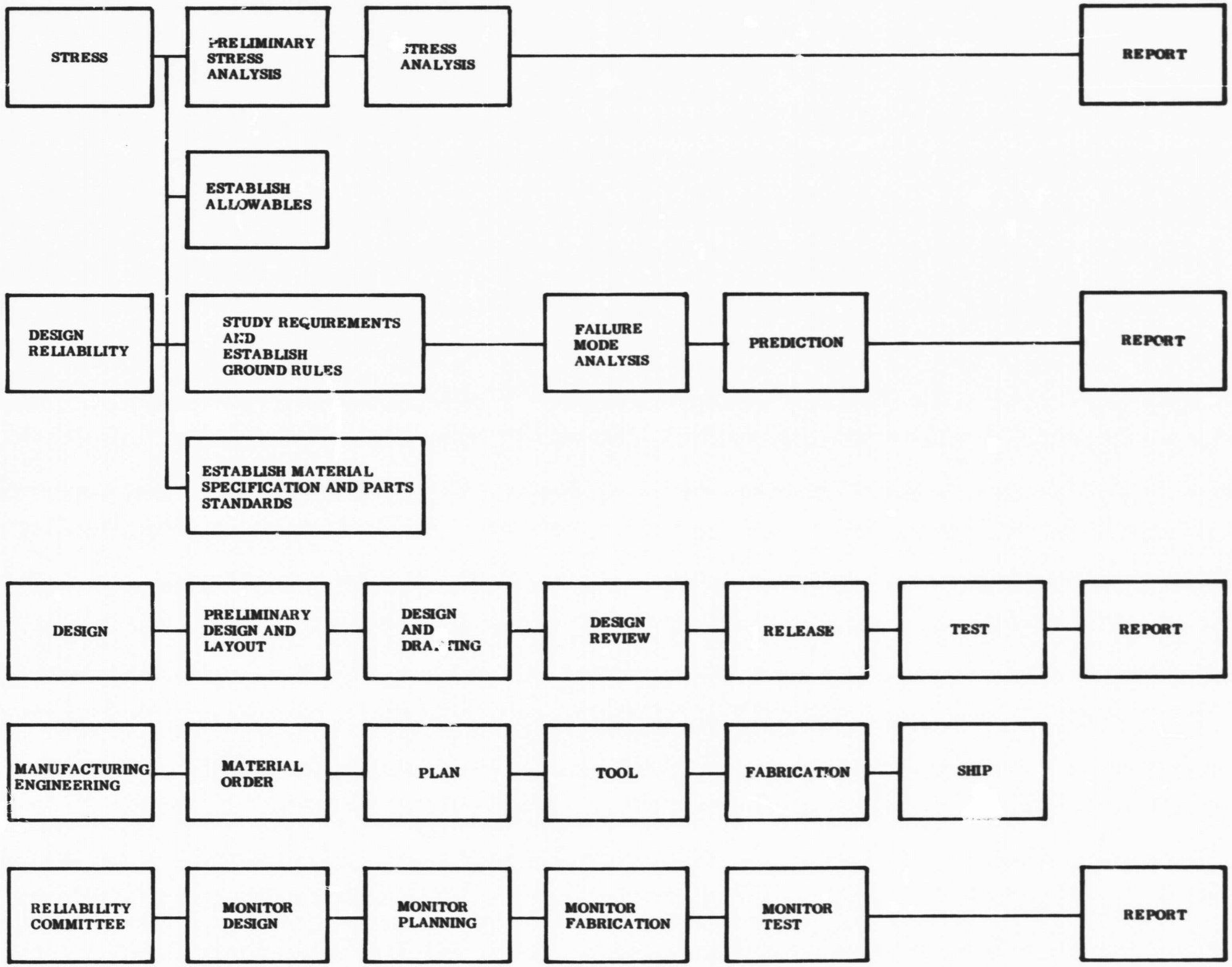


FIGURE C-1. RELIABILITY PROGRAM SEQUENCE DIAGRAM

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STANDARD PRACTICE INSTRUCTION

SUBJECT: RELIABILITY PROGRAM

CLASSIFICATION: S. P. I. 6.930
DATE ISSUED: May 1, 1963
SUPERSEDES: November 13, 1962
PLANTS AFFECTED:

1. PURPOSE

To establish Company policy governing reliability programs activated to ensure that the end product possesses the level of reliability which is in accordance with customer requirements and established Company standards.

2. POLICY

2.1 Each reliability program shall recognize the concept of inherent reliability of design and that the concentration of such effort must take place in the early design phase. This shall result in mature design of hardware suitable for production, for use in the operational environment and ease of maintenance. Every effort shall be made to ensure retention of reliability during manufacturing, testing and use phases.

2.2 The following principles shall be applied in the reliability program.

(1) Reliability is a major factor in planning, management and engineering.

(2) Inherent reliability is established by basic design and can only be improved by design changes.

(3) Improvement of inherent reliability is best achieved in the early phases of a development and test program.

(4) The reliability factor is also affected by the human element and the product is to be engineered to minimize human error in manufacturing, tests, operation and maintenance, not only by design, but also through creation of suitable operating and maintenance manuals.

(5) The inherent design reliability may be degraded as a result of variation in manufacturing, maintenance, storage, transportation and operational processes. Accordingly, Production, Quality Control and Testing are to be planned within, and strictly controlled by, Engineering requirements.

(6) Estimated reliability is to be compared with the achieved reliability at predetermined steps of design, development, production and operation.

(7) The collection, analysis and feedback of information to the proper operating activity is fundamental to the improvement of existing designs and mature future ones.

(8) The required end product reliability can only be obtained by coordination of all activities.

3. RESPONSIBILITIES

3.1 General

3.1.1 A Manager of Product Reliability has been appointed and will report to Vice President - Operations.

3.1.2 Company wide reliability committees shall be established as determined by Vice President - Operations for products designated to be governed by a planned reliability program.

3.2 Manager - Product Reliability

3.2.1 Generate reliability awareness and motivation throughout the Company.

3.2.2 Continually evaluate latest techniques in the field of reliability and military standards and effect their application to Company programs and policy.

3.2.3 Review reliability instructions and procedures and make recommendations for their acceptance.

3.2.4 Monitor reliability policy as quoted in proposals or presentations to ensure that they reflect over-all Company policy.

3.2.5 Keep management informed as to the progress of reliability programs.

3.3 Reliability Committee

3.3.1 Coordinate the efforts of all activities involved.

3.3.2 Review reliability status against estimates and standards applied to particular programs.

3.3.3 Record reliability status and special findings and distribute this information to management.

3.4 Engineering

3.4.1 Maintain competence in the design and analytical departments for the application of current reliability techniques.

3.4.2 Attain required reliability of basic design through sound design concepts and development taking into consideration results of reliability studies, failure mode analysis, calculations for quantitative reliability prediction and apportioning.

3.4.3 Establish development, qualification and reliability test plans and conduct tests as planned.

3.4.4 Conduct failure analysis and determine corrective action.

3.4.5 Establish quality requirements and specifications and prepare operating and maintenance manuals.

3.4.6 Evaluate operation of end product in use.

3.5 Manufacturing

3.5.1 Production

Produce or procure parts conforming to engineering drawings and specifications, methods, procedure and processing improvements and provide documented control over manufacturing operations.

3.5.2 Quality Control

(1) Ensure conformance to engineering drawings and specifications.

(2) Conduct quality improvement programs and vendor quality evaluations.

(3) Review documentation from Engineering, Manufacturing, and Purchasing for adequacy of data.

(4) Review qualification tests and review or conduct production tests for compliance with established engineering specifications.

3.6 All Activities

Prepare organizational instructions to implement activities in accordance with respective responsibilities as outlined above. Such instructions are to be reviewed by Manager, Product Reliability and approved by Systems and Procedures prior to publication.

Reason for Reissue: Revised throughout in line with Company's current reliability program.

Herbert Kunzel

Herbert Kunzel, President

R. S. Palmer

R.S. Palmer, Comptroller

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STANDARD PRACTICE INSTRUCTION

SOLAR

SUBJECT: FAILURE REPORTING, ANALYSIS and CORRECTIVE ACTION PROCEDURE

CLASSIFICATION: SPI - 2.490
DATE ISSUED: October 23, 1963
SUPERSEDES:

1. PURPOSE

1.1 To establish areas of responsibility and procedures for the performance of "Hardware Failure Reporting, Failure Analysis and Corrective Action", as specified in S.P.I. 6.930, paragraph 3.4.4.

2. RESPONSIBILITIES

Manufacturing

Quality Control

Sales

Engineering

3. GENERAL

3.1 When a part designed by Solar, built by Solar, or supplied by a Solar Vendor fails, a failure report shall be initiated.

3.1.1 Failure Analysis and Corrective Action shall be performed under the direction of Reliability Engineering to ensure that results are recorded for reference in maintaining established company standards and Customer required levels of reliability.

3.1.2 Malfunction or Failure reports directed to Solar Product Reliability by the Customer shall be utilized by Reliability Engineering to ensure conformance with the procedures outlined.

3.2 Definitions

(1) Failure - The inability or cessation of ability of an item to meet minimum specified performance; or any unscheduled adjustment or replacement of hardware (other than a scheduled calibration).

(2) Secondary Failure - Failure resulting from a failure of another part.

(3) Failure Report - Initial document used to report a failure and request for a failure analysis if necessary.

(4) Failure Analysis - Investigative action performed as directed by the Reliability Engineer, with support action provided by other departments as necessary.

(5) Corrective Action - Specific action initiated on failures after a failure analysis has been performed to prevent recurrence by elimination of the basic cause.

4. PROCEDURE

4.1 Manufacturing

4.1.1 Manufacturing Departments

(1) When a failure is detected by a production employee, or when there is concern that a failure may have occurred, the Department Supervisor shall be notified.

(2) The Supervisor shall immediately notify Quality Control.

4.1.2 Quality Control

(1) Upon detection or notification of a failure the Inspection Supervisor shall withdraw the failed part from production and,

1. Prepare a witholding notice in accordance with S.P.I. 4.989.

2. If a failure has occurred as defined by paragraph 3.2 (1), take the necessary action to ensure that the failed part is kept in its exact "failed condition" and present to Materials Review.

(2) Upon presentation and review of items received on a witholding notice, Materials Review shall:

1. If a failure has occurred as defined by paragraph 3.2 (1), complete a Failure Report Form, SACO 2164A in Duplicate and forward one copy to Reliability Engineering.

2. Affix a "Failure - Hold for Analysis" tag, SACO 2168, to the failed part.

(3) Upon presentation and review of Customer Returned items, Materials Review shall:

1. Provide suitable display of returned material enabling the Reliability Personnel to determine items for which failure analysis is required.

2. Affix a "Failure - Hold for Analysis" tag, SACO 2168, to the failed parts so determined by the Reliability Personnel.

4.1.3 Receiving

4.1.3.1 Upon receipt of items returned from Customer or Field Sales/Field Service Personnel, the procedures outlined in S.P.I. 1.391 (Aerospace), and S.P.I. 1.392 (Turbomachinery), shall be followed.

4.2 Sales

4.2.1 Field Sales - Aerospace

(1) When a failure in the field is detected, if a formal Reliability, Failure Analysis Feedback program is required by Contract under which items were manufactured, then the following is to be provided:

Note: Information to be provided on Solar Form, SACO 2164A, if available.

1. Customer Name
2. Name and Title of person reporting.
3. Failed item name.
4. Failed item part number.
5. Failed item serial number.
6. System type/Model/Series .
7. Replacement part number .
8. Replacement part - Serial number.
9. How failure was detected (e.g. check-out, normal operation).
10. Failure date.

11. Description of Trouble

12. Whether a failure analysis was performed.

Note: One copy of the above information should be forwarded to Solar Reliability Engineering.

(2) If the Customer performs a failure analysis, he should provide Solar with the following information, in addition to that required under paragraph 4.2.1 (1):

1. Name and Title of person performing analysis.

2. Known internal environment; i.e. temperature, pressure, etc.

3. Known external environment, i.e. temperature, pressure, etc.

4. Visible failure indications.

5. Whether failure was "primary" or "secondary".

6. If failure was secondary, what was its origin.

7. If primary, what was its origin, e.g. poor design, etc.

8. Recommended corrective action Solar should take.

9. Forward one copy of the above information to Solar, Reliability Engineering.

4.2.2 Customer Service - Turbomachinery

(1) When a failure occurs in the field and a representative of Customer Service is present, he shall initiate a Field Service Report, SACO 1773, and forward to Solar.

(2) Customer Service Representative shall tag the failed part with a "Field Returned Part" Identification tag, SACO 1333, and return the part to Solar.

(3) Upon receipt of Field Service Report(s), Customer Service department shall prepare a Failure Report, SACO 2164A, and forward one copy to Reliability Engineering.

(4) Provide a copy of all "URs" (Unsatisfactory Report), to Reliability Engineering for evaluation.

(5) Request Commercial Customers report on "Turbomachinery Malfunction Report", SACO 2114.

(6) Procedure outlined in paragraph 4.2.1 shall also apply if SACO 2114 is not available.

4.3 Engineering

4.3.1 Failure Analysis

(1) Failures occurring during any phase of "Turbomachinery - Engineering Development" shall be reported on Failure Report, SACO 2164A, and one copy forwarded to the Reliability Engineer.

(2) When part(s) supplied by outside Vendor(s) are subject to failure analysis, Reliability Engineering shall determine if an analysis by the Vendor(s) is required. If any, he shall request such analysis through Purchasing.

(3) Reliability Engineering shall ensure that the Vendor(s) is provided all possible information regarding the failure.

(4) The Reliability Engineer shall provide a copy of the failure report and any necessary instructions to the Analyzing Activity, who shall perform an analysis to determine the ultimate failure mode and cause. This shall be reported on SACO 2164.

(5) The Analyzing Activity shall solicit any assistance deemed necessary in order to complete the failure analysis.

(6) Upon completion of the failure analysis, the Analyzing Activity shall forward his report to the Reliability Engineer for review.

4.3.2 Corrective Action

(1) The Reliability Engineer shall forward a copy of the Failure Analysis report with recommendations for corrective action to the Project Engineer and Manager Product Reliability for review.

(2) The Project Engineer shall initiate corrective action and advise the Reliability Engineer of action taken.

(3) Upon notification from the Project Engineer of the necessity for corrective action, such action shall be taken by the activities so designated.

(4) Completed corrective action shall be noted on SACO 2164 and returned to the Reliability Engineer.

4.3.3 Reliability Engineer

(1) The Reliability Engineer shall be responsible for the economical actuation and maintenance of the system.

(2) The Reliability Engineer shall maintain the following records:

1. A card file of failed parts by part numbers, SACO 2167.

2. A card file of failure codes and parts that have failed in each such coded classification, SACO 2166.

3. A card file of failed parts listed by Customer, SACO 2165.

4. Completed failure analysis and corrective action reports.

(3) Development and maintenance of a "Failure Coding System" capable of providing, but not limited to:

1. Suitable display of experience data to:

a. Increase inherent reliability of new designs.

b. Determine requirements for improvement of production procedures, processes, and inspection methods.

c. Indicate necessity of preventive maintenance and its timing.

d. Indicate areas in which additional personnel training is required.

2. Identity and quantitative requirements forecast for spare parts.

3. Turbomachinery or Aerospace system reliability data, values, and confidence factors suitable for prediction and warranty determination.

(4) Conduct a monthly review of failures and distribute a summary of reliability status to departments concerned.

(5) Issue special reports on failure trends and Experience Retention Studies as deemed useful for corrective actions and for guidance in new designs.

4.4 Forms

SACO 1333	Field Returned Part Identification Tag
SACO 1773	Field Service Report
SACO 2114	Turbomachinery Malfunction Report
SACO 2164	Failure Report, Analysis and Corrective Action
SACO 2164A	Failure Report - Half Sheet of SACO 2164
SACO 2165	Customer Service Card
SACO 2166	Failure Code Card
SACO 2167	Failed Part Card
SACO 2168	Failure - Hold for Analysis Tag

Herbert Kunzel, President

R. S. Palmer, Comptroller