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SUBJECT: HIGH PRESSURE FLANGE STUDIES

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NOTICE

This document contains information of a preliminary nature and was prepared primarily for internal use at the Oak Ridge National Laboratory. It is subject to revision or correction and therefore does not represent a final report.

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1.0 ABSTRACT

During the 1956 Spring quarter, 2500 psi ring type flanges, ring gaskets, bolts, and special connectors were tested for adaptability to the aqueous homogeneous reactor. The testing procedure and results are presented.

2.0 SUMMARY

High pressure line closures were studied to obtain empirical data pertinent to the selection or design of a connector capable of withstanding sustained thermal cycling and high pressures encountered in the aqueous homogeneous reactor. Specialized stress-strain measurement techniques yielded information concerning flange deformation, ring type gaskets, bolts, and special connectors.

The results indicated that no totally acceptable connector is currently available. Most promising of the combination of components tested during this period was a 2500 psi ring type flange with an accurately machined octagonal gasket and Grade B-7* bolts.

3.0 INTRODUCTION

The fission products present in the chemical processing plant of the aqueous homogeneous reactor provide an extreme radioactivity hazard. A leak would therefore be a serious problem. In addition to this safety consideration, maintenance is difficult and must be kept to a minimum by eliminating sources of trouble and providing facility of maintenance for those which can not be eliminated.

Figure 3.1 shows the flanged joints which allow removal of the homogeneous reactor components that may need attention. These constitute an important source of possible leaks. A comprehensive series of high pressure flange tests to determine the characteristics and best application of the closures to be used was therefore begun.

During the 1956 Spring quarter a number of high pressure flange tests were made concerning the following aspects: a) flange deformation, b) bolts, c) flange gaskets, and d) special connectors. This report describes these tests and presents the results obtained. It should not be construed that this report represents the final conclusion of the high pressure flange studies. It is an intermediate report and the data presented are only part of that to be obtained.

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* ASTM 193-53T-B7.

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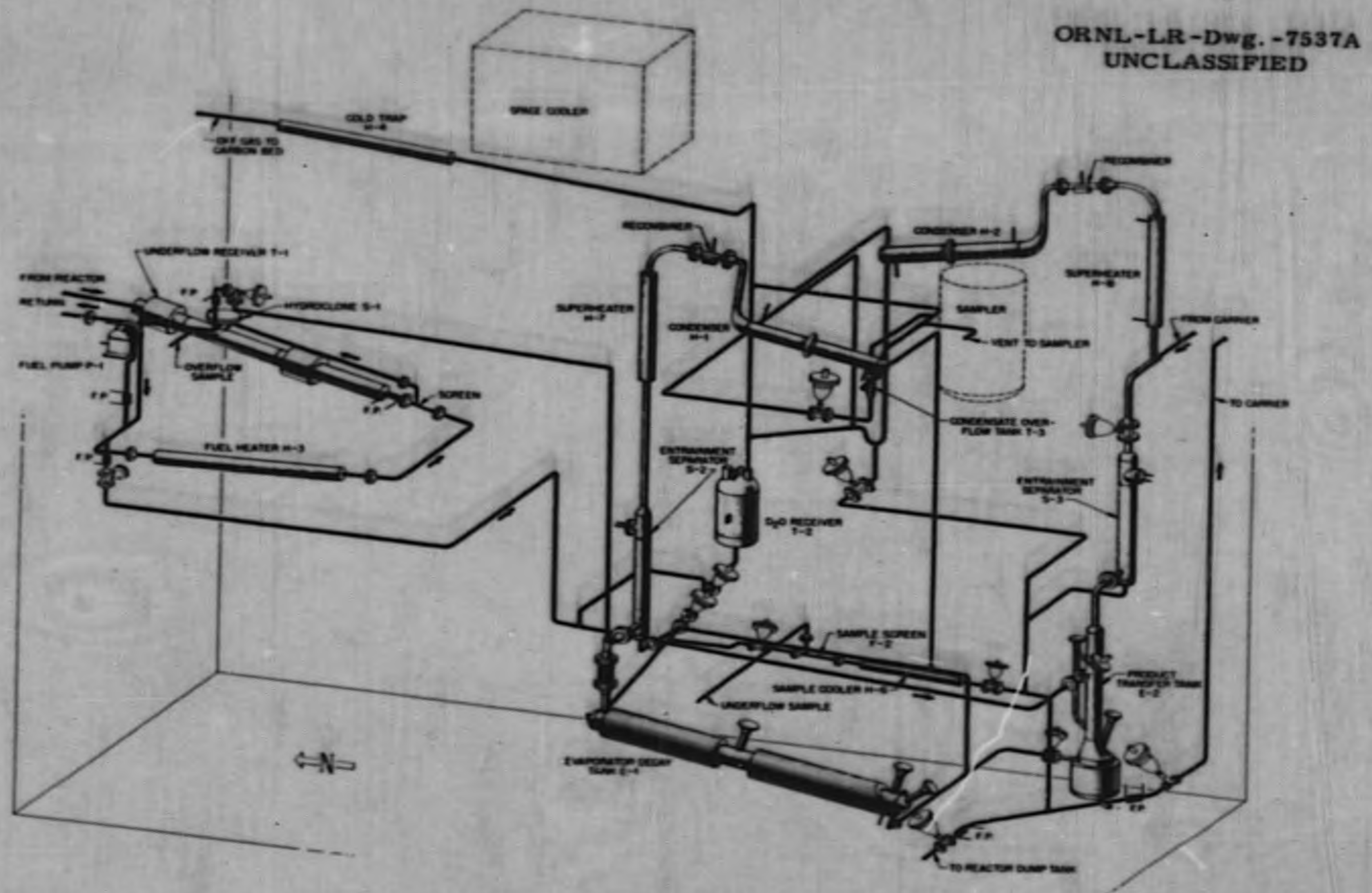

Figure 3.1

5

4.0 FLANGE DEFORMATION

Two types of deformation occur in flanges. These are the permanent grooves impressed on the flange seat by the ring and the temporary bending or flexing which occurs only while the flange is loaded. Both of these types of deformation are of interest. Unfortunately only the permanent deformation, the impressed groove, lends itself to the measurement techniques employed in these studies. This section is a discussion of the nature and implications of this permanent deformation.

The yield point of type 347 stainless steel is 40,000 psi*. This yield point is frequently exceeded on the seats of an oval gasketed flange assembly (such as shown in Figure 4.1). Permanent deformation of the flange seat, the ring gasket, or both, then occurs along the "line" of contact. When bolt loadings are maintained below a total of 60,000 pounds, the deformation may be considered within the elastic limit of the flange and ring. Higher loading produces the effect shown in Figure 4.2. The gasket begins to slip along the flange seat as the yield point is exceeded and a permanent deformation of the type shown in Figure 4.3 is produced. This ring slip has the strange property of producing no increase in bolt load when the nut is tightened. Reference to Figure 4.2 again shows that the ring slip regains its linear relationship with loading after sufficient tightening of the nut. But this linear relationship now in the inelastic range (the permanent deformation having occured at the break in the curve) and an undesirable marring of the flange seat and ring has been produced.

Because no leaks are tolerable in HRT closure and such a marring of the contact surfaces (the seat and ring) would produce a leak as the joint cools and contracts. The foregoing ring slip considerations imply a maximum permissible load of 60,000 pounds per flange if an oval ring gasket is used. The performance of an octagonal ring gasket is included in Figure 4.3 for comparison.

5.0 BOLTS

The 3/4" bolts to be used in the HRT flanged closures are a high tensile strength carbon steel with a yield point of approximately 100,000 psi. A recommended maximum stress is 20% of this value. These figures yield a safe loading of 6000 pounds per bolt.

Recall that the chemical processing plant will be thermally cycled from room temperature to 300°C*. The thermal stresses introduced at the 300°C temperature will exceed by three times the recommended safe limit of 6000 pounds load per bolt, if no preventative measures are taken. Therefore, tests were made to determine what convenient preventative measures are available.

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* Perry, Chemical Engineering Handbook, p. 1539.

* See Appendix 8.3.

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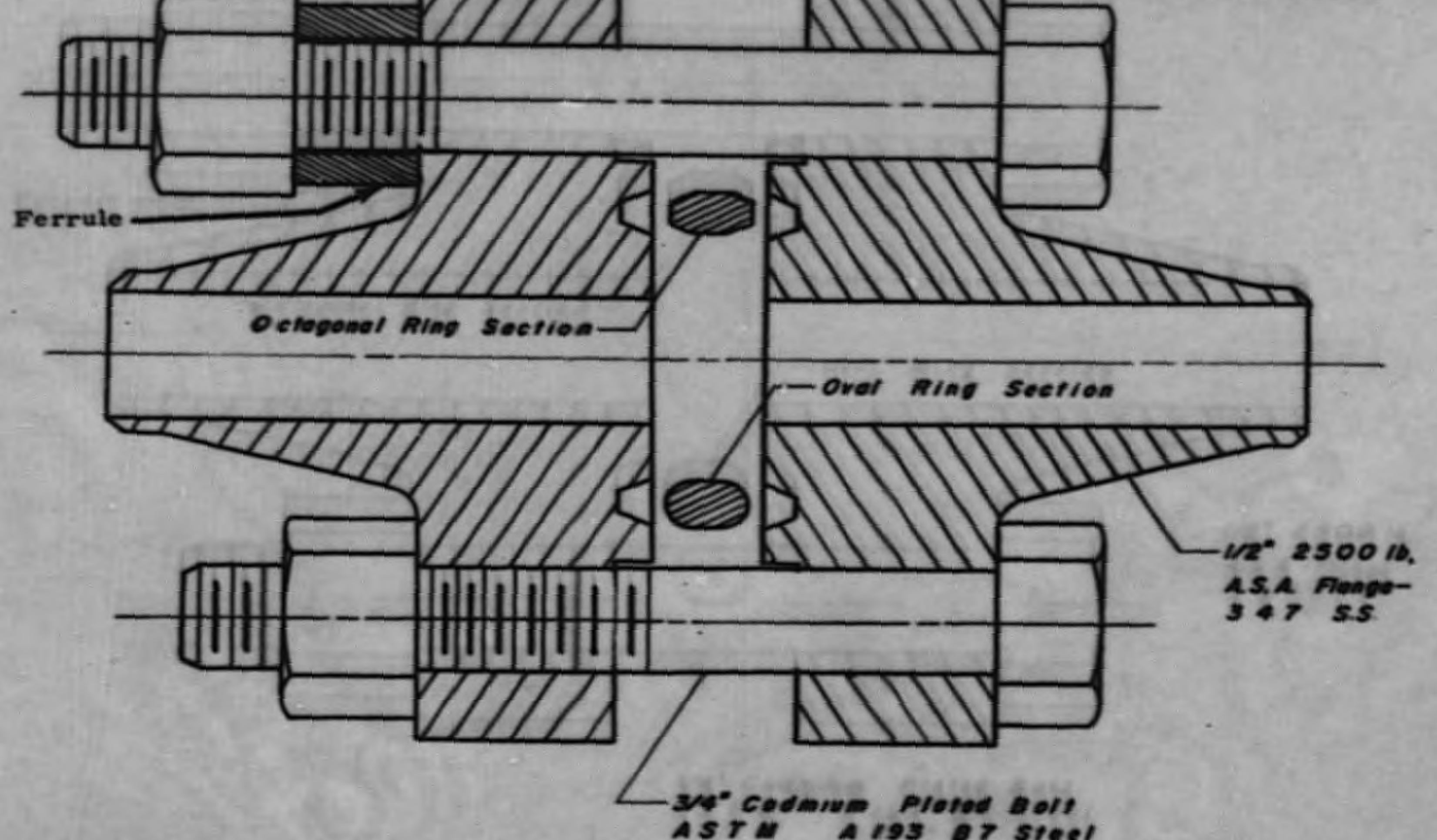

Figure 4.1 Representative High Pressure Flange Components

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Figure 4.2

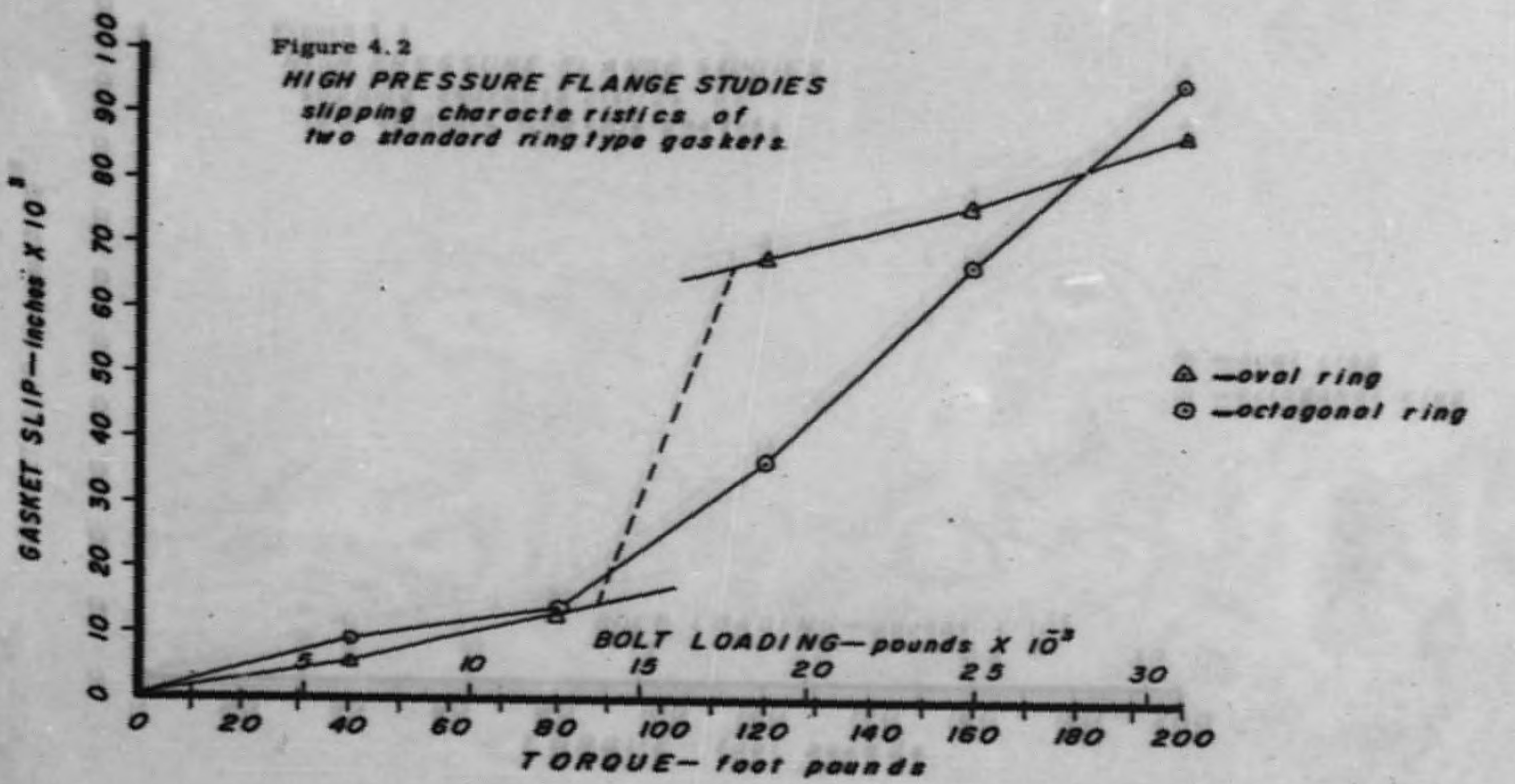

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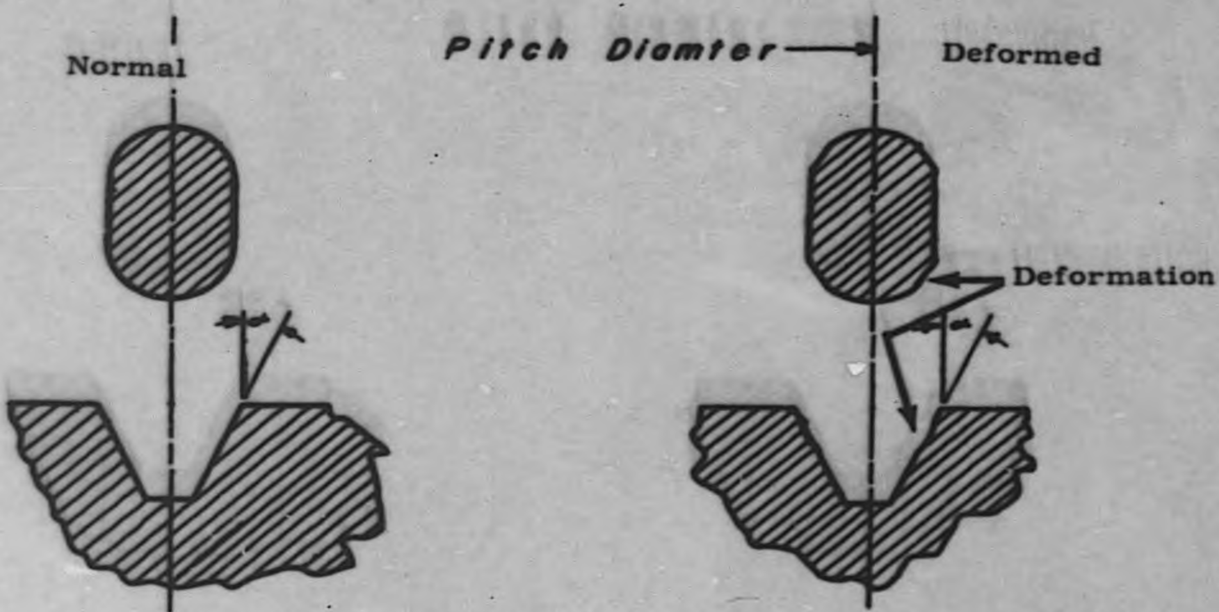
Figure 4.3

HIGH PRESSURE FLANGE STUDIES



SECTIONS THROUGH FLANGE
GROOVE and RING GASKET

Stress-strain-torque relationships, the elastic limit, and other physical characteristics of the bolt were studied.

5.1 Testing

Two 3/4" bolts were destructively tested along 3.5 inches of their length by applying even increments of torque from 0 to 350 foot pounds and measuring bolt elongation. The values obtained are listed in Table 5.1 and plotted in Figure 5.1. The inconsistency between test two and the others was traced to a crack in the bolt.

Consistency was obtained in a series of stress-strain-torque tests made within the elastic limit. These tests demonstrated that although some changes in the bolt and nut contact surfaces occured with each repeated tightening, the torque-loading relationship shown in Figure 5.1 is valid without correction for several bolt re-uses.

5.2 Reduction of Thermal Loading

Two methods of reducing thermal loading of the bolts are being investigated. These are: 1) lengthening the bolt, and 2) employing spring washers.

When a bolt is loaded it obeys Hooke's low within the elastic limit. That is: Elongation $\propto$ stress x length. It is apparent, therefore, that by doubling the length of a bolt, a given elongation is produced by only half the stress formerly required. Thus using longer bolts in a high pressure flange would significantly reduce thermal stresses. In addition, if a ferrule (see Figure 4.1) with a low thermal expansion coefficient were used, thermal stresses could be reduced to practically zero. A discussion of the application of thermally corrected flanges is found in section 7.2.

A spring washer to be placed between the flange, and the bolt head and nut, is designed, but has not been evaluated.

6.0 RING GASKETS

Application of ring type flanged closures usually require that only one line of contact be made between the flange and gasket. That is the ring gasket contacts the flange on the inner or the outer seat; usually not both. The chemical processing plant application, however, require that maximum protection from leaks be obtained. Therefore, sealing the closure on both seats is desirable. Once sealing on both seats has been obtained, the cavity between the two lines of contact can be pressurized to above the plant operating pressure. Any leak which developed would be inward and could be detected by the drop in pressure which would occur in the cavity.

Initially obtaining and then maintaining the required double seal during thermal cycling made ring gasket studies necessary.

Table 5. 1

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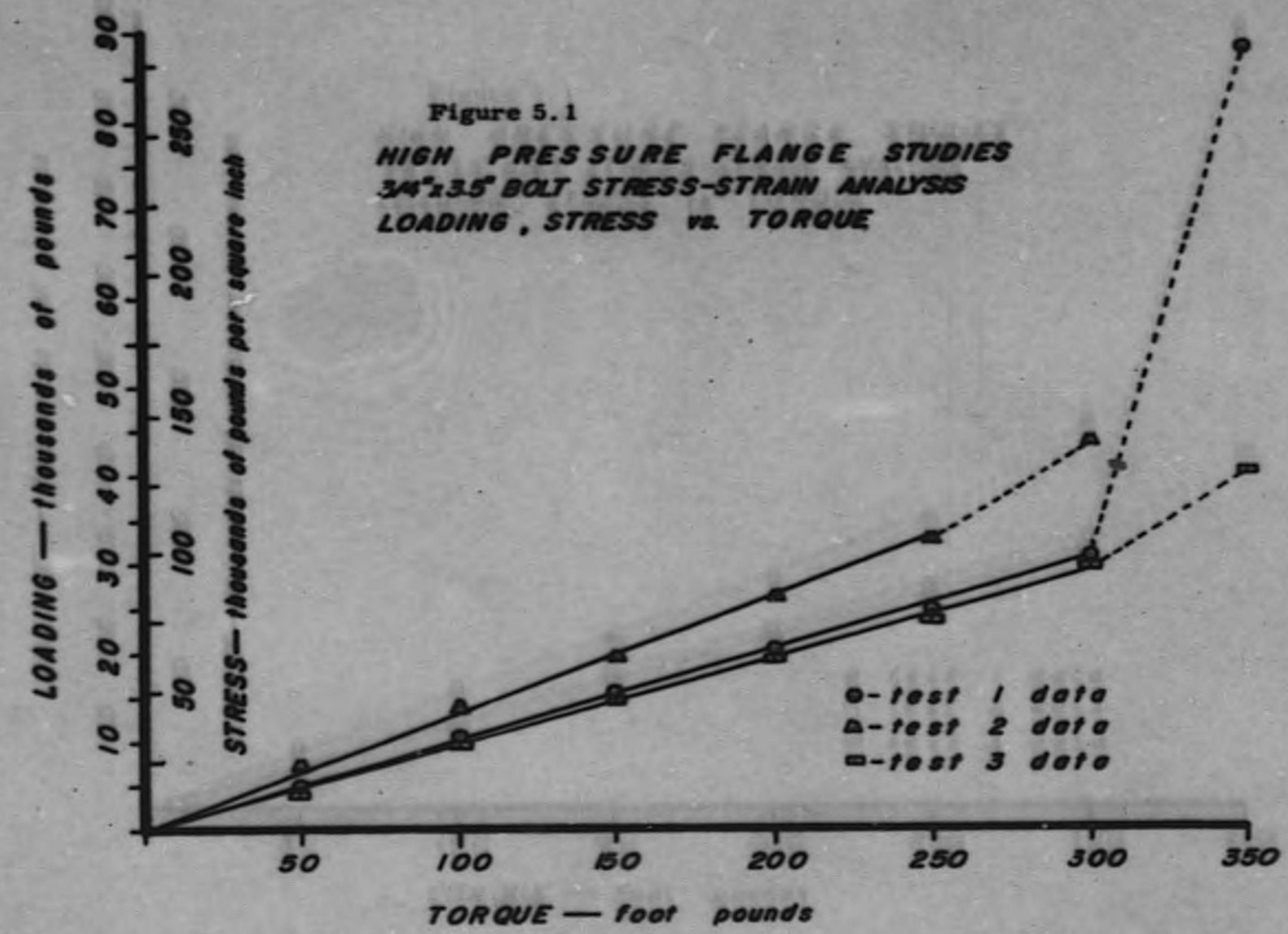
HIGH PRESSURE FLANGE STUDIES

Bolt Stress-Strain Data

TORQUE	TEST #1			TEST #2			TEST #3		
FOOT POUNDS	LOAD (lb)	STRESS (psi)	$\frac{dE}{dT}$	LOAD (lb)	STRESS (psi)	$\frac{dE}{dT}$	LOAD (lb)	STRESS (psi)	$\frac{dE}{dT}$
50	5,027	16,373	3.8	7,144	23,268	5.4	4,763	15,513	3.6
100	10,319	33,608	4.0	13,493	43,947	4.8	9,789	31,883	3.8
150	15,346	49,981	3.8	19,500	63,512	4.6	14,552	47,396	3.6
200	20,100	65,466	3.6	26,723	85,571	5.4	19,048	61,942	3.4
250	24,605	80,138	3.4	32,808	106,855	4.6	23,018	74,970	3.0
300	30,691	120,215	4.6	43,126	140,461	7.8	29,103	94,788	4.6
340	----	----	----	BOLT	BROKE		----	----	----
350	84,000	273,588	40.0	----	----	----	39,687	129,260	8.0
360	88,000	286,616	----	----	----	----	----	----	----

BOLT DESCRIPTION: CADMIUM PLATED, 3/4 in. x 3.5 in. (ENGAGED LENGTH) ASTM A193 GRADE B7.



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6.1 The Oval Ring

An oval ring of the type shown in Figure 5.1 is the gasket supplied for use in industrial high pressure flange application such as steam plants. This was the first type of ring tested. The tolerance to which these rings are made shows that re-machined rings are required. This is illustrated in Table 6.1.

Initial sealing of flanges with re-machined rings was obtained by applying bolt loadings which exceeded the 6000 pound maximum in many cases. These rings were then tested by thermally cycling the assemblies from 20°C to 300°C. Leaks developed which may be attributed to one or a combination of the following:

1) Permanent deformation of the flange seat.
2) Scarring and galling caused during tightening.
3) Warping from uneven heating.
4) Imperfections in the ring casting.

Permanent deformation of the flange seat occurred in every case of leaking.

6.2 The Octagonal Ring

Deformation of the flange seat was the major consideration in oval ring gasket leaking. Therefore, the obvious advantage of flat contact surfaces led to testing a ring gasket of octagonal cross section. This ring gasket distributes the force, thereby reducing the pressure against the flange seat to below the yield point. Now no permanent deformation can occur.

Two ring gaskets of octagonal cross section were tested by sealing and then thermal cycling them from 20°C to 300°C under 2500 psi pressure. Satisfactory performance was obtained. The shortest complete thermal cycles were two hours in duration.

6.3 Special Rings

Three special ring gaskets were designed. These gaskets are shown in cross section in Figures 6.2 and 6.3.

6.3.1 The Finned Octagonal Ring

The octagonal ring with its larger contact surface area requires a slightly greater bolt loading to effect a seal. Because it is desirable to limit initial bolt loading to 6000 pounds, a compromise between the oval and octagonal ring gasket is made in the finned octagonal ring. This gasket has not been tested to date.

6.3.2 Split Rings

The two split rings shown will make separate mechanical seals on the inside and outside flange seats. In addition to making sealing possible with less loading, the split in these rings will provide some protection against thermal warping. These rings have not yet been tested.

Table 6.1

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HIGH PRESSURE FLANGE STUDIES

Measured Pitch Diameters of Sample Lots of Ring Gaskets

COMMERCIAL RING GASKETS TOLERANCE ±0.005 in.		COMMERCIAL RING GASKET RE-MACHINED TOLERANCE ±0.002 in.	
P_D (MEASURED) (in.)	NO MEASURING	P_D (MEASURED) (in.)	NO MEASURING
1.688	2	1.684	1
1.689	3	1.685	1
1.690	8	1.686	3
1.691	3	1.687	3
1.692	7	1.688	25
1.693	0	1.689	3
1.694	3		
1.695	4		
TOTAL	30		36

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Figure 6.2

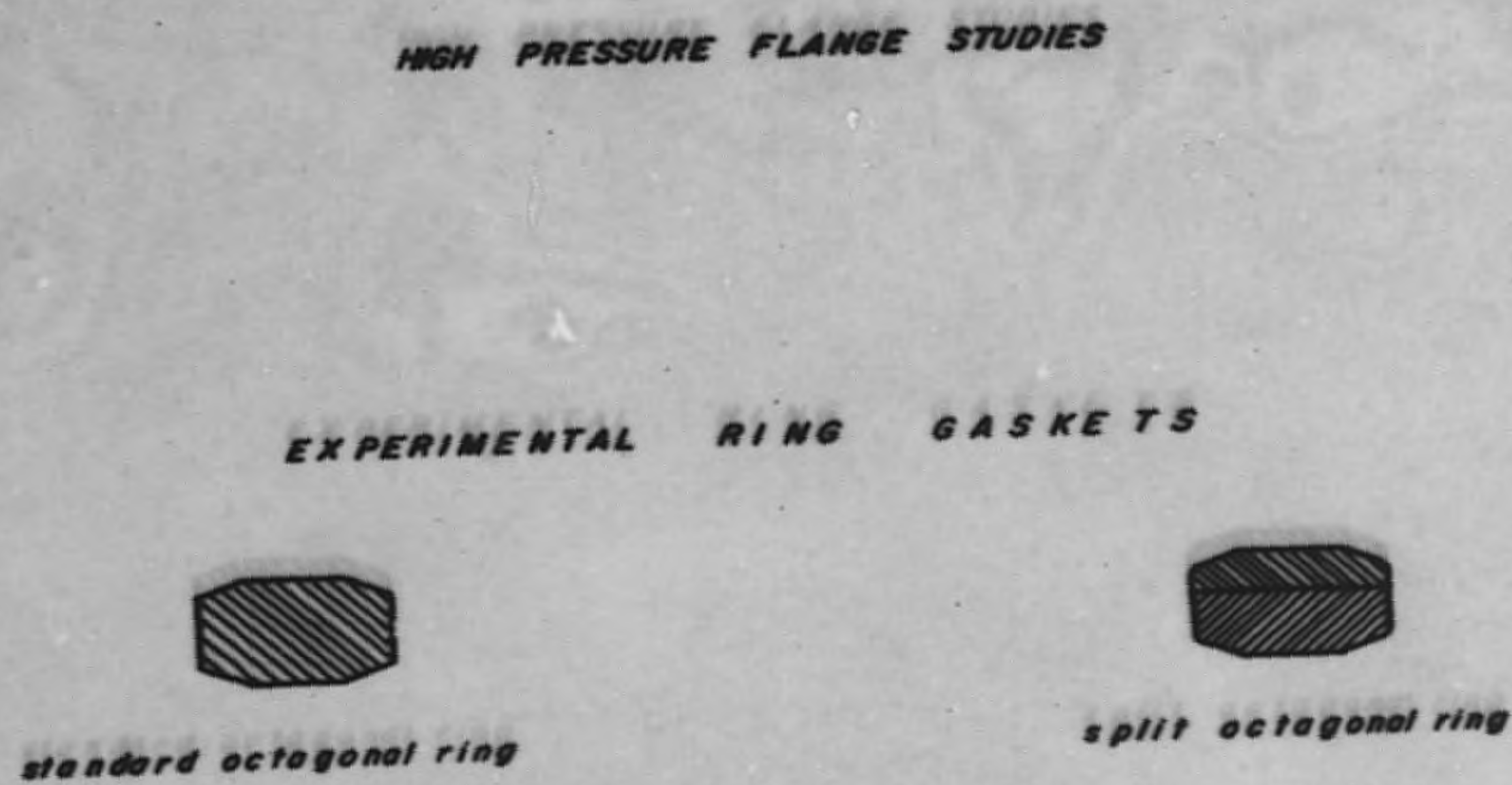

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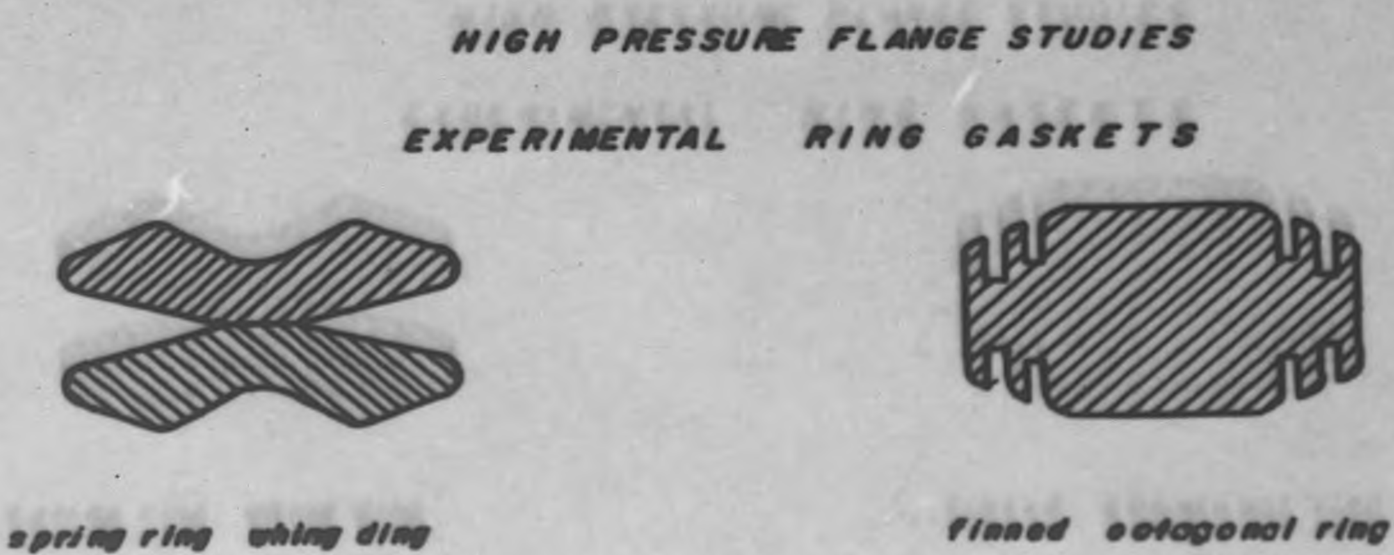
Figure 6.3



7.0 SPECIAL CONNECTORS

A modified Graloc connector and a thermally corrected flange assembly were tested. Neither has been successful to date.

7.1 The Graloc Connector

Used successfully in the petroleum industry, the Graloc connector seemed a logical connector for application in the HRT Chemical Processing Plant. Modification in the connector resulted in the closure shown in Figure 7.1. Contact on the inner sealing surface could not be obtained. Further modifications are now underway.

7.2 Thermally Connected Flange

Because thermal expansion causes an overloading of bolts, a flange which has zero relative expansion with respect to the bolt would be desirable. Such a closure was designed, built, and tested. Mechanical alignment of the components of this closure is very difficult. This will necessitate re-design before any successful tests may be made. When properly modified, this flange should prove most interesting.

8.0 APPENDIX

8.1 Stress-Strain Measurement Technique

The very small changes in lengths resulting from stresses in flange components made specialized measurement techniques necessary. This section describes these techniques.

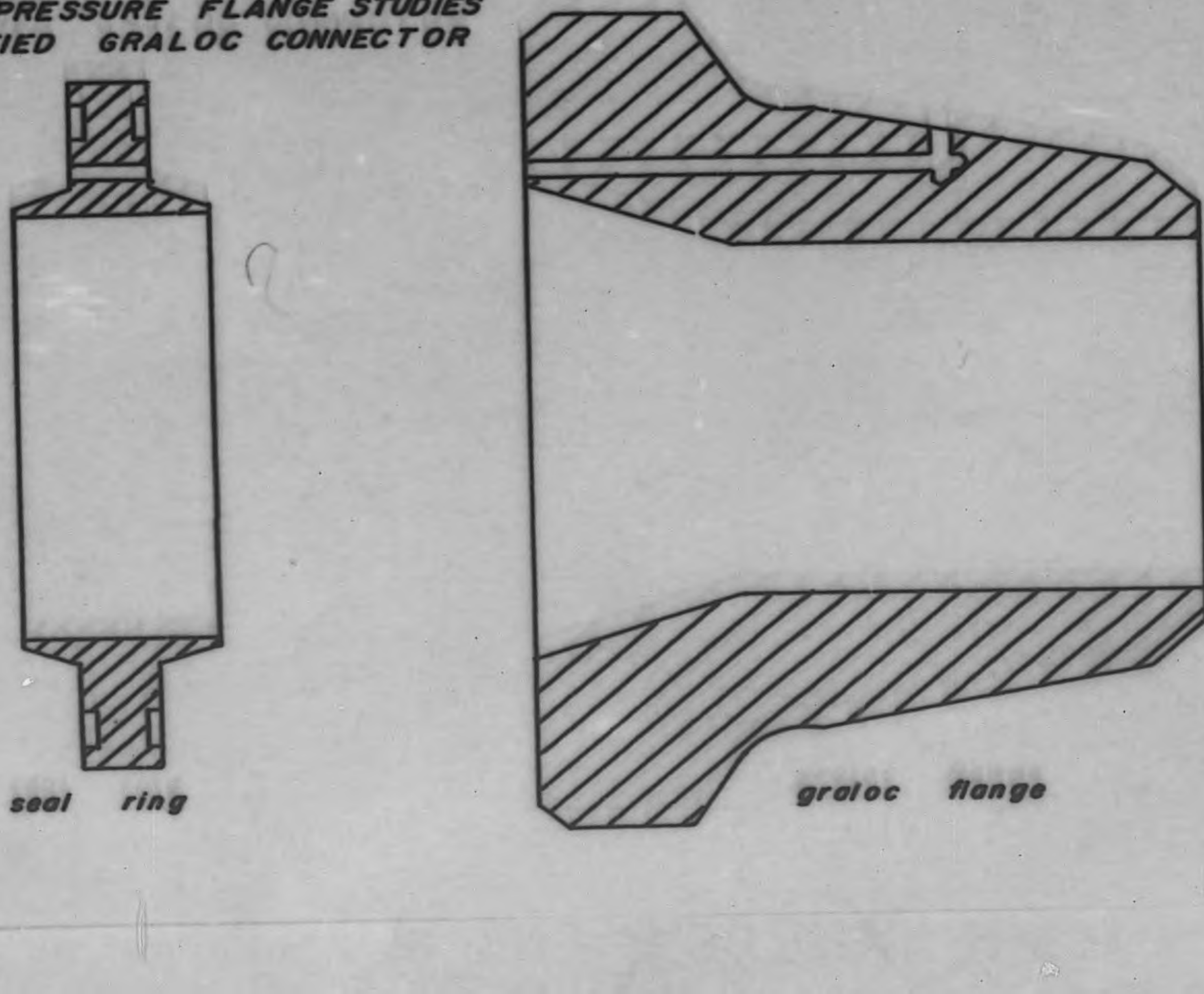
8.1.1 Measurement Methods

Two of the several methods for measuring stress-strain relationships in bolts were considered. The more accurate of the two is employment of an electronic strain gage to make direct readings. This method requires an elaborate preparation for each test, however, and was discarded because many tests were to be made.

Direct employment of the relationship $Y = \frac{\text{stress}}{\text{strain}}$, where Y is Young's Modulus, constitutes the second measurement method. Strain, in addition to being determined directly from bolt elongation, may be found as a linear function of angular nut rotation and as an approximately linear function of torque applied to the nut. Bolt elongation provides the most accurate correlation used between stress and torque. Angular rotation of the nut is too coarse a measurement device for this application.

Using Young's Modulus in conjunction with bolt elongation and torque measurement, data were accumulated and plotted. Figure 7.1 is an illustration of this.

FIGURE 7.1
HIGH PRESSURE FLANGE STUDIES
MODIFIED GRALOC CONNECTOR



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8.1.2 The Special Micrometer

To provide bolt elongation data accurate to 10^{-4} inches, a high quality 6 inch micrometer was used. The first readings were made on bolts whose ends had been carefully cut parallel. Alignment of the micrometer proved very difficult, however, and a reproducibility of readings was not obtained with the desired accuracy.

To eliminate the alignment problem, conical holes were cut in the ends of the bolts and the micrometer was fitted with spherical contacts. This provided automatic centering which proved extremely valuable during hot testing when rapid readings were necessary to prevent significant thermal expansion. Nc measureable loss of precision was found to occur due to grit and dust accumulation in the conical holes at either end of the bolts. The holes were cleaned frequently, however.

8.2 Typical Oval Ring Test Procedure and Results

Two high pressure flanges with standard oval rings were cycled from 100 deg. C. to 300 deg. C. seventeen times between March 21 and May 3, 1956. These flanges were dismantled and inspected during the past week. The following is the report of that inspection.

Run No. 2 - Summary

Description of Flanges and Rings on Disassembly

Flange # 7-A; 7-B   Unit # 7

The ring was inspected and the inner seals (top and bottom) were found remarkably uniform. The width of the seal lines was the same for both; approximately 1/32" wide. This is apparently an example of an excellent mechanical fit. One inside surface showed a slight amount of galling.

The outer lines of contact were similarly uniform and were scaled at 3/64" width.

The flanges had similarly uniform inner and outer seals. The lines of contact were: inside - 3/64" maximum and the outer was slightly greater. The grooves are deep and smooth with slight galling evident.

Flange # 3-A; 4-B   Unit # 7

The ring and flanges were inspected as above. All seal lines were of the same approximate width and as uniform as those above in all respects.

Microscopic examination of the rings showed a finish on the line of contact equal in smoothness and general homogeneity to the adjacent machined surfaces. There are scattered pits of about 1/100 inch mean diameter. These pits appeared to be imperfections in the casting. At several places grooves of about 1/500 inch wide and up to 3/4 inch long were found. The grooves originated at one edge of the contact line and crossed to about the center before returning to the same edge. None of these grooves crossed more than half the width of the line of contact; however. At one point on the ring intense heat was applied while welding a thermocouple lead to the flange. This area appeared to be a bed of small cinders under 60 magnifications.

The line of sealing appeared to be divided into two distinct parts as if two separate seals were made under different bolt loadings. The upper half of theline (the half appearing to be made under the smaller loading) has a mottled appearance indicating that only part of its surface was in contact during the hot part of the test when discoloration presumably occurs. In contrast, the bottom half of the line has retained a distinct new metal color.

Flange unit # 7, which did not leak during cycling, was reassembled and tested for sealing under 3000 psi (water) at room temperature. Under a bolt loading of 6600 pounds per bolt a leak of 120 psi per hour was recorded. At 9000 pounds load per bolt a pressure drop of only 7 psi per hour occurred. The flange was then broken and remade immediately. The test to determine the loading required to make this seal after it has been broken twice is now in progress.

8.3 Calculated Bolt Stresses Introduced By Thermal Cycling

(Refer to Table 8.1)

8.4 Co-Op Personal Evaluation

The successful co-op program from the students point of view must fulfill two major criteria: 1) it must introduce him to the principle of engineering, 2) it must allow him to apply what he has learned. Both of these requisites were amply fulfilled during this first co-op period at ORNL. I feel deeply indebted to those who provided such a very enjoyable and educational quarter.

Table 8.1

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HIGH PRESSURE FLANGE STUDIES

Calculated Bolt Stresses Induced by Thermal Cycling

CONDITIONS: Flanged Joint Assumed Fully Rigid
Reactor Temperature - 575°F
Room Temperature - 75°F
Reactor Pressure - 2000 psig

FLANGED CONDITIONS		BOLT CONDITIONS		DIFFERENTIAL THERMAL EXPANSION in. x 10³	EQUIVALENT MECHANICAL LOADINGS (lb)	
Temp. Range °F	Thermal Expansion in. x 10³	Temp. Range °F	Thermal Expansion in. x 10³		3.2 in. Bolt	6.5 in. Bolt
FLANGED JOINT; FULLY INSULATED:						
75-575	16.7	75-575	10.2	6.5	27,000	13,500
FLANGES INSULATED; BOLTS BARE:						
75-575	16.7	75-400	6.6	10.1	41,500	20,700

THERMAL STRESS EQUIVALENT TO: 55 lb per degree °F temperature change
85 lb per degree °F temperature difference between flange and bolts

END