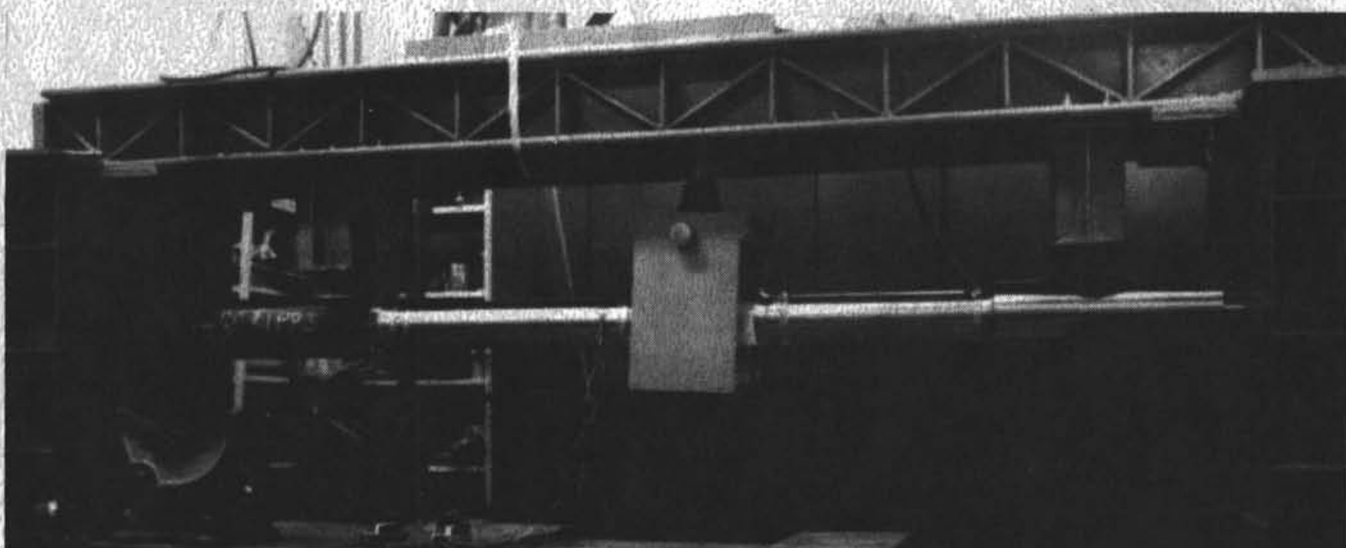
INTERNATIONAL PHASE OF OCEAN DRILLING (IPOD)
DEEP SEA DRILLING PROJECT
DEVELOPMENT ENGINEERING
TECHNICAL REPORT NO. 15

TEST AND EVALUATION OF ALUMINUM DRILL PIPE FOR DEEP WATER CORING

DESIGN AND USE OF HEAVY WALL DRILLING JOINTS FOR BENDING STRESS REDUCTION



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UNIVERSITY OF CALIFORNIA AT SAN DIEGO
CONTRACT NSF C-482
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DISCLAIMER

THE COVER PICTURE

The cover shows a horizontal heavy wall drilling joint in a test frame. The fixture applied tension loads of 400,000 pounds to the drilling joint while simultaneously bending the joint and connections to a 350-foot radius arc. Strain gages were attached to the tube wall and connection at critical points to measure stress levels. Test results proved the design acceptable for use at the critical upper end of the drill string.

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TEST AND EVALUATION OF ALUMINUM DRILL PIPE FOR DEEP WATER CORING

DESIGN AND USE OF HEAVY WALL DRILLING JOINTS FOR BENDING STRESS REDUCTION

Prepared For

NATIONAL SCIENCE FOUNDATION
Under Provisions of Contract NSF C-482

By

DEEP SEA DRILLING PROJECT
Scripps Institution of Oceanography
University of California at San Diego

February 1984

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PREFACE

Technical Report No. 15 includes two papers dealing with means of extending the drill string depth capability and lowering stresses at the top of the drill string.

The first paper titled "Test and Evaluation of Aluminum Drill Pipe for Deep Water Coring" reports on the operational use of a 2000 foot section of aluminum drill pipe in a mixed aluminum/steel drill string. The report also discusses metallurgical laboratory tests and assesses the potential of mixed strings for use in wireline coring operations to 30,000 feet.

The second technical study is entitled "Design and Use of Heavy Wall Drilling Joints for Bending Stress Reduction ". This report deals with the development and test of drilling joints used to reduce stresses at the upper end of the drill string. These stresses include static, dynamic, and bending loads. For very deep water coring operations or very slow penetration rates, 300 feet of the special joints are placed in the upper end of the drill string. The larger pipe cross-sectional area and the machined hubs design have allowed wireline coring operations to 23,000 feet with 5-inch drill pipe.

ACKNOWLEDGEMENTS

The Deep Sea Drilling Project (DSDP) gratefully acknowledges the cooperation of the Reynolds Metals Company in the testing and evaluation of aluminum drill pipe (ADP) for deep ocean wireline coring. Reynolds personnel have periodically assisted in ADP inspection and have undertaken fatigue and metallurgical tests to aid in the Project's evaluation of ADP.

Analysis and operational test results indicate the feasibility of a 30,000 foot mixed aluminum-steel drill string design for wireline coring.

The report "Test and Evaluation of Aluminum Drill Pipe for Deep Water Coring" was prepared by Mr. Don Bellows, a mechanical engineer with DSDP's Development Engineering Department.

The report "Design and Use of Heavy Wall Drilling Joints for Bending Stress Reduction" was prepared by Mr. Stan Serocki, head of DSDP's Development Engineering Department.

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TEST AND EVALUATION OF ALUMINUM
DRILL PIPE FOR DEEP WATER CORING

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SUMMARY

Analysis and operational testing show that aluminum drill pipe (ADP) can be used to extend deep ocean coring limits from 23,000 feet to 30,000 feet. The mixed aluminum-steel drill string design maintains a 4-inch drift diameter which is highly desirable for continuous wireline coring. Alternate all steel designs require tapered strings with reduced drift diameters. Fatigue and corrosion characteristics of ADP are evaluated and found to be acceptable for deep water wireline coring operations.

INTRODUCTION

The Deep Sea Drilling Project (DSDP) coring program started in 1968 as part of the Ocean Sediment Program funded by the National Science Foundation (NSF). Wireline coring is limited to about 23,000 feet by the present drill string design which is 5-inch diameter, 19.5 pounds per foot, S-135 steel. The string is not tapered and therefore maintains a constant 4-inch drift bore for efficient wireline coring operations.

Ultra-deep scientific objectives require a 30,000-foot drill string capability, preferably with a 4-inch drift bore. Such a requirement limits the use of conventional steel tapered drill strings as a practical option for a 4-inch bore. This study examines a mixed steel aluminum design which can meet the desired scientific criteria and maintain reasonable stress levels at the upper end of the drill string. The report includes a performance analysis of the mixed string including load and length limitations, resonance considerations, and an evaluation of fatigue life and corrosion. The results of laboratory metallurgical tests and operational tests are discussed.

ANALYSIS AND PROPOSED DESIGN

It has been recognized since the beginning of the Deep Sea Drilling Project that the standard 5-inch, 19.5 pound, S-135 steel drill pipe would limit drill string length to about 20,000 feet. Actually, string lengths of 23,000 feet have been deployed, but special precautions were taken such as using a picalo to reduce bending stresses in the moon pool, using new pipe at the top of the string, and operating in calm seas. The moon pool is the well immediately below the rig floor which is open to the sea and through which the drill string is deployed. The ship has a built-in horn shaped guide shoe in the moon pool which insures that the drill pipe is bent uniformly and gradually during pitch and roll motions and is not bent sharply around any edges. The radius of the guide shoe is 350 feet. Under high loading conditions a picalo is sometimes used. It is a section of tapered pipe that is used in the guide shoe and which restricts the radius of curvature of the drill pipe passing through it to about 450 feet, thereby decreasing the bending stress. The guide shoe and picalo are shown in Figure 1. Even under the special conditions noted above, the 23,000-foot string of 5-inch pipe is used at 90 to 95 percent of the yield strength.

Various options are available to permit the use of long drill strings:

1. Tapered drill string using larger diameter pipe at top of string.

2. "Tapered" drill string using same outer diameter pipe throughout but with greater wall thickness at top of string.

3. Mixed drill string using pipe of different materials, e.g., steel and aluminum.

Options 1) and 2), which were analyzed in the Mohole Report (Reference 1), have some inherent practical difficulties such as non-standard sizes, distinguishing among the various sizes and wall thicknesses in operational use, and maintaining a 4-inch drift bore for efficient coring with the present hardware. A concentrated analytical effort has been devoted to the mixed drill string using steel and ADP. An investigation was conducted as to the technical feasibility of using 5-inch ADP in conjunction with the 5-inch steel pipe already in use on the ship GLOMAR CHALLENGER. The items considered in this investigation were strength, fatigue life, static and dynamic loading, resonance, corrosion, shipboard handling, quality control at the manufacturer's extrusion plant, and in-service experience by various oil companies. It should be noted that mixed strings are not a new innovation; they have been used by many drilling companies (Reference 2). The unique aspect of this investigation is the configuration of a 30,000-foot mixed drill string while maintaining the 4-inch bore.

The ADP is manufactured by Reynolds Metals Company (RMC). Some pertinent characteristics of the pipe are (See Figure 2):

Length of joint (ft. nominal)...............	30
Outer diameter of body of pipe (in).........	5.150
Wall thickness of body of pipe (in).........	0.525
Weight per foot in seawater (lbs)...........	9.2
Diameter of tool joint (in).................	7.0
Minimum yield strength (psi)................	58,000

The aluminum is type 2014-T6. The inside diameter of the pipe is constant at 4.1 inches. The outside has a 41-46 inch long tapered transition at each end which brings the outer diameter from 5.150 inches on the body to 5.688 inches at the tool joint. The in-water weight of 9.2 pounds per foot compares with 18.8 pounds per foot for 5-inch steel pipe. More details on dimensions and weights for the ADP can be found in Reference 3.

The configuration of a 30,000-foot mixed drill string would be 5,000 feet of 5-inch steel pipe at the bottom, then 10,000 feet of 5-inch ADP, 10,000 feet of 5-inch steel, and finally 5,000 feet of 5.5 inch steel pipe. This represents a slight tapering of this mixed string. However, the 4-inch minimum bore is still maintained. (This configuration is shown schematically in Figure 3). The 5,000 feet of steel pipe at the bottom of the string is used so that the ADP does not enter the hole and become abraded in chert or basaltic intervals. Reynolds has specified that mixed strings of steel and aluminum should include at least 5 percent aluminum, which condition is obviously satisfied. The objective of the configuration described is to use each of the major sections of the string to its full capability which has been set at 90 percent of the yield strength to allow a ten percent margin for safety.

Fatigue life is also an important consideration when working from a drilling ship. Two types of dynamic stress and the resultant effect on the fatigue life of drill string material must be monitored. Axial dynamic stress results from the heave motion of the ship, and bending dynamic stress is caused by the pitch or roll of the ship. The axial dynamic stress is present during the total time that the string is deployed, but decreases in magnitude lower in the string. The bending dynamic stress is present only in that section of pipe which is passing through the guide shoe. Once a particular length of pipe has been lowered beneath the keel, the bending stress is no longer present. Therefore, it is desirable, when drilling, that the rate of penetration be such that one section of pipe is not subjected to long intervals of bending in the guide shoe.

COMPUTER MODEL

DSDP has developed a computer model of the drill string which predicts stresses, displacements, and fatigue life for any element along the length of the drill string. Figures 4 through 10 represent a run of the computer program for the 30,000-foot mixed string configuration discussed previously. Figure 4 lists the setup data that is interactively entered into the computer. It shows no heave compensation, velocity dependent hydrodynamic damping, and a Pierson-Moskowitz sea spectrum. Figure 5 shows the amplitude of displacement along the string with wave height as the curve index. Figure 6 shows the velocity along the string. The top displacement and velocity is the heave motion of the ship, and it can be seen from Figures 5 and 6 that the displacement and velocity at the bottom of the string are greater than at the top. This is due to the elastic properties of the string and the resultant stretching. Figure 7 shows the dynamic axial stress in any element of the string. This stress is caused by the heave motion of the ship. Figure 8 shows the total stress which is the sum of the static stress and the dyanmic stress. The static stress is the in-water weight of the portion of the string below the element being considered. Figure 9 shows the fatigue life and indicates that the ADP fatigue life is approximately 1000 hours in this configuration. The computer fatigue life estimate of 1000 hours for a 30,000-foot string appears to be too low based on Project experience. The fatigue prediction algorithm and the fitting of the fatigue curve are being reviewed for range of applicability. Figure 10 shows the resonance condition for the string. The fundamental resonant period is about 3.3 seconds which is in the low energy portion of the ocean wave spectrum, and therefore the string is not likely to be excited to resonance. It should be noted that the computer model, in its present form, calculates only the static and dynamic stress. The bending stress, as the pipe passes through the guide shoe, must also be taken into account and added to the static and dynamic stress. For the 30,000-foot configuration, the total stress leaves a calculated margin which is adequate for an overpull of only 50,000 pounds in the event of a stuck pipe.

OPERATIONAL TESTS

An operational test of the ADP was necessary in order to fully evaluate its suitability in long drill strings. Various aspects of operational use were to be considered including fatigue life, coatings and their effect on corrosion, handling, and inspection methods. An order was initiated with Reynolds Metals Company, the only supplier of ADP, and in November 1978, the first shipment of 42 joints of 5-inch ADP was put aboard the ship. Shortly thereafter 21 additional joints were delivered to make a total of 63 joints (approximately 1900 feet). The 63 joints were all shot peened on the interior but had various interior coatings--21 joints were coated with Drilcote, a Reynolds proprietary epoxy coating; 21 joints were coated with Dimetcote, an inorganic zinc coating produced by Ameron; and 21 joints were left without any coating. The 21 joints that were coated internally with Dimetcote were also coated externally with the same coating. The ADP was to be put in service for about 12 to 18 months; then an evaluation was to be conducted and the pipe was to be taken off the ship. However, after the initial trial period, more exposure was needed for evaluation and the pipe was left on board. As of the date of this report, the original order of ADP has been on the ship for about five years.

The ADP has a number of advantages over steel pipe, the most significant of which is weight. The 5-inch ADP weighs only half as much as the standard 5-inch steel pipe that has been used on the ship. Another distinct advantage is the lower Young's modulus of aluminum--one third that of steel. This allows the ADP to bend in a tighter arc with less stress being developed which is an important consideration when the pipe is being lowered through the guide shoe in the moon pool. Also, the ADP has better low temperature toughness and higher fatigue endurance than steel pipe. Because of the lighter weight and lower Young's modulus of the ADP, it can be rotated at higher speed before encountering critical vibrations. The price per foot of ADP is comparable to that of the steel pipe.

The operational tests have shown that there are no operational problems in handling the ADP. The pipe is made up into stands--three joints coupled together--for a total length of 90 feet. These stands are then laid horizontally in a mechanized pipe racker. Each stand must be drawn from the horizontal position to a vertical position. The bending experienced by the ADP during this process is well within the capability of the pipe. Because of extreme drill string lengths and the possibility of pipe damage caused by slips, elevators rather than slips are used with all pipe. The aluminum pipe is handled with a special 5-1/2 inch, 18 degree taper elevator modified to handle the 5.688 inch diameter below the box tool joint.

INSPECTIONS

As the performance of the ADP was of mutual interest to DSDP and Reynolds, both parties have participated in frequent inspections of the pipe. In January, 1979, Reynolds inspected the ADP. Approximately 39 joints were visually examined. Four joints were inspected by internal borescope and by ultrasonic wall thickness gaging. The borescope is an optical device which can be passed through the bore of the pipe and which enables the user to see the condition of the interior of the pipe. The borescope inspection was performed on two joints coated on the inside with Drilcote and two joints that were bare on the inside. No corrosion was observed in the coated joints. Profuse, broad but shallow pitting was observed in the two bare joints. This type of corrosion is normal for the 2014 alloy in a marine environment. The external surface of all the ADP displayed profuse shallow pitting which again is normal for 2014. Normal, light galvanic pitting was observed on the pipe surfaces next to the steel tool joints. Measured pits were less than .005 inch deep. The ultrasonic thickness gaging showed the thickness exceeding the .525 inch specified nominal thickness for new pipe. After inspection, all of the ADP was found suitable for further service.

In 1980, another inspection was performed at the Norfolk, Virginia port call by personnel from Reynolds, Richmond. Visual inspection was made on the complete test string of ADP. Corrosion and fatigue were evaluated and the wall thickness was measured using a Sonotest Model UTG-5 ultrasonic thickness gage. The ADP test string was considered to be in satisfactory condition and was put back in service.

Unlike conventional drill pipe which can become thin anywhere along the length, experience has shown that ADP will always exhibit its minimum wall at the mid-point of the length. It has never been found otherwise. This characteristic permits inspection of ADP at the rig site to determine the minimum remaining wall at the mid-length of each joint. Pi tapes (furnished by Reynolds) are used initially around the circumference at the mid-length to determine the diameter. This value, as related to the original diameter, then indicates remaining wall assuming that all wear has occurred on the outside and that the bore is free of major corrosion which can be checked with a borescope. This system is satisfactory if no more than .200 inch is missing from the O.D. at the mid-length. If O.D. loss at the mid-length exceeds .200 inch, then an ultrasonic wall thickness gage must be used.

In the fall of 1981, Reynolds requested Tuboscope to make an evaluation of the Drilcote coating that was applied to some of the ADP. Tuboscope found the coating to be generally intact and in good condition. Small patches of coating had been removed and these areas were associated with grooves or cuts in the surface of the coating. The cuts were usually long and were deduced to be caused by running the wireline for core retrieval.

In November 1981, a complete inspection of the ADP was performed jointly by DSDP and Reynolds at the Panama port call. The inspection consisted of:

1) Visual inspection of exterior and interior.

2) Measured depth of typical "deepest" pits on exterior.

3) Measured length - if not stretched, length should be 30 feet, plus 2 inches, minus 0.

4) Measured wall thickness.

The type of coating on the interior of the pipe could be determined visually. The Drilcote had a greenish cast and was very smooth and reflective. The uncoated interior had a dark, but smooth and reflective appearance. The Dimetcote (zinc) coating had a rough, pitted, non-reflective appearance. The wall thickness was measured with a Nova 201 Ultrasonic Thickness gage made by NDT Instruments of Huntington Beach, CA. The gage was calibrated on a single ring cross section of pipe before and after each set of measurements. The results of this inspection were:

1) Most of the pipe showed general pitting (about .025 inch deep maximum) on the exterior although some joints were relatively free from pitting.

2) Five joints exhibited severe exfoliation, i.e., large patches of 3 to 4 square inches each, usually on the tapered region at the pin end. It should be noted that severe does not mean deep since, even in the exfoliated areas, the wall thickness was greater than that specified for new pipe.

3) The smoothest, least corroded interiors were those coated with Drilcote. The bare interior was the next best. The pipe coated on the interior with Dimetcote

(zinc) had general pitting throughout the interior.

4) No stretching was observed on any of the joints.

5) All of the wall thickness measurements were greater than that specified for new pipe (.525 inch). This was true even for the exfoliated regions.

Twenty four joints exhibited some exfoliation ranging from minor to severe. The five joints with severe exfoliation were pulled out and shipped to Houston to be shot peened on the exterior. The remaining joints of ADP were declared suitable for continued service. This was to have been a test to determine if shot peening would decrease the exfoliation. However, all of the exfoliation was not removed by the shot peening so it was not a valid test. The shot peening would have to be done on new pipe to provide a proper test. A detailed investigation of the exfoliation problem was conducted and this is discussed in the next section.

The most recent inspection of the ADP was again a joint effort with Reynolds and DSDP and was held at the port call in Norfolk, Virginia in June 1983. The inspection consisted of:

1) Visual examination of the exterior.

2) Check of I.D. numbers to locate ten new joints that were shot peened on the exterior and delivered to the ship in September 1982.

3) Ultrasonic wall thickness measurements.

4) Borescope examination of interior of pipe.

5) Scrapings from interior of pipe to test for zinc coating.

The results of the inspection were:

1) The exterior condition of the pipe was about the same as in the 1981 inspection.

2) All of the wall thickness measurements were greater than that specified for new pipe (.525 inch).

3) Most of the pipe showed smooth reflective interiors. The non-reflective pitted interiors were examined more closely with the borescope and the pits were found to be very shallow.

4) There was no exfoliation on the ten new joints. There were some corrosion blisters and light galvanic corrosion on the aluminum adjacent to the steel tool joints.

Three joints of pipe were pulled out at this inspection and shipped to the Reynolds, Richmond facility for testing. Two of these joints had severe (as previously defined) exfoliation and will be put in the rotating beam fatigue test machine. The third joint was observed spraying water from the tool joint when the stands of pipe were being broken into singles after the previous deployment. This tool joint will be cut open to examine the interior for some feature that may have caused the entrapment of water under pressure. The scrapings from the inside of the pipe have been analyzed but they do not show anything conclusive as to the effect of the zinc coating on the interior condition of the pipe.

LABORATORY/METALLURGICAL TESTS

In the summer of 1981, exfoliation of the exterior of the ADP was detected, and in some cases, it appeared to be rather severe. One joint with extensive exfoliation was taken off the ship and sent to the Reynolds, Richmond facility for fatigue testing. Figure 11 shows a typical case of exfoliation.

Visual examination of the complete drill pipe joint showed that the exfoliation was much more severe at the ends of the pipe. On peeling back some of the surface, there was clear indication of corrosion product. Since the amount and extent of corrosion attack was observed to vary from one end of the pipe to the other, it seemed advisable to examine the depth and type of corrosion attack, as well as the thickness of the recrystallized layer along the length of the pipe. One-inch thick rings were cut at 10", 41", 43", 46", 60", and 80", from the pin end of the pipe. These samples were etched in caustic to reveal the grain structure. The recrystallized layer on the exterior surface varied from very light (0.020") in the heavy wall to substantial (0.120" to 0.130") at the 80" section. The interior surface also showed a recrystallized layer, which was much thinner and uniform along the 80" length examined. Figure 13 shows that a recrystallized layer exists at both the inside and outside surface of the pipe.

As with the pin end, one inch rings were cut and machined at 10", 41", 43", 46", 60", and 80", from the box end of the pipe. These samples were etched in caustic to reveal the grain structure. There was no discernible recrystallized structure at the 10", 41", 43", and 46" locations, while the 60" section had a recrystallized skin of >0.125".

Intergranular corrosion was seen in many of the box-end and pin-end cross-sections, with the depth of attack generally on the order of 0.020" or less. One extreme example of intergranular attack is shown in Figure 12, where the attack has penetrated to a depth of 0.044". A larger view of this same section of the pipe is shown in Figure 13, where the total depth of the recrystallized layer is 0.130".

The reason for the noted exfoliation and intergranular corrosion is not clear at this time, but improved heat treating practice is expected to prevent this from occurring in the future.

FATIGUE TESTS

In the Spring of 1981, after 18 months of service, six joints of the ADP were taken off the ship and sent to the Reynolds, Richmond facility for fatigue testing. The six joints were chosen so that, on the interior surface, two were bare, two were Drilcote, and two were Dimetcote. Tests were conducted using a rotating cantilever beam fatigue machine. The speed of rotation was controlled at 300 rpm. The bending load was applied at the end of the test specimen, resulting in a moment arm to the end of the tool joint of approximately 109 inches. Two bending loads were employed:

200,000 in-lbs (max bending stress in body of pipe = 15,000 psi)

170,000 in-lbs (max bending stress in body of pipe = 13,000 psi)

The higher bending moment was included to provide a comparison with rotating beam tests previously performed on new, unused 5-inch ADP. The lower bending moment was chosen to be equivalent to a 85 percent reduction factor to accommodate to the stress risers on the surface of the used pipe that result from abrasion, wear, and corrosion. The results of the fatigue tests are shown in Table I. As might be expected in fatigue testing, there was some scatter in the data especially for the case of new pipe where the cycles-to-failure ranged from 2.44×10^5 to 2.60×10^6. The most important observations on the used ADP were:

1) The failures occurred outside of the tool joints.

2) All fatigue failures occurred at visible stress risers.

Figure 14 shows a fatigue curve for sharply notched, round specimens of the ADP alloy 2014-T6 (Reference 4). The results of the fatigue tests from Table I have been over- plotted on the curve of Figure 14 and show very good agreement with the curve. It appears that sharply notched specimens are representative of the condition of used pipe that has been gouged and abraded during service, and therefore, contains stress risers comparable to the notches on the specimens. There is no endurance limit for non-ferrous metals--the fatigue curve continues to decline at lower stress levels regardless of whether the surrounding medium is air or water.

The test results of Table I are based upon the six joints of ADP that were chosen such that two joints were bare, two were coated with Dimetcote, and two were coated with Drilcote. Another joint of ADP, which was severely exfoliated, was also sent to Reynolds for fatigue testing to see if the exfoliation had any effect on the fatigue life. The joint was tested at a stress level of 13,000 psi and it failed at 308,000 cycles. This fits in with the results of Table I, but the more important observation is that the failure did not occur in the exfoliated area. It seems, from this one test, at least, that the exfoliation does not produce stress risers that would lead to the start of fatigue cracks.

In addition to the fatigue testing, full sections of the used 5-inch ADP were tensile tested using equipment at the Phoenix extrusion plant. All of the pipe exhibited tensile strengths in excess of the 488,000-pound value specified for new pipe.

Table II is a listing of the ADP usage by DSDP. At the stress levels (<8,000 psi) encountered during operations up to the present, it is estimated that about five percent of the fatigue life of the ADP has been consumed.

Table III is a chronological listing of the major events concerning the aluminum drill pipe during the 5-year period of usage by the DSDP.

CONCLUSIONS AND RECOMMENDATIONS

From the results of the inspections and testing as described previously, the following conclusions may be drawn concerning the ADP that has been in service for five years in the Deep Sea Drilling Project:

1) The bare, shot peened interior seems to be holding up as well as the interior coated with Drilcote. The Dimetcote appears to be the least suitable.

2) The exfoliation, although cosmetically unattractive, does not appear to affect the fatigue life.

3) The recrystallization of the exterior surface of the pipe can not be eliminated, but may be mitigated with further metallurgical modifications.

4) Every joint of the ADP, on all inspections, has shown a wall thickness at mid-length greater than that specified for new pipe.

5) A very small percentage (5%) of the fatigue life of the ADP has been consumed.

6) With the modified 5-1/2 inch, 18 degree elevators, handling of the ADP has presented no problems for the drilling crew.

The following recommendations are put forth for any future purchases and use of ADP:

1) Shot peen interior of pipe. Care must be exercised during this process; too much shot peening can cause the surface to become brittle and increase the chances of development of fatigue cracks.

2) For deep sea drilling applications, the pipe interior should be left bare.

3) Conduct scheduled inspections at approximate yearly intervals. Inspection should consist of:

   a) Visual: exfoliation, pitting, gouges, cracks on exterior.

   b) Borescope examination of interior for pitting or cracks.

   c) Wall thickness measurements with ultrasonic gage.

   d) Examination of tool joint threads as with steel pipe.

   e) Length measurement to determine stretching.

4) Maintain a usage record so as to have a running account of fatigue life consumed.

5) Set limits on wall thickness and fatigue life and remove from service those joints of pipe which exceed the limits.

In general, DSDP has received good service from the test section of 5-inch aluminum drill pipe, and based upon the computer work discussed previously, a mixed string of steel and ADP should be a viable option for reaching total depths of 30,000 feet while maintaining a 4-inch drift bore.

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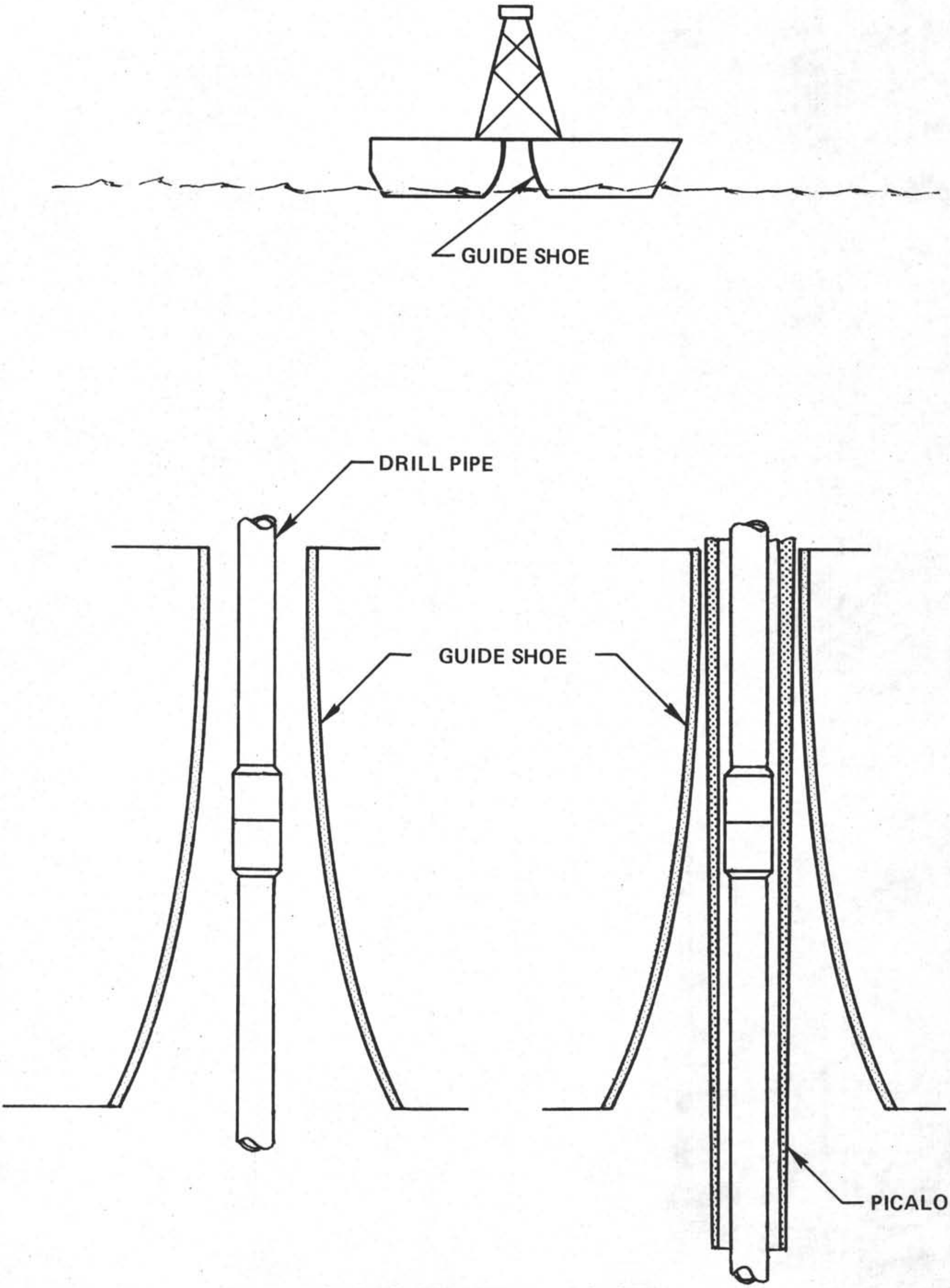
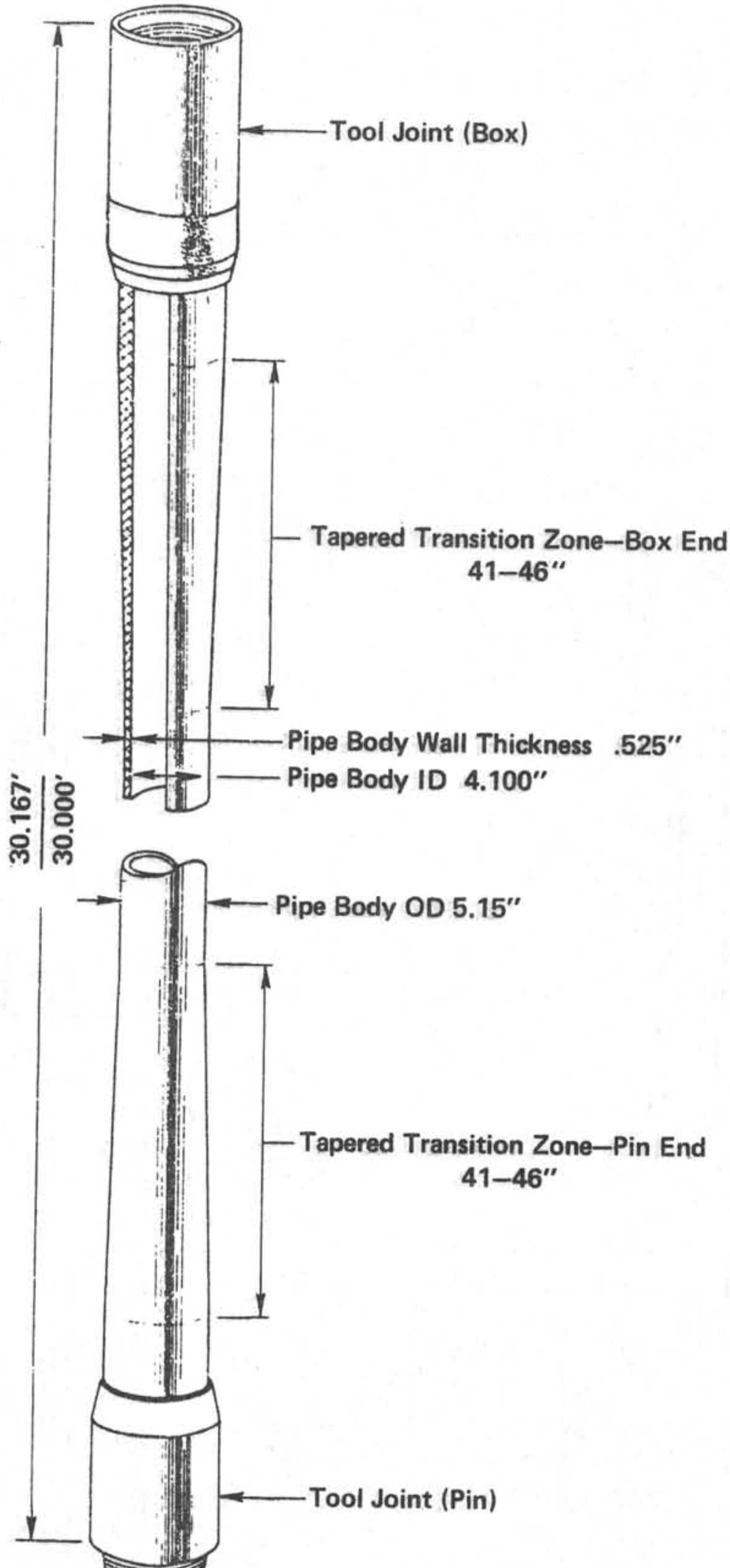
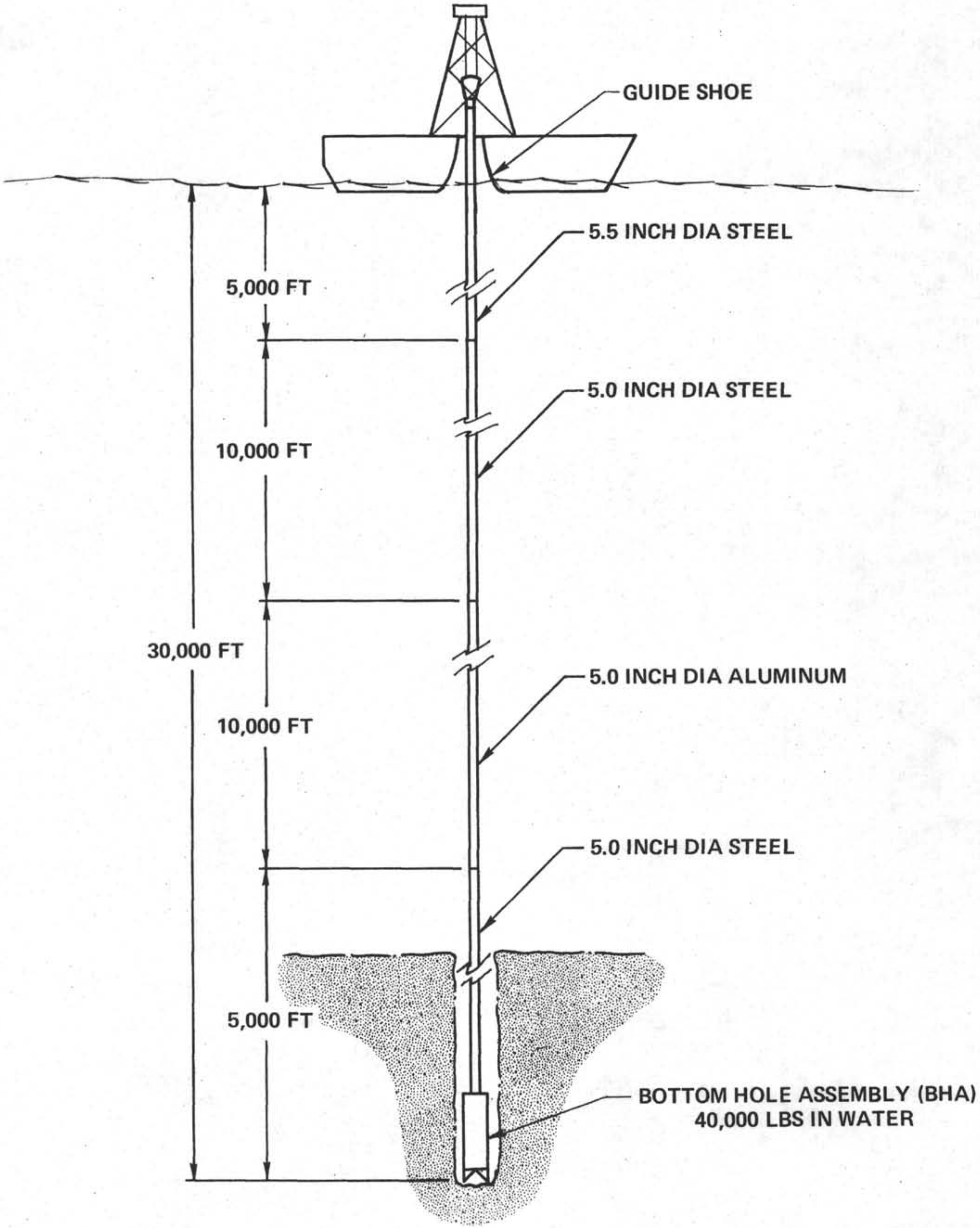


GUIDE SHOE AND PICALO SYSTEM
FIGURE 1



ALUMINUM DRILL PIPE
FIGURE 2



CONFIGURATION OF 30,000-FOOT DRILL STRING

FIGURE 3

RUN ID# 1368

GROUP	LENGTH (FT)	#ELEMENTS	RUBBER SPACING (FT)	PIPE TYPE
1	5000.0	10	7.0	10= 5.50 ST. API
2	5000.0	10	7.0	1=5 ST., PREM API
3	5000.0	10	7.0	1=5 ST., PREM API
4	5000.0	10	7.0	2=ALUMINUM, NEW
5	5000.0	10	7.0	2=ALUMINUM, NEW
6	5000.0	10	7.0	4=5 ST. 90%-WL API

H.C. TYPE =NONE
BOT ASSY DIAM = 0.867
INPUT TYPE =RANDOM SEA
HYDRO DAMPG =VELOCITY DEPENDENT
H.C. SPRING RATE = 0.005
HYDRO DAMP IN HOLE = 0
ADDED MASS FACTOR FOR JOINTS = 1.44 = 0

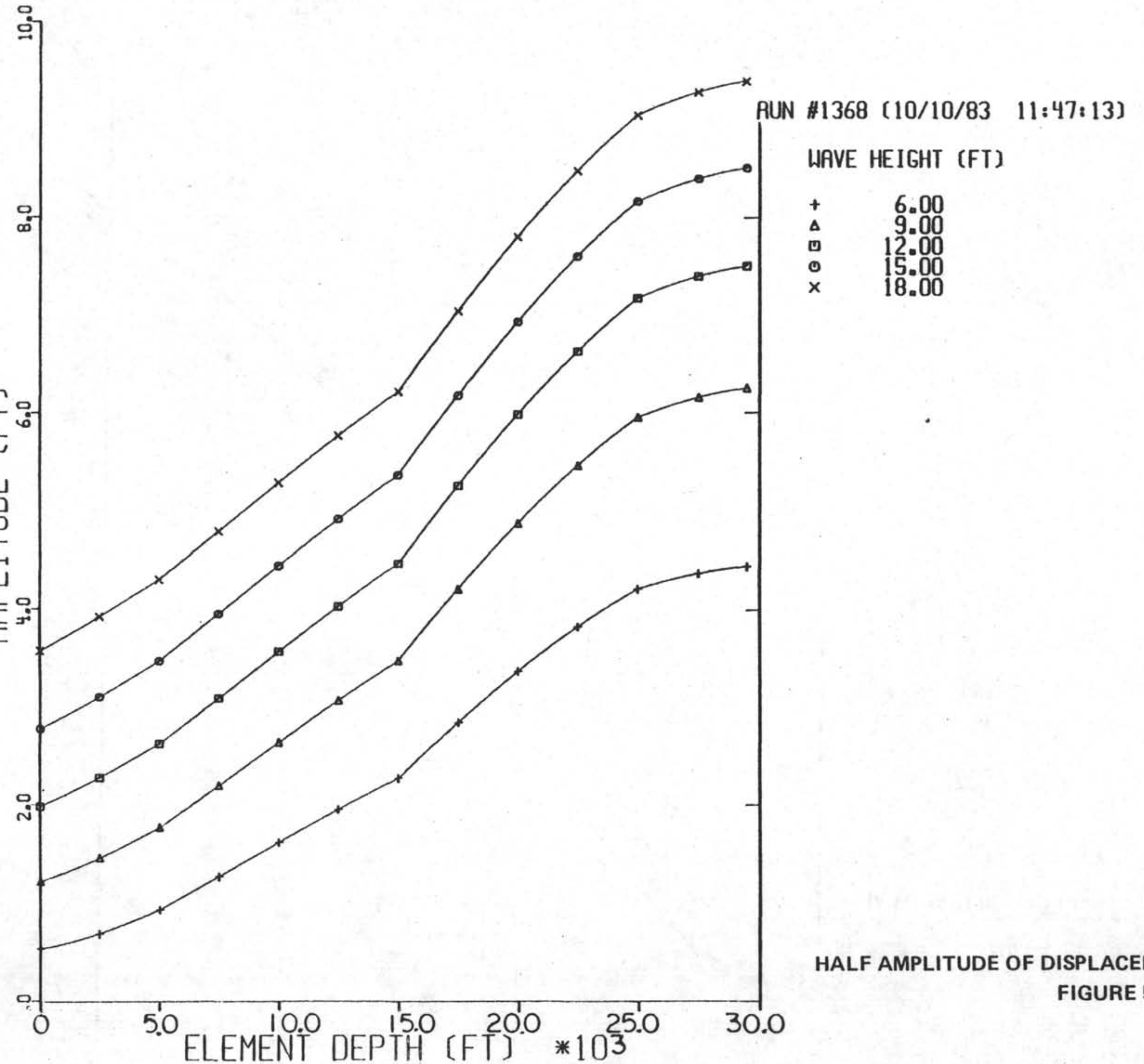
H.C. DAMPING = 2100
DS END ATTACH =FREELY SUSPENDED
SIG WAVE HT = 6
SHIP X-FCN =45 OFF BOW
#ELT GR-S = 6
= 0

BOT ASSY WT = 40000
INPUT LOCTN =TO SHIP
INT STEPS = 40
PIERSON/MOSKOWITZ.
BOT-GRND SPRING RATE = 0
ADDED BOTTOM MASS = 0
= 0

X-AXIS: ELEMENT DEPTH (FT) FROM 0.00 TO 0.00 BY 0.00
CURVE INDEX: WAVE HEIGHT (FT) VALUES 6.00 9.00 12.00 15.00 18.00
PLOT # 1: AMPLITUDE (FT) [1]
PLOT # 2: VELOCITY (FT/SEC) [2]
PLOT # 3: DYNAMIC STRESS (LB/SQIN) [3]
PLOT # 4: FATIGUE LIFE (LOG HRS) [4]
PLOT # 5: TOTAL STRESS (LB/SQIN) [5]

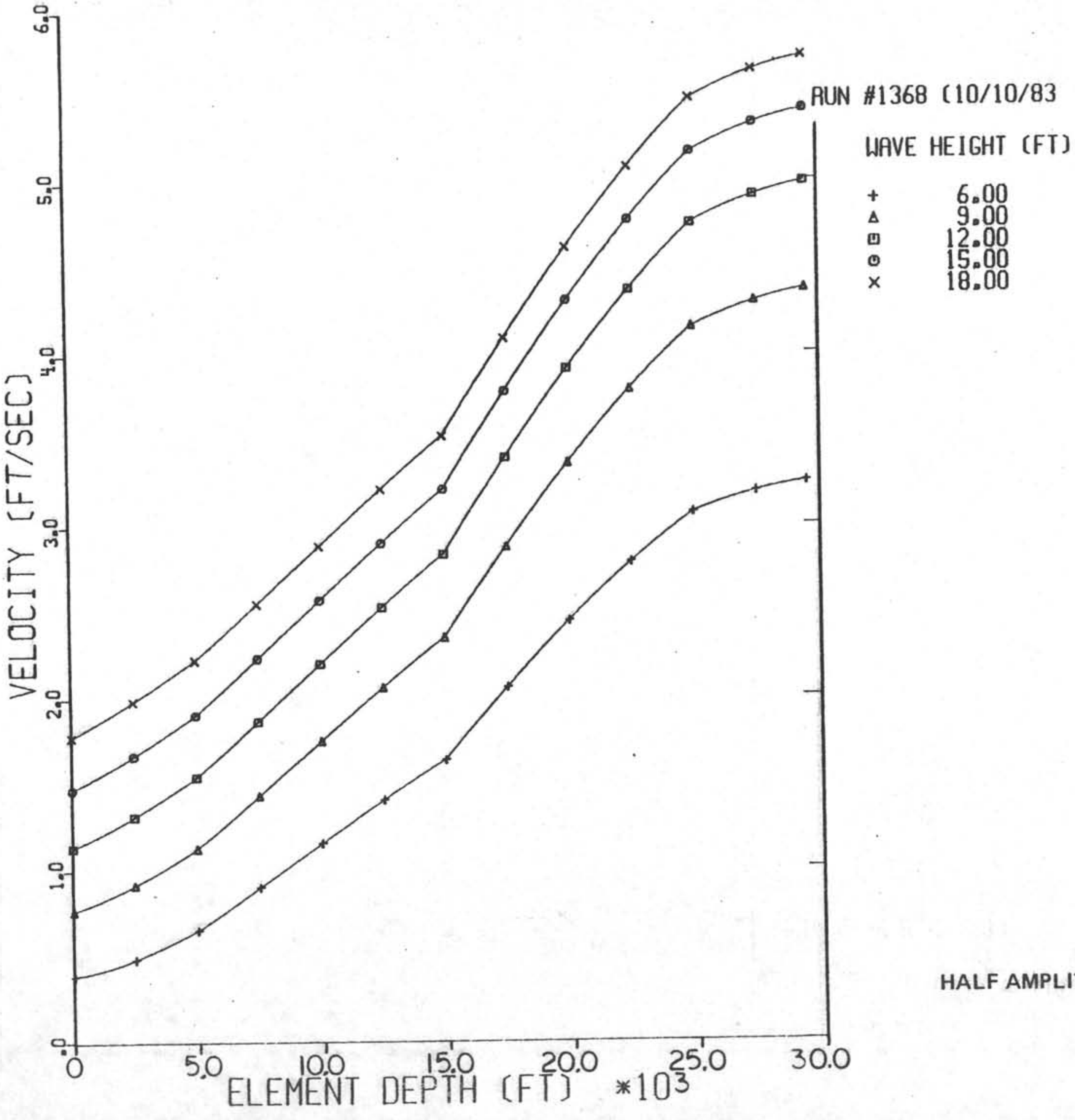
COMPUTER SETUP FOR 30,000 FT DRILL STRING

FIGURE 4



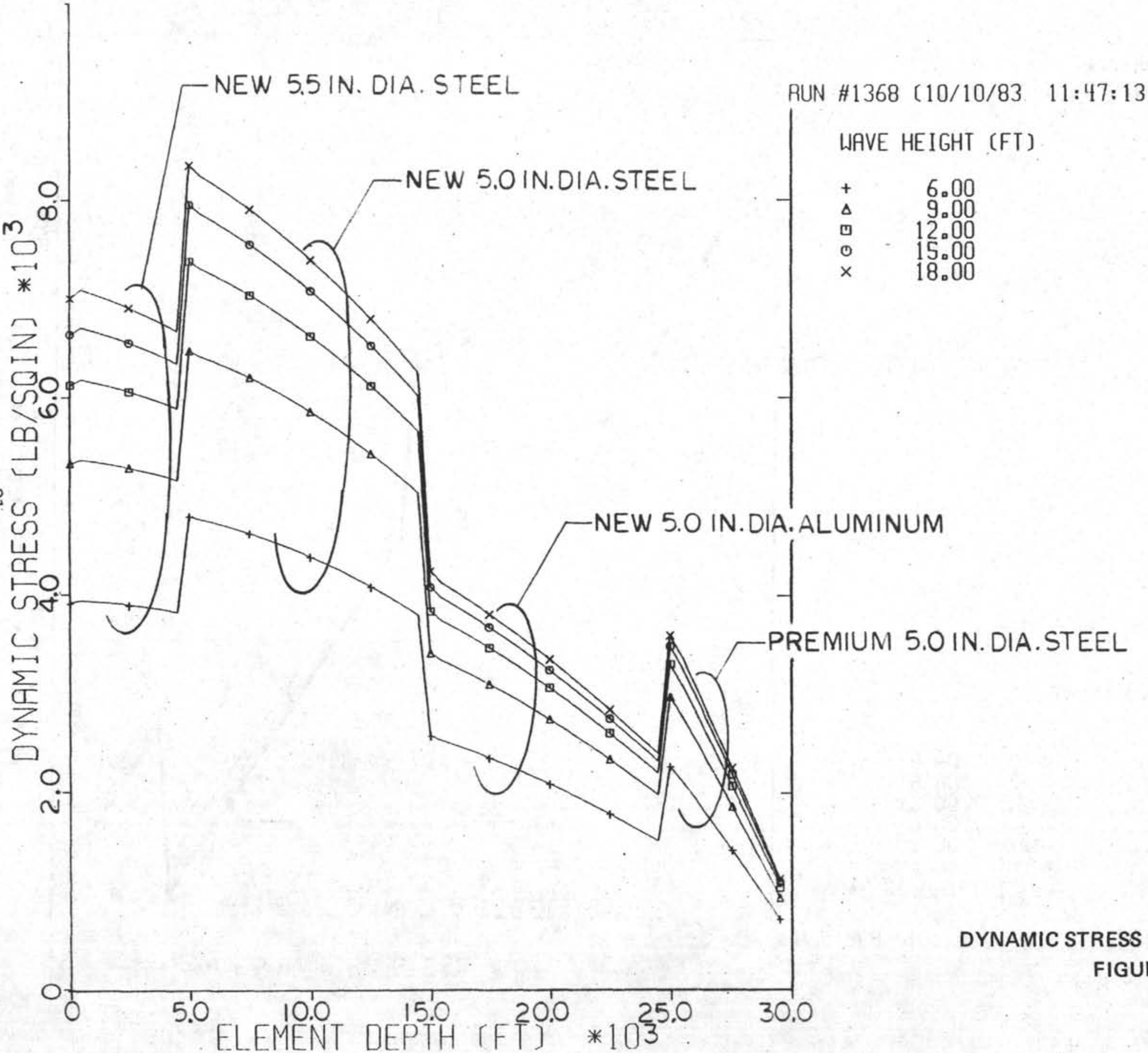
HALF AMPLITUDE OF DISPLACEMENT OF DRILL STRING

FIGURE 5



HALF AMPLITUDE OF VELOCITY OF DRILL STRING

FIGURE 6



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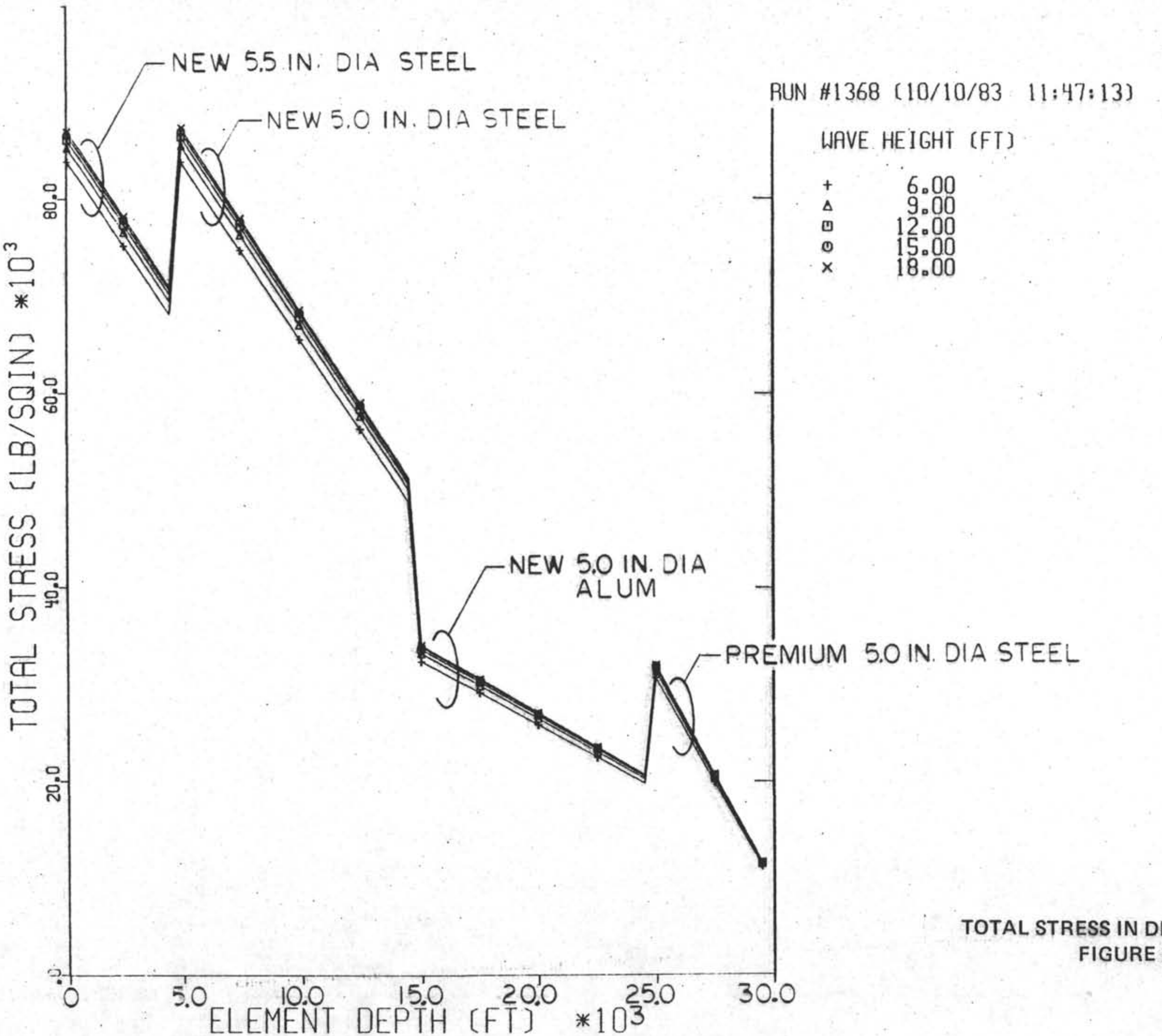
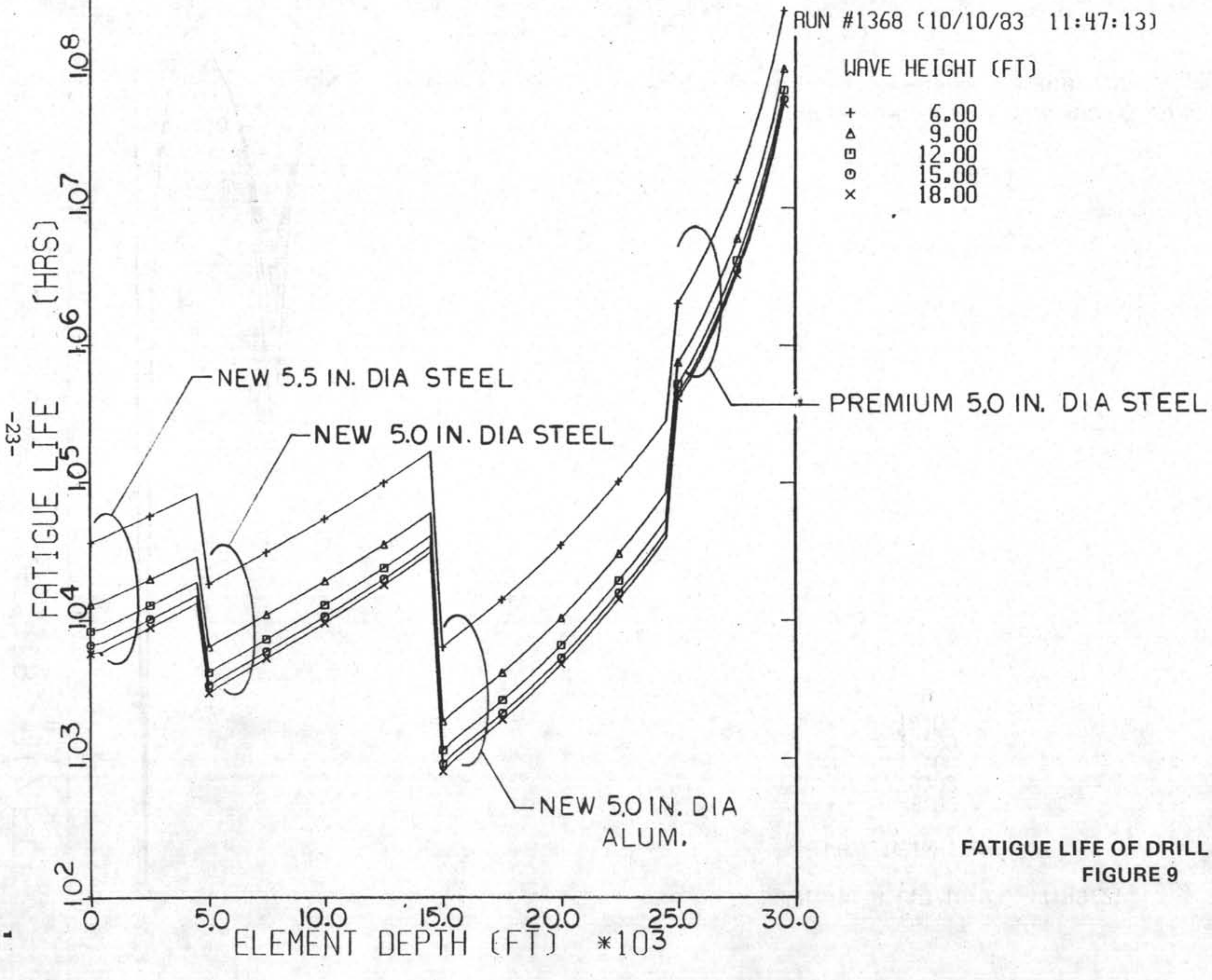
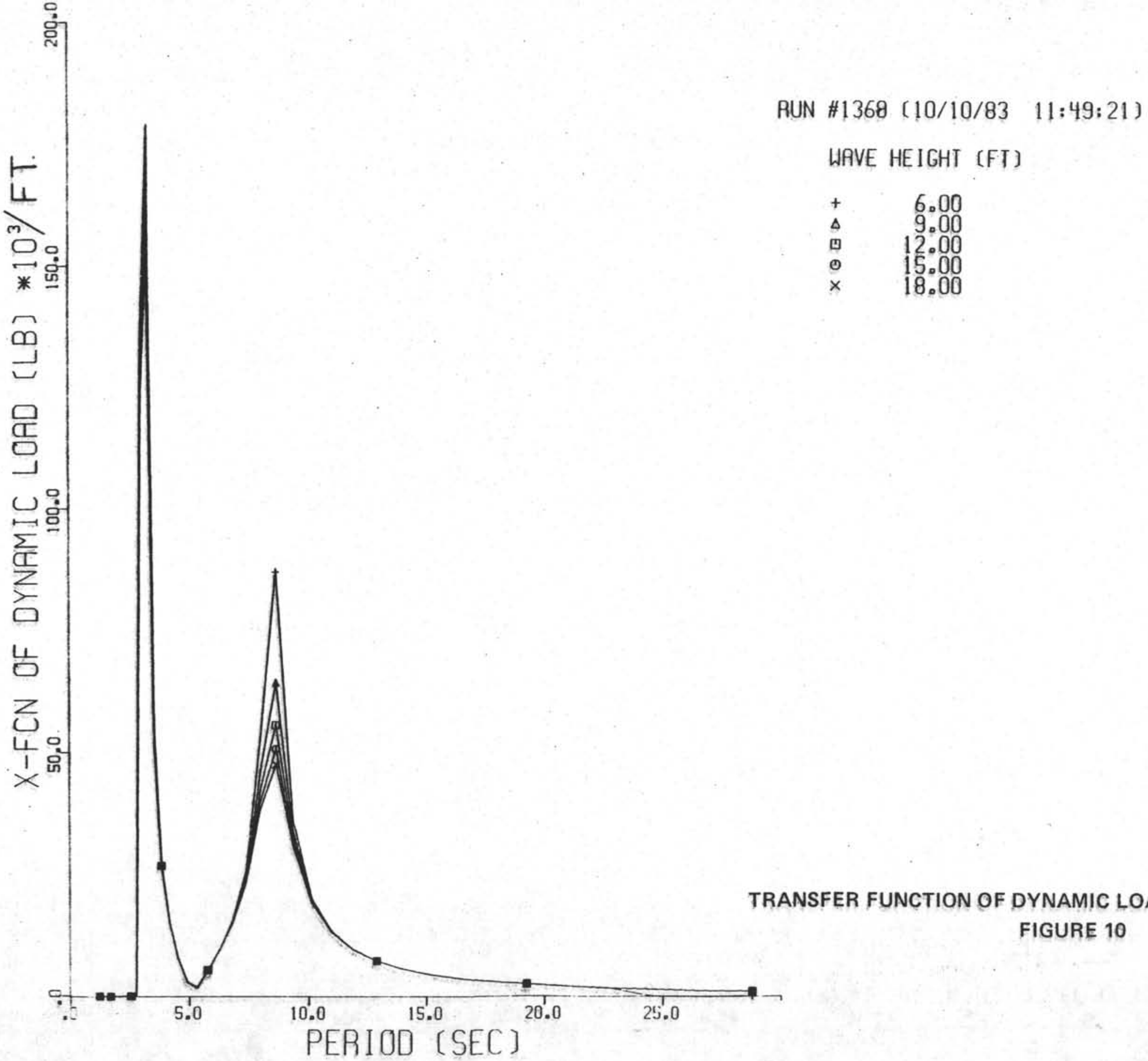
DYNAMIC STRESS IN DRILL STRING
FIGURE 7



TOTAL STRESS IN DRILL STRING
FIGURE 8



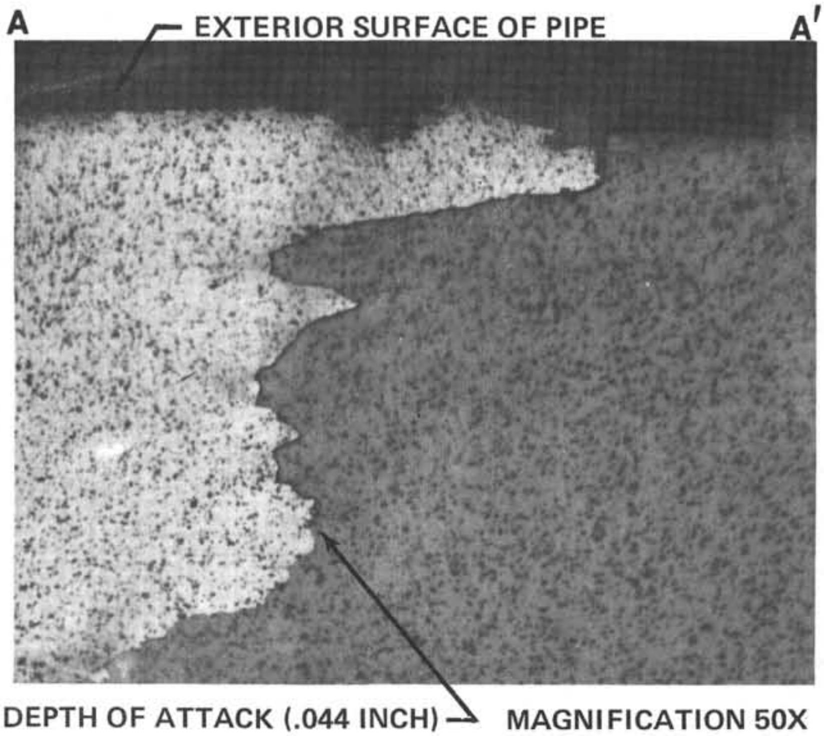
FATIGUE LIFE OF DRILL STRING
FIGURE 9

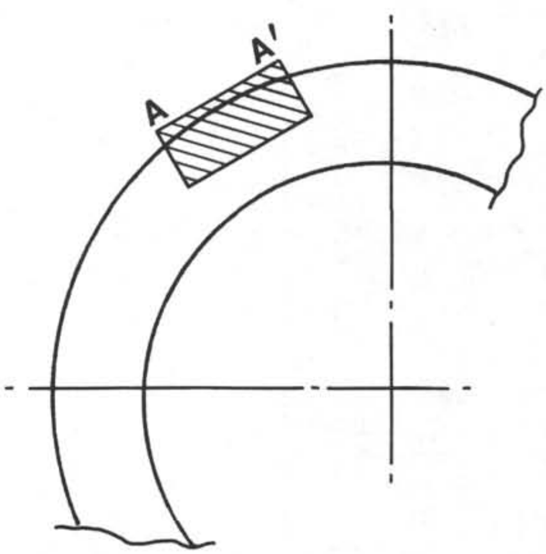


TRANSFER FUNCTION OF DYNAMIC LOADING OF DRILL STRING

FIGURE 10

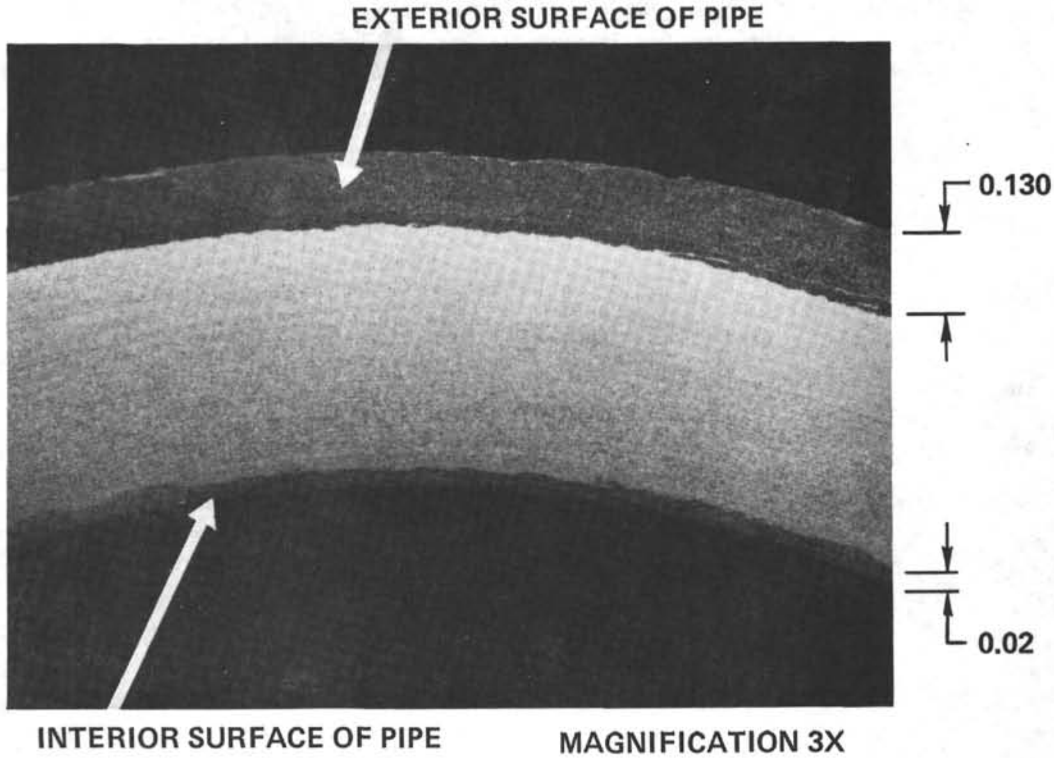
EXFOLATION OF ALUMINUM DRILL PIPE
FIGURE 11



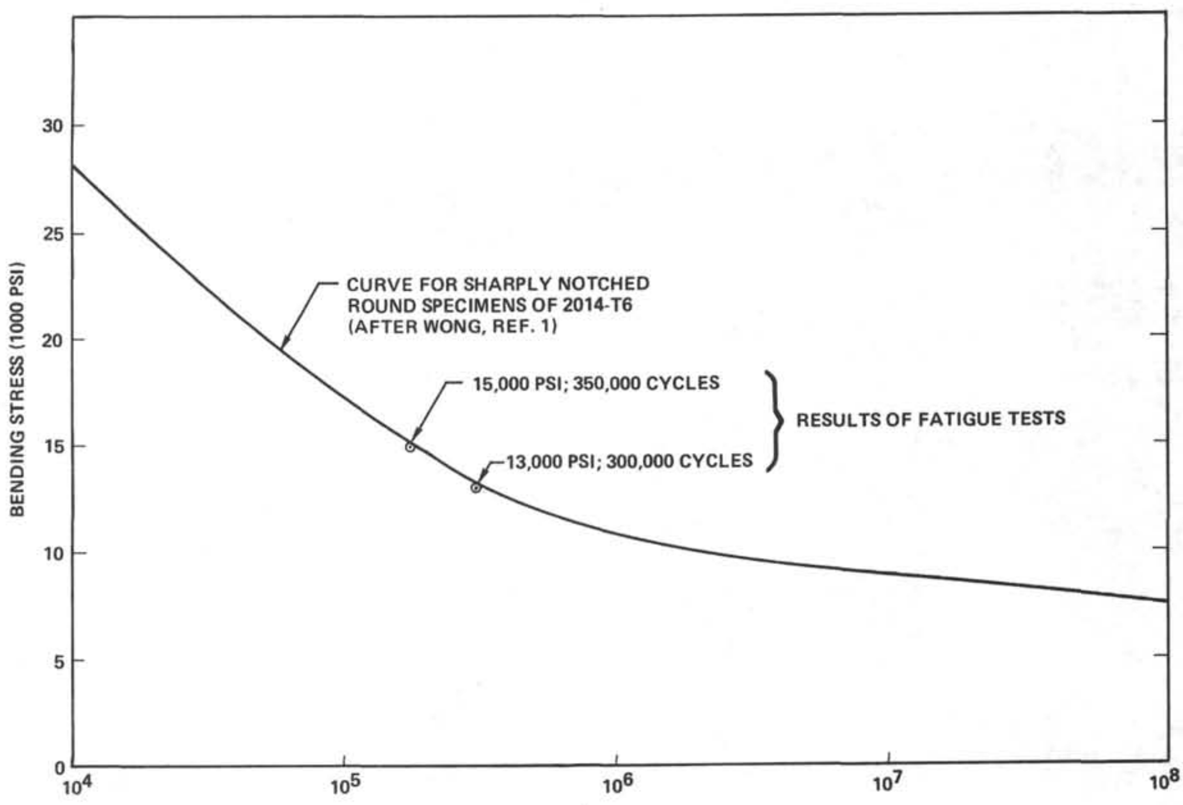


LOCATION OF SECTION

INTERGRANULAR ATTACK
FIGURE 12



RECRYSTALLIZED LAYER
FIGURE 13



FATIGUE CURVE FOR 2014-T6 ALUMINUM ALLOY
FIGURE 14

TABLE I

RESULTS OF FATIGUE TESTS

Description of 5" ADP Samples	Bending Moment In/Lbs	Cycles to Failure	Remarks
New	200,000	2.44×10^5 to 2.60×10^6	Failures occurred in transition zone near body of pipe
Dimetcote	200,000	117,400	Broke 25-1/2" from tool joint at stress riser caused by 3 pitted spots on outer surface
Drilcote	200,000	175,600	Broke 26" from tool joint at stress riser caused by one pit on outer surface
No coating	170,000	360,900	Broke 3-1/2" from tool joint in transition zone. Stress risers at 2 pitted areas on outer surface
Drilcote	170,000	500,000 *	No failure; stopped test
Dimetcote	170,000	454,700	Broke 32" from tool joint at stress riser from one pitted spot on outer surface

TABLE II

ALUMINUM DRILL PIPE USAGE ON DEEP SEA DRILLING PROJECT

LEG	DATE	WATER DEPTH (M)	LENGTH OF ADP (M)	HOURS ROTATING	HOURS (1) IN WATER
63	Nov 78	3850	375	117	277
66	Apr 79	2850	375	161	224
67	May 79	2350	375	21	147
73	May 80	4800	567	72	711
79	May 81	4000	439	150	354
80	Jun 81	1250	448	13	82
84	Feb 82	3850	402	89	205
85	Apr 82	3900	439	40	72
93	May 83	4650	540	110	270
95	Sep 83	4650	503	150	339
			TOTALS	923	2681

(1) Drill string is not rotated during periods of wireline core retrieval.

TABLE III

CHRONOLOGICAL HISTORY OF ADP USED BY DSDP

Date	Event
Nov 1978	Received 42 joints (21 coated internally with Drilcote, 21 uncoated)
Jan 1979	Inspection of ADP on board ship in California by RMC
Nov 1979	Received 21 joints (coated internally and externally with Dimetcote)
Sep 1980	Pipe unloaded at Norfolk, VA and shipped to Bellwood, VA for inspection by RMC
Feb 1981	Six joints shipped to Richmond, VA for fatigue testing by RMC
Apr 1981	Exfoliated joint shipped from Europe to Richmond, VA for examination by RMC
Sep 1981	Evaluation of Drilcote coating by AMF/Tuboscope
Nov 1981	Inspection of ADP in Panama by DSDP and RMC
Mar 1982	Five joints shot peened and put back on ship
Jun 1982	Ten new joints received
Jun 1983	Inspection of 59 joints at Norfolk, VA by DSDP and RMC. Three joints shipped to Richmond, VA for testing and examination

DESIGN AND USE OF HEAVY WALL DRILLING JOINTS FOR BENDING STRESS REDUCTION

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SUMMARY

This paper describes the design, test, and operational use of special heavy wall drilling joints. The drilling joints were developed for riserless coring and drilling, in water depths to 23,000 feet, carried out from the research vessel GLOMAR CHALLENGER. The drilling joints are designed for improved fatigue resistance at the upper end of the drill string which is exposed to reversed bending fatigue as the drill pipe rotates through a 350-foot radius (16 degrees per 100 feet curvature) bending restraint. Full scale proof test of a pipe section demonstrated that the shoulders of the connection did not separate under a combined load of 402,000 pounds axial and a bending moment sufficient to deflect the pipe to 16 degrees per 100 feet. Heavy wall drilling joints have been in routine use since early 1978.

INTRODUCTION

Deep Sea Drilling operations require significant penetration of basaltic basement rock. The maximum length of drill strings have reached 23,000 feet. A major source of fatigue is cyclic stresses generated as the drill pipe rotates in a bending restraint or guide shoe (see Figure 1). This guide shoe has been fitted between the rotary table and the keel of the vessel to restrain bending of the drill string. The shoe has a radius of curvature of 350 feet to 8 degrees maximum roll angle (half amplitude). Penetration rates in basalt are slow--on the order of 6 feet per hour in hard basalt. This slow rate of penetration through the guide shoe at high tensile loads can lead to early fatigue failure of the upper portion of the drill string with the present 5-inch, S-135, nominal 19.5 lbs/ft drill pipe.

The bending stresses generated are also a function of tension and the distance beteween supporting tool joints or drill pipe rubbers (Ref. 1). These bending stresses increase with increased tension and with increased length between supporting tool joints or rubbers. Figure 1 shows how the bending of the drill pipe is affected by this spacing. For Deep Sea Drilling operations the drill pipe is fitted with rubbers at an average spacing of 7 feet to reduce bending stresses. The effect of tension and rubber spacing is shown in Figure 2. At a top tension of 410,000 lbs. for a 20,000-foot drill string, Figure 2 shows the bending stress can vary from 22,000 psi for a 5-foot rubber spacing to 45,000 psi at a 15-foot rubber spacing.

Drill pipe rubbers as stress reducers have severe drawbacks for the control of bending stresses. The rubbers do not stay in position reliably. Slow drilling rates and heave compensation contribute to excessive wear and rubber displacement. With the heave compensator in use, the relative vertical motion betweeen the compensated drill string and the vessel is greatly increased leading to accelerated rubber wear and displacement when passing through the guide shoe. An alternate solution is the use of special heavy wall pipe with integral machined hubs replacing the "moveable" rubbers. The increased wall thickness of the heavy wall pipe would also provide for longer fatigue life. The design, test, and operational use of heavy wall drilling joints is described below.

TUBE DESIGN

The desired design was a heavy wall pipe with a hub spacing (stabilizers) to minimize bending stresses due to deflection of the tool joints while the pipe is in the guide shoe. The heavy wall pipe was to be used only at the upper section of the drill string exposed to the bending stresses in the guide shoe. The optimum design was to yield the maximum cycles to failure for the following design and handling limitations as excerpted from the request for proposal:

1) Maximum outside diameter of tool joints and hubs: 7 inches.

2) Minimum internal diameter: 4-1/8 inches (4 inch drift required for wireline coring operations).

3) Tensile load: static maximum 532,000 lbs. Average 400,000 lbs while drilling. Maximum static with 100,000 lbs overpull is 632,000 lbs.

4) Length: 30 feet minimum, 45 feet maximum.

5) Maximum weight: 2,500 lbs per joint for ease of handling.

6) Design for collar type elevators (slip type will not be used) compatible with normal rig tools.

7) Approximate maximum 400 feet of heavy wall pipe would be in drill string at any one time.

8) Design stiffness should allow for gradual radius of curvature with no stress riser at connection to 5-inch, 19.5 lbs/ft, S-135. Present pipe has 5-1/2 inch API F.H. tool joints.

Resulting design specifications will include, but not be limited to, following specifications:

1) Wall thickness.

2) Type and size of connections.

3) Recommended mechanical properties (ultimate strength, yield, etc.).

4) Hub dimensions and spacing.

5) If other than 5-1/2 inch API F.H. connection specified, please provide dimension and design of crossover subs required at upper and lower terminations of heavy wall (nominal 300-400 ft assembly).

6) Calculations and assumptions used to arrive at optimum design.

Dr. Arthur Lubinski responded to the request for proposal meeting the above requirements. The development of the design is found in Reference 1 which is included in Appendix I. His essential findings and recommendations were:

1. An O.D. of 5.725 inches (wall thickness of 0.8 inches) was selected.

2. Hub spacing to be 5 ft. A 6 or 8 foot spacing could be adopted with a shorter fatigue life.

3. If 500 ft of heavy wall pipe is used with equal bending exposure the fatigue life is 21 years (calendar operational years, not rotating hours). Rotation at a single point under continuous bending would fail the pipe in 17 days.

4. Minimum yield: 120,000 psi.

5. Tube:

   Length: 30 ft
   O.D.: 5.725 in
   I.D.: 4.125 in (4-inch drift)

6. Pipe to have 7-inch O.D. hubs.

7. Provide for elevator suspension (dual elevator handling system to be employed).

8. Notch toughness and fatigue characteristics in seawater to be the same as those of S-135 steel.

9.

   Tool Joints:

   O.D.: 8 in
   I.D.: 4.125 in (4-inch drift)

   Design to withstand:

(1) 25,300 ft-lb static bending moment applied simultaneously with a static pull of 632,000 lb.

(2) In fatigue in seawater, a fully reversed bending moment of 25,300 ft-lb applied simultaneously with a pull of 400,000 lb.

(NOTE: The final design used a 1/2-inch wall thickness, 5-1/2 inch O.D., and a 4-1/8-inch bore with a 4-inch drift. Hub and tool joint diameters were 7-3/4 inches. The changes were made following the selection of a 5-1/2 inch I.F. connection).

CONNECTION DESIGN

After design of the heavy wall pipe tube Dr. P.D. Weiner of Texas A & M University was selected to propose a design and a test program for a new tool joint (connection). An analysis was made to determine the optimum heavy wall design. The following connections were investigated (Ref. 2):

6-5/8 inch	Full Hole
5-1/2 inch	I.F.
5-1/2 inch	Regular
5-1/2 inch	Full
NC 56	
6-5/8 inch	Regular
6-5/8 inch	H-90
NC 61	
7 inch	H-90
5-1/2 inch	H-90

The optimum connection proved to be a 5-1/2-inch Internal Flush (I.F.) with an O.D. of 7-3/4 inches and an I.D. of 4-1/8 inches. Make-up torque was calculated at 53,247 ft-lb and the force to open the connection at 2,287,568 lbs. (NOTE: Industry recommendation, as per Reed Tool Company, is 44,000 lbs for make-up torque.)

Other considerations in the design are bending strength ratios and moment of inertia ratios. The bending strength ratio of a connection is the ratio of the box section modulus to the pin section modulus. The American Petroleum Institute (API) states that a bending strength ratio of 2.50:1 is generally accepted as an average balanced connection with an acceptable range of 3.20:1 to 1.90:1 depending on drilling conditions (Ref. 3). The API recommendations are for compression members where the stress arises from make-up torque and bending. The designed connection with a bending strength ratio of 2.06 is reasonable as the connection is used in tension at the top of the string and the box is not subject to wear in the hole.

The ratio of moment of inertia of the tube section to the connection (or hub) should also be optimized. Weiner (Ref. 4) found, from extensive testing of drill collar connections, that the maximum life of the connection is obtained when the ratio of the moment of inertia of the 5-1/2 inch tube body (relief) to the connection is from .28 to .32. The ratio is .19 for the design connection. The lower value is considered reasonable as the life of the connection will be extended, and the 10.4 square inches of tube body area is adequate for the tensile loads expected. The minimum yield load of the tube is 1,248,000 pounds. The fabrication drawing and specifications for the heavy wall drilling joints are included in Appendix II.

TEST PROGRAM

A static test program was undertaken by a contractor to verify the theoretical analysis (Ref. 5). Pertinent parts of Reference 5 are included as Appendix III. The general testing sequence, as excerpted from Appendix III, was as follows:

a) Torque connections to 60% of optimum torque.

b) Apply axial load of 400,000 lbs. Remove axial load.

c) Apply bending moment to equal an equivalent deflection of 16 degrees per 100 ft. Remove bending load.

d) Combined load. Apply axial load of 400,000 lb., followed by a bending moment such that the deflection at the connection is the same as that noted under part C. Remove bending load.

e) Break-out connections.

This sequence was repeated for torquing the connections to 80%, 100% and 120% of optimum torque with one exception-- Step E was not executed at the completion of 120% of optimum torque sequence.

The principal findings of the test program were:

a) At no point during the loading did the shoulders separate. The maximum loading was 402,025 lb axial combined with a bending moment to bend the test specimen to 16 degrees/100 ft arc.

b) The maximum stress concentration factor for the loading conditions of (a) above is 1.45. This geometric stress concentration is adjacent to a change of section where the tube meets the larger diameter connection.

c) The calculated pin stress at the stress relief groove for a make-up toque of 53,247 ft-lbs (120% of optimum) with 400,000 lbs tension and a curvature of 16 degrees/100 ft is 105,626 psi. The contractor's calculated stress from strain gages is on the order of 165,000 psi. The difference may be due to a localized geometrical stress concentration in the stress relief groove. No deformation was apparent and the connection did not open up under the load.

The high stress is due mainly to make up torque and the contractor has proposed to lower the torque to 27,000 ft-lbs to bring pin stresses to 68,000 psi. A make-up torque of 44,000 ft-lbs is in accord with field practice and should be used to minimize connection problems.

The pin stresses obtained by the contractor are consistent with similar work done in industry (Ref. 6). The measured stresses on the pin stress relief groove outer surface are higher than those measured on the I.D. of the pin sress relief groove (bore) by a factor of three. Recommended connection make up torque is based on a <u>nominal</u> 60,000 psi stress across the pin stress relief groove, i.e., average stress through the pin.

DSDP has followed the industry recommendation of 44,000 ft-lbs make-up torque and has received adequate service from their heavy wall joints. Reference 5 states that a make-up torque of 27,000 ft-lbs should be used so as not to exceed the stress limitations, but cautions that since only one connection was tested, it might be advisable to test more connections. Additional analysis and testing would need to be considered to verify stress distribution through the pin area and verify that the pin stress readings were valid. However, operational use has substantiated the safety of the higher make-up torque. An analysis which compares the theoretical strains and deflections of the heavy wall joints with the measured values discussed above has been included as Appendix IV. It can be seen from Table I of this comparison that the calculated and measured strain values generally differ by less than ten percent.

OPERATIONAL USE

The operational use of the heavy wall drilling joints is determined by the length of drill string deployed and the rate of penetration while drilling. The reason for using the heavy wall joints is to reduce stress at the top of the drill string and thereby increase the fatigue life. Figure 3 shows the conditions of string length and penetration rate under which the heavy wall joints should be used. The heavy wall joints are also used under certain special conditions such as during logging where the drill string is stationary in the guide shoe without rotating and during storms which produce high pitch or roll.

It is usually in borderline situations where the heavy wall joints are used--situations that are imposing high bending stresses repeatedly on the same section of drill pipe. If a storm is producing the situation, then the moving of the heavy wall joints from the casing rack to the rig floor must be done with utmost care. The joints may be stored in a casing rack or in shucks, but they must be available quickly when needed. The heavy wall joints, when used, should extend to at least below the keel of the ship, and be positioned so that a connection is not located at the keel except for passing the keel on the downward or upward movement of the drill string. It has been customary in the DSDP to use ten heavy wall joints at a time. If greater drill string length is required, the heavy wall joints are pulled up, taken out, and replaced with standard drill pipe. The replaced section is then lowered quickly through the guide shoe until the bottom of the hole is reached; then, if still necessary, the heavy wall joints are again inserted in order to drill ahead. For ease of core tally the heavy wall joints should be the same length as the standard drill pipe being used. A lubricant with high (40%-60%) metallic content, either zinc or lead, should be used when making up a connection with the heavy wall joints. The lubricant should be applied liberally to both the box and the pin threads and to the shoulders. Best-O-Life 270 is an adequate lubricant for this purpose.

Since the heavy wall joints are, by the nature of their function, highly stressed it is important that they be maintained properly and inspected periodically. A magnaflux inspection will reveal tiny fatigue cracks. A log should be kept in which is listed the hours of usage, both rotating and non-rotating. This log can be used as a guide to determine, in conjunction with the inspections, when a heavy wall joint should be removed from service. Not only must the drilling joints themselves be monitored for fatigue or cracks, but the guide shoe through which the drill string passes must also be checked periodically. Repeated use of the heavy wall joints can lead to excessive wear of the guide shoe especially at the top 2 or 3 feet. When this occurs, the worn section must be repaired. With this in mind, it might be advisable in future designs of guide shoes to make the top 5 feet thicker, or replaceable, or with rollers.

It may be noticed from the drawing of the heavy wall drilling joint in Appendix II that both of the end hubs are 18.8 inches long whereas the middle hubs are approximately 13 inches long. Experience of the DSDP has shown that cracks usually appear first in the threaded portion at either end. The longer (than necessary) length of the end hubs allows for a recut, i.e., cutting off the existing threaded portion and machining new threads. This process has been used successfully to extend the usable life of the drilling joints.

REFERENCES

1. Lubinski, Arthur, "Special Heavy Wall Drill Pipe for The Deep Sea Drilling Project", August 1975.

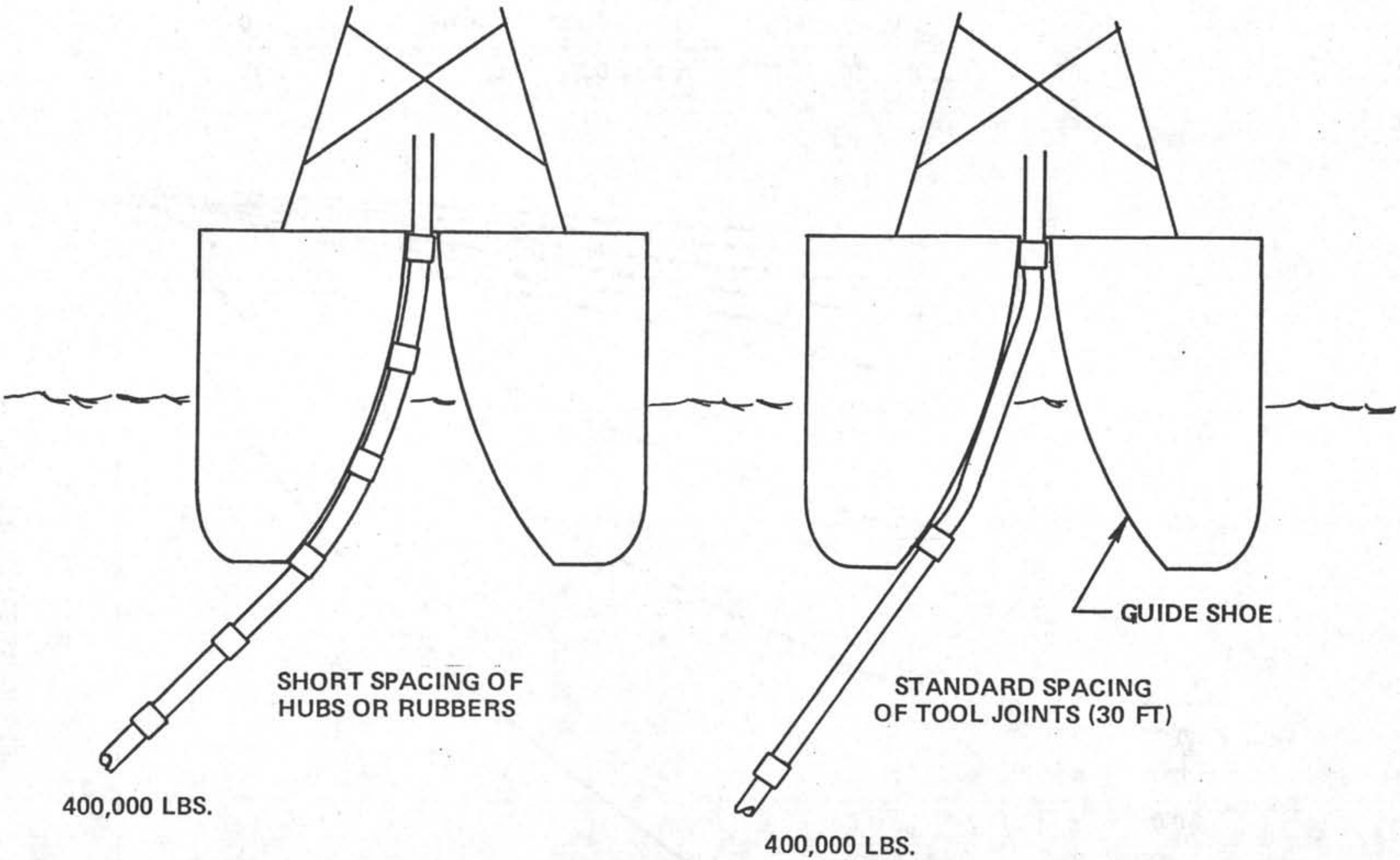
2. Memorandum to the Deep Sea Drilling Project, Mr. Adams from Weatherford/Lamb, Inc., Dr. P.D. Weiner, December 1, 1976.

3. API Recommended Practice for Drill Stem Design and Operating Limits (API RP-7G) Dallas, January 1981 and Supplement 2, 1983.

4. Memorandum Report to the Deep Sea Drilling Project from Dr. P. D. Weiner, dated May 10, 1976.

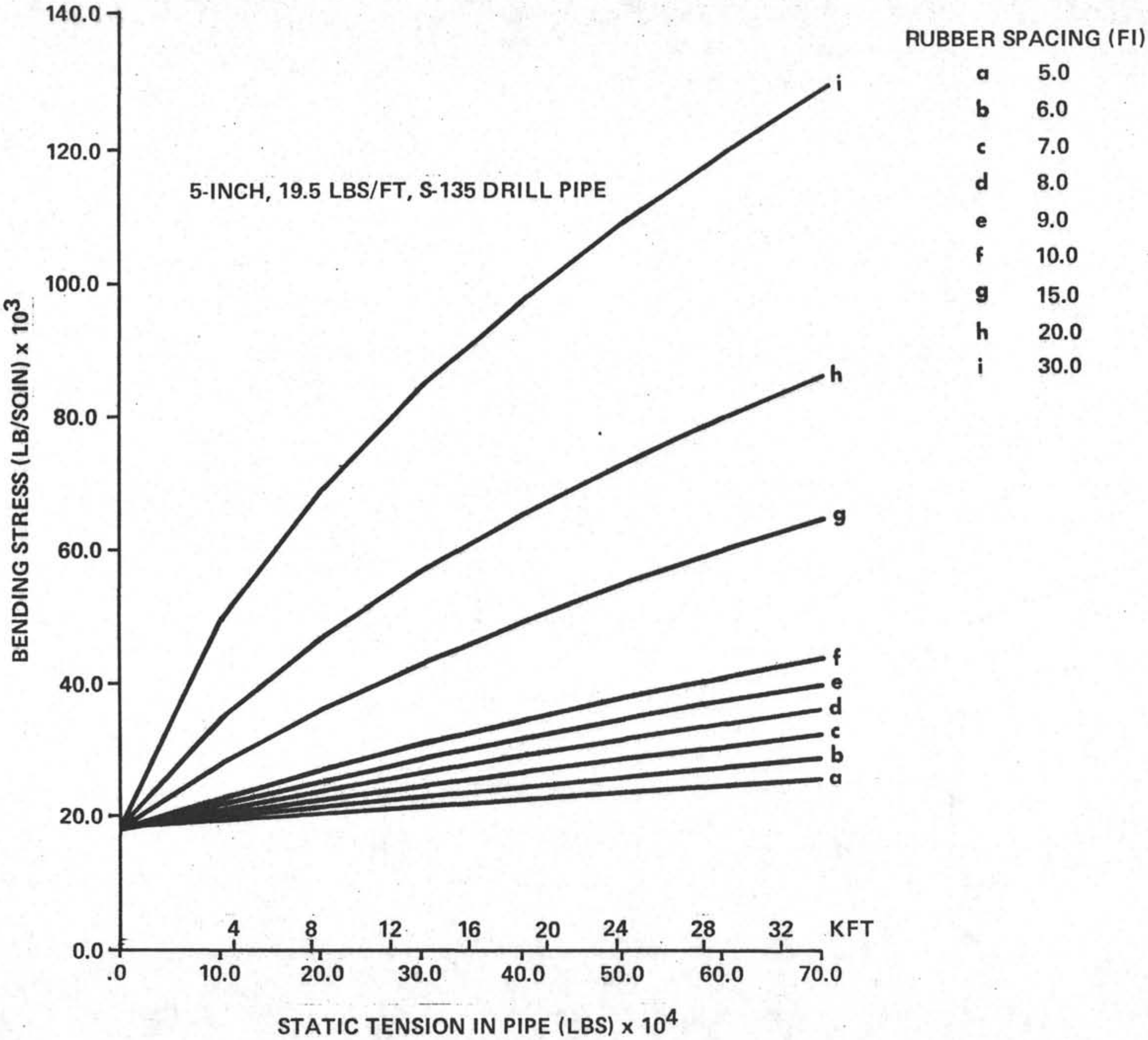
5. Eichberger, L. C., "Experimental Evaluation of Optimum 5-1/2" I.F. Connection Design". Report submitted by Weatherford/Lamb to the Deep Sea Drilling Project, Scripps Institution of Oceanography, March 1978.

6. Internal Deep Sea Drilling Project memorandum from Mr. B. W. Adams, dated May 8, 1978.



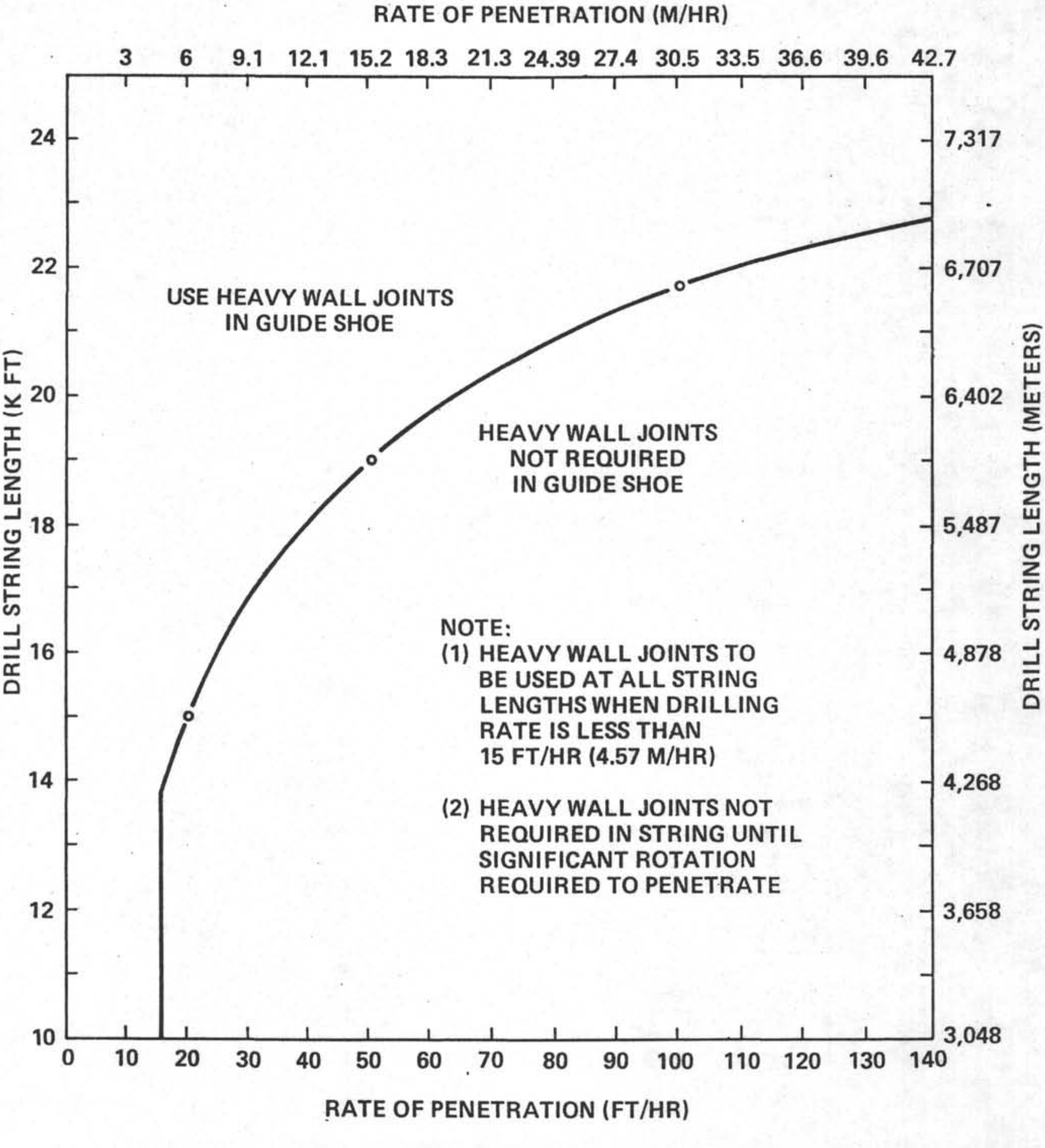
DRILL STRING IN GUIDE SHOE

FIGURE 1



DEPENDENCY OF BENDING STRESS ON RUBBER SPACING

FIGURE 2



HEAVY WALL DRILLING JOINT GUIDE

FIG. 3

APPENDIX I

SPECIAL HEAVY WALL DRILL PIPE FOR THE DEEP SEA DRILLING PROJECT

August 6, 1975

Deep Sea Drilling Project
University of California
Scripps Institution of Oceanography
La Jolla, California 92037

Subject: Transmittal of Results and Report
Independent Consultant Agreement No. 0332.

Attached is my report entitled "Special Heavy Wall Drill Pipe for the Deep Sea Drilling Project". The report is presented in two sections, namely the "SPECIFICATION" and the "DEVELOPMENT". As agreed I might, if needed, make a trip to La Jolla.

The essential findings and comments are as follows:

1. You might wish to change and/or complete the specification.

2. I have chosen an OD of 5.725 inches (wall thickness of 0.8 inches). If necessary, an OD of 5.525 inches (wall thickness of 0.7 inches) would also be satisfactory.

3. I have chosen a hub spacing of 5 feet. A spacing of 6 or even 8 feet could be adopted, with a somewhat shorter fatigue life.

4. At the bottom of the guide shoe drill pipe is always bent, except in completely calm seas. If a length (joint) of pipe were rotated and maintained under tension at the bottom of the guide shoe, a fatigue failure could be expected in this length in 17 days. If, on the other hand all of the 500 feet of heavy wall drill pipe is

subjected to equal bending exposure, then no failure should be encountered until 11 years. In normal drilling operations, in which 500 feet of heavy drill pipe are used, all lengths (joints) of pipe are subjected to essentially equal bending exposure. Lengthy rotation of the pipe without simultaneous vertical advance should be avoided.

5. The reason for using heavy wall drill pipe is to improve the fatigue resistance. However with the heavy wall drill pipe, torsional and tensile stresses become smaller, which justifies the value of minimum yield strength of 120,000 psi included in the Specification. It is rather fortunate that a yield strength smaller than 135,000 psi could be tolerated, as in the manufacturing process, thick walls would have to be heated to a higher temperature than ordinary S-135 pipe. With the same chemical composition one could not maintain the same yield value (135,000 psi). Thus to maintain the same yield value it would be necessary to increase the alloy content of steel, which in turn could make it more brittle and destroy the fatigue characteristic which we must try to maintain.

You could request tool joint design from Drilco in Houston. Please contact there S. T. (Sam) Crews, Manager of Product Engineering.

As requested I am returning ~~herewith~~ under separate cover Lockman's report and the Technical Report No. 4.

Yours very truly,

Arthur Lubinski

Arthur Lubinski
Technical Consultant

AL:imj
Attachments

SPECIAL HEAVY WALL DRILL PIPE

FOR THE

DEEP SEA DRILLING PROJECT

Report prepared by

Arthur Lubinski, Technical Consultant

August 6, 1975

SPECIFICATION

SPECIAL HEAVY WALL DRILL PIPE

Joints (lengths) of special heavy wall drill pipe shall be 30 feet long.

ID = 4.125 inches (4-inch drift)

OD = 5.725 inches

The pipe shall be provided with 7" OD hubs. The pipe and hubs shall be one piece made from a steel bar or a tube. The OD of a hub shall be 7 inches, over a length of 12 inches. The shape of machined surfaces shall be such that stress raisers are avoided. The distances (axis to axis) between hubs (or between a hub and a tool joint) shall be 5 feet.

The provision for elevator suspension shall be the same as in the drill pipe presently used in the Deep Sea Drilling Project.

The minimum yield strength shall not be less than 120,000 psi. The notch toughness, and fatigue characteristics in salt water shall be the same as those of S-135 steel. To ascertain this, bars (or tubes) from one heat will be set aside to make the product. One bar (or tube) will be heat treated. Thereafter specimens will be taken and fatigue tests conducted in conditions similar to tests made on S-135 steel for the Mohole Project.

TOOL JOINTS

OD = 7 inches

ID = 4.125 inches (4-inch drift)

The tool joints shall be able to withstand the following:

(1) a static bending moment of 25,300 ft lb applied simultaneously with a static pull of 632,000 lb.

(2) in fatigue in salt water, a fully reversed bending moment of 25,300 ft lb applied simultaneously with a pull of 400,000 lb.

Tool joints may either be (1) machined at the end of special heavy wall drill pipe lengths; or (2) welded to drill pipe lengths.

Tool joints presently used by the Deep Sea Drilling Project are satisfactory and, therefore, similar design and metallurgical properties are desirable.

DEVELOPMENT

C_o = pipe curvature at the tool joint

C = guide shoe curvature $= \dfrac{1}{350 \times 12} = 238.095 \times 10^{-6}$ in $^{-1}$

Assume first that $C_o = C$, i.e. that drill pipe centralization, thanks to hubs and tool joints, is perfect. This assumption will be removed later.

σ_B = bending stress

From Eq. (7), Ref. 1

$$\sigma_B = \frac{EDC_o}{2}$$

E = Young's modulus $= 30 \times 10^6$ lb/in^2

$$\sigma_B = \frac{30 \times 10^6 \times 238.095 \times 10^{-6} \times D}{2} = 3571.429D$$

A = cross-sectional area of pipe wall

Tension = **400,000 lb**

σ_t = tensile stress

Pipe ID = 4.125

$$\sigma_t = \frac{400{,}000}{\frac{\pi}{4}(D^2 - 4.125^2)} = \frac{509{,}296}{D^2 - 17.016}$$

To make allowance for the effect of the tensile stress σ_t on fatigue, σ_B must be multiplied by the correction factor γ (Ref. 2, Eq. (1))

$$\gamma = \frac{t}{t - \sigma_t} = \frac{1}{1 - \frac{\sigma_t}{t}}$$

σ_t = tensile stress

t = tensile strength = 150,000 psi

Thus the bending stress to be considered, i.e. the bending stress adjusted for tension is

$$\gamma\sigma_B = \frac{\sigma_B}{1 - \frac{\sigma_t}{t}}$$

which is the same as Eq. (2.42) page 99, Ref. 3, with changed notation, except that the latter contains a correction factor k (K in Ref. 3) to account for the corrosive nature of sea water. k = 0.9.

$$\gamma\sigma_B = \frac{\sigma_B}{k\ (1 - \frac{\sigma_t}{t})}$$

$$\gamma\sigma_B = \frac{3571.429\ D}{0.9\ (1 - \frac{509{,}296}{D^2 - 17.106} \times \frac{1}{150{,}000})}$$

$$= \frac{3968.254\ D}{1 - \frac{3.395305}{D^2 - 17.016}} \qquad (1)$$

Introduction of the coefficient k is not logical, as the fatigue data to be used are for sea water environment. However, as use of k yields more conservative results, k is used in the calculations of this report.

A curve drawn through points for fatigue experiments in sea water is given in Fig. 11, Ref. 4, and Fig. 1, Ref. 5. Stresses which will be considered farther in this report are of the order of 30,000 psi. For such stresses the solid curve is more conservative than the fitted dashed curve B of Fig. 1, Ref. 5. For this reason the solid curve will be used. This curve is redrawn in the Fig. 1 of this report.

In Fig. 1, the abscissa is the number of revolutions of drill pipe to failure. The ordinate is the bending stress $\gamma\sigma_B$ adjusted for tension.

The following table was obtained using Eq. (1).

Wall pipe thickness in	Outside diameter D in	$\gamma\sigma_B$	
0.3	4.725	52,006	
0.4	4.925	36,805	
0.5	5.125	32,132	
0.6	5.325	30,162	
0.7	5.525	29,285	
0.8	5.725	28,957	
0.9	5.925	28,945	(minimum)
1.0	6.125	29,130	
1.5	7.125	31,436	
2.0	8.125	34,643	
3.0	10.125	41,840	

These results are plotted in Fig. 2.

The bending stress corrected for tension, $\gamma\sigma_B$, is minimum for wall thickness of 0.9, which corresponds to D = 5.925. $\gamma\sigma_B$ is nearly the same for wall thickness of 0.8, D = 5.725, which are the values adopted here. It is believed that adequate tool joint design is feasible for an OD not exceeding the permissible value of 7 inches. However, if necessary, a pipe thickness of 0.7 inches would be acceptable too, although this would result in somewhat shorter fatigue life than calculated in this report.

For pipe of wall thickness of 0.8, various values of the spacing between hubs (and between a hub and a tool joint) will now be considered. It is assumed that the OD of hubs is the same as that of tool joints.

From Eq. (12), Ref. 1, we obtain the pipe curvature C_o (which is proportional to the bending stress) as function of the guide shoe curvature C.

$$C_o = C \frac{KL}{\text{Tanh}KL}$$

L = half distance between hubs (and between a hub and a tool joint)

K is given by Eq. (2t) Ref. 1.

$$K = \sqrt{\frac{T}{EI}}$$

T = tension in the pipe = 400,000 lb

E = Young's modulus

I = moment of inertia with respect to diameter

$$I = \frac{\pi}{64}(D^4 - 4.125^4) = \frac{\pi}{64}(5.725^4 - 4.125^4)$$

$$K = \sqrt{\frac{0.4 \times 10^6}{30 \times 10^6 \times \frac{\pi}{64}(5.725^4 - 4.125^4)}} = 0.018605 \text{ in}^{-1}$$

The above equation may also be written as follows:

$$(\gamma\sigma_B)_o = \gamma\sigma_B \frac{KL}{\text{Tanh}KL}$$

in which $\gamma\sigma_B$ corresponds to perfect centralization ($\gamma\sigma_B$ = 28,957 psi), and $(\gamma\sigma_B)_o$ corresponds to actual centralization.

Distance between hubs (and between a hub and a tool joint) ft.	L. in	$\frac{KL}{\text{Tanh}KL}$	$(\gamma\sigma_B)_o = 28957 \frac{KL}{\text{Tanh}KL}$ psi	Revolutions to failure N
0	0	1.0000	28 957	960 000
3	18	1.0371	30 030	860 000
4	24	1.0656	30 860	800 000
5	30	1.1017	31 900	730 000
6	36	1.1452	33 160	630 000
8	48	1.2527	36 274	480 000

The last column in the above table is obtained from Fig. 1. The shorter the spacing between hubs (and a hub and a tool joint), the greater is the number of revolutions to failure, i.e. the longer is the life of the pipe. A spacing of 5 feet is adopted in this project. The above table indicates how much could be gained through a shorter spacing, or lost through a longer spacing.

From now on the subscript "o" will not be used in $(\tau\sigma_B)_o$ when referring to the adopted value, $\tau\sigma_B = 31,900$ psi.

The corresponding value of revolutions to failure N = 730,000 would hold true for a length of pipe (commonly called joint of pipe) which would be permanently located at the bottom of the guide shoe. Such a length of pipe would be permanently bent, while a length located half way into the guide shoe would be subjected to bending only when the vessel angular deflection from vertical is between 4^o and 8^o; finally a length of pipe located at the top of the guide shoe would be bent only when vessel deflection is 8^o.

Heavy drill pipe should be used in such a way that every length is subjected to equal bending exposure. In other words lengths of heavy pipe should not always be run in the same sequence.

The resultant angular motion of the pitch and roll of the vessel will be referred to as pitch.

During a pitching motion of amplitude α, the pipe is rotated while drilling or coring. Therefore the angular deflection of the ship varies during one quarter of the pitch cycle between α and zero. As a result, the number of revolutions to failure doubles if the pipe is run so as to achieve equal exposure to bending, with respect to the case of the same joint of pipe being located at the bottom of the guide shoe. Thus $\tau\sigma_B$ becomes

$$\tau\sigma_B = 2 \times 730{,}000 = 1{,}460{,}000 = 1.46 \times 10^6 \text{ revolutions to failure.}$$

An estimate of Glomar Challenger pitch history (actually the combination of pitch and roll) is as follows

Pitch Interval		Fraction of Time
$7^\circ - 8^\circ$		0.001
$6^\circ - 7^\circ$		0.001
$5^\circ - 6^\circ$		0.008
$4^\circ - 5^\circ$		0.02
$3^\circ - 4^\circ$		0.05
$2^\circ - 3^\circ$		0.12
$0 - 2^\circ$		0.80
	Total	1.000

The number of revolutions to failure, taking this pitch distribution into consideration, is calculated in the following table.

Pitch Interval	Average Pitch	Revolutions to Failure	Fraction of Life Expended per Revolution
$7^\circ - 8^\circ$	7.5°	$1.46 \times 10^6 \times \frac{500}{49} \times \frac{8}{7.5}$	$\frac{1}{1.46} \times 10^{-6} \times \frac{49}{500} \times \frac{7.5}{8} \times 0.001 = 0.000063 \times 10^{-6}$
$6^\circ - 7^\circ$	6.5°	$1.46 \times 10^6 \times \frac{500}{49} \times \frac{8}{6.5}$	$\frac{1}{1.46} \times 10^{-6} \times \frac{49}{500} \times \frac{6.5}{8} \times 0.001 = 0.000055 \times 10^{-6}$
$5^\circ - 6^\circ$	5.5°	$1.46 \times 10^6 \times \frac{500}{49} \times \frac{8}{5.5}$	$\frac{1}{1.46} \times 10^{-6} \times \frac{49}{500} \times \frac{5.5}{8} \times 0.008 = 0.000369 \times 10^{-6}$
$4^\circ - 5^\circ$	4.5°	$1.46 \times 10^6 \times \frac{500}{49} \times \frac{8}{4.5}$	$\frac{1}{1.46} \times 10^{-6} \times \frac{49}{500} \times \frac{4.5}{8} \times 0.02 = 0.000755 \times 10^{-6}$
$3^\circ - 4^\circ$	3.5°	$1.46 \times 10^6 \times \frac{500}{49} \times \frac{8}{3.5}$	$\frac{1}{1.46} \times 10^{-6} \times \frac{49}{500} \times \frac{3.5}{8} \times 0.05 = 0.001468 \times 10^{-6}$
$2^\circ - 3^\circ$	2.5°	$1.46 \times 10^6 \times \frac{500}{49} \times \frac{8}{2.5}$	$\frac{1}{1.46} \times 10^{-6} \times \frac{49}{500} \times \frac{2.5}{8} \times 0.12 = 0.002517 \times 10^{-6}$
$0 - 2^\circ$	1°	$1.46 \times 10^6 \times \frac{500}{49} \times \frac{8}{1}$	$\frac{1}{1.46} \times 10^{-6} \times \frac{49}{500} \times \frac{1}{8} \times 0.80 = 0.006712 \times 10^{-6}$

Fraction of Life Expended per Average Revolution $= \overline{0.011939 \times 10^{-6}}$

Average number of revolutions

$$\text{to failure} = \frac{1}{0.011939 \times 10^{-6}} = 83.759 \times 10^6$$

The above table is explained in the text which follows.

Assume that 500 feet of heavy drill pipe are being used in proper sequence to achieve equal exposure to bending. As the height of the guide shoe is 350 ft x sin 8° = 49 ft, the previously calculated number of revolutions to failure (1.46×10^6) must be multiplied by 500/49. For an 8° pitch all the pipe in the guide shoe is bent. Consider for instance 3.5° average pitch. Then only the fraction (3.5)/8 of the pipe in the guide shoe is bent. Thus the number of revolutions to failure becomes

$$1.46 \times 10^6 \times \frac{500}{49} \times \frac{8}{3.5}$$

This number pertains to a case of constant pitch of 3.5 degrees. In other words this would have been the number of revolutions to failure if the pitch did not vary with time but remained equal to 3.5° until failure.

In such a case the fraction of life expended per cycle would have been the reciprocal of the number of revolutions to failure. As actually the 3.5 degrees average pitch occurs only a fraction 0.05 of time, this reciprocal must be multiplied by 0.05. Thus the fraction of life expended for 3.5° average pitch becomes

$$\frac{1}{1.46} \times 10^{-6} \times \frac{49}{500} \times \frac{3.5}{8} \times 0.05$$

which is entered in the table.

Fractions of life expended are additive, and their sum is entered in the above table.

The reciprocal of their sum is the average number of revolutions to failure.

Let us assume that the drilling string is rotated by 60 RPM. Then

$$84 \times 10^6 \text{ Rev} = 84 \times 10^6 \text{ sec} = \frac{84 \times 10^6}{3600 \times 24} \text{ days} = 969 \text{ days}$$

As drilling or coring is conducted only 1/4 of total time, the above number becomes

$$969 \times 4 = 3876 \text{ days} = 10.6 \text{ years}$$

It should be emphasized once again that this value is predicated upon the assumption that the heavy wall drill pipe is used in such a manner that all lengths (joints) of pipe are subjected to equal bending exposure. A pipe rotated while permanently located at the bottom of the guide shoe, will fail after 1.46×10^6 revolutions or about 1.46×10^6 sec = 17 days.

In the Specification a minimum yield strength of 120,000 psi has been written. Justification of this value is given below

$$\text{Tensile stress } \sigma_t = \frac{632{,}000 \text{ lb}}{A} = \frac{630{,}000}{\frac{\pi}{4}(5.725^2 - 4.125^2)} = 51{,}059 \text{ psi}$$

$$\text{Bending stress} = \sigma_B = \frac{EDC}{2} \times \frac{KL}{\text{Tanh}KL} = 3{,}571 \text{ D} \times \frac{KL}{\text{Tanhkl}}$$

$$= 3{,}571 \times 5.725 \times \frac{KL}{\text{Tanh}KL} = 22{,}526 \text{ psi}$$

$$\sigma_t + \sigma_B = 73{,}585$$

Using a coefficient of safety of 1.6

$$\sigma_t + \sigma_B = 1.6 \times 73{,}585 = 117{,}736 \text{ psi}$$

which shows that a minimum yield strength of 120,000 psi is adequate as far as tension and bending are concerned.

From present Deep Sea Drilling Project operations we know that 5" 19.5 lb/ft drill pipe is adequate in torsion. In heavy wall drill pipe the torsional stress will be smaller in the ratio of wall cross-sectional areas, and larger in the ratio of O. D.'s.

$$\frac{5.275}{12.378} \times \frac{5.725}{5} = 0.488$$

The ratio of yield strengths is

$$\frac{120{,}000}{135{,}000} = 0.889$$

Thus a minimum yield strength of 120,000 psi is also adequate as far as torsion is concerned.

It is written in the Specification that tool joints must be able to withstand a bending moment of 25,300 ft lb. The calculation from which this value has been obtained is as follows.

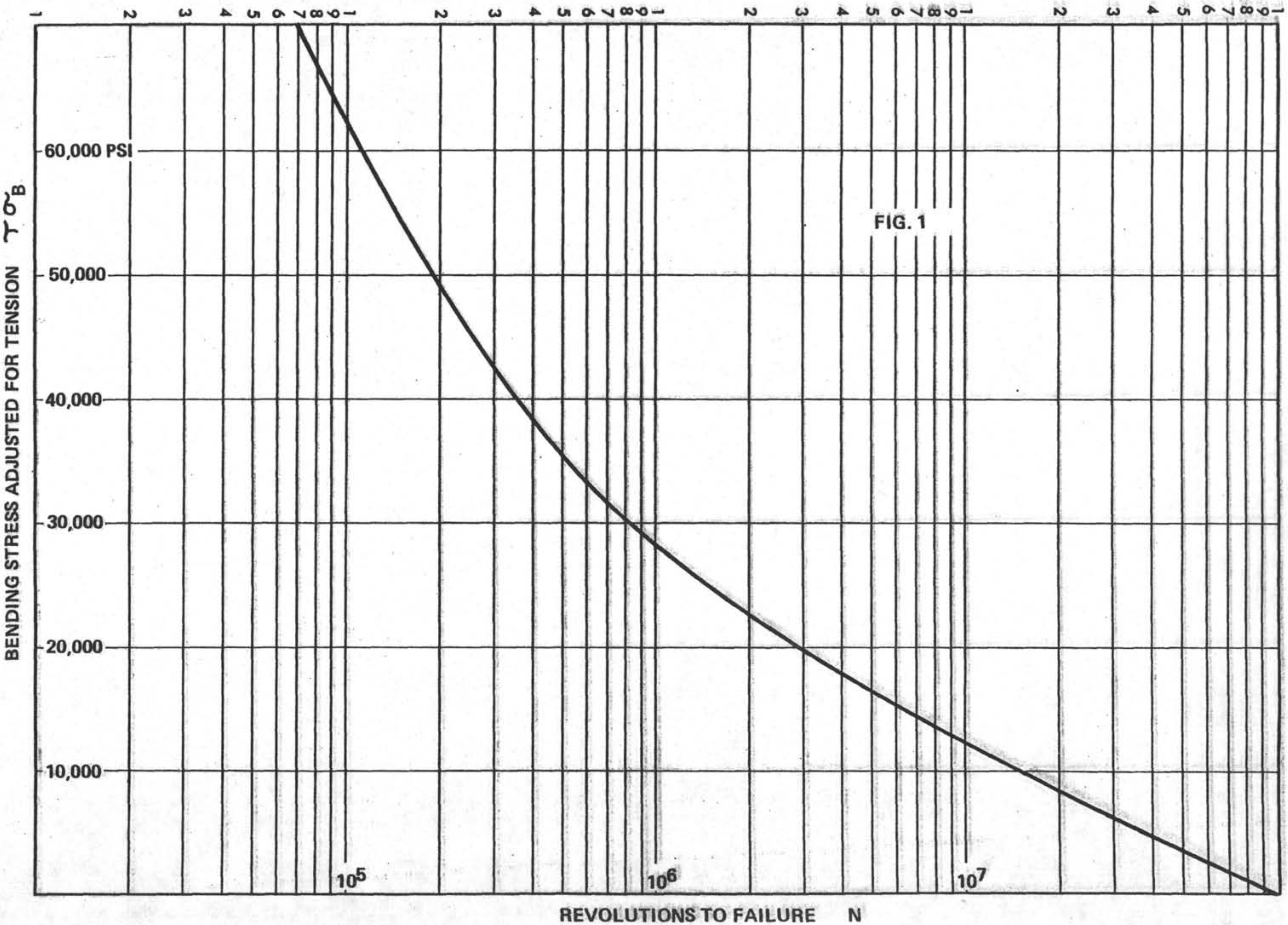
$$M_B = EIC_o$$

$$= 30 \times 10^6 \frac{\text{lb}}{\text{in}^2} \times \frac{\pi}{64} (5.725^4 - 4.125^4) \text{ in}^4 \times \frac{1}{350 \times 12} \times 1.1017 \text{ in}^{-1}$$

$$= 303{,}120 \text{ in} \times \text{lb} = 25{,}260 \text{ ft} \times \text{lb}$$

REFERENCES

1. "Maximum Permissible Dog-Legs in Rotary Boreholes" by Arthur Lubinski. Transactions of AIME 1961, Vol. 222.
2. "The Effects of Drilling Vessel Pitch or Roll on Kelly and Drill Pipe Fatigue" by John E. Hansford and Arthur Lubinski. Transactions of AIME 1964, Vol. 231.
3. "Deep Sea Drilling Project, Technical Report No. 4", Scripps Institution of Oceanography, University of California at San Diego.
4. "Drill-Pipe Fatigue Failure" by H. M. Rollins, The Oil and Gas Journal, April 18, 1966.
5. "Drill String Analysis Report for Deep Sea Drilling Project", by James C. Lockman, Global Marine Inc., February 4, 1975.



BENDING STRESS ADJUSTED FOR TENSION $T\sigma_B$
60,000 PSI
50,000
40,000
30,000
20,000
10,000
10^5
10^6
10^7
REVOLUTIONS TO FAILURE N

FIG. 1

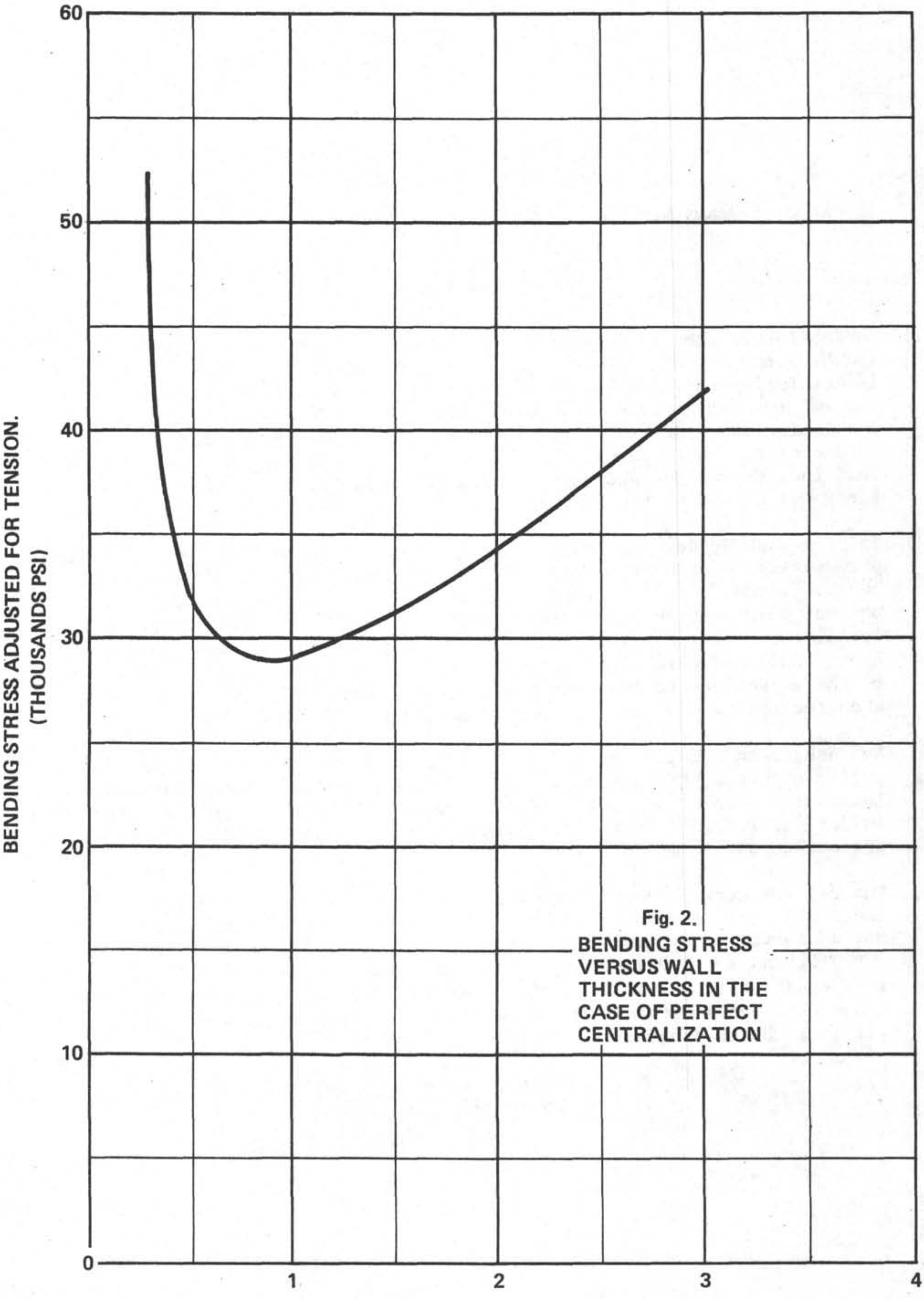


Fig. 2.
BENDING STRESS VERSUS WALL THICKNESS IN THE CASE OF PERFECT CENTRALIZATION

Some Comments on the Report,

"Special Heavy Wall Drill Pipe for the Deep Sea Drilling Project"

By

Arthur Lubinski

1) Lubinski says the introduction of the coefficient K is not logical (Development, p. 2). The fatigue data from the Rollins paper, Fig. 11, is for full-sized S-135 drill pipe testing in sea water. At 60 cycles/min and 10^6 cycles to failure, the test duration was 277h 47min. Actual drill pipe is subjected to sea water corrosion for much longer periods, so use of the coefficient to account for corrosion over a longer period than that in the fatigue tests is logical. Perhaps a K = 0.9 is too low, but K = 1.0 must be too high.

2) In paragraph 3, Development, p. 5, Lubinski states that "a length of pipe permanently located at the bottom of the guide shoe would be permanently bent...", and "a length of pipe located at the top of the guide shoe would be bent only when vessel deflection is 8°". Since the roll and pitch is oscillatory about a mean value of zero, no point in the pipe is permanently bent. Roll and pitch less than 8° does <u>not</u> cause the pipe at the <u>bottom</u> of the guide shoe to bend, while pipe at the <u>top</u> of the guide shoe bends when the roll and pitch is greater than zero.

3) Lubinski uses a factor of 2 (in revolutions to failure) to account for the cyclic nature of the pitch (or roll). This is an upperbound estimate (see Lubinski's Ref. 3, p. 105), and the lowerbound is 1.47. The factor used in Ref. 3, 1.57, is my earlier estimate for this factor, and is what I propose to use in my analysis.

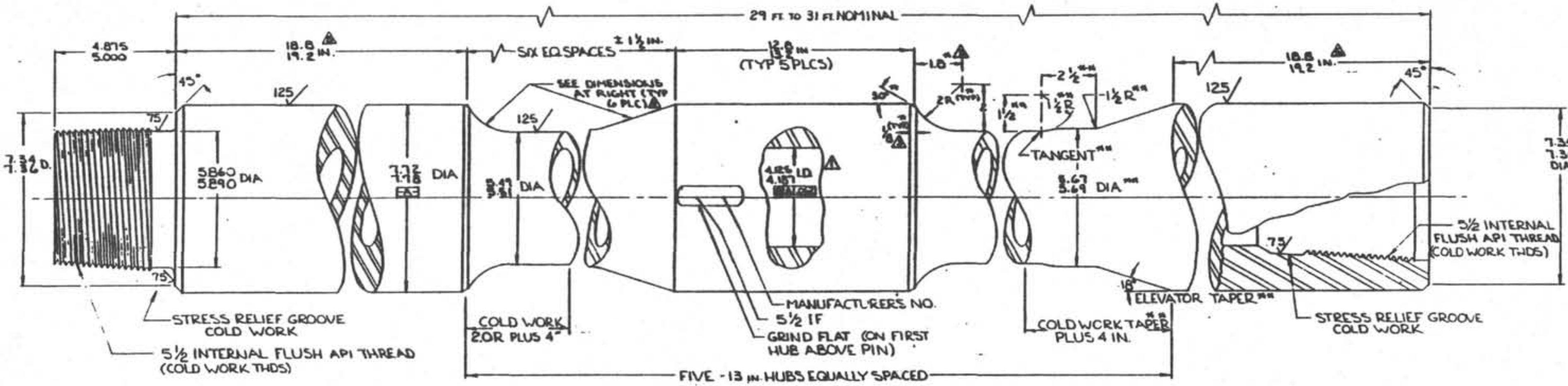
4) The fatigue curve used by Lubinski is the curve drawn in the Rollins paper (Lubinski Ref. 4). Seven of 15 test points lie below this curve, so it is not a conservative curve. The fatigue curve I used with σ = 1102.3, m = 3.82 and K = 0.9 is more conservative above 23 KSI (<2 x 10^6 cycles and less conservative below 23 KS (>2 x 10^6 cycles). If your operating experience could give us a better estimate of the fatigue curve, we can easily fit it into the program.

T. Vreeland, Jr.

APPENDIX II

FABRICATION DRAWING AND SPECIFICATION
FOR
HEAVY WALL DRILLING JOINT

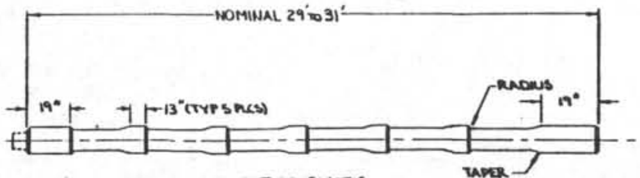
REVISIONS			
NO	DESCRIPTION	DATE	BY
01	4.125 DIA WAS 4. DIA		RK
02	TITLE WAS "OPTIMUM HEAVY WALL DRILL PIPE"		RK
03	30° x ⅜ CHAMF WAS .2R		RK
4	ELEVATOR TAPER WAS ONLY @ 1ST UPSET @ BOX	9-30-81	RK
5	1.8 was 2, 18.8/19.2 WAS 13, REDIMENSIONED	2-18-82 5-25-81	RK



* TYP ALL UPSETS TOWARD PIN END

** TYP. ALL UPSETS TOWARD BOX END

NOTE:
IF THE BAR IS TREPANNED FROM BOTH ENDS, THE MANUFACTURER MUST INSURE THE BORES MEET AT THE MIDPOINT OF THE CENTER HUB. ALLOWABLE LONGITUDINAL TOLERANCE IS ± 3.5 INCHES FROM THE CENTER HUB MIDPOINT.



LAYOUT SCHEMATIC
NO SCALE

NOTE:
PHYSICAL PROPERTIES AT THE 5.7 IN. ROUND LEVEL:

TENSILE STRENGTH	MIN 140 KSI
YIELD STRENGTH	MIN. 120 KSI
ELONGATION	MIN 14%

CHARPY (SPECIMEN TYPE A) MIN. 60 FT-LBS } AS PER
OR IZOD (SPECIMEN TYP D) MIN 40 FT-LBS } ASTM E23-72

WEIGHT IN AIR: 52.6 LBS/FT
WEIGHT IN WATER: 50.0 LBS/FT

MAT'L:
AISI 4145 STEEL OR 43XX SERIES

UNIV. OF CALIF.
DEEP SEA DRILLING PROJ
HALF
30 SEP 74 BURTON W. ADAMS
RK
HEAVY WALL DRILL JOINT
PART NO OG 0610-5 D-060605

DEEP SEA DRILLING PROJECT
SPECIFICATION: OG-0610-06
PROCUREMENT OF HEAVY WALL DRILL JOINTS
February 6, 1984

General Description and Intended use

The Deep Sea Drilling Project, which operates the drillship *GLOMAR CHALLENGER* intends to procure special design, heavy wall drill joints to run at the top end of the drill string. This design will minimize tensile and cyclical stresses in the drill pipe rotary connections that are subjected to high tensile and bending loads in the *CHALLENGER'S* guide shoe or horn. Drill pipe in the guide shoe is nominally subjected to 400,000 lbs. tensile loading in combination with 16°/100 ft. bending.

Design

The heavy wall drill joints shall be machined as per Deep Sea Drilling Project Drawing No. D-0523-05. Practice and workmanship shall be in accord with API Specification 7.

Rotary Connections

Connections

5-1/2" Internal Flush

Shoulders

They are to be 7.75 inches O.D. and a minimum of 13 inches long. Surface finish is 125 (face 75). If machining to 7.75 O.D. does not clean up the shoulder completely, the hot rolled, mill finish is acceptable. Surface imperfections are to be removed by grinding but the removal of such imperfections shall not result in stock removal in excess of 0.125 inches.

Threads

Threads must be hob cut, or cut by numerical control, and cold worked. They are to be treated witha suitable anti-galling treatment.

Stress Relief Grooves

The connections shall have stress relief grooves. The grooves shall be cold worked.

Hubs

Integrally machined hubs are to be equally spaced between the rotary connections. The hubs are 7.75-inches O.D. and 13-inches long. They will be approximately five feet on centers depending on the drill collar bar length.

The transition from the tube to the box connection is via a two inch wide, 5.68-inch O.D. elevator shoulder and an 18° taper. The transition from the tube to the elevator shoulder is a 1.5-inch radius tangent to the [illegible]. The transition from the elevator shoulder to the 18° taper is also a 1.5-inch radius, tangent to both surfaces.

The 18° taper, the 5.68-inch O.D. elevator shoulder, the 4-inches of 5.50 inch O.D. tube adjacent to the hubs and the transition radii are to be cold worked.

Marking

A flat will be milled or ground in the first hub above the pin end. The manufacturer's number and connection identification are to be stamped on the flat.

Trepanning

A. Drilling Joints

If the bar is to be trepanned from both ends, the manufacturer must ensure the bores meet at midpoint of center hub. Allowable longitudinal tolerance is ± 3.5 inches from center hub midpoint. Any step resulting from the trepanning should be removed by honing to a radius of at least .425 inches.

Drift

The joints must pass a 10-foot long 4.0 inch drift.

Material Properties

Material

AISI 4145 or AISI 43XX Series

The joints must exhibit the following mechanical properties at the 5.7-inch round level (approximately one-inch below the 7.75-inch O.D.).

Tensile Strength	-	min 140 KSI
Yield Strength	-	min 120 KSI
Elongation	-	min 14%
Toughness	-	Either Charpy or Izod impact test may be used. Tests shall conform to ASTM E23-72 Charpy (Specimen A) - min 60 ft-lbs Izod (Specimen D) - min 40 ft-lbs

At the minimum, the manufacturer will physically test each heat of steel used to demonstrate that the above specifications are met. If one heat is used, two bars will be tested. The manufacturer will supply DSDP with copies of the test results.

Inspection and Quality Control

Each joint will be completely inspected for any defects that would initiate a structural, fatigue, or corrosion failure, i.e., every square inch of tube wall is to be inspected. The manufacturer will supply DSDP with copies of the test results.

The hardness of each rotary connection pin is to be measured. Hardness measurement may be made on blanked pin. The pin hardness measurements are to be identified by the manufacturer's joint number and supplied to DSDP.

APPENDIX III

EXPERIMENTAL EVALUATION OF OPTIMUM 5-1/2" I.F. CONNECTION DESIGN

EXPERIMENTAL EVALUATION OF OPTIMUM 5-1/2" I.F. CONNECTION DESIGN

L. C. EICHBERGER
Manager,
Engineering and Technical Services

ENGINEERING AND TECHNICAL SERVICES

WEATHERFORD/LAMB U.S., INC.

Houston, Texas

March, 1978

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1.0 SUMMARY

The data for the torque make-up of the test configuration for the 120% optimum torque case is taken to be representative of the conditions which exist in the connection. The make-up stress in the stress relief groove of the connection is 1.76 - 1.81 greater than the calculated stress for the 100% optimum torque case. Tensile axial stresses are present in a region adjacent to the shoulder of the connection in the box, instead of, compressive stresses indicating the presence of local bending.

The average stress values are used to describe the conditions which exist in the connection due to an axial load. The average axial stress at the stress relief groove of the connection is 1.07 - 1.29 greater than the calculated value for an axial load of 402,025 lb. The maximum axial stress in the test configuration for an axial load occurs at the base of the fillet radius where the tube intersects the hub of the drilling sub. This stress is 1.17 - 1.40 greater than the stress in the tube.

The stresses in the connection produced by a bending moment are inconsistent, however, they represent the condition in the connector of the drilling sub when it is in contact with the horn on a drilling ship. The point of maximum stress in the test configuration is at the base of the fillet radius where the tube and hub intersect. The axial stress, at this point, is 1.18 - 1.19 greater than the bending stress in the tube at the center of the configuration.

The axial stress in the stress relief groove of the connection for a combined axial load of 402,025 lb. and a bending moment (one which produces a deflection in the test configuration of 16° per 100 ft.) is within +1957 psi and -4029 psi of the calculated value of 27,047 psi. The maximum axial stress in the test configuration for the combined loading is at the base of the fillet radius, and is greater than the axial stress in the tube at the center of the test configuration by a factor of 1.33 - 1.45.

The axial stress in the stress relief groove of the connection produced by a make-up torque of 47,124 ft-lb, an axial load of 402,025 lb. and a bending moment (one which will produce a deflection in the test configuration of 16° per 100 ft) exceeds the yield strength of the material.

2.0 CONCLUSIONS

The test results of this investigation support the following conclusions:

a) The calculated value for make-up stress in the stress relief groove of the connection, in general, is substantially lower than the stress in the connection.

b) The connection yields, when the make-up torque is equal to, or greater than, 100% optimum torque or 47,124 ft-lb, and for an axial load of 402,025 lb. and a bending moment which will produce a deflection of 16° per 100 ft. in the test configuration.

c) The maximum stress concentration factor for the test configuration subjected to an axial load of 402,025 lb. and a bending moment which produces a deflection in the test configuration of 16° per 100 ft., is 1.45.

3.0 RECOMMENDATIONS

Based on the test results of this investigation, the following recommendations are made:

a) Reduce the 100% optimum torque from 47,124 ft-lb to approximately 27,000 ft-lb. The 27,000 ft-lb of torque will produce an axial stress at the stress relief groove of the connection of 67,723 psi (Table 1), or the calculated value.

b) Since the recommendation in (a) is based on the test results of this investigation, then it is strongly recommended to test another instrumented connection in a torsion machine to validate the test results.

4.0 EXPERIMENTAL WORK

4.1 TEST FRAME

The principal function of the test frame is to hold the test configuration in position and to provide a means through which to apply an axial load and a bending moment.

Figure 1 shows the basic details of the test frame. The test frame consists of a horizontal lower frame with integral vertical end members A which supports a removable upper frame B. The lower frame and end members are made from large OD pipe, while the upper frame is a composite of welded I-beams and plate. The test configuration C is held in a horizontal position by pulling adaptors D which are attached to a fixed bearing support E at one end, and by a horizontal support F at the other end. The horizontal support allows free movement of the test configuration in the direction of the configuration's axial axis of symmetry. The pulling adaptors are fitted with self-aligning ball bushings to provide the test configuration with rotational end constraints.

The axial load is applied to the test configuration through the hydraulic cylinder G. The load, which produces the bending moment, is applied through the chain link straps H, the loading beam I and hydraulic cylinder J.

The make-up and break-out of the test configuration's connections is also done while the test configuration is in the test frame. The make-up and break-out is accomplished by the chain tong and back-up shown in Fig. 2a.

4.2 TEST CONFIGURATION

The test configuration consist of three drilling subs, as shown in Fig. 1. The drilling subs are machined from 7-3/4" OD x 4-1/8" ID, AISI 4145 Q & T, 285-341 BHN specially selected material according to specifications established by the Deep Sea Drilling Project, Scripps Institution of Oceanography. The drilling sub connection is the 5-1/2" I.F. connection with a special stress relief groove specified by Dr. P. D. Weiner. The pulling adaptor was provided with a 6" Acme (2 thread/inch) connection.

A total of forty (40) strain gages and sixteen (16) fatigue gages are installed on the drilling subs, as shown in Fig. 3. The strain gages used are a single element foil gage with a 1/8" gage length (Micro-Measurements gage EA-06-125BT-120). The majority of the gages are installed in two rows, along the axial axis of symmetry and diametrically opposed. The exception to this, is at the stress relief groove and the shoulder of the 5-1/2" I.F. connection, where a two gage 90° rosette is installed - one axial and one circumferential.

4.3 DATA COLLECTION AND MEASURING SYSTEM

The strain gages were connected to a forty (40) channel BLH Digital Strain Gage Scanning Unit 1200A. The unit has the capability of automatically scanning each channel, conditioning the signal and displaying the strain value on a LED display and printing the strain value out on a built in printer.

The Scanning Unit is interfaced to a Hewlett Packard Calculator - Computer 9825A with a Digital Plotter - Printer 9871A. Software was developed for the 9825A Computer to allow the Computer to receive and store the strain values from the BLH Strain Gage Scanning Unit on command. Software was also developed to process the data and print it out in controlled format form on the printer, as well as, to plot the data.

Deflections of the test configuration are read from five (5) Mitutoyo Dial Indications of 0.001" increments over a 2" range. The dial indicator locations are shown in Fig. 2b and in Fig. C1 in Appendix C.

The pressure to the hydraulic cylinders for the torque make-up and bending test was read on a 1,500 psi Marsh, Type 100-3 pressure gauge. The smallest division reads 10 psi, which enables an estimate of the pressure reading to 5 psi. The pressure to the hydraulic cylinder for the axial load was read on a 10,000 psi Marsh, Type 100-3 pressure gauge. The smallest division reads 100 psi, which enables an estimate of the pressure to 50 psi. The pressure gauges were calibrated on a dead weight tester before the start of the testing.

4.4 TEST PROCEDURES

The general testing sequence is as follows:

a) Torque connections to 60% of optimum torque.

b) Apply axial of 400,000 lbs. Remove axial load.

c) Apply bending moment to equal an equivalent deflection of 16° per 100 ft. Remove bending load.

d) Combined load. Apply axial load of 400,000 lb., followed by a bending moment such that the deflection at the connection is the same as that note under part C. Remove bending load.

e) Break-out connections.

This sequence was repeated for torquing the connections to 80%, 100% and 120% of optimum torque with one exception, and that is, step E was not executed at the completion of 120% of optimum torque sequence.

The detailed test procedure followed under the above steps is as follows:

The test configuration was initially assembled using Weatherford/Lamb Lub-Guard thread compound on the threads and shoulders of the connection. It was then placed in the test frame and readied for the first test sequence.

4.4.1 Make-up Torque

The make-up torque was applied to the test configuration by a chain tong which had a four (4) foot arm. The test configuration was constrained from rotating by a chain back-up. The chain tong was actuated by a hydraulic cylinder, as shown in Fig. 2a, and made-up both connections at one time. The test configuration was first made-up to a hand tight position, the strain gages were zeroed and their values were recorded. The chain tong was then attached to the drilling sub hub for torquing. Before a torque was applied, each connection was marked so that the amount of turns could be measured for a given torque. The torque was applied in consistent increments for all make-up tests. At each torque level, the weight of the chain tong was removed from the test configuration before the strain gages were read. Also the turns for each connection were noted and recorded.

4.4.2 Axial Load

The test configuration was rotated so that the strain gages on the middle drilling sub were in a vertical plane, odd numbered gages on top and even numbered gages on the bottom. In the no-load state, the strain gages were zeroed and their values were recorded. The axial load was applied in equal and consistent increments for all axial load tests. After each load increment, the strain gages were read.

4.4.3 Bending Load

In the no-load state, the strain gages were zeroed and their values were recorded. Dial indicators were positioned and set and their initial readings were noted and recorded. The bending load was applied in consistent increments for all bending tests. After each load increment, the strain gages were read and the dial indicator readings were recorded.

4.4.4 Combined Load - Axial and Bending

In the no-load state, the strain gages were zeroed and their values were recorded. An axial load of 402,025 lbs. was applied to the test configuration and the strain gages were read. The dial indicators were positioned and set and their initial readings were noted and recorded. The bending load was applied using the procedure described under Section 4.4.3.

4.4.5 Break-out Torque

Break-out torques for each 5-1/2" I.F. connection were noted and recorded. Each connection was backed out until the shoulders of the connection separated by approximately 1/2". The shoulders were inspected for damage and re-doped if no damage was detected. The test configuration was readied for the next testing sequence.

5.0 RESULTS AND DISCUSSION

5.1 GENERAL INFORMATION

Table 1 is a summary of calculated values of the axial stress in the connection for make-up torque, axial and bending loads from Appendix D. A torque of 47,124 ft-lb is the optimum torque value used in this report to establish the 60%, 80%, 100% and 120% optimum torque cases (OTC). The torque of 53,247 ft-lb is the corrected value for the optimum torque and results from a correction to an error found during the preparation of Appendix D.

The strain gages data for the 60%, 80%, 100% and 120% OTC and the corresponding axial and hoop stress values can be found in Appendix A.

Appendix B contains data on connection advancement for each torque increment and for each optimum torque case. It also contains information on strain gage misalignment due to make-up.

Deflection data due to the bending load and the combined loading of axial and bending for the 60%, 80%, 100% and 120% OTC can be found in Appendix C.

5.2 MAKE-UP STRESSES

Table 2 shows the average axial stress at the stress relief groove of the connection for each of the optimum torque cases. The values in Table 2 shows no consistency between connections and optimum torque cases.

The 120% OTC is presented as giving the truest stress values in the connection's pin and box due to make-up. The reason for this is that at each torque increment, the constraints on the test configuration were adjusted so that it would rotate freely before the strain gages were read. In previous make-ups, only the weight of the chain tong was removed from the test configuration before the strain gages were read. Since the test configuration was not checked to see if it could be rotated, then there is the possibility that the test configuration was subjected to a bending moment. The bending moment can come about by the hub of the test configuration rolling up the incline plane of the chain back-up and locking in place preventing the hub from returning to its free position when the chain tong was removed after each torque increment. This concept is supported by the test data which shows unusually high axial stress in the tube of the test configuration for all optimum torque cases except the 120% OTC, (Table A1c, A5c, A9c and A13c in Appendix A).

Figures 4 and 5, respectively, show the make-up stress in the stress relief groove of the connection for the 120% OTC to be practically the same for each connection. The slight change in shape in the initial part of the curve is probably due to overcoming tolerance differences in pin and box of connection which takes place until the shoulders of the connection come into solid contact at approximately 10,000 ft-lb.

Comparison of the axial stress in the stress relief groove in the pin of 119,240 - 122,854 psi (Table 2) with its calculated value of 67,726 psi for a 47,124 ft-lb torque and 120% OTC (Table 1), shows the axial stress to be 1.76 - 1.81 greater than the calculated.

Comparison of the axial stress in the vicinity of box shoulder (0.260" from the shoulder) of 2959 psi at point 2 and 3214 psi at point 13 for a 47,124 ft-lb torque (Table A13c, Appendix A) with the calculated box shoulder stress of -96,708 psi (Table 2), indicates the presence of local bending adjacent to the shoulder. The tensile stresses at the shoulder are accompanied by large hoop tensile stresses of 67,447 and 78,151 psi which supports the concept of local bending adjacent to the shoulder.

Before starting on a new optimum torque case, the connections of the test configuration were broke-out. The break-out torques are as follows:

Optimum Torque %	Make-up Torque ft-lb	Break-out Torque ft-lb
60	28,274	28,903
80	37,699	37,699
100	47,124	47,752

for 100 - 102% break-out to make-up torque.

The advance of each connection for a given torque can be found in Table B1 and Appendix B. Table B1 reveals consistency for the 60%, 100%, 120% OTC in the advancement of the connections during make-up. In the 80% OTC, the advancement of the connections for each torque increment falls far below those of the other optimum torque cases. The stresses in the pin, points 3 and 14, for the 80% OTC (Table A5c, Appendix A) indicates that the torque levels of 9,425, 18,850 and 28,274 ft-lbs should have been all the same. At this point in time, it can only be conjectured as to what happened.

The final advancement of the connections were measured at the maximum torque for the optimum torque cases relative to the 60% OTC, and are as follows: 0.010, 0.030, 0.048 - 0.050 of a turn, respectively, for the 80%, 100% and 120% OTC. These advancements also cause the axial alignment of the strain gage on the three drilling subs of the test configuration to be further misaligned. The axial alignment of the strain gages on the three drilling subs is given in Fig. B1 of Appendix B.

5.3 STRESSES DUE TO AXIAL LOAD

The average axial stress at point 8 in the tube for a 402,025 lb. axial load is 38,610 psi, 38,550 psi, 38,985 psi and 38,665 psi, respectively, for the 60%, 80%, 100% and 120% OTC (Table A2c, A6c, A10c and A14c in Appendix A). The calculated axial stress in the tube of 5-1/2" OD x 4-1/8" ID cross section for 402,205 lb. axial load, is 38,678 psi which is in good agreement with stress values cited above.

Locations of high stress in the test configuration due to an axial load are at points 1, 4, 5, 12, 15 and 16, which are at the transition from tube to hub in the drilling subs, as shown in Fig. 1. Listing the points in descending order of stress values is as follows: 12, 5, 16, 1, 15 and 4, for the 60% and 80% OTC and 12, 5, 16, 15, 1 and 4 for the 100% and 120% OTC. The stress at these points for an axial load of 402,025 lb. compared to the stress in the tube at point 8 is as follows: 1.17 - 1.40 for point 12, 1.16 - 1.17 for point 5, 1.16 for point 16, 1.13 for point 15, 1.12 for point 1 and 1.09 - 1.10 for point 4.

Table 3 shows the average axial stress in the stress relief groove of the connectors for axial load increments and for the optimum torque cases. There is reasonable agreement in stress values between the optimum torque cases for a given axial load with the exception of the 120% OTC. No reason can be offered for the slightly higher stresses in the 120% OTC.

Figure 6 shows a plot of the average axial stress values in Table 3 for the 120% OTC. Figure 7 is a plot of average hoop stress corresponding to the axial stress of Fig. 6. Both plots are nearly linear and both show a slight difference in stress between the two connections of the test configuration.

Comparing the average axial stress values in Table 3 for an axial load of 402,025 lb. with the calculated axial stress value of 15,942 psi (Table 1) for an axial load of 400,000 shows the axial stress to be 1.07 - 1.29 greater.

Table 4 and 5 gives the axial stress in the stress relief groove of the connector for axial load increments and for odd and even numbered gages. Odd numbered gages are located on the top of the test configuration and in a vertical plane containing the axis of symmetry of the configuration. Even numbered gages are diametrically opposite the odd numbered gages. The stress values in these tables show a difference in axial stress between odd and even numbered gages. This difference is attributable to the characteristics of the loading frame not being able to apply a pure axial load to the test configuration. The actual loading imparted to the test configuration is an uniform axial load with a superimposed bending moment, therefore, the true axial stress in the test configuration should be the average axial stress. This concept is supported by comparing the average axial stress values of 38,610, 38,580, 38,653 psi for point 8 on the tube section and an axial load of 402,025 lb. with the calculated axial stress of 38,678 psi for the same tube section and axial load. The average axial stress values are from Tables A2c, A6c, A10c and A14c in Appendix A and show good agreement with the calculated value.

5.4 STRESSES DUE TO BENDING LOAD

Tables 4 and 5 also give the axial stress in the stress relief groove of the connection for bending load increments and for odd and even numbered gages. Once again, there are differences in the absolute values of the axial stress between odd and even numbered gages. These differences are assumed to be caused by the position of the chain-link loading strap on the hubs of the drilling sub relative to the pin of the connection, see Fig. 1., and the characteristics of the connection to transmit the bending moment. These assumptions are supported by the fact that at the center of the test configuration, point 8, the difference in the absolute values of the stress for the odd and even gages, is negligible, less than 2% difference. This can be verified with reference to Tables A3a, A3b, A7a, A7b, A11a, A11b, A15a and A15b in Appendix A.

The loading arrangement used to apply the bending moment to the test configuration was chosen to simulate the conditions which the drilling subs see when they come into contact with the horn on a drilling ship.

Figures 8 and 10 show plots for the bending data given inTables 4 and 5, respectively. The plots show that the axial stress in the stress relief groove at point 3 and point 14 have the same characteristic trends, but with increasing separation of the curves for increasing bending moment.

Figures 9 and 11 show plots of the hoop stress which accompanies the axial stress at the stress relief groove of the connection during bending. The differences seen in the plots is assumed to be attributable to the reasons cited above.

The maximum stress in the test configuration occurs at point 12 for the even numbered gages and a bend moment of 301,425-lb. The stress at point 12 is 1.18 - 1.19 greater than the stress at point 8 in the tube at the center of the test configuration. This can be verified with reference to Tables A3b, A7b, A11b and A15b in Appendix A. The next highest stress occurs at point 5 and is 1.16 - 1.17 greater than the stress at point 8.

5.5 STRESSES DUE TO COMBINED AXIAL AND BENDING LOAD

Tables 4 and 5 also give the axial stress in the stress relief groove of the connection for the combined loading increments for odd and even numbered gages and for the 120% OTC. The axial stress for the combined loading of 402,025 lb. axial load and an equivalent bending moment to bend the test configuration to 16°/100 ft. is the last entry in the combined load section of Tables 4 and 5. The axial stress values from Table 5 are within +1957 psi and -4029 psi of the calculated value of 207,047 psi (Table 1) for the combined load.

When using the data in the combined load section, keep in mind that the bending moment represent the applied moment and has not been corrected for the moment due to the axial force. This would especially be true if the principle of superposition was to be used on the axial load and bending load data.

The maximum stress is at point 12 and is greater than the stress at point 8 in the tube at the center of the configuration as follows: 1.43, 1.33, 1.45 and 1.45 respectively, for the 60%, 80%, 100% and 120% OTC. This can be verified by the data given in Tables A4b, A8b, A12b and A16b of Appendix A.

Table 6 shows the axial stress in the stress relief groove of the connection for combined make-up torque and axial load and combined make-up torque, axial and bending loads. The Table shows that the axial stress for the 100% and 120% OTC is greater than 136,000 psi. The yield strength for the drilling sub material is 128,000 - 133,000 psi. Therefore, for the 100% and 120% OTC the portion of the connection at the stress relief groove has yielded.

The high axial stress in the stress relief groove of the connection is due mainly to the make-up stress. Therefore, this stress can be lower considerably by lowering the make-up torque. If the calculated value for the stress in the stress relief groove of 67,726 psi is a goal to be achieved, then Fig. 4 shows that a make-up torque of approximately 27,000 ft-lb is required. Acceptance of lowering the make-up torque would require additional testing to verify the test results of this investigation.

At no time during the loading of the test configuration did the shoulders of the connection separate.

TABLE 1 CALCULATED AXIAL STRESSES

TORQUE FT-LB	AXIAL LOAD LB	BENDING LOAD IN-LB	CONNECTION PIN STRESS PSI	BOX SHOULDER STRESS PSI
47124*	0	0	67723	- 96708
53247**	0	0	76726	-109563
0	400000	-	15942	-
0	400000	16°/100 ft	27047	-
53247	400000	16°/100 ft	105626	-

*Optimum Torque Used In This Report
**Corrected Optimum Torque

TABLE 2 AVERAGE AXIAL STRESS AT STRESS RELIEF GROOVE OF CONNECTION DUE TO MAKE-UP

TORQUE FT-LBS	AXIAL STRESS - PSI, FOR POINT LOCATION ON SPECIMEN 3(1)	14(1)	3(2)	14(2)	3(3)	14(3)	3(4)	14(4)
28275(1)	71229	80578	77037	78682	70645	74436	70343	71881
37699(2)	-	-	91770	89552	94736	101314	95380	98578
47124(3)	-	-	-	-	101580	107047	119240	122854
56549(4)	-	-	-	-	-	-	136850	139991

(1) 60% Optimum Torque Case
(2) 80% Optimum Torque Case
(3) 100% Optimum Torque Case
(4) 120% Optimum Torque Case

TABLE 3 AVERAGE AXIAL STRESS AT STRESS RELIEF GROOVE OF CONNECTION DUE TO AXIAL LOAD

AXIAL LOAD LBS	BENDING LOAD IN-LBS	AVERAGE AXIAL STRESS - PSI, FOR POINT LOCATION ON SPECIMEN 3(1)	14(1)	3(2)	14(2)	3(3)	14(3)	3(4)	14(4)
151876	0	6053	6008	6153	6157	6160	6420	6748	7396
201013	0	8278	8169	8050	8092	8171	8377	8936	9672
299286	0	12918	12776	12638	12809	12946	13568	13998	15104
402025	0	17339	17128	16690	17077	17059	18082	19304	20613

(1) 60% Optimum Torque Case
(2) 80% Optimum Torque Case
(3) 100% Optimum Torque Case
(4) 120% Optimum Torque Case

TABLE 4 AXIAL STRESS AT STRESS RELIEF GROOVE OF CONNECTION DUE TO AXIAL, BENDING AND COMBINED AXIAL AND BENDING LOAD - ODD GAGES

AXIAL LOAD LBS	BENDING LOAD IN-LBS	AXIAL STRESS - PSI, FOR POINT LOCATION ON SPECIMEN							
		3(1)	14(1)	3(2)	14(2)	3(3)	14(3)	3(4)	14(4)
151876	0	6214	6412	6396	6511	6254	6803	6676	6940
201013	0	8466	8578	8298	8479	8275	8766	8753	8894
299286	0	13118	13319	12979	13361	13127	14093	13569	13500
402025	0	17555	17726	17113	17759	17407	18755	18643	18171
0	43304	-1266	-1052	-1332	-1226	-1180	-1193	-1058	-1098
0	146552	-2690	-2502	-2674	-2809	-2561	-2786	-2261	-2446
0	198177	-4005	-3798	-4019	-4256	-3666	-4012	-3333	-3841
0	249801	-5156	-4985	-5070	-5492	-4882	-5482	-4404	-5136
0	301425	-6303	-6102	-6250	-6761	-6063	-6798	-5433	-6461
402025	43304	16457	16694	16757	17123	17321	17548	17416	17202
402025	146552	16038	16352	16236	16582	16823	17027	17163	16991
402025	198177	15663	15976	15850	16259	16503	16629	16810	16526
402025	249801	15310	15633	15540	15864	16061	16154	16444	16216
402025	301425	14769	15168	15175	15531	15643	15778	16134	15860
402025	507922	13312	13642	-	-	-	-	-	-
402025	559546	-	-	13098	13187	13556	13411	14387	13474
402025	827992	9831	9837	-	-	-	-	-	-
402025	920916	-	-	-	-	10375	9738	-	-
402025	941566	-	-	-	-	-	-	11515	24420
402025	951891	-	-	9827	9646	-	-	-	-

(1) 60% Optimum Torque Case
(2) 80% Optimum Torque Case
(3) 100% Optimum Torque Case
(4) 120% Optimum Torque Case

TABLE 5 AXIAL STRESS AT STRESS RELIEF GROOVE OF CONNECTION DUE TO AXIAL, BENDING AND COMBINED AXIAL AND BENDING LOAD - EVEN GAGES

AXIAL LOAD LBS	BENDING LOAD IN-LBS	AXIAL STRESS - PSI, FOR POINT LOCATION ON SPECIMEN							
		3(1)	14(1)	3(2)	14(2)	3(3)	14(3)	3(4)	14(4)
151876	0	5891	5604	5911	5802	6066	6033	6821	7853
201013	0	8090	7760	7803	7704	8067	7988	9119	10450
299786	0	12719	12234	12297	12257	12765	13042	14426	16708
402025	0	17123	16530	16266	16394	16711	17410	19965	23054
0	43304	976	1160	966	1098	976	1164	916	1246
0	146552	2357	2825	2509	2743	2324	2723	1938	2650
0	198177	3781	4272	3758	4114	3485	3972	3043	4022
0	249801	5018	5588	4919	5255	4645	5288	4190	5459
0	301425	6069	6781	5937	6465	5759	6570	5153	6705
402025	43304	16777	16141	16879	16619	16971	16942	18389	18135
402025	146552	17054	16450	17265	17093	17390	17350	18587	18369
402025	198177	17407	16780	17618	17426	17776	17726	18930	18821
402025	249801	17726	17080	17960	17789	18010	18036	19282	19243
402025	301425	18158	17499	18290	18155	18287	18402	19635	19694
402025	507922	19540	19015	-	-	-	-	-	-
402025	559546	-	-	20423	20631	20344	21231	21900	22783
402025	827992	23238	23018	-	-	-	-	-	-
402025	920916	-	-	-	-	23644	26123	-	-
402025	941566	-	-	-	-	-	-	26245	29004
402025	951891	-	-	23927	24804	-	-	-	-

(1) 60% Optimum Torque Case
(2) 80% Optimum Torque Case
(3) 100% Optimum Torque Case
(4) 120% Optimum Torque Case

TABLE 6 AXIAL STRESS AT STRESS RELIEF GROOVE OF CONNECTION
DUE TO COMBINED LOADING

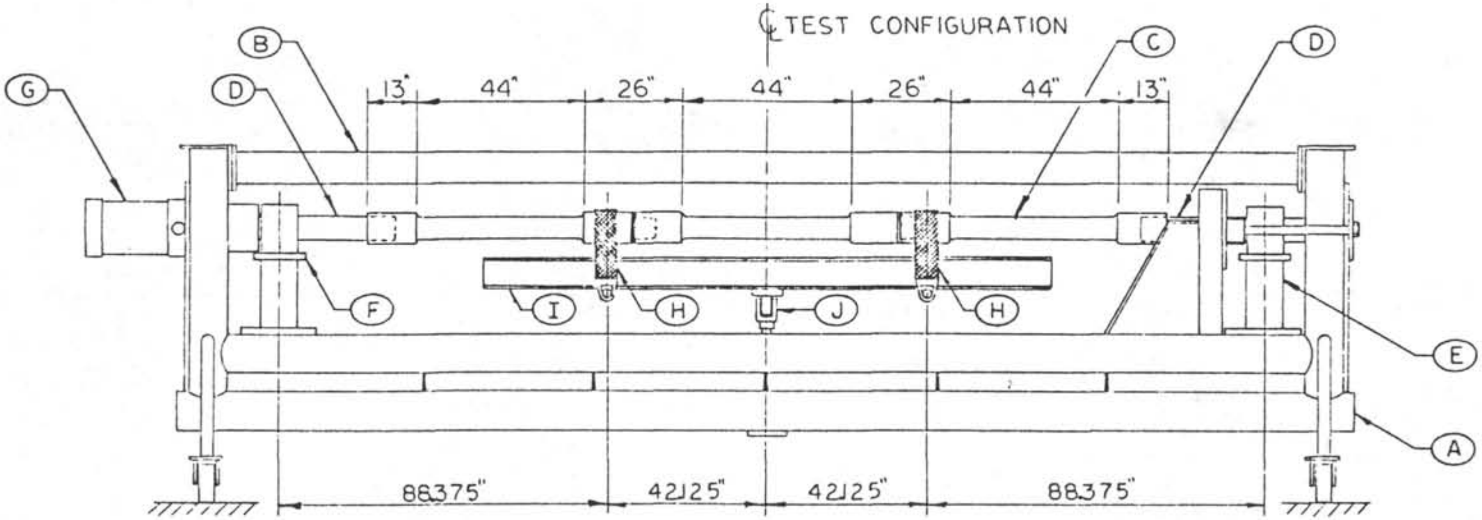
OPTIMUM TORQUE CASE %	TORQUE FT-LB	AXIAL LOAD LB	BENDING LOAD IN-LB	AXIAL STRESS PSI 3(1)	AXIAL STRESS PSI 14(1)
60	28274	402025	0	85861	79083
60	28274	402025	827992	91760	84972
80	37699	402025	0	112070	115655
80	37699	402025	951891	119307	123382
100	47124	402025	0	136299	140937
100	47124	402025	920916	142884	148978
120	56549	402025	0	156158	160604
120	56549	402025	941566	163099	168995

NOTE:

1. Numbers represent point numbers of gage locations on test configuration.

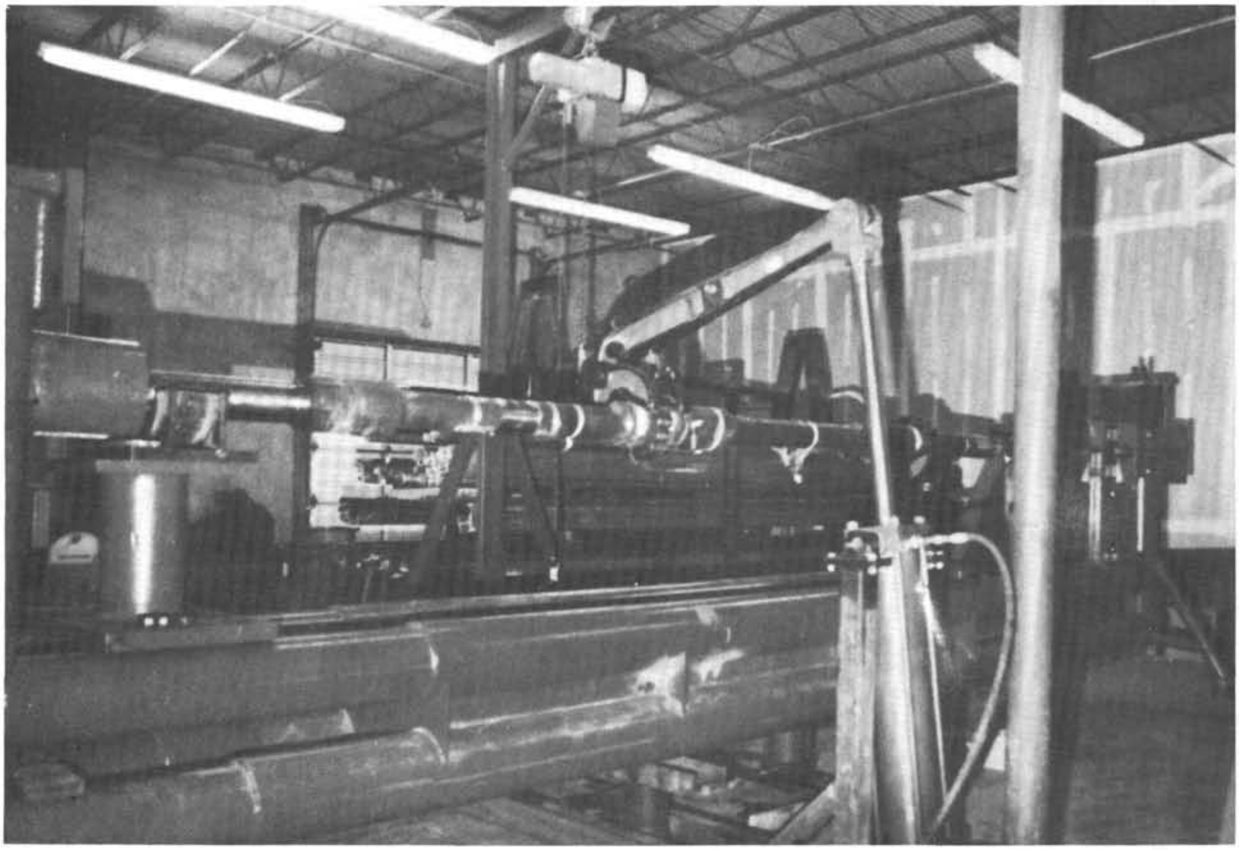
2. Make-up stresses are from Table A13c, Appendix A.

3. Axial stress values for axial and bending load are from Tables A4c, A8c, A12c and A16c in Appendix A, respectively for the 60%, 80%, 100% and 120% optimum torque case.

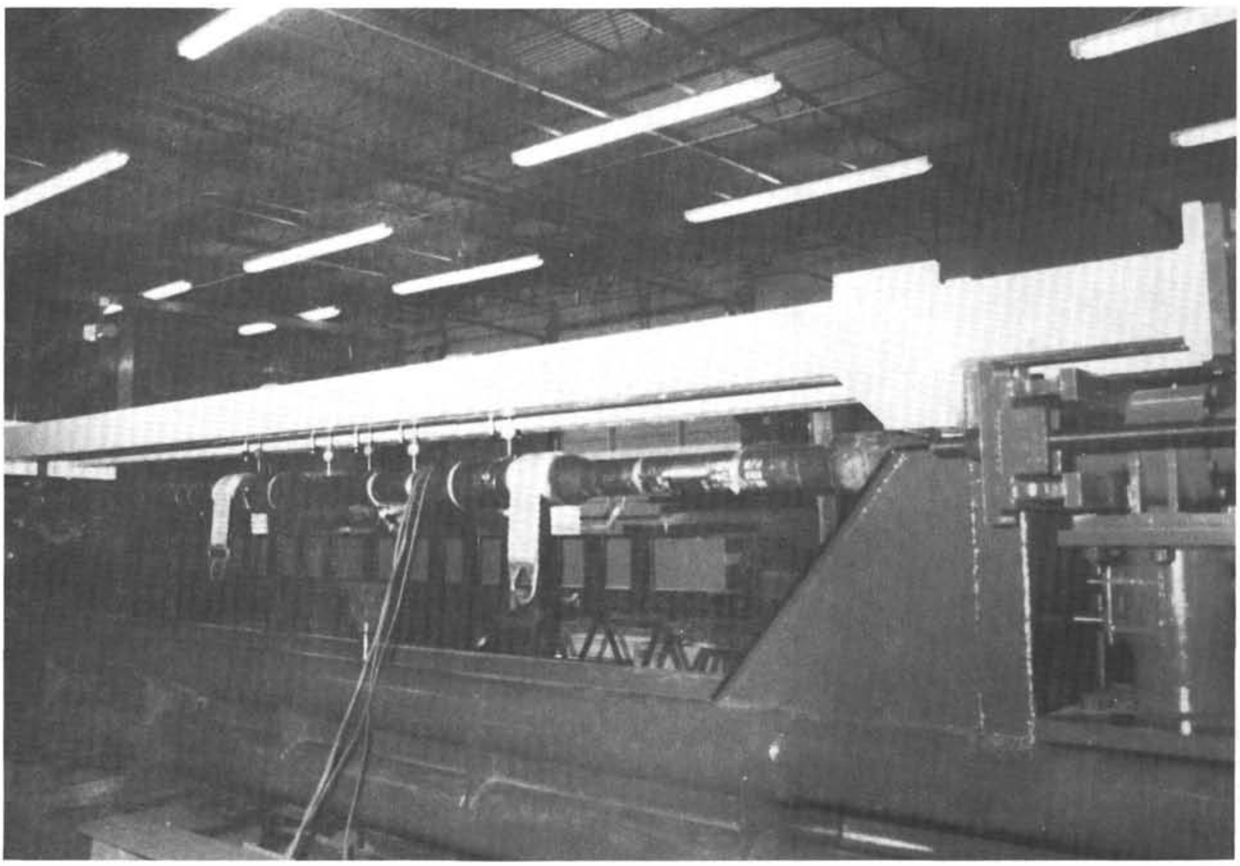


ITEM	DESCRIPTION
A	HORIZONTAL FRAME AND END MEMBERS
B	UPPER FRAME
C	TEST CONFIGURATION
D	PULLING ADAPTERS
E	FIXED BEARING SUPPORT
F	HORIZONTAL SUPPORT
G	HYDRAULIC CYLINDER (AXIAL LOAD)
H	CHAIN-LINK STRAPS
I	LOADING BEAM
J	HYDRAULIC CYLINDER (BENDING LOAD)

FIG. I TEST FRAME & SPECIMEN CONFIGURATION

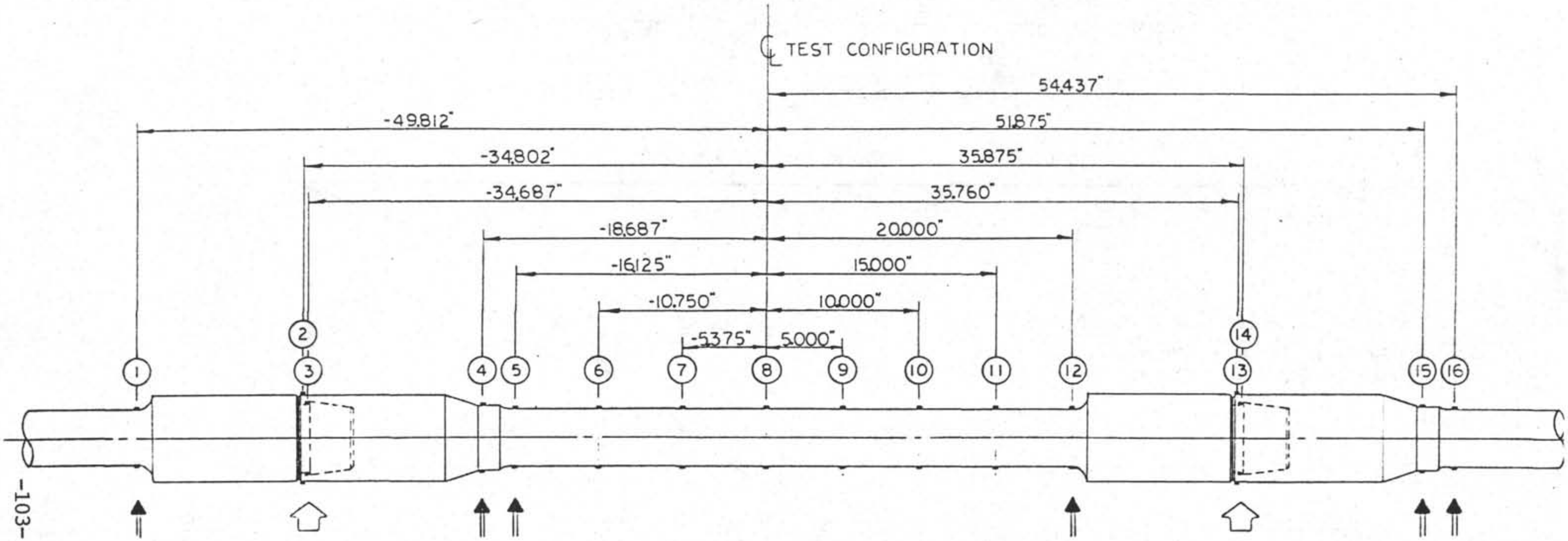


a) Make-up Mode



b) Bending or Combined Axial and Bending Mode

FIG. 2 LOAD MODES OF TEST CONFIGURATION



NOTES: 1) ALL STRAIN LOCATIONS CONSIST OF SINGLE ELEMENT GAGES ORIENTED IN THE AXIAL DIRECTION.

2) ENCIRCLED NUMBERS REPRESENT POINT LOCATION OF GAGES.

⇧ - 90° STRAIN ROSETTE

▲ - AXIAL STRAIN & FATIGUE GAGE

FIG. 3 STRAIN GAGE LOCATIONS

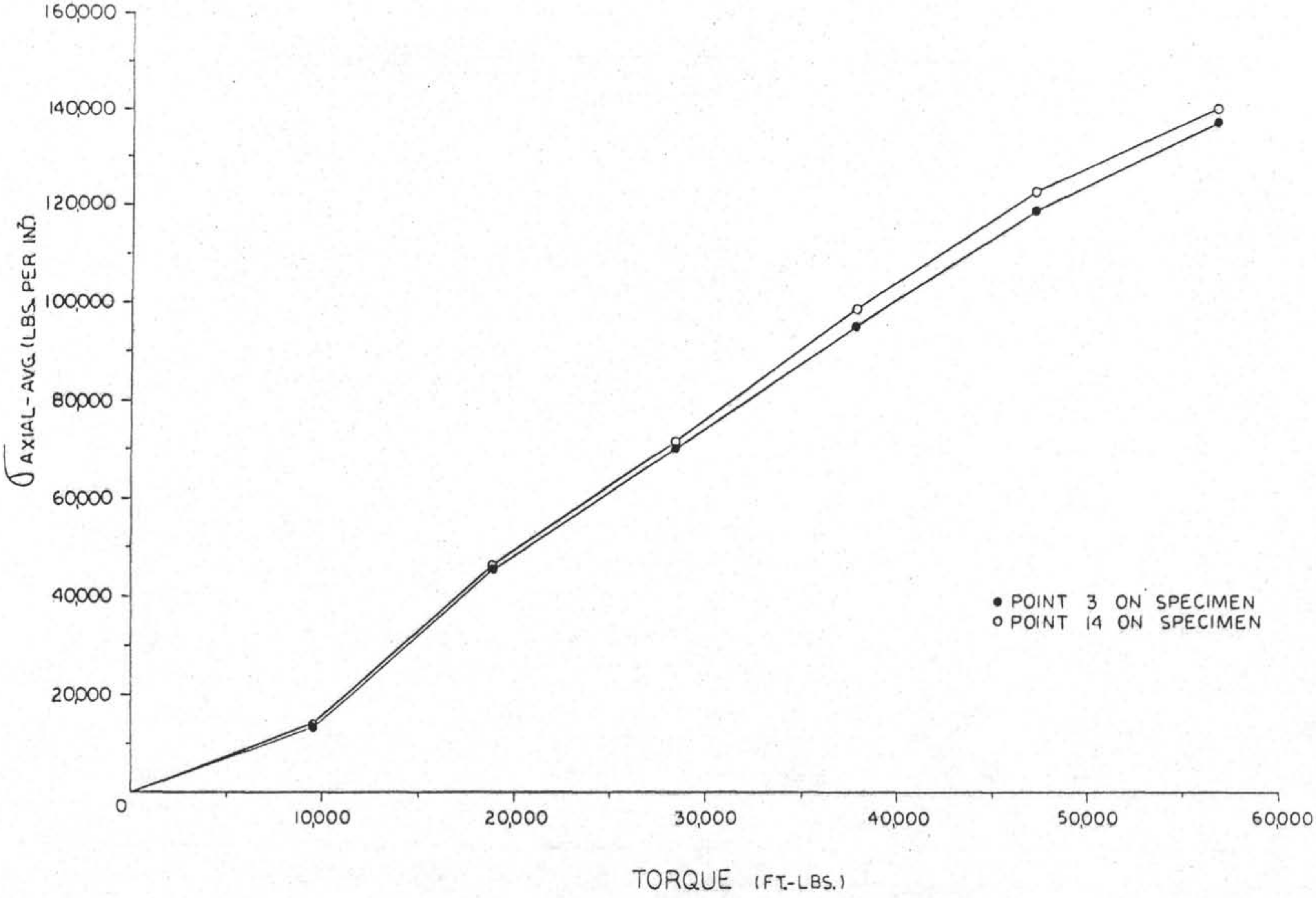


FIG. 4 AVERAGE AXIAL STRESS AT STRESS RELIEF GROOVE OF CONNECTION DUE TO MAKE-UP (120% OPTIMUM TORQUE CASE)

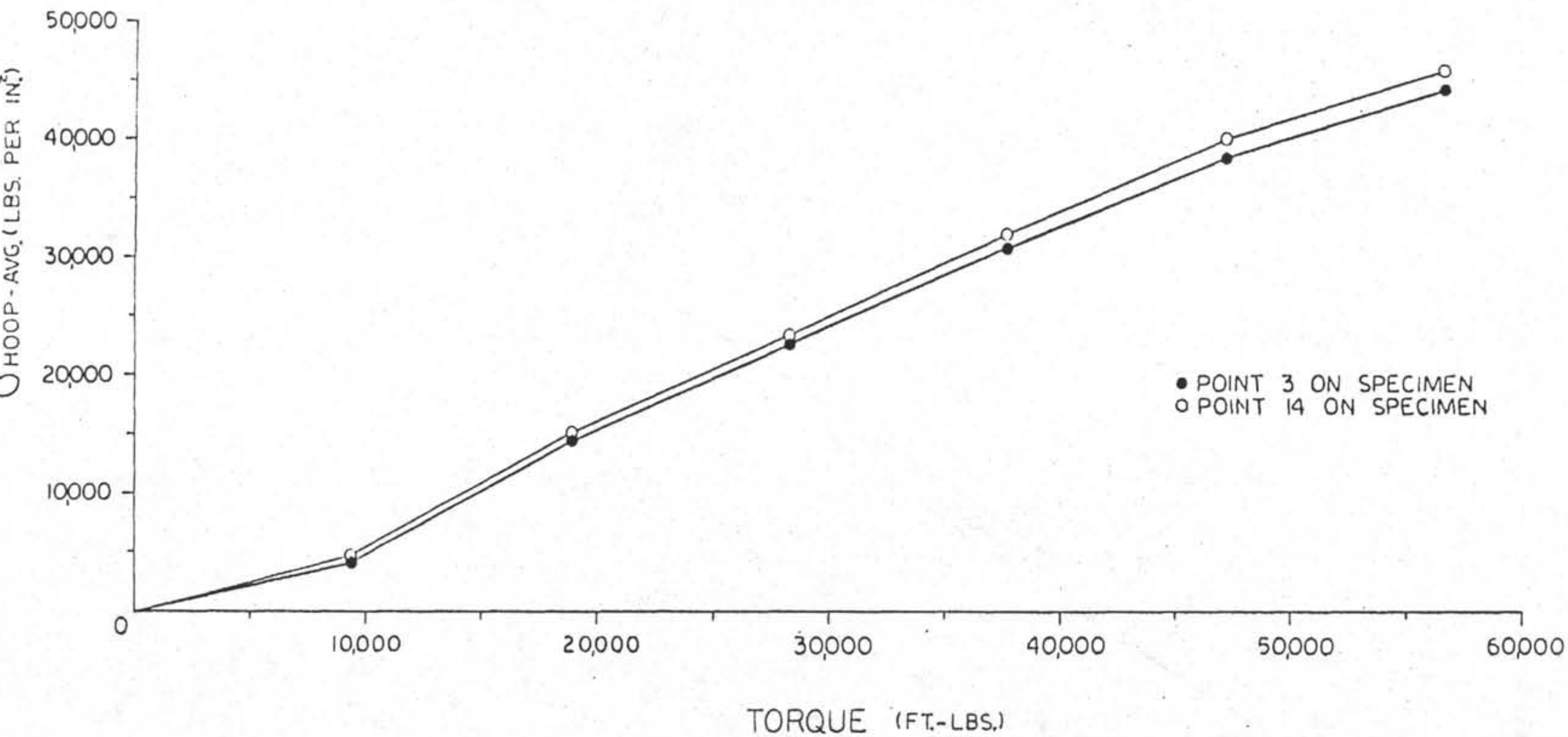


FIG. 5 AVERAGE HOOP STRESS AT STRESS RELIEF GROOVE OF CONNECTION DUE TO MAKE-UP (120% OPTIMUM TORQUE CASE)

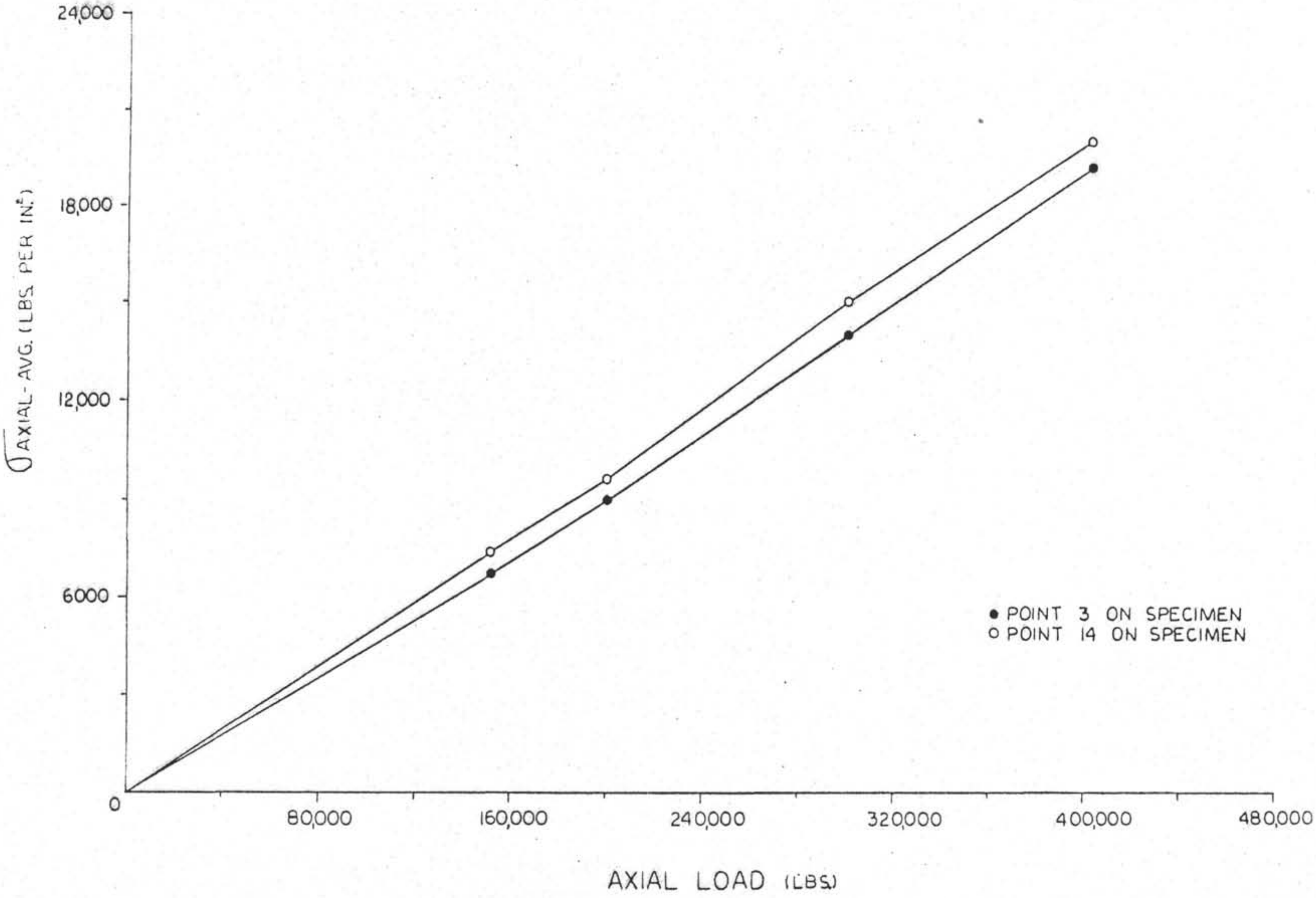


FIG. 6 AVERAGE AXIAL STRESS AT STRESS RELIEF GROOVE OF CONNECTION DUE TO AXIAL LOAD (120% OPTIMUM TORQUE CASE)

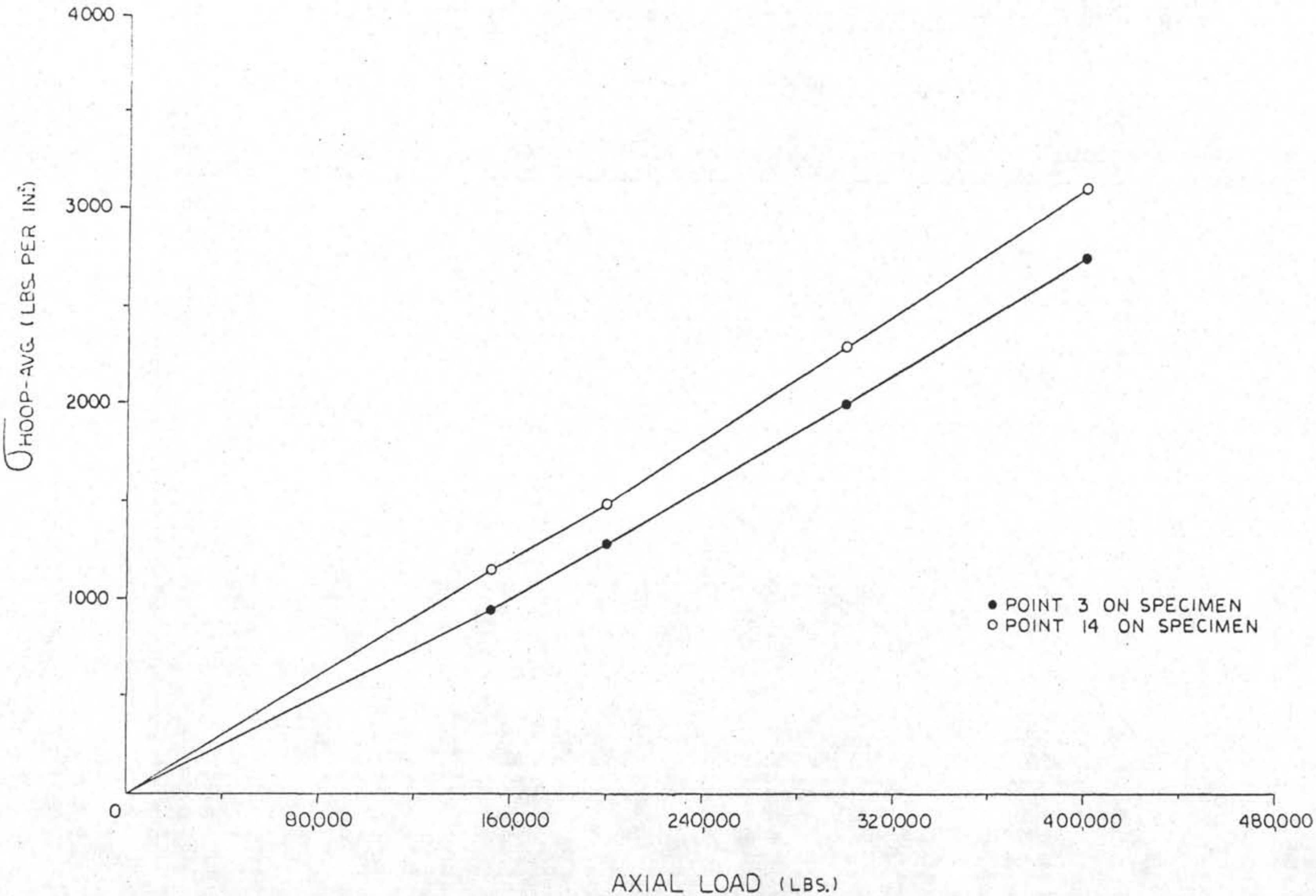


FIG. 7 AVERAGE HOOP STRESS AT STRESS RELIEF GROOVE OF CONNECTION DUE TO AXIAL LOAD (120% OPTIMUM TORQUE CASE)

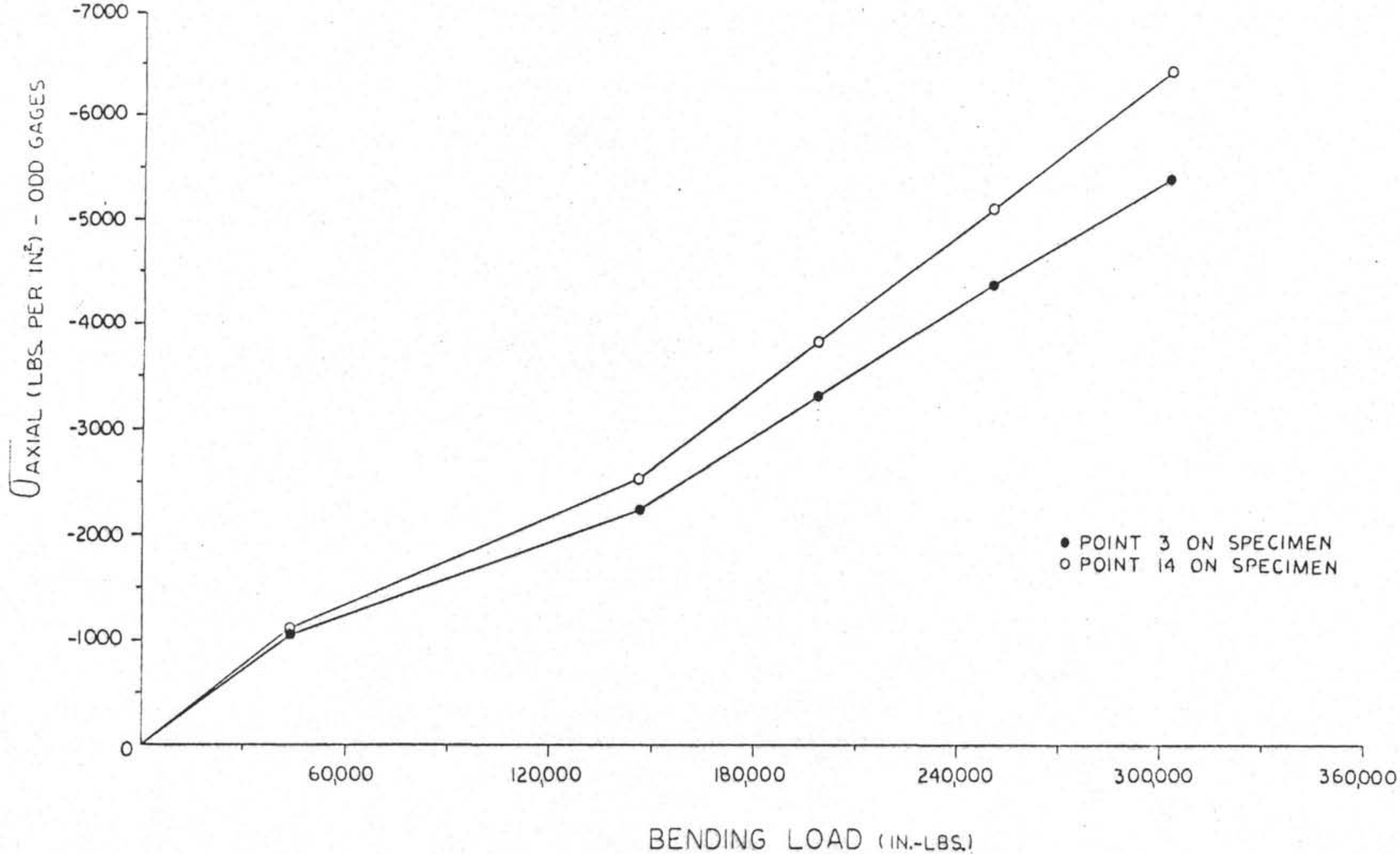


FIG. 8 AXIAL STRESS AT STRESS RELIEF GROOVE OF CONNECTION DUE TO BENDING LOAD - ODD GAGES - (120% OPTIMUM TORQUE CASE)

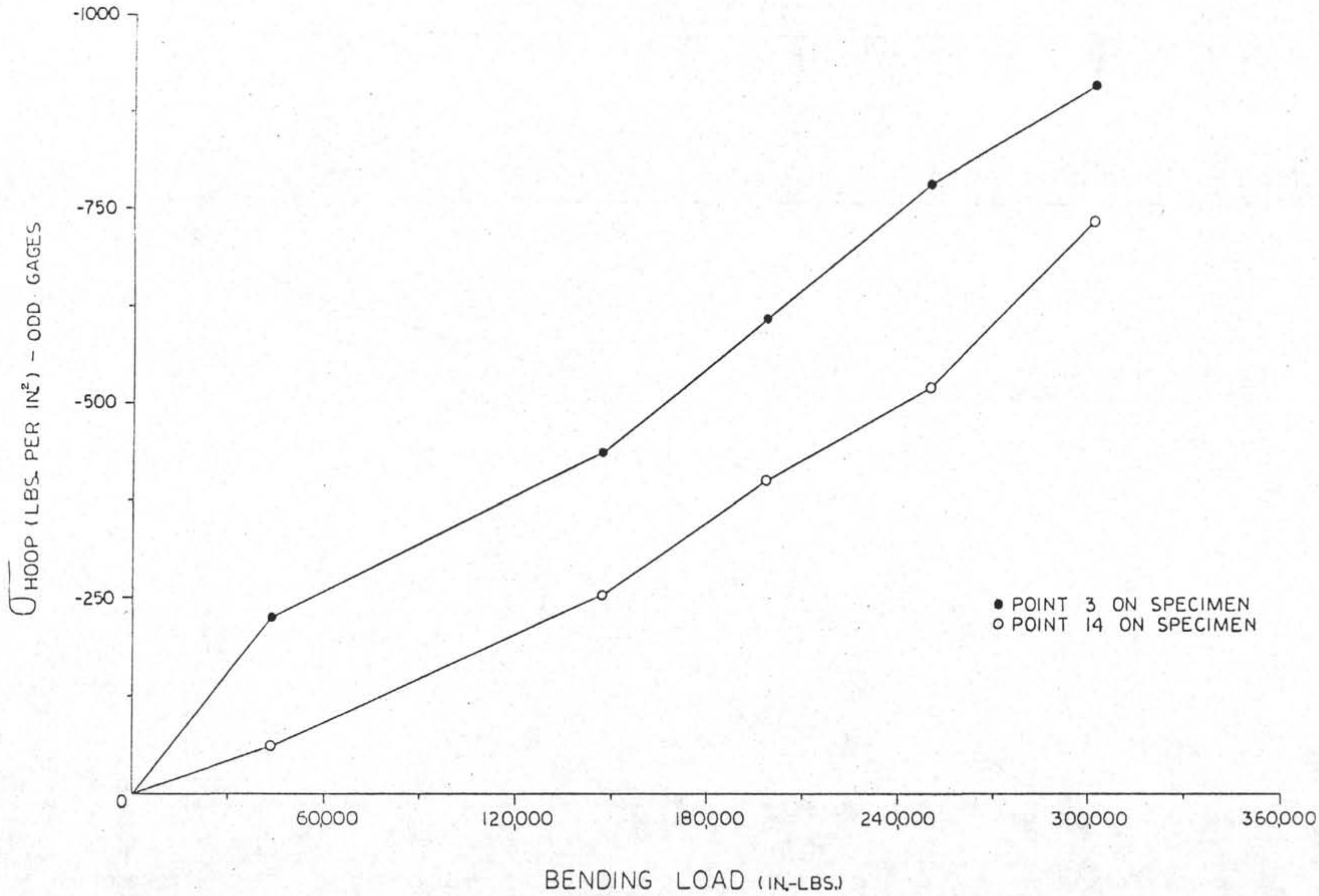


FIG. 9 HOOP STRESS AT STRESS RELIEF GROOVE OF CONNECTION DUE TO BENDING LOAD-ODD GAGES-(120% OPTIMUM TORQUE CASE)

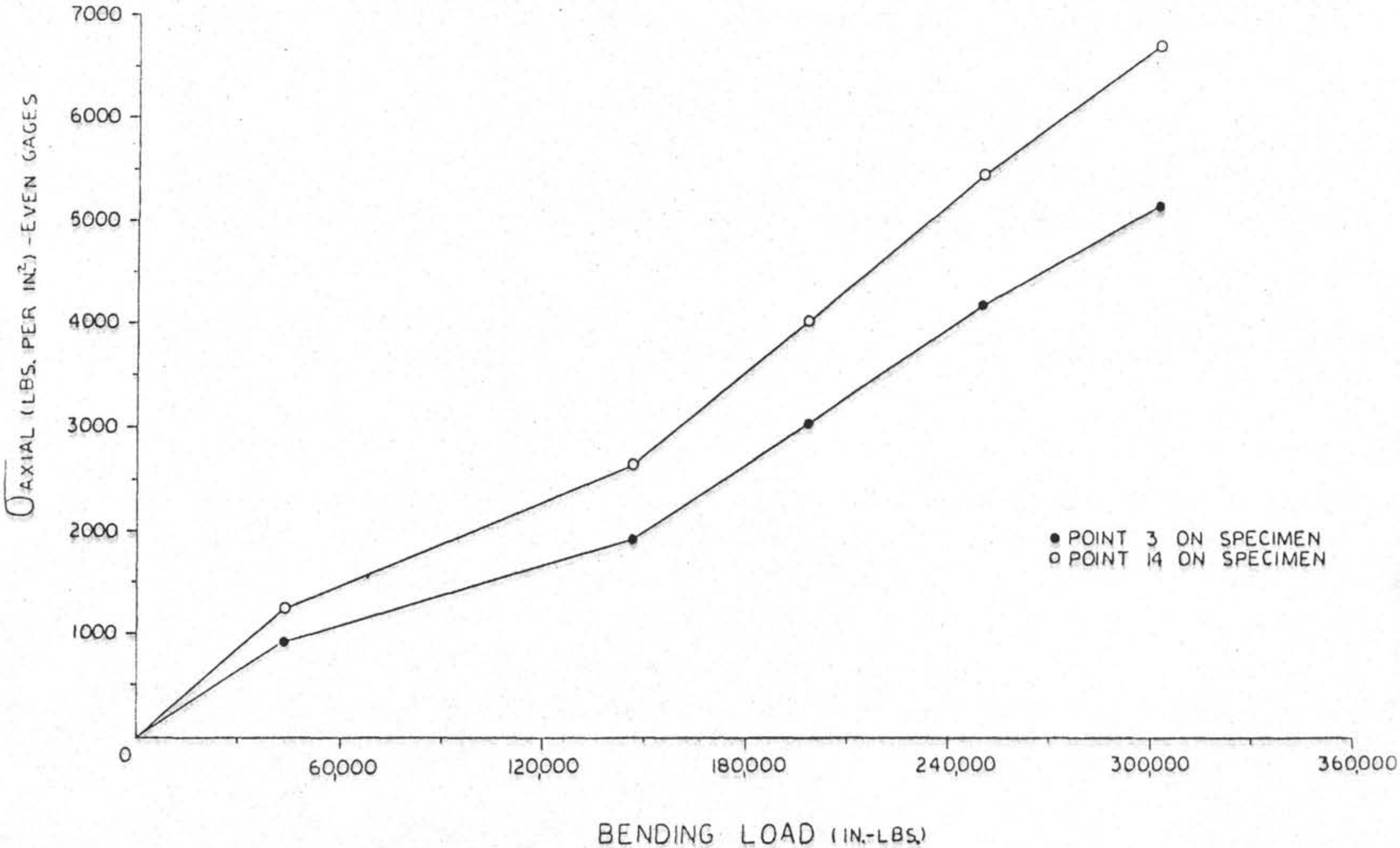


FIG. 10 AXIAL STRESS AT STRESS RELIEF GROOVE OF CONNECTION DUE TO BENDING LOAD - EVEN GAGES - (120% OPTIMUM TORQUE CASE)

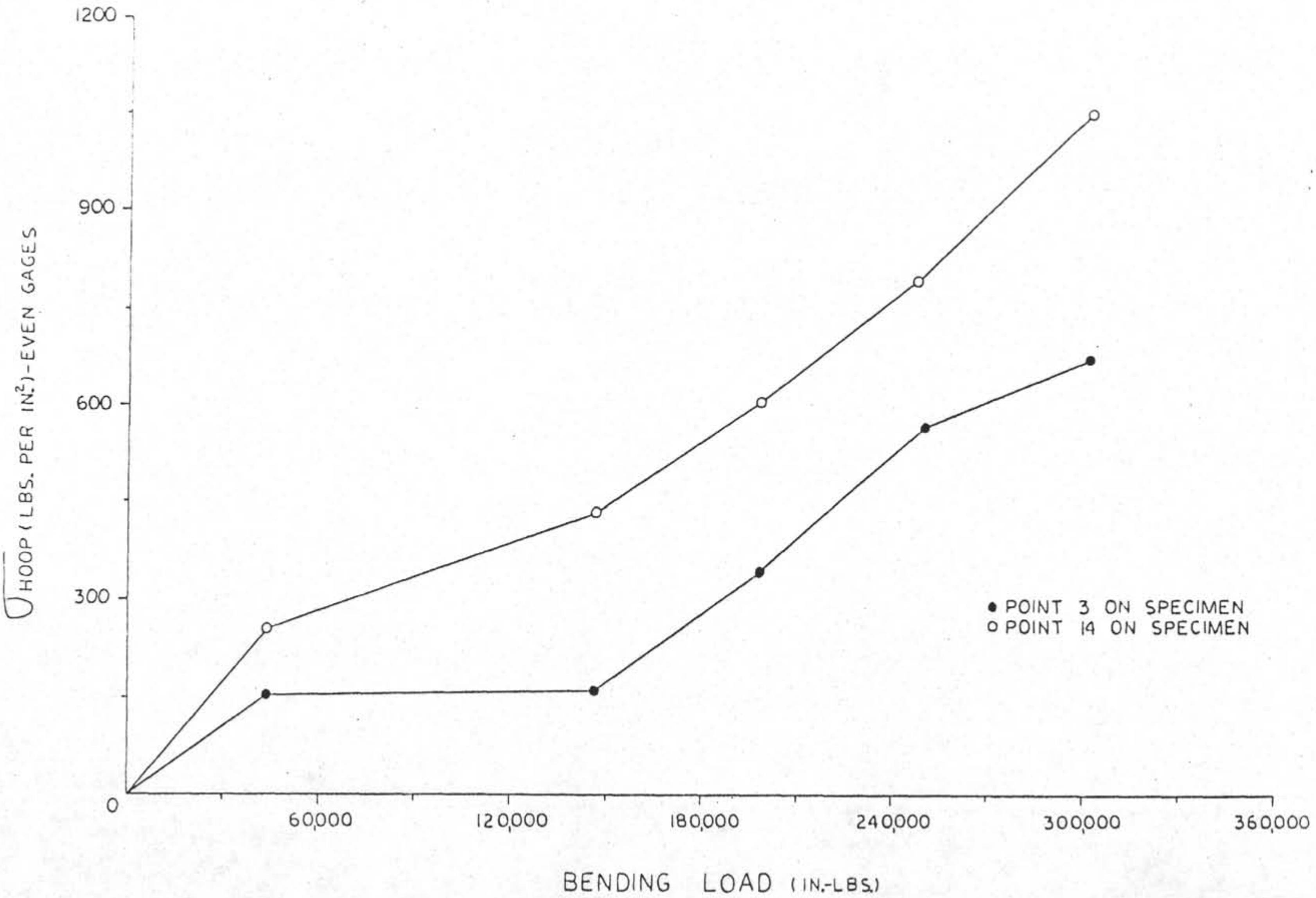


FIG. 11 HOOP STRESS AT STRESS RELIEF GROOVE OF CONNECTION DUE TO BENDING LOAD - EVEN GAGES - (120% OPTIMUM TORQUE CASE)

Appendix A has been omitted in the interest of brevity. It contains the data from all the strain gages used during the experimental evaluation, and can be found in Reference 5.

APPENDIX B

Connection Rotation For 60%, 80%, 100% And 120% Optimum Torque Cases

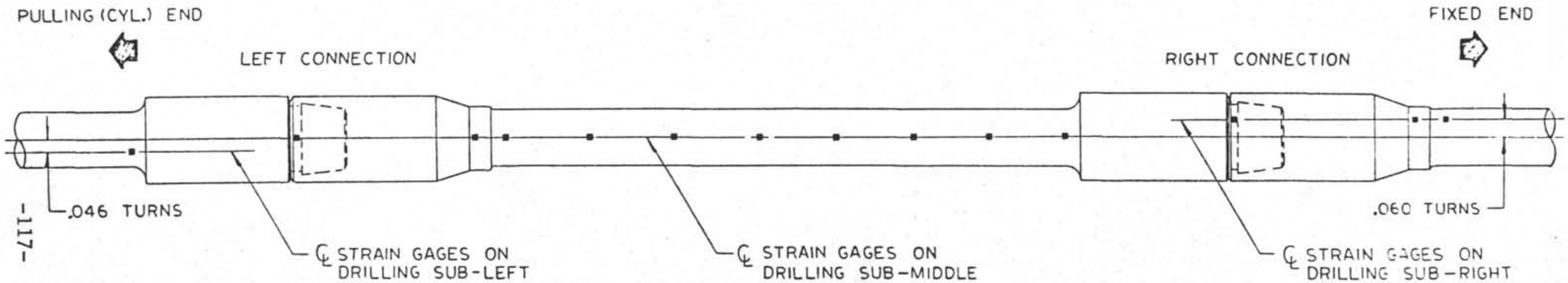


FIG. B1 STRAIN GAGE ALIGNMENT AT 60% OPTIMUM TORQUE CASE

TABLE B1 CONNECTION ROTATION DURING MAKE-UP

TORQUE FT-LBS	TURNS, FOR 5-1/2" I.F. CONNECTION							
	LEFT(1)	RIGHT(1)	LEFT(2)	RIGHT(2)	LEFT(3)	RIGHT(3)	LEFT(4)	RIGHT(4)
HAND TIGHT	-	-	-	-	0.000	0.000	0.000	0.000
9425	0.000	0.000	0.000	0.000	0.060	0.070	0.008	0.015
18850	0.020	0.025	0.000	0.001	0.085	0.090	0.028	0.035
28274(1)	0.040	0.045	0.000	0.002	0.103	0.105	0.048	0.050
37699(2)	-	-	0.010	0.012	0.113	0.122	0.063	0.065
47124(3)	-	-	-	-	0.130	0.140	0.078	0.082
56544(4)	-	-	-	-	-	-	0.096	0.100
CALCULATED ADVANCE FROM 28274 FT-LB REF.	0.000	0.000	-	-	0.030	0.025	0.048	0.040
MEASURED ADVANCE FROM 28274 FT-LB REF.	-	-	0.010	0.010	0.030	0.030	0.048	0.050

(1) 60% Optimum Torque Case
(2) 80% Optimum Torque Case
(3) 100% Optimum Torque Case
(4) 120% Optimum Torque Case

APPENDIX C

Bending Deflections For 60%, 80%, 100% And 120% Optimum Torque Cases

LIST OF TABLES

Table	
C1	Bending Deflection For 60% Optimum Torque Case
C2	Bending Deflection For 80% Optimum Torque Case
C3	Bending Deflection For 100% Optimum Torque Case
C4	Bending Deflection For 120% Optimum Torque Case

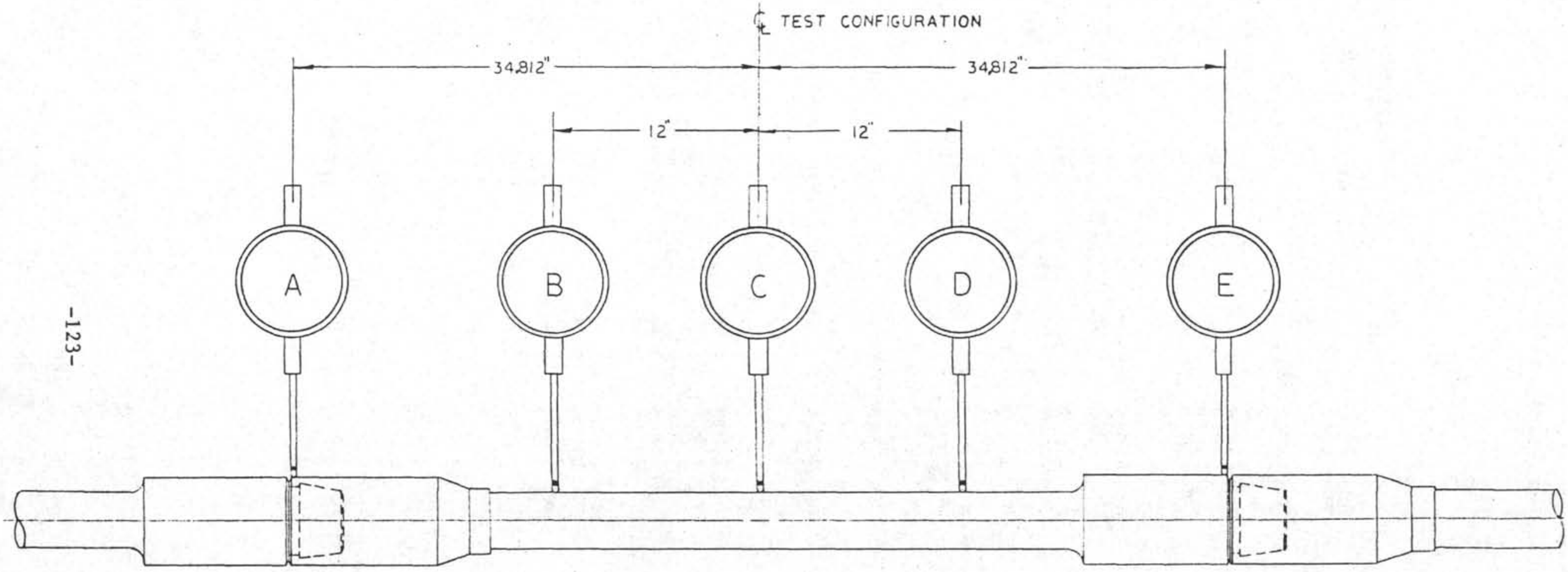


FIG. C1 DIAL INDICATOR LOCATIONS

TABLE C1 BENDING DEFLECTIONS FOR 60% OPTIMUM TORQUE CASE

AXIAL LOAD LB	BENDING LOAD IN-LB	DIAL INDICATOR READINGS (INCHES) A	B	C	D	E
0	43304	0.205	0.223	0.224	0.223	0.302
0	146552	0.471	0.520	0.525	0.520	0.472
0	198177	0.510	0.782	0.790	0.783	0.707
0	249801	0.927	1.020	1.032	1.021	0.926
0	301425	1.142	1.256	1.272	1.258	1.137
402025	43304	0.053	0.055	0.056	0.054	0.047
402025	146552	0.092	0.100	0.099	0.097	0.081
402025	198177	0.140	0.150	0.148	0.144	0.121
402025	249801	0.180	0.195	0.194	0.188	0.159
402025	301425	0.254	0.271	0.268	0.260	0.220
402025	507922	0.445	0.485	0.486	0.478	0.419
402025	827992	1.020	1.123	1.132	1.120	1.072

TABLE C2 BENDING DEFLECTION FOR 80% OPTIMUM TORQUE CASE

AXIAL LOAD LB	BENDING LOAD IN-LB	DIAL INDICATOR READINGS (INCHES) A	B	C	D	E
0	43304	0.205	0.129	0.231	0.229	0.105
0	146552	0.488	0.439	0.544	0.539	0.385
0	198177	0.744	0.719	0.827	0.818	0.635
0	249801	0.957	0.953	1.064	1.050	0.846
0	301425	1.170	1.196	1.300	1.283	1.055
402025	43304	0.049	0.052	0.053	0.050	0.043
402025	146552	0.108	0.117	0.124	0.113	0.093
402025	198177	0.155	0.168	0.165	0.161	0.135
402025	249801	0.201	0.217	0.215	0.209	0.176
402025	301425	0.251	0.271	0.268	0.261	0.220
402025	559546	0.600	0.650	0.649	0.638	0.555
402025	951891	1.182	1.289	1.294	1.275	1.133

TABLE C3 BENDING DEFLECTION FOR 100% OPTIMUM TORQUE CASE

AXIAL LOAD LB	BENDING LOAD IN-LB	DIAL INDICATOR READINGS (INCHES) A	B	C	D	E
0	43304	0.210	0.230	0.232	0.229	0.208
0	146552	0.483	0.532	0.536	0.531	0.480
0	198177	0.696	0.767	0.775	0.766	0.694
0	249801	0.926	1.021	1.031	1.019	0.925
0	301425	1.144	1.261	1.274	1.259	1.142
402025	43304	0.043	0.046	0.046	0.045	0.036
402025	146552	0.105	0.114	0.113	0.110	0.090
402025	198177	0.157	0.170	0.168	0.164	0.137
402025	249801	0.208	0.225	0.223	0.216	0.182
402025	301425	0.261	0.283	0.279	0.272	0.230
402025	559546	0.620	0.674	0.675	0.660	0.580
402025	920916	1.184	1.299	1.307	1.289	1.153

TABLE C4 BENDING DEFLECTION FOR 120% OPTIMUM TORQUE CASE

AXIAL LOAD LB	BENDING LOAD IN-LB	DIAL INDICATORS READINGS (INCHES) A	B	C	D	E
0	43304	0.206	0.223	0.235	0.221	0.201
0	146552	0.452	0.492	0.497	0.491	0.444
0	198177	0.697	0.761	0.769	0.760	0.687
0	249801	0.937	1.023	1.034	1.022	0.924
0	301425	1.146	1.251	1.264	1.246	1.128
402025	43304	0.070	0.076	0.074	0.072	0.060
402025	146552	0.100	0.109	0.107	0.104	0.087
402025	198177	0.151	0.164	0.162	0.158	0.132
402025	249801	0.202	0.218	0.216	0.210	0.178
402025	301425	0.252	0.273	0.270	0.263	0.223
402025	559546	0.603	0.654	0.655	0.643	0.564
402025	941566	1.200	1.306	1.312	1.292	1.151

APPENDIX D

Sample Calculations For 5-1/2" I.F. Connection For 7-3/4" OD X 4-1/8" ID Drilling Sub By Dr. P. D. Weiner

5-1/2 inch I.F. connection with a 7-3/4 inch outside diameter and 4-1/8 inch inside diameter.

Box Moment of Inertia

$$I_B = \pi/64\ (7.75^4 - 5.8^4)$$
$$= 121.53\ in^4$$

Pin Moment of Inertia

$$I_P = \pi/64\ (5.89^4 - 4.125^4)$$
$$= 44.87\ in^4$$

Ratio of Moment of Inertia

$$I_B/I_P = \frac{121.53}{44.87}$$
$$= 2.71$$

Box Section Modulus

$$Z_B = \frac{\pi/64\ (7.75^4 - 5.8^4)}{7.75}$$
$$= 15.68\ in^3$$

Pin Section Modulus

$$Z_P = \frac{\pi/64\ \ 5.89^4 - 4.125^4}{5.89}$$
$$= 7.62\ in^3$$

Bending Strength Ratio

$$BSR = Z_B/Z_P$$
$$= \frac{15.68}{7.62}$$
$$= 2.06$$

Box Area

$$A_B = \frac{\pi}{4}(7.75^2 - 5.8^2)$$
$$= 20.75\ in^2$$

Pin Area

$$A_P = \frac{\pi}{4}(5.89^2 - 4.125^2)$$
$$= 13.88\ in^2$$

Area Ratio

$$A_R = \frac{A_B}{A_P}$$
$$= \frac{20.75}{13.88}$$
$$= 1.5$$

Box Area at Recess

$$A_{Recess} = \frac{\pi}{4}(7.75^2 - 6.453^2)$$
$$= 14.46 \text{ in}^2$$

Box Contact Area

$$A_{Contact} = \frac{\pi}{4}(7.35^2 - 6.453^2)$$
$$= 9.72 \text{ in}^2$$

Moment of Inertia Main Beam

$$I = \frac{\pi}{64}(7.75^4 - 6.453^4)$$
$$= 91.97 \text{ in}^4$$

Moment of Inertia Box in Compression

$$I_1 = \frac{\pi}{64}\left[7.35^4 - 6.453^4\right]$$
$$= 54.29 \text{ in}^4$$

Make-Up Torque

$$T = \frac{C_1 EA}{12}\left[Q_c + 2\delta\right]\left[\frac{I}{I_1}\right]\left[\frac{P}{2} + \frac{R_t f}{\cos t} + R_s f\right]$$

Where P = .25 inches/thread

$$R_t = \frac{C + D_s - H}{4}$$ Table 9.1 API Std. 7

$$= \frac{6.189+5.564-0.216}{4}$$

$$= 2.992 \text{ in}$$

$$R_S = \frac{D+Q_c}{4}$$
$$= \frac{7.75+6.453}{4}$$
$$= 3.55 \text{ in.}$$

$$Q_c + 2\delta = 6.453 + (2)(0.2)$$
$$= 6.853 \text{ in}$$

$C_1 = 2.203 \times 10^{-4}$ for 16° deflection per 100 feet

$$\left[\frac{P}{2\pi} + \frac{R_t F}{\text{Cost}} + R_s F\right] = \left[\frac{.25}{2} + \frac{(2.99)(.08)}{30°} + (3.55)(.08)\right]$$
$$= .04 + .276 + .284$$
$$= 0.6$$

$$\text{Torque} = \frac{(2.203 \times 10^{-4})(30\times10^{6})(13.88)(6.853)(91.97)(0.6)}{(12)(54.29)}$$

$$= 53247 \text{ ft - \#}$$

Force Due to Torque

$$F_i = \frac{(T)(12)}{P/2\pi + \frac{R_t F + R_s F}{\text{Cost}}}$$

$$= \frac{(53247)(12)}{0.6}$$

$$= 1,064,957\#$$

Shoulder Stress

$$S_{shoulder} = \frac{1,064,957}{9.72}$$

$$= 109563 \text{ psi}$$

Pin Stress

$$S_p = \frac{F_i}{A_{pl}}$$

$$\text{Spin} = \frac{1,064,957}{13.88}$$

$$= 76,726 \text{ psi}$$

Stress in Box

$$S_{Bi} = \frac{F_i}{A_B}$$

$$= \frac{1,064,957}{20.75}$$

$$= 51,323$$

Box Shoulder Stress

$$S_{Bsi} = \frac{1,064,957}{9.72}$$

$$= 109,563 \text{ psi}$$

Opening Force

$$F_o = F_i \left[\frac{A_p + A_B}{A_B} \right]$$

$$= 1,064,957 \left[\frac{13.88 + (14.46+9.72)(.5)}{(14.46+9.72)(.5)} \right]$$

$$F_o = 2,287,586 \text{ lbs}$$

Change in Pin Load

$$\Delta F_p = F_e \left[\frac{A_p}{A_B + Ap} \right]$$

$$= F_E \left[\frac{13.88}{12.09 + 13.88} \right]$$

$$= F_e \left[.55 \right]$$

For $F_e = 400,000$ lbs

Change in force on the pin

$$\Delta F_p = (.55)\ (400000)$$
$$= 221{,}283 \text{ lbs}$$

Then the total force on the pin is

$$F_{pin\ total} = F_i + F_p$$
$$=1{,}064{,}957 + 221{,}283$$
$$=1{,}286{,}240 \text{ lbs}$$

The resulting stress

$$S_{pin} = \frac{1{,}286{,}240}{13.88}$$
$$= 92{,}668 \text{ psi}$$

or change in Pin Stress

$$\Delta S_{pin_A} = \frac{221{,}203}{13.88}$$
$$= 15{,}942 \text{ psi}$$

Bending effect

$$S_{Bending} = 218.125D \quad \text{per o/100 feet}$$
$$\text{or} = 3490D \text{ for } 16^{o}/100 \text{ feet}$$

The $S_{Bending} = (3490)(7.75)$

$$= 27047 \text{ psi}$$

If we consider this as an external force

in $S = \frac{F}{AB}$

$$F_e = (27047)(12.09)$$
$$= 326{,}998 \text{ lbs}$$

or on the tensile side of the beam

the $\Delta F_{p_b} = (.55)(326{,}998)$

$$= 179{,}849 \text{ lbs}$$

or the additional stress in the pin is

$$S_{pin\ Bending} = \frac{179849}{13.88}$$
$$= 12957 \text{ psi}$$

& the total pin stress

$$S_{pin\ total} = S_i + S_A + S_{Bending} = 76{,}727 + 15{,}942 + 12{,}957$$
$$= 105{,}626 \text{ psi}$$

The box compressive stress would be

$$S_B = \frac{1{,}064{,}957 - 180{,}000 - 147{,}149}{12.09}$$
$$= 61{,}026 \text{ psi}$$

5-1/2 inch full hole with 7-3/4 inch outside diameter and 4-1/8 inch inside diameter.

Box Member Moment of Inertia

$$I_B = \pi/64 \left[7.75^4 - 5.172^4\right]$$
$$= 141.93 \text{ in}^4$$

Pin Moment of Inertia

$$I_P = \pi/64 \left[5.25^4 - 4.125^4\right]$$
$$= 23.08 \text{ in}^4$$

Ratio of Moment of Inertia

$$I_B/I_P = \frac{141.93}{23.08}$$
$$= 6.15$$

Box Section Modulus

$$Z_B = \frac{\pi/64 \left[7.75^4 - 5.172^4\right]}{7.75}$$
$$= 18.31$$

Pin Section Modulus

$$Z_P = \frac{\pi/64 \left[5.25^4 - 4.125^4\right]}{5.25}$$
$$= 4.4$$

Bending Strength Ratio

$$BSR = ZB/ZP$$
$$= 18.31/4.4$$
$$= 4.16$$

Box Area

$$A_B = \pi/4 \left[7.75^2 - 5.172^2\right]$$
$$= 26.16 \text{ in}^2$$

Pin Area

$$A_P = \pi/4 \left[5.25^2 - 4.125^2\right]$$
$$= 8.28 \text{ in}^2$$

Area Ratio

$$A_R = A_B/A_P$$
$$= \frac{26.16}{8.28}$$
$$= 3.16$$

Box Area Under Recess

$$A_{Recess} = \pi/4 \left[7.75^2 - 5.906^2\right]$$
$$= 19.78 \text{ in}^2$$

Box Contact Area

$$A_{Contact} = \pi/4 \left[7.35^2 - 5.906^2\right]$$
$$= 15.03$$

Moment of Inertia Main Beam

$$I = \pi/64 \left[7.75^4 - 5.906^4\right]$$
$$= 117.34 \text{ in}^4$$

Moment of Inertia of the Box in Compression

$$I_1 = \pi/64 \left[7.35^4 - 5.906^4\right]$$
$$= 83.52$$

$$R_t = \frac{5.591 + 4.992}{4}$$
$$= 2.646 \text{ in}$$

$$R_S = \frac{7.75 + 5.906}{4}$$
$$= 3.414 \text{ in}$$

$$Q_C + 2S = 5.906 + (2)(.2)$$
$$= 6.306$$

$$C_1 = 2.203 \times 10^{-4}$$

$$\left[\frac{P}{2\pi} + \frac{R_{tf}}{\text{Cost}} + R_{sf}\right] = \left[\frac{.25}{} + \frac{(2.646)(.08)}{\text{Cost } 30^\circ} + (3.414)(.08)\right]$$
$$= .04 + .244 + .273$$
$$= .557$$

$$T = \frac{(2.203 \times 10^{-4})(30 \times 10^6)(8.28)(6.306)(117.34)(.557)}{(12)(83.52)}$$

$$= 22{,}503 \text{ ft} - \#$$

Force Due to Torque

$$F_i = \frac{(22503(12)}{.557}$$

$$= 484{,}804 \text{ lbs}$$

Shoulder Stress

$$S_{Shoulder} = \frac{484,804}{15.03}$$

$$= 32,255 \text{ psi}$$

Pin Stress

$$Spin = \frac{484,804}{8.28}$$
$$= 58,551 \text{ psi}$$

Stress in the Box

$$S_{B_i} = \frac{484,804}{26.16}$$
$$= 18,532 \text{ psi}$$

Box Shoulder Stress

$$S_{Bsi} = \frac{484,804}{15.03}$$
$$= 32,255 \text{ psi}$$

Opening Force

$$F_o = 484,804 \left[\frac{8.28 + (19.78 + 15.03)(.5)}{(19.78 + 15.03)(.5)} \right]$$
$$= 715,643 \text{ lbs}$$

Change in Pin Load

$$\Delta F_p = F_e \left[\frac{A_p}{A_B + A_p} \right]$$
$$= F_e \left[\frac{8.28}{17.4 + 8.28} \right]$$
$$= (F_e)(.322)$$

For $F_e = 400,000$ lbs

Change in Pin Force

$$\Delta F_b = (.322)(400,000)$$
$$= 128,972 \text{ lbs}$$

The total force on the pin

$$F_{pin\ i + A} = 484,804 + 128,972$$
$$= 613,776 \text{ lbs}$$

The Resulting Stress

$$S_{pin\ i + A} = \frac{613,776}{8.28}$$

$$= 74,127 \text{ psi}$$

or the change in pin stress

$$\Delta S_{pin\ A} = \frac{128,972}{8.28}$$
$$= 15,576 \text{ psi}$$

Bending Effect

$$S_{Bending} = 3490\ D$$
$$= (3490)(7.75)$$
$$= 27,047\ \text{psi}$$

If we consider this as an external force

$$Fe = (27,047)(17.4)$$
$$= 470,617\ \text{lbs}$$

Or the additional tensile force on the beam

$$F_{pb} = (.322)(470,617)$$
$$= 151,539\ \text{lbs}$$

Or the additional stress due to bending in the pin is

$$\Delta S_{pb} = \frac{151,539}{8.28}$$
$$= 18,302\ \text{psi}$$

And the total pin stress is

$$S_{total} = S_i + S_A + S_B$$
$$= 58,551 + 15,576 + 18,302$$
$$= 92,429\ \text{psi}$$

The resulting box force is

$$F_B = -484,804 + 271,028 + 319,078$$
$$= 105,302 \quad \therefore \text{Box Opens}$$

If we increase the initial pin stress to 750,000 psi

$$F_i = (8.28)(75,000)$$
$$= 621,000\ \text{lbs}$$

$$T = \frac{(621,000)(.557)}{12}$$
$$= 28,825\ \text{ft - \#}$$

Stress in the Box

$$S_{Bi} = \frac{621,000}{26.16}$$
$$= 23,738\ \text{psi}$$

Box Shoulder Stress

$$S_{Bsi} = \frac{621,000}{15.03}$$
$$= 41,317\ \text{psi}$$

Opening Force

$$F_o = 621{,}000 \left[1.476 \right] = 916{,}596 \text{ lbs}$$

The final force on the pin would be with 400,000 lbs axial load and 16 degrees per 100 foot bend is

$$F_{total} = 621{,}000 + 128{,}972 + 151{,}539 = 901{,}511 \text{ lbs}$$

$$S_{total} = \frac{901{,}511}{8.28} = 108{,}878 \text{ psi}$$

And the final box force is

$$F_{Box\ Total} = -621{,}000 + 271{,}028 + 319{,}078 = -30{,}894 \text{ lbs.}$$

.: The box is still in compression and the connection remains together.

Comparison between the two connections follows.

Type	5-1/2 inch I.F.	5-1/2 inch F.H.
O.D.	7-3/4 in	7-3/4 in
I.D.	4-1/8 in	4-1/8 in
M.O. Torque	53,247 ft-lbs	28,825 ft-lbs
F_i	1,064,957 lbs	621,000 lbs
F_o	2,287,586 lbs	916,596 lbs
I_B	121.53 in^4	141.93 in^4
I_P	44.87 in^4	23.08 in^4
A_B	20.75 in^2	26.16 in^2
A_P	13.88 in^2	8.28 in^2
S_{pi}	76,726 psi	75,000 psi
S_{Bi}	-109,563 psi	-41,317 psi
S_{PTotal}	105,626 psi	108,878 psi
S_{BTotal}	-61,026 psi	-2,055 psi
I_B/I_P	2.71	6.15
BSR	2.06	4.16
F_P	.55 Fe lbs	.322 Fe lbs

The above comparison shows that the I.F connection is better than the FH connection for the type of loading that this drill string will be receiving that is a combination of tension and bending in that since the pin will be subjected to large tensile loading and since the pin of the I.F connection is larger than that of the F.H connection and therefore is less likely to have a failure. An additional advantage of the I.F connection is the larger opening force or the amount of external loading that the connection can handle before opening.

APPENDIX IV

CALCULATED STRAINS AND DEFLECTIONS OF DRILLING SUBS COMPARED TO EXPERIMENTAL MEASUREMENTS

CALCULATED STRAINS AND DEFLECTIONS OF DRILLING SUBS COMPARED TO EXPERIMENTAL MEASUREMENTS

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May 15, 1978

Conclusions

1) The stress concentration factor at the change in section of the drilling subs, obtained from strain gauge measurements under pure axial load, and under pure bending load, should be essentially constant. A large scatter was actually observed in the concentration factor (+18% to -6% of the average value). This scatter is attributed to experimental errors which are at present unexplained. This scatter is larger than the differences between the calculated strain values and the experimental measurements.

2) The calculated strain values, based on a drilling sub with a uniform section (5.5 in. O.D. and 4.125 in I.D.) are in reasonable agreement with the strain gage measurements for combined tensile and bending loading when the geometric stress concentration at the change in section of the drilling subs is taken into account (see Table I).

3) The calculated center deflection is in good agreement with the measured deflection when the initial deflection of the test assembly due to its own weight is taken into account (see Table II).

4) The approximation of a uniform section for the drilling subs may be used in analyses for the behavior of the subs in the guide shoe, but account must be taken of the geometric stress concentrations due to the change in cross section at the hubs.

5) A more gradual transition at the change in section would significantly reduce the stress concentration factor and thereby increase the fatigue life of the drilling subs.

Summary of Results:

Calculations of the strains and deflections were made for combined tension and bending loads applied to drilling subs in the manner described in the Weatherford/Lamb U.S., Inc. report, "Experimental Evaluation of Optimum 5-1/2" I.F. Connection Design", by L. C. Eichberger (March 1978).

The drilling sub was modelled as (a) a uniform 5.5 in. O.D. x 4.125 in. I.D. tube, and (b) a tube of the same cross section with 26 in. long "tool joints" assumed to be rigid. The behavior of the uniform section tube was found to be in better agreement with the experimental measurements.

The experimental strain measurements under axial load only and bending load only showed a geometric stress concentration to exist at the end of the 44 in. long central section of the drilling subs. The measured values of this stress concentration at the point of maximum stress (gauge location 12) varied from 1.157 to 1.447 with an average value for all tests of 1.23. This variation (+18% to -6% of the average) is believed to be due to experimental error, since the stress concentration should be constant for pure tensile or bending loading.

Strains were calculated from the analytical treatment of a uniform section tube under combined tension and bending, at the tube center (location of gauge #8), and at the location of gauge #12. The calculated strains at gauge #12 were multiplied by 1.23 to account for the geometric stress concentration. These calculated strains are compared to the measured strains (for the extreme values of bending load in each combined load test) in Table I.

The total deflection of the center of the assembly under combined loading was calculated and found to be substantially less than the measured values listed in Tables C1-C4 of the Weatherford/Lamb report (for combined loading).

This is attributed to the omission of the initial deflection due to the weight of the drilling subs and the weight of the loading beam. The dial gauge readings were most probably zeroed prior to the application of the axial (or the bending) load, and the initial deflection must therefore be added to the dial gauge readings to give a comparison with the calculated deflections from an initially straight configuration.

The system weights were estimated, and an initial center deflection of 0.481 was calculated. The measured center deflection, plus 0.481 in., is compared to the calculated center deflection of the uniform tube under combined loading in Table II.

The calculated and measured bending deflections over the central portion of the test assembly are compared in Table III.

The Weatherford/Lamb report states that a bending load of 951891 in.lb. in combination with a 402025 lb. tension produces a curvature of 16°/100 ft. The slope of the drilling sub at two adjacent hubs was not measured, so it is unclear how this curvature was measured. The slope change was calculated to be 17.0°/100 ft. (hub-to-hub) under this combination of loads.

The analysis of the behavior of the drilling subs under combined loading is given in the appendix.

TABLE I

Calculated and Measured Strain Values at Gauges 8 and 12

Under Combined Loading (x 10^{-6} in./in.)

# Table	# Page	Bend. Load. in. lb.	Gauges	Gauge 8			Gauge 12		
				Calculated	Measured	% Diff.	Calculated*	Measured	% Diff.
A4a	A16	827992	Odd	748	655	-12	859	838	-2
A4b	A20	827992	Even	1830	1801	-2	2313	2583	+12
A8a	A40	951891	Odd	667	628	-6	750	698	-7
A8b	A44	951891	Even	1912	1922	+0.5	2421	2551	+5
A12a	A64	920916	Odd	687	615	-10	777	786	+1
A12b	A68	920916	Even	1891	1913	+1	2394	2782	+16
A16a	A88	941566	Odd	674	644	-4	759	820	+8
A16b	A92	941566	Even	1905	1929	+1	2412	2800	+16

* Values calculated for a uniform section were increased by 1.23 times to account for the geometric stress concentration.

From Reference 5

TABLE II

Calculated and Measured Values of Total Center Deflection

Under Combined Loads

# Table	# Page	Bending Load, in. lb.	Deflections, in. Calculated	Deflections, in. Measured*	% Difference
C1	C2	827992	1.624	1.613	-0.7
C2	C2	951891	1.867	1.775	-5.2
C3	C3	920916	1.806	1.788	-1.0
C4	C3	941566	1.846	1.793	-3.0

* Dial gauge "C" reading plus 0.481 in. calculated initial deflection

From Reference 5

TABLE III

Calculated Deflection vs. Measured Deflection Of Central Drilling Sub Under Combined Loading

# Page	# Table	x,* in.	y** measured, in.	y measured + y initial, in.	y calculated, in.	% difference
C2	C1	12	0.0105	0.016	0.014	-13
		34.812	0.0860	0.128	0.125	-2
C2	C2	12	0.0120	0.017	0.017	0
		34.812	0.1365	0.179	0.144	-20
C3	C3	12	0.0130	0.018	0.016	-11
		34.812	0.1385	0.181	0.139	-23
C3	C4	12	0.0130	0.018	0.016	-11
		34.812	0.1365	0.179	0.142	-21

* x Measured from mid-span location of dial gauge "C"

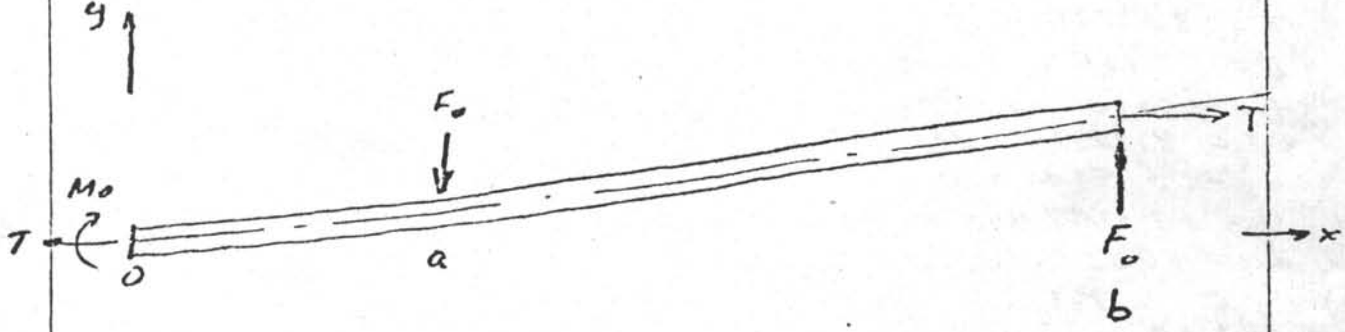
** y Deflection = reading of dial gauge "C" minus average of dial gauge readings at ± x

\# From Reference 5

①

APPENDIX

UNIFORM FLEXIBLE PIPE



BOUNDARY CONDITIONS

$x=0$: $\quad y_1(0)=0$ 1)

$y_1'(0)=0$ 2)

$y_1'''(0)=0$ (ZERO SHEAR) 3)

$y_1''(0)=M_o/EI$ 4)

$x=a$: $\quad y_1(a)=y_2(a)$ 5)

$y_1'(a)=y_2'(a)$ 6)

$y_1''(a)=y_2''(a)$ 7)

$x=b$: $\quad y_2''(b)=0$ 8)

$\Sigma M_o=0 \quad T\,y_2(b)+M_o=F_o(b-a)$ 9)

$0 \le x \le a$

$y_1=A\sinh\alpha x+B\cosh\alpha x+Cx+D \qquad \alpha=\sqrt{T/EI}$

$y_1'=A\alpha\cosh\alpha x+B\alpha\sinh\alpha x+C$

$y_1''=A\alpha^2\sinh\alpha x+B\alpha^2\cosh\alpha x$

$y_1'''=A\alpha^3\cosh\alpha x+B\alpha^3\sinh\alpha x$

$a \le x \le b$

$y_2=E\sinh\alpha x+F\cosh\alpha x+Gx+H$

$y_2'=E\alpha\cosh\alpha x+F\alpha\sinh\alpha x+G$

$y_2''=E\alpha^2\sinh\alpha x+F\alpha^2\cosh\alpha x$ (8 CONSTS + M_o = 9 UNKNOWNS

EQ.3 $\quad y_1'''(0)=A\alpha^3=0 \Rightarrow A=0$

EQ.1 $\quad y_1(0)=0 \quad D=-B \Rightarrow \quad y_1=B(\cosh\alpha x-1),\ y_1'=B\alpha\sinh\alpha x,\ y_1''=B\alpha^2\cosh\alpha$

EQ.8 $\quad E\sinh\alpha b+F\cosh\alpha b=0 \quad (\alpha\neq 0)$

EQ.4 $\quad B\alpha^2=M_o/EI \Rightarrow B=\frac{M_o}{\alpha^2EI}=\frac{M_o}{T}$

EQ.5 $\quad B(\cosh\alpha a-1)=E\sinh\alpha a+F\cosh\alpha a+Ga+H$

EQ.6 $\quad B\alpha\sinh\alpha a=E\alpha\cosh\alpha a+F\alpha\sinh\alpha a+G$

EQ.7 $\quad B\cosh\alpha a=E\sinh\alpha a+F\cosh\alpha a$

EQ.9 $\quad \underbrace{E\sinh\alpha b+F\cosh\alpha b}_{=0}+Gb+H+M_o/T=\frac{F_o}{T}(b-a)$

OR $\quad Gb+H+B=\frac{F_o}{T}(b-a)$

(2) NUMERICAL VALUES - FLEXIBLE, UNIFORM

$T = 402025$ lb.
$a = 42.125$
$b = 130.5$
$EI = 8.905 \times 10^{8}$ lb in^2
H.P. $\alpha^2 = 4.5148 \times 10^{-4}$
(1) $\alpha = 2.1248 \times 10^{-2}$
(2) $\alpha a = 0.895072$
(3) $\alpha b = 2.772865$
(4) $\sinh \alpha a = 1.019967$
(5) $\sinh \alpha b = 7.970966$
(6) $\cosh \alpha a = 1.428045$
(7) $\cosh \alpha b = 8.033449$

EQ. 8
$$F = -\frac{\sinh \alpha b}{\cosh \alpha b} E = \overbrace{-.99222}^{(8)} E$$

EQ. 7
$$B \cosh \alpha a = E(\sinh \alpha a - .99222 \cosh \alpha a) = -.397471\, E$$
$$E = \overbrace{-3.592829}^{(9)} B$$
$$F = \overbrace{3.564884}^{(10)} B$$

EQ. 6
$$\alpha B\left(\sinh \alpha a - \frac{E}{B}\cosh \alpha a - \frac{F}{B}\sinh \alpha a\right) = G$$
$$G = \overbrace{0.053458}^{(11)} B$$

EQ. 5
$$H = B\left(\cosh \alpha a - 1 - \frac{E}{B}\sinh \alpha a - \frac{F}{B}\cosh \alpha a - \frac{G}{B} a\right)$$
$$= \overbrace{-3.251918}^{(12)} B$$

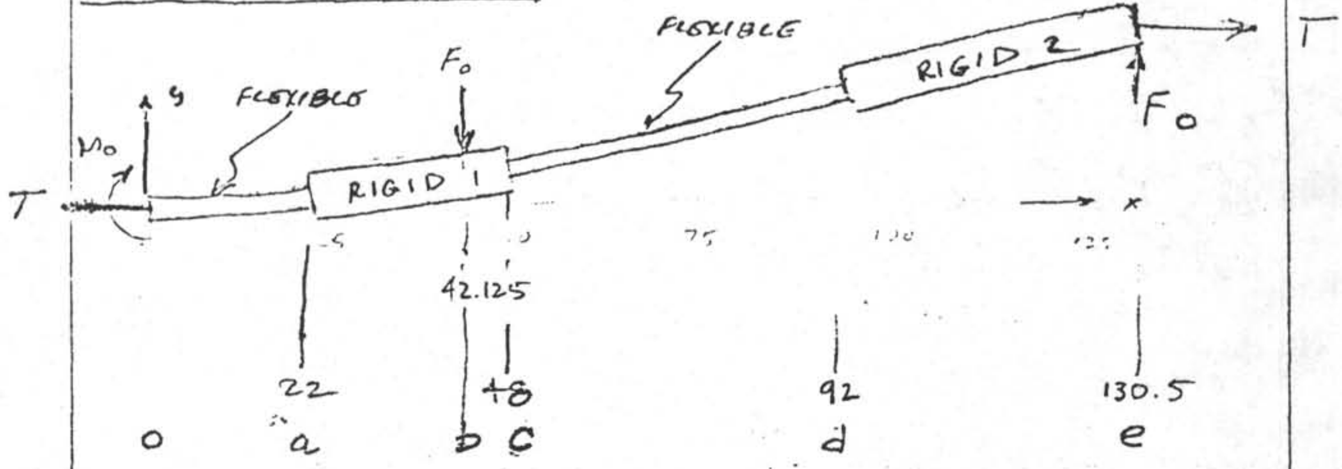
EQ. 9'
$$\left(1 + \frac{G}{B} b + \frac{H}{B}\right) B = \frac{F_0}{T}(b-a)$$
$$4.724351\, B = 88.375 \frac{F_0}{T}$$
(CHECKED) $$B = 18.7063 \frac{F_0}{T}$$

OR $$B = 0.211669 \frac{\text{BEND. MOM.}}{T}$$

$$\boxed{B = 5.2651 \times 10^{-7} \text{ (BEND. MOM.) in.lb.}}$$

3.)
/8/78

CONSIDER HUBS RIGID



$a = 22$
$b = 42.125$
$c = 48$
$d = 92$
$a = 130.5$

X=0:

$y(0) = 0$ 1)

$y'(0) = 0$ 2)

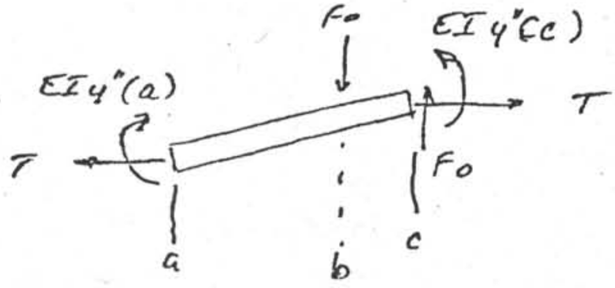
$y''(0) = M_o/EI$ 3)

$y'''(0) = 0$ 4)

RIGID 1:

$y'(a) = y'(c)$ 5)

$y(c) = y(a) + y'(a)(c-a)$ 6)

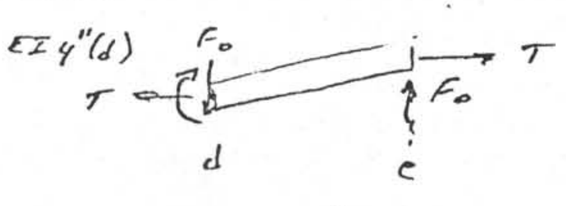


$\Sigma M_a = 0$:

$EIy''(a) + F_o(b-a) + T(y(c) - y(a)) = F_o(c-a) + EIy''(c)$

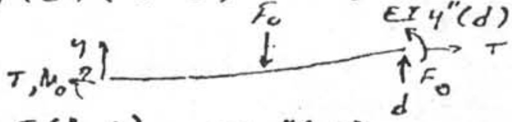
$$EI(y''(a) - y''(c)) + T(y(c) - y(a)) = F_o(c-b) \qquad 7)$$

RIGID 2 !



$$EIy''(d) + Ty'(d)(e-d) = F_o(e-d) \qquad 8)$$

$\Sigma M_o = 0$:



$$Ty(d) + M_o = F_o(d-b) + EIy''(d) \qquad 9)$$

4).

$0 \le x \le a$ $\qquad\qquad$ $\alpha^2 = \frac{T}{EI}$, $EI = \frac{T}{\alpha^2}$

$y = A\sinh\alpha x + B\cosh\alpha x + Cx + D$

$y' = A\alpha\cosh\alpha x + B\alpha\sinh\alpha x + C$

$y'' = \alpha^2(A\sinh\alpha x + B\cosh\alpha x)$

$y''' = \alpha^3(A\cosh\alpha x + B\sinh\alpha x)$

$c \le x \le d$

$y = E\sinh\alpha x + F\cosh\alpha x + Gx + H$

$y' = E\alpha\cosh\alpha x + F\alpha\sinh\alpha x + G$

$y'' = \alpha^2(E\sinh\alpha x + F\cosh\alpha x)$

4): $y'''(0) = 0 = A\alpha^3 \Rightarrow A = 0$

2) $y'(0) = 0 = C \Rightarrow C = 0$

1) $y(0) = 0 \Rightarrow$

$$\left.\begin{aligned} y &= B(\cosh\alpha x - 1) \\ y' &= B\alpha\sinh\alpha x \\ y'' &= B\alpha^2\cosh\alpha x \end{aligned}\right\} \quad 0 \le x \le a$$

3) $y''(0) = M_0/EI \Rightarrow M_0 = BT$

5) $B\alpha\sinh\alpha a = E\alpha\cosh\alpha c + F\alpha\sinh\alpha c + G$

6) $E\sinh\alpha c + F\cosh\alpha c + Gc + H = B(\cosh\alpha a - 1) + (B\alpha\sinh\alpha a)(c-a)$

$E\sinh\alpha c + F\cosh\alpha c + Gc + H = B(\cosh\alpha a + (c-a)\alpha\sinh\alpha a - 1)$

7) $\frac{T}{\alpha^2}(y''(a) - y''(c)) + T(y(c) - y(a)) = F_0(c-b)$

or using 6 ($y(c) - y(a) = y'(a)(c-a)$

$$B\cosh\alpha a - E\sinh\alpha c - F\cosh\alpha c + y(c) - y(a) = \frac{F_0(c-b)}{T}$$

or

$$\cancel{B\cosh\alpha a} - \cancel{E\sinh\alpha c} - \cancel{F\cosh\alpha c} + \cancel{E\sinh\alpha c} + \cancel{F\cosh\alpha c} + Gc + H - B(\cancel{\cosh\alpha a} - 1) = \frac{F_0(c-b)}{T}$$

7') or $\quad Gc + H + B = \frac{F_0(c-b)}{T}$

5.)

$$8)\quad T(E\sinh\alpha d + F\cosh\alpha d) + T(e-d)(E\alpha\cosh\alpha d + F\alpha\sinh\alpha d + G) = F_0(e-d)$$

or

$$8')\quad E[\sinh\alpha d + (e-d)\alpha\cosh\alpha d] + F[\cosh\alpha d + (e-d)\alpha\sinh\alpha d] + (e-d)G = F_0(e-d)/T$$

$$9)\quad y(d) + B = \frac{F_0}{T}(d-b) + \frac{1}{\alpha^2}y''(d)$$

$$E\sinh\alpha d + F\cosh\alpha d + Gd + H - E\sinh\alpha d - F\cosh\alpha d + B = \frac{F_0}{T}(d-b)$$

or

$$9')\quad Gd + H + B = \frac{F_0(d-b)}{T}$$

EQUATION SUMMARY (B, E, F, G, H UNKNOWNS)

① $B\alpha\sinh\alpha a = E\alpha\cosh\alpha c + F\alpha\sinh\alpha c + G$

② $B[\cosh\alpha a + (c-a)\alpha\sinh\alpha a - 1] = E\sinh\alpha c + F\cosh\alpha c + Gc + H$

③ $Gc + H + B = F_0(c-b)/T$

④ $E[\sinh\alpha d + (e-d)\alpha\cosh\alpha d] + F[\cosh\alpha d + (e-d)\alpha\sinh\alpha d] + (e-d)G = F_0(e-d)/T$

⑤ $Gd + H + B = F_0(d-b)/T$

SOLUTION:

③,⑤ $\quad G(d-c) = F_0(d-c)/T \Rightarrow \boxed{G = F_0/T}$

(5') $\quad H + B = -\dfrac{F_0 b}{T} \Rightarrow \boxed{H = -B - \dfrac{F_0 b}{T}}$

THEN ② BECOMES

$$B[\cosh\alpha a + (c-a)\alpha\sinh\alpha a - 1] = E\sinh\alpha c + F\cosh\alpha c + \frac{F_0}{T}(c-b) - B$$

or

$$B[\cosh\alpha a + (c-a)\alpha\sinh\alpha a] - E\sinh\alpha c - F\cosh\alpha c = \frac{F_0}{T}(c-b)$$

6) 3 EQUATIONS IN B, E, F.

(1) $B\alpha \sinh \alpha a - E\alpha \cosh \alpha c - F\alpha \sinh \alpha c = F_0/T$

(2) $B[\cosh \alpha a + (c-a)\alpha \sinh \alpha a] - E \sinh \alpha c - F \cosh \alpha c = \frac{F_0}{T}(c-b)$

(3) $E[\sinh \alpha d + (e-d)\alpha \cosh \alpha d] + F[\cosh \alpha d + (e-d)\alpha \sinh \alpha d] = 0$

$a = 22$ $\quad c-a = 26$
$b = 42.125$ $\quad c-b = 5.875$
$c = 48$ $\quad e-d = 38.5$
$d = 92$
$e = 130.5$

o $T = 402025$ lb. $\quad$ 2 $\alpha a = 0.4675$
F_0 = VARIABLE $\quad$ 3 $\alpha c = 1.0199$
$\alpha^2 = 4.5148 \times 10^{-4}$ $\quad$ 4 $\alpha d = 1.9548$
1 $\alpha = 2.1248 \times 10^{-2}$

5 $\sinh \alpha a = 0.4847$
6 $\sinh \alpha c = 1.2061$ $\quad$ (ELIMINATE F USING 3
7 $\sinh \alpha d = 3.4605$ $\quad$ ELIMINATE E FROM 1, 2)

8 $\cosh \alpha a = 1.1113$
9 $\cosh \alpha c = 1.5668$
.0 $\cosh \alpha d = 3.6021$

(1) $0.010298212 B - 0.033290921 E - 0.025628264 F = F_0/T$

(2) $1.379015143 B - 1.206149496 E - 1.566779055 F = 5.875\, F_0/T$

(R.1)

(3) $6.407209180 E + 6.43297186 F = 0$

(R.2)

$F = -0.995995213 E$

$0.010298221213 - 0.007765293 E = F_0/T$
$1.379015143 B + 0.354354943 E = 5.875\, F_0/T$

$3.891621015 B + E = 16.57942161\, F_0/T$
$1.326185786 B - E = 128.7781414\, F_0/T$

$5.2178 B = 145.357563\, F_0/T$

(CHECKED) $B = 27.858\, F_0/T$ (LARGER THAN FOR UNIFORM SECTION)

7)

DEFLECTIONS

FLEXIBLE, UNIFORM.

$y(\overset{42.125}{\cancel{\quad}}) = 8.007\ F_o/T = B(\cosh \alpha a - 1)$

$y(130.5) = E \sinh \alpha b + F \cosh \alpha b + Gb + H = 69.67\ F_o/T$

RIGID HUBS

$y(42.125) = y(22) + y'(22)(20.125) = 8.874\ F_o/T$

$y(130.5) = 60.467\ F_o/T$

BEND. MOM $= F_o(130.5 - 42.125) = 88.375 F_o$

$T = 402025$, $\frac{F_o}{T} = \frac{\text{BEND. MOM}}{(88.375)(402025)} = 2.8146 \times 10^{-8}$ BEND. MOM

FLEX, UNIFORM

$y(42.125) = 2.254 \times 10^{-7}$ (BEND. M.)

$y(130.5) = 1.961 \times 10^{-6}$ (BEND. M.)

RIGID HUBS

$y(42.125) = 2.498 \times 10^{-7}$ (BEND. M.)

$y(130.5) = 1.702 \times 10^{-6}$ (BEND. M.)

SPOT CHECK: C2 $BM = 827992$, $C = 1.132$

$y_{flex} = 1.624$

$y_{rigid} = 1.41$

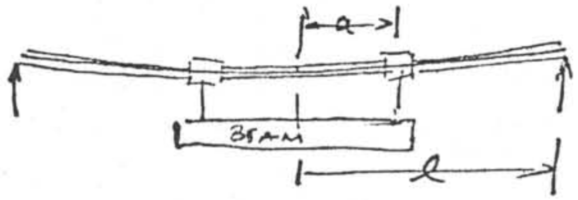
C3 $BM = 920916$, $C = 1.307$

$y_{flex} = 1.806$

$y_{rigid} = 1.567$

DIAL GAGE READINGS ARE CHANGE IN DEFL. FROM INITIAL CURVE OF DEFLECTION DUE TO PIPE WEIGHT. ESTIMATE INITIAL DEFLECTION!

8) DEFLECTION DUE TO WEIGHT OF SUBS & LOADING BEAM.



CENTER DEFLECTION W/R TO ENDS

$\delta_1 = \frac{5 w l^4}{24 EI}$ (DUE TO 5.5" O.D. × 4.125 I.D. STEEL)

$w = \frac{(7.9)(5.5^2 - 4.125^2)\pi(62.4)}{4(1728)} \cong 4$ #/in.

$EI = 8.905 \times 10^8$ in²lb

$l = 130.5$ in

$\delta_1 \simeq 0.256"$

δ_2 = DEFLECTION DUE TO HUB WEIGHT + LOADING BEAM WEIGHT

HUB: 7" O.D. × 5.5 I.D. × 26"

$W_h = \frac{(49 - 5.5^2)(7.9)(62.4)(26)\pi}{4 \cdot 1728} \cong 110$ lb.

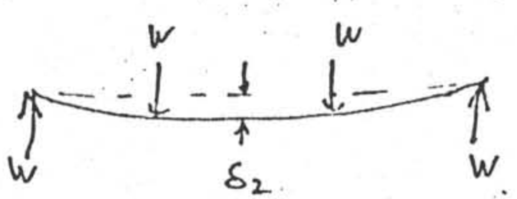
$\frac{1}{2}$ LOADING BEAM ~ 8 × 8 × $\frac{1}{4}$ "I" × 74.4"

$W_b = \frac{3 \times 8 \times \frac{1}{4} \times 74.35 \times 7.9 \times 62.4 \times \pi}{4 \times 1728} \cong 100$ lb.

$W_{LOAD\ STRAP} \cong 50$ lb.

$W_{JACK} \cong 40$ lb (½)

$W = W_h + W_b + W_{LOAD\ STRAP} + W_{JACK} \cong 300$ lb.



$$\delta_2 = \frac{W}{2EI}\left(\frac{2}{3}l^3 - la^2 + \frac{a^3}{3}\right) \qquad a = 35,\ l = 130.5,\ EI = 8.905\times10^8$$

$\delta_2 = 0.225$

δ = INITIAL MID-POINT DEFLECTION = $\delta_1 + \delta_2 = 0.481$ in.

HUB – TO – HUB SLOPE CHANGE

$y' = B\alpha \sinh \alpha x$, $y'(35) = 9.1081\times10^{-9}$ (BEND. LOAD) RADIANS
$= 5.2186\times10^{-7}$(B.L.) DEG/35 inches

WHICH GIVES 1.7892×10^{-5} (BEND. LOAD) °/100 ft.

OR 17.0°/100' AT BEND. LOAD = 951891 in. lb.

CALCULATED STRAINS

$$e_{\text{TENSILE LOAD}} = \frac{T}{AE} = \frac{402025}{(\pi/4)(5.5^2 - 4.125^2)30\times10^6} = 1289\ \mu\ \text{in/in.}$$

$$|\epsilon_{BEND}| = \frac{r\,d\theta}{ds} = \frac{Mr}{EI}$$

$$0 \le x \le a,\quad |\epsilon_{BEND}| = B\alpha^2 r \cosh \alpha x,\quad r = 2.75''$$

$$\epsilon_{axial} = 1289 \pm B\alpha^2 r \cosh \alpha x$$
$$= 1289 \pm 6.537\times10^{-4}\,(\text{BEND. LOAD}) \cosh \alpha x$$

$$(\epsilon_{axial})_{GAGE\ 8,\ x=0} = 1289 \pm 6.537\times10^{-4}\,(\text{BEND. LOAD})$$

$$(\epsilon_{axial})_{GAGE\ 12,\ x=20''} = \left[1289 \pm 7.1366\times10^{-4}\,(\text{BEND. LOAD})\right](1.23)$$

(ASSUMING THE AVERAGE MEASURED STRESS CONCENTRATION)

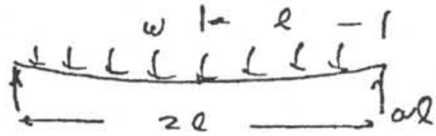
BENDING LOAD = $F_0(c-b) = 88.375 F_0$ IS LISTED IN TABULAR DATA OF WEATHERFORD/LAMB REPORT

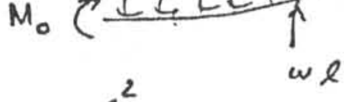
CALCULATED DEFLECTIONS

$0 \le x \le a$ (DIAL GAGES A – E), $y = 5.2651\times10^{-7}(\cosh 2.1278\times10^{-2}x - 1)$(BEND. LOAD)

MID POINT W/R ENDS (SEE p. 7)

10) MIDDLE SUB DEFLECTIONS (INITIAL) AT $x=12$, $x=34.812$



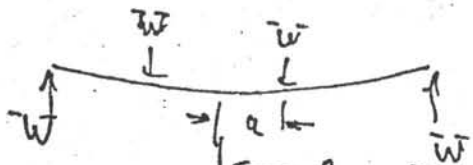


$$M_o = wl^2 - (wl)\frac{l}{2} = \frac{wl^2}{2}$$

$$EI\,y_1'' = \frac{wl^2}{2} - \frac{wx^2}{2} = \frac{w}{2}(l^2 - x^2)$$

$$EI\,y_1' = \frac{w}{2}\left(l^2x - \frac{x^3}{3}\right) + \cancel{C_1}^{\,0}$$

$$EI\,y_1 = \frac{w}{2}\left(\frac{l^2x^2}{2} - \frac{x^4}{12}\right) \Rightarrow y_1 = \frac{w}{4EI}\left(l^2x^2 - \frac{x^4}{6}\right)$$



$\underline{0 \le x \le a}$

$$EI\,y_2'' = W(l-a)\,, \quad EI\,y_2' = W(l-a)x$$
$$EI\,y_2 = W(l-a)\frac{x^2}{2}$$

$$y_2 = \frac{W}{EI}(l-a)\frac{x^2}{2}$$

$y_1 + y_2$ = INITIAL DEFLECTION

$w = 4$ #/in

H.P. $W = 300$ #

0 $EI = 8.905 \times 10^8$ in²#

1 $l = 130.5$ in

$a = 35$ in

$l - a = 95.5$ in.

2 x

$\underline{x=12}$

$y_1 + y_2 = 0.0028 + 0.0023 = .0051$

$\underline{x = 34.812}$

$y_1 + y_2 = .0229 + .0195 = .0424$

11)

GEOMETRIC STRESS CONC. FACTOR AT POINT OF MAXIMUM STRESS

$K = (\epsilon_{AXIAL})_{12} / (\epsilon_{AXIAL})_{8}$

PAGE OF TABLE	LOADING	GAGES	K READINGS	K_{AVE}	
A4	AXIAL	ODD	~~1.1984~~, 1.1734, 1.1737, 1.1769	1.18	← AXIAL
A6	AXIAL	EVEN	1.216, 1.2185, 1.2132, 1.2134	1.215	←
A10	BENDING	ODD	~~1.135~~, ~~1.149~~, 1.157, 1.153, 1.156	1.154	
A12	BENDING	EVEN	1.165, 1.182, 1.185, 1.184, 1.186	1.184	
A28	AXIAL	ODD	1.173, 1.173, 1.170, 1.170	1.172	← AXIAL
A30	AXIAL	EVEN	1.195, 1.194, 1.182, 1.1755	1.184	←
A34	BENDING	ODD	1.165, 1.157, 1.16, 1.158, 1.163	1.161	
A36	BENDING	EVEN	1.16, 1.186, 1.182, 1.178, 1.179	1.177	
A52	AXIAL	ODD	1.36, 1.365, 1.36, 1.36,	1.36	← AXIAL
A54	AXIAL	EVEN	1.42, 1.435, 1.419, 1.4078	1.42	←
A58	BENDING	ODD	1.14, 1.15, 1.154, 1.154, 1.155	1.151	
A60	BENDING	EVEN	1.21, 1.192, 1.193, 1.188, 1.185	1.194	
A76	AXIAL	ODD	1.41, 1.387, 1.4075, 1.389,	1.39	← AXIAL
A78	AXIAL	EVEN	1.447, 1.439, 1.422, 1.421	1.43	←
A82	BENDING	ODD	1.171, 1.157, 1.1595, 1.1578, 1.160	1.161	
A84	BENDING	EVEN	1.225, 1.204, 1.192, 1.192, 1.189,	1.20	
			AVE:	1.23	

THROWING OUT 4 HIGH ONES: AVE: 1.18

12)

COMBINED LOAD CASE – MAXIMUM MOMENT –

MEASURED & CALCULATED STRAIN VALUES AT GAGES 8 & 12

(THE CALCULATED STRAIN AT GAGE 12 IN A UNIFORM 5.5 O.D. x 4.125 I.D. TUBE WAS MULTIPLIED BY THE AVERAGE STRESS CONCENTRATION FACTOR OF 1.23)

$$(\epsilon_{AXIAL})_{GAGE\ 8} = 1289.25 \pm 6.537\times10^{-4}\,(\text{BEND. LOAD})$$

$$(\epsilon_{AXIAL})_{GAGE\ 12} = 1.23\left[1289.25 \pm 7.1366\times10^{-4}\,(\text{BEND. LOAD})\right]$$

+ SIGN FOR EVEN # GAGES
– SIGN FOR ODD # GAGES

TABLE I

TABLE	PQ	MOM.	GAGES	CALC. $(\epsilon)_8$	MEAS. $(\epsilon)_8$	CALC. $(\epsilon)_{12}$	MEAS. $(\epsilon)_{12}$
A4a	A16	827992	ODD	748	655 (-12%)	859	838(-2% (-2.4)
A4b	A20	827992	EVEN	1830	1801 (-2%)	2313.	2583(+12 (+11.7)
A8a	A40	951891	ODD	667	628 (-6%)	750.	698 (-7 (-6.9)
A8b	A44	951891	EVEN	1912	1922 (+0.5%	2421	2551(+5 (+5.4)
A12a	A64	920916	ODD	687	615 (-10%)	777	786(+1% (+1.2)
A12b	A68	920916	EVEN	1891	1913 (+1%)	2394	2782(+16. (+16.2)
A16a	A88	941566	ODD	674	644 (-4%)	759	820(+8% (+8.0)
A16b	A92	941566	EVEN	1905	1929 (+1%)	2412	2800(+16. (+16.1)

DIFFERENCES BETWEEN MEASURED AND CALCULATED $(\epsilon)_{12}$ VALUES MAY BE ATTRIBUTED TO UNCERTAINTY IN THE STRESS CONCENTRATION FACTOR (+16%/-2.4%) SMALLER DIFFERENCES IN $(\epsilon)_8$ VALUES ARE IN THE RANGE ±2% (EVEN GAGES) AND (-12% TO -4%) ODD GAGES.

IF A STRESS CONC. FACTOR OF 1.1382 x 1.23 = 1.40 IS USED:

WITH 1.40 STRESS CONC.

	$(\epsilon)_8$ CALC	$(\epsilon)_8$ MEAS	%	$(\epsilon)_{12}$ CALC	$(\epsilon)_{12}$ MEAS	%
A4a	748	655	-12.4	977	838	-14.2
A4b	1830	1801	-1.6	2633	2583	-1.9
A8a	667	628	-5.8	854	698	-18.3
A8b	1912	1922	+0.5	2755	2551	-7.4
A12a	687	615	-10.5	884	786	-11.1
A12b	1891	1913	+1.2	2725	2782	+2.1
A16a	674	644	-4.5	863	820	-5.0
A16b	1905	1929	+1.3	2745	2800	+2.0