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Fatigue Testing of Drillpipe

G.Y. Grondin* and G.L. Kulak,** Natl. Research Council of Canada

Summary. Twenty-nine tests in air and 27 tests in a 3.5% NaCl solution were conducted to test the fatigue strength of Grade E 4.5-in.-OD, 16.6-lbm/ft [114-mm-OD, 24.7-kg/m] drillpipe. The effects of stress range, mean stress, corrosion, and upset geometry were tested on the drillpipe body alone and on specimens that included the tool joint, upset, and drillpipe body. Stress range and mean stress effects were significant in noncorrosive and corrosive environments. The effect of upset geometry was minimal on an external upset compared with that on an internal/external upset. Among the specimens that failed, grinding marks on the drillpipe surface, which occurred during inspection at the mill, caused failure of about 60% of the specimens. Grinding created a notch effect and induced a detrimental residual-stress pattern on the pipe surface, which decreased drillpipe fatigue life. API guidelines for cumulative fatigue of Range 2 drillpipe were evaluated in light of these experimental test results and were found to be unsafe for drilling in noncorrosive conditions and possibly too conservative for corrosive conditions.

Introduction

Fatigue damage occurs when a drillpipe is subjected to sufficiently high alternating stresses, such as those created when the drillpipe rotates in the curve of a wellbore. Drillpipe fatigue failure has been a serious concern in the oil industry ever since sections of drillpipe were first joined to permit drilling at depths greater than one length of drillpipe.¹⁻⁴ This problem was addressed by imposing dogleg severity limits⁵ based on test results published in the early 1950's.⁶ These tests were satisfactory for their intended use (i.e., testing of tool-joint welds). However, these tests were not performed in a corrosive environment or under axial tension, two factors considered important in current API guidelines.⁵ During these tests, the effect of corrosion was addressed by use of simplifying assumptions with respect to the decrease in fatigue strength in a corrosive environment. The effect of mean stress was dealt with by use of a modified Goodman equation for the endurance limit and a standard Goodman equation for stresses above the endurance limit.⁷⁻⁹

Published fatigue-test data on full-sized drillpipe are limited. Tests performed by Hughes Tool Co. still represent a major contribution toward better understanding of drillpipe fatigue strength.⁶ More recently, the Chinese Petroleum Standardization Committee presented the results of an investigation on the effect of the geometry of the transition zone between the drillpipe body and the upset.¹⁰ Full-sized drillpipes with transition zones of various lengths and radii were tested in air as cantilever rotating beams under no axial load. Fatigue resistance improved as the length of the transition zone increased.

Tsukano *et al.*¹¹ shed some light on the effect of upset/pipeline-body transition-zone geometry in their investigation of the internal-upset drillpipe geometry using finite-element analysis and tests. They sought a combination of taper length and radius of runout that would cause fatigue failure in the drillpipe body rather than in the pipeline-body/upset transition zone. To verify the results of the finite-element investigation, full-sized specimens were tested in four-point rotary-bending arrangements in air at a high stress range (location of crack initiation was the only factor investigated). Recommendations were made for suitable internal-upset geometry to prevent drillpipe failure in the upset region.

The drilling industry recognizes corrosion as a main factor affecting fatigue life.^{3,8,12,13} The effect of corrosive environment on drillpipe fatigue life has been investigated with small steel samples, but this effect on full-sized drillpipe has not been compared with test results from similar drillpipes tested in air. Joosten *et al.*¹⁴ tested 3-ft [0.9-m] lengths of 3½-in. [89-mm] diameter Grade G-105 drillpipe loaded in three-point bending. Tests were conducted in air and in KCl/seawater solution. Only results of tests in a corrosive environment were published.¹⁴ Consequently, resistance in air and in a corrosive environment cannot be compared from their published data.

In 1988, Dale¹⁵ presented the results of a test program on API drillpipe steels conducted to determine the influence of drilling-fluid environment on fatigue-crack growth rate. Although the pro-

gram mainly studied fatigue in a full-sized drill collar, it also included a series of fatigue tests on coupon specimens in drilling muds of various composition. The tests, conducted at a 5-Hz frequency, showed no significant effect of drilling mud corrosivity on crack growth rate.

Helbig and Vogt¹⁶ presented the results of a study on drillpipe fatigue life in a corrosive environment that investigated the effect of heat treatment on Grades D, E, and S-135 drillpipe. Full-sized sections of drillpipe bodies were fatigue tested in two corrosive environments: tap water and 20% NaCl solution. The test results did not indicate a significant difference between normalized and quenched-and-tempered specimens or between the two test environments. From their tests on coupon specimens, Helbig and Vogt demonstrated that the speed of testing is influential when tests are conducted in a corrosive environment. Fatigue life was reduced significantly when the testing frequency was decreased from 1,000 to 100 rev/min; the amount of reduction depended on the stress range at which the tests were conducted. All the tests of Helbig and Vogt¹⁶ on full-sized drillpipe were performed in a corrosive environment. Consequently, fatigue life of full-sized drillpipe in air and in a corrosive environment cannot be compared with their data. No experimental investigation to date has specifically addressed the problem of the effect of mean stress on the fatigue life of full-sized drillpipe operating in air or in a corrosive environment.

As the search for petroleum moves into more-hostile environments that require drilling to greater depths and in more-corrosive media, the oil industry is again confronted with the problem of drillpipe fatigue. One failure is estimated to occur for every 6,500 ft [1980 m] drilled, including drillpipe separations and washouts.¹⁷ Most drillpipe failures generally are agreed to result from metal fatigue. Recent publications show that drillpipe failure is still a serious concern of drilling contractors.^{3,14,17} More research is therefore required to determine the effect of such parameters as mean stress and corrosion on drillpipe fatigue life.

Experimental Program

A preliminary test program¹⁸ on full-sized drillpipe specimens that incorporated the tool joint, upset ends, and drillpipe body showed that failures in the field could be duplicated in the laboratory. Preliminary testing was done under cyclic tension and bending.

A full program was then designed to investigate the effects of stress range, mean stress, and corrosion on drillpipe fatigue life under rotational bending. Full-sized API Grade E 4½-in.-OD, 16.6-lbm/ft [114-mm-OD, 27.4-kg/m] drillpipes with internal/external upsets were tested. Test specimens were prepared so that the tool joint, upset region, and part of the pipe body could be incorporated in the test section. Testing showed that failures in all specimens except one occurred in the drillpipe body, not the tool joint or upset region. For this reason and for reasons of economy, half the test specimens were sections taken from only the drillpipe body.

Test Setup

A rotating/bending test frame that could test full-sized drillpipe under axial preload was designed to simulate field conditions. The test

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Fig. 1—Diagram of rotating/bending test setup (all units are given in millimeters).

frame, shown as a schematic in Fig. 1, uses a simply supported rotating beam under two-point loading to provide a region of constant moment 59 in. [1500 mm] long. The test frame was designed so that specimens up to 5.3 in. [135 mm] in diameter could be rotated under a deflected configuration equivalent to a dogleg with a severity in excess of 100°/100 ft [98°/30 m] at up to 7 Hz. The tensile axial load was introduced by a preloading apparatus. A prestressing rod was inserted and centered in the drillpipe and put into compression by special end fittings attached to the pipe. The configuration was such that the pipe sustained an equal and opposite tensile force, and the whole arrangement allowed pipe rotation under the preloaded condition. The preload level was established and then monitored with strain gauges on both the prestressing rod and the pipe. These readings were taken off the rotating pipe with an electronic data logger. Ref. 19 gives full details of the prestressing arrangement, the test setup, and the testing procedure.

Three levels of axial load were investigated: zero axial preload, which is what has commonly been used in the past; 18 ksi [125 MPa] (axial load of 79 kip [350 kN]), which corresponds to the stress at the drillstring top if drilling is carried 5,000 ft [1525 m] in a 15-lbm/gal [1.80-kg/L] drilling mud, and 36 ksi [250 MPa] (axial load of 158 kip [700 kN]), which corresponds to the stress at the drillstring top when drilling to 10,000 ft [3050 m].

For corrosion fatigue tests, 3.5% NaCl solution was sprayed on the test specimens at 0.53 gal/min [2.0 L/min] during testing. Because corrosion is time-dependent, testing frequency is an important factor in a corrosive environment. In the field, drillpipe is subjected to high and low stress cycles and, consequently, is expected to last longer than pipe tested at a constant high stress range. Thus, tests in a corrosive environment were run at 1.0 Hz, or lower than the 1.5 Hz commonly used in the field. Because of the lower testing frequency, drillpipes were exposed to the corrosive environment for a longer period. Consequently, test results should correlate better with field conditions.

The test program was complemented by microscopic examination of the failed specimens to determine the origin of fatigue failure. Because residual stresses may obliterate the effect of mean stress on drillpipe fatigue life, extensive residual-stress measurements were made. X-ray diffraction was used to measure local residual stresses, which are influential during the crack initiation stage

of fatigue life. Metallographic examination of the microstructures of drillpipes from different sources was also performed.

Test Results

Noncorrosive Environment. Results from test in air are presented schematically in Fig. 2. In Fig. 2a, the mean stress was kept at 18 ksi [125 MPa] and the stress range was varied from 26 to 51 ksi [180 to 350 MPa]. The effects of mean stress are presented in Fig. 2b. Mean stress was varied from 0 to 36 ksi [250 MPa], and the stress range was kept at 42 ksi [290 MPa]. Statistical analysis of test results showed that the logarithm of fatigue life is directly proportional to the square of mean stress. Multiple-linear-regression analysis of data from tests in air was performed to obtain an expression for the effect of stress range and mean stress on drillpipe fatigue life. For tests that were interrupted before specimen failure, the number of cycles applied to the specimen was conservatively used for the regression analysis. The relation between fatigue life and mean stress and stress range can be expressed as

$$\log N = 11.9 - 3.47 \log \sigma_r - 7.85 \times 10^{-4} (\bar{\sigma})^2. \dots\dots\dots (1)$$

In SI metric units, constants 11.9 and 7.85×10^{-4} are 14.8 and 1.65×10^{-5} , respectively. This expression is applicable for the loading conditions used during the experimental investigation (i.e., stress range between 26 and 51 ksi [180 and 350 MPa] and mean stress between 0 and 36 ksi [250 MPa]). Standard deviation is 0.50 for the data around the mean regression line. The stress range and the mean stress were both found to be statistically significant at a confidence level of 95%.¹⁹

Corrosive Environment. Fig. 3 shows results from tests in a 3.5% NaCl solution. Fig. 3a presents the effect of stress range, investigated at a mean stress of 18 ksi [125 MPa]. One specimen, tested at a stress range of 23 ksi [160 MPa], failed prematurely at 422,300 cycles, compared with an average of 1,699,100 cycles for two other specimens tested under the same conditions. The premature failure was caused by a steel stencil mark on the drillpipe body. We did not include this specimen in the regression analysis because stencil marking is not generally used on the drillpipe body and the presence of the marking on the test specimen is considered exceptional. Furthermore, such practice has long been recognized as

(a) Effect of Stress Range on Fatigue Life

(b) Effect of Mean Stress on Fatigue Life

Fig. 2—S/N curves showing results of (a) stress range and (b) mean stress fatigue tests conducted in air.

detrimental to drillpipe life, and the drilling industry has been cautioned against it.² Observe that the scatter in the test results is significantly lower than the scatter observed in results from tests in a noncorrosive environment. Fig. 3b presents the effect of mean stress on fatigue life at three different levels of stress range: the square of the mean stress is plotted vs. fatigue life. Because of the lower scatter observed in the corrosion fatigue-test results and the time-consuming nature of tests in a corrosive environment (a single test can take from 1 to 16 weeks), most tests were repeated only twice.

Multiple-linear-regression analysis of results from tests in a 3.5% NaCl solution yields the following expression for mean fatigue life, N , as a function of the stress range, σ_r , and the mean stress, $\bar{\sigma}$

$$\log N = 9.56 - 2.35 \log \sigma_r - 2.63 \times 10^{-4} (\bar{\sigma})^2 \quad (2)$$

In SI metric units, constants 9.56 and 2.63×10^{-4} are 11.53 and 5.54×10^{-6} , respectively. Standard deviation is 0.12 for the data around the mean regression line. Eq. 2 is applicable in a 3.5% NaCl solution for mean stresses from 0 to 36 ksi [0 to 250 MPa], a stress range between 13 and 42 ksi [90 and 290 MPa], and a testing frequency of 1.0 Hz. Again, statistical analysis of the regression coefficients indicated that, at a confidence level of 95%, both the stress range and mean stress significantly affect fatigue life.

Fig. 4 compares the mean fatigue curve for a corrosive vs. a noncorrosive environment, with both curves at a mean stress level of 18 ksi [125 MPa]. It is evident from the curves that a corrosive environment negatively affects drillpipe fatigue life. It is also evident that fatigue life is reduced more at lower stress ranges, which is thought to be the consequence of the time dependence of the corrosion process; at lower stress ranges, the drillpipe is exposed to the corrosive environment for a longer period.

Origin of Fatigue Failures

Fatigue failures of specimens tested in corrosive or noncorrosive environments were observed to originate on the outer surface of the drillpipe. These results contradict those of previous failures in the

(a) Effect of Stress Range on Fatigue Life

(b) Effect of Mean Stress on Fatigue Life

Fig. 3—S/N curves showing results of (a) stress range and (b) mean stress fatigue tests conducted in a 3.5% NaCl solution.

field, which were reported to originate on the inner surface of the drillpipe.^{2,7,10} For the pipes we tested in a corrosive environment, fatigue cracks were expected to start on the outer surface because the outer surface (not the inner surface) of the drillpipe was in contact with the corrosive environment. Drillpipes tested in air also are more likely to fail from cracks originating on the outer surface of the drillpipe, where bending stresses are larger. However, bending stresses on the inner pipe surface are not that much lower than on the outer surface because the drillpipe wall is relatively thin. Thus, such factors as drilling fluid velocity, amount of entrained oxygen, and presence of scale may overshadow the stress difference between the outer and inner surfaces of pipe used in the field. Furthermore, the cleansing action of the drillpipe outer surface rubbing against the borehole wall is thought to lessen the effects of corrosion on the outer surface vs. the inner surface.¹³ In the laboratory, where environmental conditions are either the same on the inner and outer surfaces (tests in air) or more severe on the outer surface (tests in a corrosive environment), the outer surface is prone to fatigue-crack initiation.

All but two specimens failed in the drillpipe body away from the upset runout. The effect of upset geometry on stress concentration in the drillpipe body was investigated with the finite-element method.¹⁹ As expected, this showed that a concentration of stresses exists in the pipe body next to the transition zone between the upset and the pipe body. The stress concentration factor (ratio of maximum stress calculated in the upset region to nominal pipe-body stress) for standard internal-/external-upset geometry²⁰ was 1.17 for axial tension and 1.10 for pure bending. Examination of the upset geometry on the test specimens showed a wide variation in the length of the transition zone (from 1.0 to 3.0 in. [25 to 75 mm]). The length of the transition zone varied not only among drillpipes but also around the drillpipe perimeters. We also found that the transition zone is generally rounded, which results in a more gradual change in wall thickness and a less-severe stress concentration than for the straight-line transition assumed in the analysis. Therefore, it is possible that slight imperfections on the outer surface of the drillpipe body can overshadow the effect of stress concentration at the transition zone.

Fig. 4—S/N curve showing the effect of corrosive environment on drillpipe fatigue strength.

The stress concentration factors for the external upset are 1.19 and 1.13 for the axial tension and bending cases, respectively. Comparison of the stress concentration factors for the external upset vs. the internal-/external-upset drillpipe geometry show that the differences are minimal. Therefore, the superior performance that some drilling contractors associate with 3½-in. [98-mm] -OD (external-upset) drillpipe cannot be attributed to the upset geometry. Note, however, that the internal-upset drillpipe geometry appears to be difficult to control during manufacturing. Although 4½-in.-OD, 16.6-lbm/ft [114-mm-OD, 27.4-kg/m] drillpipe is by far the most widely used in the industry, the length of the internal upset only recently was standardized.²⁰

The improved fatigue performance observed in small-diameter drillpipe can be attributed directly to pipe diameter rather than upset configuration. For a given dogleg severity, the stress range for a small-diameter drillpipe as it rotates in the dogleg is lower than for larger-diameter drillpipe. The maximum stress in a 3½-in. [89-mm] -OD drillpipe body is equal to only 78% of the maximum bending stress in a 4½-in. [114-mm] -OD drillpipe body when both drillpipes are bent in doglegs of the same severity.

The fatigue crack in many ($\approx 57\%$ of the total) of the test specimens was started by light grinding marks on the outer pipe surface, which is ground to remove small superficial imperfections detected during the inspection phase of manufacturing. Two features are considered important when assessing the effect of grinding on drillpipe fatigue resistance: surface roughness and residual stresses in the surface layer. Experimental work by Suhr²¹ on specimens that were not under residual stress and that contained grinding marks showed that fatigue cracks almost invariably began at the root of the grooves introduced by grinding when these marks were oriented transverse to the direction of loading. Therefore, surfacing grinding was found to introduce a notch effect. Fractographic examination of our test specimens showed that fatigue cracks did begin at the root of transverse grooves left by grinding.

Surface grinding not only introduces surface roughness, but also modifies residual stresses in the surface layer.¹⁹ X-ray diffraction measurements of grinding marks showed that residual stresses varied from high tension to low compression. In contrast, residual stresses on unground surfaces were consistently compressive. Fig.

Fig. 6—Chart showing the effect of mean stress on fatigue strength of Grade E drillpipe.

Fig. 5—Graph showing residual stresses on the outer surface of a drillpipe.

5 shows the residual-stress distribution measured around the circumference of one section of drillpipe. Areas of grinding are identified in this figure. Residual stresses measured in the upset region of the drillpipe were found to be comparable with those measured in the drillpipe body. Ref. 19 gives details about measured residual stresses in different sections of drillpipe.

Fatigue cracks also began at surface pits (25% of the test specimens) present before or formed during testing in a corrosive environment, at surface notches (10%), and at surface slag inclusions (7%). Surface notches observed at the origin of fracture surfaces either were incurred accidentally, were introduced during manufacturing or during subsequent handling, or were placed there deliberately (i.e., numerals stamped on the drillpipe body).

Examination of the fracture surfaces of test specimens indicated that fatigue cracks propagate relatively quickly once the cracks have propagated through the wall thickness. This observation was substantiated by fracture-mechanics analysis, which indicated that more than 95% of total fatigue life is expended by propagating a surface crack to a through-thickness crack.¹⁹ Thus, by the time a wash-out is detected, most of the drillpipe fatigue life has been consumed.

Background: API Guidelines on Drillpipe Fatigue

Effect of Stress Range and Mean Stress in Noncorrosive Environments. Current guidelines for drillpipe fatigue evolved from the works of Lubinski⁷ and Hansford and Lubinski.^{8,9} These analytical studies used experimental fatigue data presented earlier by Bachman.⁶ Lubinski⁷ used a lower-boundary curve in the test data of Bachman to predict fatigue life. The experiments of Bachman were satisfactory for their intended use, namely testing of the tool-joint/drillpipe connection. However, these experiments were not performed in a corrosive environment or under axial tension. The S/N curve Lubinski⁷ and, later, Hansford and Lubinski⁸ used for their analytical work in a noncorrosive environment can be expressed as

$$\log N = 13.31 - 4.56 \log \sigma_r \quad (3)$$

Fig. 7—S/N curve showing the effect of stress range on drillpipe fatigue life in a noncorrosive environment.

Fig. 8—S/N curve showing the effect of stress range on drillpipe fatigue life in a corrosive environment.

The constant 13.31 becomes 17.13 when the stress range is expressed in megapascals. The endurance limit was set at 36 ksi [248 MPa] for drillpipe in a noncorrosive environment.⁷

Hansford and Lubinski used a modified Goodman diagram to account for the effect of the mean stress at the fatigue endurance level and a standard Goodman diagram at stress ranges above the endurance limit.⁷⁻⁹ The standard Goodman diagram takes the form of a straight line when the fatigue strength (stress range for a fixed number of loading cycles to failure) is plotted vs. the mean stress. The interaction equation can be expressed as

$$\sigma_f = \sigma_r [\sigma_u / (\sigma_u - \bar{\sigma})], \quad (4)$$

where σ_r is the stress range acting with mean stress $\bar{\sigma}$; σ_f , equivalent fully reversed fatigue-stress range (i.e., stress range under zero mean stress, which yields the same fatigue life as σ_r acting with $\bar{\sigma}$); and σ_u , tensile strength of the material. The fatigue life reduced by the presence of axial tension is obtained by use of the equivalent fully reversed stress range in the fatigue life equation (Eq. 3) for the fully reversed bending condition.

In a modified Goodman diagram used at the endurance level, Lubinski⁷ assumed a reduction of the endurance limit at a low mean stress to allow for slip marks on the drillpipe body (Fig. 6). This reduction corresponds to $\approx 18\%$ of the fatigue strength under zero mean stress. In addition, he assumed a cutoff on the mean stress level at 67 ksi [460 MPa]. At this cutoff, the endurance limit was assumed to be 11% larger than the fatigue strength predicted from the standard Goodman diagram. Lubinski then used a quadratic curve to interpolate between the point at zero mean stress and the point at a mean stress of 67 ksi [460 MPa]. Fig. 6 compares the standard and modified Goodman curves with the results of regression analysis on results of tests in air. Because the effect of mean stress was investigated experimentally to only 36 ksi [250 MPa], the U. of Alberta design curve in Fig. 6 should be used with caution for mean stresses > 36 ksi [> 250 MPa].

Fatigue curves for different mean stress levels can be obtained from Eq. 4. Fig. 7 compares the fatigue test U. of Alberta test results with the S/N curve derived from Eqs. 3 and 4 for the appropriate test conditions. Of the 20 test results presented in Fig. 7, eight are below the Lubinski S/N curve for a mean stress of 18 ksi [125 MPa]. This indicates that, despite the fact that the Lubinski⁷ fatigue curve represents a lower boundary to fatigue test results presented by Bachman⁶ and a further reduction of the endurance limit to account for the detrimental effect of slip marks, the Lubinski curve is not conservative compared with recent test results.

Effect of Stress Range and Mean Stress in a Corrosive Environment. When a well is drilled in a corrosive environment, a fatigue curve from tests in a noncorrosive environment cannot be used. To obtain a fatigue curve that is applicable to a severely corrosive environment, Lubinski⁷ lowered the fatigue curve from tests in a noncorrosive environment by 40% throughout. As a consequence, the

Fig. 9—Chart showing the effect of axial load on the maximum bending moment in a drillpipe.

endurance limit observed in a noncorrosive environment was also reduced by 40% for use in a corrosive environment. Lubinski recognized that this method of obtaining a fatigue curve for a corrosive environment was conservative for high stress ranges, where failure occurs early in fatigue life, and was not conservative for low stress ranges, where failure occurs at stress-range levels below the resulting endurance limit. Decreasing the fatigue stress in Eq. 3 by 40% results in the following expression for fatigue life in a corrosive environment:

$$\log N = 12.30 - 4.56 \log \sigma_r, \quad (5)$$

In SI metric units, the constant 12.20 is 16.12. Lubinski used a 22-ksi [150-MPa] endurance limit in a severely corrosive environment.

Lubinski assumed that the effect of mean stress on fatigue life in a corrosive environment was the same as its effect in a noncorrosive environment. Therefore, the Goodman diagram expressed by Eq. 4 was also used for fatigue-life prediction in a corrosive environment. Fig. 8 compares fatigue life predicted by Eq. 5 with fatigue life modified for the effect of a mean stress of 18 ksi [125 MPa] and with test results reported here in a corrosive environment. As expected, the Lubinski fatigue curve is conservative at high values of stress. However, at lower stress ranges, some data points lie below the fatigue curve, indicating that the curve on which current guidelines are based is not conservative for low stress ranges. Although this effect was expected when the work for the current guidelines was carried out, no provision was made to account for it.

To arrive at the API fatigue guidelines in their current form, it was necessary to evaluate the stresses in a drillpipe rotating in a dogleg with a superimposed axial tension corresponding to the weight of drillpipe below the dogleg. Stress cycling in the drillpipe takes place because of bending stresses that occur in the rotating drillpipe. As tension increases on the drillpipe, the middle portion of a pipe between two tool joints partially straightens and moves toward the borehole wall, thereby causing more bending near the tool joint. Under tension, the middle of the drillpipe may touch the borehole wall. From this point, the bending stresses are not expected to increase as rapidly with increasing tension. Lubinski's⁷ and Hansford and Lubinski's^{8,9} method to obtain bending stresses in a drillpipe under axial tension in a dogleg is briefly described in the following section.

Analytical Work of Lubinski

Lubinski⁷ considered a length of drillpipe loaded axially in a curved configuration. The configuration of the dogleg was assumed to be an arc of a circle (a gradual dogleg). To simulate the relatively large bending stiffness of the tool joint, the end rotation of the section of drillpipe at the tool joint was fully restrained. By considering equilibrium of a uniform section of drillpipe, Lubinski obtained

$$c_o = c_d (KL / \tanh KL), \quad (6)$$

where c_o is curvature of the drillpipe body adjacent to the support

Fig. 10—Chart showing predicted drillpipe fatigue damage in a gradual dogleg in a noncorrosive environment.

(near the tool joint); c_d , curvature of the dogleg; and L , half the length of the drillpipe. K is given by

$$K = \sqrt{T/EI}, \dots\dots\dots (7)$$

where T is tension force in the drillpipe; E , modulus of elasticity of the pipe material; and I , moment of inertia of the cross section of the drillpipe body. Once c_0 has been evaluated for a given c_d and T , the bending stress in the drillpipe body is obtained from⁷

$$\sigma_b = (c_0 E d / 2), \dots\dots\dots (8)$$

In his original work, Lubinski⁷ did not consider the effect of drillpipe-body/borehole-wall contact as the axial load is increased. This assumption, which is embodied in Figs. 6.4 and 6.5 of API RP7G,⁵ leads to overestimation of the bending stresses in the drillpipe body. Later work by Lubinski²² addressed the effect of drillpipe/hole contact on the calculated stresses. This work was found to be relevant only for Range 3 drillpipe; the Lubinski analysis did not predict contact for standard Range 2 drillpipe.

Eq. 6 represents a first-order analysis for the effect of tension force on drillpipe curvature. Because application of tension force to the drillpipe tends to straighten the pipe, the moment induced at the critical section near the tool joint will be overestimated if the initial eccentricity of the axial load is used to calculate the moment induced at the tool joint. A decrease in eccentricity (corresponding to the deflection at the middle of the drillpipe with respect to the tool joint) would reduce the calculated moment near the tool joint.

Because of the shortcomings identified in the Lubinski analysis, a more general analysis using the finite-element method has been performed on a section of drillpipe.¹⁹ The effect of increased stiffness in the upset and tool-joint area and of drillpipe contact with the hole wall and the nonlinear behavior resulting from the second-order effects in the curved pipe under tension were considered in the analysis. A section consisting of two lengths of drillpipe was analyzed. To simulate a gradual dogleg, the borehole wall was assumed to be rigid and the dogleg was assumed to be an arc of a circle. Ref. 19 presents details on the modeling and analysis of a section of drillpipe in a dogleg.

Analysis of a Range 2 drillpipe under tension was performed for dogleg severities of 5 and 10°/100 ft [5 and 10°/30 m], and the effect of axial load on bending moment adjacent to the tool joint was obtained. Fig. 9 presents moments calculated in the drillpipe body adjacent to the pipe upset for values of axial tension varying from 0 to 225 kip [0 to 1000 kN]. For a dogleg severity of 5°/100 ft. [5°/30 m], the drillpipe and borehole wall were not found to touch, given axial loading up to 225 kip [1000 kN]. However, for a dogleg severity of 10°/100 ft. [10°/30 m], the drillpipe and borehole wall touch under an axial load > 11.2 kip [>50 kN]. Fig. 9 also presents the response predicted by the first-order analysis presented by Lubinski.⁷ Under small axial loads, the Lubinski analysis underestimates the moment induced in a drillpipe. As the axial load increases,

Fig. 11—Chart showing predicted drillpipe fatigue damage in a gradual dogleg in a corrosive environment.

the moment predicted from the work of Lubinski exceeds the one predicted with the finite-element method. This results mainly from the second-order effect, which was ignored in the work of Lubinski.

Cumulative Fatigue Damage

During a fatigue test on a drillpipe, load cycles of constant stress range are applied to the test specimen until the cumulative effect of all cycles produces failure (unless the test is conducted in air at a stress range below the endurance limit). When a test is conducted under variable amplitude loading (various stress ranges), a condition that prevails in the field as a drillpipe rotates through doglegs of various severities, the cumulative effect of these events may also lead to fatigue failure. Fatigue effects of loading events other than constant amplitude are called "cumulative damage." The Palmgren-Miner rule is the simplest rule used to predict cumulative fatigue.²³ Despite this rule's inability to account for loading sequence, it is used extensively, mostly because it is simple and because more-sophisticated rules have not proved to be generally applicable to all types of service loading. With the Palmgren-Miner rule, fatigue damage sustained under one cycle of loading can be expressed as

$$D_f = (1/N), \dots\dots\dots (9)$$

where N is fatigue life under constant amplitude load cycling. Fatigue failure is expected to take place when the total damage sustained represents 100% of the fatigue life.

To develop equations for the allowable stress range, the mean fatigue curves of Eqs. 1 and 2 minus two standard deviations are used. The two standard deviations below the mean regression line approximately represent the lower 95% confidence limit of the test data and result in "safe" design curves with the same probability of failure. The resulting design equations become

$$\log N = 10.9 - 3.46 \log \sigma_r - 7.94 \times 10^{-4} (\bar{\sigma})^2 \dots\dots\dots (10)$$

for a noncorrosive environment. (In SI metric units, constants 10.9 and 7.94×10^{-4} are 13.8 and 1.67×10^{-5} , respectively). The design for a corrosive environment can be expressed as

$$\log N = 9.33 - 2.35 \log \sigma_r - 2.63 \times 10^{-4} (\bar{\sigma})^2 \dots\dots\dots (11)$$

In SI metric units, constants 9.33 and 2.63×10^{-4} are 11.3 and 5.54×10^{-6} , respectively. A comparison of Eqs. 10 and 11 indicates that safe fatigue life is higher in a corrosive than in a noncorrosive environment. This apparent paradox is only an artifact arising from the large scatter observed in the test results obtained in air and from the necessity of keeping the same level of safety in both corrosive and noncorrosive environments.

Once fatigue life for a given stress range and mean stress is evaluated from test results, the percent fatigue life expended for a number of stress cycles can be evaluated from the Palmgren-Miner rule. To derive the present cumulative fatigue guidelines,⁷ Hansford and Lubinski⁹ assumed a rotary speed of 100 rev/min and a drilling rate of 9.8 ft/hr [3 m/h], which means that 20,000 cycles of loading are

put on a drillpipe as 32.8 ft [10 m] of hole is drilled. In this paper, we retain these assumptions with respect to drilling conditions so that the cumulative fatigue curves derived by the method presented can be compared directly with existing guidelines.

With the procedure outlined herein, cumulative fatigue curves were obtained for drilling in air and in a corrosive environment. Fig. 10 presents cumulative fatigue curves for drilling in a noncorrosive environment, whereas Fig. 11 is applicable to a corrosive environment. In both figures, curves are presented for dogleg severities of 5 and 10°/100 ft [5 and 10°/30 m]. The curves corresponding to API design guidelines are also shown for comparison. Note from Fig. 10 that the API design guidelines for cumulative fatigue damage in a noncorrosive environment do not provide conservative predictions for the drillpipes we tested. Also note, however, that for a dogleg severity of 10°/100 ft [10°/30 m], the two design curves presented in Fig. 10 are in better agreement than the curves for a dogleg severity of 5°/100 ft [5°/30 m]. This difference in behavior between the two dogleg severities is a result of neglecting the contact between the drillpipe and the borehole in the Lubinski analysis, which leads to more-conservative predictions of bending stresses in the drillpipe, thus compensating for the less-conservative S/N curve used to predict fatigue life.

In contrast, Fig. 11, for fatigue in a corrosive environment, indicates that the API cumulative fatigue design curves for drilling in a corrosive environment are more conservative than the predictions resulting from our work. Again, however, the API curves give an unconservative estimate of fatigue damage at low axial load, probably as a result of the lower drillpipe bending stresses calculated at low levels of axial load by use of the Lubinski analysis.

Conclusions and Recommendations

The effect of stress range and mean stress were both found to be significant to drillpipe fatigue life.

Grinding the drillpipe surface during inspection can shorten the drillpipe fatigue life. Of 29 specimens tested in air, 13 failed because a fatigue crack developed at grinding marks. Of 27 specimens tested in a corrosive environment, 13 failed because cracks developed at grinding marks. Fractographic examinations showed evidence of a notch effect at the grinding marks. In addition, residual-stress measurements using X-ray diffraction showed that grinding removes beneficial compressive surface stresses. Therefore, grinding should be kept to a minimum during inspection, especially near the upset region, where bending stresses are high owing to the secondary effect of axial tension on a curved drillpipe. Stress-relief heat treatment of the drillpipe after the inspection stage would reduce the high-tensile residual stresses sometimes introduced by grinding.

We found that only a small portion of the total fatigue life remains once a surface crack propagates through the pipe-wall thickness. Therefore, we recommend replacing drillpipe as soon as a washout is detected because twistoff is likely to occur shortly thereafter. This practice is consistent with current drilling practices.

The effect of upset geometry was found to be minimal in external-upset compared with an internal-/external-upset drillpipe. Because the internal protective coating in the internal-upset region has been found to be damaged in specimens that failed in the field, we recommend use of external-upset drillpipes with a flush inner wall when available. Geometry of pipes with flush inner walls is less susceptible to damage by tools run down the drillpipe.

The superior performance of 3½-in. [98-mm] -OD (external-upset) drillpipe observed by some drilling contractors is likely to be a direct consequence of small drillpipe diameter. For a given dogleg severity, the stress range of a drillpipe as it rotates in the dogleg is directly proportional to pipe diameter. For this reason, smaller diameter drillpipes should be used when practical. However, to prevent rupture of drillpipe before a leak is detected, drillpipe wall thickness should be kept to a minimum. Therefore, to deter fatigue, drillpipe design must represent a compromise between a small diameter to reduce the stress range and a thin wall to allow for leak before break.

Because an endurance limit does not exist in the corrosive environments common in the field, use of the API recommended dogleg severity below which no fatigue damage is sustained should be dis-

continued. In a corrosive environment, fatigue damage occurs for any dogleg severity. Therefore, fatigue should be controlled through cumulative damage calculations. Because field environments are generally corrosive, cumulative fatigue curves for a corrosive environment should be used.

Existing cumulative fatigue curves for noncorrosive environments are unconservative compared with results we obtained. However, for a corrosive environment, the existing API cumulative fatigue curves were found to be generally very conservative.

Current knowledge of corrosion fatigue is insufficient to allow prediction of fatigue life in a given corrosive environment with results from tests in a different environment. It is premature to replace existing API guidelines for cumulative fatigue damage. Further research is needed to determine the effects of a wider variety of corrosive environments, the detrimental effect of slip marks on the pipe body, and applicability of Eq. 9 for cumulative damage calculations.

Nomenclature

- c_d = dogleg curvature, L^{-1} , in.⁻¹ [mm⁻¹]
- c_o = drillpipe curvature, L^{-1} , in.⁻¹ [mm⁻¹]
- d = diameter, L, in. [mm]
- D_f = fatigue damage
- E = Young's modulus of elasticity, m/Lt², ksi [mPa]
- I = moment of inertia, L⁴, in.⁴ [mm⁴]
- K = constant given by Eq. 7
- L = half length of drillpipe, L, in. [mm]
- N = number of cycles of loading (fatigue life)
- T = axial load, kips [N]
- σ_b = bending stress in drillpipe body, m/Lt², ksi [mPa]
- σ_f = equivalent fully-reversed stress range, m/Lt², ksi [mPa]
- σ_r = stress range, ksi [MPa]
- $\bar{\sigma}$ = mean stress, m/Lt², ksi [MPa]
- σ_u = tensile strength of material, m/Lt², ksi [MPa]

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SI Metric Conversion Factors

ft × 3.048*	E-01 = m
in. × 2.54*	E+01 = mm
in. ⁴ × 4.162 314	E-07 = m ⁴
kip × 4.448 222	E+03 = N
ksi × 6.894 757	E+03 = kPa

*Conversion factor is exact

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