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On The Fatigue Life Prediction of Die-Marked Drillpipes

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Abstract

Drillpipe fatigue damage occurs under cyclic loading conditions due to, for instance, rotation in a dogleg region. Usually, failure mechanisms develop in the transition region of the tool joints and the die-marks due to gripping systems intensify it. In this paper two approaches are presented to evaluate damage in drillpipes; FEM and Cox Regression Model.

First, Finite Element Method is used to evaluate cumulative effects of fatigue damage in a number of drilling events with respect to rotating cyclic bending and constant tension and internal pressure in a G-105 drillpipe. The results show that how die-marks or other surface crushes can reduce the fatigue life of the pipe. The presented graphs can be easily used to determine the allowable length of a G-105 drillpipe below dogleg that consumes the fatigue life of the pipe section.

In the second approach, as a case study, the Cox Regression Model, a broadly applicable and the most widely used method of survival analysis is used to evaluate the distribution of survival times for the failure data of the southern oilfields of Iran. The resultant cumulative survival and hazard functions can reliably predict the time of failure and assist the engineers to evaluate cumulative damage.

Keywords: Drillpipe; Die-mark; Cumulative Fatigue; Finite Element Method; Stress Concentration; Cox Regression.

1. Introduction

Failure due to fatigue is a very costly problem in oil and gas industry. Many investigators have previously addressed this problem, but its frequency of occurrence is still excessive. Torque and tension can be correctly predicted but computations of fatigue duration are still approximate.

Fatigue is by far the most common cause of drill stem failure. It can occur at stress levels far below normal operating stress in most drill stem components. Fatigue is a complex mechanism that is affected by many factors, such as drilling environment, dogleg severity, axial loads and etc. Fatigue damage accumulates over extended use, so sudden failure may occur any time the drill string is under load. This is one of the most insidious aspects of fatigue.

Drillpipe is subjected to cyclic stresses in tension, compression, torsion and bending. Tension and bending are the most critical of these. Bending and rotation produce an alternation between states of tension at localized points in the drillpipe, such as tool joints and areas near each upset. The major factor in drillpipe fatigue is cyclic bending when pipe is rotated in a hole that has a change in direction (a dogleg).

Hill has analyzed 76 drillstring failures from 1987 to 1990 on three continents [1]. These incidents are costly because of the loss of rig time, tubular goods and even some time losing the well.

Failure causes can be estimated as follows:

- Fatigue is the main cause in 65-percent of the failures and has a significant impact in 12-

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percent.

- Combined excessive tension and torque give failures in 13-percent of the cases.
- Low toughness of material is mentioned for only 8-percent of the failures.

Lubinski [2] has defined curves where the permissible dogleg severity, below which no fatigue damage of drillpipes may occur, can be estimated from the tensile load and the drillpipes characteristics. These curves used to prevent static failure are the basis of the “API-RP-7G” [3].
According to Palmgren-Miner’s rule the total damage is the accumulation of damage occurred during a cycle, regardless of pipe's previous load history. However, it is known that the sequence of cyclic loading could have an effect on the damage accumulation [4].
Rahman [5] recommends SCFs figures of 1.07, 1.15 and 1.6 respectively for die-marks depth of 0.0004” (0.0102 mm), 0.0012” (0.0305 mm) and 0.01” (0.254 mm) for a 3.65 inch RSA-6K drillpipe. Therefore, dies of gripping system which mark the pipe surface during making and breaking operation should be minimized as they cause stress concentrations.
Tafreshi and Dover [6] have carried out finite element analyses on several standard tool-joints. SCF figures are given for standard and modified NC46 connections. Maximum SCF may vary within a range of 3.29 to 8.56, depending on the pin or box thread profile.

2. Causes of failure in drill string

The main causes of failure in drill string are:

2.1 Critical Rotary Speed

Rotating equipment has a critical speed which varies with changes in the location of the centre of gravity, mass, alignment between the axis of rotation and gravitational force, and rotational speed. Critical speed problems include bent pipe, bottom hole assembly (BHA) connection failure, fatigue failure, washouts, and severe outer diameter (OD) wear of tool joints and tube sections.
The critical speeds are proportional with length and weight of the drillpipes and drill collars and also to Bottom Hole Assembly (BHA).

2.2 Excessive Tension

When a drill string gets stuck during drilling, operational procedures are applied to release the drill string. These procedures include working the drill string up or down, attempting to rotate the string and pumping mud through the drill bit to aid pipe release. When pulling the string out, by increasing the tension stress from the yield stress of the pipe material, the weakest section of the string or the smaller one become necking. Continuing this situation, will yields to breaking the string from that section.
This type of failure mainly cause in the upper parts of the string that is usually in tension and also bear the tension due to the weight of the string.

2.3 Fatigue Failure

Fatigue is by far the most common cause of drill stem failure, often taking place in surface notches such as slip cuts, metal tears caused by the pipe turning in the slips, or deep corrosion pits on the pipe internal diameter. It can occur at stress levels far below normal operating stress in most drill stem components.
Two results of fatigue failure are the washout and the twist-off. A washout is a place where a small opening has occurred in the pipe, usually the result of a fatigue crack penetrating the pipe wall, and drilling fluid has been forced through it. Fluid abrasion erodes the metal and enlarges and rounds off the edge of the hole. A twist off is usually caused by a fatigue crack extending around the pipe and causing the pipe to break.
Several studies confirmed that washouts occurred near the end of the Miu, closest to the tube body, the most highly stressed area of the drillpipe during drilling and the most exposed to fatigue failure [5].
Fatigue, by definition is the phenomenon in which a repetitively loaded structure fractures at a load level less than its ultimate static strength. For instance, a steel bar might successfully resist a single static application of a 300 KN tensile load, but might fail after 1,000,000 repetitions of a 200 KN load.

3. Fatigue Failure

Failure due to fatigue in drillpipes can be categorized in three groups:

- ***Pure fatigue:*** break with no previous visible cause,
- ***Corrosion fatigue:*** break due to a pitting in a corrosive environment,
- ***Notch fatigue:*** break because of a mechanical cut or notch.

3.1 Notch Fatigue

Surface imperfections, either mechanical (such as a notch) or metallurgical, greatly affect the fatigue limit. Aside from the initial distortion of the steel grain structure, a notch concentrates the stresses and speeds the breakdown of the metal structure. If a notch is within 20 inches of a tool joint, where maximum bending takes place, it can form the nucleus of an early fatigue break [7]. A longitudinal notch is less harmful than a circumferential (transverse), which leads to failure.
Some of various surface scratches that can cause

drillpipe notch failure are

- Slip marks, cuts and scratches
- Tong marks
- Spinning chain marks
- Downhole notching by formation and junk cuts

Figure 1: Slip is used to hanging the string

Tong and slip cuts are probably the worst-looking defects produced on drillpipe in the field. They are long, deep, and frequently sharp notches. Slips with worn, mismatched, or incorrectly installed gripping elements may allow one or two teeth to catch the full load, causing a deep notch and potential failure. The practice of rotating drillpipe with the slips can produce a dangerous transverse notch if the pipe turns in the slips.

Because of the logarithmic relationship between stress and fatigue life, even small increases in stress will dramatically reduce fatigue life. For example, when we read that a slip cut can increase stress from 30,000 psi on a smooth pipe to 50,000 psi at the base of the cut, the stress increase may not seem significant on S-135 drillpipe but with the pipe in seawater, fatigue life would be reduced by about 85% (see Fig. 2) [8].

4. Static Analysis

The pipe section at the dogleg region may fail under the combined effects of axial tensile stress, radial pressure, torsional stress and alternating repeated bending stress due to rotation. Failure at this section in static mode can be assessed by the method of Von Mises [9] for equivalent yielding under combined loads.

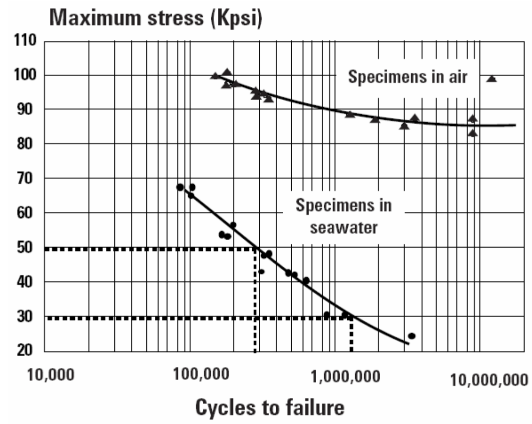

Figure 2: S-N Curve for Grade S-135 drillpipe in air and in seawater

In this paper, first, a typical slip mark is modeled in a quarter of a 5 inch G-105 drillpipe body to see how a slip crush can raise the stress. The Von-Mises equivalent stress for the 5 inch drillpipe that is loaded with a tensile axial force of 200000 lbf were obtained using ANSYS program by considering different depths of die-mark.

Then, by using an axisymmetric model, SCFs due to change in cross section area and also the stresses due to bending loading and axial loading are calculated. These stresses are then used in fatigue calculations (section 5).

4.1 Die-Mark Effects

A quarter of the pipe is modeled in ANSYS because of the symmetry condition in drillpipe. Specifications and mechanical properties of the pipe is as below,

Grade	G-105
Modulus of Elasticity	E=30×106 (psi)
Ultimate Tensile Strength	σ_u=125000psi
Yield Strength	σ_y=110000psi
Poisson's Ratio	υ = 0.3
Section Area Body of Pipe	A=5.2746 (Sq. in)
Solar Sectional Modulus	Z=11.415 (Cu. In)
Internal Diameter	ID = 4.276 (in)
External Diameter	OD = 5 (in)
Internal Pressure	Pi = 2500 (psi)

The stress concentration factor (SCF) is then defined as the ratio of the stress at the notch root to the applied stress. As it is shown in figure 4, it is ranging between 1 and 2.8 for a die-mark depth ranging between 0.1 and 1.8mm.

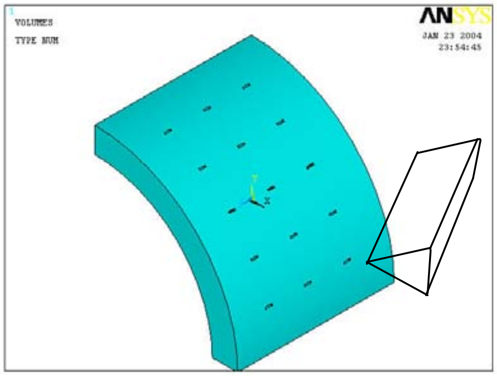

Figure 3: 3D Model in ANSYS with die-mark

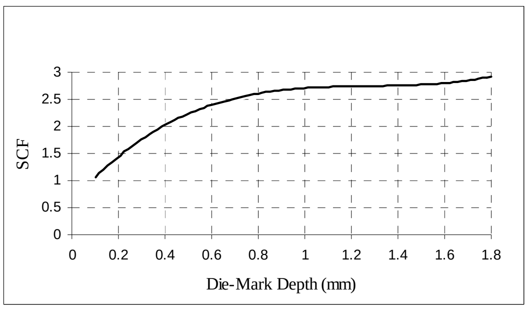

Figure 4: SCFs for Die-Mark Depth ranging from 0.1 to 1.8 mm

4.2 SCFs due to change in cross section and stresses due to bending conditions

The stress concentration factors due to change in cross section area and axial / bending load is also calculated using an axisymmetric model (see Fig. 5). Models of axisymmetric 3-D structures may be represented in equivalent 2-D form. Results from a 2-D axisymmetric analysis will be more accurate than those from an equivalent 3-D analysis [10]. Also, use an axisymmetric model is greatly reduces the time of analysis.

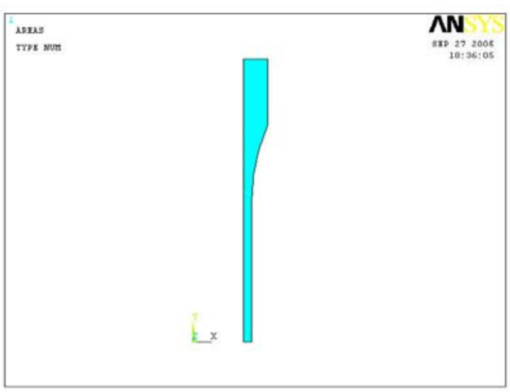

Figure 5: axisymmetric model of end section

A portion of a 12 inch long of a 5 inch drillpipe is modeled. Plane83 element is used in this study. It is used for 2-D modeling of axisymmetric structures with nonaxisymmetric loading. Examples of such loadings are bending, shear, or torsion. The element has three degrees of freedom per node: translations in the nodal x, y, and z directions.

The portion of the drillpipe in slips area is modeled as it tolerated the maximum bending loads in a curved section of a well (dogleg region) [2]. The symmetry boundary condition is applied on the upper part of the model and the bending condition is applied on the lower part as displacement boundary conditions. Also the axial force due the weight of the hanged drillstring is applied on the lower part of the model. The slip crushes are not considered in this section as they calculated from the section 4.1.

4.2.1 Axial Loading

In the axisymmetric model, with applying axial load, it has been shown that in transition area, we have the maximum stress and so the SCF due to geometry in obtained 1.128.

4.2.2 Bending Loading

Bending load was applied on the model as displacement boundary condition. In this situation, the SCF is obtained equal to 1.11

5. Fatigue Life Duration

One of the most common questions about fatigue is "How long should a component be expected to last?" The question is usually asked for the purpose of setting inspection intervals.

5.1 The Fatigue Mechanism

When a component goes into service, fatigue damage begins accumulating in small increments during cyclic loading episodes. The damage is cumulative and irreversible. The most common causes of cyclic loading in drill strings are rotating the string when some part of it is bent or buckled, and from vibration. After a period of damage, a fatigue crack forms. As stress cycles continue, the crack grows until it penetrates the wall and the part leaks, or until the remaining sound material in the part can no longer carry applied loads and it breaks. Thus, fatigue failures do not often result from events immediately preceding the failure, but from the cumulative effect of small increments of damage over long periods of past use. Because of this, fatigue failures often arrive unexpectedly, without any change in load or drilling conditions that might forewarn of a problem.

Most of the fatigue life of a drill string component will have been used up by the time a crack has formed and grown large enough to be detected by inspection. So a fatigue crack, once detected, is cause for immediate rejection of the component

The main factors that contribute to fatigue failures include [10]:

- Number of load cycles experienced
- Range of stress experienced in each load cycle
- Mean stress experienced in each load cycle
- Presence of local stress concentrations

5.2 Combined stresses of tension and bending

When a drillpipe is in pure bending, for example when it is held in a bend while we rotate it, we find that every fiber of the pipe is stretched in tension, then compressed, then stretched again. In this case, we see that each fiber is stressed alternatively in tension and in compression. If we add a high tensile pull to the pipe (due to the weight of the drill string), we will find that the stress may vary from maximum tension, but never to compression. The addition of this continuing tensile stress reduces the ability of the pipe to withstand the cyclic stress.

In this situation, we are not allowed to use the typical S-N curves. Because they are obtained for pure bending condition, and didn't consider the effect of axial tension due to the weight of the hanging drillstring and also the effects of stress concentration factors due to die-marks.

The repeated maximum tensile bending stress in typical S-N curve is then replaced by σ_{ad} which is called adjusted stress and takes the above effects into account.

The adjusted stress is calculated with the following formula [5],

$$\sigma_{ad} = \frac{\sigma_{bc}.\sigma_u}{\sigma_u - \sigma_{ac}} \qquad (1)$$

σ_{bc} and σ_{ac} are the bending and tensile stresses respectively, considering SCF, and σ_u is the ultimate tensile strength of the pipe material.

The percentage of fatigue damage of each joint of the drillpipe due to drilling through a dogleg region is calculated by:

$$D_i = \frac{n_i}{N} \times 100 \quad [\%] \qquad (2)$$

Where n_i is the number of drillpipe revolutions in a dogleg and N is the total number of cycles at a normal stress value that can cause fatigue failure.

The drillpipe revolution (n_i) can be related to actual drilling data and is calculated from the following formula [5]:

$$n_i = \frac{60Rd}{v} \qquad (3)$$

Where R is the drillstring rotational speed in rpm, d is the length of the dogleg interval and υ is the drilling rate (ft/h).

According to Miner rule, when a particular drillpipe passes through several doglegs with several severities, the cumulative damage can be evaluated as follow,

$$D_m = \sum_{i=1}^{m} D_i \qquad (4)$$

where m is the total number of doglegs in a particular drilling event.

The Miner rule can be extended to a number of drilling events to estimate the fatigue damage of a drillpipe:

$$D_{mn} = \sum_{j=1}^{n}\sum_{i=1}^{m} D_{ij} \qquad (5)$$

Where n is the total number of drilling events to be considered.

The fatigue damages of the die-marked drillpipes for different length of drillstring below dogleg, different dogleg severity values and different stress concentration factors are presented in figures 6 to 13. Such a length is considered the one that consumes 100% the fatigue life of the pipe section.

5.3 Fatigue Graphs

The ANSYS fatigue calculations rely on the ASME Boiler and Pressure Vessel Code, Section III (and Section VIII, Division 2) [13], for guidelines on range counting, simplified elastic-plastic adaptations, and cumulative fatigue summation by Miner's rule.

To analyze fatigue with ANSYS, the following data must be defined [10]:

- Maximum number of locations, events, and loadings
- Fatigue material properties
- Stress locations and stress concentration factors (SCFs)

Fatigue damage curves are presented in this section, considering dogleg severity (DLS) between 1 to 5 degrees per 100 feet. These graphs can be easily used to determine the allowable length of a G-105 drillpipe below dogleg that consumes 100% the fatigue life of the pipe section.

The drilling parameters are considered to be, R=100 rot/min, d=30ft (9.14m), υ=20ft/hr (6m/hr) throughout this study.

As it is obvious from the graphs, by increasing DLS from 1 to 5, the allowable length below dogleg will decrease. Also, we see that the current approach of fatigue analysis using smooth pipe surface is not reliable and doesn't predict fatigue failures with sufficient accuracy.

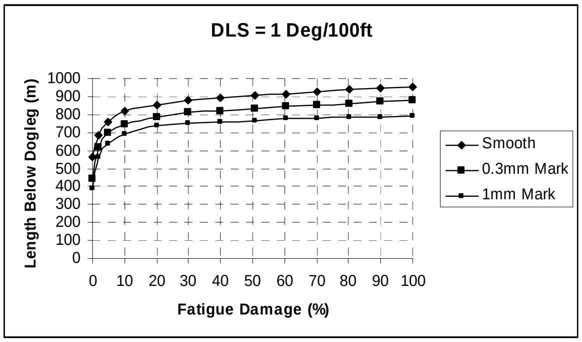

Figure 6: Fatigue Life curves, comparison between smooth pipe, 0.3 mm and 1 mm die-mark depth for DLS = 1 deg/100ft

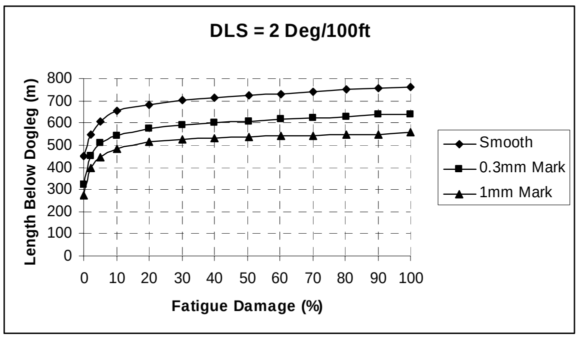

Figure 7: Fatigue Life curves, comparison between smooth pipe, 0.3 mm and 1 mm die-mark depth for DLS = 2 deg/100ft

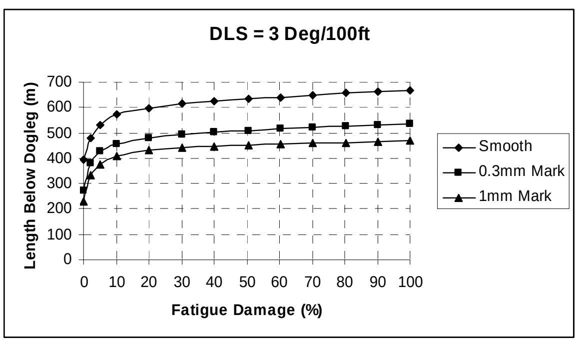

Figure 8: Fatigue Life curves, comparison between smooth pipe, 0.3 mm and 1 mm die-mark depth for DLS = 3 deg/100ft

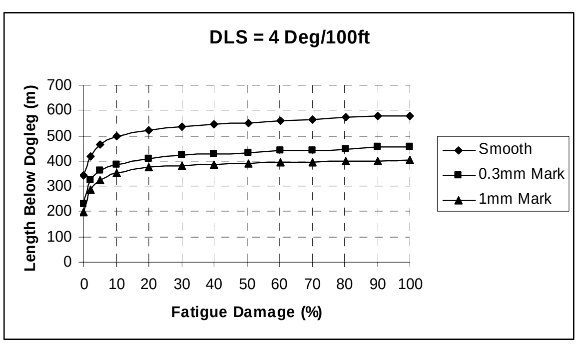

Figure 9: Fatigue Life curves, comparison between smooth pipe, 0.3 mm and 1 mm die-mark depth for DLS = 4 deg/100ft

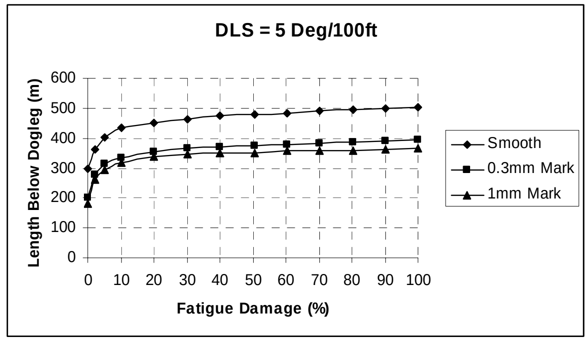

Figure 10: Fatigue Life curves, comparison between smooth pipe, 0.3 mm and 1 mm die-mark depth for DLS = 5 deg/100ft

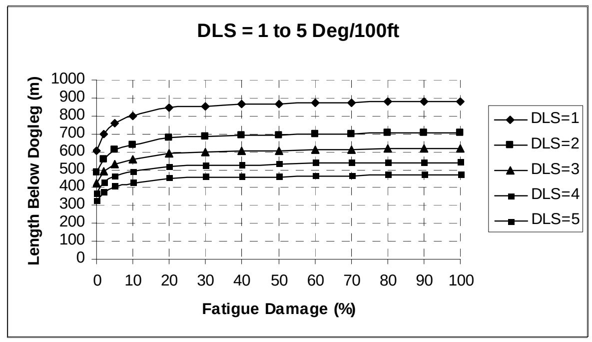

Figure 11: Fatigue curves for DLS ranging 1 to 5 deg/100ft for smooth drillpipe

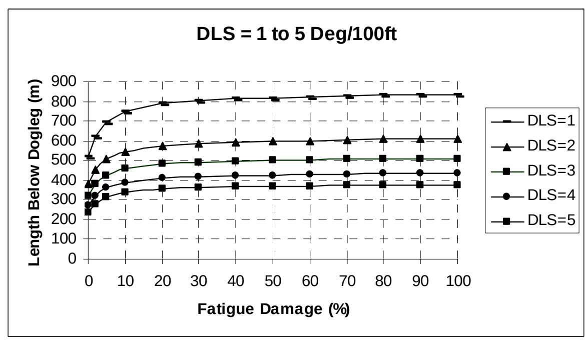

Figure 12: Fatigue curves for DLS ranging 1 to 5 deg/100ft for 0.3 mm die-marked drillpipe

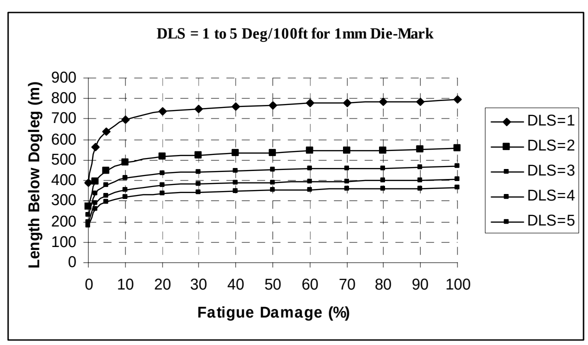

Figure 13: Fatigue curves for DLS ranging 1 to 5 deg/100ft for 1 mm die-marked drillpipe

6. Cox Regression Model

Survival analysis examines and models the time it takes for events to occur.
The Cox regression model is a standard tool in survival analysis for studying the dependence of a hazard rate on covariates (parametrically) and time (nonparametrically).
The purpose of the model is to simultaneously explore the effects of several variables on survival. This model is a well-recognized statistical technique for analyzing survival data [12].

6.1 Introduction

Survival analysis focuses on the distribution of survival times. Although there are well known methods for estimating unconditional survival distributions, most interesting survival modeling examines the relationship between survival and one or more predictors, usually termed covariates in the survival-analysis literature.
The subject of this section is to use of the Cox proportional-hazards regression model (introduced in a seminar paper by Cox [11]), a broadly applicable and the most widely used method of survival analysis, to analyze the survival data of the failures in the wells of Southern Oilfields of Iran.

6.2 Basic Concepts and Notation

The Cox-regression model is formulated through the hazard rate

$$\lambda_i(t)dt = P(T_i \in [t, t+dt] \mid safe\ at\ time\ t)$$
$$= P(failed\ at[t, t+dt] \mid safe\ at\ time\ t) \quad (6)$$

Now the Cox model assumes that the hazard for the ith component is:

$$\lambda_i(t) = h_0(t)\exp(\beta_1 X_{i1} + ... + \beta_p X_{ip}) \quad (7)$$

Note that

$$\log(\lambda_i(t)) = \log(h_0(t)) + \beta_1 X_{i1} + ... + \beta_p X_{ip} \quad (8)$$

Very similar to the models used for Poisson regression, the difference is that the baseline here is a non-parametric function.

Hazard function

$$\lambda_i(t)dt = P(T_i \in [t, t+dt] \mid safe\ at\ time\ t) \quad (9)$$

Cox model:

$$\lambda_i(t) = \lambda_0(t)\exp(\beta_1 X_{i1} + ... + \beta_p X_{ip}) \quad (10)$$

where $\lambda_0(t)$ is baseline hazard for a subject with covariates 0.
The regression coefficients β_1,... β_p represent the effects of the covariates. β_1 is the effect of X_{i1} when we have corrected for the other covariates.
β_1 may be interpreted in terms of the relative risk when the covariate X_{i1} is increased 1:

$$\frac{\lambda_0(t)\exp(\beta_1(X_{i1}+1) + ... + \beta_p X_{ip})}{\lambda_0(t)\exp(\beta_1 X_{i1}) + ... + \beta_p X_{ip})} = \exp(\beta_1) \quad (11)$$

If $\beta_1 > 0$ the risk of failure increases as X_{i1} increases, and if $\beta_1 < 0$ the risk of failure decreases as X_{i1} increases.
The quantity

$$\widehat{\beta}_1 X_{i1} + ... + \widehat{\beta}_p X_{ip} \quad (12)$$

is called the prognostic index for the ith subject.

This part of analysis is done, using SPSS 13.0 (2004) software package. In this analysis, 186 data were analyzed that 108 of them were the cases that experienced failure and the rest of them (78 data) considered as censored data (it means no failure until the end of study).
Seven parameters (including Neutral point of the drillstring; rotating speed; formation; bit size; weight on bit; mud weight and PH of drilling mud) were analyzed with this model to investigate if each of them has influence on the time of failure.
The model showed that only three parameter including ***weight on bit, rotating speed of the drillstring*** **and** ***neutral point in drillstring***; from the seven considered parameters are affecting the time of failure and based on this three parameters, it predicts the survival and hazard functions (see Fig. 14, 15).

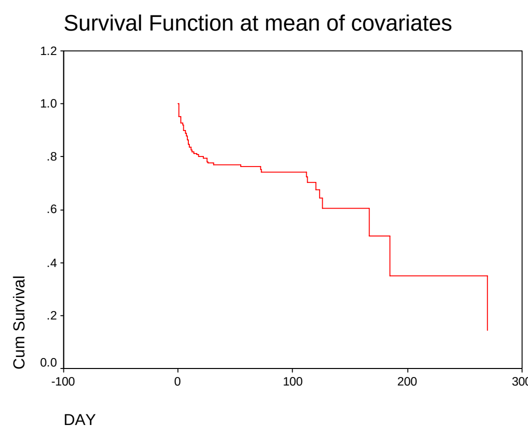

Figure 14: Survival Function Curve

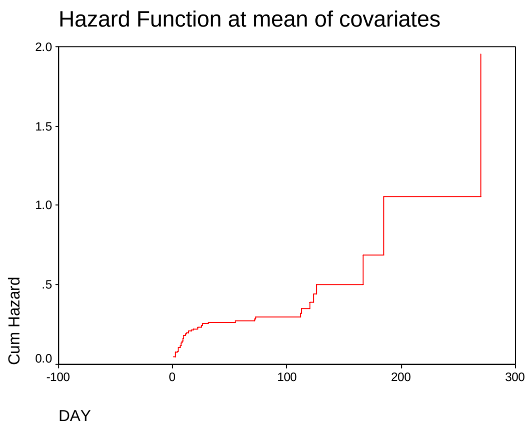

Figure 15: Hazard Function Curve

7. Conclusions

Based on the results, in can be concluded that:

1. The analysis showed that end of the internal upset taper, is the most likely area for a washout to occur.
2. The drillpipe's fatigue life depends primary on the shape of the internal upset (Miu), the degree of hole deviation (bending stresses), corrosion (metal loss) of critical high stress areas in the drillpipe, and mechanical damage to the tube upset area.
3. The stress concentration factors, versus die-mark depth were analytically determined. The SCFs was ranging 1 to 2.8 for die-mark depth between 0 and 1.8mm (0.07 inch).
4. The fatigue life of die-marked drillpipe that passing through a dogleg region was calculated, using S-N curve with equivalent alternating stress to take into account the effects of SCFs and axial tension load.
5. The results are presented graphically for G-105 drillpipes to easy calculate the allowable length of a G-105 drillpipe below dogleg that spends 100% the fatigue life of the pipe section.
6. This method of analysis could be extended to drillpipes with desirable grades and sizes.

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