USING THE API SPECIFICATION 11C AND THE STRESS RANGE DIAGRAM

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FATIGUE LIFE

The fatigue life of a reinforced plastic sucker rod is controlled by minimum stress, the stress range, and the operating temperature. The diagram, Figure 1, is part of API Specification 11C. For any minimum stress found on line O-I, a peak allowable stress is found directly above on line A-P. If a well is operating at 100 percent of this allowable range with an operating temperature of 160°F, it should cycle 7,500,000 times before expected first failure. Column B and Column D allow the stress range to be modified for different life expectancies and different operating temperatures. Figure 2 is the stress range diagram for Fiberflex fiberglass sucker rods.

Two different parts of the rod must be considered when estimating fatigue life: the rod body and the end connector system.

ROD BODY FAILURES

The rod body is a plastic, usually reinforced by unidirectional fiberglass filaments. As the rod body fatigues, the molecular bonding forces between filaments fail, causing uneven loading in the glass reinforcements. This phenomenon causes progressive failures in the fiberglass reinforcements, resulting in catastrophic failure of the rod body. Usually, in a well where no abnormal loading has occurred, the second and third failures will follow within 90 days. Abnormal conditions such as fluid pound, improper spacing, or similar problems cause localized failures. These failures cease when the problems are corrected. Nicks, cuts, and other types of mechanical rod damage will greatly reduce rod life. When a failure occurs, examine the break carefully, determine the point of origin and try to determine the cause.

END CONNECTOR FAILURES

Several types of failures can be located in the end connector: These are pin failures, adhesive failures, and pinch-off type failures.

Pin failures in glass are very similar to pin failures in steel; usually these failures are caused by improper make-up or contaminated threads. These types of failures can be easily eliminated by cleaning, lubricating, and correctly prestressing the pin-box connections. It is important to apply the correct torque during make-up; using displacement gauges is the best technique. Over-torqueing will distort the threads and shoulder faces of the pin and coupling. Under-torqueing allows the pin and coupling shoulder faces to separate under tension loads. The resulting failures are fatique failures in the pin thread root, the stress relief or unscrewing of the pin and coupling connection. Table 1 shows the ratio between the area of the rod body and the pin stress relief area for both steel and glass. By multiplying the peak stress in the rod body by the appropriate ratio, the minimum required prestress in the pin is determined. If Section A3 of API Specification 11C is adhered to, a large majority of this type of failure will be eliminated.

Adhesive failures, where the adhesive fails to properly bond to the rod, are usually caused by contaminated rods, improper preparation of the rod, or contaminated adhesive. Another type of adhesive failure is caused by voids in the adhesive. It has been the author's experience that these adhesive failures are usually defects in material or workmanship and are covered by the manufacturer's warranty.

The pinch-off type failure can be caused by several things but the main cause is the design of the internal wedge system of the end fitting. Pinch-off failures result from the rod being crushed by a high concentration of forces perpendicular to the long axis of the rod, shearing the rod body. Another cause of pinch-off failures is the rod body's low compressive strength or the loss of compressive strength due to elevated temperatures. Operating at high peak loads may also cause this type of failure. Internal cracks may develop in the rod body at the wedge system. These cracks will propagate down the rod body, resulting in ultimate failure of the rod body. This type of failure can be reduced by lowering the peak load or reducing the load range. It is the author's opinion that fatigue failures should occur in the rod body and not in the connector system.

OPERATIONAL FAILURES

Operational failures other than fatigue are usually preventable failures. These failures can happen because of improper design techniques, incorrect spacing, fluid pounding, and installing pumps that are too short. Fiberglass rod designs should be done for a minimum of two conditions: maximum anticipated fluid level or maximum anticipated pump intake pressure and the pumped-off case. The first case, with the high fluid level, gives the greatest downhole pump stroke. This case is used to calculate pump length. The following formula is used to calculate the minimum pump length:

$$MPL = DHPS + PL + 60$$

MPL = Minimum pump length in inches
DHPS = Downhole pump stroke in inches
PL = Plunger length in inches

The 60 inches is for clearance -- usually 30 inches on top and bottom.

If the fiberglass rod design is only done for the second condition, pumped off, the downhole pump stroke will be at a minimum and the pump will be too short. Pumps that are short will tag top and bottom, damage the pump, corkscrew the steel rods, and cause compression failures in the glass rods. Table 2 shows the effect of a high fluid level on a 7500-foot well.

Another problem can be improper pump bore selection. Selecting the next larger pump bore will not always increase production. Table 3 demonstrates the problem. Selecting a pump with a bore of 1.75 inches, as compared to a 1.5-inch bore, decreases production, increases peak stress, increases rod loading, and decreases rod life.

PREDICTIVE PROGRAMS

Several predictive programs are commercially available for modeling beam pumping systems and equipment loading. To predict the fatigue life of a fiberglass rod string, convert the peak and minimum loads to peak and minimum stress. Enter the manufacturer's stress range at the minimum stress and determine the allowable peak stress. The rod loading is determined by the following formula:

$$\text{PERCENT OF ROD LOADING} = \frac{\text{ACTUAL LOAD RANGE}}{\text{ALLOWABLE LOAD RANGE}}$$

ACTUAL LOAD RANGE = ACTUAL PEAK STRESS - ACTUAL MINIMUM STRESS

ALLOWABLE LOAD RANGE = ALLOWABLE PEAK STRESS - ACTUAL MINIMUM STRESS

SUGGESTED GUIDELINES

The following suggestions, if followed, should result in successful glass rod installations:

1. Check designs for proper equipment sizing of prime mover, pumping unit, rods, and downhole pump at both high and low fluid level cases.
2. Calculate rod loading, if not done by the predictive, at several places in the string. The stress range diagram applies to all rods in the string.
3. Have a knowledgeable person supervise the installation of equipment, making sure the final installation matches the design predictive.
4. Correctly space the well to prevent damage to the rods and pump.
5. Follow up the installation with a dynamometer card to compare actual loadings with predictive loadings.

If rods meeting API Specification 11C are used and the previous suggestions are followed, an accurate fatigue life should be predicted.

Table 1

ROD PIN SIZE	PIN STRESS RELIEF AREA	ROD BODY AREA STEEL	ROD BODY AREA GLASS	RATIO STEEL	RATIO GLASS
3/4	.658	.442	.601	.672	.913
7/8	.850	.601	.785	.707	.923
1	1.18	.785	1.23	.665	1.04

Table 2

WELL PARAMETERS

7500 SEATING NIPPLE DEPTH
8607 ROD STRING
1.5 INCH PUMP

C-456-256-120 PUMPING UNIT
10 STROKES PER MINUTE
.433 TUBING GRADIENT

	CONDITION 1	CONDITION 2
	FLUID LEVEL 2500 FFS	FLUID LEVEL 7200 FFS
PEAK TORQUE	325,000 IN.-LBS.	323,000 IN.-LBS.
PRODUCTION	425 BFPD	310 BFPD
PPRL	14,170 LBS.	16,930 LBS.
MPRL	4,610 LBS.	4,710 LBS.
% ROD LOADING	78%	84%
DHPS	158 INCHES	117 INCHES

MPL = DHPS + PL + 60
MPL = 158 + 60 + 60 = 278 INCHES

Table 3

WELL PARAMETERS

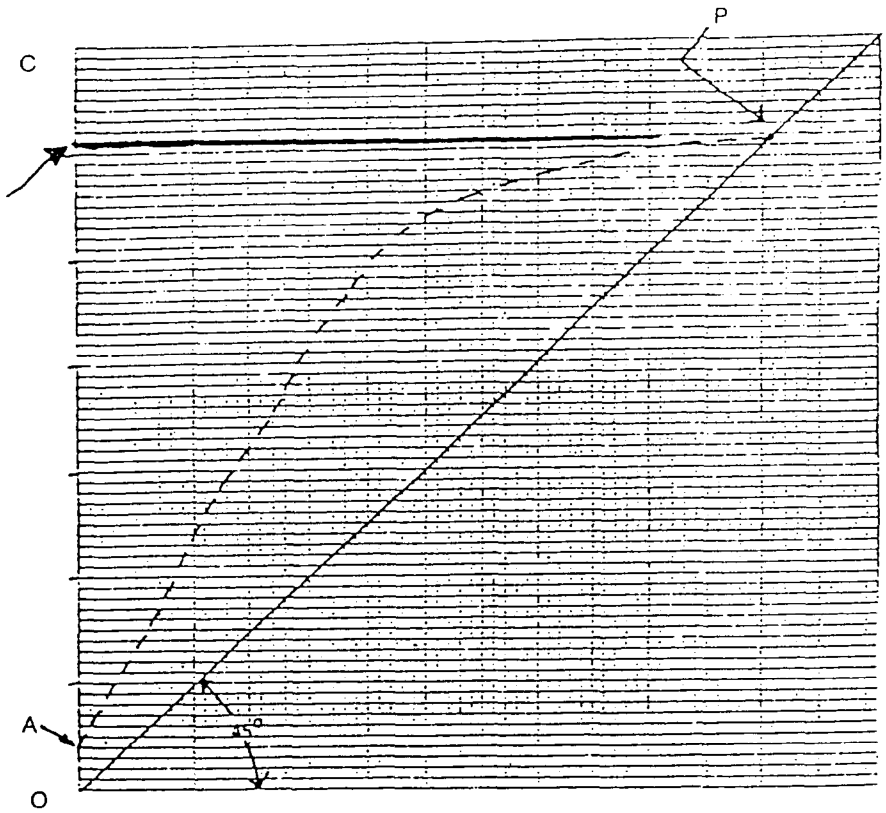
7500 SEATING NIPPLE DEPTH
8607 ROD STRING
FLUID LEVEL 7200 FFS

C-456-256-120
10 STROKES PER MINUTE
.433 TUBING GRADIENT

	CONDITION 1	CONDITION 2	CONDITION 3
	1.25 PUMP BORE	1.5 PUMP BORE	1.75 PUMP BORE
PEAK TORQUE	343,700 IN.-LBS.	323,000 IN.-LBS.	236,400 IN.-LBS.
PRODUCTION	255 BFPD	310 BFPD	288 BFPD
PPRL	15,650 LBS.	16,930 LBS.	18,600 LBS.
MPRL	4,930 LBS.	4,710 LBS.	5,550 LBS.
DHPS	138 INCHES	117 INCHES	79 INCHES
ROD LOADING	78%	84%	87%

(This diagram rated for 160°F temperature and 7.5 x 10^6 cycles to expected first failure of the rod string.)

Manufacturer's Name: ______



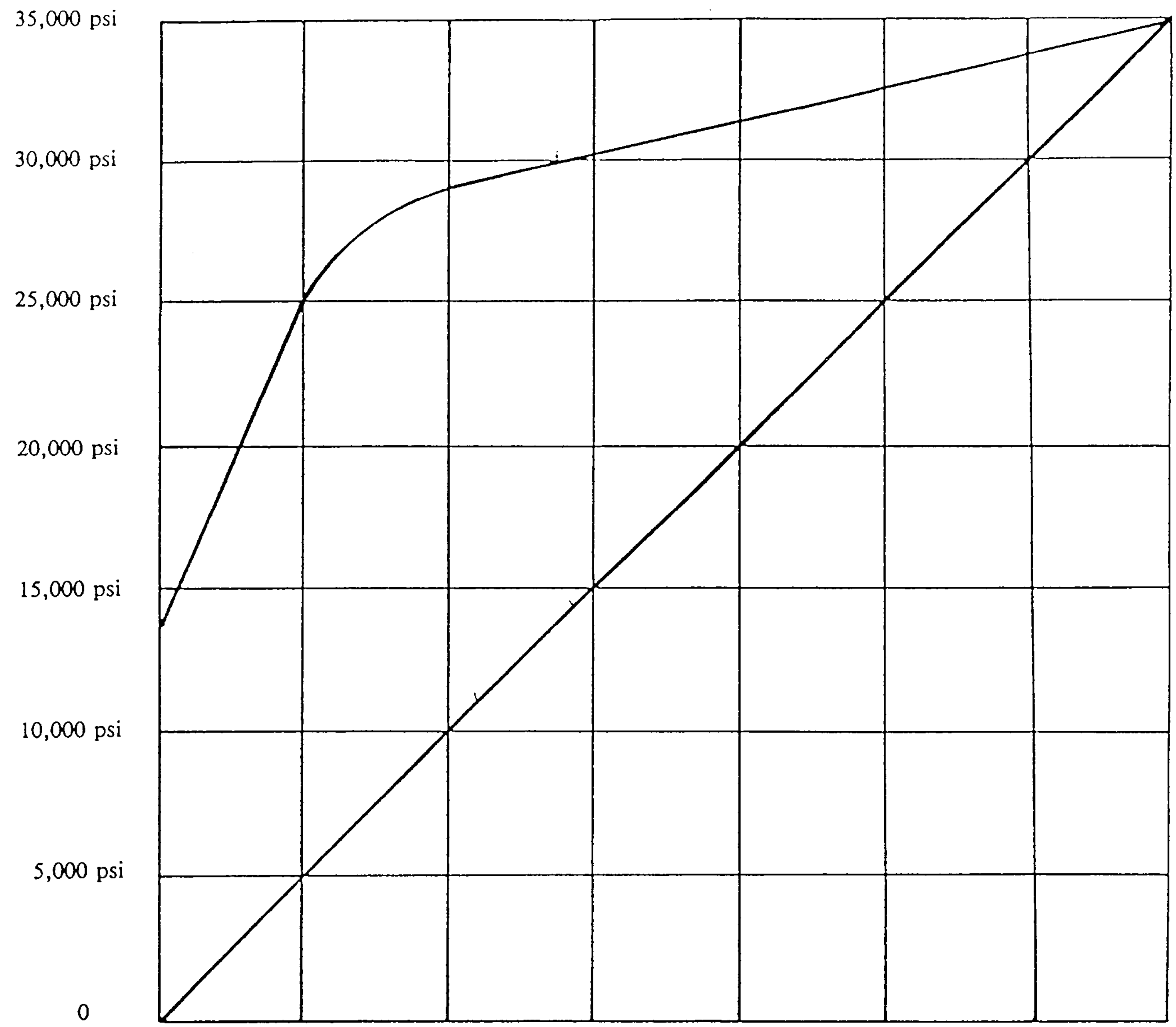
Nominal Rod Body Diameters Covered: . . . .

End Fitting Grade: ______

TABLE 2.2

A Cycles to First Expected Failure	B Allowable Range Modifier	C Operating Temperature of Rods	D Allowable Range Modifier
5 Million	______	RT	______
7.5 Million	100%	100	______
10 Million	______	120	______
15 Million	______	140	______
30 Million	______	160	100%
		180	______
		220	______
		•	______
		•	______
		Max. Op. Temp.	______
		( )	______

Figure 1 — Reinforced plastic sucker rod "BASIC STRESS RANGE DIAGRAM" and table from API Spec 11C, 1st ed., Jan. 1, 1986 (Fig. 2.1 and Table 2.2)



This stress range applies to all rods in the string.
Care should be taken to avoid compressive loading of rods.

Cycles to First Expected Failure	Allowable Range	Temperature of Rods	Allowable Range
5 Million	104%	160°	100%
7.5 Million	100%	180°	98%
10 Million	92%	200°	95%
15 Million	85%	220°	88%
30 Million	80%	240°	80%

Figure 2 — Stress Range Diagram