HOW TO MINIMIZE POLISHED ROD BREAKS

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Introduction

Sucker rod breaks have been extensively studied and documented in the oil industry. Polished rod failures, on the other hand, have not received as much attention. As a general rule, operators seem to be more tolerant of polished rod failures. But polished rods fail for reasons that can be controlled. The purpose of this paper is to identify these reasons and to discuss ways to minimize polished rod breaks.

Almost without exception, the polished rod is the strongest component of the rod string. It has the largest cross-sectional area and its material strength is at least equal to that of the sucker rods. Yet in many cases, polished rods fail with regularity while the sucker rods do not.

Surface pumping equipment can induce destructive stresses in polished rods. By analyzing polished rod failures, which usually occur at the bottom of the polished rod clamp, useful conclusions can be reached about these stresses and what can be done to control them.

Characteristics of Polished Rod Breaks

Almost all polished rod breaks are fatigue failures. Figure1(a) shows the mating cross sections of a polished rod that failed due to fatigue. Fatigue is associated with *fluctuating* loads which can cause failures even though the maximum stress resulting from these loads is less than the ultimate strength of the polished rod material.

A fatigue failure is characterized by the initiation of a small crack at a point of high stress on the surface of a part. Under fluctuating loads, the crack opens and closes repeatedly and grows in a plane perpendicular to the axis of the stress. As the crack progresses, the effective cross-sectional area is reduced until insufficient material remains to support the load.

In Figure 1(a), the crack initiation point is at the top of the cross section and the metal remaining at the rupture point is at the bottom of the cross section. In this case, a crack had progressed across approximately 80% of the cross-sectional area before the polished rod failed.

Endurance Limit

In order to understand fatigue failures, it is helpful to first understand the term *endurance limit*. When stresses resulting from fluctuating loads are below a certain critical level, the life expectancy of a part is unaffected. However, when stresses are above the critical level, life expectancy is reduced. This critical stress level is referred to as the endurance limit for the material and can only be determined experimentally.

A rotating test specimen loaded to induce a bending stress as shown in Figure 1(b) is the most widely used method to determine endurance limit. For every 360° of rotation, the bending stress reverses between maximum tension and maximum compression. By noting the number of revolutions required to produce a failure, a stress-cycle curve, sometimes referred to as an *s-N* curve, can be plotted. Figure 2 shows s-N curves for some typical materials (Ref 1). The endurance limit is that stress at which the slope of the s-N curve becomes flat or approaches zero. It should be noted that for some materials the slopes of their s-N curves never approach zero. As a result, their endurance limits are difficult to define.

Stress concentration is also an important consideration that can affect the endurance limit of a material.The endurance limit is highest if the surface of a part has a polished finish. Surface imperfections are stress raisers that cause stress concentrations which reduce the endurance limit. Some are worse than others. For example, large indentions are worse than small ones as shown in Figure 3 (Ref 3). Dents with sharp, ragged edges will create greater stress concentrations than dents with rounded edges such as a ball bearing dent in a flat plate. Surface imperfections are most detrimental to harder, higher strength steels which are commonly used for polished rods and sucker rods.

Radial compressive loads such as those in shrink fits between disks and shafts shown in Figure 12(a) also generate stress. The greater the interference of the shrink fit, the greater the stress. Therefore, polished rod clamps, by the very nature of their design, cause stress that can reduce the endurance limit of a polished rod. But these stresses can be successfully managed with polished rod clamps that are properly designed and correctly installed.

Analysis Methods

Several empirical methods are available to analyze fluctuating stresses and their potential for causing fatigue failures. The most widely used model is known as the Goodman Diagram. This model is based on the observation that the amplitude of fluctuating stresses (S_a) and their *mean* or *average* value (S_m) can be correlated to fatigue failures. Figure 4(a) represents stresses fluctuating about a mean value. As S_m increases, the magnitude of S_a necessary to cause fatigue failure decreases. The Goodman model in Figure 4(b) assumes a linear relationship between S_a and S_m. As S_m approaches zero, S_a approaches the endurance limit (S_e) of the material. When S_a is zero, S_m is equal to the ultimate tensile strength (S_u) of the material.

Polished Rod Loading

Polished rod loads fluctuate continuously between peak and minimum values. If polished rod loads are pure axial loads, fatigue failures are least likely to occur as will be shown in Example 1. When bending loads are added to axial loads, the possibility of fatigue failures increases as will be shown in Example 2. In Example 3, it will be shown that adding the effect of stress concentrations to bending and axial loads increases the likelihood of fatigue failures even more.

Example 1
Axial Loads Only

Figure 5(a) shows a dynamometer card for a beam pumped well that has a polished rod diameter of 1 1/4" (A= 1.23 sq. in.) similar to the one shown in Figure 1(a). From this card, it can be observed that the peak polished rod load (PPRL) is 24,900 pounds and the minimum polished rod load (MPRL) is 4,400 pounds. The polished rod material has an S_u = 93,000 psi and an S_e = 45,000 psi. These properties are typical of polished rods manufactured from AISI 1045 steel.

The maximum and minimum stresses in the polished rod are:

$$S_{max} = \frac{PPRL}{A} \qquad \text{(Eq. 1)}$$

$$S_{max} = \frac{24,900}{1.23} = 20,300 \text{ psi}$$

$$S_{min} = \frac{MPRL}{A} \qquad \text{(Eq. 2)}$$

$$S_{min} = \frac{4,400}{1.23} = 3,600 \text{ psi}$$

The amplitude of the fluctuating stress in the polished rod is:

$$S_a = \frac{S_{max} - S_{min}}{2} \qquad \text{(Eq. 3)}$$

$$S_a = \frac{(20,300 - 3,600)}{2} = 8,350 \text{ psi}$$

The mean value of the fluctuating stress is:

$$S_m = \frac{S_{max} + S_{min}}{2} \qquad \text{(Eq. 4)}$$

$$S_m = \frac{(20,300 + 3,600)}{2} = 11,950 \text{ psi}$$

These values of S_a and S_m are plotted and shown as the 0^o point on the Goodman Diagram in Figure 5(b). The point is well below the diagonal line.Therefore, the polished rod should not fail in fatigue.

Bending

As mentioned earlier, polished rod loads are rarely simple axial loads. In the real world, the polished rod is subjected to bending as well as axial loads. Typical conditions that can cause bending are:

(1) **Misaligned Pumping Units**. Figure 6(a) shows a plumb line suspended from the horsehead of a pumping unit. The horsehead is approximately 6 inches off the center line of the wellhead assembly. This misalignment will induce bending stresses in the polished rod and could cause premature polished rod failure. Unstable pumping unit foundations, bent structural members on the pumping unit, and improperly adjusted horseheads can cause misalignment.

Some stuffing boxes can help reduce bending stresses induced by misaligned pumping units. Stuffing boxes such as Huber's Double Pack can "flex" as the polished rod reciprocates. This flexing action can relieve some of the bending stresses that would otherwise occur if the stuffing box was rigid.

(2) **Misaligned Wellhead Assembly**. A casing head, tubing head, flow tee, and stuffing box assembly should be vertical to allow alignment with the polished rod. API Recommended Practice RP 11B suggests the assembly be vertical to within 1 1/2 inches in 20 feet, or about 0.35^{o}. Figure 6(b) shows an assembly that is approximately 1^{o} from vertical. Unless the polished rod happens to be misaligned the same amount and in the same direction, bending loads will be imposed on the polished rod.

(3) **Unlevel Carrier Bars**. Carrier bars must be level to prevent bending the polished rod. Figure 7(a) shows a carrier bar that is approximately 2^{o} out-of-level. The load on the polished rod will force the bottom of the clamp flush with the top of the carrier bar and bend the polished rod.

Leveling plates can be used to compensate for unlevel carrier bars. Huber's two-piece concave-convex leveling system shown in Figure 7(b) can make incremental corrections up to a maximum of 2^{o}.

(4) **Uneven Surfaces on the Carrier Bar**. Not only should carrier bars be level, but their top surfaces should be flat. Figure 8(a) shows the top of a carrier bar that is worn. This carrier bar will bend the polished rod. A leveling plate can also be used in this situation to correct for the uneven surfaces. Otherwise, worn carrier bars should be machined flat or replaced.

(5) **Uneven Polished Rod Clamps**. The bottom surface of a polished rod clamp should be flat, in a common plane, and perpendicular to the axis of the polished rod. Figure 8(b) is an impression block that was installed between a carrier bar and a polished rod clamp. The impression indicates one side of the clamp is putting more pressure on the carrier bar than the other.

All polished rod clamps have some end play between segments. Care should be taken to insure segments move freely on their hinges before installation and are even on the bottom after the clamp bolts have been tightened. Do not assume segments are even on the bottom just because they are even on the top.

Another way to check for uniform engagement between the polished rod clamp and the carrier bar is to test for gaps with a feeler gage or a sheet of paper as the weight of the rod string is slowly transferred to the carrier bar. Measurements should be taken before all the weight is applied to the carrier bar because under full load the polished rod and clamp will deflect and close any gaps that might exist.

Example 2
Bending and Axial Loads

Consider Example 1 with the addition of a bending load. Assume bending results from an unlevel carrier bar and an uneven polished rod clamp shown in Figure 9(a). As the polished rod load (P) is transmitted to the carrier bar, a bending stress (S_b) is generated in the polished rod as the clamp and polished rod rotate around point "A." The bending moment (M) will cause the polished rod to deflect resulting in the lateral displacement of the polished rod, carrier bar, and bridle as shown in Figure 9(b). As long as Θ is a reasonably small angle, the clamp and carrier bar will make contact at points "A" and "B."

A free-body diagram of the polished rod subjected to M and P is shown in Figure 10(a). F is the horizontal component of P after the bridle has been displaced laterally. F is roughly equal to the product of P and the ratio of the lateral displacement (Y) to the length of the bridle (L_b).

$$F = \frac{Y}{L_b} P \qquad \text{(Eq. 5)}$$

For all practical purposes, the ratio of Y/L_b is small enough that the shearing stress generated by F can be neglected. As a result, the polished rod can be treated as a cantilevered beam of length (L) with an axial load (P) and bending moment (M) acting on the free end as shown in Figure 10(b).

The equation (Ref 2) which can be used to solve for M acting alone at the free end of a cantilevered beam is:

$$M = \frac{\Theta EI}{57.3L} \qquad \text{(Eq. 6)}$$

However, Eq. 6 must be adjusted to account for P in order to determine the effective bending moment (M_e):

$$M_e = \frac{M}{\left[1 + \frac{PL^2}{8EI}\right]} \qquad \text{(Eq. 7)}$$

$$M_e = \frac{\frac{\Theta EI}{57.3L}}{\left[1 + \frac{PL^2}{8EI}\right]} \qquad \text{(Eq. 8)}$$

The bending stress (S_b) generated by M_e and added to the tensile stress (S_t), already calculated in Example 1, will equal the combined stress (S_c).

$$S_c = S_t + S_b \qquad \text{(Eq. 9)}$$

$$S_c = \frac{P}{A} + \frac{M_e c}{I} \qquad \text{(Eq. 10)}$$

$$c = \frac{d}{2} \qquad \text{(Eq. 11)}$$

$$I = \frac{\pi d^4}{64} \qquad \text{(Eq. 12)}$$

$$S_c = \frac{P}{A} + \frac{\Theta E I \quad d}{57.3L \ \ 2I \left[1 + \frac{PL^2}{8EI}\right]} \qquad \text{(Eq. 13)}$$

$$S_c = \frac{4P}{\pi d^2} + \frac{\Theta E d}{114.6L \left[1 + \frac{2.55\, PL^2}{Ed^4}\right]} \qquad \text{(Eq. 14)}$$

For a 1 1/4" diameter steel polished rod ($E = 30 \times 10^6$ psi), Eq. 14 can be reduced to:

$$Sc = 0.81P + \frac{3.27 \times 10^5\, \Theta}{L\left[1 + \frac{PL^2}{28.76 \times 10^6}\right]} \qquad \text{(Eq. 15)}$$

Where:

S_c is in Lbs per sq. in.

P is in Lbs

Θ is in Degrees

L is in Inches

Using Eq. 15 and the dynamometer card in Figure 5(a), combined stresses (S_c) were calculated for $\Theta = 0^o$, $\Theta = 1^o$, $\Theta = 3^o$, and $\Theta = 5^o$. Results are listed in Figure 11(a). Note that $S_c = S_t$ when $\Theta = 0^o$ because $S_b = 0$. New values of S_a and S_m were calculated for each value of Θ. Results are listed in Figure 11(b) and plotted on the Goodman Diagram in Figure 5(b) as points 1^o, 3^o, and 5^o.

Figure 5(b) clearly shows that the addition of bending stresses is progressively detrimental. A carrier bar tilted between 3^{o} and 4^{o} will induce enough bending stress to equal the maximum allowable stress according to the Goodman Diagram.

Stress Concentrations

Stress concentrations generated by polished rod clamps cannot be avoided. There is a noticeable absence of information on the amount and effect of stress concentration caused by polished rod clamps. However, a lot can be learned by comparing polished rod clamps to shaft and disk shrink fits illustrated in Figure 12(a). It's when stress concentration is combined with bending that the more severe problems occur.

Alternating bending stresses generate a fretting erosion at the edge of a shrink fitted member at point C. Bending, added to the radial compressive stresses caused by the interference fit between the disk and shaft is the cause of frequent shaft failures at point C. The disk-shaft assembly is very similar to a clamp and polished rod. This comparison explains why most polished rods fail at the bottom of the clamp. One way to reduce the failure rate is to groove the disk as shown in Figure 12(b). The groove allows the shaft to have a greater bending radius which, albeit a small amount, is enough to reduce the concentration of stresses at point C.

Figure 12(c) shows that bending stresses (S_b) can be amplified by a stress concentration factor (k). The magnitude of the factor depends on the S_r/S_b and l/d ratios. As the ratio of the radial stresses (S_r) to the bending stress (S_b) increases, the stress concentration factor increases. The same holds true for the ratio of the disk length (l) to the shaft diameter (d).

By using Figure 12(c), stress concentration factors can be estimated for polished rod clamps. As a result, it is possible to generate another stress line on the Goodman Diagram which will show the effect of stress concentration on the S_a and S_m values in Example 2.

<u>Example 3</u>

Bending and Axial Loads

Combined with Stress Concentration Factors

Assume:

l	= 4"	length of polished rod clamp (friction model)
d	= 1 1/4"	diameter of polished rod
μ	= 0.12	coefficient of friction between clamp and PR
P_c	= 25,000 lbs	clamp capacity ≥ point B in Figure 5(a)

Determine:

l/d = ratio of clamp length to polished rod diameter
S_r = radial stress imposed by the clamp to support P
S_{bo} = maximum bending stress if $\Theta = 0^o$
S_{b1} = maximum bending stress if $\Theta = 1^o$
S_{b3} = maximum bending stress if $\Theta = 3^o$
S_{b5} = maximum bending stress if $\Theta = 5^o$
S_r/S_{bo} = stress ratio if $\Theta = 0^o$
S_r/S_{b1} = stress ratio if $\Theta = 1^o$
S_r/S_{b3} = stress ratio if $\Theta = 3^o$
S_r/S_{b5} = stress ratio if $\Theta = 5^o$
k_o = stress concentration factor if $\Theta = 0^o$
k_1 = stress concentration factor if $\Theta = 1^o$
k_3 = stress concentration factor if $\Theta = 3^o$
k_5 = stress concentration factor if $\Theta = 5^o$

Solution:

$$\frac{l}{d} = \frac{4}{1.25} = 3.2 \qquad \text{(Eq. 16)}$$

$$S_r = \frac{P_c}{\pi\, d\, l\, \mu} = \frac{25{,}000}{(\pi)\,(1.25)(4)\,(0.12)} = 13{,}263 \text{ psi} \qquad \text{(Eq. 17)}$$

$$S_c = S_t + S_b \qquad \text{(Eq. 9)}$$

$$S_b = S_c - S_t \qquad \text{(Eq. 18)}$$

From Figure 11(a) for Point A

S_{bo} = 0
S_{b1} = 25,200 - 11,600 = 13,600
S_{b3} = 52,500 - 11,600 = 40,900
S_{b5} = 80,000 - 11,600 = 68,400

Solution (Cont)

$$\frac{S_r}{S_{bo}} = \text{N/A}; S_b = 0 \qquad \text{(Eq. 19)}$$

$$\frac{S_r}{S_{b1}} = \frac{13{,}263}{13{,}600} = 1.0 \qquad \text{(Eq. 20)}$$

$$\frac{S_r}{S_{b3}} = \frac{13{,}263}{40{,}900} = 0.3 \qquad \text{(Eq. 21)}$$

$$\frac{S_r}{S_{b5}} = \frac{13{,}263}{68{,}400} = 0.2 \qquad \text{(Eq. 22)}$$

From Figure 12(c)

$k_0 = 1.0$

$k_1 > 2.0$

$k_3 \approx 1.4$

$k_5 \approx 1.3$

Figure 13(b) shows the amplified bending stresses (kS_b) which are the product of the stress concentration factor (k) and the bending stresses (S_b) in Figure 13(a). Combined stresses (S_c) which reflect the amplified bending stresses (kS_b) are shown in Figure 14(a). Values of S_a and S_m for each value of Θ were calculated using Equations 3 and 4. The values were listed in Figure 14(b) and plotted in Figure 15.

The stress line on the Goodman Diagram in Figure 15 followed the same track as Example 2. The points moved further out. As a result, it was determined that stresses from a carrier bar tilted 3^o was just outside the maximum allowable stress line. Another way of saying the same thing is that a carrier bar tilted 2^o-3^o would dramatically reduce the life expectancy of a polished rod on this well.

Polished Rod Clamp Design

Basically, polished rod clamps can be divided into two categories— indention and friction models. Huber Fig. 1, Fig. 2, and Fig. 3 clamps are indention models. Huber's indention design is illustrated in Figures 16, 17, and 18. The I.D. of the clamp is machined slightly smaller than the O.D. of the polished rod so that the narrow lands make small indentions (Figure 18), which are well below limits set by API Spec 11B. Friction models are similar except the lands extend the full length of the clamp segments and their I.D.'s are essentially the same as the O.D. of the polished rods.

Relative to friction models, indention clamps will support greater loads at equal bolt torques as shown in Figure 19. The obvious reason is that idention clamps are not totally dependent on the coefficient of friction between the I.D. of the clamp and the O.D. of the polished rod. As a result, indention clamps are less likely to drop the rod string because many field conditions such as lubricating oils, dirt, and grease can reduce the coefficient of friction which friction clamps rely on totally.

Huber's indention design has three narrow lands in each clamp segment which support the load. The face of each land has machined micro-grooves which also increase load carrying capacity. In theory, the narrow lands result in a shorter effective length which reduces the l/d ratio and, subsequently, the stress concentration factor as shown in Figure 12(c).

The downside to indention clamps is that they create slight surface imperfections in the polished rod. However, Huber's experience indicates the stress points caused by the indentions are a small price to pay for the added load bearing and reliability benefits. Maximum indentions on Huber clamps are only two-tenths of one-thousandth of an inch which is 50 times less than API recommended limits. In addition, the indentions have all the characteristics of minimal stress raisers—smooth surfaces with well-rounded corners. One word of caution for all clamps is that care must be exercised to follow the manufacturer's torque recommendations for clamp bolts. Overtightening can result in excessive stresses.

Polished Rod Care

Polished rod clamps should never be installed on the hardened "spray metal" surface of a polished rod. The hard surface will be damaged. Both indentions and friction models will have to be excessively tightened in order to achieve rated loads. These excessive radial stresses can damage the clamp as well as the polished rod. Stress buildup in the polished rod will be excessive which will quickly lead to stress cracking and premature polished rod failure. The broken polished rod shown in Figure 1(a) was clamped off on the spray metal section. The polished rod failed after 35,000 strokes of the pumping unit.

Polished rods should be handled just as carefully as sucker rods as outlined in API's RP 11BR. Remember, all dents and bends are stress raisers that are to be avoided. Pipe wrenches should never be used on polished rods. The surface of polished rods should be protected with cardboard tubes or their original shipping containers while being transported or stored. During workovers, the stuffing box and BOP should be removed from around the polished rod before laying a polished rod down. Polished rods should be supported evenly on level wooden racks when removed from service and stored.

Conclusions

1. Bending stresses are the number one enemy of a polished rod's life expectancy. The majority of polished rod failures is the result of induced bending stresses by one or more of the causes discussed in this paper.

2. By the nature of their design, polished rod clamps create stress. Stress concentrations alone are usually not the reason polished rods fail. Two exceptions are clamps applied to the hardened "spray metal" surfaces of polished rods and grossly overtightened polished rod clamps.
3. The apparent mechanical simplicity of polished rods and polished rod clamps is misleading. It's easy to assume that no special knowledge or understanding of the equipment is required. Therefore, improved communications and cooperation between manufacturers and operators is needed to train field personnel on the proper installation and use of polished rods and clamps.
4. Properly selected and installed stuffing boxes can help reduce bending stresses.
5. Leveling plates provide assurance the carrier bar and polished rod clamp interface on a level plane to reduce unwanted bending stresses.
6. Manufacturing tolerances for the length of polished rod clamp segments should be tightened. It should not be assumed the bottom of a clamp is level just because the top is level. Lengths of clamp segments have not always been controlled to close tolerances. As a result, it is possible to have a step or offset on one end when the other end is even.

Nomenclature

A	Cross-sectional area of polished rod (in^2)
c	Distance from neutral axis to point of max stress in a beam (in)
d	Diameter of PR (in)
E	Young's modulus (30×10^6 psi)
F	Horizontal component of P @ Y
I	PR moment of inertia (in^4)
k	Stress concentration factor
L	Distance between top of carrier bar and top of stuffing box (in)
L_b	Length of horsehead bridle (in)
l	Length of PR clamp (in)
M	Bending moment in the PR at L required to deflect PR Θ^o (in-lbs)
M_e	Bending moment in the PR at L including the effect of P (in-lbs)
MPRL	Minimum polished rod load (lbs)
P	Polished rod tensile load (lbs)
P_c	Polished rod clamp capacity (lbs)
PPRL	Peak polished rod load (lbs)
PR	Polished rod
S_a	Amplitude of fluctuating stresses (psi)
S_b	Bending stress in the polished rod (psi)
S_c	Combined tensile and bending stress (psi)
S_e	Endurance limit (psi)
S_m	Mean or average value of fluctuating stresses (psi)
S_r	Radial compressive stress on the PR (psi)
S_t	Tensile stress in PR (psi)
S_u	Ultimate strength (psi)
S_{max}	Maximum stress (psi)
S_{min}	Minimum stress (psi)
Y	Lateral displacement of carrier bar (in)
π	Constant = 3.14159
Θ	Deflection of PR at clamp end (degrees)
μ	Coefficient of friction between clamp and PR

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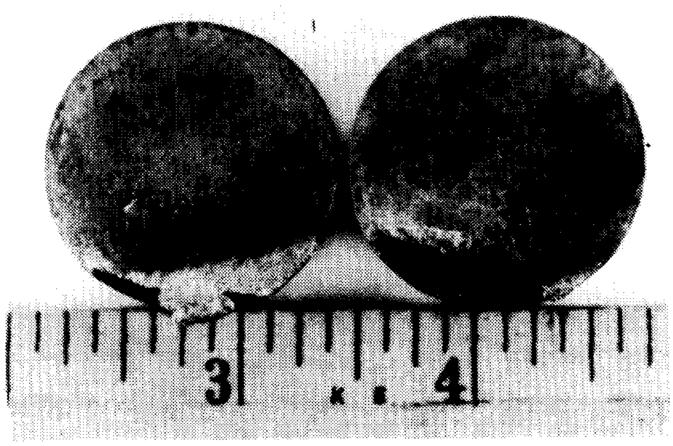


Figure 1a - Failed polished rod

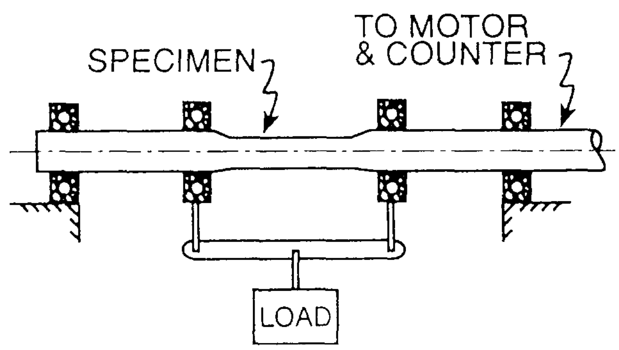


Figure 1b - Rotating beam fatigue testing machine

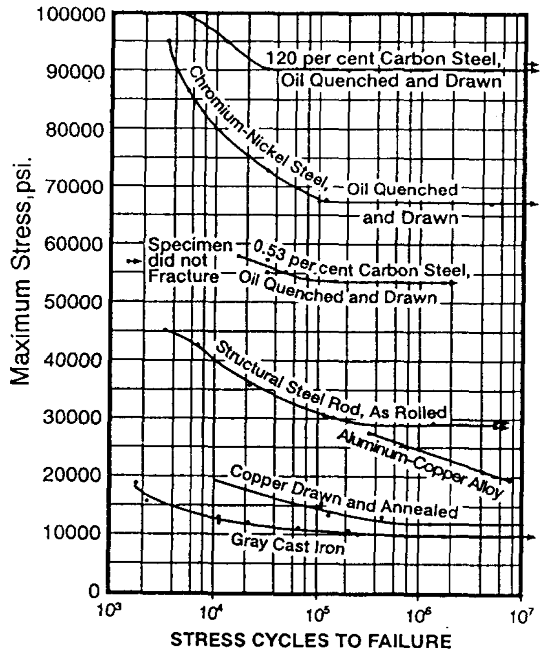


Figure 2 - Endurance limit for various metals
(Ref. 1)

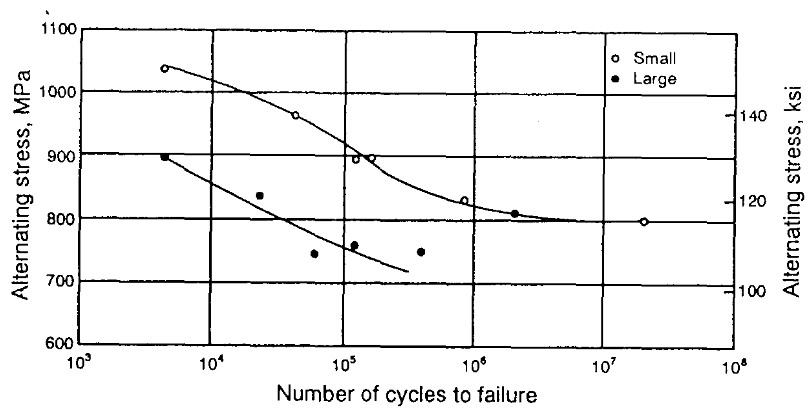


Figure 3 - Effect of surface imperfections on the endurance limit of sucker rods
(Ref. 3)

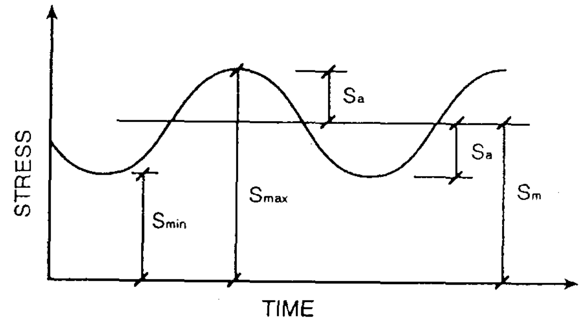


Figure 4a - Amplitude and mean values for fluctuating stresses

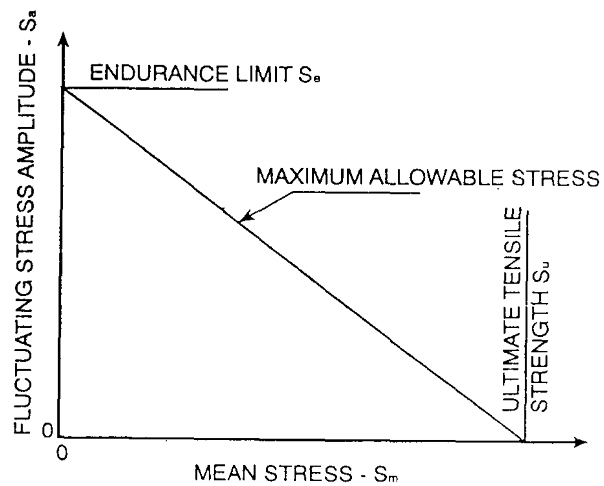
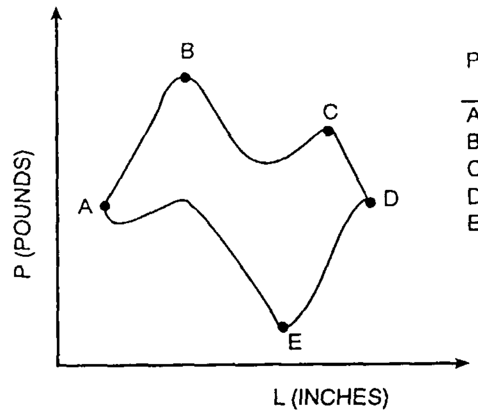


Figure 4b - Principles of the Goodman diagram



POINT	P (POUNDS)	L (INCHES)
A	14,300	20
B	24,900	84
C	20,500	204
D	14,600	236
E	4,400	112

Figure 5a - Dynamometer card for examples 1, 2 & 3

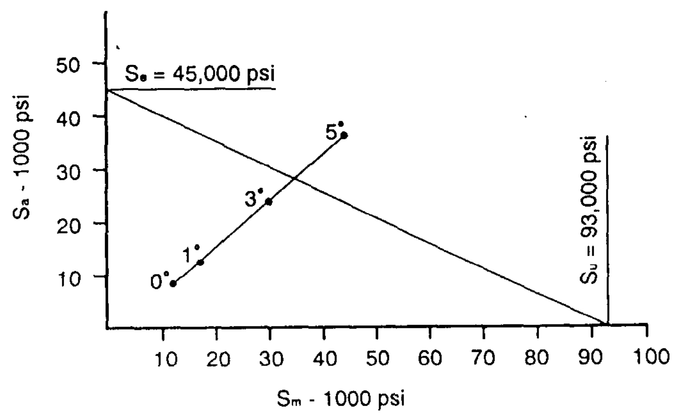


Figure 5b - Goodman diagram for examples 1 & 2

Figure 6a - Example of misalignment between the pumping unit and wellhead assembly

Figure 6b - Example of a wellhead assembly that is vertically misaligned

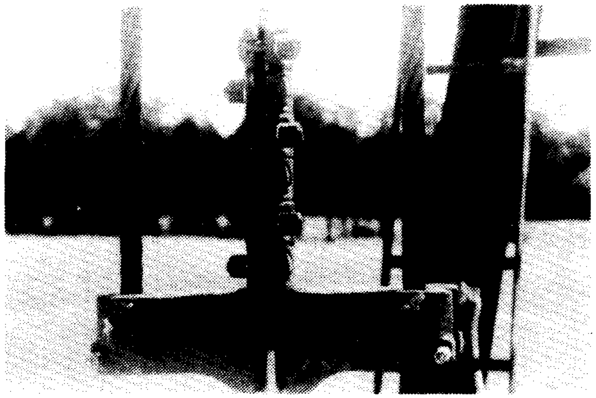


Figure 7a - Example of an unlevel carrier bar

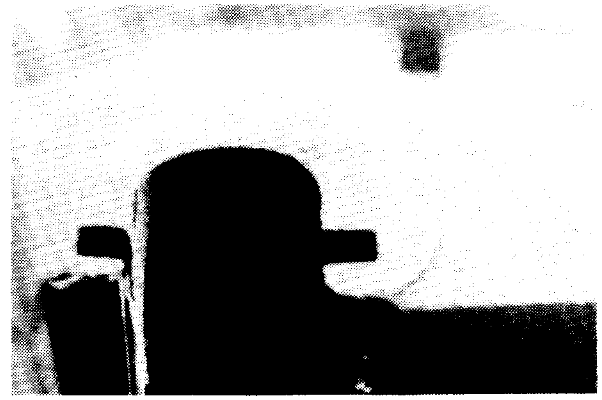


Figure 8a - Example of an imperfect surface on the top of the carrier bar

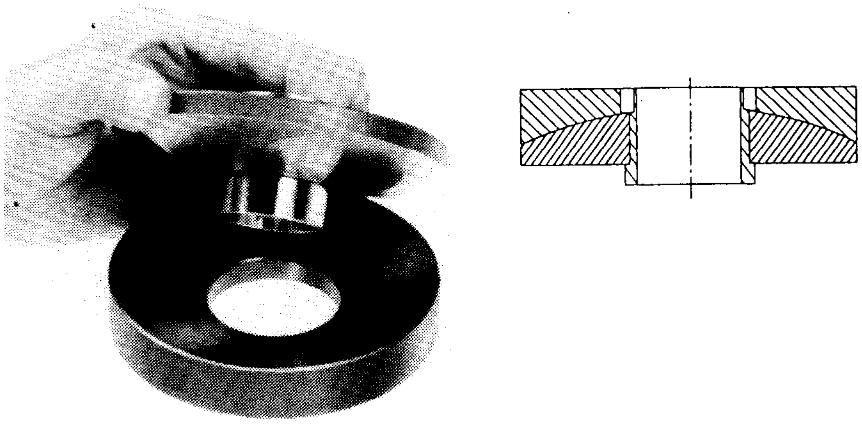


Figure 7b - Huber leveling plate

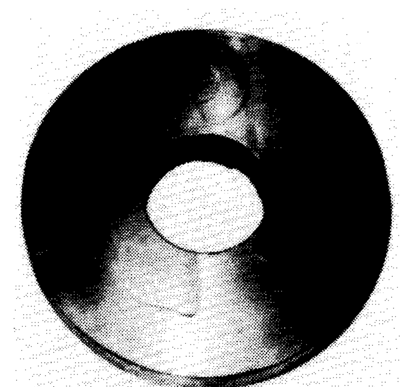


Figure 8b - Impression block showing the effect of uneven segments on the bottom of a polished rod clamp

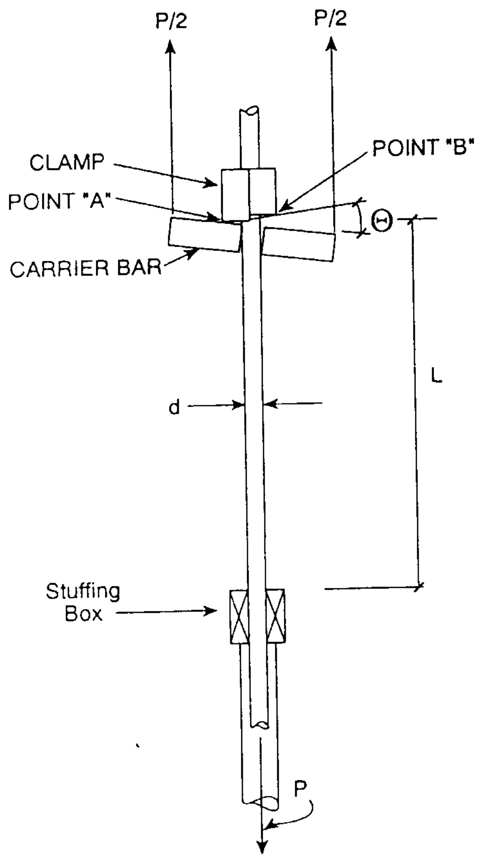


Figure 9a

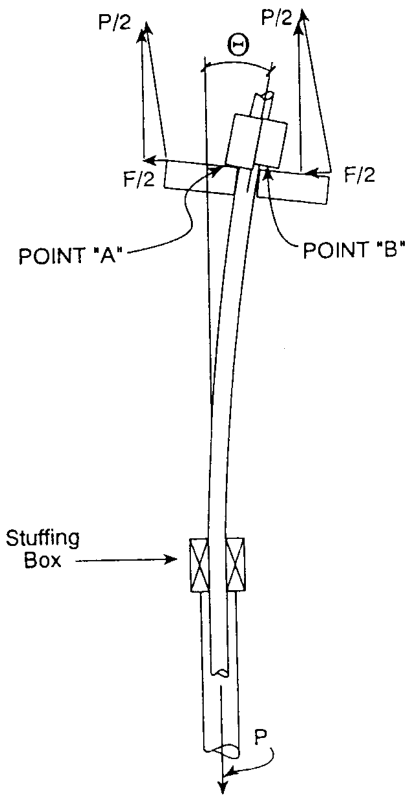


Figure 9b

Illustration of how an unlevel carrier bar and uneven segments of a polished rod clamp can induce bending moments into a polished rod

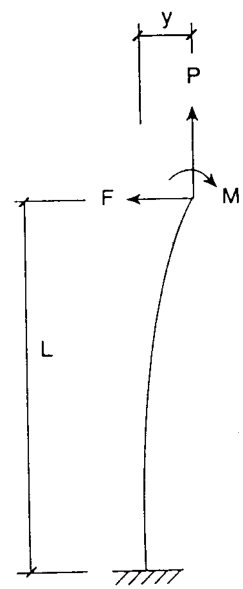


Figure 10a

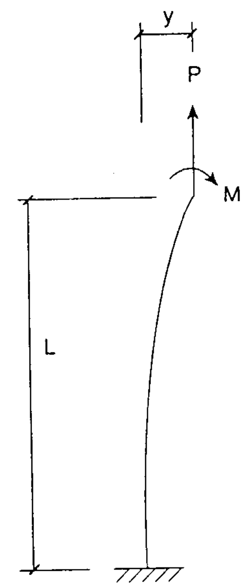


Figure 10b

Free body diagrams of polished rod in example 2

POINT	P (lbs)	L (in)	$S_c = S_t + S_b$			
			$\theta=0^\circ$*	$\theta=1^\circ$	$\theta=3^\circ$	$\theta=5^\circ$
A	14,300	20	11,600	25,200	52,500	80,000
B	24,900	84	20,300	20,700	21,800	22,900
C	20,500	204	16,600	16,700	16,800	16,900
D	14,600	236	11,800	11,900	12,000	12,100
E	4,400	112	3,600	4,600	6,600	8,600

*If $\Theta = 0^\circ$, then $S_b = 0$ $\therefore S_c = S_t$

Figure 11a - Combined bending S_b and tensile stresses S_t in the polished rod for example 2

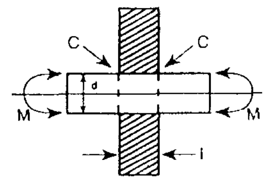


Figure 12a - Stress concentration factors example 3

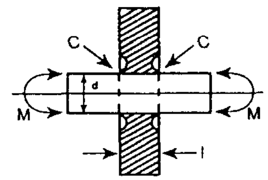


Figure 12b - Stress concentration factors example 3

Θ	Smax	Smin	Sa	Sm
0°	20,200	3,600	8,350	11,950
1°	25,200	4,600	10,300	14,900
3°	52,500	6,600	23,000	30,000
5°	80,000	8,600	35,700	44,300

Figure 11b - Amplitude S_a and mean S_m for fluctuating stresses in example 2

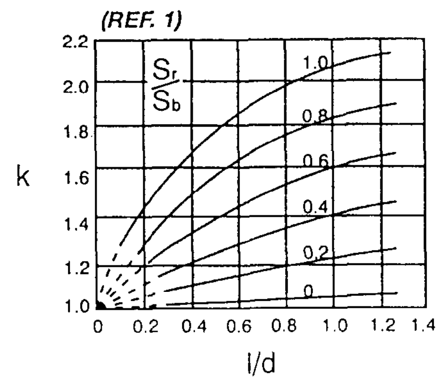


Figure 12c - Stress concentration factors example 3

From Figure 11(a) in Example 1 Where k=1.0

POINT	P (lbs)	L (in)	$S_b = S_c - S_t$ θ=0°	θ=1°	θ=3°	θ=5°
A	14,300	20	0	13,600	40,900	68,400
B	24,900	84	0	400	1,500	2,600
C	20,500	204	0	100	200	300
D	14,600	236	0	100	200	300
E	4,400	112	0	1,000	3,000	5,000

Figure 13a - Effect of stress concentration on the endurance limit of polished rods for example 3

POINT	P (lbs)	L (in)	$k S_b$ θ=0°	θ=1°	θ=3°	θ=5°
A	ʻ14,300	20	0	27,200	57,260	88,920
B	24,900	84	0	800	2,100	3,380
C	20,500	204	0	200	280	390
D	14,600	236	0	200	280	390
E	4,400	112	0	200	4,200	6,500

Figure 13b - Bending stresses corrected for stress concentration k>1.0

POINT	P (lbs)	L (in)	$S_c = S_t + k S_b$ θ=0°*	θ=1°	θ=3°	θ=5°
A	14,300	20	11,600	38,800	68,860	100,520
B	24,900	84	20,300	21,100	22,400	23,680
C	20,500	204	16,600	16,800	16,880	16,990
D	14,600	236	11,800	12,000	12,080	12,190
E	4,400	112	3,600	5,600	7,800	10,100

*If $\theta = 0°$ then $S_b = 0$ $\therefore S_c = S_t$

Figure 14a - Effect of stress concentration on the endurance limit of polished rods for example 3

Θ	Smax	Smin	Sa	Sm
0°	20,300	3,600	8,350	11,950
1°	38,800	5,600	16,600	22,200
3°	68,860	7,800	30,530	38,330
5°	100,520	10,100	45,210	55,310

Figure 14b - Amplitude S_a and mean S_m values for fluctuating stresses in example 3

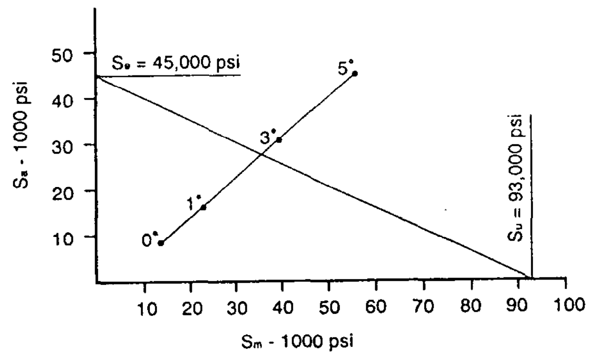


Figure 15 - Goodman diagram for example 3

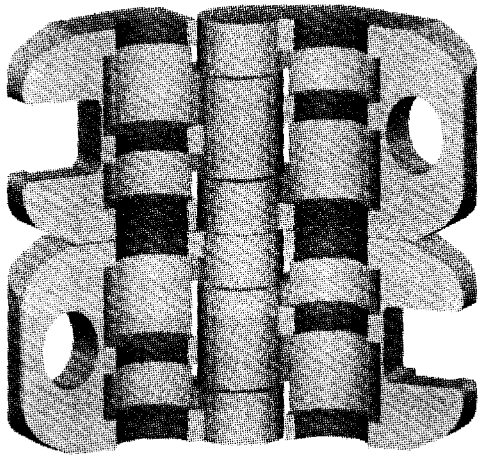


Figure 16 - Huber 1 1/4" figure 2 polished rod clamp in the open position

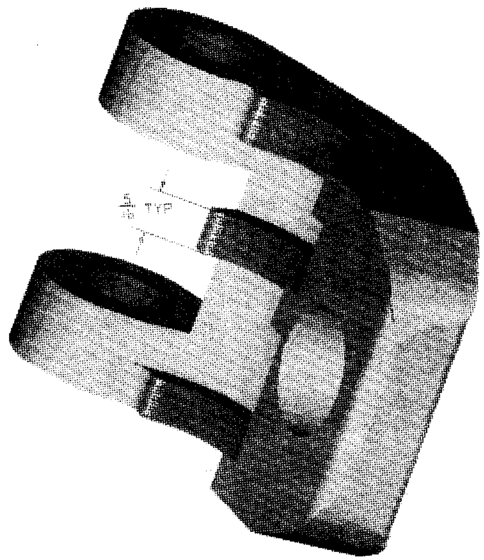
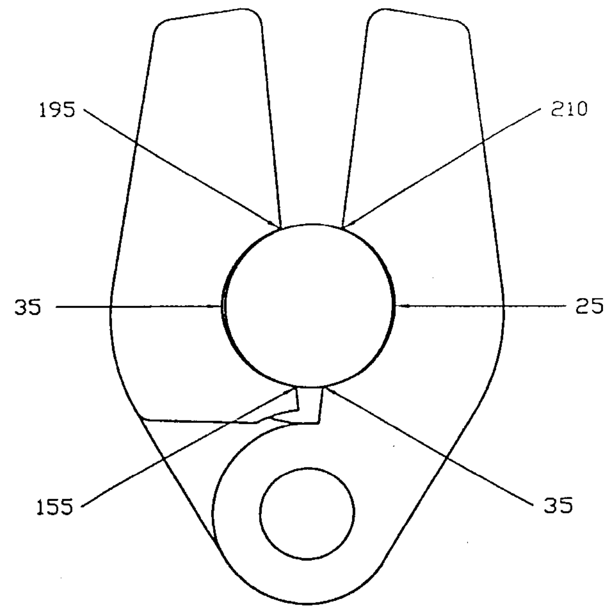


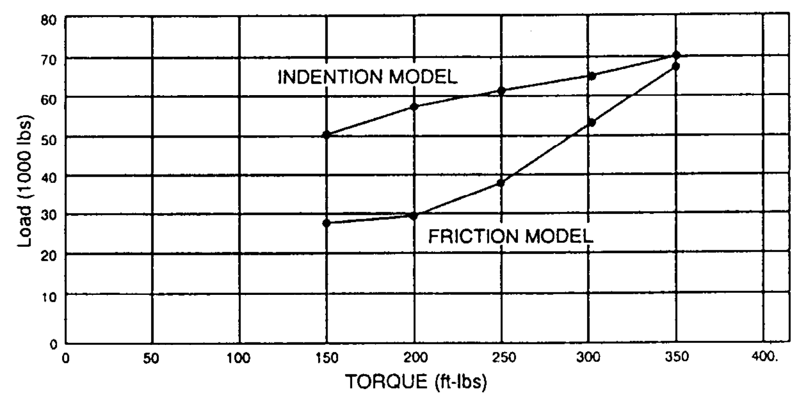
Figure 17 - Landing area closeup for one segment of a Huber 1 1/4" figure 2 polished rod clamp
Each segment of a Huber Polished Rod Clamp contains 3 radial landing areas.



Micro-inch = 1 X 10^{-6} inches
195 Micro-inches = 0.000195 inches
Max. Indention Recommended by API = 0.010 inches

Test conducted and reported by Southwestern Laboratories, Inc., Dallas TX March 1993.

Figure 18 - Polished rod indention profile in micro-inches for Huber 1 1/4" figure 2 polished rod clamp at 150 ft-lbs of clamp bolt torque



API SPEC 11B - RATED LOAD CANNOT BE GREATER THAN 75% OF CLAMP SLIPPAGE LOAD

Figure 19 - Load to slippage vs. bolt torque for 1 1/4" Huber figure 2 polished rod clamps for indention and friction grip models