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- #1 PAYOFF REEL MANDREL FAILURE -

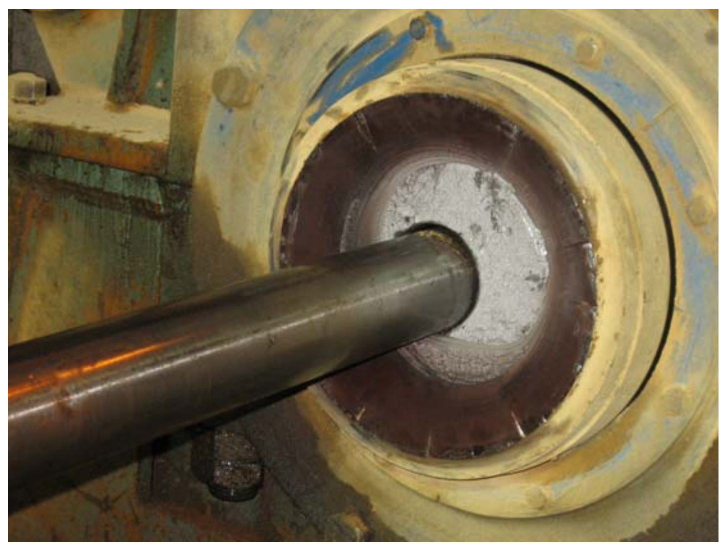
July 1, 2006

The mandrel on the #1 Payoff Reel on the ABC Line at XYZ Company broke around 8pm on Sunday, June 25, 2006. The failure was a classic fatigue failure of the metal. There were no early indications of the impending failure. The mandrel was changed and normal operations resumed at 8pm on Monday, June 26, 2006.

FAILURE ANALYSIS

The failure is shown below. Both photographs show the same pattern. The outermost elements of the shaft have some discoloration from light corrosion indicating prolonged exposure to the elements. That indicates that some cracking may have been evident for some time. The outer part of that discolored area has no rough edges and the ring pattern such as seen in the inner, more recent portion. The outer portion has been rubbed clean by the relative motion of the two portions of the shaft.

The brighter spot on the interior is a brittle fracture where the shaft finally let loose. There is a scar where on both parts where a chunk may have been dragged as the parts separated.



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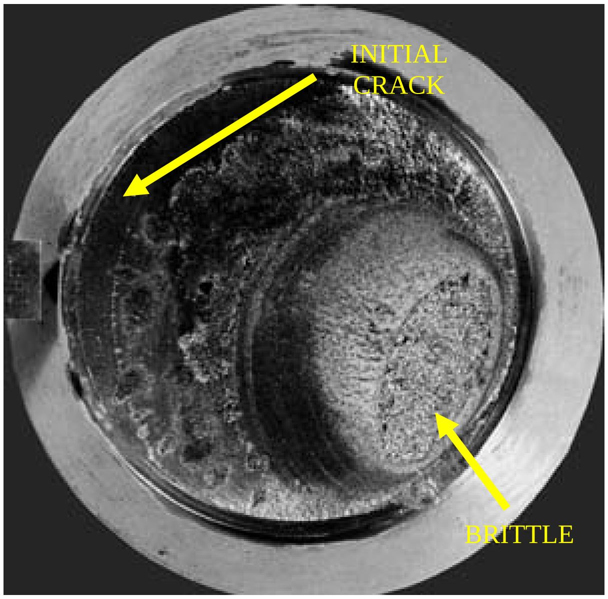

As a point of comparison, the picture shown here is a typical fatigue failure purposely produced for testing. It is right at a shoulder in the shaft. Shoulders and nicks produce what are called stress concentrations. These concentrations can raise the amount of stress on a small portion of steel by 2 or 3 times or more. If the shaft isn't properly designed, is overloaded or if the machining quality is poor then shoulders and nicks are often where fatigue failures will start.

This is typical of a shafting fatigue failure. Note the similarities to the mandrel failure. The large zone on the left where the cracking began and gradually worked across the face followed by a brittle area in the lower right where the shaft finally snapped. The initial crack was at about 10:00 and it gradually worked its way across the shaft until there wasn't enough metal to support the load. The shaft then broke as indicated by the oval in the lower right.

In the case of the mandrel shaft, the crack propagation appears to have stopped at the hole. The outermost portions of the cracking area have been rubbed down so that no sharp edges remain.

REPAIRS

The broken mandrel was replaced with a built up spare. The 'built up' spare already had bearing cartridges and the hydraulic actuator installed. Unfortunately, the outboard bearing catridge, the one furthest from the head, was installed incorrectly on the spare. The cartridge has an offset groove and holes for oil flow. It was installed backwards on this mandrel so that the oil flow would've been restricted.

The general arrangement (USS E-4-14082, zone E2) drawing shows a dimension of 8-1/4" from the end of a shaft thread to the center of the bearing and the holes. It appears that this dimension was not held. Possibly the drawing wasn't available to those who built up the shaft, but disassembly of the shaft and good practice should have noted the offset and the correct assembly direction.

CAUSE

Fatigue calculations for the mandrel design were made (see attached) and show that the mandrel can be subject to fatigue failure concerns. It is important to note that these calculations were worst case. They assume the largest coil and do not give leeway for the coil weight and resulting stresses reducing as the coil pays off. Still, the question of fatigue has to remain, especially on a mandrel that has been in service for over 40 years.

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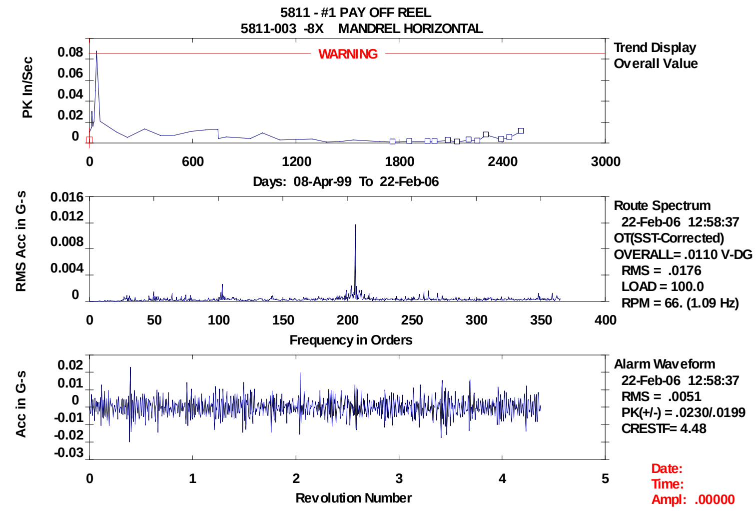
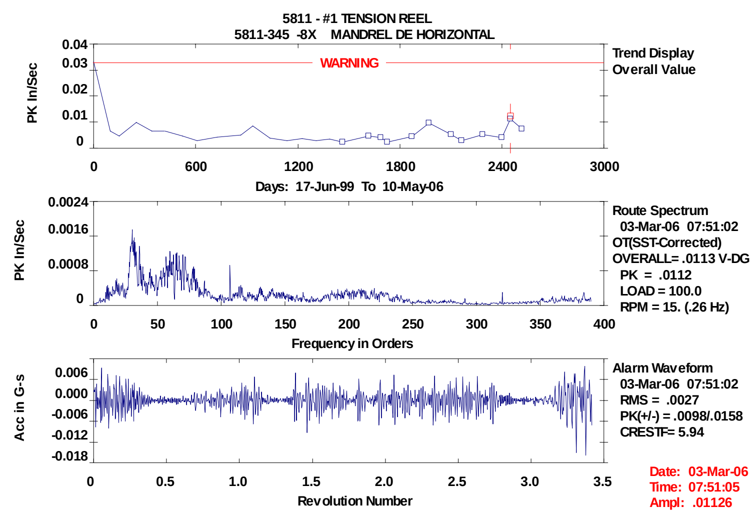
<u>Fatigue without any mitigating circumstances such as an inclusion or a gouge is a distinct possibility and the other mandrel shafts of this design are suspect.</u>

PREDICTIVE METHODS

Could this failure have been predicted? Crack detection in turbo-machinery is a well defined science. Unfortunately, the high energy levels that allow detection are lost in low speed machines. This problem becomes even more difficult in the Fairless Pay off and Tension Reels at Fairless. As noted in numerous reports, the wear plates on the pusher create dust that gets into the slider mechanism and soaks up the lubricant. This creates a stiction phenomenon in which the entire sled does not move freely. It sticks while tracking and jumps when the hydraulic cylinder pressure becomes high enough to overcome the sticking. As a result, it jumps during vibration data collection and produces signals that look like impacting. This problem is most significant on the Tension Reels but is also seen on the Pay Off Reels.

The top plot shows the latest vibration data for the bearing closest to the failure. The spikes in the time waveform (bottom) may have been due to the fatigue cracking but were attributed to the stiction. The bottom plot is from Tension Reel #1. Note how a similar pattern exists here.





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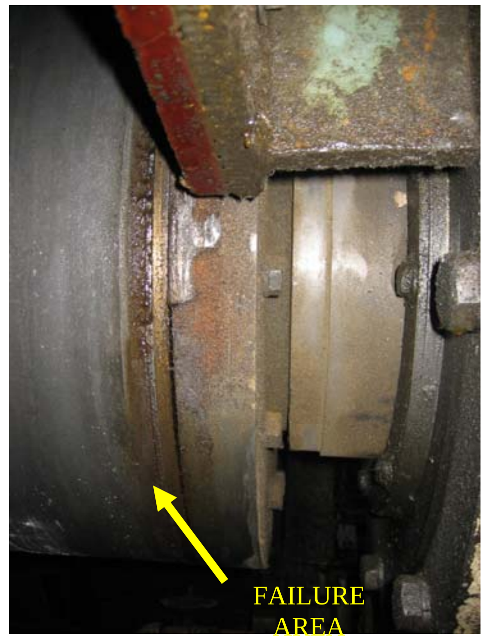
OTHER DETECTION METHODS

A simple visual inspection was unlikely to identify any problem. Most of the mandrel shaft is enclosed in either the gear case or under the reel head. The failed area is near the only exposed portion, but still behind

There are a variety of non-vibration methods to check for suspected cracks. The most practical for the mandrels would be ultrasonic inspection. This is because it requires the least disassembly. It is the same method that is used in many roll shops.



It works well in this application because the end of the mandrel is easily exposed, simply remove the reel head. Most gearboxes or other coupled shafting would require significantly more disassembly for this test.

This picture shows the area that failed on the sister reel. The actual failure took place on the shaft to the left of the head clamping plate in the center. As can be seen (or not) this area is inaccessible to simple inspection.

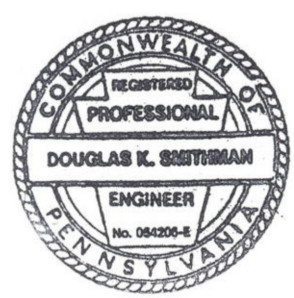
Ultrasonic crack inspection should be performed on the other reels on a semi-annual basis. This should be adequate to identify any problems before they result in a similar failure.

CONCLUSIONS

1. The mandrel failure was a case of metal fatigue.
2. The other mandrels are subject to similar failures.
3. Predictive maintenance methods currently in use at Fairless will not detect these failures.
4. Ultrasonic testing of the mandrel shafts is recommended.

Douglas K. Smithman, P.E.

July 1, 2006



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MANDREL STRESS CALCULATIONS

Given:	USS Drawing #s:	E-4-14093
		E-4-14082
	Coil Dia	= 74”
	Eye Dia	=21”
	Strip Width	=64”
	Coil Wgt	=70,000#
	S_{ut}	=125,000 psi
	S_{yt}	=100,000 psi

Find: Was the mandrel design adequate in terms of fatigue?

Solution:

Method: The endurance limit of the shaft will be determined and compared to the operating stress in the failure area.
The endurance limit will be determined using methods outlined in MECHANICAL ENGINEERING DESIGN, 3rd edition, by Joseph Shigley.
All calculations will be for the worst case; i.e., the largest coils and no allowance for weight and stress reduction as the coil pays out.

Endurance Limit – the endurance limit is the strength modified to account for several mitigating factors

$$S_e = k_a k_b k_c k_d k_e S_e^{'}$$

Each factor will be determined below.

k_a – surface factor (Fig 5-17, pg 189) for a machined surface

<u>k_a = 0.72</u>

k_b – size factor (pg 190)

<u>k_b = 0.75</u>

k_c – surface factor (Table 5-2, pg 192) for 99% reliability

<u>k_c = 0.814</u>

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k_d – temperature factor (pg 193) for less than 160^0 F

$\underline{k_d = 1.00}$

k_e – stress concentration factor

The failure occurred very close to a shoulder in the shaft. The stress concentration from this discontinuity is determined through a series of steps using data from cited figures.

$$\left.\begin{array}{l} \dfrac{D}{d} = \dfrac{14.375}{12.75} = 1.127 \\[2em] \dfrac{r}{d} = \dfrac{0.125}{12.75} = 0.01 \end{array}\right\} \quad k_t >= 3.0 \qquad \text{(Fig A-25-9, pg 667)}$$

$q = 0.95$ (Fig 5-20, pg 195)

$k_f = 1 + q(k_t - 1)$ (eqn 5-38, pg 194)
$= 1+0.95(3.0-1)$
$= 2.9$

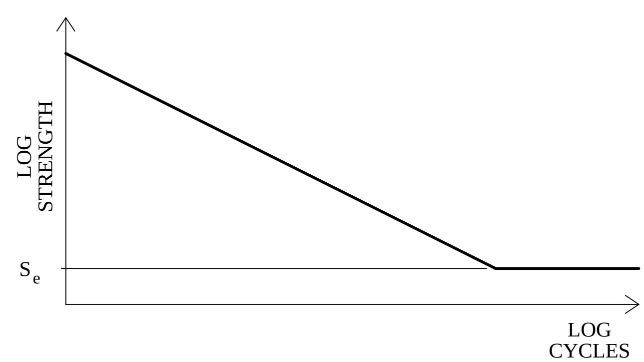
$k_e = 1/k_f$
$=1/2.9$
$\underline{=0.34}$

$S_s{}' = 0.5S_{ut}$
$=0.5(125,000)$
$=62,500$ psi

$S_e = 0.72(0.75)(0.81)(1.00)(0.34)(62,500)$

$\underline{=9,300 \text{ psi}}$

This is the endurance limit, the ‘knee’ in the S-N diagram. If the operating stress is greater than this value then fatigue will become a concern.



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Operating Stresses

There are three stresses of concern in this problem

1. Shear from the dead weight
2. Torsion form the strip tension
3. Bending from the cantilevered weight

Shear Stress

$$\sigma = \frac{F}{A}$$

$$Area_{@\,failure} = \frac{\pi}{4}(D^2 - d^2) = \frac{\pi}{4}(12.75^2 - 4^2) = 115.1 in^2$$

$$\sigma = \frac{F}{A} = \frac{70{,}000}{115} \underline{= 609\,psi}$$

Torsion Stress

$$\tau = \frac{Tc}{J}$$

T=Torque from strip tension = 74/2 (2100) = 77,700 in-lb

$$J = \frac{\pi}{32}(D^4 - d^4) = \frac{\pi}{32}(12.75^4 - 4^4) = 2569 in^4$$

$$\tau = \frac{Tc}{J} = \frac{77{,}700 in - lb(12.75 in / 2)}{2569 in^4} \underline{= 192\,psi}$$

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Bending Stress

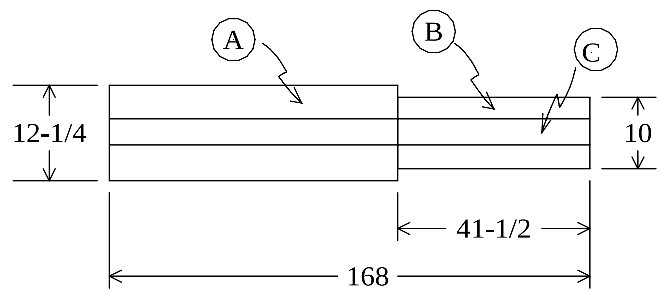
$$\sigma = \frac{M}{I/c}$$

$$I/c = \frac{\pi d^3}{32} = \frac{\pi(12.75)^3}{32} = 203in^3$$

Determining the bending moment at the point of failure will require some preliminary calculations.

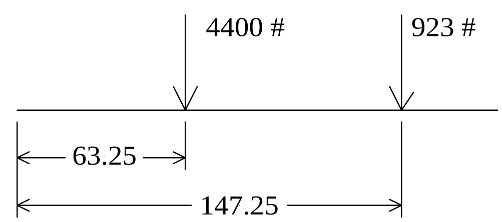
Weight and center of gravity

	Area	Vol	Wgt
A	123	15,560	4400
B	78.6	3260	923
C	16.0	2688	(760)
			4563



Round the weight to 4600 lb

$$\frac{63.25(4400) + 147.25(923)}{(4400 + 923)} = 77.8in$$



Determine the bearing reactions

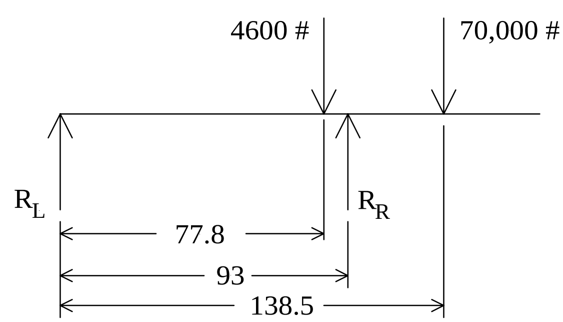
$\Sigma F = 0$

R_L – 4600 +R_R-70,000 = 0

R_L = 74,600 - R_R

R_L = 74,600 – 108,100 (from moment calc)

<u>R_L = -33,500 lb (and is acting DOWN)</u>



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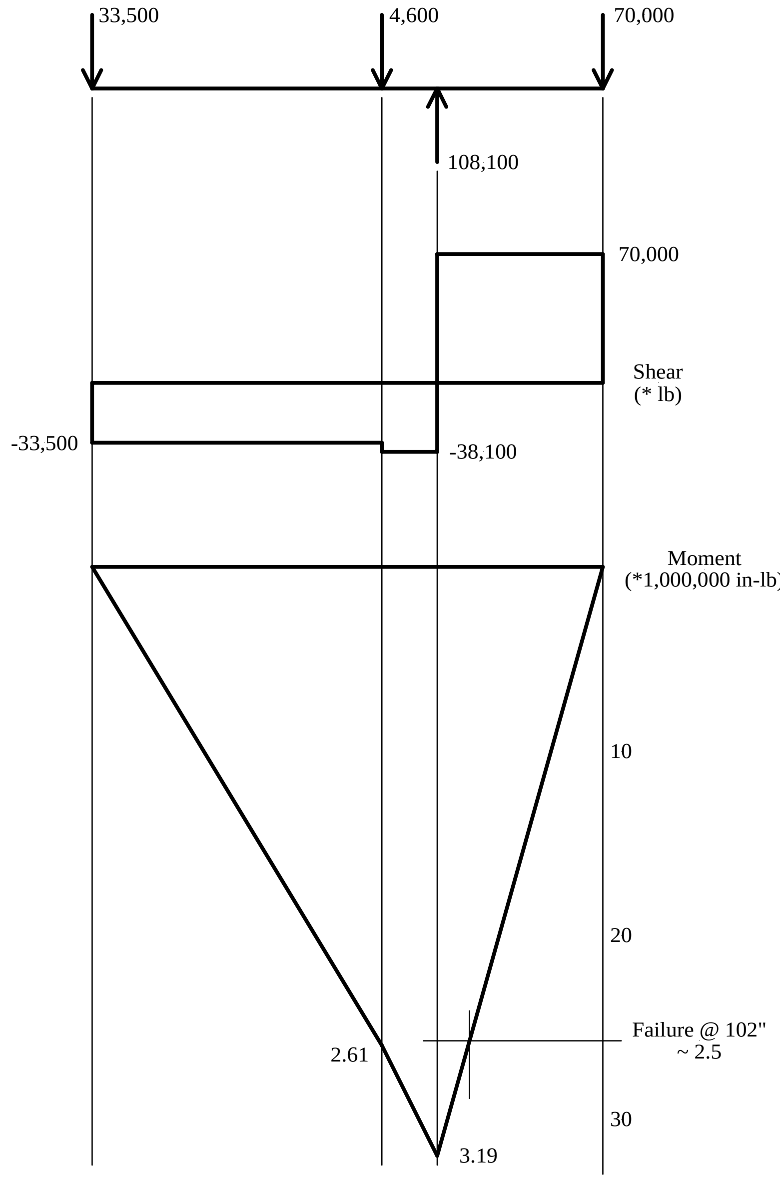
$\Sigma M_L = 0$ (CCW is positive)

$-4600(77.8) + 93R_R - 138.5(70{,}000) = 0$

$93R_R = 357{,}900 + 9{,}695{,}000$

$\underline{R_R = 108{,}100 \text{ lb}}$



Determine the moment at the failure point, 102 inches from the left end.

Graphically gives confidence that the failure occurred near the point of greatest bending stress. It is notable that the failure also occurred at the smallest section in this area.

Graphically the moment at the failure point is about:

$M_{102"} = 2.5\ (10^6)$ in-lb

An analytical check is:

70,000 (138.5-102) = 2,555,000

Solving for bending stress:

$$\sigma = \frac{M}{I/c} = \frac{2{,}500{,}000 in - lb}{203 in^3} = \underline{12{,}300 psi}$$

Because the bending stress is greater than the endurance limit; i.e.,

12,300 > 9,300

<u>the mandrel design is susceptible to fatigue failure.</u>

It should be noted that the total stress experienced will include contributions from the shear and torsion stress but that both of these were small compared to the bending stress and therefore negligible.

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It should also be noted that this calculation assumes all stress cycles are with the maximum size coil. This is obviously NOT the case. The stress will go down as the coil pays off. Also many coils are not full size. Regrettably, the distribution of loads over more than 40 years of service would be at best a swag, so worst case is assumed.

Finally, the determination of the endurance limit uses a reliability factor of 0.99. The affect of increasing reliability is to reduce the factor. As an example, 99.9% reliability uses a factor of 0.753. This reduces the endurance limit to 9,300 psi.

Conclusion:

The mandrel design is marginal in terms of fatigue.

Recommendation:

Other mandrels of this design should be checked for signs of fatigue damage. A periodic testing program should be established.