MODIFIED INTERNAL CHAMFER COUPLING

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ABSTRACT

Currently, API specs on 7/8" couplings allow for a wide variation in coupling face widths. The face width is critical when trying to achieve good rod makeup. For example, a larger face width is less likely to rotate due to the larger surface area i.e. more friction. As a result of the high number of rod pin-coupling failures, especially in the 7/8" section, a coupling with a more effective face width was developed.

INTRODUCTION

Over the last two years (2000-2001), sucker rod connections have accounted for 53% of Yates Petroleum Corporation (YPC) rod failures. The 7/8" pin-coupling connection failures account for 50% of all YPC pin-coupling connection failures. As in the rest of the oil industry, pin breaks are the most common failing component in the connection. The coupling is commonly overlooked as a possible cause for pin coupling connection failures. This oversight is probably due to two factors: 1) there are fewer failures in the coupling than in the pin 2) the outside diameter of the coupling is much larger than that of the pin placing it within the Goodman Diagram stress range.

MAKE UP PRACTICES

Normally, the pin-coupling connection is made-up according to API 11BR specifications and to the rod manufacturer's circumferential displacement (CD) values. When these practices and values are applied properly, the results are longer run times (excluding external forces like fluid pound, sticking plungers, etc.). However, even with proper rod make up procedures, YPC had a higher than expected rate of 7/8" pin failures.

VARIETY OF 7/8" FHSM COUPLINGS

Failure analysis of 7/8" pin breaks combined with couplings rejected for parallelism led to the discovery of the variety of 7/8" full hole spray-metal couplings on the market. Five 7/8" full hole spray-metal (FHSM) couplings are shown in Figure 1. All of these couplings meet API specifications (API 11B, page 10, Table 12) Figure 2. As shown in Figure 1 all have different face widths that vary from 0.043 to 0.159 inches. Those couplings with less surface area (i.e. less friction) will have a greater tendency to loosen and eventually break.

IMPROVEMENTS TO THE SUCKER ROD COUPLING

Sandia National laboratories report #1652, identified three regions as having a high potential for fatigue damage and subsequent failure: the pin neck, the first engaged pin thread and the root of the last engaged coupling thread. Figure 3. These locations correspond well to connection failures observed in the field.

To improve the pin-coupling connection, the report recommended (1) provide sufficient preload to keep the pin and coupling mating surfaces together and resist rotational motion, (2) improve the fatigue resistance of the pin and coupling by increasing the pin-coupling stiffness ratio (3) decrease the severity of the stress concentrations in the areas identified as potential fatigue failures.

Utilizing the Sandia report information, an improvement needed to be made to the 718" rod pin-coupling connection (API or Non-API) without major modifications and expense. The 7/8" FHSM coupling was chosen for the modifications because the coupling requires less modifications and minimal adjustments to the machinery. The following changes were made to the 7/8" FHSM coupling: (1) the mating surface (face width) of the coupling was enlarged by reducing the dimensions of the inside chamfer Figure **4** and (2) the dimensions of the outside chamfer were fixed thus creating a more effective mating surface (face width). Figure 5. The effective face width was increased anywhere from 23% to 55% over the current manufactured couplings.

FIELD RESULTS

At the printing of this paper, 1023 MIC couplings have been installed. The MIC couplings were installed in nine wells with a variety of well characteristics Figure 6. The couplings were installed on new and reran rods. The rod grades were high strength (HS) and grade D. The pulling unit rod tongs were used for rod make up and the torque pressures required to achieve the recommended circumferential displacement (CD) were recorded. A summary of the torque pressures is listed below.

	Torque Range	Avg. Torque
• MIC couplings with new HS rods	625 psi to 900 psi	750 psi
• MIC couplings with rerun HS rods	650 psi to 850 psi	760 psi
• MIC couplings with new grade "D" rods	650 psi	650 psi
• API 718" FHSM couplings with rerun HS rods	600 psi	600 psi
• "Bullet Nose" API 718" FHSM with rerun HS rods	570 psi	570 psi

An increase of 20% torque pressure was observed on the MIC coupling over the API coupling and 24% over the "Bullet Nose" with reran HS rods. A summary of torque pressures by well is listed in Figure 7.

SUCKER ROD CONNECTION ANALYSIS

The analysis is currently in progress and no results were available for publication at the printing of this paper. The proposed analysis is listed in Figure 8.

OBSERVATIONS

- Different torque pressures were observed with new rods, rerun rods and rod grades.
- The torque pressure values, CD, on the top and the bottom of the coupling were very consistent.
- Breakout torque pressures as high as I200 psi were observed with no connection deformations.
- No problems have been encountered with the reduced inside chamfer.
- No pin breaks or connection deformations as a result of the increased torque pressures.

CONCLUSIONS

- The API 11BR Specifications Table 12 "Pin and Box Contacts" needs to be revised to include a column for the 7/8" full hole coupling dimensions.

ACKNOWLEDGEMENTS

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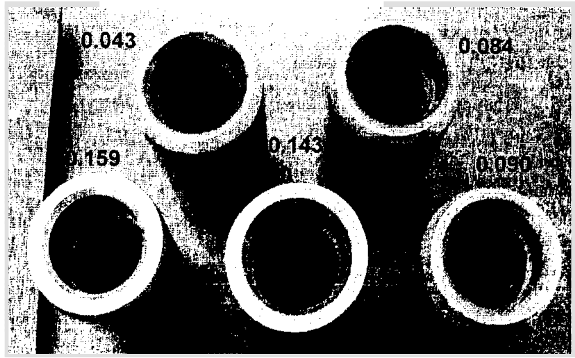


Figure 1- Coupling Face Widths

10 API RECOMMENDED PRACTICE 11B

Table 12—Pin-and-Box Contacts

Nominal Size of Rod	$^5/_8$ (15.9)	$^3/_4$ (19.1)	$^7/_8$ (22.2)	1 (25.4)	1$^1/_8$ (28.6)
Nominal Diameter Of Thread	$^{15}/_{16}$ (23.8)	1$^1/_{16}$ (27.0)	1$^3/_{16}$ (30.2)	1$^3/_8$ (34.9)	1$^9/_{16}$ (39.7)
Outside Diameter of Pin Shoulder and Box D_f +0.005 (+0.127) −0.010 (−0.254)	1.250 (31.75)	1.500 (38.10)	1.625 (41.28)	2.000 (50.80)	2.250* (57.15)
Minimum Major Diameter of Contact Faces D_c[1]	1.177 (29.90)	1.427 (36.25)	1.552 (39.42)	1.865 (47.37)	2.110 (53.59)
Minor Diameter of Contact Faces, D_m +0.015 (+0.380) −0.000 (−0.000)	1.110 (28.19)	1.253 (31.83)	1.378 (35.00)	1.566 (39.78)	1.753 (44.53)
Minimum Face Width C_f[2]	0.026 (0.66)	0.080 (2.03)	0.080 (2.03)	0.142 (3.61)	0.171 (4.34)

Notes:
All dimensions in inches followed by equivalent in mm.
See Figures 5 and 6.
Limits for pin shank diameter are the same as those for the major pin diameter, see Table 9.
*±0.015 (±0.38)
[1] $D_c = (D_f)$ min. −2 (Chamfer or Rad. On D_f) max.
[2] $C_f = (D_c - (D_m)$ max)/2

8.5 THREADS

The threaded portion of sucker rod shouldered connections and polished rod pins (9 degree cone) shall be 10 threads per inch and conform to the unified thread form with Class 2A-2B tolerances and allowances, as defined in ANSI/ASME B1.1. The design profile of the pin thread is type UNR with rounded root contour as shown in Figure 7. The thread profile of the box thread is type UN having a flat root contour with a permissible round root contour beyond the 0.25 x pitch (0.25p) flat width to allow for crest wear as shown in Figure 7. As indicated herein, sucker rod threads are straight threads (see Figure 5); polished rod threads are straight threads with the imperfect pin threads on the vanish cone (see Figure 6).

9.1.2 Calibration

9.1.2.1 Measurement standards such as thread wires gauge blocks used to calibrate equipment in 9.1.2.3 sha checked and approved at least once a year by an out agency with traceability to the National Institute of Stand and Technology, Gaithersburg, Maryland (NIST). M: gauges shall be checked and approved at least once every years of use by an outside agency with traceability to NIS

9.1.2.2 Working gauges (such as thread gauges) sha calibrated at least once per month of use. A set of wor gauges for both box and pin elements shall include the lowing as a minimum.

Figure 2

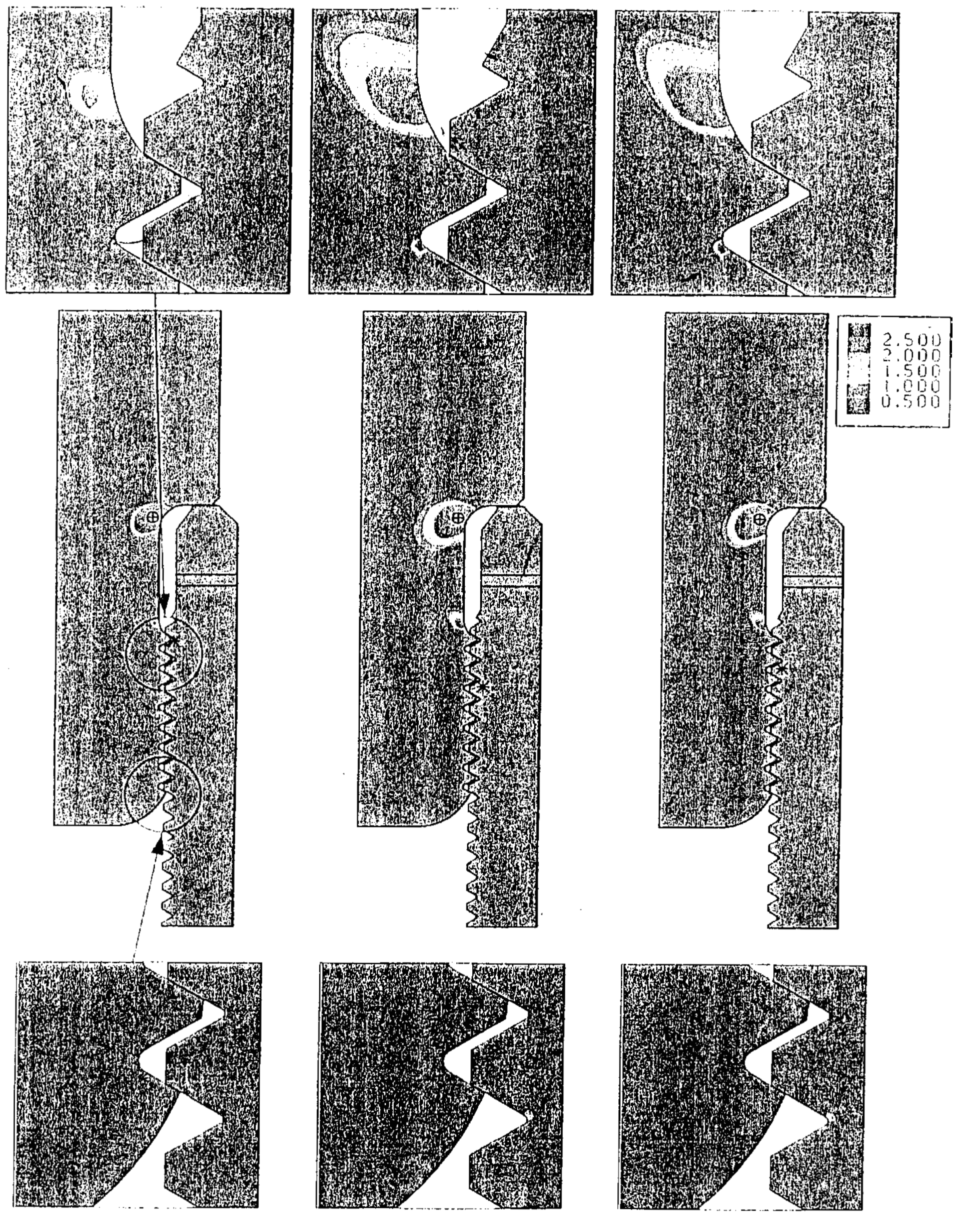
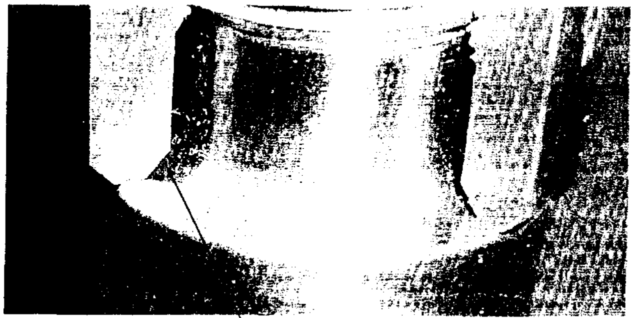


Figure 3



INSIDE CHAMFER

Figure 4

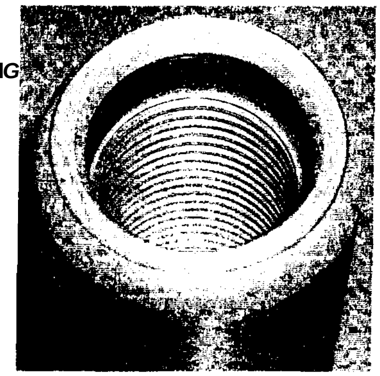


Figure 5

Well Characteristics

Well Name	Production BFPD	% Water	Pump Size	SL	SPM	Run Time	FAP %feet	Rod String (#of rodds) 1.0	7/8	314	SB	GRADE
1	73	75	1.25	167	6.5	50	77	100	206	-		T-66
2	123	10	1.50	64	10	75		-	133	170	5	D
3	154	86	1.50	146	8.4	100	754	109	114	80	8	T-66
4	143	99	1.50	165	8.1	70	641	-	140	162	10	T-67
5	95	81	2.00	146	7.4	80	396	72	95	135	12	T-68
6	15	17	1.50	169	8.25	100	40	-	160	265	5	T-69
7	58	93	1.50	144	7	73	349	101	101	108	6	N-97
8	235	94	1.50	144	7.5	100	228	10	128	170	-	T-66
9	417	99.8	1.75	192	7.75	100	234	97	104	105	9	T-66

Figure 6

FIELD SUMMARY

Date Well Installed	Name		Avg. Tourque
11/06/01 –	**Well #1**	51 cplgs, on rerun 7/8" T66 MU torque – 775 psi (had breakouts of 1000 psi) i Regular API cplgs, MU torque - 600 psi	775 ps
11/10/01 –	**Well #2**	138 cplgs, on new 7/8" grade D MU Torque – 650 psi	650 psi
11/12/01 –	**Well #3**	50 cplgs, on rerun 7/8" T66 MU Torque – 850 psi, Started @ 600 psi inc. to 750 to 850 to 900 psi Broke out previous couplings (all had small face widths) MU Torque (on these couplings) – 570 psi	850 psi
11/20/01 –	**Well #4**	191 cplgs; 49 on new 7/8" T66 (replaced, body failure) 51 cplgs on new 7/8" T66 91 cplgs on rerun 7/8 T66 MU Torque (Range 600 to 800 psi) Avg. 650 psi Started out at 800 psi drop to 600 psi	625 psi 650 psi
11/20/01 –	**Well #5**	95 cplgs, on new 7/8" T66 MU Torque – 700 psi Started out @ 800 psi – leveled out @ 700 psi	700 psi
12/03/01 –	**Well #6**	166 cplgs on new 7/8" T66 MU Torque – 775 psi	775 psi
12/28/01 –	**Well #7**	100 cplgs onl rerun 7/8" Norris 97 MU Torque – 625 psi	625 psi
01/08/02 –	**Well #8**	128 cplgs on new 7/8" T66 MU Torque – 900 psi	900 psi
01/08/02 –	**Well #9**	104 cplgs on new 7/8" T66 MU Torque – 750 psi (Had breakouts of 1100 psi)	750 psi

Total MIC Couplings – 1023

	AVG TORQUE
New T66 Torque 625 to 900 psi	750 psi
Rerun T66 Torque 650 to 850 psi	760 psi
New Grade D Torque	650 psi
Rerun Norris 97's	625 psi
Reg. API on rerun T66	600 psi
"Bullet Nose" API on rerun T66	570 psi

Figure 7

SUCKER ROD CONNECTION ANALYSIS PROPOSAL

1.0 INTRODUCTION

The sucker rod connectors will be analyzed by finite element analysis to determine the peak stresses in the assembly. Three basic sizes will be evaluated ¾", 7/8", and 1". The ¾" and 1" pins will be evaluated with two different couplings. Three different coupling will be included in the 7/8" evaluation. Nominal and low makeup conditions will be considered. The variation of the peak stress with tension will be determined for each preload.

Linearly elastic finite element analyses will be performed to determine the stresses in the connectors. The finite element model mesh will be fine enough to obtain peak stresses. Stresses parallel to the surface will be used to determine the peak stresses in the components.

The preload at the shoulder will be monitored to verify that preload is not lost at the maximum load. If preload is relieved, the load required to relieve preload will be determined. The torque required to achieve the desired preload will be estimated based on an assumed coefficient of friction.

The maximum stress concentration factor (SCF) will be determined

2.0 ANALYSIS DETAILS

Two general load cases will be included in the analysis makeup and makeup plus axial tension. The axial load will be applied in increments to the assembled connector model as a uniform pressure on the rod section at one end of the model. The center of the coupling at the other end of the model will be restrained in the axial direction. A summary of the geometry combinations follows:

¾" with Coupling 1
¾" with Coupling 2

7/8" with Coupling 1
7/8" with Coupling 2
7/8" with Coupling 3

1" with Coupling 1
1" with Coupling 2

All seven geometry combinations will be evaluated for the following load cases:

- Low Makeup
- Low Makeup + Tension Ramped to Maximum Tension
- Nominal Makeup
- Nominal Makeup + Tension Ramped to Maximum Tension

One geometry combination will be evaluated with no preload and a lateral load on the coupling.

Figure 8 - Page 1

An axisymmetric finite element model will be used to determine the stresses in the connector. FThe ABAQUS, general purpose, finite element program will be used. The element types that will be used for the analysis are:

1. Solid elements to model the pin and box. ABAQUS CAX4 (four-node axisymmetric elements) elements will be used to model the pin and the box.

2. Contact pairs, that allows the transfer of load from the pin to the box in the direction normal to the contact surface, will be used at all contact surfaces. The contact pairs model the interaction between two deforming axisymmetric bodies that may contact along their boundaries. The interface elements can be initially preloaded, if the initial geometry of the model is such that one component overlaps the other, as is the case with the made up connectors. The solution output consists of the contact pressure between parts.

3. Axisymmetric shell elements will be used to act as strain gauges in the radii of all thread roots as well as along the inner and outer diameter of the pin and box. The shell elements will have the same elastic properties as the pin and box. Since a thickness of 0.00001 inches will be specified for the shell elements, the shells will not add any additional stiffness to the model. The shell element output will be used to obtain the surface hoop stresses and the surface stresses parallel to the surface.

The axisymmetric elements will be changed to axisymmetric elements with asymmetric deformation for the lateral load case.

The assumptions which will be made in the analysis are:

1. All mating surfaces are frictionless. A coefficient of friction of 0.0 will be used for all finite element solutions.

2. Small strains and rotations

3. The material is elastic.
   Modulus of Elasticity = 30,000,000 psi
   Poisson's Ratio = 0.3

An axial interference will be specified at the external shoulder to simulate makeup. The interference will be adjusted to produce the required average stress in the pin. The makeup torque that corresponds to a given axial interference in the model will be calculated from the summation of the normal forces in the connector times the radius at which they act times a friction factor.

Shell elements will be used to obtain the peak stresses in the connector. Peak stresses will be obtained at all significant discontinuities in the connector. The stress concentration factors will be calculated for all critical locations in the connector for each load. Since the stresses and stress concentration factors at any location in the connector may depend on the connector preload, it is necessary, therefore, to determine the SCF's at different load levels.

Figure 8 - Page 2