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Testing Techniques for New Style Fittings

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Testing Techniques for New Style Fittings

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ABSTRACT

This paper will discuss testing methods for a new style of fittings. New and revolutionary designs in fittings, sometimes, do not apply to stated standards and protocol for testing. In these situations a "hybrid" testing protocol is used to validate the hydraulic conductors and fittings. Existing standards and specifications will be examined to show how the protocol is not applicable, and recommendations made to adequately cover the intent of the official testing protocol.

INTRODUCTION

With higher reliability and pressure requirements hydraulic fittings have been evolving to meet these new demands. This creates some issues with current testing protocol established by industrial standards committees across the world. The intent of the protocol is clear, however the means to accomplish validity is sometimes in question. This paper addresses some of these issues and applies it to a broad spectrum of safe evaluation and testing methods to qualify the components being tested. One particular style of fitting will be addressed in this paper, however the techniques used will be applicable to other sealing components.

This paper is based on the testing of a new revolutionary product manufactured by EPCO Products, Inc. located in Fort Wayne, Indiana. The fittings have a unique design that use a standard SAE ORB J-1926 port. However, the fitting (plugs used in testing) incorporates a seal made by engaging the standard taper at the mouth of the port with a corresponding taper under the head of the fitting to form a metal-to-metal seal at the point of contact. An O-Ring located below the taper on the fitting acts as a back up seal. This new, no-leak design did not lend itself to standard torque values stated in SAE specifications. This product required its own torque specifications (see Figure 1 below).

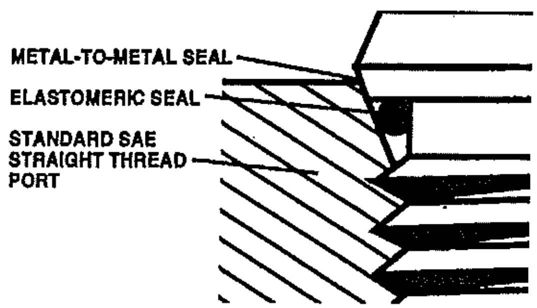

Figure 1.
Zero-Leak Gold® Fitting Cross Section
(Zero-Leak Gold® is a trademark of EPCO Products, Inc.)

The paper will use excerpts (pictures, data tables, and calculations) from the original formal report given to EPCO Products Inc.

NEW STYLE FITTING

O-Rings have been used extensively in hydraulic fittings as a primary means of containing pressurized fluid. Many new designs however utilize the o-ring as a secondary seal in the design (see Figure 2). This has posed a problem testing the new style fittings to published industry standards.

Problems arise with torque values and o-ring compression data that effect all testing protocol.

The new style fitting uses a metal-to-metal contact as the primary seal with the o-ring as the back up seal (See Fig. 1).

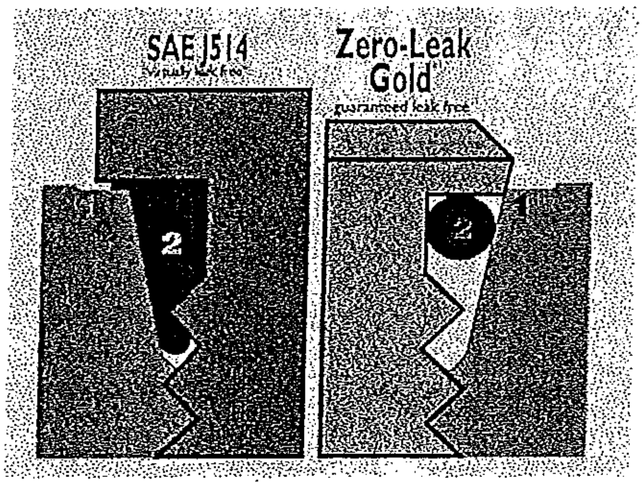

Figure 2. Difference From SAE J514 to New fitting Design.

The concern with this fitting is the metal-to-metal contact area. This contact area will exert forces to the port caused by the torque of the fitting. This area must remain in the linear region of the stress-stain diagram for both materials being used i.e. the fitting and the port. The dual seals also would need torque values established because listed SAE torque values pertain to single o-ring primary sealing fittings. For this particular fitting design the force over the contact area was calculated and a maximum torque value established. The torque value calculated was a function of the material. The values were calculated with the following relation:

$\sigma = \frac{F}{A}$ (1), where the torque is defined as $T = .2Fd$ (2), d is the screw size of the fitting, and A is the cross sectional area at the location where the load is to be applied for the calculation. Solving equation (2) in terms of Force and substituting back into equation (1) yields the result, $\sigma = \frac{T}{.2Ad}$ (3). Solving equation (3) for the Torque, T would result in the following equation, $T = .2\sigma Ad$ (4). This value is again the maximum torque and is not necessarily the torque one would want to start with. A table should now be made to define the maximum torque values for the geometry and material of the fitting identifying the weakest feature of the fitting as well the limiting torque of the fitting. The Table would be as follows:

Table 1.
Torque Values for Fitting Geometry

Fitting Size dash number	Maximum Torque Values			
	Thread Shear (ft lb)	Shoulder Compression (ft lb)	O-Ring Neck Tension (ft lb)	Socket Shear (ft lb)
-4	Eqn. (5)	Eqn. (6)	Eqn. (7)	Eqn (10)
-6	Eqn. (5)	Eqn. (6)	Eqn. (7)	Eqn (10)
-8	Eqn. (5)	Eqn. (6)	Eqn. (7)	Eqn (10)

The values are shear of the minor diameter equation (5) $\sigma_{shear} = \frac{T}{(.2)Ad(.85)}$ (5) where the area is $A = \frac{\pi}{2} L_e D_{minor}$, the compression of the shoulder contact area equation (6) $\sigma_{compression} = \frac{T}{(.2)dA}$ (6) where the area across the taper is $A = \frac{\pi}{4}(D^2_{outside} - D^2_{inside})$, the necked down area were the o-ring is installed equation (7) where the area is of the section is $A = \frac{\pi}{4} D^2_{o-ring\phi}$, and finally the value the socket will fail on a Hexagonal headed socket equation (8). The hexagonal head calculations can be important see Figure 3.

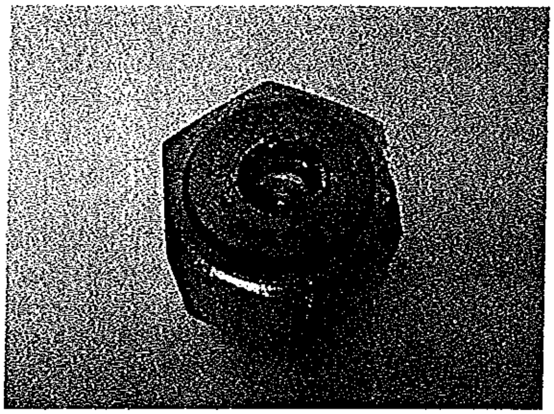

Figure 3.
Stripped out hex head plug.

The calculations for the socket strength are the depth of the socket multiplied by the width times 6 surfaces $\sigma_{Yield} = \frac{T}{.2 \times d \times A \times 6 \times .8}$ (8), where $A = L_W L_D \times 6sides$ (9) and d was defined earlier. Solving equation (8) for T the equation would be $T \leq \frac{\sigma_{Yield}.2Ad \times 6}{12}.80$ (10) the answer will be in foot-pounds. The assumption here is 80% contact area with the hex head bit.

TESTING THE FITTING

The test protocol can now be determined to begin this evaluation. The testing to be considered for this evaluation will be the proof test and impulse test outlined in both SAE and NFPA standards. The starting torque value will have to be selected. It must be understood that the value must pass proof testing as well as the impulse testing. The proof test is typically two times the working pressure for 1 minute and is a relatively inexpensive and short test. The burst test involves testing a specified number of components that must withstand four times the working pressure rating to pass (4:1 safety factor). The more risky protocol, both in expense and time is the fatigue impulse test. This test subjects the component to one million cycles at a prescribed cyclic test pressure (CTP). One million cycles at 1 Hz requires about 12 days to complete with the impulse chamber running 7 days a week 24 hours a day. This test is more expensive and to pass the fitting cannot fail in one million cycles. The torque selected must first be proof tested and then placed in the impulse fatigue chamber for one million cycles. If the fitting passes the proof, burst, and fails the impulse prior to completing one million cycles the fittings tested must be discarded and the testing started all over.

TESTING PROTOCOL

When beginning the test protocol, read the current standards available and determine which sections pertain to the fitting being tested. In the case of this paper the only sections that did not pertain was the o-ring compression values and the torque values of the fittings. The test protocol followed for this evaluation was NFPA/T2.6.1R2-1999, "Fluid power components – Method for verifying the fatigue and establishing the burst pressure ratings of the pressure containing envelope of a metal fluid power component." Since this NFPA document did not address torque values the SAE torque and proof values for o-ring boss fittings were used as a reference. The plugs were installed into (6) specially designed manifolds. The manifolds tested were made from standard industry Ductile Iron and Aluminum. Three manifolds were made of each material. The (6) manifolds were designed to install (8) different fittings allowing (3) samples of each size to be mounted. Two manifolds were designed with (3) ports on (3) sides and manifold # 3 had (2) ports on (2) sides. The manifolds allowed testing (3) samples of each SAE J1926 port sizes #2, #3, #4, #5, #6, #8, #10, and #12 on (3) manifolds in both Aluminum and Ductile Iron. The rated working pressure of Ductile Iron is 5000 psi and 3000 psi for aluminum. The project goal was to establish torque values that successfully pass the 1,000,000 cycle Endurance Test specified in paragraph 8 of NFPA /T2.6.1 R1-1991.

The SAE published values for torque were used as a bench mark for this test. To determine torque values for the plugs an iteration procedure was established. Torque values of 25%, 40%, 60%, 70% and 90% of SAE numbers would be tested in succession until successful completion of the NFPA 1,000,000-cycle Endurance Test.

For each test, the torque value was applied to the plugs and the manifold assemblies were plumbed into the impulse chamber and the Rated Fatigue Pressure Endurance Cycle Test was started (see Figure 4. Test Stand Picture and Figure 5 Impulse Circuit Schematic).

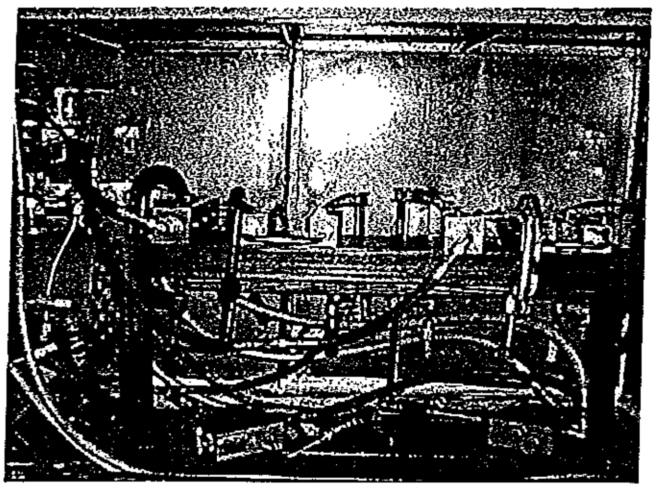

Figure 4.
Impulse/Fatigue Chamber

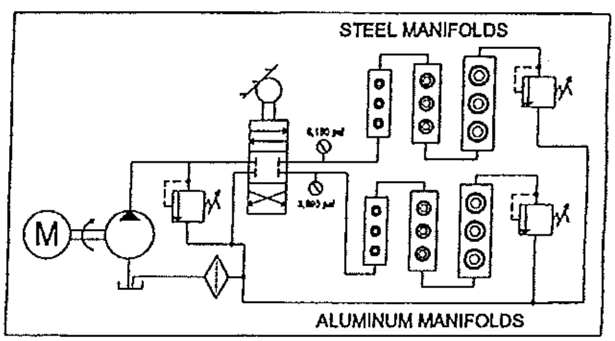

Figure 5.
Impulse Circuit Schematic

The Cyclic Test Pressure (CTP) applied was 3688 psi for Aluminum and 6156 psi for Ductile Iron. Pressure waveforms for the 1,000,000 cycle Endurance Test conformed to the characteristics specified by paragraph 8 of NFPA /T2.6.1 R1-1991 (see Figure 6. Aluminum Waveform and Figure 7 Ductile Iron waveform). The basic criteria for the wave shape is maximum rate of pressure rise must not exceed 50,000 psi/sec, the values were 49,363 psi/sec for Ductile Iron and 39,769 for Aluminum. The NFPA /T2.6.1 R1-1991, paragraph 8.4.4 states, "The Pulse Duration must be held to 100±10 milliseconds" and the tests ran at 92 milliseconds for Ductile Iron and 98 milliseconds for Aluminum.

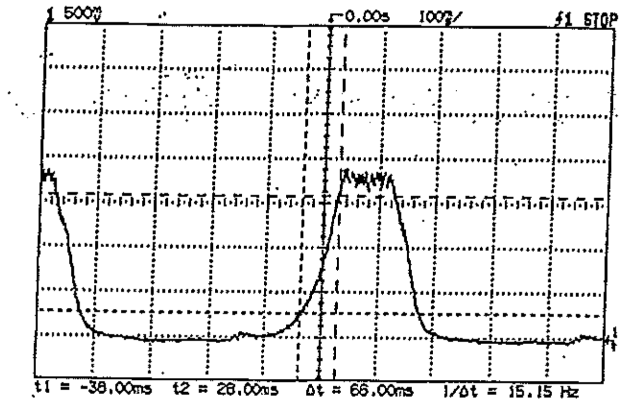

Figure 6.
Aluminum Waveform

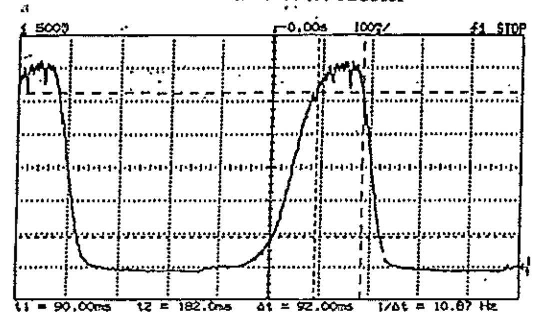

Figure 7.
Ductile Iron Waveform

Testing was started and if a failure occurred new O-Rings would be installed and the torque would be increased to the next value. This process was followed until all fittings passed the 1,000,000 cycle Endurance Test specified in paragraph 8 of NFPA

/T2.6.1 R1-1991. A failure defined by NFPA /T2.6.1 R1-1991 states " The inability to sustain a given load or to contain pressure in a pressure containing envelope". By the stated definition a leak would cause a failure. The initial test was done with 0.5 ft-lbs. torque applied to each plug. This torque was successful at sealing at static rated pressures, however it failed early into the endurance test.

Initial torque values were 25% of SAE values. There were (5) test torque trials (.5ft-lbs, 25%, 40%, and 2 trials at 60%) until the value of 60% of listed SAE values passed the 1,000,000 cycle Endurance Test (final values are listed in Table 2). It should be noted that the torque values selected must be lower than calculated in Table 1. to prevent permanent deformation of the parts.

Table 2.
Torque Values

EPCO ZERO-LEAK GOLD PLUG TESTING TORQUE VALUES		
SAE & Zero-Leak Gold plug	Torque (ft lbs)	
	MATERIAL	
Dash Size	Ductile Iron	Aluminum
#2	2	2
#3	3	3
#4	7	7
#5	9	9
#6	11	11
#8	28	28
#10	46	46
#12	51	51

The final tests conducted on the plugs were the Proof and Burst Tests. For these tests new Zero-Leak Gold plugs and O-Rings were installed. The Proof and Burst Tests were conducted in accordance with paragraphs 4.1 and 4.2 of SAE J1644 May93. Stated briefly "the Proof Pressure Test requires (3) samples to meet or exceed a ratio of 2:1 between proof and working pressure for 60 seconds minimum". The Burst Test requirement states " (3) samples be capable of withstanding the minimum of four times working pressure without failing". All plugs passed both the Proof and Burst Tests criteria (see Table 2 below).

ANALYSIS OF DATA

The 1,000,000 cycle Endurance Test concluded with no leaks from all of the plugs being tested. It was noted that during the 1,000,000 cycle Endurance Test that a slight, discernible amount of O-Ring material was observed at several test ports. This material was observed on SAE port size #10 for Aluminum and SAE #8 and #12 in Ductile Iron (see Figure 8).

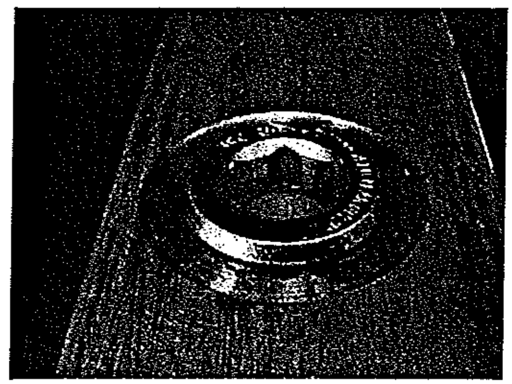

Figure 8.
Black Residue Stains Around Plug

Although this condition was noted, there were no leaks from the corresponding plugs. It should also be noted that the plugs did run continuously for 1,000,000 cycles (except for short shut downs for stand maintenance). The plugs passed the 1,000,000 cycle Endurance, Proof and Burst Tests at a value of 60% of SAE stated values with no leaks from any plugs (see Tables 1&2).

During the test O-Ring Ports were labeled and scribed with a scratch across the fitting leading into the manifold. From this mark it could be determined if any movement of the fitting in relation to manifold had occurred. There had been no movement from these marks during the test. Torque values were recorded after the test to check if values had changed (see Table 3). The torque removal values for the plugs were very high. From marks placed on plug and manifold there was no noticeable rotation of plugs and the plug threads were not distorted.

CONCLUSION

The plugs passed the Proof and Burst Tests conducted in accordance with paragraphs 4.1 and 4.2 of SAE J1644 May93, sections 4.1 and 4.2 and the NFPA/T2.6.1 R1-1991 standard titled " Fluid Power Systems and Products- Methods for Verifying the Fatigue and Establishing the Burst Pressure Ratings of The Pressure Containing Envelope of a Metal Fluid Power Component paragraph 8 titled Rated Fatigue Verification Test Program".

This paper examined testing a new style fitting that did not allow itself to be evaluated using current industry standards. It was identified that the maximum torque values had to be re-evaluated and experimentally verified following proof, burst and impulse testing requirements. The plugs tested passed testing at stated values and meet what industrial committees deem safe for use at working pressures of the fittings (plugs).

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