EXPERIMENTALLY VALIDATED MODELS OF O-RING SEALS FOR TINY HYDRAULIC CYLINDERS

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ABSTRACT
To enable simulation of tiny hydraulic systems, including predicting system efficiency, it is necessary to determine the effect of the hydraulic cylinder piston seal. For tiny cylinders whose bore is less than 10 mm, O-ring seals are convenient. Simplified models for the O-ring were used to describe piston leakage and friction and based on the models, the force and volumetric efficiencies for tiny cylinders were predicted for a range of steady state operating conditions. To validate the models, a test stand was constructed to collect experimental data for 4, 6 and 9 mm bore cylinders, which were in the form of a vertical ram with a single O-ring seal. The ram was fully extended and put under load. A needle valve was then cracked to cause the ram to descend at different speeds. Pressure, load and velocity were recorded and the data used to calculate cylinder efficiencies, which were then compared to model predictions. The model and the experiment showed essentially zero leakage. The experimental force efficiency had good agreement with the model over a range of operating conditions. The study showed that simple O-ring models for tiny hydraulic cylinders suffice for building system level simulation models.

INTRODUCTION
Large-scale hydraulic systems are well known for their high power density advantage compared to other technologies [1], which is why hydraulics are widely used in heavy-duty machines such as excavators. A recent study revealed that this power density advantage is maintained for tiny hydraulic systems [2], which makes hydraulic systems appealing for unthethered human-scale devices such as prostheses, orthoses and hand tools where high power and a small package are needed [3, 4].

Linear hydraulic cylinders are the most common actuators for a hydraulic system because they are simpler and far more efficient than rotary hydraulic motors. For most systems, cylinder efficiency is not of concern because the overall system efficiency is dominated by the efficiency of pumps and valves. In small-scale systems, however, cylinder efficiency matters because while cylinder force varies with area, cylinder loss varies with diameter. Thus cylinder efficiency models are needed at the small scale to fully understand the overall efficiency of tiny hydraulic systems.

Analytical efficiency models for small-bore hydraulic cylinders were developed in a previous paper [5]. It was shown that cylinder efficiency degrades as bore decreases, and that efficiency drops precipitously when the bore drops below 1 cm, reinforcing the need for efficiency models at small sizes. Since validated models for a clearance gap seal are readily available [6], the purpose of this work was to validate a simple model for a rubber seal.

CYLINDER EFFICIENCY MODEL
Leakage
The fluid leakage across the O-ring was modeled as

\[ q_l = 1.495 \cdot \pi \cdot B \cdot \mu^{0.71} \cdot U_r^{1.71} \cdot \sigma_m^{-0.71} \cdot \delta^{0.29} \]  

(1)
where \( B \) is the cylinder bore size, \( \mu \) is the fluid viscosity, \( U_r \) is piston speed, \( \sigma_m \) is the maximum O-ring contact pressure and \( s \) is the O-ring contact width [7, 8].

**Friction**

The cylinder force efficiency is mainly determined by the O-ring squeeze ratio, which indicates how tight the seal is. The higher the squeeze ratio, the tighter the seal and the smaller the leakage. A tighter seal, however, means more friction, which degrades force efficiency. Because of its importance, the O-ring squeeze ratio model is presented first.

The O-ring cross-sectional diameter after it is placed in the piston groove, but before the piston assembly is inserted into the cylinder block is

\[
d_s = d \cdot (1 - \delta)
\]

where \( d \) is the original O-ring cross-sectional diameter and \( \delta \) is the O-ring cross-sectional diameter reduction percentage, which can be read from a handbook chart [9]. The O-ring squeeze ratio \( \varepsilon \) is then be calculated as

\[
\varepsilon = \frac{d_s - (B - D_g)/2}{d_s}
\]

where \( B \) is the cylinder bore, \( D_g \) is the piston groove diameter and \( (B - D_g)/2 \) is the O-ring cross-sectional diameter after installation.

The O-ring friction force was modeled as

\[
f_s = \pi \cdot \mu_f \cdot B \cdot d \cdot E \cdot \varepsilon \cdot \sqrt{2 \cdot \varepsilon - \varepsilon^2}
\]

where \( \mu_f \) is the friction coefficient between the O-ring and the cylinder wall, \( B \) is the cylinder bore, \( d \) is the original O-ring cross-sectional diameter and \( E \) is Young’s modulus for the O-ring material [10]. For O-ring seals, \( \mu_f \approx 0.3 \) to 0.5 for well finished and sufficiently lubricated sealed surfaces [10] and a typical modulus for elastomeric seals is 10 MPa.

**Force Efficiency**

The pressure for an ideal cylinder is

\[
P_i = \frac{M}{A_p}
\]

where \( M \) is the load, and \( A_p \) is the piston area.

The actual pressure \( P_a \) in the cylinder is smaller than the ideal pressure because of the O-ring friction

\[
P_a = \frac{M - f_s}{A_p}
\]

Therefore, the cylinder force efficiency is

\[
\eta = \frac{P_a}{P_i}
\]

**METHODS**

**Test Apparatus**

Three sets of pistons and matching cylinder blocks were fabricated for validation testing with bore sizes 4, 6 and 9 mm (Fig. 1). For high precision, the pistons were machined from tight-tolerance precision ground rod. The cylinder block inner wall was brought to its final dimension using a reamer. Piston grooves were machined into the piston rod with dimensions according to [9], except that the piston groove diameter was intentionally made larger than that specified in the handbook to achieve 14% squeeze ratio as defined by Eqn. (3). Fillets with dimension 0.002” were cut on both the upper and lower portion of the groove to facilitate O-ring mounting. The rings were lubricated before mounting. The overall piston length was less than 10 times the bore to minimize rod bending [11] and an axial bearing was mounted at the top of the cylinder block to vertically constrain the motion of the piston and to minimize side loading on the seal.

![FIGURE 1: Pistons with O-ring seals and matching cylinder blocks](image)

The cylinder block was fixed to a rigid frame (Fig. 2). A loading block whose mass could be varied was suspended above the piston and was able to move up and down on a low-friction linear slide. The load block was carefully aligned to eliminate
side loading on the piston and piston seal. The hydraulic chamber under the piston head was connected by tubing to a needle valve that could be adjusted to control the speed of descent of the load pressing down on the other end of the piston rod as hydraulic oil passed through the valve and into a reservoir that was open to atmosphere. Another set of valves connected the cylinder to a small hydraulic axial piston pump whose purpose was to run oil from the reservoir into the cylinder, extending the piston and raising the load for the start of a test. During testing, the pump was disconnected from the circuit.

An analog output pressure gauge (PX309-300G5V, Omega Engineering) was connected between the cylinder and the valve to measure cylinder pressure and a linear potentiometer (LCP12Y, ETI Systems) was attached to the load to measure piston position. Pressure and position sensor signals were digitized using a USB data acquisition system (USB-6008, National Instruments). The piston force was determined by measuring the weight of the load block using a digital scale.

**Test Protocol**

The test protocol involved collecting data during a steady state descent of the load whose speed was determined by setting the needle valve. Test conditions covered a range of loads and a range of descent speeds. The advantages of using this protocol were that the motion was smooth compared to the flow ripple that results from pump-driven motion and that slow speeds could be attained through minimal cracking of the valve. An example cylinder pressure measurement during a load descent trial is shown in Fig. 3, which demonstrates essentially constant output. An example of slow speed motion is shown in Fig. 4 where the velocity is about 1 mm/s and the staircase profile of the position record is caused by ADC quantization.

A trial started by moving the load to its raised position using the pump after which the pump was disconnected from the circuit by closing the pump valve. Weights were added to the load block to reach the desired test condition. The needle valve was opened to the desired position and position and pressure sensors were sampled as the load descended. The position record was fit to a straight line to estimate piston velocity. The force on the piston was calculated from the load weight and the cylinder pressure from the pressure sensor. The corresponding O-ring efficiency for that test condition was calculated using Eqn. 7. Loads were applied to produce cylinder pressures from about 3.5 to 20.7 bar and speeds were set from about 1 to 20 mm/s. Before each set of tests, the system was bled to eliminate dissolved air. This was done by leaving the system under load for 24 hours.

To measure O-ring leakage, the piston was extended to its maximum height and loaded with the maximum weight with all valves closed, locking the piston in place. The initial position was estimated by collecting data from the position sensor for 5 minutes. At the 2.5 hour mark, the position sensor was sampled for 5 minutes. At the 64 hour mark (to completely eliminate air bubbles), another 5 minutes of position data was sampled. At the 88 hour mark, a final 5 minutes of position data was collected.
RESULTS

Figs. 5 through 8 compare the cylinder force efficiency as a function of pressure calculated from experiment data (the markers) to the efficiency predicted by the model (the lines) for the three sizes of cylinders and two piston speeds. The friction coefficient $\mu_f$ between the O-ring and the cylinder wall depends on the lubricating condition and because lubrication can only be estimated, we show the model as upper (dotted line) and lower (dashed line) bounds using the minimum and maximum values of $\mu_f$ friction coefficient. Fig. 8 shows efficiency as a function of bore size for two cylinder pressures.

DISCUSSION

Seal Leakage

The leakage model in Eqn. 1 predicts leakage by a moving seal, however, because the leakage is small, measuring the flow rate out the cylinder and the piston velocity with the required precision was not feasible. The experiment did show (Fig. 9) that leakage is insubstantial as the piston did not move over many hours of being under load. From this we conclude that in our case the cylinder volumetric efficiency is essentially 100% and that the overall efficiency for small bore cylinders is dominated by the seal friction.
Seal Friction

The results show that the friction model of Eqn. 4 predicted the measured piston force efficiency for all three sizes and across the entire tested operating range as can be seen most clearly in Fig. 8. Typically, an O-ring squeeze ratio between 7%-15% is an acceptable range for the O-ring to perform well [12]. A 14% squeeze ratio was selected for the cylinders tested in this study to ensure there would be sufficient friction to measure in the experiment. Figs. 5 - Fig. 7 show that the piston efficiency is close to 100% when the pressure is high, despite the high O-ring squeeze ratio, and that efficiency rolls off with lower pressure as expected. In applications where low friction was paramount, it would be possible to fabricate a cylinder with a lower squeeze ratio seal which would raise efficiency at lower pressures. Using a squeeze ratio over 15% is not recommended as the friction goes up substantially and the O-ring may become stretched. The O-ring stretch percentage is

\[ \sigma = \frac{D_g - D_i}{D_i} \cdot 100 \]  

(8)

where \(D_g\) is the mounting groove diameter, and \(D_i\) is the O-ring inner diameter. The stretch percentage is directly related to the squeeze ratio, and to avoid damage should not exceed the limit established by good design practice [9].

Eqn. 4 states that the seal friction does not change with piston velocity. This is approximately the case in Figs. 5 – 8 and implies that speed need not be taken into account when computing cylinder efficiency for cases where the seal is leak free or almost leak free.

Closer examination of the figures, and particularly Fig. 7 for 9 mm bore, reveals that efficiency does depend somewhat on speed. This is because friction coefficient \(\mu_f\) in Eqn. 4 changes, which can be explained by the Stribeck curve shown in Fig. 10 that shows the variation of the friction between two liquid lubricated surfaces [13]. In our experiment, cylinder efficiency improved with speed, which indicates that as speed increased, friction was likely moving down the Stribeck curve in the mixed friction region.

CONCLUSION

The main conclusion from this study is that a simple mathematical model for an O-ring is sufficient to describe the friction and leakage for a small hydraulic cylinder. Further, the experiments showed that for a small cylinder, an O-ring seal is essentially leak-free, which means that cylinder efficiency depends only on friction. Therefore, when developing system models for tiny hydraulic systems, small cylinders may be represented by a cylinder force efficiency modeled by Eqn. 4. The experiments also showed that the simple model is not sufficient to describe detailed behavior such as the small changes in friction that occur with piston speed.

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REFERENCES


