Design, Fabrication, and Testing of a Mechanical Timer in Application of a Stored-Heat Solar Cooker

by

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ABSTRACT
There is a large need in third-world tropical areas for a method of cooking in which users need minimal resources and traversing to heat food at night. A solution to this problem is to create a stored-heat solar cooker that may be left during the day and acquired at night to cook meals. Previously, a prototype had been built without much success in the timing of the device. This thesis aims to solve this problem by designing, building, and testing a mechanical timer. Several design choices were narrowed to the fabrication and testing of a hydraulic design similar to a gas spring. After this particular iteration of the prototype, proof of concept seems feasible. The next iterations of this timer should incorporate several design changes regarding the o-ring sealing and other various details for proper assembly and decreased cost.

Thesis Supervisor: David Wilson
Title: Professor Emeritus of Mechanical Engineering
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BACKGROUND

Although not well known, a large problem exists in third-world, tropical areas in that women have difficulties cooking meals at night time. Currently, the women venture far out of their home bounds to look for firewood and expose themselves to unspeakable violence and labor. In addition, people near the open fires inhale the toxic smoke. This toxic smoke has a casualty rate of about two million people per year. When told about the situation, Clinton's first lady angrily stated, "We have got to do something about this...".

In collaboration with MIT's Tata Center for Design and Technology, Professor David Gordon Wilson has formed a solution to this pressing issue. To allow women to cook at nighttime with minimal physical activity during the day, a stored-heat solar cooker may be used. The device simply stores heat from the sun in a sealed container of salt for roughly six hours during the day. Minimal user interaction is needed in which the owner simply places the cooker in a safe area during the day, and then retrieves it at night when he or she wishes to cook a meal or use the heat.

Professor Wilson has built a proof-of-concept prototype that successfully used a copper plate with fins as the heat-transfer device between the salt and heating surface. Improving upon this concept, many more objectives must be met before the cooker may be field tested. In particular, a significant problem with the device was the ability to aim its heating surface at the sun throughout the day. Since it is highly unlikely that these third-world tropical areas have access to electricity, a completely mechanical timer is needed to supplement the device. In addition, to achieve minimal user interaction, the timer must be automatic throughout the day and only need a reset at the beginning of each cycle. The prototype previously used an unreliable alternative to a balance wheel design [1]. Thus, this thesis aims to improve this concept by exploring different design options, fabricating the best design, and testing the viability of the timer device.

OBJECTIVES

The mechanical timer, in its most basic essence, needs to be able to correctly rotate the salt container at the sun for six hours by maintaining a constant angle rate and have a total angle travel of ninety degrees. The user will judge the starting angle for the cooker early in the day and rewind the device every cycle. Since the strongest heat intensity occurs for roughly six to eight hours of the day, it is sufficient for the device to time for six hours and travel ninety degrees. In addition to this concept, the device must also be robust and unaffected by environmental factors such as strong wind forces in the desert. The overall stored-heat solar cooker must be inexpensive (less than forty dollars) such that they may be affordable for users. This indicates that the timer fabrication must also be inexpensive. However, this thesis's prototype may be considered another first proof-of-concept device in
which manufacturing costs will be much higher than its potential mass-manufacturable form
cost in the applicable countries. This timer must be able to interface with the frame such that
the angle of the overall solar cooker may be controlled by the timer. The frame is currently
being redesigned by another undergraduate, and the interface will be designed as part of the
overall frame structure. This thesis will not discuss the interface between the two. As
previously stated in the background, the timer must also be user-friendly and avoid any use of
electrical devices. The overall objectives of the timer may be summarized in the following
three statements.

1. Correctly time the stored-heat solar cooker for about six hours with total angle of ninety
degrees and have a reset capability.
2. Withstand large wind gusts and other general environmental factors.
3. Avoid electrical devices.

PRELIMINARY DESIGNS

Based upon the objectives, several designs may be pursued. There are several
common everyday objects that demonstrate the first objective such as watches, alarms, and
sand hourglasses. In addition to these objects, it is also possible to explore devices using
hydraulic fluids such as those used in pistons. After initial research, three possible designs
seem the most viable.

Wristwatch Design

The first design is that of a wristwatch. Although most watches nowadays use quartz
crystals, luxury watches such as the Rolex brand still use an all-mechanical approach [2].
This involves the use of several gears, a balance wheel, and escapement device [3]. Since the
balance wheel concept is similar to a regular mechanical alarm, four product teardowns were
performed to research this concept further: Hermle floor clock, Taylor sixty minute
mechanical timer, Westclox Ardmore Twin Bell alarm clock, and Intermatic sixty minute
wall timer.

As seen by Figure 1, each gear train serves a purpose of movement. For example, one
gear train is to control the minute hand while another controls the hour hand. To conserve
space, several of these gear trains use the same intermediate gears. The motion of these gears
is set into action by the escapement mechanism (seen in Figure 2). This mechanism simply
pushes one tooth of a gear in a clockwise direction. Empowering the escapement mechanism
is the balance wheel. This wheel is precisely weighted and oscillates with a spiral torsion
spring (Figure 3). To begin the timing, another separate gear train is wound by the user with
its own main oscillating torsion spring (Figure 4).
FIGURE 1. EXAMPLE OF GEAR TRAINS IN PRODUCT TEARDOWNS. (A) HERMLE FLOOR CLOCK (B) WESTCLOX ARDMORE TWIN BELL ALARM CLOCK (C) INTERMATIC SIXTY MINUTE WALL TIMER (D) TAYLOR SIXTY MINUTE MECHANICAL TIMER

FIGURE 2. EXAMPLE OF ESCAPEMENT MECHANISM IN INTERMATIC SIXTY MINUTE WALL TIMER.
Now that the mechanism behind the timing is understood, the applicability for the solar cooker must be designed. To meet all three objectives, all the gear trains must be adjusted for the endgoals of each set. The balance wheel and escapement device are not easily fabricated, but can be easily taken from other timing devices. Since there is only one resulting motion in the angle of the cooker, one gear train is required for the action mechanism. Another gear train is needed for the wind-up in preparation of the main timing action. Therefore, only two gear trains are needed in total. A simple diagram can be seen in Figure 5 in which the “Special Gear” is a uniquely shaped gear with a dome-like surface to be influenced by the main spring or escapement mechanism, and the “Rod” indicates the connection gear train on a single rod to applicable outputs or inputs.
The design of the main action gear train is tricky since many factors must be taken into account. This gear train must perform the following.

1. Specified angular velocity on last gear.
2. Result in large torque force.
3. Sustain at least six hours.

The gear train associated with the wind-up mechanism must be easily wound without too much hand force. This translates to a small torque force on the input gear. Given these constraints, the resulting gear train could be very complex. A similar design to the product teardowns should be followed. As seen by the following diagrams of each mechanical product, the gear trains are flexible in design.
FIGURE 6. TAYLOR MECHANICAL TIMER GEAR TRAIN SET.

FIGURE 7. WESTCLOX ARDMORE MECHANICAL CLOCK GEAR TRAIN SET.
FIGURE 8. INTERMATIC WALL TIMER GEAR TRAIN SET.

To design a specific gear set for this particular application, the following steps are recommended:

1. Find the required number of stages and correspondingly the number of teeth to achieve a desired speed ratio.
2. Find pitch diameters.
3. Check resulting torque is desired. [4]

As seen by the gear sets in Figures 6, 7, and 8, the number of stages (indicated by N followed by a number), can be complex and repetitive. Oftentimes, the stages are repetitive simply to reverse the direction of a resulting motion. The closest gear train applicable here is the Taylor configuration is Figure 7. Based upon this gear train, several design considerations may be addressed as an example.

Since the solar cooker must travel ninety degrees in six hours, the approximate angular velocity is

\[ V = \frac{\frac{\pi}{2} \text{radians}}{6 \text{ hours}} \times \frac{1 \text{ rev}}{2\pi \text{ radians}} \times \frac{1 \text{ hour}}{60 \text{ minutes}} = 6.94 \times 10^{-4} \text{ rev/min} \]  

(1)

The angular velocity of the first and last gear may be related by the following:

\[ n_7 = \frac{N_2 N_4 N_6}{N_3 N_5 N_7} n_2 \]  

(2)

where \( n_7 \) is the desired angular velocity from equation 1. An approximate input may be estimated as the following:

\[ V_i = \frac{1 \text{ rev}}{9 \text{ sec}} = 6.67 \frac{\text{rev}}{\text{min}} \]  

(3)

15
Plugging in equations 1 and 3 into equation 2 is as follows:

\[
6.94E - 4 \frac{rev}{min} = \frac{N_2 \cdot N_4 \cdot N_6}{N_3 \cdot N_5 \cdot N_7} \cdot (6.67 \frac{rev}{min})
\]  

(4)

Since gears have whole numbers of teeth, equation 4 shows that any set of gears will work as long as this condition holds true and is approximately 0.0001 in ratio to achieve the end velocity in equation 1. Although this is simply an example, this calculation demonstrates that equation 2 may be manipulated such that a particular gear train is feasible with the objectives.

In addition to the design of the gear train, temperature compensation and humidity might need to be taken into account since tropical conditions are different than those in which the timer is being prototyped. The prototype must also be able to withstand a large torque force to balance out friction and wind torques. As seen by Figure 9, the Hermle floor clock’s gear trains and various components are complex in nature to accommodate large torques. If this design is successful though, the device would be relatively small and easy to reset by the user. In addition, many of the parts are common pieces sold by manufacturers such that the ending cost would be inexpensive.

![FIGURE 9. EXAMPLES OF VARIOUS COMPONENT COMPLEXITY IN HERMLE FLOOR CLOCK.](image)

**Hourglass Design**

The second possible design is that of a sand hourglass. These products are mostly found in gameboard sets with a defined time of a minute (shown in figure 5). Sand timers are known for reliability and accuracy. Since the pressure at the base of the sand is not affected by the height of unused sand, the device is precise in its function. In addition, these timers are known to be robust since there are very few, if any, mechanical components [5].

If this design were to be used, the objectives would change the details of this device. The most important change is the volume of sand. Based upon existing hourglass sand timers, six hours’ volume of sand is approximately more than twenty kilograms (forty-five pounds) [6]. In addition, moisture is a large problem with these types of devices since sand tends to absorb the moisture [7]. Another roadblock to this design is how to continuously pour sand in a device that constantly changes angle. If this design was successful though, the device would
be inexpensive and very reliable.

Hydraulic Design

The third possible design is to mimic the concepts of a gas spring. A gas spring is commonly used in automobiles and office chairs to support weight with damping. As seen by the diagram in figure 10, the design features a piston and sealed cylinder mechanism with a translation motion [8]. The spring effect is a result of pressure changes within the gas (typically inert nitrogen gas) and oil mediums. Since a gas spring functions through differential pressures, it is very reliable in any location. Similar to the second design, a measuring medium would have to be used. In this case, a common hydraulic fluid could be used. The volume would then be calculated with a known orifice diameter. Unlike gas springs though, this design must find a way to have a constant differential pressure rather than one dependent on the position of the piston. Gas springs are also susceptible to changes in temperature [9]. The new design avoids temperature dependency since the device will be used in tropical regions. If successful, the prototype would be easy to reset and replicate in the future.

FIGURE 10. CROSS-SECTION OF A TYPICAL GAS SPRING.

FINAL DESIGN CHOICE

The three possible designs each have their own advantages and disadvantages. However, it is reasonable to assume that the second design is not practical due to the volume
of sand needed for six hours. Both the first and third designs are feasible and can be designed according to the objectives. Since the breadth of this thesis focuses on one design, a Pugh chart was created to observe the differences between the two designs (Table 1). Since the first solar cooker prototype had a vibrator design for the timer mechanism, this design is a good base to compare the other designs.

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<th>Vibrator Design</th>
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<td>-</td>
<td>+</td>
<td>0</td>
</tr>
<tr>
<td>Cost</td>
<td>-</td>
<td>-</td>
<td>0</td>
</tr>
<tr>
<td>Complexity of Design</td>
<td>-</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Robustness</td>
<td>0</td>
<td>+</td>
<td>0</td>
</tr>
<tr>
<td>Ease of Use</td>
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<td>0</td>
</tr>
<tr>
<td>Ease of Attachment to Frame</td>
<td>+</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Total</td>
<td>-1</td>
<td>1</td>
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**TABLE 1. PUGH CHART WITH THE ORIGINAL PENDULUM DESIGN AS A BASE.**

As seen by the Pugh chart, both the wristwatch and hydraulic design have different advantages. However, the hydraulic timer design seems the most feasible and flexible for this particular application. Thus, this thesis will explore the design, fabrication, and testing of a hydraulic timer.

**DESIGN OF HYDRAULIC TIMER**

To meet all the objectives, the hydraulic timer must mimic the design of a gas spring with several other features. A complete CAD of the finished product is shown in Figures 11 and 12. These features may be organized into several components: piston mechanism, return assembly, constant-force mechanism, and housing. After multiple iterations, the final design consists of the following details.

![Figure 11. CAD of entire hydraulic timer assembly side view with transparent housings to enable view of inside piston mechanism.](image-url)
Piston Mechanism

The piston mechanism is the core of the device such that a translation motion allows hydraulic fluid to flow through an orifice to a low-pressure region. This mechanism may be organized as the following subsections: piston housing, piston base, and piston rod. Each component is labelled in Figure 13. Similar to a gas spring, the piston uses a combination of reciprocating dynamic seals and static seals at both the piston rod interface and piston base interface with the housing.

The housing for this mechanism consists of a 6061-T6 aluminum tube extrusion simply for cost and ease of fabrication. Sizing of the extrusion is based upon the volume of hydraulic fluid needed for six hours and the diameter of the orifice. Since the smallest available orifice has a diameter of 0.1016mm (0.004") (seen on Bill of Materials line nine in Appendix), the volume may be calculated.

The flow through an orifice may be approximated as the following:

\[ Q = AVK \]  \hspace{1cm} (5)

where \( Q \) represents the volumetric flow rate, \( A \) is the cross-sectional area of the orifice, \( V \) is the flow velocity, and \( K \) is a constant describing the unique flow shape of the orifice. The flow velocity may be approximated as the following:

\[ V = \sqrt{2gh} \]  \hspace{1cm} (6)

where \( g \) represents gravity and \( h \) represents the head across the orifice. To relate the differential pressure to the head, the following equation may be used:
\[ P = \frac{sg \cdot h}{2.31} \]  

where \( P \) represents the differential pressure, \( sg \) is the specific gravity of the hydraulic fluid, and 2.31 is a constant when in US customary units. The only known or desired parameters for our design will be the specific gravity, differential pressure, and cross-sectional area of the orifice. Therefore, combining equations one through three gives the following:

\[ Q = 25AK \sqrt{\frac{2.31P}{sg}} \]  

[10]. The cross-sectional area is simply given by \( \pi \cdot \left( \frac{0.04}{2} \right)^2 \) or \( 1.257 \text{E-5} \text{ in}^2 \) \( (8.11 \text{E-9} \text{ m}^2) \).

Common K values can be seen in Figure 14. The particular orifice has a shape equivalent with a K value of 0.97. Differential pressure, \( P \), is arbitrarily set as 2psi \((13.79 \text{kPa})\). This number is relatively low since our particular application does not require high pressure across the orifice. The specific gravity depends upon the hydraulic fluid. Since the device is intended for use in tropical regions, the fluid’s viscosity must be only slightly affected by temperature changes and have a defined viscosity and density. Silicone oil is a commonly used fluid for hydraulic applications and fits this description. Thus, our particular silicone oil (Bill of Materials line one and datasheet in Appendix) has a specific gravity of 0.969 at \( 25^\circ \text{C} \).

\[ Q = 25(1.257 \text{E-5} \text{ in}^2)(0.97) \sqrt{\frac{2.31\times(2 \text{ psi})}{0.969}} = 6.654E^{-4} \text{ gal/min} \times (2.52E^{-3} \text{ min/lt}) \]  

In six hours, this volumetric flow rate equates to a volume of \( 55.34 \text{ in}^3 \) \( (9.07 \text{E-4} \text{ m}^3) \). Given this volume, several dimensions of the piston housing may be found (shown in Table 2).

![Figure 14](image)
TABLE 2. POSSIBLE DIMENSIONS OF THE PISTON HOUSING GIVEN THE NECESSARY VOLUME.

For ease of manufacturing, an arbitrary choice of a 76.2mm (3") piston diameter was chosen. Using this as a guideline, the inside diameter of the housing and outside diameter of the piston base is dictated by the dynamic seal o-ring.

To ensure no leaks across the orifice, an o-ring was installed on the piston base and is expected to translate in the same motion as the housing relative to the piston base. Based on data in the McMaster catalog, a suitable o-ring is a double seal Buna-N o-ring with actual dimensions of 1.778mm (0.07") thickness, 79.5mm (3.129") outer diameter, and 75.9mm (2.989") inner diameter (Bill of Materials line two in Appendix). For a dynamic seal with 1.778mm (0.07") o-ring thickness, the o-ring “squeeze” is recommended to be fifteen to twenty-five percent of the o-ring thickness [11]. Since reduced friction is desired, a 15% squeeze on the o-ring was used. Figure 15 shows an example chart of o-ring groove sizing. Extrapolating off the chart, the housing inner diameter should be 79.38mm (3.125"). The piston base should have a diameter of 79.32mm (3.123") with an o-ring groove diameter of 76.58mm (3.015") and groove width of 2.41mm (0.095"). This information can be seen in Figure 18. For proper assembly, a thirty degree chamfer is placed at the end of the housing (described in Fabrication section) [12].

The silicone oil will be sealed within the container using two end caps on either side of the housing. These end caps are similarly fabricated out of 6061-T6 aluminum. One end cap will interface with the piston rod and contains a rod wiper and o-ring as part of the overall reciprocating dynamic seal.
Based on data in the McMaster catalog, a double seal Buna-N o-ring with a thickness of 1.778mm (0.07") is suitable for an oil application with actual dimensions of 19.15mm (0.754") outer diameter and 15.6mm (0.614") inner diameter (Bill of Materials line three in Appendix). Similar to the piston base o-ring, the gland diameter is based upon a 15% squeeze resulting in 18.67mm (0.735"). The piston rod should have a diameter of 15.88mm (0.625"), and the groove width should be 2.413mm (0.095"). For proper assembly of the o-ring, a thirty-degree chamfer is added to the o-ring gland. To ensure complete coverage of the rod and housing interface, a rod wiper is used (Bill of Materials line four in Appendix). Since the rod diameter is 15.88mm (0.625"), the appropriate wiper gland dimensions can be seen in Figure 16 as recommended by McMaster. Using these dimensions, a drawing of the endcap may be seen in Figure 20. To attach the endcaps to the housing, 6.35mm (1/4")-20 bolts are used to clamp the assembly together.

In addition to the dynamic seal, the piston base contains the small plastic orifice of diameter 0.1016mm (0.004") (Bill of Materials line three in Appendix) threaded in the center. Loctite, a brand of thread locker, is used to prevent leaks around the orifice. To connect the piston and piston rod, the piston rod has a slightly smaller diameter with a corresponding slip fit hole in the piston base such that the two may interface together. To permanently connect the two, both the piston base and rod are fabricated with C360 brass and soldered together.

**Return Assembly**

The hydraulic fluid must flow through the orifice and have a return pathway to restart the process. Thus, the return assembly adds multiple one-way check valves (Bill of Materials line ten in Appendix) to the piston base and a flow path through the piston rod such that the fluid may return to the same original reservoir without interfering in the timing process. This was easily solved by boring the inside of the rod and drilling holes on the sides. These details may be seen in Figure 17.
Constant-Force Mechanism

To actuate the device, the housing must have a mechanism to propel itself relative to the piston base. In addition, the actuation must be at a constant rate such that the timing of the device is accurate. Constant-force springs (commonly found in tape measures) solve this problem. The actuation of the device may be influenced by strong wind forces; therefore, the constant-force springs must actuate the device despite environmental factors. From personal experience and visits to third-world tropical regions, Professor Wilson estimates that a torque of 250Nm would account for these factors. Thus, the constant-force mechanism uses three constant-force springs of roughly 111.2N (25lbsf) each assuming a moment arm of 84.67mm (3 1/3") with the solar cooker frame (Bill of Materials line eleven in Appendix). To attach the springs, holders are machined out of C360 brass and soldered to the solar-cooker overall base. This can be seen in Figure 18. Each spring will contain a delrin rod that rolls around a rod placed in the holder and extend to the piston housing (shown in Figure 69). When the user is ready to reset the cooker, these springs will be unrolled by rotating a section of the solar-cooker frame.
Housing

To protect the entire assembly from harsh environmental factors, an overall housing was connected to the overall base. Since this housing does not serve a structural purpose, it is a simple, inexpensive 152.4mm (6") diameter PVC pipe. An epoxy was used to bond the housing to the overall base. This assembly can be seen in Figure 19.

FABRICATION

In total, there are ten individual types of components that must be machined. The dimensions, process plan, and results are compiled under each component’s sub-sub-section. Final assembly steps are explained at the end of this subsection.

Piston Base

Dimensions

As seen by Figure 20, the most important dimensions of the piston base to recognize are the diameter and depth of the o-ring groove.

FIGURE 18. ISOMETRIC AND TOP VIEW OF CONSTANT-FORCE SPRING TAB HOLDERS, SPRING TAB ROLLERS, SPRING TAB RODS, AND BASE.

FIGURE 19. SIDEVIEW AND ISOMETRIC VIEW OF ASSEMBLY HOUSING.

FIGURE 20. DRAWING OF PISTON BASE WITH DYNAMIC O-RING GROOVE DIMENSIONS.
### Process plan

<table>
<thead>
<tr>
<th>Step</th>
<th>Action</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Face radial surface and both ends of stock in lathe</td>
<td>Stock is C360 Brass rod, 76.2mm (3”) long, 79.375mm (3.125”) diameter</td>
</tr>
<tr>
<td>2</td>
<td>Take piece to mill</td>
<td>Clamp using soft grips</td>
</tr>
<tr>
<td>3</td>
<td>Zero mill to center of piece</td>
<td>Use edgefinder to find length of any arbitrary cord and take center of length as zero. Perform for both x and y axes.</td>
</tr>
<tr>
<td>4</td>
<td>Machine hole pattern and orifice hole 20.32mm (0.8”) depth</td>
<td>R drill bit</td>
</tr>
<tr>
<td>5</td>
<td>Take piece back to lathe</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>Machine o-ring groove 9.525mm (0.375”) from end</td>
<td>Lathe cutting tool, dimensions on sketch</td>
</tr>
<tr>
<td>7</td>
<td>Add 30 degree chamfer on one side of groove</td>
<td>Angle lathe cutting tool</td>
</tr>
<tr>
<td>8</td>
<td>Bore a 15.875mm (0.625”), 5.08mm (0.2”) depth slip fit hole to incorporate piston rod connection</td>
<td>Lathe small boring bar</td>
</tr>
<tr>
<td>9</td>
<td>Cut piece to 19.05mm (0.75”) length</td>
<td>Lathe cutting tool</td>
</tr>
<tr>
<td>10</td>
<td>Smooth edges with sandpaper</td>
<td></td>
</tr>
<tr>
<td>11</td>
<td>Tap 3.175mm (1/8”) NPT check valve hole pattern on same side as slip fit hole</td>
<td></td>
</tr>
<tr>
<td>12</td>
<td>Tap 3.175mm (1/8”) NPT orifice hole on other side</td>
<td></td>
</tr>
</tbody>
</table>

### Results

No errors in fabrication. See Figure 21 for finished piece.

![Figure 21. Finished Piston Base Piece](image-url)
**Piston Housing**

**Dimensions**

As seen by Figure 22, the most important dimension of the piston housing to recognize is the inner diameter.

![Figure 22. Dimensions of Piston Housing.](image)

**Process plan**

<table>
<thead>
<tr>
<th>Step</th>
<th>Action</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Cut stock to length</td>
<td>Stock is 6061 T-6 aluminum tube, 304.8mm (12&quot;) long, 101.6mm (4&quot;) diameter, 76.2mm (3&quot;) inner diameter</td>
</tr>
<tr>
<td>2</td>
<td>Face radial surface and both ends of stock in lathe</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>Bore inner diameter for o-ring on one side</td>
<td>Lathe large boring bar, steps of 0.508mm (0.020&quot;). Use inside micrometer for measurement (shown in Figure 24).</td>
</tr>
<tr>
<td>4</td>
<td>Flip stock end, bore inner diameter</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>Add chamfer on stock end</td>
<td>Angle lathe cutting tool</td>
</tr>
<tr>
<td>6</td>
<td>Take piece to mill</td>
<td>Clamp using soft grips.</td>
</tr>
<tr>
<td>7</td>
<td>Zero mill to center of piece</td>
<td>Use edgefinder to find length of any arbitrary cord and take center of length as zero. Perform for both x and y axes.</td>
</tr>
<tr>
<td>8</td>
<td>Drill holes for 6.35mm (¼&quot;)-20 bolts</td>
<td>#7 drill bit</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>---</td>
<td>---</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>Flip stock end, zero mill to center of piece</td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>Drill holes for 6.35mm ($\frac{1}{4}''$)-20 bolts</td>
<td></td>
</tr>
<tr>
<td>11</td>
<td>Debur, smooth edges with sandpaper on lathe</td>
<td></td>
</tr>
<tr>
<td>12</td>
<td>Tap 6.35mm (1/4'')-20 all holes</td>
<td></td>
</tr>
</tbody>
</table>

Results

When flipping the stock for boring, the concentricity of the part was not checked. As seen by Figure 23, a lip is evident inside the housing. For future prototypes, this mistake should be taken into account since it is not proper fabrication for an o-ring. For purposes of time completion, this piece was not remachined and the lip was simply sanded down until smooth. See Figure 25 for finished piece.

FIGURE 23. RESULTING BORE OF PISTON HOUSING WITH NOTICEABLE LIP FROM CONCENTRICITY ISSUE.

FIGURE 24. USE OF INSIDE MICROMETER FOR ACCURATE BORE MEASUREMENT.
Piston Rod Dimensions

As seen by Figure 26, the most important dimension of the piston rod to recognize is the diameter.

\[\phi 15.875\]

249.682

External threads of 5/8"-18

Figure 26. Dimensions of Piston Rod.
Process plan

<table>
<thead>
<tr>
<th>Step</th>
<th>Action</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Cut stock to length</td>
<td>Stock is C260 brass rod, 304.8mm (12&quot;) long, 19.05mm (0.75&quot;) diameter</td>
</tr>
<tr>
<td>2</td>
<td>Face radial surface and both ends of stock in lathe</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>Turn diameter to 15.875mm (0.625&quot;&quot;)</td>
<td>Lathe turning tool</td>
</tr>
<tr>
<td>4</td>
<td>Turn diameter on end of stock slightly below 15.875mm, 6.35mm (0.25&quot;) depth to create slip fit into piston rod connection</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>Create 15.875mm (5/8&quot;)-18 threads on other end of stock to interface with overall base</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>Debur and smooth edges</td>
<td></td>
</tr>
</tbody>
</table>

Results

No errors in fabrication.

Piston Rod Connection

Dimensions

As seen by Figure 27, the most important dimensions of the piston rod connection to recognize are the diameters of the split fit interfaces with the piston base and piston rod.

![FIGURE 27. DIMENSIONS OF PISTON ROD CONNECTOR.](image-url)
### Results

No errors in fabrication. See Figure 28 for finished piece.

<table>
<thead>
<tr>
<th>Step</th>
<th>Action</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Face radial surface and both ends of stock in lathe</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>Cut stock to length</td>
<td>Stock is C360 Brass Rod, 76.2mm (3&quot;) long, 25.4mm (1&quot;) diameter</td>
</tr>
<tr>
<td>3</td>
<td>Center drill, drill 6.35mm (1/4&quot;) hole</td>
<td>Through all</td>
</tr>
<tr>
<td>4</td>
<td>Bore end of stock for slip fit with piston rod 15.875mm (0.625&quot;)</td>
<td>Lathe small boring bar</td>
</tr>
<tr>
<td>5</td>
<td>Flip stock, turn to diameter 15.875mm (0.625&quot;) for slip fit with piston base</td>
<td>Lathe turning tool</td>
</tr>
<tr>
<td>6</td>
<td>Take piece to mill</td>
<td>Clamp using v-block</td>
</tr>
<tr>
<td>7</td>
<td>Zero mill to center of piece</td>
<td>Use edgefinder to find diameter and take center of length as zero.</td>
</tr>
<tr>
<td>8</td>
<td>Drill 3.175mm (1/8&quot;) holes</td>
<td>Through all</td>
</tr>
<tr>
<td>9</td>
<td>Rotate stock 90 degrees, drill 3.175mm holes</td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>Debur, smooth edges with sandpaper on lathe</td>
<td></td>
</tr>
</tbody>
</table>

**FIGURE 28. RESULTING SOLDERED ASSEMBLY OF PISTON BASE, PISTON ROD CONNECTION, AND PISTON ROD.**
**Piston Housing Cap One**

**Dimensions**

As seen by Figure 259, the most important dimension of the piston housing cap one to recognize is the locations of the four bolt holes to interface with the piston housing.

![Image of Piston Housing Cap One Dimensions](image)

![FIGURE 29. DIMENSIONS OF PISTON HOUSING CAP ONE.](image)

<table>
<thead>
<tr>
<th>Step</th>
<th>Action</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Place stock in waterjet</td>
<td>Stock is 6061-T6 aluminum sheet, 152.4mm (6&quot;) x 304.88mm (12&quot;)</td>
</tr>
<tr>
<td>2</td>
<td>Waterjet piece to correct dimensions</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>Debur, smooth edges</td>
<td></td>
</tr>
</tbody>
</table>

**Results**

Due to time constraints, this piece was machined on a CNC mill.

**Piston Housing Cap Two**

**Dimensions**

As seen by Figure 30, the most important dimensions of the piston housing cap two to recognize are the dimensions of the o-ring groove and shaft wiper.
**FIGURE 30. DRAWING OF PISTON HOUSING ENDCAP CONTAINING ROD WIPER AND STATIC O-RING SEAL.**

### Process plan

<table>
<thead>
<tr>
<th>Step</th>
<th>Action</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Face radial surface and both ends of stock in lathe</td>
<td>Stock is 6061-T6 aluminum rod, 25.4mm (1&quot;) length, 101.6mm (4&quot;) diameter</td>
</tr>
<tr>
<td>2</td>
<td>Center drill, drill 17.07mm (41/64&quot;) hole</td>
<td>Through all</td>
</tr>
<tr>
<td>3</td>
<td>Bore groove for shaft wiper</td>
<td>Lathe small boring bar</td>
</tr>
<tr>
<td>4</td>
<td>Bore groove for o-ring</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>Take piece to mill</td>
<td>Clamp using soft grips</td>
</tr>
<tr>
<td>6</td>
<td>Zero mill to center of piece</td>
<td>Use edgefinder to find length of any arbitrary cord and take center of length as zero. Perform for both x and y axes.</td>
</tr>
<tr>
<td>7</td>
<td>Drill holes for 6.35mm</td>
<td>F drill bit, through all</td>
</tr>
</tbody>
</table>
(1/4")-20 bolts
8
Debur, smooth edges

Results
Piece was outsourced to MIT Central Machine Shop due to time constraints.

Spring Tab Holder
Dimensions
As seen by Figure 31, the most important dimensions of the spring tab holder to recognize are the length and width of the slot for the constant-force spring.

![Figure 31. Dimensions of Spring Tab Holder.](image)

<table>
<thead>
<tr>
<th>Step</th>
<th>Action</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Face all sides of stock</td>
<td>Stock is C360 Brass 50.8mm (2&quot;) cube</td>
</tr>
<tr>
<td>2</td>
<td>Face stock to correct dimensions</td>
<td>Facing tool</td>
</tr>
<tr>
<td>3</td>
<td>Zero mill to corner of stock</td>
<td>Edgefinder tool</td>
</tr>
<tr>
<td>4</td>
<td>Drill hole for 12.7mm (1/8&quot;) rod</td>
<td>13.0969mm (33/64&quot;) drill bit</td>
</tr>
<tr>
<td>5</td>
<td>Zero mill to other corner of stock</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>Machine pocket</td>
<td>15.875mm (5/8&quot;) endmill</td>
</tr>
<tr>
<td>7</td>
<td>Machine slot</td>
<td>3.175mm (1/8&quot;) endmill</td>
</tr>
<tr>
<td>8</td>
<td>Debur, smooth edges</td>
<td></td>
</tr>
</tbody>
</table>

Results
No errors in fabrication. See Figure 32 for finished pieces.

![Figure 32. Resulting soldered assembly of spring tab holders and overall base.](image)

**Spring Tab Roller**

**Dimensions**

As seen by Figure 33, the most important dimension of the spring tab roller to recognize is the outer dimensions of the piece.

![Figure 33. Dimensions of spring tab roller.](image)

<table>
<thead>
<tr>
<th>Process plan</th>
<th>Action</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Step</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>Face radial surface and both ends of stock in lathe</td>
<td>Stock is delrin, 47.625mm (1 7/8&quot;) diameter, 152.4mm (6&quot;) length</td>
</tr>
<tr>
<td>2</td>
<td>Center drill, drill hole for spring tab rod</td>
<td>13.0969mm (33/64&quot;) drill bit</td>
</tr>
</tbody>
</table>
3 Turn diameter to 44.704mm (1.76") for press fit with constant-force spring
   Lathe turning tool
4 Cut to length
   Lathe cutting tool
5 Debur, smooth edges

Results
No errors in fabrication.

Spring Tab Rod
Dimensions
As seen by Figure 34, the most important dimension of the spring tab rod to recognize is the diameter.

![Figure 34. Dimensions of Spring Tab Rod.](image)

Process plan
<table>
<thead>
<tr>
<th>Step</th>
<th>Action</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Face radial surface and both ends of stock in lathe</td>
<td>Stock is 6061-T6 aluminum rod, 12.7mm (⅛&quot;) diameter, 152.4mm (6&quot;) length</td>
</tr>
<tr>
<td>2</td>
<td>Debur, smooth edges</td>
<td></td>
</tr>
</tbody>
</table>

Results
No errors in fabrication.

Overall Base
Dimensions
As seen by Figure 35, the most important dimensions of the overall base to recognize
are the slots for the spring tab holders.

FIGURE 35. DIMENSIONS OF OVERALL BASE.

<table>
<thead>
<tr>
<th>Process plan</th>
<th>Action</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Step</td>
<td>Action</td>
<td>Notes</td>
</tr>
<tr>
<td>1</td>
<td>Place stock in waterjet</td>
<td>Stock is C360 Brass sheet, 152.4mm (6&quot;) x 152.4mm</td>
</tr>
<tr>
<td>2</td>
<td>Waterjet to correct dimensions</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>Debur, smooth edges</td>
<td></td>
</tr>
</tbody>
</table>

Results
This piece was machined on a CNC mill due to time constraints.

<table>
<thead>
<tr>
<th>Assembly</th>
<th>Action</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Step</td>
<td>Action</td>
<td>Notes</td>
</tr>
<tr>
<td>1</td>
<td>Solder piston rod connection in slip fit hole of piston base</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>Solder piston rod in slip fit holes of piston rod connection</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>Solder spring tab holders onto overall base</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>Screw in orifice and check valves with Loctite into piston base</td>
<td>Let sit to cure</td>
</tr>
<tr>
<td>5</td>
<td>Install o-ring on piston base</td>
<td>Same side as chamfer on piston base groove</td>
</tr>
<tr>
<td>6</td>
<td>Clamp piston housing cap one with $\frac{1}{4}''$-20 bolts with Loctite</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>Fill cylinder with silicone oil</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>Place piston assembly into piston housing</td>
<td>Same side as chamfer on piston housing</td>
</tr>
<tr>
<td>9</td>
<td>Install piston housing cap two wit $\frac{1}{4}''$-20 bolts with</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>---</td>
<td>---</td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>Screw overall base onto piston rod</td>
<td></td>
</tr>
<tr>
<td>11</td>
<td>Fit constant-force springs with spring tab rollers</td>
<td></td>
</tr>
<tr>
<td>12</td>
<td>Insert spring tab rollers with constant-force springs and spring tab rods into spring tab holders</td>
<td></td>
</tr>
<tr>
<td>13</td>
<td>Extend each constant-force spring to piston housing cap one and epoxy</td>
<td></td>
</tr>
<tr>
<td>14</td>
<td>Epoxy overall housing to overall base</td>
<td></td>
</tr>
<tr>
<td>15</td>
<td>Check for leaks and let run</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Loctite</td>
<td></td>
</tr>
</tbody>
</table>

Assembly of the hydraulic timer revealed flaws in the design. All necessary parts had proper fabrication, so these errors must be attributed to the design. In particular, there were two major problems. The first is the designs of the o-ring grooves on the piston base and piston housing cap two. The depth of the gland was too shallow such that the o-ring simply slipped off when placing the piston base within the piston housing (as seen in Figure 36). In addition, the o-ring was able to freely rotate within the groove which indicates the o-ring was not stretched enough. Assembly was similarly difficult when placing the piston housing cap two onto the piston rod. Placing the o-ring and shaft wiper within the piston housing cap two was not difficult. However, due to friction with the piston rod and the sharp external threads on the end, assembly forced the o-ring and shaft wiper out of their respective grooves and tore the rubber (shown in Figure 37).

**FIGURE 36. IMPROPER ASSEMBLY OF PISTON BASE O-RING DUE TO GROOVE DESIGN.**
The second major problem discovered during assembly was the installation of the constant-force springs. Although the design of the spring tab holders can accommodate the functionality of the constant-force springs, these springs are not easy to handle and are not able to be unrolled safely by human force alone (demonstrated in Figure 38). For purposes of testing, smaller load constant-force springs were temporarily used.

**TESTING**

For purposes of testing, the overall housing was not attached to the assembly (step fourteen in assembly process). However, this piece should be attached if field-tested. In addition, testing was done with none or improperly assembled o-rings. Despite these flaws, certain tests may still be performed (assembly shown in Figure 39). The first test of the assembly was a leak test. By adding water into the assembly, any cracks or deformations...
would be immediately known. After adding metal epoxy to interface between the piston housing and piston housing cap one, no leaks were found (as shown in Figure 40).

FIGURE 39. BEST ASSEMBLY OF HYDRAULIC TIMER GIVEN DESIGN FLAWS.

FIGURE 40. ASSEMBLY OF PISTON MECHANISM CONTAINING WATER WITH NO LEAKS.
The second test involves the piston mechanism. By submerging the piston base inside the water, pressure should be “felt” when water travels through the orifice and check valves. Despite water travel around the piston base from no o-ring, significant pressure was felt when traversing the piston base. This indicates correct function by both the orifice and check valves.

RECOMMENDATIONS

With several design changes, the next iteration should be more successful in adhering to the objectives. In particular, the o-ring design for both the interface between the piston base/piston housing and piston rod/piston housing cap two should be improved. In the first situation between the piston base and piston housing, the groove on the piston base should be deep enough such that the o-ring will not slip off during assembly. This groove should also stretch the o-ring enough such that there is no radial movement. However, a fine balance between sealing and friction should be determined such that the piston base may easily translate and contain no leaks. In the second situation between the piston rod and piston housing cap two, the design should similarly slightly change such that the friction decreases and allows easier assembly. In addition, the piston housing cap two should be placed onto the piston rod before the external threads are created. These threads are inherently sharp and will tear the rubber.

Besides changes in the o-ring design, the constant-force spring mechanism should be improved such that installation is easier. The 111.2N (25lbsf) constant-force springs are necessary for reasons stated earlier. Thus, the spring tab holders should be modified for easier assembly. However, since the constant-force springs are hard to handle, any other constant force mechanism is also recommended. Normal extension or compression springs may also be used if calculations are done to account for the displacement-dependable force.

CONCLUSION

The hydraulic timer design shows promise to correctly demonstrate the original objectives. Although the cost of this prototype is significant (shown in BOM in Appendix), the manufacturability of the device may always be improved. After several design and fabrication choices, the hydraulic device is certainly one of the more promising prototypes even after several design flaws. For an optimal mechanical timer, the wristwatch design should be pursued in parallel since it also showed extreme promise. Overall, this thesis demonstrates that the hydraulic timer concept is certainly feasible and should be pursued further for the future of the stored-heat solar cooker.
APPENDIX

Bill of Materials (BOM)

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<tr>
<th>Line</th>
<th>Item Description</th>
<th>Link/Source</th>
<th>Quantity</th>
<th>Price</th>
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BOM Line 1: Silicone Oil datasheet

High Purity Silicone Oil
Polydimethylsiloxane 50 cSt.

Description
Silicone oil is a high purity, low viscosity silicone fluid with a broad range of applications ranging from personal care to automotive and electronics. It is an environmentally safe engineered product, free-flowing, non-flammable, non-toxic and non-ionic, with excellent heat stability, electrical properties, and fluid resistance. It can be used neat or in product formulations.

Product Equivalency
Dow Corning® 200 Fluid 50 cSt
GE ZF55 50
Eckerle® AT 50
Stainless Ultra Fluid 50 cSt

Properties
Non Flammable
No odor
Inert to seal o-rings and gaskets
High and low temp stability
Tasteless
Non Toxic
Ultra High Purity

Polydimethylsiloxane
CAS #: 63148-62-5

Appearance:
Crystal clear, colorless liquid

Purity, wt. %:
99.9

Viscosity:
47.5-52.5 at 25°C

Water:
max. 0.003

Molecular weight:
131.17

Nourishing point:
132.5 (98.5)

Flash Point Open Cup:
20.8

Surface Tension:
0.070

Specific Gravity:
1.4034

Hardness:
400

Coefficient of expansion:
1.86 x 10^-4

Molecular Weight:
20.35

Disclaimer: The information listed above is based on data available to the manufacturer and is considered accurate to the best of our knowledge. The use of this product is subject to the user’s own due diligence and is not guaranteed to be accurate or complete. The user is responsible for determining the suitability of this product for their specific application. The manufacturer accepts no liability for any loss, injury or damage resulting from the use of this product. Please refer to the Material Safety Data Sheet (MSDS) for additional safety information before using this product.

Source: http://www.polyquartz.com

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### BOM Line 2: Piston Base O-ring

**Double Seal Buna-N O-Ring**  
**AS568A Dash Number 041**

- **Packs of 25**  
  - In stock  
  - $11.20 per pack of 25

**Material**: Buna-N  
**AS568A Dash No.**: 041  
**Fractional Size**:  
- **ID**: 3"  
- **OD**: 3 1/8"  
**Actual Inch Size**:  
- **ID**: 2.989"  
- **OD**: 3.129"  
**Additional Specifications**: Width: 1/16" Fractional (0.070" Actual)

**Add to Order**  
**90025K435**

- You ordered 1 each on 05/02/14.

---

**Buna-N** is used in oil applications. Temperature range is -40°F to +250°F. Durometer hardness is A70. Color is black.

### BOM Line 3: Piston Rod O-ring

**Double Seal Buna-N O-Ring**  
**AS568A Dash Number 016**

- **Packs of 100**  
  - In stock  
  - $11.30 per pack of 100

**Material**: Buna-N  
**AS568A Dash No.**: 016  
**Fractional Size**:  
- **ID**: 5/8"  
- **OD**: 7/16"  
**Actual Inch Size**:  
- **ID**: 0.614"  
- **OD**: 0.754"  
**Additional Specifications**: Width: 1/16" Fractional (0.070" Actual)

**Add to Order**  
**90025K147**

- You ordered 1 each on 05/13/14.

---

**Buna-N** is used in oil applications. Temperature range is -40°F to +250°F. Durometer hardness is A70. Color is black.
BOM Line 4: Piston Rod Wiper

Double-Lip Buna-N Shaft Wiper
5/8" ID, 15/16" OD, .203" Base Height

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2 added to your order 05/11/14.

Wiper
- ID (A): 5/8"
- OD (B): 15/16"
- Base Height (C): .293"
- Overall Height (D): 1.245"
- Shaft Groove:
  - ID (E): .827"
  - OD (F): .933"
  - Base Height (G): .745"

Additional Specifications
Double-Lip Buna-N Shaft Wipers

Prevent contaminants from entering your fluid power system with these one-piece, snap-in wipers. They scrape moisture, dust, and grit from shafts. They also have great resistance to petroleum-based hydraulic fluids. Color is black.

Also known as 5400 type. While the outer lip scrapes foreign matter from the shaft, the inner lip acts as a pressure seal. Commonly used with JIC (Joint Industry Conference) cylinders. Wipers are made of Buna-N, which is recommended when using a rubber rod seal. Buna-N has excellent oil resistance. It also resists water, grease, and detergents, as well as some ammonia, alcohols, and refrigerants. Temperature range is -40° to 250° F. Durometer hardness is A90. Maximum pressure is 500 psi.

Also Available: Additional sizes. Please select 9422K12 and specify ID and OD.

BOM Line 9: Orifice

Nylon Threaded-Insert Flow-Control Orifice
.004" Orifice Diameter, Purple, 1/8 NPT Male

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You ordered 1 each on 02/11/14.

Pipe Size 1/8
Orifice Diameter 0.004"
Flow Rate @ 40 psi
- Air, scfm 0.010
- Water, gpm 0.0022
Color Purple
Max Pressure 100 psi @ 75° F
Temperature Range 40° to 120° F

Vent, bleed, or purge your system to the atmosphere at a controlled rate. Use with water, inert gas, and air. Body is nylon. Orifices are colored for easy identification. Have a hex head. Connection is NPT male.
BOM Line 10: Check Valve

**Quick-Opening Brass Check Valve**
1/8 NPTF Dryseal Male Connections, Buna-N Seal

- **Pipe Size**: 1/8
- **Length**: 1 1/4"
- **Maximum Pressure**
  - Buna-N Seal: 1,000 psi @ 70° F
  - Fluoroelastomer Seal: 1,000 psi @ 70° F
- **Temperature Range**
  - Buna-N Seal: -40° to +300° F
  - Fluoroelastomer Seal: +10° to +400° F
- **Additional Specifications**
  - Male x Male
  - For Water, Oil, and Air—Buna-N Seal

Get flow at minimal pressure levels—just 0.3 psi is required to open. Valves are often used in instrumentation applications. Body is brass. Install in any direction. Connections are NPTF (Dryseal).

BOM Line 11: Constant-Force Spring

**Stainless Steel Constant-Force Spring**
4000 Cycle Life, .025" Thick, 52.0" Long, 1.5" Wide

- **Thickness**: .025"
- **Extended Length**: 52"
- **Width**: 1.500"
- **Wound ID**: 1.77"
- **OD**: 2.23"
- **End Hole Diameter**: 0.265"
- **Load**: 24.80 lbs.

**Additional Specifications**
- 4,000-Cycle Life
- Free end has two side-by-side holes at the tip of the spring.

Typically used as retractors in tape measures and cable reels, these springs are wound into a tight coil to provide uniform force throughout extension and retraction. They can be wrapped around a shaft, spool, or rod, allowing for 1/2 extra coils on the shaft at full extension to hold the spring in place. All of these springs are made of Type 301 stainless steel. The free end has one hole for attaching a load to the spring, unless noted. Wound ID, wound OD, and load tolerances are ±15%.
BIBLIOGRAPHY


