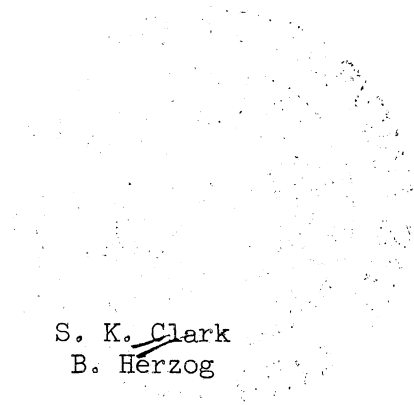


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ACTUATOR LOCK MECHANISM ANALYSIS



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ABSTRACT

Tests were conducted to determine the mean friction angle of SAE 52100 steel balls on hardened SAE 52100 steel plates. This information was used, together with the results of a theoretical force analysis, in order to specify a range of mechanical parameters inside of which the subject lock mechanism would operate according to specification.

OBJECTIVE

The objective of this study was to analyze the lock mechanism in question in order to determine the influence of the various mechanical parameters on its operating characteristics.

## INTRODUCTION

An analysis of the operating requirements for a lock mechanism shaft of a control surface actuator was requested by Messrs. Taylor, Porter, and Williams of Vickers, Inc. During the analysis and tests contact was maintained with Vickers and results were transmitted verbally or by letter. This report, then, constitutes a discussion of the analysis.

## THE PROBLEM

The locking mechanism has been isolated from the rest of the actuator and reduced to the minimum number of parts sufficient to state the problem, as shown in Fig. 1. The shaft, A, is shown in the locked position, the piston, B, blocking motion of the balls, C. A pressure,  $p$ , is exerted upon B as shown. The springs, D, react upon this piston, B, and allow travel of the piston, which depends on the pressure,  $p$ , the rate, and the preset of the springs. The parts E and F are spacers and retainers for the balls and perform no significant part in this analysis.

Basically the locking or unlocking is to be controlled by the magnitude of the pressure,  $p$ , during operation. At a pressure of approximately 1000 psi the system shall lock and unlocking shall occur near 2000 psi. Obviously, as the pressure,  $p$ , increases the piston, B, will tend to move to the left and eventually the indentation will move in position for the balls to be ejected from the groove, thus permitting motion of the shaft, A. The balls must be ejected from the groove even when a load up to 8750 lb is acting in either axial direction of shaft A.

For any value of the small slope of angle  $\theta$  on the piston some part of this axial force will be transmitted to the piston, B, by friction. Thus the motion of the piston, B, will depend on the pressure,  $p$ , the springs, and the above-mentioned friction force.

From an examination of the drawings and of the test data it is plain that one source of difficulty is in the frictional forces which are transmitted to the moving piston by the actuator shaft. This is undesirable during unlocking since it is only under this condition that the friction force must be overcome in order to initiate motion. By a proper choice of the wedge angle on the



sliding piston, wedge and friction angles may be made equal so that the component of the resultant force between ball and wedge in the direction of motion of the wedge is zero. Thus, axial actuator forces will not influence the locking mechanism in any way. This was the purpose in determining the friction angle experimentally, and from here on in this report it will be assumed that friction and wedge angles are set equal.

A simplified diagram is shown in Fig. 2, from which the following equations are written.

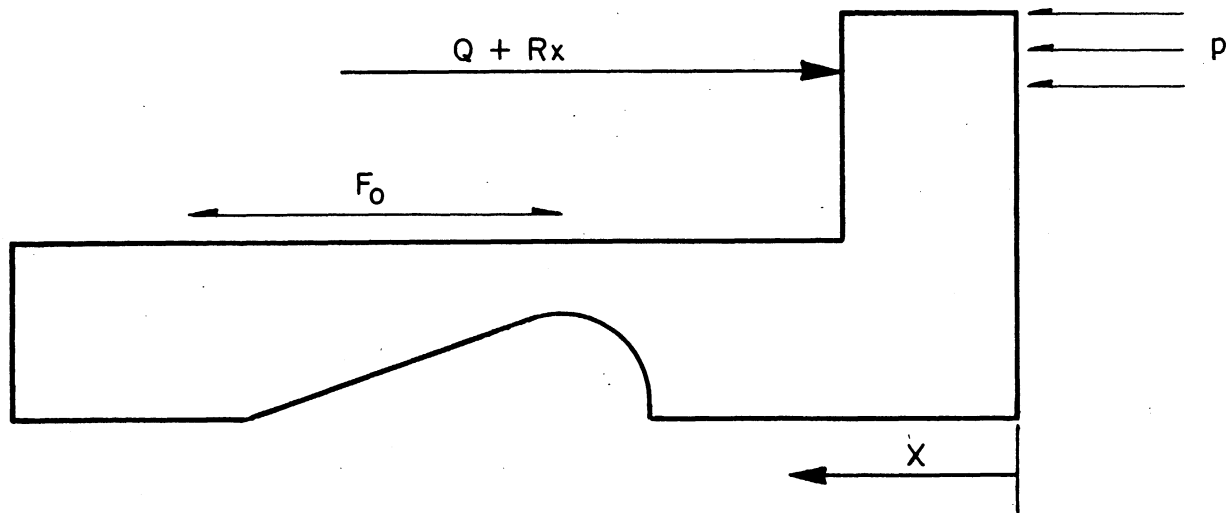


Fig. 2. Locking piston showing the active forces.

Unlocking cycle:

$$Q + Rx + (F_o)_1 = A_o \frac{p_1}{1000} \quad (1)$$

Locking cycle:

$$Q + Rx - (F_o)_2 = A_o \frac{p_2}{1000} \quad (2)$$

where  $Q$  is the amount of preset force in springs,  
 $R$  is the spring rate,  
 $F_o$  is O-ring force, a function of the pressure,  
 $p_1$  is unlocking pressure, psi, and  
 $p_2$  is the locking pressure.

The O-ring presents dry-friction forces which depend on the direction of piston travel. Hence, the subscripts 1 and 2 are used to denote the forces during the unlocking and locking cycles, respectively. The magnitude of the O-

ring force is assumed to be

$$F = 25 + 22.5 \frac{P}{1000} \text{ lb.}^* \quad (3)$$

The range of pressures shall be

$$1 < \frac{P_{1,2}}{1000} < 2 \quad (4)$$

Substituting (3) and (4) into (1) and (2) yields

$$Q + Rx + 25 + 22.5 p_1(x) = A_0 p_1(x) \quad (5)$$

$$Q + Rx - \left[ 25 + \frac{22.5 p_2(x)}{1000} \right] = A_0 p_2(x) \quad (6)$$

The equilibrium pressures (impending motion) must be linear functions of  $x$ , at the most due to spring linearity. Let

$$p_1(x) = a_1 + b_1 x$$

$$p_2(x) = a_2 + b_2 x \quad .$$

Substituting into equations (5) and (6) one obtains

Unlocking:

$$Q + Rx + 25 + \frac{22.5}{1000} (a_1 + b_1 x) = A_0 (a_1 + b_1 x) \quad (7)$$

Locking:

$$Q + Rx - 25 - \frac{22.5}{1000} (a_2 + b_2 x) = A_0 (a_2 + b_2 x) \quad (8)$$

The conditions on these equations are as follows:

$$(a) \quad (a_2 + b_2 x)_{x=0} = a_2 \geq 1000$$

since the piston must be fully locked ( $x = 0$ ) at 1000 psi or greater.

$$(b) \quad (a_1 + b_1 x)_{x=3/16} = a_1 + \frac{3b_1}{16} \leq 2000$$

since the piston must be fully unlocked ( $x = 3/16$ ) at 2000 psi or less.

\*An approximation to the true pressure vs friction force curve. See Vickers T.O. 12238-10, 2-14-56.

By equating coefficients in equations (7) and (8) four simultaneous equations are obtained:

$$Q + 25 + \frac{22.5}{1000} a_1 = A_0 a_1 \quad (7a)$$

$$R + \frac{22.5}{1000} b_1 = A_0 b_1 \quad (7b)$$

$$Q - 25 - \frac{22.5}{1000} a_2 = A_0 a_2 \quad (8a)$$

$$R - \frac{22.5}{1000} b_2 = A_0 b_2 \quad (8b)$$

By eliminating Q from (7a) and (8a), and R from (7b) and (8b), each of the resulting equations may be solved for  $A_0$  and equated to one another, giving

$$A_0 = \frac{22.5}{1000} \cdot \frac{(b_1 + b_2)}{(b_1 - b_2)} = \frac{50 + \frac{22.5}{1000} (a_1 + a_2)}{(a_1 - a_2)} \quad (9)$$

With reference to the two conditions listed previously, the maximum design freedom will be obtained by taking full advantage of them. Thus, let

$$a_2 = 1000$$

$$a_1 + \frac{3}{16} b_1 = 2000 ; a_1 = -\frac{3}{16} b_1 + 2000$$

Then equation (9) becomes

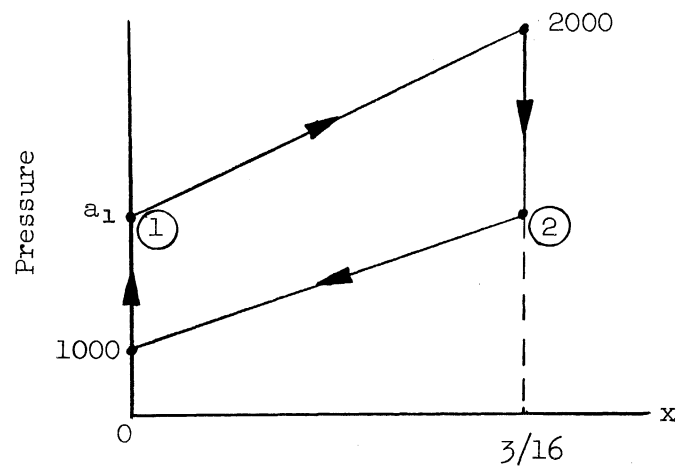
$$A_0 = \frac{22.5 (b_1 + b_2)}{1000 (b_1 - b_2)} = \frac{50 + \frac{22.5}{1000} \left( -\frac{3}{16} b_1 + 3000 \right)}{-\frac{3}{16} b_1 + 1000} \quad (9a)$$

From this it is seen that the choice of either  $b_1$  or  $b_2$  automatically determines the other one of the b constants as well as  $A_0$ , and from equations (7a) and (7b), Q and R. Equation (9a) may be solved for  $b_2$  in terms of  $b_1$  to give



$$b_2 = \frac{\left\{ \left[ \frac{50 + \frac{22.5}{1000} \left( 3000 - \frac{3}{16} b_1 \right) \right] \times 1000}{1000 - \frac{3}{16} b_1} - 22.5 \right\} b_1}{\left\{ \left[ \frac{50 + \frac{22.5}{1000} \left( 3000 - \frac{3}{16} b_1 \right) \right] \times 1000}{1000 - \frac{3}{16} b_1} + 22.5 \right\}}$$

Let us identify the points on a typical operating diagram of the mechanism as follows:



By choosing values of  $b_1$  it is now possible to plot points 1 and 2 on the operating diagram,  $A_0$ ,  $Q$ , and  $R$  all vs  $b_1$ , thus giving an overall picture. Since  $b_1$  has little significance in itself, Fig. 3 shows a plot of  $A_0$  vs  $Q$ ,  $R$ , 1, and 2 for the most favorable design, namely, when the entire available pressure range of 1000 psi is used to actuate the mechanism. From this curve values of the spring parameters and operating characteristics can be determined for any value of effective piston area. The problem then reduces to choosing a piston area which yields spring characteristics which, by trial-and-error design, are capable of being made and operated at the stress levels and in the space involved.

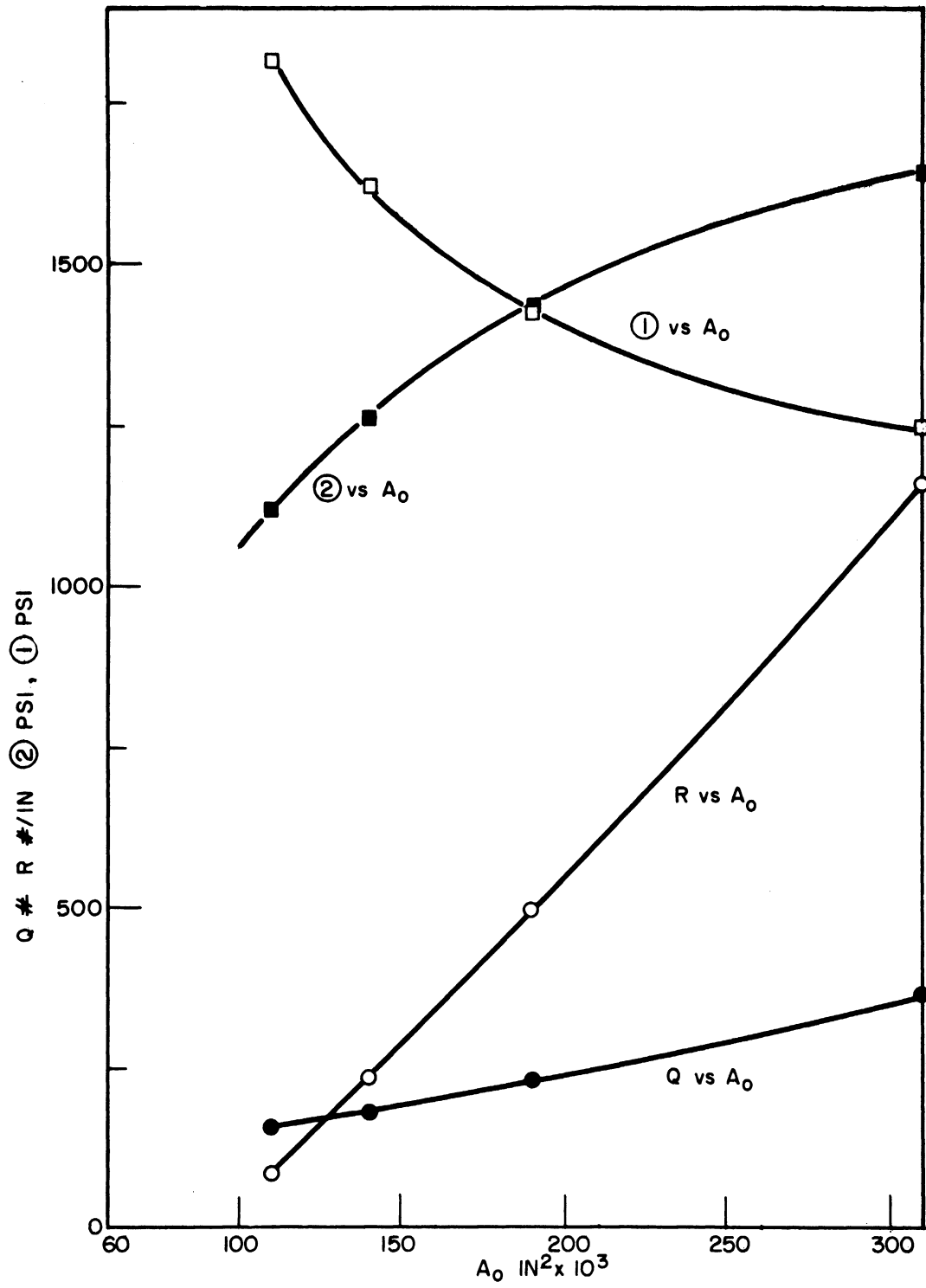


Fig. 3. Plot of spring rate, spring preset, and points on the operating diagram vs piston area for most favorable design conditions.

## APPENDIX

## Determination of Friction Angles

In order to determine the optimum wedge angle on the piston lock for the actuator, it was necessary to obtain the proper friction angle between the ball bearings and the material of the piston lock. The material specification for the piston lock is SAE 52100, to be heat treated to Rockwell "C" 60-65, and an experiment was planned to determine the proper coefficient of friction from which the friction angle might be obtained.

Six-inch-diameter, one-inch-thick blanks of this steel were prepared and heat treated for this test. One group of such blanks was obtained by the University but was not successfully heat treated to the specification. After this Vickers, Inc., obtained similar blanks and had them heat treated by their sources. Again these were not up to specification. However, tests were performed and will now be described in the following paragraphs.

The ball bearings were prepared for the tests by grinding a flat on them and they were in turn mounted in two pieces of steel with the flat side down and cemented in place. The above-mentioned heat-treated blanks were then put in between these two sets of balls as shown in the accompanying Fig. A-1. The plates A (Fig. A-1) were placed as shown over plate B, which was the test material. The forces N were applied by a testing machine to the plates A and the force F was applied by a winch and measured with a dynamometer bar placed in the connection between B and the winch. Care was taken to see that the three ball bearings above and below plate B were concentrically located. A plan of the ball arrangement in the plates A is shown in Fig. A2. The dynamometer used to measure force F consisted of an aluminum bar with strain gages mounted upon it and previously calibrated. The output of these strain gages was measured by a Wheatstone bridge and Brush Strain Recorder. Dial gages were used to determine impending motion of part B.

The coefficient of friction,  $\mu$ , or the angle of friction,  $\phi$ , may be determined from the following relationship:

$$\mu = \frac{F}{2N} = \tan \phi .$$

The accompanying table shows the results of the test conducted. The loading range was based on previous calculations which indicated that each ball will be required to carry a normal force of about 1000 lb.

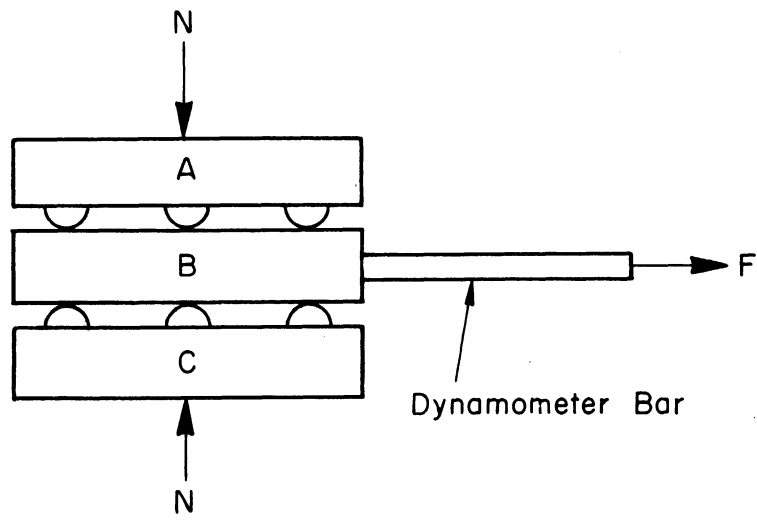


Fig. A-1.

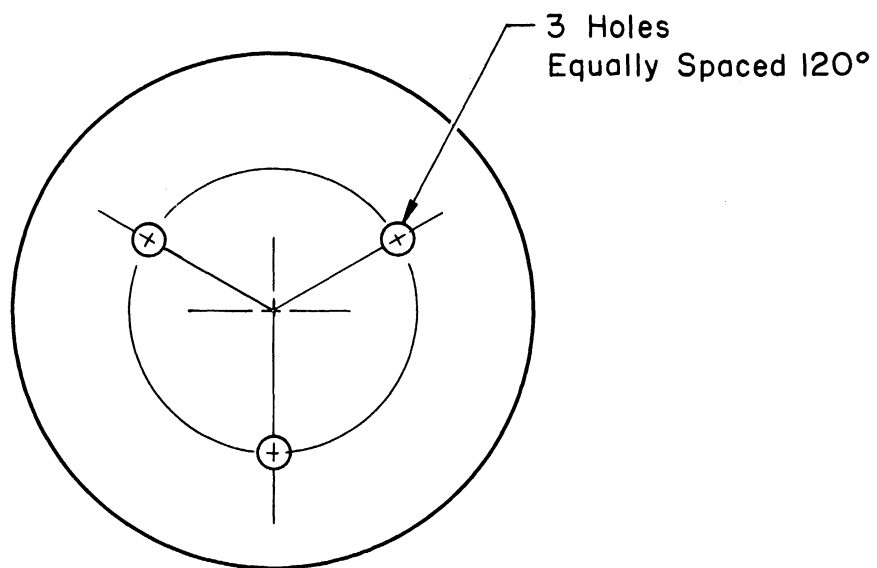


Fig. A-2.

TABLE OF DATA AND RESULTS

Run	N (lb)	F <sub>max</sub> (lb)	μ <sub>max</sub>	F <sub>0</sub> (lb)	μ <sub>0</sub>
1	600	125	.104	—	—
2	1200	275	.115	150	.063
3	1800	350	.097	—	—
4	1800	380	.105	—	—
5	2400	530	.110	250	.052
6	2400	580	.121	350	.073
7	3000	800	.134	400	.066
8	600	120	.100	90	.076
9	1200	225	.094	125	.052
10	1800	375	.104	225	.062
11	2400	500	.104	275	.057
12	3000	750	.125	375	.063
13	3600	875	.122	500	.070
14	600	150	.125	130	.108
15	1200	375	.156*	150	.063
16	1800	400	.111	275	.076
17	2400	600	.125	400	.084
18	3000	750	.125	400	.066
19	3600	825	.115	—	—

Avg: .113  
 $\phi = 6^\circ - 27^\circ$

Avg: .069  
 $\phi = 4^\circ$

\*Not used in computing average friction coefficients.

It may be noted that the results vary. At low normal force N the balls did not noticeably brinell the surface of the plate B. However, as the force N was increased, brinelling was observed. As motion of part B with respect to the plates A occurred, the force F increased due to the fact that the balls had gouged into the surface of part B and met with increased resistance. It should be noted in this respect that the material had not been hardened to the required hardness (tests indicated hardness of about Rockwell "C" 45) and in all probability the hardness some distance below the surface was even less than this, accounting for the increase in force with increased motion.

The reduced data of this experiment are shown in the accompanying table. In all there were 19 runs performed for various values of the normal load N. F<sub>max</sub> is the largest value of the force recorded and F<sub>0</sub> that which occurred at impending motion. Accordingly, two sets of values for μ were obtained, one for each of the previously mentioned loads. It will be noted that run 15 was

out of line with the rest of the data and this was not used in computing the average values. Since one may expect that the actual production models will behave similarly in this test, the maximum value  $F$  was used to determine the net friction angle, which is then in the vicinity of  $6^\circ$ .



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