

BUREAU OF MINES  
CONTRACT H0111021

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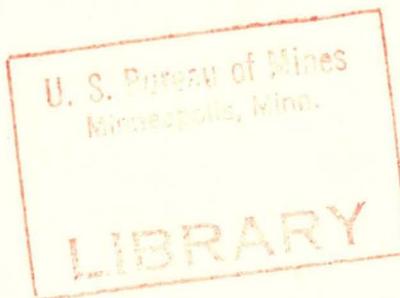
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# METHANE CONTROL PROGRAM

*Open File Report*

## DESIGN AND DEVELOPMENT OF HORIZONTAL BOREHOLE PACKERS FOR USE IN COAL BEDS



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TULSA, OKLAHOMA

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## CONTENTS

	<u>Page</u>
Abstract . . . . .	1
Introduction . . . . .	1
Conclusions . . . . .	1
Current Commercial Packers . . . . .	2
Commercial Packer Tests . . . . .	8
Prototype Packer . . . . .	8
Design Parameters . . . . .	8
Development of Design . . . . .	10
Testing of First Design . . . . .	13
Design Revisions . . . . .	16
Field Testing . . . . .	24
Optimum Rubber Properties . . . . .	25
References . . . . .	27

## ILLUSTRATIONS

<u>Figure</u>		<u>Page</u>
1	Floating Head Balloon Packer	3
2	Fixed Head Balloon Packer	5
3	Slip Set Packer	7
4	Compression Seal	7
5	Holding Power Commercial 2-3/4" Packer	9
6	First Prototype Packer	10
7	Original Prototype Packer (Photograph)	13
8	Original Prototype Packer Disassembled (Photo)	13
9	Test Bench	14
10	Test Bench (Photograph)	14
11	Revised Prototype Packer Design	17
12	Revised Prototype Packer (Photograph)	17
13	Revised Prototype Packer Disassembled (Photo)	17
14	Revised Prototype Packer in 3 in. Tubing (Photo)	18
15	Performance Comparison Original and Revised Packers 70 Durometer Rubbers	19
16	Performance Comparison Original and Revised Packers 80 Durometer Rubbers	19
17	Performance Comparison Revised Packer and Commercial Packer	20
18	Effect of Cycling on Pump Out Pressure	21
19	Comparison of 2.75 and 3.25 in. Rubbers	22
20	Comparison of Combination Rubber With 70 and 80 Durometer Rubbers	24
21	Holding Power Three Element Packer	26
22	Holding Power Three Element Packer	27

## Abstract

A new horizontal borehole packer for use in coal beds has been developed. This packer utilizes a principal of operation which has not previously been used. The resulting packer is less costly to purchase, can be easily repaired in the field with inexpensive expendable parts. This report describes the development and testing of the new packer and the evaluation of commercially available packers.

## Introduction

This work has been authorized under the Bureau of Mines Contract H0111021. The objective of the work was "to evaluate commercially available horizontal borehole packers for use in coalbeds and to design and produce a packer that is more reliable, will cost less initially and will be less costly to maintain than those presently available".

Gas pressure measurements and other testing procedures in horizontal holes may require as many as thirty packers in a 300 foot hole. These packers must be set simultaneously.

The cost of the only commercially available packer suitable for this service is quite high and the repair and replacement of expendable parts is also quite expensive.

## Conclusions

A new hydraulic packer has been developed, manufactured and tested which has demonstrated superiority over previously existing packers. This superiority was accomplished while reducing cost approximately sixty per cent over the best of other packers and reducing the weight almost as much.

This new packer, because of a unique piston design, cannot suffer a sealing element rupture. The packer can be easily disassembled in the field for maintenance using inexpensive expendable parts. The packer can undergo over forty cycles without any loss of efficiency in the sealing elements.

The packer exhibits good versatility in that over size sealing elements can be used on the mandrel to effect a seal in an oversize hole, and an extended mandrel can be utilized for more sealing elements.

These improvements have been realized with the only sacrifice being in the length of seal when compared to the packer now in use.

The packer is now on the commercial market and other applications, variations and improvements can be expected with increased usage.

### Current Commercial Packers

Engineering data on current commercial packers is virtually non-existent. Development of these packers has been an art, and as a consequence performance properties are largely unrecorded. The analysis presented here is of necessity empirical and much of the information was gathered from industry sources.

The packers presently available may be divided into two classes with two sub classes in each class. These are:

- I. Balloon Packers
  - A. Floating Head
  - B. Fixed Head
  
- II. Compression Packers
  - A. Mechanical
  - B. Pneumatic or Hydraulic

In general the Balloon Packer is characterized by an elastometric element which inflates much as a balloon to effect a seal. The compression packer relies on axial compression of an elastometric element to cause radial expansion of the element and sealing. Each packer type has unique advantages and disadvantages which are discussed individually.

#### I. Balloon Packers

All balloon packers are characterized by an inflatable elastomer sealing element fixed between two heads with a concentric pipe mandrel inside the elastomer. The sealing elements are of various compositions, the simplest form of which is an unreinforced elastomer tube. The strongest type consists of three elements: An outer rubber element, a wire braid reinforcing element and a soft rubber inner sealing element. Sealing elements are attached to the heads in three ways: (1) banding, (2) with a threaded compression fitting, and (3) by chemical bonding. With methods one and two a stress raiser is created at the points of attachment and cutting of the element is possible during installation -- this can encourage rupture of the element. The problems of stress raisers are eliminated by chemical bonding, but field replacement of the element is not possible.

All balloon packers enjoy common advantages and disadvantages. These are:

Advantages

The packer is long (three to four feet) relative to its diameter, which gives a large sealing area.

Most of the packers are inexpensive and can be repaired in the field.

Disadvantages

All of the packers are subject to failure if inflated in an oversize hole or if the element is pinched or cut by angular materials or open fractures encountered by the borehole.

Other features which are unique to the type of heads on the packer are:

A. Floating Head Packers

This packer type is illustrated in Figure One. This type of packer is characterized by the ability of the head to move on the pipe mandrel. The advantages and disadvantages of this type of packer are:

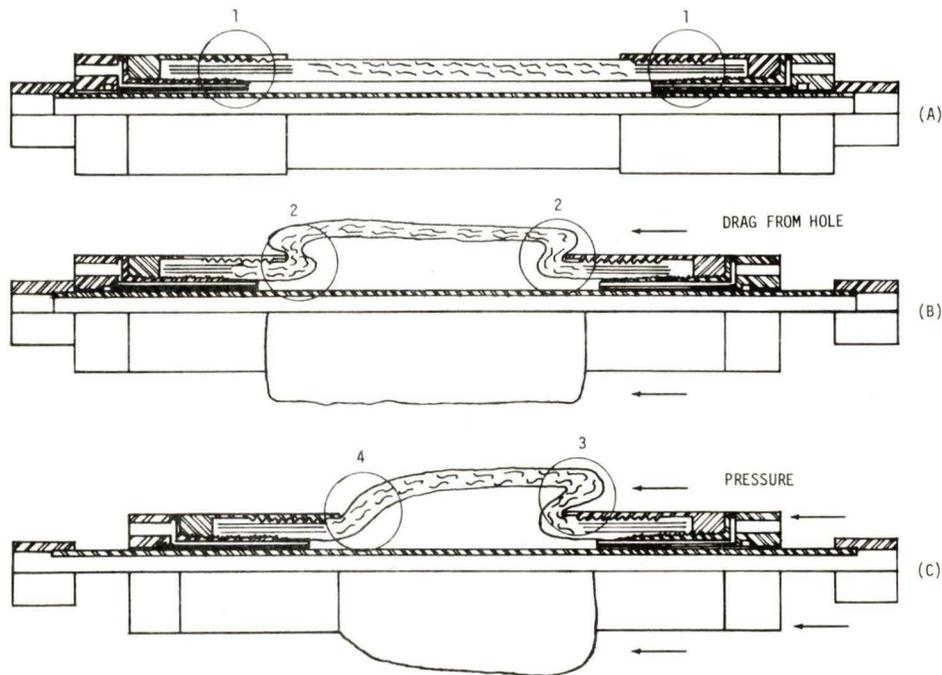


FIGURE 1 - Floating Head Balloon Packer

### Advantages

The movement of the head or heads on the mandrel permits a greater amount of expansion per unit of inflation pressure. This allows the packer to be set in larger holes with less danger of rupturing the element.

The floating head reduces the end effect stresses in the rubber and permits the packer to successfully seal a larger diameter hole than is possible with a fixed head packer.

### Disadvantages

When running in or coming out of a hole with crooked or tight spots, the head affected tends to move on the mandrel and wedge the packer in the hole as illustrated in 1(B). This problem can usually be overcome by partial inflation of the packer or placing a concentric pipe around the mandrel to limit the head travel. When the packer is wedged in the hole the rubber may be subjected to recurrent bending with weakening of the rubber. These possible recurrent bends are illustrated as points 2 in Figure 1(B).

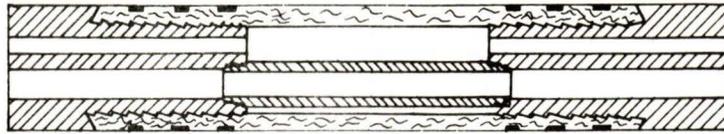
When relatively high differential pressures exist across a packer the head may move on the mandrel and result in a recurrent bend as illustrated as point 3 in 1(C). Failure can result from such recurrent bending.

Without the recurrent bend in the rubber, if the pressure outside the packer exceeds the inflation pressure, the rubber will collapse and allow leakage around the packer.

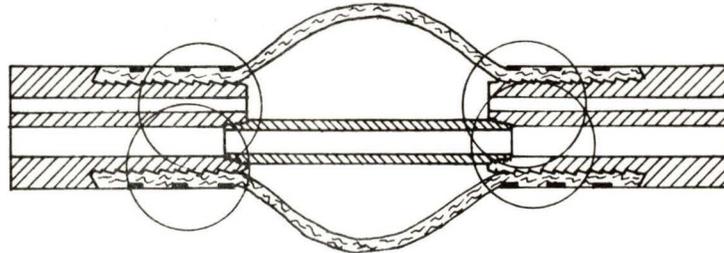
If the sealing element encounters any sharp edges such as might be located at 4 in 1(C), the rubber can be cut, resulting in a premature failure.

### B. Fixed Head Packers

This type of packer is illustrated in Figure Two. 2(A) illustrates the packer uninflated, and 2(B) illustrates the packer inflated. This type is characterized by heads which are not free to move on the mandrel. The unique advantages and disadvantages of this type are:



(A)



(B)

FIGURE 2 - Fixed Head Balloon Packer

### Advantages

With the fixed heads preventing movement of the element, there is no danger of the rubber wedging the packer in a tight or crooked spot in the hole.

### Disadvantages

An end effect stress raiser is created at the points of attachment of the sealing element to the heads (circled on 2(B) ). This is especially the case if there are sharp projections or corners at the points of attachment.

The restriction in axial movement causes an increase in stress at the ends. This stress increase in the ends is equal to:

$$s_z = \frac{u2Pa^2}{b^2 - a^2}$$

where  $s_z$  = stress in the axial direction  
 $u$  = Poisson's ratio  
 $P$  = inflation pressure  
 $a$  = inside diameter of rubber  
 $b$  = outside diameter of rubber

Since Poisson's ratio for rubber is 0.5, this stress can become quite high. As the element expands, the rate of increase of "a" exceeds that of "b" so the denominator decreases while the numerator increases causing  $s_z$  to increase rapidly. This further limits the hole diameter within which the packer can successfully operate.

One balloon packer has been successfully utilized by the Bureau, and this packer was bench tested for comparative purposes.

## II. Compression Packers

This class of packer is characterized by an elastometric sealing element which is subjected to axial compression with the resulting radial expansion of the element creating the seal.

Compression packers can be further subdivided into two sub classes which can be characterized as Mechanical and Pneumatic packers.

Compression packers as a class enjoy the following advantages and disadvantages, while features unique to each sub class are discussed individually.

### Advantages

Since no inflating fluid is utilized under the sealing element, there is no danger of a rupture of the rubber and loss of setting capability.

### Disadvantages

Rubber or a rubber like material will behave as other materials, and a long column of rubber will buckle under axial compression. This limits the length of the sealing element which can be effectively utilized. This disadvantage can be overcome in part by utilizing multiple short sealing elements with steel spacers between them.

### A. Mechanical Packers

These packers are characterized by a mechanical force being utilized to furnish the axial compression to set the sealing element. In this case tension on the pipe column is utilized to create a reactive compression force in the sealing element. One case utilizes slips which expand into the borehole wall to anchor the packer and furnish a reactive force. This packer type is illustrated as Figure Three. The second type utilizes a concentric pipe column in compression to achieve the reactive force to produce axial compression in the sealing element and is illustrated as Figure Four.

### Advantages

The slip set packer requires only one pipe column and it is in tension. The compression seal does not rely on the borehole wall for support.

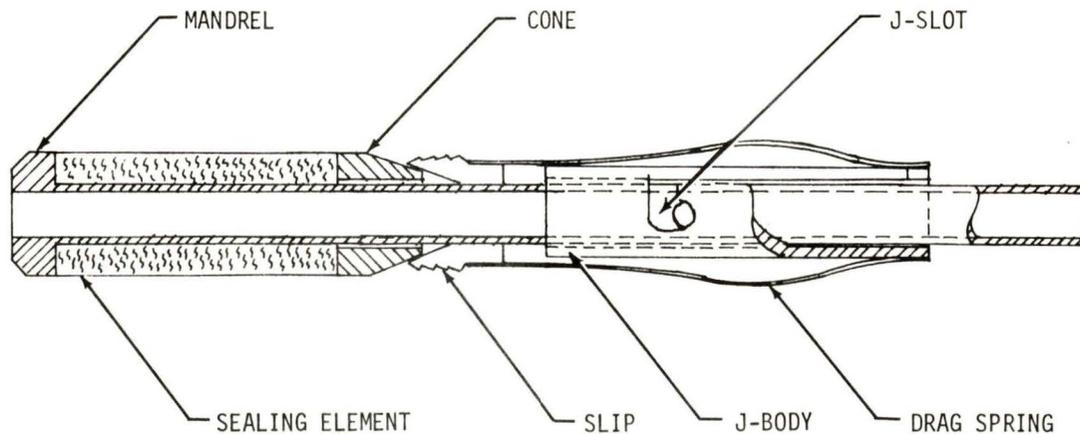


FIGURE 3 - Slip Set Packer

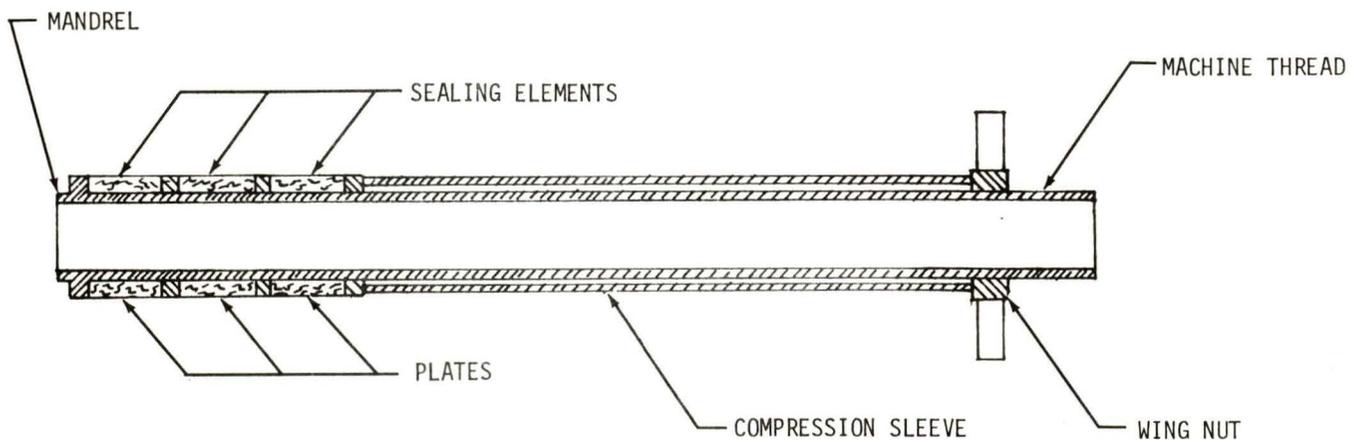


FIGURE 4 - Compression Seal

### Disadvantages

Neither of the mechanical packer types can be relied on for effective sealing when multiple packers are used. This is especially the case with the slip set packer where the first slips set will carry all of the load unless there is a mechanical failure of the borehole wall to allow additional slips to set and transfer compressive load to other sealing elements.

The slip set packer must rely on the borehole wall for the development of reactive force to compress the sealing element.

The compression seal must rely on a double pipe column, one of which is in compression, for the reactive compressive force.

This doubles the amount of pipe required with a corresponding increase in weight and number of joints.

#### B. Pneumatic or Hydraulic Packer

There are no commercial packers utilizing pneumatic or hydraulic pressure to create axial compression except the prototype packer developed under this contract. The development of this packer is discussed at length in the following pages.

Because of the obvious problems in utilizing multiple mechanical packers, they were not tested.

One criterion established for packers is the ability to contain 1,200 psi or greater setting pressure. Of the balloon packers manufactured only one can withstand pressures over about two hundred psi. This packer has only one floating and one fixed head. The sealing element is chemically bonded to the heads and the element consists of three parts. This packer was tested as a basis of comparison to the prototype packer.

#### Commercial Packer Tests

One packer was purchased for comparative testing. The testing procedure was the same as is utilized for the prototype packers. In all of the packer tests conducted with one exception, the sealing elements did not leak at pressures up to the pressure at which the packer was forced from the test pipe. Since sealing effectiveness proved to be reliable in almost all cases, the testing criteria became holding power or friction between the sealing element or elements and the containing pipe. This holding power was determined by the force required to cause the set packer to move in the pipe. This force expressed as a pressure beneath the packer is defined as "pump out pressure" and this term is used throughout the balance of this report. Figure Five illustrates this.

The test procedure and comparative results are discussed with the prototype packer.

#### Prototype Packer

##### Design Parameters

The objective of this contract was to develop, if possible, a packer that is more reliable, easily repaired in the field, less costly to purchase

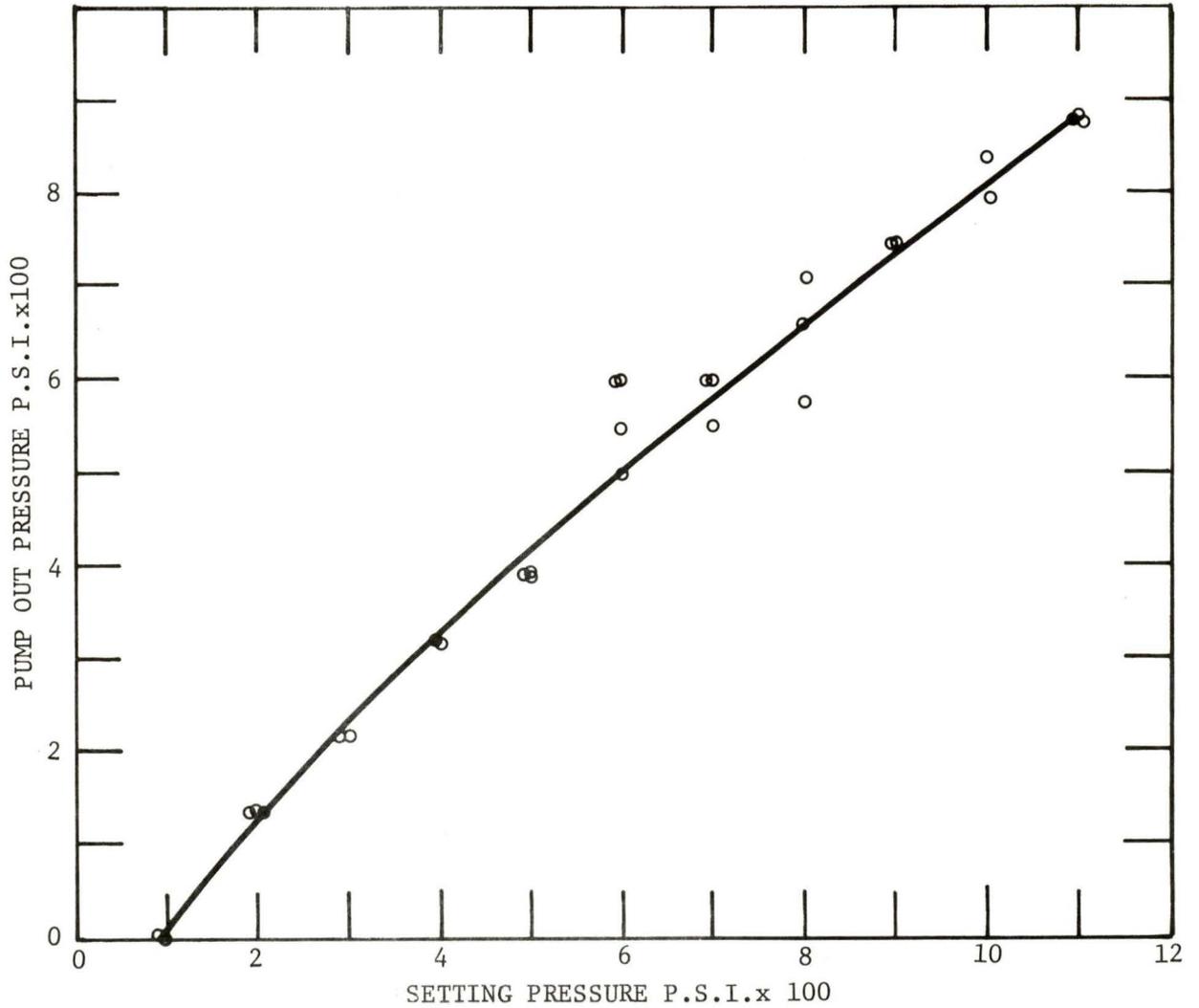


FIGURE 5 - Holding Power Commercial 2-3/4" Packer

and less costly to repair than those presently available. There is only one commercially available packer which meets the service requirements, and this packer costs \$460. A replacement element costs \$337 with the heads. If the element is returned to the vendor, replacement elements are \$202 exchange.

The principal form of failure experienced in the present packer is rupture of the sealing element. This is caused by enlargements in the borehole and/or fractures encountered which might cut the element. To eliminate this problem it was decided that a compression packer of a new type would be most satisfactory. The compression type packer would also have the advantage of utilizing low cost molded rubber sealing elements which could be easily replaced. It was considered undesirable to depend on either slips or a concentric pipe column to furnish the reactive force for compression of the elements. In order to assure independent setting of multiple packers a piston was utilized

to develop the axial compressive force and the mandrel to develop the reactive force.

The packer developed is shown in Figure Six. This packer had been developed and had limited field utilization in vertical holes prior to the Bureau of Mines contract.

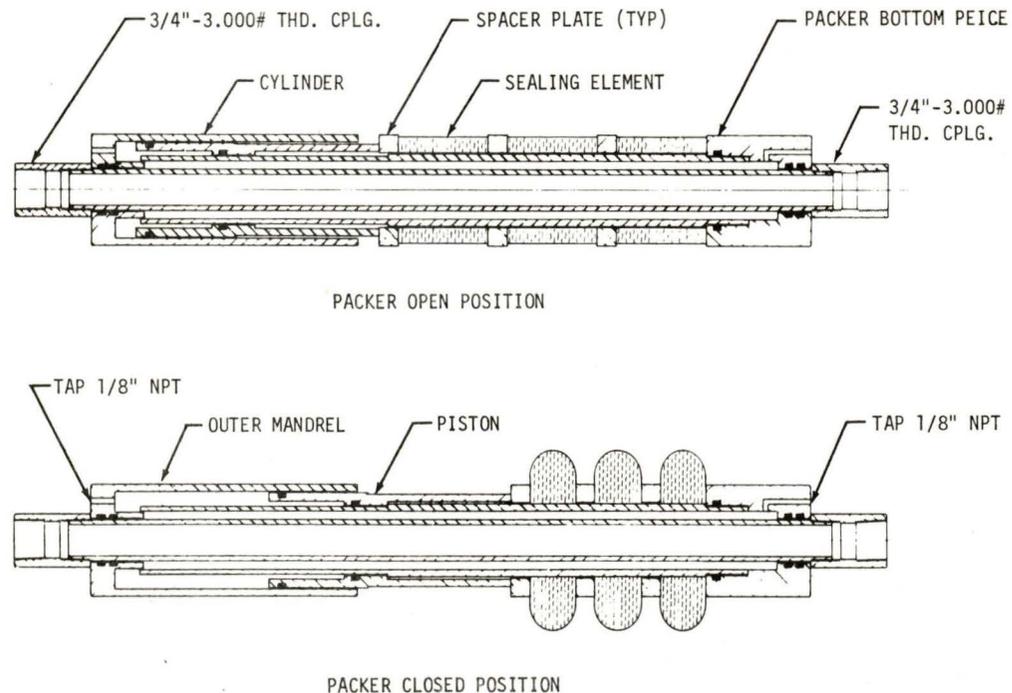


FIGURE 6 - First Prototype Packer

The original goal for this packer was to produce a packer for less than \$100. This was possible with the original design, but as development proceeded, certain shortcomings were discovered and improvements incorporated into the design which resulted in an increased cost. The final design has resulted in a cost of about \$150. Replacement sealing elements cost about \$5 each and can be easily replaced in the field.

#### Development of Design

The principal problem in the conception of this packer was how to conduct the setting fluid through the packer so multiple packers could be utilized in a hole. Various ideas were examined and then rejected because of fabrication problems and the desire to hold cost down. The scheme which was finally accepted was to utilize concentric mandrels with the annulus between the mandrels serving as the conduit for the setting fluid to the next packer. As an economy measure standard sizes of mechanical tubing were utilized wherever possible.

With the development of the basic concept of how the packer was to function, design then proceeded to the sealing elements. This is the heart of the whole packer concept.

Rubber in compression is not discussed in the usual sources of data on engineering materials. The source of rubber data of necessity has been the rubber industry itself. The work here has been based on research published by the Goodyear Tire and Rubber Company. This is cited as Reference (1).\*

In the absence of mechanical holding devices, the frictional force developed between the sealing element and the borehole wall must be sufficient to prevent the pressure beneath the packer from forcing it out of the hole. Frictional force is increased by increasing the hardness of the material, but at the cost of increased creep and permanent set. The increasing hardness also decreases the sealing effectiveness according to (2). Both the sealing effectiveness and the frictional force are increased by increasing the squeezing force.

The setting force required to force the sealing elements into contact with the wall of the borehole is a function of the hardness of the rubber, the rubber volume, and the shape factor of the element. The effect of the shape has been empirically developed in Reference (1).

The element is considered essentially incompressible and thus there is no volume change in the elastomer. The percentage of compressive deflection required to contact the borehole wall is determined by:

$$C = 100 - \frac{100 L_2}{L_1}$$

Where C = % compression (length reduction)  
L<sub>2</sub> = compressed length  
L<sub>1</sub> = original length

(1)

L<sub>2</sub> is determined by:

$$L_2 = \frac{D_1^2 - D_2^2}{D_3^2 - D_2^2}$$
(2)

Where D<sub>1</sub> = original OD of rubber  
D<sub>2</sub> = ID of rubber  
D<sub>3</sub> = hole diameter

The shape factor is described as the ratio of the loaded to the free area, or:

$$S = \frac{D_1^2 - D_2^2}{4D_1 L_1}$$
(3)

\* References are given at the end of this report.

After the required per cent compression and shape factors have been determined, the required compressive stress is found using equation (1). The compressive stress is converted to a setting pressure by the ratio of the piston area to the original rubber area, or:

$$P_s = S_z \frac{D_1^2 - D_2^2}{D_4^2 - D_5^2} \quad (4)$$

Where  $P_s$  = setting pressure on piston  
 $S_z$  = compressive stress  
 $D_4$  = OD of piston  
 $D_5$  = ID of piston

Frictional force on the borehole wall then is developed as the pressure on the piston increases above the initial setting pressure. The unit lateral force is the difference between the applied pressure and setting pressure. The available frictional force is determined by:

$$F = (P - P_s) D_3 L_2^f \quad (5)$$

Where  $F$  = frictional force  
 $P$  = applied pressure  
 $f$  = coefficient of friction

An exact numerical solution for  $F$  is impossible because  $f$  varies so widely with materials and the surface condition at the area of contact. Equation (5) does have some value to show the rate of increase in frictional force with increasing pressure and constant coefficient of friction.

Reference (1) recommends that a rubber compression piece not be longer than its diameter and that the percentage compression be limited to about 40%. These recommendations governed the length of sealing elements and the amount of piston travel which is to be allowed. The design parameter was given that the packer was to be used in a three-inch hole. This limited the original diameter of the sealing element to 2-3/4 inches. The length to diameter ratio recommendations limited the length to the same 2-3/4 inches. As a slight margin for safety we have limited the length to 2-3/8 inches. This should also improve the shape factor.

In the case of the amount of piston travel, we departed slightly for two reasons. In some circumstances it may be desirable to set the packer in an oversize hole and it may also be desirable to add additional elements which can be accomplished by using a longer mandrel. Thus we selected 3.5625 inches for the travel length of the piston. This will make possible 50% compression of three elements.

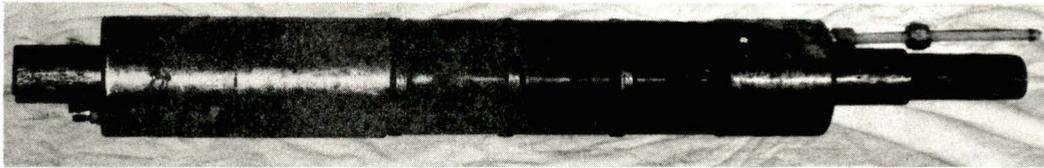


FIGURE 7 - Original Prototype Packer

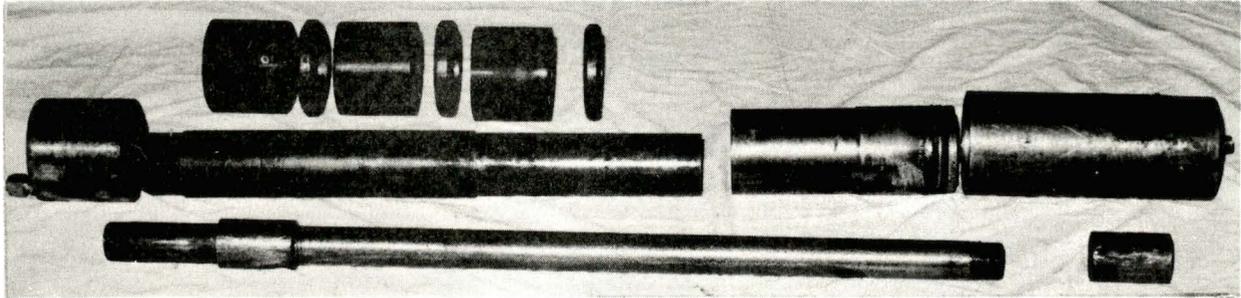


FIGURE 8 - Original Prototype Packer Disassembled

Much of the design work which was required in this packer was based upon empirical observation which was not subject to numerical analysis. Because of the intuitive and empirical factors, it was expected that numerous bugs would develop as the packer was developed. Only two major bugs were revealed, but another major revision was made to the design to obtain improvement in the performance.

#### Testing of First Design

The testing of the first prototype design was directed toward determining the sealing effectiveness and holding power of the packer in three-inch standard pipe. Figure Nine is a schematic diagram of the test bench. Two methods of determining the holding power were utilized. One was to pull the packer loose with a hydraulic ram, and the other method was to apply nitrogen pressure below the packer until the packer began to move. Both methods functioned quite well, but the nitrogen pressure method gave a more precise measurement because of greater gauge sensitivity.

Leakage was determined by pressuring the pipe with nitrogen and observing flow, if any, through a bubble tube. Figure Ten is a photograph of the test bench.

The prototype packer was tested with one, two and three sealing elements on the packer. This gave a measurement of the increase in effectiveness obtained by utilizing additional elements. Two packers in tandem were also tested to determine the reinforcing frictional effect of two packers with up to three elements each. Rubber hardness of sixty, seventy,

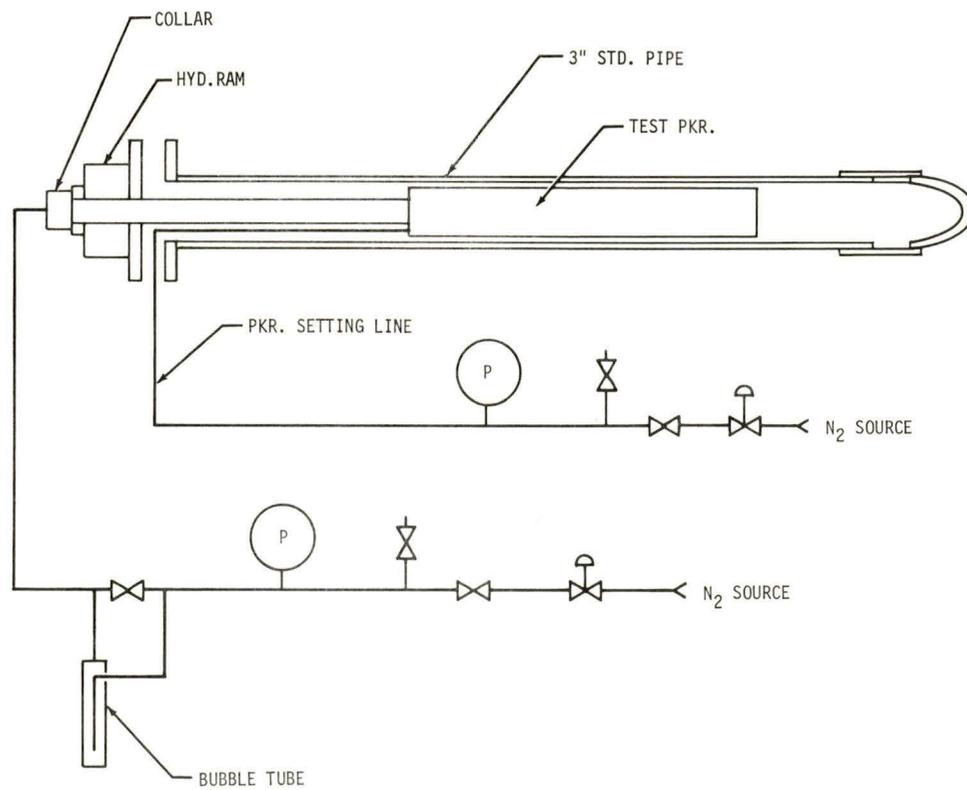


FIGURE 9 - Test Bench

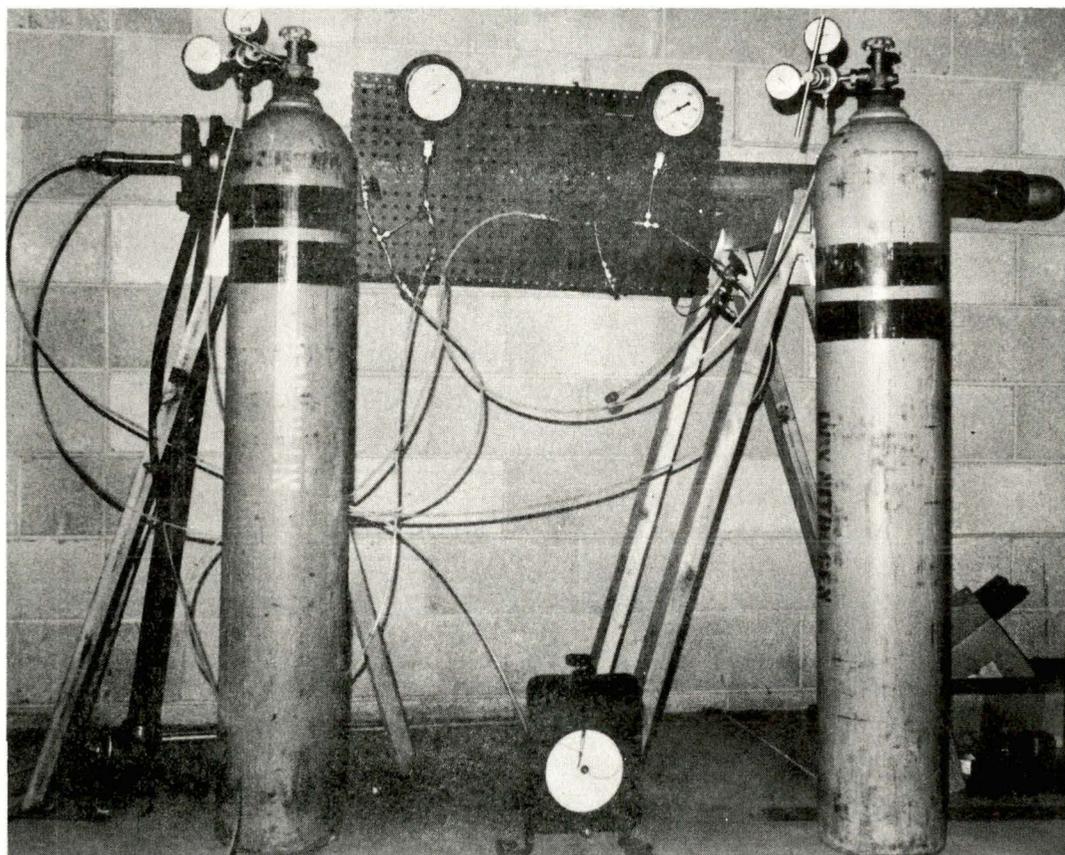


FIGURE 10 - Test Bench

eighty and ninety Shore A durometer were utilized to evaluate the effect of rubber hardness. All of the rubber elements were Nitrile rubber. This compound was selected because of its combination of compression set, abrasion, gas permeation and creep properties. This compound also exhibits good to excellent resistance to all chemicals except strong oxidizing agents.

All combinations of sealing elements except those of ninety durometer exhibited sealing characteristics which exceeded the frictional holding power of the combination. All ninety durometer sealing combinations exhibited very poor sealing characteristics except at very high setting pressures. The sealing characteristics improved as the setting pressure increased, but they were still poor. Early in the testing program ninety durometer elements were dropped as being unsatisfactory.

Early in the testing program we noted the development of rather severe discrepancies in our experimental data. There was no consistent relationship between the number of elements used and the frictional holding power. As an example, the holding power of three sixty durometer elements was less than two elements which was in turn less than one element.

This inconsistency was in direct contradiction of theory and caused us to re-examine our theory as well as our test procedure. Upon examining our testing procedure we discovered that the elements were not expanding uniformly. There was an apparent seizing of one or more rubbers on the mandrel causing them to fail to expand uniformly. In an attempt to alleviate this problem the mandrel was coated with grease to reduce the friction between mandrel and rubber. This helped, but it did not solve the problem.

Upon recognition of the problem a dry permanent lubricant was sought which offered the possibility of solving the problem. The desire for a permanent lubricant indicated that a baked on finish would be best. This would remove the requirement of field lubrication.

The first lubricant considered was tetrafluoroethylene or "Teflon" which is now available as a fuseable material for such service. After consideration of this and an epoxy material, they were both rejected in favor of Molybdenum Disulfide in an epoxy carrier. Only Teflon has a lower coefficient of friction than Molybdenum Disulfide, but the cost approaches three times that of the molybdenum compound. This material which is sixty per cent pigment (Molybdenum Disulfide) in an epoxy carrier was the first material tried and the results were quite satisfactory. Table One illustrates the improved results. Not only is the minimum setting pressure reduced, but the setting pressure imposed on the packer is more effectively converted to frictional holding power.

TABLE ONE

Setting and Pump Out Pressure

<u>Rubber Hardness</u> <u>Durometer</u>	<u>Minimum</u> <u>Setting Pressure*</u>		<u>Pump Out Pressure</u> <u>(Set with 1150 psi)</u>	
	<u>Uncoated</u>	<u>Coated</u>	<u>Uncoated</u>	<u>Coated</u>
60	190	170	200	285
70	285	255	280	370
80	300	260	400	430
90	700	600	70	110

\* Minimum setting pressure is the pressure at which the packer can no longer be moved freely in 3-inch pipe.

Design Revisions

As the importance of the shape factor, rubber volume and piston areas became more apparent, a revised design was sought which would improve these factors. The change of a single dimension could accomplish the desired result. This was the diameter of the outer mandrel. If this dimension could be reduced, all of the factors would be improved. The reduction of the mandrel diameter was conditioned by the desire to maintain a 0.75 inch internal diameter through the packer. Three design revisions were considered before one was adopted.

The revised packer design reduced the mandrel diameter from 1.75 inches to 1.3125 inches. The rubber volume then increased from 7.545 cu. in. to 10.888 cu. in. This reduced the required compression from 31.4% to 24.0%. The piston area was increased from 2.502 sq. in. to 3.680 sq. in. The shape factor of the rubber increased from 0.155 to 0.223. The ratio of piston area to load area increased from 0.788 to 0.803. All of these factors combined to more effectively utilize the available setting fluid pressure for sealing effect. An example of this is the increase in pump out pressure. The three element eighty durometer packer increased from 430 psi at 1150 setting pressure to 1,085 psi at 875 psi setting pressure.

An additional factor was the elimination of one O ring seal which reduces the potential for leakage of setting fluid.

Figure Eleven illustrates the revised design. Figures Twelve and Thirteen are photographs of the revised packer including an additional mandrel for six rubbers. From Figure Thirteen it can be seen that the revised design is somewhat simpler for field disassembly.

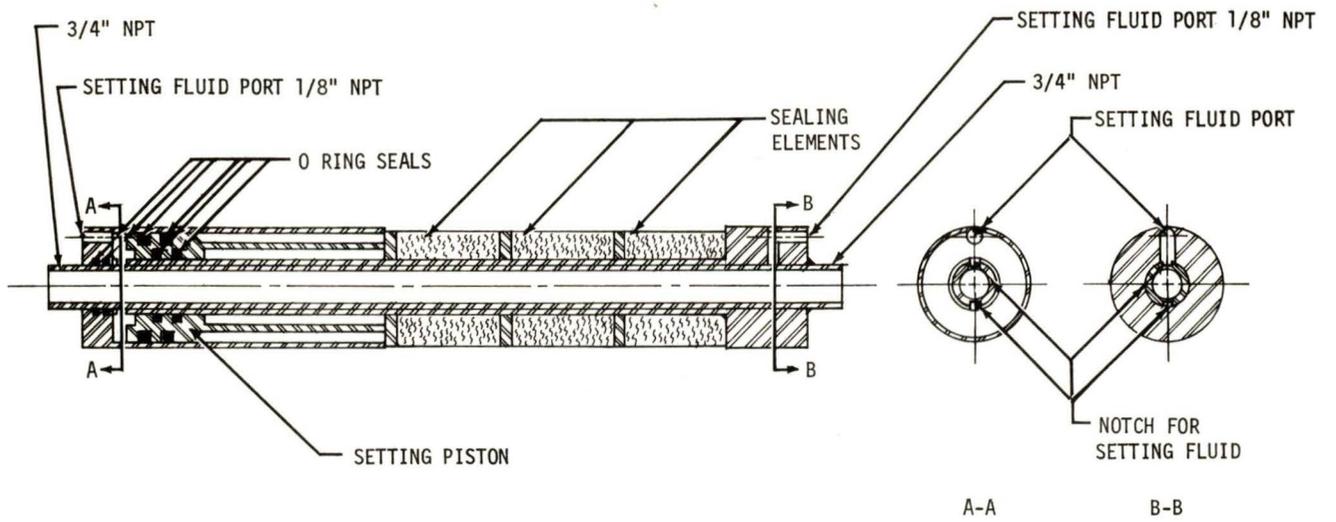


FIGURE 11 - Revised Prototype Packer Design

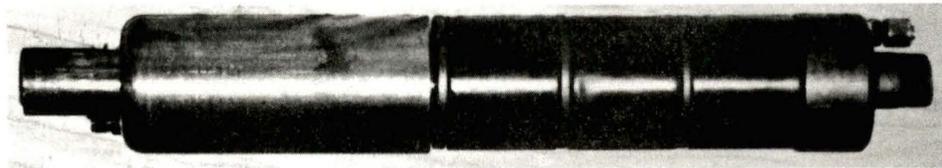


FIGURE 12 - Revised Prototype Packer

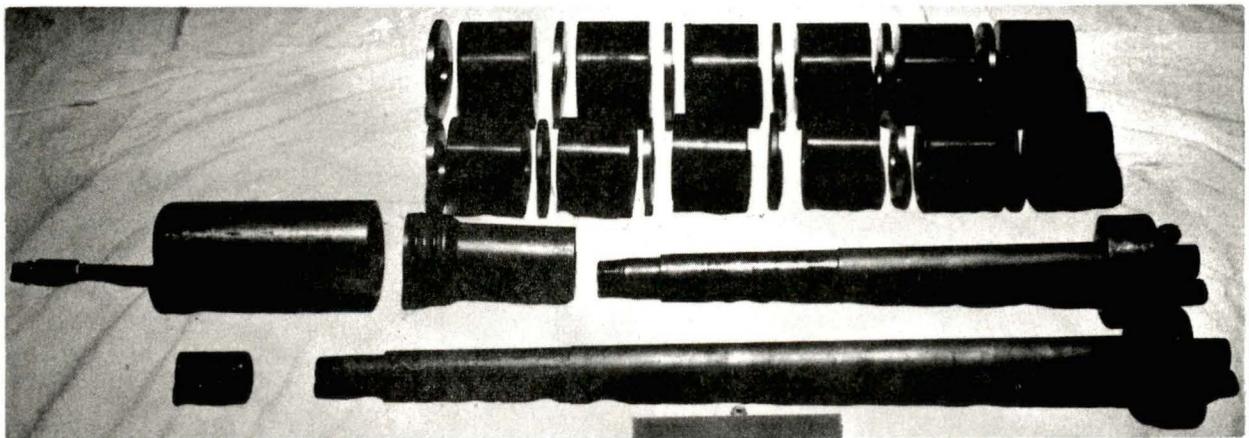
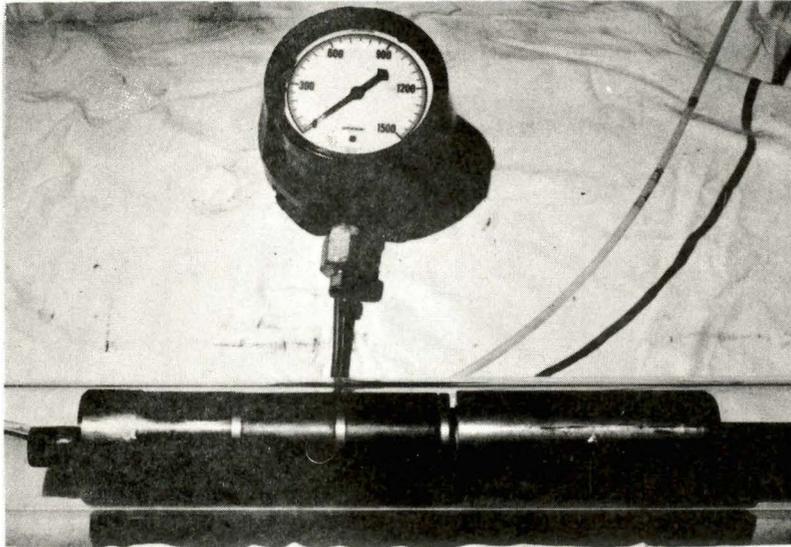


FIGURE 13 - Revised Prototype Packer Disassembled

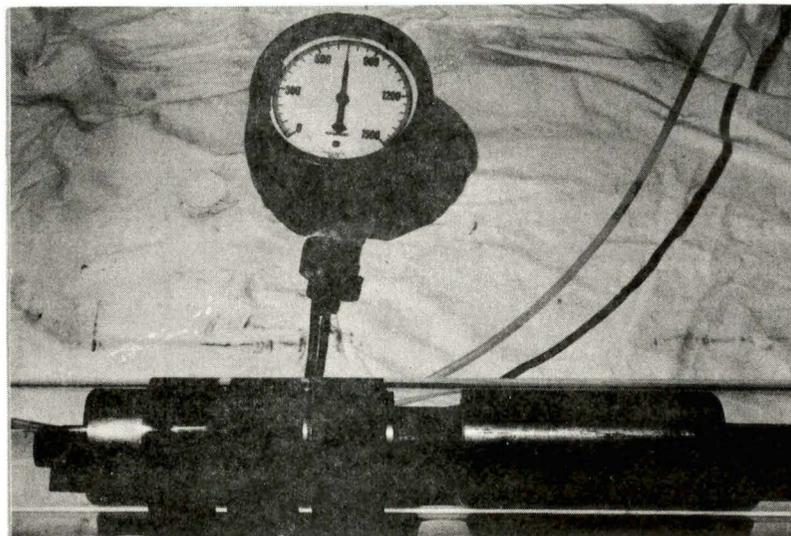
Figure Fourteen shows the revised packer in a clear plastic tube both unset and set to illustrate the setting action of the rubbers.

Figures Fifteen and Sixteen compare the pump out pressure characteristics of the original and revised prototype packers with seventy and eighty durometer elements.

Figure Seventeen compares the pump out characteristics of the revised packer and the presently available commercial packer tested.



UNSET



SET

FIGURE 14 - Revised Prototype Packer In 3 in. Tubing

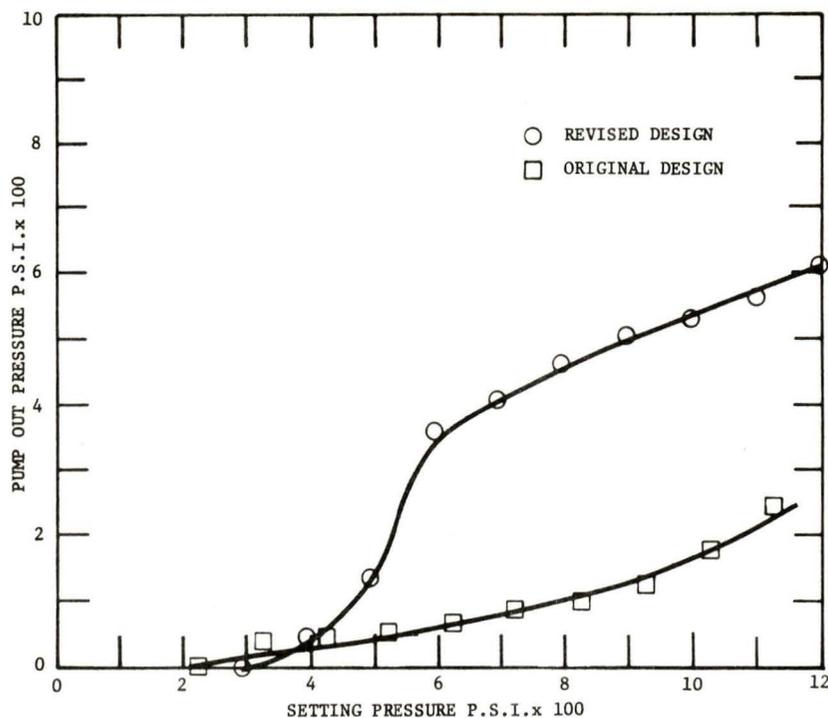


FIGURE 15 - Performance Comparison Original and Revised Packers  
70 Durometer Rubbers

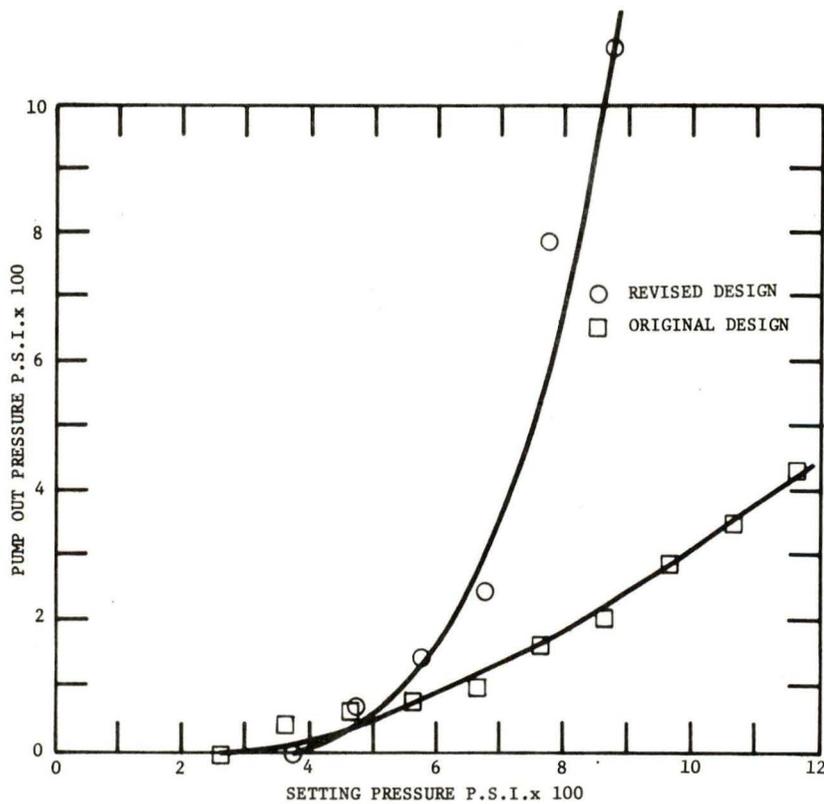


FIGURE 16 - Performance Comparison Original and Revised Packers  
80 Durometer Rubbers

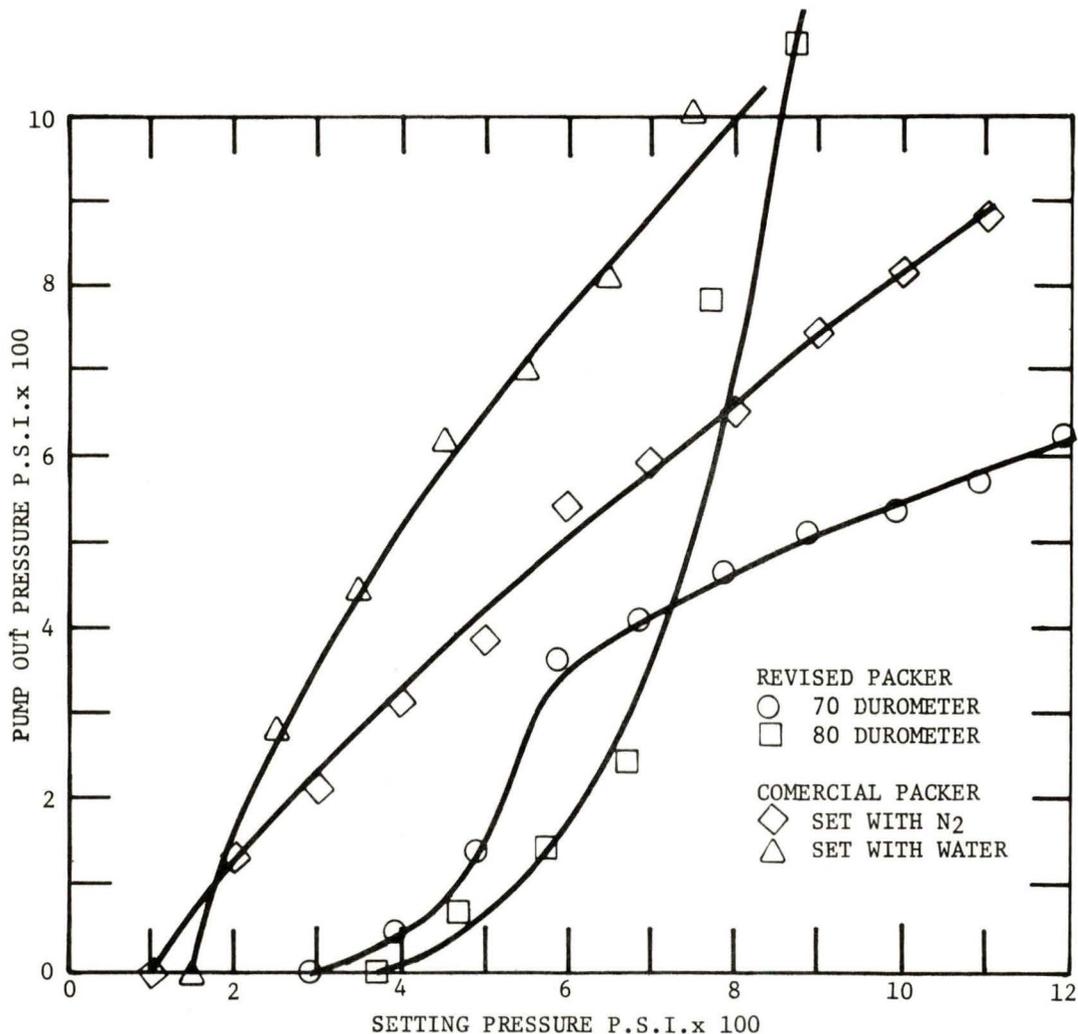


FIGURE 17 - Performance Comparison Revised Packer  
And Commercial Packer

During the testing an anomalous condition arose in that data were not repeatable. This was especially true at higher setting pressures. We also observed that there were instances where two packers in tandem did not necessarily have a higher pump out pressure than a single packer. This gave rise to some study of what the cause might be. It was thought that the age and number of cycles an element had been subjected to might have a bearing. A review of the data in chronological order indicated this to be the case. To test this theory three previously unused eighty durometer elements were placed on the packer and run through eighty-four cycles of setting and releasing. The pump out pressure was determined at each cycle. This indicated that the cycling does indeed have an effect on the pump out characteristics. Figure Eighteen illustrates this effect.

The peak efficiency of the packer appears to be between forty-five and fifty cycles. The increasing efficiency may be the result of some permanent set, creep or erosion of the rubber surface. This does show that comparisons are questionable if all of the elements are not

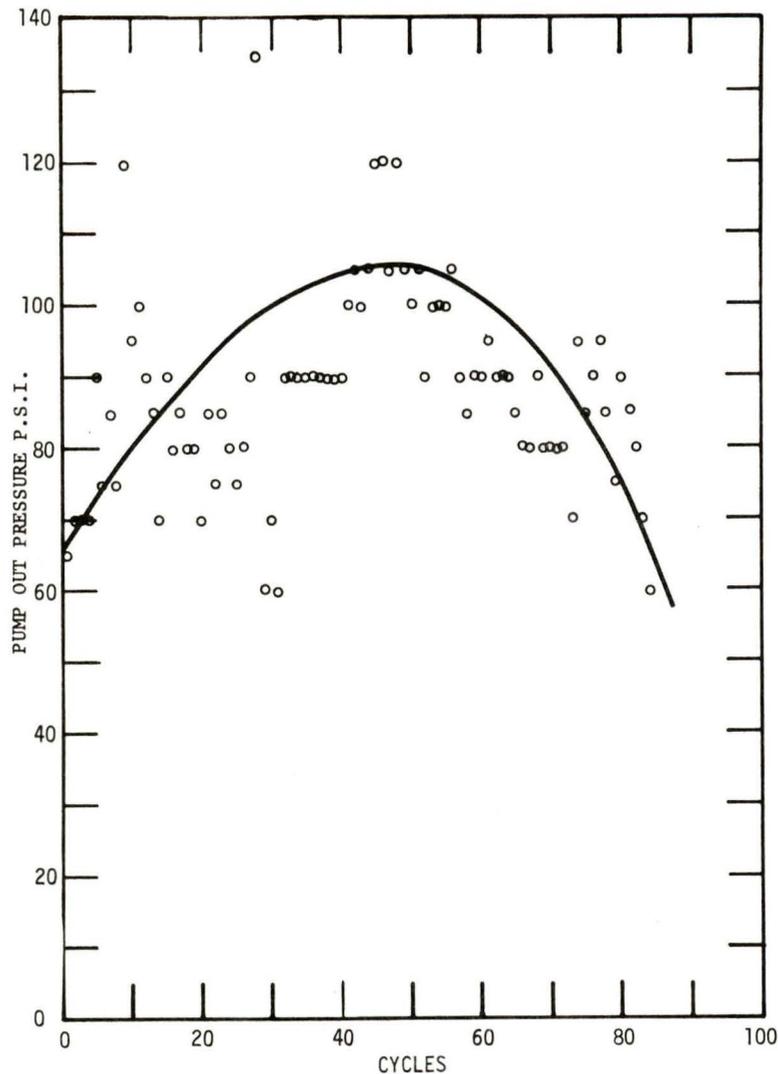


FIGURE 18 - Effect of Cycling on Pump Out Pressure

of the same age and subjected to a uniform number of cycles. This does not have a great bearing on applications in the field because of hold condition variations.

At the conclusion of the original testing variations were sought which would improve the utility of the tool. These variations were: (1) To test seventy durometer rubbers of 3.25 inch OD in 3.5 inch ID pipe; (2) test 2.75 inch rubbers composed of an eighty durometer core with a sixty durometer outer coating; and (3) test six seventy durometer rubbers on a single mandrel. These variations have led to some modification of the design which are incorporated into the production design.

Variation number one was to test larger diameter rubbers on the packer to determine if this could effectively increase the operating range of the packer without the necessity of a new packer for each change in hole size. We would expect that there would be an increase in setting pressure and a decrease in holding power with this change because the ratio of piston to rubber area would decrease. This proved to be the case but the changes were not bad. Table Two summarizes the changes which were made.

TABLE TWO

	<u>Rubber Diameter</u>	
	<u>2.75</u>	<u>3.25</u>
Piston/rubber area ratio	0.803	0.530
Shape factor	0.223	0.286
Calculated setting pressure (in pipe)	336	415
Actual setting pressure (in pipe)	290	350
Pump out pressure when set with 1,150 psi	610	332
% Compression required to set in pipe	34.0	18.6

Figure Nineteen is a comparative plot of the two conditions for seventy durometer elements. From this comparison it can be seen that at some sacrifice in efficiency the rubber size can be increased without the necessity of increasing the size of the entire packer.

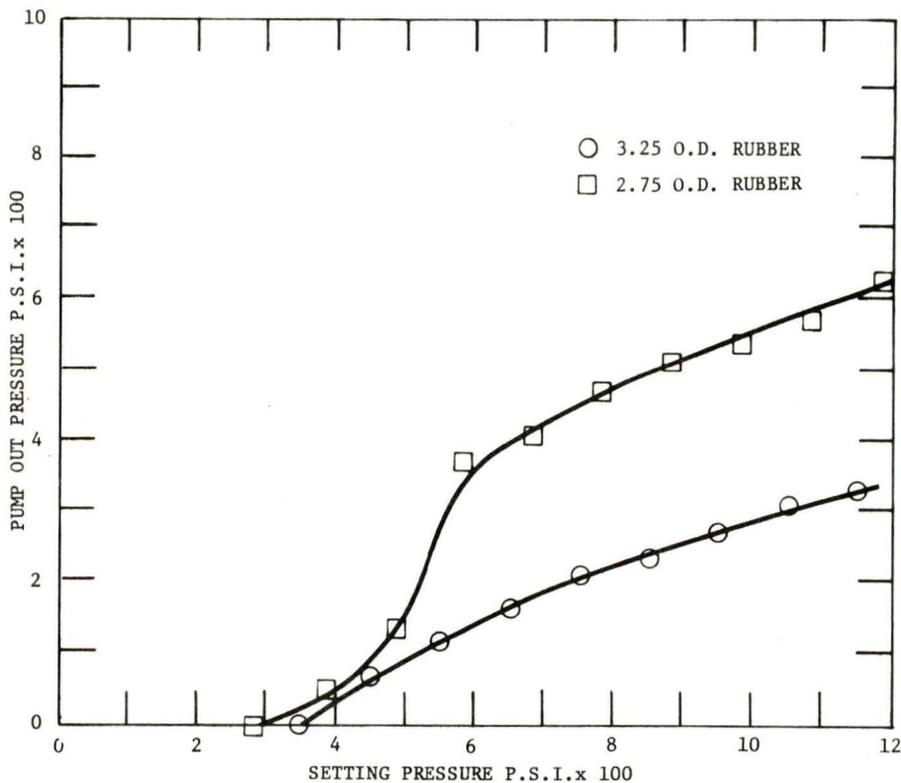


FIGURE 19 - Comparison of 2.75 and 3.25 in. Rubbers

As a result of this test the production packer has been lengthened to allow an additional spacer plate at the bottom of the mandrel. This will allow changing packer sizes by only changing rubbers and spacer plates.

Variation number two was the testing of rubbers composed of two hardnesses. The core was eighty durometer with an outer coat of sixty durometer rubber which was 1/8 inch thick. This variation created considerable delay and frustration. Our supplier of rubber products thought that he could mold the two hardnesses together, but his efforts were unsuccessful. His conclusions were that successful molding will require a core of ninety durometer rubber or harder. We can obtain the desired properties with mandrel wrapped rubber. In this process raw rubber of the desired properties is wrapped on a mandrel and then cured. Dimensional control of mandrel wrapped rubber is difficult, so the rubber is usually oversized and then machined to the desired diameter. Machining of soft rubber is a costly and difficult operation, so mandrel wrapped products are avoided whenever possible.

To test the concept eighty durometer rubbers were turned down and sixty durometer sheet rubber bonded to the cores with a rubber bonding adhesive. This proved to be satisfactory for the testing purposes, but could not be recommended as a regular procedure.

These rubbers required setting pressures equal to eighty durometer rubbers but exhibited holding power comparable to sixty durometer rubber. The sixty durometer portion extruded over the spacer plates and there was some damage to the rubber.

The concept of the soft outer portion is that it will better flow into irregularities in the borehole and create a more effective seal. The problem here is that so much of the available pressure is utilized in expanding the eighty durometer core that the effective lateral pressure is appreciably less than is achieved with seventy durometer rubber. The coefficient of friction is of course less than that achieved with either seventy or eighty durometer rubber.

We were unable to demonstrate any improvement in sealing effectiveness even in gouged pipe with this type of rubber, so we conclude that the additional cost of these rubbers would not be warranted in a production tool.

Figure Twenty is a plot comparing the combination rubber with seventy and eighty durometer rubbers.

Variation three was to test a longer mandrel tool. In this case a mandrel of sufficient length to accommodate six rubbers was fabricated. In this particular case insufficient piston travel was allowed and no

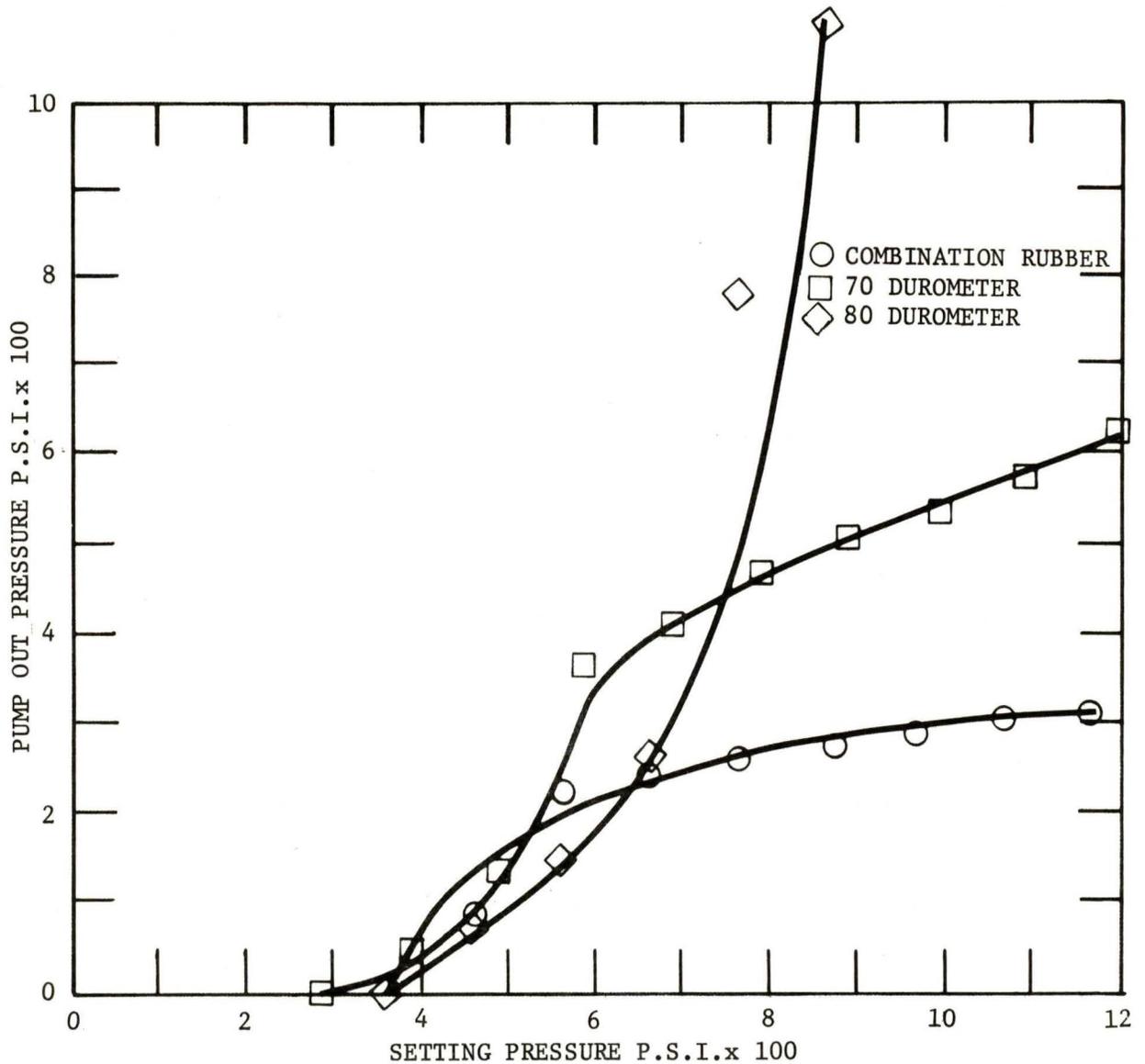


FIGURE 20 - Comparison of Combination Rubber With 70 and 80 Durometer Rubbers

improvement was noted other than having six rubbers on the mandrel. The limited piston travel prevented the full development of the desired lateral pressure.

Because in certain circumstances it may be desirable to have a packer with six rubbers, the design was further revised to provide for increased piston travel and a longer mandrel as an option with the packer. To provide this option the cylinder has been lengthened to 10.125 inches and the piston lengthened to 8.75 inches. Piston travel is limited by the length of mandrel turned down to accommodate the piston. When the piston reaches the limit of its travel, the piston encounters a shoulder on the mandrel which prevents further travel. These stops in the mandrels will prevent possible damage to the rubber or overriding of a rubber on a spacer plate and wedging in the hole.

No increase in setting pressure is required with six rubbers because the compressive force required is a function of the percentage length reduction, not the absolute length reduction as is shown in (1).

The production packer is now available with mandrels for three or six rubbers and over size rubbers. Thus for nominal additional cost a packer component system can be purchased which can be utilized in holes from three to almost four and one-half inches. Even larger diameter rubbers are possible with further reduction of setting pressure efficiency.

### Field Testing

Five packers of each of the original and revised design were field tested in horizontal holes drilled in coal. The original packers developed a serious problem of extrusion of the large piston O ring. This caused severe leakage and the testing of these packers was discontinued.

The revised packer has heavier O rings on the piston and it functioned satisfactorily.

The field testing consisted of running the packers in a three inch diameter hole and setting them. The packers remained set for periods ranging from twenty-four to seventy-two hours when they were released, retrieved and inspected. After inspection the packers were re-run and re-set.

With the revised packer the only notable occurrence was the development of a dimpled appearance of the rubbers caused apparently by irregularities in the borehole wall. This dimpled appearance was first noted after the packers had been in service for six days. There was no apparent effect on the sealing characteristics. This dimpled appearance disappeared after the packers had been out of the hole for several hours.

As a result of the problem encountered with the original design, back up rings have been incorporated into the revised packer O ring grooves and the diametrical clearance reduced from 0.03125 in. to 0.0156 in. This provides additional safety, though it may have been unnecessary.

### Optimum Rubber Properties

We also sought to determine the optimum rubber hardness. Figures Twenty-one and Twenty-two illustrate the effect of hardness on holding power.

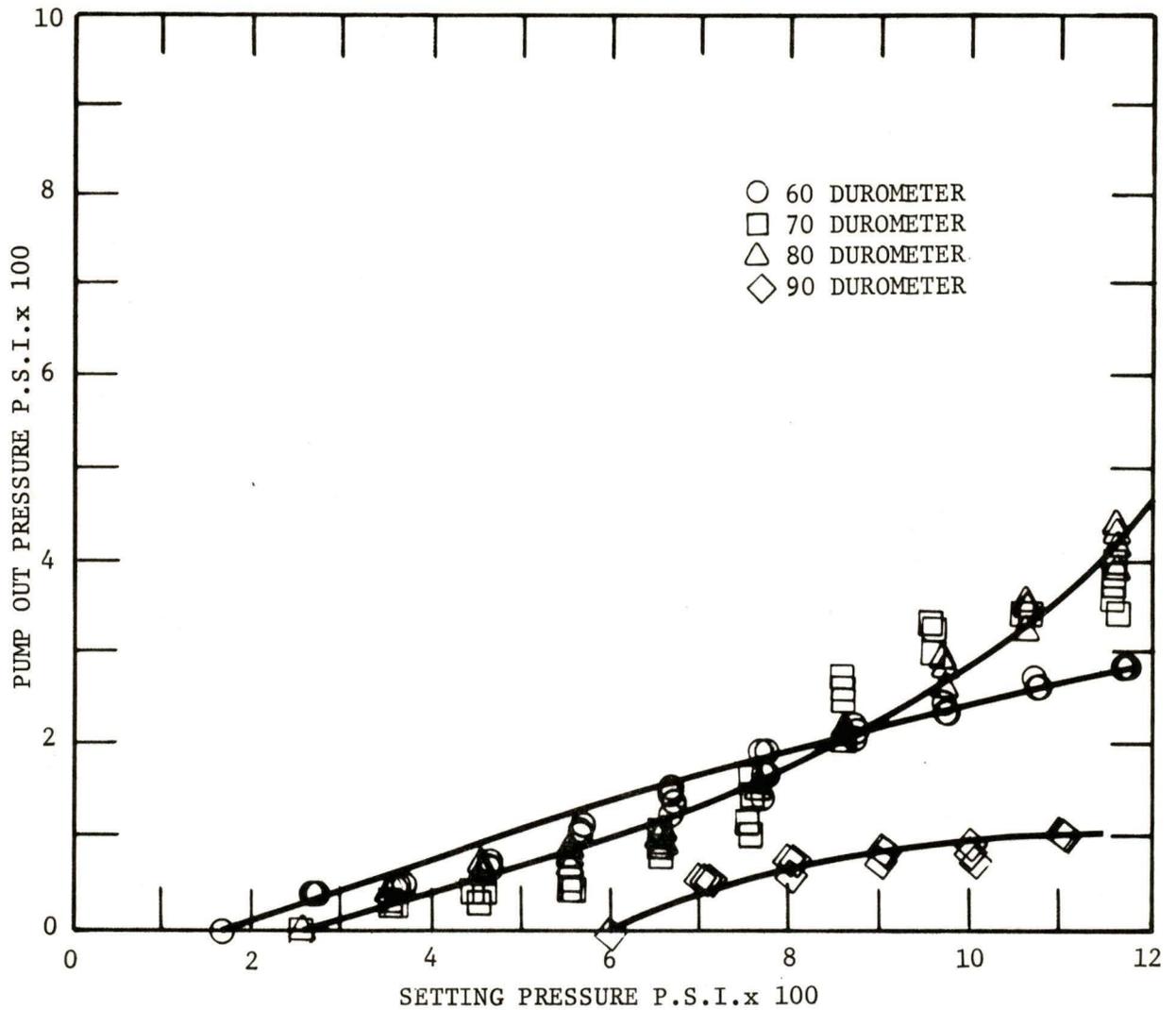


FIGURE 21 - Holding Power Three Element Packer

Figure Twenty-two especially illustrates the relative merits of the different hardness ratings. From these tests sixty and ninety durometer rubbers were rejected. Sixty durometer rubbers lack frictional power and ninety durometer rubbers lack sealing effectiveness. Another problem with the ninety durometer rubbers is that such a high pressure is required to achieve the initial setting that the differential pressure available for sealing and holding effects is seriously reduced.

At differential pressures below about five hundred psi there is little difference. At six hundred psi seventy durometer rubber is the optimum, while above eight hundred psi eighty durometer rubber is superior.

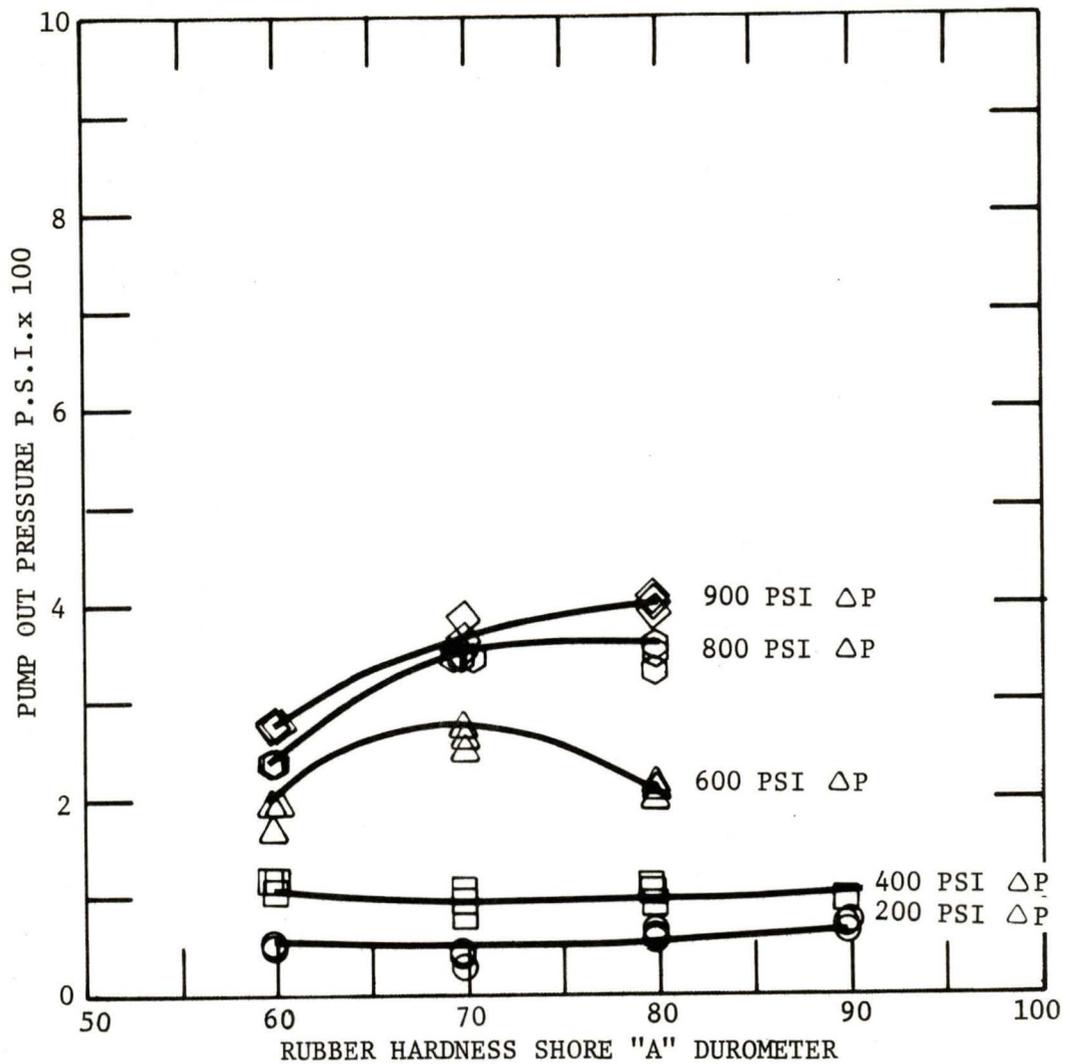


FIGURE 22 - Holding Power Three Element Packer

REFERENCES:

- (1) Handbook of Molded and Extruded Rubber, Second Edition; Good-year Tire and Rubber Company
- (2) Parker O Ring Handbook; Parker Seal Company
- (3) Design and Development of Drill Equipment; J. D. Hadden and Joseph Cervik, Bureau of Mines Technical Progress Report - 11, Methane Control Program