UNITED STATES
DEPARTMENT OF THE INTERIOR
BUREAU OF RECLAMATION

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HYDRAULIC MODEL STUDIES OF SPHERE VALVE SEAL DESIGNS

FOR

SHASTA POWER PLANT

- - - - -

By

E. S. GRAY, ASSISTANT ENGINEER

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Denver, Colorado,
June 11, 1943

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MEMORANDUM TO CHIEF DESIGNING ENGINEER

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FOR
SHASTA POWER PLANT

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By E. S. GRAY, ASSISTANT ENGINEER

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Under Direction of

J. E. WARNOCK, ENGINEER
and
R. F. BLANKS, SENIOR ENGINEER

- - -

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MEMORANDUM TO CHIEF DESIGNING ENGINEER
(E. S. Gray through J. E. Warnock)

Subject: Hydraulic model studies of sphere valve seal designs for Shasta power plant - Central Valley project.

1. Introduction. The sphere valve tests involved the 30-inch sphere valve (figure 1) upstream from the station service units at Shasta Dam. The valve proper is a spherical plug which rotates 90 degrees and is hydraulically operated by a rotor and a stator, both of which are enclosed in a cylindrical housing. There is one operating unit on each end of the spherical plug, comprising two rotors and two stators and one annular space formed by the head and the walls of the rotor housing.

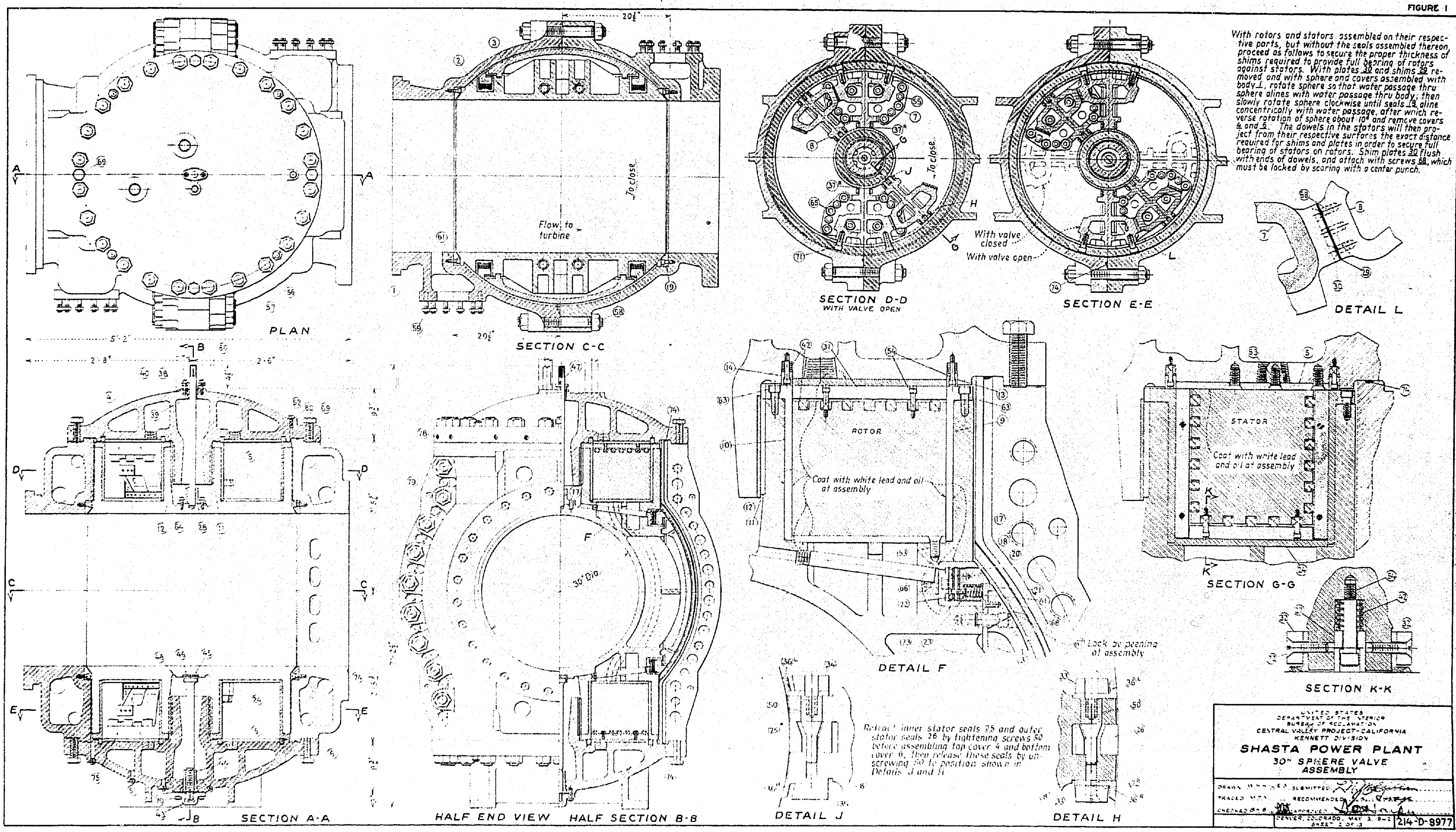
The rotor (detail F and item 7 in section D-D) is part of the plug and forces the plug to rotate in the open or closed position when actuated. Section C-C is a plan in the open position, whereas section B-B is an end view in the open position.

The stator (section G-G and item 8 in section D-D) is attached to the head of the valve, one on each end, and is stationary. The rotor moves in the hollow, cylindrical, annular spaces, two on each end of the plug. The rotor divides each space so that four chambers are formed. The rotor is actuated by oil pressure in two chambers. The direction of rotation is reversed by manipulating the three-way control valve to apply pressure in the other two chambers.

Five types of seals are used in this valve: sixteen sliding, straight bar seals; sliding-bar ring seals; the sliding spherical-bar ring seal; the cupped seal ring; and the wiping seal. In these tests consideration has been given only to the first two types.

There are four sliding, straight bar seals in each of the four annular chambers (or a total of sixteen sliding, straight bar seals): (1) the

FIGURE 1



With rotors and stators assembled on their respective parts, but without the seals assembled thereon, proceed as follows to secure the proper thickness of shims required to provide full bearing of rotors against stators. With plates 30 and shims 29 removed and with sphere and covers assembled with body 1, rotate sphere so that water passage thru sphere alines with water passage thru body; then slowly rotate sphere clockwise until seals 19 aline concentrically with water passage, after which reverse rotation of sphere about 10° and remove covers 4 and 6. The dowels in the stators will then project from their respective surfaces the exact distance required for shims and plates in order to secure full bearing of stators on rotors. Shim plates 30 flush with ends of dowels, and attach with screws 68, which must be locked by scoring with a center punch.

UNITED STATES
DEPARTMENT OF THE INTERIOR
BUREAU OF RECLAMATION
CENTRAL VALLEY PROJECT-CALIFORNIA
KENNETT DIVISION

SHASTA POWER PLANT

30" SPHERE VALVE ASSEMBLY

DRAWN [illegible] SUBMITTED [signature]
TRACED [illegible] RECOMMENDED [signature]
CHECKED [illegible] APPROVED [signature]

DENVER, COLORADO, MAY 3, 1942
SHEET 2 OF 3

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horizontal upper seal on each rotor (detail F, item 31); (2) the horizontal lower seal on each stator (section K-K, item 27), and two vertical seals on each stator; (3) the inner seal (detail J, item 25); and (4) the outer seal (detail H, item 26). All of the sixteen separate, sliding bar seals are constant-contact and spring-actuated. The vertical seal bars have circular surfaces.

There are four sliding-bar, horizontal, ring seals in each valve, one inner ring seal (detail F, item 14) and one outer ring seal (detail F, item 13) in both the upper and the lower heads. These ring seals are in constant contact against the walls of the annular space (section G-G) and are actuated by springs. Section K-K shows the details. Both rings must seal from either direction.

The remaining three types of seals were not tested. There are two sliding, spherical-bar ring seals in a vertical plane (detail F, item 18) to seal penstock pressure from the outer, sliding-bar ring seal (detail F, item 13). These seal rings are retractable, operate with hydraulic pressure applied underneath, and are spring-retracted when this pressure is relieved. There are also two cupped rings on each spherical bar (detail F, item 23), of expansive material, to seal the actuating chamber under the ring. Two ring seals are provided in each valve so that it can be sealed when the flow is in either direction or with pressures on both sides.

The wiping, flexible seals protect the sliding bar seals, wiping the surfaces ahead of the bars and also performing a sealing. These wiping seals and clamp bars are shaped to conform to the various surfaces on which they slide or scrape (section K-K, item 32, and details J and H, items 36L and 36R). In addition there are two wiping seals along the vertical, stationary, inner contact surface of the rotor (section D-D, item 37R) which simply seal the crack between the contacting surfaces where the sliding fit must be allowed for assembly purposes.

2. Problems for model study. It has been thought generally that the unit sealing pressure must be considerable to effect a seal. For design

purposes the unit pressure on the sealing surfaces has been taken as five times the unit head pressure, which means the sealing surface must be narrow to have a high unit pressure on it; otherwise the force to be applied on the seal bar must be unreasonably great.

The first part of the tests was conducted to determine the unit contact pressure required to effect a seal against any particular head. This was completely tested in the visual, sectional seal model and corroborated in the 10-inch, ring-seal model.

It was desired to test the original seal design (figure 2A) proposed for the 30-inch sphere valve. The seal bar had very small clearances to reduce leakage by way of the slot or groove, but nevertheless it was desired to observe or measure the leakage quantitatively. The quality of the seal at the seat also was to be observed for narrow or wide contact surfaces. In the case of the straight seal bars, leakage through the clearance would probably be slight because the bar is pressed against the downstream face of the groove; but in the case of the ring-seal bars, the clearances are not materially changed with the head, so that leakage might be considerable even through an allowed clearance of 0.0025 inch. Other designs, therefore, might have to be tried in tests.

3. The models. Two hydraulic models were used in which seal assemblies having full-sized cross sections could be tested. The sliding-bar seal design, for the sphere valve (figure 3C), was made to fit into the visual, sectional seal model (figure 2). The sealing unit was a full-sized section of the prototype, 2-1/2 inches long. The housing of the model was a rectangular box with transparent plates for the front and back, made of 1/2-inch Lucite plastic plate, permitting visual observation and photography of the sealing unit inside. Supply lines were connected to provide penstock pressure upstream from the seal bar and to provide pressure for operating it. Relief lines and valves from these supply lines were also connected. Bourdon gages were installed at suitable points to measure penstock pressure under the seal bar.

The seal assembly was mounted on an adjustable base inside of the box, and its position with respect to the seat could be varied without disassembling the installation. A sliding seat was provided which could be moved at any time during the test. A piezometer connection from chamber A was made through the seat to open to a hole in the top plate of the box and thence by a suitable pipe connection to a pressure gage or to a relief valve.

There was also a hand-operated lever provided for operating the plunger in the base plate of the box. The plunger could be connected to the seal bar and, when so connected, forces could be applied to the seal bar and measured. Figure 4A shows the test apparatus, and figure 2 shows the principal details.

The 10-inch, ring-seal model (figures 3 and 5) was designed to represent a circular gate in which a 10-inch diameter ring seal of prototype cross section could be installed.

The penstock supply was a 3-inch line into and through the cap plate of the model. From chamber C under the cap plate, the water was forced outwardly between the seat on the cap plate and the nose or seating surface of the seal ring. When the seal ring was forced against the seat, the penstock flow ceased and the gate was sealed.

The supply line by which the seal ring was operated was a 1/2-inch line into the bottom of the model housing which opened to an annular chamber and finally to chamber B under the seal ring through the holes provided for the seal-ring spring.

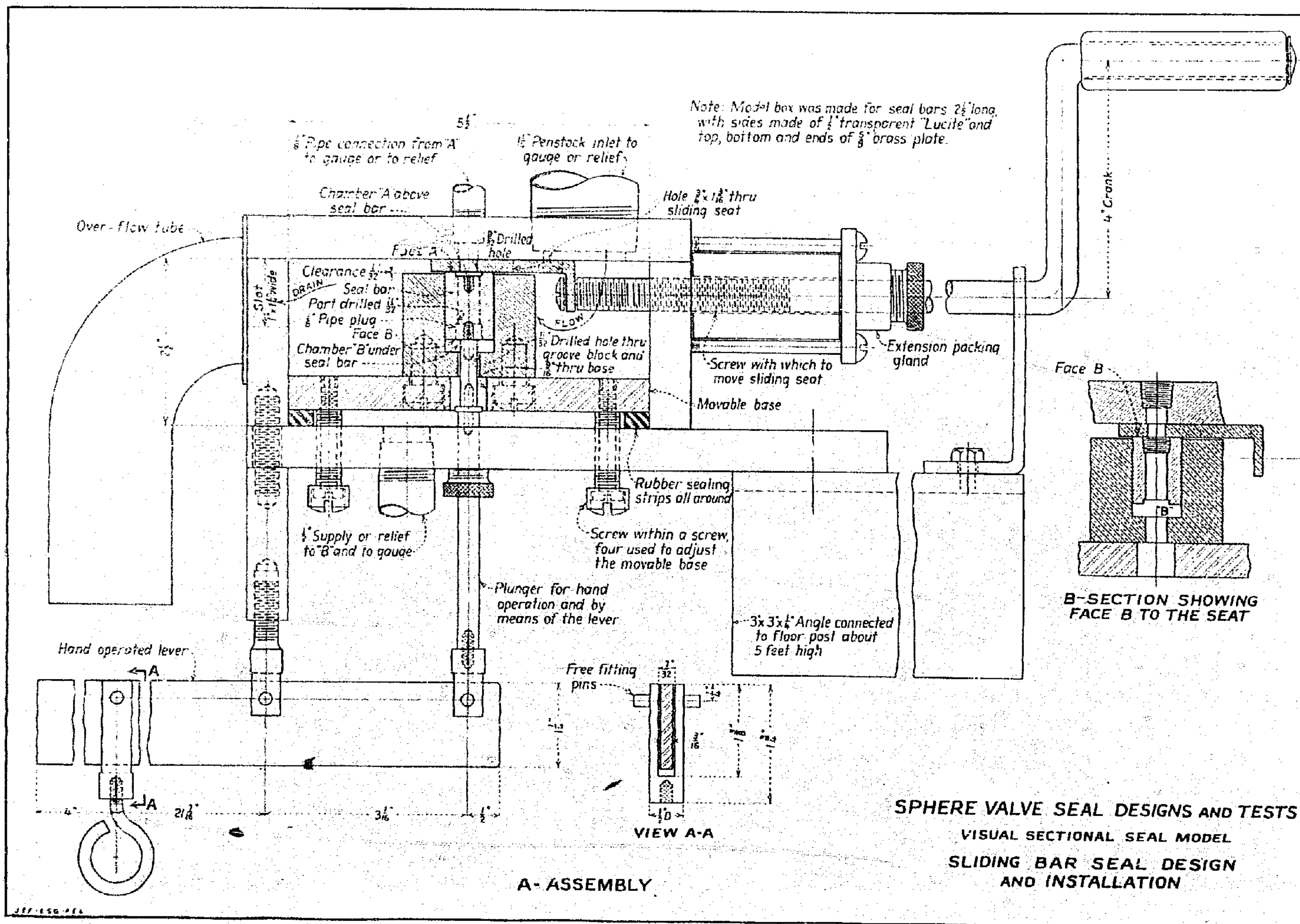
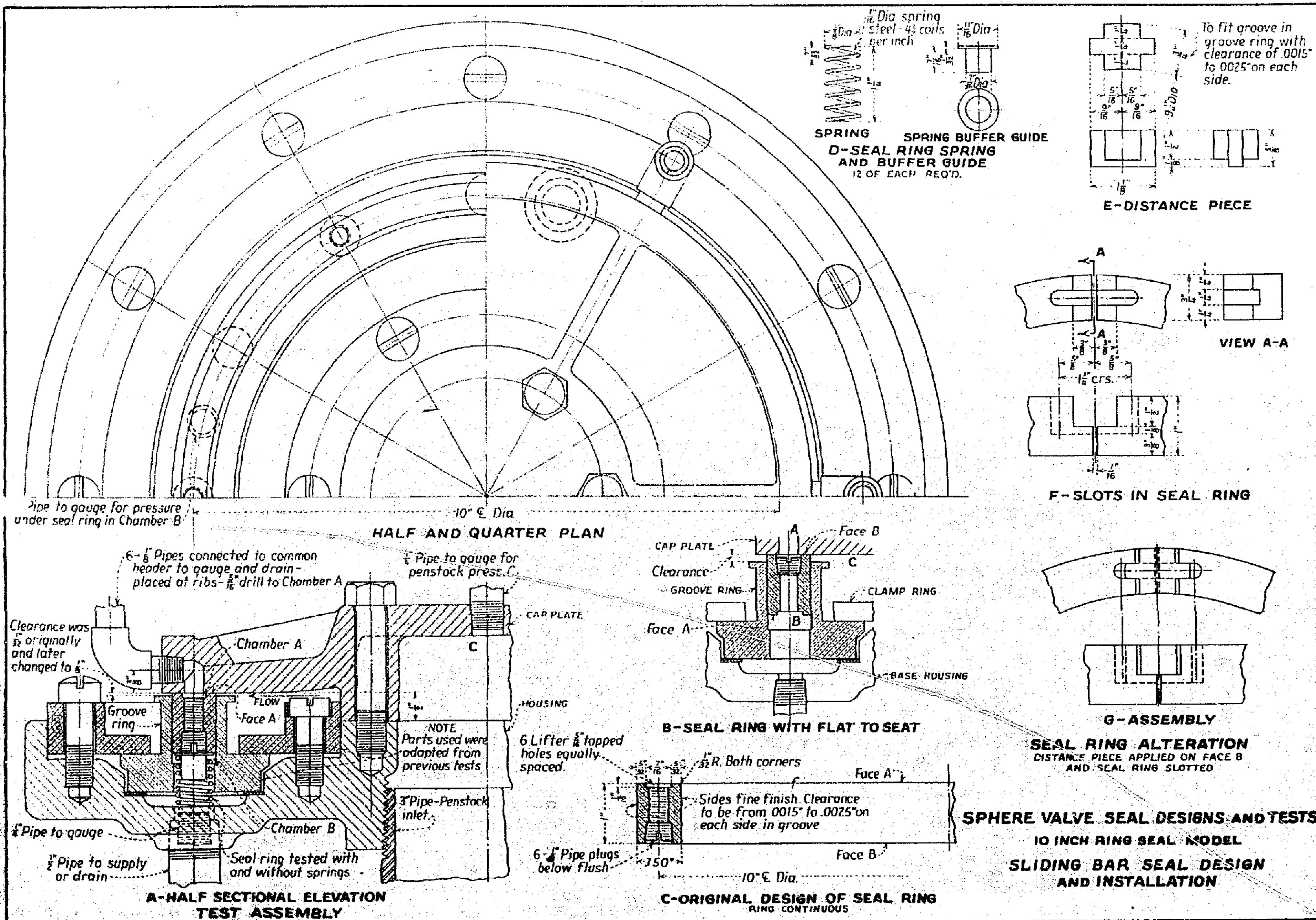
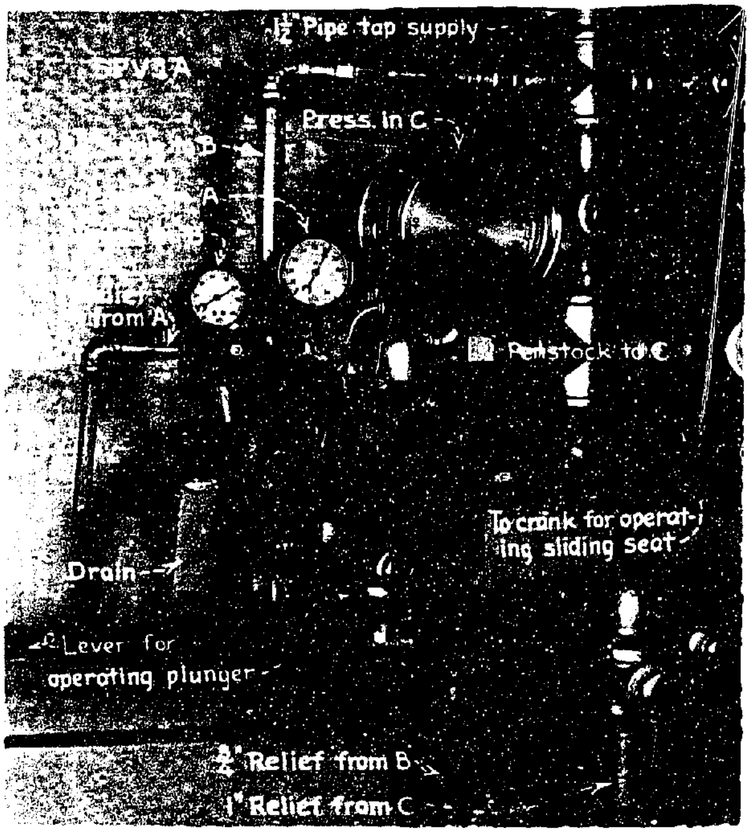

Note: Model box was made for seal bars 2½" long, with sides made of ½" transparent "Lucite" and top, bottom and ends of ⅝" brass plate.
5½"
⅛" Pipe connection from "A" to gauge or to relief
1½" Penstock inlet to gauge or relief
Chamber "A" above seal bar
Hole thru sliding seat
Over-flow tube
Drilled hole
Face A
Clearance
DRAIN
Seal bar
Part drilled
⅛" Pipe plug
Face B
Chamber "B" under seal bar
Slot
FLOW
Drilled hole thru groove block and thru base
Screw with which to move sliding seat
Extension packing gland
4" Crank
Movable base
Rubber sealing strips all around
¼" Supply or relief to "B" and to gauge
Screw within a screw, four used to adjust the movable base
Plunger for hand operation and by means of the lever
3"x3"x¼" Angle connected to floor post about 5 feet high
Hand operated lever
Free fitting pins
VIEW A-A
Face B
"B"
B-SECTION SHOWING FACE B TO THE SEAT
A-ASSEMBLY
SPHERE VALVE SEAL DESIGNS AND TESTS
VISUAL SECTIONAL SEAL MODEL
SLIDING BAR SEAL DESIGN AND INSTALLATION

FIGURE 2

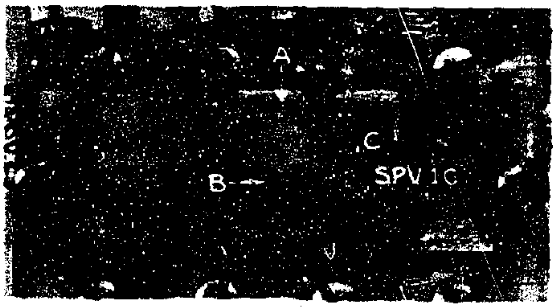


SPHERE VALVE SEAL DESIGNS AND TESTS
10 INCH RING SEAL MODEL
SLIDING BAR SEAL DESIGN AND INSTALLATION

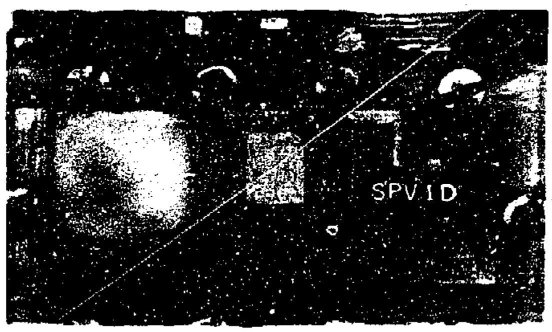
FIGURE 3



A-Test apparatus showing the sliding bar seal installed with face A (legs) of seal bar to the seat and sealed at 70 lb. per sq.in. penstock pressure, pressure in chambers A and B was 60 lb.



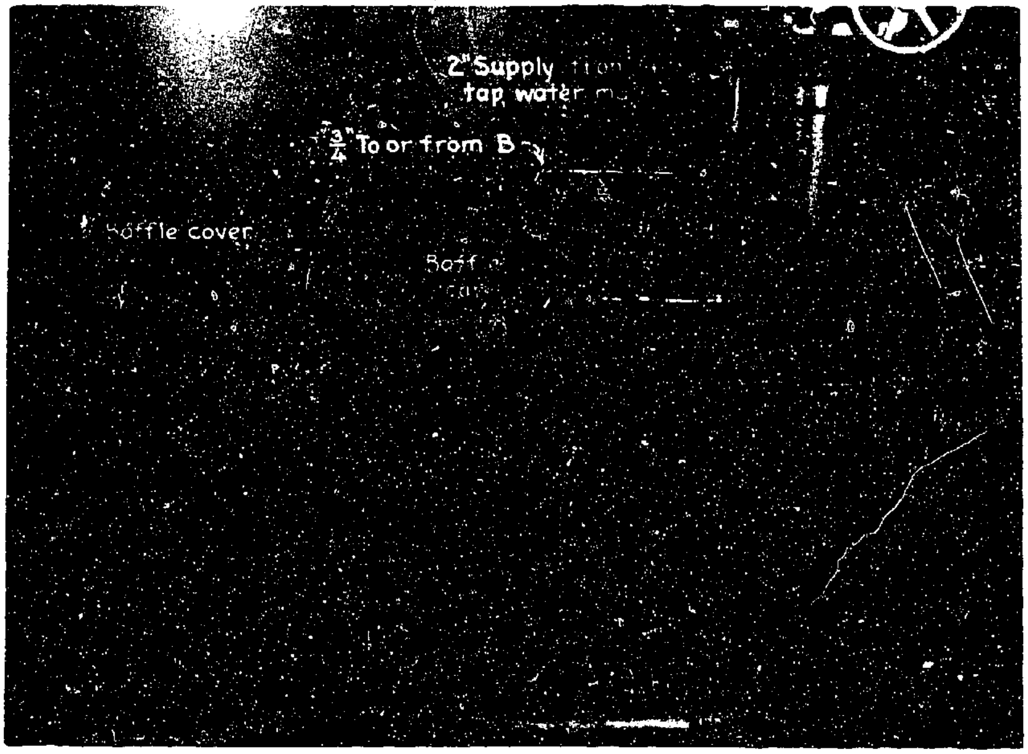
B-Face A of seal bar to the seat. Seal bar remains sealed at 70 lb. per sq.in. with pressure in chamber B reduced to zero--additional force was required to retract it.



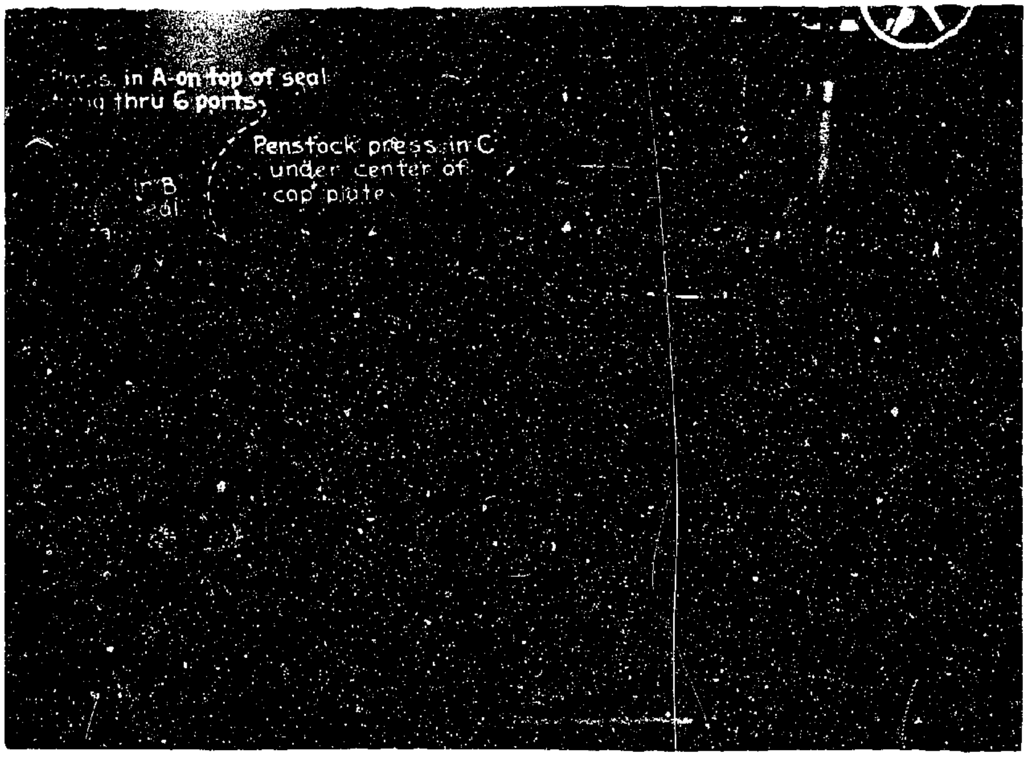
C-Seal bar retracted 1/32 inch with pressure in chamber C 70 lb. and drains to chambers A and B closed--pressure in A and B was 40 lb.

D-Seal bar retracted to floating position with a gap of less than 1/32 inch and penstock pressure of 70 lb. Pressure in chamber B was reduced to zero.

VISUAL, SECTIONAL SEAL MODEL APPARATUS



A-Test arrangement with baffle can and cover in place.



B-Baffle can and cover removed.

10-INCH RING-SEAL MODEL TEST APPARATUS

Relief lines and valves were connected to the supply lines to control the pressures and volumes. Bourdon gages were installed at suitable points. The gage measuring penstock pressure C was connected into the center of the cap plate, and the one measuring pressure in chamber B was connected into the annular chamber of the housing.

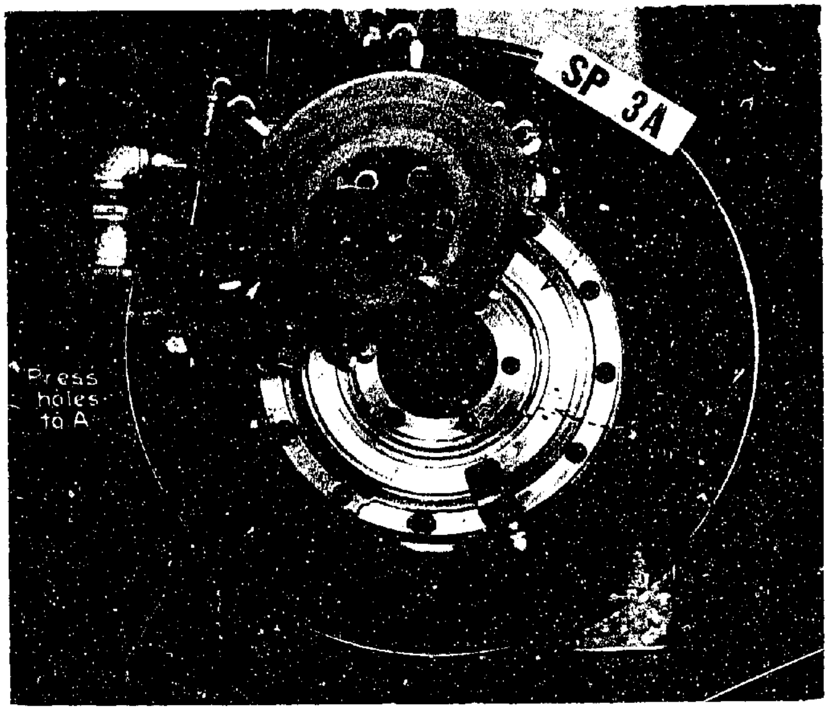
In addition, there were six pressure or relief ports opening to the seat on the cap plate and centering on the seal ring (figure 6). The six connections were brought to a common header provided with a pressure gage and a relief valve. The necessity for these pressure or relief ports was that the pressure or leakage conditions in chamber A, at the seating surfaces, might be determined.

The distance between the top of the groove ring and the seat is termed the clearance, and the distance between the nose of the seal bar and the seat is termed the gap. The clearance could be varied by shims under the cap plate. The gap was controlled by the design or by hydraulic forces acting on the seal bar.

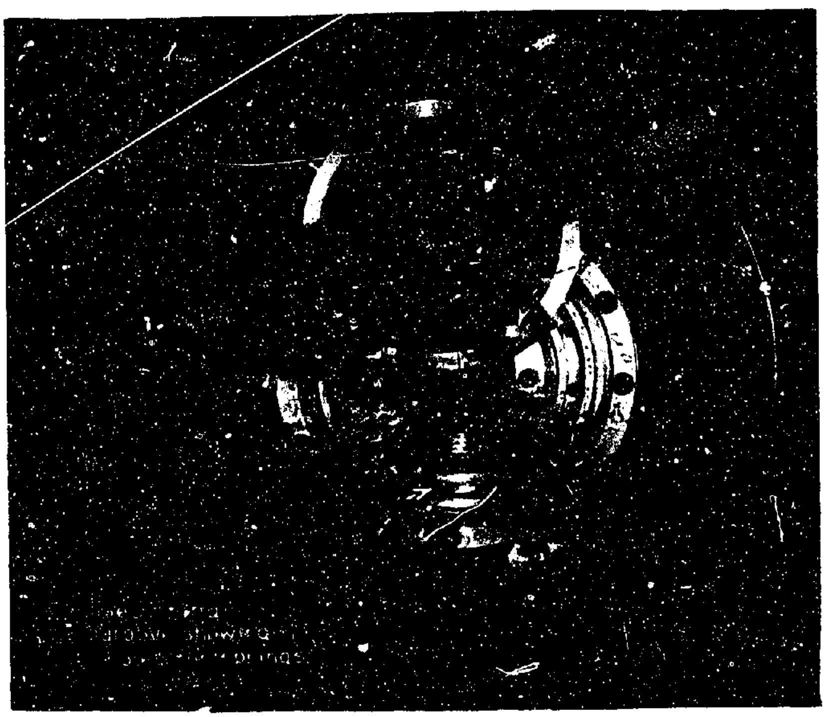
4. Operation of the models. With the available supply systems it was possible to control water pressures and large volumes over all ranges up to and including 110 pounds per square inch. Higher pressure could be obtained with a hand pump arrangement, but with practically no volume.

The tests consisted of measurements of penstock pressure in chamber C (figures 2 and 3); pressure in chamber B under the seal bar; pressure or leakage in chamber A above the seal bar; and the load required on the lever in the visual model to move or hold the seal bar in any position. Visual observation was augmented by photographs, and various changes were made to the assembly and to the seal ring or bar to determine the operational characteristics of the seal ring or bar in sealing or retracting, or of the unit as a whole.

For any penstock flow, sealing pressure in chamber B or load on the lever was determined by admitting pressure slowly in chamber B or loading the lever and observing the action of the seal bar. At the point of seal-



A-Seal ring installed with its distance piece. Cap plate inverted showing pressure holes centering on seal ring to chamber A.



B-Seal ring and distance piece removed.

10-INCH RING-SEAL MODEL TEST APPARATUS

ing, the penstock flow was stopped. The quality of the seal was determined by the amount of leakage between the sealing surfaces. Breaking pressure was determined by relieving the pressure slowly in chamber B or load on the lever and noting the point at which penstock flow began. The sealing pressure and the breaking pressure usually were not the same, due either to inherent friction in the parts or to the end friction of the seal bar (in the visual model) or to the extent of surfaces in contact, or to all factors combined.

If the valve supplying chamber B was opened too rapidly, while penstock supply was maintained, a slam of more or less violent nature was produced by the seal bar as it closed, and consequently water hammer resulted.

8. The visual, sectional, seal-model tests. The original design (figure 4) for the straight bar seals was tested in the visual, sectional model with the seal bar in two positions and under at least two conditions.

(a) With face A to the seat (figure 2A) and the port through the bar closed, sealing took place very easily, and once the seal bar was closed, pressure in chamber B or pressure on the lever could be entirely relieved without the seal breaking (figure 4B). This was done for pressures as high as 110 pounds per square inch in chamber C. For this sealed condition, however, the drain from chamber A had to be opened because pressure as high as 9/10 of the head developed in it. This pressure in chamber A was due to leakage over the upstream leg of face A. Readjusting the level of the legs produced no better sealing, nor was sealing improved when sealing pressure was applied on the seal bar with the lever.

The greatest negative pressure was developed over face A when the gap was 1/32 inch with the drain from A closed (figure 4C). Since the clearance was also 1/32 inch, face A was flush with the top of the groove block. It required an upward pull of eight pounds on the lever to retract the seal bar farther; or, a force of about 50 pounds was exerted downward on the seal bar. When the negative pressure was overcome, the bar instantly was forced to the bottom of the groove by positive water pressure above it. Sealing from this position occurred with a slam which could not be prevented normally.

(b) With face A to the seat and the port open and drain from chamber A closed, the leakage over the downstream leg of face A was observed to be about the same as before. It was also observed that the leakage through the downstream surfaces in the groove block was more than over the legs at the seat. After careful remachining of the parts and painstaking installation (both of which were done several times), it was apparent that a perfect seal could not be attained in this manner. Consequently, the parts were ground on a flat plate. Face A and face B of the seal bar and the sliding and seating surfaces on the sliding seat were ground on a glass plate, using FFF emery powder and No. 304 pumice for finishing. In retests, leakage was only in drops over face A. With the port closed the seat was moved back until only the upstream leg sealed. In this position the seal bar closed or opened with a force as small as one pound on the seal bar, applied through the plunger. The condition was the same when both legs were in contact with the seat and chamber A drained. The sides of the box were clamped against the seal bar, and leakage was reduced to a thin stream mainly coming by at the ends of the two legs.

(c) The seal bar was inverted so that face B (flat of seal bar) was now to the seat (figure 2B). The port was closed and face B was adjusted so that the downstream edge was definitely in contact. With the drain from chamber A closed, the pressure in chamber A almost equaled penstock pressure. At 70 pounds per square inch in chamber C, a force of 19 pounds was applied to the end of the lever at a ratio of 7:1 to cause sealing, while pressure in chamber B was zero. The sealing force on the bar was thus 19 x 7 = 133 lb., and unsealing pressure, if water pressure is acting over the entire seal-bar face is 3/4 x 70 x 2.5 = 131 lb., the seal-bar face being 3/4 inch wide and 2.5 inches long. The discrepancy between 133 and 131 pounds may be made up if the weight of the lever is taken into account, so that the unit contact sealing pressure was practically zero.

The leakage of one quart in 4-1/2 minutes in (c) was about one-third that in (a), where leakage was one quart in 1-1/2 minutes. From this test it was concluded that the wider, flat seal was better than the seal made by the two narrow legs, so far as the quantity of leakage was concerned, and it disproved the common belief that a small contact surface with great pressure was needed to effect a seal.

In another test the seat was withdrawn with the crank, a turn at a time, and simultaneously the weight on the lever was reduced uniformly to maintain the same unit pressure on the seating surfaces. Leakage remained practically constant during the traverse, sometimes decreasing to drops. The seal was just as effective with 1/16 inch in contact as with 3/4 inch in contact. From this test it was concluded that only line contact is necessary to make a perfect seal, because the narrow surface sealed just as well as the wide surface.

Upon clamping the sides of the box against the seal bar, leakage through the assembly was reduced to drops so that quantitative tests were abandoned.

(d) With the face B to the seat and the port open, the conditions were the same as in (c) because in the sealed position there was water pressure at A in either case. The main problem was to get the flat nose of the seal bar to seat perfectly. To do this involved the tilting of the bar in the groove block. When under a pressure test, slight tilting generally occurred which could be seen even in the 0.002-inch clearance crack. The tilting was counterclockwise (downstream) when penstock pressure of about 75 pounds per square inch was applied. This affected the lining-up process and also the area of the surface in contact.

(e) It was further desired to find the effect and to observe the results of the application of grease on the surfaces in the groove block. Two kinds of grease were used, a paraffin grease called "Ezit" and vaseline. The Ezit grease is waterproof, does not injure rubber, and has good sticking qualities; the vaseline was not tenacious.

First, only pressure in chamber B was applied up to 75 pounds per square inch for both kinds of grease. The Ezit grease was estimated to be about five times better than the vaseline in preventing a blowout and the resulting leakage.

Second, penstock pressure was applied in chambers C and B. The seal bar was crowded to the downstream wall in this case, but with Ezit grease, leakage was in drops, where with the vaseline it was in thin streams.

6. The 10-inch, ring-seal model tests. Difficulty was experienced in these tests from two sources. The first was from the scale and dirt and fine sand which continually broke loose from the old piping used; in the second, to raise or lower the seal ring was especially difficult because the slightest tilting or cocking would cause it to bind, and neither hydraulic pressure nor force would move it unless the seal ring was first straightened axially in its groove. Disassembly and cleaning was necessary a number of times to get rid of scale and dirt, but finally fine screens (200-mesh) were installed both over the penstock inlet to chamber C and over the inlet to channel B under the seal ring. Then successful operation of the model was possible.

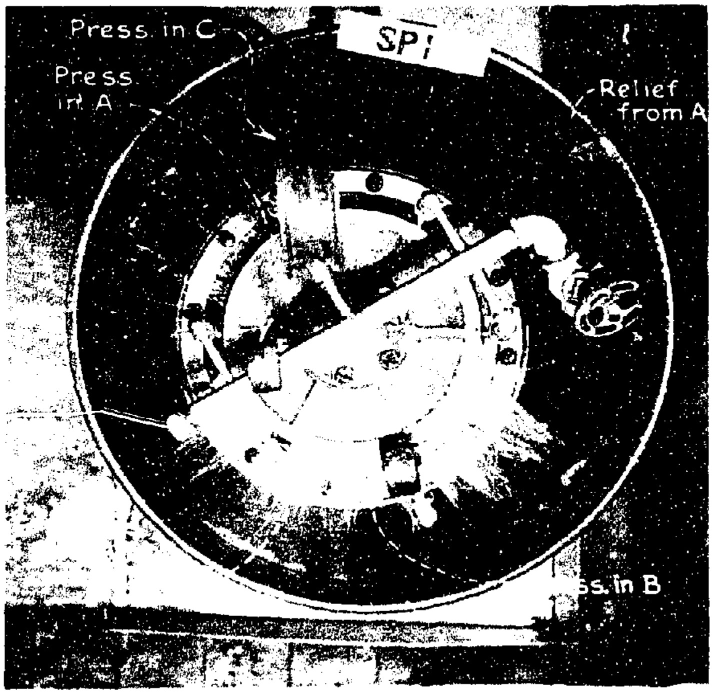
Ports through the seal ring were not contemplated in the original design for the sphere-valve seals, but there was a possibility that they might be used after results of the tests in the visual, sectional seal model with face A to the seat, and six ports were provided in the 10-inch seal ring. These also served in applying the lifter bar across any two holes for removing the seal ring. Tests, however, were conducted only with the ports closed, because from the very first it was seen that the inner edge of the seal ring sealed very well; so leakage to chamber A was not a problem. The reason for considering ports through the seal bar was that if pressure in chamber A was to be constantly a matter of fact, deliberate conditions might just as well be provided so that pressure in chamber A would always be maintained equal to that in chamber B.

The seal ring was installed with the twelve 5/8-inch diameter springs and buffer guides in place (figures 3A and D) and the ports closed. The seal ring was tested in the two positions; with face A (legs) to the seat (figure 3A) and with face B (flat) to the seat (figure 3B). The clearance was 1/32 inch.

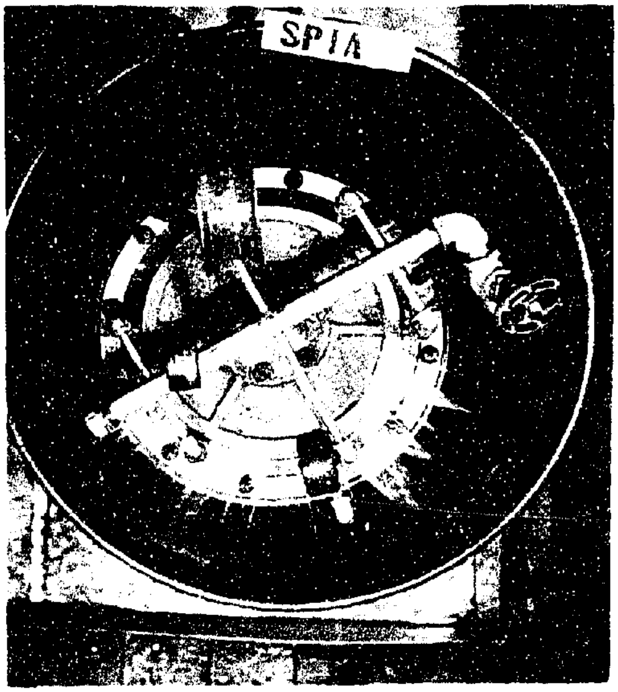
Leakage occurred in the sealed positions for both cases and sprayed out in a flat thin sheet below the cap plate (figures 7A, B, C, and D). In A and B the legs were to the seat and in C and D the flat was to the seat. It made no difference as to the amount of leakage whether pressure was applied in chamber B only or both in chambers B and C. By applying pressure in chamber B only, it was seen that leakage to the outside was greater than that toward the inside, by comparing the outside flow with that of the drain from chamber C. Consequently, a test was made with a feeler gage 0.015 inch thick to determine clearances. It was found that in the inside clearance the feeler could be inserted only at a few points, but in the outside clearance it was inserted freely practically all around. A maximum clearance of 0.0025 inch was allowable. Furthermore, pressure in chamber A would reduce to zero when a good seal was effected, but leakage would continue. It was therefore concluded that the leakage all came from the clearance around the seal ring. The leakage was also found to be about the same amount when no springs were used.

Sealing conditions at the seat were good when good contact was made, but when dirt got on the seating surfaces no amount of pressure in chamber B would cause a seal. Since no retraction was possible when the springs were used, further tests were conducted without them so that limited retraction could be obtained.

FIGURE 7

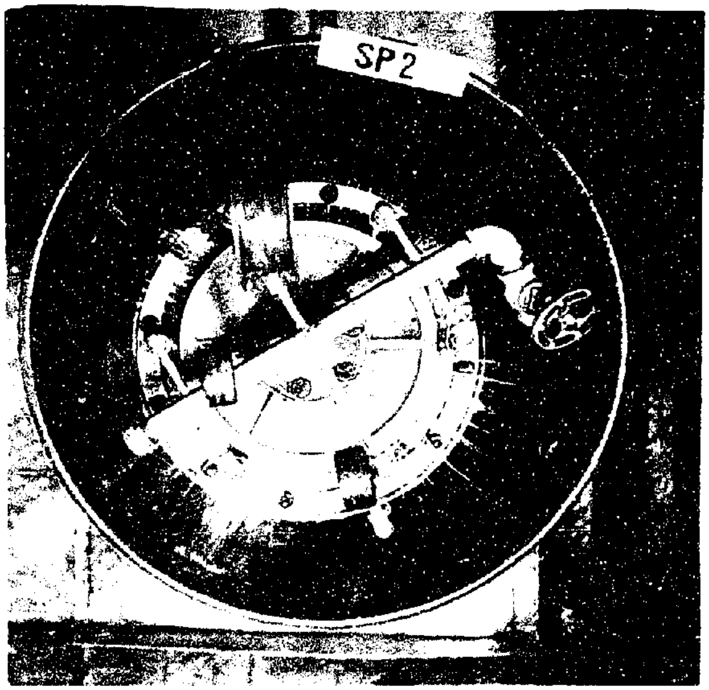


A-75 pounds per sq.in. under seal ring in chamber B only.

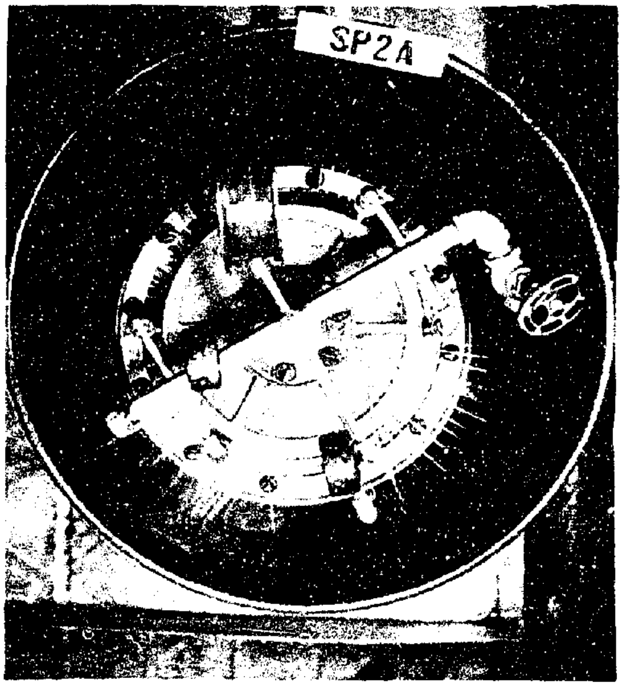


B-75 pounds per sq.in. in chamber B and in chamber C.

FACE A (LEGS) OF SEAL RING TO THE SEAT



C-75 pounds per sq.in. under seal ring in chamber B only.



D-75 pounds per sq.in. under ring in chamber B and in chamber C.

FACE B (FLAT) SEAL RING TO THE SEAT

LEAKAGE TESTS---ORIGINAL SEAL RING DESIGN

The seal ring was installed without the springs and tested with face A (legs) to the seat and with face B (flat) to the seat. The clearance was increased to 1/8 inch (figure 35).

With face A to the seat and the relief from chamber A open, for 75 pounds per square inch in chamber C the sealing and breaking pressures in chamber B were 15 and 7 pounds per square inch, respectively. Sealing occurred suddenly and with a slam; release was also sudden and with a jerk. When sealed pressure in chamber A was zero and when unsealed, the amount of retraction was small.

When the drain from chamber A was closed and the penstock pressure maintained at 75 pounds per square inch in chamber C, the seal ring rose slightly at 3 pounds per square inch in chamber B. Further pressure application in chamber B caused the seal ring to rise to within 1/64 inch of the closed position or to a gap of 1/64 inch. The negative pressure developed through the gap assisted in raising the seal ring. This same condition was very apparent in the visual, sectional, seal-model tests. As pressure in chamber B was further increased, pressure in chamber A advanced ahead of pressure in chamber B by 2 pounds per square inch, until a seal was suddenly effected at 30 pounds per square inch in chamber B.

As soon as the seal ring slammed closed, pressure in chamber A slowly reduced to practically zero. This now proved that the inner leg was almost perfectly sealed.

With face B (flat) to the seat and relief from chamber A open, for 75 pounds per square inch in chamber C, sealing pressure was 28 pounds per square inch in chamber B and breaking pressure was 24 pounds per square inch. Pressure in chamber A reduced to zero. The seal ring closed without a slam if pressure in chamber B was raised slowly, but slammed closed if it was increased rapidly. Slamming was less pronounced than when face A was placed to the seat. The retraction of the ring demonstrated that sealing was not accomplished at the upstream edge of the seal-ring nose.

When the drain from chamber A was closed, the action was similar to that when face A was to the seat. With penstock pressure maintained at 75 pounds per square inch in chamber C, sealing pressure was 30 pounds per square inch in chamber B. The pressure in chamber A led the pressure applied in chamber B by 2 pounds per square inch, until a seal was suddenly effected.

The conclusions from this last test are that the pressure necessary to cause sealing is the same for either face A or face B to the seat. Furthermore, the quality of the seal is not necessarily a function of the area in contact nor of the unit pressures of these areas, but rather the pressure necessary to make line contact, because the quality of the seal was good for either the narrow or the wide seating surfaces. The increased contact area may, however, assist in making a better seal because of the greater resistance to flow over the longer path so that leakage would be reduced.

The reason for the variation between sealing and breaking pressure from 15 to 28 pounds per square inch and from 7 to 24 pounds per square inch, respectively, is probably due to the variation in action of the water over the two faces, quicker relief being afforded from chamber A when face A is toward the seat than when face B is toward the seat, so that the unseating force is greater for the wider contacting surface.

The seal ring was changed by adding the distance piece and slotting the seal ring (figure 6B). It was decided that the ring must be made to act like a piston ring, because leakage through either the outside or the inside clearances was too great to be allowed (figure 7). The design is shown in figuros 3E, F, and G as seal-ring alteration.

With drain from chamber A open and penstock pressure of 110 pounds per square inch in chamber C, the sealing pressure was 47 pounds per square inch in chamber B and breaking pressure was 25 pounds per square inch. For a pressure of 75 pounds in chamber C the sealing pressure was 35 pounds in chamber B and breaking pressure was 23 pounds. Full release took place after pressure in chamber B was further reduced by 10 pounds.

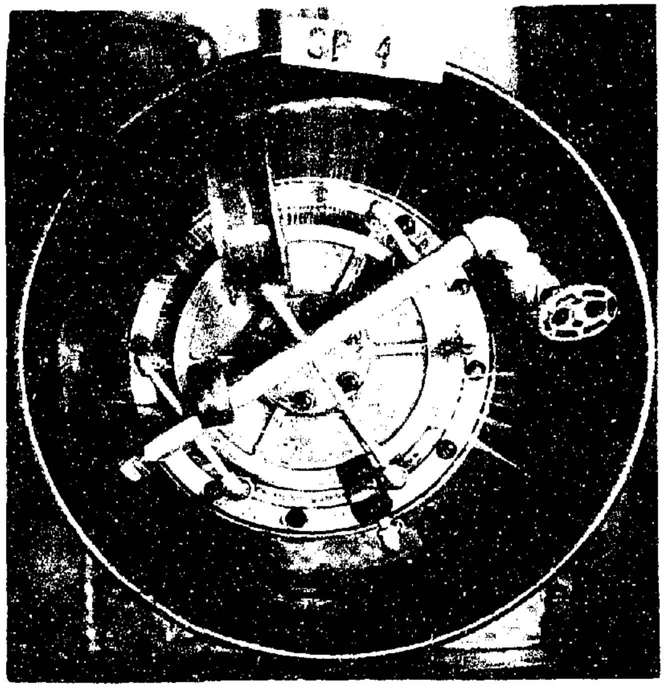
With drain from chamber A closed and penstock pressure of 110 pounds per square inch in chamber C, the sealing pressure was 50 pounds per square inch in chamber B and breaking pressure was 35 pounds per square inch. For a pressure of 75 pounds in chamber C the sealing pressure was 38 pounds in chamber B and breaking pressure was 27 pounds.

In both cases pressure in chamber A did not always become zero. In some cases additional pressure in chamber B caused a better seal and pressure in chamber A eventually dropped to zero. This condition indicated the presence of foreign matter on the seating or sliding surfaces which could be compressed or indented into the brass of the seal ring. Several times, on dismantling, the seal ring was found pitted and the edges battered.

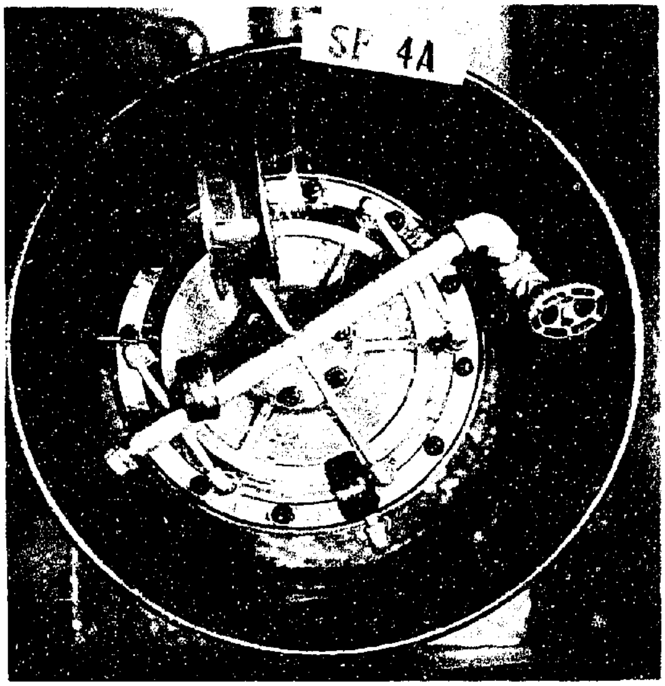
In considering the leakage through the outer clearance and through the distance piece, the action of the ring must be taken into account. When pressure in chamber B only was applied, the leakage was greater with the solid ring (figure 7A compared with figure 8A); but when penstock pressure in chamber C was also applied, the leakage was reduced by about one-half, showing that the ring had slipped outwardly (one could also hear the slippage taking place). Finally, leakage was practically stopped when the cap plate was hit with a hammer handle, causing more outward slippage, each rap decreasing the leakage. This proved that the ring would make a good seal along the outer clearance space when designed after this fashion. The improved sealing conditions are shown in figures 8B and C. In B, pressure was applied in chambers B and C and some leakage is seen, but in C, leakage has virtually stopped after rapping the cap plate with the hammer handle. It was assumed that the seal ring would slip inwardly in the same manner if the penstock pressure were reversed.

There was some water getting by the distance piece but when it was greased with Ezit before installation, almost a droptight seal was effected. Splitting the seal ring did not make its installation or removal any easier.

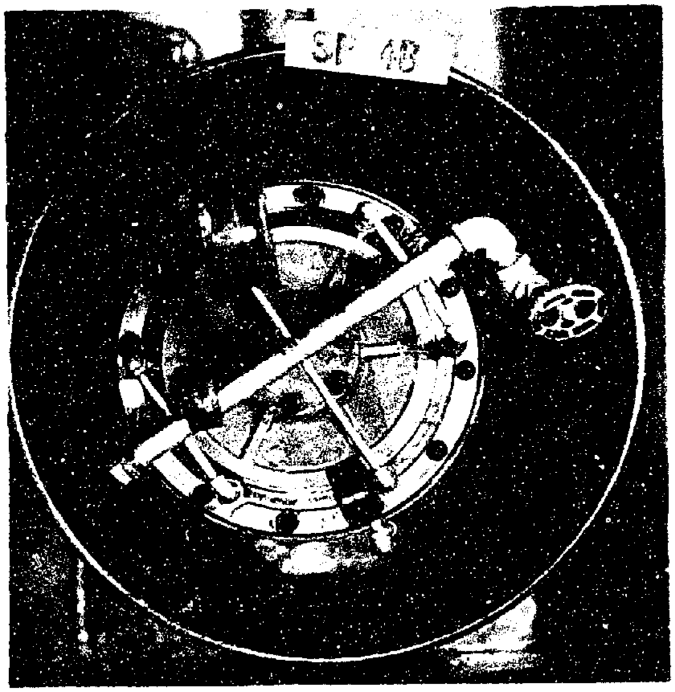
7. Unit contact sealing-pressure analysis. At least one set of computations may be made showing that the unit pressure required to maintain



A-75 pounds per sq.in. under ring in chamber B only.



B-75 pounds per sq.in. under ring in chamber B and in chamber C



C-75 pounds per sq.in. in chambers B and C. [illegible] plate was tapped with hammer handle to cause the ring to slip [illegible] ing leakage through outer clearance [illegible].

LEAKAGE TESTS---SLOTTED SEAL RING---FACE B (FLAT) TO SEAT a seal after seating has taken place is small, if not negligible.

If the pressure at which the seal ring breaks contact of 7 pounds per square inch were taken as the sealing pressure, and the unit area as 3/4 inch wide by 1 inch long along the center line of the seal ring, then the force upward is $3/4 \times 7$, or 5-1/4 pounds minus 0.3 pound weight of 1-inch length of ring, or about 5 pounds. It is assumed that two legs are taking this pressure equally because each sealed equally well; so the upward force may be divided proportionately. The outer leg has a mean circumference of 32.3 inches and the inner leg has a mean circumference of 30.5 inches, while the center-line circumference is 31.4 inches. The force on the outer leg is taken by the seat and need not be considered further. The force on the inner leg is $\frac{30.5}{31.4} \times 1/2 \times 5 = 2.4$ pounds upward. The downward pressure due to 75-pounds-per-square-inch penstock pressure on the approximately 1/32-inch radius on the inner seal-ring edge is about $1/32 \times \frac{30.5}{31.4} \times 75 = 2.3$ pounds. Since the two forces practically balance, it is concluded that once line contact is made, the sealing pressure required is almost, if not entirely, nil. If the outer leg is only touching but not pressing on the seat, then the sealing pressure in chamber B is too large and indicates that the inner leg does not seal at the upstream edge but that pressure areas exist between the seating surfaces.

The reason that computations generally cannot be made to check is that sealing does not take place at the extreme upstream edge of the seal bar. If water pressure creeps in the crevices or spaces it will create a force tending to unseat the seal bar; yet a good seal is maintained because line contact is established in the form of a contour along the high spots.

In the tests in the visual, sectional seal model the pressures applied by the lever when the seating surfaces leaked could not be made to check in computations; therefore, these are not included. After the seating surfaces were ground true, lever measurements were superfluous and not taken because the upstream edge sealed perfectly so that there was no unsealing pressure to counteract.

8. Conclusions and summary of tests. The unit pressure required to effect a seal, once line contact is made, is very small. Line contact only is necessary to form a perfect seal. A wider area may be beneficial from the standpoint of leakage if contact is not perfect, because a longer path for water flow with consequent pressure drop is thus provided.

The narrower the seating surface on the seal bar or ring, the lower will be the pressure required to bring the seal bar in contact with the seat. A rounded nose piece would therefore appear to be the most advantageous design.

The flat seating surfaces must be very clean to make a perfect seal. Accumulation of dirt on the upstream edge of a flat nose does not wash through when retracting the seal ring; therefore, a flat sealing surface is detrimental from this standpoint.

To produce good retraction of the seal bar or ring with hydraulic forces, the upstream edge of the seal bar or ring should be chamfered. The size of the hamfer as demonstrated by tests on retractable seals in gate seal designs and tests should be at least half the width of the seal bar or ring; the unbalanced force under the remaining unchamfered part produces extra sealing pressure to force the seal bar in better or continuous contact.

The split ring with it's distance piece insert produced good sealing qualities against leakage through the downstream clearance space. The design was accepted for field installation both for the inner and the outer rings.

Flat seats on the seal bars were accepted for field installation because they produced at least as good quality of seal from either direction as a seat with two narrow legs, and in some cases a seal of better quality.

Sealing should take place on the upstream edge of the seal-bar nose but may take place toward the downstream edge. In the latter case the springs provided in the design are necessary to maintain a seal.

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E. S. Gray