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THERMAL CYCLING AND LEAKAGE TESTS
OF 12-INCH VALVES FOR SODIUM SERVICE

AEC Research and Development Report

ATOMICS INTERNATIONAL

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THERMAL CYCLING AND LEAKAGE TESTS
OF 12-INCH VALVES FOR SODIUM SERVICE

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ATOMICS INTERNATIONAL

A DIVISION OF NORTH AMERICAN AVIATION, INC.
P.O. BOX 309 CANOGA PARK, CALIFORNIA

CONTRACT: AT(11-1)-GEN-8
ISSUED:

MAY 1 1960

DISTRIBUTION

This report has been distributed according to the category "Reactors - General" as given in "Standard Distribution Lists for Unclassified Scientific and Technical Reports" TID-4500 (15th Ed.), August 1, 1959). A total of 625 copies was printed.

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ABSTRACT

Tests were performed to determine the effect of thermal cycling on the across-the-seat leakage characteristics of commercially available valves considered for use in the sodium coolant system of the Hallam Nuclear Power Facility. Four 12-in. valves were cycled between 680 and 1100°F in a sodium loop and periodically tested for across-the-seat sodium leakage. The valves tested were two throttling (ball) valves and two blocking (gate) valves. The first-ball valve had a consistent leakage pattern which was primarily a function of valve temperature, but also varied with pressure differential and number of thermal cycles. The second ball valve showed high initial leakage rates which became excessive after some sodium circulation. Using data obtained with the first-ball valve, a general equation for the rate of leakage across the valve seat was obtained for this type and size of ball-valve, as a function of temperature, pressure differential, and number of thermal cycles. No sodium leakage was exhibited by either gate valve throughout the test. None of the valves exhibited any leakage through the freeze-seal along the stem.

I. INTRODUCTION

Sodium-cooled nuclear power plants require leak-tight valves which are capable of withstanding both high temperatures and severe thermal transients. For sodium service in the primary cooling system of the Hallam Nuclear Power Facility,* both throttling and blocking valves are required. These valves must remain leak-tight with sodium at 1000°F and under a pressure differential of 50 psi, both before and after thermal cycling.

Because only limited data are available on the sodium leakage characteristics of valves, tests were required to determine the suitability of four 12-in. commercially available valves for the intended service effect of thermal cycling upon the sodium leakage. The test program to be performed included: (a) thermally cycling the valves between 680 and 1100°F in a loop containing sodium for a minimum of 150 cycles, and (b) testing for leakage of sodium across the valve seat before thermal cycling and after approximately every 50 cycles. The valves to be tested included two ball valves for throttling flow and two gate valves for blocking flow. Because these valves utilized freeze seals instead of bellows on the valve stem, observations were also to be made to detect any sodium leakage along the stem.

Although the primary purpose of the test program was to determine the shut-off performance of 12-in. valves under the conditions enumerated above, a consistent overall leakage pattern was obtained only with the first throttling (ball) valve. Accordingly, this information was utilized to obtain a general correlation for leakage characteristics for this type and size of ball valve.

Similar tests are being performed on 2-in. valves for use in the auxiliary piping system of the Hallam Nuclear Power Facility. These tests are now in progress and will be reported in a subsequent report.

*"USAEC Power Reactors," Technical Information Service Pamphlet (May 1958)

Figure 1. 12-In.-Valve Test Loop

II. DESCRIPTION OF EQUIPMENT AND TEST PROCEDURE

A. TEST EQUIPMENT

The piping and equipment arrangement of Test Loop 1 which was used for the first P-K ball valve and the Crane gate valve is shown in Figure 1. The 3-in. pipe loop was rectangular in shape, approximately 10 by 10 ft in size, with a test valve, pump, heating section, and cooling section, each occupying one side. The latter side also had direct connections to a 40-gal dump tank, positioned inside the loop. All piping, tanks, and components were of Type 304 stainless steel, and all welds were made to sodium service specifications. A linear induction electromagnetic pump was installed, as shown, to circulate about 40 gpm of sodium during the cycling tests. Helium was used to provide an inert atmosphere over the sodium and for pressurizing during leak tests.

Electric resistance heaters were fastened to the outside of the piping and used for pre-heating. For thermal cycling tests, a heating transformer^{*} which used the piping as a portion of the electrical path, and immersion-type electric resistance heaters were utilized. For thermal insulation, 3-in. thick Kaylo was used. A finned-tube heat exchanger and an air blower effected the cooling portion of the cycle.

Test Loop 2, which was used for testing the second P-K ball valve and the Cooper-Alloy gate valve, was generally similar to that described above. The exception was the use of electrical immersion-type heater only, instead of a combination of electrical immersion-type heaters and a heating transformer for thermal cycling. Nitrogen, instead of helium, was used to provide an inert atmosphere over the sodium.

B. THERMAL CYCLE

The desired cycle for valve testing simulates the most severe thermal transient conditions anticipated in HNPF and was to consist of the following heating pattern: 680 to 960°F at 3°F/min, 960 to 1090°F at 6°F/min, and 1090 to 1200°F at 12°F/min. For cooling the same pattern in reverse was to be followed. During cycling the valve was to be in the fully open position and during pressure testing, fully closed. A minimum of 150 such cycles, with an across-the-seat

^{*}R. S. Baker and H. W. Lohman, "Electrical Heating Methods for Liquid Metal Systems," NAA-SR-3882 (September 15, 1959)

Figure 2. Thermal Cycle Pattern

leak check performed after approximately every 50 cycles, was desired. Additional cycles, as permitted by the testing schedule, were to be accumulated.

Subsequent re-examination of the thermal cycle indicated that the 1200°F peak temperature was too high and it was agreed that a peak temperature of at least 1050°F would be satisfactory. Since the control system was designed for the thermal cycle described previously, the actual thermal cycles differed to the extent shown in Figure 2.

To obtain a thermal cycle without any dead time on the automatic controller, the 3°F/min portion of the cycle was changed to 3.14°F/min. This resulted in a 2-hr heating period and a 2-hr cooling period. The peak temperature reached during the actual cycle was 1060°F for the P-K ball valve and 1100°F for the Crane gate valve. The second valve-test loop was used to test the second P-K ball valve and the Cooper-Alloy valve. The peak temperature reached during the cycle was 1140°F.

C. CONTROL SYSTEM

In order to vary the temperature of the sodium in a prescribed manner, with identical cycles, an automatic control system was used. The control system consisted of a cam program controller, a pneumatically controlled autotransformer with limit switches, and a step switch. The program controller contained a plastic cam which was shaped so as to give the desired time-temperature pattern. The program controller supplied an air-pressure signal to an autotransformer during heating or to the blower damper during cooling. Magnitude of the control signal was determined by the difference between the system temperature (as indicated by an immersion thermocouple in the sodium entering the test valve) and the desired program temperature (as indicated by the cam follower).

During heating, a pointer on the shaft of the autotransformer controlling one group of immersion heaters engaged a limit switch. This occurred when the autotransformer was in the full-power position and the system temperature was lagging the program temperature. The limit switch actuated the step switch which in turn successively engaged a second group of electric immersion heaters, and if required, the heating transformer. Introduction of the additional heat

eventually caused the system temperature to exceed the program temperature. To correct this condition, the autotransformer decreased the power output causing the pointer to move away from the limit switch. If the difference between the system temperature and program temperature still was not corrected when the autotransformer reached the zero-power position, the pointer engaged another limit switch which disconnected all heaters.

At the maximum cycle temperature, the blower was turned on and the control air was directed from the autotransformer to the damper. All the transformer-type and immersion-type heaters were then shut off. According to the desired cooling pattern, the damper altered the amount of heat-exchanger air flow and thereby varied the cooling rate of the system. At the minimum cycle temperature, the blower was turned off and the control air was directed to the autotransformer in order to start the heating portion of the cycle. To obtain sufficient heating capacity, several line heaters were on throughout the cycle. The entire cycling operation was completely automatic, requiring no personal attention, and capable of completing six cycles every 24 hr.

The control system described above was used in Test Loop 1. For Test Loop 2 the step switch was not required as adequate control of the heating portion of the cycle was obtained by a combination of two groups of immersion heaters. One group was introduced at the start of heating, along with a second group of immersion heaters, which were varied by a pneumatically controlled autotransformer. Since sufficient heating capacity was available with the immersion heaters alone, the need for using line heaters throughout the cycle was eliminated. Without heat addition during the cooling cycle, the heat loss of the test loop was large enough to produce the proper cooling rate without using the blower.

D. TEST VALVES

All four of the test valves were for service in 12-in. pipe lines. The ball valves are intended to serve both throttling and shut-off functions. The gate valves are intended for shut-off only. All valves had identical water-cooled frozen-sodium stem seals backed up by high temperature packing.

1. P-K Ball Valves

A drawing of the P-K ball valve body is shown in Figure 3. The valve body was cast from Type 304 stainless steel. A 10-in. diameter hollow stainless-steel sphere, Stellite faced, is contained within a carrier which moves vertically. When the carrier descends to a certain position, the lower section of the ball comes into contact with an inclined rod which is offset from the center line of the ball. This causes the ball to be rocked toward the Stellite-faced conical valve seat and, as the carrier continues to descend, the ball is forced into the valve seat effecting a shut-off.

A photo of the ball and carrier after completion of 252 thermal cycles is shown in Figure 4. The ball and carrier were quite clean with a slight sodium oxide deposit evident. The pattern of the seating-ring marks on the ball show that the ball had varied in its closing position over a circumferential distance of only about one inch.

The second P-K ball valve was similar to the first, except that the ball was 12-in. instead of 10-in. in diameter. The second valve had both a pneumatic and manual operator instead of manual only. The ball was increased in diameter in the second valve because the larger seat diameter resulted in a decreased pressure loss through the valve in the wide open position.

2. Crane Gate Valve (Split-Wedge Disc)

The Crane-gate valve is shown in Figure 5. The valve body is cast from Type 304 stainless steel. The valve is of split-wedge-disc design, has sealing and rubbing surfaces of Stellite and a manual operator.

3. Cooper-Alloy Gate Valve (Solid-Wedge Disc)

The Cooper-Alloy gate valve is shown in Figure 6. The valve body was cast from Type 304 stainless steel. The valve is of solid wedge design, has sealing surfaces of Stellite, and both a pneumatic and manual operator.

Figure 3. Drawing of P-K Ball Valve

Figure 4. Photograph of P-K Ball Valve (Ball and Carrier)

(AI)

Figure 5. Drawing of Crane Gate Valve

Figure 6. Drawing of Cooper-Alloy Gate Valve

4. Leak Test Procedure

An across-the-seat leak check was performed in the following manner (see Figure 1): The loop was emptied and the test valve was closed. The shut-off valve in the test loop was closed. To ensure sealing, a water-cooled freeze-coil adjacent to the shut-off valve was turned on. This minimized the possibility of sodium leaking past the shut-off valve and flowing in a counter clockwise direction to register as test valve leakage in the leak accumulator. The loop was then filled with sodium to a level which was well above the valve center line, as indicated by the valve fill-tank level probe. Valve temperature was then raised to the desired level and the test valve was again checked for full closure. On occasion, the valve would be fully closed at low temperature and slightly open at a higher temperature. This permitted several degrees of additional movement of the hand wheel. The valve fill tank was then pressurized. Any leakage of sodium past the test valve flowed into the leak accumulator and the subsequent level change during the elapsed time was recorded. Care was taken to ensure that the leak accumulator, and the line leading to it, were well above the melting point of sodium during leak tests. In addition, the sodium level in the valve fill tank was noted before and after the leak test. For an indicated "no leakage" test, a constant level in the fill tank constituted an additional check of the observation that no sodium had leaked either by the test valve or into the dump tank.

At a given temperature, the above procedure was repeated for increasing pressure differentials. The duration of a leak test varied from 2 min to 1 hr, depending on leak rate, with an average time of about 8 min. The number of tests made at one time were limited by the leak-accumulator volume. The sodium in the leak-accumulator was then transferred to the dump tank.

III. TEST RESULTS

A. FIRST P-K BALL VALVE

A total of 252 four-hr thermal cycles was completed on the first P-K ball valve and leak tests were performed at 0, 73, 236, and 252 cycles. Thirty such sodium leak tests were made with pressure differentials, across the valve, ranging from 5 to 50 psi and with temperatures ranging from 520 to 950°F.

In the brand-new condition, prior to circulating sodium through the valve, the valve was leak tight up to a 50 psi differential at 900°F. At 73 cycles what appeared to be a high leak rate was recorded for pressure differentials of 5 and 7 psi at 950°F. This prompted belief that the valve had been severely affected by thermal cycling and there would be no purpose in making further leak tests until the end of the program. At 236 cycles a substantial number of tests at various pressure differentials and temperature levels made a consistent valve-leakage pattern apparent. At 252 cycles an additional test gave further confirmation of this. All test data for the first P-K ball valve are tabulated in Table I.

B. SECOND P-K BALL VALVE

In the brand-new condition, before any sodium had circulated through the second P-K ball valve, the data shown in Table II were obtained. The leakage pattern confirms the existence of a zero-leakage region and has the general shape of the correlation lines, as shown in Figure 7. Except for the number of cycles variation, good agreement is indicated for the leak-rate behavior between the first P-K ball valve at 236 cycles and the second P-K ball valve at zero cycles. The leak rate observed in the second P-K ball valve at zero cycles is attributed to the markedly different physical condition of this valve.

After circulating 600°F sodium through the valve for 3 days another leak check was attempted. With the valve fully closed the leakage was so great that it was not possible to fill the pipe leading to the valve, because the sodium immediately leaked past the valve and into the leak accumulator. Several attempts were made to leak test the valve, but in each case the leakage was prohibitive and no test could be made. It appears that severe seat-distortion occurred during sodium circulation.

TABLE I
TEST DATA FOR FIRST 12-IN. P-K BALL VALVE
(10-in. diameter ball)

Number of Cycles	Pressure Differential (psi)	Temperature (°F)	Sodium Leak Rate (gph)
0	10	800	0
	20	800	0
	30	800	0
	40	800	0
	50	800	0
	10	900	0
	20	900	0
	30	900	0
	40	900	0
	50	900	0
73	5	950	29
	7	950	36
236	10	520	0
	30	520	0
	40	520	0
	50	520	0
	50	520	0
	10	600	0
	20	600	0
	30	600	0
	40	600	2.4
	10	700	8.5
	40	700	5.7
	50	700	2.3
	20	750	58
	30	750	19
	10	825	47
	20	825	51
	30	825	87
252	10	825	44

Disassembly of the second valve, after six thermal cycles, showed a fairly heavy deposit (about 0.050-in. thick) on the ball which consisted of 3.5% carbon, 4% calcium, 41% sodium, and 15% various metals. The most likely source of carbon was the organic solvent which was used to degrease the loop before filling with sodium. While the deposit on the ball may have

Figure 7. Sodium Leak Rate vs Valve Temperature for First P-K Ball Valve

(A)

TABLE II
TEST DATA FOR SECOND 12-IN. P-K BALL VALVE
(12-in. diameter ball)

Number of Cycles	Pressure Differential (psi)	Temperature (°F)	Sodium Leak Rate (gph)
0 (Before sodium circulation)	10	350	0
	30	350	0
	50	350	0
	10	485	0
	30	485	0
	50	525	0.8
	10	875	101
0 (After sodium circulation)	Prohibitively high valve leakage during loop filling procedure.		

contributed to the high leakage rate, a pressure test made with gas, after the ball and seat were thoroughly cleaned, still indicated a leakage rate equivalent to 70 gph of sodium. Shop inspection showed that a 3-in. portion of the valve seat (about 12% of the valve seat circumference) in the lower right quadrant, as viewed from upstream, was not making contact with the ball.

C. CRANE GATE VALVE

A total of 207 thermal cycles was completed with valve leak tests performed at zero cycles (before and after sodium circulation), 89, and 207 cycles. Twenty-three sodium leak tests were made with pressure differentials across the valve ranging from 10 to 50 psi and temperatures ranging between 550 and 900°F.

Table III is a tabulation of the leak test data. It can be seen that only in one of the twenty-three leak tests was there any leakage noted. The leakage rate was slight (1.7 gph) and, as the test was performed immediately after sodium circulation, it could have been the result of insufficient tightening of the valve. All other tests, at various temperatures and pressure differentials, showed the valve to be leak tight. This is especially impressive because a failure in the building air supply (at 140 thermal cycles) resulted in a loss of control air and

TABLE III
TEST DATA FOR 12-IN. CRANE GATE VALVE
(split-wedge disc design)

Number of Cycles	Pressure Differential (psi)	Temperature (°F)	Sodium Leak Rate (gph)
0 (Before sodium circulation)	10	615	0
	50	615	0
	15	620	0
	30	625	0
	10	895	0
	50	895	0
0 (After sodium circulation)	10	720	0
	30	720	1.7
89	10	805	0
	30	805	0
	50	805	0
	10	670	0
	40	660	0
	10	550	0
	40	550	0
207	10	860	0
	30	840	0
	50	840	0
	10	680	0
	30	675	0
	50	660	0
	10	900	0
	50	900	0

caused four thermal cycles between 700°F and a peak of 1280 to 1300°F. The valve appears to be leak tight under all test conditions and as a result, no correlation was found between leak rate, pressure differential, temperature, and number of cycles.

The most likely reasons for the excellent shut-off performance of this valve are: (1) the split wedge disc offers two high resistance leakage paths in series instead of the usual single path. Leakage past the first disc reduces the pressure

head, so that the remaining pressure differential is insufficient to force leakage past the second disc. (2) The valve body is compact and of relatively light construction which enables it to withstand thermal transients with but slight resultant stress and distortion.

Some additional evidence of the tight shut-off characteristics of this type of valve is to be found in the 1-in. gate valve which is located between the dump tank and the test loop. This gate valve is subjected to the same pressure differentials (and approximately the same temperatures) as the test valve during leak tests. Although this valve is several years old and has been subjected to hundreds of heating and cooling cycles during loop filling and dumping, it is for all practical purposes, leak tight.

D. COOPER-ALLOY GATE VALVE

A total of 168 thermal cycles was completed and leak tests were performed at zero cycles (before and after sodium circulation), 76 cycles, and 168 cycles. Thirty sodium leak tests were made with pressure differentials across the valve ranging from 10 to 50 psi, and temperatures ranging between 505 and 975°F.

Table IV is a tabulation of all leak test data. It can be seen that no sodium leakage took place under any of the test conditions; as a result, no correlation could be made for the variation of sodium leakage with temperature, pressure differential, and number of thermal cycles.

As in the case of the Crane gate valve, the excellent shut-off characteristics of the Cooper-Alloy gate valve appear to be the result of the solid-wedge design which offers two high resistance sodium leakage paths in series (instead of one) and the compact, relatively light construction of the valve body, which minimizes the stress and distortion resulting from thermal transients.

E. WATER-COOLED FROZEN-SODIUM STEM SEAL

All four of the test valves had identical water-cooled frozen-sodium stem seals which were backed up by high temperature packing. In addition to the heat conducted to the stem from the valve, the heat lost from the valve body enclosure raised the temperature of the air passing over the stem to

TABLE IV
TEST DATA FOR 12-IN. COOPER-ALLOY GATE VALVE
(solid-wedge disc)

Number of Cycles	Pressure Differential (psi)	Temperature (°F)	Sodium Leak Rate (gph)
0 (Before sodium circulation)	10	730	0
	30	730	0
	50	730	0
	10	730	0
	30	730	0
	50	730	0
	10	950	0
	30	960	0
	50	960	0
0 (After sodium circulation)	10	825	0
	30	825	0
	50	825	0
76	10	505	0
	30	505	0
	50	505	0
	10	700	0
	30	710	0
	50	700	0
	10	937	0
	30	935	0
	50	935	0
168	10	530	0
	30	530	0
	50	530	0
	10	690	0
	30	698	0
	50	710	0
	10	970	0
	30	975	0
	50	975	0

well above the ambient level. A cooling water flow of about 0.3 gpm maintained the temperature near the bottom of the seal (closest to the valve body) at 130°F even at the highest temperatures encountered during cycling.

The stem seals did not leak sodium at any time during thermal cycling, and during across-the-seat leak tests, the seals were subjected to a pressure differential of 50 psi and a valve temperature of 900°F for periods up to one hour. Because of the construction of the P-K ball valves the stem seals were known to have been subjected to the full leak test pressure differential. (See Figure 3.) However, with the gate valves the stem seal pressure differential depended on the amount of leakage getting past the first seat sealing surface. (See Figures 4 and 5.) As the pressure existing between the first and second sealing surface could not be measured, the actual stem seal pressure differential could not be measured either.

IV. DATA CORRELATION

A. CORRELATION OF TEST DATA FOR THE FIRST P-K BALL VALVE

Although the prime purpose of the valve test program was to determine valve shut-off performance at 1000°F and 50 psi differential (after thermal cycling) a variety of tests on the first P-K ball valve indicated a consistent overall leakage pattern. While the number of tests was not large, it was possible to derive a general correlation for this type and size of ball valve which appeared logical and fitted the test points well. In areas where data were limited, reasonable assumptions were made in order to establish the general trend.

Because of the almost perfect shut-off characteristics of the Crane and Cooper-Alloy gate valves under all test conditions, no leak rate correlation could be obtained for these valves. The imperfect physical condition of the second P-K ball valve resulted in very limited data which, before sodium circulation, generally resembled the data obtained on the first P-K ball valve. Only in the case of the first P-K ball valve was it possible to develop an overall leak rate correlation and a leakage pattern explanation. The method used is described below.

A plot of valve leakage rate vs valve temperature for all test points obtained for the first P-K ball valve is shown in Figure 7. For tests made after thermal cycling it can be seen that no leakage occurs up to temperatures of 500 to 600°F. This was considered a region of zero leakage. Above these temperatures the leakage rate increases at an extremely high rate except for those tests taken at zero cycles. Straight lines of constant pressure differential appeared to fit the test points best and the lines shown represent a good balance of positive and negative errors. In order to use a log-log plot for the test data and still utilize the zero leakage information, the data were plotted as actual leakage rate, in gph + 1. The average slope of the lines is 8.5.

From Figure 7 the leakage rate (at constant ΔP) can be expressed as:

$$R = \left(\frac{T}{T_o} \right)^{8.5} - 1 \quad \dots (1)$$

Figure 8. F vs Pressure Differential for First P-K Ball Valve

where R = the valve leakage rate in gph and T = the valve temperature in °F. T_0 = the maximum temperature at which no leakage occurs, after cycling, and is obtained by the intersection of each ΔP line with the temperature axis, in the zero-leakage region. Since each ΔP line has a corresponding T_0 , the leakage rate can be expressed in a more meaningful manner as:

$$R = \left(\frac{T}{(\Delta P + 1)F} \right)^{8.5} - 1 \quad \dots (2)$$

F is equal to $T_0 / (\Delta P + 1)$ where ΔP is the pressure differential across the valve, in psi. In order for Equation 2 to give zero leakage at $\Delta P = 0$, $(\Delta P + 1)$ was used instead of ΔP . Figure 8 shows the variation of F with ΔP . The derivation of the curves for pressure differentials below 5 psi is explained below.

Using Equation 2 a satisfactory correlation was obtained for test data taken at 236 and 252 cycles. However, the expression did not fit with as good accuracy as data taken at 0 and 73 cycles (Figure 7). This indicated that the number of cycles had some effect on the leakage rate. This variation was determined by comparing the 236 cycle data with that taken at 0, 73, and 252 cycles. The inclusion of a cycling factor resulted in the final leakage expression as follows:

$$R = \left(\frac{N}{236} \right)^{0.35} \left[\left(\frac{T}{(\Delta P + 1)F} \right)^{8.5} - 1 \right] \quad \dots (3)$$

where N is the number of thermal cycles. Leakage curves below $\Delta P = 5$ psi were obtained by cross-plotting the data of Figure 7 at three leakage rates and extrapolating to $\Delta P = 1$ psi.

Using the test data and the extrapolated data, Figure 8 then was extended to $\Delta P = 1$ psi. Below this level F becomes a function of both pressure differential and valve temperature. For R to equal zero at zero pressure differential, F becomes equal to the valve temperature for the range 750 to 950 °F. While primarily of academic interest this permits Equation 3 to be valid under all conditions up to $\Delta P = 50$ psi and $T = 950$ °F.

Figure 9. Sodium Leak Rate vs Pressure Differential for First P-K Ball Valve

Figure 9 is a plot of valve leakage rate vs pressure differential with valve temperature as a parameter. The curves were obtained from a cross-plot of Figure 7 and includes the extrapolated data. Figure 9 shows that with increasing pressure differential, the leakage rate increases, reaches a maximum, and then decreases. Below 750°F leakage does not occur until some finite pressure differential is reached. Data obtained in another investigation now in progress at AI using piston rings within a cylinder, show the leakage pattern to be identical to that described above. This would indicate that the leakage pattern phenomenon has broader implications and applications. Figure 9 can be read directly for 236 cycles, but for any other number of cycles the cycle correction factor in Figure 10 must be applied to the indicated leakage rate.

Figure 10. Correction Factor vs Number of Cycles for First
P-K Ball Valve
(For use with Figure 9)

Figure 11 is an error plot using Equation 3, that shows the ratio of computed leak rate to experimental leak rate for all thirty test points. For four of the test points, taken in the zero-leakage region after thermal cycling a resultant computed leakage rate made the ratio infinite although the absolute

Figure 11. Error Plot for First P-K Ball Valve Data Correlation

error was moderate. Fairly good correlation is indicated, as 67% of all test points fall within $\pm 12\%$ of the computed leak rate.

B. EXPLANATION OF LEAKAGE PATTERN FOR THE FIRST P-K BALL VALVE

The leakage pattern shown in Figure 9 is the combined result of the physical design of the valve, pressure differential, and temperature. At constant temperature, increasing pressure differential exerts a simultaneous two-fold effect:

(1) it increases the leakage rate, and (2) it increases the seating pressure between the ball and seat, which tends to reduce the leakage area and the leakage rate. It may be seen from the shape of the curves that these effects are equal at about 20 psi. From 0 to 20 psi the fluid-driving-force effect predominates and the leakage rate increases with pressure differential. Above 20 psi the ball-seating effect predominates and the leakage rate decreases as the pressure differential increases.

The clearance between the ball and seat can be determined in the following manner:

$$\Delta P = 2f \frac{L}{D_h} \frac{G^2}{\rho g 144} \quad \dots (4)$$

where the friction factor,

$$f = K \left(\frac{G D_h}{\mu} \right)^x \quad \dots (5)$$

Terms are defined and units are presented under Nomenclature. In this analysis the leakage area is assumed to consist of an annular space between the ball and seat. The length of the flow path is assumed to be the depth of the seat.

Substituting the physical dimensions of the valve into Equations 4 and 5, combining, and then solving, the expression for radial seat clearance becomes:

$$c = \left[\left(\frac{LK}{g \rho (M \pi d)^{2+x}} \right) \left(\frac{2}{\mu} \right)^x \left(\frac{W^{2+x}}{144 \Delta P} \right) \right]^{1/3} \quad \dots (6)$$

The equation for valve seat clearance shows that the validity of the assumptions described above will not alter the general trend and will have but a moderate effect on the actual physical result.

The friction factor exponent x (obtained from a log-log plot of f vs Reynolds number), is determined by the character of the flow; i. e., laminar or turbulent. For Reynolds numbers less than 2000 (laminar flow), $x = -1$ and $K = 16$. For Reynolds numbers greater than 2000 (turbulent flow), $x = -0.16$ and $K = 0.04$. By use of Equation 3, W (as R) can be obtained at a given temperature and ΔP ; c can then be calculated. Because of the annular geometry, the magnitude of the Reynolds number ($N_R = 2W/\pi d\mu$) and consequently the friction factor exponent was known before the radial clearance was computed. This resulted in a direct solution for radial clearance instead of a trial and error procedure which would have been otherwise required.

Figure 12 shows the variation of radial seat clearance with pressure differential, at constant temperature, as calculated from Equation 6. The radial clearances obtained are on the order of one thousandth of an inch. It will be noted that at 750°F and above the leakage area does not go to zero, because the leakage area has to be finite, at zero pressure differential, in order for leakage to occur at pressure differentials slightly greater than zero.

As pressure differential increases to 15 - 20 psi the leakage area increases and thereafter it decreases. The increase in leakage area with increasing pressure differential, between zero and 20 psi, goes counter to expectations as either a constant area, or decreasing area would be logically anticipated.

The computed points up to 10 psi were in the laminar flow region, based on a full annular leakage area, with $M = 1$. However, if the leakage area were but a small fraction of the full annulus ($M \ll 1$) the flow would become turbulent. Recomputation of the points at 1 psi, assuming turbulent flow resulted in leakage clearances in the range of two to three thousandths of an inch for all the temperatures shown. If used, this would make the leakage clearance pattern, in Figure 12, a constantly decreasing one with increasing pressure differential and would be more logical. However, since there is no way to know the actual portion of the annulus involved in leakage this is questionable, but it does offer a possible explanation.

Some support for M being considerably less than 1 ($M = 0.12$) was found during examination of the second P-K ball valve. As indicated previously, shop inspection of the second P-K ball valve showed that about 12% of the valve seat

Figure 12. Radial Clearance vs Pressure Differential for First P-K Ball Valve

(A)

circumference was not making contact with the ball. Equation 6 showed the radial clearance for the leakage area portion of the seat to be 0.00125 in. for 10 psi pressure differential, at 875 °F, and $M = 0.12$. (Reference to Figure 12 shows that this clearance compares closely with the radial clearance obtained, under the same conditions, for the first P-K ball valve after 236 cycles, and $M = 1$.)

V. CONCLUSIONS

Prior to thermal cycling, the first P-K ball valve, the Crane gate valve, and the Cooper-Alloy gate valve did not leak sodium (across-the-seat) in tests involving pressure differentials up to 50 psi and temperatures ranging from 500 to 900°F. However, the second P-K ball valve leaked at the rate of ~100 gph under a pressure differential of 10 psi at 875°F.

After thermal cycling, the Crane gate valve and the Cooper-Alloy gate valve did not leak sodium (across-the-seat) in tests involving pressure differentials up to 50 psi at temperatures ranging from 500 to 975°F. After 236 thermal cycles, the first P-K ball valve leaked sodium at the rate of 33 gph under a pressure differential of 50 psi at 900°F. After circulation of 600°F sodium through the second P-K ball valve, the leakage rate was so high that sodium leaked past the fully-closed valve during filling. Later, inspection showed that a 3-in. portion of the valve seat (about 12% of the circumference) was not making contact with the ball.

The water-cooled frozen-sodium stem seal on all four valves did not leak sodium at temperatures up to 975°F and pressure differentials up to 50 psi.

The excellent shut-off characteristics of the gate valves appear to be the result of the wedge design (split-wedge in the Crane and solid wedge in the Cooper-Alloy) which provides two high resistance sodium leakage paths in series, instead of one as in the P-K ball valves. In addition the compact, relatively light construction of the gate-valves bodies (as compared to the P-K ball valves) tends to minimize distortion resulting from thermal transients.

Although the prime purpose of this valve test program was to determine valve shut-off performance at 1000°F and 50 psi differential (after thermal cycling) a variety of tests on the first P-K ball valve at other temperatures and pressure differentials indicated a consistent overall leakage pattern. Using these additional data, a general equation for rate of leakage, across the seat, of this type and size of ball valve was derived and may be expressed as:

$$R = \left(\frac{N}{236} \right)^{0.35} \left[\left(\frac{T}{(\Delta P + 1)F} \right)^{8.5} - 1 \right]$$

Good agreement between test results and the derived equation was obtained. The overall trends indicated by this equation are believed to be applicable to ball valves in which the ball and seat fit properly in the as-received condition, although additional data are required for definite confirmation.

NOMENCLATURE

T = valve temperature (°F)

T_o = maximum valve temperature at which no sodium leakage occurs, after thermal cycling ($^{\circ}\text{F}$)

R = valve sodium leakage rate (gph)

$$\Delta P = \text{pressure differential across valve (psi)}$$

N = number of thermal cycles

L = depth of valve seat (ft)

d = diameter of valve seat (ft)

c = radial clearance between ball and seat (ft)

W = valve sodium leakage rate (lb/hr)

$$G = \text{sodium mass velocity, based on flow through leakage area, lb/hr-ft}^2 = W/dc\pi$$
$$D_h = \text{hydraulic diameter of leakage annulus (ft)} = 2 \text{ c}$$
$$N_R = \text{Reynolds number} = GD_h / \mu = 2W / \pi d \mu$$
$$\mu = \text{absolute viscosity of sodium (lb/ft-hr)}$$
$$g = \text{acceleration of gravity, ft/hr}^2 = 4.18 \times 10^8$$
$$\rho = \text{density of sodium (lb/ft}^3\text{)}$$

f = friction factor

x = friction factor exponent, for $N_R < 2000$, $x = -1$

for $N_R > 2000$, $x = -0.16$

K = constant relating to friction factor, for $N_B < 2000$, $K = 16$

for $N_R > 2000$, $K = 0.04$

M = ratio of leakage circumference/valve seat circumference

$$F = T_o / (\Delta P + 1), \text{ } (^{\circ}\text{F}/\text{psi})$$