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SUBJECT:

Thermal Cycling and Leakage Tests of 12-Inch Sodium Valves

CONTENTS:

I STATEMENT OF PROBLEM

To assist in evaluating commercially available valves for HNPF, four 12-inch valves were thermally cycled between 680°F and 1100°F in a sodium loop and periodically tested for across-the-seat sodium leakage. The valves tested consisted of two P-K ball valves, a Crane gate valve, and a Cooper-Alloy gate valve.

II SUMMARY OF RESULTS AND RECOMMENDATIONS

1. 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.

2. 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-inch portion of the valve seat (about 12% of the circumference) was not making contact with the ball.

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3. 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.

4. The excellent shut-off characteristics of the gate valves appear to be the result of the wedge design (split-wedge in the Crane, 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.

5. 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 at other temperatures and pressure differentials indicated a consistent overall leakage pattern. Using this additional data, a general equation for valve leakage was derived and may be expressed as:

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

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

III DESCRIPTION OF EQUIPMENT AND TEST PROCEDURE

1. Test Loop Description

Figure 1 shows the arrangement of test loop #1 which was used for the first P-K ball valve and the Crane gate valve. 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 and immersion-type electric resistance heaters were utilized. For thermal insulation, 3-inch 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 except for using electric immersion heat only, instead of a combination of electric immersion heaters and a heating transformer, for thermal cycling. Nitrogen, instead of helium, was used to provide an inert atmosphere over the sodium.

2. Control System Description

The automatic 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 time-temperature pattern described previously. The program controller supplied an air-pressure signal to the autotransformer during heating or to the blower damper during cooling. The 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 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 transformer and immersion heat was shut off. According to the cycle 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 hours.

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 one group of immersion heaters, which were introduced at the start of heating, and 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.

3. Test Valve Description

The ball valves are intended to serve both throttling and shut-off functions; the gate valves are intended for shut-off only. All four of the test valves had identical water-cooled frozen-sodium stem seals backed up by a packing of John-Crane Superseal.

a. P-K Ball Valve

Figure 2 is a drawing, reproduced from Assembly Drawing No. E-7508-73403, of the P-K ball valve body manufactured by P-K Industries, N. Arlington, N. J. The valve body was cast from Type 304 stainless steel. A 10-inch 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.

Figure 3 is a photo of the ball and carrier after completion of 252 thermal cycles. 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 inches in diameter instead of 10 inches; also, the second valve has both a pneumatic and manual operator instead of a manual operator only.

b. Crane Gate Valve (Split-Wedge Disc)

Figure 4, reproduced from Crane drawing No. F-2187, shows the valve body as cast from Type 304 stainless steel. The valve is of split-wedge-disc design and has sealing and rubbing surfaces of Stellite. The valve had a manual operator.

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

Figure 5, reproduced from Cooper-Alloy Drawing No. 6-9005-35-0 shows the valve body which was cast from Type 304 stainless steel. The valve is of solid wedge design and has sealing surfaces of Stellite. The valve had both a pneumatic and manual operator.

4. Description of Thermal Cycle

The desired cycle for valve testing is specified in References 1 and 2. The cycle simulates thermal transient conditions more severe than those anticipated in HNPF and was to consist of the following heating pattern: 680°F to 960°F at 3°F/min.; 960°F to 1090°F at 6°F/min.; 1090°F 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; during pressure testing, fully closed. A total of 250 such cycles, with an across-the-seat leak check performed after approximately every 50 cycles, was desired. 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 two-hour heating period and a two-hour cooling period.

The urgency of the test program required utilization of an existing experimental loop which was extensively modified and used for testing the first P-K ball valve and the Crane gate valve. It was realized that this loop could not attain the peak temperatures desired, but it was agreed that a peak temperature of at least 1050°F would be satisfactory as this temperature exceeds the peak anticipated in HNPF. 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. A 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.

5. Leak Test Procedure

Referring to Figure 1, an across-the-seat leak check was performed in the following manner:

The E-M pump was shut off and the test valve was closed down tightly. 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 circulating 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. The valve temperature was then raised to the desired level and the test valve was again checked for full closure; on occasion the valve, while fully closed at low temperature, was slightly open at a higher temperature permitting 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 two minutes to one hour, depending on leak rate, with an average time of about 8 minutes. The number of tests was limited by the leak-accumulator volume.

IV DISCUSSION OF TEST RESULTS

1. First P-K Ball Valve

A total of 250 four-hour thermal cycles were 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 psi to 50 psi and with temperatures ranging from 520°F 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 psi 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.

2. 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 6 in the Appendix. 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 as explained below.

TABLE 1

TEST RESULTS FOR FIRST 12-IN. P-K BALL VALVE (10-IN. DIA. BALL)

NO. 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	29.0
	7	950	36.0
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	58.2
	30	750	19.4
	10	825	47.3
	20	825	50.9
	30	825	87.2
252	10	825	43.7

TABLE 2

TEST RESULTS FOR SECOND 12-IN. P-K BALL VALVE (12-IN. DIA. BALL)

NO. 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	0.774
	10	875	101.5
0 (After sodium circulation)	Prohibitively high valve leakage during loop filling procedure.		

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, as 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.

Disassembly of the second valve, after a few thermal cycles, showed a fairly heavy deposit on the ball which analysis revealed to consist of 41% sodium, 4% calcium, 3.5% carbon, 15% various metals, and the remainder as chlorides. The most likely source of carbon and chlorides was the organic solvent which was used to degrease the loop before filling with sodium. While the deposit on the ball may have 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 three-inch 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.

3. <u>Crane Gate Valve</u>

A total of 207 thermal cycles were completed with valve leak tests performed at zero cycles (before and after sodium circulation), 89, and 207 cycles. The full thermal cycling program was not carried out because the valve was needed in another facility. Twenty-three sodium leak tests were made with pressure differentials across the valve ranging from 10 psi to 50 psi and temperatures ranging between 550°F and 900°F.

Table 3 is a tabulation of the leak test data. It can be seen that only in one of the twenty-three leak tests was any leakage noted. The leakage rate was slight (1.68 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 caused four thermal cycles between 700°F and 1280°F 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: One, the split wedge disc offers two high resistance leakage paths in series instead of the usual single path; any leakage past the first disc could be stopped by the second disc. Two, the valve body is compact and of relatively

TABLE 3

TEST RESULTS FOR 12-IN. CRANE GATE VALVE (SPLIT-WEDGE DISC DESIGN)

NO. 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	1.68
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

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 one-inch 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 too is, for all practical purposes, leak tight.

4. Cooper-Alloy Gate Valve

A total of 168 thermal cycles were 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 psi to 50 psi, and temperatures ranging between 505°F and 975°F.

The full thermal cycling program was not carried out because the valve was needed in another facility. Table 4 is a tabulation of all leak test data. It can be seen that no sodium leakage took place under any of the test conditions and, 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 offering two high resistance sodium leakage paths in series instead of one and the compact, relatively light construction of the valve body minimizing stress and distortion resulting from thermal transients.

5. 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 packing made of John Crane Superseal. 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 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. In view of these conditions, the stem cooling system appears oversized.

TABLE 4

TEST RESULTS FOR 12-IN. COOPER-ALLOY GATE VALVE (SOLID-WEDGE DISC)

NO. 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

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 2). 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 3 and 4). 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.

V NOMENCLATURE (As used in Appendix)

T = valve temperature, °F

T_o = maximum valve temperature at which no sodium leakage occurs (after thermal cycling), °F

R = valve sodium leakage rate, gallons per hour

ΔP = 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 = sodium mass velocity, based on flow through leakage area, lb/hr - ft^2 $= \frac{W}{dc\pi}$

D_h = hydraulic diameter of leakage annulus, ft, = 2c

N_R = Reynolds number $= \frac{GD_h}{\mu} = \frac{2W}{\pi d \mu}$

μ = absolute viscosity of sodium, lb/ft-hr

g = acceleration of gravity, ft/hr^2 = 4.18×10^8

ρ = density of sodium, lb/ft^3

f = friction factor

x = friction factor exponent, for $N_R < 2000$, $x = -1$
for $N_R > 2000$, $x = -.16$

K = constant relating to friction factor, for $N_R < 2000$, $K = 16$
for $N_R > 2000$, $K = .04$

V = sodium velocity, ft/hr

M = ratio of leakage circumference/valve seat circumference

V REFERENCES

1. IOL from D. F, Casey and M. P. Heisler to F. W. Herrmann. Subject: Preliminary Specifications for SRE Intermediate Heat Exchanger. Date: January 24, 1958.

2. IOL from R. Cygan to R. B. Gordon. Subject: HNPF Valve Evaluation. Date: April 21, 1958.

3. Heat Transmission, McAdams.

4. Original data for this report are recorded in AI Laboratory Notebooks Nos. 033451 and 060451.

VI APPENDIX

1. Correlation of Test Data

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 which appeared logical and fitted the test points well. In areas where data were limited, reliance was placed on reasoning 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.

Figure 6 is a plot of valve leakage rate versus valve temperature for all test points obtained for the first P-K ball valve. For tests made after thermal cycling it can be seen that no leakage occurs up to temperatures of 500°F 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 gallons per hour plus one. The average slope of the lines is 8.5.

From Figure 6 the leakage rate can be expressed as:

$$R = \left[\frac{T}{T_o}\right]^{8.5} -1 \qquad (1)$$

T_o is 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_o, the leakage rate can be expressed in a more meaningful manner as:

$$R = \left[\left(\frac{T}{\Delta P x F}\right)^{8.5} -1\right] \qquad (2)$$

F is equal to $T_o/\Delta P$, where ΔP is the pressure differential across the valve, in psi. See Figure 7 for the variation of F for the ΔP range of 10 to 50 psi. The derivation of the curves for pressure differentials below 10 psi is explained below.

Figure 8 is a plot of valve leakage rate versus pressure differential with valve temperature as a parameter. The curves were obtained from a cross-plot of Figure 6 for the pressure differential range of 10 to 50 psi. The curves shown below 10 psi are based on a logical extrapolation of the data to zero leakage, at zero pressure differential. Accordingly, Figure 7 has been extended in order for equation (2) to be valid down to $\Delta P = 1$ psi. It should be noted that, below $\Delta P =$ 5 psi, F becomes a function of valve temperature. Figure 8 can be read directly for 236 cycles but, for any other number of cycles, the correction factor in Figure 9 must be applied to the indicated leakage rate.

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, data taken at zero cycles and 73 cycles. (See Figure 6). This indicated that the number of cycles had some effect on the leakage rate. This variation was determined by using the data obtained at zero cycles, 73 cycles, and from Figure 8. The inclusion of this factor resulted in the final leakage expression as follows:

$$R = \left(\frac{N}{236}\right)^{.35} \left[\left(\frac{T}{\Delta P x F}\right)^{8.5} - 1\right] \qquad (3)$$

Figure 10 is an error plot showing the ratio of computed leak rate, using equation (3), to experimental leak rate for all thirty test points. For eight 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 error was moderate. Fairly good correlation is indicated as 57% of all test points fall within ±17% of the computed leak rate.

2. Explanation of Leakage Pattern for the First P-K Ball Valve

The leakage pattern shown in Figure 8 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: one, it increases the fluid pressure drop across the valve seat which tends to increase the leakage rate; two, it increases the seating pressure between the ball and seat which tends to reduce the leakage area and the leakage rate.

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

The pressure drop across the valve seat is:

$$\Delta P = 2f \rho \frac{L}{D_h} \frac{V^2}{g\,144} \qquad (4)$$

where the friction factor, $f = K \left(\frac{GD_h}{\mu} \right)^x$ (5)

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:

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

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 and Reynolds number) is determined by the character of the flow; i.e., laminar or turbulent. For Reynolds numbers less than 2000 (laminar flow), f = -1 and K = 16; for Reynolds numbers greater than 2000 (turbulent flow), f = -.16 and K = .04. By use of equation (3), W (as R) can be obtained at a given temperature and ΔP and c can then be determined.

Figure 11 shows the variation of seat clearance with pressure differential, at constant temperature, as calculated from equation (6). The leakage seat clearances are on the order of one thousandth of an inch. It will be noted that 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 up 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 at 1 and 10 psi were in the laminar flow region, based on a full annular leakage area, with M = 1. However, if the leakage area was 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 11, 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 affer a possible explanation.

Some concrete 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 circumference was not making contact with the ball. Equation (6) showed the radial clearance for the leakage area portion of the seat to be .00125 inches, for 10 psi pressure differential, at 875°F, and M = .12. Reference to Figure 11 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.

12 INCH VALVE TEST LOOP

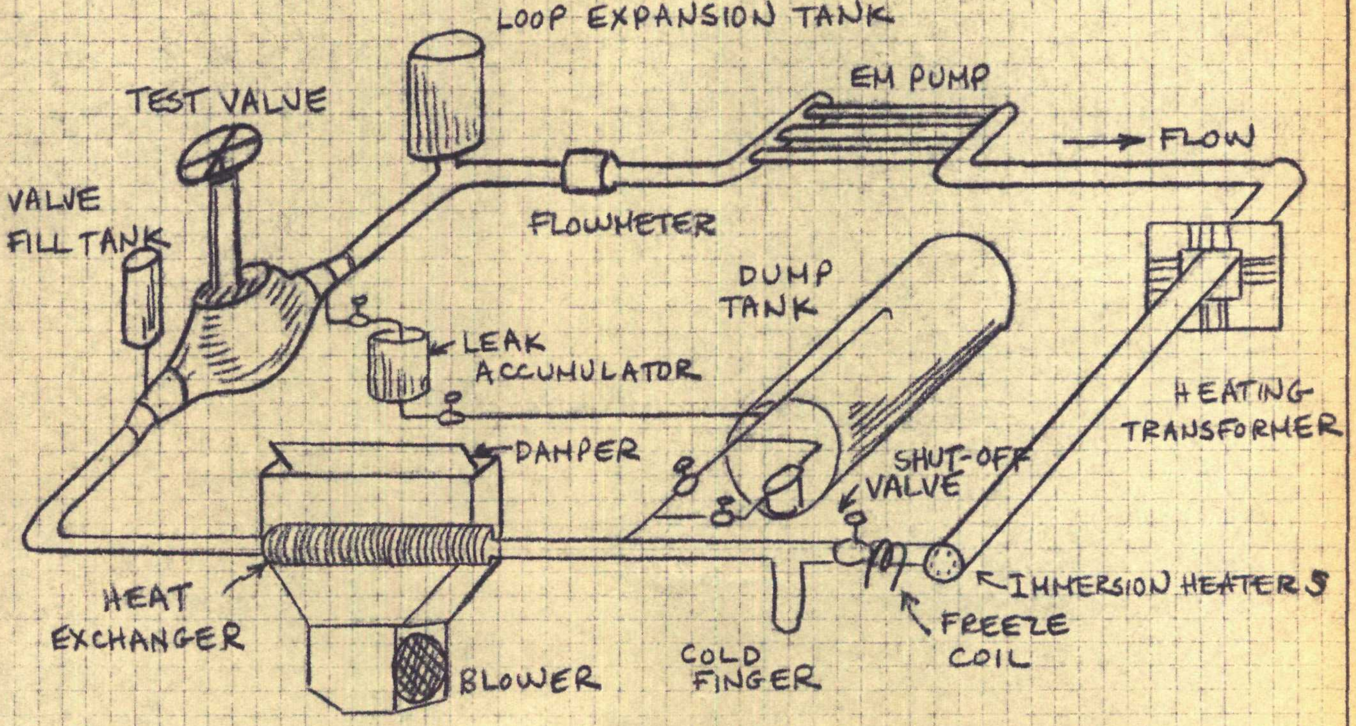

FIGURE 1

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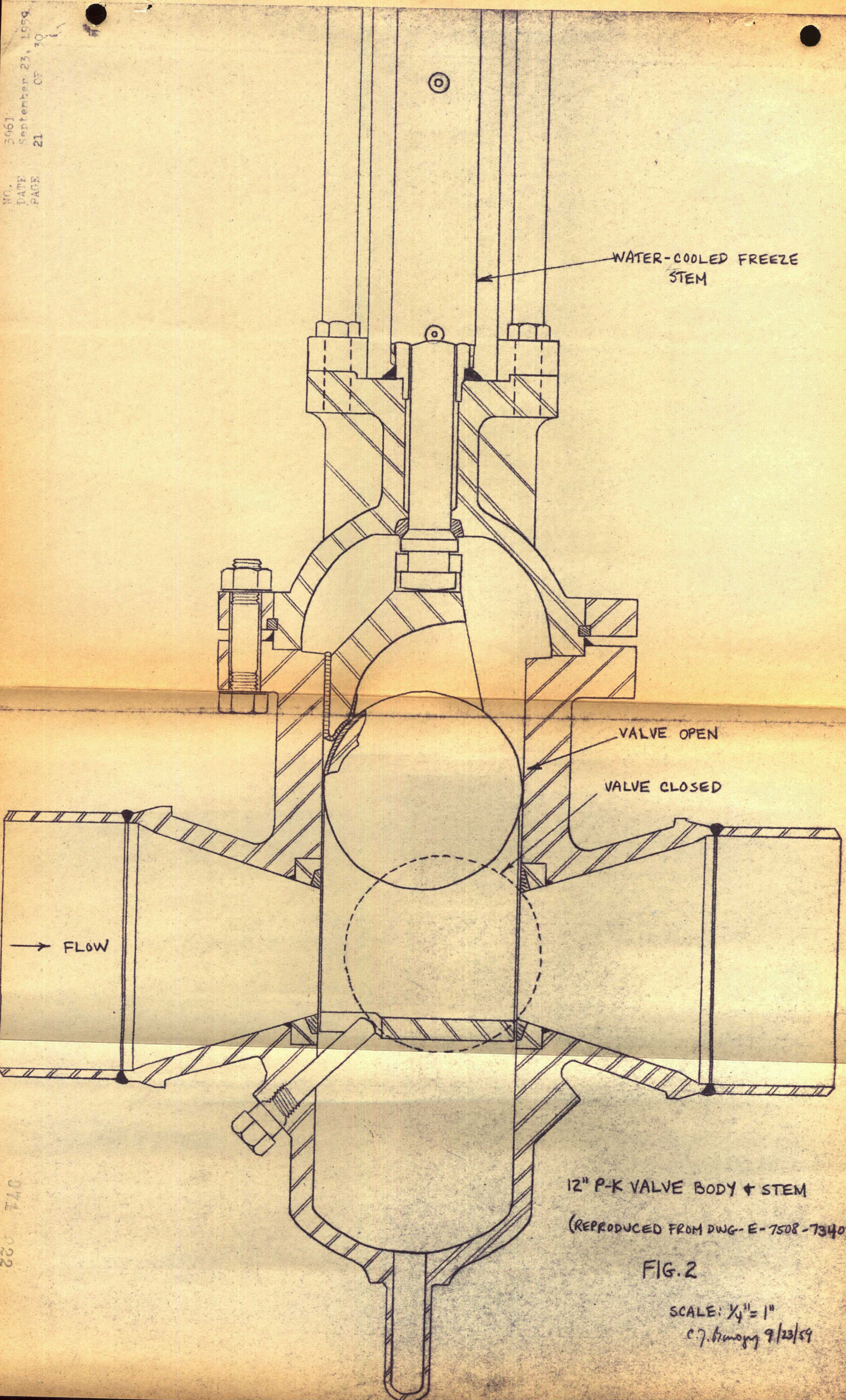

12" P-K VALVE BODY & STEM

(REPRODUCED FROM DWG-E-7508-73403

FIG. 2

SCALE: 1/4" = 1"

C.J. Kunogny 9/23/59


7508 - 5444 B
UNCLASSIFIED
PHOTOGRAPHIC DEPT. CANOGA PARK, CALIF.

12 INCH CRANE SODIUM VALVE

(SPLIT WEDGE DISC)

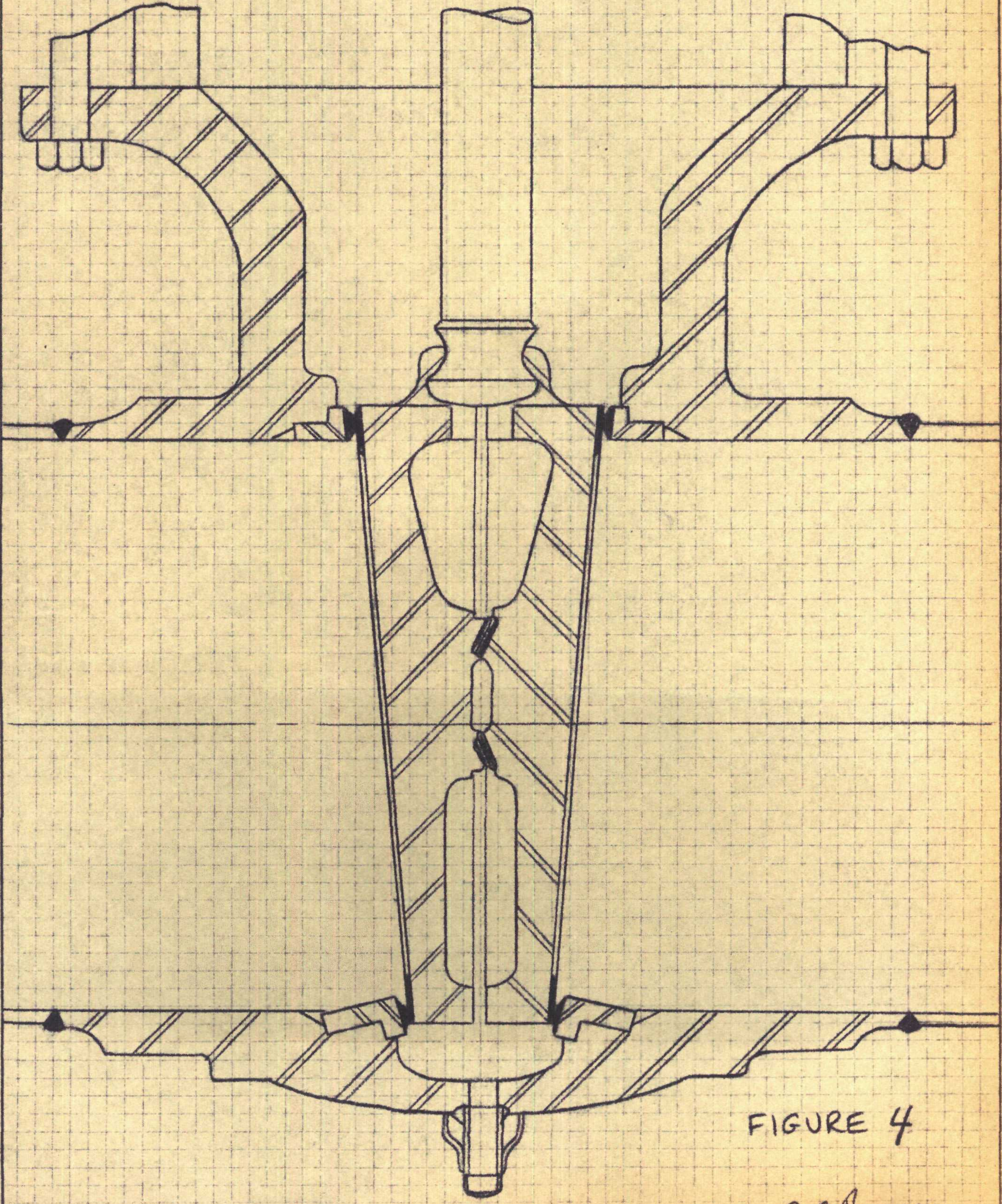

FIGURE 4

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9/23/59

PREPARED BY:	ATOMICS INTERNATIONAL A DIVISION OF NORTH AMERICAN AVIATION, INC.	PAGE NO. 24 OF 30
CHECKED BY:		REPORT NO. 3961
DATE:	12 INCH COOPER-ALLOY SODIUM VALVE	MODEL NO.

(SOLID WEDGE)

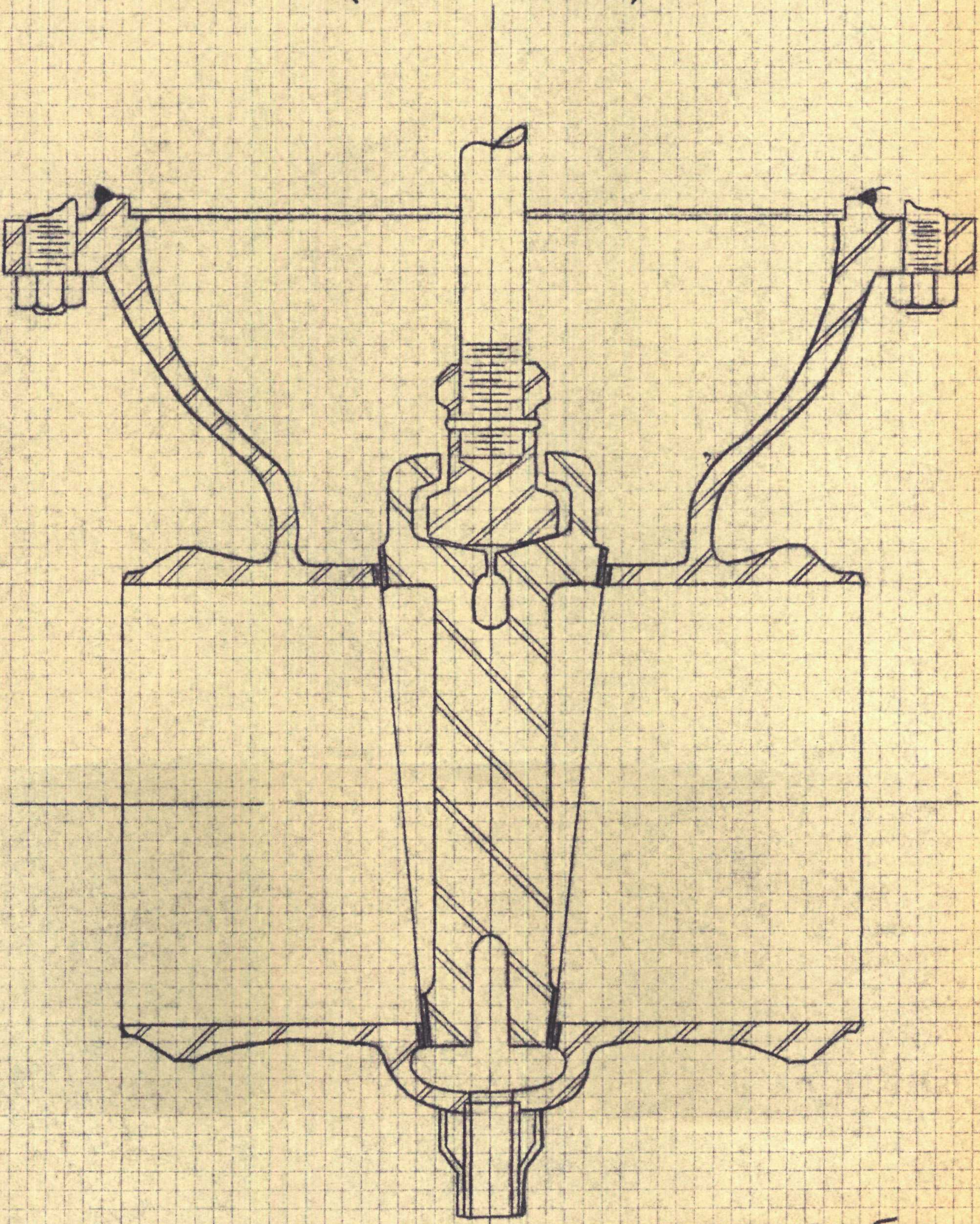

FIGURE 5

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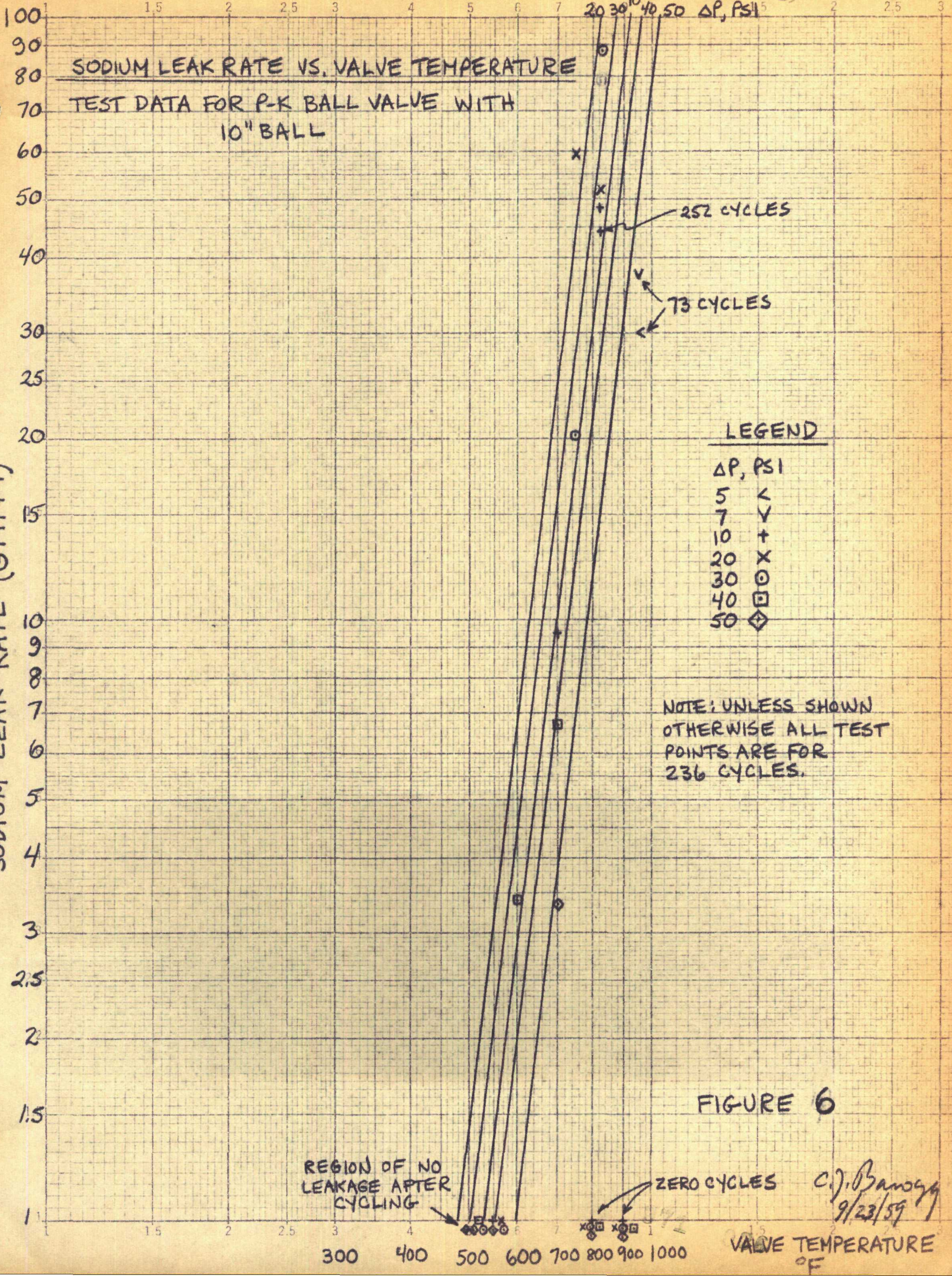
SODIUM LEAK RATE VS. VALVE TEMPERATURE
TEST DATA FOR P-K BALL VALVE WITH 10" BALL
20 30 10 40 50 ΔP, PSI
252 CYCLES
73 CYCLES
LEGEND
ΔP, PSI
5 <
7 Y
10 +
20 X
30 ⊙
40 ⊡
50 ◇
NOTE: UNLESS SHOWN OTHERWISE ALL TEST POINTS ARE FOR 236 CYCLES.
REGION OF NO LEAKAGE AFTER CYCLING
ZERO CYCLES
300 400 500 600 700 800 900 1000
VALVE TEMPERATURE °F
C.J. Banogy
9/23/59

FIGURE 6

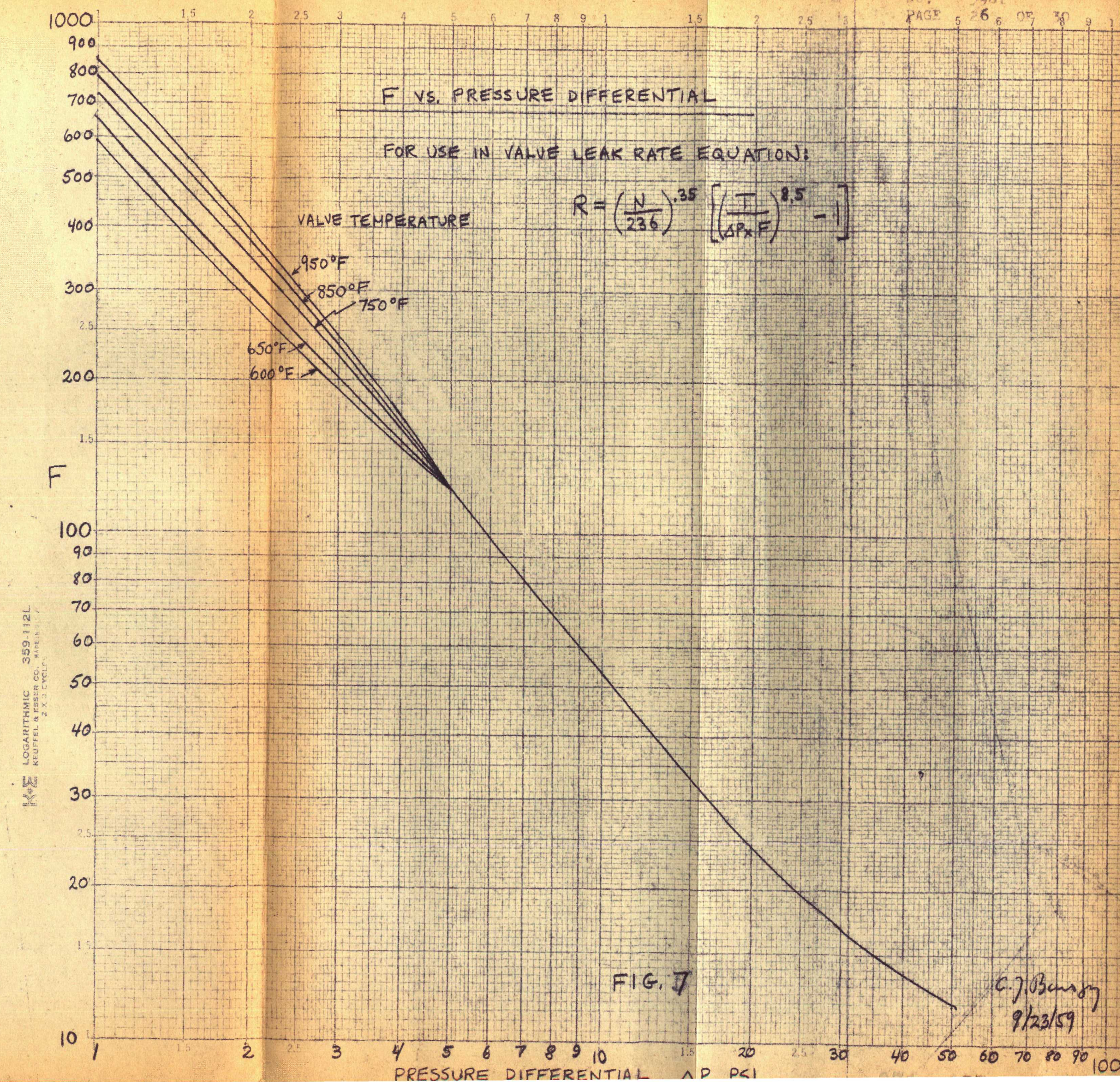

FIG. 7

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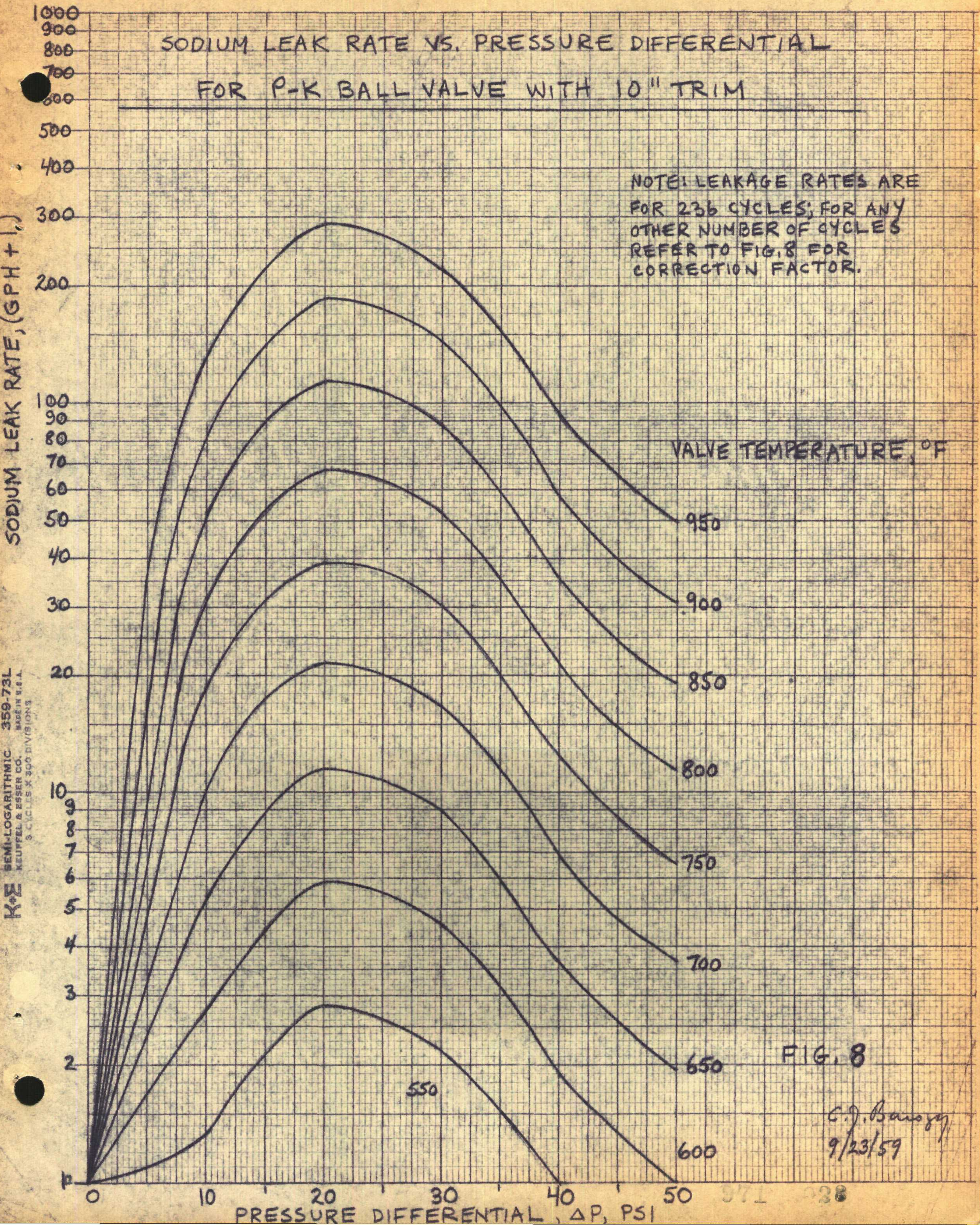

FIG. 8

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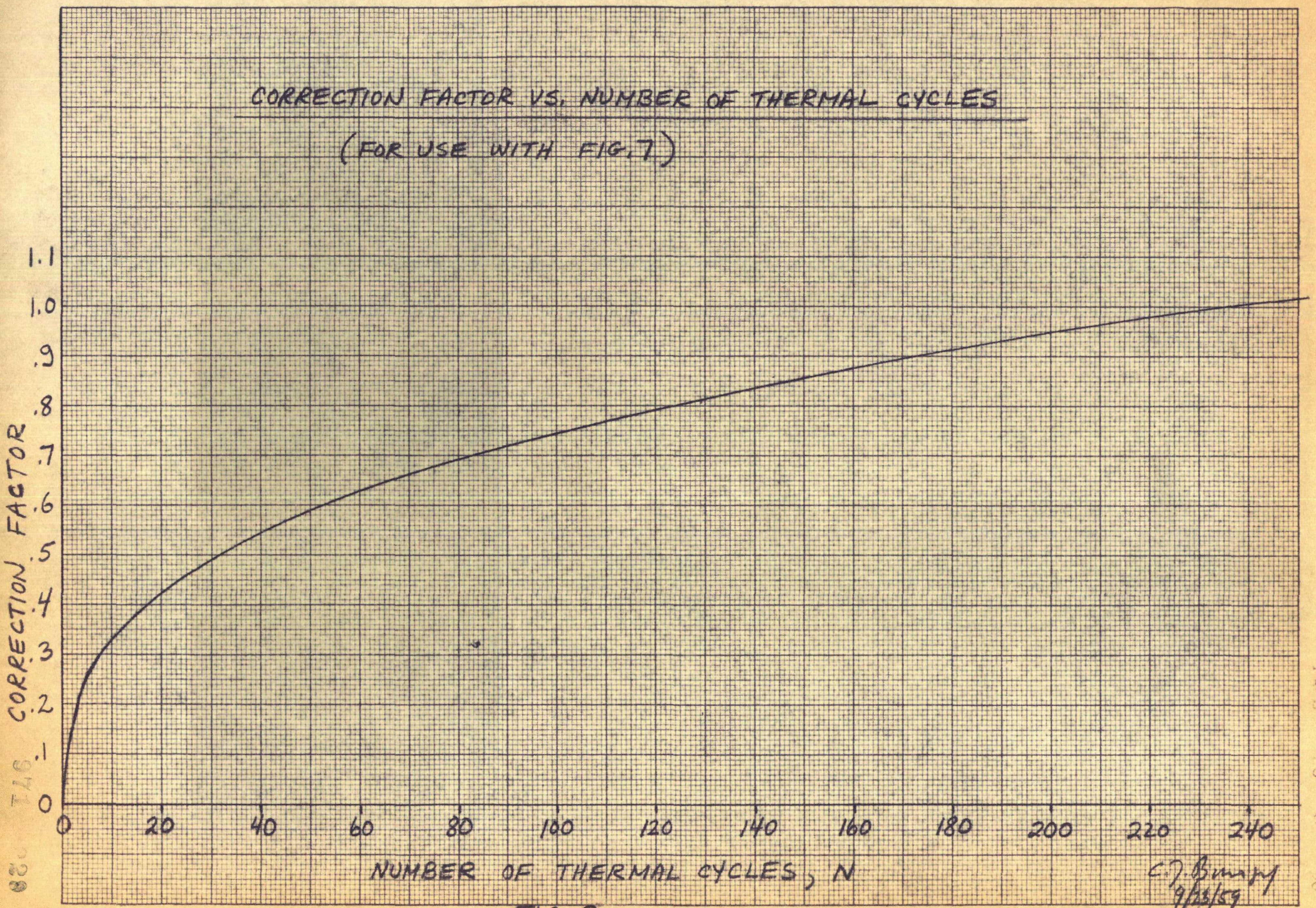
CORRECTION FACTOR VS. NUMBER OF THERMAL CYCLES
(FOR USE WITH FIG. 7)
CORRECTION FACTOR
0
.1
.2
.3
.4
.5
.6
.7
.8
.9
1.0
1.1
0
20
40
60
80
100
120
140
160
180
200
220
240
NUMBER OF THERMAL CYCLES, N
9/23/59

FIG. 9

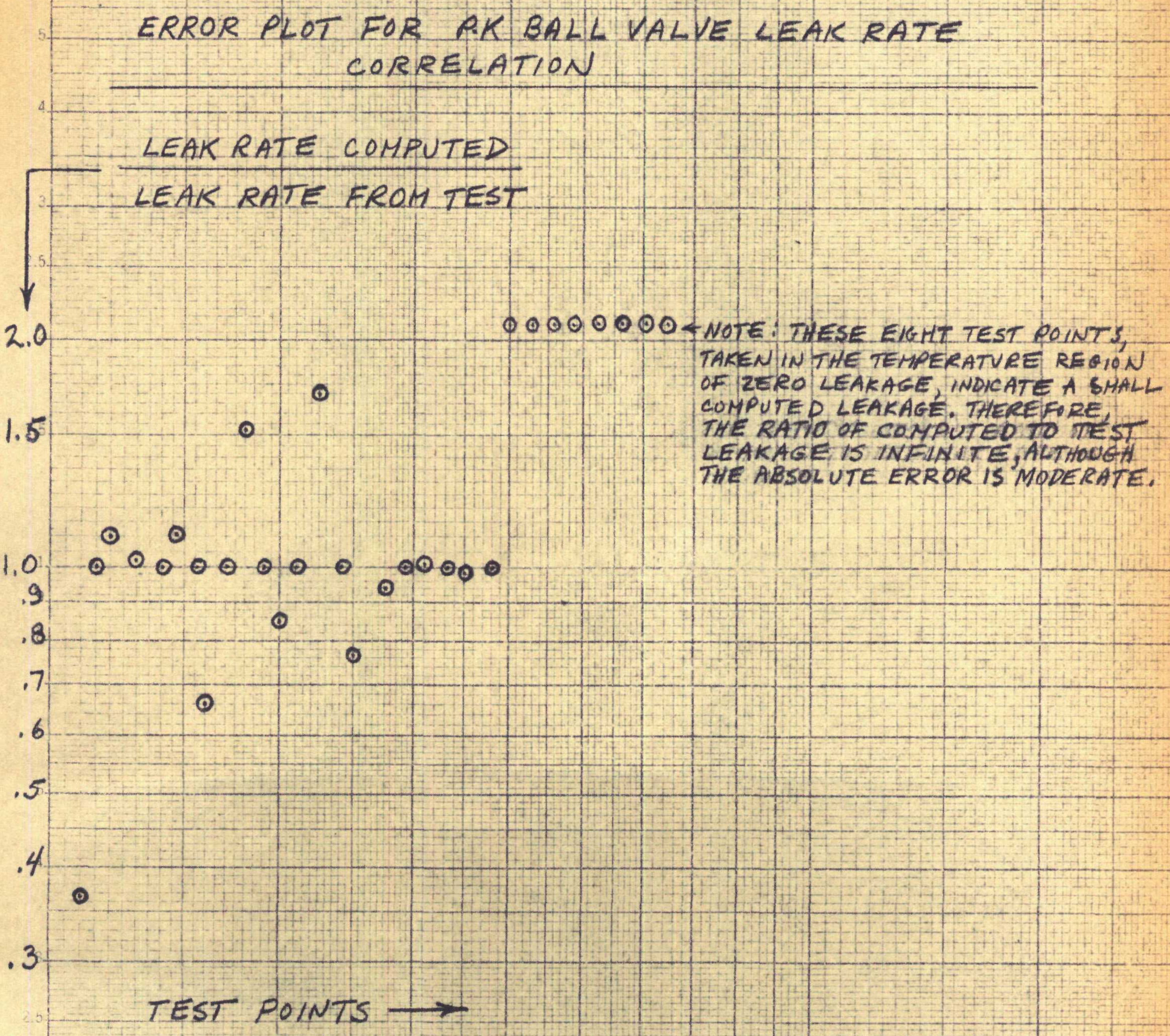

FIGURE 10

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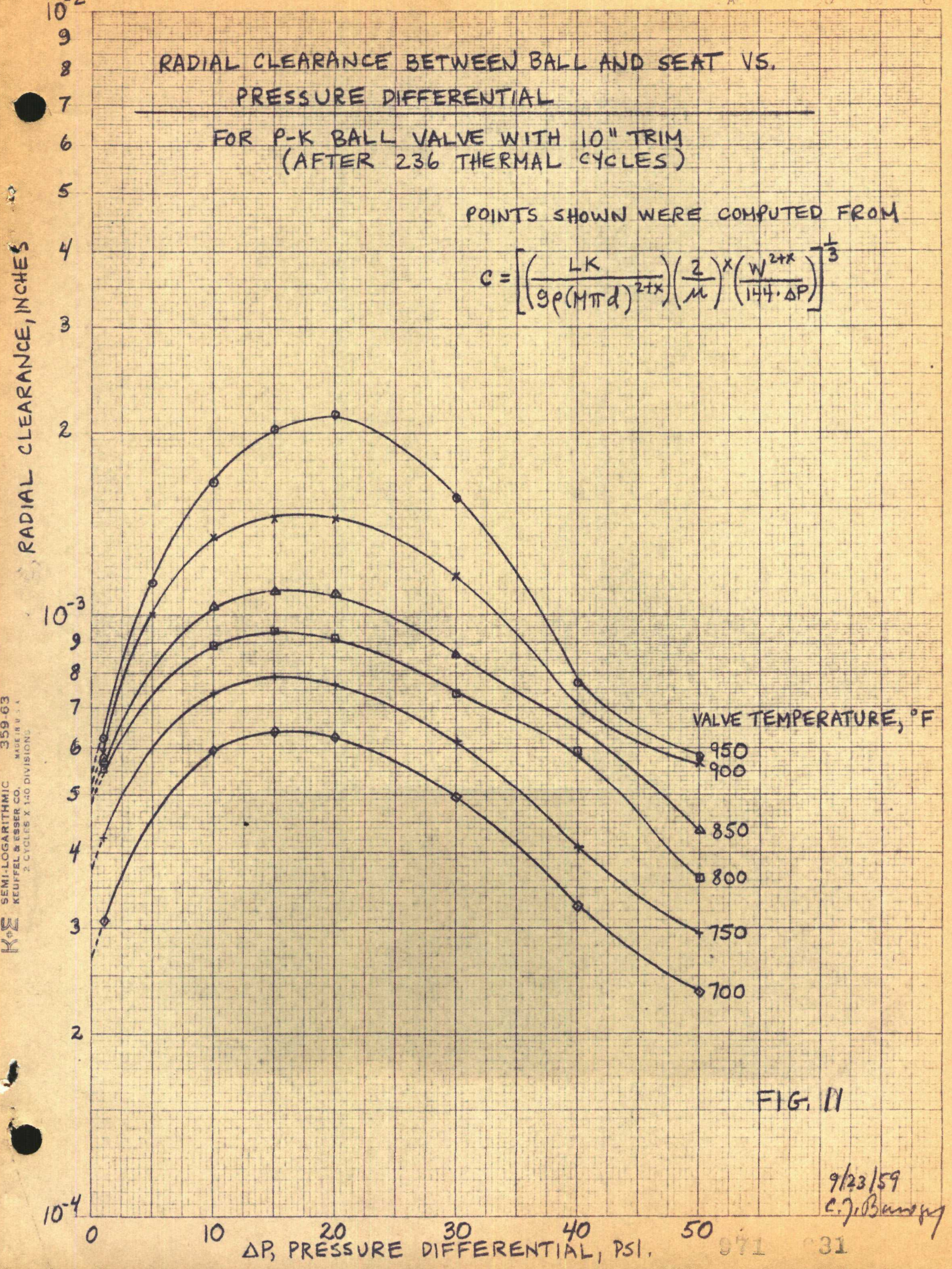

FIG. 11

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