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General Electric Company
KNOLLS ATOMIC POWER LABORATORY
Schenectady, New York

KAPL-M-KDL-1

ANALYSIS OF THE SODIUM AND WATER PERFORMANCE
OF THE ATWOOD-MORRILL SWING CHECK VALVE

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COMPONENTS SUB-UNIT

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I N T R O D U C T I O N

The Atwood and Morrill swing check valve, drawing No. 3654 F, was tested with sodium flow at KAPL. A second valve was modified for water flow tests at M.I.T.

During sodium flow tests it was observed that inconsistent flow rates were required to close the valve. An analysis was made to determine the effects of friction and valve inclination upon the sodium flow rate required to close the valve. This analysis was also correlated with the results of the M.I.T. tests.

Recommendations are made which may minimize the flow rate required to close this type of valve design.

CONCLUSIONS

It was shown that valve inclination will have a greater effect upon the back flow required to close the valve than friction. Increasing the coefficient of friction from zero to one causes an increase of approximately 15% in the flow rate required to close the valve. Doubling the angle of inclination, such as $\alpha=20^\circ$ to $\alpha=40^\circ$, causes an increase of approximately 140% in the closing flow rate. However, friction should be kept to a minimum in order to facilitate valve closing and to ensure that the disc arm returns to a true no flow position.

A satisfactory correlation of water flow test data and analytical results was obtained. The derived empirical expression for steady state conditions was used to determine the valve performance with sodium.

It is recommended that the water flow tests be extended to include the following tests:

- 1) Head Loss versus forward and reverse flow with the valve locked fully opened.
- 2) Head Loss and torque required to hold disc in fixed position versus reverse flow for various no flow angles.
- 3) Torque to hold disc in fixed position versus no flow angle with valve full of water.

The following test is recommended for inclusion in any further sodium flow tests upon this type of valve:

- 1) Head loss versus forward sodium flow rate.

It is believed that the following modifications will improve the performance of this valve design by reducing the back flow required to close the valve:

- 1) Other bearings or bearing materials be selected which minimize the coefficient of friction. (Other recommendations concerning bearing design are noted in appendix II).
- 2) Minimize weight of moving parts.
- 3) Shape the back of the disc to resemble an impulse turbine bucket to increase the hydraulic forces tending to close the valve.
- 4) Include a turning vane in the upper portion of the valve body to re-direct the back flow at a more favorable angle across the disc.

DISCUSSION

An examination was made of the data accumulated during the M.I. T. water flow test. The form in which this data was presented did not allow for evaluating for the effects of friction. Also, the torques or weights applied to obtain fictitious no flow angles obscured the hydraulic characteristics of the valve.

These torques and weights, which were added to simulate valve body inclination, also compensated for bearing and seal friction. It was noted that the frictional torques were inconsistent so that it was impossible to determine either the bearing friction or seal friction. For the purposes of analysis it was considered that the bearing coefficient of friction and seal torque were constant. Certain values for the bearing coefficient of friction and seal torque were assumed to complete the computations.

Figure 5 Graph KH-9A2620 is a plot of K_L as a function of the no flow angle using M.I.T. head loss data for forward and reverse flow in the horizontal position.

$$\text{Where } K_L = \frac{2gh_d}{V^2}$$

V = Ave. Flow Velocity - ft/sec

g = Acceleration Of Gravity - ft/sec²

h_d = Head Loss Thru Valve - ft of H₂O

The no flow angle is defined as the angle that the disc arm makes with the valve seat. See figure 1, sketch K-124, for a definition of the subdivisions of this angle and a schematic diagram of the valve construction. It was noted that the data for the forward flow with a 60° inclination of the valve also correlated with this curve but that the coefficient for reverse flow with a 60° inclination of the valve was erratic.

For the purposes of analysis it was assumed that one curve describes the hydraulic relationship between K_L and no flow angle for all valve positions and inclinations. Figure 6, Graph KH-9A2621 is a replot on log-log paper of the hydraulic relationship between K_L and no flow angle. From this curve an empirical relationship between K_L and no flow angle was obtained. This equation is:

$$K_L = 697 (\beta + 9)^{-1.646}$$

At this point an analysis was made to determine the mathematical relationship between the no flow angle and the reverse flow required to close the valve. A complete list of the assumptions used in the analysis and the procedure for analysis is in Appendix I.

It is to be noted that the analysis applies to steady state conditions only. The flow required to close the valve will be affected by either transient conditions of flow or movement of the disc. It may also be pointed out that the coefficient of kinetic friction will generally be lower than the coefficient of static friction.

To perform the analysis it was necessary to assume that all the pressure drop (head loss) is exerted across the disc and the force associated with this drop acts perpendicular to the disc face. This assumption may be refined by determining the hydraulic loss of the valve with the disc locked in the full open position. This data would allow a more accurate determination of the relationship between K_L and the no flow angle as noted above. The modified K_L would relate only the loss due to disc, and not the entire losses of the valve, to the no flow angle.

The following empirical relationship was determined for the steady state stability of the A and M swing check valve with reverse flow:

$$-W_a r_a - [\mu r (W_a + W_b) + T_s] + W_b L_{cg} \sin \theta + \frac{6.08 \times 10^{-3} P L_a D^2 Q^2}{A_p^2 (\beta + \alpha + \theta)^{1.646}} = 0 \quad (21)$$

Where the following nomenclature is used:

W_a = Applied external load	lbs.
W_b = Buoyant weight of disc and arm	lbs.
r_a = Radius of external load disc	inches
r = Shaft radius	inches
D = Disc diameter	inches
L_{cg} = Distance from shaft centerline to center of gravity of disc and arm	inches
L_d = Distance from shaft centerline to center of disc	inches
A_p = Pipe flow area	sq. inches
Q = Flow rate	GPM
T_s = Seal and packing torque	inch - lbs.
ρ = Density of fluid	lbs/ft ³
μ = Coefficient of friction	
α = Angle of inclination valve	degrees
β = Offset angle of valve seat	degrees
θ = Angle of disc arm from vertical	degrees

* See figure I Sketch K-124 for method of measuring and sign of angle.

Equation 21 may be used for forward flow if the sign of the last term is changed from plus to minus.

Graphs KH-9A2625 and KH-9A2624 show calculated results of equation 21 for reverse water flow with a valve inclination of 0 and 60° respectively. The curves are designated with a number which represents a simulated no flow angle. These angles were obtained by adding weights to overcome disc weight and seal and bearing friction during the M.I.T. water tests.

From these curves the flow required to close the valve is determined. This flow is the maximum value which is calculated for a given set of service conditions. Graph KH-9A2623 is a cross plot of the maximum values observed on Graphs KH-9A2625 and KH-9A2624 for water flow. The estimated reverse flow rates required to close the valve during the M.I.T. water tests are also plotted on graph KH-9A2623. It may be observed that close correlation between experimental and calculated values was obtained.

It is believed that a closer correlation between calculated and observed values may be obtained by giving more consideration to the variables shown by analysis to have an effect upon the flow rate required to close the valve. This may be accomplished by more accurately determining the effects of or eliminating the external seal torques and applied loads. The externally applied loads not only add torque to the system but also affect the bearing loading.

Using the physical properties of sodium at 570°F, Graph KH-9A2622 was calculated from equation 21 for a 60° valve inclination. This graph illustrates the effect of the coefficient of friction of the bearing materials upon the position of the disc arm for given flow rates. A change in the coefficient of friction from zero to one causes approximately fifteen per cent increase in the flow rate required to close the valve.

Figure 3, Graph KH9A2649 shows that the back flow required to close the valve is primarily controlled by the position of the disc in the valve. However, the angle of inclination, weight of the disc and arm, and the coefficient of bearing friction affect the position assumed by the disc. The effect of friction and the angle of inclination for a sodium system valve is shown on figure 4, Graph KH9A2648.

During the test of the A and M swing check valve with a 60° inclination in a sodium system the valve was operated approximately 1000 times. It was observed that the following back flow rates were required to close the valve on the dates indicated:

TABLE I

Date	Cycles of Continuous Operation	Back Flow Rate Required GPM	Temperature °F.	Remarks
June 11, 1952	1	115	800°F.	(Flow gradually increased until
June 12, 1952	1	120	580°F.	unstable operation of disc was noted.
August 15, 1952	1	311	580°F.	(Fixed flows suddenly admitted
September 4, 1952	1	585	580°F.	(to valve.
" " "	10	522	580°F.	(
" " "	20	470	580°F.	(
" " "	25	418	580°F.	(
" " "	30	418	580°F.	(
September 22, 1952	1	366	580°F.	(

These values may be compared with the calculated results of 445 gpm when $\mu = 0$ and 505 gpm when $\mu = 1.0$ (reference figure 9, Graph KH9A2622). This would lead one to believe that the valve was operating satisfactorily on September 4, 1952, but that the bearing friction was in effect decreasing as a result of repeated operations. This effective reduction of friction may be the result of one or more of the following:

- 1) "Run-in" or increased "wetting" of the bearing surfaces.
- 2) The disc arm aligning itself so that it does not rub on the bearing thrust faces.
- 3) Removal of any metallic or non-metallic contaminants in the bearings.

On the basis of the analysis, it would appear that the disc and arm did not assume a normal opening during the earliest tests. The results of the early tests neither compare with the M.I.T. water flow tests nor the calculated data for a sodium system. This may have been caused by some obstruction in the valve which would prevent normal opening of the disc and arm.

As noted in appendix II, the bearings after test were out of round and also had a "high spot" tending to prevent rotation of the shaft. Assuming that the "high spot" may have occurred during disassembly of the valve and that the bearings were concentric on assembly, misalignment of the bearings may have occurred during installation in the loop or may have been caused by differential thermal expansion or thermal distortion of the valve body. It is possible that these effects may have allowed the disc arm to move to a limited opening position during the early test period without seizing in this position. Continued operation of the valve probably opened up the bearing clearances, thereby allowing the disc to assume a more normal position as the test program progressed, thus resulting in higher back flow rates required to close the valve.

Bearing clearance may be opened up by wear of the bearing material or deformation of the annealed 18-4-1 HSTS bearing bushing.

To prevent the misinterpretation of test data in any additional swing check valve tests it is recommended that pressure drop through the valve versus forward flow rate be taken and compared with calculated results.

ILLUSTRATIONS

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A+M SWING CHECK VALVE DWG. NO. 3654 F
WATER FLOW REQUIRED TO CLOSE VALVE

LEGEND: ——— CALCULATED
----- ESTIMATE OF ACTUAL
MIT TEST DATA

D. B. VAIL / M. ALBERT FEB. 25, 1953

GENERAL ELECTRIC CO.
KNOLLS ATOMIC POWER LAB

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A+M SWING CHECK VALVE DWG NO 3654 F
SODIUM FLOW REQUIRED TO CLOSE VALVE

LEGEND: O-O- μ =0
X-X- μ =0.5
Δ-Δ- μ =1.0

SODIUM FLOW REQUIRED TO CLOSE VALVE, GPM

D.B. VAN ALBERT MARCH 13, 1953

909 014 KH-9A2649

16

A + M SWING CHECK VALVE DWG. NO 3654 F
SODIUM TEMPERATURE - 580°F
EFFECT OF INCLINATION + COEFFICIENT OF FRICTION ON CLOSING FLOW

GENERAL ELECTRIC CO.
KNOLLS ATOMIC POWER LAB.

DIBVALL/M. ALBERT MARCH 12, 1953

909 015 KH-9A2648

17

A+M SWING CHECK VALVE DWG. NO. 3654 F
 IN HORIZONTAL POSITION WITH WATER FLOW (FORWARD DIRECTION)
 MIT TEST DATA

GENERAL ELECTRIC CO.
 KNOLLS ATOMIC POWER LAB.

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LANTZ-VAIL/MLA
 FEB. 16, 1953

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KH-9A2620

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A+M SWING CHECK VALVE-DWG. NO. 3654 F
IN HORIZONTAL POSITION
WITH WATER FLOW (FORWARD DIRECTION)

GENERAL ELECTRIC CO.
KNOLLS ATOMIC POWER LAB.

KH-9A2621

909 017

LANTZ/ALBERT

FEB, 20, 1951

19

A + M SWING CHECK VALVE DWG. NO. 3654 F
WATER TESTS 0° INCLINATION
CALCULATED REVERSE FLOW
FOR
STEADY STATE CONDITIONS

ASSUMPTIONS - $T_s = 1 \text{ IN-LB.}$
 $\mu = 0.1$

NOTE - CURVE NUMBERS INDICATE
THAT CONSTANT TORQUE
WAS APPLIED TO HOLD THE
VALVE OPEN.

KD LANTZ - DEVAL/M.L.A. FEB 16, 1953

GENERAL ELECTRIC CO.
KNOLLS ATOMIC POWER LAB

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KH-9A2625

20

A. M. SWING CHECK VALVE DWG. NO. 3654F
WATER TESTS 60° INCLINATION
CALCULATED REVERSE FLOW
FOR
STEADY STATE CONDITIONS

ASSUMPTIONS:
 $T = 1 \text{ in.-lbs.}$
 $\mu = 0.1$

NOTE: CURVE NUMBERS INDICATE
THAT CONSTANT TORQUE WAS
APPLIED TO HOLD THE
VALVE OPEN

LANTZ-VAIL/m.l.a. FEB. 17, 1953

GENERAL ELECTRIC CO.
KNOLLS ATOMIC POWER LAB.

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A + M SWING CHECK VALVE DWG. NO. 3654 F
SODIUM TESTS 60° INCLINATION
CALCULATED REVERSE FLOW
FOR
STEADY STATE CONDITIONS

GENERAL ELECTRIC CO.
KNOLLS ATOMIC POWER LAB.

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LANTZ-VAIL/MALBERT FEB. 17, 1953

909 020 KH-9A2622

APPENDIX IANALYSIS

Appendix I

Analysis of steady State Conditions in Atwood & Morrill Swing Check Valve

The analysis is based upon the following assumptions:

- (1) Only steady state is considered.
- (2) All pressure drop in valve is exerted across disc.
- (3) Hydraulic force on disc consists only of pressure drop across disc and is considered to act at a right angle (90°) to the centerline of disc and arm. Aerodynamic lift and drag are not considered.
- (4) Force due to buoyant weight is concentrated at center of gravity of disc and arm assembly.
- (5) Hydraulic force is concentrated at center of disc.
- (6) Seal torque is constant for all positions of valve and disc.

The summation of torques about shaft ϕ must equal zero.

$$-T_a - T_f + T_h + T_w = 0 \quad (1)$$

The external applied torque is:

$$T_a = W_a r_a \quad (2)$$

The torque due to buoyant weight of disc is:

$$T_w = W_B l_{cg} \sin \theta \quad (3)$$

Friction torque always resists movement.

$$T_f = \mu r (W_a + W_B) + T_s \quad (4)$$

where T_s = constant seal torque

The hydraulic torque consists of the force associated with the pressure drop across the disc multiplied by its moment arm.

$$T_h = F_h l_{\phi} \quad (5)$$

Relating pressure drop to flow

$$K_L = \frac{2g h_d}{V^2} \quad (6)$$

$$V^2 = \frac{Q_1^2}{(\rho A_p)^2} \quad \text{where } Q_1 = \text{lb/sec} \quad (7)$$

$$h_d = \frac{\Delta P}{\rho} \quad \text{where } \rho = \frac{\text{lb}}{\text{ft}^3} \quad (8)$$

substituting 7 & 8 into 6

$$K_L = \frac{29 P A_p^2 \Delta P}{Q_1^2} \quad (9)$$

Solving for ΔP

$$\Delta P = \frac{K_L Q_1^2}{29 P A_p^2} \quad \text{Lbs/ft}^2 \quad (10)$$

From Fig. 5, graph KH-9A2621

$$K_L = 697 (\beta + \alpha + \theta)^{-1.646} \quad (11)$$

Substituting 11 into 10

$$\Delta P = \frac{697 Q_1^2}{29 P A_p^2 (\beta + \alpha + \theta)^{1.646}} \quad (12)$$

$$Q_1 \text{ Lb/sec} = Q \text{ gal/min} \times \frac{1 \text{ min}}{60 \text{ sec}} \times 0.1337 \frac{\text{ft}^3}{\text{gal}} \times \rho \frac{\text{Lb}}{\text{ft}^3}$$

$$Q_1 \text{ Lb/sec} = 0.00223 Q P \quad (13)$$

where $Q = \text{G.P.M.}$

$$A_p \text{ ft}^2 = A_p \text{ in}^2 \times \frac{1 \text{ ft}^2}{144 \text{ in}^2} \quad (14)$$

$$\Delta P \text{ Lb/in}^2 = \Delta P \text{ Lb/ft}^2 \times \frac{1 \text{ ft}^2}{144 \text{ in}^2} \quad (15)$$

Substituting 13, 14 & 15 into 12

$$\Delta P = \frac{0.2495 P Q^2}{9 A_p^2 (\beta + \alpha + \theta)^{1.646}} \quad \text{Lbs/in}^2 \quad (16)$$

Finding hydraulic force on disc

$$F_h = \Delta P A_d \quad \text{where } A_d = \text{Disc Area} \quad (17)$$

$$A_d = \frac{\pi D^2}{4} \text{ in}^2 \text{ where } D = \text{in} \quad (18)$$

Substituting 16 and 18 into 17.

$$F_h = \frac{6.08 \times 10^{-3} P D^2 Q^2}{A_p^2 (\beta + \alpha + \theta)^{1.646}} \text{ Lbs.} \quad (19)$$

Substituting 19 into 5

$$T_h = \frac{6.08 \times 10^{-3} P L_d D^2 Q^2}{A_p^2 (\beta + \alpha + \theta)^{1.646}} \text{ inch Lbs.} \quad (20)$$

Substituting 2, 3, 4 and 20 into 1.

$$-W_a a - [\mu r (W_a + W_b) + T_s] + \frac{6.08 \times 10^{-3} P L_d D^2 Q^2}{A_p^2 (\beta + \alpha + \theta)^{1.646}} + W_b \log \sin \theta = 0 \quad (21)$$

Equation 21 is the equation for stability of the A+M Swing Check Valve with reverse flow.

Stability Equation For A+M Swing Check Valve.

$$-W_a r_a - [u_r(W_a + W_b) + T_s] + \frac{6.08 \times 10^3 \rho L_d D^2 Q^2}{A_p^2 (\beta + \alpha + \theta)^{6.46}} + W_b l_{cg} \sin \theta = 0 \quad (21)$$

Nomenclature

Item	Description	Units	Values For Tests	
			Water	Sodium
W_a	Applied external load	Lbs	MIT Data.	None
W_b	Buoyant weight of disc & arm	Lbs	10.95	11.1
r_a	Radius of external load disc	Inches	5	None
r	Shaft radius	Inches	0.625	0.625
D	Disc diameter	Inches	8.5	8.5
l_{cg}	Distance from shaft Φ to center of gravity of disc & arm	Inches	5.46	5.46
l_{Φ}	Distance from shaft Φ to center of disc	Inches	5.75	5.75
A_p	Pipe flow area	Sq. inches	50	50
Q	Flow rate	G.P.M.		
T_s	Seal and packing torque	Inch Lbs.	Assumed	None
ρ	Density of fluid	Lbs/ft ³	62.4	54.9
μ	Coefficient of friction		Assumed	Assumed
α	Angle of inclination of valve	Degrees	MIT Data.	60
β	Offset angle of valve seat	Degrees	5	5
θ	Angle of disc arm from vertical	Degrees	Assumed	Assumed.

Evaluating equation 21 for water flow tests where the following values were assumed:

$$\mu = 0.1 \quad T_s = 1.0 \text{ inch-Lb.}$$

$$-5.063 W_a - 1.666 + \frac{0.063 Q^2}{(5 + \alpha + \theta)^{1.646}} + 59.8 \sin \theta = 0 \quad (22)$$

M.I.T data gave the following values for W_a

$\theta + \theta$	W_a Lbs.
2°	-1.375
8°	+0.50
12°	+1.61
20°	+3.61
30°	+5.86

The same value for W_a was used for all valve inclinations

Evaluating equation 21 for sodium flow tests.

$$-6.94 \mu + \frac{0.055 Q^2}{(5 + \alpha + \theta)^{1.646}} + 60.61 \sin \theta = 0 \quad (23)$$

where α in actual tests was 60°

and μ was assumed as:

$$\mu = 0.0, \quad \mu = 0.5, \quad \mu = 1.0$$

APPENDIX IIBEARING RECOMMENDATIONS

UNCLASSIFIEDAtwood and Morrill Co. Swing Check Valve Bearing Recommendations

Reference: Letter to S. E. Solomon from D. B. Vail, dated Jan. 26, 1953

An Examination of the Bearings After Test.

1) There is no visual evidence of material pick up between the 18-4-1 HSTS shaft and the 18-4-1 HSTS bushings. However, this does not mean that there has been no increase in the friction between these parts. Bearing development tests performed in Na and Nak since the original recommendations were made, indicate a marked increase in friction in tool steels as a function of temperature and of cycles of operation.

2) There are traces of material pick up between the stainless steel disc arm and the thrust faces of the 18-4-1 HSTS bushing. Various grades of austenitic stainless steel, which have been tested as part of our general program, exhibit high coefficients of friction.

3) There is visual evidence that the 18-4-1 HSTS has a porosity previously not noted in any bearing fabricated of 18-4-1 HSTS. To determine whether this is the result of poor material, corrosion, or some other effect will require a metallurgical analysis.

4) There is also evidence that the bushings were severely pounded to such an extent that the bushings are markedly out of round. The exact cause is not known. However, this may be the result of the forces created by the inertia forces of the liquid metal in the system.

Based upon previous test experience with bearings, the following observations are relative to the design of the swing check valve:

1) That many metals and 18-4-1 HSTS, in particular, have increased coefficients of friction with increased temperature.

2) That the coefficient of friction for 18-4-1 HSTS versus other materials indicates an increase in friction and marked variations in friction, as a function of the number of operations.

3) That the coefficient of static friction of other materials tends to increase as a function of the time at stand still.

4) That a shaft frozen by sodium oxides can be loosened by a sharp tap. Thereafter, the shaft will be free to rotate without undue drag.

The following recommendations are made relative to the bearing design and application:

1) Institute a design and development including the use of 18-4-1 HSTS roller bearings. This requires that the life and the maximum load capacity must be established for successful use of anti-friction bearings.

2) All bearing surfaces, which are required to operate, must be treated so that similar coefficients of frictions exist between the moving surfaces. The present design has several locations which do not follow this recommendation. One location is present between the disc arm