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**ATR PRIMARY COOLANT SYSTEM
CHECK VALVE PROBLEM ANALYSIS AND CORRECTION**

T. L. Bridges and T. K. Burr

IDAHO NUCLEAR CORPORATION
NATIONAL REACTOR TESTING STATION
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PLANT AND DESIGN ENGINEERING BRANCHES
ENGINEERING DIVISION

IDAHO NUCLEAR CORPORATION

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ABSTRACT

During the final stage of equipment and system testing and prior to initial operation of the Advanced Test Reactor, several deficiencies were discovered in the primary coolant system check valves. Each of these deficiencies, uncovered at different times, affected the safety and reliability of reactor operation and, consequently, required correction before startup of the reactor. Several Engineering and Operations groups were involved in contributing to a satisfactory solution in the shortest time. During the analysis, modification, and testing phase of the work, periodic status reports were submitted to the Idaho Operations Office of the USAEC. This report utilizes some of the previously written material and provides a complete documentation of events leading to the discovery of the check valve deficiencies, the resulting investigations, the solutions, and the tests conducted to verify the solutions. Recommendations also are included which should be considered in the future preparation of specifications for check valves.

SUMMARY

The analysis and correction of the problems associated with the primary and emergency pump discharge check valves used in the primary coolant system for the Advanced Test Reactor are the subject of this report. Deficiencies were discovered in both the original and replacement valves after installation and brief periods of operation. These deficiencies were:

Original Valves

Noncompliance with radiographic requirements for castings which resulted in scrapping of the valves.

Replacement Valves

- (1) Incompatible wear surfaces for bearing loads.
- (2) Valve response characteristics which resulted in insufficient reactor cooling during emergency situations and in undesirable waterhammer. No response characteristics were specified by the reactor Architect-Engineer for the original valves or by Idaho Nuclear Corporation for the replacement valves.

The original valves were rejected because they did not meet casting radiographic requirements. The replacement valves suffered bearing surface failures (galling) after short periods of operation, and the closing response characteristics resulted in excessive coolant backflow through the valves when pumps were shut down. Because of the excessive backflow prior to valve closure, waterhammer was observed during tests simulating power failure to one of three operating primary pumps. Flow coastdown tests by simulating power failure to all three operating primary pumps revealed a short period of insufficient flow to the reactor due to the short circuiting effect of slow valve closing times. Two major investigations were undertaken to determine solutions to (a) the galling problem and (b) the reactor cooling and waterhammer problem.

Two areas of galling developed: the first occurred between the disc hinge pin and its bushing and the second occurred between the damping piston rod and its guide bushing. The replacement valves contained bushings, pins, and piston rod made of 17-4 PH stainless steel with the following hardnesses:

Bushings	380-BHN (H-975)
Hinge Pins	341-BHN (H-1075)
Piston Rod	341-BHN (H-1075)
Link Pins	341-BHN (H-1075)

Calculations indicated that bearing loads were greater than allowable for the above material combinations. As a result of the subsequent investigation, the surfaces of the piston rods, link pins, and hinge pins were nitrided, providing a hardness difference greater than 500 BHN between rubbing surfaces. The

Type 304 stainless steel pistons were also nitrided. In addition, these items were coated with a solid film lubricant, Electrofilm #4396. To date, after 109 open and close cycles, there is no evidence of surface deterioration.

A design analysis indicated that a check valve closing time of from two to three seconds was required to essentially eliminate waterhammer under a 1-of-3 pump cutoff situation. To accomplish this, the valves were modified in accordance with the design analysis. Calculations were made predicting the modified valve performance, and these predictions were confirmed by tests conducted under operating conditions. Waterhammer was virtually eliminated and core cooling requirements also were satisfied under emergency conditions.

The following recommendations are suggested when designing check valves for incorporation into a hydraulic system which includes more than one pump supplying a common manifold or when flow parameters are critical.

- (1) Perform a waterhammer analysis considering all possible system and valve operating modes, and from this analysis establish the valve response characteristics necessary to prevent waterhammer.
- (2) Determine the emergency flow requirements of the system's dependent equipment to ascertain that the valve(s) action will not compromise these requirements.
- (3) When the check valve design is established, based on the system requirements, calculation of the valve bearing loads should be made and used as a criteria in the selection of bearing materials.
- (4) Test programs should be scheduled to verify the required valve performance, and in those applications where failure of the check valve would endanger personnel or equipment, instrumentation should be installed to predict deterioration of the valve performance.

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ATR PRIMARY COOLANT SYSTEM

CHECK VALVE PROBLEM ANALYSIS AND CORRECTION

I. INTRODUCTION

1. PRINCIPAL REPORT SECTIONS

The work involved in correcting the check valve deficiencies has been divided into five major topics as follows:

- (1) Galling of wear surfaces
- (2) Waterhammer and reactor core cooling requirements
- (3) Redesign of check valves
- (4) Development of test programs to verify analytical predictions
- (5) Test results.

It will become apparent in subsequent paragraphs that the events with respect to the above topics overlap or coincide with each other chronologically. To eliminate confusion which might develop from a chronological presentation, each topic is discussed separately. However, the remainder of the introduction will familiarize the reader with the sequence of events leading to the discovery of check valve deficiencies and subsequent investigations.

2. DESCRIPTION OF THE ADVANCED TEST REACTOR AND PRIMARY COOLANT SYSTEM

The Advanced Test Reactor (ATR) is a 250 megawatt (thermal) water cooled test reactor located at the National Reactor Testing Station in Southeastern Idaho. The ATR will be used as a tool to investigate the effect of intense radiation on the properties of reactor materials. Currently, the ATR is progressing through initial startup testing.

The ATR primary coolant system (PCS) is shown schematically in Figure 1.

Coolant flow at 47,000 gpm takes the following route through the primary system: Water at 170°F leaves the reactor vessel through four 18-inch-O.D. pipes and merges into a 36-inch-O.D. header which leads to the five tubular heat exchangers. Water passes through the tubes and is cooled to 130°F before entering the primary pumps. Three of the four manifolded pumps are normally in service to cool the reactor when at full power; these are horizontal, double

Fig. 1 Flow diagram of ATR Primary Coolant System.

suction, volute, centrifugal pumps. Each primary pump can be isolated with suction and discharge valves. From the pumps, the water is routed through a 36-inch-O.D. header and a butterfly flow control valve to the reactor. The water reenters the reactor through two 24-inch lines.

A side line between the heat exchangers and primary pumps connects the PCS to a surge tank designed to absorb minor system pressure fluctuations. The primary system incorporates other components for pressure control, degassing, and demineralization.

Emergency flow is provided by two 4700 gpm emergency pumps, one of which operates continuously while the reactor is at power. Their purpose is to circulate sufficient water through the heat exchangers to remove reactor decay heat following a loss of the primary pumps or during a normal shutdown. One pump motor uses commercial or diesel-electric power and the other a DC battery power supply.

The emergency and main primary pumps each have check valves located in their discharge lines. The emergency pump valves (CK-A-17 and CK-A-20)

and the main primary pump valves (CK-A-2, CK-A-5, CK-A-8, and CK-A-11) are the subjects of this report. Sectional views of the emergency pump, 14 inch, and primary pump, 20 inch, replacement check valves in their originally supplied condition are shown on Figures 2 and 3.

3. CHECK VALVE FUNCTIONS

Normal operating procedure requires that the main primary pumps start and stop with their respective discharge gate valves closed. The gate valves are motor operated and require approximately four minutes to open or close. The main function of check valves CK-A-2, CK-A-5, CK-A-8, and CK-A-11 is to limit backflow through the primary pumps in the event of pump failure or loss of commercial power. In the event of a one-pump failure during three-pump operation, the check valve of the stopped pump must close such that the piping system is not subjected to excessive pressure surges or waterhammer.

The function of the emergency pump check valves (CK-A-17 and CK-A-20) is to prevent primary coolant recirculation through the emergency pump piping and to open immediately, should the primary pumps fail.

Fig. 2 Sectional view of 14-inch check valve.

Fig. 3 Sectional view of 20-inch check valve.

4. OPERATING BACKGROUND

4.1 Initial PCS Operation

Construction of the PCS was completed in October 1964. Tests and inspections during initial operation revealed deficiencies in the design and construction of critical components of the PCS. The resulting investigations lead to modification of the heat exchangers, repair of pipe welds, redesign of the piping support system, and the rejection of 30 PCS valves due to casting deficiencies[1]. Included in the rejected valves were the emergency and primary pump check valves.

Prior to rejection of the valves, performance testing of the primary pump check valves was conducted with the assistance of a fluid hydraulics consultant. These tests indicated no valve deficiencies or adverse affects on the piping system[2].

4.2 Replacement Valves

Specifications were written by Idaho Nuclear Corporation, which placed orders for replacement of the 14-inch emergency checks, the 20-inch main checks, and the 20- and 24-inch pump suction and discharge gate valves. The check valves were received and installed in September 1967; the gate valves were not installed until February 1968.

Operation of the PCS without the discharge valves was studied and found to be feasible. Spool pieces were installed in place of the discharge valves, and ATR startup testing requiring the PCS was resumed in October 1967. A preliminary test was conducted to evaluate possible waterhammer effects resulting from a 1-of-3 pump cutoff with the new check valves. These tests were conducted with both dashpot damping valves wide open (minimum check valve damping). Waterhammer was apparent during the transient period of the test, and fairly large pressure surges developed from the check valve closure. The check valve disc was seated approximately five seconds after the motor trip.

To reduce the observed waterhammer, another 1-of-3 pump cutoff test was made, this time with the one-inch damping valve closed. The 1/2-inch damping valve remained wide open. The valve closing time (from motor trip) was increased to 15 seconds, and the previously experienced waterhammer and pipe deflection were virtually eliminated.

On November 18 and 19, 1967, portions of bolts attaching the 20-inch check valve disc lugs to the hinge pins were found in the reactor tank. Disassembly of the valves showed that the hinge pins and bushings had scored. It was assumed that torque values generated during this scoring were sufficient to shear the bolts. Three of the four valves had been operated; five of the six bolts were completely sheared with the sixth showing evidence of partial shearing.

During the repair of the check valves, consideration was given to the increased check valve closing time (15 seconds) resulting from a 1-of-3 pump cutoff and with the one-inch damping valve closed. With a 3-of-3 pump cutoff, as would occur during a commercial power outage, no driving force would be available to close the check valves, and the closing time would then be greater than 15 seconds. It appeared likely that emergency pump flow could bypass the reactor by flowing through the primary pumps and the partially open check valves. This flow was expected to be such that reactor core cooling requirements would not be satisfied during the critical period immediately after a scram.

Tests were conducted on December 11, 1967, to determine the emergency flow following the loss of power to the three operating pumps and with the check valve dashpot one-inch damping valves closed. As expected, the emergency flow did not meet the core cooling requirements established by the ATR Safety Analysis Report (SAR)[3].

Again on December 11, 1967, another 3-of-3 pump cutoff test was conducted with minimum damping, that is, both damping valves wide open on each check valve. Core coolant flow under these conditions was above the minimum requirements of the SAR. Since this damping valve setting was known to produce undesirable waterhammer, testing was continued to determine the setting of the damping valves that would prevent:

- (1) Deficient core cooling during a 3-of-3 pump cutoff
- (2) Waterhammer during a 1-of-3 pump cutoff.

With the one-inch damping valves set 1/2 open and the 1/2-inch valve wide open, emergency core coolant flow under a 3-of-3 pump cutoff was found to meet the SAR requirements. The amount of primary pipe deflection that would be produced, if a partially damped check valve was rapidly shut from a 1-of-3 pump cutoff, was unknown. With this setting, a 1-of-3 cutoff was made. The valve was observed to shut. Considerable noise and pipe vibration were observed. The check valve closure created a waterhammer of sufficient magnitude to damage some of the support hangers for the discharge pipe from the emergency pumps. Investigations utilizing nondestructive testing and stress analysis showed that no damage had occurred to the primary coolant piping as a result of the waterhammer.

During a subsequent startup of primary pump 670-M-9, it was discovered that check valve CK-A-11 was stuck partially open. Disassembly of the check valve indicated that the damping piston rod had seized in its guide bushing. This failure occurred in the 20th operational cycle of the valve. The piston rod had been Electrofilmed but was not nitrided during the corrective work previously mentioned. The components (hinge pins) affected by the recent repairs were in satisfactory condition.

Operation of the PCS was discontinued to allow further study of the bearing surface problem and to begin an intensive investigation of the PCS waterhammer and core cooling problem. The purpose of these investigations was to determine the design changes necessary to adequately improve the performance and reliability of the check valves. The scope of these investigations covered a wide range including:

- (1) A hydraulic analysis to define the waterhammer experienced in the PCS and to establish the required valve characteristics necessary to prevent waterhammer
- (2) Analytical work to determine the expected bearing loads
- (3) Redesign of the valves in order to meet the new requirements both mechanically and hydraulically
- (4) Analytical work to predict the modified valve performance
- (5) Development of test programs to verify the analytical calculations and to provide data which would predict impending valve failure.

II. DETAILED DESCRIPTION OF VALVES

1. COMPARISON OF ORIGINAL AND REPLACEMENT CHECK VALVE SPECIFICATIONS

Four 20-inch and two 14-inch replacement check valves were purchased under Idaho Nuclear Specification IN-P-1-ATR, Rev. 1. These valves were replacements for the original valves purchased by the construction contractor. The architect-engineer, under contract to the AEC, prepared the specifications for the original valves. The original specifications required that both the 20-inch and 14-inch check valves be furnished with an integral dashpot using primary coolant water to control movement of the piston in the dashpot cylinder. The specification for the replacement valves did not require a dashpot for the 14-inch check valves since the valve disc could be shut gently by stopping the emergency pump prior to startup of the primary pumps. In preparation of the specification for the replacement valves, consideration was given to the need for dashpots on the 20-inch check valves. It was decided that a dashpot was necessary to prevent slamming and fluttering of the discs.

Both the original and replacement check valve specifications contained the following design requirements:

- (1) Fluid: High purity demineralized water with a pH of 4.5 to 5.5 -- fluid is radioactive
- (2) Valve Design Pressure: 485 psig
- (3) Valve Design Temperatures:
 - Initial 70°F
 - Normal 130°F
 - Maximum 240°F
- (4) Valve Radiation Exposure: Total exposure is 5×10^7 rads at a dose rate of 500 rads per hour.

The specifications for the replacement check valves contained the following additional design requirements:

Differential Pressure

- (1) 14-inch valves
 - (a) Maximum pressure drop across the fully open valve shall be 5.0 psi at a flow of 4800 gpm.
 - (b) Valve shall be open at a differential pressure of 0.29 psi across the seat.

(2) 20-inch valves

- (a) Maximum pressure drop across the fully opened valve shall be 7.5 psi at a flow of 18,000 gpm.
- (b) Valve shall be fully open at a differential pressure of 1.21 psi across the seat.

Stability of Discs

No detectable disc flutter shall occur at the following system flow ranges:

- (1) 20-inch valves: with a differential pressure across the seat above 1.21 psi.
- (2) 14-inch valves: with a differential pressure across the seat above 0.29 psi.

Seat Leakage

Leakage across the seat shall be less than 2 gpm with a differential pressure of 225 psi across the valve.

The original and replacement check valve specifications also differed somewhat with regard to specification of materials. The specifications for the original check valves stated, "Wetted surface of the seat ring shall be nitrided to prevent galling". The specifications for the replacement check valves stated, "Seating surfaces of seat ring and other wear parts shall be nitrided or faced with other approved hard surfacing alloys to prevent galling".

2. DESCRIPTION OF REPLACEMENT CHECK VALVES BEFORE MODIFICATIONS

As shown in Figures 2 and 3, the replacement check valves are 300-pound, inclined seat, tilting disc, with inlet and outlet connections horizontal and on the same center line.

The check valves are held open by the hydraulic forces acting on the disc. Stops on the inside of the valve limit the fully open position to six degrees from horizontal. The disc closed position is 55 degrees from horizontal. The closing force for the 20-inch check valve is provided by the compressed spring in the dashpot and the weights of the disc, piston, and piston rod. The weight of the disc alone provides the closing force for the 14-inch check valves. During the pump coastdown period, hydraulic forces developing as a result of flow reversal assist valve closure.

The 14- and 20-inch check valves contained the following areas which were subjected to wearing. These areas are given by the following combinations:

20-inch Check Valves

- (1) Piston - dashpot cylinder
- (2) Piston rod - piston rod bushings
- (3) Link pins - link and clevises
- (4) Hinge pins - hinge pin bushings
- (5) Disk - seat ring

14-inch Check Valves

- (1) Hinge pin - hinge pin bushings
- (2) Disk - seat ring

Tables I and II furnish critical dimensions and materials of construction of the 20- and 14-inch check valves, respectively.

The nominal manufactured clearance of 0.025 inch between each hinge pin and hinge pin bushing was necessary to allow operating movement between the seat and disc sufficient for longitudinal translation of the disc upon opening or closing.

The dashpot furnished on the 20-inch check valves provides control of valve disc movement. The spring in the dashpot exerts a 100-pound force on the piston when the valve is in the fully open position. Two separate lines, one from the lower portion of the dashpot cylinder and the other from the cylinder base, are connected to the top of the dashpot. The lines provide a means of bleeding the dashpot fluid from one side of the piston to the other during movement of the piston. One of these lines is a one-inch line and the other is a 1/2-inch line. The one-inch line contains a one-inch globe valve and the 1/2-inch line contains a 1/2-inch needle valve. These valves provide a means of adjustment to acquire the desired damping by the dashpot. The lower inlet of the one-inch line is closed off by the piston when the valve disc is approximately 80% closed. This provides greater damping for the remainder of piston travel during final closure of the check valve. One other line, of 1/2-inch nominal size, connects the top of the cylinder to the valve body. The purpose of this line, which is not valved, is to make up water volume changes in the cylinder due to translation of the piston rod into and out of the cylinder.

TABLE I
20-INCH CHECK VALVE
DIMENSIONS AND MATERIALS OF CONSTRUCTION

Item	Critical Dimension (inches)	Material
Piston	Dia. 11.996 ^{+0.000} -0.003	ASTM A-351 Grade CF8
Dashpot Cylinder	Bore 12.003 ^{+0.003} -0.000	ASTM A-351 Grade CF8
Piston Rod	Dia. 2.500 ^{+0.000} -0.010	ARMCO 17-4 PH 341 BHN (H 1075)
Piston Rod Bushings	Bore 2.503 ^{+0.005} -0.000	ARMCO 17-4 PH 380 BHN (H 975)
Link Pins	Dia. 1.500 ^{+0.000} -0.005	ARMCO 17-4 PH 341 BHN (H 1075)
Link	Bore 1.505 ^{+0.005} -0.000	AISI #304
Piston Rod Clevis	Bore 1.505 ^{+0.005} -0.000	ASTM A-296 Grade CF8
Disc		ASTM A-351 Grade CF8
Seat Ring		AISI #316 Stellite Faced
Hinge Pins	Dia. 3.100 ^{+0.000} -0.005	ARMCO 17-4 PH 341 BHN (H 1075)
Hinge Pin Bushings	Bore 3.125 ^{+0.005} -0.005	ARMCO 17-4 PH 380 BHN (H 975)

TABLE II
14-INCH CHECK VALVE
DIMENSIONS AND MATERIALS OF CONSTRUCTION

Item	Critical Dimension (inches)	Material
Disc		ASTM A-351 Grade CF8
Seat Ring		AISI #304 Stellite Faced
Hinge Pins	Dia. 2.000 ^{+0.000} -0.005	ARMCO 17-4 PH 341 BHN (H 1075)
Hinge Pin Bushings	Bore 2.025 ^{+0.005} -0.000	ARMCO 17-4 PH 380 BHN (H 975)

3. BASIS FOR MATERIAL SELECTION OF REPLACEMENT VALVES

Prior to procurement of the replacement valves, Idaho Nuclear Engineering personnel made studies of a variety of surface conditions existing on the nitrided (Malcomized) parts of the original valves. It appeared that the nitriding process had not been uniformly controlled. This resulted in blistered surfaces on some of the pistons and valve seats, and some of these areas contained rust which flaked off during service. In other areas, the nitrided surfaces were found to be uniform and rust free, indicating that a properly controlled nitriding process had been employed. In view of this occurrence, Idaho Nuclear concluded that an anti-galling surface should be provided by some means other than nitriding. Therefore, the valve vendor proposed the 17-4 PH combinations and these were approved.

During final review of the vendor design drawings, consideration was given to the materials specified for the wearing parts. It was decided that the combinations proposed had adequate antigalling properties. The hinge pin and hinge pin bushings, which were both made of Armco 17-4 PH stainless steel, had different heat treatments. This provided a desired hardness differential which gives Armco 17-4 PH stainless steel greater galling resistance. In addition to this, it was specified that these parts be Electrolyzed, a proprietary hard chrome plating.

III. GALLING OF WEARING SURFACES

1. GALLED HINGE PINS AND BUSHINGS

On November 18 and 19, 1967, portions of the 3/8-inch-diameter bolts shown in Figure 4, which attach the check valve disc to the hinge pins, were found in the ATR vessel. Upon disassembly of the check valves, it was discovered that the hinge pins and bushings had galled, as shown in Figures 5, 6, and 7. These mating surfaces had galled to the extent that enough torque was generated to shear the 3/8-inch-diameter bolts.

The diametrical clearance between the hinge pins and hinge pin bushings provides a theoretical line contact between the parts. Because of this small contact surface, the hydraulic forces exerted on the valve disc caused a calculated high bearing stress of 24,100 psi on the hinge pins and bushings. See Appendix A for stress calculations based on static conditions. The corresponding galling stress for 17-4 PH with a hardness of 341 BHN (H 1075) against 17-4 PH with a hardness of 380 BHN (H 975) is 7000 to 10,000 psi^[4].

Fig. 4 Sheared 20-inch check valve disc bolts.

Fig. 5 20-inch check valve CK-A-11 disc lug as disassembled.

Fig. 6 Enlarged view of 20-inch check valve CK-A-11 hinge pin bushing as disassembled.

Fig. 7 Galled hinge pins of 20-inch check valve CK-A-11 as disassembled.

2. REPAIR OF HINGE PINS AND HINGE PIN BUSHINGS

The galled surfaces on both the hinge pin bushings were honed to remove all raised and sharp edges. Various platings were examined to determine which one could be used on the hinge pin surfaces to give the most desirable characteristics. The valve vendor was directed to stellite face a test piece of 17-4 PH material. This test showed the stellite material cracked upon cooling; therefore, this method was abandoned. It was then decided that a desirable hardness differential could be achieved by nitriding the hinge pins. In addition, the hinge pins were coated with Electrofilm #4396, a solid film lubricant.

Nitriding and Electrofilming were selected for the following reasons:

- (1) Nitriding increases the surface hardness to approximately 900 BHN, providing a hardness difference of greater than 500 BHN. If nitriding and service conditions (surface preparation, microstructure, atmosphere control, corrosion, case depth, core hardness) are proper, the load capacity without galling should exceed 21,000 psi[4].
- (2) Consideration of the above showed that with a nitrided depth of 0.010 to 0.015 inch and a core hardness of Rc 50, spur gears can sustain a continuous Hertz (contact) stress of 150,000 psi[5].
- (3) It was known that nitriding the surface would reduce to a slight degree the corrosion resistance of the parts. Electrofilm was added to the nitrided surface to help provide compensating corrosion resistance.

3. SUCCESSFUL OPERATION OF NITRIDED HINGE PINS

It was decided that after the 20-inch check valves were reassembled, one valve would be tested 20 cycles and would then be disassembled and inspected. The galling of other wearing parts (namely the piston rod and piston bushing of check valve CK-A-11 which occurred during the December 11, 1967, waterhammer incident on the 20th cycle) made it necessary to disassemble all four of the 20-inch check valves in order to nitride wear surfaces of the piston assembly and linkage. After disassembly of the 20-inch check valves, a complete visual inspection of the hinge pins and hinge pin bushings of all four valves showed no evidence of new galling.

The ability of the nitrided hinge pins to resist galling while subject to the high bearing loads as a result of the 20th-cycle waterhammer is significant. The severity of the loads was indicated by the elongation of the pin holes of the connecting link of valve CK-A-11, one in the amount of 0.030 inch and the other in the amount of 0.050 inch. Because nitriding appeared to be satisfactory, the treatment was applied to the remaining wear surfaces in the 20-inch check valves. The surfaces treated were:

- (1) piston
- (2) piston rod
- (3) link pins.

The hinge pins of the 14-inch check valves also were nitrided.

On February 27, 1967, check valve CK-A-2 was disassembled for inspection after it had accumulated 109 cycles. This examination consisted of:

- (1) Visual examination of wearing surfaces
- (2) Comparison of measurements with measurements previously taken
- (3) Liquid penetrant examination of nitrided surfaces.

The examination was conclusive in that there was no evidence of corrosion or new galling on any of the wearing surfaces. The visual and liquid penetrant examinations showed no evidence of surface deterioration. Measurements obtained of critical dimensions of wearing parts also gave no indication of surface wear. Figures 8 and 10 show the hinge pins of check valve CK-A-2, as removed after initial galling. Figures 9 and 11 show the same hinge pins which were nitrided and then reassembled and subjected to 109 cycles of opening and closing. It is seen that these hinge pins did not receive any additional galling or surface deterioration as a result of this cycling.

Fig. 8 Galled hinge pins of 20-inch check valve CK-A-2 as disassembled.

Fig. 9 Repaired hinge pins of 20-inch check valve CK-A-2 after 109 cycles of operation.

Fig. 10 Galled hinge pins of 20-inch check valve CK-A-2 as disassembled -- rotated 180 degrees.

Fig. 11 Repaired hinge pins of 20-inch check valve CK-A-2 after 109 cycles of operation -- rotated 180 degrees.

IV. WATERHAMMER ANALYSIS AND REACTOR CORE COOLANT REQUIREMENTS

The purpose of the waterhammer investigation was twofold. First, it was necessary to analytically verify the effects of the waterhammer as observed in the PCS piping. Secondly, it was necessary to determine the system requirements (with respect to the check valves) that must be met in order to eliminate waterhammer and still provide sufficient core cooling during emergency situations.

Arrangement of two or more pumps in parallel, having discharge check valves, is liable to produce waterhammer when some of the pumps are cut off. As the cutoff pump loses speed, its flow decreases to zero and reverses. If this reverse flow (backflow) is then suddenly stopped, as by a check valve, pressure surges develop and waterhammer results. The severity of the waterhammer depends on the following:

- (1) Check valve closing time relative to the return time of a waterhammer wave from the nearest reflection point
- (2) Fluid velocity, or the head of the pumps available to produce backflow
- (3) Pipe modulus, geometry, and length
- (4) Surge suppressors present in the piping system
- (5) Entrainment of air in the flowing fluid.

The analysis of the waterhammer of December 11, 1967, was based on previously measured characteristics of the system. The backflow through a pump for which power is lost in a 1-of-3 pump cutoff had been established in previous tests. Pressures on each side of the check valves during various closures, including the waterhammer of December 11, were used to establish the waterhammer characteristics of the downstream pipe section and to determine other characteristics of the system. Figure 12 shows the measured backflow through a secured pump during a 1-of-3 cutoff. The backflow behavior is terminated by the closure of the check valve. The effective portion of the valve closure with attendant flow stoppage can be seen to be very abrupt.

The portion of the backflow curve which is unaffected by the movement of the check valve shows a very rapid reversal of flow through the pump. This is a result of the high head of the other operating pumps. The behavior, shown in Figure 12, agrees well with the fundamental characteristics of the pump and piping system.

1. WATERHAMMER ANALYSIS

The analysis of the December 11, 1967, waterhammer was accomplished using data obtained during the incident and from previous less severe waterhammers. This permitted criteria for successful check-valve operation to be

Fig. 12 Measured flow through a secured pump during a 1-of-3 pump cutoff.

established. Figure 13 shows the location of the two pressure transducers in service at the start of the December 11, 1967, test. Figure 14 shows the pressure traces obtained from these two transducers. The pressure pulse measured on the upstream side of the check valve was sufficient to overrange and damage the transducer so that the maximum pressure was not obtained. The pressure must be traced before the positive surge is valid. The transient pressures at Test Point 2 are not the same as the pressure at the downstream side of the check valve because of intervening changes in pipe section and branch connections. No closer transducer location was available, without welding on the system. Using the equations for the transmission and reflection of pressure waves in a compressible fluid and the transducer test data, the pressure at the check valve was determined^[6]. This calculation, which is shown in Appendix C, indicated that the waterhammer waves originating at the check valve were reduced to 27% of their original value at the transducer measurement point. The maximum overpressure (above steady state pressure) recorded at Test Point 2 gives a maximum overpressure of 410 psi at the downstream side of the

Fig. 13 Location of pressure transducers during check valve testing.

Fig. 14 Pressure up and downstream of check valve CK-A-11 during December 11, 1967, water-hammer incident.

check valve. A valve closure in less time than the period of travel of a wave from the valve to and back from a reflection point in the system would have resulted in the maximum waterhammer pressure, the so-called instantaneous closure value. For the backflow at the time of the actual closure, the overpressure for "instantaneous closure" is 700 psi. This value is determined from the equation^[6].

$$\Delta H = \frac{a \Delta V}{g}$$

where

ΔH = the overpressure

a = the propagation velocity of a pressure wave in the fluid in the pipe

ΔV = the change in water velocity before the return of the first reflected wave (pipe period)

g = the acceleration of gravity in consistent units.

The period of a pipe of length "L" is the time required for a pressure wave traveling at the propagation velocity "a" to traverse the pipe and return, and is $\frac{2L}{a}$ in appropriate time units. The period of the pipe downstream of the check valve is 0.01 sec. Thus, a 10-msec closure of the check valve would be necessary to produce the maximum waterhammer value on the downstream side of the valve. The actual closure time of the valve, over the part of its stroke which is significantly effective in stopping flow, is about 100 msec. Multiple reflections, and the suddenness of the velocity change in the last few pipe periods of closure, account for the fact that the overpressure reached 58% of the theoretical maximum value. The downstream hammer was not excessively severe and is of interest primarily as a key to the behavior upstream of the valve.

The highest pressures occurred on the upstream side of the check valve where the fluid behavior is more complex. First, consider what would occur if waterhammer theory had been valid on this side of the valve. The period of the pipe on the upstream side is much the same as on the downstream side. The first reflection point is at the pump, and the period of the intervening pipe is 0.011 sec. The velocity change per pipe period is on the order of 10% greater than on the downstream side, and the pressures should be similar on both sides of the valve. Since the water is flowing away from the valve rather than toward it, the pressure changes on the upstream side are of the opposite sign. The pressure recording at Test Point 1 in Figure 14 should have exhibited the reverse of the behavior of Test Point 2, that is, a damped pressure oscillation, starting with a negative surge. The first negative surge does indeed agree with the first surge on the downstream side when correction for downstream reflections are made. The upstream initial surge dropped from 370 to a recorded 10 psig. This measured 360 psi pressure surge would have continued to an underpressure of 410 psi except that the vapor point of the water was reached. Had this not happened, the next half-cycle of pressure would have been an overpressure of about 410 psi, and the same damped pressure oscillation as at Test Point 2 would have ensued.

When the vapor point of the water was reached, a phenomenon known as water column separation occurred. The water column on the upstream side of the check valve continued to move away from the valve, leaving a water vapor bubble at the valve. Water vaporization may have occurred at other points in the pipe where the pressure reached the bubble point. Because of the very high velocity jet passing through the nearly closed check valve, and the consequent low pressure in the jet, vaporization should have occurred, preferentially at the check valve. In the presence of only slight damping, the kinetic energy of the separated column is absorbed by vaporization into the bubble. The stored energy in the bubble is almost fully converted back to kinetic energy of the water as the separated column accelerates back toward the valve, so that the separated column strikes the valve with velocity it had before the valve closure. The collapse of the vapor cavity at the valve is essentially instantaneous. Water column separation therefore caused the instantaneous-closure waterhammer value to be achieved, and the pressure wave generated by the separated water column striking the valve is governed by the equation on page 23. Although the pressure transducer at Test Point 1 was overranged by the positive pulse caused by the vapor bubble collapse, the overpressure upstream of the check valve is calculated to have reached 710 psi. In the center of collapsing bubbles, of course, very high cavitation-type pressures existed, but the average pressure in the pipe should have reached a maximum of about 900 psig. The damaged pressure transducer at Test Point 1 had a dynamic load rating of 1000 psig. The damage to the transducer is attributed to a local bubble collapse pressure in the transducer impulse line or very near its entrance.

The presence of dissolved gas in the water at the time of the large waterhammer permitted water separation to begin at a higher pressure. The bubble point pressure, at which water column separation begins, is shown in Figure 15 along with the waterhammer underpressure which gives the bubble point pressure for an initial pressure of 400 psia, and the velocity which results in this pressure for instantaneous closure in ATR piping. The figure is based on radiolytic gas. Air produces the same bubble point at $4/7$ the concentration of a radiolytic gas. The air concentration in the water during this test was unknown, but it was believed to have been high, so that water column separation initiated more easily in the incident. Noncondensable gas in the vapor bubble tends to cushion the bubble collapse, but as long as the collapse occurs before the return of the first reflected wave, the maximum waterhammer still results. This obviously occurred.

The presence of dissolved gas in the ATR primary cooling water was taken into account in setting valve performance criteria which would avoid water column separation.

1.1 Waterhammer Criteria for Future Operation

The knowledge of the waterhammer behavior of the PCS for a 1-of-3 pump cutoff as determined from measurement and analysis was used to determine safe criteria for check valve response which would prevent damaging waterhammer and ensure no loss of plant life.

As previously mentioned, a criterion for adequate core cooling is that the check valve must close fast enough to prevent loss of core coolant as a result

Note

For air, this curve may be used by multiplying air Concentration by 7/4 and entering Curve with the value so obtained.

Fig. 15 Waterhammer versus gas concentration.

of excessive backflow. The severe December 11, 1967, waterhammer occurred with a valve adjustment which gave the slowest allowable valve action for adequate core cooling. This dictated that the solution of the waterhammer problem must be a speeding of valve action. It was assured thereby that any further speeding of the valve action would improve the core cooling above the minimum requirement.

The allowable valve action criterion was based on the backflow in the most severe pump failure, the 1-of-3 cutoff, which is shown in Figure 12. The other element of the criterion was the waterhammer for an instantaneous valve closure which could be tolerated by the system. The criterion requires that the valve close in sufficient time to a point such that the flow which gives an acceptable waterhammer would never be exceeded, even if no further closure took place. The maximum allowable flow also can be seen in Figure 15 to be 6600 gpm, the maximum flow for which no column separation occurs. It may be noted in Figure 12 that this flow is reached in 2.5 seconds. The valve must be closed to 17 degrees open to hold the flow to 6600 gpm indefinitely. In summary, then, the criterion for successful valve operation was that the valve must be closed to the 17 degree open position in 2.5 seconds. Thereafter, the valve closure may proceed in any fashion without damaging the system. This criterion is quite conservative since it considers as a basis the instantaneous closure value

of waterhammer as an indication of whether water column separation occurs, whereas the waterhammer surge which precedes the water column separation is about 60% of the instantaneous closure value. There is, therefore, a margin of about 67% above the maximum flow the criterion allows.

The actual performance of the check valve far exceeded the requirements of the criterion, as noted in Section VI of this report.

2. CORE COOLING REQUIREMENTS DURING FLOW COAST-DOWNS AS RELATED TO CHECK VALVE ACTION

The "flow required" curve in Figure 16 is obtained from the ATR Safety Analysis Report^[3] and shows the requirements for core cooling in a 3-of-3 pump loss-of-flow accident.

The power curve shows the start of the reactor power drop resulting from a scram. The "flow required" curve shows the flow which is necessary to maintain the hot-stripe three-standard deviations (on hot-stripe bulk-water temperature) from burnout. After a second, the "flow required" curve decreases

Fig. 16 Core cooling required following a power loss to all operating primary pumps.

slowly as reactor decay heat decreases. The "measured flow" curve should asymptotically approach 9% of full flow, the steady state emergency flow. To supply this flow, the primary pump check valves must be closed before primary pump head drops low enough to permit significant amount of emergency flow to be diverted back through an open check valve. The "measured flow" curve shown in Figure 16 demonstrates the effect of slow check valve action. It can be seen that for the 3-of-3 pump cutoff, the check valves close after 38 seconds and that there is backflow from about 22 to 38 seconds. This flow curve was considered to be the minimum which would assure meeting the SAR required flow. With the check valve damped to produce this flow curve, the 1-of-3 pump cutoff test which resulted in the large waterhammer of December 11, 1967, was performed. Previous tests had shown that slower check valve action produced satisfactory waterhammer. Because of core cooling requirements, however, only faster valve action could satisfy both waterhammer and core cooling criteria.

V. REDESIGN OF 20-INCH CHECK VALVES

1. LIST OF CHANGES AND REASONING BEHIND THEM

The closing times of the 20-inch check valves were decreased by modifying the check valve in the following four ways:

- (1) Removed globe valve from one-inch line on dashpot cylinder
- (2) Chamfered one-inch bypass line penetration in the cylinder wall approximately 1/2 inch by 45 degrees
- (3) Blocked the valve disc so the fully open position of the disc was changed from 6 to 25 degrees from the horizontal
- (4) Increased spring load in dashpot cylinder by mounting the spring on a spacer device above the piston.

Removal of the globe valve from the one-inch bypass line on the dashpot cylinder reduced the hydraulic resistance in dashpot water flow during check valve closure. The effect on valve closure can be shown for a steady condition by application of Bernoulli's equation as follows:

For a given driving force acting on the dashpot piston, there exists a corresponding proportional pressure differential (ΔP) across the piston. With this given ΔP , the velocity of fluid in the bypass line can be determined from

$$\frac{\Delta P}{\gamma} = \frac{KV^2}{2g} + f \frac{\ell}{d} \frac{V^2}{2g} + f \left(\frac{\ell}{d} \right)_{eq} \frac{V^2}{2g} + \frac{V^2}{2g}$$

where

$$\frac{\Delta P}{\gamma} = \text{change in pressure head between bypass line entrance and exit}$$

$$\frac{KV^2}{2g} = \text{entrance loss}$$

$$f \left(\frac{\ell}{d} \right) \frac{V^2}{2g} = \text{line loss}$$

$$f \left(\frac{\ell}{d} \right)_{eq} \frac{V^2}{2g} = \text{line loss due to globe valve and elbows}$$

$$\frac{V^2}{2g} = \text{exit loss}$$

K = entrance loss coefficient = 0.5 for unchamfered entrance [7]

V = velocity of fluid

l = length of pipe = 35 inches

d = diameter of pipe = 1 inch

f = friction factor = 0.02

$\left(\frac{l}{d}\right)_{eq}$ = equivalent length of pipe in pipe diameters

= 340 for conventional globe valve^[7]

= 75 for two 90-degree elbows and one 45-degree elbow^[7].

Therefore,

$$\frac{\Delta P}{\gamma} = \frac{0.5V^2}{2g} + 0.02 \frac{35}{1} \frac{V^2}{2g} + 0.02 (415) \frac{V^2}{2g} + \frac{V^2}{2g}$$

$$\frac{\Delta P 2g}{\gamma} = 0.5 V^2 + 0.7 V^2 + 8.3 V^2 + V^2$$

$$\frac{\Delta P 2g}{\gamma} = 10.5 V^2$$

$$V = \sqrt{\frac{\Delta P 2g}{\gamma} \frac{1}{10.5}} = (0.3085) \sqrt{\frac{\Delta P 2g}{\gamma}}$$

For the case of the bypass line with the globe valve removed, the velocity of the fluid in the bypass line can be obtained using the same equation only with the omission of line loss due to the globe valve.

$$\frac{\Delta P}{\gamma} = \frac{0.5 V^2}{2g} + 0.02 \frac{35}{1} \frac{V^2}{2g} + (0.02) (75) \frac{V^2}{2g} + \frac{V^2}{2g}$$

$$\frac{\Delta P 2g}{\gamma} = 0.5V^2 + 0.7V^2 + 1.5V^2 + V^2$$

$$V = \sqrt{\frac{\Delta P 2g}{\gamma} \frac{1}{3.7}}$$

$$V = (0.52) \sqrt{\frac{\Delta P 2g}{\gamma}}$$

Thus, it is seen that the velocity of fluid in the bypass line with the globe valve removed is approximately 70% faster than it was with the globe valve.

$$\frac{(0.52) \sqrt{\frac{\Delta P 2g}{\gamma}}}{(0.3085) \sqrt{\frac{\Delta P 2g}{\gamma}}} = 1.685$$

Since the piston velocity is proportional to the velocity of the fluid in the bypass line, it also is increased by the same ratio.

Chamfering the one-inch bypass pipe penetration in the cylinder wall reduced the check valve closure time in two ways. First, it lowered the hydraulic entrance losses to water flowing through the line by reducing the entrance loss coefficient "k" from 0.50 to 0.20^[8]. Secondly, the chamfer permitted an additional 1/2 inch of piston travel before the entrance to the one-inch bypass line is completely closed by the piston.

Changing the fully open disc position from 6 to 25 degrees from the horizontal reduced the closure time by reducing piston travel from 5.4 to 3.1 inches. It was realized that this would increase the pressure drop across the check valve from normal operating flow (14,700 gpm) from 1.5 to about 6.0 psi. This was considered tolerable. This new fully open disc position was achieved by installing a piston stop device above the piston and around the spring in the dashpot such that it would contact the top of dashpot when the valve reached the new fully open position.

The spring load was increased by mounting the spring on a spacer incorporated inside the piston stop device. Since the force from the compressed spring is one of the closing forces, this increased force would cause the check valve to close faster. The spring was compressed such that the combined spring force and weight of the piston and piston rod would be approximately 500 pounds less than the vertical force applied to the piston rod clevis during normal flow. This was desired for two reasons. First, the check valve would remain in the fully open position and not flutter during normal flow. Secondly, the check valve would commence to close almost immediately after loss of pump power. Figure 17 shows the 20-inch check valve after modification.

2. ANALYTICAL WORK TO PREDICT MODIFIED VALVE PERFORMANCE

A dynamic analysis was performed to predict the modified 20-inch check valve closing time for each of the following four cases:

- (1) 1-of-1 pump cutoff
- (2) 3-of-3 pump cutoff
- (3) 2-of-3 pump cutoff
- (4) 1-of-3 pump cutoff.

Each analysis was made in two parts. The first part considered valve closure from the fully open position to the point at which the one inch bypass line was nearly closed by the piston. The second part considered the remainder of the closure.

The closing time for the first part was determined using a finite difference technique over small time increments. The velocity of the piston was assumed

Fig. 17 Sectional view of modified 20-inch check valve.

for the end of the time increments. Based on the assumed velocity, the forces acting on the dashpot piston and piston rod were determined. The resultant of these forces times the piston travel for the time increment gives the work done on the piston and piston rod for the time increment. The velocity of the piston at the end of the time increment was determined by equating the work done on the piston and piston rod to the change in kinetic energy of the piston and piston rod. This calculation was repeated using a new assumed velocity until the calculation and assumed velocities were in agreement.

It was not possible to use the finite difference technique for the remaining part of valve closure since the forces acting on the check valve disc were not known to the required accuracy. It was assumed for the remainder of closure that the piston traveled at a constant velocity because of an average closing force. This constant velocity was determined by equating the assumed average closing force to the damping forces of the dashpot.

The results of this analysis indicated that the modified check valves would close fast enough to prevent waterhammer and also to assure adequate reactor core cooling. Sample calculations, determining the check valve closing time for a 1-of-3 pump cutoff, are shown in Appendix C.

VI. TEST PROGRAM TO VERIFY THE ADEQUACY OF THE MODIFIED CHECK VALVES

1. DESCRIPTION OF TEST PROGRAM

As described herein, check valve design criteria were determined from the waterhammer and core coolant analysis, a redesign was established based on this criteria, and calculated predictions of the modified valve closing times were made. To verify the accuracy of the calculations and the ability of the modified valves to meet the new criteria, a series of performance tests was necessary.

Insofar as could be determined by the analytical work, the only plant condition which contained any element of risk was the 1-of-3 pump cutoff situation. Therefore, this shutdown condition was gradually approached through the following sequence of tests:

Test No. 1 -- 1-of-1 pump cutoff

Test No. 2 -- 3-of-3 pump cutoff

Test No. 3 -- 2-of-3 pump cutoff

Test No. 4 -- 1-of-3 pump cutoff

For each of the above tests, the 1/2-inch damping valve was arbitrarily set 1/2 open.

The PCS piping and check valves were extensively instrumented to provide data necessary to evaluate the effect of each transient condition on the piping system. Thirty-five signals from devices sensing flow, pressure, strain, and deflection were recorded instantaneously on light-galvanometer type oscillographs and magnetic tape. After each test, the data were assessed and compared to analytical predictions and previously established criteria before proceeding to the next test.

2. TEST RESULTS

The tests were successfully completed. These indicated that the modified check valves met the two PCS requirements, that is, prevention of waterhammer and assurance of adequate core cooling under each pump shutdown tested. Results of these tests are presented in the form of curves and tables on the following pages.

Figures 18, 19, 20, and 21 provide a comparison of measured versus calculated valve closing times for the four shutdown situations. These curves

Fig. 18 Measured versus calculated valve response during a 1-of-1 pump cutoff.

Fig. 19 Measured versus calculated valve response during a 3-of-3 pump cutoff.

Fig. 20 Measured versus calculated valve response during a 2-of-3 pump cutoff.

Fig. 21 Measured versus calculated valve response during a 1-of-3 pump cutoff.

indicate that the modified valves performed as expected and that the calculation method described in Section 5 was adequate for design purposes.

At no time did the recorded data or visual observation ever indicate or predict (for subsequent tests) abnormal or dangerous system conditions.

Figure 22 shows the line pressure fluctuations upstream and downstream of the check valve resulting from the most severe shutdown situation, a 1-of-3 pump cutoff. The reduction of the pressure surges with the modified valve can be seen by comparing Figure 22 with Figure 14 used in the waterhammer discussion.

Fig. 22 Pressure up and downstream of a modified 20-inch check valve during a 1-of-3 pump cutoff.

Strain levels, reflecting the effect of internal pressure and pipe deflection, decayed steadily after motor trip and with only minor oscillations occurring as the check valve disc was seated. The oscillations, as expected, were greatest during the 1-of-3 pump cutoff test; Figure 23 represents typical recorded traces of strain and deflection signals during this transient period. Stresses were calculated in the vicinity of Test Points 1, 2, and 3 and are tabulated below Figure 23.

Pipe deflection was measured in vertical runs of the 20-inch primary pump and 14-inch emergency pump discharge lines. Measurements were made along two horizontal axes, 90 degrees apart, referred to as North-South and East-West. Maximum peak-to-peak deflections for the 1-of-3 pump cutoff situation also are tabulated below Figure 26.

With the modified valve, the emergency reactor core coolant flow was maintained above the minimum requirements established by the ATR Safety Analysis Report. Measured versus required emergency core coolant flow is shown in Figure 24.

MAXIMUM PEAK TO PEAK DEFLECTIONS—ONE OF THREE PUMP CUTOFF:

20" Pri. Pump Disch. Line:	North-South — 0.26 Inches
	East-West — 0.14 Inches
14" Emerg. Pump Disch. Line:	North-South — 0.02 Inches
	East-West — 0.05 Inches

MAXIMUM STRESSES AT POINTS ①, ②, ③ ABOVE—ONE OF THREE PUMP CUTOFF:

20" Pri. Pump Disch. Line:	① 9590 psi
	② 5660 psi
	③ 7120 psi
14" Emerg. Pump Disch. Line:	① 11,150 psi
	② 10,200 psi
	③ 9,230 psi

YIELD STRENGTH, ASTM 376-TP 304: 28,680 psi @ 130° F.

Fig. 23 Deflection and strain occurring in primary coolant piping during a 1-of-3 pump cutoff.

Fig. 24 Measured core flow following a power loss to all operating primary pumps.

3. RESULTS OF ADDITIONAL CHECK VALVE TESTS

The similarity of each modified check valve was determined by subjecting each valve to several identical shutdown situations and comparing the closing times of each valve. Closing times for identical situations never differed between any two valves by more than 16%, indicating reasonable similarity.

The effect of the 1/2-inch damping valve on the performance of the modified check valve was determined by subjecting a check valve to duplicate 1-of-3 pump cutoffs, once with the 1/2-inch valve closed and once with the 1/2-inch valve wide open. The closing time with the 1/2-inch valve closed was only 3% longer than with the 1/2-inch valve wide open. However, the differential pressure measured across the damping piston was increased approximately 20% with the 1/2-inch valve closed. This indicated that for reduced bearing loads, it was preferable to set the 1/2-inch damping valves wide open.

The 1-of-3 pump cutoff tests were repeated after each check valve had accumulated approximately 100 cycles of operation. These tests indicated that the valve closing times had not changed during this period of operation.

VII. CONCLUSIONS AND RECOMMENDATIONS

The necessity of the check valves in the ATR primary coolant system and the function of these valves have always been recognized. However, it is apparent from the preceeding discussions that inadequate criteria were furnished for the proper design and/or selection of these valves.

The authors conclude that the function of a check valve should be defined with respect to the specific requirements of the system in which it will be used. Based on this conclusion, the following recommendations for the selection or design of a check valve are made:

- (1) Perform a waterhammer analysis considering all possible system and valve operating modes, and from this analysis establish the valve response characteristics necessary to prevent waterhammer.
- (2) Determine the emergency flow requirements of the system's dependent equipment to ascertain that the valve(s) action will not compromise these requirements.
- (3) When the check valve design is established, based on the system requirements, calculation of the valve bearing loads should be made and used as a criteria in the selection of bearing materials.
- (4) Test programs should be scheduled to verify the required valve performance, and in those applications where failure of the check valve would endanger personnel or equipment, instrumentation should be installed to predict deterioration of the valve performance.

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APPENDIX A

CONTACT STRESS CALCULATIONS

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APPENDIX A

CONTACT STRESS CALCULATIONS

Two cylindrical bodies with different radii have approximately a line contact before loads are applied^[9]. The area of contact becomes a narrow rectangle of width $2b$ after a load of q pounds per unit length is applied. The value of b is given by

$$b = \sqrt{2q \Delta / \pi}$$

in which q is the load per unit length of the contact area. The value of Δ is given by

$$\Delta = \frac{1}{(1/2R_1) + (1/2R_2)} \left[\frac{1-\mu_1^2}{E_1} + \frac{1-\mu_2^2}{E_2} \right].$$

The maximum principal stress is given by

$$\sigma = \frac{b}{\Delta}$$

where

R_1 and R_2 = radii of curvature of the cylindrical surfaces as shown

σ = principal contact stress

E_1 and E_2 = Moduli of elasticity for bodies 1 and 2

μ_1 and μ_2 = Poisson's ratio for bodies 1 and 2

$$2R_2 = 3.125 \text{ inch}$$

$$2R_1 = 3.100 \text{ inch}$$

$$\mu_1 = \mu_2 = 0.3 \text{ in./in.}$$

$$E_1 = E_2 = 29 \times 10^6 \text{ psi}$$

$$\Delta = \frac{1}{\frac{1}{3.100} - \frac{1}{3.125}} \times \left[\frac{1 - (.3)^2}{29 \times 10^6} + \frac{1 - (.3)^2}{29 \times 10^6} \right]$$

$$\Delta = \frac{1}{0.32258 - 0.3200} \times \left[\frac{2 (0.91)}{29 \times 10^6} \right]$$

$$\Delta = \frac{2 (0.91)}{(0.00258) 29 \times 10^6} = 2.43 \times 10^{-5}$$

Assumed Load Conditions

$$\frac{4.5(4)}{2} = P$$

$$q = \frac{2P}{4.5}$$

$$q = 0.445P$$

Assumed loading conditions show a region of highest stress where galling took place.

Determination of "P" (bearing load). It was estimated that a conservative pressure differential of 250 psi was acting on the valve disc as the disc was closing and nearly closed. This resolves to 50,000 pounds for each hinge pin.

$$"P" = 50,000 \text{ lb}$$

It must be noted that this is a conservative value since action of the dashpot was neglected.

Worst Condition

$$q = 0.445 (50,000) = 22,250 \text{ lb/in.}$$

$$b = \sqrt{2 (q) \Delta / \pi}$$

$$b = \left[\frac{(2) (2.225) \times 10^4 (2.43 \times 10^{-5})}{\pi} \right]^{\frac{1}{2}}$$

$$b = 0.586$$

$$\sigma = \frac{b}{\Delta} = \frac{5.86 \times 10^{-1}}{2.43 \times 10^{-5}} = 24,100 \text{ psi}$$

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APPENDIX B

WATERHAMMER WAVE REFLECTION CALCULATION

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APPENDIX B

WATERHAMMER WAVE REFLECTION CALCULATION

The equations for the transmission and reflection of pressure waves in a compressible fluid, such as waterhammer waves, at a point of change in conduit properties may be derived from the one-dimensional equations of motion and continuity for compressible fluids, using the compressibility of the pipe-fluid system. This is customary in waterhammer theory^[6], and the results are

$$s = \frac{2A_1/a_1}{A_1/a_1 + A_2/a_2}$$

$$r = \frac{A_1/a_1 - A_2/a_2}{A_1/a_1 + A_2/a_2}$$

where

s = transmission coefficient (fraction of the wave transmitted)

A = pipe cross-sectional area

a = wave propagation velocity in the pipe

r = reflection coefficient (fraction of the wave reflected).

Subscripts 1 and 2 refer to conditions upstream and downstream, respectively, in the direction of wave travel. Negative coefficients refer to a change in sign of the pressure wave (possible only upon reflection and not upon transmission).

The equations for transmission and reflection coefficients at a branch point in a piping system are the same, except that the parameter A_2/a_2 for the

downstream branch is replaced by $\sum_{i=2}^n A_i/a_i$ for n downstream pipes.

Figure 25 is a schematic diagram of the ATR primary pumps and their manifolds. At each branch point or change in section, the reflection coefficient is shown. Because of the complicated geometry of the pumps, no coefficient can be determined. It is known from pressure measurements that the reflection in the pumps is large and the transmission small. These facts are deduced from the observation that no evidence of pressure surges coming from the upstream side of the check valve was seen on the pressure transducer downstream of the check valve. These surges would have come through the two open-pump systems in the 1-of-3 pump cutoff. If the transmission coefficient of a pump were 1, pulses of 1/3 of the measured pulses should have been superimposed on the traces from the downstream transducer at Test Point 2. The fact that none was observed indicates that the transmission through one pump must be less than 1/4.

Fig. 25 Schematic flow diagram of primary pump manifold showing reflection coefficients.

APPENDIX C

SAMPLE CALCULATIONS FROM ANALYSIS PREDICTING CLOSING TIMES FOR 20-INCH CHECK VALVES

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APPENDIX C

SAMPLE CALCULATIONS FROM ANALYSIS PREDICTING CLOSING TIMES FOR 20-INCH CHECK VALVES

This analysis was completed in two parts. The first part was completed using a finite difference technique over small time increments. The forces acting on the piston and piston rod were determined for each time increment. This force depends on the assumed velocity of the piston at the end of the time increments and was arrived at in the following manner. The Roman numerals correspond to those on the sample calculations sheet shown in Figure 30, page 69.

I. First choose a time increment, ΔT , (sec).

II. Assume velocity of piston at the end of the time increment, VP (ft/sec).

III. Determine piston travel (PT) in feet from the time increment. This is given by the average velocity times the time increment.

$$PT = \frac{(VP_i + VP_{i-1}) \Delta T}{2}$$

where

i = time increment index

VP_i = piston velocity at end of ΔT

VP_{i-1} = piston velocity at beginning at ΔT .

IV. Convert piston travel to inches.

$$pt = 12 PT$$

V. Determine piston position (PP) at end of time increment. This is given by the piston position for the end of the previous time increment (PP_{i-1}) plus the piston travel in inches for present time increment.

$$PP = PP_{i-1} + pt$$

VI. Disc position (α), degrees from horizontal, at end of time increment from Figure 26. Figure 26 relates disc position to piston position. From the geometry of the linkage system the piston position is related to the disc position by the following equation.

$$PP = 5.489 - \left\{ 4.5 \cos \left[\sin^{-1} \left(\frac{7.03 \cos (\alpha - 16^\circ 15') - 5.5}{4.5} \right) \right] - 7.03 \sin (\alpha - 16^\circ 15') \right\}$$

Fig. 26 20-inch check valve disc position in degrees from horizontal versus piston position.

VII. Determine velocity of fluid (VF) in one-inch bypass line, ft/sec. Area of one-inch bypass line = 0.85 in.^2 . This was adjusted to allow for the flow through the 1/2 inch bypass line and annulus between the piston and cylinder. The one-inch and 1/2-inch bypass lines are shown in Figure 17.

Assume a piston velocity of 1 in./sec.

$$\text{Area of piston} = \frac{\pi(12)^2}{4} = 113 \text{ in.}^2$$

$$V_1 A_1 = V_2 A_2 \quad V_1 = \frac{V_2 A_2}{A_1}$$

$$\text{Velocity of fluid in one-inch bypass line} = \frac{(1 \text{ in./sec}) 113 \text{ in.}^2}{0.85 \text{ in.}^2} =$$

$$133 \text{ in./sec} = 11.1 \text{ ft/sec.}$$

Find pressure head differential using Bernoulli's equation, considering the one-inch bypass line.

$$\frac{\Delta P}{\gamma} = \frac{1.2 V^2}{2g} + \frac{35 f V^2}{2g} + 75 f \frac{V^2}{2g}$$

where

$$\frac{1.2 V^2}{2g} = \text{entrance and exit head loss}$$

$$\frac{35 f V^2}{2g} = \text{line head loss due to line}^{[7]}$$

$$\frac{75 f V^2}{2g} = \text{line head loss due to elbows}^{[7]}$$

$$\frac{\Delta P}{\gamma} = \frac{1.2V^2}{2g} + \frac{110 (f) V^2}{2g}$$

Find friction factor (f) from Moody diagram for Reynolds number (R_e).

$$R_e = \frac{VD}{\nu} = \frac{11.1(1)}{(0.93 \times 10^{-5})(12)} = 9.95 \times 10^4$$

$$f = 0.018$$

$$\frac{\Delta P}{\gamma} = \frac{1.2V^2}{2g} + \frac{110 (0.018) V^2}{2g} = \frac{3.18 V^2}{2g}$$

Now find the pressure head differential considering the annulus between the piston and cylinder.

$$\frac{\Delta P}{\gamma} = \frac{1.5 V_1^2}{2g} + f \frac{\ell}{D} \frac{V_1^2}{2g}$$

$$\ell = 2-1/2 \text{ in. } D = 0.008 \text{ in. } \frac{\ell}{D} = 310$$

Assume $f = 0.02$ V_1 = velocity of fluid through annulus

$$\frac{\Delta P}{\gamma} = \frac{1.5V_1^2}{2g} + \frac{310 (0.02)V_1^2}{2g} = \frac{7.7 V_1^2}{2g}$$

By equating the two pressure head differentials, the velocity of fluid through the annulus can be solved in terms of velocity through the one-inch bypass line.

$$\frac{7.7 V_1^2}{2g} = \frac{3.18 V^2}{2g}$$

$$V_1 = \sqrt{\frac{3.18 V^2}{7.7}} = \sqrt{\frac{3.18 (11.1)^2}{7.7}} = 7.1 \text{ ft/sec.}$$

Now check assumed friction factor

$$R_e = \frac{VD}{\nu} = \frac{7.1 (0.008)}{0.930 \times 10^{-5} (12)} = 5.1 \times 10^2$$

$f = 0.02$ is a poor value, now use $f = 0.10$

$$\frac{\Delta P}{\gamma} = \frac{1.5 V_1^2}{2g} + \frac{310 (0.1) V_1^2}{2g} = \frac{33V_1^2}{2g}$$

Again solve for velocity of fluid through annulus.

$$\frac{33V_1^2}{2g} = \frac{3.18V^2}{2g}$$

$$V_1 \sqrt{\frac{3.18}{33}} = V$$

$$V_1 = \sqrt{\frac{3.18}{33}} (11.1)^2 = 3.44 \text{ ft/sec}$$

Now by knowing the velocities of fluid through both the one-inch bypass line and annulus and area of the annulus, the equivalent area to be added to the one-inch bypass line can be determined.

$$VA = V_1A_1 \quad A = \frac{V_1A_1}{V}$$

where

A = equivalent area to be added to one-inch bypass line

A_1 = area of annulus = $\pi(12) (0.009) = 0.34 \text{ in.}^2$

V_1 = velocity through annulus

V = velocity through one-inch bypass line

Therefore

$$A = \frac{3.44 \text{ ft/sec } 0.34 \text{ in.}^2}{11.1 \text{ ft/sec}} = 0.11 \text{ in.}^2$$

Now do the same to allow for flow through 1/2-inch bypass line.

$$\frac{\Delta P}{\gamma} = \frac{1.5 V_2^2}{2g} + \frac{80 (f) V_2^2}{2g} + \frac{415 (f) V_2^2}{2g}$$

$$\frac{\Delta P}{\gamma} = \frac{1.5 V_2^2}{2g} + \frac{495 (f) V_2^2}{2g}$$

Assume $f = 0.02$

$$\frac{\Delta P}{\gamma} = \frac{1.5 V_2^2}{2g} + \frac{495 (0.02) V_2^2}{2g}$$

$$\frac{\Delta P}{\gamma} = \frac{11.4 V_2^2}{2g}$$

Now equate differential pressure heads and solve for velocity through 1/2-inch bypass line (V_2) in terms of velocity through one-inch bypass line (V)

$$\frac{11.4 V_2^2}{2g} = \frac{3.18 V^2}{2g}$$

$$V_2 = \sqrt{\frac{3.18 V^2}{11.4}} = \sqrt{\frac{3.18 (11.1)^2}{11.4}} = 5.8 \text{ ft/sec.}$$

Now check assumed friction factor.

$$R_e = \frac{VD}{\nu} = \frac{5.8 (0.5)}{(12) 0.93 \times 10^{-5}} = 2.6 \times 10^4$$

Assumed friction factor is correct. As with the annulus, the equivalent area to be added to the one-inch bypass line to correct for flow through 1/2-inch bypass line can now be determined.

$$VA = V_2 A_2$$

$$A = \frac{V_2 A_2}{V}$$

$$A_2 = \text{Area of 1/2-inch line} = 0.302 \text{ in.}^2$$

$$A = (0.302) \frac{5.8}{11.1} = 0.16 \text{ in.}^2$$

Now the effective area of the one-inch bypass line is equal to

$$0.85 \text{ in.}^2 + 0.11 \text{ in.}^2 + 0.16 \text{ in.}^2 = 1.13 \text{ in.}^2$$

It was realized that the effective area would vary slightly for different velocities of the piston since friction factors are a function of velocity. However, it was felt the change in effective area was negligible, thus permitting the use of 1.13 in.² throughout.

Knowing the effective area (A_{eff}) of the one-inch bypass line and area of the piston, the velocity of fluid (V_F) in the one-inch bypass line can be solved for in terms of velocity of piston (V_P).

$$V_F (A_{\text{eff}}) = V_P (A_2)$$

$$V_F = \frac{V_P (A_2)}{A_{\text{eff}}}$$

$$A_{\text{eff}} = \text{effective area of one-inch bypass line} = 1.13 \text{ in.}^2$$

$$A_2 = \text{area of piston} = \frac{\pi(12)^2}{4} = 113 \text{ in.}^2$$

$$V_F = \frac{V_P 113 \text{ in.}^2}{1.13 \text{ in.}^2} = 100 V_P$$

VIII. Determine Reynolds number (R_e) for one-inch bypass line.

$$R_e = \frac{V d}{\nu} = \frac{V_F (1.05)}{(0.93 \times 10^{-5})(12)} = V_F (9.4 \times 10^3)$$

IX. Determine friction factor (f) from Moody diagram for corresponding R_e .

X, XI, and XII.

Find force on piston due to pressure differential (F_P) caused by line losses.

$$F_P = \Delta P (\text{area})$$

$$\text{Area} = \frac{113 \text{ in.}^2}{144 \text{ in.}^2/\text{ft}^2} = 0.785 \text{ ft}^2$$

$$\frac{\Delta P}{\gamma} = \frac{K V_F^2}{2g} + \frac{f \ell}{d} \frac{V_F^2}{2g} + f \left(\frac{\ell}{d} \right)_{\text{eq}} \frac{V_F^2}{2g} + \frac{V_F^2}{2g}$$

Where

$$\frac{\Delta P}{\gamma} = \text{pressure head differential}$$

$$\frac{KVF^2}{2g} = \text{entrance loss}$$

$$f \frac{\ell}{D} \frac{VF^2}{2g} = \text{line loss due to pipe}$$

$$f \left(\frac{\ell}{D} \right)_{eq} \frac{VF^2}{2g} = \text{line loss due to elbows}$$

$$\frac{VF^2}{2g} = \text{exit loss}$$

VF = velocity of fluid

K = entrance loss coefficient = 0.2^[8]

f = friction factor

ℓ = length of line

D = diameter of line

$\left(\frac{\ell}{D} \right)_{eq}$ = equivalent length of pipe in pipe diameters
= 75 for two 90-degree elbows and one 45-degree elbow^[7]

$$\gamma = 62.4 \text{ lb/ft}^3$$

$$2(g) = 64.4 \text{ ft/sec}^2$$

$$\frac{\Delta P}{62.4} = \frac{(0.2) VF^2}{64.4} + \frac{35 (f) VF^2}{(1) 64.4} + \frac{75 (f) VF^2}{64.4} + \frac{VF^2}{64.4}$$

$$\Delta P = \frac{62.4}{64.4} \left[1.2 (VF)^2 + f(110) VF^2 \right]$$

$$\Delta P = \frac{62.4}{64.4} (VF)^2 \left[1.2 + f(110) \right]$$

X. Let A = f(110)

XI. Let B = A + 1.2

This is not true when the entrance to the one-inch bypass line is partially closed by the piston. The entrance loss is still $\frac{(0.2) V^2}{2g}$, where V is the velocity

of the fluid at the entrance. But this velocity is greater than in the one-inch bypass line as shown by

$$V(A) = VF(A_1)$$

$$V = \frac{VF(A_1)}{A}$$

where

V = velocity at entrance

A = area of entrance

VF = velocity in one-inch bypass line

A_1 = area of one-inch bypass line.

Now substituting the velocity at the entrance in terms of the areas and velocity in the one-inch bypass line into the entrance loss term, the result is

$$\text{Entrance loss} = \frac{0.2 (A_1)^2 (VF)^2}{A^2 2g}$$

Therefore, when the entrance is partially closed the value of B is given as follows:

$$B = A + 1 + \frac{0.2(A_1)^2}{A^2}$$

Going on with the determination of ΔP ,

$$\Delta P = \frac{62.4}{64.4} (VF)^2 B$$

XII. Force on piston (FP) = ΔP (area of piston).

$$FP = \frac{62.4}{64.4} (VF)^2 B (0.785) = 0.76 (VF)^2 B$$

XIII. FP was determined for the end of the time increments. Now it is necessary to determine an average value (FPA) based on values of FP.

$$FPA = \frac{FP_i + FP_{i-1}}{2}$$

XIV. and XV.

It also was necessary to determine the force acting on the piston due to a pressure differential caused by acceleration losses. It can easily be shown that acceleration of fluid in the dashpot cylinder need not be considered. First consider ΔP necessary for accelerating the mass of fluid in the dashpot.

Force = mass x acceleration

$$\text{Force} = \Delta P (A_1)$$

$$A_1 = \text{area of piston} = 113 \text{ in.}^2$$

$$\text{Volume of fluid} = \frac{\pi(1)^2}{4} (1) = 0.785 \text{ ft}^3$$

$$\text{Mass of fluid in dashpot} = (0.785) \frac{62.4}{32.2} = 1.52 \text{ slugs}$$

Let the piston and fluid in dashpot cylinder accelerate by an amount $\bar{A}$.

$$\Delta P (A_1) = 1.52 \bar{A}$$

$$\Delta P = \frac{1.52 \bar{A}}{A_1}$$

Now give the same consideration to the fluid in the one-inch bypass line. Volume of fluid = $(1.06) 35 = 37.1 \text{ in.}^3 = 0.0215 \text{ ft}^3$.

$$\text{Mass of fluid} = (0.0215) \frac{62.4}{32.2} = 0.0416 \text{ slugs}$$

$$\Delta P A_2 = 0.0416 \bar{A}_2$$

$$A_2 = 0.85 \text{ in.}^2 = \frac{A_1}{113}$$

For a given acceleration ($\bar{A}$) of piston, acceleration of fluid ($\bar{A}_2$) in one-inch bypass line is given by

$$\bar{A}_2 = 133 \bar{A}$$

$$\Delta P \frac{A_1}{133} = 0.0416 (133) \bar{A}$$

$$\Delta P = \frac{(0.0416) (133)^2 \bar{A}}{A_1}$$

The ratio of the two results indicates that for the same acceleration of the piston, the pressure differential due to the acceleration of the fluid in the one-inch bypass line is 484 times that due to the fluid in the dashpot cylinder.

$$\frac{\frac{0.0416 (133)^2 \bar{A}}{A_1}}{\frac{1.52 \bar{A}}{A_1}} = 484$$

Thus, acceleration of the fluid in the dashpot cylinder need not be considered.

XIV. Acceleration of fluid in the one-inch bypass line (Acc) is given for the time increment as follows:

$$\text{Acc} = \frac{VF_i - VF_{i-1}}{\Delta T}$$

XV. As already shown,

$$\Delta P (A_2) = (0.0416) \text{Acc.}$$

$$A_2 = 0.85 \text{ in.}^2$$

$$\Delta P = \frac{(0.0416) \text{Acc.}}{0.85}$$

Force of piston (FA) due to this pressure differential is given by

$$FA = \Delta P (\text{area of piston}) = \frac{0.0416 (\text{Acc}) (113)}{0.85} = 5.53 \text{ Acc.}$$

XVI. Total force on piston (FT) due to pressure differential is given by

$$F_T = F_A + F_{PA}.$$

XVII. Obtain (Q) at end of time increment from Figure 12.

XVIII. Find drag force acting on check valve disc. This force which acts parallel to flow was determined as follows:

First equate the net force due to pressure drop across the check valve to drag force.

$$\Delta P (A_1) = F_{\text{Drag}} = \frac{C_D \rho A_2 V^2}{2}$$

ΔP = pressure drop across check valve

A_1 = flow area = 284 in.²

C_D = drag coefficient

ρ = density of fluid

A_2 = projected area of disc normal to flow = 3.0 ft² sin α

V = velocity of fluid

$$\Delta P = \frac{C_D \rho A_2 V^2}{2 A_1}$$

$$\text{Put in } V^2 = ZQ^2$$

where

Q is in gpm and Z is the appropriate constant

$$\Delta P = \frac{C_D \rho A_2 Z Q^2}{2 A_1}$$

Now let $K = \frac{C_D \rho Z}{2 A_1}$

$$\Delta P = K A_2 Q^2$$

$$K = \frac{\Delta P}{A_2 Q^2} = \frac{\Delta P}{(3.0) \sin \alpha Q^2}$$

With this relationship and values of α and ΔP for various values of Q from previous flow tests performed on the check valves, it was possible to obtain a plot of K versus α for values of α from 10 to 40 degrees. This plot is given in Figure 27.

$$F_{\text{Drag}} = \Delta P A_1 = K A_2 Q^2 A_1 = K Q^2 (3.0) (\sin \alpha) 284 = 852 K Q^2 \sin \alpha$$

XIX, XX, and XXI.

After the drag force (F_{Drag}) was determined, it was necessary to sum moments about the hinge pin and solve for the vertical component of the axial force in the link which was applied to the lower end of the piston rod.

XIX. The resultant drag force was assumed to act at a point in the center of the disk. Summing moments about the hinge pin, the moment arm (D) for the drag force was given by $3.25 \sin \alpha + 3.0 \cos \alpha$. A plot of this moment arm (D) versus α is given in Figure 28.

Fig. 27 Drag coefficient versus disc position.

Fig. 28 Drag moment arm versus disc position.

XX. Let F = force in link. Still summing moments about hinge pin, the following is obtained:

$$F_{\text{Drag}}(D) = F(\cos \theta) 7.02 \cos (\alpha - 16.25^\circ) - F(\sin \theta) 7.02 \sin (\alpha - 16.25^\circ) .$$

The vertical component (FV) of F is given by

$$FV = F \cos \theta, F = \frac{FV}{\cos \theta} .$$

Now,

$$F_{\text{Drag}}(D) = FV \frac{\cos \theta}{\cos \theta} 7.02 \cos (\alpha - 16.25^\circ) - FV \frac{\sin \theta}{\cos \theta} 7.02 \sin (\alpha - 16.25^\circ)$$

$$F_{\text{Drag}}(D) = FV 7.02 \left[\cos (\alpha - 16.25^\circ) - \tan \theta \sin (\alpha - 16.25^\circ) \right]$$

where

$$\theta = \sin^{-1} \left[\frac{7.02 \cos (\alpha - 16.25^\circ) - 5.5}{4.5} \right] .$$

$$\text{Let } R = 7.02 \left[\cos (\alpha - 16.25^\circ) - \tan \theta \sin (\alpha - 16.25^\circ) \right]$$

Figure 29 is plot of R versus α .

Fig. 29 "R" versus disc position.

XXI. Determine vertical component (FV) of force in link.

$$F_{\text{Drag}} (D) = FV (R)$$

$$FV = \frac{F_{\text{Drag}} (D)}{(R)}$$

XXII. Determine average value (FVA) of the vertical force acting on the lower end of the piston.

$$FVA = \frac{FV_i + FV_{i-1}}{2}$$

XXIII. Determine the total upward force (FTU) acting on the piston.

$$FTU = FVA + FT$$

XXIV. Determine the force due to compressed spring (FS) at end of time increment.

$$\text{Spring constant} = 60 \text{ lbs/in.}$$

Spring is compressed 5.4 inches when the check valve is in the fully open position. Therefore, the spring exerts a 325-pound force when the check valve is in the fully open position. The spring force at the end of each time increment is given by:

$$FS = 325 - 60 (\text{total piston travel}).$$

$$\text{Total Piston Travel} = \text{Piston Position} - 2.3$$

Therefore

$$FS = 325 - 60(PP - 2.3).$$

XXV. Determine the average spring force (FSA) for the time increment.

$$FSA = \frac{FS_i + FS_{i-1}}{2}$$

XXVI. Determine the combined force (FTD) due to the average spring force and weights of piston and piston rod.

$$FTD = 112 + FSA$$

XXVII. Determine the net force (F) acting on the piston and piston rod.

$$F = FTD - FTU$$

XXVIII. Determine work done (W) on piston and piston rod.

$$W = F \text{ (Piston Travel)}$$

XXIX. From an energy balance, the final velocity of the piston was determined for the time increment and checked against the assumed final velocity of the piston for the time increment.

Work = change in kinetic energy

$$W = \frac{\text{Mass}}{2} (V_i^2 - V_{i-1}^2)$$

$$\text{Mass of piston and piston rod} = \frac{112}{32.2} = 3.5 \text{ slugs}$$

$$V_i^2 - (V_{i-1})^2 = \frac{2W}{3.5}$$

$$V_i^2 = \frac{W}{1.75} + (V_{i-1})^2$$

$$V_i = \sqrt{\frac{W}{1.75} + (V_{i-1})^2}$$

These last two steps, XXVIII and XXIX, were not done because it was noted that greater accuracy was needed for the net force acting on the piston and piston rod than could be obtained. The reason for this is that the force necessary for accelerating the piston and piston rod is very small compared to the other forces involved as was the case with the force necessary for accelerating the fluid in the dashpot cylinder. This can be shown by looking at the maximum acceleration of the piston.

$$\text{Maximum acceleration} = \frac{0.062 \text{ ft/sec}}{.25 \text{ sec}} = 0.248 \text{ ft/sec}^2$$

Force = mass x acceleration

$$\text{Force} = \frac{112 \text{ lbs } (0.248 \text{ ft/sec}^2)}{32.2 \text{ ft/sec}^2} = 0.863 \text{ lb}$$

The accuracy of the assumed velocity of the piston was determined from the magnitude of the net force acting on the piston and piston rod, as shown in the last row of the sample calculation sheet (Figure 30). Large negative values at this net force indicate that the assumed piston velocity was too fast. Likewise, large positive values indicate the assumed piston velocity was too slow.

The second part of this analysis was done by assuming that the piston traveled at a constant velocity due to an average closing force. This constant velocity was determined by equating the assumed average closing force to the damping forces of the dashpot. The average closing force for the remainder of the check valve closure was assumed to be 900 pounds.

$$900 \text{ lbs} = \Delta P (\text{area of piston})$$

$$900 \text{ lbs} = \Delta P 0.785 \text{ ft}^2$$

$$\Delta P = \frac{900}{0.785} = 1,145 \text{ lb/ft}^2$$

$$\frac{\Delta P}{\gamma} = \frac{1.5 V^2}{2g} + f \frac{\ell}{D} \frac{V^2}{2g} + f \left(\frac{\ell}{d} \right)_{\text{eq}} \frac{V^2}{2g}$$

where

$$\frac{1.5 V^2}{2g} = \text{entrance and exit loss for 1/2-inch bypass line}$$

$$\frac{\ell}{D} = 80 \text{ for 1/2-inch line}$$

$$\left(\frac{\ell}{D} \right)_{\text{eq}} = 340 \text{ for valve in 1/2-inch bypass line}$$

$$\left(\frac{\ell}{D} \right)_{\text{eq}} = 75 \text{ for two 90 degree elbows and one 45 degree elbow [7]}$$

$$\frac{\Delta P}{\gamma} = \frac{1.5 V^2}{2g} + f (495) \frac{V^2}{2g}$$

$$\frac{\Delta P (2g)}{\gamma} = V^2 (1.5 + 495 f)$$

Assume $f = 0.021$

$$\frac{\Delta P (2g)}{\gamma} = V^2 [1.5 + 495 (0.021)]$$

I	ΔT (sec)	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.10	0.10	0.10	0.10	0.10	
II	VP (ft/sec)	0.00	0.00	0.07	0.066	0.06	0.062	0.10	0.116	0.118	0.12	0.13	0.128	0.134	0.14	0.15	0.156	0.156	0.164	0.164	0.156	0.152
III	$PT = \frac{VP_i + VP_{i-1}}{2} \Delta T$ (ft)	0.0	0.0	.00875	.00825	.0075	.00775	.02025	.02225	.0225	.02975	.031	.03075	.0325	.03425	.0355	.03625	.0156	.0160	.0164	.0160	.0158
IV	$p_t = PT$ (I2) (in)	0.0	0.0	0.105	0.099	0.09	0.093	.243	.2665	.270	.357	.372	.368	.390	.411	.426	.435	.1872	.192	.1968	.192	.1896
V	$PP = PP_{i-1} + p_t$ (in)	2.3	2.3	2.405	2.399	2.390	2.393	2.636	2.660	2.663	3.020	3.035	3.031	3.421	3.832	3.847	3.856	4.043	4.048	4.245	4.24	4.237
VI	α from Figure 26	25°	25°	26°	26°	25.8°	25.9°	28°	28.2°	28.2°	31.3°	31.3°	31.3°	35.3°	39.2°	39.4°	39.5°	40.5°	40.5°	43.3°	43.3°	43.3°
VII	VF=100 VP	0.007	0.00	7.0	6.6	6.0	6.2	10.0	11.6	11.8	12.0	13.0	12.8	13.4	14.0	15.0	15.6	15.6	16.4	16.4	15.6	15.2
VIII	$RE = VF (9.4 \times 10^3)$	0	0	6.6×10^4	6.2×10^4	5.6×10^4	5.8×10^4	9.4×10^4	1.1×10^5	1.1×10^5	1.1×10^5	1.2×10^5	1.2×10^5	1.2×10^5	1.3×10^5	1.4×10^5	1.4×10^5	1.4×10^5	1.5×10^5	1.5×10^5	1.5×10^5	1.5×10^5
IX	f from Moody Diagram	0	0	.0195	.0198	.0200	.0200	0.018	0.018	0.018	0.018	0.018	0.018	0.017	0.017	0.017	0.017	0.017	0.017	0.017	0.017	0.017
X	$A = 110f$	0	0	2.145	2.18	2.20	2.20	1.98	1.98	1.98	1.98	1.98	1.98	1.87	1.87	1.87	1.87	1.87	1.87	1.87	1.87	1.87
XI	$B = A + 1.2$	1.2	1.2	3.345	3.38	3.40	3.40	3.18	3.18	3.18	3.18	3.18	3.18	3.07	3.23	3.23	$\frac{+1.63}{3.50}$	$\frac{+4.80}{3.67}$	$\frac{+1.8}{3.67}$	$\frac{+6.0}{7.87}$	$\frac{+6.0}{7.87}$	$\frac{+6.0}{7.87}$
XII	$FP = .76B(VF)^2$ (lbs)	0	0	124.6	111.5	93.0	99.0	240.0	325	337	348	408	395	419	482	552	648	676	750	1,615	1,450	1,380
XIII	$FPA = \frac{FP_i + FP_{i-1}}{2}$ (lbs)	0	0	62.3	56.0	46.5	49.5	169.5	212	218	342.5	372.5	366.0	407	450.5	485.5	533.5	662	699	1,182	1,100	1,065
XIV	$Acc = \frac{VF_i - VF_{i-1}}{\Delta T}$ (ft/sec ²)	0	0	28.0	26.4	24.0	24.8	15.2	21.6	22.4	0.8	7.2	4.0	2.4	2.4	6.4	8.8	0	8.0	0	-8.0	-12.0
XV	$FA = 5.53 Acc$ (lbs)	0	0	155	146	127	137	84.0	119.5	124.0	4.5	40.0	22.0	13.0	13.0	35.0	49.0	0	44.0	0	-44.0	-66.0
XVI	$FT = FA + FPA$ (lbs)	0	0	217.3	202	173.5	186.5	253.5	331.5	342.0	347.0	412.5	388.0	420.00	463.5	520.5	582.5	662	794	1,182	1,056	999
XVII	Q from Figure 2 (gpm)	12,500	8,500	4,200	4,200	4,200	4,200	1,800	1,800	1,800	0	0	0	*reverse -2,100	-3,700	-3,700	-3,700	-4,100	-4,100	-4,500	-4,500	-4,500
XVIII	$F Drag = K(\sin \alpha) Q^2$ (lbs) K from Figure 27	1,350	625	178	178	174	176	45	45	45	0	0	0	-133	-556	-575	-575	-744	-744	-940	-950	-950
XIX	D from Figure 28 (in)	4.085	4.085	4.11	4.11	4.109	4.109	4.17	4.17	4.17				4.325	4.38	4.385	4.395	4.395	4.395	4.41	4.41	4.41
XX	R from Figure 29 (in)	6.6	6.6	6.55	6.55	6.54	6.54	6.41	6.41	6.41				6.05	5.95	5.85	5.85	5.8	5.8	5.75	5.75	5.75
XXI	$FV = \frac{F Drag}{Q} \left(\frac{Q}{2} \right)$ (lbs)	835	387	112	112	109.5	111	29	29	29	0	0	0	-95	-410	-432	-432	-564	-564	-722	-730	-730
XXII	$FVA = \frac{FV_i + FV_{i-1}}{2}$ (lbs)	1000	611	249.5	249.5	248.2	249	70	70	70	15	15	15	-47.5	-255.5	-263.5	-263.5	-498	-498	-643	-647	-647
XXIII	$FTU = FT + FVA$ (lbs)	1000	611	466.8	451.5	421.7	435.5	323.5	401.5	412	362	427.5	403	372.5	208	257.0	319.0	164.0	296	539	409	352
XXIV	$FS = 325 - 60(pp - 23)$ (lbs)	325	325	318.7	319	319	319	305	303	303	282	281	281	258	234	232	232	223	223	208	208	208
XXV	$FSA = \frac{FS_i + FS_{i-1}}{2}$	325	325	321.3	322.0	322.0	322.0	312	311	311	292.5	292.0	292.0	269.5	246	245	245	227.5	227.5	215	215	215
XXVI	$FTD = FSA + 112$	437	437	433.3	434	434.0	434.0	424.0	423.0	423.0	404.5	404.0	404.0	381.5	358	357	357	339.5	339.5	327	327	327
XXVII	$F = FTD - FTU$	-675	-175	-33.5	-17.5	12.3	-1.5	100.5	21.5	11.0	42.5	-8.5	1.0	9.0	150	100	38	175.5	43.5	-212	-82	-25
	Comments	didn't move-OK		fast	fast	slow	OK	slow	slow	OK	slow	fast	OK	OK	slow	slow	OK	slow	OK	fast	fast	OK

Fig. 30 Sample calculation sheet.

$$V^2 = \frac{(1,145) (64.4)}{(62.4) (1.5 + (0.021) 495)}$$

$$V = 9.96 \text{ ft/sec.}$$

Check assumed friction factor (f)

$$R_e = \frac{VD}{\nu} = \frac{9.96(0.5)}{(12) 0.93 \times 10^{-5}} = 4.46 \times 10^4 \quad \text{O. K.}$$

Area of 1/2-inch line = 0.302 in.². This has to be adjusted to allow for flow through annulus. From the first part of this analysis for a piston velocity of 1 in./sec, the velocity through the annulus was determined to be 3.44 ft/sec. The corresponding velocity through the 1/2-inch bypass line was found to be 5.8 ft/sec. The area of the annulus is 0.34 in.². The equivalent area to be added to the 1/2-inch bypass line to allow for flow through the annulus is $\frac{3.44}{5.8} (0.34) = 0.202$ in.². The area of the 1/2-inch bypass line is 0.302 in.². Thus, the total effective area of the 1/2-inch bypass line is $0.302 + 0.202 = 0.504$ in.². The velocity of the piston is now determined from $VA = V_1 A_1$ $V = \frac{V_1 A_1}{A}$

where

V = velocity of piston

A = area of piston

V_1 = velocity of fluid in 1/2-inch bypass line

A_1 = effective area of 1/2-inch bypass line

$$V = \frac{9.96(0.504)}{113} = 0.0445 \text{ ft/sec} = 0.534 \text{ in./sec.}$$

At the end of the first part of this analysis, using the finite difference technique, the piston position was determined to be 4.237 inches at 1.95 sec. The piston velocity was then 0.152 ft/sec. From this position, the piston has to travel $4.500 - 4.237 = 0.263$ inches before the one-inch bypass line is completely closed off by the piston. The average piston velocity for this 0.263 inch of travel was assumed to be the average of that determined at the end of the finite difference technique and that determined for the last part of closure, and is given by:

$$\frac{0.152 + 0.0445}{2} = 0.098 \text{ ft/sec} = 1.176 \text{ in./sec.}$$

The time for this 0.263 inch of piston travel is $\frac{0.263 \text{ in.}}{1.176 \text{ in./sec}} = 0.22 \text{ sec.}$ Therefore, the piston position is 4.5 inches at 2.17 sec.

For complete closure of the check valve, the piston had to travel an additional 0.80 inch at a velocity of 0.534 in./sec. The time required for this piston travel is $\frac{0.80 \text{ in.}}{0.534 \text{ in./sec}} = 1.50 \text{ sec.}$ Therefore, the check valve would be completely closed at 3.67 sec.

As can be noted from Figures 18, 19, 20, and 21, the second part of this analysis was in much greater error than the first part. It is felt this error was due to the average closing force used for the last portion of closure. It appears that the drag force acting on check valve disc for the last portion of closure was much greater than was estimated. This did not present as much of a problem for the first part of the check valve closure since the drag force acting on the check valve was determined using data from flow tests previously performed on the check valves. These data were not obtained for the last portion of check valve closure; hence, the drag force acting on the check valve disc for the last portion of closure had to be estimated based on the drag force determined for the first part of closure.