

University of Minnesota
St. Anthony Falls Hydraulic Laboratory

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TESTS OF THE WOOLLEY VALVE
FOR THE
JAMES H. CAMPBELL PLANT, UNIT NO. 3

by

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CONTENTS

	Page
List of Figures	ii
List of Tables	iv
Introduction	1
The Woolley Valve	1
The Static Test Program in the Laboratory	3
A. Static Calibration of Valve Compression Springs	4
B. Mechanical Friction Tests	4
C. Opening Response to Hydrostatic Loading	8
D. Leakage Tests	10
Steady State Flow Tests	10
A. The Steady State Flow Facilities	10
B. Vorticity Observations	13
C. Slow Opening and Closing Vane Loading Tests	14
D. Combined Vane Loading and Headloss Tests	19
Dynamic Tests	27
A. Interim Valve Modifications and Adjustments	27
B. Final Adjustments, Modifications, and Test Criteria	27
C. Final Dynamic Flow Tests	29
Conclusions	34
Figures 1 - 33	35

LIST OF FIGURES

- Fig. 1 General Plan View of Proposed Intake System.
- Fig. 2 General Nature of the Woolley Standard Synchro-Chek Valve.
- Fig. 3 Installation of the Full Scale Woolley Valve at the Campbell Plant Intake - Lake Michigan.
- Fig. 4 Original Design of the Woolley 1:4 Scale Valve Model and Elbow Model Attachment to the Header.
- Fig. 5 Approximate Dimensions of the Full Scale Woolley Standard Synchro-Chek Valve (from W. J. Woolley Co., Bulletin 103-73).
- Fig. 6 Damping Control Mechanism of the Model Woolley Valve as Originally Supplied. (Adapted from Woolley Drawing No. 35909)
- Fig. 7 Damping Control Mechanism of the Woolley Valve Model as Finally Supplied. (Adapted from Woolley Drawing No. 35909, Revision of 5-30-79)
- Fig. 8 Schematic of the Time-Delay Hydraulic Circuit of the Woolley Valve. (Adapted from Woolley Drawing No. 36236, 3-9-79)
- Fig. 9 Test Stand for Calibration of Valve Compression Springs.
- Fig. 10 The Woolley Valve Model with External Test Loading Spring.
- Fig. 11 Woolley Valve Tests. Static Forces to Open and Close the Model Valve.
- Fig. 12(a, b, c, d) Set-up for Hydrostatic Loading Tests.
- Fig. 13 Woolley Valve Tests. Potentiometer Linearity.
- Fig. 14 Static Opening Tests of the Woolley Valve with a 150 Pound Spring.
- Fig. 15 Static Opening Tests of the Woolley Valve with a 125 Pound Spring.
- Fig. 16 Plan View of the Test Set-up.
- Fig. 17 Elevation of the Test Set-up for Woolley Valve 1:4 Scale.
- Fig. 18(a) Installation of the 1/4 Scale Model of the Woolley Valve in the Laboratory Volumetric Basin. Installation with Basin Drained.
- Fig. 18(b) Installation of the 1/4 Scale Model of the Woolley Valve in the Laboratory Volumetric Basin. Installation with Basin Filled and Valve Discharging.

LIST OF FIGURES (Cont'd)

- Fig. 19(a) Discharge Controls for the Woolley Valve Tests. Outside Piping and Control Valves.
- Fig. 19(b) Discharge Controls for the Woolley Valve Tests. Inside Instrumentation and Readout Station.
- Fig. 20 Opening and Closing Response of the Woolley Valve. (Spring rate = 120.7 lbs/in; spring rod setting X = 1.3 in; undamped pressures at Tap #2)
- Fig. 21 Opening and Closing Response of the Woolley Valve. (Spring rate = 120.7 lbs/in; spring rod setting X = 1.3 in; damped pressures at Tap #2)
- Fig. 22 Opening and Closing Response of the Woolley Valve to Imposed Pressure Differentials.
- Fig. 23 Opening and Closing Response of the Woolley Valve to Imposed Pressure Differentials.
- Fig. 24 Opening and Closing Response of the Woolley Valve to Imposed Differential Pressures.
- Fig. 25 Discharge Coefficient Versus Vane Opening as Measured by ΔH_{1-2} . (Data from Table II - Tests of 3/13/79)
- Fig. 26 Discharge Coefficient Versus Vane Opening as Measured by ΔH_{1-2} . (Data from Table IV - Tests of 3/28/79)
- Fig. 27 Discharge Coefficient Versus Vane Opening as Measured by ΔH_{1-3} . (Data from Table IV - Tests of 3/28/79)
- Fig. 28 Values of Discharge Versus Vane Opening and Vane Torque. (Data from Table I - Tests of 3/13/79)
- Fig. 29 Values of Discharge Versus Vane Opening and Vane Torque. (Data from Table III - Tests of 3/28/79)
- Fig. 30 Hypothetical Opening Response Under Various Head Differentials.
- Fig. 31 Opening and Closing Response of Woolley Valve with Time-Delay Mechanism.
- Fig. 32 Opening Response to Cycling Pressure for Woolley Valve with Time-Delay Mechanism.
- Fig. 33 Opening Response to Cycling Pressure for Woolley Valve with Time-Delay Mechanism.

LIST OF TABLES

- Table I Woolley Valve Tests - Data of 3/13/79.
- Table II Woolley Valve Tests - Computed Discharge Coefficient Tests of 3/13/79.
- Table III Woolley Valve Tests - Data of 3/28/79.
- Table IV Woolley Valve Tests - Computed Discharge Coefficients Tests of 3/28/79.

Project Report No. 181

THE WOOLLEY VALVE STUDIES

Introduction

The innovative use of fixed screens to reject solids in condenser cooling water drawn from Lake Michigan has been considered a practical solution for the lake withdrawal system proposed at the James H. Campbell, Unit No. 3 of the Consumer Power Company. The withdrawal point, which is 3,500 feet offshore and approximately 30 feet below the lake surface, is considered relatively free of screen-plugging solids under normal conditions, but during winter, frazil ice plugging is possible. To assure continued operation under these unusual conditions it was considered necessary to provide the system with an opened water (non-screened) auxiliary intake. This auxiliary function is to be provided by installing a relief valve at the outer or stub end of two of the four header pipes which are to constitute the intake system as shown in Fig. 1. These valves are to open and provide auxiliary intake water whenever screen plugging resulted in a selected level of pressure reduction within the header. The selected level proposed for the valve operation was 12 inches additional headloss. The valve deemed most appropriate for use was a modified version of a Synchro-check valve made by the W. J. Woolley Company of River Forest, Illinois.

The Woolley valve has been marketed for many years as a pump discharge check valve, but its performance under wave conditions, including 100 year storms, at the proposed site were unknown. In order to clarify the valve performance it was decided to conduct tests of a 1:4 scale-model valve at the St. Anthony Falls Hydraulic Laboratory of the University of Minnesota. The material which follows describes the Woolley valve, the three part test program (static, steady state flow, and dynamic flow), the test facilities, the test results, and the conclusions and recommendations resulting from the tests.

The Woolley Valve

The Woolley valve is a dual-vaned, centrally-hinged check valve which in general applications has the configuration and features shown in Fig. 2.

In the proposed Campbell Plant installation the valve is to be positioned as shown in Figs. 3 and 4, wherein the valve vanes, when closed, will be in the horizontal plane and the hinge axis is to be parallel to the axis of the header pipe. The valve vanes would normally fall downward or open under the influence of gravity but are restrained by the damper or restoring spring arrangement shown in Fig. 2. The spring also keeps the valve closed under modest differential pressure heads.

In the full-scale system the valve will have a nominal size of 72 inches, which is measured by the standard pipe size of the inlet elbow connecting the valve to the end of the 96 inch horizontal header pipe. The elbow makes a 90° turn and includes a 72 inch to 96 inch expansion. The general dimensions of a full-scale standard 72 inch Synchro-chek valve are shown in Fig. 5. The general dimensions of the 1:4 scale-model valve are shown in Fig. 4.

The restoring spring arrangement initially provided with the 1:4 scale-model valve was slightly different than that shown in Fig. 2. The spring system was externally housed as shown at the left in Fig. 4 and the main internal features were as shown in Fig. 6. In this the spring force was applied to the valve actuating rod, to a direct connected gear rack, to a driver gear, to a torsion bar, and thence to the valve vane. As a consequence of the tests, the spring system was later modified with internal features as shown in Fig. 7. In this modification the spring force was applied to the actuating rod, to a clevis arm, to a gear bracket, to a driver gear, to a torsion bar, and thence to the valve vane.

Each Woolley valve is intended to provide relief flow discharge ranging from 0 to 232 cfs whenever the valve is exposed to abnormal pressure differentials. A normal pressure differential head imposed by the operating intake system has been established as 9 inches. The valve is designed to begin opening whenever an additional differential of 12 inches (21 inches total) is imposed on the valve. The valve shall also maintain the full maximum discharge of 232 cfs when fully opened and under a differential head of 27 inches.

An auxiliary control mechanism is to provide the following hydraulic and functional performance:

- A. Remain closed for all piezometric head differences up to 21 inches.
- B. Exhibit a valve opening characteristic for piezometric head differences in excess of 21 inches as follows:
 - 1. Five (5) minute delayed opening for a continuous piezometric head difference of 12 inches in addition to 21 inches.
 - 2. Thirty (30) second delayed opening for a continuous piezometric head difference of ten (10) feet in addition to 21 inches.
- C. Permit an inflow of 104,000 gpm, for a piezometric head loss through the inlet valve of 27 inches.
- D. Close upon return of headloss to 21 inches.

The restoring or damper springs shown in Figs. 6 and 7 can be adjusted to provide the selected 21 inch pressure head (nominal 12 inch) additional response on opening. The time response of the valve is provided by the leakage rate of hydraulic oil through a piston-cylinder arrangement. The piston is loaded by the valve vanes acting through the vane torsion bars through the driver gear and thence through a gear rack connected to the piston. The dual piston arrangement and hydraulic circuits are shown in Fig. 8. The time-delay mechanism is housed in the appendage at the right of the valve in Fig. 4.

The Static Test Program in the Laboratory

The unknowns relating to the valve performance were judged to be susceptible to clarification by conducting tests of a model valve under conditions which simulated the exposure of the full-scale valve.

Initial tests include the following: (1) static calibration of the valve compression springs, (2) measurement of friction break-away forces under torque loading, (3) valve opening response to hydrostatic loading, and (4) a leakage test. These tests were conducted using small special assembled test facilities in the Laboratory.

Subsequent steady state and dynamic flow tests were conducted on the valve in Laboratory flow facilities and these tests are separately

described later in this report.

A. Static Calibration of Valve Compression Springs

Three different valve compression springs were supplied with the Woolley valve with nominal spring rates of 100, 125, and 150 pounds per inch of compression. These springs were calibrated separately with a gravity dead load weight system, as shown in Fig. 9. The spring housing and spring were removed from the valve and clamped to a support stand. A washer was placed on top of the spring and a loading rod extended through the washer, spring, and support stand to the weights below (Fig. 9). The dimension "Y" was measured accurately for "no load" conditions and for five different weight loads from 78 to 294 pounds and the spring rates computed. The measured spring rates were determined to be 90.9, 120.7, and 152.9 pounds per inch. Each of these values are the average of the five weight measurements which varied from 89.7 to 91.6, 120.0 to 122.0, and 150.0 to 154.7 pounds per inch, respectively. These spring rates were used in determining forces in the mechanical friction tests.

B. Mechanical Friction Tests

When originally received, the vane action of the Woolley valve model was observed to be sticky and high in internal frictional resistance to manually imposed opening or closing forces. The valve was then set up with a controlled and measurable external loading system for opening the valve vanes. The loading system which is shown in the photos of Fig. 10 consists of the following components.

- a. An attachment which clamps around the valve spring housing and which supports the external force lever loading system.
- b. The lever system forces an adjustable pin against the outboard end of the valve actuating rod (see Fig. 6) and in turn compresses the valve spring.
- c. The lever system derives its force from a spring balance hooked to the lever arm. The rotary dial of the balance reads from 0 to 30 pounds and revolves through two revolutions to apply a maximum spring force of 60 pounds.

- d. The spring balance is supported from above by an adjustable screw mechanism which applies and holds any desired valve spring loading on the spring adjuster rod.

To measure the vane angles, an adjustable protractor with a vernier and a bubble level was used in direct contact with the top of the vane. Before making these measurements, the valve was leveled across the top. With the valve closed, the driver gear vane (see Fig. 6) is open 0° and the idler gear vane is 0.5° open. With the valve wide open, the driver gear vane is open 89° and the idler gear vane, 87.5° .

An additional measure of vane angle was provided by suitably attaching an electrical potentiometer to the outboard end of one of the two hinge torsion bars of the valve. The potentiometer was contained within the housing provided for the time-delay mechanism previously referred to. The potentiometer was grease-packed to permit its use underwater. The potentiometer was made a part of an electrical circuit which included a paper chart recorder for external readout of vane angle position during test operations.

In order to evaluate the internal friction of the valve, the compression spring was first removed from the valve assembly and the external loading and measuring system shown in Fig. 10 was assembled for direct compression loading of the spring adjuster rod.

The valve vanes were forcibly returned manually to the fully closed or zero degrees of vane angle position. The external loading system was then brought into a zero loading contact with the end of the valve spring adjuster rod. Then the screw mechanism supporting the measuring balance was gradually raised to impose an increasing axial compression load on the spring adjuster rod. Increasing applied load failed to produce a smooth progressive movement of the valve vanes until a substantial load of 130 pounds had built up. At this point the frictional resistance of the valve mechanism was exceeded and the valve vanes suddenly moved to a new position and came to rest. Because the balance-lever loading system had only a limited travel before unloading, it could not apply a continuous load on the valve mechanism so only a limited angular travel occurred

in the valve vane motion before coming to rest. The angular position of the valve vanes was then reset to an arbitrary angle of about 2° and the loading system was re-adjusted to begin a new build-up of compression force again starting from zero. In this manner the frictional resistance was measured with the corresponding vane angle starting positions and break-away force values. The resulting measurements are plotted on Fig. 11.

Somewhat similar load tests were then made for a valve closing operation beginning with a 90° or wide open angle setting. In this case the valve assembly included the stiffest compression spring having a spring rate of 152.9 pounds per inch. In this case the closing force was supplied by the compression spring. The compression spring was in turn gradually loaded by screwing in the spring adjuster rod and carefully noting the external rod length at the time that the valve vanes achieved a frictional breakaway. The spring load at the time of breakaway was calculated from the spring rate value together with the spring compressed length as interpreted from the external length of the rod at breakaway. The values derived from these tests are also plotted on Fig. 11. The friction values appear random in nature.

Because of the extremely high friction forces encountered in both opening and closing the valve, it was judged necessary to re-align the various movable components of the valve. The valve was returned to the W. J. Woolley Company and re-aligned before any further testing.

After being re-aligned, the friction forces were reduced to such an extent that the weight of the valve vanes would open the valve. To keep the valve closed, compression was applied to the valve spring by turning in the spring adjuster rod. As the valve would open due to the weight of the vanes, the external loading system could no longer be used to measure forces involved in opening the valve. The procedure followed was to turn in or tighten the spring adjuster rod until the valve vanes were closed. Then the spring load was gradually reduced by turning out the adjuster rod until a light movement of the valve vanes was detected. At this breakaway point the external length of the rod, X , and vane angles

were measured. With X measured at breakaway and having pre-determined X at zero spring load, the force was computed using the appropriate spring rate. The forces were measured at about 15° increments from $0 - 60^\circ$. Just beyond 60° the end of the threaded valve vane adjusting rod was reached and the valve vanes would fall free to the fully opened position. Two different springs with rates of 120.7 and 90.9 pounds per inch were used in these tests. Tests were conducted with the valve mounted in the test box as shown in Fig. 10, but without the external loading system. An additional test variable was introduced by conducting the tests with and without snow packed around the valve as shown in Fig. 12 (b). The corresponding air temperatures in the test box were 67° and 32°F . The resulting forces versus valve vane angles are presented in Fig. 11.

Forces necessary to close the valve vanes were determined by using the same procedures followed in previous tests before the valve was re-aligned. The compression spring was gradually loaded by screwing in the spring adjuster rod and carefully measuring the rod length, X , and vane angles at the frictional breakaway point. Valve closing tests were conducted with the two springs used in the opening tests and at the same temperatures. Measurements were made starting with vane angles of about 88° and continuing up to 2° or almost closed. The resulting forces are also presented in Fig. 11. Examination of the results plotted in Fig. 11 supported the following conclusions.

- a. The forces required to open and close the valve were reduced considerably after the valve was re-aligned.
- b. After re-alignment, the forces necessary to open the valve were considerably lower than the forces to close the valve. This was primarily due to the weight of the valve vanes helping to overcome friction when the valve was being opened, but worked against the force closing the valve.
- c. The valve operated quite smoothly after re-alignment, both in opening and closing modes.

- d. The normal range of temperatures to be encountered in the full-scale lake installation does not appear to significantly influence the frictional force response of the model valve.
- e. The force data of Fig. 11 for the re-aligned valve appear orderly and rational. It should prove useful in design considerations for the full scale valve. Initial and long-term control of friction appears to be essential to good valve performance.

C. Opening Response to Hydrostatic Loading

To establish the response characteristics of the valve to an imposed differential water head, a test facility was constructed, as shown in Fig. 12(a). The 1:4 scale-model of the valve was mounted in the container box, as shown in Fig. 12(b). Initial friction test data, as was discussed earlier, were made with the set-up shown in Fig. 12(b). Subsequently, a wood flange plate was sealed to the top of the valve and a 12 inch diameter standpipe was attached and sealed to the flange, as shown in Fig. 12(c) to support a water column on the top side of the valve vanes. Additionally, a gauge glass was tapped into the valve chamber above the valve vanes to indicate and permit measurement of the static head existing in the standpipe. Leakage through the valve vane seals was sufficient to require a continuous water supply to maintain the standpipe head. Figures 12(a) and 12(d) show this 4 inch supply line and its control. Leakage through the valve was routed from the underside of the valve to a floor collector channel which conveyed it in turn to a flow weighing tank not shown. The container box permitted valve response measurements at either room temperature or alternately at a temperature reduced to near 32°F by subjecting the valve to an overnight snowpack before testing.

The opening response of the valve vanes to hydrostatic loading was observed while gradually increasing the head in the standpipe by use of the overhead supply control valve. Observation of a critical opening response of the valve was noted by a sudden falling

off of the head in the gauge glass and a sudden small change in the length of the spring compression rod. The latter change was detectable on a dial gauge mounted as shown in Fig. 12(a). Additional evidence of valve plate movement could be observed on the chart of the recorder shown in the foreground in Fig. 12(a). This chart (Fig. 13) showed the position of the potentiometer angle indicator affixed to the end of one of the valve hinge torsion bars. The total valve movement was only about 1° on the chart recorder and about 0.005 in. on the dial gauge. The movement occurred as a fairly sudden opening to a new stable position.

Once the valve vanes had opened, increasing the inflow would not bring the head back up to the head required to open the valve, as the valve capacity was more than could be provided by the 4 in. inflow line. How far the valve would open under a constant head was not established.

Tests as described above were completed for a wide variety of initial spring loading conditions for two different compression springs and for two different temperature conditions. The test results are presented in Figs. 14 and 15. Also shown are the spring loads in pounds corresponding to the X values and computed using the X value for no spring load and the measured spring rate. In tests of opening response using the 125 #/in. (measured at 120.7 #/in.) spring with the arrangement shown in Fig. 6 and the spring adjuster rod turned in so $X = 0$, not enough spring load could be applied to keep the valve vanes closed with 21 inches of hydrostatic loading. In order to run the tests, a 1.5 in. spacer was placed inside the spring housing next to the spring, as shown in Fig. 6. This has the effect of increasing X at no spring load from 3.20 in. to 4.70.

Examination of the results presented in Figs. 14 and 15 supports the following conclusions.

1. The data on hydrostatic head necessary to open the valve vanes versus X or spring loading plots in a reasonable

orderly manner. It should be useful in design considerations for the full scale valve.

2. Slight differences are noted between data taken at 64°F and 32°F but not enough to warrant the drawing of more than one line through the data. These differences are more noticeable at lower opening heads.
3. In comparing the spring load for the same head on the valve between Fig. 14 (152.9 #/in. spring) and Fig. 15 (120.7 #/in. spring) the latter is about 40 pounds higher. No explanation of this is readily available.

D. Leakage Tests

Initially it was observed that, even under high spring loadings and forcible closure of the valve vanes, the valve vanes were in contact with the resilient vane seal in only a portion of the seating length (a thin metal blade could be slipped through the seal for 75 percent of the seat length). Around the circular outer portion of the valve vane the vane overlapped the resilient seal a distance of about ½ inch but on the straight side of the vane the overlap was virtually zero. In consequence, when water was poured into the top side of the valve, it leaked through the seals at a high rate. This was reduced after realignment but the valve still leaked considerably. At about 20 in. of head on the valve, the leakage was measured in the Laboratory weighing tanks and found to be 0.288 cfs at 64°F and 0.286 cfs at 32°F. The temperature does not appear to affect the leakage rate. This information is presented in Fig. 15 as the leakage tests were made with 120.7 #/in. spring and spacer installed.

Steady State Flow Tests

A. The Steady State Flow Facilities

The later tests of the Woolley valve nearly all involved substantial flow and were conducted in a specially assembled facility mounted in the west volumetric basin at the St. Anthony Falls Hydraulic Laboratory. The installation is shown in the general plan

view of Fig. 16, the general elevation of Fig. 17, the valve and elbow detail of Fig. 4, and the photos of Figs. 18 and 19. The described mounting was intended to provide a test environment approximating the site flow constraints available at the intake of the Campbell Plant in Lake Michigan. The valve in the model approximates a 72 inch diameter full scale valve attached to an 8 foot diameter full scale header pipe.

The header pipe in full scale normally supports a row of seven attached screened inlet tees. In the model only one of these screen taps was provided, and the screen had been supplanted with a 12 inch control valve. Space limitations of the test set-up confined the header length to 5 feet before entering the connecting 24 inch discharge pipe. The discharge piping reduced to 12 inches and a separate 4 inch alternate. Each of these was provided with a rapid acting control valve and a flow metering orifice at the discharge point into a free surface tailwater control tank. The discharge system is shown in Figs. 17 and 19. The orifices were earlier provided with flow calibrations using the Laboratory's gravimetric tanks.

The upper pool level was maintained by supplying a flow slightly in excess of test needs through valved control of the 12 inch supply pipe shown in Fig. 16. The excess water discharged in a weir action over the crest of a 7 foot diameter overflow cylinder as shown in Fig. 17. This provided a stable and calm simulation of the lake surface for a wide variety of flow conditions together with a valve lip submergence of 7 ft-5 inches (29 ft + full scale).

Submergence of the 12 inch and 4 inch discharge lines below the surface of the tailwater control tank assured a constant operating head with a full pipe flow at all rates of flow from maximum to minimum. The differential head between the head and tailwater pools was about 18 feet. This head was in excess of that required to model maximum design flow for the Woolley valve which is 232 cfs full scale (7.25 cfs in a Froude model).

The valve as tested was fitted with a housed system for varying the length of the rod supplying compression to the damping spring which, in turn, loads the valve vanes. To expedite vane angle adjustment during the test program the valve rod mechanism was externally connected to an angle gear drive and vertical extension handle. This handle extended above the pool surface and could be adjusted from an overhead access bridge, as shown in Fig. 18.

An attached housing was also provided for a fluid time delay mechanism but the mechanism was not included during these essentially steady-state tests. This housing contained an electrical potentiometer attached to the end of one of the two hinge bars of the valve. The output of the potentiometer was connected to a paper chart recorder through a bridge circuit for external readout of valve vane angular position during test operations.

Headloss values in the valve tests were inferred from pressures measured with three pressure taps located as shown in Figs. 4 and 17. Pressure Tap #1 was located in the basin wall and provided a direct measure of the depth of the water in the basin. Tap #2 was located about five inches below the top of the valve vanes when closed and midway between the two valve vanes when in the opened position. Tap #2 provided a rough measure of the pressure just downstream of the valve. Tap #3 was located three inches downstream of the end of the 90° elbow. Differential pressure measurements between Taps #1 and #3 are considered a meaningful measure of the head drop across the valve and elbow as a unit.

Pressure values at Tap #2 were read with a strain-gauge pressure transducer mounted in the gauge line just outside of the valve as shown in Fig. 4. The backside of the transducer was fitted with a plastic tube which extended above the free surface of the pool and vented to atmosphere. In the static no-flow state this transducer then read the head of water at Tap #2 or about 8 ft-3 inches.

The pressure difference between Taps #1 and #3 was read with a manometer.

B. Vorticity Observations

The generation of vortices are a fairly common occurrence at water intakes located below a free water surface. In most cases they represent a problem only when the strength of the vortex is such that air is sucked into the intake. Such air may initiate troublesome pressure transients and system instability. Air sucking may be prevented by fitting the intake with a suitable hood. It was considered desirable to make vorticity observations on the modelled Woolley valve installation to determine whether vorticity was a serious problem and whether a protective hood needed to be developed in the model.

Test observations for vorticity were made on February 22, 1979. These observations were initiated with the pool level as shown in Fig. 17, and with the 12 inch discharge valve wide open and providing a maximum discharge. The vanes of the Woolley valve were positioned wide open.

No vorticity was observed for the initial maximum flow condition, but a modest level of turbulence existed in the test basin because of the inflow into the basin. Since a general turbulence usually inhibits the formation of vortices, the inflow was terminated and observations were continued under a gradually falling head in the basin.

A small vortex of about 3 inch surface diameter and $\frac{1}{2}$ inch stem diameter did form after the pool had fallen about one foot. This vortex did suck a small stream of air into the valve. The vortex existed for about a minute and then disappeared. Several small, weak, non-air drawing vortices did form and rapidly disappear as the pool level dropped 5 feet over a period of about 20 minutes at which time tests were terminated. The observed vortices were not considered detrimental to further valve testing and an available anti-vortex hood was not installed on the valve.

A similar demonstration test with similar findings was repeated on March 6, 1979 for visitors from Consumers Power Company, Commonwealth Associates, and Johnson Division UOP. It was generally agreed that air sucking vortices were infrequent and weak and unlikely to pose a problem in full scale operations. Full scale operations would be additionally protected by a proposed hood for interception of debris.

C. Slow Opening and Closing Vane Loading Tests

In order to promote a better understanding of the response of the spring loaded vanes of the Woolley valve to pressure loadings developed by normal pressures and flow processes, two series of tests were conducted. In the first of these tests, which are described here, the valve was initially subjected to an increasing differential head which initiated opening and a falling head which initiated closure. Rates of change of head in these tests were quite slow and nearly steady-state flow conditions were approximated for obtaining the co-related values which characterized a slow valve performance. In these tests the hydraulic time-delay mechanism was not installed.

A general requirement for the full-scale valve response was that the valve should not open for imposed valve differential pressure conditions which are consistent with normal operation of a clean intake screen system (a 9-inch differential water head at the valve as established by other tests). However, the valve should open if plugging of the screens imposes an excessive additional head drop (an additional differential pressure at the valve of 12 inches as determined by other system criteria or a total differential of 21 inches). The valve also fully opens with another 12 inches additional head and should begin closure when the differential head falls below 21 inches. Details of the opening and closing response criteria were given previously on pages 2 and 3.

It was further specified that the model valve should pass the maximum design model discharge of 7.25 cfs when subjected to a

maximum available differential pressure head of 27 inches of water. (it should be noted that the model valve was designed to operate in response to full scale differential head values of 21 and 27 inches of water while processing scale-modelled discharges.)

The foregoing pressure criteria relate to slowly imposed or essentially static pressure values. Additional criteria with regard to valve response under dynamic or rapidly changing head conditions have also been evolved and will be considered in a subsequent discussion of a dynamic testing program in which the valve was equipped with time-delay controls.

For the slow rate flow response tests, the Woolley valve was placed in a submerged mounting in the test basin as shown in Fig. 17. Differential pressure heads were imposed on the valve by selected operations of the 12 inch screen closure valve (hereinafter referred to as the S/C valve) within the basin and the 12 inch discharge valve on the external 12 inch piping system. These large valves were opened and closed by pneumatic actuators controlled by small manually operated valves. By first fully opening the S/C valve, gradually opening the discharge valve, and then gradually closing the S/C valve, any desired differential pressure head could be progressively imposed on the Woolley valve vanes. A reverse sequence of control valve changes could then progressively impose diminishing differential heads across the Woolley valve until closure was completed.

The differential pressure imposed on the vanes of the Woolley valve by the above sequence of operations was conveniently measured by employing the strain gauge transducer connected to Tap #2 as previously described and shown in Fig. 4.

In a typical test run the above described control valve operations were initiated in order to impose a slowly increasing and then a slowly decreasing differential pressure. Throughout this sequence a strip-chart of the differential pressure head was recorded together with the response of the valve vanes as measured by their angular position. Figure 20 is typical of the data recorded by this procedure.

The following system and valve characteristics may be noted in Fig. 20.

1. The pneumatically actuated flow control valves were somewhat irregular and jumpy in action and did not permit a smooth and gradual change in the differential pressure imposed on the Woolley valve. Freezing weather conditions did not permit the use of hydraulic actuation of the flow control valves to provide smoother action.
2. Flow conditions in the discharge systems impose a slightly oscillating pressure on the underside of the valve vanes even before the valve opens and a strongly oscillating pressure after the vanes open.
3. The damping characteristics of the valve vane system are such that the valve vanes appear to slowly seek a position consistent with some average value of the pressure. When the average value of the pressure remains constant, the valve vanes achieve a stable and non-fluttering position.
4. Initial opening and especially final closure of the valve vanes are quite rapid, being less than about two seconds.

The high frequency response of the strain gauge pressure transducer produced an erratic record which made evaluation of the mean pressure difficult. In order to improve readout, electrical damping of the transducer signal with a one-second time constant was used and resulted in new pressure charts typified by Fig. 21.

Using the described techniques and facilities, six test events were recorded and concurrent values of the differential pressure and vane openings in degrees were taken from the chart records. These values were in turn plotted as shown in Figs. 22, 23, and 24.

In the latter tests the water temperature was approximately 33^oF and the spring values and spring compression test setting, X, are as shown in the figures. The spring values involved were a nominal 125 pounds per inch (measured 120.7 pounds per inch) and a nominal 150 pounds per inch (measured 152.9 pounds per inch) as obtained in the earlier static tests.

The spring compression is measured by an arbitrary adjustment setting X on the valve as shown in Fig. 6. The adjustment X and the observed related spring compression loadings and incipient static opening values are shown in Fig. 14 and 15.

Figure 22 shows the opening and closing response of the Woolley valve for two sets of tests with a spring of 120.7 pounds per inch set with X equal 1.3 inches. (From Fig. 15 the incipient opening condition for a 21 inch imposed static head would occur with $X = 1.3$ inches and would involve a compression spring load of ≈ 420 pounds.) The data of Fig. 22 in one case relate to the valve exposed to an arbitrary maximum ΔP value of about 30 inches based on a charted event of the character of Fig. 21. In the second case ΔP achieved a maximum value of about 43 inches.

It should be noted in Fig. 22 that the breakaway or opening ΔP was very close to the value statically determined in Fig. 15, but the pressure fell significantly and then stabilized as the valve started to open. Further increases in ΔP were accompanied by an increasing vane opening. At an arbitrary halt in ΔP at about 30 inches, decreasing ΔP values were then imposed. Little change in the angle of opening of the vanes occurred under the decreasing pressure until a critical pressure condition was reached and then the vanes quite suddenly moved to complete closure.

In the second test, also shown in Fig. 22, the increasing pressure conditions were continued to a higher ΔP before beginning to decrease. The initial opening conditions for this test were not clearly defined but the values for larger vane openings roughly duplicated the previous test before continuing up to an arbitrary maximum of ΔP 43 inches and opening at $\approx 63^\circ$. Closure under diminishing ΔP again began with little vane angle response until a critical value of ΔP triggered a rapid closure

In additional tests with the same spring, and shown as Fig. 23, the X setting was adjusted to 0.6 inches. (From Fig. 15 the incipient valve opening condition with $X = 0.6$ should occur with $\Delta P = 26$ inches and should involve a compression spring load of 490 pounds.)

It should be noted from Fig. 23 that the breakaway or opening did not occur at $\Delta P = 26$ inches but rather at a value of $\Delta P = 24$ inches. Two repeats of this test condition, as shown in Fig. 23, indicate the general trend of the vane response but show considerable instability existing in opening response. Closure on the other hand shows a considerable reproducibility in the response although again little closure response occurs until a critical value of ΔP is reached.

A third test condition was then imposed by replacing the spring with a rate of 120.7 pounds per inch with one having a rate of 152.9 pounds per inch and setting $X = 0.95$ inches (from Fig. 14 the incipient opening condition for a 21 inch imposed static head should occur with $X = 0.95$ inches and should involve a compression spring load of ≈ 370 pounds). Figure 24 shows the opening and closing response of the valve for these conditions in three separate tests involving some variations in the maximum ΔP imposed. The general shape of the response curves of Fig. 24 resemble those in Figs. 22 and 23 but the breakaway conditions on opening depart considerably from the static test values of Fig. 14. Closure also occurs at much less than the design value of 9 inches.

A review of Figs. 22, 23, and 24 support the following conclusions:

1. The nature of the spring loading and friction of the valve vane system together with inherent dynamic surging pressures in the piping system downstream of the valve leads to significant variations in the opening response of the valve.
2. The nature of the spring loading system for the valve vanes provides adjustments which permit the valve to open at a pressure which roughly approximates the specified value of 21 inches of water head. However, this spring adjustment is too stiff to permit the valve to fully open at the specified maximum head of 27 inches. This raises a question whether or not the valve can pass the design discharge.

3. The nature of the spring loading system for the valve vanes leads to a rapid snap-action closure of the valve.
4. It appears that the spring loading system as tested is incapable of satisfying all of the specified operating conditions.
5. Additional tests are necessary to establish the discharge or headloss constraints of the valve.

D. Combined Vane Loading and Headloss Tests

The foregoing tests supplied considerable data on response of the Woolley valve to selected flow loading conditions. Because of cold weather conditions, the tests precluded manometric measurements of flow discharge and manometric measurements of headloss using pressure Taps #1 and #3. With the advent of occasional above-freezing weather conditions a new program of tests was evolved to supplement the slow opening and closing tests previously discussed.

In the first of these tests conducted on March 13, 1979, the 12 inch inlet valve (S/C) remained closed and all flow control was by adjustments with the 12 inch discharge valve. These flows were then evaluated using the discharge orifice and outside manometer shown in Figs. 17 and 19. Concurrent with the discharge determination, values of vane angle position were read with the potentiometer, and values of the pressure Tap #2 were read with the strain gauge transducer.

The following procedures were observed in running these tests:

1. The spring with the 120.7 #/in. rating was installed in the Woolley valve and the spring mechanism was in accord with Fig. 6. The time-delay mechanism was not installed during these tests.
2. The spring compression was relaxed so that the valve vanes came to the wide open position ($\approx 88.25^\circ$) and rested on the vane stops.

3. The basin pool was filled to the normal lake stage.
4. The 12 inch inlet valve (S/C) was closed and the 12 inch discharge valve was opened fully.
5. Concurrent readings were then recorded for the flow discharges, the vane angle opening, and the ΔP between normal pool and Pressure Tap #2. The values were entered in the data table. No spring loading force was involved in this test.
6. The discharge valve was closed and the flow brought to zero.
7. The spring compression rod of the Woolley valve was rotated with the remotely controlled adjusting system until the spring was brought to a zero compression condition and then 10 additional turns were made on the valve rod ($\frac{1}{2}$ inch of compression).
8. The 12 inch discharge valve was quickly opened to the wide open condition.
9. The Woolley valve responded to the flow and opened to a stable position. Concurrent values of flow discharge, vane angle opening and the ΔP were recorded and entered in the data table. The concurrent spring force was calculated and entered in the table (see footnote of Table).
10. The 12 inch discharge valve was closed and the flow brought to zero.
11. The 12 inch discharge valve was quickly opened approximately halfway.
12. Items 9 and 10 were repeated in sequence.
13. The 12 inch discharge valve was quickly opened one quarter open.
14. Items 9 and 10 were repeated in sequence.
15. The spring was compressed an additional $\frac{1}{2}$ inch.

16. Items 8 through 15 were repeated.
17. Items 8 through 15 were repeated again.
18. Items 8 through 15 were repeated again.
19. Items 8 through 15 were repeated again.
20. Under Item 19 the initial Woolley valve condition before imposing flow was essentially a closed position.

The data acquired by the foregoing procedure were tabulated as shown in Table I. The ΔP values of Table I were then converted to equivalent ΔH values expressed as feet of headloss and then related to the discharge Q in cfs by the arbitrary discharge equation $K = Q/\sqrt{\Delta H}$. These computed values are listed in Table II and graphically plotted in Fig. 25.

Studies of the data leading to Fig. 25 led to the decision that additional tests should be conducted to establish the ΔH values between pressure Taps #1 and #3 as well as between Taps #1 and #2. These tests were also to include data with weaker and stronger spring adjustments. The resulting tests which were conducted on March 28, 1979 yielded data which are listed in Table III and gave computed values which are listed in Table IV and graphically plotted in Figs. 26 and 27.

Figures 26 and 27 are useful generalized data for future valve design modifications or for system discharge determinations. It should be noted that the ΔH values resulting from the tests are not headloss values in the conventional sense since they represent pressure head differentials rather than energy head differentials. This interpretation is also the reason why the values of ΔH_{1-2} (involving a flow constriction in the vicinity of Tap #2) are always larger than ΔH_{1-3} (involving a large pipe cross-section at Tap #3).

Additional useful information can be derived from the data of Table I, II, and III by alternate plottings which give emphasis to the relation between the vane opening and the torque required to maintain the vane opening. Torque values may be derived from the data when it is recognized that the pitch diameters of the gears shown in Fig. 6 are 2 inches, thus permitting the recorded spring

forces to act at a one inch radius arm on the vanes. For these particular dimensional conditions the recorded spring force values are also vane torque values in inch pound units. Fig. 28 employs this reasoning to graphically depict the relation between vane opening and the applied torque under the imposed flow conditions of the tests of 3/13/79 as recorded in Table I. Fig. 29 similarly depicts the relation for the imposed flow conditions of the tests of 3/28/79 as recorded in Table III.

Study of the findings of the tests of 3/13/79 and 3/28/79 led to the conclusion that the available data would serve to establish a new design for the valve torque mechanism and further tests with the mechanism of Fig. 6 would not be necessary.

TABLE I

Woolley Valve Tests - Data of 3/13/79

Procedure Item	Q (cfs)	Vane Opening (degrees)	ΔP (in. of water from pool surf. to Tap #2)	Spring compression x dimension (inches)	Spring Force* (pounds) [Also torque in inch pounds]
4-5	8.2	88.25°	14.4	0	0
6-10	8.2	76°	19.3	½	31.25
11-12	6.6	75°	13.9	½	29.2
13-14	3.6	73°	4.3	¾	25.0
15-16	8.2	73°	20.9	1	85.4
	6.2	69°	13.9	1	77.4
	3.0	62°	7.5	1	62.4
17	8.2	69°	26.2	1½	137.1
	5.8	63°	19.8	1½	125.1
	3.0	54°	11.0	1½	106.1
18	8.2	67°	29.4	2	193.8
	6.2	61°	26.8	2	180.8
	3.6	52°	18.7	2	162.8
19	8.0	62°	42.8	2½	243.5
	6.2	57°	35.3	2½	233.5
	3.3	48°	21.4	2½	214.5

* Spring Force = spring compression at fully open x 120.7 $\frac{\text{lb}}{\text{in}}$ - $\left[\frac{(90^\circ - \text{vane opening})}{90^\circ} \right]$
 $\times 1.55" \times \frac{120.7 \text{ lb}}{\text{in.}}$

TABLE II

Woolley Valve Tests - Computed Discharge Coefficient
Tests of 3/13/79

$$K = Q/\sqrt{H}$$

Vane Opening (degrees)	Q (cfs)	ΔH Taps 1-2		K_{1-2}
		(inches)	(feet)	
88.25	8.2	14.4	1.20	7.48
76	8.2	19.3	1.61	6.46
75	6.6	13.9	1.16	6.13
73	3.6	4.3	.36	6.01
73	8.2	20.9	1.74	6.21
69	6.2	13.9	1.16	5.76
62	3.0	7.5	.62	3.79
69	8.2	26.2	2.18	5.55
63	5.8	19.8	1.65	4.51
54	3.0	11.0	.92	3.13
67	8.2	29.4	2.45	5.24
61	6.2	26.8	2.23	4.15
52	3.6	18.7	1.56	2.88
62	8.0	42.8	3.57	4.24
57	6.2	35.3	2.94	3.61
48	3.3	21.4	1.78	2.47

TABLE III

Woolley Valve Tests - Data of 3/28/79

Q (cfs)	Vane Opening (degrees)	ΔP Tap #2 (in. of water)	ΔH Taps #1-#3 (in. of water)	Spring* Force (pounds) [Also torque in inch pounds]
8.3	88.25	14	8.4	0
7.0	88.25	10	6.0	0
5.2	88.25	6.0	2.7	0
4.1	88.25	3.0	2.1	0
8.3	81	16	11.1	15
7.1	81	13	8.2	15
5.2	81	7.5	4.7	15
4.1	81	4.5	3.1	15
8.3	78	18	12.7	39
7.2	77	14	10.1	37
5.2	76	10	5.9	35
4.0	75	4.0	3.5	33
8.3	74	23	15	91
7.1	70	19	13.5	83
5.2	68	13	9.0	79
4.1	67	9	6.4	76
8.2	70	25	18.1	143
7.1	66	23	17.4	135
5.2	64	18	11.1	131
4.1	57	17	10.1	116
8.0	64	35	22.8	191
7.1	60	38	24.0	183
5.2	57	27	16.2	176
4.1	52	25	15.6	166
7.9	63.5	43	25.7	250
7.0	63	30	20.5	249
5.2	58.5	27	16.2	240
4.1	58	18	10.5	239
6.2	38	135	98.5	258
5.2	37	105	77	256
4.1	36	70	34	253

*Spring Force = spring compression at 88.25° x 120.7 lb/in.

$$- \left[\frac{88.25^\circ - \text{vane opening}}{90^\circ} \right] \times 1.55" \times 120.7 \text{ lb/in.}$$

TABLE IV
 Woolley Valve Tests - Computed Discharge Coefficients
 Tests of 3/28/79

$$K = Q/\sqrt{H}$$

Vane Opening (degrees)	Q (cfs)	ΔH_{1-2}		K_{1-2}	ΔH_{1-3}		K_{1-3}
		(inches)	(feet)		(inches)	(feet)	
88.25	8.3	14	1.17	7.68	8.4	0.70	9.92
88.25	7.0	10	0.83	7.67	6.0	0.50	9.90
88.25	5.2	6.0	0.50	7.35	2.7	0.22	10.96
88.25	4.1	3.0	0.25	8.20	2.1	0.17	9.80
81	8.3	16	1.33	7.19	11.1	0.92	8.63
81	7.1	13	1.08	6.82	6.2	0.52	8.59
81	5.2	7.5	0.62	6.58	4.7	0.39	8.31
81	4.1	4.5	0.37	6.70	3.1	0.26	8.07
78	8.3	18	1.50	6.78	12.7	1.06	8.07
77	7.2	14	1.17	6.67	10.1	0.84	7.85
76	5.2	10	0.83	5.70	5.9	0.49	7.42
75	4.0	4.0	0.33	6.96	3.5	0.29	7.41
74	8.3	23	1.92	3.37	15	1.25	7.42
70	7.1	19	1.58	5.64	13.5	1.12	6.69
68	5.2	13	1.08	5.00	9.0	0.75	6.00
67	4.1	9	0.75	4.73	6.4	0.53	5.61
70	8.2	25	2.08	5.68	18.1	1.51	6.68
66	7.1	23	1.92	5.13	17.4	1.45	5.90
64	5.2	18	1.50	4.25	11.1	0.92	5.41
57	4.1	17	1.42	3.44	10.1	0.84	4.47
64	8.0	35	2.92	4.68	22.8	1.90	5.80
60	7.1	38	3.16	3.99	24.0	2.00	5.02
57	5.2	27	2.25	3.47	16.2	1.35	4.48
52	4.1	25	2.08	2.84	15.6	1.30	3.60
63.5	7.9	43	3.58	4.17	25.7	2.14	5.40
63	7.0	30	2.50	4.43	20.5	1.17	5.36
58.5	5.2	27	2.25	3.47	16.2	1.35	4.48
58	4.1	18	1.50	3.35	10.5	0.87	4.38

Dynamic Tests

A. Interim Valve Modifications and Adjustments

As a consequence of the tests of March 1979 at the St. Anthony Falls Hydraulic Laboratory, the Woolley valve was returned to the W. J. Woolley Company for modifications and additions. The valve was returned to the Laboratory on May 28 and the following changes were noted:

1. The earlier linear force rod system (valve actuating rod) shown in Fig. 6 was replaced by a variable torque arm linkage clevis arm, as shown in Fig. 7. This included an adjustable setting for establishing the maximum valve vane opening. In subsequent tests this was set to provide a maximum opening of about 64° .
2. The driver pinion gear in Fig. 7 was provided with an extension and external lever to permit manual application of opening and closing forces as needed in the test program.
3. The hydraulic control circuit with control valves and accumulator, as shown in Fig. 8, was temporarily mounted externally on the time-delay mechanism housing for ready access and adjustment.
4. A bourdon pressure gauge was included in the time-delay circuit, as shown in Fig. 8, to aid in clarifying the operation of the circuit.
5. The valve was provided with the damper spring shown in Fig. 7 having the 152.9 lbs/in. rate value. This spring was retained throughout all subsequent tests.

B. Final Adjustments, Modifications, and Test Criteria

A number of preliminary tests were conducted early in June 1979, and resulted in a demonstration and coordination meeting of concerned parties at the Laboratory on June 13, 1979. Tentative procedures were established at that time for completing the tests, but subsequent

tests indicated that internal friction in the time-delay mechanism was excessive. This mechanism was then shipped to the W. J. Woolley Company for rework (consisting mainly of replacing seal rings and sealing tapes) and was returned on June 27, 1979.

Among other findings of the early June tests was evidence that the new variable torque conditions provided by the revised mechanical linkage shown in Fig. 7 did not provide adequate closure force. Tests with the time-delay mechanism removed from the Woolley valve showed that closure could not occur with normal spring adjustments if the angle of opening exceeded about 48° .

Tests with the reconditioned valve indicated inadequate valve response under design flow conditions (only partial opening after 25 minutes of flow exposure). The delay cylinder was disassembled at the Laboratory and the return spring, shown in Fig. 8, was subjected to loading tests. These tests established a spring rate value of about 20 lbs/in. Measurement of the piston-cylinder components in the delay cylinder indicated that the spring was under about 40 pounds of compression when the time-delay piston was seated and under about 50 pounds of compression when the piston was fully unseated. This spring was replaced by an available spring at the Laboratory, having a rate of about 5 lbs/in. The replacement spring provided a compression of about $10\frac{1}{2}$ pounds when the piston was seated and a compression of about 13 pounds when the piston was fully unseated. The latter spring was employed in all subsequent tests.

The damper spring of the valve (152.9 lbs/in. spring rate) in the final tests was adjusted to produce opening response to an imposed differential head of 21 inches of water with the time-delay mechanism disconnected from the valve. The head in this case was provided by gradually increasing the stage in the volumetric basin until a head of 21 inches existed above the top side of the closed valve vanes while the 12 inch discharge control valve remained open. The damper spring under these conditions had an X value of 2-5/16 inches (approximately 160 lbs).

The final imposed objective test conditions were as follows:

1. The normal system operating exposure of the Woolley valve was to be 9 inches of differential head. No opening action was to occur under this head or heads up to an additional head of 12 inches or a total of 21 inches.
2. At 21 inches of differential head the valve should begin to open after 30 seconds of reaching 21 inches.
3. The valve vane should begin to open for values of ΔH in excess of 21". The degree of opening is to be dependent on the magnitude of the ΔH in excess of 21". For the prototype design flow of 232 cfs (7.25 cfs model flow) the total ΔH is not required to exceed 27" (21"+6"). From Fig. 27, the vane opening for 7.25 cfs flow through the model valve is estimated to be 60°.
4. The conditions of Items 2 and 3 above and the performance criteria previously given on page 3 serve to define two points on a hypothetical curve depicting the time-response for vane opening. The two points so defined are Point A with $\Delta H = 33$ inches when opening time is 5.5 minutes and Point B with $\Delta H = 141$ inches when opening time is 1.0 minutes. Figure 30 is a rough estimate of the type of curve of ΔH versus time that might result.

C. Final Dynamic Flow Tests

1. Single Cycle Tests

For the final tests, pressure differential head ΔP across the Woolley valve was measured with the recording pressure sensor located at manifolded Pressure Tap #3, as shown in Fig. 17. The back side of the diaphragm type sensor was exposed to atmosphere. The test head was imposed by employing the full overflow head in the volumetric basin as shown in Fig. 17. The tests were initiated with the 12 inch inlet control valve (S/C) wide open and the 12 inch discharge control valve closed. The pressure sensor was zeroed under these conditions.

The initial differential pressure head imposed in the test resulted from quickly and fully opening the 12 inch

discharge control valve. This results in a differential head on the Woolley valve of about 15 inches (see Fig. 31). Subsequent head increases were then made by staged closures of the 12 inch inlet control valve (S/C).

A number of tests were run and chart recorded as typified in the example of Fig. 31. The following imposed conditions and observed response are significant.

- (1) Control of the imposed differential head was by manual incremental manipulation of the 12 inch flow inlet valve (S/C) and led to a somewhat ragged pressure curve.
- (2) The opening response curve showed an immediate slight response to the initial pressure loading of about 15 inches ΔP .
- (3) True opening response began as it should when the pressure approximated 21 inches of head.
- (4) The initiation of true opening was much more rapid than specified (about 14 degrees in 10 seconds) indicating substantial leakage or blow-by in the hydraulic time-delay circuit. It is speculated that this may be due to low or non-sealing hydraulic pressure valves in the rack cylinder during the initial head buildup. (Figure 31 does not show the visually read pressure buildup values in the first 45 seconds of true opening but records from other runs show that slow rate opening was not achieved until a pressure buildup of about 10 psi occurred.)
- (5) Following the rapid rate of opening discussed in Item (4) in Fig. 31, the rate rather suddenly diminished to a low value presumed indicative of a normal time-delay function. This rate of rise led to an increase of about 10 degrees (a total of 24^o) in opening angle in about 105 seconds.

- (6) Following the normal delayed opening function, the valve rapidly opened to about 64° in an additional 70 seconds.
- (7) Subsequent staged reduction in ΔP to the initial value failed to produce any evidence of valve closure despite fall-off to zero in the hydraulic gauge pressure.
- (8) A test run, similar to that shown in Fig. 31, but with the #2 delay valve closed, showed essentially the same response as in Fig. 31 with the #2 valve open wide.

In a separate flow test, it was established that the 1:4 model Woolley valve with a pressure differential head of 26 inches and with a vane opening position of 64° would pass a discharge of 8.4 cfs. This value is in excess of the 7.2 cfs required at 27 inches of differential head in order to satisfy Froude modelling criteria. It is reasonable to expect that the full scale valve would pass an even greater discharge than is represented by scaling the 8.4 cfs of the model.

2.. Wave Simulation Tests

Limited flow tests were also made on the response of the Woolley valve to differential pressure cycling. These tests are represented by the chart records of Figs. 32 and 33.

Figure 32 simulates a 12 foot or 144 inch differential pressure head cycle imposed on the Woolley valve with a cycle period of about 25 seconds. The basic pressure control employed the 12 inch discharge control valve in a wide open position and the 12 inch inlet control valve (S/C) cycled from a wide-open condition to a pre-tuned closure which would produce the desired 144 inch differential on the pressure recording. (Note that the pressure scale in Figs. 32 and 33 are different than in Fig. 31.) The following imposed test conditions and observed response are significant:

- (1) Control of the imposed differential head was with manual manipulation of the operator of the 12 inch inlet valve (S/C). A high degree of control was neither possible nor necessary.
- (2) The zeroing of the pressure sensor was with static conditions provided by closure of the 12 inch discharge control valve.
- (3) The initial plateau in the pressure record resulted from full opening of both the inlet and discharge control valves and provided about a 20 inch differential pressure head, ΔP , across the Woolley valve.
- (4) Imposition of additional differential pressure by throttling the inlet valve led to an immediate indication of a fairly rapid opening of the vanes of the Woolley valve to about 12° over a period of about 5 seconds. As with the single opening event previously described, this rapid initial opening indicates an abnormal leakage or blow-by in the time-delay hydraulics.
- (5) The sustained high pressures near the peak of the pressure wave record caused a continuing but smaller rate of opening of the vanes of the Woolley valve. The rate of this opening in Fig. 32 is considerably greater than the rate shown in Fig. 31 and this is to be expected because of the higher differential head involved.
- (6) A sudden rapid fall-off in the differential head produced a concurrent drop in vane opening angle but the closure was not completed in the sag of the pressure curve. It should be noted that the imposed pressure wave is not a true simulation of a gravity pressure wave in a lake setting.

The lake wave varies plus and minus from its normal at-rest or zero level, whereas the test wave varies from a high positive value to a lower positive value relative to an at-rest or zero value. As a result of the test condition, the vane record shows a progressive opening of the valve as the wave action continues. It is possible that a corresponding fully developed negative response would keep the valve closed. This is because the negative pressure would impose a closing force to the hydraulic control mechanism.

- (7) Because of the progressive opening of the Woolley valve, the limitations of the head-discharge conditions of the test set-up led to a gradual diminishing of the pressure peak of the wave cycles.

Figure 33 shows the response of the Woolley valve to a cycling of the differential head with a lower wave amplitude (40 to 60 inches of head rather than 144 inches) and a shorter period (about 10 seconds rather than 25 seconds) than shown in Fig. 32. This is a more realistic simulation of a 100 year storm wave. The following imposed test conditions and observed response are significant:

- (1) The opening response of the valve is substantial in the first second of pressurization.
- (2) After the first second of pressurization, the opening response is slow but progressive.
- (3) The test is not conclusive.

Following completion of the cycling tests, a decision was made by Commonwealth Associates to terminate the tests.

Conclusions

The modified Woolley valve exhibits the following flow response characteristics:

1. The valve can be adjusted to initiate opening action at any selected head differential value including the specified value of 21 inches. Opening will not occur for heads less than the selected setting.
2. The time-delay mechanism, although operating, does not properly delay the valve opening for the initial 10 or 20 degrees of opening, but settles out to specified rate values for larger angles of opening.
3. The valve will open to a specified 64° of opening within 5 minutes when exposed to a 33 inch differential head.
4. At a 64° angle of opening and a 27 inch differential head the valve will pass the design discharge (232 cfs full scale, 7.2 cfs model).
5. The damper spring linkage mechanism, when set to provide a full opening of 64° , will return to closed only from an opening position of 50° or less.

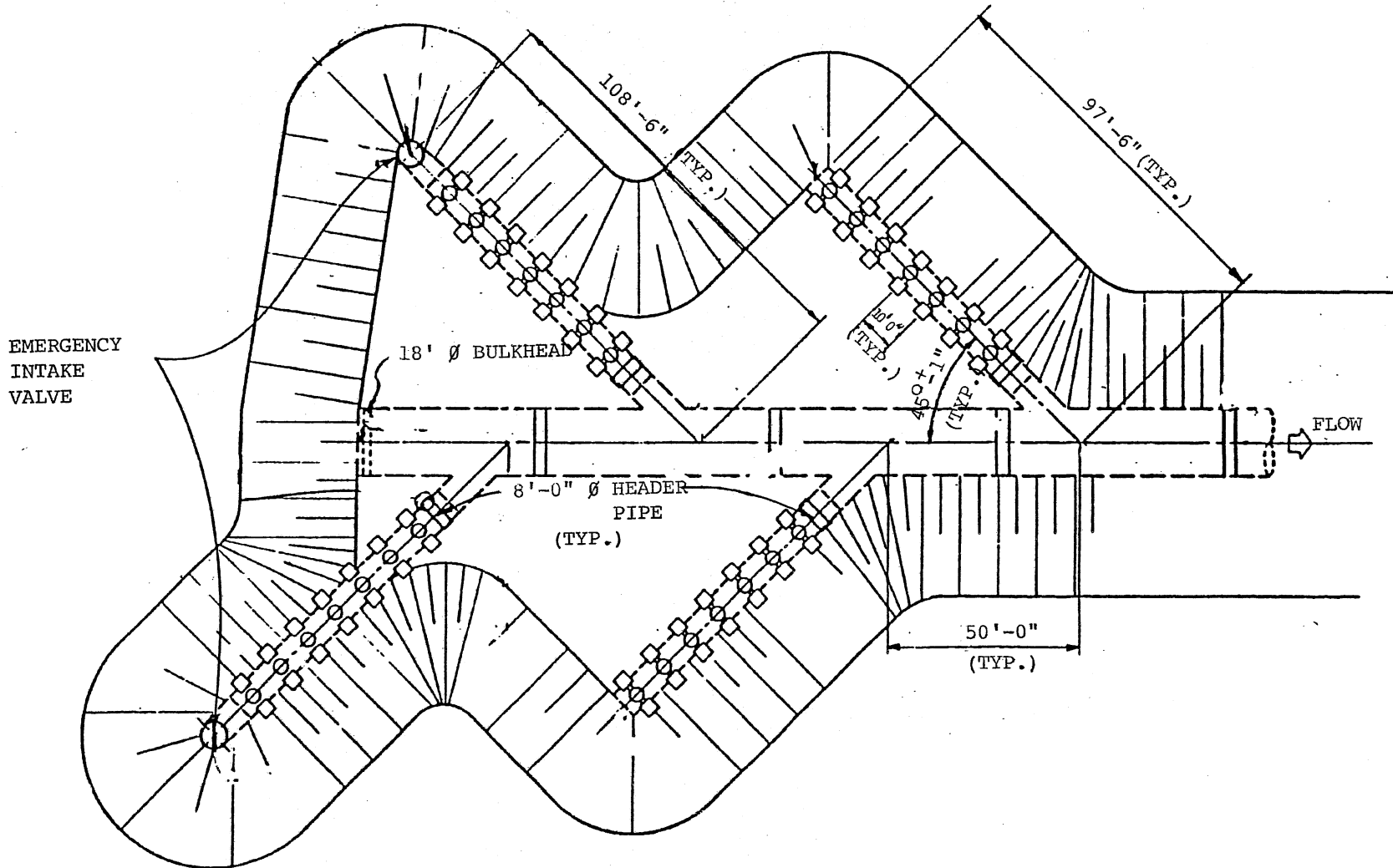


Fig. 1 - General Plan View of Proposed Intake System.

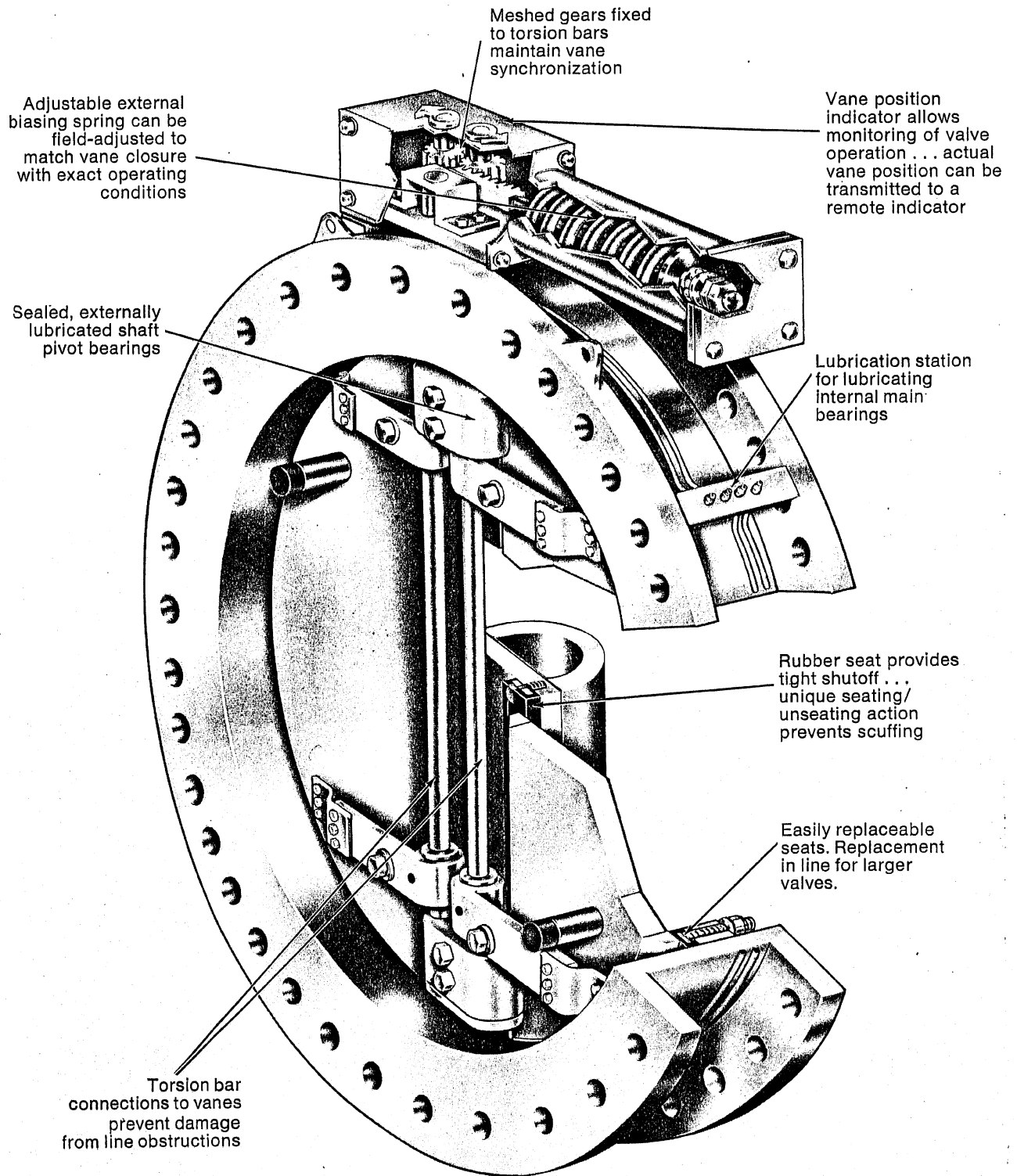


Fig. 2 - General Nature of the Woolley Standard Synchro-Chek Valve.

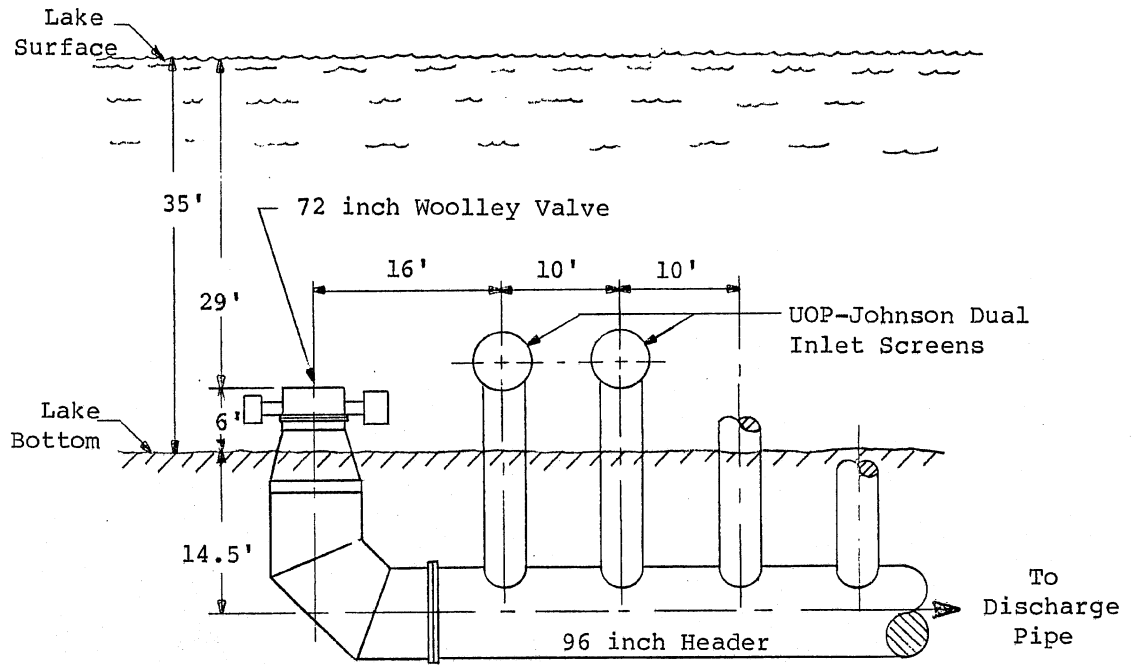
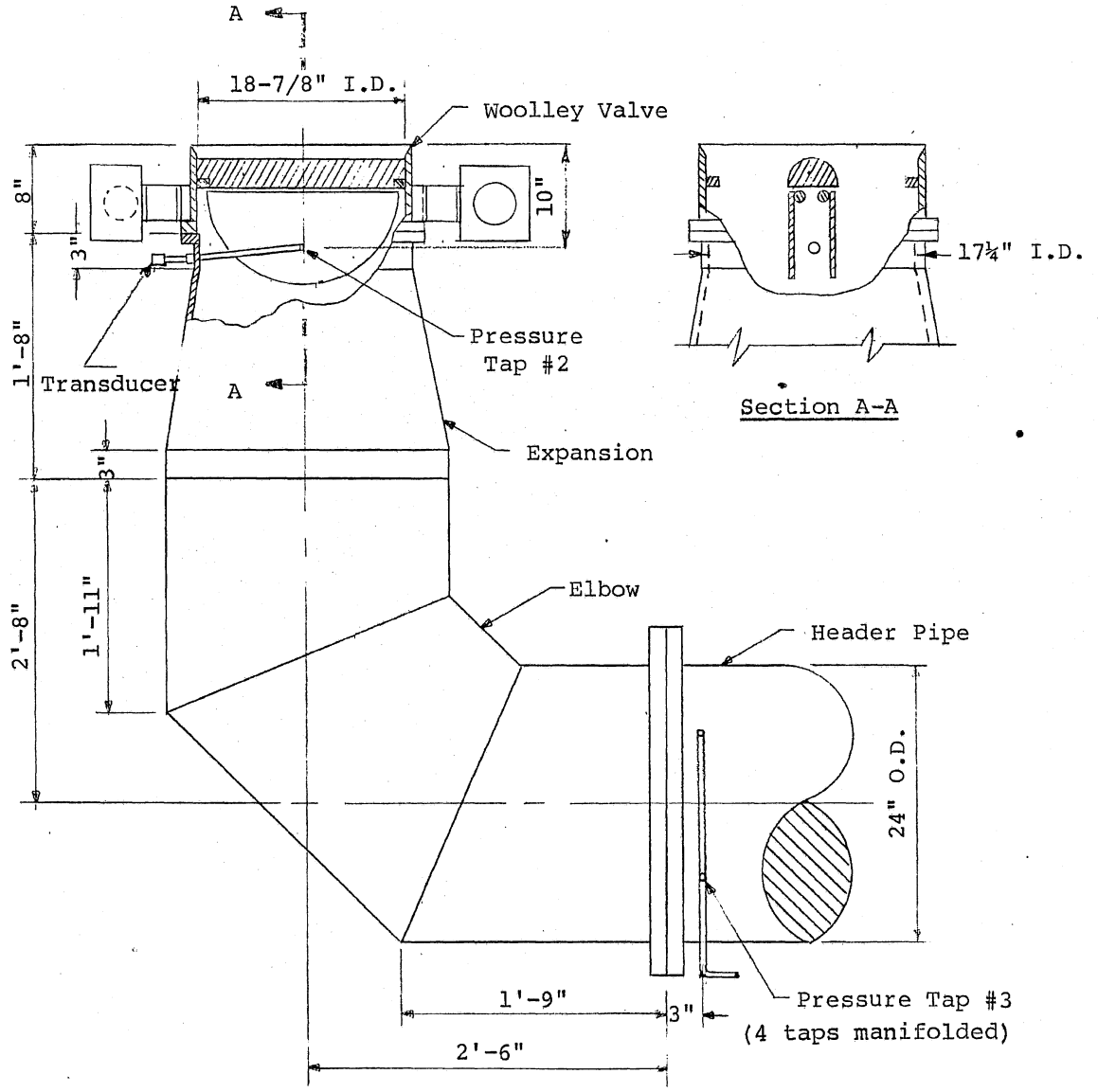
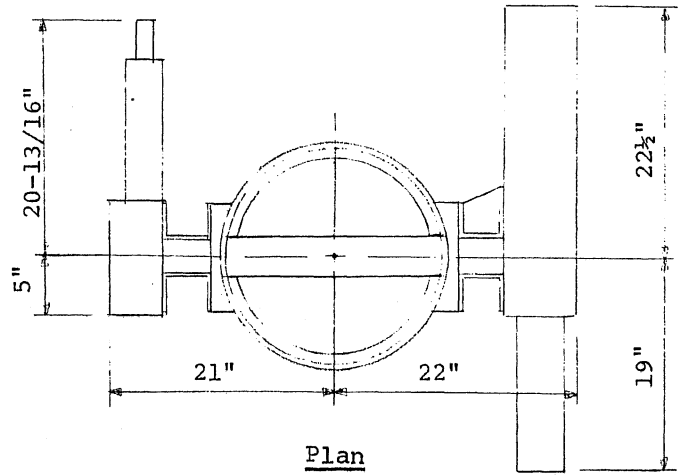


Fig. 3 - Installation of the Full Scale Woolley Valve at the Campbell Plant Intake - Lake Michigan.



Elevation

Fig. 4 - Original Design of the Woolley 1:4 Scale Valve Model and Elbow Model Attachment to the Header.

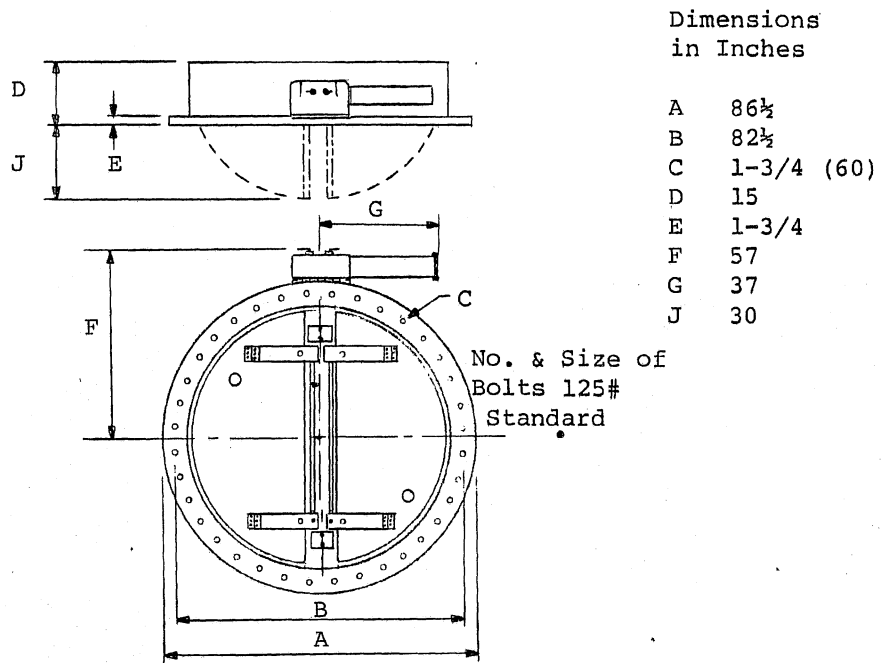


Fig. 5 - Approximate Dimensions of the Full Scale Woolley Standard Synchro-Chek Valve (from W.J. Woolley Co., Bulletin 103-73).

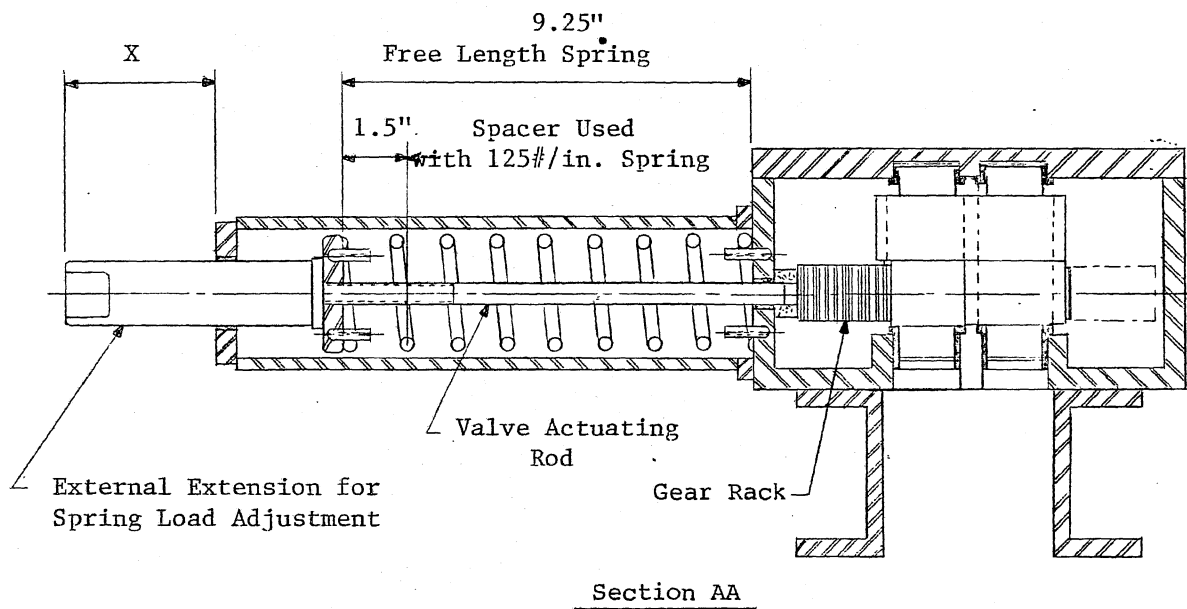
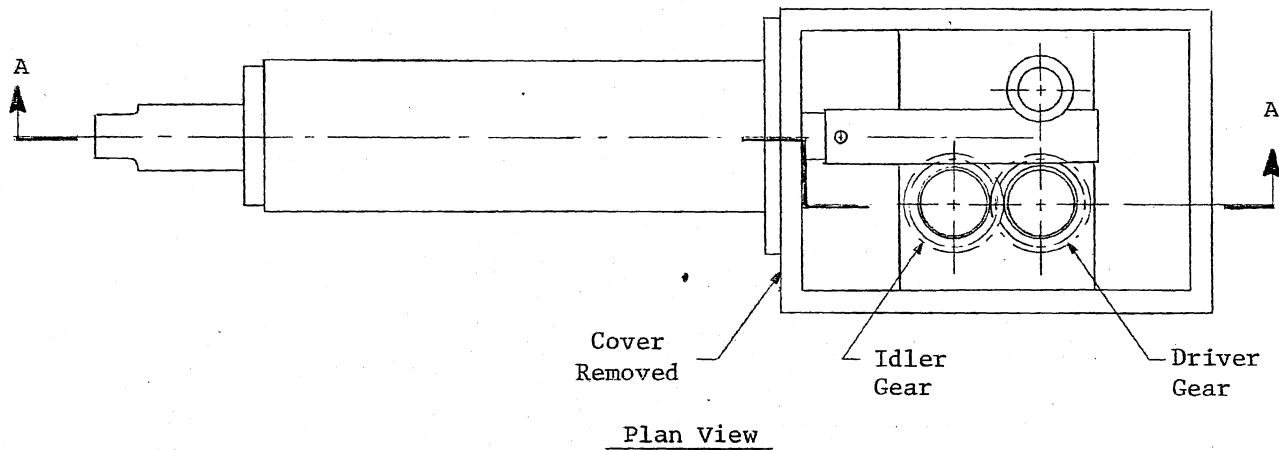


Fig. 6 - Damping Control Mechanism of the Model Woolley Valve as Originally Supplied.
(Adapted from Woolley Drawing No. 35909)

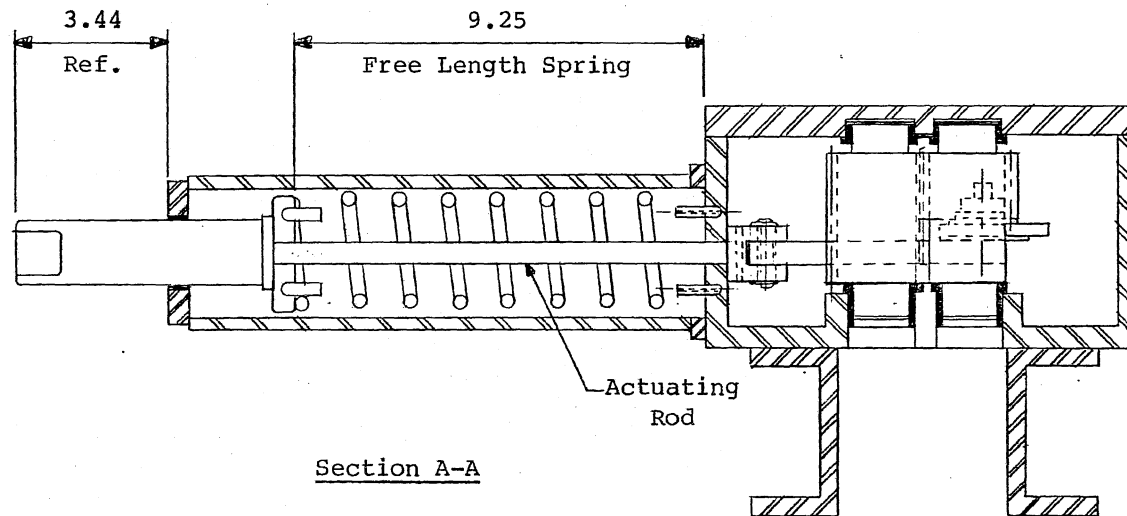
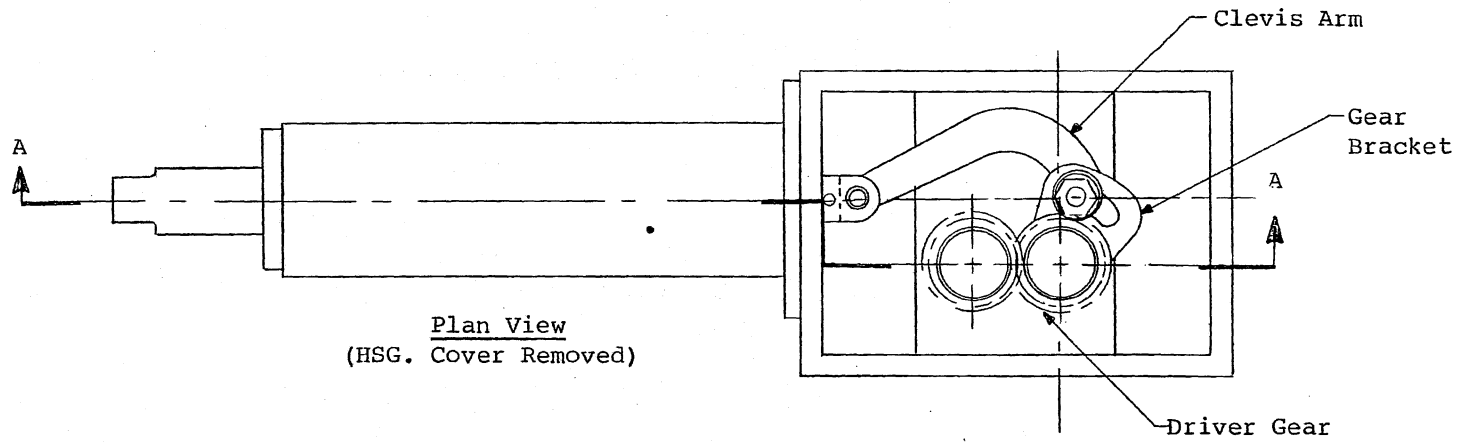


Fig. 7 - Damping Control Mechanism of the Woolley Valve Model as Finally Supplied.
(Adapted from Woolley Drawing No. 35909, Revision of 5-30-79)

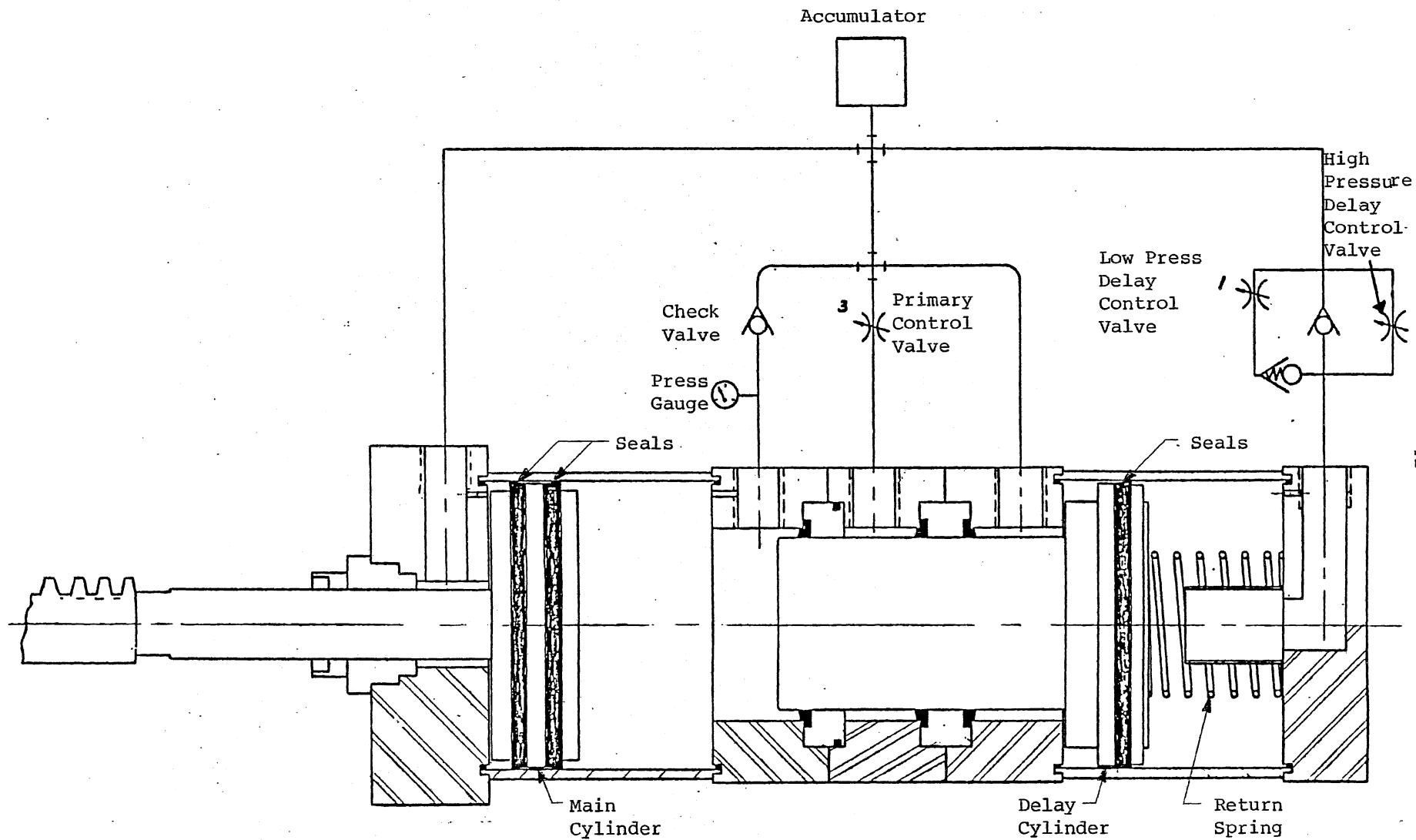


Fig. 8 - Schematic of the Time-Delay Hydraulic Circuit of the Woolley Valve.
 (Adapted from Woolley Drawing No. 36236, 3-9-79)

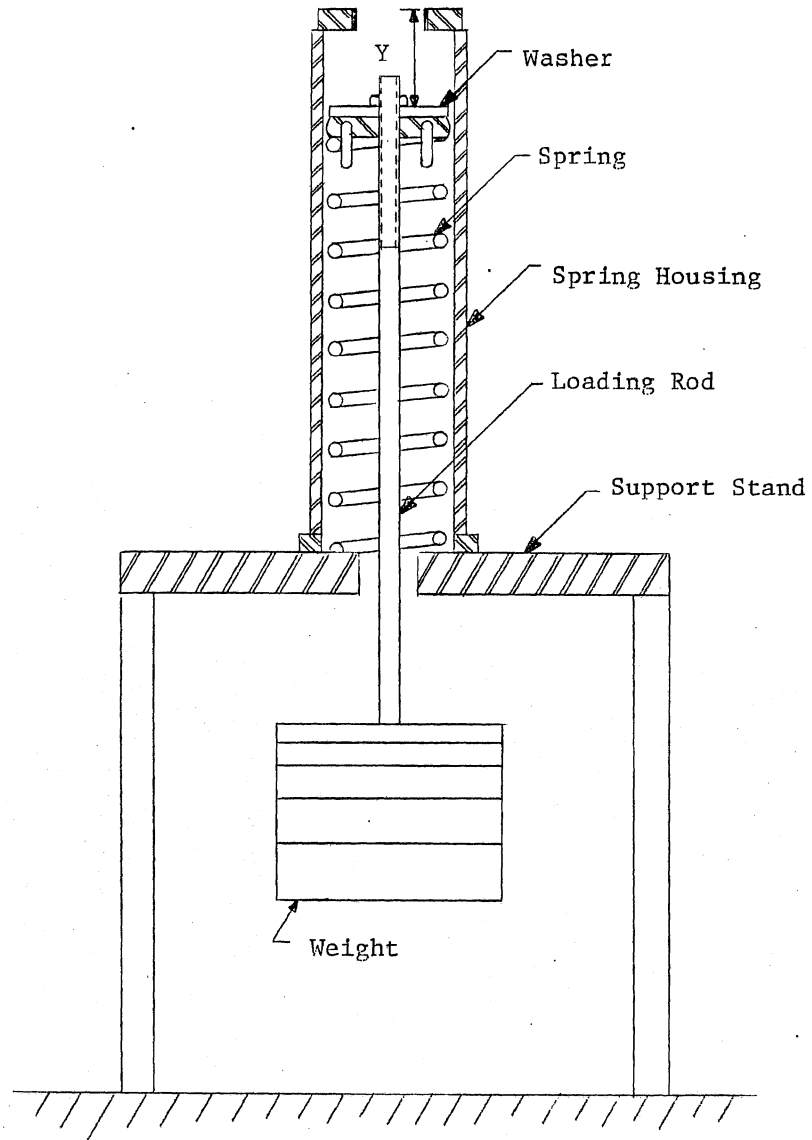


Fig. 9 - Test Stand for Calibration
of Valve Compression Springs.

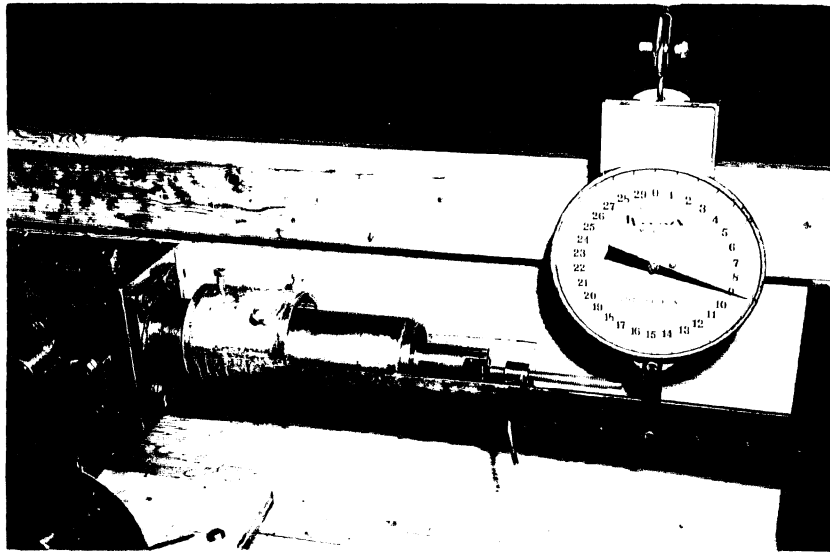
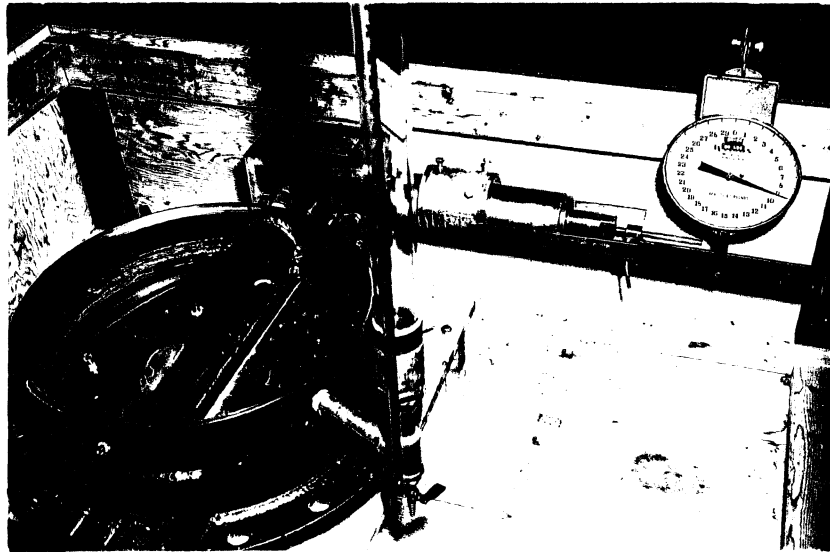
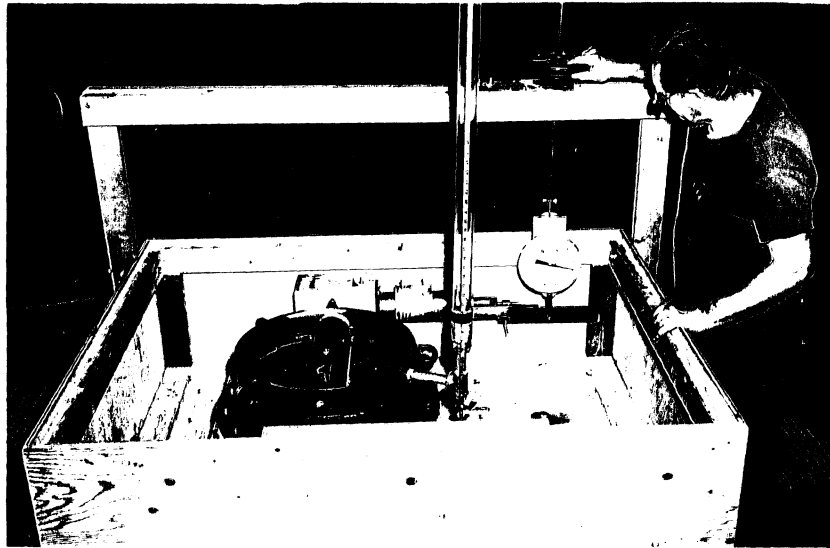


Fig. 10 - The Woolley Valve Model with External Test Loading Spring.

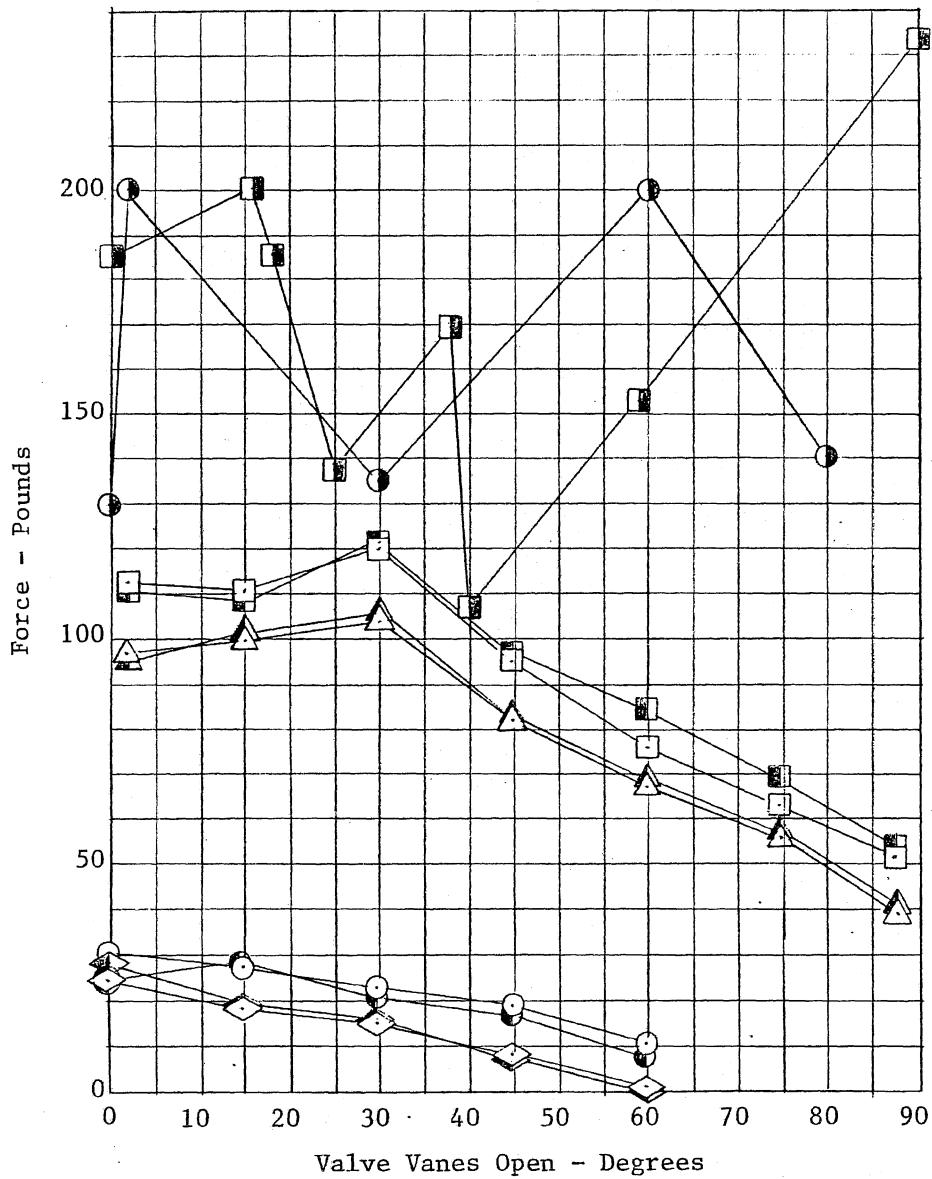


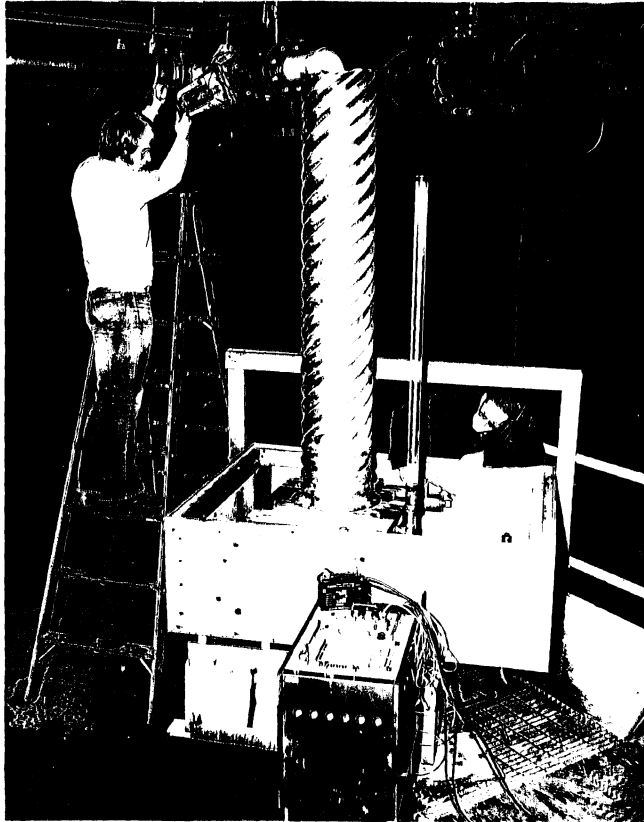
Fig. 11
 Woolley Valve Tests.
 Static Forces to Open and Close the Model Valve

Before Re-alignment

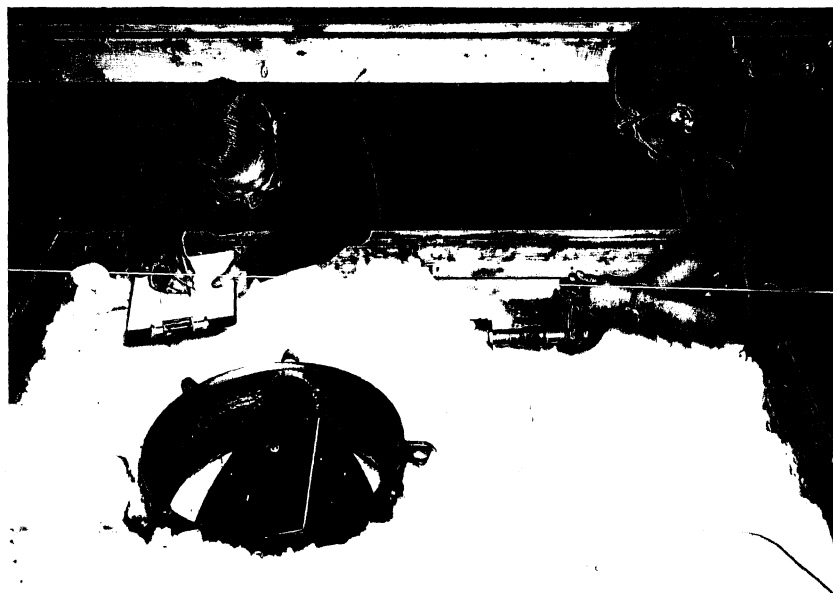
	Air Temp. °F	Vane Motion	Spring Used lbs/in.
○—○	68	Opening	scale
◻—◻	68	Closing	152.9

After Re-alignment

◇—◇	67	Opening	120.7
◀—▶	32	Opening	120.7
○—○	67	Opening	90.9
◐—◐	32	Opening	90.9
△—△	67	Closing	120.7
▲—▲	32	Closing	120.7
◻—◻	67	Closing	90.9
◼—◼	32	Closing	90.9

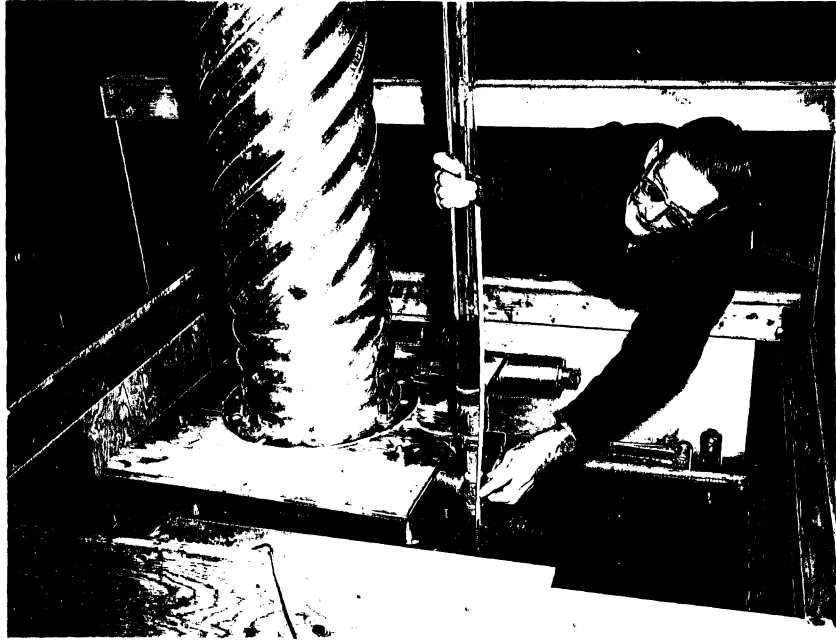


(a)

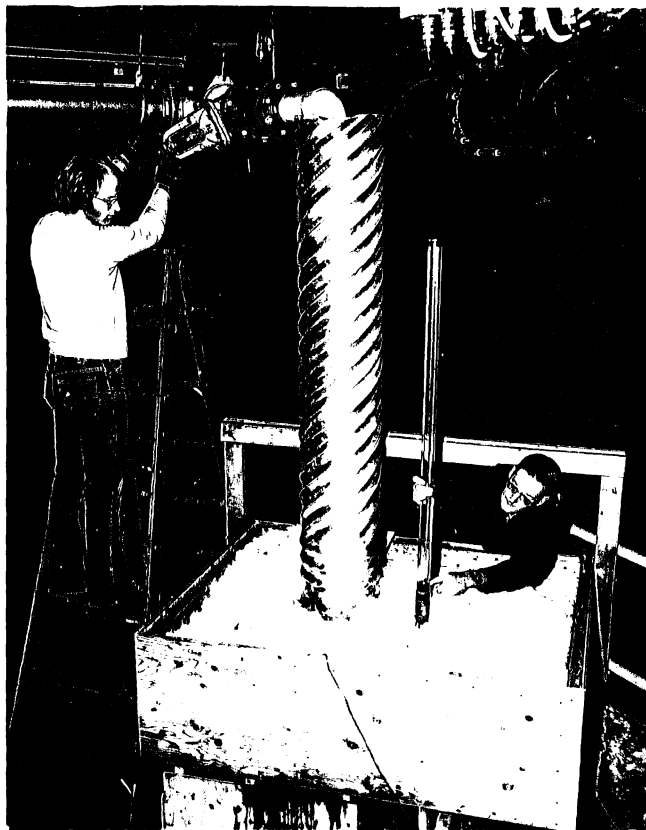


(b)

Fig. 12 - Set-up for Hydrostatic Loading Tests.



(c)



(b)

Fig. 12 (Cont.) - Set-up for Hydrostatic Loading Tests.

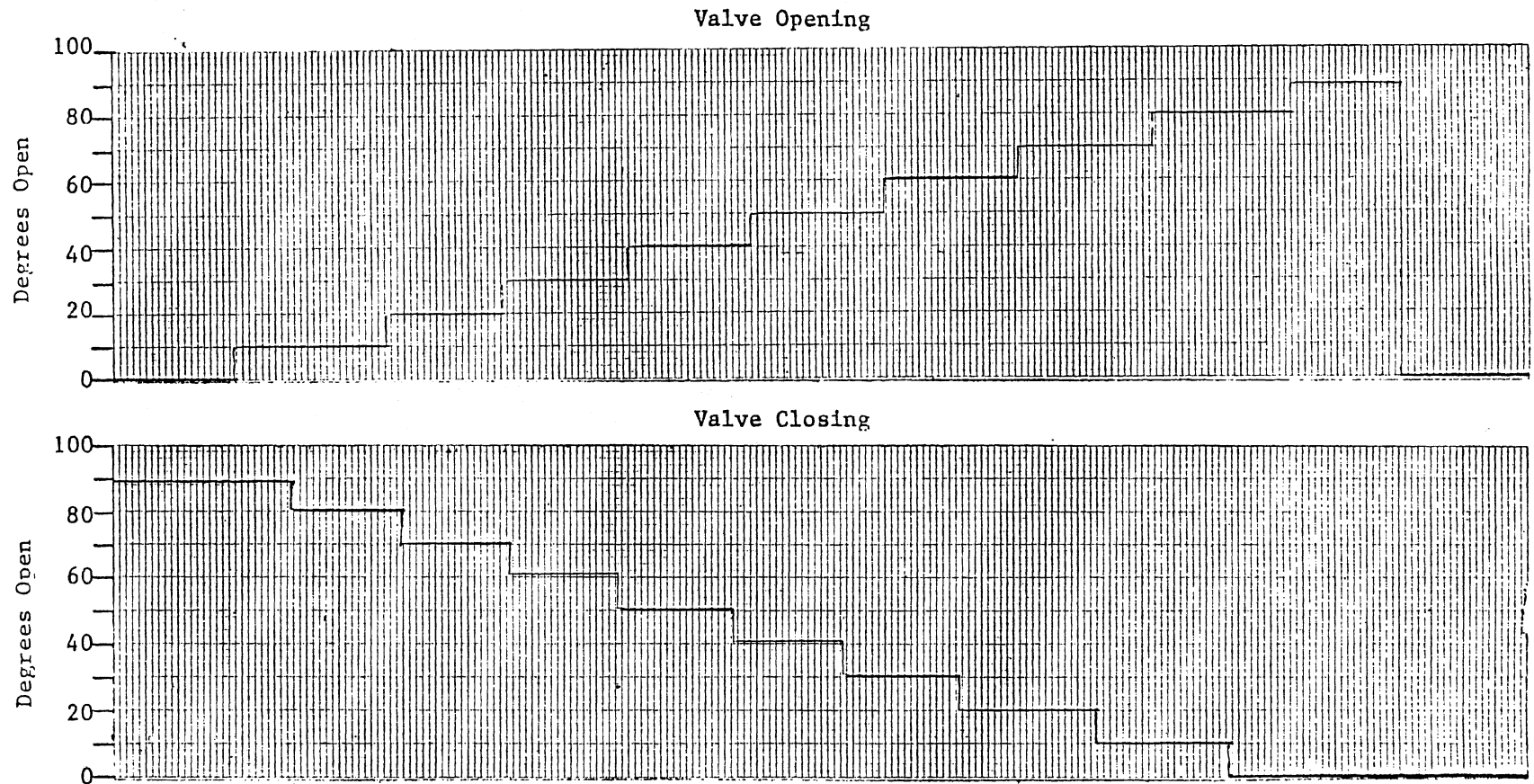


Fig. 13
 Woolley Valve Tests.
 Potentiometer Linearity

Readings on Sanborn recorder with valve vanes set open at 10° increments (average of two vanes). With valve closed, driver gear vane is open 0° and idler gear vane is 0.5° open. With valve wide open, driver gear vane is open 89° and idler gear vane is open 87.5° .

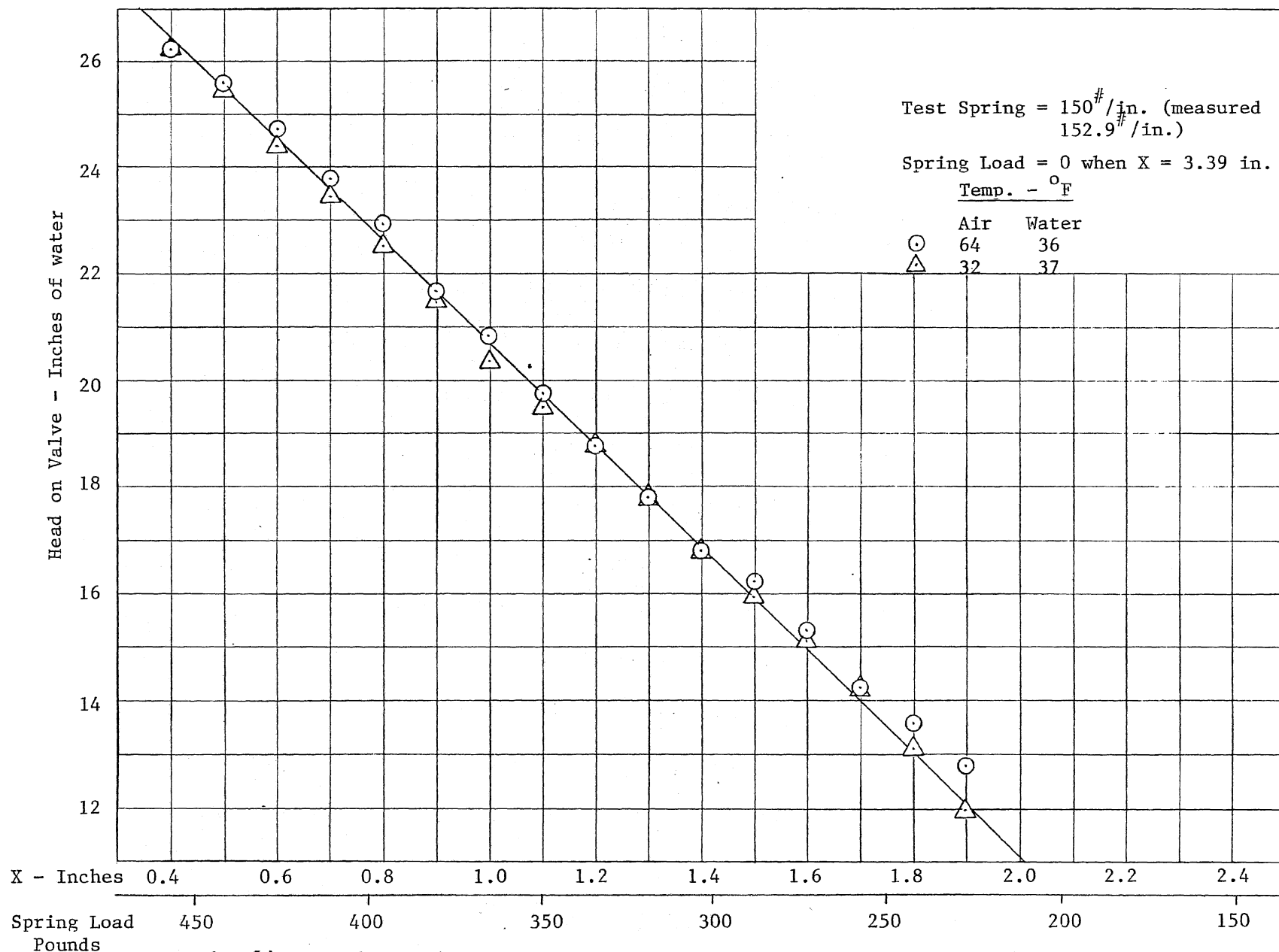


Fig. 14 - Static Opening Tests of the Woolley Valve with a 150 Pound Spring.

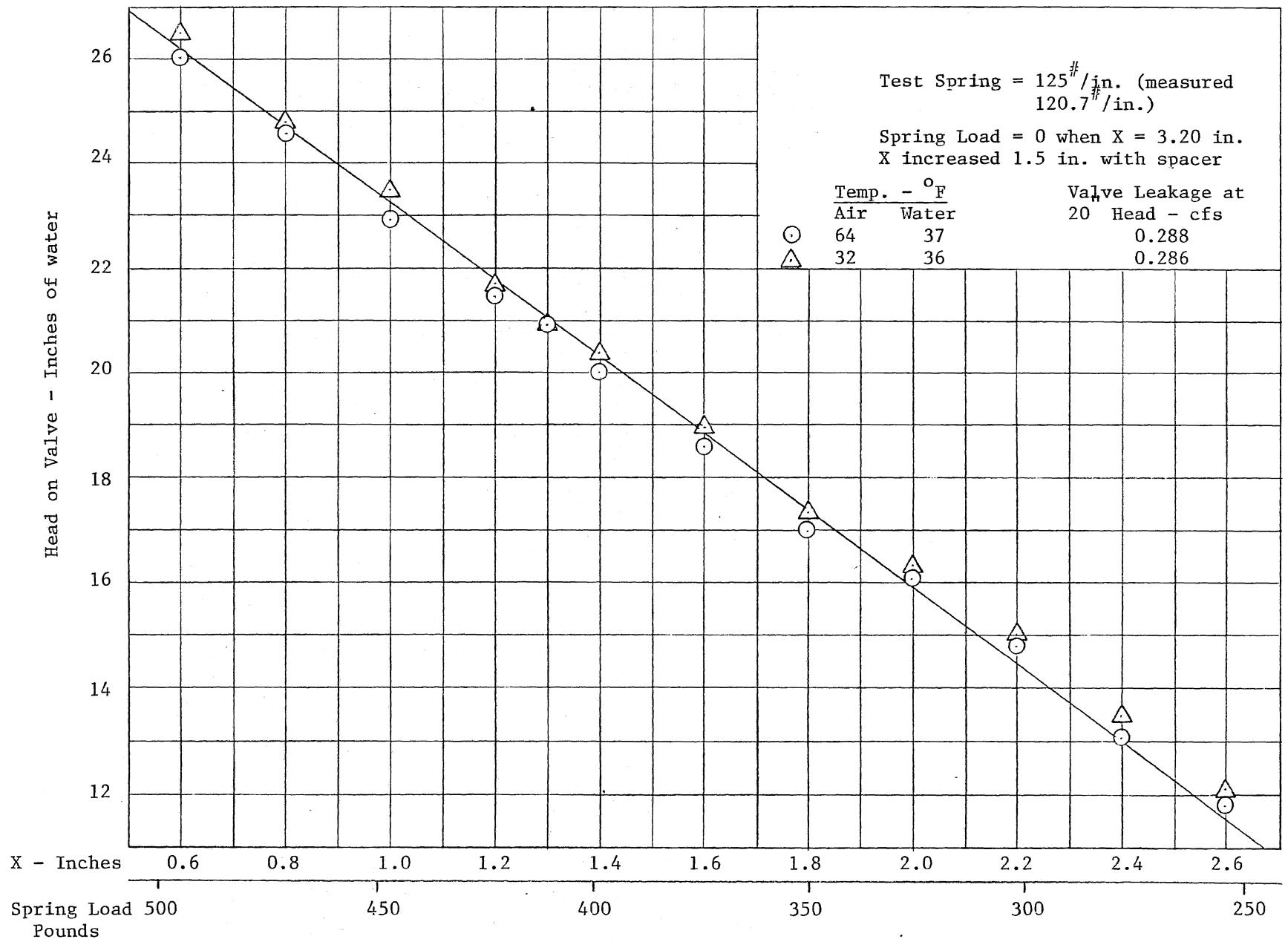


Fig. 15 - Static Opening Tests of the Woolley Valve with a 125 Pound Spring.

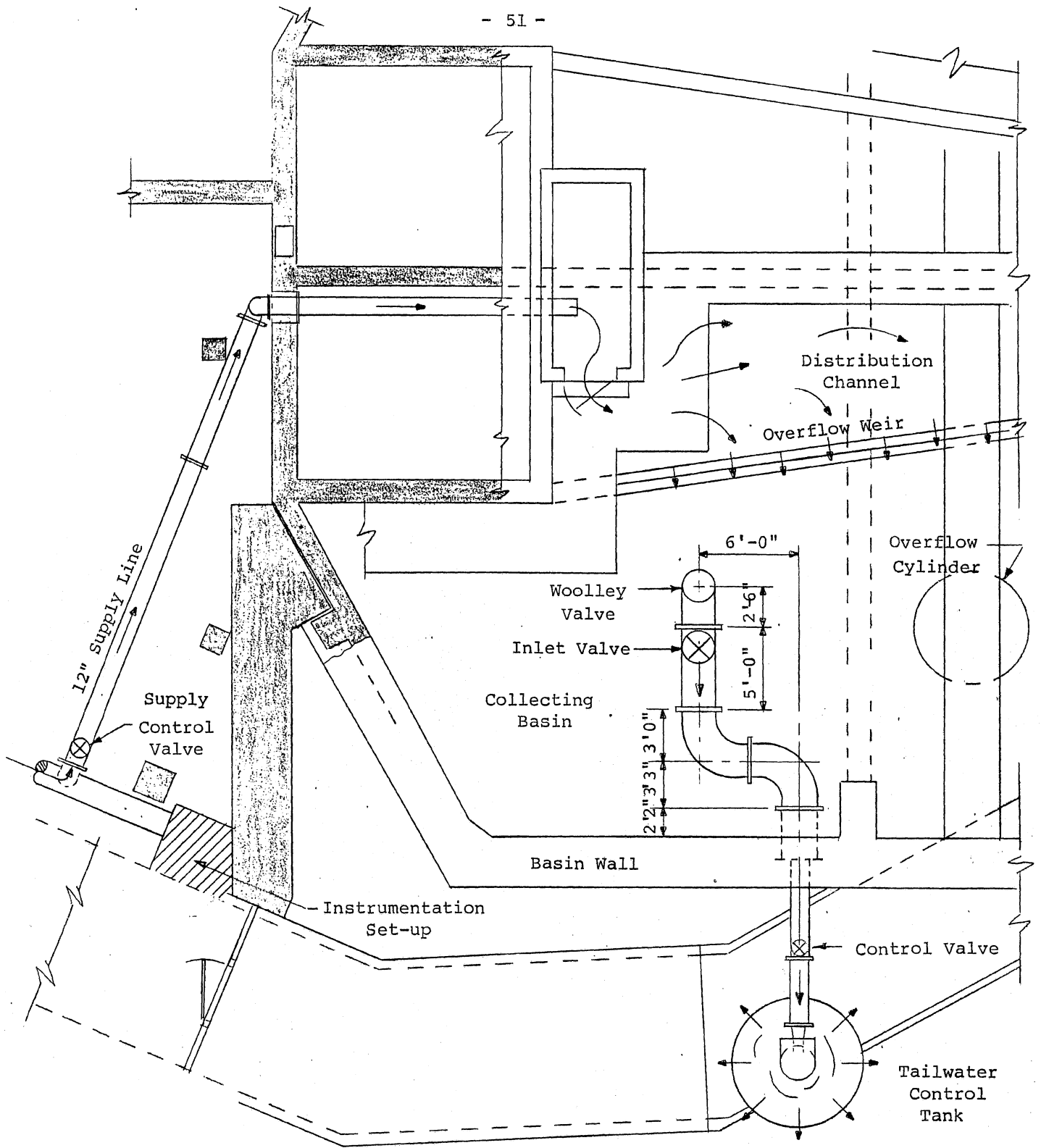


Fig. 16- Plan View of the Test Set-up.

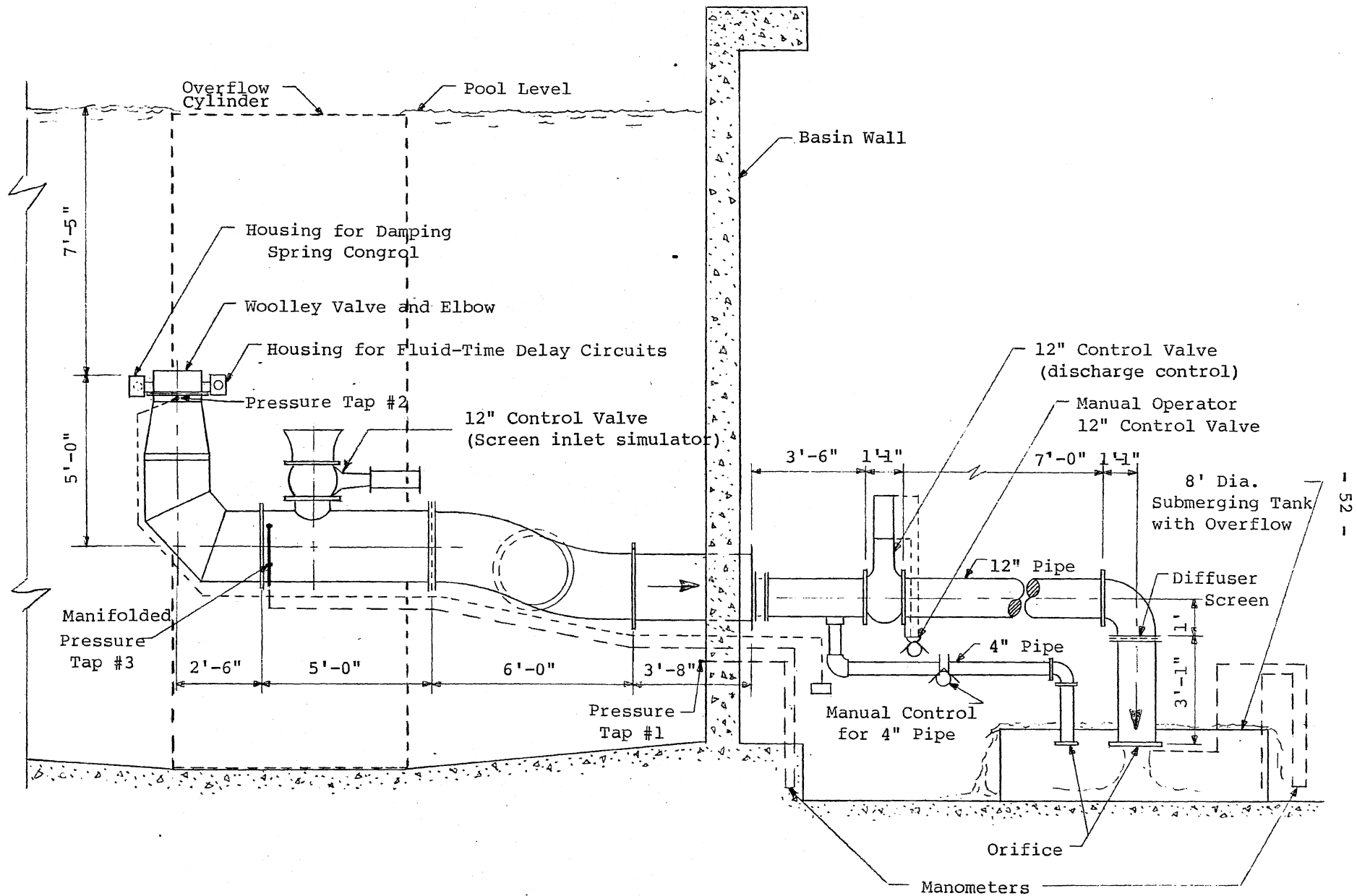
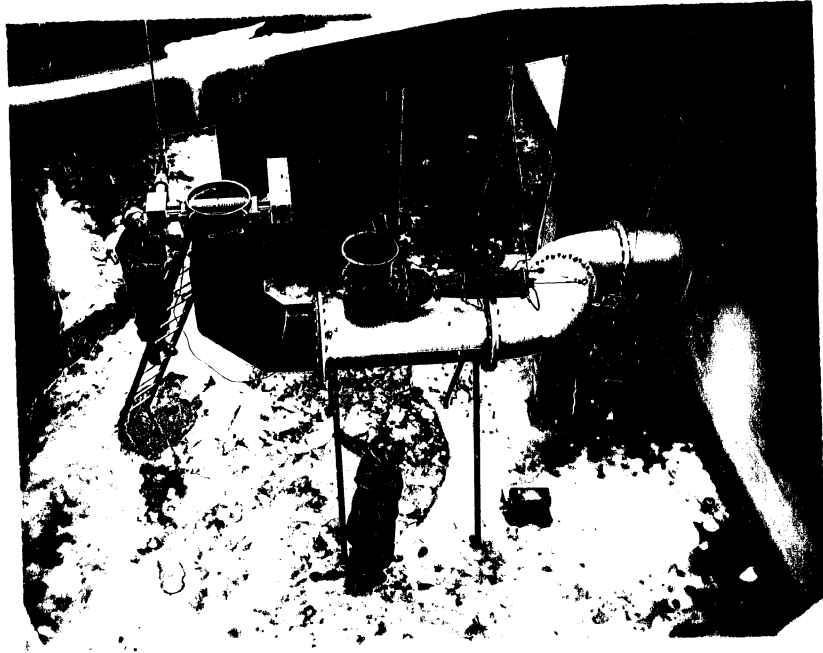
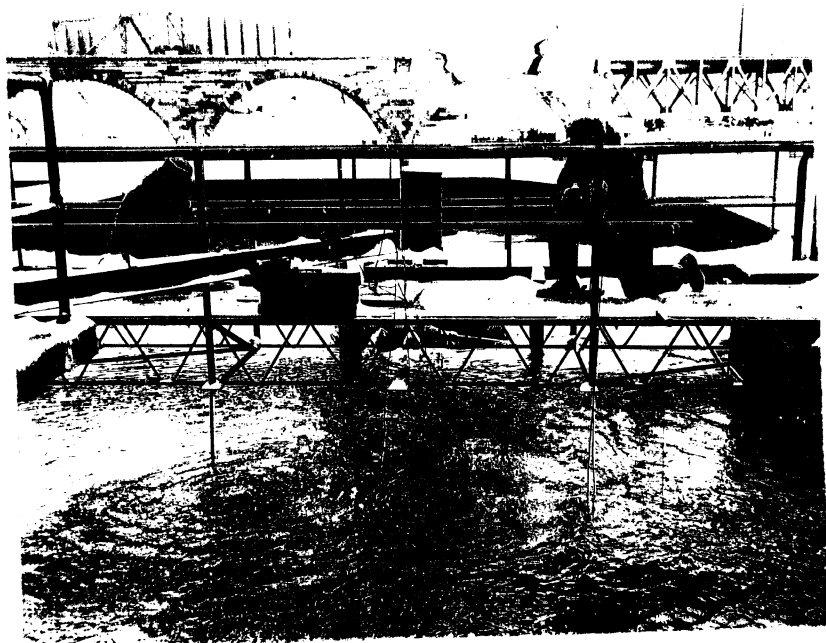


Fig. 17- Elevation of the Test Set-up for Woolley Valve 1:4 Scale.



(a) Installation with Basin Drained

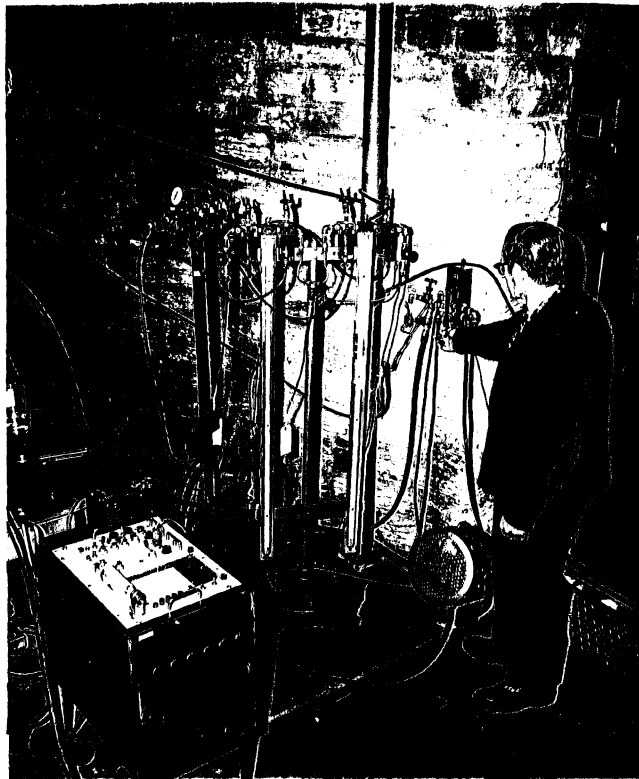


(b) Installation with Basin Filled
and Valve Discharging

Fig. 18 - Installation of the 1/4 Scale Model of the Woolley Valve in the Laboratory Volumetric Basin.



(a) Outside Piping and Control Valves.



(b) Inside Instrumentation and Readout Station.

Fig. 19 - Discharge Controls for the Woolley Valve Tests.

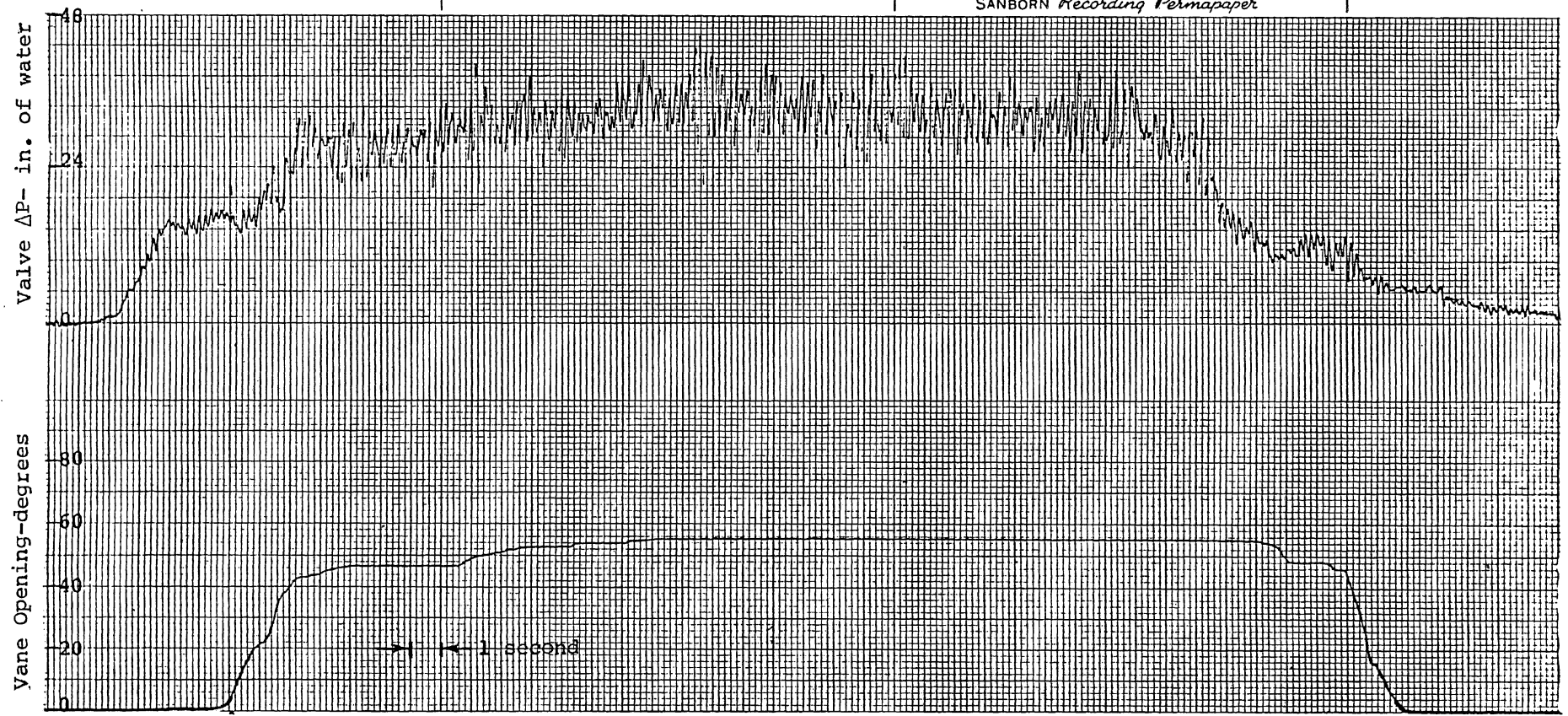


Fig. 20 - Opening and Closing Response of the Woolley Valve.
(Spring rate = 120.7 lbs/in; spring rod setting X = 1.3 in;
undamped pressures at Tap #2)

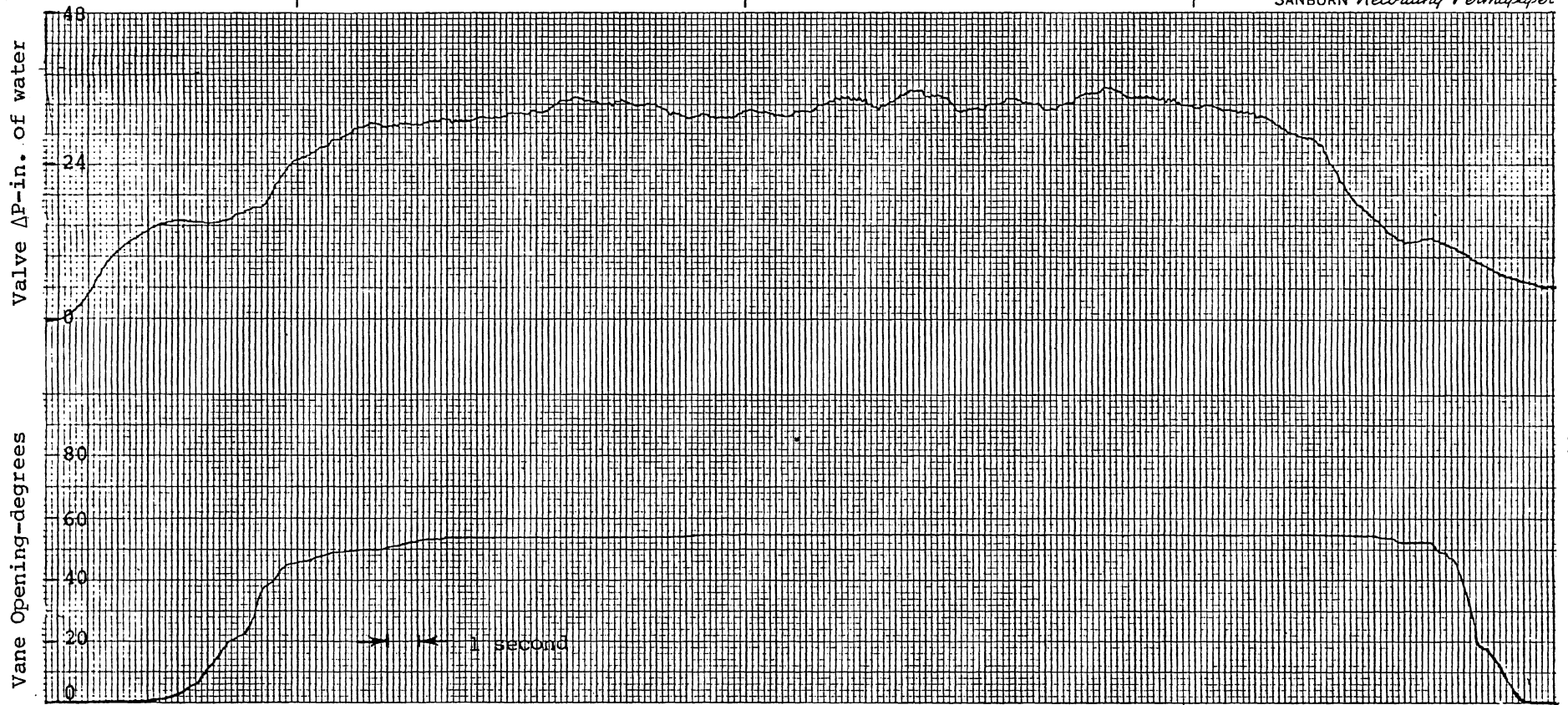


Fig. 21 - Opening and Closing Response of the Woolley Valve.

(Spring rate = 120.7 lbs/in; spring rod setting X = 1.3 in;
damped pressures at Tap #2)

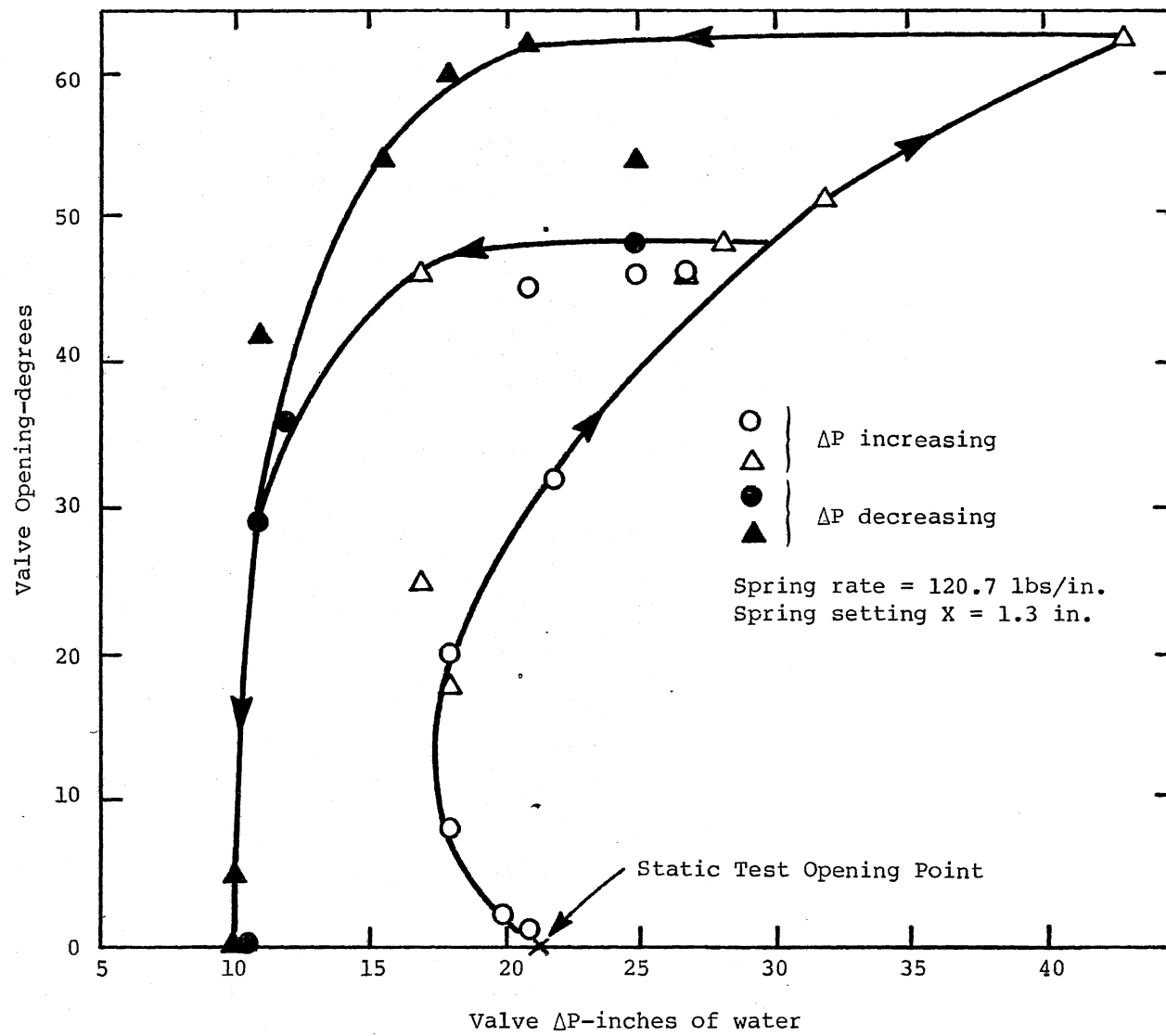


Fig. 22- Opening and Closing Response of the Woolley Valve to Imposed Pressure Differentials.

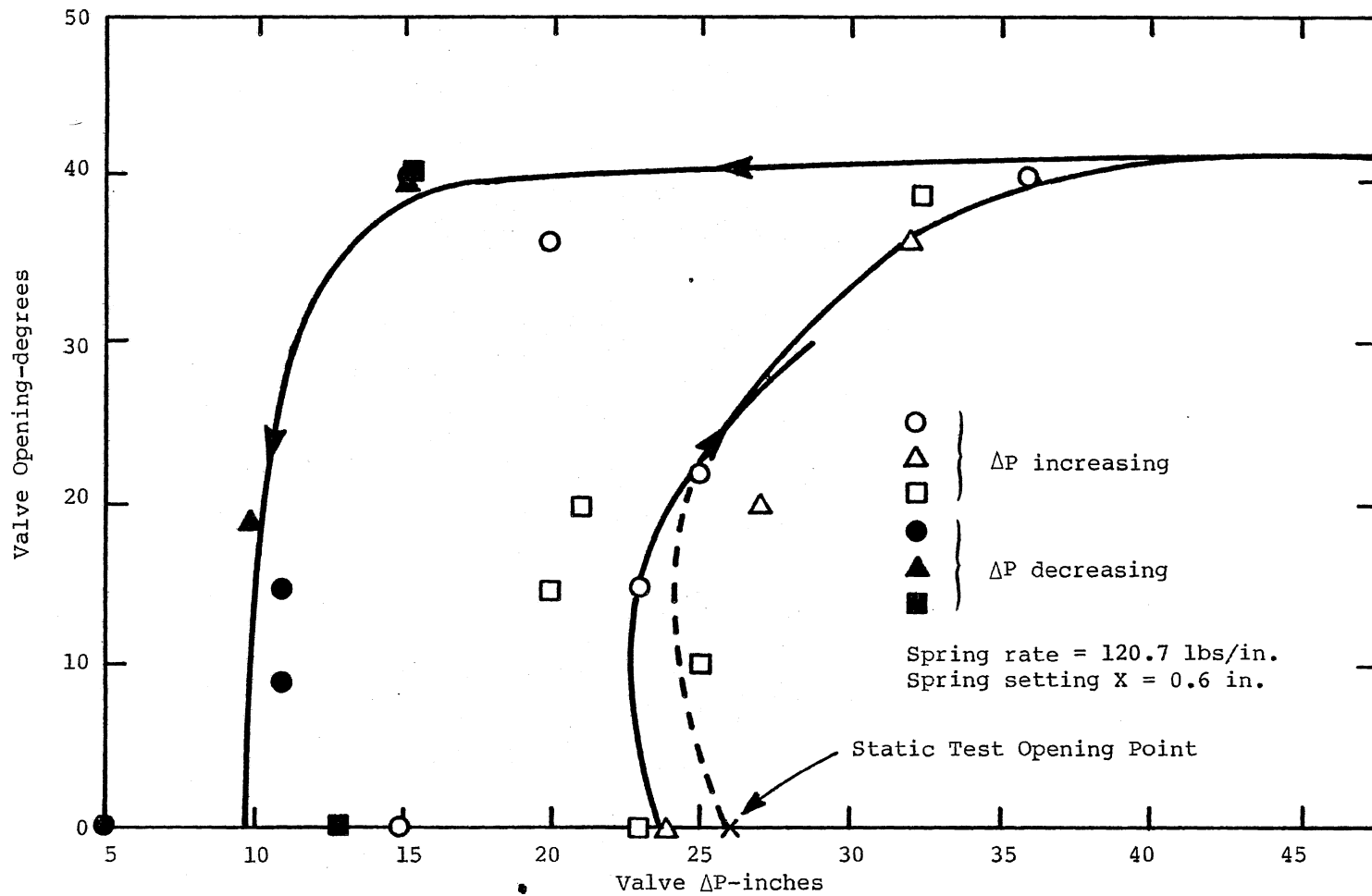


Fig. 23 - Opening and Closing Response of the Woolley Valve to Imposed Pressure Differentials.

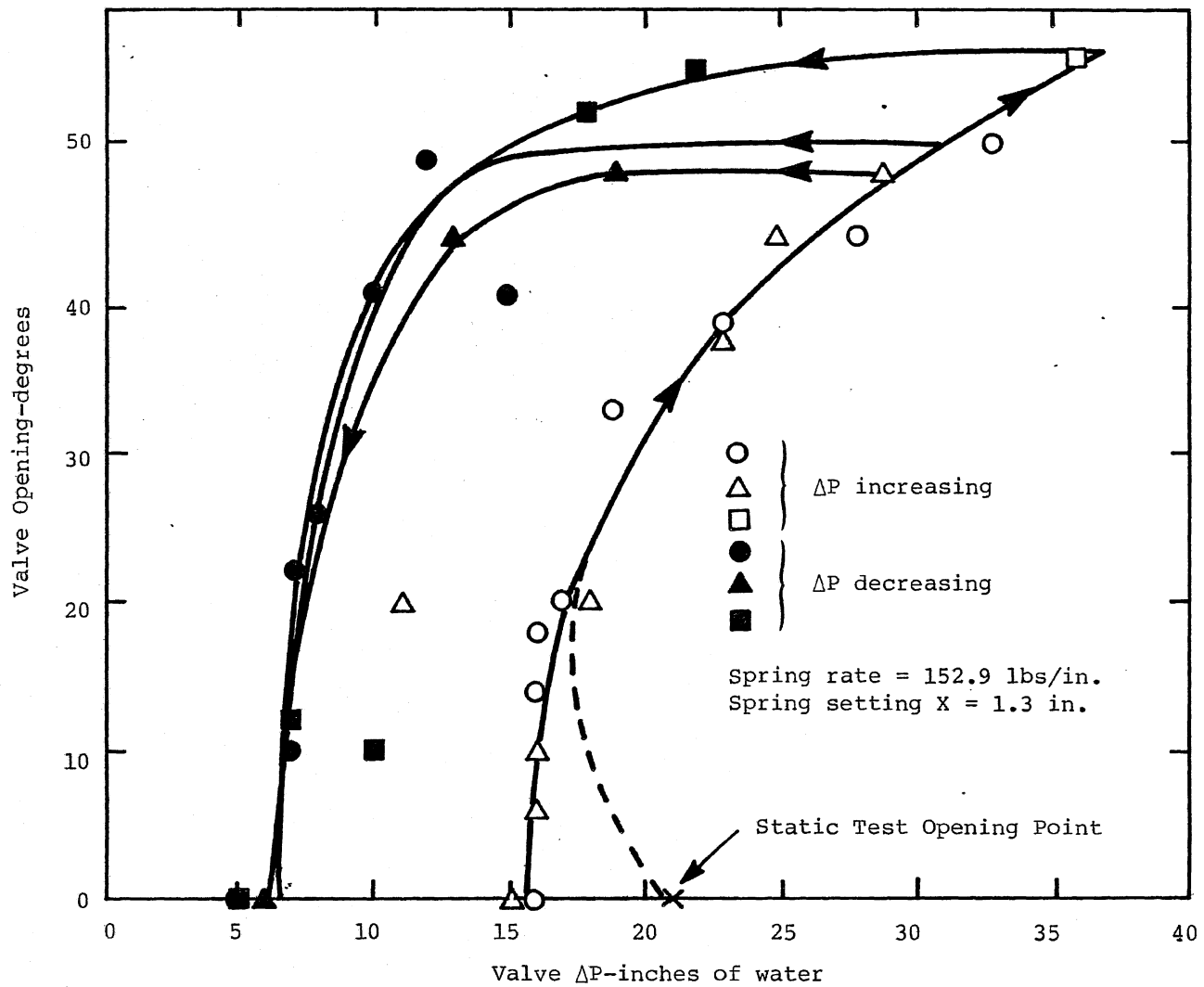


Fig. 24 - Opening and Closing Response of the Woolley Valve to Imposed Differential Pressures.

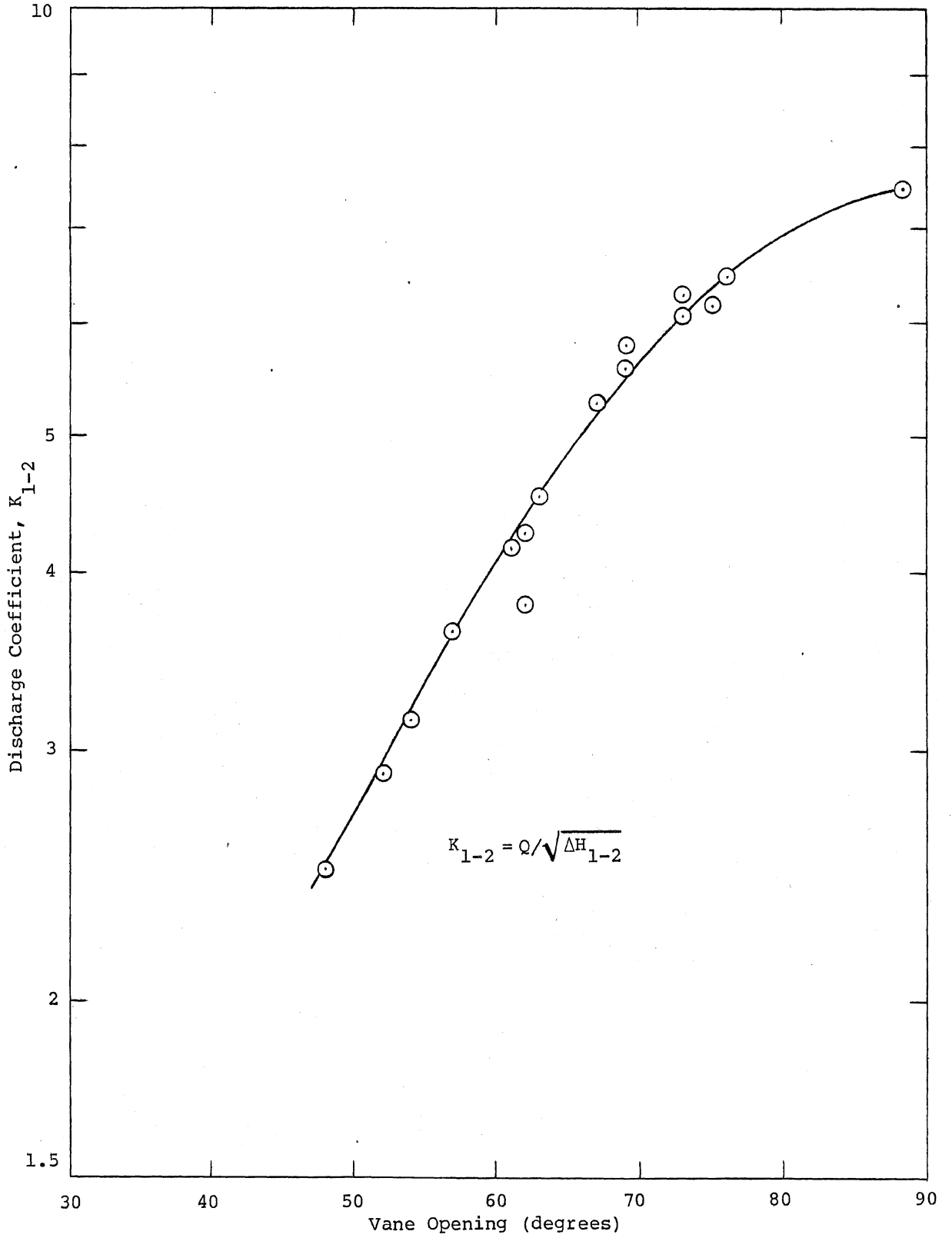


Fig. 25 - Discharge Coefficient Versus Vane Opening as Measured by ΔH_{1-2} . (Data from Table II - Tests of 3/13/79)

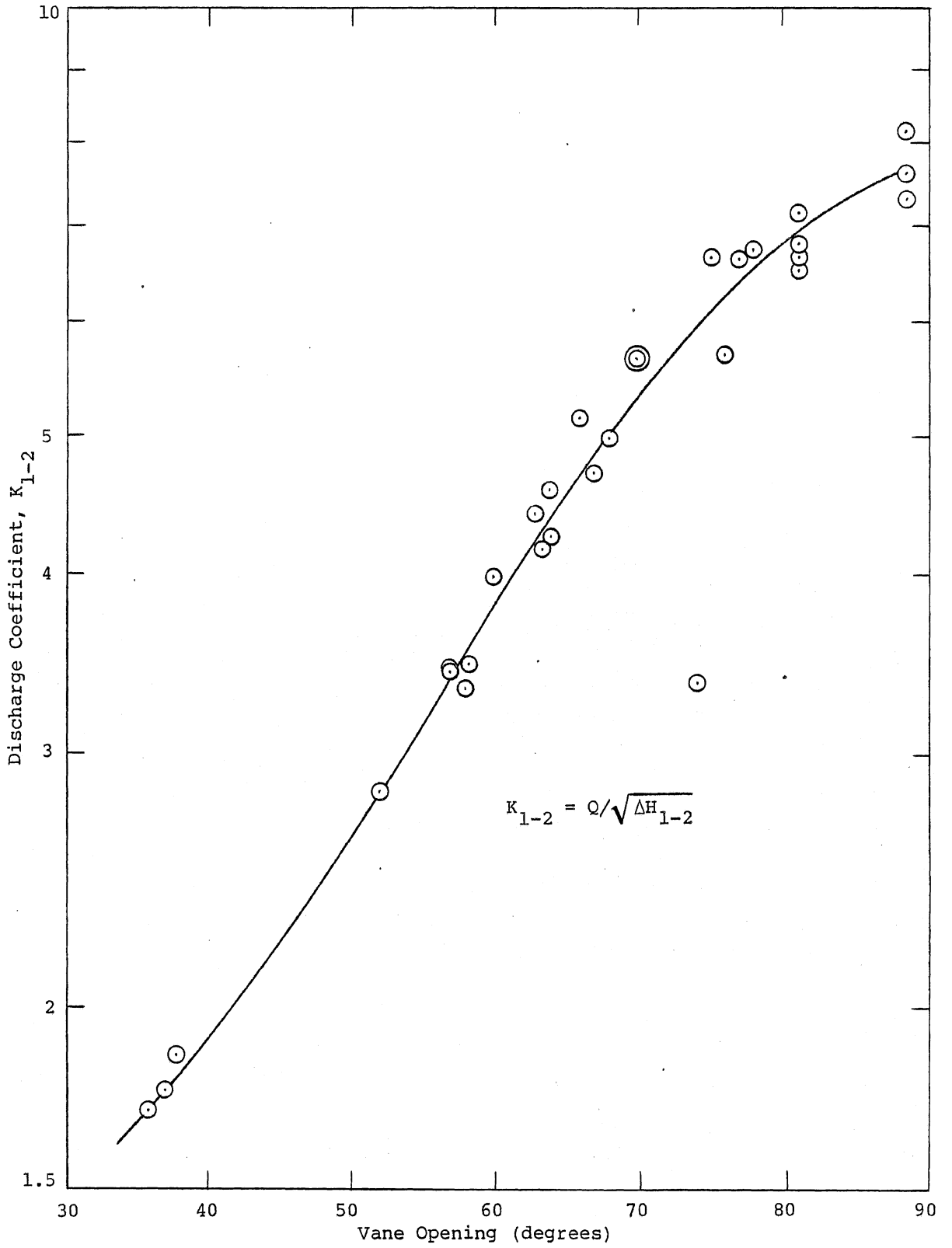


Fig. 26 - Discharge Coefficient Versus Vane Opening as Measured by ΔH_{1-2} . (Data from Table IV - Tests of 3/28/79)

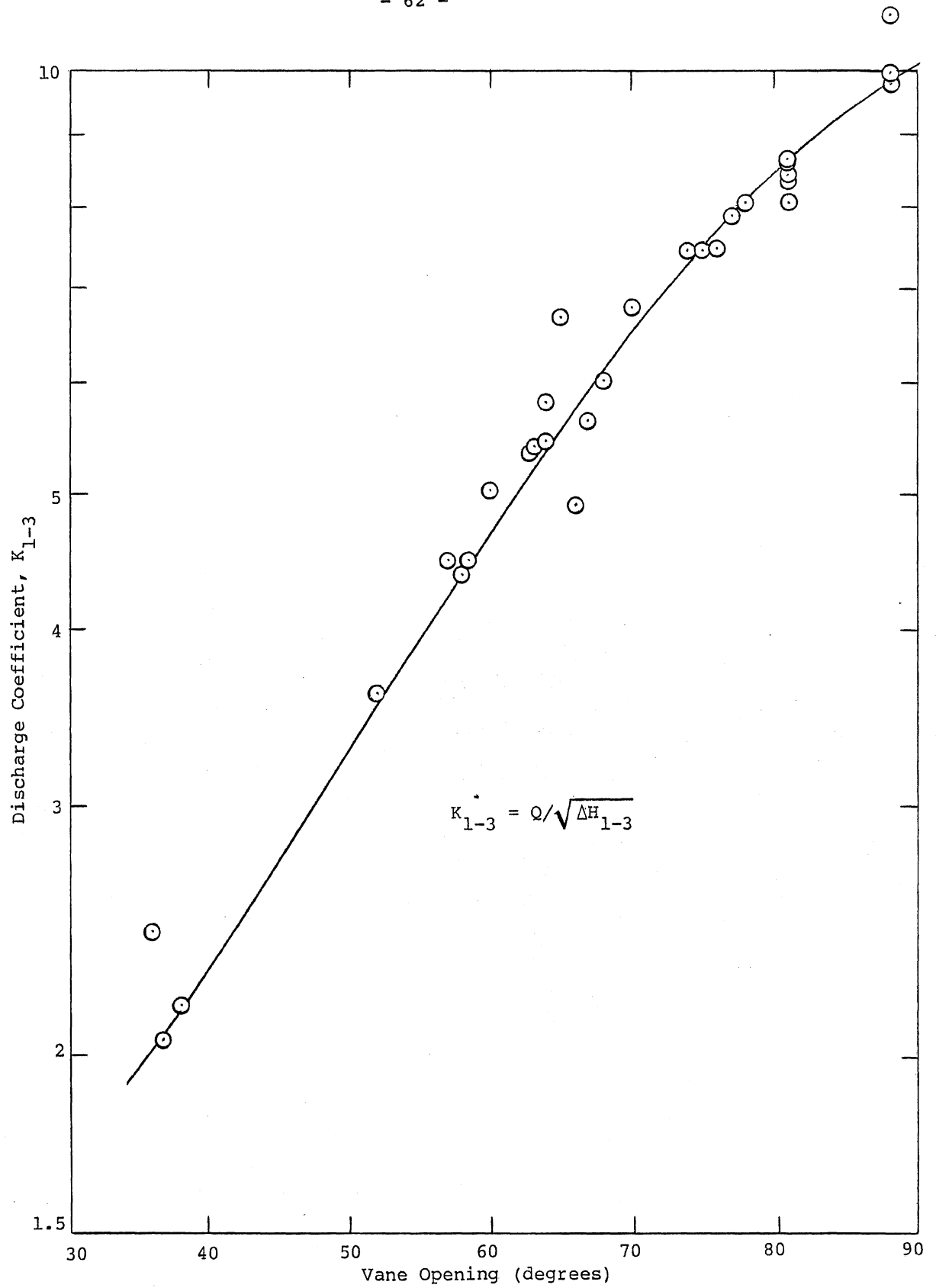


Fig. 27 - Discharge Coefficient Versus Vane Opening as Measured by ΔH_{1-3} . (Data from Table IV - Tests of 3/28/79)

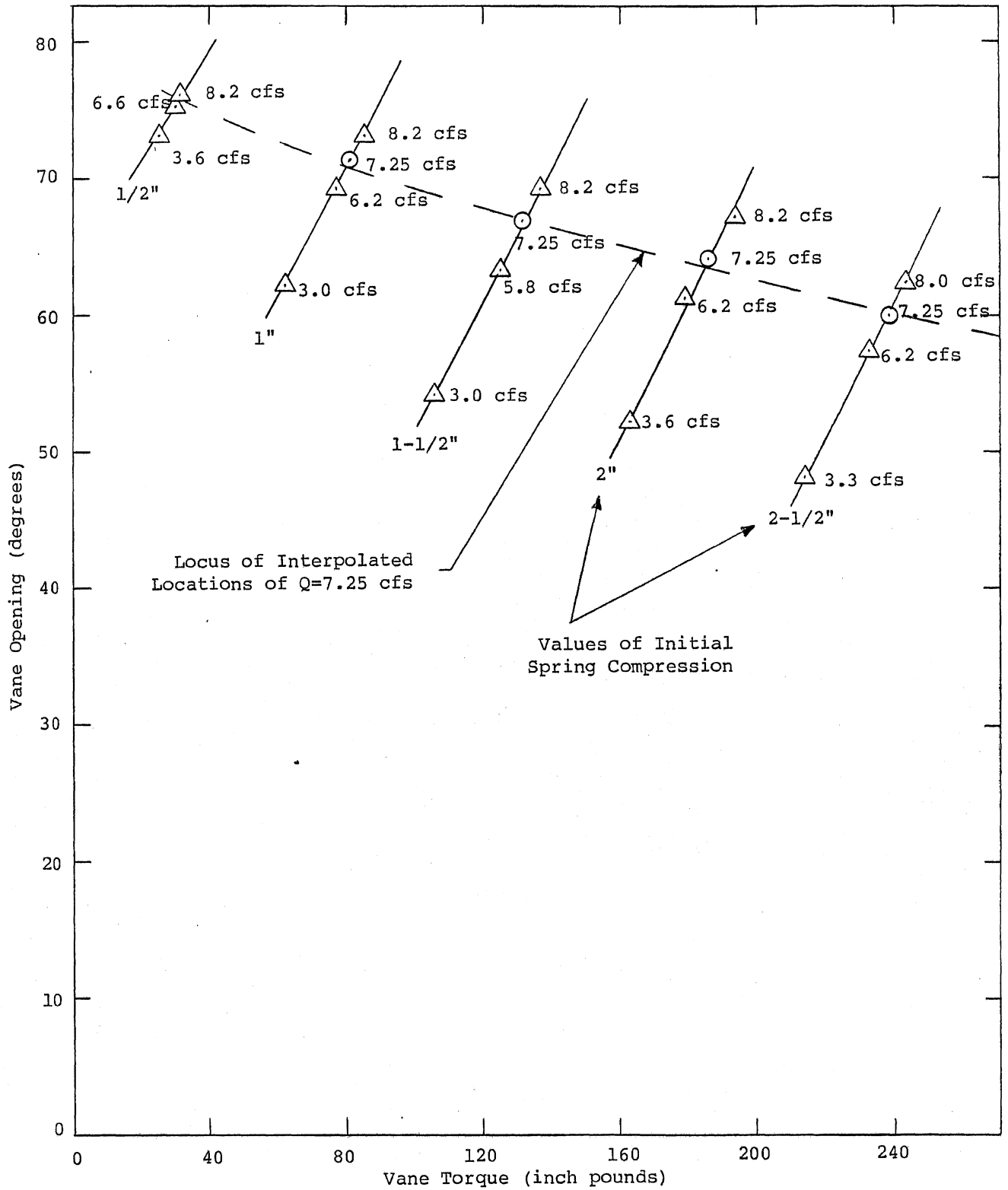


Fig. 28 - Values of Discharge Versus Vane Opening and Vane Torque.
(Data from Table I - Tests of 3/13/79)

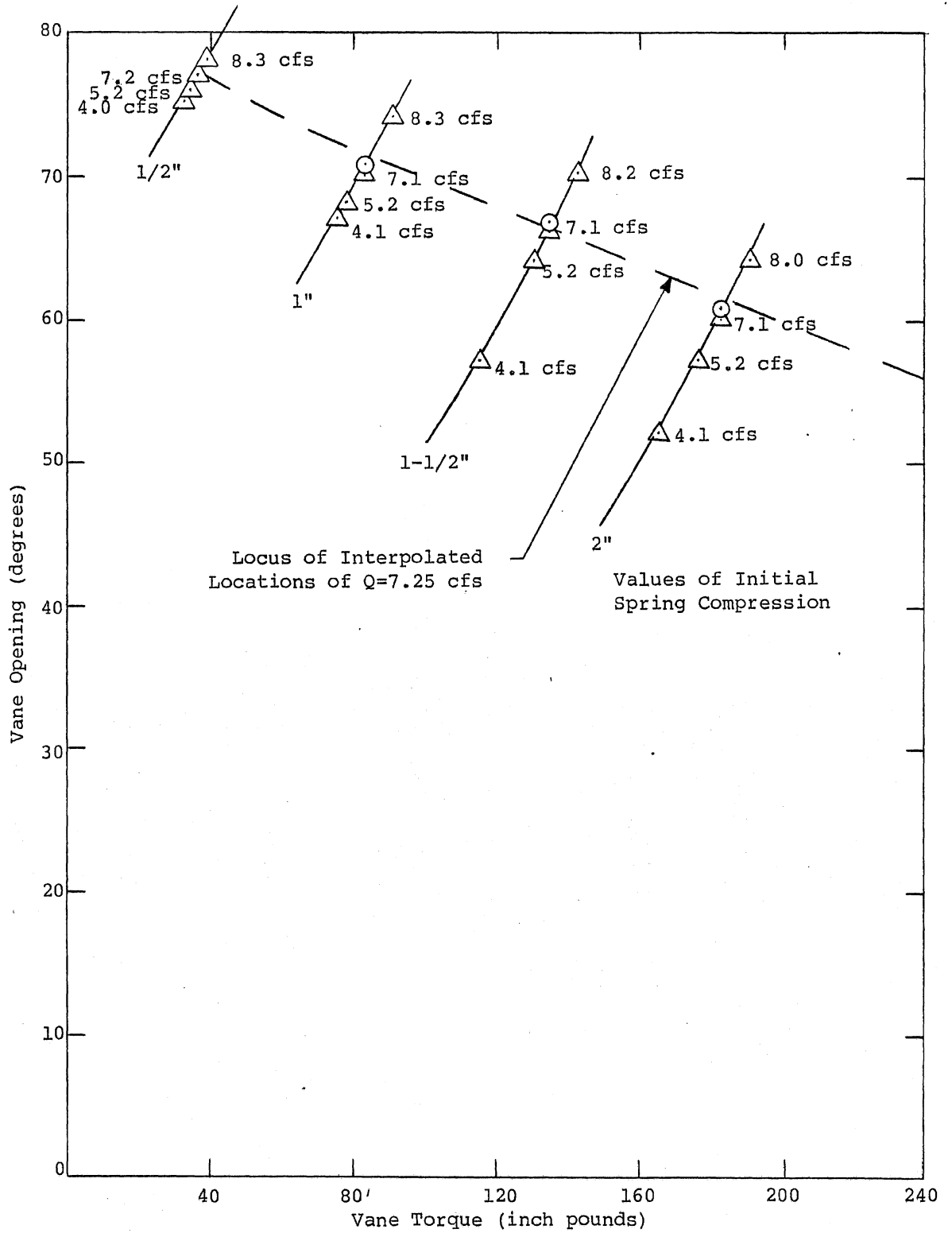


Fig. 29 - Values of Discharge Versus Vane Opening and Vane Torque.
(Data from Table III - Tests of 3/28/79)

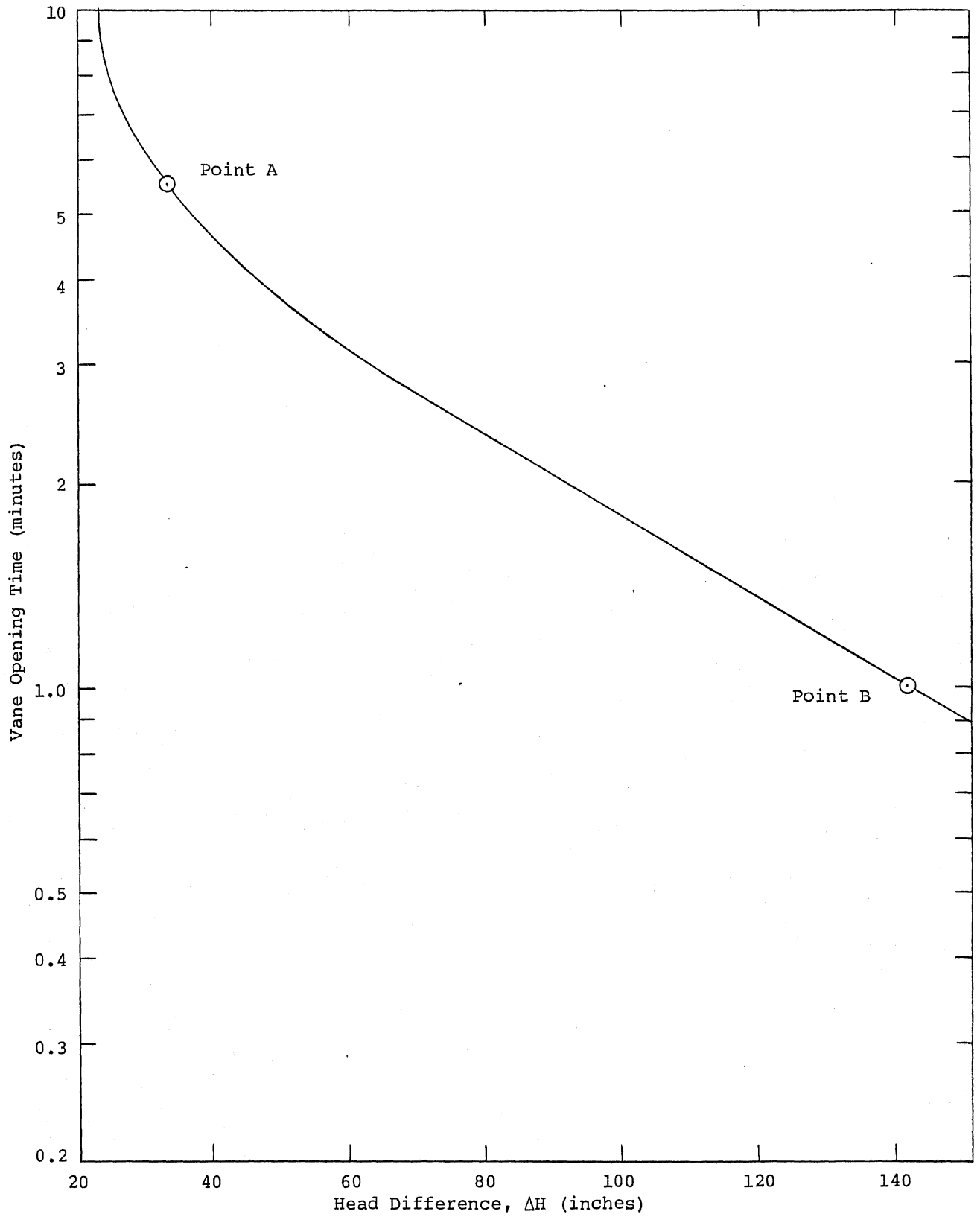


Fig. 30 - Hypothetical Opening Response Under Various Head Differentials.

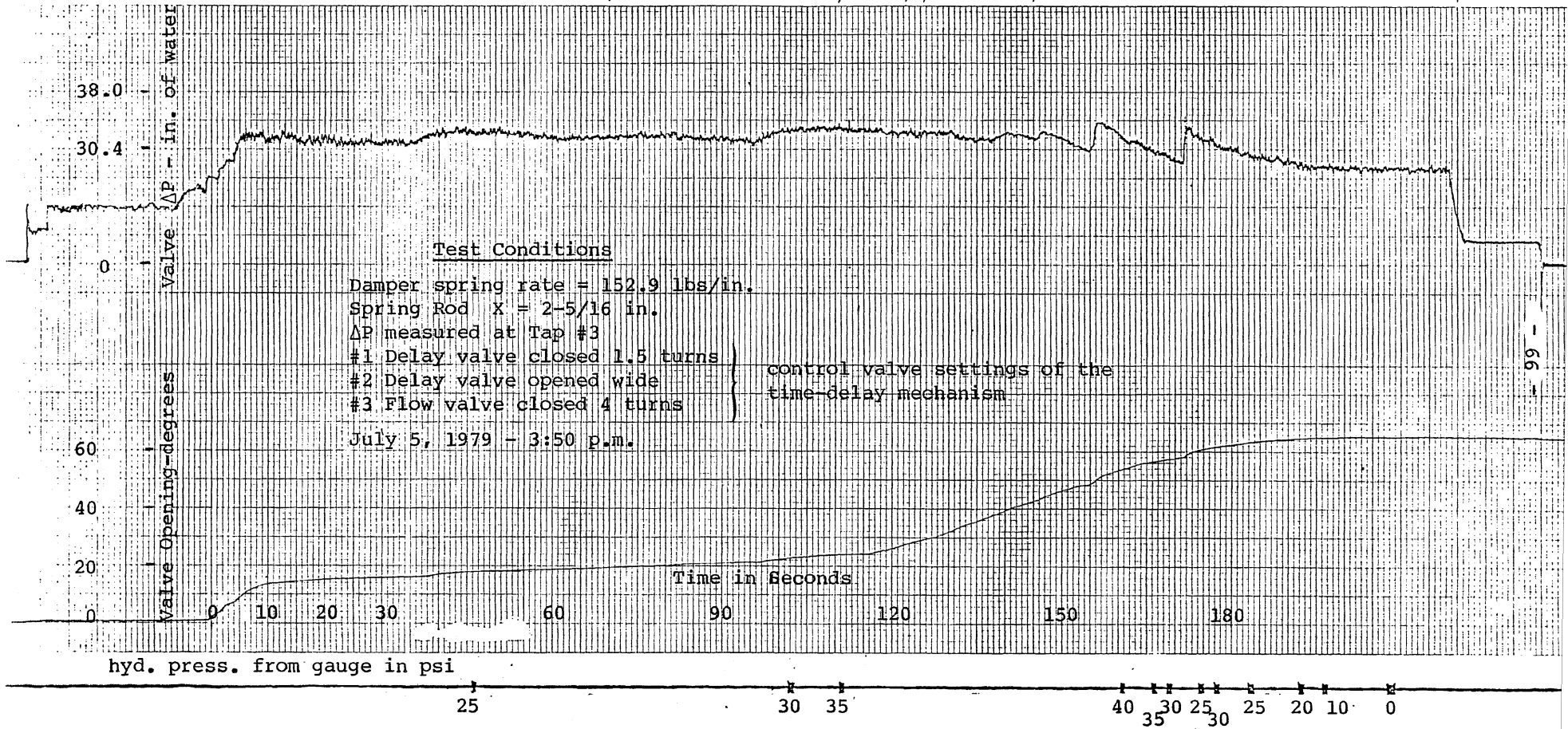


Fig. 31 - Opening and Closing Response of Woolley Valve with Time-Delay Mechanism.

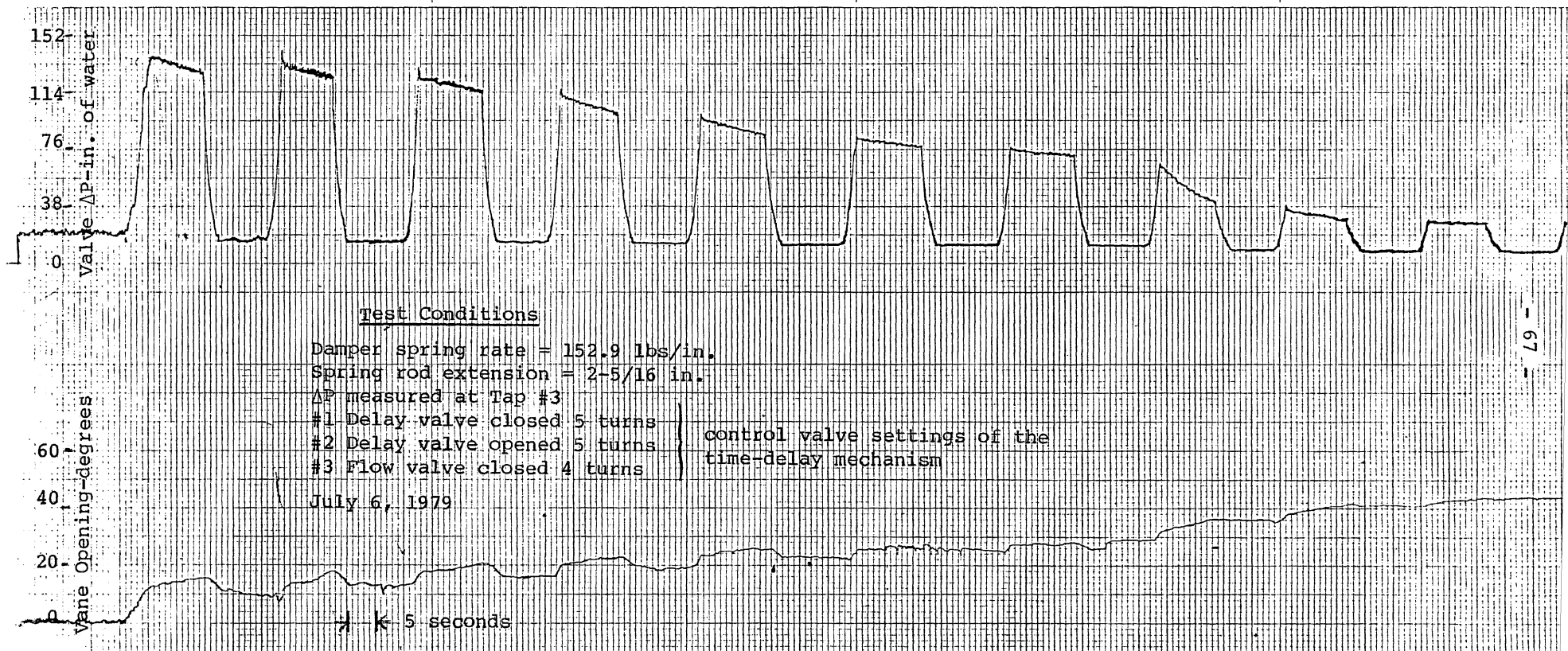


Fig. 32- Opening Response to Cycling Pressure for Woolley Valve with Time-Delay Mechanism.

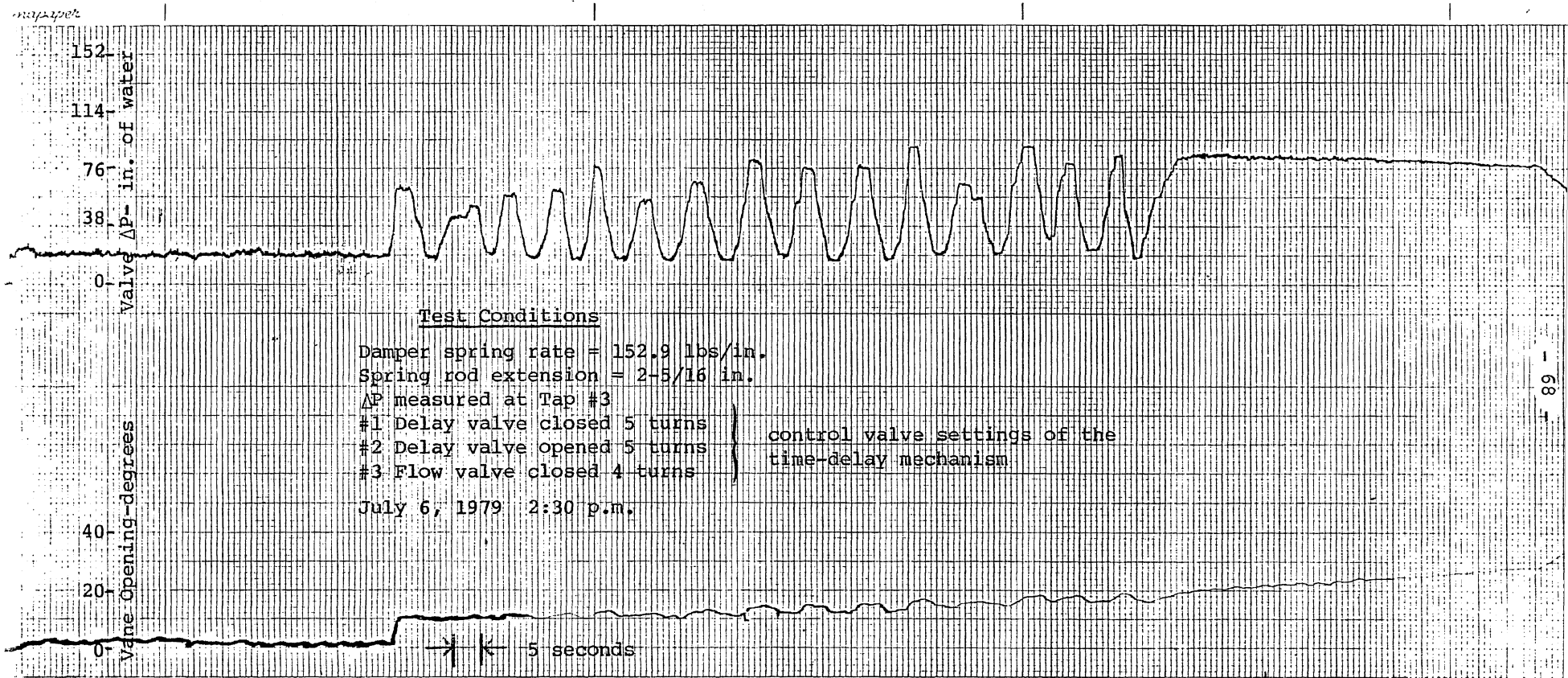


Fig. 33 - Opening Response to Cycling Pressure for Woolley Valve with Time-Delay Mechanism.