Prediction of Check Valve Performance and Degradation in Nuclear Power Plant Systems

Final Report
September 1987 - April 1988

Prepared by M.S. Kalsi, C.L. Horst, J.K. Wang

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ABSTRACT

Degradation and failure of swing check valves and resulting damage to plant equipment has led to a need to develop a method to predict performance and degradation of these valves in nuclear power plant systems. This Phase I investigation developed methods which can be used to predict the stability of the check valve disk when piping disturbances such as elbows, reducers, and generalized turbulence sources are present upstream of the valve within 10 pipe diameters. Major findings include the flow velocity required to achieve a full open, stable disk position, the magnitude of disk motion developed with these upstream disturbances with flow velocities below full open conditions, as well as disk natural frequency data which can be used to predict wear and fatigue damage. Reducers were found to cause little or no performance degradation. Elbow effects must be considered when located within 5 diameters of the check valve, while severe turbulence sources have significant effect at distances to 10 diameters.

Clearway swing check designs were found to be particularly sensitive to manufacturing tolerances and installation variables making them likely candidates for premature failure. Reducing the disk full opening angle on these clearway designs results in significant performance improvement.

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EXECUTIVE SUMMARY

The purpose of this Phase I investigation was to determine the feasibility of a technical approach in predicting and quantifying the stability or instability of swing check valve disks in piping systems which have significant flow disturbances within 10 pipe diameters upstream of the valve. Specifically, the effects of elbows, reducers, and turbulence sources have been investigated. A series of over 2,000 flow tests were performed to measure the performance of 3-inch and 6-inch swing check valves subjected to these upstream disturbances.

The following is a summary of major findings from the test program:

1. Disk stability for swing check valves can be predicted if the valve disk angle, the type and proximity of the upstream disturbance, and the flow velocity through the valve are known.

2. To account for the presence of an upstream flow disturbance, some velocity margin over the minimum velocity needed to suppress disk fluctuations (and fully open it against the backstop) in an undisturbed, uniform pipe flow is required. The necessary margins for various flow disturbances were developed and are included in this report.

3. Clearway swing check valves are most susceptible to the effects of upstream disturbances such as elbows and turbulence producing devices. (A clearway swing check is one in which the disk, at its full-open position is lifted up into the bound area and nearly out of the path of the flow stream through the valve body). Such valves may never achieve disk stability under certain flow conditions.

4. One of the important findings from this research was that upstream reducers have no measurable effect on either velocity required over baseline to fully open the valve or on disk stability. This holds even when the reducer is located immediately adjacent to the check valve. (A reducer is defined as an element which reduces the pipe size as one approaches the valve in the direction of flow.)

5. Some specific results from Phase I tests are that at proximities of 3 to 5 pipe diameters upstream, elbows require an increase in flow velocity over baseline of 10 or 15 percent to fully open the valve disk. Proximities of 0 to 1 diameter will require up to 50 percent higher flow velocity over baseline to fully open the disk, except for clearway designs which will require velocities more than 100 percent higher than baseline.

6. The amplitude of disk motion which occurs before fully seating the disk increases as the flow disturbance is brought closer to the valve. The maximum disk fluctuations in the case of a severe turbulence source can reach as high as 16 degrees and for elbows up to 9 degrees. The reducers have negligible effect on disk fluctuations.

7. Suitable nondimensional parameters were chosen in the test program to check the validity of the results when scaling from one size to another. The 3-inch and 6-inch test results show that reasonable predictions can be made while scaling to other sizes.

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INTRODUCTION

Accelerated degradation of check valve internals can lead to their failure to perform, leading to serious safety consequences or causing extensive damage to other plant components and systems. The primary cause of premature degradation is continuous fluctuation of the disk when it is not firmly held against the backstop by the fluid forces pushing on it. The severity of the degradation depends upon the type and severity of disturbances in the fluid system that are present upstream of the valve which disturb the uniform flow and create velocity skew or fluctuations.

Disk fluctuation can be eliminated by ensuring that the valve is sized to create sufficient flow velocity through it to fully open the disk against its backstop. However, many valve manufacturers lack accurate data to make minimum velocity (V_{min}) recommendations concerning their valves. A generalized V_{min} formula has recently been developed (References 5, 23) for swing check valves which has been shown to correlate well with experimental results covering a wide range of sizes and pressure ratings. It should be pointed out, however, that this formula is applicable only to installations in which fully developed pipe flow, unaffected by upstream disturbance, is entering the valve.

In order to predict the performance of check valves in practical installations at nuclear power plants, additional data are needed to properly account for the effect of flow disturbances that are commonly present in close proximity (less than 10 pipe diameters) upstream of the check valve. The minimum velocity, V_{min}, for an ideal pipe flow may not be sufficient to hold the disk fully open under such conditions. One of the objectives of this research is to develop techniques which can be used to predict whether disk stability can be achieved in the presence of such upstream disturbances by increasing the velocity above V_{min} and to develop the velocity margins necessary to suppress disk fluctuations.

Another important objective of this effort is to develop techniques to quantify the magnitude and severity of disk fluctuations under various flow conditions in the presence of these upstream disturbances when the velocity through the valve is insufficient to hold it fully open. These techniques will aid in predicting (1) whether a swing check valve installed with a known upstream disturbance at a defined proximity will suffer from accelerated degradation, and (2) the severity of disk motion which causes that degradation of the internals. Eventually, the results from the successful completion of the research effort envisioned under Phase I and II should be useful in predicting suitable maintenance intervals during which the degradation is within acceptable limits. Within these limits the check valve should continue to perform without compromising the safety or reliability of its operation.

This report summarizes the results of the Phase I effort in which the performance of swing check valves with certain selected upstream flow disturbances was systematically evaluated. In a broad sense the types of flow disturbances that can be found upstream of the check valves fall into two distinct categories:

1. Piping elements such as elbows, reducers, and tees which are quite specific in their geometry,

2. Turbulence sources which includes a large variety of devices capable of creating velocity fluctuations in the flow stream. This includes control valves

of various types - e.g., single ported globe valve, double ported globe valve, multiple hole cage guided trim valves, butterfly valves - pumps, and orifices, etc.

From the first category of flow disturbances, elbows and reducers were selected for Phase I testing since these are the most commonly used piping elements upstream of check valves. Phase I testing also included the investigation of an upstream turbulence source. In order to generically characterize the effect of the various types of turbulence sources described above, orifice plates with multiple holes were used. By varying the size and number of holes, the two important characteristics of turbulence, e.g., eddy size (or scale of turbulence) and the intensity of turbulence could be controlled independently. Multiple-hole orifice plates have the additional advantage of creating uniform turbulence across the pipe cross-section, which provides data of more general utility than some specific throttling devices, e.g., butterfly valves that have been used in the past (References 5, 24), which introduce a velocity skew along with turbulence. Extensive testing was done using several perforated plates to develop upper bounds of disk fluctuation under the most severe turbulent conditions. These bounding disk fluctuation results can be used to make conservative predictions about check valve degradation. Additionally, some interesting trends were found which show how the size of holes in the orifice plates affects the severity of disk fluctuations.

It should be pointed out that until recently there had been no reported results of research directed at systematic evaluation of upstream disturbances on check valve performance, even though some unique configurations have been investigated (References 13, 24, 25). Reference 5 reports the results of the first such effort towards the systematic evaluation of upstream disturbances completed by the investigators in the present research. That work included investigation of elbows and a throttled butterfly valve as a turbulence source. However, the velocity increments used in those tests were relatively coarse and they did not permit an accurate determination of the velocity at which the disk begins to impact the backstop and when it finally reaches its fully open position. Secondly, the butterfly valve introduces a velocity skew along with intended turbulence which prevented any useful velocity margin data over V_{min} to be extracted from these tests. The Phase I tests were designed to specifically overcome these limitations and to develop more general, upper bound data for turbulence as well as reducers.

The tests were performed on two different sizes of swing check valves (3- and 6-inch) with an adjustable stop to simulate the disk angle variations found in valves made by different manufacturers. An important consideration in this Phase I effort was to use methods which can allow the results to be scaled and applied to other sizes. All together, over 2,000 tests explored the effects of elbows, reducers, and turbulence sources upstream of the check valve.

Phase I effort also included a thorough review of the current state of the art in fluid/structure interaction techniques, and their capabilities and limitation to predict disk response to upstream turbulence if the turbulence characteristics are known.

This report summarizes the results of the experimental and theoretical investigation completed under Phase I.

OBJECTIVES

The objective of the Phase I effort was to determine the feasibility of a technical approach in predicting and quantifying the stability or instability of the swing check valve disks in practical piping systems which have a significant flow disturbance within 10 pipe diameters upstream of the valve.

More specifically, the objectives were to

1. Develop a general approach which can be used to modify the V_{min} formula to account for the orientation and proximity of an upstream elbow;

2. Develop a general approach to account for the effect of an upstream reducer on the V_{min} formula;

3. Develop techniques for predicting the amplitude and frequency of check valve disk motion in the presence of an upstream turbulence source. These results can be used in a Phase II program to develop quantitative wear and fatigue prediction techniques;

4. Establish velocity margins above V_{min} necessary to fully open the disk and suppress disk oscillations due to 1, 2, and 3 above.

A test program was devised to meet these objectives. The tests were performed on 3-inch and 6-inch swing check valves supplied by MCC Pacific Valve Company. The valves are similar to that shown in Figure 1. The tests were performed at Kalsi Engineering, Inc.'s flow test facility in Sugar Land, Texas (Figures 2 and 3). Water at outdoor ambient temperature was used as the flow media at velocities between 1.5 and 25 feet per second. All together, over 2,000 tests explored the effects of elbows, reducers, and turbulence upstream of the check valve.

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TEST PROGRAM

Phase I testing was divided into six major areas as follows:

1. **Baseline Tests:** To establish check valve performance with uniform, fully developed flow unaffected by upstream disturbances.

2. **Elbow Tests:** To determine the effect of elbow orientation and proximity on check valve performance. Elbow up and down orientations were tested at upstream distances from 0 to 5 pipe diameters. Based upon earlier test results, the effects of elbows beyond 5 pipe diameters are known to be insignificant.

3. **Reducer Tests:** Similar in nature to the elbow tests, a pipe reducer was installed upstream at distances from 0 to 5 diameters. A reducer is defined as an element that decreases the pipe size when moving in the direction of flow.

4. **Turbulence Tests:** Turbulence was developed by flowing through a perforated plate upstream of the check valve. A selection of plates was available with different numbers and sizes of holes. Geometric scaling of plate geometry was maintained between both valves. The turbulence source was located from 1.5 to 10.5 diameters upstream of the check valve.

5. **Disk Natural Frequency and Damping Tests:** A series of tests were run to measure the disk/hinge arm natural frequency and damping under actual flowing conditions. The data from these tests will be be used in Phase II to develop predictive techniques to quantify wear and fatigue of the hinge pin and disk stud connection, the two known weakest areas in swing check valves.

6. **Tapping Force Tests:** To quantify the magnitude and duration of disk impacts with the backstop, a load cell was incorporated in an adjustable backstop mechanism. Transient disk contact with the backstop could then be recorded as well as the steady-state load when the disk is pegged in the fully open condition.

SCALING CONSIDERATIONS

One of the important considerations in Phase I work was to properly take the pertinent nondimensional scaling parameters into account. Two sizes of valve, 3-inch and 6-inch, were tested so that data can be used to check the validity of how the results scale up between the two sizes using the nondimensional parameters chosen.

The following table lists the dimensional and nondimensional parameters that were used in the tests:

D = Nominal size of the valve; diameter of the upstream pipe

L = Distance between the valve end and upstream disturbance

L/D = Nondimensional distance between the valve end and upstream disturbance; also referred to as proximity of the disturbance

$\Delta\theta$ = Disk fluctuation angle

θ = Total fluid impingement angle against the disk (see Figure 1)
= [90° - (the angle swept by disk from fully closed to fully open position)]

V = Flow velocity through the valve, based on valve inlet area

V' = Velocity required to fully open the valve with an upstream disturbance

V_{min} = Minimum flow velocity required to fully open the disk against the backstop without any upstream disturbance

V'/V_{min} = Nondimensional velocity ratio used to express velocity margins needed over V_{min} to stabilize the disk in the presence of an upstream disturbance

C_{up} = (Same definition as ratio V'/V_{min}) A nondimensional velocity margin factor to account for upstream disturbances

Standard short-radius 90-degree elbows and concentric reducers manufactured by Victaulic Company were used in these tests. They were dimensionally inspected to determine what, if any, deviations exist in the geometrical scaling between the 3- and 6-inch sizes. It was found that the ratio of mean elbow radius and the inside diameter of the elbow varied by approximately ± 10 percent around the mean value of 1.075. The angle of the concentric reducers was also found to be consistent: between 14 and 15 degrees for both reducers, which equates to a total included angle of about 30 degrees.

Geometrical similarity was also maintained in the design of various multiple-hole orifice plates used in tests on the 3-inch and 6-inch valves. The size and number of holes used in these plates is shown in Table 3. The hole size governs the size of eddies emitted by an orifice plate. The ratio between the eddy size and physical dimensions of the check valve disk is an important nondimensional scaling parameter for fluid/structure interaction caused by turbulence.

TEST SPECIMENS

The valves used throughout the test program were 3-inch, 300-pound and six-inch, 600-pound clearway swing check valves. Dimensional and physical data are presented for both valves in tables 1A and 1B. Both valves were modified to incorporate an LVDT (Linear Variable Differential Transformer) type of displacement transducer for measurement of disk/hinge arm motion and an adjustable backstop assembly (Figure 4). The adjustable backstop takes the place of the fixed backstop which is normally cast integrally into the bonnet in these valves. The adjustment mechanism allows the stop position to be varied, thus simulating different full-open disk positions covering the range of variations found in valves made by different manufacturers. In addition, the backstop assembly incorporates a load cell which provides direct measurement of the loads developed whenever the disk contacts the stop.

Table 1A
VALVE DATA FOR 3-INCH VALVE

Valve Type	3", 300# Swing Disk Check
Manufacturer	MCC Pacific Valve Company

Weight and Dimensional Data

Disk Weight (incl. nut)	1.94 lbs.
Hinge Arm Weight	0.72 lbs.
Hinge Pin Diameter	0.375"
Disk O.D.	3.90"
Seat Bore Diameter	3.0"
Seat Tilt From Vertical	3°
Full Open Angle	73°

Table 1B
VALVE DATA FOR 6-INCH VALVE

Valve Type	6", 300# Swing Disk Check
Manufacturer	MCC Pacific Valve Company

Weight and Dimensional Data

Disk Weight (incl. nut)	8.94 lbs.
Hinge Arm Weight	3.38 lbs.
Hinge Pin Diameter	0.500"
Disk O.D.	6.90"
Seat Bore Diameter	6.0"
Seat Tilt From Vertical	3°
Full Open Angle	70°

INSTRUMENTATION

Design Features
The load cell apparatus incorporates a spring to preload the stop rod against the load cell, thus taking up any clearance that might exist. This feature prevents unwanted load amplification that can be caused by impact of the stop rod against the load cell if any clearance is present between them. A static seal was used to isolate the load cell from the valve interior. This results in a fictitious disk-to-stop load in response to valve internal pressure. To minimize this load, the pressure effective area was made as small as practical resulting in a pressure induced load of 2 pounds at 75 psig.

Disk motion measurement was accomplished by attaching the LVDT core directly to the hinge arm with a stiff plastic tube capable of transmitting push/pull motion while retaining the ability to flex slightly to accommodate kinematic restraints of the arrangement. This connecting link transformed the rotary motion of the hinge arm into linear displacement of the LVDT core. Calibration of the apparatus against disk angle proved the mechanism to be nearly linear with a worst case error of ± 0.5 degrees. Under certain flow conditions turbulence in the valve can buffet the connecting link causing displacement of the LVDT core not associated with disk motion. This phenomenon was eliminated by placing a protective aluminum tube over the connecting link to shield it from the turbulence.

Flow rates were calculated by measuring the pressure drop across an orifice plate. The entire orifice meter run is constructed in accordance with the requirements of ANSI/API 2530. Three different orifice sizes were used to provide flow accuracy of ± 0.5 percent over a range of 1.3 to 27 feet per second (water). This flow range exceeds that required for the present test program.

The following parameters were measured and recorded during each of the tests performed: disk motion, backstop load, flow orifice ΔP (for flow velocity), and pressure upstream of the check valve. In addition, during the turbulence tests, a second pressure measurement was added to monitor pressure drop across the perforated plate. Each of the analog signals was acquired through a 12-bit analog-to-digital converter, then stored on disk. A test log was also printed out documenting important test parameters and summary test data.

TEST RESULTS

Identical tests were run on both 3-inch and 6-inch swing check valves. The flow velocity range spanned 1.5 to 25 feet per second, using water. Flow velocity increments were varied to adequately characterize disk motion through the velocity range. Therefore, in the velocity range where the disk begins to tap against the stop, relatively fine velocity increments were taken, typically 0.1 to 0.2 fps. Once the disk was pegged, the velocity increments were increased to 2 fps. This methodology allowed a more precise determination of the velocity at which tapping begins and the velocity at which the disk becomes pegged against the stop. On average, 20 separate tests were needed to cover the 1.5 to 25 fps range at one disk stop angle.

Data were taken at stop angles of 53 degrees, 63 degrees, and 73 degrees for the 3-inch valve and 50 degrees, 60 degrees, and 70 degrees for the 6-inch valve. The 73- and 70-degree positions represent the as-manufactured full-open angle for the 3-inch and 6-inch valves respectively.

BASELINE TESTS

Baseline tests establish performance data when the valve is presented with, fully developed pipe flow. To provide these flow conditions, 20 diameters of straight pipe incorporating a section of flow straightening vanes was installed upstream. This upstream configuration exceeds ANSI/API Standard 2530 requirements for a flow metering piping run. Results of the baseline tests for the 6-inch valve are presented in Figure 5, which shows peak-to-peak disk motion reported as $\pm 3\sigma$ (standard deviations) from the mean disk position plotted against flow velocity. Data for all three stop positions are shown. Inspection of Figure 5 shows that the highest levels of disk motion reaching 1.5 to 2 degrees, occur between 4 and 6 fps. We can conclude that disk fluctuation values of about 2 degrees (peak to peak) are the lowest that can be expected, even with the best upstream conditions we can create. This minimum amount of disk motion must be expected whenever V_{min} has not been reached. Figure 5 can be used to point out other important aspects of valve behavior. First, disk motion occurs over a much wider range of velocities for the 70-degree stop angle than at 50 degrees and 60 degrees. This can be seen by measuring the velocity range over which disk fluctuations exceed 1 degree[*]. Table 2 shows the velocity range and span in fps for the three stop angles. This range is also indicated in Figure 5.

Table 2

6-INCH VALVE DISK MOTION

Stop Angle	Velocity Range	Velocity Span, fps
50°	3.8 - 4.9	1.1
60°	3.8 - 6.5	2.7
70°	3.8 - 13.8	10.0

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[*] Because of mechanical clearances in the LVDT assembly are variable, we have established a 1° fluctuation level ($\pm 0.5°$) as the threshold below which we consider the disk actually seated against the backstop.

The wide velocity span over which disk motion occurs at the 70-degree stop angle is inherent in the design of a clearway swing check valve. As was reported in Reference 5, this is due in large part to the shielding effect of the valve body/bonnet area when the valve is opened to the clearway position. Figure 6 shows in cross-section how the clearway design causes the disk to be lifted very nearly out of the normal flow stream through the valve. Measurements of the two valves show that the 6-inch disk protrudes 0.8 inches into the flow stream while the 3-inch disk has no protrusion. Data from the 3-inch valve baseline tests are presented in Figure 7. Again, the maximum fluctuation levels are in the 2-degree range and the range over which disk motion occurs is much wider at the 73-degree stop angle than at 63 degrees or 53 degrees.

ELBOW TESTS
Tests were run with elbows at proximities of 0, 1, 3, and 5 diameters upstream of the check valves and oriented both up and down. Tests were not run at distances beyond 5d because results reported in Reference 5 showed that at distances of 5 diameters and beyond, the effects of elbows become very small, approaching the performance achieved with baseline conditions. To avoid confusion, Figure 8 shows sketches representing the definitions used in this text for elbow up and elbow down orientations.

A detailed review and comparison will be made of the elbow test results at 0d for the 3- and 6-inch valves. Remaining data will be covered later in this section.

Disk Fluctuations for Elbow Up, 0d
Figures 9A and 9B show the results of disk fluctuation vs. flow velocity tests for the 3- and 6-inch valves. Maximum disk motion for both valves occurred when the stop was positioned at the clearway positions of 73- and 70 degrees for 3- and 6-inch valves respectively. Values of approximately 9- and 6 degrees were recorded for the 3- and 6-inch valves respectively. The levels of fluctuation at the 60°/63° and 50°/53° stop positions were lower than those reached at the clearway position. As seen in the baseline data, the velocity range over which disk motion occurs is much wider at 70°/73° than at the lower angles. Most interesting, however, is the fact that for the 3-inch valve a stable, fully open disk position was never achieved at the 73-degree clearway stop position even at velocities of 24 to 25 fps. This phenomenon did not occur during elbow tests reported in Reference 5 which showed the 3-inch valve reaching a fully open condition at 18 fps. This demonstrates the extreme sensitivity of the clearway design to minor changes in flow conditions or actual positioning of the disk near the nominal 73-degree position. The 3-inch valve, with no disk projection into the flow stream, is very sensitive to physical changes in the piping setup or conditions inside the valve. The 6-inch valve, with some disk projection into the flow stream, is less sensitive to these changes.

Disk Fluctuations for Elbow Down, 0d
Figures 10A and 10B show the elbow down test results for the 3- and 6-inch valves. The same conclusions are reached as for the elbow up case, except that the maximum disk fluctuations are not as high. This is due to the fact that the highest levels of turbulence exist along the inner radius of an elbow (Reference 6); thus, the elbow down orientation positions this more turbulent portion of the flow further from the valve disk than the elbow up orientation. Once again the 3-inch valve exhibits the inability to reach a stable, fully open position.

Figure 11 compares the maximum disk fluctuation for both valves at each location tested. To normalize the data, both horizontal axes have been offset to account for what

amounts to a built-in length of straight pipe between the flange face and the disk region inside the valve. This distance measures 2d and 1.5d for the 3- and 6-inch valve respectively. The highest levels of disk motion occur at elbow proximities of 0d and generally decrease as the elbow is moved further from the valve.

Disk Tapping Zones

Another important aspect of check valve behavior is disk tapping. Disk tapping occurs when the flow velocity is not quite sufficient to firmly seat the disk against the stop, so that fluctuations in disk motion cause the disk to repeatedly tap against the stop. Tapping stops when flow velocity is raised to the point where the disk becomes firmly seated or pegged. By carefully determining the flow velocity at which tapping begins[*] and the velocity at which the disk becomes pegged, the tapping "zone" can be defined in feet per second.

By examining the elbow test data, the following observations about disk tapping can be made:

1. The onset of tapping begins at lower flow velocities when the elbow is oriented up than when oriented down. This results in wider tapping zones for elbow up installations compared to elbow down. This is particularly evident at 0d and 1d and less so at 3d and 5d.

2. The tapping zone is widest with the stop angle set at 70°/73°, typically spans a magnitude of 2 to 4 fps.

3. The tapping zone is very narrow when the stop position is decreased to 60°/63° and 50°/53°. At these stop positions, the onset of tapping very nearly coincides with the velocity at which maximum disk motion occurs. A very small increment in flow velocity is then required to peg the disk; typically less than 1 fps at 60°/63° and less than 0.5 fps at 50°/53°.

Disk tapping plays an important role in fatigue and impact wear of the disk stud and wear of the hinge pin. Once again, the clearway design is seen to be most susceptible to damage because of the wider tapping zone and the fact that the zone lies at velocities in the 12 to 15 fps range commonly encountered in check valve installations.

A predictive wear and fatigue model should be developed by combining this data with wear data for specific material combinations gathered as part of a Phase II program.

Velocity Margin Over V_{min} Required for Elbows

Figures 12 and 13 show the results for 3- and 6-inch valves for the velocity required for full disk opening at each stop angle plotted against the nondimensional distance to the elbow, L/D. For both valves, it can be seen that at distances of 0d to 1d, the velocity required to fully open the disk can be as much as 75 percent higher than baseline in the clearway position, and, as noted earlier, may never actually become stable in the case of the 3-inch valve at 0d. At 3d and beyond, the velocity margin required drops to 10 percent over baseline conditions.

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[*] The onset of tapping was defined as four disk-to-stop contacts in ten seconds.

At the lower disk opening angles, the velocity margins required decrease markedly. At 60°/63°, a 35 percent margin is required at 0d and 1d and 10 percent elsewhere. At 50°/53°, a 15 percent margin is required at 0d and 1d and 10 percent elsewhere.

REDUCER TESTS
A common piping system element found upstream of check valves is the pipe reducer. Many reducer geometries exist with various ratios of inlet-to-outlet size and overall length, as well as being fabricated or forged, concentric or eccentric. The sizes chosen for these tests were 4 x 3 for the 3-inch valve and 8 x 6 for the 6-inch valve. This selection provides a reduction of one nominal pipe size in both cases, and maintains geometric similarity in both cases by providing a reduction ratio of 0.75. The reducers were 3.5 and 5 inches long in their overall length for the 3-inch and 6-inch valves respectively. The actual slope of the wall in the reduction section was close to 15 degrees (30 degrees total included angle) for both reducers.

Tests were run with the reducer placed at the same distances from the valve as the elbows: 0, 1, 3, and 5 diameters. Figure 14 shows disk fluctuation data compiled from tests of the 6-inch valve with reducer at 0d. Interestingly, it looks very much like the baseline test data of figures 5 and 7 with maximum disk fluctuations in the 2- to 2.5-degree range occurring between 4 and 6 fps. Velocities at which the disk reaches the fully open position are very close to baseline levels. Data from the other proximities (1, 3 and 5d) as well as the 3-inch tests show very similar performance. Unlike elbows, there is no characteristic increase in disk instability as the reducer is brought closer to the valve. This is easily seen by reexamining figures 11 - 13, which show summary data for the reducer tests. In Figure 11, it can be seen that maximum disk fluctuations for both valves at any distance are well within the levels of baseline fluctuations shown by the dashed lines.

Figures 12 and 13 show the velocity required to fully open the disk at the different stop angles. In each case, for both valves and at all stop angles and distances, the required velocity is within ± 5 percent of baseline. These results are in agreement with expected behavior based on theoretical formulations in Reference 7 where it is shown that, in fact, a reducer will tend to decrease the level of axial velocity variations by an amount approximately equal to λ^2 where λ = ratio of the contraction. For the reducers tested here, the axial velocity variation could be expected to be reduced by $\lambda^2 = (0.75)^2$, or 56 percent of the original velocity fluctuations. This example applies to an ideal case, neglecting viscous effects, and compressibility and allowing no geometric discontinuities which will otherwise disturb the flow field. This also assumes no contribution of radial velocity variations to disk instability. Although reduction of this level will not be realized in a real application, it does show that an upstream reducer will tend to reduce axial flow fluctuations or turbulence intensity and resultant disk motion as observed in our tests.

Velocity Margin over V_{min} for Reducers
To summarize the reducer test results, it can be stated that the reducers do not have any detrimental effect on the stability of the swing check valve disk. There is essentially no velocity margin required above V_{min} to suppress disk oscillations due to this type of flow disturbance. In fact the reducers have a beneficial effect of decreasing the turbulence intensity that is responsible for causing disk fluctuations.

TURBULENCE TESTS

Test results reported in Reference 5 concluded that using a partially closed butterfly valve as a throttling device generated high levels of turbulence. However, flow over the canted disk of the butterfly valve resulted in a highly skewed velocity profile entering the check valve. The superposition of velocity skew and turbulence masked the individual effects of the two independent phenomena, making it impossible to determine the velocity margin over V_{min} due to turbulence effects alone. To provide controllable levels of turbulence without velocity skew for this series of tests, a special set of perforated plates were constructed which could be placed at a variety of upstream locations.

When considering turbulence sources which might affect check valve performance, the search quickly becomes overwhelming. Many piping system elements are sources of turbulence. Control valves of all types, pumps, pipe fittings such as tees and elbows, all contribute to turbulence which will affect check valve performance. Characterizing the effects of many different variations of control valves, pumps, or other throttling devices on an individual basis would be an impossibly large task. Instead, we took the approach of using a relatively simple but generic source of turbulence which could be easily modified or scaled.

Multi-Hole Orifice Plate as a Turbulence Source

The device settled upon for generating turbulence is a plate of stainless steel with a pattern of drilled holes captured in a holding fixture fitted with pipe stubs. This allowed for easy axial positioning of the turbulence source with respect to the check valve as well as interchangeability of the perforated plates. Figure 15 shows the plate holder and an assortment of plates used for the 3- and 6-inch valve tests, respectively. Data regarding perforated plate geometry are presented in Table 3.

Table 3

PERFORATED PLATE DATA

3-Inch Valve		6-Inch Valve	
No. of Holes x Hole Diameter (inches)	Area Ratio*, α	No. of Holes x Hole Diameter (inches)	Area Ratio*, α
36 x 3/16	.13	36 x 3/8	.13
61 x 3/16	.23	67 x 3/8	.26
3 x 7/8	.24	3 x 1 3/4	.24
4 x 7/8	.33	4 x 1 3/4	.33

* Area Ratio = Total Area of Holes/Pipe Flow Area

One important characteristic of turbulence which we wanted to simulate is the size of the turbulent eddies developed by different devices. Caged control valves can have numerous small holes through which the flow will pass. The smaller (3/16 inch and 3/8 inch) holes were chosen as representative of that class of device and most likely the smallest eddies to be encountered. At the other end of the spectrum, we chose the largest hole size for a three-hole orifice geometry. Three seemed to be the least which could be

used without introducing too much bias or unevenness into the flow field across the pipe. Selection of the total flow area in the orifice, which is determined by the number and size of holes, was guided by the pump's ability to deliver the required flow while still preventing cavitation at the perforated plate. To prevent excessive cavitation, the cavitation index was continuously monitored during the tests. The cavitation index was calculated according to the formula:

$$\sigma = \frac{P_{down} - P_{vap}}{P_{up} - P_{down}}$$

where

σ = Cavitation Index

P_{up}, P_{down} = pressures upstream and downstream of the perforated plate, psig

P_{vap} = Vapor pressure of fluid, psig

For water at 100°F, $P_{vap} \approx 1$ psig

So the formula simplifies to

$$\sigma = \frac{P_{down} - 1}{\Delta P}$$

Therefore, a smaller cavitation index, σ, means higher levels of cavitation.

Work by Tullis with single and multi-hole orifices, skirted control valves, and butterfly valves (References 8,9) has helped define a range of operating regimes with differing cavitation characteristics ranging from incipient cavitation at the mild end to choked flow at the severe end. Incipient cavitation can be characterized by sounds similar to the crackling of frying bacon or popping corn. This is a very mild level of cavitation. For the perforated plates used in these tests, this level of cavitation was typically reached with cavitation indices of about $\sigma = 0.5$. All tests were performed with $\sigma = 0.7$ as a minimum to guarantee no influence of cavitation.

Turbulence tests were run at flow velocities from 2.5 to 18 fps. Given the pump characteristics and cavitation criterion, higher flow velocities could not be developed. Nevertheless, 18 fps was enough to fully open the disk against the backstop in all cases. Tests were run at all three stop angles and at proximities of 1.5, 2.5, 4.5, and 10.5 diameters upstream. In general, the 3-hole and 36-hole plates were used to cover the flow ranges from 2 to 9 fps, and the 4-hole and 61/67 hole plates were used to cover the range from 7 to 18 fps. It should be remembered that pressure drop across any perforated plate increases with the square of the flow velocity. Thus, at the low end of the two velocity ranges just mentioned, plate ΔP was in the 2 to 4 psi range while at the high end of the ranges the pressure drop was in the 25 to 35 psi range. Detailed discussion of results for the 3-inch valve at 1.5d and 73-degree stop angle will be covered, with the remaining data summarized later in this section.

Disk Fluctuation Data, 3-inch Valve

Figure 16 shows the disk fluctuation data for the 3-inch valve with the stop angle at the 73-degree clearway position and turbulence source at 1.5d. The overall plot represents data from all four of the perforated plates used in the test program. Some important trends should be noted:

1. Peak disk fluctuations are considerably higher than for any of the elbow tests, reaching levels of over 15 degrees (peak to peak, 3σ).

2. The highest levels occur in the 4 to 6 fps range with the 36-hole plate. As shown in Table 3 data for various orifice plates, this plate has the smaller (3/16") diameter holes.

3. The velocity required to fully open the disk is in the 17 to 18 fps range based upon disk fluctuation levels dropping to under 1 degree. This agrees closely with the baseline data.

4. The darkened symbols represent those tests where tapping of the disk against the stop begins as flow velocity is increased. Compared to the elbow tests, the velocity zone over which tapping occurs is wider. This is the zone over which accelerated fatigue and impact wear occurs.

Figures 17, 18, and 19 show the fluctuation data for the remaining proximities of 2.5, 4.5, and 10.5d at the 73-degree stop angle. On each plot an upper bound curve has been drawn for disk fluctuations at each proximity.

The upper bound data for the most severe disk fluctuations as a function of both flow velocity and proximity of the turbulence source shown in previous figures are presented in a more comprehensive manner in Figure 20. As one would expect, at any given velocity, maximum fluctuations decrease as the turbulence source is moved away from the valve. Also, at all proximities the maximum fluctuations occur at around 6 fps.

To complete a set of turbulence data at all stop angles, the 53-degree and 63-degree fluctuation data at 2.5d is shown in Figures 21 and 22 respectively. We see that peak fluctuations are lower than at the 73-degree stop angle, and the velocity range over which disk motion occurs gets progressively narrower as the stop angle is decreased. Additionally, the velocity range between onset of tapping (darkened symbols) and disk seating (fluctuations less than 1 degree) is much narrower as well. These trends were also present in the elbow data. Even though disk fluctuations are much higher than were seen for the elbow tests, the velocity margin required over baseline to firmly seat the disk is very similar, approximately 30 percent at 63 degrees and 15 percent at 53 degrees.

Disk Fluctuation Data, 6-inch Valve

Figures 23 through 29 represent the disk fluctuation results for the 6-inch valve with various orifices and their proximities. All the trends noted for the 3-inch valve data still apply to the 6-inch valve with the exception that at 2.5d the peak fluctuations are slightly higher at velocities above 6 fps than at 1.5d. This can be seen in Figure 27 where the data are summarized. A comparison between Figure 27 and Figure 20 shows that the overall magnitudes of the disk fluctuations are approximately the same for both the 3-inch and 6-inch valves, confirming the validity of scaling up of the results in these tests.

Summary of Maximum Disk Fluctuations for
Elbows, Reducers, and Turbulence Sources

Figure 30 is an overall summary and comparison of maximum disk fluctuation data for all of the upstream disturbances tested for both valves at each proximity. Clearly, the turbulence source has the most severe effect on disk stability, with fluctuation magnitudes reaching as high as 16 degrees at the closest proximity. As expected, the disk fluctuations created by the turbulence source continue to decrease as the proximity from the check valve is increased, but even at 10.5d the magnitude is as high as with elbows at 5d. One can also see that both the 3-inch and the 6-inch valves show overlapping data which provides some confirmation of the fact that disk motion expressed in nondimensional terms of fluctuating angle $\Delta\theta$ is a valid scaling approach. Comparisons made against published data by others show that our results bound the reported magnitudes of disk fluctuations with other turbulence sources which include control valves, throttled butterfly valves, and pumps as well as the effect of larger check valves (References 5, 10, 13, 24, 25). Therefore, the upper bound data developed from Phase I experiments could be used in predicting maximum wear or fatigue that can be caused by disk motion.

The elbow data for both valves shows that the overall disk motion is considerably less than with a turbulence source and is typically in the range of 3 to 5 degrees. The maximum fluctuations at the closest proximity of 0d are between 6 and 9 degrees range for the elbow up condition, which is the most severe elbow orientation. As reported in previous elbow results (Reference 5), the disk fluctuations become negligible beyond 5d.

One of the most interesting results was the performance of the check valve with reducers as an upstream disturbance. As shown in this figure, the disk fluctuations with the reducers even at the closest proximity to the valve are within the baseline fluctuations that were measured with 20d length of upstream straight pipe which included a flow straightening section. The results show that the reducer employed to install a smaller size check valve in a larger diameter upstream pipe is not a detrimental disturbance in such configuration. One should be cautioned, however, that the reverse situation where a reducer is being used to install a larger size check valve in a smaller size upstream pipe (which in fact makes the reducer function as a flow expander) is known to be a definite source of turbulence. Fortunately, for swing check valves such installations are rare in typical power plant systems.

DISK NATURAL FREQUENCY AND DAMPING TESTS

By combining data for amplitude of disk motion with disk natural frequency data, estimates for hinge pin wear and disk stud impact wear and fatigue can be made. To measure the disk natural frequency under flowing conditions, a simple test was run on each valve where steady state flow conditions were established with no upstream disturbance. A pushrod was introduced through the bonnet and used to quickly disturb the disk from its equilibrium position. A strip chart recording was made of the resulting response to the disturbance. Figure 31 shows a section of a typical recording. As is evident from this recording, the disk response is relatively highly damped. By measuring successive peaks from a number of tests, the damping ratio is calculated as 0.5 ± 0.1 for the 6-inch valve and 0.3 ± 0.1 for the 3-inch. The natural frequency tests were performed at a number of flow velocities which correspond to different equilibrium positions of the disk. The results are presented in Figure 32.

An analytical formulation for to predict natural frequency was also developed by considering the disk as a single degree of freedom spring-mass system. This is presented in Appendix A and the results from it for the 3-inch and the 6-inch valves are plotted in Figure 32 for comparison. Another comparison was made against work by Sununu (Reference 10) which suggests that the disk natural frequency is proportional to the eddy frequency generated in normal turbulent pipe flow. This frequency can be calculated as follows:

$$f_{eddy} = \frac{0.04\ V_{avg}}{R_{pipe}}$$

Test data published in Reference 10 for 2-inch and 4-inch valves is also plotted in Figure 32 for comparison. Two trends are apparent. First is a trend of increasing disk natural frequency with decreasing valve size. Second is an increasing natural frequency with increasing velocity. Unfortunately, not enough data are available to draw firm conclusions about the trends. Other factors such as effective disk weight and valve internal geometry may have some effect on the natural frequency. In fact there does not appear to be a linear relationship between natural frequency and velocity for majority of the data as the previous formula suggests. Nonetheless, these measurements give a reasonable quantification of disk natural frequency and form a basis for performing wear and fatigue estimates.

DISK-TO-STOP IMPACT LOAD TESTS
To calculate impact fatigue life of the disk stud/nut assembly when the disk is tapping against the stop, estimates of the impact force can be made based upon the transfer of kinetic energy of the moving disk and hinge arm assembly into strain energy absorbed by the contacting elements. With the load cell apparatus incorporated in the stop mechanism, actual load measurements were made for the 3-inch valve to characterize the load signature when tapping. It was found that the data acquisition rate had to be increased considerably - from 40 hz (which is sufficient for the measurement of displacement and pressures) to 4 khz - to properly capture the load/time history.

Figure 33 shows the results of one such measurement made on the 3-inch valve under the following conditions: flow velocity of 17.5 fps, an upstream elbow oriented up at a proximity of 1d, the valve stop angle set at 73 degrees. A number of such measurements were made under the same flowing conditions, with the highest recorded loads being 35 pounds. These measurements are, of course, with the clear plastic bonnet and the adjustable stop mechanism which was being used in the test instead of the actual steel bonnet with integral stop. In all cases, the contact duration was between 0.003 to 0.0035 seconds. To use this load data, it is necessary to account for the relative stiffnesses of the test valve bonnet/stop assembly and the actual valve. These stiffnesses can be determined by testing or analysis. One method would be to apply known loads to each assembly and measure the displacements. Appendix B documents an analysis method and a calculation in which the relative stiffnesses are estimated. Based on the 35-pound load measured and this analysis approach, it is estimated that the actual impact load with steel bonnet having integral stop arrangement will be approximately 270 pounds for the 3-inch valve. The impact force results thus obtained can be used in predicting impact wear between the disk/stud nut and the stop as well as fatigue life of this assembly.

DISCUSSION

CLEARWAY SWING CHECK PERFORMANCE PROBLEMS

A very important result of the elbow tests conducted in this project is the apparent discrepancy between these tests and those reported in Reference 5 for the minimum velocity requirements for the 3-inch valve with stop angle at the 73-degree clearway position and elbows at 0d. The earlier tests reported values of 17 fps for elbow down and 18 fps for elbow up. The current test program did not fully seat the disk at velocities as high as 25 fps. Possible reasons for that difference can be categorized as follows:

1. Differences in flow velocities/conditions;

2. Variations in inlet piping setup;

3. Minor differences in physical location of stop position(s).

The first item is unlikely because no evidence of velocity based errors shows up in other tests. All equipment was calibrated and, in fact, was the same in both tests.

The second item could conceivably contribute if, for example, the elbow was tilted slightly as it was bolted to the valve. This would cause the flow to enter the valve at a different angle relative to the nominal flow axis, in effect changing the full open angle of the valve. This possibility exists because of the flexibility of the elbow/valve connection and the difficulty in establishing the elbow orientation (using a bubble level) at this 0d proximity. Finally, physical changes made to the valve between test programs could have affected the absolute position of the adjustable stop. Since different assemblies were used for the adjustable stop devices, there could have been some offset introduced at each stop position. Although great care was taken to prevent this, a difference between assemblies of 1 degree is a possibility.

The second or third items could create enough of a difference in this valve to account for the observed phenomenon. The large 73-degree opening angle of this valve, combined with the clearway design in which the disk is completely shielded from direct flow impingement, makes it very susceptible to these physical changes. It is important to realize that there are real-world analogies to both of these test related items. First, a real piping installation will be as likely to have small angular alignment variations between valve and pipe as the test setup, if not more so. A pipe could very easily be welded to a valve or flange in such a way as to introduce a couple of degrees of misalignment between the pipe and valve centerlines. Second, valve manufacturing tolerances are such that the full open angle of the disk can vary by as much as 5 degrees, which has been observed to be a typical variation in product lines made by most manufacturers. This is considerably more than the variation between test setups. Even a comparison between the 3-inch and the 6-inch valve (both products of MCC Pacific Valve Company) in the full open clearway position for the valves used in our tests shows a significant difference in the velocity required to fully open the disk.

For these reasons, sufficient velocity margins should be used for clearway designs. To avoid possible problems, it is best to lengthen the backstop to ensure the disk protrusion into the flow stream by at least 10 degrees, which has been found to provide much better disk stability under various flow conditions and upstream disturbances.

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VELOCITY MARGIN REQUIRED FOR ELBOWS AND REDUCERS

One of the project objectives was to develop appropriate modifications to the minimum velocity formula to account for upstream elbow and reducer effects and develop the velocity margins necessary to fully open the disk under these conditions. The data presented in Figures 12 and 13 form the basis for these modifications. The appropriate margins for upstream disturbances are most conveniently expressed as a multiplying factor:

$$V'_{min} = C_{up} V_{min}$$

where V'_{min} = minimum velocity required to fully open the disk with an upstream disturbance

V_{min} = Minimum velocity required to fully open the disk with no upstream disturbance

C_{up} = The velocity margin factor accounting for an arbitrary upstream disturbance.

Table 3 presents values for the factor C_{up} for the different upstream disturbances. Keeping in mind some of the test variables, we have chosen to use factors we feel represent the worst case conditions. Although this may result in somewhat higher velocities being required in certain instances, the error must be conservative to ensure safe performance.

Table 3
VELOCITY MARGIN FACTORS FOR
UPSTREAM DISTURBANCES, C_{up}

	Elbows			Reducers		
	50°	60°	70°	50°	60°	70°
0 to 1d	1.2	1.4	1.8	1.05	1.05	1.05
1 to 3d	1.2	1.2	1.4	1.05	1.05	1.05
3 to 5d	1.1	1.1	1.1	1.05	1.05	1.05
Beyond 5d	1.0	1.0	1.0	1.0	1.0	1.0

THEORETICAL PREDICTION OF
TURBULENCE INDUCED DISK MOTION

Discussion of the turbulence test results always involves the issue of relating laboratory results to the many different real-world situations which exist. One approach, of course, would be to test all the different types of turbulence producing devices individually for their effect on check valve performance. An effort this extensive would involve many man-years of test and analysis time and would invariably remain incomplete as new devices arrive on the scene. An alternate method would be to use turbulence characteristics of the flow to predict check valve disk

response. Research in the area of fluid-structure interaction has resulted in a general formulation of the type given below which can be used to predict disk response to a turbulent flow stream (References 11, 12, and 14 through 22).

$$X_{rms} = \left[\frac{\pi}{2} x \frac{S_F(f)}{K^2} x f_n x Q x J^2(f) \right]^{1/2}$$

where X_{rms} = Root mean square displacement response, inch

$S_F(f)$ = Power spectral density of the fluctuating forces over the disk area, lb^2/hz. This can be obtained by multiplying the power spectral density of pressure field, expressed in psi^2/hz, by the disk area, in^2

K = Slope of the fluid dynamic force on the disk vs. linear displacement, lb/in (see Appendix B)

f_n = Natural frequency of the disk/arm assembly dangling in the flow stream

Q = Transmissibility at resonance = 1/2 C_c

C_c = Critical damping ratio

$J(f)$ = Joint acceptance for the generalized force power spectra at frequency, f

The disk natural frequency and damping were measured, the results of which are discussed in this report. An alternate analysis method to calculate the natural frequency of the disk is shown in Appendix A, which also includes a method to determine effective stiffness, K, for the disk fluid dynamic force vs. position curve. Thus, analytical methods have been developed which can provide a reliable quantification for calculating f_n, K, as well as Q; and the remaining variables are S_F and $J(f)$. These pertain to the turbulence power spectral density and how the pressure fluctuations in the turbulent flow stream are time correlated across the disk area. Joint acceptance factor, $J(f)$, can have a possible maximum value of 1 for perfect span-wise correlation of pressure fluctuations and a minimum value of zero for random fluctuations having no phase correlation whatsoever.

Preliminary analysis of pressure transducer data to determine the turbulence power spectral density has been performed, and a general trend of higher disk motion with increasing power spectral density of the pressure field in the turbulent flow has been found to exist. However, considerably more effort is needed in this area to better correlate disk motion to turbulence phenomena by using this power spectral density approach.

A thorough review of literature in the area of fluid structure interaction revealed that, even though the theoretical formulations based on power spectral measurements for predicting mechanical response of an elastic system due to turbulent flow have been generally well established, considerable work is still under way to develop reliable

correlation between theoretical predictions and actual tests (References 11, 12 and 14 through 22). State of the art in this very active area of research still relies heavily upon sound experimental measurements which then form the basis to refine the theoretical models that predict structural response. Vibration response of cylindrical tubes subjected to cross flow and axial flow is one subject that has been studied extensively by many different researchers because of its obvious importance to the nuclear power reactors, heat exchangers, and condensers. This has resulted in significant advancements in the development of analytical models, but they are limited to the cylindrical tube structures. Even for this well-studied structure, not much data regarding power spectral density of the pressure field are available in the open literature.

Therefore, recognizing the limitations of the current state of the art in this field and the long-term research effort needed to develop reliable analytical methods to predict disk motions, the effort under Phase I put more focus on developing the needed information by systematic experimentation using a turbulence source of generic nature. In order to ensure that the results have general applicability, suitable nondimensional scaling factors were used in our test program.

As presented in detail in the previous sections, many tests were conducted with multi-hole orifice plates in which different sizes and number of holes were used to control the two important characteristics of turbulence: eddy size and turbulence intensity. Proximity of the turbulence source was varied up to 10.5d from the valve. Extensive data were generated for both 3-inch and 6-inch valves which are presented in Figures 16 through 26, 28, and 29. The most important results from these turbulence tests, however, are presented in the family of bounding curves which relate the maximum disk fluctuation experienced as a function of flow velocity and proximity of the turbulence source and are shown in Figures 20, 27, and 30. Any data available from published sources quantifying the disk response due to various turbulence sources (References 5, 10, 13, 24, 25) were reviewed and compared against these bounding curves. It was found that in no case did the maximum disk fluctuation exceed the values plotted in this family of bounding curves.

Thus, these bounding curves, which were produced from our Phase I effort, can be used to predict maximum swing check valve disk response with a severe turbulence source present at a defined proximity upstream of the valve.

RESULTS AND CONCLUSIONS

Phase I has resulted in the development of extensive data and techniques that can be used to predict the stability of swing check valves under three different types of upstream disturbances: elbows, reducers, and turbulence sources. The major conclusion from Phase I is that the swing check valve disk performance can be predicted. The specific predictions include:

- Whether disk stability can be achieved for a swing check valve of known disk angle with an upstream disturbance at a defined proximity.

- The increase in velocity required over V_{min} to achieve stability when a disturbance is present upstream of the valve;

- The maximum disk fluctuations that can be expected due to these upstream disturbances. To predict the worst case disk fluctuations caused by a severe upstream turbulence source, upper bound curves have been developed;

- The range of velocity over which the disk fluctuations can cause tapping against the backstop which results in impact wear and fatigue of the disk-to-hinge arm connection. The zone of velocity over which the disk is free-swinging without impact has also been defined for various configurations of valves and upstream disturbances. In the free-swinging zone, only hinge pin wear is of concern.

In addition to these major conclusions which relate to the specific objectives of Phase I, the important results obtained are summarized below:

1. Stability of clearway swing check valves can be very sensitive to manufacturing tolerances and the physical alignment of piping to the valve. Clearway designs with closely coupled elbows may not attain a stable fully open disk position with any flow velocity.

2. Decreasing the fully open disk angle, ϕ, by 10 degrees in clearway designs greatly enhances disk stability at the fully open position. It also reduces the velocity required to fully open the valve;

3. Upstream reducers have little or no disturbing effect on disk stability or velocity required to fully open the valve. Reducers are used in the sense that it reduces the flow area as flow approaches the valve;

4. Maximum disk motion due to an upstream turbulence source can be much higher than for upstream elbows, depending upon the severity of the source. When calculating wear, turbulence sources out to 10d upstream must be considered;

5. The minimum velocity margins over V_{min} for elbows depend upon the disk opening angle, elbow orientation, and the proximity. Table 3 shows the minimum margins necessary.

6. The minimum velocity margin required to fully open the disk for non-clearway valves with an upstream turbulence source is 10 percent higher than baseline conditions, assuming the velocity field remains uniform across the pipe section. Higher velocity margins might be required in the case of turbulence combined with velocity skew effects.

RECOMMENDATIONS

To achieve the eventual goal of quantifying degradation trends when swing check valves are used in less than fully open conditions, which is the case with many valves in nuclear power plants, further research is necessary. Phase I has resulted in providing definite information regarding flow conditions and the upstream disturbances which can cause accelerated wear. It has also produced data regarding maximum disk fluctuation, disk tapping zone, and free-swinging zone, which provide the basis for developing quantitative predictions of degradation.

Research should be continued to use the disk stability information from Phase I to develop quantitative fatigue and wear life predictions. This can serve as a basis to make objective decisions regarding suitable maintenance intervals over which check valve degradation is within acceptable safe limits and which will not jeopardize its operation. Phase II research, aimed at developing such predictive techniques, verifying them in controlled laboratory conditions, and correlating them with actual plant data should be conducted. The eventual goal of this research effort is to improve the safety and reliability of check valves in nuclear power plant service properly accounting for the expected degradation trends.

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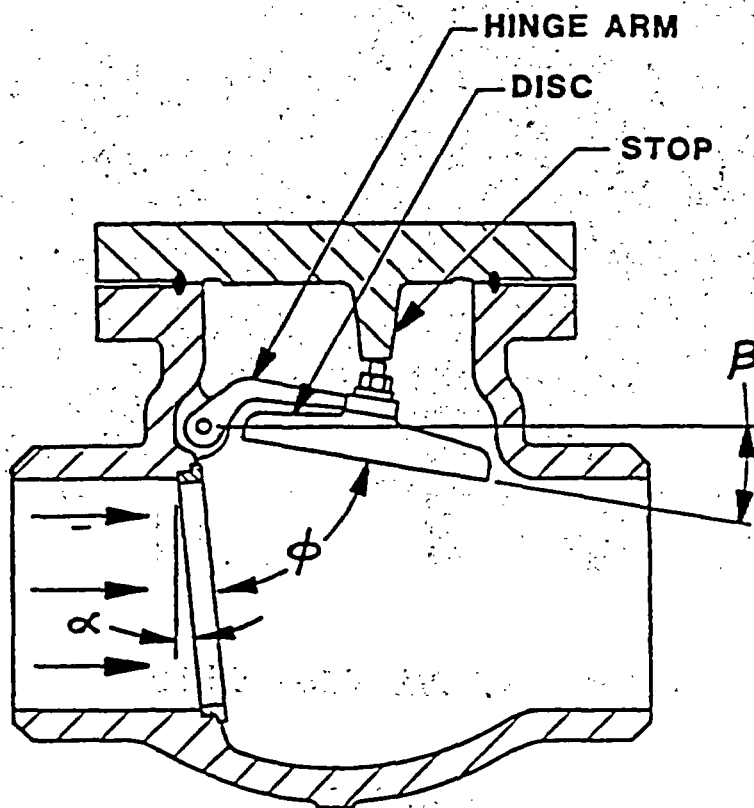
α = SEAT PLANE TILT WITH RESPECT TO VERTICAL
β = DISC ANGLE WITH RESPECT TO HORIZONTAL
θ = (α + β) = TOTAL IMPINGEMENT ANGLE
φ = DISC OPENING ANGLE

CHECK VALVE CONFIGURATION

FIGURE 1
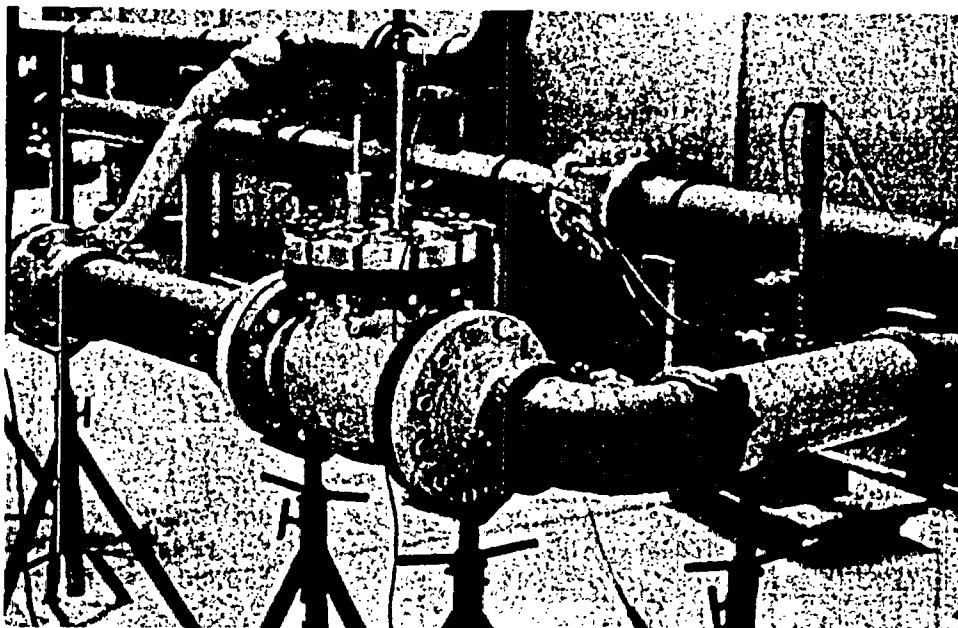
CHECK VALVE TEST SETUP

FIGURE 2

PUMP

ORIFICE METER

TEST CHECK VALVE

DISCHARGE HOSE

DOWNSTREAM THROTTLING VALVE

UPSTREAM LENGTH OF STRAIGHT PIPE

DOWNSTREAM LENGTH OF STRAIGHT PIPE

29

BYPASS RETURN FLOW

PRIMARY RETURN FLOW

ADJUSTABLE PIPING SECTION

BYPASS VALVE

STRAIGHTENING VANES

STRAIGHTENING VANES

UPSTREAM ELBOW (VERTICAL UP)
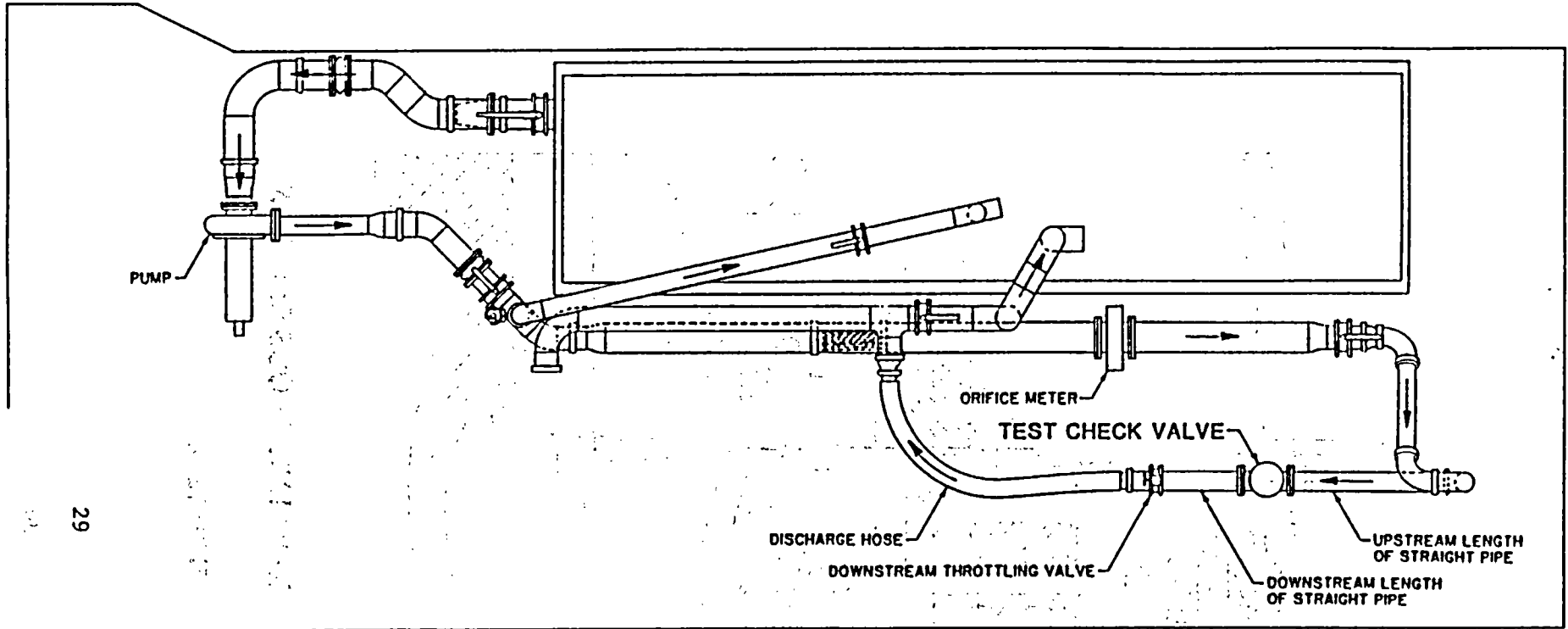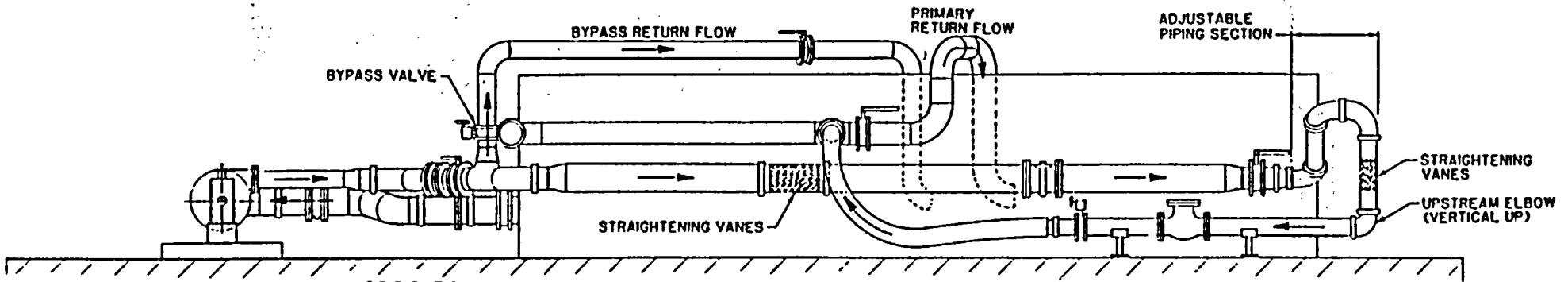
KALSI ENGINEERING, INC. FLOW TEST FACILITY

FIGURE 3

ADJUSTABLE BACKSTOP
WITH LOADCELL

LVDT FOR MEASURING
DISC MOTION
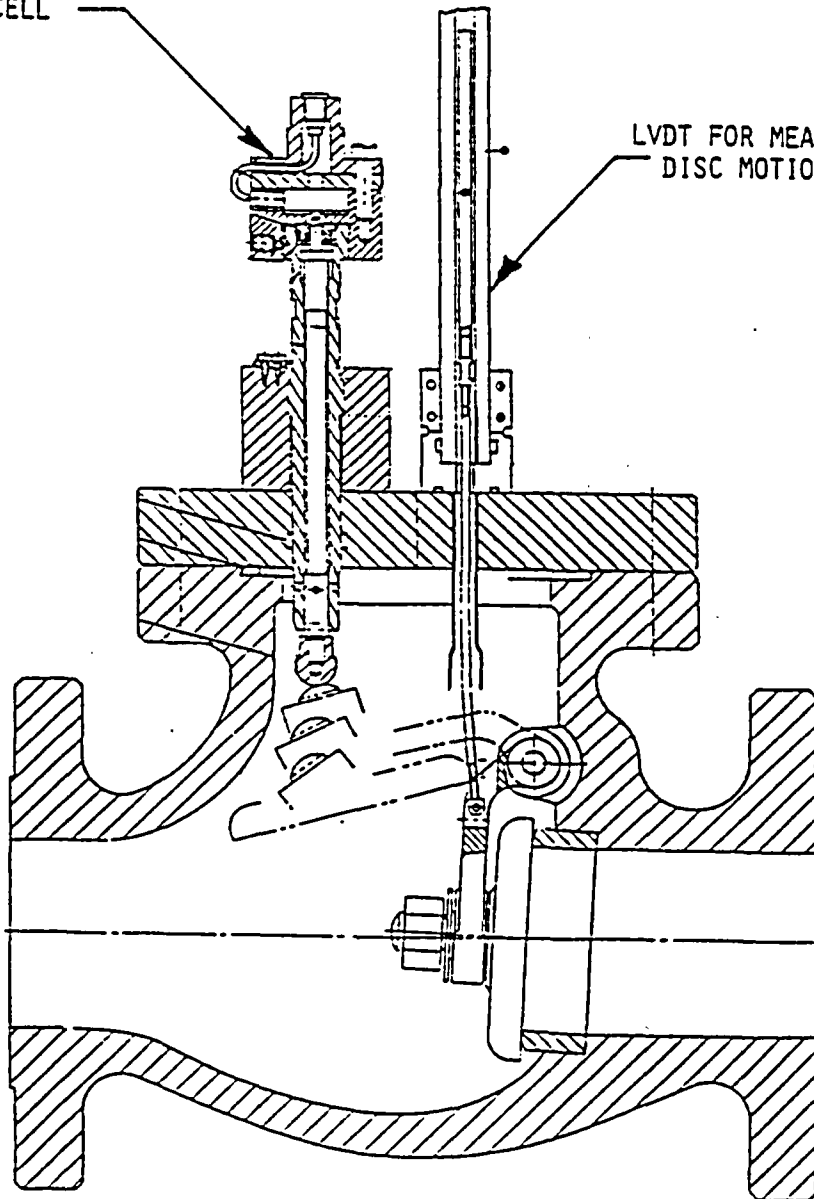
INSTRUMENTED VALVE CROSS SECTION

FIGURE 4

DISK FLUCTUATION vs FLOW VELOCITY

Baseline, 20d upstream (6 Inch)
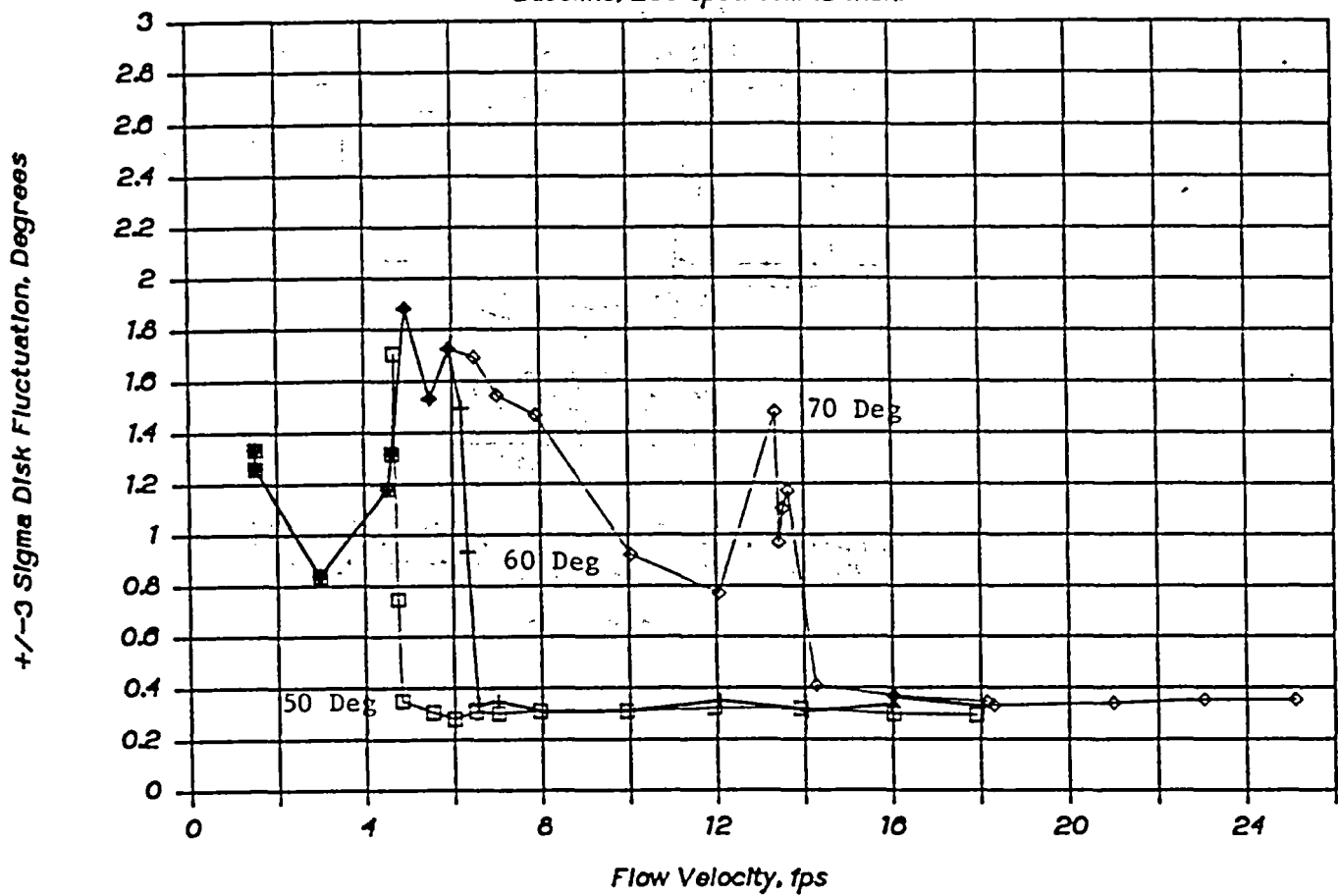
COMPOSITE PLOT OF DISK FLUCTUATIONS FOR 6" VALVE, BASELINE TESTS

FIGURE 5
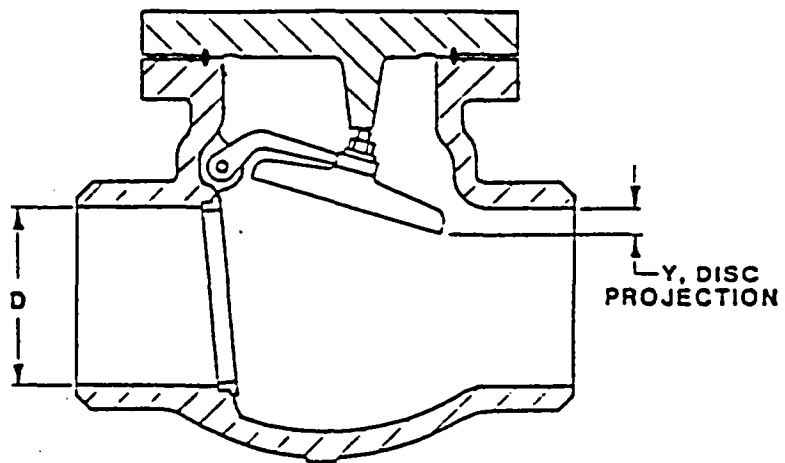
D, IN.	Y, IN.	Y/D
3	0	0
6	0.8	.133

DISK PROJECTION COMPARISON
CLEARWAY SWING CHECK TEST VALVE

FIGURE 6

DISK FLUCTUATION vs FLOW VELOCITY

Baseline (20d pipe upstream)(3 Inch)
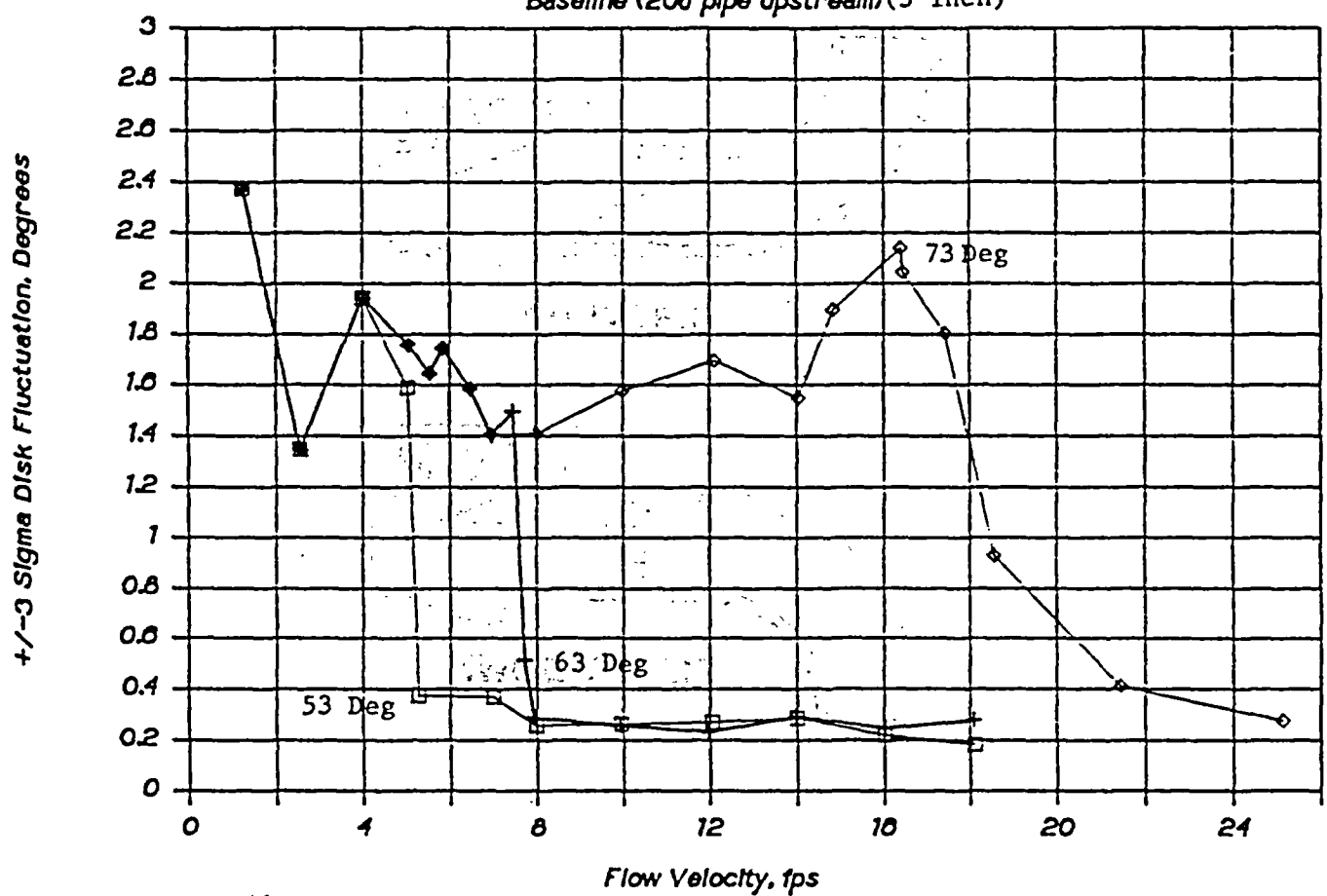
COMPOSITE PLOT OF DISK FLUCTUATIONS FOR 3" VALVE, BASELINE TESTS

FIGURE 7
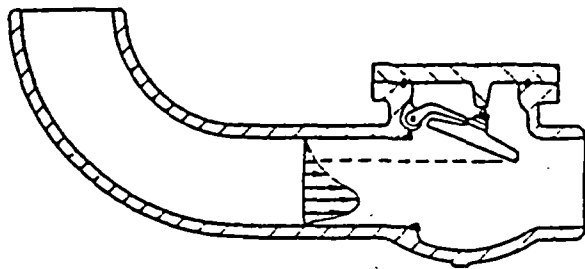
ELBOW UP
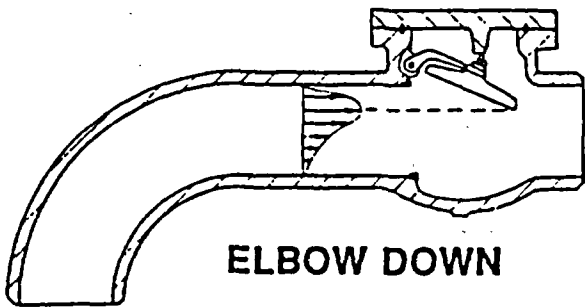
ELBOW DOWN

DEFINITION OF ELBOW ORIENTATION

FIGURE 8

DISK FLUCTUATION vs FLOW VELOCITY
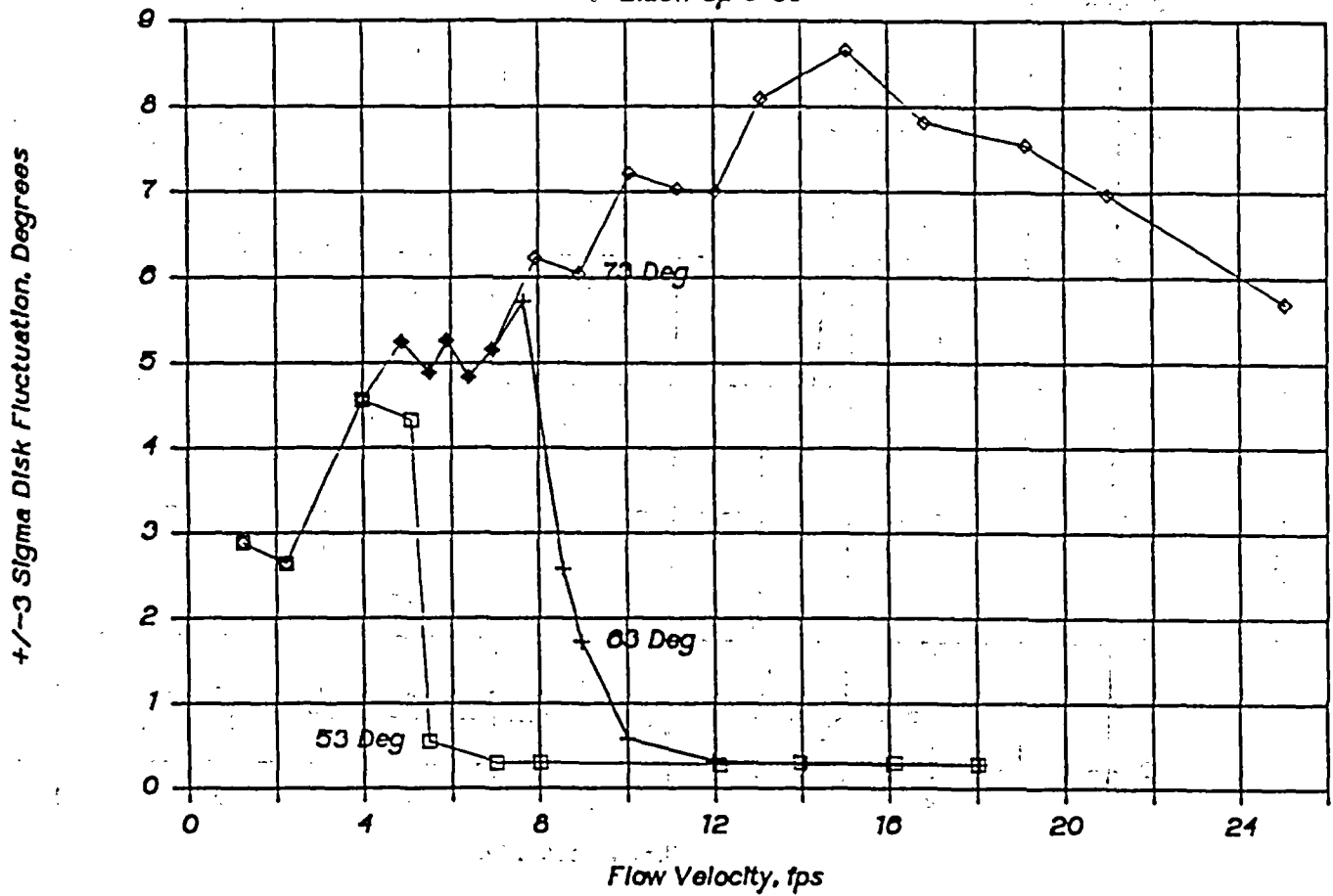
Elbow Up @ 0d

COMPOSITE PLOT OF DISK FLUCTUATIONS FOR
3" VALVE AT VARIOUS DISK STOP ANGLES

FIGURE 9A

DISK FLUCTUATION vs FLOW VELOCITY
Elbow Up @ 0d (6 Inch)
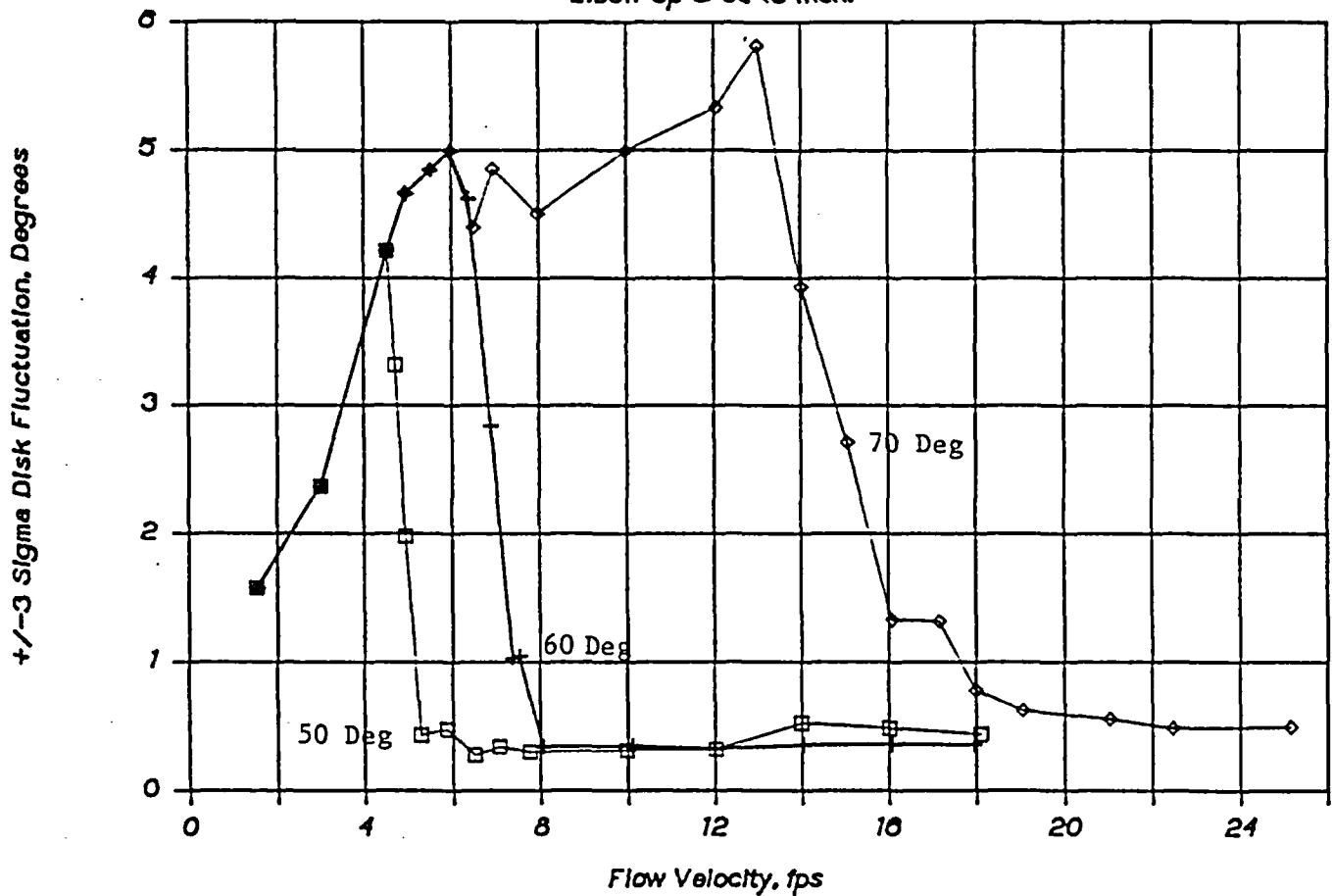
COMPOSITE PLOT OF DISK FLUCTUATIONS FOR 6" VALVE AT VARIOUS DISK STOP ANGLES

FIGURE 9B

DISK FLUCTUATION vs FLOW VELOCITY
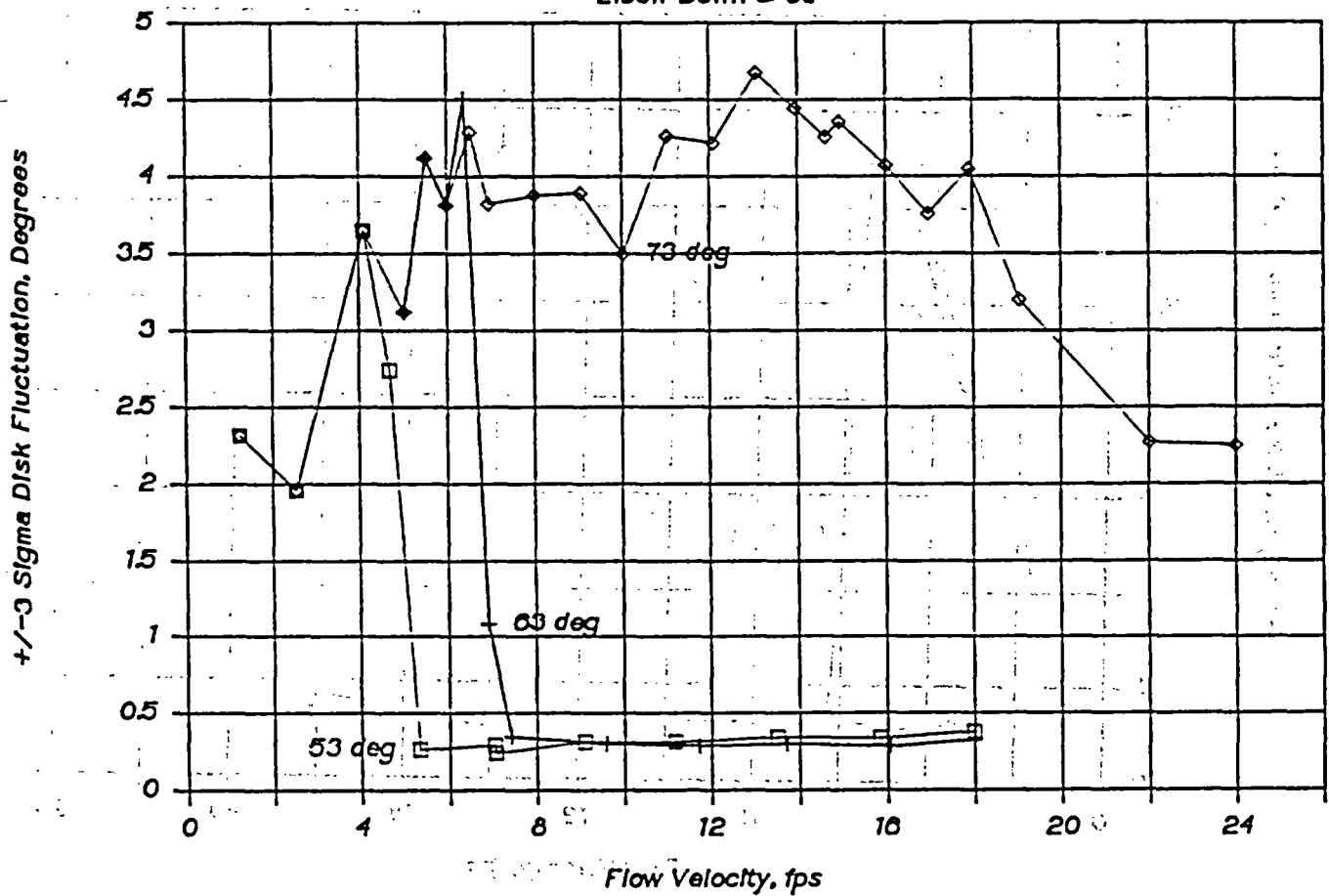
Elbow Down @ 0d

COMPOSITE PLOT OF DISK FLUCTUATION FOR
3" VALVE AT VARIOUS DISK STOP ANGLES

FIGURE 10A

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DISK FLUCTUATION vs FLOW VELOCITY

Elbow Down @ 0d (6 Inch)
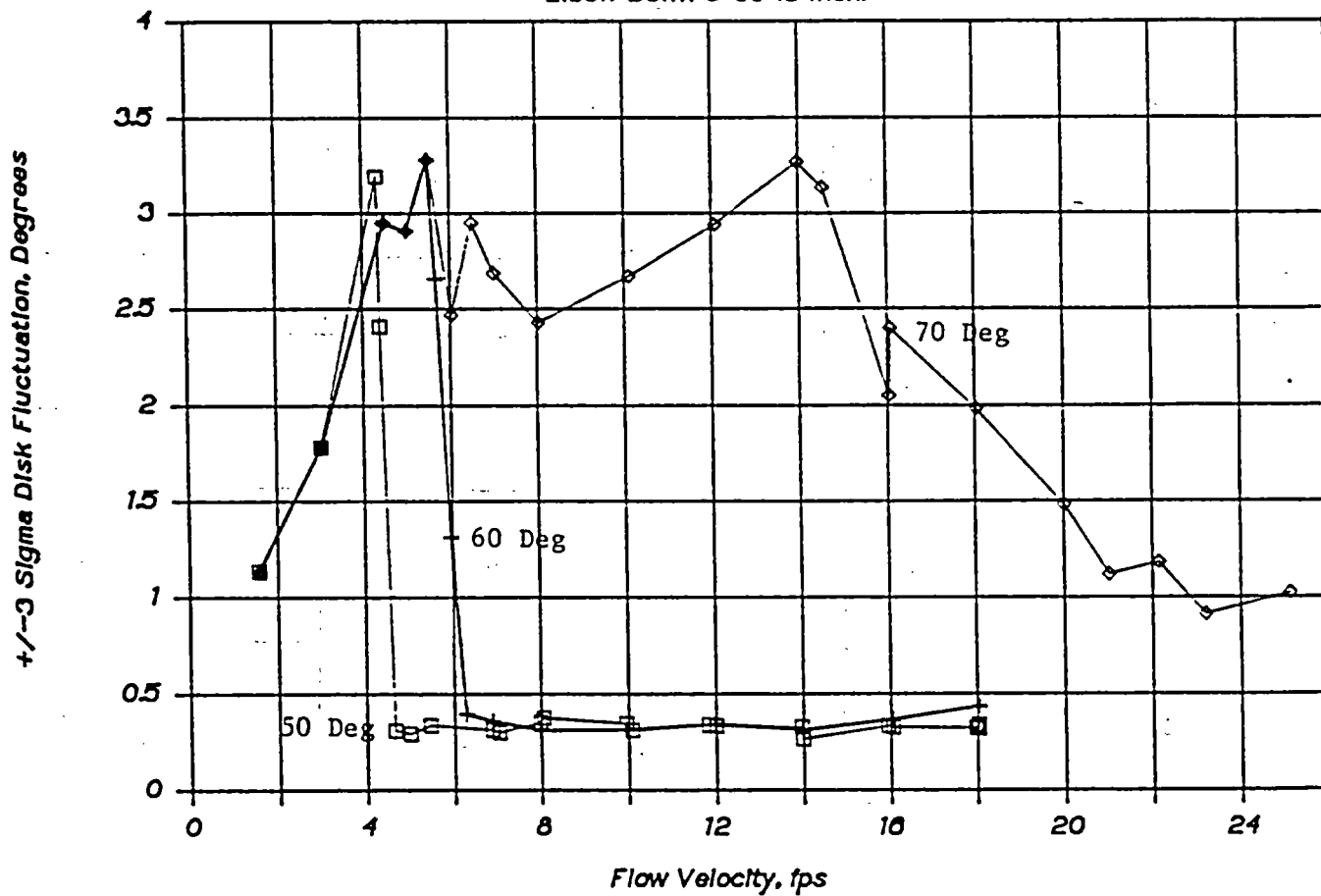
COMPOSITE PLOT OF DISK FLUCTUATIONS FOR 6" VALVE AT VARIOUS DISK STOP ANGLES

FIGURE 10B
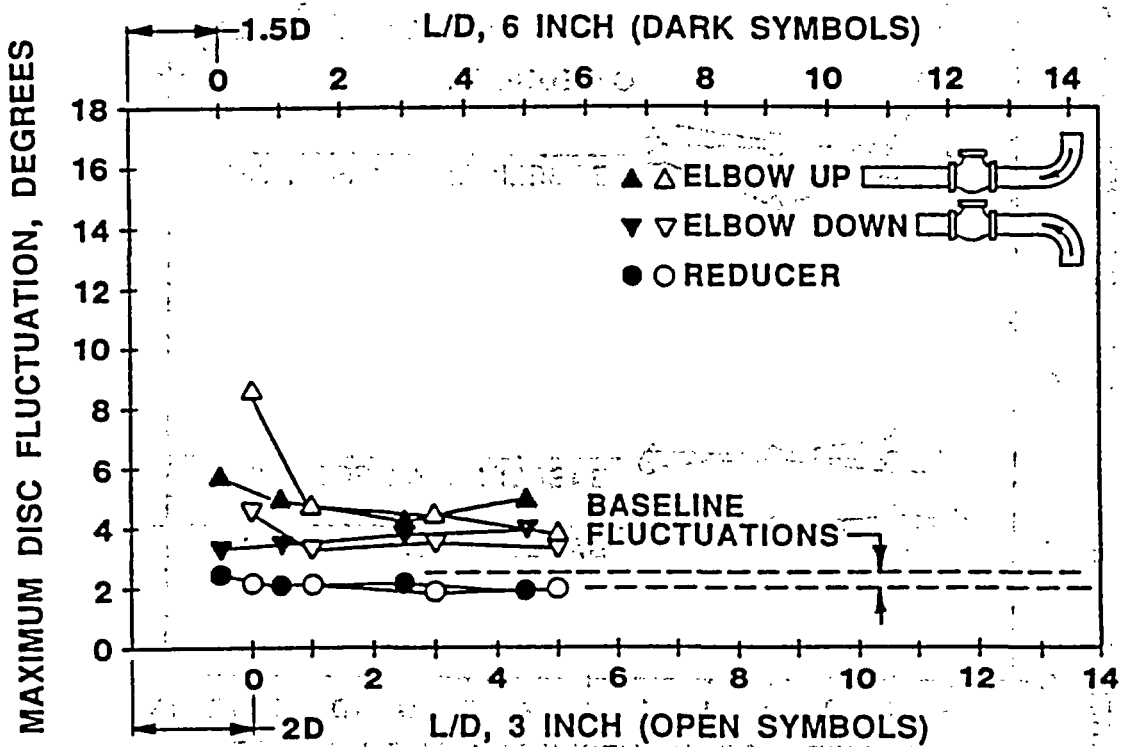
MAXIMUM DISK FLUCTUATION VS. DISTANCE TO
UPSTREAM DISTURBANCE

FIGURE 11

ELBOW AND REDUCER TEST RESULTS
3 INCH SWING CHECK VALVE.

FIGURE 12

40

ELBOW AND REDUCER TEST RESULTS
6 INCH SWING CHECK VALVE

FIGURE 13

41
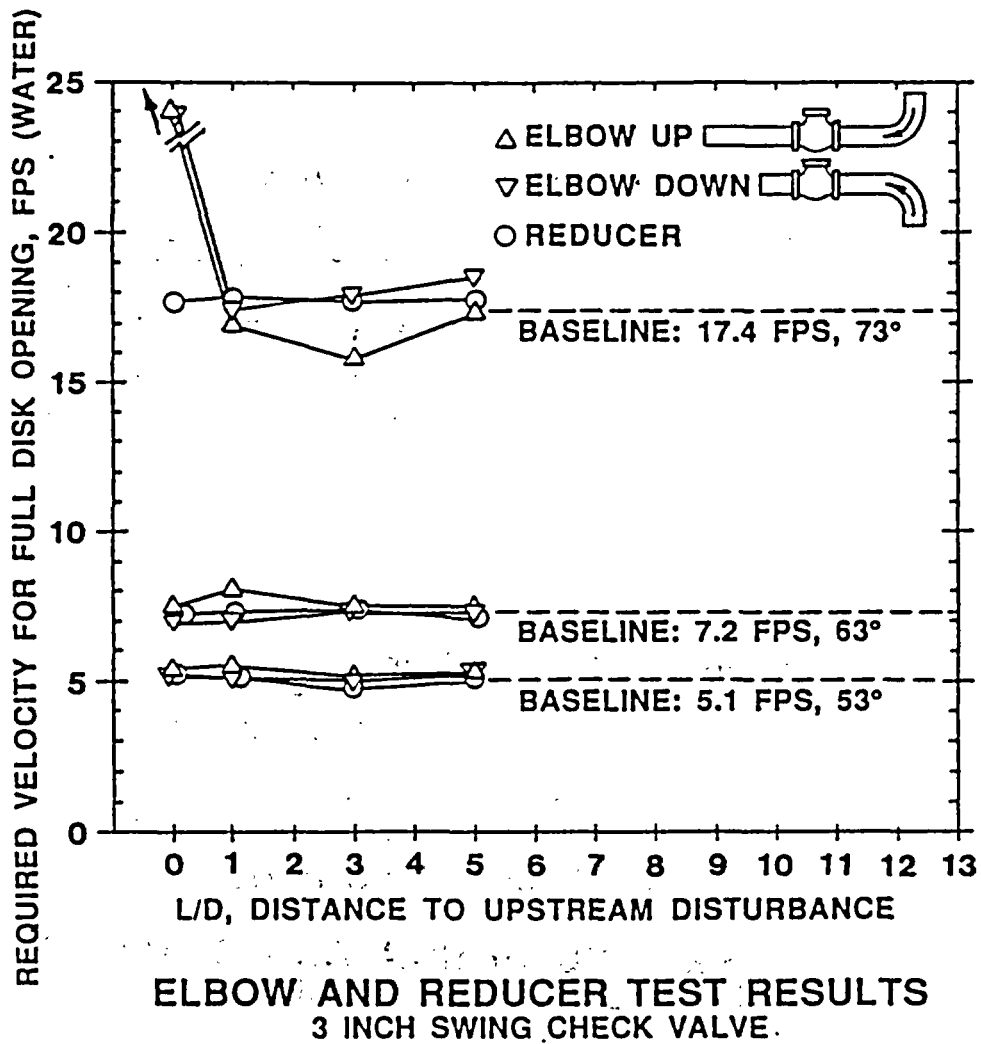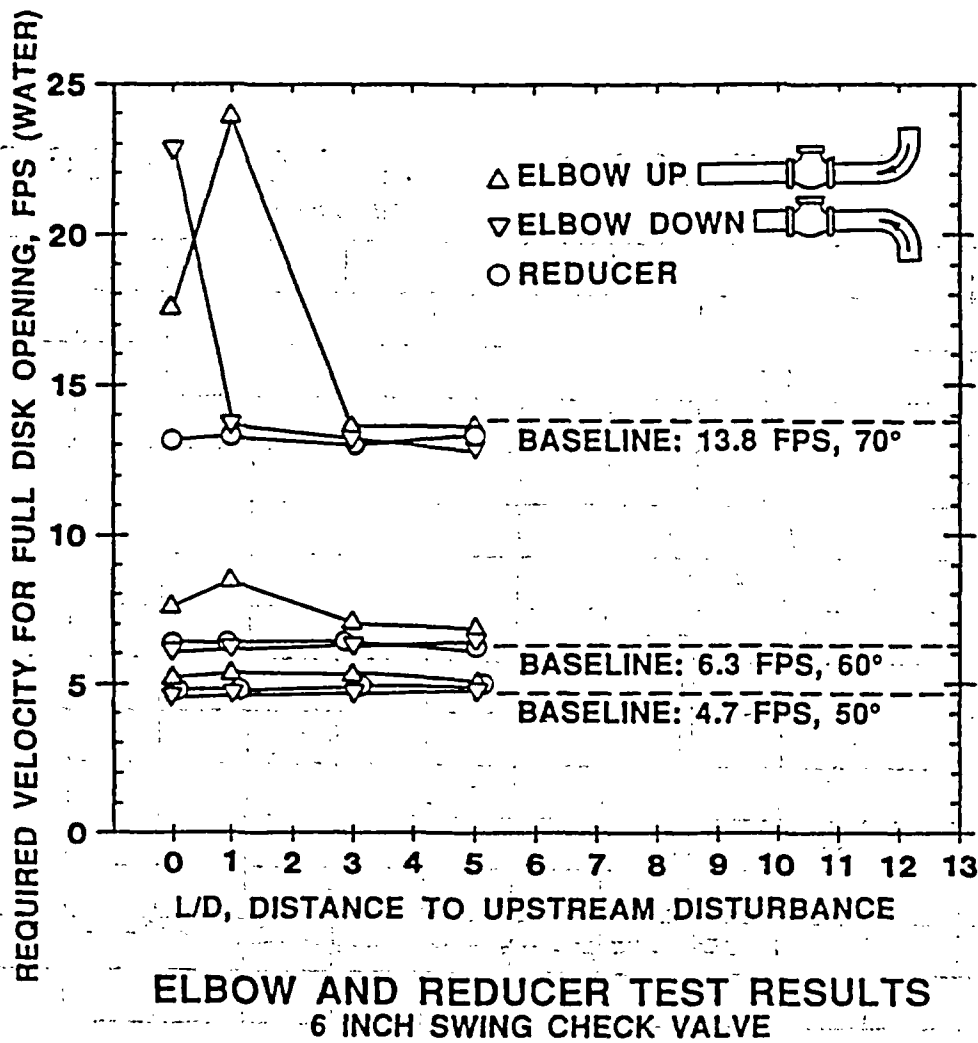
DISK FLUCTUATION vs FLOW VELOCITY

8x6 Reducer @ 0d (6 Inch)
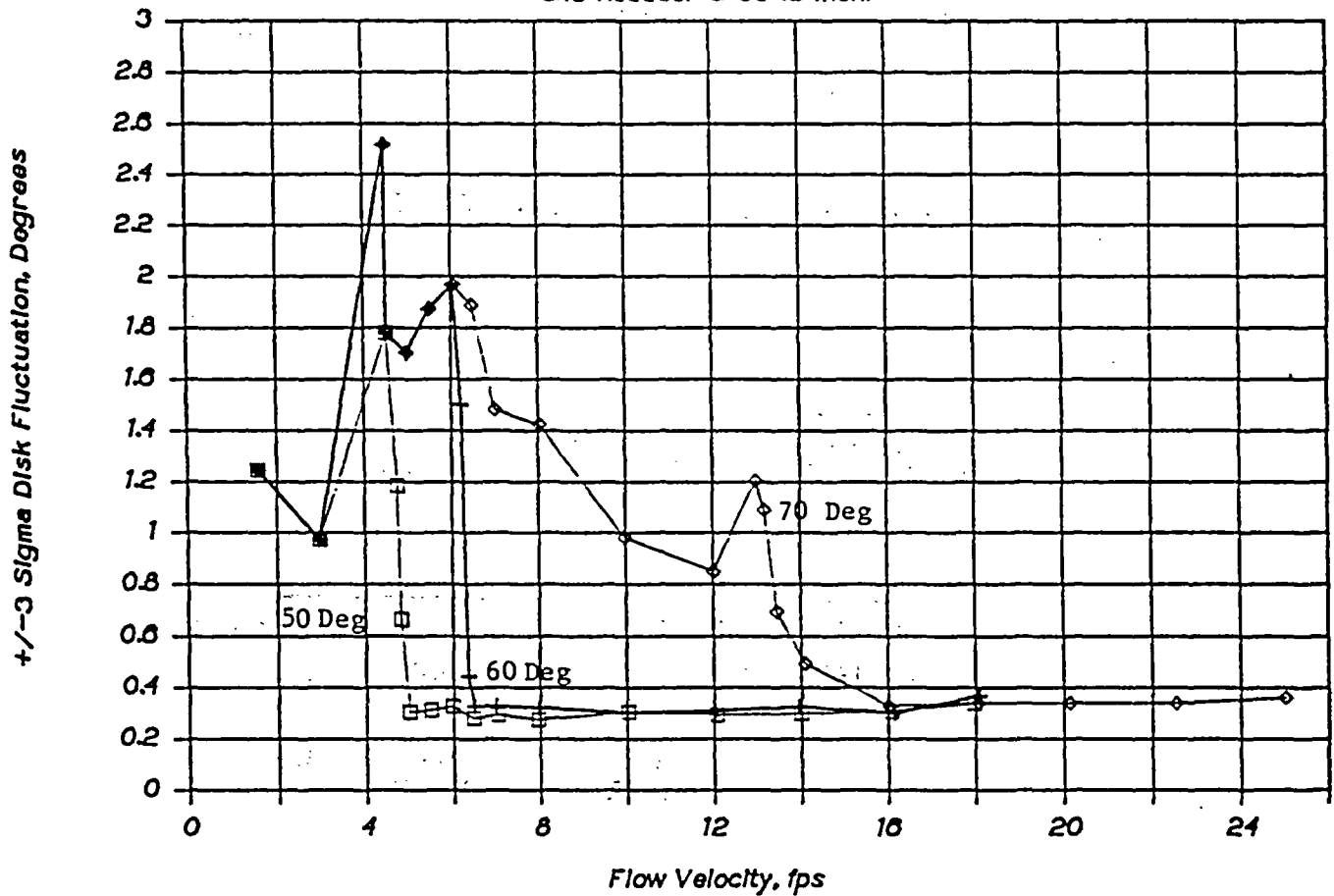
**COMPOSITE PLOT OF DISK FLUCTUATIONS FOR
6" VALVE WITH 8" x 6" REDUCER AT 0D**
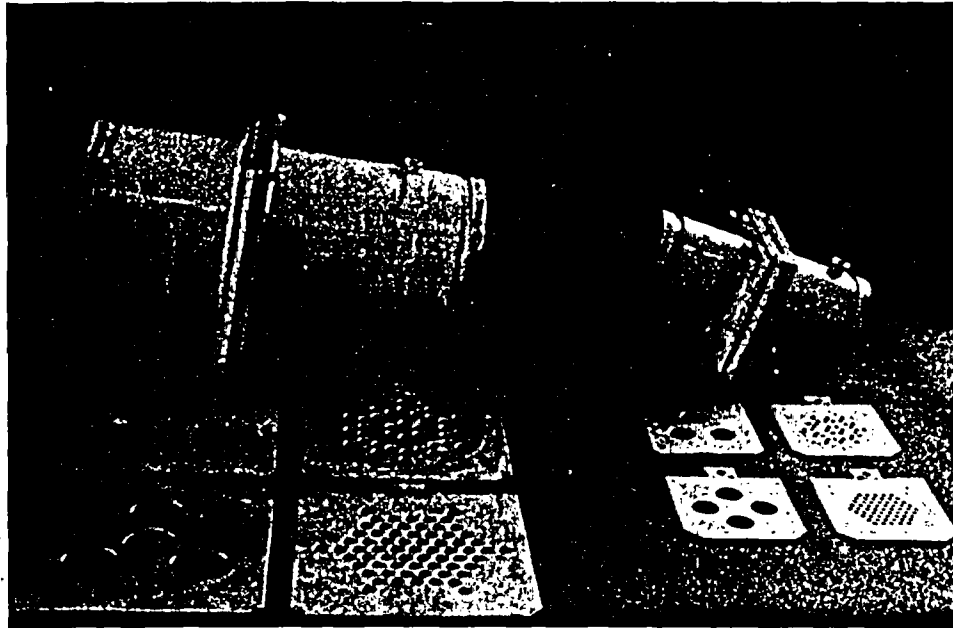
FIGURE 14

**3 & 6 INCH PERFORATED
PLATE HOLDERS**

FIGURE 15
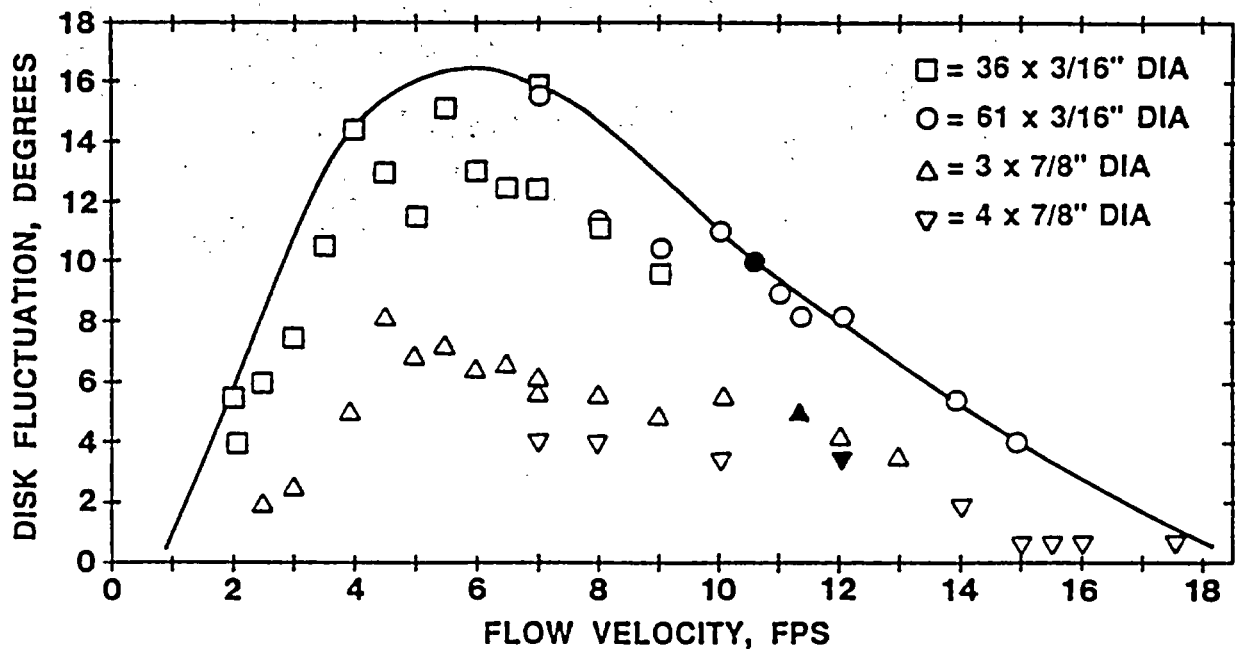
DISK MOTION OF 3 INCH VALVE
DUE TO TURBULENCE

PERFORATED PLATES AT 1.5 DIAMETERS UPSTREAM
STOP ANGLE AT 73 DEGREES

FIGURE 16
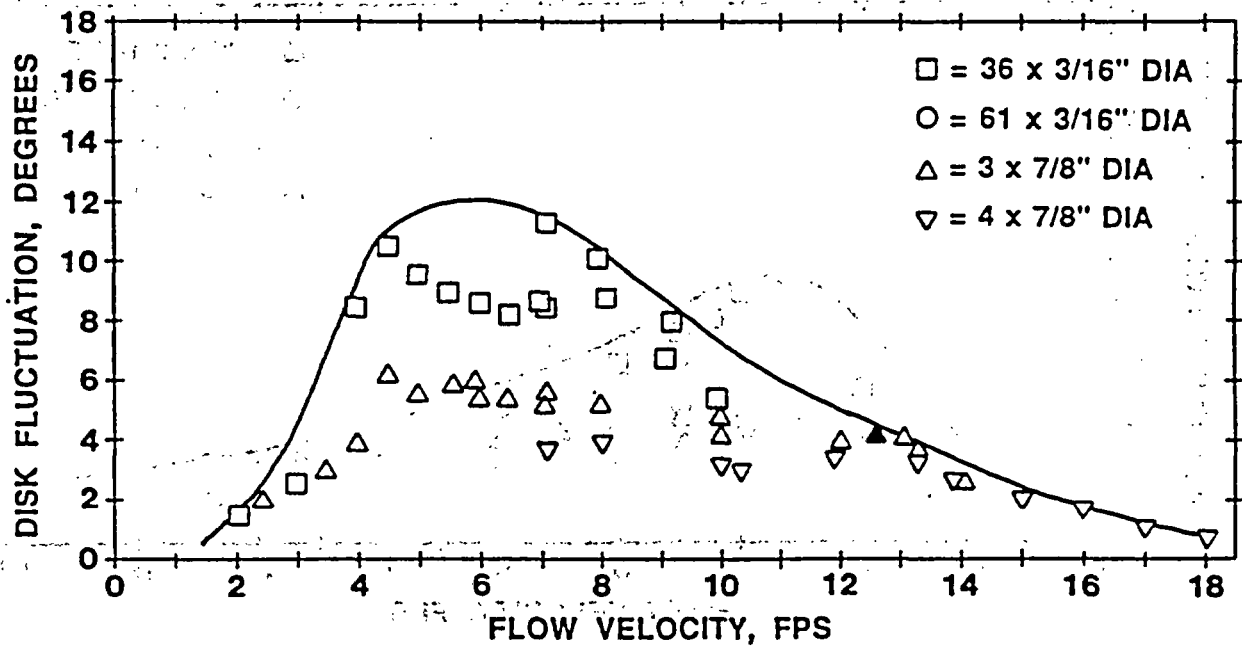
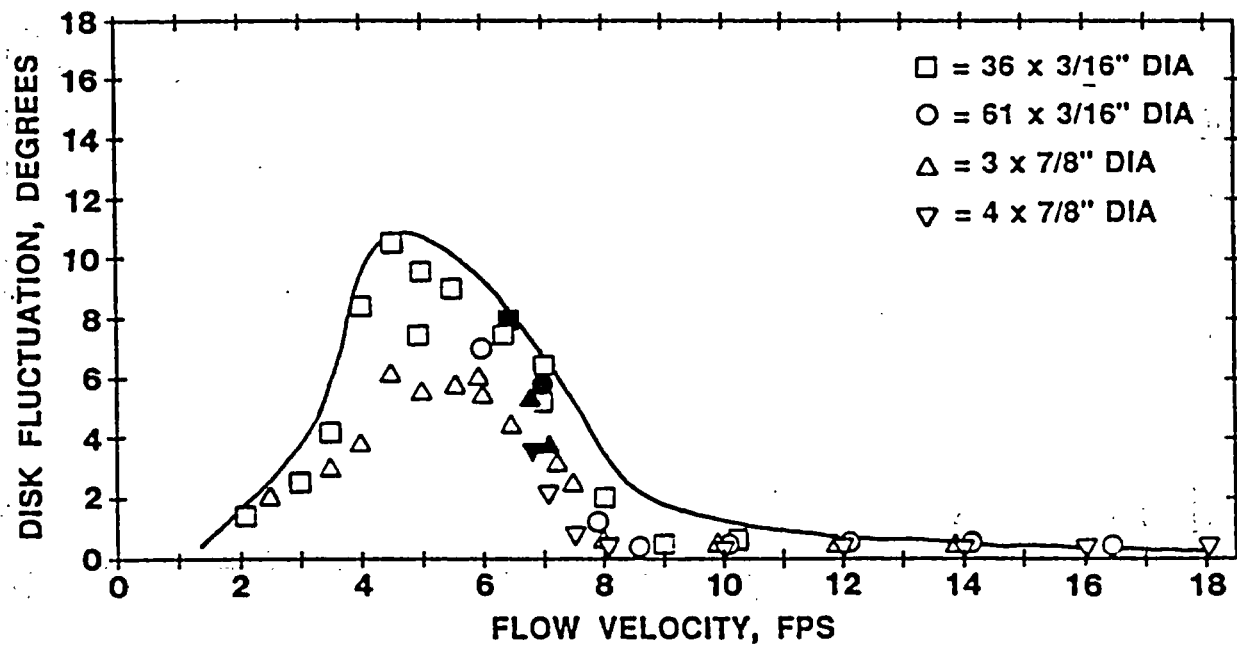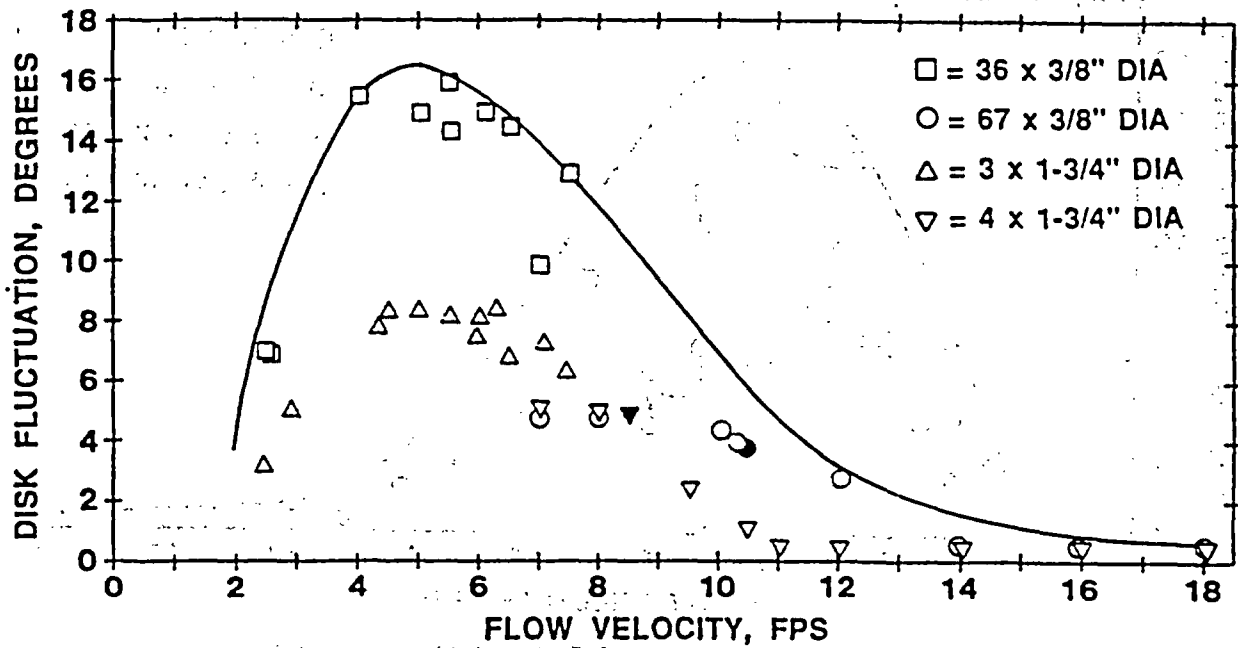**DISK MOTION OF 3 INCH VALVE**
DUE TO TURBULENCE
PERFORATED PLATES AT 2.5 DIAMETERS UPSTREAM
STOP ANGLE AT 73 DEGREES

FIGURE 17
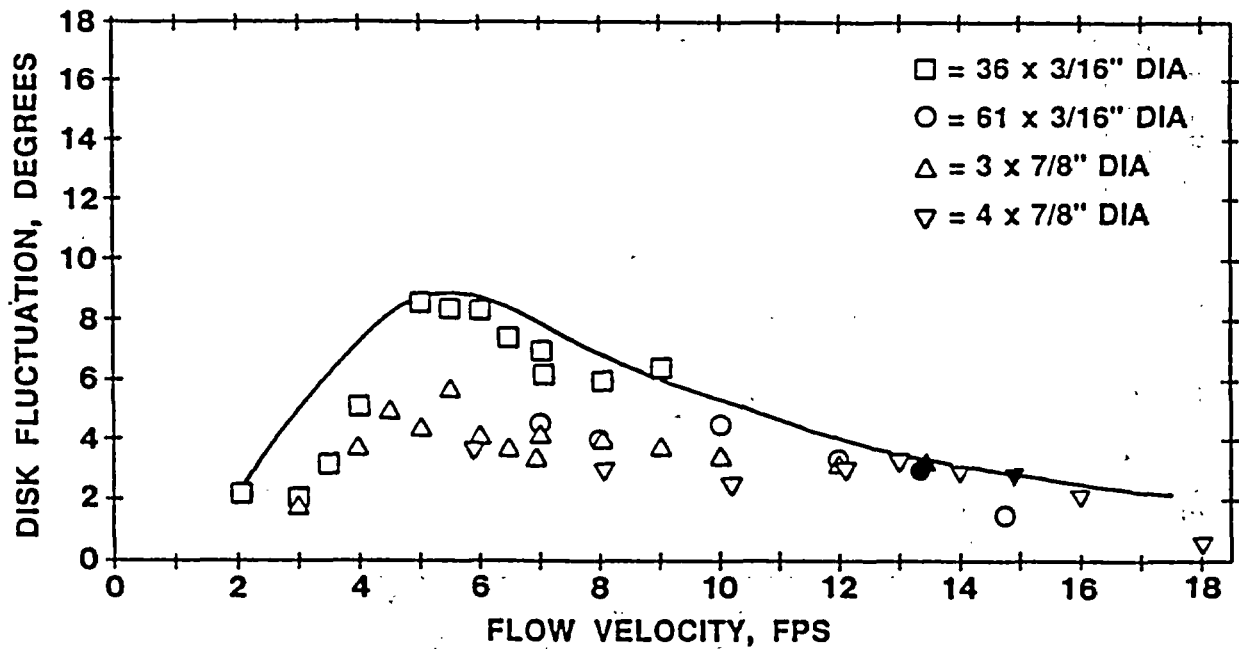
Chart legend:
- □ = 36 x 3/16" DIA
- O = 61 x 3/16" DIA
- △ = 3 x 7/8" DIA
- ▽ = 4 x 7/8" DIA

Y-axis: DISK FLUCTUATION, DEGREES (0 to 18)
X-axis: FLOW VELOCITY, FPS (0 to 18)

DISK MOTION OF 3 INCH VALVE
DUE TO TURBULENCE
PERFORATED PLATES AT 4.5 DIAMETERS UPSTREAM
STOP ANGLE AT 73 DEGREES
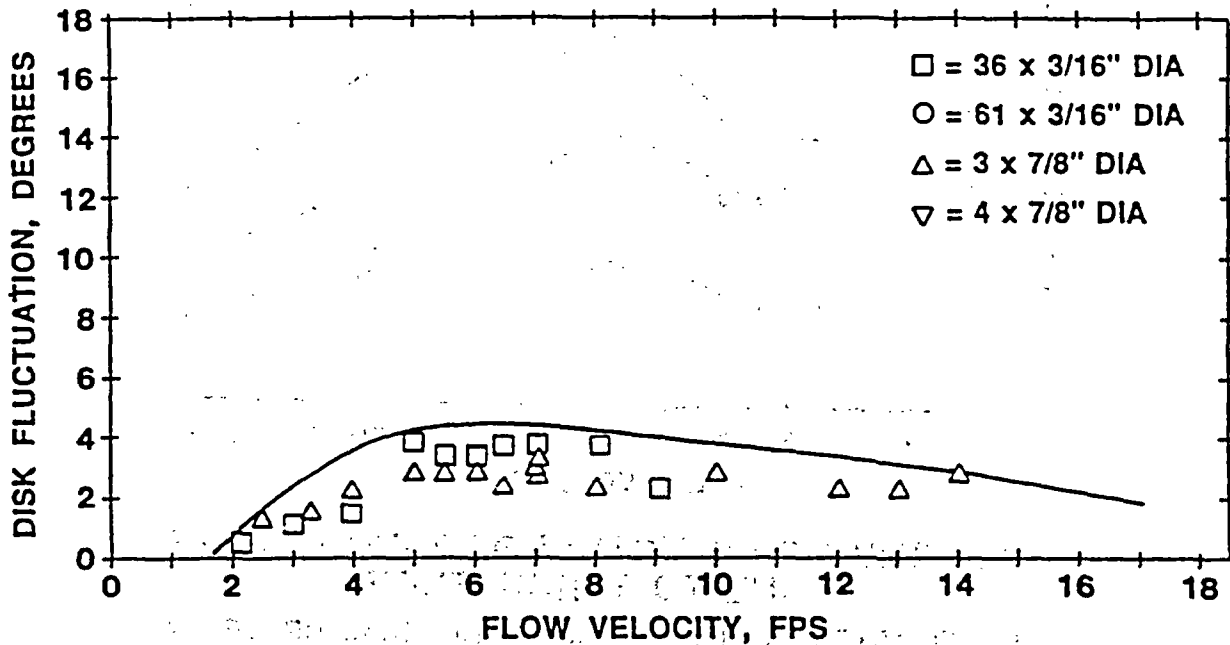
FIGURE 18

DISK MOTION OF 3 INCH VALVE
DUE TO TURBULENCE
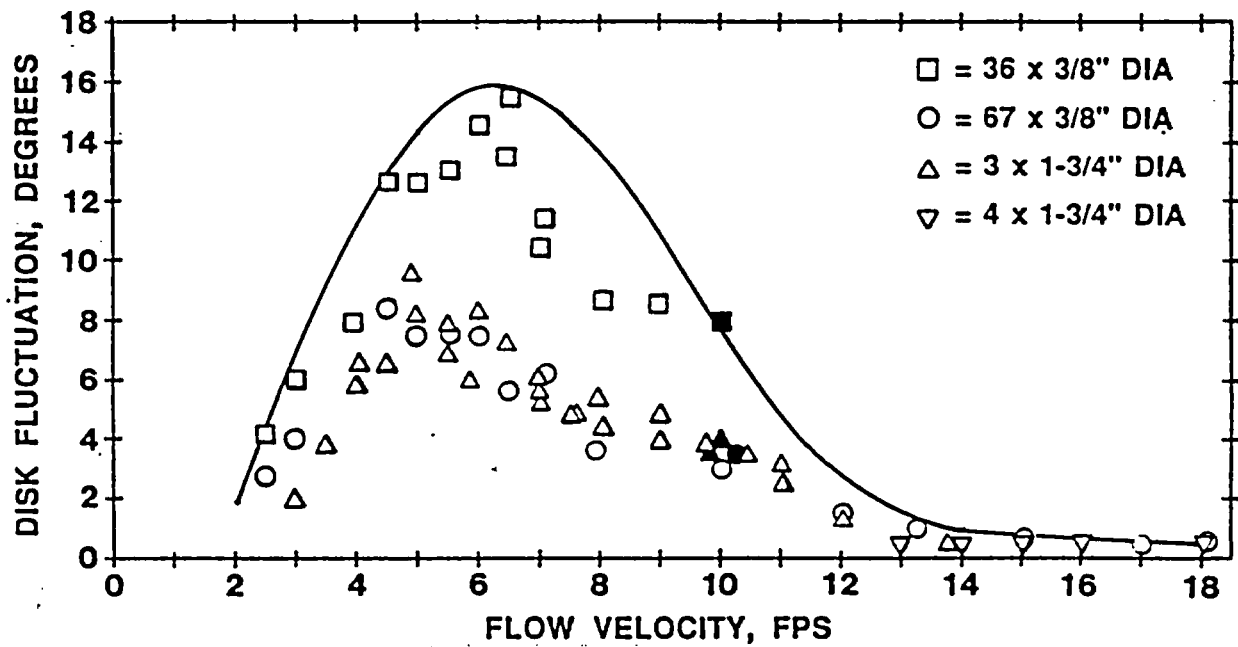PERFORATED PLATES AT 10.5 DIAMETERS UPSTREAM
STOP ANGLE AT 73 DEGREES

FIGURE 19
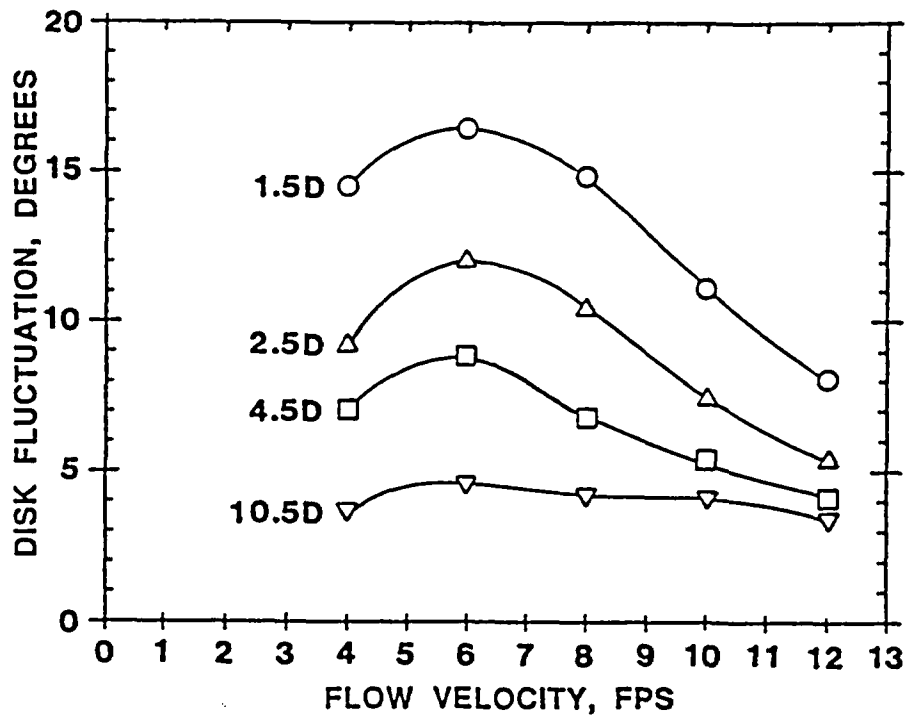
MAXIMUM DISK MOTION OF 3 INCH VALVE
DUE TO TURBULENCE

PERFORATED PLATES AT 1.5 TO 10.5 DIAM.UPSTREAM
STOP ANGLE AT 73 DEGREES

FIGURE 20
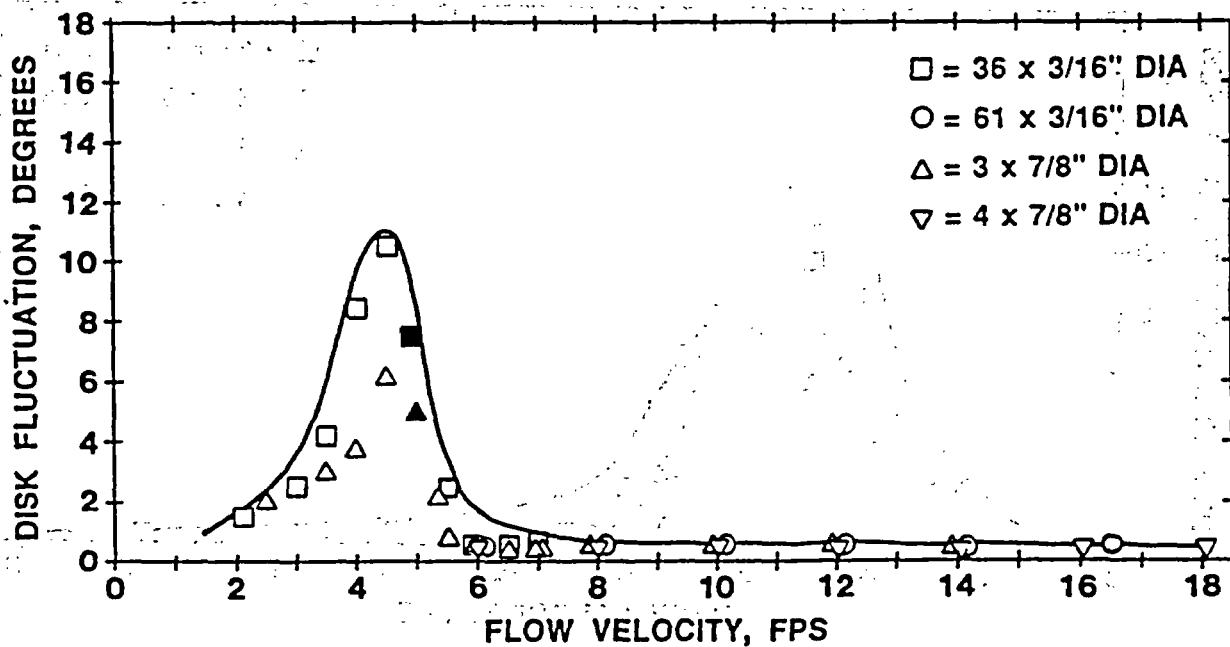
DISK MOTION OF 3 INCH VALVE
DUE TO TURBULENCE

PERFORATED PLATES AT 2.5 DIAMETERS UPSTREAM
STOP ANGLE AT 53 DEGREES

FIGURE 21

DISK MOTION OF 3 INCH VALVE
DUE TO TURBULENCE

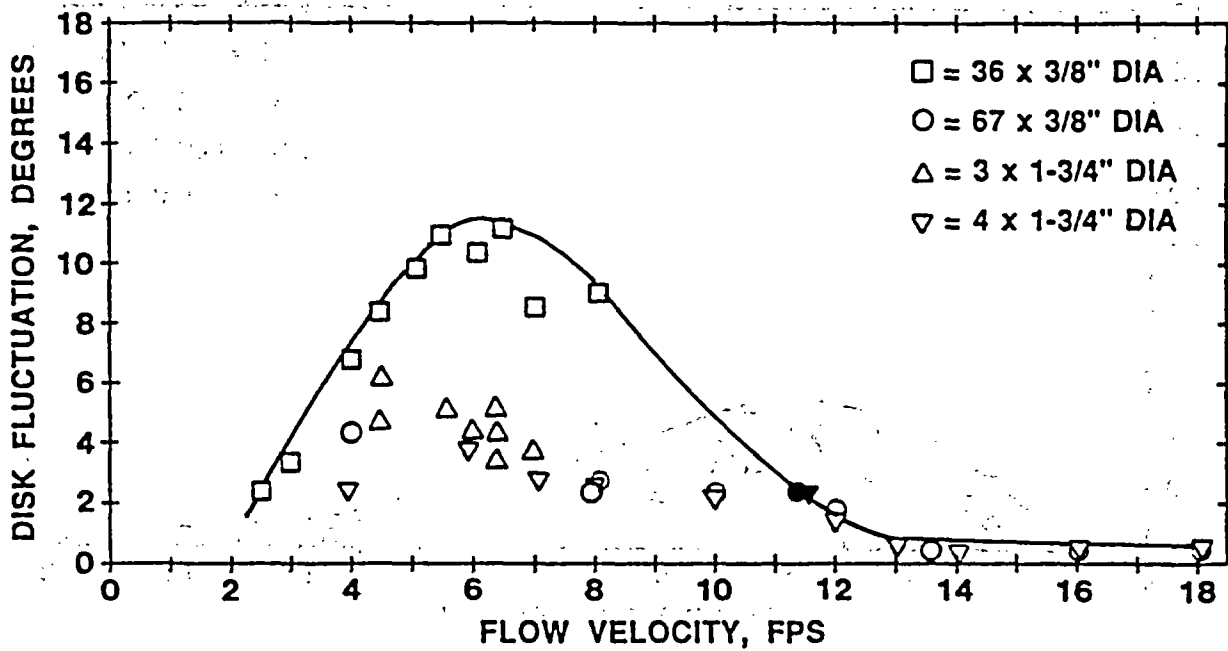PERFORATED PLATES AT 2.5 DIAMETERS UPSTREAM
STOP ANGLE AT 63 DEGREES

FIGURE 22

DISK MOTION OF 6 INCH VALVE
DUE TO TURBULENCE
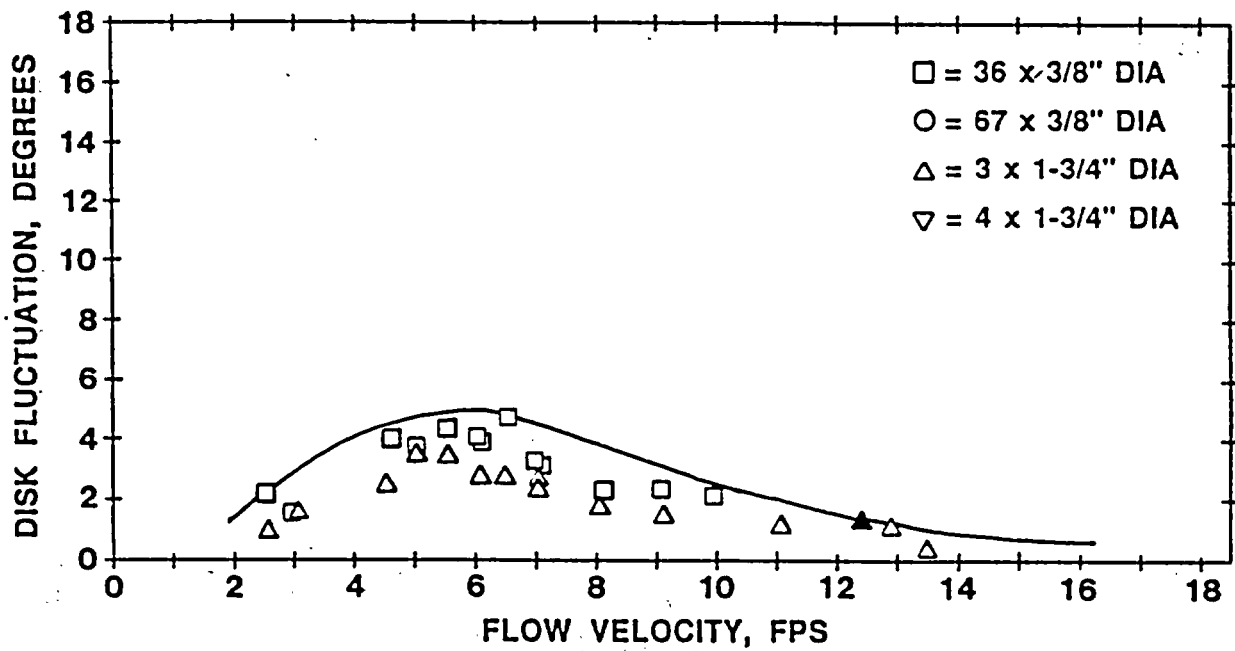PERFORATED PLATES AT 1.5 DIAMETERS UPSTREAM
STOP ANGLE AT 70 DEGREES

FIGURE 23

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DISK MOTION OF 6 INCH VALVE
DUE TO TURBULENCE
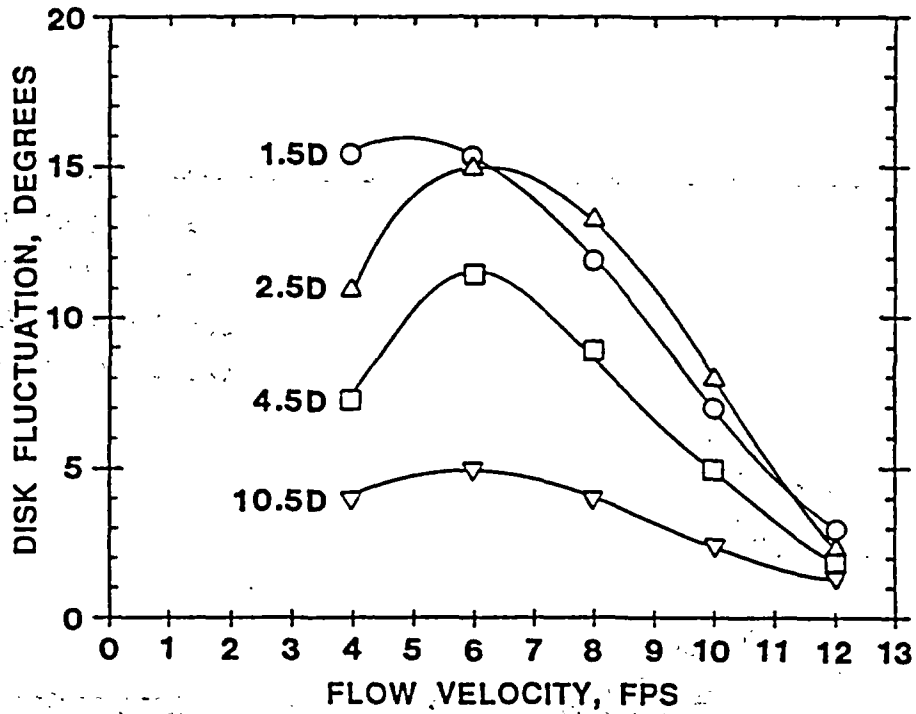PERFORATED PLATES AT 2.5 DIAMETERS UPSTREAM
STOP ANGLE AT 70 DEGREES

FIGURE 24

DISK MOTION OF 6 INCH VALVE
DUE TO TURBULENCE
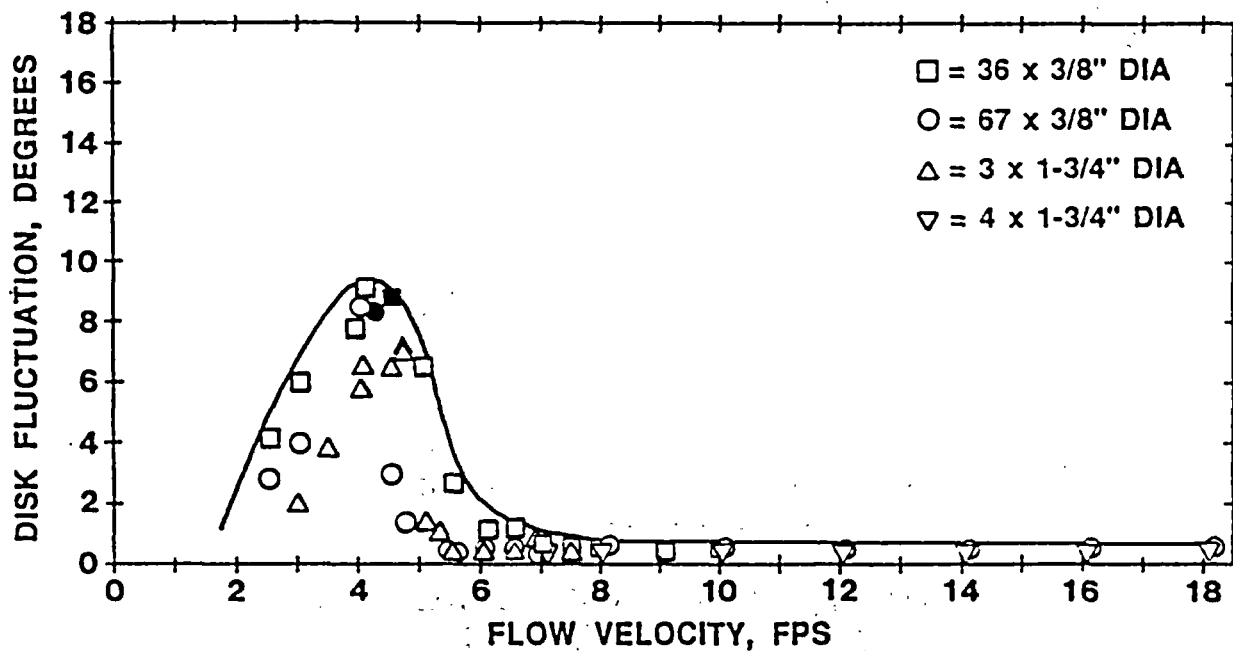PERFORATED PLATES AT 4.5 DIAMETERS UPSTREAM
STOP ANGLE AT 70 DEGREES

FIGURE 25

DISK MOTION OF 6 INCH VALVE
DUE TO TURBULENCE

PERFORATED PLATES AT 10.5 DIAMETERS UPSTREAM
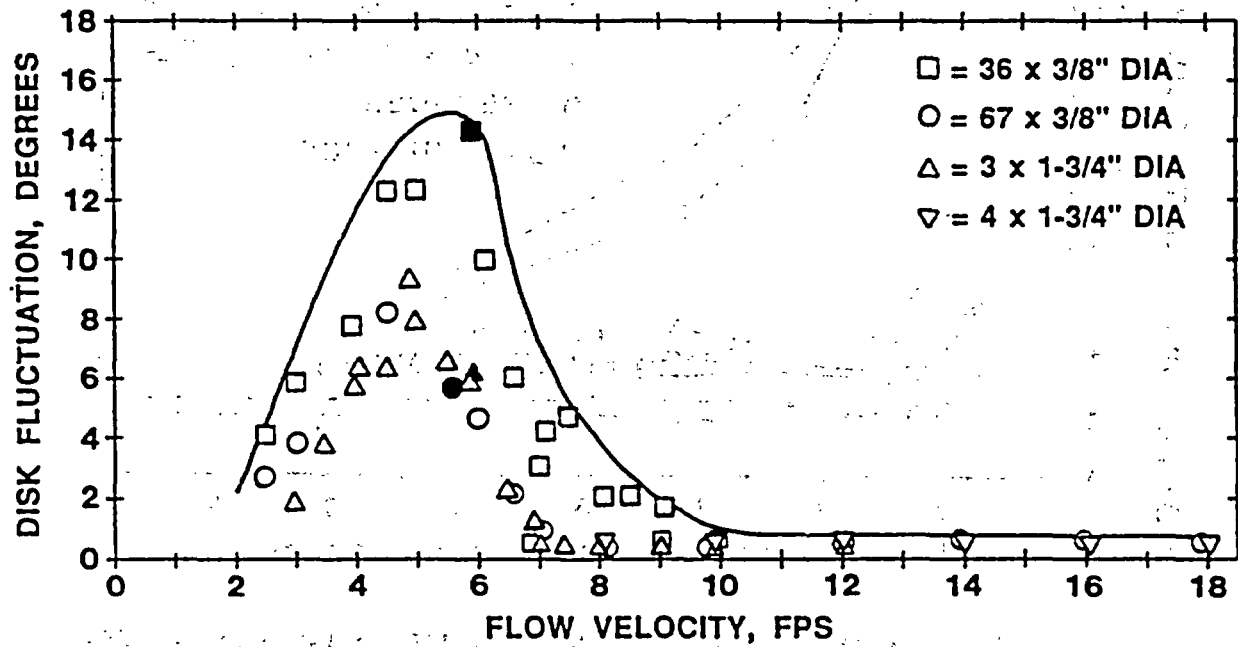STOP ANGLE AT 70 DEGREES

FIGURE 26

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MAXIMUM DISK MOTION OF 6 INCH VALVE
DUE TO TURBULENCE

PERFORATED PLATES AT 1.5 TO 10.5 DIAM. UPSTREAM
STOP ANGLE AT 70 DEGREES

FIGURE 27
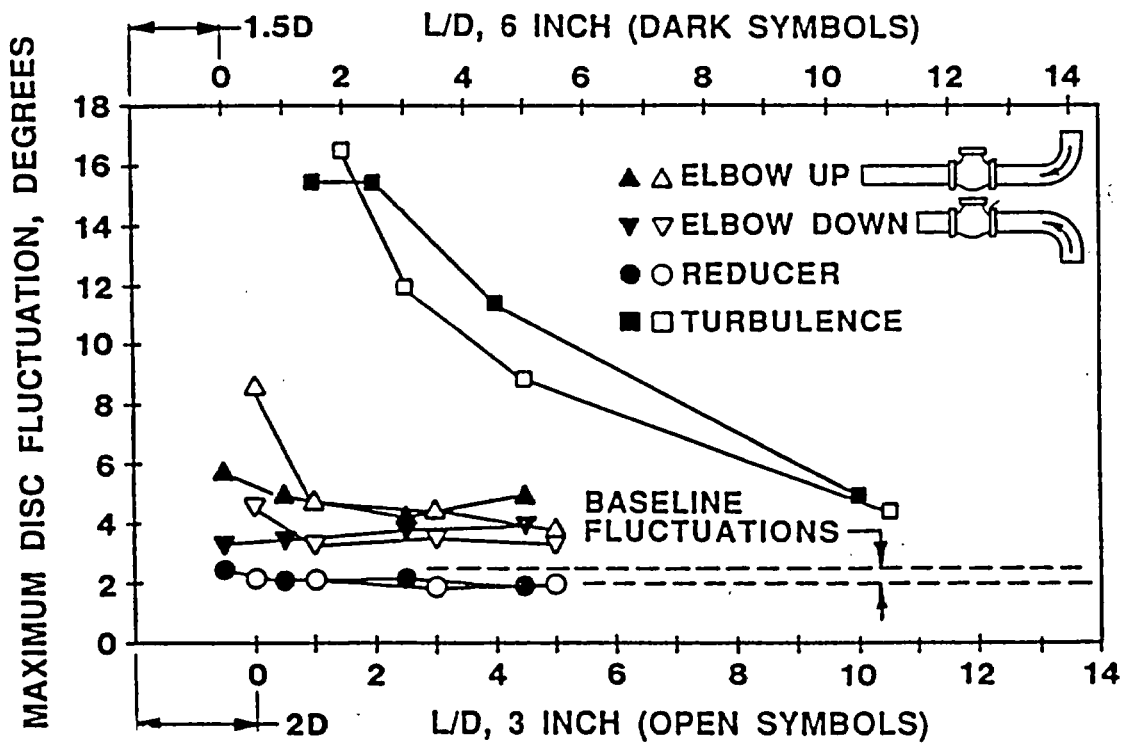
DISK MOTION OF 6 INCH VALVE
DUE TO TURBULENCE

PERFORATED PLATES AT 2.5 DIAMETERS UPSTREAM
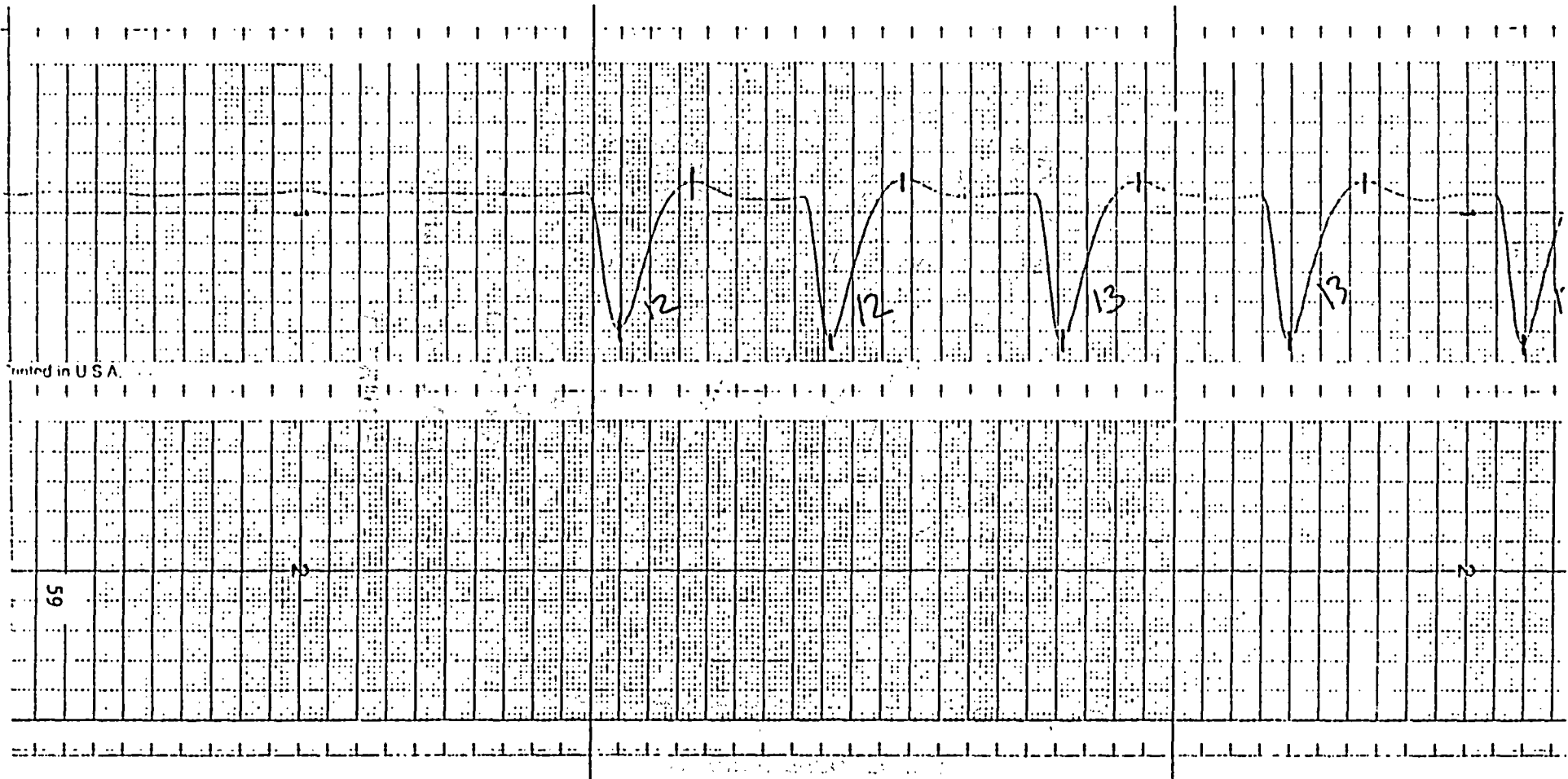STOP ANGLE AT 50 DEGREES

FIGURE 28

DISK MOTION OF 6 INCH VALVE
DUE TO TURBULENCE

PERFORATED PLATES AT 2.5 DIAMETERS UPSTREAM
STOP ANGLE AT 60 DEGREES
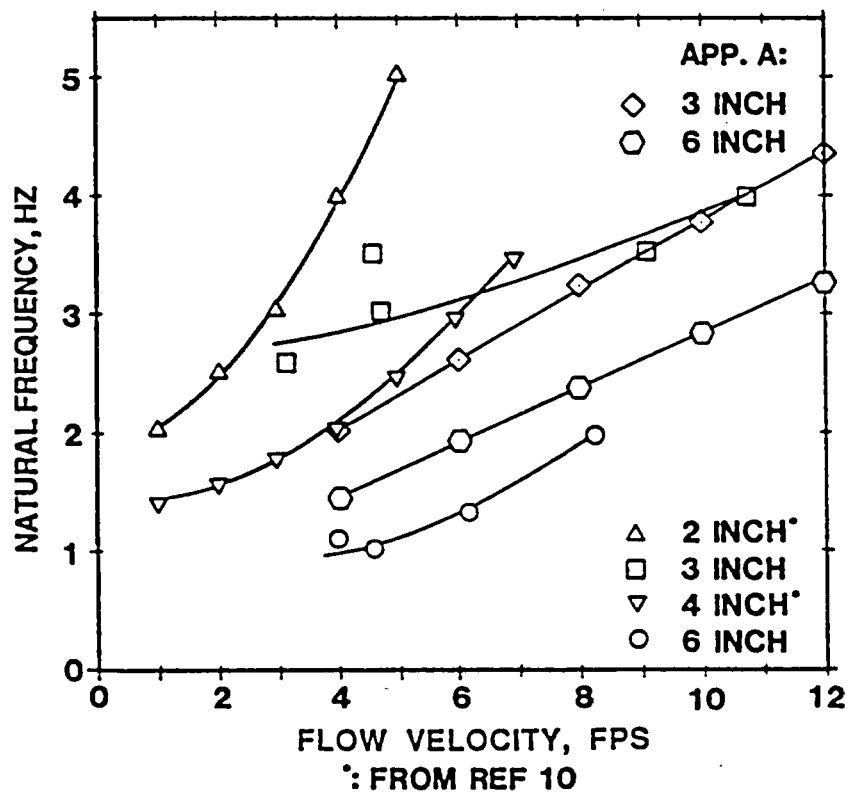
FIGURE 29

MAXIMUM DISK FLUCTUATION VS. DISTANCE TO UPSTREAM DISTURBANCE

FIGURE 30

58

DISK RESPONSE TO DISTURBANCE, 6" VALVE
NATURAL FREQUENCY TESTS

FIGURE 31

APP. A:

◇ 3 INCH
⬡ 6 INCH

NATURAL FREQUENCY, HZ

5

4

3

2

1

0

△ 2 INCH*
□ 3 INCH
▽ 4 INCH*
○ 6 INCH

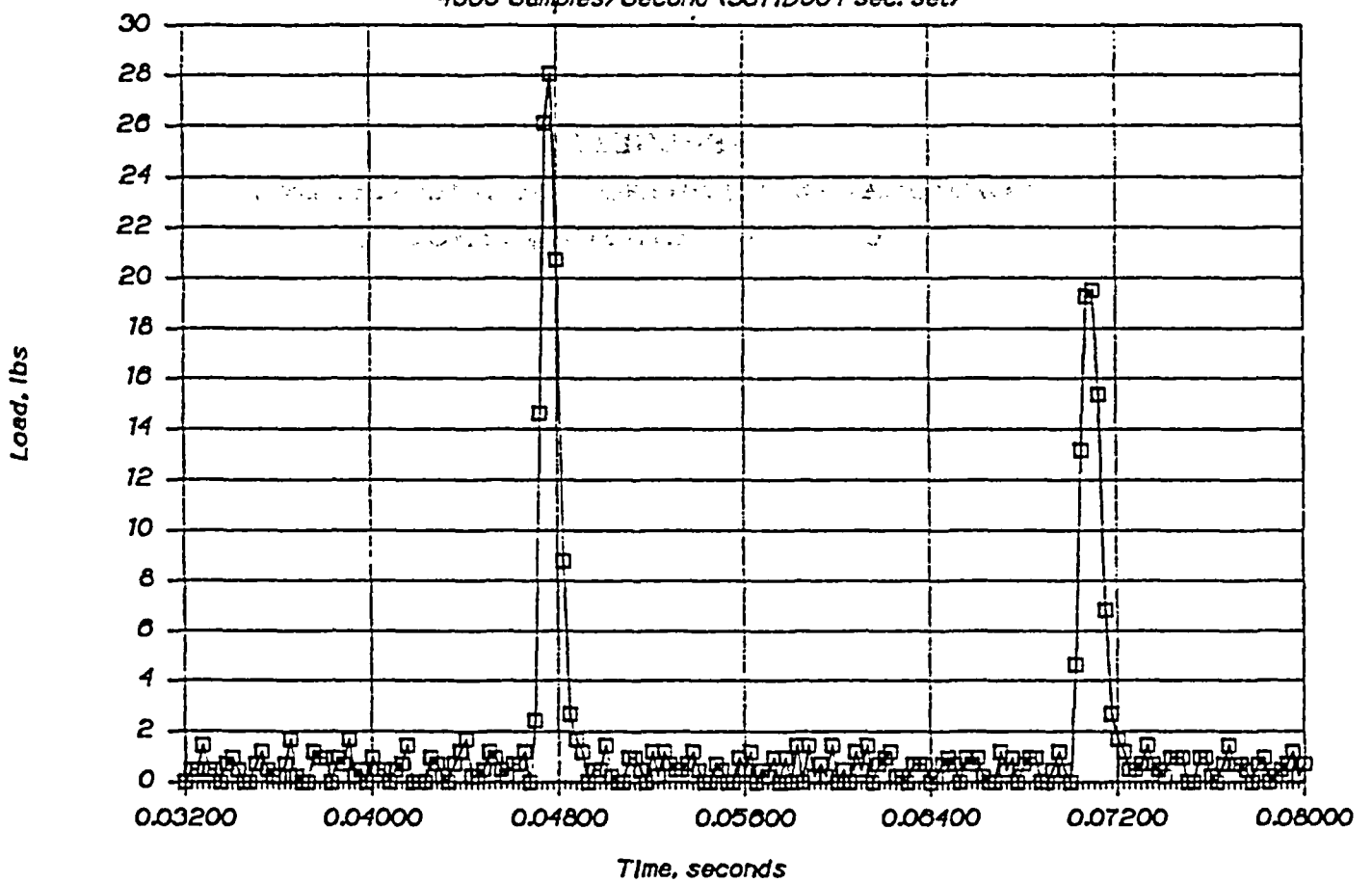0   2   4   6   8   10   12

FLOW VELOCITY, FPS
*: FROM REF 10

DISK NATURAL FREQUENCY

FIGURE 32

Tapping, Stop @ 73 deg, 17.5 fps

4000 Samples/Second (3UT1D004 sec. set)

DISK-TO-STOP CONTACT LOADS, 3" VALVE

FIGURE 33

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APPENDIX A
FORMULATION OF SWING DISC CHECK VALVE
DISC NATURAL FREQUENCY

APPENDIX A
FORMULATION OF SWING DISC CHECK VALVE
DISC NATURAL FREQUENCY

The natural frequency of a single degree of freedom spring-mass system is given by:

$$f_n = \frac{1}{2\pi} \sqrt{\frac{K_{stiff}}{M}}$$

where f_n = natural frequency in hertz

K_{stiff} = system spring rate, force/displacement

M = system mass

For the check valve, effective spring rate can be expressed as the force required to open the disc to some equilibrium angle, θ. An expression for this force as developed in Reference 5 is:

$$F = kA\rho V^2 \sin\theta \, \sin\theta/2$$

where

F = fluid force acting on the disc

k = an empirical constant dependent on valve geometry

A = disc area

V = fluid mean velocity

θ = fluid impingement angle (see Figure 1)

$$K_{stiff} = \frac{1}{R} \frac{dF}{d\theta} \quad \text{where R = radius from hinge pin to disc } \mathcal{C}_L$$

$$\therefore K_{stiff} = \frac{1}{R} kA\rho V^2 \frac{d}{d\theta}\left(\sin\theta \, \sin\frac{\theta}{2}\right)$$

$$K_{stiff} = \frac{kA\rho V^2}{R} \cdot \left(\cos\theta \, \sin\frac{\theta}{2} + \frac{1}{2} \cos\frac{\theta}{2} \sin\theta\right)$$

Finally,

$$f_n = \frac{1}{2\pi} \cdot \sqrt{\frac{kA\rho V^2 Z}{RM}}$$

where $Z = \left[\cos\theta \, \sin\frac{\theta}{2} + \frac{1}{2} \cos\frac{\theta}{2} \sin\theta\right]$

From the results of baseline tests reported earlier in this document, the relationship between velocity and θ can be measured. These values are tabulated below for both valves, along with the resulting natural frequencies. Pertinent valve data are:

	3"	6"	
A (ft^2)	0.067	0.26	
ρ (lb/ft^3)	62.4	62.4	(water, 70°F)
R (ft)	0.23	0.42	
M (lbs)	2.3	9.5	(1/2M$_{hinge\ arm}$ + M$_{disc}$)
k	2.0	2.0	

3" VALVE

Velocity fps	θ deg	Z	fn hertz
4	48	0.61	1.98
6	30	0.47	2.60
8	25	0.40	3.20
10	22	0.36	3.80
12	20	0.33	4.36

6" VALVE

Velocity fps	θ deg	Z	fn hertz
4	53	0.63	1.44
6	34	0.51	1.94
8	27	0.43	2.38
10	24	0.39	2.83
12	22	0.36	3.27

APPENDIX B
METHOD FOR ESTIMATING IMPACT FORCE
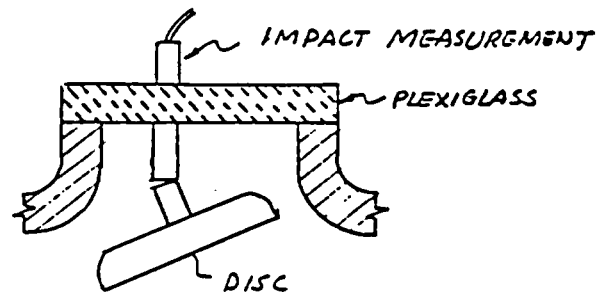### FROM TEST RESULTS

APPENDIX B
METHOD FOR ESTIMATING IMPACT FORCE
FROM TEST RESULTS

Check valve testing uses a plexiglass bonnet for test observation of disc motions. The stiffness of this plexiglass bonnet is different from the actual steel bonnet stiffness; therefore, the time history of impact force measurements from the tests will be different from the actual disc impact in service for the same amount of disc energy before impact. However, this difference can be adjusted based on the following estimations:
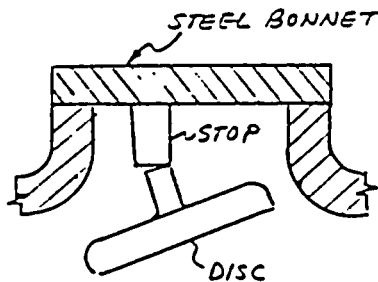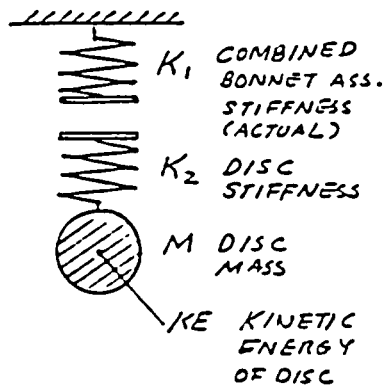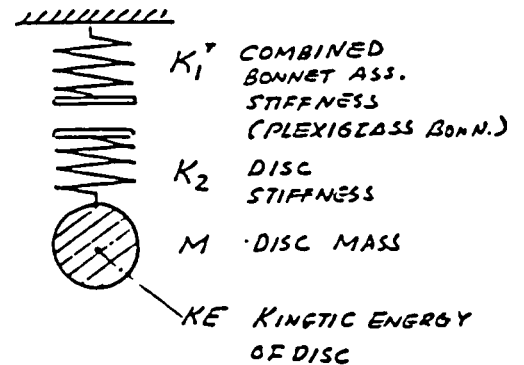
ACTUAL VALVE ASSEMBLY   TEST ASSEMBLY

The assemblies can be described in the following math models:

ACTUAL VALVE ASSEMBLY   TEST ASSEMBLY

The stiffnesses can be estimated from the design dimensions and material properties. Then from the test results we can estimate the impact in the actual valve assembly as:

Let KE = Kinetic energy before impact

K_A = Combined stiffness for actual valve assembly

$$= \frac{1}{\dfrac{1}{K_1} + \dfrac{1}{K_2}}$$

K_T = Combined stiffness for test assembly

$$= \frac{1}{\dfrac{1}{K'_1} + \dfrac{1}{K_2}}$$

At peak of impact force, the KE is completely converted to the potential energy, PE, (assuming no permanent deformation as:

$$PE = KE = \frac{1}{2} K \delta^2 = \frac{1}{2} K \left(\frac{P}{K}\right)^2 = \frac{1}{2} \frac{P^2}{K}$$
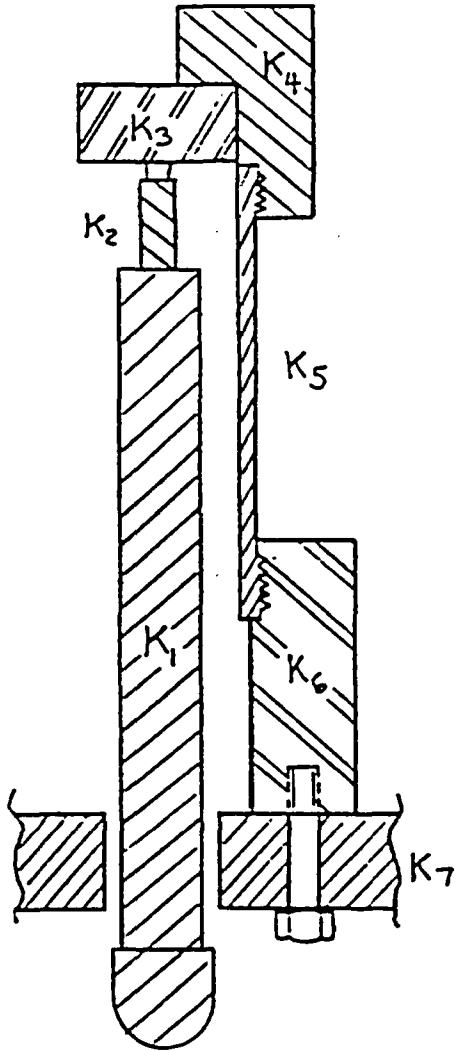
where P = Peak force

K = Combined stiffness

If P_A is the actual impact force and P_T is test measured peak force, then we have

$$PE = KE = \frac{1}{2} \frac{P_A^2}{K_A} = \frac{1}{2} \frac{P_T^2}{K_T}$$

$$\therefore P_A = P_T \sqrt{\frac{K_A}{K_T}}$$

$$K_1 = \frac{EA}{L} = \frac{30 \times 10^6 \times \frac{\pi}{4}(3/8)^2}{7}$$

$$= 473,343 \text{ lb/in} \leftarrow$$

$$K_2 = \frac{EA}{L} = \frac{30 \times 10^6 \times \frac{\pi}{4}(3/16)^2}{0.5}$$

$$= 1,656,699 \text{ lb/in} \leftarrow$$

$$K_3 = \frac{200}{.001} = 200,000 \text{ lb/in} \leftarrow$$

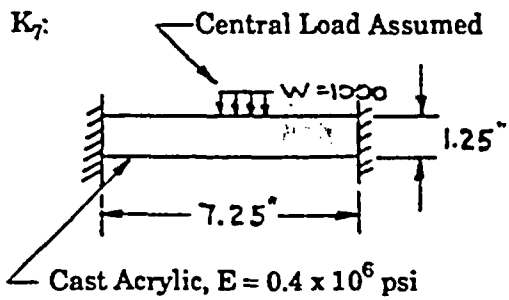$$K_4 = \frac{EA}{L} = \frac{30 \times 10^6 \times \frac{\pi}{4}(2^2 - 1^2)}{1}$$

$$= 70,685,835 \text{ lb/in} \leftarrow$$

$$K_5 = \frac{EA}{L} = \frac{30 \times 10^6 \times \frac{\pi}{4}(3/4^2 - 3/8^2)}{2.5}$$

$$= 3,976,078 \text{ lb/in}$$

$$K_6 = \frac{EA}{L} = \frac{30 \times 10^6 \times \frac{\pi}{4}(2\,1/4^2 - 13/16^2)}{1.5}$$

$$= 69,151,854 \text{ lb/in}$$

K_7:

—Central Load Assumed

$W = 1000$

$1.25"$

$7.25"$

Cast Acrylic, $E = 0.4 \times 10^6$ psi

$$Y_{max} = \frac{-W\,a^2}{16\pi\,D}, \quad a = \text{radians}$$

$$D = \frac{E\,t^3}{12\,(1 - v^2)}$$

$$\therefore Y_{max} = \frac{-1000\left(\dfrac{7.25}{2}\right)^2}{16\pi\,\dfrac{30 \times 10^6\, 1.25^3}{12\,(1 - 0.3^2)}}$$

$$= 0.00004872\,\text{in}$$

$$\therefore \quad K_7 = 20{,}524{,}968\ \text{lb/in}$$

$$K_T = \cfrac{1}{\dfrac{1}{K_2} + \dfrac{1}{K_2} + \cdots + \dfrac{1}{K_7}}$$

$$= \frac{1}{10^{-6}\,(2.1126 + 0.6036 + 5 + 0.014 + 0.252 + 0.0145 + 0.049)}$$

$$= 10^{-6} \times 8.0457$$

$$K_T = 124{,}290\ \text{lb/in} \quad \leftarrow$$

$$K_{Disc} = \frac{AE}{L} = \frac{\dfrac{\pi}{4}\,(0.6875)^2 \times 30 \times 10^6}{1.5} = 7{,}424{,}467\ \text{lb/in}$$

$$\gg K_T$$

If K_{Disc} is close to the actual total stiffness of the disc and bonnet assembly (i.e. $K_{Bonnet} \gg K_{Disc}$), then

$$F_{Actual} = F_{Measured}\left(\sqrt{\frac{K_{Disc}}{K_T}}\right)$$

$$= 35\sqrt{\frac{7{,}424{,}467}{124{,}290}} = 270\ \text{lb}$$

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13 ABSTRACT (200 words or less)

Degradation and failure of swing check valves and resulting damage to plant equipment
has led to a need to develop a method to predict performance and degradation of
these valves in nuclear power plant systems. This Phase I investigation developed
methods which can be used to predict the stability of the check valve disk when
piping disturbances such as elbows, reducers, and generalized turbulence sources
are present upstream of the valve within 10 pipe diameters. Major findings include
the flow velocity required to achieve a full open, stable disk position, the magnitude
of disk motion developed with these upstream disturbances with flow velocities below
full open conditions, as well as disk natural frequency data which can be used to
predict wear and fatigue damage. Reducers were found to cause little or no performance
degradation. Elbow effects must be considered when located within 5 diameters of
the check valve, while severe turbulence sources have significant effect at distances
to 10 diameters.

Clearway swing check designs were found to be particularly sensitive to manufacturing
tolerances and installation variables making them likely candidates for premature
failure. Reducing the disk full opening angle on these clearway designs results
in significant performance improvement.

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Swing check valves, check valve degradation, disk fluctuations, upstream
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PREDICTION OF CHECK VALVE PERFORMANCE AND DEGRADATION IN NUCLEAR POWER PLANT SYSTEMS

MAY 1988