CONTENTS

INTRODUCTION

This report documents Carolina Power & Light Company's review of operability of the containment purge and vent valves and their ability to close during a design basis accident to assure containment isolation. This review is required by NUREG-0737, II.E.4.2, *Containment Isolation Dependability.*

The report follows the guidance presented in the Nuclear Regulatory Commission letter dated February 11, 1985 (attached).

(1427PSA/crs)

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3/4/85

NLS-85-122

UNITED STATES
NUCLEAR REGULATORY COMMISSION
WASHINGTON, D. C. 20555

FEB 11 1985

Docket No.: 50-400

Mr. E. E. Utley, Executive Vice President
Power Supply & Engineering & Construction
Carolina Power & Light Company
P. O. Box 1551
Raleigh, North Carolina 27602

Dear Mr. Utley:

Subject: Request for Additional Information on Containment Purge and Vent Valve Operability

Continued staff review of the Shearon Harris, Unit 1 OL application has resulted in the need for additional information, as delineated in the enclosure, in the area of containment purge and vent valve operability.

Please inform the NRC Project Manger of your schedule for responding on this issue.

Sincerely,

George W. Knighton

George W. Knighton, Chief
Licensing Branch No. 3
Division of Licensing

Enclosure:
As stated

cc: See next page

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Operability Qualification of Purge and Vent Valves

Demonstration of operability of the containment purge and vent valves and the ability of these valves to close during a design basis accident is necessary to assure containment isolation. This demonstration of operability is required by NUREG-0737, "Clarification of TMI Action Plan Requirements," II.E.4.2 for containment purge and vent valves which are not sealed closed during operational conditions 1, 2, 3 and 4.

1. For each purge and vent valve covered in the scope of this review, the following documentation demonstrating compliance with the "Guidelines for Demonstration of Operability of Purge and Vent Valves" (attached, Attachment #5) is to be submitted for staff review:

   A. Dynamic Torque Coefficient Test Reports (Butterfly valves only) - including a description of the test setup.

   B. Operability Demonstration or In-situ Test Reports, (when used)

   C. Stress Reports

   D. Seismic Reports for Valve Assembly (valve and operator) and associated parts.

   E. Sketch or description of each valve installation showing the following (Butterfly valves only):

      1. direction of flow

      2. disc closure direction

      3. curved side of disc, upstream or downstream (asymetric discs)

      4. orientation and distance of elbows, tees, bends, etc. within 20 pipe diameters of valve

      5. shaft orientation

      6. distance between valves

   F. Demonstration that the maximum combined torque developed by the valve is below the actuator rating.

2. The applicant should respond to the "Specific Valve Type Questions" (attached) which relate to his valve.

3. Analysis, if used, should be supported by tests which establish torque coefficients of the valve at various angles. As torque coefficients in butterfly valves are dependent on disc shape aspect ratio, angle of closure flow direction and approach flow, these things should be accurately represented during tests. Specifically, piping installations (upstream and downstream of the valve) during the test should be representative of actual field installations. For example, non-symetric approach flow from an elbow upstream of a valve can result in fluid dynamic torques of double the magnitude of those found for a valve with straight piping upstream and downstream.

4. In-situ tests, when performed on a representative valve, should be performed on a valve of each sinze/type which is determined to represent the worst case load. Worst case flow direction, for example, should be considered.

   For two valves in series where the second valve is a butterfly valve, the effect of non-symetric flow from the first valve should be considered if the valves are within 15 pipe diameters of each other.

5. If the applicant takes credit for closure time vs. the buildup of containment pressure, he must demonstrate that the method is conservative with respect to the actual valve closure rate. Actual valve closure rate is to be determined under both loaded and unloaded conditions and periodic inspection under tech. spec. requirements should be performed to assure closure rate does not increase with time or use.

Specific Valve Type Questions

The following questions apply to specific valve types only and need to be answered only where applicable. If not applicable, state so.

A. Torque Due To Containment Backpressure Effect (TCB)
For those air operated valves located inside containment, is the operator design of a type that can be affected by the containment pressure rise (backpressure effect) i.e. where the containment pressure acts to reduce the operator torque capability due to TCB. Discuss the operator design with respect to the air vent and bleeds. Show how TCB was calculated (if applicable).

B. Where air operated valve assemblies use accumulators as the fail-safe feature, describe the accumulator air system configuration and its operation. Discuss active electrical components in the accumulator system, and the basis used to determine their qualification for the environmental conditions experienced. Is this system seismically designed? How is the allowable leakage from the accumulators determined and monitored.

C. For valve assemblies requiring a seal pressurization system (inflatable main seal), describe the air pressurization system configuration and operation including means used to determine that valve closure and seal pressurization have taken place. Discuss active electrical components in this system, and the basis used to determine their qualification for the environmental condition experienced. Is this system seismically designed?

D. Where electric motor operators are used to close the valve has the minimum available voltage to the electric operator under both normal or emergency modes been determined and specified to the operator manufacturer to assure the adequacy of the operator to stroke the valve at accident conditions with these lower limit voltages available? Does this reduced voltage operation result in any significant change in stroke timing? Describe the emergency mode power source used.

E. Where electric motor and air operator units are equipped with handwheels, does their design provide for automatic re-engagement of the motor operator following the handwheel mode of operation? If not, what steps are taken to preclude the possibility of the valve being left in the handwheel mode following some maintenance, test etc. type operation?

F. For electric motor operated valves have the torques developed during operation been found to be less than the torque limiting settings?

GUIDELINES FOR DEMONSTRATION OF OPERABILITY OF PURGE AND VENT VALVES

OPERABILITY

In order to establish operability it must be shown that the valve actuator's torque capability has sufficient margin to overcome or resist the torques and/or forces (i.e., fluid dynamic, bearing, seating, friction) that resist closure when stroking from the initial open position to full seated (bubble tight) in the time limit specified. This should be predicted on the pressure(s) established in the containment following a design basis LOCA. Considerations which should be addressed in assuring valve design adequacy include:

1. Valve closure rate versus time - i.e., constant rate or other.
2. Flow direction through valve; ΔP across valve.
3. Single valve closure (inside containment or outside containment valve) or simultaneous closure. Establish worst case.
4. Containment back pressure effect on closing torque margins of air operated valve which vent pilot air inside containment.
5. Adequacy of accumulator (when used) sizing and initial charge for valve closure requirements.
6. For valve operators using torque limiting devices - are the settings of the devices compatible with the torques required to operate the valve during the design basis condition.
7. The effect of the piping system (turns, branches) upstream and downstream of all valve installations.
8. The effect of butterfly valve disc and shaft orientation to the fluid mixture egressing from the containment.

DEMONSTRATION

Demonstration of the various aspects of operability of purge and vent valves may be by analysis, bench testing, insitu testing or a combination of these means.

Purge and vent valve structural elements (valve/actuator assembly) must be evaluated to have sufficient stress margins to withstand loads imposed while valve closes during a design basis accident. Torsional shear, shear, bending, tension and compression loads/stresses should be considered. Seismic loading should be addressed.

Once valve closure and structural integrity are assured by analysis, testing or a suitable combination, a determination of the sealing integrity after closure and long term exposure to the containment environment should be evaluated. Emphasis should be directed at the effect of radiation and of containment spray chemical solutions on seal material. Other aspects such as the effect on sealing from outside ambient temperatures and debris should be considered.

E 1
A 2

The following considerations apply when testing is chosen as a means for demonstrating valve operability:

Bench Testing

A. Bench testing can be used to demonstrate suitability of the in-service valve by reason of its traceability in design to a test valve. The following factors should be considered when qualifying valves through bench testing.

1. Whether a valve was qualified by testing of an identical valve assembly or by extrapolation of data from a similarly designed valve.

2. Whether measures were taken to assure that piping upstream and downstream and valve orientation are simulated.

3. Whether the following load and environmental factors were considered

a. Simulation of LOCA
b. Seismic loading
c. Temperature soak
d. Radiation exposure
e. Chemical exposure
d. Debris

Bench testing of installed valves to demonstrate the suitability of the specific valve to perform its required function during the postulated design basis accident is acceptable.

1. The factors listed in items A.2 and A.3 should be considered when taking this approach.

In-Situ Testing

In-situ testing of purge and vent valves may be performed to confirm the suitability of the valve under actual conditions. When performing such tests, the conditions (loading, environment) to which the valve(s) will be subjected during the test should simulate the design basis accident.

NOTE: Post test valve examination should be performed to establish structural integrity of the key valve/actuator components..

PURGE AND VENT VALVES
OPERABILITY REVIEW FORM

Plant: Shearon Harris Nuclear Power Plant Unit No. 1

Utility: Carolina Power & Light Company

A/E: Ebasco Services Incorporated

I. VALVE IDENTIFICATION

- System Served: NORMAL CONTAINMENT PURGE
- Valve Tag Number(s):
  - Inside Containment 2CP-B1SA-1 & 2CP-B5SA-1
  - Outside Containment 2CP-B2SB-1 & 2CP-B6SB-1
- Ebasco Specification: CAR-SH-BE-35

	Valve Data(*)	Actuator Data(*)
- Manufacturer :	BIF	BETTIS
- Model:	0657	N721C-SR60-12
- Serial Number:	SEE NOTE BELOW	SEE NOTE BELOW
- Type:	LUG WAFER BUTTERFLY	PNEUMATIC, SPRING RETURN
- Size:	8"	12

(*) additional valve/actuator data is provided on the Pump and Valve Operability Review forms.

NOTE: THE SERIAL NUMBERS WILL BE PROVIDED AFTER THE FABRICATION IS COMPLETED**

** FABRICATION COMPLETE SERIAL NUMBERS ARE
2CP-B1SA-1 SN-N67926-1 : 2CP-B5SA-1 SN.-N67926-2
2CP-B2SB-1 SN.-N67927-1 : 2CP-B6SB-1 SN.-N67927-2

RM Yarrow 5/16/85
[illegible]

II. OPERABILITY DEMONSTRATION

1. Purge and Vent Valve Operability Documentation References: (identify documents by title, number, revision and page no. as applicable, to aid review).

A. Dynamic Torque Coefficient Test Report(s) (Butterfly valves only) - including description of the test setup:
1) "DYNAMIC TORQUE CALCULATION OF BUTTERFLY VALVE" REPORT NO. DT-67926 REV. A.
2) "HYDRODYNAMIC AND HEADLOSS TEST OF 12"-150B BUTTERFLY VALVE WITH DIRECTLY CONNECTED SHORT RADIUS ELBOW UPSTREAM" TEST REPORT NO. TR-0650-43 DATED 2.24.82

B. In-situ Test Reports (indicate N/A if not used for operability demonstration):

N/A

C. Stress Report(s):

"SEISMIC ANALYSIS OF BUTTERFLY VALVES FOR EBASCO/CAROLINA POWER AND LIGHT" REPORT NO - N-67926 REV. A

D. Seismic Report(s) for Valve Assembly (valve and operator) and associated parts:

SAME AS ITEM C.

E. Sketch or Description of Valve Installation(s) (Butterfly valves only) showing the following:
(1) direction of flow; (2) disc closure direction;
(3) curved side of disc, upstream or downstream (asymetric discs); (4) orientation and distance of elbows, tees, bends, etc., within 20 pipe diameters

of valve; (5) shaft orientation; (6) distance between valves;

A WORST CASE INSTALLATION IS SHOWN IN THE FIGURE IN ITEM 1.E. ON PAGE 38 OF DYNAMIC TORQUE REPORT NO. DT-67926 REV. A. SEE ATTACHMENT 'C' FOR ACTUAL PIPING LAYOUTS IN ISOMETRIC FOR COMPARISON.

Note: If a worst case installation is analyzed, identify and include documentation that establishes it as a worst case.

F. Documentation that the maximum torque developed by the valve is below the actuator rating: SEE DYNAMIC TORQUE CALCULATION REPORT NO. DT-67926 REV. A, PAGES 2,3,18 THRU 37

2. Specific Valve Type Questions [Refer to Attachment A] — ITEMS 5.2 THRU 5.6 ARE NOT APPLICABLE. ITEM 5.1 IS ADDRESSED IN ITEM 4, PAGE 40 OF REPORT NO. DT-67926 REV. A. AN EXPLANATION WHY CONTAINMENT BACK PRESSURE WILL NOT AFFECT THE CLOSING TORQUE OF OPERATOR IS GIVEN. SEE ATTACHMENT 'B' FOR ACTUATOR AND PNEUMATIC PIPING CONNECTIONS.

3. Analysis Review:

The reviewer is to answer the following questions, identify and provide the reference documentation. In addition, justification for any "NO" answers is to be included with the reference(s).

a. Is the analysis supported by tests which establish torque coefficients of the valve at various angles?

YES [X]   NO [ ]

Reference: AIF TEST REPORT NO. TR-0650-43, "HYDRODYNAMIC AND HEAD LOSS TEST OF 12"-150LB. BUTTERFLY VALVE WITH DIRECTLY CONNECTED SHORT RADIUS ELBOW UPSTREAM" DATED 2.24.82

b. For Butterfly valves, were disc shape aspect ratio, angle of closure, flow direction and approach flow accurately represented during tests? Specifically, piping installations (upstream and downstream of the valve) during the test should be representative (or worst case) of actual field installations.

YES [X] NO [ ]

Reference(s) REPORT NO. DT-67926 REV. A DATED 11.22.83 PAGE 13 FOR WORST CASE

4. In-situ Tests:

In-situ tests, when performed on a representative valve, should be performed on a valve of each size/kind which is determined to represent the worst case load.

In-situ Test Reference(s) ____________________

In-situ Tests Not Performed [X]

5. Valve Closure Time:

a. Is credit taken for closure time vs. the buildup of containment pressure? (Method to be conservative with respect to actual valve closure rate).

YES [X] NO [ ]

Reference(s) REPORT NO. DT-67926 REV. A DATED 11.21.83 PAGES 7 THRU 12.

b. Has valve closure rate been determined under loaded conditions?

YES [X] NO [ ]

Reference(s) REPORT NO. DT-67926 REV. A DATED 11.22.83 PAGES 7 THRU 12

II. CONSIDERATIONS FOR DEMONSTRATION OF OPERABILITY

The reviewer is to address the following considerations, identify and provide the reference documentation for each (use additional sheets if necessary):

1. Valve closure rate versus time - i.e. constant rate or other.

   Reference(s) REPORT NO. DT-67926 REV. A. USES CONSTANT VALVE CLOSURE RATE. SEE PAGES 7 THRU 12, 14, 15 AND 40

2. Flow direction through valve; differential pressure across valve.

   Reference(s) REPORT NO. DT-67926 REV. A - FLOW DIRECTION IS MARKED ON THE FIGURE ON PAGE 38; DIFFERENTIAL PRESSURES ACROSS THE VALVES AT DIFFENT ANGLES OF DISC ARE CALCULATED ON PAGES 18 THRU 37

3. Single valve closure (inside containment or outside containment valve) or simultaneous closure. Establish worst case.

   Reference(s) INSIDE AND OUTSIDE CONTAINMENT VALVES ARE IN SERIES WITH RESPECT TO FLOW. SINGLE VALVE CLOSURE GIVES HIGHER FLOW RATE RESULTING IN HIGHER DYNAMIC TORQUE REQUIREMENT. THEREFORE SINGLE VALVE CLOSURE IS THE WORST CASE.

4. Containment back pressure effect on closing torque margins of air operated valves which vent pilot air inside containment.

   Reference(s) REPORT NO. DT-67926 REV. A, PAGE 40, ITEM 4 EXPLAINS WHY CONTAINMENT BACK PRESSURE WILL NOT AFFECT THE CLOSING TORQUE OF OPERATOR. ALSO, SUFFICIENT MARGIN BETWEEN THE MAXIMUM REQUIRED DYNAMIC TORQUE AND ACTUAL CAPACITY OF ACTUATOR EXISTS. (SEE PAGE 2 OF REPORT)

5. Adequacy of accumulator (indicate N/A if not used) sizing and initial charge for valve closure requirements.

   Reference(s) N/A

6. Torque limiting devices (indicate N/A if not used) - are the settings compatible with the torques required to operate the valve during the design basis condition?

   Reference(s) N/A

7. The effect of the piping system (turns, branches) upstream and downstream of all valve installations.

   Reference(s) THE FIGURE ON PAGE 38 OF THE REPORT DT-67926 REV.A GIVES THE WORST CONDITION COMPARED TO THE ACTUAL PIPING SYSTEM

8. The effect of butterfly valve disc and shaft orientation to the fluid mixture egressing from the containment.

   Reference(s) REPORT NO. DT-67926 REV.A, PAGE 13 DESCRIBES THE WORST CASE VALVE AND DISC ORIENTATION CONSIDERED IN THE ANALYSIS.

In addition to the above review, sealing integrity after closure and long term exposure to the containment environment should be considered under the mechanical equipment environmental qualification program - refer to Pump and Valve Operability Forms for this information.

Reviewer: Print Name - Sign/Date Y. JAGANNATH - Y. Jagannath 12/11/84

Checker: Print Name - Sign/Date E. BORKOWSKA - E Borkowska 12/11/84

PURGE AND VENT VALVES
OPERABILITY REVIEW FORM
ATTACHMENT A

The following Questions apply to specific valve types only and need to be answered only where applicable. If not applicable, state so. A resonse is expected for each item.

5.1 Torque Due to Containment Backpressure Effect (TCB)

For those air operated valves located inside containment, is the operator design of a type that can be affected by the containment pressure rise (backpressure effect), i.e., where the containment pressure acts to reduce the operator torque capability due to TCB. Discuss the operator design with respect to the air vent and bleeds. Explain in detail how TCB was calculated (if applicable).

5.2 Where air operated valve assemblies use accumulators as the fail safe feature, describe the accumulator air system configuration and its operation. Discuss active electrical components in the accumulator system, and the basis used to determine their qualification for the environmental conditions experienced. Is this system seismically designed? How is the allowable leakage from the accumulators determined and monitored? Is the accumulator size and initial charge adequate for valve closure.

5.3 For valve assemblies requiring a seal pressurization system (inflatable main seal), describe the air pressurization system configuration and operation including means used to determine their qualification for the environmental condition experienced. Is this system seismically designed?

5.4 Where electric motor operators are used to close the valve has the minimum available voltage to the electric operator under both

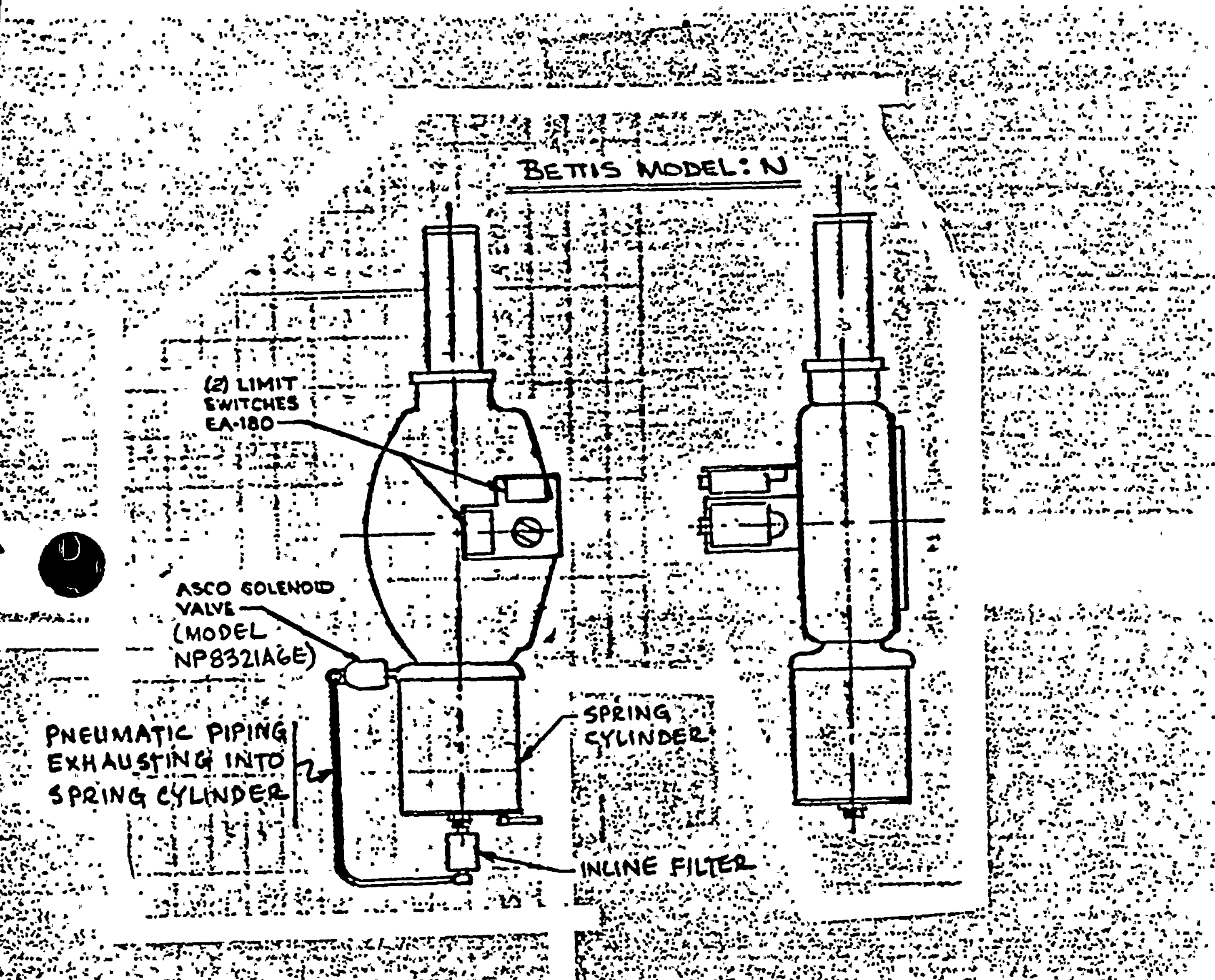
normal or emergency modes been determined and specified to the operator manufacturer to assure the adequacy of the operator to stroke the valve at accident conditions with these lower limit voltages available? Does this reduced voltage operation result in any significant change in stroke timing? Describe the emergency mode power source used.

5.5 Where electric motor and air operator units are equipped with handwheels, does their design provide for automatic re-engagement of the motor operator following the handwheel mode of operation? If not, what steps are taken to preclude the possibility of the valve being left in the handwheel mode following some maintenance, test etc. type operation?

5.6 For electric motor operated valves have the torques developed during operation been found to be less than the torque limiting settings?

ATTACHMENT 'B'

PURGE AND VENT VALVES OPERABILITY REVIEW FORM
ITEM II.2 - SPECIFIC VALVE TYPE QUESTIONS, ITEM 5.1

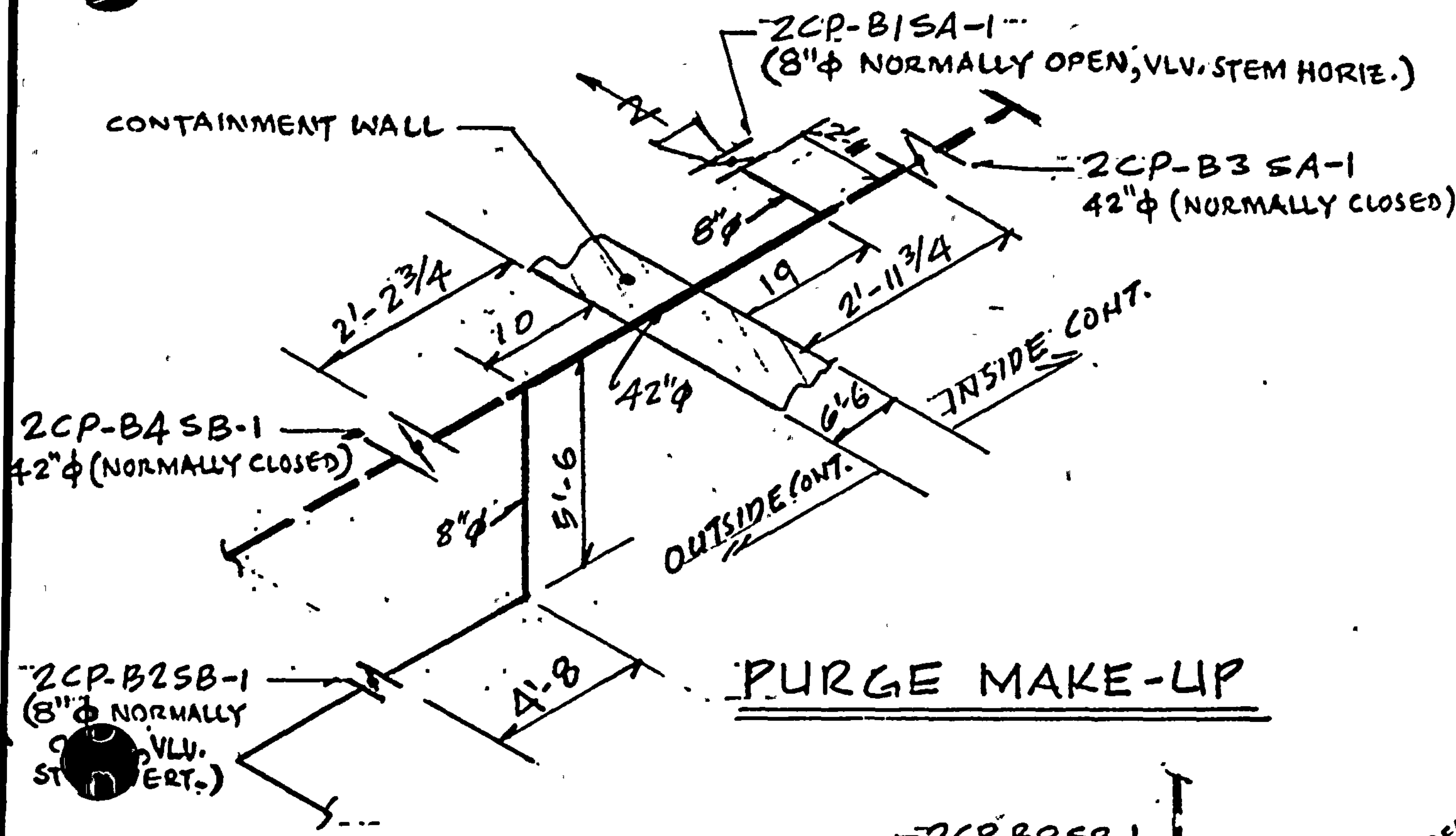


BETTIS ACTUATOR MODEL N721C-SR 60

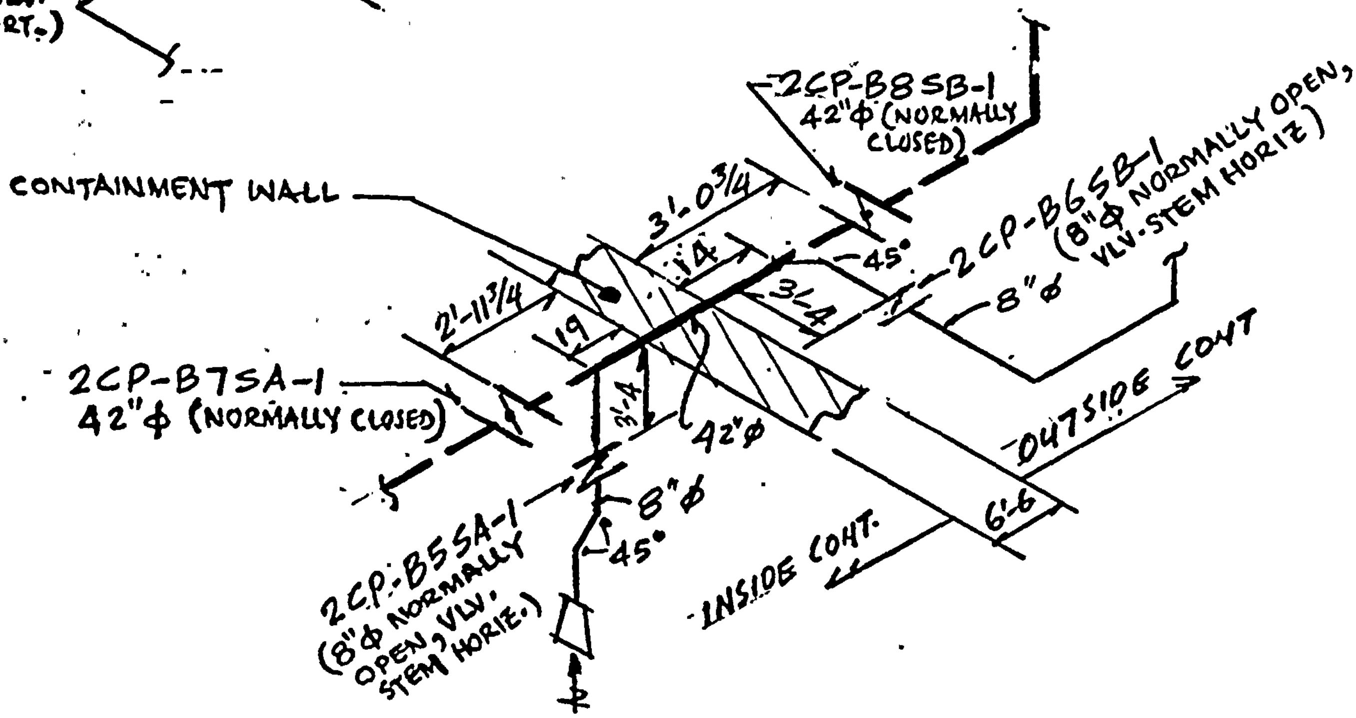
ATTACHMENT 'C'

PURGE AND VENT VALVES OPERABILITY REVIEW FORM

ITEM II.E



PURGE MAKE-UP



NORMAL CONT. PURGE

B I F A UNIT OF GENERAL SIGNAL
1600 DIVISION ROAD
WEST WARWICK, R.I. 02893

QUALIFICATION OF CONTAINMENT PURGE BUTTERFLY VALVES UNDER LOCA CONDITION.

DYNAMIC TORQUE CALCULATION OF BUTTERFLY VALVE

NUCLEAR

PREPARED FOR:

EBASCO SERVICES INCORPORATED
SHEARON HARRIS NUCLEAR POWER PLANT

VALVE SIZE: 8 INCH
EBASCO CONTRACT NO. NY-435211 & 435212
BIF ORDER NO.: N67926-U/N67927-U
EBASCO IDENTIFICATION NO. 2CP-B1SA
2CP-B2SB
2CP-B5SA
2CP-B6SB

Prepared by: Debendra K. Dias
Date: Nov. 8, 1983
Checked by: Dezso Szilagyi
Date: Nov. 21, 83
Approved by: Dezso Szilagyi
Date: Nov. 22, 83

REPORT NO. DT-67926
REVISION A

B I F A UNIT OF GENERAL SIGNAL
1600 DIVISION ROAD
WEST WARWICK, R.I. 02893

QUALIFICATION OF CONTAINMENT PURGE BUTTERFLY VALVES UNDER LOCA CONDITION.

DYNAMIC TORQUE CALCULATION OF BUTTERFLY VALVE

PREPARED FOR:

EBASCO SERVICES INCORPORATED
SHEARON HARRIS NUCLEAR POWER PLANT

VALVE SIZE: 8 inch
EBASCO CONTRACT NO. NY-435211 & 435212
BIF ORDER NO.: N67926-U/N67927-U
EBASCO IDENTIFICATION NO. 2CP-B1SA
2CP-B2SB
2CP-B5SA
2CP-B6SB

NUCLEAR

Prepared by: Debendra K. Das Debendra K. Das

Date: June 1, 1983

Checked by: Antonio M. Amaral Antonio M. Amaral

Date: June 6, 1983

Approved by: Richard Sciapito /BIF

Date: 6/30/83

REPORT NO. DT-67926

TABLE OF CONTENTS

REVISION (A) TO THE DYNAMIC TORQUE CALCULATION REPORT DT-67926

This revision is prepared to answer the questions presented in EBASCO letter dated 8/22/83 regarding the original Dynamic torque calculation. The response to these questions forms the basis of Revision (A) and are inserted in this report starting with page 38. These pages may be referred to for further details.

SUMMARY

This report contains the dynamic torque analysis of an 8 inch butterfly valve. The analysis is performed for LOCA (loss of Coolant Accident) per Ebasco Specification, reference 1 on page four of this report. The analytical procedure and the assumptions are outlined in the section beginning on page five.. Dynamic torque calculations have been performed for the valve at various angles of opening.

The results of the analysis presented on pages two and three of the report indicate that the dynamic torques developed under the specified flow conditions are less than the torque capability of the valve operator. Therefore, the operator is capable of providing sufficient torque to bring the valve from fully open to fully closed position in the event of a LOCA. In the seismic and stress analyses of this valve, the design torque used is greater than the maximum dynamic torque, thus qualifying the valve for LOCA conditions.

SUMMARY OF RESULTS

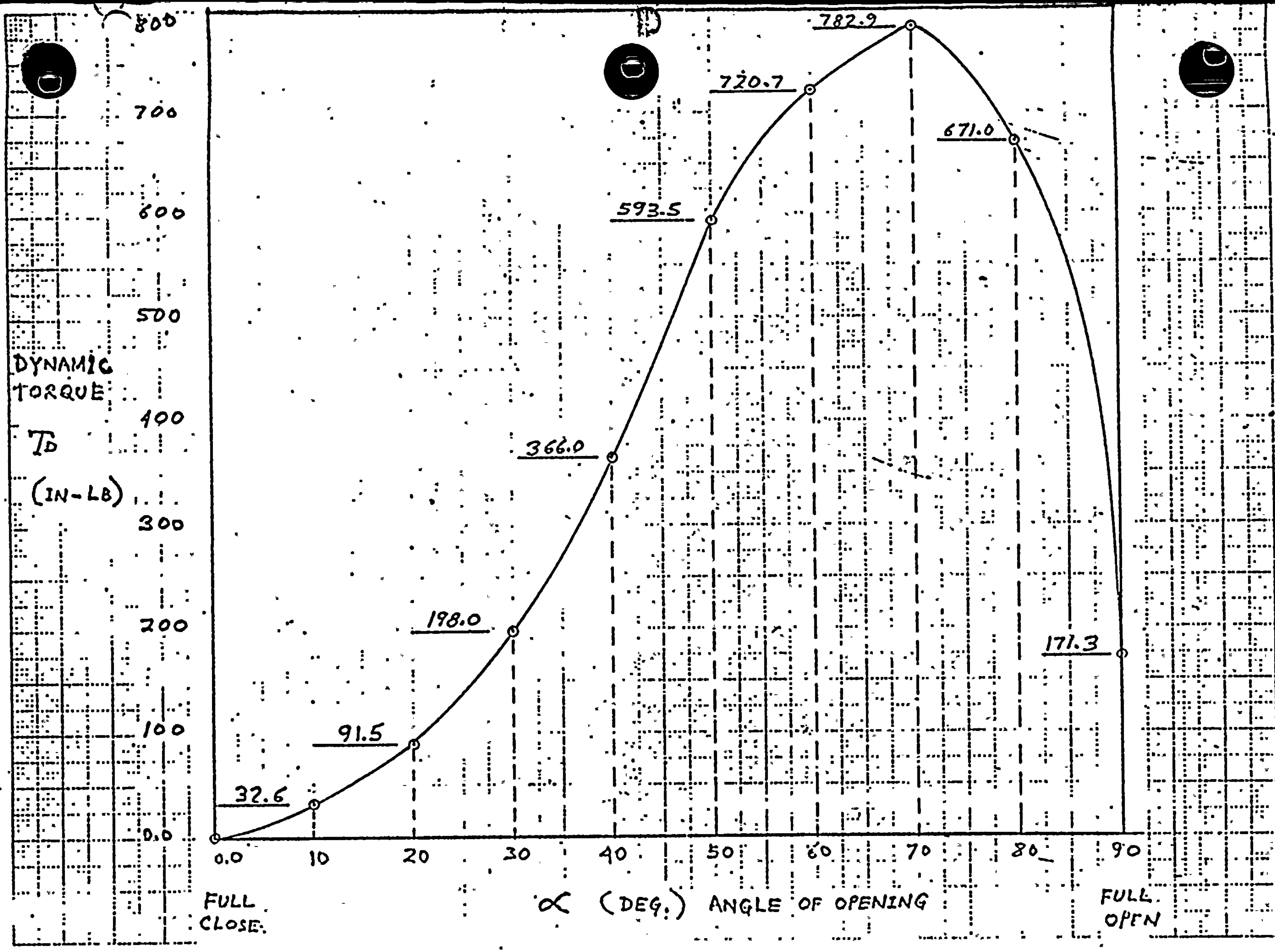
Table - 1, 8 inch Valve

Angle deg.	Dynamic Torque In-lb
90 (Full open)	171.3
80	671.0
70	783.0
60	720.7
50	593.5
40	366.0
30	198.0
20	91.5
10	32.6
0.0 (Full closed)	0.0*

T_{Net} = 1648 in-lb *

* At full closed position the dynamic torque is zero and the net (T_{Net}) torque is due to seating and bearing friction.

NOTE: The design torque used in the Seismic analysis for this valve is 1648 in-lb, which is greater than the maximum dynamic torque of 783 in-lb. Therefore the design is safe against the dynamic torque under LOCA condition.



DYNAMIC TORQUE
T_D
(IN-LB)
800
700
600
500
400
300
200
100
0.0
782.9
720.7
671.0
593.5
366.0
198.0
91.5
32.6
171.3
0.0
10
20
30
40
50
60
70
80
90
FULL CLOSE
α (DEG.) ANGLE OF OPENING
FULL OPEN

REFERENCES

1. Ebasco Letter dated Aug. 3, 1982 with attachment containing pressure, density, and flow data.

   Ebasco data, Fig. 6.2.1.1, for containment temperature rise.

2. ANSI/AWWA C504-80, AWWA Standard for Rubber-Seated Butterfly Valves. American Water Works Association, Colo.

3. Beard, C., Final Control Elements, Valves and Actuators, First Edition, Rimbach Publications, 1969.

4. Hutchison, J. W., ISA Handbook of Control Valves, 2nd Edition.

5. Stress report and torque sizing of B I F Butterfly valves, Job No. N-67926

B I F Test Report for Dynamic Torque and Head Loss Tests of Cast Iron Streamline Disc versus Fabricated Flat Plate Disc dated May 13, 1974.

7. B I F Test Report #TR-0650-43, Hydrodynamic and Headloss Test of 12" - 150 Lb. Butterfly Valve with directly connected short radius elbow upstream, dated 2/24/82.

8. Lyons, J. L., Lyon's Valve Designer's Handbook, Van Nostrand Reinhold Co., NY, 1982.

9. Crane Technical Paper No. 410, 1981 printing.

ANALYTICAL PROCEDURE FOR DYNAMIC TORQUE CALCULATION

The valve analysed in this report is a primary containment isolation butterfly valve used in the purge system. Valve size considered here is 8 inch. During the normal operation, the valve is in full open position and should close completely in case of an accident. In the event of a LOCA (loss of Coolant Accident), the valve has to close against ascending differential pressure. During the closing operation, the valve disc will be in semi-open positions and will experience fluid dynamic forces due to uneven pressure distribution across the faces of the disc. The flow through the valve causes aerodynamic effect on the disc that gives rise to the dynamic torque. This dynamic torque is given by the formula:

$$T_D = C_T \ (\Delta P)\ D^3 \qquad \text{(Ref. 2)....................(1)}$$

Where T_D = Dynamic Torque (in.-Lb.)

C_T = Coefficient of dynamic torque obtained from test (Dimensionless constant) (Ref. 7)

ΔP = Differential pressure across the valve (psi)

D = Disc diameter (in.).

During the closing operation of the valve, C_T and ΔP will be changing for varying closing angles of the disc. The dynamic torque will tend to close the valve whereas the shaft bearing friction torque will oppose it. The bearing friction torque is given by the formula:

$$T_b = \frac{\pi D^2}{4} \left[f_b \ (d/2) \ \Delta p \right] \qquad \text{(Ref. 2)............(2)}$$

T_b = Shaft bearing friction torque (Lb.-in.)

D = Valve Port diameter (in.)

f_b = Bearing friction coefficient (dimensionless constant) (Ref. 5)

d = Shaft diameter (in.)

Δp = Differential pressure (psi)

Therefore, the net unbalanced torque is

$T_N = T_D - T_b$

The differential pressure Δp across the valve shall be calculated from the flow rate established earlier under LOCA Condition. The equation used will be the one for sub-sonic gas flow recommended by the Fluid Controls Institute:

$$Q_S = 963\, C_V \sqrt{\frac{P_1^2 - P_2^2}{GT_1}} \qquad \text{(Ref. 3 and 4)} \ldots\ldots\ldots\ldots\ldots \text{(3)}$$

Where Q_S = Gas flow in SCFH

P_1 = Valve upstream pressure (psia)

P_2 = Valve downstream pressure (psia)

G = Specific gravity (air = 1 at 60^oF and 1 atm. pressure)

T_1 = Upstream temperature in ^{o} Rankine

C_V = Valve coefficient = $\frac{29.9D^2}{\sqrt{K_V}}$

D = Valve Port diameter (in.)

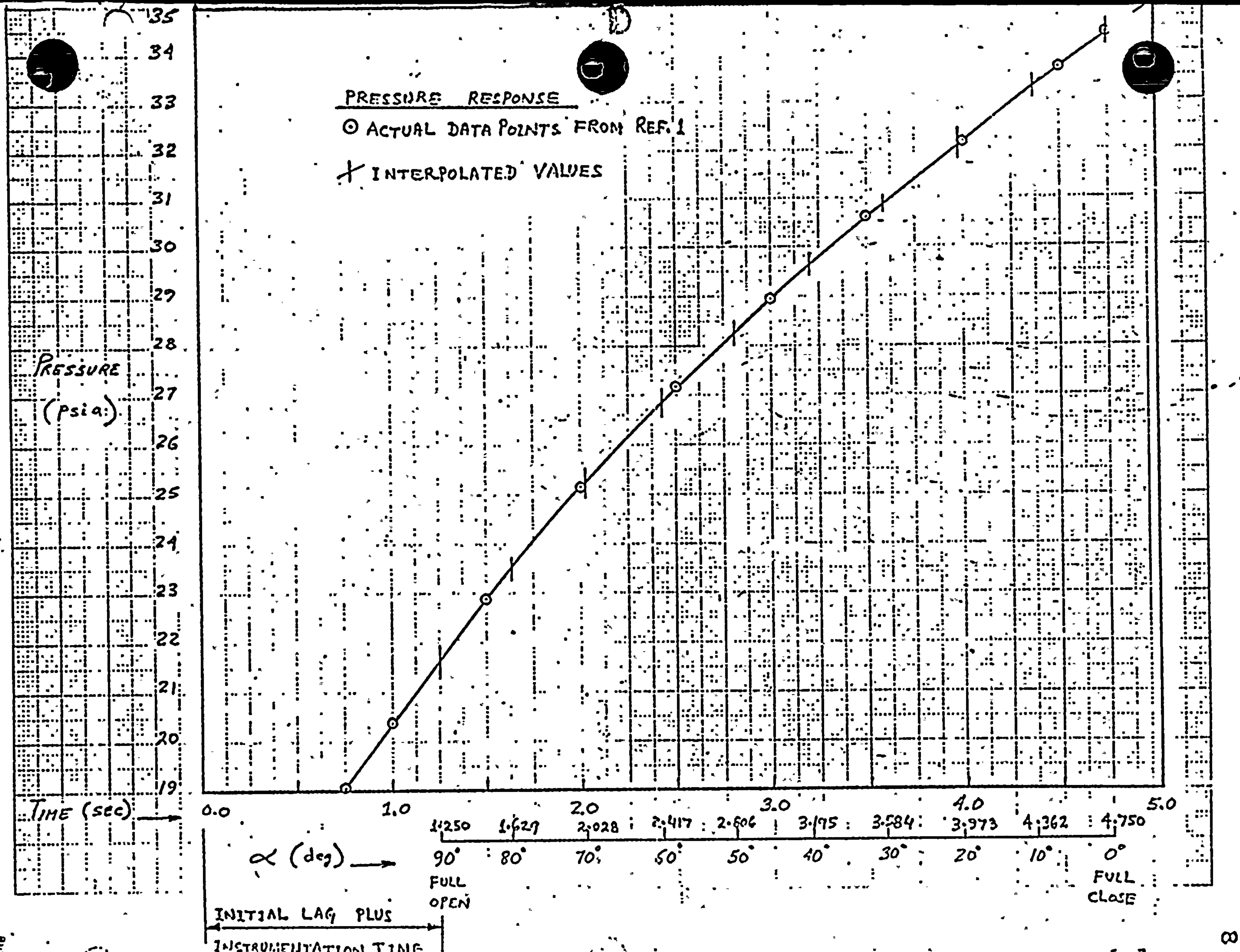
K_V = Coefficient of flow (dimensionless constant) (Ref. 7)

$$Q_S = Q_A \cdot \left[\frac{520\, P_1}{14.7\, T_1} \right] \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(4)}$$

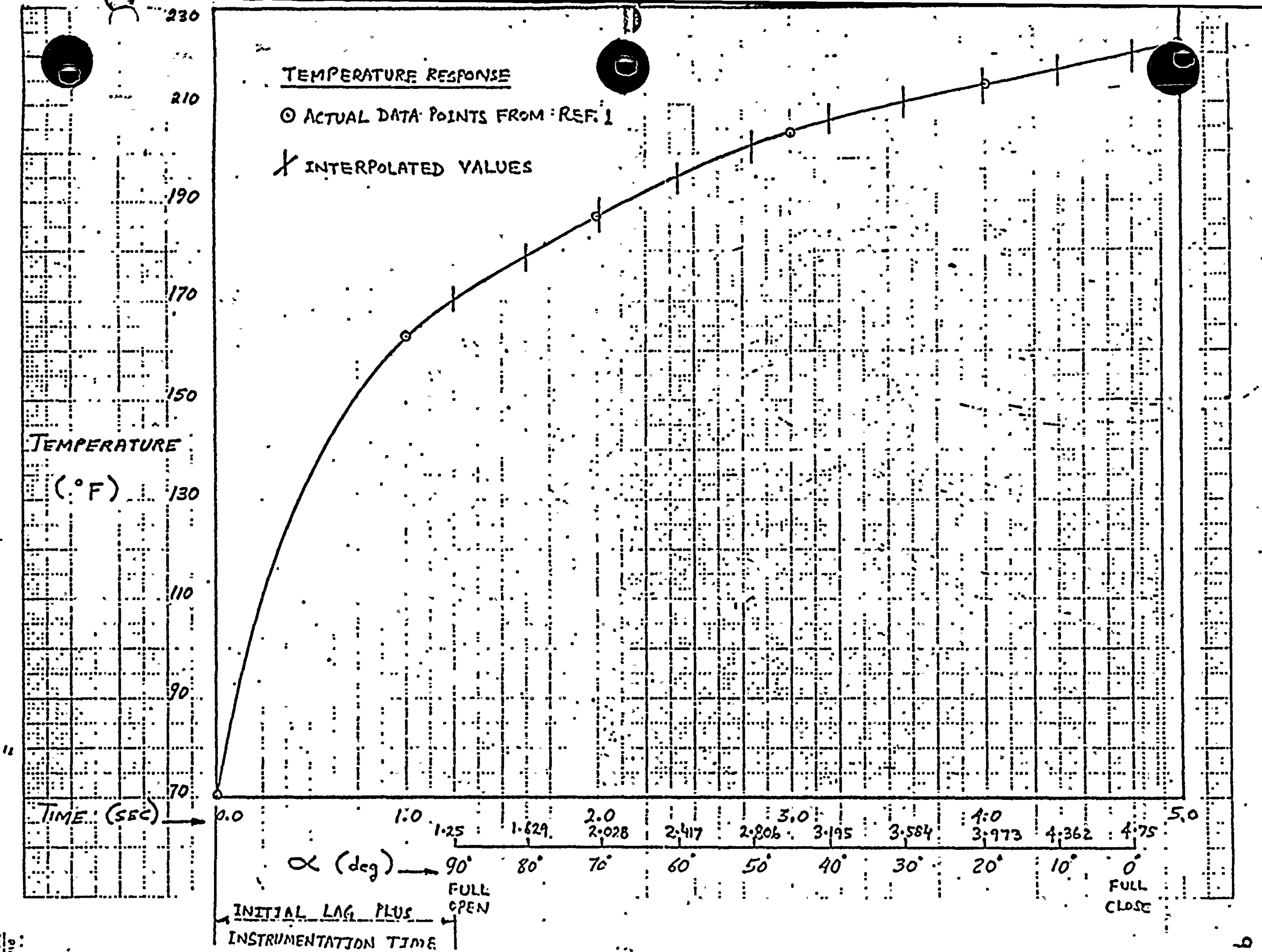
Where Q_A = Actual flow rate in ft^3/hr

Ebasco recommends that the containment isolation signal which initiates valve closure is energized at 0.75 second into the LOCA and the time delay (instrumentation time) before the signal reaches the solenoid valve of the operator so that the butterfly valve starts to close is given to be 0.5 second. Time of closure from the full-open position to full-close position is 3.5 seconds. Therefore, from the onset of LOCA to the full closure of the valve the time duration is 4.75 seconds.

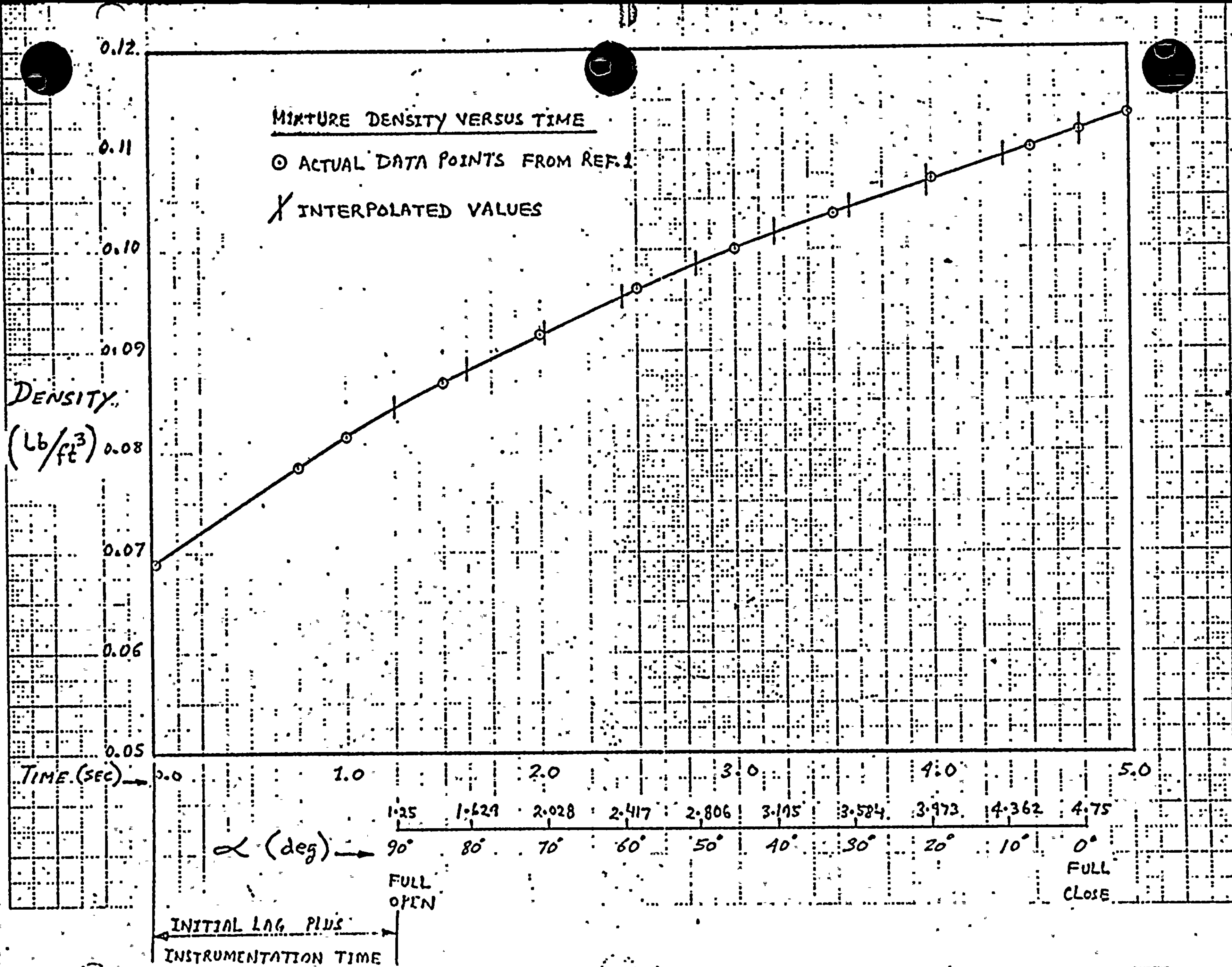
Using this time period we have abstracted the pressure, density, flowrate, and temperature response under the LOCA condition from Ebasco data (Reference 1). The enlarged plots for the period of interest are shown on pages 8 thru 11. The period of closure of the valve has been divided into nine equal divisions each of 0.389 second duration representing 10.00 degree of closure of the butterfly valve at a uniform rate. Using these divisions the interpolated values of pressure, temperature density and volumetric flow are extracted from the plots on pages 8, thru 11. These interpolated values are presented on page 12.



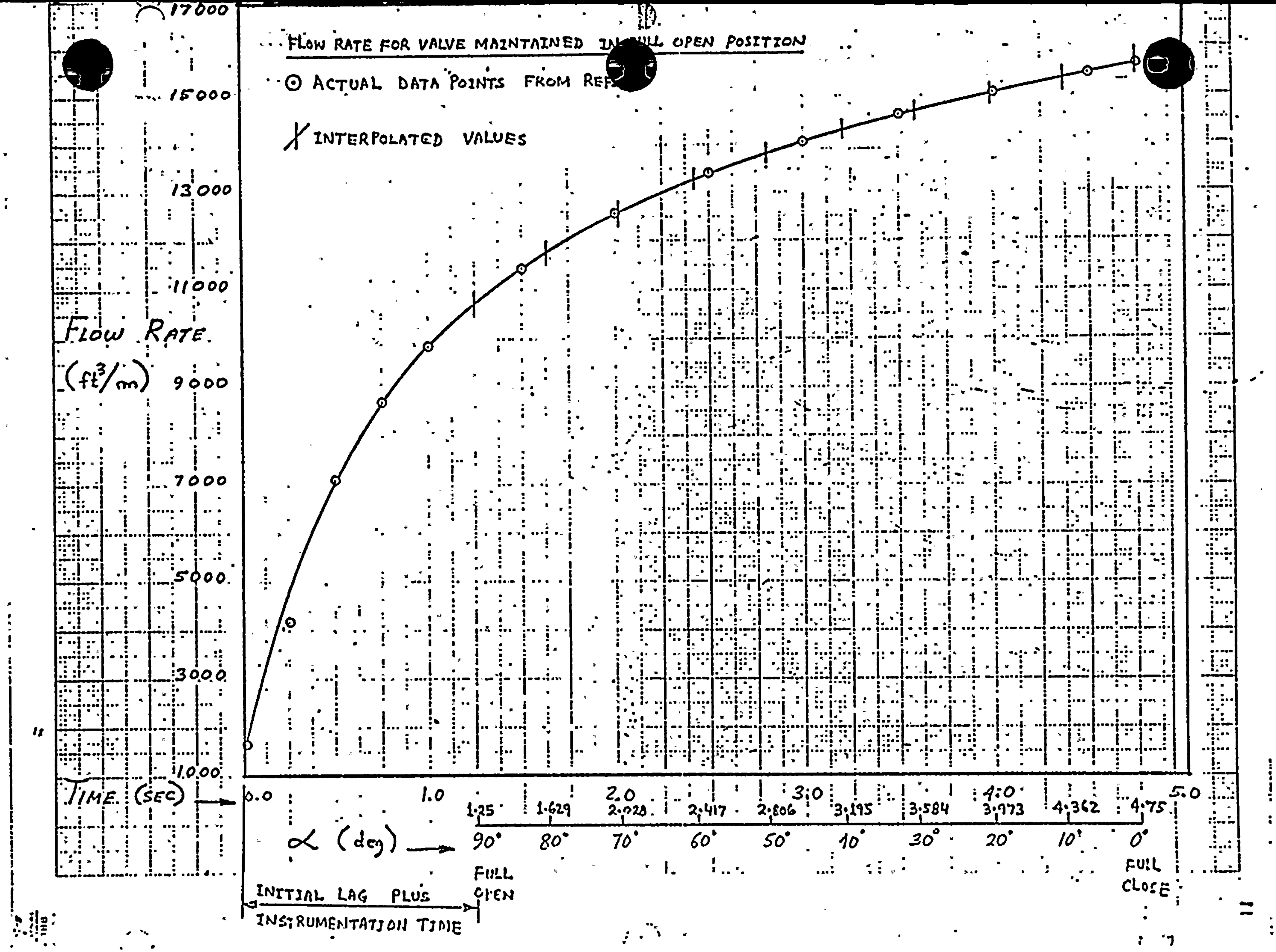
PRESSURE RESPONSE
ACTUAL DATA POINTS FROM REF. 1
INTERPOLATED VALUES
PRESSURE (psia)
TIME (sec)
0.0
1.0
2.0
3.0
4.0
5.0
1.250
1.629
2.028
2.417
2.806
3.195
3.584
3.973
4.362
4.750
α (deg)
90°
80°
70°
60°
50°
40°
30°
20°
10°
0°
FULL OPEN
FULL CLOSE
INITIAL LAG PLUS INSTRUMENTATION TIME
19
20
21
22
23
24
25
26
27
28
29
30
31
32
33
34
35



TEMPERATURE RESPONSE
ACTUAL DATA POINTS FROM REF. 1
INTERPOLATED VALUES
TEMPERATURE (°F)
230
210
190
170
150
130
110
90
70
TIME (SEC)
0.0
1.0
2.0
3.0
4.0
5.0
1.25
1.629
2.028
2.417
2.806
3.195
3.584
3.973
4.362
4.75
α (deg)
90°
80°
70°
60°
50°
40°
30°
20°
10°
0°
FULL OPEN
FULL CLOSE
INITIAL LAG PLUS INSTRUMENTATION TIME



MIXTURE DENSITY VERSUS TIME
ACTUAL DATA POINTS FROM REF. 1
INTERPOLATED VALUES
DENSITY (Lb/ft3)
0.12
0.11
0.10
0.09
0.08
0.07
0.06
0.05
TIME (SEC)
0.0
1.0
2.0
3.0
4.0
5.0
1.25
1.629
2.028
2.417
2.806
3.195
3.584
3.973
4.362
4.75
α (deg)
90°
80°
70°
60°
50°
40°
30°
20°
10°
0°
FULL OPEN
FULL CLOSE
INITIAL LAG PLUS INSTRUMENTATION TIME



FLOW RATE FOR VALVE MAINTAINED IN FULL OPEN POSITION
ACTUAL DATA POINTS FROM REF
INTERPOLATED VALUES
FLOW RATE (ft³/m)
17000
15000
13000
11000
9000
7000
5000
3000
1000
TIME (SEC)
0.0
1.0
2.0
3.0
4.0
5.0
1.25
1.629
2.028
2.417
2.806
3.195
3.584
3.973
4.362
4.75
α (deg)
90°
80°
70°
60°
50°
40°
30°
20°
10°
0°
FULL OPEN
FULL CLOSE
INITIAL LAG PLUS INSTRUMENTATION TIME

TABLE - 2

Time (Sec.)	Angle α (deg.)	Pressure P_1 (Psia)	Temp. T_1 (^oF)	Density Lb/ft^3	Full open flowrate Q_F (ft^3/min)
1.250	90 Full Open	21.65	170	0.0845	10700
1.629	80	23.52	179	0.0830	11780
2.028	70	25.25	187	0.0918	12600
2.417	60	26.85	195	0.0955	13300
2.806	50	28.25	202	0.0985	13800
3.195	40	29.63	207	0.1015	14300
3.584	30	30.85	210	0.1040	14700
3.973	20	32.10	214	0.1065	15000
4.362	10	33.28	217	0.1092	15350
4.750	0 Full Closed	34.35	220	0.1117	15627

Coefficient of flow K_v and the dynamic torque coefficient C_T for different angles of valve opening are obtained from the test report reference 7. BIF has conducted extensive tests on different types of disc geometry and and shaft orientation with respect to the direction of flow which are summarized in reference 6 and 7. The test medium is water and no air test is undertaken. Reference 6 is for two types of discs; namely, cast iron streamline disc and fabricated flat plate disc. Measurements have been made for dynamic torque coefficient and flow coefficient for both flatside upstream and flatside downstream of the disc. The comparison indicates that the disc orientation of flatside downstream always causes higher dynamic torque. Reference 7 incorporates a directly connected short radius elbow upstream to study the effect of flow non-uniformity on dynamic torque. Several tests have been performed with shaft vertical and shaft horizontal, counter clockwise opening and clockwise opening, with flatside upstream and flatside downstream. These test data are also compared with that of a straight pipe without any elbow upstream of the valve. A careful study of these experimental results indicate that the most severe case is a vertical shaft orientation (i.e. perpendicular to the plane of the elbow) with flatside of the disc downstream with a clockwise rotation of the disc.

This orientation results in approximately 30% increase in maximum dynamic torque coefficient than that would be obtained for a straight pipe. In this report, this most severe case is used to obtain torque coefficients at various angle of valve opening. This approach results in higher torque values and represents the worst condition. The test data are presented in the tabular form.

TABLE-3

Angle α Deg.	K_V	C_T
90	0.52	0.185
80	0.62	0.560
70	1.00	0.400
60	1.70	0.225
50	3.70	0.125
40	8.60	0.070
30	20.00	0.035
20	60.00	0.015
10	230.00	0.005
0.0	Closed	0.0

The volume flow rate through the valve is presented earlier. This is the flow rate for valve in fully open position. However, the valve is closing gradually and the flow rate should decrease accordingly and when the valve is fully shut the flow rate should reduce to zero. Therefore, we have to obtain the percentage of full open flow corresponding to the appropriate percentage of opening. References 3, 4, and 8 A provide such information. In reference 3, page 38, the flow characteristic of a butterfly valve is presented. This is a plot of percent of flow versus percent open which shows an equal percentage curve for the first 25% of flow a linear curve thereafter for the remaining 75% of flow. In reference 4, page 166, and reference 8 A, page 226, the flow characteristic of butterfly valve is shown to fall between the linear and equal percentage curve. Therefore, from these plots the fraction of maximum flow at a percentage opening can be determined. Before deciding whether to use linear or equal percentage curve some careful consideration has been given to determine which one should give the worst dynamic torque. Upon some reflection, it is observed from equation (1) that the dynamic torque increases when the pressure drop increases. It is also apparent from equation (3) that the pressure drop is greater when the flow rate is greater. This is achieved by using the linear curve which predicts higher flow than the equal percentage curve. Therefore, on the basis of this argument following flow rates are established for different degree of opening of the butterfly valve.

TABLE-4

Time (Sec.)	Full open flow Q_F (ft^3/M)	Angle α (deg.)	Percentage open (%)	Percentage flow Q_A $ft^3/min.$	$ft^3/Sec.$
1.250	10700	90 Full Open	100	10700	178.3
1.639	11780	80	89	10484	174.7
2.028	12600	70	78	9828	163.8
2.417	13300	60	67	8911	148.5
[illegible]	13800	50	56	7728	128.8
3.195	14300	40	44	6292	104.9
3.584	14700	30	33	4851	80.9
3.973	15000	20	22	3300	55.0
4.362	15350	10	11	1689	28.2
4.750	15627	0.0 Full Close	0	0.0	0.0

When the valve shuts off completely, the flow through the valve ceases therefore the dynamic torque vanishes. In this position, the differential pressure across the valve disc is the containment absolute pressure minus the atmospheric pressure. This is equal to the gage pressure inside the containment. Thus the necessary torque to completely close the valve and maintain it in the fully-shut condition against the existing differential pressure is due to the sum of the shaft bearing friction torque and the rubber seat friction torque called the seating torque.

The shaft bearing friction torque is presented as equation 2 earlier. The seating torque is given by

$$T_s = C_S D^2 \quad \text{(Ref. 2)} \;.......... \quad (5)$$

Where

T_s = Seating or unseating torque (in-lb)

C_S = Coefficient of seating or unseating torque (Ref. 5)

D = Valve port diameter (inch)

With all data available, the necessary calculation is performed using equation [illegible] through (5). Dynamic torque is calcualted for each angular postion to determine its maximum value and at what angle it occurs. For nine equal divisions of closing period representing 10° each (90° to 0°) altogether 10 sets of calculation are to be made. For this repetitive type of work, a computer program is written following the methodology described earlier. In order to validate the computer program, hand calculation of several test cases are performed in the beginning. Subsequently, the computer results are presented including the input and output. Comparisons between manual calculation and computer results show full agreement and therefore verifies validity of the computer program.

SAMPLE CALCULATION

VALVE SIZE: 8 Inch

Medium: AIR-STEAM MIXTURE

Valve opening angle of 90 degree occurring at 1.25 second

Inlet pressure from pressure curve = 6.95 + 14.7 = 21.65 psia.

Inlet temperature from temperature curve = 170 + 460 = 630 °R

Density from the density curve = 0.0845 lb/ft³.

Full open volume flow rate from flowrate curve = 178.3 ft³/s

Percentage flow at percentage opening = (178.3) 1.00 = 178.3 ft³/s

Flow rate in SCFH $Q_s = (0.642)10^6 \left[\frac{520(21.65)}{14.7(630)}\right] = 780292$ ft³/hr.

Valve coefficient $C_v = \frac{29.9 D^2}{\sqrt{K_v}} = \frac{29.9(7.05)^2}{\sqrt{0.52}} = 2061$; $K_v = 0.52$ (Ref. 7)

Specific gravity $G = \frac{0.0845}{0.0766} = 1.103$ based on air wieght density at 60°F and 1 atm. pressure.

Downstream pressure $p_2 = \sqrt{21.65^2 - \left[\frac{(0.7803)10^6}{963(2.061)10^3}\right]^2 (1.103)(630)} = 19.008$ psia

Therefore pressure drop $\Delta p = p_1 - p_2 = 2.642$ psi

Dynamic torque $T_D = C_T \Delta p \; d^3 = 171.3$ in-lb $C_T = 0.125$ (Ref. 7 elbow effect plus most adverse shaft orientation and disc rotation)

The shaft friction torque $T_b = \frac{\pi}{4}(7.05)^2 \left[0.25 \left(\frac{2.575}{2}\right)(2.642)\right]$

$= 11.28$ in-lb (1)

Therefore the net unbalanced torque is $T_N = T_D - T_b =$ 160.02 in-lb

This is a set of calculation for one valve angle.
Similar calcualations are performed for different angles and presented in subsequent pages.

:(1) The shaft friction torque is negligibly small. Therefore, no further calculation of this torque would be made. Since this is subtracted from the dynamic torque to obtain the net torque at any angular position, this approach is conservative.

SAMPLE CALCULATION

VALVE SIZE: 8 Inch

Medium: AIR - STEAM MIXTURE

Valve opening angle of 80 degree occurring at 1.639 second

Inlet pressure from pressure curve = 8.82 + 14.7 = 23.52 psia

Inlet temperature from temperature curve = 179 + 460 = 639 °R

Density from the density curve = 0.0880 lb/ft3

Full open volume flow rate from flowrate curve = 196.3 ft³/s

Percentage flow at percentage opening = (196.3) 0.89 = 174.7 ft³/s

Flow rate in SCFH $Q_s = (0.629)10^6 \left[\frac{520(23.52)}{14.7(639)}\right] = 819000$ ft³/hr

Valve coefficient $C_v = \frac{29.9D^2}{\sqrt{K_v}} = \frac{29.9(7.05)^2}{\sqrt{0.62}} = 1887.4$ $K_v = 0.62$ (Ref.7)

Specific gravity $G = \frac{0.0880}{0.0766} = 1.149$ based on air wieght density at 60°F and 1 atm. pressure.

Downstream pressure $p_2 = \sqrt{23.52^2 - \left[\frac{(0.819)10^6}{963(1.8874)10^3}\right]^2 (1.149)(639)} = 20.10$ psia

Therefore pressure drop $\Delta p = p_1 - p_2 = 3.42$ psi

Dynamic torque $T_D = C_T \Delta p \ d^3 = 671$ in-lb $C_T = 0.56$ (Ref. 7 elbow effect plus most adverse shaft orientation and disc rotation)

SAMPLE CALCULATION

VALVE SIZE: 8 Inch

Medium: AIR- STEAM MIXTURE

Valve opening angle of 70 degree occurring at 2.028 second

Inlet pressure from pressure curve = 10.55 + 14.7 = 25.25 psia.

Inlet temperature from temperature curve = 187 + 460 = 647 °R

Density from the density curve = 0.0918 lb/ft3

Full open volume flow rate from flowrate curve = 210 ft^3/s

Percentage flow at percentage opening = (210) 0.78 = 163.8 ft^3/s

Flow rate in SCFH $Q_s = (0.5897)10^6\left[\dfrac{520(25.25)}{14.7(647)}\right] = 814066$ ft^3/hr

Valve coefficient $C_v = \dfrac{29.9D^2}{\sqrt{K_v}} = \dfrac{29.9(7.05)^2}{\sqrt{1.0}} = 1486$; $K_v = 1.0$ (Ref.7)

Specific gravity $G = \dfrac{0.0918}{0.0766} = 1.198$ based on air wieght density at 60°F and 1 atm. pressure.

Downstream pressure $p_2 = \sqrt{25.25^2 - \left[\dfrac{(0.814066)10^6}{963(1.486)10^3}\right]^2 (1.198)(647)} = 19.664$ psia

Therefore pressure drop $\Delta p = P_1 - P_2 = 5.586$ psi

Dynamic torque $T_D = C_T \Delta P \; d^3 = 782.9$ in-lb $C_T = 0.4$ (Ref. 7 elbow effect plus most adverse shaft orientation and disc rotation)

SAMPLE CALCULATION

Valve Size: 8 Inch

Medium: Air-Steam Mixture

Valve opening angle of 50 degree occurring at 2.806 second

Inlet pressure from pressure curve = 13.55 + 14.7 = 28.25 psia

Inlet temperature from temperature curve = 202 + 460 = 662 ^{o}R

Density from the density curve = 0.0985 Lb/ft^3

Full open volume flow rate from flowrate curve = 230 ft^3/S

Percentage flow at percentage opening = (230) 0.56 = 128.8 ft^3/S

$$\text{Flow rate in SCFH } Q_S = (0.4637)10^6 \left[\frac{520(28.25)}{14.7(662)}\right] = 699946 \; ft^3/hr$$

$$\text{Valve coefficient } C_V = \frac{29.9D^2}{\sqrt{K_V}} = \frac{29.9(7.05)^2}{\sqrt{3.7}} = 772.6 \quad [K_V = 3.7 \text{ (Ref.7)}]$$

$$\text{Specific gravity } G = \frac{0.0985}{0.0766} = 1.286$$ based on air weight density at 60^{o}F and 1 atm. pressure.

$$\text{Downstream pressure } P_2 = \sqrt{28.25^2 - \underbrace{\left[\frac{(0.69995)10^6}{9.63(0.7726)10^3}\right]^2 (1.286)(662)}_{\text{TERM-A}}} = 6.68 \text{ psia}$$

However a downstream pressure P_2 = 6.68 psia is physically not possible since the lowest possible downstream pressure should be at least atmospheric i.e 14.7 psia. As a matter of fact the downstream pressure of the valve should be more than 14.7 psia since there is piping and exit losses occurring in the system after the valve and the gaseous mixture eventually exhausts into the atmosphere.

Therefore the Term A, shown above, which is proportional to the valve pressure loss is becoming too large to give this non-physical result.

Term A is larger than a practically possible value because either the flow rate Q or the temperature T_1 or the specific gravity G being used here are larger than their actual values. As a result of which, what this is essentially indicating is that, to maintain a flow rate of 699946 ft^3/hr through the valve the pressure must drop from 28.25 psia at the valve inlet to 6.68 psia at the valve outlet. But physically the maximum possible pressure drop under the worst situation is from 28.25 psia at the inlet to 14.7 psia at the outlet.

Using this maximum available pressure drop we can, in fact, calculate the maximum flow possible through the valve.

P_1 = 28.25 psia $\quad$ P_2 = 14.7 psia (Lowest possible)

$$Q_S^2 = \left[28.25^2 - 14.7^2\right]\left[\frac{\{(963)(772.6)\}^2}{(1.286)(662)}\right] = 3.784 \times 10^{11}$$

Therefore Q_S = 615152.43 ft^3/hr (maximum possible)

Versus 699946 ft^3/hr

Hence the maximum pressure drop, taking downstream pressure of 14.7 psia is

$$\Delta p_{max} = 28.25 - 14.7 = 13.55 \text{ psi}$$

Maximum possible dynamic torque $T_D = C_T \Delta p_{max} \, d^3$ = 593.5 in-lb $\quad C_T$ = 0.125 (Ref.7 elbow effect plus most adverse shaft orientation and disc rotation)

SAMPLE CALCULATION

Valve Size: 8 Inch

Medium: Air-Steam Mixture

Valve opening angle of 40 degree occurring at 3.195 second

Inlet pressure from pressure curve = 14.93 + 14.7 = 29.63 psia

Inlet temperature from temperature curve = 207 + 460 = 667 ^{o}R

Density from the density curve = 0.1015 Lb/ft^3

Full open volume flow rate from flowrate curve = 238.3 ft^3/S

Percentage flow at percentage opening = (238.3) 0.44 = 104.9 ft^3/S

$$\text{Flow rate in SCFH } Q_s = (0.37752)10^6 \left[\frac{520\ (29.63)}{14.7\ (667)}\right] = 593430 \ ft^3/hr$$

$$\text{Valve coefficient } C_v = \frac{29.9D^2}{\sqrt{K_v}} = \frac{29.9(7.05)^2}{\sqrt{8.6}} = 506.8 \quad \left[K_v = 8.6 \quad (\text{Ref.7})\right]$$

$$\text{Specific gravity } G = \frac{0.1015}{0.0766} = 1.325$$ based on air weight density at $60^{o}F$ and 1 atm. pressure.

$$\text{Downstream pressure } p_2 = \sqrt{29.63^2 - \underbrace{\left[\frac{(0.59343)10^6}{963(0.5068)10^3}\right]^2(1.325)(667)}_{\text{TERM-A}}}$$

In the above expression $P_1^2 = 29.63^2 = 878$

Term A = 1307 ; Therefore Term A $>$ P_1^2

This means that the flow rate is too high as explained earlier in the last page.

Calcualte the maximum possible flow for comparison.

P_1 = 29.63 psia; P_2 = 14.7 psia (lowest possible).

$$Q_s^2 = \left[29.63^2 - 14.7^2\right]\left[\frac{\{(963)(507)\}^2}{(1.325)(667)}\right] = 1.785 \times 10^{11}$$

Therefore Q_s = 422515 ft^3/hr (maximum possible)

Versus 593430 ft^3/hr

[illegible]ce the maximum pressure drop, taking downstream pressure of 14.7 psia is

$\Delta p_{max.}$ = 29.63 - 14.7 = 14.93 psi

Maximum possible dynamic torque $T_D = C_T \Delta p_{max.} d^3 = 366$ in-Lb $C_T = 0.07$ (Ref.6 elbow effect plus most adverse shaft orientation and disc rotation)

It should be noted that the calculations presented on pages 22 thru 24 represent the upper limit of the valve pressure drops and the resulting dynamic torques. In reality, the downstream pressure of the valve should be much greater than 14.7 psia. Precise information on the density, temperature, and the flow rate of the mixture with respect to time is necessary to determine the actual downstream pressure. However, the dynamic torque values, obtained with the present conservative approach, are much smaller than the operator torque capability. Moreover, the valve design torque in sizing the shaft, key, taper pin, etc. is much higher than the calculated dynamic torque. Due to this reason, no further refinement of the analysis is necessary.

VALVE 9:43 WED 25 MAY 83 26

ENTER THE VALVE DIAMETER IN INCHES
?7.05

ENTER THE NUMBER OF DATA SETS
?10

FOR EACH DATA SET ENTER THE FOLLOWING DATA IN ITS
RESPECTIVE ORDER SEPARATED BY A COMMA OR A BLANK.

A) UPSTREAM PRESSURE IN PSIG (P)
B) UPSTREAM TEMPERATURE IN DEG. F (T)
C) DENSITY IN LB/FT**3 (RO)
D) ACTUAL FLOW RATE IN FT**3/SEC (QA)
E) LOSS COEFFICIENT (KV)
F) TORQUE COEFFICIENT (CT)

ENTER DATA FOR SET NO. 1
?6.95 170 .0845 176.3 .52 .185

ENTER DATA FOR SET NO. 2
?8.82 179 .0880 14_74.7 .62 .56

ENTER DATA FOR SET NO. 3
?10.55 187 .0916 163.8 1 .4

ENTER DATA FOR SET NO. 4
?12.15 195 .0955 148.5 1.7 .225

INPUT DATA

ENTER DATA FOR SET NO. 5
?13.55 2021_ .0985 128.8 3.7 .125

ENTER DATA FOR SET NO. 6
?14.95 207 .1015 104.9 8.6 .07

ENTER DATA FOR SET NO. 7
?16.15 210 .1040 80.9 20 .035

ENTER DATA FOR SET NO. 8
?17.4 214 .1065 55 60 .015

ENTER DATA FOR SET NO. 9
?18.58 217 .1092 28.2 230 .005

ENTER DATA FOR SET NO. 10
?19.65 220 .1117 0.0 CLOSED 0.0

THE INPUT IS AS FOLLOWS: (ECHOING BACK THE INPUT) INPUT DATA VERIFICATION

SET NO.	P PSIG	T DEG. F	RO LB/FT**3	QA FT**3/SEC	KV	C_T
1	6.9	170.0	0.0845	178.3	0.52	0.185
2	8.8	179.0	0.0880	174.7	0.62	0.560
3	10.6	187.0	0.0918	163.8	1.00	0.400
4	12.1	195.0	0.0955	148.5	1.70	0.225
5	13.6	202.0	0.0985	128.8	3.70	0.125
6	14.9	207.0	0.1015	104.9	8.60	0.070
7	16.1	210.0	0.1040	80.9	20.00	0.035
8	17.4	214.0	0.1065	55.0	60.00	0.015
9	18.6	217.0	0.1092	28.2	230.00	0.005
10	19.6	220.0	0.1117	0.0	CLOSED	0.0

D YOU WISH TO MAKE ANY CHANGES?
NO

CALCULATION AT ANGLE = 90 DEG. OCCURING AT TIME = 1.25

VALVE DIAMETER	DIA =	7.050 INCHES
ABSOLUTE UPSTREAM PRESSURE	P1 =	21.6 PSIA
ABSOLUTE UPSTREAM TEMPERATURE	T1 =	630.0 DEG. R
FLOW RATE IN SCFH	QS =	780292. FT**3/HR
VALVE COEFFICIENT	CV =	2060.9
SPECIFIC GRAVITY	G =	1.103
CALCULATED DOWNSTREAM PRESSURE	P2 =	19.0 PSIA
PRESSURE DROP ACCROSS THE VALVE	DP =	2.642 PSI
DYNAMIC TORQUE	TD =	171. LB-IN

CALCULATION AT ANGLE = 80 DEG. OCCURING AT TIME = 1.639

VALVE DIAMETER .... DIA = 7.050 INCHES

ABSOLUTE UPSTREAM PRESSURE .... P1 = 23.5 PSIA

ABSOLUTE UPSTREAM TEMPERATURE .... T1 = 639.0 DEG. R

FLOW RATE IN SCFH .... QS = 818675. FT**3/HR

VALVE COEFFICIENT .... CV = 1867.4

SPECIFIC GRAVITY .... G = 1.149

CALCULATED DOWNSTREAM PRESSURE .... P2 = 20.1 PSIA

PRESSURE DROP ACCROSS THE VALVE ...... DP = 3.416 PSI

DYNAMIC TORQUE .... TD = 670. LB-IN

CALCULATION AT ANGLE = 70 DEG. OCCURING AT TIME = 2.028

VALVE DIAMETER .... DIA = 7.000 INCHES

ABSOLUTE UPSTREAM PRESSURE .... P1 = 25.3 PSIA

ABSOLUTE UPSTREAM TEMPERATURE .... T1 = 647.0 DEG. R

FLOW RATE IN SCFM .... QS = 614065. FT**3/HR

VALVE COEFFICIENT .... CV = 1486.1

SPECIFIC GRAVITY .... G = 1.198

CALCULATED DOWNSTREAM PRESSURE .... P2 = 19.7 PSIA

PRESSURE DROP ACCROSS THE VALVE ...... DP = 5.586 PSI

DYNAMIC TORQUE .... TD = 783. LB-IN

CALCULATION AT ANGLE = 60 DEG. OCCURING AT TIME = 2.417

VALVE DIAMETER .... DIA = 7.000 INCHES

ABSOLUTE UPSTREAM PRESSURE .... P1 = 26.8 PSIA

ABSOLUTE UPSTREAM TEMPERATURE .... T1 = 600.0 DEG. R

FLOW RATE IN SCFH .... QS = 779207. FT**3/HR

VALVE COEFFICIENT .... CV = 1139.8

SPECIFIC GRAVITY .... G = 1.247

CALCULATED DOWNSTREAM PRESSURE .... P2 = 17.7 PSIA

PRESSURE DROP ACCROSS THE VALVE ...... DP = 9.142 PSI

DYNAMIC TORQUE .... TD = 721. LB-IN

CALCULATION AT ANGLE = 50, DEG. OCCURING AT TIME = 2.806

VALVE DIAMETER .... DIA = 7.000 INCHES

ABSOLUTE UPSTREAM PRESSURE .... P1 = 20.3 PSIA

ABSOLUTE UPSTREAM TEMPERATURE .... T1 = 662.0 DEG. R

FLOW RATE IN SCFH .... QS = 699946. FT**3/HR

VALVE COEFFICIENT .... CV = 772.6

SPECIFIC GRAVITY .... G = 1.286

CALCULATED DOWNSTREAM PRESSURE .... P2 = 6.7 PSIA

WARNING: DOWNSTREAM PRESSURE IS LESS THAN ATMOSPHERIC PRESSURE.
TAKING DOWNSTREAM PRESSURE P2 = 14.7 PSIA

MAXIMUN FLOW RATE .... QSMAX = 610167. FT**3/

PRESSURE DROP ACCROSS THE VALVE ...... DP = 13.000 PSI

DYNAMIC TORQUE .... TD = 093. LB-IN

```
CALCULATION AT ANGLE =    40    DEG.    OCCURING AT TIME =  3.19

VALVE DIAMETER ...................... DIA =  7.050 INCHES

ABSOLUTE UPSTREAM PRESSURE ........... P1 =   29.6   PSIA

ABSOLUTE UPSTREAM TEMPERATURE ........ T1 =  667.0   DEG. R

FLOW RATE IN SCFH .................... QS =    593430.   FT**3/HR

VALVE COEFFICIENT .................... CV =    506.8

SPECIFIC GRAVITY ..................... G =   1.320

WARNING: NEGATIVE ARGUMENT ... SQRT(     677.9 -    1306.9 )
TAKING DOWNSTREAM PRESSURE  P2 = 14.7 PSIA

MAXIMUN FLOW RATE .................... QSMAX =    422302.   FT**3/

PRESSURE DROP ACCROSS THE VALVE ...... DP =  14.930   PSI

DYNAMIC TORGUE ....................... TD =    366.   LB-IN
```

CALCULATION AT ANGLE = 30 DEG. OCCURING AT TIME = 3.583

VALVE DIAMETER .... DIA = 7.000 INCHES

ABSOLUTE UPSTREAM PRESSURE .... P1 = 30.8 PSIA

ABSOLUTE UPSTREAM TEMPERATURE .... T1 = 670.0 DEG. R

FLOW RATE IN SCFH .... QS = 474370. FT**3/HR

VALVE COEFFICIENT .... CV = 332.3

SPECIFIC GRAVITY .... G = 1.308

WARNING: NEGATIVE ARGUMENT ... SQRT(931.7 - 1998.9)
TAKING DOWNSTREAM PRESSURE P2 = 14.7 PSIA

MAXIMUN FLOW RATE .... QSMAX = 287774. FT**3/HR

PRESSURE DROP ACCROSS THE VALVE ...... DP = 16.100 PSI

DYNAMIC TORQUE .... TD = 198. LB-IN

CALCULATION AT ANGLE = 20 DEG. OCCURING AT TIME = 3.972

VALVE DIAMETER .... DIA = 7.000 INCHES

ABSOLUTE UPSTREAM PRESSURE .... P1 = 32.1 PSIA

ABSOLUTE UPSTREAM TEMPERATURE .... T1 = 674.0 DEG. R

FLOW RATE IN SCFH .... QS = 333077. FT**3/HR

VALVE COEFFICIENT .... CV = 191.9

SPECIFIC GRAVITY .... G = 1.390

WARNING: NEGATIVE ARGUMENT ... SQRT(1030.4 - 3034.7)
TAKING DOWNSTREAM PRESSURE . P2 = 14.7 PSIA

MAXIMUN FLOW RATE .... QSMAX = 172229. FT**3/HR

PRESSURE DROP ACCROSS THE VALVE ...... DP = 17.400 PSI

DYNAMIC TORQUE .... TD = 91. LB-IN

CALCULATION AT ANGLE = 10 DEG. OCCURING AT TIME = 4.361

VALVE DIAMETER .... DIA = 7.030 INCHES

ABSOLUTE UPSTREAM PRESSURE .... P1 = 33.3 PSIA

ABSOLUTE UPSTREAM TEMPERATURE .... T1 = 677.0 DEG. R

FLOW RATE IN SCFH .... QS = 176336. FT**3/HR

VALVE COEFFICIENT .... CV = 98.0

SPECIFIC GRAVITY .... G = 1.426

WARNING: NEGATIVE ARGUMENT ... SQRT(1107.6 - 3377.7)
TAKING DOWNSTREAM PRESSURE P2 = 14.7 PSIA

MAXIMUN FLOW RATE .... QSMAX = 90693. FT**3/HR

PRESSURE DROP ACCROSS THE VALVE .... DP = 18.580 PSI

DYNAMIC TORQUE .... TD = 33. LB-IN

lve in full closed position. Angle $\alpha = 0^\circ$

Upstream pressure = 19.65 + 14.7 = 34.35 psia at 4.75 Sec.

Downstream pressure = Atmospheric = 14.7 psia, valve fully shut; downstream is explosed to atmosphere; no flow in the pipeline.

Differential pressure $\Delta p = 34.35 - 14.7 = 19.65$ psi

Flow rate is zero since the valve is fully closed. Therefore the dynamic torque is zero.

Friction torque at the shaft bearing is

$$T_b = \frac{\pi}{8} (D^2) \cdot (f_b \cdot d) \, \Delta p \qquad \text{(Eqn. 2)}$$

$$= \frac{\pi}{8} (7.05)^2 (0.25)(0.875)(19.65) = 84 \text{ in-lb}$$

Valve seating torque due to rubber friction is

$$T_s = C_s D^2 = 25(7.05)^2 \qquad \text{(Eqn. 4)}$$

$$= 1243 \text{ in-lb}$$

Net torque $T_N = T_b + T_s = 1327$ in-lb

Actual valve torque used in selecting the operator and seismic analysis report is 1648 in-lb. This is based upon a higher differential pressure. Since this torque is greater than the static torque calculated above, this is more conservative and we adopt this as our final value.

Therefore, the net torque for static condition is $T_{net} = 1648$ in-lb.

RESVISION (A) TO DYNAMIC TORQUE REPORT

NO. DT-67926

Prepared in response to EBASCO's comments on the above report, outlined in their letter dated 8/22/83.

EBASCO COMMENT #1

A point by point response to Items 1.A,B,C,D,E,1,2,3,4,5,6,F,3,4 & 5 of the operability qualification of purge and vent valves and Item 1 thru 8 of Guidelines for Demonstration of Operability of Purge and vent lines are required by EBASCO.

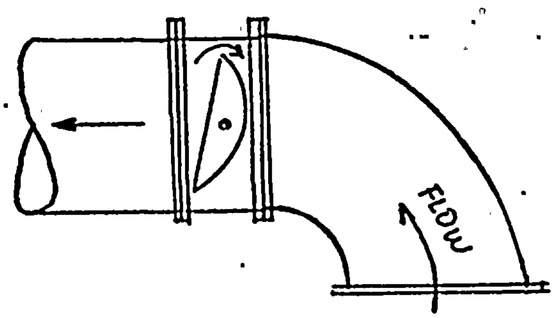
A. Dynamic Torque Coefficient Test Reports are attached. The report gives the description of the test setup.

B. Not used

C. Stress report (included in the Seismic Report)

D. Seismic Reports for valve assembly and associated parts are attached.

E. Description of each valve with shaft and disc orientation and the orientation of the fittings in the whole piping system is not known to BIF, since the piping design is beyond the scope of BIF. Therefore, we present below the description of the valve installation that BIF has tested to determine the dynamic torque coefficients. The dynamic torque coefficients obtained with a Short radius 90° elbow directly upstream of the valve is higher than that obtained with straight pipe configuration and is assumed to represent the worst condition in the evaluation of dynamic torque.



1. Direction of flow is from the elbow towards the valve.
2. Disc closer direction is clockwise rotation
3. Curved side of the disc is upstream
4. The elbow is directly attached to the upstream of the valve. The elbow is a 90^{0} short radius elbow.
5. Shaft orientation is vertical. Elbow is in horizontal plane.
6. No other valve present

   Combinations 1 thru 6 results in highest C_T. See BIF Test Report TR-0650-43.

F. Under the response to EBASCO comment #2 it is shown that the maximum torque developed by the valve is below the actuator rating.

3. Analysis is supported by tests. The torque coefficients have been determined, considering the angle of closure, flow direction, non-symmetric flow from an elbow upstream of valve. See two BIF test reports attached.
4. No In-situ test performed
5. BIF has used 3.5 second as the maximum closing time, which is the actual closing time of the valve. In addition to that 1.25 second of instrumentation time is used which lets the containment pressure build up. As higher pressure means higher dynamic torque the analysis is conservative.

Item 1 thru 8 of the Guidelines of Demonstration of Operability.

Operability

1. A constant rate of valve closure is used.
2. Flow direction is from the 90° elbow to the valve. The elbow creates the flow non-uniformity that increases the dynamic torque. Flow impinges on the curved side of the disc. Δp values are given in this report at different angular positions.
3. Single valve closure. The valve inside containment is exposed to the containment pressure rise given in EBASCO letter dated Aug. 3, 1982. The calculation is based on this pressure on the valve inside containment and is assumed to be the worst case.
4. Containment back pressure will not affect the closing torque of spring to close air operator. The cylinder piston will have inside containment pressure at both sides - thus it will balance off, then the spring force will close the valve with its full force.
5. No accumulator used.
6. No torque limiting device is used for the operator.
7. The piping upstream is considered to be an elbow and downstream is straight pipe.
8. Effect of the disc and shaft orientation has been taken into account during the test. See attached test reports.

EBASCO Comment #2

In response to the inconsistency pointed out by EBASCO for 60^o to 90^o position the following calculations are presented which are identical for 10^o to 50^o positions and therefore are consistent with each other.

Between 60^o to 90^o the torque coefficients peaks at 80^o. If the downstream pressure is assumed to be atmospheric, as done for 10^o to 50^o, the the dynamic torque at 80^o will be maximum and those at 60^o, 70^o, and 90^o will be less than this. Therefore it is only necessary to calculate the torque at 80^o opening.

Valve opening of 80^o occurs at 1.639 second

Upstream pressure of the valve from pressure curve = 23.52 psia

Minimum possible downstream pressure of the valve = 14.7 psia

Maximum pressure differential p_{max} = 23.52 - 14.7 = 8.82 psi

Maximum dynamic torque $T_{D\,max} = C_{T\,max} \quad p_{max} \; d^3$

$= 0.56(8.82)(7.05)^3 = \underline{1731 \text{ in-lb}}$

Actual torque capacity of the actuator given by the manufacturer Bettis company is 2570 in-lb. The actuator torque, being greater than the macimum dynamic torque is adequate to close the valve in the event of a LOCA.

In the original seismic analysis report the static torque value used in calculating the stresses in various components was 1648 in-lb. Under the dynamic condition the torque increased to 1731 in-lb, which is a 5% increase on the previous value. The effect of this increase has been taken in to account by revising the original seismic report with the maximum torque value. This report marked N-67926 Rev.A is attached to this package.

BIF had stated in its quotation that, the flow data for the valve under LOCA condition would be supplied by EBASCO and BIF would perform the dynamic torque analysis using EBASCO supplied flow data. Therefore, the flow calculation under Comment #2 of EBASCO letter dated 8/22/83 is beyond the scope of this contract.

EBASCO Comment #3

References 2 thru 4, and 8 are open literatures, e.g. journal and books. References 5 thru 7 are BIF reports which are attached with this package.

EBASCO Comment #4

Equation (4) is the conversion of actual flow to flow under standard condition of 60°F and 1 atmospheric pressure.

P_1, T_1, and Q_A are actual pressure, temperature, and flow rate of the valve inlet condition. P_S = 14.7 psia, T_S = 520°R, and Q_S are the same parameters at the standard condition.

For further clarification, refer to, Crane Technical paper No. 410, 1981 printing, page 4-9, Example 4-16.

EBASCO Comment #5

BIF to ascertain that the valve closure period is 3.5 seconds. All valves will be tested for speed of closure at BIF before shipment. Valves above 3.5 sec. will be rejected by BIF Q.C. However, BIF does not have any test data of speed of closure of spring loaded cylinder when it is bled into higher than atmospheric pressure. BIF's recommendation is to pipe the exhaust port of the solenoid valve to atmosphere to insure the 3.5 sec. closure period.

EBASCO Comment #6

On page 14, reference 10 should have been reference 8. This has been corrected in this revision.

EBASCO Comment #7

The maximum dynamic torque recalculated under EBASCO Comment #2 with a valve downstream pressure of 14.7 psia is shown to be 1731 in-lb. This is lower than the operator capacity of 2570 in-lb. The valve components are reanalyzed with the higher dynamic torque in seismic report N-67926 Rev. A and are shown to be safe.

APPENDIX

EBASCO SERVICES INCORPORATED

EBASCO

Two World Trade Center, New York, N.Y. 10048

HARRY GRUEN
839-3752

(C)

August 3, 1982
File:9Q-BE-35

Rec. 8/16/82

8-16-82

Mr Daniel P Cyronak Jr
BIF
1600 Division Road
West Warwick, Rhode Island 02893

cc: D SZILAGYI
P.L.O.F (NG7
(NG09

Dear Mr Cyronak:

SUBJECT: SHEARON HARRIS NUCLEAR POWER PLANT
PERFORMANCE OF 8" NORMAL CONTAINMENT
PURGE BUTTERFLY VALVE
CONTRACT NO. NY-435211 & 435212

REFERENCES:
1. NUREG-0737 Item II.E.4.2 - Staff Interim position of October 23, 1979
2. Table of flow through 8" containment purge butterfly valve for worst LOCA pressure transient case
3. NRC Question No. 480.40

In order to satisfy the Reference 1&3 requirements we must establish operability of the subject valve under the most severe design basis accident flow conditions.

We request your confirmation that the required valve torque during maximum flow conditions shown in Reference 2 will not exceed the capacity of the valve operator and that the valve will close and remain tightly closed.

The following is pertinent data you may need to perform the analysis:

1. The containment isolation signal which initiates valve closure is energized at 0.75 seconds into the LOCA when the containment pressure reaches 4.5 psig.

2. Processing time for the signal so that the solenoid valve loses power is 0.50 seconds.

3. The butterfly valve operability shall be demonstrated where the valve is 30° and 50° open taking no more than 3.5 seconds for the solenoid valve to bleed the instrumentation air and close the butterfly valve. (BIF is to indicate at which point during the closing cycle the highest dynamic torque will be experienced.)

NUCLEAR

Should you require more information don't hesitate to contact us.

Very truly yours,

J Berenberg
Supervising Engineer
Mechanical Engineering

By: W Bielawski/H Gruen

Enclosure
HG:am

cc: L I Loflin
L H Martin
J L Willis
R M Parsons

NUCLEAR

PURGE VALVE 2CP-B1SA

FOR WORST LOCA PRESSURE TRANSIENT CASE

minus 14.7 for PSIG

Time secs.	Containment pressure, psia	Temp °F	Mixture density lbm/ft³	Flow Through the purge line, cfm
0.0	14.8410		0.0690	1684.1
0.25	15.6067		0.0723	4171.9
0.50	17.449		0.0756	7104.0
0.75	19.0012		0.0786	8714.8
1.00	20.4054		0.0815	9856.9
1.5	22.9289		0.0869	11464.0
2.0	25.1647		0.0917	12585.0
2.5	27.1488		0.0960	13415.4
3.0	28.9469		0.0999	14068.6
3.5	30.6054		0.1035	14604.2
4.0	32.1655		0.1069	15058.3
4.5	33.6407		0.1101	15451.8
[illegible]	34.3538		0.1117	15626.9
[illegible]	35.0648		0.1133	0.0
6.0	37.4830		0.1185	0.0
7.0	40.0838		0.1242	0.0
8.0	42.6072		0.1297	0.0
9.0	44.5073		0.1338	0.0
10.0	45.9028		0.1368	0.0
11.0	47.3242		0.1399	0.0

LOCA DEHLG MIN SI

NUCLEAR

FLOW IS TYPICAL FOR VALVES:
2CP-B3SA-1
2CP-B5SA-1
2CP-B6SA-1
SEE SPEC CAR-SH-BE-35
PAGES 43, 44, 47 + 48 OF 59

DS#
5100 SCFM 43 = 2CP-B1SA - N67926
...0 SC... 44 = 2CP-B2SB - N67927
A... 47 = 2CP-B5SA - N67926
ACFM 48 = 2CP-B6SB - N67927

BETTIS ACT.
N721C-SR60-12
VALVE TORQUE 1770 in#
SIZING TORQUE 1805 in#

MAX END TORQUE
2570 in# • CATALOG
PER IEEE
+ 18 1/2 Los

NUREG-737

II.E.4.2. ATTACHMENT 1, OCTOBER 23, 1979* INTERIM POSITION FOR CONTAINMENT PURGE AND VENT VALVE OPERATION PENDING RESOLUTION OF ISOLATION VALVE OPERABILITY

Once the conditions listed below are met, restrictions on use of the containment purge and vent system isolation valves will be revised based on our review of your responses to the November 1978 letter on this subject justifying your proposed operational mode. The November 1978 letters to all licensees identified certain events related to containment purging of concern to the NRC and requested commitments to either cease purging or justify purging operations. The revised restrictions can be established separately for each system.

(1) Whenever the containment integrity is required, emphasis should be placed on operating the containment in a passive mode as much as possible and on limiting all purging and venting times to as low as achievable. To justify venting or purging, there must be an established need to improve working conditions to perform a safety-related surveillance or safety-related maintenance procedure. (Examples of improved working conditions would include deinerting, reducing temperature,** humidity, and airborne activity sufficiently to permit efficient performance or to significantly reduce occupational radiation exposures.)

(2) Maintain the containment purge and vent isolation valves closed whenever the reactor is not in the cold shutdown or refueling mode until such time as you can show that:

(a) All isolation valves greater than 3-in. nominal diameter used for containment purge and venting operations are operable under the most severe design-basis-accident (DBA) flow-condition loading and can close within the time limit stated in the technical specifications, design criteria, or operating procedures. The operability of butterfly valves may, on an interim basis, be demonstrated by limiting the valve to be no more than 30° to 50° open (90° being full open). The maximum opening shall be determined in consultation with the valve supplier. The valve opening must be such that the critical valve parts will not be damaged by DBA-LOCA (loss-of-coolant accident) loads and that the valve will tend to close when the fluid dynamic forces are introduced, and

(b) Modifications, as necessary, have been made to segregate the containment ventilation isolation signals to ensure that, as a minimum, at least one of the automatic safety injection actuation signals is uninhibited and operable to initiate valve closure when any other isolation signal may be blocked, reset, or overridden.

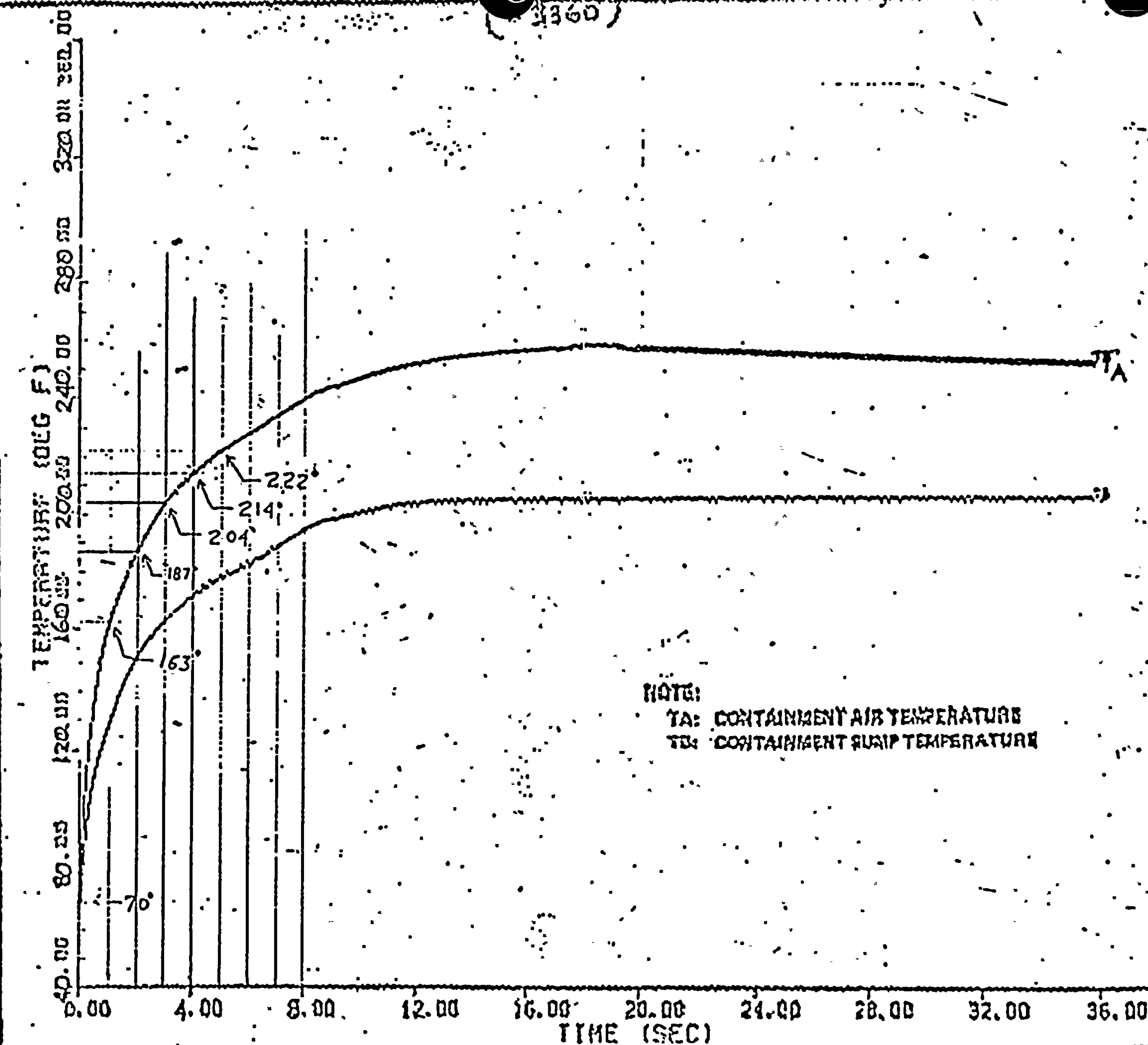
*Previously referred to as DOE Interim Position.

**Only when temperature and humidity controls are not in the present design.

NUCLEAR

USE THIS PLOT FOR CONT. TEMPS. DURING LOCA —
PER TELECON WITH T. DHIH (APPLIED PHYSICS)
(X3360)

TO: RICK REAPITO, BIF
FROM: Y. JAGANNATH, EAASCO



SHEARON HARRIS NUCLEAR POWER PLANT
Carolina Power & Light Company
FINAL SAFETY ANALYSIS REPORT

CONTAINMENT TEMPERATURE FOR MOST SEVERE HOT LEG BREAK (DEHLG)

FIGURE 6.2.1-1

EBASCO SERVICES INCORPORATED EBASCO

Two World Trade Center, New York, N.Y. 10048

August 22,1983
File.: 9Q-BE-35
Reply requested by: Sept 12,198[illegible]

Mr Dan Cyronak
BIF
1600 Division Road
West Warwick, Rhode Island 02893

Dear Mr Cyronak: 8-30-83

cc: R. Riccapita
LOF-T

SUBJECT: CAROLINA POWER & LIGHT COMPANY
SHEARON HARRIS NUCLEAR POWER PLANT
PO NY-435211 & 212
EBASCO SPEC CAR-SH-BE-35 REV. 7
OPERABILITY QUALIFICATION OF
CONTAINMENT PURGE VALVES -67930.U

REFERENCE:
1) Dynamic Torque Calculation of Butterfly Valves dated 6-30-83
2) Ebasco letter to BIF dated April 8,1983
3) Ebasco letter to BIF dated September 1,1982
4) Sketches SK-1,2 & 3 dated 8-16-83

We are disapproving the reference 1 report due to the following:

1) The report does not address the problem in its entirety. The reference 2 & 3 letters had enclosed sketches as well as NRC requirements that have to be met as regards the operability of the subject valves. We request a point by point response to Items 1.A,B,C,D,E,1,2,3,4,5,6,F,3,4&5 of the Operability Qualification of Purge and Vent Valves attachment (Item 2 will be responded to by the client) and Item 1 thru 8 of the Guidelines for Demonstration of Operability of Purge and Vent Lines attachment. These responses should be made part of the report.

2) The argument presented on page 22 thru 25 and pages 32 thru 36 points to an inconsistency in the entire approach to the report-for the 10^{o} to 50^{o} valve disc position, 14.7 PSIA (atmospheric) is taken as (P_2) the downstream pressure while for the 60^{o} to 90^{o} valve disc position 17.7,19.7,20.1 and 19.0 PSIA respectively is used for (P_2) as downstream pressure.

We calculate that when 14.7 PSIA (atmospheric) is used, as in the 10^{o} to 50^{o} position, with the 80^{o} valve disc position the dynamic torque (T_D) exceeds the design torque (1727 in-lb verus 1648 in-lb). (As a clarification please be advised that the flow data that were initially transmitted to you in the Aug 3,1982 letter was based on a fixed "K" of 2.92, for conservatism, with atomospheric pressure considered upstream of the valve.)

NUCLEAR

We are therefore enclosing (reference 4) sketches SK-1,2 & 3 dated 8-16-83 which depict valve verus duct arrangements for your use and suggest that you recalculate and take credit for the upstream and downstream ductwork pressure drops in order to reduce the flow rates to the extent that it will show not to produce undue forces on the valve disc. It should be noted that only seismic ductwork can be taken as credit for additional resistance.

3) Attach copies or excerpts of reference 2 thru 8 listed on page 4 of the report to report.

4) Equation (4) on page 6 is not referenced to a source. Provide reference.

5) Provide qualification that the valve will indeed close in 3.5 seconds from the receipt of signal to the solenoid valve to the full-close position.

6) Page 14 refers to reference 10 however is not listed on page 4 of the report.

7) Values should be quantified rather than using terms "much higher" and "much smaller" on page 25.

Should you have any comments or need more information, please advise.

Very truly yours,

J Berenberg
Supervising Engineer
Mechancial Engineering

W Bielawski H Gruen
By: W Bielawski/ H Gruen

HG:rob

cc: L I Loflin
L H Martin
J L Willis
R M Parsons

TR-0650-43

2-24-82

B I F

A UNIT OF GENERAL SIGNAL

TEST REPORT

HYDRODYNAMIC AND HEADLOSS TEST OF 12" -150B BUTTERFLY VALVE WITH DIRECTLY CONNECTED SHORT RADIUS ELBOW UPSTREAM.

TEST PERIOD: May & June, 1980

TEST ENGINEER: K. Kormos

APPROVED BY: D. Szilagyi D Szilagyi 2-25-82

Report prepared by: K. Kormos, January 20, 1982

mh

PURPOSE OF THE TEST

To establish Flow (K_V) and hydrodynamic torque (C_t) coefficient of a Butterfly Valve with directly connected short-radius elbow.

INTRODUCTION

The test was conducted in the B I F Hydraulic Lab. Test setup is shown on Attachment #1.

Prior to the eblow test the 12" pipe headloss without and with installed valve was measured in straight pipe (Attachment #2). The taps used to measure headloss at the elbow test were located at the same distance from the valve as at the straight pipe test. Plotted headloss curve: Attachment #2A.

The test valve shaft was provided with an adaptor for torque wrench, with a safety torque arm and a pointer for setting the valve disc to the desired test angle. The valve body- mounted heavy steel plate, was provided with holes for 5/8" bolts which were limiting the torque arm motion and was also used to fix the valve disc in any desired position. Valve and elbow dimensions: Attachment #2. Six elbow tests were performed as follows:

I. Vertical shaft: (Attachment #3) Attachment:

1.	Flat side of disc upstream, CCW Opening:	#5
2.	Flat side of disc upstream, CCW Opening:	#6
3.	Flat side of disc downstream, CCW Opening:	#7
4.	Flat side of disc downstream, CW Opening:	#8

For plotted data see Attachment #11, 12, 13 & 14

II. Horizontal shaft: (Attachment #4) Attachment:

1.	Flat side of disc upstream, CCW Opening:	#9
2.	Flat side of disc downstream, CCW Opening:	#10

For plotted data see attachment #15, 16, 17 & 18.

TEST PROCEDURE

Details of the operation of the Hydraulic Lab are not described here. The Lab provides the means to collect water in a scale mounted 50,000 lb. tank and to measure collecting time accurately. The accuracy of this flow rate measuring method is better than ±0.1%.

The flow rate thru the test valve was set by manipulating electrically driven valves. There were two limiting factors: the upstream pressure must not drop under 60 PSI (lower pressure would overload the pumps); and the maximum differential could not be higher than the range of the manometer (50" Mercury).

For every 5° setting of the disc from 10° position to 90°, the procedure was the same as follows:

1. The disc was set to the desired angle by wedging the safety torque-arm between the 5/8" bolts.

2. Flow was measured by collecting more than 45,000 pounds of water and reading the differential pressure on the manometer at least 15 times during the run.

3. From the test data (weight of water, running time and average of the means meter readings), the flow rate, velocity, valve headloss and flow coefficient were calculated.

4. The disc position was held by the torque wrench while the wedges were removed. The opening and closing torque was read in motion of the disc by turning the disc position a few degrees under and above the original setting and reading the torque wrench when the pointer passed the setting.

5. From torque wrench readings, the dynamic torque coefficient was calculated.

All test and calculated data were recorded on log sheets, and the coefficients also on graph paper.

EQUATION USED FOR CALCULATIONS

Differential Pressure thru Valve: P_v PSI

Differential Pressure thru Pipe: P_p INHG

Differential Pressure thru Valve & Pipe: P_s INHG

$$P_v = \frac{P_s - P_p}{2.204}$$

Flow rate: Q GPM

Weight of collected water :: W LB

Specific weight of water at water temp.: ρ_T LB/FT^3

Collecting time: t SEC.

$$Q = \frac{448.83\,W}{\rho_T\, t} = K_1 \frac{W}{t}$$

Fluid velocity: V FT/SEC

Port Area: A FT^2 A = 0.739 FT^2

$$V = \frac{Q}{448.83A}$$

Bearing torque: T_B INLB

Bearing Dia. d IN d = 1.5"

Disc area: A_D IN^2 A_D - 106.14 in^2

Friction factor: μ = 0.08 (Teflon per D.S.)

$$T_B = \mu A_D P_v \frac{d}{2}$$

Dynamic torque: T_D

Opening torque wrench reading: T_O

Closing torque wrench reading T_C

$$T_D = \frac{T_O + T_C}{2}$$

Dynamic torque coefficient: C_t

Disc diameter: D IN D = 11.625"

$$C_t = \frac{T_D}{P_V D^3}$$

Flow coefficient: K_V

Acceleration of gravity: g g - 32.17 FT/SEC^2

$$K_V = P_V \frac{144 \times 2g}{V^2 \rho_T} = K_2 \frac{P_V}{V^2}$$

Line Temp. °F	K_1	K_2	W_T
76	7.220	149.054	62.1582
77	7.221	149.076	62.1493
78	7.223	149.097	62.1405
79	7.223	149.118	62.1316
80	7.225	149.139	62.1228
81	7.226	149.163	62.1131
82	7.227	149.186	62.1034
83	7.228	149.209	62.0936
84	7.229	149.233	62.0839
85	7.231	149.256	62.0742
86	7.232	149.282	62.0636

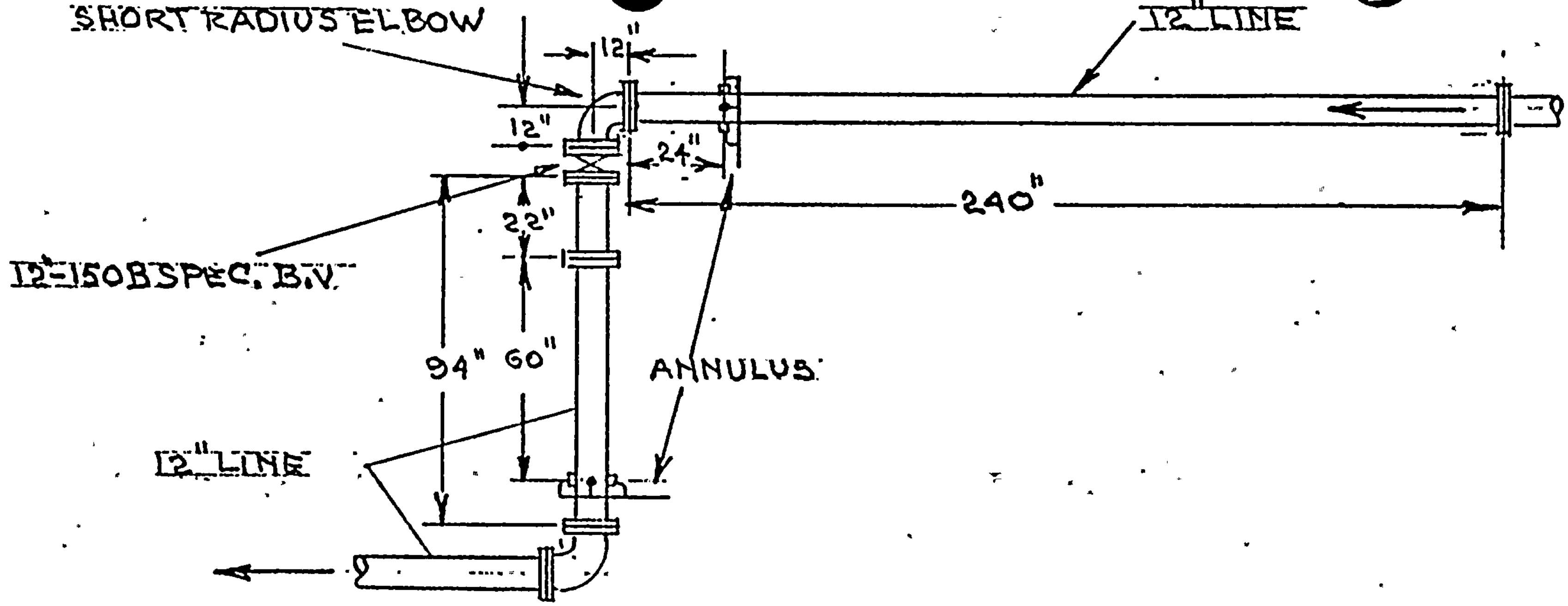
NOTES:

1. Torque readings on a calibrated hand held torque wrench is not better than ± 10% because it is impossible moving the disc for reading opening and closing torque at any disc position with the same moving speed, and reading it when the pointer passes the test position; the torque wrench pointer vibrates and there is also a parallax problem.

2. The here published test data should not be used for extrapolation if another disc shape is different from the tested one.

   Test in straight pipe with attached 8 x 12 increaser upstream of the valve points to shape effect. (see test report).

(not issued on 2-26-82)



TEST SETUP.

ELBOW UPSTREAM OF B.V.

ATTACHMENT II

FORM 1162A

BIF
A UNIT OF GENERAL SIGNAL

TEST SETUP
BIF HYDRAULIC LAB.

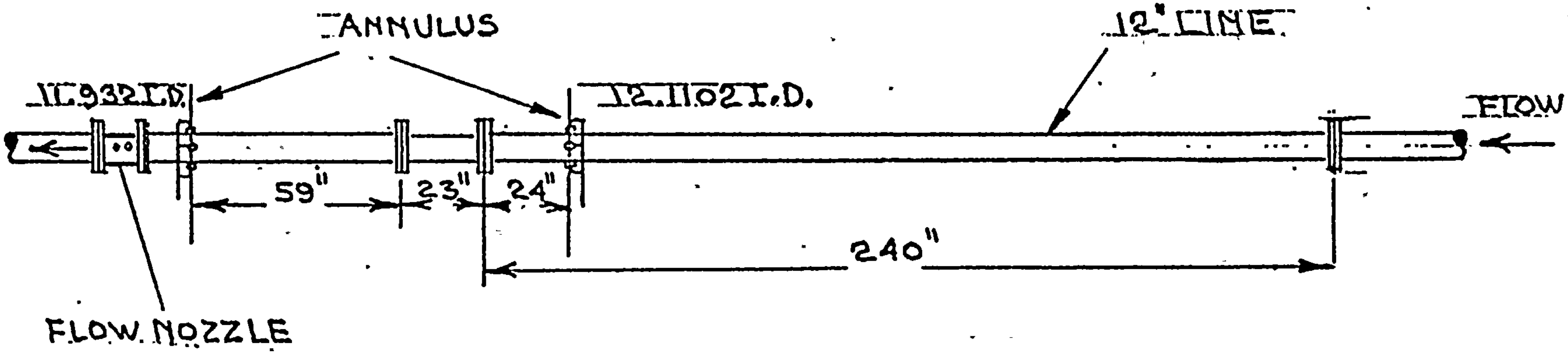
DR K.K.
DATE 6-17-80

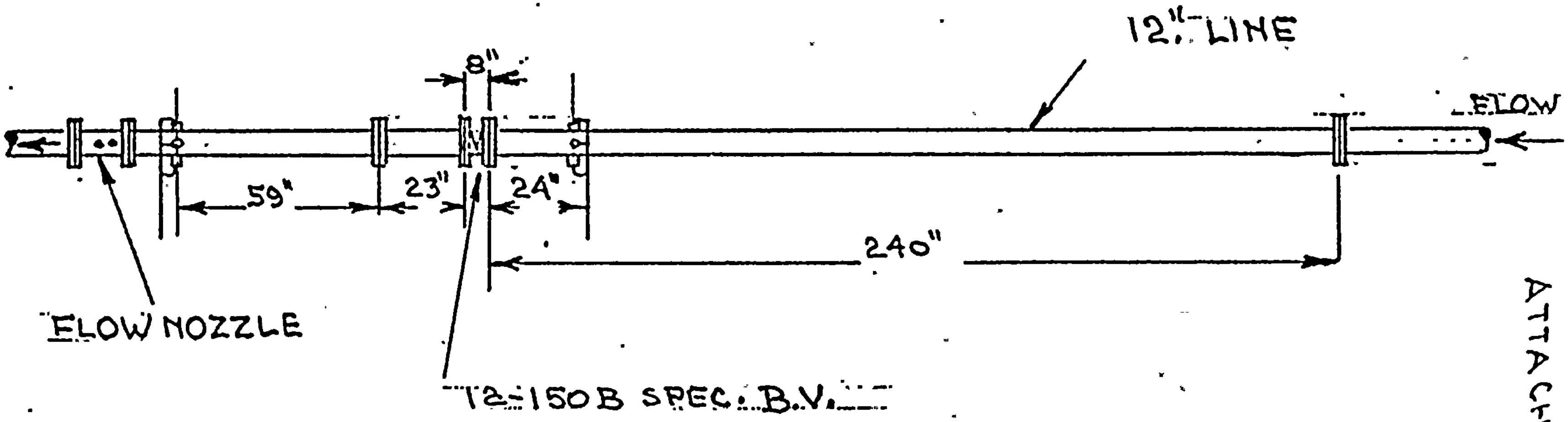
DSKK-200617

FORM 1162A

BIF A UNIT OF GENERAL SIGNAL

12" PIPE HEADLOSS TEST SETUP.

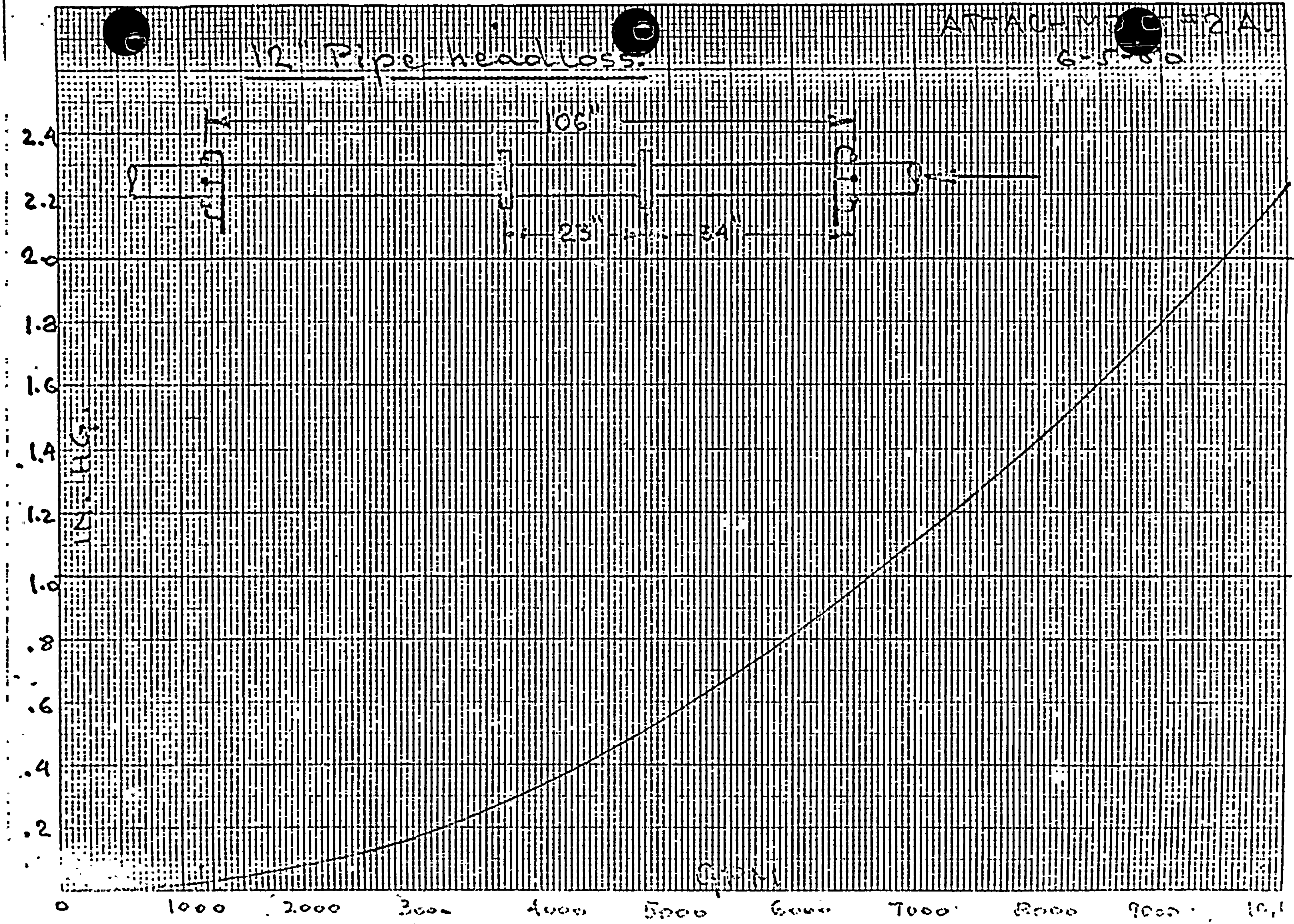




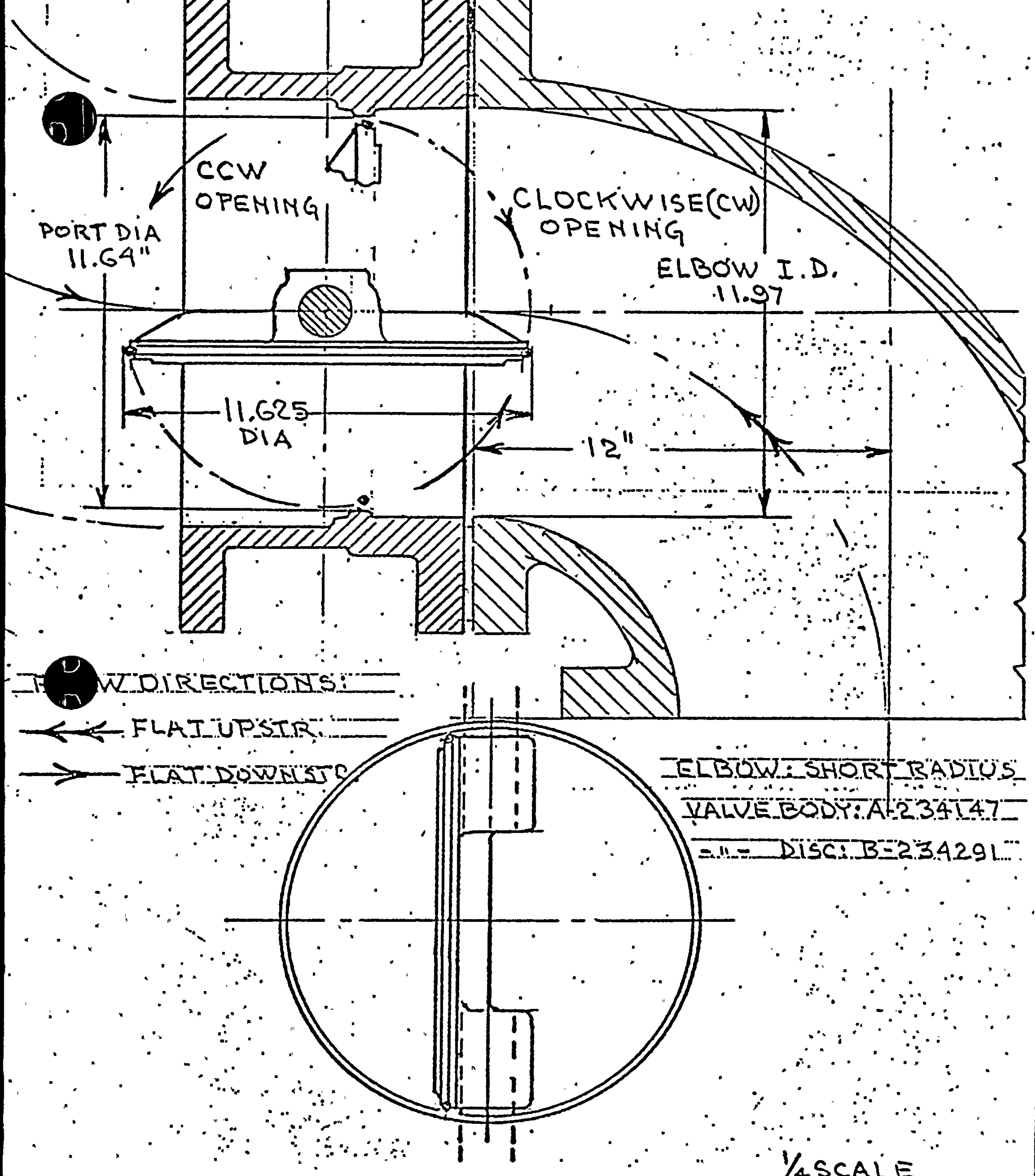
12" PIPE & B.V. HEADLOSS TEST SETUP.

ATTACHMENT #2

TEST SETUP BIF HYDRAULIC LAB.	
DR K.K.	DATE 6-4-80
DSKK-800604	


12" Pipe headloss
106"
23"
34"
2.4
2.2
2.0
1.8
1.6
1.4
1.2
1.0
.8
.6
.4
.2
0
1000
2000
4000
7000

ATTACHMENT #3.

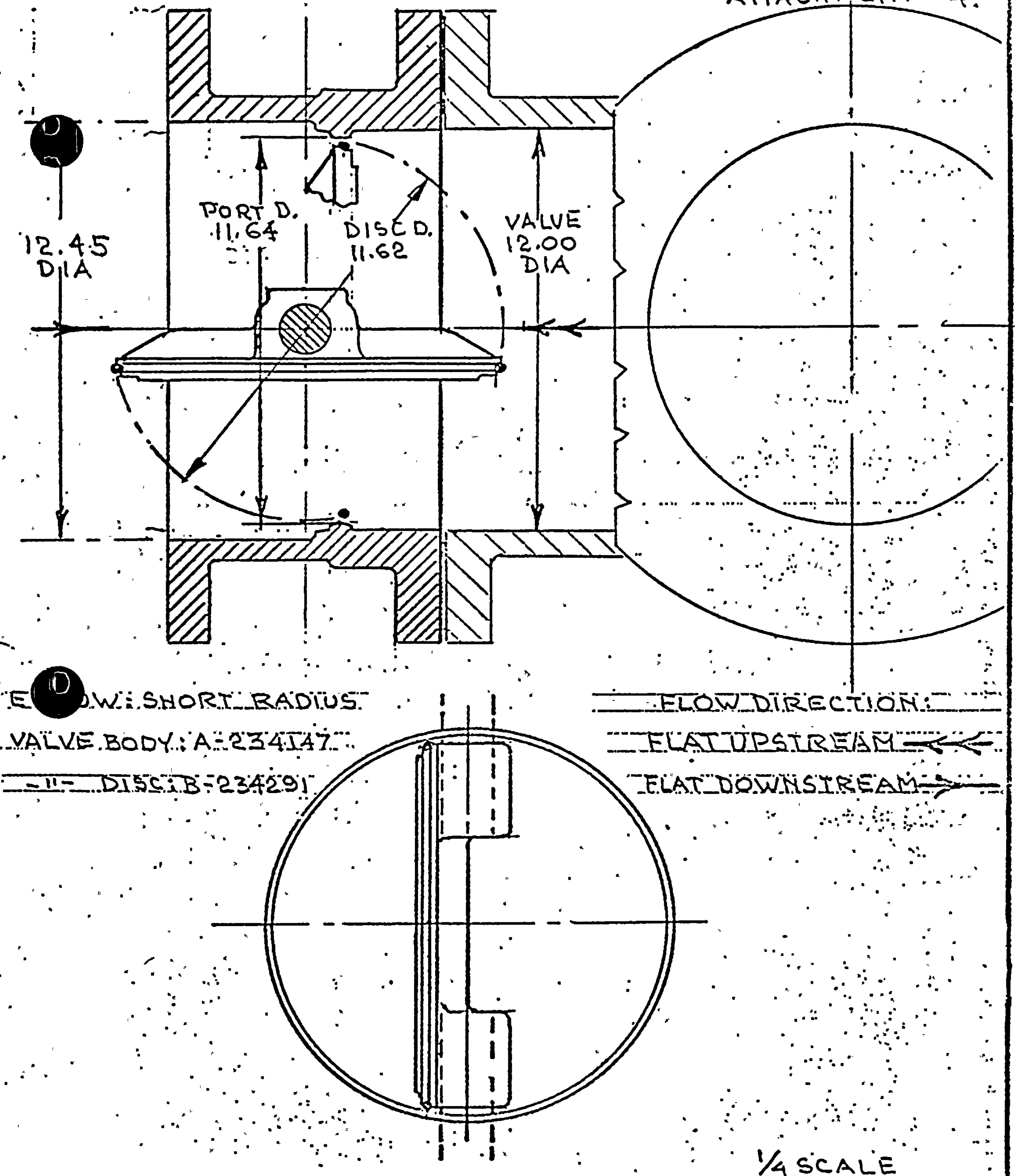


	B	C	D	E	F

BIF A UNIT OF GENERAL SIGNAL	12-150 B.V. WITH ELBOW VERTICAL SHAFT	DR K.K.	DATE 6-26-80

ORM 2162A

ATTACHMENT #4.



	B	C	D	E	F

BIF
A UNIT OF GENERAL SIGNAL

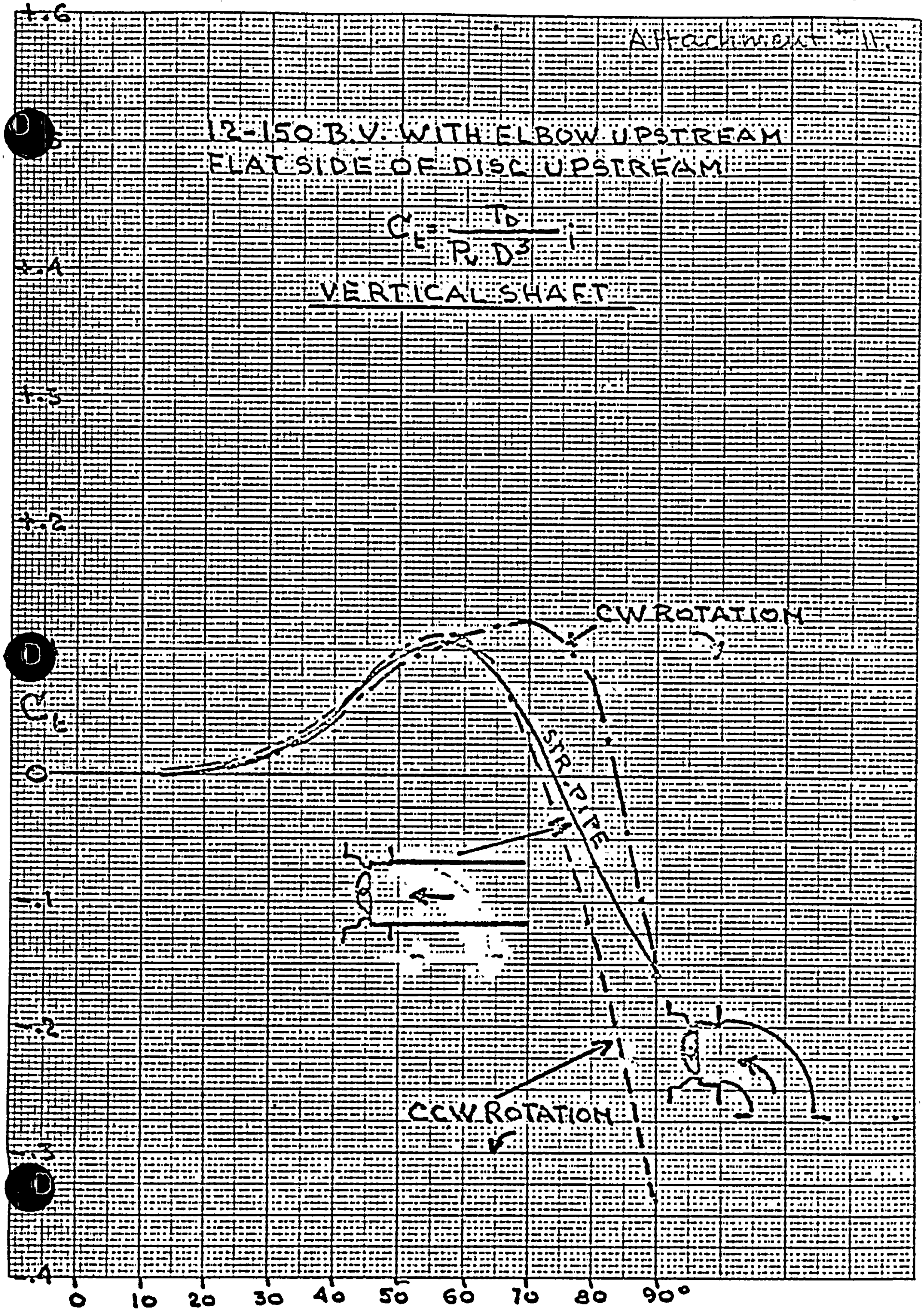
12-150 B.V. WITH ELBOW
HORIZONTAL SHAFT

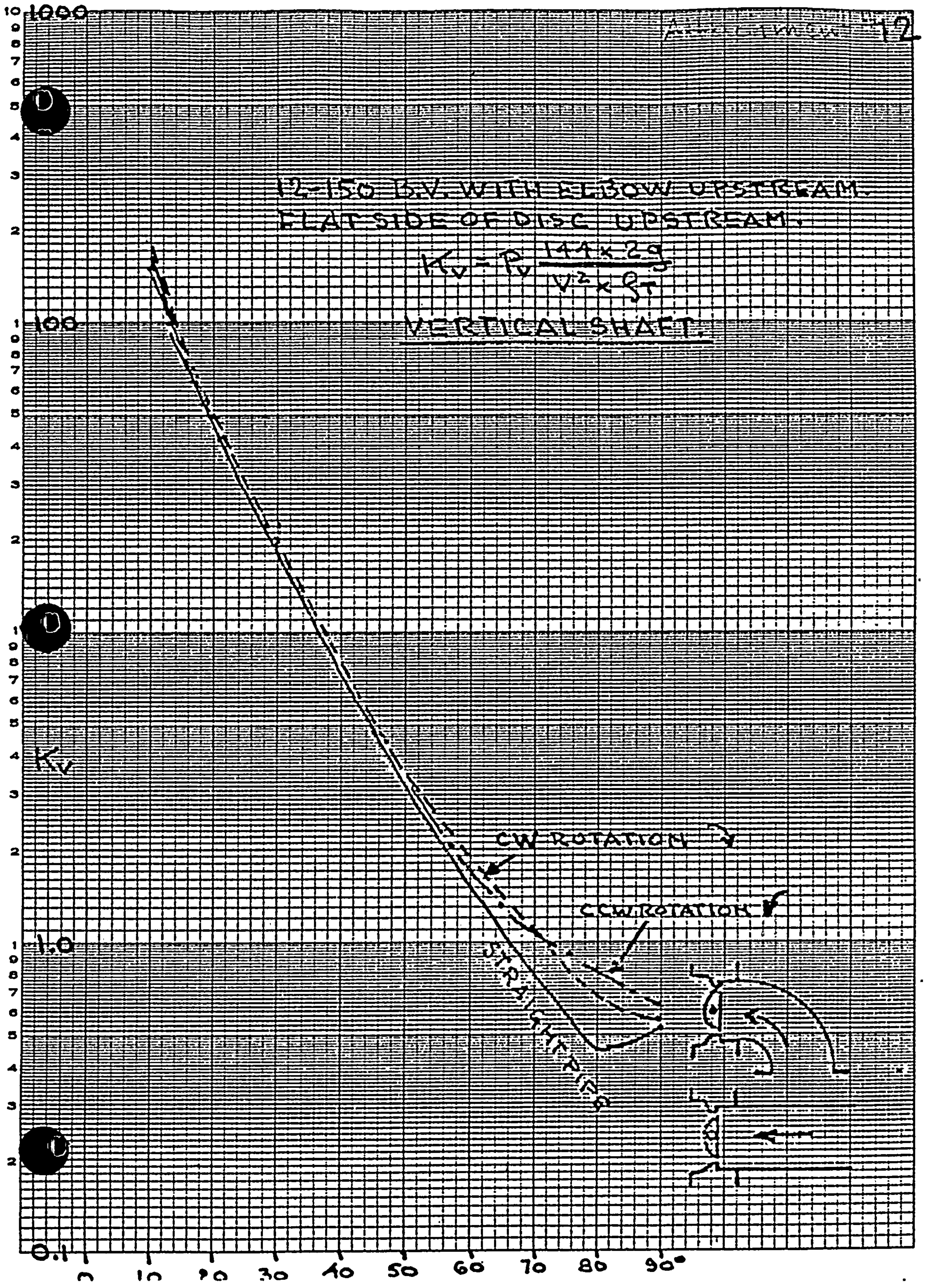
DR K.K.

DATE 6-26-60

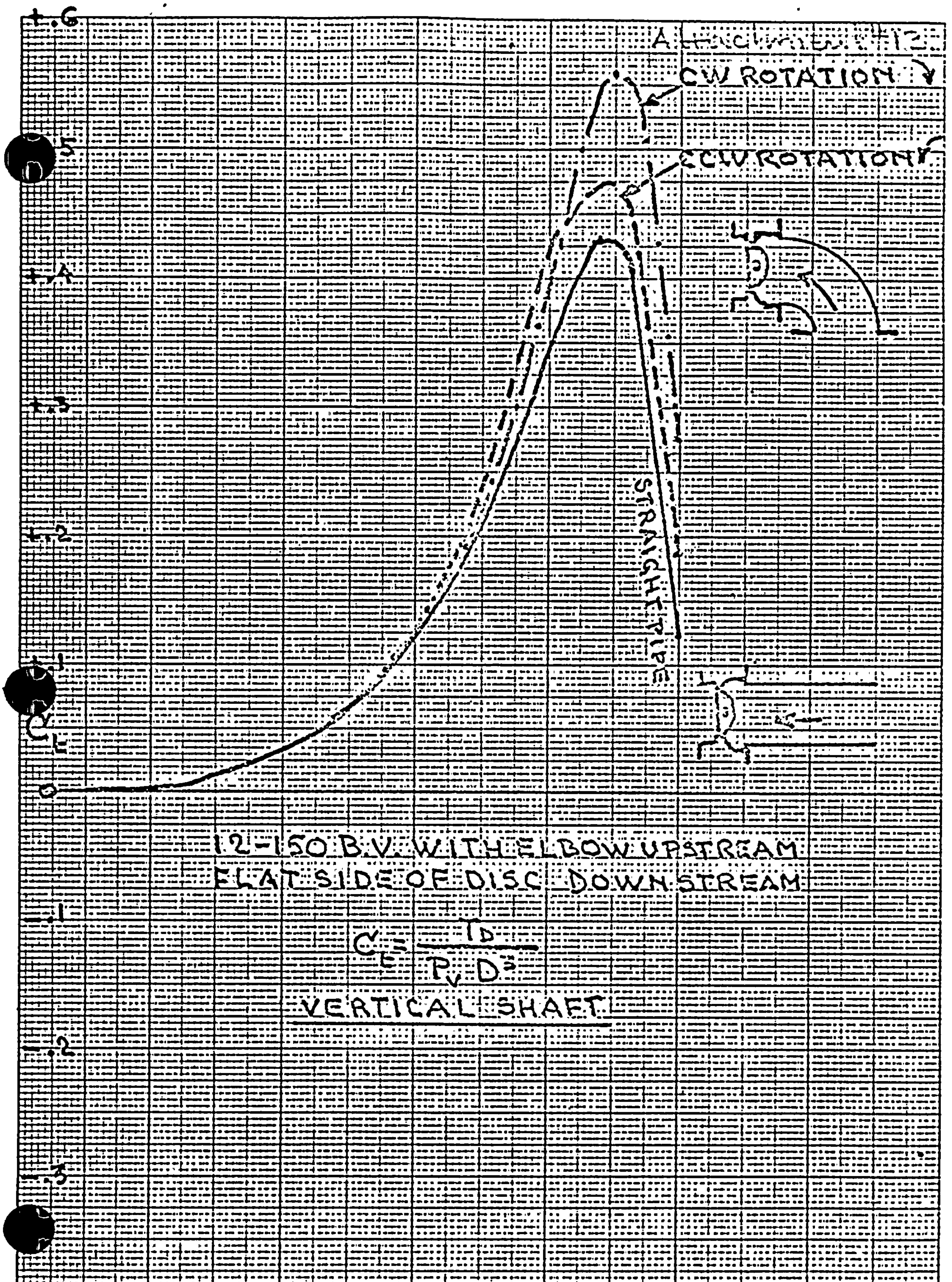
FORM 12A

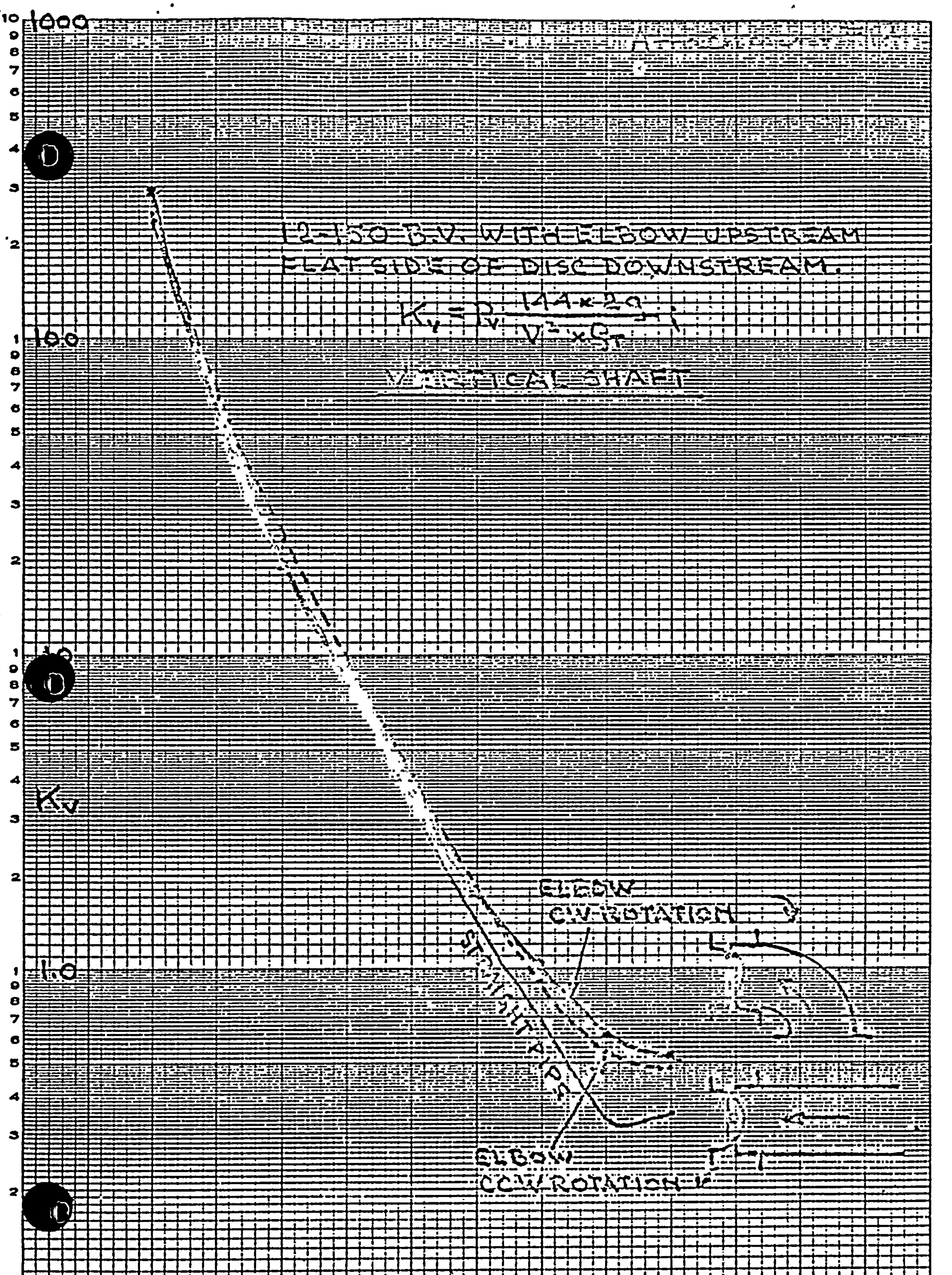
Attachment 11



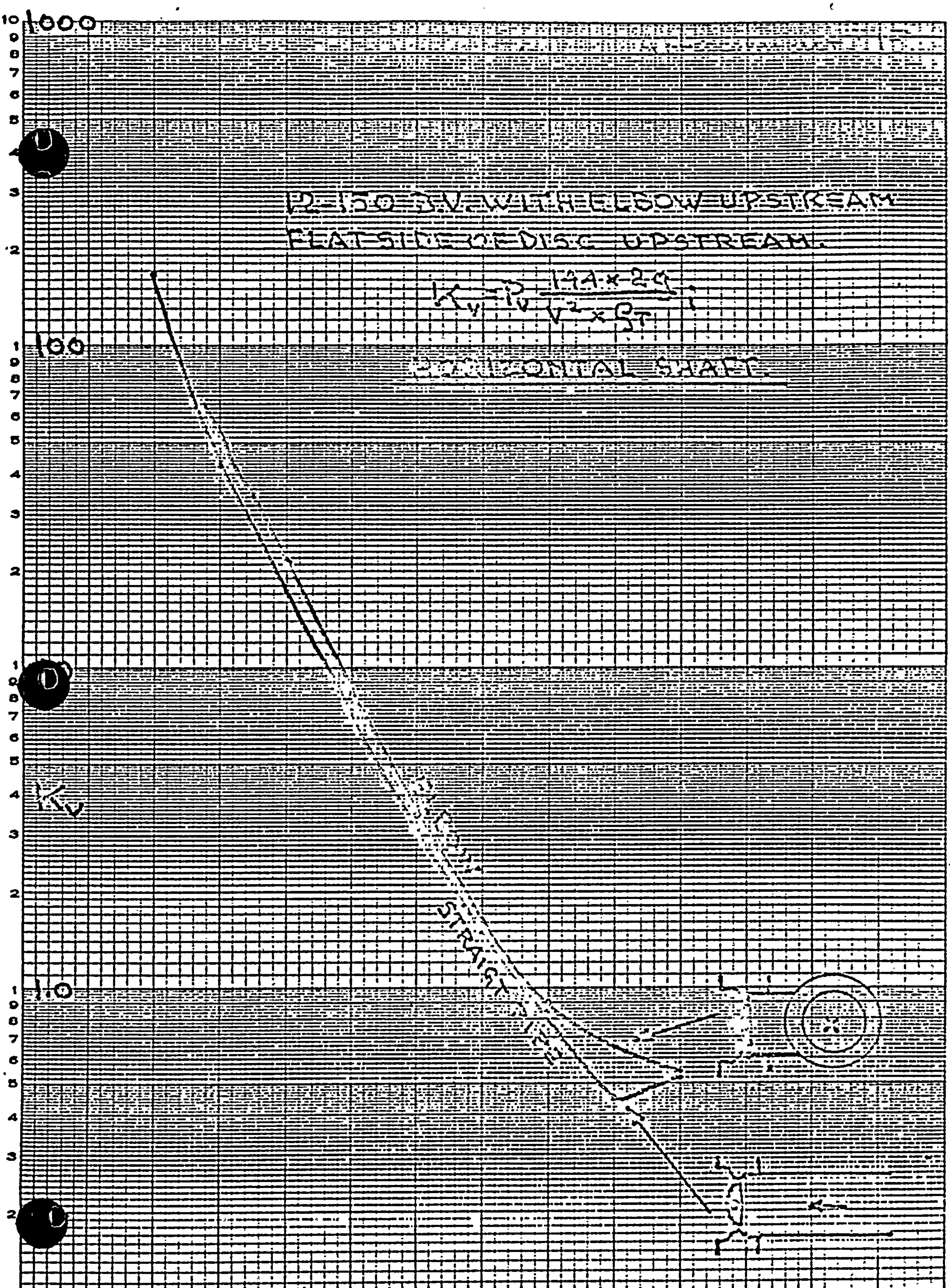


12
12-150 B.V. WITH ELBOW UPSTREAM.
FLAT SIDE OF DISC UPSTREAM.
$K_V = P_V \frac{144 \times 2g}{V^2 \times \rho_T}$
VERTICAL SHAFT.
CW ROTATION
CCW ROTATION
STRAIGHT PIPE
K_V
1000
100
10
1.0
0.1
0
10
20
30
40
50
60
70
80
90°

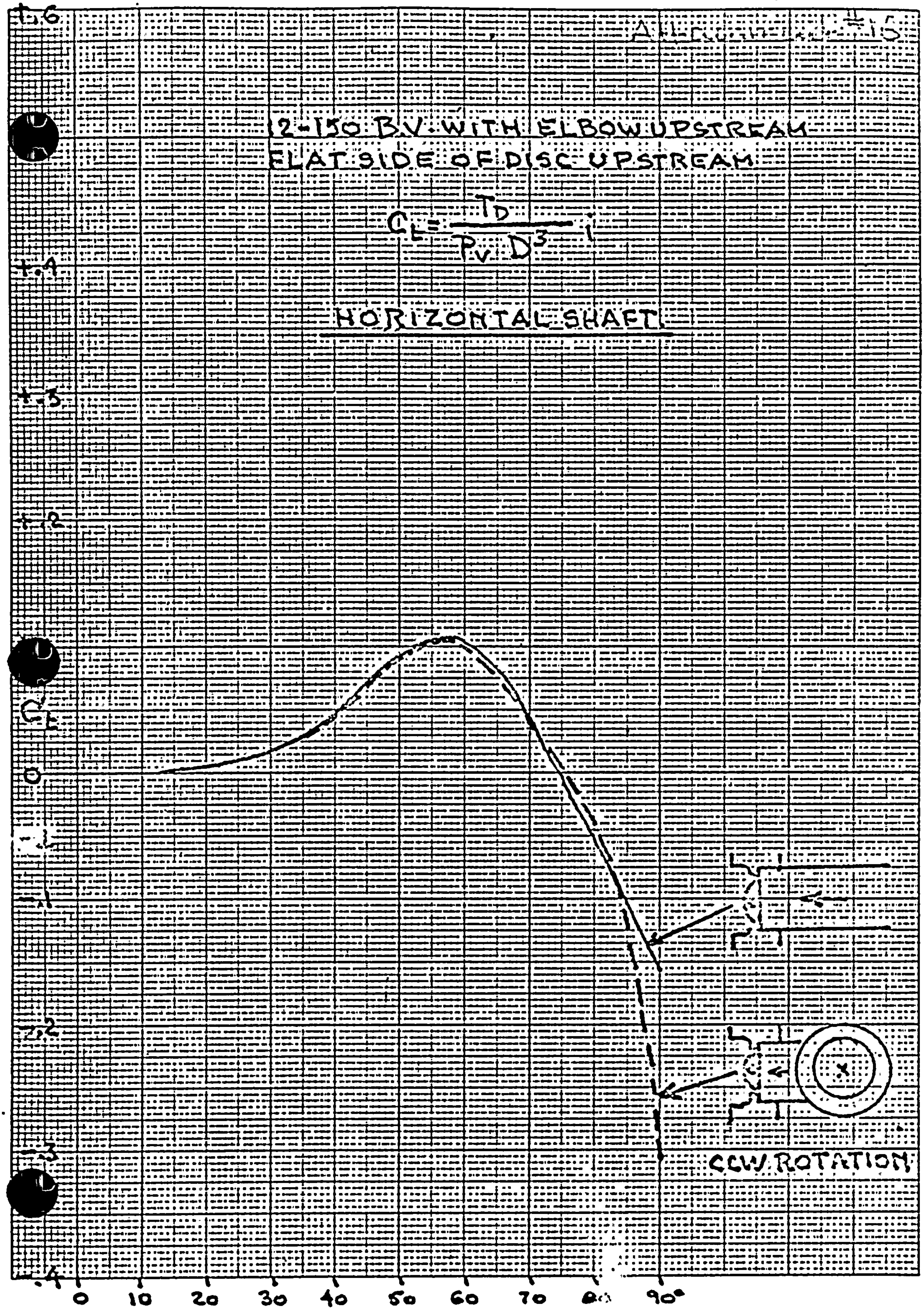

ATTACHMENT 12
CW ROTATION
CCW ROTATION
STRAIGHT PIPE
12-150 B.V. WITH ELBOW UPSTREAM
FLAT SIDE OF DISC DOWNSTREAM
$C_T = \frac{T_D}{P_V D^3}$
VERTICAL SHAFT
C_T
+.6
+.5
+.4
+.3
+.2
+.1
0
-.1
-.2
-.3
-.4
0
10
20
30
40
50
60
70
80
90°

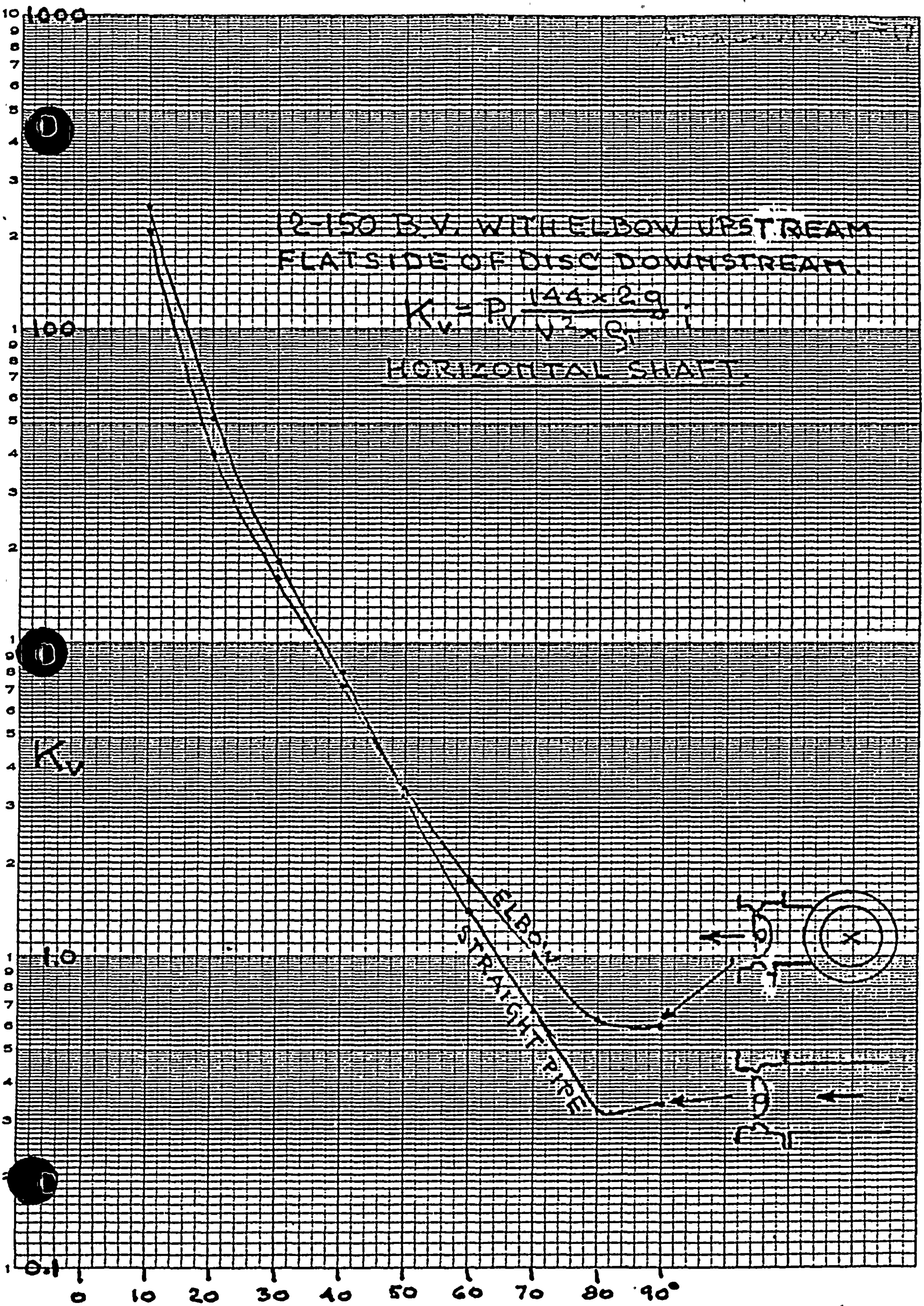


12-150 B.V. WITH ELBOW UPSTREAM
FLAT SIDE OF DISC DOWNSTREAM.
$K_v = P_v \dfrac{144 \times 2g}{V^2 \times S_T}$
VERTICAL SHAFT
ELBOW
CW ROTATION
ELBOW
CCW ROTATION
K_v
1000
100
10
1.0
0.1
0
10
20
30
40
50
60
70
80
90°

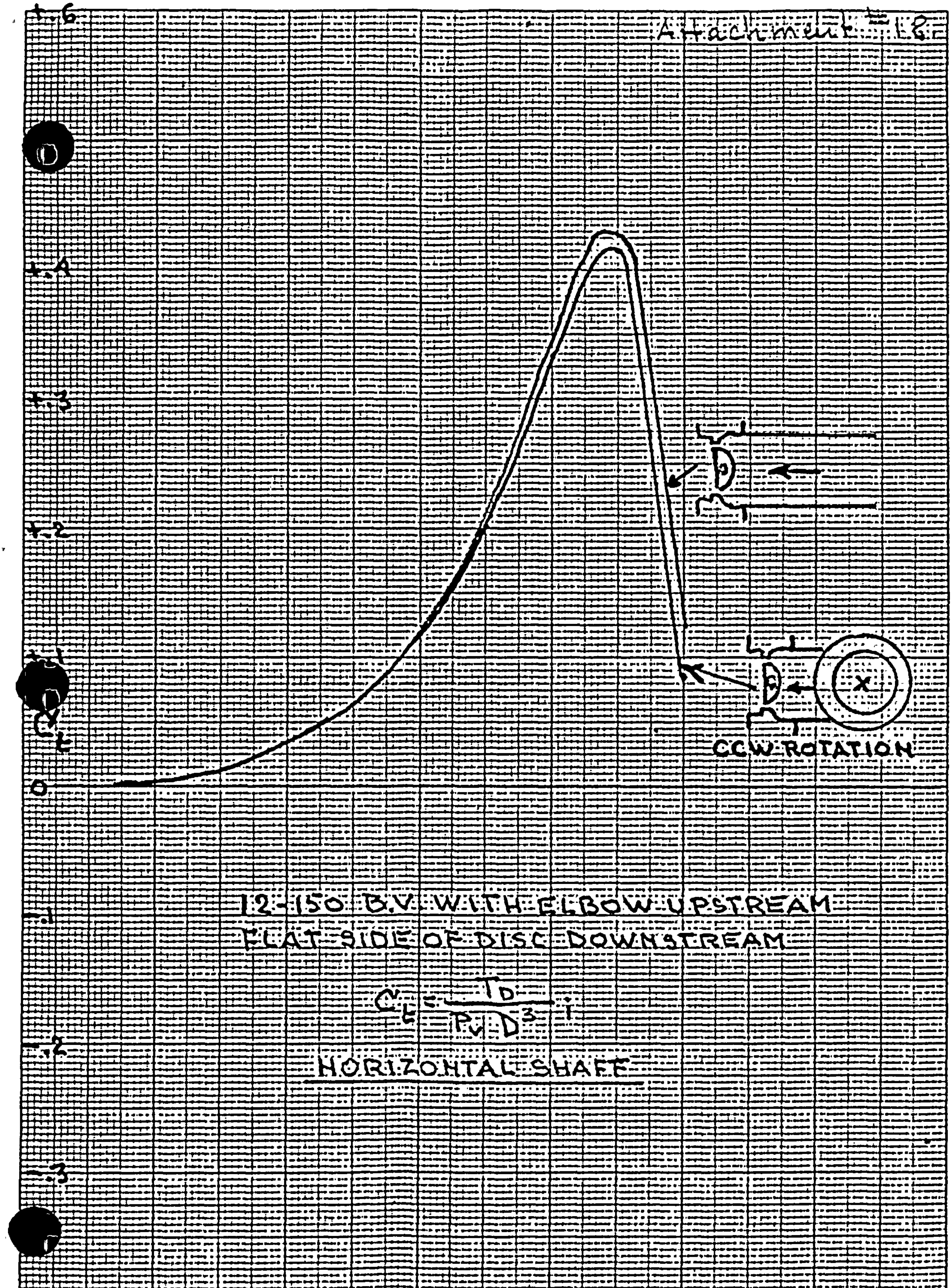


R-150 B.V. WITH ELBOW UPSTREAM
FLAT SIDE OF DISC UPSTREAM.
$K_V = P_V \frac{144 \times 2g}{V^2 \times \rho_T}$
HORIZONTAL SHAFT
STRAIGHT
K_V
1000
100
10
1.0
0.1
0
10
20
30
40
50
60
70
80
90°


ATTACHMENT #5
12-150 B.V. WITH ELBOW UPSTREAM
FLAT SIDE OF DISC UPSTREAM
$C_L = \frac{T_D}{P_V D^3}$
HORIZONTAL SHAFT
C_L
+.6
+.4
+.3
+.2
0
-.1
-.2
-.3
-.4
0
10
20
30
40
50
60
70
80
90°
CCW ROTATION



12-150 B.V. WITH ELBOW UPSTREAM
FLATSIDE OF DISC DOWNSTREAM.
$K_v = P_v \dfrac{144 \times 2g}{V^2 \times \rho_1}$
HORIZONTAL SHAFT.
K_v
ELBOW
STRAIGHT PIPE
1000
100
1.0
0.1
0
10
20
30
40
50
60
70
80
90°


Attachment #18
+.6
+.4
+.3
+.2
+.1
C_t
0
-.1
-.2
-.3
-.4
0
10
20
30
40
50
60
70
80
90°
CCW ROTATION
12-150 B.V. WITH ELBOW UPSTREAM
FLAT SIDE OF DISC DOWNSTREAM
$C_t = \frac{T_D}{P_v D^3}$
HORIZONTAL SHAFT

B I F

A UNIT OF GENERAL SIGNAL

TEST REPORT

DYNAMIC TORQUE & HEAD LOSS TESTS OF CAST IRON STREAMLINE DISC VERSUS FABRICATED FLAT PLATE DISC

Prepared By: F. E. Hart

Approved By: Dezso Szilagyi

5-13-75

1600 Division Rd.
West Warwick, RI 02893

TABLE OF CONTENTS

ABSTRACT

Performance tests were conducted on two 12" Class 150-B Butterfly Valves of the Model 0658 design with sewage rings. One valve was constructed using the standard streamlined cast iron disc, the other a fabricated flat plate disc.

The B I F Hydraulic Laboratory was utilized to perform tests on each valve. A 12" pipe line was used to mount each valve on individually to perform its series of tests. With flow passing through the valve, the disc was set at a specific angle of opening. These angles were 5^{o}, $7\frac{1}{2}^{o}$, 10^{o} and each 5^{o} increment to the full open position of 90^{o}. At each setting, torque measurements were made with a torque wrench to determine the amount of torque required to move the disc from the set position and return the disc to the set position. After determining the torque, the disc was held in the set position while the flow rate was measured. This was accomplished by diverting the flow to a weighing tank. Weighing the amount of water in the tank and measuring the time required to obtain the amount of water. The differential pressure was measured at each setting by using a well type single tube mercury manometer.

Each valve was subjected to two series of test runs, one with the flat side of the disc upstream and one with the flat side of the disc downstream.

These series of test runs resulted in determining the dynamic torque and head loss for each disc positioned in the pipe line in both positions.

PROCEDURE

Tests were performed to determine the dynamic torque coefficient (C_T) and the coefficient of flow (K_V) of a 12" Class 150-B valve, Model 0658, with sewage ring.

Two valves were used for the test. One valve was constructed with a standard streamlined cast iron disc (Part No. B-189325-1), the other with a fabricated flat plate disc (Part No. C-216994).

Prior to the valve test runs a pipe calibration run was made, without the test valve installed, to determine the pipe head loss. For the calibration run, a 20" water micro "U" tube manometer was used to measure the differential pressure. With the determined head loss, a weighing tank was filled with a timer connected so that flow rate could be determined. The results of this run appear on Test Log #1 and Graph #1.

Each valve was installed in the pipe line assembly (Fig. 1) in two positions; one position with the flat side of the disc downstream, the other position with the flat side upstream. Each position of the valve was with the shaft in the vertical position to eliminate the effect of hydrostatic torque.

For test purposes a lever operator was installed on the valve with a socket adaptor located over the center of the shaft for a torque wrench. Each valve was tested at 5°, $7\frac{1}{2}^\circ$, 10° and each 5° increment to 90° for

both positions. A mounting plate made up part of the lever assembly to permit "C" clamps to be used to position the lever arm in the specified positions.

The first test valve installed contained a cast iron disc with the flat side of the disc downstream. Two mercury well single leg manometers (one 50", the other 100"), were connected to the pipe at points indicated on the log sheets. The manometer used was determined by the differential encountered for the different angle settings.

Water was introduced into the test line and air was bled from the manometers, manometer lines and test line. The manometers were zeroed with no flow through the test line.

A run consists of setting the disc at one of the desired angles and adjusting the flow through the test valve with the control valves located downstream. After the flow had stabilized, torque readings were taken and recorded, T_1 being the torque required to move the disc from the set angle and T_2 the torque required to return the disc to the set angle. The torque reading procedure was repeated until the torque readings themselves repeated and stabilized. Following the torque reading procedure the flow rate was measured. This was accomplished by diverting the flow through the line to a weighing tank by use of the vertical switchway. A timer was hooked up to determine the length of time the switchway diverted the flow to the weighing tank. With this data, the flow rate was determined. While the weighing tank was being filled, differential pressure

readings were taken from the mercury well manometer. The average of these readings appear on the test log under ΔP_S.

This run procedure was followed for each desired angle setting and the results recorded on the log sheets. At the completion of the series of runs, the valve was relocated in the pipe line 180° from the first position resulting in the valve disc being located with the flat side upstream. Again the system was bled of all air and the same series of runs were performed trying as close as possible to maintain the same differential pressure.

For the second test valve with the flat plate fabricated disc installed, the same series of runs were performed with the valve located in the same two positions in the line. The results of these tests appear on the following logs and graphs.

Cast iron disc with flat side downstream, Test Log #2, Graph #2 & 2A.
Cast iron disc with flat side upstream, Test Log #3, Graph #3 & 3A.
Fabricated disc with flat side downstream, Test Log #4, Graph #4 & 4A.
Fabricated disc with flat side upstream, Test Log #5, Graph #5 & 5A.

CONCLUSION

For valves positioned with the flat side of the disc downstream, the coefficient of flow with the disc full open, for the flat plate disc was 1.58 times greater than the cast iron disc. The dynamic torque coefficient, with the disc full open, was 1.14 times greater in the fabricated disc than in the cast iron disc with no negative torque experienced from zero to ninety degrees. Each disc reached a maximum torque at seventy degrees.

For valves positioned with the flat side of the disc upstream, the coefficient of flow with the disc full open, for the flat plate disc was 1.82 times greater than the cast iron disc. The dynamic torque coefficient, with the disc full open, for the flat plate disc was 1.89 times greater than the cast iron disc. A negative torque was experienced in the flat plate disc at 60^{o} and at 75^{o} for the cast iron disc and continued up to the 90^{o} position. A maximum torque was reached at 45^{o} for the flat plate disc and 25^{o} for the cast iron disc.

The same comparative results should be obtained on other discs tested under similar conditions and having the same disc diameter to thickness ratio (Fig. 2).

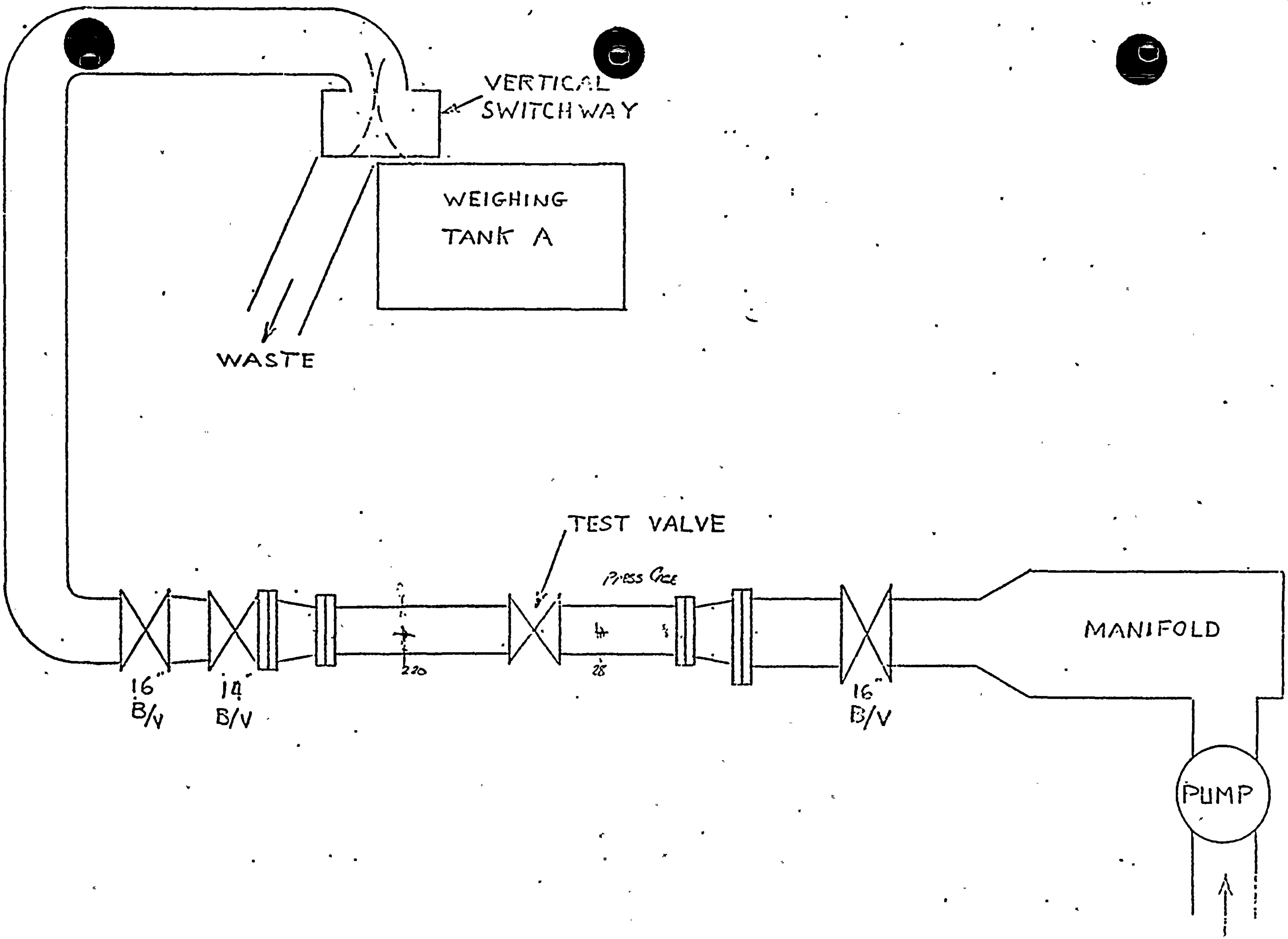
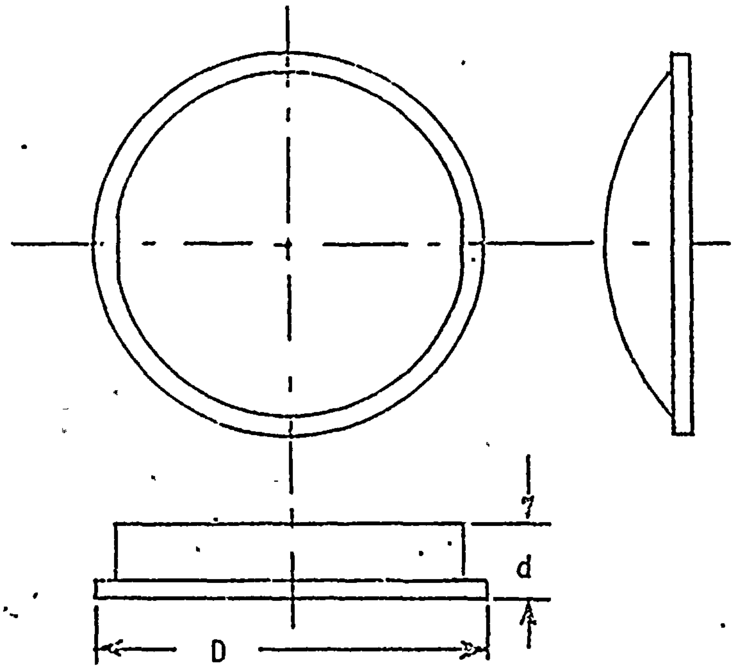

VERTICAL SWITCHWAY
WEIGHING TANK A
WASTE
TEST VALVE
220
28
MANIFOLD
16" B/V
14" B/V
16" B/V
PUMP

FIG. 1

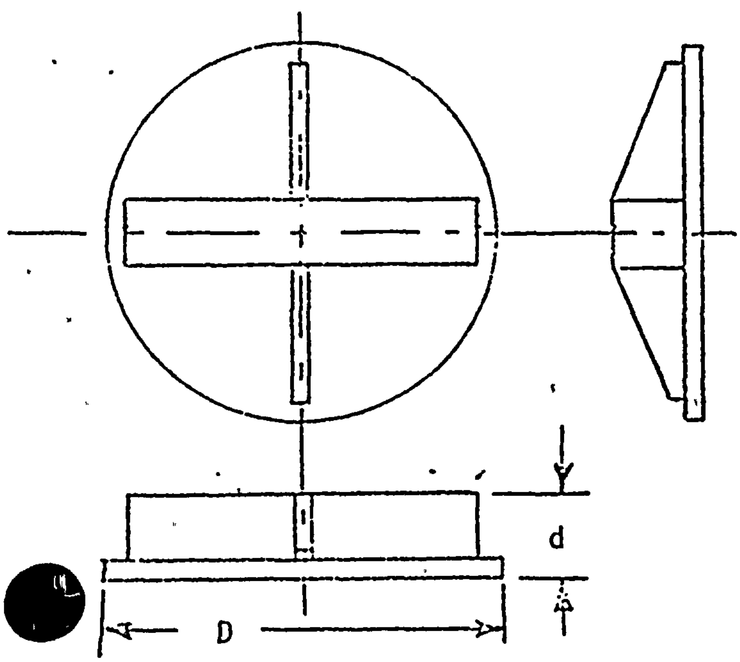
DISC DIAMETER TO THICKNESS RATIO

STREAMLINED CAST IRON DISC



$\frac{D}{d} = \cancel{4.6}\ 4.21$

FABRICATED FLAT PLATE DISC



$\frac{D}{d} = \cancel{4.6}\ 4.21$

FIG. 2

CALCULATIONS

DIFFERENTIAL PRESSURE

$$\Delta P_V = \frac{\Delta P_S - \Delta P_P}{2.204}$$

ΔP_V = differential pressure of the valve (PSI)

ΔP_S = differential pressure of test valve and pipe (IN. Hg.)

ΔP_P = differential pressure of pipe (IN. Hg.)

2.204 = conversion factor (IN. Hg. wet to PSI)

FLOW RATE

$$Q = \frac{W}{t}(0.01607)(7.4805)(60)$$

Q = flow rate (GPM)

W = water collected in weighing tank (LBS.)

t = time required to fill weighing tank (SEC.)

0.01607 = conversion factor for water in cubic feet per pound

7.48 = conversion factor for water in gallons per cubic foot

1/60 = conversion factor (seconds to minutes)

FLUID VELOCITY

$$V = \frac{Q}{(60)(7.5)A}$$

V = fluid velocity (FT. PER SEC.)

Q = flow rate (GPM)

60 = conversion factors (minutes to seconds)

7.48= conversion factor (gallon to cubic feet)

A = area of valve port (ft.^{2})

DYNAMIC TORQUE

$$T_D = \frac{T_1 - T_2}{2} \times 12$$

T_D = average dynamic torque (IN. LBS.)

T_1 = torque in opening direction (FT. LBS.)

T_2 = torque in closing direction (FT.LBS.)

12 = conversion factor (FT. LBS. to IN. LBS.)

COEFFICIENT OF DYNAMIC TORQUE

$$C_T = \frac{T_D}{\Delta P_V\ D^3}$$

C_T = coefficient of dynamic torque (dimensionless)

T_D = average dynamic torque (IN. LBS.)

ΔP_V = differential pressure across the valve (PSI)

D = disc diameter (IN.)

COEFFICIENT OF FLOW

$$K_V = \Delta P_V \ \frac{144}{V^2}\ \frac{2g}{w}$$

K_V = coefficient of flow (dimensionless)

ΔP_V = differential pressure across the valve (PSI)

144 = conversion factor square feet to square inches

V = fluid velocity (FT. PER SEC.)

g = acceleration due to gravity (FT. PER SEC.^{2})

w = specific weight of water (LBS. PER FT.^{3})

TEST LOG #1

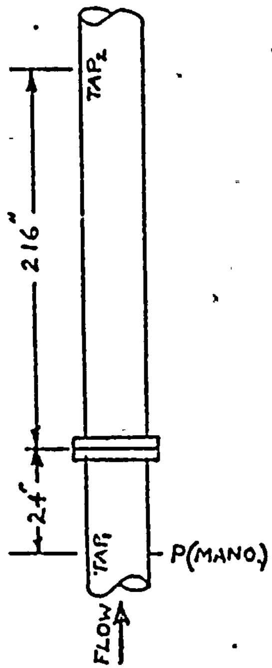
BIF HYDRO-LAB CALIBRATION LOG

TEST #1

TYPE METER:- PIPE LOSS CALIBRATION RUN FOR 12" B/V TEST — CALIBRATED BY:- A.J.I. & H.C.

SIZE: ——— X — BETA:- — S/N:- — ORDER No.:- 03422-R — DATE:- 4/3/75

PIPING SKETCH	RUN No.	TIME OF DAY	TANK USED	WEIGHT (#) START	WEIGHT (#) STOP	TIME (SEC'S)	TEMPERATURE °F LINE	TEMPERATURE °F ROOM	TEMPERATURE °F MANO	GPM	MANO. USED	DIFF. PK-PK (%)	No. DIFF. R'DGS	PRESS. (PSI) THROAT	PRESS. (PSI) MANO.	AVG. DIFF.	ΔC (%)	ΔPp " Hg	X 1,000 = R_D
CONDITION OF PIPE:-	1	A.M. 8:44	A	653#	49561#	55.653 55.653	79	73	73	6328	20" MW				64	15.528	15.43	1.228	
LEAK IN LOOP:-	2	8:50	A	760#	49100	68.623 68.623	79	73	73	5084	20" MW				64	10.090	10.03	0.798	
ECCENTRICITY:-	3	9:17	A	766#	49170#	89.451 89.451	79	70	73	3905	20" MW				64	6.041	6.00	0.477	
MISMATCH:-	4	9:27	A	647#	49806#	127.720 127.720	79	70	73	2778	20" MW				65	3.105	3.09	0.246	
CONDITION OF METER:-	5	9:37	A	746#	49705#	221.342 221.342	79	70	73	1596	20" MW				64	1.064	1.06	0.084	



FORM NO. 121

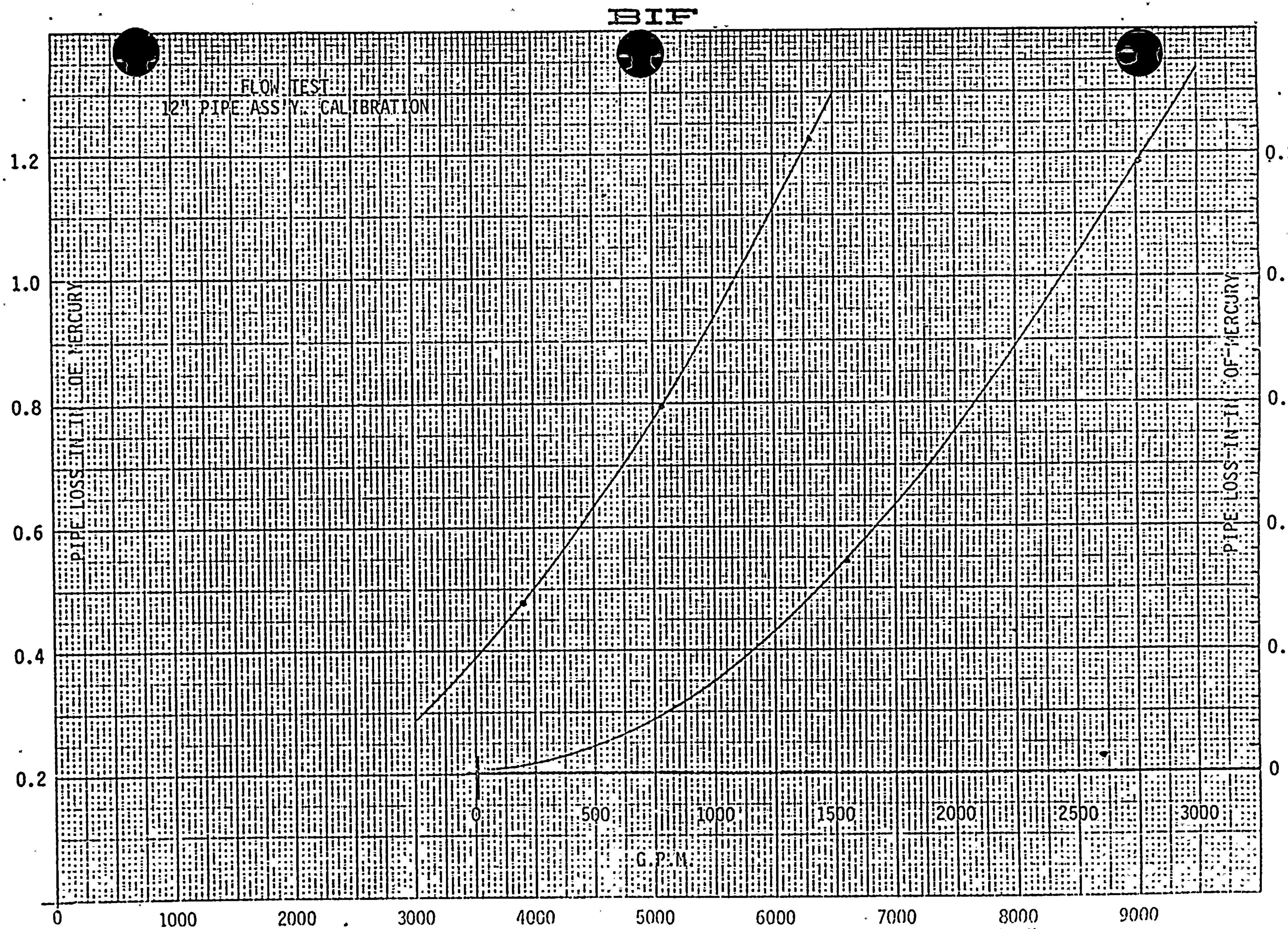
100 INCH MERCURY SINGLE LEG:— 100 WM
50 INCH MERCURY SINGLE LEG:— 50 WM
20 INCH MERCURY MICRO-U-TUBE:- 20 MM

100 INCH WATER U-TUBE:— 100 UM
20 INCH WATER MICRO-U-TUBE:- 20 MW
ΔC % : CORRECTED AVERAGE DIFFERENTIAL

METER FACTOR = $\frac{0.0997(d^2)}{\sqrt{1-\beta^4}}$

R_D FACTOR = $\frac{26,528}{D}$

* - APPROX. C


BIF
FLOW TEST
12" PIPE ASS'Y. CALIBRATION
PIPE LOSS IN IN. OF MERCURY
1.2
1.0
0.8
0.6
0.4
0.2
0
1000
2000
3000
4000
5000
6000
7000
8000
9000
G.P.M.
0
500
1000
1500
2000
2500
3000
PIPE LOSS IN IN. OF MERCURY

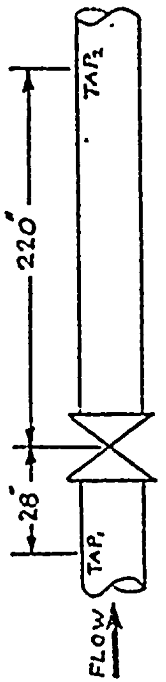
ST LOG #2

BIF HYDRO-LAB CALIBRATION LOG

D = 11.58 IN.
A = .795 FT.^{2}
CALIBRATED BY:-

TYPE METER:- TEST #2 12" B.V. 150-B W/CAST IRON DISC FLATSIDE DOWNSTREAM
SIZE: ——— X BETA:- S/N:- ORDER NO.:- 03422-R DATE:- 4-4-75

PIPING SKETCH	α	TIME OF DAY	TANK USED	WEIGHT (#) START	WEIGHT (#) STOP	TIME (SEC'S)	TEMPERATURE °F LINE	TEMPERATURE °F ROOM	TEMPERATURE °F MANO	ΔP_S "Hg	ΔP_F "H2	ΔP_V "H2	ΔP_V PSI	T_1 " H	T_2 " H	Q GPM	T_D " H	P_1 PSI	V F/S	C_T	K_V
CONDITION OF PIPE:-	5°	2:20	"A"	650#	10510#	181.297 181.297	80	67	66	72.911	0.010	72.900	33.070	53	20	392.10	198	63	1.19	.003	3460
LEAK IN LOOP:-	7 1/2	2:35		760#	17551#	179.280 179.280	80	57	66	61.555	0.018	61.530	27.910	58	7	675.93	306	63	2.05	.007	981
ECCENTRICITY:- 0	10°	2:43		775#	24784#	180.373 180.374	80	67	67	58.92	0.038	58.880	25.710	63	-9	960.06	432	65	2.91	.010	466
MISMATCH:-	15°	2:53		764#	32645#	179.921 179.921	80	69	67	58.162	0.072	58.090	26.350	77	-28	1519.04	630	64	4.61	.015	183
CONDITION OF METER:-	20°	3:04		758#	48903#	179.636 179.636	80	70	69	55.717	0.117	55.600	25.220	90	-38	1934.45	768	61	5.88	.019	108
	25°	3:30		710#	50009	143.854 143.855	80	68	67	48.318	0.180	48.138	21.840	95	-48	2473.02	858	65	7.51	.025	57.4
	30°	3:43		770#	49195#	126.390 126.389	80	69	67	35.187	0.210	34.877	15.820	95	-58	2765.12	918	62	8.40	.037	33.2
	35°	3:56		765#	49512#	111.044 111.044	80	70	69	27.706	0.300	27.406	12.430	95	-73	3168.18	1,008	64	9.63	.052	19.9
	40°	4:05		774#	49370#	103.742 103.742	80	71	69	20.974	0.350	20.624	9.350	95	-78	3415.46	1038	62	10.38	.071	12.9
	45°	8:40		745#	49625#	91.932 91.932	79	60	59	15.346	0.450	15.896	7.210	98	-89	3837.37	1116	62	11.66	.099	7.89
	50°	8:47		772#	49341#	90.472 90.471	79	62	59	11.658	0.460	11.198	5.080	91	-78	3874.48	1014	61	11.77	.128	5.45
	55°	8:53		769#	49990#	84.594 84.594	79	63	59	10.008	0.540	9.550	4.330	91	-81	4199.82	1032	60	12.76	.153	3.95
	60°	9:13		761#	49581#	75.175 75.175	79	64	60	8.272	0.675	7.597	3.440	93	-88	4686.98	1086	61	14.24	.203	2.25
	65°	9:20		774#	49551#	69.674 69.673	79	66	60	6.804	0.785	6.019	2.730	91	-91	5053.61	1092	58	15.36	.257	1.72
	70°	9:25		774#	49780#	61.648 61.648	79	67	60	5.785	1.010	4.695	2.120	95	-95	5737.19	1140	60	17.44	.346	1.03
	75°	9:31		765#	49940#	61.611 61.611	79	68	60	3.926	1.025	2.901	1.310	78	-68	5760.43	876	60	17.51	.430	.635
	80°	10:37		768#	49450#	60.599 60.599	79	73	65	3.228	1.045	2.183	0.990	56	-52	5799.10	648	57	17.63	.421	.473
	85°	10:50		774#	49506#	61.484 61.484	79	74	65	2.728	1.010	1.718	0.779	38	-34	5720.33	432	58	17.39	.357	.383
	90°	10:55		770#	49735#	61.783 61.783	79	74	65	2.837	1.000	1.887	0.856	22	-18	5719.86	240	57	17.38	.180	.421



FORM NO 121

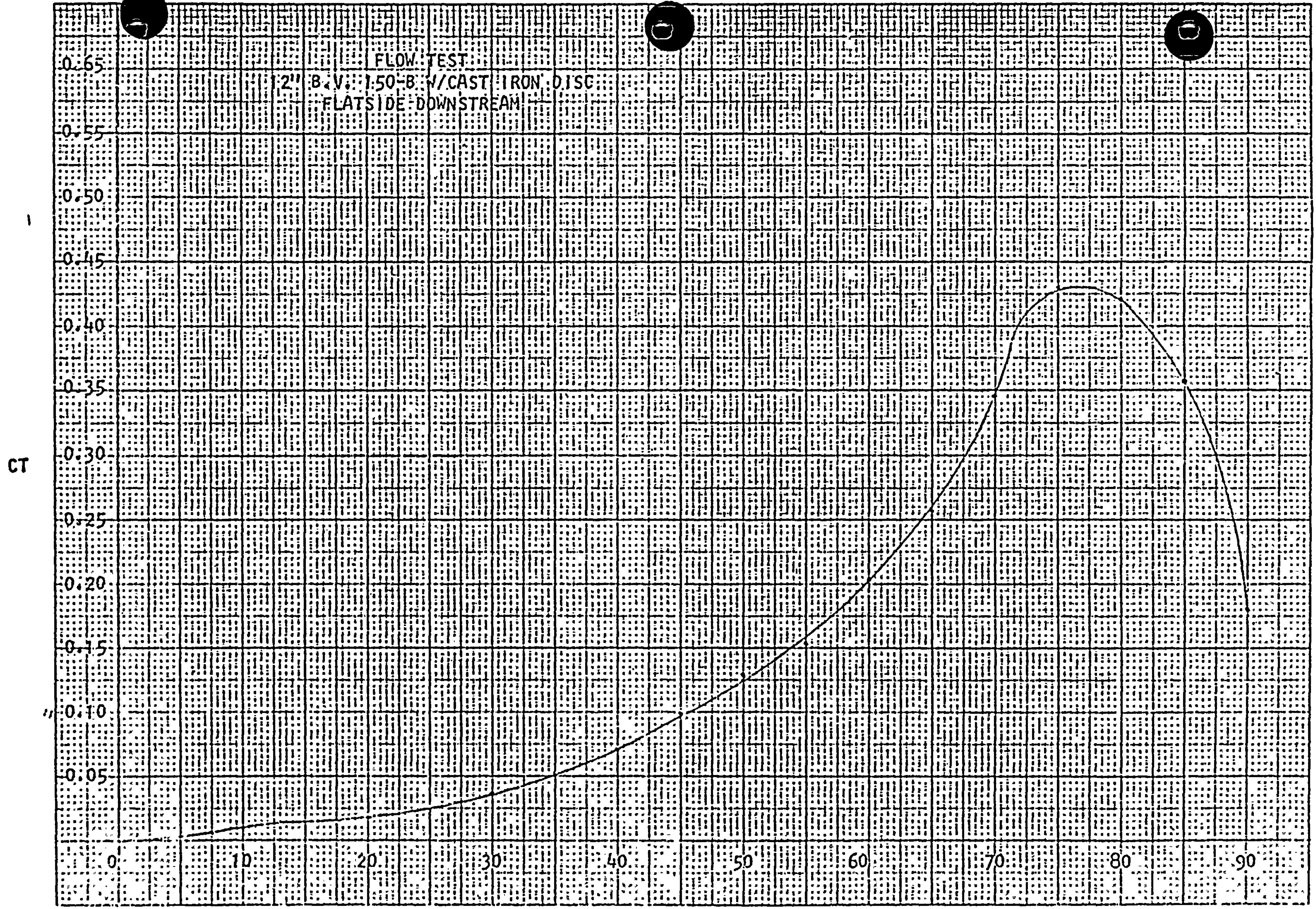
100 INCH MERCURY SINGLE LEG:— 100 WM
50 INCH MERCURY SINGLE LEG:— 50 WM
20 INCH MERCURY MICRO-U-TUBE:- 20 MM

100 INCH WATER U-TUBE:——— 100 UM
20 INCH WATER MICRO-U-TUBE:- 20 MW

METER FACTOR = $\frac{0.0997(d^2)}{\sqrt{1-\beta^4}}$

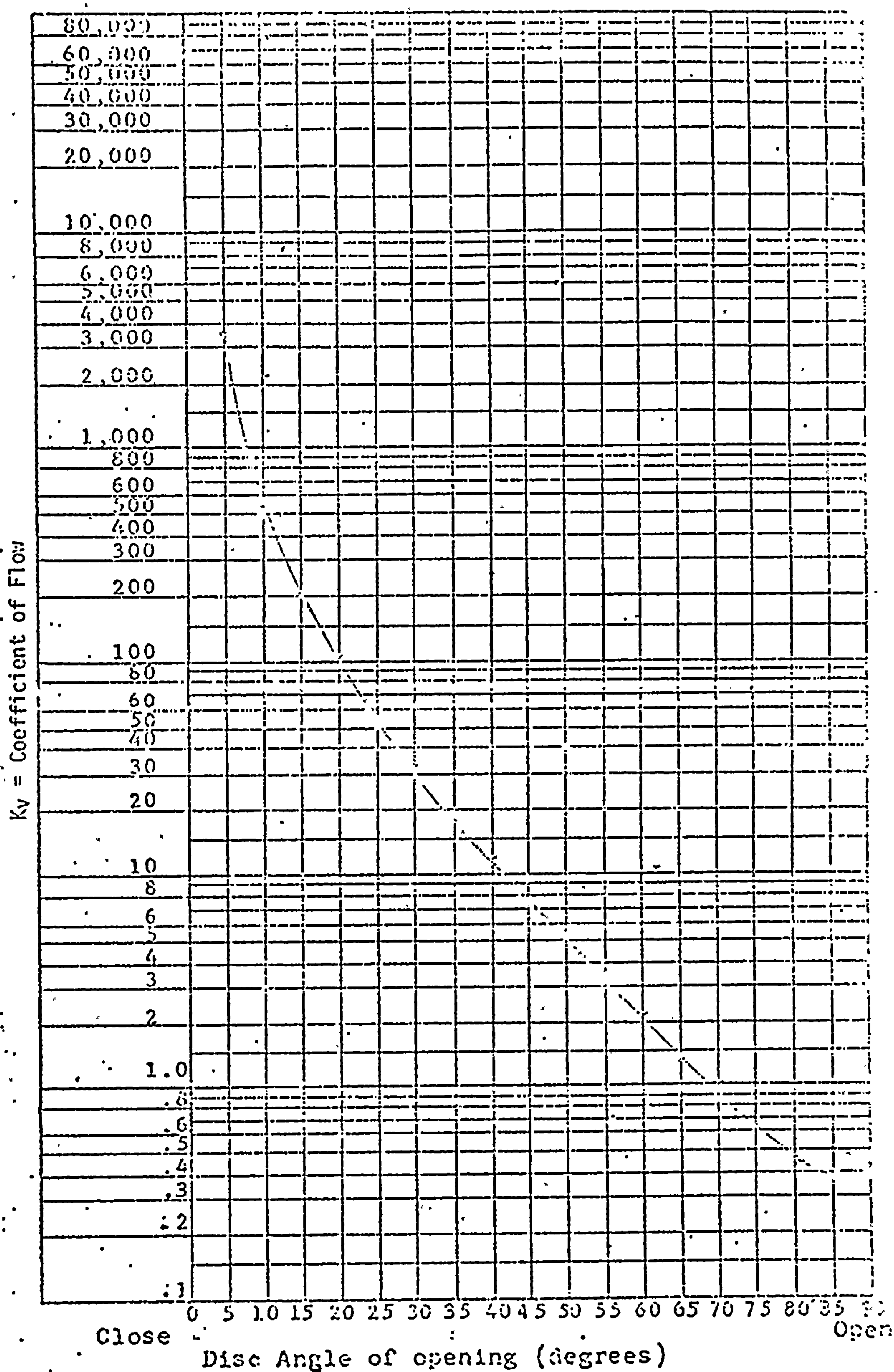
R_D FACTOR = $\frac{26.523}{D}$

* = APPROX. C


FLOW TEST
12" B.V. 150-B W/CAST IRON DISC
FLATSIDE DOWNSTREAM
0.65
0.55
0.50
0.45
0.40
0.35
0.30
0.25
0.20
0.15
0.10
0.05
CT
0
10
20
30
40
50
60
70
80
90

GRAPH #2

CA[illegible]RON DISC
FLAT SIDE DOWNSTREAM



TEST LOG #3

BIF HYDRO-LAB CALIBRATION LOG

TEST_#3

D = 11.58 IN.
A = .795 FT.^{2}

TYPE METER:- 12" B.V. 150-B W/CAST IRON DISC FLATSIDE UPSTREAM

CALIBRATED BY:-

SIZE: ——— X BETA:- S/N:- ORDER No.:- 03422-R DATE:- 4-7-75

PIPING SKETCH

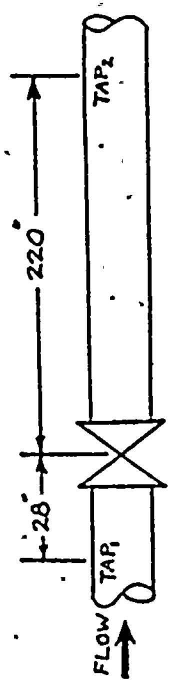
CONDITION OF PIPE:-

LEAK IN LOOP:-

ECCENTRICITY:-

MISMATCH:-

CONDITION OF METER:-



α	TIME OF DAY	TANK USED	WEIGHT (#) START	WEIGHT (#) STOP	TIME (SEC'S)	TEMPERATURE °F LINE	TEMPERATURE °F ROOM	TEMPERATURE °F MANO	ΔP_S "Hg	ΔP_P "Hg	ΔP_T "Hg	ΔP_V PSI	T_1 "H	T_2 "H	Q GPM	T_D "H	P_1 PSI	V F/S	C_T	K_V
5°	10:30	A	671#	12647#	180.102 180.102	78	70	68	73.925	.011	73.914	33.530	73	-15	479.91	528	64	1.45	.010	2371
7 1/2°	10:47		662#	18611#	180.092 180.092	78	70	69	55.949	.019	55.93	25.370	58	-14	719.67	432	64	2.18	.010	793
10°	10:58		746#	25480#	179.996 179.997	78	68	68	57.639	.035	57.604	26.130	64.	-6	1021.66	420	65	3.10	.010	404
15°	11:13		666#	39516#	180.114 180.114	78	68	67	58.721	.080	57.641	26.150	72	-1	1556.73	438	65	4.73	.010	173
20°	11:26		770#	50067#	179.951 179.952	78	69	67	55.313	.132	55.181	25.030	73	-7	1977.13	480	65	6.01	.012	103
25°	11:35		744#	49825#	146.636 146.636	79	68	67	48.495	.199	48.300	21.910	80	-25	2415.69	630	65	7.34	.018	60.4
30°	11:43		704#	49379#	129.093 129.093	79	68	68	34.927	.240	34.680	15.730	68	-30	2721.27	588	64	8.27	.024	34.2
35°	12:46		759#	49523#	116.518 116.518	79	74	69	27.423	.290	27.130	12.310	56	-34	3020.47	540	62	9.18	.028	21.7
40°	12:56		760#	49367#	108.537 108.537	79	74	70	20.91.7	.330	20.580	9.340	54	-36	3232.14	540	63	9.82	.037	14.4
45°	1:06		760#	49583#	98.571 98.571	80	74	70	16.409	.395	16.010	7.260	47	-36	3574.74	498	63	10.86	.044	9.15
50°	1:13		759#	49746#	98.957 98.957	80	74	70	11.735	.394	11.340	5.140	38	-29	3572.75	402	63	10.86	.050	6.48
55°	1:20		745#	49750#	90.051 90.052	80	73	70	10.065	.480	9.580	4.340	38	-33	3927.54	426	63	11.94	.063	4.52
60°	1:28		748#	49925#	83.201 83.201	80	73	70	8.396	.560	7.830	3.550	35	-33	4265.82	408	63	12.96	.074	3.14
65°	1:35		765#	49540#	77.320 77.320	80	73	70	6.793	.642	6.150	2.790	32	-30	4552.75	372	63	13.84	.085	2.16
70°	1:43		749#	49841#	71.809 71.809	80	72	70	5.742	.750	4.990	2.260	28	-28	4934.02	336	62	15.00	.095	1.49
75°	1:48		766#	49418#	74.983 74.983	80	72	70	3.813	.676	3.130	1.420	15	-15	4682.81	180	62	14.23	.081	1.04
80°	1:57		757#	49610#	70.974 70.974	80	72	70	3.225	0.755	2.470	1.1204	8	-7	4967.76	90	62	15.10	.051	0.73
85°	2:06		773#	49550#	70.013 70.013	80	71	70	2.631	0.782	1.849	0.838	-2	2	5028.11	-24	62	15.28	-.018	0.53
90°	10:00		777#	49485#	65.606 65.606	78	66	68	2.855	0.830	1.975	0.896	-13	13	5358.28	-156	58	16.29	-.112	0.503

100 INCH MERCURY SINGLE LEG:— 100WM

50 INCH MERCURY SINGLE LEG:— 50WM

20 INCH MERCURY MICRO-U-TUBE:- 20MM

100 INCH WATER U-TUBE:——— 100UM

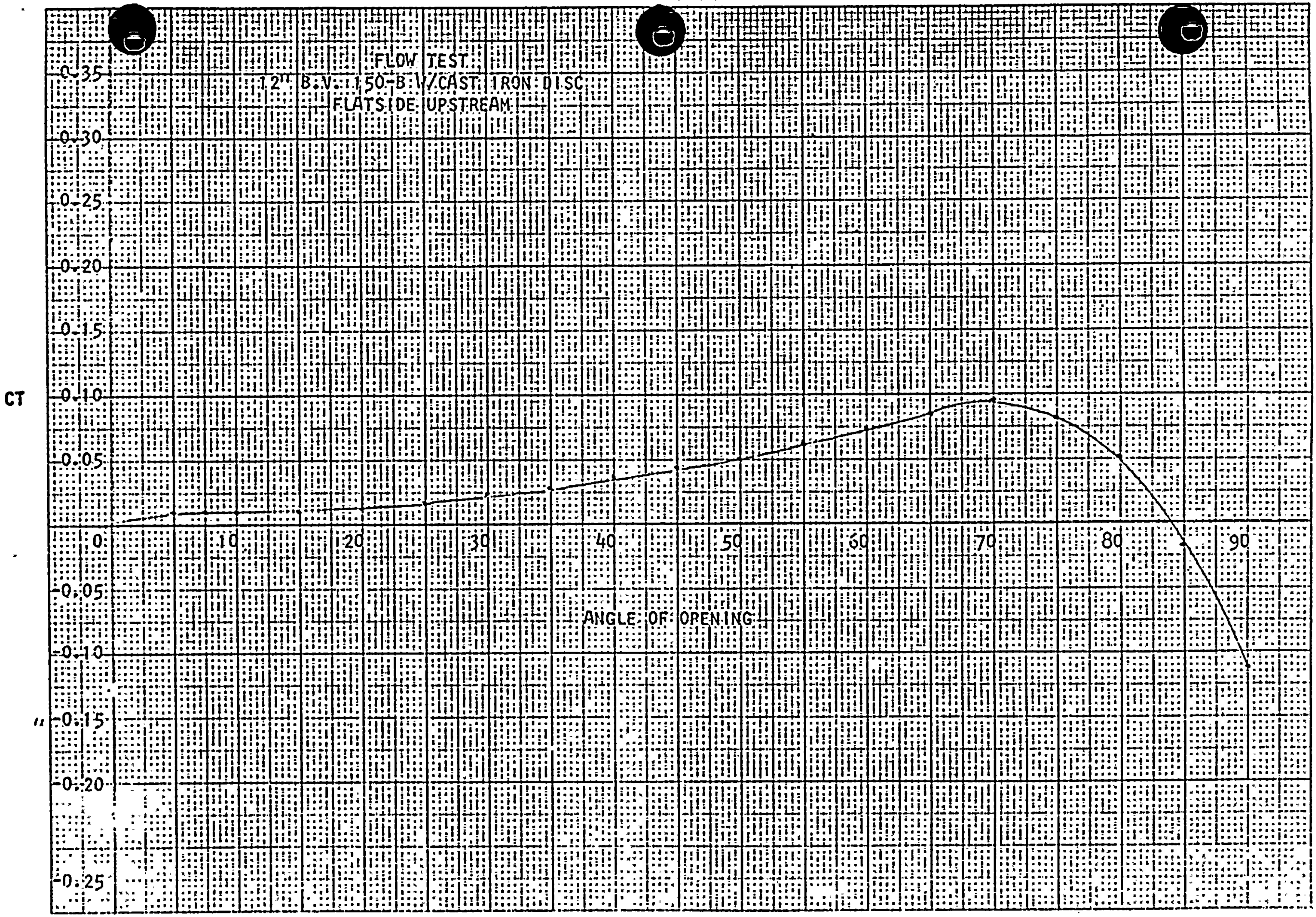
20 INCH WATER MICRO-U-TUBE:- 20MW

METER FACTOR = $\frac{0.0997(d^2)}{\sqrt{1-\beta^4}}$

R_D FACTOR = $\frac{26,528}{D}$

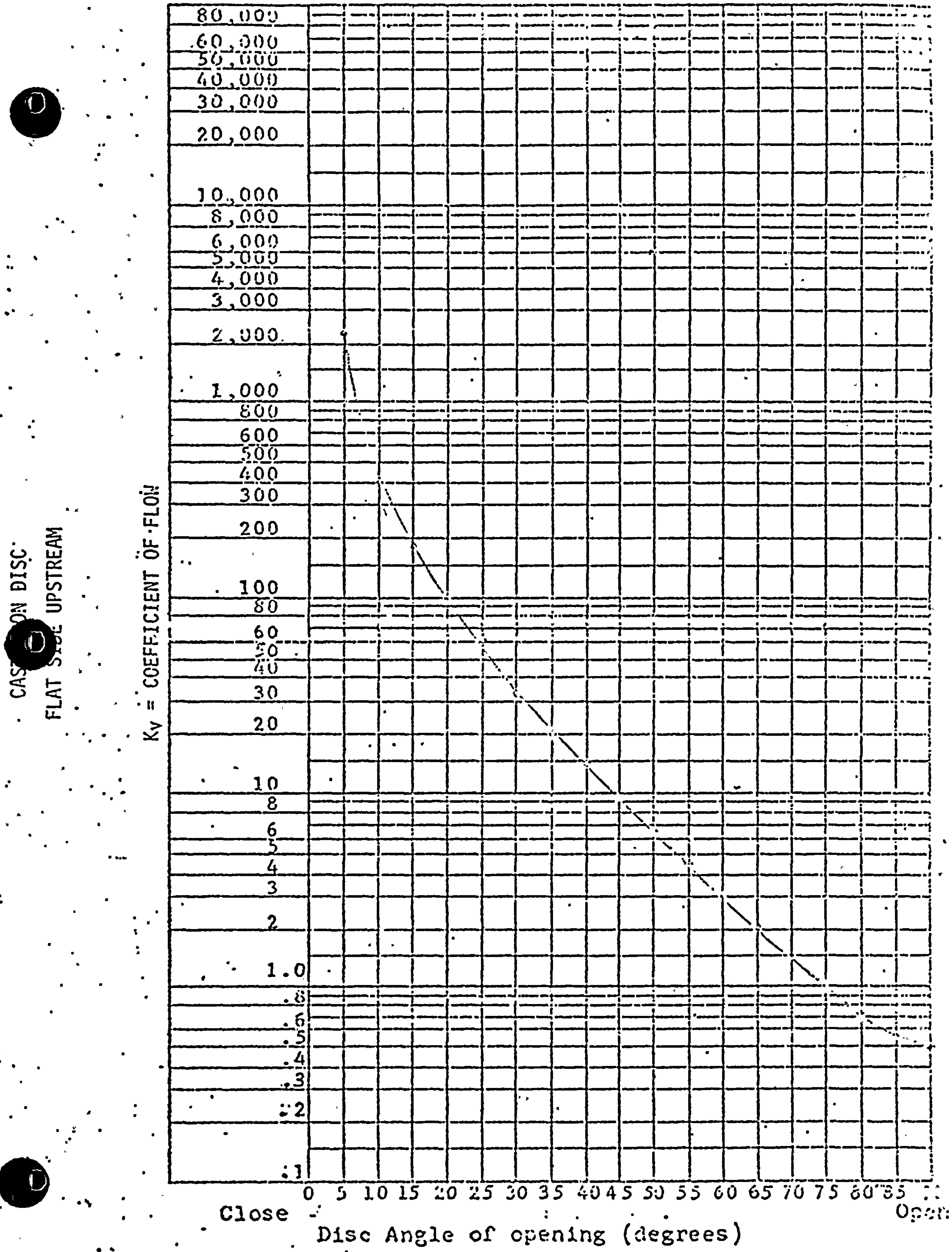
* = APPROX. C

FORM NO. 121



FLOW TEST
12" B.V. 150-B W/CAST IRON DISC
FLATSIDE UPSTREAM
CT
0.35
0.30
0.25
0.20
0.15
0.10
0.05
0
10
20
30
40
50
60
70
80
90
-0.05
-0.10
-0.15
-0.20
-0.25
ANGLE OF OPENING

GRAPH #3


CAS ON DISC
FLAT S E UPSTREAM
Kv = COEFFICIENT OF FLOW
80,000
60,000
50,000
40,000
30,000
20,000
10,000
8,000
6,000
5,000
4,000
3,000
2,000
1,000
800
600
500
400
300
200
100
80
60
50
40
30
20
10
8
6
5
4
3
2
1.0
.8
.6
.5
.4
.3
.2
.1
0 5 10 15 20 25 30 35 40 45 50 55 60 65 70 75 80 85
Close
Open
Disc Angle of opening (degrees)

TEST LOG #4

BIF HYDRO-LAB CALIBRATION LOG

TEST #4

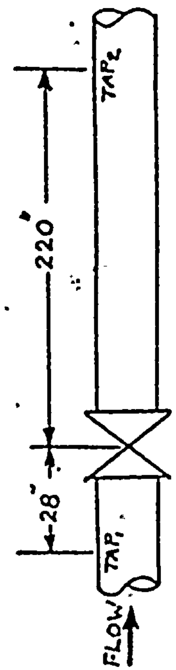
D = 11.58 IN.
A = .795 FT.^{2}

TYPE METER:- 12" B.V. 150-B W/ FABRICATED DISC FLATSIDE DOWN STREAM

CALIBRATED BY:-

SIZE: ——— X BETA:- S/N:- ORDER No.:- 0-3422-R DATE:- 4-8-75

PIPING SKETCH	α	TIME OF DAY	TANK USED	WEIGHT (#) START	WEIGHT (#) STOP	TIME (SEC'S)	TEMPERATURE °F LINE	ROOM	MANO	ΔP_s "Hg	ΔP_r "Hg	ΔP_v "Hg	ΔP_v PSI	T_1 "#	T_2 "#	Q GPM	T_D "#	P_1 PSI	V F/S	C_T	K_v
CONDITION OF PIPE:-	5°	11:08	"A"	659#	10324#	179.732 179.733	80	72	69	71.032	.012	71.023	32.220	51	3	388.10	288	65	1.17	.009	3446
LEAK IN LOOP:-	7 1/2°	11:17		765#	16950#	180.082 180.082	80	72	69	61.734	.025	61.710	27.990	56	-10	648.65	396	65	1.97	.009	1070
ECCENTRICITY:-	10°	1:14		766#	23009#	180.075 180.076	80	63	68	57.784	.033	57.755	26.200	68	-14	891.47	492	63	2.71	.012	530
MISMATCH:-	15°	1:25		744#	39035#	180.070 180.070	80	63	68	57.594	.075	57.519	26.097	45	-3	1534.70	288	62	4.66	.007	178
CONDITION OF METER:-	20°	1:36		769#	49348#	178.986 178.986	80	63	68	54.315	.127	54.183	24.583	47	-13	1958.84	360	63	5.95	.009	103
	25°	1:47		767#	49404#	136.979 136.979	80	68	68	48.663	.213	48.450	21.987	54	-14	2562.60	408	63	7.79	.011	53.8
	30°	1:53		770#	49387#	120.140 120.140	80	69	68	35.016	.274	34.740	15.760	51	-20	2920.58	426	63	8.87	.017	29.7
	35°	1:58		765#	49476#	102.494 102.495	80	69	68	27.898	.365	27.533	12.497	54	-26	3430.03	480	63	10.42	.024	17.09
	40°	2:07		757#	49604#	95.425 95.425	80	68	63	20.887	.420	20.460	9.283	54	-32	2694.40	516	63	11.23	.039	10.9
	45°	2:15		755#	49602#	87.496 87.496	80	68	63	16.519	.500	16.019	7.268	57	-39	4029.66	576	63	12.25	.051	7.20
	50°	2:21		704#	49762#	87.143 87.143	80	70	63	11.705	.590	11.119	5.043	57	-40	4062.99	582	63	12.35	.074	4.91
	55°	2:45		627#	49573#	75.626 75.626	80	72	63	10.636	.670	9.966	4.521	67	-54	4671.05	726	63	14.20	.103	3.33
	60°	2:53		765#	49880#	70.405 70.405	80	74	69	8.660	.780	7.830	3.575	79	-67	5034.77	876	63	15.30	.157	2.26
	65°	3:03		767#	49605#	68.254 68.253	80	73	69	6.547	.819	5.723	2.598	84	-75	5164.15	954	63	15.69	.238	1.56
	70	3:11		755#	49622#	62.873 62.873	81	73	69	5.715	.961	4.751	2.156	93	-82	5609.45	1050	63	17.09	.313	1.10
	75	3:17		708#	49811#	68.056 68.055	81	72	69	3.858	.830	3.028	1.373	70	-58	5207.27	768	63	15.83	.360	.814
	80	3:25		750#	49738#	67.875 67.785	81	72	69	3.296	.830	2.466	1.118	57	-46	5215.85	618	63	15.85	.359	.661
	85	3:31		764#	49433	70.348 70.348	81	72	70	2.869	.765	2.104	.954	43	-32	4993.09	450	63	15.17	.303	.615
	90°	11:41		652#	49631#	73.159 73.159	80	71	68	2.860	.719	2.141	.971	29	-23	4831.82	312	63	14.68	.206	.668



FORM NO. 121

100 INCH MERCURY SINGLE LEG:— 100 WM
50 INCH MERCURY SINGLE LEG:— 50 WM
20 INCH MERCURY MICRO-U-TUBE:- 20 MM

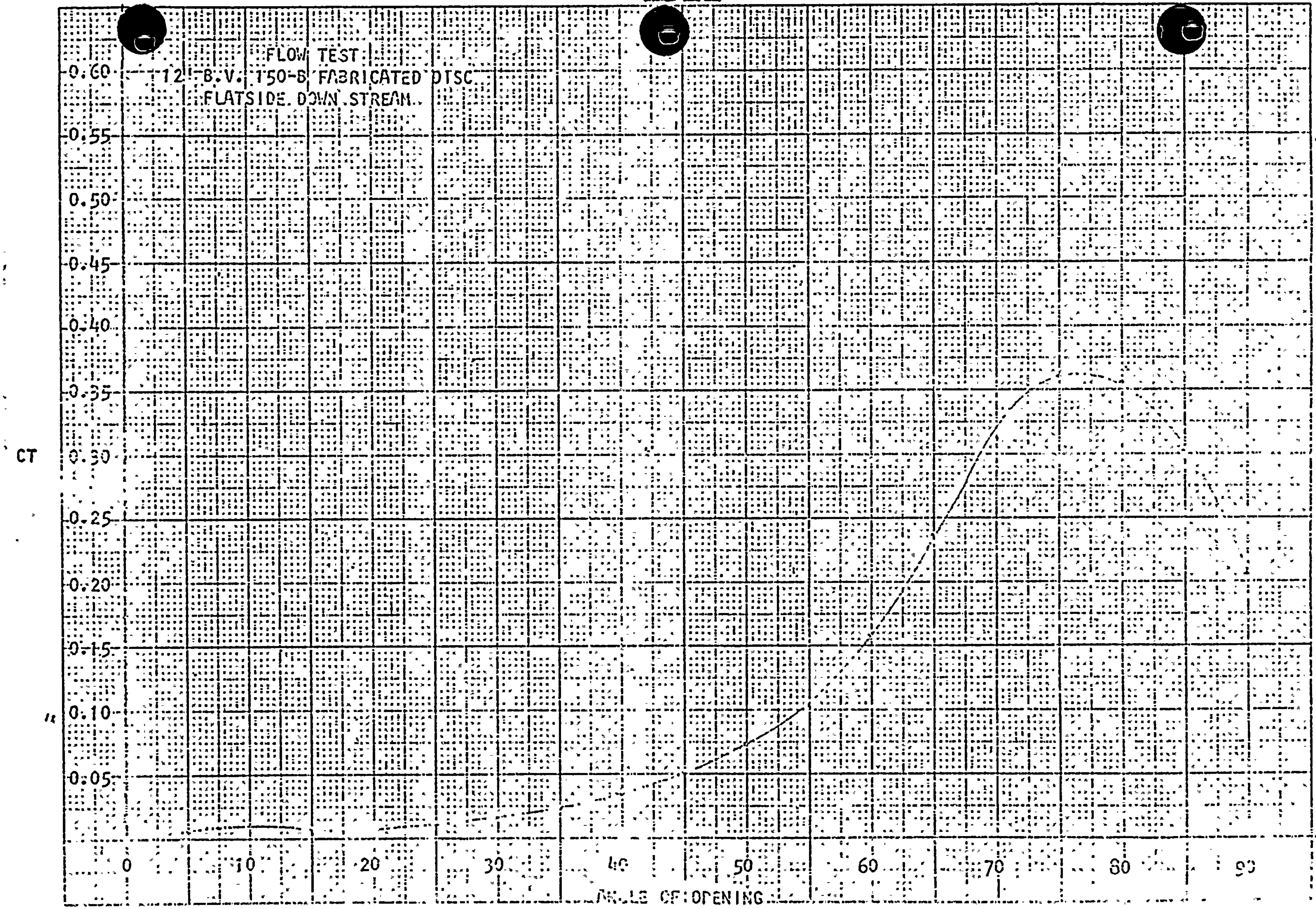
100 INCH WATER U-TUBE:——— 100 UM
20 INCH WATER MICRO-U-TUBE:- 20 MW

METER FACTOR = $\frac{0.0997(d^2)}{\sqrt{1-\beta^4}}$

R_D FACTOR = $\frac{26,528}{D}$

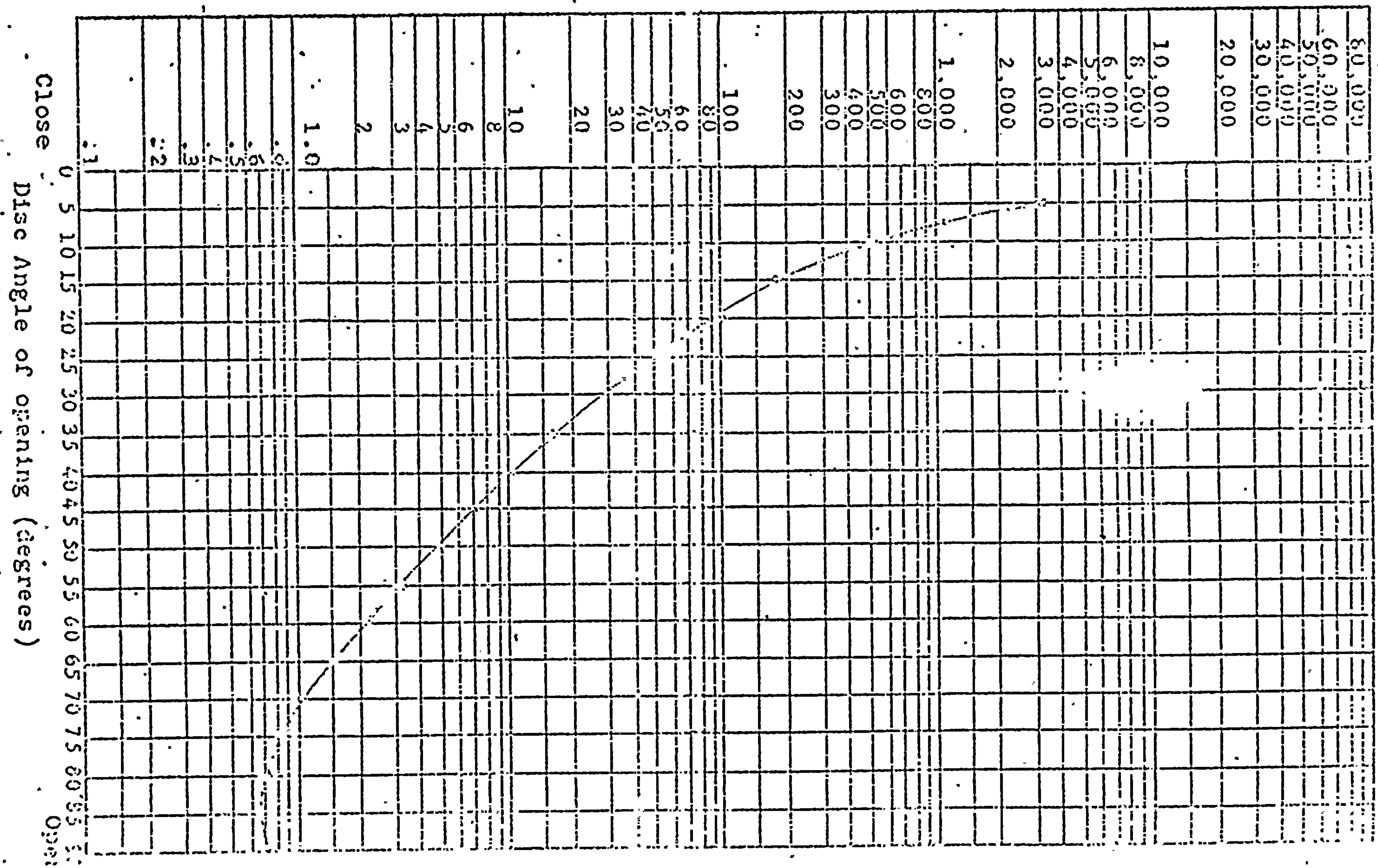
* = APPROX. C

BIF


FLOW TEST
12" B.V. T50-B FABRICATED DISC
FLATSIDE DOWN STREAM
CT
0.60
0.55
0.50
0.45
0.40
0.35
0.30
0.25
0.20
0.15
0.10
0.05
0
10
20
30
40
50
60
70
80
90
ANGLE OF OPENING

FABRICATED DISC

FLAT S DOWNSTREAM



TEST LOG #5

BIF HYDRO-LAB CALIBRATION LOG

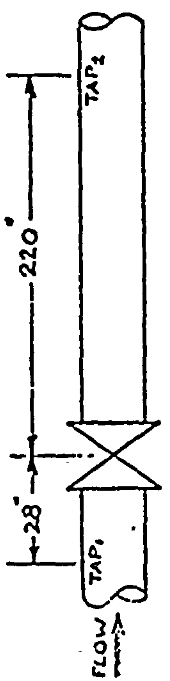
TEST #5

TYPE METER:- 12" B.V. 150-B W/FABRICATED DISC FLATSIDE UPSTREAM

SIZE: ——— X BETA:- S/N:- ORDER NO.:-

D = 11.58 IN.
A = .795 FT.^{2}
CALIBRATED BY:-
DATE:- 4-9-75

PIPING SKETCH	α	TIME OF DAY	TANK USED	WEIGHT (#) START	WEIGHT (#) STOP	TIME (SEC'S)	TEMPERATURE °F LINE	TEMPERATURE °F ROCM	TEMPERATURE °F MANO	ΔPs "H2O	ΔPp "HG	ΔPv "HG	ΔPv PSI	T1 /#	T2 /#	Q GPM	TD "#	P1 PSI	V F/S	CT	Kv
CONDITION OF PIPE:-	5°	9:40	"A"	730#	11025#	180.323 180.324	80	72	70	73.322	.014	73.308	33.261	49	26	412.04	138	62	1.25	.002	3155
LEAK IN LOOP:-	7 1/2°	9:53		767#	18927#	180.170 180.170	80	70	70	61.336	.025	61.311	27.818	40	24	727.45	96	63	2.21	.002	846.1
ECCENTRICITY:-	10°	10:07		766#	25724#	180.092 180.092	80	70	70	57.857	.037	57.830	26.238	49	10	1000.19	234	63	3.04	.005	422.1
MISMATCH:-	15°	10:19		643#	41228	180.141 180.142	80	70	70	57.806	.087	57.719	26.188	69	-7	1626.00	456	63	4.94	.011	159.3
CONDITION OF METER:-	20°	10:33		666#	49375#	167.453 167.453	80	70	70	54.099	.148	53.951	24.478	69	-7	2099.35	456	63	6.38	.011	89.36
	25°	10:42		726#	49367#	131.573 131.578	80	70	70	47.204	.230	46.974	21.313	69	-13	2667.99	492	63	8.11	.014	48.17
	30°	10:53		644#	49600#	118.070 118.070	80	70	69	34.629	.285	34.344	15.582	65	-17	2992.51	492	63	9.09	.020	27.9[illegible]
	35°	11:00		766#	49493#	103.119 103.119	80	70	69	27.849	.365	27.484	12.470	64	-25	3410.08	534	63	10.30	.027	17.2[illegible]
	40°	11:07		766#	49506#	94.547 94.545	80	70	69	20.621	.425	20.196	9.163	66	-30	3720.54	576	63	11.31	.040	10.64
	45°	11:20		767#	49455#	84.610 84.610	80	70	69	16.250	.530	15.725	7.134	69	-35	4153.07	624	63	12.62	.056	6.65[illegible]
	50°	11:26		754#	45100#	78.473 78.473	80	70	69	11.538	.510	11.028	5.003	56	-28	4078.26	504	63	12.39	.064	4.83[illegible]
	55°	11:35		767#	49757#	78.247 78.247	80	70	68	10.076	.630	9.446	4.286	57	-31	4518.65	528	60	13.73	.079	3.377
	60°	11.44		757#	49760#	72.970 72.959	80	63	68	8.317	.720	7.597	3.447	47	-22	4846.71	414	60	14.73	.077	2.361
	65°	12.44		770#	49565#	70.797 70.797	80	74	68	6.764	.760	6.004	2.724	32	-13	4974.27	270	63	15.12	.063	1.771
	70°	12.48		756#	49324#	68.496 68.496	80	74	69	5.743	.820	4.923	2.233	16	-1	5169.08	102	63	15.71	.029	1.344
	75°	12.55		760#	49832#	74.045 74.044	81	74	69	4.034	.705	3.329	1.510	-10	2	4783.07	-72	63	14.54	-.030	1.061
	80°	1:02		765#	49475#	74.129 74.130	81	74	69	3.388	.690	2.698	1.224	-21	8	4742.41	-174	62	14.41	-.091	.875
	85°	1:08		635#	49594#	73.735 73.734	81	74	69	2.940	.620	2.320	1.052	-31	16	4483.22	-282	62	13.62	-.172	.842
	90°	1:14		775#	49480#	80.097 80.097	81	72	69	2.999	.591	2.408	1.092	-38	22	4389.60	-360	62	13.34	-.212	.912



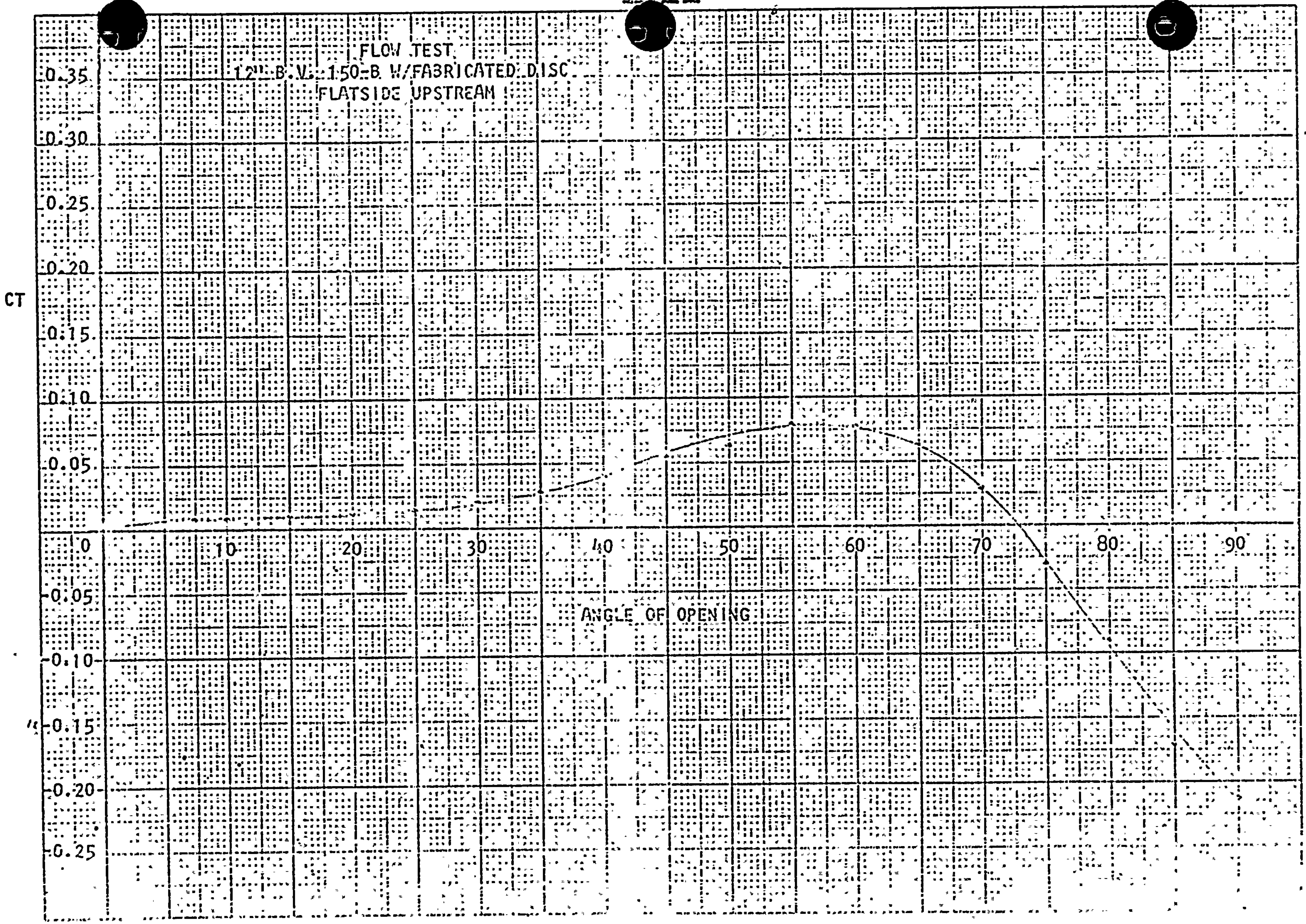
FORM NO. 121

100 INCH MERCURY SINGLE LEG:— 100 WM
50 INCH MERCURY SINGLE LEG:— 50 WM
MERCURY MICRO-U-TUBE:- 20 MM

100 INCH WATER U-TUBE:— 100 UM
20 INCH WATER MICRO-U-TUBE:- 20 MW

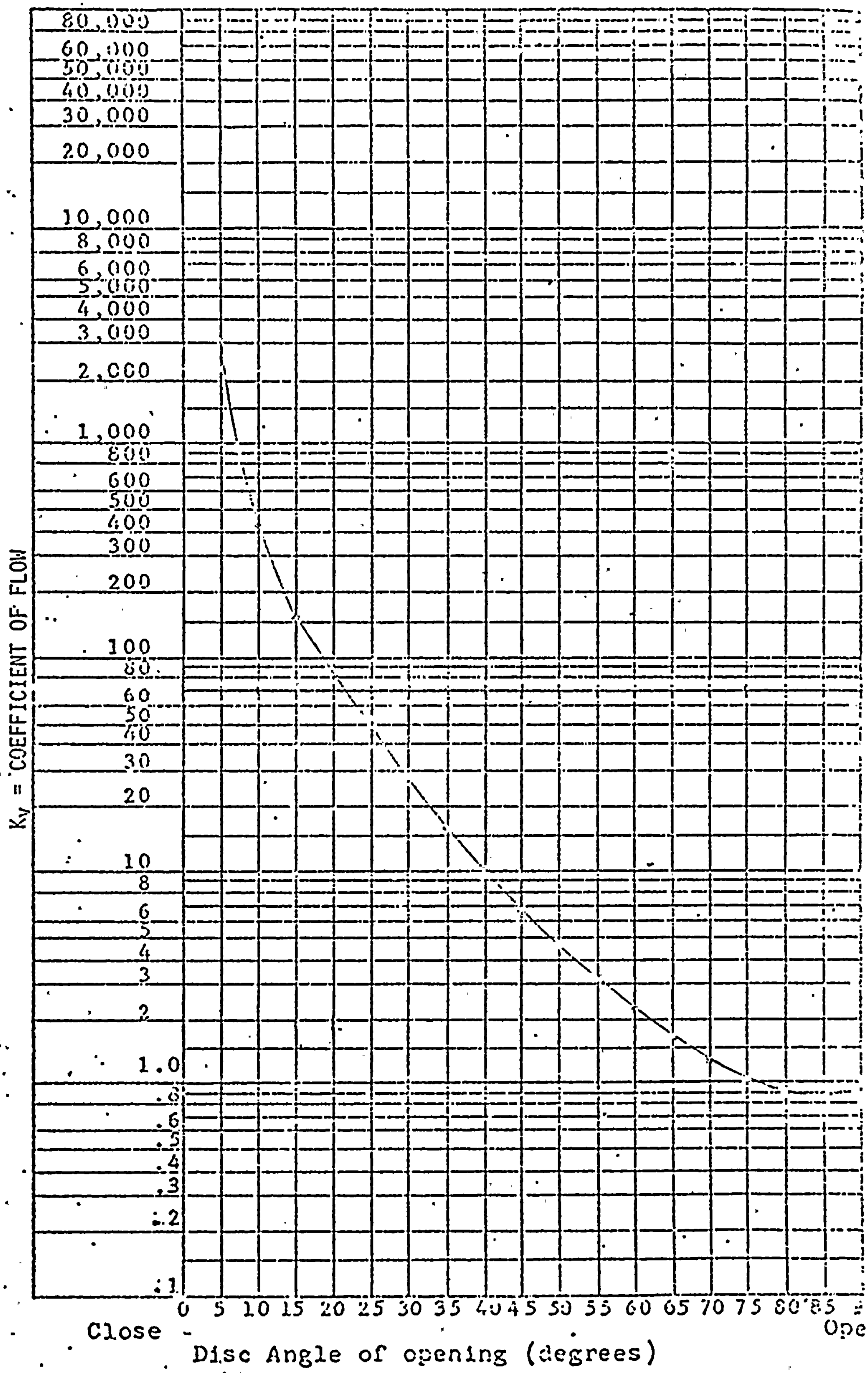
METER FACTOR = $\frac{0.0997(d^2)}{\sqrt{1-\beta^4}}$

R. FACTOR = 26.528



FLOW TEST
12" B.V. 150-B W/FABRICATED DISC
FLATSIDE UPSTREAM
CT
0.35
0.30
0.25
0.20
0.15
0.10
0.05
0
10
20
30
40
50
60
70
80
90
-0.05
-0.10
-0.15
-0.20
-0.25
ANGLE OF OPENING

FAB[illegible]TED.DISC
FLAT S[illegible]E UPSTREAM



B I F A UNIT OF GENERAL SIGNAL
1600 DIVISION ROAD
WEST WARWICK, R.I. 02893

SEISMIC ANALYSIS
OF
BUTTERFLY VALVES
FOR

EBASCO/CAROLINA POWER AND LIGHT

Prepared by: D K Das
Date: 10/28/83
Checked by: Dezso Szilagyi
Date: 11/3/83
Approved by: Dezso Szilagyi
Date: 11/3/83

SEISMIC ANALYSIS REPORT NO. N-67926
REVISION NO. A

B I F A UNIT OF GENERAL SIGNAL
1600 DIVISION ROAD
WEST WARWICK, R.I. 02893

SEISMIC ANALYSIS OF BUTTERFLY VALVES FOR

EBASCO/CAROLINA POWER AND LIGHT

Prepared by: DKDas
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SEISMIC ANALYSIS REPORT NO. N-67926

B I F A UNIT OF GENERAL SIGNAL

1600 DIVISION ROAD

WEST WARWICK, R.I. 02893

SEISMIC ANALYSIS OF BUTTERFLY VALVES

FOR

EBASCO/CAROLINA POWER AND LIGHT

VALVE SIZE 8 INCH

REPORT NO. N- 67926

BIF SHOP ORDER NO.	CUSTOMER IDENT. NO.	OPERATOR
N67926,27,48,49,73, 74,98 & 99	Ebasco Data Sht.#43,44, 47 & 48	Bettis Pneumatic N721C-SR60-12

SUMMARY OF REVISION A

NO.	DESCRIPTION					
1	Summary of results revised					
	Oper. Supp. Brack.	stress was	2588 psi ;	Revision(A)	2607	psi
	Oper. Attach. Bolt.	" "	18094 " ;	" "	18152	"
	Oper. Shaft	" "	14048 " ;	" "	14690	"
	Oper.Shaft Pin	" "	11470 " ;	" "	12048	"
	Oper.Shaft Key	" "	8916 " ;	" "	9365	"
	Valve body	" "	1358 " ;	" "	1359	"
	Brack.Plt.Weld	Load "	749 Lb/in ;	" "	753	Lb/in

4 Torque was 1648 in-Lb; Revision (A) Value is 1731 in-Lb.

8 M_x was 8408 Lb-in; Revision(A) Value 8491 Lb-in

9 was 2096 psi; Revision(A) Value 2096 psi

prin. was 2588 psi; Revision(A) Value 2607 psi

10 M_x was 8408 Lb-in; Revision(A) Value 8491 Lb-in

1 was 8833 psi; Revision(A) Value 8906 psi

prin. was 18094 psi; Revision(A) Value 18152 psi

16 f_t was 578 Lb/in; Revision(A) Value 583 Lb/in

f_s was 641 Lb/in; Revision(A) Value 646 Lb/in

f_r was 749 Lb/in; Revision(A) Value 753 Lb/in

17 W was 0.078 inch; Revision(A) Value 0.07844 ~~0.7844~~ inch

19 T was 1648 in-Lb; Revision(A) Value 1731 in-Lb

was 12529 psi; Revision(A) Value 13171 psi

t was 14048 psi; Revision(A) Value 14690 psi

SUMMARY OF REVISION A (Con't)

NO.	DESCRIPTION
	Summary of results reviewed
20	T was 1648 in-Lb; Revision(A) Value 1731 in-Lb
	was 11470 psi; Revision(A) Value 12048 psi
21	T was 1648 in-Lb; Revision(A) Value 1731 in-Lb
	S_b was 7386 psi Revision(A) Value 7758 psi
	was 3693 psi; Revision(A) Value 3879 psi
	prin. was 8916 psi; Revision(A) Value 9365 psi
23	was 479 psi; Revision(A) Value 480 psi
23 & 24	prin. was 1358 psi; Revision(A) Value 1359 psi

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SUMMARY OF RESULTS

REVISION A

ITEM	MATERIAL	ALLOWABLE (PSI)	CALCULATED (PSI)
perator Support Bracket	SA 516GR.70	17500	2607
perator Attachment Bolts	SA 193GR.B7	25000	18132
perator Attachment Plate	SA 516GR.70	17500	9246
perator Shaft	SA 564TP630	21720	14690
perator Shaft Shear Pin	SA 564TP630	21720	12048
perator Shaft Key	AISI 10.18 C.D	17500	9365
'alve Body	SA 516GR.70	17500	1359
racket Plate Weld	E 7018	9600 Lb/In.	753 Lb/In.

er Frequency f_{min} = 33 Hz f_{xy} = 423 Hz

f_t = 95 Hz

f_{xz} = 320 Hz

ANALYTICAL PROCEDURE

1. The natural frequencies of the valve operator vibrating on the the operator support are calculated. The operator is supported as a mass at the end of a cantilever beam. Since the operator can have both translation and rotation this is a two degree of freedom system. The bending and torsional frequencies of the operator system in mutually orthogonal planes are calculated. Minimum allowable natural frequency is maintained above 33Hz.

2. A static seismic analysis is performed which considers the stresses due to seismic load, dead weight, design pressure and maximum operator torque. The analysis is based on all above loads acting simultaneously, and the magnitude of resulting stresses are added together to give the worst possible loading situation.

Stresses are considered in:

a) Operator support bracket
b) Operator attachment bolts
c) Bracket plate
d) Bracket plate weld
e) Operator Disc and shaft
f) Shear pins
g) Shaft key
h) Valve body

The maximum stress limits for different materials are taken from the ASME Boiler & Pressure Vessel Code Section III.

REFERENCES

1. Ebasco Services Incorporated specification No. CAR-SH-BE-35 Revision - 3 and IEEE-344 (1975)

2. ASME Boiler & Pressure Vessel Code, Section III, 1977.

3. Oberg, E., and Jones, F., Machinery's Handbook, Nineteenth Edition, Industrial Press, Inc. 1973.

4. Roark and Young, Formulas for Stress and Strain, Fifth Edition McGraw Hill, 1975.

5. Blodgett, O., Design of Welded Structures, The James F. Lincoln Arc Welding Foundation, Fifth Printing 1972.

6 B. I. F. Drawings

Name	Number
General Arrangement Drawing	A- 902844
Disc Fab. Drawing	B- 903013

7. Operator Drawing and Catalog

Name	Number
Operator Drawing	G.H. Bettis Dwg. No. SPC-7825 Rev.A, and SK 1600
Catalog	G.H. Bettis Catalog #HD478

Inch Lug Wafer Valve with Bettis N721C-SR60-12 Operator

B General arrangement drawing A- 902844 (Pneumatic)

Job Identification Ebasco Data Sheet No. 43,44,47 & 48

BIF Shop Order Number N- 67926,27,48,49,73,74,98 & 99

Loads

Design Pressure= 45 Psi

Design Temperature= 366 °F

MAX. DYNAMIC: Torque= 1731. Lb-in A

Differential Pressure= 45 Psi

Assembly Weight= 286 Lb (Ref: 6, Gen. Arrang. Dwg.)

Operator Weight= 143 Lb (Ref: 7, Operator Dwg.)

Disc Weight = 14 Lb (Ref: 6, Disc Fab. Dwg.)

Seismic Load Factors - OBE

Horizontal = 3 g Vertical = 4 g (1 g due to self weight)

Seismic Load Factors - SSE (Ref: 1)

Horizontal = 3 g Vertical = 4 g

The operator is in position- B

NATURAL FREQUENCY CALCULATION:

The natural frequency will be that of the operator acting as a mass at the end of a cantilever beam. Since the mass can have both translation and rotation this is a two degree of freedom system. The beam supporting this mass is the operator support bracket. The length of this equivalent beam is the distance from the origin to the end of the support bracket.

FIG. 1

M = operator mass

Jx,Jy,Jz = mass moments of inertia about x,y,z Axes (Appendix B)

L = 6.75 inch

M = 0.37 $lb\text{-}S^2/in$ Jx = 127.88 $lb\text{-}in\text{-}S^2$

Jy = 126.14 $lb\text{-}in\text{-}S^2$ Jz = 3.985 $lb\text{-}in\text{-}S^2$

Natural frequencies are calculated for motion in the XZ and XY planes and for a torsional mode (rotation about X axis).

For motion in the XZ plane the natural frequencies are (Appendix A)

$$f_{1,2} = \frac{1}{2\pi}\sqrt{\left[\frac{W_1^2 + W_2^2}{2}\right] \pm \sqrt{\left[\frac{W_1^2 + W_2^2}{2}\right]^2 - \left[W_1^2W_2^2 - W_{12}^2W_{21}^2\right]}}$$

$W_1^2 = 4EI_y/J_yL$ $\qquad$ $W_2^2 = 12EI_y/ML^3$

$W_{12}^2 = 6EI_y/J_yL^2$ $\qquad$ $W_{21}^2 = 6EI_y/ML^2$

The operator support cross section is shown below.

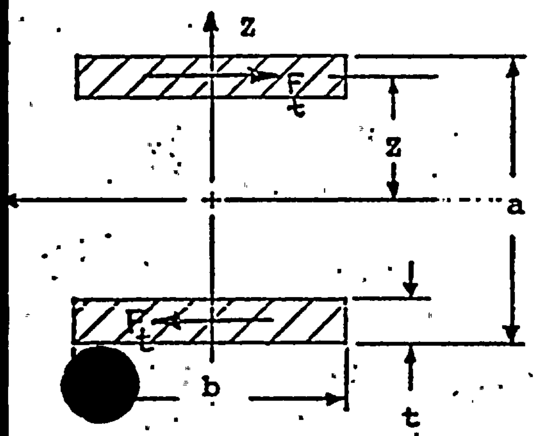


FIG. 2

z =	3.3125	inch
a =	8.0	inch
b =	4.0	inch
t =	1.375	inch

$I_y = 2\ (bt^3/12 + bt\,z^2) = 122.4\ In^4$

$I_z = 2\ (tb^3/12) = 14.67\ In^4$

$E = 29\text{x}10^6$ Psi $\qquad$ $L = 6.75$ inch

$W_1^2 = 16.67\text{x}10^6$ $\quad$ $W_2^2 = 374\text{x}10^6$ $\quad$ $W_{12}^2 = 3.7\text{x}10^6$ $\quad$ $W_{21}^2 = 1263\text{x}10^6$

$f_{xz} = 320$ Hz

Negative sign gives the minimum frequency.

motion in the Xy plane the formulas are the same except J_z and are used.

$J_z = 3.985$ Lb-s^2-in $\quad I_z = 14.67$ In^4

$W_1^2 = 63.26\times10^6$ $W_2^2 = 44.86\times10^6$ $W_{12}^2 = 14.06\times10^6$ $W_{21}^2 = 151\times10^6$

$f_{xy} = 423$ Hz

A third possible mode is primarily a torsional mode of the operator on the operator support bracket. The torsional reaction is a shear force on the valve feet.

$f_t = \frac{1}{2\pi}\sqrt{K_T / J_x}$ $\quad J_x = 127.88$ $\quad K_T = M_t/\theta = 2F_tZ/\theta$

$\theta = F_tL^3/3EIZ.$ $\quad$ Where $I = tb^3/12 = 7.33$ In^4.

$K_t = 6Z^2IE/L^3 = 45.52\times10^6$ Lb-In.

$f_t = 95$ Hz

The minimum frequency is greater than 33 Hz.

<u>LOADS</u>

The seismic load factors are:

	SSE	OBE	
Horizontal	3g	3g	
Vertical	3±1=4g max.	3±1=4g max.	(1g due to self weight)

The design pressure and temperature are:

$P = 45$ Psi $\quad T = 366$ ^oF

STRESS ANALYSIS

OPERATOR SUPPORT BRACKET

Stresses are calculated in the operator support bracket by calculating the reactions to the seismic loading, dead weight and operator torque at the intersection of the valve body and support bracket. Moments and forces can act in all possible directions. For the worst case forces and moments must act such that the combined loadings are additive in magnitude.

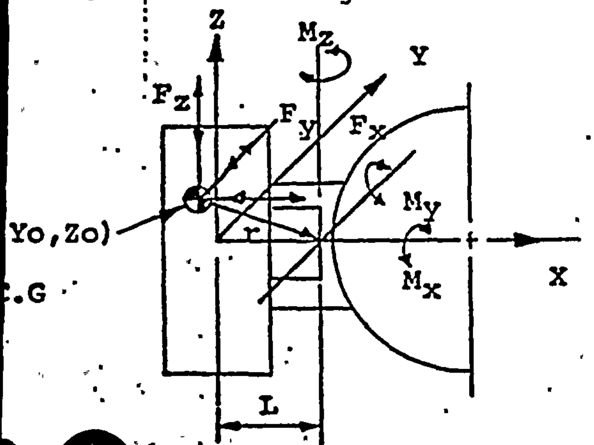


FIG.3

CG=Location(Xo,Yo,Zo)

T=Operator Torque

W=Operator Weight

g_h, g_v=Horizontal & Vertical Seismic Coefficients

Inertia and dead load $\vec{F}=\hat{i}Fx+\hat{j}Fy+\hat{k}Fz=\hat{i}(\pm g_h W)+\hat{j}(\pm g_h W)+\hat{k}\left[\pm(g_v+1)\right]W$

$|F_x|=$ 429 lb $|F_y|=$ 429 lb $|F_z|=$ 572 lb

Moment arm $\vec{r}=\hat{i}(L+Xo)+\hat{j}Yo+\hat{k}Zo$ and Moment $\vec{M}=\vec{F}\times\vec{r}=\hat{i}Mx+\hat{j}My+\hat{k}Mz$

$\vec{r}=\hat{i}$ (6.75) + $\hat{j}$ (2.39) + $\hat{k}$ (12.57)

Resultant Mx=(±FyZo)-(±FzYo)+ operator torque T=FyZo+FzYo+T

$|Mx|=$ 429x12.57 + 572x2.39 + 1731 = 8491 Lb-in (A)

Similarly $My=F_xZ_o+(L+Xo)Fz=$

$|My|=$ 9253 Lb-in

and Mz=FzYo+Fy(L+Xo)=

$|M_z|=$ 4263 Lb-in

The absolute value of each term is calculated and added together.

From Fig. 2 the operator support cross section at the valve body has the properties

$Iy = 122.4 \text{ in}^4$ $\quad A = 2bt = 11 \text{ in}^2$

$Iz = 14.67 \text{ in}^4$

Normal Stress: $\sigma = \pm \dfrac{Fx}{A} \pm \dfrac{My\,a}{2Iy} \pm \dfrac{Mz\,b}{2Iz}$ Psi

Add the magnitude of all the stresses for the worst condition.

$\sigma = 39 + 302 + 581 = 922$ Psi

Shear Stress: $\tau = \pm \dfrac{\left(\frac{Mx}{2}\right)\left[3\left(\frac{b}{2}\right) + 1.8\left(\frac{t}{2}\right)\right]}{8\left(\frac{b}{2}\right)^2\left(\frac{t}{2}\right)^2} \pm \dfrac{\sqrt{Fy^2 + Fz^2}}{A}$

(Ref:4)

$\tau = 2031 + 65 = 2096$ Psi A

Principal Stress: $\sigma \text{ prin.} = \dfrac{\sigma}{2} \pm \sqrt{\left(\dfrac{\sigma}{2}\right)^2 + \tau^2}$

$\sigma_{prin} = 2607$ Psi A

The material of the support bracket is SA 516 Gr. 70

The allowable design stress Sm is 17500 Psi.

Maximum stress developed σ prin. = 2607 Psi A $<$ Allowable stress Sm= 17500 Psi

Therefore the design is safe for the specified seismic loads.

OPERATOR ATTACHMENT BOLTS

The operator is attached to the valve feet by four bolts as shown below in figure 4.

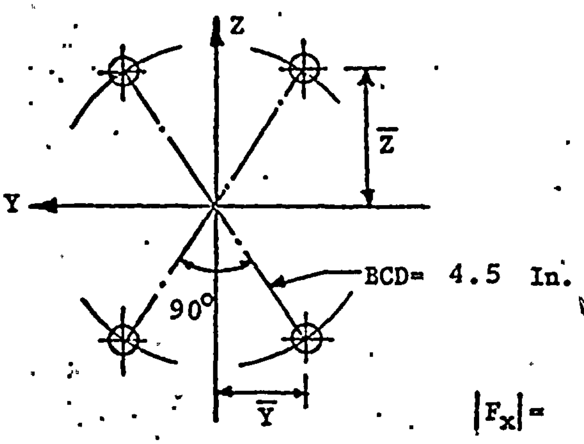


Fig. 4

No. of bolts N = 4
Size of the bolt=0.5 inch
Nom. dia -13 UNC

The bolt root area= 0.126 In^2 (Ref:3)

$\overline{Y}$ = 1.59 In. $\overline{Z}$ = 1.59 In.

The forces acting on the bolts are

$|F_x|$ = 429 Lb, $|F_y|$ = 429 Lb, $|F_z|$ = 572 Lb

The moments at the bolts are calculated using the same formulas as for the valve feet described on page 8 except L = 3.94 inch.

$|M_x|$ = 8491 Lb-In. A

$|M_y| = \pm F_x Z_o \pm F_z (L + X_o) =$ 7646 Lb-In.

$|M_z| = \pm F_x Y_o \pm F_y (L + X_o) =$ 2716 Lb-In.

The tensile force on the bolt is $F_t = \pm \frac{F_x}{4} \pm \frac{M_y}{4\overline{Z}} \pm \frac{M_z}{4\overline{Y}}$.

F_t = 1736 Lb

The tensile stress in the bolt is $\sigma = \frac{F_t}{A} =$ 13782 psi

Shear Stress:

$$\tau = \left[\frac{\sqrt{F_Y^2 + F_Z^2}}{NA} + \frac{M_X}{NRA} \right]$$ where R= Bolt circle radius = 2.25 inch

τ = 1418 + 7488 = 8906 Psi A

Principal Stress:

$$\sigma_{prin.} = \frac{\sigma}{2} \pm \sqrt{\left(\frac{\sigma}{2}\right)^2 + \tau^2}$$

$\sigma_{prin.}$ = 6891 + 11260 = 18152 Psi A

[illegible] material is SA 193 Gr. B7

Allowable stress is Sm = 25000 psi

Principal stress $\sigma_{prin.}$ = 18152 psi < Allowable stress Sm = 25000 psi A

Therefore the design is safe for the specified seismic loads.

BRACKET PLATE ANALYSIS (BOLT CIRCLE WITHIN THE VALVE FEET)

The bracket plate is welded to the valve feet as shown below. The bolt orientation drawn below gives the worst loading on the plate due to the tensile pull on the bolt.

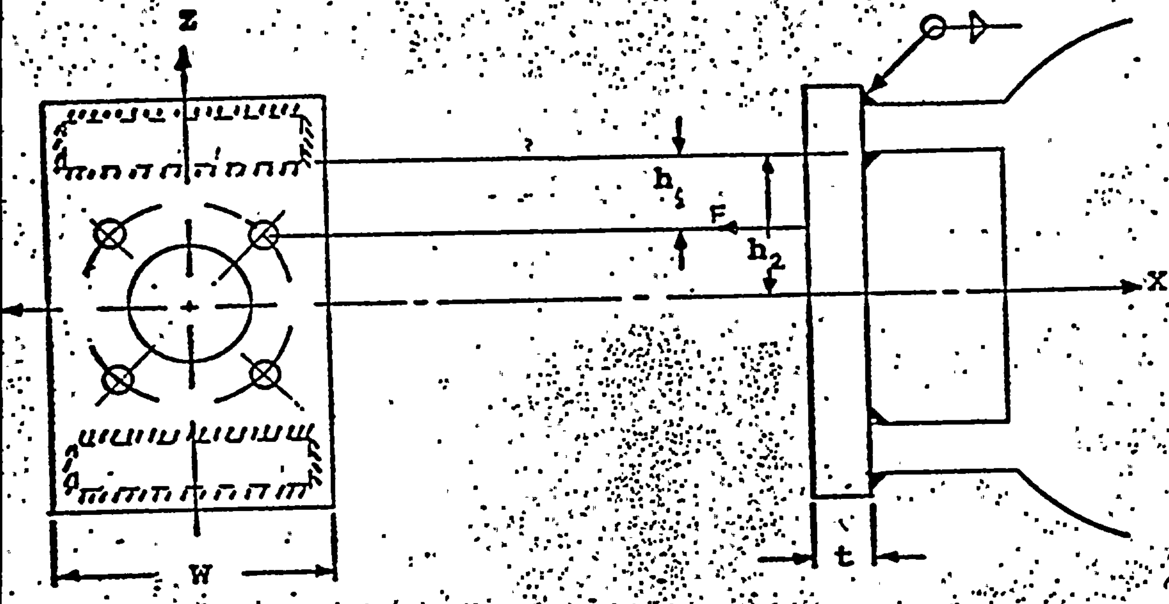


FIG. 5

The moments acting on the bracket plate are calculated using the same formulas as for the operator support except $L = 3.565$ inch

$M_y = F_x Z_o + F_z (L + X_o) = 7432$ Lb-in

$t = 0.75$ inch

$W = 4.5$ inch

$h_1 = 1.035$ inch

$h_2 = 2.625$ inch

$M_z = F_x Y_o + F_y (L + X_o) = 2555$ Lb-in

Bending stress in the plate due to bolt load at the weld juncture:

$$\sigma_b = \frac{F_{bolt}\,(h_1)\quad 2}{(1/6)\,Wt^2}$$

Where F_{bolt} = Maximum bolt stress x Ar

$= 13782 \times 0.126 = 1736$ lb

$\sigma_b = 8518$ Psi

Shear stress $\tau = \dfrac{\sqrt{F_x^2 + F_y^2}}{2Wt} + \dfrac{M_y}{h_2 W t} + \dfrac{M_z\,(1.5W+0.9t)}{W^2 t^2}$

$\tau = 90 + 839 + 1665 = 2594$ Psi

Principal stress $\sigma_{prin.} = \dfrac{\sigma_b}{2} \pm \sqrt{\left(\dfrac{\sigma_b}{2}\right)^2 + \tau^2}$

$\sigma_{prin.} = 4259 + 4987 = 9246$ Psi

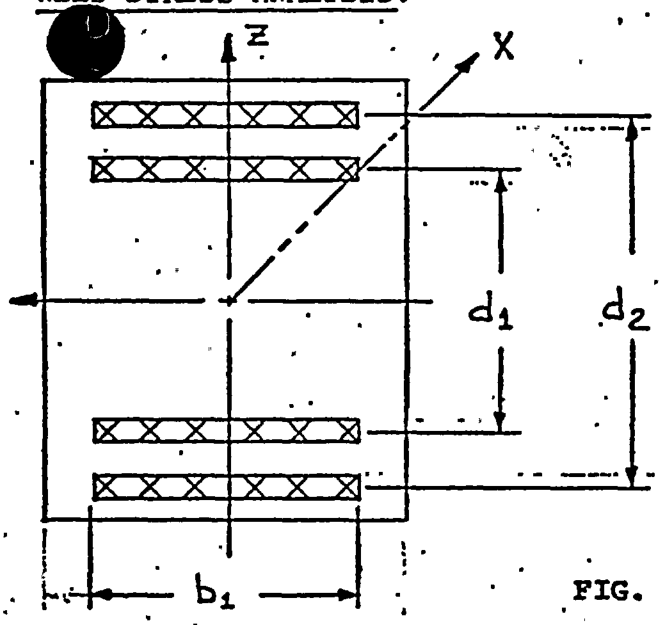
The plate material is SA 516 gr 70

Allowable stress is $S_m = 17500$ Psi

Principal stress $\sigma_{prin.} = 9246$ Psi $<$ Allow. stress $S_m = 17500$ Psi

Therefore the design is safe for the specified seismic loads.

WELD STRESS ANALYSIS:



b_1 = 4.0 inch

d_1 = 5.25 inch

d_2 = 8.0 inch

FIG. 6

The moments acting on the weld are calculated using the same formulas as for the operator support except L = 3.19 inch.

$M_y = F_x Z_o + F_z \ (L+X_o) =$ 7217 Lb-in

$M_z = F_x Y_o + F_y \ (L+X_o) =$ 2394 Lb-in

Treating the weld as a line (Ref.5) total length of the weld $A_W =$ 16 inch.

X-direction axial tension $= \frac{F_x}{A_W} =$ 27 lb/in

Y-direction shear $= \frac{F_y}{A_W} =$ 27 lb/in

Z-direction shear $= \frac{F_z}{A_W} =$ 36 lb/in

Section modulus S_w of the weld:

S_{wy} about Y-axis $= b_1 \quad (d_1 + d_2) \quad = \quad 53 \quad in^2$

S_{wz} about Z-axis $= \dfrac{2\, b_1^2}{3} \quad = \quad 10.667 \quad in^2$

Longitudinal force per unit length carried by the weld are given by

$$f_y = \frac{M_y}{S_{wy}} = \quad 136 \quad lb/in$$

$$f_z = \frac{M_z}{S_{wz}} = \quad 224 \quad lb/in$$

Torque on the weld due to M_x:

Torsional section modulus $J_w = J_{w1} + J_{w2}$

$$= \frac{b_1^3 + 3b_1 d_1^2}{6} + \frac{b_1^3 + 3b_1 d_2^2}{6} \quad in^3$$

$$= \quad 65.8 \quad + \quad 138.7 \quad = \quad 204.5 \quad in^3$$

The weld force is $f_t = f_{t1} + f_{t2} = M_x \left[\dfrac{d_1}{2J_{w1}} + \dfrac{d_2}{2J_{w2}} \right]$ lb/in

$$f_t = 8491 \left[0.0399 + 0.0288 \right] = 583 \quad \text{A} \quad \text{lb/in}$$

Total normal load carried by the weld $f_n = \dfrac{F_x}{A_w} + f_y + f_z$ lb/in.

$$f_n = 27 + 136 + 224 = 387 \text{ lb/in}$$

Total shear load carried by the weld $f_s = \dfrac{F_y}{A_w} + \dfrac{F_z}{A_w} + f_t$ lb/in

$$f_s = 27 + 36 + 583 = 646 \text{ lb/in} \quad \text{A}$$

The resultant load $f_r = \sqrt{f_n^2 + f_s^2} = 753$ A lb/in

Required leg size $W = \dfrac{\text{Actual force}}{\text{Allowable force}}$

Allowable force for E7018 electrode in fillet weld is 9600 lb/in

Therefore the leg size is $W = \frac{f_r}{9600} = 0.07844$ inch (A)

Actual leg size W_a provided = 0.5625 inch

Hence actual leg size W_a = 0.5625 inch > Required leg size W = 0.07844 inch (A)

Therefore the weld size is adequate for the specified seismic conditions.

DISC AND SHAFT ANALYSIS:

W_d = Disc weight = 14 Lb

G_m=Resultant seismic acceleration acting on the disc

$$G_m = \sqrt{g_h^2 + g_v^2} = 5$$

As the section modulus of the shaft is always less than that of the disc hub combination if failure would occur it would be at the shaft.

D_d = Disc diameter = 7.05 inch

D = Diameter of the shaft = 0.875 inch

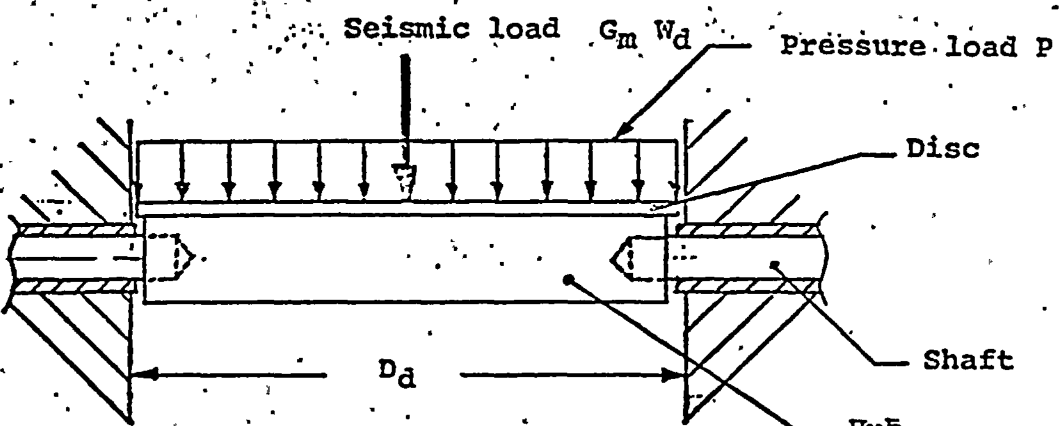


FIG. 7

$$\text{Total shear force } F = G_m W_d + \frac{\pi D_d^2 P}{4} = 70 + 1757 = 1827 \text{ Lb}$$

Cross sectional area $A = \pi D^2/4 = 0.6013 \text{ in}^2$

$$\text{Shear stress } \tau_1 = \frac{F}{2A} = 1519 \text{ Psi}$$

The shear stress in the operator shaft is developed by the operator torque T. $T = 1731$ Lb-in A

Shear stress $\tau_2 = \dfrac{TD}{2J}$ where $J = \pi D^4/32 = 0.0575$ in^4

$\tau_2 = 13171$ Psi A

Total Shear stress $\tau_t = 1519 + 13171 = 14690$ Psi A

The shaft material is SA 564 TP630 Age hardened at 1075°F

Allowable stress is $S_m = 36200$ Psi

Allowable shear stress $S = 0.6\ S_m = 21720$ Psi (Ref: 3)

Shear stress $\tau_t = 14690$ Psi A $<$ Allowable stress $S = 21720$ Psi

Hence the design is safe for specified seismic loads.

OPERATOR SHAFT PIN

The disc attaches to the shaft with taper pins. These pins are subjected to double shear due to the operator torque.

The shear area is an ellipse with major axis b and minor axis d. (Appendix C)

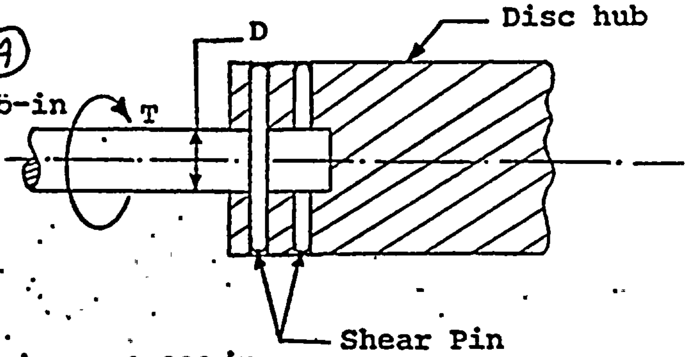


FIG.8

(A)

T = operator torque = 1731 Lb-in

D = shaft diameter = 0.875 in

N_p = number of pins = 2

d = mean diameter of the taper pin = 0.232 in

shear area $A_s = (\pi/4)\, d \sqrt{dD} = 0.0821 \text{ in}^2$

Shear stress $\tau = \dfrac{2T}{D} \times \dfrac{1}{2N_pA_s} = 12048$ Psi (A)

The pin material is: SA 564 TP630 Age-Hardened at 1075°F

Normal design allowable stress S_m = 36200 Psi

The allowable pin shear stress $S = 0.6\, S_m$ = 21720 Psi (Ref:3)

Actual shear stress τ = 12048 Psi $<$ Allowable shear stress S = 21720 Psi

(A)

Therefore the design is safe for the specified operator torque.

Shaft Key Analysis

The valve shaft is attached to the operator adapter through key whose dimensions are t x b x L;

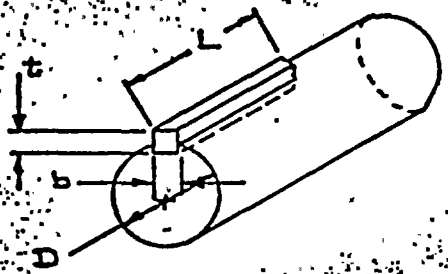


FIG: 9

t = 0.1875 inch

b = 0.1875 inch

L = 5.44 inch

D = 0.875 inch

T = operator torque = 1731 Lb-in A

The shear stress in the key is $\tau = \frac{2T}{bLD}$ = 3879 Psi A

The bearing stress in the key is $Sb = \frac{4T}{tLD}$ = 7758 Psi A

Principal Stress $\sigma_{prin.} = \frac{S_b}{2} + \sqrt{\left(\frac{S_b}{2}\right)^2 + \tau^2}$ = 9365 Psi A

Key material: AISI 1018 Cold drawn   Allowable Stress Sm= 17500 Psi

The allowable shearing stress S = 0.6 Sm = 10500 Psi

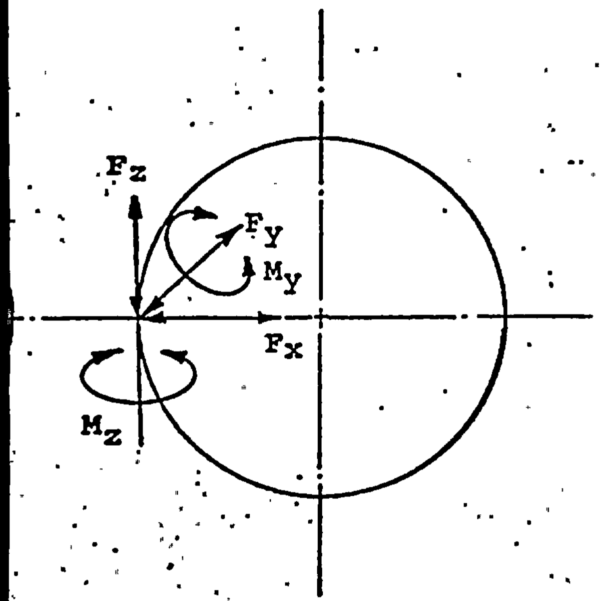
Shear Stress τ = 3879 Psi A < Allowable Stress S = 10500 Psi

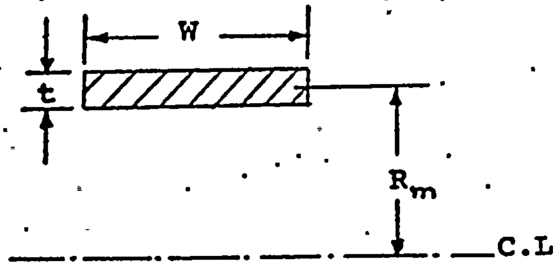
Principal Stress $\sigma_{prin.}$ = 9365 Psi A < Allowable Stress S_m = 17500 Psi

Therefore the design is safe for the specified operator torque.

VALVE BODY ANALYSIS:

The valve body is modeled as a ring subjected to forces and moments as shown in the figures below.. (Ref. 4)





Valve body cross section

$W =$ 4.0 inch

$t =$ 2.685 inch

$R_o =$ 6.75 inch (outer)

$R_m =$ 5.4075 inch (mean)

FIG. 10

Membrane stresses are developed due to the internal pressure and F_X.

$$\sigma_m = \frac{PR_m}{t} + \frac{F_X}{2Wt}$$

$$= 91 + 20 = 111 \text{ Psi}$$

Shear stresses are generated due to the application of force F_Y and moments M_X and M_Z. Contributions of each of these are computed and added together on the next page.

$$\frac{\left(\frac{M_z}{2}\right)\left[3\left(\frac{W}{2}\right) + 1.8\left(\frac{t}{2}\right)\right]}{8\left(\frac{W}{2}\right)^2\left(\frac{t}{2}\right)^2} + \frac{F_Y}{2Wt} + \frac{(M_x/2z)}{2Wt} \quad \text{Psi}$$

τ = 400 + 20 + 60 = 480 Psi (A)

Bending stresses in the valve body are caused due to the radial load F_x and the moment M_y.

$$\sigma_b = \frac{0.239 F_x R_m}{\frac{1}{6} W t^2} + \frac{(M_y/2)}{\frac{1}{6} W t^2} \quad \text{Psi}$$

σ_b = 115 + 963 = 1078 Psi

Therefore the total normal stress is obtained by the summation of the membrane and bending stress.

$\sigma_n = \sigma_b + \sigma_m$ = 1078 + 111 = 1189 Psi

Principal stress $\sigma_{prin.} = \frac{\sigma_n}{2} \pm \sqrt{\left(\frac{\sigma_n}{2}\right)^2 + \tau^2}$

$\sigma_{prin.}$ = 595 + 764 = 1359 Psi (A)

The material of the valve body is SA 516 Gr. 70

The allowable design stress S_m is 17500 Psi

Maximum stress developed $\sigma_{prin.}$ = 1359 A psi $<$ Allowable stress S_m = 17500 psi

Therefore the design is safe for the specified seismic loads.

APPENDICES

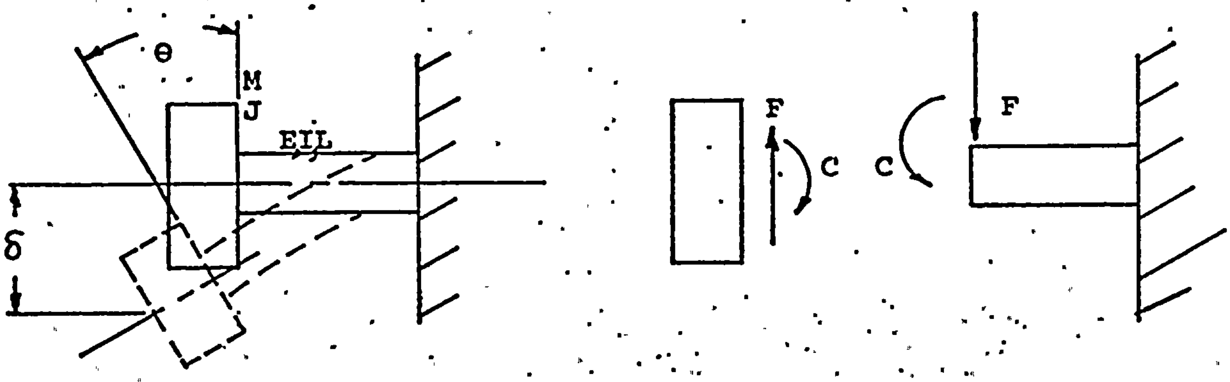
A. NATURAL FREQUENCY FORMULA

B. INERTIA FORMULAS AND TABULATION OF RESULTS

C. SHEAR PIN AREA

Appendix A - Natural Frequency

Model of a mass with rotary inertia on the end of a cantilever am. This is a two degree of freedom system.



$$\delta = FL^3/3EI + CL^2/2EI$$

$$\theta = FL^2/2EI + CL/EI \qquad F = \frac{12EI}{L^3}(\delta - \theta L/2)$$

or $$C = \frac{4EI}{L}(\theta - 3\delta/2L)$$

tions of motion

$$J\ddot{\theta} = -C = -4EI\,\theta/L + 6EI\delta/L^2$$

$$M\ddot{\delta} = -F = -12EI\delta/L^3 + 6EI\,\theta/L^2$$

Using $\theta = \bar{\theta}e^{iwt}$ $\delta = \bar{\delta}e^{iwt}$ for simple harmonic motion

or $\theta\,(-W^2 + W_1^2) - \bar{\delta}W_{12}^2 = 0$

$-\bar{\theta}\text{-}W_{21}^2 + \bar{\delta}(-W^2 + W_2^2) = 0$

$W_1^2 = 4EI/JL$ $W_2^2 = 12EI/ML^3$ $W_{12}^2 = 6EI/JL^2$

$W_{21}^2 = 6EI/ML^2$

For $\bar{\theta} \neq 0$, $\bar{\delta} \neq 0$

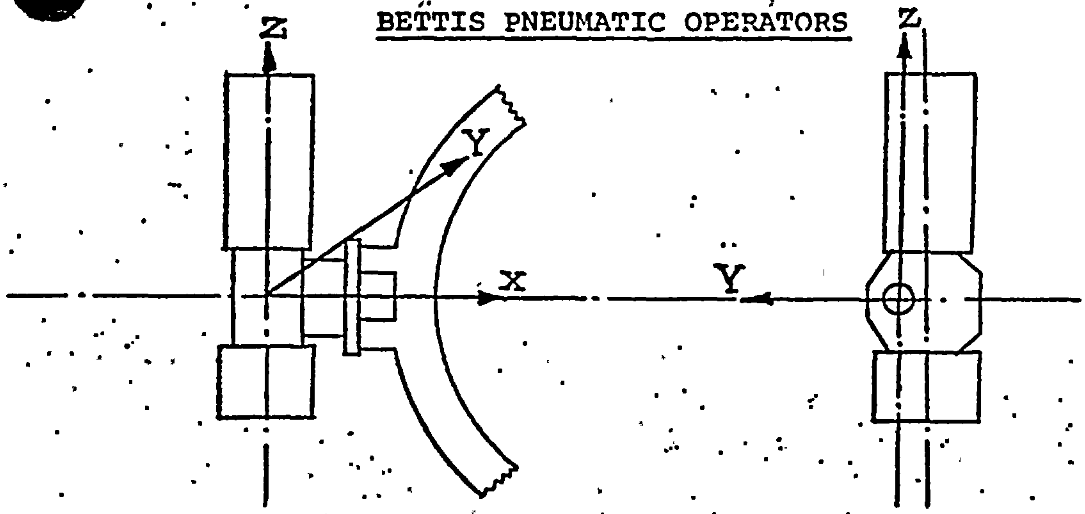
$$W^4 - (W_1^2 + W_2^2)\,W^2 + (W_1^2W_2^2 - W_{12}^2W_{21}^2) = 0$$

$$f = W/2\pi$$

$$f_{1,2} = \frac{1}{2\pi}\sqrt{\frac{W_1^2 + W_2^2}{2} \pm \sqrt{\left[\frac{W_1^2 + W_2^2}{2}\right]^2 - (W_1^2W_2^2 - W_{12}^2W_{21}^2)}}$$

APPENDIX B - INERTIA FORMULAS FOR OPERATORS

BETTIS PNEUMATIC OPERATORS



The X Y Z coordinate system is located in line with the axes of symmetry of the valve. The center of gravity location of different components of the operator are given below.

Total Operator	(X_o, Y_o, Z_o)
Spring Cylinder	(X_s, Y_s, Z_s)
Guide Cylinder	(X_g, Y_g, Z_g)
Cylinder Support	(X_p, Y_p, Z_p)

Total Operator Mass = M_o

Spring Cylinder mass = M_s

Guide Cylinder mass = M_g

Cylinder Support mass = M_p

The mass moment of inertia of the cylinders and the cylinder support about the axes parallel to the coordinate system X,Y and Z through their respective center of gravity are:

Spring cylinder of length L_s and radius R_s

Moment of inertia $J_{xs} = J_{ys} = M_s \ (R_s{}^2/4 + L_s{}^2/12)$

$J_{zs} = (M_s) \ (R_s{}^2)/4$

Guide cylinder of length L_g and radius R_g

Moment of inertia $J_{xg} = J_{yg} = M_g \ (R_g{}^2/4 + L_g{}^2/12)$

$J_{zg} = (M_g) \ (R_g{}^2)/4$

The cylinder support is equivalent to a rectangular prism of height H_p and cross sectional dimensions W_x and W_y.

Moment of inertia $J_{xp} = (M_p/12) \ (H_p{}^2 + W_x{}^2)$

$J_{yp} = (M_p/12) \ (H_p{}^2 + W_y{}^2)$

$J_{zp} = (M_p/12) \ (W_x{}^2 + W_y{}^2)$

Finally the moment of inertia of the entire operator about the coordinate axes X,Y and Z are:

$$J_x = J_{xs} + M_s(Y_s^2+Z_s^2) + J_{xg} + M_g(Y_g^2+Z_g^2) + J_{xp} + M_p(Y_p^2+Z_p^2)$$

$$J_y = J_{ys} + M_s(X_s^2+Z_s^2) + J_{yg}+M_g(X_g^2+Z_g^2) + J_{yp}+M_p(X_p^2+Z_p^2)$$

$$J_z = J_{zs} + M_s(X_s^2+Y_s^2) + J_{zg}+M_g(X_g^2+Y_g^2) + J_{zp}+M_p(X_p^2+Y_p^2)$$

W Ope	M_o	X_o	Y_o	Z_o	M_s	X_s	Y_s	Z_s	M_g
143	0.37	0.0	-2.39	12.57	0.25	0.0	-2.5	18.42	0.01
Lb	$\frac{Lb-S^2}{Inch}$	Inch	Inch	Inch	$\frac{Lb-S^2}{Inch}$	Inch	Inch	Inch	$\frac{Lb-S^2}{Inch}$

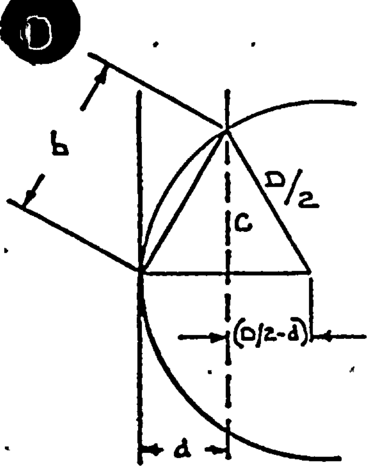
X_g	Y_g	Z_g	M_p	X_p	Y_p	Z_p	L_s	R_s	L_g
0.0	-2.5	-9.94	0.11	0.0	-2.13	0.0	30.00	3.75	12.5
Inch	Inch	Inch	$\frac{Lb-S^2}{Inch}$	Inch	Inch	Inch	Inch	Inch	Inch

R_g	H_p	W_x	W_y	$J_{xs}=J_{ys}$	J_{zs}	$J_{xg}=J_{yg}$	J_{zg}	J_{xp}	J_{yp}
2.12	11.12	5.62	8.6	38.38	0.88	0.14	0.011	1.423	1.81
Inch	Inch	Inch	Inch	$Lb-S^2In$	$Lb-S^2In$	$Lb-S^2In$	$Lb-S^2In$	$Lb-S^2In$	$Lb-S^2In$

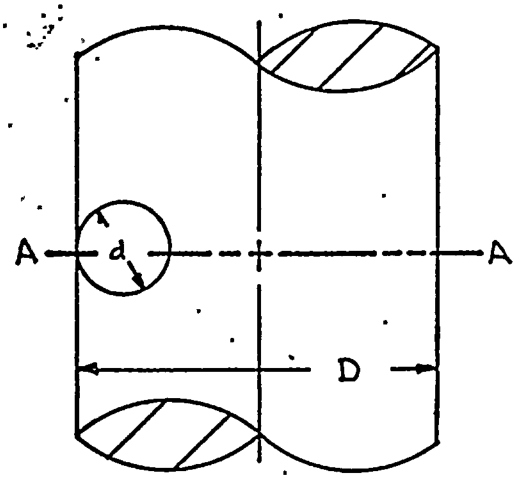
J_{zp}	J_x	J_y	J_z						
0.97	127.88	126.14	3.985						
$Lb-S^2In$	$Lb-S^2In$	$Lb-S^2In$	$Lb-S^2In$						

(Ref: 7 Operator Drawing and Catalog)

Appendix C - Shaft Pin Area



Sect. A-A



d=mean pin diameter

D=shaft diameter

Taking the shear area as an ellipse with semi-major axis b/2 and semi-minor axis d/2

$$A = \frac{\pi}{4} bd$$

$$c^2 = \frac{D^2}{4} - (\frac{D}{2} - d)^2 = dD - d^2$$

$$b^2 = d^2 + c^2 = d^2 + dD - d^2 = dD$$

$$b = \sqrt{dD}$$

$$A = \frac{\pi}{4} d \sqrt{dD}$$