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EVALUATION OF 18-INCH CONTAINMENT ISOLATION VALVES FOR DRESDEN STATION, UNITS 2 & 3, AND QUAD CITIES STATION, UNITS 1 & 2

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REVISION CONTROL SHEET

SUBJECT: EVALUATION OF 18 INCH CONTAINMENT ISOLATION VALVES FOR DRESDEN STATION, UNITS 2 & 3 AND QUAD CITIES STATION, UNITS 1 & 2

REPORT NUMBER: COM-0708-03

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EFFECTIVE PAGE(S)	REV	PREPARED	ACCURACY CHECK	CRITERIA CHECK	EFFECTIVE PAGE(S)	REV	PREPARED	ACCURACY CHECK	CRITERIA CHECK
ii-vi	0	DG	TSH	DKM	7.1-7.7	0	DG	TSH	DKM
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2.1	0	DG	TSH		9.1-9.2	0	DG	TSH	
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4.1-4.2	0	DG	TSH		11.1	0	DG	TSH	DKM
5.1-5.2	0	DG	TSH		App.A	0	*	DG	TSH
6.1-6.3	0	DG	TSH	DKM	App.B		**	**	TL

QEP-001.1-00

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ABSTRACT

Hydrodynamic testing and stress analysis was performed by NUTECH to evaluate stress level margins in critical components of the 18-inch butterfly valves manufactured by the Henry Pratt Company and used as drywell purge and vent containment isolation valves at Dresden Station, Units 2 & 3, and Quad Cities Station, Units 1 & 2.

The purpose of this evaluation was to:

1. Determine the torque values for these valves during closing at various mass flow rates and incremental valve disc angles.
2. Verify that the valves tend to close under flow conditions.
3. Determine the worst case stress level margins existing in the critical load-carrying structural members of the valve during the postulated closing event.

The first stage of the evaluation consisted of hydrodynamic testing at 1/3 scale. The valve used for testing was geometrically similar to the 18-inch butterfly valve and was tested in a facility that reproduced the postulated air flow resulting from containment pressure venting through the valve to atmosphere. During the test, shaft torque values were measured at eight valve disc angles (separated

by 10° increments) covering the range of positions from open to shut and under three sets of flow conditions ranging from a containment pressure of 62.7 psia (sonic) to 20 psia (subsonic). The second stage of the evaluation consisted of an analysis of stresses in the valve shaft, pin, key and actuator arm. This analysis was performed using, as the loading condition, the valve shaft torque values determined in the 1/3-scale valve test scaled to full scale.

The results of the testing indicated that the flow induced hydrodynamic torque tends to close the valve up to the angle where the valve disc contacts the valve seat. The results of the stress analysis indicate that the worst case stress level margins in the valve load-carrying structural members are acceptable.

It is concluded from this testing and analysis that the critical internal components of the Pratt Model 2FII 18-inch butterfly valve will retain structural integrity if subjected to the flow induced loads resulting from a postulated design basis Loss of Coolant Accident when used as a purge and vent containment isolation valve at Dresden Station, Units 2 & 3. and Quad Cities Station, Units 1 & 2.

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1.0 INTRODUCTION

In References 1, 2 and 3, the Nuclear Regulatory Commission requested Commonwealth Edison Company to respond to generic concerns regarding containment purging or venting during normal plant operation and provided guidelines for operability of containment isolation valves used for purging and venting. These operability guidelines included:

1. Demonstrating that the containment isolation valve actuators have sufficient torque capability to stroke the valves from full open to full closed within the technical specification time limit against design basis Loss of Coolant Accident containment pressure.

2. Ensuring that the valve structural elements have sufficient stress margins to withstand the concomitant loads imposed while closing.

The purge and vent containment isolation valves at Dresden Station, Units 2 & 3, and Quad Cities Station, Units 1 & 2, are butterfly valves manufactured by the Henry Pratt Company. Discussions between the Henry Pratt Company, Commonwealth Edison Company and NUTECH (Reference 4) regarding the first operability guideline detailed above resulted in a statement from the Henry Pratt Company that these butterfly valves would tend to close under the postulated conditions.

In order to address the second operability guideline, testing and analysis was conducted to evaluate the stress margins inherent in the valves under the postulated flow conditions. A series of 1/3-scale model tests were conducted at FluiDyne Engineering Corporation laboratories to determine torque values as a function of valve disc angle and flow rate. Analysis of the critical load-carrying components of the valve was performed utilizing the FluiDyne torque values as input. The analysis included calculation of hydrodynamic forces and a simplified stress analysis considering bending, shear and torsional shear loadings. Stress margins were calculated for the load bearing components utilizing standard stress allowables.

This report, prepared for Commonwealth Edison Company, presents the results of the 1/3-scale model test and stress analysis performed on the Dresden Station, Units 2 & 3, and Quad Cities, Units 1 & 2, 18-inch containment isolation valves and provides verification of acceptable stress level margins in their critical internal structural components under a postulated design basis Loss of Coolant Accident. The report summarizes the test objectives, facility, modeling and results and the stress analysis design criteria, loading conditions, methods and results. APPENDIX A is the FluiDyne Engineering Corporation data report. APPENDIX B is the NUTECH stress analysis.

2.0 COMPONENT DESCRIPTION

The valves used at Dresden Station, Units 2 & 3, and Quad Cities Station, Units 1 & 2, for containment purging and venting are 18-inch balanced butterfly valves with external pneumatic/spring (air to open/spring to close) valve actuators. These Pratt Model 2FII valves are constructed with a cast iron body, a 2-1/4 inch diameter type 304 stainless steel shaft and nylon bearings. The valve is mounted in horizontal runs of pipe with the shaft vertically oriented. The actuators are aligned horizontally and are attached to the valve shaft through a lever arm which is keyed to the shaft with a cold drawn steel key. The valve disc is attached to the valve shaft with one stainless steel pin.

Each plant uses two of these valves in series in the drywell purge and vent line. These valves are located outside containment and serve the function of containment isolation valves. In the postulated event that these valves are open for purging or venting and a design basis Loss of Coolant Accident occurs, these valves must be capable of closing within the technical specification time limits and provide containment integrity.

3.0 TEST OBJECTIVES AND DESCRIPTION

The objective of the 1/3 scale test of the Pratt Model 2FII butterfly valve was to ascertain representative torque coefficient values as a function of valve disc angle over a range of air mass flow rates. The test program was conducted in accordance with the NUTECH test specification (Reference 5).

During the first stage of a postulated design basis Loss of Coolant Accident when the subject containment isolation valves are closing, the containment environment will be a mixture of air and steam. Analysis indicates that a discharge of air through the valve results in approximately the same torque values and slightly higher flow rates than does a discharge of steam under the same conditions (Reference 6). Therefore, for conservatism, the test program was run using an all air discharge through the valve.

The initial screening tests were run using three different upstream pressures (approximately 62.7, 38.0 and 20.0 psia venting to atmosphere through the test valve) with test valve disc settings between 8° and 78° from fully open in 10 degree increments. The test pressures were selected using the following criteria:

1. The largest test pressure was to simulate the highest containment transient pressure (62.7 psia) (Ref. 7)

2. The smallest test pressure was to be sufficiently low to ensure subsonic flow through the valve (20 psia). Sonic flow would occur when upstream pressure exceeds 26.9 psia.

3. The intermediate pressure would be between the other two pressures but would still ensure sonic flow (38.0 psia).

The final test runs were comprised of setting the valve disc at the two angles that resulted in the largest torque values in the initial screening tests and then decreasing the upstream pressure in 4 psia increments from 62.7 psia down past a pressure that yielded a maximum torque value. This set of runs was performed to ensure that the maximum torque value for the system was measured, since the initial screening test runs had large gaps in the pressure range.

4.0 TEST FACILITY

The test facility employed for these tests was located at FluiDyne Engineering Corporation's Medicine Lake Aerodynamics Laboratory. This facility (depicted in Figure 1 of APPENDIX A) was used to supply a known air flow rate with a uniform flow distribution to the test valve. High pressure dried air was supplied at 500 psi to a control valve where it was throttled, metered through a choked standard ASME long-radius flow nozzle and discharged into a stagnation chamber. Uniform air flow from the stagnation chamber then flowed through the 6-inch test valve and a 92.5 inch long, 6-inch straight discharge pipe to atmosphere. The stagnation chamber was comprised of a sufficient number of baffles and expansion areas to ensure uniform flow from the stagnation chamber outlet.

The 6-inch test valve was a standard Pratt Model 2FII valve with a custom fabricated scaled valve disc and shaft (depicted in Figure 2 of APPENDIX A). The disc was fabricated at Fluidyne on a contour machining mill utilizing dimensions received from actual field measurements of an 18-inch Model 2FII valve which were scaled to 1/3 size. The shaft was fabricated to accept the disc and was keyed to an adjustable face plate which provided disc angles in increments of 10 degrees. In this way, the flow path through the valve was virtually an exact 1/3-scale representation of the 18-inch valve.

Test instrumentation consisted of:

(1) pressure taps connected to the ASME flow nozzle, to measure air mass flow rate

(2) a Seegers bourdon-tube pressure gauge to measure upstream total pressure

(3) a shielded iron-constantan thermocouple probe to measure upstream total temperature

(4) a Heise bourdon-tube pressure gauge to measure total pressure at the valve

(5) a shielded iron-constantan thermocouple probe to measure total temperature at the valve

(6) six static pressure taps downstream of the valve connected to multiple tube mercury manometers and

(7) a strain gauge torque meter.

The readings of the pressure gauges and mercury manometers were recorded on Polaroid film. The outputs from the ASME nozzle, thermocouples and torque meter were recorded on the test facility data system.

Test calibration records for this instrumentation are presented in of APPENDIX A.

5.0 TEST PROCEDURE

1. The valve disc angle, α, was set by the lab mechanic and verified by the test engineer at each position. All valve pressures were run at this setting before α was changed.

2. The barometric pressure was measured (Hass mercury barometer) before each run and was used in the data reduction.

3. Before each test, the cameras on the P_{t1} and P_{t2} (total pressure) gauges were checked and cocked.

4. Pre-run outputs of electronic instrumentation were recorded using the digital data system.

5. At the direction of the test engineer, air flow was initated by a mechanic opening a manual control valve. The total pressure at the test valve was the independent variable monitored by the valve operator. The control pressure was observed using a 0-200 psi Heise differential pressure gauge.

6. When steady flow was established, all data was simultaneously recorded (torque meter and thermocouples on data system and pressure gauges and manometer on Polaroid film). Flow was then terminated.

7. Post-run outputs were recorded.

8. All results and inputs were verified for transcription accuracy or any electronic anomalies.

9a. For run numbers 1-24, one set of data per discharge was taken.

9b. For run numbers 25-30, only the torque meter output and total pressure at the test valve was recorded. The output was recorded at approximately 4 psi intervals of P_{t2}. All other procedures were as described above.

6.0 SCALING CONSIDERATION

Investigations into the hydrodynamic torque characteristics of butterfly valves (Reference 8) indicate that non-dimensional torque coefficients developed for symmetric butterfly valves can be applied to varying sizes of valves with similar geometries. The basis for scaling the 1/3-scale model torque results up to full size follows:

$$\text{Torque} = C_T \rho V^2 D^3$$

Where

C_T	=	Torque coefficient
ρ	=	Density of air
V	=	Velocity of air at valve disc edge
D	=	Diameter of valve opening

For a perfect gas, which air is assumed to be, the density and velocity can be determined as follows:

$$\rho = \frac{P}{RT}$$

$$V = M \sqrt{\gamma g R T}$$

M	=	Mach Number
P	=	Static Pressure
R	=	Universal gas constant
T	=	Temperature of the air
γ	=	Ratio of specific heats
g	=	Gravitational constant

Therefore:

$$\text{Torque} = C_T \left(\frac{P}{RT}\right) \left(M\sqrt{\gamma gRT}\right)^2 D^3 = C_T P M^2 \gamma g D^3$$

Since the 1/3-scale model duplicates the relative geometry of the full size valve, the pressure characteristics of the 1/3-scale flow path should be identical to those of the full size valve for the same inlet and discharge pressures. Therefore, the ratio of static to total pressure $\frac{P}{P_t}$ and the ratio of static to discharge pressure $\frac{P}{P_{atm}}$ at the valve disc should be identical for the scaled and full size valves.

Since P_{atm} is assumed to be the same, for the same total (containment) pressure, the static pressure at the valve disc is the same for the scaled and full size valves. Since the flow characteristics are the same, the Mach number M is also identical. A final assumption is that at any given disc angle, C_T remains constant for all valve sizes of the same geometry.

The scaling ratio is calculated as follows:

$$\frac{\text{Torque}_{18''}}{\text{Torque}_{6''}} = \frac{C_T \left(\frac{P_{18''}}{P_t}\right) M^2 \ \gamma g \ (D_{18''})^3}{C_T \left(\frac{P_{6''}}{P_t}\right) M^2 \ \gamma g \ (D_{6''})^3}$$

($\frac{P}{P_t}$, C_T, M, γ, g all identical for 6" and 18" valves)

For a given total pressure P_t in the scaled and full size valve:

$$\frac{\text{Torque}_{18''}}{\text{Torque}_{6''}} = \left(\frac{D_{18''}}{D_{6''}}\right)^3$$

Therefore the scale factor for torque is

$$\left(\frac{18}{6}\right)^3 = 27 \quad .$$

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7.0 TEST RESULTS

Table 7-1 summarizes the data from the test program. Included in this table are the measured valve disc angle, total pressure at the test valve and torque. For the subsonic test runs, the calculated torque coefficients are also tabulated. Torque values for the final two sets of tests are presented in Figure 5 of the data report (APPENDIX A). The final two sets of tests were run in an effort to determine experimentally the maximum torque value that could be obtained from the test facility. Table 7.2 presents the maximum torque values measured in the first phase of the testing scaled to full size for each of the three containment pressures. In addition, the maximum torque scaled to full size for the second phase of the testing is presented.

Based on the form of the scaling factor for torque developed in Section 6.0, it was possible to calculate full size valve torque values without the use of specifically determined torque coefficients. However, as a check on the validity of the tests results obtained, torque coefficients were calculated for the P_t = 20 psia case, for which the assumption of incompressible flow is valid, and were compared to the torque coefficients developed in Reference 8. The two sets of torque coefficients are plotted in

Figure 7-1. They are relatively close, thus providing a check that the subsonic torque values from this test are reasonable.

Several negative torque values (tending to open the valve) occur in the test data. At the 8° from full open angle, the case with subsonic flow over the valve disc (P_t = 20 psia) exhibited a positive torque value while the two sonic cases (38.0 and 62.7 psia) exhibited relatively small negative torque values. This may be explained by likening the valve disc to an airfoil in which the valve shaft is at mid-chord. The transition from subsonic to sonic flow causes the aerodynamic center, through which the hydrodynamic force acts, to move from the leading surface of the valve disc (positive torque) to the approximate location of the valve shaft. This location can yield a small positive or negative torque. At the 68° and 78° angles, negative torque values were also measured. These angles correspond to the orientation at which the valve disc progressively contacts and deforms the valve seat. At these orientations, the friction and seat-induced forces become relatively large as the seat material resists further deformation by exerting a force against the valve disc as the seat attempts to restore its original shape.

7.1 Torque Coefficients

Torque coefficients were calculated as follows for the subsonic ($M < 0.7$) flow case (P_t = 20 psia):

Data from test: Torque

Total pressure: P_t

Static pressure: P_s

From Bernoulli's equation: $P_t = P_s + 1/2\ \rho V^2$

or $\rho V^2 = 2(P_t - P_s)$

also $\text{Torque} = C_t \rho V^2 D^3$

rearranging and combining

$$C_T = \frac{\text{Torque}}{2(P_t - P_s)\ D^3}$$

7.2 Friction

It was intended in this test to ensure that the values of friction in the valve were relatively small so as to allow the most accurate determination of hydrodynamic torque. As presented in Section 4.0 of the APPENDIX A data report, the measured torque to overcome friction was less than 10% of the maximum hydrodynamic torque values except at a valve disc angle of 78° at which point the valve disc was contacting the valve seat. It was conservative to minimize the friction in the test valve because, in the full size valve, friction will resist the forces that tend to shut the valve thus lowering the actuator force and the bending applied to the shaft by the actuator. The effect of friction in the test was to counteract the flow-generated hydrodynamic torque, thus reducing in absolute value the torque measured by the torque meter. For this reason, the absolute value of the torque meter output was increased by the static friction-generated torque values, and the resulting torque values are presented in Table 7-1.

As an example, the 1/3-scale torque value for the case with valve disc angle of 38° and P_t = 38 psia was calculated as follows:

Torque meter reading:	80.1 in-lbs.
Friction at 38°	7.0 in-lbs.
	87.1 in-lbs.

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TABLE 7-1

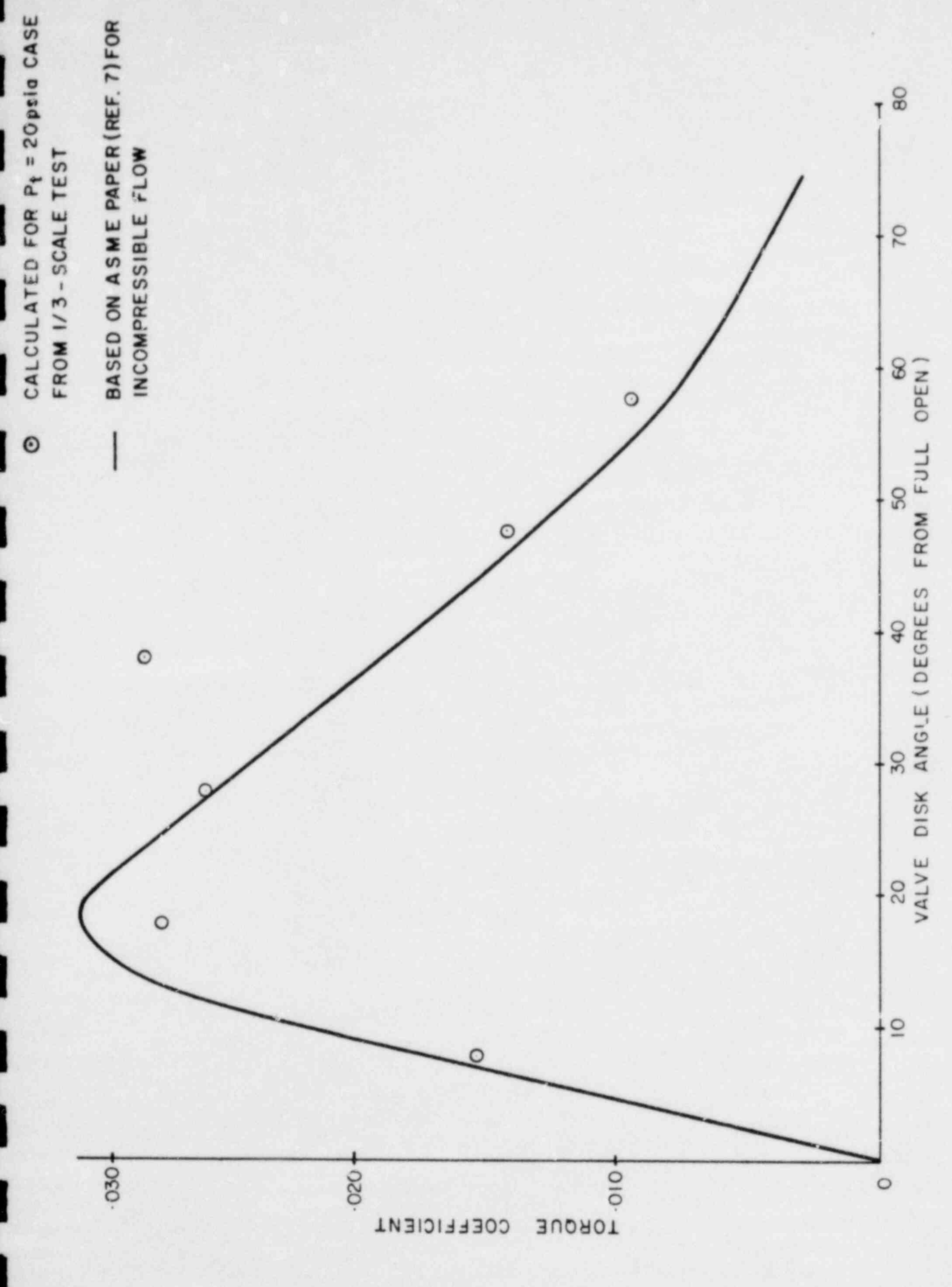
SUMMARIZED TEST DATA

Angle (degrees)	P_t (psia)	Torque (in-lbs)	C_T
8	63.32	-20.0	
18	62.53	62.4	
28	62.23	58.9	
38	62.46	71.6	
48	63.06	64.9	
58	62.96	43.8	
58	62.86	49.8	
68	63.26	-25.1	
78	62.23	-76.2	
78	62.93	-70.4	
8	38.32	-13.9	
18	38.13	35.3	
28	38.28	62.6	
38	38.46	87.1	
48	38.46	76.6	
58	37.96	37.1	
58	37.86	37.0	
68	37.86	-12.4	
78	38.23	-49.9	
8	20.22	36.3	.0152
18	20.03	65.9	.0271
28	20.23	66.5	.0255
38	18.36	50.6	.0278
48	20.16	37.1	.0140
48	20.06	36.8	.0141
58	19.76	23.6	.[illegible]
68	19.86	-8.3	-.00[illegible]
78	20.03	31.5	.0126

TABLE 7-2

MAXIMUM TORQUE RESULTS

P_t (psia)	Valve Disc Angle (degrees)	Maximum Full Scale Torque (ft-lbs)
62.7	18	140
38.0	38	196
20.0	18	148
32.0-35.0	38	216



CALCULATED FOR P_t = 20psia CASE FROM 1/3 - SCALE TEST
BASED ON ASME PAPER (REF. 7) FOR INCOMPRESSIBLE FLOW
TORQUE COEFFICIENT
.030
.020
.010
0
10
20
30
40
50
60
70
80
VALVE DISK ANGLE (DEGREES FROM FULL OPEN)

FIGURE 7-1

8.0 STRESS ANALYSIS DESIGN CRITERIA, LOADING CONDITION AND ANALYTICAL METHODS

The stress analysis of the valve is presented in APPENDIX B. The design criteria, loadings and analytical methods used are presented below.

8.1 Design Criteria

The purpose of the analysis was to analytically determine that the stress levels in the valve load-carrying structural members were within limits that would preclude yielding for those members as the valve is closed against a flow rate generated by a postulated design basis Loss of Coolant Accident. This criteria would ensure that during closing, the active valve parts would not deform.

8.2 Loading Condition

The loading condition considered in the stress analysis of the butterfly valve included hydrodynamic torque and valve actuator restraining force. Dead weight and seismic forces were considered to be negligible.

Torque loading of the valve shaft was based on the torque coefficients determined in the 1/3-scale valve test. Full size torque values were calculated from the experimentally determined 1/3-scale torque values shown in Tables 7-1 and 7-2 to which a conservative factor was added.

Valve actuator force was conservatively calculated based on the assumption that the actuator balanced the hydrodynamic torque generated in the valve at each valve disc angle. The valve actuator is comprised of an air cylinder/spring combination. The spring is attached to a piston inside the air cylinder and forces the valve closed when there is no air in the cylinder. To open the valve, the cylinder is pressurized with air such that the air pressure on the piston overcomes the spring force. Upon a containment isolation signal, the air in the cylinder is bled out through an orifice thus permitting the spring to gradually close the valve. If the flow induced hydrodynamic torque is positive (to close the valve), the effect is to compress the air in the cylinder. The upper curve in Figure 8-1 represents the torque available from the air compression force in the cylinder which can resist this closing torque. If the flow induced hydrodynamic torque is negative (to open the valve), th effect is to compress the spring. The lower curve in Figure 8-1 represents the torque available from the spring force which can resist this opening torque. Since the upper and lower curves envelope the calculated hydrodynamic torque, during a postulated design basis Loss of Coolant Accident, the valve will close at the normal operating rate governed by the initial air pressure, orifice size and spring constant.

8.3 Analytical Methods

The butterfly valve was analyzed to determine the stress level margins in the valve load-carrying active components during the postulated flow condition.

The analysis consisted of determination of bending, torsion and shear loads on the valve shaft, key, pin and actuator arm at valve disc angles ranging from 8° to 78° from full open. Hydrodynamic torque values were conservatively based on experimental test results reported in Section 6.0 of this Report. Bending and torsional moments and shear forces were calculated at the actuator arm attachment, upper and lower bearings and the pin. The maximum shear stress due to combined bending, torsion and shear was then calculated and compared to a maximum shear stress allowable of 1/2 yield strength to generate safety factor values.

AVAILABLE TORQUE IN SYSTEM

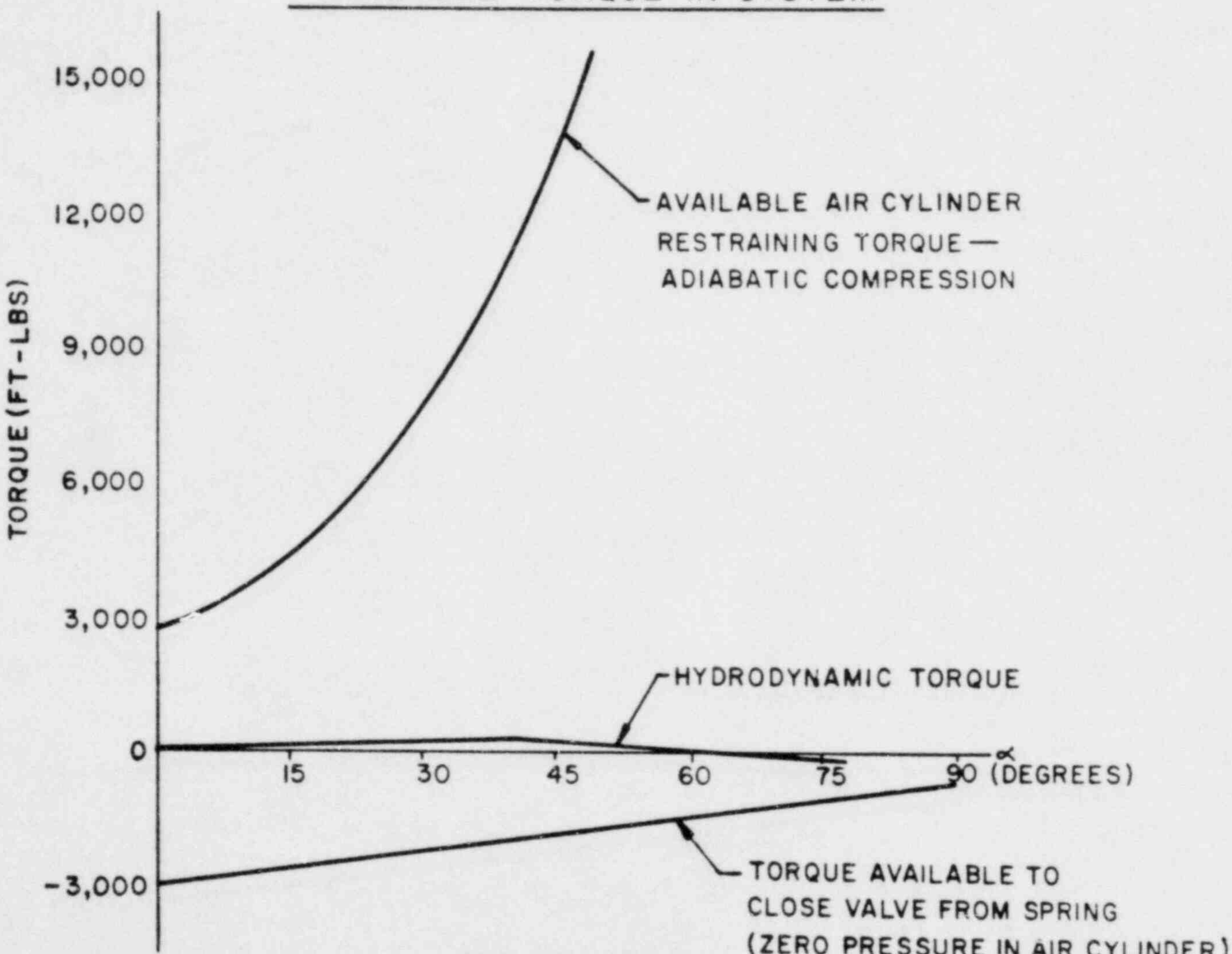


FIGURE 8-1

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9.0 STRESS ANALYSIS RESULTS

The stress and safety factor values for the stress analysis are presented in Table 9-1. The critical location in the valve was determined to be the shaft at the valve disc-to-shaft pin, where a conservatively calculated factor of safety of 1.33 was calculated.

TABLE 9-1

SUMMARY OF STRESS RESULTS AND FACTORS OF SAFETY

STRESS AND FACTOR OF SAFETY (in brackets)

SHAFT:

VALVE ANGLE \ LOCATION	KEY	UPPER BEARING	PIN	LOWER BEARING
8°			11.235 ksi (1.34)	
18°		3.110 ksi (4.8)	11.257 ksi (1.33)	2.607 ksi (5.75)
28°		3.055 ksi (4.91)	7.555 ksi (1.99)	2.106 ksi (7.12)
38°	4152.7 ksi (3.6)	4.291 ksi (3.50)	10.194 ksi (1.47)	1.010 ksi (14.9)
48°		3.331 ksi (4.5)	10.711 ksi (1.40)	0.439 ksi (34.1)

KEY:	Shear Stress	1.293 ksi
	Comp. Stress	2.586 ksi
PIN:	Unit Working Stress	2.032 ksi
ACTUATOR ARM:	Bending Stress	1.755 ksi

10.0 CONCLUSIONS

The stress analysis of the Pratt Model 2FII 18-inch butterfly valve included as APPENDIX B indicates that the loads and stresses imposed upon the active load-carrying components during a valve closure under a postulated design basis Loss of Coolant Accident event are within acceptable limits and that stress margins are sufficient to ensure no deformation of the active valve parts will occur when the valve is used as a containment isolation valve in the Dresden Station, Units 2 & 3 or Quad Cities Station, Units 1 and 2. The loads used in the stress analysis were based upon results from the Fluidyne Engineering Corporation 1/3-scale valve test, included as APPENDIX A.

11.0 REFERENCES

1. Nuclear Regulatory Commission letter from Mr. Thomas Ippolito to Mr. Cordell Reed (Commonwealth Edison Company) "Containment Purging During Normal Plant Operation", dated November 29, 1978.

2. Nuclear Regulatory Commission letter from Mr. Darrell G. Eisenhut to All Light Water Reactors, "Containment Purging and Venting During Normal Operation - Guidelines for Valve Operability" dated September 27, 1979.

3. Nuclear Regulatory Commission letter from Mr. Dennis L. Ziemann to Mr. D. Louis Peoples (Commonwealth Edison Company), "Containment Purging and Venting During Normal Operation", dated October 23, 1979.

4. NUTECH letter COM-0708-01 from Mr. D. A. Gerber, to Mr. H. L. Gustin (Commonwealth Edison Company), "Henry Pratt Company Meeting Report", dated December 10, 1979.

5. NUTECH Test Specification COM-0708-02, "Determination of Torque Coefficients for the Dresden 2/3 and Quad Cities 1/2-18 Inch Butterfly Containment Isolation Valves", dated February 8, 1980.

6. NUTECH Hydrodynamic Analysis of Pratt 18 Inch Butterfly Valve, Revision 0 dated December 3, 1979, File No. 64.801.0004.

7. Dresden Nuclear Power Station, Unit 2 - Final Safety Analysis Report.

8. T. Sarpkaya, "Torque and Cavitation Characteristics of Butterfly Valves," Journal of Applied Mechanics, Transactions of the ASME, Number 60-WA-105, December 1961, pp. 511 - 518.

APPENDIX A

FluiDyne Test Data Report

Measurement of Disc Torque at Various

Airflow Rates in a 6-Inch Butterfly Valve

FLUIDYNE ENGINEERING CORPORATION

MEASUREMENT OF DISC TORQUE AT VARIOUS AIRFLOW RATES IN A 6-INCH BUTTERFLY VALVE

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Nutech Engineering Services Agreement 8 February 1980
FluiDyne Report 1240
March 1980

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SUMMARY

This report presents the results of tests conducted to determine the torque on the shaft of a modified 6-inch Pratt model 2FII butterfly valve at various airflow rates and valve disc positions. The tests were performed by FluiDyne Engineering Corporation for Nutech in the Channel 7 test stand at FluiDyne's Medicine Lake Aerodynamic Laboratory.

The test valve was obtained by replacing the standard disc with a disc geometrically similar to that in a particular 18-inch valve. Tests were made at disc settings from 8° to 78° (nearly closed) in 10° increments. The total pressure of the approach flow was varied from 62.7 psia to 20 psia, with the flow exhausting to atmosphere through a straight pipe downstream of the valve. Torque on the disc shaft was measured using a strain-gage torque meter.

Test results for each vane setting include valve shaft torque, airflow total pressure, total temperature and mass flow rate, and six static pressures in the pipe just downstream of the valve. The effective open area of the valve was calculated from the airflow quantitites.

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LIST OF SYMBOLS

A	Cross section area, in^2
C_D	Discharge Coefficient, dimensionless
K	Critical flow factor, R°$^{1/2}$/sec.
M	Mach number, dimensionless
P	Pressure, static unless otherwise specified by subscript, psia
R_N	Reynolds number, dimensionless
T	Temperature, °R
W	Flow rate, lbm/sec
α	Valve angle, degrees

Subscripts

1	ASME meter conditions
2	Valve approach conditions
t	Total conditions
eff	Effective

1.0 INTRODUCTION

This test program was conducted to measure the torque and airflow characteristics of a 6-inch butterfly valve. The test valve was a 1/3 scale simulation of an 18-inch isolation valve in a nuclear power plant. The disc and shaft in the 6-inch valve were designed and fabricated by FluiDyne using disc contours defined by Nutech.

The test program was defined by Nutech Test Specification COM-0708-02, "Determination of Torque Coefficients for the Dresden 2/3 and Quad Cities 1/2 18-inch Butterfly Containment Isolation Valves." Technical liaison for Nutech was performed by Mr. Dave Gerber.

This report describes the test apparatus, test conditions, data acquisition and analysis procedures, and presents the test results. Test conditions and major test results are tabulated in Figure 5 and are plotted in Figures 6-9. Detailed data and calculations are tabulated in the Appendix.

2.0 TEST APPARATUS

2.1 Channel 7 Facility

The tests were performed in Channel 7 at FluiDyne's Medicine Lake Aerodynamics Laboratory. Channel 7 is a test stand (operating since 1956) normally used for determining thrust and flow rate performance of scale-model exhaust nozzles for jet or rocket engines. For the present tests, Channel 7 was used only to supply a known flow rate, with a uniform flow distribution, to the valve assembly. A 92.5-inch long, 6-inch diameter pipe was installed downstream of the valve exhausting to atmosphere.

The basic arrangement of this facility is indicated in Figure 1. High-pressure dried air from the facility 100 psi storage system was throttled at the control valve, metered through a choked standard ASME long-radius flow nozzle, and discharged into a stagnation chamber. Uniform flow from the stagnation chamber was obtained at the entrance to the test valve.

Facility instrumentation was provided to measure the flow rate, the total temperature and pressure upstream of the valve, and six static pressures just downstream of the valve. Details are described in Section 3.0.

2.2 Butterfly Valve

Figure 2 is an assembly drawing of the valve and torque meter assembly. As indicated, the valve was a standard 6-inch Pratt model 2FII valve, with a custom-fabricated disc and shaft. The disc is defined in Figure 2. Photographs of the valve assembly are shown in Figure 4.

The end of the shaft was keyed into an adjustable face-plate to provide incremental (10°) disc settings. The 0° position was defined as fully-open.

3.0 DATA ACQUISITION AND CALCULATIONS

The following subsections describe data acquisition and analysis procedures used in the present test program. Station notations are defined in Figure 1. A computer program for data reduction, written in BASIC language, is included in the Appendix.

3.1 Flow Rate

The actual mass flow rate through the test valve was determined with a choked ASME long-radius metering nozzle at Station 1.

$$W_1 = W_2 = \frac{K_1 C_{D_1} A_1 P_{t_1}}{\sqrt{T_{t_1}}}$$

The meter discharge coefficient (C_{D_1}) was calculated as a function of throat Reynolds number, using a semi-empirical equation.

$$C_{D_1} = 1 - 0.184\, R_{N_1}^{-0.2}$$

C_{D_1} varied between 0.987 and 0.993 for the present tests.

The critical flow factor, K, was calculated as a function of total pressure and total temperature. The equation for K, applicable to the range of P_t and T_t normally encountered in the present test facility, was obtained from Reference 1:

$$K = .53160 + (P_t + 16.9)\,[1.581 - .00834\,(T_t - 520)] \times 10^{-5}$$

P_t is in units of psia, and T_t is in °R.

A_1, the meter geometric throat area, had three different values, depending on the open area in the test valve. They were: 4.905 in^2, .8028 in^2, and .1955 in^2. Meter pressure, P_t, was measured with a Seegers bourdon-tube pressure gauge and recorded on Polaroid film. T_t was measured using a shielded iron-constantan thermocouple probe. The thermocouple output was recorded on the facility digital data system. Flow rates calculated for the present tests varied from .257 to 28.5 lb_m/sec.

The effective flow area of the valve was calculated as

$$A_{eff} = \frac{W_2 \sqrt{T_{t_2}}}{P_{t_2} K_2}$$

K_2 was evaluated, using a previous equation defining the critical flow factor, as a function of P_{t_2} and T_{t_2}. The above equation for A_{eff} therefore assumes choked flow through the valve.

3.2 Condition at Test Valve

The total pressure at the valve, P_{t_2}, was measured with a Heise bourdon-tube pressure gauge and recorded on Polaroid film. T_{t_2} was measured using a shielded iron-constantan thermocouple probe and the facility data system.

Static pressures were measured at 3, 6, and 9 inches downstream of the valve in the 6-inch diameter pipe at two circumferential positions (see Figure 1). Static pressures were measured using multiple tube mercury manometers (atmospheric reference) and recorded on Polaroid film. Pressures were reduced to absolute pressures (psia) and are listed on the computer output sheets.

3.3 Torque Measurement

The torque developed at the shaft end was measured with an existing FluiDyne torque meter. This device is a cylindrical tube wired to sense small strains in the torsional direction only. In general, all other bending moments and shear stresses (if they exist) cancel in the stress measurement output. The torque meter output was recorded on the facility data system.

Prior to testing, the torquemeter was loaded with known moments in the ranges and directions to be encountered during testing. A curve of known applied moment versus readout signal was generated to give torque as a function of torquemeter output. Positive moments are defined to be in the direction tending to close the valve. After the tests the calibration was repeated.

The post-test calibration agreed to within 0.05% of the pre-test calibration used to reduce the test data.

4.0 PRESENTATION OF RESULTS

Test conditions and major test results are tabulated in Figure 5 and are plotted in Figures 6-9. Detailed data and calculations are contained in the Appendix.

Figure 5, sheet 1, presents the results for tests at valve settings of 8° to 78°, for nominal values of P_{t_2} = 62.7, 38 and 20 psia. A few repeat runs were made either to demonstrate data repeatability or because of incomplete data acquisition. Extra runs are denoted by *. The tabulation includes run number, valve angle, measured torque, air total temperature and total pressure at the valve, measured mass flow rate, and the effective open area of the valve.

Figure 5, sheet 2, presents the results of tests during which valve angle was fixed (at 38° and 48°) and P_{t_2} was varied between 62.7 and 20 psia.

Measured torque values are plotted versus valve angle in Figure 6. The symbols denote nominal values of P_{t_2} = 62.7, 38 and 20 psia. The dashed curve, shown for reference only, is an approximate prediction of torque using an incompressible flow relation from Reference 2. The present data at P_{t_2} = 20 psia compare reasonably well with the prediction. The Mach number through the valve is approximately 0.7 for P_{t_2} = 20 psia. With P_{t_2} = 38 and 62.7 psia, the flow through the valve is choked (M = 1).

At valve angles greater than about 65°, the measured torque was negative, i.e., was in the direction to resist closing of the valve. This resistance is attributed to friction of the disc on the rubber valve seat, and is greater than the aerodynamic torque which approaches zero as α approaches 90°. At the conclusion of the test program, the frictional torque on the valve with no flow was measured. This measurement was accomplished by disconnecting the torque meter, and applying a torque-wrench to the end of the shaft. The torque was recorded when rotation of the shaft was first noticed. The apparent torque due to valve friction at various valve settings is tabulated below.

α, degrees	Torque, in-lbs	α, degrees	Torque, in-lbs
8	5	48	7
18	6	58	7
28	7	68	7
36	7	78	30

Referring to Figure 6, maximum values of torque appear to be obtained at valve angles near 40°. Additional tests (Figure 5, sheet 2) were then made in which pressure was varied to search for the maximum torque at fixed valve settings. The results are plotted in Figure 7. The previous data from Figure 6 are included for comparison.

Flow rates and corresponding effective flow areas are plotted in Figures 8 and 9. The reduction in A_{eff} at P_{t_2} = 20 psia, compared to the higher pressure results, is due to unchoking of the valve.

REFERENCES

1. Reimer, R. M., "Computation of the Critical Flow Function, Pressure Ratio, and Temperature Ratio for Real Air." ASME Paper #62-WA-177. 1962.

2. Sarpkaya, Turgut, "Torque and Cavitation Characteristics of Butterfly Valves." Journal of Applied Mechanics, Trans. ASME, December 1961, pp. 511-518.

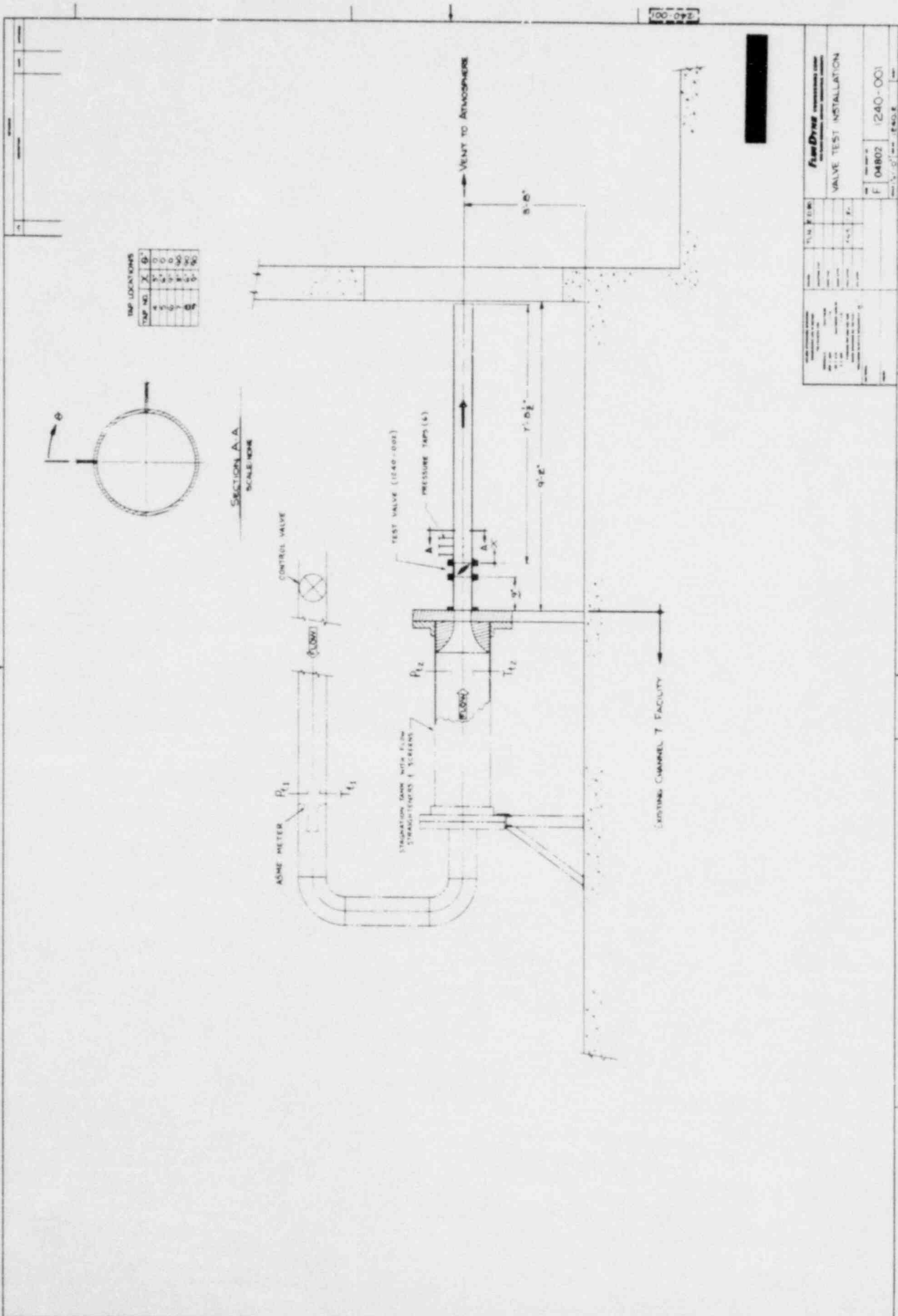

FIGURE 1. VALVE TEST INSTALLATION

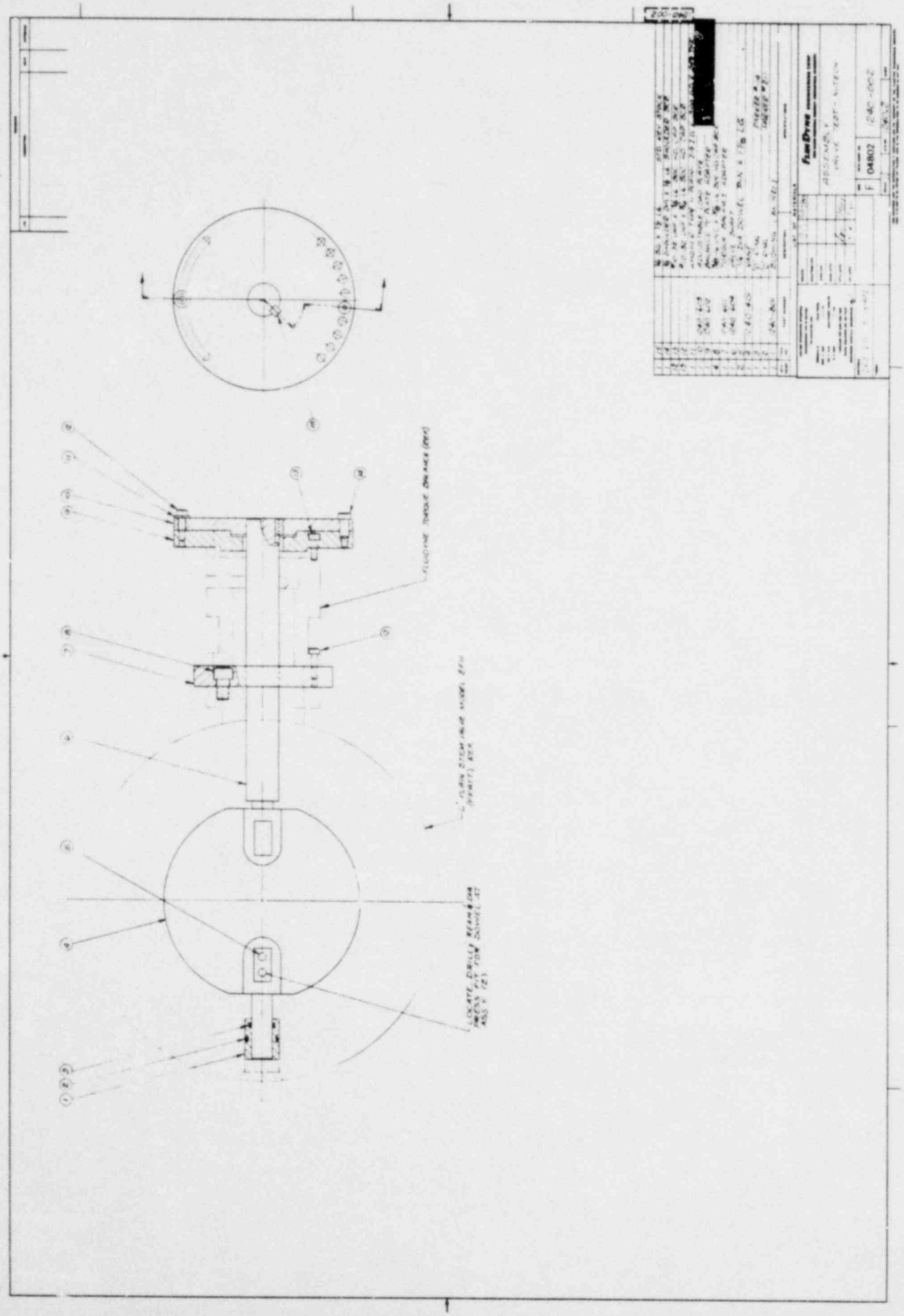

FIGURE 2. VALVE TEST ASSEMBLY

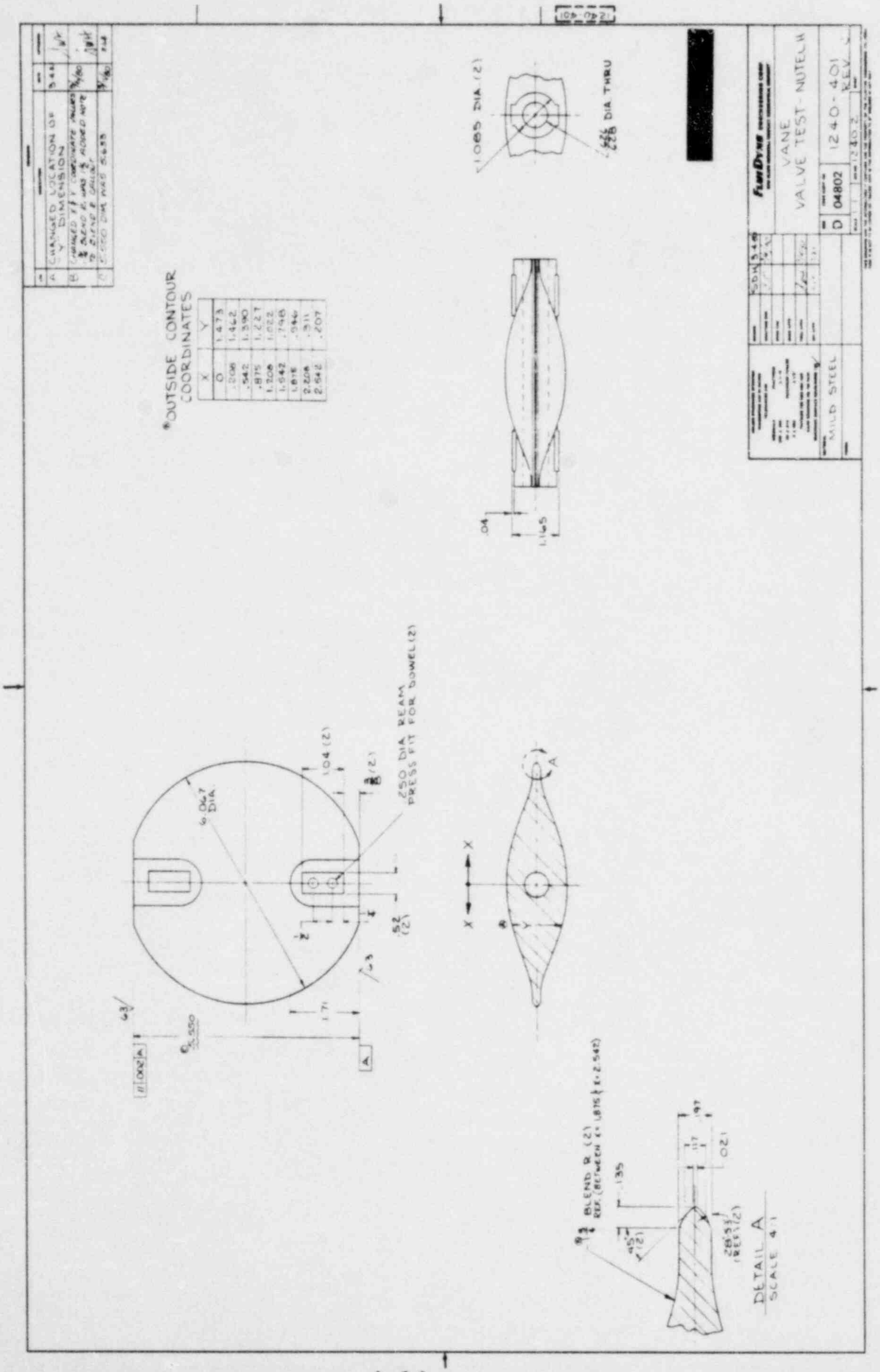

FIGURE 3. VANE (DISC)

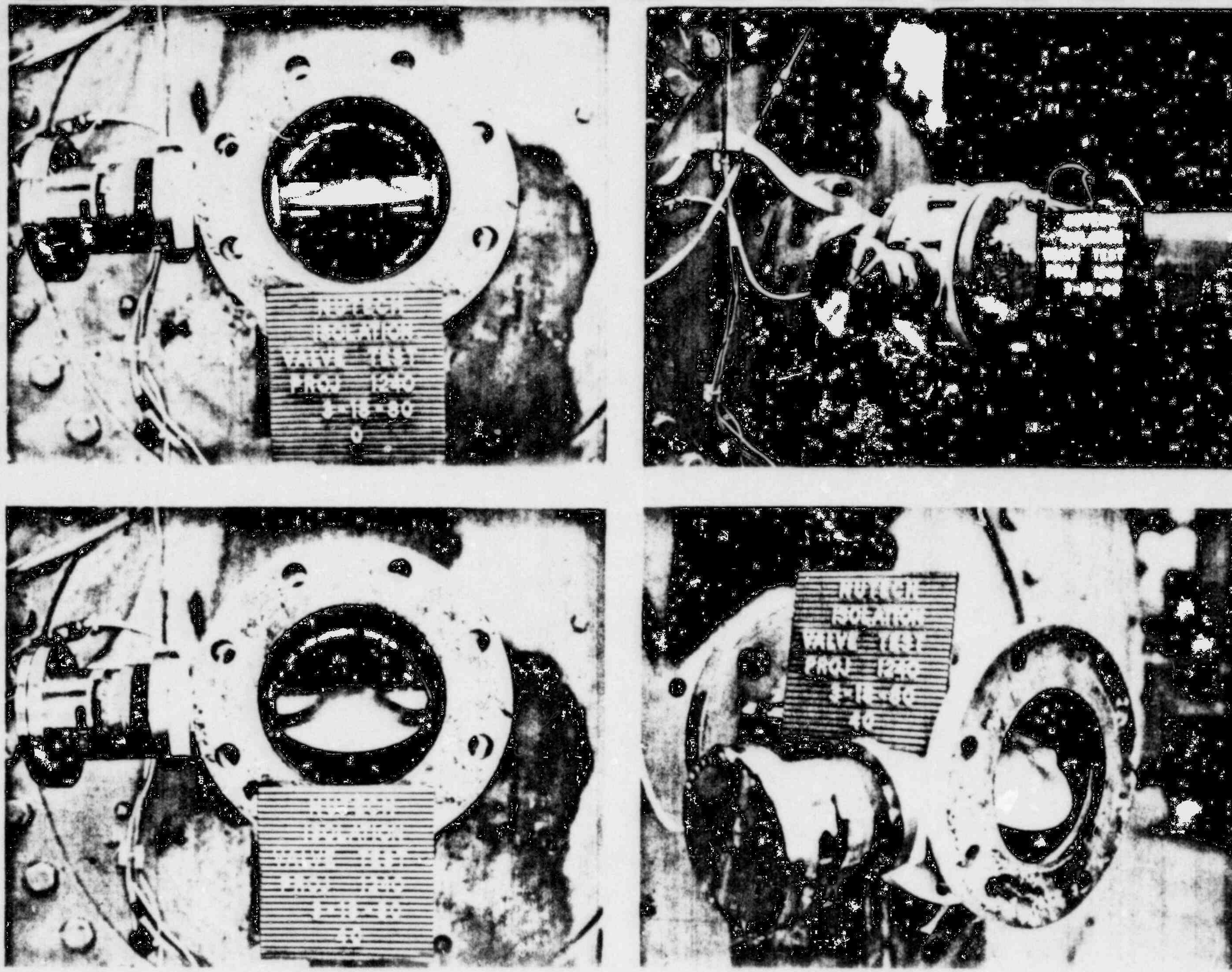

FIGURE 4. PHOTOGRAPHS OF VALVE TEST SETUP

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Run Number	(degrees) α	(in-lbs) Torque	(psia) P_{t_2}	(°F) T_{t_2}	(lbs/sec) W	(in^2) A eff
1.1	8	-15	63.3	64	28.5	19.3
2.1		-9	38.3	60	17.4	19.4
3.1		31	20.2	61	8.60	18.2
4	18	56	62.5	61	27.1	18.6
5		29	38.1	61	16.4	18.4
6		60	20.0	66	7.82	16.8
7	28	52	62.2	66	21.5	14.9
8		56	38.3	66	13.2	14.9
9		60	20.2	67	6.18	13.2
10	38	65	62.5	70	16.1	11.2
11*		80	38	70	-	-
11.1		80	38.5	71	9.84	11.1
12		44	18.4	74	3.84	9.07
13	48	58	63.1	71	11.1	7.59
14		70	38.3	72	6.63	7.51
15		30	20.2	73	2.98	6.41
15.1*		30	20.1	73	2.87	6.20
16*	58	45	63	68	-	-
16.1*		37	63.0	67	6.24	4.27
16.2		43	62.9	68	6.23	4.27
17*		30	38.0	73	3.69	4.21
17.1		37	37.9	68	3.70	4.21
18.1		17	19.8	68	1.63	3.55
19	68	-18	63.3	69	2.29	1.56
20*		-5	63	69	-	-
20.1		-5	37.9	70	1.35	1.54
21		-1	19.9	68	.596	1.30
22*	78	-46	62.2	72	.432	.300
22.1		-40	62.9	82	.433	.301
23		-17	38.2	70	.257	.290
24		2	20.0	66	-	-

* extra run, no charge.

FIGURE 5. RUN SCHEDULE AND MAJOR TEST RESULTS (Sheet 1 of 2)

IMAGE EVALUATION
TEST TARGET (MT-3)

MICROCOPY RESOLUTION TEST CHART

IMAGE EVALUATION
TEST TARGET (MT-3)

6″

MICROCOPY RESOLUTION TEST CHART

Run Number	(degrees) α	(in-lbs) Torque	(psia) P_{t_2}
25	38	56	62.7
25.1		56	58
25.2		58	54
25.3		63	50
26		68	46
26.1		74	42
26.2		79	38
26.3		79	38
27		89	35
27.1		89	32
27.2		79	29
27.3		74	26
28	48	52	62.7
28.1		56	58
28.2		58	54
28.3		62	50
29		65	46
29.1		67	42
29.2		68	38
29.3		66	35
30		64	32
30.1		61	29
30.2		52	26
30.3		33	20

FIGURE 5. RUN SCHEDULE AND MAJOR TEST RESULTS
(Sheet 2 of 2)

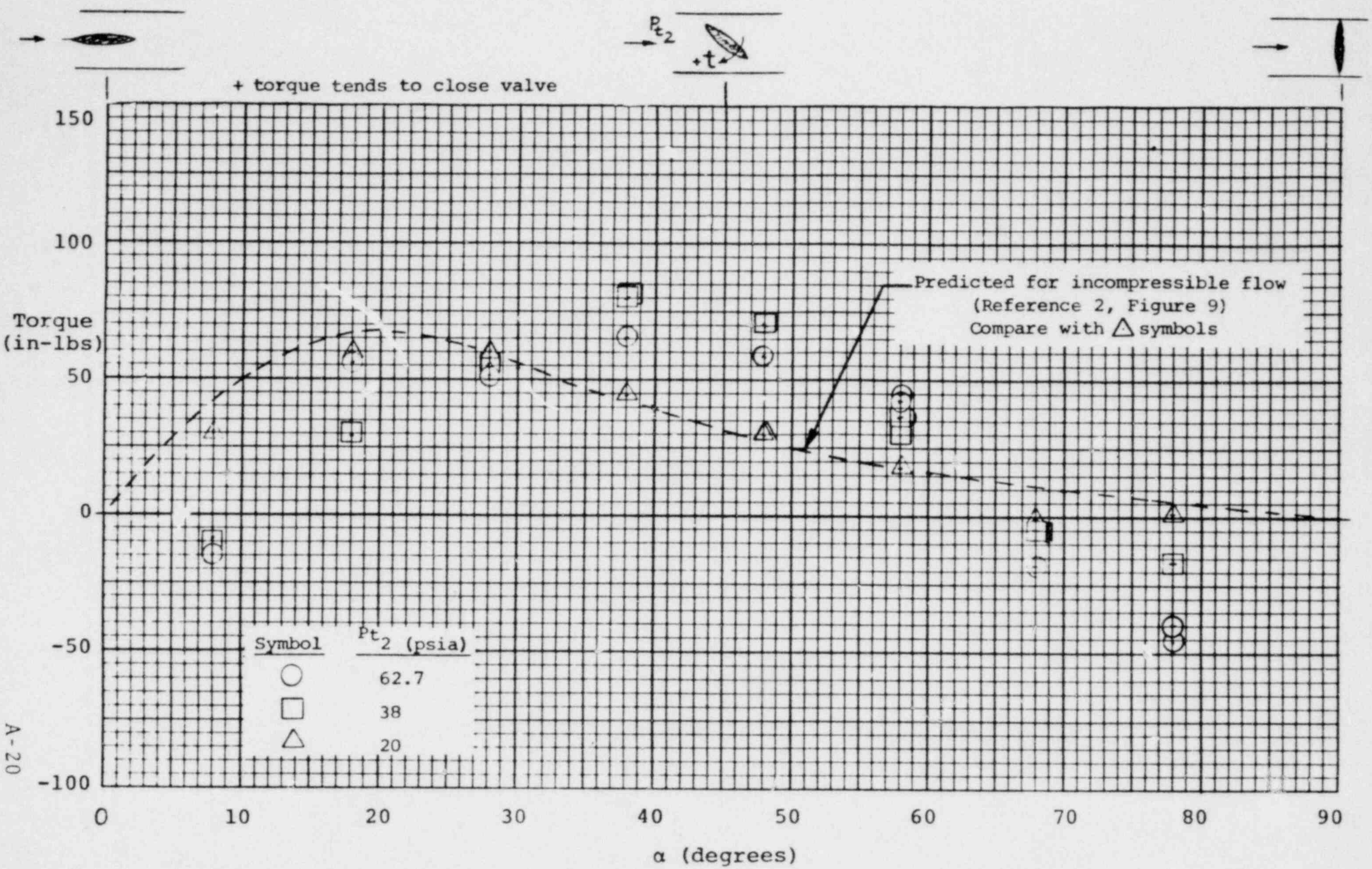
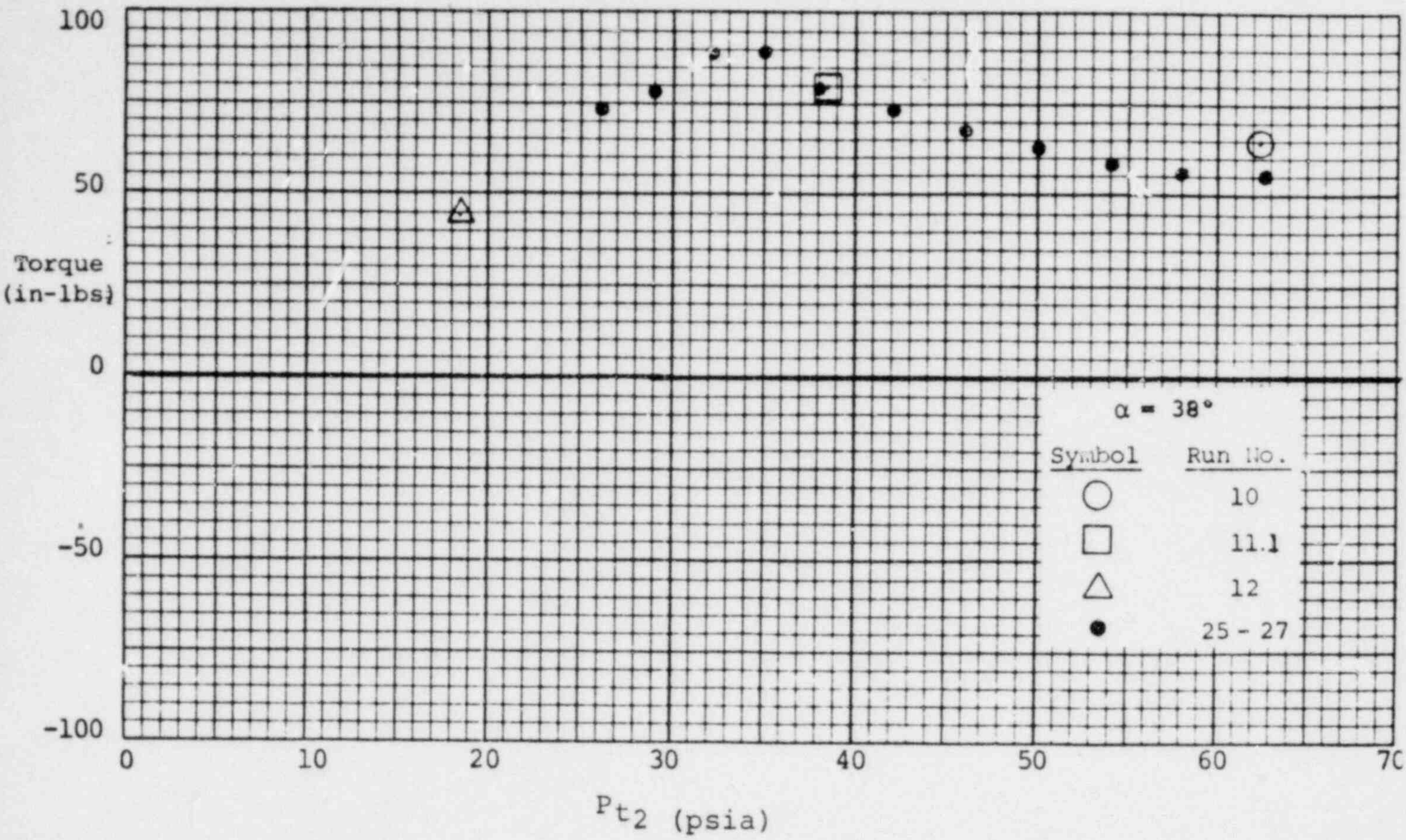


FIGURE 6. TORQUE VERSUS VALVE ANGLE

FLUIDYNE ENGINEERING CORPORATION



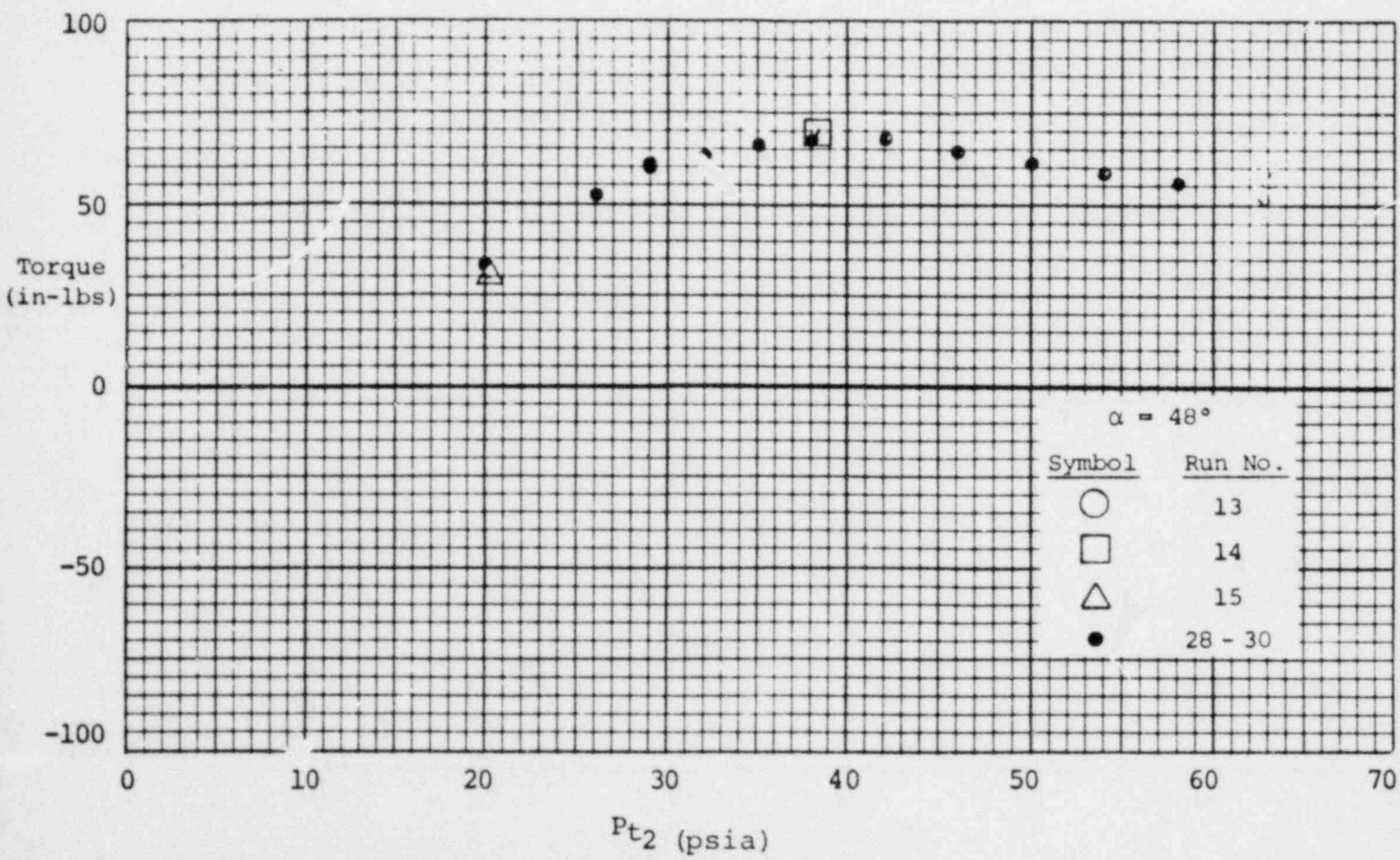


FIGURE 7. TORQUE VERSUS TOTAL PRESSURE

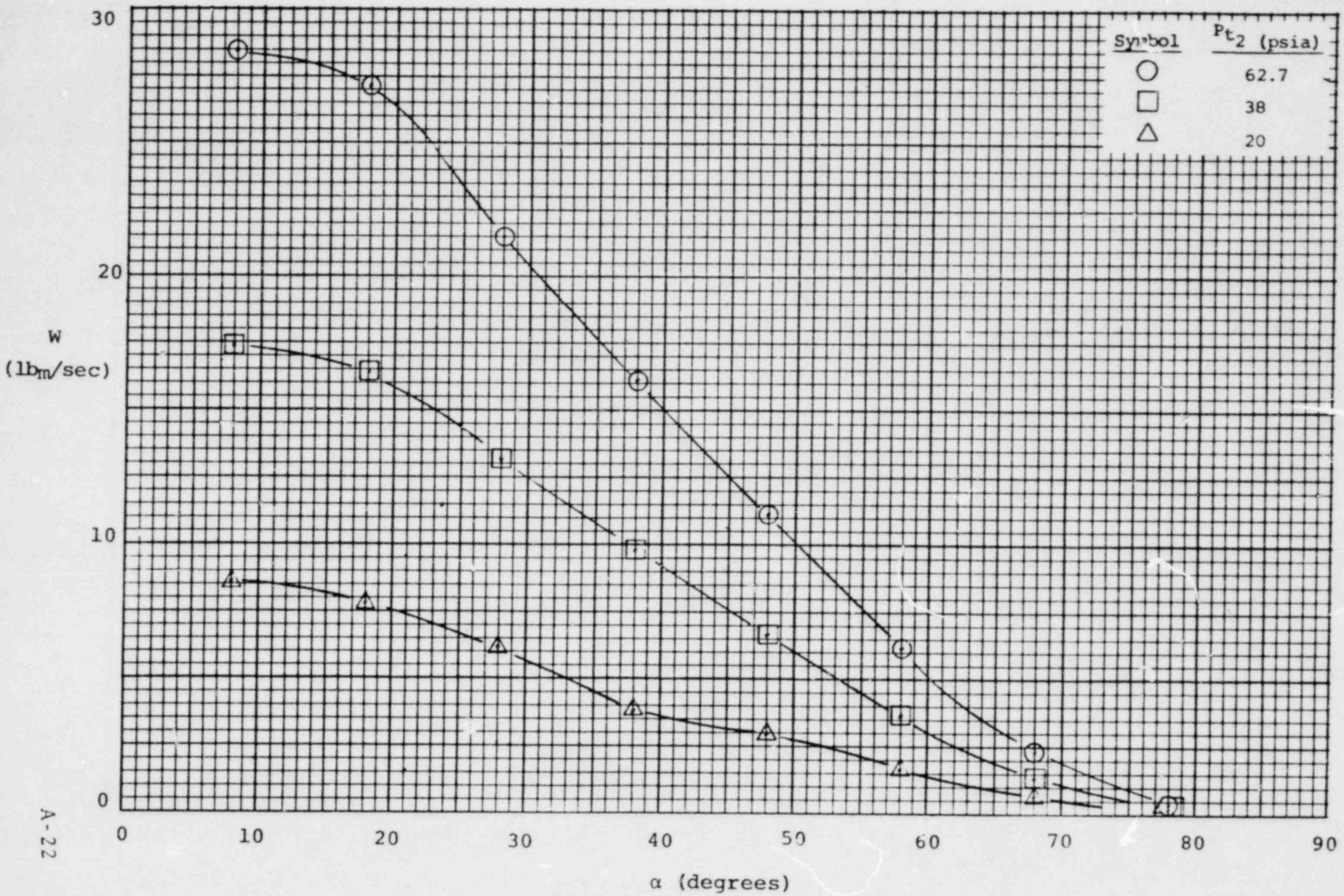


FIGURE 8. FLOW RATE VERSUS VALVE ANGLE

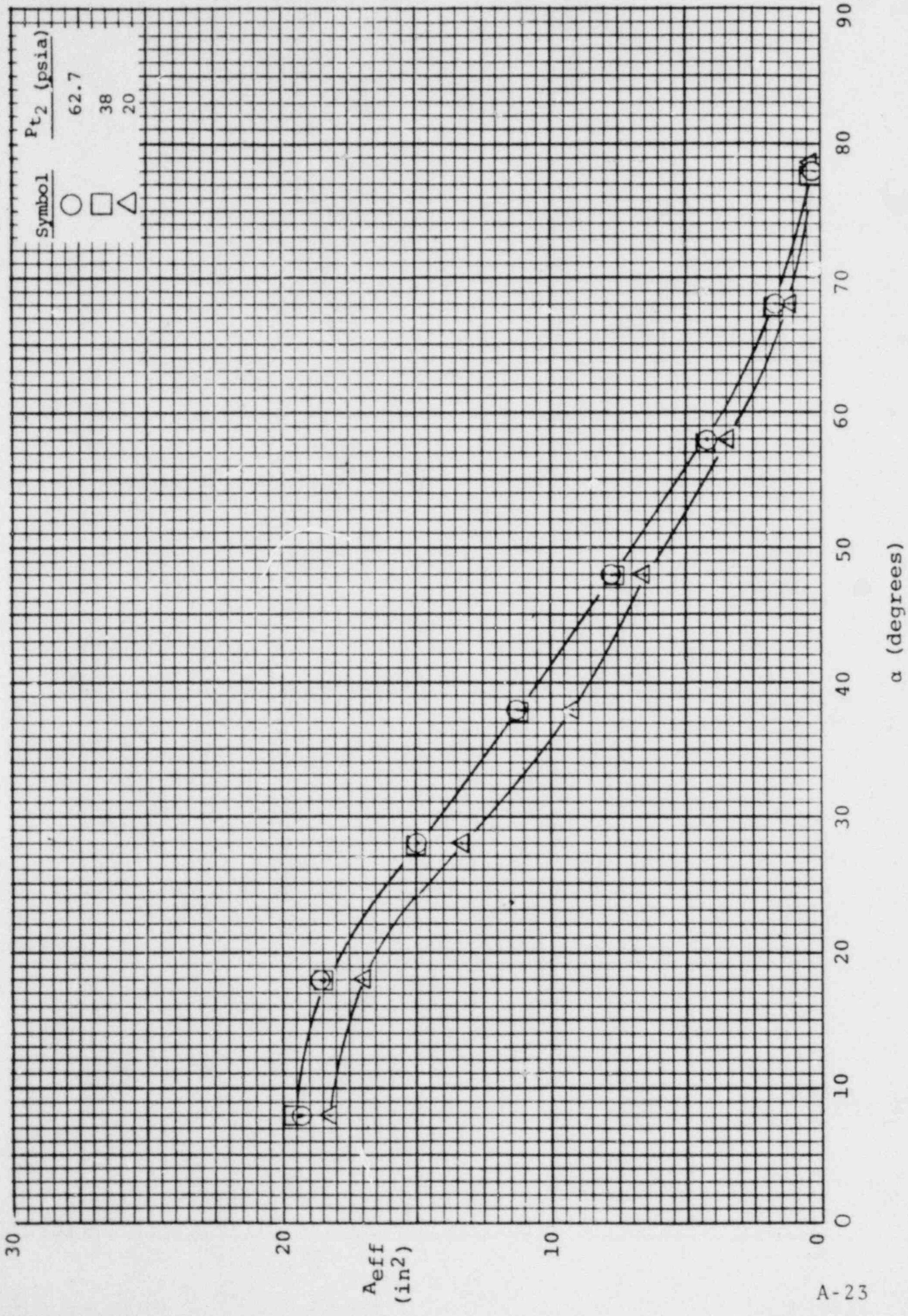


FIGURE 9. EFFECTIVE FLOW AREA VERSUS VALVE ANGLE

Test Procedure for Job 1240

1. Valve disc angle, α, was set by lab mechanic and double-checked by test engineer at each point. All valve pressures were run at this setting of α before α was changed.

2. The barometric pressure was measured (Hass mercury barometer) before each run and was used in the data reduction.

3. Before each test, the cameras on the P_{t1} and P_{t2} pressure gauges were checked and cocked.

4. Pre-run outputs of electronic instrumentation were recorded using the digital data system.

5. At the direction of the test engineer, airflow was started by a mechanic opening a manual control valve. The total pressure at the test valve was the independent variable monitored by the valve operator. The control pressure was observed using a 0-200 psi Heise differential pressure gauge.

6. When steady flow was established, all data were simultaneously recorded (torquemeter and thermocouples on data system pressure gauges and manometer photos on Polaroid film). Flow was then shut down.

7. Post-run outputs were recorded.

GL/RGB

8. All results and inputs were double-checked for transcription accuracy or any electronic anomalies.

9a. For run numbers 1-24, one set of data per blow was taken.

9b. For run numbers 25-30, only the torquemeter output was recorded. The output was recorded at 3-4 psi intervals of P_{t_2}. All other procedures were as described above.

JG/RGB

```
 70 REM DR1240A DATA REDUCTION PROGRAM - ISOLATION VALVE TEST
 80 REM 2/25/80 - G.F.
 95 SELECT PRINT 215(80)
    :S=0
100 READ D1,A1,N9,D$
110 READ N,B,P1,P2,I1,P4,P5,P6,P7,P8,P9,T1,T2,H0,V1
120 B=B*.49115
121 PRINT HEX( 0A0A0A)
122 IF S>0  THEN 127
123 PRINT HEX( 0E),"                           FLUIDYNE ENGINEERING CORPORATI
    ON"
124 PRINT HEX( 0A)
125 PRINT TAB( 24);"PROJECT 1240 - VALVE TESTS"
126 PRINT HEX( 0A0A)
127 PRINTUSING  551  ,P1,P2,I1,P4,P5,P6,P7,P8,P9
128 PRINT HEX( 0A)
130 P1=P1+B
140 P2=P2+B
150 P4=(I1-P4)*.489+B
160 P5=(I1-P5)*.489+B
170 P6=(I1-P6)*.489+B
171 P7=(I1-P7)*.489+B
172 P8=(I1-P8)*.489+B
173 P9=(I1-P9)*.489+B
175 Z=459.69
180 T1=T1+Z
190 T2=T2+Z
200 R1=.11486E8*P1*D1*(.8333*T1+198.6)/T1^2
210 C1=1-.184*R1^(-.2)
220 K1=.53160+(P1+16.9)*(1.581-.00834*(T1-520))*10E-6
230 W1=K1*C1*A1*P1/SQR( T1)
250 T1=T1-Z
260 T2=T2-Z
320 PRINT TAB( 8);"DATE  ";D$
330 PRINT HEX( 0A)
340 PRINTUSING  450  ,N
350 PRINTUSING  460  ,V1
360 PRINTUSING  470
370 PRINTUSING  480  ,W1,H0
380 PRINTUSING  490  ,P4,P5,P6
385 PRINTUSING  500  ,P7,P8,P9
390 PRINT HEX( 0A)
400 PRINTUSING  510  ,B
410 PRINTUSING  520  ,P1,P2
420 PRINTUSING  530  ,T1,T2
430 PRINTUSING  540  ,I1,A1
440 PRINTUSING  550  ,K1,C1,R1*1E-06
450 %              RUN NUMBER    ###.##
460 %              VALVE POSITION - ALPHA(DEGREES)=  -##.#
470 %              (ZERO DEG. - VALVE IS FULLY OPEN)
480 %              FLOW RATE (LBS/SEC)=   -##.###    TORQUE(IN-LBS)= -#
    ##.##
```


Gd /rcb

PROGRAM LISTING

```
490 %                 P4,P5,P6(PSIA)=     -###.##  -###.##  -###.##
500 %                 P7,P8,P9(PSIA)=     -###.##  -###.##  -###.##
510 %                 BAROMETER (PSIA)=   ##.###
520 %                 PT1(PSIA)=          -###.##           PT2(PSIA)=    -##
    #.##
530 %                 TT1(DEG.F)=         -###.##           TT2(DEG.F)=   -##
    #.##
540 %                 I1(INCHES OF HG.)=   ##.##            A1(SQ.IN.)=    ##.
    ###
550 %                 K1,C1,R1=           -#.##### -#.##### -##.#####
551 %       INPUTS  -###.## -###.## ##.## ##.## ##.## ##.## ##.## ##.## #
    #.##
560 S=S+1
    :IF S < 3  THEN  570
    :PRINT HEX( OC)
    :S=0
570 IF N9 = N  THEN  600
    :GOTO  110
600 END
```

3/19/80 DR1240A

LINE NUMBER CROSS REFERENCE

```
0110 -  0570
0127 -  0122
0450 -  0340
0460 -  0350
0470 -  0360
0480 -  0370
0490 -  0380
0500 -  0385
0510 -  0400
0520 -  0410
0530 -  0420
0540 -  0430
0550 -  0440
0551 -  0127
0570 -  0560
0600 -  0570
```


GF / 263

3/19/80 DR1240A

VARIABLE CROSS REFERENCE

```
A      - 0121 0124 0128 0330 0390
A()    - 0460
A0     - 0121 0126
A1     - 0100 0230 0430
A1()   - 0540
B      - 0110 0120 0120 0130 0140 0150 0160 0170 0171 0172 0173 0400
C      - 0480 0560
C1     - 0210 0230 0440 0550
D$     - 0100 0320
D1     - 0100 0200
E      - 0220 0440
E8     - 0200
F      - 0530 0530
G      - 0470 0530 0530
H0     - 0110 0370
I1     - 0110 0127 0150 0160 0170 0171 0172 0173 0430
I1()   - 0540
K1     - 0220 0230 0440 0550
N      - 0110 0340 0460 0570
N9     - 0100 0570
P1     - 0110 0127 0130 0130 0200 0220 0230 0410
P2     - 0110 0127 0140 0140 0410
P4     - 0110 0127 0150 0150 0380 0490
P5     - 0110 0127 0160 0160 0380 0490
P6     - 0110 0127 0170 0170 0380
P6()   - 0490
```


GF/RAB

```
P7     - 0110 0127 0171 0171 0385 0500

P8     - 0110 0127 0172 0172 0385 0500

P9     - 0110 0127 0173 0173 0385

P9()   - 0500

R      - 0450

R()    - 0510

R1     - 0200 0210 0440 0550

S      - 0095 0122 0460 0480 0480 0560 0560 0560 0560

T1     - 0110 0180 0180 0200 0200 0220 0230 0250 0250 0420

T2     - 0110 0190 0190 0260 0260 0420

V1     - 0110 0350

W1     - 0230 0370

Z      - 0175 0180 0190 0250 0260
```


RGB

FLUIDYNE ENGINEERING CORPORATION

PROJECT 1240 - VALVE TESTS

INPUTS 236.00 49.20 85.00 57.10 53.60 51.30 52.40 53.20 49.80

DATE 3-14-80

RUN NUMBER 1.10
VALVE POSITION - ALPHA(DEGREES)= 8
(ZERO D[illegible] - VALVE IS FULLY OPEN)
FLOW RATE (LBS/SEC)= 28.453 TORQUE(IN-LBS)= -15.0
P4, P5, P6(PSIA)= 27.77 29.48 30.60
P7, P8, P9(PSIA)= 30.06 29.67 31.33

BAROMETER (PSIA)= 14.126
PT1(PSIA)= 250.12 PT2(PSIA)= 63.32
TT1(DEG. F[illegible] 66.75 TT2(DEG. F)= 63.83
I1(INCHES [illegible] HG.)= 85.00 A1(SQ. IN.)= 4.905
K1, C1, R1= 0.53567 0.99337 16.50934 $\times 10^6$

INPUTS 138.20 24.20 84.80 78.95 76.80 75.30 76.05 76.66 74.50

DATE 3-14-80

RUN NUMBER 2.10
VALVE POSITION - ALPHA(DEGREES)= 8
(ZERO DEG. - VALVE IS FULLY OPEN)
FLOW RATE (LBS/SEC)= 17.366 TORQUE(IN-LBS)= -8.9
P4, P5, P6(PSIA)= 16.98 18.03 18.77
P7, P8, P9(PSIA)= 18.40 18.10 19.16

BAROMETER (PSIA)= 14.126
PT1(PSIA)= 152.32 PT2(PSIA)= 38.32
TT1(DEG. F)= 60.96 TT2(DEG. F)= 60.37
I1(INCHES OF HG.)= 84.80 A1(SQ. IN.)= 4.905
K1, C1, R1= 0.53426 0.99270 10.20120

INPUTS 61.50 6.10 84.80 84.60 83.65 83.40 84.40 83.60 83.35

DATE 3-14-80

RUN NUMBER 3.10
VALVE POSITION - ALPHA(DEGREES)= 8
(ZERO DEG. - VALVE IS FULLY OPEN)
FLOW RATE (LBS/SEC)= 8.604 TORQUE(IN-LBS)= +31.3
P4, P5, P6(PSIA)= 14.22 14.68 14.81
P7, P8, P9(PSIA)= 14.32 14.71 14.83

BAROMETER (PSIA)= 14.126
PT1(PSIA)= 75.62 PT2(PSIA)= 20.22
TT1(DEG. F)= 59.65 TT2(DEG. F)= 60.53
I1(INCHES OF HG.)= 84.80 A1(SQ. IN.)= 4.905
K1, C1, R1= 0.53306 0.99161 5.08146

[illegible] / RGB

FLUIDYNE ENGINEERING CORPORATION

PROJECT 1240 - VALVE TESTS

INPUTS 222.80 48.40 85.00 60.15 54.90 52.40 54.00 55.90 53.40

DATE 3-14-80

RUN NUMBER 4.00
VALVE POSITION - ALPHA(DEGREES)= 18
(ZERO DEG. - VALVE IS FULLY OPEN)
FLOW RATE (LBS/SEC)= 27.091 TORQUE(IN-LBS)= +56.4
P4, P5, P6(PSIA)= 26.28 28.85 30.07
P7, P8, P9(PSIA)= 29.29 28.36 29.58

BAROMETER (PSIA)= 14.132
PT1(PSIA)= 236.93 PT2(PSIA)= 62.53
TT1(DEG. F)= 61.16 TT2(DEG. F)= 61.43
I1(INCHES OF HG.)= 85.00 A1(SQ. IN.)= 4.905
K1, C1, R1= 0.53559 0.99332 15.85915

INPUTS 129.60 24.00 84.85 80.40 77.20 75.60 76.85 77.95 76.25

DATE 3-14-80

RUN NUMBER 5.00
VALVE POSITION - ALPHA(DEGREES)= 18
(ZERO DEG. - VALVE IS FULLY OPEN)
FLOW RATE (LBS/SEC)= 16.351 TORQUE(IN-LBS)= +29.3
P4, P5, P6(PSIA)= 16.30 17.87 18.65
P7, P8, P9(PSIA)= 18.04 17.50 18.33

BAROMETER (PSIA)= 14.132
PT1(PSIA)= 143.73 PT2(PSIA)= 38.13
TT1(DEG. F)= 62.84 TT2(DEG. F)= 61.40
I1(INCHES OF HG.)= 84.85 A1(SQ. IN.)= 4.905
K1, C1, R1= 0.53410 0.99261 9.58016

INPUTS 55.10 5.90 84.80 84.70 84.30 83.90 84.25 84.15 83.80

DATE 3-14-80

RUN NUMBER 6.00
VALVE POSITION - ALPHA(DEGREES)= 18
(ZERO DEG. - VALVE IS FULLY OPEN)
FLOW RATE (LBS/SEC)= 7.819 TORQUE(IN-LBS)= +59.9
P4, P5, P6(PSIA)= 14.18 14.38 14.57
P7, P8, P9(PSIA)= 14.40 14.45 14.62

BAROMETER (PSIA)= 14.138
PT1(PSIA)= 69.23 PT2(PSIA)= 20.03
TT1(DEG. F)= 66.84 TT2(DEG. F)= 66.12
I1(INCHES OF HG.)= 84.80 A1(SQ. IN.)= 4.905
K1, C1, R1= 0.53291 0.99143 4.56899

PAB

FLUIDYNE ENGINEERING CORPORATION

PROJECT 1240 - VALVE TESTS

INPUTS 175.50 48.10 84.90 71.90 68.10 65.80 70.60 69.70 66.60

DATE 3-14-80

RUN NUMBER 7.00
VALVE POSITION - ALPHA(DEGREES)= 28
(ZERO DEG. - VALVE IS FULLY OPEN)
FLOW RATE (LBS/SEC)= 21.477 TORQUE(IN-LBS)= +51.9
P4, P5, P6(PSIA)= 20.49 22.35 23.47
P7, P8, P9(PSIA)= 21.13 21.57 23.08

BAROMETER (PSIA)= 14.138
PT1(PSIA)= 189.63 PT2(PSIA)= 62.23
TT1(DEG. F)= 69.12 TT2(DEG. F)= 66.29
I1(INCHES OF HG.)= 84.90 A1(SQ. IN.)= 4.905
K1, C1, R1= 0.53471 0.99298 12.44338

INPUTS 102.70 24.15 84.80 85.90 82.45 81.70 84.55 83.40 82.00

DATE 3-14-80

RUN NUMBER 8.00
VALVE POSITION - ALPHA(DEGREES)= 28
(ZERO DEG. - VALVE IS FULLY OPEN)
FLOW RATE (LBS/SEC)= 13.213 TORQUE(IN-LBS)= +55.6
P4, P5, P6(PSIA)= 13.60 15.28 15.65
P7, P8, P9(PSIA)= 14.26 14.82 15.50

BAROMETER (PSIA)= 14.138
PT1(PSIA)= 116.83 PT2(PSIA)= 38.28
TT1(DEG. F)= 67.71 TT2(DEG. F)= 66.26
I1(INCHES OF HG.)= 84.80 A1(SQ. IN.)= 4.905
K1, C1, R1= 0.53363 0.99228 7.69340

INPUTS 40.70 6.10 84.80 85.35 84.90 84.35 85.50 84.40 84.40

DATE 3-14-80

RUN NUMBER 9.00
VALVE POSITION - ALPHA(DEGREES)= 28
(ZERO DEG. - VALVE IS FULLY OPEN)
FLOW RATE (LBS/SEC)= 6.182 TORQUE(IN-LBS)= +59.5
P4, P5, P6(PSIA)= 13.86 14.08 14.35
P7, P8, P9(PSIA)= 13.79 14.33 14.33

BAROMETER (PSIA)= 14.138
PT1(PSIA)= 54.83 PT2(PSIA)= 20.23
TT1(DEG. F)= 67.82 TT2(DEG. F)= 67.13
I1(INCHES OF HG.)= 84.80 A1(SQ. IN.)= 4.905
K1, C1, R1= 0.53268 0.99101 3.60994

FLUIDYNE ENGINEERING CORPORATION

PROJECT 1240 - VALVE TESTS

INPUTS 128.70 48.20 71.70 87.30 72.80 71.65 78.20 79.40 77.70

DATE 3-14-80

RUN NUMBER 10.00
VALVE POSITION - ALPHA(DEGREES)= 38
(ZERO DEG. - VALVE IS FULLY OPEN)
FLOW RATE (LBS/SEC)= 16.126 TORQUE(IN-LBS)= +64.6
P4,P5,P6(PSIA)= 6.63 13.72 14.28
P7,P8,P9(PSIA)= 11.08 10.49 11.32

BAROMETER (PSIA)= 14.263
PT1(PSIA)= 142.96 PT2(PSIA)= 62.46
TT1(DEG.F)= 71.48 TT2(DEG.F)= 70.23
I1(INCHES OF HG.)= 71.70 A1(SQ.IN.)= 4.905
K1,C1,R1= 0.53397 0.99257 9.32619

INPUTS 73.10 24.20 73.15 75.15 71.20 69.90 76.45 72.00 70.85

DATE 3-14-80

RUN NUMBER 11.10
VALVE POSITION - ALPHA(DEGREES)= 38
(ZERO DEG. - VALVE IS FULLY OPEN)
FLOW RATE (LBS/SEC)= 9.842 TORQUE(IN-LBS)= +80.1
P4,P5,P6(PSIA)= 13.28 15.21 15.85
P7,P8,P9(PSIA)= 12.65 14.82 15.38

BAROMETER (PSIA)= 14.263
PT1(PSIA)= 87.36 PT2(PSIA)= 38.46
TT1(DEG.F)= 70.33 TT2(DEG.F)= 70.60
I1(INCHES OF HG.)= 73.15 A1(SQ.IN.)= 4.905
K1,C1,R1= 0.53316 0.99180 5.71535

INPUTS 20.10 4.10 73.05 73.90 73.35 72.95 74.05 73.20 73.00

DATE 3-14-80

RUN NUMBER 12.00
VALVE POSITION - ALPHA(DEGREES)= 38
(ZERO DEG - VALVE IS FULLY OPEN)
FLOW RATE (LBS/SEC)= 3.835 TORQUE(IN-LBS)= +43.6
P4,P5,P6(PSIA)= 13.84 14.11 14.31
P7,P8,P9(PSIA)= 13.77 14.19 14.28

BAROMETER (PSIA)= 14.263
PT1(PSIA)= 34.36 PT2(PSIA)= 18.36
TT1(DEG.F)= 76.93 TT2(DEG.F)= 74.10
I1(INCHES OF HG.)= 73.05 A1(SQ.IN.)= 4.905
K1,C1,R1= 0.53233 0.99009 2.21197

RGB

FLUIDYNE ENGINEERING CORPORATION

PROJECT 1240 - VALVE TESTS

INPUTS 84.00 48.80 67.00 77.60 66.00 62.70 75.65 68.55 65.00

DATE 3-14-80

RUN NUMBER 13.00
VALVE POSITION - ALPHA(DEGREES)= 48
(ZERO DEG. - VALVE IS FULLY OPEN)
FLOW RATE (LBS/SEC)= 11.064 TORQUE(IN-LBS)= +57.9
P4, P5, P6(PSIA)= 9.08 14.75 16.36
P7, P8, P9(PSIA)= 10.03 13.50 15.24

BAROMETER (PSIA)= 14.265
PT1(PSIA)= 98.26 PT2(PSIA)= 63.06
TT1(DEG. F)= 71.46 TT2(DEG. F)= 71.22
I1(INCHES OF HG.)= 67.00 A1(SQ. IN.)= 4.905
K1, C1, R1= 0.53331 0.99199 6.41065

INPUTS 44.80 24.00 67.00 69.00 67.00 65.85 71.90 66.40 66.25

DATE 3-14-80

RUN NUMBER 14.00
VALVE POSITION - ALPHA(DEGREES)= 48
(ZERO DEG. - VALVE IS FULLY OPEN)
FLOW RATE (LBS/SEC)= 6.630 TORQUE(IN-LBS)= +69.6
P4, P5, P6(PSIA)= 13.28 14.26 14.82
P7, P8, P9(PSIA)= 11.86 14.55 14.63

BAROMETER (PSIA)= 14.265
PT1(PSIA)= 59.06 PT2(PSIA)= 38.26
TT1(DEG. F)= 72.52 TT2(DEG. F)= 72.19
I1(INCHES OF HG.)= 67.00 A1(SQ. IN.)= 4.905
K1, C1, R1= 0.53272 0.99113 3.84328

INPUTS 12.40 5.90 67.00 68.40 67.75 67.05 69.20 67.25 67.10

DATE 3-14-80

RUN NUMBER 15.00
VALVE POSITION - ALPHA(DEGREES)= 48
(ZERO DEG. - VALVE IS FULLY OPEN)
FLOW RATE (LBS/SEC)= 2.978 TORQUE(IN-LBS)= +30.1
P4, P5, P6(PSIA)= 13.58 13.89 14.24
P7, P8, P9(PSIA)= 13.19 14.14 14.21

BAROMETER (PSIA)= 14.265
PT1(PSIA)= 26.66 PT2(PSIA)= 20.16
TT1(DEG. F)= 75.42 TT2(DEG. F)= 73.44
I1(INCHES OF HG.)= 67.00 A1(SQ. IN.)= 4.905
K1, C1, R1= 0.53223 0.98958 1.72279

1/1/91 RGB

FLUIDYNE ENGINEERING CORPORATION

PROJECT 1240 - VALVE TESTS

INPUTS 11.40 5.80 67.00 68.40 67.70 67.05 69.15 67.30 67.05

DATE 3-14-80

RUN NUMBER 15.10
VALVE POSITION - ALPHA(DEGREES)= 48
(ZERO DEG. - VALVE IS FULLY OPEN)

FLOW RATE (LBS/SEC)=	2.868		TORQUE(IN-LBS)=	+29.80	
P4, P5, P6(PSIA)=	13.58	13.92	14.24		
P7, P8, P9(PSIA)=	13.21	14.11	14.24		
BAROMETER (PSIA)=	14.265				
PT1(PSIA)=	25.66		PT2(PSIA)=	20.06	
TT1(DEG. F)=	74.39		TT2(DEG. F)=	73.16	
I1(INCHES OF HG.)=	67.00		A1(SQ. IN.)=	4.905	
K1, C1, R1=	0.53222	0.98951	1.66237		

JGB

FLUIDYNE ENGINEERING CORPORATION

PROJECT 1240 - VALVE TESTS

INPUTS 321.00 48.70 67.00 69.60 69.10 67.20 74.10 69.60 65.90

DATE 3-21-80

RUN NUMBER 16.10
VALVE POSITION - ALPHA(DEGREES)= 58
(ZERO DEG - VALVE IS FULLY OPEN).
FLOW RATE (LBS/SEC)= 6.243 TORQUE(IN-LBS)= +36.80
P4, P5, P6(PSIA)= 12.99 13.23 14.16
P7, P8, P9(PSIA)= 10.79 12.99 14.80

BAROMETER (PSIA)= 14.263
PT1(PSIA)= 335.26 PT2(PSIA)= 62.96
TT1(DEG F)= 68.09 TT2(DEG F)= 66.71
I1(INCHES OF HG)= 67.00 A1(SQ IN)= 0.802
K1, C1, R1= 0.53693 0.99250 8.92263

RGB

FLUIDYNE ENGINEERING CORPORATION

PROJECT 1240 - VALVE TESTS

INPUTS 321.00 48.60 67.00 69.60 69.20 67.30 74.10 69.80 65.95

DATE 3-17-80

RUN NUMBER 16.20
VALVE POSITION - ALPHA(DEGREES)= 58
(ZERO DEG. - VALVE IS FULLY OPEN)
FLOW RATE (LBS/SEC)= 6.228 TORQUE(IN-LBS)= +42.8
P4,P5,P6(PSIA)= 12.99 13.18 14.11
P7,P8,P9(PSIA)= 10.79 12.89 14.77

BAROMETER (PSIA)= 14.263
PT1(PSIA)= 335.26 PT2(PSIA)= 62.86
TT1(DEG.F)= 70.52 TT2(DEG.F)= 68.23
I1(INCHES OF HG.)= 67.00 A1(SQ. IN.)= 0.802
K1,C1,R1= 0.53686 0.99249 8.86908

INPUTS 185.70 23.60 67.00 68.90 69.20 68.00 70.80 68.15 67.55

DATE 3-17-80

RUN NUMBER 17.10
VALVE POSITION - ALPHA(DEGREES)= 58
(ZERO DEG. - VALVE IS FULLY OPEN)
FLOW RATE (LBS/SEC)= 3.695 TORQUE(IN-LBS)= +37.0
P4,P5,P6(PSIA)= 13.33 13.18 13.77
P7,P8,P9(PSIA)= 12.40 13.70 13.99

BAROMETER (PSIA)= 14.263
PT1(PSIA)= 199.96 PT2(PSIA)= 37.86
TT1(DEG.F)= 71.03 TT2(DEG.F)= 67.98
I1(INCHES OF HG.)= 67.00 A1(SQ. IN.)= 0.802
K1,C1,R1= 0.53483 0.99167 5.28319

INPUTS 74.20 5.50 67.00 67.80 67.80 67.30 69.30 67.50 67.15

DATE 3-17-80

RUN NUMBER 18.10
VALVE POSITION - ALPHA(DEGREES)= 58
(ZERO DEG. - VALVE IS FULLY OPEN)
FLOW RATE (LBS/SEC)= 1.627 TORQUE(IN-LBS)= +16.6
P4,P5,P6(PSIA)= 13.87 13.87 14.11
P7,P8,P9(PSIA)= 13.13 14.01 14.19

BAROMETER (PSIA)= 14.263
PT1(PSIA)= 88.46 PT2(PSIA)= 19.76
TT1(DEG.F)= 70.87 TT2(DEG.F)= 67.91
I1(INCHES OF HG.)= 67.00 A1(SQ. IN.)= 0.802
K1,C1,R1= 0.53317 0.99020 2.33820

JL
RGB

FLUIDYNE ENGINEERING CORPORATION

PROJECT 1240 - VALVE TESTS

INPUTS 186.00 23.70 67.00 68.90 69.30 68.00 70.80 68.20 67.55

DATE 3-21-80

RUN NUMBER 17.00
VALVE POSITION - ALPHA(DEGREES)= 58
(ZERO DEG. - VALVE IS FULLY OPEN)
FLOW RATE (LBS/SEC)= 3.685 TORQUE(IN-LBS)= +30.10

P4, P5, P6(PSIA)=	13.33	13.13	13.77
P7, P8, P9(PSIA)=	12.40	13.67	13.90

BAROMETER (PSIA)= 14.263

PT1(PSIA)=	200.26	PT2(PSIA)=	37.96
TT1(DEG. F)=	75.42	TT2(DEG. F)=	73.44
I1(INCHES OF HG.)=	67.00	A1(SQ. IN.)=	0.802

K1, C1, R1= 0.53475 0.99166 5.23437

FLUIDYNE ENGINEERING CORPORATION

PROJECT 1240 - VALVE TESTS

INPUTS 109.90 49.00 67.00 68.60 68.05 67.90 70.75 68.25 67.40

DATE 3-17-80

RUN NUMBER 19.00
VALVE POSITION - ALPHA(DEGREES)= 68
(ZERO DEG. - VALVE IS FULLY OPEN)
FLOW RATE (LBS/SEC)= 2.293 TORQUE(IN-LBS)= -18.1
P4,P5,P6(PSIA)= 13.48 13.75 13.82
P7,P8,P9(PSIA)= 12.43 13.65 14.06

BAROMETER (PSIA)= 14.265
PT1(PSIA)= 124.16 PT2(PSIA)= 63.26
TT1(DEG.F)= 68.82 TT2(DEG.F)= 68.60
I1(INCHES OF HG.)= 67.00 A1(SQ.IN.)= 0.802
K1,C1,R1= 0.53373 0.99085 3.29852

INPUTS 59.10 23.60 67.00 67.40 67.65 67.30 68.60 67.70 67.00

DATE 3-17-80

RUN NUMBER 20.10
VALVE POSITION - ALPHA(DEGREES)= 68
(ZERO DEG. - VALVE IS FULLY OPEN)
FLOW RATE (LBS/SEC)= 1.350 TORQUE(IN-LBS)= -5.4
P4,P5,P6(PSIA)= 14.06 13.94 14.11
P7,P8,P9(PSIA)= 13.48 13.92 14.26

BAROMETER (PSIA)= 14.265
PT1(PSIA)= 73.36 PT2(PSIA)= 37.86
TT1(DEG.F)= 69.93 TT2(DEG.F)= 69.69
I1(INCHES OF HG.)= 67.00 A1(SQ.IN.)= 0.802
K1,C1,R1= 0.53295 0.98983 1.94364

INPUTS 18.60 5.60 67.00 67.20 67.25 67.20 67.45 67.30 67.20

DATE 3-17-80

RUN NUMBER 21.00
VALVE POSITION - ALPHA(DEGREES)= 68
(ZERO DEG. - VALVE IS FULLY OPEN)
FLOW RATE (LBS/SEC)= 0.596 TORQUE(IN-LBS)= -1.3
P4,P5,P6(PSIA)= 14.16 14.14 14.16
P7,P8,P9(PSIA)= 14.04 14.11 14.16

BAROMETER (PSIA)= 14.265
PT1(PSIA)= 32.86 PT2(PSIA)= 19.86
TT1(DEG.F)= 68.67 TT2(DEG.F)= 67.76
I1(INCHES OF HG.)= 67.00 A1(SQ.IN.)= 0.802
K1,C1,R1= 0.53235 0.98807 0.87341

FLUIDYNE ENGINEERING CORPORATION

JOB ______ CODE 1240 SHEET NO. 1 OF ____

BY GJ DATE 3/20/80

COMPONENT ______ REF. DRWG. ______ CHK.D. BY ______ DATE ______

SUBJECT Unchoked cases - Determining Mass flow REV. BY ______ DATE ______

$\therefore \dot{m}_1 = \dfrac{C_{D_1} A_1 K_1 P_{t_1}}{\sqrt{T_1}} \left(A^*/A\right)_1$ if $\lambda < 1.89$ in meter, or $P_s/P_{t_1} > .528$; flow unchoked

Run 21 $P_s/P_t = .595 \Rightarrow M = .89 \Rightarrow A/A^* = 1.011$

$\rightarrow A^*/A = .989$

$\therefore \dot{m}_1 = (\dot{m}_{1_{Rd}})(.989) = (.603)(.989) = .596$ (lbs/sec)

GJ 76B

FLUIDYNE ENGINEERING CORPORATION

PROJECT 1240 - VALVE TESTS

INPUTS 82.20 48.00 67.00 67.30 67.20 67.05 67.50 67.25 67.10

DATE 3-17-80

RUN NUMBER 22.00
VALVE POSITION - ALPHA(DEGREES)= 78
(ZERO DEG. - VALVE IS FULLY OPEN)
FLOW RATE (LBS/SEC)= 0.432 TORQUE(IN-LBS)= -46.2
P4,P5,P6(PSIA)= 14.08 14.13 14.20
P7,P8,P9(PSIA)= 13.98 14.10 14.18

BAROMETER (PSIA)= 14.231
PT1(PSIA)= 96.43 PT2(PSIA)= 62.23
TT1(DEG.F)= 69.64 TT2(DEG.F)= 71.75
I1(INCHES OF HG.)= 67.00 A1(SQ.IN.)= 0.195
K1,C1,R1= 0.53330 0.98891 1.26183

INPUTS 43.40 24.00 67.00 67.00 67.10 67.10 67.20 67.15 67.05

DATE 3-17-80

RUN NUMBER 23.00
VALVE POSITION - ALPHA(DEGREES)= 78
(ZERO DEG. - VALVE IS FULLY OPEN)
FLOW RATE (LBS/SEC)= 0.257 TORQUE(IN-LBS)= -16.9
P4,P5,P6(PSIA)= 14.23 14.18 14.18
P7,P8,P9(PSIA)= 14.13 14.15 14.20

BAROMETER (PSIA)= 14.231
PT1(PSIA)= 57.63 PT2(PSIA)= 38.23
TT1(DEG.F)= 68.62 TT2(DEG.F)= 69.70
I1(INCHES OF HG.)= 67.00 A1(SQ.IN.)= 0.195
K1,C1,R1= 0.53272 0.98772 0.75603

INPUTS 5.20 5.80 67.00 67.00 67.00 67.00 67.05 67.00 67.00

DATE 3-17-80

RUN NUMBER 24.00
VALVE POSITION - ALPHA(DEGREES)= 78
(ZERO DEG. - VALVE IS FULLY OPEN)
FLOW RATE (LBS/SEC)= ~~0.088~~ TORQUE(IN-LBS)= +1.5
P4,P5,P6(PSIA)= 14.23 14.23 14.23
P7,P8,P9(PSIA)= 14.20 14.23 14.23

BAROMETER (PSIA)= 14.231
PT1(PSIA)= ~~19.43~~ PT2(PSIA)= 20.03
TT1(DEG.F)= 77.40 TT2(DEG.F)= 65.50
I1(INCHES OF HG.)= 67.00 A1(SQ.IN.)= 0.195
K1,C1,R1= 0.53212 0.98467 0.24946

JT 2613

FLUIDYNE ENGINEERING CORPORATION

PROJECT 1240 - VALVE TESTS

INPUTS 82.60 48.70 67.00 67.30 67.20 67.05 67.50 67 25 67.10

DATE 3-17-80

RUN NUMBER 22.10
VALVE POSITION - ALPHA(DEGREES)= 78
(ZERO DEG. - VALVE IS FULLY OPEN)
FLOW RATE (LBS/SEC)= 0.433 TORQUE(IN-LBS)= -40.4
P4,P5,P6(PSIA)= 14.08 14.13 14.20
P7,P8,P9(PSIA)= 13.98 14.10 14.18

BAROMETER (PSIA)= 14.231
PT1(PSIA)= 96.83 PT2(PSIA)= 62.93
TT1(DEG.F)= 70.84 TT2(DEG.F)= 82.09
I1(INCHES OF HG.)= 67.00 A1(SQ.IN.)= 0.195
K1,C1,R1= 0.53329 0.98892 1.26331

HJ rLB

Approximate Calculation of "Predicted" Torque

Assumes: incompressible flow
use C_{T_3} plot in Ref 2, for axially-symmetric (not 2D) valve.

the factor 2260 is derived on next page.

$\alpha°$	C_{T_3}	$T \approx 2260 C_{T_3}$ (in-lbs)
0	0	0
10	.022	50
20	.030	68
30	.024	54
40	.0175	40
50	.011	25
60	.007	16
70	.004	9
80	.002	5
90	0	0

FLUIDYNE ENGINEERING CORPORATION

JOB ______ CODE ______ SHEET NO. 2 OF 2

COMPONENT ______ REF. DRWG. ______ BY RGB DATE ______

CHK.D. BY ______ DATE ______

SUBJECT ______ REV. BY ______ DATE ______

For $P_t = 20$ psia

$P_{exit} \simeq 14.2$ psia

Use NACA 1135 Compressible Flow Tables.

$\lambda = P_t / P_{exit} = 20/14.2 = 1.408$

$M = .72 \qquad \rho/\rho_t = .7814 \qquad V/a^* = .7508$

$a^* = 49 \sqrt{T^*} = 49 \sqrt{.8333\, T_{t_2}}$

$T_{t_2} \simeq 70°F = 530°R \qquad a^* = 1030$ ft/sec

$V = 773$ ft/sec

$\rho_t = .0765 \frac{lbs}{ft^3} \left(\frac{20}{14.7}\right) = .104$ lbs/ft³

$\rho = .0813$ lbs/ft³

$T = C_{T_3}\, \rho V^2 D^3$

$in \cdot lbs = (\ )\ \frac{lbs}{ft^3} \frac{sec^2}{g\, ft} \frac{ft^2}{sec^2}\, ft^3\ 12 \frac{in}{ft}$

$T\,[in\, lbs] = C_{T_3} \frac{\rho\,[lbs/ft^3]}{g\ \ 32.174\ ft/sec^2}\ V^2\,[ft/sec]^2\ D^3\,[ft^3]\ 12$

$T = C_{T_3} \frac{.0813}{32.174}\ 773^2\ (.5)^3\ 12 = \underline{\underline{2260\ C_{T_3}}}$

GL

FLUIDYNE ENGINEERING CORPORATION

PRESSURE GAGE RECORD

0-300 psi press gauge

MANUFACTURER Seegers

SERIAL NUMBER 7104

Calibration #1

Performed By: M. Watschke {1159-III}

Date: 7-27-78

Room Temp. At Test

Dead Weight Pressure	Gage Reading Increasing	Decreasing	
0	0	0	0
30	30.05	30.07	-.06
60	60.05	60.05	-.05
90	90.00	90.07	-.08
120	120.00	120.00	0
150	150.05	150.08	-.06
180	180.00	180.07	-.04
210	210.0	210.05	-.03
240	240.03	240.10	-.06
270	270.00	270.08	-.04
300	299.05		

Calibration #2

Performed By: JO 1240

Date: 2/28/80

Room Temp. At Test

Dead Weight Pressure	Gage Reading Increasing	Decreasing
0	0	0
30	29.95	29.90
60	60.00	60.00
90	90.00	89.95
120	119.95	120.00
150	150.00	150.00
180	180.00	180.05
210	209.90	209.95
240	240.00	240.00
270	269.90	269.90
300	295.95	

Calibration #3

Performed By:

Date:

Room Temp. At Test

Dead Weight Pressure	Gage Reading Increasing	Decreasing

Calibration #4

Performed By:

Date:

Room Temp. At Test

Dead Weight Pressure	Gage Reading Increasing	Decreasing

Use Record:

Date	Date	Date	Date
Job No.	Job No.	Job No.	Job No.
Engr.	Engr.	Engr.	Engr.
Date	Date	Date	Date
Job No.	Job No.	Job No.	Job No.
Engr.	Engr.	Engr.	Engr.

RGB

DIRECTIONS FOR PERFORMING CALIBRATIONS

1. Place gage in a room with controlled temperature one hour before calibrating.
2. Set the pressure to full scale of the gage.
3. Close the supply valve and let the system set for one (1) minute. If the pressure has dropped 0.1% of full scale, check for, and fix all leaks.
4. Exercise the gage through 4 cycles from zero to full scale before performing the calibration. (Be careful not to surge the gage with rapid pressure changes).
5. Starting at zero pressure, take readings (pictures) at each designated pressure point. Take each succeeding reading on increasing pressure until full scale is reached. Then take each succeeding reading on decreasing pressure until zero is reached. (Note: Do not allow the pressure to vary more than 10% of full scale while changing the weights on the dead weight tester.)

FLUIDYNE ENGINEERING CORPORATION

P2   0-200 psi press gauge

PRESSURE GAGE RECORD

MANUFACTURER: HEISE   SERIAL NUMBER: 57471

Calibration #1

Performed By: B.G. & B.K.

Date: 8-14-76 (1090)

Room Temp. At Test:

Dead Weight Pressure	Gage Reading Increasing	Gage Reading Decreasing
0	0	+.05
20	20.0	20.0
40	40.0	40.0
60	60.0	60.0
80	80.0	80.0
100	100.0	100.0
120	120.0	120.0
140	140.0	139.95
160	160.0	160.0
180	180.0	180.0
200	200.0	

Calibration #2

Performed By: M.W & BK

Date: 5-23-77

Room Temp. At Test:

Dead Weight Pressure	Gage Reading Increasing	Gage Reading Decreasing
0	0	0
20	20.0	20.05
40	40.0	40.05
60	60.0	60.05
80	80.0	80.0
100	100.0	100.05
120	120.0	120.0
140	140.0	140.0
160	160.0	160.1
180	180.0	180.15
200	200.0	

Calibration #3

Performed By: JO   JOB 1240

Date: 2/27/80

Room Temp. At Test:

Dead Weight Pressure	Gage Reading Increasing	Gage Reading Decreasing
0	0	0
20	20.0	20.05
40	40.0	40.0
60	60.0	60.05
80	80.0	80.0
100	100.0	100.0
120	120.0	120.0
140	140.0	140.0
160	160.0	160.05
180	180.0	180.0
200	200.0	

Calibration #4

Performed By:

Date:

Room Temp. At Test:

Dead Weight Pressure	Gage Reading Increasing	Gage Reading Decreasing

Use Record:

Date	Date	Date	Date
Job No.	Job No.	Job No.	Job No.
Engr.	Engr.	Engr.	Engr.
Date	Date	Date	Date
Job No.	Job No.	Job No.	Job No.
Engr.	Engr.	Engr.	Engr.

DIRECTIONS FOR PERFORMING CALIBRATIONS

1. Place gage in a room with controlled temperature one hour before calibrating.
2. Set the pressure to full scale of the gage.
3. Close the supply valve and let the system set for one (1) minute. If the pressure has dropped 0.1% of full scale, check for, and fix all leaks.
4. Exercise the gage through 4 cycles from zero to full scale before performing the calibration. (Be careful not to surge the gage with rapid pressure changes).
5. Starting at zero pressure, take readings (pictures) at each designated pressure point. Take each succeeding reading on increasing pressure until full scale is reached. Then take each succeeding reading on decreasing pressure until zero is reached. (Note: Do not allow the pressure to vary more than 10% of full scale while changing the weights on the dead weight tester.

*** FLUIDYNE ENGINEERING CORPORATION ***

Post test Calib 3/20/80

BALANCE = TORQUE METER
BRIDGE NO. = 2
CALIBRATION NO. = 80.03
DIRECTION OF LOAD(DEG.)= 0
LOAD = ROLL
STATION = 0
X = 0.000
Y = 0.000

JOB NO. 1240.00
DATE = 3/20/80
LOCATION = CALIBRATION ROOM
SIGN = NEG
EXCITATION (VOLTS)= 10.020
BALANCE TEMP(DEG F)= 74.0

ALL CALCULATIONS ARE IN MILLIVOLTS

W(LBS)	RDG	NET	LIN(1)	DELTA	LIN(2)	DELTA	INCRE	HYST
0	9.7048							
-40	8.8054	-0.8994	-0.903	0.004	-0.896	-0.002	0.8994	0.000
-80	7.9070	-1.7978	-1.797	-0.000	-1.792	-0.005	0.8984	0.000
-120	7.0108	-2.6940	-2.690	-0.003	-2.687	-0.006	0.8962	0.000
-160	6.1130	-3.5918	-3.583	-0.008	-3.582	-0.009	0.8978	0.000
-200	5.2368	-4.4680	-4.476	0.008	-4.477	0.009	0.8762	0.000
-160	6.1220	-3.5828	-3.577	-0.005	-3.582	-0.000	0.8852	0.000
-120	7.0212	-2.6836	-2.682	-0.001	-2.687	0.003	0.8992	0.006
-80	7.9172	-1.7876	-1.787	-0.000	-1.792	0.004	0.8960	0.009
-40	8.8154	-0.8894	-0.891	0.002	-0.896	0.007	0.8982	0.014
0	9.7132							

LOADING LEAST-SQUARES EQUATION= 4.85489E-01 + 4.48763E+02(LB*V/MV)
MAXIMUM LINEAR DEVIATION = 0.189 %

UNLOADING LEAST-SQUARES EQUATION= -1.53701E-01 + 4.47701E+02(LB*V/MV)
MAXIMUM LINEAR DEVIATION = 0.188 %

AVERAGE LEAST-SQUARES EQUATION= 8.10742E-02 + 4.47759E+02(LB*V/MV)
MAXIMUM LINEAR DEVIATION = 0.212 %
MAXIMUM HYSTERESIS = 0.212 %
ZERO SHIFT = 0.188 %

CALIBRATION BY WM. MOILANEN

GG

RGB

*** FLUIDYNE ENGINEERING CORPORATION ***

BALANCE = TORQUE METER
BRIDGE NO. = 2
CALIBRATION NO. = 80.04
DIRECTION OF LOAD(DEG.)= 0
LOAD = ROLL
STATION = 0
X = 0.000
Y = 0.000

JOB NO. 1240.00
DATE = 3/20/80
LOCATION = CALIBRATION ROOM
SIGN = NEG
EXCITATION (VOLTS)= 10.020
BALANCE TEMP(DEG F)= 74.0

ALL CALCULATIONS ARE IN MILLIVOLTS

W(LBS)	RDG.	NET	LIN(1)	DELTA	LIN(2)	DELTA	INCRE	HYST.
0	9.7150							
-40	8.8106	-0.9044	-0.905	0.001	-0.904	0.000	0.9044	0.000
-80	7.9120	-1.8030	-1.803	0.000	-1.802	-0.000	0.8986	0.000
-120	7.0132	-2.7018	-2.700	-0.001	-2.699	-0.002	0.8988	0.000
-160	6.1136	-3.6014	-3.597	-0.003	-3.597	-0.004	0.8996	0.000
-200	5.2228	-4.4922	-4.495	0.003	-4.494	0.002	0.8908	0.000
-160	6.1194	-3.5956	-3.595	-0.000	-3.597	0.001	0.8966	0.002
-120	7.0160	-2.6990	-2.698	-0.000	-2.699	0.000	0.8966	0.001
-80	7.9138	-1.8012	-1.801	0.000	-1.802	0.001	0.8978	0.001
-40	8.8106	-0.9044	-0.904	0.000	-0.904	0.000	0.8968	0.001
0	9.7070							

LOADING LEAST-SQUARES EQUATION= 3.72632E-01 + 4.46623E+02(LB*V/MV)
MAXIMUM LINEAR DEVIATION = 0.076 %

UNLOADING LEAST-SQUARES EQUATION= 3.33556E-01 + 4.46822E+02(LB*V/MV)
MAXIMUM LINEAR DEVIATION = 0.070 %

AVERAGE LEAST-SQUARES EQUATION= 3.31098E-01 + 4.46600E+02(LB*V/MV)
MAXIMUM LINEAR DEVIATION = 0.093 %
MAXIMUM HYSTERESIS = 0.052 %
ZERO SHIFT = -0.178 %

CALIBRATION BY:WM. MOILANEN

GT.

RGB

*** FLUIDYNE ENGINEERING CORPORATION ***

Calibration used during Test 3/12/80

BALANCE = TORQUEMETER
BRIDGE NO = 1
CALIBRATION NO = 72.02
DIRECTION OF LOAD(DEG.)= 0
LOAD = ROLL
STATION = 0
X = 0.000
Y = 0.000

JOB NO 1240.00
DATE = 3-12-80
LOCATION = CH 7
SIGN = NEG
EXCITATION (VOLTS)= 9.963
BALANCE TEMP(DEG F)= 70.0

ALL CALCULATIONS ARE IN MILLIVOLTS

W(LBS)	RDG.	NET	LIN(1)	DELTA	LIN(2)	DELTA	INCRE	HYST.
* 0	3.5500							
-100	1.3272	-2.2228	-2.232	0.009	-2.234	0.012	2.2228	0.000
-200	-0.8976	-4.4476	-4.439	-0.008	-4.441	-0.006	2.2248	0.000
-300	-3.1020	-6.6520	-6.646	-0.005	-6.647	-0.004	2.2044	0.000
-400	-5.3044	-8.8544	-8.853	-0.000	-8.853	-0.000	2.2024	0.000
-500	-7.5052	-11.0552	-11.060	0.005	-11.060	0.004	2.2008	0.000
-400	-5.3012	-8.8512	-8.852	0.000	-8.853	0.002	2.2040	0.002
-300	-3.0994	-6.6494	-6.647	-0.001	-6.647	-0.001	2.2018	-0.003
-200	-0.8969	-4.4468	-4.443	-0.003	-4.441	-0.005	2.2026	-0.007
-100	1.3146	-2.2354	-2.238	0.003	-2.234	-0.000	2.2114	-0.003
0	3.5230							

LOADING LEAST-SQUARES EQUATION= 1.12905E+00 + 4.51394E+02(LB*V/MV)
MAXIMUM LINEAR DEVIATION = 0.083 %

UNLOADING LEAST-SQUARES EQUATION= 1.56051E+[illegible] + 4.51959E+02(LB*V/MV)
MAXIMUM LINEAR DEVIATION = 0.049 %

AVERAGE LEAST-SQUARES EQUATION= 1.30084E+00 + 4.51578E+02(LB*V/MV)
MAXIMUM LINEAR DEVIATION = 0.109 %
MAXIMUM HYSTERESIS = [illegible] %
ZERO SHIFT = -0.244 %

CALIBRATION BY: M.K.

GF RGB

*** FLUIDYNE ENGINEERING CORPORATION ***

BALANCE = TORQUEMETER
BRIDGE NO = 1
CALIBRATION NO = 72.01
DIRECTION OF LOAD(DEG.)= 0
LOAD = ROLL
STATION = 0
X = 0.000
Y = 0.000

JOB NO. 1240.00
DATE = 3-12-80
LOCATION = CH.7
SIGN = NEG.
EXCITATION (VOLTS)= 9.963
BALANCE TEMP(DEG F)= 70.0

ALL CALCULATIONS ARE IN MILLIVOLTS

W(LBS)	RDG.	NET	LIN(1)	DELTA	LIN(2)	DELTA	INCRE	HYST.
0	3.5616							
-100	1.3328	-2.2288	-2.235	0.007	-2.221	-0.007	2.2288	0.000
-200	-0.8780	-4.4396	-4.437	-0.002	-4.425	-0.013	2.2108	0.000
-300	-3.0884	-6.6500	-6.638	-0.011	-6.630	-0.019	2.2104	0.000
-400	-5.2764	-8.8380	-8.840	0.002	-8.834	-0.003	2.1880	0.000
-500	-7.4750	-11.0366	-11.041	0.004	-11.039	0.002	2.1986	0.000
-400	-5.2668	-8.8284	-8.828	-0.000	-8.834	0.006	2.2082	0.011
-300	-3.0564	-6.6180	-6.621	0.003	-6.630	0.012	2.2104	0.020
-200	-0.8530	-4.4146	-4.414	-0.000	-4.425	0.011	2.2034	0.022
-100	1.3532	-2.2084	-2.207	-0.001	-2.221	0.012	2.2062	0.027
0	3.5660							

LOADING LEAST-SQUARES EQUATION= 1.56264[illegible] 4.52575E+02(LB*V/MV)
MAXIMUM LINEAR DEVIATION = 0[illegible]

UNLOADING LEAST-SQUARES EQUATION= 6.34339E-03 + 4.51423E+02(LB*V/MV)
MAXIMUM LINEAR DEVIATION = 0.043 %

AVERAGE LEAST-SQUARES EQUATION= 7.61120E-01 + 4.51948E+02(LB*V/MV)
MAXIMUM LINEAR DEVIATION = 0.179 %
MAXIMUM HYSTERESIS = 0.204 %
ZERO SHIFT = 0.039 %

CALIBRATION BY: M. F

KJ. RCT

Preliminary Cal.b 2/25/80

*** FLUIDYNE ENGINEERING CORPORATION ***

BALANCE = TORQUE METER
BRIDGE NO. = 2
CALIBRATION NO. = 56.02
DIRECTION OF LOAD(DEG.)= 0
LOAD = ROLL
STATION = 0
X = 0.000
Y = 0.000

JOB NO 1240.00
DATE = 2/25/80
LOCATION = CAL. RM.
SIGN = NEG.
EXCITATION (VOLTS)= 10.022
BALANCE TEMP(DEG F)= 72.0

ALL CALCULATIONS ARE IN MILLIVOLTS

W(LBS) (IN-LBS)	RDG.	NET	LIN(1)	DELTA	LIN(2)	DELTA	INCRE	HYST.
0	-1.6368							
-40	-2.5346	-0.8978	-0.898	0.000	-0.886	-0.011	0.8978	0.000
-80	-3.4300	-1.7932	-1.791	-0.002	-1.782	-0.010	0.8954	0.000
-120	-4.3184	-2.6816	-2.683	0.002	-2.678	-0.003	0.8884	0.000
-160	-5.2142	-3.5774	-3.576	-0.000	-3.574	-0.003	0.8958	0.000
-200	-6.1062	-4.4694	-4.469	-0.000	-4.470	0.000	0.8920	0.000
-160	-5.2080	-3.5712	-3.571	0.000	-3.574	0.003	0.8982	0.005
-120	-4.3130	-2.6762	-2.672	-0.003	-2.678	0.002	0.8950	0.007
-80	-3.4084	-1.7716	-1.773	0.001	-1.782	0.010	0.9046	0.019
-40	-2.5110	-0.8742	-0.874	0.000	-0.886	0.012	0.8974	0.024
0	-1.6114							

LOADING LEAST-SQUARES EQUATION= 2.53601E-01 + 4.49044E+02(LB*V/MV)
MAXIMUM LINEAR DEVIATION = 0.051 %

UNLOADING LEAST-SQUARES EQUATION= -1.08921E+00 + 4.45917E+02(LB*V/MV)
MAXIMUM LINEAR DEVIATION = 0.082 %

AVERAGE LEAST-SQUARES EQUATION= -4.27002E-01 + 4.47437E+02(LB*V/MV)
MAXIMUM LINEAR DEVIATION = 0.255 %
MAXIMUM HYSTERESIS = 0.437 %
ZERO SHIFT = 0.568 %

CALIBRATION BY:WM. MOILANEN

*** FLUIDYNE ENGINEERING CORPORATION ***

BALANCE = TORQUE METER
BRIDGE NO. = 2
CALIBRATION NO. = 56.03
DIRECTION OF LOAD(DEG.)= 0
LOAD = ROLL
STATION = 0
X = 0.000
Y = 0.000

JOB NO. 1249.00
DATE = 2/25/80
LOCATION = CAL. RM.
SIGN = NEG.
EXCITATION (VOLTS)= 10.022
BALANCE TEMP(DEG F)= 72.1

ALL CALCULATIONS ARE IN MILLIVOLTS

W(LBS)	RDG.	NET	LIN(1)	DELTA	LIN(2)	DELTA	INCRE	HYST.
0	-1.5864							
-40	-2.4814	-0.8950	-0.894	-0.000	-0.889	-0.005	0.8950	0.000
-80	-3.3778	-1.7914	-1.790	-0.001	-1.786	-0.005	0.8964	0.000
-120	-4.2698	-2.6834	-2.685	0.002	-2.682	-0.000	0.8920	0.000
-160	-5.1680	-3.5816	-3.580	-0.000	-3.579	-0.002	0.8982	0.000
-200	-6.0630	-4.4766	-4.476	-0.000	-4.476	-0.000	0.8950	0.000
-160	-5.1628	-3.5764	-3.578	0.001	-3.579	0.003	0.9002	0.004
-120	-4.2702	-2.6838	-2.680	-0.003	-2.682	-0.000	0.8926	0.001
-80	-3.3668	-1.7804	-1.782	0.001	-1.786	0.005	0.9034	0.009
-40	-2.4704	-0.8840	-0.884	0.000	-0.889	0.005	0.8964	0.010
0	-1.5734							

LOADING LEAST-SQUARES EQUATION= -1.87638E-02 + 4.47740E+02(LB*V/MV)
MAXIMUM LINEAR DEVIATION = 0.049 %

UNLOADING LEAST-SQUARES EQUATION= -6.28869E-01 + 4.46354E+02(LB*V/MV)
MAXIMUM LINEAR DEVIATION = 0.079 %

AVERAGE LEAST-SQUARES EQUATION= -3.21496E-01 + 4.47062E+02(LB*V/MV)
MAXIMUM LINEAR DEVIATION = 0.129 %
MAXIMUM HYSTERESIS = 0.220 %
ZERO SHIFT = 0.290 %

CALIBRATION BY:WM. MOILANEN

HCJ RGB

*** FLUIDYNE ENGINEERING CORPORATION ***

BALANCE = TORQUE METER
BRIDGE NO = 2
CALIBRATION NO = 56.04
DIRECTION OF LOAD(DEG.)= 0
LOAD = ROLL
STATION = 0
X = 0.000
Y = 0.000

JOB NO 1240.00
DATE = 2/25/80
LOCATION = CAL. RM
SIGN = POS.
EXCITATION (VOLTS)= 10.022
BALANCE TEMP(DEG F)= 72.1

ALL CALCULATIONS ARE IN MILLIVOLTS

W(LBS)	RDG	NET	LIN(1)	DELTA	LIN(2)	DELTA	INCRE	HYST
0	-1.5508							
40	-0.6566	0.8942	0.895	-0.001	0.896	-0.002	0.8942	0.000
80	0.2416	1.7924	1.791	0.001	1.792	0.000	0.8982	0.000
120	1.1370	2.6878	2.686	0.001	2.687	0.000	0.8954	0.000
160	2.0302	3.5810	3.581	-0.000	3.583	-0.002	0.8932	0.000
200	2.9262	4.4770	4.477	-0.000	4.478	-0.001	0.8960	0.000
160	2.0308	3.5816	3.583	-0.002	3.583	-0.001	0.8954	-0.000
120	1.1460	2.6968	2.688	0.008	2.687	0.009	0.8848	0.010
80	0.2390	1.7898	1.793	-0.003	1.792	-0.002	0.9070	-0.001
40	-0.6540	0.8968	0.897	-0.001	0.896	0.000	0.8930	0.001
0	-1.5536							

LOADING LEAST-SQUARES EQUATION= -9.82779E-03 + 4.47700E+02(LB*V/MV)
MAXIMUM LINEAR DEVIATION = 0.032 %

UNLOADING LEAST-SQUARES EQUATION= -1.22428E-01 + 4.47800E+02(LB*V/MV)
MAXIMUM LINEAR DEVIATION = 0.187 %

AVERAGE LEAST-SQUARES EQUATION= -5.23266E-02 + 4.47673E+02(LB*V/MV)
MAXIMUM LINEAR DEVIATION = 0.205 %
MAXIMUM HYSTERESIS = 0.230 %
ZERO SHIFT = 0.062 %

CALIBRATION BY:WM. MOILANEN

KJ REB

*** FLUIDYNE ENGINEERING CORPORATION ***

BALANCE = TORQUE METER
BRIDGE NO = 2
CALIBRATION NO. = 56.05
DIRECTION OF LOAD(DEG.)= 0
LOAD = ROLL
STATION = 0
X = 0.000
Y = 0.000

JOB NO. 1240.00
DATE = 2/25/80
LOCATION = CAL. RM
SIGN = POS.
EXCITATION (VOLTS)= 10.022
BALANCE TEMP(DEG F)= 72.2

ALL CALCULATIONS ARE IN MILLIVOLTS

W(LBS)	RDG.	NET	LIN(1)	DELTA	LIN(2)	DELTA	INCRE	HYST.
0	-1.5526							
40	-0.6554	0.8972	0.897	-0.000	0.899	-0.001	0.8972	0.000
80	0.2430	1.7956	1.793	0.001	1.795	0.000	0.8984	0.000
120	1.1348	2.6874	2.690	-0.002	2.691	-0.003	0.8918	0.000
160	2.0354	3.5880	3.586	0.001	3.587	0.000	0.9006	0.000
200	2.9308	4.4834	4.483	0.000	4.483	0.000	0.8954	0.000
160	2.0334	3.5860	3.587	-0.001	3.587	-0.001	0.8974	-0.000
120	1.1406	2.6932	2.692	0.001	2.691	0.001	0.8928	0.002
80	0.2470	1.7996	1.796	0.003	1.795	0.004	0.8936	0.005
40	-0.6540	0.8986	0.900	-0.002	0.899	-0.000	0.9010	0.001
0	-1.5470							

LOADING LEAST-SQUARES EQUATION= -3.92646E-02 + 4.47171E+02(LB*V/MV)
MAXIMUM LINEAR DEVIATION = 0.065 %

UNLOADING LEAST-SQUARES EQUATION= -2.39392E-01 + 4.47610E+02(LB*V/MV)
MAXIMUM LINEAR DEVIATION = 0.067 %

AVERAGE LEAST-SQUARES EQUATION= -1.40301E-01 + 4.47396E+02(LB*V/MV)
MAXIMUM LINEAR DEVIATION = 0.098 %
MAXIMUM HYSTERESIS = 0.128 %
ZERO SHIFT = -0.124 %

CALIBRATION BY: WM. MOILANEN

KGB

APPENDIX B

NUTECH Stress Analysis of 18-Inch

Butterfly Valve

STRESS ANALYSIS OF 18" BUTTERFLY VALVE

REFERENCES:

1. T. Sarphara, "Torque & Cavitation Characteristics of Butterfly Valve", ASME Journal, Dec. 1961
2. Richard H.F. Pao, "Fluid Mechanics" John Wiley & Sons, Inc. 1961 Pg 123
3. Dresden Station Maintenance File No. 1601-20, "General Arrangement" | Δ
4. S.P. Timoshenko and D.H. Young, "Elements of Strength of Materials", D. Van Norstrand Company, Inc., 1968 Pg. 185
5. M.F. Spotts, "Design of Machine Elements", Prentice-Hall, Inc., 1971. Pg 123
6. Robert C. Jovinall, "Engineering Consideration of Stress, Strain and Strength", McGraw-Hill Book Company Pg 244
7. Henry Pratt Co., Drwg No. F-41, "Parts and Material Specifications", Dec. 10, 1968
8. Oberg, Jones, "Machinery's Handbook", 19th Ed. Pg 382, Pg 1132-1140 Pins
9. Fluidyne Engineering Corporation Test Report
10. NUTECH Calculation, "Hydrodynamic Analysis of 18" Pratt Butterfly Valve", Revision 1, dated 12/10/79, Pg 9, File no. 64.801.0004

Project Dresden Nuclear Power Station File No. 64.801.0006
Owner Commonwealth Edison Company
Client Commonwealth Edison Company

SUMMARY:

SHAFT:

STRESS and FACTOR OF SAFETY (in brackets)

LOCATION / VALVE ANGLE	A	B	C	D	Page
8°			11,235 (1.34)		22-24
18°		3,110 (4.8)	11,257 (1.33)	2,607 (5.75)	11-15
28°		3055 (4.91)	7,555 (1.99)	2,106 (7.12)	16-18
38°	4152.7 (3.6)	4,291 (3.50)	10,194 (1.47)	1,010 (14.9)	3-9
48°		3,331 (4.5)	10,711 (1.40)	439 (34.1)	19-21

KEY: Shear Stress 1,293 psi

Comp. Stress 2,586 psi

PIN: Unit Working Stress 2,032 psi

ACTUATOR ARM: Bending Stress 1,755 psi

Revision	0					Page 2 of 25
Prepared By/Date	TSH/4-9-80					
Checked By/Date	DAG 4-9-80					

SHAFT:

LOADS ON VALVE SHAFT

1. Max. Hydrodynamic Torque tending to shut the valve

   The valve test (Ref. 9) reveals that the max. torque generated in the valve is about 200 ft-lbs. Here we use $T_R = 300^{ft\text{-}lbs}$ for conservatism. And assuming that the max. torque occurs as disk at 38° from full open position (from Ref. 9)

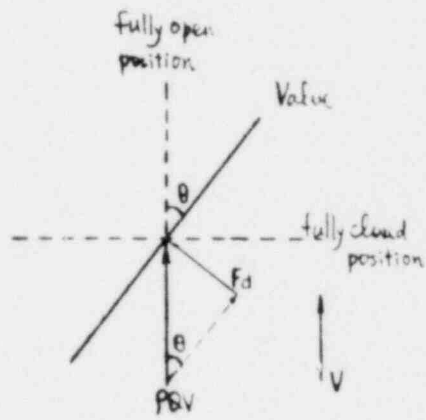
2. Restraining Torque by Actuator

   $T_a = T_R = 300^{ft\text{-}lbs} = 3600^{in\text{-}lbs}$

3. Piston Force generating restraining torque T_a

   $F_a = \frac{T_a}{L} = \frac{3600}{10} = 360^{lbs}$

   where L = actuator arm = 10"



4. Force due to discharged air (Ref. 2)

   $F_d = \rho Q V \sin\theta = \frac{\gamma}{g} A V^2 \sin\theta$

   where γ = specific weight of air at 20°F, 63 psia = $0.354^{lb}/_{ft^3}$

   g = gravitational acceleration = 32.2 ft/sec²

   A = disk area = $\frac{\pi}{4}\left(\frac{18}{12}\right)^2 = 1.767$ ft²

   V = Air Velocity at valve = 300 ft/sec (see Insert A, next page)

   θ = 38° (see item 1 above)

   $\therefore F_d = \frac{0.354}{32.2}(1.767)(300)^2 \sin 38° = 1076$ lbs

Insert A

From page 10 of Ref. 10:

$$\frac{A_{eff}}{A_{nom}} = \left(\frac{C_{c_1} + C_{c_2}}{2}\right)\left[1 - \sin 38^\circ\right]$$

$$= \left(\frac{0.776 + 0.570}{2}\right)(1 - \sin 38^\circ)$$

$$= 0.259$$

Assuming sonic velocity at valve throat, the velocity upstream V_u is:

$$\dot{m} = \text{constant} = \rho V_u A_u = \rho V_T A_T$$

with V_u = velocity upstream
V_T = velocity at valve throat
A_u = area upstream
A_T = area at throat

Assume incompressible flow → ρ constant

$$V_u A_u = V_T A_T$$

$$V_u = V_T \left(\frac{A_T}{A_u}\right) = V_T \left(\frac{A_{eff}}{A_{nom}}\right)$$

$$= \sqrt{\gamma g_c R T}\ (.259)$$

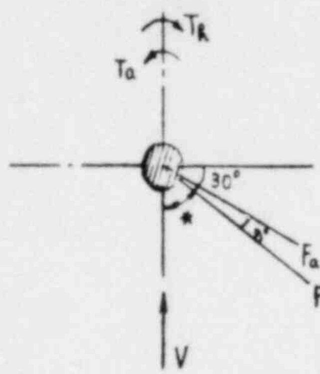
$$= \sqrt{(1.4)(32.2)(53.3)(560)}\ (.259)$$

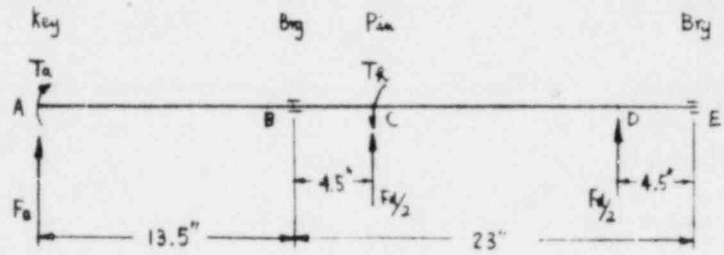
This assumes conservative temperature of 100°F

$$V_u = (1160)(.259)$$

$$= 300.4\ \text{ft/sec}$$

5. Find max. bending moment and shear force :





(Dimensions from Ref. 3)

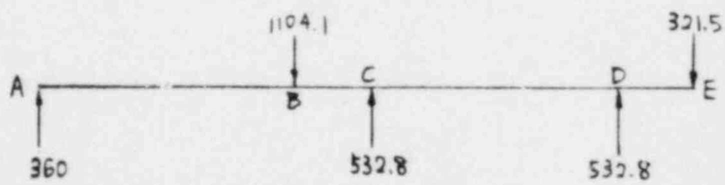
* The angle between piston force and flow direction = 60° obtaining from Ref 3 top view and position 1

<a> In the plane containing F_a :

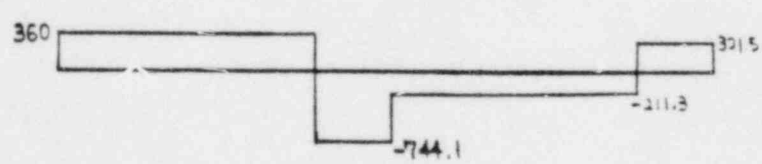
$$F_{C_1} = F_{D_1} = \frac{F_d}{2}\cos 8^\circ = \frac{1076}{2}\cos 8^\circ = 532.8^{\text{lbs}}$$

$$R_{B_1} = \frac{1}{23}\left[360 \times 36.5 + 532.8 \times 18.5 + 532.8 \times 4.5\right] = 1104.1 \text{ lbs}$$

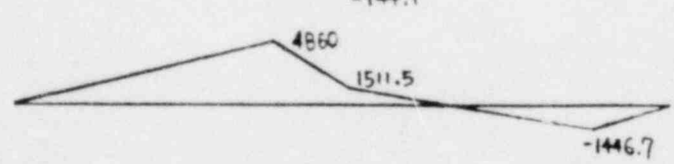
$$R_{E_1} = 360 + 532.8 \times 2 - 1104.1 = 321.5 \text{ lbs}$$



SFD



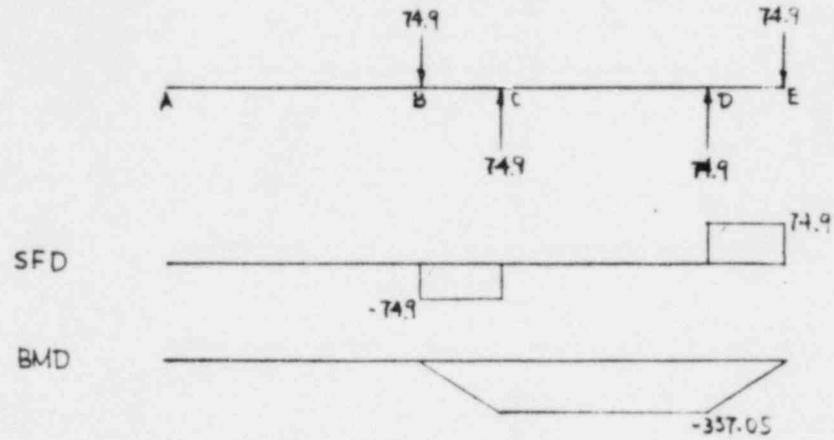
BMD



<b> In the plane perpendicular to Fa:

$$F_{C_2} = F_{D_2} = \frac{F_d}{2}\sin 8^\circ \quad \frac{1076}{2}\sin 8^\circ = 74.9^{\,lbs}$$

$$R_{B_2} = R_{E_2} = 74.9^{\,lbs}$$



<C> From <a> & <b> above, we found

max. bending moment $M_{max} = 4860^{\,in\text{-}lb}$ at B

shear force at B $V_B = \sqrt{(744.1)^2 + (74.9)^2} = 747.9^{\,lbs}$ (to the right of B)

bending moment at C $M_C = \sqrt{(1511.5)^2 + (337.05)^2} = 1548.6^{\,in\text{-}lbs}$

shear force at C $V_C = \sqrt{(744.1)^2 + (74.9)^2} = 747.9^{\,lbs}$ (to the left of C)

bending moment at D $M_D = \sqrt{(1446.7)^2 + (337.05)^2} = 1485.4^{\,in\text{-}lbs}$

shear force at D $V_D = \sqrt{(321.5)^2 + (74.9)^2} = 330.1^{\,lbs}$ (to the right of D)

bending moment at A $M_A = 0$

shear force at A $V_A = 360^{\,lbs}$

STRESS CALCULATIONS:

1. At B: $M_{max} = 4860$ in-lbs, $T = 3{,}600$ in-lbs, $V_B = 747.9$ lbs

Bending Stress $\sigma = \frac{MC}{I} = \frac{32M}{\pi D^3} = \frac{32(4860)}{\pi(2.25)^3} = 4346$ psi

Torsional Stress $\tau' = \frac{Tr}{J} = \frac{16T}{\pi D^3} = \frac{16(3600)}{\pi(2.25)^3} = 1610$ psi

Shear in bending $\tau'' = \frac{4V}{3A}$ (Ref. 4)

A = Cross Sectional Area $= \frac{\pi}{4}(2.25)^2 = 3.976$ in²

$\tau'' = \frac{4(747.9)}{3(3.976)} = 250.8$ psi

Total Shear $\tau = \tau' + \tau'' = 1610 + 250.8 = 1860.8$ psi

Using ASME Code for transmission shafting (Ref. 5)

$$S = \frac{0.5\,S_y}{FS} = \sqrt{\left(C_m \frac{\sigma}{2}\right)^2 + (C_t\,\tau)^2}$$

where S = max shear stress due to combined bending and torsion

S_y = Yield Stress = 30,000 psi for Stainless Steel Type 304 (from ASME Code)

FS = Factor of Safety

C_m & C_t = Shock and fatigue factors

$C_m = C_t = 1.5$ for a suddenly applied load on a stationary shaft

hence $S = \sqrt{\left(1.5\,\frac{4346}{2}\right)^2 + (1.5 \times 1860.8)^2} = \underline{4291}$ psi

$FS = \frac{0.5\,S_y}{S} = \frac{15{,}000}{4291} = \underline{3.50}$

nutech

San Jose, California

Project Dresden Nuclear Power Station File No. 64.801.0006

Owner Commonwealth Edison Company

Client Commonwealth Edison Company

2. Now we look at the stress concentration at shaft pin (at C)

$d/D = \frac{1}{2.25} = 0.44$, $M_C = 1548.6$ in lb, $T = 3{,}600$ in lb, $V_c = 747.9$ lb

Stress concentration factor due to bending $k_b = 1.86$ (Ref. 6)

$$\sigma = K_b \cdot \sigma_{nom} = K_b \cdot \frac{M}{\pi D^3/32 - dD^2/6} = (1.86)\frac{1548.6}{0.2745} = 10{,}493 \text{ psi}$$

where $\frac{\pi D^3}{32} - \frac{dD^2}{6} = \frac{\pi(2.25)^3}{32} - \frac{1\cdot(2.25)^2}{6} = 0.2745 \text{ in}^3$

Stress concentration factor due to torsion $K_t = 1.38$ (Ref 6)

$$\tau' = K_t \cdot \tau_{nom} = K_t \cdot \frac{T}{\pi D^3/16 - dD^2/6} = (1.38)\frac{3600}{1.3928} = 3567 \text{ psi}$$

where $\frac{\pi D^3}{16} - \frac{dD^2}{6} = \frac{\pi(2.25)^3}{16} - \frac{1\cdot(2.25)^2}{6} = 1.3928 \text{ in}^3$

Assume stress concentration factor associated with τ'', $K_S = 3.0$

$$\tau'' = K_S \frac{4V}{3A} = (3.0)\frac{4(747.9)}{3(3.976)} = 752.4 \text{ psi}$$

$$\tau = \tau' + \tau'' = 3567 + 752.4 = 4319.4 \text{ psi}$$

Using ASME Code, $C_m = C_t = 1.5$

$$S = \sqrt{\left(\frac{C_m\sigma}{2}\right)^2 + (C_t\tau)^2} = 1.5 \times \sqrt{\left(\frac{10{,}493}{2}\right)^2 + (4319.4)^2} = \underline{\underline{10{,}194}} \text{ psi}$$

$$FS = \frac{0.5\,S_y}{S} = \frac{15{,}000}{10{,}194} = \underline{\underline{1.47}}$$

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3. Keyway effect in shaft (at A)

$M_A = 0 \qquad T = 3{,}600 \text{ in-lbs} \qquad V_A = 360 \text{ lbs}$

Bending stress $\sigma = 0$

Torsional Stress $\tau' = \frac{16T}{\pi D^3} = \frac{16(3600)}{\pi(2.25)^3} = 1609.6 \text{ psi}$

Shear in bending $\tau'' = \frac{4V}{3A} = \frac{4(360)}{3(3.976)} = 120.7 \text{ psi}$

$\tau = \tau' + \tau'' = 1609.6 + 120.7 = 1730.3 \text{ psi}$

Apply stress concentration factor of 1.6 (Ref 6 p.252), and use ASME Code:

$S = (1.5)(1.6)\sqrt{(\frac{\sigma}{2})^2 + \tau^2} = (1.5)(1.6)(1730.3) = \underline{\underline{4152.7 \text{ psi}}}$

$FS = \frac{0.5S_y}{S} = \frac{15000}{4152.7} = \underline{\underline{3.6}}$

4. At D: $M_D = 1485.4 \text{ in-lbs} \qquad T = 0 \qquad V_D = 330.1 \text{ lbs}$

$\sigma = \frac{32M}{\pi D^3} = \frac{32(1485.4)}{\pi(2.25)^3} = 1328.4 \text{ psi}$

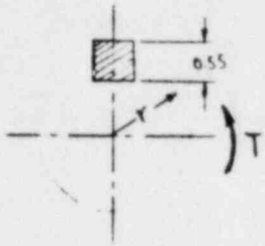
$\tau' = 0$

$\tau'' = \frac{4V}{3A} = \frac{4(330.1)}{3(3.976)} = 110.7 \text{ psi}$

$S = (1.5)\sqrt{(\frac{\sigma}{2})^2 + \tau^2} = (1.5)\sqrt{(\frac{1328.4}{2})^2 + (110.7)^2} = 1010 \text{ psi}$

$FS = \frac{0.5S_y}{S} = \frac{15{,}000}{1{,}010} = \underline{\underline{14.9}}$

SQUARE KEY: (at A) AISI Cold Drawn Steel



Assume key dimensions 0.55" x 0.55" x 4.5" long (Ref. 7)

Force at shaft surface: $F = \frac{T}{r} = \frac{3600}{1.125} = 3200$ lbs

Area in shear for key: $0.55 \times 4.5 = 2.475$ in²

Shear stress in key: $S_s = \frac{3200}{2.475} = \underline{\underline{1293}}$ psi O.K

Area in shaft for key: $\frac{0.55}{2} \times 4.5 = 1.2375$ in²

Compressive stress in key: $S_c = \frac{3200}{1.2375} = \underline{\underline{2586}}$ psi O.K

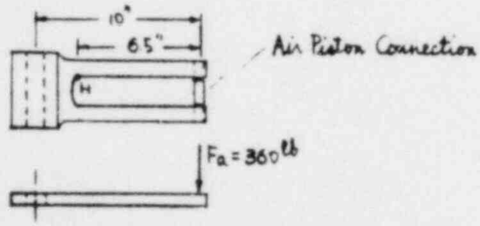
TAPERED PIN: (at C) Stainless Steel 18-8 Type 304 — $S_y = 30{,}000$ psi

Unit working stress on the pin in shear (Ref. 8)

$$S_u = \frac{1.27 \cdot T}{D \cdot d^2} = \frac{1.27 \cdot (3600)}{(2.25) \cdot (1)^2} = \underline{\underline{2032}} \text{ psi} \quad \text{O.K}$$

where d = mean dia of pin = 1"

ACTUATOR ARM:



Moment at H = $360 \times 6.5 = 2340$ in-lbs

Assume the arm has rectangular cross-section

$$\sigma = \frac{MC}{I} = \frac{M\left(\frac{h}{2}\right)}{bh^3/12} = \frac{6M}{bh^2} = \frac{6 \cdot (2340)}{(2 \cdot 1) \cdot 2^2} = \underline{\underline{1755}} \text{ psi} \quad \text{O.K}$$

Since the maximum stresses may not occur exactly at the point of maximum hydrodynamic torque, we must calculate stresses on either side of $\alpha = 38°$:

I. At $\alpha = 18°$

• $T = 135$ ft-lbs from Ref. 9

For conservatism multiply T by 1.5 → $T_a = 202.5$ ft-lbs.

• $F_a = \frac{T_a}{L} = \frac{202.5(12)}{10} = 243$ lbs. Lever arm $L = 10''$

• $\frac{A_{eff}}{A_{nom}} = \left(\frac{C_{C_1} + C_{C_2}}{2}\right)(1 - \sin\alpha)$ (Ref. 10)

$= \left(\frac{0.862 + 0.572}{2}\right)(1 - \sin 18°)$

$= .495$

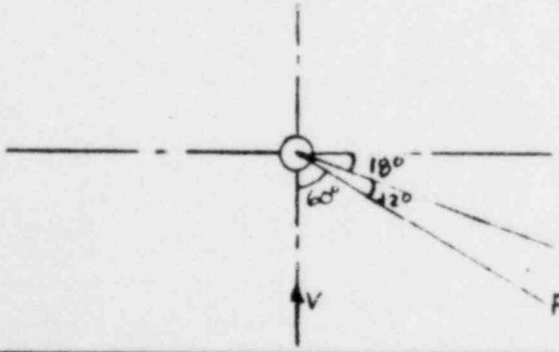
• $V_u = V_T\left(\frac{A_{eff}}{A_{nom}}\right) = (1160)(.495)$

$= 574.2$ ft/sec

• $F_d = \frac{\gamma}{g} A V^2 \sin\alpha$

$= \left(\frac{.354}{32.2}\right)(1.767)(574.2)^2 \sin(18°)$

$= 1979.2$ lbs.

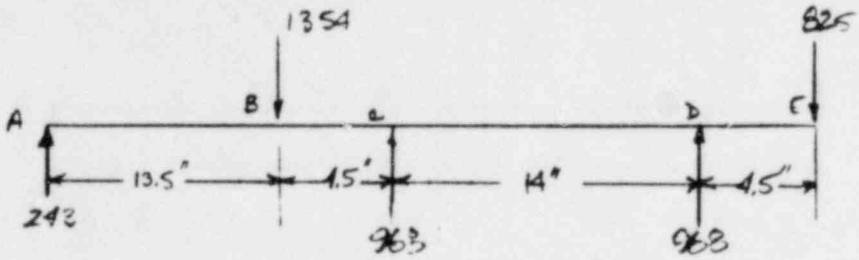


(a) In the plane containing F_a:

$$F_{c_1} = F_{D_1} = \frac{F_d}{2}\cos 12^\circ = \frac{1979.2}{2}\cos 12^\circ = 968 \text{ lbs.}$$

$$R_{B_1} = \frac{1}{23}\left[(243\times 36.5)+(968\times 18.5)+(968\times 4.5)\right] = 1354 \text{ lbs.}$$

$$R_{E_1} = 243 + (2\times 968) - 1354 = 825 \text{ lbs}$$



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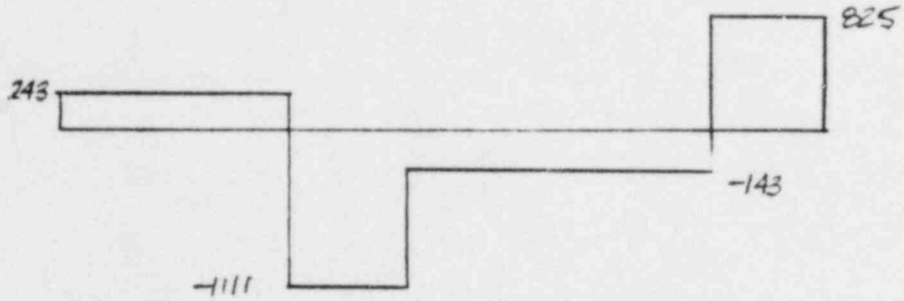

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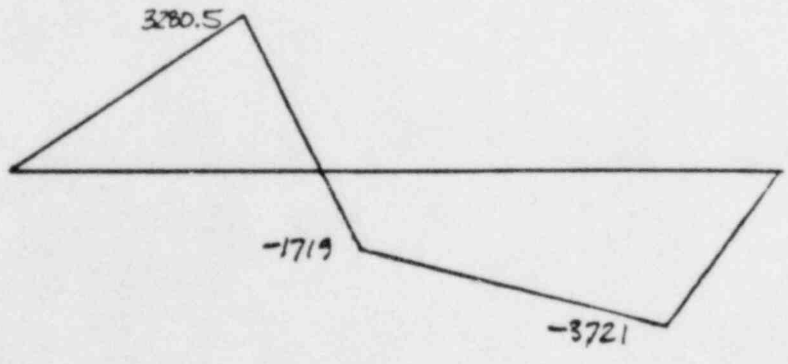
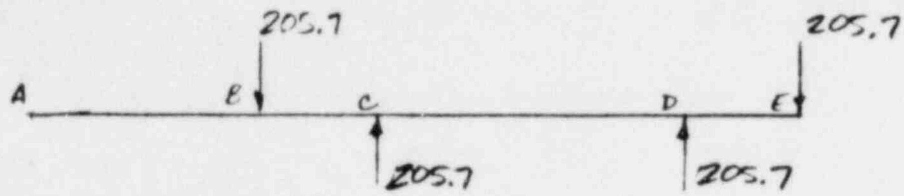


(b) In the plane perpendicular to F_a:

$F_{C_z} = F_{D_z} = \frac{F_d}{2} \sin 12^\circ = \frac{1979.2}{2} \sin 12^\circ = 205.7$ lbs.

$R_{B_z} = R_{E_z} = 205.7$ lbs.



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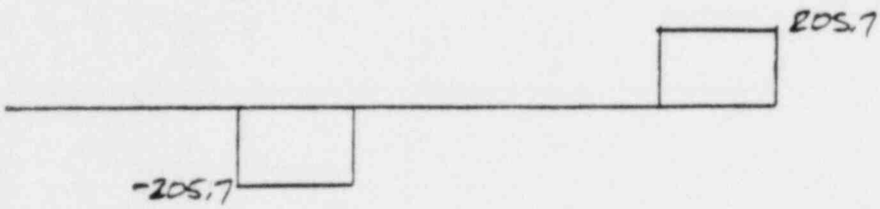

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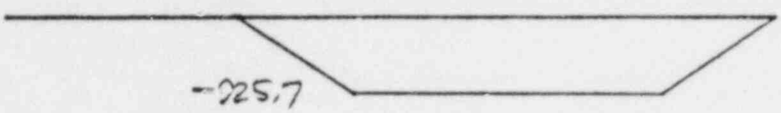


From (a) and (b):

bending moment at B $\quad M_B = 3280.5$ in-lbs.

shear force at B $\quad V_B = \sqrt{(1111)^2 + (205.7)^2} = 1129.9$ lbs. (to the right of B)

bending moment at C $\quad M_C = \sqrt{(1719)^2 + (925.7)^2} = 1952.4$ in-lbs.

shear force at C $\quad V_C = \sqrt{(1111)^2 + (205.7)^2} = 1129.9$ lbs (to the left of C)

bending moment at D $\quad M_D = \sqrt{(3721)^2 + (925.7)^2} = 3834.4$ in-lbs

shear force at D $\quad V_D = \sqrt{(825)^2 + (205.7)^2} = 850.$ lbs. (to the right of D)

STRESS CALCULATIONS

1. At B $M_B = 3280.5$ in-lbs $T = 2430$ in-lbs $V_B = 1129.9$ lbs.

Bending stress $\sigma = \frac{32M}{\pi D^3} = \frac{(32)(3280.5)}{\pi(2.25)^3} = 2933.5$ psi

Torsional stress $\tau' = \frac{16T}{\pi D^3} = \frac{(16)(2430)}{\pi(2.25)^3} = 1086.5$ psi

Shear in bending $\tau'' = \frac{4V}{3A} = \frac{(4)(1129.9)}{3(3.976)} = 378.9$ psi

Total shear $\tau = \tau' + \tau'' = 1086.5 + 378.9 = 1465.4$ psi

$S = \sqrt{(C_m \frac{\sigma}{2})^2 + (C_t \tau)^2}$ $C_m = C_t = 1.5$

$S = 1.5\sqrt{(\frac{2933.5}{2})^2 + (1465.4)^2}$

$S = \underline{3110 \text{ psi}}$

$FS = \frac{15,000}{3110} = \underline{\underline{4.8}}$

2. At C $M_c = 1952.4$ in-lbs $T = 2430$ in-lbs $V_c = 1129.9$ lbs

Bending stress $K_b \cdot \sigma_{nom} = K_b \cdot \frac{M}{\pi D^3/32 - dD^2/6} = \frac{1.86(1952.4)}{0.2745} = 13,229.4$ psi

Torsional stress $K_t \cdot \tau_{nom} = K_t \cdot \frac{T}{\pi D^3/16 - dD^2/6} = \frac{(1.38)(2430)}{1.3928} = 2407.7$ psi

Shear in bending $K_s \frac{4V}{3A} = (3.0) \cdot \frac{4(1129.9)}{3(3.976)} = 1136.7$ psi

Total shear $\tau = \tau' + \tau'' = 2407.7 + 1136.7 = 3544.4$ psi

$S = 1.5\sqrt{(\frac{13,229.4}{2})^2 + (3544.4)^2} = \underline{\underline{11,256.7 \text{ psi}}}$

$FS = \frac{15,000}{11,256.7} = \underline{\underline{1.33}}$

At D

$M_D = 3834.4$ in-lbs $T = 0$ in-lbs $V_D = 850$ lbs

Bending stress $\sigma = \frac{32M}{\pi D^3} = \frac{(32)(3834.4)}{\pi(2.25)^3} = 3428.9$ psi

Torsional stress $\tau' = \frac{16T}{\pi D^3} = 0$ psi

Shear in bending $\tau'' = \frac{4V}{3A} = \frac{(4)(850)}{3(3.976)} = 285.$ psi

Total shear $\tau = \tau' + \tau'' = 0 + 285 = 285.$ psi

$S = 1.5\sqrt{\left(\frac{3428.9}{2}\right)^2 + (285)^2} = \underline{2607 \text{ psi}}$

$FS = \frac{15{,}000}{2607} = \underline{\underline{5.75}}$

II. At 28°

• $T = 135$ ft-lbs. $\times 1.5 = 202.5$ ft-lbs.

• $F_a = \frac{(202.5)(12)}{10} = 243$ lbs Lever arm $L = 10''$

• $\frac{A_{eff}}{A_{nom}} = \left(\frac{C_{c_1} + C_{c_2}}{2}\right)(1 - \sin\alpha)$ (Ref. 10)

$= \left(\frac{0.817 + 0.569}{2}\right)(1 - \sin 28°)$

$= .368$

• $V_u = V_T \frac{A_{eff}}{A_{nom}} = (1160)(.368) = 427$ ft/sec

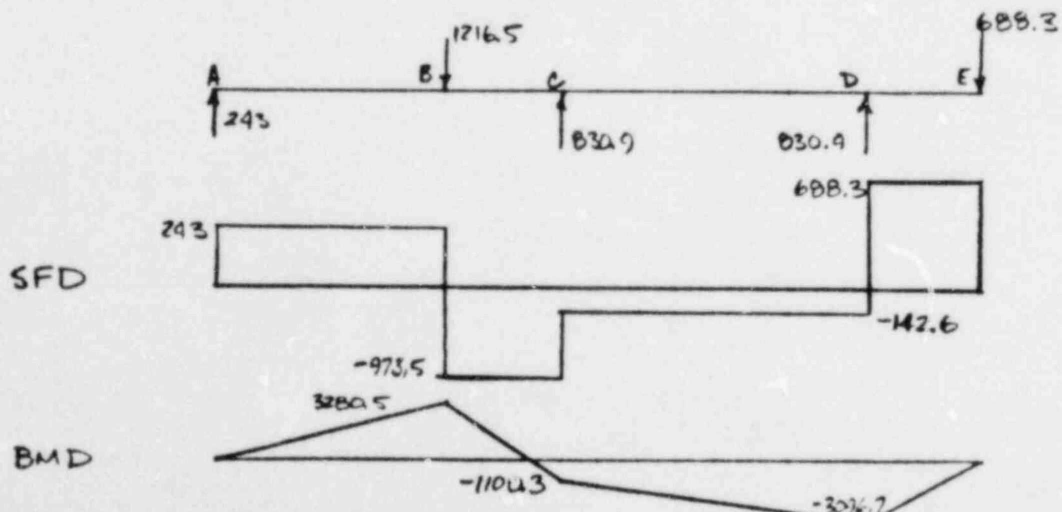
• $F_d = \frac{\gamma}{g} A V^2 \sin\alpha = \left(\frac{.354}{32.2}\right)(1.767)(427)^2 \sin 28° = 1662.8$ lbs.

(a) In the plane containing F_a:

$F_{C_1} = F_{D_1} = \frac{F_d}{2} \cos 2° = \frac{1662.8}{2} \cos 2° = 830.9$ lbs.

$R_{B_1} = \frac{1}{22}\left[(243 \times 36.5) + (830.9 \times 18.5) + (830.9 \times 4.5)\right] = 1216.5$ lbs.

$R_{E_1} = 243 + (2 \times 830.9) - 1216.5 = 688.3$ lbs.



(b) In the plane perpendicular to F_a:

$$F_{C_2} = F_{D_2} = \frac{F_d}{2} \sin 2^\circ = 29 \text{ lbs.}$$

Since this number is so small (3.5% of the number in (a)), this plane will be ignored.

STRESS CALCULATIONS

At B $M_B = 3280.5$ in-lbs $T = 2430$ in-lbs $V_B = -973.5$

Bending stress $\sigma = \frac{32M}{\pi D^3} = \frac{(32)(3280.5)}{\pi(2.25)^3} = 2933.5$ psi

Torsional stress $\tau' = \frac{16T}{\pi D^3} = \frac{16(2430)}{\pi(2.25)^3} = 1086.5$ psi

Shear in bending $\tau'' = \frac{4V}{3A} = \frac{(4)(973.5)}{3(3.976)} = 326.5$ psi

Total shear $\tau = \tau' + \tau'' = 1086.5 + 326.5 = 1413$ psi

$S = 1.5\sqrt{\left(\frac{2933.5}{2}\right)^2 + (1413)^2} = \underline{\underline{3055}}$ psi

$FS = \frac{15,000}{3,055} = \underline{\underline{4.91}}$

At C $M_C = 1100.3$ in-lbs. $T = 2430$ in-lbs. $V_C = 973.5$ lbs

Bending stress $K_b \cdot \sigma_{nom} = k_b \cdot \frac{M}{\pi D^3/32 - dD^2/6} = 1.86 \frac{1100.3}{0.2745} = 7455.6$ psi

Torsional stress $K_t \cdot \tau_{nom} = K_t \cdot \frac{T}{\pi D^3/16 - dD^2/6} = 1.38 \frac{2430}{1.3928} = 2407.7$ psi

Shear in bending $K_S \cdot \frac{4V}{3A} = (3.0)\frac{4(973.5)}{3(3.976)} = 979.4$ psi

Total shear $\tau = \tau' + \tau'' = 2407.7 + 979.4 = 3387.1$ psi

$S = 1.5\sqrt{\left(\frac{7455.6}{2}\right)^2 + (3387.1)^2} = \underline{\underline{7,555}}$ psi

$FS = \underline{\underline{1.99}}$

At D

$M_D = 3096.7$ in-lbs. $T = 0$ in-lbs. $V_D = 688.3$ lbs.

Bending stress $\sigma = \frac{32M}{\pi D^3} = \frac{(32)(3096.7)}{\pi(2.25)^3} = 2769.2$ psi

Torsional stress $\tau' = \frac{16T}{\pi D^3} = 0$ psi

Shear in bending $\tau'' = \frac{4V}{3A} = \frac{4(688.3)}{3(3.976)} = 230.8$ psi

Total shear $\tau = \tau' + \tau'' = 0 + 230.8 = 230.8$ psi

$S = 1.5\sqrt{\left(\frac{2769.2}{2}\right)^2 + (230.8)^2} = \underline{2105.6 \text{ psi}}$

$FS = \frac{15{,}000}{2105.6} = \underline{7.12}$

III. At 48°

- $T = 157.5 \text{ ft-lbs.} \times 1.5 = 236 \text{ ft-lbs.} = 2832 \text{ in-lbs}$

- $F_a = \frac{2832}{10} = 283.2 \text{ lbs}$

- $\frac{A_{eff}}{A_{nom}} = \left(\frac{C_{c_1} + C_{c_2}}{2}\right)(1 - \sin\alpha) = \left(\frac{0.776 + 0.570}{2}\right)(1 - \sin 48°)$ (Ref. 10)

  $= .173$

- $V_u = V_T \left(\frac{A_{eff}}{A_{nom}}\right) = (1160)(.173) = 200.7 \text{ ft/sec}$

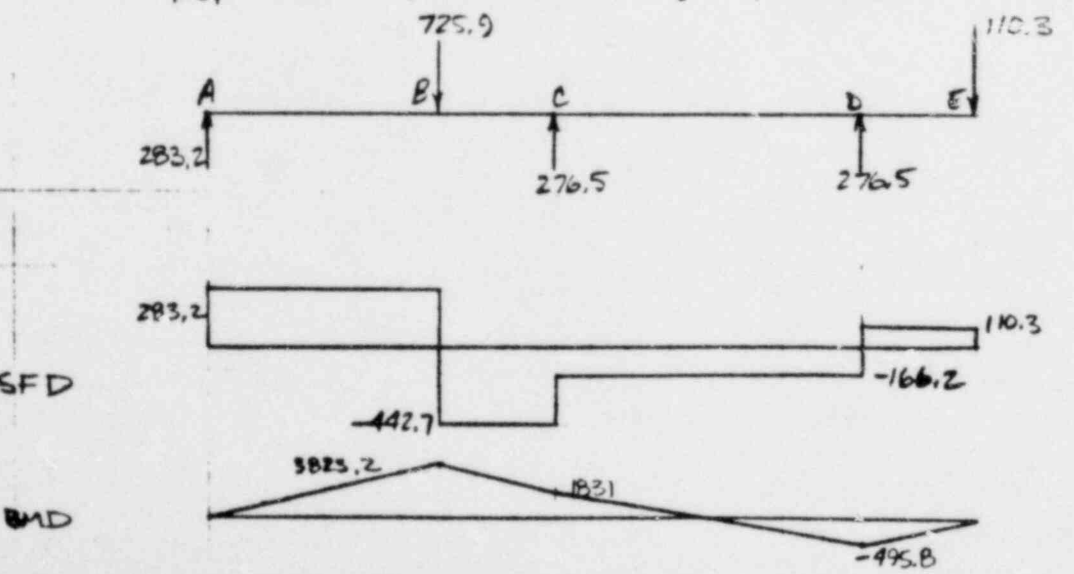
- $F_d = \frac{\gamma}{g} A V^2 \sin\alpha = \left(\frac{.354}{32.2}\right)(1.767)(200.7)^2 \sin(48°) = 581.5 \text{ lbs.}$

(a) In the plane containing F_a:

$F_{C_1} = F_{D_1} = \frac{F_d}{2} \cos 18° = \frac{581.5}{2} \cos 18° = 276.5 \text{ lbs.}$

$R_{B_1} = \frac{1}{23}[(283.2 \times 36.5) + (276.5 \times 18.5) + (276.5 \times 4.5)] = 725.9 \text{ lbs.}$

$R_{E_1} = 283.2 + (2 \times 276.5) - 725.9 = 110.3 \text{ lbs.}$



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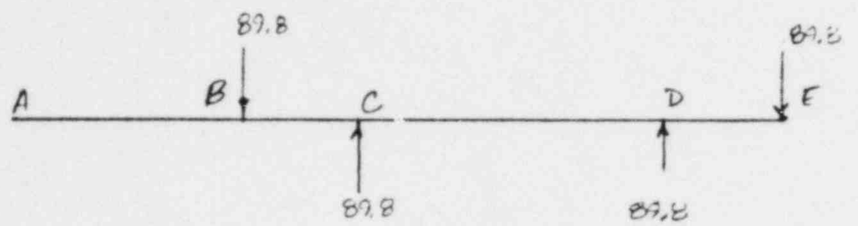
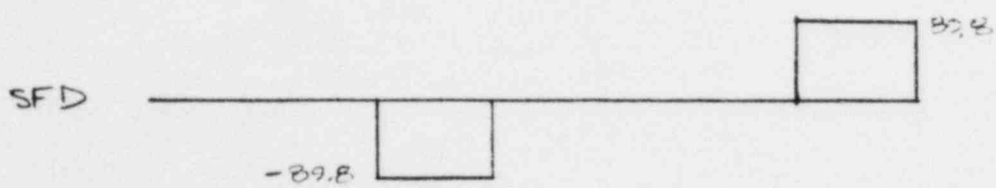
Client	Commonwealth Edison Company	

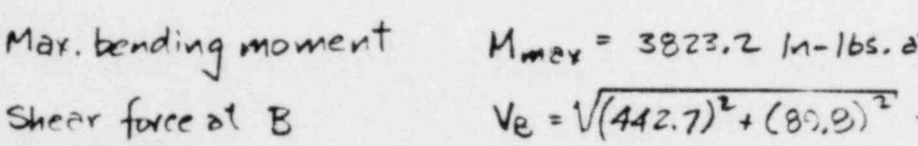
(b) In the plane perpendicular to Fa:

$F_{C_2} = F_{D_2} = \frac{F_d}{2} \sin 18^\circ = \left(\frac{581.5}{2}\right) \sin 18^\circ = 89.8$ lbs.

$R_{E_1} = R_{E_1} = 89.8$ lbs.







From (a) and (b):

Max. bending moment $M_{max} = 3823.2$ in-lbs. at E

Shear force at B $V_B = \sqrt{(442.7)^2 + (89.8)^2} = 451.7$ lbs.

Bending moment at C $M_C = \sqrt{(1831)^2 + (404.1)^2} = 1875.1$ in-lbs.

Shear force at C $V_C = \sqrt{(442.7)^2 + (89.8)^2} = 451.7$ lbs.

Bending moment at D $M_D = \sqrt{(495.8)^2 + (404.1)^2} = 639.6$ in-lbs.

Shear force at D $V_D = \sqrt{(166.2)^2 + (89.8)^2} = 188.9$ lbs.

STRESS CALCULATIONS

At B: $M_B = 3823.2$ in-lbs $T = 2832$ in-lbs $V_B = 451.7$ lbs.

Bending stress $\sigma = \frac{32M}{\pi D^3} = \frac{32(3823.2)}{\pi(2.25)^3} = 3418.8$ psi

Torsional stress $\tau' = \frac{16T}{\pi D^3} = \frac{16(2832)}{\pi(2.25)^3} = 1265.9$ psi

Shear in bending $\tau'' = \frac{4V}{3A} = \frac{(4)(451.7)}{3(3.976)} = 151.5$ psi

Total shear $\tau = \tau' + \tau'' = 1265.9 + 151.5 = 1417.4$ psi

$S = (1.5)\sqrt{\left(\frac{3418.8}{2}\right)^2 + (1417.4)^2} = \underline{\underline{3330.9 \text{ psi}}}$

$FS = \frac{15,000}{3330.9} = \underline{\underline{4.5}}$

At C: $M_C = 1875.1$ in-lbs. $T = 2832$ in-lbs. $V_C = 451.7$ lbs.

Bending stress $k_b \cdot \sigma_{nom} = k_b \cdot \frac{M}{\pi D^3/32 - dD^2/6} = (1.86)\frac{1875.1}{0.2745} = 12,705.6$ psi

Torsional stress $k_t \cdot \tau'_{nom} = k_t \cdot \frac{T}{\pi D^3/16 - dD^2/6} = 1.38 \frac{2832}{1.3925} = 2806$ psi

Shear in bending $k_s \cdot \tau''_{nom} = (3.0)\frac{(4)(451.7)}{(3)(3.976)} = 454.4$ psi

Total shear $\tau = \tau' + \tau'' = 2806 + 454.4 = 3260.4$ psi

$S = 1.5\sqrt{\left(\frac{12,705.6}{2}\right)^2 + (3260.4)^2} = \underline{\underline{10,710.9 \text{ psi}}}$

$FS = \frac{15000}{10,710.9} = \underline{\underline{1.40}}$

At D: $M_D = 639.6$ in-lbs. $T = 0$ in-lbs. $V_D = 188.9$ lbs.

Bending stress $\sigma = \frac{32M}{\pi D^3} = \frac{(32)(639.6)}{\pi(2.25)^3} = 572$ psi

Total shear $\tau = \tau' + \tau'' = 0 + \frac{(4)(188.9)}{3(3.976)} = 0 + 63.3 = 63.3$ psi

$S = (1.5)\sqrt{\left(\frac{572}{2}\right)^2 + (63.3)^2} = \underline{439.4 \text{ psi}}$

$FS = 15,000/439.4 = \underline{\underline{34.1}}$

IV. $\alpha = 8^\circ$

- TORQUE = 70 ft-lbs (REF. 9)
- FOR CONSERVATISM, MULTIPLY BY 1.5

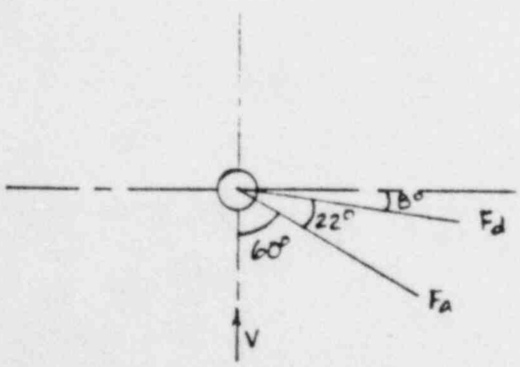
$$T_h = (1.5)(70) = 105 \text{ ft-lbs} = 1260 \text{ in-lbs.}$$

- $F_a = \frac{T}{L} = \frac{1260}{10} = 126 \text{ lbs.}$
- ASSUME SONIC VELOCITY AT VALVE THROAT (THIS IS CONSERVATIVE SINCE $\frac{P}{P_t} = 0.735 > 0.525$)
- $\frac{A_{eff}}{A_{nom}} = \left(\frac{C_{c_1} + C_{c_2}}{2}\right)(1 - \sin\alpha) = \left(\frac{0.904 + 0.578}{2}\right)(1 - \sin 8^\circ)$ (REF. 10)

$$= 0.638$$

- $V_u = V_T \left(\frac{A_{eff}}{A_{nom}}\right) = (1160)(0.638) = 740 \text{ ft/sec}$
- $F_d = \frac{\gamma}{g} A V^2 \sin\alpha = \left(\frac{35.4}{32.2}\right)(1.767)(740)^2 \sin 8^\circ$

$$= 1480.5 \text{ lbs.}$$



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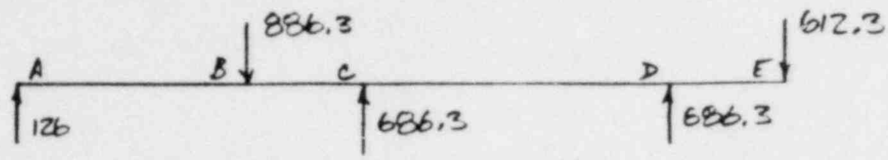
San Jose, California

Project: Dresden Nuclear Power Station — File No. 64.801.0006

Owner: Commonwealth Edison Company

Client: Commonwealth Edison Company

(a) IN THE PLANE OF F_a:

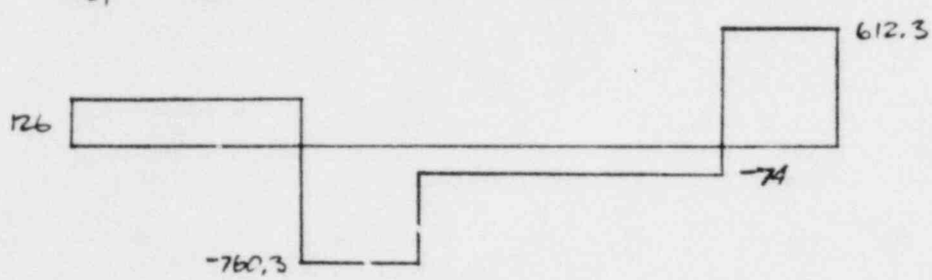


$$R_{C_1} = R_{D_1} = \frac{F_d}{2}\cos 22^\circ = \frac{1480.5}{2}\cos 22^\circ = 686.3 \text{ lbs.}$$

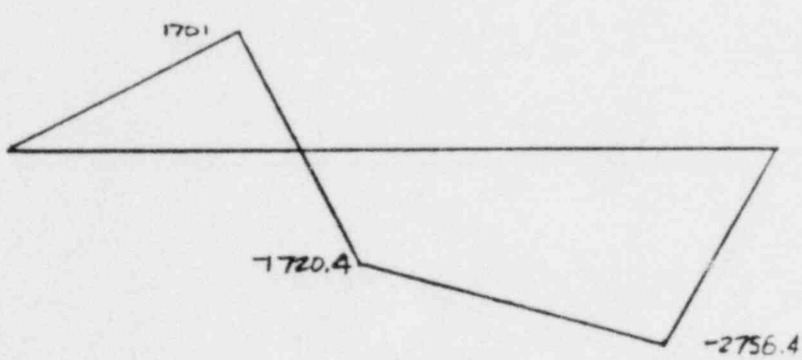
$$R_{B_1} = \frac{1}{23}\left[(126 \times 36.5) + (686.3 \times 18.5) + (686.3 \times 4.5)\right] = 886.3 \text{ lbs.}$$

$$R_{E_1} = \left[126 + (2 \times 686.3) - 886.3\right] = 612.3 \text{ lbs.}$$

SFD



BMD



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San Jose, California

Project Dresden Nuclear Power Station File No. 64.801.0006

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CONCLUSION:

It seems the stresses on the affected parts such as shaft, square key, tapered pin and actuator arm are within allowable stresses according to the average strength of industrial materials.

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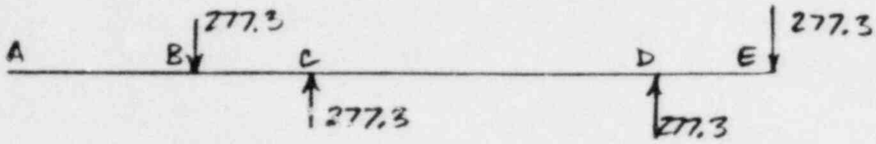
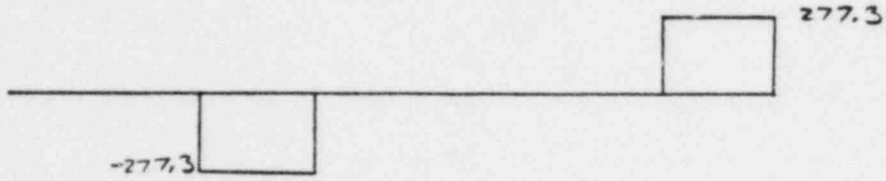
(b) IN THE PLANE PERPENDICULAR TO F_a:



$$R_{C_2} = R_{D_2} = \frac{F_d}{2}\sin 22^\circ = \frac{1480.5}{2}\sin 22^\circ = 277.3 \text{ lbs.}$$

$$R_{B_2} = R_{E_2} = 277.3 \text{ lbs.}$$

SFD



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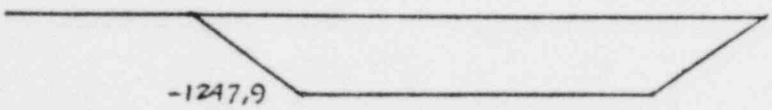

(c) STRESS CALCULATIONS

BENDING MOMENT AT C $\quad M_c = \sqrt{(1720.4)^2 + (1247.9)^2} = 2125.3$ in-lbs.

SHEAR FORCE AT C $\quad V_C = \sqrt{(760.3)^2 + (277.3)^2} = 809.3$ lbs.

AT C:

BENDING STRESS $\sigma = K_b \cdot \frac{M}{\pi D^3/32 - dD^2/6} = (1.86)\frac{2125.3}{.2745} = 14{,}401$ psi

TORSIONAL STRESS $\tau' = K_t \cdot \frac{T}{\pi D^3/16 - dD^2/6} = (1.38)\frac{1260}{1.3928} = 1248.4$ psi

SHEAR IN BENDING $\tau'' = K_s \cdot \frac{4V}{3A} = (3.0)\frac{(4)(809.3)}{(3)(3.976)} = 814.2$ psi

TOTAL SHEAR $\tau = \tau' + \tau'' = 1248.4 + 814.2 = 2062.6$ psi

$$S = (1.5)\sqrt{\left(\frac{14{,}401}{2}\right)^2 + (2062.6)^2} = \underline{11{,}235 \text{ psi}}$$

$$FS = \frac{15{,}000}{11{,}235} = \underline{1.34}$$