ATTACHMENT E

REPORTED TURBINE COMPONENT DAMAGE

AT BVPS UNIT 1 RELATED TO CONDUCT OF

OPERATIONAL SURVEILLANCE TEST (OST) 1.26.1

TS Test

DUQUESNE LIGHT COMPANY
Beaver Valley Power Station

REPORTED TURBINE COMPONENT DAMAGE AT BVPS UNIT 1 RELATED TO CONDUCT OF OPERATIONAL SURVEILLANCE TEST (OST) 1.26.1

A review of unusual steam flow distribution patterns, steam pressure changes and load disturbances occurring during OST 1.26.1, leading to turbine component damage.

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INTRODUCTION

Operational Surveillance Test (OST) 1.26.1 is a periodic test conducted to insure that the main turbine throttle, governor, reheat stop, and intercept valves are responding smoothly and freely to imposed test signals that causes a specific valve to move through its full travel from full open to full closed. These valves are vital devices for the control of steam flow, either for load control or for limiting turbine rotor overspeed. The periodic testing, inspection, maintenance and repair of these devices is of major operational concern.

OST 1.26.1 is not a passive test. It imposes unusual stresses and forces on certain turbine components and has caused equipment damage to occur that was not clearly recognized or understood at the time the testing program was initiated. In this report, each valve test (throttle, governor, reheat stop and intercept) is reviewed separately and in the sequence required by OST 1.26.1. Since steam flows are being interrupted with the exercising of these valves, the problem is usually one of determining how the steam flows are redistributed and what pressure drop and velocity changes are occurring and what forces are resulting. Historically, the equipment damage has been disguised mainly because damage is the cumulative effect of repeated testing and usually not the result of single events.

Modifications have been incorporated, or will be incorporated, that will better resist some of the potential damage, but it should be recognized that OST 1.26.1 imposes a severe service requirement on various turbine components and that methods to increase the life expectancy of certain main steam turbine components be investigated.

The sequence of valve testing and the order in which they are reviewed in this report are as follows:

PART A Governor Valve Test

PART B Throttle Valve Test

PART C Reheat Stop and Intercept Valve Test

Of the above three (3) specific valve tests, PART C (Reheat and Intercept) imposes the severest challenge to the integrity of the Main Steam Turbine System. Approximately $400,000 was expended during the most recent refueling outage (December 1987 - March 1988) to repair the cumulative damage to the four (4) moisture separator reheaters. These expenses were directly attributed to valve testing. The next most severe challenge comes from PART A (Governor). Expenditures in this instance are not specifically budgeted to Planned Maintenance activities. However, usual restoration costs include rebabbiting two (2) HP turbine bearings and HP rotor steam seal strip restoration. PART B (Throttle) is the least challenging surveillance test and no restoration costs have been associated with this test.

PART A - GOVERNOR VALVE TESTING

The function of the turbine governor valves are to accurately control the steam flow rates directed to the inlet nozzle chambers of the main turbine. The governor valves perform that function in a manner so as to match the inlet steam flow rate to the electrical load imposed on the main generator. An additional function of the governor valves are to prevent the frequency (speed) of the turbine generator from changing beyond a specified value. In the event of turbine generator overspeed, the governor valves are designed to rapidly close to limit the rotor overspeed. Thus, the governor valves are an integral part of the turbine overspeed protection system.

Each governor valve controls the steam flow to one (1) of four (4) turbine nozzle chamber. The governor valves are located external to the main turbine HP cylinder in two (2) steam chests located on opposite sides of the HP turbine. There are four (4) governor valves - two (2) in each steam chest. The nozzle chambers are positioned internally in the HP cylinder and the four (4) nozzle chambers each span an arc of ninety degrees (90°). Figure A1 depicts this arrangement schematically.

The normal governor valve opening sequence and valve lifts to produce 100% power are tabulated below:

1st - Valves #2 and #3	100% Lift (together)
2nd - Valve #1	38% Lift
3rd - Valve #4	0%

The governor valve nomenclature is identical to that appearing in Figure A1.

Surveillance testing requires each governor valve be observed during its full travel to open and close smoothly without sticking. The testing sequence for the governor valves are tabulated below:

1st - Valve #3	Close - Open
2nd - Valve #2	Close - Open
3rd - Valve #1	Close - Open

Governor valve #4 is not directly stroked but will move to the open position (but not full open) as a consequence of either Valve #3 closing or Valve #2 closing. With electrical output constant, as Valve #3 (or #2) reopen, Valve #4 will reclose. Consequently, all governor valves are stroked. The surveillance test is conducted with the load reduced 5%. It is important to note that load reductions beyond 10% will introduce a "double shock" to the control stage blading.* The region where governor valve testing can be conducted safely is limited to a range of 90% to 100% of full load. (This is at variance to the value of 84% to 100% of full load that appears in the Turbine Vendor's Instruction Book.) (See Figure A2).

On those occasions when the main HP turbine was disassembled for a planned inspection and/or maintenance, the damage reported has been similar - HP Bearings No. 1 and No. 2 wiped and light to medium rotor seal strip rubs on all rows wherever spring backed seals are applied. This situation has led to rebabbiting the bearings and restoring or replacing the damaged seals. These are all rather costly and time consuming maintenance functions.

The cause of this component damage observed was traceable directly to governor valve surveillance testing, coupled with the unanticipated dynamic response of the HP rotor to the steam forces acting on the turbine blading in the control (first) stage of the turbine during the specific governor valve testing interval.

Appendix A provides a method for calculating the resultant forces acting on the HP rotors. These steam forces are relatively very large compared to the HP rotor weight. For example, the static weight of the HP rotor between Bearings No. 1 and No. 2 equals 108,000 lbs., while a situation may exist (~670 MW) when Governor Valves No. 2 and No. 3 are completely open (Valves No. 1 and No. 4 being closed) where the resultant steam forces developed (106,000 lbs.) almost equals the static weight of the rotor, but in the upward direction. Whenever that or similar situations develop in the HP turbine, the rotor bearings become very lightly loaded. Figure A3 represents the actual initial condition (95% load) required to perform the governor valve surveillance test. Under these conditions, it can be observed that not only are Bearings No. 1 and No. 2 very lightly loaded (less than 30 psi), but the resultant bearing contact angle is about 40° from the 6 o'clock position, suggesting that the bearings are operating in a region of dynamic instability.

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* A "double shock" occurs whenever in one revolution of the rotor, a turbine blade passes through an active (steam flowing) nozzle arc and then enters an inactive (zero steam flowing) nozzle arc region then re-enters an active region and finally passes through a second inactive arc. The double impulse to the blade may reinforce the vibratory stresses in the blade to the point that failure may occur. Since Unit 1 has always performed the surveillance test at 95% load, it has not experienced a control stage blade failure attributable to this test.

The large rotor bearing displacements occurring at Bearing No. 1 were first observed by the plant operators while performing the governor valve surveillance tests per OST 1.26.1. The typical bearing vibration patterns observed are shown on portions of the Control Room strip chart recorder, reproduced in Figures A4, A4.1 and A4.2. Additional engineering analysis, supported by supplementary field data, clearly confirmed that movement was occurring within the No. 1 bearing. Figure 5A records the relative position of Bearing No. 1 center-line during the governor valve test. It is noteworthy to observe that the greatest bearing motion occurs during the governor Valve No. 3 testing interval. This is in agreement with calculated predictions.

Additional confirming evidence of bearing stability problems was supplied by the turbine vendor who provided the following reply to an Engineering Department inquiry:

> "BB296 Turbines were originally designed with sleeve bearings. Valve test studies indicated that light bearing loads would exist under certain conditions. Stability studies, however, indicated that the rotors would be stable with the light bearing loads.
>
> "Several years after the first BB296 Turbines were built, additional study indicated that the bearings on newer units would become totally unloaded during valve test. A decision was made to install tilting pad bearings on all future BB296 Turbines. When this decision was made, existing units had not exhibited any vibration difficulty during valve test and they were not retrofitted with tilting pad bearings."

As noted above, in order to eliminate the HP rotor dynamic instability and to decrease the HP rotor displacements during the governor valve testing internals, tilting pad type bearings should replace the original sleeve type bearings. Tilting pad bearings will permit high bearing contact angles to be accepted without introducing "oil whip", "rotor whirl", or other similar instabilities. Observations at BVPS Unit 2 during valve testing appears to confirm this fact.

Installation of tilt pad bearings is estimated to require a one (1) year lead time and expenditures of about $300,000 for bearings and associated hardware alone. The earliest installation date would coincide with R8 (late 1989 or early 1990) and a schedule HP turbine maintenance outage. Labor cost have not been estimated.

The periodic inspection of governor valve assemblies has, from time-to-time, disclosed cracks developing in a part of the valve assembly (mufflers). These failures are related to flow induced vibrations that may occur whenever a valve operates close to its seat. The failures are <u>not</u> related to governor valve testing. Modifying the valve lift and overlap characteristics has essentially controlled the incidence of muffler cracking.

PART B - THROTTLE VALVE TESTING

The throttle valves function to shut off the supply of main steam to the turbine. Normally, the throttle valves are either fully closed or fully opened, leaving to the governor valves the function of actually metering the steam flow to the HP turbine during normal operation. The one occasion the throttle valves control steam flow is during a unit start up. This is accomplished by means of limited capacity pilot valves that are an integral part of the throttle valve (valve within a valve) assembly. During a turbine rotor speed excursion, the throttle valve will be actuated to trip (close) at a predetermined rotor speed. Consequently, the throttle valves provide a redundant or backup overspeed protection system.

As previously indicated in Figure A1, there are four (4) throttle valves each attached to opposite ends of either one of two (2) steam chests. The four (4) throttle valves normally operate in unison. Valve testing requires that the throttle valves be stroked to close (and open) sequentially in their normal numbered sequence No. 1, No. 2, No. 3, and No. 4.

No observable damage appears to be attributable to the conduct of this test. During the throttle valve testing interval, the largest throttle valve steam flows appear in the left side steam chest. First, when the No. 1 TV is closed and later when the No. 3 TV is closed. When the No. 1 TV is closed, the maximum flows occurs at No. 3 TV. Likewise, when the No. 3 TV is closed, the maximum flows occurs at No. 1 TV. This can be clearly seen in the tabulations appearing in Table B1 and Table B2. The increase in flows above the normal steam flow is 1.60x or a 2.6x increase in pressure drop through the throttle valve. Assuming a 2% throttle valve pressure drop, the resulting pressure drop increases to 5.1%. This is not considered a significant increase and may explain the lack of visible damage to any of the throttle valves. Replacement of bushings, gaskets and locking pins is considered normal maintenance and not attributable to valve testing.

A summary of the steam flow relationship at the throttle valves, during valve testing appears below:

Total Steam Flow to TV (LB./HR.)

	No. 1 TV Closed	No. 3 TV Closed
No. 1 TV	0	6.4×10^6 (Max.)
No. 3 TV	6.4×10^6 (Max.)	0
No. 2 TV	2.10×10^6	2.0×10^6
No. 4 TV	2.0×10^6	2.0×10^6

Maximum throttle valve flows do not develop when either No. 2 or No. 4 throttle valves are closed. In addition, the power generated does not vary significantly during valve testing as indicated by the MW recorder. As noted earlier, PART B - throttle valve testing is the least challenging surveillance test and no restoration costs are associated with this test.

The supplier of the turbine throttle valves has notified users that if the throttle valves are closed for a period of 15 minutes or longer, then reopened, there exists an interval of 15 minutes when valve sticking is likely to occur, resulting in possible failure of the valve to close. The vendor recommends that if any throttle valve is held closed for more than ten (10) minutes, upon reopening, it should be stroked by means of the test button.

Addressing the above concerns, the valve surveillance testing is sequenced in a manner such that throttle valve testing always follows governor valve testing. This assures that the governor valves tested are in working order and available for overspeed protection in the event of difficulties with the throttle valves.

Until the root cause of valve sticking is identified and corrected, the additional testing should not create any additional threat to the mechanical reliability of the throttle valves.

PART C - REHEAT STOP AND INTERCEPT VALVE TESTING

The primary function of the reheat stop and intercept valves are to block the flow of steam into the low pressure turbines during either a rotor speed excursion or load rejection (turbine trip) transient. A significant volume of high pressure steam accumulates in, (1) the HP turbine exhaust piping, (2) the moisture separator-reheater tanks and, (3) the turbine cross-over piping during normal operation. The energy content of this stored steam is such that if allowed, during a load transient, to flow uncontrolled into the low pressure turbines, would increase the potential overspeed of the turbine-generator rotor system to undesirable limits. Like the turbine governor valves, intercept valves are sensitive to speed changes and can, under certain circumstances, modulate the steam flows to the low pressure turbines. The reheat stop valves are a back-up or redundant overspeed protection system, similar to the main turbine throttle valves. They are, therefore, a vital part of the total overspeed protection system.

One (1) reheat stop and intercept valve combination is located in each of the four (4) cross-over pipes as near to the low pressure turbine inlets as practical. Figure C1 shows this arrangement schematically.

Surveillance testing of the reheat stop and intercept valves follows the following sequence:

1. LP-1 (Left Side), TEST 1IRL Close - Open
2. LP-2 (Left Side), TEST 2IRL Close - Open
3. LP-1 (Right Side), TEST 1IRR Close - Open
4. LP-2 (Right Side), TEST 2IRR Close - Open

This surveillance testing results in a most severe challenge to the mechanical integrity of the main steam turbine and a concomitant detriment to its thermodynamic efficiency. The accumulated component damage associated with valve testing appears to be centered primarily in the internals of the moisture separator-reheaters. An extensive review and discussion of the observed damage and the necessary repairs appears in a companion report.* A brief summary describing the nature of the damage is extracted from the report:

* Deahna, S. T., "Beaver Valley Power Station No. 1 and No. 2 Moisture Separator-Reheater Status Report", NED, August 23, 1988.

"During the Unit 1 sixth refueling outage, two (2) types of damage were discovered in the MSR's. The first type of damage was the failure of the reheat steam inlet hemi-head partition plate. The hemi-head partition plate isolates the top half of the U-tube bundle from the bottom half. Its integrity is required to ensure proper reheat steam flow. The second type of damage is the failure of shellside closure plates. The integrity of the closure plates is required in order to direct the turbine cycle steam through the cheveron type moisture, separating sections of the MSR without leaking around the reheater."

As a consequence of the shellside closure plate failures, debris was carried into the crossover pipe, then passed through the intercept and stop valves, and rested in the turning vanes at the inlet to the low pressure turbine. Fortunately, these fragments did not appear to have damaged any low pressure turbine blades.

To better understand the actual flow distribution patterns and pressure drops being encountered during this specific valve testing, data was recorded during a required OST 1.26.1, when the intercept and stop valves were exercised. The results (pressures and flows) appear in Figure C1.1, C1.2, C1.3 and C1.4.

An examination of the above test results clearly show that during the intercept and reheat stop valve testing interval, the turbine crossover (X-O) pipe flows patterns undergo significant changes. The tabulation below indicates the maximum individual relative pipe flows are achieved whenever one (1) set of IV/RSV are closed. The maximum X-O flow increases are about X1.76 normal and appear in the opposite side X-O pipe feeding the same low pressure turbine. For example, when the intercept and reheat stop valves in X-O pipe -1A are closed, the maximum flow appears in the opposite X-O pipe -1B.

This increases in steam flows, steam velocities and pressure drops during this test are a direct precursor of the premature failures in the MSR shellside closure plates. The following tabulation demonstrates the fact that there are large forces to absorb during valve testing:

TEST	X-O PIPE	X-O FLOW	P (TEST)	P (NORMAL)
1IRL	(-1B)	X 1.69	70.1	29.7
2IRL	(-1D)	X 1.71	74.0	34.0
1IRR	(-1A)	X 1.87	54.4	29.4
2IRR	(-1C)	X 1.75	72.5	33.7

The failure of the reheat steam inlet hemi-head partition plate is also related to the occurrence of excessive pressure differences, this time across the divider plate. A response from the MSR vendor states in part the following:

> "...the cause of distress for both the hemi-head partition plate and closure plates is the abnormally high pressure drops developed during interceptor valve testing..."

The normal pressure drop across the divider plate was estimated by the vendor to be 40 PSID. MSR tests at BVPS Unit 1 on June 20, 1988, suggest that at 100% load a pressure drop of 63 PSID develops. The vendor further indicated that during intercept and reheat valve testing, the pressure drop exceeds 106 PSI. Comparable test data is not available from Unit 1. However, it is anticipated that a pressure drop of 106 PSI will be exceeded. The extensive repairs and modifications to the MSR's, to reduce the incidence of failures caused by these pressure drops, are detailed in the companion report previously cited.

The reach of effects from intercept and reheat stop valve testing is very wide. For example, the generating capability of the unit fell from about 780 MW to 720 MW (See Figure C2) as each valve was exercised. At the same time, total steam flows to one (1) LP turbine increased by 10-11%. By reference to Figure C1.1, it was observed that during TEST 1IRL, the steam flow to LP-2 increased by a factor of X1.104. As a result of this increase, the LP last row blading in LP-2 experienced a load increase of 5% above the normal levels. Note this increase is after the load had been reduced 5% in order to conduct OST 1.26.1. Continuing with TEST 1IRL, the pressure in the -1A cross-under and cross-over piping (no flow/valves closed) increased to 232.2 PSIG or within about 18 PSIG of challenging the MSR relief valve. Finally, the pressure distribution at the HP turbine exhaust region creates a pressure and thrust unbalance that causes the rotor position to move in one direction and then in the other direction.

The above transients are some of the observed effects from valve testing that may challenge the mechanical integrity of the main turbine.

The mechanism of the progressive tube failures (leaks) experienced in the reheat section of the MSRs is not clearly understood. However, it is believed that OST 1.26.1 is a contributor to the tube failures. When valves are exercised (closed), severe cycle steam flow distortions occur which, when coupled to the low heat transfer rates from the reheater, may be creating unusual strains in the tube bundles.

The vendor observed that after surveying the repaired MSR tanks, estimated that replacement reheater tube bundles may be required in about three (3) years. The total replacement cost would be about $4,000,000.

CONCLUSIONS AND RECOMMENDATIONS

THROTTLE VALVE TESTING

Little, if any, direct turbine component damage is attributed to throttle valve testing. Valve sticking has been reported as a problem on some units (not at BVPS-1). By following the vendor's recommendations during valve movements, incidents of valve sticking should be eliminated or avoided.

GOVERNOR VALVE TESTING

The dynamic instability of the HP turbine rotor leads to accumulated HP bearing damage and HP turbine seal rubs during the governor valve testing time interval. Both the vendor and the experience at BVPS-2 (with tilt-pad bearings) support the conclusion the replacement of the sleeve type bearings at BVPS-1 with tilt-pad bearings will improve the stability sufficiently to mitigate the damage occurring. This modification will be expensive and require dismantling the HP turbine.

INTERCEPT AND REHEAT VALVE TESTING

Intercept and reheat stop valve testing has resulted in extensive damage to the MSR tank internals, which poses a direct threat to the main turbine from debris leaving the MSR tanks. Efforts have been made to correct design deficiencies in the MSR to mitigate the damage. During valve testing, the relative flows to the LP turbine changes drastically, depending on which set of valves are closed. Other than velocity and pressure changes observed at the inlet to the MSRs and LP turbine, the following events were also observed to occur during the testing interval:

Sudden loss of 60-70 MW of electrical load

Swings in rotor position and thrust bearing loads

Overloading of the last row turbine blades

Heat load unbalance between Condenser A and B of about X 1.25

Feedwater heater level alarms at 2nd, 4th, and 5th heaters

Situation created for high erosion rates in cross-over and cross-under pipes

The propensity for serious turbine component damage is clearly present during the conduct of this test. Recommendations are to avoid, defer or decrease the frequency of this testing until the results of this testing are better understood, or an alternative to testing is developed.

REFERENCES

1. Operational Surveillance Test (OST) 1.26.1

2. Technical Specification 3.3.4 (BVPS Unit 2)

3. Analysis of HP Turbine Valve Cycling, No. 1 Main Turbine Generator, 6FEB87, Bentley Nevada, 03/03/87

4. Field Report, Unit Maintenance Contract, PGSD Westinghouse Electric Corporation, 06/20/88

5. Deahna, S. T., Beaver Valley Power Station No. 1 and No. 2 Moisture Separator-Reheater Status Report, NED, 08/23/88

6. Letter, Westinghouse Electric Corporation, W. L. Shoff, 07/14/88

7. Customer Advisory Letter, Westinghouse Electric Corporation, 09/30/87

FIGURE NO. A1

STEAM CHEST - NOZZLE BLOCK - VALVE ARRANGEMENT

(VIEW TOWARD GENERATOR FROM GOVERNOR END)

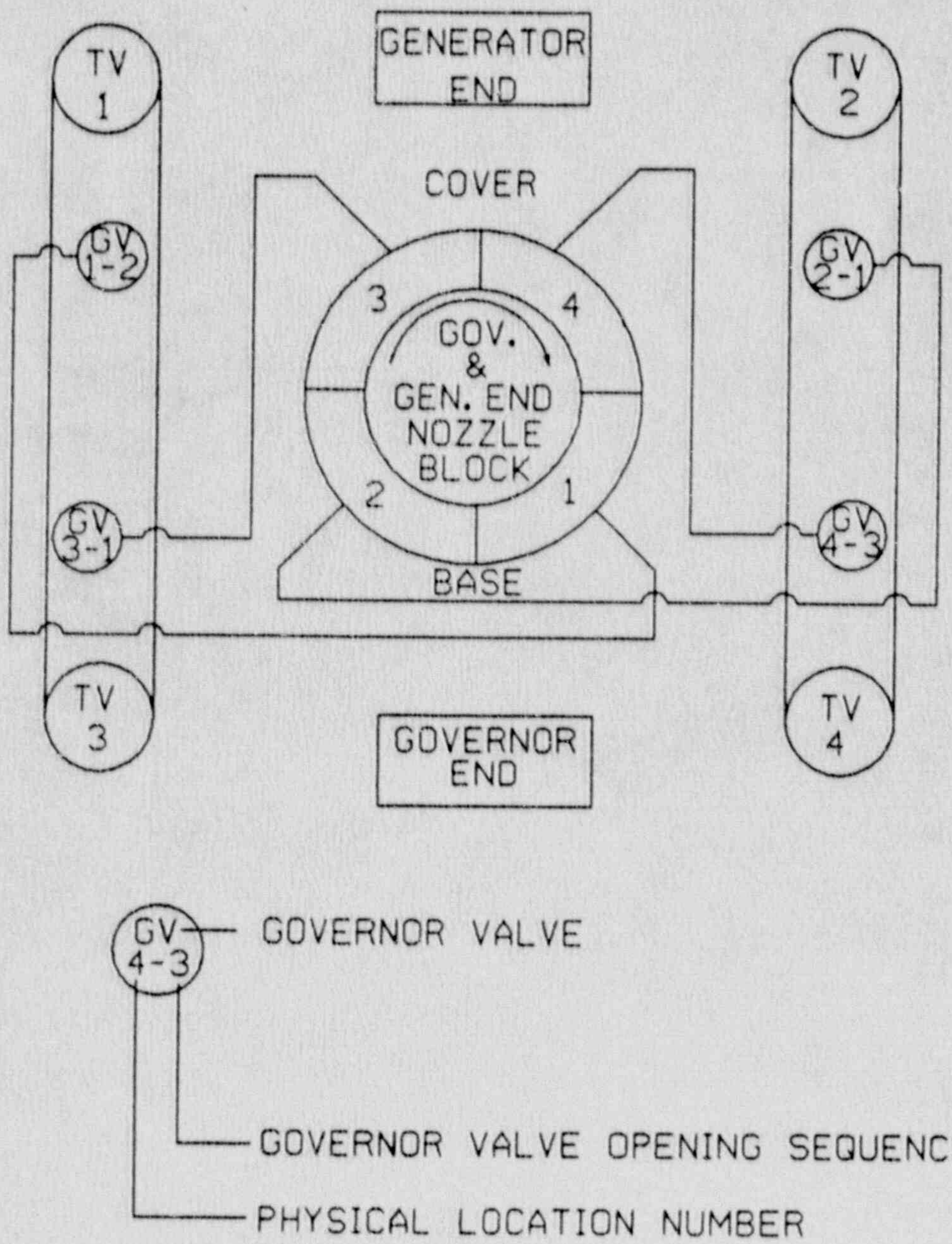
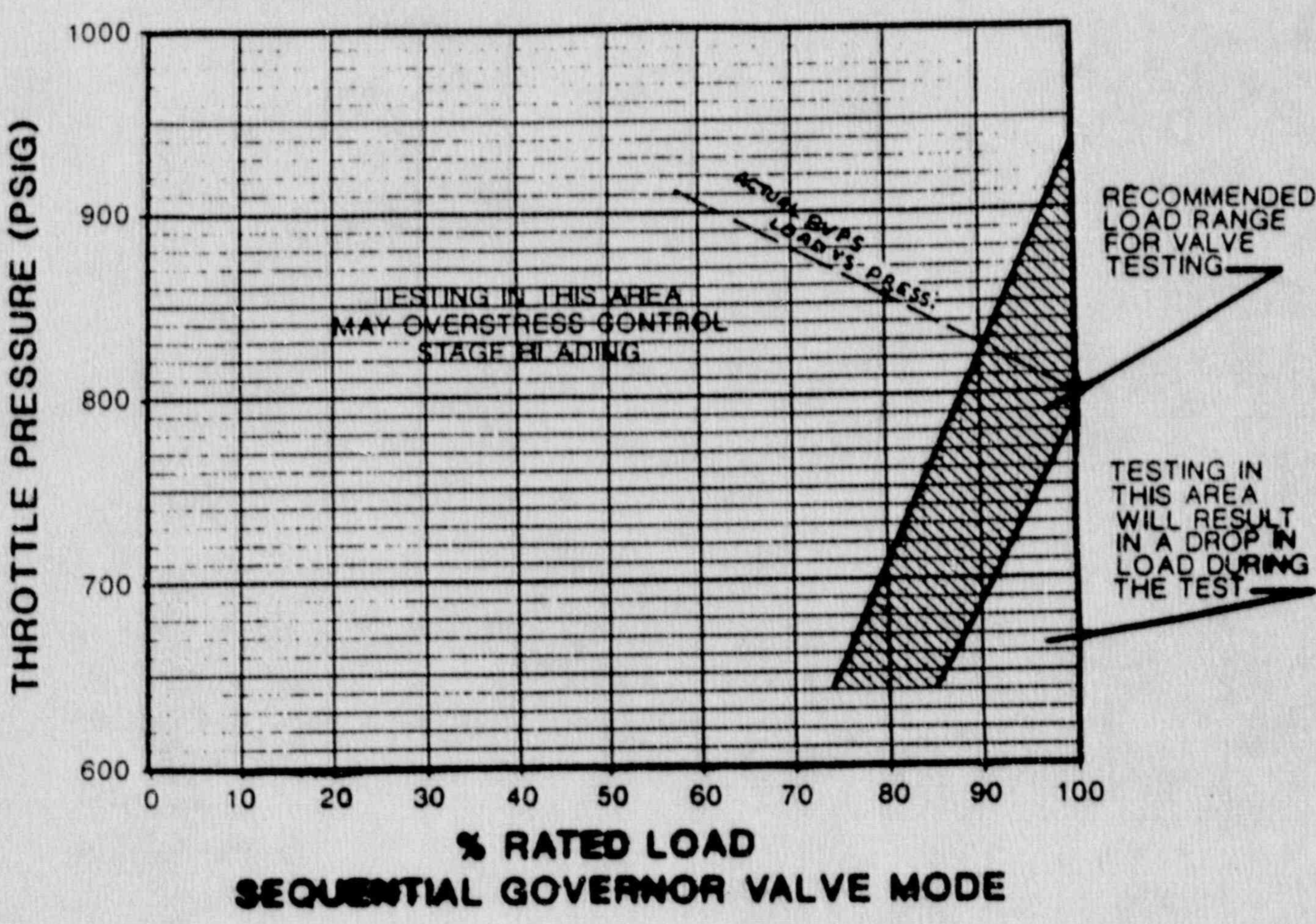


FIGURE NO. A2

RECOMMENDED LOAD RANGE FOR TESTING MAIN STEAM INLET VALVES



Reference: OST 1.26.1

FIGURE NO. 43

GOV VALVE POSITIONS AT START OF OST 1.26.1 (95% LOAD)

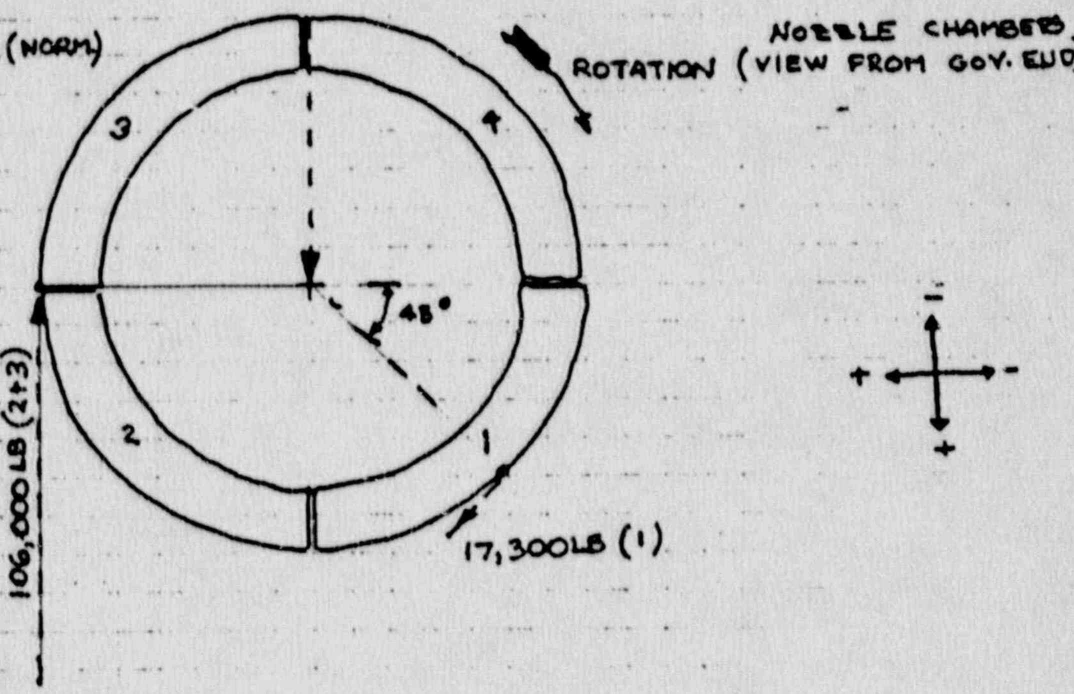
NO. 1 - PART OPEN

NO. 2 - FULL OPEN

NO. 3 - FULL OPEN

NO. 4 - CLOSED (NORM)

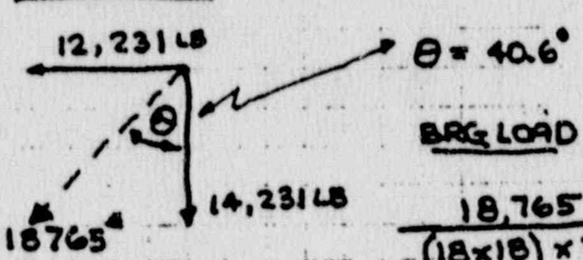
108,000 LB (WT. OF ROTOR BETWEEN BRGS 1 & 2)
ROTOR OVERHANG NOT CONSIDERED



RESULTANTS

VERTICAL	HORIZONTAL
(2+3) -106,000 LB	(1) +12,231 LB
ROTOR +108,000 LB	
(1) + 12,231 LB	
+ 14,231 LB	

RESOLUTION



BRG LOAD (18"x18") =

$$\frac{18,765}{(18 \times 18) \times 2} = \underline{28.9\ LB/IN^2}$$

OBSERVATION: BRGS (1 & 2) BECOME VERY LIGHTLY LOADED - PERHAPS UNSTABLE

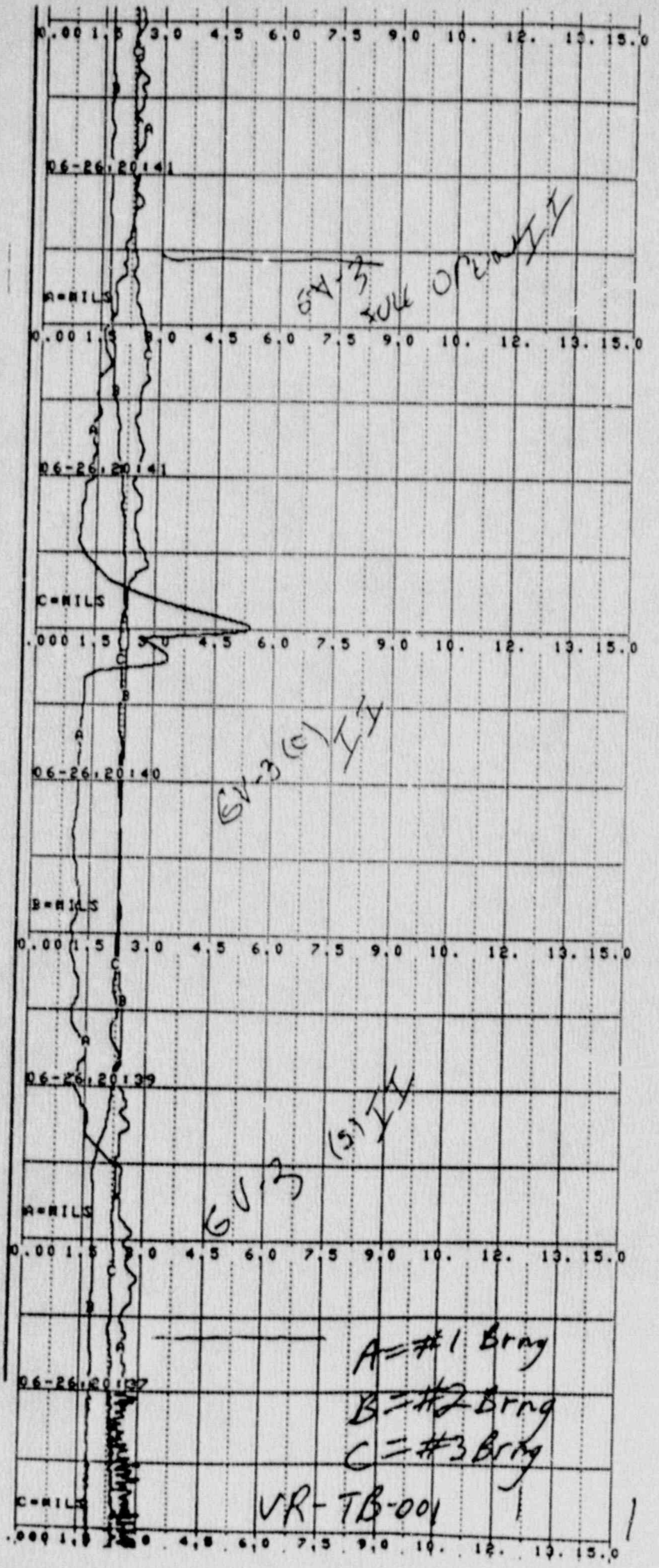


FIGURE NO. A4

TYPICAL RESPONSE OF GV-3 DURING OST 1.26.1 ON HP ROTOR DISPLACEMENT

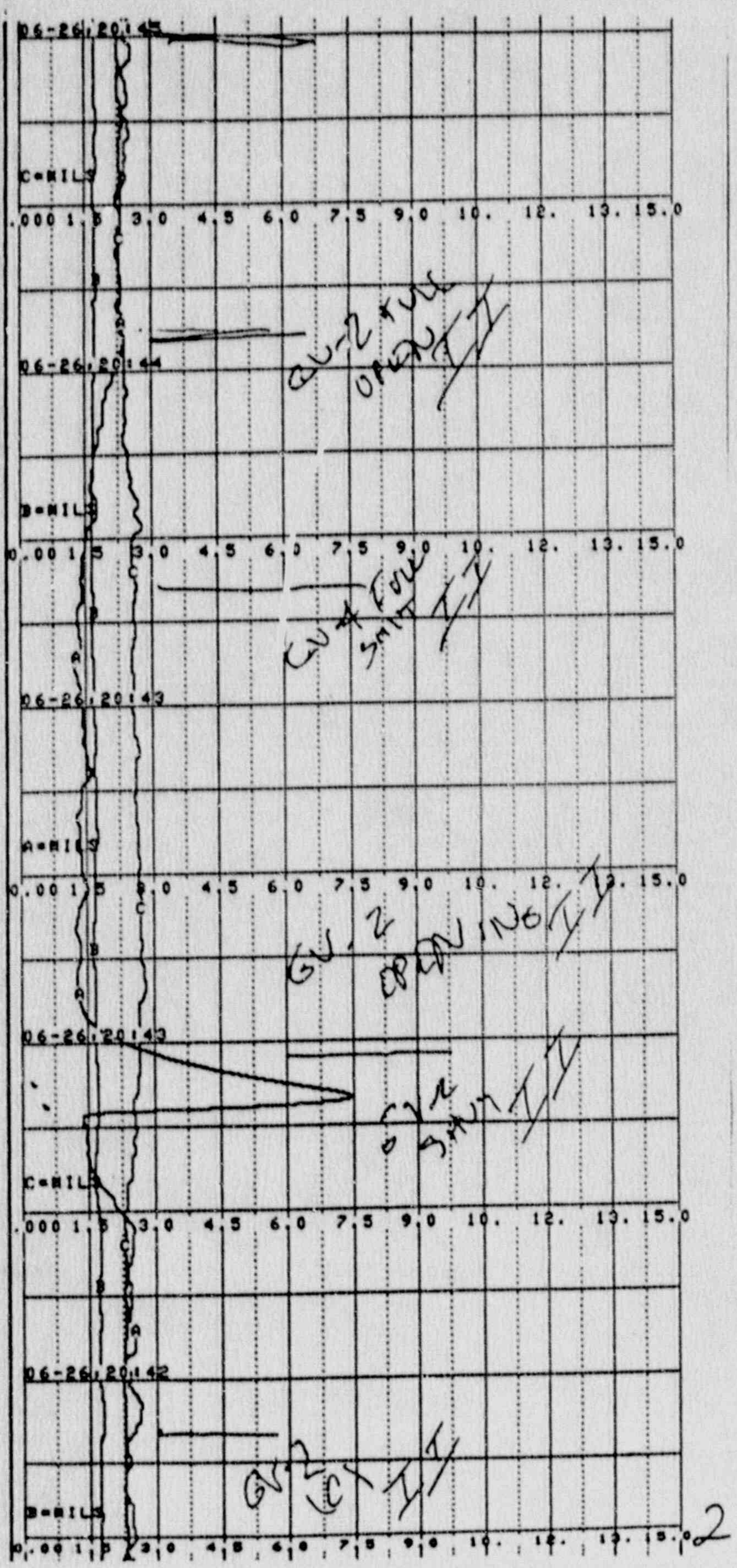
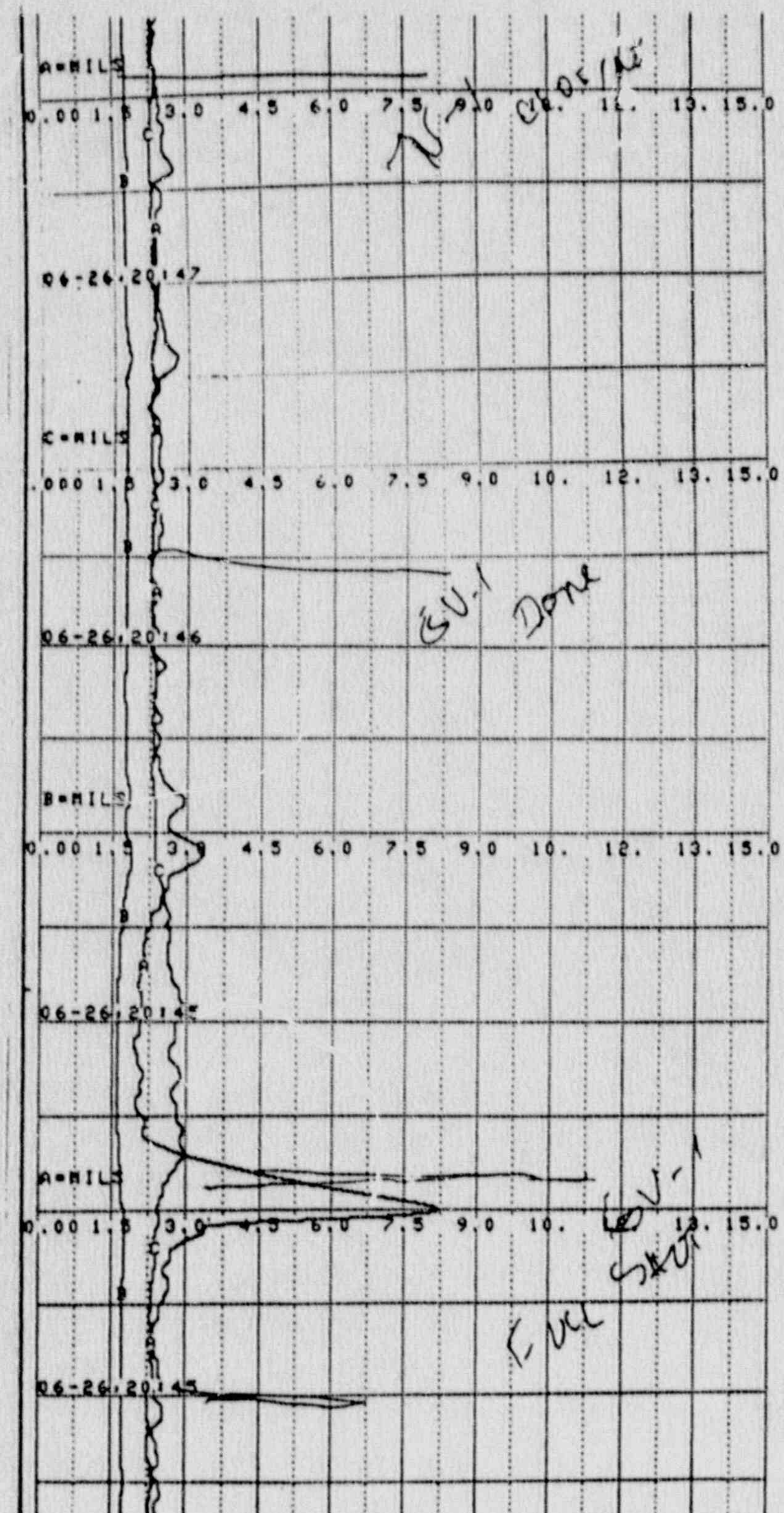


FIGURE NO. A4.1

TYPICAL RESPONSE OF GV-2 DURING OST 1.26.1 ON HP ROTOR DISPLACEMENT

FIGURE NO. A4.?



TYPICAL RESPONSE OF GV-1 DURING OST 1.26.1 ON HP ROTOR DISPLACEMENT

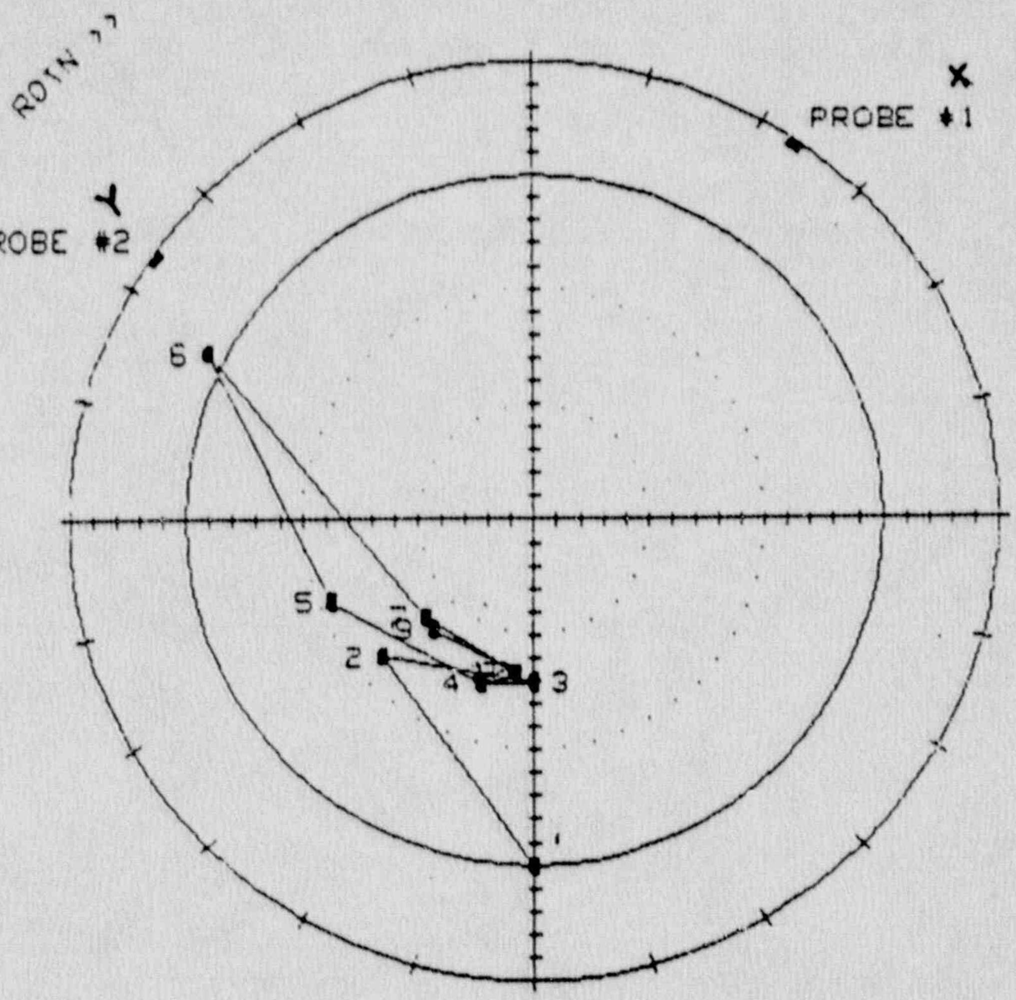
FIGURE NO. A5

BENTLY
NEVADA
CORP.

PLOT NO: 06-001

PLANT ID: BEAVER VALLEY
TRAIN ID: UNIT #1
MACHINE ID: HP
PROBE #1 ID: 1XD (RELATIVE)
PROBE #2 ID: 1YD (RELATIVE)

RUN: 1 DATE: 6 FEB 87 TIME: 4:55 PM



SHAFT CENTERLINE PLOT

BEARING CLEARANCE =30.0 MILS
PLOT RADIUS =20.00 MILS/ 4.00 VOLTS
PROBE #1 SCALE FACTOR = 200 mV/MIL
TIC SPACING = 1.00 MILS
PROBE #2 SCALE FACTOR = 200 mV/MIL

#	RPM	TIME	DC VALUES (Volts) PROBE #1	PROBE #2	REMARKS
1	20-50		-14.13	-12.74	PREVIOUS DATA-12 SEPT 86
2	1800		-13.39	-10.63	100 % LOAD-12 SEPT 86
3	1800	1605	-12.83	-11.83	810 MW GROSS-6 FEB 87
4	1800	1655	-13.10	-11.45	#1 GOV VALVE-6 FEB 87
5	1800	1655	-13.25	-10.00	#2 GOV VALVE-6 FEB 87
6	1800	1655	-12.10	-7.90	#3 GOV VALVE-6 FEB 87
7	1800	1655	-12.90	-10.75	#1 THROTTLE-6 FEB 87
8	1800	1655	-12.85	-11.65	#2 THROTTLE-6 FEB 87
9	1800	1655	-12.95	-10.85	#3 THROTTLE-6 FEB 87
10	1800	1655	-12.85	-11.65	#4 THROTTLE-6 FEB 87

FIGURE C1

CROSS - OVER & CROSS - UNDER PIPE ARRANGEMENT
RELATIVE CROSS - OVER PIPE FLOW DURING
VALVE TESTING

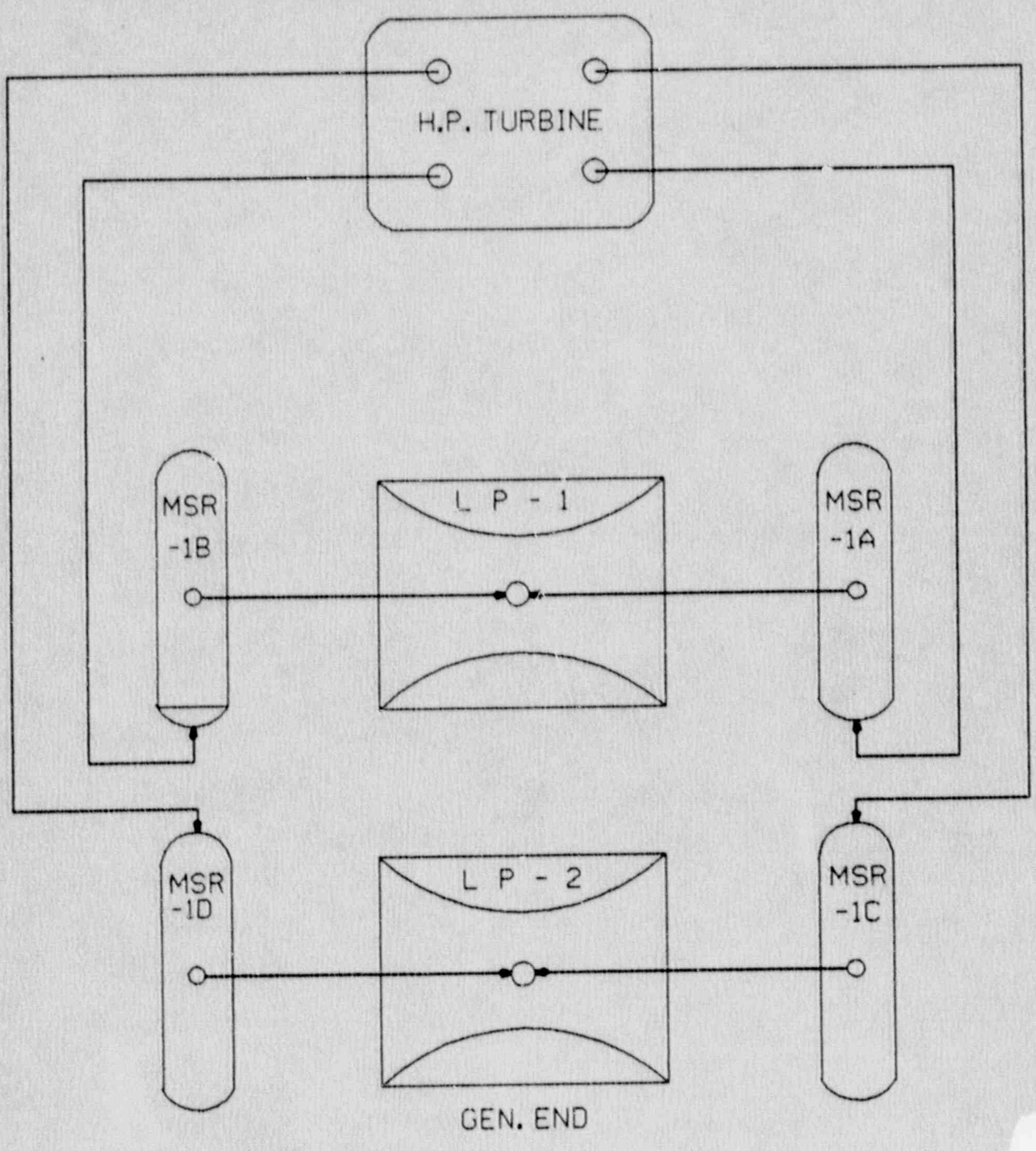
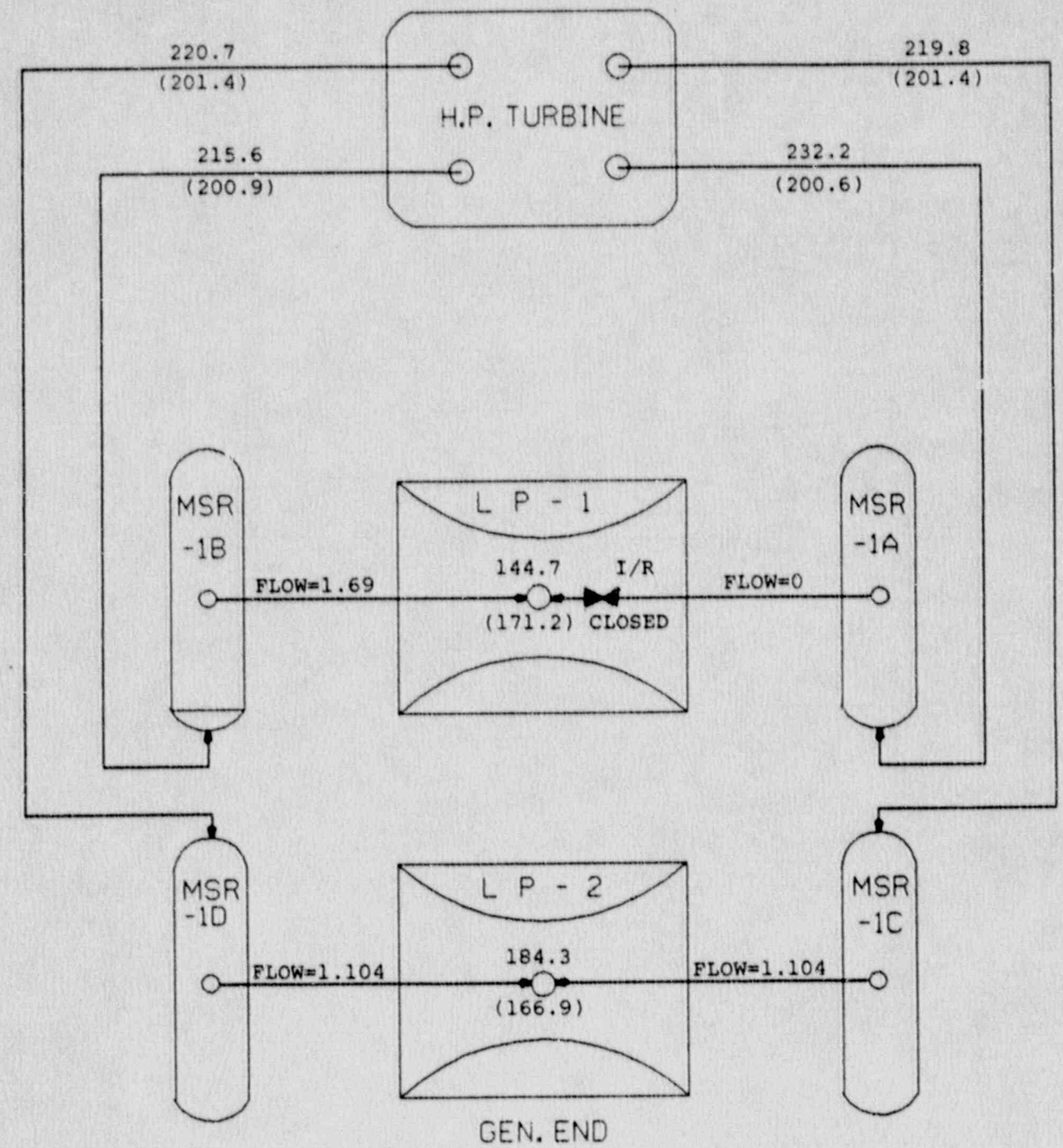


FIGURE C1.1

CROSS - OVER & CROSS - UNDER PIPE ARRANGEMENT
RELATIVE CROSS - OVER PIPE FLOW DURING
VALVE TESTING (TEST 1IRL)

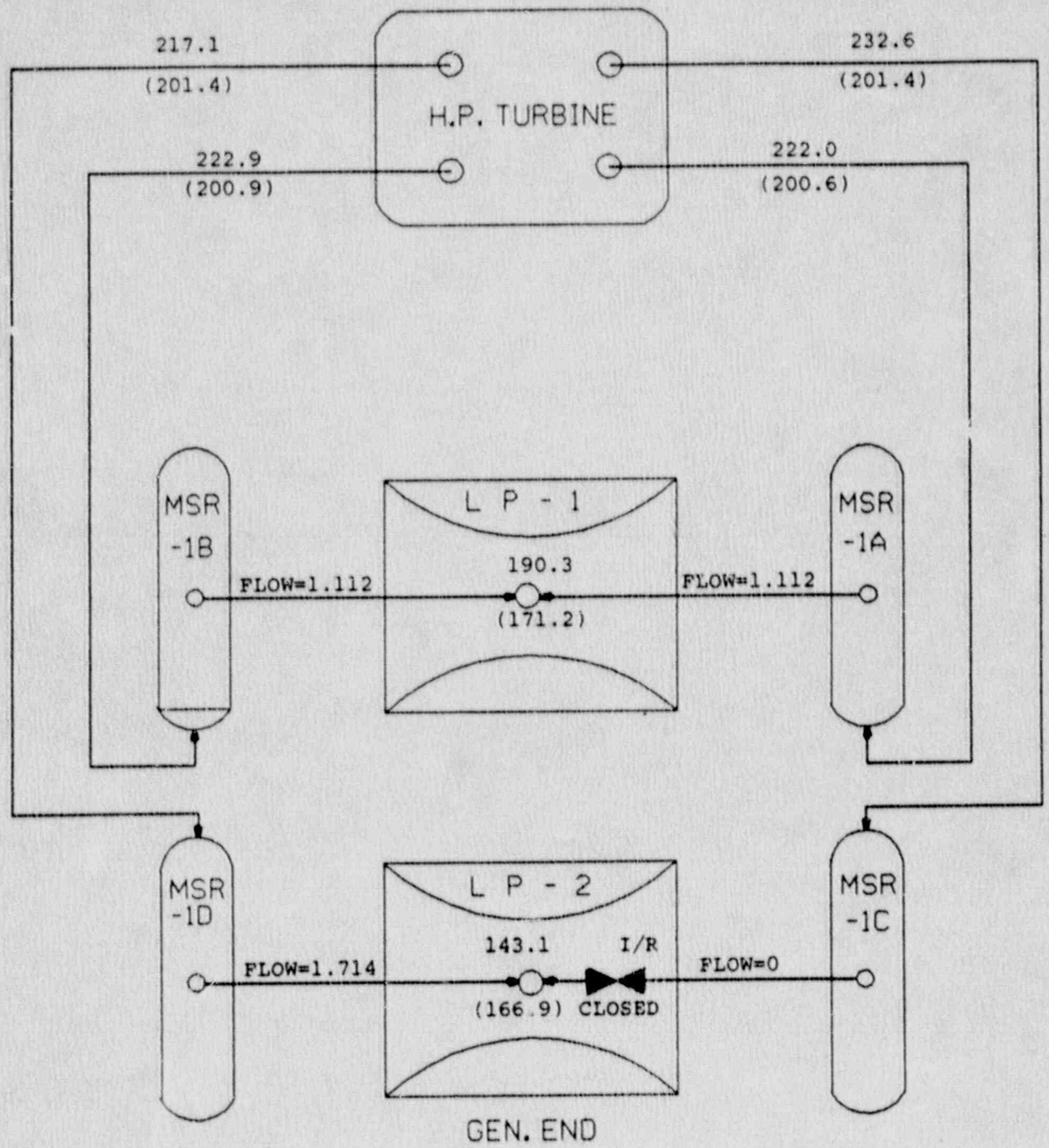


NOTES

1. All pressure in PSIG.
2. Values in parenthesis are for steady-state conditions at 95% load.
3. Steady-state Flow is 1.00 for <u>each</u> X-O pipe.

FIGURE C1.2

CROSS - OVER & CROSS - UNDER PIPE ARRANGEMENT
RELATIVE CROSS - OVER PIPE FLOW DURING
VALVE TESTING (TEST 2IRL)

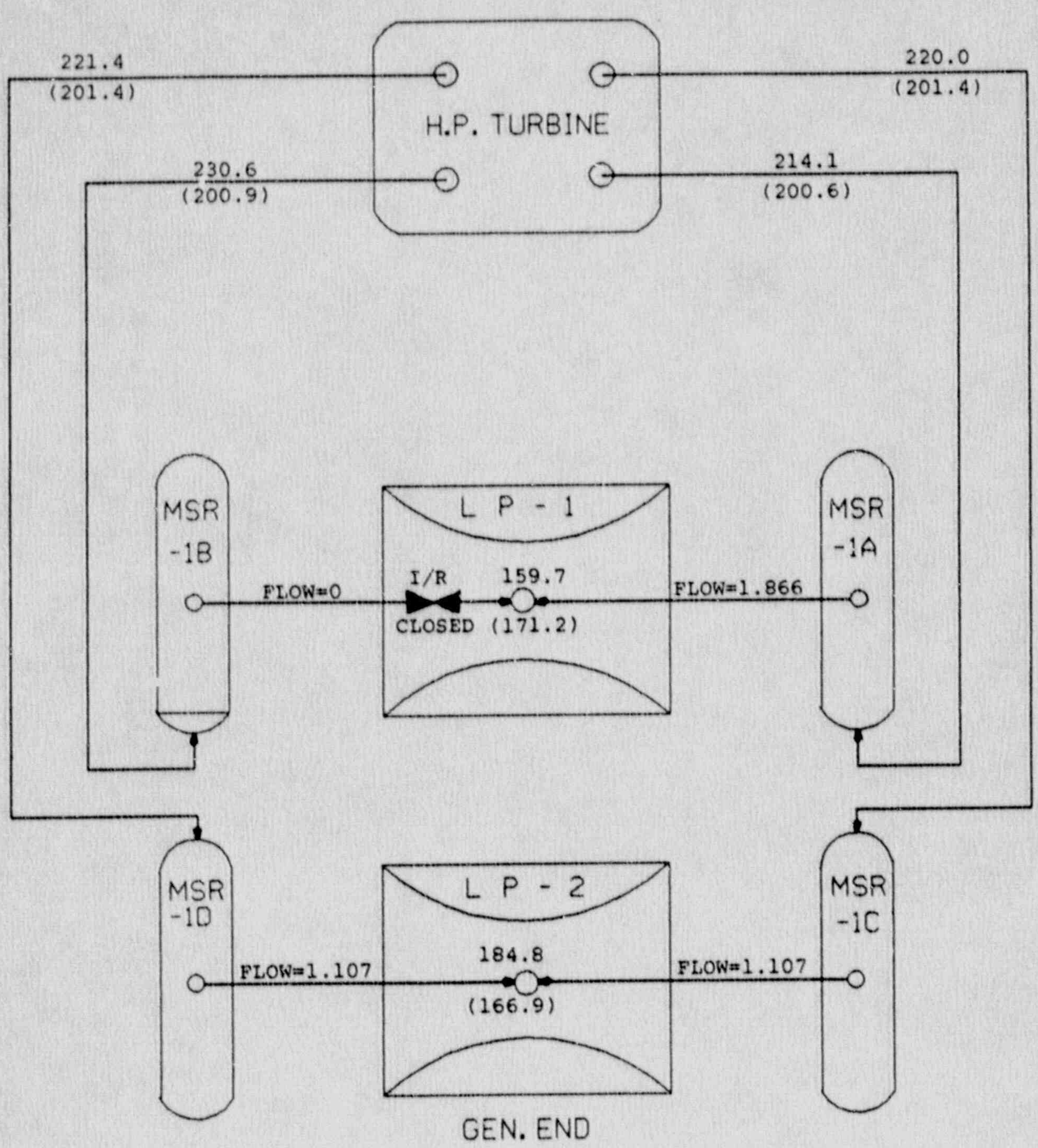


NOTES

1. All pressure in PSIG.
2. Value in parenthesis are for steady-state conditions at 95% load.
3. Steady-state Flow is 1.00 for each X-O pipe.

FIGURE C1.3

CROSS - OVER & CROSS - UNDER PIPE ARRANGEMENT
RELATIVE CROSS - OVER PIPE FLOW DURING
VALVE TESTING (TEST 1IRR)

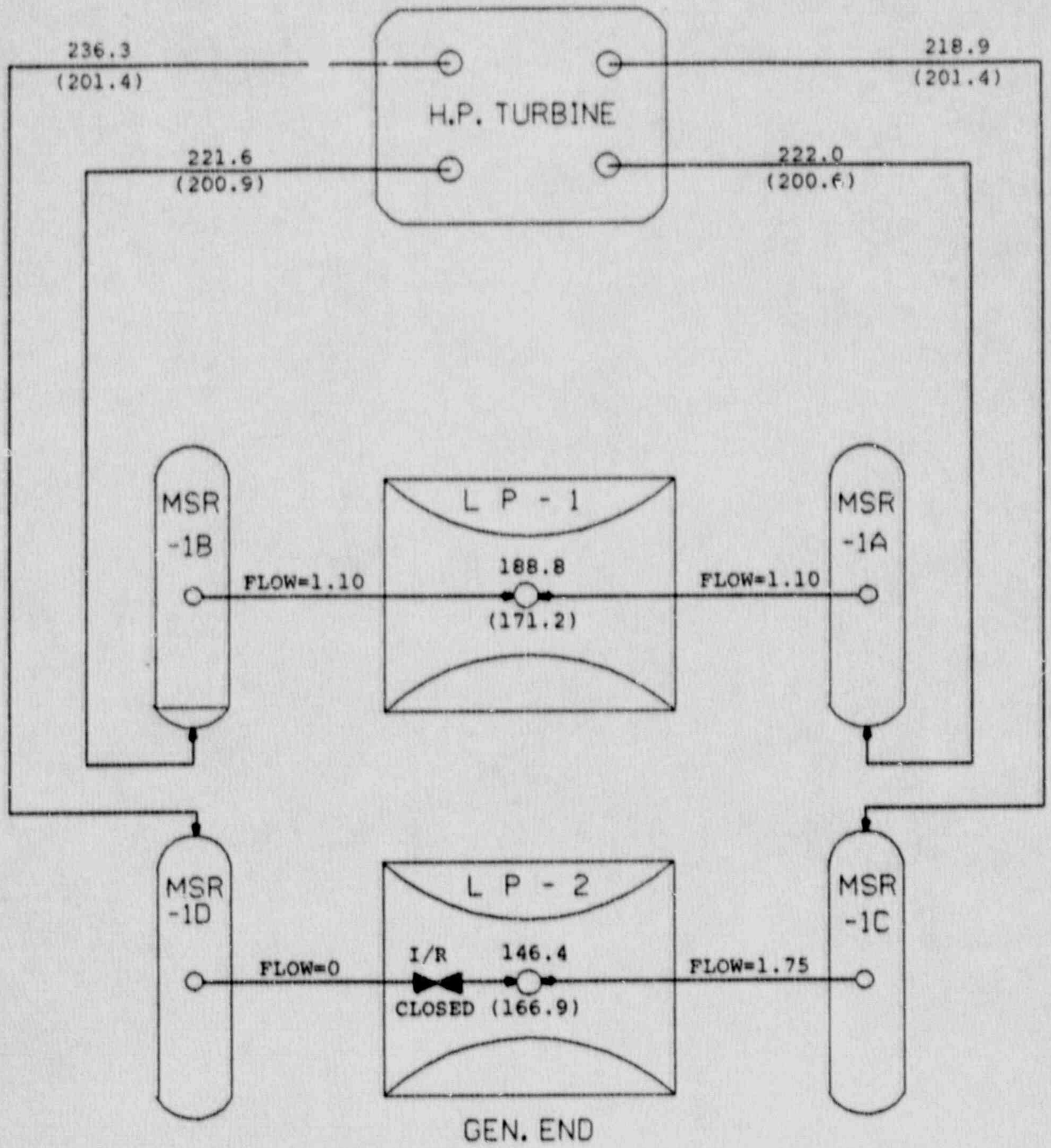


NOTES

1. All pressures in PSIG.
2. Values in parenthesis are for steady-state conditions at 95% load.
3. Steady-state Flow is 1.00 for each X-O pipe.

FIGURE C1.4

CROSS - OVER & CROSS - UNDER PIPE ARRANGEMENT
RELATIVE CROSS - OVER PIPE FLOW DURING
VALVE TESTING (TEST 2IRR)



NOTES

1. All pressures in PSIG.
2. Values in parenthesis are for steady-state conditions at 95% load.
3. Steady-state Flow is 1.00 for each X-O pipe.

FIGURE NO. C2

TYPICAL RESPONSE OF ELECTRICAL OUTPUT (MW) DURING OST 1.26.1
FROM INTERCEPT AND REHEAT STOP VALVE TESTING

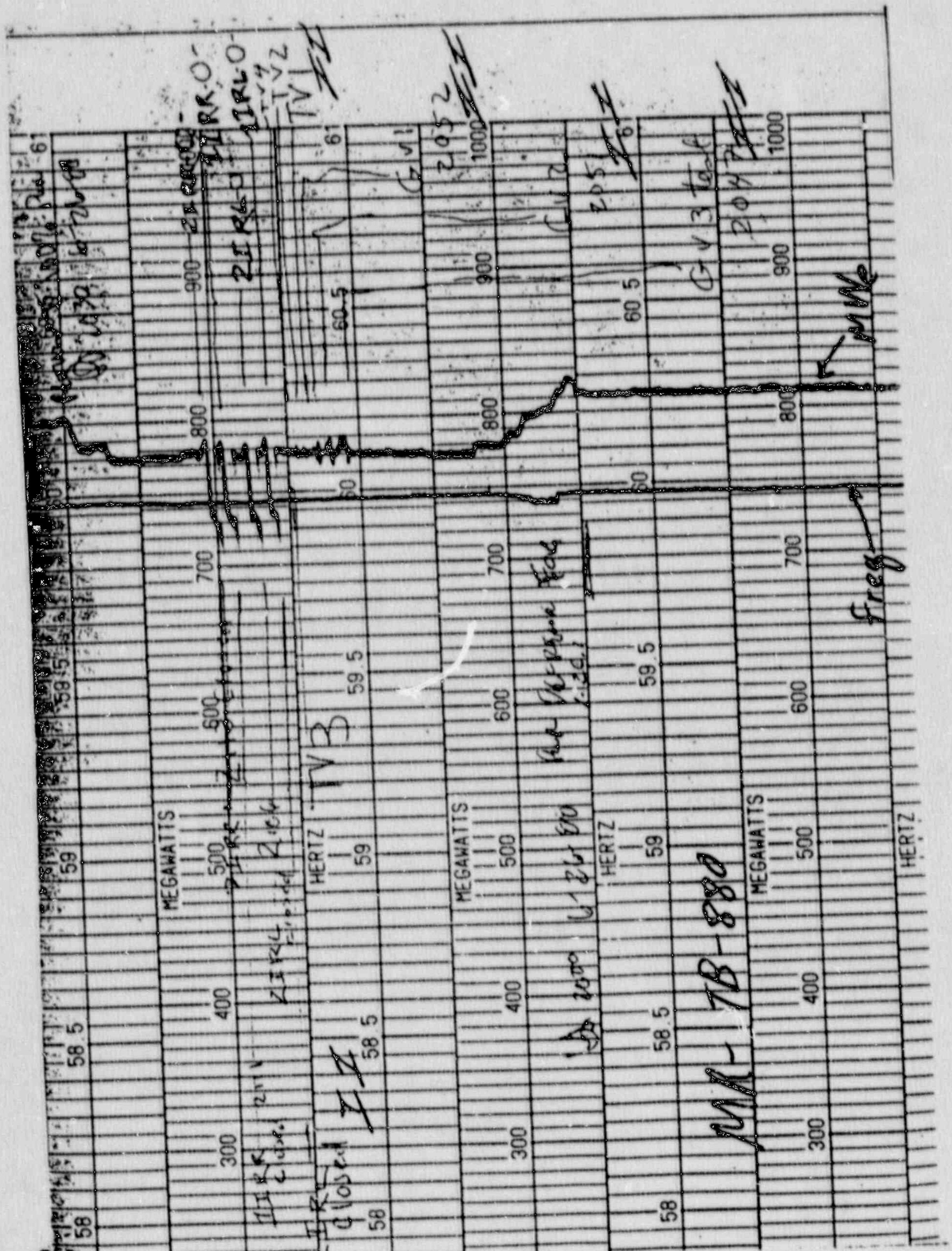


TABLE B1

ESTIMATED STEAM FLOW RELATIONSHIPS DURING TESTING OF THROTTLE VALVES (LB./HR.)

95% LOAD (NO. 1 TV CLOSED)

FLOW TO → / FLOW FROM ↓	NO. 1 GV	NO. 2 GV *	NO. 3 GV	NO. 4 GV (CLOSED)	TOTAL FLOWS THR. VALVES
NO. 1 TV (CLOSED)	0				0
NO. 2 TV		2.0×10^6			2.0×10^6
NO. 3 TV	2.4×10^6		4.0×10^6		6.4×10^6 (MAXIMUM)
NO. 4 TV		2.0×10^6		0	2.0×10^6
TOTAL FLOWS GOV. VALVES	2.4×10^6	4.0×10^6	4.0×10^6	0	10.4×10^6

* If two (2) TV supply (1) GV, the flows are assumed to be supplied equally from each TV.

TABLE B2

ESTIMATED STEAM FLOW RELATIONSHIPS DURING TESTING
OF THROTTLE VALVES (LB./HR.)

95% LOAD (NO. 3 TV CLOSED)

FLOW TO → / FLOW FROM ↓	NO. 1 GV	NO. 2 GV *	NO. 3 GV	NO. 4 GV (CLOSED)	TOTAL FLOWS THR. VALVES
NO. 1 TV	2.4×10^6		4.0×10^6		6.4×10^6 (MAXIMUM)
NO. 2 TV		2.0×10^6			2.0×10^6
NO. 3 TV (CLOSED)			0		0
NO. 4 TV		2.0×10^4		0	2.0×10^6
TOTAL FLOWS GOV. VALVES	2.4×10^6	4.0×10^6	4.0×10^6	0	10.4×10^2

* If two (2) TV supply (1) GV, the flows are assumed to be supplied equally from each TV.

APPENDIX A

CALCULATED HP ROTOR LOADS AT BEARING NO. 1 AND NO. 2 DUE TO HP CONTROL STAGE OPERATION

(1) Power Developed = Torque x Speed of Shaft

Torque = Radius x Force

(2) $$\text{Power Developed, KW} = \frac{\text{Force (Lb.)} \times \text{Radius (Ft.)} \times \text{Speed (RPM)} \times 2\pi}{44247 \ \frac{\text{(Ft.-Lb.)}}{\text{(KW-Min.)}}}$$

(2') $$\text{Power Developed, KW} = \frac{\text{Steam Flow (Lb./Hr.)} \times \text{Energy to Work (BTU/Lb.)}}{3412 \ \text{(BTU/KW-Hr.)}}$$

Substitute (2) in (2')

(3) $$\frac{\text{Force} \times \text{Radius} \times \text{Speed} \times 2\pi}{44247} = \frac{\text{(Steam Flow} \times \text{Energy to Work)}}{3412}$$

(4) $$\text{Force} = \frac{\text{Steam Flow} \times \text{Energy to Work} \times 44247}{\text{Radius} \times \text{Speed} \times 2\pi \times 3412}$$

$$= \frac{7042 \times \text{Steam Flow} \times \text{Energy to Work}}{3412 \times \text{Radius} \times \text{Speed}}$$

$$= \frac{7042 \times \text{KW}}{\text{Radius} \times \text{Speed}}$$

LET:

Shaft Speed = 1800 RPM
Radius of Stage = 2.725 Ft. (32.70 In.)

(4') $$\text{Force (Lb.)} = \frac{7042 \times \text{KW}^{\star}}{2.725 \times 1800} = 1.435 \ \text{KW},$$ (Force Acting at Centroid of Nozzle Arc)

* For Estimating KW, See Appendix A1

APPENDIX A1

CALCULATED OUTPUT (KW) OF HP CONTROL STAGE

ACTUAL PLANT DATA - 95% LOAD FOR OST 1.26.1

				REFERENCE SOURCE
8	Steam Gen. Flow, PPH	11,097,917		P-250 Computer
9	Blow-Down Flow, PPH	?		
10	Reheater Flow, PPH	722,339		Heat Balance - W
11	Throttle Flow, PPH	10,375,578		8 - 10
12	Stem Leakage, PPH	409		Heat Balance - W
13	No. of Gov. Valves Open	#2 + #3 Full	#1 Part.	#4 Closed
14	Nozzle Flow, PPH	7,988,635	2,386,534	Trial Calc. **
15	Throttle Press., PSIA	844	844	P-250 or VB
16	Gov. ΔP% (1.00-ΔP/100)	0.980 (2%)	0.777 (22.5%)	Assumed or Derived
17	Steam Chest Press., PSIA	827.1	656.	(at Nozzle Chamber)
18	Inlet Enthalpy, B/Lb.	1197.1	1197.1	Table
19	Inlet Moisture, %	0.0	0.8	Assumed 0%
20	Inlet Entropy, B/Lb./°F	1.415	1.431	Table
21	Exit Enthalpy, B/Lb.	1157.5	1177.5	Table
22	Exit Moisture, %	6.2	3.7	Table
23	Exit Press.(1st.Stg),PSIA	516.8	516.8	P-250 or VB
24	Pressure Ratio, ρ	1.600	1.270	17/23
25	Isentropic Drop, B/Lb. ΔH	39.6	19.6	18 - 21
26	Flow Coefficient, ϕ	0.930	0.700	Assumed Curve(ϕ vs ρ)
27	Nozzle Height Corr., Ht	1.050	1.050	Assumed
28	Moisture Corr., Hm	1.040	1.040	Assumed
29	K-Factor Lb/H-In^2-Lb/In^2	50.3	50.3	Table
30				
31	Nozzle Height, In.	3.400	3.400	Ref. W 739-J-791
32	Stage Mean Dia., In.	65.400	65.400	Ref. W 739-J-791
33	Nozzle Gauging, λ	0.270	0.270	Assumed
34	Wheel Velocity,Ft./Sec. ν	513.8	515.8	ν = 32 x Speed/229.1
35	Isentropic Vel. Ft/Sec, C	1408.3	990.8	C = 223.8 $\sqrt{25}$
36	Velocity Ratio, ν	0.365	0.518	34/35
37	Percent Admission	50	25	
38	Effective Nozzle Area,In^2	189.078	94.539	31 x π x 32 x 33 x 37
39				
40	Stage Efficiency, %/100 η	0.800	0.880	Estimated (η vs ν)
41	Energy to Work, B/Lb.	31.68	17.25	η = 40 x 25
42	Output, KW	74,173	12,065	14 x 41/3412.75
43				
44	Forces on Rotor Mid-Span, * Lb.	106,438	17,300	See Below *
45	Forces on Rotor Direction, °	225° CW	135° CW	12 O'clock = Zero°

* Forces on Rotor Mid-Span 44 = 1.435 x 42, Lb.

** Nozzle Flow = 17 x 26 x 27 x 28 x 29 x 38, Lb./Hr.