C.D.I. TECH NOTE NO. 84-19

MARK I WETWELL TO DRYWELL DIFFERENTIAL PRESSURE LOAD AND VACUUM BREAKER RESPONSE FOR THE DUANE ARNOLD ENERGY CENTER UNIT 1

Revision 0

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SUMMARY

Mark I wetwell to drywell vacuum breaker (VB) actuation velocities during the chugging phase of a postulated loss of coolant accident (LOCA) are predicted. Data collected during the full scale test facility (FSTF) test series is used to conservatively predict the differential pressure load across the VB. Adjustment is made for plant-unique drywell volumes with a vent dynamic model validated against FSTF test data. The predicted differential pressure load is used to drive a valve dynamic model with the plant-specific VB valve characteristics. The valve dynamic model, validated against full scale test data, conservatively predicts actuation velocities. These velocities are predicted on a plant-unique basis, and presented in this report.

Application of the above methodology to the Duane Arnold Energy Center Unit 1 results in a negative differential pressure peak of 1.11 psid, applied across installed 18-inch GPE internal vacuum breakers, and a predicted maximum closing impact velocity of 8.68 rad/sec.

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1. INTRODUCTION

The Mark I long term containment program included the construction of a full scale test facility (FSTF) modeling a 1/16th sector of a Mark I torus and ring header, with eight downcomers. A series of tests simulating a loss of coolant accident (LOCA) demonstrated a chugging phenomenon occurring at the ends of the downcomers. Continuum Dynamics, Inc. (C.D.I.) was requested to examine the FSTF geometry and develop a vent acoustic model for predicting the differential pressure across wetwell to drywell vacuum breakers during the chugging phenomenon. Concurrently, C.D.I. developed a valve dynamic model that includes the hydrodynamic effects of pressure alleviation across the valve disc when the valve is partially open. These two efforts are summarized in Sections 2 and 3, respectively, of this report.

These methodologies have recently been reviewed and accepted by the Nuclear Regulatory Commission (Ref. 1). This report documents the application of these methodologies to the Duane Arnold Energy Center Unit 1 (hereafter referred to as Duane Arnold).

2. FORCING FUNCTION METHODOLOGY

This section of the report summarizes the methodology used to define plant-unique wetwell to drywell Mark I vacuum breaker differential pressure forcing functions from FSTF data. Additional details of the analysis may be found in Refs. 2 and 3.

During the Mark I FSTF test series, wetwell to drywell vacuum breaker actuation was observed during the chugging phase of a postulated LOCA. This observation lead to the development of a methodology defining the plant-unique pressure loading function acting across a vacuum breaker during the chugging phenomenon. The methodology idealized the FSTF as an interconnection of simple acoustic elements and modeled the chugging phenomenon as a condensation process occurring at the exit of each downcomer across the steam water interface. The FSTF drywell airspace pressure time history data was used with a vent dynamic model to compute the consistent condensation source velocity time history during chugging. The FSTF ring header pressure time history data was then used to validate the methodology.

For plant-unique applications the most important parameter controlling the magnitude of the vent pressure oscillations (and hence the VB forcing function) was determined to be the ratio of the drywell volume to main vent area. These forcing functions are specified as time histories of the differential pressure across the valve disc, using the time segment of actual FSTF data that generated the most conservative condensation source strength.

The steps taken in the development of the plant-unique forcing function model are shown in Figure 2-1. Step 1 involves the development of analytical models for: the unsteady motion in the steam vent system (characterized as shown in Figure 2-2); the dynamics of condensation across the steam water interface (schematically shown in Figure 2-3); and the dynamics of the suppression pool and the wetwell airspace (idealized as shown in Figure 2-4). In the analysis the condensation source is a velocity time history representing the transport of steam into water at the steam water interface.

STEP

1	DEVELOP A DYNAMIC MODEL OF THE VENT SYSTEM, STEAM WATER INTERFACE AND POOL SLOSH WITH THE CONDENSATION RATE AT THE INTERFACE UNKNOWN.
2	USE MEASURED DRYWELL PRESSURE TO DETERMINE THE CONDENSATION RATE.
3	WITH THE CONDENSATION RATE DETERMINED, PREDICT UNSTEADY PRESSURES AT OTHER VENT LOCATIONS TO VALIDATE THE MODEL.
4	USE THE CONDENSATION SOURCE AT THE VENT EXIT TO DRIVE DYNAMIC MODELS OF MARK I PLANTS TO DETERMINE PLANT-UNIQUE VACUUM BREAKER FORCING FUNCTIONS.

Figure 2-1. Steps in determining plant-unique vacuum breaker forcing functions.

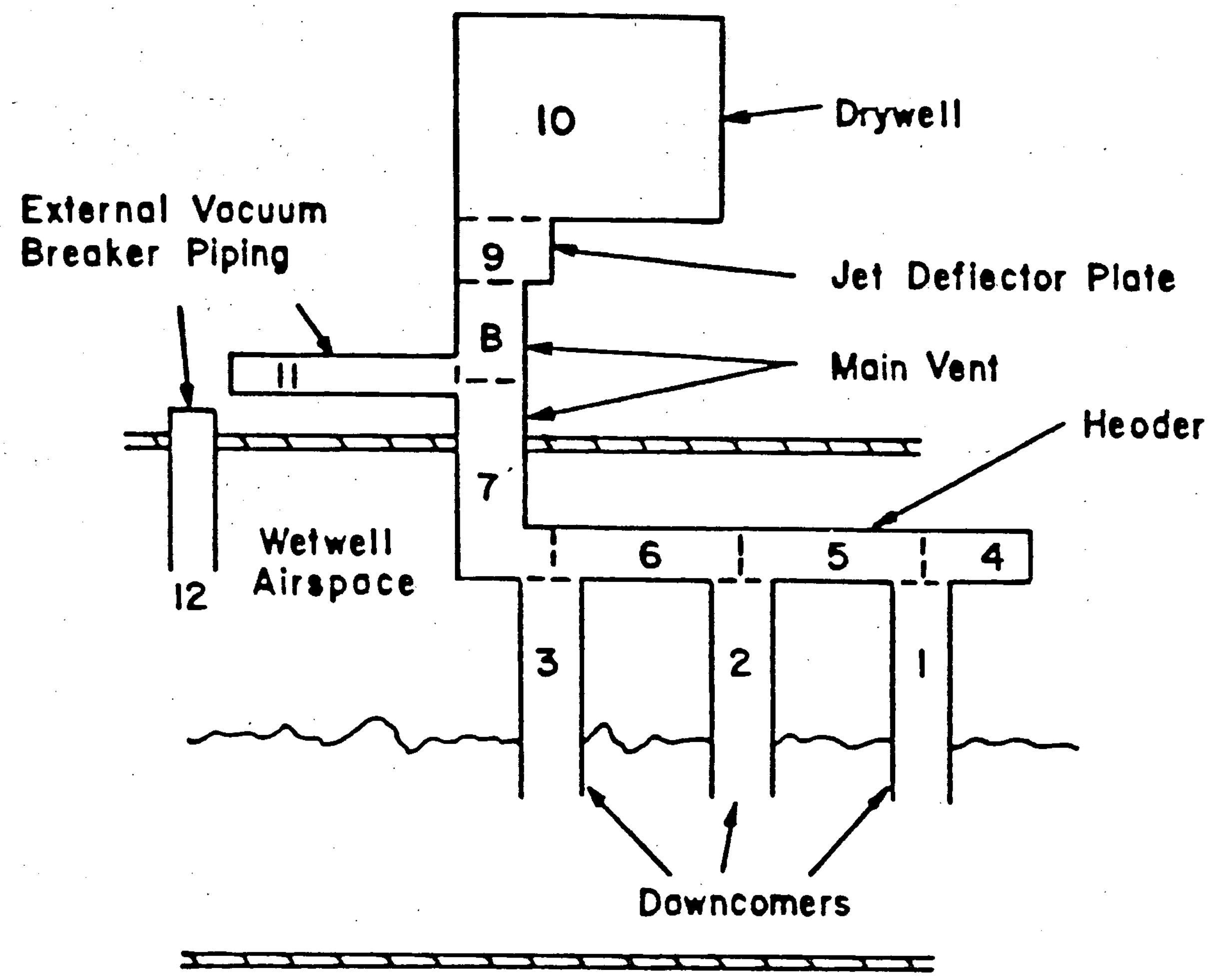

Figure 2-2. Schematic model of the vent system depicted by 12 dynamic components.

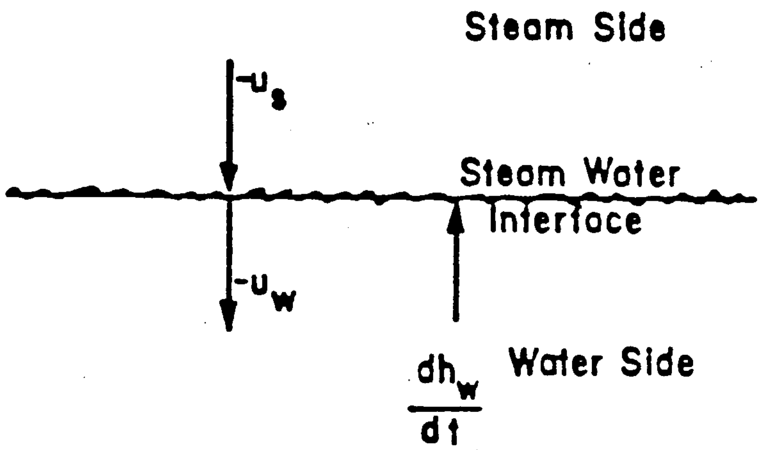

Figure 2-3. Details of the steam water interface.

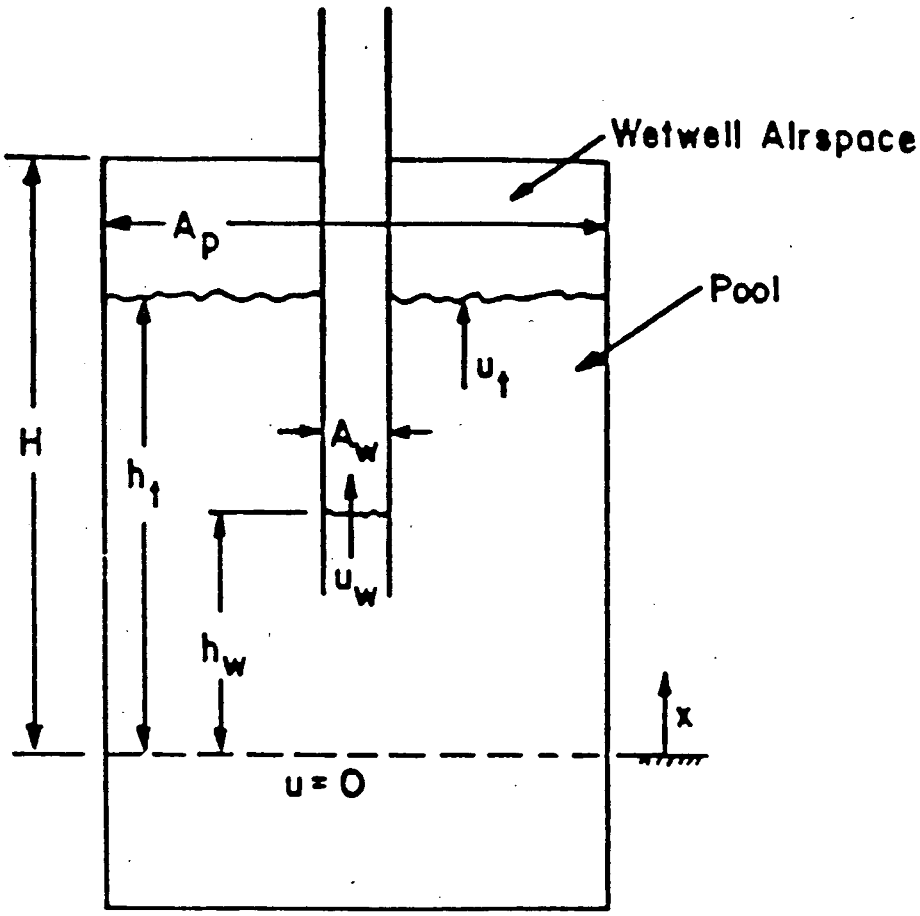

Figure 2-4. Details of the pool dynamic model around each downcomer.

For the purposes of step 1, this velocity time history is assumed to be unknown. The steam dynamics in the vent system are governed by one-dimensional acoustic theory (in the configuration used here, element 3 in Figure 2-2 is nulled). Jump conditions across the steam water interface are the Rankine-Hugoniot relationships. A one-dimensional model of the suppression pool (assigning an equal share of the wetwell airspace volume and pool area to each downcomer) was developed to account for the compression of the airspace with the lowering of the steam water interface in the downcomers.

For plants with external lines connecting the vacuum breakers to the main vent and the wetwell airspace (elements 11 and 12 in Figure 2-2), additional analysis and bounding linearized loss coefficients obtained from subscale acoustic tests (Ref. 4) are included in the vent model to conservatively predict the differential pressure across the VB disc. Internal vacuum breakers are attached at the main vent intersection with the ring header, element 7 of Figure 2-2. The same condensation source velocity time history is assumed to act at the end of each downcomer.

Step 2 involves determining the condensation source velocity time history by using the FSTF measured drywell pressure time history data during the period of most severe chugging.

Step 3 involves validation of the model in the FSTF by using the condensation source velocity time history determined in step 2 to predict the pressure elsewhere in the FSTF. A prediction of the ring header pressure time history was made and compared with experimental data. To bound the negative pressure peaks, a load factor of 1.06 was used to multiply the predicted results to match the largest pressure data spike. To identify the origin of the nonconservatism in the vent dynamic model, the input parameters to the model were varied by wide margins without altering the results (Ref. 5). The origin of the nonconservatism appears to result from the assumption of applying an averaged condensation source of each downcomer exit. This assumption was required because sufficient independent data sets do not exist to determine the condensation source at the exit of each downcomer independently.

Step 4 applies the modified condensation source velocity time history to the plant-unique vent dynamic model. The key assumption is made that the condensation source at the end of a downcomer is plant independent. The amount of steam condensed per chug per downcomer is assumed to be the same between the FSTF and Mark I plants. This assumption is supported by the observation that the condensation rate is fixed by local conditions at the vent exit, such as steam mass flow rate, noncondensibles, and thermodynamic conditions. These local conditions will vary only slightly between plants.

The only plant characteristics which are changed in a plant-unique calculation are the ratio of drywell volume to main vent area and the pool submergence. All lengths, areas and system flow and pool parameters are retained at their FSTF values in a plant-unique calculation. Thus, gross depressurization, controlled by drywell volume, is corrected on a plant-unique basis, while high frequency ring out at the vent natural frequency is not plant-unique and is essentially taken to be that of the FSTF. The plant drywell may be treated as a capacitance or as an acoustic volume composed of two right circular cylinders standing end to end. The acoustic volume model results in a more conservative forcing function for Duane Arnold.

The plant characteristic parameters given in Table 2-1 were used to compute the differential pressure time history across the vacuum breakers in Duane Arnold. Figure 2-5 shows the resulting differential pressure time history, without addition of the pressure resulting from the submergence head.

TABLE 2-1

Plant Characteristic Parameters
for Duane Arnold Energy Center Unit 1

FSTF Parameter	
Main vent area/downcomer area ratio	0.99
Main vent length	37.32 ft
Header area/downcomer area ratio	1.47
Header length	15.0 ft
Downcomer area	3.01 ft^2
Downcomer length	10.8 ft
Vent/pool area ratio	0.045
Plant-Specific Parameter	
Drywell volume/main vent area ratio	775.72 ft
Submergence head	3.0 ft water
Lower drywell volume length	50.21 ft
Lower drywell volume area	1724.0 ft^2
Upper drywell volume length	51.90 ft
Upper drywell volume area	373.6 ft^2

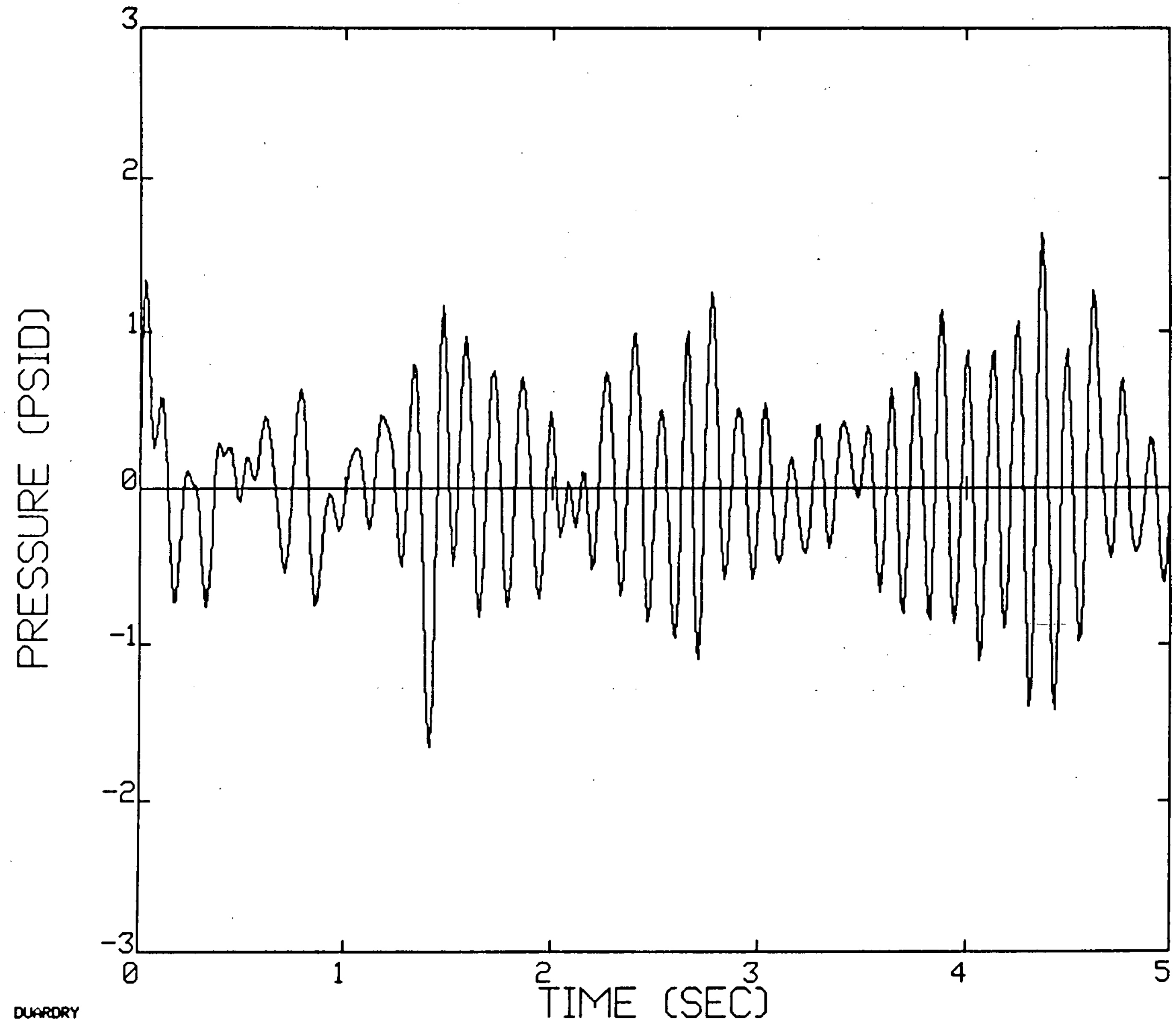

Figure 2-5. Differential pressure time history predicted across a vacuum breaker at the main vent-ring head junction in Duane Arnold. Submergence head has not been added. a) 0 - 5 seconds.

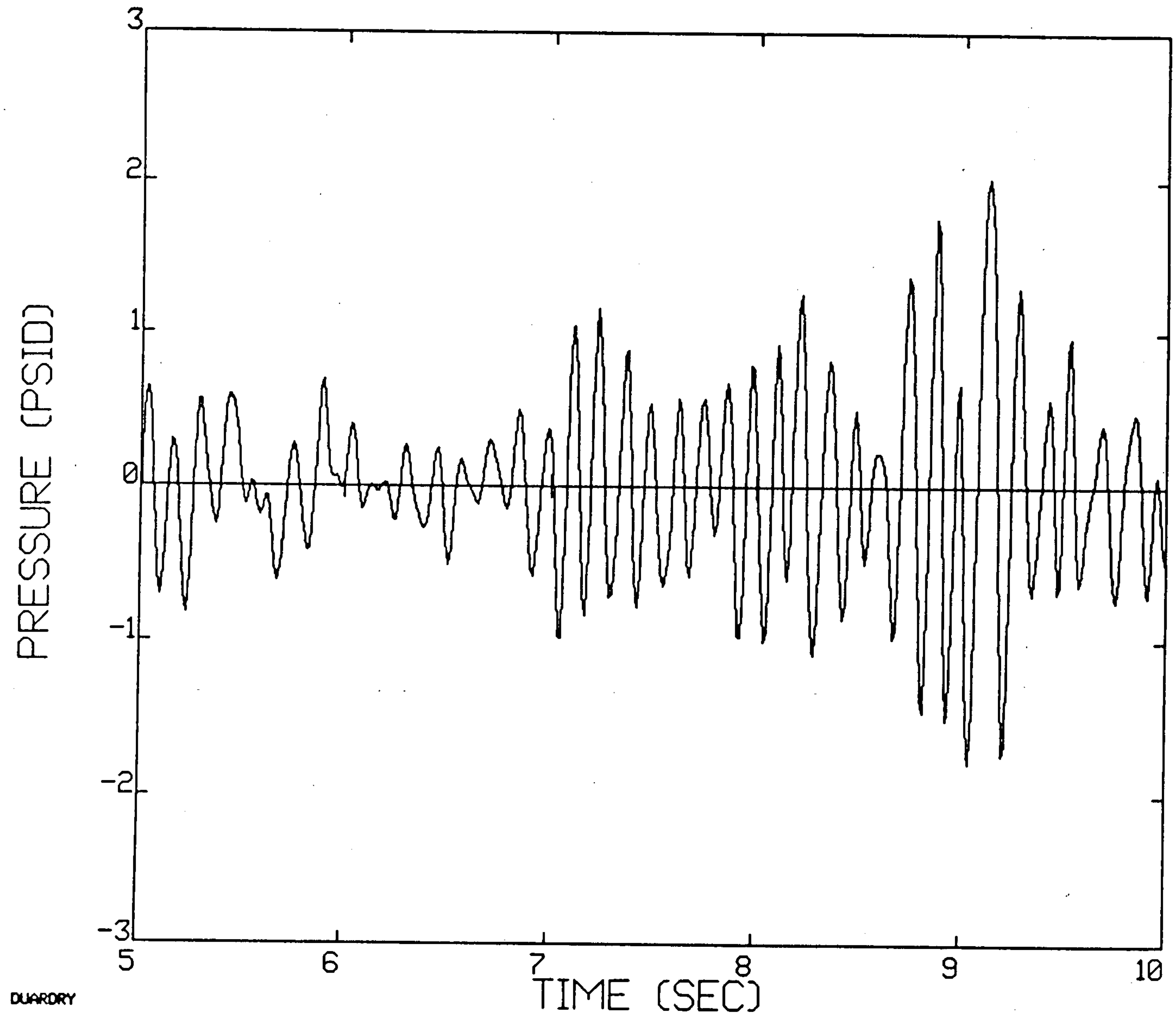

Figure 2-5b. 5 - 10 seconds.

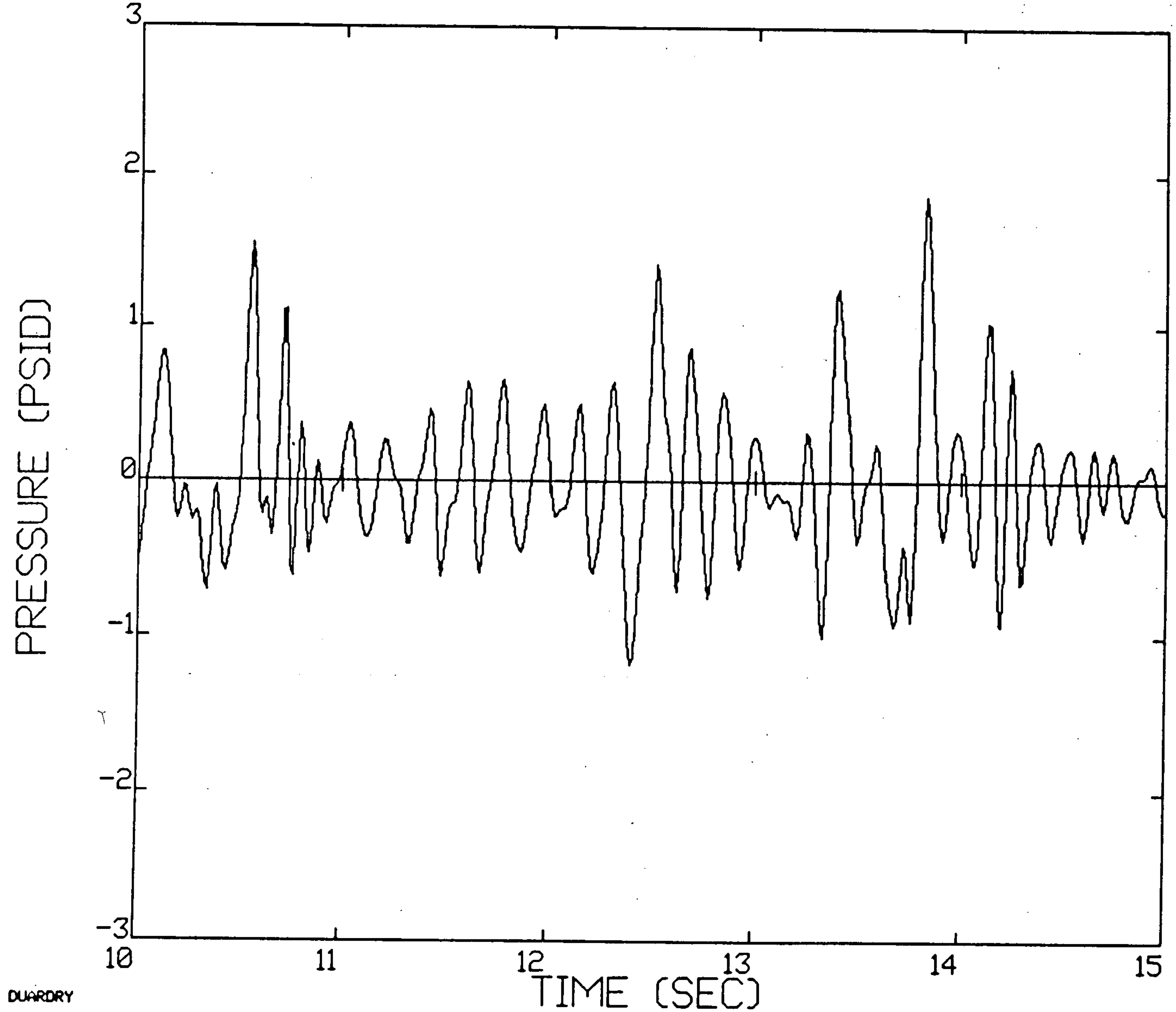

Figure 2-5c. 10 - 15 seconds.

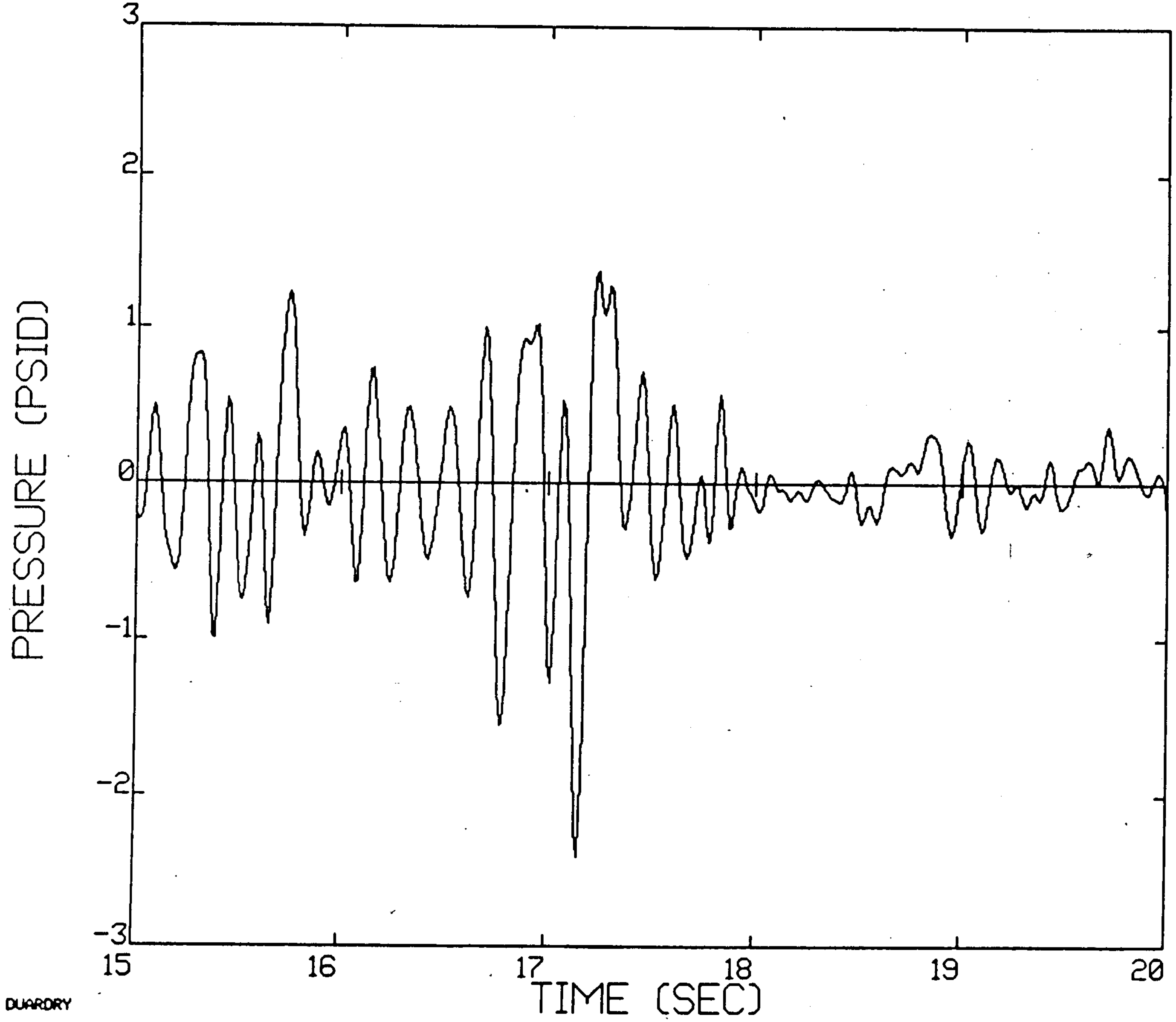

Figure 2-5d. 15 - 20 seconds.

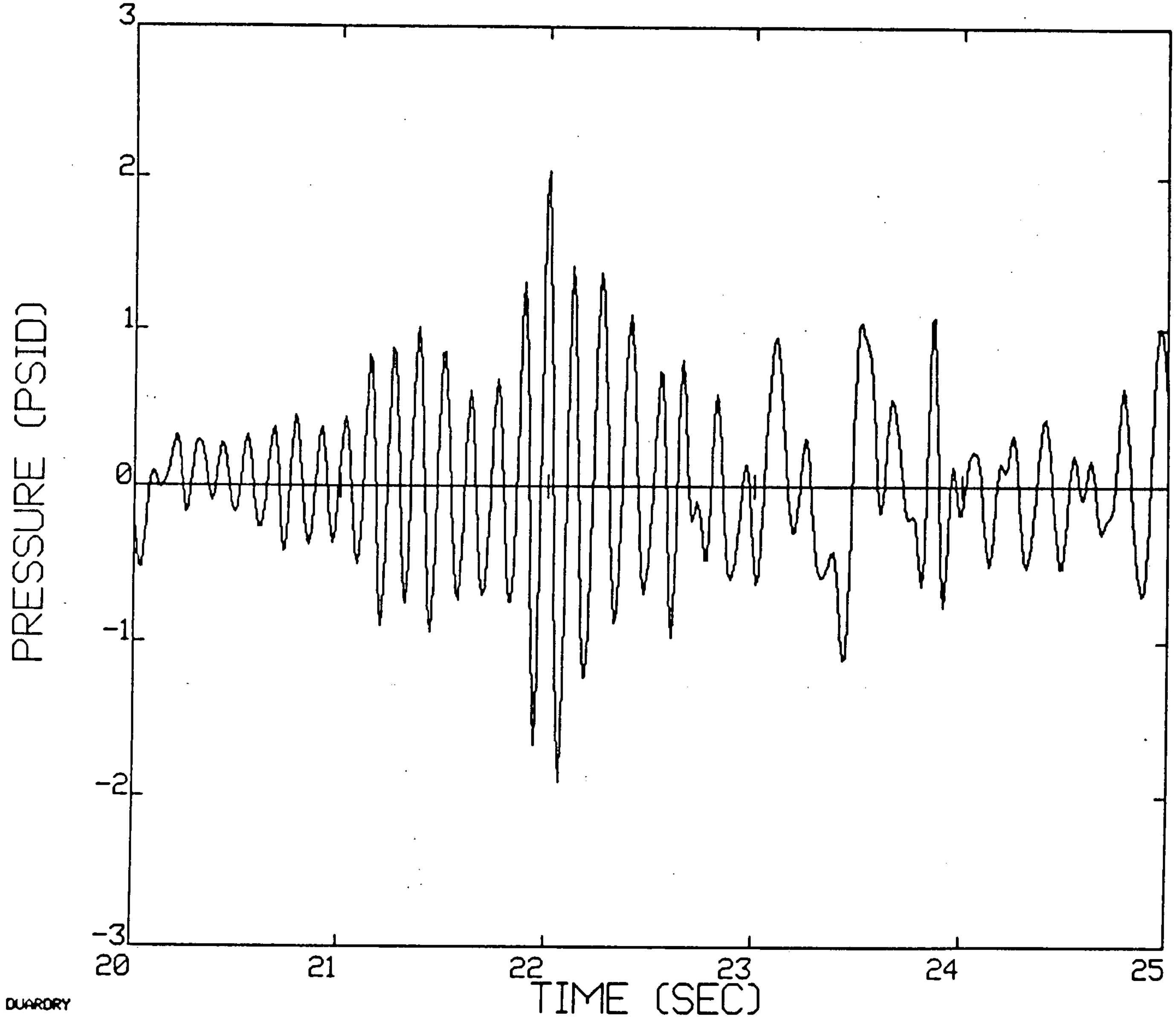

Figure 2-5e. 20 - 25 seconds.

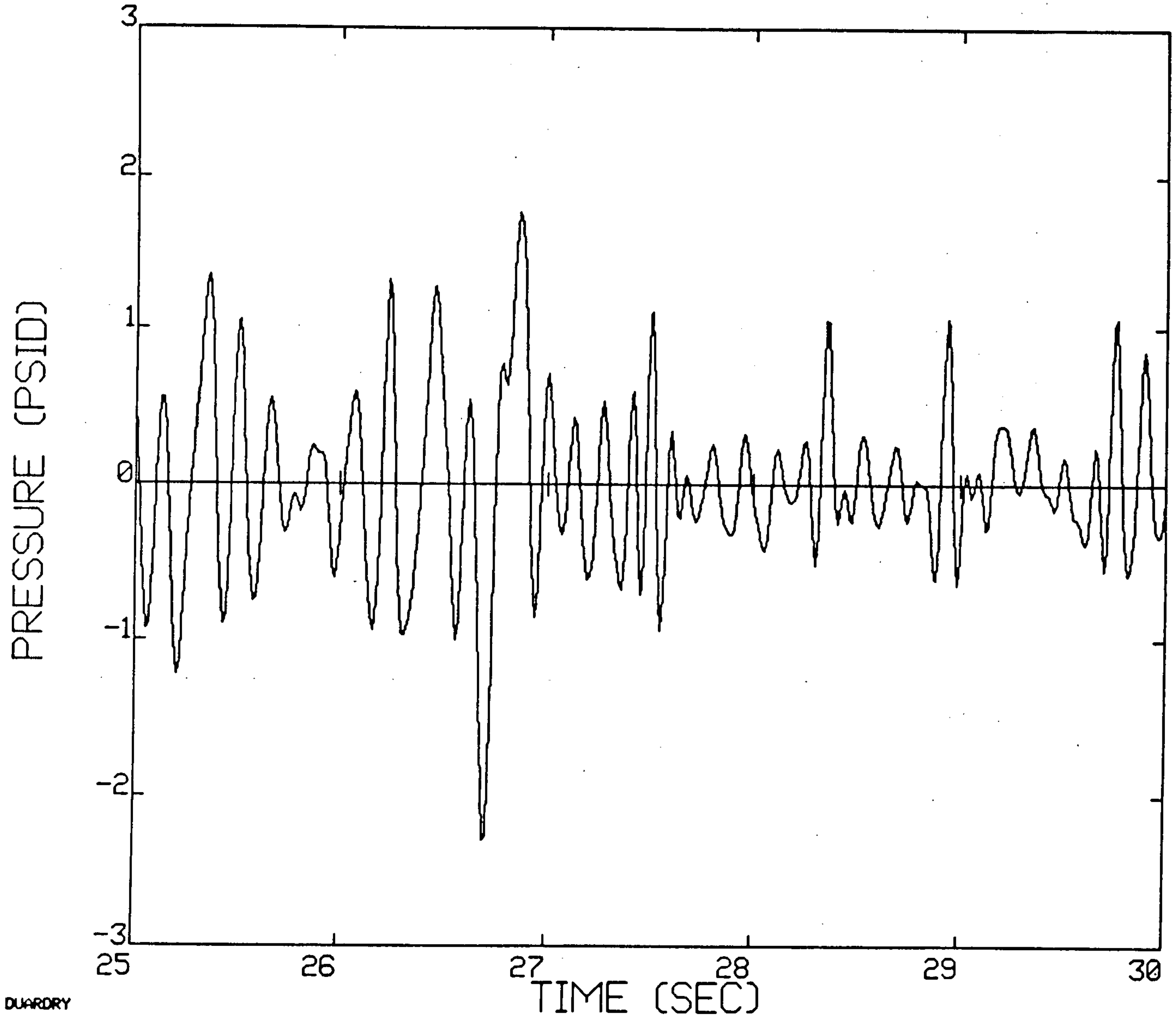

Figure 2-5f. 25 - 30 seconds.

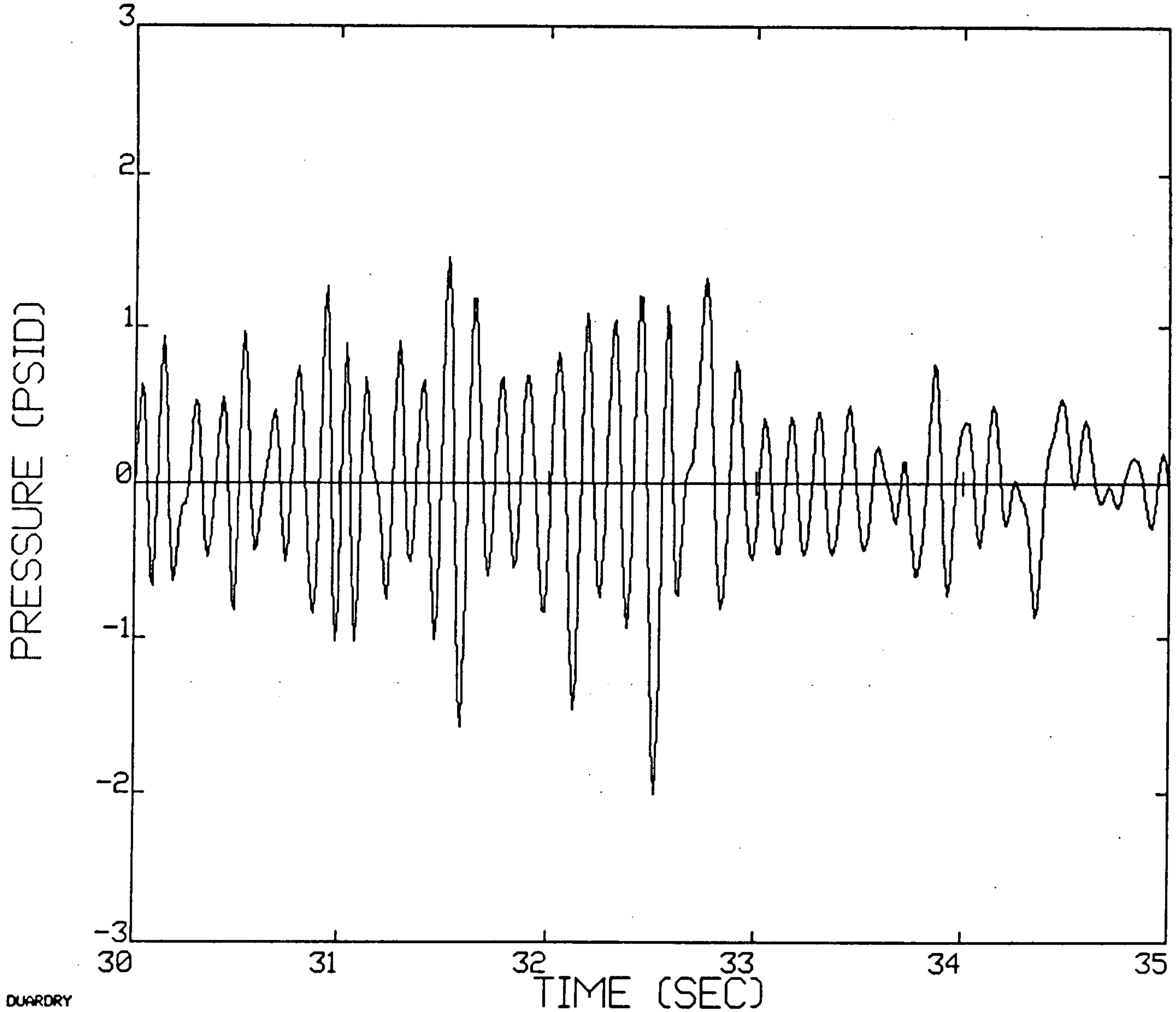

Figure 2-5g. 30 - 35 seconds.

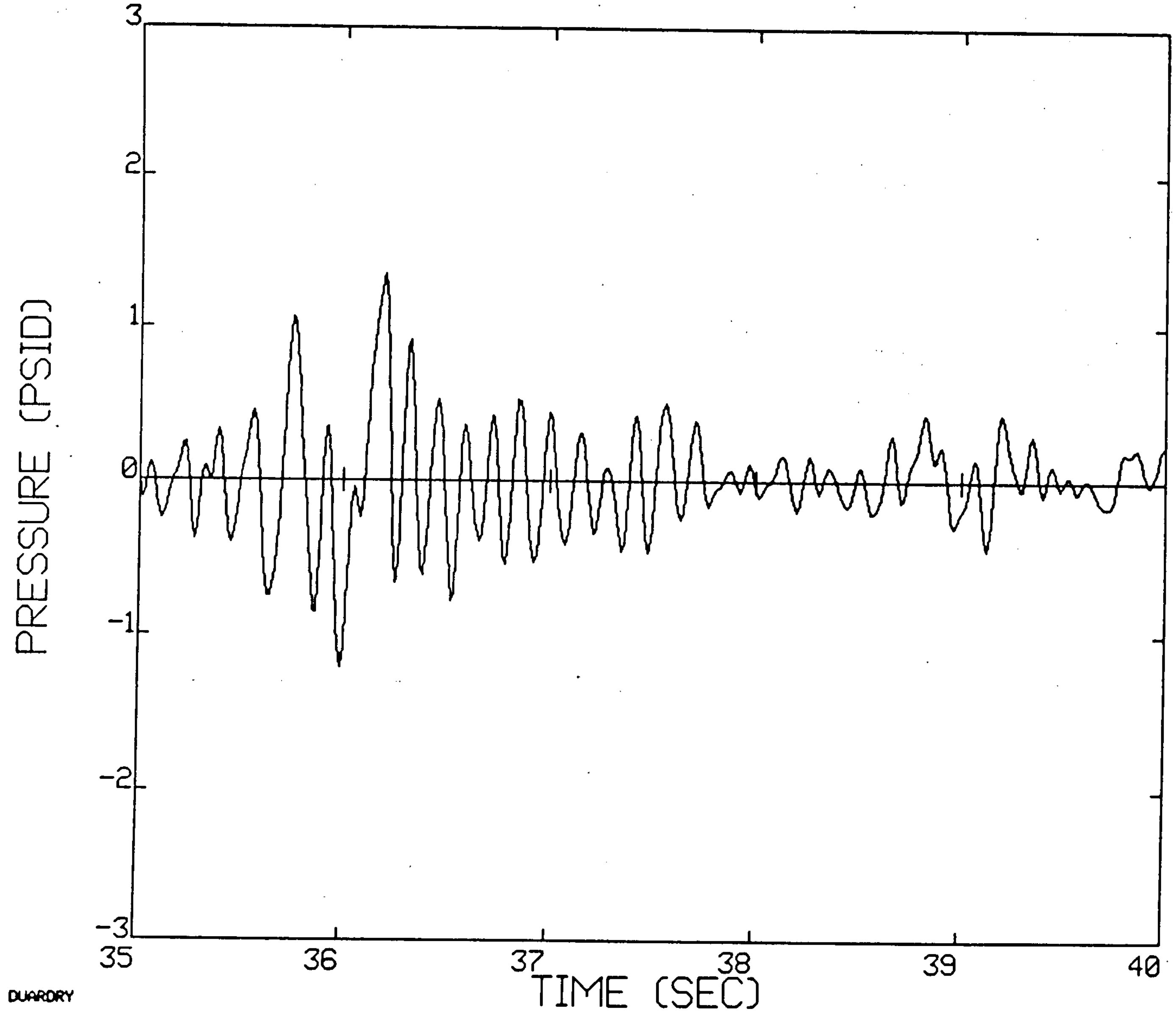

Figure 2-5h. 35 - 40 seconds.

3. VACUUM BREAKER METHODOLOGY

This section of the report summarizes the methodology used to construct the Mark I vacuum breaker valve dynamic model including hydrodynamic effects. Additional details of the analysis may be found in Ref. 6.

During the Mark I shakedown tests, the vacuum breaker displacement time history was recorded. A methodology was developed that uses the differential pressure forcing function across the VB (computed by the vent dynamic model) and includes the effect of torque alleviation as a consequence of fluid flow through the opened valve. With the valve in an open position, the differential pressure acting across the valve disc is less than the applied pressure, because of flow across the face and around the edges of the open disc. The purpose of the analysis is to take credit for the reduction of static pressure across the valve disc as a consequence of flow.

Hydrodynamic torque reduction is estimated using the following procedure:

1) A linear analysis for the flow field on either side of an arbitrarily moving disc permits the solution for the local pressure and velocity in the vicinity of the valve disc.

2) The flow is modeled as a mathematical combination of sources and sinks around the circumference of the open disc, with the local pressure obtained in step 1 used to evaluate the strength of the sources and sinks.

3) The complete response of the valve to this resulting flow and to the applied differential pressure obtained from the vent dynamic model is then calculated. In all cases, the inclusion of the hydrodynamic torque tends to reduce the actual differential pressure and hence load acting on the valve disc.

Comparison of the valve dynamic model with Mark I FSTF test data from blowdown SDA allows validation of the valve dynamic model (Ref. 6) since both valve disc displacement and differential pressure across the valve disc were measured. Results from Ref. 6 demonstrate a conservatism of over 12% in maximum predicted impact velocity and a slope of 1.39 from a least squares fit of the measured and predicted data to a straight line, with a Bernoulli torque factor of 2.25 (Figure 3-1). By increasing the Bernoulli torque factor slightly, to 2.55, the conservatism in maximum predicted impact velocity is reduced to 5% and the least squares fit slope is reduced to 1.24 (Figure 3-2). The larger value of Bernoulli torque factor is used in the Duane Arnold application.

The characteristics of the VB valve in Duane Arnold are given in Table 3-1. An application of the valve dynamic model with these characteristics and the differential pressure forcing function determined in Section 2 results in the computed valve response shown in Figure 3-3 for valve disc angle and Figure 3-4 for valve disc velocity. A summary of results appears in Table 3-2.

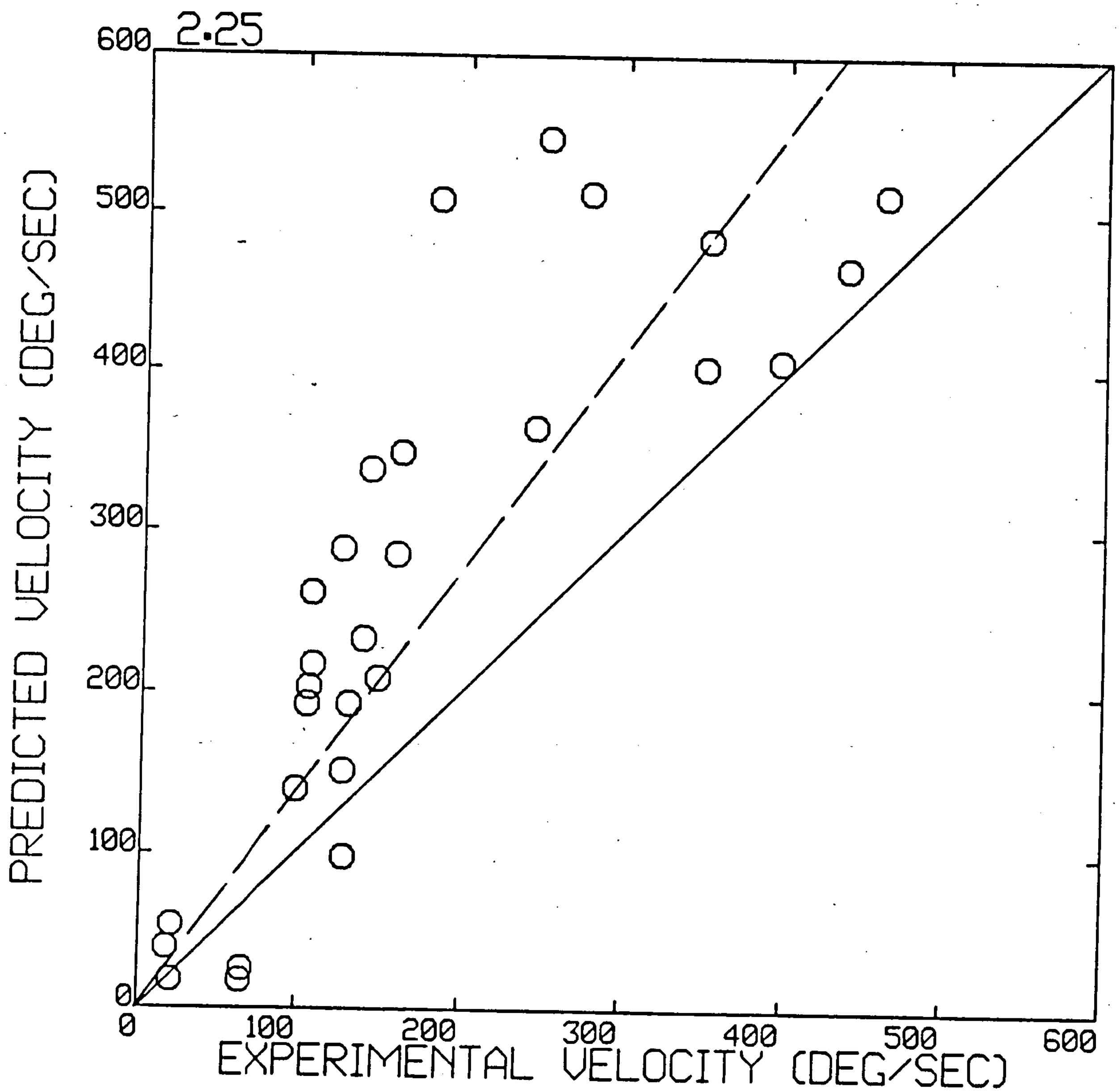

Figure 3-1. Comparison of experimental and predicted closing impact velocities for run SDA, Bernoulli torque factor = 2.25 (Ref. 6). The solid line is for minimized conservatism; the dashed line is the least squares linear fit.

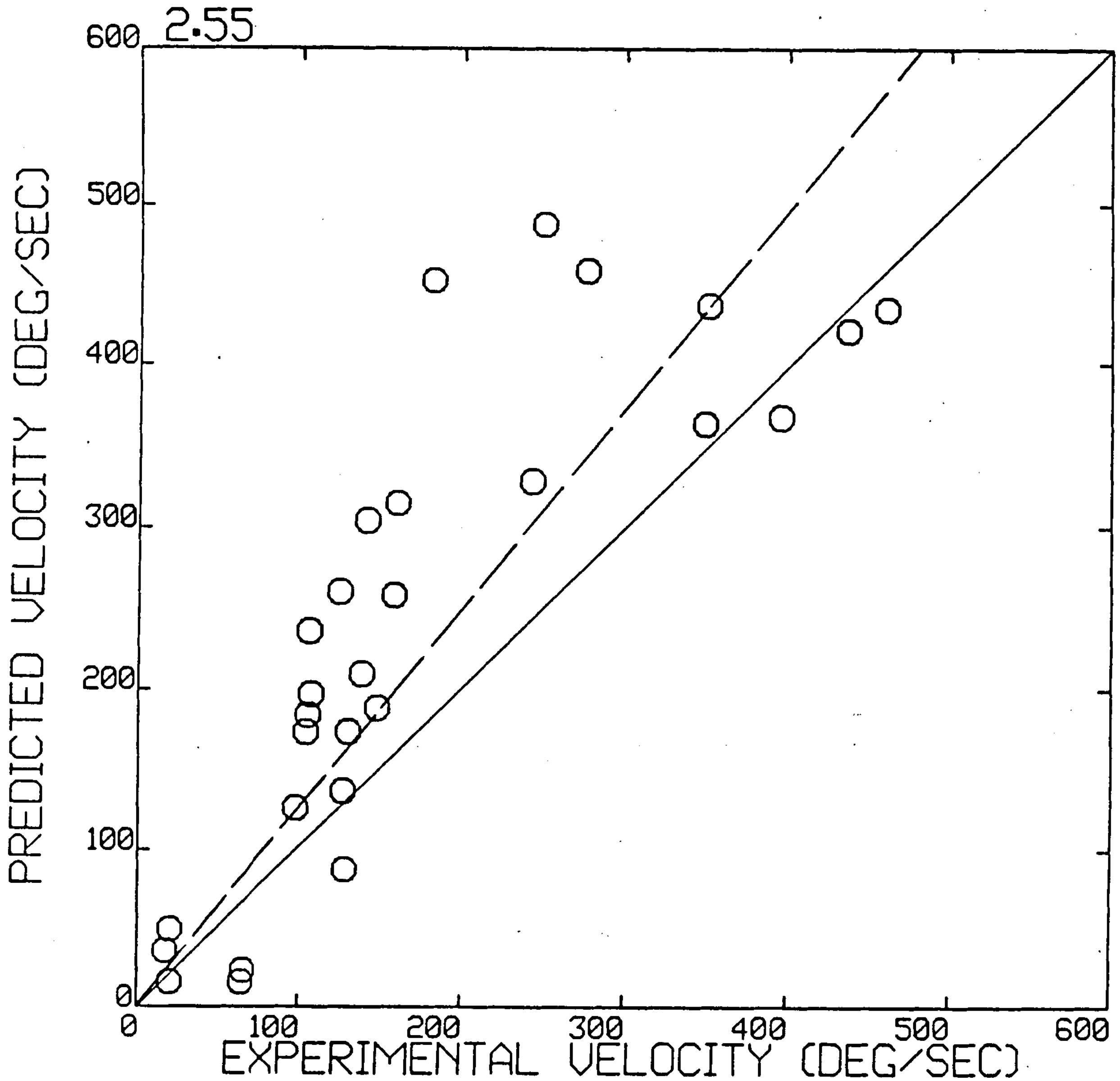

Figure 3-2. Comparison of experimental and predicted closing impact velocities for run SDA, Bernoulli torque factor = 2.55. The solid line is for minimized conservatism; the dashed line is the least squares linear fit.

TABLE 3-1

Vacuum Breaker Valve Characteristics
for Duane Arnold Energy Center Unit 1

Vacuum breaker type	18" GPE internal	
System moment of inertia	20.08	in-lb-sec^2
System weight	49.84	lb
System moment arm	10.85	in
Disc moment arm	11.47	in
Disc area	375.83	in^2
System rest angle	0.0	rad
Seat angle	0.07	rad
Body angle	1.39	rad
Seat coefficient of restitution	0.6	
Body coefficient of restitution	0.6	
Magnetic latch set pressure	0.25	psid

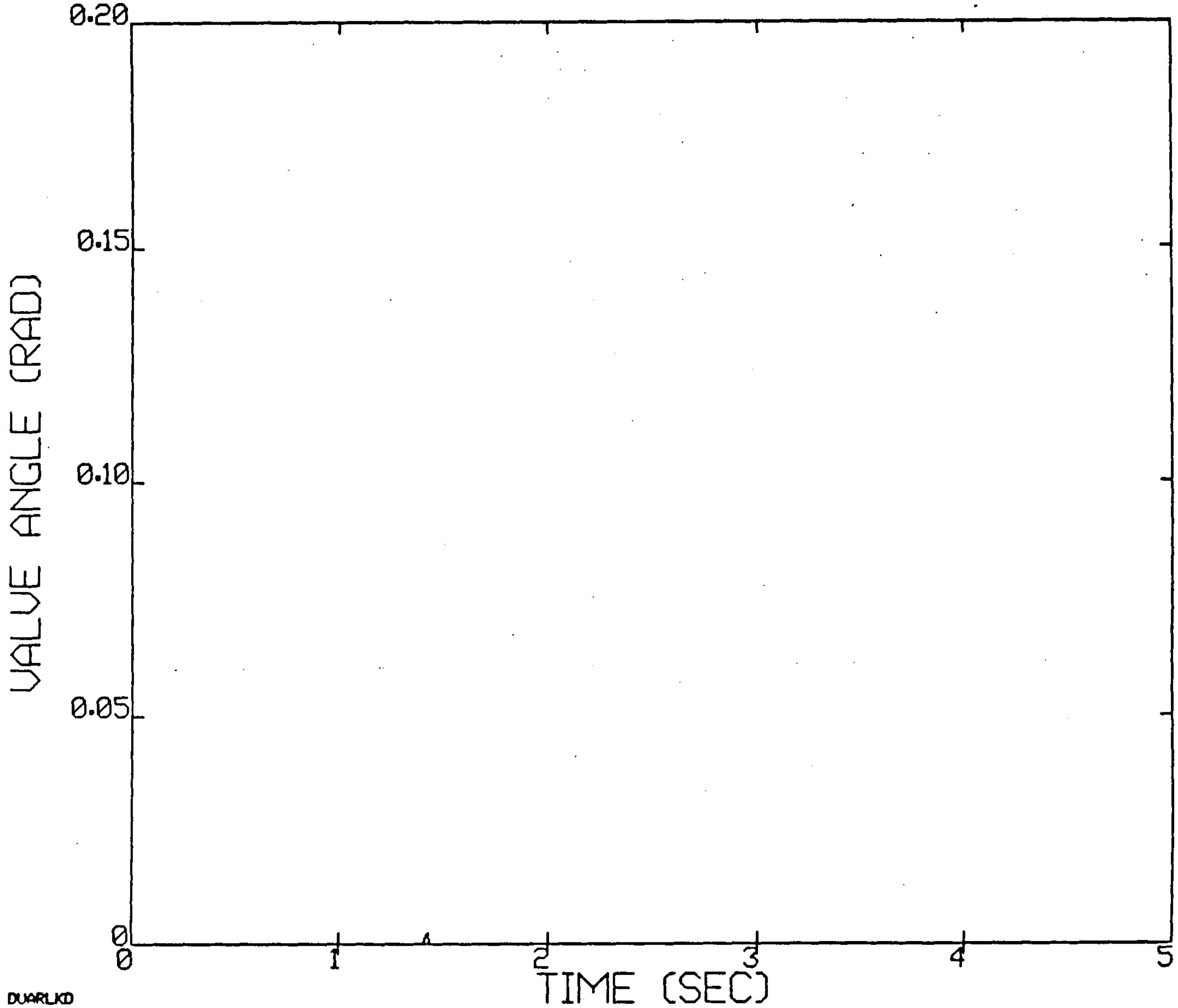

Figure 3-3. Valve angle response to the differential pressure time history shown in Figure 2-5 for the vacuum breaker in Duane Arnold. a) 0 - 5 seconds.

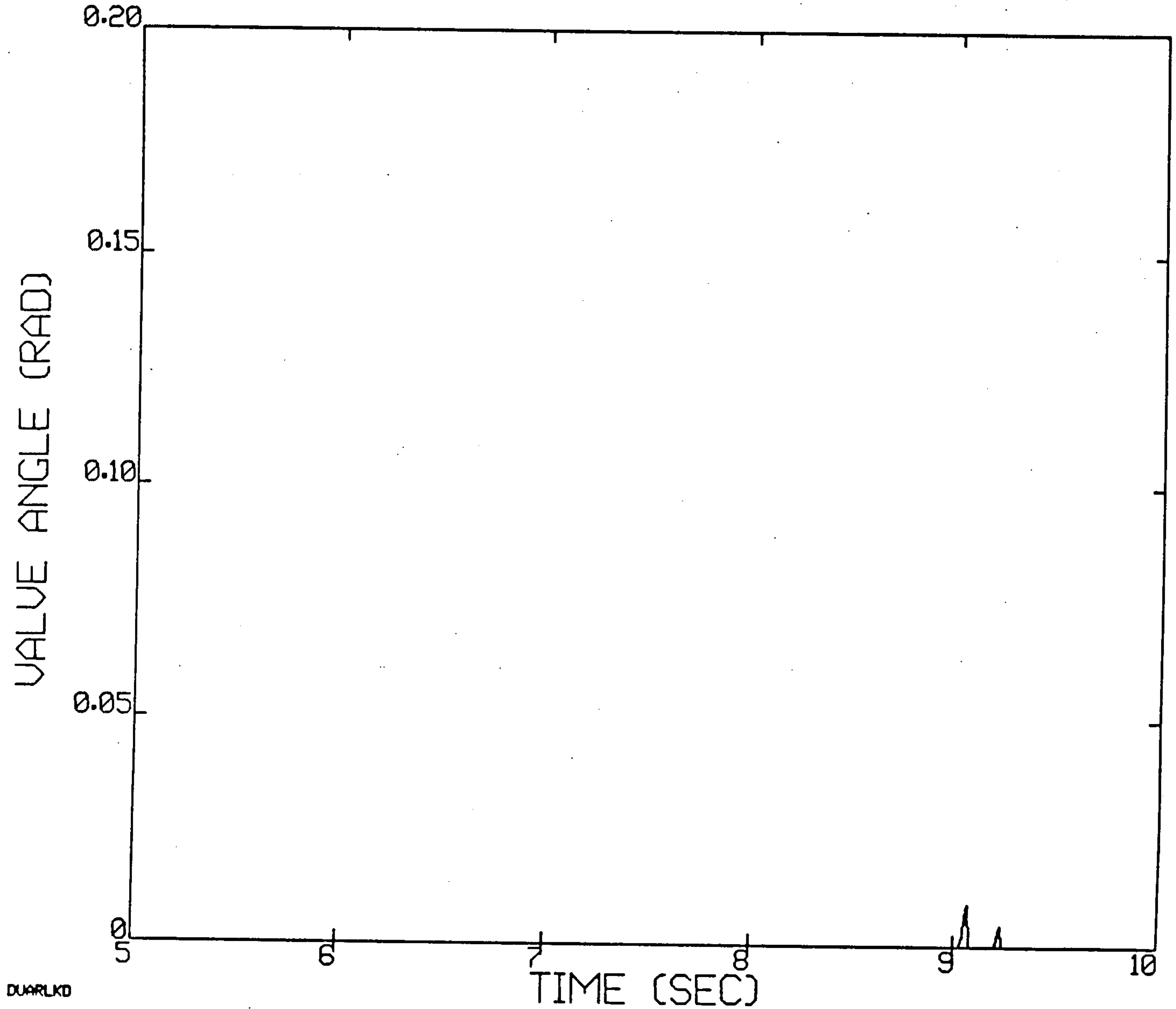

Figure 3-3b. 5 - 10 seconds.

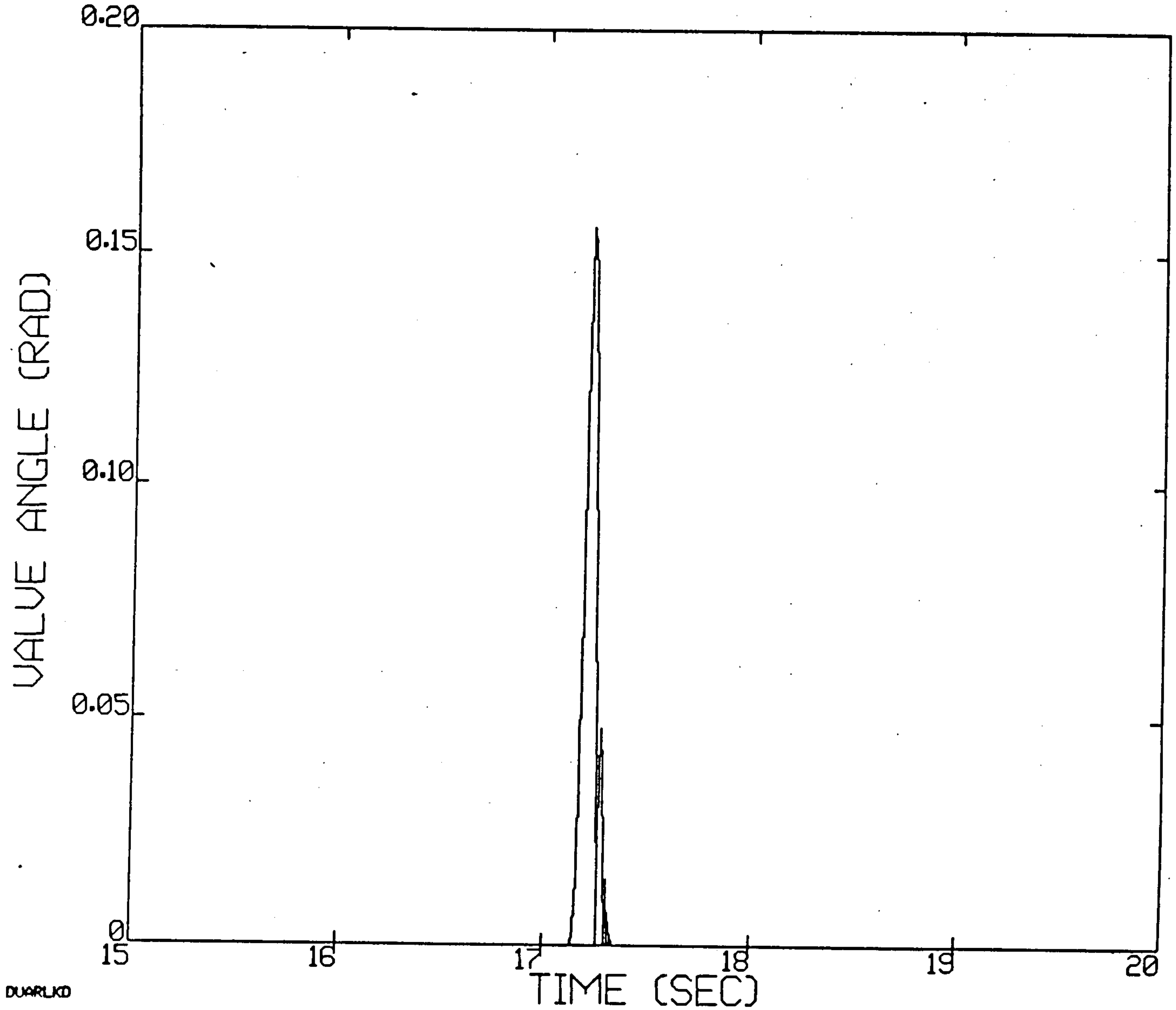

Figure 3-3c. 15 - 20 seconds (no response between 10 - 15 seconds).

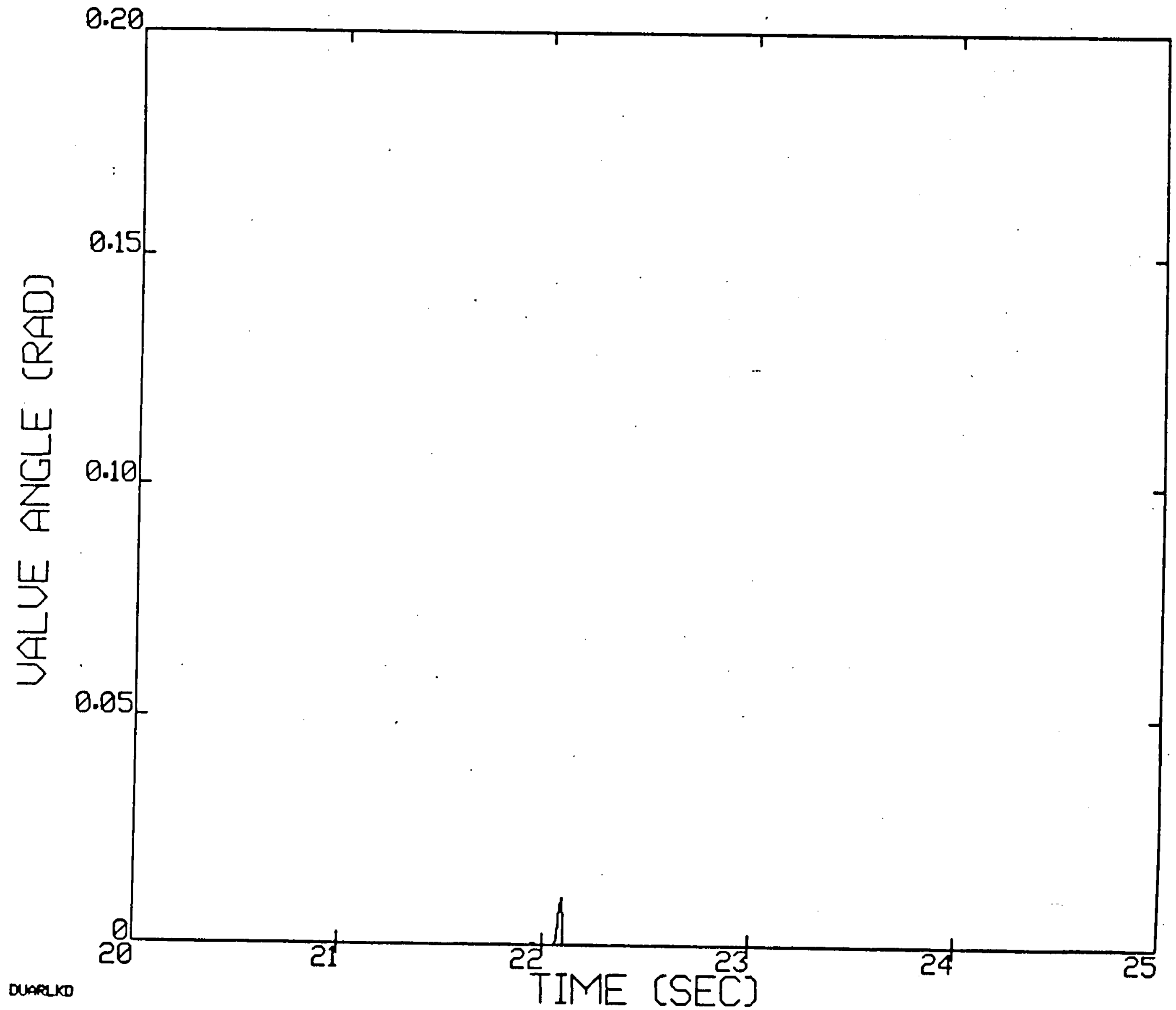

Figure 3-3d. 20 - 25 seconds.

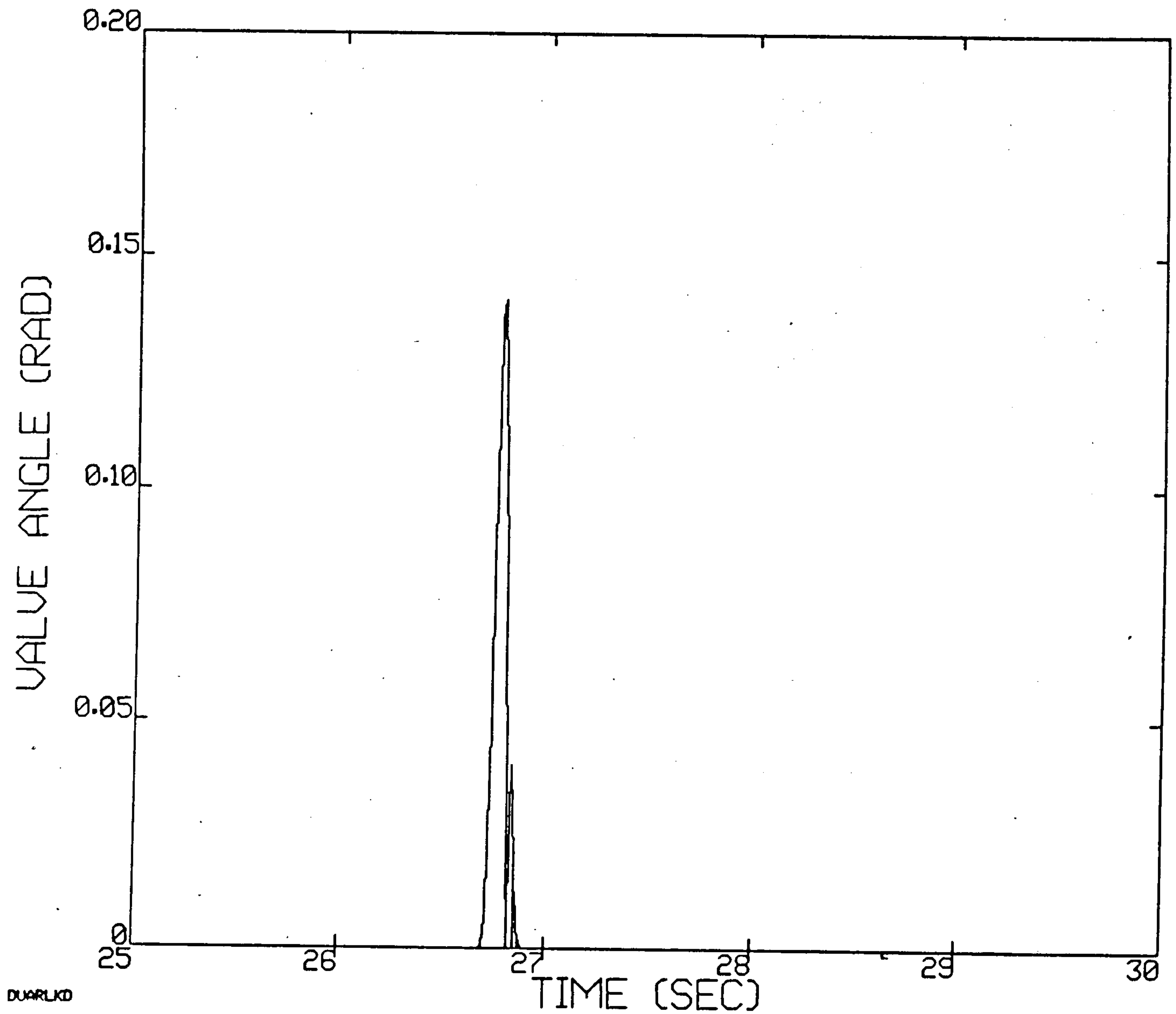

Figure 3-3e. 25 - 30 seconds.

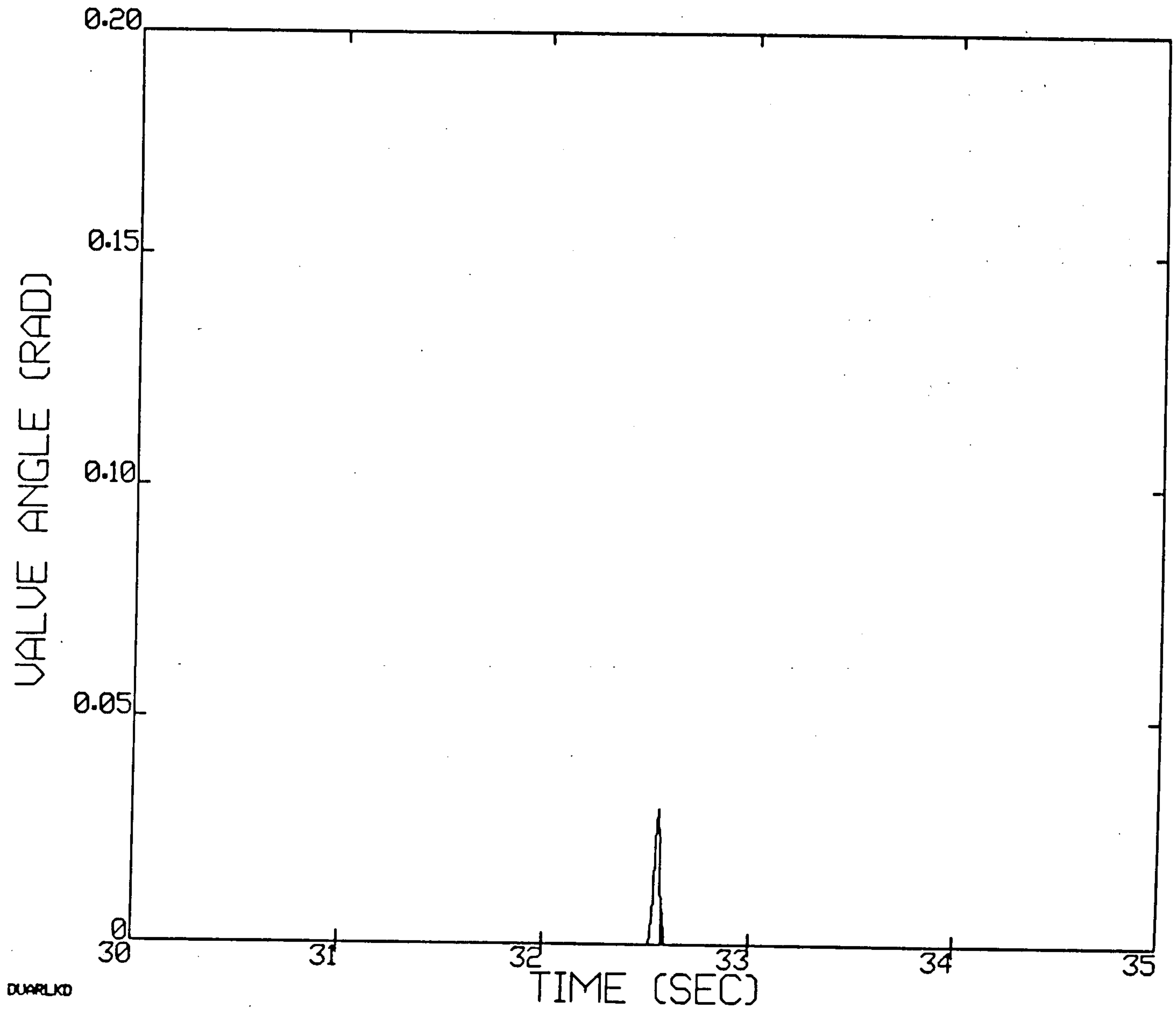

Figure 3-3f. 30 - 35 seconds (no response above 35 seconds).

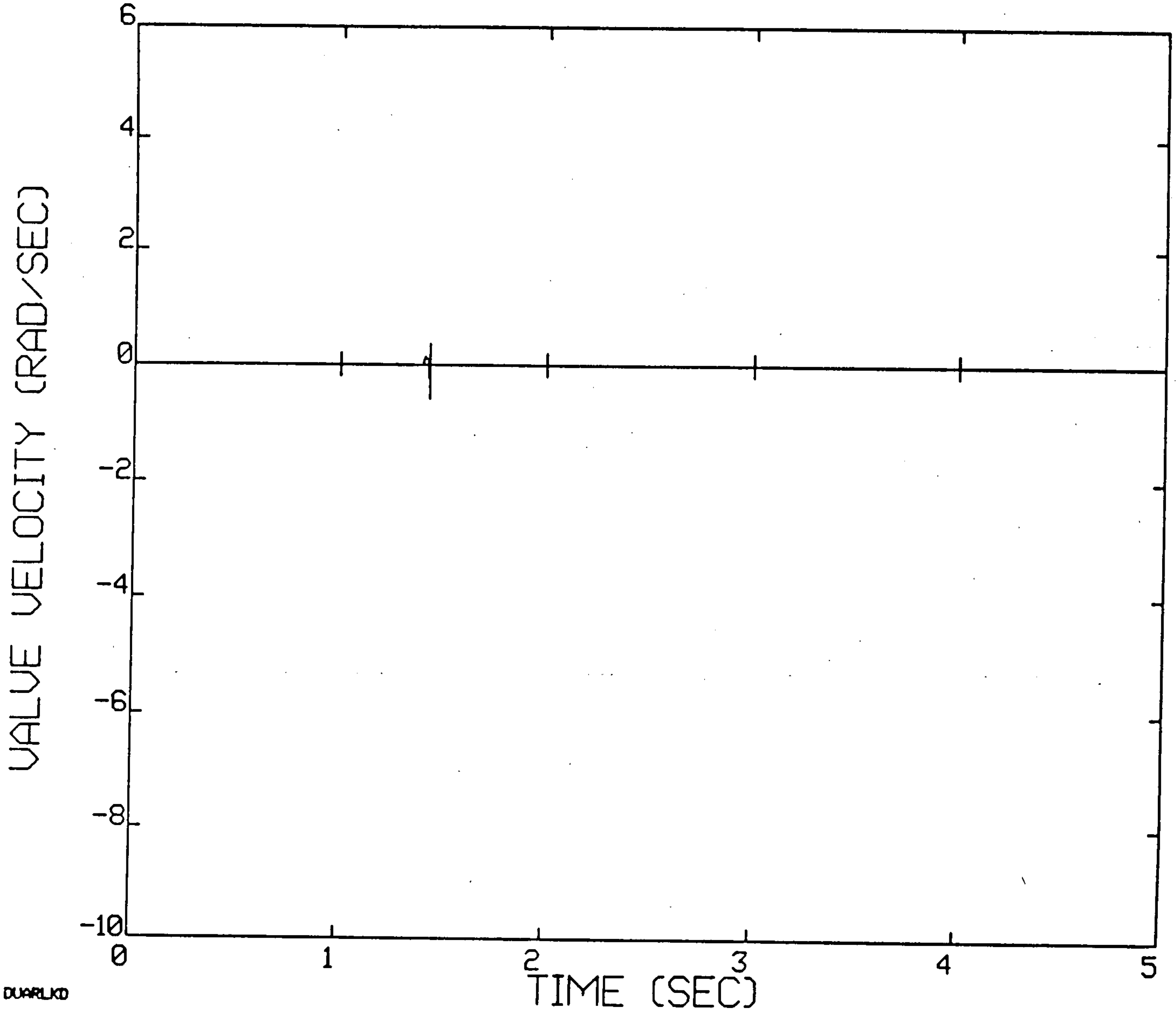

Figure 3-4. Valve velocity response to the differential pressure time history shown in Figure 2-5 for the vacuum breaker in Duane Arnold. a) 0 - 5 seconds.

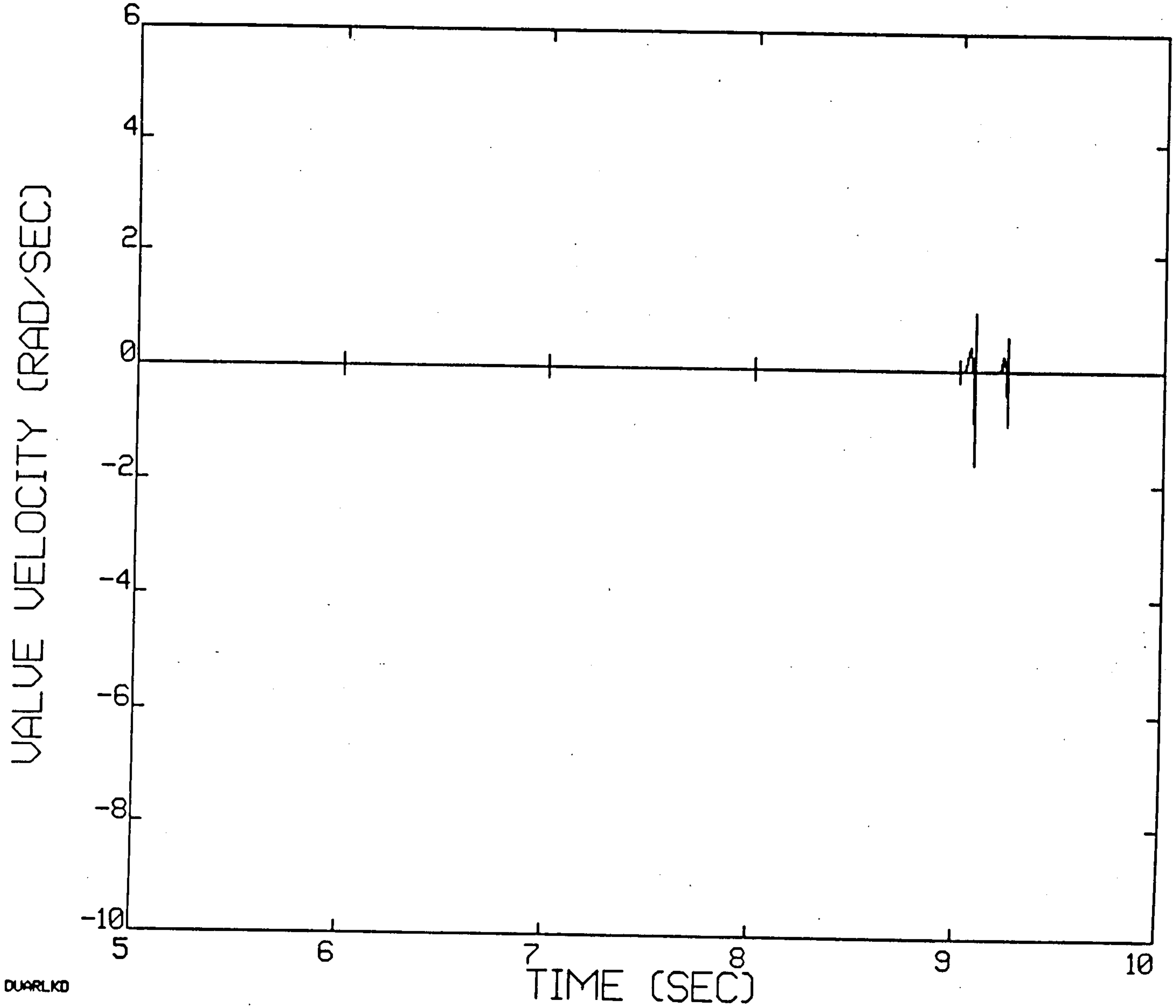

Figure 3-4b. 5 - 10 seconds.

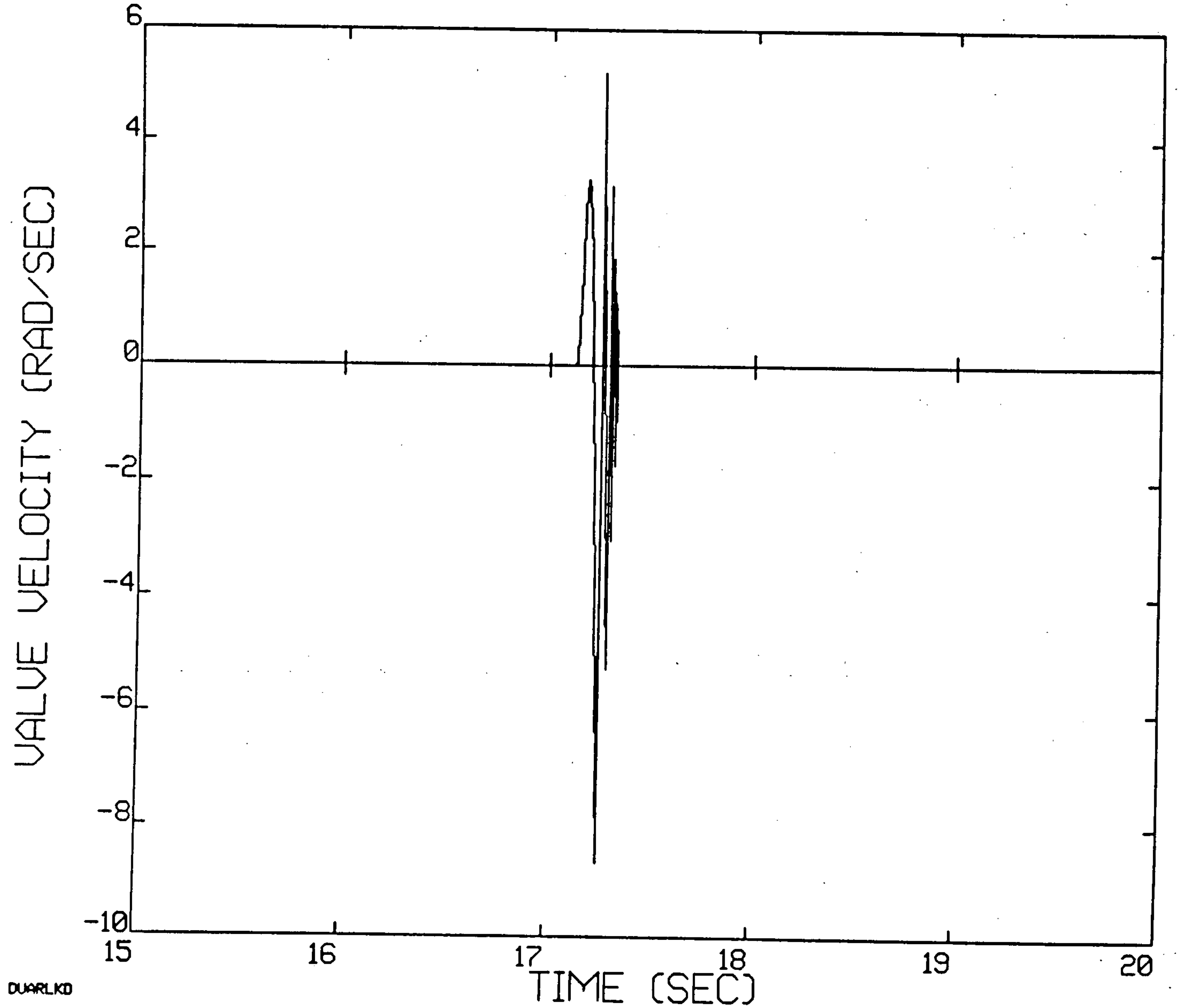

Figure 3-4c. 15 - 20 seconds.

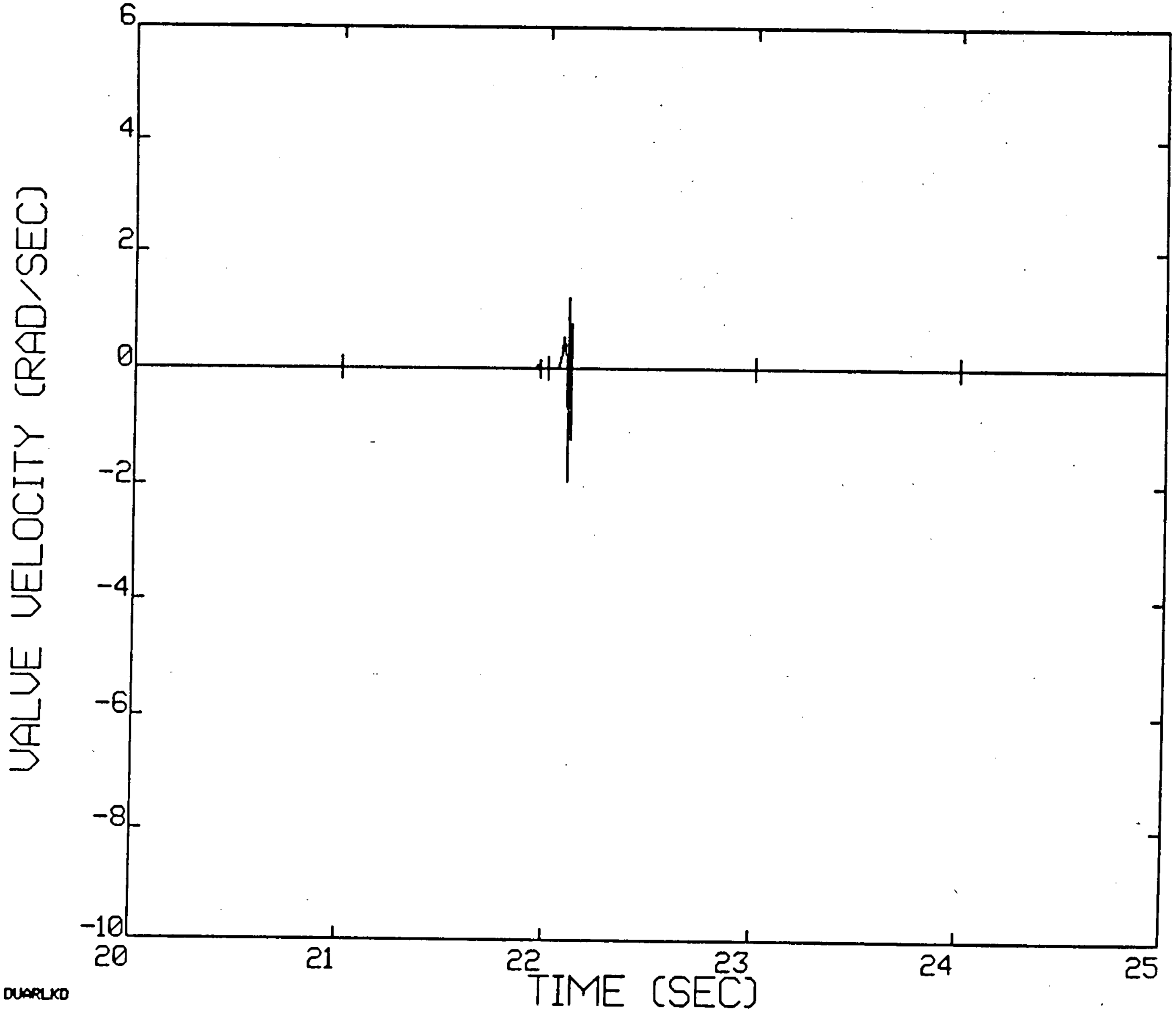

Figure 3-4d. 20 - 25 seconds.

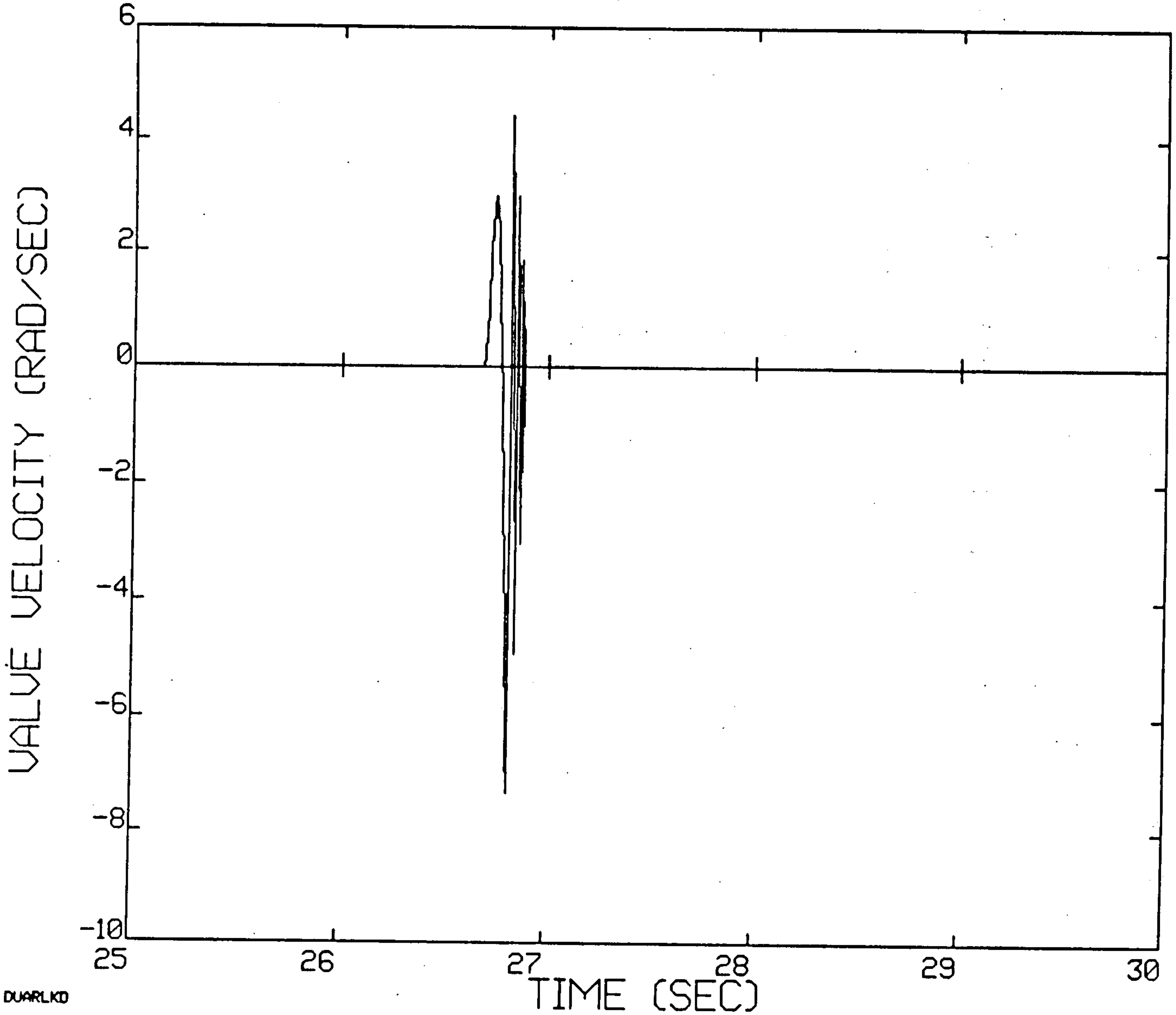

Figure 3-4e. 25 - 30 seconds.

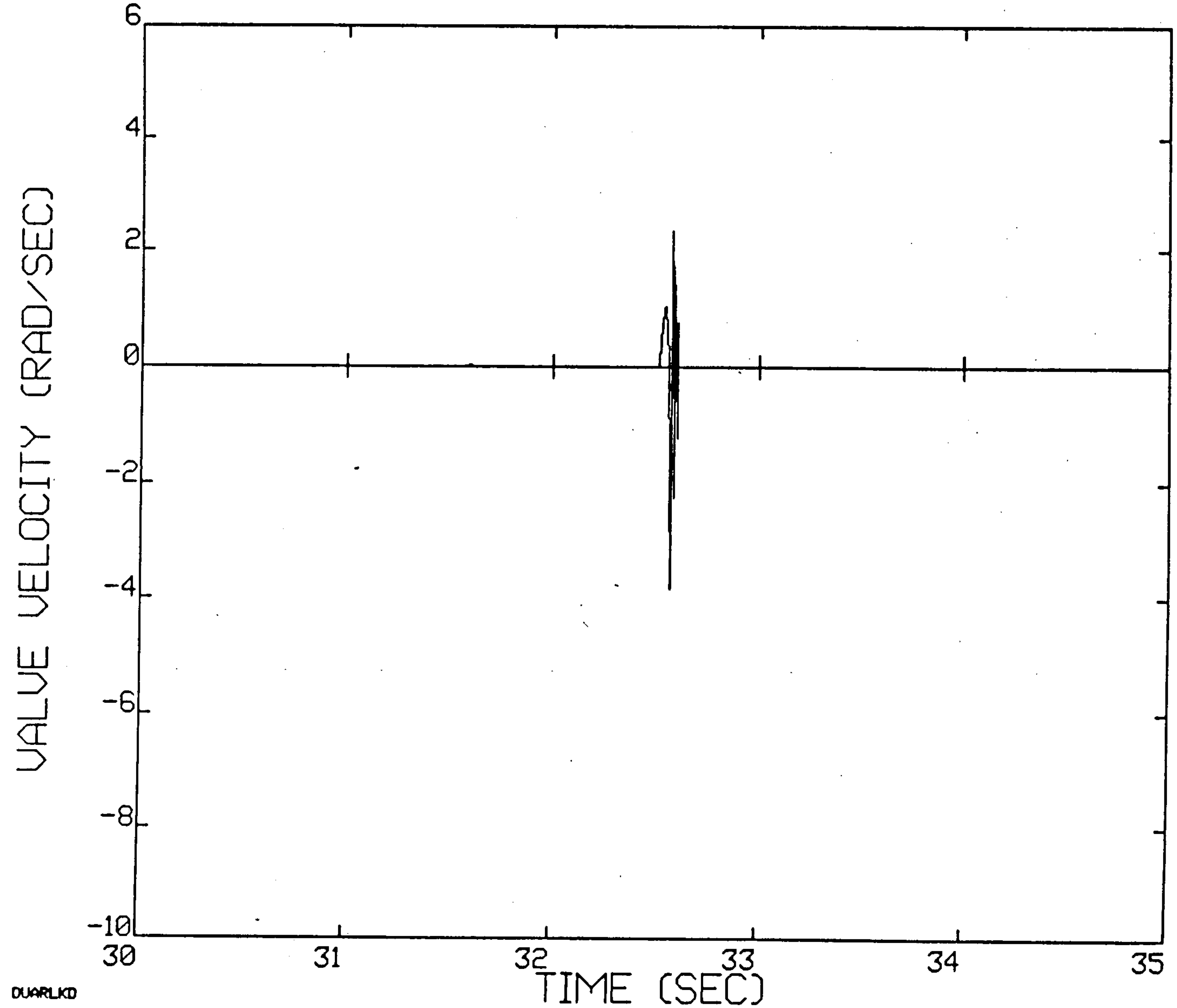

Figure 3-4f. 30 - 35 seconds.

TABLE 3-2

Vacuum Breaker Valve Response
for Duane Arnold Energy Center Unit 1

Maximum closing impact velocity	8.68	rad/sec
Maximum opening angle	0.156	rad
Number of closing impacts above 1 rad/sec	15	

4. REFERENCES

1. Safety Evaluation by the Office of Nuclear Reactor Regulation on the Acceptability of the Analytical Model for Predicting Valve Dynamics, issued by F. Eltawila, NRC, December 24, 1984.

2. "Mark I Containment Program; Mark I Wetwell to Drywell Vacuum Breaker Functional Requirements," General Electric Company Report No. NEDE 24802, April 1980.

3. "Mark I Wetwell to Drywell Vacuum Breaker Load Methodology, Revision 0," Continuum Dynamics, Inc. Report No. 84-3, February 1984.

4. "Mark I Experimental Determination of External Line Losses for Definition of External Vacuum Breaker Loads, Revision 2," Continuum Dynamics, Inc. Report No. 81-2, September 1984.

5. "Responses to NRC Request for Additional Information on Mark I Containment Program Wetwell to Drywell Vacuum Breaker Load Methodology, Revision 0," Continuum Dynamics, Inc. Technical Note No. 84-11, October 1984.

6. "Mark I Vacuum Breaker Improved Valve Dynamic Model, Revision 0," Continuum Dynamics, Inc. Technical Note No. 82-31, August 1982.