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Vibration and Noise Control of Reciprocating Compressors and Their Application

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APPLICATION OF NOISE SOURCE IDENTIFICATION TECHNIQUES
AND RESULTANT MODIFICATIONS ON A RECIPROCATING
COMPRESSOR

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INTRODUCTION

In recent years more emphasis has been placed upon noise reduction at the source of compressor noise rather than the use of more conventional noise control techniques.

Conventional noise treatment wisdom usually calls for controlling vibration of the outer structure of the machine either by means of stiffness or damping treatments. Such approaches are justified while dealing with, for example, hermetic types of compressors. These are characterized by a fairly light structural outer shell. Usually these kinds of treatment only reach the prototype stage and very seldom become applied in production. Prohibitive production costs and durability problems generally outweigh the sound quality improvement. Another conventional approach might involve the application of an acoustical enclosure. This offers a quick solution to particular problems of compressor application in air conditioning units. The main disadvantages lie in the inconsistency of noise suppression performance caused by open side leakage, the blanket coupling with the compressor, and high material cost. The resolution of these problems are largely beyond the control of the compressor manufacturer. In these cases of noise control, the nature of the noise source is not essential in treating the structure's response, which is one of the major advantages.

When dealing with noise control of semi-hermetic, reciprocating compressors or new design concepts of compressors having significantly different outer

structures and having different methods of coupling the compressor to the outer structure, additional approaches to noise control are advisable. As an example, typical semi-hermetic compressors already display very high structural stiffness and are very massive. In such a case, stiffness and damping control are not practical. Also, the never ending search for further improvement of performance and reduction of production cost in most cases results in higher noise levels. This is further aggravated by strict noise emission requirements of the European and Japanese markets. In view of all these factors, it becomes evident that different approaches to the solution of noise problems are needed.

The most obvious approach would be to treat the fundamental noise source itself. This concept is even more desirable when a new product is being developed since fewer restrictions are imposed on the noise control engineer.

The first major step from a noise control view is to identify the major forcing mechanisms and their contribution to noise emission. Then, by utilizing detailed knowledge of the experimental techniques leading to the identification of structural part responses, the basic cycle of noise identification is completed. The main emphasis of this paper will be the identification techniques themselves and source modification rather than the structural response aspects of control. More specifically, we will examine the usefulness of time domain investigation in view of the working character of reciprocating compressors or any compressor displaying repetitive signals of the nonstationary character. The key aspects of this technique will be presented in more detail with examples and validity checks of measurements and their interpretation. The technique of utilizing time domain measurements was used in several cases of noise control projects during the past four years. However, this process will be illustrated on a single noise control project involving a semi-hermetic compressor with the discus type discharge valve.

DESCRIPTION OF MEASUREMENT HARDWARE AND SETUP

The compressor used in the noise reduction testing was an in-line, three cylinder, semi-hermetic type driven by an overhung, four-pole, 15 horsepower, induction motor. A cross-sectional view of the compressor is shown in Figure 1a. Figure 1b is a perspective view of the standard valve design as it is

attached to the valve plate. As shown, the disc type valve is positioned in the closed position, seated against the valve plate. Three curved washer type springs provide the valve with a preload force. Discharge plenum pressure acts on the spring side of the valve through the hole in the center of the retainer. As the valve moves to the open position, it is guided on its outside diameter by the retainer. When the valve opens, the gas in the cylinder escapes past the valve and retainer, flowing into the discharge plenum.

A timing mark was generated using a rotary incremental, optical encoder coupled directly to the crankshaft of the test compressor. The encoder provides a reference transistor-transistor logic (TTL) level pulse once every crank revolution as well as 720 TTL level pulses every crank revolution. While operating the compressor at the test conditions, a proximity probe monitored the piston's position within 0.050 inch of TDC for the test (#1) cylinder. The reference pulse from the encoder was aligned with the proximity probe's dynamic measure of actual TDC. Therefore, the data gathered was accurately referenced to TDC in the time domain history of either a Bruel & Kjaer 2032 two-channel Fast Fourier Transform (FFT) analyzer or a Norland 3001 four-channel digital storage oscilloscope. Also, the 720 TTL level pulses were used to directly time the analog to digital (A to D) circuit in the oscilloscope causing data to be accurately sampled every 1/2 degree of crank rotation.

Knowing the pressure history in the cylinder and immediately behind the discharge valve allows for the direct measure of the acting pressure differential on the valve. Cylinder pressure was monitored using a piezoelectric type pressure transducer mounted in a pressure tap sensing pressure at the edge of the cylinder at TDC. Also, the absolute pressure at bottom-dead-center (BDC) was measured using a strain gage type pressure transducer. This transducer was mounted in a pressure tap exposed to cylinder pressure only when the piston was within 0.090 inch of BDC. Otherwise, the tap was exposed to the crankcase (suction) pressure. Consequently, the strain gage pressure transducer was not subjected to large pressure or temperature variations thereby providing a reference at BDC for the dynamic pressure signal from the transducer measuring cylinder pressure. A piezoelectric pressure transducer was used to sense the dynamic pressure in the plenum immediately behind the discharge valve (retainer pressure). This pressure signal was referenced to the static pressure measured in the discharge plenum. By using post

processing techniques, the actual pressure differential acting on the discharge valve could then be determined.

Valve displacement was monitored by relating the bending stress at the center of the discharge spring to the displacement of the valve. From the fully closed position to approximately 80% of the fully open position, this relationship was linear. During the last 20% of the displacement, this relationship became increasingly non-linear. Also, impact with the retainer (valve stop) caused perturbations in the strain gage signal. Computerized post processing of the strain signal was required to relate the dynamic strain to the actual displacement.

Piezoelectric accelerometers were used to monitor the impact and vibration events occurring on the valve plate, retainer, head, and crankcase. Prior testing on other types of compressors revealed that these four locations were most important in identifying impact events in the compressor. The valve plate accelerometer was mounted on the top side of the plate halfway between the #1 and the #2 cylinder on an axis parallel to the center line of the crankshaft. The retainer accelerometer was mounted to the top side of the #1 cylinder's retainer on a diameter equivalent to that of the discharge valve. The head accelerometer was mounted on the top of the discharge head near the oil pump side edge on an axis parallel to the center line of the crankshaft. The crankcase accelerometer was mounted on the side opposite the discharge service valve at the point perpendicular to the intersection of the center line of the crankshaft and the center line of the #2 cylinder.

TEST RESULTS

Initial noise testing of the compressor in the reverberant room resulted in a sound power level of 86 dBA. The spectra itself displayed relatively flat response across the frequency range of interest (Figure #2). Further testing of vibration at different locations on the compressor structure and preliminary experimental modal analysis revealed that the low frequency vibration around 500-600 Hz was associated with rigid body motion of the compressor supported on helical springs. The peak centered around 1600 Hz was related to the local resonance of the side of the crankcase. The high frequency range represents typical structural response of highly stressed structures. Later, measurement of driving point accelerance of different major components also

displayed increases in acceleration and the number of modes as the frequency increased.

The trace of cylinder pressure, retainer cavity pressure, and discharge valve displacement in the time domain are shown in Figure #3. The purpose of this measurement was to establish the sequencing of work cycle events as related to the timing. Four major timing events of the discharge valve are marked by vertical lines A, B, C, and D. They are respectively the beginning of discharge valve opening, full valve opening, start of valve closing, and again valve fully closed.

Since the head acceleration was a suitable measurement location for frequencies related to the rigid body motion, it was used to determine if there was any transient event associated with their source. Figures #4 and #5 show the results of partial windowing of the time domain vibration signal as

Appendix for a detailed explain

By extending the length of the window past the second cylinder discharge valve opening, strong frequency components around 600 Hz are evident (Figure #5). Time enhancement was then done off the top-dead-center of cylinder #1. This also strongly indicated a low frequency occurrence near the time of second cylinder discharge (Figure #6). The presence and timing of these low frequencies was found to correspond with the discharge valve opening of the second cylinder. Figure #7a shows the retainer vibration response as measured in the vertical direction. There was indication of two distinguishable impulses correlating with full opening and closing of the discharge valve. Windowing the second impact revealed that we were dealing with retainer response. Later modal analysis confirmed that the bending mode of the retainer was responsible. It was also noticed that the first impact contained far less vibrational energy than the closing impact (Figure #7b).

Vibration response measured on the side of the crankcase and valve plate shows a similar pattern. Three major impact responses related to compressor structural resonances were discovered timed to the discharge valve opening and closing (Figure #8). It was apparent that the noise problem of the compressor is mainly caused by valve impact as the major source of excitation of the structure.

Three major options were available for noise reduction. First, reduce the magnitude of the forcing mechanisms--valve impact; second, decouple the

transmission path between impactor location and the outer body casting; and last, stiffen the body casting structure. The last option did not appear feasible because the casting was already very stiff and noise spectra showed that we were not dealing with highly predominant resonances. Decoupling also appeared to be very difficult to accomplish in view of the sealing problem the high pressure head assembly presents.

Analytical study of the discus valve impact mechanism suggested that the main parameters governing impact magnitude were the velocity of the impacting valve, stiffness of the valve and valve seat, and the mass of the valve.

During this time period in the sound program, major modifications were made to the discus valve and retainer to satisfy other performance requirements. The new design resulted in a much lighter valve. Since it was theorized that the pressure differential acting across the valve was a governing factor in valve impact velocity, design changes modifying the flow character were also made. The final design was widened, to restrict the gas flow through the outside perimeter. Eight additional discharge holes were provided. Overall flow area was decreased. As part of this modification, the bending stiffness of the retainer bridge was increased too. This final version of the retainer is also shown in Figure #9.

Pressure measurements taken inside the cylinder and above the valve for the nonrestricted and the restricted cavity (with and without the eight holes) are shown in Figures #10 and #11. The lower portion of the graph also displays the vibration response of the retainer. There is evidence of vast improvement of vibration response on the valve opening impact, correlating with the pressure differential drop as compared to the nonrestrictive retainer. There is also a noticeable decrease in pressure pulsation inside the cylinder.

Figures #12, #13, and #14 show vibration responses of the valve retainer, valve plate, and side of the crankcase respectively, with the former style retainer design and the final concept of the restrictive type retainer. Total sound power dropped to 82.5 dBA as is seen in Figure #15.

OVERALL PERFORMANCE RESULTS

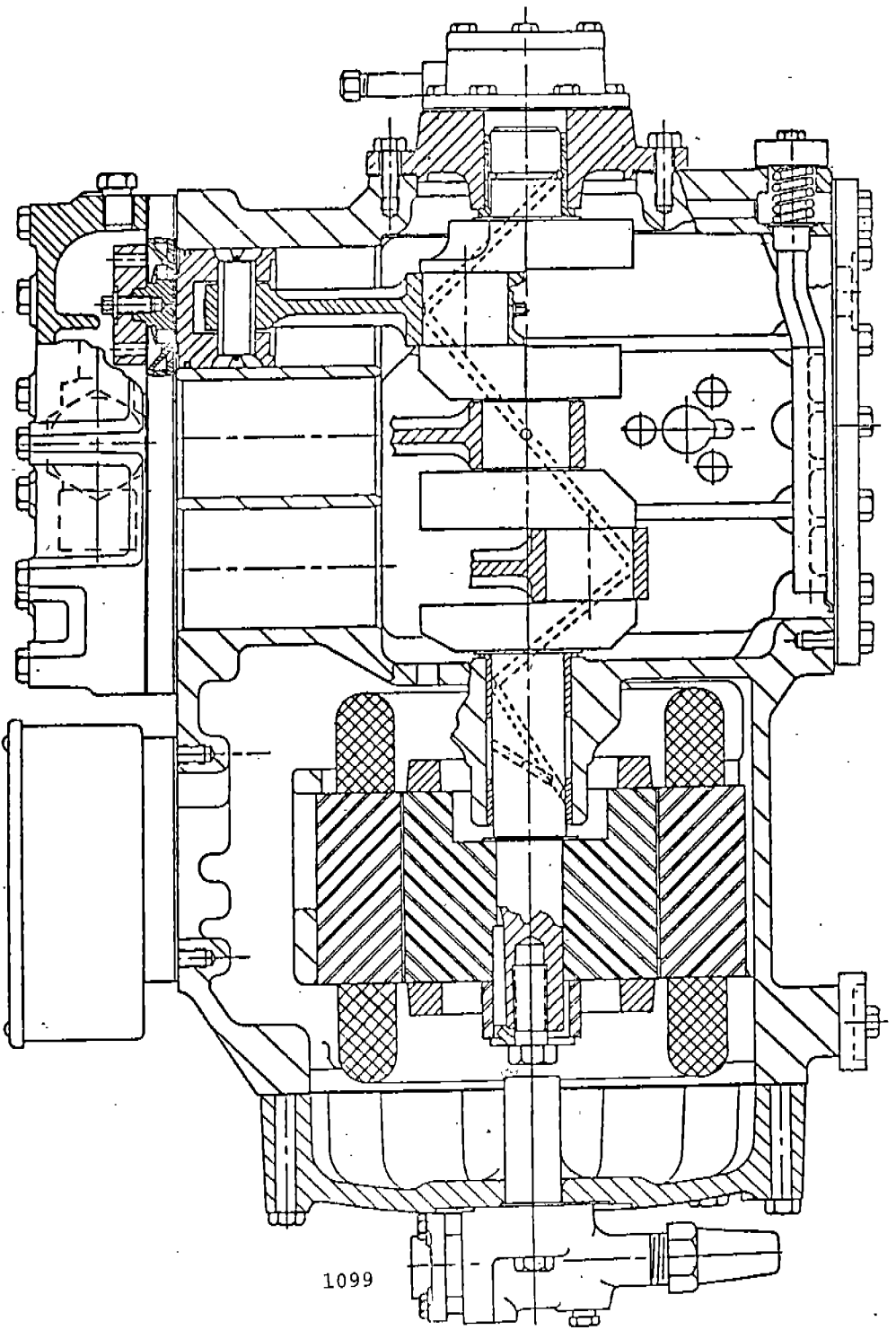
An important consideration in any redesign is the effect of that redesign on performance. For

refrigeration compressors, performance is evaluated by measuring the capacity (Btu/hr) and energy efficiency ratio (Btu/watt-hr) for a specified system operating condition. The test compressor compressed refrigerant R-502 from 26.8 psia to 262.6 psia with the inlet gas temperature at 65°F. This corresponds to a system condition of -25°F saturation temperature for the evaporator, 110°F saturation temperature for the condenser, 5°F subcooling of the liquid refrigerant after the condensing process and 90°F superheating of the refrigerant gas after the evaporation process. The ambient temperature in the test facility was 95°F. At these conditions, the test compressor operated with the standard valving at a capacity of 44,030 Btu/hr and an EER of 5.12 Btu/watt-hr. Installing the redesigned muffled type solid post valving actually increased the capacity to 44,930 Btu/hr and the EER to 5.20 Btu/watt-hr. Performance was positively affected by the valve redesign.

CONCLUSIONS

1. Discharge valve impact during opening and closing was identified as a major source of the compressor noise at higher frequencies.
2. Rigid body motion of the compressor in the low frequency range was associated with the process of abrupt discharge valve opening.
3. The treatment of the noise source itself was very effective in the noise reduction of the compressor. Modification of discharge valve flow mechanisms resulted in 4 dBA noise reduction.
4. The analysis of the vibration signal in the time domain was the major tool in identification of the noise source. The techniques of partial windowing, signal enhancement, and timing detection of major events were used.
5. The treatment of the main source of noise also had a positive effect on the overall performance of the compressor. EER was increased by 1.5% and capacity was increased by 2.0%.

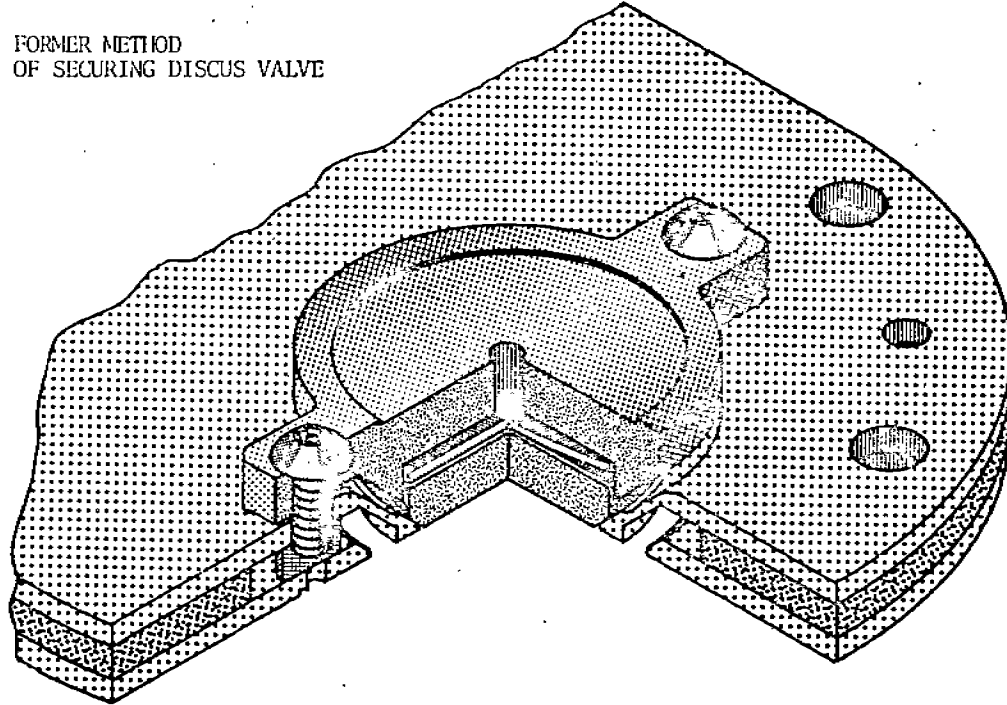
Figure 1a: Sectional View of Test Compressor



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Figure 1b: 3D View of Former Valve Assembly

FORMER METHOD
OF SECURING DISCUS VALVE



1100

Figure 2: Overall Sound Power Level of Original Compressor

3DS1-1500-TFD 84G-Q4069A 60HZ 460V R502 -25/110/65 0033-86-01 2-4-86 *
 OLD STANDARD DISCUS

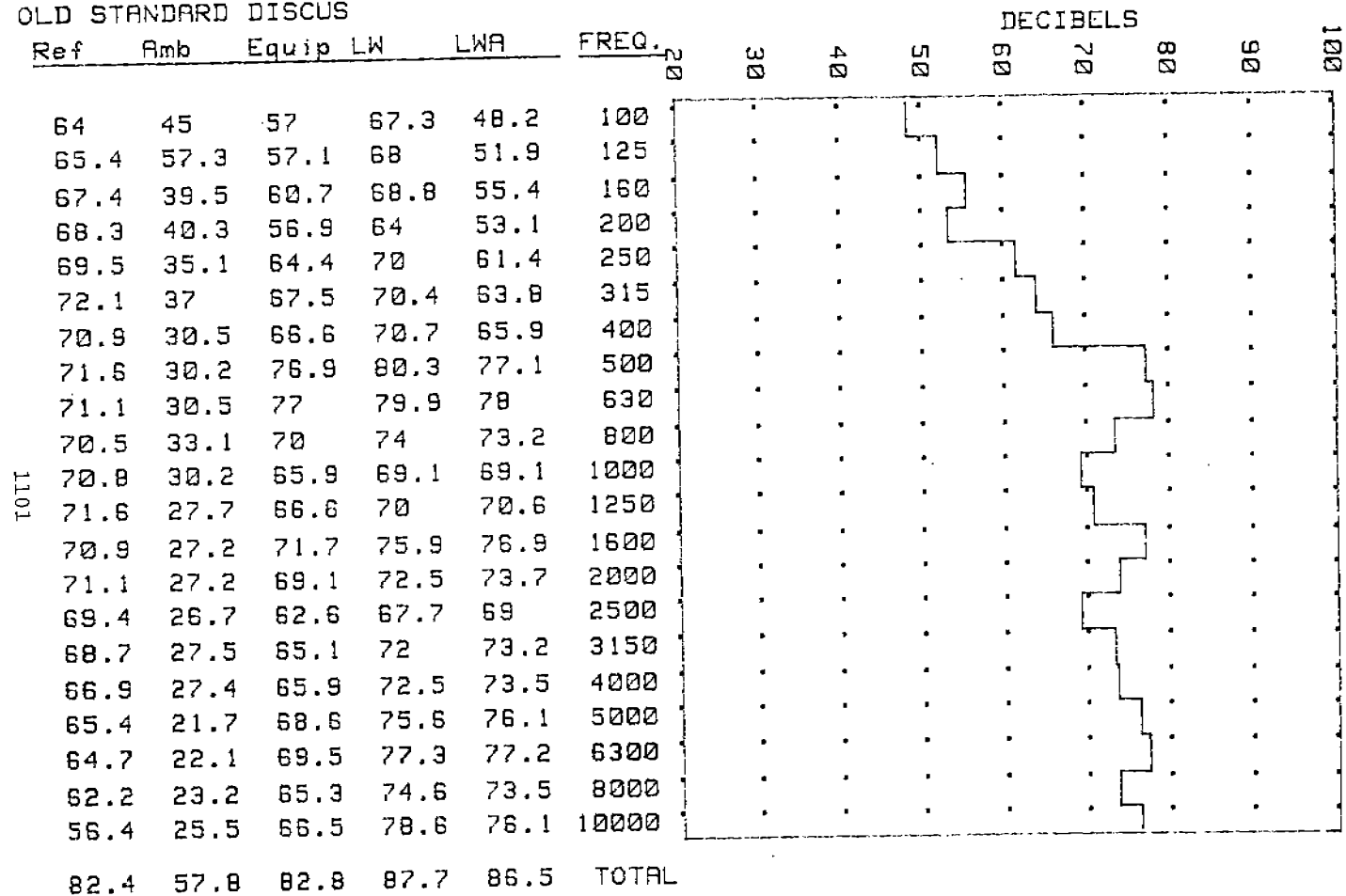


Figure 3: Discharge Valve Related Events - Original Design

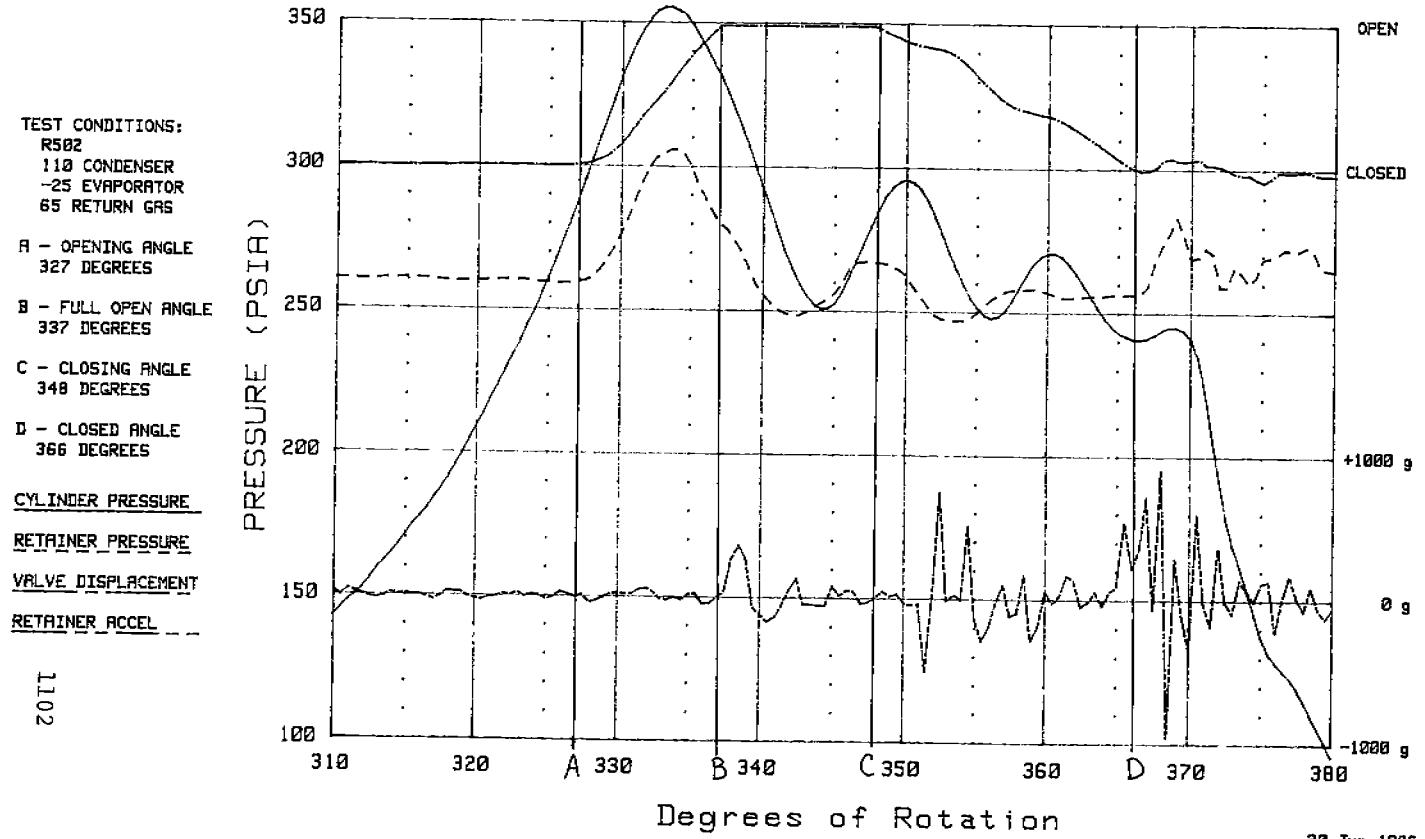
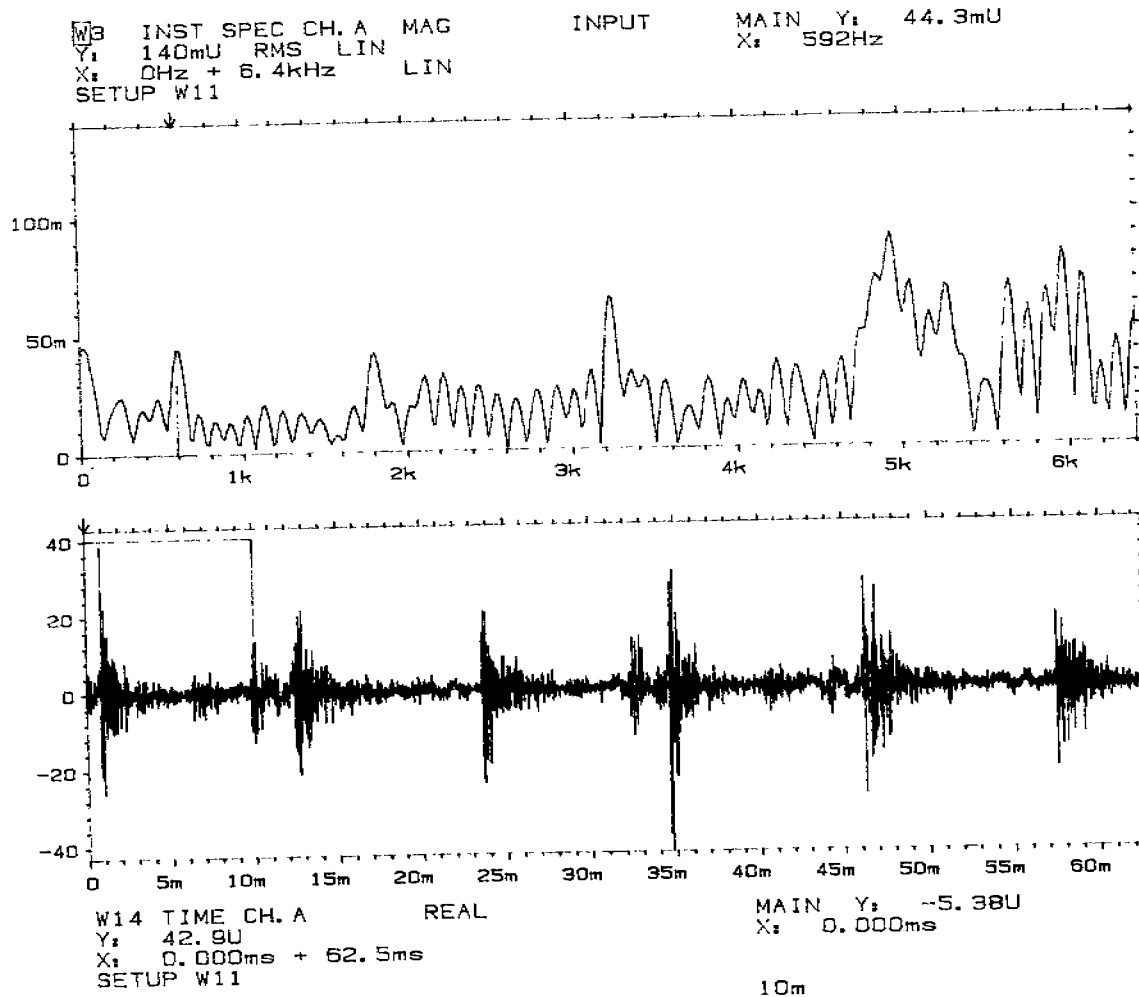


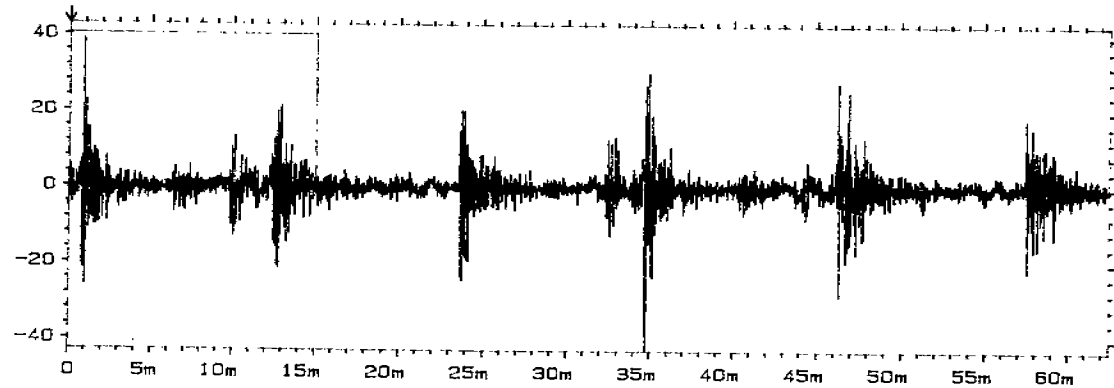
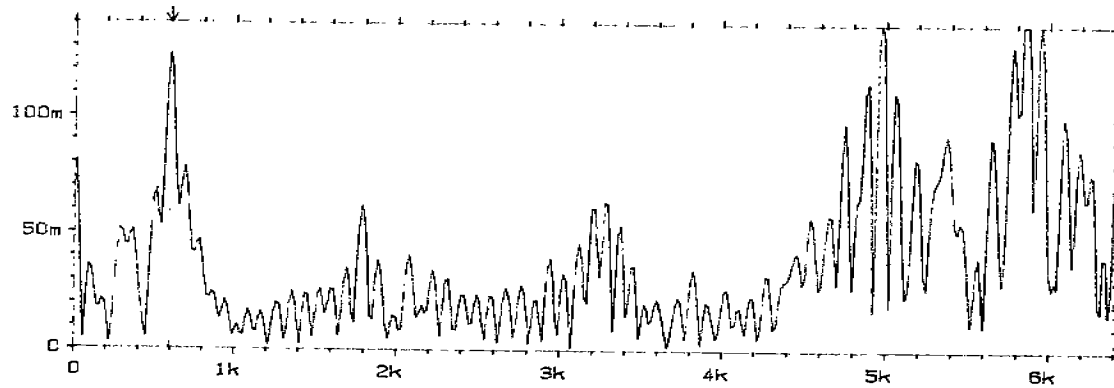
Figure 4: Partial Windowing of Acceleration Data - Top of the Head



1103

Figure 5: Longer Window of Data in Figure 4

W3 INST SPEC CH. A MAG INPUT MAIN Y: 126mU
Y: 140mU RMS LIN X: 592Hz
X: 0Hz + 6.4kHz LIN
SETUP W11

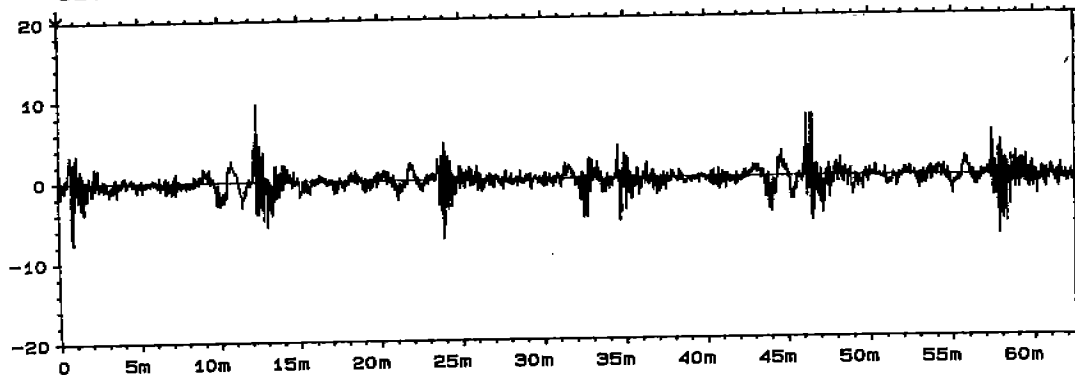


W14 TIME CH. A REAL MAIN Y: -5.38U
Y: 42.9U X: 0.000ms + 62.5ms
X: 0.000ms
SETUP W11

1104

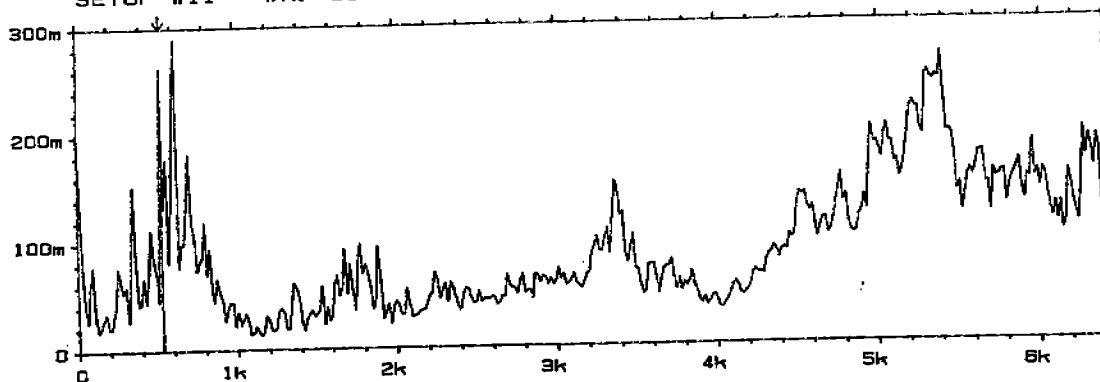
W3 ENH TIME CH.A REAL
Y: 20.0U
X: 0.000ms + 62.5ms
SETUP W11 #A: 30

MAIN Y: -1.79U
X: 0.000ms



W14 AUTO SPEC CH.A [] INPUT
Y: 300mU RMS LIN
X: 0Hz + 6.4kHz LIN
SETUP W11 #A: 30

MAIN Y: 263mU
X: 528Hz

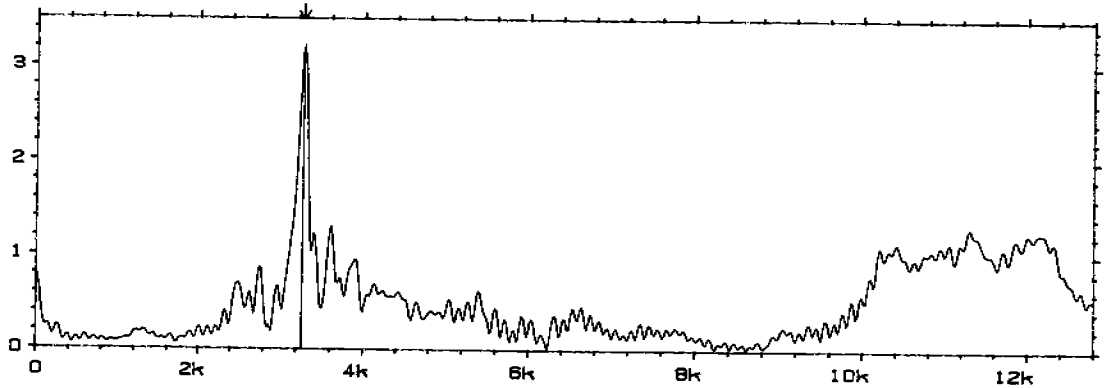
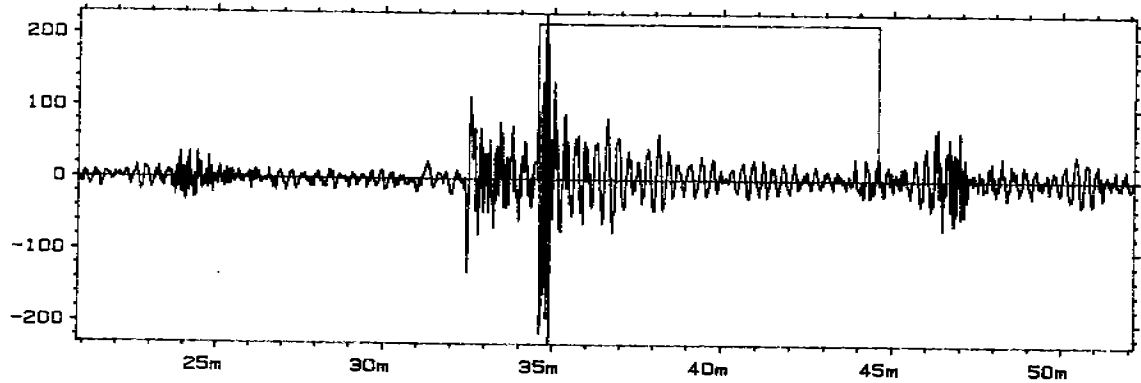


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Figure 6: Time enhancement of Acceleration - Top of the Head

Figure 7a: Partial Windowing of Impulse Response - Retainer

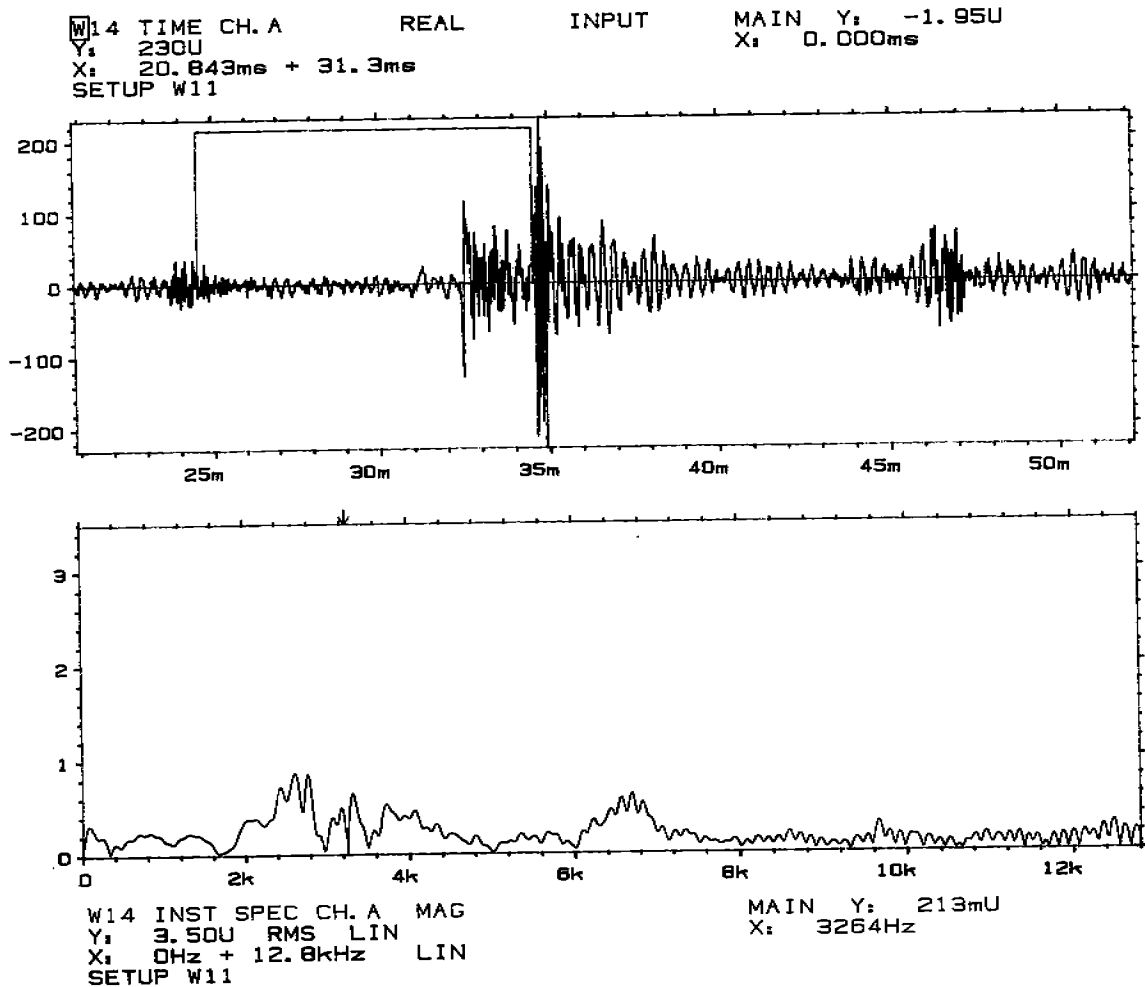
W14 TIME CH. A REAL INPUT MAIN Y: -1.95U
Y: 230U X: 0.000ms
X: 20.843ms + 31.3ms
SETUP W11



W14 INST SPEC CH. A MAG MAIN Y: 3.17U
Y: 3.50U RMS LIN X: 3264Hz
X: 0Hz + 12.8kHz LIN
SETUP W11

1106

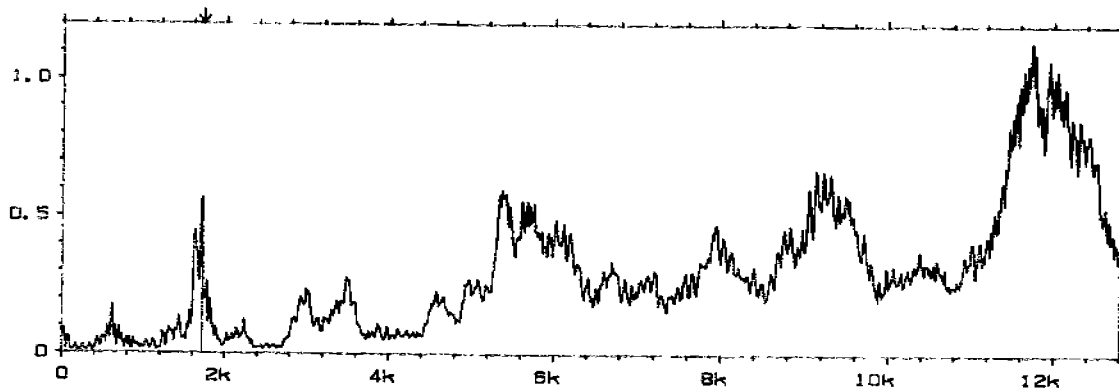
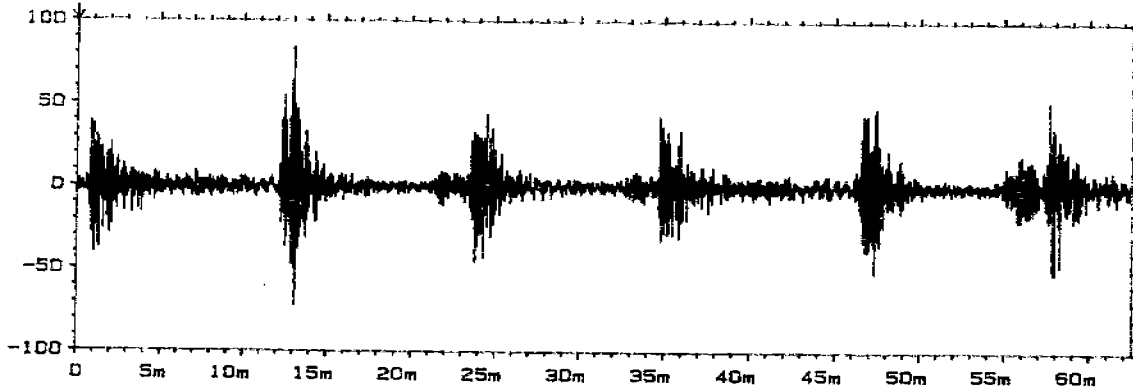
Figure 7b: Partial Windowing Before Major Impulse Response - Retainer



1107

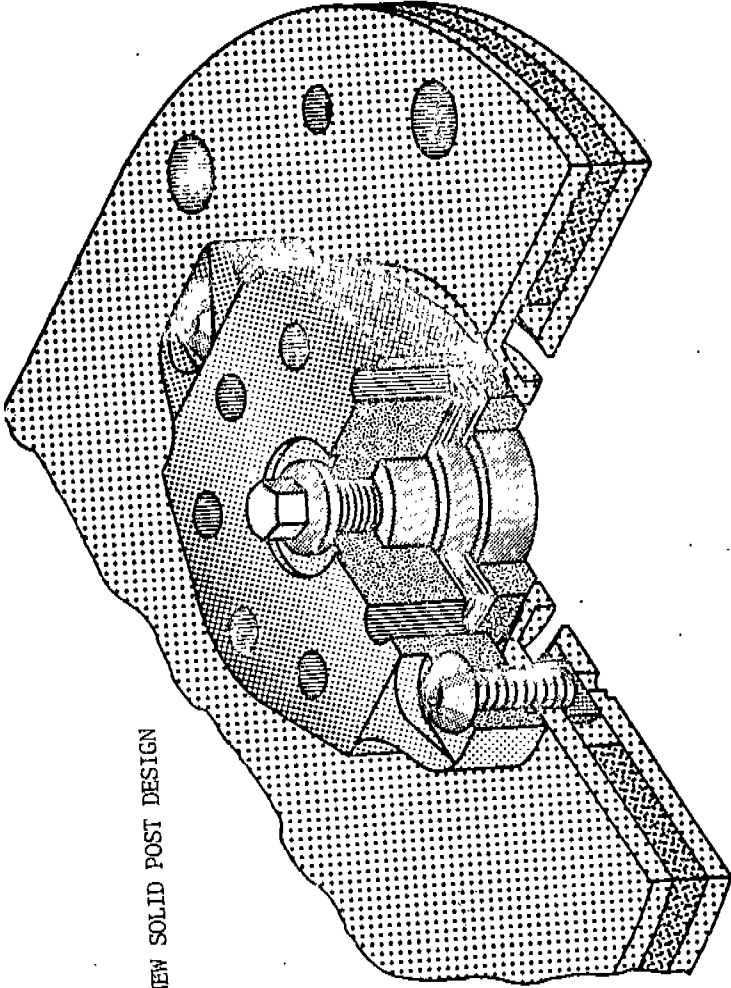
Figure 8: Acceleration - Side of the Crankcase

W14 TIME CH. B REAL INPUT MAIN Y: -977mU
Y: 100U
X: 0.000ms + 62.5ms
SETUP W11 X: 0.000ms



W16 AUTO SPEC CH. B
Y: 1.20U RMS LIN
X: 0Hz + 12.8kHz LIN
SETUP W11 #A: 30

MAIN Y: 565mU
X: 1712Hz



NEW SOLID POST DESIGN

Figure 9: 3D View of Final Valve Assembly

Figure 10: Discharge Valve Related Events - Solid Post Design

TEST CONDITIONS:
 R502
 110 CONDENSER
 -25 EVAPORATOR
 65 RETURN GAS

A - OPENING ANGLE
 327 DEGREES

B - FULL OPEN ANGLE
 333.5 DEGREES

C - CLOSING ANGLE
 344.5 DEGREES

D - CLOSED ANGLE
 364.5 DEGREES

CYLINDER PRESSURE
RETAINER PRESSURE
VALVE DISPLACEMENT
RETAINER ACCEL - - -

0111

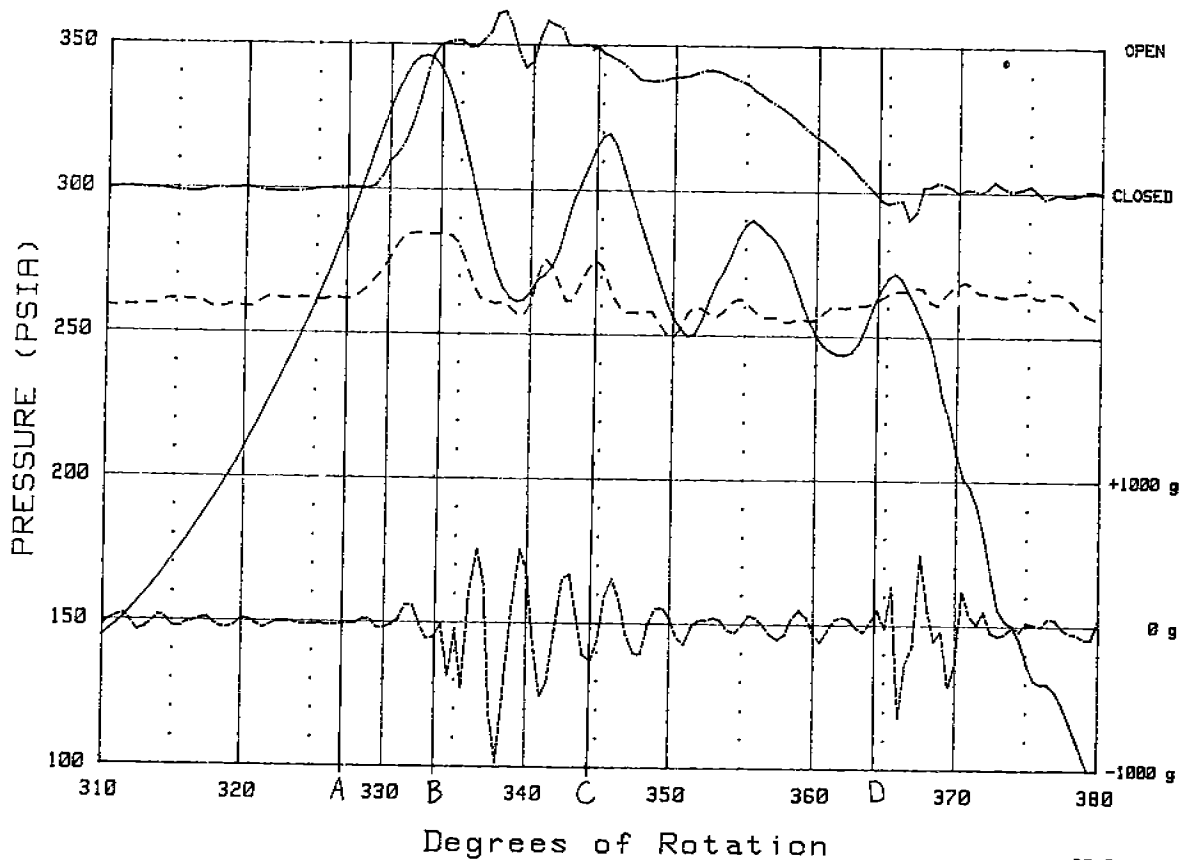


Figure 11: Discharge Valve Related Events - Restrictive Solid Post Design

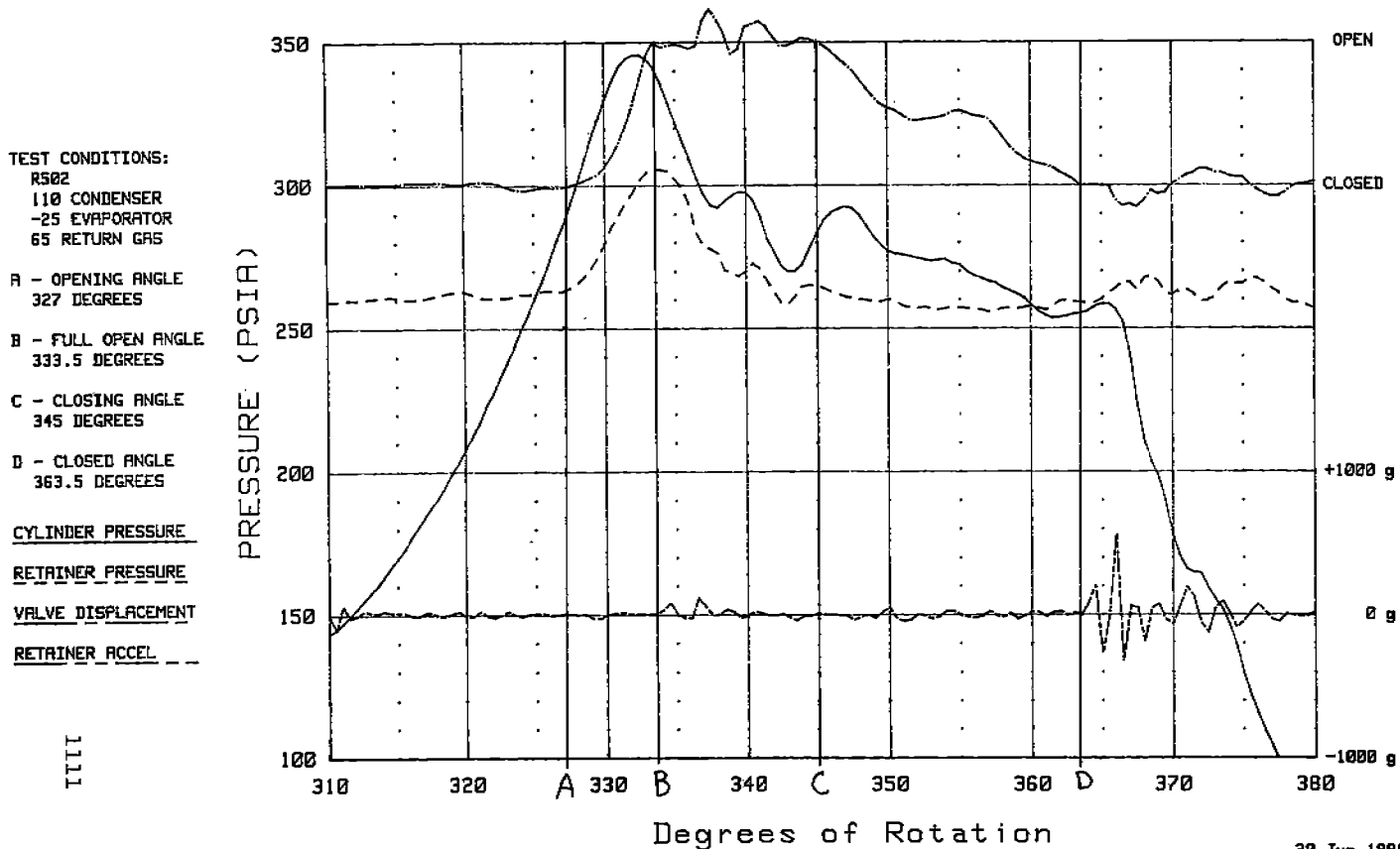
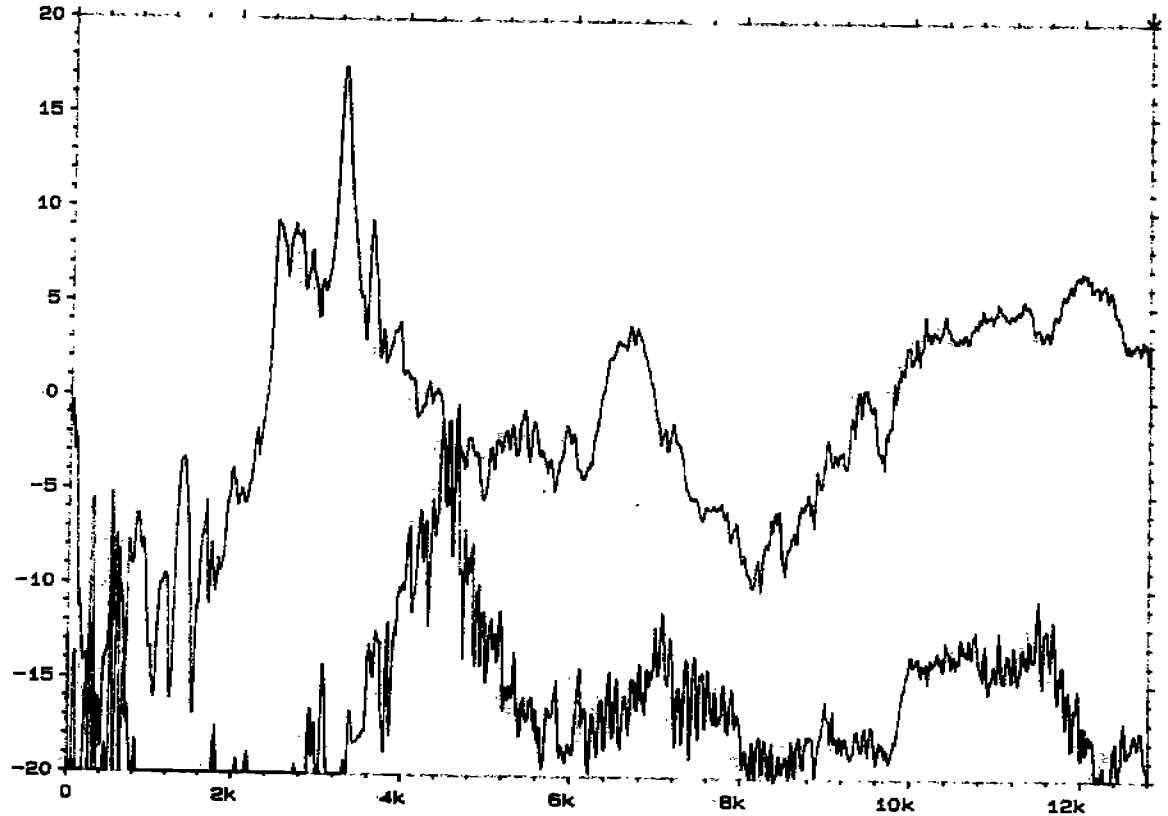


Figure 12. Comparison of Acceleration Data of Retainer for Former and Final Design

W14 AUTO SPEC CH.A [] INPUT MAIN Y: 2.6dB
Y: 20.0dB / 1.00U RMS 40dB X: 12800HZ
X: 0Hz + 12.8kHz LIN
SETUP W11 #A: 30

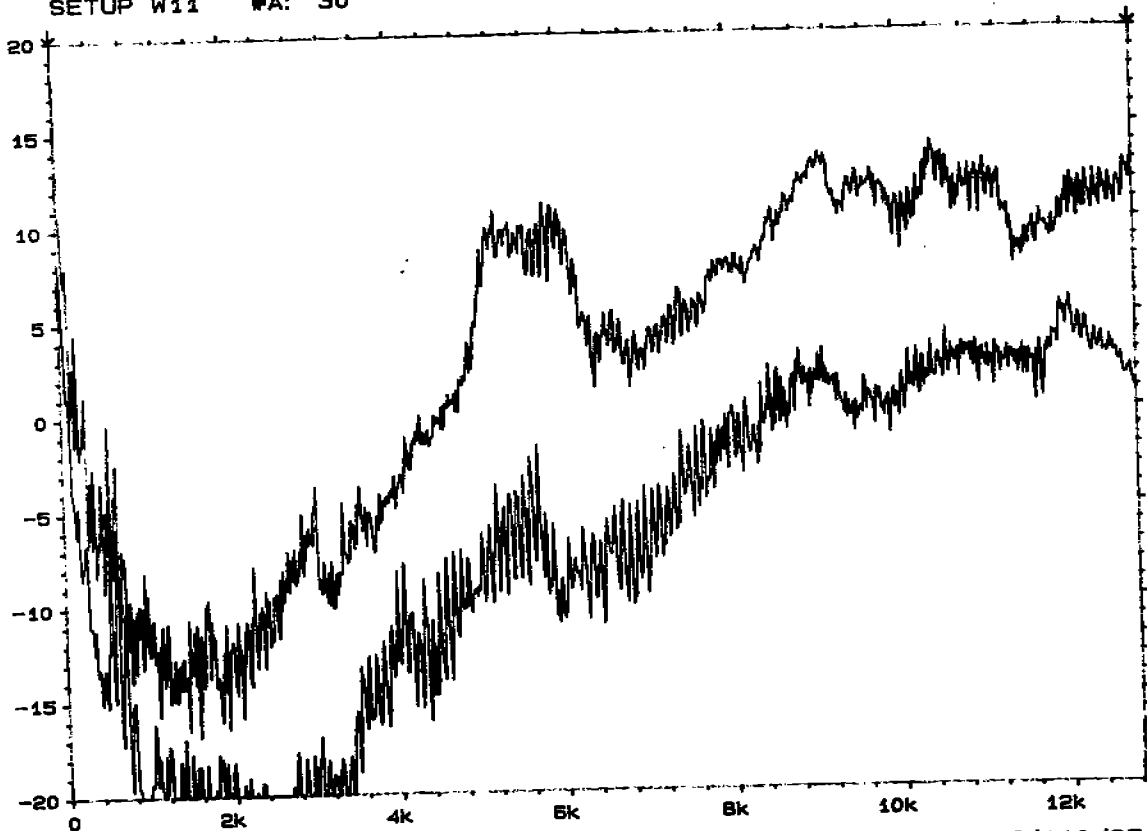


1112

3DS1-1500-TFD 846-Q4069 60HZ 460V R502 -25/110/65
STD DISCUS 33-86-29 4-15-86 TSQ A-RET AC B-CC ACC

Figure 13: Comparison of Acceleration Data of Valve Plate for Former and New Design

W14 AUTO SPEC CH.A [] INPUT MAIN Y: 13.4dB
Y: 20.0dB /1.00U RMS 40dB X: 12800HZ
X: 0Hz + 12.8kHz LIN
SETUP W11 #A: 30

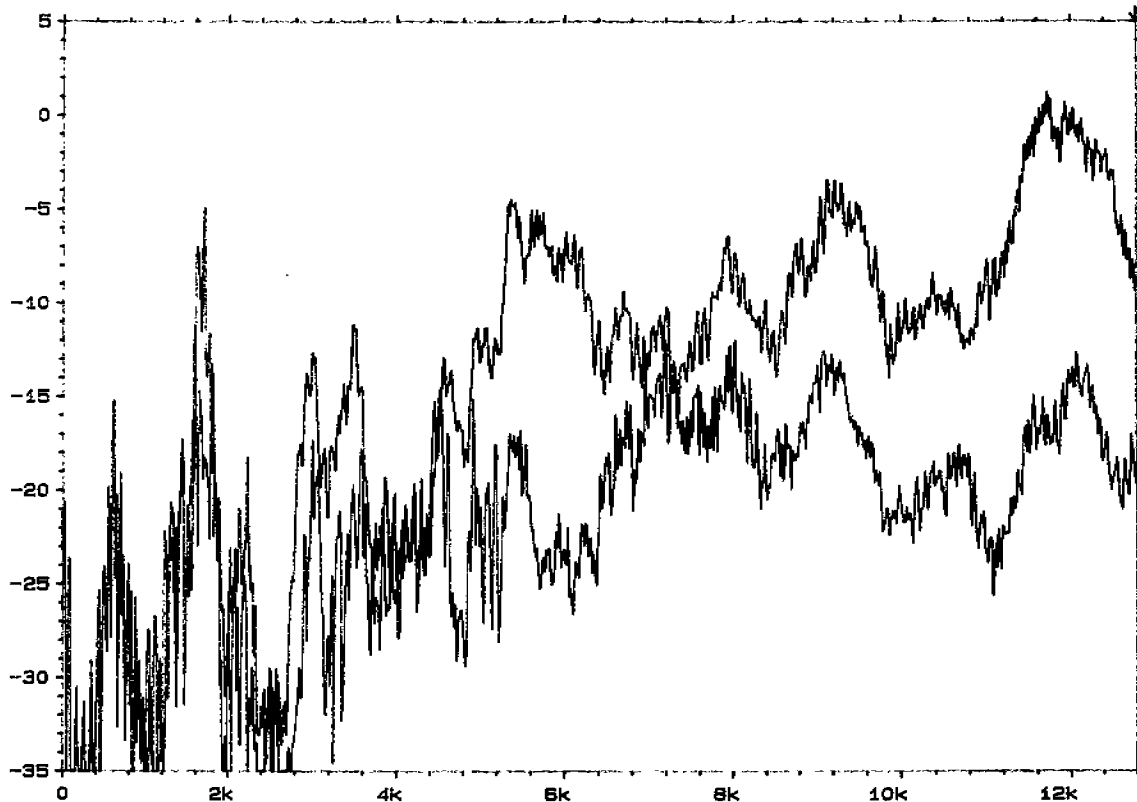


1113

3DS1-1500-TFD 846-Q4069 60HZ 460V R502 -25/110/65
STD DISCUS 33-86-30 4-15-86 TSQ A=VP ACC B=CYL 1

Figure 14: Comparison of Acceleration Data of Crankcase for Former and Final Design

W14 AUTO SPEC CH.B [] INPUT MAIN Y: -8.7dB
Y: 5.0dB /1.00U RMS 40dB X: 12800HZ
X: 0HZ + 12.8KHZ LIN
SETUP W11 #A: 30

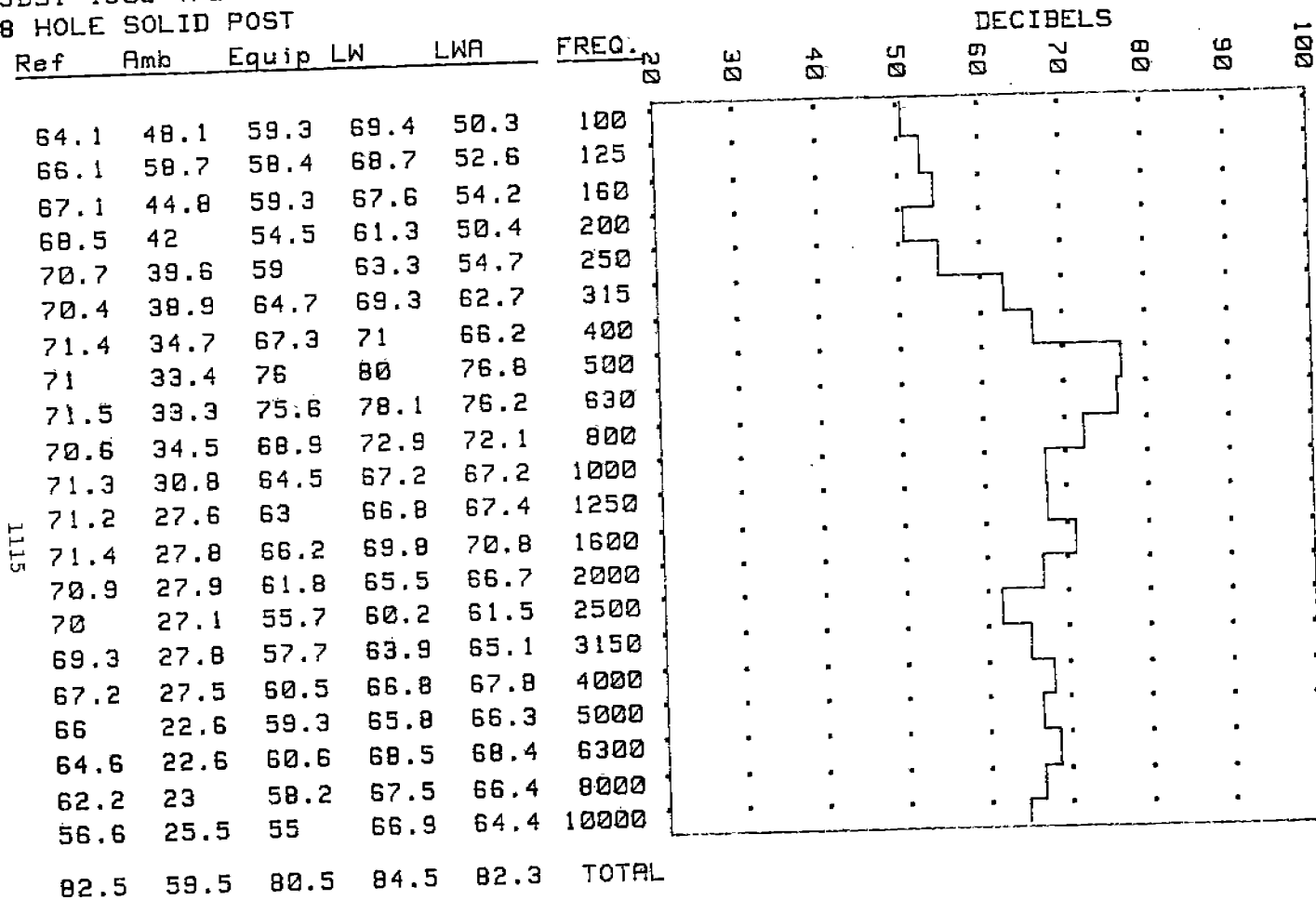


1114

3DS1-1500-TFD B4G-Q4069 60HZ 460V R502 -25/110/65
STD DISCUS 33-86-29 4-15-86 TSG A=RET AC B=CC ACC

Figure 15: Overall Sound Power Level of Final Design of Compressor

3DS1-1500-TFD B4G-Q4069A 60HZ 460V R502 -25/110/65 003-06-02 2-4-86 *
 8 HOLE SOLID POST



1115