DOWNHOLE VIBRATION MONITORING & CONTROL SYSTEM PHASE I FINAL REPORT

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Submitting Organization	APS Technology, Inc. 800 Corporate Row Cromwell, CT 06416

DISCLAIMER

ABSTRACT

The deep hard rock drilling environment induces severe vibrations into the drillstring, which can cause reduced rates of penetration (ROP) and premature failure of the equipment. The only current means of controlling vibration under varying conditions is to change either the rotary speed or the weight-on-bit (WOB). These changes often reduce drilling efficiency. Conventional shock subs are useful in some situations, but often exacerbate the problems.

The objective of this project is development of a unique system to monitor and control drilling vibrations in a 'smart' drilling system. This system has two primary elements:

- The first is an active vibration damper (AVD) to minimize harmful axial, lateral and torsional vibrations. The hardness of this damper will be continuously adjusted using a robust, fast-acting and reliable unique technology.
- The second is a real-time system to monitor drillstring vibration, and related parameters. This monitor adjusts the damper according to local conditions. In some configurations, it may also send diagnostic information to the surface via real-time telemetry.

The AVD is implemented in a configuration using magnetorheological (MR) fluid. By applying a current to the magnetic coils in the damper, the viscosity of the fluid can be changed rapidly, thereby altering the damping coefficient in response to the measured motion of the tool.

Phase I of this program entailed modeling and design of the necessary subsystems and design, manufacture and test of a full laboratory prototype. Phase I of the project was completed by the revised end date of May 31, 2004. The objectives of this phase were met, and all prerequisites for Phase II have been completed.

Table of Contents

List of Figures

Executive Summary

The deep hard rock drilling environment induces severe vibrations into the drillstring, which can cause reduced rates of penetration (ROP) and premature failure of the equipment. The only current means of controlling vibration under varying conditions is to change either the rotary speed or the weight-on-bit (WOB). These changes often reduce drilling efficiency. Conventional shock subs are useful in some situations, but often exacerbate the problems.

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Phase I of this program had several tasks, each of which has been completed successfully. These tasks were:

1. **Analyze requirements for DVMCS**: This task was completed using modifications of APS Technology's WellDrill program, and the anticipated range of vibrations was mapped.
2. **Develop specifications for DVMCS:** Done. Specifications given below.
3. **Prepare top-level design for DVMCE:** Done. See **Figure 10.**
4. **Analyze design to predict performance.** Done. Combined with Task 3.
5. **Characterize properties of damper *via* testing.** Done. The test results are given below. The damper is fully capable of performing as required in a drilling environment.
6. **Complete preliminary economic, market and environmental analysis.** Done. See **Appendix B: Marketing Study.**
7. **Develop preliminary financing plan.** Done, and included as part of marketing study.
8. **Complete development and testing plan for Phase II.** Done. See **Appendix C: Preliminary Project Plan for Phase II**.

Introduction

The key component of this DVMCS is the Active Vibration Damper (AVD). The AVD is designed to isolate and dampen drillstring vibration using a magnetorheological (MR) fluid. Minimizing vibration increases the life of downhole electronic sensors and keeps the drill bit on bottom increasing the rate of penetration (ROP).

The MR damper consists of electronics that measure and monitor vibrations, and a spring-fluid damper that controls the vibration. The damper properties are continuously modified to provide optimal damping characteristics for the vibrations present.

MR fluid is a “smart’ fluid whose viscous properties are changed and controlled by passing a magnetic field through it. MR components have no moving parts, rapid response times and low power requirements. The damping properties can thus be optimized to detune the drillstring from resonant vibration.

MR fluid damping is currently being used in such diverse applications as sophisticated automotive suspensions[1] and earthquake protection systems for buildings and bridges.[2]

The MR Damper has two effects that combine to increase ROP and reduce vibration. First, the damper isolates the drillstring section below the damper from above it. Second, it optimizes the damping based upon the excitation forces such that vibration is significantly reduced. The combination allows the bit to respond more quickly to discontinuities on the cutting surface, while maintaining surface contact.

Separating the bit from the rest of the drillstring with a spring-damper assembly reduces the effective mass that must respond to discontinuities of the drilled surface. Reducing the mass increases the first critical speed of the drillstring attached to the bit, while the adaptive damping reduces the magnitude of vibration at the resonance. This provides a much wider Zone A as shown in **Figure 2**, which is based on a simple model of the damper. For a tricone bit, Zone A now covers a range of 0 – 220 rpm, a significant improvement compared to the 0–80 rpm shown in **Figure 1**.

Modeling (Task 1)

Analyze requirements for DVMCS using WellDrill^{SM}

Review Sources of Vibration

The major sources of vibration that are likely to influence the bottom hole assembly (BHA) in general and the bit in particular, were evaluated. They were characterized, as described below, by their anticipated frequency range and amplitude. The results for these properties are illustrated below in **Figure 3** and **Figure 4** in **Appendix A: Figures** and discussed below in the **Results and Discussion** section.

There are a number of drill vibration sources that could potentially reduce ROP and cause vibration damage to sensors and collars. These are:

Bit excitations from the cones and blades on the bit.

There are multiple cones or blades on the bit. As these travel over discontinuities at the base of the borehole, they produce excitation forces on the drillstring. The excitation is a multiple of the bit speed. For example, a tricone bit has three cones that each strike the discontinuity every revolution of the bit. Therefore a bit rotating at 100 rpm is excited 3 times per revolution, or 300 cycles/min (5 Hz). PDC bits usually have 3 to 8 blades resulting in 3 to 8 excitations per bit revolution.

Tricone bits also have hardened teeth that strike the borehole base as the cones rotate. These also setup drill string excitations based on the number of teeth.

Forward Whirl

Forward whirl is a lateral vibration excited by imbalances within the drillstring. The imbalance may come from machining features of the collar such as hatch pockets, or they may be due to bent collars. The imbalance causes a 1x excitation that may excite lateral vibrations along the drillstring.

Backward Whirl

Backward whirl is cause by the friction between the drillstring and the borehole. If there is sufficient contact force and rotary speed then the collars begin to whirl around the borehole in a counter clockwise direction. The frequency of the whirl depends on the OD of the collars and the ID of the borehole.

Excitation = Collar OD / (Borehole ID – Collar OD)

Mud Motors

Mud motors have an internal rotor that has an eccentric orbit within the stator. This creates an imbalance force on the drillstring. The excitation is a multiple of the motor speed times the number of lobes on the rotor.

Stabilizers

Stabilizers have blades that contact the borehole. The excitation is a multiple of the rotary speed times the number of blades. Straight blades cause more vibration than angled blades.

Stick slip

Stick slip is cause by the friction between the collars or stabilizers and the borehole resulting from the gravitational forces along the drillstring, which may cause the element to hang up. As the drillstring rotates, the drillstring begins to wind up until there is enough force to break free of the friction, then the drillstring spins at a high angular velocity.

Modify WellDrill and analyze effect of active damper

The WellDrill program was modified to include the effect of an active damper upon the behavior of the drillstring. The predicted effects are quite dramatic, as can be seen in the figures, which represent modeling at 30,000 lbs. WOB, which are typical of the range of calculations. Specifically:

1. **Bit contact**: With an active damper adjusted to the appropriate range, bit contact can be made virtually constant, as can be seen in **Figure 5**. Thus, the bit is continually drilling while downhole, as compared to other cases in the figure, in which it is bouncing for a significant fraction of the time.
2. **WOB**: As seen in **Figure 6**, the WOB applied to the bit with an optimized damping remains constant at the desired level. With non-optimized damping, even when the bit is in contact with the formations (see **Figure 5**), the force exerted on the cutters may be far from the nominal value.
3. **Acceleration**: The AVD will also reduce the magnitude of shock and vibration in the bit and the entire drillstring. This can be seen in **Figure 7**, which plots the bit acceleration. With optimum damping, the bit remains relatively stationary.
4. **ROP**: The net effect of the improved bit contact and WOB control is a significant enhancement of the ROP, as can be seen in **Figure 8**. While the ROP does increase again at very high damping levels, what is happening in that case approaches ‘hammer drilling.’ (This can be seen by the acceleration plot in **Figure 7**.) Under these conditions, while ROP may be high, bit life is highly attenuated.
5. **Damper stroke**: As can be seen in **Figure 9**, these enhancements to the drilling process result from the motion of the damper. When in the optimal range of damping coefficients, damper motions are relatively small (± 0.5” from the null position.) In an underdamped or overdamped mode, these motions increase, and are less effective at controlling the bit.

Design

Specification (Task 2)

The specifications for the DVMCS were evaluated based on anticipated drilling conditions and the requirements determined from the modeling. These specifications are given in **Table 1** below.

Table 1: DVMCS Design Specifications

Parameter	Value
Collar OD (in):	6.75
Collar ID (in):	2.00
Overall Length (ft):	25
Temperature (^oC):	-20 to 175
Maximum static WOB (klbs.):	75
Maximum instantaneous WOB (klbs.)	120
Measured WOB Accuracy:	± 1% of full scale
Measured WOB Resolution (lbs):	25
Maximum Torque (kft-lbs):	50
Measured Torque Accuracy	± 1% of full scale
Measured Torque Resolution (ft-lbs)	20
Maximum shock sensed (g)	1000
Shock Resolution (g)	0.25
Spring Rate (lbs/in):	30,000
Damping (lbs-sec/in):	TBD
Tensile Yield Load (klbs):	TBD
Dogleg Severity (deg/100ft):	TBD

Mechanical Design (Task 3) and Analysis (Task 4)

DVMCS

The DVMCS was designed to meet the specifications of Task 2, and, in parallel, its mechanical performance and survival was modeled using finite elements analysis and APS Technology's WellDrill program. The overall design is shown in **Figure 10**.

Test Equipment

After much consideration, it was decided that a standard shake table would be an unsatisfactory test mechanism for this tool. A shake table would drive one end of the DVMCS unit at a given frequency, with an essentially inflexible driver. The other end of the device would either be fixed or free. None of these is a realistic representation of the downhole operating conditions.

In operation, the DVMCS would be supported and loaded by the entire drill string above it. Considerable weight would be applied from above, and this loading would have both resilience and damping. The damping would result from the intrinsic damping in the drill string itself, from the hydraulic damping of the drilling fluid and from contact with the borehole walls. At the bit, the driving force will result from the interaction of the bit and the irregular bottom of the hole. This interaction will have a primary frequency (*e.g.*, triple the rotation rate for a tri-cone bit), but may have other harmonics as well if there is more than one high point on the well bottom. In addition, the well bottom is not completely rigid, but can respond to the bit by flexing or being drilled away. (If not, there would be no point in drilling.)

To simulate these conditions, both for the testing of the MR valves and the entire DVMCS, we designed the test bench shown in **Figure 11**. The test piece (5) is supported by linear bearings (4) on a large load frame (6). At the 'uphole' end, to the left, a large pneumatic cylinder (1) applies a force simulating the loading from the drill string above the tool. The damping of the drill string motion is simulated by two hydraulic cylinders (2) configured to produce adjustable damping. To mimic the driving force of the bit's interaction with the bottom of the well, a lower assembly (7) is provided. In this assembly, a cam (8) is rotated by a variable speed gear motor (9) at rates simulating the drillstring rotation rate. The cam, which is supported by ball bearings, has a configuration that can mimic a variety of degrees of irregularity of the well bottom.

The first step was the testing of the MR valves to determine their properties and efficiency. For this purpose, a subset of this test equipment was employed. The two hydraulic cylinders from the test stand were filled with MR fluid. As the pneumatic cylinder moved, it drove the hydraulic cylinders to pump fluid through the valve at different pressures and flow rates.

We designed an MR valve test assembly (**Figure 12**) to use in this configuration. The MR fluid was forced in through the port in the left end cap (13) and flowed through the narrow annulus between the outer housing (1) and the inner mandrel (2) and exit through two ports in the right end cap (3). Four magnet coils (not shown) with bucking fields were wound in the grooves in the mandrel. By applying electric current to the valve circuits, the ability of the valve to stop flows of different pressures was measured, as was the amount of electrical power needed to operate the valve. Performance was monitored by measuring both the flow in and out and the pressure at three points in the valve (8). The design allows the central portion of the valve to be shifted relative to the outer part by using different spacers (10), so that the efficiency of the MR circuit was studied over the range of motion anticipated downhole (~4").

In this manner, the same basic test apparatus use for the valve testing and the Phase I damper testing can be used for the full-scale prototype testing in Phase II.

Electrical Design (Task 3)

The design of the prototype control electronics was completed and the prototype electronics was manufactured, then used in the test. The circuit diagram is shown below in **Figure 13**.

Experimental (Task 5)

Determination of damping coefficient

Using the instrumented damper test piece, (see **Figure 14**) the damping coefficient was calculated over a range of operating values as follows.

The coefficient of damping is, in most simplistic terms, force/velocity. In our case, the relevant velocity is the velocity at which the valve mandrel moves relative to the housing, and the force is the fluid pressure multiplied by the area upon which that pressure acts.

The test apparatus records pressure at several points, as well as time needed to actuate the cylinders and the distance the cylinder travels in that time. Velocity and force can then be calculated from the measured values and the known system geometry.

A_b = Cylinder bore area = 4.91 in^2

A_r = Cylinder rod area = 0.79 in^2

A_v = Valve flow area = 0.49 in^2

A_p = MR valve piston area = 6.48 in^2

Force is the measured pressure across the valve, P (lbs/in^2), multiplied by the piston area of the valve, A_p, in this case 6.48 in^2.

The fluid is driven through the damper by two cylinders. The average flow of the fluid through the valve is the volume of fluid introduced divided by the time, t, taken to displace this volume. The fluid volume, V, is the area of each cylinder displacing the fluid, in this case (A_b - A_r) or 4.12 in^2, multiplied by the two cylinders, then multiplied by the length of travel, L; *i.e.*, V (in^3)= 8.25 $in^2 \cdot L$

Therefore, the flow rate, Q, is:

$$Q\ (in^3/s) = 8.25 \cdot L / t$$

The valve velocity, v, is the speed at which the valve extends as fluid flowing into the chamber forces it to move to accommodate the change in volume. This is the flow rate divided by the piston area, or:

$$v\ (in/s) = Q / A_p = 1.27\ L / t$$

The damping coefficient, c, is then:

$$c\ (lbs/in/s) = P \cdot A_p / v = (6.48/1.27) \cdot P / L / t = 5.09 \cdot P \cdot t / L$$

Varying loads were applied to the test damper, corresponding to different values of WOB, and the damping coefficients were determined as above. The results of these measurements are shown in and **Figure 15**, and discussed in the **Results and Discussion** section below.

Testing of DVMCS prototype

The laboratory prototype was initially assembled in February (See **Figure 16** to **Figure 22**.) During the assembly, however, several key subassemblies became galled together. The galling was attributed to several factors:

- While hard steels were used for the components, extensive surface treatments, which will be used in downhole prototypes, were considered unnecessary. This proved to be an erroneous assumption.
- Assembly of the prototype requires threading pieces into long blind holes with very tight tolerances, making misthreading fairly easy.
- Alignment problems were complicated by the horizontal assembly of the tool, which tended to accumulate all of the tolerances on one side. Also, torquing the threads across long sections applied side forces which can further misalign the components.

After the components were disassembled or cut apart, and evaluated, a detailed design review was held, and several changes were made in materials, surface treatments and tolerances. One component, the MR valve mandrel, was redesigned into a three-piece assembly. This will facilitate its integration into the tool, and make it easier and less costly to replace the part when the threaded areas wear in use. The new design is shown in **Figure 10**

This revised prototype was tested by applying varying forces (WOB) on the 'uphole' end of the device, while driving the 'bit' end *via* the cam system, described above using a matrix of values, which is summarized below in **Table 2**. At each set of conditions, the current being applied to the AVD was varied over a wide range, and the effect on the motion of the tool was recorded. These data were analyzed and the characteristics of the damper determined, as reported in **Results and Discussion**, below.

Table 2: DVMCS Test Matrix

Parameter	Values
Vibration amplitude	0.708"
Excitation frequency	0.5 – 2.0 Hz
WOB	5,000 and 10,000 lbs.
Drillstring mass and damping	2 values (not calibrated)

Results and Discussion

Modeling (Task 1)

The results of the modeling of bit-induced vibration are given in **Figure 3** and **Figure 4**. The ranges of vibrations were used in optimizing the design of the DVMCS.

Performance of the Damper (Task 5)

Determination of the Damping Coefficient

The results of the initial damper testing are given in **Figure 15**. For a given electrical power applied to the MR valve coils, the damping coefficient is plotted as a function of the effective WOB applied to the system.

It will be noted that as WOB increases, the damping coefficient decreases. This results from the nonlinear behavior of the MR fluid. With increasing load (pressure) the fluid flows through the cell at higher velocities. Under the influence of these velocities, and their resulting shear forces, the magnetic particles in the fluid are more readily separated from one another. This results in a lower viscosity, and hence a lower damping coefficient.

It will be noted that for each curve (other than the zero power curve), there is a minimum WOB value. At this point, the impedance of the damper is such that the pressure applied cannot move the fluid through it and motion stops. Note that in the downhole tool, the static WOB will be supported by the Belleville spring stack; the damper will only need to react to the variations caused by shock and vibration. Therefore, in the downhole application, these high damping levels may not be necessary.

Also, **Figure 15** shows that even with no power applied, the damping coefficient is in the range of 10-20,000 lbs/in/s, which may be higher than is desirable for optimum performance. We are, therefore, reconfiguring the damper and increasing the clearance between the poles of the magnets to shift its performance to a slightly lower range. The new version was used in the testing of the full DVMCS.

The results of testing the redesigned damper are shown in **Figure 24**. As can be seen from the figure, increasing the current through the damper greatly increases its damping coefficient over the range of frequencies studied. This confirms the analysis made on the earlier design.

Testing of the full DVMCS

The intent of the vibration testing was to:

1. Determine the performance of the MR damping fluid for the downhole damping application
2. Characterize the MR Damper for spring rate and damping as a function of vibration frequency, amplitude and WOB. Develop damping coefficients for algorithms to be used in the software for the tool.
3. Evaluate the seals and compensation system
4. Determine the power requirements of the system

Graphs summarizing the results of this testing are shown in Appendix A, **Figure 24-Figure 36**. While the analysis of the large volume of data* is ongoing, and will form the basis for the development of the feedback control algorithms in Phase II, there are several observations that can be made now.

1. The initial round of tests demonstrates the effectiveness of using the MR fluid as a vibration damping medium.
2. The amount of damping depends on the current passed through the MR fluid and the gap of the damper. Increasing the current through the MR fluid significantly increases the damping (**Figure 24**)
3. The dynamic stiffness of the damper is a combination of the stiffness of the springs, the amount of damping and the vibration frequency. As the damping increases the dynamic stiffness of the AVD increases. (**Figure 26** & **Figure 30**). [Note: Some of the data displayed in these figures was taken during different setups. The hydraulic damping controls for WOB and string damping are difficult to control and may not exactly repeat their settings from run to run. The data in some of the figures may, therefore, be offset from one frequency to the next.]
4. Extensive data were gathered on the variation of the AVD performance over the frequency range. Parameters studied included: relative displacement (**Figure 27** and **Figure 31**); WOB applied to bit (**Figure 28** and **Figure 32**); total system damping (**Figure 29** and **Figure 33**). Some combination of these parameters will be used in Phase II to drive the feedback algorithm.
5. To optimize the damping, different gap sizes will be evaluated. This includes gaps both larger and smaller than the original 0.060”.
6. The AVD test bed can apply a maximum WOB of 10,000 lbs. This limits the amount of stroke that will be imposed on the AVD test piece. For the additional testing in Phase II, the spring rate of the Belleville springs will be reduced to 5,000 lbs/in. This provides a stroke of ± 2”, or 50% of the maximum stroke.

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* See **Figure 34 - Figure 36** for a small sample of the raw data. Each plot represents just a few minutes’ data at a single combination of parameters.

7. The coil and the potting held up well under test conditions. There had been a concern that the potting might erode from the velocity of the MR fluid passing over the potted coils, but this was not the case in the laboratory testing.

Conclusions

The overall objectives of this phase of the project have been met.

- The DVMCS environment has been modeled and the tool specifications set.
- Extensive mechanical modeling demonstrated that the DVMCS will have a significant, even dramatic, effect on the drilling process.
- The AVD component of the DVMCS has been designed, tested, modified and re-tested. The second design has shown that it can provide the necessary range of damping under downhole conditions, at acceptable power levels, to produce the effects mentioned above.
- The design of the downhole prototype will include significant segments of the laboratory unit, and the downhole tool may include some of the actual laboratory hardware.
- Sufficient data has been gathered to develop effective feedback algorithms, but additional data will be taken in Phase II to optimize the design.

Units

To be consistent with standard oilfield practice, English units have been used in this report. The conversion factors into SI units are given below.

1 ft.	=	0.30480 m
1 g	=	9.82 m/s
1 in.	=	0.02540 m
1 klb.	=	4448.2 N
1 lb.	=	4.4482 N
1 rpm	=	0.01667 Hz
1 psi	=	6984.76 Pa

References

[1] *cf.*, *e.g.*, "Magnetic Ride Control," *GM Tech Links,* Vol. **4**, No.1, *pp.* 1-2, January, 2002, http://service.gm.com/techlink/html_en/pdf/200201-en.pdf .

[2] B.F. Spencer Jr., S.J. Dyke, M.K. Sain and J.D. Carlson, "Phenomenological Model of a Magnetor-heological Damper," Journal of Engineering Mechanics, ASCE, **123** 230-238, 1997, http://www.nd.edu/~quake/papers/MRD.Journal.pdf .

Appendix A: Figures

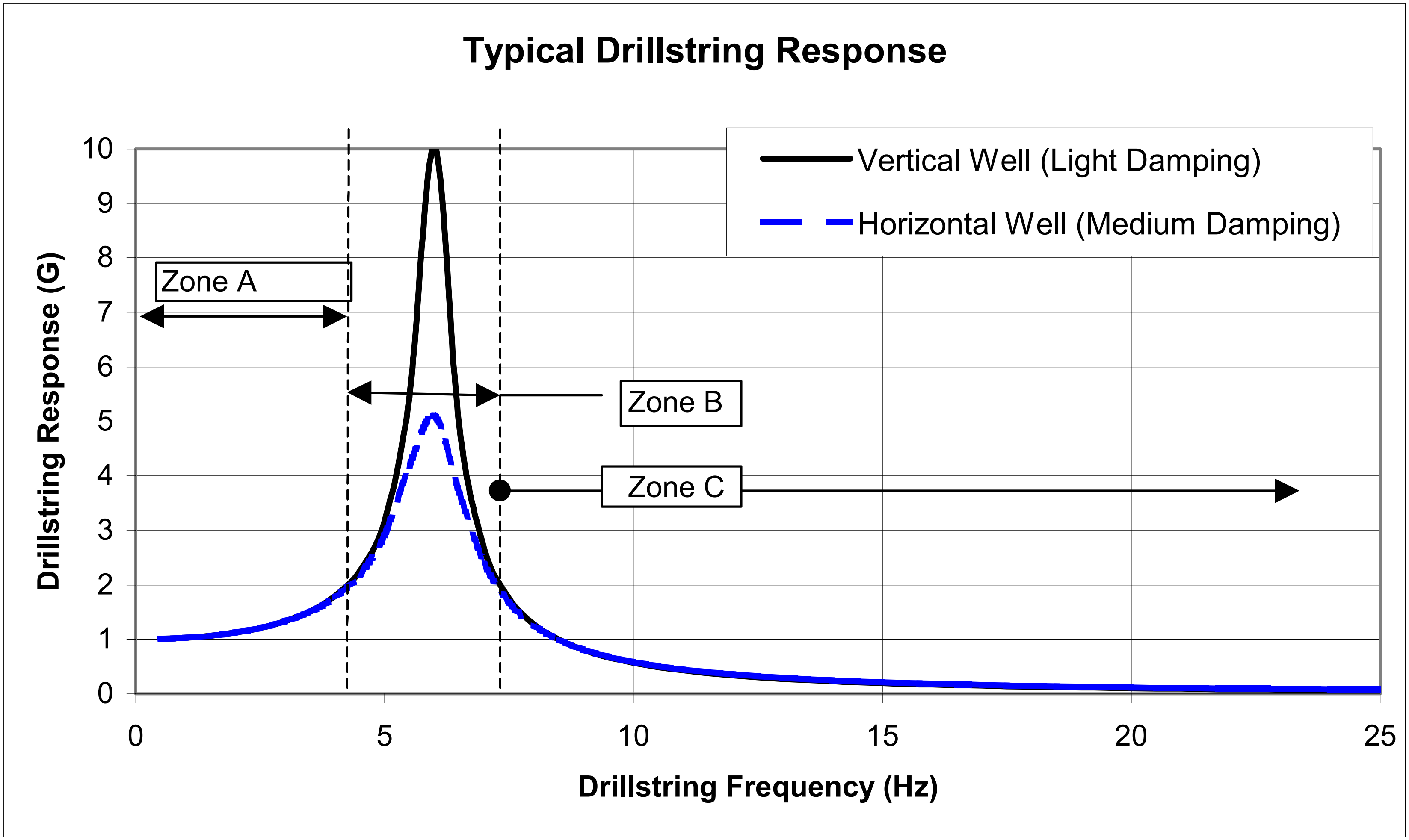


Figure 1: Frequency response of a typical drillstring

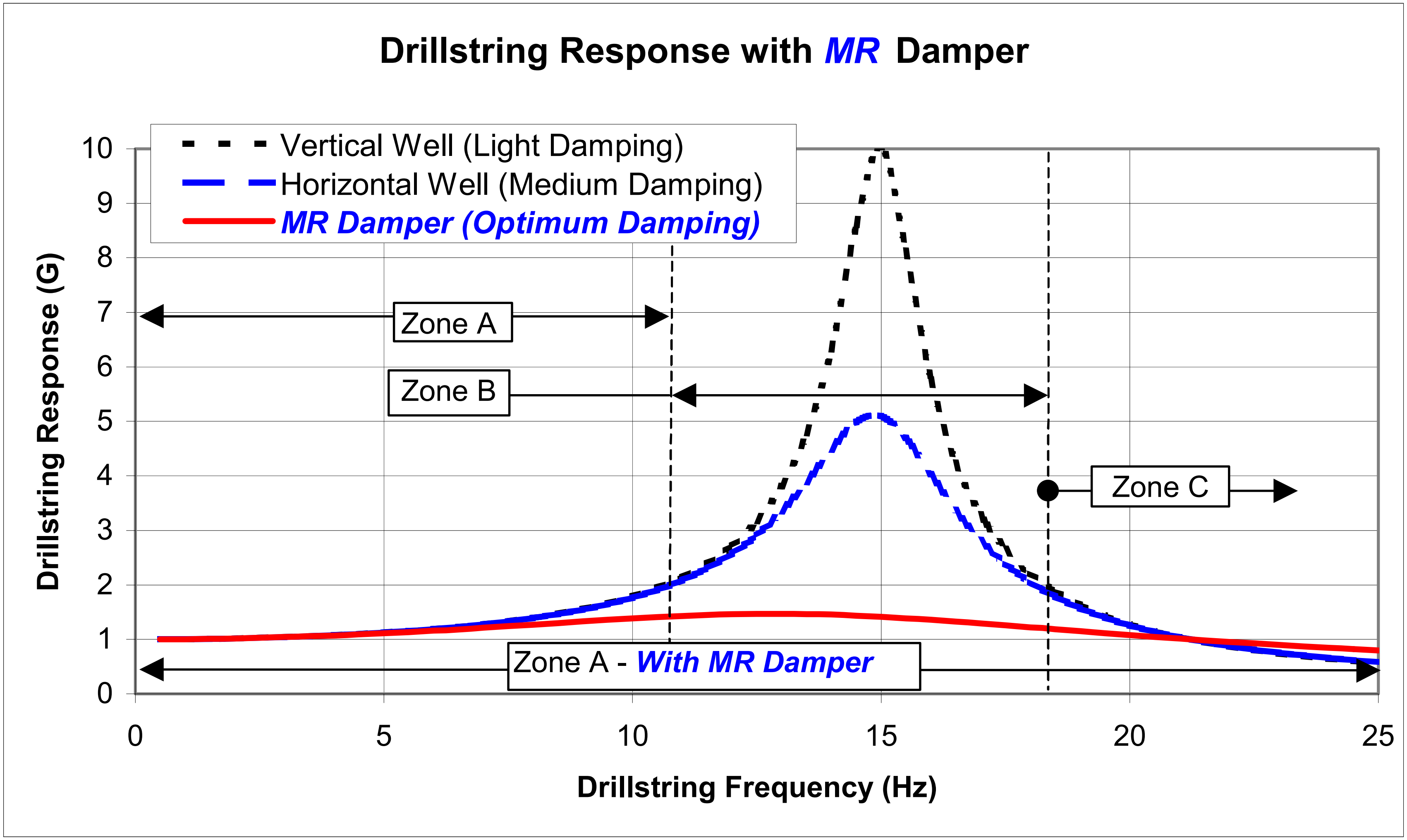


Figure 2: Drillstring response with an active MR damper in use

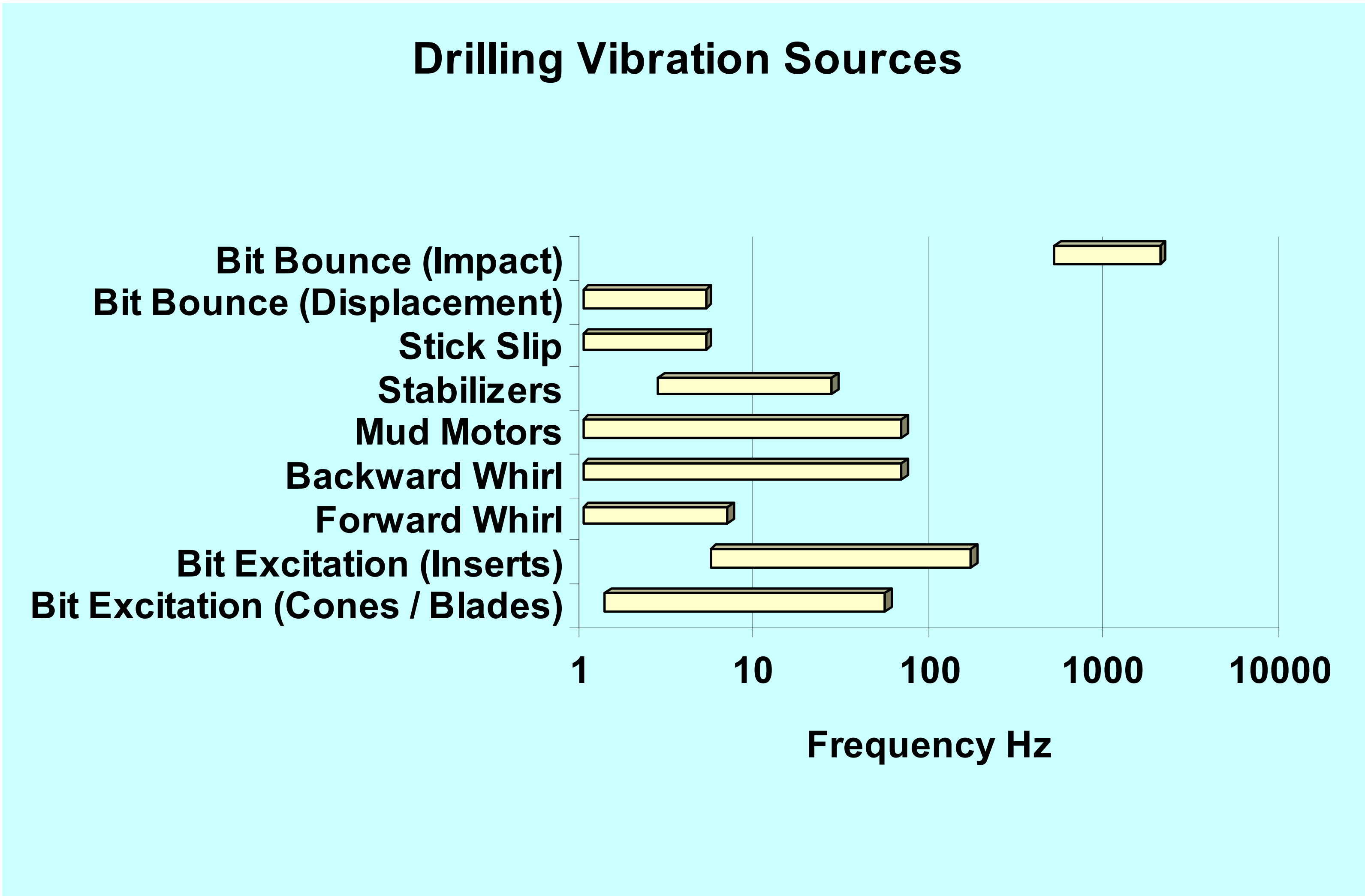


Figure 3: Frequency distribution of common sources of bit vibration

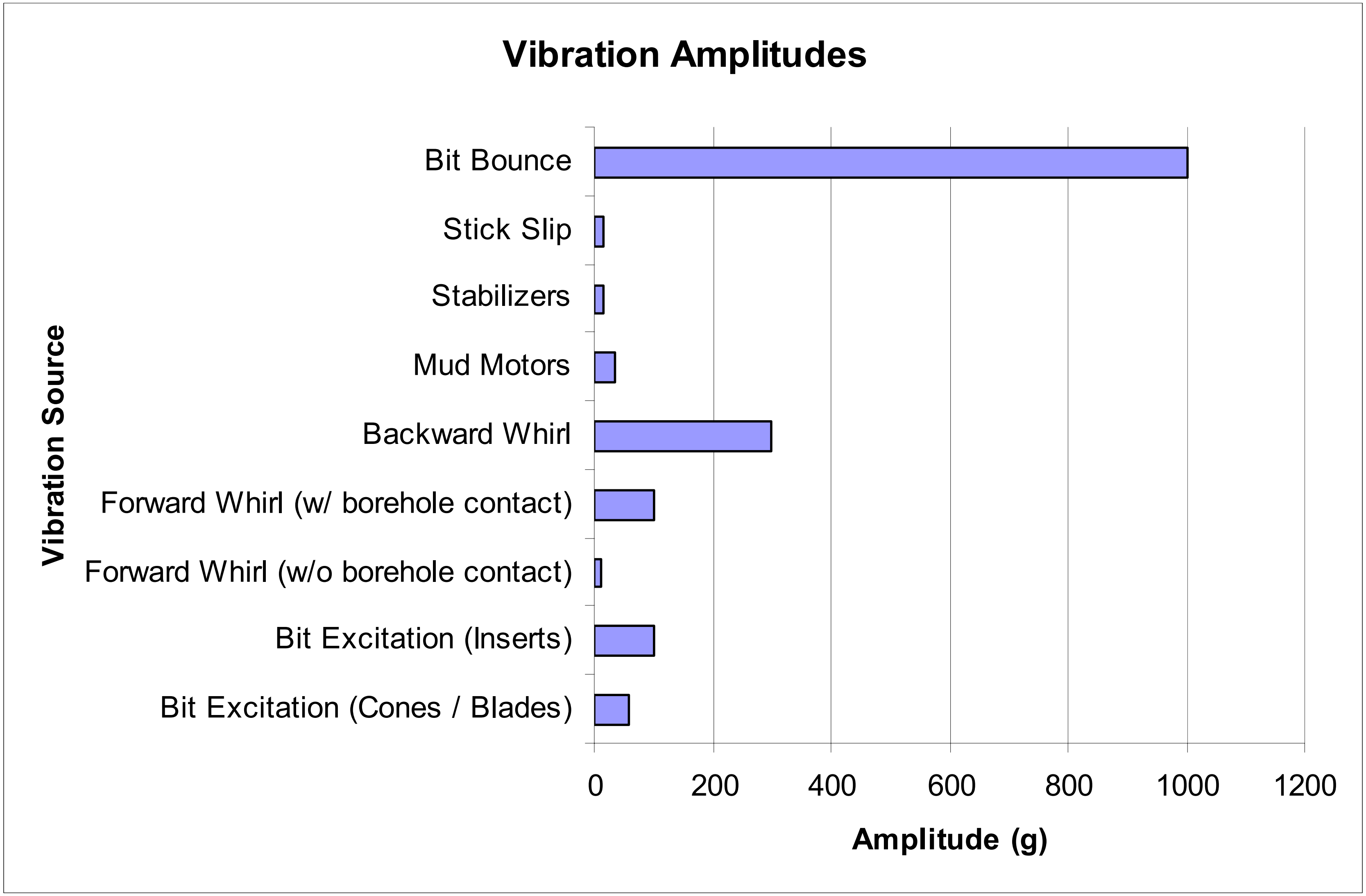


Figure 4: Range of amplitudes of common sources of bit vibration

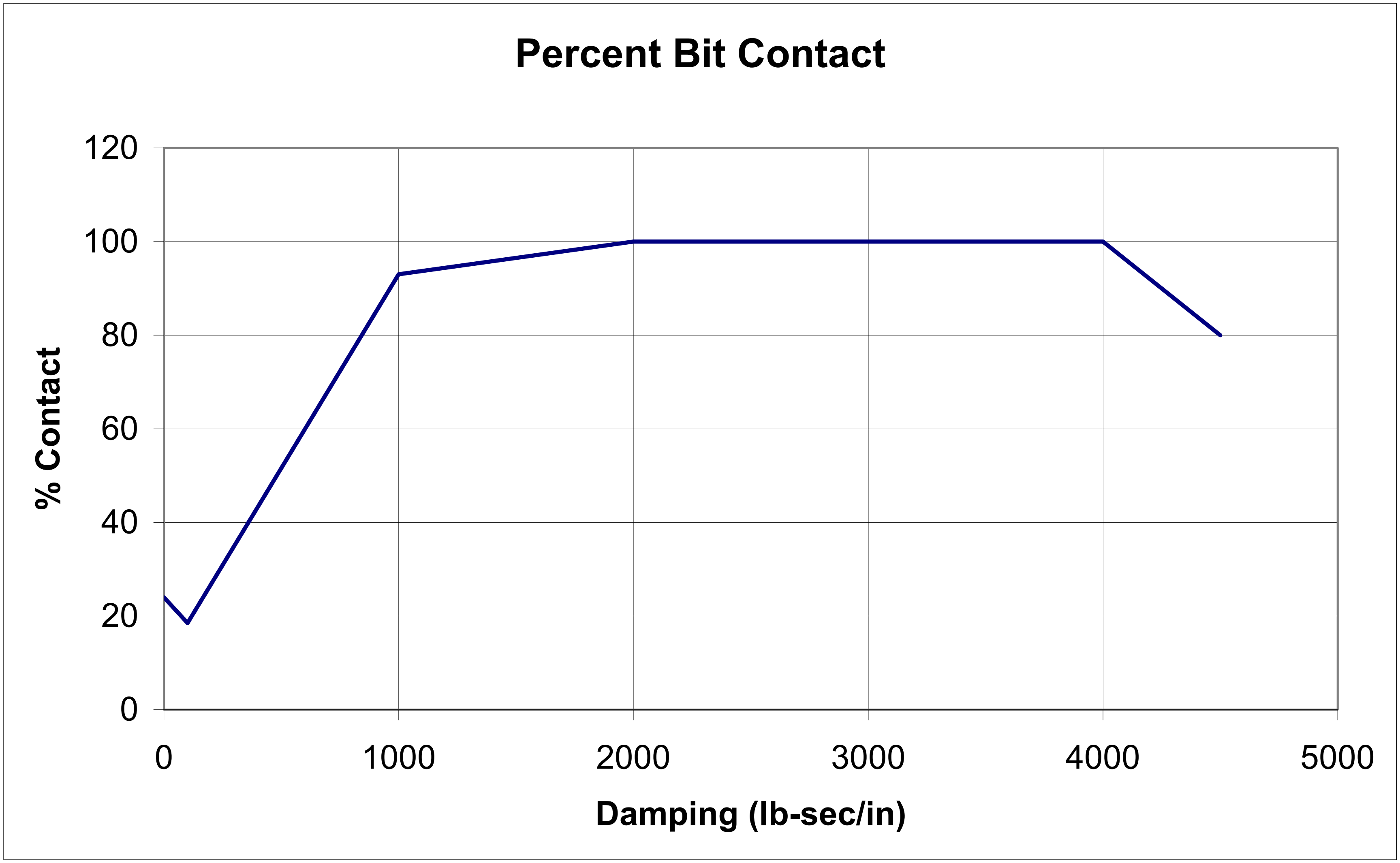


Figure 5: Bit contact : 30,000 WOB - 30,000 in-lb spring rate

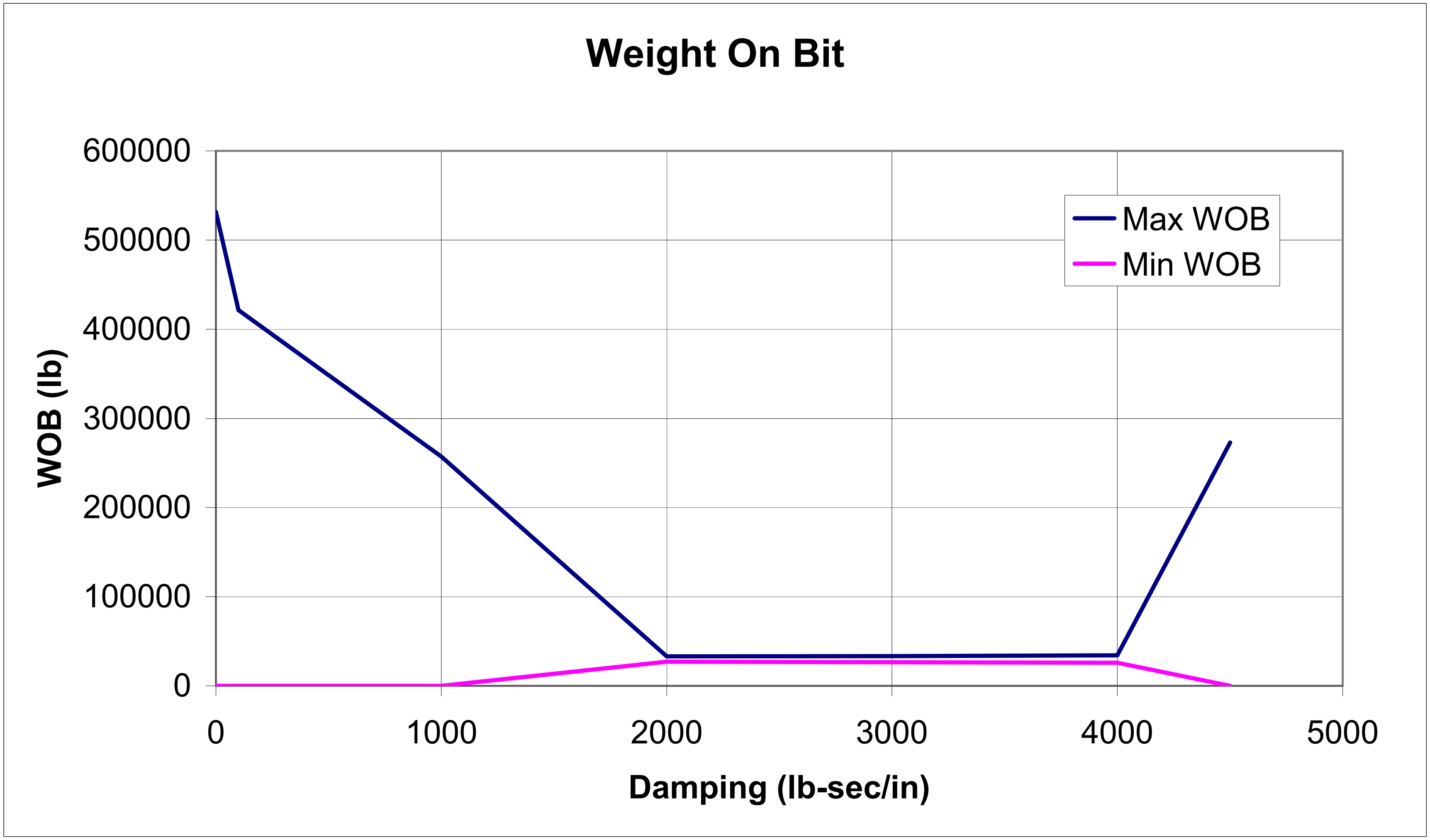


Figure 6: Measured WOB: 30,000 WOB - 30,000 in-lb spring rate

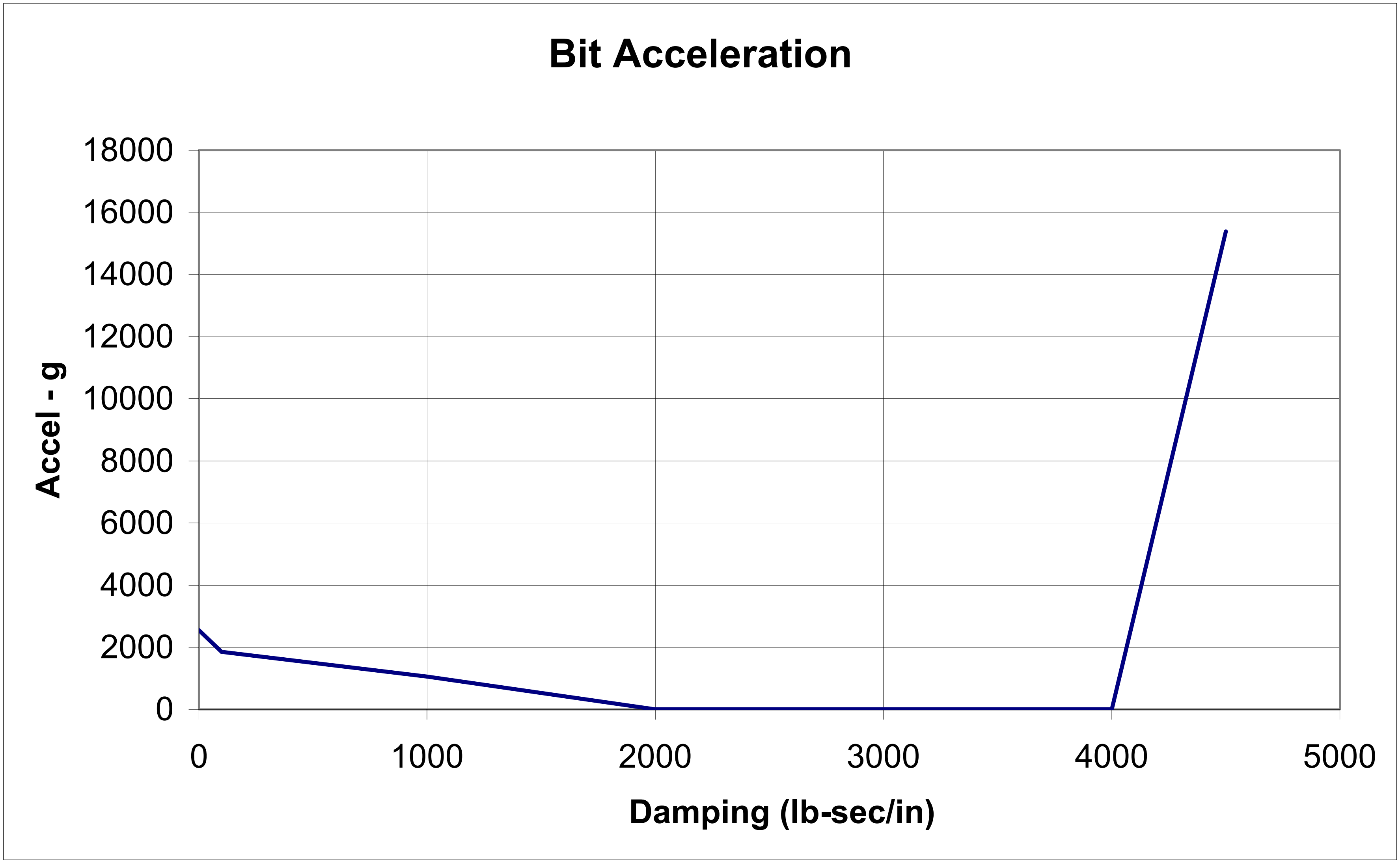


Figure 7: Bit acceleration: 30,000 WOB - 30,000 in-lb spring rate

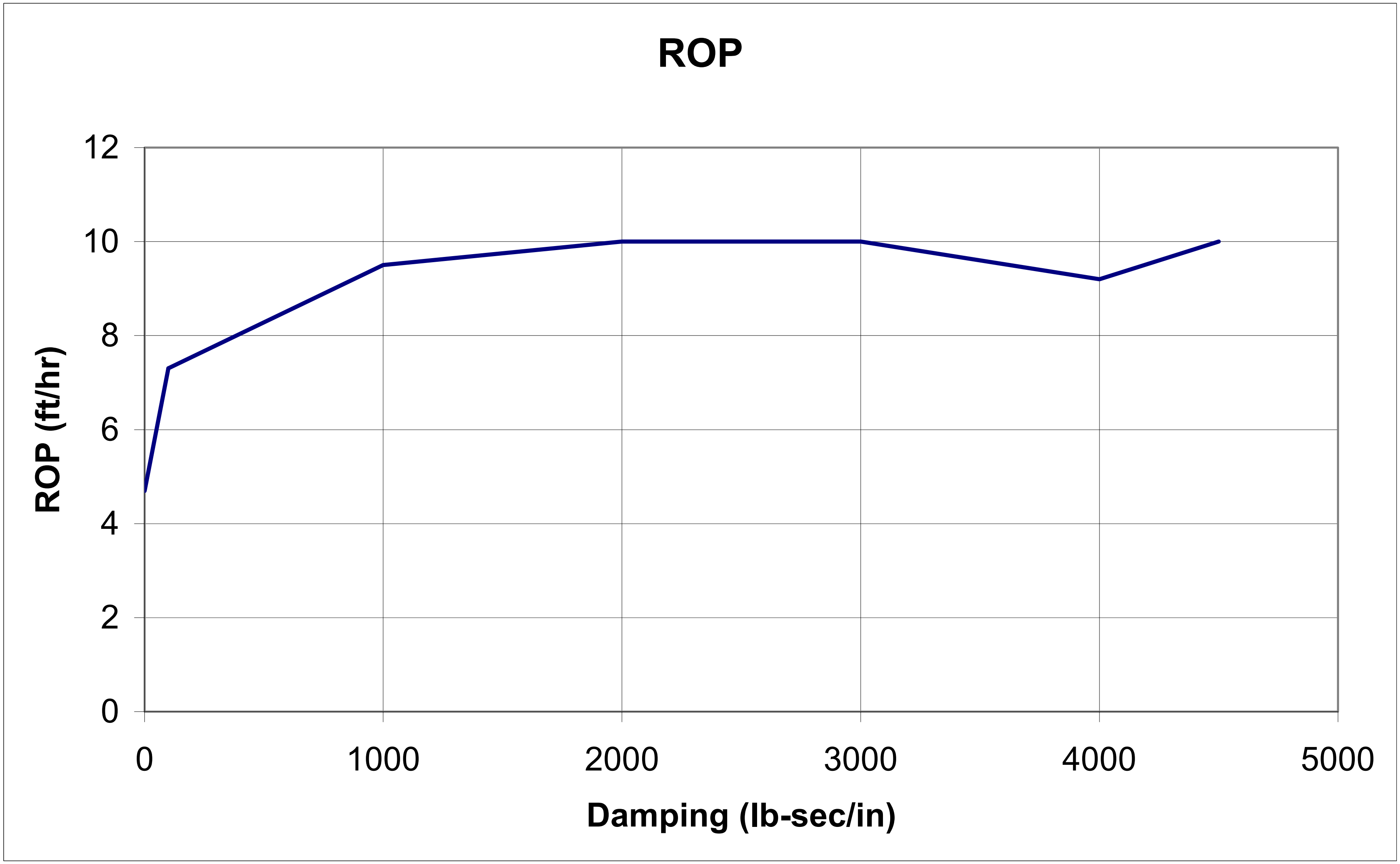


Figure 8: ROP: 30,000 WOB - 30,000 in-lb spring rate

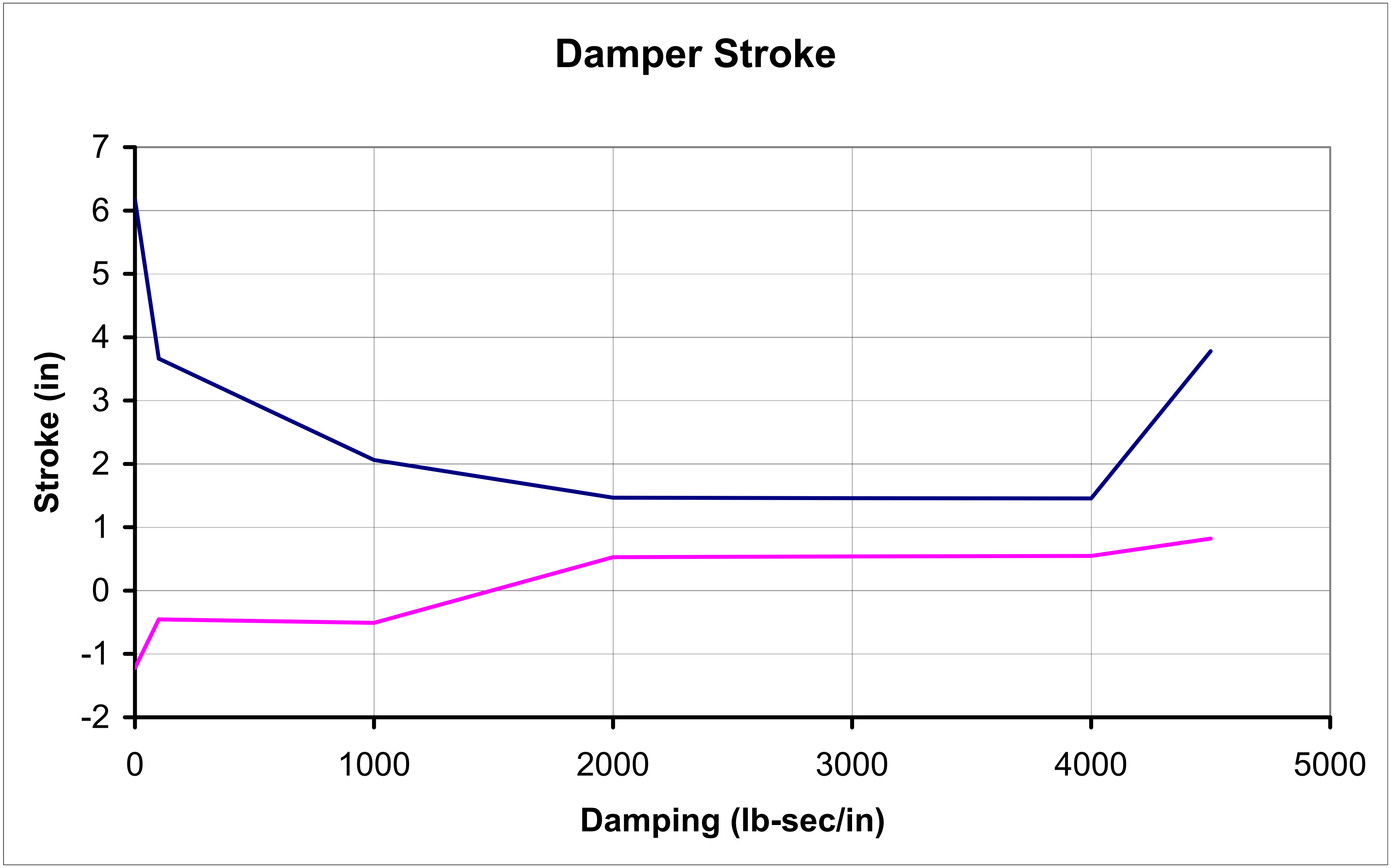
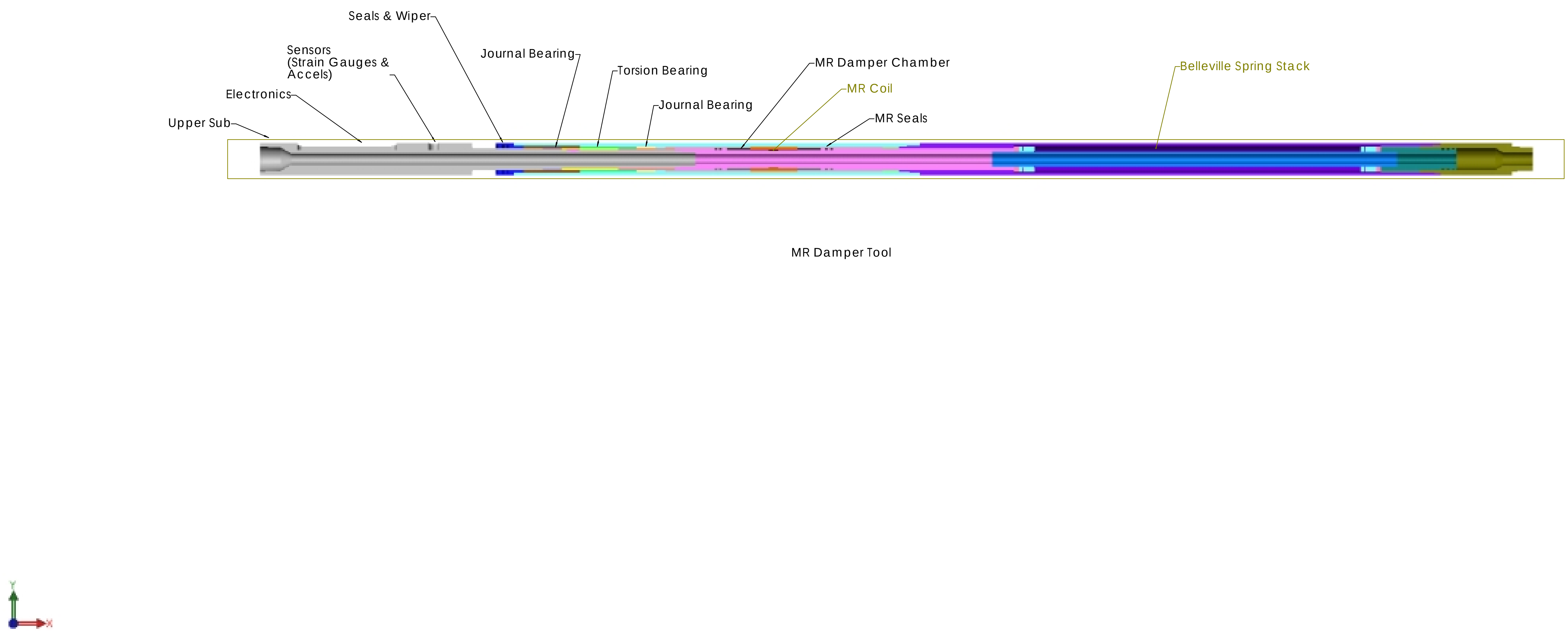
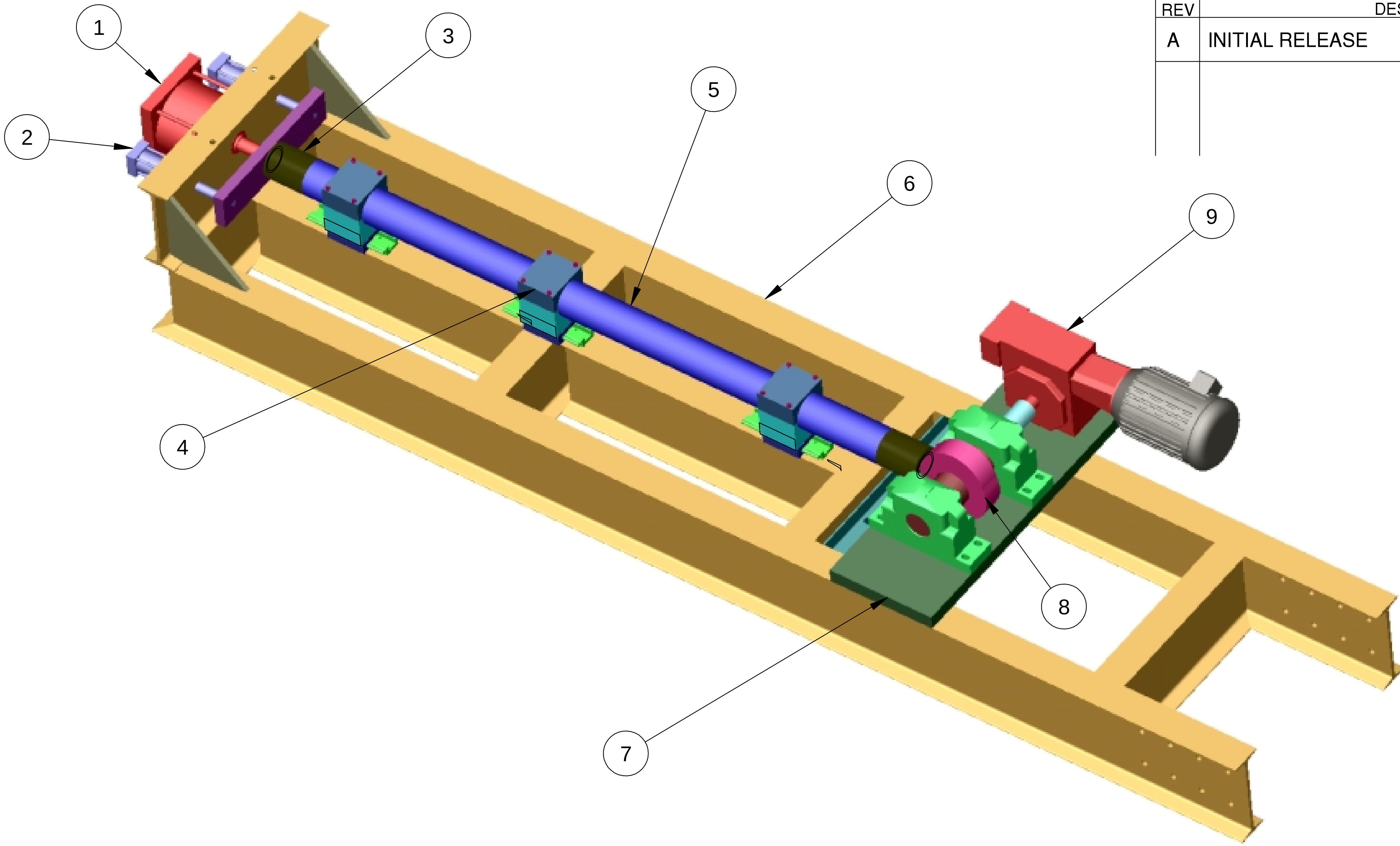


Figure 9: Damper movement: 30,000 WOB - 30,000 in-lb spring rate



MR Damper Tool



REVISIONS

REV	DESCRIPTION	DRAWN	CHECKER
A	INITIAL RELEASE	TFR 06/13/03	CAP 06/13/03

COMPANY CONFIDENTIAL

NOTES: U.O.S.

MATERIAL:	HEAT TREAT:	FINISH:
N/A	N/A	N/A

CAD FILE SK-10300 A TEST ASSY. - DRILLING VIBRATION MONITORING & CONTROL SUB.SLDDRW

UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN INCHES & APPLY AFTER FINISH
PART MUST BE FREE OF BURRS AND/OR FLASH
BREAK SHARP EDGES APPROX .005
FILLET RADII .020 MAX
SURFACE FINISH 63 MAX

APPROVALS	BY	DATE
DRAWN	RZASA	06/13/03
CHECKED	CAP	06/13/03
ENGRG	FISH	06/13/03
MFG	APS	XX/XX/XX

TOLERANCES
.XXX ±.005 FRACTIONS ±1/64
.XX ±.01 ANGLES ±2°
INTERPRET DWG PER ASME Y14.5M-1994
DIMS IN PARENTHESIS ARE REF ONLY
DO NOT SCALE DWG

APS TECHNOLOGY
800 CORPORATE ROW
CROMWELL, CT 06416-2072

TEST ASSY. - DRILLING VIBRATION MONITORING & CONTROL SUB

SIZE	DWG NO	REV
B	SK-10300	A

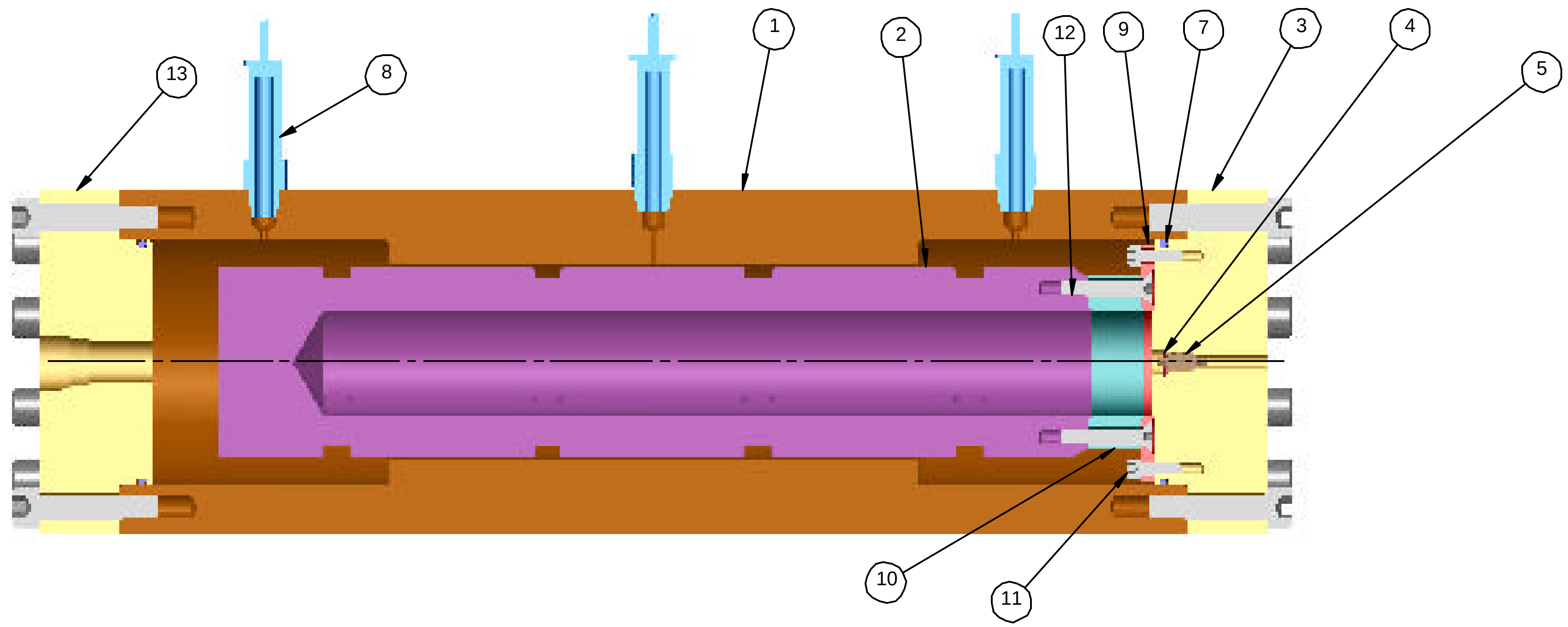
SCALE: NONE
SHEET 1 OF 1

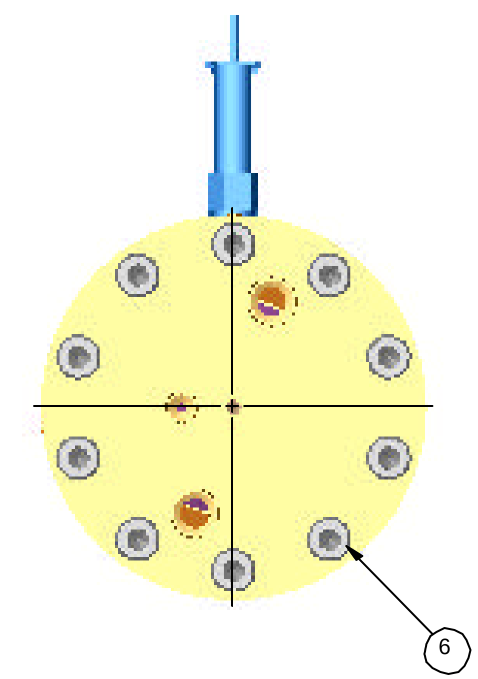
PROJECT # XXX-XXX

COMPANY CONFIDENTIAL

REVISIONS

REV	DESCRIPTION	DRWN	CHKD
B	CHANGE PRIOR TO RELEASE	TFR 07/14/03	TFR 07/14/03





ITEM NO.	QTY.	PART NO.	DESCRIPTION
1	1	T-10550	HOUSING, PRESSURE
2	1	T-10554	MANDREL
3	1	T-10551	CAP,PRESSURE HOUSING
4	1	80358	RETAINING RING, N5000-50
5	1	80671	HEADER 8-PIN
6	20	XXXXX	SCREW SHC 1/2-13 X 2.25 Mc 91251A721
7	2	XXXXX	O'RING -245 GALLAGER
8	3	LAB	TRANSDUCER, PRESSURE
9	1	T-10552	PLATE, ADAPTOR
10	1	T-10553	SPACER
11	8	80208	SCREW SHC 1/4-20 X .75
12	8	XXXXX	SCREW, SH FLAT 5/16-18 Mc91253A589
13	1	T-10564	CAP, FIXTURE

ITEM QTY P/N DESCRIPTION

13 8 XXXXX SCREW SH FLAT 5/16 X 18
McMASTER 91253A581 USED WHEN SPACER
IS REMOVED

PROPRIETARY & TRADE SECRET INFORMATION

ANY USE OR REPRODUCTION OF THIS DOCUMENT, IN PART OR WHOLE, WITHOUT EXPRESS WRITTEN PERMISSION OF APS TECHNOLOGY IS PROHIBITED

UNLESS OTHERWISE SPECIFIED

DIMENSIONS ARE IN INCHES & APPLY AFTER FINISH
PART MUST BE FREE OF BURRS AND/OR FLASH
BREAK SHARP EDGES APPROX .005
FILLET RADII .020 MAX

SURFACE FINISH 125 MAX.

APPROVAL	BY	DATE
DRAWN	RZASA	07/14/03
CHECKED	CAP	07/14/03
ENGRG	DEB	07/14/03
MFG	BWP	07/14/03

TOLERANCES
.XXX ±.005 FRACTIONS ±1/64
.XX ±.01 ANGLES ±2°

INTERPRET DWG PER ASME Y14.5M-1994
DIMS IN PARENTHESIS () ARE REF ONLY
DO NOT SCALE DWG

APS TECHNOLOGY
800 CORPORATE ROW
CROMWELL, CT 06416-2072

FIXTURE -
MR FLUID TEST

C DWG NO T-10549 REV B

SCALE: 1/2 SHEET 1 OF 1

MATERIAL:	HEAT TREAT:	FINISH:
N/A	N/A	N/A

APS CAD FILE NAME: T-10549 A FIXTURE, MR FLUID TEST.SLDDRW

NOTES: UOS

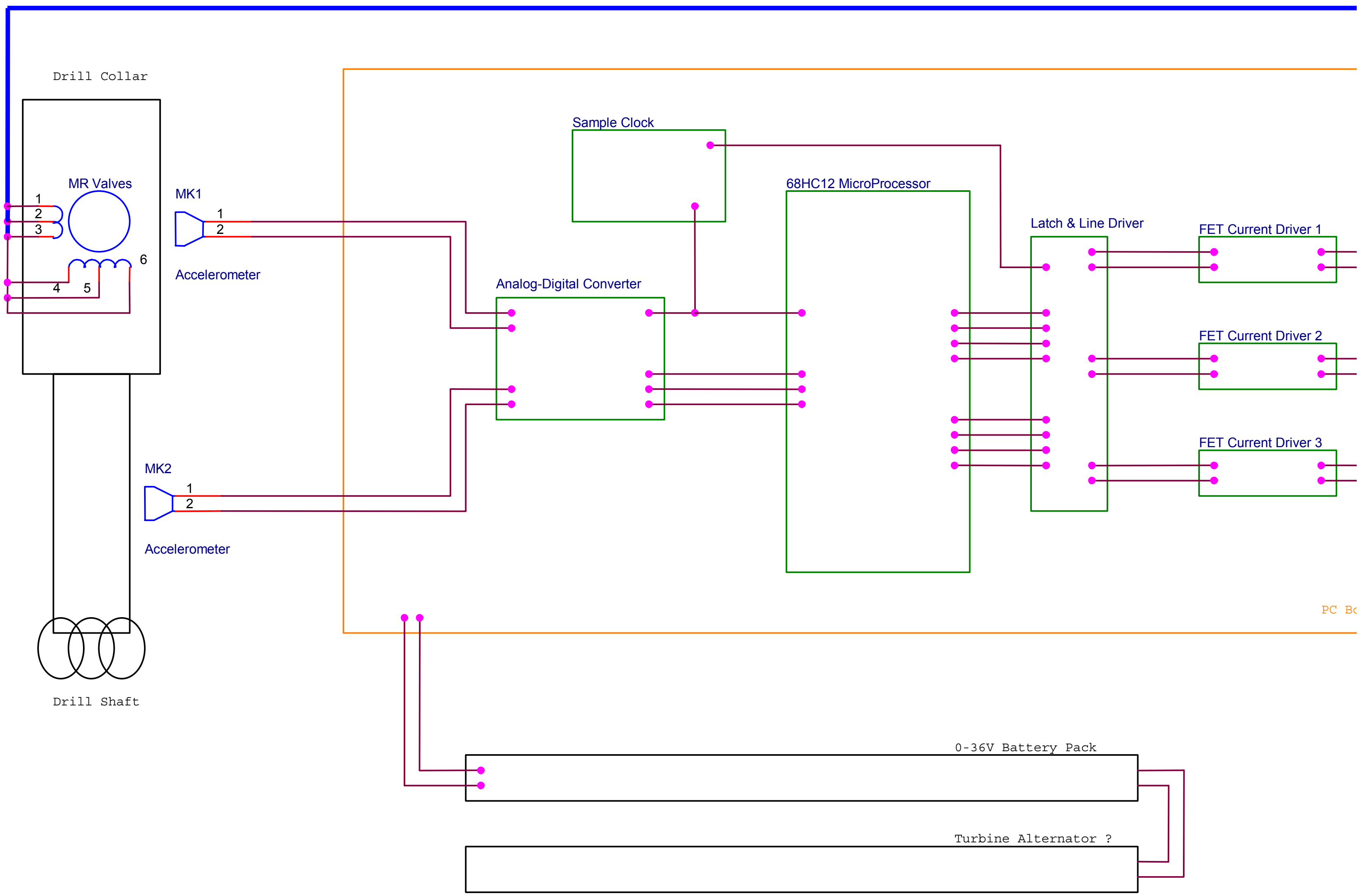


Figure 13: Electronics circuit diagram of test equipment

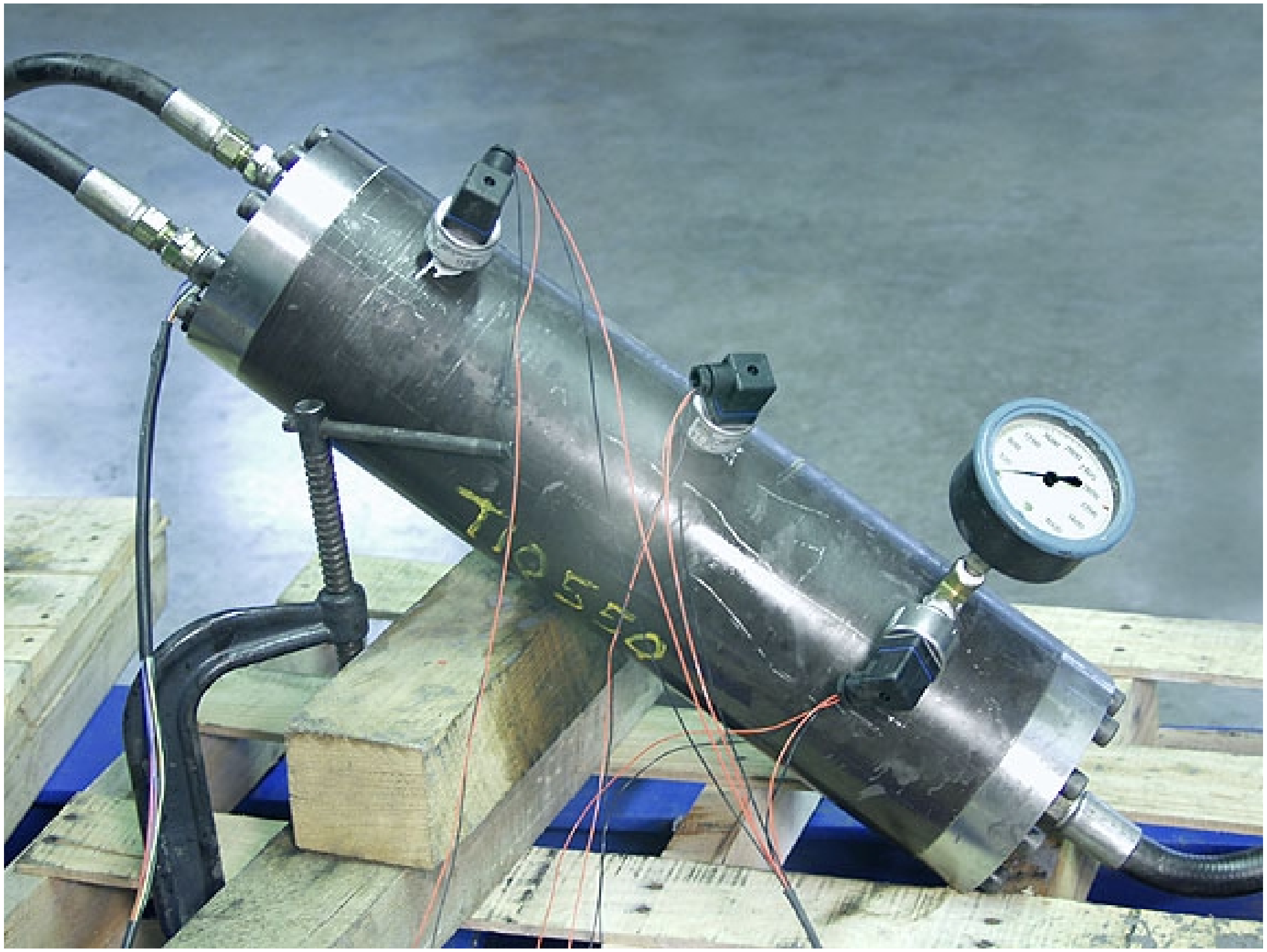


Figure 14: MR valve under test

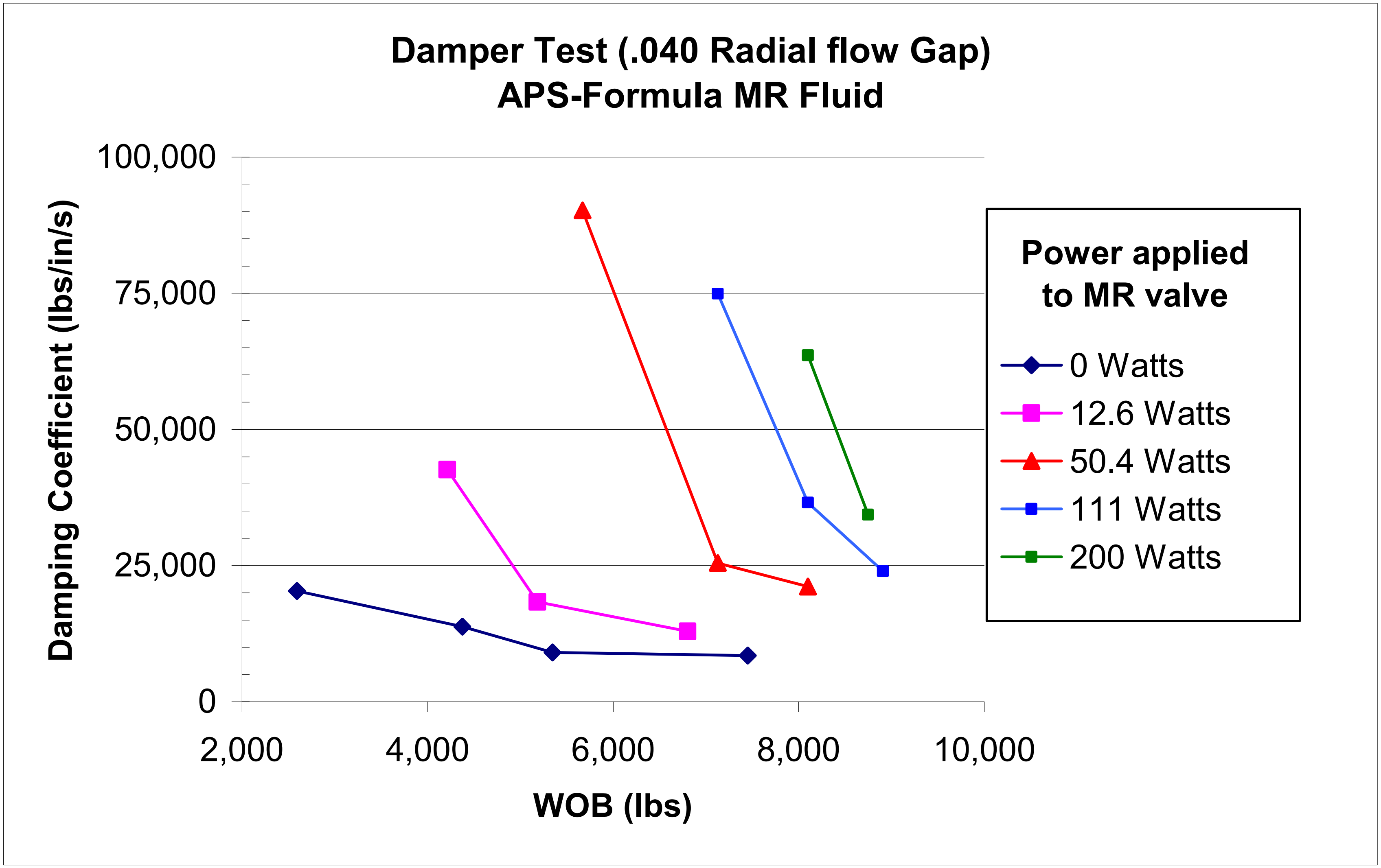


Figure 15: Results of initial damper testing

Figure 16: Assembling torsional bearing section

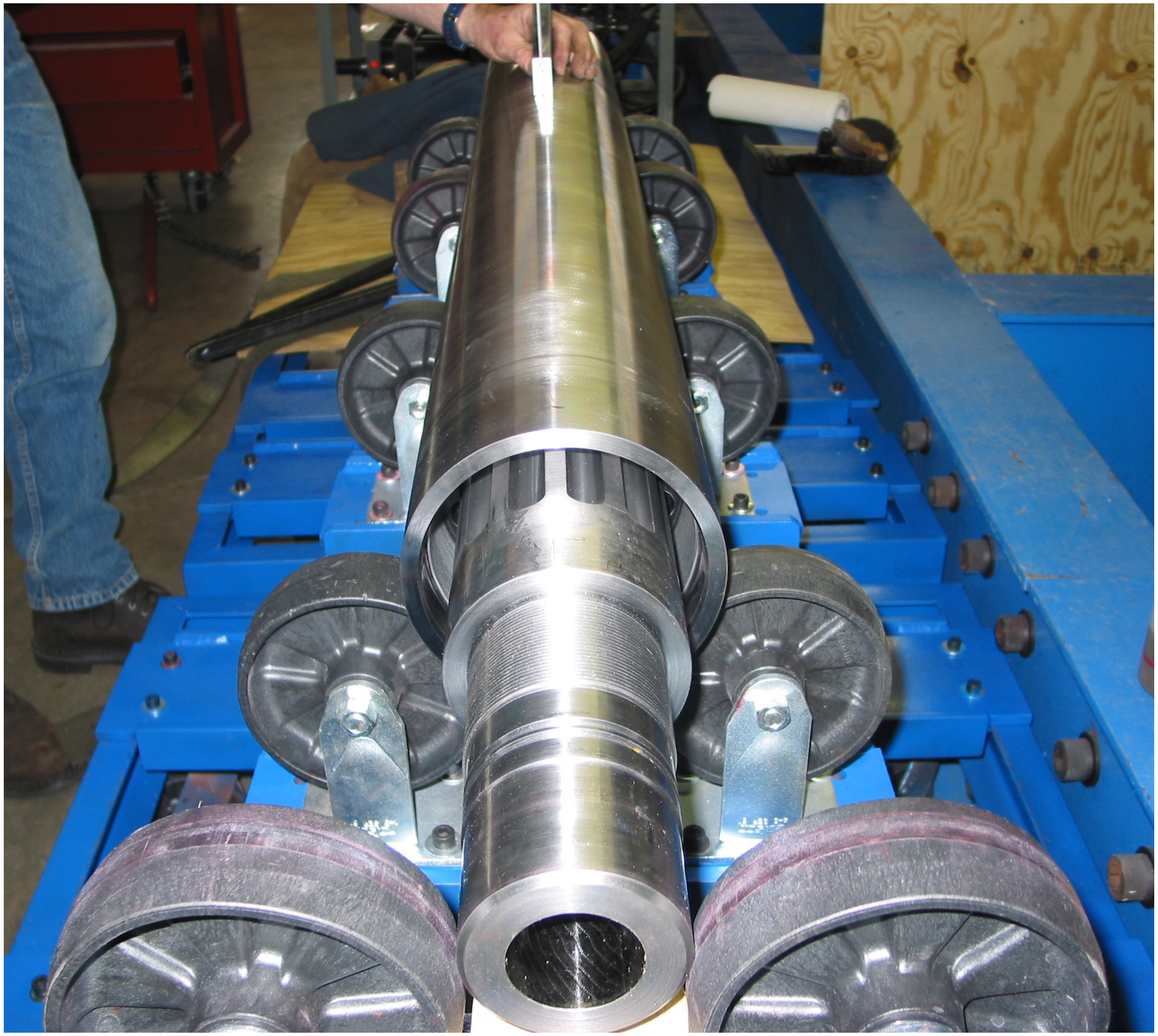


Figure 17: Prototype showing torsional bearing with races for ball bearings

Figure 18: Torsional bearing with ball bearings in place

Figure 19: Assembled bearing section

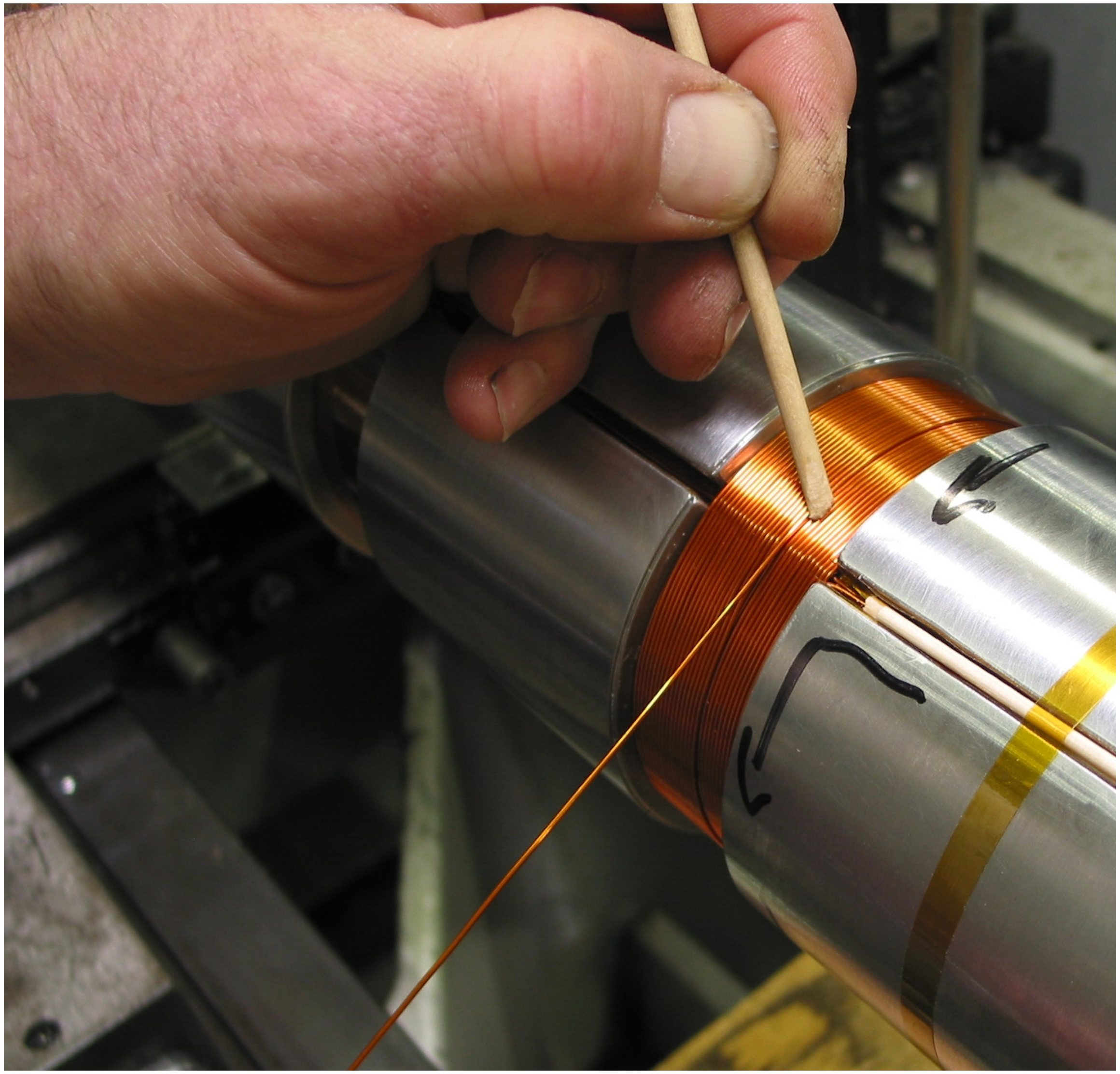


Figure 20: Winding MR magnet coil on damper

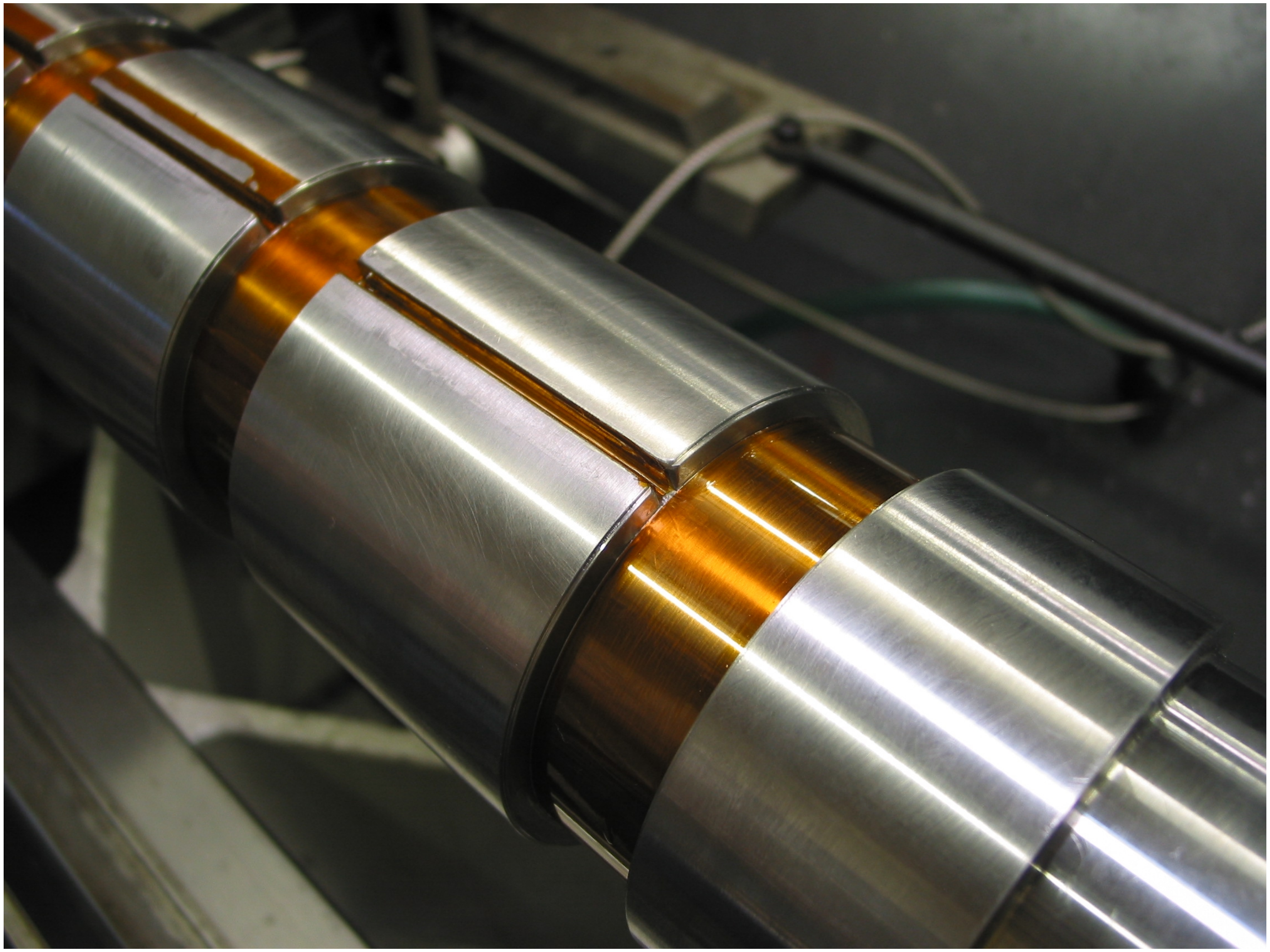


Figure 21: MR damper section showing magnet coils before potting

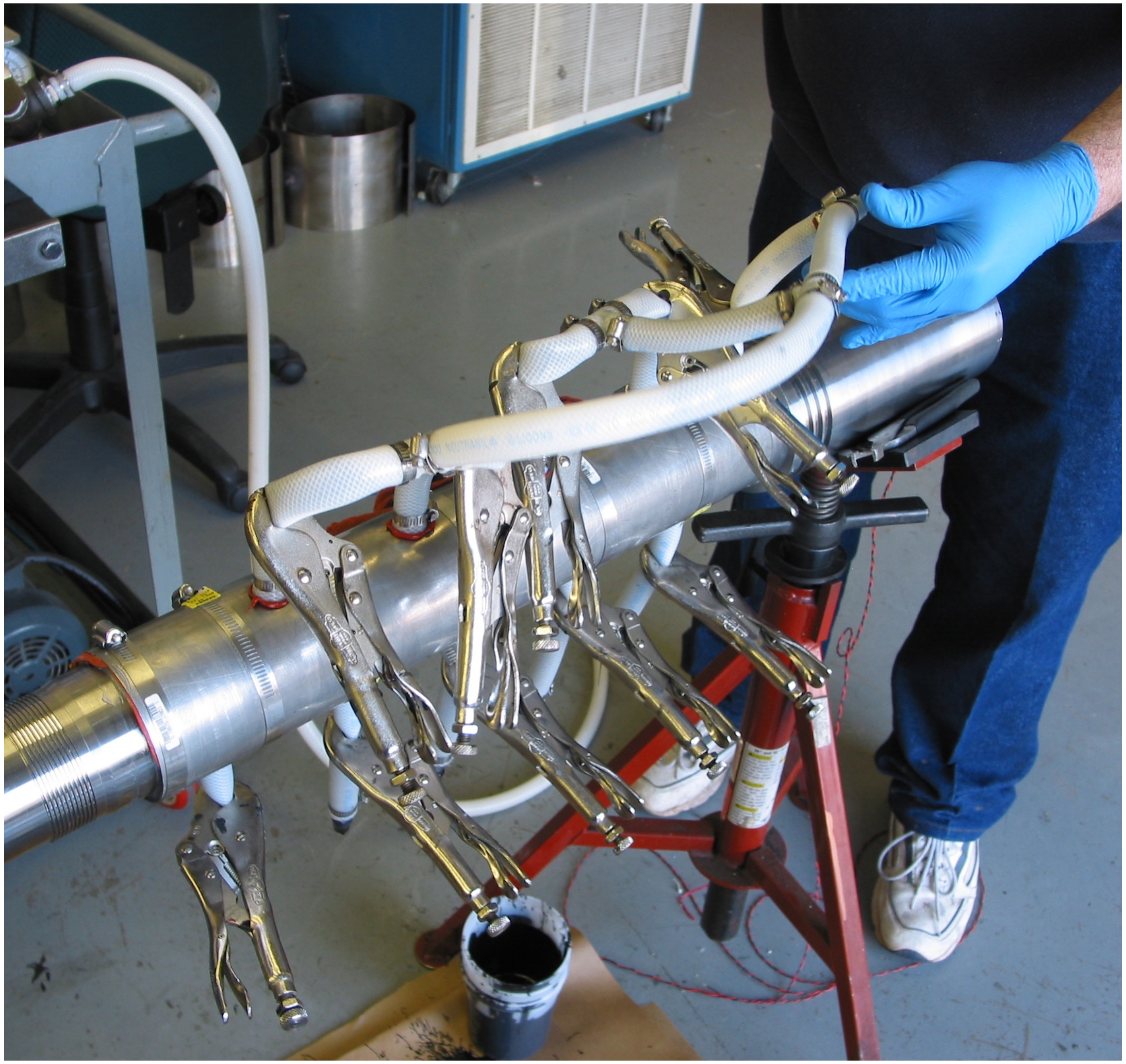


Figure 22: MR valve coils being potted

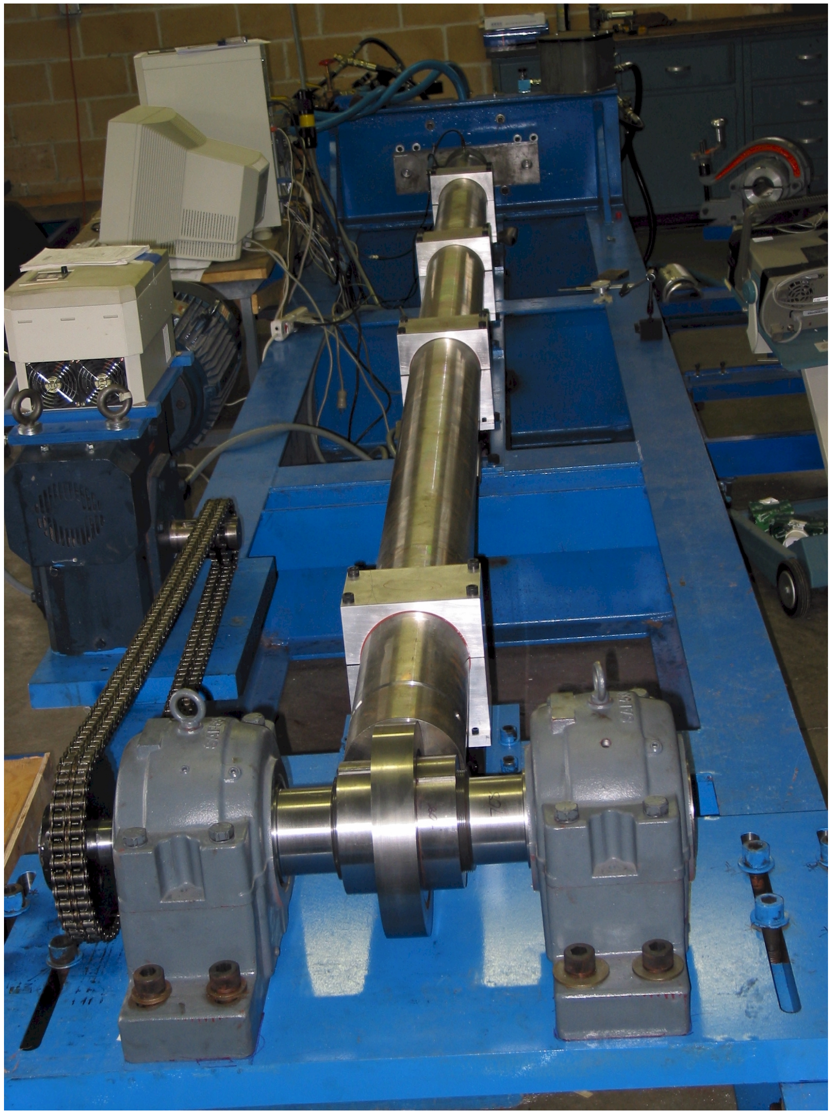


Figure 23: DVMCS laboratory prototype under testing on test bench

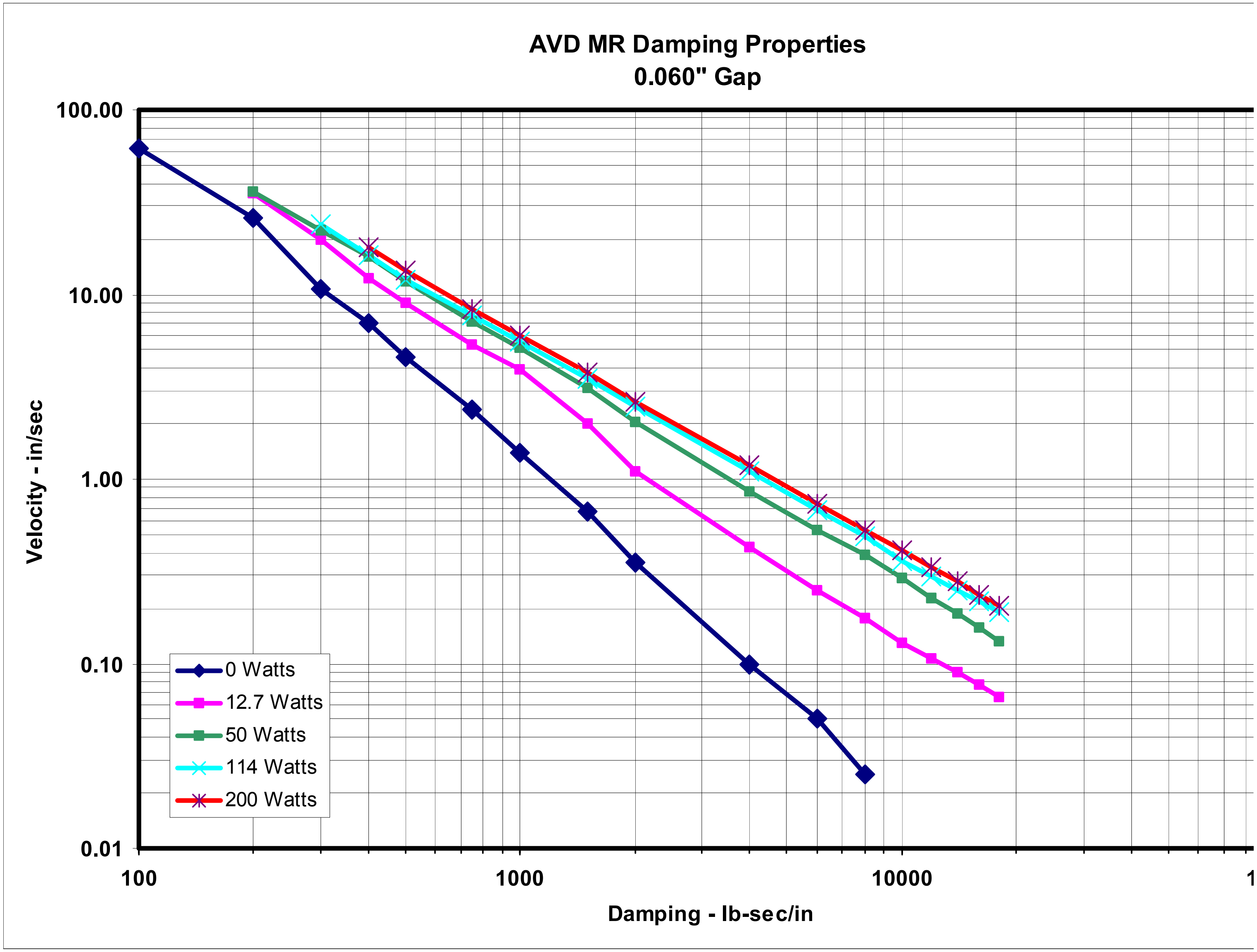


Figure 24: Damping vs. velocity

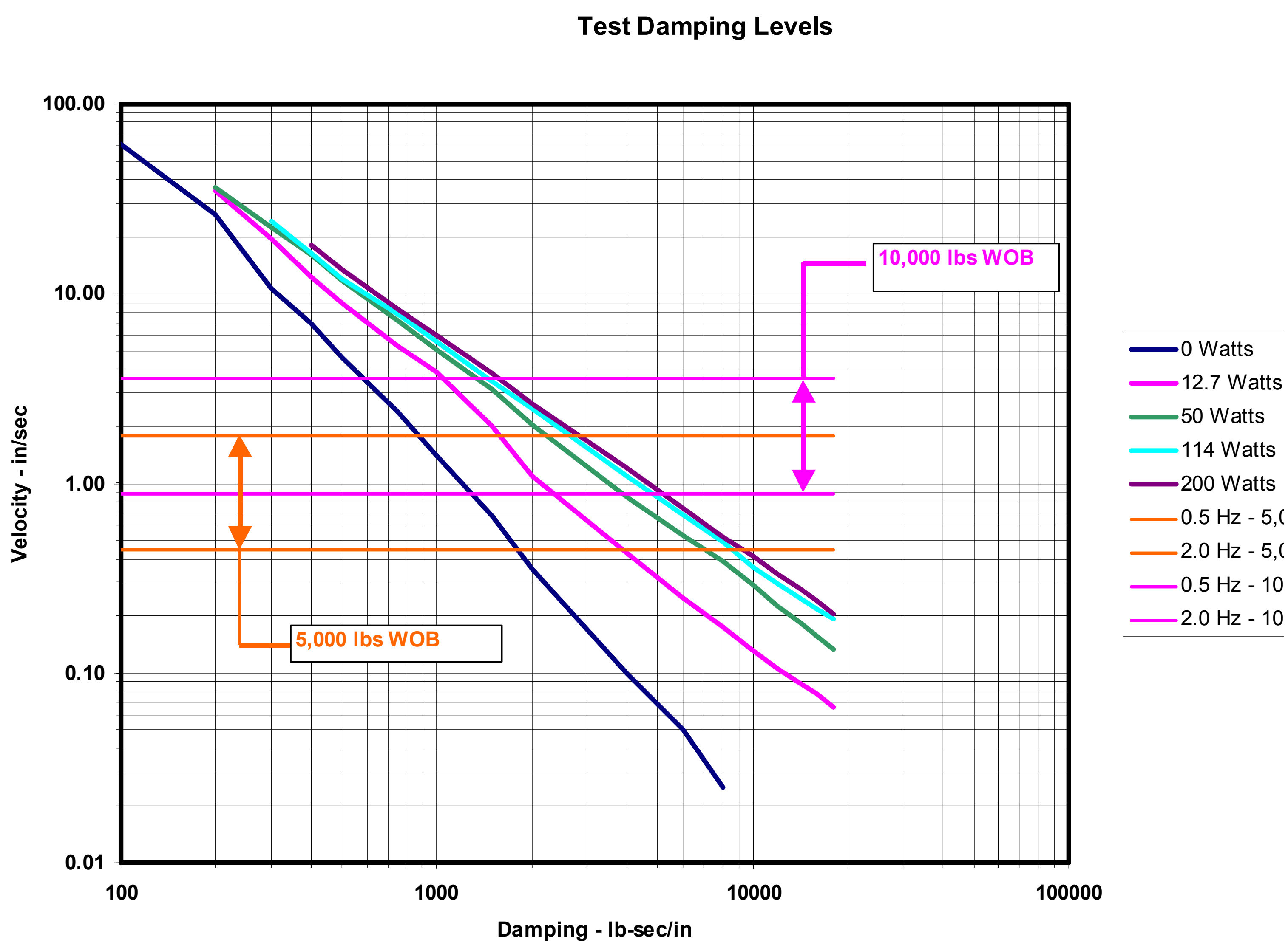


Figure 25: Test damping levels

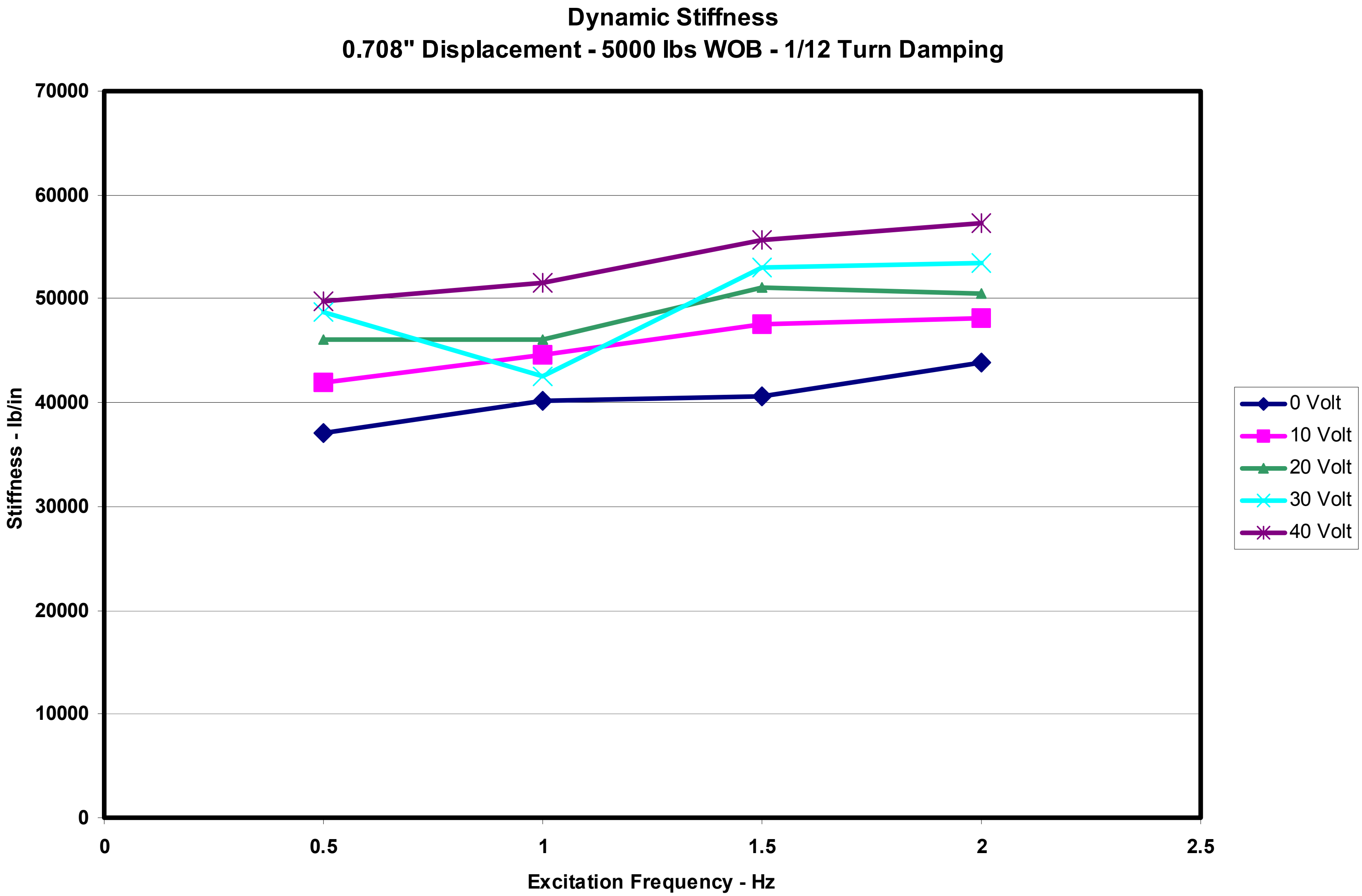


Figure 26: 5,000 lbs. WOB - Dynamic stiffness

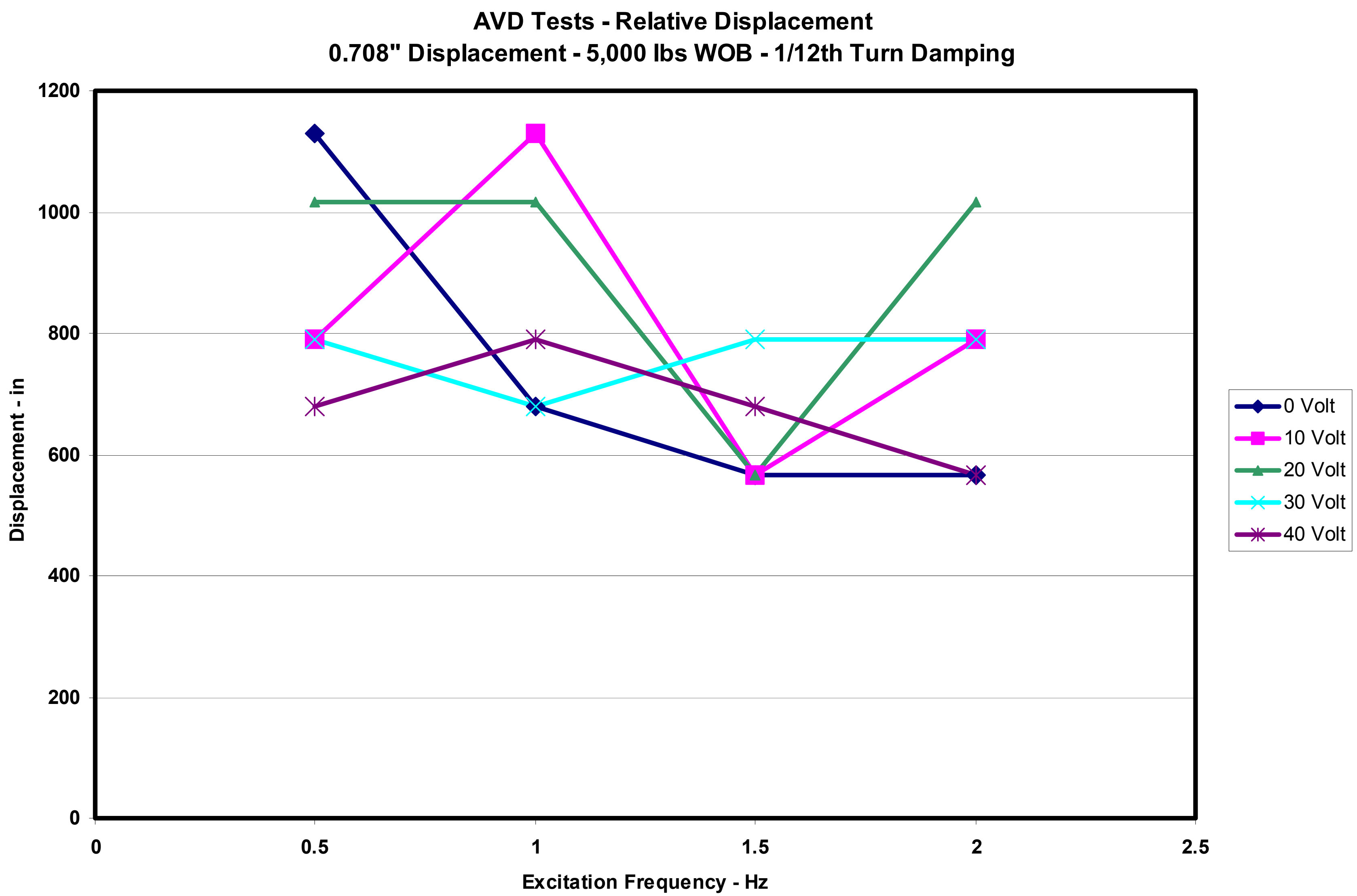


Figure 27: Relative displacements

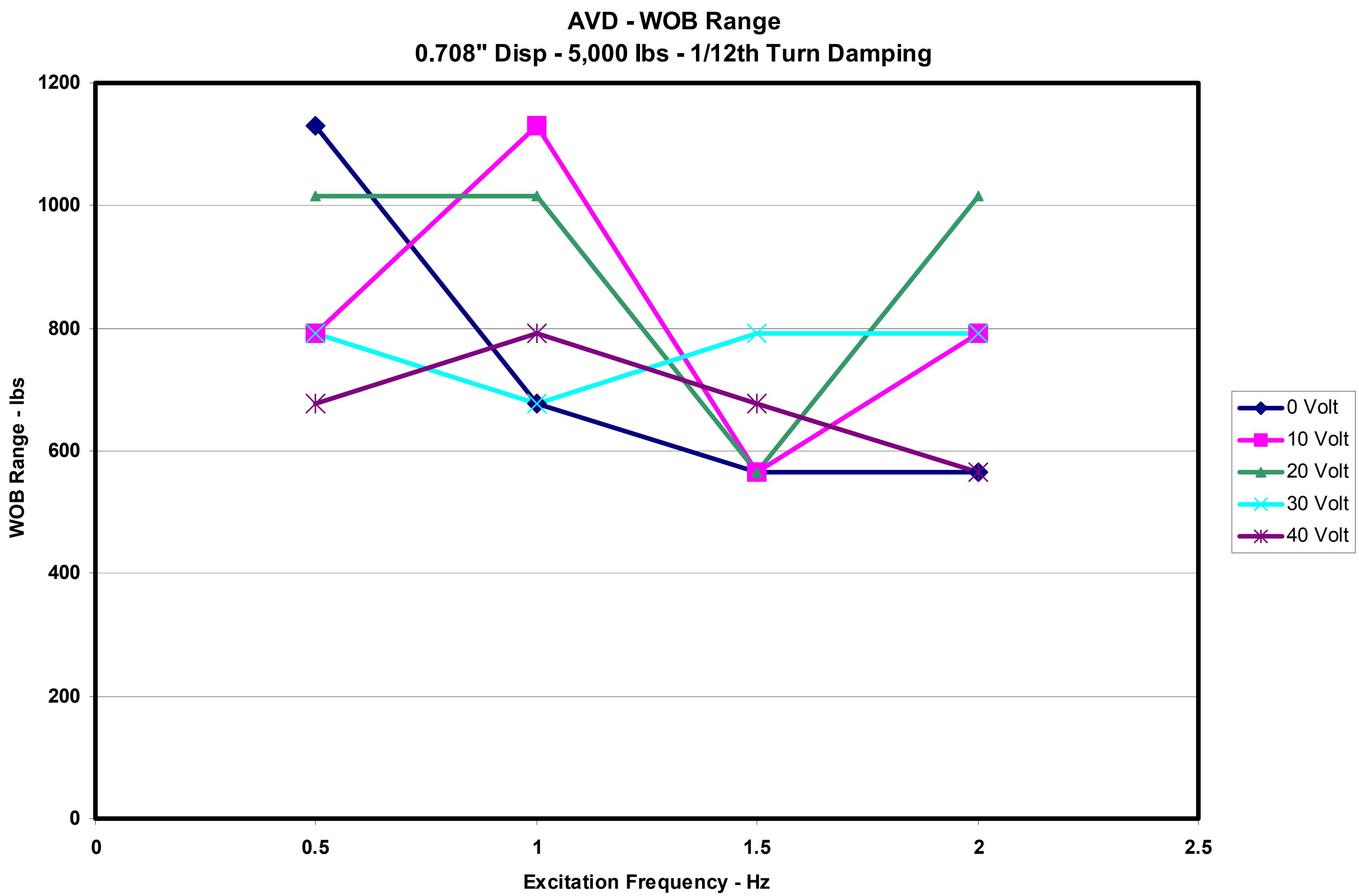


Figure 28: Effect of AVD on WOB vs. excitation frequency

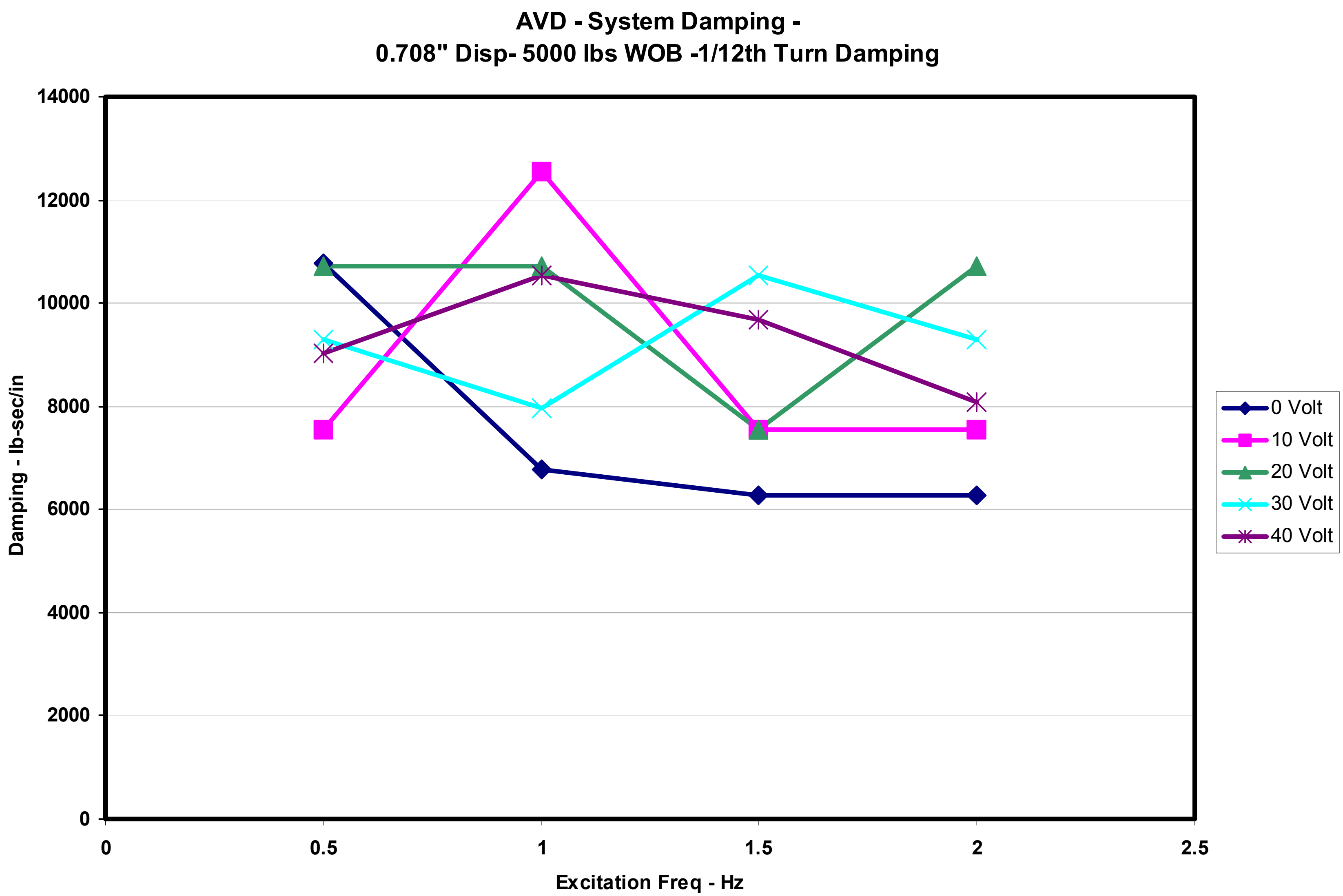


Figure 29: System damping vs. frequency

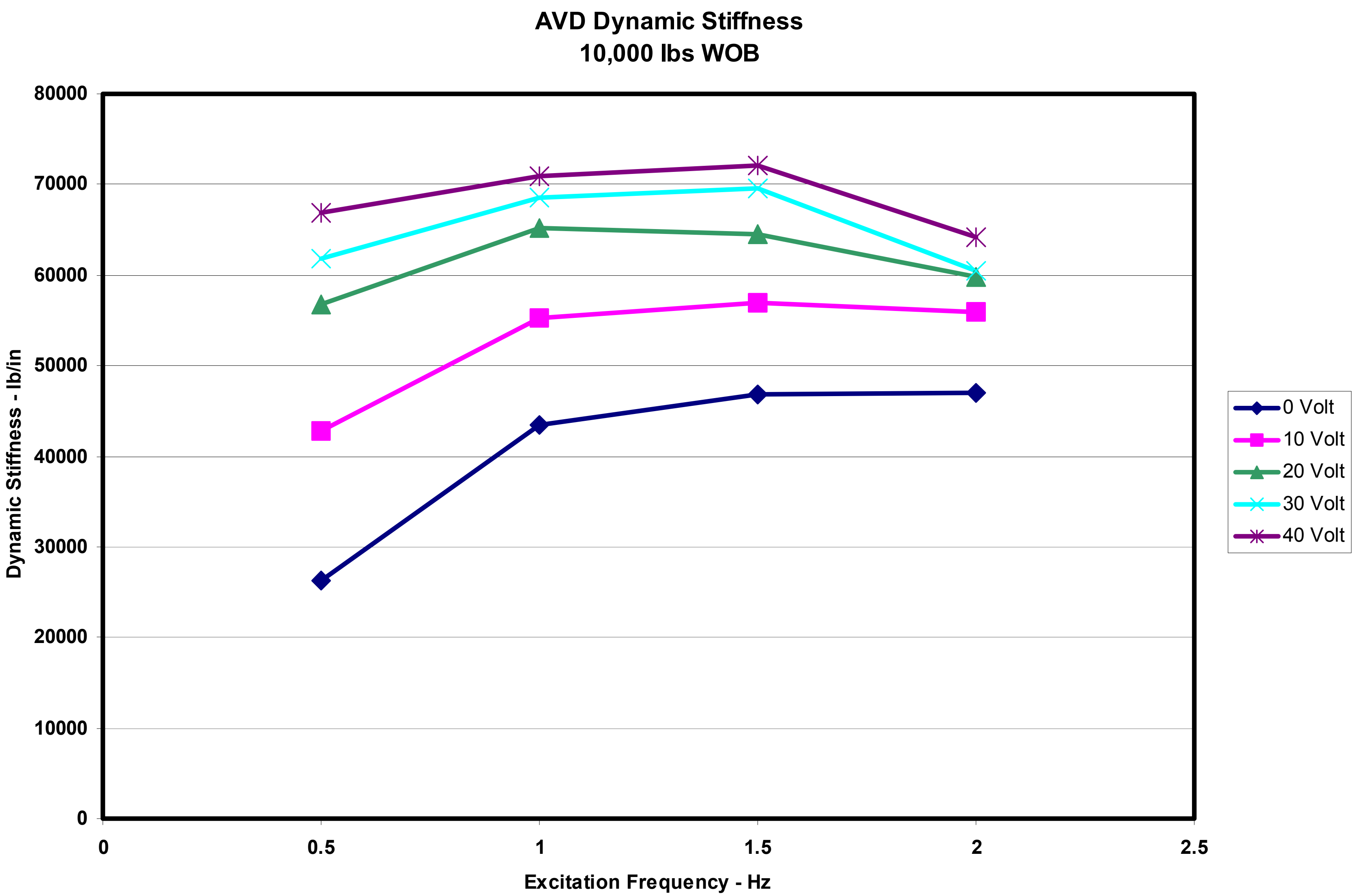


Figure 30: 10,000 lbs. WOB - AVD dynamic stiffness

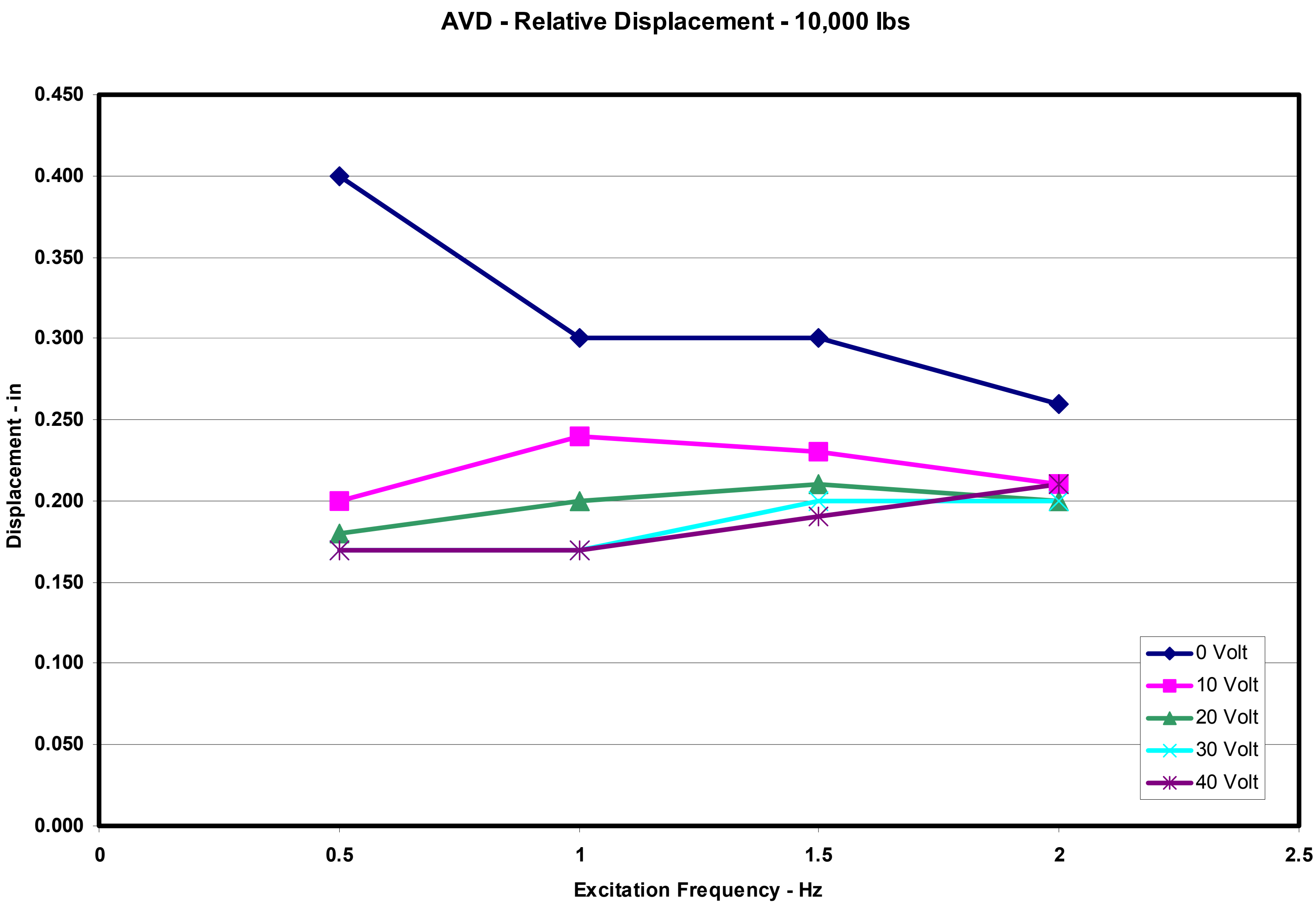


Figure 31: Relative displacement

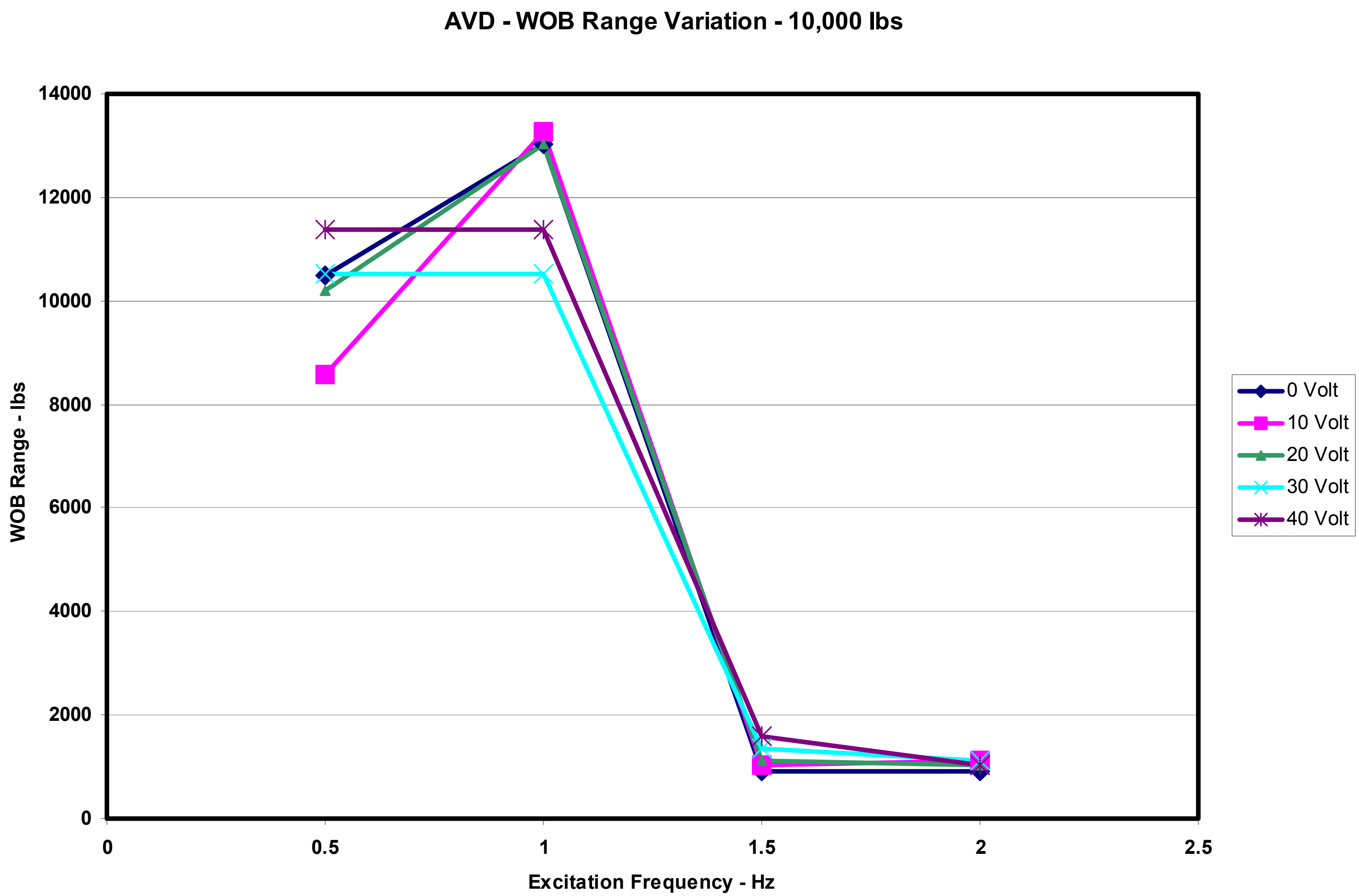


Figure 32: WOB range vs. frequency

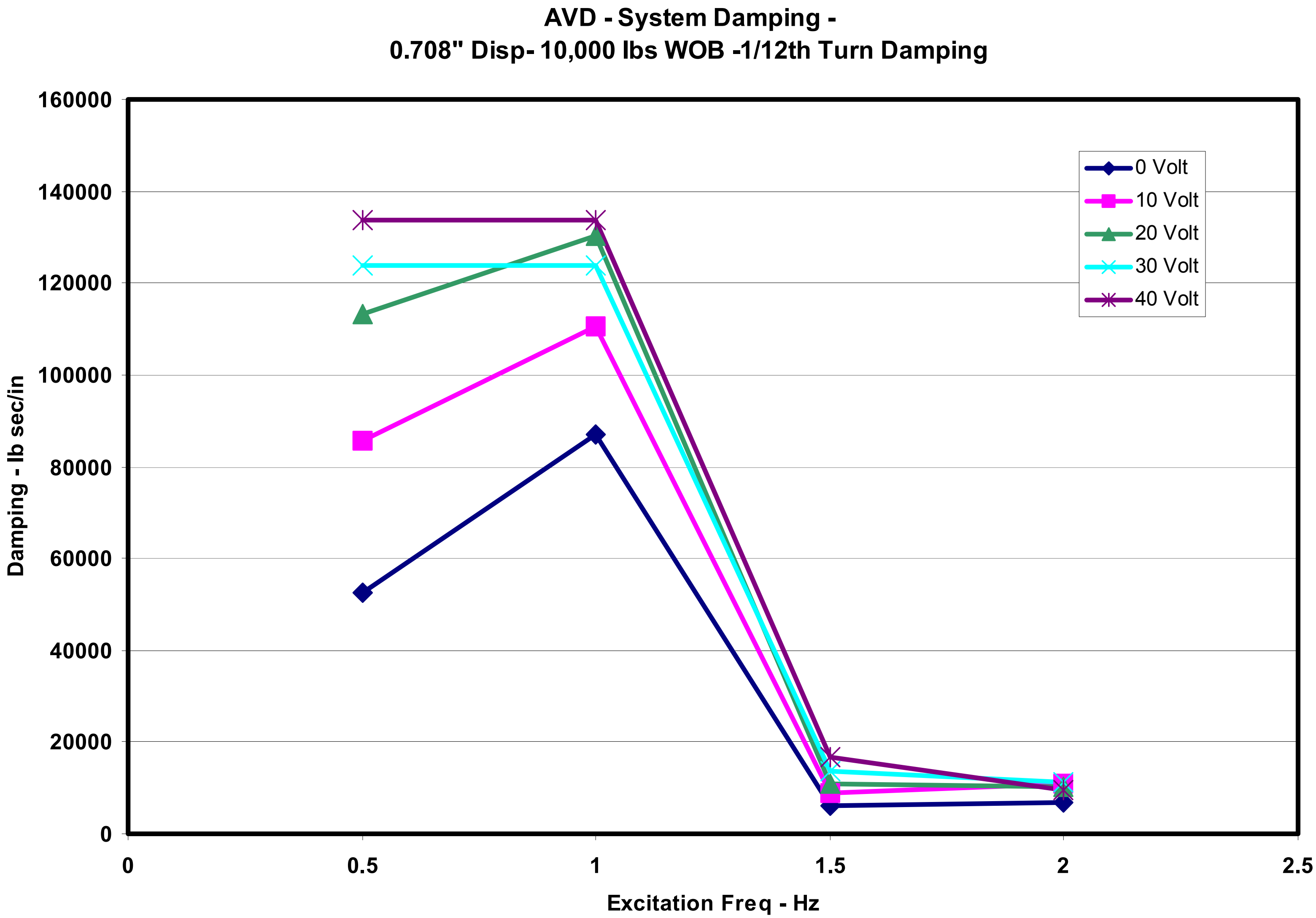


Figure 33: System damping

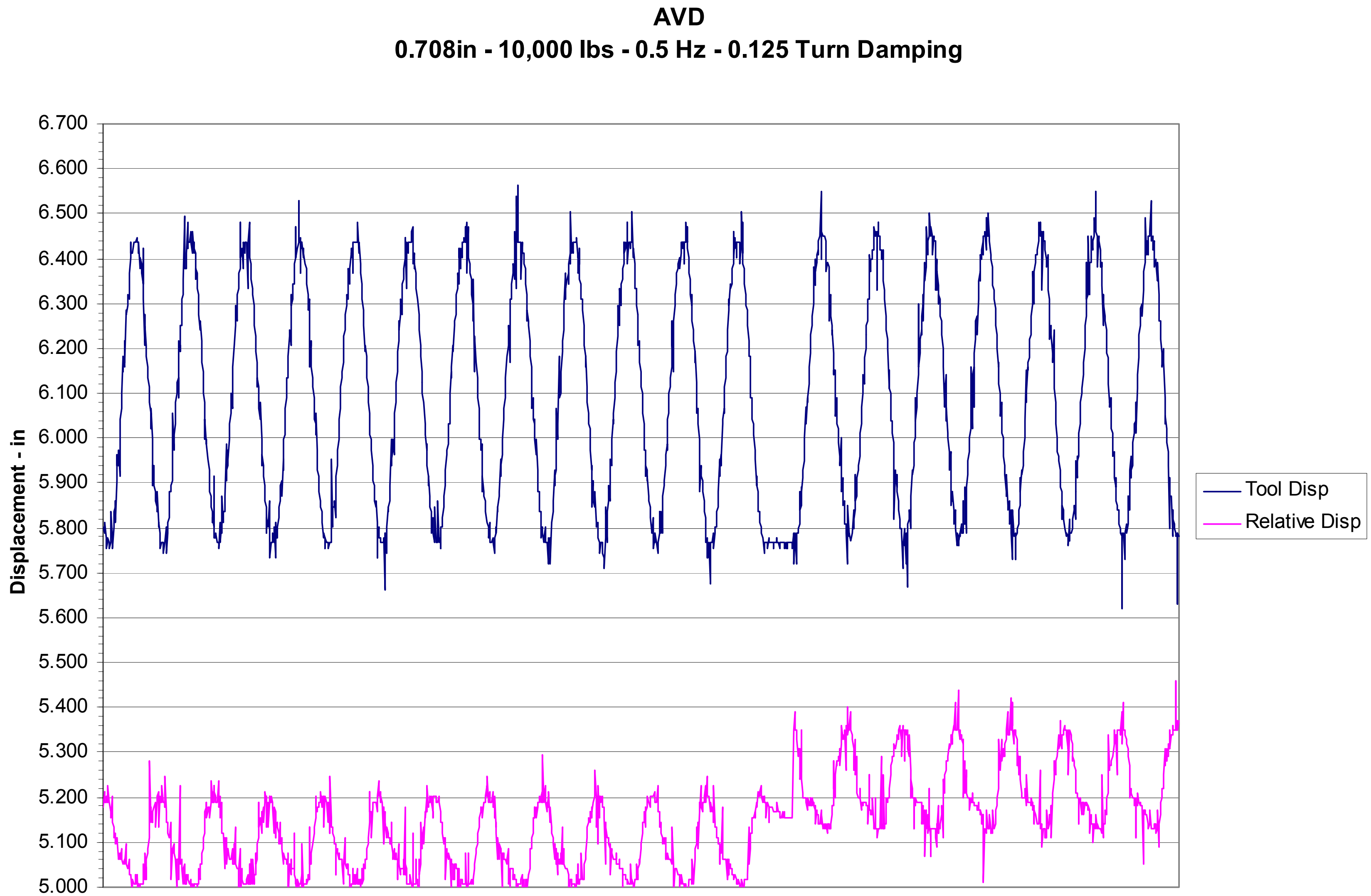


Figure 34: RAW DATA: Displacements

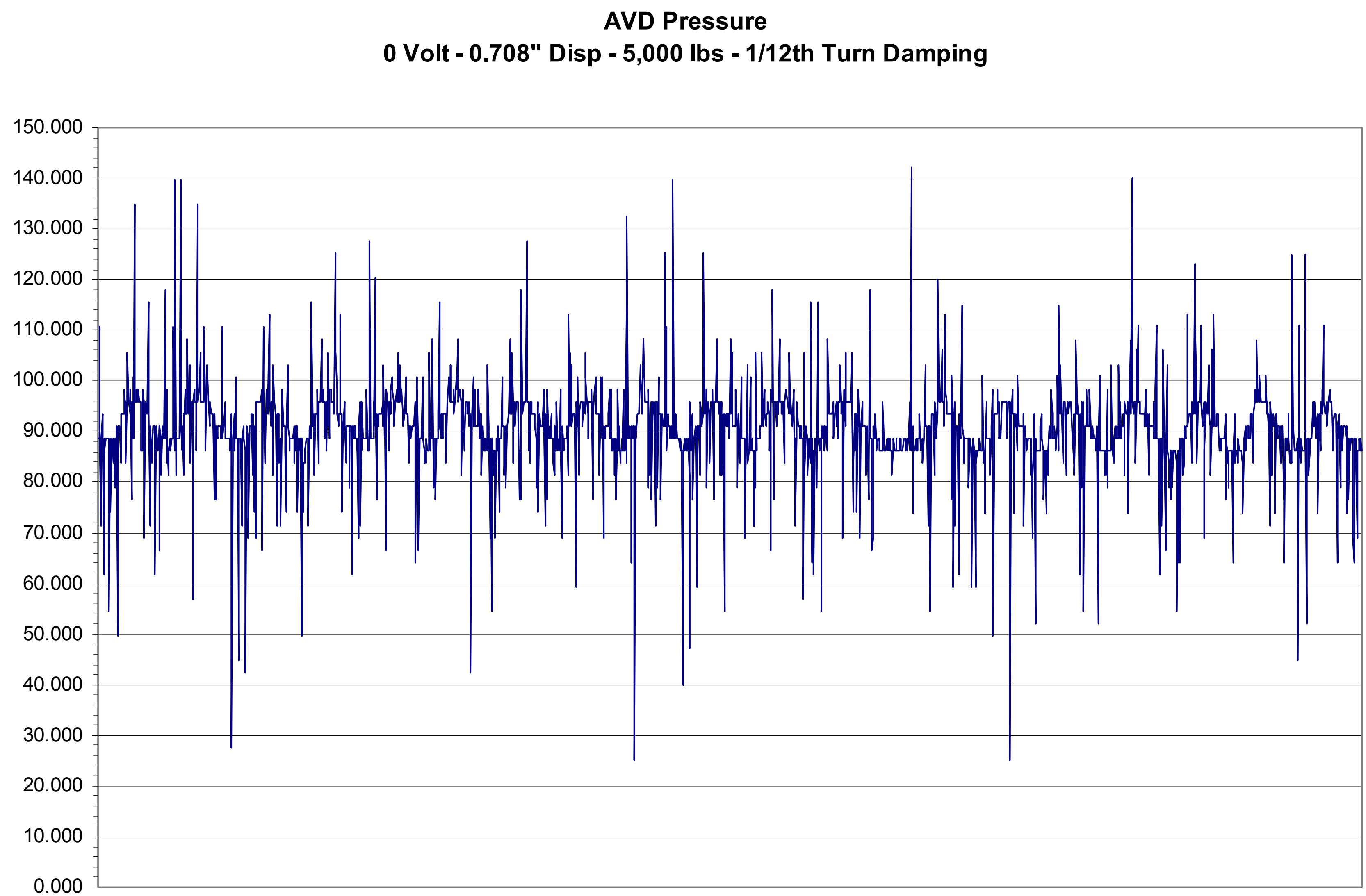


Figure 35: Pressure

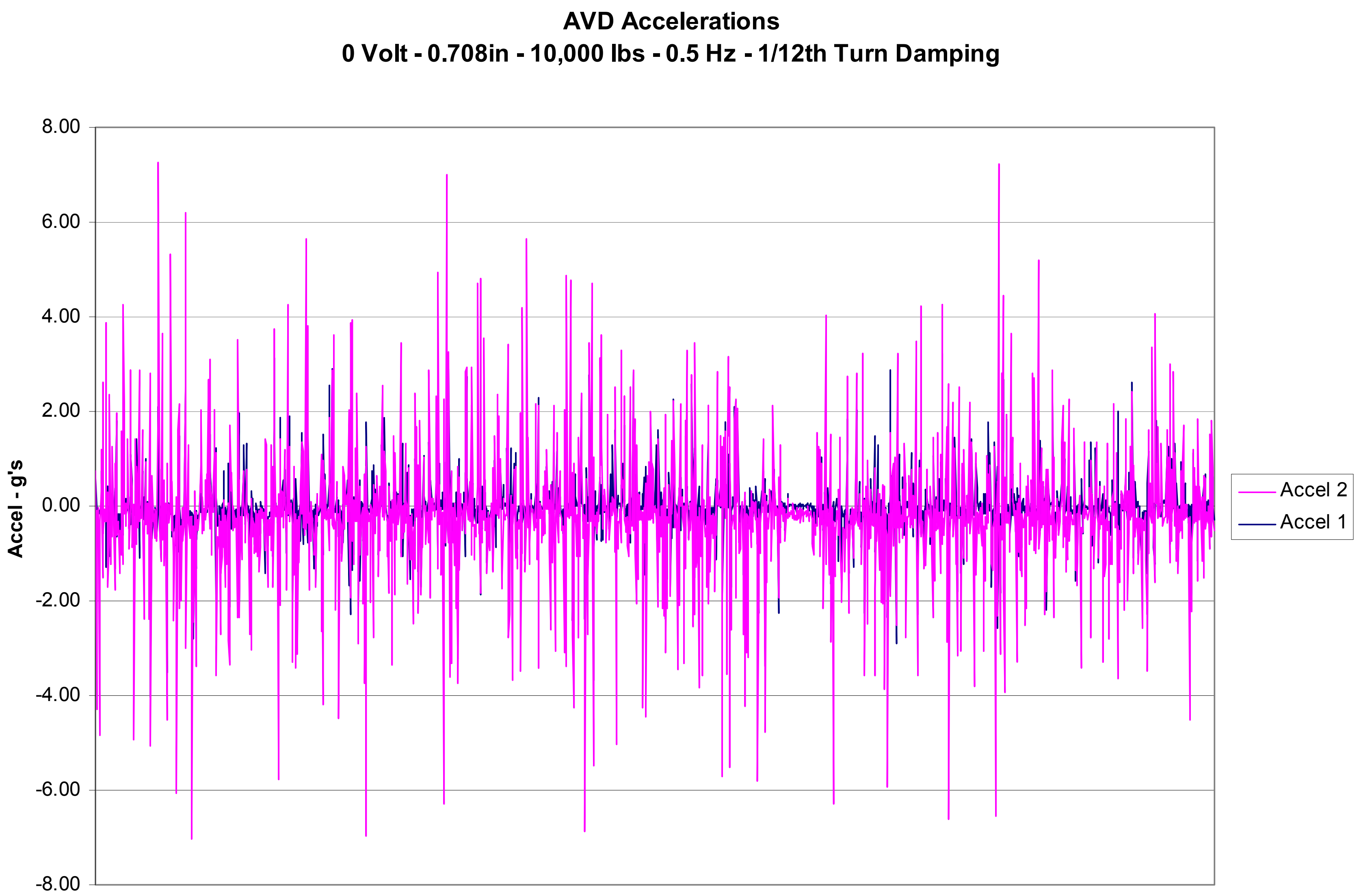
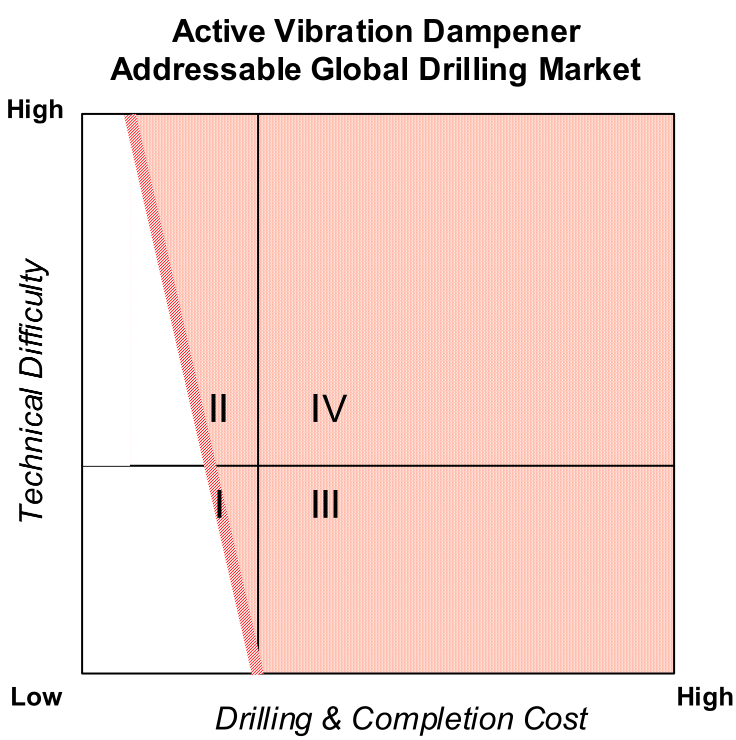


Figure 36: Accelerations

Appendix B: Marketing Study

In parallel with the technical development, APS Technology continues to refine its understanding of the application and market potential of the AVD system. In addition to the obvious advantages to the economics of ‘deep drilling’, as envisaged by the “DEEPTREK” program, AVD is perceived to have significant benefits to much of the current footage drilled worldwide.

A recent study of the worldwide drilling market, carried out in cooperation with Spears and Associates of Tulsa, Oklahoma, attempted to characterize the drilled footage by both cost & technical difficulty. Using the assumptions of that study DVMCS was determined to have significant applicability to over 61% of the projected footage for 2004.



APS/Spears Market Analysis

Total Applicable Footage: ***186M Ft***

Percentage of Overall Market: ***61%***

2004 Market Value: ***$356 Million***

The DVMCS is a unique technology in a massively lucrative market niche. No competitive technology currently exists or is known to be in development. Eliminating "Bit Bounce" has the potential to make major impacts on Rate of Penetration and the additional potential to significantly decrease trouble costs through significantly reducing harmful vibrations within critical elements of the BHA.

- By keeping the bit on bottom in continuous contact with the rock face, the instantaneous ROP is increased significantly. Initial modeling indicates that a 20% improvement in on-bottom cutting time can be achieved. Bits of all descriptions will spend more time actually cutting rock!
- Eliminating bit bounce will also greatly reduce the shock and vibration being imparted to all components of the BHA. Reduced shock and vibration will allow all downhole components – bit, mud motors, rotary steering devices, MWD/LWD tools – to function more reliably greatly increasing MTBFF (Mean-Time-Between-Field-Failures), greatly reducing the number of non-productive trips and thereby increasing effective on bottom drilling time. More time drilling and less time tripping equals increased 'effective' ROP.
- Vibration damage costs the MWD/LWD industry in excess of $500M per year in direct costs. If performance concessions are included in the calculation, the cost to the industry exceeds $750M. Costs to the operators in non-productive time are incalculable. Providing a new technology capable of reducing shock levels and increasing actual ROP will open major new revenue opportunities to APS.

One Trip Wells (OTW)

Many major oil companies believe that the next 'step-function' change in the economics of the industry will come about through the advent of "One Trip Wells". The ability to drill an entire well from top to bottom with one BHA in one trip requires many advances in all aspects of current drilling and completion practices. Two major drivers are:

- Rotary Steerable/Rotary Steerable Motors capable of maintaining full directional control of each phase of the well
- ***Significant increases in the reliability of the components that make up the rest of the BH, particularly MWD/LWD components****

(*APS emphasis)

The SPE took note of this development in a recent announcement of a Forum:

> Designer wells have become commonplace. Casing while drilling, real-time formation evaluation and 3D rotary steerable drilling are now being routinely applied. The single-trip gravel pack is emerging, "smart" wells are proliferating, the use of expandable tubulars continues to grow, and chemical applications for borehole integrity are evolving. While these technologies certainly address and solve specific problems, our industry has not advanced beyond a process that requires numerous round trips to drill and complete a well.

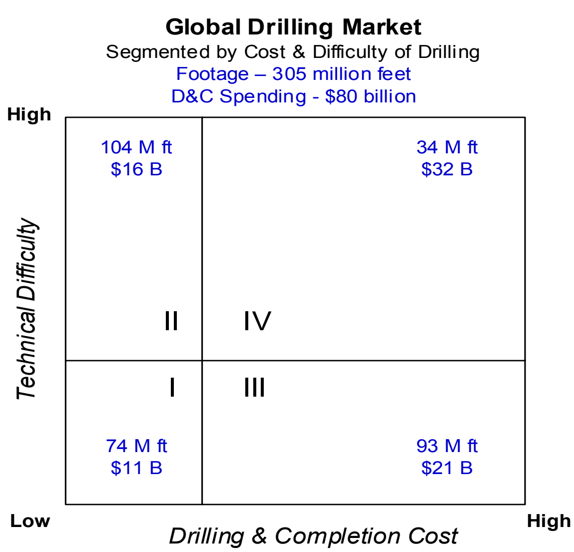
This forum will explore the integration of drilling and completion technologies and techniques, seeking to identify those that will be required to drill and complete a well in fewer trips, ultimately leading to a true one-trip well. Starting with the "average" well, where we have unlimited experience, know-how, confidence, and routinely and reliably construct such wells every day, we will extend the concept to more challenging opportunities, such as deepwater, and HP/HT environment.

Market Size

APS recently contracted Spears and Associates of Tulsa, OK to examine the market potential of a number of new products and services. The reports conclusions (DVMCS) are attached but some discussion of the methodology is in order.

Spears and Associates are well known for their reviews of industry activities and are respected forecasters of future activity. APS requested that Spears take their existing analysis of Rig Activity and Drilled Footage and break out the numbers in a more novel manner.

Total Footage, while useful as an indicator of overall industry health, is not indicative of how and where new technology might be applied. No single new technology or service is applicable to every foot drilled by the industry, nor is every segment of the industry expanding/contracting at the same rate. For the purposes of this study the drilled footage was broken out into four (4) basic categories:



SEGMENT I: Shallow land drilling as is typically found in North America, South America and other <5000' applications. Also includes some shallow water development drilling. These holes have low rig costs and are technically simple to drill.

SEGMENT II: More technically difficult wells, but still with fairly low drilling costs, such are found in the horizontal drilling plays of the Austin Chalk or in certain Middle East and African provinces.

SEGMENT III: Can include offshore development drilling from jack-ups, like the Gulf of Mexico Shelf, deep land drilling and some international offshore work. Rig costs are higher, but well profiles are still technically simple.

SEGMENT IV: These high cost, technically challenging wells include all deepwater drilling, high temperature/high pressure drilling and deep GOM Shelf work. Also includes remote or international exploration operations.

Within each category, drilled footage was further broken down into, "Straight-hole" and "Deviated" footage. Each new technology is then judged by its applicability to the footage drilled in that category.

The DVMCS is clearly applicable in all categories, even for some of the more technically challenging of the lower cost drilling projects.

Category IV: 100% of the High Cost-Technically Challenging wells would benefit significantly from the application of vibration damping services. ROP improvements provide rapid payback in high cost drilling environments while the vibration damping provides valuable protection for the increasingly sophisticated drilling tools deployed in technically challenging wells.

Category III: High drilling costs brought on by location and rig requirements will enable rapid returns from improved ROP, Although lower technology requirements will not enable maximum returns from the equipment protection aspects of the service, 100% of these wells will benefit from this technology.

Category II: 50% of wells drilled in this environment will benefit from DVMCS Services. While ROP benefits might not be as readily calculated, vibration protection will be beneficial for LWD and other advanced drilling tools.

Category I: 10% of wells drilled in this environment could benefit from DVMCS services. ROP benefits are minimized from low overall costs, vibration protection will be greatly beneficial MWD & other advanced drilling tools.

APS Marketing Plans

APS has neither the corporate structure nor the international network to deploy DVMCS as rapidly as the market will demand. Consequently, APS is actively seeking a commercial partner with a more global distribution.

In addition to providing funds to accelerate development and enable an accelerated prototype/pilot build in the USA, the ideal partner will also have an existing network of support bases throughout the USA (to support initial introduction) and internationally (to facilitate growth).

Discussions have been initiated with major service companies and bit manufacturers. These presentations have been very favorably received and we have been encouraged to maintain contact. Publication of a full set of test data from Phase I results is expected to ‘firm up’ interest and takes discussions to the next level

Environmental Analysis

The only component of the DVMCS which could conceivably be released into the environment is the oil used in the tool. The DVMCS contains about 8 liters of Mobil SHC 600 Series oil. About 2 liters of this has iron particles added to convert it to an MR fluid. The MSDS for this oil lists it “practically non-toxic” and “practically non-irritating” in all categories. While release of this oil downhole could conceivable contaminate the drilling mud, the quantities involved are essentially negligible and pose no hazard. The iron particles in the MR fluid are commonly found in the drilling mud, either as a ferrite additive or the byproduct of wear from drill bits and other components.

By improving drilling efficiency, eliminating some trips and reducing the time to drill a well, the use of the DVMCS could reduce the environmental impact of oil and gas exploration and production, but this effect is virtually impossible to quantify.

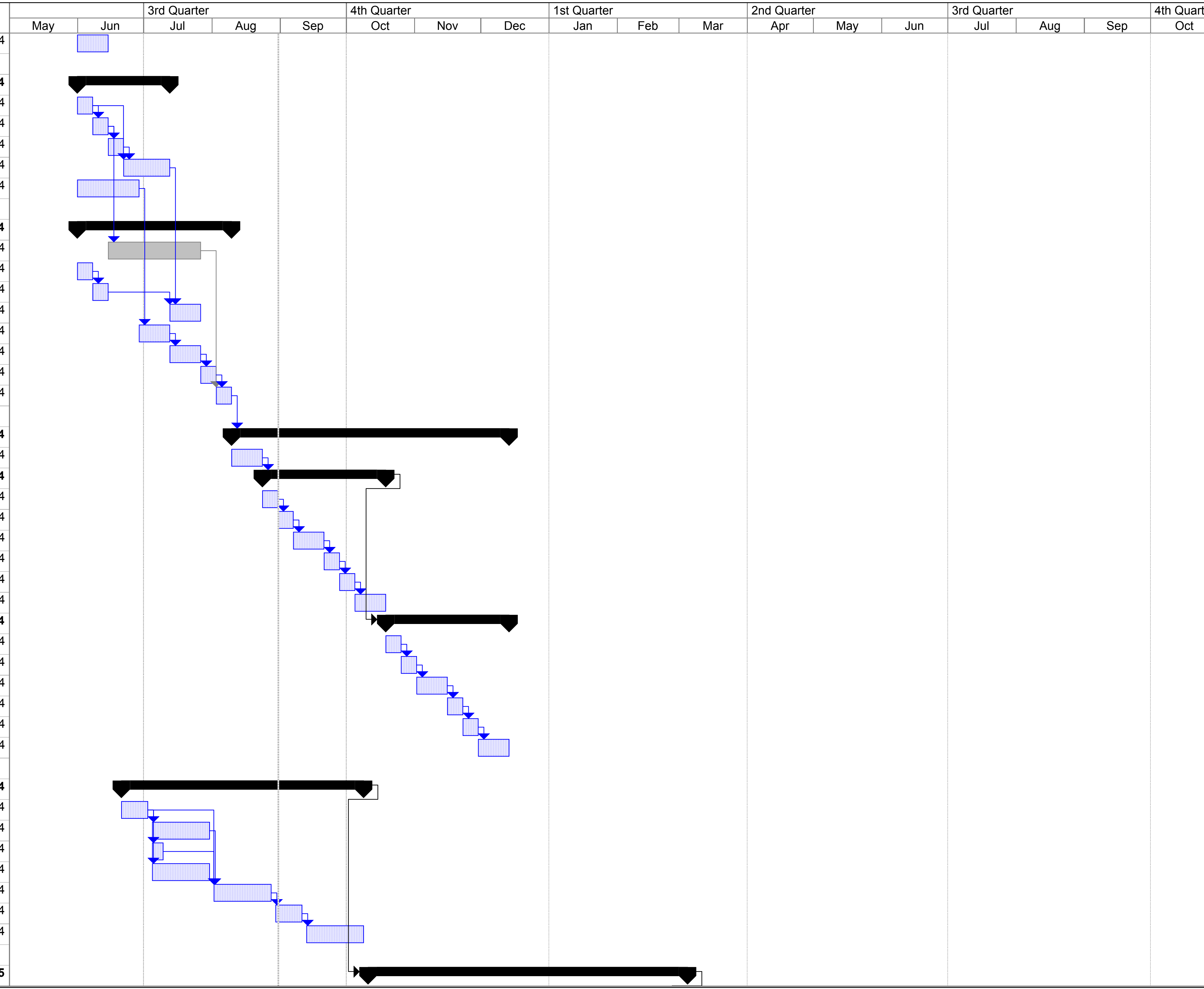
Appendix C: Preliminary Project Plan for Phase II

The project plan for Phase II has some small adjustments from that submitted with the original project proposal, namely:

- Task 10: Development and testing plan has been removed as redundant, since the plan developed at the end of Phase I (below) already exists.
- Task 13: Laboratory prototype construction and testing was partially completed as part of the damper testing in Phase I. We have reworked this task to reflects its new role as a revision and expansion of the earlier tests.

A Gantt chart for Phase II is given below.

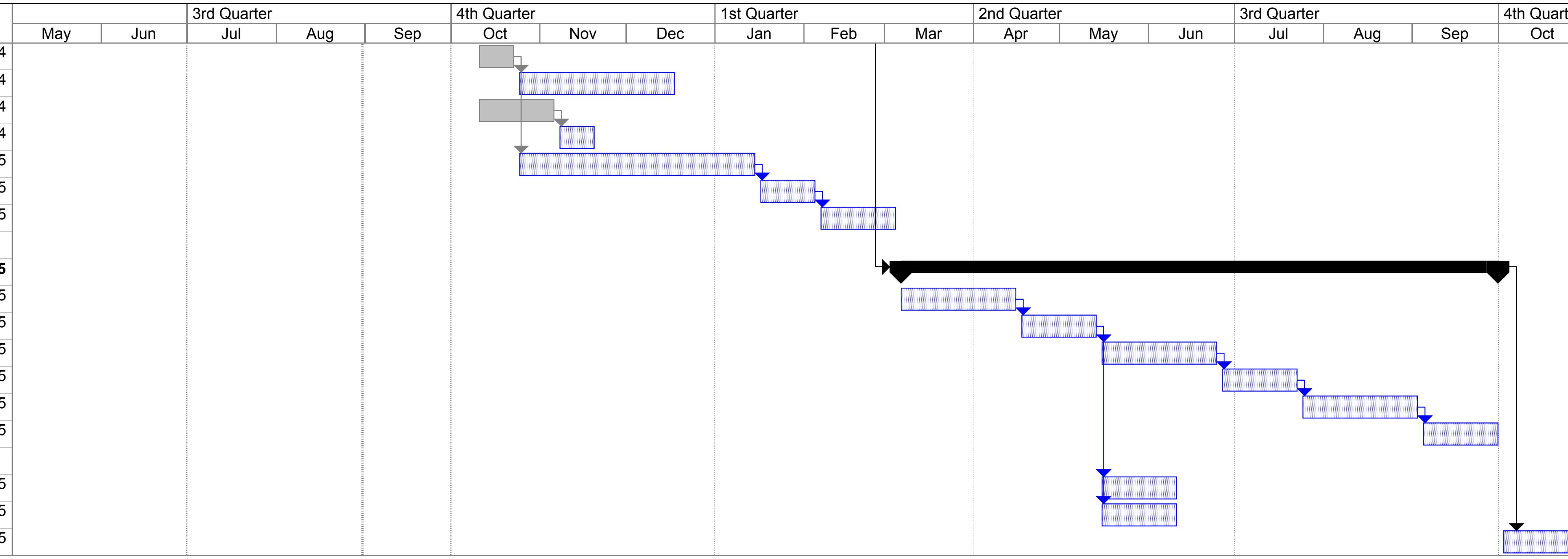
ID	Task Name	Duration	Finish
1	Final Phase II Plan	2 wks	Mon 6/14/04
2			
3	**Update Lab Prototype Design**	**30 days**	**Mon 7/12/04**
4	Include load cells	1 wk	Mon 6/7/04
5	Redesign Belleville stack	1 wk	Mon 6/14/04
6	Collar through ports	1 wk	Mon 6/21/04
7	Detail	3 wks	Mon 7/12/04
8	Feedback circuit	4 wks	Mon 6/28/04
9			
10	**Rebuild Lab Prototype**	**50 days**	**Mon 8/9/04**
11	Order Bellevilles	6 wks	Mon 7/26/04
12	Disassemble	1 wk	Mon 6/7/04
13	Clean	1 wk	Mon 6/14/04
14	Install load cells	2 wks	Mon 7/26/04
15	Install feedback circuit	2 wks	Mon 7/12/04
16	Reassemble	2 wks	Mon 7/26/04
17	Fill	1 wk	Mon 8/2/04
18	Shake out	1 wk	Mon 8/9/04
19			
20	**Laboratory Testing**	**90 days**	**Mon 12/13/04**
21	First Gap Setting	2 wks	Mon 8/23/04
22	**Second Gap Setting**	**40 days**	**Mon 10/18/04**
23	Disassemble	1 wk	Mon 8/30/04
24	Clean	1 wk	Mon 9/6/04
25	Reassemble	2 wks	Mon 9/20/04
26	Fill	1 wk	Mon 9/27/04
27	Shake out	1 wk	Mon 10/4/04
28	Test	2 wks	Mon 10/18/04
29	**Third Gap Setting**	**40 days**	**Mon 12/13/04**
30	Disassemble	1 wk	Mon 10/25/04
31	Clean	1 wk	Mon 11/1/04
32	Reassemble	2 wks	Mon 11/15/04
33	Fill	1 wk	Mon 11/22/04
34	Shake out	1 wk	Mon 11/29/04
35	Test	2 wks	Mon 12/13/04
36			
37	**Field prototype design**	**80 days**	**Fri 10/8/04**
38	Top level	2 wks	Fri 7/2/04
39	Instrument module	4 wks	Fri 7/30/04
40	Turbine-alternator interface	1 wk	Fri 7/9/04
41	Electical interconnection	4 wks	Fri 7/30/04
42	Seals and bearings	4 wks	Fri 8/27/04
43	Linear displacement transducer	2 wks	Fri 9/10/04
44	Detail	4 wks	Fri 10/8/04
45			
46	**Field Prototype Manufacture**	**105 days**	**Fri 3/4/05**



Project: AVD Phase II
Date: Tue 8/31/04

Task | Split | Progress | Milestone | Summary | Project Summary | External Tasks | External Milestone | Deadline

ID		Task Name	Duration	Finish
47		Order materials	2 wks	Fri 10/22/04
48		Machine subs	8 wks	Fri 12/17/04
49		Order electronics boards	4 wks	Fri 11/5/04
50		Assemble and test boards	2 wks	Fri 11/19/04
51		Machine damper parts	12 wks	Fri 1/14/05
52		Assemble	3 wks	Fri 2/4/05
53		Laboratory test	4 wks	Fri 3/4/05
54				
55		**Field Testing**	**150 days**	**Fri 9/30/05**
56		First test - Drilling laboratory	6 wks	Fri 4/15/05
57		Analysis & possible rework	4 wks	Fri 5/13/05
58		Second test - shallow well	6 wks	Fri 6/24/05
59		Analysis & possible rework	4 wks	Fri 7/22/05
60		Third test	6 wks	Fri 9/2/05
61		Analysis	4 wks	Fri 9/30/05
62				
63		Revised economic, market analysis	4 wks	Fri 6/10/05
64		Revised financing plan	4 wks	Fri 6/10/05
65		Final report	4 wks	Fri 10/28/05



Project: AVD Phase II
Date: Tue 8/31/04

Task | Progress | Summary | External Tasks | Deadline
Split | Milestone | Project Summary | External Milestone