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Final Report

DOWNHOLE VIBRATION MONITORING & CONTROL SYSTEM

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DOWNHOLE VIBRATION MONITORING & CONTROL SYSTEM FINAL REPORT

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ABSTRACT

The objective of this program is to develop a system to both monitor the vibration of a bottomhole assembly, and to adjust the properties of an active damper in response to these measured vibrations. The key feature of this system is its use of a **magnetorheological fluid** (MRF) to allow the damping coefficient to be changed extensively, rapidly and reversibly *without the use of mechanical valves*, but only by the application of a current.

Phase I of this program, which entailed modeling and design of the necessary subsystems and design, manufacture and test of a full laboratory prototype, was completed on May 31, 2004.¹ Much of the effort was devoted to the design and testing of the MRF damper, itself.

The principal objectives of Phase II were: more extensive laboratory testing, including the evaluation of different feedback algorithms for control of the damper; design and manufacture of a field prototype system; and, testing of the field prototype in a drilling laboratory. Phase II concluded on January 31, 2006, and a final report was issued.²

Work on Phase III of the project began during the first quarter, 2006, with the objectives of building precommercial prototypes, testing them in a drilling laboratory and the field; developing and implementing a commercialization plan.

All of these have been accomplished. The Downhole Vibration Monitoring & Control System (DVMCS) prototypes have been successfully proven in testing at the TerraTek drilling facility and at the Rocky Mountain Oilfield Test Center (RMOTC.) Based on the results of these tests, we have signed a definitive development and distribution agreement with Smith, and commercial deployment is underway.

This current version of the DVMCS monitors and controls axial vibrations. Due to time and budget constraints of this program, it was not possible to complete a system that would also deal with lateral and torsional (stick-slip) vibrations as originally planned; however, this effort is continuing without DOE funding.

Table of Contents

Executive Summary	4
Table of Figures	5
Summary of Phase I Results.....	6
Task 1 – Modeling of Downhole Vibration	6
Task 2 – Specifications.....	7
Tasks 3 & 4 – Mechanical Design & analysis	8
DVMCS	8
Test Equipment	8
Task 3 – Electrical Design	8
Task 5 – Testing	8
MRF Damper.....	8
DVMCS	9
Results & Discussion	10
Summary of Phase II Results.....	11
Task 2 – Complete redesign of prototype systems	11
Redesign of laboratory prototype	11
Design of feedback system	11
Intermediate prototype design.....	11
Task 3 – Build mockups for laboratory testing	12
Task 4 – Test mockups in laboratory and analyze performance.....	12
Task 5 – Redesign mockups based on test results	13
Task 6 – Build revised mockup.....	13
Task 7 - Retest mockups in laboratory and analyze performance	13
Task 8 – Refit laboratory mockup for use in drilling laboratory	13
Task 9 – Test in drilling laboratory	14
Task 10 – Complete procurement of long-lead items	15
Task 11: Update economic, market and environmental analyses.....	15
Economic & Market Analysis.....	15
Environmental analysis	16
Task 12 – Update financing plan	16
Phase III Technical Report	18
Preparation for Field Testing	18
Field Testing	18
Analysis	19
Rate of Penetration	20
Vibration.....	20
Burst data.....	21
Conclusions.....	21
Commercialization	21
Marketing and Sales	21
Manufacturing	22
Intellectual Property	22
Units	23
References.....	24
Figures	25

Executive Summary

The objective of this program is to develop a system to both monitor the vibration of a bottomhole assembly, and to adjust the properties of an active damper in response to these measured vibrations. The key feature of this system is its use of a **magnetorheological fluid** (MRF) to allow the damping coefficient to be changed extensively, rapidly and reversibly *without the use of mechanical valves*, but only by the application of a current.

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Table of Figures

Figure 1: Bit contact : 30,000 WOB - 30,000 in-lb spring rate	25
Figure 2: Measured WOB: 30,000 WOB - 30,000 in-lb spring rate	26
Figure 3: Bit acceleration: 30,000 WOB - 30,000 in-lb spring rate	27
Figure 4: ROP: 30,000 WOB - 30,000 in-lb spring rate	28
Figure 5: Top-level design of DVMCS laboratory prototype	29
Figure 6: Design of test bench and related material	30
Figure 7: MR valve test fixture.....	31
Figure 8: Electronics circuit diagram of test equipment.....	32
Figure 9: Results of initial damper testing	33
Figure 10: DVMCS laboratory prototype under testing on test bench	34
Figure 11: 10,000 lbs. WOB - DVMCS dynamic stiffness.....	35
Figure 12: Dynamic Range of MRF damper including demagnetization circuit	36
Figure 13: Redesigned MRF damper section.....	37
Figure 14: Concrete block design used for TerraTek testing.....	38
Figure 15: Effect of DVMCS on rate of penetration	39
Figure 16: Effect of DVMCS on WOB variation	40
Figure 17: Field prototype valve assembly mounted in flow loop loading frame (left) with output monitors (right)	41
Figure 18: Field test lithology from offset log.....	42
Figure 19: Preliminary Analysis of Effect of DVMCS on ROP	43
Figure 20: Effect of DVMCS on vibrations in shale	44
Figure 21: Effect of DVMCS on vibrations in siltstone	45
Figure 22: Burst data Fourier analysis - Lower vibration sensor	46
Figure 23: Burst data Fourier analysis - Upper vibration sensor	46

Summary of Phase I Results

The overall task of Phase I was to define the DVMCS and evaluate the feasibility of using an MRF damper for this purpose. More complete details can be found in the Phase I Final Report.¹

Task 1 – Modeling of Downhole Vibration

The initial task was to adapt APS's proprietary WellDrill^{SM 3} software to model the effect of the active damping on drillstring response. Typical results are shown in **Figure 1** to **Figure 4**. They demonstrate that if the damping coefficient can be maintained in the optimal range for the particular drilling conditions:

- the bit can be kept on bottom (**Figure 1**);
- the weight-on-bit (WOB) can be maintained at the desired value (**Figure 2**);
- Vibrations generated at the bit (and then propagated through the drill string) are minimized (**Figure 3**); and,
- The rate of penetration (ROP) can be increases (**Figure 4**).

Task 2 – Specifications

The Specifications defined in Phase I (and later refined) are the following:

Table 1: DVMCS Design Specifications

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

Tasks 3 & 4 – Mechanical Design & analysis

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 5**.

Test Equipment

To simulate downhole conditions, both for the testing of the MRF valves and the entire DVMCS, APS designed the test bench shown in **Figure 6**. 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.

Before testing the entire DVMCS (in Phase II), it was important to evaluate the performance of the MRF valve itself. To that end, we designed a special test fixture (shown in **Figure 7**.)

Task 3 – Electrical Design

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 8**.

Task 5 – Testing

MRF Damper

Varying loads were applied to the test damper, using the test bench (**Figure 6**), corresponding to different values of WOB, and the damping coefficients were determined as describe in **Reference 1**. The results of these measurements are shown in **Figure 9**, and discussed in the **Results & Discussion** section below. The testing indicated that an acceptable range of damping coefficients could be obtained using the power available downhole.

DVMCS

The first attempt to assemble the prototype DVMCS ran into difficulties, particularly the galling of several parts. 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 facilitates its integration into the tool, and makes it easier and less costly to replace the part when the threaded areas wear in use. The new design is shown in **Figure 13**.

This revised prototype was tested (see **Figure 10**) 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 Active Vibration Damper (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 & Discussion**, below.

Table 2: DVMCS Test Matrix

<u>Parameter</u>	<u>Values</u>
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 & Discussion

A detailed discussion of the Phase I results is given in Appendix A of **Reference 1**. What follows is a summary of the primary conclusions.

1. This initial round of tests demonstrated the effectiveness of using the MR fluid as a vibration damping medium.
2. The amount of damping depended upon the current passed through the MR fluid and the gap of the damper. Increasing the current through the MR fluid significantly increased the damping (**Figure 9**.)
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 11**).
4. Extensive data were gathered on the variation of the AVD performance over the frequency range. Parameters studied included: relative displacement; WOB applied to bit; total system damping. A combination of these parameters was used in Phase II to drive the feedback algorithm.
5. To optimize the damping, different gap sizes were evaluated in Phase II, including 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 limited the amount of stroke that can be imposed on the AVD test piece. For the additional testing in Phase II, the spring rate of the Belleville springs was reduced to 5,000 lbs/in. This provided a stroke of ± 2 ”, or 50% of the maximum stroke.
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.

Summary of Phase II Results

Complete details of Phase II can be found in the Phase II Final Report². A summary of key results is given below. **Task 1** consisted of a redefinition of tasks, which is reflected in the results given below.

Task 2 – Complete redesign of prototype systems

Redesign of laboratory prototype

The redesigned valve now has the coils located in the non-reciprocating portion of the tool. This lends itself to a more reliable electrical connection and better protection of the coils from the abrasive MR fluid.

Design of feedback system

The laboratory results of Phase I were analyzed to determine which particular feedback algorithms, and which input data, were likely to result in the most efficient control of the DVMCS. Two algorithms were identified for further study using the laboratory prototype:

- *Minimum WOB variation.* In this algorithm, the relative motion of the two halves of the DVMCS is used as a proxy for the approximate WOB. The algorithm minimizes the change in DVMCS motion, thereby keeping WOB constant.
- *Hardening algorithm.* The first algorithm had some difficulties near the extremes of motion. Under certain conditions, it caused the DVMCS to ‘lock up’ and effectively remove all damping from the system. To remedy this possible problem, a ‘hardening algorithm’ was developed, which uses a quadratic factor, also based on the relative motion of the DVMCS. This approach, described in **Reference 2, Appendix B**, is the one employed in the Phase II and Phase III prototypes.

Intermediate prototype design

The design of the field prototype evolved considerably over the course of Phase II. Among the areas added or changed in the design were:

- Addition of a battery-powered, self-contained unit to record accelerations at the bit. (This was for evaluation purposes, and will likely not be a part of the commercial tool.)
- Addition of a battery to the DVMCS to preserve the absolute position when the tool is powered down.
- Elimination of the WOB sensor, as the absolute deflection of the DVMCS will be used as a proxy for this measurement.
- Development of a connector to transfer power and data between the turbine-alternator unit and the DVMCS sub.

According to the revised Statement of Work, the laboratory prototype design was further modified to allow the prototype to be run in the drilling laboratory at TerraTek. Among the changes made were:

- Replaced the instrumented “bit” element with the appropriate bit box.
- Installed the battery-operated vibration monitoring sub described above, both above and below the DVMCS sub
- Installed the internal motion controller input.
- Manufactured new upper sub to interface with the TerraTek commutator.

Task 3 – Build mockups for laboratory testing

The revised laboratory mockup was built with the modifications described above. It was tested on APS’s test bench, using the sweep algorithm. During testing, it was noted that the motor and gear train of the test bench (**Figure 6**) could not provide sufficient power at higher frequencies without overheating or tripping the controller circuit breaker. Testing was therefore limited to 1.5 Hz. The initial results indicated a deviation from earlier static testing and the model predictions. Given these two problems, testing of the feedback algorithms was postponed until the causes were better understood.

A new gearbox/motor combination was designed and installed in the test bench. A viscometer was procured, and was modified to include magnetic coils, to study the properties of APS’s ‘home-made’ MR fluid vs. the Lord commercial fluid. Testing showed no differences in the fluids.

Task 4 – Test mockups in laboratory and analyze performance

A preliminary analysis of the dynamic test data showed that the ranges of the dynamic stiffness of the damper were 11% (at 5,000 lbs. WOB) and 80% (at 10,000 lbs. WOB). These were significantly below the ranges predicted by the modeling and earlier static tests. Analysis of the test data showed that the field in the damper gaps was only 60-90% of that required for saturation of the MR properties of the fluid. The increase in viscosity of the fluid was also less than was predicted by the Lord literature.⁴

The residual magnetic fields around the components were measured to be quite high, even in low coercivity ‘nonmagnetic’ materials. The low coercivity means, however, that a small reverse field can remove this residual magnetism. Based on these results, the laboratory prototype external control circuitry was modified to include a demagnetization algorithm, and the earlier testing was repeated. The results were markedly improved and are presented in **Figure 12**.

It was concluded that the addition of the demagnetization circuit was the most critical change required in the prototype design.

Task 5 – Redesign mockups based on test results

Based on the above observations, several changes were made in the design.

- The coil circuit was modified to demagnetize the valve components when the field is to be reduced.
- In order to allow a wider choice of materials for the valve components, the valve was moved *above* the Belleville spring. This decreased the amount of load the valve must bear and allowed the alloys to be chosen on the basis of their magnetic properties and resistance to damage by the mud, rather than particularly on their strength.
- The damper design was ‘inverted,’ putting the coils in the outer housing and leaving the mandrel a constant diameter. Calculations show that this geometry will generate much higher fields. It also greatly simplifies assembly, since the mandrel, which must be carefully slid into the housing, will have an even surface. A sketch of the new design is shown in **Figure 13**. The coils are the orange features in the sketch.

Task 6 – Build revised mockup

The revised mockup was constructed with the changes described above.

Task 7 - Retest mockups in laboratory and analyze performance

The reworked prototype, with external coils, was tested on our laboratory test bench, (**Figure 6**.) The test data were analyzed by APS analytical specialist, Mark Wassell. The dynamic range of the revised damper circuit was somewhat less than that of the original. This was attributed to two factors:

- The reworked mandrel was slightly smaller than the original, which resulted in a larger gap. This, in turn, reduced the ‘power off’ damping coefficient and also reduced the applied field for a given current.
- The reworked mandrel still had some of the internal structure of the original coil winding slots. While these were filled in, the interfaces might have interfered with the magnetic flux lines, decreasing the efficiency of the magnetic circuit.

Despite these results, it was decided to go ahead with the testing using the current design. Since the ‘power off’ coefficient was quite low, the gap was reduced to increase the dynamic range. Time considerations prevented the manufacture of a new mandrel and the testing proceeded with the existing one.

Task 8 – Refit laboratory mockup for use in drilling laboratory

The new and revised components of the prototype were manufactured and assembled with the changes described above. The test procedure and matrix were developed, and four test formations were designed and built. These each consist of a slab of hard granite mounted at an angle of 10° within a larger hard concrete block. (See **Figure 14**) The contrast in hardness at the inclined interfaces was designed to induce significant vibra-

tion in the drilling, which will serve as a test of the efficiency of the damper and its feedback algorithms.

Task 9 – Test in drilling laboratory

The tests were run at the TerraTek drilling laboratory from January 23-27, 2006. A summary of the results is included as **Attachment F** of **Reference 2**. The conclusions reached from analysis of the data include:

- The vibration levels measured through the tests were fairly benign, in the 5 – 25 g level.
- Significant bit wear occurred during the tests. For the conventional collar (*without* the DVMCS), the ROP at the end of the tests was 60% of the ROP at the start of the tests. To account for this, the ROP rates were corrected based on an assumed linear degradation of the bit for each hole drilled. Other drilling results, such as WOB, TOB acceleration, have not been corrected for bit wear.
- Even with the benign drilling conditions the DVMCS showed its ability to improve ROP. (See **Figure 15***) ***While drilling through concrete, the DVMCS improved ROP by 10 – 15%. For granite the improvement was up to 11%.*** While drilling through granite at 120rpm with 15,000 WOB the ROP actually dropped by 2%; however, lighter damping might improve this situation. The lighter damping case was not run.
- The optimum drilling condition occurs when the DVMCS internal travel is ~0.17". If the drilling becomes too "smooth," the ROP actually drops. There appears to be an optimum travel and WOB fluctuation that produces the best ROP. It is believed that some vibration may improve cutting efficient, but this has not been proven.
- The DVMCS tool significantly decreased the WOB fluctuations compared to the conventional drill collar. (See **Figure 16**.) ***The DVMCS reduced the WOB fluctuation by 60%.*** This should be even more beneficial under high vibration conditions.
- The DVMCS displacement was 0.25" for the power off state and 0.13" at full power. The dynamic stiffness varied from 4,200 lb/in to 17,800 lb/in. These ranges should significantly improve for the commercial tool. Upon disassembly of the DVMCS tool, it was found that the brass bobbins for the damper coils had become slightly magnetic. This would have reduced the performance of the magnetic coils.

The tool was disassembled after the completion of the tests to inspect the seals and bearings. Both the bearings and the seals were still in good working order.

* For this, and the figure that follows, data are normalized to the data measured in runs without the DVMCS. Each curve is labeled by the rock type, RPM and WOB. (e.g., C-120-15 represents drilling through concrete at 120 rpm and 15,000 lbs. WOB.)

Task 10 – Complete procurement of long-lead items

The testing described above resulted in some considerable changes in the design of the prototype. These include the following:

- The internal position monitor (LVDT) did not produce sufficient signal within the collar to control the feedback loop, and it was necessary to rely upon an external sensor during the TerraTek testing. This necessitated a complete redesign of the LVDT, which is complete and used in the field prototype tools. Tests with a ¼-scale model were very promising and the revised design was included in the later prototypes.
- Assembly of the laboratory prototype, and its conversion to the drilling lab prototype, demonstrated that the current design was overly complicated, extremely difficult to assemble, and not commercially viable.
- APS's commercial partner indicated a strong preference for a system which can be integrated into their existing shock subs. A significant part of the DVMCS (bearings, Belleville springs, etc.) is very similar to a shock sub, with the key addition being the active feedback and control system.

Based on these considerations, the use of our existing prototype in the field was not considered practical and began a redesign. The result was a much improved tool. In particular:

- The hydraulic compensation system was modified so that it uses a single reservoir at the top of the tool. By eliminating the lower reservoir, the bottom end of the tool became essentially identical to a standard shock sub. This permits its integration into existing products and will greatly reduce manufacturing costs.
- Reconfiguration of the tool permits greater flexibility in assembly and maintenance. In particular, it will not be necessary to depressurize the MR fluid section to perform routine maintenance.
- The part count has been significantly reduced.

As a result of the redesign, we had to reduce the number of parts for Phase III that were to be ordered during Phase II.

Task 11: Update economic, market and environmental analyses

Economic & Market Analysis

A "bottom up" economic model was prepared for the DVMCS and was included as **Appendix H of Reference 2**. In this model, typical drilling costs were input for the particular market. Some basic assumptions were made about the improvement provided by the DVMCS, how these savings were to be shared between the service company and its clients. These were used to generate an anticipated revenue per job. The estimated number of jobs per year is also estimated.

The following were included to estimate costs: cost of purchasing the units; number of units needed to support one job; cost of money; anticipated repair costs and schedule; overhead, *etc.*

On this basis, one can calculate the ROI and payback period for our customer (*i.e.*, the oilfield service or supply company) on purchasing systems. Preliminary calculations show that, with reasonable, conservative assumptions, the DVMCS represents a very attractive investment, even for simple vertical well drilling. In deep, hard rock or offshore drilling the paybacks are enormous. One example of the results of the model is shown below in **Table 3**

To balance this model, APS also commissioned a “top down” model for several APS products and potential products. This model, produced by Spears & Associates, looked at the total domestic drilling market projections, estimated the value of this product, including rig time savings, repair and replacement costs for damaged products, *etc.* It also assumes reasonable sharing of these savings among the end user, service company and supplier (*i.e.*, APS.) It also indicated an enormous market for this product. An excerpt of this study, dealing with the DVMCS, was attached as **Appendix H of Reference 2**.

These studies have been confirmed by the level of customer interest and the signing of a worldwide development and marketing agreement with Smith Services.

Environmental analysis

The DVMCS is, by and large, a standard piece of oilfield hardware. The only nonstandard substance is the magnetorheological fluid (MRF). The MRF consists of iron filings suspended in a high-temperature synthetic oil, and therefore poses no more environmental risks than the oil itself, which are minimal. The development of the DVMCS, therefore, poses no significant environmental risks.

Task 12 – Update financing plan

The commercialization of the DVMCS and its financing have been resolved by the development & marketing agreement with Smith Services.

Table 3: ROI calculation for DVMCS

REVENUE CALCULATION					
TYPICAL JOB	QUANTITY	RATE	PER JOB		ANNUAL
			TIME	COST	
DAYS	25.0				
COST PER DAY		\$ 28,000			
TOTAL DRILLING COST				\$ 700,000	
NO OF TRIPS	7.0				
HOURS PER TRIP		12.0			
TRIPPING HOURS			84.0		
MAINTENANCE HOURS/DAY	2.0				
MAINTENANCE HOURS			50.0		
DRILLING HOURS			466.0		
AVD SAVINGS					
ROP INCREASE		10%			
DRILLING HOURS SAVED			42.4		
TRIPS AVOIDED	2				
TRIPPING HOURS SAVED			24.0		
NET TIME REDUCTION			66.4		
COST SAVINGS				\$ 77,424	
PERCENT TO SERVICE CO		50%			
SERVICE CO GROSS REV				\$ 38,712	
NUMBER OF JOBS	12				
ANNUAL REVENUE				\$ 464,545	
COST CALCULATION					
COST OF TOOL	\$ 200,000				
TOOLS/JOB	2.5				
TOTAL TOOL COST	\$ 500,000				
USEFUL LIFE (years)		5			
DEPRECIATION (s/l)				\$ 100,000	
INTEREST RATE	6.0%				
INTEREST COST				30,000	
TOOL REFURBISHMENT	\$ 12,000				
DRILLING HOURS/REFURB		500			
NUMBER OF REFURBS/YR		11			
MAINTENANCE COSTS				\$ 132,000	
DEPLOYMENT COSTS/JOB	\$ 3,000				
TOTAL DEPLOYMENT				\$ 36,000	
TOTAL ANNUAL COST				\$ 298,000	
PAYBACK CALCULATION					
NET PROFIT				\$ 166,545	
ROI				33.3%	
PAYBACK (YEARS)				3.00	

Phase III Technical Report

The basic objectives of Phase III were to build and test prototype systems in the field and to finalize the commercialization plan. While a number of delays limited the amount of field testing accomplished, the tests that were run showed the significant impact of the DVMCS on the efficiency of drilling. Based on these results, APS has moved forward with the commercialization of the tool in conjunction with Smith Services.

Preparation for Field Testing

The precommercial prototypes were tested successfully in the flow loop prior to shipment for the field test. The test arrangement had the active damper placed in a special fixture that could vary the load applied. Instrumentation measured the movement of the damper, flow, pressure, etc. The pre-test objectives, which were all met, included:

- Verifying that the damping levels were within the specified limits over a range of flows.
- Confirming the performance of the alternator.
- Testing a downlinking scheme that would be used in the field test to change the mode. The field test plans included running at 0, $\frac{1}{3}$, $\frac{2}{3}$ and full power, and switching to the automatic algorithm. The downlinking operated as planned.

Photos of the system under pre-test are shown in **Figure 17**.

Field Testing

The DVMCS prototypes were field tested at the Rocky Mountain Oilfield Test Center (RMOTC)⁵ from July 17-22, 2008. The interval drilled is known, based on offset well information, for hard drilling conditions that produces a lot of drill string vibration. This interval consisted of shale and silt stone (**Figure 18**). The typical ROP for this interval is between 5 – 15 ft/hr. A base line test was run first, without the AVD in the bottom hole assembly (BHA). The test BHA was slick assembly and consisted of:

Table 4: Bottomhole Assembly for RMOTC Test

Item	Length (ft.)	OD(in.)
F35 Tricone Bit	0.7	8 ³ / ₄
Bit Sub	3	6 ¹ / ₄
Drill Collar	30	6 ³ / ₄
Crossover	3	6 ³ / ₄
VMS	3	6 ³ / ₄
DVMCS	26.36	6 ³ / ₄
VMS	2.52	6 ³ / ₄
Crossover	3	6 ³ / ₄
APS MWD with Gamma	30	6 ³ / ₄
Drill Collars	420	6 ¹ / ₄
Jars	30	6 ¹ / ₄
Drill Collars	60	6 ¹ / ₄

Downhole data measurements were made using two APS Vibration Monitoring Subs (VMS), mounted below and above the DVMCS. Each sub measured and recorded the vibration in three orthogonal orientations, two at right angles to each other in the lateral direction and one in the axial direction. The tool's memory recorded the data for downloading at the surface after each test run. The data consist of the maximum, minimum and root mean square (RMS) vibration levels for each of the three accelerometers, taken in successive 4 second intervals.

In addition, the VMS recorded four data bursts. These were triggered by preset vibration severities. The first time the sensors noted levels exceeding, 25, 50, 75 and 100g, the VMS recorded a 20 second burst of data taken at 1000 Hz, including 8 seconds of data from before the triggering event. These bursts help identify the nature of a particular vibration.

The BHA also included an APS MWD tool with gamma to correlate the changes in formation to the offset wells, and monitor the directional control of the well. Surface sensors recorded the RPM, WOB and ROP vs. depth during the test.

The operating mode of the DVMCS was preset at the surface and then modified *via* changes in flow rate. The tool was run in automatic adjustment mode (the planned commercial approach), and at 0, ¹/₃, ²/₃, and full damping. The latter were used to measure the sensitivity of the results to the use of the feedback loop.

Analysis

Much of the data analysis was performed by our commercialization partner, Smith Services, using their standard drilling analysis software. (This constituted a portion of their "in-kind" contribution to this project.) Initial test analyses compared the measured ROP using the DVMCS in its various settings with that in the control run without the DVMCS. Care was taken to maintain the other drilling parameters as identical as possible during the two runs, so that the differences can be ascribed to the DVMCS.

Rate of Penetration

A summary of the ROP data is given in **Figure 19**.^{*} It plots the ROP measured when drilling the shale and siltstone formations with the DVMCS to that drilled without it. Several interesting conclusions can be drawn from this graph.

- The DVMCS can produce *dramatic* increases in the ROP. (We do not contend that the 70% increase seen in the shale at $\frac{1}{3}$ power is typical.)
- When the control power was turned off, and the DVMCS was operating as a standard shock sub, the ROP showed little change in the shale and actually *decreased* in the siltstone.
- The optimum damping level seems to be ~35% for the shale and ~55% for the siltstone. This result, coupled to the one above highlights the need for an *adaptive* system for maximum performance.
- The feedback algorithm appears to be optimized for the siltstone, since the ROP produced with the feedback is higher than any of the fixed damping results. This does not appear to be the case for the shale, however. While the 51% increase in ROP is impressive, better results seemingly could have been attained using a fixed $\frac{1}{3}$ power. Further refinement of the algorithm is ongoing during current field testing.

Vibration

Vibration comparisons[†] are shown in shale in **Figure 20**, and in siltstone in **Figure 21**. In the shale, the axial vibration is generally small, but shows some dependence on the damping level, decreasing at lower values. The uphole vibration is significantly lower than that near the bit. The lateral vibration is somewhat greater. In addition, although the DVMCS is not designed to dampen lateral vibrations, it seems to have an effect. The uphole vibration appears to go into some sort of resonance at $\sim\frac{2}{3}$ full scale. Note also, that the primary objective of the DVMCS is to reduce vibration **at the bit** to enhance drilling. The reduction of vibration in the BHA as a whole is a secondary objective.

The situation in the siltstone shows some different behavior. The axial vibration is again small, but the vibration at the bit decreases with increasing damping. Both the uphole and downhole sensors show some form of lateral resonance near $\frac{2}{3}$ full power.

These results show that fixed damping shock subs can occasionally have unexpected, and negative, impacts, and provide further evidence of the need for an adaptive damper.

^{*} Note that in the figures, the DVMCS is referred to by its commercial name of AVD (“Active Vibration Damper”).

[†] Note that the downhole vibration monitor was not functioning during the full-scale damping run.

Burst data

The burst data gives some interesting results. A Fourier analysis shows strong responses at both 1x and 3x the 100 rpm rotary speed, in both the downhole (**Figure 22**) and uphole (**Figure 23**) data. Additionally the downhole sensor displays has several resonances between 100 Hz 140 Hz. These are believed to be induced by the tricone bit inserts. In the uphole tool, however, these higher frequency resonance are reduced. They appear to have been filtered out by the DVMCS tool.

Conclusions

In addition to the analyses above, APS's partners at Smith Services did a proprietary analysis of the data, including various parameters – e.g., lithology, rock hardness, energy employed, etc. Some of their conclusions are given below:

- In the general case, without lithology selection, the section drilled with the automatic algorithm had the highest ROP, highest WOB, highest compressive strength and the highest performance. This was the best mode in this testing.
- For shales, the algorithm section had the highest ROP, highest WOB, and highest compressive strength; the 1/3 damping section had the highest drillability for performance. For the siltstones, the algorithm section had the highest WOB, highest compressive strength and the highest performance; full power yielded the highest ROP. In general, the DVMCS amply demonstrated its ability to improve the speed and efficiency of the drilling process over a range of conditions.

Commercialization

Marketing and Sales

APS and Smith Services have signed an exclusive worldwide marketing agreement. Smith has bought the commercial prototypes and has committed to minimum purchases going forward.

Over the past several years, APS has presented several papers^{6,7,8} describing the progress of this program. In addition, a joint paper with Smith Services has been submitted for the 2008 SPE Annual Technical Conference & Exhibition.

Manufacturing

At present, we are refining the manufacturing, maintenance and servicing of the DVMCS, with the aims of:

- reducing manufacturing complexity and cost;
- improving reliability; and,
- minimizing the maintenance requirements.

The overall objective is to improve the cost/benefit ratio for both the service provider and the end user.

Intellectual Property

APS has been issued a patent⁹ on this technology, which supplements an earlier patent¹⁰ covering the use of MRF in downhole applications. Several divisional applications, and international applications are also in process. We also signed a non-exclusive licensing agreement with Sandia National Laboratory covering their related patent¹¹ and technology.

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

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- ² Cobern, Martin E., "Downhole Vibration Monitoring & Control System: Phase II Final Report," April 30, 2006 (This report does not appear to be posted on the Information Bridge.)
- ³ APS Technology, Inc., "WellDrill Technical Data Sheet," <http://aps-tech.com/tds/APS-WellDrill.pdf>
- ⁴ cf., e.g., Lord Corporation Technical Data Sheet "MRF-140 CG" <http://www.lordfulfillment.com/upload/DS7012.pdf>
- ⁵ Under RMOTC Cooperative Research and Development Agreement (CRADA) 2007-078
- ⁶ Cobern, Martin E., and Wassell, Mark E., "[Drilling Vibration Monitoring & Control System](#)," at the National Gas Technology Conference II, Phoenix, AZ, February, 2004.
- ⁷ "Laboratory Testing of an Active Drilling Vibration Monitoring & Control System," at the American Association of Drilling Engineers National Technical Conference, Houston, April, 2005, paper [AADE-05-NTCE-25](#)
- ⁸ Cobern, Martin E., Perry, Carl A., Barbely, Jason A., Burgess, Daniel E. and Wassell, Mark E., "Drilling Tests of an Active Vibration Damper," at the SPE/IADC Drilling Conference, February 2007, Amsterdam, paper [SPE/IADC-105400](#)
- ⁹ Wassell, Mark E., Turner; William E., Burgess; Daniel E., and Perry; Carl A., "System and method for damping vibration in a drill string," US Patent # [7,219,752](#), issued May 22, 2007.
- ¹⁰ Wassell, Mark E., "Magnetorheological fluid apparatus, especially adapted for use in a steerable drill string, and a method of using same," US Patent # [6,257,356](#), issued July 10, 2001.
- ¹¹ Raymond; David W. and Elsayed; Mostafa A., "Controllable magneto-rheological fluid-based dampers for drilling," US Patent # [7,036,612](#), issued May 2, 2006.

Figures

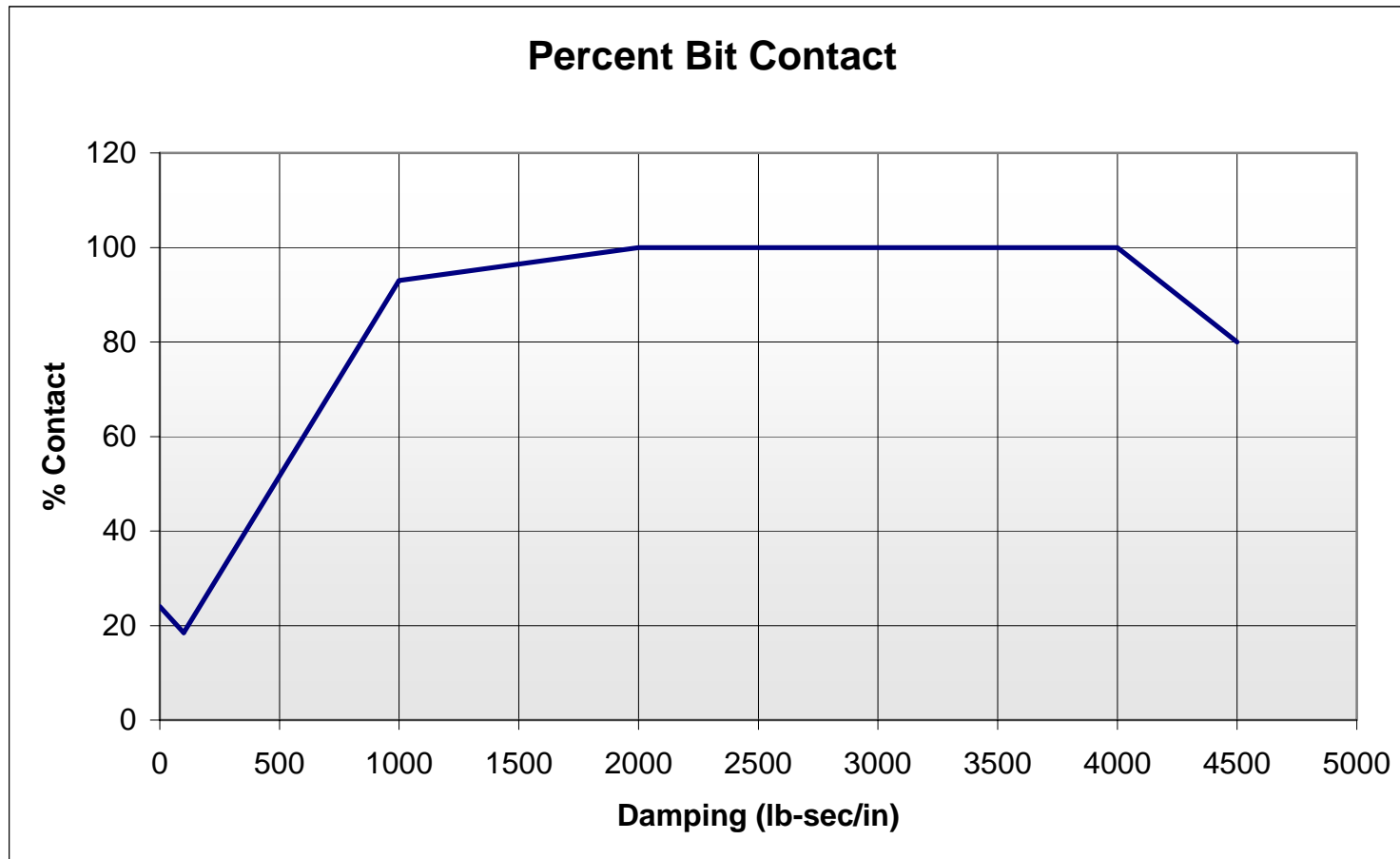


Figure 1: Bit contact : 30,000 WOB - 30,000 in-lb spring rate

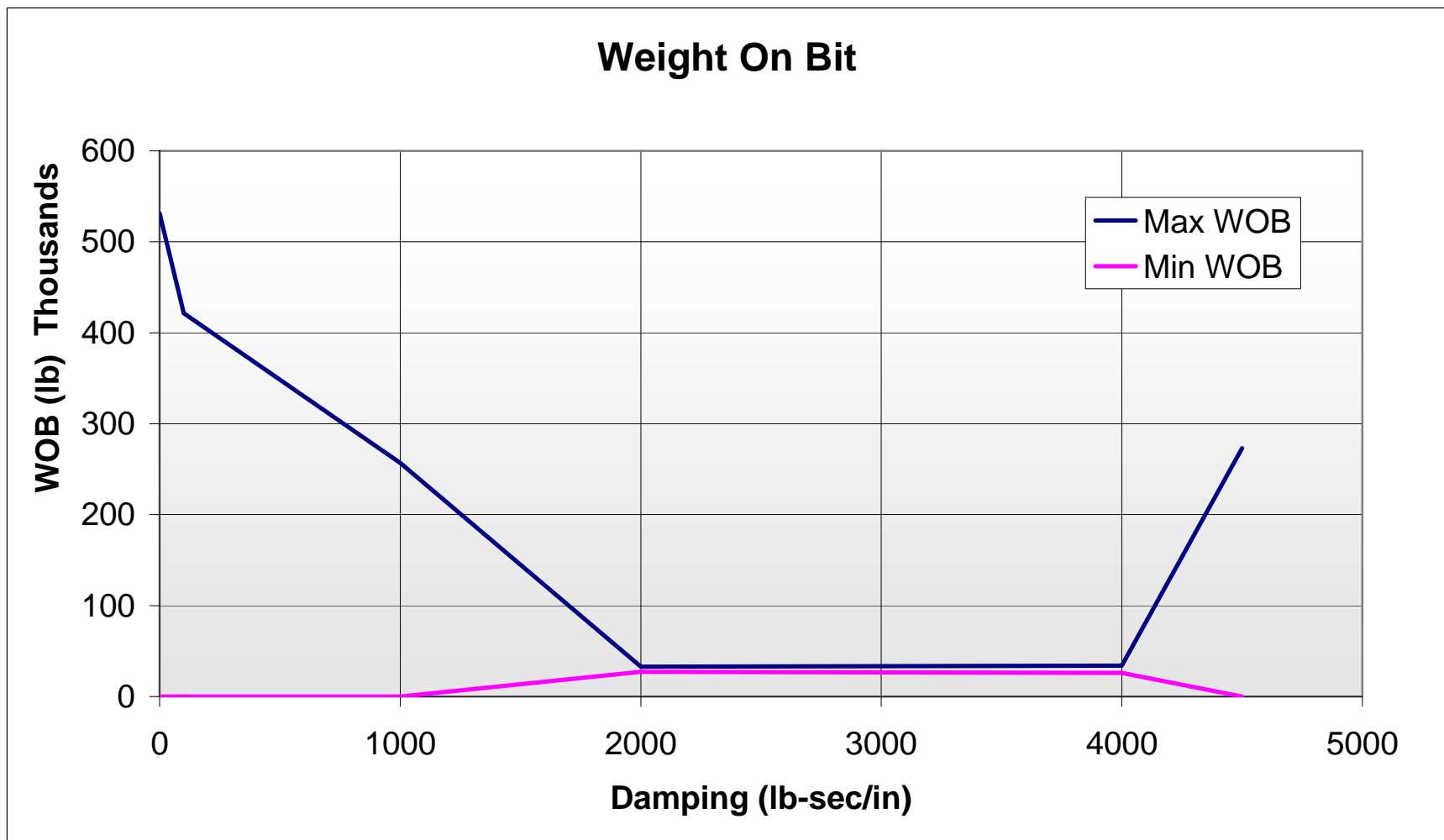


Figure 2: Measured WOB: 30,000 WOB - 30,000 in-lb spring rate

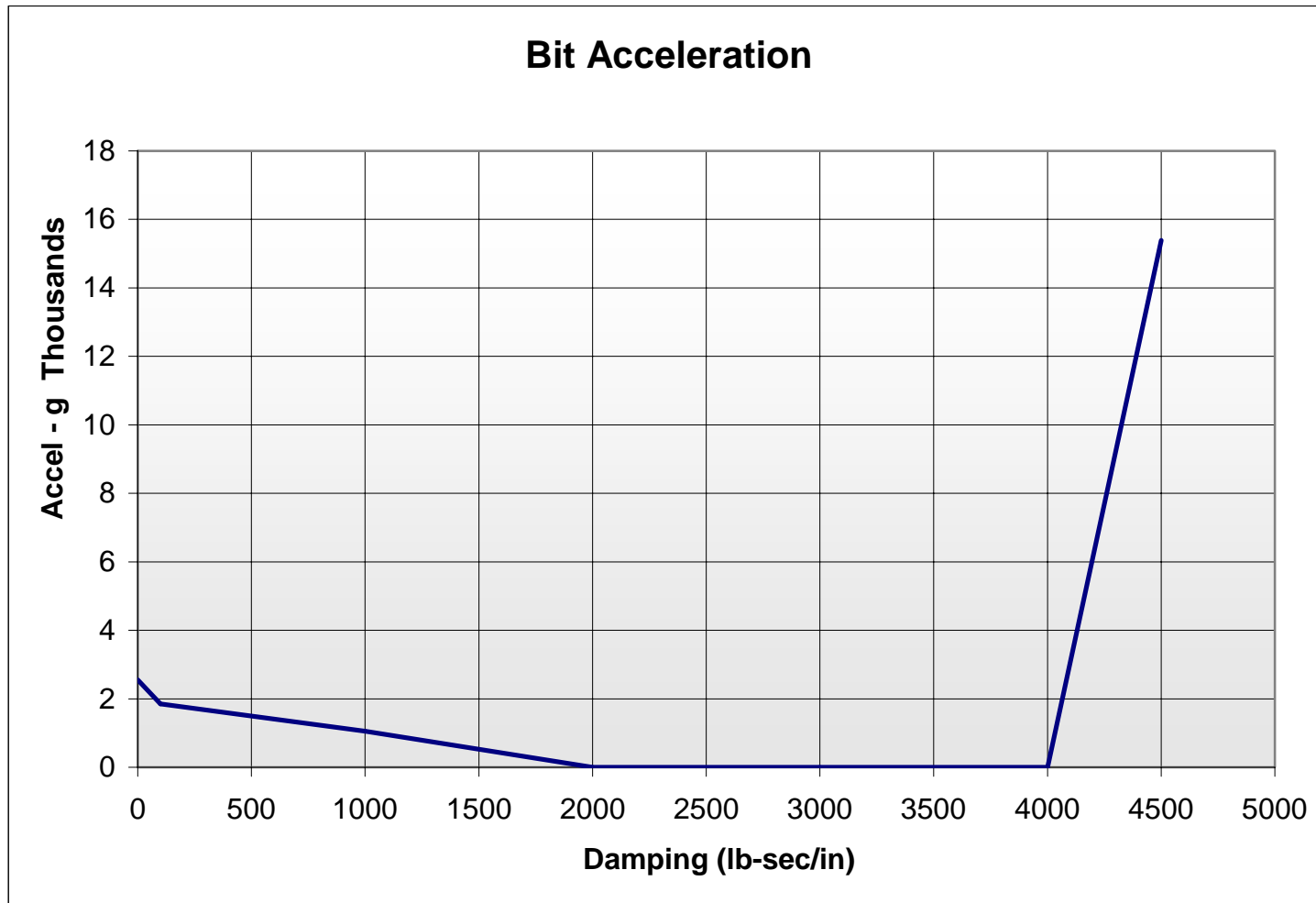


Figure 3: Bit acceleration: 30,000 WOB - 30,000 in-lb spring rate

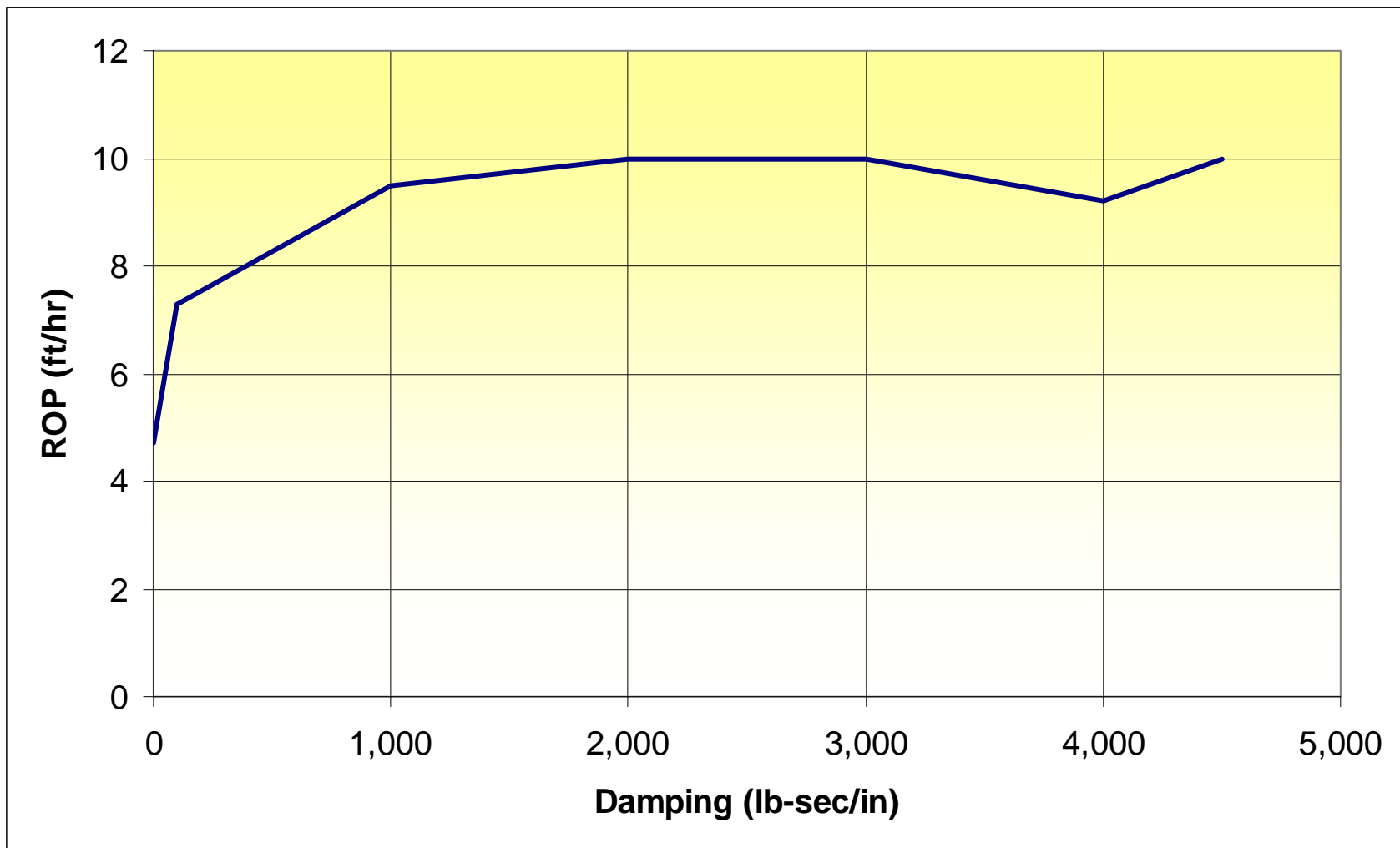


Figure 4: ROP: 30,000 WOB - 30,000 in-lb spring rate

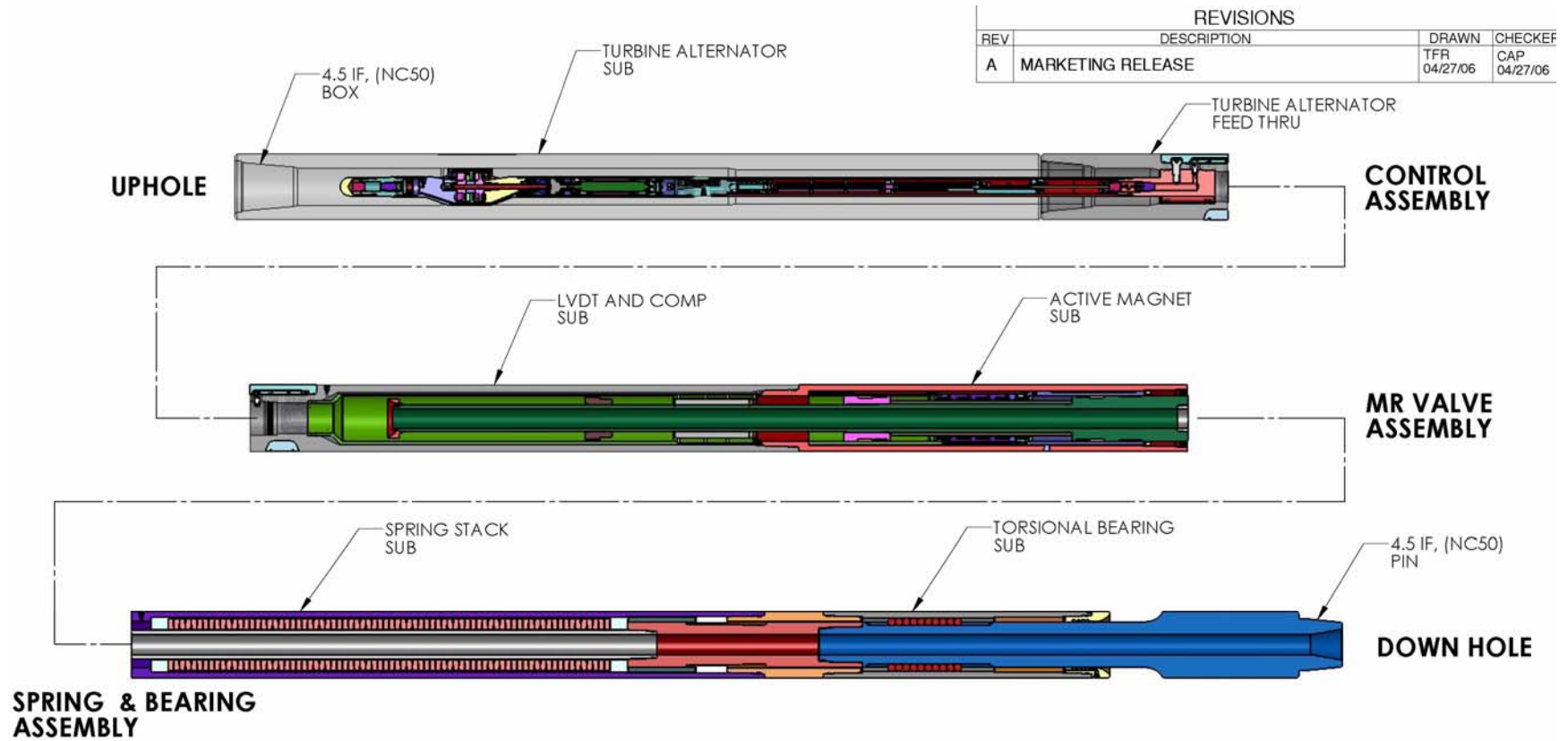


Figure 5: Top-level design of DVMCS laboratory prototype

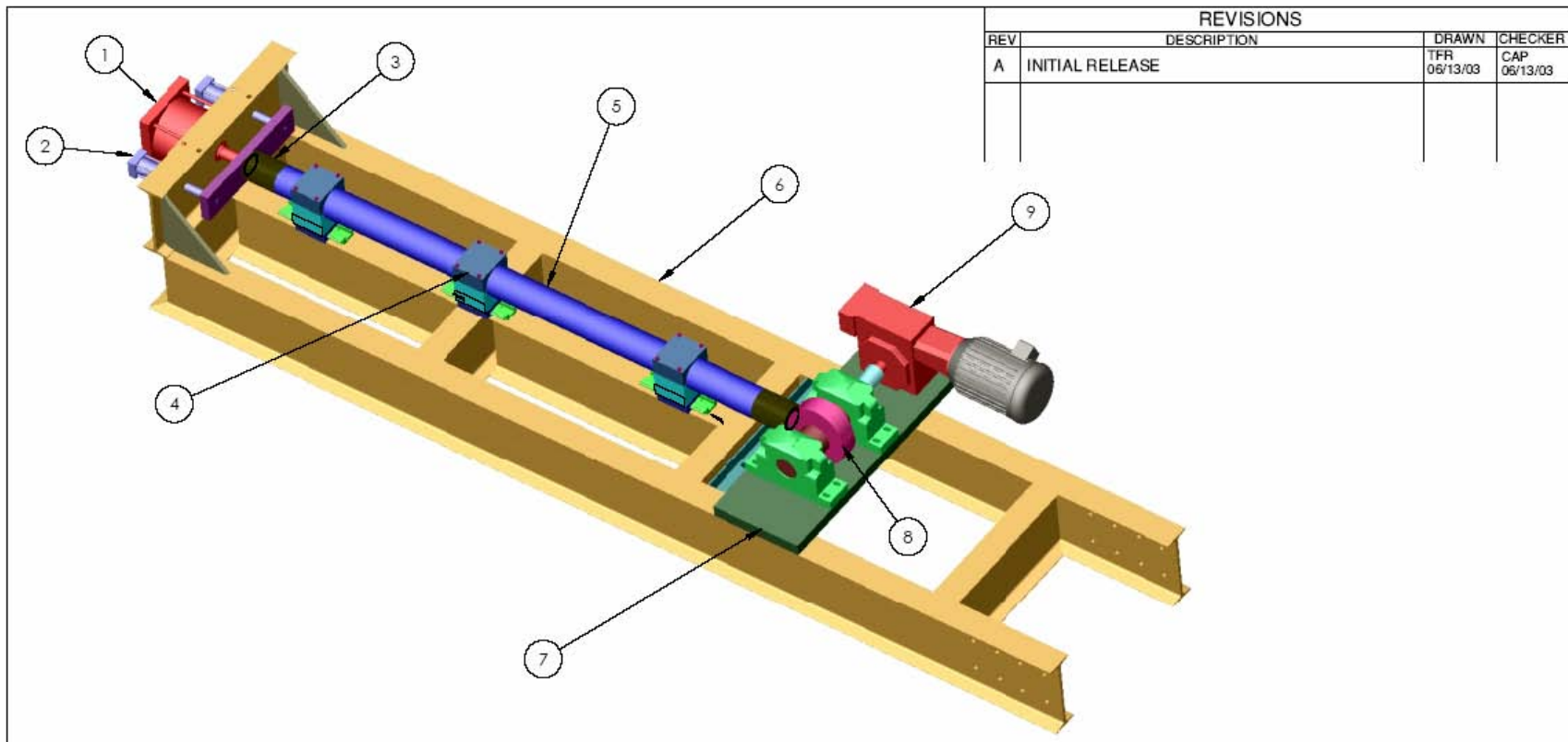


Figure 6: Design of test bench and related material

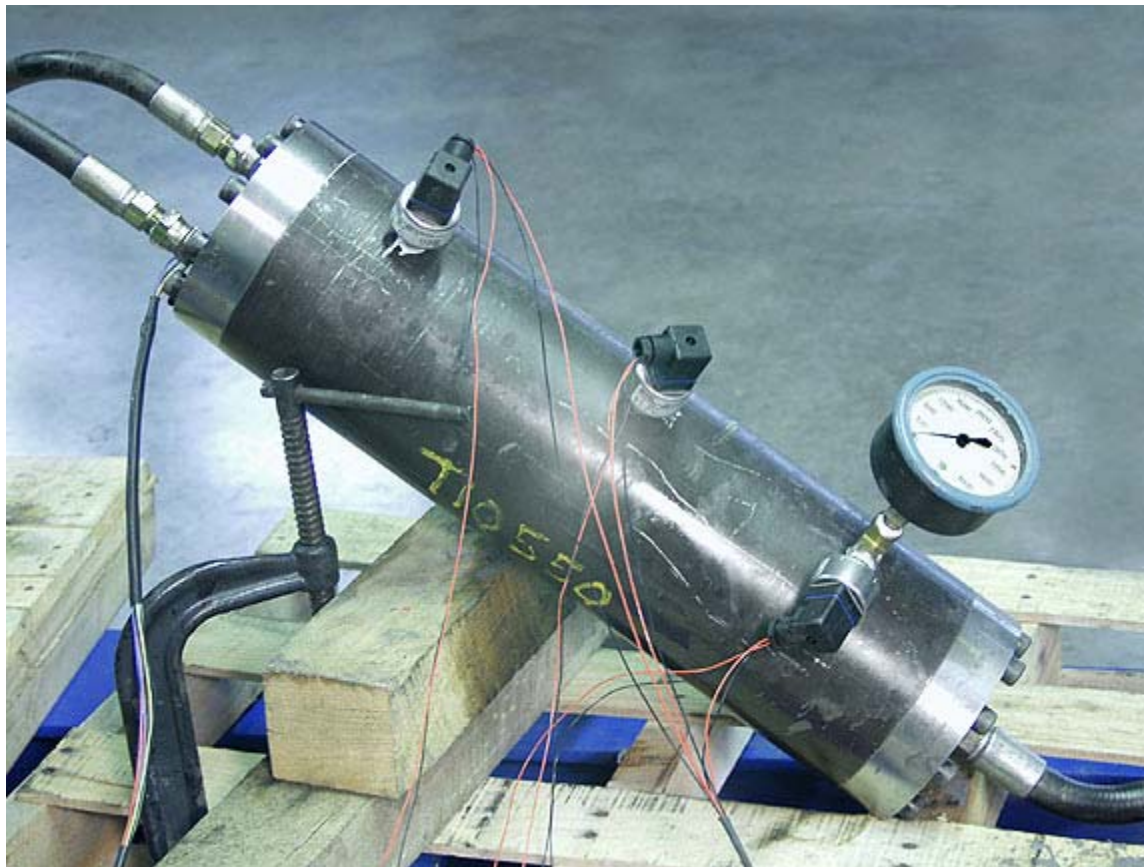


Figure 7: MR valve test fixture

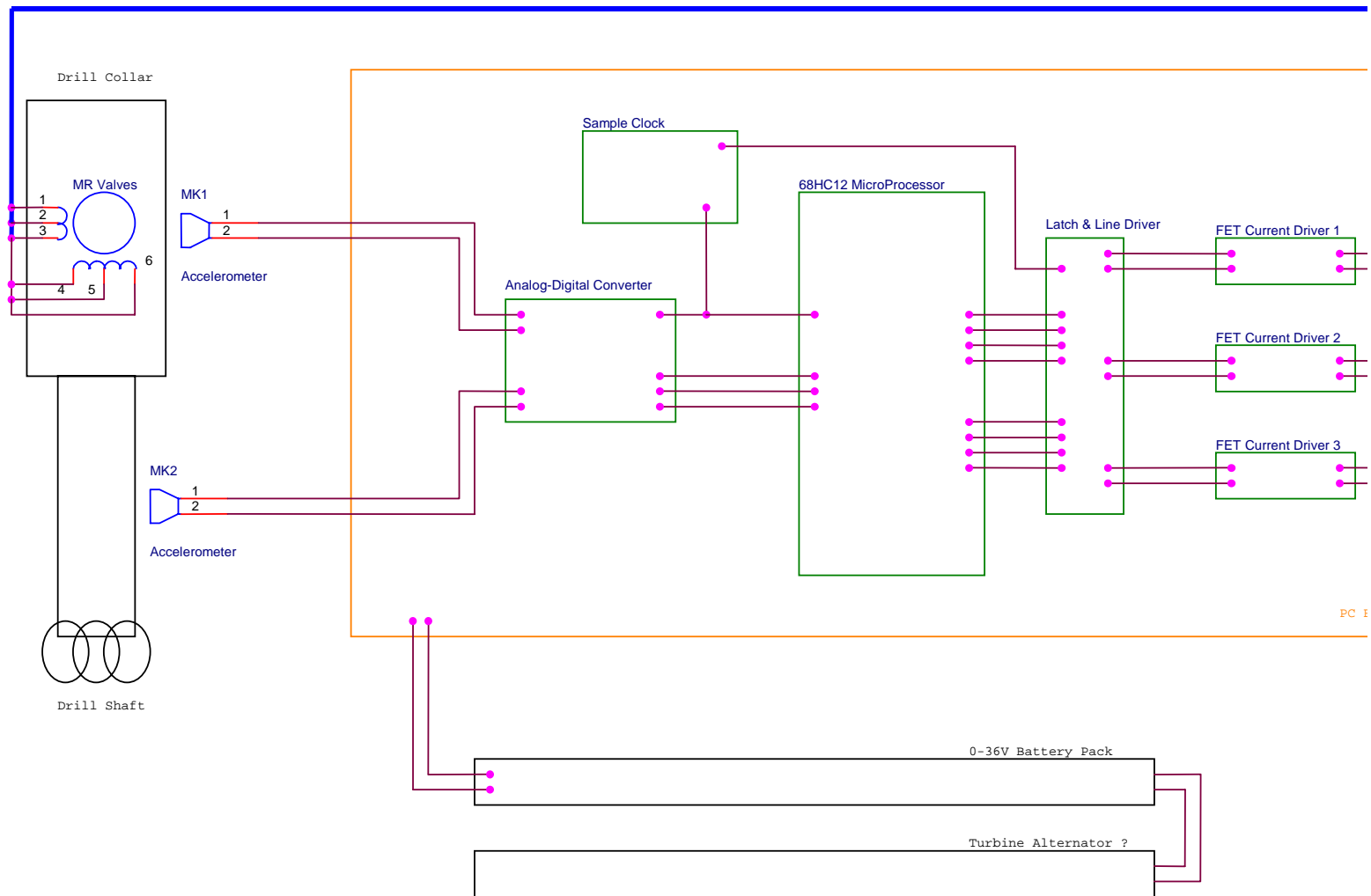


Figure 8: Electronics circuit diagram of test equipment

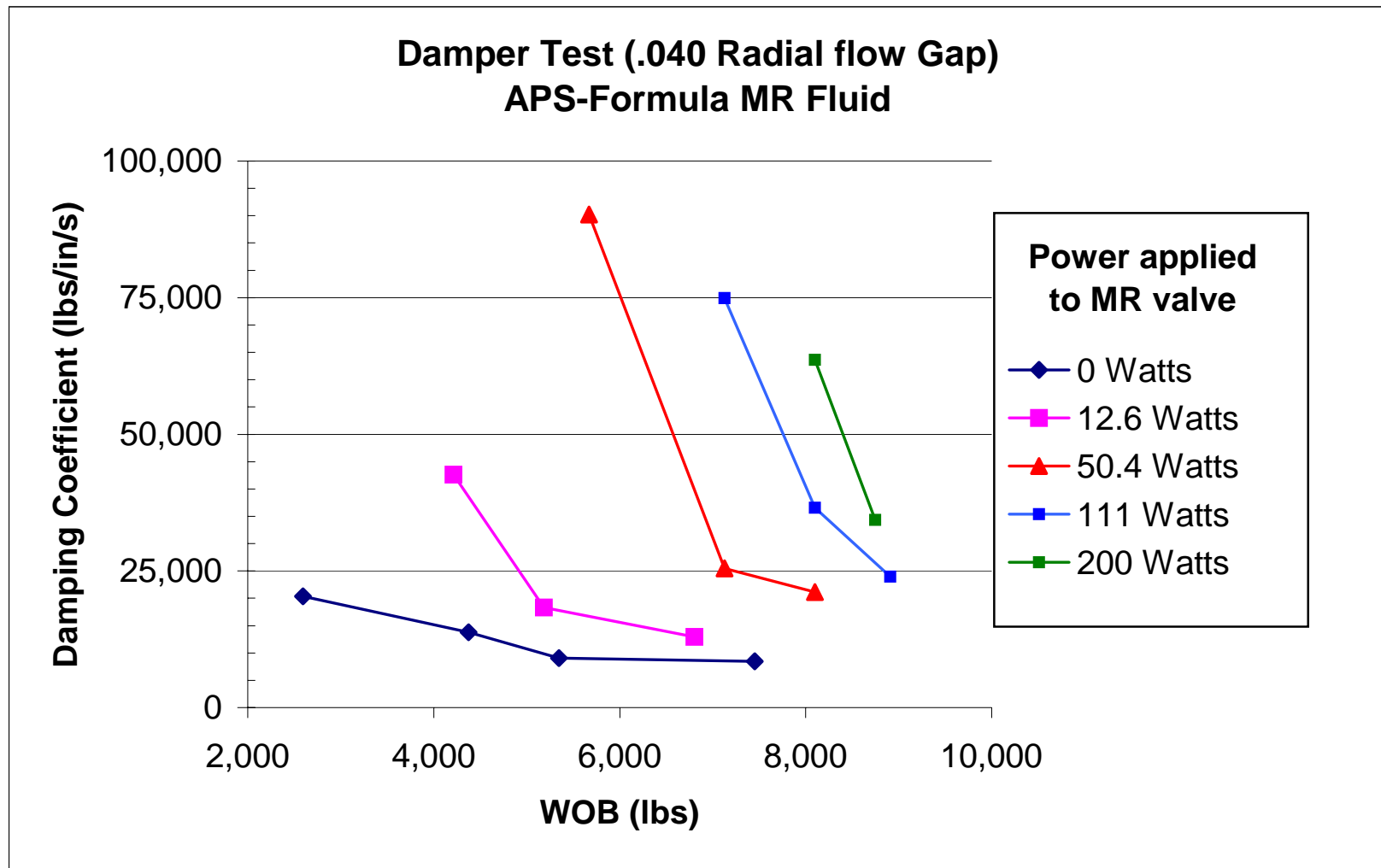


Figure 9: Results of initial damper testing



Figure 10: DVMCS laboratory prototype under testing on test bench

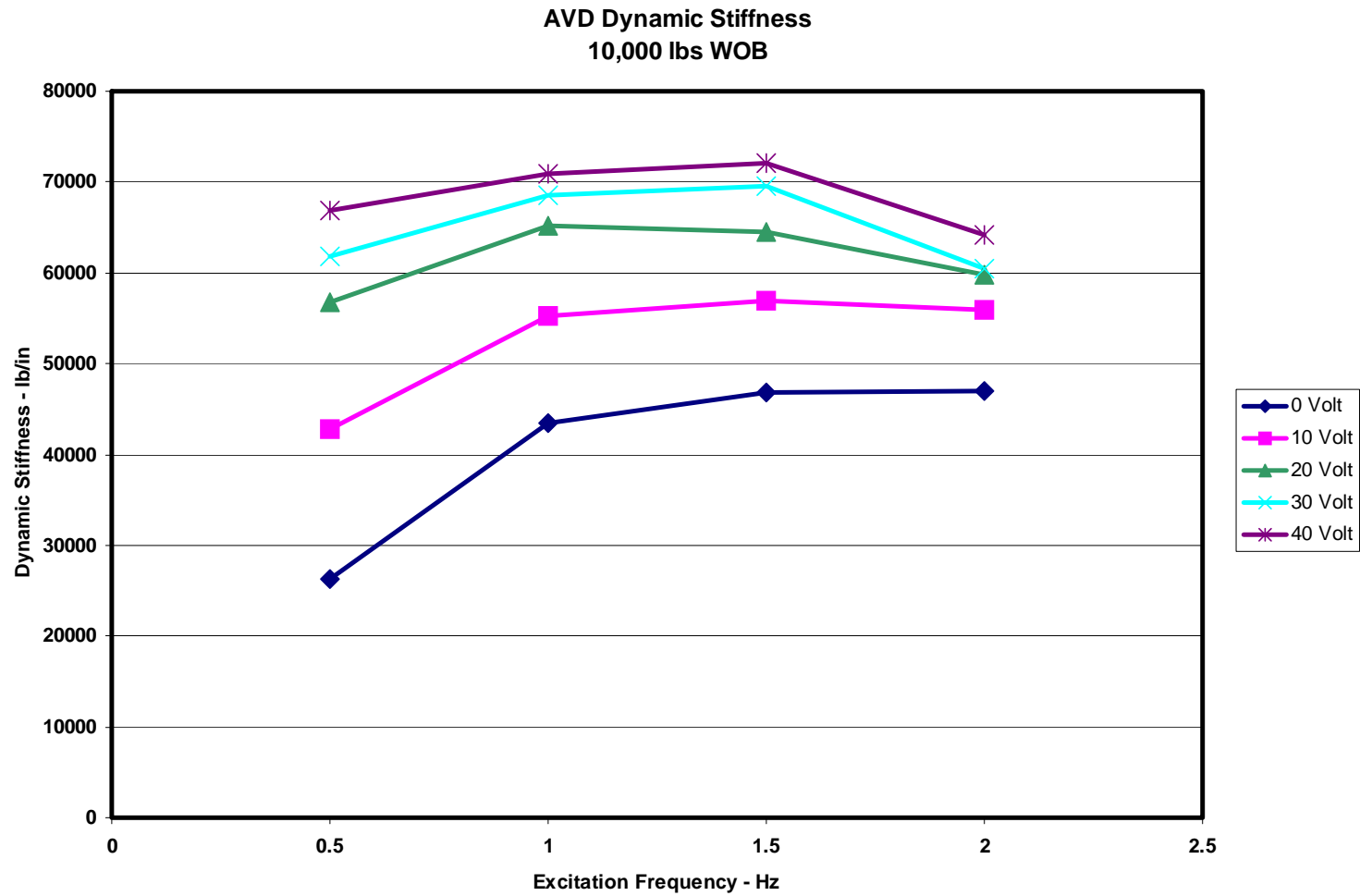


Figure 11: 10,000 lbs. WOB - DVMCS dynamic stiffness

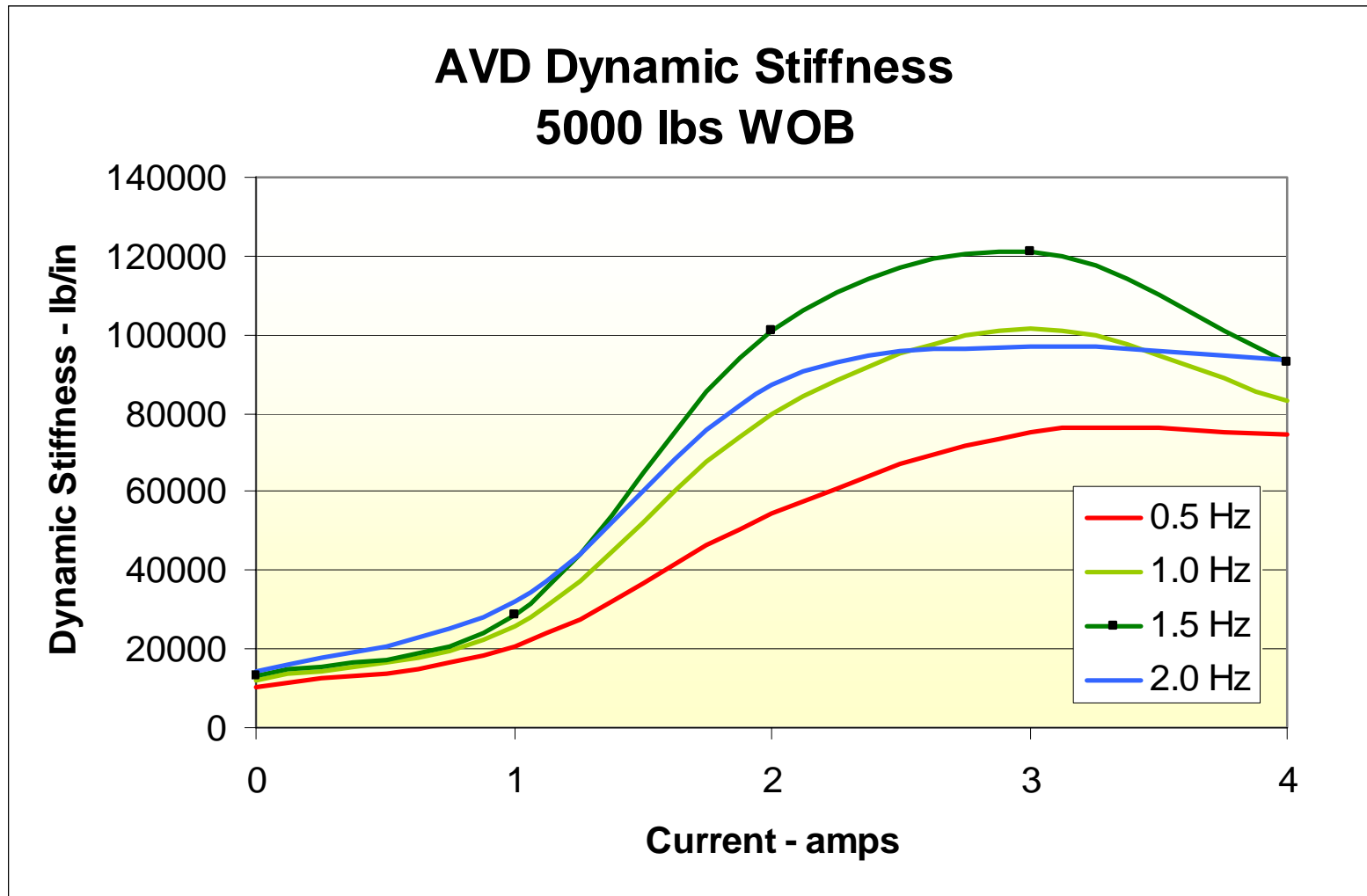


Figure 12: Dynamic Range of MRF damper including demagnetization circuit

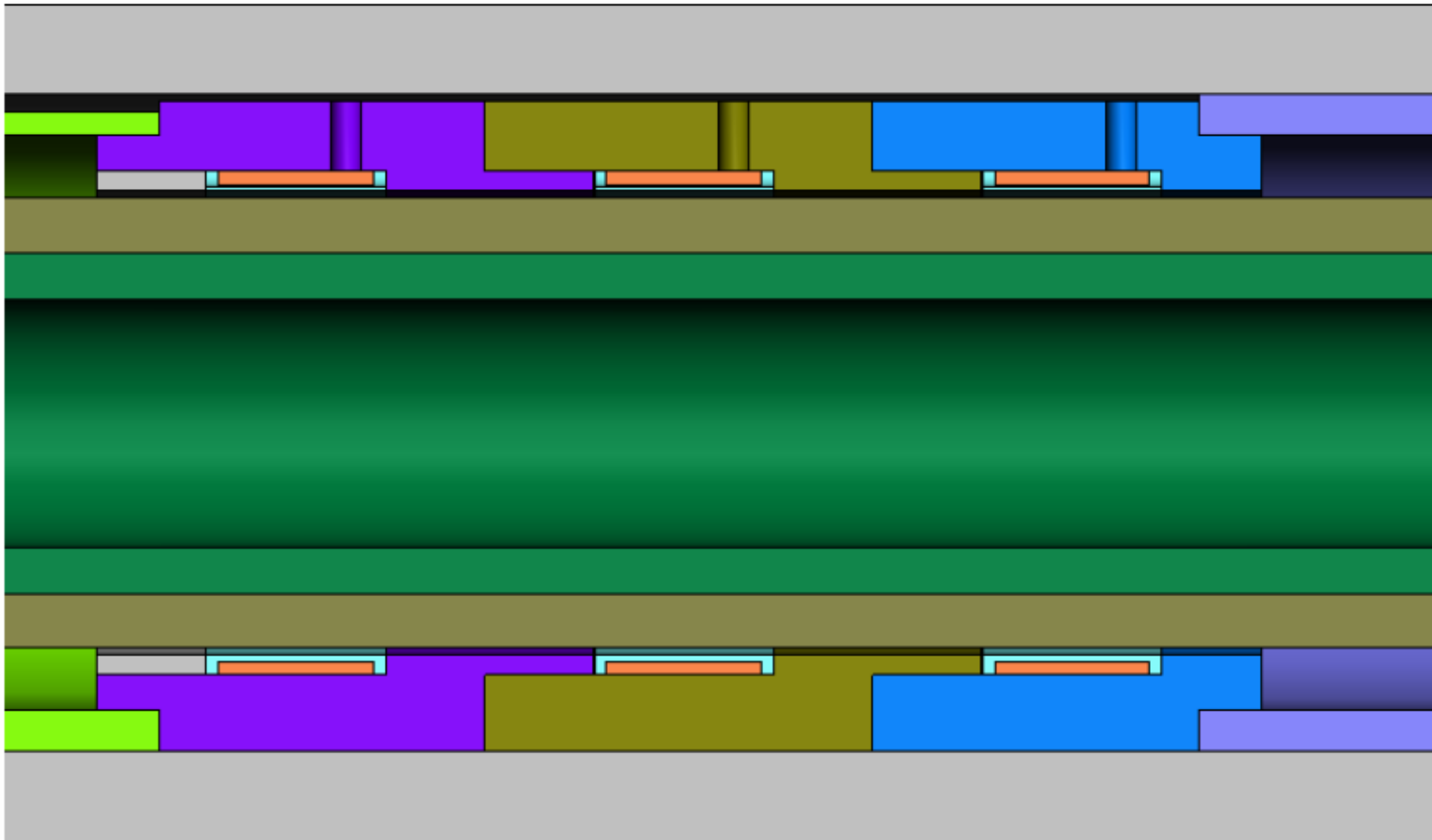


Figure 13: Redesigned MRF damper section

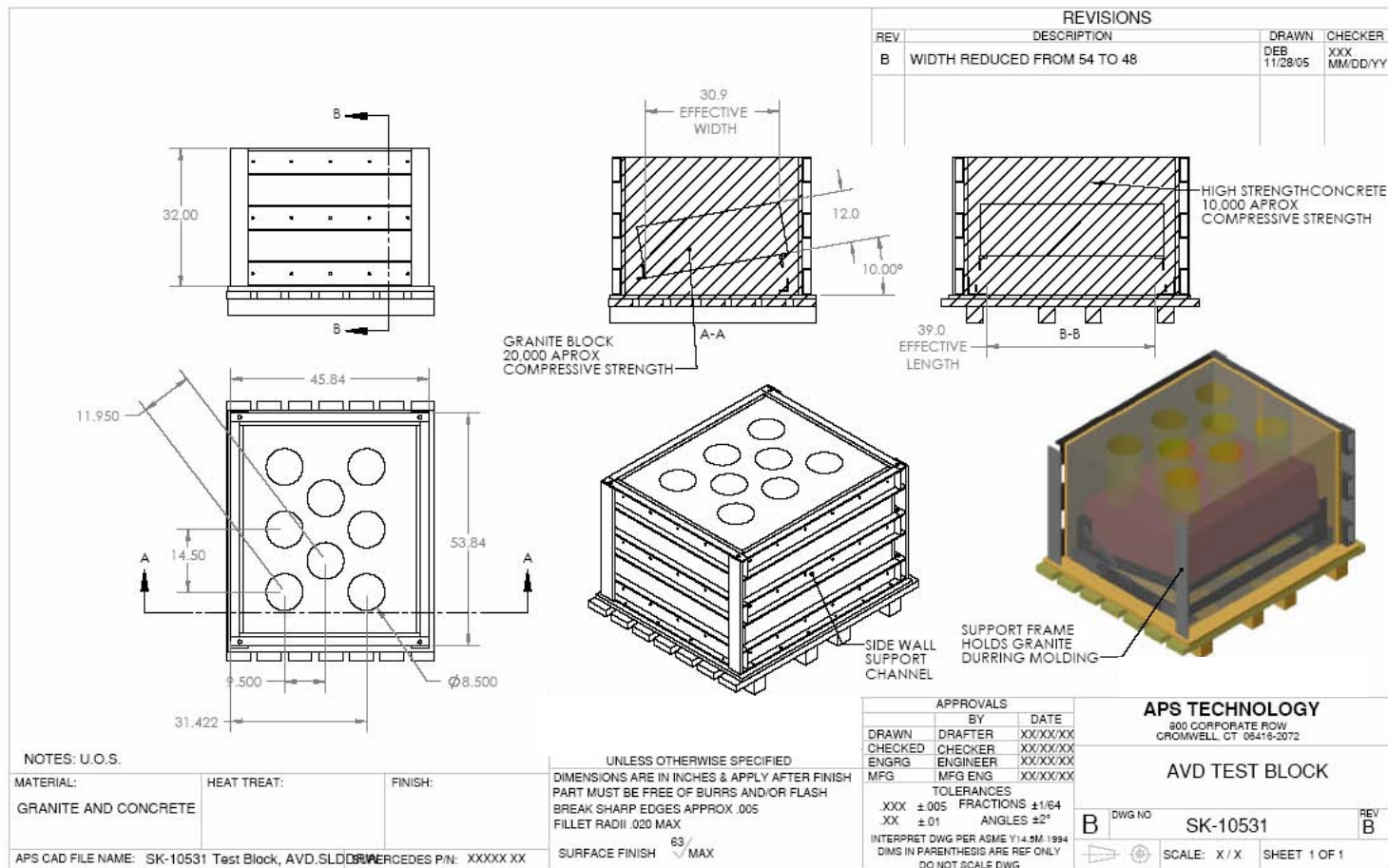


Figure 14: Concrete block design used for TerraTek testing

DVMCS TerraTek Test ROP Relative Improvement*

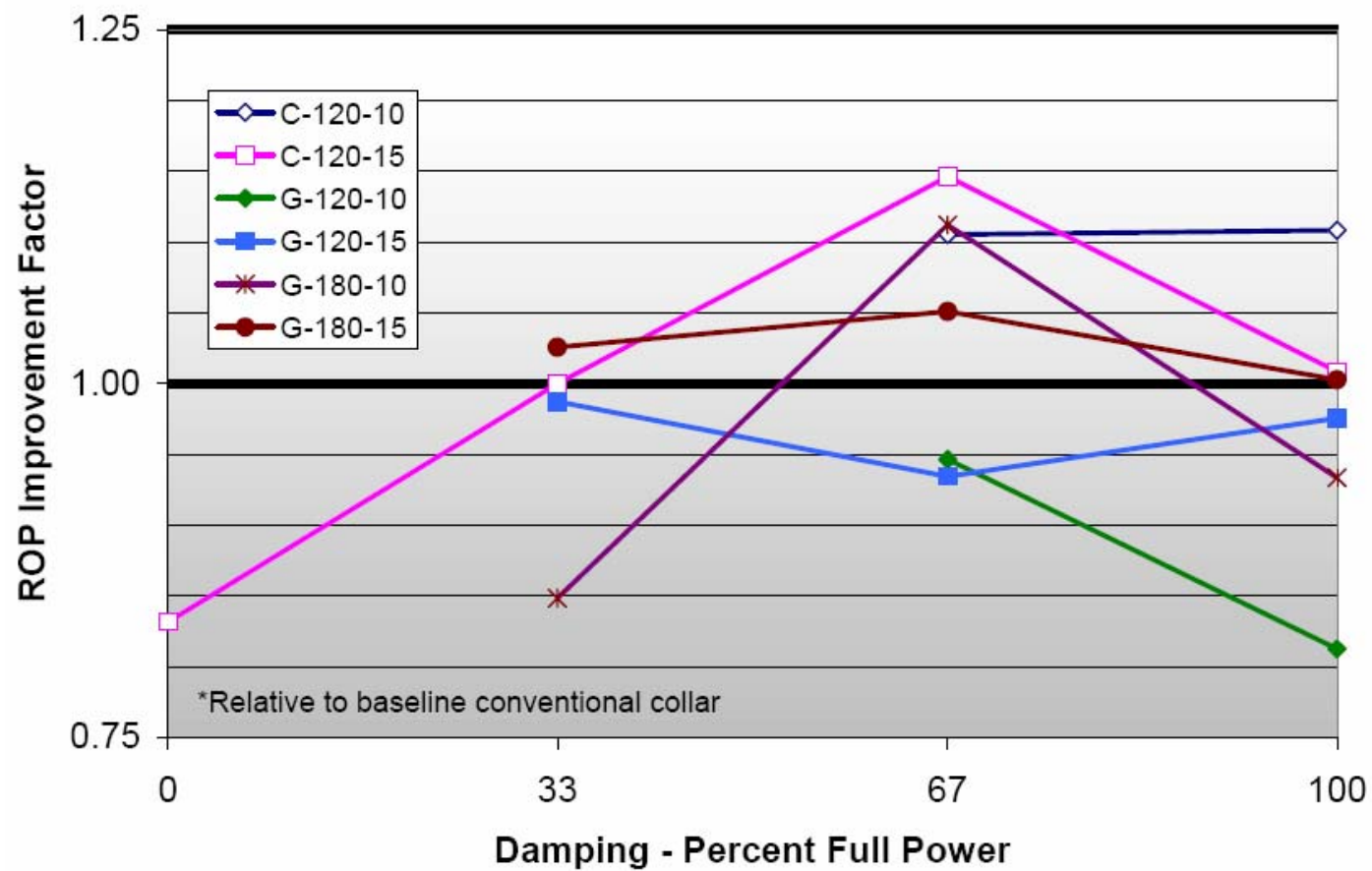


Figure 15: Effect of DVMCS on rate of penetration

DVMCS TerraTek Test Relative WOB Variation*

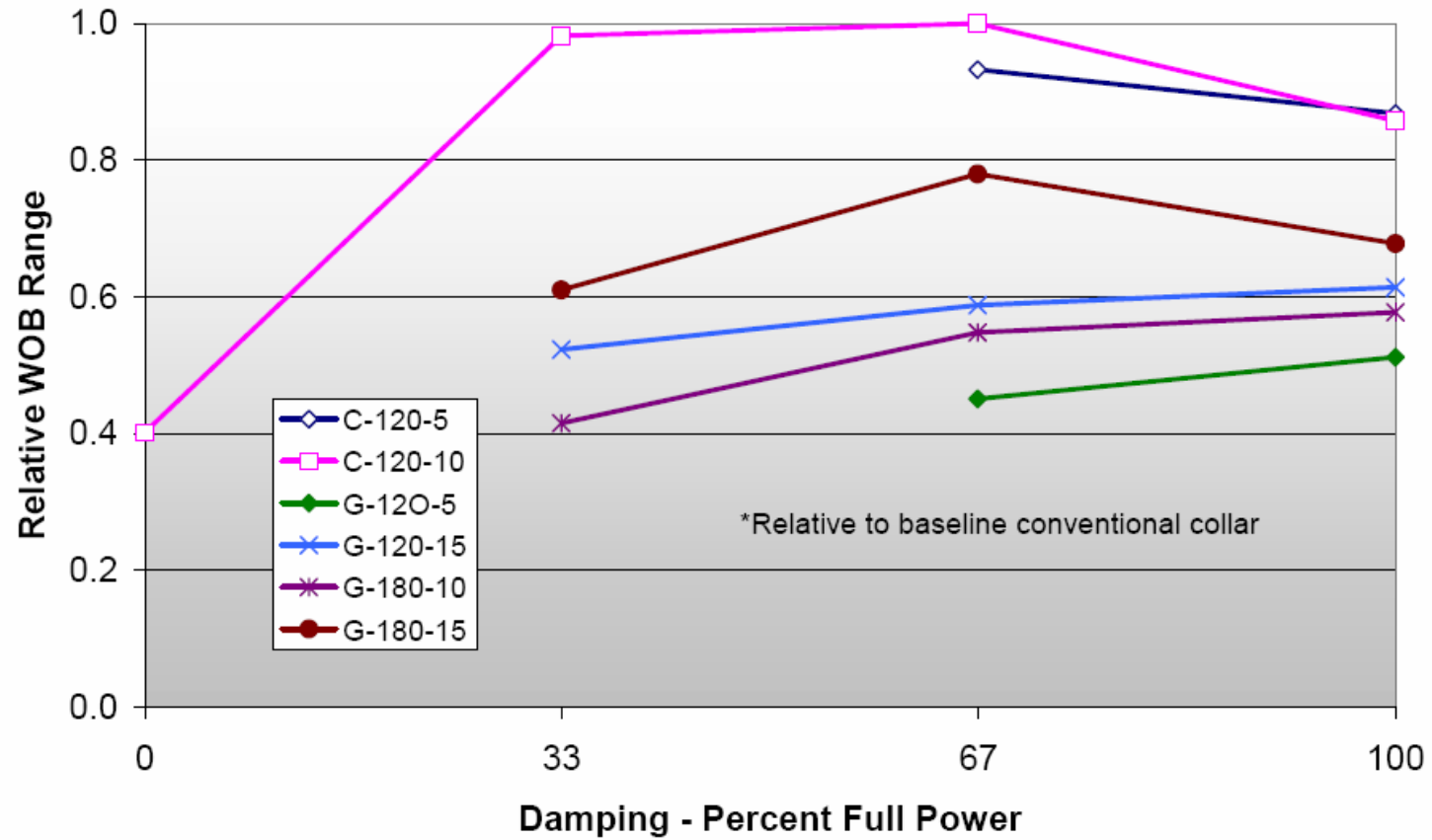
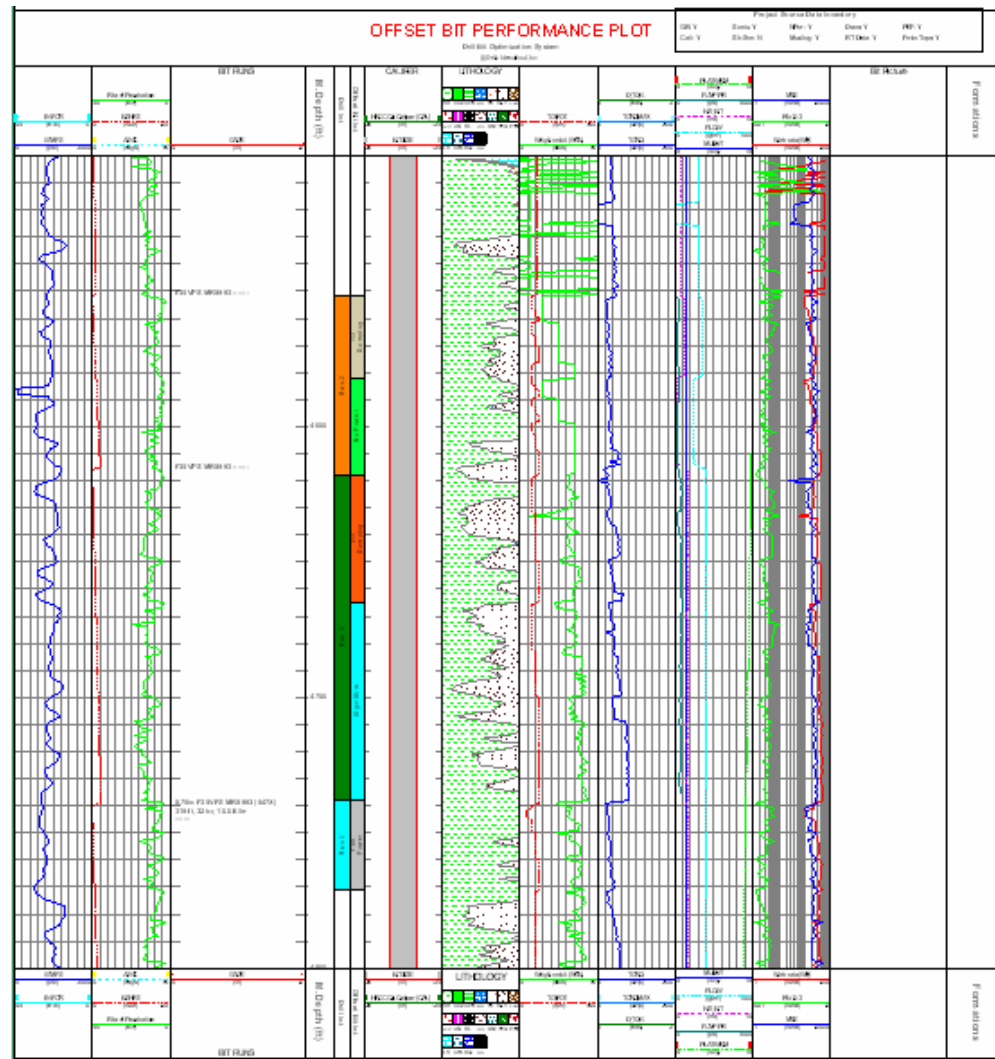


Figure 16: Effect of DVMCS on WOB variation



Figure 17: Field prototype valve assembly mounted in flow loop loading frame (left) with output monitors (right)



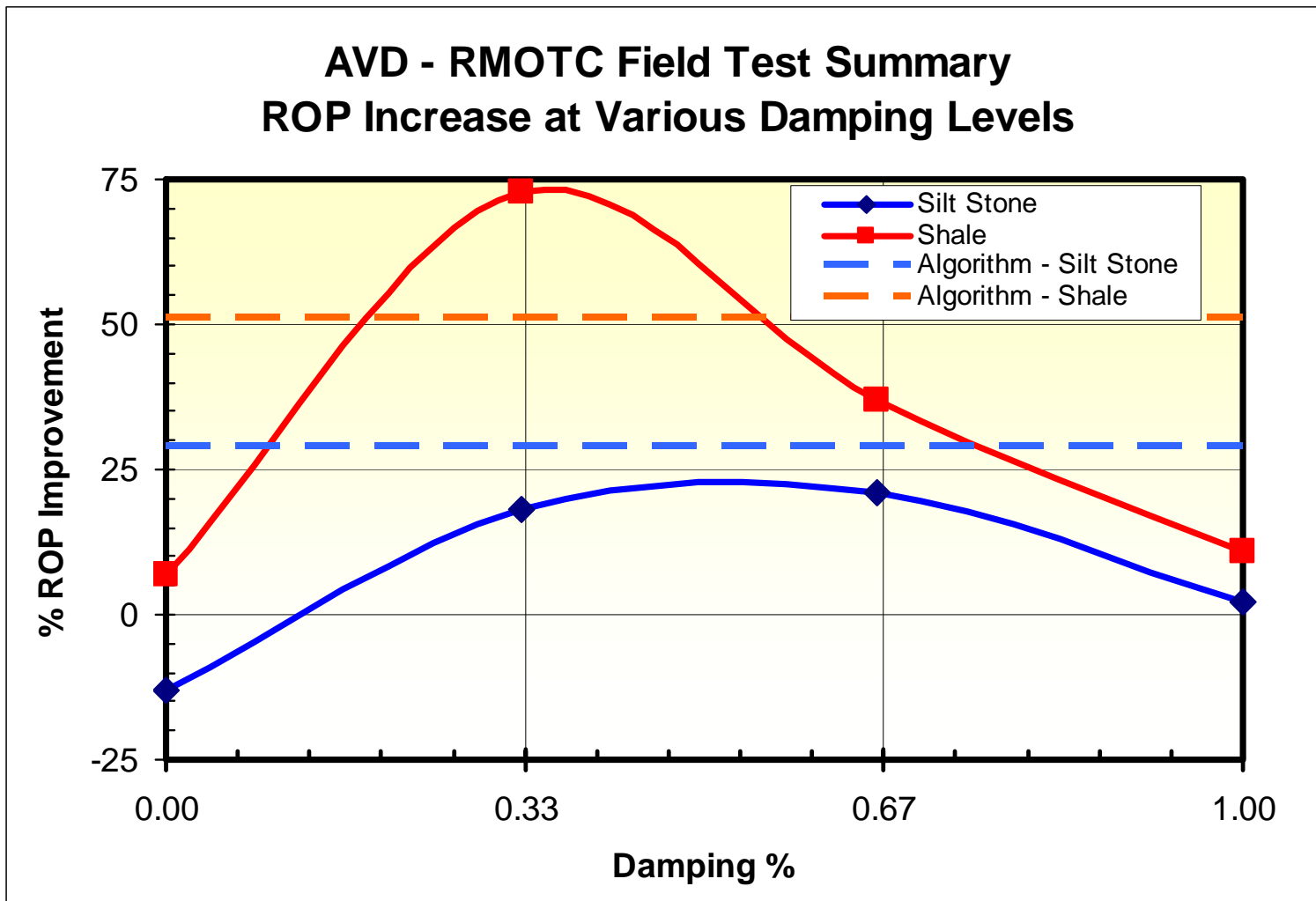


Figure 19: Preliminary Analysis of Effect of DVMCS on ROP

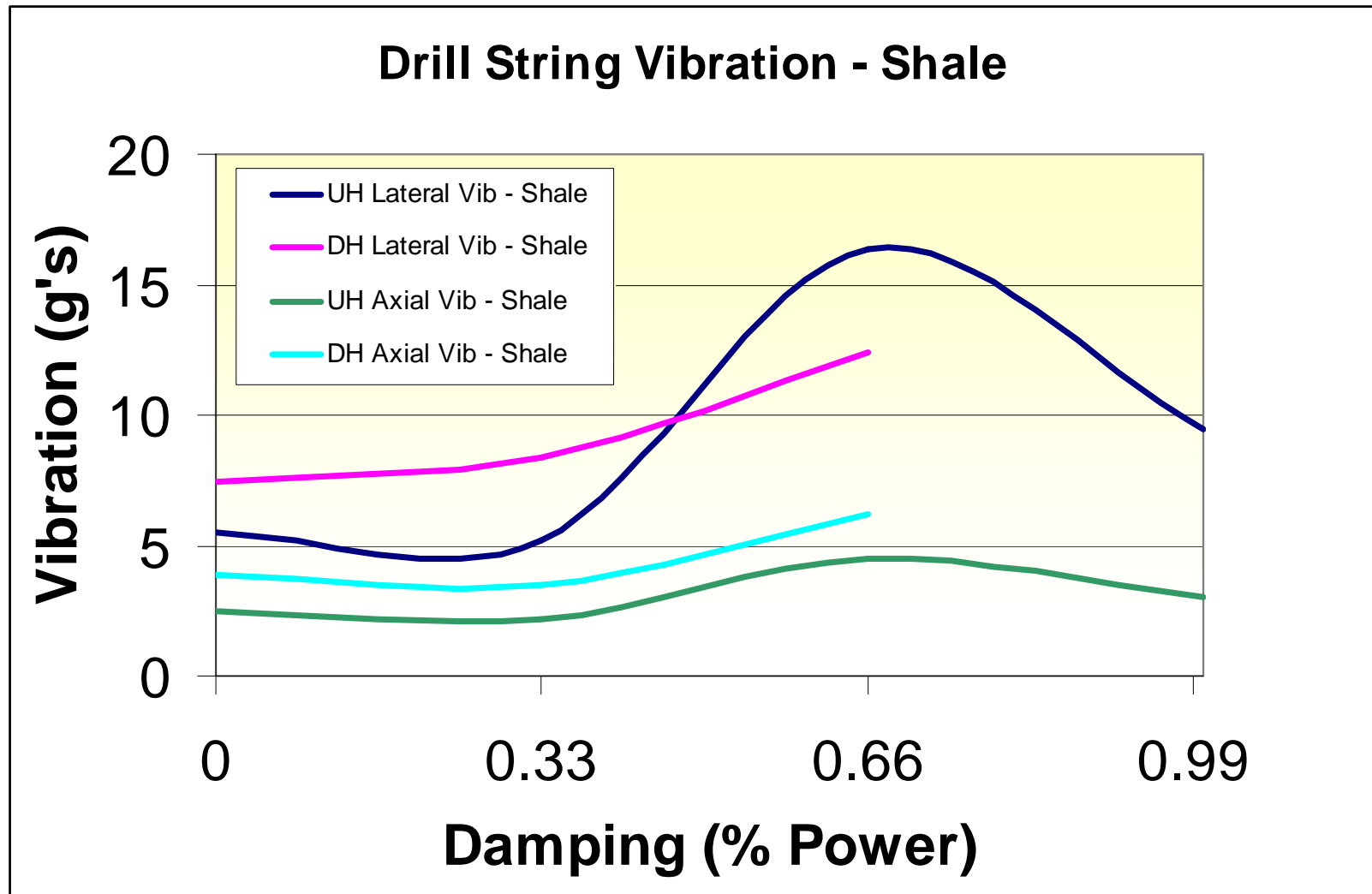


Figure 20: Effect of DVMCS on vibrations in shale

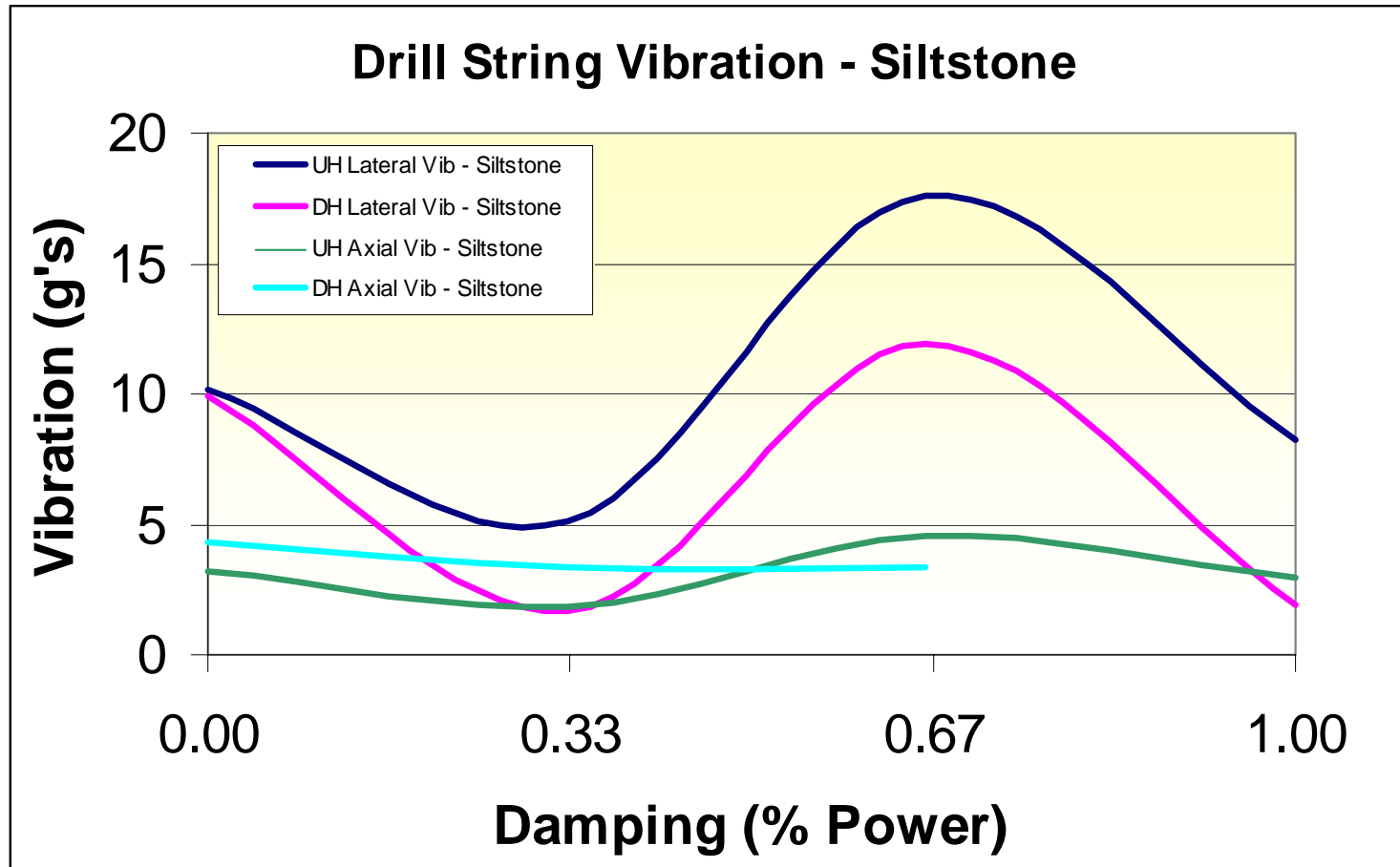


Figure 21: Effect of DVMCS on vibrations in siltstone

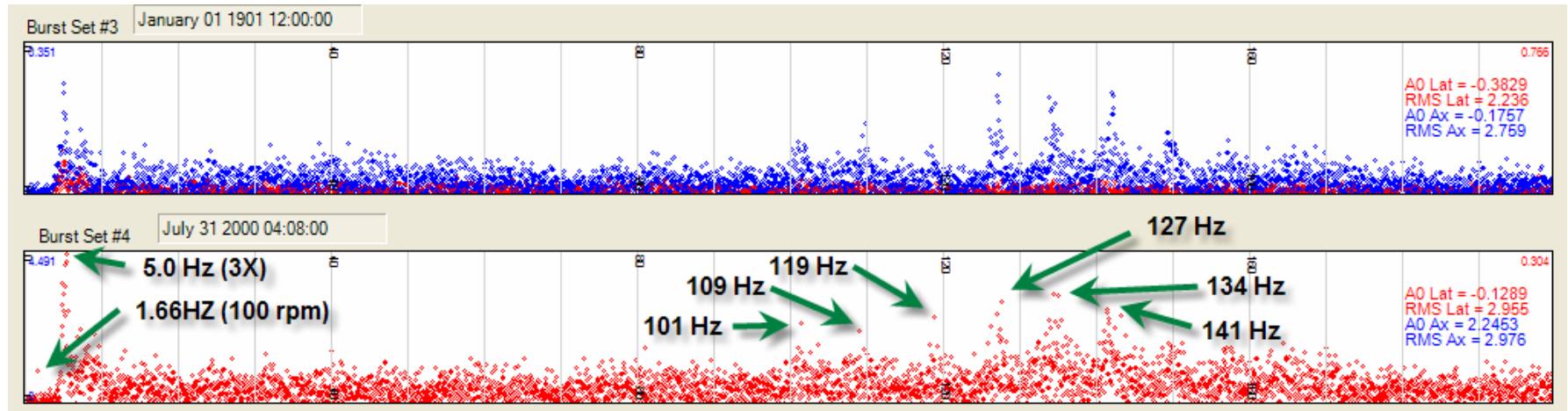


Figure 22: Burst data Fourier analysis - Lower vibration sensor

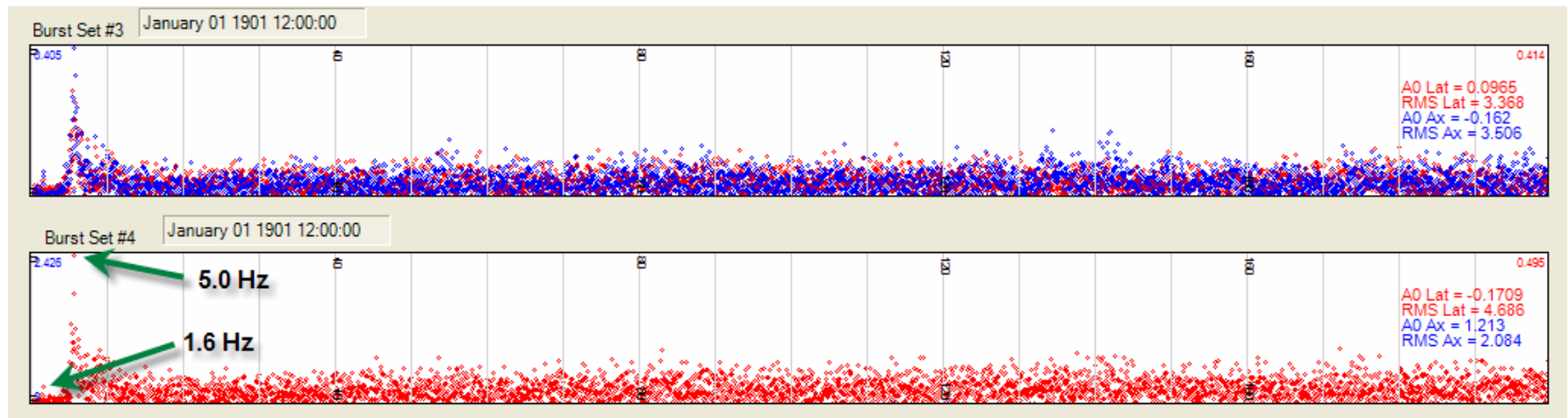


Figure 23: Burst data Fourier analysis - Upper vibration sensor

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