DRILLING VIBRATION MONITORING & CONTROL SYSTEM*

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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.

APS Technology is developing a unique system to monitor and control drilling vibrations in a 'smart' drilling system. This system has two primary elements:

- The first is a multi-axis active vibration damper 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 3-axis 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.

This paper is an interim report on the design, modeling and laboratory testing of the Drilling Vibration Monitoring & Control System (DVMCS) over the first year of the program.

Background

The drilling environment, and especially hard rock drilling, induces severe vibrations into the drillstring. The result of drillstring vibration is premature failure of the equipment and reduced ROP. The only means of controlling vibration with current monitoring technology is to change either the rotary speed or the weight on bit (WOB). Changes of the rotary speed and/or WOB to reduce vibration often have a negative effect on drilling efficiency.

Shock subs are not a universal solution, as they are designed for one set of conditions. When the drilling environment changes, as it often does, shock subs become ineffective and often result in increased drilling vibrations, exacerbating the situation. As one study[1] concluded: "Most of the

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shock subs tested showed a definite reduction in the axial accelerations experienced in the drillstring *above the shock sub…. The accelerations at the bit were little affected*, but even at the same accelerations the dynamic forces at the bit were probably reduced. Clearly the best place to run the shock sub is near the bit to minimize both axial and lateral accelerations. Even though it provides some benefit in terms of reducing axial vibrations when run at the top of a packed BHA, *it increases the risk of encountering high lateral vibrations when run in this position. These vibrations may cause more problems in terms of fatigue damage than will be offset by the reduction in the axial vibrations*." [*emphasis added*.]

Drillstrings develop vibrations when run at critical rotary speeds, and these vibrations are difficult to control due to the strings' long length and large mass. Operating at a critical speed imparts severe shock and vibration damage to the drillstring, fatigues drill collars and rotary connections. Vibrations also cause the drillstring to lift off bottom, reducing ROP. The effect of axial, lateral and torsional (stick-slip or bit whirl) vibration upon drilling have been documented in both the laboratory[2] and the field[2,3]

The natural frequencies of the drillstring often fall in the range excited by typical drilling speeds, between 0.5 Hz and 10 Hz depending on the BHA and length of the drillstring. There are many sources that excite drillstring vibrations, including bits, motors, stabilizers and drillstring imbalance. For example, a tricone bits imparts an excitation frequency of three times the rotary speed. If rotating between 120 and 180 rpm, the excitation frequency is 6 – 9 Hz. Mud motors are also significant sources of excitations on the drillstring. The rotor of the mud motor moves in an eccentric orbit that oscillates several times per revolution. Depending on the lobe configuration of the motor, excitations occur between 1 and 30 Hz.

The best situation for a drillstring is to operate below its lowest critical speed. By staying below this first critical speed, the drillstring is not excited by drilling frequencies and the bit maintains contact with cutting surface of the borehole. In **Figure 1**, this safe range is shown as **Zone A**. In this example, with a fundamental natural frequency of 6 Hz, this zone extends to 4 Hz, corresponding to a rotary speed to 80 rpm or less for a tricone bit. **Zone B** is the resonant range that results in high levels of vibration. Shock and vibration damage and low ROP occur in this zone. **Zone C** lies above the first critical speed of the drillstring. Vibrations levels are reduced compared to Zone A and B; however the bit does not maintain continuous contact with the drilling surface, since the natural frequency of the drillstring is lower than the excitations of the bit preventing it from reacting to vibrations. This discontinuous contact with the drilling surface of the borehole greatly reduces the ROP.

At present, there are no systems that provide real-time monitoring of downhole vibrations, *etc.*, that can operate at 175° C. Neither is there an active shock attenuation system to reduce the deleterious effects of drill bit vibration. The combination of these two features will represent a unique advance in the technology of drilling. These benefits will be particularly evident in deep gas drilling.

Objectives

The DVMCS combines the ability to measure and monitor the downhole drilling environment and to actively control and damping drilling induced vibrations. The ability to actively monitor and control the drilling vibration results in the following benefits:

- Increased ROP by keeping the bit in contact with the cutting surface.
- Increased bit life by eliminating shock and vibration damage.

- Increased MWD / LWD sensor life.
- Reduced number of trips needed to complete a well.

The above combine to provide reduced drilling costs and time to complete the well.

Description of the Damper

The key component of this DVMCS is the MR Damper. The MR Damper 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 [4] and earthquake protection systems for buildings and bridges.[5]

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

Axial Vibration Modeling

APS has developed an analytical program to analyze drillstring axial vibration. This analytical model includes the MR fluid damper, the portion of the drillstring above the damper and the portion below the damper to the bit. The section of drillstring above the damper is considered to be free floating. The program models the spring stiffness and the damping. The damping can be altered to account for various vibration conditions. For example, the damping may differ in the upward and downward directions. The vibration forcing functions are input as displacements of the bit. The height and number of vibrations per revolution are specified. A sample screen is shown below in **Figure 3**.

One of the objects of this program is to reduce vibration both above and below the damper. This is accomplished by smoothing out the discontinuities encountered by the bit as it drills ahead.

The model has the capability to simulate the drilling environment by looking at the depth of cut along the discontinuities. It can also calculate the percentage of time that the bit is in contact with the formation. It is found that, under some conditions, the vibrations increased the discontinuities of the well bottom as a result of resonant conditions. In the ideal situation, the damper changes the force applied by the bit, allowing it to reduce the discontinuities and smooth out the drilling.

We considered a wide range of spring stiffness and damping properties to determine the optimum values. Spring rates of 10,000 to 60,000 lbs/in were analyzed. The best compromise was found at 30,000 lb/in for the 6 3/4 " tool. Lower spring rates would require large displacements for the damper, while stiffer springs do not provide enough motion for the damper to effectively reduce vibration.

Several damping concepts were analyzed.

- A first approach was to have a damper provide high damping in the upward direction and low damping in the downward direction. It was found, however, that this increased the vibration by gouging out the troughs of the discontinuities, leading to increased displacements at the bit.
- Another method investigated was to provide increased damping at high bit acceleration levels and less damping at lower acceleration levels. This gave improved results.
- The constant damping approach has so far provided the best solution. With the proper damping level, the damper can smooth out the discontinuities and provide smooth drilling. The optimum damping values are quite different for different drilling conditions. Varying WOB and ROPs require different damping coefficients which therefore must be constantly adjusted to provide optimum drilling conditions. The DVMCS system is designed to provide this adjustment
- Additional control methods are still being investigated. One is a constant force damper that might provide optimum damping over a wider range.

Damping Results

The figures below show a sampling of the results of the damping calculations, for a 30,000 in.-lb. spring constant at nominal 5,000 and 30,000 lbs. WOB. In each figure, the blue line represents the maximum value observed and the red line the minimum. Comparing **Figure 4** to **Figure 5**, it is seen that the damping is optimum in the range of 500-1800 lb.-sec./in. at 5,000 lbs. WOB, but must be in a range of 2,000-4000 at 30,000 lbs. WOB. The balance of the figures show the 30,000 lbs. WOB results; the 5,000 lbs. results were similar, but at the lower spring constant values. It can be seen that in the optimum damping regime, all of these are greatly improved.

Figure 6 and **Figure 7** show two measures of bit bounce – the acceleration of the bit and the fraction of time it is in contact with the formation.

Figure 8 shows the change in the ROP with the damper. The ROP is approximately double that of that with no damping.

Figure 9 shows the amount of movement needed in the damper; a critical parameter in the design of the system.

Calculating and Monitoring the Severity of the Vibration Modes

One of the drawbacks to existing vibration monitors is their inability to differentiate between the various vibration modes. While there are capable of recording maximum vibration levels, they are not very good at determining the mode of vibration that is responsible for the measurement. Only when the tool is returned to the surface and the data analyzed can the vibration modes can be determined, and by then it is too late to be of any use. Knowing the mode of vibration is extremely useful in providing corrective drilling modifications to eliminate the vibration.

There are several modes of vibration responsible for poor drilling and tool damage. Listed in order of decreasing severity these are:

- **Backward whirling** – occurs when the drill collar contacts the borehole, causing it to race around the hole in a direction opposite that of the rotation of the collar. The speed of the backward whirl depends on the difference between the diameter of the borehole and the collar outer diameter.
- **Stick-Slip** – occurs when a section of the rotating drillstring is momentarily caught by friction against the borehole, then releases . Stick-slip can be severe enough to stop the rotation at the bit. When the friction is released, the collar rotation speeds up dramatically. This creates large centrifugal accelerations. Often stick slip and backward whirl occur in combination, exciting one another.
- **Bit Bounce** – results from axial drillstring vibrations caused by discontinuities at the bit. As the bit rides up and over the discontinuities, the drillstring reacts. At drillstring resonances, these vibrations become severe
- **Lateral Vibration or Whip** – occurs when a section of the drillstring between two stabilizers or supports is in resonance. This mode of vibration is self limited by the borehole, but can result in shock damage from contact with the wall. It also causes drill collar and connection fatigue.

We have developed a method to directly quantify the various vibration modes, with parameters that can easily be transmitted to the surface. The system uses four accelerometers and a magnetometer mounted in the sensor sub of the damper. By using different combinations of the accelerometer outputs, whirling, stick slip, bit bounce and lateral vibrations can be differentiated from each other.

Three of the accelerometers are mounted in the collar 120 degrees from each other and oriented to measure radially. The fourth accelerometer is mounted in the axial direction. A magnetometer is also mounted in one of the pockets. The pockets can also include strain gages to measure WOB and TOB, thus providing a complete downhole drilling diagnostics tool.

Mounting the accelerometers in the radial direction (**Figure 10**) permits different vibration modes to be calculated directly. This would not be true if they were mounted in the tangential direction. Tangential accelerometers read *zero* acceleration at constant rotary speed; and are therefore only sensitive to rotational accelerations. Radial accelerometers measure centrifugal force that is directly related to rotational speed. From this, rotary speed, stick-slip and backward whirling can be directly calculated. The magnetometer is used as a backup to measure rotary speed. Axial vibration measurement requires only the single axial accelerometer.

These parameters are calculated as follows:

Rotary Speed:

The centripetal acceleration, $A_c(t)$, is calculated as follows:

$$A_c(t) := \frac{A_1(t) + A_2(t) + A_3(t)}{3}$$

Since $A_c(t) = \omega^2(t) \cdot r$, where r is the sensor radius and ω is the angular rotation rate in rad/sec. Therefore:

$\omega(t) = \sqrt{A_c(t) / r}$, and the instantaneous RPM is given by:

$$RPM = \left(60 / 2\pi\right) \cdot \omega(t)$$

Stick-Slip:

The stick-slip effect is given the maximum of the rotary speed, as determined above:

$\omega_{SS} = \max\,[\omega(t)]$

Backward whirl is determined by the peak of the following expression:

$$A_w(t) := A_1(t) + A_2(t) \cdot \cos(120 \cdot \deg) + A_3(t) \cdot \cos(240 \cdot \deg)$$

Lateral Vibration (Whip) has two components. The acceleration in the x-direction is given by:

$$A_x(t) := \left(\frac{A_2(t) - A_c(t)}{\cos(30 \cdot \deg)} - \frac{A_3(t) - A_c(t)}{\cos(30 \cdot \deg)} \right) \cdot \frac{1}{2}$$

The y-acceleration is given by:

$$A_y(t) := \left(A_1(t) - A_c(t) + \frac{-A_2(t) + A_c(t)}{\sin(30 \cdot \deg)} + \frac{-A_3(t) + A_c(t)}{\sin(30 \cdot \deg)} \right) \cdot \frac{1}{3}$$

Thus, the magnitude of the lateral vibration is given by the vector sum:

$$A_{Lat}(t) := \sqrt{A_x(t)^2 + A_y(t)^2}$$

Collar Orbits can be displayed by plotting the x and y components of the lateral vibration over time.

Axial Vibration is simply the output of the axial accelerometer.

Tool Design

An overview of the DVMCS is shown below in **Figure 11.** The tool has many features of a conventional shock sub, including: a stack of Belleville washers to support the weight applied to the bit; bearings to absorb the axial and torsional loads, *etc*. The key feature is the adjustable damper which utilizes MR fluid. By applying an electromagnetic field to the fluid in the damper, it will be possible to continuously adjust its viscosity, and thereby the damping factor of the tool.

The key features of MR damper are shown in **Figure 12**, the test piece used in the testing described below. The MR fluid will be in the volume between the mandrel (2) and the housing (1). A series of coils wrapped in the grooves in the mandrel will create bucking fields, which will be strongest in the gaps between the coils. The MR fluid in these areas will become more viscous as a function of the field strength, thereby varying the damping of the motion of the mandrel relative to the housing. Other aspects of this test piece will be described below.

Experimental

Test Bench

The design of a test bench to evaluate the performance of the DVMCS was a significant task in itself. A simple vibration table would not suffice.

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 13**. 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, can have a configuration to mimic a variety of degrees of irregularity of the well bottom.

Initial Test Results

[*Note*: Testing is currently underway, and we anticipate having additional results at the time this paper is presented.]

Before testing the entire system, we are evaluating the performance of the MR damper itself. The test piece of **Figure 12** is mounted on the test bench, as shown in **Figure 14**. The test bench pneumatic-hydraulic system is used for force the MR fluid around the mandrel to simulate the motion of the inner section of DVMCS relative to the collar. Referring to **Figure 13**: the pneumatic pressure in the cylinder (1) is increased until the hydraulic cylinders (2) begin to move, driving the MR fluid through the test piece. Now, referring to **Figure 12**: The MR fluid enters the cylinder through the end cap (13), and flows around the mandrel (2), exiting through ports in the other end cap (3). A known volume of fluid is forced through the assembly with each stroke of the hydraulic pistons. Transducers (8) monitor the pressure at various points in the assembly. The time required for the passage of this fluid is measured. This procedure is repeated with different levels of power applied to the coils on the mandrel.

The damping coefficient is defined by the ratio of force over velocity. With the known cross-sectional area on which the pressure acts and the measured pressure, the force can be calculated. The time needed for the passage of a known volume of fluid through a known area gives the fluid velocity, and the damping coefficient can thus be calculated.

Some preliminary results are shown in **Figure 15**. As can be seen, the damping coefficient increases rapidly with applied power, and decreases roughly linearly as the pressure increases. The fluid pressure is proportional to the force applied, and these early results indicate that the damper will be able to support approximately 6,000 lbs. With much of the WOB supported by the spring stack, the damper will be able to provide the necessary damping at typical WOB values. The damping value with the power off, however, is somewhat higher than desired, making the damper too stiff for some situations. The gaps between the mandrel and housing are being adjusted to optimize the damper response.

Later, testing will be done on the full mockup of the DVMCS system (**Figure 11**) and monitor its response to various bit vibrations under simulated downhole conditions.

Conclusions

Although the project is still underway, we can draw some preliminary conclusions.

- The DVMCS should be able to provide actively controlled damping under typical downhole conditions.
- Results of the preliminary testing will be used to refine the MR damper geometry.
- The response of the drillstring with the DVMCS has been modeled, and these models will be refined based on the results of further testing.
- Sensors to monitor and characterize the downhole vibrations have been developed.
- Design of the full system is nearly complete.

References

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[2] *cf.*, *e.g.*, M.W. Dykstra, D.C.-K. Chen, T.M. Warren & S.A. Zannoni, “Experimental Evaluations of Drill Bit and Drill String Dynamics,” **SPE 28323**, presented at the 61st SPE ATCE, New Orleans, September 25-28, 1994.

[3] *cf.*, *e.g.*, S.L. Chen, K. Blackwood and E. Lamine, “Field Investigation of the Effects of Stick-Slip, Lateral and Whirl Vibrations on Roller Cone Bit Performance, **SPE 56439**, presented at the 58th SPE ATCE, Houston, October 3-6, 1991.

[4] *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 .

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

Figures

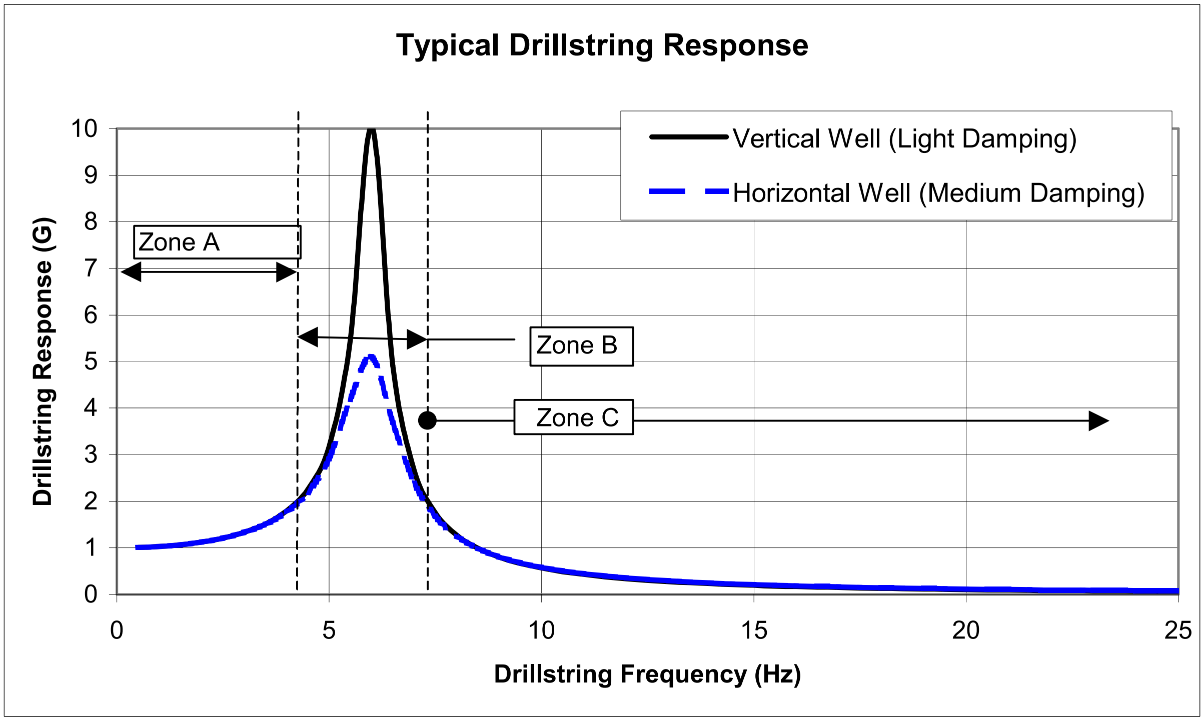

Figure 1: Frequency response of typical drillstring

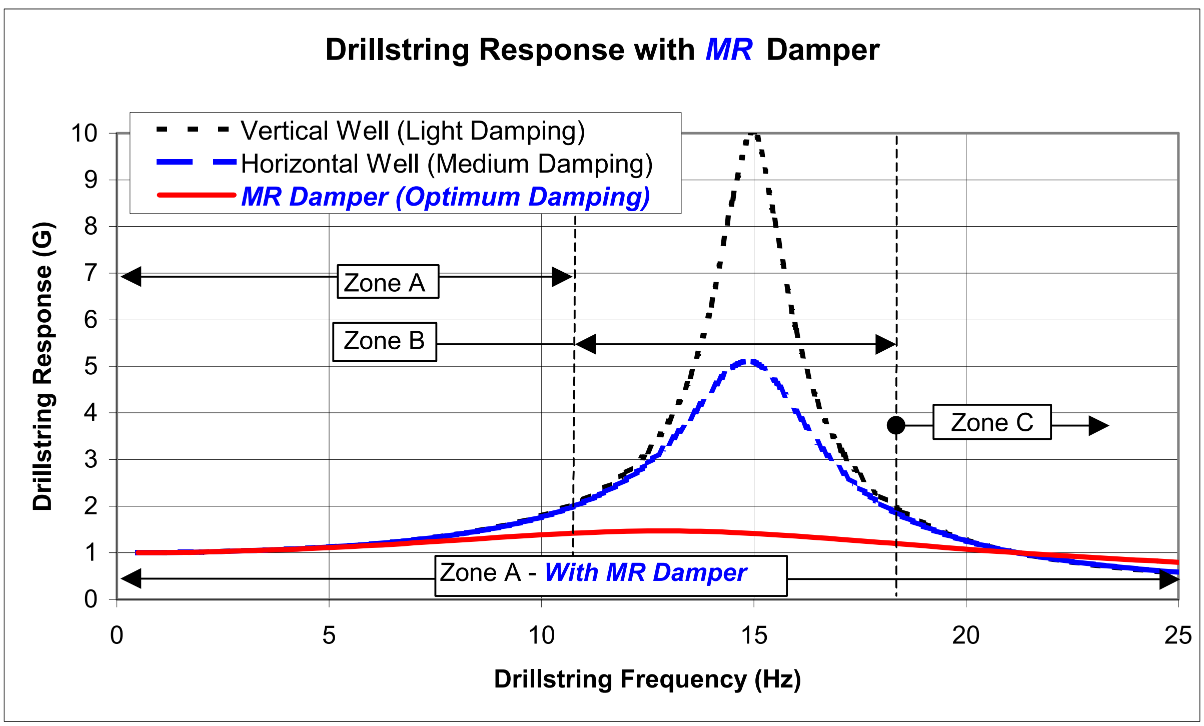

Figure 2: Drillstring response with an active MR damper in use

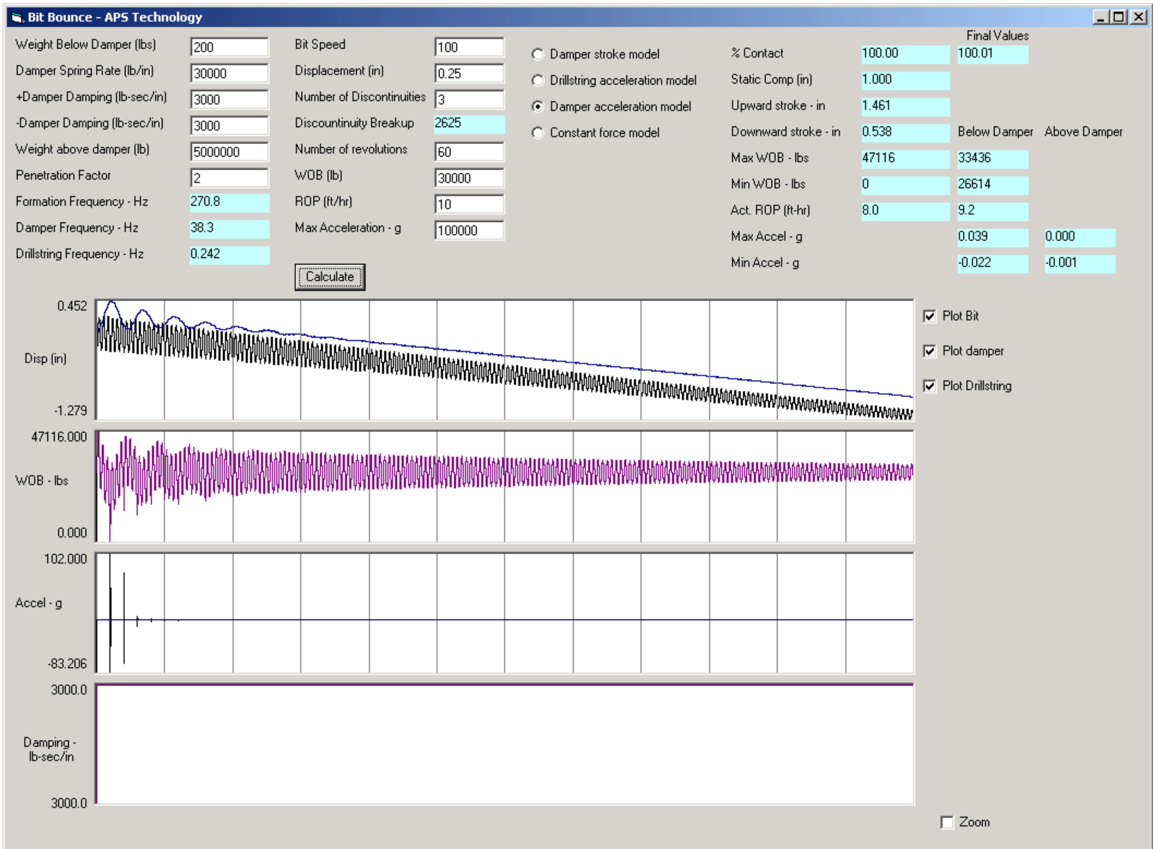

Figure 3: Sample screen from analysis program

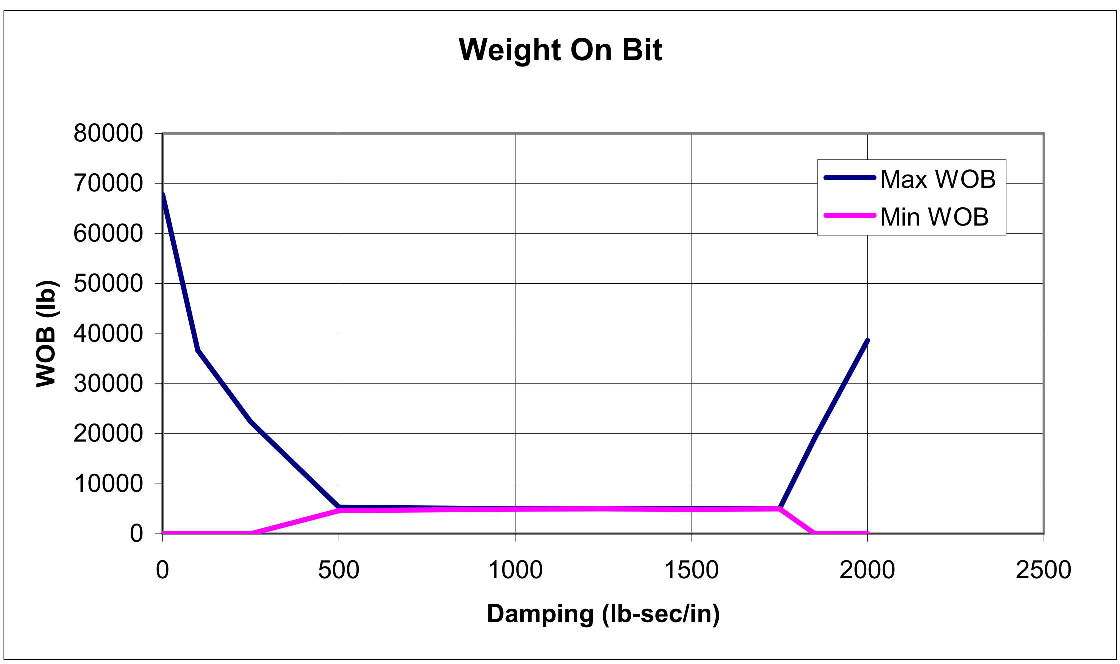

Figure 4: Measured WOB: 5,000 WOB - 30,000 in-lb Spring Rate

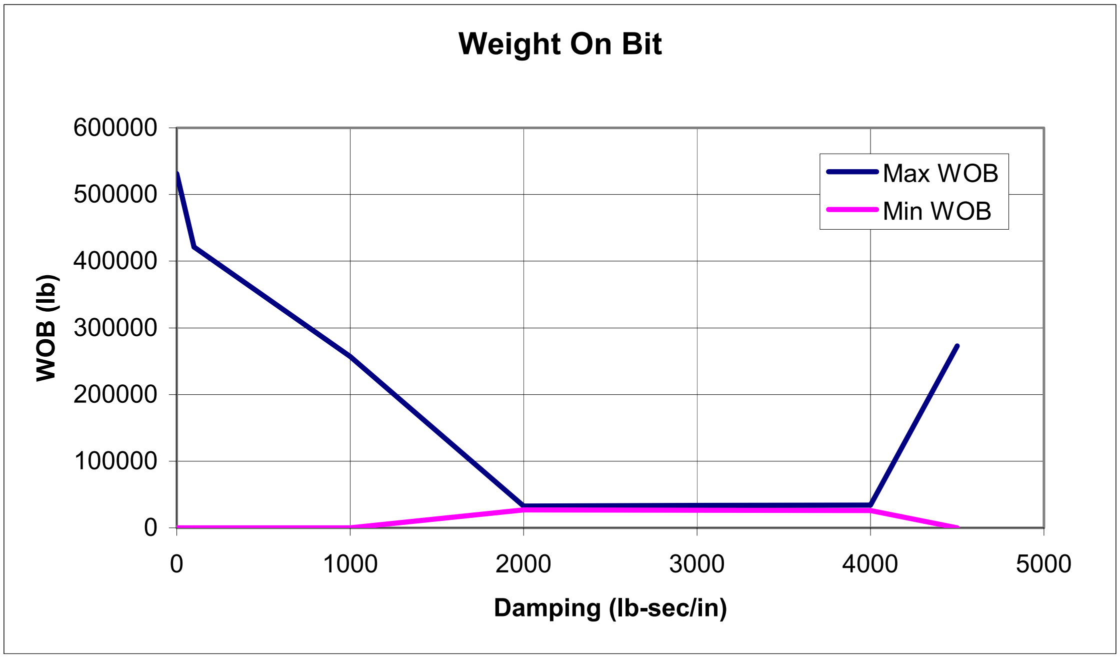

Figure 5: Measured WOB: 30,000 WOB - 30,000 in-lb Spring Rate

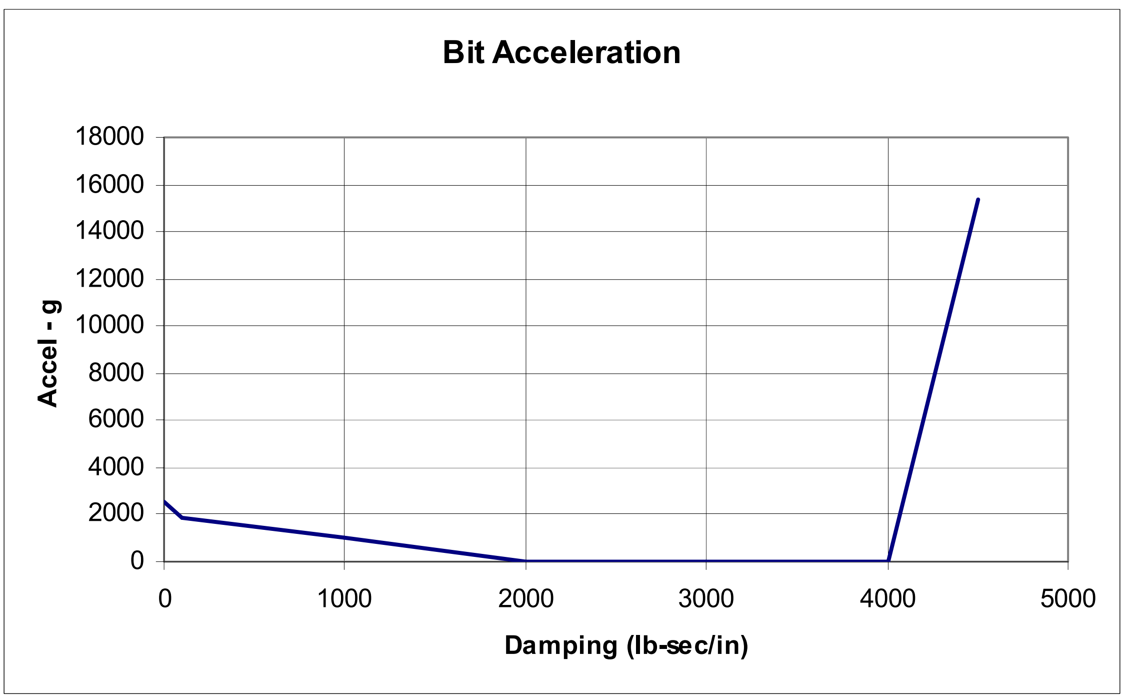

Figure 6: Bit acceleration: 30,000 WOB - 30,000 in-lb Spring Rate

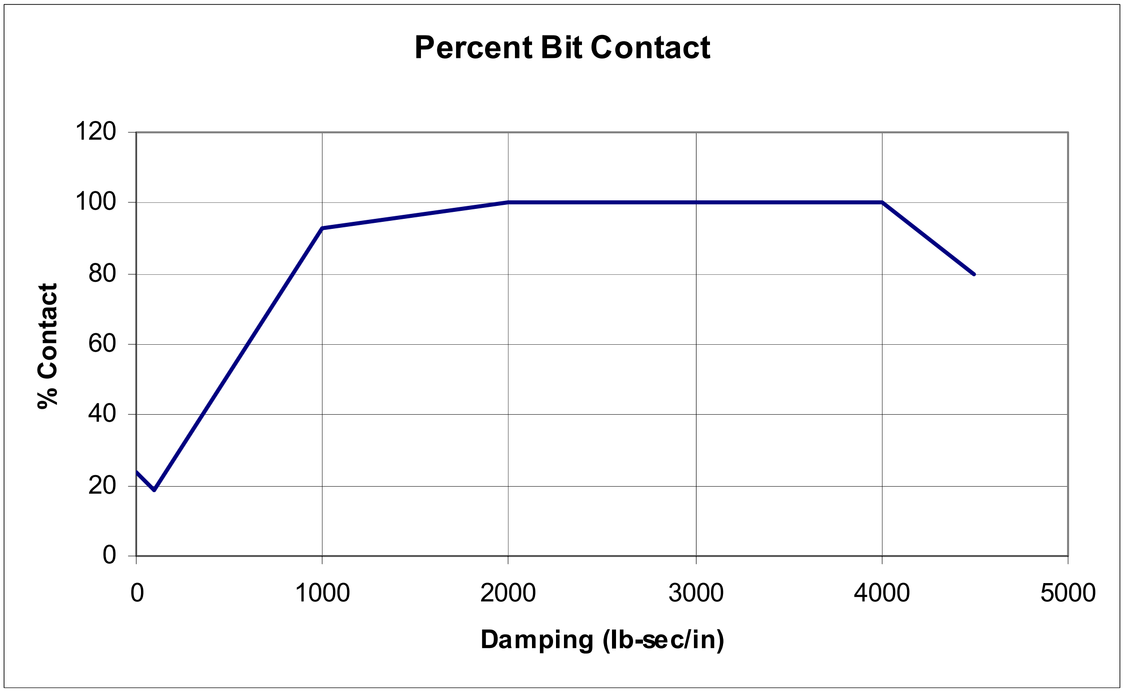

Figure 7: Bit contact : 30,000 WOB - 30,000 in-lb Spring Rate

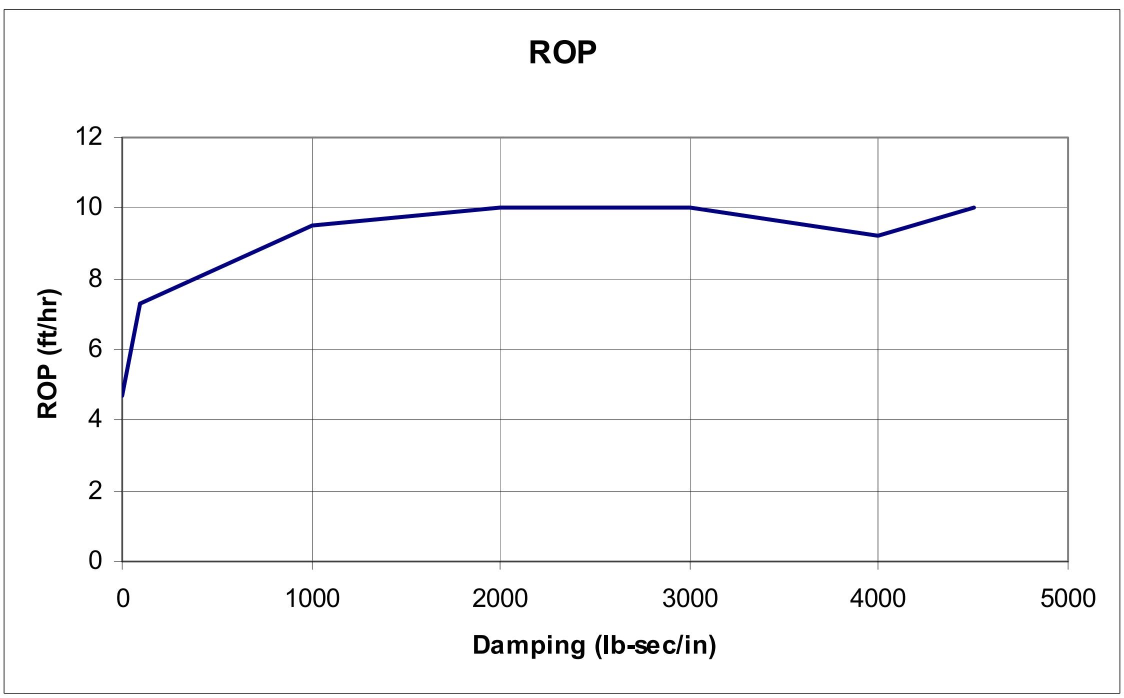

Figure 8: ROP: 30,000 WOB - 30,000 in-lb Spring Rate

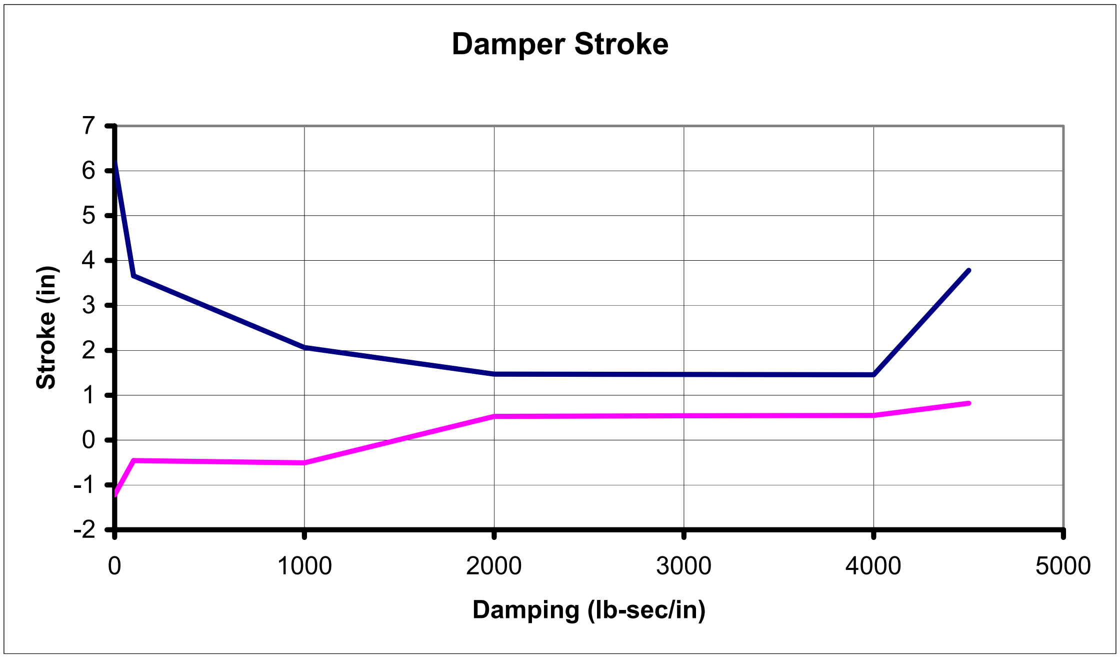

Figure 9: Damper movement: 30,000 WOB - 30,000 in-lb Spring Rate

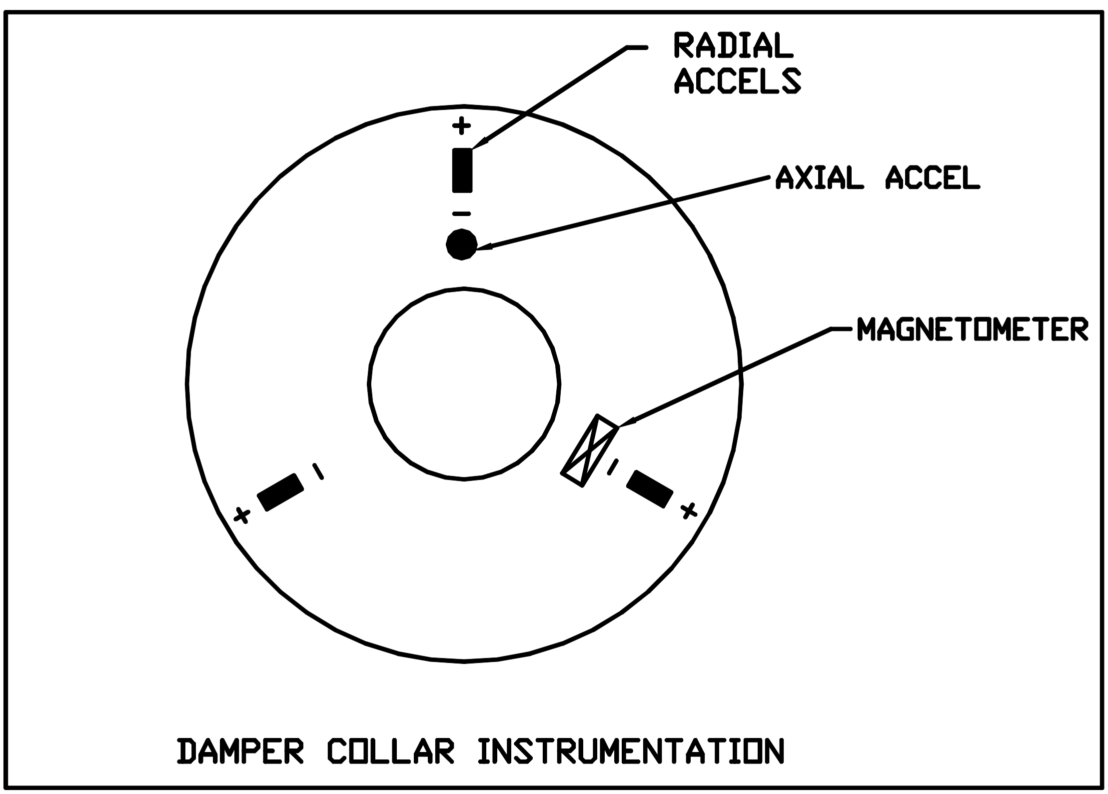

Figure 10: Drill Collar Sensors

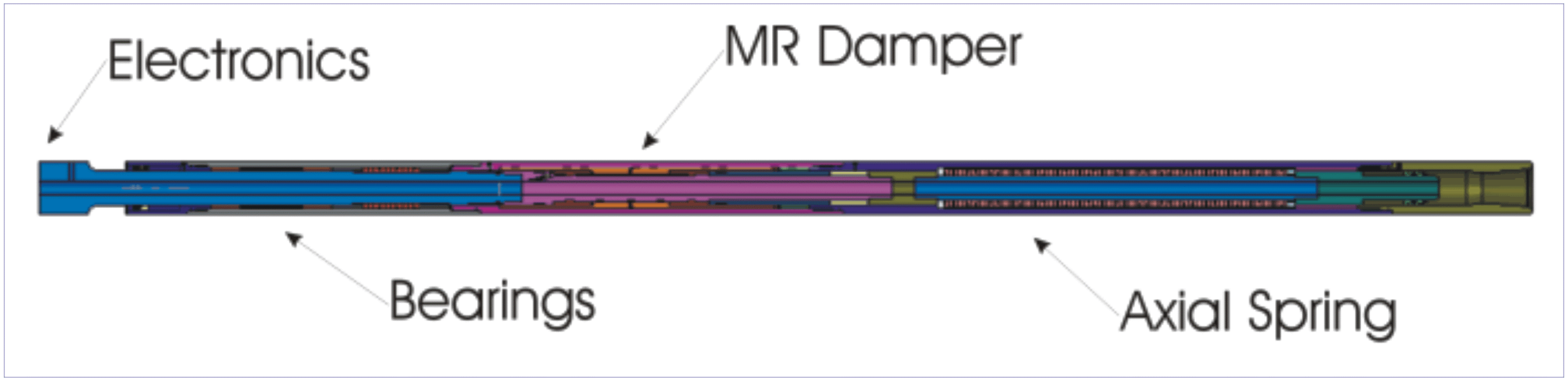

Figure 11: Schematic of DVMCS Tool

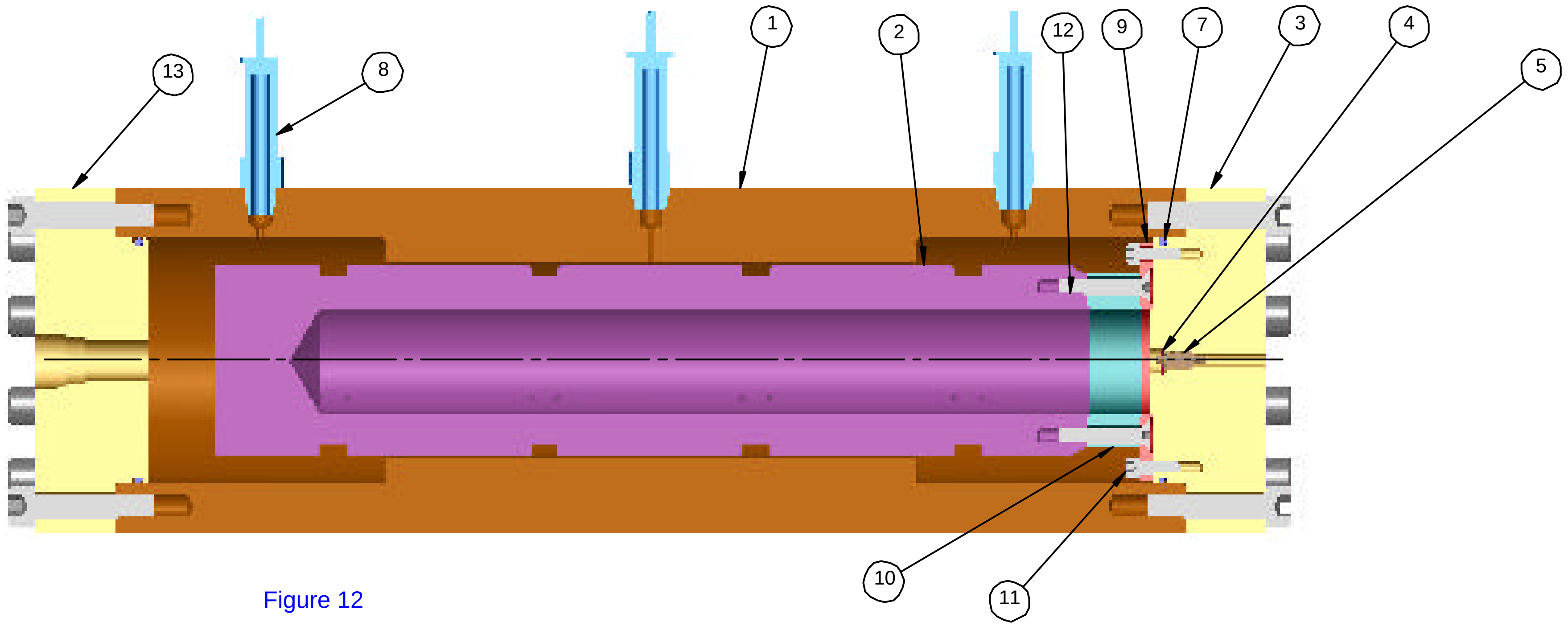

Figure 12

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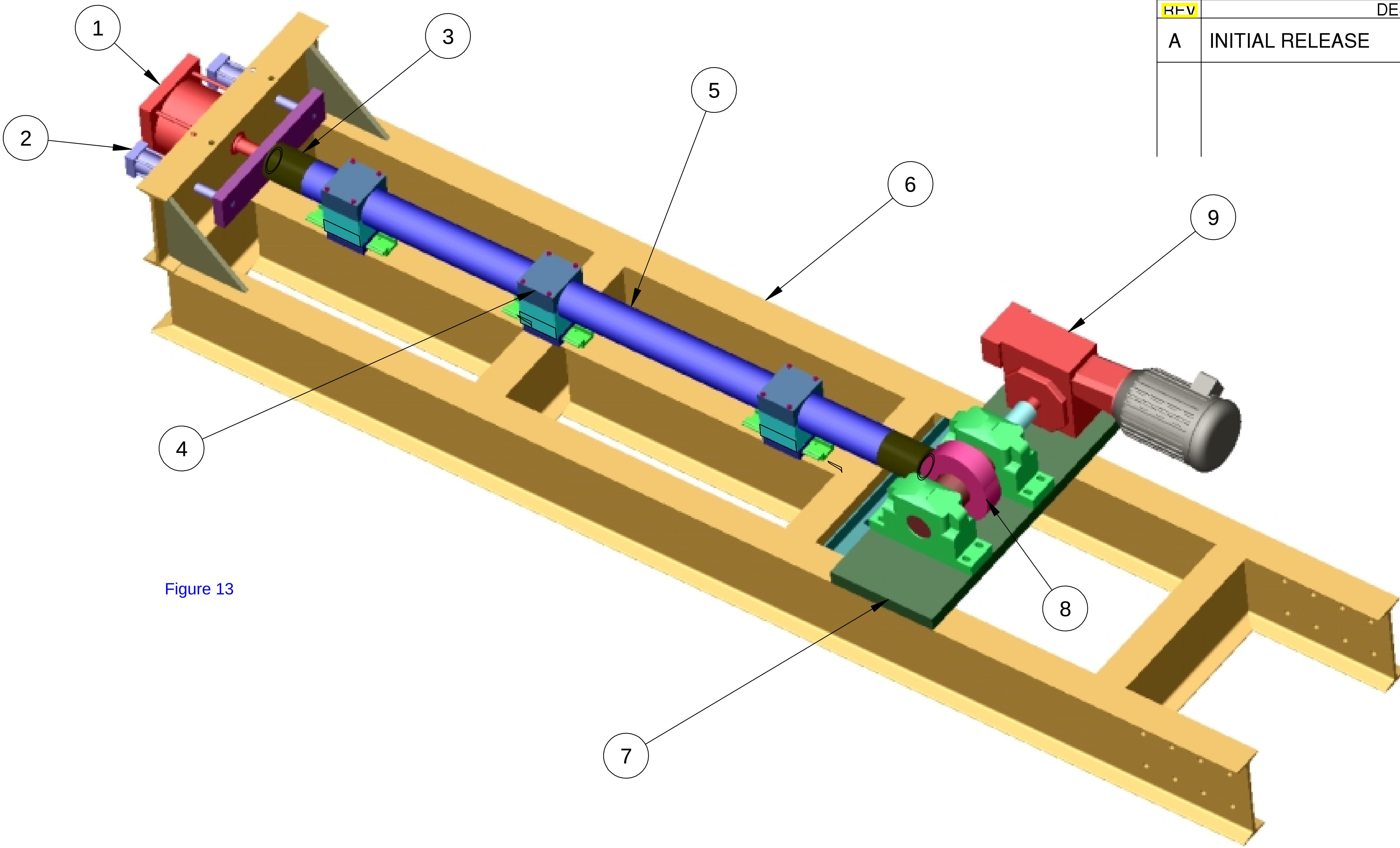

Figure 13

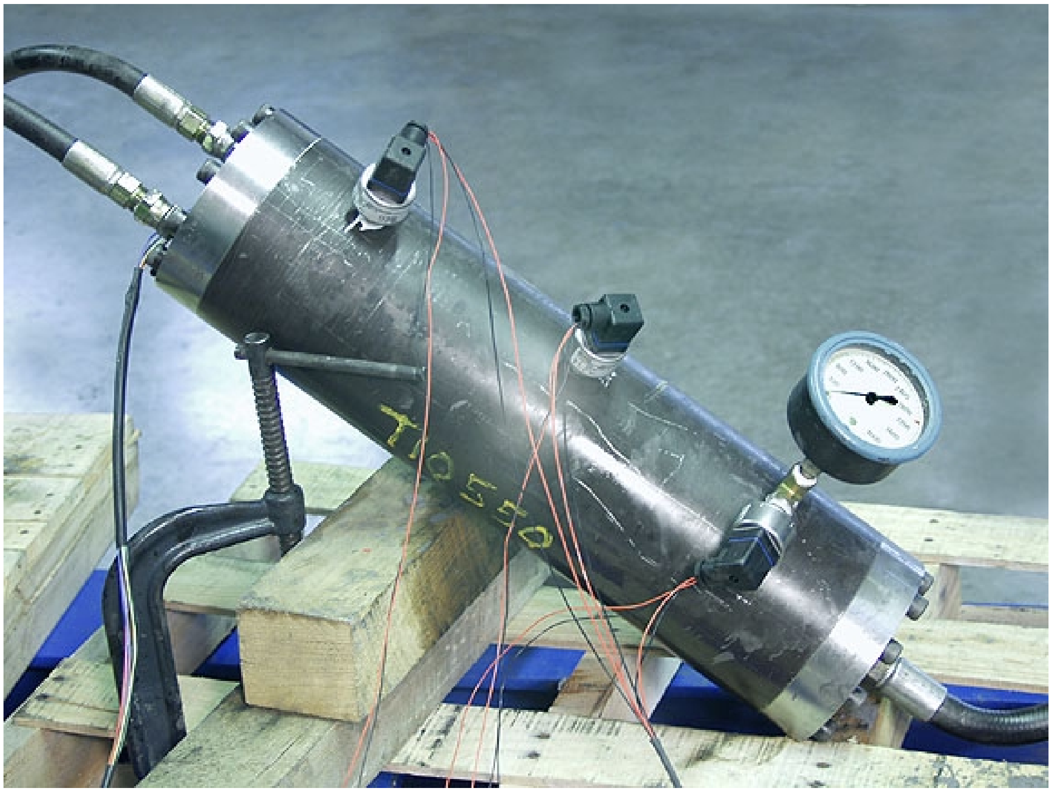

Figure 14: Photo of MR Valve Test Fixture on Test Bench

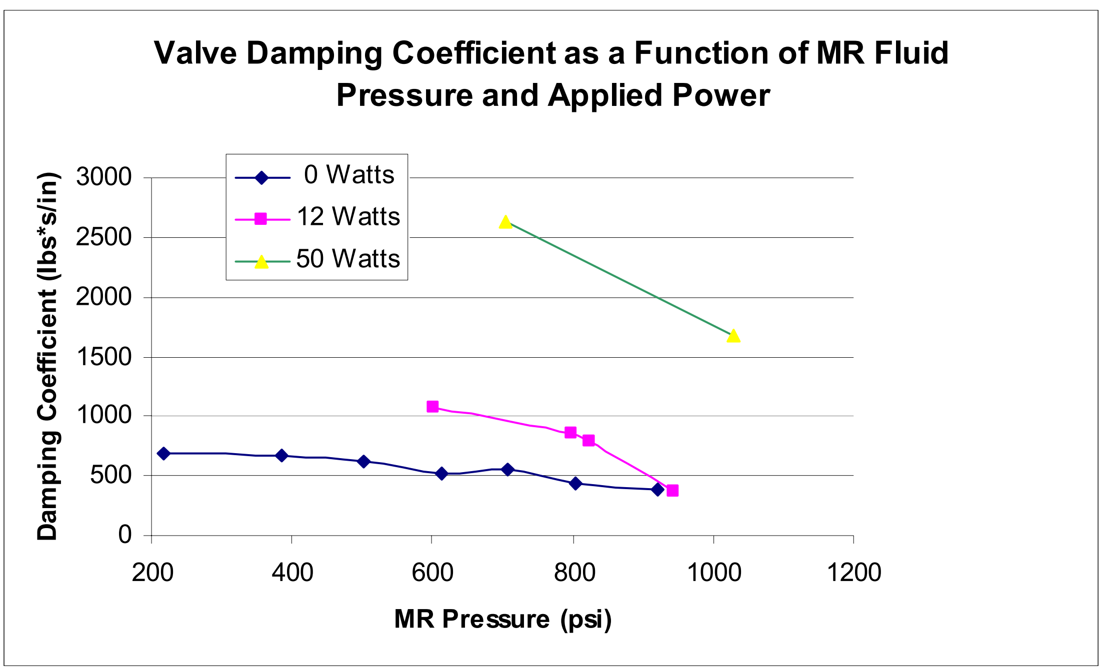

Figure 15: Preliminary Data: Measurement of MR Damper Damping Coefficient