DOWNHOLE VIBRATION MONITORING & CONTROL SYSTEM QUARTERLY PROGRESS REPORT #10

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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. 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.

The principal objectives of Phase II are: 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 drilling laboratories and test wells.

As a result of the lower than expected performance of the MR damper noted last quarter¹, several additional tests were conducted. These dealt with possible causes of the lack of dynamic range observed in the testing: additional damping from the oil in the Belleville springs; changes in properties of the MR fluid; and, residual magnetization of the valve components. Of these, only the last was found to be significant. By using a laboratory demagnetization apparatus between runs, **a dynamic range of 10:1 was achieved for the damper**, more than adequate to produce the needed improvements in drilling. Additional modeling was also performed to identify a method of increasing the magnetic field in the damper.

As a result of the above, several changes were made in the design. Additional circuitry was added to demagnetize the valve as the field is lowered. The valve was located to above the Belleville springs to reduce the load placed upon it and offer a greater range of materials for its construction. In addition, to further increase the field strength, the coils were relocated from the mandrel to the outer housing.

At the end of the quarter, the redesign was complete and new parts were on order. The project is approximately three months behind schedule at this time.

Table of Contents

Table of Contents	3
Executive Summary	4
Design	4
Redesign of laboratory prototype	5
Design of feedback system	5
Field prototype design.....	5
Experimental.....	5
Retesting of DVMCS prototype	5
Analysis	6
Other	8
Units	8
References	9
Appendix A: Magnetic Testing	10
Appendix B: MR Fluid Testing	17
Appendix C: AADE Paper.....	27

Executive Summary

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As a result of the lower than expected performance of the MR damper noted last quarter¹, several additional tests were conducted. These dealt with possible causes of the lack of dynamic range observed in the testing: additional damping from the oil in the Belleville springs; changes in properties of the MR fluid; and, residual magnetization of the valve components. Of these, only the last was found to be significant. By using a laboratory demagnetization apparatus between runs, **a dynamic range of 10:1 was achieved for the damper**, more than adequate to produce the needed improvements in drilling. Additional modeling was also performed to identify a method of increasing the magnetic field in the damper.

As a result of the above, several changes were made in the design. Additional circuitry was added to demagnetize the valve as the field is lowered. The valve was located to above the Belleville springs to reduce the load placed upon it and offer a greater range of materials for its construction. In addition, to further increase the field strength, the coils were relocated from the mandrel to the outer housing.

At the end of the quarter, the redesign was complete and new parts were on order. The project is approximately three months behind schedule at this time.

Design

Redesign of laboratory prototype

Complete.

Design of feedback system

The algorithms are installed in the system.

Field prototype design

Based on the results of the retesting, analysis and modeling performed this quarter, two significant changes were made in the design of the field prototype:

- The valve structure was inverted, putting the coils into the outer housing and leaving the mandrel with an uninterrupted diameter. This provides several advantages:
 - A stronger field can be generated in the gaps using this arrangement.
 - The uniform diameter of the mandrel is easier to manufacture and greatly simplifies testing.
- The MR valve component has been moved to a position above the Belleville spring stack. This will reduce the mechanical load applied to this component and allow a wider choice of alloys for its construction. The alloy can now be chosen on the basis of its magnetic properties and resistance to corrosion and erosion, without the emphasis on mechanical strength required by the earlier design.
- A demagnetization circuit was added to the damper design.

At the end of the quarter, the revised design was complete and parts were on order for the laboratory prototype MR valve, and the revised field prototype sub. Assuming the testing of the new valve is satisfactorily completed in the next quarter it will be incorporated into the field prototype.

Experimental

Retesting of DVMCS prototype

The testing conducted during the last quarter dealt with the problems identified in the previous report.

- The prototype was tested with the oil removed from the Belleville spring stack, which showed that viscous damping in the spring stack had a negligible effect on the overall damping, ruling this out as a cause of the lack of range. The tests did show, however, that the spring rate was slightly higher than assumed. The modeling was modified to reflect this fact.
- 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. These tests and analyses are summarized in **Appendix A**, below.

- Finally, the properties of both the Lord and 'home made' MR fluids were analyzed for their viscosity properties. The report is included as **Appendix B**. In short, the tests showed that the home made MR fluid was comparable in properties to the commercial one, and that it allowed us to adjust the magnetic properties of the fluid by varying the ration of iron to base oil.
- Based on these results, the laboratory prototype AVD external control circuitry was modified to include an demagnetization algorithm, and the earlier testing was repeated. The results were markedly improved and are presented below un the **Analysis** section.

Analysis

The above tests led to several obvious conclusions:

- The damping of the inert components is not a significant factor in the overall damping coefficient of the AVD.
- The MR fluid used does not change its properties in a manner that would explain the lack of dynamic range observed in the first tests.
- The most significant cause of the results observed earlier was the residual magnetization of the components of the damper.

As described above, a demagnetization algorithm was built into the laboratory control apparatus and our earlier dynamic testing was repeated. The demagnetization process was extremely effective as illustrated by **Figure 1**, below.

Figure 1: Effect of demagnetization on performance of MR damper

With the demagnetization circuit in place, a dynamic range of 7:1 was achieved at lower frequencies, and 10:1 at higher frequencies, as seen in **Figure 3**, below. This represents a significant improvement over the earlier results.¹

Figure 2: Performance of MR damper with demagnetization circuit operating

Based on these 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 decreases 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 below in **Figure 3**. The coils are the orange features in the sketch.

Figure 3 Schematic of redesigned MR damper section

Other

A paper on this project was submitted to the American Association of Drilling Engineers National Technology Conference. A copy is attached as **Appendix C**. The paper was presented after the end of the period (on April 5, 2005) and received an enthusiastic response.

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 APS Technology, Inc., “Downhole Vibration Monitoring & Control System Quarterly Progress Report #9,” **41664R09**, issued 28 January 2005.

Appendix A: Magnetic Testing

Daniel E. Burgess

February 25, 2005

OBJECTIVE

The magnetic testing was performed to investigate the properties of ferromagnetic materials in a magnet field as relates to the axial valve of the Down Hole Vibration Damper.

TESTING

The testing consisted of 3 sets of tests the first was to examine how materials behaved in a magnetic field. The second to test what happens if a strong magnet is placed into a variable shunt devise, and the third testing on the test mandrel used in our DVM valve.

The first test involved using a common coil to charge equal length samples with the identical amperages and measuring the resulting magnetic fields. Peak magnetic field strength was recorded with various current settings. Then the coil was turned off and residual field strength was recorded both in and out of the fixture (outside the fixture does not take into account the residual field in the fixture itself).

	1018	12L14	4140
Initial reading (kGauss)	0.006	0.108	0.1
Coil at 1 amp (kGauss)	1.7	1.73	1.6
Coil Off (kGauss)	0.35	0.35	0.45
Coil 2 amps (kGauss)	2.5	2.9	2.4
Coil Off (kGauss)	0.39	0.35	0.59
Residual measured outside (kGauss)	0.36	0.14	0.54
Percent of Highest field residual	14%	5%	23%
Comments		Low residual as measured outside leads me to believe that the short circuit bar was retaining a high enough field to alter off state measurements

Test Coil

The measurement device is pictured here. A steel frame shorts the poles with a fixed gap at one end maintained by a spacer with a slot that provides space to insert the probe. The coil is a wound bobbin designed and produced for the RSM solenoids. The current in each test is precisely the same and coil orientation is the same.

Gap Spacer

Short Circuit Frame

Sample

Probe

Sample

This is how residuals were measured outside the frame. A steel desk was used to short the poles with the probe between the desk and the sample, the highest attainable reading was recorded.

Steel Desk

Probe

The second test performed was to test if it was possible to use the coils to shunt or bias a piece of “magnetically soft” iron to short out the ends of a permanently biased magnetic bar that runs through it’s center. The goal here being that we could use 4140 or some other high strength steel, but most likely having a high residual field, through the center of the valve; and band around it with a magnetically soft material to shunt out that field.

Above is a sketch of the test setup used. A bar magnet, made of Alnico 5 being 2.5 inches long with an OD of .5 inches, was installed in two separate fixtures. The first fixture made of nylon and having the same OD and ID as the second. The flux was then measured using a loop of HyMU80 a material that has very high permeability and very low coercivity. This approximates the field reaching the OD of the fixture in a very low permeability gap such as if it were in Air. The gauss reading in the “air gap” was **1.365 kGauss**. The bar magnet was then placed in the second fixture which comprised a HyMu80 barbell shaped core wrapped with a coil having an OD of 1 inch an ID of .7 inches and a length of 2 inches. The flux was again measured using the same setup as before. (The loop was passed through a demagnetizing coil between each test.) The gauss level through the HyMu80 loop was **3.64 kGauss**. The coil was then energized with .5 amps (which should generate approximately 1.1 kGauss) so that the field generated would match the field inside the bar magnet in effect repelling the field out into the loop, which gave it a reading of **5.12 kGauss**. It should be noted that the field generated was higher than the sum of individual readings of the magnet and electromagnet. When energized in the opposing direction which should create a short in the bar magnet, the field measured in the HyMu80 loop was **0.00 kGauss**. And when power was cut the residual field was **1.34 kGauss** which is much less than the original reading, however if the bar magnet is removed and reinstalled the reading is the same as before.

The third round of testing consisted of testing the MR coil to try and determine what the residual fields were and how to remove them. The highest residuals were actually found to be in the extension tubes where they neck down close to the mandrel. The shorter of the tubes had a field of 260 Gauss the longer was 350 gauss. Unfortunately these relatively high fields are at the point of greatest impact on the MR fluid (at the entry to the valve). Based on the testing of materials and knowing how the 4140 behaves magnetically we recognized that this area would be very difficult to demagnetize. However we were able to change the geometry of the part to make the strong field have negligible effects on the MR fluid. In the test piece we opened up the bore from .078 clearances to .2575 clearances. This is 3 times more clearance and assuming that the field falls off with the square of the distance this should reduce the field strength to just 10% of it's original strength. In the new design we have further increased the clearance between the

mandrel and tubes to .502 inches. So the magnetic field effect should be only about 2.4% of the original field, or 8.5 gauss.

The second highest residual fields occurred within the valve itself and were approximately 150 gauss as measured in air (later tests showed this to be 1.4 kGauss in the MR fluid). This is still high enough to thicken the fluid to the point where excessively high damping will be created. This cannot be solved by material selection (as any material will have a residual field) or by geometry because for obvious reasons the valve geometry needs to develop a high field in a tight gap. Thus in order to move to a lower off state residual field you would have to perform some type of demagnetization technique to remove any residual field. With this in mind some tests on the damper section were performed using trace amounts of MR fluid to concentrate the field. In this test we reversed the field applied to the coils but held the current low so that the reversed field exactly matched the residual field. (The 4 amp field used during testing resulted in a residual of approximately 1.4 kGauss; this was equivalent to a 1.26 amp field.) We found that this created a near zero field at the point we had been measuring but other points had residual field of lower magnitude than the initial residual field but with reverse polarity, and still high enough to cause potential problems (approximately 200 gauss). To eliminate these fields we again reversed the polarity of the coil again and fed it an even lower current (.1 amps was found to work well). This canceled out the reverse polarity residual and left us with field that ranged from -20 Gauss to +60 Gauss; this is only 4% of the initial residual field and is not enough to create any appreciable damping. This is how we stumbled upon the stepwise demagnetization technique. We attempted to repeat the results using 120 VAC voltages but found that at 60 Hz the impedance is too great to put the required current through the coil, this meant it was not a viable solution, especially downhole.

ANALYSIS

In the first experiment we discovered that different materials will have different field strength when charged by the same coil with the same power. This is the effect of the permeability of the material but also the saturation density. The permeability is similar to resistance; it tells us how easily the field will flow through the material. The saturation density is simply the maximum flux that can be transmitted. In these tests it is very unlikely we got anywhere near the saturation point. A third property is the coercivity, which we had wrongly assumed was a representation of how much residual field would remain after the magnet had been turned off. However in the course of these experiments we found that the residual field even in low coercivity materials can be quite high. Observations during the second test showed that the HyMu80 used in testing was retaining a field, quite a high field of approximately .35 kGauss. Similar to the values attained in testing materials such as 1018 or 4140. However the field was easily reversed. Coercivity is a measure of how easy it is to create and or reverse a field in the material. Fields weaker than the coercivity point will have no magnetic effect on the material. Coercivity is not a measure of maximum residual field, rather it is the minimum field required to develop a response in the material, and therefore represents the lowest possible field that will remain after demagnetizing. The lower the coercivity the material has the easier it is to reverse the field and therefore to demagnetize also weaker fields still affect it so it can be demagnetized to a lower level.

Mild steels are not magnetically soft enough for the applications we are attempting here. 1018 has far too high a residual field. The 12L14 (which is a variant of low carbon steel to which lead and phosphorus are added to improve machining) may be a good candidate however finding published magnetic properties on this alloy has proven difficult since it is not considered to be among the “electrical irons” which are optimized for magnetic performance. The ideal material would have a high permeability, a high saturation density, a low coercive force, and low residual magnetism. Below is a chart published by Carpenter Steels.

Magnetic Property Selector

Click on a **category** below to see a listing of datasheets

	Sensitivity (permeability)	Strength (flux density)	Cost
Highest	Nickel-Iron	Iron-Cobalt	Iron-Cobalt
	Silicon Iron	Iron	Nickel-Iron
	Iron	Silicon Iron	Ferritic Stainless
	Iron-Cobalt	Ferritic Stainless	Silicon Iron
Lowest	Ferritic Stainless	Nickel-Iron	Iron

This shows that the best materials for us to use would be the irons and silicon irons. The silicon irons are a bit higher in strength and corrosion resistance and may therefore be the better choice.

The experiment with the magnet inside the coil proved interesting. It was possible to create a zero gauss field quite easily, and it was possible to create a very high field without much power. In addition it leads to the belief that we could counter a weak residual field with another weak field of opposite polarity. This conclusion is reached because the residual field after the power reversal is so much lower than the residual field before reverse power was applied. Before applying power the magnetic field was 3.64 kGauss and after reversing the field it dropped to 1.34 kGauss. That means the shunt is conducting approximately 2.3 kGauss preventing it from being exposed to the HyMu loop. This is most likely caused by the internal residual fields within the HyMu shunt continue to be polarized in such a way that it drained off some of the flux preventing it from getting to the test loop.

The results from the shunting test lead us to research various demagnetization techniques. A paper by a company called Annis who make demagnetization equipment lists the several methods available. We could heat it past the curie temperature of iron (the Curie temperature is the temperature at which a material loses its ferromagnetism) however the Curie temperature for iron is over 700°C so this option will not work on our valve. Another method is to exactly oppose the field with another field. This would prove difficult since it requires sensors located at strategic points within the mr valve section. It is however very close to what we successfully attempted during the third experiment. The final option is to expose the steel to a field of cyclically reversing polarity with a slowly diminishing intensity. Most commonly an AC field is used, however our findings were that at frequencies that we would get from the alternator the voltage would be insufficient to drive the required current. So we adapted a military technique called “stepwise deperming” that was developed during WWII (and is still used today) to demagnetize submarines. It uses a DC source that is switched in such a way as to reverse the polarity after each cycle and a voltage regulator to reduce the current gradually. We developed a

circuit to switch DC voltage back and forth while lowering the amplitude continuously. This method uses our available power and does not require sensors in the valve nor much additional circuitry.

Our stepwise deperming circuit has been shown to work well during preliminary testing. It was not ready to do a full size test, however we accomplished the same thing manually and were able to bring the sub down to near zero gauss. The system was driven as follows: -4 amps, +3.75, -3.5, +3.25, -3, +2.75, -2.5, +2.25, -2, +1.75, -1.5, +1.25, -1, +0.9, -0.8, +0.7, -0.6, +0.5, -0.4, +0.3, -0.2, +0.1, -0.05, +0.02, -0.01 amps. The entire series of steps took approximately 2 minutes to complete by hand, the circuit running at 10 Hz could do a similar series in 1.25 seconds. A series such as this could be performed at every pumps on / power up to start the tool fresh after every new stand was made up. If a sudden reduction is needed during routine drilling an abbreviated series such as -2, +1, -0.5, +0.1 could be done in a mere fraction of a second.

CONCLUSIONS

1. All materials retain a residual field, but the proportions of residual fields vary from one material to the next and are often not published.
2. Coercivity is not a measure of maximum residual field, rather it is a representation of the minimum field required to develop a response in the material, and therefore represents the lowest field that will remain after demaging.
3. A permanent magnet setup can be used to generate high magnetic fields with low power consumption; however power will be required to go to an off state.
4. A variable shunt device can be used to minimize the effects of residual fields on the fluid but cannot eliminate them without the use of power.
5. Regardless of what material is used, to attain an off state of near zero Gauss we will have to perform a demagnetizing technique
6. The best demagnetizing technique available to us down hole is to periodically reverse the fields applied and gradually lower the field intensity to demagnetize the materials (stepwise deperming).
7. Downhole ready circuits to demagnetize the sub have been successfully built and tested, and we have shown that the technique does indeed work.

Appendix B: MR Fluid Viscosity Testing

Joe Nord

February 25, 2005

OBJECTIVE

Determination of the optimum mixture of magnetorheological (MR) powder and oil to use in the Active Vibration Damper (AVD) tool. It is important that the MR mixture have a broad range of viscosities that increase as an increasing magnetic field is passed through it. It is desirable for the fluid to be as close to Newtonian as possible meaning that it's viscosity isn't affected by shearing. We will also compare the viscosity of the homemade MR fluid with a commercially available Lord brand MR fluid.

TESTING

Equipment

Figure 4 shows a model of the test fixture and **Figure 2** shows a photo of the apparatus. An electromagnet sends the magnetic field through steel mounting brackets and into the aluminum sample container. A rotational viscometer was used to determine the viscosity. The viscometer spindle was lowered into the sample container, which measures the torque required for the spindle to rotate at different speeds. A vane spindle with 4 blades (**Figure 3**) was used, which works better than the traditional disc spindle for fluids that have solids suspended in them.

Figure 1: Diagram of magnetic circuit for sample container

Figure 2: Photo of the viscometer test apparatus

Figure 3: 4-vane spindle

A gaussmeter was used to determine the field through the sample container. **Figure 4** below shows the results. At relatively low flux levels the fluid became too viscous to measure. The highest current we could send to the coil was 2 amps, which relates to 240 gauss. Even though the fluid is highly viscous at this flux level, the Lord literature says it is not yet magnetically saturated at this point.

Figure 4: Magnetic flux levels observed in viscometer test container

The rotational viscometer displays three outputs: viscosity, percentage of torque, and rotational speed (RPM). The viscosity is programmed to be accurate when using a 500 mL beaker and a spindle guard. The high cost of the MR fluid and the difficulty involved in putting a magnetic field through a container of this size led to the use of a much smaller sample container (approx 37 mL). Appendix A shows the formulas that were used to convert the torque and rotational speed output of the viscometer into apparent viscosity, shear rate, and shear stress.

Procedures

Initial testing on the AVD sub was performed using an 8:1 mass ratio MR fluid. *i.e.*, 8 parts MR powder mixed with 1 part Mobil 624. Viscosity testing was performed with the 8:1 mixture, 6:1, 4:1, 2:1, and a Lord brand silicone-based MR fluid. All of the homemade MR mixtures ran well in the viscometer. The Lord fluid was much more difficult. An important step in determining the apparent viscosity is a plot of the log of the angular velocity versus the log of the torque. For the Lord fluid, these plots were not straight lines. Most sloped downward and then upward while others never even sloped upward. The slope determines the flow behavior index, which is a measure of how it behaves under shear. A flow behavior index above 1 becomes more viscous under shear and an index less than 1 is less viscous under shear. The flow behavior index was so difficult to determine for the Lord fluid that it is unlikely that the calculated apparent viscosity, shear rate, and shear stress are correct.

Plots were created (see **Results**, below) of the flow behavior index (**Figure 5**), consistency index (**Figure 6**), and yield stress (**Figure 7**) as a function of the magnetic flux in the sample container. The Lord fluid is shown as a dashed line to show that it may not be accurate. The flow behavior index is a measure of how the viscosity reacts to shear. As the MR mixture ratio was increased, the flow behavior index increased and became closer to 1 (closer to Newtonian). All of the flow

behavior indexes were below 1 so the apparent viscosity reduced as the shear rate increased. The consistency index in **Figure 6** is an index of the viscosity at low shear rates. As the MR mixture ratio was increased the consistency index increased. For the magnetic flux levels that we were able to send into the sample cup the consistency index had a broader range as the mixture ratio increased. It is unclear if this would change under higher levels of magnetic flux. The yield stress in **Figure 7** is the shear stress required to initiate flow. If the shear stress of the viscometer spindle is lower than the yield stress of the fluid then the spindle will not rotate. The yield stress also has higher values and a broader range as the mixture ratio is increased.

In order to compare the homemade mixtures to the Lord brand MR fluid plots using the raw viscometer data were created. **Figure 8 - Figure 11** show the viscosity output versus the rotational speed for different magnetic flux levels. The absolute viscosities are based on using a 500 mL beaker and spindle guard, so they are not accurate. The curves do, however, show good comparative data. For all flux levels, the 6:1 MR mixture ratio plot was closest to the Lord fluid.

RESULTS

Figure 5: Comparison of Flow Behavior Index

Figure 6: Comparison of Consistency Index

Figure 7: Comparison of Yield Stress

Figure 8: Comparison of Viscometer Output at 0.4A (164 Gauss)

Figure 9: Comparison of Viscometer Output at 0.7A (195 Gauss)

Figure 10: Comparison of Viscometer Output at 1 A (218 Gauss)

Figure 11: Comparison of Viscometer Output at 2 A (240 Gauss)

CONCLUSIONS

The fluid with the highest ratio of MR powder had the best viscosity properties. The higher ratio fluids had a broader range of viscosities over the same range of magnetic flux, and they were closer to being Newtonian. However as the ratio is increased the viscosity increases. If a lower viscosity is needed to give the minimum damping in the tool then a lower mixture may be needed. The 6:1 and 4:1 homemade mixtures were the most similar to the Lord brand MR fluid. Lord is highly experienced with MR fluid so they have probably done additional optimization based on the cost of powder, rate of settling out, and other factors. Taking Lord's experience into account it is recommended that we use the 6:1 ratio fluid in the AVD tool. The 4:1 mixture also had viscosities close to the Lord brand fluid, but 6:1 is more suitable because of its wider range of viscosities.

APPENDIX A: VISCOSITY CALCULATIONS

angular velocity = ω
RPM = N

$$\omega := 2 \cdot \pi \cdot \frac{N}{60}$$

A plot of the Log of Torque v. Log of angular velocity determines the flow behavior index (η) and a constant A. The slope is η and the y intercept is the Log of A. This was proven with the calibration fluid. η was equal to 1 which means that it is Newtonian. The constant A provided the correct viscosity 5060 cP using additional equations listed below. For a Newtonian fluid the consistency factor (K) is equal to the viscosity (μ).

flow behavior index = η
consistency factor = K

$$K := \frac{A}{2 \cdot \pi \cdot Rb^2 \cdot L \cdot \left[\frac{2}{\eta \cdot \left[1 - \left(\frac{Rb}{Rc} \right)^{\frac{2}{\eta}} \right]} \right]^{\eta}}$$

Spindle radius = Rb
Effective Spindle Length = L
Cup radius = Rc

shear rate = SR

$$SR := 2 \cdot \frac{\omega}{\eta \cdot \left[1 - \left(\frac{Rb}{Rc} \right)^{\frac{2}{\eta}} \right]}$$

shear stress at the spindle = SS

$$SS := \frac{T}{2 \cdot \pi \cdot Rb^2 \cdot L}$$

apparent viscosity = μ_a

$$\mu_a := K \cdot SR^{\eta - 1}$$

Appendix C: MR Damper Retesting

Mark E. Wassell

February 25, 2005

SCOPE

Disassembly of after the previous tests showed that the MR valve components took a high permanent magnetic set. The permanent set resulted in thickening of the MR fluid even without coil current being applied. A method of de-magnetizing the valve components was developed to correct this problem. The original tests were then rerun.

CONCLUSIONS

1. The de-magnetization provides excellent dynamic stiffness range for the performance of the AVD tool.
2. The de-magnetization method significantly reduces the residual magnetism in the valve components. Hall Effect sensors were installed between coils to measure the field strength during testing. The demagnetized materials had residual field strengths of approximately 400 gauss. This reduction is sufficient to thicken the MR fluid. De-magnetization after applying full power brings the residual magnetism back down to acceptable limits.
3. The ratio of the dynamic stiffness is 7:1 at low frequencies and 10:1 at higher frequencies. This is a significant improvement over the initial testing of 1.5: 1 and 2:1. These values are for the 410 material. Better material choices will give even wider stiffness ratios.
4. The maximum dynamic stiffness either peaks or plateaus at ~ 3 amps.

DISCUSSION

Test Parameters

- MR Fluid – 6:1 mixture
- Gap – 0.031”
- Valve materials – 410SS
- Cam displacement – 0.708”

AVD Dynamic Stiffness 5000 lbs WOB

AVD De-Magnetization

Appendix D: AADE Paper

Laboratory Testing of an Active Drilling Vibration Monitoring & Control System

Martin E. Cobern & Mark E. Wassell, APS Technology, Inc.

This paper was prepared for presentation at the AADE 2005 National Technical Conference and Exhibition, held at the Wyndam Greenspoint in Houston, Texas, April 5-7, 2005. This conference was sponsored by the Houston Chapter of the American Association of Drilling Engineers. The information presented in this paper does not reflect any position, claim or endorsement made or implied by the American Association of Drilling Engineers, their officers or members. Questions concerning the content of this paper should be directed to the individuals listed as author/s of this work.

Abstract

The deep, hard rock drilling environment induces severe vibrations, which can cause reduced rates of penetration and premature failure of the equipment. 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. This system has two primary elements:

- An active vibration damper (AVD) to minimize harmful vibrations, whose hardness is continuously adjusted.
- A real-time system to monitor drillstring vibration and related parameters. This monitor adjusts the damper according to local conditions.

The AVD is designed to have several favorable effects on the time needed to drill a well. By keeping the bit in constant contact with the well bottom, and maintaining the actual weight applied the bit (WOB) at the optimum level, the instantaneous rate of penetration (ROP) is increased. Additionally, by reducing the levels of vibration throughout the bottomhole assembly (BHA), the operating life of all downhole components (bits, motors, MWD systems, etc.) is increased, thereby reducing the number of trips required for a particular well. These advantages will apply in all wells, but their value increases disproportionately in deep drilling.

An earlier paper¹ reported on the design and modeling of this system. After briefly reviewing these, we present preliminary laboratory tests that illustrate the ability of the AVD to adjust to a range of downhole conditions. Field test prototypes are being designed and built, and will be field tested in 2005.

Introduction

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 WOB. These changes may move the drilling parameters away from their optimum value and thus may 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 study² concluded: "Most of the 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 or torsional (stick-slip or bit whirl) vibration upon drilling have been documented in the laboratory³ and the field^{2,4}.

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 (**Figure 1**). There are many sources that excite drillstring vibrations, including bits, motors, stabilizers and drillstring imbalance. For example, a tricone bit imparts a primary 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. Shocks from bit bounce and collar impacts against the borehole result in higher frequency vibration.

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 2**, 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.

Principles of Operation

The AVD consists of electronics that monitor vibrations and other drilling parameters, and a spring-fluid damper that controls the vibration. The damper properties are **continuously modified** to provide optimal damping characteristics for the conditions present. A key innovation in the AVD is the use of magnetorheological fluid (MRF) as the means of varying the damping coefficient of the AVD*.

MRF is a "smart" fluid whose viscous properties are changed by passing a magnetic field through it. MRF 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.

MRF damping is currently being used in such diverse applications as sophisticated automotive suspensions⁵ and earthquake protection systems for buildings and bridges.⁶

The AVD modifies the properties of the BHA in two ways that combine to increase ROP and reduce vibration. First, the damper isolates the drillstring section below the damper from that 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 the desired surface contact force.

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 3**, 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 2**.

The practical effect of these changes are shown in the following figures, which are based on BHA models performed with APS's WellDrill™ software. When the damping force is optimized, which for this case is in the range of 200-300 lb.-sec./in., the bit remains in contact with the formation (**Figure 4**), the WOB remains constant (**Figure 5**), bit vibration is essentially eliminated (**Figure 6**), and the ROP increases (**Figure 7**).

Tool Design

An overview of the AVD tool is shown below in **Figure 9**. 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 difference is that the damping coefficient is continually adjustable by varying the magnetic field applied to the MRF. The details of the MRF damper design are shown in

Figure 10, [This drawing is of an earlier test piece, but the configuration is essentially unchanged in the prototype tools. The MRF 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 MRF 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.

The MR damper control algorithm utilizes displacement measurements taken in real time during the drilling operation. Based on this information, the damping properties are continuously modified throughout the drilling process. The intent is to both reduce the motion of the bit relative to the well bottom and smooth out the vibrations above the damper. A hardening damper algorithm was developed as the simplest and most robust method to control damping (**Figure 8**). This method increases damping levels as the damper sections displace relative to one another. For small displacements and low WOB, a low level of damping is provided. As the deflection increases, due to higher either WOB or larger vibration levels, the damping is increased. This method was shown analytically to provide proper damping levels over a wide range of conditions with the initial prototype design of the AVD tool uses a Belleville spring stack with a compound spring rate. This provides better isolation at various levels of vibration and WOB compared to shock subs having a linear spring rate. This provides increased deflection for the damping module at low levels of vibration and improved support at high levels.

* US Patent #6,257 356 B1; additional patent applied for.

Test Bench

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

In operation, the AVD will be supported and loaded by the entire drill string above it. Considerable weight is applied from above, and this loading would have both resilience and damping. The damping will 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 is 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, we designed the test bench shown in **Figure 11**. The prototype (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 well bottom, 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 configurations that mimic a variety of degrees of irregularity of the well bottom.

Test Results

The first test performed was a static test, to measure the damping coefficient of the AVD under various stationary conditions. In this test, a mockup of the damper element was mounted on the test stand, and the hydraulic pistons were used to drive a known volume of MRF through the damper with different voltage levels across the magnet circuit. By measuring the pressures and the time required for this flow, the damping coefficient could be derived from the ratio of the force applied to the fluid velocity.

As shown in **Figure 12**, 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 indicated that the damper would 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, was somewhat higher than desired, making the damper too

stiff for some situations. The gaps between the mandrel and housing were then adjusted to optimize the damper response.

In the next phase of testing, the full laboratory prototype, including the Belleville springs and bearings, was mounted in the test bench and driven by the cam. A sample of the results is shown in **Figure 13**, which plots the dynamic stiffness of the AVD as a function of the current applied and the drive frequency. The dynamic stiffness of the damper is a combination of the stiffness of the springs and the variable damping applied by the AVD. This combination is a function of the frequency of the driving vibration. As the applied current increases, the dynamic stiffness of the AVD rises.

The ability of the damper to reduce bit vibration and bit bounce is shown in **Figure 14**, which plots the maximum motion of the damper collar (connected to the bit) relative to the central mandrel, (connected to the upper drillstring). The driving displacement from the cam was 0.7. As the damping is increased, the maximum motion converges toward a level consistent with the bit's remaining in constant contact with the cam (which simulates the irregular bottom of the well). The damper is thus operating as it was designed to.

[Note: These figure combines data taken during different setups. The hydraulic damping controls for WOB and string damping were difficult to control and may not *exactly* repeat their settings from run to run. The data in the figures may, therefore, be slightly offset from one frequency to the next.]

Conclusions

Laboratory testing of the AVD indicates that it is capable of providing the variable damping necessary to control bit bounce, maintain uniform WOB and increase drilling ROP. There are several areas that are still under investigation and development, including:

- Increasing the dynamic range of the damper. The approximately 2:1 variation seen in **Figure 13**, while significant, is not repeated under all conditions, and is less than predicted by the modeling. Further refinements of the damper design are being studied and implemented.
- The hardening algorithm must be tested on the laboratory prototype to demonstrate that it performs as it is modeled.
- A field prototype has been designed and is being manufactured, with some parts of its design on hold pending the results of the testing described above.

Once these open issues have been resolved, it is anticipated that a prototype AVD tool will be tested, first in drilling laboratories and then in the field, in the second half of 2005.

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Nomenclature

AVD	=	<i>active vibration damper</i>
BHA	=	<i>bottomhole assembly</i>
LWD	=	<i>logging-while-drilling</i>
MRF	=	<i>magnetorheological fluid</i>
MTBFF	=	<i>mean time between field failures</i>
MWD	=	<i>measurements-while-drilling</i>
ROP	=	<i>drilling rate of penetration</i>
rpm	=	<i>revolutions per minute</i>
WOB	=	<i>weight on bit</i>

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FIGURES

Figure 1 - Drilling Vibration Sources

Figure 2: Frequency response of typical drillstring

Figure 3: Drillstring response with an AVD in use

Figure 4: Bit contact : 30,000 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: ROP: 30,000 WOB - 30,000 in-lb Spring Rate

Figure 8 - AVD Hardening Damper

Figure 9: Schematic of AVD Tool

Figure 10: (Next page) Detail view of the adjustable damping element design

Figure 11:(Second following page) AVD Test Bench

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Figure 13

Figure 12: Variation of Damping Coefficient vs. Pressure and Power in Initial Testing

Figure 13: 10,000 lbs. WOB - AVD dynamic stiffness

Figure 14: Maximum relative motion of the damper during testing