DOWNHOLE VIBRATION MONITORING & CONTROL SYSTEM QUARTERLY PROGRESS REPORT #9

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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.

The redesign and upgrade of the laboratory prototype was completed on schedule and it was assembled during the last period. Testing was begun during the first week of October. Initial results indicated that the dynamic range of the damping was less than predicted and that the maximum damping was also less than required. A number of possible explanations for these results were posited, and test equipment was acquired to evaluate the various hypotheses. Testing was just underway at the end of this period.

The project is approximately two months behind schedule at this time.

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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. 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.

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Design

Redesign of laboratory prototype

Complete.

Design of feedback system

The algorithms are installed in the system.

Field prototype design

The field prototype design is essentially complete, with a few details yet to be resolved. The electronics boards have been designed, laid out and ordered; most were received by the year end. The mechanical parts have not been approved for manufacture, however, pending further analysis of the test results. See below.

Experimental

Retesting of DVMCS prototype

The revised prototype was tested using the sweep algorithm. During testing, it was noted that the motor and gear train 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 our earlier static testing and the model predictions. Given these two problems, testing of the feedback algorithms was postponed until the causes are better understood.

A new gearbox/motor combination was specified and ordered, and is scheduled for delivery in January.

A viscometer was procured, and was modified to include magnetic coils, to study the properties of our 'home-made' MR fluid vs. the Lord commercial fluid. Testing was begun at the end of the period; initial results show no differences in the fluids.

A gaussmeter was acquired to verify the calculated field strengths in the gaps of the valve, and to test for any residual magnetization of components, which could effect the results. It was received at the end of the quarter.

We have not yet identified a laboratory to evaluate the magnetic properties (especially, permeability and coercive force) of the various alloys used in the prototype. It was decided that a permeability meter was beyond the budget of this program.

We also tested the damper prototype with the MR fluid drained from the damper element, to give an indication of the damping coefficient of the oil in the Belleville spring assembly. Results were obtained at the end of the period, but have not yet been analyzed.

Analysis

A preliminary analysis of the dynamic test data was completed in October. This analysis showed variations in the dynamic stiffness of the damper varied over ranges from 11% (at 5,000 lbs. WOB) to 80% (at 10,000 lbs. WOB). These are significantly below the ranges predicted by the modeling and earlier static tests. As a result of these findings, further testing and parts procurement for the Phase II prototype were put on hold pending their resolution.

Several explanations were posited for these findings:

- It is possible that the passive parts of the system – the oil-filled Belleville springs, the compensation system, and general friction – themselves provide significant dynamic stiffness, which reduces the relative effect of the AVD. To study this effect, we will drain the MR fluid from the valve and test the system response using only the passive components.
- The difference in the results could be attributed to differences in the magnetic permeability of the alloys used in the damper construction, which result in lower magnetic fields than predicted. To study these results, several parallel approaches will be used.
 - The magnetic fields in the gaps will be measured directly using a magnetometer to be purchased for this purpose.
 - Additional modeling will be performed and alternate coil designs studied.
 - The permeability of the alloys will be measured directly. To this end, we will either buy or rent a permeability meter, or send the materials to a laboratory for analysis.
- We have been using a 'home made' MR fluid for our testing, since the commercial fluid displayed unsatisfactory settling properties. It is possible that this fluid does not display the range of viscosities under the influence of the magnetic field that the commercial fluid does. To evaluate this possibility, we are purchasing a commercial viscometer, and modifying it to allow the measurement to be made while applying a magnetic field to the fluid.

Further analysis of the test data was conducted in November. This analysis showed that the field in the damper gaps is 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. We continue to pursue evaluation of these results.

The results of this analysis are in Appendix A, which also compares the results with the magnetic modeling of the damper.

The equipment needed to evaluate these potential sources of the differences was procured (see above) and testing begun.

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

Appendix A: Test Results & Analysis

Scope

The following are the test results and the comparison to performance analysis of the AVD damper. These tests include bench testing of the valve assembly and the full scale prototype tests. Prototype tests were performed with gap sizes of 0.042" and 0.030".

Conclusions

1. Based on the test data of the valve damper and the AVD prototype the APS MR fluid does not appear to be optimized. In its off-state condition the fluid viscosity is much higher than either Lord's water based and oil based fluids. In the charged-state the yield stress is much lower than the Lord fluids.
2. The high off-state viscosity combined with low yield stress of the APS MRF results in narrow performance band for the AVD tool. In the off-state the damper stiffness is higher than desired. When charged, the damping stiffness only increases slightly
3. Lord's water based MR fluid gives the best overall properties. It has a low viscosity and the highest yield strengths. The silicone based material has a slightly higher viscosity and has 70% of the water based yield strength.
4. Analysis shows that given the silicone based MR fluid properties the AVD tool should function with wide range of damping properties.

Recommendations

1. Vibration testing of the prototype tool should be performed using Lord's MRF-33AG silicone based fluid. This has a temperature capability of 150C and its properties are known.
2. Impact tests should also be performed. The setup of the cam driven vibration test will always perform better in the off-state. Impact tests could be setup to cover a wide range of loadings.

Discussion

There are two components that determine how the MR fluid performs; 1) the fluid viscosity in the fluid's off state and 2) the fluid yield stress as a function of the applied current. The fluids should have a low viscosity in the off state and become much harder in the charged state. Figure 1 shows the comparison of the various MR fluids. These include Lord's waterⁱ and oil basedⁱⁱ fluids and the APS fluid. It also compares the viscosity to basic water and light and heavy weight oil

Figure 1

This figure shows that the APS MR fluid is much more viscous compared to the Lord fluids.

Figure 2 shows the APS fluid yield stress as determined from the bench test of the damper. The results show that the fluid plateaus at a charge of 2.5 amps. Figure 3 shows the comparison of the APS fluid to the Lord MR fluids. The Lord water based MR fluid achieves the highest yield strength. The Lord silicone based fluid achieves 70% of the water based fluid's yield strength. Based on the test results, the APS fluid only develops 15% of the yield strength of Lord's water based fluid.

Figure 2

Figure 3

The AVD tool performance was analyzedⁱⁱⁱ using the MR Fluid properties presented in Figure 2 and Figure 3. Figure 4 gives the expected performance based on using Lord's MRF-33AG fluid. This figure shows a wide range of performance between the off-state and on-state conditions.

Figure 4

Figure 5 gives the results for the AVD using APS's MR fluid. Because of the fluid's higher viscosity and its lower yield strength, the tool has much lower difference between the off-state and on-state conditions.

Figure 5

A finite element model predicted the charge generated by the coils that charge the MR fluid. The amount of charge developed is a function of:

- Number of coil windings
- Available power (amps, voltage)
- Permeability of the valve material
- B-H properties of the MR fluid
- Valve gap

Figure 6 gives the valve field strength as a function of the valve gap.

A larger valve gap is desirable to minimize the off-state damping of the tool. This, however, requires more electrical power to fully charge the MR fluid. It also reduces the on-state damping. Therefore the valve gap must be optimized based on the available electrical power and the required damping range.

Figure 6

Valve Analysis with a 0.060" Gap

Figure 7

Figure 8

Figure 9

Figure 10

Figure 11

Figure 12

Valve Analysis with a 0.042" Gap

Figure 13

Figure 14

Figure 15

Figure 16

Figure 17

Valve Analysis with a 0.030" Gap

Figure 18

Figure 19

Figure 20

Figure 21

Figure 22

Figure 23

Valve Testing Results

0.030" Valve Gap

5,000 WOB – 1/16 Turn Damping – 0.5 Hz – 0 A

Figure 24

Figure 25

5,000 WOB – 1/16 Turn Damping – 0.5 Hz – 1 A

Figure 26

Figure 27

5,000 WOB – 1/16 Turn Damping – 0.5 Hz – 2 A

Figure 28

Figure 29

5,000 WOB – 1/16 Turn Damping – 0.5 Hz – 3 A

Figure 30

Figure 31

5,000 WOB – 1/16 Turn Damping – 0.5 Hz – 4 A

Figure 32

Figure 33

ⁱ MRF-241EF, Lord Corporation

ⁱⁱ MRF-33AG, Lord Corporation

ⁱⁱⁱ “Designing with MR Fluids” Engineering Note, Lord Corporation