# DOWNHOLE VIBRATION MONITORING & CONTROL SYSTEM
# QUARTERLY TECHNICAL REPORT #5

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## DISCLAIMER



## 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 entails modeling and design of the necessary subsystems and design, manufacture and test of a full laboratory prototype.

The project continues to advance, but is behind the revised (14-month) schedule. Tasks 1-3 (Modeling, Specification and Design) are all essentially complete. The test bench for the Test and Evaluation (Tasks 4 & 5) has been designed and constructed. The design of the full-scale laboratory prototype and associated test equipment is complete and the components are out for manufacture.

Barring any unforeseen difficulties, laboratory testing should be complete by the end of March, as currently scheduled. We anticipate the expenses through March to be approximately equal to those budgeted for Phase I.

# Table of Contents

# Executive Summary

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# Modeling (Task 1)

The initial modeling is essentially complete, as described in the Second Quarterly Technical Report[1]. Analysis of the predicted response of the particular mechanical design (**Task 3**, below) in relation to the testing (**Task 5**), which is part of **Task 4** has begun. An enhanced version of the ANSYS program has been procured to assist in this next level of modeling.

# Design

### *Specification (Task 2)*

**Complete**. See Third Quarterly Progress Report.[2]

### *Mechanical Design (Task 3)*

**Complete**. The design for both the laboratory test piece and the test equipment for use with the bench are complete, and parts are on order.

### *Electrical Design (Task 3)*

**Complete**. The electrical design was presented in the second quarterly report[1]. The circuit boards and software needed to control the laboratory prototype have been designed and are out for manufacture.

# Experimental (Task 5)

A prototype damper assembly is complete has been tested on the test bench. Initial results demonstrate that it can provide the required range of damping coefficients with acceptable power levels. The damper is being modified slightly to improve this performance and will be tested in January 2004.

Using the instrumented damper test piece, the damping coefficient was calculated over a range of operating values as follows.

### *Determination of damping coefficient*

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

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

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|---|---|
| $A_b$ = Cylinder bore area | = 4.91 $in^2$ |
| $A_r$ = Cylinder rod area | = 0.79 $in^2$ |
| $A_v$ = Valve flow area | = 0.49 $in^2$ |
| $A_p$ = MR valve piston area | = 6.48 $in^2$ |

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

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

Therefore, the flow rate, Q, is:

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

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

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

The damping coefficient, c, is then:

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

Varying loads were applied to the test damper, corresponding to different values of WOB, and the damping coefficients were determined as above. The results of these measurements are shown in **Figure 1** below.

### *MR fluid*

The MR fluid purchased from Lord Rheonics has been performing to specification. We noticed, however, that the magnetic particles have a tendency to settle out of suspension very rapidly, leaving a heavy sludge that is difficult to disperse when the fluid must be used. We have, therefore, developed a high-temperature MR fluid with somewhat better settling characteristics. This fluid is also less expensive than the Lord fluid. At present, we are doing our tests with both, to verify their suitability for this application.

### *Testing of DVMCS prototype*

As noted above, the design for both the laboratory test piece and the test bench are complete. The design and manufacture are taking somewhat longer than anticipated. All parts are expected to be in house in mid- February, and, barring any unforeseen delays, we should be able to complete the testing by the end of March as planned.

## Results and Discussion

Testing of the prototype MR damper was completed during this period, as described above. The performance as a function of applied WOB and current applied to the MR valve are plotted in **Figure 1**, below. For a given electrical power applied to the MR valve coils, the damping coefficient is plotted as a function of the effective WOB applied to the system.

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

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

Also, **Figure 1** shows that even with no power applied, the damping coefficient is in the range of 10-20,000 lbs/in/s, which may be higher than is desirable for optimum performance. We are, therefore, reconfiguring the damper and increasing the clearance between the poles of the magnets to shift its performance to a slightly lower range. The new version will be tested in January.

## Conclusions

The initial tests of the valve demonstrate that it can perform as required for the application. Some minor modifications to improve its performance will be conducted in January. The full DVMCS laboratory prototype is being manufactured and will be ready for testing in early March.

The project is progressing, but behind schedule, and a four-month extension of Phase I has been approved. Based on the current status, we should be able to complete the laboratory testing by the end of March, as scheduled, provided there are no unforeseen delays. Our expenditures through March will be very close to the total budget for Phase I, or perhaps slightly above. The anticipated costs for the entire project remain unchanged.

## 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.

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| 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] “Downhole Vibration Monitoring & Control System: Quarterly Technical Report #2, Report 41664R02, April 2003.

[2] “Downhole Vibration Monitoring & Control System: Quarterly Technical Report #3, Report 41664R03, July 2003

## Appendix: Figures

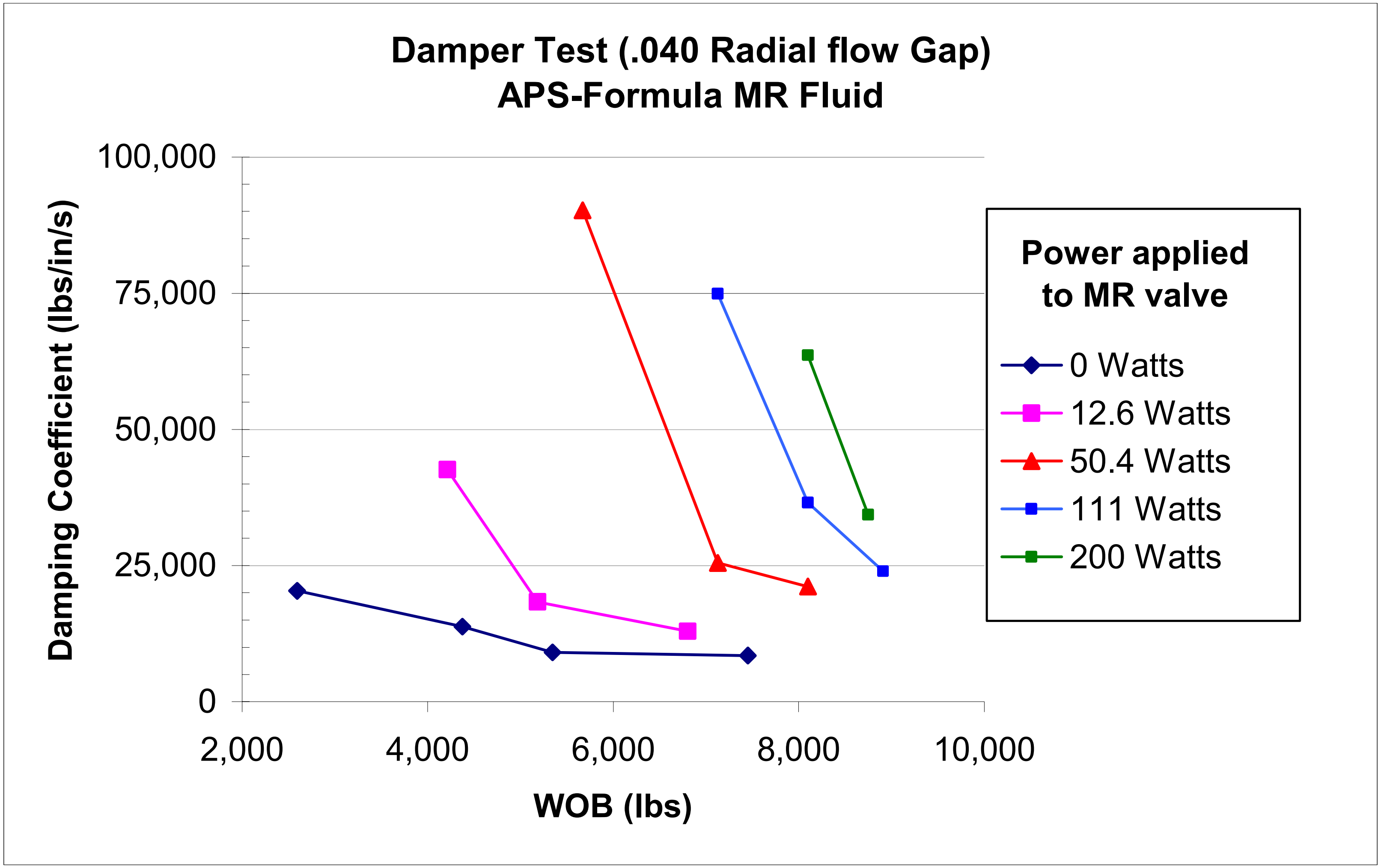


**Figure 1: Results of Initial Damper Testing**