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Final Report

NOVEL HIGH-SPEED DRILLING MOTOR FOR OIL EXPLORATION & PRODUCTION

Submitted by:

APS Technology, Inc.

7 Laser Lane

Wallingford, CT 06492

Prepared for:

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National Energy Technology Laboratory

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NOVEL HIGH-SPEED DRILLING MOTOR
FOR OIL EXPLORATION & PRODUCTION

FINAL TECHNICAL REPORT

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DISCLAIMER

ABSTRACT

The overall objective of this effort is to develop, build and test a high-speed drilling motor that can meet the performance guidelines of the announcement[1], namely: “The motors are expected to rotate at a minimum of 10,000 rpm, have an OD no larger than 7 inches and work downhole continuously for at least 100 hours. The motor must have common oilfield thread connections capable of making up to a drill bit and bottomhole assembly. The motor must be capable of transmitting drilling fluid through the motor.” To these goals, APS would add that the motor must be economically viable, in terms of both its manufacturing and maintenance costs, and be applicable to as broad a range of markets as possible.

APS has taken the approach of using a system using planetary gears to increase the speed of a conventional mud motor to 10,000 rpm. The mud flow is directed around the outside of the gear train, and a unique flow diversion system has been employed. A prototype of the motor was built and tested in APS’s high-pressure flow loop. The motor operated per the model up to ~4200 rpm. At that point a bearing seized and the performance was severely degraded. The motor is being rebuilt and will be retested outside of this program.

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Executive Summary

The overall objective of this effort is to develop, build and test a high-speed drilling motor that can meet the performance guidelines of the announcement[1], namely: "The motors are expected to rotate at a minimum of 10,000 rpm, have an OD no larger than 7 inches and work downhole continuously for at least 100 hours. The motor must have common oilfield thread connections capable of making up to a drill bit and bottomhole assembly. The motor must be capable of transmitting drilling fluid through the motor." To these goals, APS would add that the motor must be economically viable, in terms of both its manufacturing and maintenance costs, and be applicable to as broad a range of markets as possible.

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Regarding the specific tasks for Year 2:

1. The laboratory prototype has been built. (See **Section 2.1**, below.)
2. The test facility is complete. The flow loop was installed and then moved to APS's new facility. The dynamometer and related test equipment are complete. (See **Section 2.2**, below.)
3. Laboratory testing was performed, with the results as described above and in **Section 2.3**, below.
4. Testing in a drilling laboratory was deferred, pending development of an appropriate drill bit.

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1.0 Analysis & Design

The objective of this phase was to arrive at a viable design for the motor. The design completed during this phase has proven to have some difficulties, particularly with the ability to manufacture the gears. A partial redesign is underway, while key components and materials are being ordered.

1.1 Evaluation & Design of Power Section

The first task of this program was to consider the primary source of drilling power – the motor. Although is seems likely that APS's final choice will be a PDM, APS also evaluated the possibility of using a drilling turbine. This analysis included studies of the anticipated power and torque generated at various speeds, and the effect of the gearing on the output.

An extensive study was conducted of various motor and turbine options. These were evaluated on several bases. In particular, the gear ratio required to reach 10,000 rpm should not be excessive, the power per unit length and the torque delivered at 10,000 rpm should be as large as possible. Based on these criteria, APS selected three motor designs as the most likely candidates.

For completeness, APS also evaluated available drilling turbines. A small-diameter turbodrill did not deliver sufficient torque, and was extremely long compared to the motors. For reference, APS also include some larger (9 ½") turbodrills, which, although quite long, deliver significant torque with a relatively low gear ratio required. The results of the study are summarized below in **Table 1**:

Table 1: Comparison of Motors & Turbines for Suitability

Configuration	Model	Rotary Speed	Input Torque	Power	Length	HP/Length	Required Gear Ratio	Torque @ 10,000 RPM	Rank
		(rpm)	(ft-lb)	(hp)	(in)	(hp/in)		(ft-lb)	
4:5	DD675457.0	300	6060	284.1	210.0	1.35	33.3	181.8	**1**
2:3	675M2380	527	3172	261.2	210.0	1.24	19.0	167.2	**2**
4:5	675M4570	293	5200	238.1	202.5	1.18	34.1	152.4	**3**
Turbodrill	TO2-172	705	579	78.0	443.0	0.18	14.2	40.8	**0**
Turbodrill (9.5" OD)	TOR-240KE	660	5000	515.6	382.0	1.35	15.2	330.0	**NA**
3:4 (9.5" OD)	962M3460	265	8800	364.4	228.0	1.60	37.7	233.2	**NA**

Originally, the choice was for the 4:5 model DD675457.0. Later in the program, it was decided to reduce the gear stages from three to two and use a higher speed power section. The mud motor chosen was a 2:3 configuration, model 6-3/4" P300XL with a rotary speed of 450 rpm, input torque of 2,660 ft lb, 208 HP, and 27.8" length. See **Section 1.2**, below.

1.2 Gearing Design

In order to meet the design goals of the Announcement [1], while simultaneously producing a system that is commercially viable, flexibility is essential. The system must be compatible with a wide range of power sections, and serve the broadest market, including that for bits requiring 10,000 rpm or higher to operate efficiently.

To address this need, APS chose the path of mating a conventional mud motor power section to a compact, efficient gearbox, which will increase the motor's speed to the required 10,000 rpm. By choosing different motors and/or different gear ratios, the same system can address a greater variety of drilling requirements.

After a preliminary study of various gearing systems, it became clear that ***planetary gears*** offered the best solution for this application. They are compact and have relative high ratios, and these can be easily modified. (See **Figure 1** & **Figure 2**) Within this design range, two options were studied. The first was a conventional design, in which the mud flows through the center of the gear shaft. We refer to this as the 'flow through' design. The other options diverts the flow around the outside of the gears, and then back on axis at the bit box; this is the 'flow over' design. Each has its advantages and disadvantages, which are summarized in **Table 2,** below.

The performance of the 'flow through' design was felt to be inadequate. In order to achieve reasonable gear ratios, the central bore had to be made quite small, leading to unacceptable velocities. When the bore was opened to reduce the flow velocity, the total gear ratio fell significantly, as shown below in **Table 3**.

By comparison, the 'flow over' design (**Figure 3**) routes the drilling mud along the outside of the housing, where the greater radius generates greater cross-section for a given thickness. This allows more useful room for the gears, and results in higher ratios, as shown below in **Table 4**. Note that this ratio is achieved even with a lower-gain first stage, which is designed to reduce the torque loading on the gears.

Given the clearly superior performance of the 'flow over' design, it was chosen for use in the conceptual design, **completing this task**. Its choice, however, required the solution of several difficulties introduced by this design. In particular, the requirement for matching the rotational speed of the mud flow to the rapidly rotating bit box required the development of the 'vortex of doom', described below in **Task** below, and additional sealing problems, discussed in **Task 1.4**.

The optimum gear design was found to be a 2 stage gearbox instead of 3 stages. (See **Figure 4**.) A low pitch (11 teeth/in) gives the gears the strongest teeth possible. Each stage has a gear ratio of 4.66 for a total gearbox ratio of 21.8.

Table 2: Comparison of 'Flow Through' vs. 'Flow Over' Gear Designs

Advantages	Disadvantages
'Flow Through' Design	
Simple fluid flow path	Lower, less flexible, gearing ratio per stage → more stages
All bearings can be run in oil	Higher fluid velocities in tubulars
Drive components are simple and strong	Must draw power from motor to accelerate flow
	Requires a minimum of 3 rotating seals with large diameters
'Flow Over' Design	
Higher ratio per stage → fewer stages; greater versatility and choice of motors.	Complex fluid flow geometries
Smaller main seal diameter → lower surface speeds; only 2 rotating seals	Lower bearing must be mud lubricated or sealed on large diameter
Fluid is accelerated by pressure drop → no motor power lost	High fluid velocities at flow diverter; diverter is a weaker section

Table 3: Performance of 'flow through' gearbox with differing bore diameters

Flow Through Gearbox Design (108 Ft/s @ 600 GPM)					
	Ring PD	Planet PD	Sun PD	Ratio for this Stage	Total Ratio
Stage 1	5.875	1.625	2.625	3.24	3.24
Stage 2	5.875	1.625	2.625	3.24	10.49
Stage 3	5.875	1.625	2.625	3.24	**33.95**

Flow Through Gearbox Design (58 Ft/s @ 600 GPM)					
	Ring PD	Planet PD	Sun PD	Ratio for this Stage	Total Ratio
Stage 1	5.875	1.375	3.125	2.88	2.88
Stage 2	5.875	1.375	3.125	2.88	8.29
Stage 3	5.875	1.375	3.125	2.88	**23.89**

Table 4: Performance of 'flow over' gearbox

Flow Over Gearbox Design					
	Ring PD	Planet PD	Sun PD	Ratio for this Stage	Total Ratio
Stage 1	4	1	2	3	3
Stage 2	4	1.4375	1.125	4.56	13.67
Stage 3	4	1.4375	1.125	4.56	**62.26**

1.3 Overall Conceptual Design

The overall conceptual design encompasses the power section and gearbox described above, and must combine them into a survivable, manufacturable product. To do so, several problems must be addressed. The seals, coupling, casing and bearings are discussed below, but one of the most critical aspects of the design is the management of the flow around the gearbox.

In the absence of any mechanism to change the flow patterns, the drilling mud would be traveling in an essentially axial direction when it encountered, and passed through ports in, the bit box, which is rotating at 10,000 rpm. This mismatch of rotational velocities would require that the rotating part impart significant energy to the fluid flow, most likely enough to stall the motor. In addition, these high relative velocities would lead to turbulent flow and premature wear of the system.

The solution proposed involves two elements. First, a helical fin is machined into the outer diameter of the gearbox. This helix changes its pitch from an initial axial alignment by ~1º/in., reaching an angle of ~45º at the exit, so that the flow has been smoothly turned until it has approximately equal axial and rotational components.

To further increase the angular velocity, the flow is now directed into a conical section (the 'vortex of doom') which gradually decreases the radius of the flow. By conservation of angular momentum, the rotation velocity increases inversely with the radius of flow, until it nears (but may not reach) 10,000 rpm at the entrance to the bit box assembly.

Finally, the bit box has several 'scoops' machined into its upper surface, which collect the flowing mud and give it its final acceleration to the matching angular velocity. This last interaction is the only one that will take any energy from the drilling motor. The first two steps take their energy from the flowing fluid itself, in the from of a pressure drop.

The overall design of the gearing system is shown below in **Figure 3**. An analysis of the flow patterns, pressure drops, velocities, *etc*. is given in **Appendix A: Vortex Pressure Drop & Velocity Study.** The conclusion is that the overall vortex system generates a pressure drop of 35 psi and an maximum velocity of 90 ft/sec at the exit. These are considered acceptable numbers, but the design and modeling will be refined as the project progresses.

1.4 Sealing System Design

Seal designs have been evaluated for both the high speed (~10,000 rpm) and low speed (~500 rpm) sections of the high-speed motor. The downhole, high-speed seal under consideration is a modified Type 8 face seal that will be a joint design between APS and John Crane, The seal will be pressure compensated with silicon carbide (SiC) *vs.* tungsten carbide (WC) seal elements with an acceptable surface speed of less that 4000 ft/min.

The uphole low-speed seal is a carbon graphite (CGr) filled polytetrafluoroethene (PFTE, or Teflon) ring-loaded seal being designed by APS. This seal will be pressure compensated and will operate with and acceptable surface speed of less than 100 ft./min.

1.5 Flexible Coupling & Casing

After discussions with BICO Drilling Tools, a leading motor company, APS agreed upon a design of a flexible coupling to serve as a constant velocity joint between the motor and gearbox. It is illustrated below in **Figure 4**. It may be necessary to add a marine-type bearing to the top of the gearbox to simultaneously constrain the driveshaft to an axial position (to reduce vibration) while allowing the mud to flow past it.

1.6 Bearings

Bearing designs have been evaluated for all four sections of the high speed motor.

- The low speed input shaft bearing under consideration will support the mud motor flex coupling. It will be an elastomer-lined design, lubricated with drilling mud, with a 400 lb. radial load rating and operate to speeds of 500 rpm.
- The primary high speed bearing set will consist of angular contact ball bearings capable of supporting a 17,500 lb. dynamic axial load with a 14,000 rpm limit. The bearing set will be spring suspended and lubricated with synthetic oil.
- High-speed bearings lubricated with drilling mud will be of the ceramic staggered disk design, to support up to 2,000 lbs. radial and axial loads, capable of operating up to 10,000 rpm.

1.7 Critical Frequencies & System Damping

It was not possible to conduct this analysis in the absence of any laboratory test data. If the project goes forward (see **Section 3.0**, below) this analysis will be performed.

1.8 Test Equipment Design

The primary test equipment used in this project is a flow loop. The flow loop consists of a large structural steel baseplate to support the entire high-speed motor assembly. Flow is supplied by an 850 hp diesel-driven National Oilwell triplex pump capable of flows to 750 gpm and 1500 psi. The flow loop was installed at APS's facility in Cromwell, as shown in **Figure 5** to **Figure 9**.

The high speed motor is connected to a 500 hp Kahn dynamometer. Flow exiting the high speed motor is slung into a collector where it is pumped back into the main supply tank. The HSM will be instrumented to measure temperatures, delta pressures, flow, rpm, torque and vibrations. All measurements will be collected on a computer using LabView data acquisition software.

2.0 Laboratory Testing

2.1 Laboratory System

The laboratory prototype was built per the design shown in **Figure 1** to **Figure 4.** There were several problems in the manufacture, which required slight redesigns of bearings, seals, *etc*., but these were resolved and the parts manufactured.

The assembly of the motor for testing is shown in **Figure 10** to **Figure 16**

2.2 Test Facility

The flow loop was installed at APS's facility in Cromwell, as shown in **Figure 5** to **Figure 9**. In June, 2007 it was moved to APS's new facility in Wallingford. In the new arrangement, the control panel and the test areas of the flow loop were moved indoors, making the loop much more useful for testing. A dynamometer was constructed for testing the output of the HSM.

The motor was installed in a special pipe section with a collector for the exiting mud flow (see **Figure 17**.) It was connected to the dynamometer and other sensors as described below for the laboratory testing.

2.3 Laboratory Testing

The HSM was installed in the flow loop, and instrumented with flowmeters, pressure gauges, thermocouples and the dynamometer. The intention was to slowly bring the pressure up at approximately constant flow rate, and observe the speed, power and torque generated by the HSM. The test matrix is shown in **Table 5.**

As the pump pressure was increased, the motor speed increased linearly to ~4200 rpm. (See blue curve in **Figure 18.)** At that point, there was a loud noise and the output speed and motor torque decreased precipitously. The test continued, and generated the data shown in the red curve in **Figure 18.**

Table 5: Test Data from High Speed Motor Run

OUTPUT SPEED	UPSTEAM PRESSURE	MUD MOTOR DIFF PRESS	MUD MOTOR TORQUE*	FLOW*	MUD MOTOR HP	DOWNSTREAM PRESSURE
1980	100	65	163	91	61	35
4265	200	155	388	196	315	45
SPEED CHANGED DRASTICALLY AT THIS POINT. PROBABLY WHEN PLANET GEAR BEARING SIEZED.						
680	100	70	175	31	23	30
983	150	125	313	45	58	25
1222	200	170	425	56	99	30
700	100	74	185	32	25	26
1200	200	170	425	55	97	30
2076	500	441	1103	95	436	59
2652	800	721	1803	122	910	79
2200	600	540	1350	101	565	60

Upon disassembly, it was discovered that the thrust bearing between the sun gears had been overloaded and failed. The metal debris and ball bearings were found in the gearbox. One of the planet gears had jammed as a result of the thrust bearing debris' getting in between the smooth bearing and the planet. (See **Figure 19** and **Figure 20**) This placed extreme loads on the motor. The HSM was cleaned and serviced, and the bearings were replaced in anticipation of retesting. The thrust bearing was replaced with a solid thrust bearing to prevent the same failure. Testing could not be completed within the time frame of this project but will resume in Q2-08. Despite the unfortunate failure of a subsidiary component, the motor was functioning as designed, and seems likely to have reached the targeted 10,000 rpm.

2.4 Drilling at Test Facilities or Test Wells

APS were unable to complete this phase of the project for several reasons. The design and manufacture of the HSM took considerably longer than was anticipated, so there was no time for field testing. Also, APS were not able to locate a suitable drill bit that could rotate at 10,000 rpm.

3.0 Future Plans

APS plans to continue the testing of the HSM to verify its ability to reach the design speed. Parallel activities under this overall program are working toward the realization of drill bits to operate at comparable speeds[2]. When that occurs, the HSM will be in position to drive them.

At the same time, APS is investigating other applications of the geared motor design used in the HSM. It may be possible to use it with a lower gain to drive more conventional bits, or other downhole equipment.

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] U.S. Department of Energy, National Energy Technology Laboratory, Funding Opportunity Master Announcement No. DE-PS26-04NT15450-0, "Oil Exploration and Production Program Solicitation," issued November 4, 2003

[2] Alan Black & Arnis Judzis, "Smaller Footprint Drilling System for Deep And Hard Rock Environments; Feasibility of Ultra-High Speed Diamond Drilling," U.S. DOE Contract #FC26-03NT15401, Technical Report, October 1, 2004.

Figures

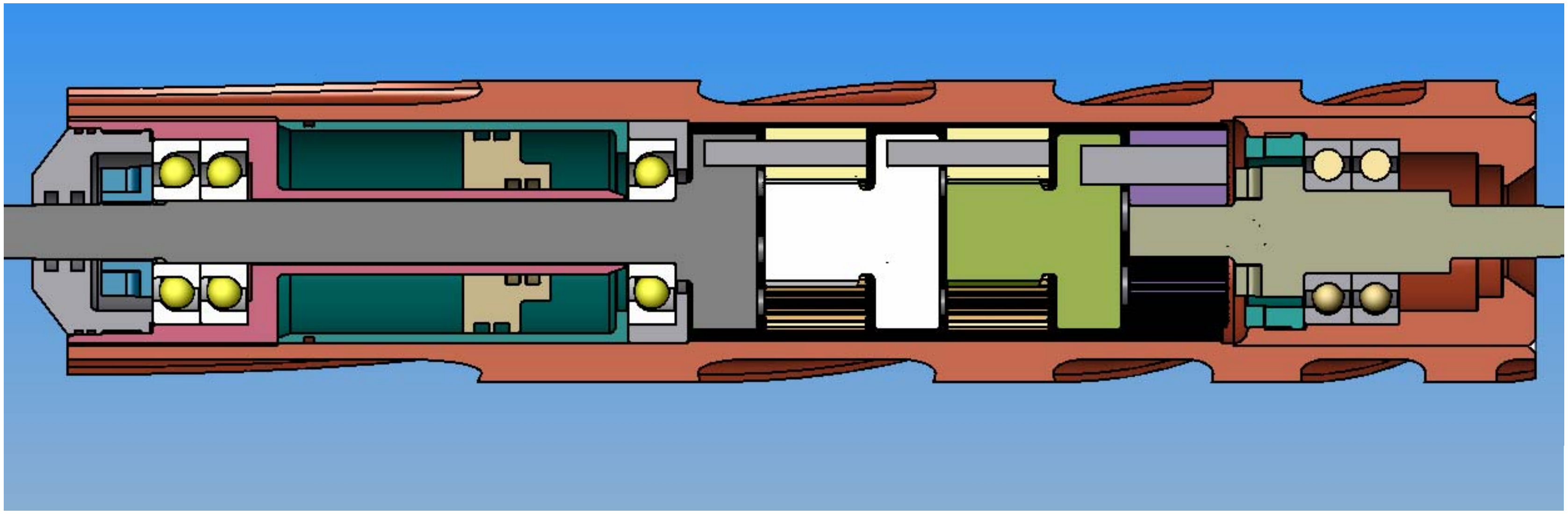


Figure 1: Cross-section through the bearings and gears

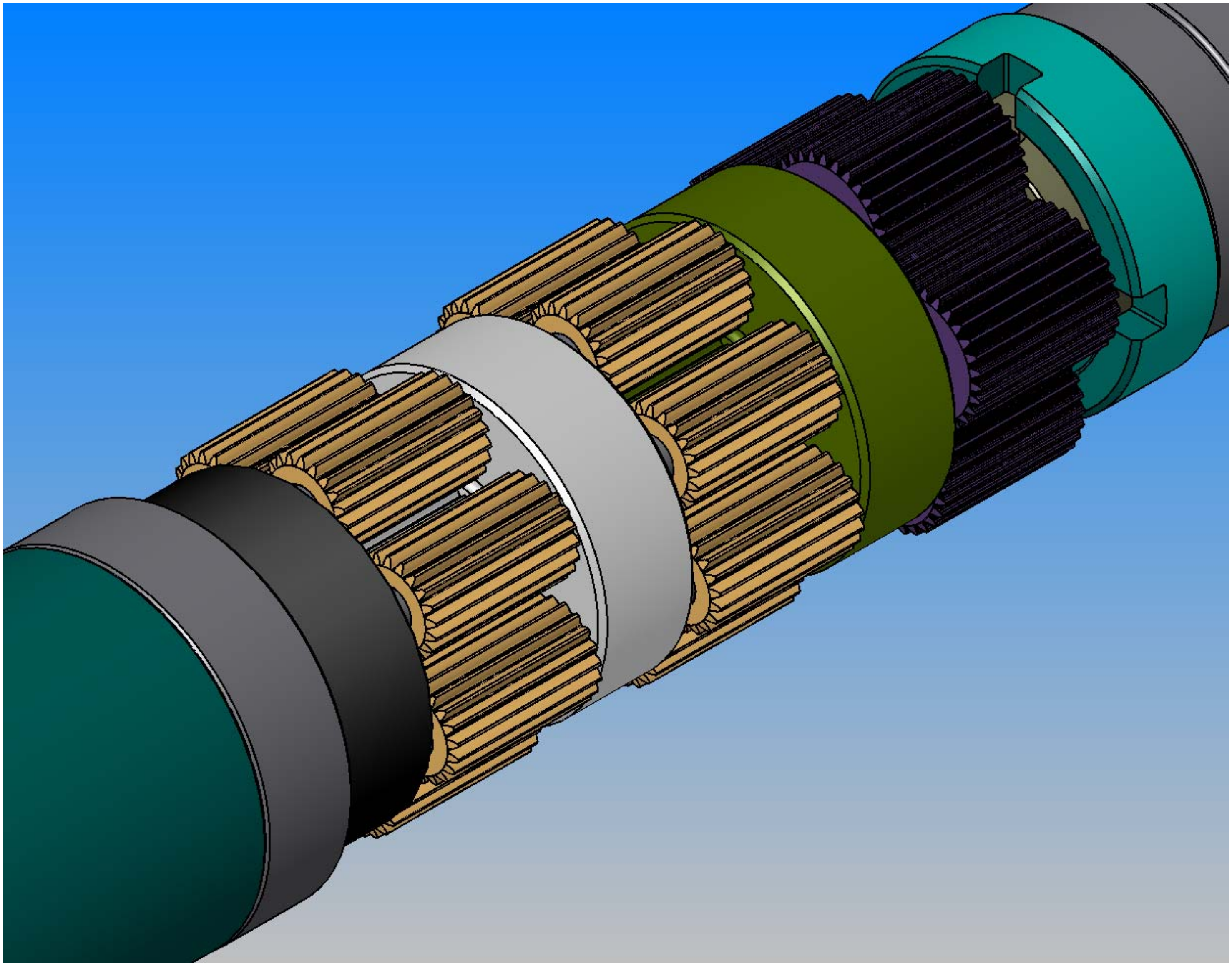


Figure 2: Detail of the planetary gears (earlier 3-stage design)

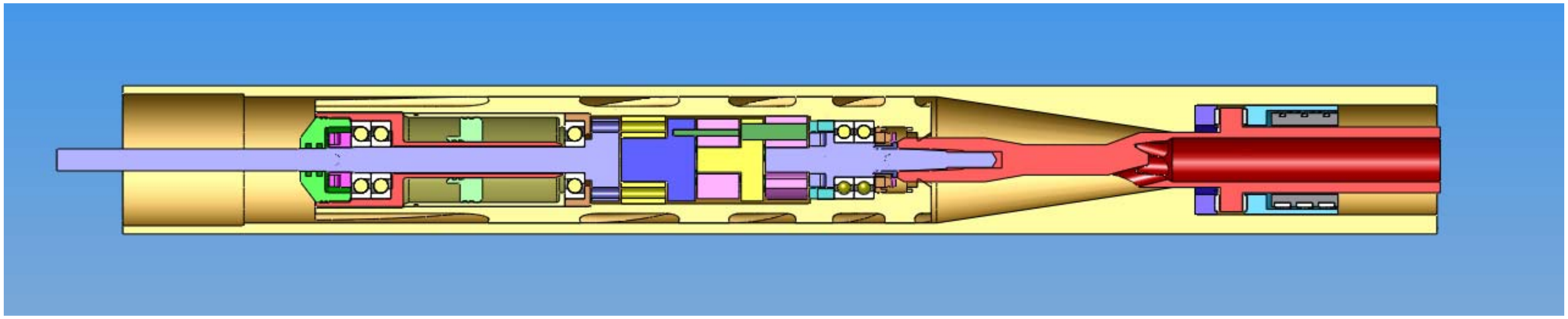


Figure 3: 'Flow Over' Gearbox Design

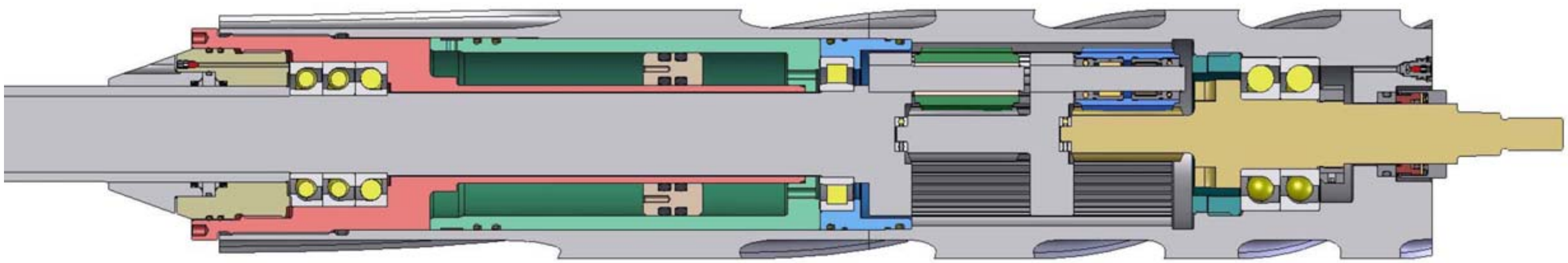


Figure 4: Detail of revised (two-stage) gearbox

Figure 5: Delivery of pump

Figure 6: Delivery of motor & frame

Figure 7: Placement of motor

Figure 8: Placement of pump

Figure 9: Storage tank (recycled from soda bottler)

Figure 10: Parts of High Speed Motor awaiting assembly

Figure 11: Planetary gears ready for assembly

Figure 12: Planetary gears during assembly

Figure 13: Variable pitch helical fin element

Figure 14: 'Vortex of Doom' to match angular velocities of mud and rotor

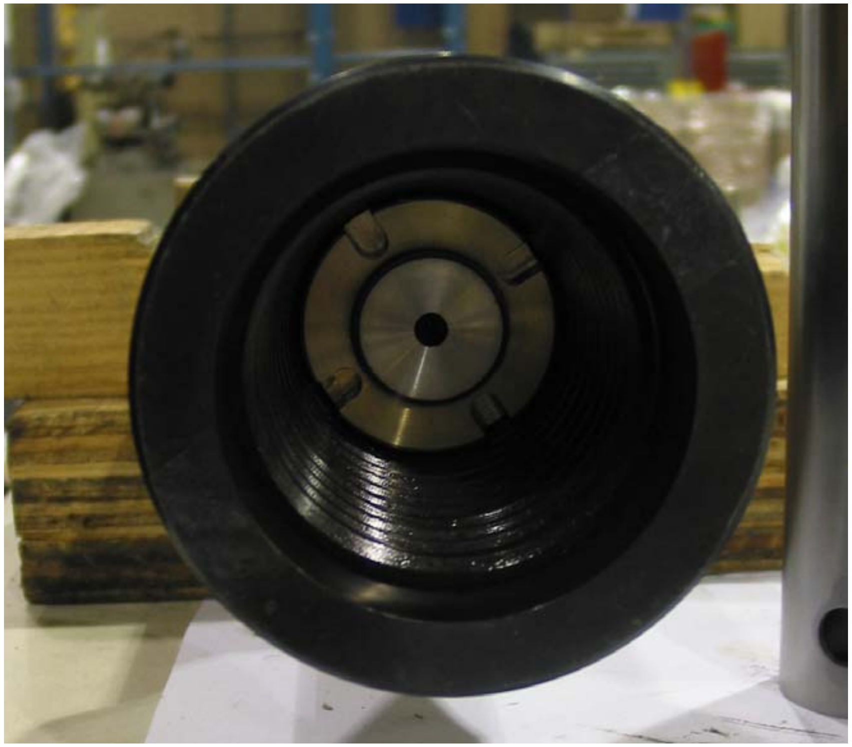


Figure 15: End view of assembled motor

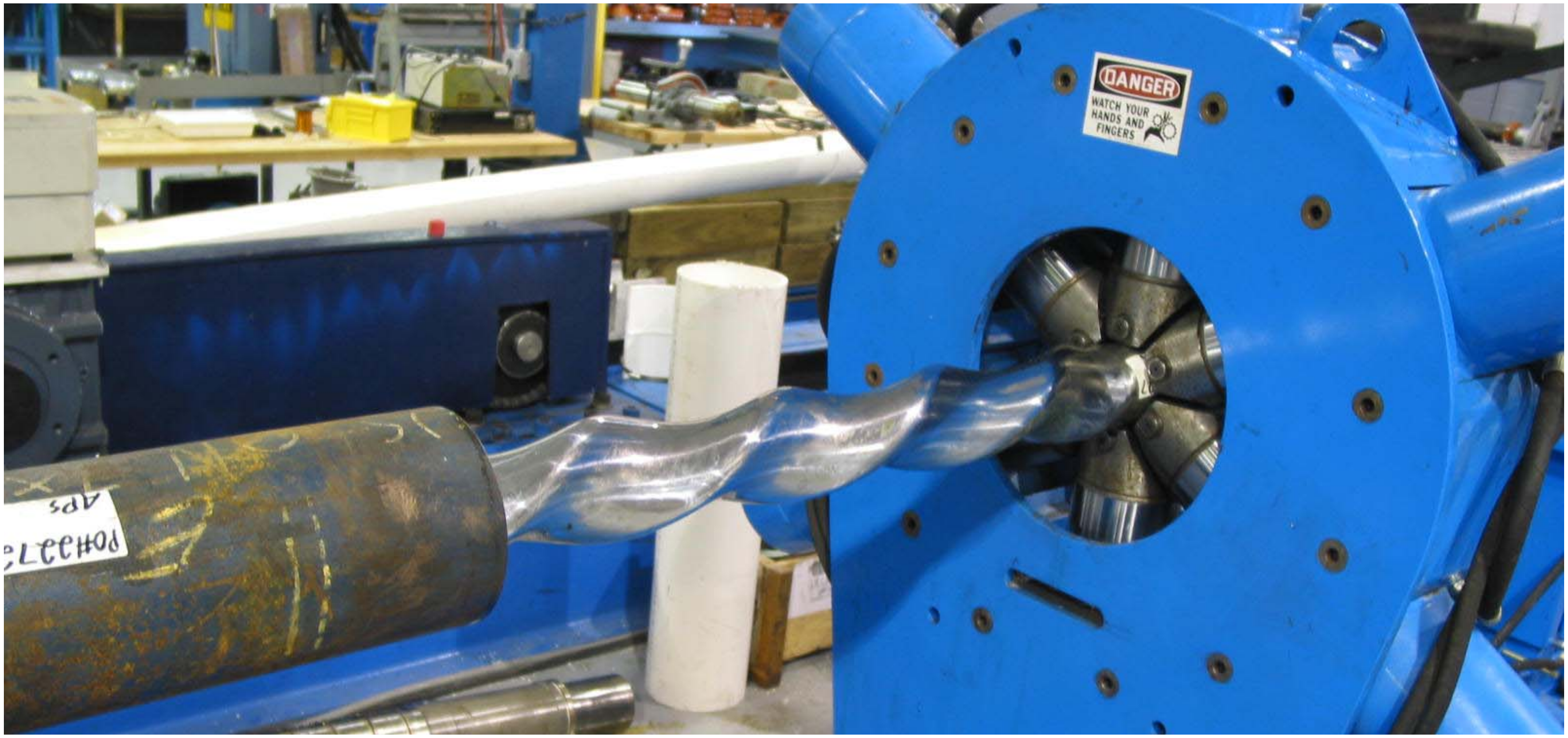


Figure 16: Installing the rotor in the power section

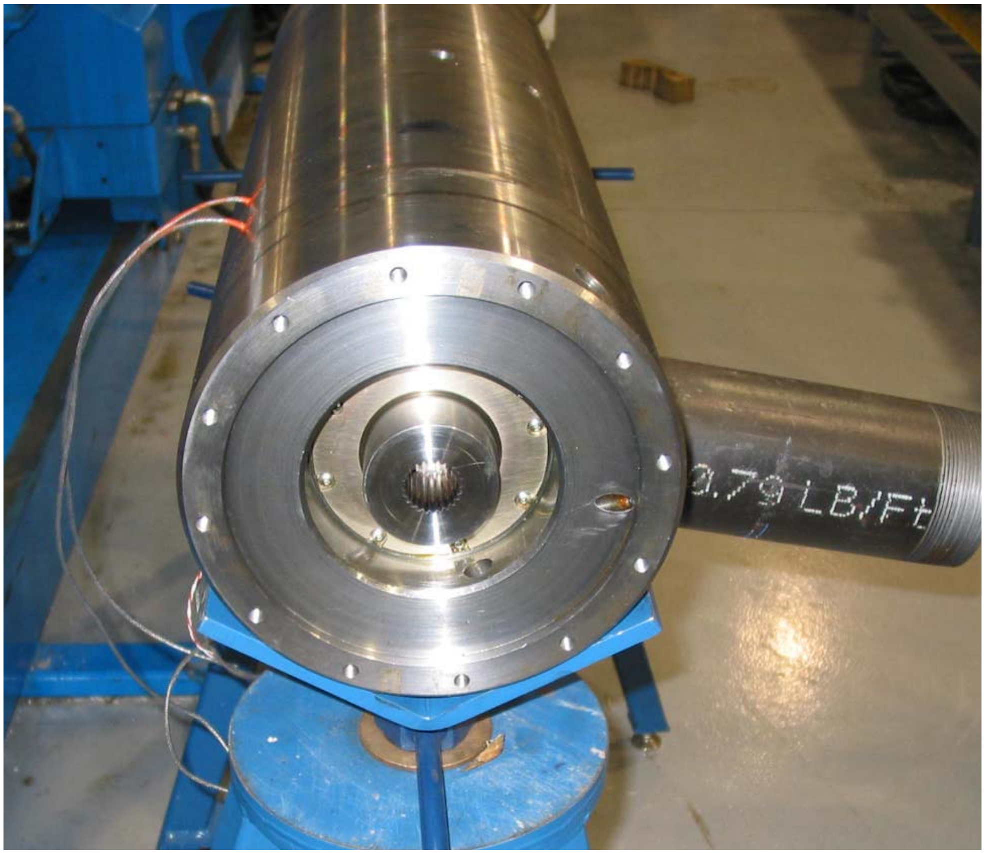


Figure 17: Assembled motor in test section showing output mud collector

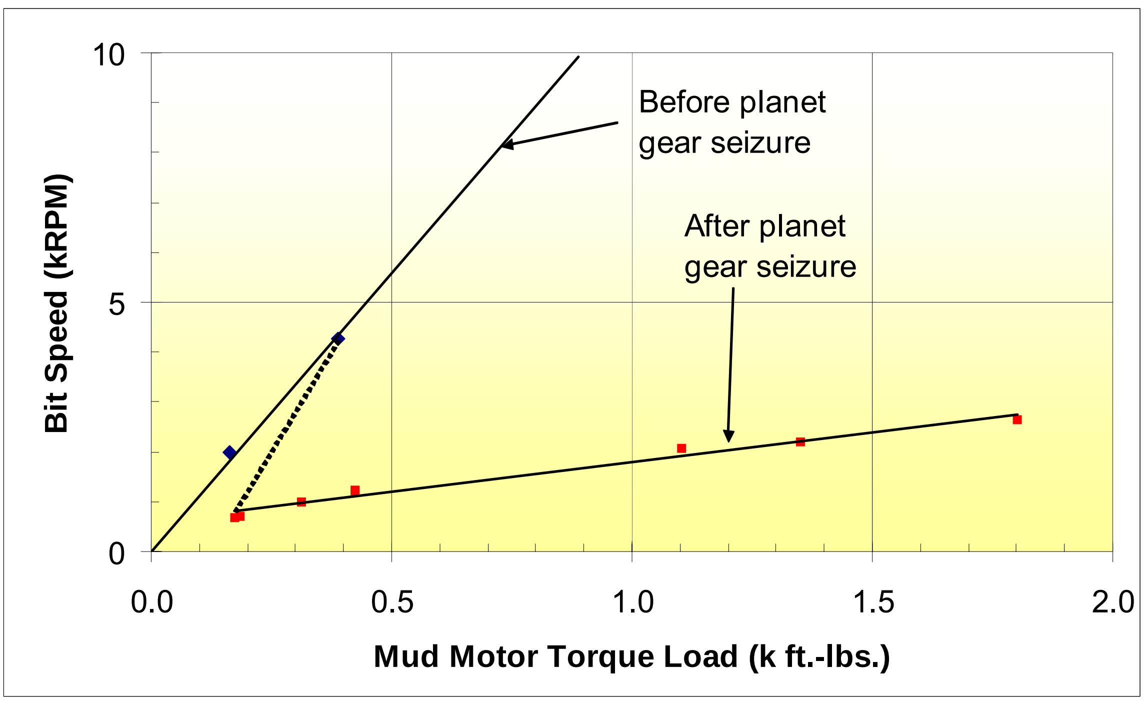


Figure 18: HSM speed vs. mud motor torque

Figure 19: Planet gear showing inner surface wear

Figure 20: Section of planet gear showing wear

Appendix A: Vortex Pressure Drop & Velocity Study

M Wassell

February 7, 2005

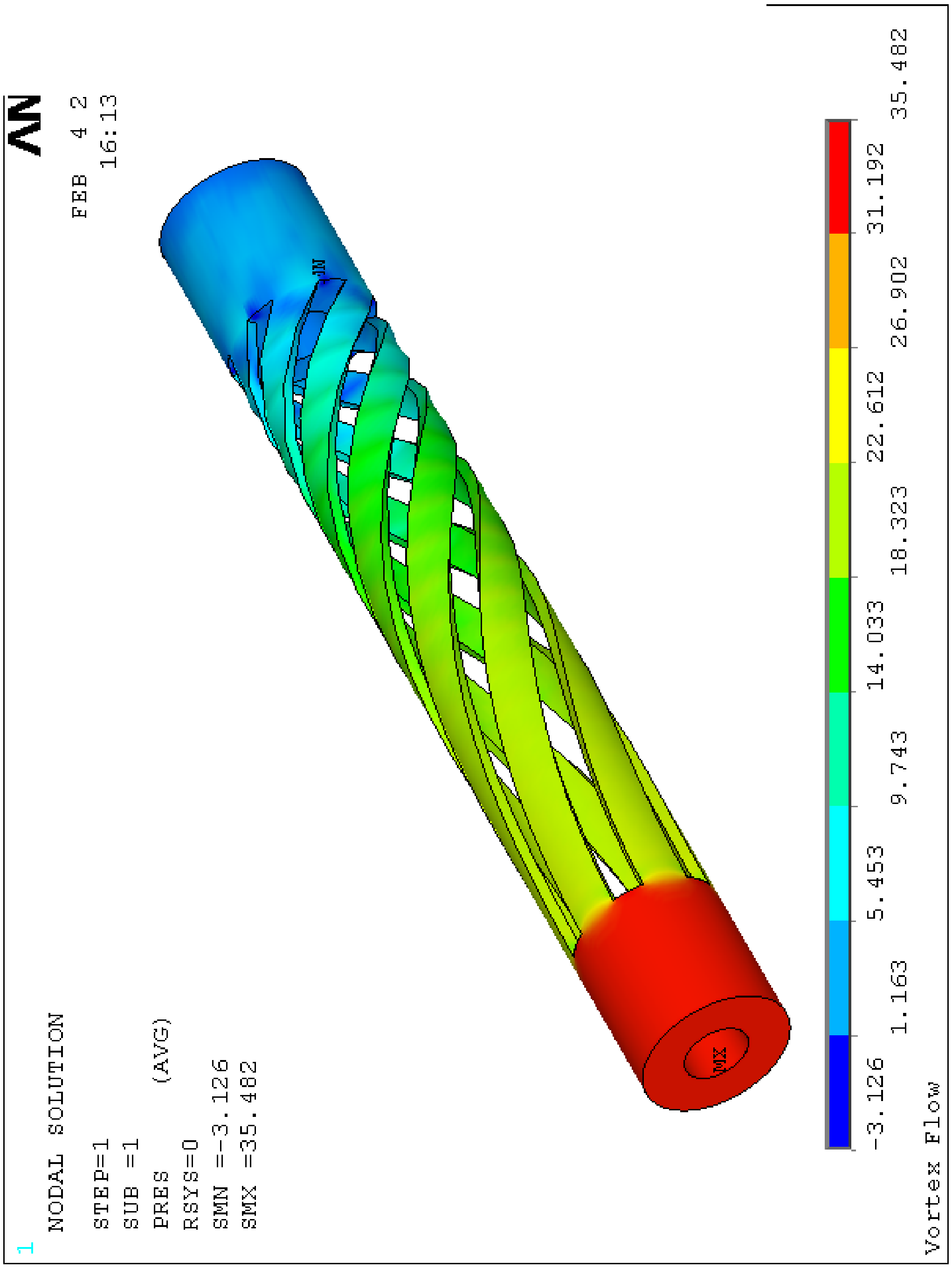
Scope

This analysis is an analysis of the pressure drop and the velocities generated through the outer vortex. The flow is based on water based mud at a flow rate of 400 GPM. The mud velocity at the inlet is 6 ft/sec (400 GPM, 5.75”OD x 2.75”ID).

Conclusions

1. The pressure drop across the vortex is 35 psi.
2. The maximum velocity of 742 in/sec (90 ft/sec) occurs at the exit.
3. Most of the circumferential flow occurs at the OD. There is little circumferential flow at the ID. This is opposite of what is desired since the shaft is rotating and the outer casing is not.
4. There will be some large pressure spikes across the bit as the inlet ports on the flow diverter pass through the wakes coming out of the vortex. This will develop a frequency of 1000 Hz at 10,000 rpm and 6 vortex slots.

Results



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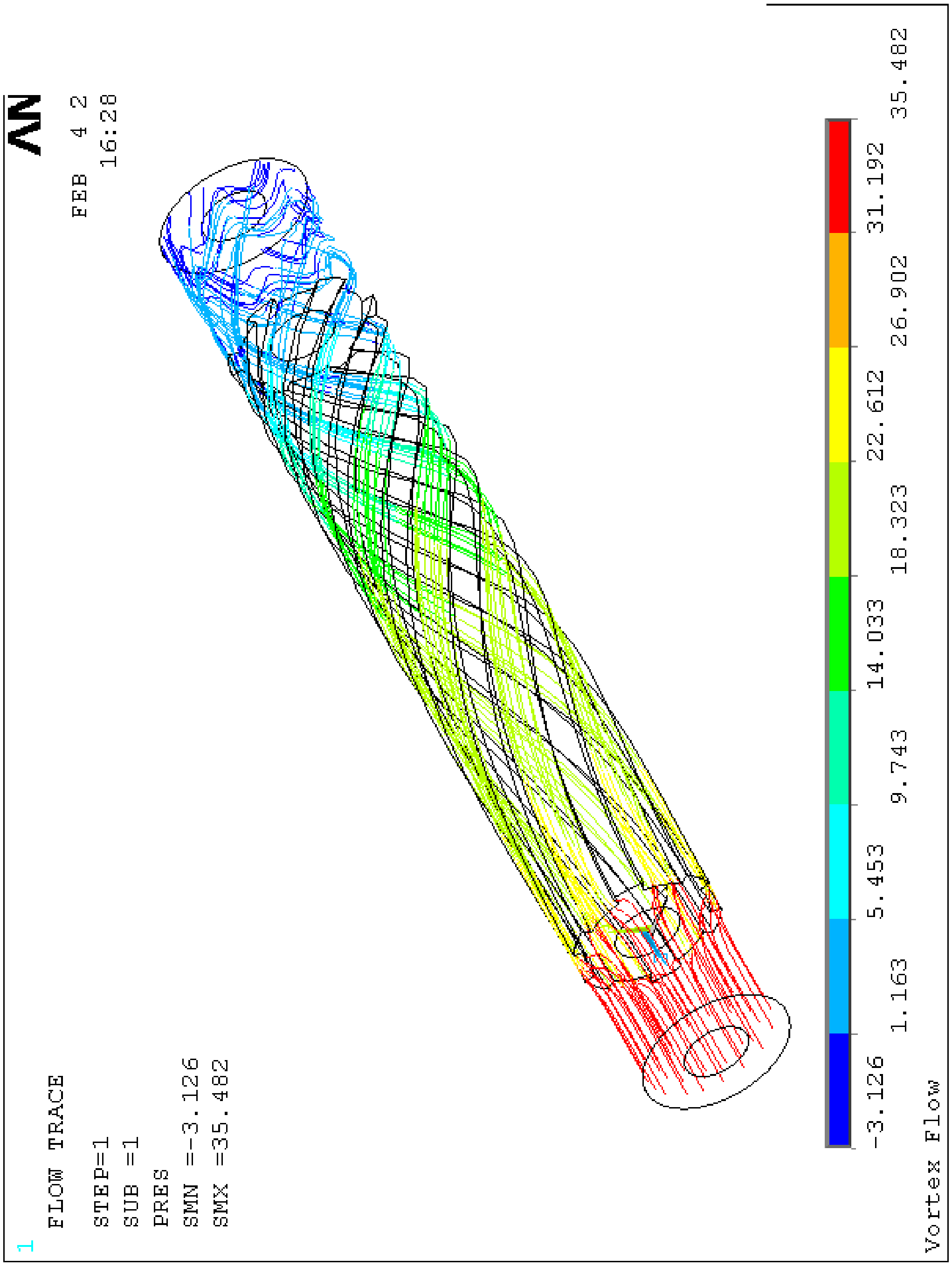
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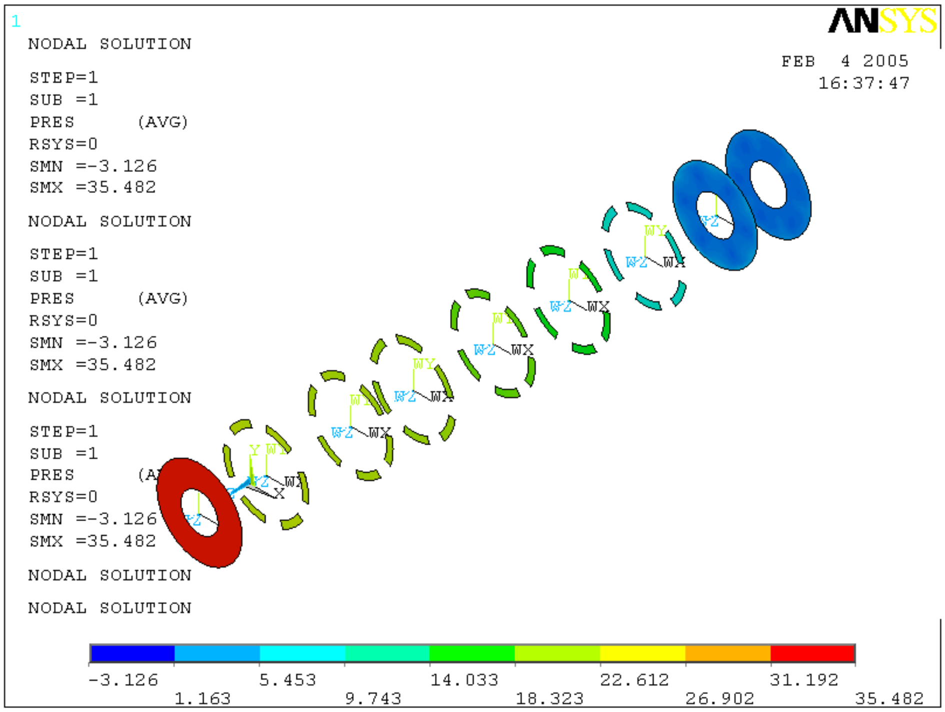
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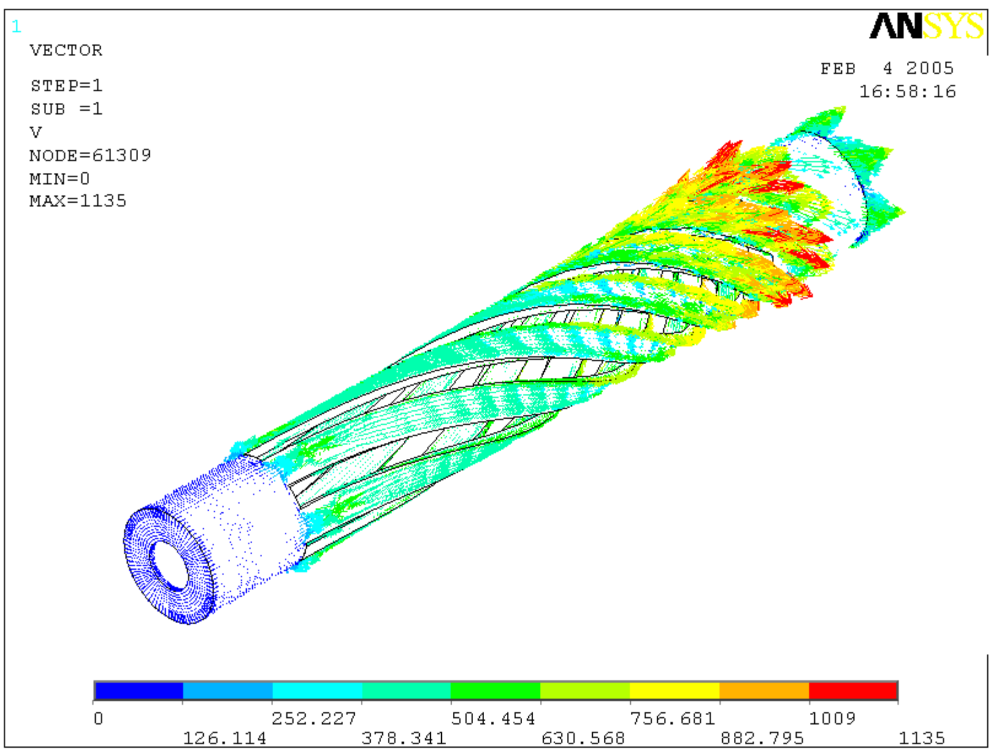
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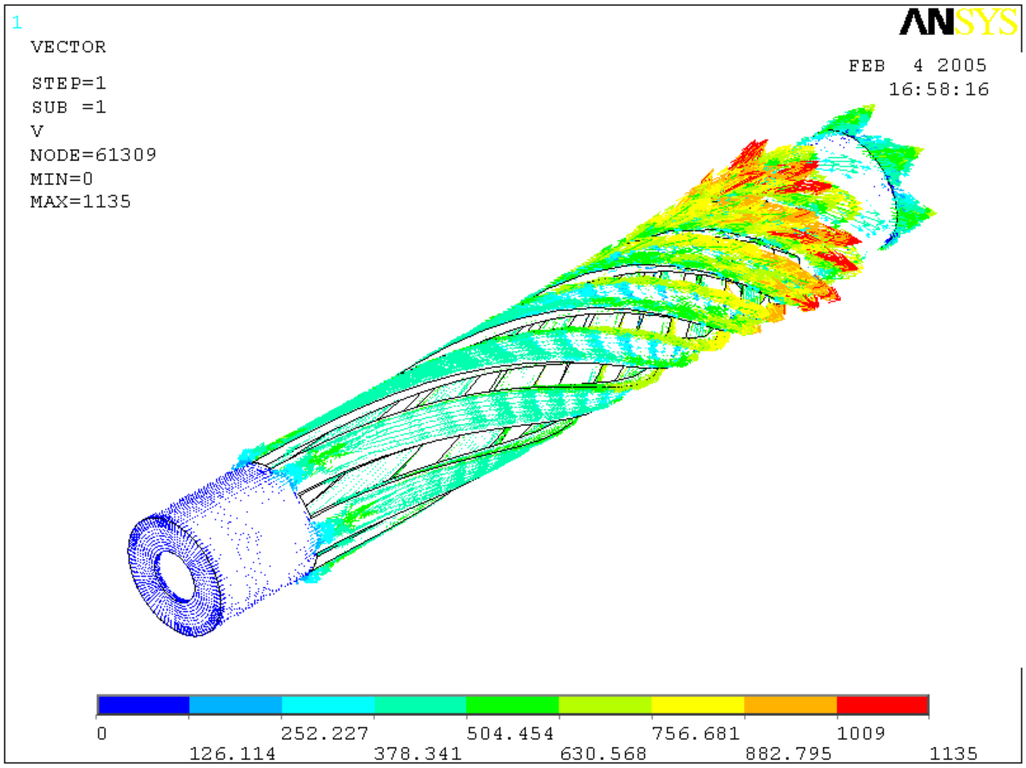
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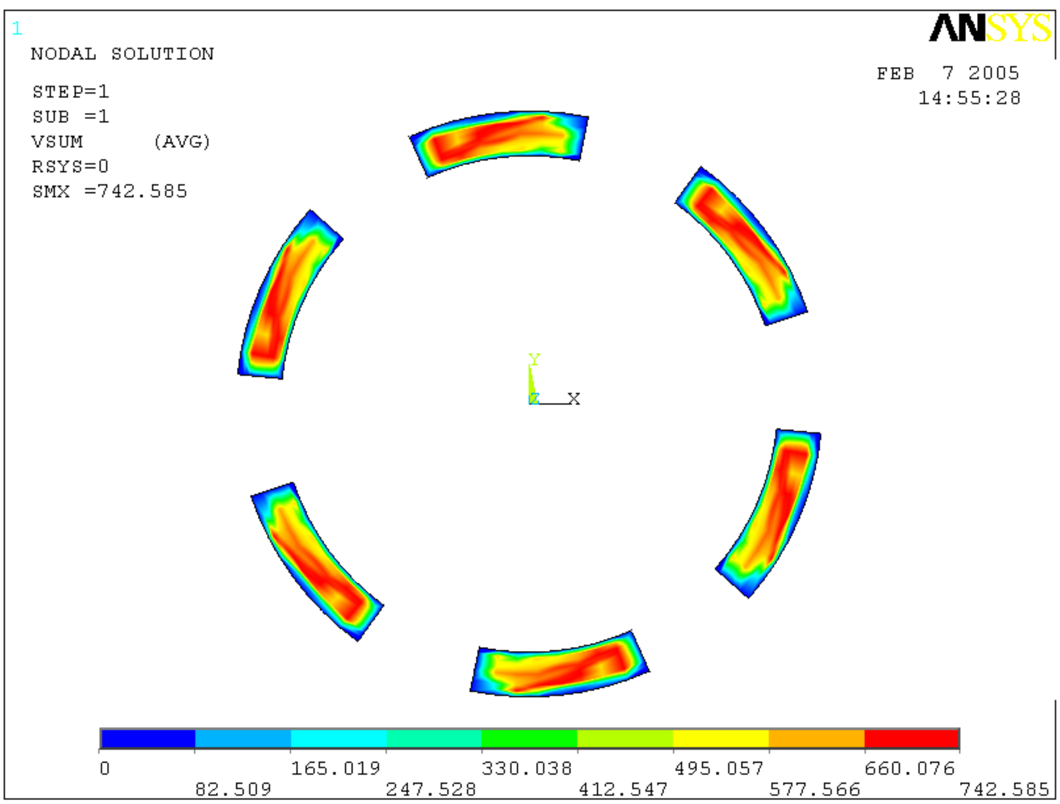
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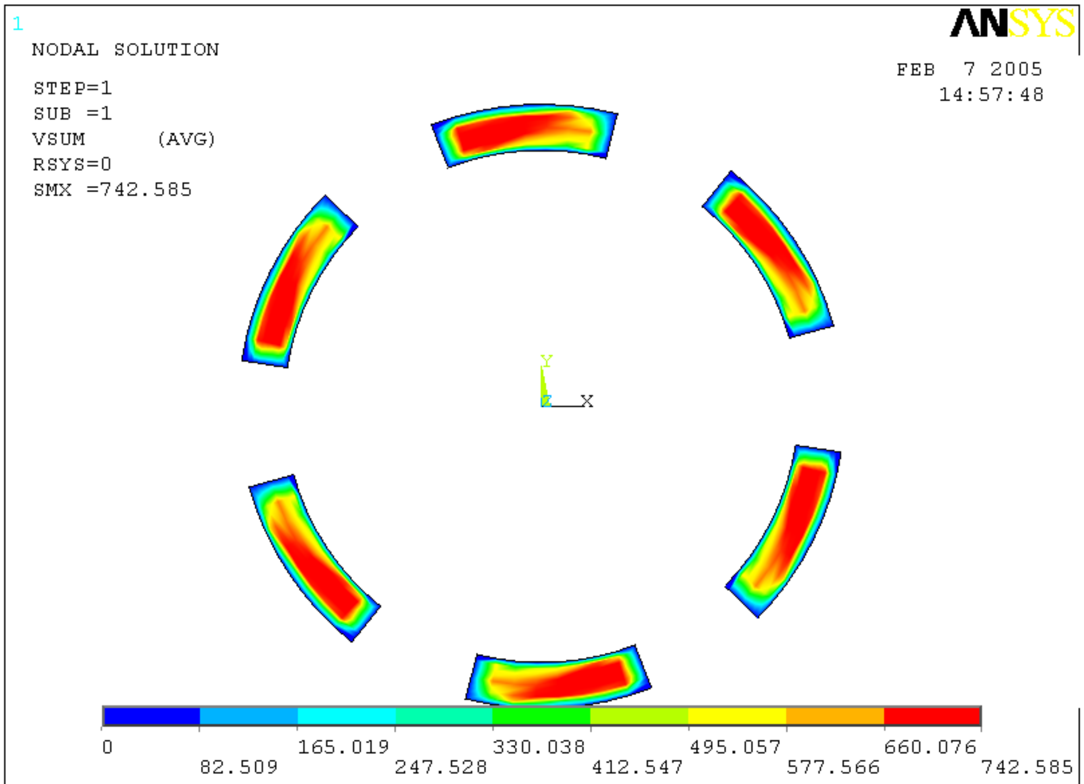
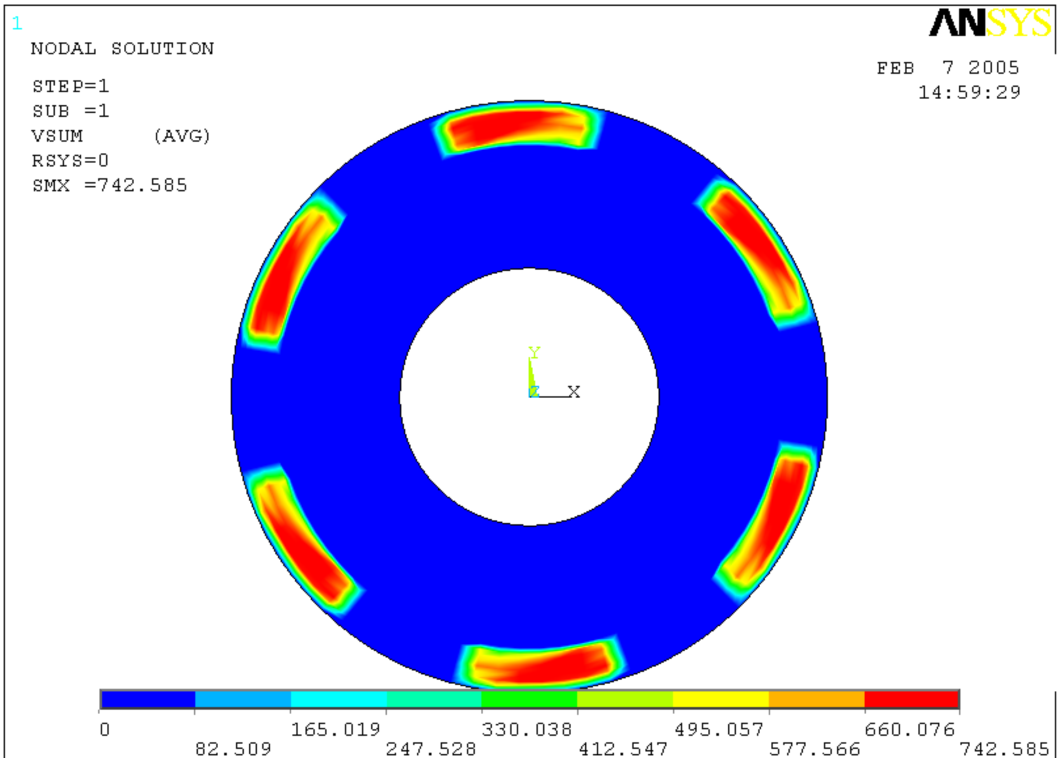
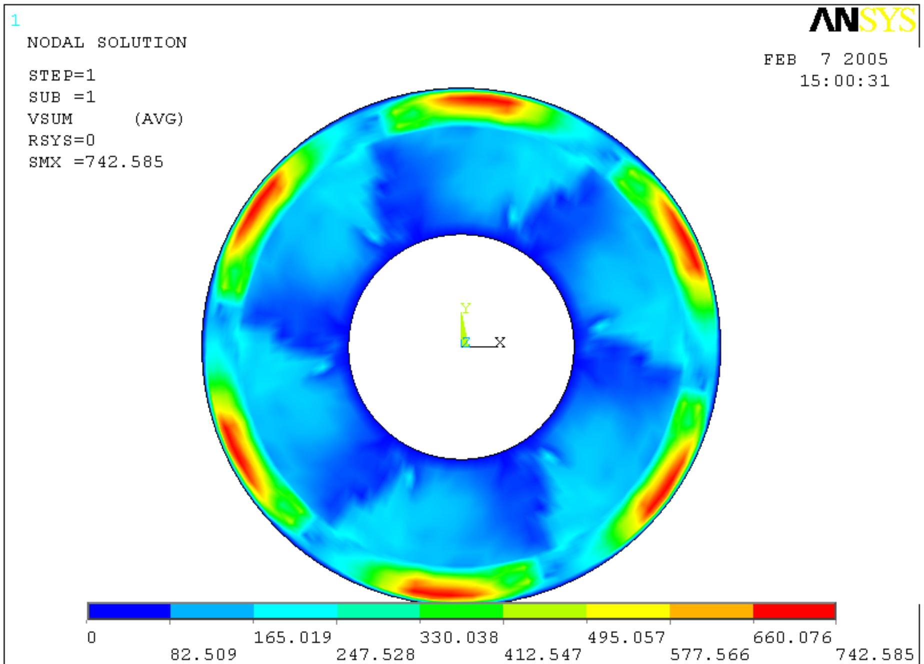


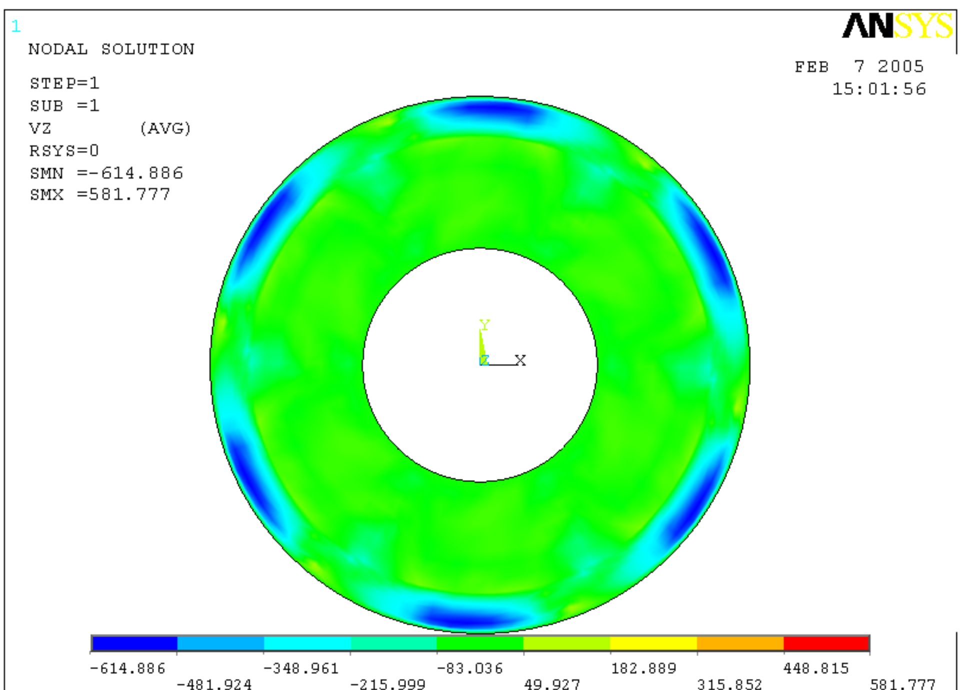
Total Fluid Velocity - 0.200" upstream of the exit



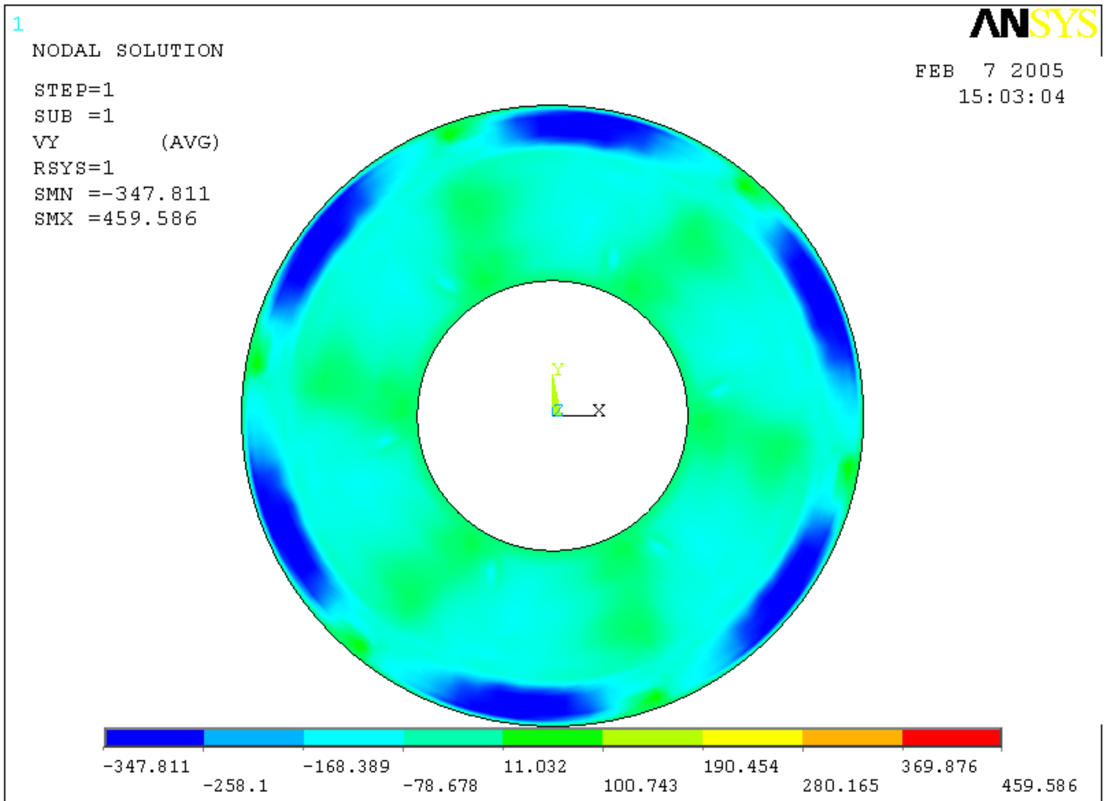
Total Fluid Velocity - At the exit



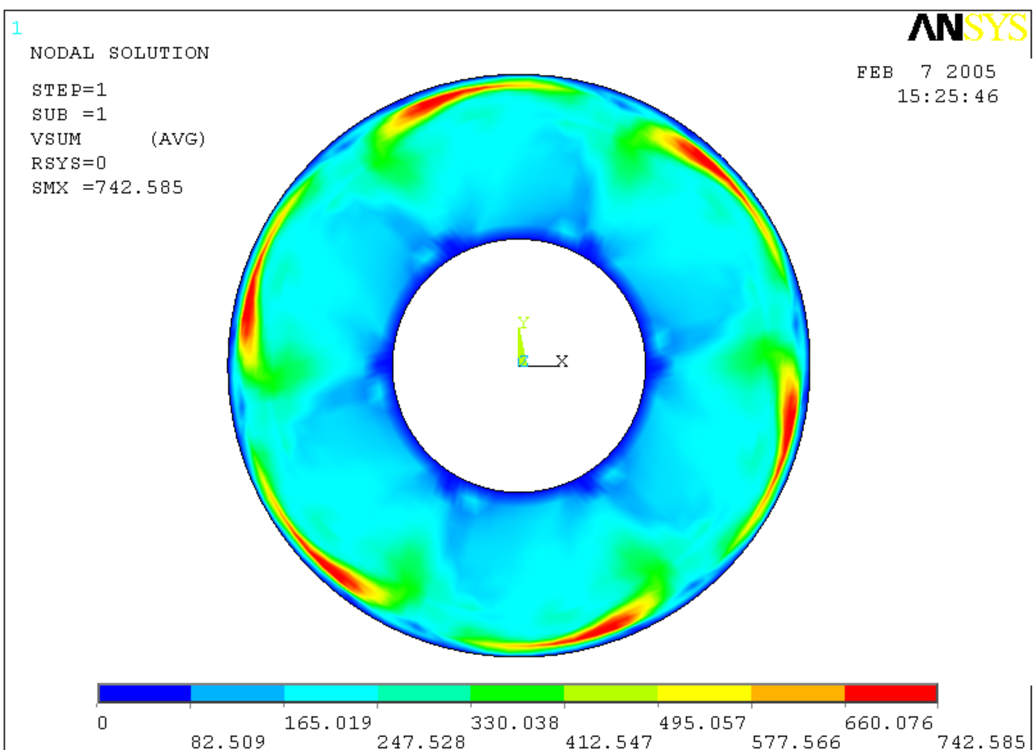
Total Fluid Velocity - 0.500" downstream of the exit



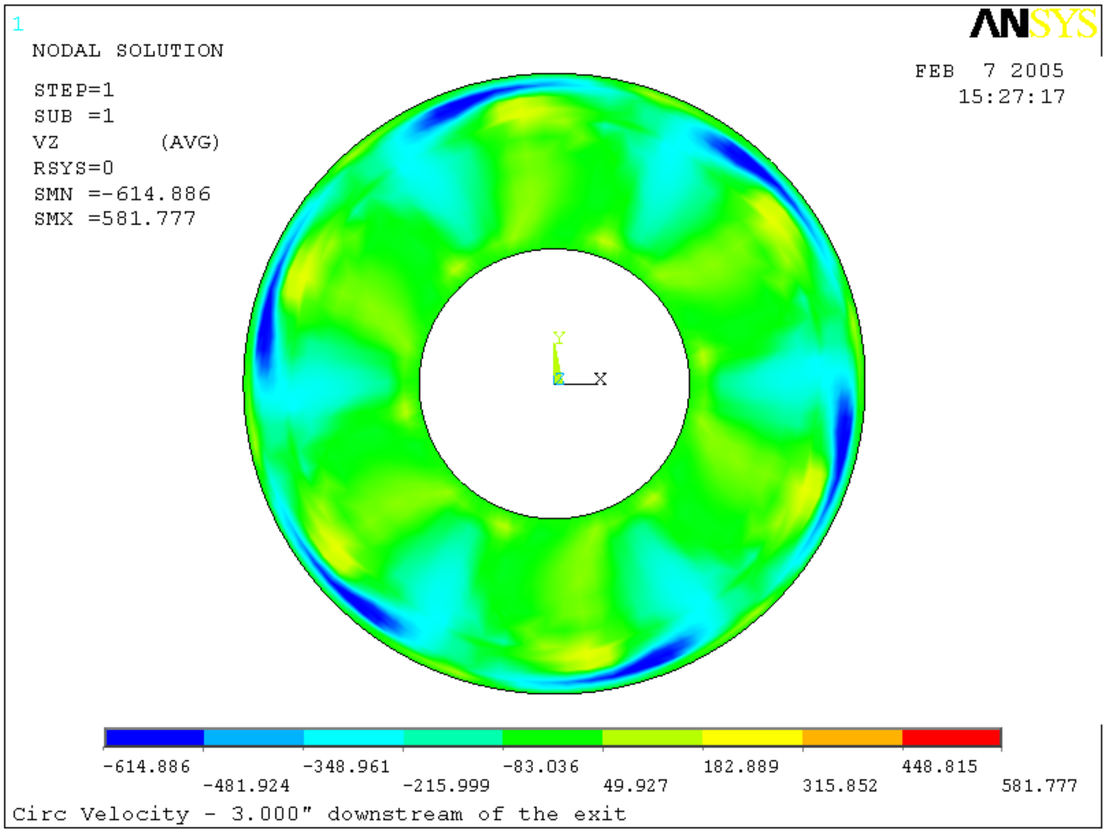
Axial Fluid Velocity - 0.500" downstream of the exit

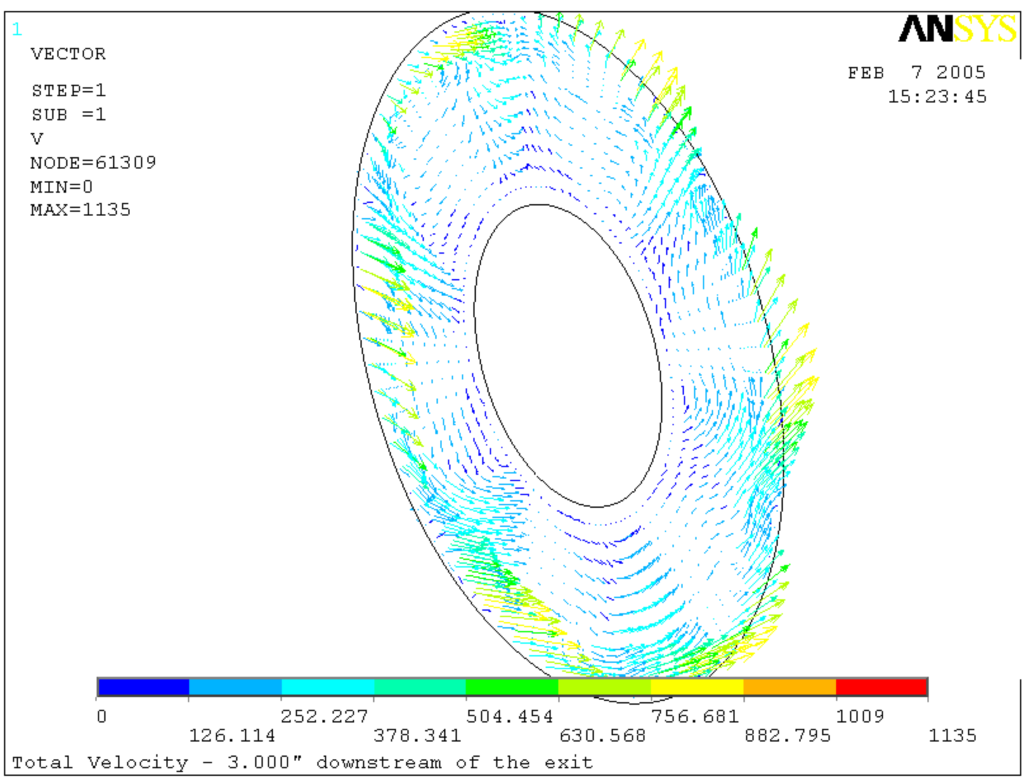


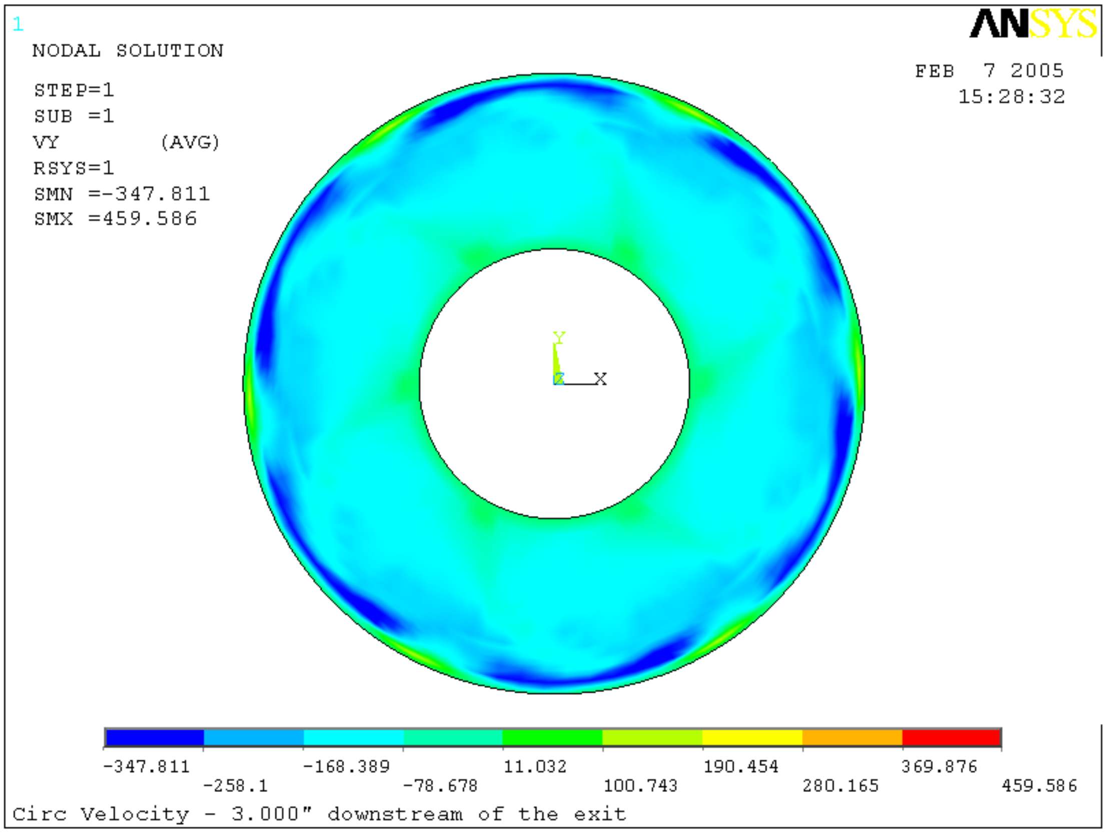
Circumferential Fluid Velocity - 0.500" downstream of the exit

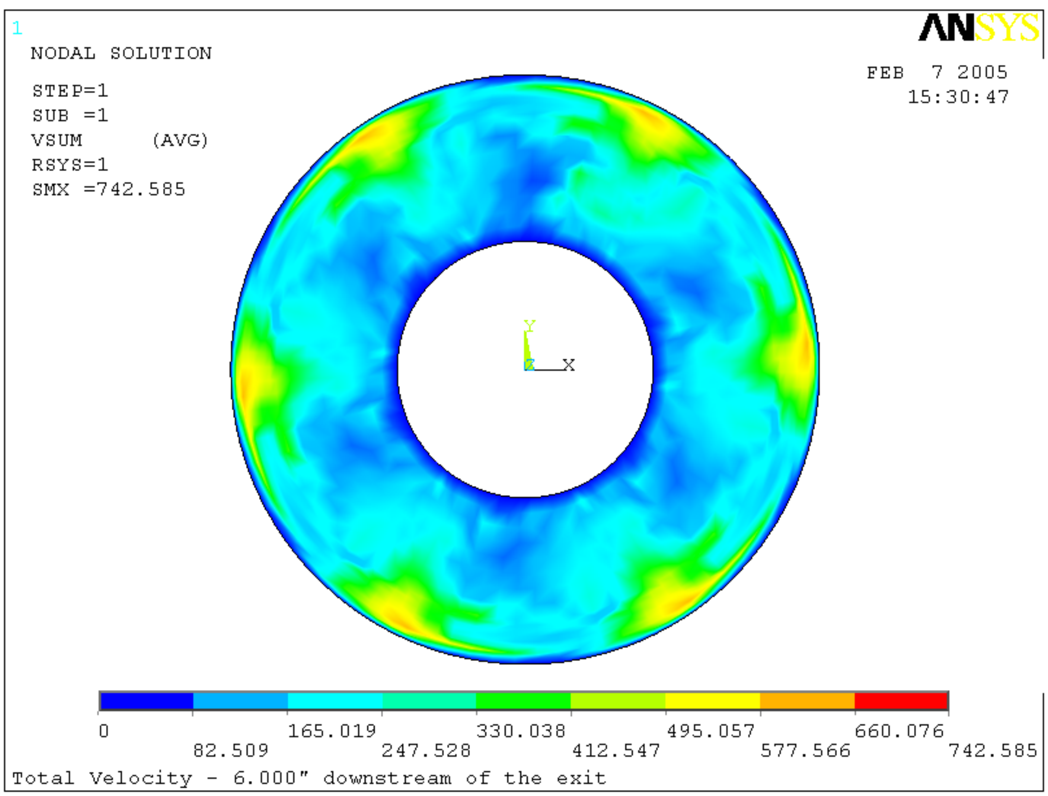


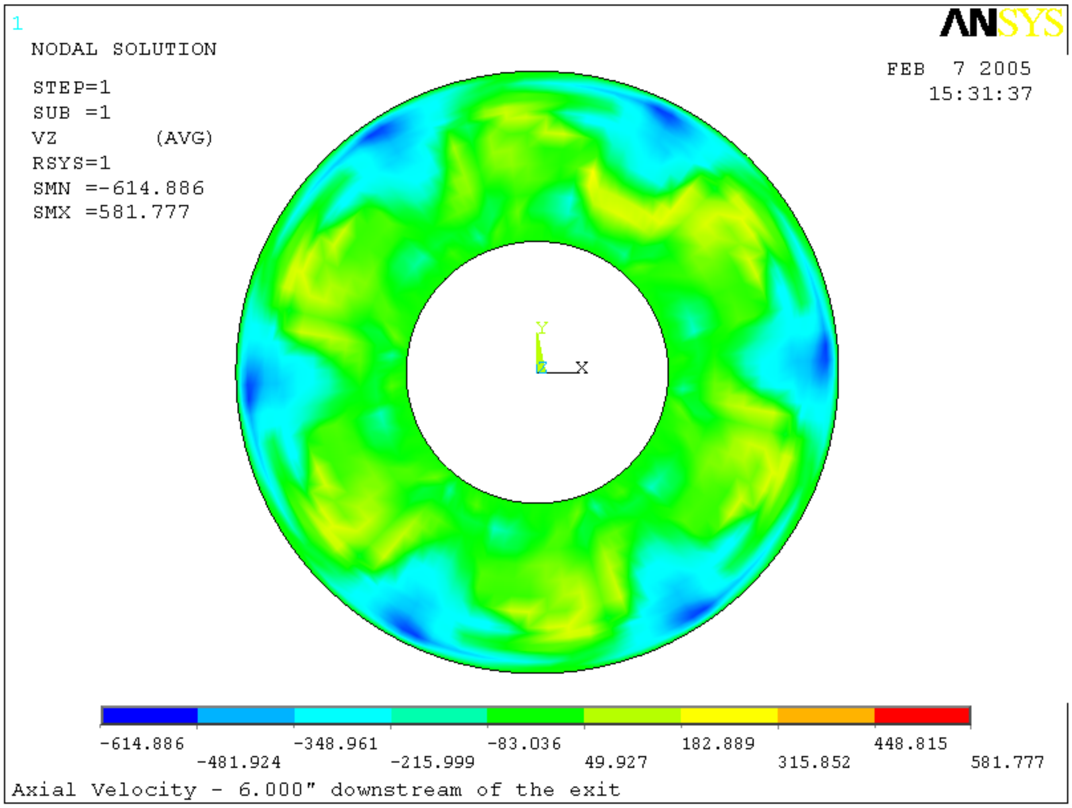
Total Velocity - 3.000" downstream of the exit

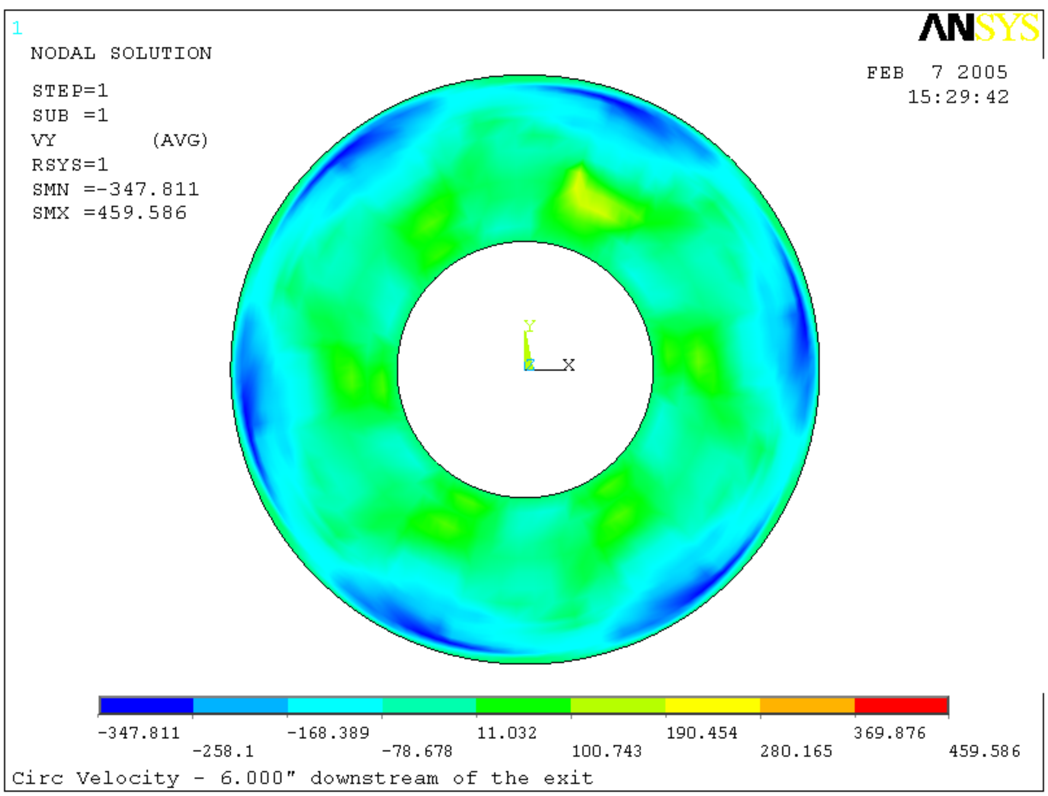

1
NODAL SOLUTION
STEP=1
SUB =1
VZ (AVG)
RSYS=0
SMN =-614.886
SMX =581.777
ANSYS
FEB 7 2005
15:27:17
Y
Z X
-614.886
-481.924
-348.961
-215.999
-83.036
49.927
182.889
315.852
448.815
581.777
Circ Velocity - 3.000" downstream of the exit

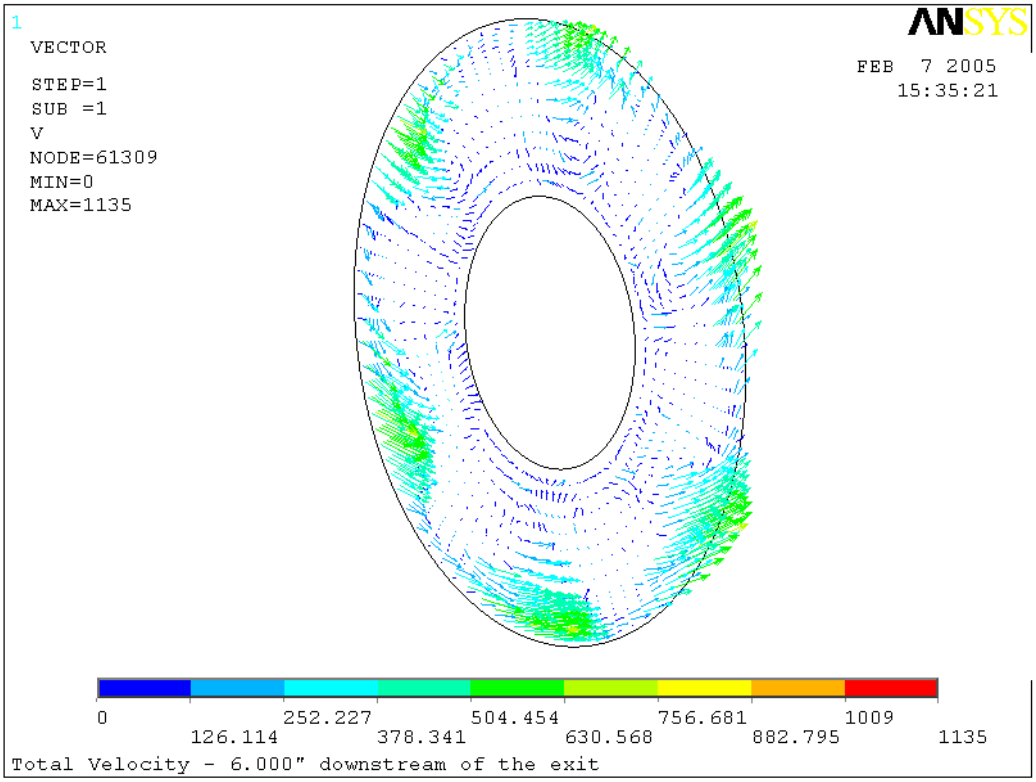

1
VECTOR
STEP=1
SUB =1
V
NODE=61309
MIN=0
MAX=1135
ANSYS
FEB 7 2005
15:23:45
0
126.114
252.227
378.341
504.454
630.568
756.681
882.795
1009
1135
Total Velocity - 3.000" downstream of the exit


1
NODAL SOLUTION
STEP=1
SUB =1
VY (AVG)
RSYS=1
SMN =-347.811
SMX =459.586
ANSYS
FEB 7 2005
15:28:32
Y
X
-347.811
-258.1
-168.389
-78.678
11.032
100.743
190.454
280.165
369.876
459.586
Circ Velocity - 3.000" downstream of the exit


1
NODAL SOLUTION
STEP=1
SUB =1
VSUM (AVG)
RSYS=1
SMX =742.585
ANSYS
FEB 7 2005
15:30:47
Y
X
0
82.509
165.019
247.528
330.038
412.547
495.057
577.566
660.076
742.585
Total Velocity - 6.000" downstream of the exit


1
NODAL SOLUTION
STEP=1
SUB =1
VZ (AVG)
RSYS=1
SMN =-614.886
SMX =581.777
ANSYS
FEB 7 2005
15:31:37
Y
X
-614.886
-481.924
-348.961
-215.999
-83.036
49.927
182.889
315.852
448.815
581.777
Axial Velocity - 6.000" downstream of the exit


1
NODAL SOLUTION
STEP=1
SUB =1
VY (AVG)
RSYS=1
SMN =-347.811
SMX =459.586
ANSYS
FEB 7 2005
15:29:42
Y
X
-347.811
-258.1
-168.389
-78.678
11.032
100.743
190.454
280.165
369.876
459.586
Circ Velocity - 6.000" downstream of the exit


1
VECTOR
STEP=1
SUB =1
V
NODE=61309
MIN=0
MAX=1135
ANSYS
FEB 7 2005
15:35:21
0
126.114
252.227
378.341
504.454
630.568
756.681
882.795
1009
1135
Total Velocity - 6.000" downstream of the exit

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National Energy Technology Laboratory

626 Cochrans Mill Road
P.O. Box 10940
Pittsburgh, PA 15236-0940

3610 Collins Ferry Road
P.O. Box 880
Morgantown, WV 26507-0880

One West Third Street, Suite 1400
Tulsa, OK 74103-3519

1450 Queen Avenue SW
Albany, OR 97321-2198

2175 University Ave. South
Suite 201
Fairbanks, AK 99709

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