# NOVEL HIGH-SPEED DRILLING MOTOR FOR OIL EXPLORATION & PRODUCTION

# INTERIM PROGRESS REPORT #1

| | |
|---|---|
| Starting date: | October 1, 2004 |
| Ending Date: | March 31, 2005 |
| Principal Author: | Martin E. Cobern |
| Date Issued: | May 10, 2005 |
| DOE Award Number: | DE-FC26-04NT15501 |
| Submitting Organization | APS Technology, Inc.<br>800 Corporate Row<br>Cromwell, CT 06416 |

## DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

## 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, we 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.

Phase I is proceeding on or ahead of schedule, with significant progress on all but one of the eight tasks.

## Table of Contents

Executive Summary ..... 4
1.0 Analysis & Design [1st year] ..... 5
1.1 Evaluation & Design of Power Section ..... 5
1.2 Gearing Design ..... 5
1.3 Overall Conceptual Design ..... 8
1.4 Sealing System Design ..... 8
1.5 Flexible Coupling & Casing ..... 9
1.6 Bearings ..... 9
1.7 Critical Frequencies & System Damping ..... 9
1.8 Test Equipment Design ..... 9
2.0 Laboratory Testing [2nd year] ..... 10
2.1 Laboratory System ..... 10
2.2 Test Facility ..... 10
2.3 Laboratory Testing ..... 10
2.4 Drilling at Test Facilities or Test Wells ..... 10
Units ..... 10
References ..... 11
Figures ..... 12
Table of Figures ..... 12
Appendix A: Vortex Pressure Drop & Velocity Study ..... 14
Scope ..... 15
Conclusions ..... 15
Results ..... 16

# 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[2], 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, we 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.

Phase I is proceeding on or ahead of schedule, with significant progress on all but one of the eight tasks. Specifically:

1. The choice of power section is complete.
2. The gearing design is complete.
3. The overall design concept is basically complete, but is being refined.
4. The high-speed and low-speed seals have been chosen and their design is being completed.
5. The flexible coupling has been chosen, but an additional marine bearing may be needed and is being designed.
6. The type of bearing for each of the four required has been specified and their design is progressing.
7. The work on evaluating critical frequencies will begin in the next quarter, along with the design of the damping mechanisms.
8. The flow loop has been designed.

## 9. Project Tasks & Progress

### *1.0 Analysis & Design [1st year]*

The objective of this phase is to arrive at a viable design for the motor. This design effort will include a number of studies. At the end of this Phase, we will have fully modeled designs for the motor power section, bearing packs, and the gear.

1.1 Evaluation & Design of Power Section

As stated in the application, This first task will be to consider the primary source of drilling power – the motor. Although is seems likely that our final choice will be a PDM, we will also evaluate the possibility of using a drilling turbine. This analysis will include studies of the anticipated power and torque generated at various speeds, and the effect of the gearing on the output (see **Task 1.2**)

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, we selected three motor designs as the most likely candidates.

For completeness, we also evaluated available drilling turbines. A small-diameter turbodrill did not deliver sufficient torque, and was extremely long compared to the motors. For reference, we 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 (rpm) | Input Torque (ft-lb) | Power (hp) | Length (in) | HP/Length (hp/in) | Required Gear Ratio | Torque @ 10,000 RPM (ft-lb) | Rank |
|---|---|---|---|---|---|---|---|---|---|
| 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** |

With the selection or these motors, this task is **complete**. Some work will be necessary in later stages to refine the coupling of the power section to the gearing *etc*.

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 has chosen 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. 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.

**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 |

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**.

**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** |

By comparison, the ‘flow over’ design 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.

**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** |

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 the difficulties outlined in **Table 2**, above. 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’, described below in **Task 1.3** below, and additional sealing problems, discussed in **Task 1.4**

## 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 1**. 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 under consideration is a carbon graphite (CGr) filled polytetrafluoroethene (PFTE, or Teflon) ring-loaded seal being desgned 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 we 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 2**. 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

This task is not yet underway, but will begin in the coming quarter. The critical speed analysis will analyze the shafts and bearings to determine the natural frequencies and determine the forced vibration response. Sources of imbalance include; shaft and gearing imbalance, bit imbalance (especially in the worn condition, or if the bit is to lose a cutter), drilling forces and gear forces.

### 1.8 Test Equipment Design

The primary test equipment used in this project will be a flow loop. While initially planned for purchase and assembly in Year 2 of the project, it has become clear that it will be needed by the end of the first year, as we expect to have prototype components to test.

The flow loop will consist of a large structural steel baseplate that will support the entire high-speed motor assembly. Flow will be supplied by an 850 hp diesel-driven National Oilwell triplex pump capable of flows to 750 gpm and

1500 psi. The high speed motor will connected to a 500 hp Kahn dynamometer. Flow exiting the high speed motor will be slung into a collector where it will be 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.

A schematic of the flow loop test facility follows below in **Figure 3**.

## *2.0 Laboratory Testing [2nd year]*

### 2.1 Laboratory System

No effort to date.

### 2.2 Test Facility

See Section 1.8 , above.

### 2.3 Laboratory Testing

No effort to date.

### 2.4 Drilling at Test Facilities or Test Wells

No effort to date.

# 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] 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

# Figures

## *Table of Figures*

Figure 1: ‘Flow Over’ Gearbox Design ..... 12
Figure 2: Constant velocity joint assembly ..... 12
Figure 3: Flow Loop Design ..... 13

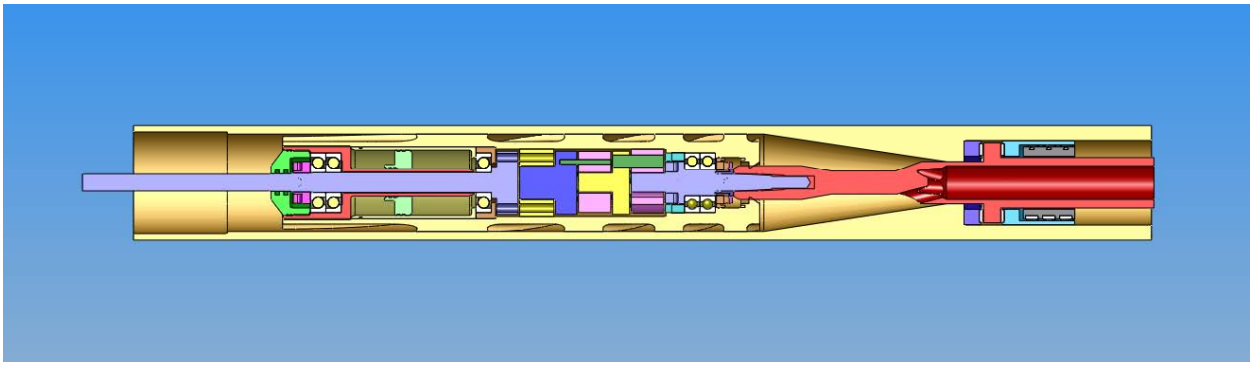


**Figure 1: ‘Flow Over’ Gearbox Design**

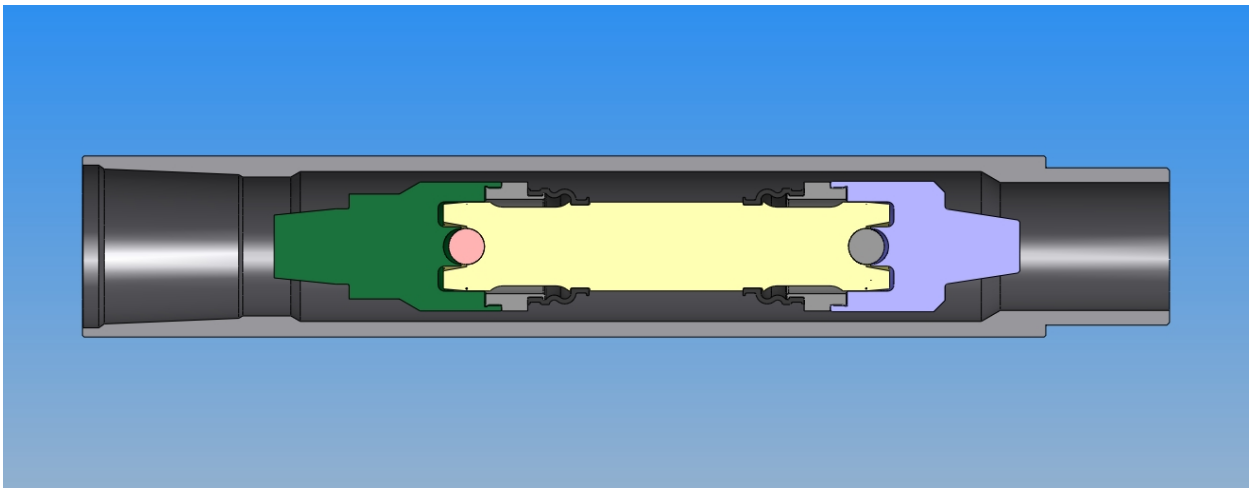
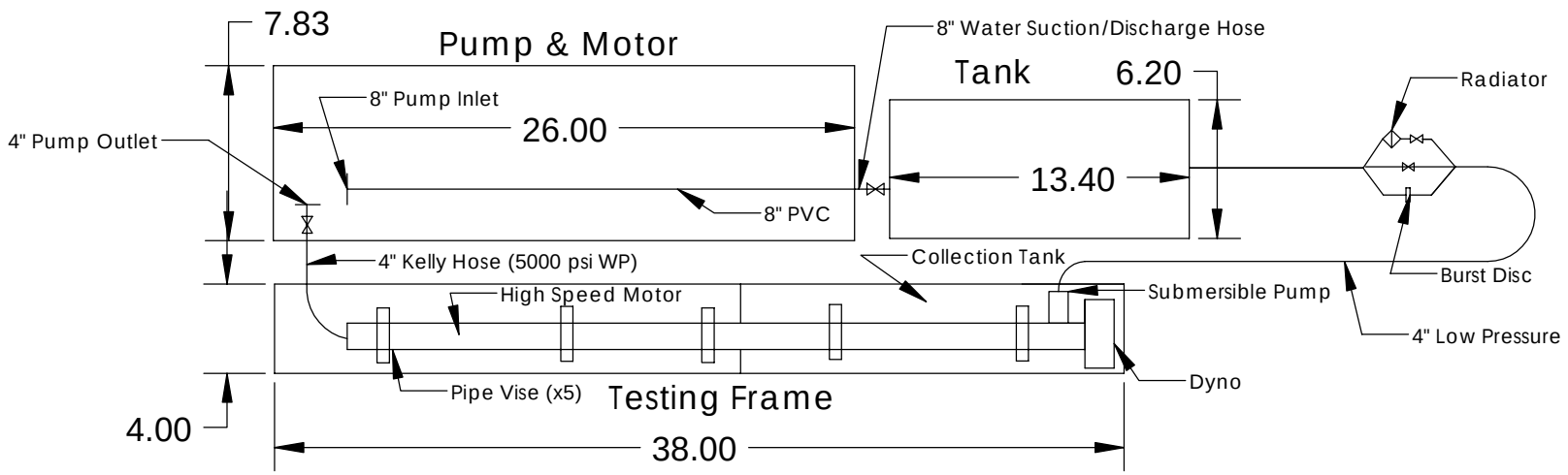


**Figure 2: Constant velocity joint assembly**


7.83
Pump & Motor
8" Water Suction/Discharge Hose
Tank
6.20
Radiator
8" Pump Inlet
26.00
4" Pump Outlet
13.40
8" PVC
4" Kelly Hose (5000 psi WP)
Collection Tank
Burst Disc
High Speed Motor
Submersible Pump
4" Low Pressure
Dyno
Pipe Vise (x5)
Testing Frame
4.00
38.00

# Appendix A: Vortex Pressure Drop & Velocity Study

**M Wassell**

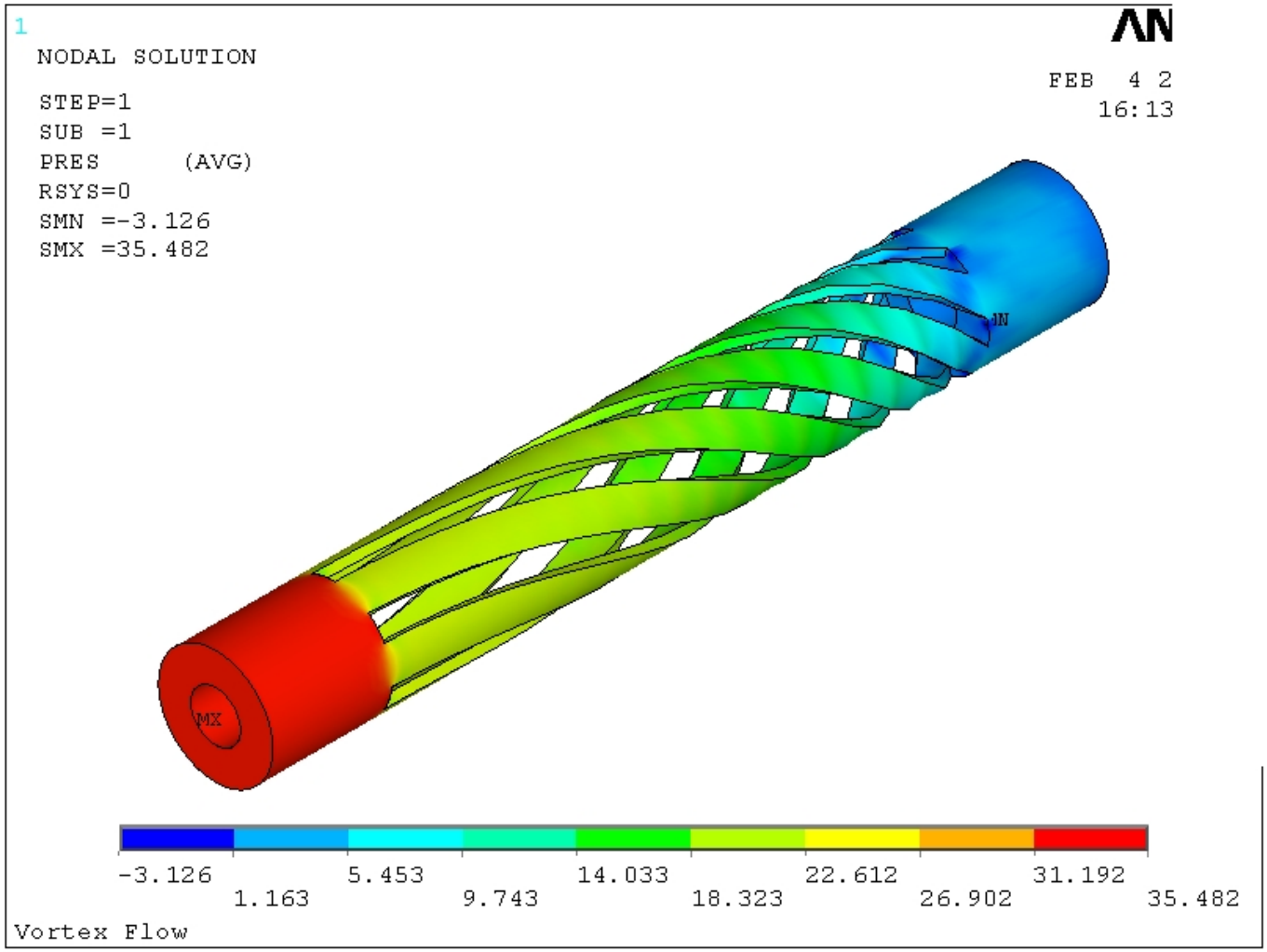
***February 7, 2005***

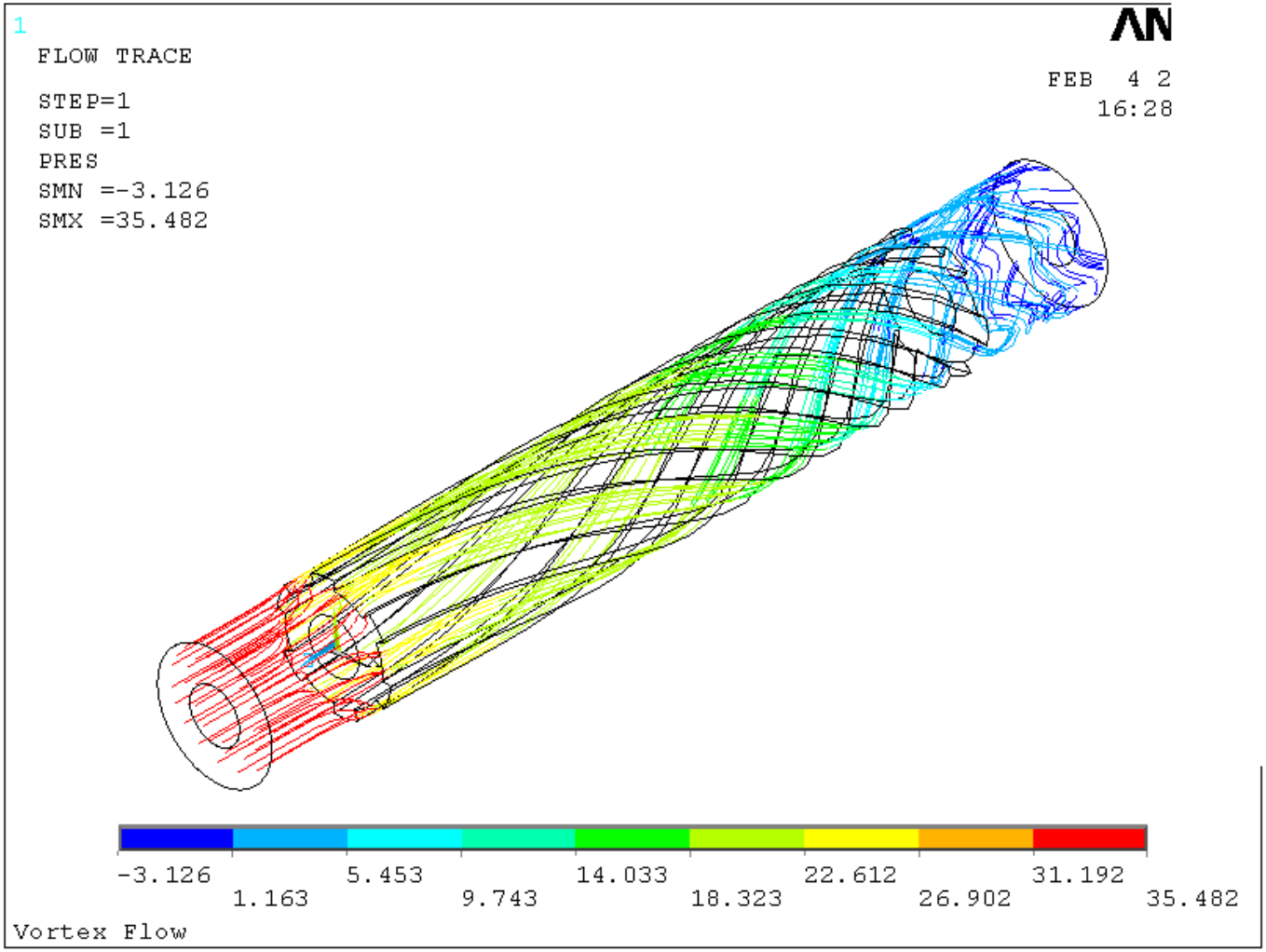
### *Scope*

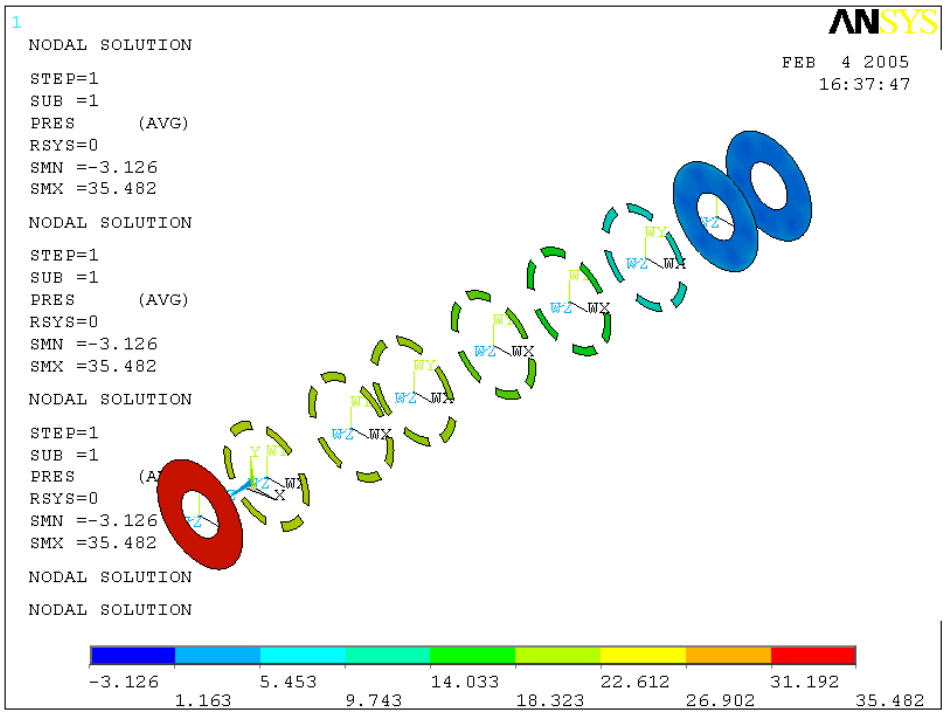
This analysis is an analysis of the pressure drop and the velocities generated through the outer vortex. The flow is based on water based mud at a flow rate of 400 GPM. The mud velocity at the inlet is 6 ft/sec (400 GPM, 5.75”OD x 2.75”ID).

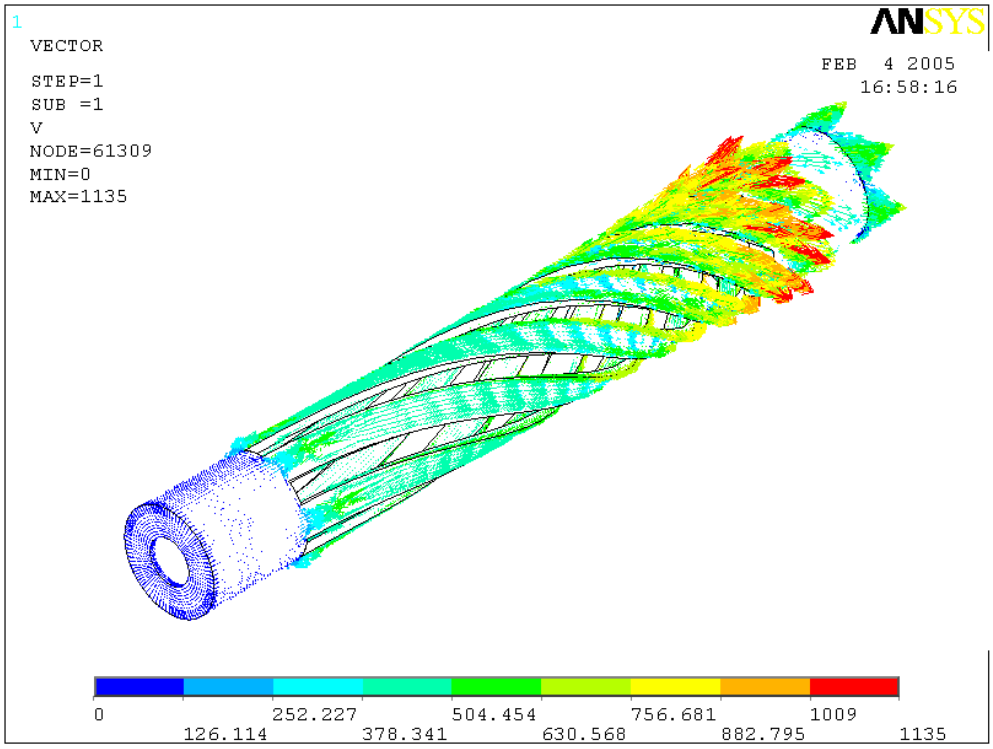
### *Conclusions*

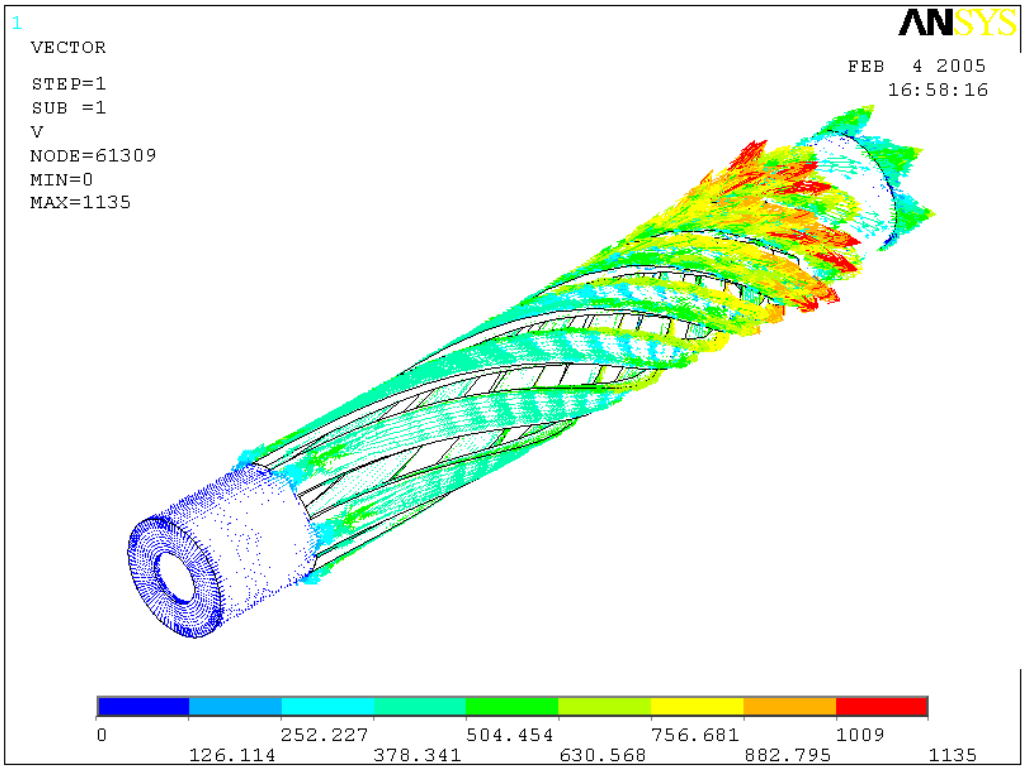
1. The pressure drop across the vortex is 35 psi.
2. The maximum velocity of 742 in/sec (90 ft/sec) occurs at the exit.
3. Most of the circumferential flow occurs at the OD. There is little circumferential flow at the ID. This is opposite of what is desired since the shaft is rotating and the outer casing 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.

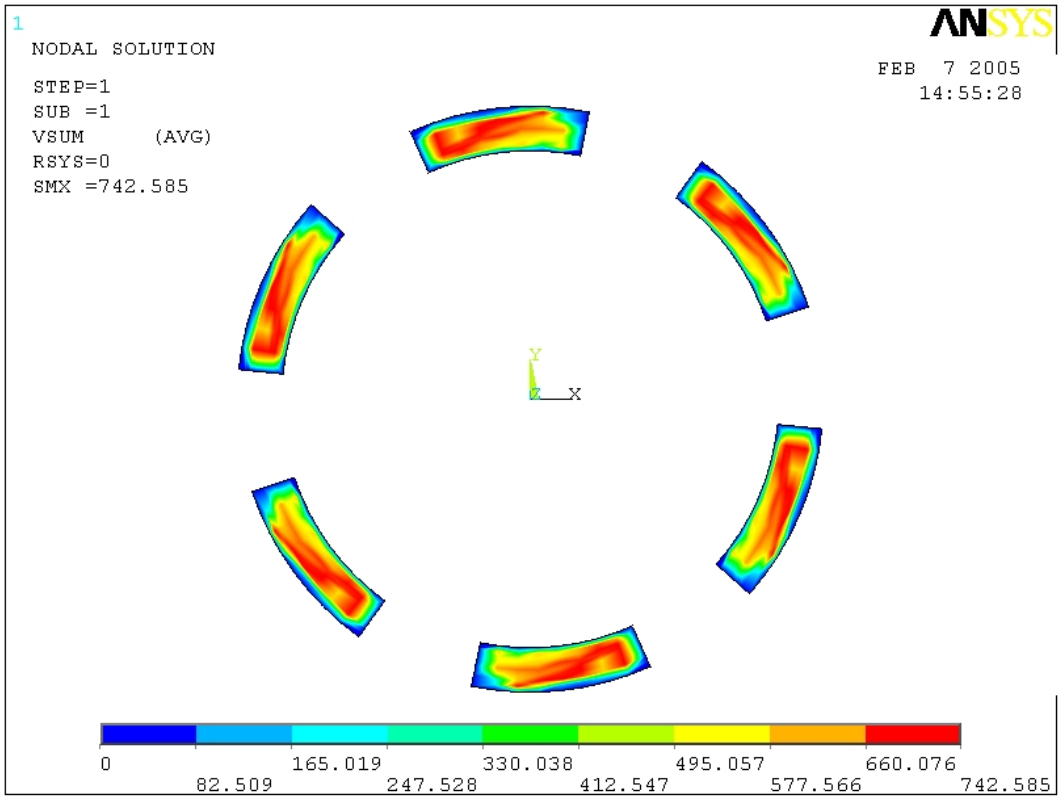

1
NODAL SOLUTION
STEP=1
SUB =1
PRES (AVG)
RSYS=0
SMN =-3.126
SMX =35.482
FEB 4 2
16:13
MN
MX
-3.126
1.163
5.453
9.743
14.033
18.323
22.612
26.902
31.192
35.482
Vortex Flow

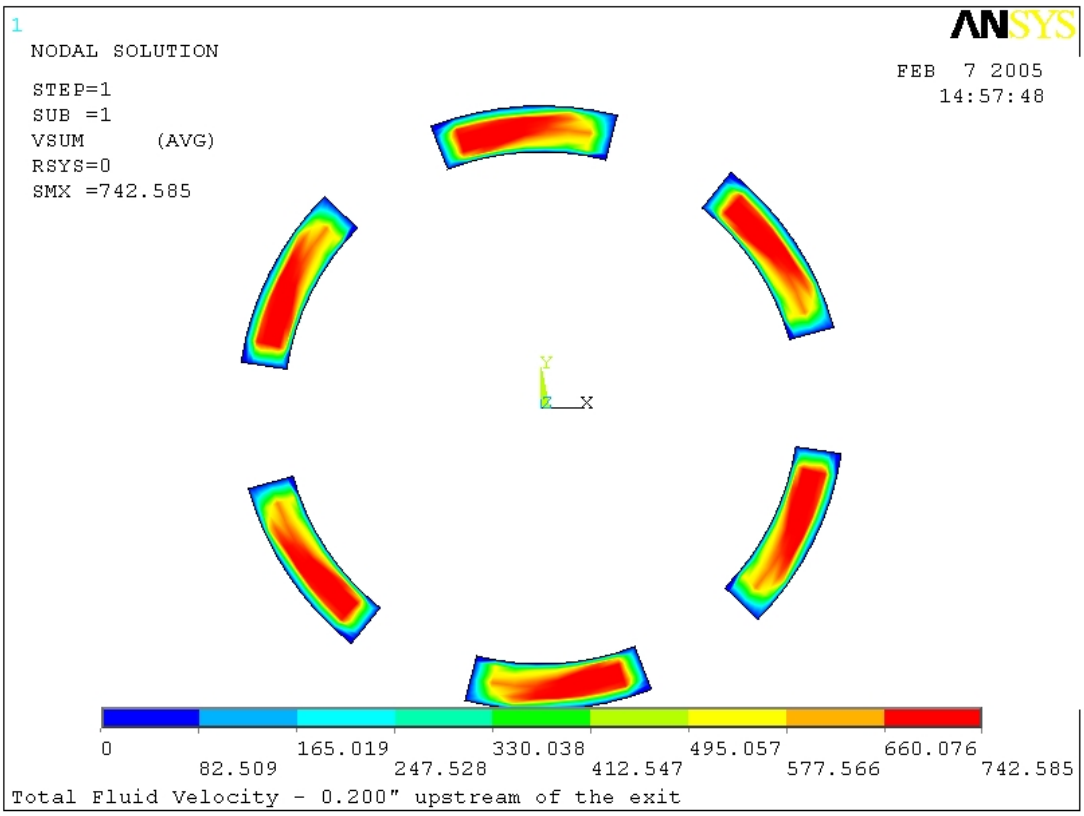

1
FLOW TRACE
STEP=1
SUB =1
PRES
SMN =-3.126
SMX =35.482
FEB 4 2
16:28
-3.126
1.163
5.453
9.743
14.033
18.323
22.612
26.902
31.192
35.482
Vortex Flow

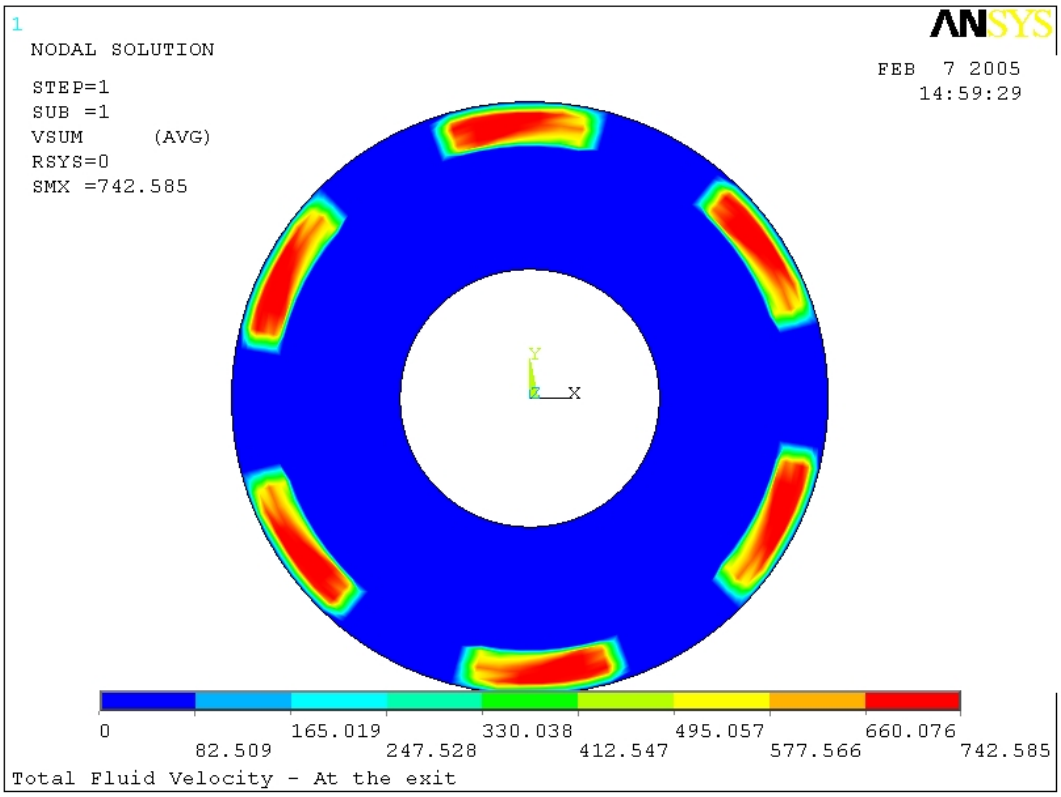

1
NODAL SOLUTION
STEP=1
SUB =1
PRES (AVG)
RSYS=0
SMN =-3.126
SMX =35.482
NODAL SOLUTION
STEP=1
SUB =1
PRES (AVG)
RSYS=0
SMN =-3.126
SMX =35.482
NODAL SOLUTION
STEP=1
SUB =1
PRES (A
RSYS=0
SMN =-3.126
SMX =35.482
NODAL SOLUTION
NODAL SOLUTION
ANSYS
FEB 4 2005
16:37:47
-3.126
1.163
5.453
9.743
14.033
18.323
22.612
26.902
31.192
35.482

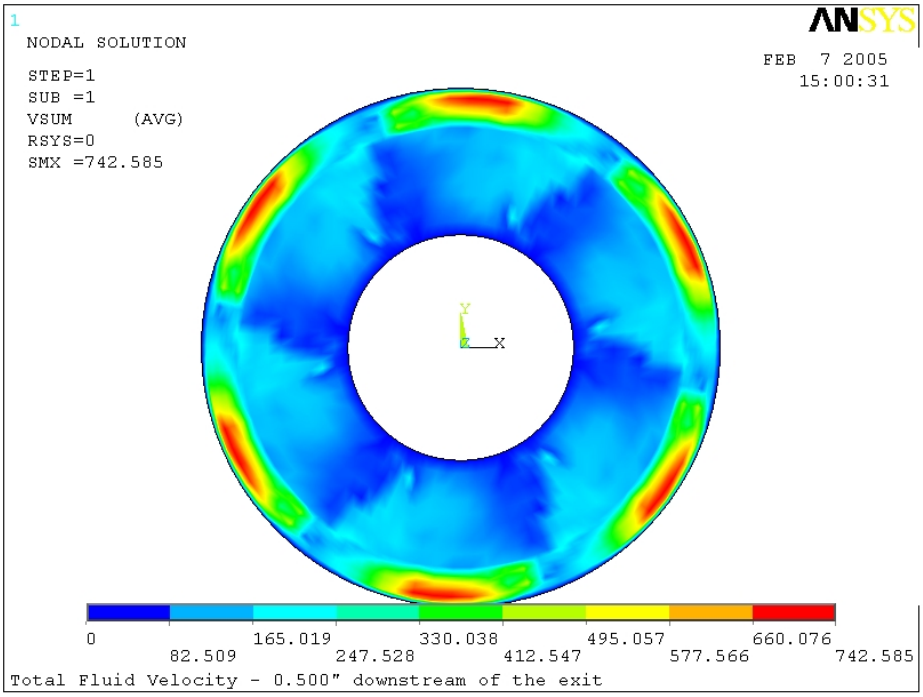

1
VECTOR
STEP=1
SUB =1
V
NODE=61309
MIN=0
MAX=1135
ANSYS
FEB 4 2005
16:58:16
0
126.114
252.227
378.341
504.454
630.568
756.681
882.795
1009
1135

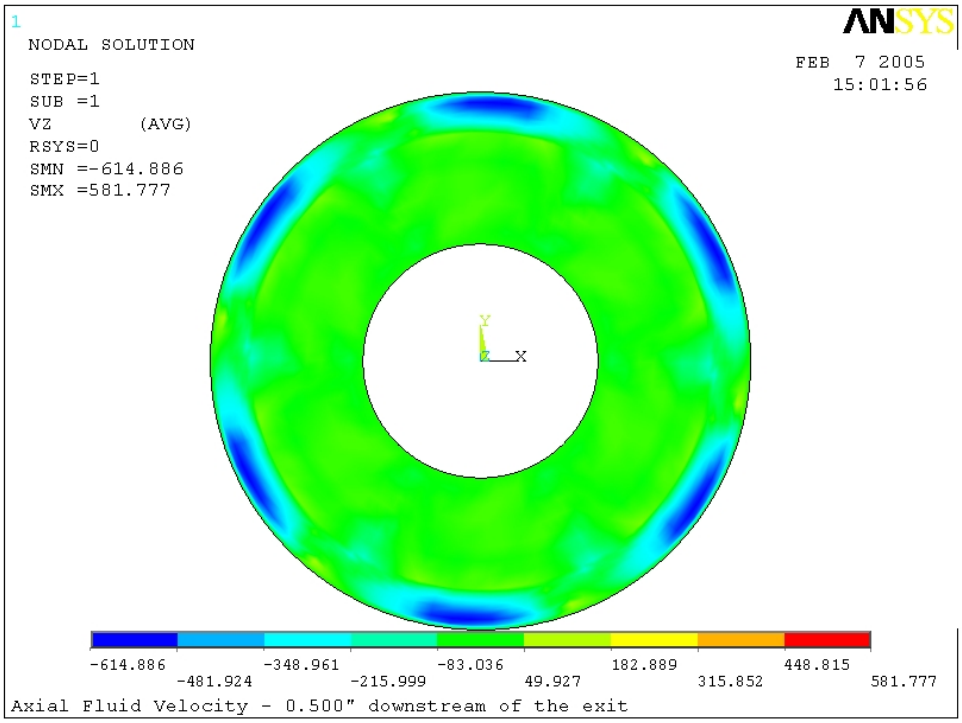

1
VECTOR
STEP=1
SUB =1
V
NODE=61309
MIN=0
MAX=1135
ANSYS
FEB 4 2005
16:58:16
0
126.114
252.227
378.341
504.454
630.568
756.681
882.795
1009
1135

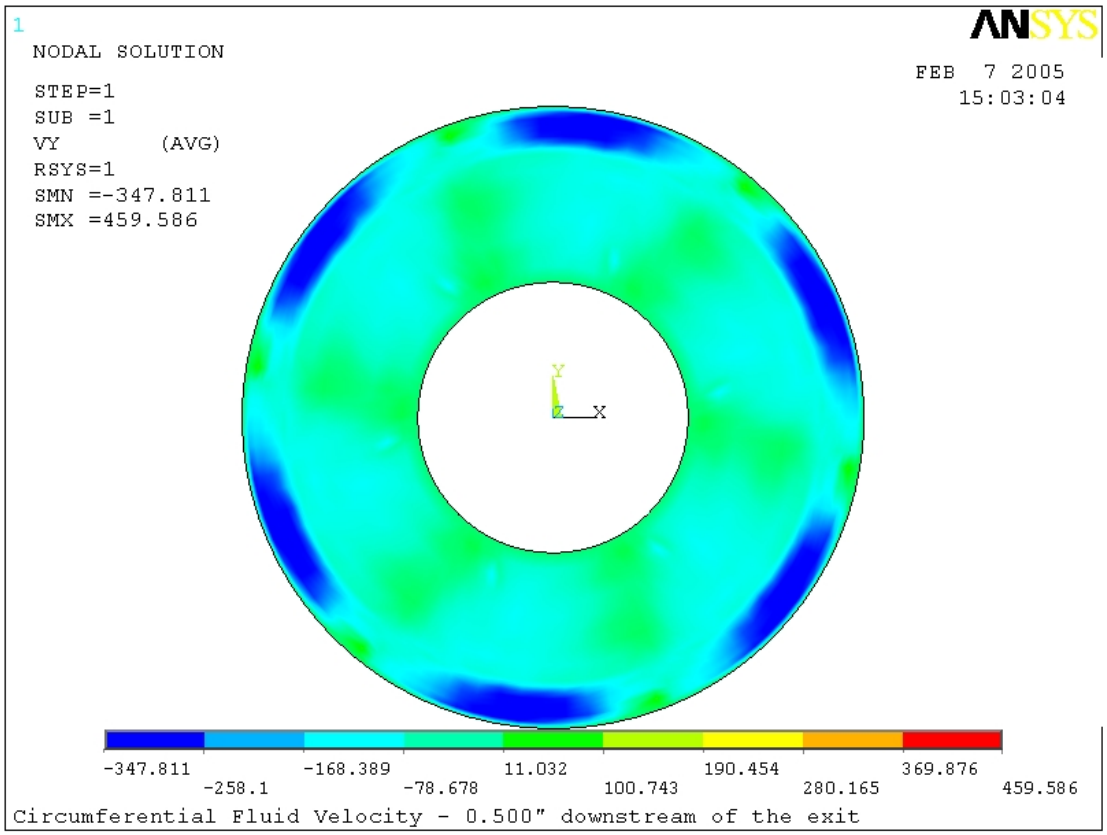

1
NODAL SOLUTION
STEP=1
SUB =1
VSUM (AVG)
RSYS=0
SMX =742.585
ANSYS
FEB 7 2005
14:55:28
Y
Z
X
0
82.509
165.019
247.528
330.038
412.547
495.057
577.566
660.076
742.585

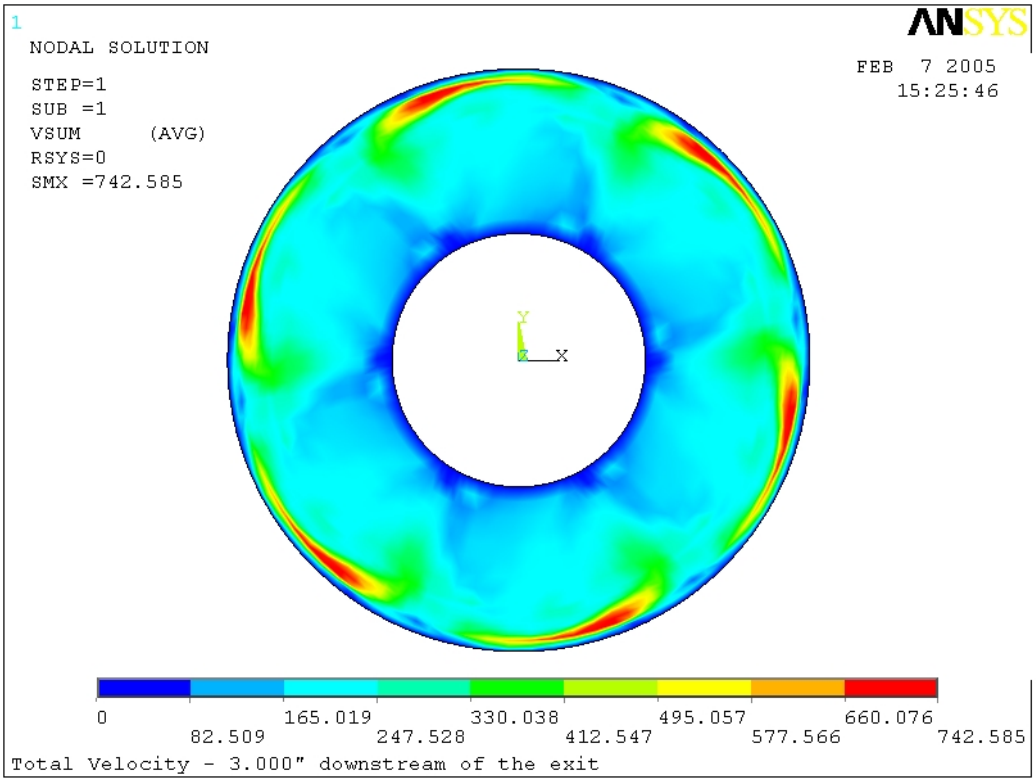

1
NODAL SOLUTION
STEP=1
SUB =1
VSUM (AVG)
RSYS=0
SMX =742.585
ANSYS
FEB 7 2005
14:57:48
Y
Z X
0
82.509
165.019
247.528
330.038
412.547
495.057
577.566
660.076
742.585
Total Fluid Velocity - 0.200" upstream of the exit

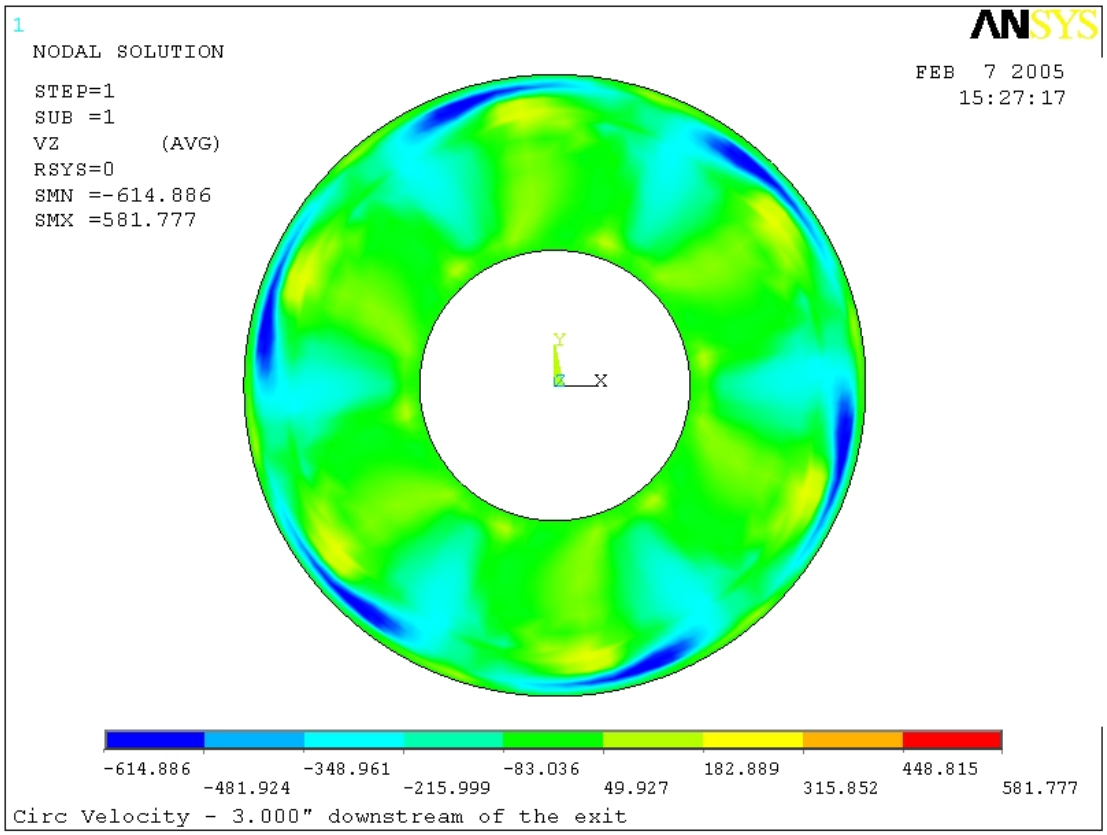

1
NODAL SOLUTION
STEP=1
SUB =1
VSUM (AVG)
RSYS=0
SMX =742.585
ANSYS
FEB 7 2005
14:59:29
Y
Z X
0
82.509
165.019
247.528
330.038
412.547
495.057
577.566
660.076
742.585
Total Fluid Velocity - At the exit

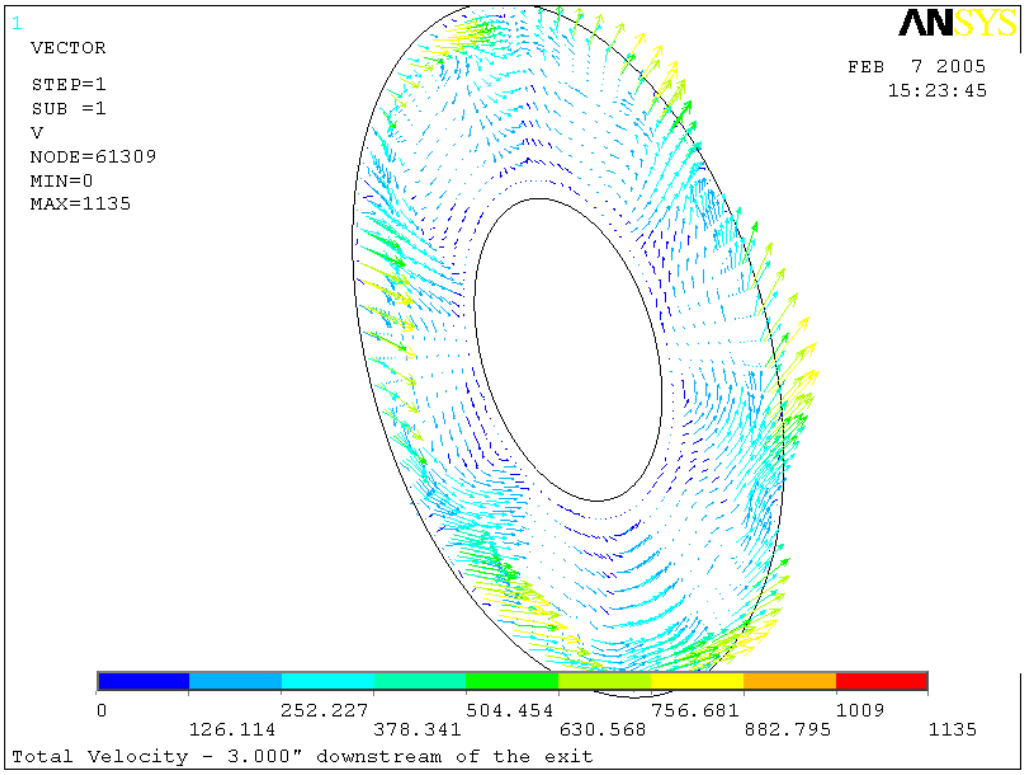

1
NODAL SOLUTION
STEP=1
SUB =1
VSUM (AVG)
RSYS=0
SMX =742.585
ANSYS
FEB 7 2005
15:00:31
Y
Z X
0
82.509
165.019
247.528
330.038
412.547
495.057
577.566
660.076
742.585
Total Fluid Velocity - 0.500" downstream of the exit

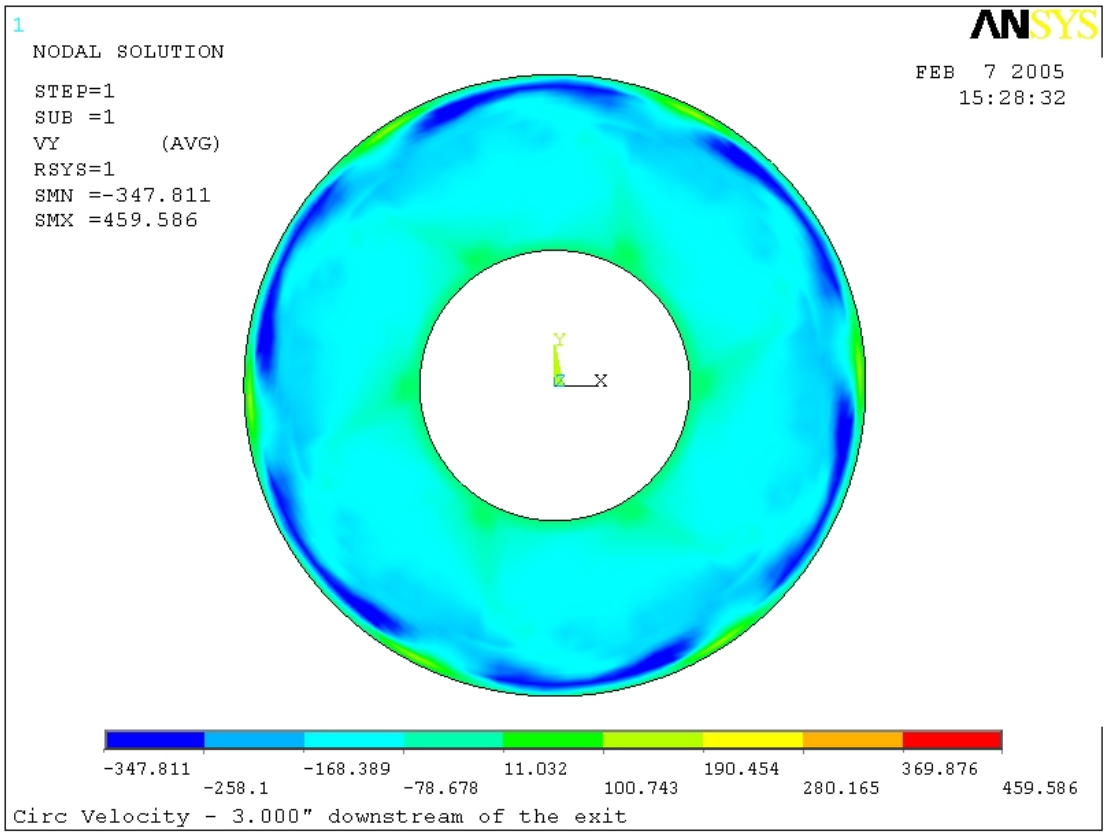

1
NODAL SOLUTION
STEP=1
SUB =1
VZ (AVG)
RSYS=0
SMN =-614.886
SMX =581.777
ANSYS
FEB 7 2005
15:01:56
Y
Z X
-614.886
-481.924
-348.961
-215.999
-83.036
49.927
182.889
315.852
448.815
581.777
Axial Fluid Velocity - 0.500" downstream of the exit

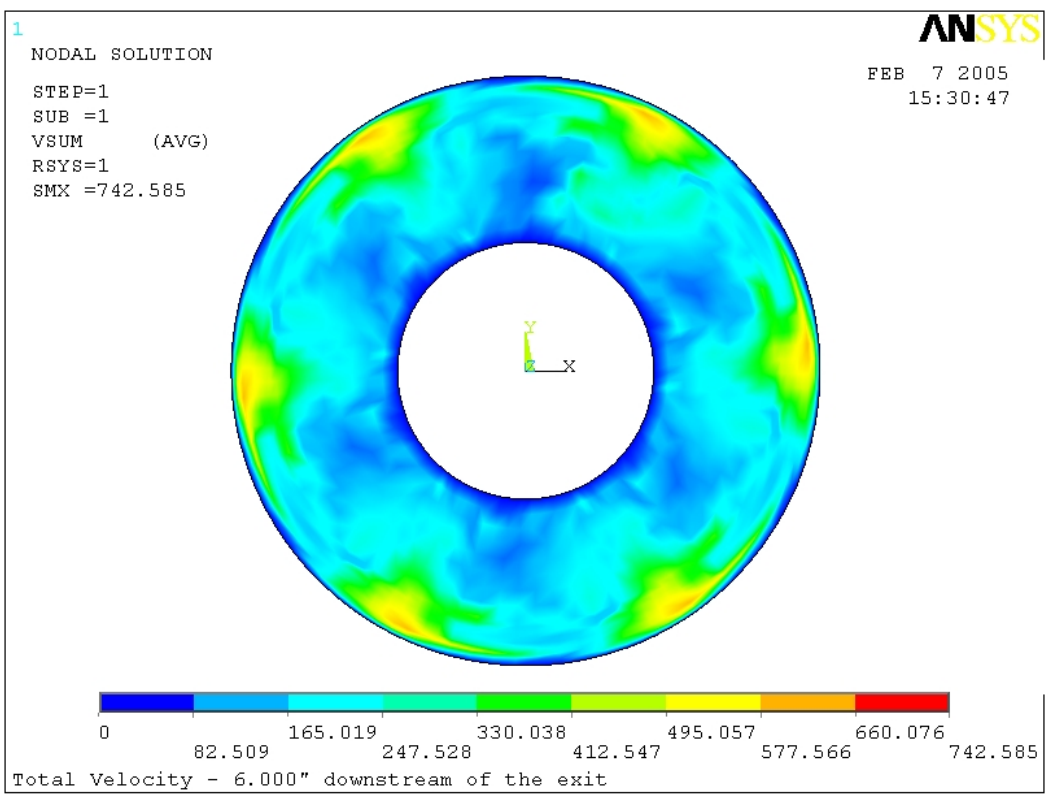

1
NODAL SOLUTION
STEP=1
SUB =1
VY (AVG)
RSYS=1
SMN =-347.811
SMX =459.586
ANSYS
FEB 7 2005
15:03:04
Y
Z X
-347.811
-258.1
-168.389
-78.678
11.032
100.743
190.454
280.165
369.876
459.586
Circumferential Fluid Velocity - 0.500" downstream of the exit

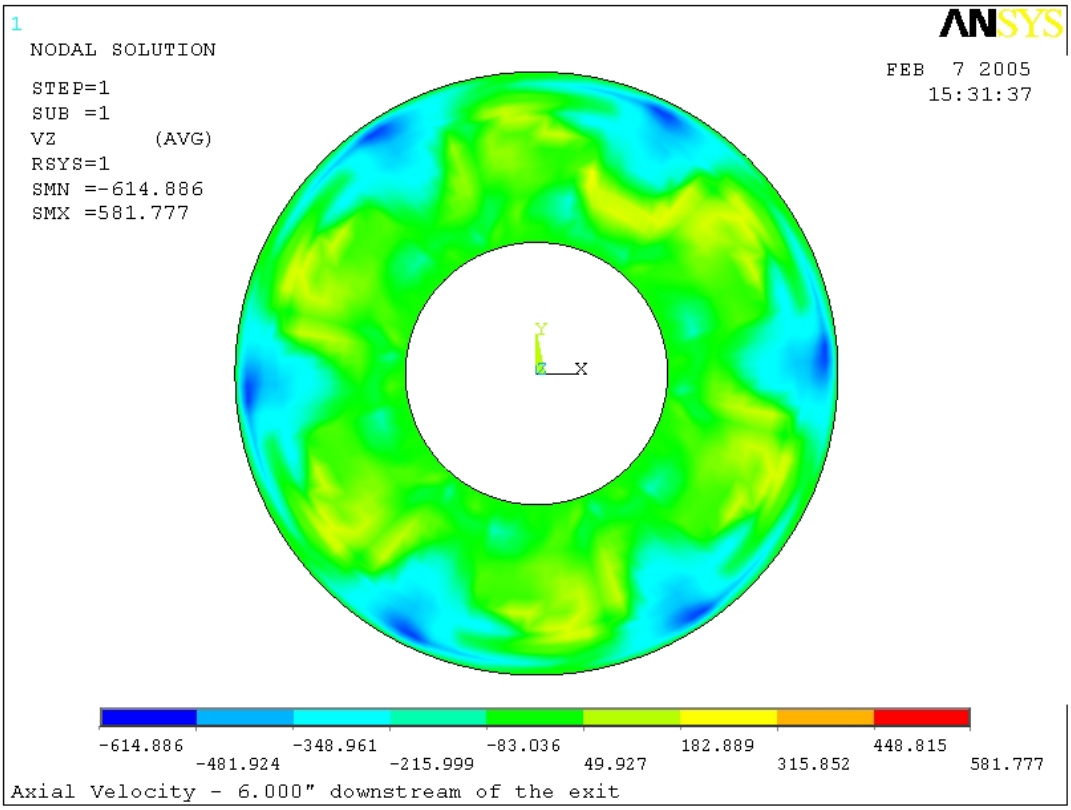

1
NODAL SOLUTION
STEP=1
SUB =1
VSUM (AVG)
RSYS=0
SMX =742.585
ANSYS
FEB 7 2005
15:25:46
Y
Z X
0
82.509
165.019
247.528
330.038
412.547
495.057
577.566
660.076
742.585
Total Velocity - 3.000" downstream of the exit

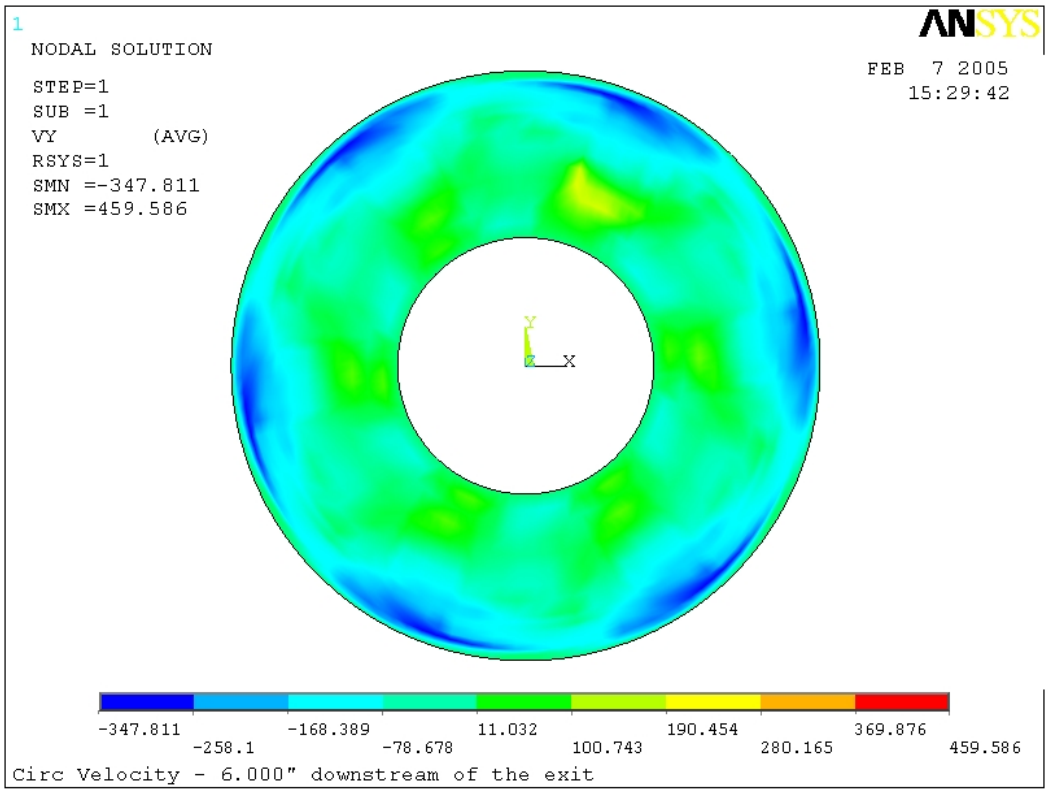

1
NODAL SOLUTION
STEP=1
SUB =1
VZ (AVG)
RSYS=0
SMN =-614.886
SMX =581.777
ANSYS
FEB 7 2005
15:27:17
Y
Z
X
-614.886
-481.924
-348.961
-215.999
-83.036
49.927
182.889
315.852
448.815
581.777
Circ Velocity - 3.000" downstream of the exit

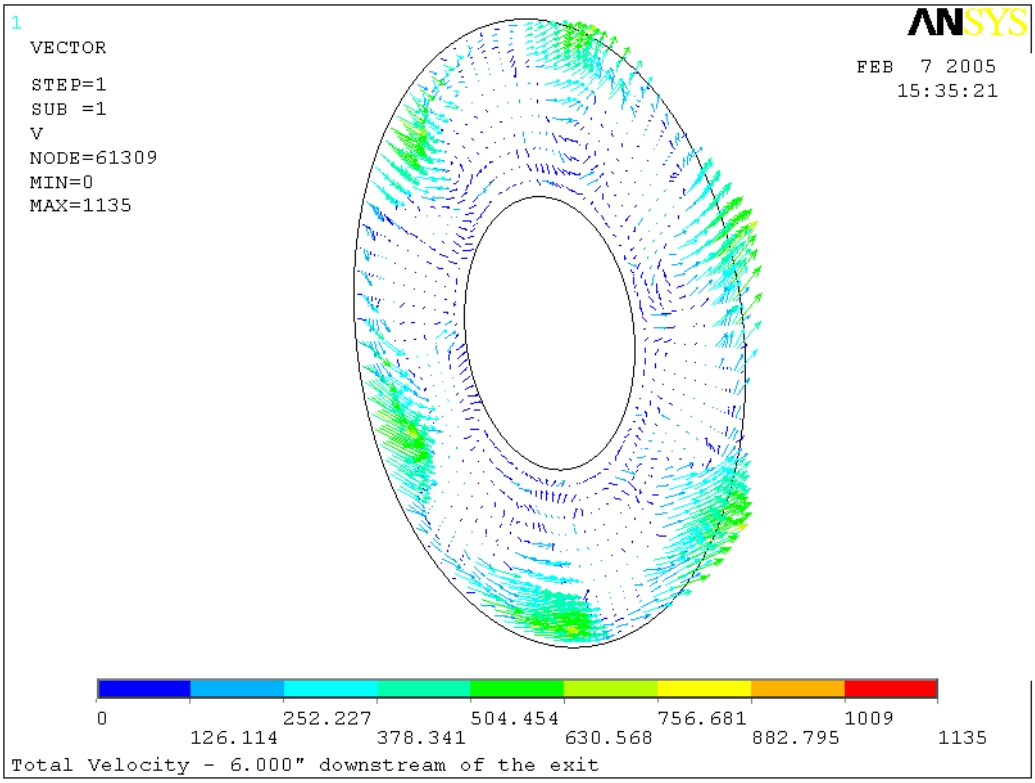

1
VECTOR
STEP=1
SUB =1
V
NODE=61309
MIN=0
MAX=1135
ANSYS
FEB 7 2005
15:23:45
0
126.114
252.227
378.341
504.454
630.568
756.681
882.795
1009
1135
Total Velocity - 3.000" downstream of the exit


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
Z 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
Z 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
Z 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
Z 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