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CFD SIMULATIONS OF TURBODRILL PERFORMANCE WITH ASYMMETRIC STATOR AND ROTOR BLADES CONFIGURATION

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ABSTRACT

Numerical simulations of a down hole turbine motor (Turbodrill) are presented which are optimised for Coiled Tube Turbodrilling in deep hard rocks mineral exploration applications. A Turbodrill is a type of hydraulic axial turbomachinery which has a multistage of stators and rotors. It converts the hydraulic power provided by the drilling fluid to mechanical power through turbine motor while diverting the fluid flow through the stator vanes to the rotor vanes. This paper presents computational fluid dynamics (CFD) simulations of a single stage Turbodrill performance with different rotation speeds and mass flow rates. As a result optimum design parameters are proposed for gaining the required rotation speed and torque for hard rocks drilling.

NOMENCLATURE

c absolute velocity
p pressure
U blade speed
w relative velocity
(x, r, r_θ) cylindrical axes
α absolute velocity angle
β relative velocity angle
ρ density

INTRODUCTION

The authors have recently proposed Coiled Tube (CT) Turbodrilling technology for drilling deep hard rocks for mineral exploration applications (Mokaramian et al., 2012). Drilling small size holes as fast as possible and collecting reliable samples at the surface are the main objectives in mineral exploration drilling. CT drilling can meet these objectives with several advantages over conventional drilling methods.

Coiled tube itself cannot rotate and therefore a down hole motor is needed to provide mechanical power and rotation to the bit. There are many special design criteria to be considered for successful operation of down hole motors in CT drilling (Beaton and Seale, 2004; RIO, 2004; IT, 2007). One major concern is that it is often difficult to produce enough weight on bit (WOB) to maximize the rate of penetration (ROP) for optimised drilling. In an environment where WOB is limited (as with CT drilling); high rotation speed is the key driver for ROP (Beaton and Seale, 2004). Amongst available down hole motors, Turbodrills (turbine motors) are the best choice to be used for small size CT drilling in hard rocks (Mokaramian et al., 2012): this results in a smooth borehole with little vibrational effects during drilling. The turbine motor section has a multistage of stators and rotors which converts the hydraulic power provided by the drilling fluid to mechanical power through the turbine motor while diverting the fluid flow through the stator vanes to the rotor vanes. The drilling fluid here for the purpose of small coiled tube drilling for mineral exploration is considered to be pure water which is pumped from surface pump with different flow rates and pressures.

TURBODRILL CHARACTERISTICS

In general, the down hole turbine motor is composed of two sections: turbine motor section and bearing section, i.e. thrust-bearing and radial support bearing (Eskin and Maurer, 1997). Figure 1 shows a typical Turbodrill assembly and the fluid flow path through turbine stages.

The activating drilling mud or freshwater is pumped at high velocity through the motor section, which, because of the vane angle of each rotor and stator (one stage), causes the rotor to rotate the shaft of the motor which is connected to the bit. The energy required to change the rotational direction of the drilling fluid is transformed into rotational and axial (thrust) forces. This energy transfer is seen as a pressure drop in the drilling fluid. The thrust is typically absorbed by the thrust bearings. The rotational force causes the rotor to rotate relative to the housing. In practice, multiple stages are stacked coaxially until the desired power and torque is achieved.

TURBODRILL DESIGN

When designing a hydraulic multistage turbine, it is assumed that each turbine stage is identical (i.e., that the flow rate, pressure drop, rotary speed, generated torque, When designing a hydraulic multistage turbine, it is assumed that each turbine stage is identical (i.e., that the flow rate, pressure drop, rotary speed, generated torque, and power transmitted to the shaft are the same for each of the stages).

The well-known method of building velocity triangles (and polygons) is used when designing the blade unit profile (see Figure 2). This method is useful for visualizing changes in the magnitude and direction of the fluid flow due to its interaction with the blade system. Fluid enters the stator with absolute velocity c_1 at angle α_1 and accelerates to an absolute velocity c_2 at angle α_2. All angles are measured from the axial (x) direction. From the velocity diagram, the rotor inlet relative velocity w_2, at an angle β_2, is found by subtracting, vectorially, the blade speed U from the absolute velocity c_2. The relative flow within the rotor accelerates to velocity w_3 at an angle β_3 at rotor outlet.

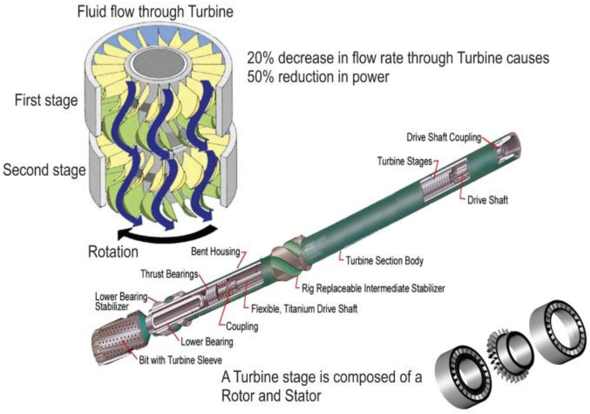


Figure 1: Turbodrill assembly and fluid flow through turbine stages, after (Beaton and Seale, 2004).

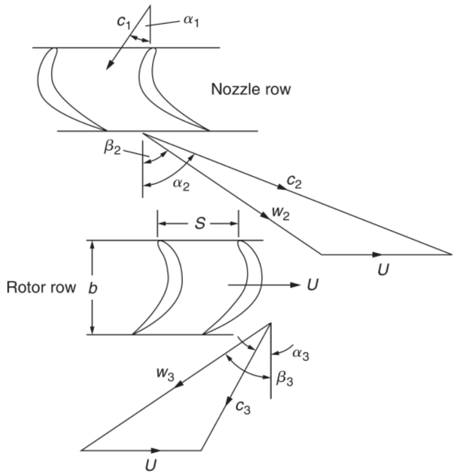


Figure 2: Turbine Stage Velocity Diagrams (Dixon and Hall, 2010).

NUMERICAL SIMULATIONS

Turbodrill is a type of hydraulic axial turbomachinery and while fluid is flowing in the Turbodrill stage, the inlet flow angle and inlet structure angle as well as outlet flow angle and outlet structure angle are different for the stator and rotor blades. The flow angles cannot be directly considered as the blade profile's structure angles, so it is necessary to modify the calculated flow angles from preliminary Turbodrill design methodology based on non-dimensional turbine parameters and velocity polygons, to obtain the structure angles. As a result, through the analysis, in order to improve the performance of the Turbodrill, the flow angles calculated according to the design parameters should be modified to get the structure angles toward reducing flow separation.

Turbodrill design and simulation with ANSYS

TurboSystem is a set of software applications and tools for designing turbomachinery in the ANSYS Workbench environment (ANSYS, Copyright © 2011). In this study some of these tools have been used for Turbodrill design and simulation which include: *ANSYS BladeGen* (a geometry creation tool for turbomachinery blades design), *ANSYS TurboGrid* (a meshing tool for turbomachinery blade rows), *ANSYS CFX* (a general purpose CFD software suite that combines an advanced solver with powerful pre- and post-processing capabilities).

Geometry and Meshing

The shroud and hub (shaft) diameter of the Turbodrill is fixed for the applications of this study as it needs to be utilized by the small size CT for fast drilling. As a result, the shroud (housing) and hub (shaft) diameter were set to be 7 and 3 cm, respectively. Consequently, the spanwise height is 2 cm and there is no shroud tip between the blade tip and housing, i.e. the blades are connected to the housing.

There are two models for geometry generation of the blades in ANSYS BladeGen: Angle/Thickness mode and Pressure/Suction mode, in which the blade profile properties are displayed and defined in various views and blade sections.

After building the geometry of blade's row on the stator and rotor, the geometry data was transferred to the Turbo mesh cell of a TurboGrid system. ANSYS TurboGrid is a powerful tool that creates high quality hexahedral meshes, while preserving the underlying geometry. In this study after a mesh resolution study, a relatively fine mesh resolution of about 5×10^5 numbers of hexahedral elements for this small domain (one blade of stator and one blade of rotor) was generated to obtain high quality output simulation results within reasonable computational time.

Physics Definition

After importing the mesh into the ANSYS CFX-Pre, other elements of the simulation including the boundary conditions (inlets, outlets, etc.) and fluid flow properties are defined. In this study, because of periodic symmetry between blades on the stator row and also on the rotor row and having the same number of blades on each row i.e. pitch ratio of 1, the simulation was conducted for one blade of stator and one blade of rotor interacting with each other. The results show velocity vectors and pressure profiles on the meridional view of the flow passage at 50% span. In simulations of one Turbodrill stage some of the physical and fluid parameters were set as following:

Analysis type: Steady State,

Fluid type: Water (997 Kg/m^3),

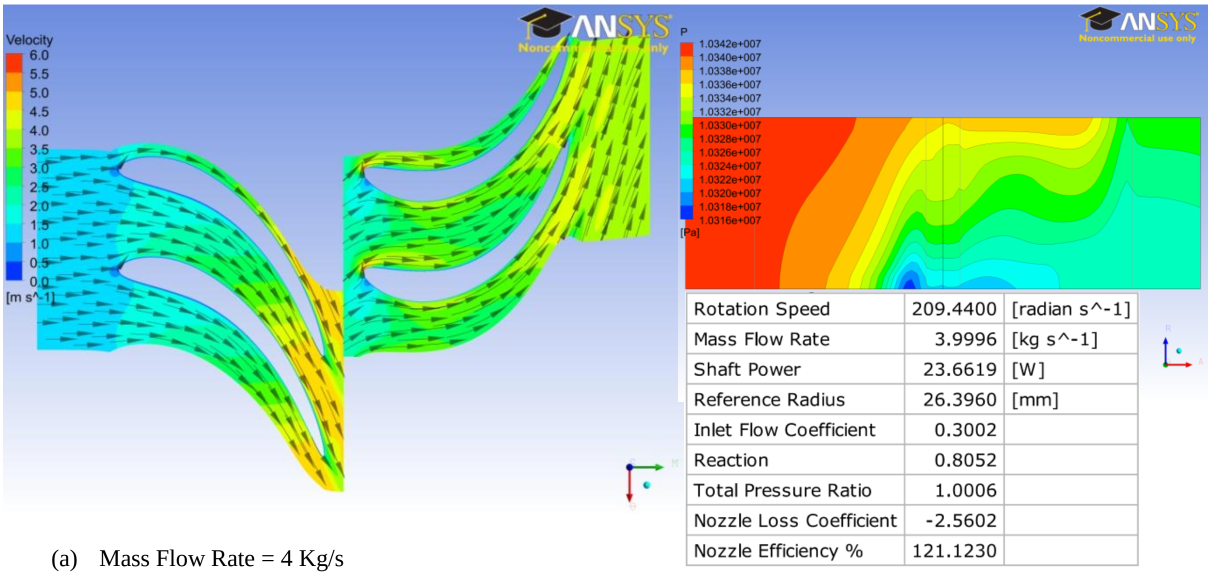
Turbulence model: K-Epsilon,

Boundary conditions: P-Total in inlet and mass flow rate for outlet.

SIMULATION RESULTS

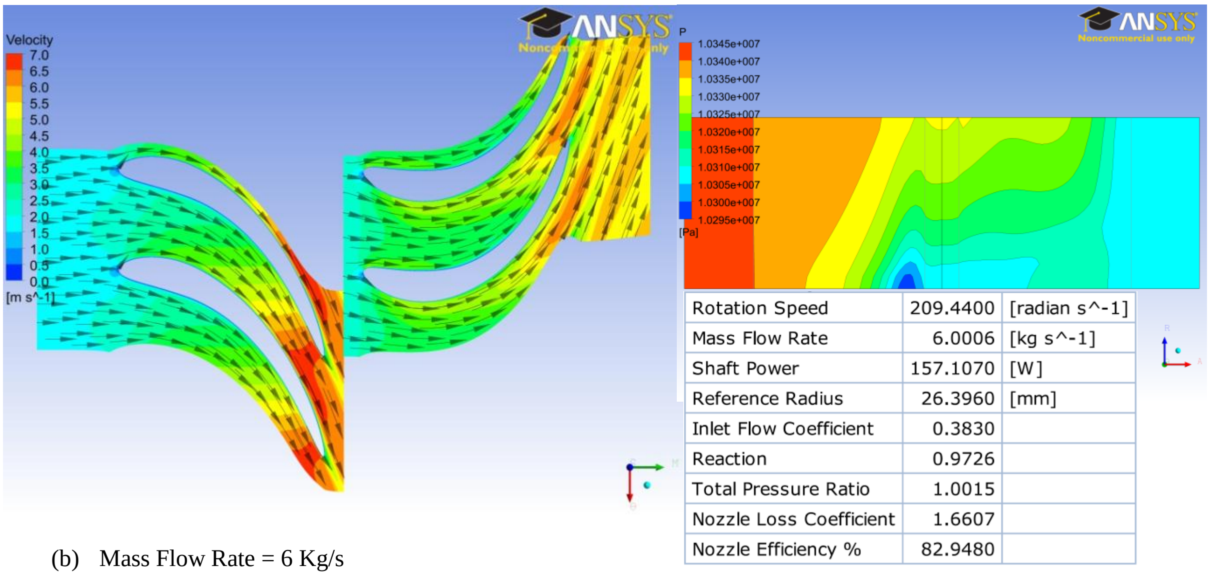
In this study for CFD simulations of a one stage small size Turbodrill used to generate high rotation speed and output power, several blades configurations (inlet and outlet angles, number of blades on each row, etc.) with asymmetric profiles on stator and rotor were simulated. Different flow rates were considered to investigate how velocity vectors, pressure profile, and output power and other performance parameters are affected.

Here, only the results of a few simulations corresponding to one set of blades configuration (16 blades on each row, having highly skewed blades) testing with three different mass flow rates and for three rotation speed of 2,000, 3,000 and 4,000 rpm (revolution per minute) are presented. The simulation results are shown in Figures 3, 4, and 5 for 2,000, 3000, and 4,000 rpm, respectively.



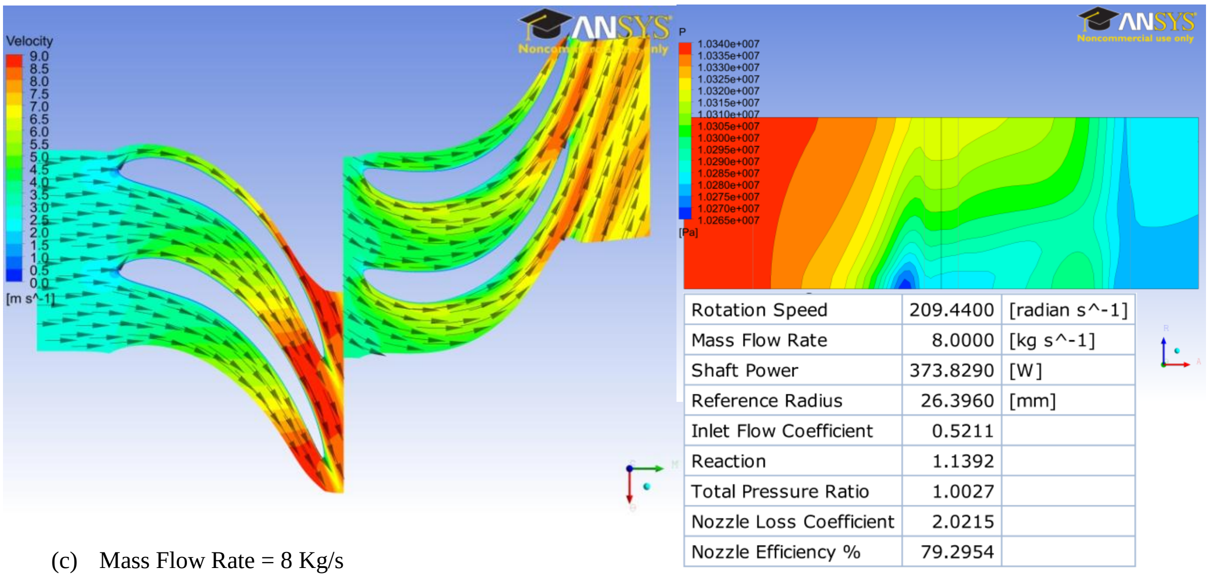
Rotation Speed	209.4400	[radian s^-1]
Mass Flow Rate	3.9996	[kg s^-1]
Shaft Power	23.6619	[W]
Reference Radius	26.3960	[mm]
Inlet Flow Coefficient	0.3002	
Reaction	0.8052	
Total Pressure Ratio	1.0006	
Nozzle Loss Coefficient	-2.5602	
Nozzle Efficiency %	121.1230	

(a) Mass Flow Rate = 4 Kg/s



Rotation Speed	209.4400	[radian s^-1]
Mass Flow Rate	6.0006	[kg s^-1]
Shaft Power	157.1070	[W]
Reference Radius	26.3960	[mm]
Inlet Flow Coefficient	0.3830	
Reaction	0.9726	
Total Pressure Ratio	1.0015	
Nozzle Loss Coefficient	1.6607	
Nozzle Efficiency %	82.9480	

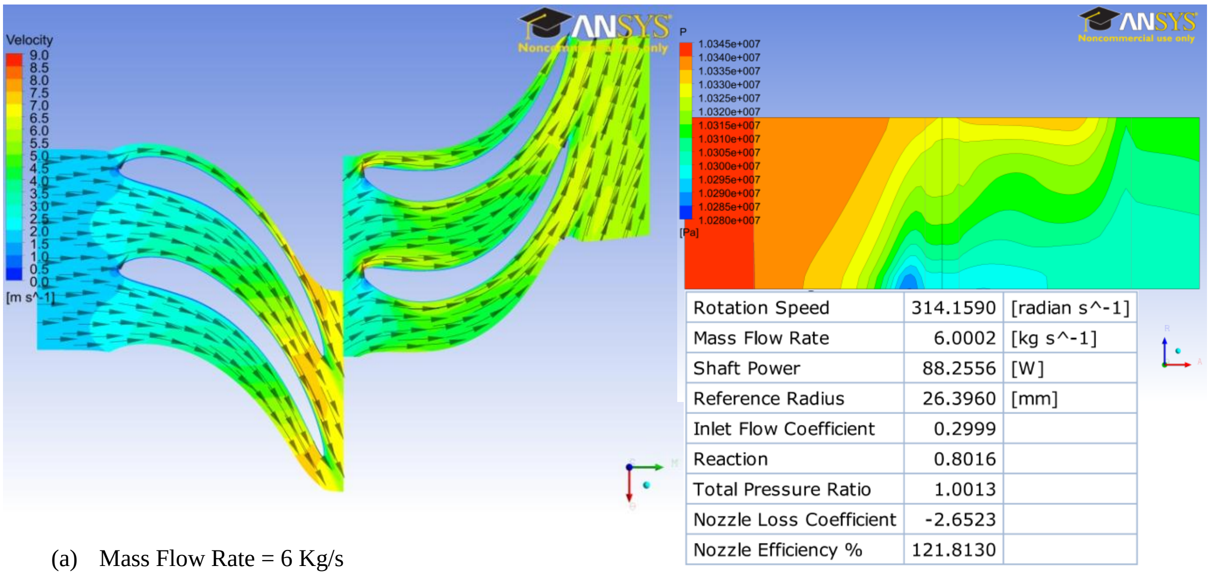
(b) Mass Flow Rate = 6 Kg/s



Rotation Speed	209.4400	[radian s^-1]
Mass Flow Rate	8.0000	[kg s^-1]
Shaft Power	373.8290	[W]
Reference Radius	26.3960	[mm]
Inlet Flow Coefficient	0.5211	
Reaction	1.1392	
Total Pressure Ratio	1.0027	
Nozzle Loss Coefficient	2.0215	
Nozzle Efficiency %	79.2954	

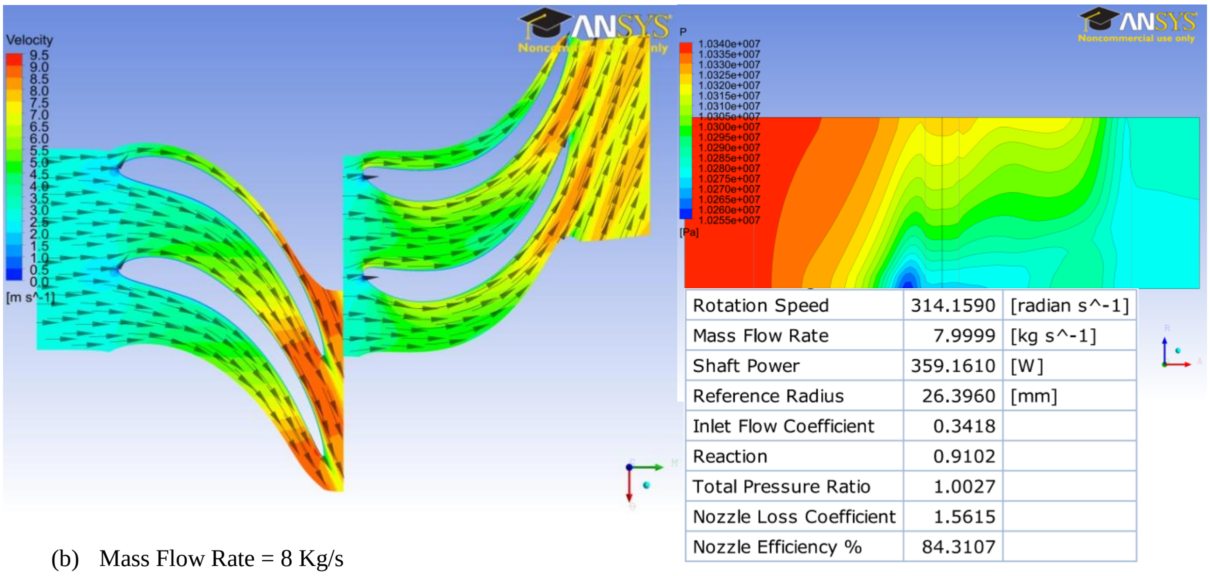
(c) Mass Flow Rate = 8 Kg/s

Figure 3: Turbodrill stage simulation with 2,000 rpm rotation speed for three different mass flow rates of 4, 6, and 8 kg/s.



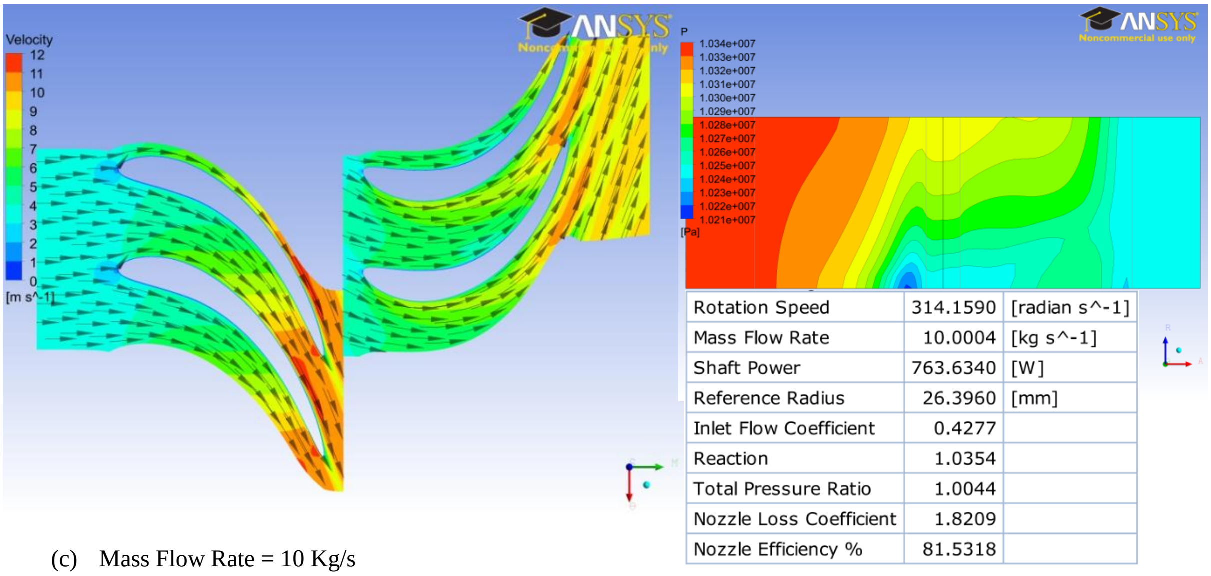
Rotation Speed	314.1590	[radian s^-1]
Mass Flow Rate	6.0002	[kg s^-1]
Shaft Power	88.2556	[W]
Reference Radius	26.3960	[mm]
Inlet Flow Coefficient	0.2999	
Reaction	0.8016	
Total Pressure Ratio	1.0013	
Nozzle Loss Coefficient	-2.6523	
Nozzle Efficiency %	121.8130	

(a) Mass Flow Rate = 6 Kg/s



Rotation Speed	314.1590	[radian s^-1]
Mass Flow Rate	7.9999	[kg s^-1]
Shaft Power	359.1610	[W]
Reference Radius	26.3960	[mm]
Inlet Flow Coefficient	0.3418	
Reaction	0.9102	
Total Pressure Ratio	1.0027	
Nozzle Loss Coefficient	1.5615	
Nozzle Efficiency %	84.3107	

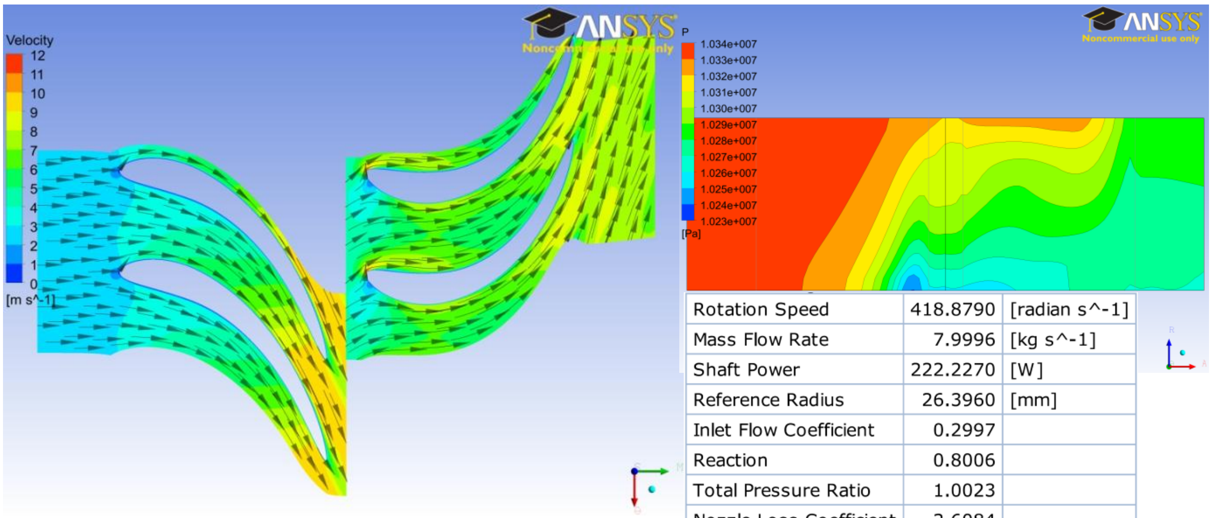
(b) Mass Flow Rate = 8 Kg/s



Rotation Speed	314.1590	[radian s^-1]
Mass Flow Rate	10.0004	[kg s^-1]
Shaft Power	763.6340	[W]
Reference Radius	26.3960	[mm]
Inlet Flow Coefficient	0.4277	
Reaction	1.0354	
Total Pressure Ratio	1.0044	
Nozzle Loss Coefficient	1.8209	
Nozzle Efficiency %	81.5318	

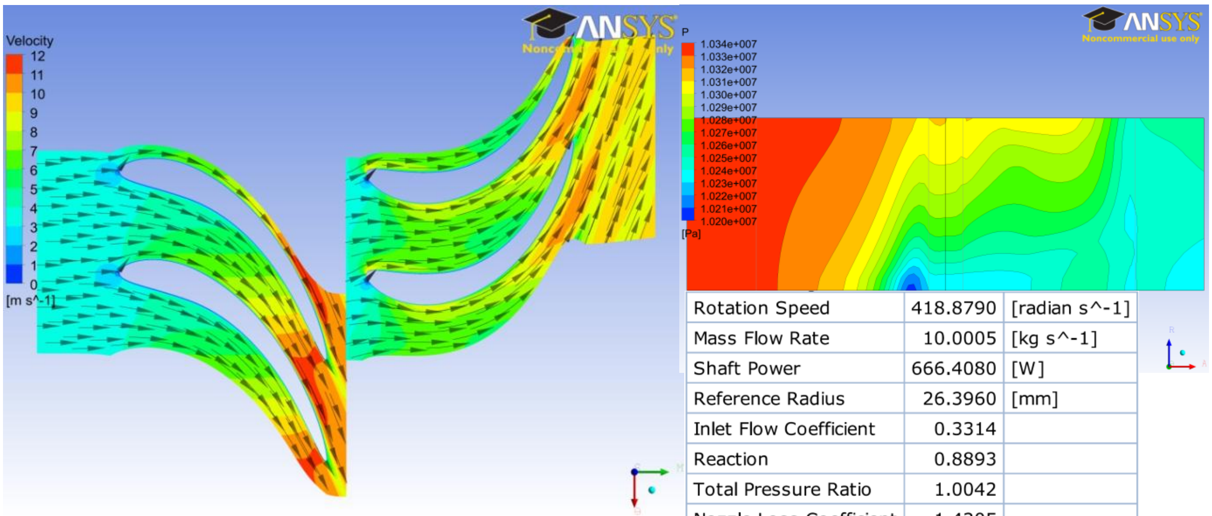
(c) Mass Flow Rate = 10 Kg/s

Figure 4: Turbodrill stage simulation with 3,000 rpm rotation speed for three different mass flow rates of 6, 8, and 10 kg/s.



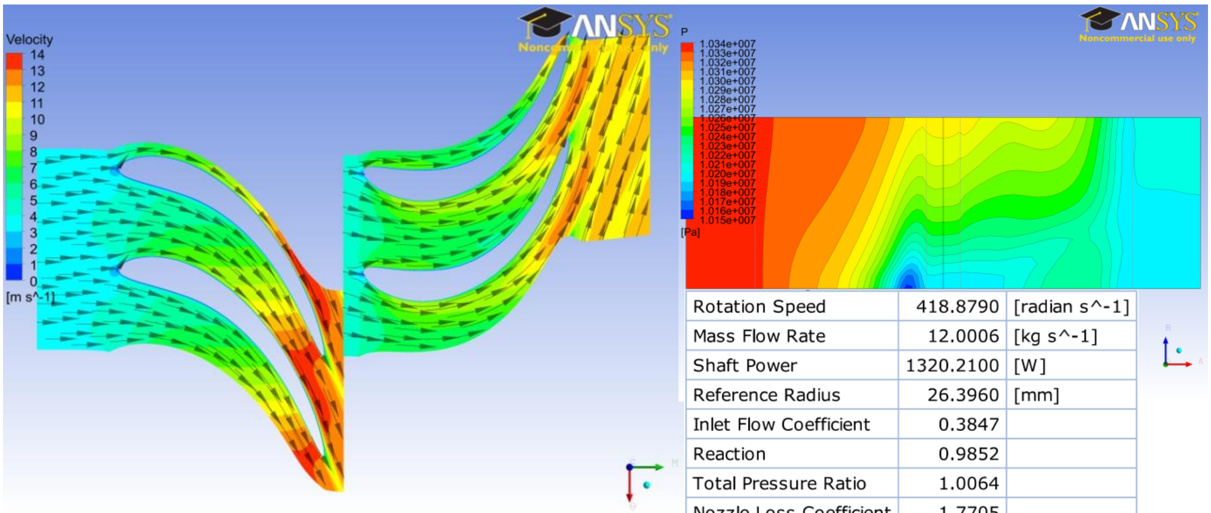
Rotation Speed	418.8790	[radian s^-1]
Mass Flow Rate	7.9996	[kg s^-1]
Shaft Power	222.2270	[W]
Reference Radius	26.3960	[mm]
Inlet Flow Coefficient	0.2997	
Reaction	0.8006	
Total Pressure Ratio	1.0023	
Nozzle Loss Coefficient	-2.6084	
Nozzle Efficiency %	121.2250	

(a) Mass Flow Rate = 8 Kg/s



Rotation Speed	418.8790	[radian s^-1]
Mass Flow Rate	10.0005	[kg s^-1]
Shaft Power	666.4080	[W]
Reference Radius	26.3960	[mm]
Inlet Flow Coefficient	0.3314	
Reaction	0.8893	
Total Pressure Ratio	1.0042	
Nozzle Loss Coefficient	1.4205	
Nozzle Efficiency %	85.9561	

(b) Mass Flow Rate = 10 Kg/s



Rotation Speed	418.8790	[radian s^-1]
Mass Flow Rate	12.0006	[kg s^-1]
Shaft Power	1320.2100	[W]
Reference Radius	26.3960	[mm]
Inlet Flow Coefficient	0.3847	
Reaction	0.9852	
Total Pressure Ratio	1.0064	
Nozzle Loss Coefficient	1.7705	
Nozzle Efficiency %	82.2993	

(c) Mass Flow Rate = 12 Kg/s

Figure 5: Turbodrill stage simulation with 4,000 rpm rotation speed for three different mass flow rates of 8, 10, and 12 kg/s.

Figure 6 shows the simulation results for water mass flow rates of 6 and 8 Kg/s at different rotation speeds. Overall, the simulation results show that the Turbodrill performance is highly dependent on the mass flow rate of the drilling fluid. As the mass flow rate is increased the expected optimum rotation speed, generated power and torque of the Turbodrill are increased. Also, for each specific rotation speed there is a minimum mass flow rate below which, the entering velocity vectors into the rotor are not effective in producing rotational effects on the rotor blades (the velocity vectors are no longer in the right direction). This means that, while the mass flow rate of the drilling fluid reduces, the output rotation speed, the power and consequently shaft torque will drastically decrease to allow the velocity vectors entering rotor be aligned with the rotor inlet angle (to be effective in producing rotational effects on the rotor blades). The effects of low mass flow rates on the velocity vectors entering the rotor can be seen in part (a) of three different rotation speed (i.e. for mass flow rates of 4, 6, and 8 Kg/s with rotation speed of 2,000, 3,000, and 4,000 rpm, respectively).

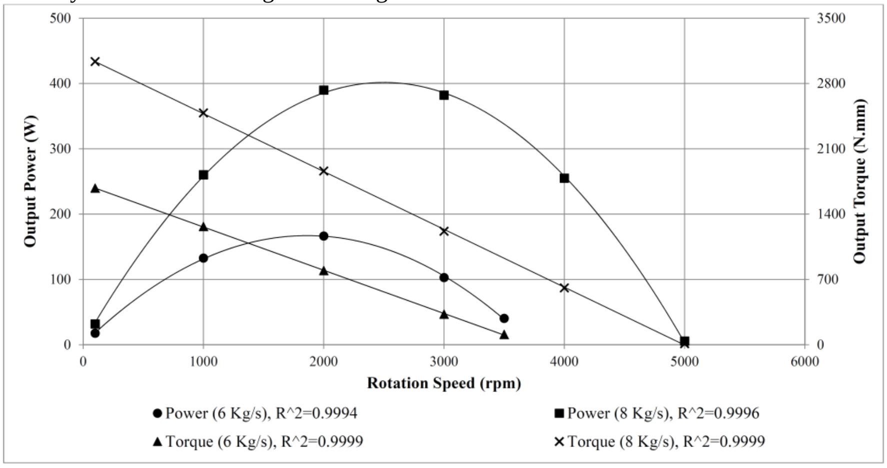


Figure 6: One stage Turbodrill simulation results for water mass flow rates of 6 and 8 Kg/s at different rotation speeds.

In actual field conditions, there are limitations on the mass flow rate that can be pumped from surface down through the CT to bottom hole to actuate the Turbodrill and because of huge pressure losses through drilling assembly, the mass flow rates of higher than 10 Kg/s of water with the specified CT assembly is not applicable. The practical, mass flow rate is about 8 Kg/s and below. As a result, with this Turbodrill specifications (which is highly skewed) and specified CT drilling assembly and its limitations, the optimum rotation speed that can be achieved using water as drilling fluid is less than 4,000 rpm.

CONCLUSIONS

The results of numerical simulations for a Turbodrill stage performance using the ANSYS TurboSystem software applications were presented. CFD simulations of one stage Turbodrill performance with asymmetric blades profiles on stator and rotor were carried out with different mass flow rates and rotation speeds. The simulation results included in this paper for the specific blades configuration showed that the Turbodrill performance is highly dependent on the mass flow rate of the drilling fluid, i.e. as the mass flow rate increases the expected optimum rotation speed of the Turbodrill and consequently the output power and torque will increase. The results also indicated that using water as drilling fluid, with the presented Turbodrill specifications the maximum optimum Turbodrill rotation speed cannot exceed 4,000 rpm in practical field applications for the small size CT drilling.

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