Computational Fluid Dynamics Modeling and Verification of a Submersible Pump-Turbine Runner

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1 National Renewable Energy Laboratory
2 Obermeyer Hydro, Inc.

Presented at HydroVision International
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Computational Fluid Dynamics Modeling and Verification of a Submersible Pump-Turbine Runner

For presentation at Hydrovision International July 23-25, 2019, Portland, OR

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Abstract:

An innovative reversible pump-turbine runner with a 180-degree flow direction reversal has been designed and optimized using computational fluid dynamics (CFD) modeling. The reversible pump-turbine configuration provides a cost-effective means of achieving adequate submergence below tailwater, thereby permitting higher-speed turbine and generator operation and reduced equipment size. Results from CFD modeling and analysis of the hydraulics and runner design are presented. The benchmarks for these parameters are the performance parameters of conventional pump turbines with scroll-case turbine inlets. The model efficiencies were independently verified, and a sensitivity analysis of the results is presented. The results do not reveal a region of pump instability that is characteristic of pumps and pump turbines with scroll cases.

Introduction:

An innovative reversible pump-turbine runner with a 180-degree flow direction change has been designed and optimized using computational fluid dynamics (CFD) modeling tools. Water flows to and from the reversible pump turbine through vertical coaxial conduits above the pump-turbine assembly. The inner conduit is removable and connected to the penstock and upper reservoir. A steel-lined vertical excavated shaft connects to the tailrace and lower reservoir. A “flow inverter” efficiently connects the low-pressure side of the reversible pump turbine to the outer annulus of the vertical water passageway or “well.” The associated motor generator is submersible and located below the pump turbine(s). This configuration allows for a cost-effective method of achieving adequate submergence below tailwater in a conventionally constructed large diameter well. With increased submergence, the turbine and generator equipment size is reduced without the expense of an underground powerhouse.

Description of the Runner:

The runner is analogous to a conventional Francis-type pump-turbine runner (USBR, 1976). The primary difference is that the pump discharge is upwards into an axial diffuser rather than radially outwards through guide vanes and stay vanes into a scroll case. The axial diffuser provides greater design latitude as it can be as long as needed to maintain moderate pressure gradients to suppress diffuser stall.

In the meridional plane, the flow changes direction 180-degrees from straight down into the runner to straight up out of the runner. In a conventional Francis-type pump turbine, the flow has a 90-

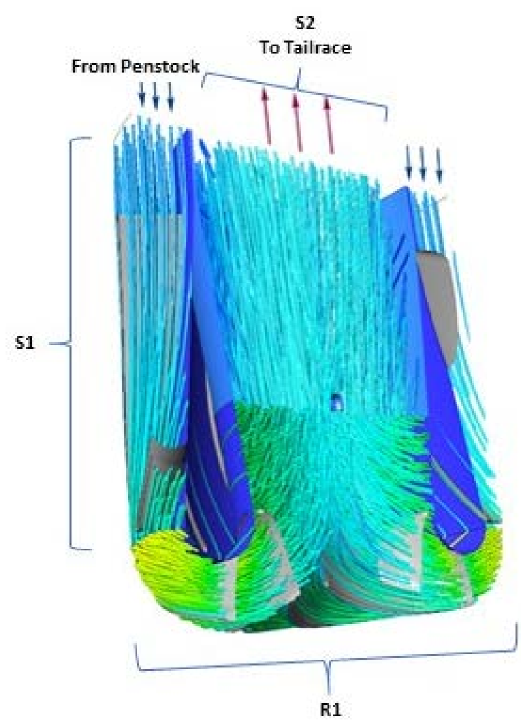
degree change in direction in the meridional plane from axially into the runner (pump mode) to radially out from the runner (turbine mode). CFD results indicate minimal changes in efficiency associated with the 180-degree, rather than 90-degree, change in direction in the meridional plane.

In keeping with Francis turbine nomenclature, we are referring to the runner hub as the crown, despite its position on the bottom of the runner. We are referring to the opposing rotating water passageway element as the band, even though its form is less band-like than in the case of a Francis turbine. Attachment of the runner to the shaft is a splined connection. The splined connection facilitates use of a shaft of minimum diameter, which minimizes the diameter and maximizes the reliability of the outboard and inboard mechanical face seals, respectively. The stationary seal faces of both mechanical shaft seals are mounted to the spherical self-aligning shells of the upper guide-bearing assembly. This ensures that the face seals' stationary faces remain in alignment even during transient events.

Description of the Model:

CFD modeling was completed using ANSYS CFX V.15. Flow through the pump-turbine runner was modeled by the steady-state Reynolds-averaged Navier-Stokes equations (Reynolds, 1895). Turbulence closure was provided by the Spalart-Allmaras model (National Aeronautics and Space Administration, 2019).

Figure 1: Velocity Streamlines



The CFD model consists of three reference frames: two stationary and one rotational. The first stationary reference frame (S1) is the outer annulus providing inflow from the upper reservoir to the runner, including guide vanes. Inflow, in turbine mode, is defined with 300 meters of static pressure head with fully developed flow. The rotational reference frame (R1) consists of the runner rotating about the vertical axis. The circumferential-averaged flow leaves S1 and enters R1. Taking advantage of axis symmetry (as is commonly done in turbomachinery models (Dawes, 1999)), only a single rotor blade and guide vane were modeled. Sliding mesh interfaces were used between the rotating and stationary domains. Flow leaving R1 is again circumferentially averaged flowing into the inner annulus directing outflow to the lower reservoir (S2). The outflow from S2 is defined as 1 atmosphere of average static pressure. Velocity streamlines and pressure distribution are graphically displayed in Figure 1 to show lack of flow separation and regions of excessive shear rate or low pressure.

The following equations were used to calculate changes in pressure head, the turbine mode efficiency, and hydrodynamic power, respectively:

$$Change\ in\ Pressure\ Head\ (Pa): \Delta h = \frac{|\dot{m}_{in} - \dot{m}_{out}|}{\rho g}$$

$$Efficiency: \eta = \frac{\rho \tau \omega}{\dot{m} \Delta h}$$

$$Power: P = \tau \omega$$

Where ρ is the average water density at the inlet, $\dot{m}$ is the mass flow rate, g is the gravitational constant, ω is the rotational speed of the turbine, Δh is the pressure head, and τ is the torque measured in the z direction on the runner blades.

CFD Results:

Initial CFD modeling showed hydraulic efficiency of approximately 95% in pump mode and 94.5% in turbine mode. Tables 1–3 and Figures 2 and 3 show efficiency results over a range of speed and flow rates. Each pump turbine would be designed to optimize speed, flow, and efficiency for site-specific conditions. The best efficiency point for the modeled conditions was empirically found at a rotational speed of 850 rpm.

Table 1: Pump Turbine Characteristics

	PUMP MODE	TURBINE MODE
Head [m]	300	300
Discharge [m³/s]	42.3	37.0
Speed [rpm]	741.6	857.7
Shaft Power [MW]	111.0	106.1
Best Efficiency [%]	95.168	94.551
$\sigma_{critical\ estimated}$	0.19	0.23

Table 2: Turbine Mode CFD Modeling Results

TURBINE MODE (600 rpm)				
Tip speed	**Mass flow** [kg/s]	**Shaft Power** [MW]	**Hydraulic Efficiency** [%]	**Head** [m]
1.024	43000	89.919	85.818	294.980
1.069	42000	84.698	87.464	270.940
1.105	41000	78.383	89.728	253.310
1.139	40000	73.066	91.241	238.540
1.174	39000	67.831	92.268	224.410
1.206	38000	63.453	92.857	212.890
1.238	37000	58.890	93.628	201.840
1.269	36000	54.713	94.254	192.240
1.299	*35000*	*50.768*	*94.551*	*183.490*
1.328	34000	47.042	94.447	175.570
1.361	33000	43.363	93.816	167.090
1.394	32000	39.714	92.821	159.260
1.428	31000	36.104	91.162	151.880
1.470	30000	32.222	89.250	143.190

Table 3: Pump Mode CFD Modeling Results

PUMP MODE (777 rpm)						
Tip speed	Mass flow [kg/s]	Shaft Power [MW]	Hydraulic Efficiency [%]	Total Efficiency [%]	Torque [J, N*m]	Head rise [m]
1.657	46000	92.275	92.693	90.629	1.13E+06	189.09
1.598	45000	96.086	93.57	91.486	1.18E+06	203.18
1.546	44000	99.656	94.227	92.128	1.22E+06	217.03
1.501	43000	102.94	94.678	92.568	1.27E+06	230.48
1.460	42000	105.89	95.002	92.886	1.30E+06	243.57
1.424	41000	108.52	95.157	93.037	1.33E+06	256.13
1.390	*40000*	*111*	*95.168*	*93.048*	*1.36E+06*	*268.56*
1.360	39000	113.28	94.973	92.858	1.39E+06	280.52
1.332	38000	115.84	94.444	92.34	1.42E+06	292.76
1.309	37000	117.25	94.005	91.911	1.44E+06	302.92
1.286	36000	118.96	93.454	91.372	1.46E+06	314.04
1.264	35000	120.2	92.987	90.916	1.48E+06	324.76
1.245	34000	121.08	92.435	90.376	1.49E+06	334.74
1.228	33000	121.57	91.892	89.845	1.49E+06	344.25

Figure 2: Turbine Mode CFD Modeling Results

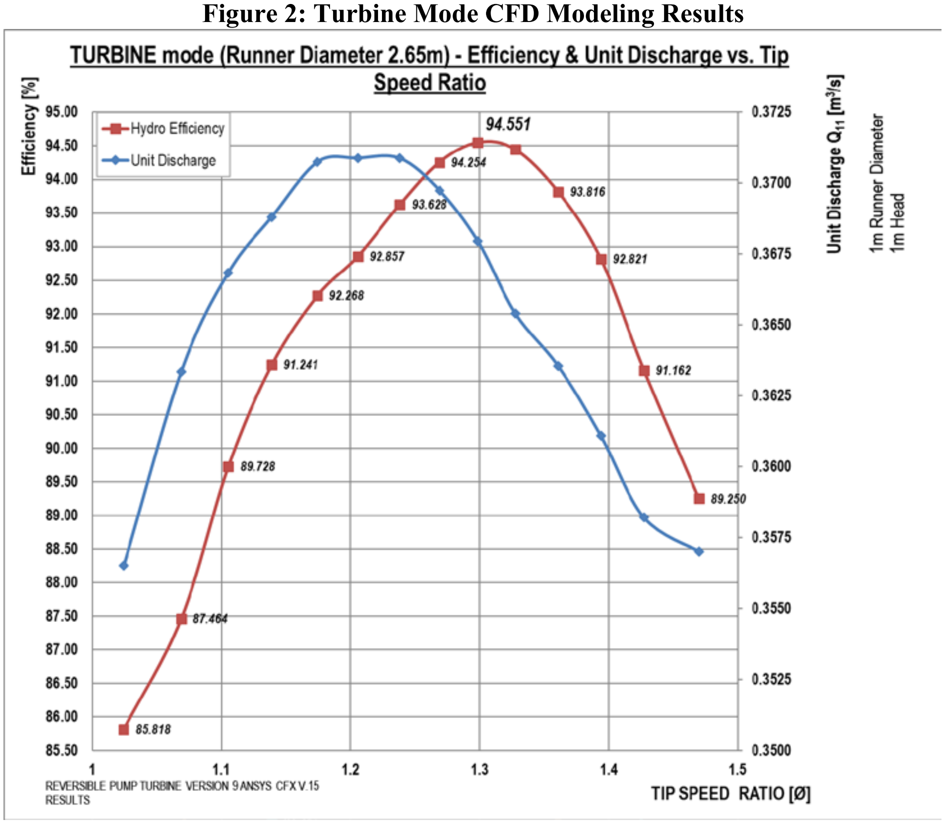
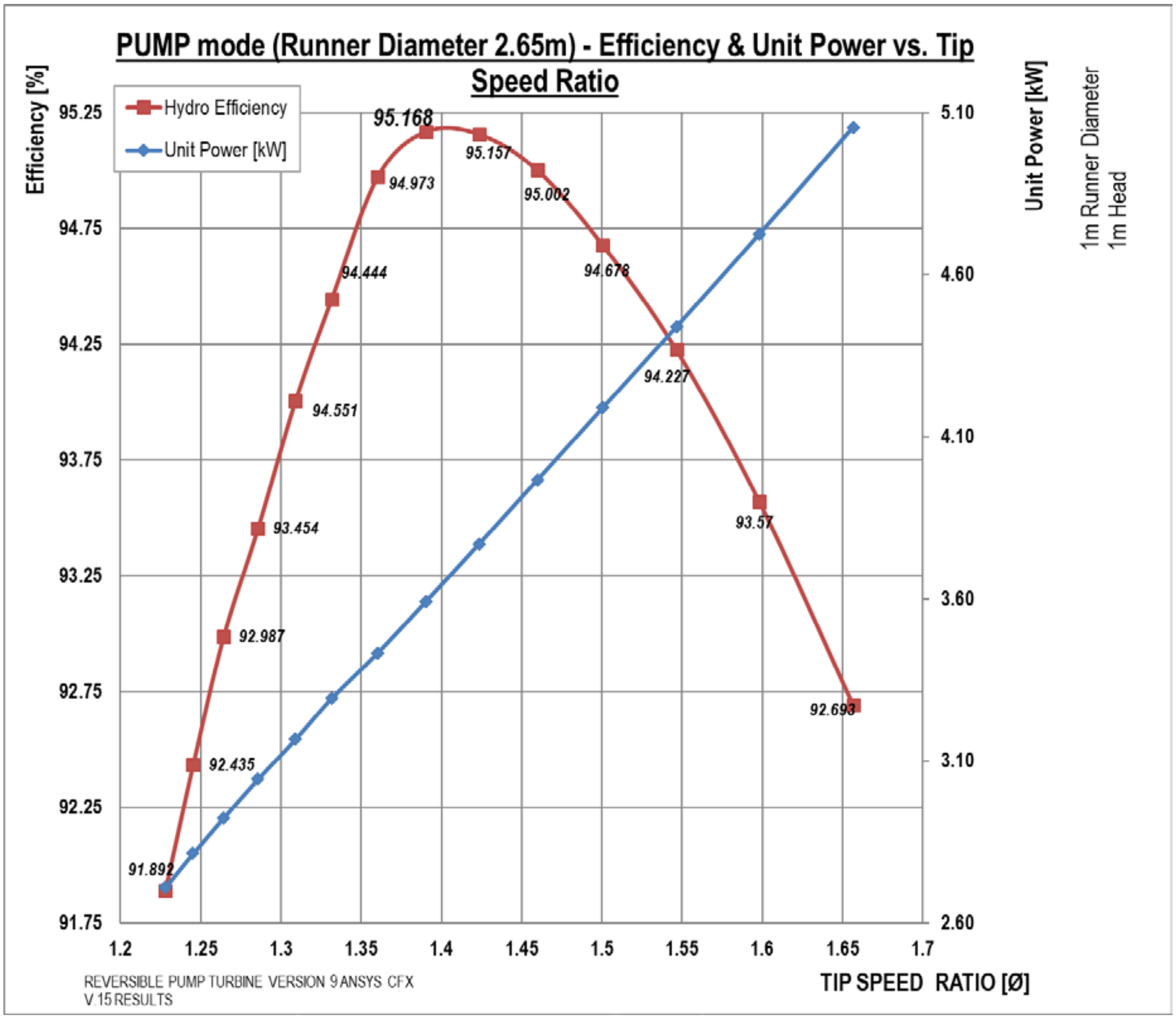


Figure 3: Pump Mode CFD Modeling Results



These efficiencies are equivalent or higher than traditional scroll-case pump turbines. To verify the CFD modeling effort, a model sensitivity study was completed by the National Renewable Energy Laboratory.

Independent CFD Verification:

Several parameters in the CFD model were adjusted to see the impact on CFD efficiency results. An initial mesh convergence study was done, and 2.9 million nodes were found to adequately capture the dynamics of the system using efficiency as a convergence metric. Approximately 1000 CPU hours were needed for RMS residual convergence to 1e-5 in steady-state analyses. Turbulence models, mesh interconnections, and domain interface definitions were modified in order to test model robustness. The impact on efficiency due to these changes is shown in Table 4.

Table 4: Sensitivity Analysis Results

Changes Made to Original Model	Resulting Final Efficiency
No Change – Base Case 850 rpm	94.31%
Outflow Pressure Averaging Changed to Circumferential Averaging	94.32%
No Mesh Intersection Allowed	94.31%
Turbulence Model Changed to Shear Stress Transport with Curvature Correction	93.93%
Turbulence Model Changed to k-omega	93.56%
Turbulence Model Changed to k-omega with Curvature Correction	93.53%
Turbulence Model Changed to k-epsilon	94.31%
Rotation Rate Changed to 800 rpm	94.20%
Rotation Rate Changed to 900 rpm	92.08%

Additionally, a transient analysis was performed to verify the lack of flow separation seen in the steady-state results. The analysis was done over four full rotations of the turbine using one-degree time steps. The transient analysis results suggested hydrodynamics that are not strongly unstable and predicted efficiencies of approximately 94.3%, similar to the steady-state analysis.

Conclusions:

This work has presented the results of a CFD analysis for an innovative pump-turbine design. This design is similar to a Francis turbine, but employs a 180-degree flow direction change. This fundamental system change allows for increased submergence, higher operation speeds, smaller equipment size, and subsequent reductions in overall cost.

Initial CFD modeling results show stable and consistent flow through the turbine. Optimized operating conditions were found empirically for arbitrary site conditions, though they would be changed dependent on installation location. Pump efficiencies were initially found to be approximately 95%, while turbine efficiencies were found to be around 94.5%. Independent verification of the turbine-mode efficiency was conducted at the National Renewable Energy Laboratory. Specific parameters concerning the turbulence of the flow, and model interface connections were changed to test the robustness of the design to modeling changes. For all of the investigated steady-state model simulations, efficiencies were found to be greater than 93%. A transient analysis showed stable flow throughout transient operation, and further supports turbine-mode efficiency values of greater than 93%.

The results of this turbine model design and CFD analysis present a novel concept for a hydrodynamic pump turbine. The presented design affords decreased cost of energy production, without a loss in efficiency as compared to the modern state-of-the-art (Walters, 1977).

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Authors' Bios:

Nikhar Abbas: Nikhar is a Ph.D. student at the University of Colorado, Boulder. He is currently working with the National Renewable Energy Laboratory on control of fixed and floating offshore wind turbines. He holds an M.S. in mechanical engineering and a B.S. in environmental engineering, both from the University of California, San Diego.

Eliot Quon, Ph.D.: Eliot is a researcher at the National Renewable Energy Laboratory. His research includes high-fidelity modeling of the atmospheric boundary layer, wind plants, and marine and hydrokinetic devices. He is also developing improved engineering models of wind turbine wakes and wake interactions.

Claudiu Iavornic, Ph.D.: Claudiu is an engineer with expertise in hydrodynamic machinery research, mechanical design, as well as computational simulations. He was involved in a multitude of large-scale engineering projects and holds a number of U.S. patents in the field of hydroturbines.

Henry Obermeyer: Henry is a graduate of the Colorado School of Mines and is a registered professional engineer. He holds a number of patents for improvements in water control gates and hydroturbines.