



Hydro Green Energy®

Final Technical Report

HYDRO GREEN ENERGY

**DEVELOPMENT OF NEW, LOW-HEAD HYDROPOWER TURBINE – MODELING &
LABORATORY TEST**

EERE AWARD: DE-EE0005426

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List of Acronyms

HGE – Hydro Green Energy

MSI – Mechanical Solutions Incorporated

HDR – HDR Engineering Consultants Incorporated

USACE – United States Army Corps of Engineers

BEP – Best Efficiency Point

DOE – Department of Energy

EERE – Department of Energy Efficiency and Renewable Energy

MBT – Modular Bulb Turbine™

FIT – Feed in Tariff

CFD – Computational Fluid Dynamics

FEA – Finite Element Analysis

CAD – Computer Aided Design

ASME – American Society of Mechanical Engineers

STLE - Society of Tribologists and Lubrication Engineers

IEC – International Electromechanical Code

BOM – Bill of Materials

PTC – Performance Test Code

psi – Pounds per Square Inch

psia – Pounds per Square Inch Atmospheric

Lbm/s – Pounds mass per second

Lbf/in² – Pounds force per square inch

Rpm – Revolutions per minute

Gpm – gallons per minute

Cfs – cubic feet per second

HP – horse power

KW – Kilowatts

MW – Megawatts

M - meters

Ft - feet

In - inches

Executive Summary

Hydro Green Energy (HGE) is a privately held hydropower project development company.

Our leadership team holds over 35 years of combined experience in the U.S. power sector, over 15 years in the hydropower sector and has developed more than 10,000 MW of power generation projects.

HGE was the first company in the United States to license, fabricate, install and operate a grid-connected hydrokinetic (zero-head) in-stream power project. HGE won the 2009 Presidents Award from the National Hydropower Association for that project. Our first low-head hydropower project is expected to come on-line in 2018. HGE presently hold the development rights to approximately 250 MW of hydropower capacity at existing non-powered dams.

The objective of this project was that HGE would complete the design, fabrication and laboratory testing of a scaled, vertically stackable, low cost, low-head hydropower turbine called the Modular Bulb Turbine™ (MBT).

The project goals were to design, model and test the modular bulb turbine for installation in numerous HGE low-head hydropower projects at non-powered USACE dams.

The Preliminary Design Criteria of the MBT was: Efficiency = 81-83%; 3-4 Blades; speed ≤ 150 rpm; Design Head = 10 feet; Design Flow 1100-1200 cfs; Generator exit power = 700-750 kW.

The overall Project Concept was to design a Chevy instead of a Ferrari which meant that we wanted a low cost, robust, reliable, acceptable performance instead of a high cost, complex, high maintenance design in a conventional system.

The conclusions from the testing were the following:

- The characteristics of the 16-inch model (~1:6 scale) at the Best Efficiency Point (BEP) prior to the test were estimated as follows:
 - Flow rate: 46 cfs
 - Gross head: 18.4 ft
 - Blade tip diameter: 1.33 ft
 - Rotational speed: 769 RPM
 - Power Estimate: 52 kW
- Alden Lab indoor test loop unable to operate at sub-scale design point therefore the Best Efficiency Point (BEP) was most likely not observed
- Cavitation was observed approaching design point during multiple test runs – tail water level tested at 6 inches up to 24 inches.

- Braddock full scale has higher tail water static pressure – runner setting ~11+ feet below tail water which means that there would be no anticipated cavitation
- Alden Lab tests mostly conducted below design point in the low torque range. Cavitation was observed as the test approached high torque range.
- Synchronous belt that was used is less efficient at below design conditions. Since the test was conducted at below design conditions there was a negative impact to observed efficiency. The belt used was sized for high torque tests which we were not able to observe without cavitation.
- Alden sub-scale best manually recorded/observed efficiency at 0.722 (<50 datas)
- Alden sub-scale digitally recorded average efficiency at 0.702 (10,000 data points)
- Alden projected full scale efficiency at 0.775 (based on sub-scale digital data)
- MSI projects operation at design point (and belt design point) would have yielded as much as another 4-6% efficiency
- Based on this information MSI believes that the full scale BEP would have been ~ 0.805

All members of the team agreed that for a first sub-subscale test the turbine performed significantly better than most sub-scale designs that are tested for a first time. One comment was that some machines break or generate no power at all on their first sub-scale runs. The team believed that the test was a success. After the test it was recommended to the Department of Energy (DOE) to proceed to the full scale design and testing.

Introduction

The Project Objective was that HGE would complete the design, fabrication and laboratory testing of a scaled, vertically stackable, low-head hydropower turbine called the Modular Bulb Turbine™ (MBT).

One of the current challenges in the US hydropower industry is that there no longer are US based turbine manufacturers that design hydraulic turbines for low head projects. Decades ago this was not the case with over ten US based manufacturers. Most of the remaining untapped hydropower potential in the continental US is located at low head sites. Since turbines are a large component of the project costs, reduction in these costs help make a project more economically viable. HGE has permits for many potential low head hydropower sites where existing licenses and designs for conventional hydropower projects had existed in the past but the projects were never built. HGE believes a part of the reason for this is that the required individual customization of conventional hydropower sites does not lend itself well to non-Federal hydropower development. In other words, the federal government can afford to pay a high capital cost for a custom conventional design however, private independent power producers (IPP) and developers have multiple federal laws and restrictions from Public Utility Commissions on what they are allowed to be paid for their power. Thus, the cost of producing the power must be lower.

The Project Goals were to design, model and test the MBT for installation in numerous HGE low-head hydropower projects at non-powered USACE dams.

In order to solve the low head problem the design requirements were:

- Keep it inexpensive, but robust – Chevy vs. Ferrari
- To have a known cost of product for turbine of this size
- Removable/stackable modules – minimum downtime and maximize capacity factor
- Lower Cost even if it means less efficiency
- Minimize capital cost through use of existing infrastructure
- Minimize capital cost through standardization
- Use readily available OEM parts – minimize specials
- Minimize excavation (environmental impact)
- Minimize RPM (fish impact)
- Ease of Assembly & Maintenance

These requirements led HGE to the concept of the MBT. At this stage of laboratory sub-scale testing future commercialization plans depend on a successful full scale test and proven longer term operation of a full scale design. There are also opportunities for additional design optimization which would include ongoing iterations of CFD and FEA work to continue to refine and increase the efficiency of the turbine.

Background

The current state of the art in hydraulic turbine designs is advanced machines from countries that typically have either feed-in-tariffs (FIT) or high power costs. However, in recent years in the US, contracted long term wholesale power prices set by the price of natural gas have been declining and therefore the energy only benefit of a hydropower project is lower than in previous years. There are other benefits besides energy produced at hydropower plants such as capacity, black start, frequency response and other ancillary services just to name a few. It is the belief of some in the US hydropower industry that this situation causes foreign manufacturers of low head hydraulic turbines to not fully understand the US market and misprice their equipment for US projects on the high side. Additionally, there are no longer any US based low head turbine manufacturers.

The Project Objective was that HGE would complete the design, fabrication and laboratory testing of a scaled, vertically stackable, low-head hydropower turbine called the MBT. HGE would also complete a summary report that includes the laboratory testing results and analysis of the tests.

The Project Goals were Design, model and test modular bulb turbine for installation in numerous HGE low-head hydropower projects at non-powered USACE dams.

The project team members were Mechanical Solutions Inc.(MSI) and HDR Engineering Consultants (HDR).

MSI relies on the expertise of its experienced technical staff to perform their portion of the proposed project for DOE effectively. Dr. Edward Bennett, MSI's Director of Fluids Engineering, lead the hydraulics aspects of the project, and William Marscher, P.E., MSI's Technical Director, both provided oversight and lead the mechanical phases of the project. They were assisted in their efforts by additional qualified MSI engineers, e.g. Principal Engineer Paul Boyadjis.

Prior to joining MSI, Dr. Bennett was the Director of TurboPump Assembly Design and Analysis at Aerojet Inc. He also was the founder of ASC Turbo (now part of ANSYS/ CFX) and worked for Fluent, Inc. in the past as a consulting engineer. Dr. Bennett recently was selected as the Associate Technical Editor for the Journal of Fluids Engineering published by ASME, and in August 2009 the ASME Fluids Engineering Division awarded Dr. Bennett with the Gopalakrishnan-Flowserve Pump Technology Award. Nominees for the award must have demonstrated outstanding achievement in hydraulic machinery technology as documented through publications and the testimonials of their peers and co-workers.

Bill Marscher was past Chief Mechanical Engineer for Worthington and Dresser Pump Divisional Engineering. He is known for successfully troubleshooting flow-induced and other fluid machinery vibration problems and has over 40 years of experience in the design, analysis,

and evaluation of fluid machinery. Mr. Marscher is a recognized expert in the area of hydraulic machinery mechanical design, as well as vibration analysis and testing. He regularly delivers the vibration and rotordynamics short courses at the Texas A&M Pump Symposium, he was co-author of the vibration chapter of Sank's Pumping Station Design Handbook, 2nd Edition, is co-author of the book Centrifugal Pump Design & Performance, published by Oxford University Press, and also wrote the Lubrication chapter of the Modern Marine Engineer's Manual, 2nd Edition. He is a past board chair for the Machinery Failure Prevention Society (a division of the Vibration Institute) and a past President of the Society of Tribologists and Lubrication Engineers (STLE).

Mr. Boyadjis is past Head of Analytical Engineering for Ingersoll-Rand (and later IDP) in Phillipsburg, NJ, and he graduated Magna Cum Laude from Lehigh University 20 years ago. Mr. Boyadjis pioneered the use of P-Type finite element analysis in fluid machinery casings and internal components. In his years at MSI, he has analyzed fluid machinery and their systems in a variety of challenging applications, analyzing them "up-front" as well as successfully evaluating and fixing problems in the field. Applications have included all types of fluid machinery including vertical turbine pumps, high-flow axial pumps, hydraulic turbines, boiler feed pumps and rocket turbopumps, as well as multi-stage centrifugal compressors and turbines. In addition to his finite element and machinery mechanical expertise, Mr. Boyadjis is an expert in the Pro/Engineer solids modeling program, which is likely to be used in this project.

Mr. Kelly is a professional engineer with 25 years of diverse engineering work experience. Currently working as a structural analyst for MSI, he has been involved with a variety of rotating equipment applications. His areas of expertise are dynamics, vibration, and stress analysis, and he has performed both theoretical and field troubleshooting work. Recent projects which he has led include the vibration analysis of floodwater pumping stations for the City of New Orleans, and a large concrete subsurface wastewater tunnel / dewatering pumping station. A licensed professional engineer in the State of New Jersey, Bill has a B.S degree in Engineering Science from the Pennsylvania State University, and an M.S. degree in Mechanical Engineering from Florida Atlantic University. He has authored multiple machinery-related technical papers as well as a portion of the Pump Handbook, 4th Edition.

HDR is a global engineering consulting firm. HDR specializes in engineering, architecture, environmental and construction services. While HDR is most well-known for adding beauty and structure to communities through high-performance buildings and smart infrastructure, we provide much more than that. We create an unshakable foundation for progress because our multidisciplinary teams also include scientists, economists, builders, analysts and artists.

Alden Research Laboratories has been a recognized leader in the field of applied fluid dynamic consulting. We use this expertise to support the World's environmental and energy future through independent evaluation and optimization of new and existing technologies; providing practical value-added solutions to industry and government agencies; delivering a quality product that meets or exceeds our client's expectations; developing staff to be company and industry leaders; and using our staff's expertise to help shape regulations that are relevant to our clients.

During the proposed Sub-topic 1.1 project, MSI performed the hydraulic design and analysis of the proposed 2.35 m runner diameter MBT hydro-turbine using advanced CFD software and techniques. MSI also performed a mechanical analysis of the impeller that included the application of advanced FEA and rotordynamic code, and selected the optimum materials from which to manufacture the turbine. Additionally, MSI performed a CFD analysis of the scale model of the hydro-turbine that was constructed and evaluated during this project, compared the two sets of analytical data, and performed an iteration of the CFD analysis of the scale model turbine.

Throughout this project, MSI both coordinated their efforts and interacted closely with the HGE team to ensure that the best design is achieved in an efficient manner. Once the preliminary design was realized, appropriate descriptive plots which included but were not limited to pressure gradient profiles, pressure drop across different parts of the MBT, velocity streamlines, velocity vector graphs and x-y graphs of speed, power and efficiency were sent and a teleconference was conducted to discuss the design with the HGE team.

Alden Research Laboratory, Inc. (Alden) is contracted by Hydro Green Energy, LLC (HGE) to conduct engineering performance tests of a new turbine designed specifically for low head applications. This work is funded by a grant HGE received from the Department of Energy (DOE) Funding Opportunity Announcement for Advanced Hydropower Development (DE-FOA-0000486), Topic Area 1.1. Hydro Green has assembled a team of experienced hydro industry experts which includes Mechanical Solutions Inc. (MSI) and HDR Engineering (HDR) to assist with the development and testing of this new innovative turbine. MSI's role is to design the hydraulic and mechanical turbine components, while HDR's role is to provide general oversight and turbine design expertise for the project. Alden's role is to provide turbine model testing services.

MSI recently completed the hydraulic design of the turbine which was provided to Alden on November 7, 2012. With hydraulic design complete, efforts related to preparing for model tests have been initiated, and this study plan is the first step in that process. This study plan outlines scaling from the prototype turbine to a model scale turbine, test configuration, and measurements. HGE should use the final model testing study plan to initiate the design and fabrication of the model scale turbine

Standards

The model scale testing was performed in accordance to the American Society of Mechanical Engineers Performance Test Codes on Hydraulic Turbines (ASME PTC 18-1992), due to the nature of Alden's test facility which utilizes a reservoir and penstock, similar to field conditions. In addition, Alden's standard test procedures, (QMSM-01) were followed. Also, to the extent possible, Alden followed guidelines provided in the International Electrotechnical Commission International Standard (IEC 60193) for model acceptance tests of hydro turbines.

Facility Description

It was initially planned that the model turbine to be tested would be installed at Test Line 3 in the Hooper Low Reynolds Number Facility (also known as building No. 2) shown on Figure 1 and Figure 2. Water for the facility is supplied from a 150-acre reservoir through a 40-inch penstock

that provides a static head of up to 28 feet. The facility has a maximum flow capacity of approximately 80 cfs and a permanently installed Venturi flow meter. Head and flow can be adjusted using valves installed in the test line.

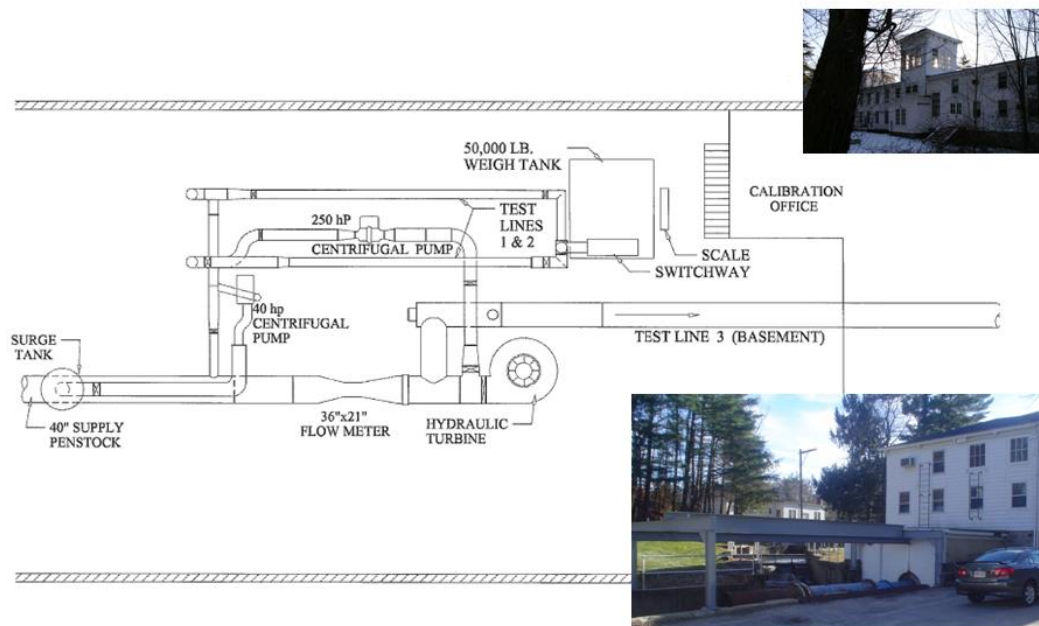


Figure 1: Plan View and Photographs of Alden's Hooper Test Facility

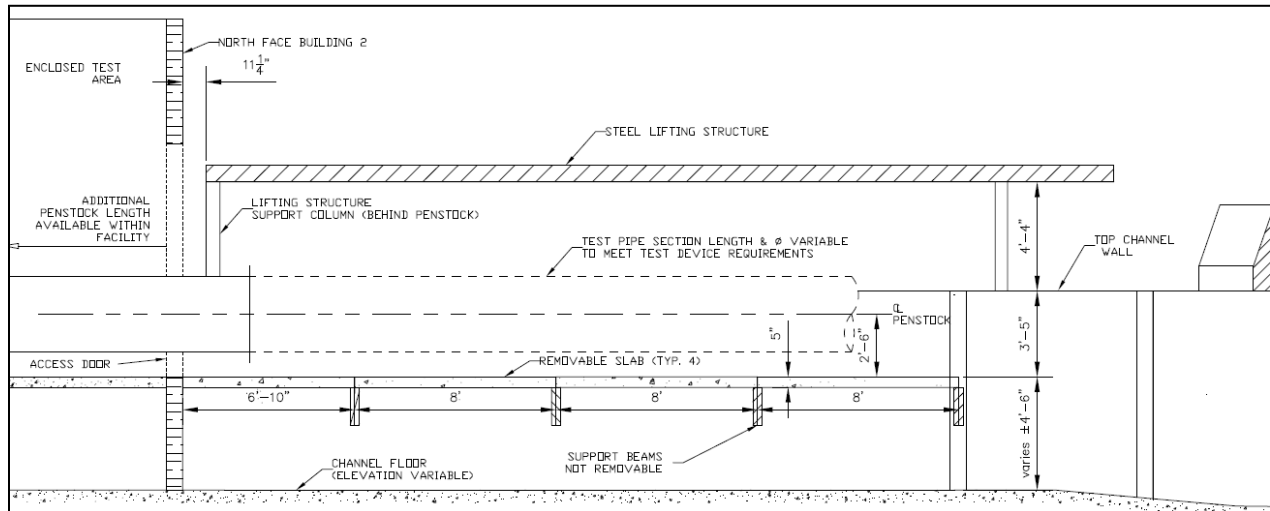


Figure 2: Cross Section of Testing Area

At the downstream terminus of the Hooper test facility enclosure, the Test Line 3 pathway continues outside as shown on Figure 2. Outside of the building, the piping is located in an engineered channel recessed into the surrounding topography. The channel continues downstream before discharging back into the natural stream channel. Approximately 35 ft downstream of the building, a vehicular bridge crosses over the channel, which effectively limits component installation further downstream. An overhead lifting gantry is installed over the majority of the outdoor piping area to assist in installation of both piping elements and turbine

components. Test Line 3 is composed of many individual piping sections (see Figure 1) with upstream water control. This provides flexibility to modify the configuration and components (centerline, diameter, etc...) of the water supply to the turbine, if required.

The existing test loop has a permanently installed 36-inch by 21-inch master Venturi meter for the purposes of measuring flow rate. This meter is calibrated and would have been used for flow measurement during testing. All other required instrumentation would have been installed specifically for the HGE model turbine testing.

Model Scaling

The prototype turbine design provided by MSI was used as a basis for model scaling. HDR recommended a 16-inch diameter model turbine which satisfies the minimum diameter of 11.8 inches for axial flow turbines recommended by IEC 60193. Based on this recommendation and prototype characteristics, the minimum flow rate required to achieve an acceptable Reynolds number was calculated. Following evaluation of Reynolds number, the prototype turbine characteristics were scaled using affinity laws to determine the model design conditions.

The prototype turbine geometry provided by MSI is shown on Figure 3. MSI also provided the following prototype turbine design information:

- Flow rate: 37 cms (1,307 cfs)
- Gross head: 10.4 ft
- Blade tip diameter: 8.2 ft (2.5 m)
- Rotational speed: 94 RPM
- Power estimate: 839 kW
- Estimated efficiency: 71%

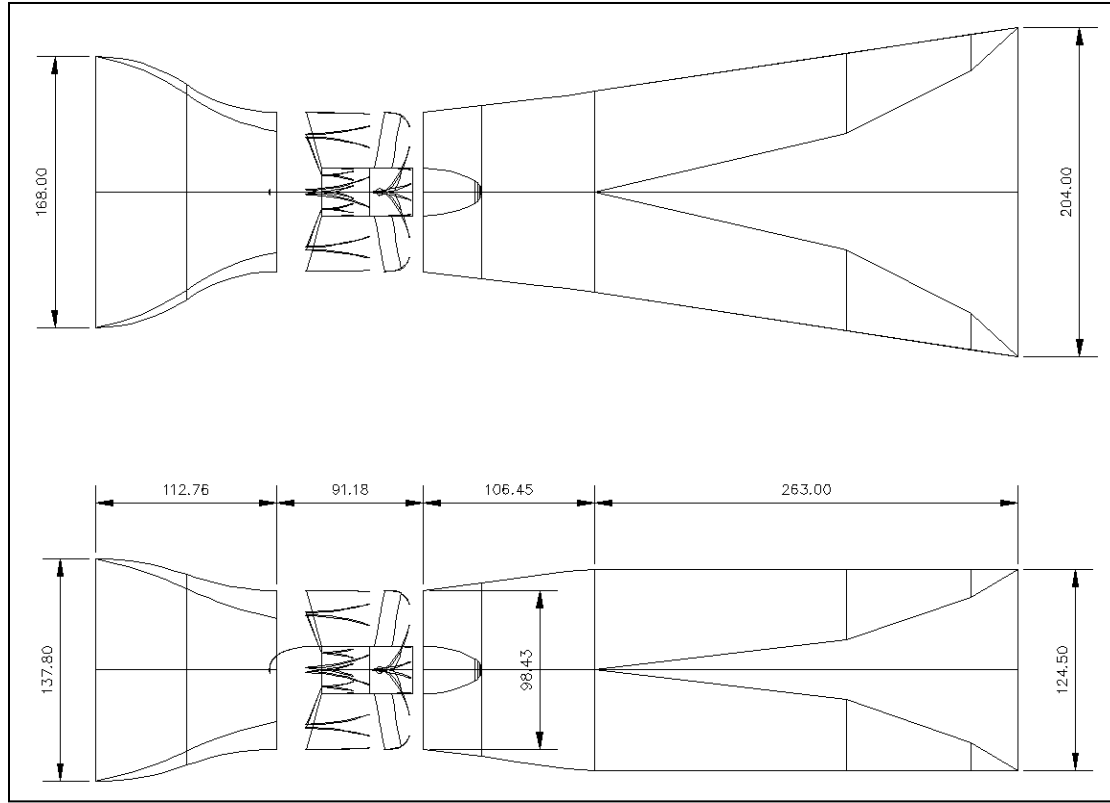


Figure 3: Section and plan view of prototype geometry (provided by MSI November 6, 2012)

Equation 1 illustrates the formula for Reynolds number calculation. Using Equation 1, and the minimum recommended Reynolds number (4×10^6), the minimum required flow rate for a 16-inch diameter model was found to be 46 cfs.

$$Re = \frac{Du}{\nu} \quad (1)$$

Where:

Re = Reynolds Number

D = Reference Diameter (defined in IEC60193)

u = Peripheral Velocity at Reference Diameter

With a known diameter and flow rate, the affinity laws (Equations 2, 3 and 4) were utilized to estimate the remaining model characteristics.

$$\frac{Q_p}{n_p D_p^3} = \frac{Q_m}{n_m D_m^3} \quad (2)$$

$$\frac{n_p D_p}{\sqrt{h_p}} = \frac{n_m D_m}{\sqrt{h_m}} \quad (3)$$

$$\frac{P_p}{n_p^3 D_p^5} = \frac{P_m}{n_m^3 D_m^5} \quad (4)$$

Where p denotes prototype and m denotes the model and:

P = power

h = head

n = rotational speed

Q = flow rate

D = diameter

Using the affinity equations, characteristics of the 16-inch model (~1:6 scale) have been estimated as follows:

- Flow rate: 46 cfs
- Gross head: 18.4 ft
- Blade tip diameter: 1.33 ft
- Rotational speed: 769 RPM
- Power Estimate: 52 kW

The head and flow values for the model turbine are close to the upper range the test facility; therefore, further analysis was required to verify facility performance for the final agreed-upon model turbine test configuration.

Model Design and Fabrication

HGE or other team members are responsible for the turbine model design and fabrication. The model turbine should be designed to accommodate the requirements and recommendations provided in this study plan. Alden assisted HGE to assure the model turbine is designed to accommodate the required instrumentation outlined in this study plan. Initial requirements for the model include the following:

- Design must accommodate required instrumentation. This includes an external shaft designed to accommodate a breaking system, torque measurements, and speed measurements.
- Unit must be self-supporting between the upstream and downstream flow connections

Test Configuration

Ultimately, due to delays and a back end time constraint, the original test loop was not available for the test. At that point, the test was moved to a second indoor test loop at the Alden facility. As discussed later some compromises were unavoidable such a lower back pressure on the turbine due to a shallower testing tailrace. The HGE turbine is a horizontally-oriented unit consisting primarily of an inlet section, turbine section, and a draft tube section. Several design parameters have to be considered to develop the test configuration. These parameters include:

- Approach flow conditions

- Tailwater elevation
- Turbine setting (center line elevation)
- Operational ranges (min/max water levels, head, flow)
- Physical limitations

The turbine inlet section is composed of a square section transitioning to a round section as it approaches the turbine. Therefore, conduit approaching and upstream of the turbine inlet must be square in profile, (see Figure 4). Furthermore, to reduce the potential for separation of flow and to provide uniform approach flow, the square conduit extended upstream for approximately 19 feet without any contractions, expansions, or bends. At the connection to Alden's 36-inch diameter penstock, a round-to-rectangular transition piece was installed and have a total length of approximately 6 ft. Downstream of the transition piece was an approximately 19 feet of rectangular conduit 22.5 inches high. This rectangular conduit connected to the turbine model inlet.

Flow characteristics at the outlet must also been considered to provide representative conditions to the prototype unit. A tailrace structure consisting of a discharge tank was constructed downstream of the draft tube. The tank structure was approximately 8 ft by 10 ft in plan and allow for a 6-inch flow depth over the turbine outlet and 14 inch flow depth between the turbine invert and discharge box invert.

A model diameter of 16 inches was used and the corresponding total length of the model was 7.77 ft, based on the prototype dimensions provided by MSI. The centerline of the testing apparatus and model remained consistent without any vertical variation in the testing area. Some physical limitations have been considered in developing the testing approach. The installed lifting apparatus situated over the testing area is rated for a maximum load of 5 tons. Approximately 15 ft downstream of the terminus of the concrete platform is a bridge crossing limiting component installation. In addition, the invert of the concrete channel floor begins to transition from level to sloped in this general area.

Results and Discussion

For the proposed Sub-topic 1.1 project, MSI designed a compact, efficient, low-head hydro-turbine which satisfied the HGE team's performance objectives. MSI performed the preliminary hydraulic design of the propeller, the blading, the shroud, and the inlet and outlet ducts' fixed vanes and water flow passages, to optimize the initial performance of the runner. MSI's mechanical finite element analysis (FEA) of the propeller consisted of vibration analysis and of impeller stress and deflection analysis. The impeller stress and deflection analysis addressed the centrifugal loadings and the worst-case pressure loadings. To prevent problem resonances from occurring when the new turbine was to be operated, analytical modal analysis was applied to predict the natural frequencies and the mode shapes of vibration of the impeller design. MSI's mechanical FEA helped to mitigate the net risk of the turbine project by enabling a robust design to be produced efficiently, and by preventing potential vibration problems from occurring after the machine is constructed and operated, when such problems become far more costly to remedy.

A detailed design optimization of the turbine stage was performed for the low-head design pitch using the ANSYS/CFX Design suite of CFD tools. The performance optimization focused on the attainment of maximum power, especially during the anticipated worst case operating conditions. MSI performed a CFD analysis of the one-sixth scale hydro-turbine model that was constructed during this project, and additionally, performed an iteration of the CFD analysis. MSI's use of CFD analysis assured the accuracy of the representative physical model of the turbine to the scaled model which was tested at Alden Research Laboratory.

Due to the typical nature of its projects, MSI maintains a very high level of computational resources that would befit a much larger company. MSI's high performance computing resources include ten 64 bit multi-processor servers running Linux. MSI's HPCC (high performance computing cluster) has a total of 72 Intel Xeon processor cores running at 3.33 GHz and 288 GB of 1333 MHz DDR3 memory. The cluster's compute nodes communicate via a 20 gigabit / sec high-speed interconnection. Another system has eight Intel Xeon processor cores running at 3.2 GHz, 64 GB of memory, and an NVidia Tesla C2070 supercomputing GPU card, with 448 cores and 6 GB of DDR5 memory. Three additional systems each have eight Intel Xeon processor cores running at 3.2 GHz and with 64 GB of memory.

MSI performed its project tasks through the expert application of state-of-the-art tools in computational fluid dynamics (CFD) analysis, finite element analysis (FEA), and three dimensional (3D) computer aided design (CAD) modeling. For example, MSI has a suite of finite element general-purpose structural analysis computer programs, which includes the ANSYS CFX and FLUENT fluid dynamics codes, and the full version of ANSYS Multiphysics with fluid / structure interaction modeling capability. MSI has multiple seats of the Pro/ENGINEER solid modeling computer software package, with the Pro/MECHANICA FEA package. MSI's specialized design / analysis software includes turbine-generator system sizing and optimization tools, the University of Virginia ROMAC rotordynamic programs, and the ACTIS tribological component behavior programs. MSI also possesses a suite of material

behavior tools which includes the ASME Boiler and Pressure Vessel Code, Section II set of programs.

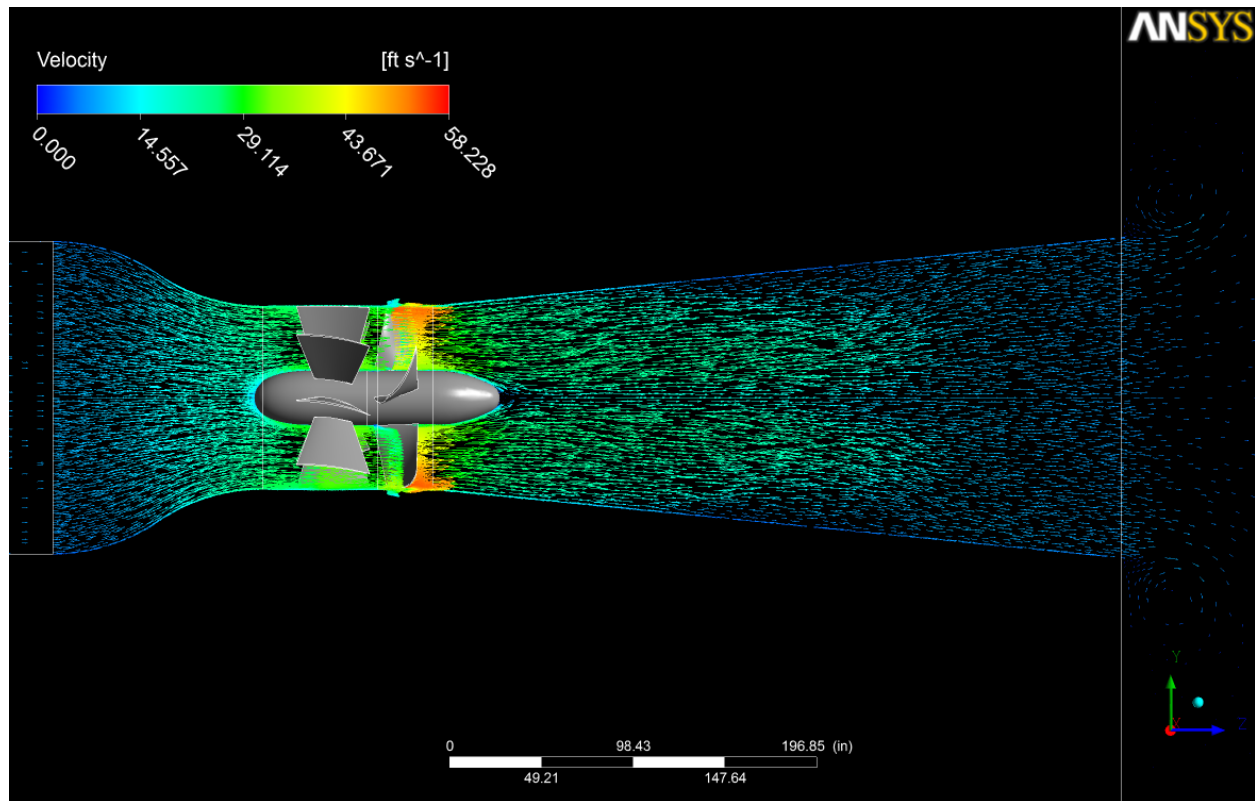


Figure 4 CFD of Full Scale Turbine Flowpath

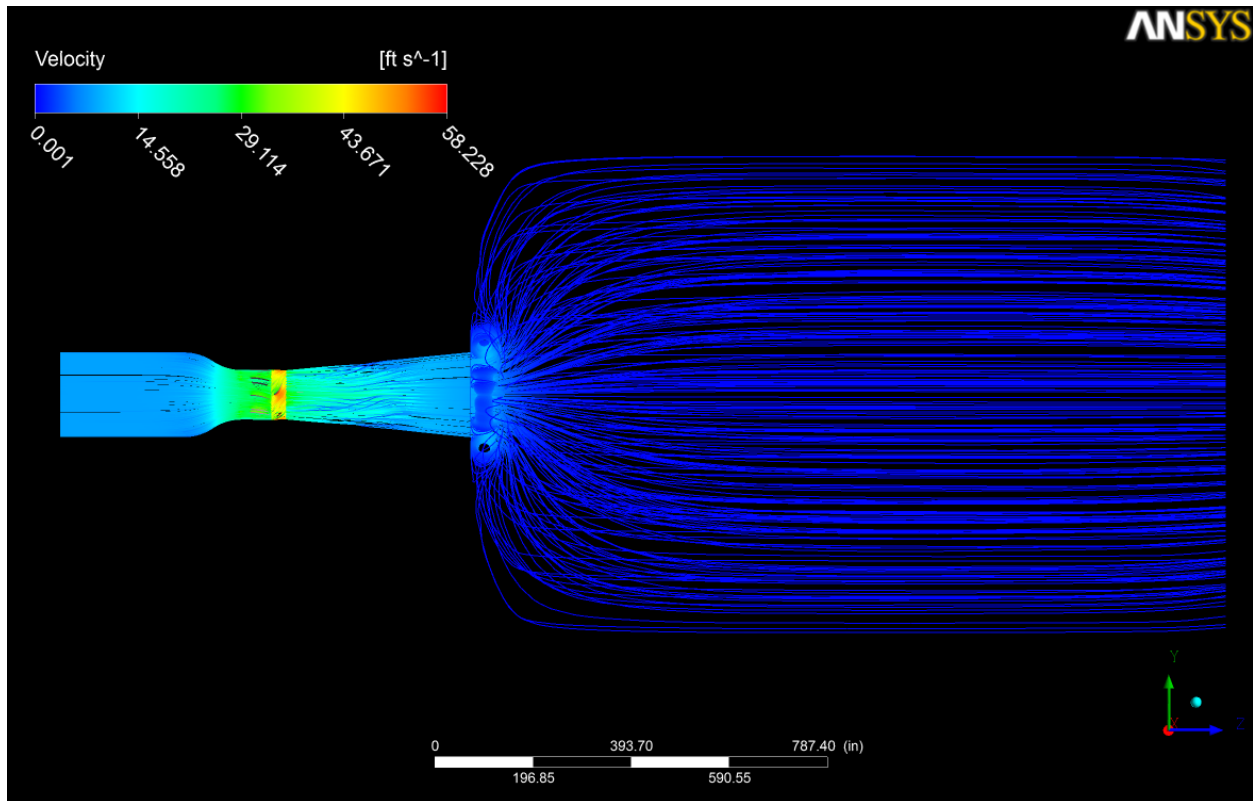


Figure 5 CFD of Full Scale Turbine Flow System - Pool to Pool

Initial design parameters were required to be modified during the CFD runs in order to achieve the desired performance. Adjustments that were made during the modeling runs were:

- Focus On Reducing Delta H to Meet Requirement
- Revised Shroud Iteration Reduced Flowrate Relative To Initial Run
 - 1133 cfs to 1060 cfs
- Turbine Efficiency – 86%
- Reduced Flowrate And Delta H Insures Requirement Will Be Met
 - Accounts For Losses In Inlet and Draft Tube
- Optimization Required Updated Stator And Rotor To Obtain Maximum Efficiency At Desired Flow And Head

Discussion of Laboratory Sub-Scale Test Plan (taken directly from Test Plan prior to test)

Instrumentation

The following parameters and associated instruments were used during the turbine performance testing:

- Flow Rate – Venturi meter
- Head – differential pressure (DP) transducers and pressure taps
- Speed – tachometer sensor
- Torque – torque sensor
- Speed Control – friction brake (assuming no generator)
- Dynamic Pressure – Absolute conditions

The flow rate was measured using the permanently installed 36-inch by 21-inch Master Venturi meter located in the existing 40-inch supply penstock. The Venturi is periodically calibrated using the gravimetric method (45,000 lb weight tank) and a transfer standard Venturi. The Master Venturi performance is characterized by plotting the discharge coefficient versus pipe Reynolds number.

Three piezometer measurements were recorded for each test point to measure head. The inlet head was measured using two yoked taps located in the 26-inch pipe, 10 ft upstream of the round-to-rectangular pipe transition connecting the penstock to the square conduit section. Inlet head (Tap 1) was measured using a DP transducer referencing a water column set at the centerline of the rectangular-to-round transition piece located immediately upstream of the turbine; therefore, inlet pressure was relative to the centerline of the turbine. Tap 2 was the differential head between the inlet piezometer (described above) and the pressure tap located in the draft tube. Tap 3 provided the differential head between the inlet piezometer and a pressure tap located in the tailwater box.

A water-cooled pneumatic friction breaking system was utilized to allow operation of the turbine at a predetermined set speed for evaluation of head and flow conditions associated with a particular speed. Turbine rotational speed was controlled manually by increasing the air pressure on the brake while monitoring the turbine rpm. Torque was measured using a rotary torque sensor installed on an external shaft connected mechanically to the turbine runner. Turbine shaft speed was monitored by an optical speed sensor and manually recorded. The sensor includes a multi-tooth sprocket mounted on an external shaft connected to the turbine runner with an optical sensor. The optical sensor will be displayed near the turbine to allow for adjustment of the brake friction to maintain a target turbine rotational speed. The testing was performed in accordance with ASME PTC 18-1992 as described above. ASME PTC 18-1992 provides procedures and methods for field testing hydraulic turbines to a minimum of $\pm 2\%$ uncertainty in calculated efficiency values. Alden anticipates that the actual uncertainty for the tests will be less than $\pm 1\%$ in calculated efficiency. The actual uncertainty is dependent on the accuracy of data collection equipment and instruments.

Testing

Turbine performance is dependent on a variety of variables including flow rate, head, and unit speed. The testing focused on varying the head and flow rates for a predetermined speed to

develop a hill chart and ultimately determine the best efficiency point (BEP). Initial testing was focus on evaluating the turbine at the estimated design BEP. However, due to limitations of the the tailwater tank, BEP was never achieved. The number of testing points allowable was a function of the allotted 3 week test duration. Finally, the unit was tested in over speed conditions to observe behavior and measure pressure pulsations.

Testing commenced with a system shakedown to ensure proper system installation. After checking all installation components, water was introduced into the system to equalize the line pressures. Prior to the test run, the control valve were adjusted to reach the desired test condition. After steady conditions were achieved, the differential head recorded by the pressure taps was averaged over two minutes to obtain a precise head reading. Once the system reached a steady state, initial measurements were taken and recorded for a target head and flow condition. Upstream valves were then adjusted to change the head and flow for the next test condition. Holding the speed steady, measurements of torque were collected while the head and flow rates were varied through adjustment of upstream valves. Flow and head on the test turbine was controlled by a valve located upstream of the initial round-to-rectangular transition piece.

Flow rate was measured using the installed Venturi meter located upstream of the installation location. Standard DP transducers output 4-20 milliamp signals which are sent to Alden's CAL 3DA software. CAL 3DA translates the 4-20 milliamp signal to a flow rate ultimately recording the flow rate in gallons per minute. Head was measured using DP transducers with taps located on a common side wall upstream and downstream of the turbine. A reference tap elevation matching that of the turbine axis will be used. The DP transducer signal were monitored and recorded using standard PC equipment with an analog to digital (A/D) board and Test Point software used by Alden's Instrumentation Department. A DP transducer installed between the unit inlet and outlet was used for time average changes in pressure between the two locations.

Following initial data collection, dynamic pressure measurements were taken to evaluate the absolute pressure including variance, pressure peaks, and fluctuations associated with overspeed conditions. The unit was allowed to operate without a load or braking system restraining the rotational speed. During this time, the existing differential pressure transducers and pressure taps were utilized; however, recording frequency increased to monitor for peak pressure loadings.

Turbine speed was controlled utilizing a breaking system coupled to the turbine shaft or external upper shafting system. The breaking system was manually operated and speed outputs were taken by the tachometer system.

References

American Society of Mechanical Engineers (ASME). ASME PTC 18-2002. Hydraulic Turbines and Pump-Turbines Performance Test Codes, An American National Standard. 2003.
International Electrotechnical Commission (IEC). IEC 60193. Hydraulic Turbines, Storage Pumps and Pump-Turbines Model Acceptance Tests. 1999.

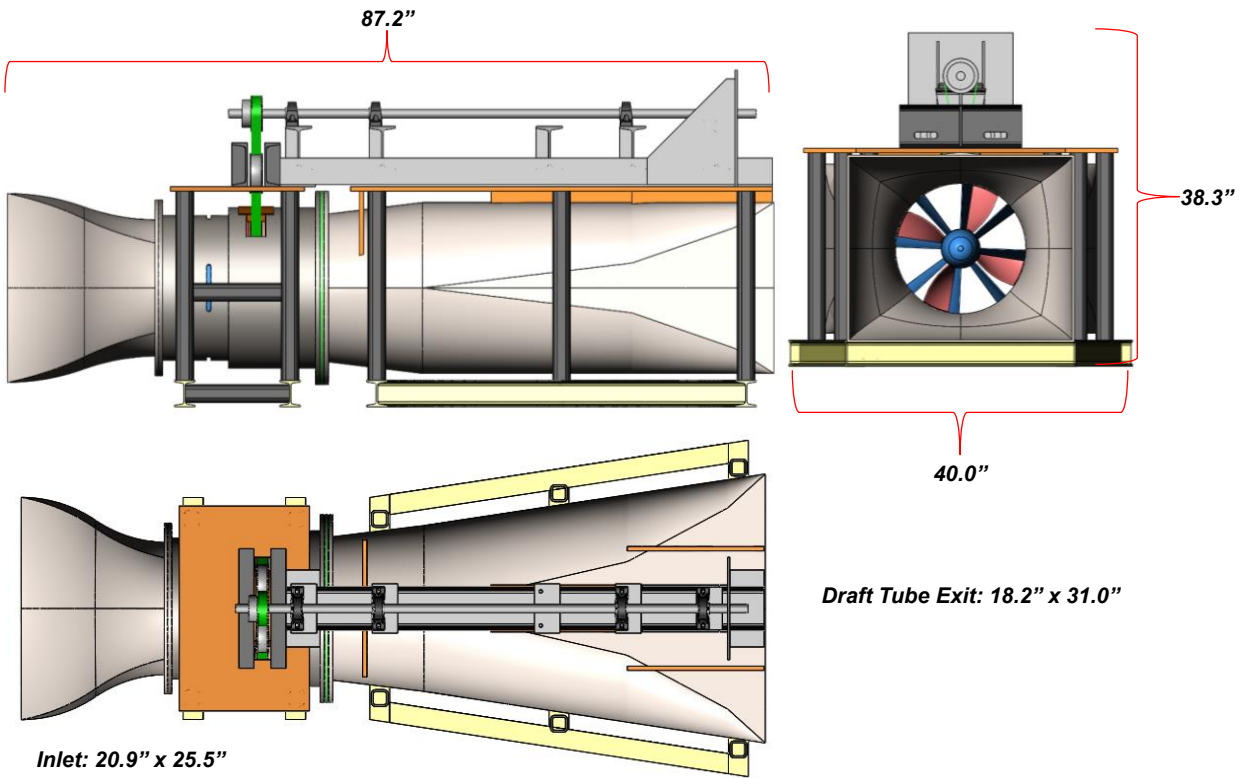


Figure 6 Side Elevation, Plan and Front Elevation of Sub Scale Turbine Test Rig



Figure 7 Sub Scale Turbine Test Loop at Alden Labs

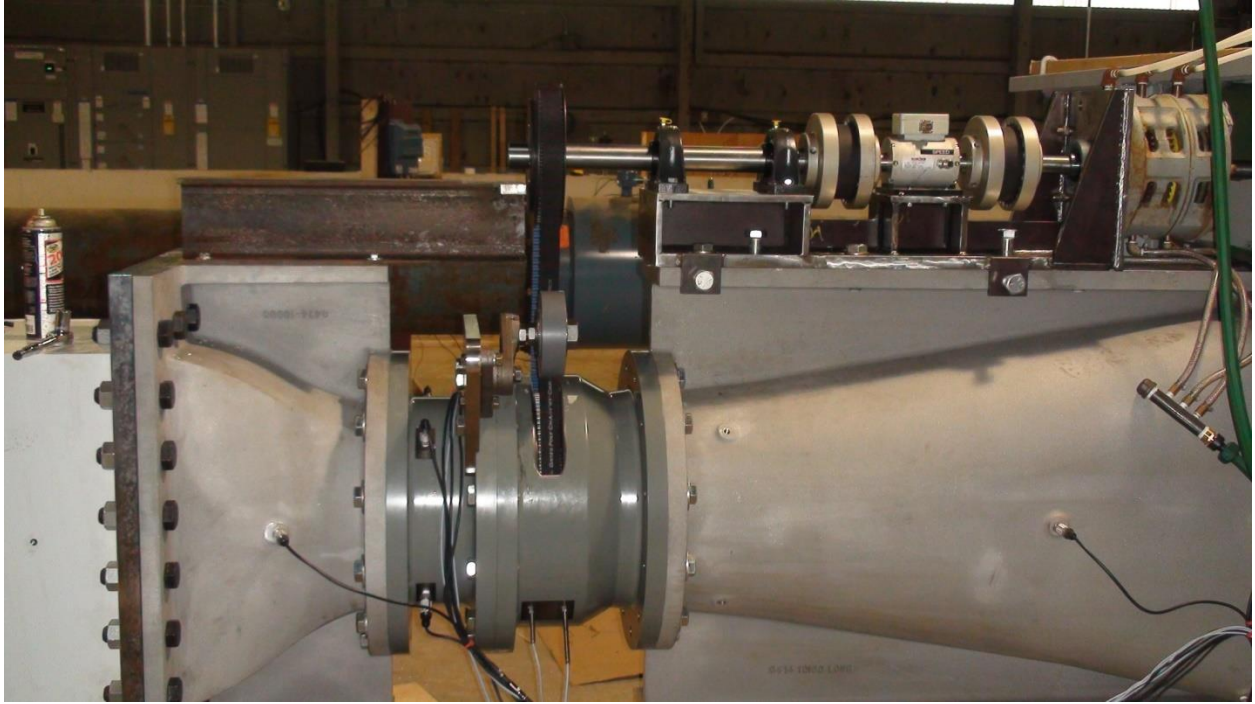


Figure 8 Sub Scale Turbine – Inlet at Left, Draft Tube at Right



Figure 9 Test Loop “Tailrace/Lower Pool”



Figure 10 Running Tests and Data Collection

Laboratory Test Plan Summary

- Test run using American Society of Mechanical Engineers Test Codes on Hydraulic Turbines (ASME PTC-18 2002)
- Turbine scaling via International Electrotechnical Commission (IEC) International Standard 60193
- Turbine runner diameter = 16 inches (1/6th scale)
- Test loop – 50 ft x 14 ft – 24 in. diameter pipe
- Tank flume 8 ft x 12 ft
- Real-time data collected via:
 - Flow rate – magnetic flow meter – opposite side of loop from turbine
 - Head – differential pressure transducers
 - Torque – torque sensor
 - Speed – tachometer within torque sensor
- Manually controlled pneumatic brake to adjust turbine speed

	Full Scale	Sub-scale
Flow, cfs	1,307	31.2
Head, ft	9.4	10.0
Runner Diameter, ft	8.2	1.3
Speed, RPM	94	638

Turbine Power, kW	840	21.3
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Figure 11 Comparison of Sub Scale and Full Scale Turbine Design Parameters

Sub Scale Test Results

- Initial goal to test BEP, then develop curves and high risk runs at end if time permitted
- Alden Lab indoor test loop unable to operate at sub-scale design point – high torque
- Cavitation observed approaching design point during multiple test runs – tail water level tested at 6 inches up to 24 inches.
- Braddock full scale has higher tail water static pressure – runner setting ~11+ feet below tail water = no cavitation
- Alden Lab tests mostly conducted below design point – low torque range
- Synchronous belt less efficient at below design conditions – sized for high torque tests
- Alden sub-scale best manually recorded/observed efficiency at 0.722 (<50 datas)
- Alden sub-scale digitally recorded average efficiency at 0.702 (10,000 data points)
- Alden projected full scale efficiency at 0.775 (based on sub-scale digital data)
- MSI projects operation at design point (and belt design point) would have yielded as much as another 4-6% efficiency
- Full Scale Best Efficiency Point (BEP) would have been ~ 0.805

Mechanical Solutions Lab Test Executive Summary (taken directly from MSI Executive Summary):

On September 29, 2014, Alden Laboratories conducted a series of tests on a propeller turbine which was designed and assembled by a joint Hydro Green Energy, MSI and HDR Incorporated team. Hydro Green served as the project lead, MSI developed the inlet, fixed gates, runner blades, and mechanical design, while HDR developed the draft tube. MSI performed all of the CFD analyses on the full scale and laboratory scaled assemblies.

The full scale design featured a 2.5 meter diameter runner, while the scaled test employed a 0.380 meter diameter runner. As noted in the Alden HE Report, the model characteristics of the test were the following:

1. Flow rate = 31.2 cfs
2. Net head = 10 feet
3. Rotational speed = 638 rpm
4. Alden estimated efficiency = 78%

The CFD calculations conducted by MSI used slightly different conditions. At the best efficiency point, MSI obtained the following:

1. Flow = 30.2 cfs
2. RPM = 618

3. Net Head = 8.96 feet
4. Efficiency = 0.827.

It should be noted that the CFD calculations do not address leakage or mechanical losses, so Alden's number become reasonable. The first series of tests conducted by Alden were run at 618 rpm. The pressure transducer readings in the draft tube indicated that the gauge pressure was negative in the draft tube. The exit pressure was only slightly higher than atmospheric. The net head was also much larger, closer to 14.8 feet. The efficiency was much lower than BEP due to an obvious explanation. The draft tube serves to minimize the exit velocity, in order to extract the maximum energy from the system. This creates a very low pressure at the runner exit. If the pressure is low at the draft tube exit, it will be even lower at the runner exit. This creates cavitation that deteriorates the performance of the turbine. MSI recommended a minimum exit draft tube pressure of 16.75 psia. The net head should have been limited to ten feet. This would have produced significantly different results.

On September 29, 2014, the test was rerun at a much lower rpm (450), and this changed results significantly. The turbine now operated in a region where cavitation did not deteriorate the performance of the turbine. The max efficiency reported by Alden was 72.2%, much higher compared to the previous week's test. This result still deviated from a higher expectation. The result for the departure can be found in the following:

1. The belt drive was sized for 41 hp. The test points were run at an approximate horsepower level of 10 hp. At 25 % of its sized condition, the efficiency of the belt drops precipitously per the manufacturers product data sheet.

The fact that the belt drive has much higher losses at lower torque ratings demonstrate that the turbine performance was quite close to the predicted CFD performance. It demonstrated the importance of maintaining the correct back pressure. The actual unit is intended to be situated such that the top of the draft tube lies two feet below the surface of the water. That will set the back pressure at 19.245 psia, approximately 4.4 psia higher compared to our test. If the belt can operate at the proper setting, its loss will be extremely low, and the efficiency of the turbine will be much higher.

Based upon these results, as well as the predicted power loss curve supplied by the belt manufacturer, a redesign of the draft tube is not warranted. The fact that the cavitation was present and potent demonstrates that the draft tube effectively diffuses the flow, and reduces dynamic pressure. If the turbine operated outside of the cavitation region at the high speed (618 rpm), then it would demonstrate that the draft tube was ineffective at diffusing the pressure. It is important to maintain a two foot level of water over the top of the draft tube. This should maintain a high enough back pressure to permit service at full load without the deleterious effects of cavitation. No changes were ever contemplated for the inlet. In general, the inlet losses are quite low based upon the CFD, so no redesign is warranted. There are no indications whatever from the test of a poorly performing inlet. The belt losses alone justify the difference in the results.

Test Results Summary Table

The larger set of automatically recorded data included over 10,000 averaged data points taken over a three day testing period. The data required processing to remove data associated with unstable conditions and transition periods between target test conditions. During testing and under certain speed, head and flow conditions, surging of the system was observed. These unstable conditions were likely conditions outside the turbine's operating range and may represent cavitation. An analysis of the data was completed which evaluated the standard deviation (SD) of each consecutive grouping of data based on the recorded speed. Data sets which had greater than two SD among the group of five speed recordings were removed from the data set. The remaining data set was condensed into averages of the five data sets. Finally, the standard deviation of the efficiency data was calculated and all data sets in which the efficiency was outside of one SD were removed. Throughout the report, this will be referred to as the "reduced data set".

Note - *** = results based on the reduced data set.

	Uncertainty, %
Alden Observed Top Efficiency – Sub-scale	0.722
Alden Digitally Averaged Efficiency – Sub-scale***	0.702
MSI Estimated Synch. Belt Efficiency Loss Range	4-6%
Maximum Measurement Uncertainty	0.93
Calculated BEP at Design Point (sub-scale)	~0.75 to 0.77
Alden Estimated Full Scale Efficiency***	0.775
Estimated BEP at Design Point (full scale)	≥0.805

Figure 12 Sub Scale Test Results

Accomplishments

While the project was successful, no external publications were made nor were any patents applied for. HGE's intellectual property includes multiple U.S. and international patents pending and issued not based on this work and prior to the start of this project.

Conclusions

The sub-scale lab test was deemed successful. There were no mechanical failures of the turbine. Given the limited budget the turbine operational efficiency actually came very close to the predicted efficiency for the operating conditions in the lab. Considering the limitations of the lab test loop, the conclusion is that no additional wholesale redesign of the turbine is warranted. Given more time and a larger budget there is the possibility to increase the efficiency of the turbine. However, in order to test the turbine at the proper sub-scale conditions corresponding to the actual setting of the full scale turbine, significant and costly alterations would need to be made by this lab or an alternate suitable testing lab would need to be located.

Hydro Green Energy believes that this is the only technology addressing large river/utility scale systems (up to 30-40 MW) at navigational locks & dams. HGE has 250 MW of lock & dam permits from FERC which would enable wide scale adoption of this technology. Due to the simple and robust design this turbine is ideally suited for modular systems. There are at least 16,000 MW of unpowered dams in the U.S. as estimated by the DOE.

Recommendations

Based on the results of the sub-scale test at the end of the award period, it is HGE's recommendation to proceed with full scale design/testing with the award that has already been selected for negotiation by DOE.

References

Alden Research Laboratory Report, Test Plan, October 2013

Mechanical Solutions Inc., Test Results Analysis Executive Summary, October 2014.

Alden Research Laboratory Report, Testing of One Hydroelectric Turbine Serial Number 10001, December 2014

Hydro Green Energy, HGE Lab Tests Final Presentation, December 2014.

American Society of Mechanical Engineers (ASME). ASME PTC 18-2002. Hydraulic Turbines and Pump-Turbines Performance Test Codes, An American National Standard. 2003.

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