# Application of CFD Turbine Design for Small Hydro Elliott Falls, A Case Study

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## ABSTRACT

*The Elliott Falls small-hydro generating station is located in Norland, Ontario, Canada. Commissioned in 1990, the plant originally comprised two vertical axis, variable-pitch cooling pumps acting as turbines. After less than 10 years, severe blade cavitation and hub seal failure necessitated remedial action. Various alternatives were considered including replacement of the pumps with new, off-the-shelf small-hydro turbines. A solution was implemented that involved replacement of the existing stay vanes, hub and blades with a custom runner and distributor. Both were designed using state-of-the-art computational fluid dynamics (CFD) software technology. The paper describes the original design of the power plant, the problems encountered, the solutions adopted, and the results of the repair/upgrade. It will be demonstrated that the use of CFD for the design of custom small-hydro turbines is appropriate.*

## Background

The Elliott Falls small-hydro generating station is located in south-eastern Ontario on the Gull River just north of the town of Norland. The site was originally developed at the beginning of the 20$^{th}$ century to supply power to a nearby cement factory. The plant was decommissioned in 1928.

The original plant was an open-flume type development with two sets of four 25-inch runners each mounted on a horizontal shaft. A separate exciter turbine was also provided. Each set of four turbine runners was connected to a 350 kW generator giving a total plant capacity of 700 kW.

In 1988 the Ontario Ministry of Natural Resources released the site for development as part of the government's initiative to encourage private development of small hydro sites. At the time, long-term power purchase agreements were being offered by the former Ontario Hydro, which provided the incentive to re-develop the site. Elliott Falls Power Corporation was the successful bidder out of the thirteen companies that submitted proposals. The redeveloped plant was commissioned in 1990.

The hydropower site at Elliott Falls was redeveloped within the confines of the old open-flume and powerhouse structures. Two used vertical axis variable-pitch propeller cooling water pumps were refurbished and modified slightly to work as turbines. The modifications included shortening the vertical portion of the pumps (to accommodate the redevelopment of the site) and adjusting the upstream edges of the vanes in each of the pump's right-angle mitre bend (which were originally the downstream edges when the units were operated as pumps).

Advice provided by experts at the time indicated that the pumps-as-turbines would produce approximately 330 kW each. A financial assessment of the project clearly indicated that developing the site using the pumps-as-turbines provided a higher rate of return than purchasing new turbines. Additionally, the low-cost pumps significantly reduced the cost of the project and, therefore, facilitated financing of the project. As part of the design, provisions were made to allow replacing the pumps with properly suited turbines at a later date.

It was not foreseen, however, that the performance of the pump-as-turbines would deteriorate significantly over time. Severe blade cavitation, as shown in the following photograph, occurred on the downstream faces of the blades close to the leading edges. In addition, the variable-pitch adjusting mechanisms seized in both units due to faulty hub seals, which allowed water to leak into the hubs and cause corrosion. As a result, remedial action was required.

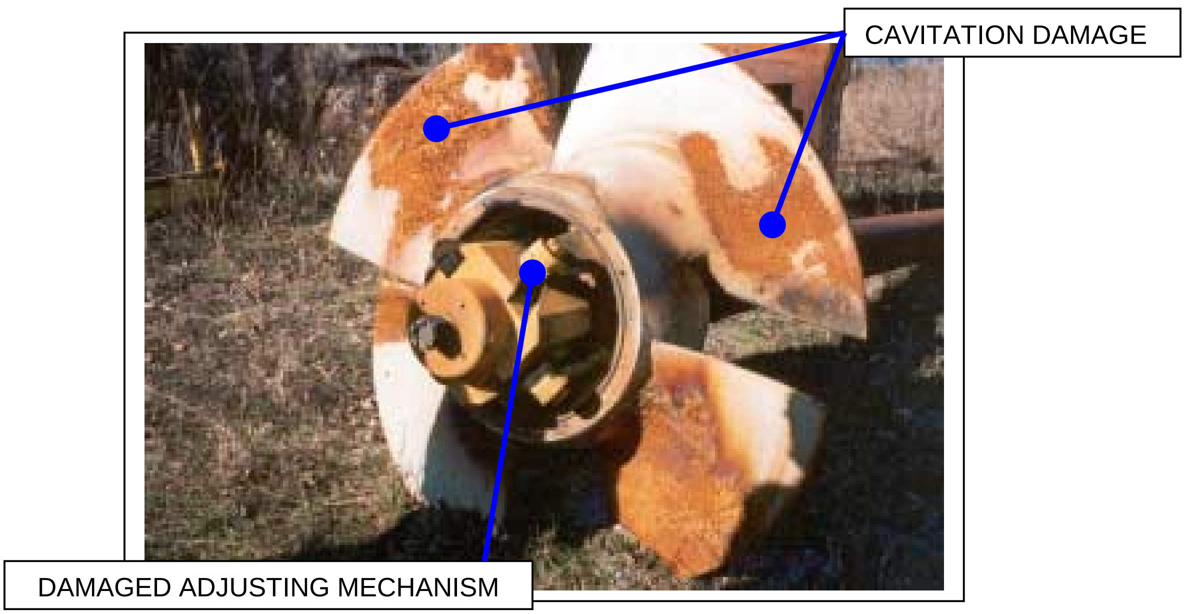



*Figure 1: Photograph of old blade and hub assembly showing cavitation damage and damaged adjusting mechanism (runner diameter is 5 feet).*

Two repair options were considered:

1. Replacing the entire pump assemblies with off-the-shelf Kaplan turbines and making the necessary civil works modifications; and
2. Custom designing, using computational fluid dynamics (CFD) design software, new variable-pitch propeller runners with fixed geometry distributors to replace the two pump impeller and stay vanes assemblies.

While the former option offered somewhat higher overall efficiency, the latter option was selected as it proved to be significantly more attractive from a return on investment perspective.

## Design of New Runner and Distributor

Designing a suitable replacement runner and distributor involved carefully identifying the key design criteria and adopting a cost-effective CFD design procedure appropriate for small hydro.

### *Design criteria*

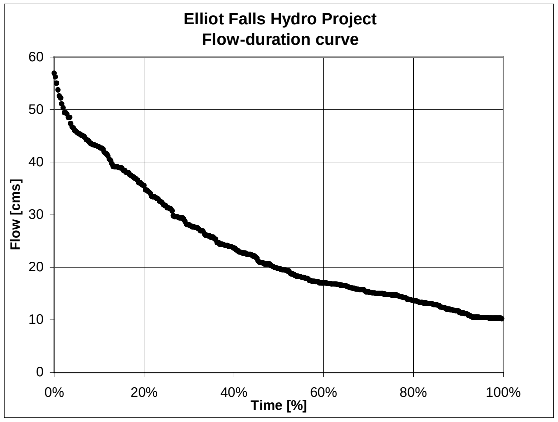
After carefully considering the requirements for the design of the replacement runner and distributor, the following key design criteria were adopted:

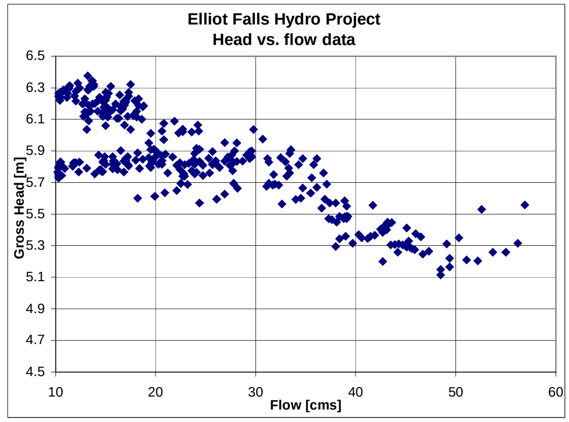
- The new components were to replace the existing pump runner and distributor without the need to modify the remaining water passages or the existing generators (i.e., the existing rotational speed of 330 rpm had to be maintained).
- The design would be optimised for maximum energy production rather than maximum power.

The latter criterion necessitated an evaluation of the available flow and gross head conditions to determine the most appropriate design parameters for the new runner and distributor.

Historical flow and gross head data were analysed to determine the most appropriate $n_{11}$ (unit speed) and $Q_{11}$ (unit flow) values for the new turbine design.

As is evident from Figure 2 (b), the gross head varies significantly. The variations are due to changing upstream water levels (winter vs. summer water levels in the lake upstream of the site) and the hydraulic characteristics of the river (tailwater elevation and upstream river surface gradient between the lake and the plant vs. flow). Consideration of the variations in gross head was a very important part of the design process.




*Figures 2 (a) and 2 (b): Data collected at the site over 5 years of operation.*

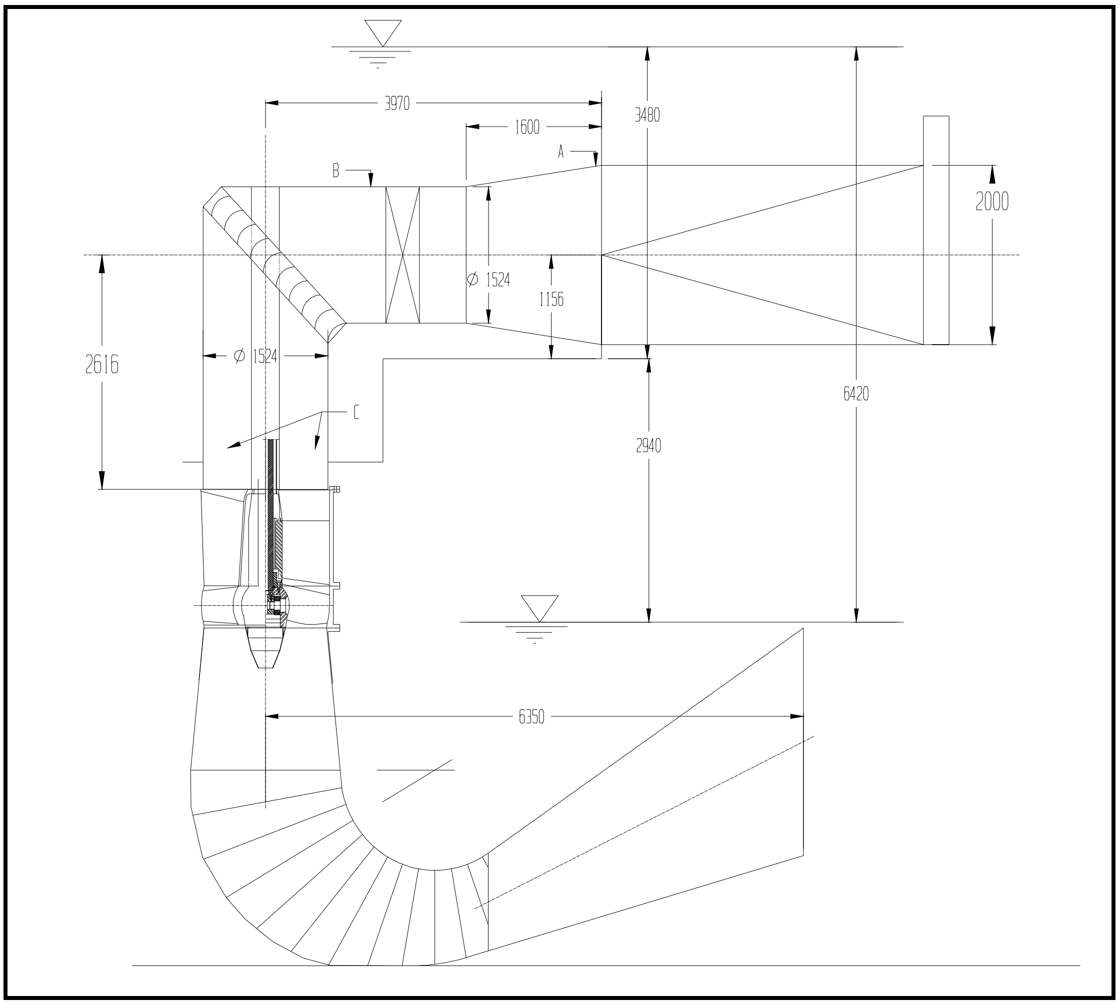



*Figure 3: Elevation showing existing water passages and key dimensions*

The headloss vs. flow relationship for the existing water passages (refer to Figure 3) was determined using conventional, empirical hydraulic engineering equations. A summary of the results is presented in the following table.

**Table 1: Summary of headloss calculations based on conventional methods.**

| *Flow* | *Headlosses* | *Gross Head* | *Net Head* | $n_{11}$ | $Q_{11}$ |
|---|---|---|---|---|---|
| 5 | 0.21 | 6.3 | 6.09 | 199 | 0.94 |
| 7 | 0.4 | 6.3 | 5.90 | 202 | 1.33 |
| 9 | 0.64 | 5.5 | 4.86 | 223 | 1.89 |
| 11 | 0.94 | 5.1 | 4.16 | 241 | 2.50 |

Evaluation of the available data led to the selection of the design net head and flow of 4.9 m and 11 cms respectively and target design values for $n_{11}$ of 222 and $Q_{11}$ of 2.3. The selection of the target design values took into account the frequency of operation at different head and flow conditions, limitations imposed by geometric restrictions (60-inch throat diameter and minimum practical hub size), required shaft speed of 330 rpm and other characteristics such as exposure to cavitation and hydraulic instabilities.

***Description of CFD design process***

Selecting an appropriate procedure for the CFD design process that balanced effort (cost) and results (optimum design) was important given the budgetary and scheduling constraints. The adopted procedure is illustrated in the Figure 4 [3].

**Design Results**

The adopted CFD design procedure and established target design values for $n_{11}$ and $Q_{11}$ led to an optimised design with the following key characteristics:

Runner throat diameter = 1.47 m
Shaft speed = 330 rpm
Turbine setting (max) = 0.5 m above tailrace level
Number of runner blades = 4
Number of distributor vanes = 7
Hub ratio = 0.37
Peak efficiency ($n_{11}$ = 190, $Q_{11}$ = 1.85) = 91% (hydraulic efficiency)

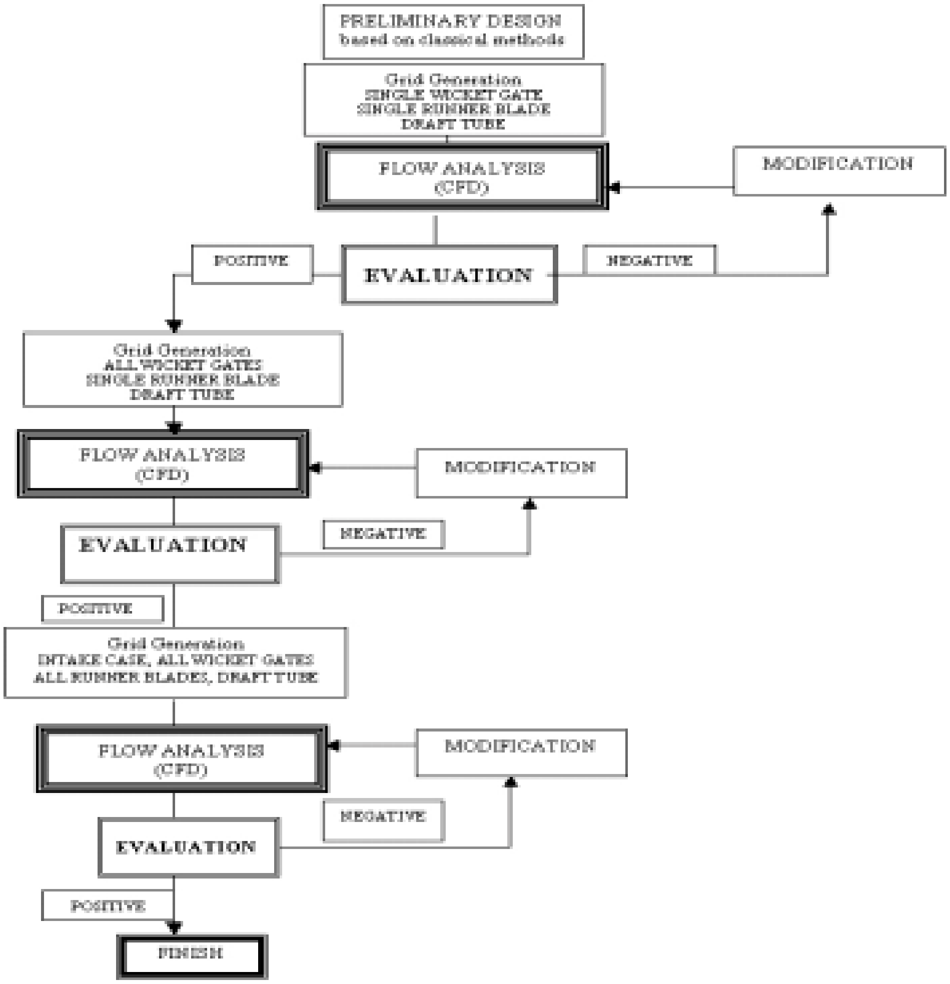



*Figure 4: CFD design procedure adopted for the Elliott Falls runner/distributor assembly*

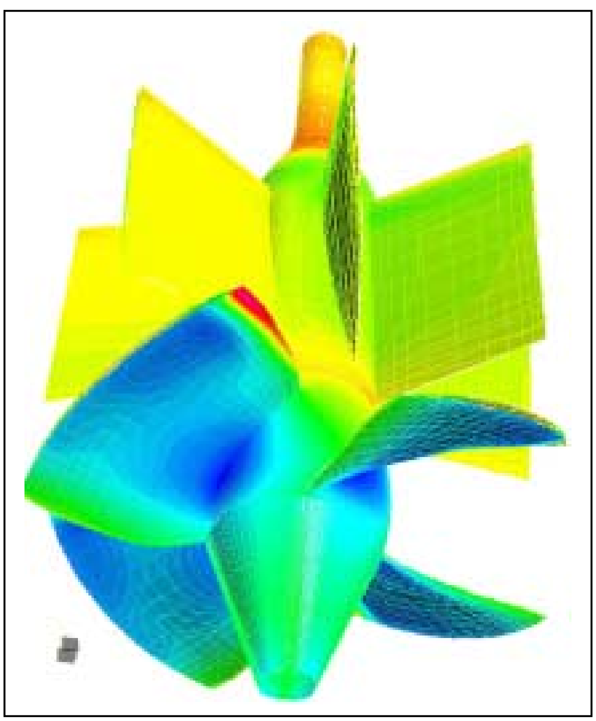


*Figure 5: Static pressure distribution of the runner/distributor assembly*

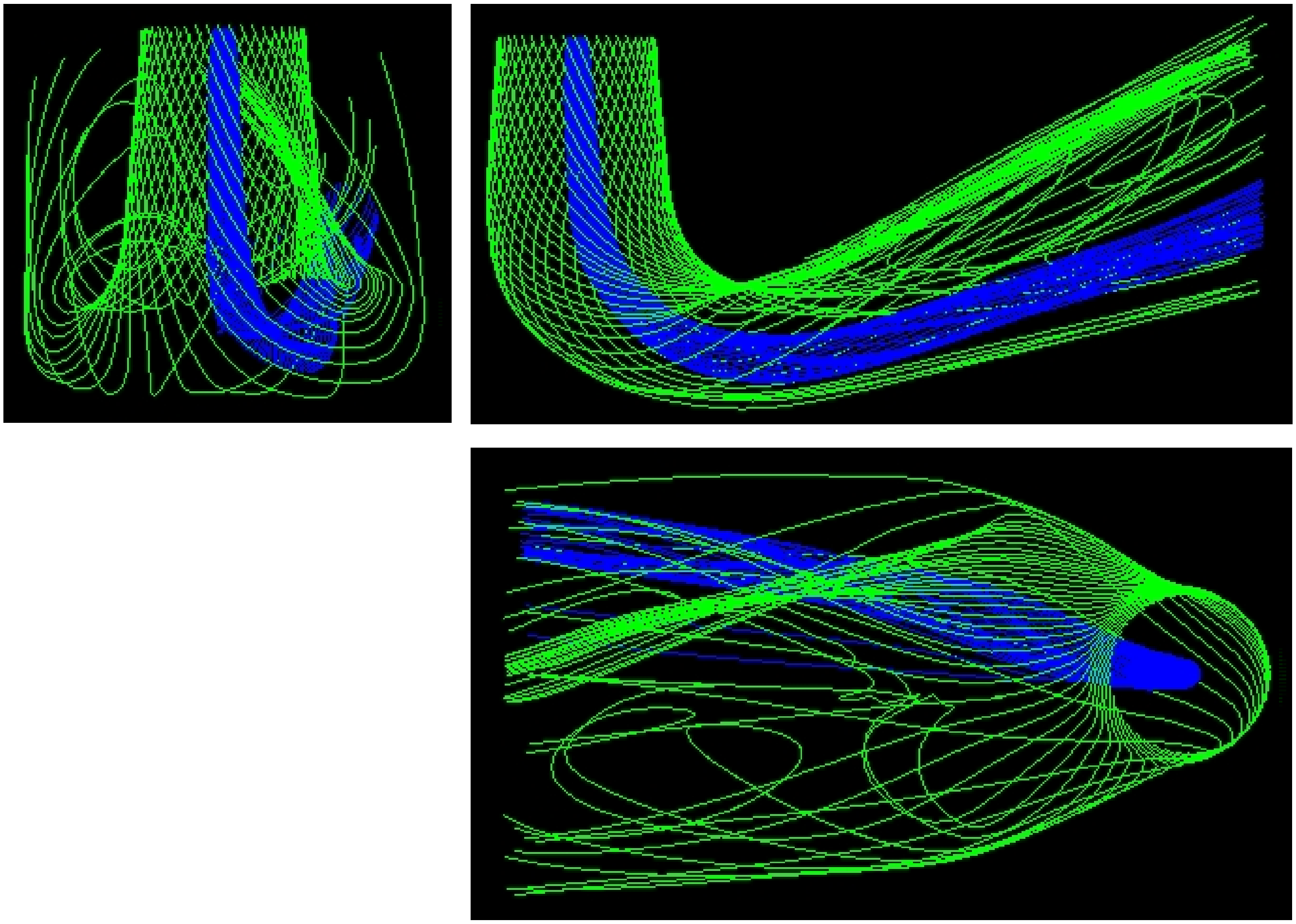


*Figure 6: Draft tube flow - CFD results for the existing draft tube with the new runner/distributor assembly*

**Field Testing of New Runner/Distributor**

The expected power output of the replacement runner and distributor assembly was originally found to be satisfactory, however, the performance of the refurbished turbine did not improve as expected after reaching approximately 350 kW (turbine output).

A field test was conducted which included measuring power output, blade pitch and the elevation of the hydraulic grade line (in clear tubing connected to the turbine water passage) at Taps #1, #2 and #3 as shown on Figure 3.

Based on the assumed water passage headloss to Tap #1 (trashrack, inlet and gradual contraction), the approximate velocity head component of the drop in the hydraulic grade line to Tap #1, and hence the approximate flow, could be derived for each test point. Based on the approximate flow, the average velocity head at Taps #2 and #3 (same cross-sectional area) could be calculated and deducted from the drop in hydraulic grade line across the elbow to estimate the elbow head losses.

Based on the estimated flow and net head downstream of the elbow, the efficiency of the runner distributor assembly was calculated.

Figure 7 shows the efficiency of the replacement runner distributor assembly as designed (expected efficiency) and as estimated based on the field test (calculated test efficiency).

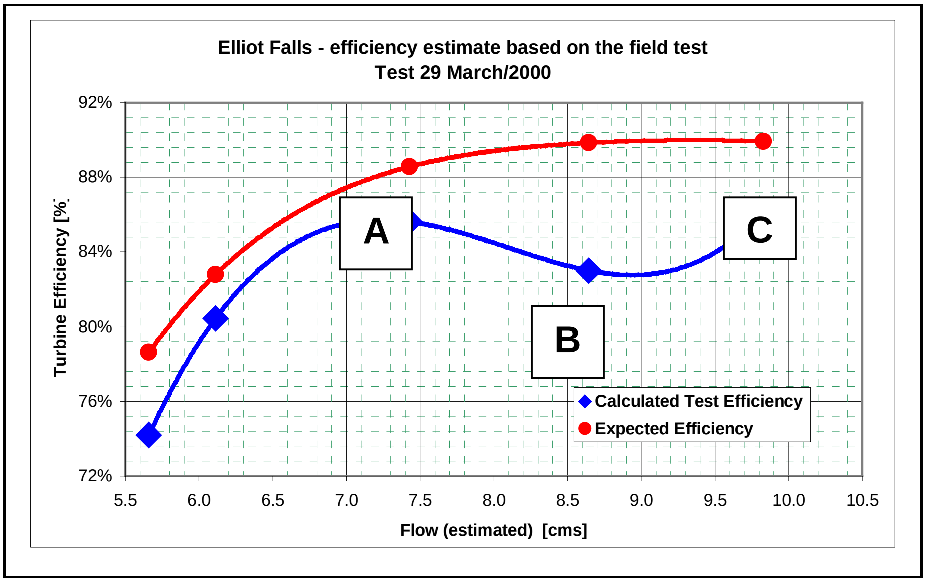



*Figure 7: Estimated Efficiency of the runner/distributor/draft tube assembly*

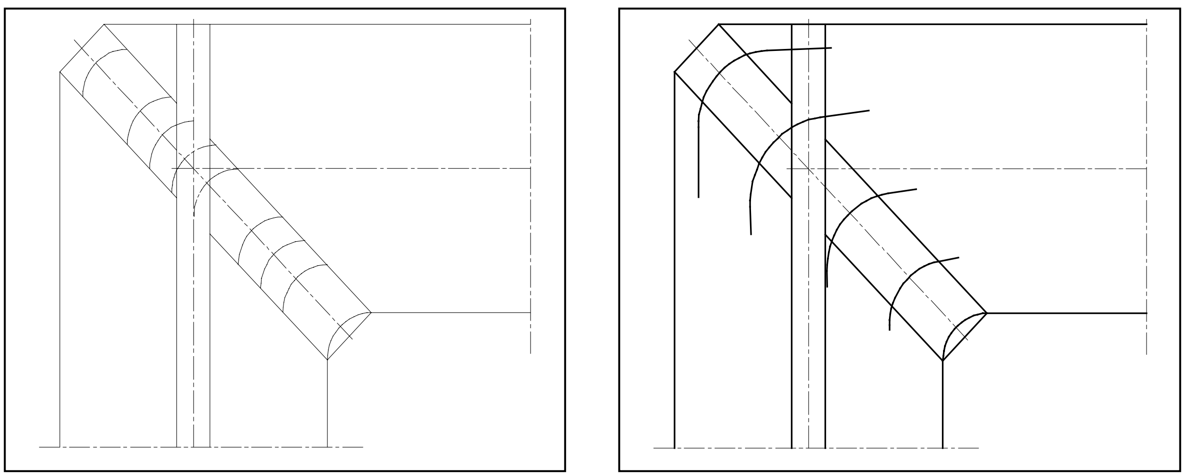
During the field test, cavitation noise was noticed at the elbow turning vane area. As the turbine runner blades were opened and the generator developed more power, the noise and vibration increased, reaching the highest level at Point B (Figure 7). With increased flow through the turbine, the static pressure (inside the penstock) at Tap #2 reached atmospheric pressure. As soon as the critical Point B (Figure 7) was passed, air was sucked into the flow passage through Tap #2 and the vibration and cavitation noise vanished almost immediately.

Based on this observation it was hypothesised that the noticeable efficiency drop between Points A and B (Figure 7) resulted from a very low local pressure zone, which takes place somewhere within the area of the elbow turning vanes and that the low pressure zone resulted in cavitation. It was further hypothesised that the resulting violent flow disturbances must cause significant energy losses in addition to what had been anticipated in the original headloss evaluation using conventional hydraulic engineering methods.

A decision was made to use CFD to analyse and, if possible, design an improved turbine elbow.

**Analysis and Design of Improved 90° Turbine Elbow**

Extensive CFD analysis was conducted independently on two computers using different grid topologies. Flow simulation conducted on both computers showed a very low-pressure zone at the downstream side of the elbow turning vanes close to the top of the penstock. Moreover the flow leaving the elbow turning vanes and entering the turbine distributor was non-uniform causing a power imbalance of approximately 30% between runner blades (different pressure distribution on each runner blade).



(a) (b)

*Figure 8: Comparison between existing (a) and new (b) elbow turning vanes*

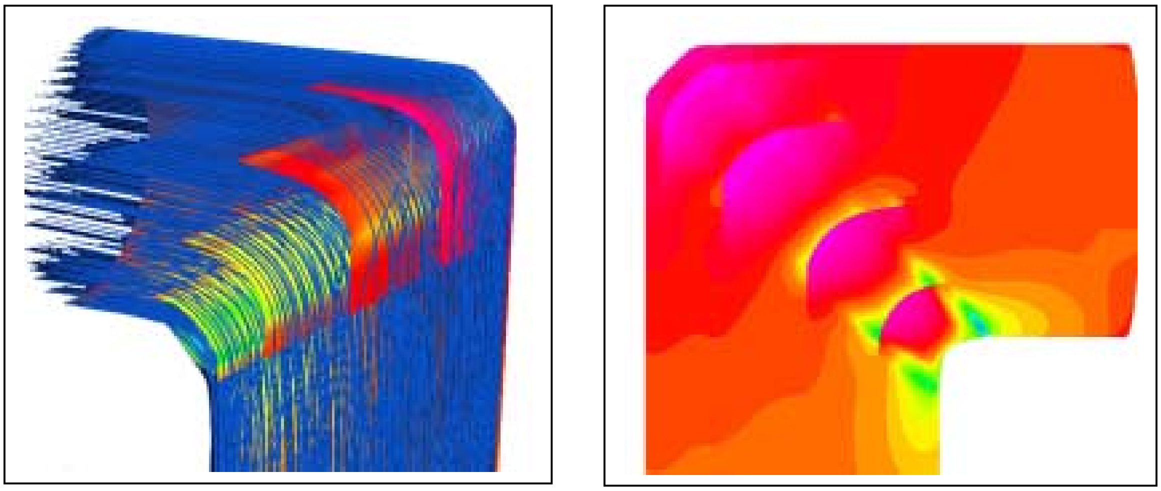


*Figure 9: Results of the flow simulation for the new elbow turning vanes*

In designing the new elbow turning vanes the design was optimised to:

1) Minimise the hydraulic loss at the elbow
2) Achieve uniform velocity distribution at the distributor inlet.

After repeated CFD simulations for various geometric options for improved elbow turning vanes, a new design was adopted (Figure 8 (b) and 9).

**Anticipated Results of New Turbine Elbow Design**

The CFD analysis showed clearly that the hydraulic losses occurring in the turbine elbow turning vanes were not the only reason for performance limits of the new runner and distributor assembly. The pattern of the flow leaving the turning vanes and entering the distributor area had, in this case, a significant impact on the power output and, therefore, the hydraulic efficiency of the turbine unit.

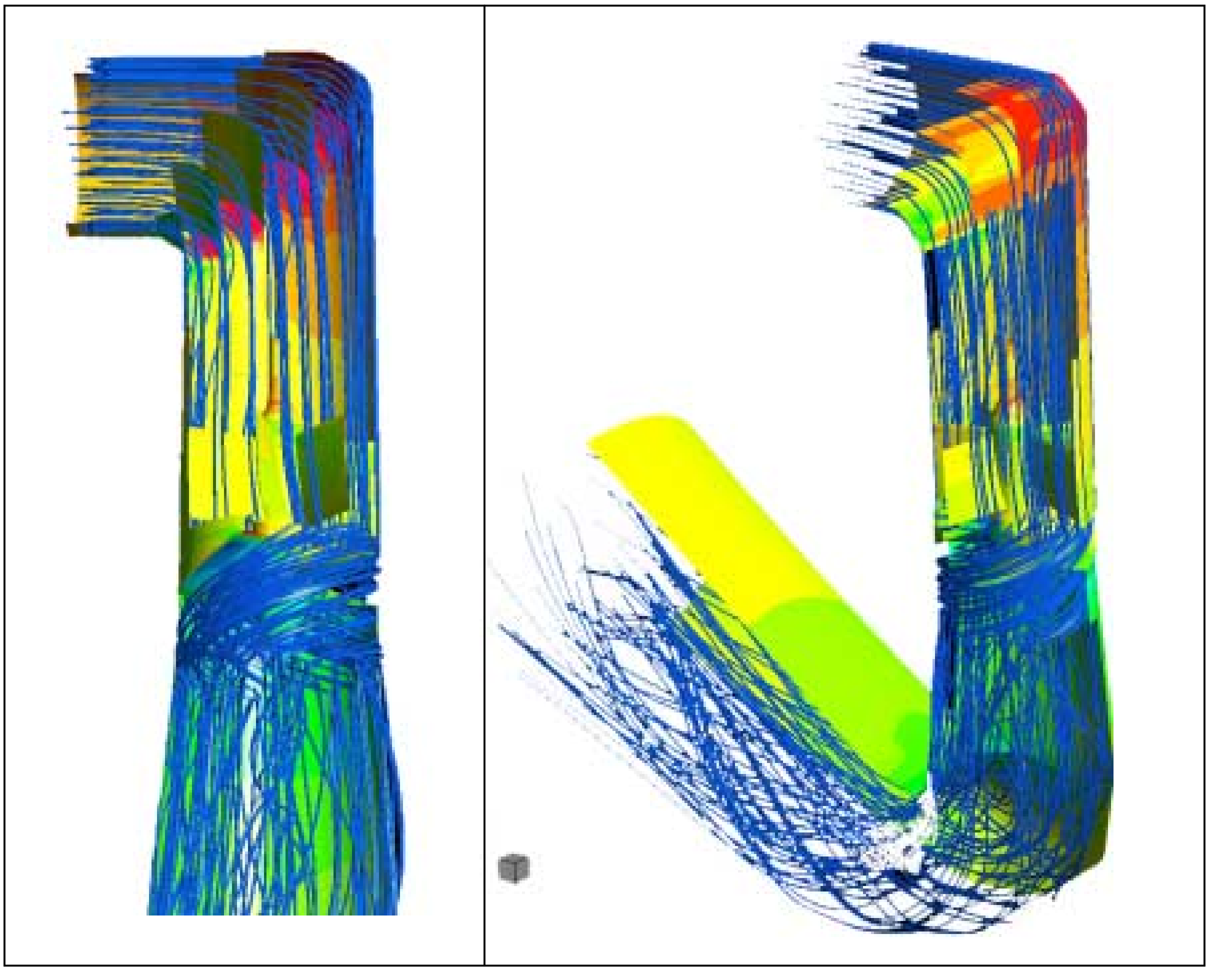


*Figure 10: Geometry of the entire turbine was used to evaluate the performance of the unit equipped with the new elbow turning vanes*

The CFD design optimisation of the turbine elbow turning vanes resulted in the following expected improvements:

1) Reduction in turbine intake losses (elbow turning vanes plus distributor) from 13.4% to 6.7%.
2) Reduction in the imbalance between runner blade outputs from 30% to 6%.
3) Increase of 12% in the turbine runner output.
4) Overall unit efficiency increase of 7%.
5) Elimination of the possibility for cavitation at the turning vanes area.

**Field Testing after Installation of the Improved Turbine Elbow**

Preliminary performance testing after installation of the improved turbine elbow has indicated that actual improvements meet or exceed those anticipated using CFD. Power output has increased by at least 12%.

**Continued Investigations**

A detailed analysis of the entire hydraulic flow path from the trash racks to the draft tube exit (including the intake, transition, butterfly valve, turbine elbow, distributor, runner and draft tube) is being conducted to determine if any further improvements can be made. Results will be available in February 2001.

**Conclusions**

Based on the Elliott Falls experience, the following can be concluded:

- Appropriate use of CFD is cost-effective.
- CFD is an appropriate tool for the design of hydraulic components in small-hydro plants including custom-designed turbines and other hydraulic structures such as high-velocity pipe elbows.
- CFD can be a valuable and accurate tool for evaluating the headloss, cavitation zones and the resulting reduction in plant output associated with poor performing hydraulic structures.

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Jacek Swiderski, P. Eng., owns and operates Swiderski Engineering based in Ottawa. He designed, based on the CFD, replacement runner and distributor and turbine elbow turning vanes.

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