International techno-scientific conference
HYDRAULIC TURBOMACHINES IN HYDRO POWER AND OTHER INDUSTRIAL APPLICATIONS

Czorsztyn, 18 - 20 November 2000

PRACTICAL IMPLEMENTATIONS OF COMPUTATIONAL FLUID DYNAMICS IN THE DESIGN OF HYDRAULIC TURBINES - VIRTUAL HYDRAULIC LABORATORY -

Jacek Swiderski

Swiderski Engineering
Ottawa, Canada

Joseph N. Martin

Norcan Hydraulic Turbine Inc.
Carleton Place, Canada

Abstract: *The authors created and implemented a universal design method for water turbines including inlet structure, distributor, runner and the draft tube. The interactive geometry editor is coupled, via data files, with commercial 3D viscous flow analysis software.*
The paper presents a general structure of the design algorithm as well as an illustrated example of the design optimization process. Particular application of a newly design Francis runner is presented in a form of comparison between CFD predicted and field performances.

1. INTRODUCTION

The calculations of the fluid flow through the rotation machinery by the Computational Fluid Dynamics methods (CFD) always raised a question: how well would their results reflect the performances of the real machine. Model tests of the water turbines as well as other experiments conducted on various fluid flow aspects, when compared to the theoretical predictions, resulted in the high degree of trust to the CFD. Some groups are presently using CFD as a fundamental and ultimate tool to determine and evaluate performances of the newly designed turbines.
The article herein presents how the authors, coming from the assumption on the trustworthiness of the CFD results, are using it for design purposes in the small hydropower field.

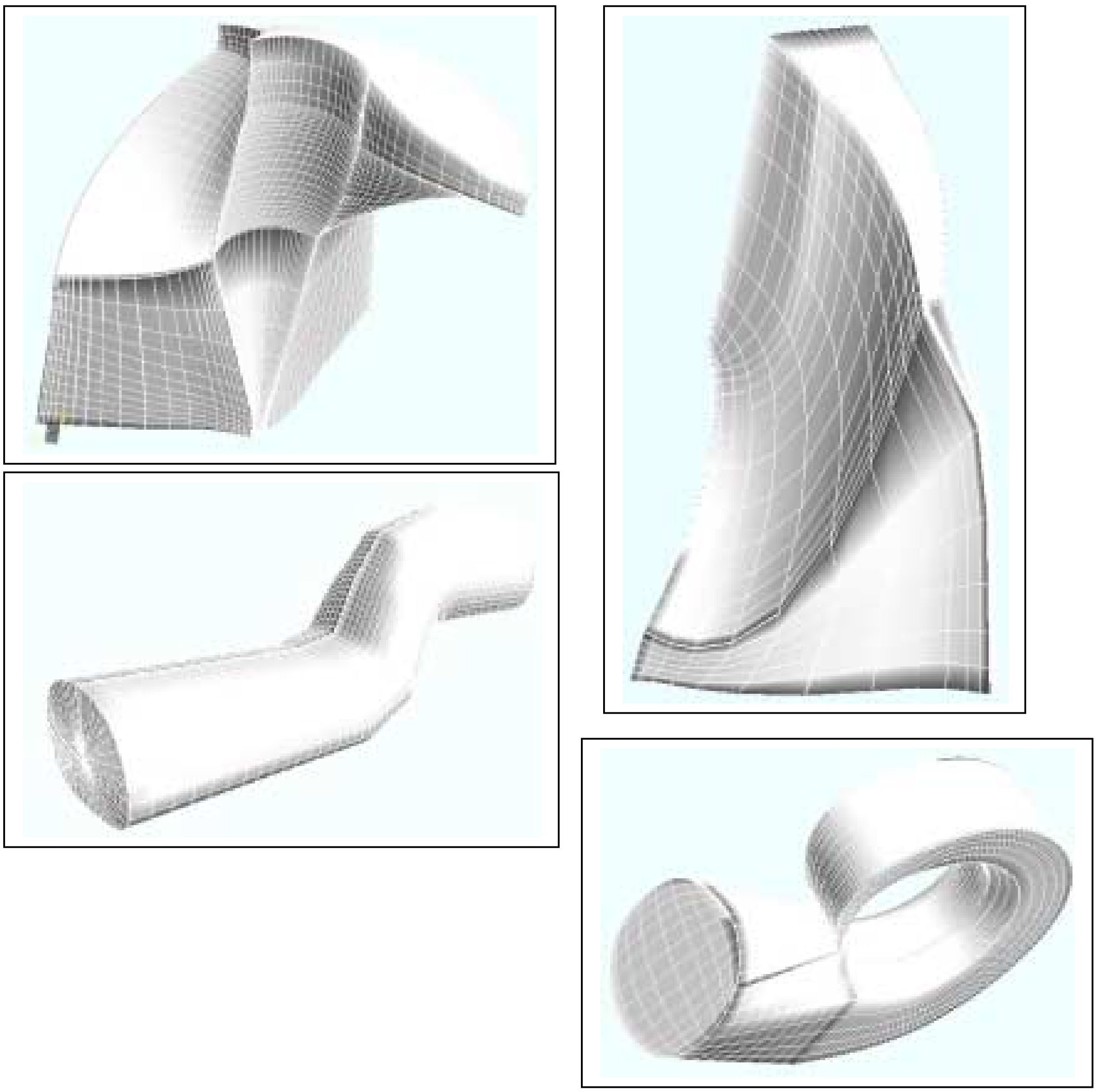
2. BASIC INFORMATION

Two aspects of the commercial CFD software package (TASCflow) had a key influence on the decision regarding the creation, for the industrial use, of the virtual hydraulic laboratory:

a) multi-block domain
b) capability of conducting calculations in the Multiple Frame of References

or each component of the hydraulic system of the water turbine the library of standard topologies is developed. Shapes of draft tubes, runner blades, wicket gates and variety of others intake structures are permanently stored in the database.

In order to create a structure representing an entire flow passage system of the turbine, the procedures for automatic block connecting was used. The most complicated structures were as large as sixty blocks.



DWG 1. Examples of grids for different shapes.

The goal of the first phase of testing is the calibration of the virtual model. Based on the fundamental data and measurements conducted in the existing power plant, the flow simulation is conducted for the existing geometry (including the turbine runner). Comparison of the operating parameters of both systems (real and the theoretical) is provided for the coefficients of flow, power and the hydraulic efficiency.

Q11 = Q/D^2/Sqrt(H)	- unit flow
n11 = n*D/Sqrt(H)	- unit speed
η	- hydraulic efficiency
D [m]	- turbine throat diameter
Q [m^3/s]	- flow
H [m]	- net Head as defined by the IEC code
n [rpm]	- turbine shaft speed
$\Omega = \pi$*n/30	

Value of the total pressure at the selected section is determined as a mass averaged total pressure

Ptot = Σ(ptot(i,j,k)*m(i,j,k0)/Σm(i,j,k)

The runner power output is calculated by the equation:

N = Np+Nt

where:

Np – power output resulting from the static pressure difference on opposite sided of the runner blade (pressure and suction sides)

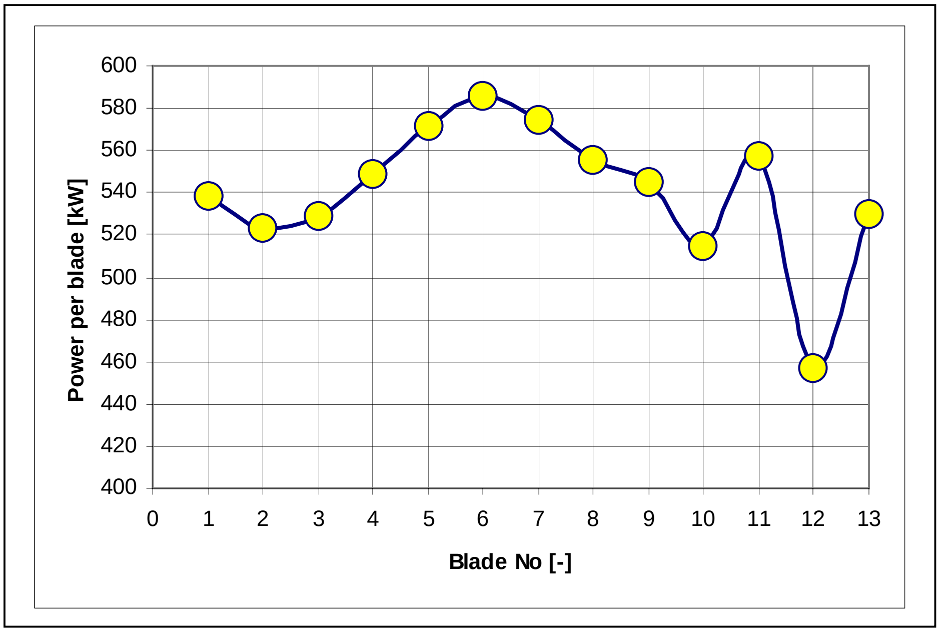
$$Np = \Omega * \sum_{\substack{i=ni \\ j=nj}}^{\substack{i=1 \\ j=1}} [p(i,j,1)* A(i,j,1)-p(i,j,nk))*A(i,j,nk)]*r(i,j,k)$$

Nt – power output resulting from the action of friction forces acting upon the runner blade

$$Nt = \Omega * \sum_{\substack{i=ni \\ j=nj}}^{\substack{i=1 \\ j=1}} [fv(i,j,1)* A(i,j,1)-fv(i,j,nk))*A(i,j,nk)]*r(i,j,k)$$

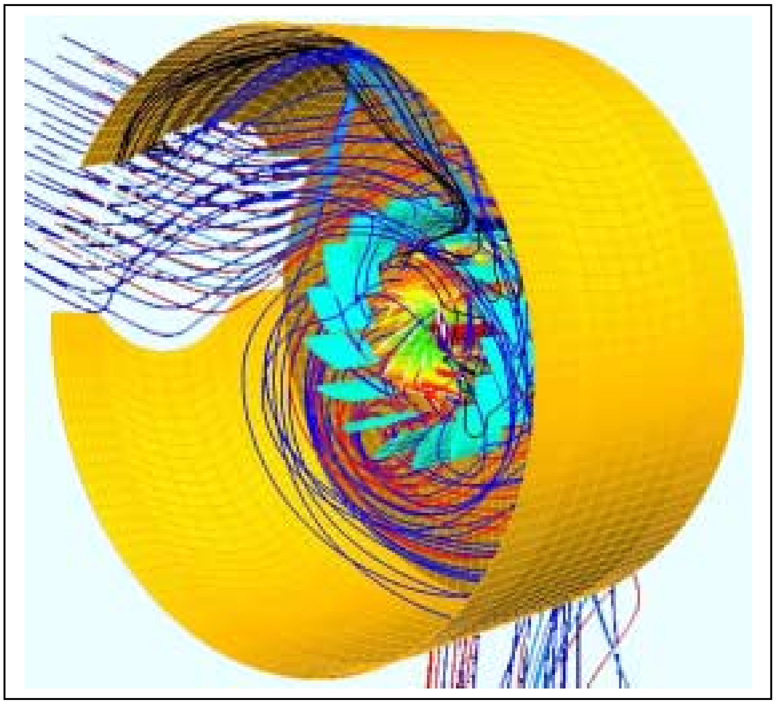
p(i,j,k) - static pressure at node (j,i,k)
fv(i,j,k) - viscosity tangent to the blade surface at node (j,i,k)
A(i,j,k) - blade surface area around node (i,j,k)
r(i,j,k) - radius at the node (i,j,k)
k = nk - index of the wall surface - suction sude of the blade
k = 1 - Pressure side of the blade

The above equations relate to the single runner blade; during the full flow simulation, the complete flow passage is analyzed (all wicket gates and runner blades), therefore the total power output is calculated as a sum of outputs produced by each blade. Mostly, at the preliminary phase of the calculations, only single blade-to-blade flow space is analyzed – the total output is then calculated by multiplying the number of blades by the output calculated for the single blade.
In case of the analysis with the “frozen rotor” option, the pressure distribution on different blades is always different depending on the angular position of the runner. In such a case, the instantaneous power output of the runner is calculated as a sum of outputs of all blades. Drawing 2 shows a graph representing output variation of the runner blades at a certain angular position of the runner.



DWG 2. Instantaneous runner power output calculated for each blade

The results of calculations are compared with the data collected during basic measurements, in order to assure that the conditions of the flow simulation correspond to the real circumstances. The turbine runner is usually the most significant source of energy loss, however, very often, partial responsibility can be attributed to the inlet structure, which may create highly uneven inlet conditions to the distributor (see Dwg. 3)



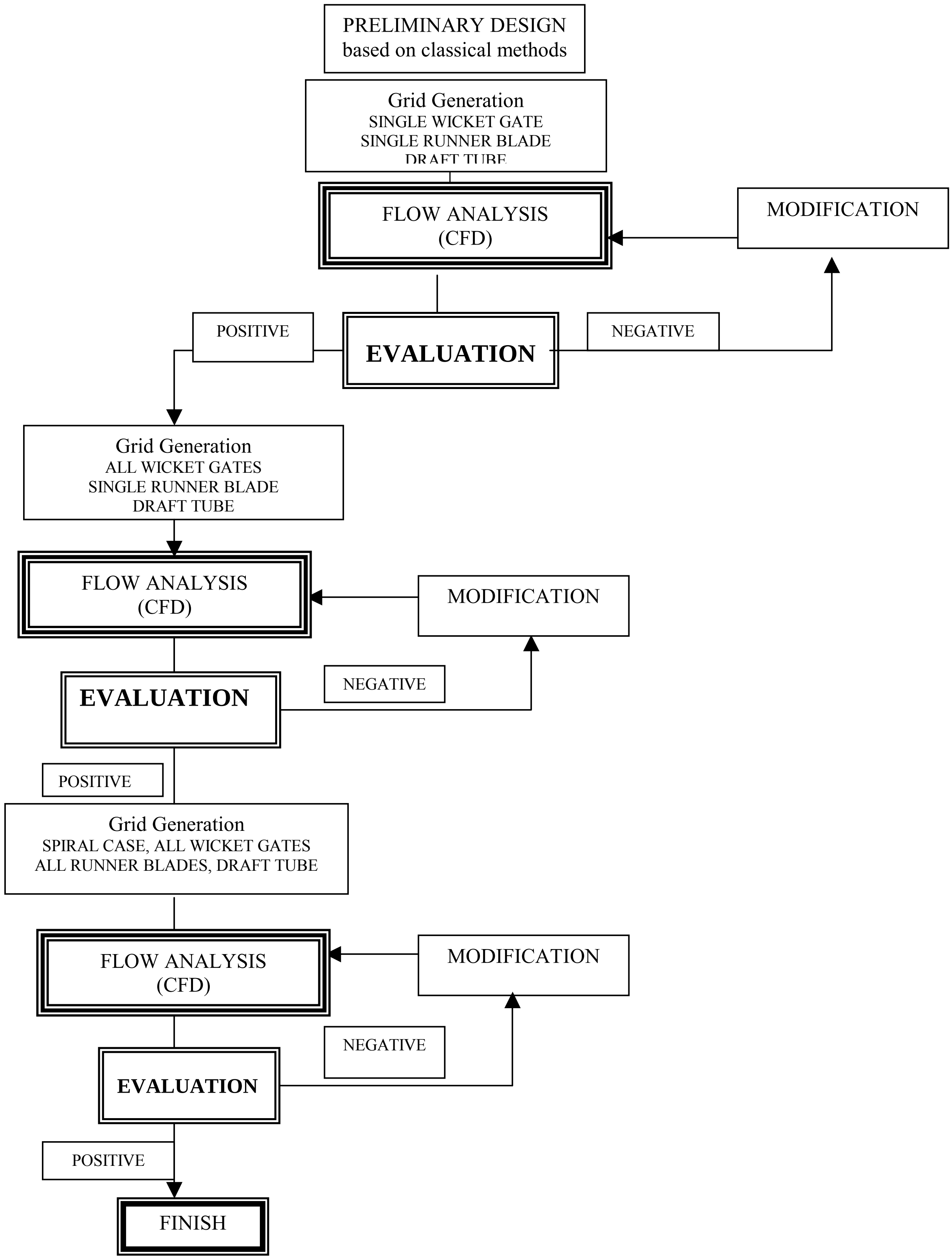
Dwg. 3 Example of highly non-uniform distributor inlet – selected streamlines in the structurally simple intake structure

In the practice of the system applications in last two years only once the modification into an intake structure proved itself to be economically useful. In the other cases the improvements of existing machines dealt with the designing and changing the runner and small changes in the draft chest area.

3. APPLICATIONS - OPTIMALISATION

The problem of the designing a replacement runner for an existing water turbine can be solved by using an algorithm presented below. The process of designing the runner blade is done by subsequent modifications of the runner blade shape until a needed performance of the whole turbine is achieved. Each replacement runner has to have relatively better operational parameters than the old one. It should lead to the higher electric power production and lower costs of maintenance. Therefore the optimized process of designing leads to:

A) maximization of both: power output and efficiency
B) minimization of the cavitation coefficient
C) assurance of stability in the widest possible operational range (variations of the head and flow)

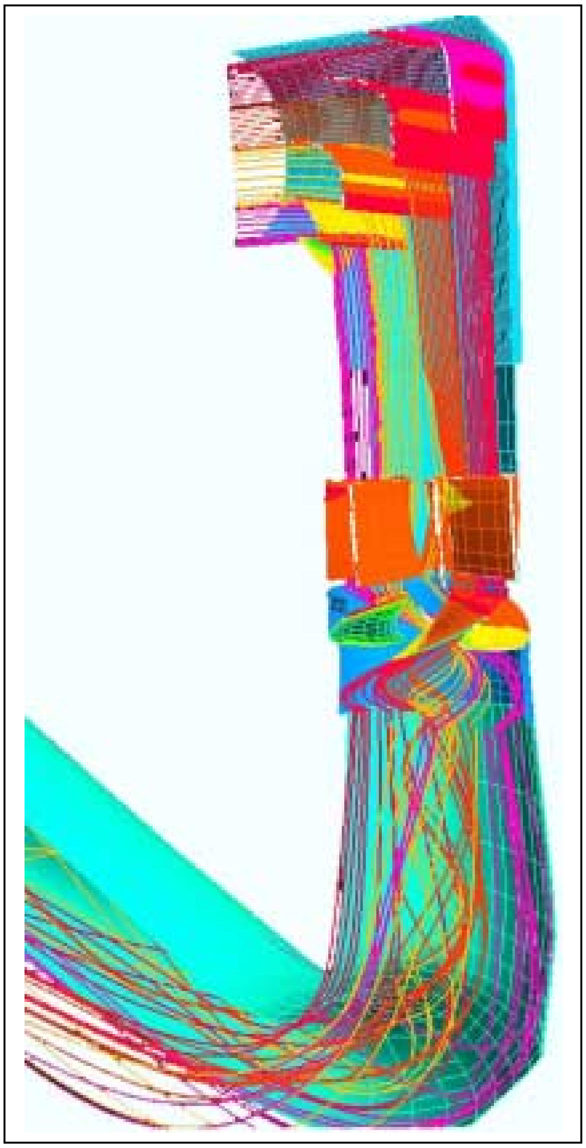


Dwg. 4 Design optimisation process

4. SELECTED EXAMPLES

4.1 Vertical semi-Kaplan turbine

The vertical shaft turbine equipped with a four bladed Kaplan runner, elbow draft tube, distributor and seven fixed wicked gates. The inlet elbow (90 deg) equipped with turning vanes is an integral part of an analyzed turbine flow passage as it is shown on the DWG. 5.
An existing turbine has been in operation, for the last 5 years, (approx. 40% of operational time) in a powerful cavitation area. Moreover the capacity of the installed generator was not fully utilized.
As the effect of the preliminary analysis the decision about designing the system wicked gates-runner was made. The intake structure consisted of horizontal penstock, elbow with the turning vanes and the vertical, short section of a penstock connected directly to the distributor.

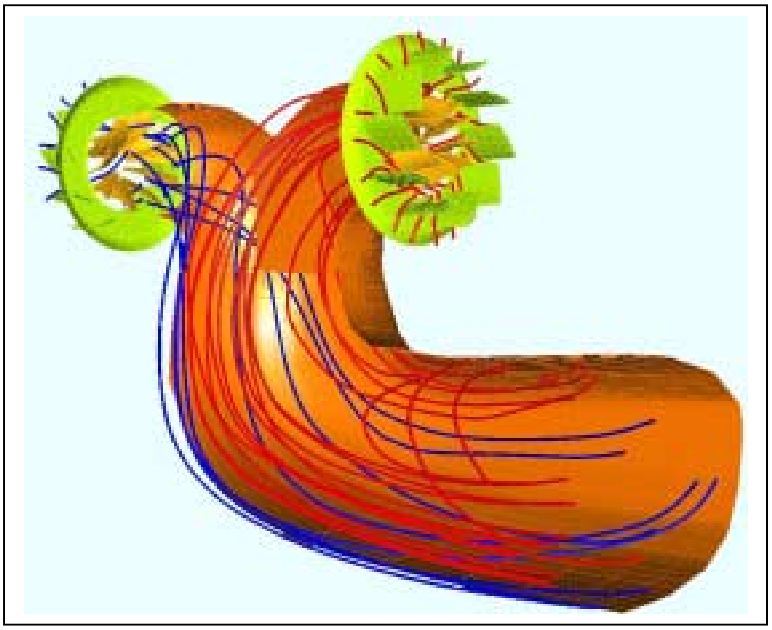


As a result of the flow analysis through the entire turbine, it was discovered that in the area of the elbow's turning vanes a very high hydraulic losses should occur. The upgrade of the turbine involved a new design of the runner, distributor and the turning vanes at the intake elbow. This modernization resulted in turbine output increase by 25%.

Dwg.5 Analysed flow passage of the Axial Flow turbine equipped with the regulated runner blades and the fixed stay vanes

4.2 Double horizontal Francis turbine

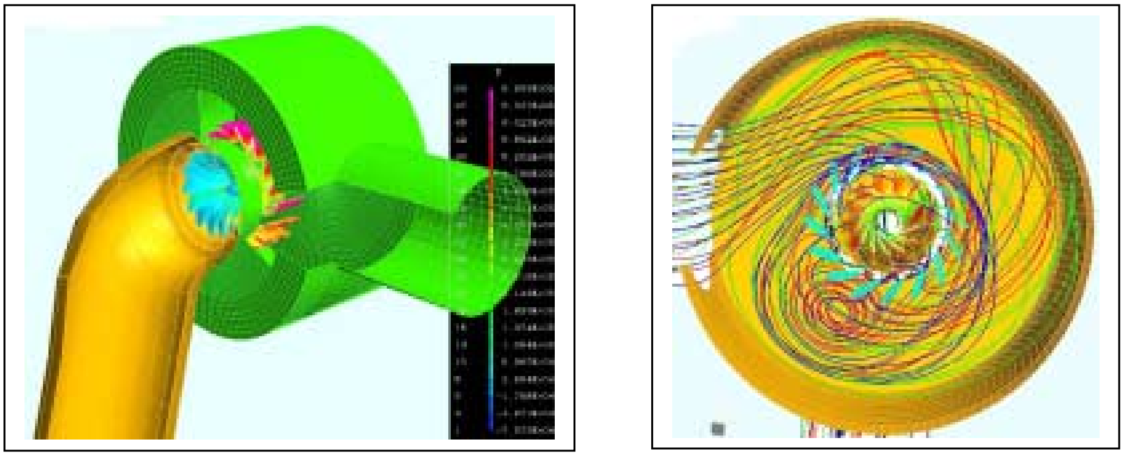
Replacement of the runner-distributor assembly in the double - horizontal Francis turbine (DWG 6). The open flume intake structure, camel-back type suction side of the turbine and the common draft tube. During the preliminary design stage, the simplified model was used: single wicket gate and blade flow passage at each side, plus full shape of the suction part of the unit (draft chest and draft tube). The final prediction of the turbine performances under different loads and heads were done based on the analysis results for the entire system (intake, all gates and all blades, draft chest, draft tube)



Dwg. 6 Configuration of the double - horizontal Francis turbine - hydraulic efficiency estimated based on the CFD results of such a structure is between 65% and 80% depending on the draft chest design. The runner efficiency has marginal influence on the efficiency of the entire unit.

4.3 Francis turbine in a cylindrical pressure case

A replacement runner for a Francis turbine set in a steel pressure case (DWG 4 and below). A high setting of the turbine and hydraulically complicated inlet structure caused difficulties in achieving client’s expectations of a 25% increase of the maximal power. The replacement of the whole turbine could not be economically justified. A full flow simulation as for the configuration shown on the Dwg. 7, was very difficult due to high instabilities of the flow. Careful observation of the results of the iteration steps (Dwg. 7b) and the operator’s comments about instability in the turbine performances led to assumed conclusion that the instabilities could not be sourced at the draft tube, but at the pressure case. (DWG 7b)

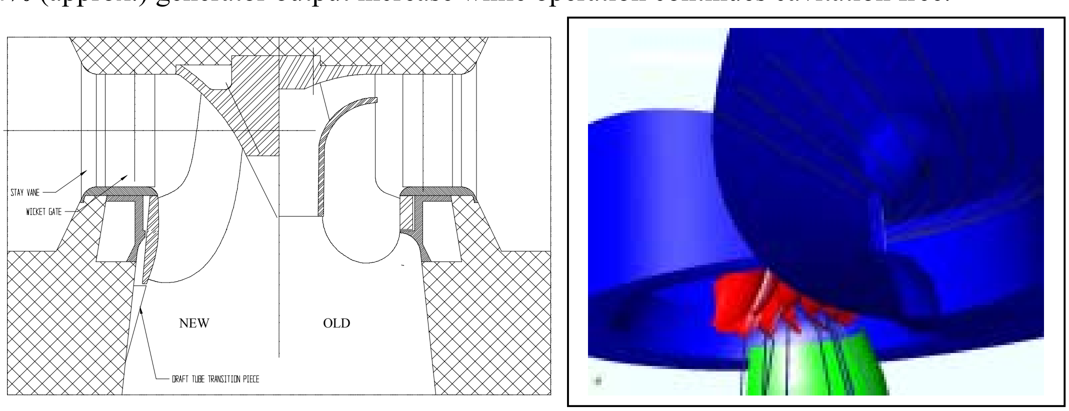


(a) (b)

Dwg. 7 Francis Turbine in a cylindrical pressure case.

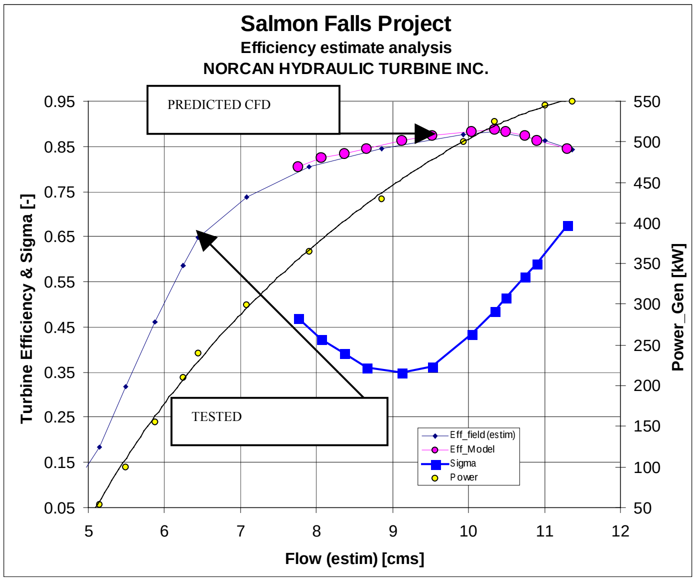
4.4 Vertical, semi-spiral case Francis turbine upgrade

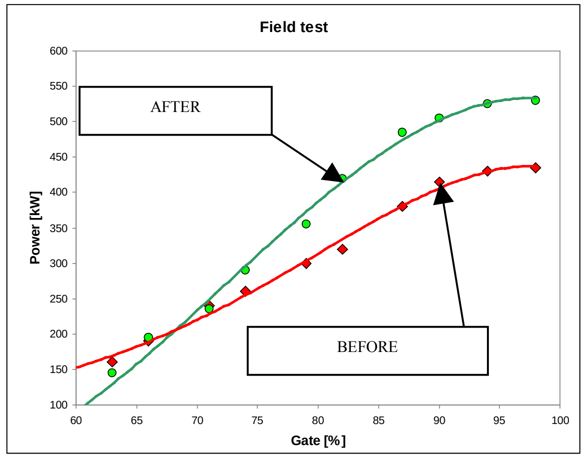
This is a classical example of a modern turbine upgrade, which has resulted in a significant power output increase . The new runner designed with the help of CFD was placed with in the existing system of stationary components. A relatively efficient existing spiral case, draft tube and the distributor did not create too many contstraints on the power increase. A very high turbine setting was, however, responsible for bad cavitation - repairs of the eroded runner blades had to be performed every odd year. The newly designed runner and the throat ring modification, achieved a 24% (approx.) generator output increase while operation continues cavitation free.



Dwg. 8 Replacement runner for the vertical Francis turbine .

a) comparison of the old and new runners outline (the throat modification)





Rys. 9 Expected and achieved - performances of the upgraded turbine

References

1. CFX - TASCflow Computational Fluid Dynamics Software, version 2.10 - manuals - AEA Technology, Engineering Software, 2000.
2. Gulliver J.S., Arndt R. : Hydropower Engineering Handbook, McGraw-Hill, inc., New York 1991

Authors

Jacek Świderski P. Eng.
Swiderski Engineering
1100 Ambleside Dr. Suite 1209
Ottawa, Ontario, K2B, 8G6
Canada
Tel (613) 829 - 8248
Email: jacek@achilles.net

Joseph N. Martin
Norcan Hydraulic Turbine Inc.
51 Bruce Cr
Carleton Place, Ontario,
Canada
Tel: (613) 257 - 4755
Email: norcan@primus.ca