Optimal Design of Hydraulic Turbine Distributor

Ahmed ALNAGA, Jean-Louis KUENY
National Polytechnical Institute of Grenoble
LEGI-ENSHMG
1025 rue de la piscine, BP95, 38402, Saint Martin d'Hères
France
ahmed.alnagat@honeywell.com
Jean-Louis.Kueny@epfl.ch

Abstract: *This work concerns the development of an automatic iterative procedure for optimal design of hydraulic turbine distributors. This procedure based on the geometry parameterization of the distributor to facilitate the fully automatic generation of the design by modifying the geometry parameters, and Evolutionary Algorithms to define the best design parameters using optimal functions (regrouped fixed objectives) evaluated from CFD results to decide the design quality. This procedure is successfully applied in two examples of Francis turbine distributors, the first with a specific speed equal to 81 and the second with a specific speed equal to 48. We obtained two new geometries with better efficiency and performance compared to the initial designs.*

Key-Words: Hydraulic turbines, geometry parameterization, flow simulation, evolutionary algorithms.

Nomenclature

Term	Symbol and definition
Low pressure external runner radius	R_e [m]
Radius	R[m]
Flow rate	Q [kg.m^{-3}]
Head	H [m]
Hydraulic energy	E [J.kg^{-1}]
Discharge coefficient	φ_e [-]=$Q/\pi R_e^3\omega$
Energy coefficient	ψ_e [-] = $2E/(R_e^2\omega^2)$
Specific speed	N [rnd/mn]
Unit specific speed	N_q [-]=$N\,Q^{0.5}/H^{0.75}$
Tangential velocity component	C_u [m/s]
radial velocity component	C_r [m/s]
Entrainment velocity	U [m/s] = $R\,\omega$

Indices
0: Runner outlet
1: Runner inlet
2: Distributor outlet
3 : Guide vanes inlet
4: Stay vanes outlet
5: Stay vanes inlet
6: Spiral casing outlet
Subscripts
^: nominal
nom: nominal
max: maximal
d: distributor
Abbreviation
GV: Guide vanes
SV: Stay vanes
CFD: Computational Fluid Dynamics
OF: Objective Function

1 Introduction

The design optimization procedure is described as an iterative process based on previous and initial designs; it consists in four steps (shown in figure 1) [1]:

· Geometry parametrization: it is necessary to establish a certain number of parameters able to represent the geometric entity that we want to improve. The most important factor is their number. A high parameters number may increase the shape manipulation complexity, where a low number may provide a poor and a limited range of feasible solutions. Further we describe the design procedure and parametrization of the Francis turbine distributor developed and adopted in the optimization process;

· Design performance evaluation: the design performance evaluation is obtained in two steps consisting in the domain discretization (grid generation) and the CFD simulation.

We used commercial tools for grid generation and design evaluation.

· Objective Function: there is no unique definition of an optimum solution. Moreover, the best solution changes from one case to another bearing in mind that engineering applications may involve complex multidisciplinary tasks.

· Optimization Technique: the relationship between the geometric parameters and their evaluation by CFD simulations is largely non-linear. So non-linear optimization techniques must be considered to drive the improvement. The optimization techniques we used are based on Evolutionary Algorithms.

Because the high number of the design evaluations, the different steps in the design optimization scheme are automatized by means of scripting files, relaying the different steps of geometry generation, domain discretization, simulation, etc.

The automation of the optimization sequence is accomplished by standard input-output exchange file formats handling the geometry and hydraulic parameters.

A design optimization cycle dealing with 3D shapes and CFD simulations usually requires a large computational effort for a single run. The RANS fluid flow simulations used for the geometry evaluation represents the most time-consuming step of the sequence here used, where 90% of the overall time-effort is expended in. Although a single run of the sequence requires a significant computational effort, the geometric parameterization and reduced number of parameters allow handling 3D shapes and limits the required design time.

Although the optimization based on EAs considered here requires large number of evaluations, it presents advantages such as global search and minimization so we do consider then suitable for this type of applications. ahmed.alnaga@honeywell.com .

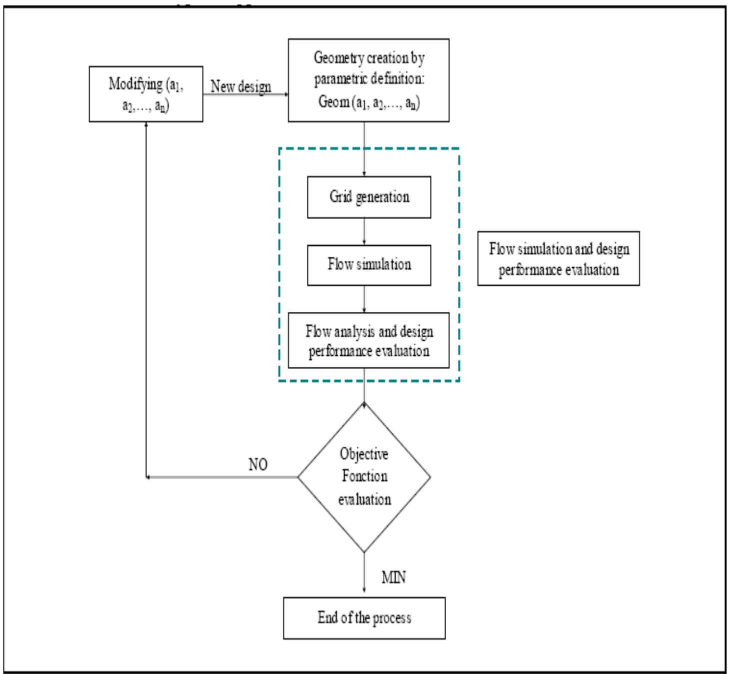
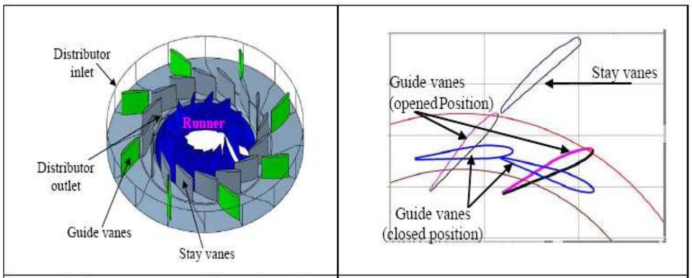


Figure 1: The design optimization procedure

2 Geometry generation

The first step in the optimization process is the geometry generation. According to the constant spanwise section of distributor, the parametric definition of the distributor can be reduced to a planar (r,q) representation (see figure 1.b). The 3D distributor shape is obtained by an extrusion of the 2D profile in the axial direction (z) (figure 1.a). (The parametric definition uses the 2D profile property to reduce the number of design parameters.)

In this part, we depict the objectives aimed in the design of the distributor. Then we describe the design procedure and the distributor parametrization that we developed and used to generate the geometry of the distributor in the optimization process [2].



1.a. Distributor + Runner of Francis turbine (3D view)

1.b. Distributor in planar (r,θ) representation (2D view)

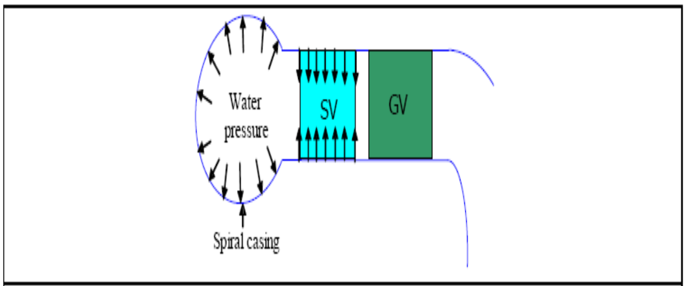


Figure 2 : spiral casing + distributor (meridianal view)

2.1 Design objectives

The distributor (Guide vanes and stay vanes) of reaction turbine is the element that ensures, for a nominal operation, the most favourable flow condition at runner inlet. It is placed between the Spiral-Casing and the runner in hydraulic turbines (see figures 1.a and 1.b).

The distributor is constituted mainly by two sets:

1. The stay vanes;
2. The guide vanes.

The distributor ensures two principal functions:

- Adapting of the turbine power to the operating conditions, by flow-rate monitoring;
- At the runner inlet, the favourable flow conditions at nominal speed.

The main objectives in the distributor design are:

1. Obtaining a maximum torque on the turbine shaft that corresponds to the highest energy provided to the turbine through the angular momentum given by Euler's equation:
$E=\Delta(UC_u)$
2. Avoiding shape deformation of the Spiral-Casing where the maximum pressure of the whole machine is located. (see figure 2);
3. The distributor has to be designed so as to minimize both the interferences with the flow and ensure that the flow reaches the runner with appropriate hydraulic angles;
4. Adjustment of flow-rate and the mechanical power by the manipulation of the angular position of the guide vanes;
5. Limitation of the energy losses at each opening rate of the distributor;
6. Resistance to extreme efforts (reduction of operating pressures);
7. Easy control of guide vanes, without excessive efforts;
8. Minimization of the manufacture cost.

2.2 Design procedure

First, we show the needed parameters used in the distributor design.

As the distributor consists in two parts (Stay vanes and Guide vanes) we will design each part separately, starting initially with the guide vanes and afterwards dealing with the stay vanes.

a. design parameters of guide vanes (see figures 4 and 5)

- Nominal radius of the distributor R_d.

The nominal radius of the guide vanes fixes the rotating axes of guide vanes blade. The choice of this radius is very important since it determines the construction, the weight and the cost of the turbine; it should be as small as possible.

- The number of guide vanes blade Z_d.

Z_d depends primarily on the dimension of the turbine. A small turbine will have a small number and vice-versa.

- Width of the distributor b_d.

Real width of the distributor is considered starting from statistics based on experiments.

- The flow angle α_{2d}.

The flow angle α_{2d}, formed by the blade guide vanes skeleton at its trailing edge and the peripheral direction, is shown in figure (4).

We have α_{2d} =atan(C_{r2}/C_{u2}) (figure 4) .

To determine the angle α2d at the distributor outlet, we follow these steps:

We have;

Erunner=g H_{net}-Σ of energetic losses (in spiral casing, SV, GV and runner)

With;

H_{net} is the net head.

Erunner is the Hydraulic energy transformed in the runner

We use Euler's equation between the distributor outlet (runner inlet) and the runner outlet.

Neglecting the energetic losses in the runner (U0 C_{u0} =0), we get:

Erunner= U_2 Cu_2 - U_0 Cu_0 = g H_{net} - Σ of energetic

losses (in spiral casing, SV, GV and runner)
Hence:

$$C_{u2} = \frac{g\,H_{net}}{U_2}$$

At the outlet of the distributor we have:
Q=Cr2 Surface(distributor outlet), with Surface(distributor outlet) =2 π $R_{d2}\, b_d$
Hence:

$$C_{r2} = \frac{Q}{2\pi R_{d2}\, b_d}$$

- Profile type and the blade characteristic dimensions.

Several types of profiles exist: Some with no camber chords, other with cambered chords.

The profile may be symmetrical or asymmetrical about the chord. It may also be hollow or with full section. The profile form may be selected among those recommended and tested from experimental benchmarks intended for aeronautics, such German profiles GÖTTINGEN or American NACA. One may resort profiles forms analytically expressed. The choice is very important especially to satisfy best the conditions of flows (less flow disturbing) at the runner inlet. In our case we chose a right skeleton profile with a NACA004 thickness distribution (see figure 3).

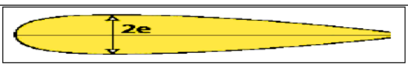


Figure 3 : Symmetric NACA profile

- Length of the guide vanes blade Ld:

The length of the profile of the blade is given by the rotating axis position of the blade guide vane:

$$L_d = \frac{2\pi R_d}{K\, Z_d} \quad \text{with } 0<K<1$$

- The angle of maximum opening γdmax:

The angle of maximum opening from the closed position γdmax depends on the maximum rate flow Qmax.

- Radius at guide vanes inlet and outlet R_{d3}, R_{d2}.

The guide vanes inlet and outlet radius are related to the maximum opening angle. It is fixed in such a manner to avoid all guide vanes overflow.

Ultimately, the design procedure of the guide vanes is obtained as follows:

1. Define the initial data related to turbine exploitation: H_{nom}, Q_{nom}, Q_{max} and N;
2. From the initial design, we fix the radius of the distributor outlet Rd1, and the distributor width b_d;
3. We use the Euler's relation to define the nominal flow angle $\hat{\gamma}_{2d}$ at the distributor outlet, that gives the guide vanes nominal opening angle;
4. After we define the guide vanes maximal opening angle γ_{dmax}, that is related to the maximal flow-rate Q_{max};
5. We estimate the number of the guide vanes blade Z_d (in our case we used the same number with the intial design);
6. We estimate the radius R_d that fixes the guide vanes rotating axis;
7. We define the blade guide vanes length;
8. At the end we chose the profile type and we define the inlet guide vanes radius.

b. Stay vanes parameters

For the stay vanes design, we start with the chord design. For this purpose, we use a 3rd degree Bezier curve, and we impose a thickness equal on both sides of the chord for profile design, with semi-circular leading and trailing edges.

The stay vanes design parameters are: α_{inlet} (Flow angle at inlet), α_{outlet} (Flow angle at outlet), (R_{d5}-R_{d4}), δR (between the stay vanes and the guide vanes)= (R_{d4}-R_{d3}), and the angular position of the stay vanes trailing-edge θ_{BFavd} and leading-edge θ_{BAavd}(see figures 4 and 5).

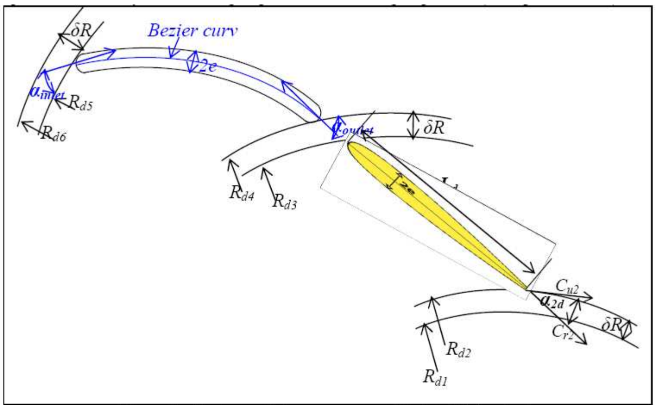


Figure 4: distributor design parameters (blade to blade view).

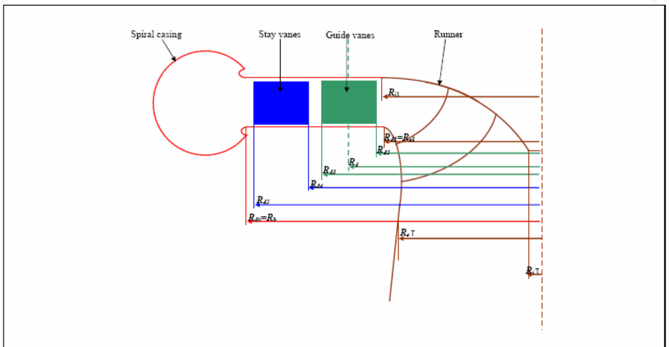


Figure 5: distributor design parameters (meridian view)

c. Examples

In this part, we present a few distributor design examples (guide vanes and stay vanes) by modifying some mentioned above parameters (see figure 6).

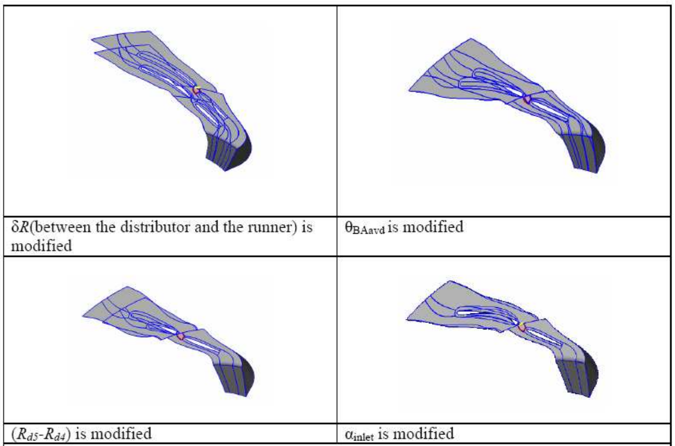


Figure 6: Distributors obtained by design parameters modification.

The design parameters defining the distributor geometry are expressed as a sequence of parameters in an exchange file.

Some part of the design parameters constituting the geometry are used as design variables in the optimization process, while some remained fixed, in order to meet the constraints imposed by the original design, and other design parameters remain fixed by tests in order to minimize the design parameters.

The choice of the design parameters is motivated mainly by the operating conditions and the machine constraints. The constraints differ from the studied turbine components to others and must be accurately chosen to ensure fair optimized solutions.

The parametric representation of the distributor uses 24 design parameters to generate the distributor geometry.

The design parameters fixed by tests and initial design are: Z_d(GV), Z_d(SV), δR(GV-SV)=(R_{d4}-R_{d3}), R_{d1} , e, and b_d .The design parameters, used as design variables in the optimization process, are: δR (Runner-distributor)= (R_{d2}-R_{d1}), (R_{d5}-R_{d4}), α_{BAavd}, α_{BFavd} and θ_{BAavd}.

3 Design Performance Evaluation

The evaluation of a geometry is performed by evaluating the prescribed OF, from hydraulic performances. The evaluation is obtained from a two step process: grid generation and CFD simulation.

a. Grid generation

The domain discretization step must be automatized as well. The automatic grid generator has to ensure grids robustness and the quality for the range of generated shapes. In this case, we use the commercial aided tool Autogrid5®. Autogrid5 is a grid generator specially developed by (NUMECA®) for turbomachinery applications. Among the features, it provides the capability to automate the generation of the grid for new blade shapes by defining the topology parameters. Autogrid uses a predefined template file where main parameters defining the suitable mesh to be applied, i.e. size, blocks number, topology (H-O-H, H-I), clustering, Tip-Hub clearance, etc., are stored. The same template file is applied for the geometries generated during the optimization process.

This template file is set for a given turbine design components and the grid topology parameters are adapted according to the specifics of current design.

The generation of the grid is a critical step in the optimization procedure. The robustness of the grid generator allows obtaining valid and consistent results for optimization success. For this reason, it is necessary to determine accurately the proper size and topology of the grid being generated.

To determine the appropriate grid size we consider the conclusions intuitively drawn by Denton, where he quotes the main hydraulic parameters in turbomachinery could be captured with a coarse grid. Similar assumptions were applied by Benini. Thus we must consider that for design optimization purposes instead of large grids and accurate simulations, coarse adapted grids and fast simulations achieving good performances to capture the mean flow tendencies are preferable.

The minimization searching path depends on the evaluation of the OF computed from the simulations results. The accuracy of such results is necessarily obtained with the precision given by the grid. We survey and check the influence of the mesh in the optimization during the optimization and finally a check is performed over the optimized solutions by performing the simulations using more fine grids probing that same OF tendencies can be obtained.

The domain discretization of the distributor domain is obtained from Autogrid5. The structured grid provided by Autogrid is made up by five blocks

with (H-O-H) topology for the stay vanes and by five blocks with (H-C-H) topology for the Guide vanes. This topology is found to provide the most robust grid configuration for a large number of checked geometries.
The quality of the grid is checked at each run, avoiding unwanted grid angles and grid dependence. The minimum (≤ 20 degrees) orthogonality and aspect-ratio criteria are mainly surveyed. Figure 7 presents the typical grid topology thus generated.
The grid is constituted by approximately 400000 nodes. This grid dimension allows obtaining quality grids with a limited grid size. Moreover, this size provides good convergence rate on the CFD simulations.

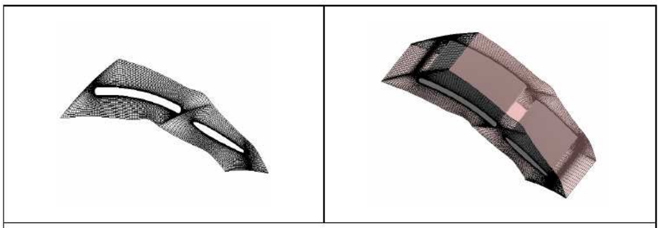


Figure 7: Distributor grid

b. CFD simulation

Regarding the flow simulations, the optimizations are been performed using the commercial solver Fine/Turbo®. RANS simulations and (k-ε) turbulence models are used. Steady-state simulations over a single blade-to-blade channel conditions are computed, where usual periodicity and symmetry conditions are used [5].
The flow conditions are imposed at boundaries related to Mass Flow Rate, flow angles at inlet and Averaged Static Pressure at the outlet. At the interface between the stay vanes and guide vanes we used a Frozen-Rotor interface.
The automatic execution of the simulations is accomplished by coupling the grid generator with the CFD solvers. The CFD simulation setup is defined in template files running in a preprocessing sequence. Once the problem is set, the simulation solver is run until either convergence or stopping criteria is achieved.
The results are post-processed and the OF is evaluated.

4 Objective function and constraint

Optimization is the process of maximizing or minimizing a desired objective function with the combination of independent variable parameters while satisfying some constraints.
The formulation of the OF used in each optimization must be adequately chosen in relation with the component and its functionality.
In the optimization domain, two main approaches may be distinguished for achieving the optimum solution. One is the SOO, when single or multiple objectives are grouped in a unique expression to be optimized. This approach *a priori* implies definition of the importance of each factor in the set. The other approach is called a MOO, this time no relation is established between the different objectives, thus each of the objectives is optimized independently [6].
In our case for the optimization of Francis turbine distributor we used the first approach (SOO). Hence the optimization problem is formulated as minimizing of single-objective function $f(x)$ in the form:

$$f(x) = \sum_{i=1}^{n} w_i f_i(x)$$

Where n is the number of objective functions f_i, w_i represents weight coefficient that balances each objective in the set with respect to the others and x is the optimization (vector) parameter, $x \in X$, where X is parametrization space. Weight vector is determined by the designer. Thus for each combination of weights we get another optimal solution. For this work all optimizations, performed by aggregate weighted methods, use an arbitrary definition of the weights. However, other research approach uses variable adjusted weights. If the constant approach provides a single Pareto Optimal solution, because of the invariant values of the weights, the second approach provides information similar to that obtained using multiobjectives optimization. For such cases it is possible to identify the influence of the different objectives in the definition of the optimal solution.
The Objective function used in the distributor optimization intends to minimize the energetic losses, to meet fluid flow constraints and to minimize the cost of construction. Thus a single expression taking into account both objectives is proposed as OF. The first term is related to the energetic losses computed as the inlet outlet Total Pressure variation, the second term is the outlet

angle variation geometry and the third is the surface of the distributor written as:

$$FO = C_1 \frac{\Delta P_t}{\rho g H} + C_2 \Delta\beta_{\max} + C_3 \frac{Surface_{SC}}{Surface_{ref}}$$

where:
- ΔP_t is the difference of the total pressure between the inlet and the outlet of the distributor, which represents the energetic losses in the distributor;
- Δbmax represents the maximum change between the relative flow angle at the distributoroutlet;
- $Surface_{sc}$ is the spiral casing surface that represents the construction cost;
- $Surface_{ref}$ is the reference spiral casing surface;
- C_1, C_2, and C_3 are the weight coefficients.

Note that improving one objective is equivalent to set the other coefficients to zero.

Constraint

The constraint fixed in the distributor design is to obtain at the distributor outlet a (UC_u) outlet (kinetic moment) that corresponds to the objective fixed (UC_u nominal for the initial existing geometry).

5 Optimization algorithm

The design optimization examples presented here are performed by EASY® (Developed by NTUA) software using Single-Objective optimization.

EASY uses EAs (Evolutionary Algorithms) and ANN (Artificial Neuronal Network) to find optimum solution [8] and [9].

6 Applications

We present here two examples for two distributor design optimizations for two existing industrial Francis turbines with specific speed equal to 81 (rapid turbine) and 48 (slow turbine), in order to show the effectiveness of the optimal design procedure developed and used for the distributor of Francis turbine is independent of the geometry shape.

First, we make analysis of the initial existing geometry of Francis turbines. The objective of this step is to find the nominal operating conditions of each turbine. Around these conditions will be our objective afterwards to make the new optimized design of the distributor.

6.1 Application1: Distributor design optimization of Francis turbine *Nq*81

a. Analysis of the initial turbine *Nq*81

We start with the description of the Francis turbine ***Nq*81**, it's an existing industrial turbine, which characteristics are presented in table 1:

$H_n\,[m]$	87.5	Z_d (Runner)	13
$Q_n\,[m^3/s]$	60	Z_d (GV)	24
$N\,[rnd/\min]$	300	Z_d (SV)	24
$n_q\,[-]$	81		

Table 1: Francis turbine N_q**81** characteristics

The complete Francis turbine is represented in the follow figure:

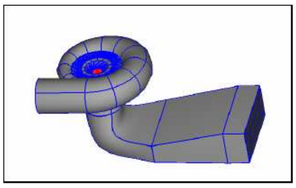


Figure 8: Francis turbine N_q**81**

CFD calculations are performed for guide-vanes opening (a_{2d}) of 16°, 20°, 22°, 24°, 28° and 31.4° and for flow-rate varying between 30m3/s and 100m3/s in three steps [3] [4] and [7]:

1. Spiral casing/stay vanes computation;
2. Distributor/runner computation;
3. Draft tub computation.

The grid and calculus parameters for each step are the following.

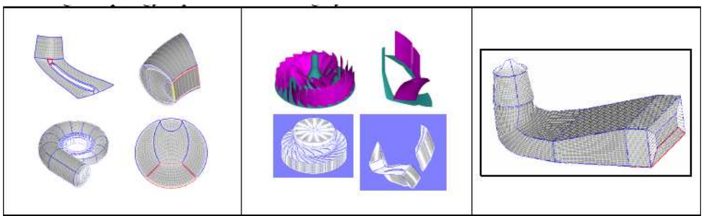
Grid generation

The domain discretization of the Spiral casing (figure 9.a) is obtained from IGG® (commercial tools) a structured butter fly multi-blocs grid topology is provided with approximately 400000 nodes.

The domain discretization of the stay-vanes is obtained from Autogrid5. A structured H-O-H multi-blocs topology with approximately 1200000 nodes is generated.

The domain discretization of the guide-vanes and runner is obtained from Autogrid5. A structured H-O-H multi-blocs topology with approximately 800000 nodes over a single bladeto-blade channel is

computed, where usual periodicity and symmetry conditions are used (the
grid of the stay vanes, guide vanes and runne is shown in figure 9.b). Finally the domain
discretization of the draft tube (figure 9.c) is obtained from IGG. A structured butter fly multiblocks grid topology is provided with roughly 400000 nodes.



a. Spiral casing grid | b. Distributor + runner grid | c. Draft tube grid

Figure 9: Geometry grid

Flow simulation

Regarding the flow simulations, the calculus are been performed using a commercial solver
FineTurbo. RANS simulations and (k-ε) turbulence models are used with Steady-state simulations.

For the Spiral casing/stay vanes calculus at the interface between the stay vanes and spiral casing, we used a NMB (Non Matching Boundary) interface.

For the Distributor/runner calculus at the interface between the stay vanes and guide vanes, we used a Frozen-Rotor interface while at the interface between the guide vanes and runner we used a mean stage interface.

The flow conditions for each calculus are imposed at boundaries related to Mass Flow Rate flow angles at inlet and Averaged Static Pressure at the outlet.

Flow analysis

In figure 10 we show the efficiency-hill of the Francis turbine obtained from numerical simulation.

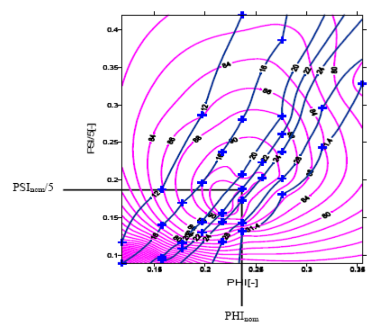


Figure 10: Efficiency-hill of turbine N_q81

The nominal operating condition for the Francis turbine corresponds to the point: PHI_{nom}= 0.298 and PSI_{nom}= 3.26, that is for H_{nom}= 88.941m (not far from 87.5m) and Q_{nom}=60m3/s at the opening of 22°<=> UC_u=825 J/kg.

The distributor design optimization constraint is set to be this result.

b. Distributor design optimization

The main settings applied to the distributor design optimization process are described in the following.

Performances Evaluation

We used the same grid and simulation parameters described in section 2.

Objective Function

The OF use for this example is defined as follow.

$$FO = C_1 \frac{\Delta P_t}{\rho g H} + C_2 \Delta\beta_{\max} + C_3 \frac{Surface_{SC}}{Surface_{ref}}$$

With the weights C_1 = 100, C_2 = C_3 = 1 obtained from an initial evaluation so to balance the different objectives.

Optimization

The distributor design optimization performed uses the SOO configuration of EAs. The optimizations for both cases are stopped when reaching the maximum number of evaluations, which is set to 400. The computational resources are the same (a single-processor computer).

The time required for the optimization to converge is estimated to 15 days to 21 days.

Results

Now we show the results obtained for the distributor design optimization of the Francis turbine *Nq*81.

Figure 11 shows a comparison between the UC_u obtained by numerical calculation at the outlet of the distributor for each geometry created in the optimization process and the *UCu* for the initial existing geometry. It appears clearly that along the optimization process each designed geometry has approximately the same *UCu* as the initial existing geometry.

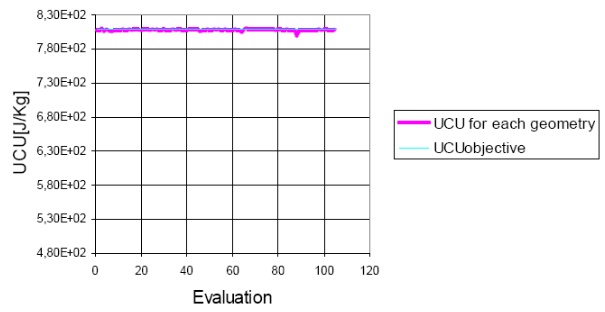


Figure 11 : Comparison of UC_u at distributor outlet with UC_u objective

Figure 12 shows the evolution of the total objective function and each objective alone along the optimization process. We can see a good convergence of the total objective function, with minimization of the energetic losses, the maximum change between the flow angle at the distributor outlet and the minimization of the spiral casing surface (decreasing of 5%). This convergence is reached around 100 evaluations.

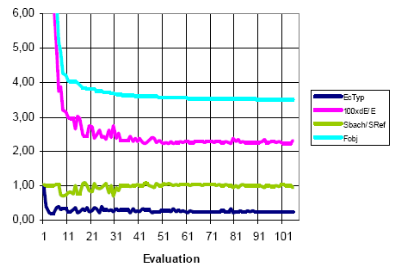


Figure 12 : Objectif fonction N_q81

Figure (13) shows a comparison between the distributor initial geometry and the optimized one.

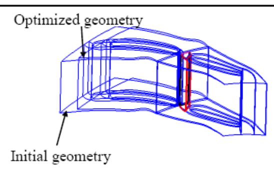


Figure 13 : Initial and optimized distributors (N_q81=)

We present in figure 14 a comparison between the efficiency of initial runner/initial distributor and initial runner/optimized distributor. An efficiency improvement of the optimized geometry distributor is shown.

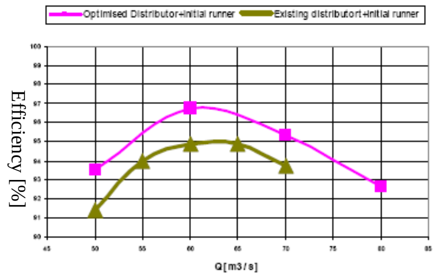


Figure 14: Efficiency comparaison (Nq81)

6.2 Application2: Distributor design optimization of Francis turbine *Nq*48

a. Analysis of the initial turbine *Nq*48

Characteristics of Francis turbine ***Nq***48 are presented in the table 2.

$H_n\,[m]$	66.5	Z_d (Runenr)	13
$Q_n\,[m^3/s]$	1.25	Z_d (GV)	12
$N\,[rnd/\min]$	1000	Z_d (SV)	12
$n_q\,[-]$	48		

Table 2: Characteristics of Francis turbine N_q48

Calculations are performed for guide-vanes opening of 20°, 24°, 28°, 31.2° and 35° and for flow-rate various between 1.0 m3 /s and 1.6 m3/s in the same fashion like turbine ***Nq***81.

We used the same grid and flow simulation parameters defined in Francis turbine *Nq*81 as well. The nominal operating condition for the Francis turbine corresponds to the point:

PHI_{nom}= 0.298 and PSI_{nom}= 2.247, that is for H_{nom}= 68.548 (not far from 66.5m) and

Q_{nom}=1.25 m3/s at the opening of 24°<=> UC_u=625 J/kg.

The distributor design optimization objective is set to be this result.

b. Distributor design optimization

The main settings applied to the distributor design optimization process are the same use with the Francis turbine distributor *Nq*81. While the OF used is:

$$OF = 25\frac{\Delta P_t}{\rho g H} + \left(\Delta\beta_{\max}\right)^2 + \frac{Surface_{SC}}{Surface_{ref}}$$

Results

Figure 15 shows a comparison between the *UCu* obtained by numerical calculation at the outlet of the distributor for each geometry created in the optimization process and the kinetic moment that constituts our objective. No significant difference is hence noticed as can be seen.

Figure 16 shows the evolution of the total objective function and each objective alone along the optimization process. We notice a good convergence of the total objective function, with minimization of the energetic losses, the maximum change between the flow angle at the distributor outlet and the minimization of the spiral casing surface (decreasing of 3%).

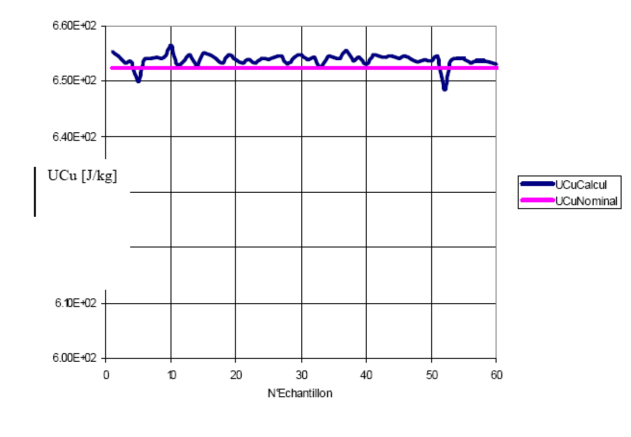


Figure 15 : Comparison of UC_u at distributor outlet with UC_u objective (turbine N_q48)

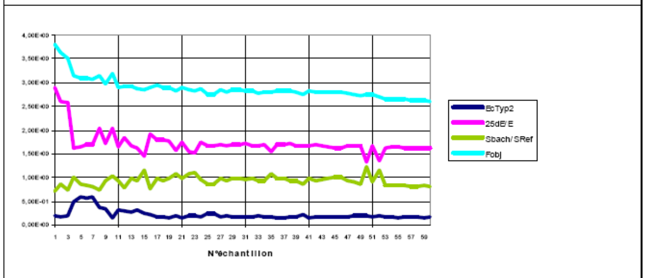


Figure 16 : Objectif function versus optimization process (N_q48)

Figure 17 shows a comparison between the initial geometry of the distributor and the optimized one.

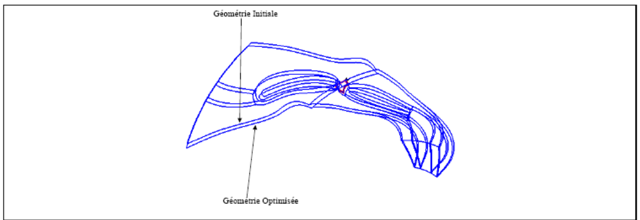


Figure 17 : Initial and optimized distributor (turbine N_q48)

Figure 18 represents a comparison between the efficiency of initial runner with initial distributor and initial runner with optimized distributor. Here also, an significant improvement of the efficiency of the optimized geometry distributor is noticed.

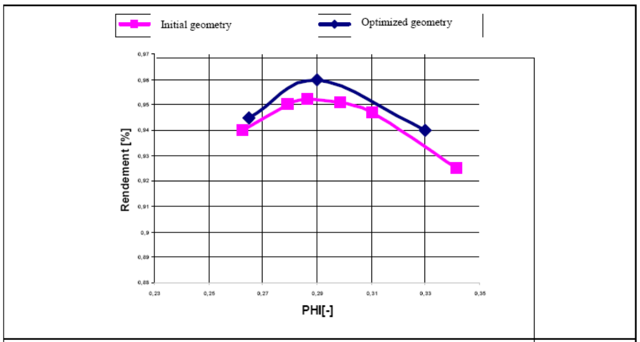


Figure 18 : Efficiency comparison (N_q48)

6 Conclusion

We have developed a specific fully automatic Francis turbine distributor optimization process.

To test its effectiveness, we performed two examples of Francis turbine distributor optimization, the first with specific speed equal to 81 and the second with specific speed equal to 48. Optimization objective concerned three parameters. The first parameter is related to the energetic losses computed as the inlet/outlet total pressure variation. The second parameter, related to flow disturbance, is the outlet angle variation geometry. And the third parameter, related to the geometry cost, corresponds to the spiral casing surface. In both cases, we did obtain two new geometries with best efficiency and performance from the initial ones.

Optimization time was reasonable. Efficiency improvement reached approximately 1.5 %, while spiral casing surface reduced to (3-5) % and energetic losses to (1-2) %. Furthermore, we believe that this optimization procedure is extendable to other hydraulic turbines in a straight forward manner.

[1] Avellan F. Pedretti C. Tomas L. Ferrando L., Kueny JL., Surface parametrization of a francis runner turbine for optimum design, Proceedings of the XXIInd IAHR Symposium on Hydraulic Turbomachinery and Systems, 2004.

[2] Bovet TH., Distributor of hydraulic turbine, publication N°114, Lausanne 1970.

[3] Denton JD, Turbomachinery design using cfd, AGARD Lecture Series 195 (1994).

[4] Benini E., Optimal navier-stokes design of compressors impellers using evolutionary computation, International Journal of

Computational Fluid Dynamics 17 (Oct. 2003), no. 5,357–369.
[5] Alex GUEDES, Jean-Louis KUENY, Gabriel Dan CIOCAN, François AVELLAN, Unsteady rotor-stator analysis of a hydraulic pump-turbine – CFD and experimental approach,
21st IAHR Symposium September 2002, Lausanne
[6] Randy L., Sue E H., Practical genetic algorithms, john wiley & son, 1998.
[7] Kueny J-L., CFD analysis report in Francis turbines, (EPFL) Lauzanne November 2002.
[8] Ll. Ferrando, J.L. Kueny, F.Avellan: Optimal Francis Runner Design: Preliminary
Tests.Design Optimization International Conference. March 2004. Athens Greece.
[9] Ll. Ferrando, J.L Kueny, C. Pedretti, L.Tomas : Surface Parameterization of a Francis Runner Turbine for Optimum Design. 22nd IAHR Symposium on Hydraulic Machinery and Systems. July 2, 2004 Stockholm Sweden.