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Projet
Caractéristiques des pompes fonctionnant en turbines

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ABSTRACT

Thanks to the partnership between OFEN, Sulzer, EIG and HEVS important improvements for the determination of performance of pumps as turbines and for the use of small variable speed generators in small power plants have been done.

The method proposed is based on the determination of a model which computes the total head losses in pump operation with the help of the pump geometry. Then, this model is applied to determine the characteristics in turbine operation.

The pump model has been divided in elements connected in series and in parallel from the inlet to the exit of the pump. The calculation of hydraulic losses in each element (friction losses, bend losses, gradual expansion and contraction, sudden expansion or contraction) has been performed and the sum of those gives the total head losses through the pump.

For this project, 4 pump geometries have been used in order to verify the model. One belongs to the EIG and measurements have been performed to determine the geometry and the pump and turbine characteristics. The other 3 geometries and their characteristics have been provided by Sulzer Pumps.

As results, the model of losses allows a prediction of the head at the best efficiency point (BEP) in pump operation with a shift between 4 and 5 %.

In turbine operation, by comparing the turbine characteristics calculated and the experimental data, a shift from 2 to 5% is observed for the head at BEP.

The next step of the project will be to improve the model in order to determine the flow rate at the BEP, then with the help of the head already predicted, the efficiency will be determined.

A software programmed in Excel is going to be prepared in the next step and it will be available for free use.

At the HEVS, a laboratory test rig has been built and is composed of a permanent magnet synchronous generator as well as a frequency converter developed at this institution. The test rig allows testing the control algorithms. The results show a good efficiency at part load operation. A paper concerning this generator has been published.

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SYMBOLS

A	area, cross section
A1q	impeller inlet throat area (trapezoidal: A1q = a1 b1)
A2q	area between vanes at impeller outlet (trapezoidal: A2q = a2 b2)
A3q	diffuser inlet throat area (trapezoidal: A3q = a3 b3)
a	distance between vanes
b	width of vane
b2	impeller outlet width
c	absolute velocity
c_u	circumferential component of the velocity
c_{3q}	average velocity in diffuser throat $c_{3q} = Q_{Le}/(Z_{Le}\ A_{3q})$
D, d	diameter
d_{3q}	equivalent diameter of volute throat
d_b	arithmetic average of diameters at impeller or diffuser e.g. $d_{1b} = 0.5\,(d_1 + d_{1i})$; defined such that: $A1 = \pi d_{1b}\ b_1$
d_m	geometric average of diameters at impeller or diffuser, e.g. $d_{1m} = \sqrt{0.5(d_{1a}^2 + d_{1i}^2)}$
e	vane thickness
g	acceleration due to gravity (g = 9,81 m/s2)
H	head
Hp	static pressure rise in impeller
k	rotation of fluid in between impeller and casing
L	length
n	rotational speed (revolutions per minute)
n_q	specific speed (min-1, m3/s, m)
p	static pressure
p^*	wet perimeter
p_{amb}	ambient pressure
Q	flow rate, volumetric flow
Q_{La}	flow rate through impeller: $QLa = Q + Q_{sp} + Q_E + Q_h = Q/\eta_v$
Q_{Le}	flow rate through diffuser: $Q_{Le} = Q + Q_{s3} + Q_E$
Q_E	flow rate through axial thrust balancing device
Q_h	flow rate through auxiliaries (mostly zero)
Q_{sp}	leakage flow rate through impeller neck ring
Q_{s3}	leakage flow rate through inter-stage seal
Q*, q*	flow rate referred to flow rate at best efficiency point: $q^* = Q/Q_{opt}$
Re	Reynolds number. Channel: $\mathrm{Re} = cD_h/\nu$. Blade: $\mathrm{Re} = wL/\nu$

R_o	bend radius
r	radius
r_{3q}	equivalent radius of volute throat area
s	gap width
t	pitch $t = \pi d / Z_{La}$ (or Z_{Le})
u	circumferential velocity $u = \pi d\, n/60$
w	relative velocity
w_{1q}	average velocity in impeller throat area $w_{1q} = Q_{La}/(Z_{La}\ A_{1q})$
Z_{La}	number of impeller blades
Z_{Le}	number of diffuser vanes (volute: number of cutwaters)
α	angle between direction of circumferential and absolute velocity
α	opening angle of a diffuser
β	angle between relative velocity vector and the negative direction of circumferential velocity
ΔH	hydraulic losses
δ_0	bend angle
ζ	loss coefficient
η_{vol}	volumetric efficiency
θ	angle used for the law section of the spiral casing
λ	angle between vanes and side disks (impeller or diffuser)
ρ	density
ν	kinematic viscosity
τ	vane blockage factor

BEP	Best efficiency point
PAT	Pump as turbine

Subscripts

0	bend
1	impeller blade leading edge (low pressure)
2	impeller blade trailing edge (high pressure)
3	diffuser vane leading edge or volute cutwater
a,m,i	outer, mean, inner streamline
B	blade angle (impeller, diffuser, volute cutwater)
d	discharge nozzle
fr	friction
GC	gradual contraction

GE	gradual expansion
hyd	hydraulic
i	element i of the volute
La	impeller
Le	diffuser
m	meridian component
max	maximum
min	minimum
opt	operation at maximum efficiency (BEP)
r	radial
SC	sudden contraction
SE	sudden expansion
sp	gap, leakage flow
Sp	volute
th	theoretical flow conditions (flow without losses)
tot	total pressure = static pressure + stagnation pressure
u	circumferential component
*	dimensionless quantity: all dimensions are referred to d_2 e.g. : $b_2^* = \mathrm{b2/d2}$

Introduction

In the hydraulic power generation, turbines are built to fit the hydraulic specifications of a site, especially high power turbines. In the case of small sites which generate little power, a very interesting solution, if the efficiency is not primordial, is the use of centrifugal pumps as turbines (PAT). Indeed, a pump slightly modified is a good turbine which reaches a good efficiency. It is also a fact that serial pumps show an interesting cost reduction for a power generation site.

Another important application of these pumps as turbines is the energy recovery in water supply systems. If a reverse running pump is installed alongside such a valve, most of the dissipated energy can be recovered [14]. Some sewage-treatment outfalls may also have potential for energy recovery using pumps as turbines. For example, a demonstration project built in 1993 at Nyon uses a pump as turbine to produce up to 210 kW electrical power from a sewage outfall [4].

Pumps manufacturers sell their machines with the hydraulic characteristics in pump mode, which have been obtained by experimental tests. Nevertheless they do not know the machine's performances in turbine mode (expensive tests). The main objective of this project is to predict the characteristics of the pump working as a turbine (without any experimental test) thanks to the pump characteristic and the pump geometry.

State of the art

Several methods have been suggested for predicting the turbine performance based on the data for pump performance at best efficiency, but they produce widely varying results. None of these methods give an accurate prediction for all pumps, at most they just give a first estimate of the turbine performance.

Several authors have published papers on turbine performance prediction using the best efficiency values. In 1962, Childs [5] stated that the turbine best efficiency point and pump best efficiency for the same machines are approximately equal. Stepanoff [13] proposed that the head ratio and the flow rate ratio in turbine and pump operation are in inverse proportion to the pump efficiency square and to the pump efficiency, respectively. Alatorre-Frenk et al.[1] presented formulations based on fitting equations to a limited number of PAT data. Other relations were proposed by Engel [6] and Sharma [12].

Several other authors have published equations that relate the head and flow ratios for pump and turbine operation to the pump or turbine specific speed, see for instance Buse [2], Lewinsky [10] and Grover [7].

Burton et al. [3] have studied the prediction of pumps as turbines using the ratio method, which is a tool for designing centrifugal pumps, whereby the best operating point is determined by the interaction of impeller and volute characteristics. The general equations for impeller and volute characteristics in turbine mode are derived, and they are used to obtain the best operating point.
Rodrigues et al. [11] studied the prediction of a pump as a turbine with CFD and found deviations within 5% to 10% for most of the analyzed parameters (total head, hydraulic torque, hydraulic efficiency).

Hydraulic losses in hydraulic machines

Figure 1 shows the types of hydraulic losses which intervene during the pump and turbine operation of a classic centrifugal pump. The losses can be divided in three types:

1) A circulation loss in pump operation only, see Figure 1 a, which is due to the finite number of blades and which caused a shift of the ideal Euler line.

2) Losses proportional to the flow rate (Q^2) , see Figure 1 b, which can be divided in friction losses when the flow passes through the impeller channels and the casing, and minor or local losses (abrupt changes of section, gradual changes of section, bend effect).

3) Incidence or shock losses see Figure 1 c, which occur when there is a misalignment between the direction of flow and the angle of the volute casing and the runner blade angles. This misalignment invokes a residual velocity vector Ws which is the reason for decreasing efficiencies of machines for flows other than the design flow. The BEP flow rate is located near the minimum of the shock losses.

Adding or subtracting all these losses, for turbine and pump respectively, to the ideal Euler line will lead to the actual performance curve.

In this first phase of the project, the two first types of losses will be modeled. These will allow the prediction of the head of the BEP.

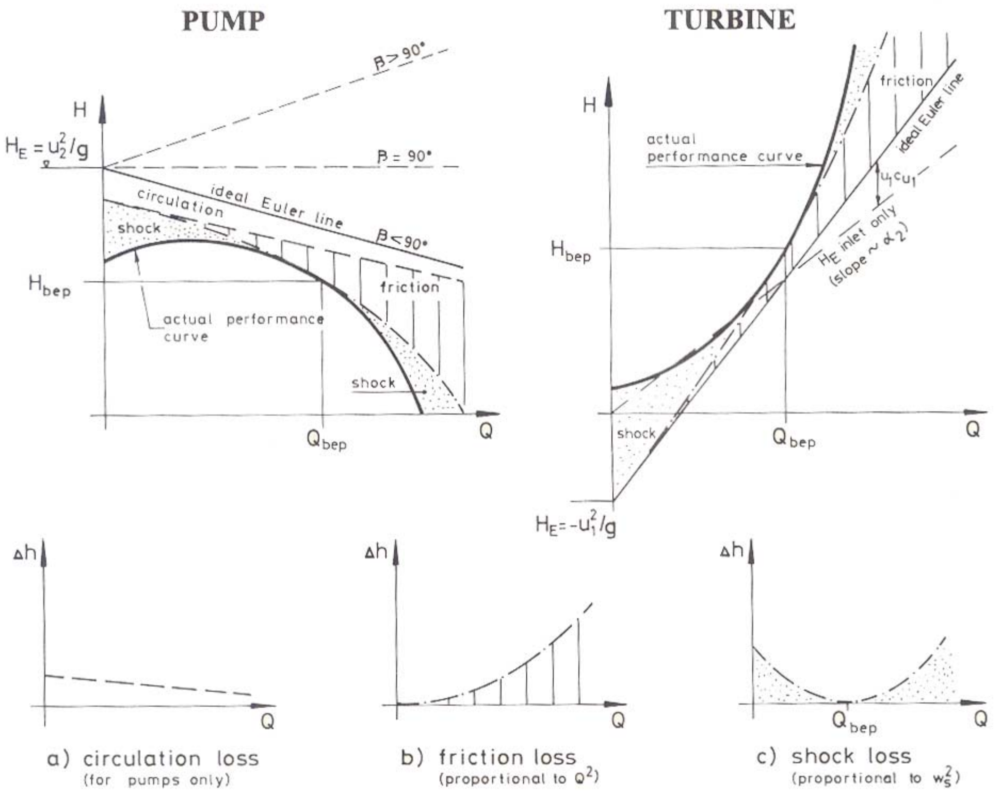


Figure 1: Types of hydraulic losses in a hydraulic turbo-machine

Characteristics in pump and turbine

The diagram in Figure 2 shows the procedure followed in this project to determine the curve predictions in pump and in turbine operation. These curves will be compared with experimental data to determine the deviation of head and flow rate for the BEP.

In summary, the starting point for the prediction is to determine the theoretical pump and turbine characteristics (H-Q) with the help of the pump geometry. Then the loss model proposed is subtracted or added to the pump or turbine characteristics respectively to get the prediction curves.

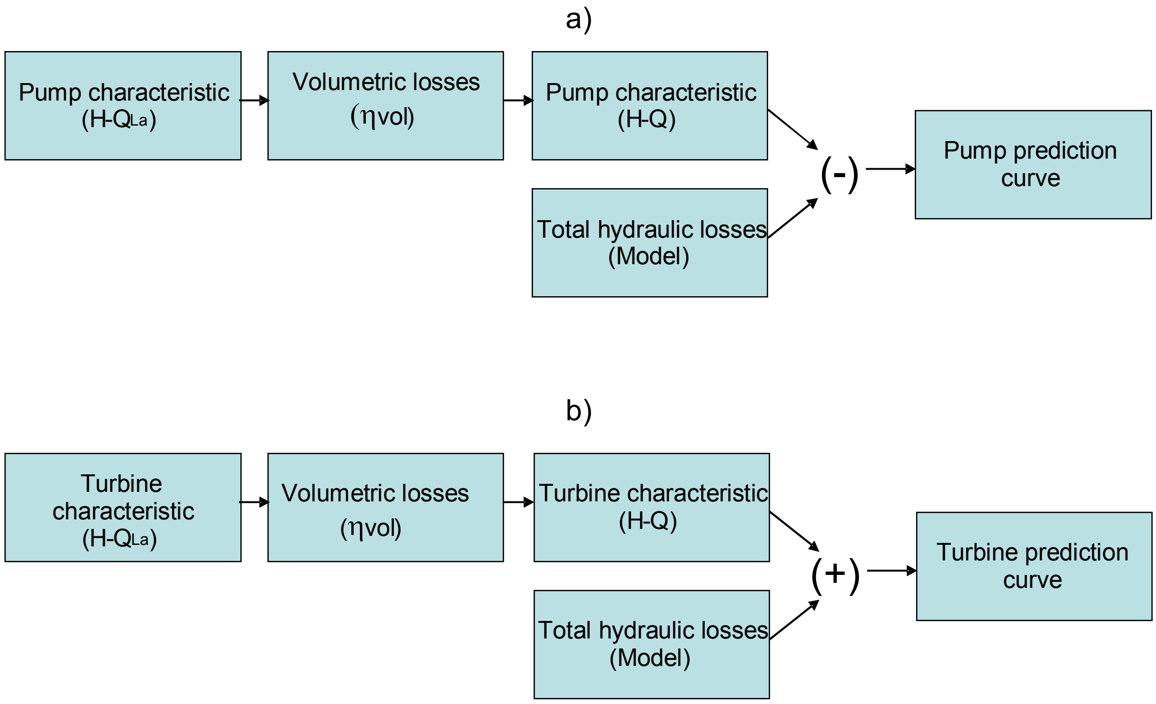


Figure 2 a) procedure for predicting the pump characteristics (H-Q), b) procedure for predicting the turbine characteristic

DETERMINATION OF CHARACTERISTICS IN PUMP OPERATION

<u>Euler Head</u> ($H - Q_{La}$)

The fundamental equation for hydraulics machines, the Euler relation, is:

$$gH_{th} = u_2 c_{u2} - u_1 c_{u1} \qquad \text{Eq. 1}$$

Under the assumption of a pump with an infinite number of vanes, the theoretical head is written as (for the details of formulas, see Gülich [8]):

$$H_{th-\infty} = \frac{u_2^2}{g}\left\{1 - \frac{Q_{La}}{A_2 u_2 \tan\beta_{2B}}\left[1 + \frac{A_2 d_{1m}^* \tan\beta_{2B}}{A_1 \tan\alpha_1}\right]\right\} \qquad \text{Eq. 2}$$

The influence of the finite number of vanes (circulation losses) and the vane blockage on the flow can be better represented by adding a factor called the slip factor γ and a blockage factor τ_2 at the pump exit (see Equation 4 and 5).

$$H_{th} = \frac{u_2^2}{g}\left\{\gamma - \frac{Q_{La}}{A_2 u_2 \tan\beta_{2B}}\left[\tau_2 + \frac{A_2 d_{1m}^* \tan\beta_{2B}}{A_1 \tan\alpha_1}\right]\right\} \qquad \text{Eq. 3}$$

where

$$\gamma = f_1 (1 - \frac{\sqrt{\sin \beta_{2B}}}{z_{La}^{0.7}}) k_w$$

with Eq. 4

$$k_w = 1 - \left(\frac{d_{1m}^* - \varepsilon_{Lim}}{1 - \varepsilon_{Lim}} \right)^3 \text{ and } \varepsilon_{Lim} = \exp\left(- \frac{8.16 \sin \beta_{2B}}{z_{La}} \right)$$

and

$$\tau_2 = \left\{ 1 - \frac{e Z_{La}}{\pi d_2 \sin \beta_{2B} \sin \lambda_{La}} \right\}^{-1} \qquad \text{Eq. 5}$$

Note: if an axial flow at the inlet of the pump is assumed, i.e. $\alpha_1 = \pi / 2$. The Equation 3 becomes:

$$H_{th} = \frac{u_2^2}{g} \left\{ \gamma - \frac{\tau_2 Q_{La}}{A_2 u_2 \tan \beta_{2B}} \right\} \qquad \text{Eq. 6}$$

Leakage flow losses

The pressure difference ΔH_{sp} along the gap for the inwards leakage is (see Figure 3):

$$\Delta H_{sp} = H_p - k^2 \frac{u_2^2}{2g} (1 - \frac{d_{sp}^2}{d_2^2}) \qquad \text{Eq. 7}$$

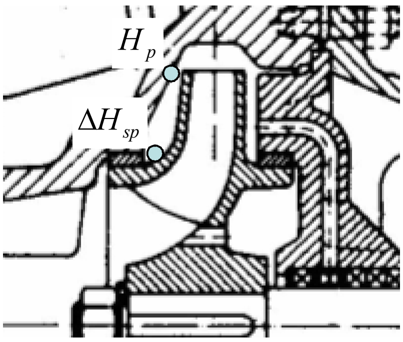


Figure 3: gap for the inwards leakage

with

$$H_p = 0.75 H \quad \text{for} \quad n_q < 40$$

$$k = 0.9 y_{sp}^{0.087}, \quad y_{sp} = \mathrm{Re}_{u_2}^{0.3} \frac{s d_{sp}}{d_2^2} \sqrt{\frac{s}{L_{sp}}}, \quad \mathrm{Re}_{u2} = \frac{u_2 r_2}{\nu} \qquad \text{Eq. 8}$$

k (rotational factor) representing the rotation of fluid between impeller and casing.

The velocity through the gap according to the number of chambers (labyrinths) is:

$$c_{ax} = \sqrt{\frac{2g\Delta H_{sp}}{\zeta_{EA} + \lambda \frac{L_{sp}}{2s} + \Sigma \left(\frac{d_{sp}}{d_{si}}\right)^2 \left(\frac{s}{s_i}\right)^2_i \left(\zeta_k + \lambda_i \frac{L_i}{2s_i}\right)}}$$ Eq. 9

The flow rate Q_{sp} along the gap is:

$$Q_{sp} = \pi d_{sp} s c_{ax}$$ Eq. 10

Finally the pumping volumetric efficiency is:

$$\eta_v = \frac{Q}{Q_{La}} = \frac{Q}{Q + Q_{sp}}$$ Eq. 11

The head relation shown in Eq. 6 is function of Q_{La}, the flow rate flowing through the impeller. By taking into account the leakage losses calculated according to Eq. 10, the head can be written in function of Q, the flow rate through the machine as:

$$H_{th} = \frac{u_2^2}{g}\left\{\gamma - \frac{\tau_2 Q / \eta_v}{A_2 u_2 \tan \beta_{2B}}\right\}$$ Eq. 12

For comfort, all pump characteristics in this study will be represented in function of Q which is obtained thanks to the volumetric efficiency η_v.

DETERMINATION OF CHARACTERISTICS IN TURBINE OPERATION

Theoretical head

From the general equation of turbo machines (Eq. 1), Eq. 13 can be determined according to Gülich [8]. The main characteristic of this equation is that the inflow angle α_2 in turbine mode can be determined from the volute (or diffuser) geometry with the help of the blocked cross section A_{3q}.

$$gH_{th} = u_2^2 \left\{\frac{\mathrm{Q}}{\mathrm{u}_2\, z_{Le}\, \mathrm{A}_{3q}}\left(\frac{r_3}{\mathrm{r}_2}\cos\alpha_{3B} + \frac{d_1^*\, \eta_{\mathrm{v}}\, z_{Le}\, A_{3q}}{\mathrm{z}_{\mathrm{La}}\, \mathrm{A}_{1q}}\cos\beta_{A1}\right) - d_1^{*\,2}\right\}$$ Eq. 13

with

$$\beta_{A1} = \arcsin\frac{A_{1q}}{b_1 t_1},\ \alpha_{3B} = \arcsin\frac{a_3}{t_3}$$ Eq. 14

For calculation of geometrical parameters of the volute, a representation of the unfolded volute can be drawn (see Figure 4).

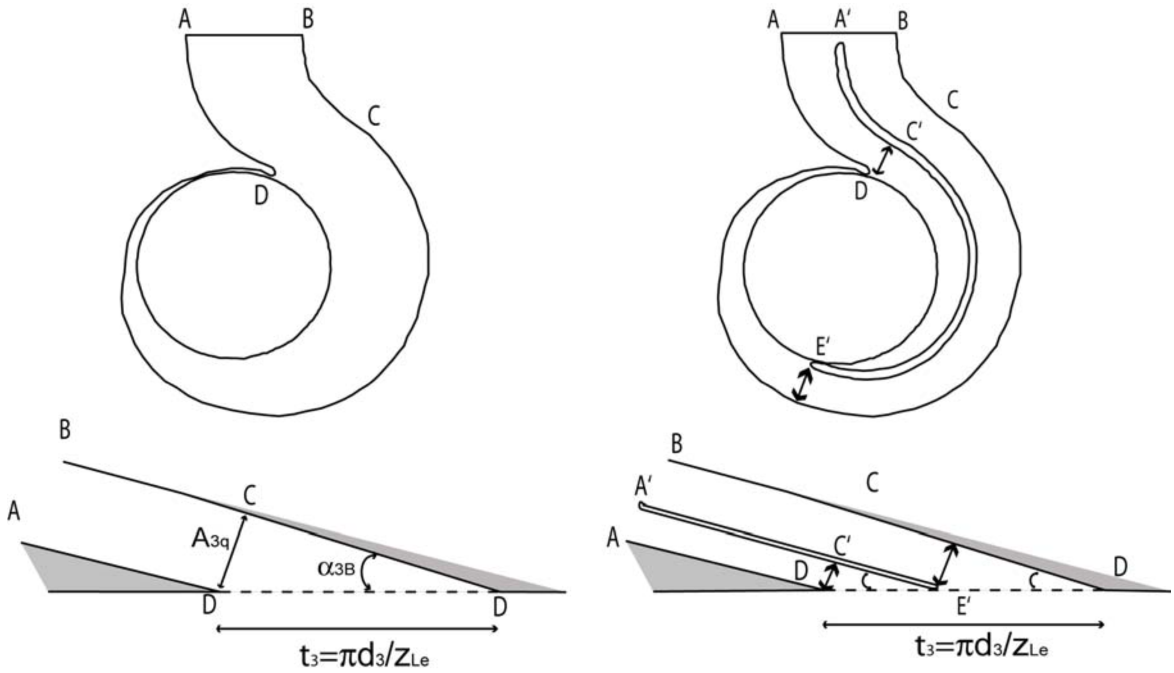


Figure 4: simple volute and double volute with their unfolded representation to determine α_{3B}

Leakage flow losses

In turbine mode, the Equations 7 to 10 can be used to calculate the leakages, taking into account that at this time the flow rate through the runner Q_{La} is:

$$Q_{La} = Q - Q_{sp} \qquad \text{Eq. 15}$$

The volumetric efficiency is:

$$\eta_v = \frac{Q_{La}}{Q} = \frac{Q - Q_{sp}}{Q} \qquad \text{Eq. 16}$$

Model of the hydraulic losses

Losses through the pump channels are obtained with the help of relations well known for friction and minor losses. The channels and sections of the pump have been decomposed in simple elements in order to be able to use the Idel'cik [9] relations.

HYDRAULIC LOSSES THROUGH THE RUNNER

According to the principle of superposition of losses, every channel in the runner is submitted to 3 types of losses: friction losses ΔH_{fr}, bend losses ΔH_{bend} and gradual expansion losses in pump mode ΔH_{GE} or gradual contraction losses in turbine mode ΔH_{GC}.

Figure 5 shows a representation of one runner channel in which an average path of length L_m has been determined with the help of balls following the channel form. The curve angle of the path is δ_0 and the radius R_0.

An important parameter to calculate the losses is the hydraulic diameter D_H defined as:

$$D_H = \frac{4A}{p^*} \quad \text{Eq. 17}$$

with the wet perimeter p^* and the blocked exit or throat area A (A_{1q} and A_{2q} in Figure 9).

Since the hydraulic diameter varies along the vanes, a weighted average is used for the calculation

$$D_{HEQ} = \frac{D_{H1} + 2\overline{D}_H + D_{H2}}{4} \quad \text{Eq. 18}$$

where

$\overline{D}_H$ is the hydraulic diameter at midpoint.

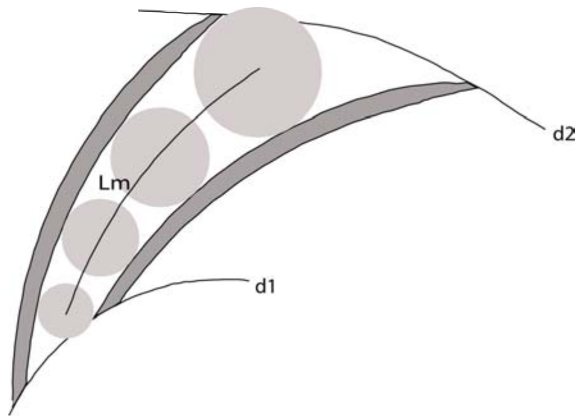


Figure 5: Representation for the modeling of a runner channel

HYDRAULIC LOSSES THROUGH THE VOLUTE

The volute is divided into elements according to the volute angle θ_i (0<θ_i<360°) which starts at the cutwater at $\theta_1 = 0°$ (Figure 6). The diffuser part at the volute outlet can also be divided in several elements.

As was done for the runner, friction losses, bend losses and gradual expansion or contraction losses for each element can be added and a total head loss in the volute determined.

In the case of a double volute there are two cutwaters. Thus the volute is divided in two channels. Each channel can be divided in elements in which losses can be determined.

For each element of the volute, the hydraulic diameter can be determined with the help of cross sections of the volute and with the wet perimeters. The path length and the radius curve for each element help to determine losses according to Equations 20 to 32.

The flow rate through each element is governed by the law of section for a spiral casing and is:

$$Q_i = \frac{\theta_i}{360} Q \qquad \text{Eq. 19}$$

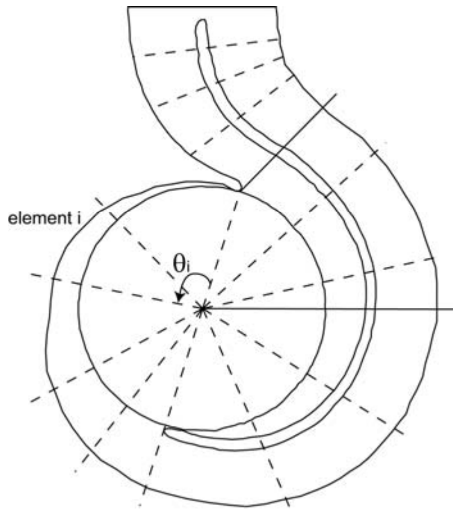


Figure 6: Division of the volute into elements according to the angular position θ_i

MODELING OF THE COMPLETE PUMP

The pump can be modeled as a set of elements or channels connected in series and parallel from its inlet to its exit.

Figures 7 and 8 show a simplified scheme with the main hydraulic losses (friction and minor losses) in the runner and in the volute, in pump mode and in turbine mode respectively.

The assumptions for the pump and turbine models are: constant static pressure around the diameters d1, d2 and d3 (volute) and same pressure through section Dd (discharge nozzle).

These assumptions imply the same head losses through each runner channel and through each volute channel (if double volute).

Then, the runner is modeled as a set of channels in parallel representing the lines of head losses according to the number of vanes. The volute is also modeled in 2 parallel lines (if double volute).

Thus the total head losses are obtained by the sum of head losses through one runner channel and through one volute channel.

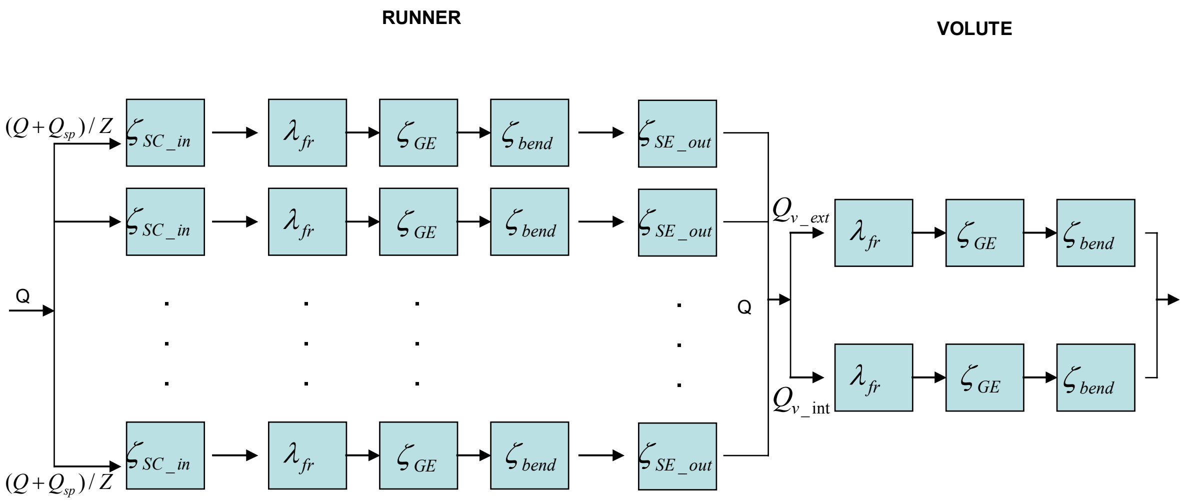


Figure 7: scheme for the model of the hydraulic losses in a pump

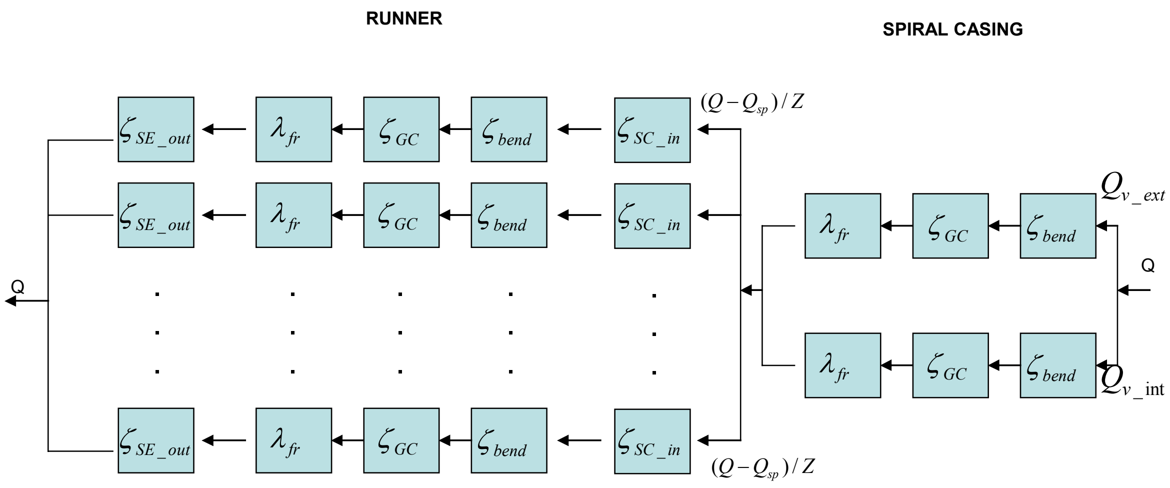


Figure 8: scheme for the model of the hydraulic losses in a turbine

Friction losses (ΔH_{fr})

The head friction loss in a pipe is:

$$\Delta H_{fr} = \frac{\lambda L}{D_H}\frac{c^2}{2g} \qquad \text{Eq. 20}$$

with λ, the friction factor calculated as:

$$\lambda = \frac{0.3086}{\left[\log_{10}(0.135 * \overline{\Delta} + \frac{6.5}{\mathrm{Re}})\right]^2} \quad \text{Colebrook's formula} \qquad \text{Eq. 21}$$

and $\overline{\Delta}$, the relative roughness coefficient (Δ / D_H)

Sudden contraction and expansion losses ($\Delta H_{SC}, \Delta H_{SE}$)

Sudden contraction and expansion losses occur at the inlet and at the exit of the runner channels in both pump and turbine mode.

The areas used to calculate these losses are those called throat area or blocked area A_{1q}, A_{2q} (taken normal to the vanes), see Figure 9.

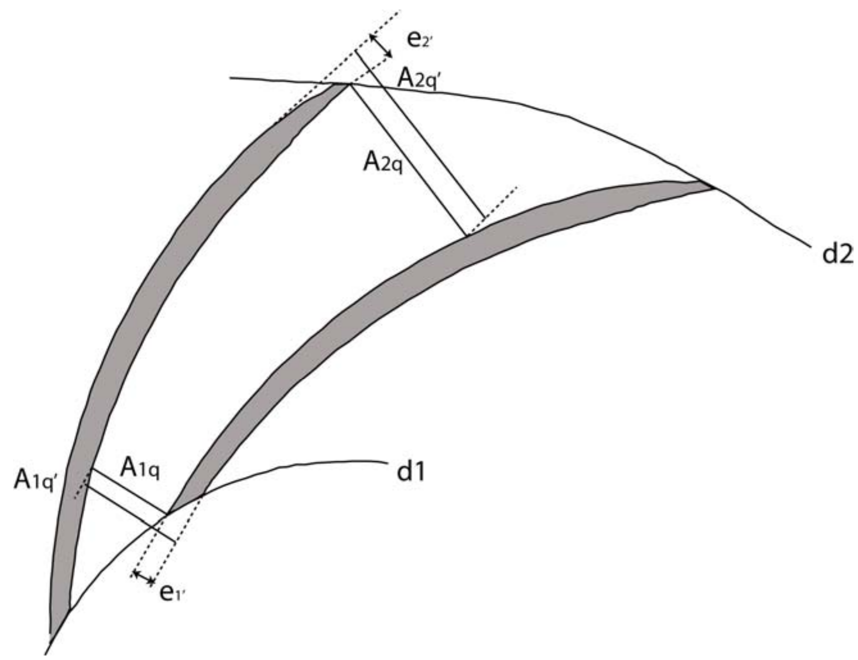


Figure 9: Areas used to calculate the sudden expansion and contraction losses in the runner

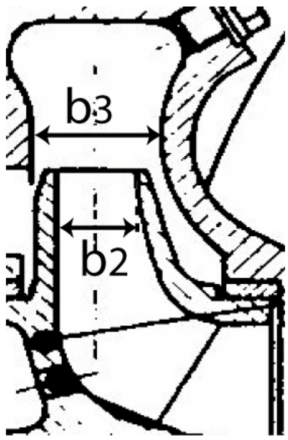


Figure 10: dimension to calculate sudden expansion or contraction between de volute and the runner

The relations used to calculate the sudden contraction losses and the sudden expansion losses in pump mode are:

$$\Delta H_{SC} = \zeta_{SC} \frac{c^2}{2g}, \ \Delta H_{SE} = \zeta_{SE} \frac{c^2}{2g} \quad \text{Eq. 22}$$

The sudden contraction losses coefficient ζ_{SC} and the sudden expansion losses coefficient ζ_{SE} in pump mode are (see Idel'cik [9]):

$$\zeta_{SC} = 0.5(1 - \frac{A_{1q}}{A_{1q'}}) \qquad \text{Eq. 23}$$

$$\zeta_{SE1} = \left(1 - \frac{A_{2q}}{A_{2q'}}\right)^2 \qquad \text{Eq. 24}$$

$$\zeta_{SE2} = \left(1 - \frac{A_{2q''}}{A_{3q'}}\right)^2 \qquad \text{Eq. 25}$$

with:

$$\begin{aligned} A_{1q} &= a_1 b_1 \\ A_{1q'} &= (a_1 + e_{1'}) b_1 \\ A_{2q} &= a_2 b_2 \\ A_{2q'} &= (a_2 + e_{2'}) b_2 \\ A_{2q''} &= (\frac{2\pi r_2}{Z_{La}} - e_2) b_2 \\ A_{3q'} &= (\frac{2\pi r_2}{Z_{La}} - e_2) b_3 \end{aligned} \qquad \text{Eq. 26}$$

Note: The same relations but taking into account the flow direction in turbine mode is used to calculate the sudden expansion and contraction losses in turbine mode.

Gradual expansion and gradual contraction losses (ΔH_{GE}, ΔH_{GC})

Gradual expansion losses occur through the runner channels and through the volute channels in pump mode.

Gradual contraction losses occur in the runner channels and in the volute channels in turbine mode.

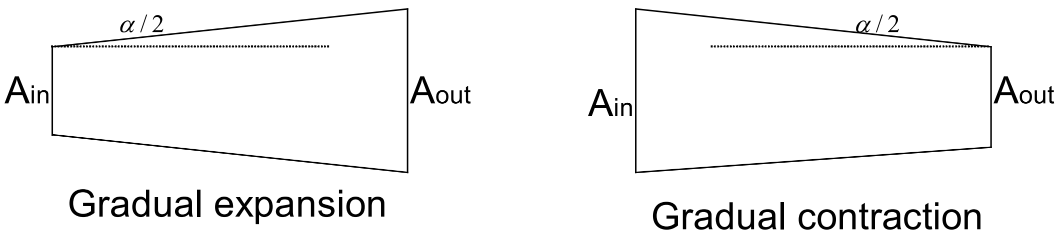
The relations used to determine these losses are:

$$\Delta H_{GE} = \zeta_{GE} \frac{c^2}{2g} \text{ and } \Delta H_{GC} = \zeta_{GC} \frac{c^2}{2g} \qquad \text{Eq. 27}$$

The gradual expansion and contraction coefficients are determined as (see Idel'cik [9]):

$$\zeta_{GE} = 3.2(\tan\frac{\alpha}{2})^{1.25}(1 - \frac{A_{in}}{A_{out}})^2 \qquad \text{for } 0 < \alpha < 40° \qquad \text{Eq. 28}$$

$$\zeta_{GC} = 0.8 \sin\frac{\alpha}{2}(1 - \frac{A_{in}}{A_{out}}) \qquad \text{for } 0 < \alpha < 40° \qquad \text{Eq. 29}$$



5.4 Changes of flow direction losses (ΔH_{bend})

The losses due to the bend geometry occur in the runner and along the volute, for pump and turbine operation and are written as:

$$\Delta H_{bend} = \zeta_{bend} \frac{c^2}{2g}$$

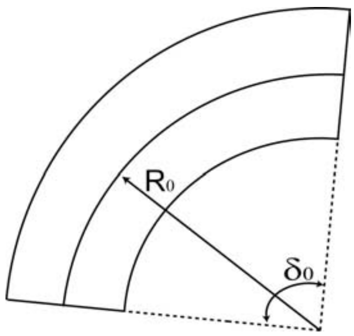


Figure 11: bend losses

The coefficient of bend losses is (see Idel'cik [9]):

$$\zeta_{bend} = A_1 B_1 C_1 \qquad \text{Eq. 30}$$

where

$$A_1 = \begin{cases} 0.9 \sin\delta° & \text{for } \delta° \leq 70° \\ 0.7 + 0.35\dfrac{\delta°}{90} & \text{for } \delta° \geq 100° \end{cases} \qquad \text{Eq. 31}$$

$$B_1 = \begin{cases} \dfrac{0.21}{\left(\dfrac{R_o}{D_H}\right)^{2.5}} & \text{for } 0.5 \le \dfrac{R_o}{D_H} \le 1 \\ \dfrac{0.21}{\left(\dfrac{R_o}{D_H}\right)} & \text{for } \dfrac{R_o}{D_H} > 1 \end{cases}$$ Eq. 32

C_1 depends on the ratio of a_0 / b_0. For square and circular sections C_1=1.

TOTAL HYDRAULIC LOSSES

The relation leading to the total hydraulic losses of the pump and of the pump operating as turbine respectively is showed in Eq. 33.

$$\Delta H_{Total\_Loss} = (\Delta H_{fr} + \Delta H_{GE/GC} + \Delta H_{SE/SC})_{runner} + \sum_i^n (\Delta H_{fr} + \Delta H_{GE/GC} + \Delta H_{SE/SC})_{volute}$$ Eq. 33

where n is the number of elements in which the volute pump has been divided.

Pumps used

Four pump geometries have been used to test the model proposed. The main characteristics of these pumps are shown in Table 1.

Pump No 1, a pump from Biral, belongs to the EIG. A test rig has been built in the hydraulic laboratory in order to measure the pump and turbine characteristics (head, efficiency and power output). Because the geometry of the Biral pump was not known, it has been dismounted and measurements of the runner and of the volute have been performed, see Figures 12 to 14.

The plans of the three other geometries were provided by Sulzer Pumps with their respective measured characteristics as pumps and as turbines.

Pump No	Specific speed (n_q)	Number of vanes (Z_{La})
1 (Pump from Biral, EIG)	40	6
2 Sulzer pump	19	5
3 Sulzer pump	28.7	6
4 Sulzer pump	31.6	6

Table 1: Some specifications of pumps used in the project

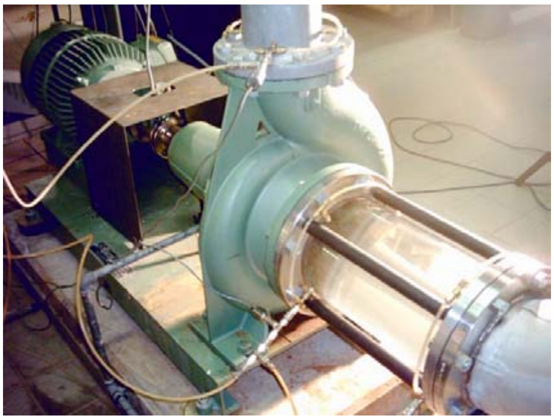


Figure 12: Test rig of PAT at the EIG (Biral pump)

Figure 13: Volute geometry measurements

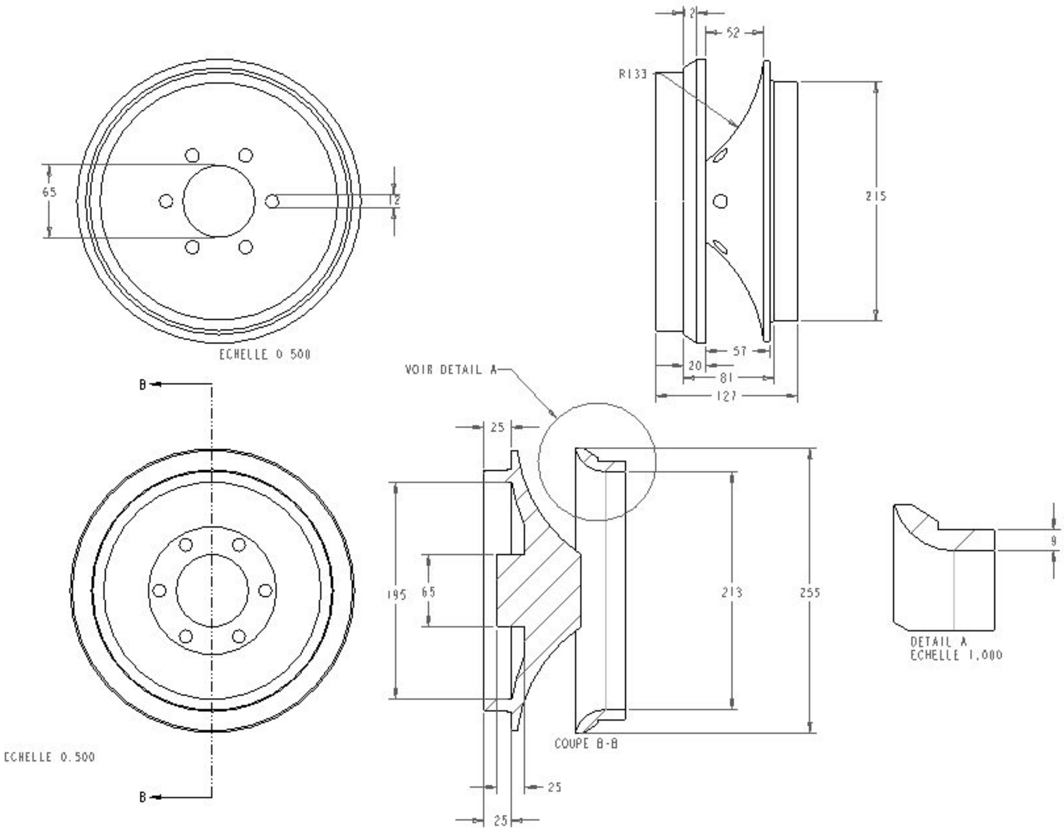


Figure 14: Geometry of the runner of the Biral pump measured by the EIG

Results

PUMP OPERATION

Figures 15 to 18 show the characteristics (H* - Q*) calculated in pump mode and the total head losses calculated in order to determine a pump prediction curve. The head and flow rate axis are non dimensional (divided by the head and flow rate of the BEP).

It is observed that the predicted pump characteristics curves fit the experimental data very well in the zone near the BEP. Deviations from 4% to 5% between the predicted head and the head measured at BEP are obtained for the tested pumps.

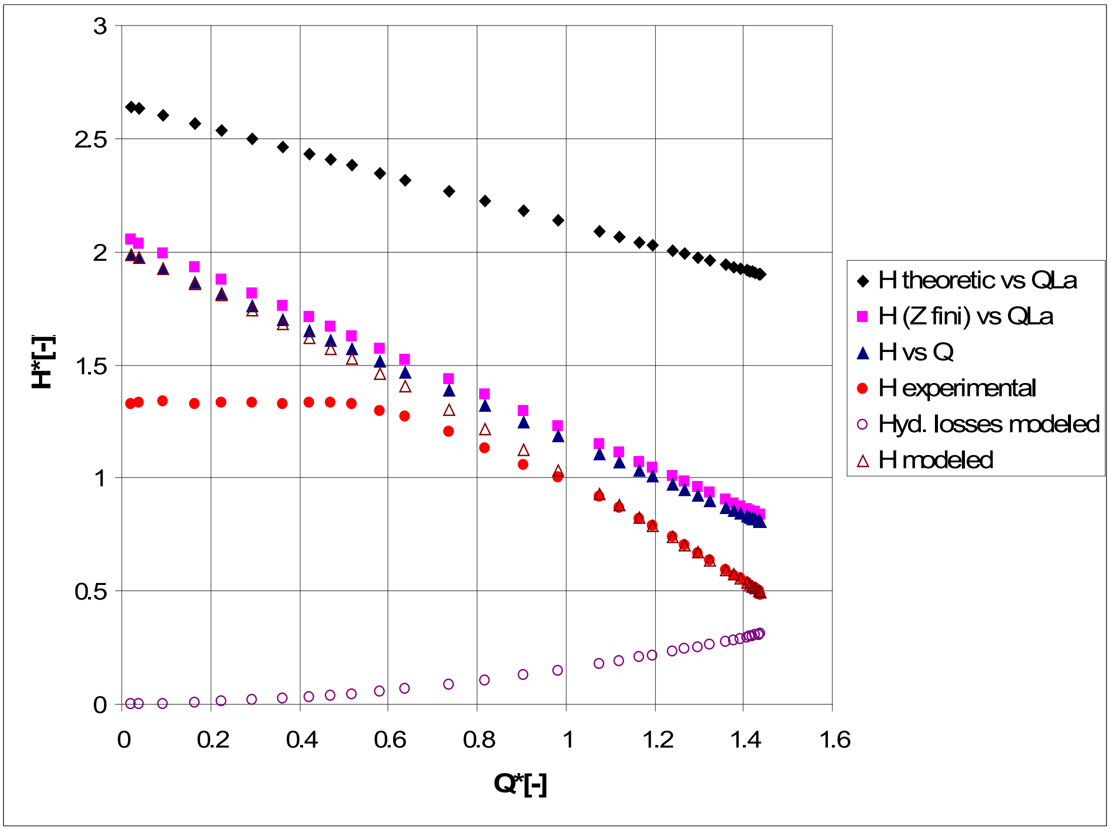


Figure 15: Modeled pump characteristics and hydraulic losses for Pump No 1 (BIRAL pump)

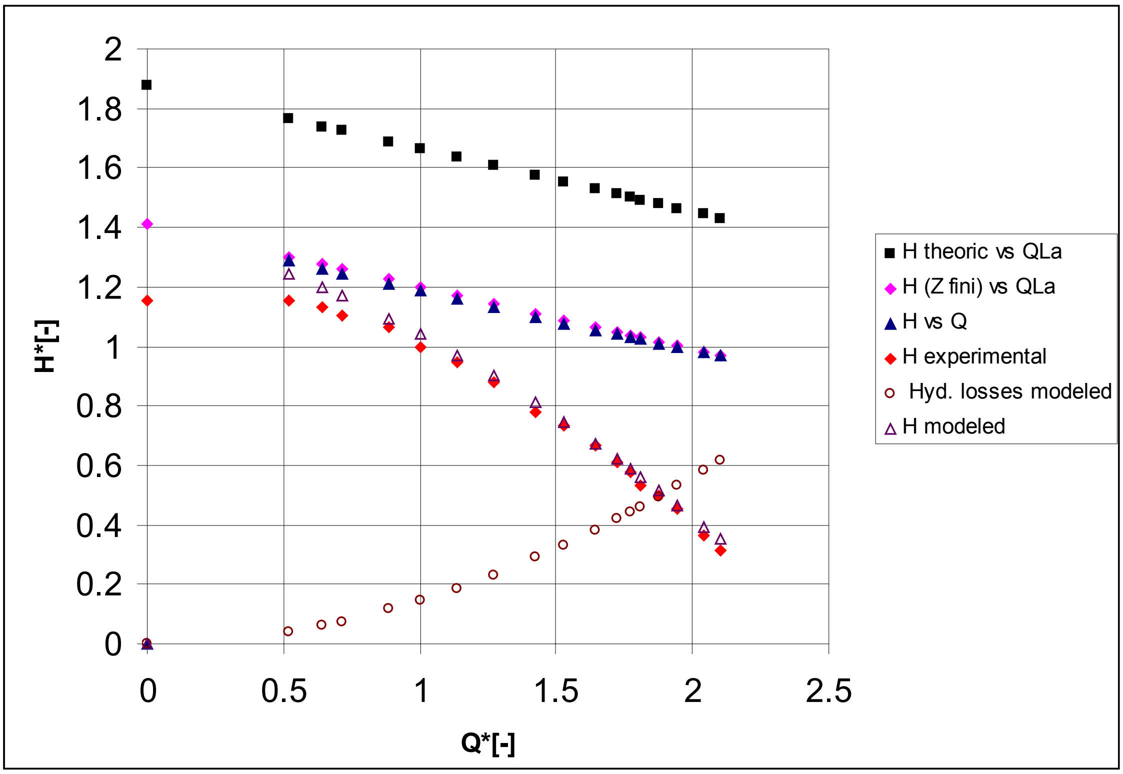


Figure 16: Modeled pump characteristics and hydraulic losses for Pump No 2

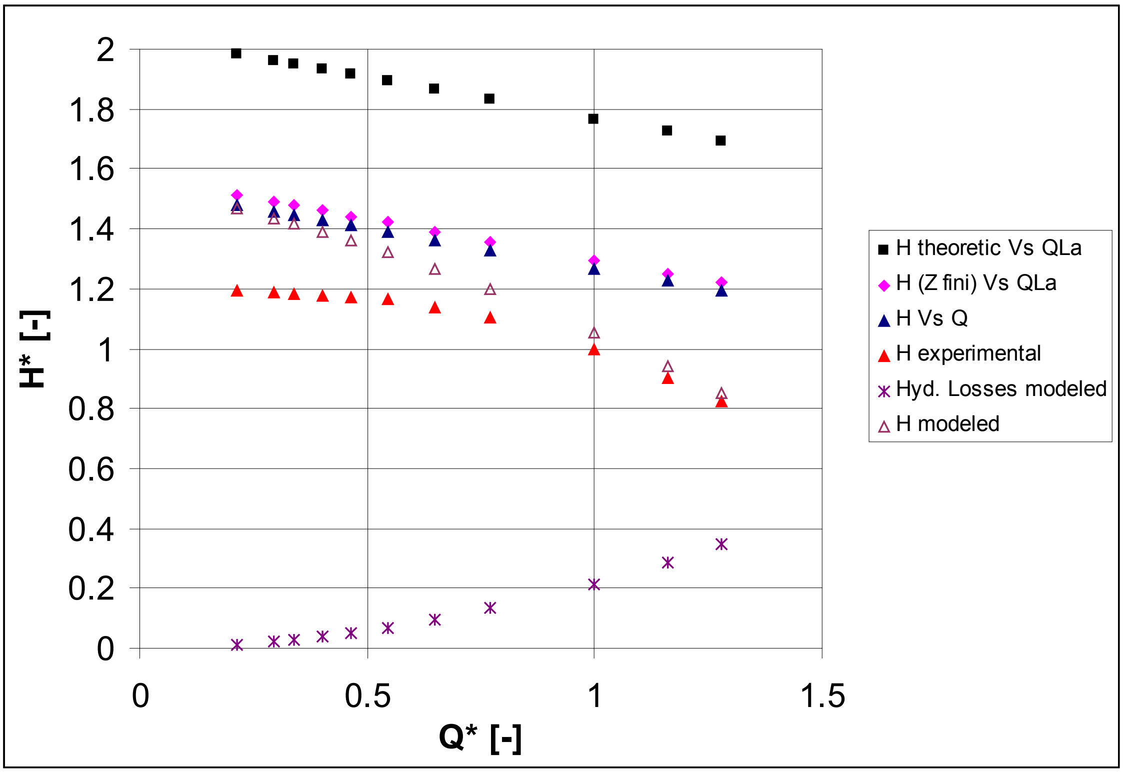


Figure 17: Modeled pump characteristics and hydraulic losses for Pump No 3

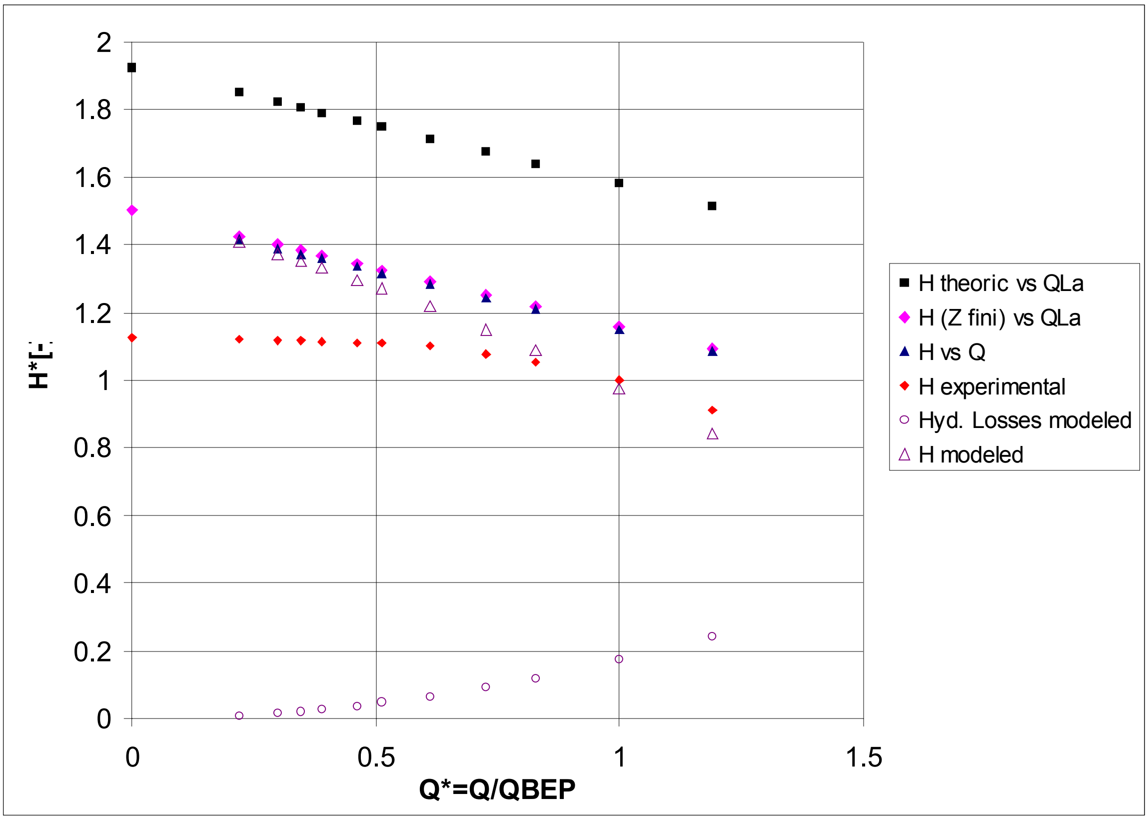


Figure 18 Modeled pump characteristics and hydraulic losses for Pump No 4

TURBINE OPERATION

Prediction of the head

Figure 19 to Figure 22 show the turbine characteristics and the modeled hydraulic head losses of the four pumps studied.

The predicted turbine curves (H modeled) fit the experimental head curve at the BEP with deviations going from 2.5% to 4.6%. To obtain these results a correction factor, which is a common factor for all pumps, has been used to correct the hydraulic losses modeled. Apparently the model under evaluates the losses in turbine mode. Improvements on this point must be done in the second phase of the project.

Prediction of the flow rate

The model proposed computes the hydraulic losses without the incidence losses, which occur when the pump-turbine operates off the BEP. According to Figure 1, in order to get the BEP flow rate, the minimum of the incidence losses must be calculated. That is another point to study in a second phase of the project.

Prediction of the efficiency

Once the prediction of the flow rate at the BEP has been calculated, the determination of the global efficiency is easy, see Equations 34 and 35.

$$\eta_{global\,BEP} = \eta_{hyd_{BEP}} \eta_{vol} \eta_{mec}$$ Eq. 34

with

$$\eta_{hyd\,BEP} = \frac{H_{th_{BEP}}}{H_{BEP}}$$ Eq. 35

The volumetric efficiency η_{vol} which is almost constant for all flow rates is calculated by the software with the Equation 16. The mechanical efficiency η_{mec} can be approximated by 0.98.

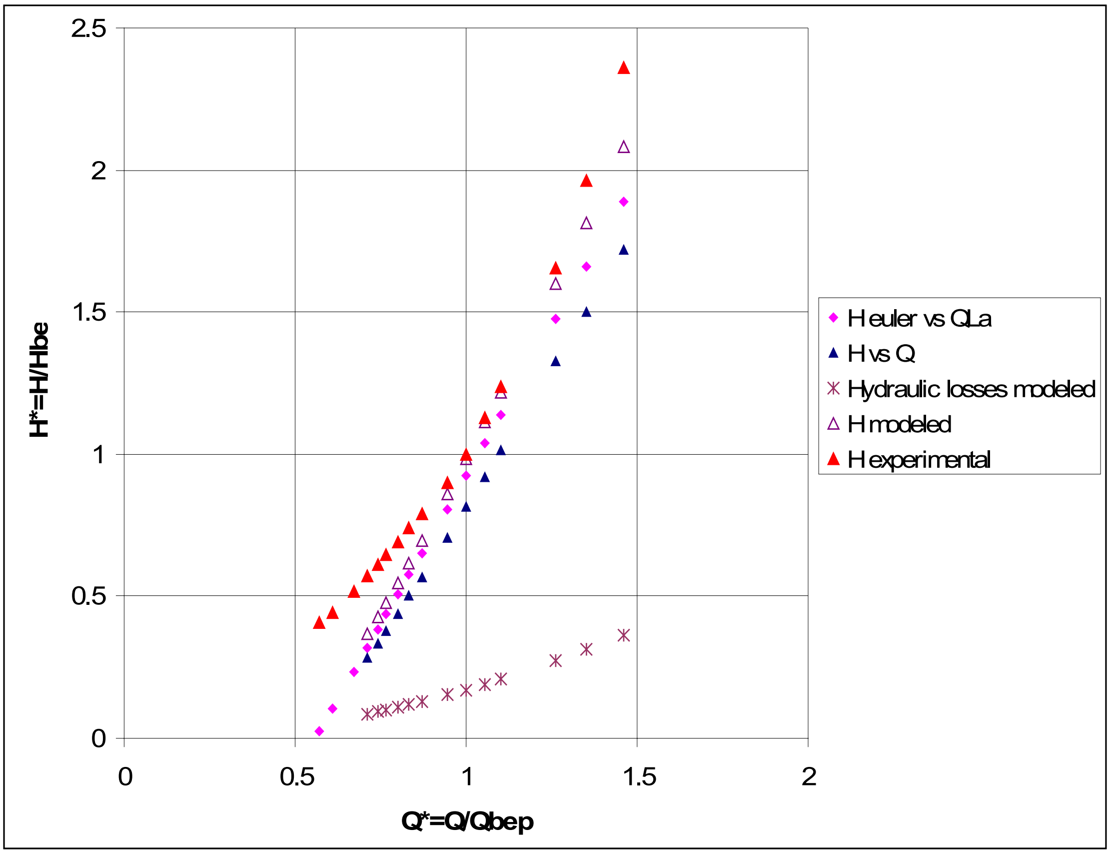


Figure 19: Modeled turbine characteristics and hydraulic losses for Pump No 1

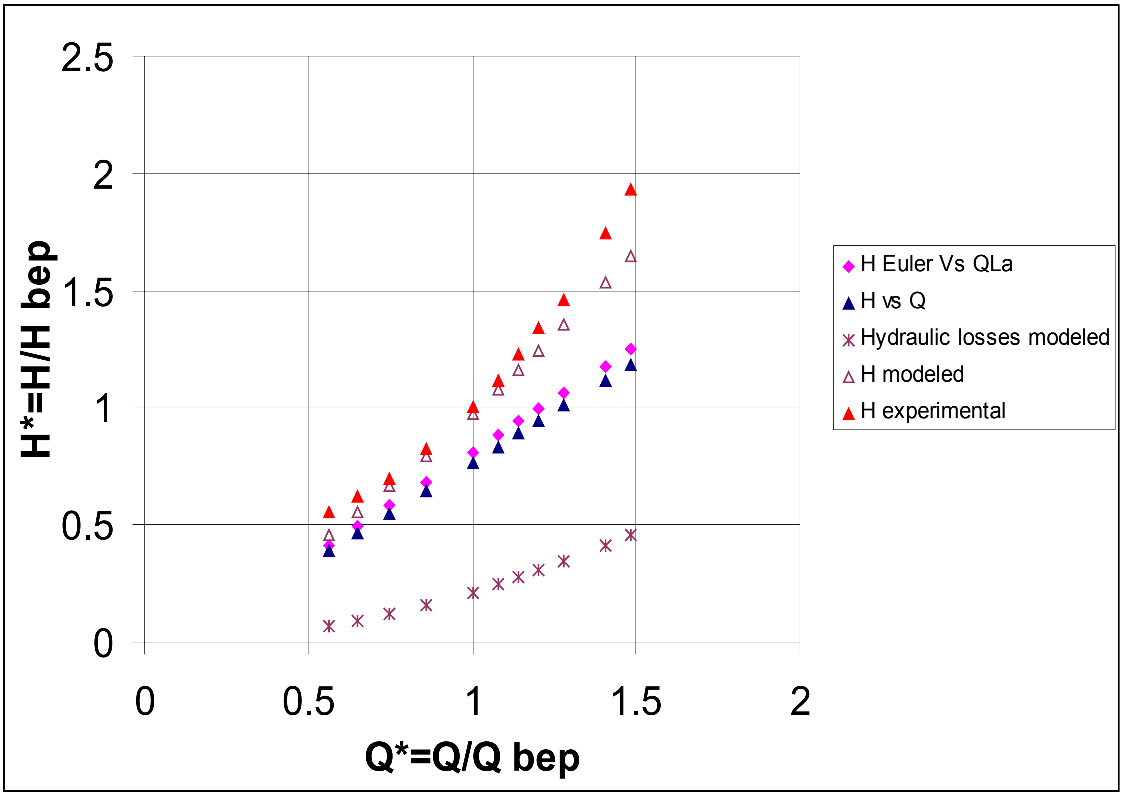


Figure 20: Modeled turbine characteristics and hydraulic losses for Pump No 2

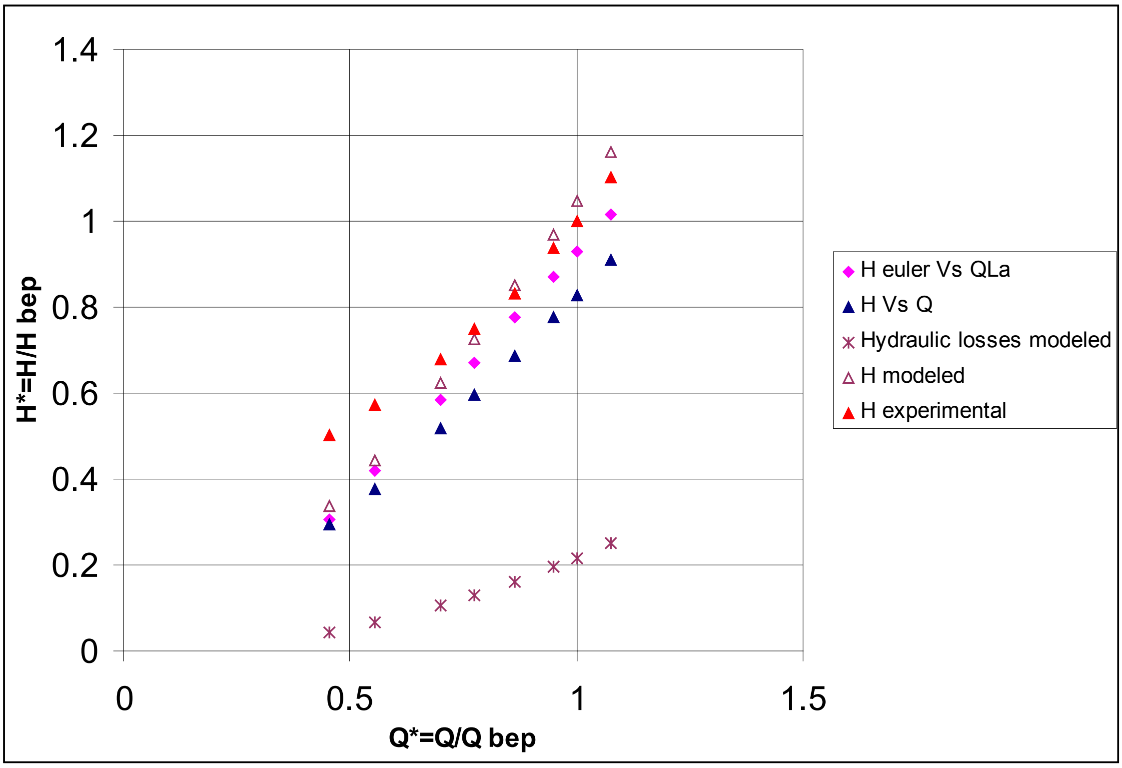


Figure 21: Modeled turbine characteristics and hydraulic losses for Pump No 3

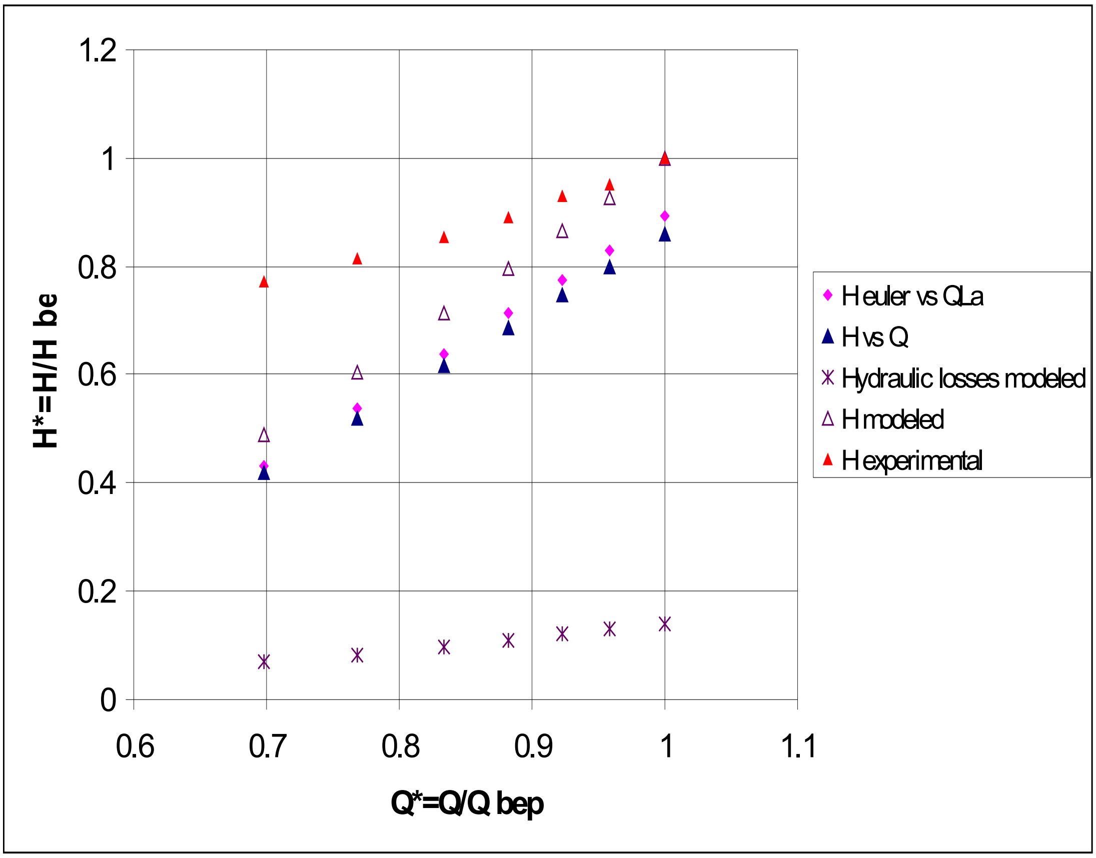


Figure 22: Modeled turbine characteristics and hydraulic losses for Pump No 4

Exploitation du moteur synchrone en vitesse variable

A la *Haute Ecole Valaisanne* (*HEVs*), une installation de laboratoire a été réalisée; elle comprend **une machine synchrone à aimants permanents et un convertisseur de fréquence** développé à la *HEVs*. L'objectif des essais avec 2kW de puissance était de tester différents **algorithmes de réglage** du système à vitesse variable pour la partie onduleur/redresseur du côté réseau et du côté machine (voir Figure 23).

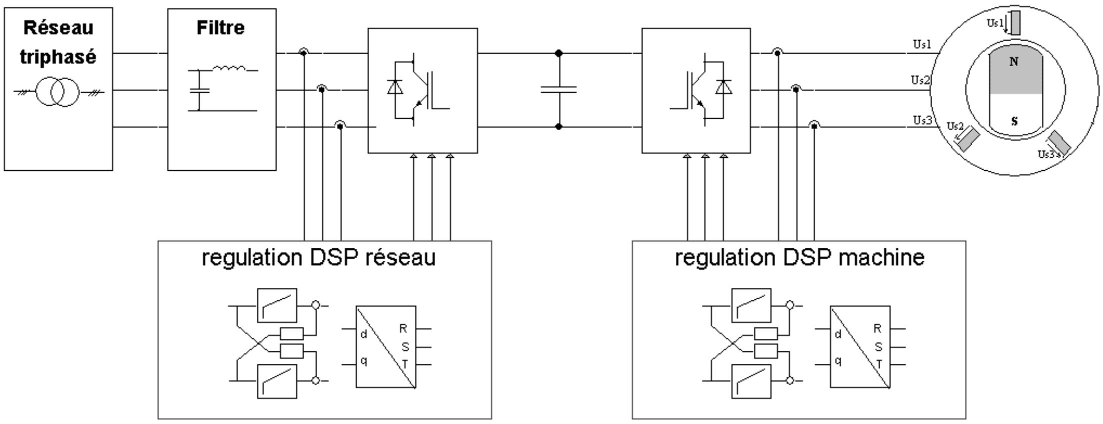


Figure 23 : Schéma bloc du système d'entraînement comprenant un convertisseur de fréquence 4 quadrants et une machine synchrone à aimants permanents

Les algorithmes de régulation se basent sur la théorie du réglage vectoriel. Ils étaient implémentés dans un système à processeur de signal DSP. La régulation de vitesse s'effectue sans capteur de vitesse. La fréquence de commutation des transistors IGBT était choisie à 20kHz pour diminuer les dimensions du filtre du côté réseau. Le système montre un bon rendement en charge partielle (rendement maximal : onduleur 94%, machine 90%, voir Figure 24), avec courants sinusoïdaux et facteur de puissance 1.

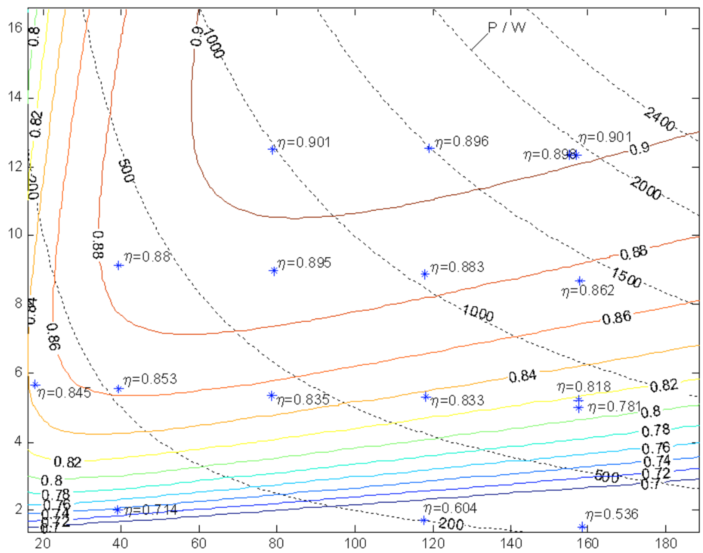


Figure 24 : Diagramme du rendement en fonction du couple M [Nm] et de la vitesse angulaire ω [rad/s]

Conclusions

Le modèle de pertes hydrauliques proposé pour la prédiction de la caractéristique de fonctionnement en turbine d'une pompe, a fourni des résultats très concluants pour la suite.

Le modèle de pertes a tout d'abord permis de prédire la hauteur de refoulement en pompe au point de meilleur rendement avec une erreur de 4 à 5% par rapport aux mesures.

Le même modèle de pertes appliqué au fonctionnement en turbine nous permet de prédire la chute au point de meilleur rendement avec un écart de 2 à 5 %. Une surestimation des pertes hydrauliques données par le code a été remarquée dans les résultats. Une légère amélioration du modèle doit être effectuée pour obtenir des résultats avec une meilleure précision.

Reste à déterminer le débit en turbine au point de meilleur rendement, donnée essentielle pour le calcul du rendement en turbine. Celui-ci sera obtenu par la modélisation des pertes par incidence dont le minimum se trouve à un débit proche du débit de meilleur rendement. Un modèle de ces pertes est déjà en cours d'évaluation.

Un logiciel programmé sur une feuille EXCEL sera préparé et mis à disposition pour libre utilisation des constructeurs de pompes et des particuliers.

Évaluation du projet (phase 1) et perspectives pour 2006

Les travaux sur la partie hydraulique du projet ont débuté en 2004, mais ce n'est que depuis la moitié de l'année 2005 que les moyens ont pu être réunis pour que l'équipe EIG-HEVS puisse travailler d'arrache-pied sur le projet, notamment avec l'engagement officiel du Dr. Jorge Arpe par l'EIG. Dans ce laps de temps, la modélisation des pertes hydrauliques a pu être testée sur 4 géométries de pompe. Plusieurs réunions d'ordre technique avec Sulzer Pumps ont été menées lesquelles ont servi à l'amélioration du modèle. Même si la démarche pour établir ce modèle semble être très mathématique, c'est le passage obligé pour obtenir des meilleures prédictions des performances des PAT autres que celles établies par d'autres méthodes, qu'elles soient empiriques ou analytiques.

Sulzer Pumps, partenaire du projet, est très satisfait des résultats à ce stade du projet et a manifesté d'ores et déjà son désir de continuer dans une deuxième phase du projet. De même, des pourparlers sont en bonne voie avec les Services Industriels de Genève. Ils sont intéressés à participer au projet et à appliquer la méthode pour le dimensionnement des PATs pour la récupération d'énergie dans le retour d'un circuit alimentant des utilisateurs de pompes à chaleur.

Par rapport à l'étude de la génératrice synchrone effectuée à l'HEVS, les résultats obtenus en 2004 ont été valorisés par la publication d'un article, voir référence ci-dessous. Rappelons que, à la suite d'un projet précédent, une pompe Sulzer fonctionnant en turbine à vitesse variable et utilisant une électronique de puissance développée à Sion a été installée sur l'eau potable de la Ville de Sion en 1995. Grâce à la présente étude, l'électronique ainsi que les algorithmes de réglage ont pu être améliorés et adaptés aux derniers développements de l'électronique.

Nos objectifs pour 2006, si un financement complémentaire est accepté, se centrent sur l'ajout d'un modèle des pertes par incidence à notre modèle actuel des pertes hydrauliques qui nous permettra de déterminer le débit du point de meilleur rendement, ainsi nous pourrions estimer le rendement global d'une PAT. Pour cela il serait aussi important de disposer d'autres géométries de pompes à d'autres vitesses spécifiques à tester pour valider définitivement notre modèle.

Travaux publiés dans le cadre du projet

Biner H.-P., Dubas M., Germanier A.: "Small variable speed power plant on a drinking water network", in Proc. European Conference on Power Electronics, Intelligent Motion and Power Quality PCIM, pp. 420-424, Nuremberg 2005.

Danssmann E. : Pompe fonctionnant en turbine. Détermination d'un modèle de pertes pour une pompe et validation pour le fonctionnement en turbine. Travail de diplôme 2004. Ecole d'ingénieurs de Genève.

Collaboration nationale

Les partenaires du projet sont:

- Sulzer Pumps à Winterthur : Philippe Dupont
- Ecole d'ingénieurs de Genève : Jean Prénat, Jorge Arpe
- Haute Ecole Valaisanne : Michel Dubas, Hans-Peter Biner

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