Methodology to determine the parameters of the hydraulic turbine governor for primary control

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Integration of large amounts of new renewable energies such as wind and solar power represents a challenging task as far as the power network stability is concerned. Due to extended operations range, fast response time and high flexibility, hydro power plants are usually well suited to provide primary, secondary or tertiary frequency control services, see Fig. 1, and contribute to power network reliability. All generating units that supply to the electricity market must demonstrate the compliance with Transmission System Operators, TSO. Thus, a provider of primary control, also called Frequency Containment Reserve FCR, has to offer both positive and negative control power at any time for a stable and secure operation of the grid. The appropriate determination of the PID parameters of the hydraulic turbine governor is required to quantify the realistic primary control capability of the hydraulic power plant.

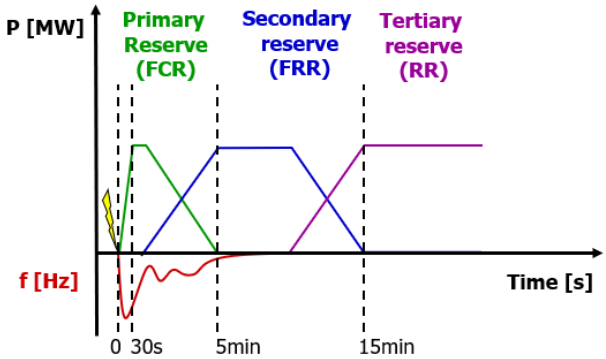
In this paper, different methods to determine the PID parameters using either the so-called Ziegler-Nichols stability limit or the system time constants were applied to 40 hypothetic different Francis turbines with specific speeds Nq ranging between 20 and 130. The layout of the hydraulic power plant comprising an upstream reservoir, a single penstock, a turbine, a tailrace tunnel and a downstream reservoir was adapted for each specific speed according to the net head and the dimensions of the Francis turbine. The nominal data and a realistic empirical turbine performance hill chart of each Francis turbine were defined with the new RENOVHydro library of SIMSEN software.

For each test case, a transient simulation of the response of a Francis turbine driven by standard PID governor to a frequency variation of 200 mHz is performed with SIMSEN software and the system response is analysed in time domain. With this systematic approach applied to a large range of hydraulic turbines, a methodology was determined to easily define the PID parameters of the hydraulic turbine controller.

1. RENOVHydro methodology

The RENOVHydro project [12] is dedicated to the renovation of an existing hydroelectric power plant and an independent assessment of a high number of civil and electromechanical potential modifications using a unique methodology. Thus, energy and economic indicators such as annual energy generation, annual amount of turbined/pumped water, energy coefficient, investment cost, profitability and ancillary services for each renovation option can be analysed to identify the technical trends according to a given political, economic and environmental context.

In order to automatically assess the primary control potential of the renovated hydroelectric power plant, it is necessary to have a simple and robust methodology to deduce the parameters of a PID controller. In this paper, two different methods based on the limit of stability and on the system time constants are compared. The SIMSEN model of the hydraulic power plant and the control system are described in the following sections.



Time response for different frequency control reserves

Type	*Action*	*Time response*
Primary (FCR: Frequency Containment Reserve)	- Frequency control: turbine governor - Voltage control: synchronous generator voltage regulator	< 30 seconds
Secondary (FRR: Frequency Restoration Reserve)	- Frequency / active power regulation: turbine governor with TSO active power set point - Voltage / reactive power regulation: voltage governor with TSO reactive power set point	< 5 minutes
Tertiary (RR: Replacement Reserve)	Human: →dispatching	< 15 minutes

Description of the actions for power network control

Fig. 1. Definition of the different frequency control reserves.

2. Description of the generic test case

A generic test case is used to test the two different methods using either the Ziegler-Nichols stability limit or the system time constants proposed in this paper and determine the parameters of the PID controller.

2.1 Dimensioning of the generic hydraulic power plant

These methods were applied to 40 Francis turbines with water head from 30 to 500 mWC, see Fig. 2 (left). The selected turbine can be represented as function of the IEC speed factor n_{ED} and the IEC discharge factor Q_{ED}, see Fig. 2 (right).

With the new library of SIMSEN software developed in the RENOVHydro project, the dimensioning of the spiral casing, the runner and the draft tube for each Francis turbine using statistical laws [3, 4, 5, 7, 8, 13, 18, 19] requires knowledge of only four parameters:

- **Mechanical power P_m**: fixed arbitrary to 50 MW or to 300 MW.
- **Rated head H_n**: computed with the specific speed Nq = $f(n_{ED}, Q_{ED})$ [14].
- **Year of commissioning**: fixed arbitrary to 1990.
- **Frequency of the electrical grid F_{Grid}**: fixed arbitrary to 50 Hz.

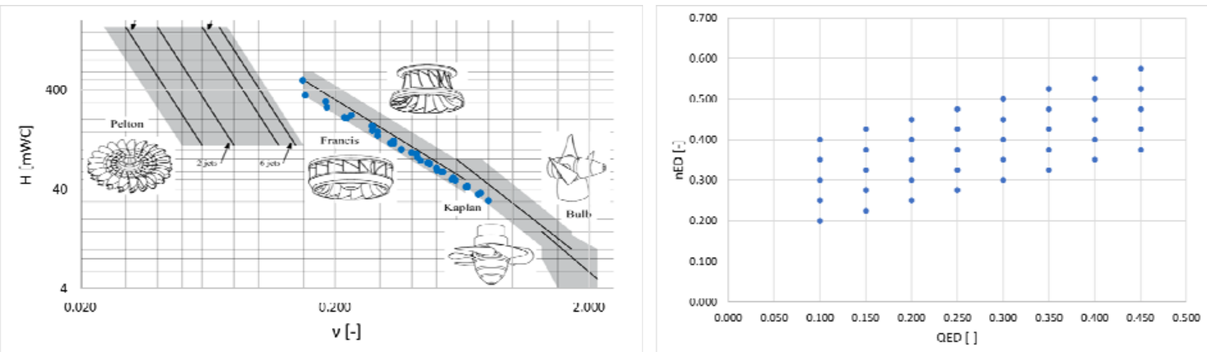


Fig. 2. Illustration of the 40 selected Francis turbines (blue points) as function of the Head H and specific speed ν (Left), the IEC speed factor n_{ED} and IEC discharge factor Q_{ED} (Right).

This first dimensioning defines the complete geometry of a turbine (spiral case, runner, draft tube) and estimates the rated data (rated discharge, rated rotational speed, peak efficiency, reference diameter of the runner, generator and runner inertia). This information was validated by comparing the geometries with existing hydraulic installations described in the Henry's book [10]. The maximum error found on more than 50 test cases was a maximum of 10%.

The performance hill chart of a hydraulic machine is very often derived from measurements on a reduced-scale physical model. In this case study, the 40 Francis turbines selected do not belong to specific projects and therefore the performance hill chart must be estimated. In the RENOVHydro project, a methodology was applied to generate a performance hill chart with a polynomial bi-variate functions based on Hermite polynomials [6] for a given IEC speed factor n_{ED} and IEC discharge factor Q_{ED} at best efficiency points. An example of estimated hill chart is illustrated in Fig. 3 and compared with experimental measurement on a reduced scale model of a Francis turbine.

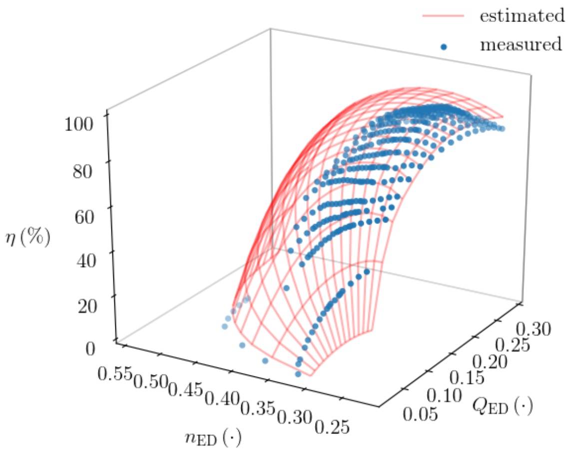


Fig. 3. Performance hill chart from the database versus experimental measurement on reduced scale model of a Francis turbine (Nq = 43).

The layout of the generic hydraulic power plant features by an upstream reservoir, a penstock, a Francis turbine, a tailrace tunnel and a downstream reservoir, see Fig. 4. The main dimensions are defined by the following rules:

- **Upstream reservoir:** The volume of this reservoir is assumed to be infinite. The water elevation is fixed to 1.04 x rated head of the Francis turbine to compensate for head losses in the penstock.
- **Penstock:** The length of the penstock is assumed to be equal to 3 x the rated head of the Francis turbine. The diameter is fixed to 1.5 x the reference diameter of the Francis turbine.
- **Francis turbine:** The dimension, the performance hill chart and the inertia of the turbine and the generator are defined by the new library of SIMSEN software [6][12]. The closing time of the guide vane are fixed arbitrary to 2 x the mechanical time constant of the turbine.
- **Tailrace tunnel:** The length of the penstock is assumed to be equal to the rated head of the Francis turbine. The diameter is fixed to 1.5 x the reference diameter of the Francis turbine.
- **Downstream reservoir:** The volume of this reservoir is assumed to be infinite. The water elevation is fixed to 0 masl (reference elevation).

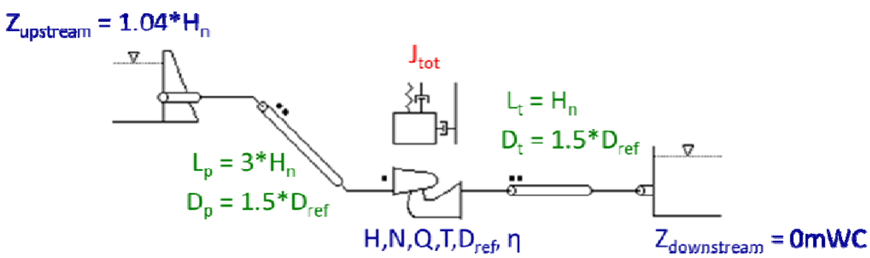


Fig. 4. Dimensioning rules defining the layout of the hydraulic power plant.

Finally, all this dimensioning procedure was applied to two different rated powers: 50 and 300 MW. The different dimensioning rules defined above lead to the following ranges:

- Mechanical time constant: $T_m = \frac{J_{tot}\omega^2}{P} = [5.5 - 9.6]$
- Water starting time constant: $T_w = \frac{Q}{H}\sum\frac{L}{gA} = [0.9 - 2.6]$
- Hadley criteria [22]: $Hadley = \frac{T_m}{T_w} = [2.35 - 9.36]$
- Wave reflection time: $T_r = \frac{2L_{penstock}}{a_{penstock}} = [0.15 - 2.5]$

2.2 Description of the control system

The control system used in the SIMSEN model is illustrated in Fig. 5. The control system is a PID turbine governor with both speed and power control loops combined with the permanent droop. An Anti-Reset Windup is used to limit the integral contribution of the PID when a saturation is reached. Finally, the output signal of the PID controller is limited by a rate limiter to guarantee a system response that fulfils the opening/closing laws of the turbine and by a saturation to remain within physical limits of the distributor.

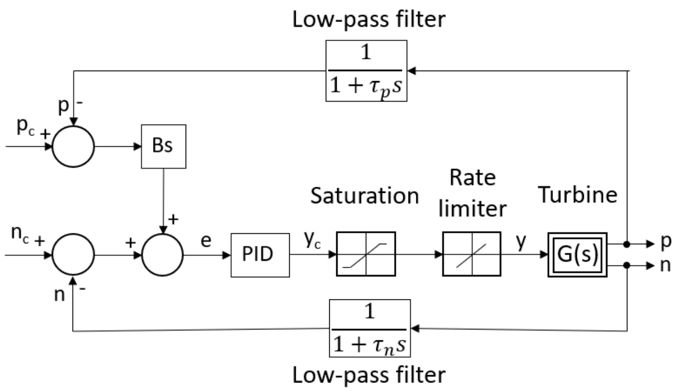


Fig. 5. SIMSEN model of the control system.

- ***pc*:** Dimensionless power set point.
- ***p*:** Dimensionless mechanical power generated by the Francis turbine.
- ***nc*:** Dimensionless speed set point.
- ***n*:** Dimensionless rotational speed of the Francis turbine.
- ***Bs*:** Permanent droop. $B_s = \frac{\Delta f}{f_n} / \frac{\Delta P}{P_n}$. *Bs* defines the contribution to primary reserve and is fixed to 4%.

The block diagram of the PID series used in this study case is presented in Fig. 6, where *K* is the proportional gain, *Ti* is the integral time constant, *Td* is the derivative time constant and *m* is the filter of the derivative term, range between 5 and 10. The block diagram of the PID controller in parallel is defined in Appendix 9.3.

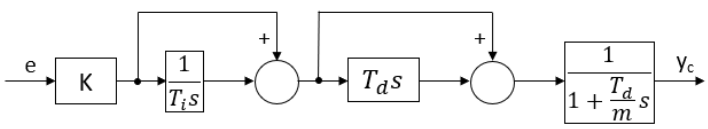


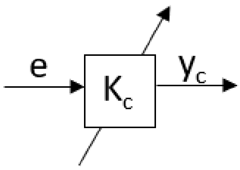
Fig. 6. Block diagram of the PID controller in SIMSEN software.

3. Methods to define the PID controller parameters

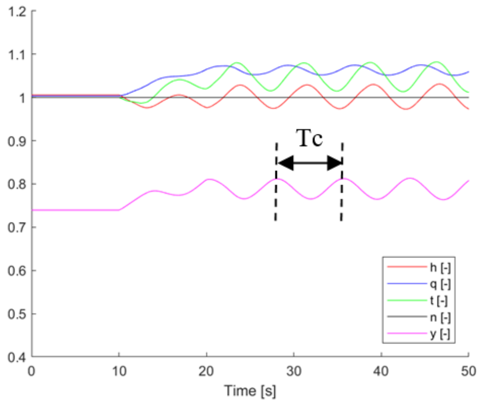
3.1 Ziegler-Nichols methods

The Ziegler–Nichols method is a heuristic method of tuning a PID controller. This method is divided in two different steps:

- The integral time constant *Ti* and the derivative time constant *Td* are set to zero. The proportional gain *Kc* is then increased until it reaches the limit of stability, see Fig. 7.



Block diagram of the proportional controller



Transient behaviour of the Francis turbine

Fig. 7. First step of the Ziegler-Nichols method: Limit of stability.

- According to the proportional gain obtained in the first step *Kc* and the oscillation period *Tc*, the *K*, *Ti*, and *Td* parameters of the PID controller in series are defined. A large number of tuning formulas exists in the literature [14] and the most popular of them are defined in the following table:

Table 1. Tuning formulas for different methods.

	PID parameters in serie		
Method name	**K**	**Ti**	**Td**
Classic Ziegler-Nichols	0.6 · Kc	0.5 · Tc	0.125 · Tc
Pessen Integral Rule	0.7 · Kc	0.4 · Tc	0.15 · Tc
Some Overshoot	0.33 · Kc	0.5 · Tc	0.33 · Tc
No Overshoot	0.2 · Kc	0.5 · Tc	0.33 · Tc

3.2 Time constant method

The guidelines to determine the controller parameters for turbine governor were established by Ransford [16], Hovey [11] and Paynter [15]. System performance is predicted on the basis of the mechanical time constant and the water starting time constant. A more elaborate method has been presented by Chaudry [2], who has also incorporated the effect of permanent droop *Bs*, as well as that of turbine and load self-regulation [20]. Hagihara et al. [9] expanded these methods to a PID governor in parallel for hydroelectric power plant on isolated grid. The advantage to define the parameter for a parallel controller structure is to eliminate the interdependence of these three parameters *Kc*, *Ki* and *Kd*. The equations of Hagihara et al. cannot be used directly in this case study because the hydroelectric power plant is assumed to be connected to an infinite electrical grid, but they serve as a basis for reflection to defined new equations. Other authors have also used this approach to determine parameters of the PID controller for primary control [1] [21].

4. Comparison of the methods

The test for primary control capability defined by the Swiss TSO, Swissgrid, is based on a frequency linear variation of 200 mHz in 10 seconds. The output power variation resulting from primary control response must be delivered within 30 seconds and remain between minimum and maximum threshold as depicted in Fig. 8.

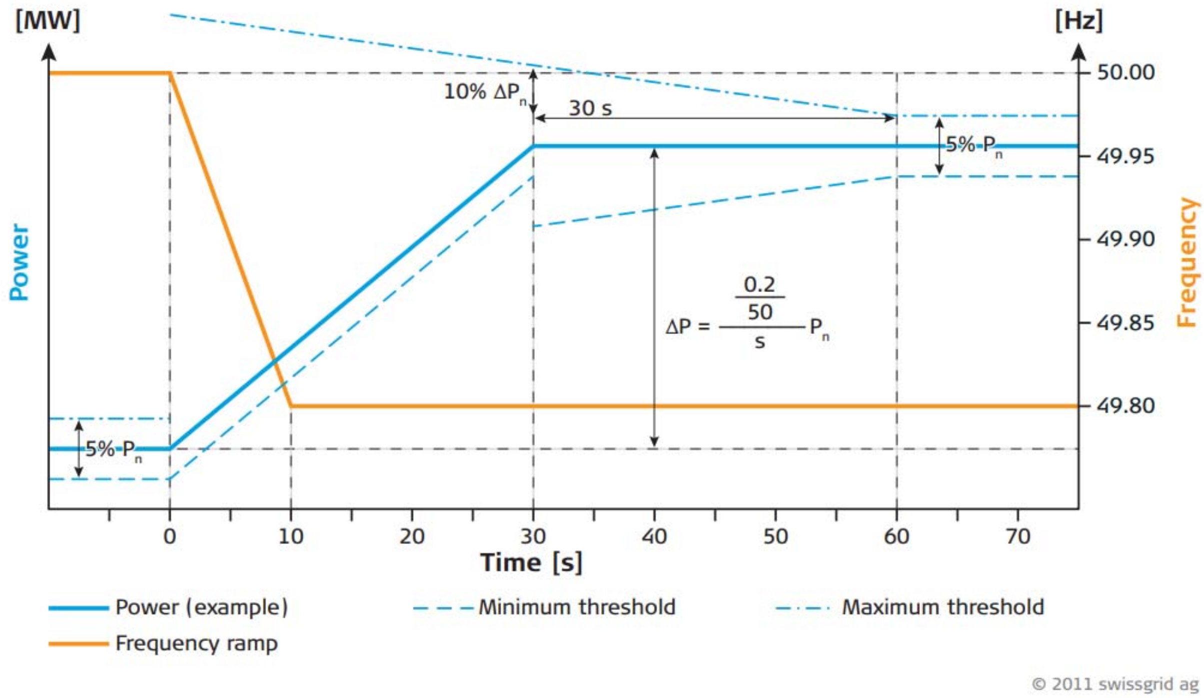


Fig. 8. Test for primary control capability defined by Swissgrid [17]

For this study case, the permanent droop was fixed to 4%. Thus, a frequency variation of 200 mHz induces a power variation of 10%, i.e. 5 MW if the mechanical power is fixed to 50 MW. In order to compare the 40 different Francis turbines, the system response for primary control capability are classified into 5 categories:

Swissgrid criteria validated (green point)	The output power response of the Francis turbine fulfilled the threshold defined by Swissgrid. The primary control is validated.
No respect of the minimum power (red point)	The output power response of the Francis turbine does not respect the minimum threshold imposed by Swissgrid.
Small oscillation of power (pink point)	Small oscillations of the output power response occur. This solution is not interesting for the grid stability.
Slow response of the PID controller (blue point)	The system response is too long and does not fulfil the minimum threshold defined by Swissgrid.
Unstable response of the PID controller (black point)	The output power response is unstable. The stability of the grid is compromised.

The following subsections present the results obtained with the PID parameters of the methods presented in Table 1.

4.1 Classic Ziegler-Nichols method

The parametrization proposed by the Classic Ziegler-Nichols method leads to results illustrated in Fig. 9. For small IEC discharge factor Q_{ED}, small oscillations of power occur and the stability of the grid is not guaranteed. This problem of instability is due to the proportional gain K which is too high. The second problem of this method is the minimum power for high IEC discharge factor Q_{ED}. The response of the output power to a frequency variation of 200 mHz is illustrated in Fig. 10 for two different Francis turbines.

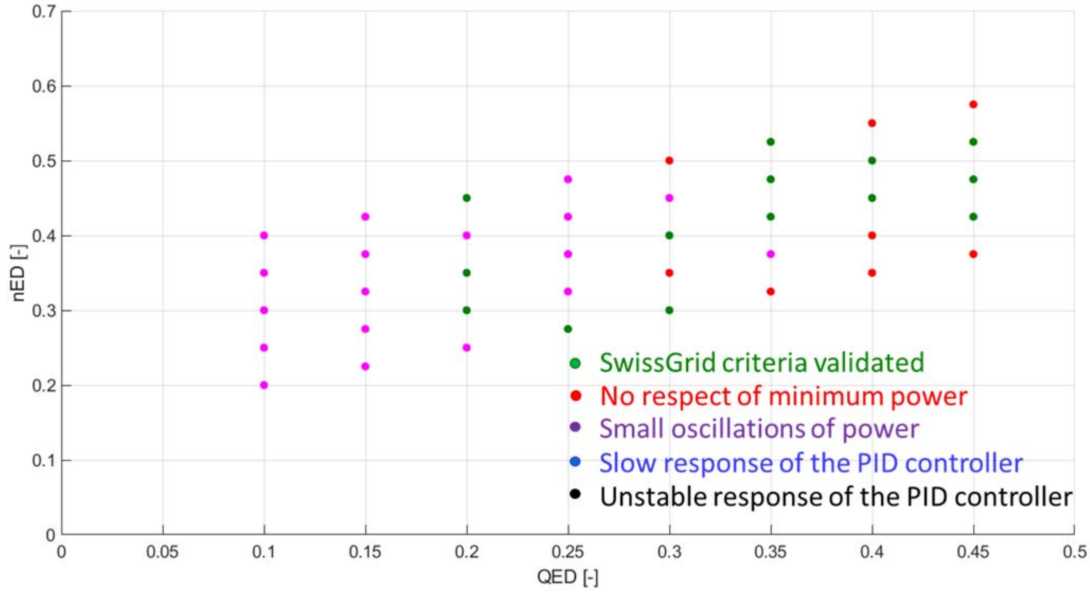
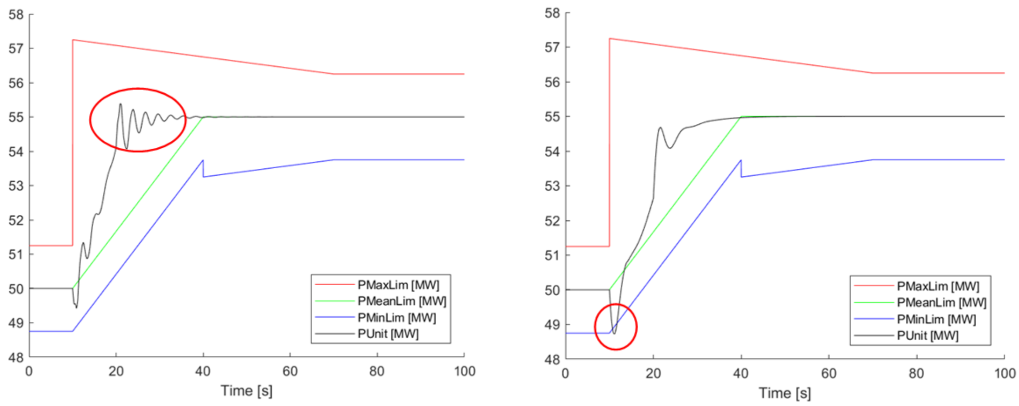
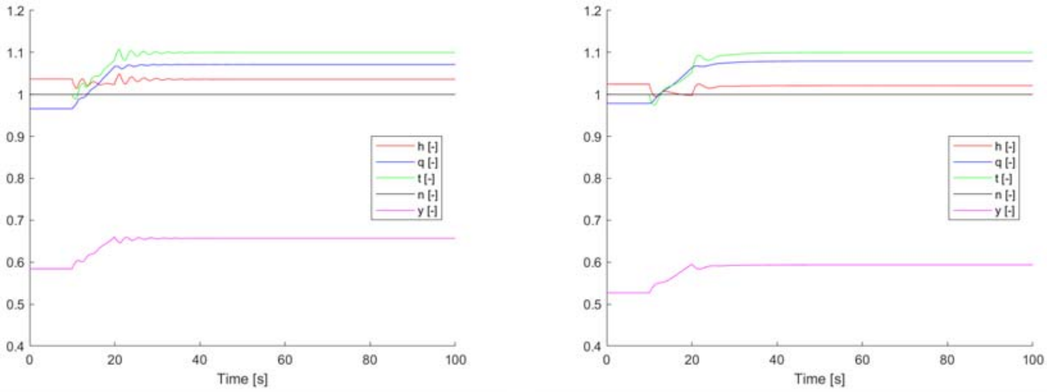


Fig. 9 Summary of the response to a frequency variation of 200 mHz for the 40 different Francis turbines (P_m = 50 MW), using Classic Ziegler-Nichols parameters.



a) n_{ED} = 0.400, Q_{ED} = 0.100 (Nq = 42.07) *b) n_{ED} = 0.375, Q_{ED} = 0.450 (Nq = 83.66)*

Fig. 10 Transient simulation of the response of the output power to a frequency variation of 200 mHz (P_m = 50 MW), using Classic Ziegler-Nichols parameters.



a) n_{ED} = 0.400, Q_{ED} = 0.100 (Nq = 42.07) *b) n_{ED} = 0.375, Q_{ED} = 0.450 (Nq = 83.66)*

Fig. 11 Transient behaviour of the Francis turbine after a frequency variation of 200 mHz (P_m = 50 MW), using Classic Ziegler-Nichols parameters.

4.2 Pessen Integral Rule method

The parametrization proposed by the Pessen Integral Rule Ziegler-Nichols method leads to results illustrated in Fig. 12. The increase of the proportional gain and the decrease of the integral time constant compared to the previous method makes the system less stable for the small IEC discharge factor Q_{ED}. The problem of minimum power for high IEC discharge factor Q_{ED} is still present, see Fig. 13.

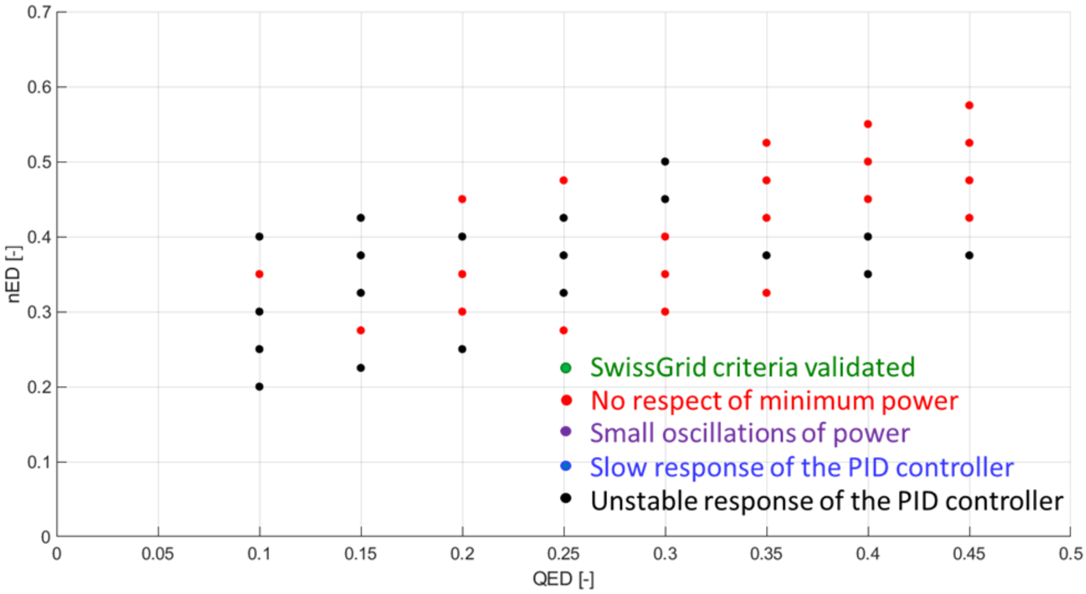
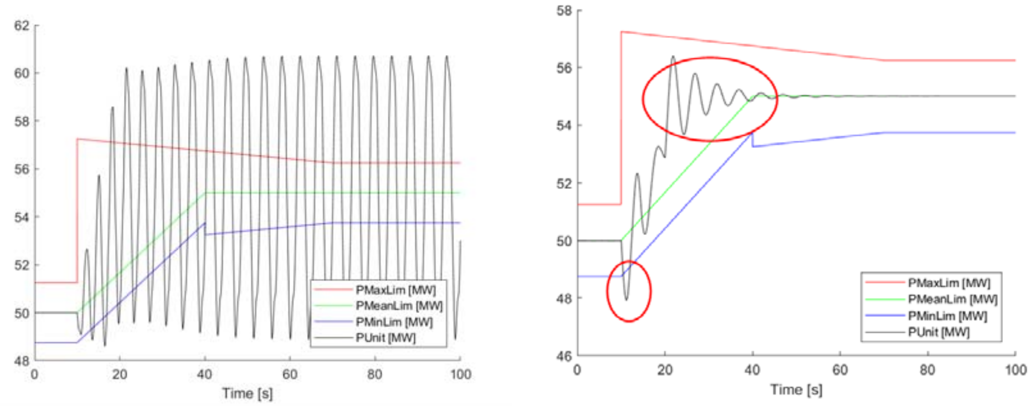
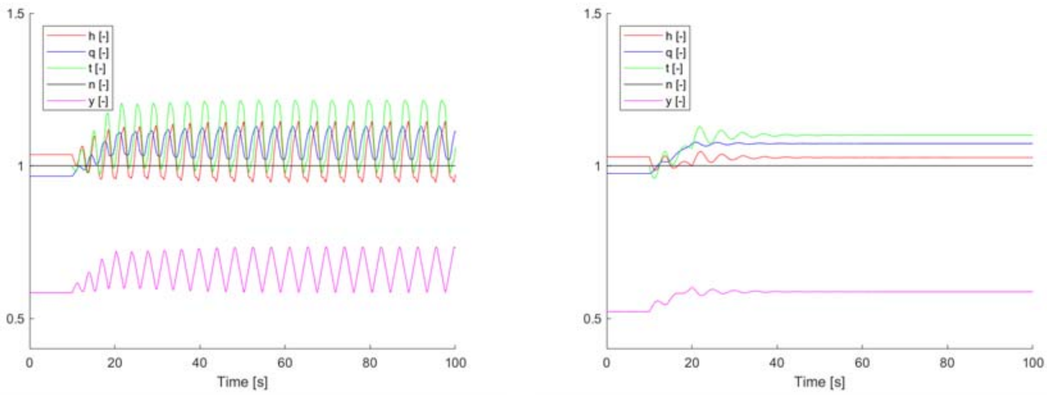


Fig. 12 Summary of the response to a frequency variation of 200 mHz for the 40 different Francis turbines (P_m = 50 MW), using Pessen Integral rule parameters.



a) n_{ED} = 0.400, Q_{ED} = 0.100 (Nq = 42.07) b) n_{ED} = 0.375, Q_{ED} = 0.450 (Nq = 83.66)

Fig. 13 Transient simulation of the response of the output power to a frequency variation of 200 mHz (P_m = 50 MW), using Pessen Integral rule parameters.



a) n_{ED} = 0.400, Q_{ED} = 0.100 (Nq = 42.07) b) n_{ED} = 0.375, Q_{ED} = 0.450 (Nq = 83.66)

Fig. 14 Transient behaviour of the Francis turbine after a frequency variation of 200 mHz (P_m = 50 MW), using Pessen Integral rule parameters.

4.3 Some Overshoot method

The Some Overshoot method proposes a proportional gain divided by 2 compared to the classic Ziegler-Nichols method. The small oscillations of the mechanical power are therefore reduced, but are still present, see Fig. 16. The multiplication of the derivative time constant by a factor of 2.6 solved the problem of non-compliance with the minimum power requirement, see Fig. 15.

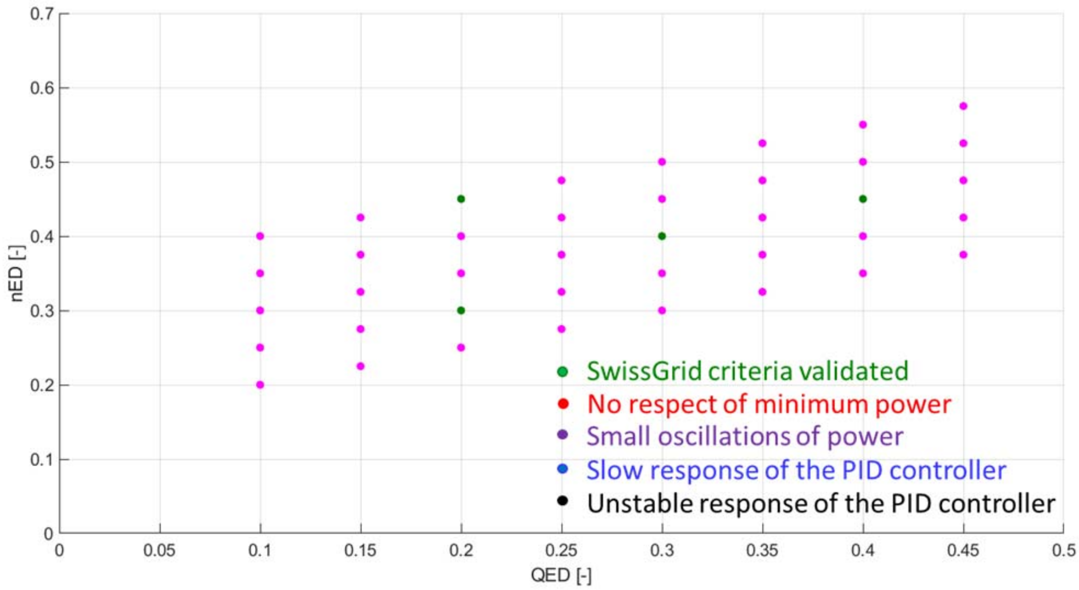
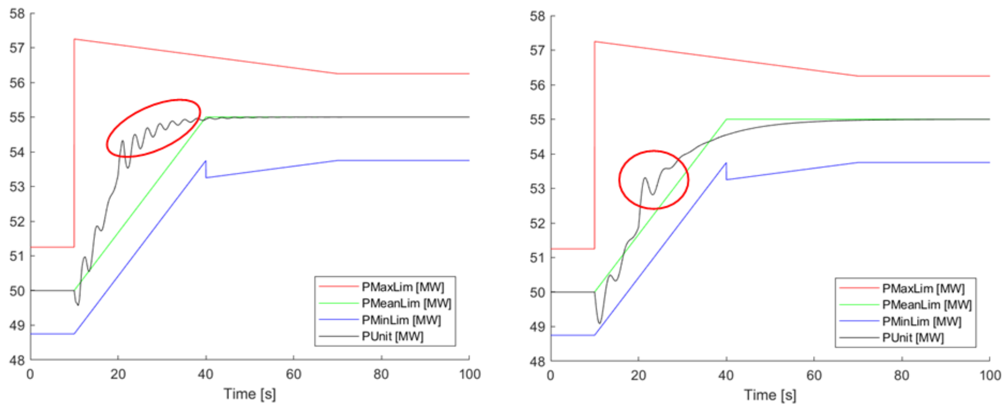
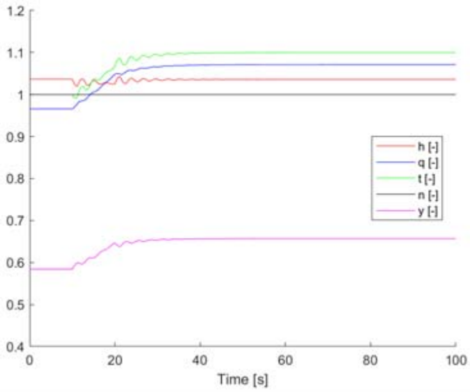


Fig. 15 Summary of the response to a frequency variation of 200 mHz for the 40 different Francis turbines (P_m = 50 MW), using Some Overshoot parameters.

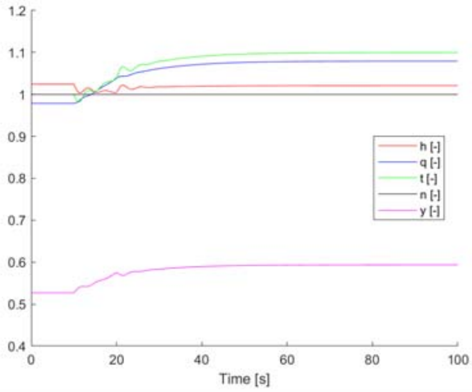


a) n_{ED} = 0.400, Q_{ED} = 0.100 (Nq = 42.07)   *b) n_{ED} = 0.375, Q_{ED} = 0.450 (Nq = 83.66)*

Fig. 16 Transient simulation of the response of the output power to a frequency variation of 200 mHz (P_m = 50 MW), using Some Overshoot parameters.



a) $n_{ED} = 0.400$, $Q_{ED} = 0.100$ *(Nq = 42.07)*



b) $n_{ED} = 0.375$, $Q_{ED} = 0.450$ *(Nq = 83.66)*

Fig. 17 Transient behaviour of the Francis turbine after a frequency variation of 200 mHz ($P_m = 50$ *MW), using Some Overshoot parameters.*

4.4 No Overshoot method

For the No Overshoot method, the proportional gain is further reduced. Thus, the small oscillations have disappeared and the majority of the 40 test cases comply with the limitations imposed by Swissgrid. However, with this small value of the proportional gain, the system response is sometimes too slow, see Fig. 18.

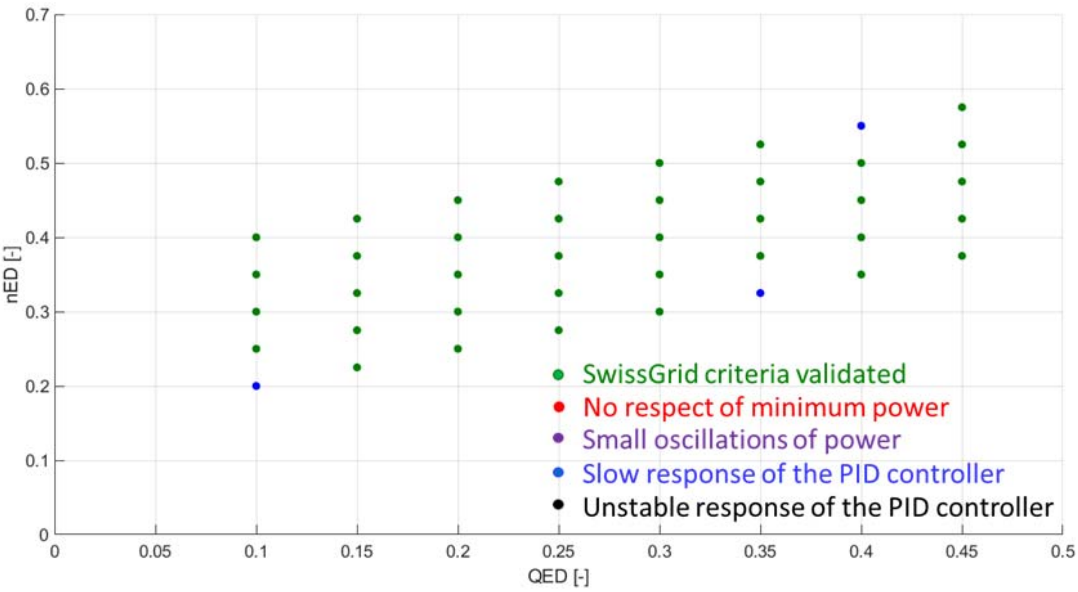
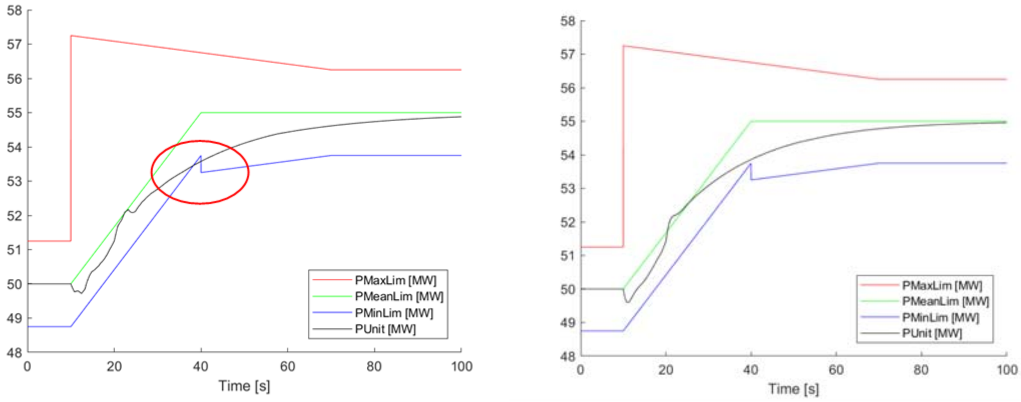
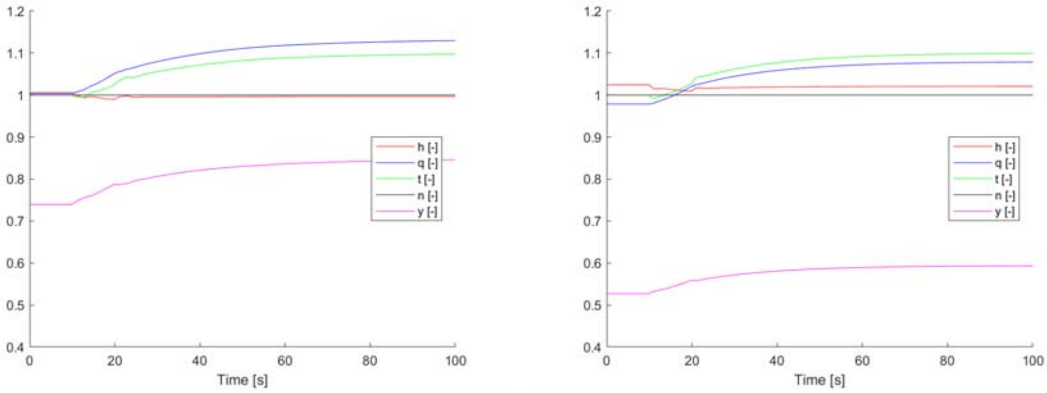


Fig. 18 Summary of the response to a frequency variation of 200 mHz for the 40 different Francis turbines ($P_m = 50$ *MW), using No Overshoot parameters.*



a) n_{ED} = 0.200, Q_{ED} = 0.100 (Nq = 21.04) *b) n_{ED} = 0.375, Q_{ED} = 0.450 (Nq = 83.66)*

Fig. 19 Transient simulation of the response of the output power to a frequency variation of 200 mHz (P_m = 50 MW), using No Overshoot parameters.



a) n_{ED} = 0.200, Q_{ED} = 0.100 (Nq = 21.04) *b) n_{ED} = 0.375, Q_{ED} = 0.450 (Nq = 83.66)*

Fig. 20 Transient behaviour of the Francis turbine after a frequency variation of 200 mHz (P_m = 50 MW), using No Overshoot parameters.

4.5 Method with Optimized parameters

With the analysis carried out on the previous methods, the PID parameters were optimized in order to solve the various problems highlighted in the previous sections.

- $K = 0.3 \cdot K_c$,
- $T_i = 0.5 \cdot T_c$,
- $T_d = 0.114 \cdot T_c$.

With these new equations, the system response fulfills the guarantees imposed by Swissgrid for each Francis turbine. The response of the output power to a frequency variation of 200 mHz is illustrated in Fig. 22 for two different Francis turbines.

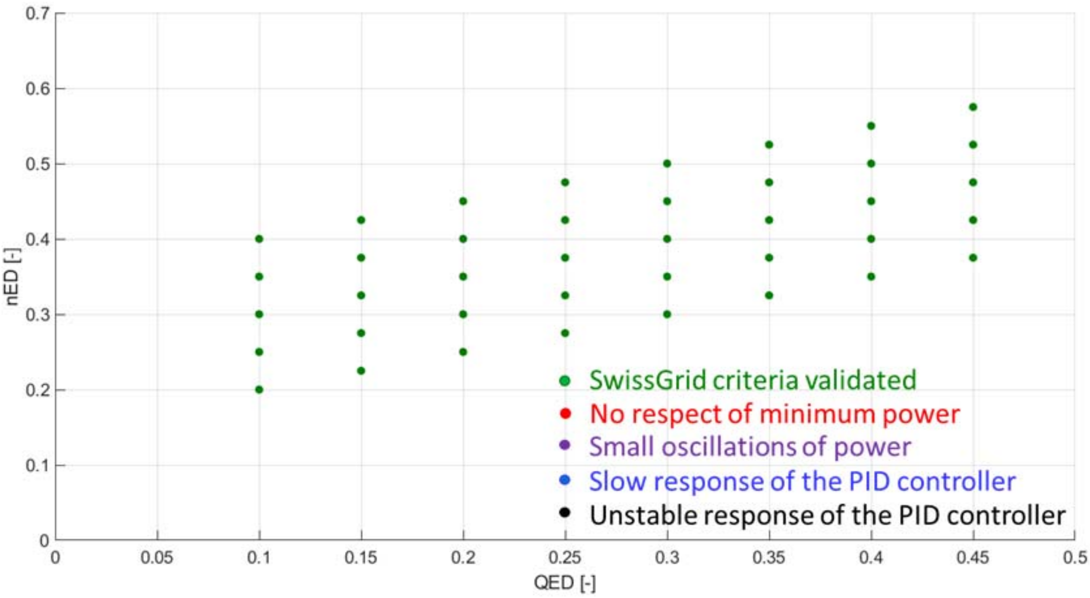
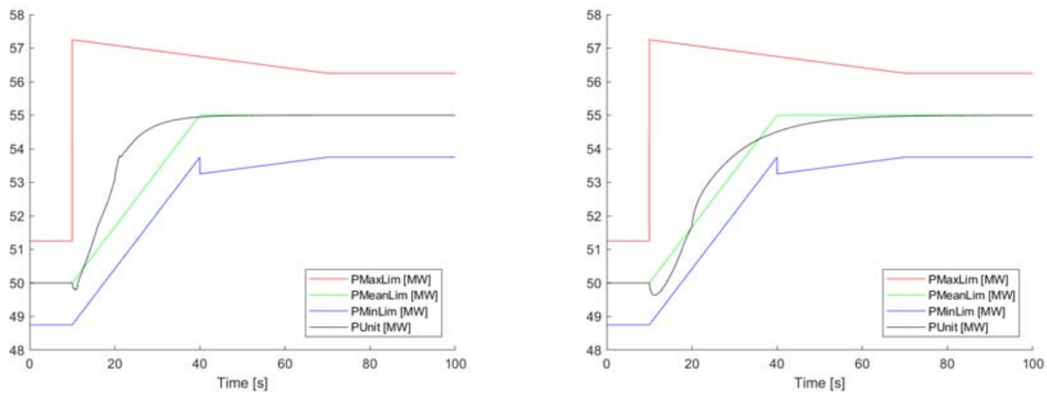
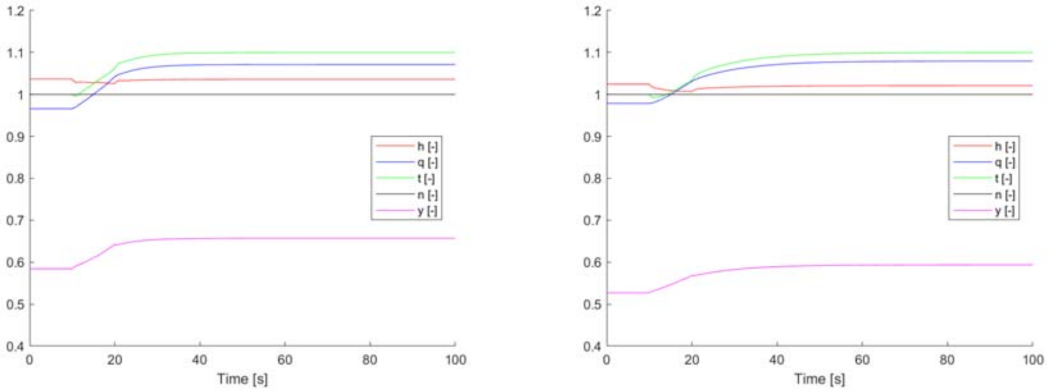


Fig. 21 Summary of the response to a frequency variation of 200 mHz for the 40 different Francis turbines (P_m = 50 MW), using optimized parameters.



a) n_{ED} = 0.400, Q_{ED} = 0.100 (Nq = 42.07) *b) n_{ED} = 0.375, Q_{ED} = 0.450 (Nq = 83.66)*

Fig. 22 Transient simulation of the response of the output power to a frequency variation of 200 mHz (P_m = 50 MW), using optimized parameters.



a) n_{ED} = 0.400, Q_{ED} = 0.100 (Nq = 42.07) *b) n_{ED} = 0.375, Q_{ED} = 0.450 (Nq = 83.66)*

Fig. 23 Transient behaviour of the Francis turbine after a frequency variation of 200 mHz (P_m = 50 MW), using optimized parameters.

4.6 Time constant method

The aim of this method is to define new equations to obtain the optimum parameters defined by the Ziegler-Nichols method in the previous section for the 40 Francis turbines with a rated power fixed to 50MW. The linear regressions underlying these new equations are illustrated in Appendix 9.4. The new equations for the PID parameters in parallel are defined below and are valid for a turbine hill chart with guide vane openings divided by the guide vane opening at the best efficiency point:

- Proportional gain: $K_{p,50MW} = 0.0347 \cdot \frac{T_m^3}{T_w} + 6.8627$
- Integral gain: $K_{I,50MW} = 0.7852 \cdot \frac{T_m}{T_w} + 0.7169$
- Derivative gain: $K_{d,50MW} = 2.9441 \cdot T_m - 11.9600$

These equations are a function of mechanical time constant, despite the fact that the natural inertia of the hydraulic turbine plays no role in the transient behaviour because the system is connected to an infinite electrical grid. Nevertheless, *Tm* is representative of the influence of the mechanical power relative to the rotational speed.

Moreover, the increase of the rated power of the Francis turbines to 300 MW implies an increase in inertia and therefore in mechanical time constant. Thus, in order to maintain stable PID parameters for these more powerful hydraulic turbines, the above parameters have been multiplied by a correction constant equal to 0.6.

- Proportional gain: $K_{p,300MW} = 0.6 \cdot K_{p,50MW}$
- Integral gain: $K_{I,300MW} = 0.6 \cdot K_{I,50MW}$
- Derivative gain: $K_{d,300MW} = 0.6 \cdot K_{d,50MW}$

As these parameters are defined for a PID controller in parallel, they should be converted into a coefficient for PID controllers in series (see Appendix 9.2). It is interesting to note that this correction constant only applies to the proportional gain K for PID controller in series. With these new equations, the system response of each Francis turbine with a rated power fixed to 50 MW fulfils the limitations imposed by Swissgrid, see Fig. 24. The transient simulation of the response of the output power for 4 Francis turbines with a rated power fixed to 300 MW are illustrated in Appendix 9.5.

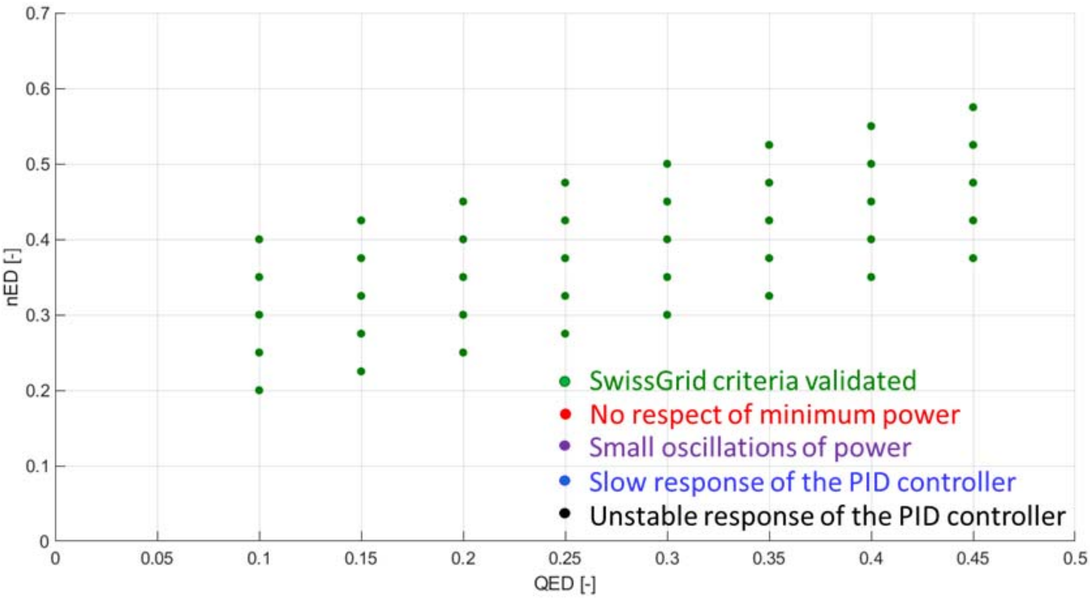
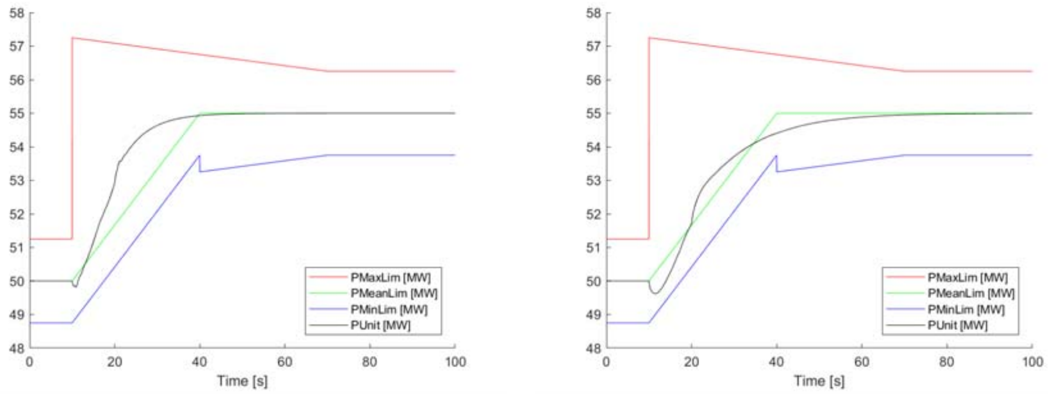
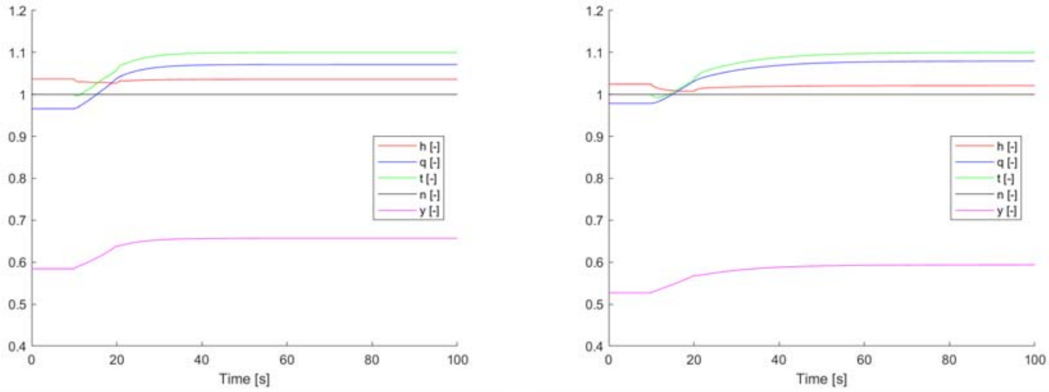


Fig. 24 Summary of the response to a frequency variation of 200 mHz for the 40 different Francis turbines (P_m = 50 MW), using time constant method.



a) n_{ED} = 0.400, Q_{ED} = 0.100 (Nq = 42.07) *b) n_{ED} = 0.375, Q_{ED} = 0.450 (Nq = 83.66)*

Fig. 25 Transient simulation of the response of the output power to a frequency variation of 200 mHz (P_m = 50 MW), using time constant equations.



a) n_{ED} = 0.400, Q_{ED} = 0.100 (Nq = 42.07) *b) n_{ED} = 0.375, Q_{ED} = 0.450 (Nq = 83.66)*

Fig. 26 Transient behaviour of the Francis turbine after a frequency variation of 200 mHz (P_m = 50 MW), using time constant equations.

5. Conclusion

The assessment of the frequency primary control capability of a hydraulic power plant requires the determination of the PID parameters of the control system. Two different methods to assess these parameters were compared: the Ziegler-Nichols method using the stability limit and the time constant method using the mechanical time constant and the water starting time. For each method, the test for primary control capability defined by Swissgrid was applied to 40 different Francis turbines with water head from 30 to 500 mWC. For each Francis turbine, a SIMSEN model was realised with a realistic performance hill chart and a dimensioning defined by statistical laws.

The systematic analysis of the system response of each turbine were used to determine optimized parameters for each method. The Ziegler-Nichols method is robust and can be applied regardless of the mechanical power of the Francis turbine. The time constant method is based on the geometric quantities of the layout and avoids a search for the stability limit. A correction constant must be applied depending on the power of the hydraulic turbine.

With these new equations, the maximum primary ancillary service capabilities of the hydraulic power plant compatible with Transmission System Operator requirements can be easily assessed by gradually reducing the permanent droop. This approach will be further extended to isolated grid operation.

6. Acknowledgments

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7. Nomenclature

ARW	Anti-reset Windup	*Kd*	Derivative gain (parallel)	P_m	Mechanical power [W]
Bs	Permanent droop [%]	*Ki*	Integral gain (parallel)	*Q*	Discharge [m^3/s]
D_{ref}	Reference diameter [m]	*Kp*	Proportional gain (parallel)	*Tc*	Oscillation period (stability limit) [s]
Fgrid	Frequency of the grid [Hz]	*N*	Rotational speed [rpm]	*Ti*	Integral time constant (series) [s]
H_n	Rated head [mWC]	*n*	Dimensionless speed [-]	*Td*	Derivative time constant (series) [s]
J_{tot}	Total inertia [kgm^2]	*nc*	Dimensionless speed set point [-]	*Tm*	Mechanical time constant [s]
K	Proportional gain (series)	*p*	Dimensionless power [-]	*Tw*	Water starting time [s]
Kc	Proportional gain (stability limit)	*pc*	Dimensionless power set point [-]	*ν*	IEC specific speed [-]

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9. Appendix

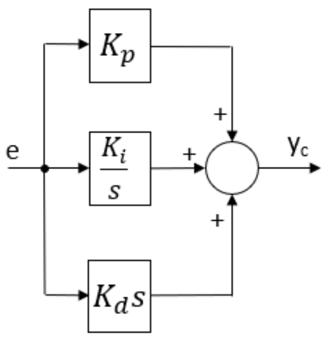
9.1 Dimensionless number

- Specific speed $N_q = \dfrac{N_n (Q_n)^{1/2}}{(H_n)^{3/4}}$,
- IEC speed factor $n_{ED} = \dfrac{n \cdot D_{ref}}{\sqrt{gH}}$,
- IEC Discharge factor $Q_{ED} = \dfrac{Q}{{D_{ref}}^2 \cdot \sqrt{gH}}$,

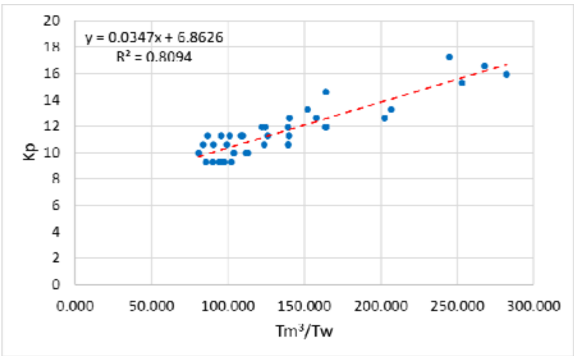
9.2 Equations to convert PID parameters

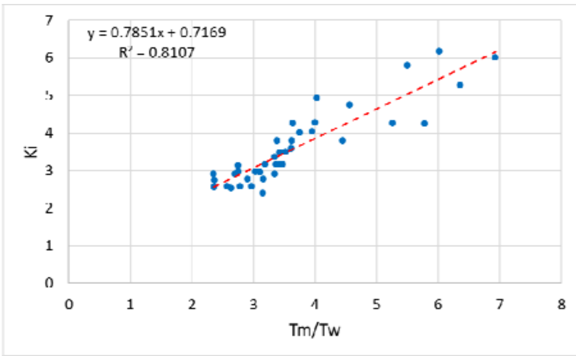
Parallel to series	Series to parallel
$K = \dfrac{K_p + \sqrt{K_p^2 - 4K_i K_d}}{2}$	$K_p = K\left(1 + \dfrac{T_d}{T_i}\right)$
$T_i = \dfrac{K}{K_i}$	$K_i = \dfrac{K}{T_i}$
$T_d = \dfrac{K_d}{K}$	$K_d = K \cdot T_d$

9.3 Block diagram of the PID controller in parallel



9.4 Linear regression for the proportional, integral and derivative gain optimized with the Ziegler-Nichols method





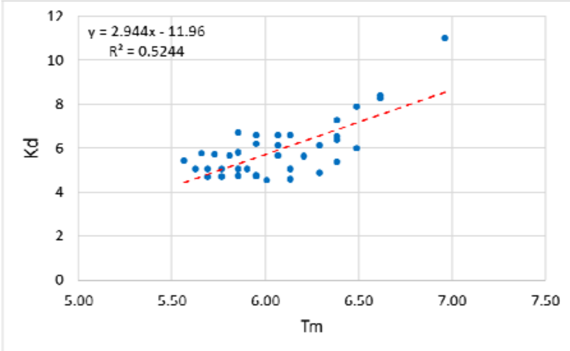
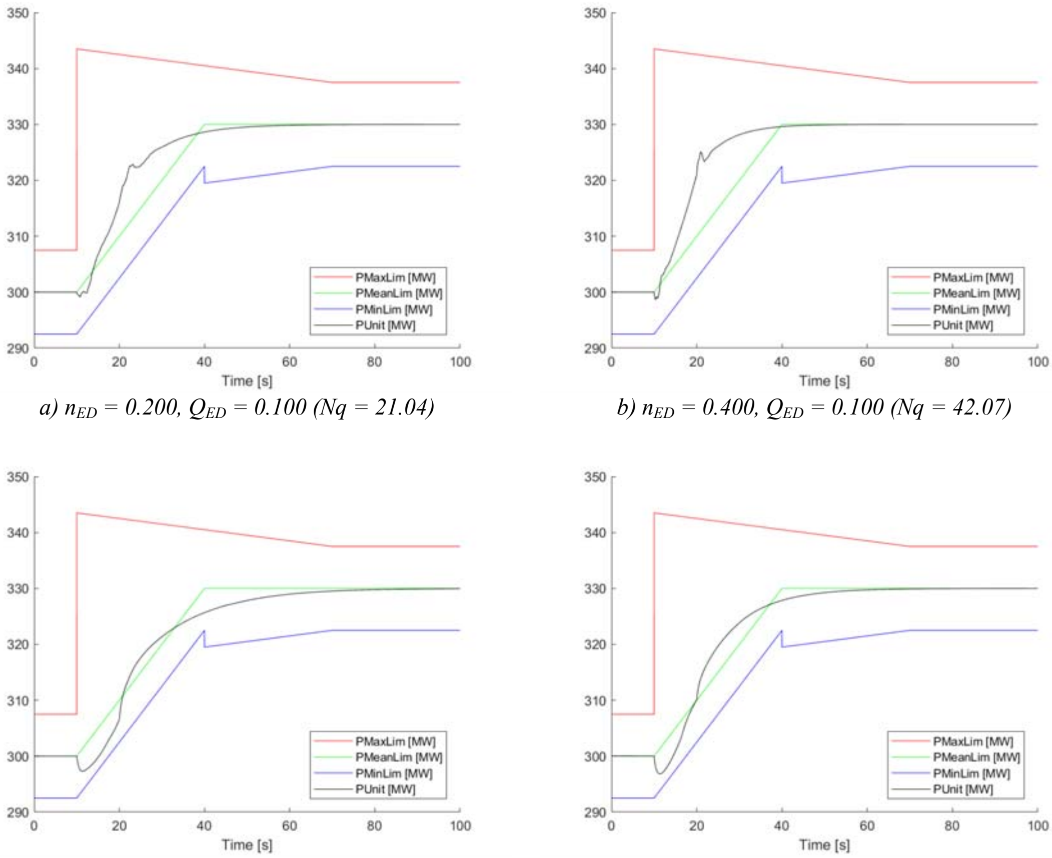


Fig. 27 Linear regression for the proportional, integral and derivative gain optimized with the Ziegler-Nichols method

9.5 Transient simulation of the response of the output power to a frequency variation of 200 mHz (Pm = 300 MW), using time constant equations.



a) n_{ED} = 0.200, Q_{ED} = 0.100 (Nq = 21.04)

b) n_{ED} = 0.400, Q_{ED} = 0.100 (Nq = 42.07)

c) n_{ED} = 0.375, Q_{ED} = 0.450 (Nq = 83.66)

d) n_{ED} = 0.525, Q_{ED} = 0.350 (Nq = 103.3)

Fig. 28 Transient simulation of the response of the output power to a frequency variation of 200 mHz (P_m = 300 MW), using time constant equations.

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