Primary frequency control in isolated power grids

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Abstract - Satisfactory operation of a power system is predominantly assured by keeping voltage levels and frequency in narrow tolerance bands. As any other system, a power system is frequently subject to changes in operating conditions, which find their cause in faults or the loss of a major generating unit. With the goal of controlling the two variables, specific devices are often equipped in generating units. Frequency regulation, the topic covered in this work, is directly related to the active power balance in the system. In order for a unit to be able to control its mechanical power and consequently system frequency, a device called a speed governor is installed. Performance of said speed governors varies widely depending on the type of resource a power plant is based on. The usual practice in power systems is to rely on thermal power plants to conduct primary frequency control, especially in isolated or weaker grids where few generators exist.

The core objective of this work is to determine to what degree hydroelectric power plants can contribute to primary frequency regulation in isolated grids with a high share of wind power. This goal is achieved by first defining the relevant dynamic models of hydraulic turbines and speed governors and then establishing parameter requirements that lead to optimum governor performances. Lastly, transient stability simulations are performed in order to form a conclusion about the influence of hydroelectric speed governing in system frequency control.

Index Terms - Transient Stability, Dynamic Models, Turbine-Governor Systems, Speed Governor Tuning

I. INTRODUCTION

Power systems are prone to suffer from disturbances, which may lead to wide excursions of its variables and cause equipment damage. Disturbances may be either small or large depending on their strength. The former occur as continuous small load changes in the system and can usually be interpreted as a small-signal stability problem. The latter are related to events of a severe nature, such as transmission line short-circuit, the tripping of a generator or a high amount of load and fall in the category of the transient stability problem. The larger a power system, the stronger and more resilient it is with respect to the effects of large disturbances, which cause the most significant excursions of system variables [1].

Controllers are used to contribute to the safe operation of the power system by maintaining system voltages, frequency, and other system variables within acceptable limits. Regulation of frequency is closely related to active power control while voltage regulation is closely related to reactive power control. These two regulation solutions can therefore be treated separately for a large class of problems [1].

The fact that system frequency and active power balance are tied together implies that a change in active power demand at one point in the network is propagated throughout the system as a change in frequency. The availability of many generators supplying power, distributed throughout the system, implies that a means to allocate power demand change to the generators should be provided. A controller denominated speed governor is usually installed on each generating unit. It has the capability to control the unit’s mechanical power and the system frequency, a function commonly labelled as load-frequency control or primary frequency control. A unit’s mechanical power is regulated by acting on the amount of resource available for energy transformation [1].

The main resources used for electrical energy generation are the kinetic energy of water and the thermal energy extracted from fossil fuels or nuclear fission. The generating unit, or prime mover, converts one resource into mechanical energy which is turned converted to electrical energy by synchronous generators [1]. The usual practice is to resort to thermal power plants for the system frequency control duty to the detriment of the hydroelectric plants. The explanation resides in the inner nature of the process of energy conversion happening in each power plant. While for thermal power plants the principle of operation is the heated and high pressure steam which is converted into rotating energy in a steam turbine, in hydroelectric power plants mechanical motion is caused by the kinetic energy of water fed from the dam through a penstock.

Naturally, the time constants of the processes associated with energy conversion in thermal power plants are slightly smaller than those of a hydroelectric plant.

Nevertheless, in some cases, such as weaker or isolated power grids, hydroelectric power plants exhibit a significant contribution to system frequency regulation [2]. The main objectives of this work are to illustrate the nature of the hydroelectric power plants speed governors in primary frequency control and to establish relationships between governor parameters and its hydro unit characteristic values.

Finally, transient stability simulations are performed in an isolated grid with a high share of wind power, in order to observe to what degree hydroelectric power plants are capable of contributing to frequency regulation.

II. HYDRAULIC TURBINES AND GOVERNING SYSTEMS

A. Generalities

Four basic elements constitute hydraulic power plants, which are indispensable for the process of energy conversion from water: a way of creating a head level (a forebay), a
penstock to convey water, a hydraulic turbine and an electric generator. These essential elements are depicted in Fig. 1.

![Fig. 1 - Schematic of a hydroelectric power plant [1].](image1)

The performance of a hydraulic turbine is influenced by the characteristics of the water column feeding the turbine, namely, the effects of water inertia, water compressibility and pipe wall elasticity in the penstock. Water inertia effect causes changes in turbine flow to lag behind changes in turbine gate opening or closing. The effect of pipe wall elasticity gives rise to travelling waves of pressure and flow in the conduits, a phenomenon commonly known as the “water hammer” [1]. Water hammer is defined as the occurrence of pressure fluctuations, caused when the system undergoes a change from one operational steady-state condition to another, that is, upon a change in the rate of water flow following a gate closing or opening. Pressure waves consequently travel along the penstock, subjecting pipe walls to great stresses. The classical solution to this water hammer problem is to insert a device called a surge tank, a large tank that is usually located between the conduit and the penstock, in which water flows, converting kinetic energy to potential energy [2].

B. Modelling

Turbine-governor models are designed to give representations of the effects of hydroelectric power plants on power system stability. Accurate mathematical modelling of the hydraulic components includes the dynamic representation of penstock, surge tank if existent and the hydraulic losses of those elements. A functional diagram of the representation of a hydro turbine and its speed governor within the power system environment is portrayed in Fig. 2.

![Fig. 2 - Functional block diagram of hydraulic turbine generating system [1].](image2)

1) Hydraulic Turbines

Precise modelling of hydraulic turbines requires that transmission-like wave effects, which take place in the elastic-walled pipes conducting the compressible fluid, are included. It has been observed that the speed of propagation of said travelling waves is approximately equal to the speed of sound in water, which is 1497 meters per second. Consequently, nonlinear travelling wave models are especially important for hydro power plants with long penstocks, which is not the case in the hydro site study demonstrated in this work [1].

A nonlinear model is described, as it is more appropriate for large signal time domain simulations or transient stability studies. A block diagram for dynamic simulations of a hydraulic turbine with penstock, assuming unrestricted head and tailrace and without surge tank is shown in Fig. 3. The penstock is modelled assuming an incompressible fluid and rigid conduit of length $L$ and cross-section $A$. Penstock head losses $h_I$ are proportional to flow squared and $f_p$ is the head loss coefficient, usually ignored [4].

Pertaining to the laws of momentum, the rate of change of flow in the conduit is given by [4]

$$\frac{dq}{dt} = (h_0 - h - h_I)\frac{A}{L}$$  \hspace{1cm} (1)

where,

- $q$ = turbine flow rate, in m$^3$/sec
- $A$ = penstock cross-section area, in m$^2$
- $L$ = penstock length, in m
- $a_g$ = acceleration due to gravity, in m/sec$^2$
- $h_0$ = static head of water column above turbine, in m
- $h$ = head at the turbine admission, in m
- $h_I$ = head loss due to friction in the conduit, in m.

![Fig. 3 - Hydraulic turbine nonlinear model assuming inelastic water column [4].](image3)
Taking \( h_0 \) as the base head value \( h_{base} \), which is equal to reservoir head minus the tailrace head, and setting \( q_{base} \) as the turbine flow rate with gates fully open, real gate position \( \bar{g} \) equal to 1 pu, expression (1) in per unit yields

\[
\frac{d\bar{q}}{dt} = \frac{(1-\bar{h} - \bar{h}_r)}{T_W} \tag{2}
\]

where variables with superbars are per unit ones. The term \( T_W \), given in seconds, is called the water time constant or water starting time in the penstock and represents the time required for a head \( h_{base} \) to accelerate the water in the penstock from standstill to the flow rate \( q_{base} \). It is written as

\[
T_W = \left( \frac{L}{A} \right) \frac{q_{base}}{h_{base} \cdot \cdot \cdot} \tag{3}
\]

From here on, the penstock head losses \( h_l \) are not taken into consideration. Turbine characteristics define base flow as a function of head and real gate position, \( q = f(gate, head) \), which is expressed in per unit as

\[
\bar{q} = \bar{g} \sqrt{\bar{h}} \tag{4}
\]

Mechanical power available from an ideal hydraulic turbine is the product of hydraulic head available and water flow rate, multiplied by appropriate conversion factors. Real turbine efficiency is not 100\%, a fact that is accounted for by subtracting the no load flow, that weights the turbine fixed no load power losses, from the net flow giving the difference as the effective flow. A term representing the speed deviation damping effect is also to be included, which is a function of gate position. The per unit turbine mechanical power \( P_m \), on generator MVA base, is thus expressed as

\[
\bar{P}_m = A_t \bar{h} (\bar{q} - \bar{q}_{nl}) - D_{turb} \bar{g} \bar{\Delta} \bar{w} \tag{5}
\]

In the above formula, \( \bar{q}_{nl} \) corresponds to the per unit no load flow, \( D_{turb} \), the turbine damping coefficient typically takes values in the range of 0.5 to 2.0 and \( A_t \) is a proportionality factor assumed constant, calculated using turbine MW rating and generator MVA base

\[
A_t = \frac{1}{\bar{h}_r (\bar{q}_r - \bar{q}_{nl})} \frac{\text{Turbine MW rating}}{\text{Generator MVA rating}} \tag{6}
\]

where \( \bar{h}_r \) is defined as the per unit head at rated flow (usually 1 pu) and \( \bar{q}_r \) is the per unit flow at rated load.

Next, a linear model is described. Linearizing the basic penstock and turbine relationships, (4) and (2) as presented in the nonlinear model, and neglecting friction losses, Fig. 3 simplifies to the block diagram in Fig. 4.

As can be inferred from Fig. 4, change in mechanical power output in terms of gate position, speed deviation and constructive parameters is expressed as [4]

\[
\Delta P_m = A_t (1 - T_1) \frac{\Delta \bar{g}}{1 + T_2 s} - D_{turb} \bar{g} \Delta \bar{w} \tag{7}
\]

where,

\[
\bar{g}_0 = \text{per unit real gate opening at operating point},
\]

\[
T_1 = (\bar{q}_0 - \bar{q}_{nl}) T_W, \text{ in s.}
\]

\[
T_2 = \frac{\bar{g}_0 T_W}{2}, \text{ in s.}
\]

\[
\bar{q}_0 = \text{per unit steady state flow rate at operating point.}
\]

Note that \( \bar{q}_0 = \bar{g}_0 \) in (7). If the damping term \( D \) is neglected a similar expression to the usual classical turbine-penstock transfer function is achieved

\[
\Delta \bar{P}_m = \frac{1 - \bar{g}_0 T_W s}{1 + \bar{g}_0 T_W s} A_t \tag{8}
\]

where the term \( \bar{g}_0 T_W \) can be seen as an approximation to the effective water starting time constant for small perturbations around a specific operating point. This means that, contrary to the nonlinear model where \( T_W \) is determined using rated values of head and flow, therefore making it a fixed value, when using the linear model the effective water starting time corresponds to the chosen operating condition and needs to be adjusted each time that operating condition is modified.

2) Speed Governors

The governing system is responsible for assuring turbine-governor primary speed regulation and therefore frequency and active power, upon detecting load variations. The control mechanism includes equipment such as relays, servomotors, pressure or power amplifying devices, levers and linkages. The speed governor normally actuates on the governor controlled gates which in turn regulates the water inlet to the turbine. Fig. 5 depicts the general hydraulic governing system.

The primary speed control function consists on feeding back the speed error to control gate position. The controller compares the speed measured in the prime mover with the reference speed (set point input). The resulting error from this comparison is a signal that the governor uses to command the actuator, which in turn acts on the control device, operating the gate position. Since the goal is to ensure satisfactory and stable parallel operation of multiple generating units, speed controllers are provided with a droop characteristic, each. The purpose of this steady state droop is to make sure that load will be equally shared between all units, otherwise they would compete with each other trying to control system frequency to

Fig. 4 - Linear turbine model with inelastic water column [4].

Fig. 5 - Hydraulic plant governing system schematic [3].
its own setting. For each unit, it determines the amount of change in output that the unit produces in response to a change in speed. A typical value for the permanent speed droop is 5%, which means that, assuming the generator is supplying an isolated load, a speed deviation of 5% causes a 100% change in power output [1].

The setting introduced in the last paragraph however, is not enough for the role of ensuring stability operation of the unit. Consequently, a requirement for a large transient droop with a relatively long resetting time is defined. This requirement is met by feeding back real gate position into a transient gain reduction compensator as show in Fig. 6. In this fashion, gate movement will be limited or delayed, in order to allow for water flow to catch up and therefore the power delivered by the turbine. This delay on gate position movement depends on the reset time \( T_R \), during which the effect of temporary droop prevails over the permanent droop, leaving the latter to determine the steady state response of the unit. Evidently, the governor will exhibit high droop setting \( R_T \), hence low gain during transient condition, and the aforementioned permanent droop \( R_P \) or high gain in steady state mode [1].

The block diagram of Fig. 6 represents the hydro governing system, with all signals given in per unit, inferred from the above transfer functions, which by association with the turbine model in Fig. 3 provides a unified nonlinear model of hydro turbine and governing system. It should be noted that gate movement is rated limited (equal to the reciprocal of the time taken for the gates to move from fully open to fully closed position), a direct consequence of the fact that if the gate is closed too rapidly the resulting pressure may severely damage the penstock [1]. Linearizing the model of Fig. 6 and combining it with the linear turbine model of expression (8) results in the schematic of Fig. 7.

![Fig. 6 - Hydraulic turbine governor model [6].](image)

The load profile for the Summer Peak case, as stated before, is equal to 75.9 MW and 27.6 Mvar. The units dispatch follows by the power flow simulation lead to the production profile in Table I.

<table>
<thead>
<tr>
<th>Table I. Generation profile for Summer Peak case.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Injection Active Power [MW]</td>
</tr>
<tr>
<td>Graminhais</td>
</tr>
<tr>
<td>Ribeira Grande</td>
</tr>
<tr>
<td>Pico Vermelho</td>
</tr>
<tr>
<td>Waste Incineration</td>
</tr>
<tr>
<td>Furnas</td>
</tr>
<tr>
<td>Caldeirão</td>
</tr>
<tr>
<td>Total</td>
</tr>
</tbody>
</table>

Total losses for this scenario are 1.18 MW, corresponding to 1.54% of total injected active power.

The load profile for the Winter Valley case, as indicated before, is equal to 34.7 MW and 10.9 Mvar. The units dispatch followed by the power flow simulation lead to the production profile in Table II.

Total losses for this scenario are 0.75 MW, corresponding to 2.11% of total injected active power.
In order to optimize the performance of the speed governor of Fig. 8, the parameters dashpot reset time \( T_R \) and the temporary droop \( R_T \), can be varied. The effects of varying these two parameters are studied for units with different values of water time constant \( T_W \) and mechanical starting time \( T_M \). The performance factors analysed, based on the eigenvalue technique, are the time constants corresponding to real eigenvalues and the damping ratios corresponding to complex eigenvalues. The set of parameters chosen to perform the first simulation is expressed in Table III.

### Table III. Base governor parameters.  

<table>
<thead>
<tr>
<th>( T_W ) [s]</th>
<th>( T_M ) [s]</th>
<th>( K_d )</th>
<th>( T_g ) [s]</th>
<th>( R_p )</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.0</td>
<td>6.0</td>
<td>1.0</td>
<td>0.2</td>
<td>0.05</td>
</tr>
</tbody>
</table>

The value of \( K_d \), which as indicated before represents the load damping, is assumed to be 1, to account for the variation of load power with the network’s frequency. The effects that varying the parameters \( T_R \) and \( R_T \) have on the performance factors presented above are shown quantitatively in Table IV.

### Table IV. Effect of governor parameters for a unit with \( T_W = 2.0 \) and \( T_M = 6.0 \).  

<table>
<thead>
<tr>
<th>Governor Parameters</th>
<th>Eigen Values</th>
<th>Damp. Ratio</th>
<th>Time Constants [s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>( R_T )</td>
<td>( T_R ) [s]</td>
<td>( \lambda_1 ) Mode 1</td>
<td>( \lambda_2 ) Mode 2</td>
</tr>
<tr>
<td>0.40</td>
<td>2.0</td>
<td>0.121 + j 0.549</td>
<td>-0.589</td>
</tr>
<tr>
<td>0.60</td>
<td>6.0</td>
<td>-0.049 + j 0.606</td>
<td>-3.921</td>
</tr>
<tr>
<td>0.40</td>
<td>10.0</td>
<td>-0.081 + j 0.638</td>
<td>-5.929</td>
</tr>
<tr>
<td>0.40</td>
<td>3.0</td>
<td>0.043 + j 0.559</td>
<td>-3.902</td>
</tr>
<tr>
<td>0.55</td>
<td>3.0</td>
<td>-0.04 + j 0.646</td>
<td>-5.586</td>
</tr>
<tr>
<td>0.70</td>
<td>3.0</td>
<td>-0.08 + j 0.309</td>
<td>-5.555</td>
</tr>
<tr>
<td>0.60</td>
<td>2.0</td>
<td>0.026 + j 0.450</td>
<td>-3.619</td>
</tr>
<tr>
<td>0.60</td>
<td>6.0</td>
<td>-0.181 + j 0.475</td>
<td>-5.652</td>
</tr>
<tr>
<td>0.60</td>
<td>10.0</td>
<td>-0.217 + j 0.515</td>
<td>-5.658</td>
</tr>
<tr>
<td>0.40</td>
<td>8.0</td>
<td>-0.07 + j 0.625</td>
<td>-3.926</td>
</tr>
<tr>
<td>0.55</td>
<td>8.0</td>
<td>-0.18 + j 0.528</td>
<td>-5.707</td>
</tr>
<tr>
<td>0.70</td>
<td>8.0</td>
<td>-0.24 + j 0.447</td>
<td>-5.573</td>
</tr>
</tbody>
</table>

The process of optimizing governor parameters consists in finding a compromise between a well damped oscillation behaviour and an acceptable speed of response. From Table IV the following conclusions, relating the choice of parameters \( T_R \) and \( R_T \) to their effects on the performance factors, can be taken:

- For the range of temporary droop \( R_T \) that is of interest, this parameter has a very small effect on the time constants of the system. In fact, for a fixed low value of \( T_R \), increasing \( R_T \) speeds up the response of the system (by reducing time constant of mode 3) and for a fixed high value of \( T_R \), increasing \( R_T \) slows down the response, as time constant of mode 3 increases. On the other hand, the impact on the damping ratio is visible, with an increase in \( R_T \) resulting in smaller oscillation behaviour.
- As for the dashpot reset time \( T_R \), both the damping ratio of mode 1 and the time constant of mode 3 are strongly influenced by this parameter’s fluctuation. Increasing this parameter is followed by an increase in both the damping ratio.
of mode 1 and the time constant of mode 3, the latter resulting in slower response speed.

A suitable way of extending the analysis, in order to cover a wider variation of parameters, is to determine the loci of damping ratio and time constants. Essentially a locus represents a set of points whose location satisfies a specific condition. As has been constated both the damping ratio and the time constants are functions of the parameters $R_T$ and $T_R$, hence, it is possible to represent curves along which these performance factors are constant, in a $T_R$-$R_T$ plane. The locus representing the contour lines for the condition of constant damping ratio, for a unit with those parameters of Table III, is shown in Fig. 9.

![Fig. 9 - Constant damping ratio locus for a unit with $T_W = 2.0$ and $T_M = 6.0$.](image)

The contour line representation in Fig. 9 indicates that the system only becomes stable for values of $R_T$ larger than about 0.3 and for values of $T_R$ larger than about 1.5. Although both parameters influence the damping ratio, as seen before, when increasing them above a certain threshold, approximately 8 for $T_R$ and 0.8 for $R_T$, the effect starts becoming negligible. Increasing them would only slow down the rate of response.

The previous analysis is extended in order to cover other combinations of water time constant $T_W$ and mechanical starting time $T_M$, as these are the hydro plant related variables that most influence a unit’s performance. Illustrations of constant damping ratio loci for variations of these parameters are shown in Fig. 10 and Fig. 11.

![Fig. 10 - Constant damping ratio locus for a unit with $T_W = 1.0$ and $T_M = 6.0$.](image)

Fig. 11 - Constant damping ratio locus for a unit with $T_W = 1.0$ and $T_M = 4.0$.

The locus in Fig. 11 differs from that of Fig. 10 by a reduction in the mechanical starting time $T_M$, which is accompanied by a contraction in the stability margin of the system, that is, the contour lines exhibit a slight shift in the direction of increasing $R_T$. The impact this shift has on the study of the optimum performance of the system is the increase of the parameters $R_T$ and $T_R$ in order to maintain an adequate value of damping ratio.

The linear model analysis presented so far suggests that relationships exist between optimum choices of $T_R$ and $R_T$ with unit parameters machine starting time $T_M$ and water time constant $T_w$, as seen in the previous loci of constant damping ratio. Although explicit formulas could be obtained for these functions, a rather different approach is taken in this work. Optimum choice of parameters is presented in the next section for the hydroelectric nonlinear model derived in section II, which corresponds to the HYGOV model in PSS/E, by performing governor simulations and improving the frequency stability following a trial and error method. Nevertheless, two formulas pertaining to the linear model are presented here, which were developed by P. Kundur in his work [7]:

$$R_T = (5 - (T_W - 1) \times 0.5) \times T_W$$  \hspace{1cm} (9)

$$T_R = (2.3 - (T_W - 1) \times 0.15) \times \frac{T_M}{T_W}$$  \hspace{1cm} (10)

Expressions (9) and (10) show that the dashpot reset time $T_R$ depends solely on $T_W$, while the temporary droop $R_T$ depends on both $T_W$ and $T_M$. In the next section, it is acknowledged that $T_R$ should also be adjusted when $T_M$ fluctuates and the results given from these equations are confronted with the results obtained by the trial and error optimization.

### B. HYGOV Model Analysis

The HYGOV model is pictured in Fig. 12. The parameters that remain unchanged throughout the simulations are given in Table V. The fact that a Pelton turbine is assumed in the simulation is accounted for in the value of $D_{turb}$. A Francis turbine, on the other hand, usually has $D_{turb} = 0.5$. VELM represents the absolute maximum gate movement velocity.
Governor simulations are performed assuming there is a load step change in the unit and observing how the speed governor controls the frequency and the mechanical power of the unit. The load step change is an increase of 10% of the unit’s power capacity. It is also assumed that the unit is initially generating 70% of its nominal power, which is how these tests should usually be performed [9]. Fig. 13 and Fig. 14 depict the speed of the unit and its mechanical power responses, respectively, when varying the governor parameters $T_R$ and $R_T$, for a unit with $T_M = 2 \times 1,435$ and $T_W = 0.89$.

The results in the figures above expose the general idea behind trial and error optimization. It was performed through observation of the speed oscillations of the unit, upon experiments with different combinations of parameters $R_T$ and $T_R$. Close observation of Fig. 13 reveals that the light blue curve with $R_T = 0.85$ and $T_R = 1.1$, is the optimum choice. All the other curves, except for the red coloured one with $R_T = 0.60$ and $T_R = 1.2$, exhibit a slightly lower rate of response. Although the curve with $R_T = 0.60$ and $T_R = 1.2$ is faster, a small overshoot appears between $t = 6$ and $t = 8$ seconds, hence, it is not considered an optimum response.

Simulations of the same nature as the above are extended to cover units with different values of $T_M$ and $T_W$. A summary of the results is presented in Table VI, where the optimum choices obtained by simulating the nonlinear model HYGOV are confronted with the optimum choices calculated by using P. Kundur’s equations (9) and (10).

<table>
<thead>
<tr>
<th>Table VI. Optimum parameters choices for several units.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Unit Parameters</td>
</tr>
<tr>
<td>$T_W$ [s]</td>
</tr>
<tr>
<td>0.89</td>
</tr>
<tr>
<td><strong>0.89</strong></td>
</tr>
<tr>
<td>0.89</td>
</tr>
<tr>
<td>0.89</td>
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<tr>
<td>0.89</td>
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<tr>
<td>0.89</td>
</tr>
<tr>
<td>0.89</td>
</tr>
<tr>
<td>1.0</td>
</tr>
<tr>
<td>2.0</td>
</tr>
<tr>
<td>3.0</td>
</tr>
<tr>
<td>4.0</td>
</tr>
</tbody>
</table>

The case in Table VI highlighted in bold matches that of Fig. 13 and Fig. 14, which corresponds to the machine and site parameters to be used in the network simulations of section V. As mentioned before, when performing trial and error optimization on the HYGOV model, not only the parameter $R_T$ needs to be adjusted but also $T_R$. This is immediately observed, as an increase in both $T_W$ and $T_M$ is accompanied by an increase in the dashpot reset time constant $T_R$. The temporary droop $R_T$ follows the same pattern in both optimization approaches, an increase of $T_M$ causes a decrease of $R_T$ while an increase of $T_W$ is followed by an increase in $R_T$. This proportional relationship between $T_M$ and $R_T$ is a direct consequence of the effect that opening the gate of the turbine has on delaying the flow of water through it. Augmenting $T_W$ has the impact of making this effect more pronounced, contributing to stronger oscillations of the mechanical power and therefore the speed of the unit. Speed responses comparing the influence of optimum parameters by both methods are shown in Fig. 15, for the highlighted case in Table VI and for the last case of the same table.
In both cases, the dashed curves represent the responses when parameters obtained by Kundur formulas are used, while the solid line curves depict the responses obtained by using the parameters from trial and error simulation. For the unit with $T_W = 0.89$ and $T_M = 2.87$, while the optimum temporary droop $R_T$ from both methods is similar, $T_R$ obtained by trial and error simulations is much lower than Kundur’s, which causes the response to reach steady state faster. Identical behaviour occurs for the case with $T_W = 4.0$ and $T_M = 2.87$, although the difference in $T_R$ is much larger.

V. NETWORK DYNAMIC SIMULATIONS

The dynamic simulations of the grid are performed using the software PSS/E. Given the São Miguel power system a series of contingencies are simulated independently, which are chosen in accordance with incidents information provided by EDA.

A. Foros 60kV/30kV Transformer Fault

The contingency of the transformer in Foros substation is analysed for the winter valley scenario. The initial conditions pertaining to this scenario can be found in Table II.

The fault is a three phase short circuit happening at $t = 0.1 \, \text{s}$, lasting until $t = 0.2 \, \text{s}$, as it is assumed that the protection takes a total of 100 ms to eliminate the fault, taking the transformer out of service. The result is that the load connected to the 30kV bus prior to the fault is now disconnected, which accounts for a load loss of 2.43 MW and 0.6 Mvar. The chosen bus to observe frequency regulation by the speed governors in the system is the Caldeirão power plant 60kV bus, which is shown in Fig. 16. The response CTCL6 (A) is related to the situation that considers the two hydro speed governors in service, while the response CTCL6 (B) reflects the situation of exclusion of the hydro units speed governors and load shedding active. CTCL6 (C) is the same as CTCL6 (A) but with load shedding inactive and finally CTCL6 (D) is similar to CTCL6 (B) also with load shedding off.

Responses for both cases are very similar, as seen in Fig. 16. The frequency regulation is almost entirely assumed by the Caldeirão unit, DEGOV model, which is a direct consequence of the low strength of the contingency. No load nor motor shedding occur in this situation, as we are dealing with an over frequency response.

B. Pico Vermelho Contingency

The contingency of the Pico Vermelho geothermal power plant is simulated for the winter valley and the summer peak scenarios. In both scenarios the power generated by this utility is the same, 16 MW and no reactive power, meaning the generation power lost due to this contingency is the same for both cases.

1) Summer Peak

Initial conditions for this scenario can be found in Table I. Again, frequency variation is observed on the 60 kV bus of the Caldeirão power plant, which is presented in Fig. 17. CTCL6 (A) refers to the situation that considers the two hydro speed governors in service and load shedding active. CTCL6 (B) reflects the situation of exclusion of the hydro units speed governors and load shedding active. CTCL6 (C) is the same as CTCL6 (A) but with load shedding inactive and finally CTCL6 (D) is similar to CTCL6 (B) also with load shedding off.

Fig. 17 reveals that when load shedding is active, cases A and B exhibit similar behaviour, although the maximum transient frequency deviation is higher when hydro speed governors are not considered. A more pronounced difference exists between responses C and D, when load shedding is off, where it is visible that the inclusion of the hydro speed governors contributes significantly to frequency regulation. Naturally the steady state frequency deviation is larger when not
considering load shedding, as the system speed governors have to control system frequency on their own.

2) Winter Valley

Initial conditions for this scenario can be found in Table II. It should be noted that the contingency for this scenario results in a loss of generated power of almost 50%. Frequency variation is observed on the 60 kV bus of the Caldeirão power plant, which is presented in Fig. 18. CTCL6 (A) refers to the situation that considers the two hydro speed governors in service. CTCL6 (B) also reflects the situation that considers the hydro speed governors in service, but with Kundur optimum \( T_R \) and \( R_T \) parameters. CTCL6 (C) pertains to the situation that considers only the modelling of the thermal units speed governors.

![Fig. 18 - Frequency response on the 60kV bus of the Caldeirão power plant for the Pico Vermelho contingency. CTCL6 (A) – HYGOV considered. CTCL6 (B) – HYGOV considered with Kundur optimum parameters. CTCL6 (C) – HYGOV not considered.](image)

A slight delay of response is observed for case B, with Kundur parameters, compared to case A that uses governor parameters obtained by trial and error optimization. This is expected, as seen before in Fig. 15 of section IV. The response for case C, which excludes hydro speed governors, exhibits a more oscillating behaviour than the cases which consider hydro speed governors and stabilizes in a frequency value above 50 Hz. This is an indication that load shedding is larger for case C.

Simulation of this contingency was also performed when not considering load shedding. Hydro pumps protections, on the other hand, were still considered to be active. The results for frequency variations in the 60 kV bus of the Caldeirão power plant are shown in Fig. 19.

As expected, when load shedding is not considered, the steady state frequency deviations are more pronounced. For all three cases only the shedding of the hydro power plant motors occur. The beneficial contribution of the hydro governing system for frequency regulation (cases A and B), is reflected in a smaller final frequency deviation and less steep maximum transient frequency deviation in the first 5 seconds of the responses. It should also be noted that when the hydro governor models are used with Kundur optimum parameters, case B, the time to reach the steady state is much larger than that of the remaining cases.

![Fig. 19 - Frequency response on the 60kV bus of the Caldeirão power plant for the Pico Vermelho contingency considering load shedding off. CTCL6 (A) – HYGOV considered. CTCL6 (B) – HYGOV considered with Kundur optimum parameters. CTCL6 (C) – HYGOV not considered.](image)

C. Caldeirão Unit 5 Contingency

The contingency of the Caldeirão Unit 5 is analysed for the winter valley scenario. The initial conditions pertaining to this scenario are shown in Table II. The fault is the three phase short circuit happening at \( t = 0.1 \) s, lasting until \( t = 0.2 \) s, taking the unit out of service originating a loss of 9,92 MW of generation power. This unit tripping makes the hydroelectric power plant the only one capable of performing primary frequency regulation.

Frequency variation is observed on the 60 kV bus of the Caldeirão power plant, which is presented in Fig. 20. Three cases are compared. CTCL6 (A) refers to the situation that considers the two hydro speed governors in service. CTCL6 (B) also reflects the situation that considers the hydro speed governors in service, but with Kundur optimum \( T_R \) and \( R_T \) parameters. CTCL6 (C) pertains to the situation that does not consider the inclusion of hydro speed governors, which means frequency regulation is assured by load shedding on its own.

![Fig. 20 - Frequency response on the 60kV bus of the Caldeirão power plant for the Caldeirão Unit 5 contingency. CTCL6 (A) – HYGOV considered. CTCL6 (B) – HYGOV considered with Kundur optimum parameters. CTCL6 (C) – HYGOV not considered.](image)

Fig. 20 reflects that the response obtained by using the trial and error optimum parameters in the hydro speed governors is slightly faster than the remaining two.

D. Caldeirão Unit 6 Transformer Contingency

This contingency is analysed for the summer peak scenario. Initial conditions are shown in Table I. The fault originates the tripping of the transformer and the unit, with 13,8 MW being
lost. Frequency variation is observed on the 60 kV bus of the Caldeirão power plant, which is presented in Fig. 21. Four cases are simulated. CTCL6 (A) refers to the situation that considers the two hydro speed governors in service and load shedding on. CTCL6 (B) also reflects the situation that considers the hydro speed governors in service, but with Kundur optimum \( T_R \) and \( R_T \) parameters, also with load shedding on. CTCL6 (C) and CTCL6 (D) are equivalent to cases A and B, respectively, with load shedding off.

![Fig. 21 - Frequency response on the 60kV bus of the Caldeirão power plant for the Caldeirão Unit 6 transformer contingency. CTCL6 (A) – HYGOV considered with load shedding ON. CTCL6 (B) – HYGOV considered with Kundur optimum parameters and load shedding ON. CTCL6 (C) – HYGOV considered with load shedding OFF. CTCL6 (D) - HYGOV considered with Kundur optimum parameters and load shedding OFF.](image)

Inspection of Fig. 21 shows that for this contingency changing the hydro speed governor parameters has a minor impact in frequency regulation, especially when load shedding is not considered. For all the cases, the frequency steady state value is reached at about \( t = 25 \) s.

**E. Pelton vs Francis Performance**

The purpose of this last sub-section is to get an idea of how the type of hydro turbine affects system frequency regulation when a fault occurs. The Pico Vermelho contingency for the summer peak scenario is chosen to achieve this comparison. Frequency variation is observed on the 60 kV bus of the Caldeirão power plant, presented in Fig. 22. Load shedding is not considered. CTCL6 (A) represents the situation with the hydro plant equipped with Francis turbines, while CTCL6 (B) corresponds to the already simulated case of the Pelton hydro turbines. The optimum governor parameters for the Francis turbines are \( T_R = 2.1 \) and \( R_T = 0.8 \).

VI. CONCLUSION

In short, the objectives proposed in section I were well accomplished. Although exact formulas relating the optimum hydraulic governor parameters \( T_R \) and \( R_T \) with the unit parameters \( T_W \) and \( T_M \) could not be determined, relationships, for their ranges of interest, were still found. The optimum parameters obtained by trial and error optimization of the HYGOV model were confronted with the optimum parameters suggested by Kundur in his paper. As Kundur optimum parameters are obtained by studying the linear model, their applicability when performing nonlinear simulations is limited and therefore, were found to be less accurate than the parameters obtained by trial and error optimization. This was then confirmed in section V for the contingencies of Pico Vermelho, in the winter valley scenario, and Caldeirão Unit 5, also in the winter valley scenario, where using Kundur optimum parameters slightly worsens the quality of the frequency response. Generally speaking, the results from section V reflect that the inclusion of the hydroelectric speed governors in situations when there is only one or no thermal speed governor in service (winter valley cases) is substantially more beneficial than in situations with more power generating capacity (summer peak cases) and thermal speed governors in service. This is also recognized when load shedding is off, no matter the amount of thermal generation. Overall, consideration of the hydroelectric speed governors led to slightly more damped first frequency oscillations and to higher rates of frequency stabilization. This is translated as a positive and desired impact of the hydroelectric speed governing devices in primary frequency control.

REFERENCES