SSR Studies in Argentina for the Bahía Blanca Generating Plant

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Abstract — **This paper describes studies made for the characterization of SSR [1] problems in the South Region of the Argentine Interconnected System, where Bahía Blanca Thermal Power Plant (CTEBB, 2 x 310 MW) is connected in the mid point of one of three 500 kV AC series compensated transmission lines. The description contains the field tests made to obtain the spring-mass model of the turbine generator shafts and the studies for the induction generator effect, torsional interaction and accumulative fatigue life expenditure calculations of turbine generator shafts, using the Alternative Transients Program (ATP) and auxiliary programs (specially in the subject of torsional fatigue loss of life calculations). An additional contribution of this work is the way in which the lack of manufacturing data for the T-G units was overcome.**

Keywords: SSR, Series Compensation, Induction Generator Effect, Torsional Interaction, Fatigue Life Expenditure, Torsional Model Identification for T-G Shafts, Field Tests, ATP.

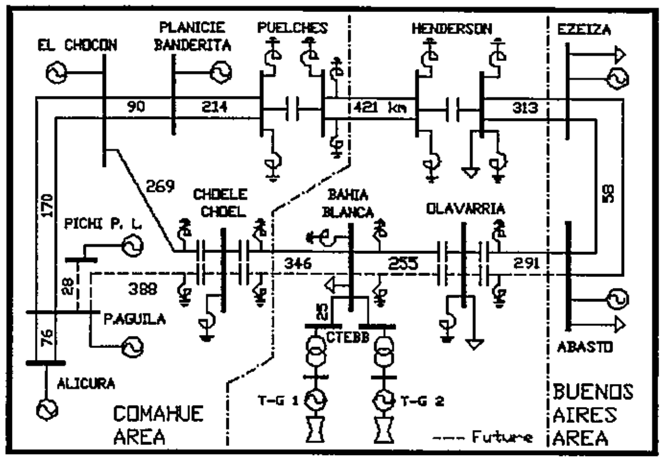

Fig. 1. Transener S.A. 500 kV AC Southern Transmission System.

I. INTRODUCTION

Fig. 1 shows a schematic representation of the southern 500 kV transmission system owned by the transmission company Transener S.A. and the most important connected generating plants. Three lines 40% series compensated link over 1100 km the south-west power exporting Comahue region, where important hydroelectric generating stations are located, to the distant large consumption centers of the country located around Buenos Aires. In 1999 a fourth line 40% series compensated from the Piedra del Aguila hydroelectric plant to Abasto will be incorporated.

Since series capacitor banks in Choele Choel and Olavarría were installed, CTEBB have been exposed to the risk of SSR phenomena. Initial study results based on precarious mechanical data showed serious problems of torsional interaction. During the commissioning of T-G 1, tests to obtain a reliable shaft spring-mass model for studies were carried out and new studies were accomplished.

This article describes the tests carried out on the T-G 1, the methods used for diagnosing the SSR problems, the study results and the subsequent recommendations.

II. METHODOLOGY

A. *Induction Generator Effect (IGE)*

The study was done to determine if the subsynchronous currents would be reinforced by the selfexcitation phenomenon, which would take place if the two following conditions are present simultaneously:

1.- The series circuit which is constituted by the generator and the external network is resonant at a subsynchronous frequency, f_e.

2.- The resistance of this series circuit computed at the same subsynchronous frequency f_e is negative.

The method used in this case, based on the Frequency Scanning Technique [2, 3], was carried out using the ATP [4] and an auxiliary program (for more details, see II.B). Circuit series reactance and resistance were calculated and plotted, for frequencies in the 5-50 Hz range stepped 0.1 Hz (Fig. 2 depicts an example of their frequency dependence). For the frequencies which made the reactance null, the sign of the respective resistance was observed in order to examine the existence of selfexcited oscillations.

B. *Torsional Interaction (TI)*

The analysis was carried out in order to detect the occurrence and maintenance of the TI effect in a generating unit, which occurs when the two following conditions coexist:
1.- The electric natural frequency of the subsynchronous current is complementary of one of the T-G shaft natural torsional frequencies.
2.- The total damping of the oscillatory excited mode is negative or zero. This total damping is the mechanical modal damping (always positive) plus the apparent damping provided by the electric system (Appendix 1).

By using frequency scanning technique, it was observed if the frequency corresponding to null reactance (resonance condition) was equal or very close to the complementary frequency of any natural mode of shaft torsional oscillation. If it was, the sign of the total damping was examined.

Fig. 2 shows the resistance, the reactance and the apparent damping vs. electrical frequency for the TI worst case.

In the analysis, the transmission 500 kV network was represented by means of distributed parameters, the loads were assumed to be constant impedances, the other generating plants were represented by means of their Thevenin equivalent circuits with their direct axis subtransient reactances, the transformers of the CTEBB were represented by their resistances and reactances (calculated using the scanning frequency), and the CTEBB generating units were represented by their induction generator equivalent circuits, with frequency dependent parameters (Appendix 1, (6)).

Using the ATP "frequency scan" option, network equivalent impedances were obtained for each frequency, as measured from the 500 kV terminals of the CTEBB transformers.

The auxiliary program calculates the frequency dependent impedance for the CTEBB generators and transformers, adds it to the network impedance obtained with ATP and calculates the total frequency dependent damping for each mode of interest.

For these studies the modal model of the shaft mechanical system was obtained as is described in III.C.

C. *Expected fatigue life expenditure*

To predict total fatigue life expenditure (FLE) for T-G shafts it is necessary to forecast the stresses that they would be exposed to along all their life expectancy. Therefore it is required to assume a hypothesis about the incidents that could affect the units, simulating each of them. Finally the damages are accumulated in order to do a diagnosis.

Transient Torques

Time simulation was fulfilled by ATP program. The shaft stresses were obtained using the "source 59" model (Appendix 2), which allows an adequate representation for shaft torsional oscillations (see mechanical model also in III). Transient torsional torques were determinated for all shaft

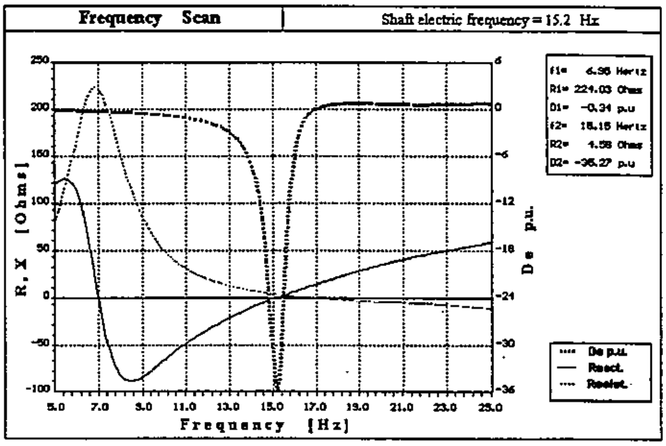

Fig. 2. Frequency Scan for the CTEBB units.

Fatigue life expenditure calculations for each incident

sections, for assumed disturbances, in several configurations corresponding to possible network expansion, with one or two T-G being in service in the CTEBB, for several types, places and times of application and clearing of faults, successful or not successful breaker reclosures, etc. The event predictions were based on the fault statistics available for the system.

The calculation of FLE for each incident was fulfilled following the methodology indicated in [5], which was specially useful because it involves very few shaft steel data.

The method takes data of fatigue strength of the steel given as stress-life (*S-N* diagram). These are obtained from destructive tests in which the material is alternatively subjected to equal tension and compression stresses. s (or σ) is the pure alternative stress amplitude and N is the number of cycles that the material can stand without failure.

This curve can be estimated from static tensile properties for small, polished and unnotched specimens, as it is shown in Fig. 3, curve A. It is established by determining the half-cycle and 10^6 cycles amplitudes (points A1 and A2, respectively) using only the ultimate tensile strength for the steel, S_u.

It can be noticed that the half cycle amplitude is related to the true fracture strength for a tension test, increasing the stress up to the fracture. The second point is the endurance limit, σ_e (from a practical point of view, amplitudes less than this value are considered to cause no fatigue damage).

To convert an uniaxial stress-life curve to a surface shear stress-life curve for torsion, the axial stress at every value of N is divided by $\sqrt{3}$ (curve B in Fig. 3), taking into account the von Mises theory.

$$\tau = \sigma / \sqrt{3} \tag{1}$$

The surface shear stress τ is calculated from the torque T obtained for each part of the shaft with the ATP, using the following relationship valid for a transversal uniform section of a hollow shaft:

$$\tau = \frac{16}{\pi} \frac{D_{ext}}{D_{ext}^4 - D_{int}^4} T \quad (2)$$

D_{ext} y D_{int} are, respectively, the external and internal diameter of the most stressed cross-section for each part of the shaft.

Palmgren-Miner's Law establishes that the rate of FLE caused by n_i cycles of a shear stress τ_i, whose fatigue strength is N_i cycles, is the quotient between n_i and N_i. Consequently, a fatigue failure will occur on the material when the following relationship is verified:

$$\sum_i (n_i / N_i) = 1 \quad (100\,\%) \quad (3)$$

The *S-N* curves correspond to pure alternative stresses applied to the material (mean stress zero). To use them in cases of complex load histories requires at first to determine a fictitious load history composed by pure alternative stresses that would produce the same damage. For this, the real history must be broken down into a number of separate events or cycles. To perform this task of "cycle counting" the Rain-Flow technique may be used. In this way the real stresses history is replaced by a number of cycles, each one characterized by its mean value τ_m and its alternative amplitude τ_a. Fig. 4 shows a short stress-time history and the cycles wich would be counted by the Rainflow method.

For each cycle "c" the *S-N* curve is modified using the mean value, as it is shown in curve C of Fig. 3. In this way it is possible to determine the number of cycles to fatigue failure entrying to the curve only with τ_a. To calculate the point C2 the Goodman's Law can be used (the formula for C1 is evident).

Additional effects, or fatigue strength reduction factors, must be considered: stress concentration due to notches, size effect between laboratoty specimen and large components, and surface finish. It can be seen in curve D of Fig. 3. At $N=10^6$, using an effective factor *K*, the stress concentration effect due to notches and other fatigue strength reduction factors are taken into account to modify the *S-N* curve. At $N=10^3$ a breakpoint was introduced. Thus, the curve D is a better approximation for the plastic or elastic behavior of the material, depending on the number of cycles. (It is considered mainly plastic before $N=10^3$, low cycle region, where corrections aren't applied, and elastic over $N=10^3$, high cycle region. The reduction is fully applied at $N=10^6$).

Then, the corresponding life value N_c, considering all fatigue strength reduction factors, can be obtained entrying the curve D with the pure alternative amplitude τ_a.

Finally, the loss of life produced by all the cycles is:

$$FLE\ per\ incident = \sum_c (1 / N_c) \quad (4)$$

The program PERVIDSN follows this methodology and was developed in FORTRAN language. This program uses as input data the torques appearing in the outputs of the ATP program and the mechanical data of the shaft. The output gives the loss of life percent for each section.

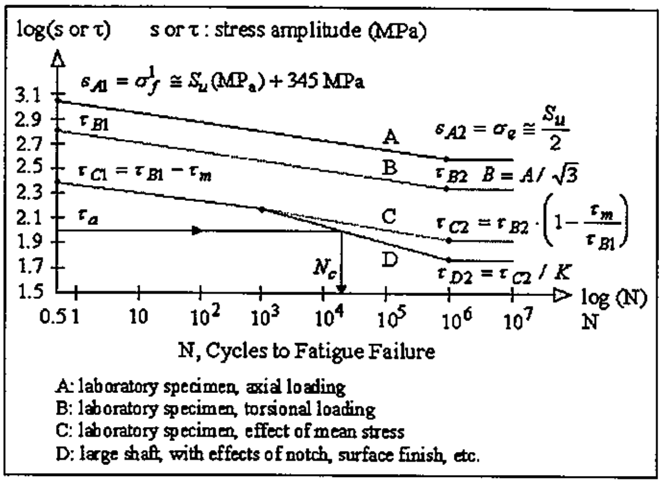

Fig. 3. Stress-Life Curves

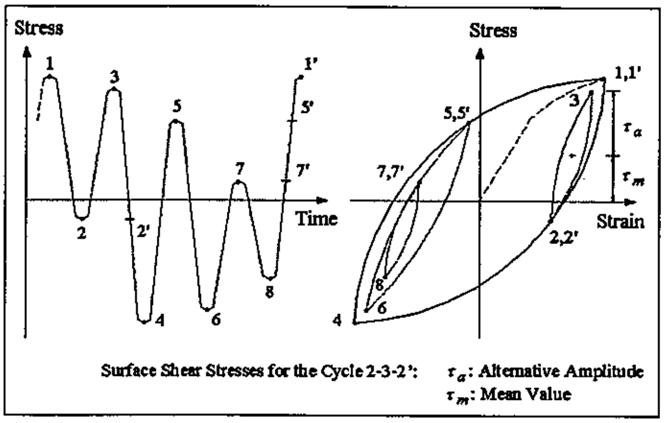

Fig. 4. Fatigue Cycle Counting by the Rainflow Method

III. SHAFT MODEL

A. Antecedents

The T-G shaft train consists of five great masses: steam turbine stages of high (HP), intermediate (IP), and low pressure (LP), generator (GEN) and main rotating exciter (EXC).

For the development of a low order shaft discrete spring-mass model, suitable for torsional oscillations studies, the following information supplied by the manufacturer was available: inertia coefficients for the five masses, frequencies of the first four oscillation modes and shaft profile drawings. Mechanical damping data were not available.

From precarious shaft profile data and mass inertias, studies performed by a consultant led to modal frequencies different from those supplied by the manufacturer. This is shown in Table 1. Because of these differences, and in order to develop a more reliable shaft mathematical model, it was considered necessary to do field tests. The results are shown in the same table.

B. Tests description

Field tests were performed by ABB Power Systems on the T-G 1 in Nov. 1989. The target was to identify the shaft torsional mode frequencies and the shaft damping modal coefficients. In the Fig. 5 the five points where the instantaneous speed measures were taken are shown. It was physically impossible to access to the sections IP-LP and LP-GEN of the shaft. Self-adhesive pulse bands were used. At each measuring point, two diametrically opposite transducers were installed, thus enabling compensation for radial movements of the shaft. A reference pulse system for the elimination of errors due to pulse band pitch errors was implemented.

As low load tests (unit synchronization and other switching) were planned for the comissioning of the T-G 1, they were used to excite the shaft torsional oscillations. In addition, Bahía Blanca 150 MVAr bus reactor switching (Fig. 1) was used for 45, 70, 85 and 100% generator rate power load levels.

The frequency identification was made in situ during the measurements, using a FFT analyzer.

The modal damping evaluation was made by two different methods. The first one is presented in [6]. The second method consists of extracting the modal components by digital filtering. The time constant σ^{-1} is calculated from the filtering resulting waves by measuring the time interval from the initial peak value up to the moment in which the value of the enveloping curve falls to 37% of it.

The frequencies given in Table 1 (first three oscillation modes) were obtained with a precision of 0.1 Hz. The precision was limited to 0.3 Hz at mode 4 due to the low level signals obtained for this mode.

For total damping, very different values were obtained, even for the same T-G load level. Table 2 shows extreme values for synchronization and full load, using both methods.

Torsional responses obtained at the described tests were, in general, of little amplitude and in some cases the signal/noise relationship was not enough to make them useful. Nevertheless, minimal values have been identified with the exception of mode 4.

C. Shaft model preparation

Fig 5 shows a spring-mass model for the shaft obtained with an auxiliary program which calculates the natural torsional frequencies (eigenvalues) and the corresponding mode shapes (eigenvectors) from characteristic matrix of free oscillations. Spring coefficients were adjusted to make the resonance frequencies match the ones determined at the field tests, assuming the inertia coefficients supplied by the manufacturer as being exact.

The same program calculates the following modal coefficients for each mode: inertia (H_m), stiffness (K_m) and damping (D_m), based on the following expressions [2]:

$$Hm = \sum_{j} Hj\,\theta j^2 \;;\; Km = 2\,Hm\,fm \;;\; Dm = 4\,Hm\,\sigma m \qquad (5)$$

where H_j is the inertia coefficient for the j mass and θ_j is the amplitude of the mode shapes of interest for each j mass. In Table 3 the values for the first three modes are shown.

D_m coefficients are applied directly to the torsional interaction analysis, as indicated in Appendix 1.

For time simulations, the damping coefficients associated with the real masses of the shaft were determined applying the inverse modal transformation to the matrix of modal damping (the modal damping matrix is assumed to be diagonal, using coefficients from Table 3).

Table 1. Shaft torsional frequencies, according to manufacturer, theoretical calculations and field tests.

Data Source	Mode Frequencies (Hz)			
	f_{m1}	f_{m2}	f_{m3}	f_{m4}
Manufacturer	13.6	18.3	31.5	61.9
Initial Studies	16.5	24.1	30.1	70.9
Field Tests	14.0	20.8	34.8	59.8

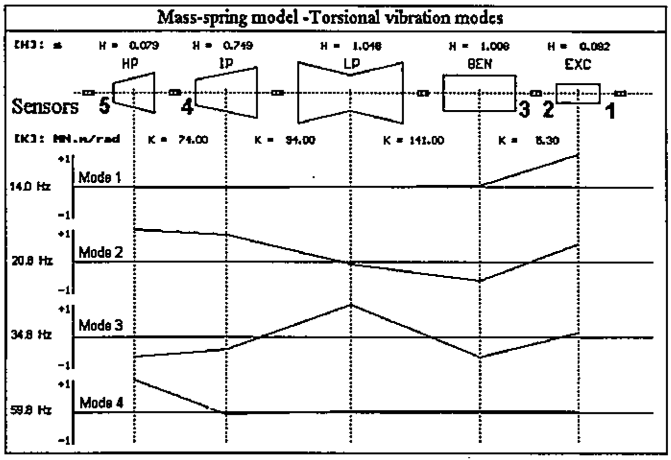

Fig. 5. Shaft Spring-Mass Model - Mode Shapes.

Table 2. Identified mechanical modal damping ranges using two different methods for extreme loading conditions.

Mode	σ_m (rad/s)	
N°	Synchronization	Full Load
1	0.68 - 0.73	0.72 - 0.75
2	0.66 - 0.74	0.70 - 0.95
3	0.77 - 0.85	0.80 - 1.10

Table 3. Modal equivalent parameters.

Mode	σ_m	H_m	D_m
N°	(rad/s)	(s)	(pu)
1	0.68	5,828.25	15,852.83
2	0.66	2.58	6.82
3	0.77	3.72	11.46

A model based only on mutual damping coefficients was adopted (see Appendix 2). Mutual terms are identified from the damping matrix in real coordinates. Neglecting the dashpot damping terms, the modal transformation is then applied to examine if the corresponding modal dampings are equal or a little lower than those measured. For this case, it was necessary to adjust repeatedly the mutual coefficients to satisfy this condition. As can be observed in Fig. 5, mode 1 is only conditioned by the GEN-EXC part and mode 4 by the HP-IP part. This fact makes the task easier.

D. *Comparison between the models obtained from field tests and theoretical calculations.*

The impact of the tests on the SSR studies was very important, because they implied substantial changes in the T-G shaft model. Table 1 shows that the values of the torsional resonance frequencies identified during the tests do not coincide neither with those supplied by the manufacturer nor with those obtained by calculation. A maximum difference of 3.3 Hz was reached between what was measured and what was indicated by the manufacturer in mode 3, which was the one most critical. The frequency values further far from reality are those obtained by calculation.

Results were much more surprising as concerning mechanical dampings of the shaft. At Table 2 non loading damping coefficients are 0.6 rad/s over, which is around 6 times greater than the upper limit of the range of typical values indicated in [7].

IV. STUDY RESULTS

A. *Induction generator effect and torsional interaction*

As some of the parameter values were not reliable (including the high damping measured and the lack of data about the local load behavior), these results are not definitive. They were considered just to provide a first evaluation of the subsynchronous resonance problem in order to specify the actual phenomenon severity and a tentative SSR countermeasure.

It should be noted that the studies covered a wide range of scenarios, such as different generation conditions of the CTEBB, with one or two units on line, different 500 kV system configurations and load levels including the future fourth line, several degrees of series capacitive compensation, different contingencies due to 500 kV lines and/or capacitor banks outages, variations of the model parameters of the induction generator equivalent circuit. The total number of cases studied was 104.

Manifestation of the induction generator effect was not detected, even when wide ranges of parameter variations of the CTEBB generating units were considered.

The risk of TI was only detected for the third oscillation mode. In ten of the 104 cases analyzed the total damping was negative. In all the cases the two CTEBB units had to be on service, and the capacitive compensation reduced to a 20% (with only one of the two modules in Choele-Choel and only one of the two modules in Olavarría and minimal load condition). In two cases the total damping appears to be reduced to zero threshold, for 30% compensation (outage of a module in Choele Choel simultaneously with maximal load).

In addition to this, the case in which the modal damping for mode 3 was 50% lower was also considered. This situation would impose a TI problem to the future system, when the forth line were in service.

About the influence of local load behavior, [7] may be consulted.

B. *Fatigue life expenditure*

Ninety cases of incidents, concerning fault types and their location, number of units in service in the CTEBB and the state of development of the transmission system, were characterized. All of the cases were simulated in order to determine each critical point torsional response at the four sections of the shaft, producing 360 outputs.

Because of the loss of life calculations, the LP-GEN section (between the LP stage rotor and the generator rotor) was identified as the most stressed one by torsional fatigue.

The total FLE evaluated was 6.3%, having been considered the number of expected faults, the different transmission system stages, and assuming a life expectancy of 25 years for the T-Gs.

There were also considered cases in which the damping coefficients in the source 59 model were 50% reduced. The subsequent total FLE was approximately 80% greater.

V. CONCLUSIONS AND RECOMMENDATIONS

Unlike earlier studies, the results described here showed that SSR phenomena in the CTEBB T-G units is not too severe. Also the calculated FLE resulted considerably small, within the life expectancy of the units. However, uncertainties of the values of some parameters, induced the authors to recommend to install a protection relay and a monitoring system, to carry out more complete electrical and mechanical tests with higher exciting disturbances levels, to perform more rigorous studies on TI and FLE, in order to adjust the protections properly and to determine if additional countermeasures are necessary, and to record the FLE in each of the disturbances that could affect the integrity of the T-G shafts.

VI. REFERENCES

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APPENDIX 1

The induction generator equivalent circuit was represented by an emf in series with the following frequency dependent impedance:

$$Z_g\left(f_e\right)=\frac{R_r}{s}+j\,X_r\left(\frac{f_e}{50}\right) \tag{6}$$

where:

f_e (Hz) Subsynchronous electric scanning frequency

R_r (Ohm) Generator effective resistance

s (pu) Slip ($= \frac{f_e-50}{f_e}$)

X_r (Ohm) Generator effective reactance

The value of R_r wasn't available. In order to overcome this difficulty, typical values (0.02-0.03 pu) and empirical formulas from [8] were used:

$$R_r = 2.0\cdot\left(r_2 - r_1\right)\ ;\ f_e < 20\text{ Hz} \tag{7}$$

$$R_r = 1.7\cdot\left(r_2 - r_1\right)\ ;\ 20 \le f_e \le 40\text{ Hz} \tag{8}$$

where r_2 and r_1 are the generator negative and positive sequence resistances, respectively.

So, a R_r range between 0.02 and 0.057 pu was considered in the studies.

The value of X_r was approximated by means of the generator negative sequence reactance delivered by the manufacturer, with a ± 30% tolerance. This tolerance was determinant for the range considered.

Several combinations of different values for these two parameters were used, in order to find critical cases.

For each oscillation mode, the apparent damping was calculated by [2] :

$$D_e\left(f_e\right)=-\frac{f_e}{2f_m}\ \frac{R}{R^2+X^2} \tag{9}$$

where:

D_e (pu) Apparent damping

R (pu), X (pu) Net system resistance and reactance as viewed from the generator neutral

f_m (Hz) Shaft natural torsional frequency

As the two T-G units are identical they were modeled as a single unit. The effective mechanical modal damping σ_{mef} was calculated as follows [7]:

$$\frac{1}{\sigma_{mef}}=\frac{1}{2}\left[\frac{1}{\sigma_1}+\frac{1}{\sigma_2}\right] \tag{10}$$

The total damping for the same oscillation mode is the mechanical modal damping plus the apparent damping provided by the electric system:

$$D_t = D_m + D_e \tag{11}$$

APPENDIX 2

CTEBB T-Gs electromechanical model: data for time simulations (source 59 model [4]):

364.7 MVA, 20 kV rms, 50 Hz, 2 poles, 3000 rpm
Field current which will produce the rated armature voltage on the air gap line (direct axis): 1080 A

Manufacturer's data:

R_a	X_L	X_d	X_q	X'_d	X'_q	X''_d
0.0014	0.155	1.76	1.76	0.27	0.351	0.18
X''_q	T'_{d0}	T'_{q0}	T''_{d0}	T''_{q0}	X_0	
0.252	5.9	1.4	0.029	0.04	0.092	

Units: R(pu), X(pu), T(s)

Spring-Mass Model:

Mass	Mech. Torque fraction	Inertia Mkg.m^2	Mutual Damping N.m/(rad/s)	Stiffness MN.m/rad
HP	0.29	5.875E-4	3510.424	74.0
IP	0.45	5.535E-3	3140.906	94.0
LP	0.26	7.743E-3	4212.509	141.0
GEN		7.450E-3	975.528	5.3
EXC		6.800E-4		