Transactions of the Korean Nuclear Society Autumn Meeting
Yeosu, Korea, October 24-26, 2018

Rotordynamic Analysis of the AM600 Turbine-Generator Shaftline

Tshimangadzo Mudau*, Robert M. Field
Department of Nuclear Engineering, KEPCO International Nuclear Graduate School
45014 Haemaji-ro, Seosaeng-myeon, Ulju-gun, Ulsan, 689-882 Republic of Korea
Corresponding author: tmudau92@gmail.com

1. Introduction

This paper investigates the design configuration of the Turbine-Generator (T-G) shaftline for the Advanced Modern 600 (AM600) turbine cycle [1], a Turbine Island design to be coupled with a medium sized nuclear reactor plant in the range of 1800~2000 MWt.

The AM600 is unique to Pressurized Water Reactor (PWR) plants in that it is designed for a medium sized reactor but has a single LP turbine rotor. In addition to the economic advantages of the simplified design and smaller profile for this shaftline, there are additional benefits related to resistance to grid and machine torsional vibration excitation.

The targeted markets for this design are characterized by grid systems which are smaller and less robust than those found in more mature markets. The simplified design here, with a stiffer configuration, is expected to show fewer resonant frequencies in the range of forced excitation. Such vibrations can result in fatigue damage in shaft components, such as the turbine blades, couplings, and retaining rings [2]. Thus this design is expected to be more appropriate for the intended use than more conventional designs.

Here, rotordynamic analysis results are reported for the AM600 T-G shaftline using CATIA V5 for physical modeling and the ANSYS Workbench for dynamic response modeling [3]. The torsional natural frequencies and separation of these frequencies will be evaluated with respect to the dominant excitation frequencies for the targeted markets (50 Hz for general grid disturbances and 100 Hz for negative sequence currents).

The modeling here includes a detailed configuration of the turbine and generator rotors but does not include modeling of the blading. Results are considered to indicative of the potential for the AM600 design but not definitive. That will require follow-on analysis which includes modeling and responses for the blading.

ANSYS results were verified by use of simplified theoretical calculations by using mass-spring approach.

2. Method

2.1 Rotordynamics Modelling in ANSYS

Two concepts that were developed during conceptual design of the T/G were considered in the analysis.

The first concept (Fig. 1a) of the AM600 turbine generator system consists of one LPT cylinder with 1.6 m long last stage blades for the 50 Hz market. (Note that these blades are not modeled in the current analysis).

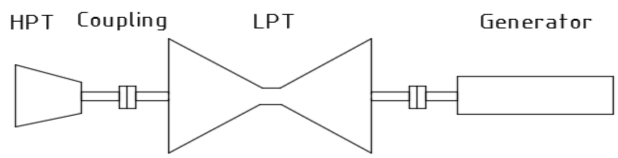

Fig. 1a: T/G with One (1) LPT Cylinder

The second concept (Fig.1b) with two (2) LPT cylinders is for countries with low sink temperatures and thus high volumetric exhaust flow.

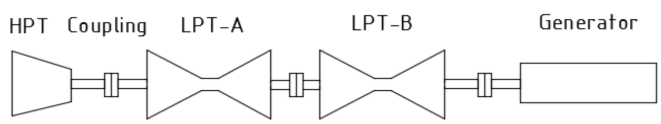

Fig. 1b: T/G with Two (2) LPT Cylinders

The rotordynamic behaviour assessment begins with the set-up of an analytical model for the coupled shaftline. The model in Fig. 2 below was developed in CATIA V5. This design models the single cylinder configuration. Due to the large span of this rotor, the modeling considers a welded drum type Low Pressure Turbine (LPT) rotor. This type of construction has lower mass than alternatives (e.g., monoblock or ruggedized disk rotors).

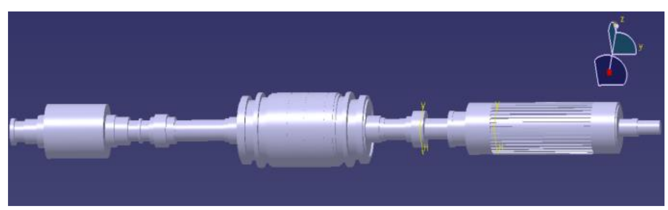

Fig. 2: AM600 T/G Shaftline

2.2 Shaft Modelling

To prepare an accurate shaftline model, detailed information is required. In the simulation, the shaftline is broken into smaller pieces using a process called slicing. Each piece will have a cross-section divided according to stiffness and mass attributed to the associated material properties as defined in the ANSYS Workbench.

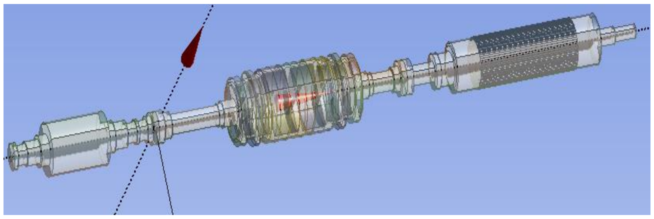

Fig. 3: Sliced Shaftline

Table 1 provides the shaftline material properties for forged 12% chrome steel.

Table 1: Rotor Material Properties

Property	Value	Units
Density	7750	kg/m^3
Modulus of Elasticity	185	GPa
Poison's Ratio	0.3	-
Max. Allowable Stress	130	MPa
Yield Strength	178	MPa

2.3 Modal Analysis of the 3D Shaftline

The 3D shaftline torsional analysis was conducted in ANSYS 19. Fine quadratic meshing was employed which generated 131,016 elements and 230,378 nodes. Boundary conditions were defined by constraining the sliced shaft in its center line to ensure that only torsional modes were calculated.

3. Results

3.1 Finite Element Method (FEM) Results in ANSYS

The preceding section outlines procedures that were followed for the modal analysis of the shaftline with single cylinder LPT welded drum rotor. A second case, a shaftline with two (2) LPT cylinders was also considered. For comparison, each of these cases was analyzed for both welded drum and the monoblock configurations. Results for the first seven torsional modes are summarized in Table 2 below.

Table 2: Torsional Natural Frequencies of the Shaftline

	Welded Drum Type LPT Frequency (Hz)		Monoblock Type LPT Frequency (Hz)	
Mode	1 LP Cylinder	2 LP Cylinders	1 LP Cylinder	2 LP Cylinders
1	0	0	0	0
2	19.1	14.4	18.6	14.0
3	26.1	24.9	24.2	23.9
4	156.6	30.0	67.8	27.2
5	192.2	188.1	181.0	78.5
6	198.4	189.0	191.7	81.9
7	219.9	194.8	208.6	196.4

3.2 Confirmation Using Simplified Model

The lumped mass-spring method [4] was used with the shaftline properties given in Table 1 to perform a 'hand calculation' confirmatory analysis. The HPT, LPT, and generator rotor were modelled as disks connected by a spring system. Fig. 3 illustrates the simplified model of the single cylinder monoblock rotor as vibrating system with three degrees of freedom.

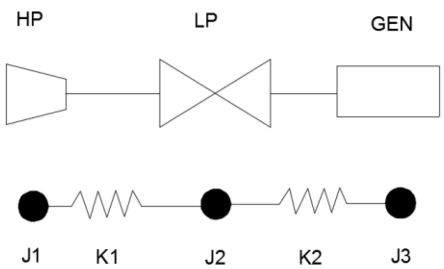

Fig. 4: T/G Mass-Spring Shaftline Model

For this mass-spring system, the general equation of motion is given as follows [5]:

$$[M]\{\ddot{x}\} + [C])\{\dot{x}\}) + [K])\{x\} = \{F\ (t)\}$$

Where M, C, K represent mass, damping, and stiffness matrices, F constitutes external force, x and its derivation present displacement, velocity, and acceleration.

The eigenvalue method was then used with the properties given in Table 3 on the simplified linear equation of motion for free and undamped vibration which is given as follows:

$$(-\omega_{nf}^2[I] + [K])\{\emptyset\}) = \{0\}$$

Where:

ω_{nf} = the torsional natural frequency (rad/s),

I = the moment of inertia matrix ($kg.m^2$),

K = stiffness matrix (Nm/rad), and

$\emptyset$ = the mode shape (-)

Table 3: Shaftline Mass- Spring Modelling Properties

Mass	Inertia moment ($kg.m^2$)	Span (-)	Torsional Stiffness (Nm/rad)
1	59,042	1-2	7.29E+8
2	140,625	2-3	1.43E+9
3	74,864		

The eigenvalues were then calculated for the three degrees of freedom system which generated the matrix size up to 3X3 and thus resulted in first three modes torsional natural frequencies as shown and compared with ANSYS results in Table 4.

Table 4: Comparison of Torsional Eigenvalues

Mode	Shaftline with Single LTP Cylinder Monoblock Rotor		
	ANSYS Results (Hz)	Calculated Results (Hz)	Δ (%)
1	0	0	0
2	18.6	18.2	2.1
3	24.2	23.9	1.1

4. Discussion

The results of the torsional vibration analysis provide indications which are promising for the intended purpose of the AM600 T-G shaftline. First, the natural torsional frequencies are few and widely separated. Second, the mode shapes show how the shaftline will respond (deform) at those particular frequencies [6].

For the three modes that were calculated for the validation of the FEM in ANSYS software, the comparison between the ANSYS and the calculated torsional natural frequencies results shows good results, with agreement to within 3%. This lends confidence to the modeling effort.

The Mode 1 frequency was 0 Hz for all cases, which indicates that the shaftline was rigid and freely rotating. For all seven modes that were extracted, the torsional frequencies increased from 0 to 220 Hz. The torsional frequencies of the T/G shaftline with a single LP cylinder rotor were higher with greater separation than those for with double cylinder LP rotors. This provides confirmation that detailed design and tuning of the single cylinder LPT design can result in a machine that provides more robust resistance to torsional excitation. Additionally, the monoblock type rotors were found to have lower frequencies than those of welded drum type rotors.

These results agree with theory which indicates that the torsional frequencies increase with a decrease in the mass of the system.

For the purpose of analysis and evaluating the results against the frequency exclusion ranges, this paper focuses on the mode frequencies which are below and above the exclusion ranges. The following frequencies and deformations were observed:

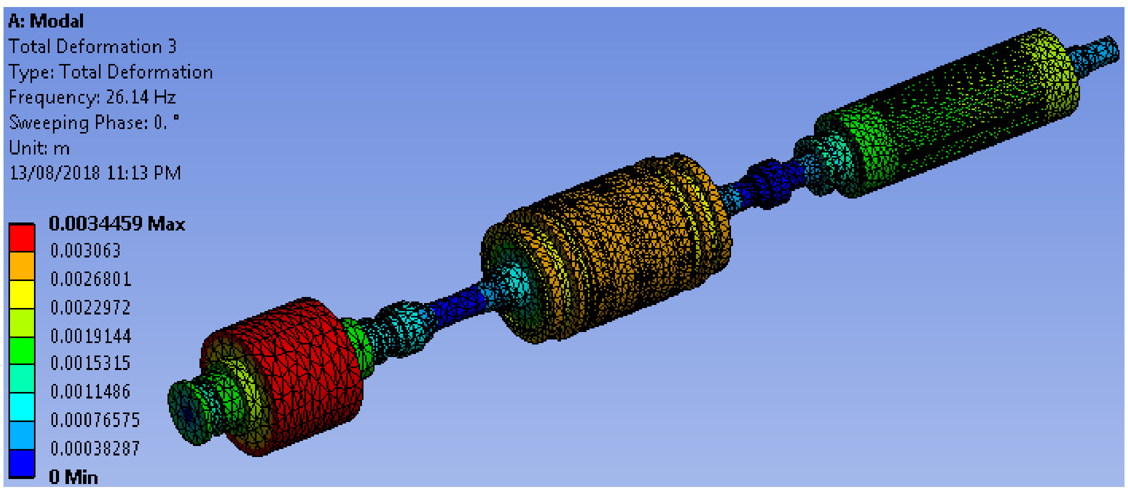

Fig. 5: Concept 1 Mode 3 Torsional Natural Frequency

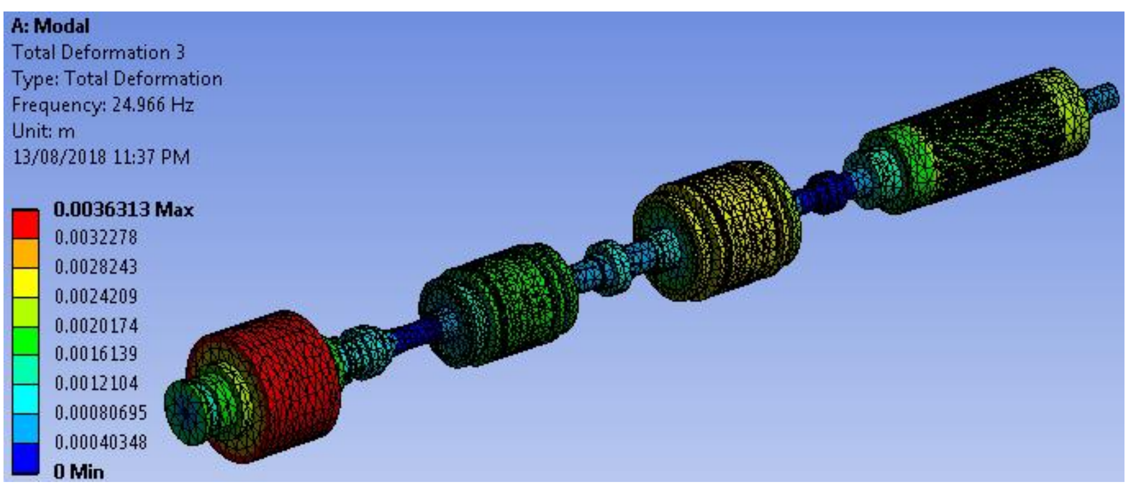

Fig. 6: Concept 2 Mode 3 Torsional Natural Frequency

5. Conclusion and Future Work

5.1 Conclusions

The torsional natural frequencies and the mode shapes of the shaftline were successfully generated using the combination of CATIA and ANSYS modelling. The most critical frequencies for the target markets are those near one and two times the grid frequency (i.e., 50 Hz and 100 Hz).

From the ANSYS analysis results, the frequencies that are closest to 50 Hz are 30 Hz and 67.8 Hz in Mode 4 of the double cylinder welded drum type rotor and Mode 4 of the single cylinder monoblock rotor, respectively. The frequencies that are closest to 100 Hz are 81.9 Hz and 156.6 Hz in Mode 6 of the double cylinder monoblock type rotor and Mode 4 of the single cylinder welded drum rotor, respectively. This shows excellent separation for the simplified models considered here.

Good fidelity was demonstrated between the complex ANSYS model and ‘hand calculations’ based on first principles.

5.2 Future Work

This first effort at rotordynamic analysis of the AM600 T-G shaftline produced results which met expectations that the number and separation of torsional eigenvalues would be promising for the AM600 design concept as related to robust performance in less than robust grids.

L-0 Blade Modelling – It is understood that participation of the large and massive L-0 blades does play a major role in torsional vibration response. It is expected that the addition of these blades (and blade rows) to the model will introduce additional eigenvalue frequencies. Follow-on activities for future rotordynamics analysis will include mass and elastic properties of the long L-0 blades (and other rows as necessary). This will include blade attachment modelling to account for stress distribution between the shaftline and the blades.

Transient Response due to 3 Phase Short Circuit Torque **–** The detailed modelling described above will provide a useful basis for torsional analysis of excitation associated with three phase short circuit faults. Also known as per unit air gap torque, the three phase short circuit torque can generate momentary torque as high as twelve (12) times the generator rated torque and can result in damage of turbine blades, couplings, and retaining rings. In future research, the maximum shear stresses will be evaluated on the LP rotor section near the generator.

REFERENCES

[1] Field, R. M., *AM600: A New Look at the Nuclear Steam Cycle*, Nuclear Engineering and Technology 49 (2017), pp 621-631.

[2] Chen, Y., "An Investigation of Excitation Method for Torsional Testing of a Large-Scale Steam Turbine Generator," *Journal of Vibration and Acoustics*, pp. 163-177, 2004.

[3] ANSYS Workbench Version 19.0. (2019). "User guide, training and help manual ".

[4] Newland, E., Mechanical Vibration Analysis and Computation, New York: Dover Publications, 2006.

[5] Vysoky, R., "Current Capabilities of Modal Analysis of Aircraft Propeller in ANSYS Mechanical Environment," Advances in Military Technology (AiMT), Vol. 12, No. 1 (2017).

[6] Walker, D., "Steam Turbine-Generator Torsional Vibration Interaction With the Electrical Network," Electric Power Research Institute (EPRI) Report No.1011679, 2005.