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Torsional and Electrical Dynamic Interactions of Compressor Drive Motors and Gas Turbine Driven Power Generators

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ABSTRACT

Load induced power fluctuations in a pair of synchronous motor driven reciprocating compressor trains have been shown to interact with a gas turbine driven power generation source. Electrical current pulsations predicted at the generator terminals were evaluated to determine the response of both mechanical torsional systems and the potential for micropitting of the generator drive gear teeth. The compressor and generator torsional mechanical systems were modeled along with an electrical model of the interconnecting power system in a time domain analysis of the dynamic interactions.

INTRODUCTION

A natural gas cycling facility is being constructed as an initial production system at the Point Thomson field on Alaska's North Slope. Condensate produced from the reservoir will be sent by pipeline to join an existing pipeline at Badami and from there to the well-known Trans-Alaska oil pipeline. Approximately 200 MMCSFD of natural gas will be returned to the reservoir at 10,000 psi via two parallel, two stage, 10,000 HP reciprocating gas injection compressors. Power generation for the site is provided by four gas turbine driven generators. Two gas fueled units are rated at 7.9 MWe while the other two dual fuel units are fitted with waste heat recovery and produce slightly less power due to additional exhaust losses. The compressor trains consume approximately two thirds of the facility's electrical load from the power generation system that is isolated from a power grid due to the remoteness of the site and operates in island mode.

Initial engineering reviews indicated the potential for large electrical current pulsations on the 13.8 kV bus due to the normal torsional characteristics of the reciprocating compressor trains. These current pulsations could interact with the generator and potentially exciting torsional natural frequencies of the gas turbine driven trains. One of the main concerns was the generator drive's epicyclic gearing being exposed to torsional variations that could cause micropitting of the gear teeth. As part of the initial engineering for the compressor trains, the motor manufacturer performed initial screening calculations for current pulsation. It was predicted that the current pulsation amplitudes could exceed 20 percent under some operating conditions. Later, revised analyses by the motor manufacturer showed reduced values. Further analyses were required using a more refined calculation method to examine the expected interactions with the gas turbine generator trains.

Various compressor operating conditions were simulated via marriage of electrical and mechanical systems into a single digital model to determine the worst case scenario for evaluation and risk assessment. Results showed excitations imposed upon the generator torsional system at frequencies of compressor speed and its harmonics. Employing the refined methods, overall current and torque oscillations of approximately 4 percent of average values were predicted with the dominant frequency being 6 Hz, which is equal to compressor speed of 360 rpm. This is significantly lower than original predictions that exceeded 20 percent.

This paper describes the studies, results and recommendations the project team used to quantify the electrical current pulsation induced excitations and design precautions implemented to assure long-term reliable operation. The first part below provides the groundwork for understanding the concern for micropitting the generator drive train's gear teeth. The second part discusses the typical individual torsional analyses of the compressor and generator trains. The third part illustrates the combined electrical and mechanical model and results derived for the interactions of two separate torsional systems coupled by an electrical bus.

GAS TURBINE DRIVEN GENERATORS – GRID-BASED VERSUS ISLAND-MODE

Gas turbine driven electric generators are installed worldwide to provide power to a variety of markets, but the two main sector divisions are the oil and gas industry and industrial power generation. In both of these markets, turbine generators span a continuum of applications that are bookended at the one end by those that are run against electrical grids and at the other end by those that are run in isolated island mode to provide the complete electrical energy needs for a particular location.

The differences in reactive loads experienced at the generator terminals between grid-based versus island-mode installations can lead to differences in the operating characteristics in terms of dynamic load fluctuations. In general, grid-based installations tend to be more stable, provided the grid is stable, and do not experience much in the way of dynamic loading at steady-state conditions. Island-mode installations run the gamut from stable to dynamic depending upon the equipment and processes at a site, but they generally tend to experience more dynamic loading than grid-based installations. Though these dynamic loads are usually well within the design capabilities of the electrical and mechanical sub-systems of the turbomachinery, they sometimes are significant enough in magnitude, frequency or duration to warrant closer engineering review.

The Point Thomson project is one such program that received additional engineering review. The items that led to this further consideration were:

- This is an island-mode application where four gas turbine generator packages will provide power for the entire site through a main 13.8 kV switchgear electrical bus.
- Two 10,000 hp synchronous motors driving two reciprocating compressors will make up a significant portion of the plant load seen at the electrical bus.
- The two reciprocating compressors will at times run with their mechanical linkages in phase and this is predicted to cause a mechanical torque pulsation that will feed back into the synchronous motors, then to the common electrical bus and then be experienced as torque pulsations by the mechanical components of the generator sets.
- The initial preliminary estimates of the resultant power fluctuations experienced at each generator shaft end indicated that they could be as high as 1.675 MW peak-to-peak at a frequency of 6 Hz – potentially for a significant portion of the equipment's operating life.
- The site is in a remote, environmentally sensitive location on the North Slope of Alaska.

GAS TURBINE GENERATOR SETS WITH EPICYCLIC REDUCTION GEAR DRIVES

As mentioned above, four gas turbine generator sets were provided to the Point Thomson site (see Figure 1). Each of these features a custom-designed epicyclic reduction gear drive that sits in between the engine and generator. The epicyclic gearbox arrangement is advantageous in that it allows for compact, in-line transmission of power from engine to generator (see Figure 2). Since it is one of the core sub-systems of the drive train, it was important to give it a robust review for this project.

From a gear design perspective, gas turbine generator sets are considered to be one of the smoothest running industrial applications having the smallest and least frequent dynamic excursions from nominal load induced on the gearing by the driving and driven equipment. This is reflected in the design phase by applying low overload/application factors as multipliers to the nominal design loads. The higher and more frequent the typical dynamic loads experienced by a gear unit in various applications, the higher the multipliers are to the nominal design loads. This covers the gross effects of externally induced dynamic loading on common load-dependent failure modes like fatigue (both surface and bending modes) and, to a lesser extent, scuffing and wear. A typical factor for the Point Thomson gas turbine driven generators is 1.1 (Radzevich and Dudley, 1994.)

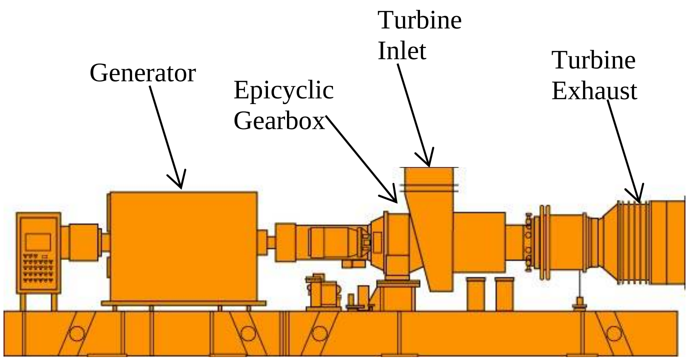

Figure 1. Schematic of a Gas Turbine Generator Package at Point Thomson (Courtesy of Solar Turbines)

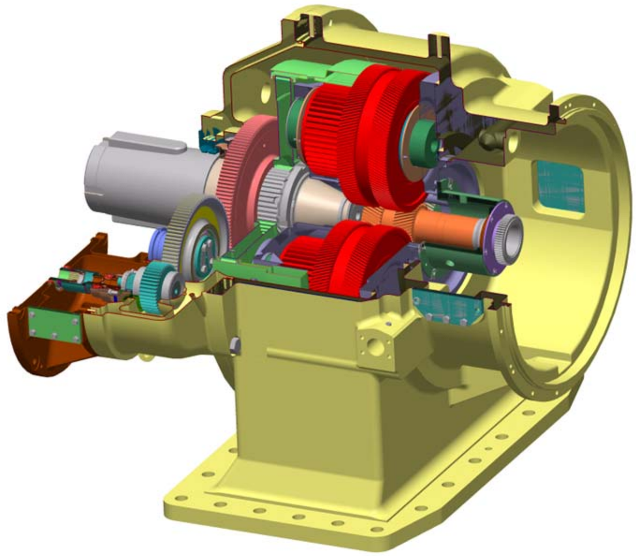

Figure 2. Schematic of an Epicyclic Reduction Gear Drive at Point Thomson (Courtesy of Solar Turbines)

In addition to the gross effects of dynamic loading on common load-dependent gear failure modes, if it is severe enough, dynamic load changes can also disrupt the Elasto-Hydrodynamic Lubrication (EHL) film thickness between the gear teeth. This is particularly true in epicyclic gear units that have floating members that adjust their running positions in response to changing load vectors during dynamic load changes.

Testing of a typical gas turbine with an epicyclic reduction gear drive reveals that this adjustment to running position occurs regularly with instantaneous load step changes during normal operation of the equipment. Figure 3 shows two data plots from one of these tests. The bottom plot shows power versus time for various load steps that occurred during the testing of this package and the top plot shows resulting vibration amplitudes recorded from the reduction gear drive's case-mounted accelerometer during the same period of testing. Note that for each load change shown on the bottom plot, there is a corresponding spike in acceleration on the reduction gear drive housing shown in the top plot. Over the years, related testing on various epicyclic gear units equipped with proximity probes has verified that these instantaneous changes in accelerometer readings correlate to instantaneous positional shifts of the floating internal gear members in response to changes in load vectors.

Normally, these positional adjustments of the floating gear members are seen as a good thing – floating capability is a mandatory feature that must be part of any sound epicyclic gear design if it is to have any chance at achieving an adequate service life. But experience in various other gear industries indicates that these shifts in loaded gear member position can also be accompanied by disruptions to the EHL film between the mating teeth (Heidenreich and Herr, 2012.) If these disruptions are severe and frequent enough, they can lead to another gear tooth failure mode called *micropitting*.

CONSIDERATION OF FAILURE MODES – IMPORTANCE OF MICROPITTING

Dynamic loading contributes to gross effects on common load-dependent gear failure modes that are easily accounted for, but dynamic loading can also have effects on more difficult to assess, less load-dependent failure modes like micropitting. Table 1 gives a listing of the five main gear tooth failure modes that could be affected by dynamic load fluctuations with a comparative summary of the mechanisms, the predictability, how they are accounted for in design rating of a gear set, and the probability of occurrence as a primary failure mode. Of all of the failure modes on the list, micropitting was seen as the one that needed the most careful consideration for the Point Thomson Project. The potential to have relatively large magnitude dynamic load fluctuations occurring for significant portions of the operating life of the equipment, along with the related positional adjustments of an epicyclic gear drive's floating members, meant that there was a high potential for disruption of the EHL film thickness over these periods. This is why micropitting was seen as the failure mode having the most increased probability of occurring: unfortunately it also has the least degree of predictability.

Micropitting is a fatigue failure of the meshing surfaces of a pair of gears characterized by smaller pits than are experienced when macropitting occurs (Drago, et al. 2010.) Pits on the order of 10 to 20 microinches in size can originate when local lubricant film thickness is insufficient. Localized contact of surface asperities fatigues the gear material. The surface asperities are a function of the gear manufacturing processes and range in size based upon the finishing techniques employed. Typically, it is expected that the EHL film will be thick enough to prevent contact between the asperities on one gear with those on a mating gear. Externally imposed dynamic loads that could disrupt the EHL film thickness was a major concern for the Point Thomson Project.

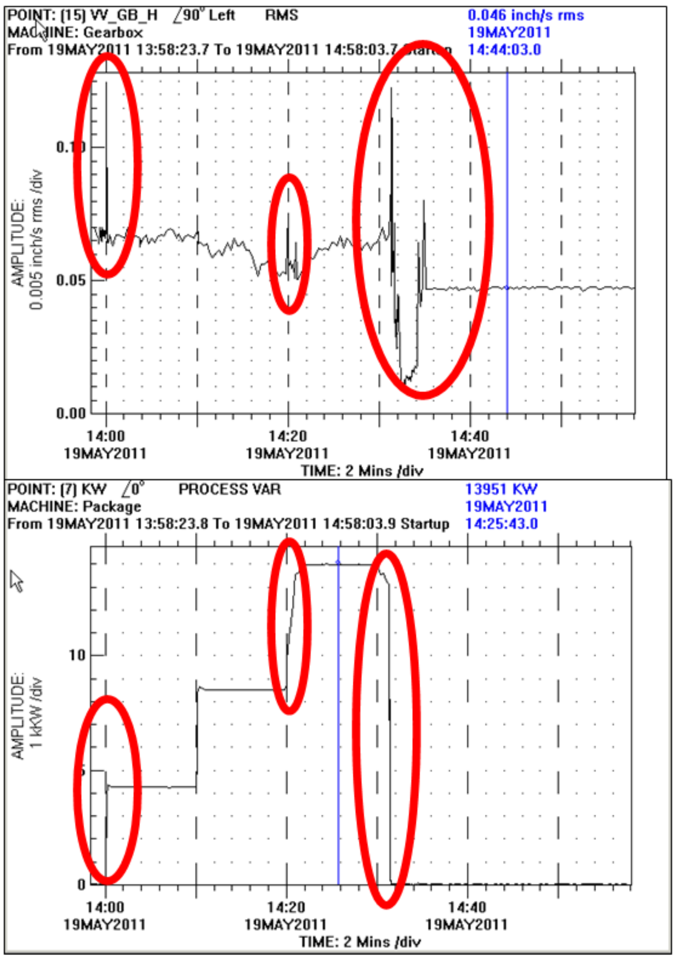

Figure 3. Test of Turbine Generator w/ Epicyclic Gear Unit Load Changes vs. Vibration (Courtesy of Solar Turbines)

Much has been written about micropitting over the years and a full treatment of it is beyond the scope of this paper, but there are several summary points that should be made:

- Micropitting is a relatively recent concern for gearing as it has probably only been recognized as a separate failure mode since the early 1990's. It occurred in gear applications since the 90's. In most of these occurrences, it was usually called *grey staining* or *frosting* and was incorrectly identified as a wear mechanism or sometimes confused with the more well-understood failure mode of macropitting.
- Many of the things that influence micropitting are well understood but there is not yet a reliable, internationally recognized method to predict a gear design's susceptibility to this failure mode. The best approach available today is to compare ways to make a design more robust against this failure mode.
- Micropitting can occur in many different types of applications with completely different operational characteristics, but certain industries see it more frequently than others.
- Turbomachinery applications are not known to be very susceptible to this failure mode, but certain industries (for example, wind energy power generation, where the gear trains are regularly exposed to relatively large magnitude dynamic load fluctuations for significant portions of the operating life) are known to experience a relatively high frequency of micropitting issues.

Table 1. Failure Modes of Gear Teeth

Name	**Mechanism**	**Predict-ability**	**Effect of Load Fluctu-ation**	**Probability of Occurrence ***
Tooth Breakage	Bending Fatigue	Good	Increases Ka	Slightly Increased
Macro pitting	Surface fatigue (sub-surface)	Good	Increases Ka	Slightly Increased
Scuffing	Adhesion	Fair	Little Effect	Slightly Increased
Wear	Abrasion	Poor	Disrupts EHL	NOT Increased
Micro pitting	Deformation/ Cracking of Surface Asperities	Poor	Disrupts EHL	Increased

*As primary mode of failure

Taking all of the above into consideration, it was deemed appropriate for the Point Thomson project to pursue some methods to mitigate risk and reduce the potential of encountering micropitting issues. One of the most important and proven ways to make a gear design more robust against micropitting is to put the finish-ground gear through a final superfinishing stage using a chemically accelerated vibratory process (Arvin, et al. 2002, Bell, et al. 2012, Errichello, 2011, Michaud, et al. 2011, Winkelman, et al., 2010.)

SUPERFINISHING – IMPROVING MICROPITTING FACTOR OF SAFETY

Micropitting is known to be influenced by:

- Operating conditions (e.g., load, speed and sliding temperature)
- Lubricant conditions (e.g., viscosity, additive packages and cleanliness)
- Surface conditions (e.g., roughness, lay, texture)

For a given application, any number of the above items cannot be changed (e.g., load or speed or lubrication), but usually some of the items can be improved to reduce the risk of micropitting. And as mentioned above, one of the proven methods of increasing a gear design's capacity to withstand micropitting is to improve the surface through superfinishing. For a given design that is finish-ground, superfinishing can be used to improve the surface and greatly improve the margin against micropitting. Surface roughness quality is usually described by a *Ra* value which stands for *roughness average* and is one of several parameters used in gear tooth machined surface evaluation (Talati, 2011.)

Typically, precision high-speed turbomachinery gearing is case-carburized and finish-ground. The grinding process usually achieves a surface finish with a roughness of between 32 and 20 micro-inches Ra. The surface is characterized by long shallow peaks and valleys in the direction of the grinding wheel travel (see Figure 4) and the carburized peaks represent sharp, discontinuous, brittle asperities that can serve as initiation sites for micropitting.

Chemically accelerated vibratory superfinishing, is a benign finishing process that removes these peaks and leaves the surface of the gear tooth smooth, neutral and free of initiation sites for micropitting. Surface finishes achieved with superfinishing are typically on the order of less than 4 micro-inches Ra; less than 1 micro-inch is readily achievable, depending upon the needs of the application.

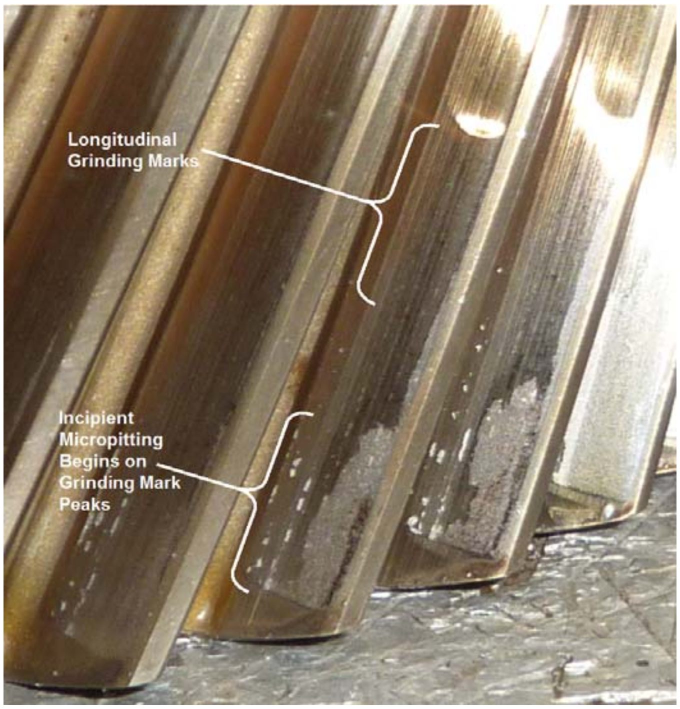

Figure 4. Carburized Ground Gear – Micropitting Initiates on Grinding Mark Peaks (Courtesy of Solar Turbines)

ISO/TR 15144-1 (2010) provides a method for evaluating and calculating a safety factor against micropitting. Equation (1) is as follows:

$$S_\lambda = \frac{\lambda_{GF,\,min}}{\lambda_{GFP}} \geq S_{\lambda,\,min} \tag{1}$$

Where,

S_λ is the safety factor for micropitting
$\lambda_{GF,min}$ is the minimum specific lubricant film thickness in contact area
λ_{GPF} is the permissible specific lubricant film thickness
$S_{\lambda,min}$ is the minimum required safety factor for micropitting

One thing to note about this calculation is that it assumes that the permissible specific film thickness is known. Thus, it assumes that there is a way to calculate it or to know if a given design is on the cusp of micropitting. However, one can also use this equation to calculate a comparison of micropitting safety factors for gears finished with a superfinishing process versus as-ground finishes. Since the specific film thickness is defined as the actual EHL film thickness divided by the effective mean surface roughness, in a given application where the actual EHL film thickness is the same regardless of finish, the film thicknesses cancels out. What is left is the ratio of the reciprocals of the two different effective mean surface roughness values for before and after superfinishing. This yields a comparative safety factor against micropitting. Another way to look at it would be as a margin of increase. Table 2 compares two typically achievable surface roughness values for superfinishing (e.g. Ra=4 and Ra=1) against two typically achievable surface roughness values for as-ground gears (e.g., Ra=32 and Ra=20).

Thus, if an as-ground gear design at 32 micro-inch surface roughness were on the cusp of micropitting (i.e., the safety factor 1.0), switching to an superfinishing process that achieved a surface roughness of 4 micro-inches would yield a safety factor of 8.0 which is an 8X margin of increase.

Table 2. Micropitting Safety Factor Comparison (ref. ISO 15144-1, 2010)

Compare SF = (1/R_a, SF) / (1/Ra, Ground)		
	Ratio SF to 32 micro-inches	**Ratio SF to 20 micro-inches**
@R_a, SF = 10	3.2	2.0
@R_a, SF = 4	8.0	5.0
@R_a, SF = 1	32.0	20.0

MECHANICAL SYSTEMS

Reciprocating Compressors

The two parallel, two-stage, four cylinder, natural gas compressors are a four-throw, horizontally-opposed design driven at 360 RPM by direct coupled synchronous motors. The synchronous drive motors are a single bearing design that attaches to the compressor flywheel. First stage suction pressure is approximately 2,600 psi and final discharge pressure is 10,000 psi. Capacity control is achieved via a recycle arrangement capable of 100 percent flow. At design operating conditions, the compressors consume approximately two-thirds of the electrical power being generated on site.

The action of the reciprocating mechanisms in the injection compressors produces a crank effort torque that has a significant amplitude variation through one revolution of the compressor crankshaft. This, in turn, creates an oscillatory angular variation of the drive motor rotor during each revolution. The oscillatory torsional energy acts as a forced vibration excitation applied to the hard-coupled synchronous drive motor and causes electrical current pulsation that is fed back into the power supply system.

It is common to perform a torsional analysis for most critical machinery trains to determine torsional natural frequencies and system response to operating loads under typical and worst case conditions. Required separation margins are +/-10 percent for compressor speed and +/-5 percent for other compressor orders up to the tenth. The first torsional natural frequency (TNF) was determined to be 44.9 Hz which was well situated between the 7th and 8th compressor orders as shown in Figure 5. The calculated separation margins are -6.4 and +6.9 percent respectively. The machine is torsionally stiff; therefore, the effect on current pulsation is primarily rigid body motion with the inertia helping to control the pulsation. System damping is very low and thus has a negligible effect. The compressor manufacturer performed the torsional analysis and considered the compressor train to be acceptable from a torsional vibration standpoint.

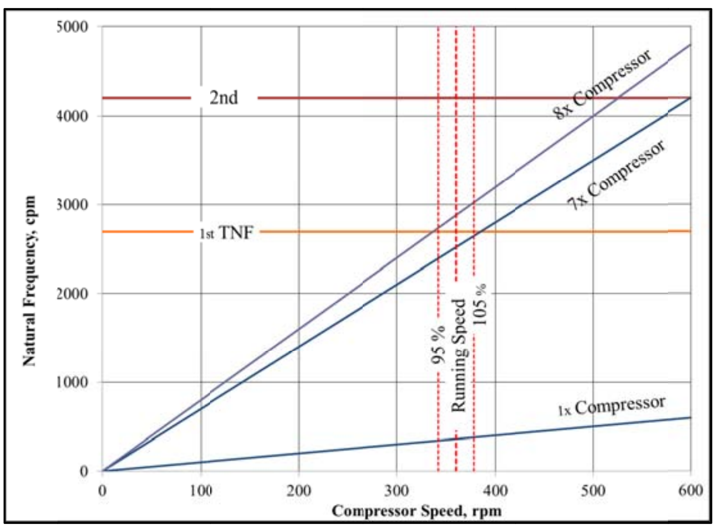

Figure 5. Campbell Diagram for Injection Compressor

According to API 618 (2007) for reciprocating compressors, the electrical current pulsation limit is 66 percent of full load torque for synchronous and 40 percent for induction machines. A limit is also published in NEMA MG1 (2011) and is 66 percent for both synchronous and induction machines. These values provide compressor and motor vendors with a target for robust machine design, but do not necessarily reflect reasonable values for the Point Thomson facility's island mode design. Initial predictions by the motor manufacturer indicated current pulsations could exceed 20 percent and could be higher under some operating conditions (e.g., crank end valve failure or discharge pressure approaching piping relief valve settings. Minimizing the current pulsation at the source and evaluating its impact on other systems is desirable.

Normal practice is that current pulsation on a motor driven reciprocating compressor train is done with the crank effort data (neglecting mass-elastic effects). The motor manufacturer then calculates the current pulsation using the entire train inertia as a rigid body to smooth the current pulsation.

Much later in the project the compressor torsional analysis is done to obtain the true torque pulsation on the motor shaft is obtained. The displacement amplitude of the motor rotor is also then available, and provides the most accurate means to determine current pulsation because it accurately represents the electrical lag (rotor to rotating magnetic field).

The error using the rigid body assumption normal method is small for the normal case that the first torsional natural frequency is well above the fourth harmonic of compressor speed. Note that due to flywheel effect, it is normally only the first four harmonics that are of interest for current pulsation considerations. If the first torsional natural frequency of the compressor train were low (certainly if less than 4x) then the error would be large and the only way to calculate a reasonably accurate current pulsation would be to use the output of the torsional analysis (rotor angular displacement FFT).

A robust design is needed. Even though there are no unloaders and there are no alternate operating conditions, four compressor conditions, (listed below) were considered. The last two are valve failure cases which are not uncommon but are severe cases that the compressor may need to operate at for a limited period of time until it can be shut down for maintenance.

- Normal full load operation
- Operation at final discharge relief valve pressure (max power case)
- Operation with a failed 1st stage discharge valve
- Operation with a failed 2nd stage discharge valve

The compressor manufacturer's torsional analysis included calculated current pulsation amplitudes for these four cases of: 4.4; 5.1; 9.3; and 15.9 percent, respectively. While the normal full load operation value of 4.4 percent was below the original values predicted by the motor manufacturer, further investigation was warranted to determine the interactions of the current pulsations with the turbine and generator torsional system.

Gas Turbine Generators

Site power for the Point Thomson facility is provided by four gas turbine-driven electrical generators. The single shaft turbines are connected to the 1800 RPM generators by an epicyclic gearbox. Under normal operating conditions for 66 percent of the year, four generators are online. Waste heat recovery is employed to provide process and building heat. The four generators are connected to a 13.8 kV switchgear bus.

The turbine manufacturer performed a torsional analysis of the generator drive train that revealed the torsional natural frequencies shown in Table 3.

Two areas of interference between the second and fourth torsional natural frequencies and typical potential excitations are shown on the Campbell Diagram in Figure 6. The separation margins were +1.0 and -9.5 percent, respectively, while the required separation margin was +/- 10 percent. Forced torsional vibration analyses performed by the turbine vendor showed the shaft stresses due to these interferences to be acceptably low. In these analyses it is common to assume forcing functions of between 0.5 and 2.5 percent of the transmitted torque at 1x and 2x of shaft speeds and line frequencies.

Table 3. Turbine Generator Torsional Natural Frequencies

Mode	Frequency, Hz	Frequency, cpm
1	19.15	1149
2	60.61	3637
3	116.02	6961
4	228.70	13722

The initial torsional analysis by the turbine vendor did not take into account excitation from interactions between the injection compressor drive motors and the generators. However, predicted current pulsations imposed upon the electrical system by the compressors became a concern that needed to be addressed. The motor and compressor manufacturers' current pulsation amplitude predictions were scrutinized closely. While no flaws were pointed out in the analysis techniques, it was determined that a more refined and robust analysis would be required to assess the impact of the current pulsations on the gas turbine generator drive trains.

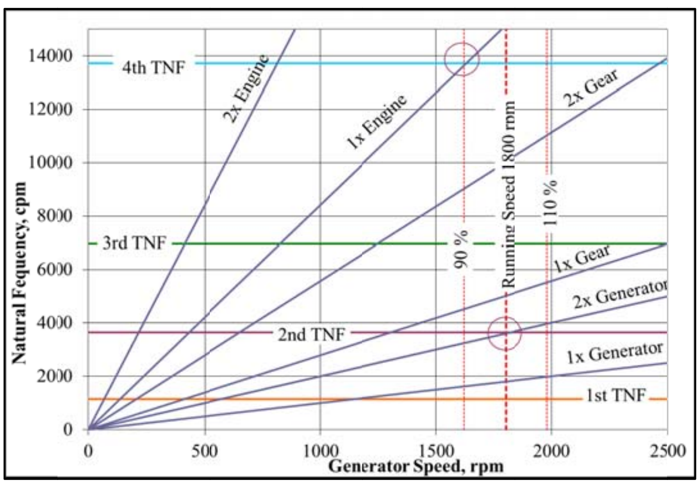

Figure 6. Campbell Diagram for Gas Turbine Generator

PRIOR MODELING HISTORY

Creating coupled electrical-mechanical digital models of machinery to study the effects of steady state and transient dynamics during operation has been performed previously in the industry but is not as common a subject as the usual torsional analyses expected to be performed for the Point Thomson gas compressors and power generators. Several publications have dealt with variable speed drive motors, both of LCI and PWM design (Szolc and Pochanke 2011, Hutten et al., 2008, Rotondo et al., 2009, Feese and Maxfield 2008, Sihler 2009.) The typical excitations studied deal with harmonics generated within the power conversion electronics and the response of a single mechanical system. The analysis techniques include modal frequency domain and time domain methods of finding solutions to sets of differential equations that represent electrical and mechanical portions of the overall system being studied. For the Point Thomson machinery, a technique was required that would provide a time-step solution that dynamically coupled two mechanical torsional mass elastic models (compressor train and gas turbine generator train) with three sets of electrical models (synchronous motor, power generator and balance of plant electrical system.) The required results would illustrate the interactions between excitations originating in the compressor and motor and resulting forced torsional vibration at the generator drive gear's shafting. The selected analysis tool represents and solves the relevant differential equations in the time domain. It is capable of handling multiple electromagnetic and electromechanical systems and their controls. It is analogous to a process simulator used in studying and designing process units in the oil and gas industry (Dommel, 1969, Wilson, 2005.)

ANALYSIS AND RESULTS

Initial torsional modeling by the compressor and turbine manufacturers employed discrete, lumped mass elastic models of the compressor and generator trains. Natural frequencies, mode shapes and steady state forced responses were determined by common Eigen analysis and modal methods. The generator and motor manufacturers also modeled the electrical characteristics of their machines during design. The project's Engineering Procurement and Construction (EPC) contractor modeled the balance of plant electrical system in its design phase. All of these independent models were integrated into one overall system model as shown in Figure 7. Since the frequencies of concern involved the lowest torsional natural frequencies of the two mechanical systems, both mass elastic mechanical models were reduced to fewer equivalent lumped inertial masses and torsional springs. The compressor and generator torsional models originally consisted of 18 and 10 lumped masses respectively, which were reduced to 4 and 5 lumped masses (Figure 8.) This reduced the computation time for the time step analysis without sacrificing accuracy of the results in the frequency range of interest. The overall system model was exercised to verify it produced the same first torsional natural frequencies as the initial analyses by the compressor and turbine manufacturers.

The crank effort torque at the compressor flywheel and drive motor interface shown as a time wave in Figure 9 was applied as an excitation input to the overall system model while operating at a normal steady state condition. The dominant frequency seen in the time wave is 6 Hz which correlates to compressor running speed of 360 rpm. Higher harmonics can be observed in the time wave also. Table 4 illustrates the harmonic content of the torque calculated as a vector addition of the crank effort for each individual compressor throw without mass elastic effect. This was used by the motor manufacturer to calculate the initial current pulsation amplitudes. Table 5 shows the output of the compressor manufacturer's torsional analysis for the torque in the motor shaft on the motor side of the flywheel. There are some differences in Tables 4 and 5 due to the flywheel smoothing the low harmonics and the magnification of the high harmonics (especially near the compressor first torsional natural frequency between the seventh and eighth orders of running speed.)

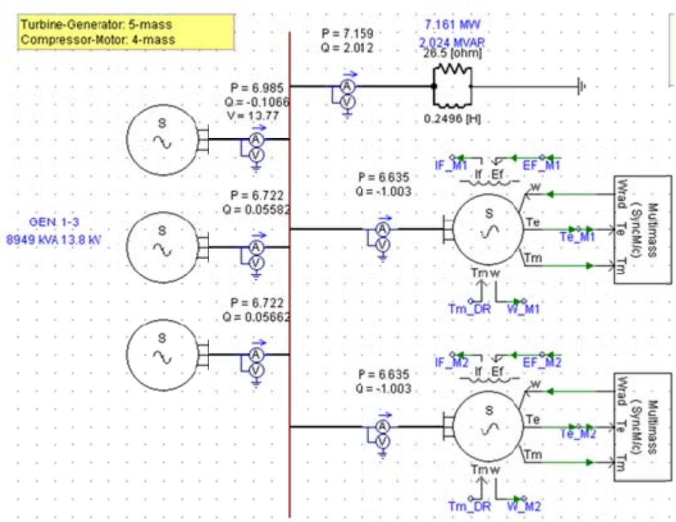

Figure 7. Combined System Model for Three Turbine Generators and Two Compressors Online (Courtesy of ABB)

Table 6 shows the resulting vibratory peak-to-peak (P-P) displacement at the motor rotor expressed in P-P electrical degrees. Note there are 180 electrical degrees between adjacent poles in an electrical motor. As this synchronous motor has 20 poles, there are 20 times, 180 equals 3,600 electrical degrees in one full revolution of the motor. In this motor, the rotor lags the rotating magnetic field by 36 electrical degrees at full load. The vibratory displacement is 1.6 deg which is 1.6 divided by 36, which equals 4.44 percent. Caution, the torque vs. lag of the motor is not linear with load and current; however, this does give a good first order approximation of the current pulsation along with the harmonic content. The motor manufacturer's refined calculations revealed a current pulsation of 4.4 percent at normal operating load which shows good agreement with the compressor torsional analysis. From Table 6 the rotor displacement harmonic strength shows that current pulsation is primarily at 1x motor speed but there is significant current pulsation at the third and fourth harmonics. The current pulsations in the main 13.8 kV bus in turn act at the electric generator terminals and its magnetic field as a forced torsional excitation of the gas turbine and generator drive train.

Comparison of Tables 3 and 6 shows a +6.39 percent separation margin between the third harmonic of motor rotor vibratory displacement and the first torsional natural frequency of the gas turbine generator train. This excitation from the compressor was not considered in the original torsional analysis by the gas turbine manufacturer. Applying this excitation

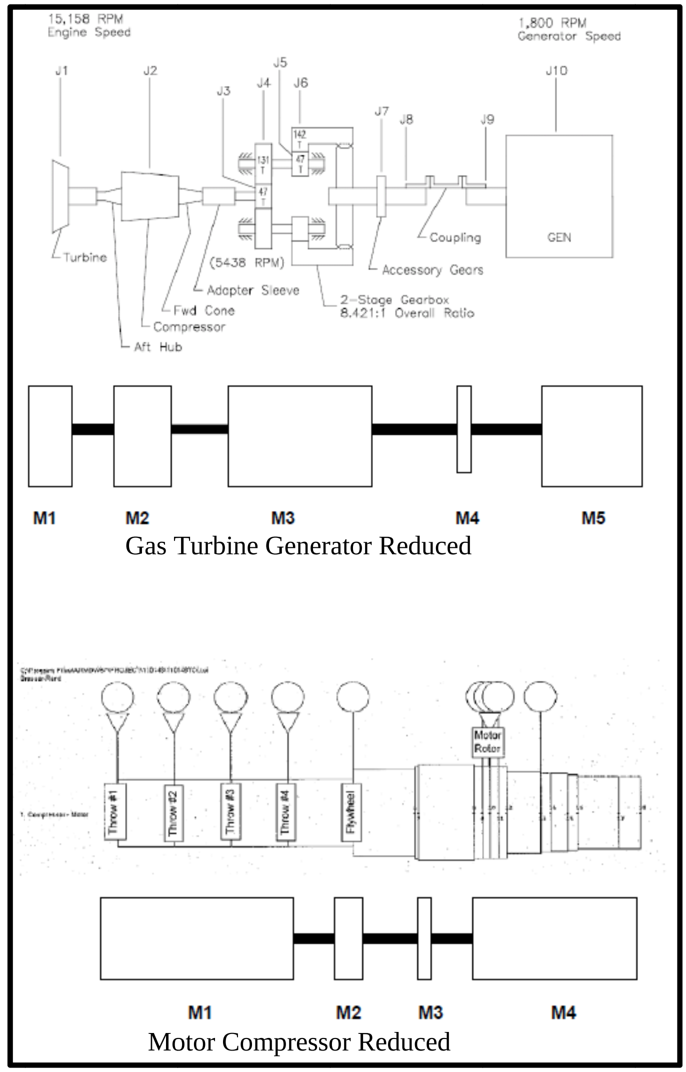

Figure 8. Reduced Torsional Models (Courtesy of ABB, Solar Turbines and Dresser-Rand Company)

independently to the gas turbine generator forced vibration model in the manufacturer's analysis did not predict excessive shaft stress. However, the concern for micropitting of remained.

Since there are two parallel compressors represented in the overall compression system, the relative phasing between the compressors was included as a variable in the analysis. Other variables studied included number of generators online, number of compressors online and balance of plant load in summer vs. winter (7.2 vs. 4.5 MW). These variables along with the four operating conditions listed above were employed to determine best and worst case scenarios to bracket the anticipated current pulsation range. While the analysis tool is capable of delivering outputs of any calculated electrical or torsional mechanical variable as a function of time, the outputs of interest for understanding the current pulsation severity and its impact on generator drive gearing are the dynamic signals of the power at the generator terminals and the torque in the shafting between the gearbox output and generator input.

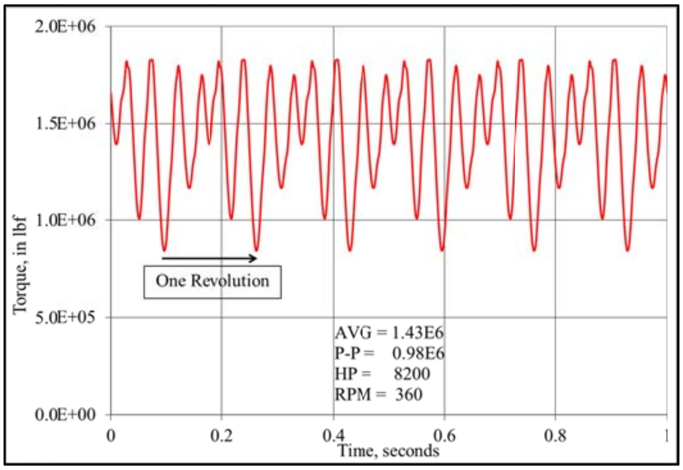

Figure 9. Compressor Crank Effort Torque Time Wave

Table 4. Crank Effort Torque Harmonic Content at 360 RPM

Harmonic	**Frequency**	**4HHE-VL**
	cpm	**Crank Effort FFT, %**
1	360	9.04
2	720	0.00
3	1080	14.56
4	1440	20.96
5	1800	3.22
6	2160	0.00
7	2520	0.54
8	2880	1.91
Total		34.50 %

Table 5. Compressor Torsional Output FFT of Shaft Motor Torque as Percent of Transmitted Torque

Harmonic	**Frequency**	**4HHE-VL**
	cpm	**Torsional Shaft FFT, %**
1	360	7.60
2	720	0.91
3	1080	15.11
4	1440	26.68
5	1800	5.64
6	2160	1.01
7	2520	5.14
8	2880	20.50
Total		52.75 %

The system model was exercised and a worst case *normal* operating condition was found to be when the two compressors were in phase, i.e., each compressor reaches top dead center of cylinder number one at the same time. This is referred to as Case 1 and is for a winter facility electrical load with one fully loaded turbine and two turbines sharing the remaining facility load equally. Figure 10 illustrates the power fluctuations predicted at the generator terminals for this case for a one second time period. Figure 11 illustrates the shaft torque time wave for the same time period as shown in Figure 8. Both figures show a dominant 6 Hz frequency component (reciprocating compressor speed) with higher harmonic content very noticeable. The higher harmonics were determined to be 18 and 24 Hz by FFT analysis which correlates with the third and fourth harmonics of compressor speed. The slight amplitude differences in the three traces in each of these figures is due to the unequal loading on the generators.

Table 6. Compressor Torsional Output FFT of Motor Rotor Vibratory Displacement in P-P Electrical Degrees

Harmonic	**Frequency**	**4HHE-VL**	**4HHE-VL**
	cpm	**Rotor P_P Displacement, deg**	**Harmonic Strength, %**
1	360	1.230	3.42
2	720	0.038	0.11
3	1080	0.254	0.71
4	1440	0.282	0.78
5	1800	0.040	0.11
6	2160	0.004	0.01
7	2520	0.024	0.07
8	2880	0.068	0.19
Total		1.6	4.44 %

Note: Harmonic strength is the rotor P-P displacement/36 deg rotor to magnetic field full load lag; harmonics are added by vector addition due to phase angles.

Table 7 lists some of the results developed by exercising the model for several of the cases that were investigated. The impact of relative phasing (TDC of cylinder number 1) of the two compressors was investigated. This demonstrated how the various response frequencies' amplitudes changed from in phase to out-of-phase. For in phase operation of the compressors, Case 1, peak to peak power oscillations at the generator terminals were predicted to be 0.28 MW. This is 3.9 percent of the generator power - considerably lower than the initial predictions using other less sophisticated techniques but that matches fairly well with the 4.4 percent from the compressor manufacturer's torsional analysis. The P-P torque oscillation in the generator/gear shafting is shown to be 13,460 in-lbf while the transmitted torque is 342,000 in-lbf. Thus the torque oscillation is 3.9 percent of the transmitted torque.

Table 7 is separated into several groupings of cases with some data repeated for clarity. The first group (cases 1, 15.1 through 15.5, and 9) compares winter loads for various phasing arrangements of the two compressors with three generators online. Having two compressors in phase yields the highest amplitude for normal operation. At different phasing of the compressors there can be addition or cancelling of harmonics in the resulting power oscillations seen at the generator terminals. Overall, the cases in the first group show the variation in system response to the variation in the two compressors' relative phasing for the possible phasing between motor poles for each compressor.

FFT analyses of the torque time waves at the first generator input shaft are illustrated in Figure 12 for these cases. The 6 Hz component is seen to have a monotonically decreasing amplitude from in phase to out of phase operation. However, the 18 and 24 Hz components are the highest for an in phase condition but do not show the same decreasing pattern as the 6 Hz component. If the relative phasing between the compressors could be accurately controlled for each motor start-up, the overall torque oscillations could be minimized at an angle of 144 degrees between top dead center of cylinder number 1 for the two compressors. There is not a reliable method to assure that the two compressors always will operate with the same relative phasing so the design must be robust enough to be continually subjected to the worst case condition (case1).

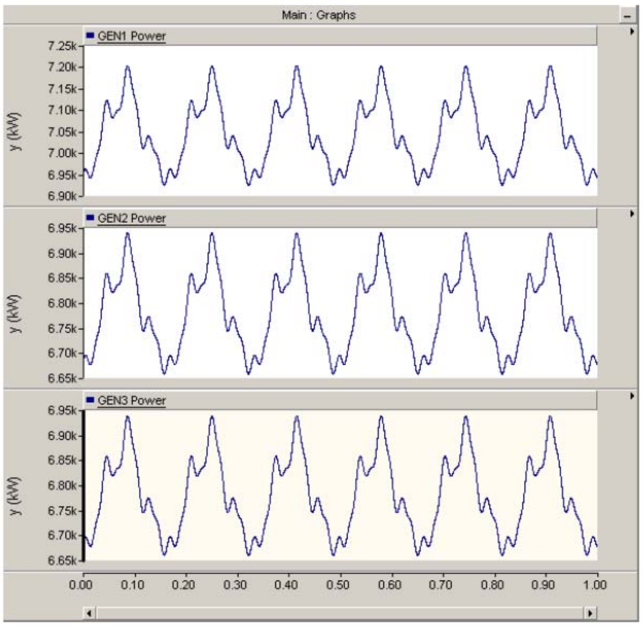

Figure 10. Power Oscillations at Each Generator's Terminals for Case 1

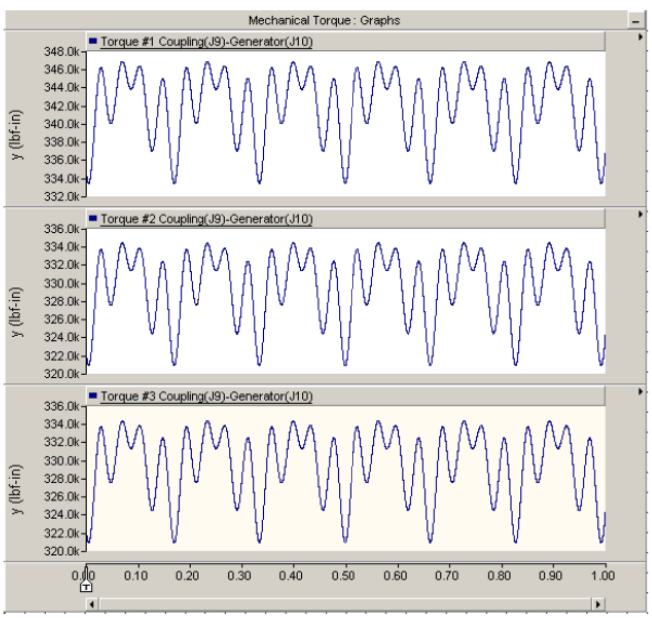

Figure 11. Torque Oscillations in Shafting between Gearboxes and Generators for Case 1

The second group in Table 7 compares winter and summer loads and four vs. three generators online. While four generators will be online for 66 percent of the time, the three generator online case shows higher amplitude results and is thus the controlling case. The third group compares compressor valve failure scenarios. The higher amplitudes shown would only exist for a short time until the compressor is shut down.

The last group compares the original case 1 with a case that includes a lower first natural frequency for the gas turbine generator train. Since the gas turbine generator system's first torsional natural frequency was 19.2 Hz, the torsional excitation at 18 Hz played a role in how severe the response was predicted because of the insufficient separation margin referenced above. Considering torsional excitations of the gas turbine generator train that are due to forcing functions rooted in another remotely located machine is not common. However, for the Point Thomson Project, it is necessary. By reducing the torsional stiffness of the coupling between the generator and the gearbox, the first torsional natural frequency of that system is reduced to 15.66 Hz yielding a separation margin of -12.96 percent. The torque oscillations in that shafting are reduced to 10,100 in-lbf but the power oscillations are almost constant. The power oscillations were reduced by only 1 percent but the resulting torque oscillations were reduced by 25 percent. This is due to the softer coupling transmitting less oscillatory torque to the gear by somewhat isolating it from the generator inertia.

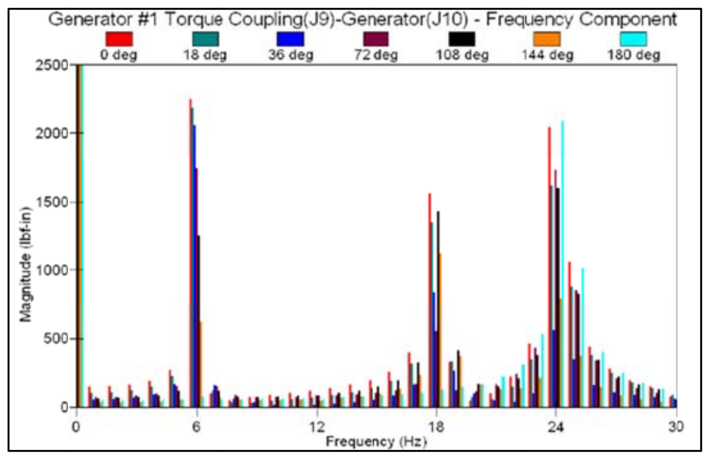

Figure 12. FFT Analyses Results of Torque Time Wave as a Function of Compressor Relative Crank Phasing

POTENTIAL SYSTEM MODIFICATIONS

Now that an analysis tool was available to evaluate the system, various system modifications could be investigated to assess effectiveness in reducing the response amplitudes. A risk assessment was conducted with all involved parties to evaluate each potential modification. Due to the project timeline when the risk assessment was performed, several of the modifications were not deemed feasible (marked as NF below) without considerable redesign of the facility.

The potential modifications were categorized as follows:

- Improve gear tolerance to torsional oscillations by
  - Improving tooth surface finish (superfinishing)
  - Increase oil film thickness on gear teeth
  - Gearbox modifications – frame size increase – NF
- Isolate oscillatory energy within injection compressor by
  - Increasing polar mass inertia – NF

- Compressor modifications – 6-throw vs. 4-throw crankshaft – NF
- Motor modifications – two bearing motor with torsionally softer shaft; induction motor – NF
- Coupling modifications – torsionally soft and/or damper couplings – NF

- Reducing oscillatory energy in the electrical system by
  - using VFD motors for compressor drivers – NF
  - Using static electrical filters to detune frequency – NF
- Isolate oscillatory energy within the generator train (prior to the gearbox) by
  - Increasing mass inertia of the generator windings – NF
  - Generator modifications – flywheel addition – NF
  - Judiciously torsionally tune coupling between gearbox and generator
- Provide spare gearbox and store on site

SELECTED MODIFICATIONS

- Improve gear teeth surface finish with superfinishing
  - A margin increase of 8x was achieved (see Figure 13)
- Increase lube oil viscosity to increase EHL film thickness
  - Change the lube oil for the gas turbine generator from ISO VG 32 to ISO VG 46
- Increase polar mass inertia of the compressor flywheel which was implemented early in the project
  - The flywheel polar mass inertia was increased as much as possible within the starting constraints of the motor prior to investigating the current pulsations
- Tune the torsional stiffness of the coupling between the gearbox and generator to provide additional separation margin between the first torsional natural frequency and the external excitation at 18 Hz
  - The coupling torsional stiffness was reduced from 122E6 in-lbf/radian to 54E6 in-lbf/radian
- Provide a spare gearbox stored onsite
  - Due to the remoteness of the Point Thomson site on the North Slope of Alaska, and the long delivery time for a new gearbox, a new spare gear box was purchased. Any anticipated gear failure because of micropitting was considered to be contained within the gearbox casing

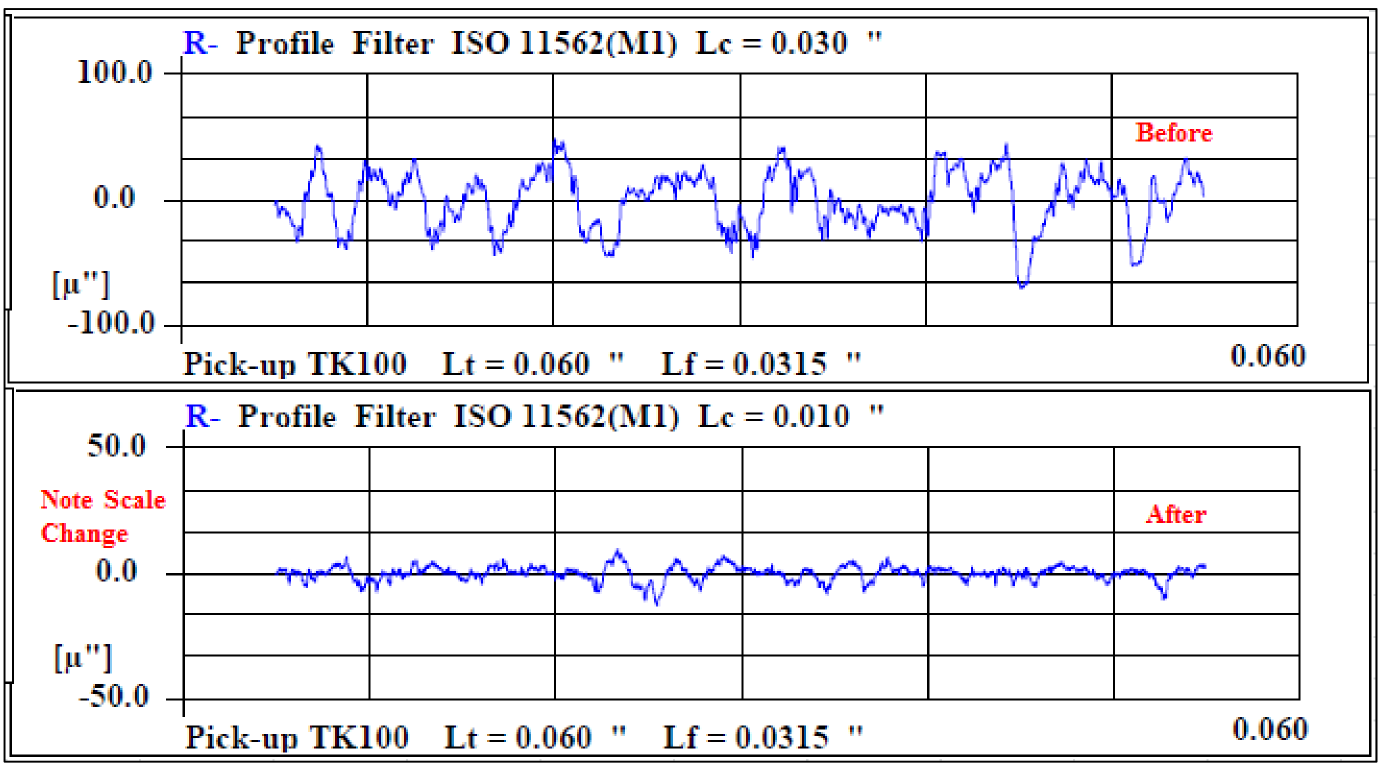

Figure 13. Example of Tooth Surface Roughness *Before* and *After* Superfinishing (Courtesy of Solar Turbines

Figure 13 shows actual measurement examples of one of the Point Thomson epicyclic gear's tooth surface finish *before* and *after* chemically accelerated vibratory superfinishing. The upper trace *before* superfinishing has an Ra of 19.3 while the lower trace *after* superfinishing has an Ra of 2.4. This example illustrates an increase in safety factor for micropitting of 8.1x according to a ratio of Equation (1) applied *before* and *after* superfinishing. While the example may not show measurements at exactly the same tooth profile location, sufficient *before* and *after* measurements were recorded to confirm the safety margin improvement was achieved. The *after* trace is on a different vertical scale, such that a more pronounced effect is actually achieved than is evidenced in a visual review of Figure 13.

Table 7. Partial Summary of Cases Investigated

Case	Facility Electrical Load	No of Comprs. Online	Two Compressor Phase Angle Shift of Motor Torque Pulsations degrees	No of Gens. Online	Peak-to-Peak Generator Power Oscillation kW	Peak-to-Peak Generator/ Gear Torque Oscillation in-lbf	Compressor Operation
1	Winter	2	0 In Phase	3	278	13460	Normal
15.1	Winter	2	18	3	249	11933	Normal
15.2	Winter	2	36	3	225	8170	Normal
15.3	Winter	2	72	3	196	10309	Normal
15.4	Winter	2	108	3	159	9959	Normal
15.5	Winter	2	144	3	107	5980	Normal
9	Winter	2	180 Out of Phase	3	68	7310	Normal
1	Winter	2	0 In Phase	3	278	13460	Normal
2	Summer	2	0 In Phase	3	272	13032	Normal
5a	Winter	2	0 In Phase	4	220	10470	Normal
7	Winter	1	0 In Phase	3	157	7578	Normal
16	Summer	2	0 In Phase	3	380	16500	1st Stage Valve Failure
16	Summer	2	180 Out of Phase	3	200	11400	1st Stage Valve Failure
17	Summer	2	0 In Phase	3	540	21500	2nd Stage Valve Failure
17	Summer	2	180 Out of Phase	3	350	16100	2nd Stage Valve Failure
1	Winter	2	0 In Phase	3	278	13460	Normal
18	Winter	2	0 In Phase	3	275	10100	Normal

CONCLUSIONS

Based upon the analysis of the current pulsations for the combined electrical and mechanical systems the following conclusions can be stated:

- Significant electrical system current pulsations are not normally present nor a concern in machinery systems similar to those at Point Thomson since reciprocating compressor loads are typically small compared to grid size. Thus they are not normally checked when dynamic analyses such as torsional natural frequency and forced vibration calculations are made. The fact that this phenomenon exists for the Point Thomson Project is unique because this is an *electrical island* with the reciprocating compressor being the major load on the generator.
- Electrical system pulsations do not normally cause significant torsional vibration, but in this case a torsional resonance condition exacerbated the situation as very little damping is available to control maximum response amplitudes.
- The turbine and gear manufacturer expressed concern for micropitting failure in the gearbox in its drive train due to power fluctuations imposed upon the generators' terminals by the injection compressor drive motors.
- Current and power pulsations have been adequately modeled between the injection compressor drive motor and the power generators. The predicted amplitudes are much lower than originally reported by the motor manufacturer but were sufficient to warrant closer engineering review.
- The modeled system's worst case power pulsation was predicted to be 0.28 MW under normal operating conditions.
- The effects of the predicted 0.28 MW power pulsation could be reduced by generator train coupling stiffness tuning.
- Relative phasing (misalignment of TDC) of the two injection compressors during operation can reduce the power oscillations. However, the degree of misalignment cannot be controlled, so a worst case scenario of in phase operation must be assumed for continuous operation.
- Superfinishing can be used as a benign method to improve gear tooth surface finish and increase the margin of safety against micropitting.
- Operating with all four gas turbine generators online reduces the predicted torsional excitation experienced at the gear

teeth. However, maintaining sufficient load on the turbine to achieve emissions requirements must be considered.

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