Proceedings of the First Middle East Turbomachinery Symposium
February 13-16, 2011, Doha, Qatar

VSDS Motor Inverter Design Concept for Compressor Trains avoiding Interharmonics in Operating Speed Range

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ABSTRACT

During operation of compressor trains by a variable speed drive system (VSDS), integer and non-integer harmonic currents are generated in the inverter. Via the electrical system of the inverter and the motor, an excitation torque is transferred across the motor air gap into the main mass of the motor rotor. The frequency of this excitation may cause torsional resonances. Due to the rapid increase in excitation frequency of integer harmonics, intersections with relevant torsional natural frequencies (TNFs) can in general be avoided within the operating speed range. In contrast, the intersections of the non-integer harmonic excitation frequencies, also called interharmonics, with TNFs within the operating speed range may have an essential impact on the vibration behaviour of the rotating equipment. This aspect has to differentiate between two train configurations. The first are direct driven trains and the second, trains including an intermediate gear. For direct driven trains, only fatigue problems have to be considered. In trains with an intermediate gear, on top of that, interaction of torsional and lateral movement may have a negative effect on the lateral vibration behaviour of the gear rotors.

This paper will present as a first step an example of simulating a fully coupled electrical and mechanical VSDS train where the interharmonics intersect the main TNFs within the operating speed range. Furthermore, basic considerations are made with regard to operation experience of torsional/ lateral vibration interaction in trains with gears.

However, the main focus of this publication is on a simple but effective method for turbo compressor applications that allows avoiding main resonances within the operating speed range caused by intersections of interharmonic excitations with relevant TNFs. This method is based on detailed knowledge of the inverter behaviour and possible design options of the motor itself. This in-depth understanding was developed by correlating numerical and experimental results based on dynamic torque measurements of real turbo compressor trains. During this investigation the mechanically relevant torsional excitations were identified. Therefore, the different types of inverters and their corresponding characteristics had to be analyzed and understood in detail. This knowledge, in combination with possible motor designs, with regard to the number of pole pairs and the most common train configurations (direct driven and/or trains including intermediate gears), is incorporated in this report.

INTRODUCTION

Application Range of VSDS Driven Trains

Rotating equipment in the turbomachinery industry traditionally uses mechanical drivers as prime movers – partially because it's always been that way and because process and mechanical engineers have confidence in their equipment and might have reservations about yet not installed electrical equipment. Ongoing discussions about energy efficiency, equipment availability, operability and avoidance of green house gas emissions (GHG) has lead to a steadily increasing use of electric motors, either as fixed speed or as variable speed drivers. The industry has reacted with an array of electrical drive systems that can beneficially replace gas turbines and gas engines as prime movers of compressors and pumps.

In smaller power ratings the electric motor is unchallenged in all industrial fields, including the oil & gas industry: the low voltage induction motor is practically the only driver for pumps, compressors, fans, and hoists. Its simplicity, robustness, and performance is unmatched by any other drive in most all applications. In Megawatt power ratings, however, electric motors are challenged by gas turbines. Detailed driver selection studies are the rule when it comes to find the best suited driver for a given application. With the introduction of electronic variable speed drives in the late '70s to the industry the benefits of fixed speed electric motor drives have been significantly enhanced and these additional features are most often the reason for their selection:

- Soft start and fully torque controlled operation over a wide speed range
- Dynamic & accurate speed control via electronic variable speed controllers
- Ability to ride through brief power bus disturbances
- Energy efficiency above 95 percent also in part-load mode and related speed range

- Shaft speeds in excess of the customary 3000. or 3600.rpm dictated by the power system frequency, eliminating step-up gears in many cases
- Insensitivity to frequent start/stop cycles and ability to (re)start fully loaded compressors
- Instantaneous starting capability provides process flexibility
- Lower in GHG and noise emissions

Suppliers of such motors and many engineering contractors have the experience, know-how, and tools to select and recommend the optimum variable speed drive system for a given application. Some typical VSDS applications in the oil & gas industry are:

- Gas storages, which require a very wide speed range of 30 – 105 percent and a power up to 30 MW. The compressors in this application are often direct driven which requires a motor speed > 3600 rpm.

- Pipeline compressor drivers are often used in the power range of 10 MW – 20 MW. The corresponding motor operating speed is in the range of 1500 rpm or 1800 rpm. For this application the avoidance of interharmonics is crucial because the grid frequency and the motor frequency are close together and this may lead to a limited speed range.

History

After the technology's potential to realize variable speed operating envelopes in combination with high efficient electric drives was discovered, it was installed more frequently. However, it turned out that the ecologic and economic advantages of all-electric compression with VSDS driven trains come along with a technical issue to solve.

For many years occasionally high lateral vibration in intermediate gears as well as coupling or shaft end damage occurred and was reported to the industry (Corcoran, et al., 2010), (Kita, et al., 2008), (Kocur and Corcoran, 2008), (Naldi, et al., 2008), (de la Roche and Howes, 2005), (Feese and Maxfield, 2008). Measurements revealed high torque oscillation amplitudes, initially caused by torque oscillations generated in the inverter. These travelled across the motor air gap towards the rotating equipment and excited the fundamental TNF of the entire train.

Variable speed drive systems rectify alternating line current (AC) of 50 Hz and/or 60 Hz, to direct current (DC), and invert the DC to a variable frequency AC current in order to operate the motor at variable speeds. The electrical conversion from line side to the motor side, quite small harmonic distortion of the inverter output current causes forced torsional vibration. Due to small amplitudes, this is outside the resonances of a well endurable load for the train components. Unfortunately, the vibration is amplified when the excitation frequency of torque ripples match a TNF with a suitable mode shape to excite the train. These can then be high enough to either transfer the torsional vibration energy into lateral pinion vibration through the gear, or even exceed the component's fatigue lifetime capacity.

The turbocompressor manufacturers historically dealt mainly with torsionally easy to handle gas and steam turbine drives and fixed-speed electric motors. They now had to close the ranks with the electric drive equipment manufacturers to gain ground in bringing the two disciplines, electrical and mechanical engineering, together.

Technical Impact of Torsional Resonances Excited by VSDS

In principle, generated harmonic torque oscillations may have an essential impact on the torsional vibration behavior of the entire train. Consequently, the train-responsible party, mostly the compressor manufacturer, must carry out detailed analyses to examine the operational condition of the rotating equipment in order to do a proper engineering design. Therefore, close collaboration of driver and compressor manufacturer in designing and engineering of such a VSDS-driven train is essential, as also stated by Hudson (1992).

First of all, it is an essential task to avoid fatigue in the torque-transmitting elements. Torsional excitation may cause fatigue which could eventually lead to a catastrophic failure of torque transmitting elements. During the engineering phase of any project the occurring peak torque and the corresponding torque capability of each individual train component has to be evaluated. Furthermore, in systems including intermediate gears, elevated lateral vibration of the pinion and bull-gear rotors could also occur. Due to the fact that torsional and lateral vibrations are coupled via the gear mesh, excitations have to be examined to avoid higher lateral amplitudes and/or to avoid clattering in gears in addition to possible fatigue problems.

Based on the authors' experience, excessive high lateral vibrations caused by a torsional excitation were observed in some cases. It is due to the coupled movement in lateral and torsional direction, a more or less plausible behaviour. Nevertheless, the authors have, in some cases, also observed high torsional vibration amplitudes and, simultaneously, only few microns of relative shaft corresponding to the torsional excitation frequency have been observed.

Ultimately, only a dynamic torque measurement during string testing and/or during commissioning would be able to identify the potential risk of a failure. The alternative is to have a reliable strategy for the engineering. One specific will be presented as the main topic of this paper, but also other options will be discussed which positively influence the aspects.

GENERATION OF TORQUE RIPPLES IN VSDS

Principle of Generation of Torque Ripples

The motor air gap torque is generated by the rotor flux in combination with the stator current. For a perfect sinusoidal stator current waveform and a perfect air gap field, the motor torque would be constant. Using a VFD, the motor current waveform is not perfectly sinusoidal. The AC-DC-AC conversion adds torsional excitation frequencies to the system. Integer harmonics and interharmonics excitation generated in the converter cause torque oscillations in the motor air-gap. This effect cannot be disregarded due to the fact that interharmonic excitations can be of such frequency that they can generate torsional resonances in the operating speed range.

The air gap torque ripple generated by the VFD characteristic can be split in two categories (Figure 1):

- Integer harmonic torque excitation
- Interharmonic torque excitation

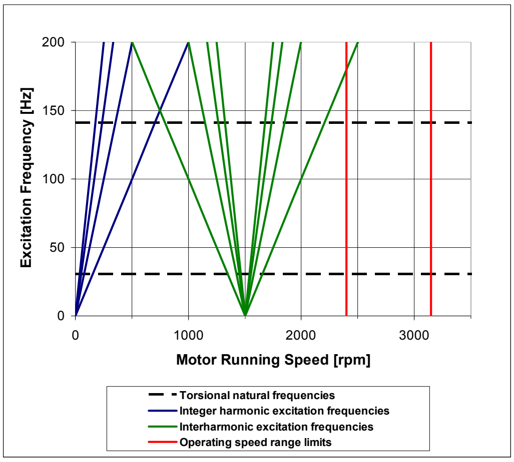

Figure 1. Typical VFD Campbell Diagram.

The excitation frequencies are written for the integer harmonic torque harmonics as $f_{exc\text{-}h} = C_1 * f_{do}$ and for the interharmonics as $f_{exc\text{-}i} = |(C_2 * f_{do} - C_3 * f_l)|$.

The integer harmonics are directly proportional to the motor stator current frequency and therefore the motor speed. The characteristics depend on the converter topology e.g. VSI or LCI and the pulse number of the motor side rectifier. The amplitude of the air gap torque ripple depends, just to mention the main factors, on:

- Switching device characteristic
- Motor impedance
- Motor voltage
- Motor cable characteristic
- PWM characteristic for VSI drives

The non-integer harmonics are caused by the not perfect DC current (LCI drives) or DC-voltage (VSI drives). This means the characteristic of the line side inverter is modulated on the motor currents by the motor side rectifier. Because of this modulation, the frequency of the interharmonics depends on the line frequency, the motor frequency and the pulse number of the motor and the line side rectifier. The amplitude is influenced by the same parameters as the integer harmonics plus additional parameters of the DC-link and the line side:

- DC-link capacitance / inductance
- Line side characteristic (harmonic pre-load, frequency-dependant impedance, etc.)
- Transformer impedance

The resulting torque ripple frequency of the interharmonic air gap torque varies within the range of operation depending on the parameters explained before. Nevertheless, this may result in an interaction with the TNF of the mechanical string even if the amplitude is much lower than the amplitude of the integer harmonics.

Because of the large number of parameters influencing the amplitude of the air gap torque ripple the prediction of specific amplitude is complicated and only possible with tolerances. But knowing the drive and motor type the torque ripple frequencies over the complete speed range can be predicted easily even without any simulation.

Nevertheless, it has to be pointed out that this kind of behaviour is inherent to all state of the art inverters in the entire market. It varies only with regard to interharmonic frequencies and excitation magnitude for each particular configuration.

Typical Induction Motor Design

As the motor pole number is decisive for the motor speeds where resonances occur, it is essential to know which type of motor can be installed for which kind of application. The main components of an induction motor are the stator winding and the rotor. In general 2 rotor types are used:

- Slipring rotor
- Squirrel cage

In the turbomachinery industry the squirrel cage induction motor is the common choice for the low and midsize power ratings due to hazardous area operation and reduced maintenance. The induction motor can be built in 2, 4, 6, … pole design (equivalent to number of pole pairs of 1, 2, 3, …) and this leads to synchronous speed of:

$$n_{syn} = (f_l / N_{PP}) * 60 \qquad (1)$$

The difference between the synchronous speed of an induction motor and the actual speed under load conditions is called “slip”. A typical value for slip of motor speed in the megawatt range is 0.2 percent. Due to the simple and very robust design, the induction motor is widely used in the turbomachinery industry for applications in the range up to 15 MW.

Typical Synchronous Motor Design

The following rotor types for synchronous motors are typically used in the turbomachinery industry:

- Solid cylindrical rotor
- Laminated cylindrical rotor
- Salient pole rotor
- Permanent magnet rotor

The solid cylindrical rotor is mostly used in high-speed or high-power driven applications. The salient pole rotor is the most cost effective design and widely used for DOL started motors. The synchronous motor can be designed in a way to increase the power factor of the supply line in DOL applications. Permanent magnet rotors are typically used under specific requirements and for lower power range.

Overview of Relevant Frequency Converter Types

In the turbomachinery industry 2 frequency converter types are typically used:

- Voltage source inverter (VSI)
- Load commutated inverter (LCI)

Both types have specific advantages and disadvantages and the selection is based on power and voltage range, complexity and the reference situation. In general VSI are used for the lower power ratings up to 25 MW and the LCI is the preferred solution for the highest power ratings up to 120 MW. In the range of 15 to 25 MW both topologies can be used. The VSI can be used with all motor types and topologies with different pulse numbers. The VSI will generate motor voltage in block form. The resulting motor current depends on the stator inductance, the cable parameter, the pulse number and the control characteristic. Nevertheless, the current total harmonic distortion (THD) of VSI drives is lower than the current THD of LCI drives. As explained before, this leads to a lower torque ripple which may be advantageous for the overall compressor string design.

The LCI instead can be used for synchronous motors only. Also for the LCI topologies, different pulse numbers are available. The motor current is generated in a block form, results finally in higher current THD.

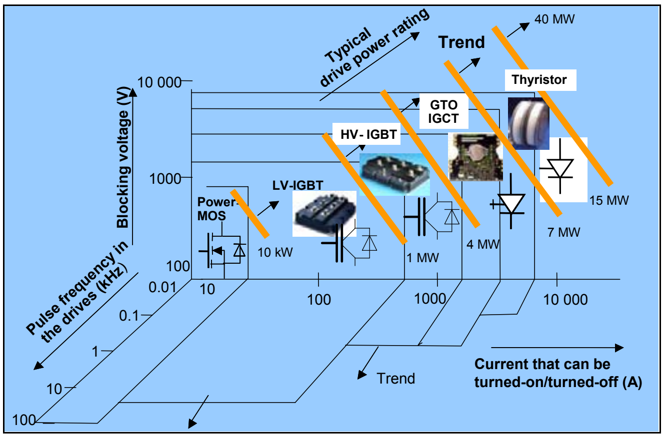

Figure 2. Power Semiconductors.

EXPERIENCE WITH AND CONSEQUENCES OF INTERHARMONIC EXCITATION

Simulation of a Full-Coupled Electrical and Mechanical Model

To ensure reliable operation of a proposed 75 MW refrigerant compressor train in an all electric driven liquefied natural gas plant (LNG), the effect of the full coupled electrical and mechanical VSDS train, including the electrical grid had to be investigated as already published by Hütten, et al. (2008). In addition, with respect to torsional oscillations, two or more identical trains in parallel operation were investigated in detail in order to determine their interaction. This paper details multiple issues associated with the electrical and mechanical interaction and additionally the interaction of trains in parallel operation with respect to torsional oscillations during converter operation.

The maximum excitation of the first torsional mode occurs when at least one of the trains operates at a frequency at which the fundamental TNF can be stimulated by the excitation torque $|6*(f_{do}-f_i)|$ (lowest order of expected interharmonics) generated in the inverter. TNFs of higher orders for conventional train configurations (motor-compressor, motor-gear-compressor) typically cannot be excited by an excitation at the motor air gap due to low amplification of the mechanical system.

The observed interactions of the electrical and mechanical system of two or more VSDS were all at an acceptable level based on this analysis. Especially the electrical model of the inverter, the electrical motor, the existing electrical grid in combination with the torsional damping represents a multiple parameter system. To demonstrate the mechanical integrity of the train a sensitivity study was carried out. The systems sensitivity was investigated with respect to a half- and fully loaded train, different short circuit capacities, different levels of torsional system damping, and finally, in regard to a variation of polar wheel-angle. Each individual parameter varies in a certain range for realistic operating conditions. Therefore, for such a system a "worst-worst case" scenario could always be created by an accumulation of individual worst case scenarios, although it is actually an quite unlikely situation.

Comparison of Simulated and Measured Results

Several VSDS driven trains were investigated on a numerical basis. For some of them, measured results of dynamic torques are available for correlation. The analytical investigation is generally able to identify the relevant interharmonic excitations. The TNF and the corresponding motor speed can be determined with satisfactory accuracy. But nevertheless, based on the authors experience of correlating results of various simulated and observed torsional turbomachinery systems, the peak torque amplitude in the state of resonance condition cannot be predicted with sufficient accuracy in order to carry out a fatigue analysis on a numerical basis only. Therefore, it can be concluded that further uncertainties exist in the electrical and also in the mechanical model. To compensate for these uncertainties the service factors are conservatively selected. This guarantees safe design while accepting the drawback of over-engineering. For an accurate prediction of the occurring dynamic torque in the train elements, the magnitude of torque excitations including realistic tolerances are essential. Further investigations and improvements of the electro-mechanical simulation model are of course an ongoing task.

Experience with Interaction of Torsional and Lateral Vibrations

For trains including an intermediate gear and/or for trains with an integrally geared type compressor, the torsional and lateral vibration system are coupled in movement via the gear mesh. In such a case, the lateral vibration spectrum also presents frequency components of the torsional excitation frequency.

Higher lateral vibrations were observed in the field with trains featuring gears. This kind of observation is not only reflected by the authors' experience, but is also published in other literature sources (Kita, et al., 2008), (Naldi, et al., 2008).

At a first glance it seems to be plausible that high torque fluctuation also produces high lateral vibrations. This is the reason why only concerns regarding high dynamic torques are raised, when high lateral gear shaft vibrations are also evident. In contrast, cases could also be observed where high torque fluctuations were measured, although only insignificant lateral vibrations of the gear pinion and/or the bull gear could be seen.

Generalized, one can only conclude that if TNFs can be measured in the lateral vibration spectrum, dynamic torque oscillation will with high probability be present in the train. However, low levels of radial vibrations do not necessarily mean low levels of dynamic torque fluctuation in the train components. Having the physical relationships in mind, this issue boils down to the influence of the dynamic oil film stiffness of the gear bearings.

Torsional and lateral vibration records from a centrifugal compressor train with VSI, induction motor and gearbox are available and allow a focus on coupling of lateral and torsional vibration. The train was equipped with a flange-type measuring device with integrated strain gauges at the low speed coupling and lateral vibration probes at motor shaft, both gear shafts and compressor shaft. Figure 3 shows the dynamic torque measurement (blue line) and motor speed (green line) plotted versus time. The inverter was programmed to ramp up and then down again quite slowly with a rate of 0.17 rpm per second through the resonance with the interharmonic excitation frequency with the highest harmonic distortion. In total, the resonance was crossed four times. As another variable, two compressor power conditions were tested by varying the suction pressure to ascertain if the inverter power has an influence on the torsional vibration amplitude. For later comparison of lateral vibration increases, it should be noted that the dynamic torque amplitude in resonance is not constant, but about 15 percent higher with higher inverter power. More interesting in this context is the question of how the gear lateral vibration depends on the different static load conditions.

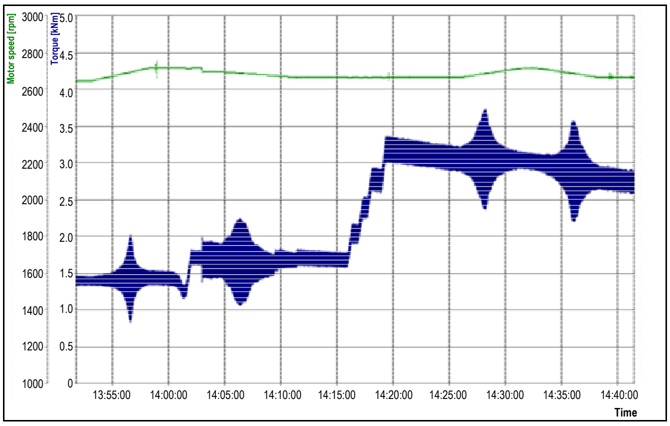

Figure 3. Motor Speed and Dynamic Torque at Low Speed Coupling.

The Waterfall plots of lateral vibration probes of motor and compressor do not present any subsynchronous vibration at first TNF (30 Hz), whereas the plots of gear low speed (Figure 4) and high speed shaft do (Figure 5).

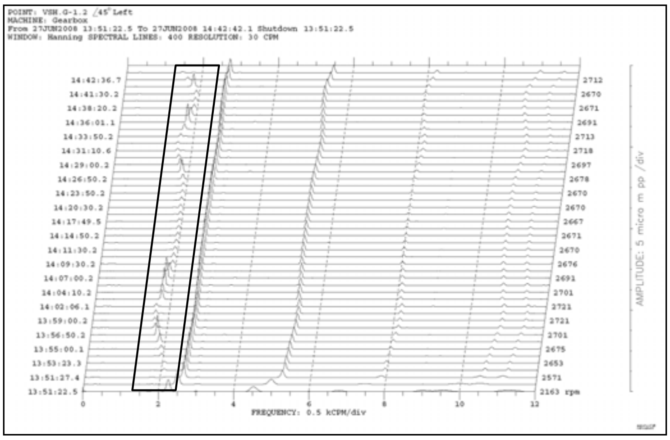

Figure 4. Waterfall Plot: Relative Shaft Vibrations of Low Speed Gear.

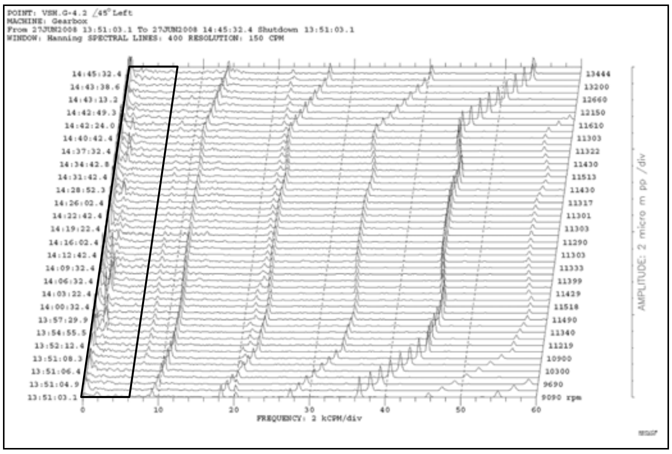

Figure 5. Waterfall Plot: Relative Shaft Vibrations of High Speed Gear.

Although the dynamic torque amplitude is significant, the maximum subsynchronous lateral vibration amplitude increase in resonance is limited to 7 µm at low speed shaft and 8 µm at the high speed shaft for the whole test run. The comparison of the vibration behavior with indirectly varied bearing oil film pressure can be conducted using trend plots too. For orientation, Figure 6 gives the motor speed for a selected time range. Figure 7 and 8 are trend plots of gear box low speed and high speed shaft respectively with overall vibration level and same time range. In these, it is quite clearly recorded that in the resonance crossing with lower power and oil film stiffness, the lateral vibration increase is about two times higher compared to the resonances with higher motor power and oil film stiffness.

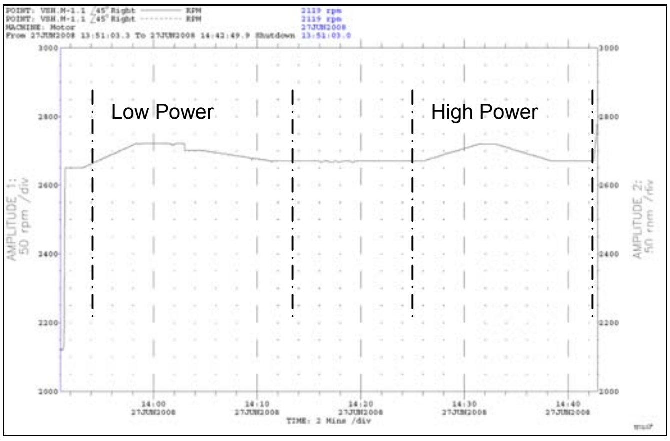

Figure 6. Trend of Motor Speed.

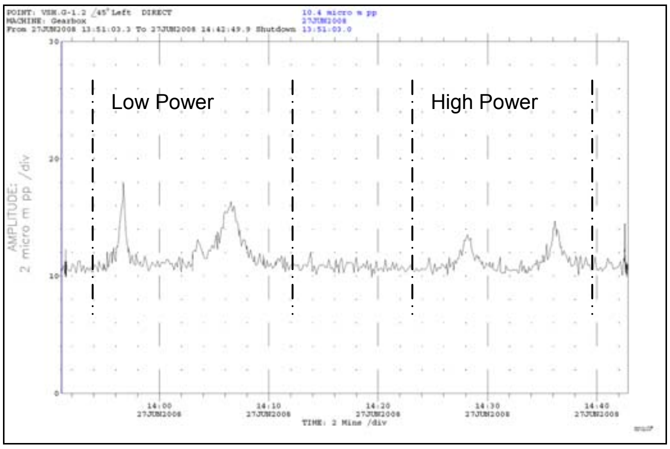

Figure 7. Trend of Relative Shaft Vibration of Low Speed Gear.

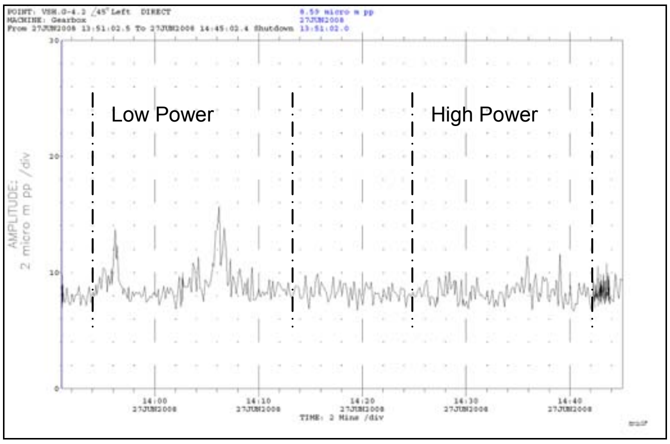

Figure 8. Trend of Relative Shaft Vibration of High Speed Gear.

The vibration records document by doubling power a 50 percent lower lateral vibration increase in resonance. The measurements made for this particular compressor train support the common opinion that high lateral gear vibration is no proof of high torsional vibration.

The lessons learned from this train are that the usual application of lateral vibration probes only, requires high vigilance of the operation personnel to notice torsional excitation in the lateral vibration records. In contrast to direct driven trains, geared trains retain the possibility to detect this excitation even though the lateral vibration can be of very low level. Unfortunately, due to the power-dependency of gear lateral vibration, a generalized transfer function for a torque amplitude estimation from lateral vibration is not yet available.

Excitation Pattern of a VSI

Several case studies of vibration issues related to torque excitation caused by inverter fed motors have already been published. Here, two of our own typical examples of case studies are presented which were used in order to get an in-depth understanding of the operating behaviour of the individual inverter types. This information is needed to realize the new train design concept that is lastly the main focus of this publication.

In the first case, a 1.5 MW 12-pulse VSI inverter feeds the induction motor of a single-shaft radial compressor train with intermediate gear. During run-up with constant acceleration the dynamic torque measurement at low speed coupling recorded amplitudes as shown in Figure 9. As the inverter speed control actively accelerates the train, the harmonic and interharmonic excitation lead to dynamic torque peaks at speeds, where torsional resonance occurs. The blue line is the torque measurement and the green line shows the motor speed, both versus time. Some torque peaks can be observed: one single major amplitude at about 2700 rpm and some peaks quite close together especially in the low speed range.

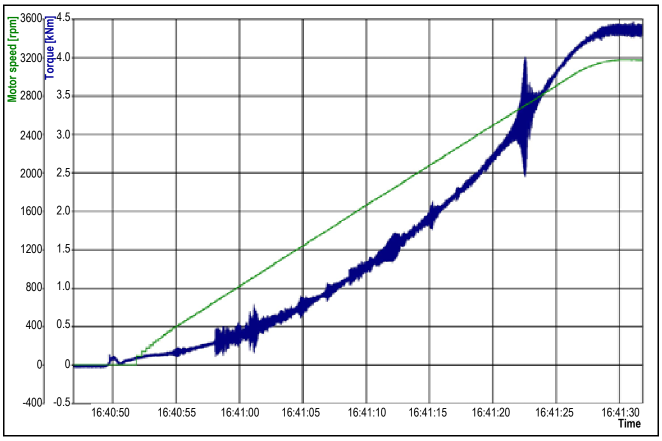

Figure 9. Trend of Motor Speed and Dynamic Torque at Low Speed Coupling.

It is practical to plot these together with the TNFs and excitation frequencies in a Campbell diagram. Figure 10 represents this diagram with the first two TNFs (dashed lines) and torque ripple excitation frequencies (solid blue lines) versus motor operating speed. At motor speeds where TNF and excitation frequency intersect, a resonance is present. The red circles indicate the relevant resonances from Figure 9.

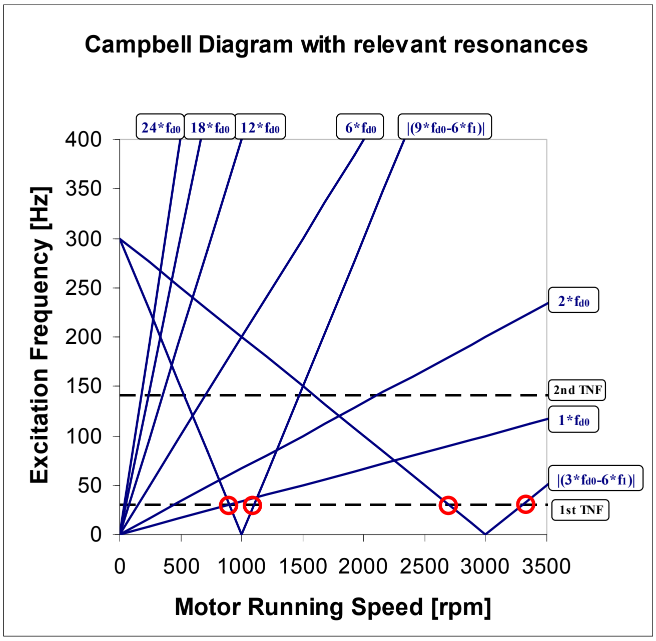

Figure 10. Campbell Diagram with Relevant Resonances.

The diagram reveals that the by far dominating torque peak at 2700 rpm is caused by the $|3*f_{do}-6*f_l|$ interharmonic excitation frequency exciting the 1st TNF. This excitation should therefore, as major resonance, not fall within the operating speed range. Apart from this, secondary amplitudes at about 10 percent peak-to-peak the motor rated torque are observed. The first natural frequency is stimulated by the $|9*f_{do}-6*f_l|$ interharmonic excitation frequency and $1*f_{do}$ at about 900 rpm. Due to its mode shape, the first natural frequency for this train configuration commonly leads to the highest torque amplification for excitation at the motor air gap. Torque amplitudes at other speeds, due to their limited amount, are considered not relevant for the train design. These results leveraged the confidence to use the numerically derived excitation frequencies for this inverter type for the later described train design concept.

Excitation Pattern of a LCI

The following example is related to a 16 MW 12-pulse LCI driven synchronous motor train, connected to a 50 Hz grid, including intermediate gear and a single shaft radial compressor. The train has been designed to operate in a speed range of 1260 rpm to 1890 rpm. In order to get the required information with regard to relevant interharmonic excitation the train was equipped temporarily with strain gauges at the low speed coupling for a dynamic torque measurement. During the measurement program the train was ramped-up slowly with an acceleration rate of 0.2 up to 0.5 rpm per second. The measured dynamic torque (blue line) and the motor speed (green line) are shown versus time in Figure 11. Four main torsional resonances could be observed. These resonances correlate with the fundamental TNF and the expected interharmonic torque excitation of the 6- and 12-order. Based on that result, it means that higher orders of interharmonic excitations are not relevant regarding torsional excitation within this system.

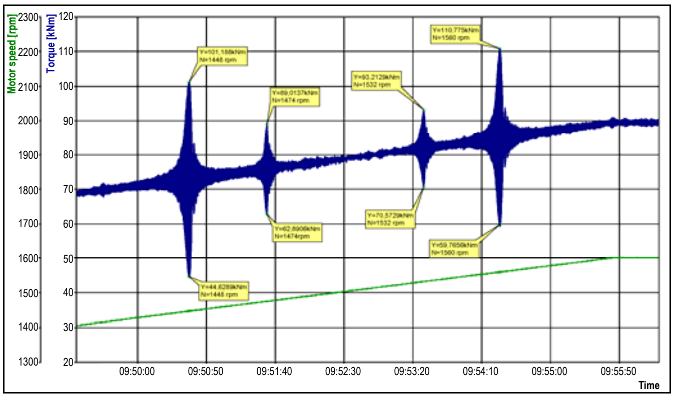

Figure 11. Ramp up of 16 MW LCI Driven Train.

In the next step it is essential to know what happens to the torque amplitude by running in resonance conditions. In order to get an understanding of the behaviour of the vibration system each individually observed resonance was entered for a period of time. For one example of these tests see Figure 12. The observed torque amplitude was generally higher during continuous operation in contrast to crossing the resonance. However, the maximum torque amplitude achieved stationary conditions. This information is of paramount importance with regard to a worst case operating scenario of the fatigue design of the train components running in resonance condition permanently.

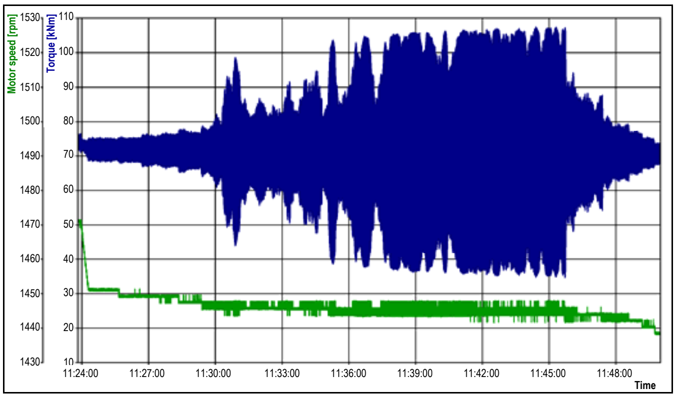

Figure 12. Stationary Operation in Resonance of 16 MW LCI Driven Train.

For this particular case, the measured torque amplitudes at torsional resonance condition were above the expected, based on the state of the art electro-mechanical simulation. Nevertheless, the applied service factors considering the simulation uncertainties ensure that the mechanical train components are capable of withstanding the observed dynamic torque amplitudes permanently. Therefore, the train can be operated without any operational restrictions.

CONVENTIONAL STRATEGIES OF DEALING WITH INTERHARMONIC EXCITATION

If a resonance with an interharmonic excitation is detected and countermeasures are found to be necessary, one can today choose from a wide range of proven alternatives. These can be sorted into one of the following categories:

- Damping increase
- Excitation reduction
- Torque transmitting component fatigue capability increase
- Resonance avoidance

What follows, they are presented and discussed with their inherent advantages and disadvantages to give an overview.

Inverter Control Setup

Although the basic root cause was not exactly the same in all of the case study papers mentioned in the introduction, for all of these cases modifications of the setup of the inverter control or inverter control type change finally reduced the excitation torque amplitudes sufficiently. De la Roche and Howes (2005) describe the case of a motor driven reciprocating compressor with motor shaft failure on one of two trains. Interestingly, by inverter control parameter adjustments it was possible to reproduce the vibration behavior with the second train as well. For the first train, inverter software parameter change was able to increase, as well as satisfyingly decrease the torque oscillations. They and Corcoran and Kocur (2008) as well, mention the speed feedback into the inverter control to be a contributing factor to the overall vibration. It is considered able to amplify the oscillation, when the speed control counteracts the speed fluctuation initiated by the torsional vibration.

When torsional vibration is present, it is adequate to first exhaust the remaining room for improvement in the inverter control for optimization.

Designing Components Robust Enough to Withstand Torque Ripples

According to the applicable paragraphs of API617 7^{th} edition, if excitation mechanisms are known to act in a compressor train, the train responsible party shall conduct a stress analysis. This shall show permissible amplitudes compared to high cycle fatigue capabilities of the train components. It would be therefore satisfying to design the relevant train components in such a way that they can withstand the occurring dynamic torsional load over the train's lifetime. This method can be used during project engineering as well as for commissioned trains. Necessarily, the dynamic torque amplitude must be known from torque measurement records, or it must be sufficiently and accurately predicted by torsional analysis. This dynamic torque oscillation value is then the input for a high cycle fatigue analysis of couplings, shafts and shaft ends, as well as gear teeth, that manufacturers use to check and design their components adequately. It can therefore be ensured that all components are capable of withstanding the dynamic torque amplitudes in resonance for continuous operation. The benefit for the operator is that he can operate the train with the whole speed range specified, although resonances are present only at certain speeds.

Doing the engineering using such a method means using a torsional simulation program. Independently, whether the simulation model is a harmonic forced response simulation or a coupled electrical/mechanical simulation, both deliver the dynamic torque response within the train elements of interest at the detected resonant conditions. These results vitally depend on the damping assumption made in the analysis and the accuracy of the expected dynamic air gap torque excitation magnitude. If uncertainties regarding the above aspects are present, service factors need to be conservatively defined. Torque measurements of suitable trains ideally support the definition. Safe and sound engineering is then ensured. From the authors' point of view, there is still some need for refinement of simulation models. It promises for the future service factors to be reduced to appropriate figures, thus preventing over engineering.

Even if a torsional vibration issue were to be detected not until commissioning and high cycle fatigue were the most critical issues, would it still be possible to qualify the train components using dynamic torque measurement values as input for the component check.

Individual Exclusion Speed Ranges at Resonance Condition

The torsional resonances described before can also be avoided by using individual exclusion ranges. It is practicable to determine resonances of relevance, i.e. with a dynamic torque amplitude probable to exceed a component's high cycle fatigue capability, by torque measurement at the manufacturers test bed facilities or during commissioning. The countermeasure is then to implement the identified resonant motor speeds into the inverter speed control. These speeds plus a separation margin including tolerances and uncertainties are blocked. It is by this, of course, not possible to exclude resonances from the speed range, but it limits the time of train operation within it or close to it to a minimum. In a variable speed performance map, blocked speed ranges can be illustrated as the areas shown in Figure 13. It must be clear that for the plant operator, a blocked speed range, even if it is small, always is a limitation of production flexibility. It should be the aim of the equipment supplier to adequately determine the necessary speed ranges to be blocked, while keeping the above in mind.

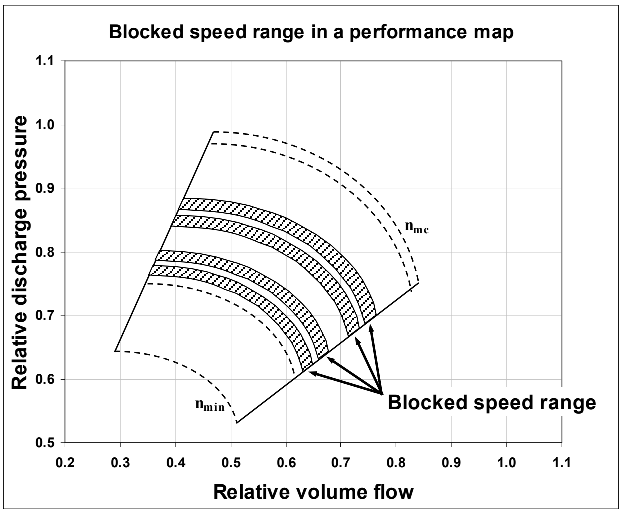

Figure 13. Blocked Speed Ranges in a Performance Map.

At a first glance thinking about avoiding torsional resonances within the operating speed range by using individual exclusion ranges seems to be a simple and effective solution. However, a lot of parameters and uncertainties have to be born in mind by setting the real problem solving exclusion speed ranges.

For one single train installed, these include the accuracy of the dynamic torque measurement itself, changing of material properties over life cycle and/ or due to temperature and local grid frequency variation, just to name few. The last parameter can, especially in countries with high grid frequency fluctuation, become the decisive parameter for the blocked speed range. Inverters that have excitation frequencies with a subtraction term of $|C1*f_{d0}-C2*f_l|$, where f_{d0} is the VFD output frequency and f_l the now variable grid frequency, the motor speed, where a resonance occurs, also becomes a variable. As a consequence, blocked speed ranges would necessarily increase, unless the grid frequency was considered in a speed control algorithm as an additional parameter, which is possible. Practically spoken, the motor speed is adjusted, when the grid frequency fluctuation causes a resonance. Of course, motor speed and grid frequency must be continuously measured.

In case of multiple identical trains installed, efforts increase. If a torsional measurement is going to be done for one train only, it should be critically discussed, how material property uncertainties between the train components are reflected in the determination of the blocked speed range(s).

Damping in Control Loop

Active damping of the load or the process using the VFD is standard in some industries. Also for compressor strings there are approaches to use the VFD for active damping (Naldi, et al., 2008). The challenge for the compressor trains is the low frequency of the switching devices and the limits of the specific topologies. Another challenge is the identification of the train characteristic. The control loop needs as input parameter the actual status of the train. Therefore, additional sensors are most likely needed.

The conventional multi parameter control system of an inverter is extended by an additional damping control loop. This feature should be considered in detail during engineering and also during the commissioning process in order to achieve a reliable operation. As long as additional sensors need to be installed and are crucial for the functionality, redundancy is essential.

Damper Coupling using Elastomeric Elements

Occurring torque response in resonance condition is mainly determined by the magnitude of the torque excitation and the mechanical amplification. Therefore, torsional damping is a significant influence parameter for the overall system. One of the common methods to reduce the torsional vibration amplitude within a resonance is to increase system damping. Couplings with elastomeric elements are often used in such cases. This coupling consists of a steel design combined with integrated elastomeric elements. The main task of these elastomeric elements is to absorb torsional vibration energy by compressed deformation of the elements in contrast to solid steel couplings. It is of utmost importance that this coupling be located close to the node of the mode shape of the fundamental TNF to be most effective in increasing system damping. The significantly reduced torsional stiffness of such couplings helps to achieve that. In addition the non-linear behavior of the torsional stiffness of the elastomeric elements has to be considered in the torsional analysis. Stiffness characteristics vary with transmitted power and operating temperature. During operation, the damper blocks absorb vibration energy. As a consequence of the material properties, the rubber elements degrade over time due to heating up and environmental factors. The main disadvantage of this kind of coupling is usually increased maintenance for reliable operation, in contrast to solid steel couplings. Due to this fact elastomeric couplings are principally not allowed per specification of the operators for some applications.

As long as a train design concept does not avoid torsional interharmonic excitations caused by frequency converter within the operating speed range, the train components have to withstand torsional resonance amplitudes at certain speeds permanently. The authors are convinced that elastomeric elements are able to dissipate vibration energy what finally means dampening the torsional system. Nevertheless, the damper elements also have a limited capability in absorbing energy. Therefore, using such kind of coupling may help to limit the torque amplitude during crossing resonances for a short period of time. However, for VSDS driven trains it may happen that the train is running in condition of a torsional resonance for a longer period. Unfortunately, the occurring magnitude of the torsional resonance amplitude can be excessively high. Therefore, running in resonance condition for a longer period may overload the elastomeric elements. Figure 14 and 15 shows an overloaded elastomeric coupling and one of its elastomeric elements. Coupling failure like that in a VSDS driven train are, in most cases, not necessarily caused by a poor coupling capability, but are quite often caused by the train behaviour itself (Corcoran and Kocur, 2008).

It can be concluded that it is an important task to minimize the dynamic torque oscillation by designing the VSDS train in a proper way in order to achieve a reliable compressor operation.

Figure 14. Overloaded Elastomeric Coupling.

Figure 15. Overloaded Elastomeric Damper Element.

NEW TRAIN DESIGN CONCEPT TO AVOID INTERHARMONICS IN OPERATING SPEED RANGE

Basis of the New Train Design Concept

In principle, the basis of the new train design concept to avoid interharmonics within the operating speed range is an in-depth understanding of inverter behavior, motor design and finally, the various compressor train configurations and their corresponding torsional behavior. First of all, it is essential to work out the details of the mechanically relevant torsional excitations caused by the individual inverter types. This task has been done on a numerical basis by simulating the electrical and mechanical system. Due to the fact that the occurring torque amplitude in a state of resonance condition is connected with tolerances, although using state-of-the-art coupled electrical and mechanical simulation models, correlation with dynamic torque measurements are of vital importance. This is to separate the relevant excitation frequencies from the insignificant. In the next step, possible motor designs of induction motors and/or synchronous motors with regard to different numbers of pole pairs have to be considered. Lastly, the engineering process of the motor manufacturer itself should enable the flexibility to adapt the required operating speed range for the individual application in order to avoid the relevant torsional excitations within the operating speed range of the driver. The new train design concept will be presented on an exemplified basis for a current source inverter in the following chapter.

Introduction of the Design Concept on Basis of a Current Source Inverter (LCI)

To explain the developed design concept in order to avoid interharmonic excitations caused by the inverter within the operating speed range, a compressor train driven by a synchronous motor fed by a 12-pulse LCI is considered. On the basis of numerical investigations, the 6-pulse and 12-pulse current source inverter generates the following interharmonic excitations:

$$|6*(f_{do}-f_l)|$$
$$|12*(f_{do}-f_l)|$$
$$|18*(f_{do}-f_l)|$$
$$|24*(f_{do}-f_l)|$$

The main difference in characteristics for a 6- and 12-pulse design is related to torsional excitation magnitude only. Typically the higher the number of pulses the lower the excitation magnitude.

For these types of inverter 6-, 12-, 18-, 24- and 36-pulse designs are currently being used in the field of compressor train applications. In order to consider all these inverter types, the mechanical relevant harmonic and interharmonic excitations have to be identified out of the known multitudinous theoretical excitation potentials.

Identification of the Relevant Interharmonic Excitations of a 12-pulse LCI

By numerical investigations of various types of inverter, numerous harmonic and interharmonic excitations can be identified. In order to identify the VSDS generated mechanical relevant excitations, dynamic torque measurements are indispensable. As described in chapter *Excitation Pattern of a LCI,* torsional resonance conditions of a 12-pulse LCI driven train could be observed. The measured resonances and its corresponding motor speeds can be projected into a Campbell diagram in order to identify the relevant excitations as can be seen in Figure 16. Based on these results, it is obvious that for this particular inverter only 6- and 12-order interharmonics are of relevance with regard to mechanical excitation of the torsional system. Therefore, only these excitations have to be considered in the train design concept.

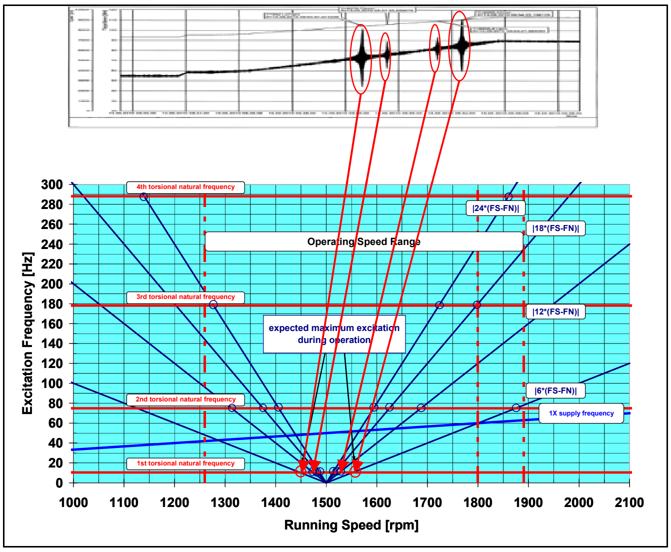

Figure 16. 12-pulse Current Source Inverter (LCI).

In the next developing step of the strategy, the mechanically relevant harmonic and interharmonic excitations can be presented in a Campbell diagram as seen in Figure 17.

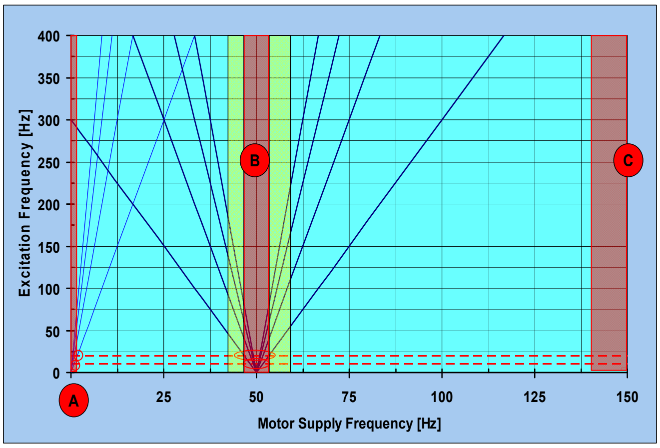

Figure 17. Campbell Diagram of a 12-pulse LCI Driven Compression Train.

The excitation frequency is presented on the vertical axis whereas the motor supply frequency is shown on the horizontal axis of this diagram. The incorporated harmonic excitation lines are of 12, 18, 24 and 36 times of motor supply frequency. The presented characteristic of the interharmonic excitation lines are as has already been described. Typically, all these interharmonics intersect the horizontal axis at grid frequency (for this example 50 Hz). The frequencies are considered as absolute values. therefore the excitation lines are given as V-lines in a Campbell diagram. The intersections of the excitation lines of the above mentioned interharmonics and the TNF range are all in immediate vicinity. All excitation lines are incorporated into the diagram, although we have learned that only the 6- and 12-order interharmonics are of mechanical relevance. However, due to the fact that a separation margin of 10 percent as specified in the API617 has to be considered, the higher order interharmonic do not effect the required exclusion range.

In a next step the typical fundamental TNF range has to be identified based on experience. For this particular example a range of 10 to 20 Hz for the fundamental TNF has been considered. Most train configurations within the authors references are within this range and/or could be moved into this range by tuning the coupling stiffness, given that some kind of mechanical necessity is present. For conventional train configurations like direct driven trains (motor-compressor) and/or trains including an intermediate gear (motor-gear-compressor), the first TNF can be significantly excited at motor main mass due to the corresponding torsional mode shape. It has to be pointed out that for train configurations considering a drive through motor, also the second torsional critical speed is of vital importance.

The intersections of the mechanically relevant excitations and the range of TNFs can easily be identified. Therefore, exclusion ranges can be set at these ranges of motor supply frequencies. To make it obvious, exclusion ranges are generally marked in red in the following figures. The first exclusion range (A) at lowest supply frequencies is related to the harmonic excitations. In order to avoid resonances with the interharmonic excitations the second exclusion range (B) has to be set. These excitations are located in the immediate vicinity of the grid frequency. In compliance with the requirements of API617, an additional separation margin of 10 percent is considered for the exclusion ranges (A) and (B). Whereas the first exclusion range is typically not relevant for continuous turbocompressor operation, the second exclusion range (B) could be in conflict with the operating speed range. Based on this graphic presentation it becomes obvious that the interharmonic excitations are of technically higher importance. The third range (C), also marked in red, is determined by the individual maximum supply frequency of this particular application which can be generated in the inverter. That means, it describes the current frequency limit of the inverter for standard applications.

Based on these considerations, the supply frequency ranges which should be avoided are separated. In addition, the ranges of motor supply frequency without resonances caused by inverter operation are identified. Therefore, these are the supply frequency ranges which should be used for designing compressor trains.

In order to design compressor trains, it is at this stage necessary to transfer the driver supply frequencies into a motor speed related diagram. Transferring electrical supply frequencies into mechanical running speeds of a motor means that the number of pole pairs of the motor has to be considered. The impact with regard to allowable or excluded motor speed ranges is clearly presented by Figure 18.

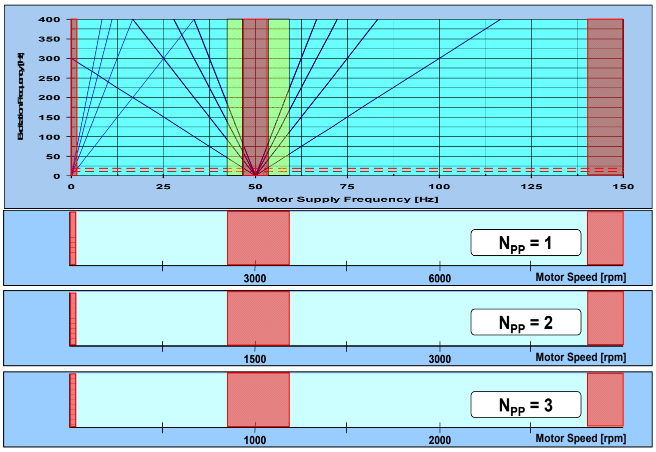

Figure 18. Exclusion Ranges Transferred into Motor Speeds.

The well known relation between operating speed and inverter supply frequency and number of pole pairs of a motor is defined as follows:

$$n_{op} = (f_{do} / N_{PP}) * 60 \qquad (2)$$

The determined exclusion ranges are transferred into a motor speed related diagram for the chiefly used number of pole pairs 1, 2 and 3. Focused on the most relevant exclusion range of the resonance conditions caused by interharmonics, it becomes obvious that the center of the most important exclusion range shifts from 3000 rpm to 1500 rpm and finally to 1000 rpm for a number of pole pairs of 1, 2 or 3, accordingly. In order to consider this information during the proposal and/or execution phase of a project, the information has to be transferred into a motor speed related bar diagram as can be seen in Figure 19. For achieving a reliable engineering process, a more or less simple illustration of the complex background information is essential.

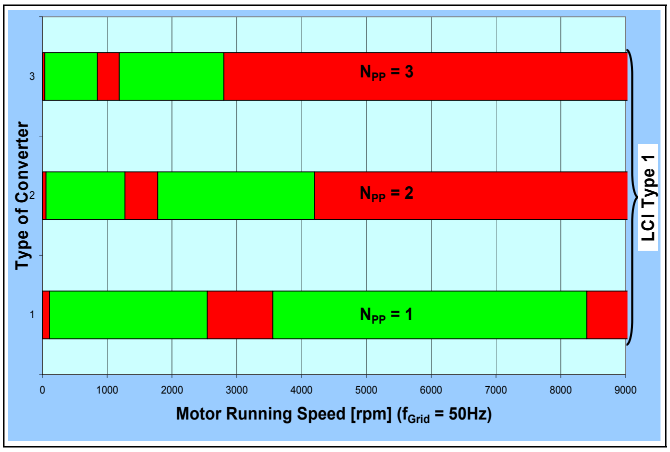

Figure 19. Bar Diagram of Motor Speeds for 12-pulse LCI.

Figure 19 collects all the relevant information in order to avoid torsional resonances caused by a 12-pulse LCI operation. The allowable speed ranges are marked in green, whereas motor speeds which should be excluded are marked in red. It goes without saying that additional separation margins have to be considered to achieve a trouble free operation with regard to torsional excitation mechanisms, for example running speed related excitations as specified in API617. Based on this illustration, it becomes obvious that theoretically for all common used driver speed ranges solutions can be realized.

Consideration of Direct Driven Trains

A typical train configuration of a direct motor driven turbo compressor is shown in Figure 20. For this particular example a large synchronous motor and a barrel type compressor are coupled via a diaphragm coupling.

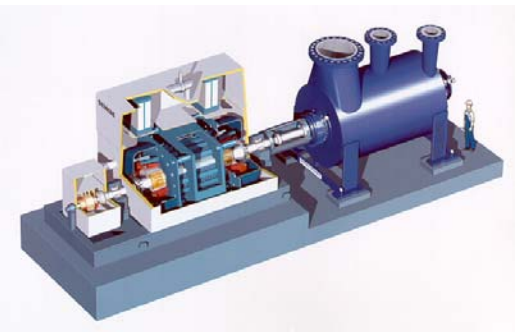

Figure 20. Typical Direct Driven Turbocompressor Train.

For the design process of such compressors, the typical application range with regard to compressor speed has to be considered. For larger direct driven centrifugal compressors the minimum operating speed could be at about 2000 rpm whereas for smaller direct driven compressors the maximum operating speed can be significantly higher, up to 8000 rpm and even higher for special applications. For direct driven trains, speed ranges between approx. 2000 and 4000 rpm can be realized by a load commutated inverter in combination with a driver considering a number of pole pairs of 2 as illustrated in Figure 21. Such a train design would be able to avoid main torsional resonances within operating speed range. Currently a 4-pole motor (number of pole pairs of 2) operated at frequencies higher than 60 Hz represents an uncommon design in the turbomachinery industry. Nevertheless, it offers the opportunity of a torsional resonance free operation by using a reliable standard motor design. Whereas for applications with operating speeds above 3500 rpm a 2-pole motor (number of pole pairs of 1) should be used. A motor with a number of pole pairs of 3 offers the design opportunity to realize driver speeds between 1200 rpm and about 2800 rpm. Lastly, it should be pointed out that this particular example is related to a grid frequency of 50 Hz.

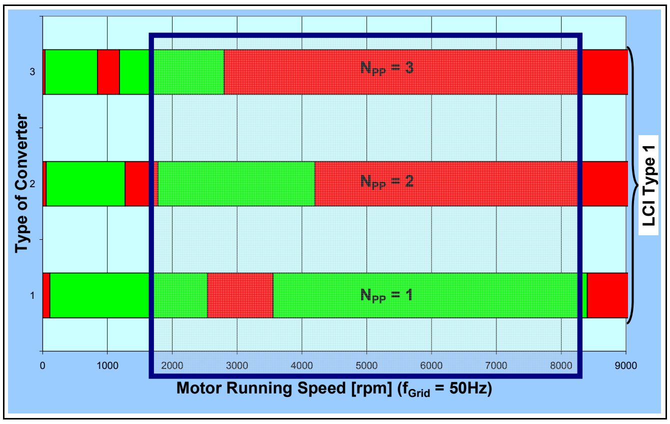

Figure 21. Typical Speed Range of Direct Motor Driven Compressor Trains.

Consideration of Compressor Trains including Intermediate Gear

Figure 22 shows a typical motor driven compressor train including an intermediate gear used as speed increaser. That means that for this kind of application the speed range of the driver is normally lower in contrast to direct driven trains.

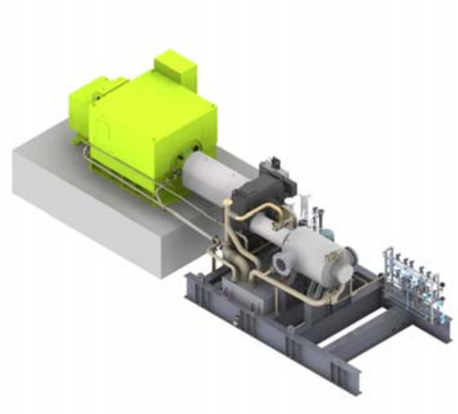

Figure 22. Typical Motor Driven Compressor Train including Intermediate Gear.

A typical application range of motor driven compressor trains with an intermediate gear is between 600 rpm and up to 3600 rpm, which can also be seen in Figure 23 for 50 Hz grid application. Whereas for direct driven trains the motor and compressor speed has to be identical, for trains including an intermediate gear the motor speed can be chosen almost independently from compressors optimal design speed. The main advantage for this train configuration is to tune the gear ratio in order to adapt the driver and the compressor speed. Accordingly, it is possible to use a motor speed range at lower or higher speeds in order to find a torsional resonance free train design solution for the particular application. However, for compression trains including an intermediate gear, the minimum operating speed and the fundamental TNF could run into a design conflict with fulfilling a separation margin of 10 percent related to running speed excitations as specified in API617. This essential design criterion has to be considered by selecting an adequate minimum operating speed which allows fulfilling the requirement. Otherwise, it can be helpful to tune the torsional stiffness of the low speed coupling in order to fulfill demand of separation margin.

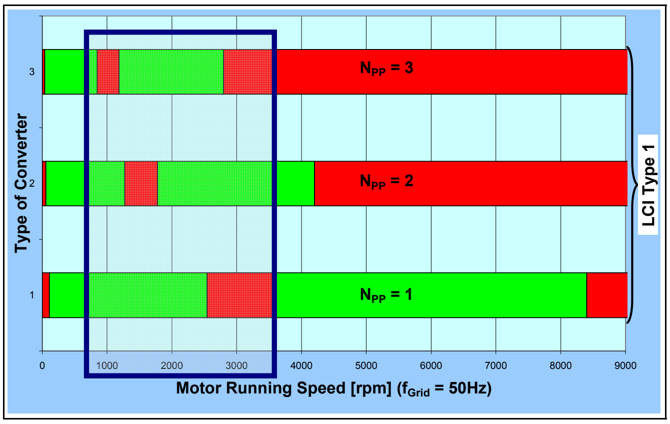

Figure 23. Typical Speed Range of Motor Driven Compressor Trains including Intermediate Gears.

As shown in Figure 23, speed ranges of about 600 rpm up to 2500 rpm can be realized by using a 2-pole motor design, whereas with a 4-pole motor design, a resonance free operation can be achieved for a driver speed range between 1800 rpm and 3600 rpm or even higher. A motor design considering a number of pole pairs of 3 can be used for applications with a driver speed range from about 1200 rpm up to 2800 rpm. Eventually, the optimal solution has to be determined case by case.

Transferring the New Train Design Concept to Relevant Inverter and Motor Types

The described designing strategy in order to avoid torsional resonances excited by inverter behavior can easily be adapted to all the other in-depth understood inverter applications. Figure 10 shows the behavior of the 12-pulse voltage source inverter. Several excitations are able to produce a torsional resonance condition but only few of them are able to produce higher torsional peak-to-peak amplitudes of more than 10 percent of the motor rated torque. Based on the dynamic torque measurements of a real train with a 12-pulse voltage source inverter under realistic load and electrical grid conditions, the mechanically relevant torsional excitation frequencies of the voltage source inverter can be indentified. The torsional resonances with a torque amplitude above 10 percent of the rated motor torque are defined as mechanically relevant and should be therefore considered in the new train design concept. On the basis of the simulation results, the measured main resonances can be correlated with interharmonic excitation frequencies. It has to be pointed out that the inverter behavior is always product and manufacturer-specific. Therefore, the relevant excitations have to be identified individually.

Separated into different types of inverters and their corresponding behavior, considering different number of pole pairs in combination with a realistic estimate of relevant TNFs, an inverter-motor-selection bar diagram can be prepared as can be seen in Figure 24. It is essential to prepare such a bar diagram separately for 50 and 60 Hz electrical grid frequencies.

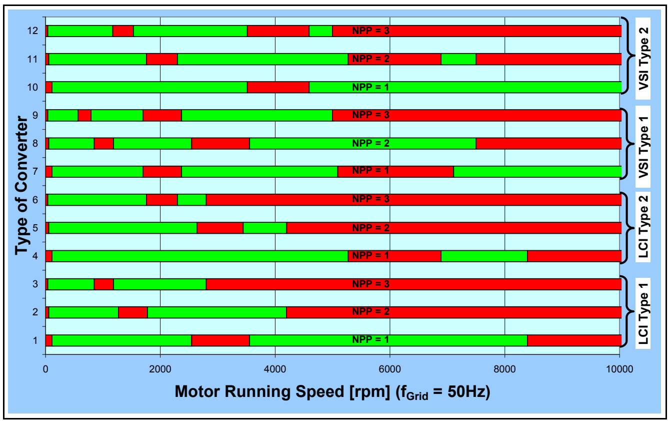

Figure 24. Bar Diagram of Motor Running Speed for VSDS Motor-Inverter-Selection-Tool.

For proposal and also for order execution phase, avoiding torsional resonances have to be considered in an early stage of the process. In order to achieve that target, a Motor-Inverter-Selection-Tool has been prepared and has already been integrated in the engineering process (Gallelli and Hütten, 2009).

Impact of Design Strategy

The design strategy of a specific project has to cover the technical and commercial impact of the VFD and motor selection. It is beneficial to optimize the overall compressor string instead of optimizing single components. Therefore, close cooperation between mechanical and electrical engineers is required to end up with the best fit for the individual train design. The above mentioned bar diagram for motor-inverter-selection leads to an operating speed range without the necessity of blending out frequencies, which is important for the overall process performance of the plant. In some cases, a slightly more expensive solution is required. In this case, a mutual agreement between process owner, EPC, and the compressor string supplier is required.

SUMMARY

- Within the turbomachinery industry, several cases of torsional resonance issues caused by inverter fed motor compressor trains have been published. The unfavorable combination of motor, inverter and speed range can always lead to train designs with resonances.
- The excited torsional vibration may cause fatigue problems of torque transmitting elements of the train. In addition, the torsional excitation may also lead to higher radial shaft vibrations and/or gear clattering of pinion and bull gear shaft in intermediate gears.
- Due to the coupled movement in torsional and lateral direction, a higher lateral gear vibration caused by torsional excitation is quite obvious behaviour. Nevertheless, also relatively low radial shaft vibrations of a few microns were observed in combination with measured dynamic torques of up to 80 percent of the motor rated torque.
- Simulation investigations of fully coupled electro-mechanical models considering the existing electrical grid, the inverter, the electrical motor and the mechanical model of the train including mass moments of inertias, the torsional stiffnesses and the mechanical damping have been carried out. The numerical investigation is generally able to identify the relevant interharmonic excitations. The TNF and the corresponding motor speed can be determined with adequate accuracy.
- Based on correlating simulated and experimental results there exist further uncertainties in the entire electro-mechanical model. Due to these, the occurring torsional amplitude in resonance condition cannot be predicted with sufficient accuracy. Appropriate service factors are necessary to compensate for this, ensuring safe and reliable design.
- The pros and cons of the various well known strategies of dealing with interharmonic excitations have been presented within this publication.
  a) Improvement of inverter control setup.
  b) Designing components more robustly in order to withstand dynamic torque permanently.
  c) Using individual exclusion ranges at resonance condition.
  d) Implementing "torsional damping" in the control loop of the inverter.
  e) Using damper coupling with elastomeric elements
- Additionally, this paper presents a new train design concept in order to avoid the mechanically relevant torsional resonances excited by the inverter within the selected operating speed range.
- The new train design strategy considers the individual behaviour of the inverters. In order to identify the mechanically relevant interharmonic excitations, correlation of analytical and experimental results of the torsional train behaviour are essential.

CONCLUSIONS

- Under consideration of the relevant individual behaviour of the inverters in combination with possible designs of the motors, number of pole pairs main torsional resonances can be avoided in operating speed range.
- The inverter behaviour is product and manufacturer-specific. Therefore, the relevant excitations have to be identified individually.
- One additional important aspect is that the engineering process of the motor allows realizing motor designs with individually required speed ranges.
- For direct driven trains, the driver speed has to be identical to the compressor speed. Nevertheless, for typical turbo compressor applications between 2000 rpm and 8000 rpm a solution can designed with motors having a number of pole pairs of 1, 2 or 3.
- For trains including a intermediate gear, there usually exists more potential design options due to the fact that the driver and the compressor speed can be selected independently from each other. In order to find a torsional resonance free solution, the driver speed can be adapted to compressor speed by the gear ratio.
- The presented pragmatic new train design concept offers reliable mechanical solutions in order to avoid interharmonic excitation within the motor speed range. However, the commercial aspect has to be evaluated individually.
- This concept does <u>not</u> need any individual adjustments during testing and/or commissioning and does <u>not</u> lead to operational restrictions of the compressor train.
- Ultimately, for designing VSDS driven turbo compressor trains, a close collaboration between the manufacturer of the driver and compressor is of vital importance.

NOMENCLATURE

AF	=	Amplification factor	
AC	=	Alternating current	
C_1, C_2, C_3	=	Constants associated with a particular VFD design	
DC	=	Direct current	
EPC	=	Engineering and procurement contractor	
f_{do}	=	VFD output frequency	[Hz]
f_l	=	Electrical line frequency	[Hz]
f_{exc-h}	=	Frequency of harmonic torque excitation	[Hz]
f_{exc-i}	=	Frequency of interharmonic torque excitation	[Hz]
f_{Ti}	=	Relevant torsional natural frequency	[Hz]
GHG	=	Green house gas	
i	=	Gear ratio	[-]
J_M	=	Mass moment of inertia of the motor	
J_C	=	Mass moment of inertia of the compressor	
LCI	=	Load commutated inverter	
LNG	=	Liquefied natural gas	

n_{min}	=	Minimum operating speed	[rpm]
n_{mc}	=	Maximum continuous speed	[rpm]
n_{op}	=	Motor operating speed	[rpm]
n_{syn}	=	Synchronous speed	[rpm]
N_{PP}	=	Number of pole pairs	[-]
THD	=	Total harmonic distortion	
TNF	=	Torsional natural frequency	
TNFs	=	Torsional natural frequencies	
VFD	=	Variable frequency drive	
VSDS	=	Variable speed drive system	
VSI	=	Voltage source inverter	

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ACKNOWLEDGEMENTS

The authors would like to thank the management of Siemens AG for authorizing the publication of this study. Additionally, we would like to thank Kopflex for permission for using photos of coupling failures related to the published matter.