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CONSIDERATIONS IN THE DESIGN OF VFD MOTOR-DRIVEN COMPRESSORS

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ABSTRACT

The various drivers for process compressor trains are designed in accordance with the pertinent process requirements. In the case of variable speed applications for process compressors, Steam turbines or Gas turbines are typically applied as a standard practice.

However, improvements in inverter technology have made Variable Frequency Drive (VFD) motors a viable solution for variable speed applications. Motor drivers are well suited for constant speed applications, but VFD motor drive systems represent a relatively new technology for process compressor applications, so the technical differences and associated risks need to be well understood by Users, EPC contractors and Turbomachinery OEM's.

Any misunderstanding or miscommunication among any of these parties could lead to a worst case scenario of machine damage and unexpected plant shutdowns.
This paper addresses the basics of VFD technology from a practical as well as technical point of view with the overall aim of helping to pre-empt worst case scenarios.

INTRODUCTION

The speed of conventional motor drivers is specified by the line frequency and motor type, such as induction or synchronous. When the line frequency is constant, the speed of a conventional motor will of course be constant. To meet the process requirement in a constant speed mode, the compressor is usually operated by suction throttle control, or discharge bypass control, or inlet guide vane control. But these control modes may be at a disadvantage compared with variable speed controls in terms of operating range and/or power consumption. For this reason, i.e., to improve performance, VFD motor systems have recently gained ground as drivers for process compressor applications.

VFD systems rectify AC (50Hz or 60Hz) to DC and then invert DC to variable frequency AC. AC-DC-AC conversion causes distortion of the electrical current wave as a logical phenomenon, and then air-gap torque pulsation is generated as a result. When this torque pulsation frequency is coincident with the natural frequency of torsional vibration, torsional resonance vibration is caused, thus exposing the weakest parts, such as couplings, to damage and the worst case scenario alluded to above.

To avoid such an occurrence, it is incumbent upon all parties concerned, i.e., Users, EPC contractors and OEM's to fully understand VFD technology from both practical and technical points of view. This will enable OEM'S to properly incorporate VFD's into the design of their compressor trains. To promote this understanding among all parties, the basic phenomenon of torsional resonance vibration caused by VFD motor drives is explained from a practical and technical point of view, using theoretical analysis, actual measurement results and Lessons Learned.

VFD SYSTEM

The VFD system is an adjustable motor speed system to control an AC motor by varying input frequency and is typically described as cited in Figure 1.

① Line AC input converts to DC.
② DC inverts to modulated quasi-sinusoidal AC output by switching element.
③ Motor is operated as per modulated AC output frequency.

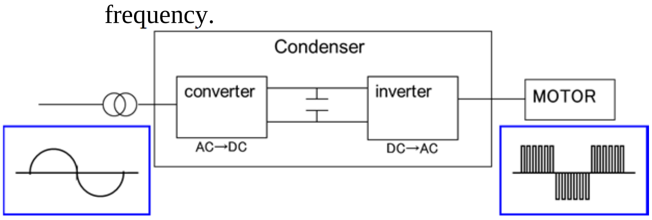


Figure 1; Typical VFD system schematic

AC-DC-AC conversion is achieved by electrical switching action resulting in the generation of some ripple / distortion / pulsation on the AC output wave. This is a very fundamental and unavoidable phenomenon of AC-DC-AC conversion. This characteristic represents the major difference between conventional motor drivers and VFD motor drivers.

There are two main types of VFD systems, and the magnitude of the caused ripple frequency on AC output is different for each type and make. In the main, LCI and VSI (with PWM) are applied to process compressor applications.

LCI = Load Commutated Inverter
VSI = Voltage Source Inverter
PWM = Pulse Width Modulation

Ripple type of VFD system output is divided into three categories and called as follows.

① Integer harmonics
- ✓ Proportional to operating frequency
- ✓ Theoretical aspect

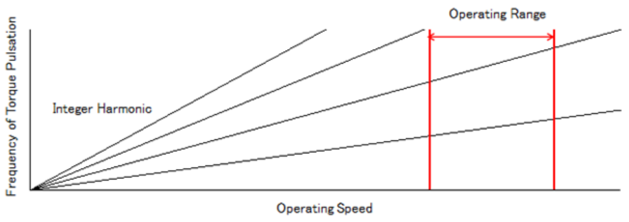


Figure 2; Typical example of Integer harmonics (LCI)

$Fr = n \times Fm$ (example for LCI)

Fr	= Frequency of torque ripple	(Hz)
n	= Integer multiple of 6	(-)
Fm	= Operating frequency	(Hz)

② Non-integer harmonics
- ✓ Interaction of line and operating frequency
- ✓ Theoretical aspect

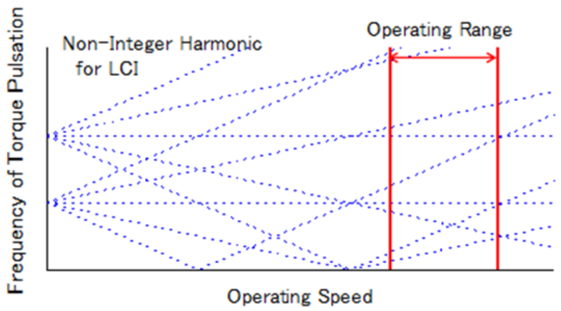


Figure 3; Typical example of Non-integer harmonics (LCI)

$Fr = | n \times Fm \pm k \times Fn |$ (example for LCI)

k	= Integer multiple of 6	(-)
Fn	= Line frequency	(Hz)

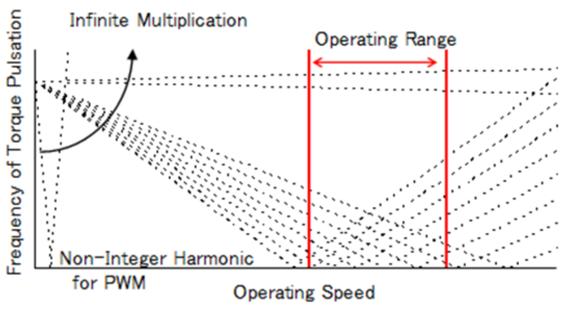


Figure 4; Typical example of Non-integer harmonics (PWM)

$| Fc \pm m \times Fm |$ (example for LCI)

Fc	= Carrier frequency	(Hz)
m	= Natural number	(-)

③ Asynchronous harmonics
- ✓ Complicated interaction of control loop and parameter setting
- ✓ Broadband aspect like the white noise figure

TORSIONAL RESONANCE VIBRATION

Torsional resonance vibration occurs when exiting frequency, such as torque ripple frequency caused by a VFD system, is coincident with torsional natural frequency of the compressor train. As we know, torsional damping is very small and torsional measurement devices are not usually installed, in this scenario, if torsional resonance vibration suddenly occurs the weakest part of the compressor train can be damaged without any notice. Figure 5 shows the typical mechanism of torsional resonance vibration caused by the VFD system.

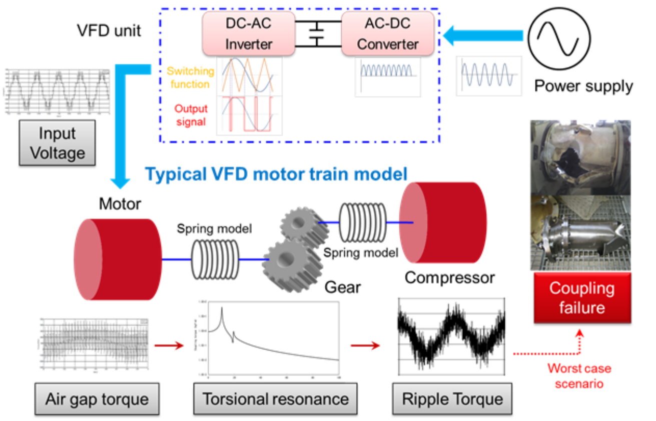


Figure 5; Typical mechanism of torsional resonance vibration

① Smooth AC sine wave is input to the VFD system.
② AC-DC-AC conversion generates distorted AC sine wave and it is input to the motor as drive frequency.
③ Accordingly, the motor air gap torque has a ripple component.
④ This superimposed ripple frequency accidentally coincides with the natural frequency of the compressor train.
⑤ As a result, torsional resonance vibration is caused.
⑥ When this resonant torque exceeds the yield strength of the weakest part, it is damaged and the compressor train goes into an emergency trip.
⑦ The compressor train cannot resume operation until the damaged part is repaired or replaced.
⑧ During this period, the plant incurs partial or total production downtime.

Torsional resonance needs to be averted during the engineering stage. For conventional compressor trains without VFD motor drives, this can be achieved by adjusting the coupling design. In the case of VFD motor driven compressors, torsional resonance can be minimized, but practically speaking, it is virtually impossible to completely eliminate same due to the presence of so much exiting frequency. Figure 6 illustrates an example of torsional resonance in LCI applications. Lower order resonance can be averted with an adjustment in coupling design. But higher order resonance cannot be fully avoided, but from a practical viewpoint, it may be considered acceptable due to its negligible impact.

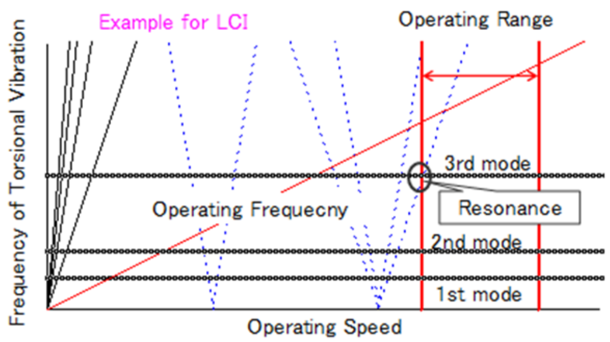


Figure 6; Example of torsional resonance in LCI applications

Figure 7 shows an example of torsional resonance in PWM applications. The torsional resonance cannot be avoided due to so much excitation, so theoretical analysis needs to be carried out to check its impact.

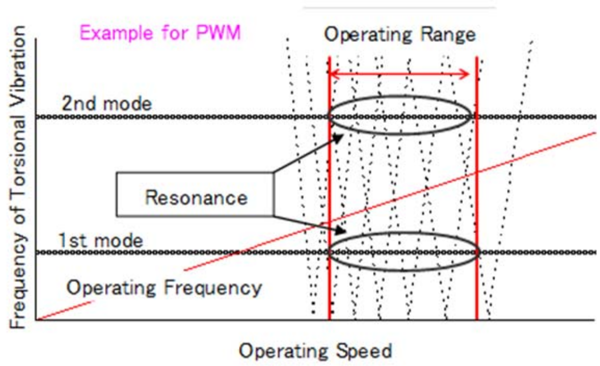


Figure 7; Example of torsional resonance in PWM applications

Figure 8 depicts an example of the mechanism for asynchronous harmonics. Integer harmonic and Non-integer harmonics can be obtained from the VFD motor manufacturer because they are simply checked with the theoretical design data of the VFD system. On the other hand, asynchronous harmonics are generated by the complex control loop, its parameter setting and the interaction to the characteristics of the entire compressor train. This is deeply related to the VFD motor vendor's proprietary know-how which usually cannot be disclosed to others. If precise estimation of asynchronous harmonics data is needed, an in-depth detailed analysis has to be carried out taking into account total train characteristics with verification testing. Requiring this degree of detailed work is not practical, so the VFD motor manufacturers will prefer to provide the client with general asynchronous harmonics data based on their experience. This data is not precise, but it is useful for the engineering of compressor trains.

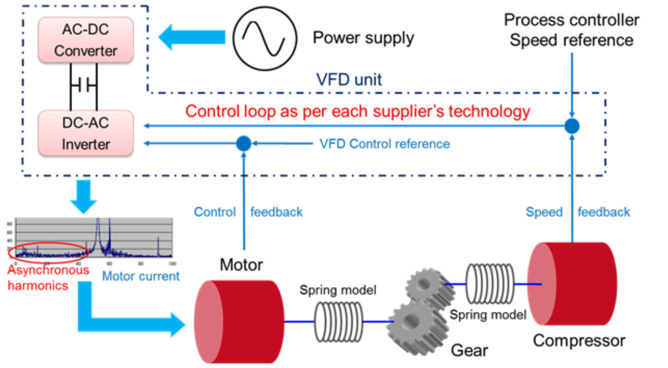


Figure 8; Example of mechanism for asynchronous harmonics

In cases where it is not possible to achieve suitable separation margin for torsional resonance, one needs to evaluate the effects of torsional resonance. Torsional response analysis is an effective means of evaluation. Figure 9 shows an example of torsional frequency response analysis results.

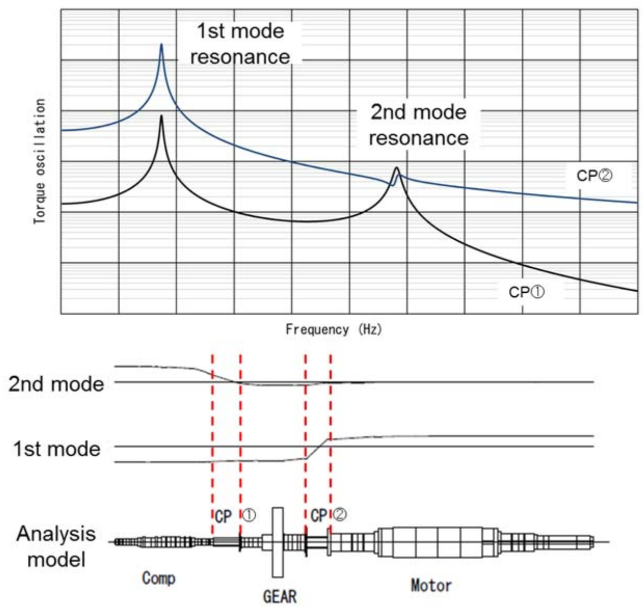


Figure 9; Example of torsional frequency response analysis

A single compressor is connected via speed increasing gear to the VFD motor. Two couplings are installed between the compressor and gear, gear and VFD motor. The upper graph illustrates the torsional frequency response analysis result. The X-axis shows frequency and the Y-axis shows torque oscillation at the coupling. The lower diagram shows the analysis model and torsional vibration mode at the 1st and 2nd resonance frequencies. The coupling is in the most oscillated position for both resonance frequencies. This is a very common configuration for all compressor trains with the coupling usually being the most oscillated part, i.e., the weakest for torsional resonance. Therefore, coupling design is one of the key components for torsional vibration.

In this example, a simplified quantitative evaluation is presented to facilitate a better understanding.

Damping ratio 1% is assumed as ζ =0.01.
Amplification factor of torsional vibration AF=1/(2xζ)=50.
Input torque ripple is assumed as 1% of motor torque.
Oscillated torque at resonant point Tosc=1%x50=50%
Total torque T=100%(motor rated torque)+50%=150%.

The above simplified calculation shows that a very small torque ripple caused by the VFD motor is amplified by torsional resonance resulting in very large torque.
In this case, 150% torque is presented to the coupling, so that the coupling may be damaged with fatigue fracture if the fatigue factor of safety is smaller than 1.5, as 1.25 is the minimum safety fatigue factor recommended by API 671.

If this characteristic of VFD motor driven compressor trains is not clearly understood among all concerned parties, the worst case scenario is likely to happen. In point of fact, the Author endured a bad experience in applying a VFD motor drive system due to the lack of the above mentioned knowledge. Therefore, the Author is willing to share that experience as a Lesson Learned to promote better understanding for all parties.

LESSON LEARNED

Outline of coupling failure

Figure 10 outlines the coupling failure event.
The VFD motor drives two compressors via speed increasing gear. The Motor rated power is 13.7MW and VSI (PWM) is applied.
After 2,000Hrs operation from plant start-up, the coupling between the gear and LP compressor was damaged due to fatigue fracture.

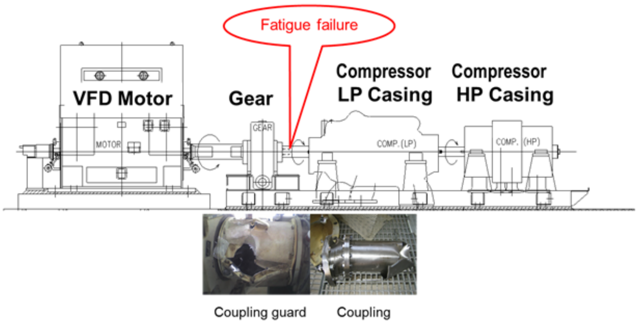


Figure 10; Outline of coupling failure event

Engineering stage

This was first time for compressor vendor to apply a VFD motor drive system for a process compressor application. At that time, the torsional resonance phenomenon associated with VFD motor drives was not known among the parties concerned. The majority of VFD experience was for driving steel mill machinery, and precise torque control was required for that application. So they adapted the latest VFD system technology incorporating all knowledge acquired with that application.

After engineering had started, the VFD motor OEM provided the compressor manufacturer with integer harmonics data only. The compressor manufacturer did not request any other data due to unfamiliarity with VFD systems. The usual torsional analysis was carried out with resonance of integer harmonics being overlooked. No other torsional analysis was undertaken because the need for that was not recognized. Of course, the compressor manufacturer had diligently studied the VFD system to the fullest extent possible, but it was not enough because the risk inherent in VFD systems was unknown in the process compressor engineering field.

Observation and action during shop test

During shop testing, non-synchronous lateral vibration was observed at the gear shaft; refer to Figure 11.

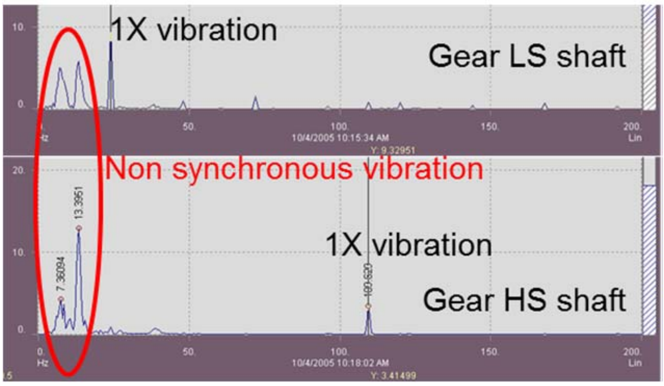


Figure 11; Non-synchronous lateral vibration at gear shaft

The frequency of non-synchronous lateral vibration was very close to the natural frequency of train torsional vibration. At that time, the cause was suspected to be the torque ripple generated by the motor. So, the compressor manufacturer requested additional data about the VFD system. The VFD motor OEM then informed that non integer harmonics might be the cause of the torsional resonance. The VFD motor OEM had performed VFD simulations to check if torsional resonance had occurred. Subsequently, both parties recognized that torsional resonance due to VFD non integer harmonics was the cause of non-synchronous lateral vibration.

Before shipping to site, additional analysis was conducted to confirm the soundness of all machines for job site operation. The VFD motor OEM had simulated air gap torque considering all actual train characteristics; refer to Figure 12.
The Compressor manufacturer had performed steady state torsional frequency analyses using simulated air gap torque, per Figure 13, and obtained the expected ripple torque value for jobsite operation. Also, the VFD motor OEM and End User conducted independent analyses to verify the simulations. Finally, fatigue evaluation was performed and no harmful effects being detected; refer to Figure 14.

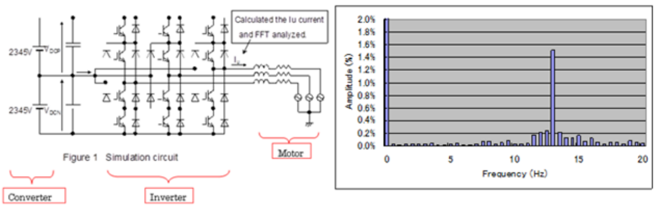
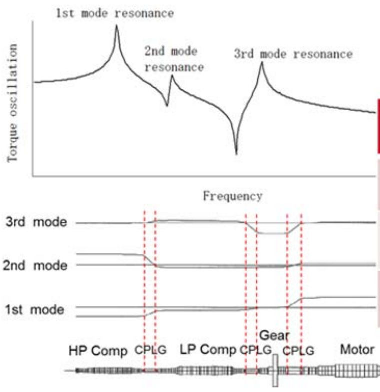
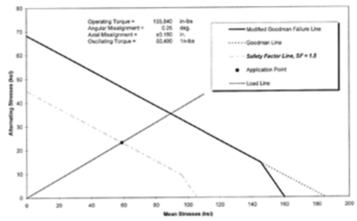


Figure 12; Simulation of air gap torque



Coupling	Ripple torque
Motor-Gear	19.7%
Gear-LP comp	32.4%
LP comp-HP comp	6.9%

Figure 13; Steady state torsional frequency response analysis



Coupling	SF
Motor-Gear	1.75
Gear-LP comp	1.5
LP comp-HP comp	1.75

Figure 14; Fatigue evaluation

Root cause analysis 1st step

After 2,000Hrs operation from plant start-up, the coupling between the gear and LP compressor was damaged. The damaged coupling was the smallest SF, i.e., the generated torque ripple was the largest value. This fact suggested that the large torque ripple generated was greater than the simulation result.

At first, to check the actual phenomenon, field measurements were conducted considering time limitation. Figure 15 shows the measurement arrangement. The damaged coupling was replaced with a new one equipped with a measuring element and then restarted. The shaft angular displacement of the gear shaft was measured with a laser measurement device. This data was compared with the gear LS shaft lateral vibration data. Asynchronous component was identified both angular displacement and lateral vibration of gear LS shaft corresponding to 1st torsional natural frequency. Lateral vibration value of gear LS shaft was rapidly increased at some timing.

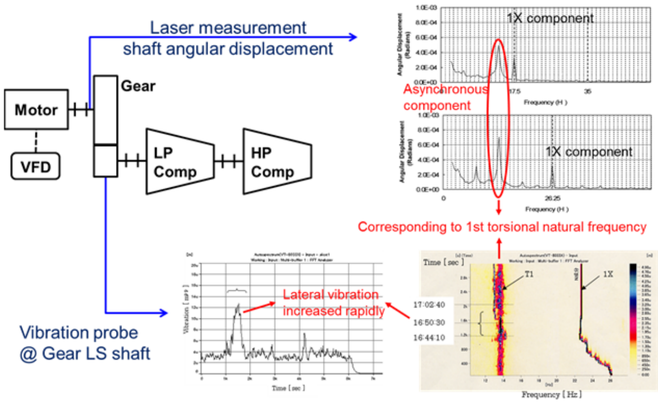


Figure 15; Field measurement 1

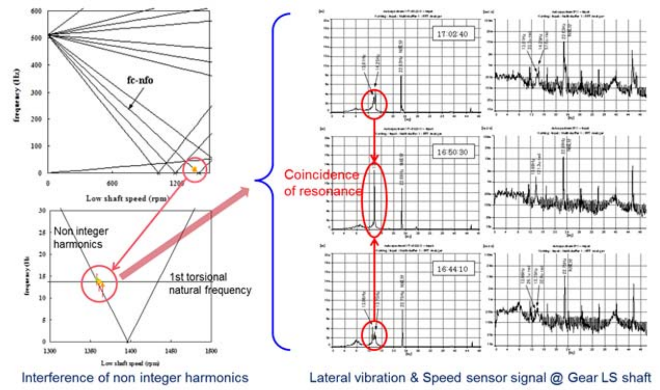


Figure 16; Interaction of non integer harmonics

Figure 16 shows the interaction between non integer harmonics and 1st torsional natural frequency. The Campbell diagram on the left plots the operating point indicating non integer harmonic and 1st torsional natural frequency. The graph on the right illustrates lateral vibration and speed sensor signal at the gear LS shaft. The result of this measurement indicated that lateral vibration was rapidly increased when the non integer harmonics component coincided with the 1st torsional natural frequency component. This phenomenon suggested that the non integer harmonic component was changed with a small change in the operating condition and severe resonance was occurred intermittently, and large oscillated torque was generated at that timing. So the coupling was damaged after 2,000Hrs operation.

To verify the above assumption, the damaged coupling was subjected to detailed investigation, to wit: visual examination, chemical analysis, micro analysis, hardness testing, SEM analysis and optical metallographic analysis; refer to Figure 17. By this investigation, the material issue was not observed, but the evidence of high cycle fatigue was found. Moreover, FEA analysis by coupling vendor supported that investigation result.

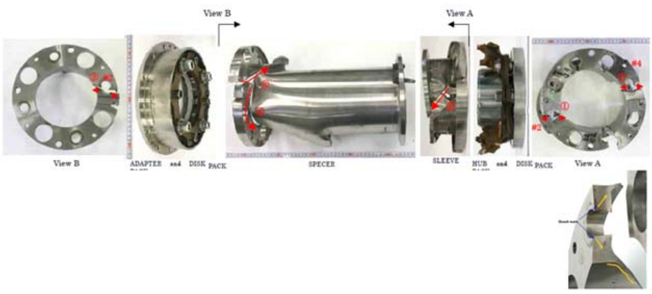


Figure 17; Investigation phot of damaged coupling

In accordance with field measurement 1 and the damaged coupling investigation, the damage phenomenon was identified as fatigue fracture. The next step was to identify the cause of fatigue fracture, and then to find a solution.

Root cause analysis 2nd step

To check the actual operating condition more precisely, strain gauge measurements and VFD signal measurements were carried out. The preparation work required additional time, so these measurements were not conducted at the same time as the 1st step. Figure 18 shows the strain gauge measurement arrangement and its result. Figure 19 shows the correlation between torsional and lateral vibration.

The strain gauge measurement unit was installed on the coupling between the motor and gear. The graph on the right illustrates a water fall plot. The X axis shows torque oscillation frequency and the Y axis shows torque oscillation. The asynchronous component corresponding to the 1st natural frequency was observed all the time, i.e. during start-up, normal operating speed and coast down. The torque oscillation value changed in accordance with changes in the operating condition. A higher torque oscillation value was observed during speed up; this suggested that it was generated not by non- integer harmonics, but by asynchronous harmonics.

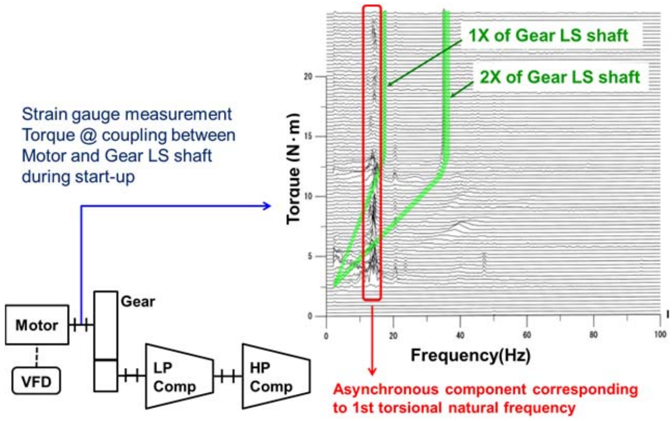


Figure 18; Field measurement 2 (Strain gauge)

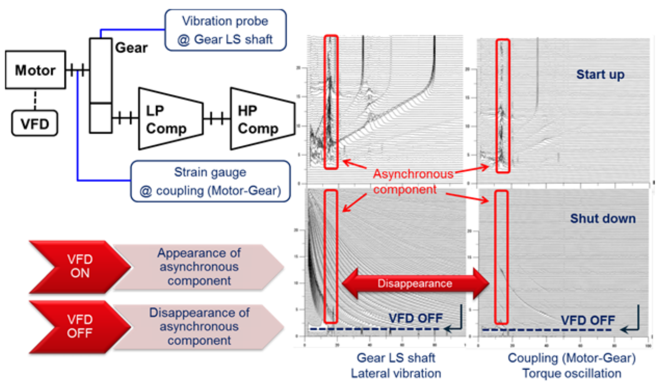


Figure 19; Field measurement 2
(Correlation between torsional and lateral vibration)

The above graph shows the correlation between torsional vibration and lateral vibration at the gear LS shaft. Torsional vibration means torque oscillation measured by strain gauge. The upper graph shows the start-up condition, and the lower graph shows the shutdown condition by VFD system trip. The left graph shows lateral vibration at the gear LS shaft and the right graph shows torsional vibration at the coupling. The X axis shows vibration frequency and the Y axis shows vibration value, accordingly. The tendency of both graphs was very similar even if its magnitude was different. During start-up, an asynchronous component was observed clearly at both vibrations. However, the asynchronous component disappeared during shut down; this meant that the exiting force causing the asynchronous component had disappeared just after the VFD system went off. This was clear evidence that the asynchronous component of torsional and lateral vibration was caused by the VFD system.

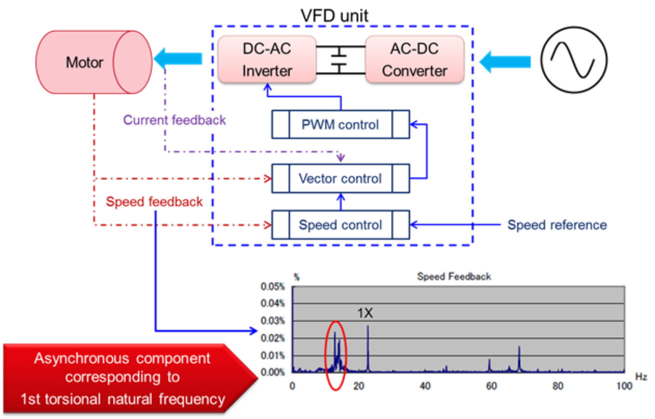


Figure 20; Field measurement result of VFD signal

Figure 20 shows the field measurement result of the VFD signal. The above sketch depicts a simplified VFD control loop diagram. The lower graph shows the feedback signal from motor speed to the speed control module. Many VFD signals were measured and found the asynchronous component of one signal, the speed feedback signal, corresponding to the 1st natural frequency. The asynchronous component frequency of the speed feedback signal did not change at any time, even if the motor speed was changed. But the signal value did change according to the change in operating conditions. This fact suggested that the speed feedback signal might be caused by torsional resonance vibration.

Based on the above mentioned considerations, the following root cause analysis is presented; refer to Figure 21.

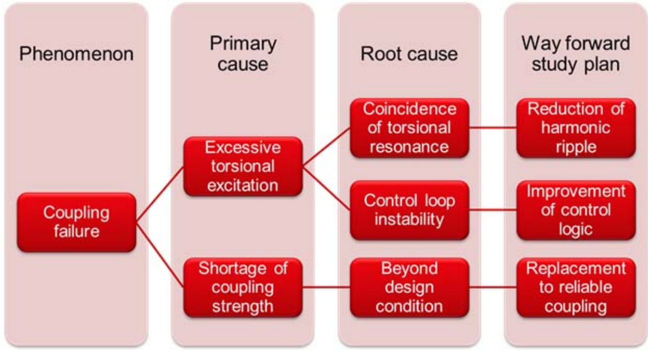


Figure 21; Root cause analysis

Way forward action

Reduction of harmonic ripple

Non integer harmonics could not be avoided, so it was important to minimize the harmonic ripple as much as possible. To determine the best parameter setting, VFD control simulation was required.

Improvement of control logic

An asynchronous component other than non -integer harmonics was observed by field measurement; detailed VFD control simulation was required to identify the interaction between the control loop and entire train characteristics.

Figure 22 shows the result of detailed VFD control simulation. The asynchronous component was reproduced in the speed feedback signal by this simulation model, so that improvement of the control loop and parameter setting could be investigated by using this simulation model.

Figure 23 shows the improvement result of the VFD control loop. To simplify the control loop, the feedback signal from motor to VFD controller was omitted, because the precise torque control was not required for the process compressor, as opposed to steel mill machinery. In addition, the vector control was replaced to a simple open loop control called V/F control, which was realized as reverse technology for control engineer. From a mechanical engineering point of view, the advanced complex control was not always the best solution, because the uncertainty sometimes might cause unexpected issues. After this modification of control loop simulation, the asynchronous component was remarkably reduced. This result suggested that the simplified V/F control could be a good solution of the issue.

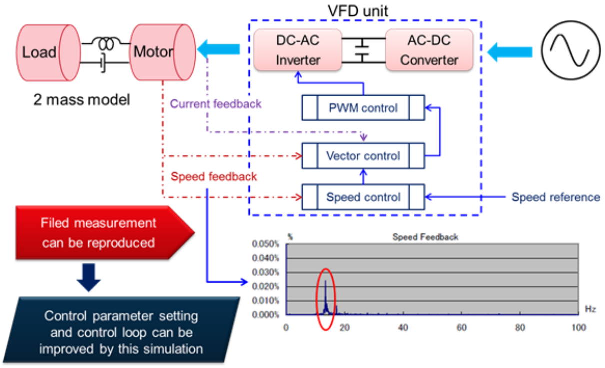


Figure 22; VFD control simulation

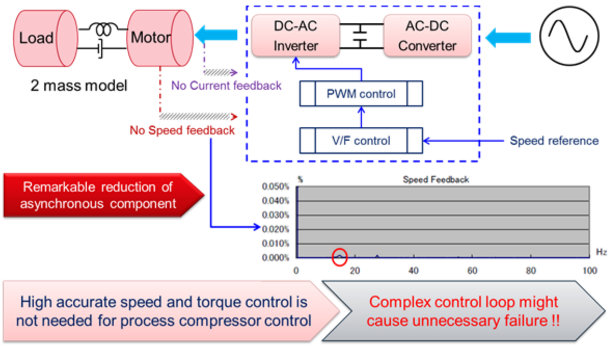
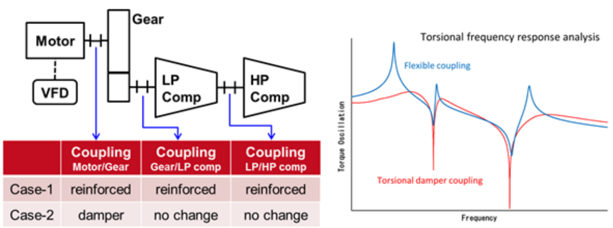


Figure 23; Improvement of VFD control loop

Replacement to reliable coupling

Another solution was to reinforce the coupling strength to withstand the torque ripple. One alternative was to reinforce the coupling simply, i.e. increasing coupling rating; the other was to apply a torsional damper coupling to absorb the torque ripple oscillation. A comparative study was performed per Figure 24. Torsional damper coupling was better solution.



Improvement of coupling SF	Coupling Motor/Gear	Coupling Gear/LP comp	Coupling LP/HP comp	Judgement
Original	1.75	1.5	1.75	---
Case-1	1.5 times	1.5 times	1.5 times	fair
Case-2	3.1 times	2.5 times	2.3 times	good

Figure 24; Study of coupling reinforcement

Way forward action

While the above mentioned remedial action studies were being conducted, the plant continued to operate under partial load with constant monitoring of the operating conditions. Figure 25 shows one practical monitoring method to be gear lateral vibration. In accordance with the field measurements, a good correlation between torsional vibration and lateral vibration at the gear shaft was verified.

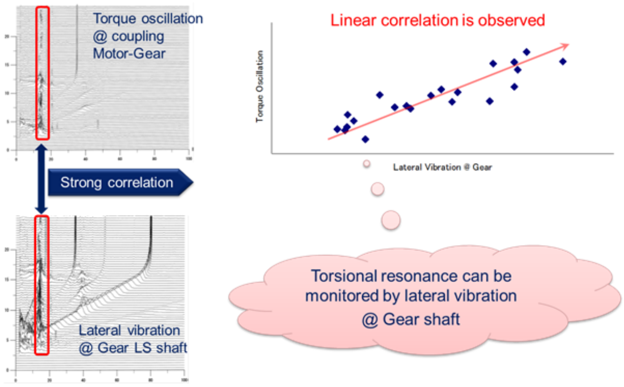


Figure 25; Monitoring method

During this period, the spare coupling was likewise prepared for emergency use. VFD control software was modified according to the simulation and demonstration testing conducted by the VFD motor OEM. The torsional damper coupling was manufactured for contingency use.

The result of remedial action

Figure 26 shows the result of remedial action. The control loop and setting was replaced with an improved control loop verified by control simulation and demonstration testing. After taking this remedy action, a remarkable reduction in torque ripple was verified, and since that time, the plant has been operating without additional any issue.

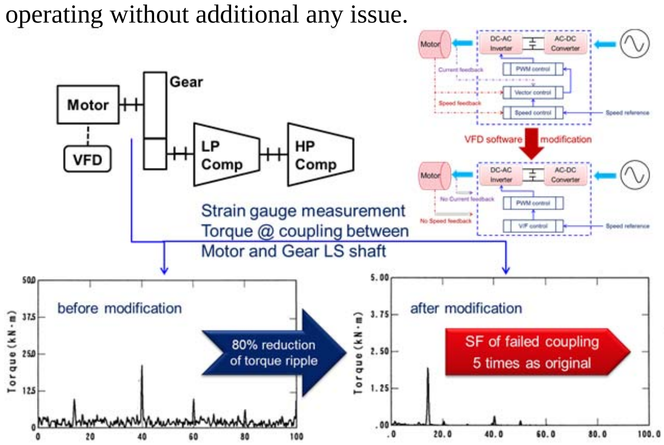


Figure 26; Result of remedy action

Recommended practice

The following recommended practices can serve as a guideline for leading the project to a successful outcome:

① Close collaboration work among all involved parties will avert misunderstandings and miscommunication.
② All information should be shared among, and understood by all involved parties.
③ The verification process should be mutually agreed before initiating engineering and manufacturing.
④ Torque measurement during field operations, especially commissioning, can minimize risk.
⑤ Parameter setting of the VFD controller should be finalized to monitor the commissioning operation.

EXAMPLE OF TORSIONAL VIBRATION MEASUREMENT

Shop measurement

Figure 27 shows shop measurement arrangements during Full load (FL), Full speed (FP) and Full pressure (FP) testing. The compressor was driven by a gas turbine with a helper motor controlled by a VFD system.

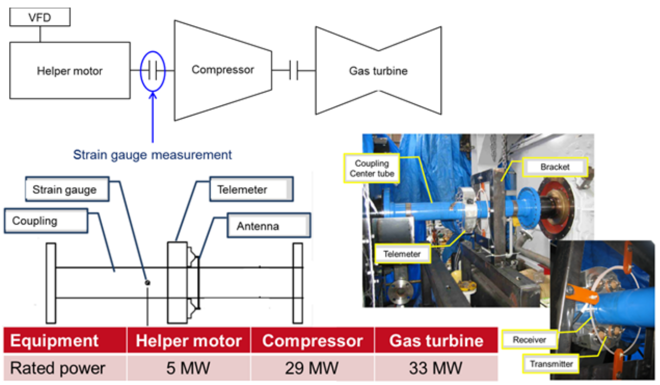


Figure 27; Arrangement of shop measurement

During the engineering stage, compressor manufacturer and VFD motor OEM had worked closely together to eliminate the risk of torsional resonance as much as possible. FL/FS/FP testing was required on that project, so the compressor manufacturer could check the rated operating condition and verify the soundness of the equipment. No abnormal phenomena were detected during that testing, but one interesting phenomenon was observed. Figure 28 shows the change of torsional natural frequency during operation. This train was driven by gas turbine and VFD helper motor, so that the power of the VFD system could be switched off during operation. When the VFD control was switched off, the 1st torsional natural frequency was suddenly changed. This change was suspected to have been caused by the electrical spring function of the VFD controller, and it was subjected accordingly rotor dynamic analysis. This phenomenon was simulated by the change of the electrical spring coefficient of the VFD motor.

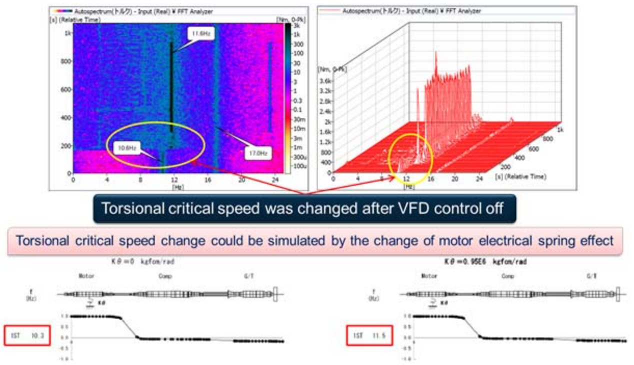


Figure 28; The change of torsional natural frequency

Field measurement

During replacement project of driver to VFD motor from other type of driver like gas turbine, more careful engineering and verification is required. Before replacement of driver, the compressor train was operated as former driver, so that the verification test of whole train could be possible at replacement stage only. If something happened at that time, the plant could not be restarted up as planned. Figure 29 shows the arrangement of field measurement and its result. This replacement project was successfully completed without any issue and delay as planned.

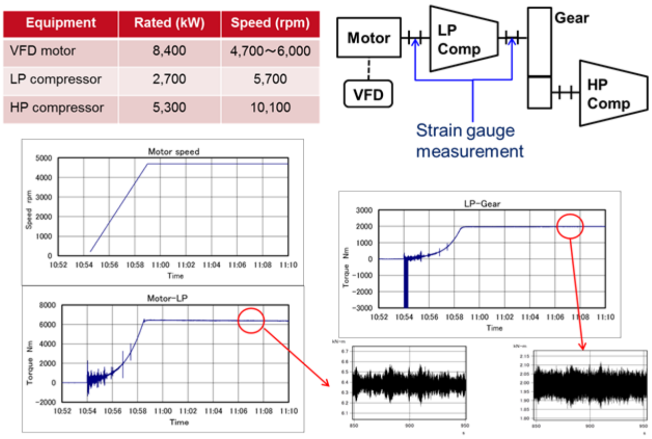


Figure 29; Arrangement of field measurement and result

After completing field measurements, the field measurement results were compared with the analysis results for verification purposes; refer to Figure 30.

The 1st, 2nd and 3rd torsional resonances were absorbed and they matched with the analysis results. Other components of line frequency and operating frequency were also observed. Asynchronous components were observed, but their magnitudes were small enough and verified to have no harmful effect on the operation.

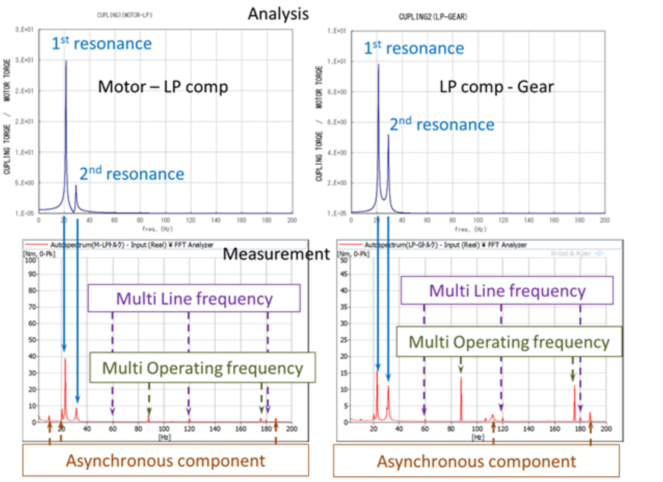


Figure 30; Comparison between analysis and measurement

A major uncertainty of torsional resonance analysis is the damping ratio. In this field measurement, the damping ratio was identified, per Figure 31. The measured damping ratio was 0.75% and the coupling strength was evaluated and its soundness thus verified.

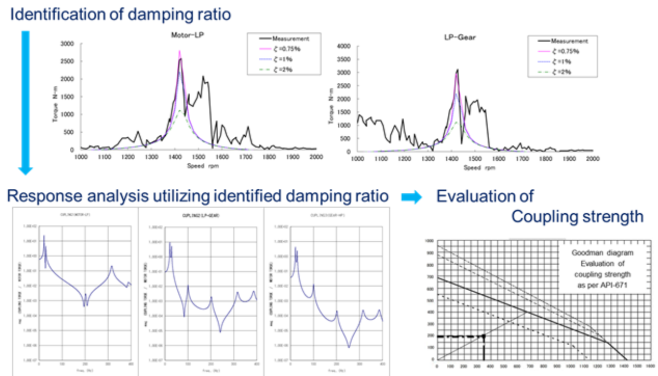


Figure 31; Identification of damping ratio

MONITORING OF TORSIONAL VIBRATION

The monitoring device for torsional vibration is not equipped usually, because suitable engineering can offset and virtually eliminate the risk. On the other hand, there has been a recent increase in the number of papers about torsional vibration caused by VFD motor drives. As mentioned above, this trend might be caused by the lack of knowledge of torsional resonance. If so, then the fatal worst case scenario could be avoided if monitoring units for torsional vibration are installed.

The torque meter is one of the easiest solutions if it is implemented at the start of engineering. The torsional vibration can be directly measured. But some consideration to lateral vibration is required and presents some concern for long term operation.

The measurement of shaft angular displacement is one lower cost solution. But reliability for long term operation is not so good and some ageing due to environmental conditions can occur.

The measurement of gear lateral vibration is one practical and economical solution because a vibration probe is usually installed. But it is an indirect measurement and some difficultly due to mode shape can arise. Notwithstanding this possible disadvantage, this is can be an effective solution because whenever the monitoring can be stated if needed.

To understand the relationship between gear lateral vibration and torsional vibration, analysis modeling is presented in Figure 32. Gear pinon and wheel are connected via oil film at gear teeth. Some factor shall be assumed based on actual measurement results. Figure 33 shows the comparison between analysis and measurement results. Good agreement was verified at lower mode torsional vibration.

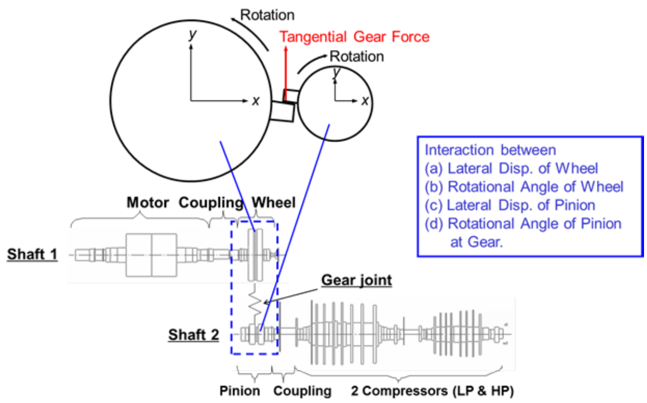


Figure 32; Analysis model of lateral and torsional vibration

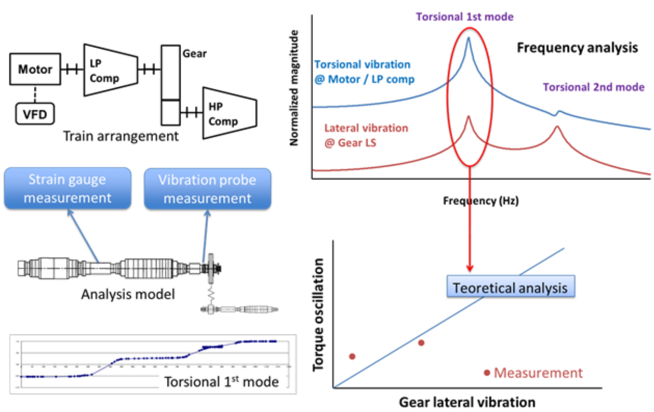


Figure 33; Comparison between analysis and measurement

PREVENTIVE DESIGN METHODOLOGY

Preventive design is important to avoid future problems related to torsional resonance caused by VFD systems. The Most important thing is to avoid resonance as much as possible. This can be achieved with a sound coupling design, but all resonance cannot be totally avoided: refer to Figure 34. In that circumstance, the weakest part, usually the couplings, shall be evaluated by fatigue design using torsional response analysis. Also, mechanical torsional damping is another solution.

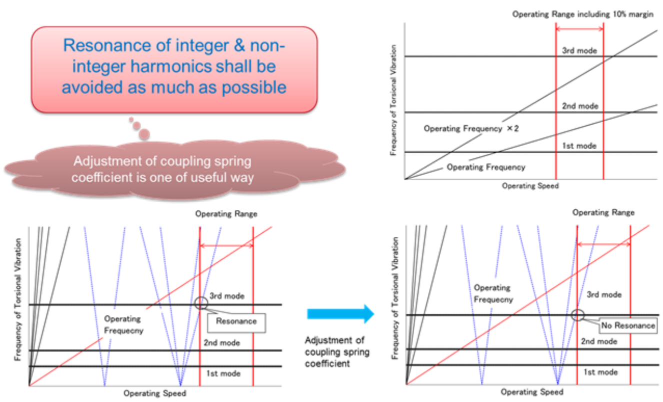


Figure 34; Avoidance of torsional resonance

From the VFD control point of view, basically three types of application are presented; refer to Figure 35.

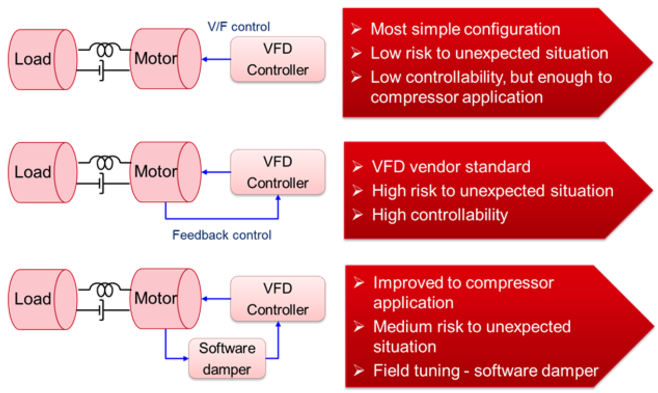


Figure 35; Typical three types of VFD control

The First one is the simplest control and is recommended for process compressor control. The controllability of speed and torque may not be so good compared to the following two systems, but it has enough capability for effective process compressor control. The simple control loop can eliminate any unexpected and unexperienced issue as much as possible compared to the following two more complex control loops.

The second one is the standard control system of VFD motor OEM which provides good controllability for speed and torque. But some risk may be hidden in the technology beyond current experience and knowledge due to the interaction between the complex control loop and the entire train characteristics.

The third control application is an improved version of the second one considering torsional resonance for process compressor applications. This seems be a good one, but some risk may be hidden same as second one.

It is important that all involved parties fully understand the characteristics of each control loop in order to select the one most suitable for the specific process compressor requirement.

CONCLUSION

As described above, misunderstanding and miscommunication among the involved parties due to lack of knowledge relative to VFD motor systems could lead to machine damage and in the worst case an unexpected plant shutdown.

① VFD systems generate torque ripple
② Torque ripple can cause machine damage and plant shutdown
③ All interested parties need to fully understand the characteristics of VFD motor driven compressor trains
④ Close collaboration among all concerned parties is key to success.
⑤ Verification work during engineering and testing is essential.

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