ASIA TURBOMACHINERY & PUMP SYMPOSIUM
23-26 FEBRUARY 2021
SHORT COURSES: 22 FEBRUARY 2021

PRUDENT DESIGN FOR TRANSIENT TORSIONAL VIBRATION IN MECHANICAL DRIVES

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ABSTRACT

In addition to the conventional analysis to identify system torsional natural frequencies for rotating machinery train, there may be a requirement to perform transient analysis if the system will be exposed to transient torsional excitation, as identified. A transient torsional vibration of machinery train installed at site may go undetected until a certain number of events occur after which a low cycle fatigue failure could occur. Some of the transient events are not possible to avoid such as the start of synchronous motor. Meeting the machine design criteria to sustain high oscillating torque during transient events is difficult in terms of cost and effort.

From mechanical design perspective, paper emphasizes that based on frequency of occurrence of transients such as short circuit events, what prudent action can be taken to protect main driveline. To minimize the torsional oscillations startup of synchronous motor driven train, driven inertia can be kept lower as much as possible by avoiding overdesign of the gear boxes and rotating machines to maintain high stiffness. Pros and cons of using elastomeric type damper couplings, fluid couplings are also discussed as tool to mitigate the high torsional oscillations during transient conditions such as startup.
With published API guidelines and API recommended practices, the paper addresses when and how to tackle the design challenges of rotating machinery -mechanical drive train. With latest advent on power electronics, usage of various online protection and monitoring tools such as IGBT (insulated-gate bipolar transistor), partial discharge and flux monitoring system etc. which may mitigate the occurrence of short circuit events to high extent.

INTRODUCTION

All rotating machinery systems experience torsional oscillations to some degree during startup, shutdown, and continuous operation. Consequently, the torsional response characteristics of rotating equipment should be analyzed and evaluated to ensure the system's reliability. Severe torsional vibrations often occur with the only indication of a problem being gear noise or coupling wear. Excessive torsional vibrations can result in gear wear, gear tooth failures, key failures, shrink fit slippage, and broken shafts in severe cases. The severity of the torsional oscillations and stresses depends upon the relationship between the operating speed and excitation frequencies of unsteady torques and the torsional natural frequencies and mode shapes of the shaft system (critical speeds). The difference between these frequencies is referred to as the separation margin. The magnitude of the stress also depends upon the amplification factor on resonance and the stress concentration factors which mainly related to shaft element design. Unlike lateral vibration, there is no high viscous damping present in a torsional train. (Dr. Ronald L. Eshleman 1977,)
From start up to full operating speed, a high -speed electric motor driven turbomachinery drive train has to pas s through few high

torsional oscillations. Event of startup of a synchronous motor, AC-DC-AC conversion in a VFD drive, sudden re-closure of breakers, 2 phase and 3 phase short circuit produce very high torsional oscillations. These are known as transient conditions and this is the regime where most of component failure occurs. To avoid coupling and keyway failures (Jon R. Mancuso et al 2001) due to high torsional oscillations, most of time the trains are designed with higher robustness than actually required.
The objective of this tutorial session is to present challenges and mitigation of such issues with prudent design considerations.
The tutorial paper provides –

- brief synopsis of transient torsional oscillations with various types of electrical drives,
- various calculation methods to evaluate fatigue life of component,
- API mandates and recommended practices on addressing transient torsional vibration.
- various design methods of components and preventive maintenance of electrical machines to mitigate component failures du e to short circuit.

TORSIONAL EXCITATION IN ELECTRIC MOTOR DRIVEN TURBO-MACHINERY TRAIN

Continuous torsional excitation in a motor driven turbo -machinery train

The continuous excitation sources of the torsional vibration for machinery trains are the following:
An excitation torque generated with 1or 2 times line frequency is common for direct on line (DOL) driven induction motors. These excitation frequencies are to be separated from the 1st and 2nd torsional natural frequencies otherwise these natural frequencies significantly excited and lead to significant and potential damages. (Michael Glasbrenner et al 2016)

An inter-harmonic frequency from variable frequency drives (VFD motor driven trains) is source of continuous excitation. Variable frequency drives are potential source of both transient and continuous torsional excitation. Whereas harmonic excitation frequencies have to be considered regarding transient excitation during start-up, the inter-harmonic excitation frequencies caused by VFD can lead to continuous vibratory torques during a steady state operation. Based on the experimental results, inter-harmonics of 6th and 12th order are important for the torsional train design particularly for couplings. These continuous vibratory torques may reach to maximum values of 10% to 35% of steady state torques for coupling.

Running speeds and gear characteristics such as unbalance, pitch line run-out and cumulative pitch error as addressed in API 684. The potential magnitude of torsional excitation from motors and high-quality gear units are usually low. A harmonic forced vibration response is carried out as torsional simulation with an alternating excitation torque. Based on experience and various issues reported in industrial fraternity, 1st and 2nd torsional natural modes can significantly be excited at both driver (usually motor) and gear unit(s). In majority of cases, vibratory torques are found to be lower than 30% of steady state torque.

Now a days, with more robust design practice, these vibrations usually are found to be lower than 10% of steady state torque. Based on experiences and calculations, torsional modes higher than 4th mode are usually neither significantly excited at driver nor at gear units.

Transient excitation in a motor driven turbo-machinery train

Excitations that are transient in nature such as the starting of a system driven by a synchronous motor, the acceleration of a variable frequency drive motor, a restart of an induction motor after a power interruption or an electrical fault in a motor or generator .These have the potential to introduce electrical excitation, which is converted to torque in the motor, any of which may excite torsional natural frequencies.

Some transient torsional vibration excitations may be caused by just one action, such as the breaker re-closure of an electric motor or the short circuit of a motor. Other transient torsional vibration may be caused by a rather limited number of events, such as the starting of a synchronous motor till it reaches full speed. Still others may occur more frequently, such as the start and variable speed operation of a variable frequency drive. A failure of a power transmission component may occur in a single transient torsional vibration event. A transient torsional vibration may go undetected until a certain number of events occur after which a low cycle fatigue failure could occur. This potential failure can even occur years after the initial startup of the machinery.

TRANSIENT EXCITATIONS DUE TO VARIOUS ELECTRICAL DRIVES

Following paragraphs provide short notes on Transient excitation with various types of drives

AC Induction motor DOL (Direct on line) start

Induction motors are typically less expensive than synchronous motors, but the operational cost is higher due to the lower power factor (typically 0.75 to 0.85) where the utility cost is based on apparent power rather than the actual power. When electrical induction motors are started direct on-line, they generate a considerable pulsating torque. There is an inter-relationship between the electric motor and the mechanical system which is effectively a multi-mass oscillatory system large transient of both torque and speed occur immediately on start-up due to excitation of the oscillatory system.

These pulsating torques that arise in electric motors can be separated into two categories; those that occur when the machine is running at constant speed and those generated when the machine is accelerating. AC motors generate fluctuating torques is in the first instants after power is initially applied to them. Such transients take much longer to decay in induction motors. Thus, it can be seen that duration of decay of fluctuating torque is a function of train / driven part inertia.

VFD Variable Frequency Drive Excitations

Variable speed drive motors are controlled by variable frequency drives (VFD) which alter motor speed by varying the electrical frequency supplied to the motor's terminals. The process of generation of torsional ripples in VFD motor are as follows - Smooth AC sine wave is input to the VFD system. AC-DC-AC conversion generates distorted AC sine wave and it is input to the motor as drive frequency. Accordingly, the motor air gap torque has a ripple component. (Vijay Ganesan et al 2016). This superimposed ripple frequency accidentally coincides with the natural frequency of the compressor train. As a result, torsional resonance vibration is caused. A schematic presentation of VFD drive is shown in Figure 1**.**

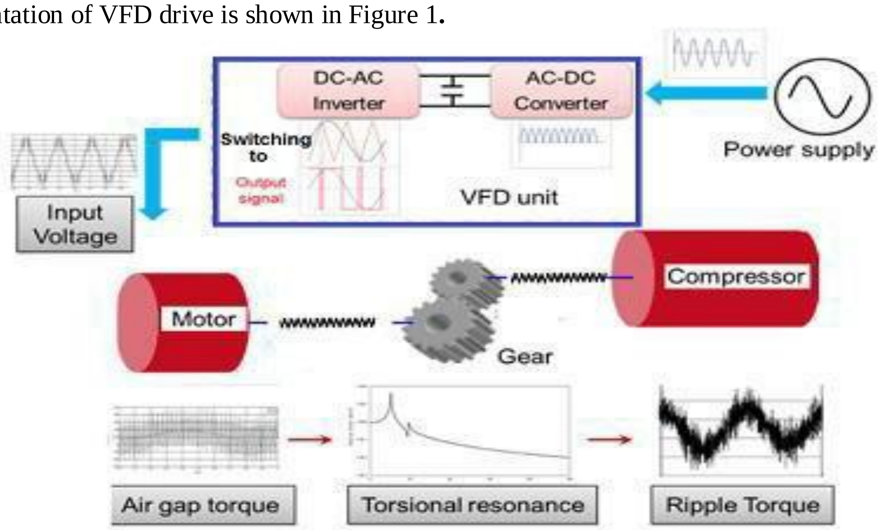

Figure 1 VFD drive and torsional ripples (courtesy Masayuki Kita 2016- https://doi.org/10.21423/R1FT43)

Torsional resonance vibration occurs when exiting frequency, such as torque ripple frequency caused by a VFD system, is coincident with TNF (torsional natural frequency) of the compressor train. Torsional resonances occur when energy at multiples of mechanical running speed and electrical harmonics from the VFD intersect at TNF. In the speed range between zero and 10 percent of maximum speed, most variable frequency drives produce fluctuating torques that are several times larger than those generated over the remainder of the speed range. Because of this, most users choose to begin the active operating range at a speed above 10 percent. In that case, due to larger time taken to reach above 10% of speed, there shall be high torsional oscillations. (Harley Tripp et al 1978) Normally, damping ratio 1% is assumed by OEM (Original equipment manufacturer) as $\zeta=0.01$. Amplification factor of torsional vibration $AF=1/(2x\zeta)=50$. 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 exerted to the coupling which may be damaged with fatigue fracture. As torque ripple generates cyclic torques, it can be well visualized that duration of torque ripple can reduce the fatigue life of component exponentially. (Dr. Luis San Andres - Note 9 Lecture –TAMU Texas)

Because of the large weight involved, the inertia of the driven is many times greater than the inertia of the motor. For the first torsional mode, the motor core is typically near an anti-node and acts like a torsional pendulum. The driven, on the other hand, is usually near the node and acts as an anchor as shown in below Figure 2.

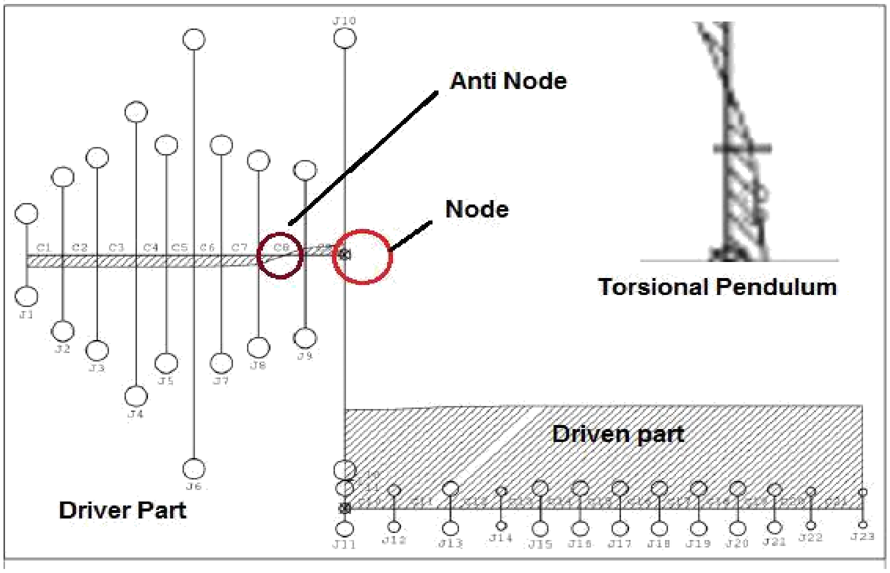

Figure 2 Example of Torsional Pendulum in torsional model

In a torsionally stiff, lightly damped system, the first torsional mode is very sensitive to any harmonic excitation or sudden speed adjustments from the VFD motor. (Timo P. Holopainen et al 2013) It has been also found that torsional resonance due to VFD non-integer harmonics can cause non-synchronous lateral vibration in rotors.

Synchronous Motor drive

Unlike induction motors, synchronous motors are not self-starting, and they are normally equipped with squirrel- cage windings which provide starting torque and also provide damping during steady state running. The stator's magnetic field begins rotating at synchronous speed virtually instantaneously after power is applied. With the stator field rotating and the rotor at rest, alternating forward and reverse torques are applied to the rotor (Malcolm E. Leader- Dyrobes technical paper) . This causes the rotor to swing back and forth These windings are utilized to accelerate the motor as an induction motor from zero speed to a speed slightly less than synchronous speed. When synchronous speed is approached, DC field voltage is applied, and the rotor is pulled into synchronism. It is known that synchronous motor rotors contain salient poles that are magnetic protrusions enclosed by field coils. A schematic model and torsional model of synchronous motor driven turbo -compressor is shown in Figure 3.

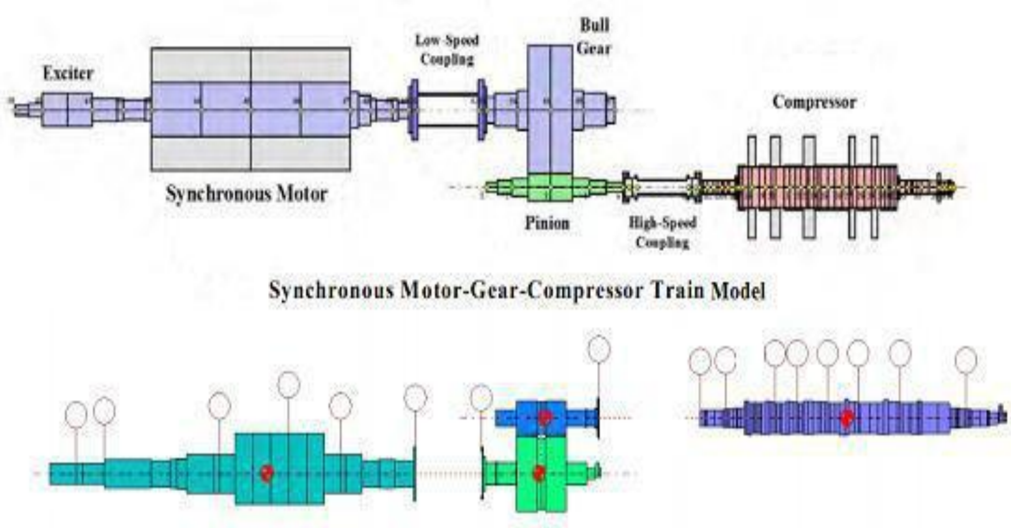

Figure 3 – Synchronous motor drive train schematic and torsional model. courtesy Dyrobes

The frequency of the torque pulsations is the frequency at which the stator's rotating magnetic field passes a rotor pole. Since the stator's magnetic field rotates at synchronous speed, the excitation frequency is a function of the difference between synchronous speed and rotor speed, which is known as slip speed. Specifically, excitations occur at twice slip frequency where slip frequency is defined by the following equation:

Fslip =FL *(Ns -N) / Ns

Where:
- Fslip is Slip frequency (Hz)
- FL is Line frequency (Hz)
- Ns is Synchronous speed (rpm)
- N is Rotor speed(rpm).

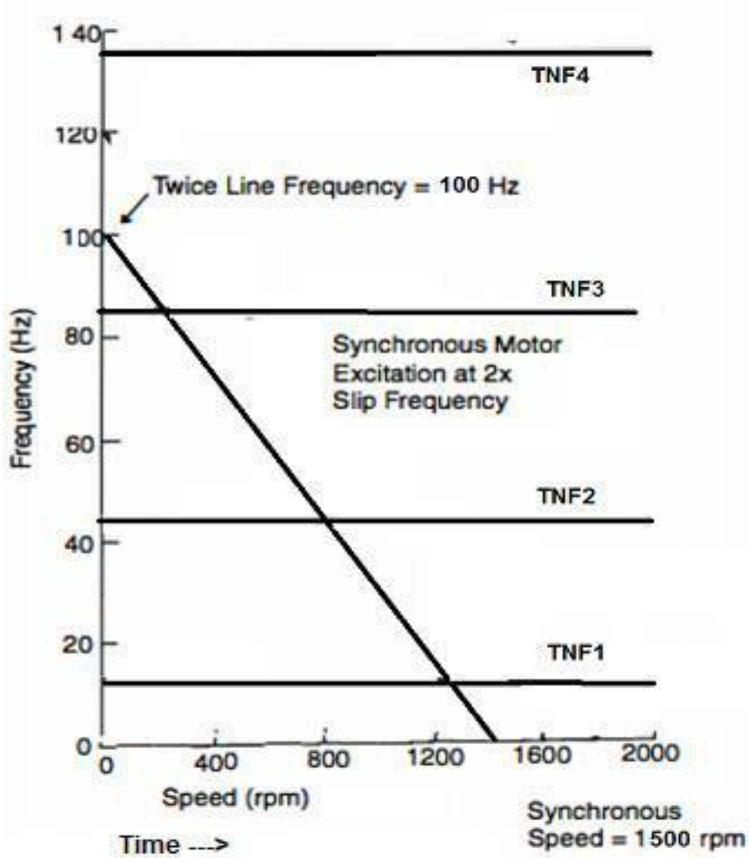

Figure 4 – Synchronous motor start up diagram

All synchronous motors produce torque pulsations between start and synchronous speed. These pulsations start at twice line frequency (120 Hz or 100 Hz) and decay to zero Hz when the motor synchronizes. Thus, all torsional resonances between 0 and 7,200 CPM /6000 CPM will be excited every time the motor is energized. As shown in Figure 4, that at time zero, oscillatory and load torques are applied simultaneously. This causes the train to accelerate in speed. Then, at very small-time steps, a new speed is calculated along with the torsional oscillation amplitude. Ultimately, the alternating torque amplitudes are converted to the shear stresses in the shafts. One important calculation can be deduced from this analysis is the time to reach full speed as seen in Figure. After synchronization, the motor torque is balanced by the load torque and there are no more considerable alternating torque pulsations from the motor. (Gerald K. Mruk et al 1978)

These torque pulsations can be several times the rated full-load torque of the motor and may cause major failures in components of machinery train. As shown in Figure 5, the train has to undergo torsional excitation till the motor reaches its synchronous speed. While passing through the torsional excitation, low and high-speed shaft line components may face high torsional ripples as shown in Figure 6.

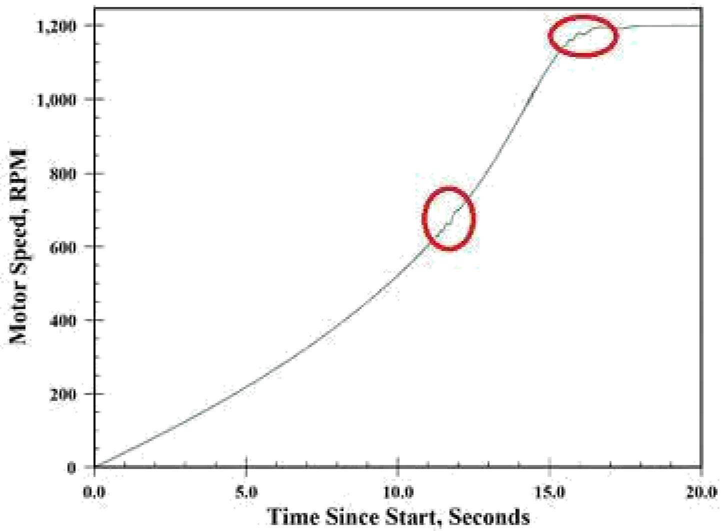

Figure 5 Torsional ripples during start up as marked in circles

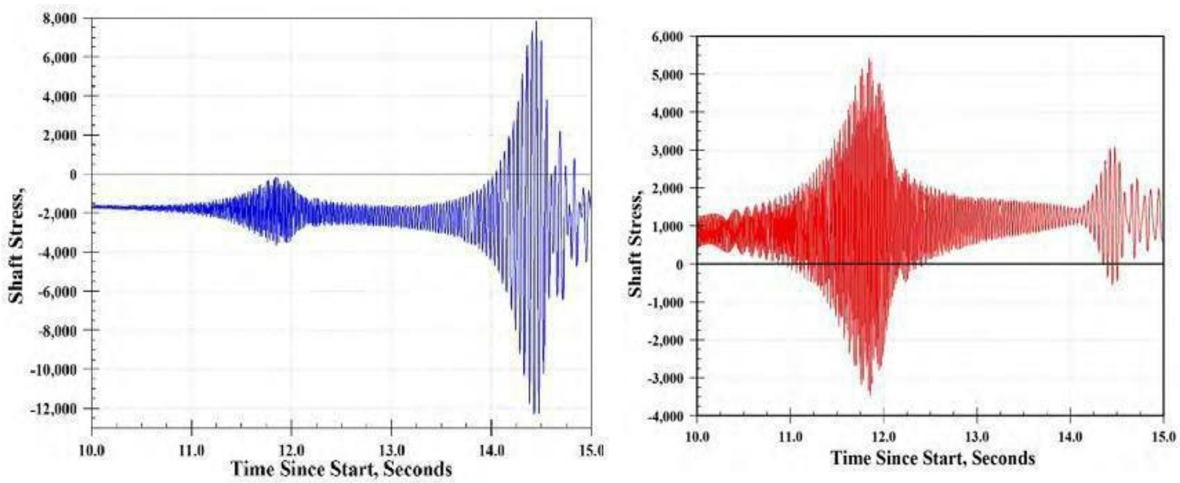

Figure 6 Stresses at Low and High-speed shaft lines during torsional ripples (courtesy -Dyrobes)

TRANSIENT TORSIONAL VIBRATION DURING SHORT CIRCUIT

Following are the types of short circuit which can happen in electrical drive **-** Phase -to-phase short-circuit-This can be caused by insulation breakdown in the motor between windings due to degradation, overtemperature, or overvoltage events.

Phase-to-earth short-circuit-This can be caused by insulation breakdown between a motor winding and the motor casing; usually due to degradation, overtemperature, or overvoltage events

Failure mechanism during a short circuit event

During short circuit torque can be at such a high level and of an instantaneous nature that both the coupling and shaft stressing criteria may not be satisfied. (Tuomo Aho et al 2016) A detailed model of a progressive failure is not normally undertaken to determine the exact nature of the damage that would be expected i.e. whether the shaft would fully shear into two pieces or sustain keyway twist, coupling shear and hub slip etc. Normally, during an event of short circuit, a coupling would fail in following modes or combination of all:

a) Local yielding of membranes would occur
b) Drive bolts would fail
c) Spacer would yield and eventually fail.

In a variable speed drive if the short circuit event occurs at full speed, the response may be significantly lower compared to events occurring at lower speed in operating map of machine. In that case the fatigue life is not infinite, but there is not cross sectional yielding of the shaft.

Short circuit torque values

Short-circuit air gap torque value is provided by motor vendors. In the event of a short-circuit, the motor shaft experiences an instantaneous peak torque as the energy stored in the stator coils dissipate through the air gap, then decays over a small amount of time. The airgap torques at fault conditions. High-speed two-pole induction motor designs shall have electromagnetic short-circuit torque levels comparatively much higher than a four-pole induction motor. When motors experience large pulsating torques, as is the case during short circuits, the peak torque is often expressed in the peculiar units of "per unit" (pu). For instance, if a short circuit caused a peak torque that was five times the motor's rated torque, its magnitude would be specified as 5.0 pu. The typical magnitudes of short circuit torque are as follows:

- Excitations at line frequency: @7.0 pu
- Excitations at twice line frequency: @3.5 pu.

According to various studies conducted so far by experts, the excitation torques generated by short circuits is significantly reduced if a variable frequency drive is in operation when the short occurs. Besides this, due to mechanical impedance associated with motor rotor construction actual torsional oscillation at motor rotor keyway side due short circuit is lesser than what calculated by formulas. Air gap gathers motor electromagnetic torques (change of electro -magnetic field in motor during short circuits), which generate responses of train sections by impact on mass -elastic model. (E. G. Hauptmann et al. 2013). It is to be considered that 2-phase short circuit and 3-phase short circuit do not occur at the same time.

Figure 7 shows how peak excitation torques are generated and high oscillatory twisting occurs in machinery component during the short circuit event.

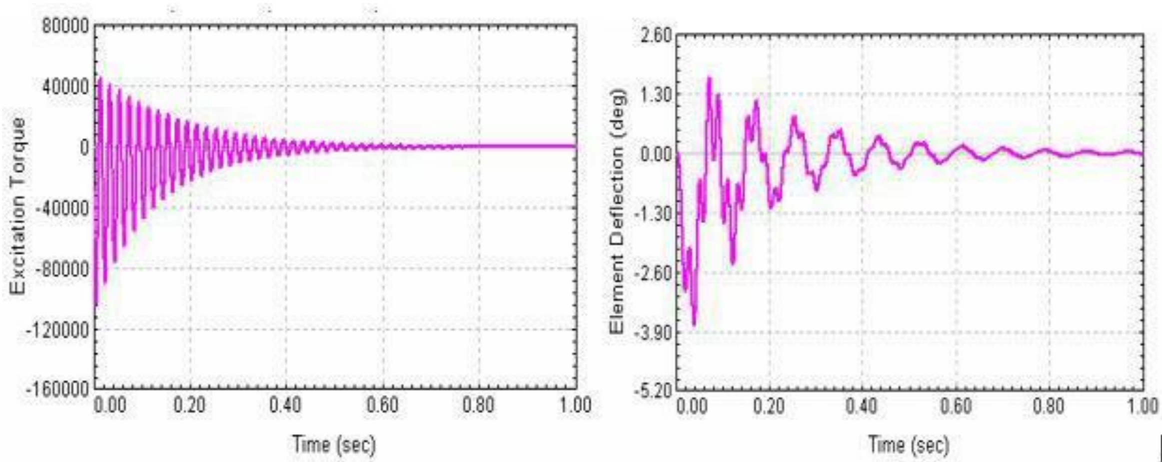

Figure 7 -Short circuit time marching and element deflection curve, courtesy – Dyrobes user manual

FATIGUE LIFE ASSESSMENT FOR COMPONENTS IN MACHINERY TRAIN

In torsional analysis study, strength assessment of affected part of train is carried out as per API requirement however in case of very stress due to torsional transients such analysis is not fruitful as they may indicate failure in design. Hence a Fatigue life assessment analysis is performed for the transient conditions to ensure that shaft-end coupling, and drive component ratings are not exceeded for immediate failure / short lived integrity.

Low cycle Fatigue [LCF] and High Cycle Fatigue [HCF]

Failures that occur at less than 1000 cycles are called low cycle fatigue and high cycle fatigue after 1000 cycles to failure. Above 1 million cycles, the component is assumed to have infinite life at that stress which is known as endurance limit Se, the material is assumed to be safe for continuous operation for its design life. Figure 8 shows a simplified diagram of fatigue life cycle. Fatigue caused by cyclic or random loading and unloading of component in terms of stress are separated in two regions – High cycle and Low cycle. In high cycle fatigue region, the stress is low and primarily elastic and low cycle fatigue where there is significant plasticity. It is to be noted that low cycle fatigue is also lead to crack growth (Anders Ekberg- Chalmers lecture on solid mechanics) Whether using stress/strain-life approach or using crack growth approach, complex or variable amplitude loading is reduced to a series of fatigue equivalent simple cyclic loadings using a technique such as rain flow analysis. The difference between low cycle fatigue (LCF) and high cycle fatigue (HCF) is based on deformations. LCF is characterized by repeated plastic deformation (i.e. in each cycle), whereas HCF is characterized by elastic deformation. Both infinite or finite fatigue life is possible and can be analyzed

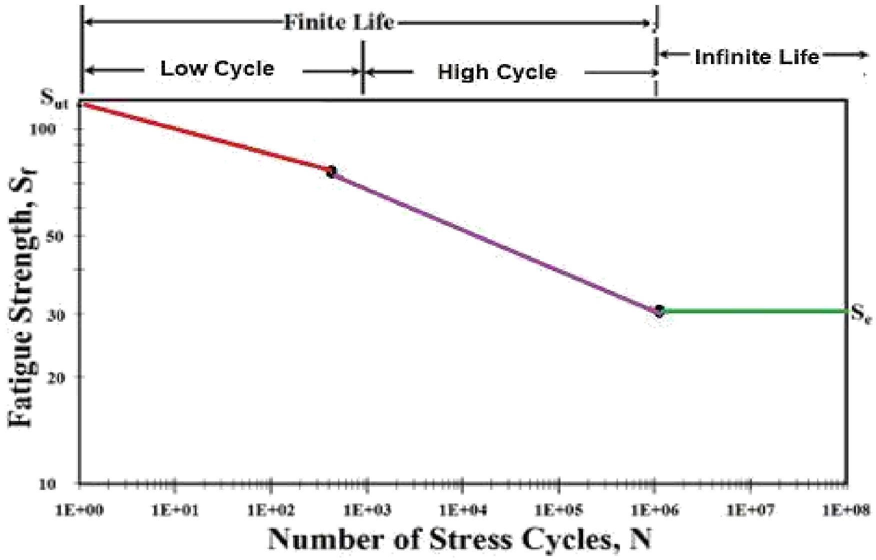

Figure 8 Fatigue life cycle

- LCF - Only finite fatigue life is possible and should be analyzed using LCF-criteria Stress or Strain. Stresses close to (or at) the yield limit, Small stress increment - large strain increment
- HCF -Elastic material Small strain increment - large stress increment
- Best "resolution" if strains are employed in fatigue model.

During transient periods of torsional vibrations are so high that stresses are generated close to yield limit of material. Hence, Methods of calculation LCF are discussed, the fatigue calculations mathematically return an answer for fatigue life, but the primary basis for assessment should be the shear stress across the shaft section. The fatigue calculation will be removed where the component fails the cross-sectional stress criteria.

Miners rule – Rule of Accumulative damage

As based on time history curve, during torsional transients, amount of stress and its time duration are known, a cumulative fatigue analysis such as Miner's rule can be used to precisely determine the life of shaft or component of a turbomachinery train.

$$\sum_{i=1}^{k} \frac{n_i \times S_i}{N_i \times S_i} = C$$

Where Si and ni are stress levels and the number of cycles to failure at the ith stress

As seen in the equation, Miner's Rule states that there are different stress levels and the number of cycles to failure at the ith stress, The number of cycles at stress Si, is ni. The ratio of n to N is the damage fraction, the amount of life that is used up by stress Si. When the sum of damage fractions is greater than 1.0, failure is said to occur. C is the damage index
Miner's rule assumes that the damage done by each stress repetition at a given stress level is equal, meaning the first stress cycle at a uniform stress level is as damaging as the last. Miner's rule operates on the hypothesis that the portion of useful fatigue life used up by a number of repeated stress cycles at a particular stress is proportional to the total number of cycles in the fatigue life, if that were the only stress level applied to the part as shown in Figure 9.
Miner's Rule does not take into account the sequencing or order in which in the cyclic loads are applied. For example, if loads are applied in the plastic region, the endurance limit is no longer in effect. For example, if a large cyclic load is applied for certain time and then smaller cycle loads are applied still the material may fail although damage index is showing less than 1. (Siemens PLM Community technical article)
To overcome the drawbacks of the Miner's rule model, probabilistic models can be used in data analysis for accelerated life testing.

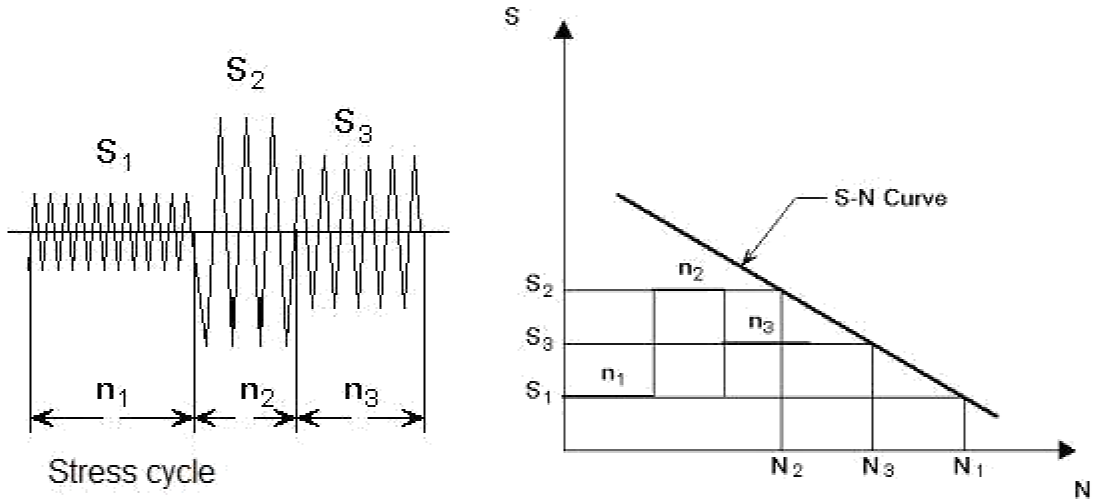

Figure 9 – Miners rule plot of cumulative number of effective cycles (courtesy CAE associates)

Coffin-Manson Law: Low Cycle Fatigue

For the elastic part, the relationship between strain amplitude and fatigue life can be approximated by

$$\varepsilon a^{el} = 1.75\,(\frac{\sigma UTS}{E})Nf^{-0.12}$$

The fatigue life in the plastic part can be approximated by

$$\varepsilon a^{pl} = 0.5D^{0.6}Nf^{-0.6}$$

Where

D is the ductility, defined as
D = ln (A0/ Afrac) which is close to εfrac
σUTS – Ultimate tensile stress
Nf – number of fatigue cycle
εa^{el} and εa^{pl} are elastic and plastic strains

The Coffin – Manson relationship is combination of elastic and plastic strains and shown as -

$$\varepsilon a^{e} = 1.75\,(\frac{\sigma UTS}{E})\,Nf^{-0.12} + 0.5D^{0.6}Nf^{-0.6}$$

Strain-life (ε-N) method

A mechanical part is often exposed to a complex, often random, sequence of loads, large and small. In order to assess the safe life of such a part using the fatigue damage or stress/strain-life methods the following series of steps is usually performed:

o Complex loading is reduced to a series of simple cyclic loadings using a technique such as rain flow analysis; o A histogram of cyclic stress is created from the rain flow analysis to form a fatigue damage spectrum;
o For each stress level, the degree of cumulative damage is calculated from the S-N curve;
o The effect of the individual contributions is combined using an algorithm such as Miner's rule.

Due to the proportionality between stress and strain, high cycle fatigue can also be expressed as strain amplitude vs. number of cycles. High cycle fatigue can be approximated by equating the total strain to just the elastic strain. Using this approximation,

1⁄2Δεelastic ≡ σf'⁄E(2Nf) ^-b
where

$\Delta\varepsilon$elastic is the change in elastic strain per cycle
$\sigma f'$ is a parameter that scales with tensile strength obtained by fitting experimental data
E is the Young's modulus
Nf is the number of cycles to failure
b is the slope of the log-log curve again determined by fitting
The Figure 10 shows high cycle fatigue as the right-most linear portion

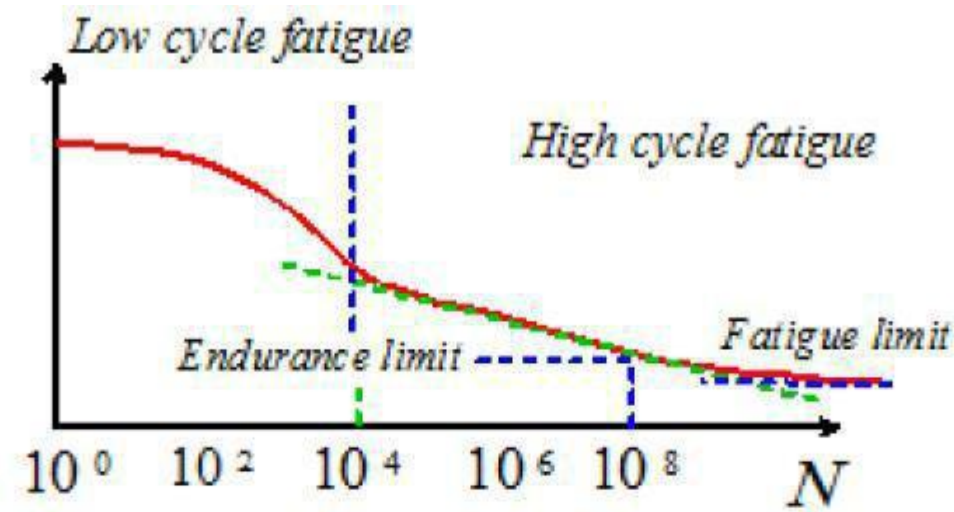

Figure 10 -Fatigue cycle vs. strain curve with Low and High cycle fatigue regions (Courtesy – Applied Mechanics of Solids (A.F. Bower) Chapter 9:

Rain flow counting method

The "Rain flow" counting method is used in the analysis of fatigue in order to reduce a spectrum of varying stress into a set of simple stress reversals. (Tom Irvine notes rev G). The counting is carried out as per standard ASTM E 1049-85 (2005) - Rain flow Counting Method. Its importance is that it allows the application of Miner's rule in order to assess the fatigue life of a structure subject to complex loading.

<u>Procedure:</u>

- Reduce the time history to a sequence of (tensile) peaks and (compressive) troughs.
- Imagine that the time history is a template for a rigid sheet (pagoda roof).

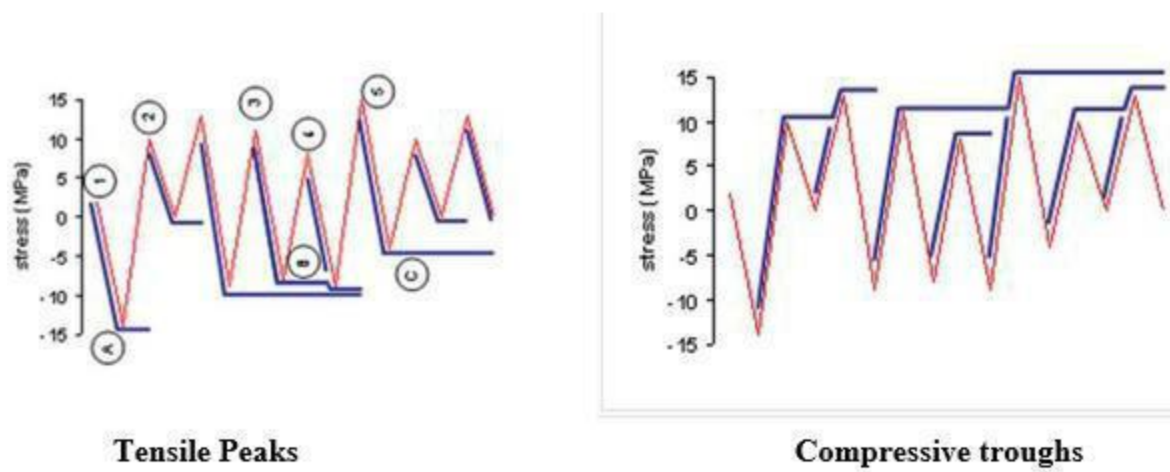

Figure 11 - Rain flow counting from stress cycle, Picture courtesy - www.aerospacengineering.net

- Each tensile peak (left in fig 11) is imagined as a source of water that "drips" down the pagoda.
- Count the number of half-cycles by looking for terminations in the flow occurring when either: o
It reaches the end of the time history;
- It merges with a flow that started at an earlier tensile peak; or
- It flows opposite a tensile peak of greater magnitude.
- For compressive troughs step 4 above is repeated. (Right in Figure 11)

- Assign a magnitude to each half-cycle equal to the stress difference between its start and termination.
- Pair up half-cycles of identical magnitude (but opposite sense) to count the number of complete cycles.

Although most engineering structures and components are designed such that the nominal stresses remain elastic, local stress concentrations often cause plastic strains to develop in regions around them. The strain -life method assumes that the smooth specimens tested in strain control simulate fatigue damage in local region around the stress concentration.

Use of the strain-life analysis method is limited to situations where crack initiation and the growth consume the majority of the service life. Strain-life curves and cyclic stress -strain curves are needed for this analysis Fatigue usually starts at the surface so that the quality of the surface finish is very important. The surface finish becomes even more important as the strength of the material increases. All mechanical components are structures contain some form of stress concentrators which can cause cracks to form. The theoretical stress concentration depends on geometry and relates the local maximum stress to the nominal or average stress through a stress concentration factor. Small stress concentrations are less effective in fatigue than predicted by Kt. A fatigue notch factor (effective stress concentration in fatigue) is used to account for this effect. It is related to the size of the local stress gradient and material strength

Strain raisers in component design

It is imperative to study the cumulative effect of all stress / strain raisers in torsional vibration analysis. Stress Concentration Factors Kt for Keys for key-seats cut by standard end-mill cutters, with a ratio of r/d = 0.02, Peterson's charts give

- Kt = 2.14 for bending
- Kt = 2.62 for torsion without the key in place
- Kt = 3.0 for torsion with the key in place

Keeping the end of the key seat at least a distance of d/10 from the shoulder fillet will prevent the two stress concentrations from combining both effects. The fillet radii of the motor shaft must be onerous from rotor drawing. Unless the torsional analysis reports are satisfactory, key way milling should not carry out.

The stress concentration factor for a keyway can be determined from the chart given in Peterson handbook – page 409 Chart 5.2

In addition to other factors described in machine design handbooks, Surface Finish, ka, Size Effect, kb: Reliability Factor, kc is also to be considered.

Combining the above factors generates the modified endurance limit is calculated which gives the realistic information on design:

S'e = ka x kb x kc x Se

API MANDATES ON TRANSIENT TORSIONAL VIBRATION ANALYSIS

API 617 8th edition

For centrifugal compressors – mandates that the supplier shall perform a transient torsional vibration analysis for synchronous motor driven units, using a time- transient analysis. The maximum torques are to be used to evaluate the peak torque capability of coupling components, gearing, and interference fits of components such as coupling hubs.

The transient analysis shall generate the maximum torque as well as a torque vs time history for each of the shafts in the compressor train. The torque vs. time history is used to develop a cumulative damage fatigue analysis of shafting, keys, and coupling components. An appropriate cumulative fatigue algorithm shall be used to develop a value for the safe number of starts. The safe number of starts shall be as agreed by the purchaser and supplier. For VFD driven equipment trains, the supplier shall report excitations along with -

- Integer orders of the drive output
- Sidebands of the pulse width modulation.

The transient stress vs speed for selected shaft sections with high cycle fatigue (HCF) (endurance limit) and the low cycle fatigue (LCF) limits shall be identified along with Results of the damage accumulation calculations as a function of one start. Predicted number of starts to failure for each shaft, coupling(s), and gear mesh.

c) A summary shall be included identifying that the shafting, coupling(s), and gear mesh have a finite number of fault cycles.

API 610 11th edition

Centrifugal pumps for refinery services mandates that if the driver is a synchronous motor rated 500 kW (670 HP) or higher, a transient torsional analysis shall be performed.

API 619 5th edition

Screw compressors for refinery services - The standard mandates that vendor shall perform a transient torsional vibration analysis for synchronous driven units and/or variable speed motors. The acceptance criteria for this analysis shall be mutually agreed upon by the purchaser and the vendor. Transient torsional analysis is to be carried out for all synchronous motor-driven units.

API 546 3rd Edition

Brushless Synchronous Machines - 500 kVA and Larger- lays the design guideline and states that the motor should be able to start at 80 percent terminal voltage. As air gap torque varies with voltage-squared, the air gap torque is reduced to a factor of (0.80) ^2 or 64 percent.

It also states that a 10 percent margin needs to be maintained between the mean motor torque and load torque for all speeds between zero and synchronous speed. Thus, if the train load torque (inclusive of the driven power including gear box and bearing power loss with windage loss etc.) at the point where the load torque curve is at closest proximity with the mean torque curve is, say, 0.45 P.U., then the mean motor torque needs to be at least 0.45/ {(0.80) ^2/ 0.90} = 0.78 P.U. at this condition to satisfy these requirements.

API 541 5th Edition

Form-wound Squirrel Cage Induction Motors—375 kW (500. HP) and Larger. - mandates as when specified, the assigned vendor(s) shall perform a steady-state and transient torsional and stress analysis of the complete electrical and mechanical equipment train Most point addressed in this standard is the "Design Review"—this is a comprehensive review meeting where the detailed electrical and mechanical designs are discussed, and any electrical/mechanical analyses are presented. This is recommended for critical motor/driven equipment trains, ratings above 3000 hp, or for new manufacturer designs (prototypes). This meeting is not the "order co-ordination meeting." The purchaser and vendor representatives meet after preliminary engineering is complete (normally 4 – 6 weeks after placement of order) to review designs and coordination with other associated equipment.

API RP 684 2nd Edition

API Standard Paragraphs Rotodynamic Tutorial: Lateral Critical Speeds, Unbalance Response, Stability, Train Torsional and Rot or Balancing, recommends following and addresses that only transient analysis normally performed is that of a synchronous startup of a synchronous motor. The document lists all possible transient faults for VFD, and induction motors however concludes as - these fault conditions are rarely analyzed for motor drives. The electrical switch gear must detect such faults so rapidly so as not to endanger the motor or connected machinery, however as a precaution it is prudent to avoid system torsional natural frequencies at one- and two-times electrical line frequencies.

To analyze the effect of transient torsional vibrations, an appropriate cumulative fatigue algorithm shall be used to develop a value for the safe number of starts. The safe number of starts shall be as mutually agreed by the purchaser and vendor.

STRATEGY TO ADDRESS TRANSIENT TORSIONAL VIBRATION DURING ENGINEERING PHASE

The simulation of torsional vibration based on electric drives is performed in two steps, separating the mechanical and the electrical part of the system. VFD non-integer excitation data are to be provided by VFD manufacturer. The torsional vibration analysis (TVA) is usually carried out by the packager and, based on its results, source current pulsation analysis (SCPA) is performed by the electric motor manufacturer

Consideration of Damping

Normally 1% damping is considered as conservative rule. We need to consider pump impellers as external dampers. The connection can be modeled as linear dashpots between the disk and ground with damping proportional to the disk's rotational velocity as:

$\tau ext = cext \times \omega$

where

τext is excitation torque , cext is external damping and ω is disk rotational velocity in rad /sec

Fluid-film journal bearings also produce external damping as they generate a squeeze -film effect. Gear meshes exhibit such motion with coupling between lateral and torsional modes. This can cause the bearing to be a significant, even predominant, damping source, even 10% of the critical damping coefficient. (Mark A. Corbo 1988)

One primary source of damping in turbo machinery is the torque versus speed characteristic of the driven load. If resisting torque increases with speed, the load is a positive-damping source and damping coefficient is instantaneous value of the torque -speed curve at that point. (Mark A. Corbo et al 1996)

Number of vanes of pump driven by a VFD Motor –The six-step inverter uses six silicon-controlled rectifiers (SCRs) to change the power from DC to AC. Firing of the SCRs is controlled electronically to produce a stepped waveform that simulates a sinusoidal variable-frequency waveform. The six-step inverter causes 5th, 7th, 11th, 13th harmonics on the current waveform and 6th, 12th, 18th harmonics on the output torque. A rotating impeller produces a circumferentially varying pressure field, between vanes, at its exit. The periodic pressure fluctuations interact with the stator, usually a volute or a diffuser. This rotor/stator interaction (RSI) can combine and amplify pressure pulsations and pump or piping vibrations. When an even vane number impeller is paired with a double volute, vane pass pressure pulsations can reinforce each other. It is imperative that we always consider od d numbers of vanes so that odd number VPF does not get coincides with 6X harmonics of VFD driven motor.

Number of starts based on fatigue life estimation - In the case of synchronous motors start, certain stresses may exceed the yield strength of the material in shear. If the potential number of transient torsional events can be reliably estimated, then it is acceptable practice to design for a limited number of starts to avoid exceptionally large shafts and couplings Even starting a motor once a week for 20 years would only result in 1040 starts. This philosophy is recognized and accepted in API RP 684 standard.

Use of software logic - If at final torsional study concludes that the torque capability of the train components regarding the infinite life fatigue is lower than the initially assumed torques, an additional measure such as the realization of mini- exclusion ranges within the operating speed range may be necessary. This can be implemented in plant with separate software logic. To exactly locate the TNF, it is then important to have full load string test with variable speed – load combination. When the TNF values are found in TNF measurement at OEM works, the software can generate logic to ensure that machine operates outside the forbidden band of +/- 10% of as found concerned modes of TNF.

Momentary torque in coupling design – This limit as defined by API 671 4th edition -point 3.25 can be useful to define material yield strength of coupling. It is to be noted that Peak torque rating is 133% of maximum continuous torque rating. Peak torque rating corresponds to a factor of safety of 1.15 with respect to the components' material yield strength. Maximum momentary torque rating is normally 176% of maximum continuous torque rating. Maximum momentary torques is defined as torsional impact type loads which may occur for one brief duration (such as short circuits). Maximum momentary torque rating is determined by the manufacturer to be the torque capacity that a coupling can experience without ultimate failure, where localized yielding (damage) of one of its components may occur. A coupling can withstand this occurrence for one brief duration. After that, the coupling should be inspected and possibly replaced. This is also sometimes called the short circuit torque rating. This information should be used with careful deliberation while analyzing the torsional system for a short circuit response.

ACTIONS TO MITIGATE HIGH TORSIONAL STRESSES AT TRANSIENT CONDITIONS

Before start of purchase of machinery and design of train, if there is concern of incipient short circuit events, the electrical fault analysis is to be carried out although not addressed in relevant API standard during initial design phase. The reason for that is so the results of the analysis, can be considered during the design phase of the equipment. If the equipment is largely complete and through the manufacture and test phase and such analysis is carried out on that stage, the results of the analysis would then need to be considered after equipment has already been manufactured, with potential for significant commercial impact(s) should any design changes be desired given the results of the analysis. Couplings, as part of turbomachinery train have a bigger influence for design optimization and mitigating high torsional oscillation affecting the drive train. (Stephen R. Locke et al 2013) Here are the plausible design actions which can be exercised to mitigate the effect of transient events during machinery train detailed design. A more detailed techno- commercial treatment of this subject can be found in tutorial by Martin D. Maier et al 2016.

Possibility of optimizing the inertia of driven machinery

One of the major issues encountered from end user specification where API bulleted points are placed from "if applicable "to mandatory requirement. For example, a synchronous motor driven compressor driven inertia is increased by using an API bulleted clause to size the gear box according to motor name plate rating where extra margins are taken over 10% API margin. With this high inertia of driven equipment, the train takes a larger time to reach synchronous speed and dwelling time in torsional frequency becomes more.

Soft coupling

A coupling with rubber blocks in compression as shown in a generic Figure 12 (elastomeric type damper coupling) generally has a lower torsional stiffness than a steel flexible disc coupling and provides additional damping. This type of coupling technically works as torsional damper. (Mark Corbo et al 2000, 2002) The damping, limits the dynamic torque when operating near resonance. The torsional stiffness of the coupling is non-linear and sensitive to shore durometer (hardness) of the rubber blocks. Although they are a "tried and tested" option, the disadvantage of elastomeric couplings is that they are viewed as high maintenance items as the blocks need to be replaced approximately every 3-5 years. Elastomeric couplings are also temperature sensitive, dimensionally larger, heavier and produce more windage compared to conventional flexible element couplings.

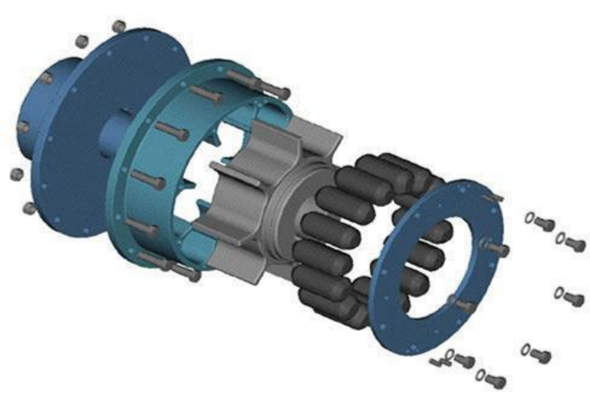

Figure 12 – elastomeric damper type soft coupling (Courtesy Renold Hi-tech coupling)

The Table1 shows the comparison of soft / near soft and stiff couplings where properties are variable over ambient / working temperature as a sample case.

Coupling	soft; rubber-in shear			steel-spring	stiff: disc pack
	cold	nominal	Warm		
k_C – kNm/rad	136	106	72	124	15,117
1st TNF, ω_0 - Hz	5.58	4.93	4.04	14.75	77.98

Table 1 – effect of torsional properties of couplings in various ambient temperatures

In this context, API RP 684 2nd edition quotes as Elastomeric couplings are more complex to model as the torsional stiffness is nonlinear. The torsional stiffness of an elastomeric coupling is a function of the transmitted torque. A typical elastomeric coupling torsional stiffness versus torque curve is shown in Figure 4-8. Reproduced below as Figure -13.

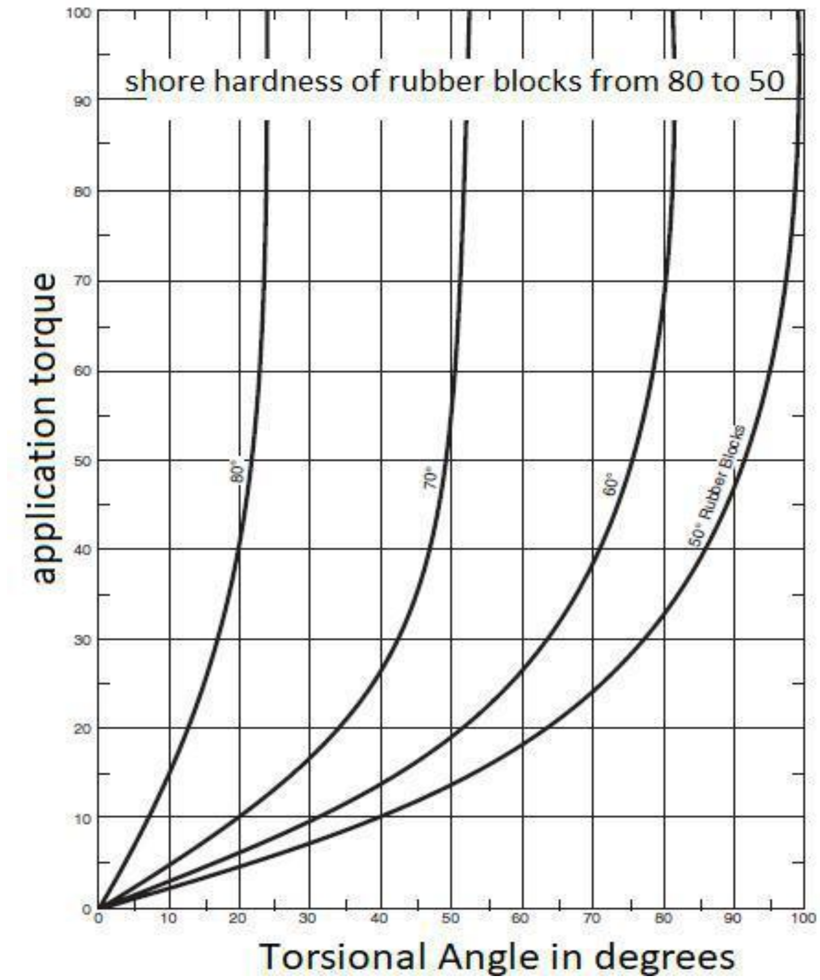

Figure 13 - Typical Non-linear Stiffness vs. Torque for an Elastomeric Coupling (source API RP 684 2nd edition)

It can be seen from Figure 13 that the coupling stiffness is dependent upon the elastomer used in the coupling. The couplingmanufacturer can provide the stiffness at a given torque or provide an explanation of how to calculate the stiffness from the nonlinear torque deflection plot. In addition, for transient conditions, the coupling vendor can recommend appropriate stiffness and damping to be used fora transient analysis. Also, if the torque supplied by the driver has a significant variation then the system may have to be analyzed with the minimum and maximum torque conditions to define a range of frequencies for a given mode. (Oliver Doidge 2018). However, many issues have been resolved by using such type of coupling when electric supply line had some fluctuations causing high torsional oscillations. (Martin Leonhard et al 2001)

Hydrodynamic (Fluid) coupling

The most practiced and believed opinion is that hydraulic couplings should be treated as zero spring rate elements that effectively divide the assembly into two independent torsional systems. But in reality, zero spring rate model is only an approximation since, as, restoring forces in the fluid yield a small but finite spring rate. These restoring forces are generated by centrifugal forces in the fluid and are, therefore, proportional to the square of running speed based on the following equation for effective stiffness: kh = D^5 • RPM^2 / Constant

Where

Kh is Hydraulic coupling spring rate,
D is Outside diameter of coupling impellers
RPM is Speed of driving member (rpm).

Compared to other drive elements, the boundary value of the stiffness is very low. For example, any type of coupling used the stiffness is ten times higher than that of the hydrodynamic coupling. Even stiffness of highly flexible couplings is higher by a factor of 3 at an identical nominal torque.

This coupling has the beneficial aspect of effectively dividing a torsional system into two independent systems. (Mark Corbo et al 1996). This permits the isolation of vulnerable components from significant sources of vibration such as synchronous motors. In addition to its isolation characteristic, the fluid coupling also pro vides damping to the machine by virtue of viscous friction between the impellers and coupling fluid.

c = Constant * (1+1/n^2) *HP / RPM^ 2

Where:

c = Damping coefficient,
n -Order number of modes being excited
HP -Average power transmitted through coupling,
RPM -Speed of driving shaft (rpm).

Only issue with hydraulic coupling is limit in installed power range – approximately 6 MW as driver power has been tested so far . Another issue with hydraulic coupling is higher journal loading on DE shafts due to high overhung and reliability of associated bearings.

Variable speed planetary gear drive -By the use of variable planetary gear the motor torque during start-up is approx. 18 % of the rated torque at full speed. This reduces starting torque and allows a reduced voltage start-up of the motor. This drive consists of a planetary gear and a hydrodynamic torque converter in the" power-split" branch as shown in Figure 14. This device is mechanical replacement of VFD drive.

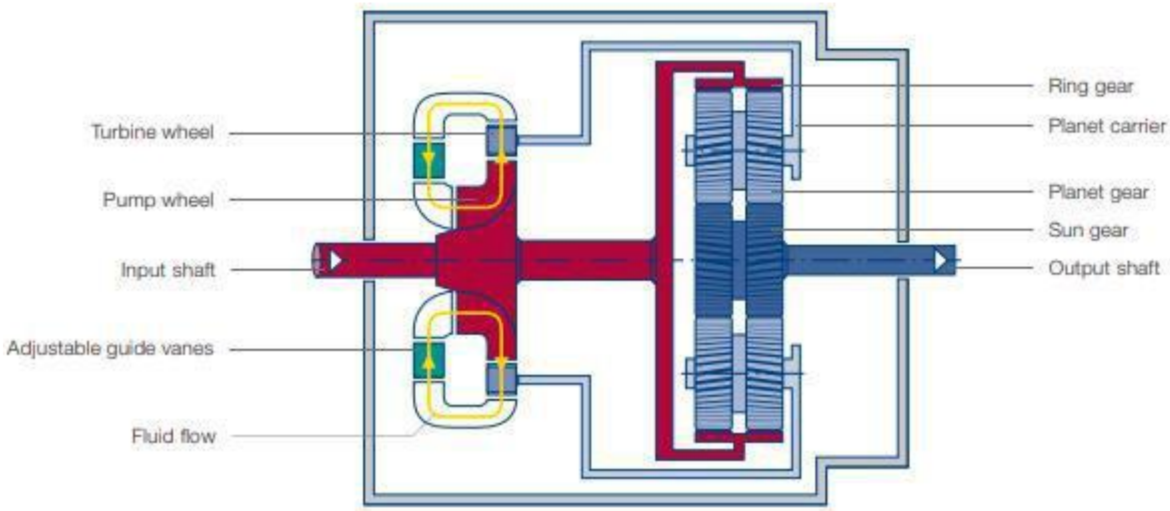

Figure 14 -Variable speed planetary gear drive (Source Vorecon) for illustrative purpose

Large spacer coupling - This type of device is used to mitigate high torsional oscillations particularly encountered in synchronous motor drive trains. As the coupling spacer length is increased as shown in Figure 15, the 1st two torsional modes go down and shifts away from forcing frequencies. The below equation for a hollow pipe holds for a coupling spacer.

Φ = TL/Ip. G, Kt = T / Φ = Ip.

G/L Where

Φ is the angle of twist in radians when a torque T is applied.
Ip is polar moment of inertia and
G is shear modulus of material

It can be seen that increasing the length of spacer L shall lower the torsional stiffness of spacer and overall TNF of train. Negativity in design are - Higher overhung and higher windage loss that lead to high coupling enclosure temperature. Hence, designer should explore possibility of using quill-shaft coupling. The thin shaft torsionally isolates the high-speed compressor from forcing frequencies such as gear mesh frequencies or driver pulsations.

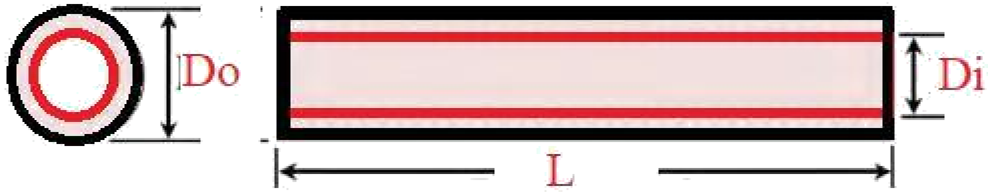

Figure 15 – large spacer (tubular shaft) of a coupling

Shot peening –

This process is very useful to increase the fatigue life of a localized area where chances of fatigue failures are high . This is a specialized process and type of parent material , process of application are of prime importance **.**

Keyed versus keyless connection

Keyed connection with shaft and coupling hubs are relatively simple and reliable when transmitting smooth, consistent power. However, they are sometimes found to be inadequate when vibratory, shock, or reversing loads are present.

Keyless connections rely on a clamping force to hold a shaft in a bore. A keyed connection will provide a positive stop until failure, whereas a keyless connection could allow slippage between the two mating parts if it exceeds the design torque. Failure of shaft ends with keyway are more damaging as keyed connection need to be design with proper fillet, depth and stress concentration factors. A keyless connection can transmit more torque than a keyed shaft due to more shaft surface contact.in comparison to the contact area of a shrink fit connection to a keyed connection.

To protect driven parts of train from high torsional oscillations, Integral flange couplings are also a very favourable choice which are forged design from shaft as shown in Figure 16.

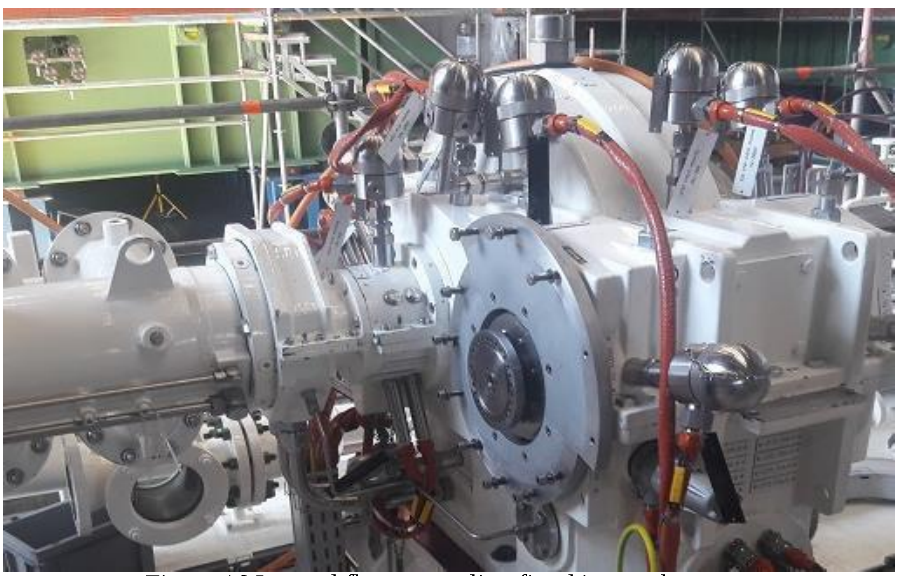
Figure 16 Integral flange coupling fitted in gear box

Mechanical Fuse type (shear pin type) coupling

A shear pin arrangement consists of a series of pins located in 2 flanges, the pins having a calculated groove diameter, positioned on a common pitch circle. A bearing is included to control rotation after disconnection the shear pin type couplings are designed to physically disconnect the driven and driver in such point of time where high shear stresses generated can be detrimental to keyways and crucial parts of machinery train. The construction of shear pin type coupling is shown in Figure 17. Thus, they protect the components from torque overload. They are designed to control the peak torque the train component might see during a torsional transient. The shear pins can be solid or hollow with a pre-determined design value. At higher than design, the coupling halves are disengaged but do not wobble as they rotate on radial ball bearings.

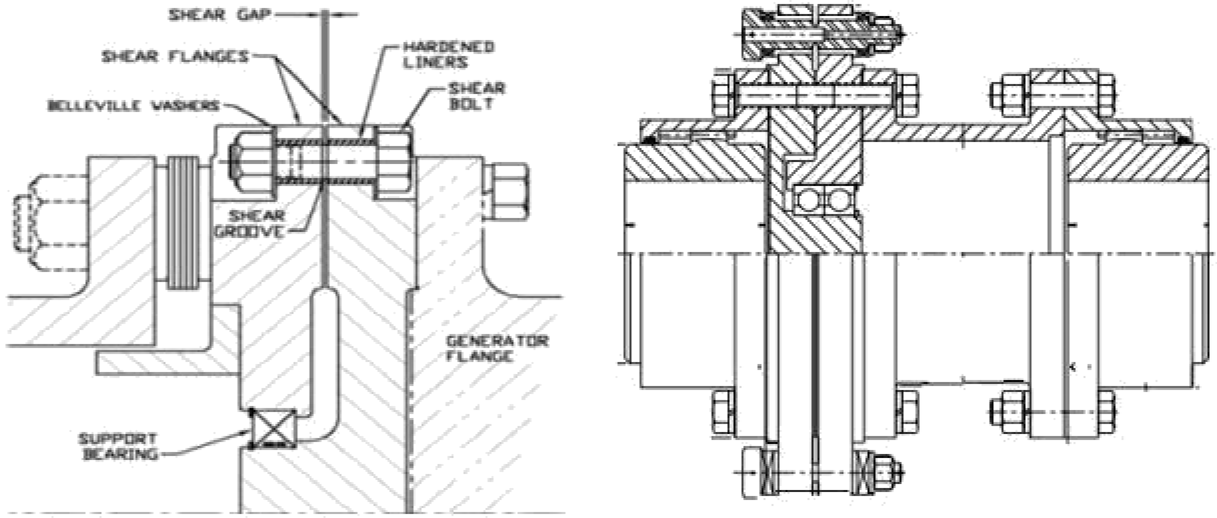

Figure 17-Shear pin type coupling with support / Ball bearing design (courtsey Ameridrive left & Kopflex right)

Once this event is occurred, the machine is shut down and shear pins are replaced, and operation can be restarted after proper identification of root cause and visual checking of ovality of coupling bolts and shea pin holes. This arrangement drastically minimizes the equipment downtime.

To ensure disconnection accuracy, the precise shear of multiple pins concurrently is required. This requires t rue position of the pin holes in both flanges to ensure all pins have equal torsional loading. The rigidity of the shear pin arrangement is resultant upon the fit of the shear pins within the holes and the robustness of the bearing arrangement. A flexible joint can cause bending fatigue upon the shear pins and an increased face run out; imposing axial thrust loads upon attached machinery.

Following preliminary guideline are (but not limited to) to be considered before selecting this type of mechanical fuse / torque limiter couplings –

Shear pins shall be designed for maximum operating / rated torque only and should fail simultaneously (all shear pins together) in case of torque reversal during a short circuit at any operating speed. However, it is to be noted that in normal trip scenario (which shall happen during operation of compressors with a residual flux in motor stator), these shear pins should not fail or deteriorate. This means that these shear pins are provided only for component protection in the event of short circuit.

The failure of shear pins shall be fast enough and at only such torsional value where other components such as key way, coupling hubs and spacers remain completely non-responsive. There may be some distortion in shim packs during this type of failure. The couplings with shear pins shall be designed with an arrangement to prevent spacer oscillation after shear pin breakage by means of radial ball bearing arrangement. The shear pin arrangements should ensure that pin replacements are fast and easy. For safety, the shear pins shall be wired together so that they do not get dislodged. The coupling also incorporates an anti-fly system via a bearing arrangement internal to the coupling. This ensures no other drive -train components are damaged by flailing parts when the pins are sheared during a motor fault event

Shear pins the failure (breakage) occurs at the ultimate strength not at the yield strength. OEM should use the most conservative failure theory to jointly validate the shear pin design by coupling manufacturer. Shear pins shall not fail prematurely due to fatigue. This means they shall fail when required per design.

During a short circuit response analysis, when we find response result shows that for both 3 phase and 2 phase faults, the mean shaft shear stress is below the shear yield strength of the shafts. However, the standard flexible coupling peak torque limit is exceeded and for the 2-phase fault, the motor shaft localized stress limit is also exceeded then it is imperative to select torque limiter coupling.

After selecting the above type coupling OEM shall submit a confirmation that change in coupling type shall not shift TNFs (Torsional natural Frequencies) more than 3% of original value as reported in earlier torsional analysis studies.

Installation / Fitment requirement of shear pin type couplings

The finite element analysis considering pure shear loading effect and the various levels of combined shear and bending loads (S. Sankar et.al 2011) shows in Figure 18 that the maximum stress is found along the neck region of the shear pin. The shear stress for the pure shear loading, in an analysis of installation was found lesser with negligible misalignment and it is found to be much higher for 200 um misalignment which exceeds the nominal shear stress for subject shear pin. (Jon R. Mancuso et al 2010)

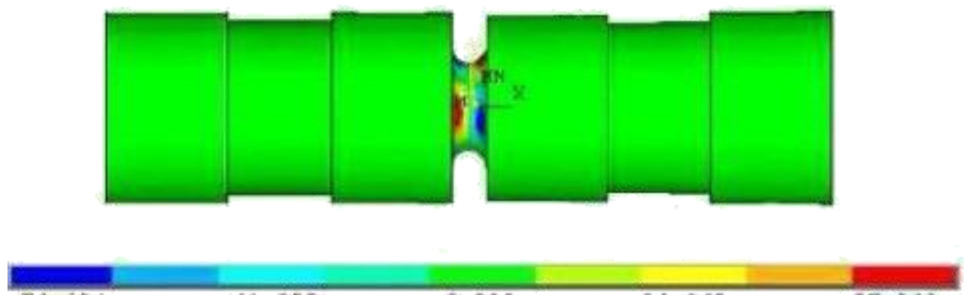

Figure 18 -Sample Shear pin stress profile of neck. (source – S. Sankar et al 2011)

Further, according to maximum shear stress theory the material shear stress is lesser than von Mises stress which is more than the material shear stress. It is well understood that the frequent failure of the shear pin happens due to the combined bending and shear loading which caused reversed fatigue loading in the shear pin. Due to misalignment and incorrect neck diameter, un warranted fatigue loading is produced with combined bending and shear load, which causes the pre-mature failure of the shear pin.

ELECTRICAL SYSTEM DESIGN ASPECTS

When short circuit torque values are provided by motor OEM it is imperative that OEM confirm that Motor shaft and coupling with keyways / shrunk fit / integrally flanged can withstand 2 phase and 3 phase short circuit event.
While drafting project specification for scope of rotor-dynamic analysis following aspects are considered - The motors have enough protections as IP65 rating to have short circuited due to external ingress, 3 phase balance is ensured by monitoring, IGBT (Insulated Gate Bipolar Transistor) protection has been employed. In case of variable speed applications, the motor is galvanically isolated from the supplying grid by a VFD. The motor is protected by the integrated VFD protection system which is based on the current measurement; in the event of short-circuit high over-current is detected resulting in a system trip, which usually takes place within milliseconds or in shorter time. Based on these considerations, project specification sometimes does not ask for a transient short circuit torsional analysis.

VFD type and protection

The currently used VSI designs provide many advantages over their LCI predecessors, including low power system harmonics, low reactive power demand, and low torque pulsations in the motor air gap and shaft. VSI designs have lower harmonic distortion and air gap torque ripple than LCI
Event time line of a Case when a fault current is passed thru a VFD –
The drive will detect an output short circuit and turn off the IGBTs. This limits the output current to about 1.8 times the drive rating, and it clears in milliseconds.
If there are internal short circuits in the drive, then the drive relies on the Overcurrent Protective Devices (OCPD) ahead of the drive. If adequate fuses are used, these will clear quickly, and the fuse let -through will be according to the proven and published curves for those fuses. Currents will be further limited by any AC line reactor or DC choke that may be used.
Here are some possible effects for internal short circuits in VFD drive:
* Most common -If an input diode shorts, OR the DC capacitors short, OR two IGBTs (insulated gate bipolar transistor) (in one leg short, THEN a line to line short results on the drive input, the fuses clear, and there is no output current.
* Rare if there is one IGBT that shorts AND the other IGBTs are not shorted AND there is a ground fault on the phase with the shorted IGBT, then, a Half Wave rectified dc short results
The fuses or ground fault relay are relied upon to clear. The bond wires in the IGBTs or diodes may clear before the fuses do. During short circuit the current rises so rapidly exceeding the (di/dt) rating of the IGBT and thus destroying it. Any Normal VFD will protect the motor from overloads. In every power semiconductor like thyristor or transistor once there is a trigger from the gat e, the expected output current will reach immediately. The full conduction gradually increases to the full surface area of the semiconductor in very short period.
In case of short circuit, the current density on the portion of the semiconductor nearer to the gate is so huge as to melt it. Generally, this is prevented by adding additional reactors in series with the VSD output so that the Di/Dt will be within limits. Some VSD manufacturer have built in reactors and test certificates to show this short circuit with standing capability can be got from them. The short circuit current is determined by the total line impedance from the transformer to distribution point until motor and that of the supply transformer rating itself. In present Oil and Gas scenario, most of case VSI drives are used with multiple IGBT protection with cell by pass features which improves the reliability and availability to very high level. The protection relays along with VFD system have fault level – 5-10 kAmp and they are essentially type tested. Hence, any incoming disturbances from feeder and VFD are controlled by very fast switching of IGBT. (Dara O'Sullivan Analog devices)

In built IGBT and Short-Circuit Capability-

The short-circuit withstand time of an IGBT is related to its transconductance or gain and the thermal capacity of the IGBT die. Higher gain leads to higher short-circuit current levels within the IGBT, so clearly lower gain IGBTs will have lower short-circuit levels. Schematic of an IGBT (Insulated Gate Bipolar Transistor) is shown in Figure 19.
In present Oil and Gas scenario, most of case VSI drives are used with multiple IGBT protection with cell by pass features which improves the reliability and availability to very high level. The protection relays along with VFD system have fault level – 5-10 kAmp and they are essentially type tested. Hence, any incoming disturbances from feeder and VFD are controlled by very fast switching of IGBT.

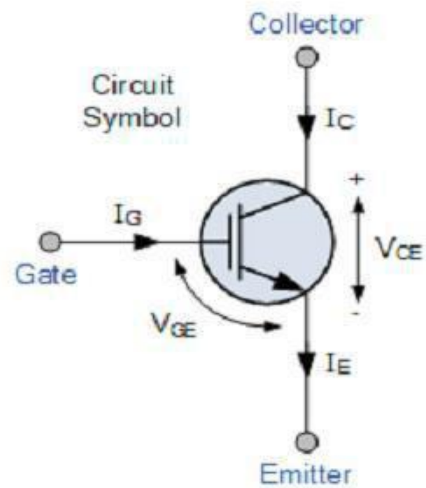

Figure 19 – Schematic of an IGBT, (Courtsey - https://www.electronics -

tutorials.ws) ***Rotor Construction for synchronous motors***

There are two major types of rotor construction: laminated pole and solid pole. Laminated pole motors generate lower levels of pulsating air gap torque than solid pole motors which results in lower startup torques and associated stress levels. However, laminated pole motors are subject to mechanical stress limitations which limit them to low speed applications or started or stopped frequently against a seized drive. Solid pole motors are cheaper in comparison to laminated pole motor. The low- speed shaft (comprising motor shaft, low speed coupling and bull gear), always has higher torques and usually has the highest steady and alternating torques from the synchronous motor. The startup for synchronous motors started across -line, keyed shaft ends are usually avoided due to fatigue considerations. Hydraulic fit shaft ends are frequently limited by slip capacity. Integral flanged shaft ends are best suited for the motor and gear shafts as they provide adequate torque capacity with minimal fatigue concerns. This avoids any stresses associated with shrunk-on hubs or keyways.

PREVENTIVE MAINTENANCE AND MONITORING OF MOTOR HEALTH TO MITIGATE SHORT CIRCUIT

The instruments are available as an early warning device to indicate the condition of the machine and allow the user to act before it fails. The IEEE state that 80% of electrical failure can be prevented by predictive / preventive monitoring / maintenance. Following are the devices which are used / can be used as preventive measures for short circuit to happen inside motor.

Motor Current Signature analysis – (Normally mandated in Project specification by end user)
The most common causes of induction motor failures are due to breaks in rotor bars or shorted circuit rings due to rotor eccentricity. Motor Current Signature Analysis (MCSA) is used to accurately measure faults in the induction motor rotors through changes in the rotor magnetic field.
Current Signature Analysis is important in tool in preventing:

- o Broken bars that cause rotor core damage due to temperature increases and arcing. o Premature degradation of bearings due to torque and speed oscillations on the rotor.
- o Stator failure when broken rotor bars can lift out of the rotor slot due to centrifugal forces.
- o Rotor eccentricity issues due to unbalanced magnetic forces

It is based on the general principle that any disturbance that the motor is subjected to will show an electrical signal in the stator current. For example, a broken rotor bar will result in a counter-rotational magnetic field, which will be reflected in the stator current. Current Signature Analysis determined from the spectrum of the signals embedded in the stator current, and by understanding induction motor and load dynamics it is possible to determine the nature of rotor problems. It uses the fact that broken rotor bars will provide a tell-tale signature in the frequency spectrum with spectral lines at harmonics of twice the slip frequency above and below the

50 Hz or 60 Hz fundamental frequency. This is usually 1-2Hz on either side of the fundamental frequency (depending on the slip). As soon as the amplitude exceeds a certain empirically determined threshold (relative to the amplitude of the fundamental), it indicates a failure condition may be likely.
Torsional resonances may incur in induction drives due to inadvertent loose wiring in electric motor or large harmonic torques produced by the inverter. (Clay McClinton et al 2014) The torsion can affect the rotor-load system and resonances may arise in the rotor end-rings of fabricated-cage induction motors. Since the high-frequency speed oscillations are difficult to detect, torsional resonances may go unnoticed and lead to drive fault on the long run. There is a non-invasive technique for early detection of high - frequency speed fluctuations in case of mechanical resonance, by using motor current signature analysis. The rotor slot harmonics are originally used here to detect drive speed fluctuations.

Partial discharge monitoring – (Normally mandated in Project specification by end user)
Partial discharges (PD) are small electrical current sparks that occur in the high voltage electrical insulation in stator windings whenever there are small air gaps or voids in or on the surface of the insulation. As a stator winding deteriorates from coil winding vibration in the slot, operation at high temperatures, or contamination, the partial discharge activity increases. Thus, partial discharge monitoring can be used to detect the primary causes of stator winding failure and generally give two or more years of warning of a machine failure. So, this condition monitoring system can prevent events of short circuit.
Partial discharge monitoring can be performed as an off-line quality control test during machine outages. An off-line instrument / probe can accurately pinpoint the source of partial discharge (PD) in a particular slot in generators and motors.

Online / offline Rotor flux monitoring system- (Normally mandated in Project specification by end user). Rotor magnetic flux variations are the result of the degradation of the rotor winding insulation by mechanisms like load cycling, thermal aging and contamination which may lead to short circuit events in turbomachinery trains. Rotor flux monitoring using air gap flux probes is the most powerful means to determine if turn-to-turn shorts have occurred in the rotor winding of synchro nous motors. This information is critical in planning maintenance, explaining abnormal vibrations and verifying new and rewound rotor integrity. A turn-to-turn short is the most frequent rotor insulation failure mechanism and can result in:
Thermal imbalance of the rotor pole, leading to increased mechanical
vibration Magnetic imbalance in the flux resulting in mechanical vibration
Rotor ground insulation failure
The flux monitoring solutions that include a flux probe designed specifically for the air gap distance in between the rotor and stator sending data to the instrument and diagnostic software. Flux monitoring identifies shorted rotor turns through the measurement of the local magnetic field emanated from each rotor pole to identify:

- Shorted turns on a rotor pole
- Variation of pole's physicalposition
- Out of round rotor shape
- Changes in the air gap between rotor and stator

CONCLUSIONS –
Transient torques / torsional vibrations are short lived but carry very high stress values. However, it is not prudent to overdesign the components to tackle these issues which adds extra capital cost, more operating cost such as bigger oil coolers or costly materials of construction.
A comprehensive review meeting should be carried out between driver and driven equipment supplier where the detailed electrical and mechanical designs are discussed, and electrical/mechanical analyses are presented. After that a preliminary train torsional analysis should be submitted to end user just after ordering phase. The preliminary report should be 90% accurate.
It should be noted that a short circuit fault may cause significant sparking in the hazardous area where the machinery is installed in an oil and gas plant, refinery or petrochemical complex. Hence a robust electrical protection along with condition monitoring system should be implemented.
In addition, proper shaft alignment must be addressed at operating conditions. The torsional vibrational behavior of shafts with parallel misalignment is studied by Diangui Huang 2005. The analytical results show that parallel misalignment will excite torsional vibration at 1x rotating frequency. The experiments show that if parallel misalignment exists, torsional vibrations at 1x rotating frequency are obvious, accompanied by smaller torsional vibrations at 2x, 3x, 4x, 5x, 6x and natural torsional frequency.
Taking consideration of electrical protection, statistical analysis of reported failure and number of starts stops, evaluating healthiness of existing grid can smoothen the process and optimal design can be proposed.
During second leg of analysis re- assurance in the design margins, taking full credit for the material certificate property values and using a proper (not conservative) reliability factor can give an increased safety factor.
During a motor terminal short circuit event, the drive will trip almost immediately (within 10ms) and is therefore protected from large fault currents. The drive itself can be ignored from the fault analysis.
The risk of a short circuit event can be further mitigated by ensuring that installation and maintenance procedures take proactive measures to minimize the risk such as:

- Ensuring terminals are correctly connected/tightened and that no tools are left within any terminal box.

- Ensuring that cables are run and protected appropriately from accidental damage and that the insulation is regularly inspected.

The final take away from this tutorial is to maintain a diligent prudence to be exercised to establish a good understand of subject machinery train between OEM and End User and EPC company . Occurrence rating criteria of a failure in FMEA. This shall avoid overdesigning the components / costlier material of construction . Higher lubrication oil consumption , larger cooler requirement also come up as domino effect of overdesigning the components .

NOMENCLATURE

AF	= Amplification factor
API	= American Petroleum Institute
cext	= External damping
D	= Outside diameter of hydraulic coupling impellers
Du	= Ductility
$\Delta\varepsilon$ elastic	= Change in elastic strain per cycle
E	= Young's modulus
[εa〗 ^el	= Elastic strains
[εa〗 ^pl	= Plastic strains
Fl	= Line frequency (Hz)
Fslip	= Slip frequency (Hz)
Φ	= Angle of twist in radians when a torque T is applied
G	= Shear modulus of material of coupling / shaft
HP	= Average power transmitted through coupling
ζ	= Damping ratio
IGBT	= Insulated Gate Bipolar Transistor
IEEE	= Institute of Electrical and Electronics Engineers
Ip	= Polar moment of inertia
Kh	= Hydraulic coupling spring rate
Kt	= Stress Concentration Factors
ka	= Factor for Surface Finish
kb:	= Factor for Size Effect
kc	= Factor for Reliability
LCI	= Load commutated inverter
MCSA	= Motor Current Signature Analysis
Ns	= Synchronous speed (rpm)
N	= Rotor speed Rotor speed (RPM)
Nf	= Number of fatigue cycle
Ni	= Number of cycles at stress Si
OEM	= Original Equipment Manufacturer
Pu	= Excitation per unit
σUTS	= Ultimate tensile stress
σf	= Parameter that scales with tensile strength obtained by fitting experimental data
TNF	= Torsional natural frequency in Hz or CPM as applicable
τext	= Excitation torque
ω	= Disk rotational velocity

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ACKNOWLEDGEMENTS

Author wishes to tender sincere thanks to Mr. Kirtikumar Tripathi for reviewing preventive maintenance aspects of electrical part, Mr. Ganesh Kumar and Mr. Viljoen Christo for internal review and suggestions for structuring of this tutorial. Author tenders sincere thanks to Dr. Timo P. Holopainen for clarifying understanding on electro-mechanical aspectAuthor also acknowledge the support of Mr. Jack Xu to pull this tutorial up to expected standard.