Table of Contents

REVAMPING A GAS COMPRESSOR DRIVE TRAIN FROM 7000 TO 8000 HP WITH A NEW SYNCHRONOUS MOTOR DRIVER AND A CONTROLLED SLIP CLUTCH MECHANISM

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ABSTRACT

The successful application of synchronous motors requires special design consideration to account for their speed-dependent characteristics, such as the large oscillating torque levels that occur during startup. This paper focuses on the torsional design and analysis process associated with revamping a 7000 hp synchronous motor-driven compressor train with an 8000 hp synchronous motor driver. Torsional analyses indicated that the replacement motor's startup torque would produce unacceptable levels of torsional vibration. An accurate overtorque protection slip clutch was integrated into the low-speed coupling assembly to limit startup peak torque to about three times the normal operating torques.

The paper details multiple issues associated with revamping the compressor drive train, including the selection of the replacement driver and low-speed coupling assembly, torsional vibration analysis, and the mechanical operation of the controlled slip clutch mechanism. The paper also presents the acquired torsional field measurements that were used to confirm the clutch's operation.

INTRODUCTION

Over the last half-century, industrial gas producers have made huge capital investments in large horsepower machinery. Synchronous motors remain a popular driver for many of these applications. In response to market demands, many older machines have been rerated to increase their output. However, over time, even equipment that has provided reliable service for decades needs to be replaced. Major revamps of machinery are often undertaken for a variety of reasons, again, most of them market-driven. Most improvements are made to increase the equipment's efficiency and reliability and, of course, to reduce maintenance.

Unlike the engineering of new equipment, the revamping of an existing drive train often poses unique challenges. In most cases, the modified equipment has to coexist with the available plant infrastructure. Typically, this means that the new equipment must fit within the existing plant real estate and be able to be integrated into the available power grid.

This paper focuses on the motor selection, torsional design, and analysis process associated with revamping two existing 7000 hp synchronous motor-driven industrial gas compressor trains with 8000 hp synchronous motor drivers. Unlike other types of motor drivers, the successful application of synchronous motors requires special design consideration to account for their speed-dependent characteristics, such as the oscillating torque performance that occurs during startup. Typical of many revamp projects, however, the requisite engineering studies were still being completed after the replacement equipment had been ordered. In this application, the final torsional analysis determined that the selected motor would create unacceptable levels of torsional vibration during startup. To remedy this problem, several different design alternatives were evaluated via torsional vibration analysis. The successful solution to the problem involved modifying the low-speed coupling assembly to include Voith Turbo's Safeset® autoreset coupling. This coupling, a variation of a proven torque limiter coupling design, permits slippage when a preset threshold torque is reached, thereby isolating the drive train from excessive levels of torque. In addition to computer simulations, the performance of the coupling was verified via torsional field measurements. The two new motors and controlled slip couplings were installed in the fourth quarter of 2003 and have been providing reliable service. A summary of the end-user's operational experiences with the revamped compressor trains is also presented in this paper.

While the successful application of torque-limiting devices in steel making machinery is not new, its utility in solving torsional vibration problems in process machinery is just now being recognized, particularly in revamp applications. It is the authors' hope that the entire turbomachinery community will benefit from this thorough review of the logistical and technical issues associated with this revamp project and that the "lessons learned" will be of some value for future applications.

OVERVIEW OF THE REVAMP PROJECT—THE PLANT'S PERSPECTIVE

Background

Two identical three-stage integrally geared gas compressors were installed in 1975 at an air separation plant in the USA that supplied oxygen and nitrogen gas to a pipeline system. Initially, excess production was liquefied and sold to the merchant market. Seven thousand horsepower (7000 hp) laminated rotor synchronous motors were used to drive the compressor trains. These motors were selected due to their relatively high efficiency, lower starting inrush current, together with a severe penalty for power factor (kVA demand) from the local utility. Over time, pipeline sales increased and the plant was "debottlenecked" to affect a substantial increase in its saleable product output. The output from the gas compressors was increased, which in turn raised the horsepower requirement of the electric motors.

Fourteen identical compressors were later installed at other locations. These trains were driven by 7500 hp synchronous motors with laminated rotors. Increasing the gas flow and motor horsepower to the shaft also "debottlenecked" these locations.

By the year 2000 the two original 7000 hp motors were nearing end of life due to being run in an overloaded condition. A study was made of the maintenance history of the motors in the company system to determine the best options for the purchase of new electric motors. At that time, the company had over 50 synchronous motors ranging in size from 1000 to 24,000 hp, with motor speeds ranging between 327 to 1800 rpm. These evaluated motors included both laminated and solid pole rotor designs.

Motor and Low-Speed Drive Coupling Selection Criteria

The criteria for selection of the motors and required modifications for this location were driven by the following:

- The new motors would have to start across the line like the original motors and therefore motor inrush had to be limited in the design.
- The new motors would be selected for a requirement of minimum motor shop visits during the projected life of 30 to 35 years.
- The new motors would have to be synchronous in order to avoid paying a high monthly penalty for power factor (kVA demand).
- The new motors would meet the current mechanical and electrical specifications.
- The efficiency of the new motors must be at least 1½ percent higher than the original motor efficiency.
- The new motors would have to be able to replace both the 7000 hp and 7500 hp motors with no foundation/sole plate modification to provide a solution for the other 14 compressor trains.
- The new motors would be 8000 hp to eliminate the possibility of overloading them under all foreseeable current and future compressor operating conditions.
- A torsional vibration analysis of the revamped compressor train with the new motor driver and selected low-speed coupling assembly must be performed to ensure that it will provide 5000 starts.
- Elastomeric couplings were not to be used, as they have been known to require maintenance about every five years. Furthermore, there were also concerns regarding the shelf life of the replacement elastomeric elements.

Motor Selection and Drive Train Design

After review of the maintenance records and discussion with motor vendors it was decided that the new motors would have solid pole rotors, as the design met the criteria established for this revamp. Solid pole synchronous motors from various vendors had been in use since the late 1970s and had experienced a minimal number of problems. Experience had shown that a laminated rotor synchronous motor would normally require a motor shop visit approximately every seven years to do rotor repairs. Moreover, the solid pole rotor's relatively compact design permitted the vendor to supply the 8000 hp replacement motor with a base plate that had the same footprint as the 7000 hp motors. When needed, an adapter plate could be furnished to also allow the new motors to replace the other 7500 hp motors.

Two new motors were purchased and the compressor vendor was contracted to do a torsional study and to interface with the motor vendor during design. There was a problem with getting a viable solution due to the increased motor horsepower and use of the solid pole rotor. The compressor vendor proposed three solutions. However each solution was, in turn, withdrawn after it was determined that it failed to either resolve potential long-term service/reliability concerns, meet the 5000-start requirement, or both.

During this time period, motors were also being replaced on gas compressors at other air separation plants. One solution to a similar problem was solved by the use of a controlled slip clutch coupling. The compressor vendor was contacted and it was suggested that a solution be tried by integrating a controlled slip clutch coupling into the low-speed coupling assembly. A new torsional analysis was performed and all parties accepted the proposed solution. Two controlled slip clutch couplings were ordered and all material was shipped to the plant awaiting either a motor failure or a scheduled shutdown, to install the new motors and couplings. In addition, a repair kit for the controlled slip coupling with oil, charging pump, and spare parts was ordered and shipped to the plant.

The two new 8000 hp motors and controlled slip couplings were installed in the fourth quarter of 2003. At that time, the compressor vendor sent personnel to the site to acquire field torsional data. The sensor suite was configured to measure the dynamic torque on the compressor input shaft (straingauges) and the differential slip across the coupling (keyphasors). The measurements confirmed that the coupling was performing as designed; however, the slip occurred at about 80 percent of its threshold slip setting. As this discrepancy did not jeopardize the torsional performance of the drive train, no attempt was made to adjust the coupling's slip torque threshold. Both couplings have provided reliable service since commissioning.

Assessment of Revamp and Recommendations for Similar Trains

The revamp of the two air compressors with new 8000 hp motors and controlled slip couplings is considered a success as all the plant's criteria were met and the equipment has been in operation with few problems. One issue did occur when one of the controlled slip couplings rotated during an "event" and sheared the brass pin. The oil drained out and the motor freewheeled until it was shut down. The motor, compressor, and coupling system was inspected and no problems were found. A new brass shear pin was installed and the controlled slip coupling was recharged with oil. The motor was restarted without incident. The most likely cause was a disturbance on the utility power. The two gas compressors are being operated in excess of 7000 hp.

The use of controlled slip couplings has become an accepted practice, as one was recently installed on a revamped 13,000 hp compressor train in the second quarter of 2003 and another one is scheduled for installation on an 11,000 hp gas compressor train in the second quarter of 2005. The technology is proven and is currently considered as a viable solution for potential torsional vibration problems that may arise on compressor revamp projects where solid pole rotor synchronous motors are used to replace turbines, induction motors, or laminated rotor synchronous motors. Moreover, this device has been recognized for its ability to provide overload protection during startups, electrical short circuits, or any other electromechanical abnormality that may be encountered during service.

GENERAL STARTING CHARACTERISTICS OF SYNCHRONOUS MOTORS

Synchronous motors as compressor drives are normally used for higher power applications, i.e., 10 MW (13,000 hp) and more. This kind of motor offers the opportunity to balance the electrical power factor of a plant. In other words, electrical idle power can be compensated.

From the mechanical point of view, the torque occurring in the air gap between rotor and stator of the motor must be thoroughly analyzed. Transient torques that arise in electric motors can be separated into two categories: those that occur when the machine is running at operating speed, and those that occur when the train is being started. Excitations that are transient in nature, such as the starting of a synchronous motor, any kind of electrical fault, or torque ripples from variable speed drive systems (VSDS), may excite torsional natural frequencies. Some of these events cannot possibly be avoided, e.g., excitations caused by starting motors and integer and noninteger harmonics from VSDS. Hence, in principle, when evaluating the occurring peak torque caused by any kind of excitation mechanism, the realistic probability of occurrence of the considered loading condition has to be taken into account. This evaluation requires a good understanding of the electrical grid condition in the field. Nevertheless, this paper focuses on the torsional characteristics of synchronous motors during asynchronous starting. A discussion of this transient behavior and its influence on the torsional dynamics of drive trains is presented herein.

Why it Occurs

The name synchronous motor is derived from the fact that the rotor rotates in synchronism with the rotating magnetic field produced by the three phases of electrical power supply. Hence, the speed is directly proportional to line frequency and the number of pole pairs. During acceleration of the train with the motor average torque, a strong oscillating torque is developed because of slippage between the rotor and stator fields.

The mean and the oscillating torques are superimposed and describe the main starting characteristics. The frequency of the torque pulsation is the frequency at which the stator's rotating field passes a rotor pole. Since the stator's magnetic field rotates at synchronous speed, the excitation frequency is a function of difference between synchronous speed and rotor speed, which is commonly known as the slip speed.

The synchronous motor speed and the slip frequency are defined in Equations (1) and (2) as follows:

$$n_{syn} = f_{grid} / N_{pp} \quad \textbf{(1)}$$

(Definition of synchronous speed)

$$f_{slip} = f_{grid} * \left(n_{syn} - n \right) / n_{syn} \quad \textbf{(2)}$$

(Definition of slip frequency)

where:

n_{syn} = Synchronous speed (rpm)
n = Rotor speed during starting (rpm)
f_{grid} = Electrical grid frequency (Hz)
N_{PP} = Number of pole pairs (-)

The motor starting characteristic including mean and alternating torque are often illustrated in a diagram like Figure 1. Torque amplitudes are given as a function of motor speed. Curves in the diagram are related to the considered motor for this project. The assumed starting conditions onsite are as follows:

- SCC = 195 MVA @ 4160 V Bus

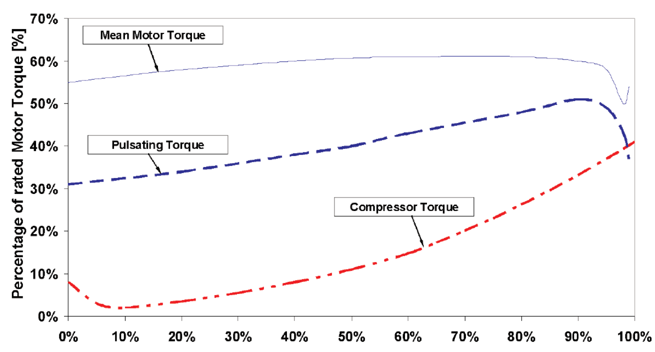

Figure 1. Synchronous Motor Torque Versus Speed Characteristics Given by the Manufacturer (SCC = 195 MVA @ 4160 V Bus).

Table of Contents

When the electric power is switched on, the motor generates fluctuating torque. This kind of transient torque at line frequency dies out rapidly. During speed increase, the frequency of the oscillating torque is double the slip frequency. This means that it is equal to twice the line frequency initially at standstill and it goes down to zero when the motor is synchronized. Accordingly, as the motor accelerates to its synchronous speed, the torsional excitation frequency decreases toward zero. Therefore, during such startup, the torsional system is excited at its resonant frequencies below twice line frequency (that means: up to 100 Hz for 50 Hz line frequency, and up to 120 Hz for 60 Hz line frequency).

After approximately 97 percent of rated speed is reached, a direct current (DC) field is applied to the rotor, producing a "pull-in" torque and the rotor "pulls-in-step." At this point, the rotor is synchronized with the rotating field in the stator and is able to produce torque at synchronous speed. If the field excitation is applied before the synchronizing speed is reached, an additional large component of pulsating torque is created at a single slip frequency.

The severity of the train's torsional response as the electrical excitation stimulates each natural frequency mainly depends on the distribution of mass moments of inertia and stiffnesses, the mode shapes, the magnitude and frequency of excitation, location of excitation, and the system damping. To determine the occurring transient torques, a thorough torsional simulation analysis of the entire train must be performed. This analysis, which should be an essential part of the standard design process of the rotating equipment vendor, must consider the most realistic transient air gap torques for the local electrical grid condition (API Standard 546, 1997). The benefit of this procedure is that well-known potential torsional vibration problems can be avoided.

Figure 2 presents a Campbell diagram for a typical synchronous motor-driven train. This diagram provides a graphic overview of the rotor system's torsional natural frequencies and its corresponding frequencies of potential excitation. In this example, the first three torsional natural frequencies (eigenfrequencies) are given along with the excitation line corresponding to double slip frequency. It can be easily seen that motor excitations interfere with the first three natural frequencies, which are all below twice the line frequency (120 Hz). The motor speeds at which resonance with the natural frequencies occurs can be calculated as follows:

$$n_{critical\ motor\ speed} = n_{syn} * \left(1 - \frac{f_{Ti}}{2 * f_{grid}}\right) \qquad \textbf{(3)}$$

where:

n_{syn} = Synchronous speed (rpm)
$n_{critical\ motor\ speed}$ = Motor speed at torsional resonance (rpm)
f_{grid} = Electrical grid frequency (Hz)
f_{Ti} = Torsional natural frequency, i = 1, 2, 3, ... (Hz)

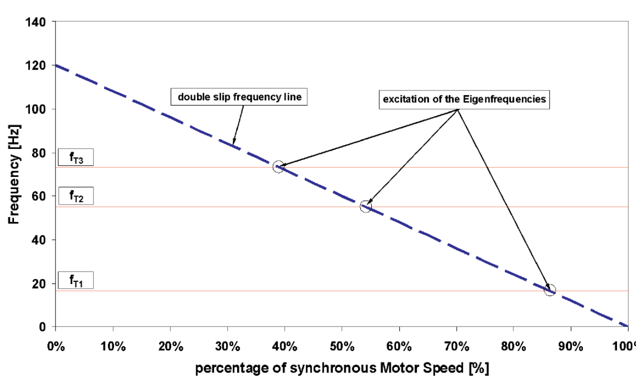

Figure 2. Campbell Diagram Including Synchronous Motor Excitation Line.

The fundamental natural frequency is of primary importance for large synchronous motor-driven drive trains. During resonance, this mode of vibration is capable of facilitating large amplifications of dynamic torque that can result in a catastrophic failure of one, or more, drive components. In general, fundamental torsional frequencies for typical motor-driven turbocompressor trains range between 8 Hz and 25 Hz. Consequently, depending upon the grid frequency, the motor speed at which torsional resonance occurs is usually between 75 percent up to 93 percent its synchronous speed (e.g., 75 percent for 50 Hz and 93 percent for 60 Hz).

Types of Rotor Construction (Solid and Laminated)

The available types of synchronous motors can be separated into two designs: the solid pole rotor, and the laminated pole rotor. In terms of their characteristic mean and pulsating torque startup behavior, these kinds of motor designs are quite similar. But nevertheless, these designs are different in magnitude of average and slip torque versus speed. Compared with laminated pole designs, solid pole rotors produce a significantly higher torsional excitation during starting. However, the general construction of the laminated rotor offers many design alternatives to optimize its starting characteristics. These alternatives are not available for solid pole designs. As mentioned earlier, the customer elected to purchase a solid pole synchronous motor.

Torsional Vibrations and Mechanical Consequences

During startup, the motor has to generate a mean torque to accelerate the driven system. The actual acceleration rate of the system mainly depends on the driver's starting torque, the mass moments of inertia of the train components, the breakaway torque, and, finally, the load torque (includes friction and windage losses). To determine the occurring loads during startup, a transient simulation analysis is required. This analysis takes into account the torsional model of the entire train, including distribution of inertias and stiffnesses, system damping, and motor characteristics.

Rapid starting with high acceleration minimizes the time period in which the system is in a state of resonance. Under these conditions, the resonant response will hopefully not have time to build up to steady-state levels. In other words, for slower motor startups, the system stays at a resonant frequency for a longer period of time, allowing amplitudes and stresses to be amplified. Unfortunately, synchronous motors with a higher amount of mean torque normally show a higher magnitude of pulsation torque, too. This fact should always be taken into account. Based on the authors' experience in the analysis and testing of many synchronous motor driven trains, it has been observed that motor starts with lower levels of average torque, which also tend to have reduced levels of exciting torque, typically produce lower peak torques during resonance. In practice, by reducing the motor's voltage during starting, this phenomenon has been effectively utilized to reduce the peak torques throughout the train. Alternatively, direct-online (DOL) starting at full voltage, where the power supply is assumed to have an infinite short circuit capacity (SCC), produces the "worst case" motor excitations. This theoretical condition produces the maximum torsional response during resonance and is often used in torsional startup simulations to determine the minimum value for the number of "safe starts" that a drive can accrue during its service life.

STRATEGIES TO REDUCE PEAK TORQUES DURING STARTUP

Reduced Voltage Starting

Synchronous motors are often started using several reduced voltage methods (e.g., autotransformer starting, reactor starting, ...). In most cases, the electrical grid onsite requires this method of starting. The maximum allowable starting current is typically specified to prevent power supply overload and/or excessive voltage drop.

With reduced voltage starting, the motor-driven train is started on a lower voltage and switched to full voltage before, or after, synchronization. This approach to lessening the motor's impact on the local power supply has a fortunate consequence in that it often reduces the transient torsional loads experienced during startup. Therefore, as stated above, reducing excitation potential of the pulsation torque can lower the peak torques during resonance. Recognizing that the level of the motor torque is proportional to the square of the voltage, this method of starting also reduces the average torque that is used to accelerate the drive train to its synchronous speed. Whether, or not, it is helpful to reduce the voltage as a means to reduce the peak torsional loads should be clarified in a torsional simulation analysis. In theory, there is a minimum starting voltage where the increased torsional amplifications facilitated by longer dwell times at resonance "balance out" the positive effects of the lower excitation torque. Based on the design of hundreds of synchronous motor-driven trains, it has been the authors' experience that a lower level of voltage normally reduces the peak torque. Of course, the optimal voltage must also satisfy the "pull out" torque and minimum acceleration requirements for the entire startup sequence.

Soft-Starter (Starting Inverter)

In addition to reduced voltage starting, the starting inverter, often referred to as a soft-starter, or load-commutated-inverter (LCI), is commonly employed to protect the power supply line. In contrast to the asynchronous starting methods, the motor is running within the synchronism from zero speed up to operating speed by starting the synchronous motor with an inverter. Afterwards, the motor is switched so that it is drawing power directly off of the electrical grid. Therefore, by starting the motor in a synchronous manner, its pulsating torque with double slip frequency is totally eliminated.

To start the motor, inverters synthesize the time-varying electrical voltage from a series of integer and noninteger harmonic waveforms. Integer harmonics coincide with multiples of the motor supply frequency, whereas the noninteger harmonics are generated by a difference between line frequency and supply frequency. Owing to imperfections in the synthesized electrical supply waveforms, these devices induce torque ripples with magnitudes that are usually in a range of less than 1 percent and up to 20 percent of the motor's rated torque. The resulting transient response should be evaluated via torsional analysis. Nevertheless, compared with DOL starting, torsional simulations have demonstrated that the maximum torsional response can be greatly reduced with this method of starting. It is worth noting that the inverter-induced harmonic torques disappear once the motor is switched onto the local electrical grid.

The use of an LCI provides an elegant, but a rather expensive means of reducing the motor torques during startup. As mentioned earlier, the customer did not want to use an LCI.

Minimizing Torsional Stiffness of Steel Coupling

In most cases, the initial drive train design is formulated without the benefit of torsional vibration analysis. Once complete, the proposed torsional system is analyzed. Afterwards, the system may have to be tuned if adequate separation margins do not exist between its torsional natural frequencies and potential excitations. Furthermore, coupling tuning may be necessary to reduce the occurring transient torques caused by synchronous motor starting.

Adjusting the torsional stiffness of the coupling is a common method of tuning the torsional system. Turbocompressor trains are mostly designed to operate supercritical in terms of the fundamental torsional natural frequency related to the minimum driver speed. It can be seen in Figure 3 that reducing a train's torsional stiffness tends to decouple the main masses of the torsional system (as example: driver and compressor), which, in turn, leads to lower dynamic torque levels.

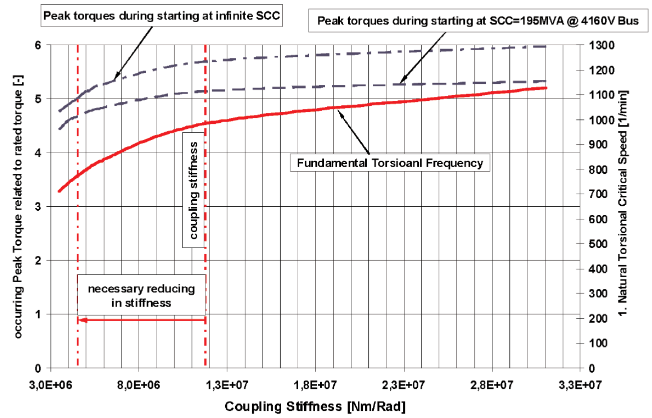

Figure 3. Results of the Torsional Stiffness Variation Map.

For a given distance between shaft ends (DBSE), the minimum stiffness can be achieved by using a solid coupling spacer. Materials with adequate strength and lower shear modulus than steel can be used to minimize the stiffness. For example, fabricating the coupling out of titanium can reduce its torsional stiffness, as ratio of the shear modulus values of titanium and steel is 0.534 ($G_{titanium}/G_{steel}$ = 53.4 percent).

However, reducing a coupling's torsional stiffness by reducing its spacer diameter also tends to lower its transferable torque capability in terms of static and dynamic torque. Therefore, the absolute minimum stiffness that can be realized is mostly limited by the required peak torque capability of the coupling.

As will be shown in a later section, the compressor vendor initially considered this approach to reduce the train's startup torques. Unfortunately, the torsional calculations determined that the startup torques would still be excessive.

Elastomeric Coupling with Low Stiffness and High Damping

One of the common methods to reduce the torsional vibration amplitude within a resonance is to increase system damping. Elastomeric couplings are often used in such cases.

This style of coupling typically consists of a steel design combined with an integrated elastomeric element. The main tasks of these elastomeric elements are reducing significantly the torsional stiffness and, in addition, increasing the damping in the system in contrast to a solid steel coupling. The torsional stiffness of such elastomeric elements leads to nonlinear behavior. Stiffness characteristics vary with transmitted power, as well as operating temperature.

To be effective, these dampers need to be located at a position with high twisting angle. Due to the fact that the normal stiffness distribution leads to vibration nodes and, therefore, to high torsional displacements within the coupling, the coupling locations tend to be "well suited" for these types of damping devices.

During operation, the elastomeric elements absorb vibration energy. As a consequence of the material properties, the rubber elements degrade over time due to thermal loading and environmental factors. Compared with steel couplings, the main disadvantage of this design is that it requires increased maintenance to provide reliable operation.

Particularly for higher horsepower applications, it should be noted that elastomeric couplings have a tendency to be extremely heavy devices. This fact should always be born in mind as it may have a negative influence on the lateral rotordynamics of the equipment on either side of the coupling. Finally, in some applications elastomeric couplings are principally not allowed, per customer specification.

Integrating Controlled Slip Clutch

Synchronous motor-driven trains are designed to transmit the rated power, as well as the dynamic torques that occur during starting. In most cases, the dynamic torque requirements are fairly high and, therefore, govern the design of the drive components (selection of shaft sizes, gearing, fits, etc.). Hence, the train's steady-state design factors are usually fairly high and can easily accommodate a modest increase to its rating.

The state-of-the-art aerodynamic technologies offer a potential for upgrading existing compressors of older design within the same compressor frame size. These upgrades require higher rated torques and a larger driver may be necessary. As pointed out, the components are often capable of transferring increased amounts of static (driving) torque, but their capacity to withstand dynamic loads is limited. If the more powerful motor increases the dynamic peak torques during starting, a controlled slip clutch presents the opportunity to limit the dynamic torques during starting. As only a coupling hub needs to be exchanged, the main parts of the original coupling and the gear set can often be reused. This assumes, of course, that these reused components are capable of providing reliable operation at the upgraded power rating.

Furthermore, this kind of safety coupling offers an overload protection against transient excitations such as those that might occur during starting when a motor is switched on to the power grid, as a consequence of an electrical fault that may occur during continuous operation (e.g., short circuit, network change over, reconnection after power interruption, ...). Especially the load cases like network change over and reconnection to power line may lead to excessive high air gap torques of up to 20 times of rated torque.

CONTROLLED SLIP CLUTCH

Background

The basic design principle of the "autoreset" controlled slip clutch is to transmit the torque through a friction joint where the torque capacity is controlled by hydraulic pressure (Appell, 1997). This basic layout of the torque-limiting coupling is shown in Figure 4. This torque limiting principle has been implemented in various critical turbomachinery applications, including aeroderivative-driven power generation units. Moreover it has been implemented on extremely large synchronous motor-driven applications, such as a 100 MW (134,000 hp) wind tunnel drive.

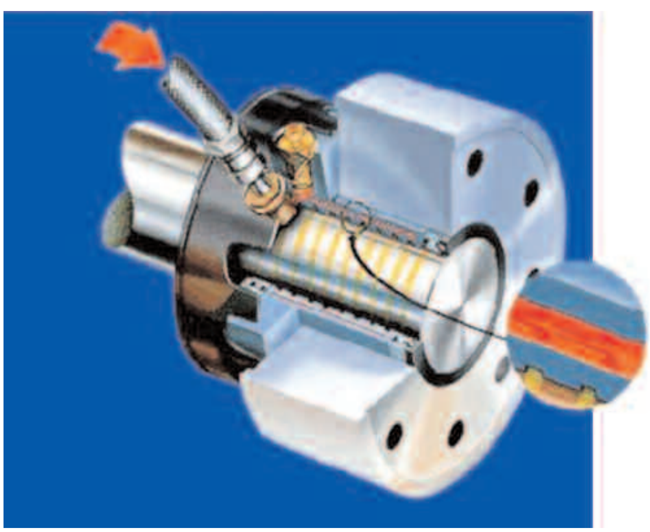

Figure 4. Torque Limiter Coupling.

Although the implementation of the controlled slip clutch is new to turbocompressor drives, it has been applied on refiner drives in the wood and pulp industries since 1987 (Figure 5). In some of these applications the startup peak torque is experienced on a daily basis.

Figure 5. Typical Paper Refiner Drive.

Mechanical Overview and Operating Principles

The controlled slip clutch uses the mean of a controlled hydraulic pressure to achieve a desired slip torque. The simplified hub's torque carrying capability can be calculated using the following equation (ANSI/AGMA 9003-A91, 1991):

$$T = \pi \times \left(P - P_0\right) \times L \times \mu \times \frac{OD}{2} \quad \textbf{(4)}$$

where:

P = Pressure (psi or MPa)
P_0 = Pressure at which the friction surface grips (psi or MPa)
L = Hydraulic cavity length (inches or mm)
μ = Coefficient of friction
OD = Friction diameter (inches or mm)

The clutch consists of two welded sleeves to create a sealed clamping disc (Figure 6). The disc is mounted on an intermediate flanged sleeve. The keyed ring is lightly shrunk on the motor shaft along with the shear ring. For the purpose of this paper, the keyed ring and the shear ring are referred to as the driver side while the clamping disc and the flanged sleeve are referred to as the driven side. The driver and driven sides of the clutch are centered to each other by means of a high performance ball bearing. An accurate slip torque is achieved between the two sides by controlling the hydraulic pressure inside the clamping disc. During normal operation, the controlled slip clutch acts as a rigid connection transmitting 100 percent of the torque. During startup, the clutch's driver and driven sides will slip relative to each other every time the torque exceeds the clutch's preset slip torque limit. While slipping, the clutch will transmit torque at a level equal to its preset slip torque. In typical applications, the clutch will slip up to 5 degrees and it is capable of slipping in both directions.

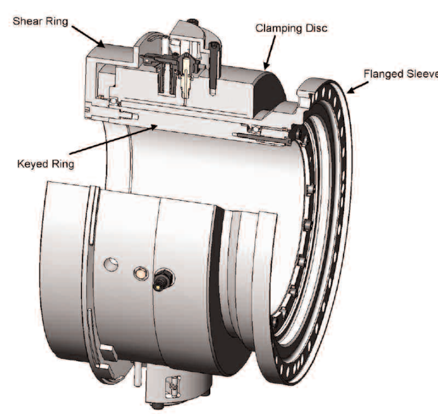

Figure 6. Cross-Section of Clutch Assembly.

Protection Against Transient Torques During Startups, Shutdowns, and Overloads

The controlled slip clutch is designed to limit the peak oscillating torque within the design speed range by sliding a short distance. The clutch implements a spring loaded trigger mechanism that automatically resets every time the machine is turned off. The trigger mechanism extends under the influence of centrifugal force. The mechanism is fully retracted at zero speed and fully extended at speeds within its design speed range. Figure 7 shows the cross-section of the trigger mechanism and its position at different speeds. Position "A" shows the trigger position while running below the design speed range. At this position, the clutch will only slip and release due to abnormal overtorque. Position "B" shows the trigger position within the design speed range. At this position, the clutch will slide a short distance after exceeding the predetermined peak torque limit. Once the machine is turned off, the trigger mechanism retracts (Position "A") and resets its position.

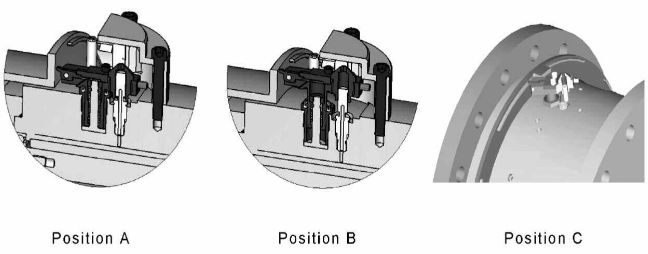

Figure 7. Clutch Trigger Mechanism Position During Operation.

The design also incorporates a complete release capability for catastrophic overtorques produced from abnormal transients, such as motor malsynchronization and compressor surge. In these rare occurrences, the controlled slip clutch will release and completely disengage the driver side from the driven side (Figure 7, Position "C"). Resetting the clutch requires replacing the shear tubes and pressurizing the clutch to the desired pressure. The typical time required to reset the clutch is less than 30 minutes. Without this added feature, a catastrophic failure of the compressor train is likely which could put it out of service for an extended period of time.

Mechanical Requirements, Clutch Selection, and Integration

The controlled slip clutch is selected and designed based on the slip torque level required for the application. Once the slip torque is specified (a common slip torque level includes 2PU, 2.5PU, 3PU) the clutch size and geometrical dimensions are set based on the interface constraints. Typically, the clutch is designed to fit within the confined space of a flexible coupling's hub. The weight and inertia of the clutch are closely matched to that of a standard flexible coupling hub.

The controlled slip clutch is installed on the low-speed portion of the drive train, between the electric motor-driver and the load (gearbox, compressor, etc.). In this circumstance, there are two possible locations for the clutch: the first as a flanged connection at the electrical motor shaft, and the second as a center spacer between the flexible coupling halves. The first location offers the simplest solution, as the clutch can be designed to replace the flexible coupling's motor-side hub. In this configuration, the clutch bolts directly to the flexible coupling without the use of any special adapters. Figure 8 presents a typical installation as a flanged connection.

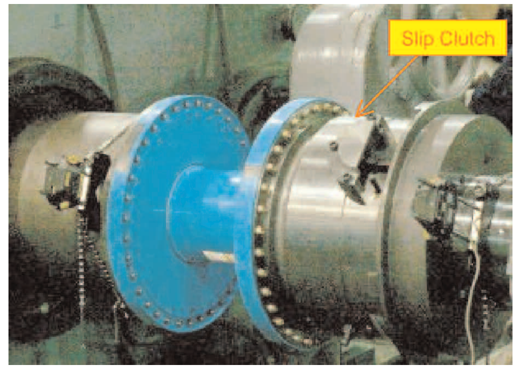

Figure 8. Typical Clutch Installation. (Photo taken during flexible coupling laser alignment.)

Installation and Operation

For a flanged design, the controlled slip clutch is installed by fitting it onto the electric motor shaft. Alternatively, when designed as a spacer-type connection, it is simply bolted in position. Regardless of its arrangement, no special tooling is required to install the clutch.

The controlled slip clutch is set by connecting a high-pressure hydraulic pump to the oil charge port (Item 1 of Figure 9) and unseating the shear tube to permit pressurization of the oil sleeve cavity (Item 2 of Figure 9). As shown in Figure 10, the required oil pressure is selected from the clutch-specific torque versus pressure calibration diagram. Once pressurized, the shear tube is tightened to seal the pressure inside the clutch after which the hydraulic pump is removed. The clutch is ready for operation.

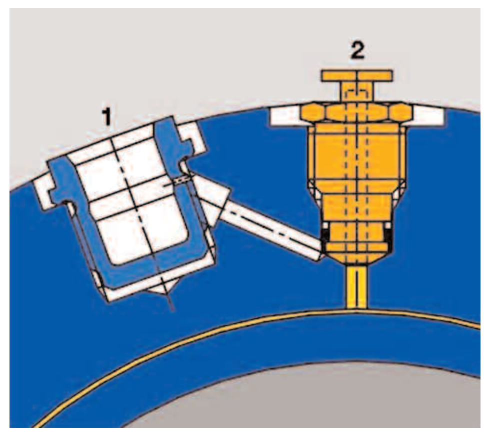

Figure 9. Cross-Section of Clutch's Charging Port.

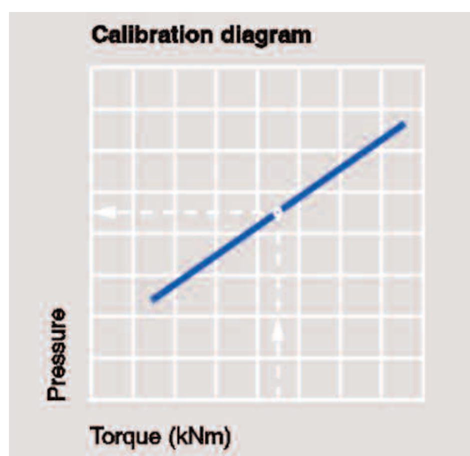

Figure 10. Typical Calibration Diagram.

Maintenance and Reliability

The controlled slip clutch requires minimum service, especially in continuous operating applications, such as a compressor drive. For this type of application, the recommended maintenance interval of the clutch matches that of the major drive components. Typical maintenance requires checking the level of the clutch's lubrication oil. The clutch's hydraulic pressure does not need to be checked. The recommended overhaul interval is 5000 starts, which, for this application, is equivalent to 25 years of compressor operation.

The main factors affecting the torque carrying capability of the clutch mechanism include, but are not limited to, the coefficient of friction, speed, manufacturing tolerances, and temperature. The design and selection of the clutch assembly are based on extensive tribological research to quantify the variables that influence the friction and wear of its internal contact surfaces, the results of which could be the subject of another technical paper. The clutch's material, heat treatment, and surface finish, as well as its friction surface lubricant and interface pressure, are carefully selected and controlled to achieve consistent static and dynamic coefficients of friction. Moreover, each clutch is statically calibrated to achieve even tighter slip torque accuracy. When necessary, this calibration can also be modified to accommodate higher-speed applications (speeds greater than 3000 rpm). The clutch's driver and the driven sides are dynamically balanced as separate components and checked for balance at several relative locations as an assembly.

TORSIONAL VIBRATION ANALYSIS

Among numerous other requirements, to ensure that synchronous motor-driven drive trains provide reliable service, a comprehensive torsional vibration analysis must be performed. The analysis must address all aspects of operation, which includes system startup, steady-state operation, and transient events. API Standard 684 (1996) provides an excellent summary of the requisite procedures and design considerations that must be undertaken to successfully accomplish a comprehensive torsional vibration analysis that will result in a sound drive train design. Briefly stated, as a minimum, a torsional vibration analysis of a synchronous motor-driven train must include the following elements:

- Undamped torsional natural frequency and mode shape analysis
- Synchronous motor startup torsional vibration analysis
- Steady-state torsional vibration analysis at system resonances

While it is beyond the scope of this paper to provide a complete treatment of the subject torsional vibration analysis, its salient elements will be discussed in the context on presenting the simulations performed for this air compressor train. There are many exceptional references that are readily available on this subject, the API's Standard 684 (1996) being one of them.

Creating the Torsional Model

Unlike most lateral rotordynamic models, torsional models must include a mathematical idealization that accurately describes the dynamics of the complete drive train. As such, the requisite engineering data often come from several vendors. For example, the motor, coupling, and compressor vendors provided the data that were used to create the models of the subject compressor train. In more complicated arrangements of equipment, there can easily be four, or more, vendors who will need to provide torsional data for their hardware. Accordingly, each vendor must be made aware of what data are required and how their information will be utilized in the torsional vibration analysis. Once collected, it is the analyst's job to decipher the information and come up with the torsional mathematical model. This model, often referred to as the mass-elastic data, is essentially an assemblage of interconnected massless torsional springs (often arranged sequentially) that is connected to concentrated inertias. The resulting mathematical model is arranged into a system of equations, commonly know as the equations of motion. Figure 11 presents a simplified torsional mass-elastic system, illustrating this concept. Depending upon the analyses performed, these equations can be solved to determine the system's natural frequencies and mode shapes, time-transient response, or steady-state response. Anwar and Colsher (1979) provide a detailed presentation of several methods of solution that are commonly used to facilitate these analyses.

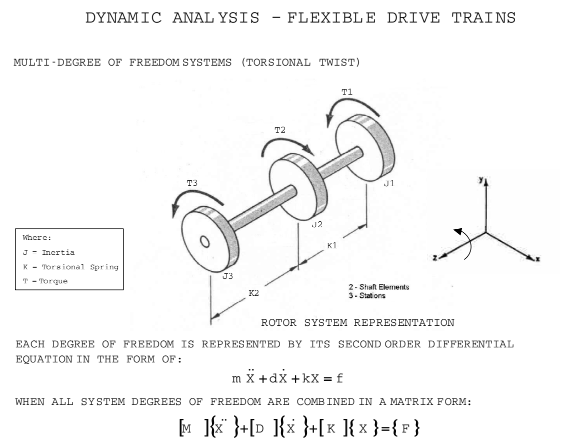

Figure 11. Simple Torsional Mass-Elastic System and Equations of Motion.

Torsional Model and Analysis
Results of System without Clutch

As mentioned earlier, a common strategy to reduce the transient torsional vibrations during a synchronous motor startup is to lower the system's fundamental torsional twist natural frequency. In most arrangements this can be accomplished by utilizing a "torsionally soft" coupling to connect the motor driver with the driven equipment. Prior to selecting the slip clutch mechanism, this approach was investigated by the compressor vendor. A presentation of the torsional model and results are presented herein.

Figure 12 presents the torsional mass-elastic system for the equipment arrangement with the "torsionally soft," steel, low-speed coupling. The arrangement evaluated is a three-branched system, whose low-speed branch operates at the motor's synchronous speed of 1200 rpm. Referring to this graphic, the torsional elements (springs) are arranged sequentially from left-to-right. Each torsional element is bounded by a left and right-hand side station. The balloons and inverted triangles represent station-based inertia and external torque input, respectively.

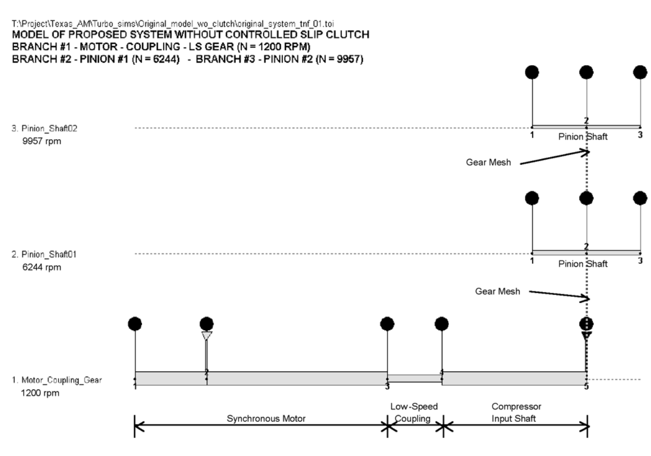

Figure 12. Torsional Model of System without Clutch.

Branch #1 contains the low-speed drive components, namely the synchronous motor, low-speed coupling, and compressor input shaft and bull gear. The inertias of the motor rotor's exciter and core are located at stations #1 and #2, respectively. The main drive coupling is represented by torsional element #3 and is bounded by stations #3 and #4. The inertia of the compressor bull gear is located at station #5. The torsional mass-elastic data for the compressor pinion assemblies are contained in branches #2 and #3. Each of these branches is connected to branch #1 via a torsional spring at the centerline of the pinion-to-gear mesh. The torsional stiffness values of these elements account for the flexibility of the geared connections. The inertias of the compressor wheels are located at the extremities of each pinion branch.

Figure 13 presents the torsional natural frequency results in the context of a torsional critical speed map (Campbell diagram). The diagram provides a graphical presentation of the torsional natural frequencies (horizontal lines) along with the excitation frequency of the synchronous motor driver as a function of speed to expeditiously identify interferences (intersections or resonance conditions). This map presents interferences with the first five modes of vibration (Modes #1 to #5). The critical speed map has been annotated with the corresponding motor speeds at each interference point. Referring to this figure, note that there are interferences with the first four torsional twist natural frequencies that will enter into resonance with the synchronous motor excitations during startup. Figure 14 presents the torsional twist mode of vibration corresponding to the system's fundamental torsional twist natural frequency. An examination of this mode shape reveals where, if excited, the maximum torsional response is most likely to occur.

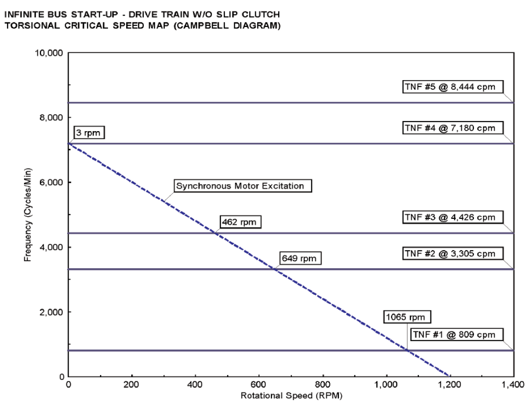

Figure 13. Torsional Critical Speed Map (System without Clutch).

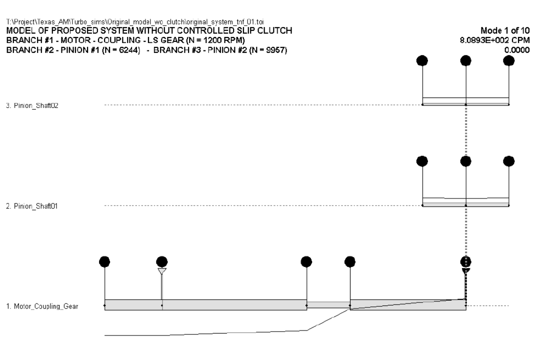

Figure 14. Fundamental Torsional Twist Mode of Vibration (System without Clutch).

For example, if Mode #1 is excited (Figure 14) the maximum dynamic torque, as identified by its node (location where it crosses the shaft centerline), is likely to occur in the compressor's input shaft. The node also provides a qualitative indication of which torsional spring rates have the greatest influence on the corresponding torsional natural frequency. Springs closest to the node will have the greatest influence on the frequency. Accordingly, as one can see, by virtue of its close proximity to the nodal point, in addition to the compressor input shaft, the torsional stiffness of the low-speed coupling has a significant influence on this train's fundamental torsional natural frequency.

Typically, as a consequence of its potential to facilitate large amplifications of the dynamic torques/angular motions, the fundamental torsional twist mode of vibration is commonly regarded as the "mode of concern" in this particular type of drive train configuration. The higher frequency torsional modes of vibration involve some degree of torsional twist in the pinions and, despite being resonant with the synchronous motor torques during motor startup, are fairly "well isolated" from these excitations by the sizable inertia of the compressor's bull gear. Furthermore, these modes of vibrations are also fairly "well damped." Estimates of the modal damping for this arrangement of equipment vary between 2 to 5 percent of the critical value. For the purposes of this study, the authors have used a value of 3½ percent for this configuration of equipment. This value is consistent with those presented by Chen, et al. (1983), as well as by Anwar and Colsher (1979). It should be noted, however, that the selection of the appropriate modal damping value(s) is often viewed as an "experience based" endeavor, and therefore should be consistent with the engineer's understanding of the train's design/operation and the end-user's requirements. Fortunately, however, in contrast to steady-state response calculations, the time-transient response at, or near, resonance is much less sensitive to small-to-moderate variations of modal damping (Smalley, 1983).

Torsional response analyses were performed to calculate the dynamic torques along the drive train, resulting from the input (synchronous motor) and load (compressor) torques, using a nonsynchronous time transient analysis. This type of analysis considers the synchronous motor's speed-dependent mean and pulsating torque characteristics as well as the compressor countertorque (load). Thus, the analysis captures the speed-dependent loading behavior of the motor and compressor to accurately simulate the dynamic torsional response at various locations along the drive train. Newmark's linear response solution algorithm was utilized to simulate the time-transient response of the compressor drive train (Anwar and Colsher, 1979). The full voltage (infinite bus) startup characteristics for the synchronous motor, as well as the compressor load torques, are presented in Figure 15. As mentioned earlier, it should be recognized that the synchronous motor starting characteristics are typically application-specific. Accordingly, with certain synchronous motor designs, the magnitude of the oscillating component torque can reach as high as 120 percent of the motor's rated torque, while others may remain below 70 percent. As such, it is imperative that the time-transient simulations utilize the vendor-supplied startup data. Moreover, the full-load (infinite bus) motor starting characteristics should be used, as this will produce the most severe time-transient response.

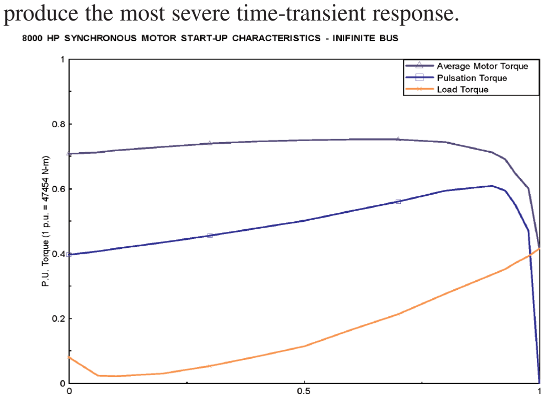

Figure 15. Synchronous Motor Torque Versus Speed Characteristics (Infinite Bus).

Figures 16 and 17 present the simulated torsional response of the drive train. The upper plot on Figure 16 presents the simulated motor speed versus time for the compressor drive train. Referring to this graph, the motor's synchronous speed is achieved in approximately 13.5 seconds. The remaining plots on Figures 16 and 17 present the simulated dynamic torques for the motor drive extension, low-speed coupling and compressor input shaft, respectively. Note that both of these figures clearly illustrate the presence of large dynamic torques at approximately 11 seconds into the startup sequence.

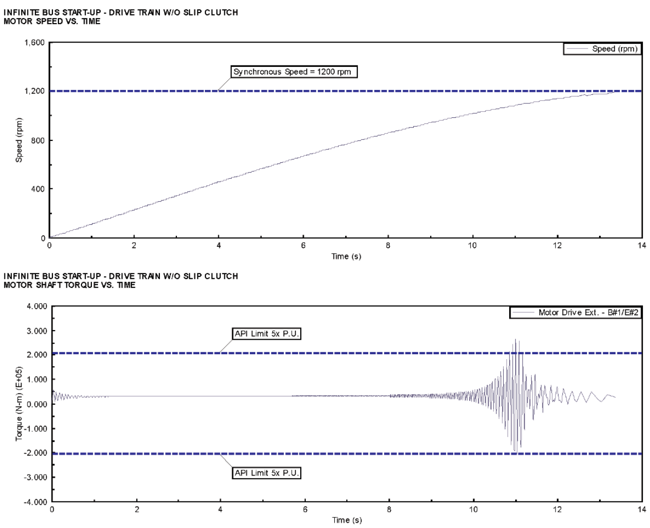

Figure 16. Synchronous Motor Startup Response (System without Clutch).

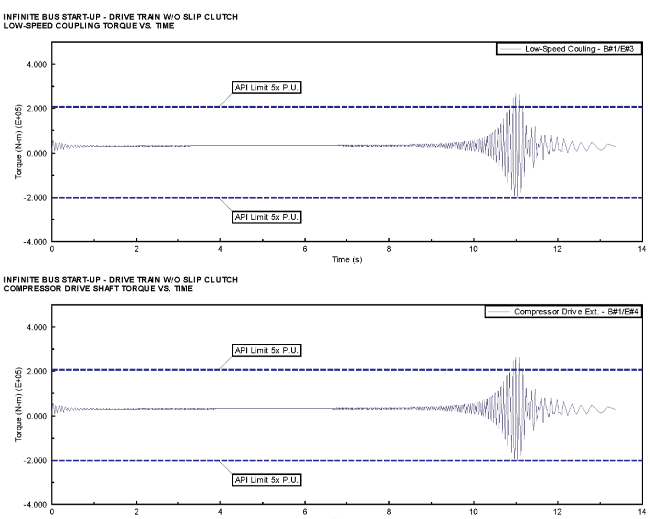

Figure 17. Synchronous Motor Startup Response (System without Clutch).

Referring again to the upper graph on Figure 16, 11 seconds also corresponds to approximately 90 percent of the motor's synchronous speed and is consistent with the resonance with the drive train's fundamental torsional twist mode of vibration that is shown on the critical speed map of Figure 13. Furthermore, referring to the motor startup performance shown in Figure 15, the resonance at 90 percent of synchronous speed is also coincident with the slip speed at which the motor's oscillating torque magnitude reaches its maximum value.

In addition to numerous other torsional performance criteria, the API specifies that, throughout the drive train, the maximum dynamic torque must not exceed five times the motor rated torque for a full voltage start (API Standard 546, 1997). The original 7000 hp rating was used to establish this torque limit. As can be seen from the time-transient simulation results, this limit is clearly exceeded during resonance. Subsequent torsional fatigue calculations also determined that the simulated response would not meet the end-user's requirement of 5000 starts. Recognizing that the motor's starting characteristics, as well as the train's layout, cannot be changed, the only ways to reduce the startup torques would be to dramatically reduce the train's fundamental torsional twist natural frequency, and/or incorporate a torque-limiting clutch mechanism at the drive end of the motor. To further reduce the train's fundamental torsional natural frequency would require utilizing a torsionally-soft, elastomeric coupling. This type of retrofit was not viewed favorably by the end-user as this style of coupling requires periodic maintenance. Furthermore, owing to its increased weight and inertial characteristics, the coupling could also have a negative influence on the lateral rotordynamics of the motor and compressor components.

Torsional Model and Analysis Results of System with Slip Clutch Mechanism

Similar to the previous drive train configuration, a mass-elastic model of the modified system with the slip clutch mechanism was prepared to assess its torsional dynamics. Figure 18 presents the torsional mass-elastic system for this equipment arrangement. For the purposes of performing natural frequency and response calculations, owing to its relatively high torsional stiffness and low inertia, the mass-elastic characteristics of the slip clutch mechanism have been combined with those of the low-speed coupling. These data are represented by element #3 of branch #1 and its adjoining left- and right-hand side stations.

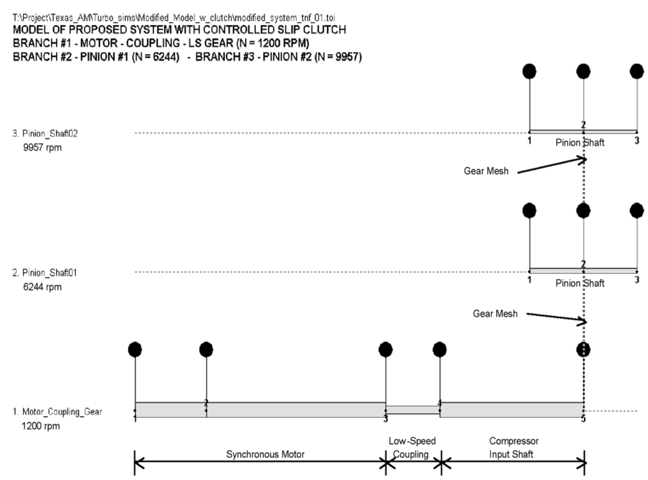

Figure 18. Torsional Model of System with Slip Clutch.

Figure 19 presents the torsional natural frequency results in the context of a critical speed map. Referring to this figure, similar to the previous model, note that there are interferences with the first four torsional twist natural frequencies that will enter into resonance with the synchronous motor excitations during startup. Figure 20 presents the train's fundamental torsional twist mode of vibration. Albeit with slightly different frequencies, the remaining mode shapes are quite similar to those of the previous model.

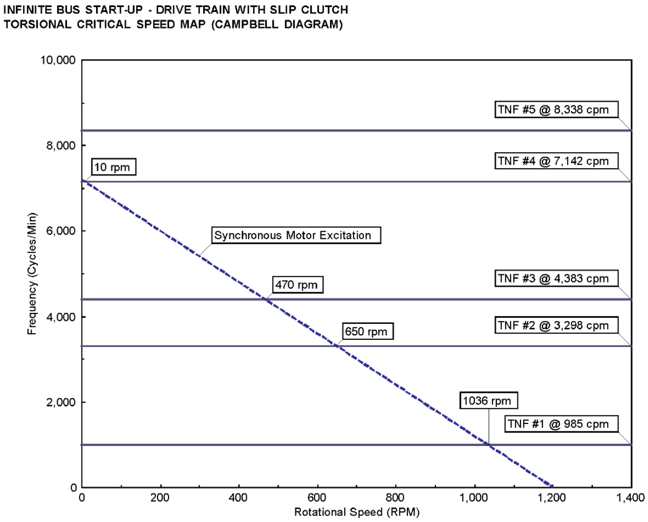

Figure 19. Torsional Critical Speed Map (System with Clutch).

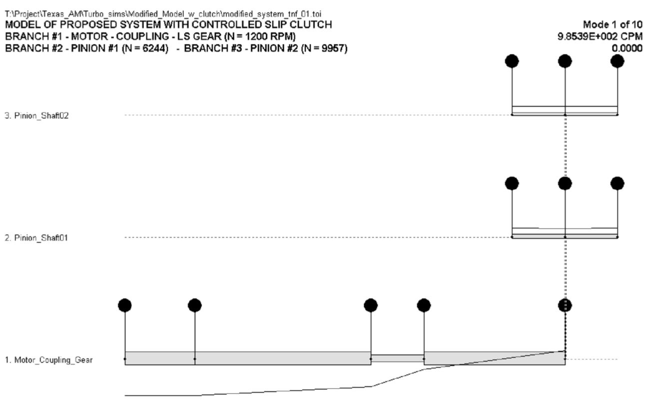

Figure 20. Fundamental Torsional Twist Mode of Vibration (System with Clutch).

The performance of the clutch component, by virtue of its ability to permit slippage, constitutes a nonlinear system and, therefore requires a slight modification to the Newmark numerical integration procedure to simulate its behavior. An overview of the modified numerical integration solution procedure is presented herein. For this application, the slip clutch was set to slip when the instantaneous torque exceeded three times the motor's rated torque (145 kNm [107,000 ft-lb]). Below this value, the coupling behaves linearly, and the aforementioned linear time-transient solution algorithm is employed (Anwar and Colsher, 1979). The slip torque threshold is monitored at every solution timestep. Once the slip torque threshold is reached, the Newmark damping and stiffness system matrices are augmented to reflect a torsional system that is no longer elastically connected through the clutch assembly. In this state, continuity across the clutch element is maintained by computing an external frictional torque that is applied to its left- and right-hand side stations. The value of this torque is determined by taking the product of the ratio of the clutch's kinematic and static coefficients of friction and its slip torque threshold. For this simulation, the clutch's frictional torque was set to 113.93 kNm (84,000 ft-lb). As long as the clutch continues to slip, the uncoupled equations of motion are integrated and the frictional torque is applied to the ends of the clutch element. While slipping, the elastic connection across the clutch is reintroduced into the torsional equations of motion the instant the differential angular velocity across the clutch changes sign. At this point, the linear numerical integration solution routine continues with the original equations of motion until the next time the slip torque limit is exceeded.

Figures 21 and 22 present the simulated torsional response of the modified drive train with the slip clutch assembly under full voltage (infinite bus) startup conditions. The upper plot on Figure 21 presents the simulated motor speed versus time for the compressor drive train. The motor's synchronous speed is achieved in approximately 13.7 seconds. The remaining plots on Figures 21 and 22 present the simulated dynamic torques for the motor drive extension, low-speed coupling and compressor input shaft, respectively. Note that both of these figures clearly illustrate the presence of large dynamic torques at approximately 11 seconds into the startup sequence; however, the maximum torsional amplitude is controlled by the clutch's slip torque threshold. Accordingly, the maximum dynamic torque values do not exceed the five times the motor rated torque limit (API Standard 546, 1997). Moreover, subsequent torsional fatigue calculations also determined the simulated response would meet the end-user's requirement of 5000 starts.

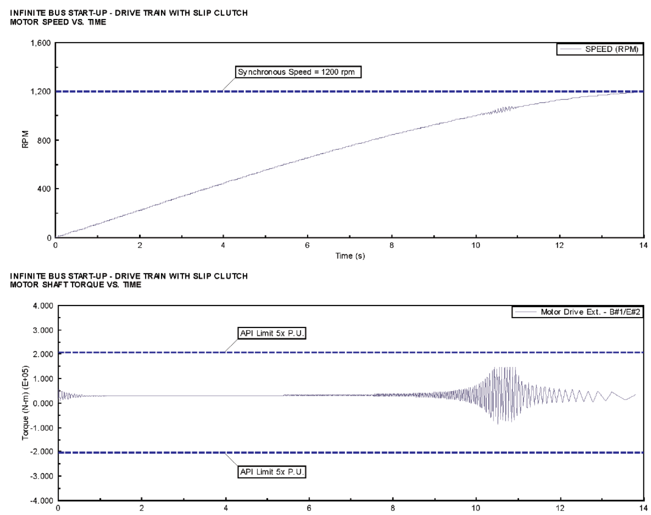

Figure 21. Synchronous Motor Startup Response A (System with Clutch).

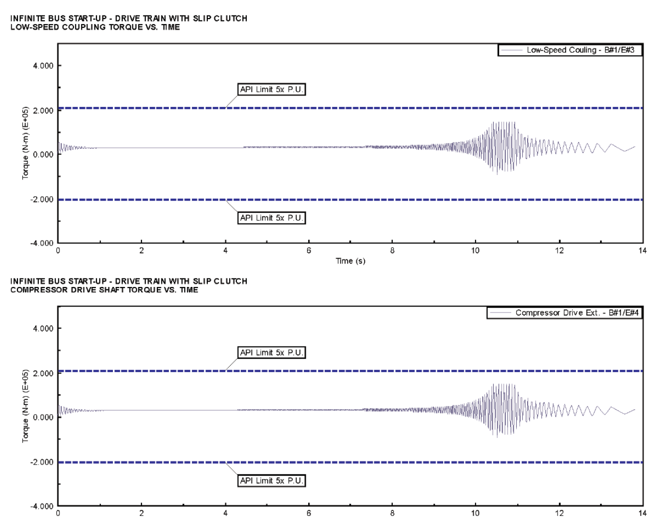

Figure 22. Synchronous Motor Startup Response B (System with Clutch).

Table of Contents

Finally, as shown in Figure 23, the performance of the slip clutch mechanism was also quantified by evaluating its cumulative slip during the startup. The clutch begins to slip when its maximum slip torque setting is reached at approximately 10.4 seconds. From this point forward, the "stepped" appearance of the simulated slip indicates that the clutch disengages (vertical slope) and reengages (horizontal slope) approximately seven, or eight, times as the train passes through the resonance. Based on these data, the cumulative slip for the entire startup was estimated to be 4.2 degrees. The slip clutch vendor determined this performance to be well within the acceptable limits for their device.

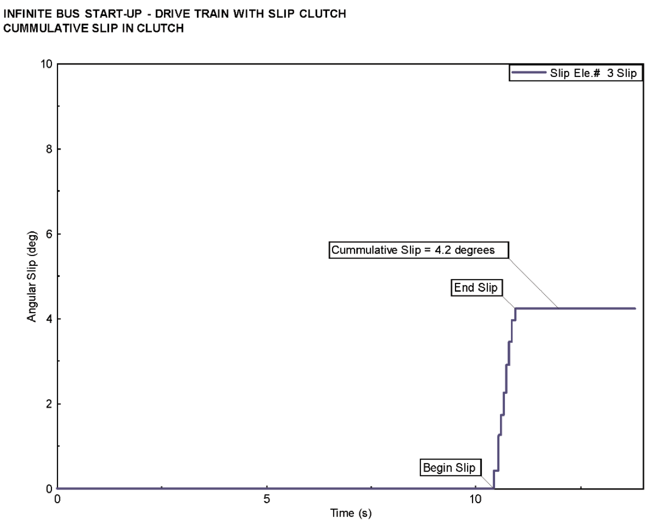

Figure 23. Synchronous Motor Startup Response C (System with Clutch).

FIELD TESTING AND DATA ANALYSIS

Purpose

Torsional vibration problems may be caused by various sources. Most of them are well understood and can be determined within a torsional simulation analysis.

In some cases, gear vibrations and/or noise may give the first indication of a torsional vibration problem due to the fact that torsional and lateral movements are coupled via the gear mesh. Under these conditions clattering of the gear teeth may be heard as the gear teeth oscillate within their backlash. A fatigue failure of train components during operation can be seen as worst indication of a torsional vibration problem.

To avoid any kind of risk related to the train modifications it was the intention of the compressor manufacturer to measure the occurring torque in the field. In principle the used controlled slip clutch was well understood. But nevertheless, the function of the torque limiting effect should be demonstrated by field testing including torsional vibration measurement. This test should also verify that the modified train will be safe over the whole life cycle. Static and dynamic friction and slip torques as well are complex mechanisms that are difficult to predict accurately. Consequently, it was essential to check whether the real slip torque correlates with the predicted figure to enhance the confidence in this design.

Sensors and Data Acquisition

The core element of the measurement method was the coupling equipped with a straingauge. A full bridge circuit was applied to the coupling spacer (Figures 24 and 25). The straingauge was wired to a sensor signal amplifier. The output of the sensor signal amplifier was transmitted via a telemetry system to a nonrotating evaluation unit. The torque equivalent output of the evaluation unit as well as the later in particular described keyphasor signals were sampled by a digital data acquisition recorder and additionally stored on a tape recorder (Figure 26).

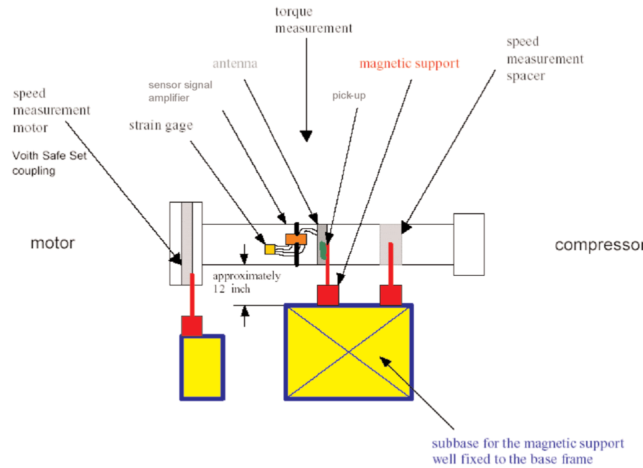

Figure 24. Schematic of Torque Measurement Configuration.

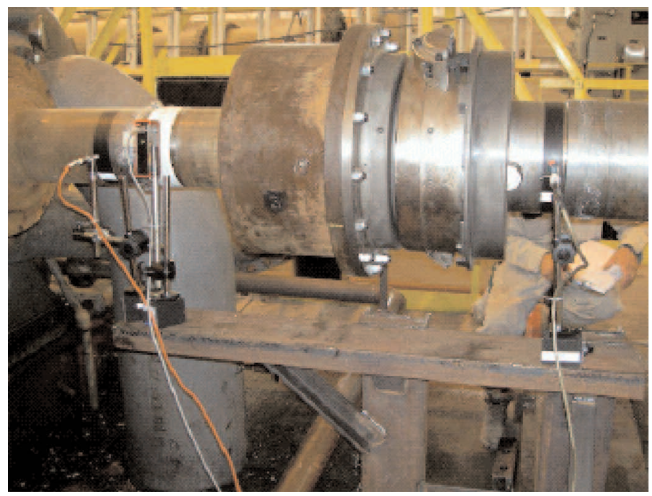

Figure 25. Torque Measurement Configuration.

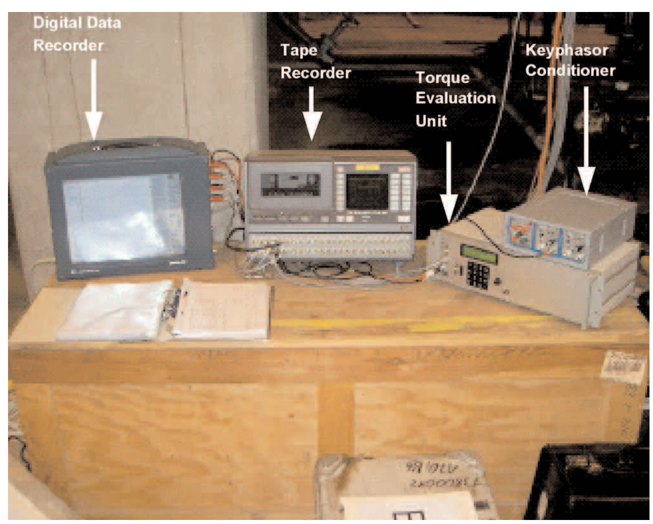

Figure 26. Data Acquisition System.

The straingauge is able to measure the torsional deflection within the spacer material. Based on the torsional flexibility this deflection is proportional to the torque. To achieve an unambiguous signal the straingauge was installed far away from a stress intensified zone as can be seen in Figure 24. The system is prepared to detect the torque transmitted from the motor via the coupling into the compressor on an instantaneous basis. In other words, the system is capable of observing the quasistatic and the dynamic torque during starting and afterwards during continuous operation. This is the most reliable method to obtain accurate response frequencies and stresses during a rapid start of a motor.

In addition to torque, the drive train also had instrumentation to measure the slip angle of the clutch during starting. To get an impression of this angle two keyphasor signals were installed; one on the motor shaft and the other on the coupling spacer (refer to Figures 24 and 25 speed measurement motor/spacer). Based on an adequate data acquisition frequency the differential angle between motor shaft and coupling spacer could be measured in a sufficiently precise manner.

Presentation of Test Plan,
Acquired Data, and General Observations

In November 2003 the dynamic torque measurements were carried out onsite. Four starts were recorded by the installed equipment. During start number 1 and 2 the train did not achieve synchronous speed due to general plant reasons. However, start number 3 and 4 were successful and the motor brought the compressor train to synchronous speed. The observed responses for both successful starts are similar in shape and level of peak torques.

To give an impression of the characteristics, the measured results of the fourth run are shown in Figure 27. Figure 27 demonstrates the synchronous motor start, which in principle represents a shape of the transient torque as expected. To provide a better evaluation of the behavior of the controlled slip clutch, Figure 28 zooms into the resonance condition where the motor air gap torque with double slip frequency excites the train's fundamental torsional twist mode of vibration. Referring to this figure, note that there are eight main torque peaks. Seven of them are on the same peak torque level. Additionally, the shapes of these peaks show some indication of nonlinear behavior, similar to a slip-stick mechanism. The detected peak torque values are summarized in Table 1. The clutch was designed to limit the startup peak torque at 145 kNm (107,000 ft-lb). During field testing the clutch slipped seven times at a torque of about 120 kNm (88,500 ft-lb) and limited the peak torque. The evaluation of the slip torque level shows a deviation of ± 2 percent (Table 1). That means in fact, that the range is between 117 kNm (86,300 ft-lb) up to 121 kNm (89,200 ft-lb). Considering the complexity of the mechanical interactions at the clutch's slip interface, the relatively small variation in the measured slip torque of ± 2 percent is quite impressive, and moreover, verifies the accuracy of the device.

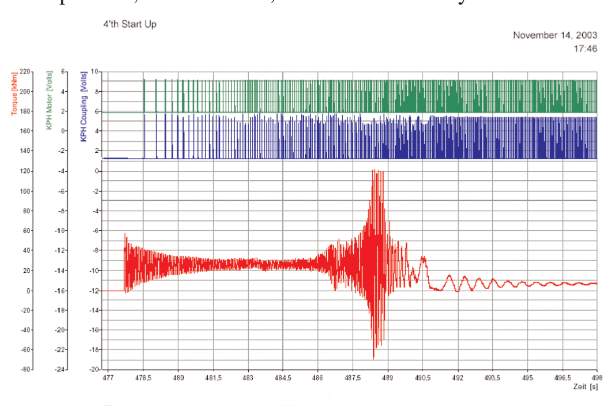

Figure 27. Measurement Results of Starting the Synchronous Motor.

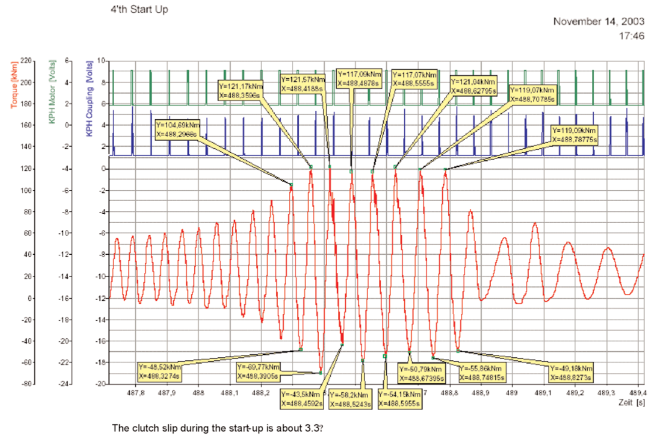

Figure 28. Measurement Results Motor (Zoomed in the Resonant Condition).

Table 1. Determination of Peak Torques During Resonance Condition.

Peak Number	Peak Torque	Deviation from average Peak Torque
1	121.17 kNm (89,370 ft-lb)	+1,4%
2	121.57 kNm (89,665 ft-lb)	+1,7%
3	117.09 kNm (86,361 ft-lb)	-2,0%
4	117.07 kNm (86,346 ft-lb)	-2,0%
5	121.04 kNm (89,274 ft-lb)	+2,2%
6	119.07 kNm (87,821 ft-lb)	-0,3%
7	119.09 kNm (87,836 ft-lb)	-0,3%
Average	119.44 kNm (88,094 ft-lb)	

The torque level in reverse direction were measured in a range from -48 kNm ($-35,400$ ft-lb) up to -70 kNm ($-51,600$ ft-lb). No indication of slipping in the opposite direction was observed due to its relatively low level. By means of the two keyphasor signals a difference angle between motor and coupling of 3.3 degrees was measured.

Based on these field test results it is proven that this slip torque protection works during startup. The slip torque was in fact 120 kNm (88,500 ft-lb), whereas the clutch was adjusted to limit the torque at about 145 kNm (107,000 ft-lb). The deviation between predicted and measured torque is about 20 percent. About 50 percent of this deviation can be attributed to differences in the diametrical tolerances of the clutch's calibration mandrel and that of the electric motor's drive extension shaft. If required, the clutch could have been recalibrated in

Table of Contents

the field. However, this procedure was not determined to be necessary, as the clutch's design slip torque was not exceeded and its measured angular slip was found to be acceptable.

To check the plausibility of the strain measurement, the torques at steady-state (continuous) operation were compared to the electrical power input. The measured results correlate well with the electric and related thermodynamic power. The deviation is in a range of about ±2 percent. Based on this comparison it can be stated that the field test was a reliable diagnosis.

Comparison with Computer Simulation Results

To facilitate a direct comparison of the measured torque values with the simulation results, the nonlinear time-transient response startup simulations were rerun with a reduced slip torque setting of 120 kNm (88,500 ft-lb). Similar to the previous analyses, the simulation utilized the same infinite bus motor starting characteristics, modal damping parameter, load torques, and slip coupling coefficients of friction. Figure 29 presents time-waveform data for the measurement (top) and simulation (bottom) data. An initial inspection of these data shows that there is relatively good agreement between the two sets of results. The measurement data indicate that the clutch begins to slip around 10.75 seconds, while the simulation predicts initial slippage about 0.25 seconds earlier (10.50 seconds). During resonance, the maximum and minimum values of the predicted torsional oscillations are almost identical to the measured values.

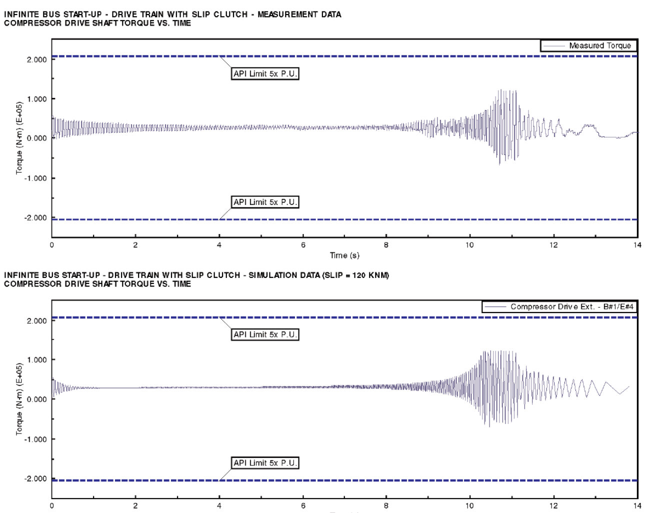

Figure 29. Measurement Versus Simulation Results (System with Slip Clutch).

The only notable difference in the two sets of data is that the simulation predicts about twice as many slip-stick events during resonance. Accordingly, as shown in Figure 30, the predicted slip of 6.9 degrees is about twice as the measured value (3.3 degrees). Nevertheless, the simulation results are considered to be quite acceptable. It is likely that further refinements of the simulation slip-stick algorithm, and/or the simulation input data would provide even better predictions. Possible refinements include:

- Implementation of a damping model at the simulated slip-stick interface.
- Reduction in the time-transient response integration timestep.
- Refinement of the motor and load startup data.
- Refinement of the static and kinetic coefficients of friction for the slip clutch.
- Refinement of the torsional mass-elastic model.

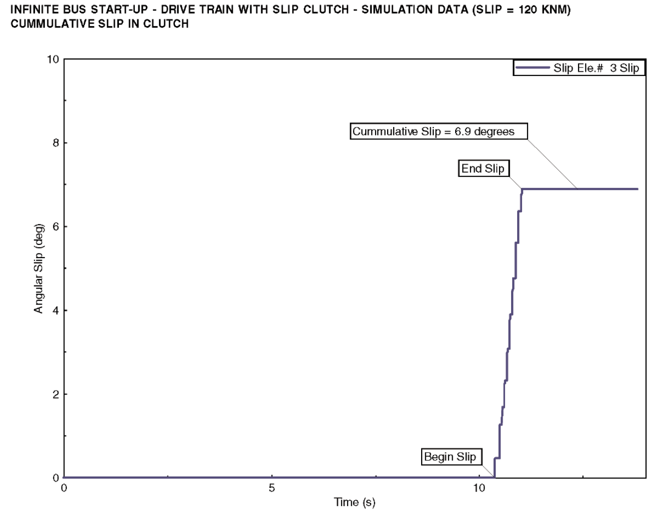

Figure 30. Simulation Clutch Slip During Startup.

SUMMARY AND CONCLUSIONS

*Review of Revamp Project and Lessons Learned—
The Plant's Perspective*

The main objective in revamping the compressor trains was to extend the life of the plant while being able to operate at substantially higher than design production. The second objective was to reduce the amount of electrical power consumed per unit volume of product produced. The third objective was to reduce the amount of projected maintenance for the compressor train. The fourth objective was to minimize the amount of modification required to install the new motors. The fifth objective was to design the revamped compressor train for 5000 starts. The final objective was to minimize the *risk* of the revamp.

The first three objectives were achieved by using a higher horsepower solid pole rotor synchronous motor whose design was optimized for this application. The fourth objective was achieved by having the motor frame designed to fit the existing sole plates and duplicating all-important dimensions of the original motor. The fifth and final objective was achieved by using the controlled slip clutch coupling. Accordingly, the revamp met the objectives and is considered a success.

The revamp of the two 7000 hp gas compressors was part of a larger program of replacement or upgrading of critical components that are reaching end of life. The replacement of the motors went very smoothly during the turnaround. Installation of the controlled slip clutch coupling consisted of fitting it onto the motor shaft, positioning it for correct axial location, and then charging the clutch with oil up to the design pressure. Once charged, the clutch coupling is fixed onto the motor shaft where it serves as the motor-side, low-speed coupling hub. The performance of the slip clutch mechanism was verified via torsional field measurements. To date, both clutches have continued to provide reliable service.

Some of the lessons learned from this revamp are:

- The use of the slip clutch type coupling is a viable solution to limiting the transient torsional torque during synchronous motor startup. This was verified by test during startup after the new solid pole rotor synchronous motors and slip clutch type couplings were installed. Risk of mechanical failure is reduced to an acceptable level.
- The use of a solid pole rotor synchronous motor has many operational advantages, but there is a penalty to pay compared to using induction or laminated rotor synchronous motors. The penalty is the more sophisticated replacement coupling sets that are required. This is considered a reasonable engineering tradeoff.

• A complete set of motor data has to be sent to the group doing the torsional analysis. The motor specification package was upgraded during this project as a result of "lessons learned."

• The final design of the motor and coupling set may take three or four iterations.

• The motor vendor, compressor vendor, and the group doing the torsional analysis should agree to a final solution before the motor is released for manufacture.

• The total time from start of project to having the components ready to install takes about 18 to 24 months.

• The time to change out a motor varies between two and three weeks for a motor installed on sole plates and three to four weeks for a motor whose frame is grouted into the foundation. This includes removal of the old equipment, cleanup, installation, alignment, and startup. Most of the time there will be electrical/instrument work such as power cable replacement, switchgear relaying upgrades, and compressor instrumentation upgrades going on while the motor is being installed.

From an operator's perspective, in addition to improved performance, the proof of a successful revamp project is not having any "reminders" that the work was done (e.g., special starting requirements, additional maintenance items, etc.). The revamped compressor trains have been in continuous operation since the fourth quarter of 2003 at the increased production rates. The work was considered a success and other compressor trains have been upgraded since then.

With the exception of the one event where a brass safety pin was sheared during a utility disturbance, there have been no additional incidents and/or special requirements placed on the drive trains' operation by virtue of installing the controlled slip clutches. It is strongly suggested that a repair kit be purchased and stored at the site where controlled slip clutch coupling is in service. A repair kit generally consists of a charging pump with pressure gauge; a torque wrench for installation of the sealing plug; and oil, spare gaskets, and brass shear pins. In addition, it is recommended that the pressure setting for the controlled slip clutch coupling be marked on the coupling guard so that there is no question as to what pressure the device needs to be set at in the event of having to recharge the unit.

General Remarks and Additional Observations

As with any new technology, the biggest hurdle for the controlled slip clutch is industry acceptance. Although the controlled slip clutch is a well-proven product in the pulp and paper industries, the turbomachinery community, being somewhat conservative, tends to be very cautious when it comes to implementing new technologies. Considering the investment in equipment, as well as the financial and logistical consequences of an unplanned outage, this characteristic is completely understandable.

It is the intention of this paper to provide the turbomachinery community with a comprehensive accounting of the torsional vibration problems, as well as the trials and solutions thereof, encountered during a typical revamp project. It is the authors' understanding that *all parties* involved were satisfied with the outcome of this project. Accordingly, when appropriate, the controlled slip clutch should be considered as a viable means to solve torsional vibration problems for new and used equipment. Since the implementation of the controlled slip clutch on this project, the technology has been successfully applied on different compressor trains at various locations.

Additional observations, recommendations, and conclusions are as follows:

• State-of-the-art aerodynamic designs and analyses offer a viable potential for upgrading existing compressors of older design within the same compressor frame size.

• By carefully applying the appropriate technologies, it is possible to maintain the equipments' original design safety factors when upgrading its performance.

• Nonlinear torsional vibration analyses can be used to accurately predict the behavior of slip clutch mechanisms. It is imperative that the torsional analyst be provided with a complete set of engineering data from all of the equipment vendors to obtain a comprehensive understanding to the drive train's dynamics.

• The use of slip clutch mechanism should also be considered in new applications, especially if the dynamic load levels are decisive in determining the dimensional requirements of the train components. In these situations, by reducing dynamic load levels, the clutch may allow designers to optimize components based on steady-state operating conditions, rather than transient events.

• Compared to other alternatives (LCI, transformers, elastomeric couplings, etc.), slip clutch mechanisms can offer a relatively low cost solution for difficult synchronous motor-induced torsional vibration problems.

• The implementation of a sound engineering approach, along with a complete understanding of the drive train equipment and full cooperation among the involved parties, are vital components to obtain a successful drive train design.

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