SELECTION, DESIGN AND FIELD TESTING OF A 10,500 HP VARIABLE SPEED INDUCTION MOTOR COMPRESSOR DRIVE

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Mr. Hudson graduated with a B.S.M.E. from Newark College of Engineering (1965). He has been a taskforce member on the 4th, 5th, and 6th editions of the API 617 Specification for Centrifugal Compressors, the API task force on Quality Improvement, and the API task force on rotordynamics. He presently is a member of the API 671 Coupling Task Force. He has participated in previous symposia as a discussion leader for the topics of compressor maintenance and compressor performance. He has published papers on torsional vibration, lateral vibration and has presented papers at the 1992 and 1994 Turbomachinery Symposia. He is a registered Professional Engineer in the State of Wisconsin. He holds one United States patent.

ABSTRACT

The cooperative effort is described of equipment manufacturers, the process plant designer and the ultimate user to specify and design a compressor system that was driven by a 10,500 horsepower induction motor that was controlled by a variable frequency drive (VFD) system.

A design methodology is presented which avoids the potential of VFD system harmonics from coinciding with system mechanical torsional frequencies. Certain VFD harmonic frequencies that could not be detuned from system torsional frequencies were evaluated for steady state synchronous response.

Included are torsional field test data that support the design methodology that was used to avoid torsional resonance and predict torsional system response. Additional test data are presented where the VFD was switched in and out of the system.

INTRODUCTION

Efforts are described, as taken by BHP-Copper (end user) at the San Manual Copper Smelter, the end user, Monsanto Enviro-Chem Systems, Inc. (engineering design firm), the vendor who designed and sold the sulfuric acid plant, the vendor of the variable speed drive system and A-C Compressor Corporation (compressor company) to specify, procure, design, and test a compressor system driven by a 10,500 hp variable speed induction motor.

The compressor is used to extract gas from a smelter in a copper smelting facility and functions as the main blower in a new sulfuric acid plant. The new sulfuric acid plant was built to accommodate increased smelter capacity and to allow for rebuilding of the existing 22-year old sulfuric acid plant.

Reliability is of utmost importance. Outages of the main compressor result in shutdowns of the smelting operation. Along with reliability is the necessity to operate as efficiently as possible in an ever more competitive marketplace. Economy of scale favored building one large acid plant rather than two 50 percent capacity plants. Also, the nature of the smelting operation is not one of base load operation. The smelter has a smelting furnace and multiple converters which are charged with fresh concentrate and matte respectively until the molten matte and copper is ready to pour. The furnace off gas flow is steady but the number of converters in the stack vary from zero to three. The ability to accommodate swings in gas volume supplied to the compressor can more easily be controlled by one compressor rather than multiple compressors operating in parallel.

The most efficient method of accommodating variable gas flow to the compressor is speed control. An electric induction motor was the choice of the end user to drive the compressor. The concept of a variable frequency electric drive (VFD) was investigated and found to be cost justified based on a comparison of the power savings when operating a 10,500 hp motor on speed control as opposed to operating with suction throttling or variable inlet guide vanes. A reactor starting system was also incorporated in the electric motor system as a backup method to start the motor in the event of a VFD failure.

The end user chose to require that the VFD system would operate in four of the following modes:

Mode 1. Normal VFD operation with a range of speed control from 40 percent to 100 percent speed (24 Hz to 60 Hz electrical frequency variation).

Mode 2. If maintenance was required for the VFD, the user wanted the ability to bring the VFD to 60 Hz, synchronize the VFD to the line frequency and transfer the motor to the line power supply. The VFD would be deenergized and available for maintenance. This transfer was referred to as the "hot sync" condition.

Mode 3. Once maintenance was completed on the VFD, the user wanted to be able to bring the VFD to 60 Hz and switch the motor from line power to VFD operation. This transfer was referred to as the "cold sync" condition.

Mode 4. If the VFD should malfunction during operation, the user wanted the ability to have the compressor restarted automatically with a reactor start as soon as the inlet guide vanes closed to their minimum position. This was referred to as the emergency trip condition.

The engineering design firm wished to comply with the user's design requirements, however, there was concern about the mechanical design of such a system. The company had previously been involved with a VFD compressor system that had serious mechanical vibrations that limited the plant capacity to 85 percent of plant rating for close to two years before the problem was finally resolved. The author's company was the compressor system supplier who eventually resolved the problem. The details of this problem are described in an earlier article [1]. In addition, the compressor company was aware of other similar compressor system vibration problems attributed to a VFD [2].

The end user previously had problems with starting a 12,200 hp synchronous motor that was installed in an oxygen plant at the site of the proposed VFD installation. Needless to say, no one wanted to experience start up delays on this VFD installation.

PROCUREMENT INVESTIGATION

The end user was concerned that there was no operating experience of similar size VFD installations in similar service. The engineering design company and the end user investigated various VFD and compressor manufacturers and chose those they deemed most knowledgeable of the effect of the VFD system on the mechanical system vibration.

The compressor vendor had, since the resolution of the problem in [1], investigated a number of VFD suppliers and had become more knowledgeable of the part that VFD drives could play in exciting system torsional vibration. The investigation had been taken to such an extent that the compressor company had prepared a purchase specification document that set limits on the electrical excitation developed by the VFD.

A number of VFD suppliers were investigated to determine if the VFD could accommodate the four operating modes identified previously along with other electrical system requirements.

The VFD and drive motor are an interrelated package. The chemical company chose to assume purchase responsibility for these items so that they could coordinate the VFD electrical requirements with the electrical distribution system being engineered for the entire sulfuric acid plant. The compressor company was awarded the compressor, gear, coupling and lubrication system portion of the contract, partially because of their knowledge of the VFD interaction with the compressor system. In addition, the compressor company was retained to perform a torsional response analysis that required modelling of the torsional excitation of the VFD. As part of the torsional consulting agreement, The compressor manufacturer supplied its purchase specification for VFD systems to minimize the torsional excitation from the VFD. The chemical company included portions of this document with their purchase specifications for the VFD.

After the compressor order was awarded, the engineering design company established a short list of two VFD suppliers. When the final VFD bids were submitted, one vendor's bid was modified from his previous offer to bring his offering into compliance with the specification's requirements. The modified offering increased the cost of the VFD. A VFD supplier was chosen and the design phase began.

DESIGN

Gear and Coupling Specification Requirements

The first phase of the compressor scope of the project was to specify and order long lead time components such as the speed increasing gear and couplings. Based on past problems, it was decided to design the coupling and speed increaser as if a torsional resonance would be excited. Most compressor systems of this type have a first mode natural frequency between 15 and 25 Hz when disc or diaphragm type couplings are used. It was assumed that the magnitude of peak to peak oscillating torque would be 10 percent of rated torque. Since the possibility of exciting resonance might exist between 90 to 100 percent rated speed, the components were designed for steady state rather than transient torsional vibration. In the case of the gear, this meant adding an additional torque capacity over and above the normal requirements of full rated power that would account for the dynamic vibratory torque which would be superimposed on the mean transmitted torque.

Since there was not a large amount of experience with extensive system torsional analysis using VFDs and seeing how an elastomeric damper style coupling was used to solve a VFD related torsional vibration problem, it was decided to utilize an elastomeric damper style coupling for this design. The estimates of dynamic vibration levels were a consideration in the coupling selection. The damping of the vibration is accomplished by hysteresis of the rubber blocks which creates heat. Proper identification of the vibratory torque and its frequency will insure that excess heat is not generated in the rubber blocks. Excessive temperature of the elastomeric blocks will shorten the life of the rubber compound. A Holset coupling was specified with an 18 in spacer between the motor and gear shaft ends. The 18 in spacer allowed for ease of maintenance for installation of the coupling blocks and provided a high level of damping with a double set of blocks. Refer to Figure 1 for a drawing of the coupling.

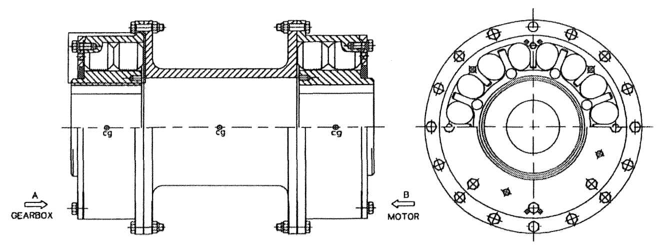

Figure 1. Low Speed Resilient Coupling.

One of the concerns regarding the use of the elastomeric coupling was the potential unbalance introduced by the coupling, even though the motor speed was 1785 rpm. The coupling for the application weighed 939 lb. The coupling vendor was questioned about the manner in which the couplings were balanced. They responded that each of the components, excluding the blocks, are individually balanced. The blocks are formed in precision molds and no effort is made to weigh balance them. No assembly balance is performed on the assembled coupling. It was stated that the spider segments of the hub shrunk on the spacer shaft and the floating hub, both of which were machined by numerically controlled milling machines, would be precise and concentric to the bores or locating fits within 0.002 in. It was also stated that the precision molded blocks that fit between the spider segments would keep the segments concentric. As a precaution, the coupling specification invoked an old API balance criteria which required that the unbalance should not create a force greater than 10 percent of the coupling weight. When this was applied to the component weights, a limit of 0.006 in eccentricity of the spacer was calculated. This eccentricity limit of the assembled coupling was added to the purchase specifications for the coupling.

MODIFICATION OF TORSIONAL PROGRAM

The compressor vendor has extensive experience with analytical simulation of transient torsional response of compressor trains with synchronous motors. The program results have been verified by several field tests of synchronous motors startups. It was decided to modify the portion of the program that calculated the direct and quadrature torques of a synchronous motor with a function generator that could model a complex VFD harmonic excitation. The most complex function generator that could be handled by the program could accommodate 10 input wave forms.

The VFD selected was a 12 pulse drive. The VFD vendor provided graphical and tabular data on 20 possible harmonic excitations. Ten harmonics were identified as being associated with the drive output frequency (F_{OUT}). Those harmonics were 6/12/18/ 24/30/36/42/ 48/54 and 60 times the drive output frequency. Two harmonics were identified with the 60 Hz input line frequency (F_{IN}). These harmonics were six times F_{IN} and 12 times F_{IN}. Finally, there were eight harmonics that were associated with the line input frequency (F_{IN}) and the drive output frequency (F_{OUT}) as follows.

$6 (F_{IN} + F_{OUT})$	$12 F_{IN} + 6 F_{OUT}$
$6 (F_{IN} - F_{OUT})$	$12 F_{IN} - 6 F_{OUT}$
$6 F_{IN} + 12 F_{OUT}$	$12 (F_{IN} + F_{OUT})$
$6 F_{IN} - 12 F_{OUT}$	$12 (F_{IN} - F_{OUT})$

Of the 20 possible harmonics only eight were identified to have torsional oscillations greater than 1 percent of rated torque. These harmonics were associated with 6 F_{OUT}, 12 F_{OUT}, 18 F_{OUT}, 24 F_{OUT}, 36 F_{OUT}, 48 F_{OUT}, 60 F_{OUT} and 12 (F_{IN} - F_{OUT}). Refer to Figures 2, 3, and 4 for vendor data provided for this contract.

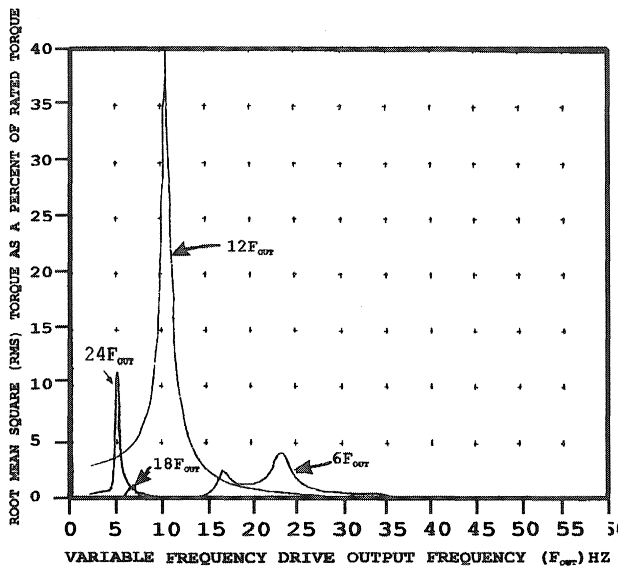

Figure 2. Variable Frequency Drive Harmonic Torsional Pulsation Vs *the Output Frequency of the Drive.*

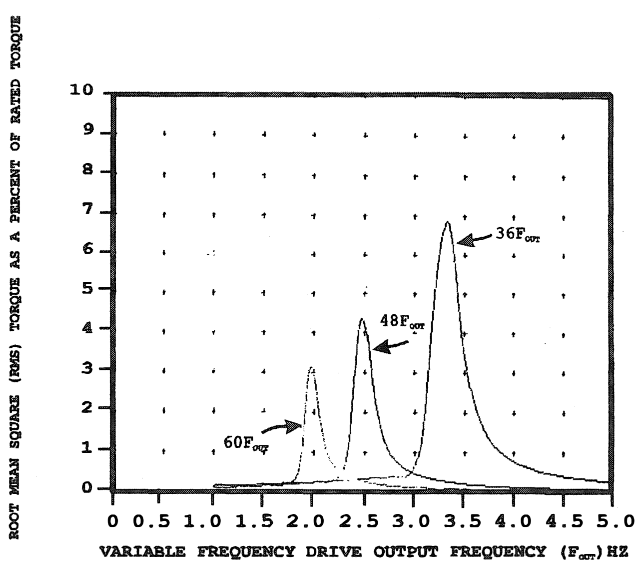

Figure 3. Variable Frequency Drive Harmonic Torsional Pulsation Vs *the Output Frequency of the Drive.*

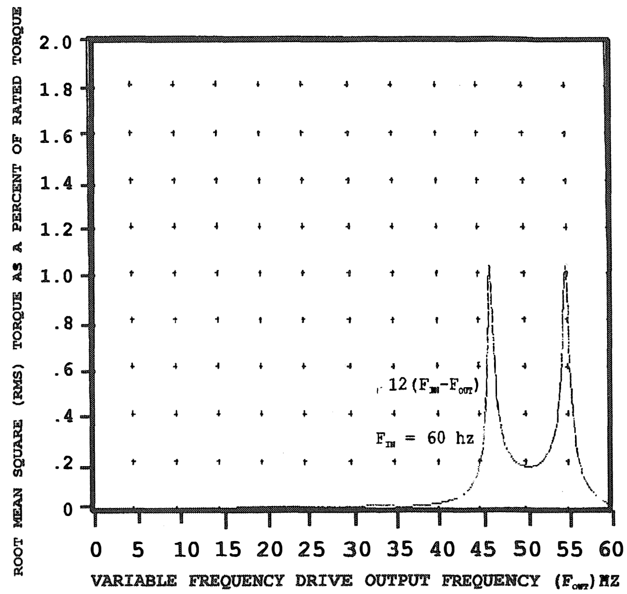

Figure 4. Variable Frequency Drive Harmonic Torsional Pulsation Vs *the Output Frequency of the Drive.*

The function generator model was developed to combine all eight inputs into one complex output wave form that would be used as the oscillating torque produced by the VFD and fed to the motor in the torsional simulation program. Refer to Figure 5 for the function generator block diagram.

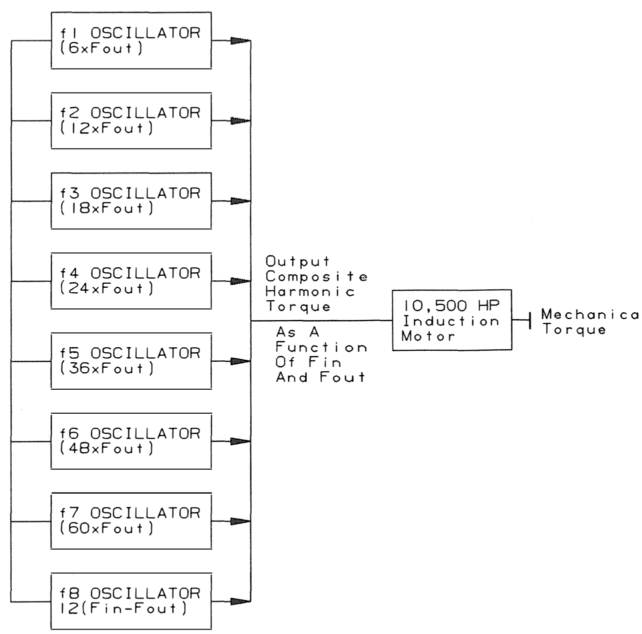

Figure 5. Block Diagram of the Function Generator for Composite Harmonic Torque Produced by the Variable Frequency Drive.

SYSTEM TORSIONAL TUNING

Realistically a VFD system cannot be tuned to avoid a condition of torsional resonance. The following discussion will show that compressor systems with first mode frequencies, the only mode usually excited by torsional oscillations at the motor, can be excited into a resonant condition at either a very low range of drive output frequencies between 0 and 5.0 Hz or at the high range of drive output frequencies usually above 50 Hz. The location of resonance can be easily calculated as follows:

For VFD harmonics that are a direct function of input or output frequencies, resonance may occur when the torsional natural frequency equals the harmonic frequency.

$N_f = K F_{OUT}$
N_f = torsional natural frequency (Hz)
K = constant associated with the harmonic (6/12/18-etc.)
F_{OUT} = drive output frequency (Hz)

Therefore the drive frequency that can excite torsional resonance can be found by

$$F_{OUT} = \frac{N_f}{K} \quad (1)$$

Similarly for harmonics that are associated with the difference between the line frequency and the drive output frequency, resonance may occur when the torsional natural frequency equals the harmonic frequency

$N_f = K_1 F_{LINE} - K_2 F_{OUT}$

F_{LINE} = synchronous electrical line frequency (normally 50 Hz or 60 Hz)

Therefore the drive frequency that can excite torsional resonance can be found by

$$F_{OUT} = \frac{K_1 F_{LINE} - N_f}{K_2} \quad (2)$$

Normally VFD's are not designed to operate at very low frequencies of 0 to 10 Hz but can be expected to operate over a wide range of frequencies from 15 to 60 Hz (and even super synchronous frequencies above line frequency if so designed). Therefore potential torsional vibration in the low frequency range can be considered a transient vibration while potential torsional vibration in the higher frequency ranges must be addressed as a steady state vibration.

SYSTEM TORSIONAL DESIGN

The compressor system was first analyzed with a finite element based program to calculate torsional natural frequencies, mode shapes and component strain energies. A schematic of the system is shown in Figure 6. A system with elastically linear couplings, such as gear or dry type couplings, is straight forward. Once the system geometry is defined, shaft spring constants and component inertias can be readily calculated. Once the coupling vendor spring rate data is obtained, the torsional natural frequencies can be calculated. When an elastomeric coupling is introduced into the system, the calculation of natural frequencies for a variable speed drive or constant speed drive with variable load becomes more complex. The elastomeric elements in the couplings have stiffness characteristics which are non linear varying with the degree to which the elastic blocks are compressed. Under light load the couplings are torsionally softer than they are under high load. Refer to Figure 7 for a plot of stiffness *vs* torque for the coupling that was used for this contract. For a compressor system, where the compressor design is simple enough so that it approximates the fan laws, the torque applied to the coupling is approximately proportional to the square of the speed. If the torsional mode in consideration has a large amount of strain energy associated with the location of an elastomeric coupling, then the natural frequency will be altered by the change in spring rate.

Because the torsional excitation was much higher for the harmonic of the type defined by Equation (1), the goal was to tune

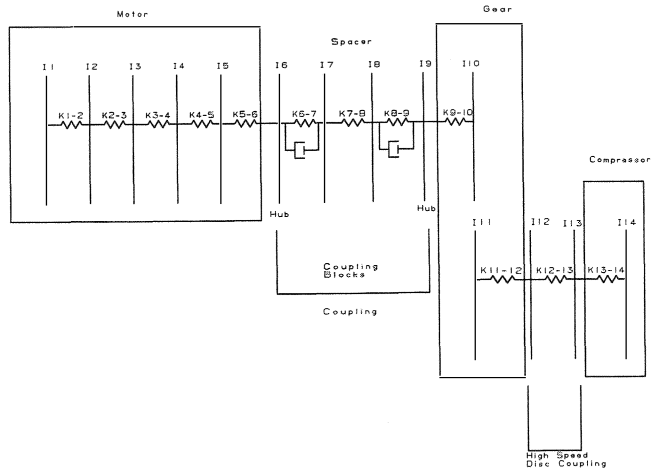

Figure 6. Torsional Schematic of the Compressor System.

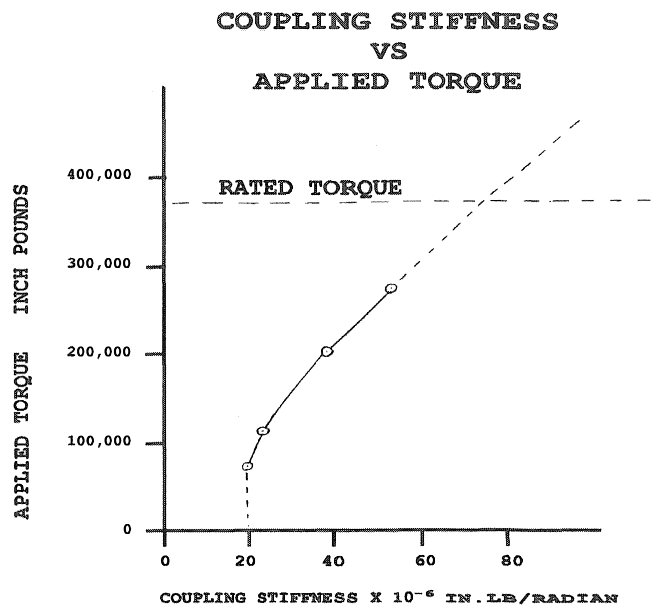

Figure 7. Coupling Stiffness × 10^{-6} in lb/radian.

the first mode frequency such that it did not coincide with an excitation peak from a harmonic.

The data provided for the coupling reveals that the stiffness is nearly constant from a torque of 0 to 80,000 in lb. This stiffness was 20 × 10^6 in lb/radian. Rated torque of the motor can be calculated by:

$$T_{rated} = \frac{63025\ HP}{N}$$

$$T_{rated} = \frac{63025\ (10500)}{1785} \quad (3)$$

$$T_{rated} = 370735\ in/lb$$

where
T_{rated} = Torque (in/lb)
HP = Horsepower
N = Speed (rpm)

A level of 80,000 in/lb would represent 21.6 percent of rated torque. Using the fan law relationship the speed associated with a partial level of torque can be calculated by the following equations.

$$T = T_{rated} \times \left(\frac{N}{N_{rated}}\right)^2 \tag{4}$$

T = Torque at speed (N)
T_{rated} = Rated driver torque

$$\therefore N = N_{rated} \sqrt{\frac{T}{T_{rated}}} \tag{5}$$

N = Speed (rpm)
N_{rated} = Rated speed

This particular case reveals that 80,000 in/lb would be required at 829 rpm. Therefore the coupling spring rate would remain constant from 0 to nearly 50 percent speed. The system torsional natural frequencies associated with this stiffness are tabulated in Table 1(A).

Table 1. Original Torsional Analysis Results.

	(A) 20 E6*	(B) 40.3 E6*	(C) 58.0 E6*	(D) 72.0 E6*
MODE	FREQUENCY (HZ)			
1	6.08	7.76	8.59	9.22
2	31.09	32.39	33.26	34.05
3	95.77	96.70	96.94	97.14
4	108.68	142.36	159.99	172.65
5	184.51	185.08	185.82	187.51
6	254.48	254.75	254.92	255.08
7	300.04	300.17	300.24	300.28
8	360.21	385.50	406.06	427.65
9	449.67	470.28	487.37	493.40
10	493.52	493.55	493.70	506.75
11	739.28	739.28	739.28	739.28
12	868.60	868.60	868.60	868.61
13	1288.37	1293.49	1297.80	1302.59

***HOLSET COUPLING STIFFNESS IN LB/RAD**

The first mode occurs at 6.08 Hz. Using Equation (1), it can be seen that the first mode natural frequency would be excited at a drive output frequency of 1 Hz or less by all harmonics associated with K F_{OUT}. This is an extremely low frequency that is passed through very rapidly. From Figures 2 and 3 it can be seen that the lowest drive output frequency associated with a resonant peak is 2.0 Hz. Therefore based on the VFD vendor supplied data, transient excitation of the first mode is unlikely.

It was also decided to evaluate whether any of the resonant peaks associated with a given harmonic might coincide with a mechanical torsional resonant frequency above the first mode. From Figure 2 it can be seen that the harmonic associated with 6 F_{OUT} has a peak that occurs at a drive output frequency of 26.6 Hz. Similarly, the 12 F_{OUT} harmonic peaks at 11.75 Hz, the 18 F_{OUT} harmonic peaks at 7.5 Hz and 24 F_{OUT} harmonic peaks at 6.1 Hz. For these harmonic peaks the excitation frequency from the VFD to the motor would occur at the product of the harmonic order times the drive output frequency where the resonant peak occurs. Therefore, for the 6th harmonic peaking at 26.6 Hz the excitation frequency would be 6 × 26.6 or 159.6 Hz. Likewise the excitation frequency for the 12th harmonic peak would be 141 Hz, for the 18th harmonic peak it would be 135 Hz and for the 24th harmonic peak it would be 146.4 Hz.

All of these cases occur at or below a drive output frequency of 26.6 Hz which is 44 percent of rated speed for a drive with a maximum output frequency of 60 Hz. It has already been shown that the Holset coupling spring rate is constant between 0 and 50 percent speed for this configuration. Thus the 135 Hz, 141 Hz, 146.4 Hz and 159.6 Hz excitation frequencies associated with the 18th, 12th, 24th and 6th harmonics respectively can be compared to the torsional natural frequencies found in Table 1(A). It can be seen that the fourth mode occurs at 108.7 Hz and the fifth mode at 184.5 Hz. A separation margin of 10 percent below 135 Hz would be 121.5 Hz and 10 percent above 159.6 Hz would be 175.6 Hz. Therefore the fourth and fifth modes had adequate separation from the excitation frequencies associated with harmonic peaks.

The other harmonic to be evaluated was associated with 12 F_{OUT}-12 F_{IN}. Using Equation (2) the drive output frequency that would have a harmonic excitation that would coincide with the first mode of 6 Hz would be

$$F_{OUT} = \frac{[(12)\,(60)] - 6}{12} = 59.9 \text{ Hz}$$

It can be seen from Figure 4 that the torsional excitation associated with this harmonic at this frequency is less than 0.1 percent of rated torque.

Note that this calculation is not 100 percent precise because of the coupling stiffness used to calculate the 6 Hz natural frequency. A stiffer coupling would raise the first mode frequency. It will be shown later that a coupling stiffness of 72×10^6 in lb/radian, the coupling stiffness associated with 100 percent torque at rated speed, produced a first mode frequency of 9.2 Hz. Using 9.2 Hz instead of 6 Hz in the above equation reduces the output frequency by 0.25 Hz.

The harmonic associated with (12 F_{OUT}- 12 F_{IN}) has two spikes which occur at 45.3 Hz and 53.3 Hz. In order to determine whether these spikes would coincide with torsional natural frequencies, calculations had to be performed at coupling stiffnesses that were associated with operation at 45.3 Hz and 53.3 Hz (75.5 percent and 88.8 percent of rated speed respectively). The torque at these speeds can be calculated using Equation (4). The torque at 75.5 percent speed would be

$$\begin{aligned} T &= 370735\,(0.755)^2 \\ &= 211328 \text{ in/lb} \end{aligned}$$

Using Figure 7, the coupling stiffness was determined to be 40.3×10^6 in lb/radian. The torsional natural frequencies calculated using this stiffness are shown in Table 1(B).

The excitation frequency associated with this harmonic can be calculated as follows

$$\text{Excitation frequency Hz} = [(12)\,(60) - (12)(45.3)\,] = 176.4 \text{ Hz}$$

It can be seen from Table 1(B) that a fifth mode is calculated to be 185.07 Hz. An excitation frequency of 176.4 Hz would be 95.3 percent of the fifth mode. This mode has 98.3 percent of the system strain energy and 97.1 percent of the kinetic energy within the motor core. This means that tuning of this frequency could only be accomplished by redesign of the motor core. The motor detail dimensions were rechecked and verified to be correct. It was decided to evaluate whether or not this was cause for concern when the response analysis results were obtained.

In a similar manner the excitation frequency at the second peak of the same harmonic is 80.4 Hz. The coupling stiffness at 88.8 percent of rated speed was determined to be 58×10^6 in lb/radian. The torsional natural frequencies associated with this stiffness are shown in Table 1(C). The excitation frequency of 80.4 Hz is located between the second and third mode torsional natural frequencies.

In addition to evaluating whether or not the compressor system would have a torsional natural frequency that would coincide with a resonant peak in a harmonic excitation from the VFD, the end user also had specified that the system operating in mode 4 operation (motor direct coupled to the electrical line using inlet guide vanes (igv's) as a flow control device) should meet API 617 separation criteria. API 617 requires that no torsional natural frequency may be within 10 percent of the minimum operating speed and + 10 percent of maximum operating speed. A calculation of the torsional natural frequencies were made at rated torque (370735 in lb). The coupling stiffness for this torque, obtained from Figure 7 is 72.0×10^6 in lb/radian. The torsional natural frequencies for this stiffness are contained in Table 1(D).

During operation with igv control, the horsepower and proportional to it the torque, could vary between 40 and 100 percent rated torque. This means that the compressor system torsional frequencies could vary between the frequencies shown in Table 1(A), for minimum coupling stiffness, and Table 1(D), for maximum coupling stiffness. It can be seen that the second mode did not have the separation margin required by API. This mode could be effectively increased by stiffening the high speed coupling stiffness. The coupling was modified. The revised torsional natural frequencies for stiffnesses previously defined in Tables 1(A) thru 1(D) are shown in Tables 2(A) through 2(D). For the most part the frequency most effected by the high speed coupling stiffness increase is the second mode.

Table 2. Torsional Analysis Results with the High Speed Coupling Stiffness Increased to 56.5×10^6 in lb/rad.

	(A)	(B)	(C)	(D)
	20 E6*	40.3 E6*	58.0 E6*	72.0 E6*
MODE	FREQUENCY (HZ)			
1	6.15	7.92	8.82	9.51
2	34.83	35.97	36.74	37.44
3	95.77	96.70	96.95	97.14
4	108.69	142.37	160.00	172.66
5	184.51	185.08	185.82	187.51
6	254.48	254.75	254.92	255.08
7	300.04	300.17	300.24	300.28
8	360.21	385.50	406.06	427.65
9	449.68	470.29	487.49	505.54
10	507.20	507.22	507.26	508.29
11	739.28	739.28	739.28	739.28
12	949.70	949.70	949.70	949.70
13	1288.37	1293.49	1297.80	1302.59

***HOLSET COUPLING STIFFNESS IN LB/RAD**

The results of the torsional frequency calculations revealed that there were no torsional natural frequency that would coincide with a resonant peak of the 6th, 12th, 18th or 24th order of the KF_{OUT} harmonic. The first peak of the harmonic associated with 12 F_{OUT}-12 F_{IN} was within 10 percent of the fifth mode torsional frequency. The second peak of the same harmonic did not coincide with a torsional resonance. For mode 4 operation, the system met API 617 requirements for torsional separation. It was also determined that the first mode torsional natural frequency had the potential to be excited between 0 and 1 Hz drive output frequency from the torsional excitation associated with the 6th , 12th, 18th, and 24th order of the K F_{OUT} harmonic.

Also, it was determined that the first mode torsional natural frequency had the potential to be excited from torsional excitation associated with the (12 F_{OUT}-12 F_{IN}) harmonic at a drive output frequency of 59.5 Hz.

TORSIONAL RESPONSE SIMULATION

The calculation of the system natural frequencies and the degree of separation from harmonic response peaks would indicate that the VFD system should be free from excessive torsional vibration. In order to prove this, a torsional response simulation was performed.

In order to evaluate the synchronous torsional response of the system, the response program must be capable of correctly modeling the mechanical system to obtain frequencies and mode shapes and correctly modeling the excitation mechanism that is being imposed upon the torsional system. The finite element program that was used to determine the torsional natural frequencies has a response option, however, the excitation could only be modeled as a constant frequency and constant magnitude. Shown previously, the excitation from the VFD is much more complex than this. The compressor company herein uses a second torsional response program to calculate the transient torsional response of compressor trains with synchronous motor drives. The synchronous motor torsional excitation varies in frequency from twice line frequency at zero motor speed to 0 Hz when the motor is synchronized to line frequency. The magnitude of the direct and quadrature axis torques are calculated from the parameters defined for the electrical motor circuit. The accuracy of the transient torsional simulation program has been verified both for excitation magnitudes and responsive frequencies by field strain gauge tests.

It was decided to modify the transient torsional program and replace the excitation of the synchronous motor calculation with the composite simulation of the VFD harmonic excitation represented by the function generator previously discussed. The response calculation of a transient torsional analysis is influenced by the rate of acceleration of the forcing function. For this analysis, a number of calculations were made at various rates of acceleration from 0 to 60 Hz. The final calculation was performed at an acceleration rate slow enough to allow the excitation to maintain a constant level of response torque.

It has been shown previously that the use of a nonlinear coupling required the torsional frequency to be calculated several times depending upon the level of torque applied to the coupling. Unfortunately the response program does not have the capability to alter the coupling stiffness as the system is accelerated to full speed, therefore, the response calculations had to be performed several times at different coupling stiffnesses. The results of the response analysis are reviewed in the area that has been most accurately represented by the choice of coupling stiffness.

The first response run was performed with a coupling stiffness of 20.0×10^6 in lb/radian. This stiffness would be valid from 0 to 30 Hz drive output frequency or 0 to 50 percent speed. The rate of acceleration was constant over a time period of 60 seconds. The input excitation torque revealed a peak to peak excitation of 12 percent of rated torque at a drive output frequency of 5.5 Hz. There was also a broad band of excitation torque from a drive output frequency of 9 Hz to 13 Hz. The peak to peak magnitude was 17 percent of rated torque. There was no mechanical resonance identified throughout the 30 Hz range of drive output frequency. A very low level of torsional vibration was found thru 0 to 7 Hz of drive output frequency. The torsional vibration frequency throughout this range was 6 Hz, the first torsional frequency. The magnitude of this vibration was 2-1/2 percent of rated torque.

The second response run was performed with a coupling stiffness of 40.3×10^6 in lb/radian. This run was used to evaluate the system response thru the first resonant peak of the harmonic associated with 12 F_{IN} - 12 F_{OUT}. The drive output frequency was varied from 40 to 50 Hz in a time period of 20.4 seconds. The highest level of excitation torque was 1.8 percent of rated torque. No measurable torsional response was calculated.

The third response run was performed with a coupling stiffness of 58×10^6 in lb/radian. This run was used to evaluate the system response thru the second resonant peak of the harmonic associated with 12 F_{IN} - 12 F_{OUT}. The drive output frequency was varied from 49 to 60 Hz in a time period of 22 seconds. The highest level of excitation torque was 1.4 percent of rated torque. There was no measurable torsional response found in this calculation. This calculation verified that the fifth torsional natural frequency would not be excited.

A run was made with a coupling stiffness of 72×10^6 in lb/radian that would simulate the coupling stiffness at 60 Hz with rated torque applied. One calculation was made at a fixed frequency of 60 Hz. The excitation torque was less than 0.01 percent of rated torque. The response torque was less than 1/2 percent of rated torque.

A final run was made varying the frequency from 0 to 60 Hz with the coupling stiffness set at 72×10^6 in lb/radian. While this run is not precisely accurate throughout the 0 to 60 Hz drive output frequency range, it is a representative display of how the system would respond to the defined VFD excitation. Refer to Figure 8.

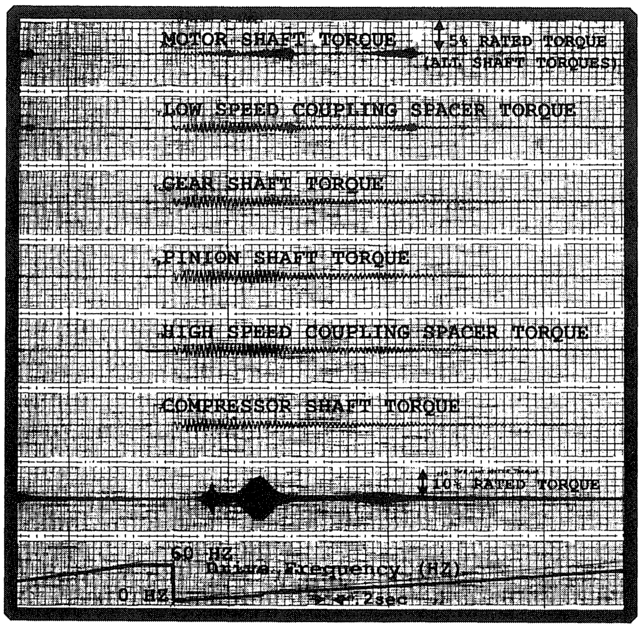

Figure 8. Torsional Response Simulation of the Compressor System.

The conclusion reached from the torsional response analysis was that the system would be free of torsional response that could lead to low cycle or high cycle fatigue damage of power train components.

FIELD TEST RECOMMENDATIONS

One of the concerns regarding the analysis was the validity of the excitation torque data supplied by the VFD vendor. During the four years that the compressor vendor was qualifying VFD vendors, it was determined that very few VFD vendors had the ability to analyze the output wave form and verify that the measured harmonic excitation torque agreed with the calculated data. One VFD vendor had previously published a technical article identifying that this was achievable [3]. While the vendor supplying the VFD on this contract acknowledged that testing was possible, they reported that comparisons of wave form test data had never been performed on their product.

It was recommended that the VFD vendor perform field tests on the VFD output waveform and compare the results to the data supplied by the vendor for the harmonic torsional excitation.

If the vendor could not perform such a test, it was recommended that a field test be performed on the compressor train to verify that there was no harmful torsional vibration in the range of 0 to 60 Hz output drive frequency.

Further it was recommended that the process system should be set up so that the compressor could be field tested to rated power and speed prior to commissioning the system on the process gas. This also would allow optimum tuning of the VFD controls and verify that no torsional vibration problems existed prior to process commissioning.

SHOP TESTING OF THE COMPRESSOR

The compressor specification required that the compressor be factory tested with the contract speed increaser including the high and low speed couplings. The mechanical testing was specified to be per API 617. The compressor was required to be performance tested per ASME Power Test Code PTC 10. The testing was to be performed as near to full load as possible. The A-C Compressor test driver capacity would allow a maximum power level of 9000 hp.

All the testing went very well with the exception of a startup vibration on the low speed shaft of the gear. The first time the compressor was started, the gear reached nearly 6 mil vibration level at a speed approaching 1785 rpm rated input speed. The gear had been previously no load tested at the factory without problems. The magnitude and characteristics of the vibration were indicative of a large unbalance. A dial indicator was placed on the turn of the outer spider of the Holset coupling and it was found to be running out in excess of 25 mil. The spacer weighed 445 lb and each of the outer spiders and end plates weighted 247 lb for a total non piloted weight of 939 lb. The unbalance at each shaft end was

$$\left[\frac{445 \text{ lb}}{2} + 247 \text{ lb}\right] (16) = 7512 \text{ in oz/in or } 7.512\ \frac{\text{in/oz}}{\text{mil}}$$

With the hub running out 25 mil the unbalance at the gear shaft end was 93.9 in oz. or 5.9 in/lb. The force associated with this unbalance at 1785 rpm was 531 lb.

The solution to the problem was to install a pilot ring to center the floating spider on the coupling hub. The cover plate was remachined to install a shrink fit brass pilot ring into the cover plate. Brass was chosen to allow minor relative rotational motion between the hub and the floating spider. The hub was skin cut and the parts detailed for a 0.001 in clearance between the hub and brass pilot ring turns.

The coupling was modified and reinstalled. The first startup after the modification had vibration levels on the gear slightly in excess of one mil.

TORSIONAL FIELD TEST

The user decided to have the compressor train field tested to prove that no harmful torsional resonances occurred in any of the four operating modes. Holset Engineering Services, an engineering service company was contracted to instrument and record torsional field test data.

The compressor company had been party to many torsional tests in which strain gauges were used on the low speed coupling spacer and transmitted by telemetry from the rotating strain gauge. The engineering service company had recently introduced the compressor vendor's products to technology which could measure torsional vibration by laser Doppler techniques. The engineering firm had used a torsional vibration meter to measure the presence of torsional vibration, however, they had not used two laser torsiographs on either side of a coupling to establish phase relationships and torsional deflection across a coupling. It was desired to see if laser torsiographs could compare to the accuracy of the data obtained from telemetry strain gauge testing. There would be an obvious advantage to laser torsiographs in that they could be easily set up to measure high speed along with low speed shafting. The mass of the strain gauge telemetry unit does not have significant effect on the level of vibration on low speed shafts but would be a major effect on a high speed shaft of lighter mass. Also the centrifugal forces associated with the attached telemetry unit becomes a safety concern on high speed shafting. The engineering firm was also interested in the concept of testing with dual laser torsiographs and agreed to establish a testing program which would use and compare strain gauge and laser torsiograph technology for simultaneous measurements.

The engineering firm provided the instrumented strain gauge on the low speed coupling spacer spool. Speed measurements of the low or high speed shafting was accomplished with a digital optical tachometer. Torsiograph measurements were made with torsional vibration meters. Two meters were used. One meter was placed on either end of the low speed or high speed coupling. Data were recorded on a digital audio tape recorder and analyzed on a dynamic signal analyzer. The test instrumentation is sketched in Figure 9.

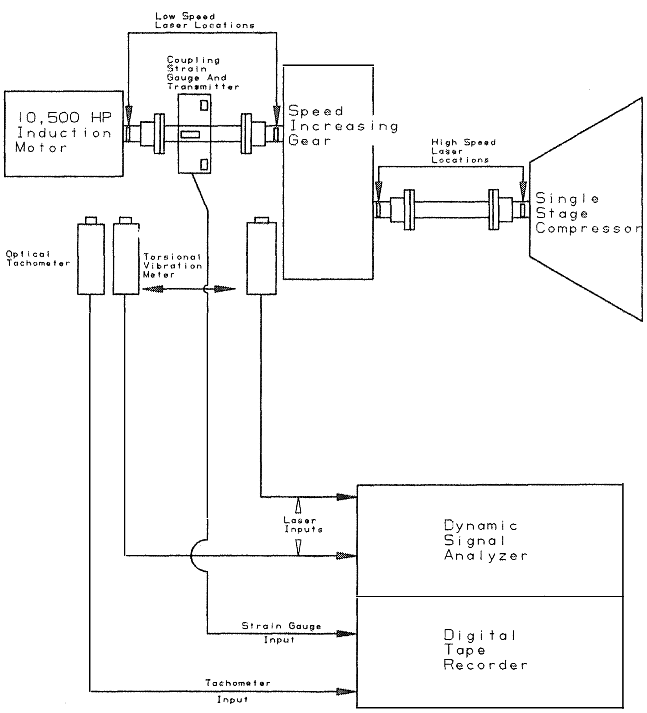

Figure 9. Instrumentation of Compressor System.

The first test sequence was planned to gather data on all the operating modes with the instrumentation located on the low speed shaft between the motor and gear. The following tests were planned.

1. *Normal VFD Start.* The rate of acceleration would remain constant from 0 to 32 Hz of the VFD output frequency. This was a computer controlled start that lasted 40 sec to bring the motor from 0 to 960 rpm. The motor speed was held constant at 960 rpm while the igv was opened to a position that would develop rated torque at 1785 rpm.

2. *Ramp Acceleration From 32 to 60 Hz VFD Output Drive Frequency.* The VFD had a feature that would allow a programmed rate of acceleration from 32 to 60 Hz. The rate of acceleration was set as low as possible and yet maintain computer control. The acceleration rate was two minutes to change from 960 rpm to 1785 rpm motor speed. If the igv position at 60 Hz did not fully load the motor once 1785 rpm was reached then the igv would be adjusted to obtain rated torque.

3. *Test Mode 2 "Hot Sync" Operation.* A test of the transfer from the VFD at 60 Hz to direct connected line frequency.

4. *Test Mode 3 "Cold Sync" Operation.* A test of the transfer from the direct connect line frequency to the VFD synchronized to 60 Hz.

5. *Ramp Deceleration From 60 to 32 Hz.* This test is the reverse of test 2 above with the deceleration period fixed at two minutes.

6. *Ramp Acceleration.* Test 2 was repeated.

7. *Emergency Trip.* The VFD was de-energized and the equipment was allowed to coast until the igv closed to the minimum position. When the igv closed, the permissive emergency restart circuit was satisfied and the motor was started from a roll.

8. *Dwell at Fixed Speed of Interest.* If any resonant conditions were detected during the acceleration or deceleration tests then the speed would be set to control the speed to a fixed value and determine maximum torsional response at a fixed speed.

After these tests were performed, the laser torsiographs would be moved from the low speed to high speed shafting so response tests could be performed on the high speed shafting. Four additional tests were performed on the high speed shaft as follows:

9. *Normal VFD start.* This test was identical to test 1.

10. *Ramp acceleration.* This test was identical to test 2.

11. *Ramp deceleration.* This test was identical to test 5.

12. *Normal VFD shutdown.* This test was set to allow the computer to control the deceleration from 32 to 0 Hz.

TEST RESULTS

All of the planned tests were verbally coordinated between personnel located at the compressor and at the location where the VFD controls were located. These two locations were separated by about 300 yards. Personnel located at the compressor would initiate the test and watch the output of the strain gauge and torsiograph on a dual channel oscilloscope.

The test of the normal VFD start from 0 to 32 Hz drive output frequency did not reveal any resonant conditions. Throughout this range of drive output frequencies, the vibratory torque levels monitored by the strain gauge did not exceed one percent of rated torque.

The test of the ramp acceleration from 32 to 60 Hz identified a point of minor resonance at a motor speed of 1240 rpm, (20.67 Hz). The VFD output frequency for this motor speed is 41.8 Hz. The resonance occurred on an order multiple of 1.717 times the motor speed. The factor of 1.717 corresponds to the speed increaser ratio. Refer to Figure 10 for a Campbell diagram of this response. The resulting frequency was 1.717 times 20.67 Hz or 35.5 Hz. The frequency of 35.5 Hz is the calculated second torsional natural frequency of this system. The magnitude of this resonance is approximately 1.4 percent of rated torque.

During all of the ramp acceleration or deceleration tests, no significant resonances were recorded by either the strain gauge or torsiograph.

The first transient test was the mode 2 "hot sync" transfer from the VFD to line frequency. This switch created a short transient vibration shown in Figure 11. After the transfer there were approximately 6 cycles of vibration at a frequency of 8.4 Hz. The peak oscillating torque created by this transient was ± 26.4 percent of rated torque. It is interesting to note that physically there were no signs of the transfer. If one were not watching the strain gauge output signal one would never realize the transfer had been made. The mean torque measured by the strain gauge hardly wavered (Figure 11).

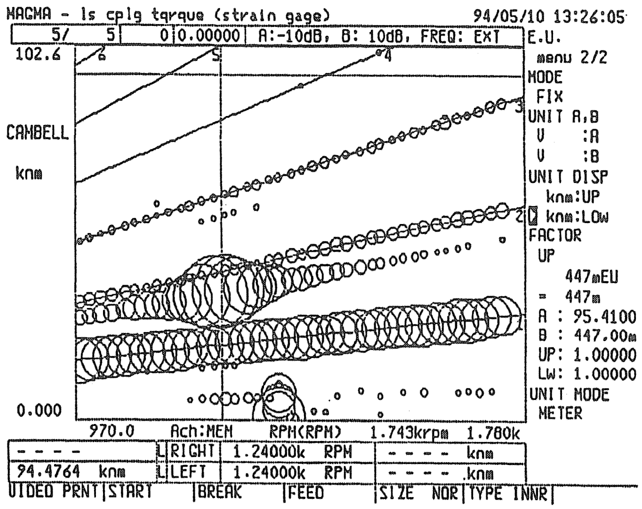

Figure 10. Campbell Diagram of Second Mode Torsional Response Measured by the Low Speed Coupling Strain Gauge.

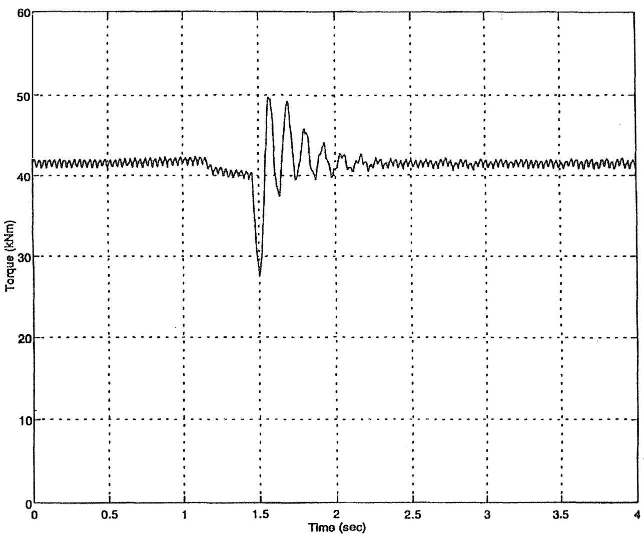

Figure 11. Transient Response Measured by the Low Speed Strain Gauge During the "Hot Sync" Transfer From the Variable Frequency Drive to Line Frequency.

The second transient test was the mode 3 "cold sync" transfer from the line to the VFD. Unlike the "hot sync" test a loud thud was heard during the cold sync. During the transfer the mean torque dropped to approximately five percent of rated torque before power was restored. When power was restored to the VFD the first mode torsional natural frequency was excited twice as if the VFD applied two bursts of power. The first torsional response lasted four cycles at a frequency of approximately five Hz. The second torsional response also lasted four cycles at a frequency of approximately 6.7 Hz. A plot is shown in Figure 12 of the cold sync response. This test clearly reveals the influence of mean torque on the elastomeric coupling stiffness and its effect on the first torsional natural frequency. The maximum level of dynamic torque was ± 22.7 percent of rated torque.

The third transient test was the emergency trip. A plot of the strain gauge signal for this event is shown in Figure 13. After the power was reapplied, the first mode torsional natural frequency

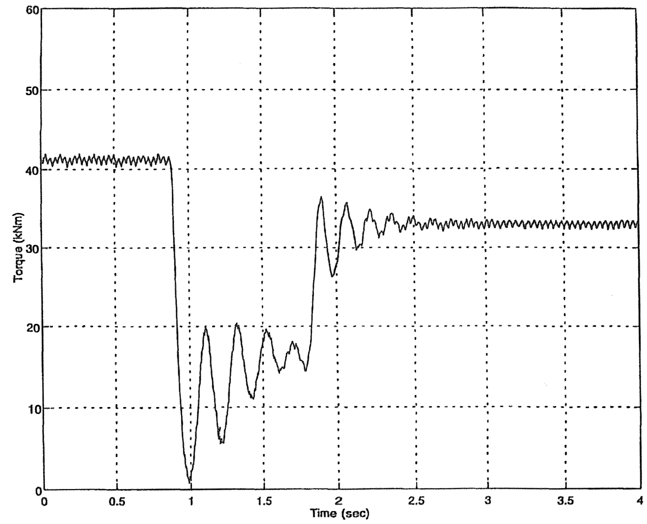

Figure 12. Transient Response Measured by the Low Speed Strain Gauge During the "Cold Sync" Transfer From Line Frequency to the Variable Frequency Drive.

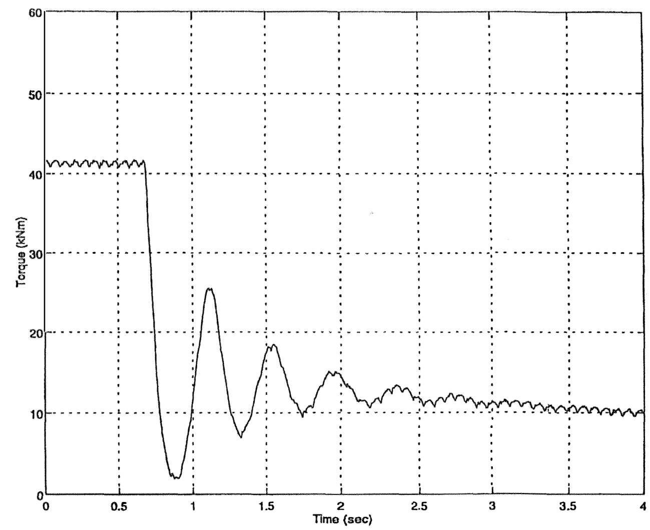

Figure 13. Transient Response Measured by the Low Speed Strain Gauge During the Emergency Trip and Across the Line Restart.

was excited for 3-1/2 cycles. The frequency of oscillation was 2.3 Hz. The magnitude of torque was ± 28.6 percent of rated torque.

After the transient tests, the laser torsiographs were relocated to the high speed shafting. No conditions of resonance were found during the starting from 0 to 32 Hz drive output frequency or the acceleration or deceleration runs between 32 and 60 Hz.

COMPARISON OF LASER TORSIOGRAPH AND STRAIN GAUGE DATA

The strain gauge gave data to evaluate both steady state and transient torsional vibration. The laser torsiographs did not provide data on steady state or ramp acceleration or deceleration tests because the dynamic torque levels were quite low. The torsiographs would have to detect deflections less than 0.01 degrees.

The laser torsiographs were also unable to lock onto abrupt transient torsional vibration. The laser torsiograph appears to be a good survey instrument to determine if there are high levels of steady state torsional vibration, however, the strain gauge has been proven to be a superior instrument for capturing transient events and conditions having a wide range of dynamic signal strength.

OPERATIONAL EXPERIENCE

The installation has operated successfully since its startup in April 1994.

SUMMARY AND CONCLUSIONS

• VFDs can generate substantial torsion excitation to mechanical systems that can cause damage to power transmission components.

• Investigation into the excitation mechanism has identified that most VFD manufacturers can quantify harmonic excitation from the VFD, however, all manufacturers have not supported analytical prediction with test data.

• The compressor company has generated a purchase document that sets limits on harmonic excitation of drives in the normal VFD operation range.

• A design methodology has been developed to compare VFD harmonic excitation frequencies to mechanical system torsional natural frequencies. Should a condition of resonance be predicted either harmonic resonance or mechanical system tuning should be pursued to avoid a condition of resonance.

• An analytical method has been developed to model a complex harmonic waveform from a VFD and to include this excitation in the mechanical system torsional response simulation.

• A 10,500 hp centrifugal compressor system driven by a current source inverter variable frequency drive was analyzed using the technology identified above. It was analyzed to be free of torsional response that would damage mechanical components.

• The process valving was positioned to allow no load and full load tuning of the VFD system prior to commissioning the plant.

• Torsional testing was conducted using strain gauge and laser torsiographs. All the modes of operation planned by the user were tested. No steady state or transient torsional resonance conditions were found that would damage power transmission components.

• The use of the elastomeric coupling did not appear to offer an advantage for operation under normal VFD operating conditions. However, the damping introduced by the elastomeric coupling was beneficial in reducing the magnitude and duration of torsional vibration due to transient electrical power source switching.

REFERENCES

1. Hudson, J. H., "Lateral Vibration Created By Torsional Coupling of a Centrifugal Compressor System Driven By a Current Source Drive for a Variable Speed Induction Motor," *Proceedings of the Twenty-First Turbomachinery Symposium,* Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 113-123, (1992).

2. Tripp, H., Kim, D., and Whitney, R., "A Comprehensive Cause Analysis of a Coupling Failure Induced By Torsional Oscillations in a Variable Speed Motor," *Proceedings of the Twenty-Second Turbomachinery Symposium,* Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 17-23, (1993).

3. Grgic, A., Werner, H., and Prenner, "Large Converter-Fed Adjustable Speed AC Drives for Turbomachines," *Proceedings of the Twenty-First Turbomachinery Symposium,* Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 103-112, (1992).