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TORSIONAL ANALYSIS OF A 20-CYLINDER HYPERCOMPRESSOR TRAIN

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ABSTRACT

The capacity of plants for the production of low-density polyethylene is increasing worldwide and is continuing to challenge designers to provide the larger compression machinery required. The expected level of reliability and availability is very high, and the cylinders and frames of these reciprocating compressors are specially designed to withstand this heavy-duty service. All design aspects have to be properly considered, including the torsional analysis that represents a very critical element.

This article considers a train with 2 compressors, each configured with 10 cylinders, with an electric motor drive between them. One machine is directly coupled to the motor while the other has a flexible coupling.

This represents the largest reciprocating compressor train in the world, with a total motor power of 33 MW.

NOMENCLATURE

c_d	Size factor
C_i	Damping coefficient for the shaft interval *i*
C_{Mi}	Damping coefficient for the mass *i*
$EL_{T,C}$	Compressor crankshaft conventional torsional endurance limit
FEM	Finite element method
J_i	Polar moment of inertia of the lumped mass *i*
K_i	Torsional stiffness of the shaft interval *i*
Ldpe	Low density polyethylene
m	Maximum compressor load torque harmonic order
$M_{a,i}$	Resonance amplifier for shaft interval *i*
n	Train lumped mass total number
RPM	Revolutions per minute
TNF	Torsional natural frequency
TVA	Torsional vibration analysis
UTS_{shaft}	Ultimate tensile strength for crankshaft material
θ_i	Angular displacement of the lumped mass *i*
ω	Torsional natural frequency

INTRODUCTION

The production of Ldpe pellets requires pressures of up to 350 MPa (50,000 psi) depending on the grade of polymer. The highest pressures are usually necessary to produce material with high quality optical properties for plastic films.

The compression system consists of: a booster that collects the recovery plant system gas and a primary compressor to raise the ethylene feed up to the recycle pressure of 25-32 MPa (these two compressors are generally combined in a multi-service machine); and a secondary or hyper-compressor that raises the pressure up to the required value of 120-350 MPa (17,500-50,000 psi), usually in two stages (Fig. 1).

Fig. 1 –Hypercompressor installation

Although many solutions have been applied to this application, generally the compressor is driven by an electric motor.

Solutions have ranged from 8+4, 6+6, 6+8 to today's 10+10 cylinder arrangements to meet the challenges of the great increase of capacity from 10,000 kg/h in 1960, to the current level of 170,000 kg/h.

Advanced calculation methods are applied in order to obtain the highest levels of safety and reliability. The rotordynamic behavior of the train is a key determinant in the reliable operation of the entire system.

SYSTEM

The lineup of the system under consideration has a single-bearing electric motor installed between the two 10-cylinder frames that make up the first and second stages.

The motor has a rigid coupling on one side (2nd stage frame) and a flexible coupling on the other (1st stage frame).

The coupling (Fig. 2) has flexible membranes to transmit the torque and impart the proper elasticity to the system.

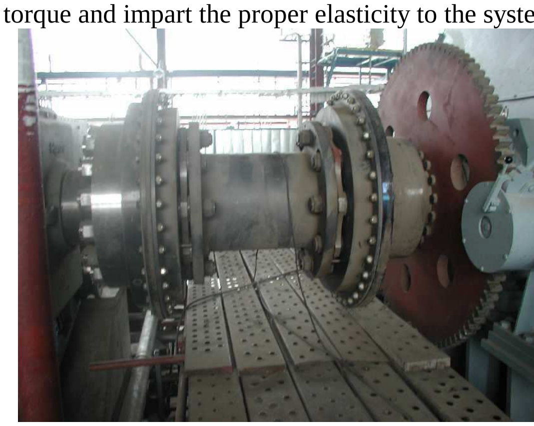

Fig. 2 – Multi-membrane flexible coupling

This coupling design is consistent with previous experience with special attention given to the crank phase and associated cylinder positions.

The design of the entire train must be optimized accounting for all critical conditions that will impact the operation of the system including:

- Dynamic loads acting on foundations
- Pulsations and related shaking forces
- Irregularity degree
- Torsional stresses

All these factors are considered and a detailed analysis from a torsional point of view is required to ensure that the individual units will operate reliably when coupled together.

PRINCIPLES OF TORSIONAL VIBRATION ANALYSIS

The general procedure for the analysis of torsional vibrations may be divided into the following steps:

- Reduction of the real shaft into an ideal model consisting of lumped masses (moments of inertia) and torsional springs
- Calculation of the undamped natural frequencies
- Calculation of the harmonic components of the torque acting on the lumped masses
- Calculation of the forced vibrations and the associated stresses

Modeling

For the analysis of the shaft line torsional behavior, an equivalent model is used to represent the real system geometry. According to the experience on hypercompressor trains, the equivalent model must be accurate enough to closely match the real system behavior at least for frequencies below the 20th multiple of the maximum rotating speed.

A model with a larger number of elements permits the evaluation of a higher number of vibration modes. In general these additional modes are not significant since they are related to high frequency / low amplitude exciting harmonics.

If the train components are accurately modeled, the undamped TVA is able to predict the train TNFs with a small margin of error with respect to the real damped ones. This is because most equipment has a low level of torsional damping. Thus the model is simplified in order to reduce the analysis calculation effort, but at the same time it is representative of the real system.

Basically the simplified model is represented by a number of flywheels connected to each other by means of massless shaft intervals with an appropriate torsional stiffness (Fig. 3).

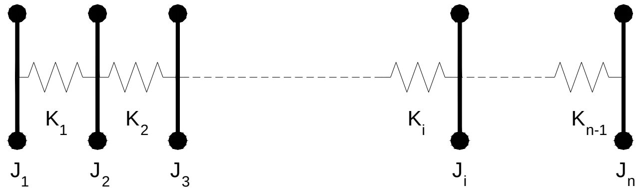

Fig. 3 – Equivalent system mass elastic model

The mass elastic model is developed on the basis of experience and with the guidance described in [2] and [3].

The compressor equivalent model consists of a series of flywheels representing both the crankshaft and the connected components (connecting rod, piston rod, crosshead). The mass elastic model of the crankshaft is created by lumping the inertia at each throw and calculating the equivalent torsional stiffness between throws (by means of a FEM analysis).

The coupling is modeled as a single torsional spring (vendor-supplied torsional stiffness) with the inertias of the associated coupling halves at each end. These inertias are added to the compressor and electric motor shaft end flanges respectively.

The gear wheel, exciter and rotor are shrunk onto the electric motor shaft. A dedicated lumped mass for each of these elements, as well as for the connecting flanges, is used.

The coupling and electric motor manufacturers provide the respective torsional interval stiffness.

Undamped torsional natural frequencies

Once the mass elastic model is completely defined, the next step is the calculation of the train's undamped TNFs.

The goal is to locate TNFs away from potential excitation frequencies that might come from both the driving and the driven machines.

Even though API618 standard [1] is not applicable to hypercompressor machines, it is considered as a guideline during the design stage.

API618 prescribes that TNFs be placed ±10% outside of any compressor running speed and ±5% outside of any other multiple up to the 10th harmonic. In addition, since the driver is an electric motor, it is recommended that the TNF be placed ±10% outside of the net frequency and ±5% outside of its second multiple.

Hypercompressor trains usually respect the API618 separation with respect to compressor harmonics since their particular geometry combined with low operating speed gives a 1st TNF higher than the 10th harmonic.

Moreover TNFs assume higher values as their order increases, and in practice only the first two or three modes of vibration are usually investigated.

Referring to Fig. 3, TNFs come from the resolution of the following differential equation system:

$$
\begin{cases}
J_1 \cdot \ddot{\theta}_1 + K_1 \cdot (\theta_1 - \theta_2) & = 0 \\
J_2 \cdot \ddot{\theta}_2 + K_1 \cdot (\theta_2 - \theta_1) + K_2 \cdot (\theta_2 - \theta_3) & = 0 \\
\ldots & \ldots \\
J_i \cdot \ddot{\theta}_i + K_{i-1} \cdot (\theta_i - \theta_{i-1}) + K_i \cdot (\theta_i - \theta_{i-1}) & = 0 \\
\ldots & \ldots \\
J_n \cdot \ddot{\theta}_{n1} + K_{n-1} \cdot (\theta_n - \theta_{n-1}) & = 0
\end{cases}
$$

The calculation of the undamped natural frequencies is a problem of eigenvalues and eigenvectors. Eigenvalues represent the natural frequencies while eigenvectors represent the mode shapes associated to the natural frequencies.

In case the TNFs fall outside the net frequency acceptable ranges of API618, individual train components should be changed or modified in accordance with the following guidelines:

- Modify the coupling torsional stiffness by an appropriate tuning of the spacer diameters
- Modify the electric motor shaft drive end diameters

The first point is the primary and easier solution. Generally the coupling is the most flexible shaft interval and the primary influence on the 1st TNF. However, excessive stiffness enhancement could result in an element as torsionally stiff as the motor shaft intervals. As a consequence, if further tuning is required to raise the 1st TNF, modifications to the motor shaft geometry should be considered jointly with the manufacturer.

In any case, the compressor crankshaft geometry is never modified to accommodate torsional application requirements.

Damped steady state torsional response

Best practice suggests performing a steady state stress analysis even though reference [1] recommends making it only in case of interference between TNFs and excitation harmonics.

The analysis should be performed for all operating load conditions including possible capacity control.

The system (Fig. 4) to calculate the damped force vibrations includes the associated effects of damping:

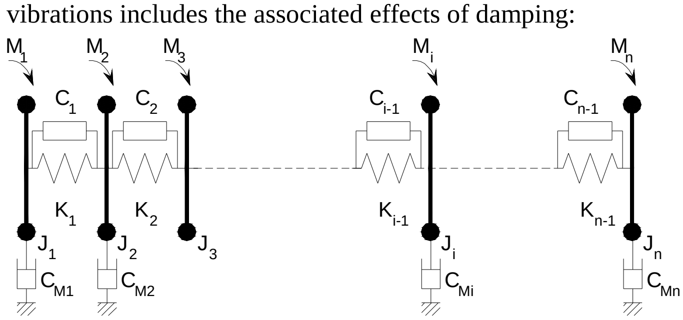

Fig. 4 – Damped Response Mass Elastic Model

The relevant system of differential equations is:

$$\begin{cases} J_1\ddot{\theta}_1 + K_1(\theta_1-\theta_2) + C_1(\dot{\theta}_1-\dot{\theta}_2) + C_{M1}\dot{\theta}_1 & = M_1 \\ J_2\ddot{\theta}_2 + K_1(\theta_2-\theta_1) + K_2(\theta_2-\theta_3) + C_1(\dot{\theta}_2-\dot{\theta}_1) + C_2(\dot{\theta}_2-\dot{\theta}_3) + C_{M2}\dot{\theta}_2 & = M_2 \\ \dots & \dots \\ J_i\ddot{\theta}_i + K_{i-1}(\theta_i-\theta_{i-1}) + K_i(\theta_i-\theta_{i-1}) + C_{i-1}(\dot{\theta}_i-\dot{\theta}_{i-1}) + C_i(\dot{\theta}_i-\dot{\theta}_{i+1}) + C_{Mi}\dot{\theta}_i & = M_i \\ \dots & \dots \\ J_n\ddot{\theta}_{n1} + K_{n-1}(\theta_n-\theta_{n-1}) + C_{n-1}(\dot{\theta}_n-\dot{\theta}_{n-1}) + C_{Mn}\dot{\theta}_n & = M_n \end{cases}$$

The torsional dampening is assumed to be hysteretic. For the shaft interval *i* it is calculated as:

$$C_i = \frac{K_i}{M_{a,i}\cdot\omega}$$

The resonance amplifier $M_{a,i}$ is assumed to be 40 for all shaft intervals in accordance with field test measurements.

The general expression of the steady state applied torques *Mi* includes a vibrating component superimposed on an average torque level as per the following formula:

$$M_i = M_{i,mean} + M_{i,1}\cdot\sin(\omega t+\alpha_1) + \dots + M_{i,m}\cdot\sin(m\cdot\omega t+\alpha_m)$$

Through a Fourier analysis, the compressor load torque is decomposed into a series of sinusoidal curves whose frequency is a multiple of the compressor running speed and whose modulus generally decreases as the harmonic order increases. The higher the number of harmonics considered, the closer the approximation to the input waveform becomes. Best practice suggests limiting the calculations to 25 harmonics (thus m=1 to 25 in the above formula).

Each applied torque must be properly phased with respect to the relative crankpin angular position on the crankshaft.

The electric motor driving torque is by design, supposed to be constant during steady state load conditions. But in reality, there is a small fluctuating torque component due to the interactions between the current pulsation of the windings and the shaft mechanical irregularity degree that can be neglected in this analysis.

The solution to the problem is based on the *principle of linear superposition*. Thus, the motion produced by the varying torque of a single cylinder is the sum of the motions that would be produced by the separate harmonic components of the torque curve if these were assumed to be acting alone. Moreover, the motion produced by a group of cylinders is the sum of the motions produced by the separate cylinders.

The system is composed of *n* linear differential equations with n parameters, tri-diagonal and symmetrical. Its solution is transferred to the solution of *m* differential equation systems (each of them related to the different harmonics of the exciting forces).

The solution of this system is the amplitude and the phase of the angular displacement of each mass.

The torsional stress on each spring element is given by:

$$\tau_i = k_i\,\frac{\vartheta_i-\vartheta_{i+1}}{W_{Ti}}$$

where W_{Ti} is the torsional constant of the spring element (for a cylindrical shaft = $\pi D^3/16$)

Within each shaft interval the torsional stresses should be calculated at the minimum cross sectional torsional resisting area:

- For intervals consisting of a series of cylindrical elements (e.g., electric motor shafts), the minimum area corresponds to the minimum diameter;
- For shaft intervals with a variable shape cross sectional area (e.g., crankshaft intervals), the minimum resisting area is usually given by either the crankpin or main journal.

A sensitivity analysis should always be carried out to take into account all the possible variances affecting the model data, due primarily to:

- Fluctuations of the rated running speed (e.g., net frequency oscillations). These shift the harmonics of the running speed.
- Variations of the mass elastic data (i.e., coupling stiffness, rotor inertia, etc.). These shift the TNFs.

A rigorous approach would require considering the combination of the independent variations of all of the above variables (running speed, all inertias and stiffnesses). This would result in a very large number of calculations. Another way to consider the relative position of the harmonics and TNFs is to calculate the torques and stresses over a range of ±10% of the rated speed. The highest values within that range are compared to the corresponding endurance limits.

The torsional stress is a function of time, and half of the difference between the maximum and minimum values gives the so-called alternating shear stress ($\Delta\tau$). For a given train, $\Delta\tau$ is strictly related to the relative position of the TNFs and exciting harmonics.

The particular cylinder arrangement of the hypercompressor crankshafts results in significant amplitude only for certain harmonics, which is related to the number of cylinders (i.e., in case of 6 cylinders frame, 3x harmonics are excited while, on 8 cylinders type, only 4x harmonics are significant). If different frame sizes are used for the 1st and 2nd compression stages, significant harmonics are given by both of them.

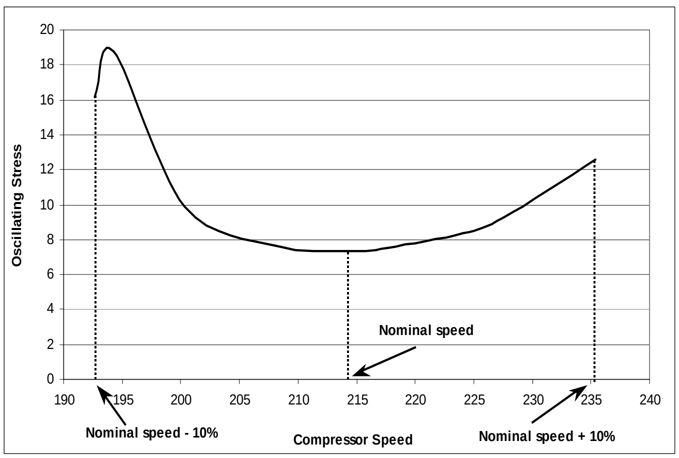

Fig. 5 – Typical shaft interval alternating stress

The TVA gives output diagrams (Fig. 5) for each rotor interval (for coupling interval the vibratory torque plot is preferred).

Transient torsional response

The transient torsional analysis requires a different and more time-consuming calculation method since it requires a direct integration of the motion equations. In this case the solution is time dependent.

The train model is the same as that used for the evaluation of damped steady state system response (Fig. 4).

Reduced versions are sometimes preferred to minimize the computer time required to perform the numerical solution of the motion equation. These reduced versions are obtained by lumping some of the adjacent masses and finally tuning one or more spring elements so that the reduced train 1st and 2nd TNF match those of the actual system. Thus in this case the TVA results are valid only for the unmodified rotor model intervals.

The differential equations of motions are similar to those of the steady state analysis, but they have different exciting torques.

Compressor loads M_i are given in a Fourier series up to the 25th harmonic. Inertia and gas pressure components are separately input since it is assumed that train speed variations affect only the inertia components while the gas cycle contribution is assumed to remain unchanged.

Driver applied torques M_i cannot be given in a Fourier series since transient excitations are not periodic.

Indicating with ω_1, ω_2,…, ω_n the angular rotational speed of each mass, where $\omega_i = \dot{\theta}_i$, the system equations can be rewritten as:

$$\begin{cases} \dot{\omega}_i = \dfrac{1}{J_i}\begin{pmatrix} M_i - C_{Mi}\omega_i - \left(C_{i-1}(\omega_i - \omega_{i-1}) + K_{i-1}(\theta_i - \theta_{i-1})\right) + \\ -\left(C_i(\omega_i - \omega_{i+1}) + K_i(\theta_i - \theta_{i+1})\right) \end{pmatrix} \\ (i = 1, \dots, n) \end{cases}$$

The above system consists of $2n$ 1st order differential equations which are functions of the $2n$ variables "$\theta_1,\dots,\theta_n, \omega_1, \dots, \omega_n$":

$$\begin{cases} \dot{\theta}_i = \omega_i \\ \dot{\omega}_i = \dot{\omega}_i(t, \theta_1, \dots, \theta_n, \omega_1, \dots, \omega_n) \end{cases}$$

Given the initial conditions it is possible to make a numerical integration of the system: Runge-Kutta 4th order with constant time step δt is the recommended integration method.

The appropriate choice of the time step is essential to correctly evaluate the high frequency transient vibrations. Its value must be related to the complexity of the train (number of lumped masses) and to the mass elastic values. In the case of a model with few masses (about 6) and if the shaft interval stiffnesses are uniform, a time step of 0.001seconds should be sufficient. Otherwise a time step of 0.0001seconds or less should be used.

The transient analysis output is generally represented by the torque and stress plots for each rotor interval as a function of time.

However the main results of the transient analysis are given in terms of :

- Peak shear stress in coupled machines
- Peak torque on couplings

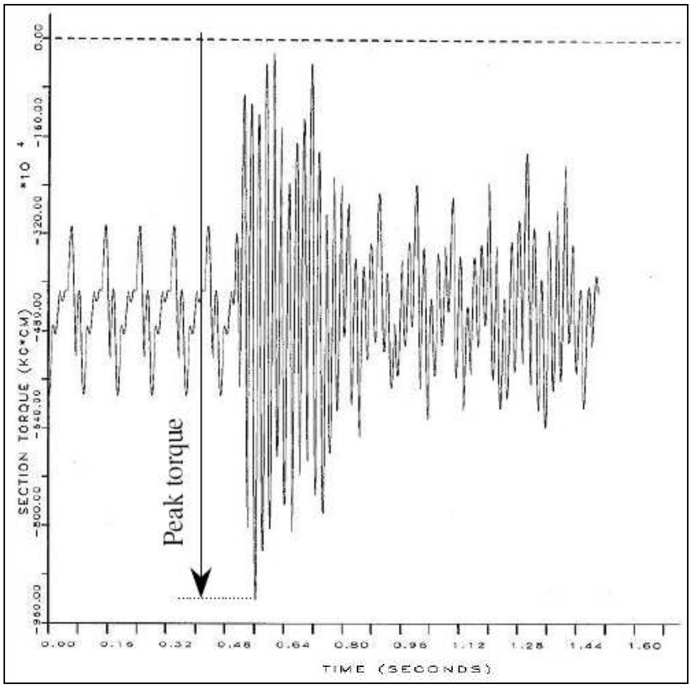

Fig. 6 – Three-phase short circuit TVA output

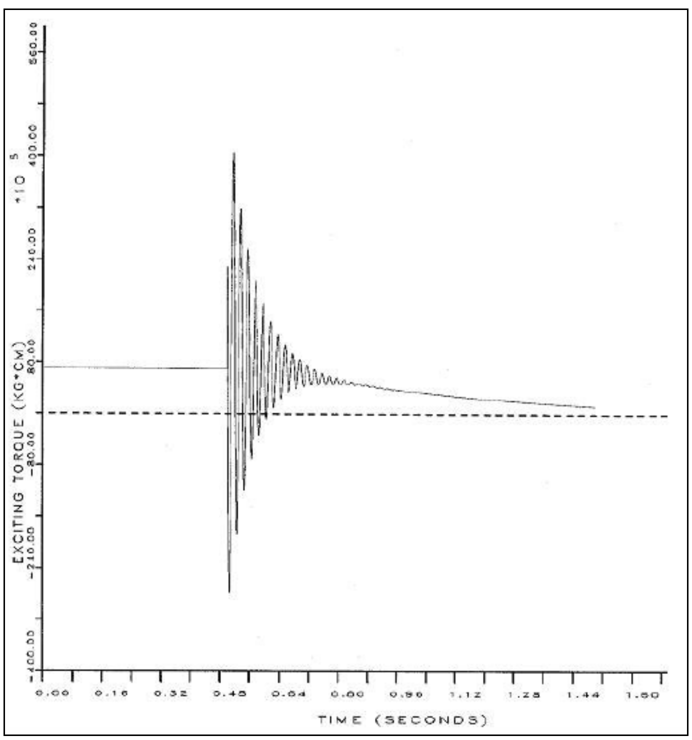

Fig. 7 – Three-phase short circuit exciting torque

ACCEPTANCE CRITERIA

For the compressor crankshaft, best practice suggests using a conventional torsional endurance limit $EL_{T,C}$ that depends only on its material properties (UTS_{shaft}) and does not take into account the bending loads coming from the cylinder throws.

This limit is referred to a stress calculated on a plane section without stress raisers, and thus it is the same for any crankshaft geometry and dimension. These characteristics introduce a significant simplification of the TVA with a remarkable reduction of computational and analysis time.

Within each crankshaft interval, the torsional shear stresses are calculated on the areas with minimum torsional resistance (crankpin or main journal cross-sections) and are compared to the conventional limit $EL_{T,C}$.

According to the Smith diagram, the mean shear stress has no effect on the shear fatigue resistance of the crankshaft until the maximum shear stress (mean + alternating) remains below the yield stress τ_{YS}. For this reason the mean shear stress can be neglected.

The compressor crankshaft conventional torsional endurance limit ($EL_{T,C}$) is calculated as:

$$EL_{T-C} = K \cdot UTS_{shaft} \cdot c_d$$

The factor K is based on extensive hypercompressor experience and takes into account the stress concentration factor, surface roughness factor and safety factor, while the size factor c_d is assumed to be 0.6 [4].

If $\Delta\tau_{max}$ is the maximum unintensified alternating torsional shear stress calculated over ±10% of the compressor running speed, the acceptability criterion is

$$\Delta\tau_{max} < EL_{T,C}$$

The bending stress contribution is neglected because of the short distance between the compressor throw centerline and adjacent main journal bearings.

The highest stress/torque amplitudes acting on both the driver and coupling must be compared to the endurance limits of each of the respective components.

In case the alternating torques and stresses are not within the allowable endurance limits a redesign must be carried out in order to achieve a wider separation margin between the TNFs and the high amplitude compressor harmonics.

The main concern is for the 1st TNF position because higher natural frequency orders are associated with high order compressor harmonics whose modulo is usually negligible.

The redesign activities must follow the prescriptions previously explained.

The optimum situation is achieved when the 1st TNF is located half way between two subsequent high amplitude compressor harmonics.

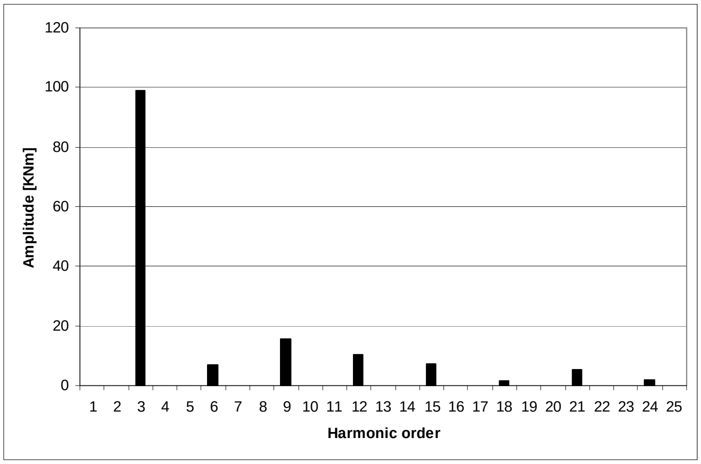

Fig. 8 – Torque harmonic content of a 12-cylinder compressor

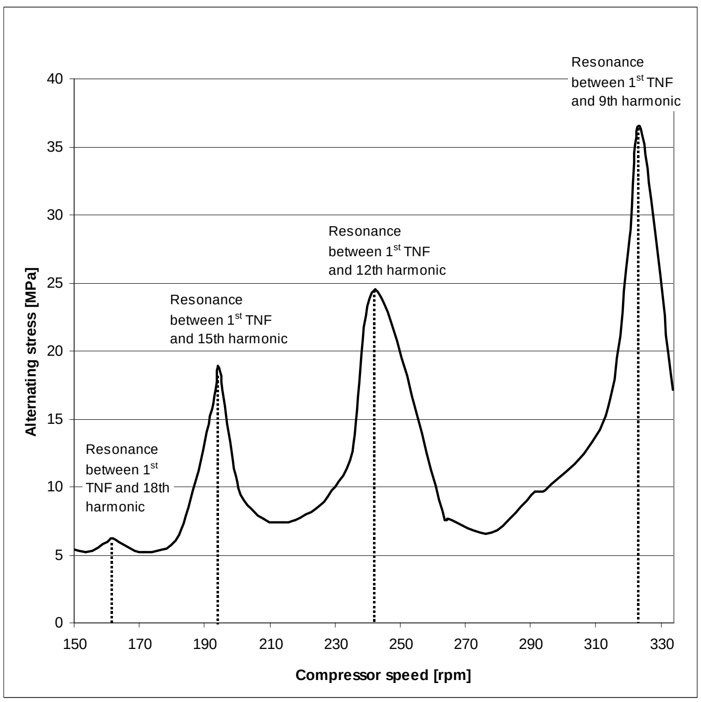

Fig. 9 – Alternating stress on the shaft of a 12-cylinder machine

For transient events, the compressor peaks are compared to the conventional allowable limits of the crankshaft as well as those of the electric motor shaft and coupling spacer.

In addition, the torques and stresses of the coupling and electric motor should be verified in detail by their manufacturers to assess operational reliability since they have special design and construction characteristics.

20 CYLINDER TRAIN TORSIONAL ANALYSIS

The compressor train investigated in the following analysis consists of an electric motor installed between two compressor units of 10 cylinders each.

The train speed is within the range of values normally used in this type of application. The motor has a rigid coupling on one side (2nd stage frame) and a flexible coupling on the other (1st stage frame).

As for the general case, the analysis requires the determination of the equivalent shaft, natural frequencies, steady state and transient exciting torques and relevant stresses.

Equivalent shaft

The mass elastic model (Fig. 10) was developed based on previous experience and was extrapolated from similar arrangements proven for up to 14 cylinders.

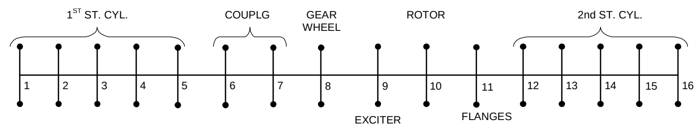

Fig. 10 – Mass elastic model

The coupling torsional stiffness is designed to have an acceptable separation margin with respect to the compressor excitation frequencies.

Torsional natural frequencies

The 1st TNF is 11.4 times the compressor speed. This implies an adequate separation (Fig. 11) with respect to the compressor exciting harmonics, and in particular with the highest and most adverse harmonic (10x).

Exciting torques

The significant compressor exciting harmonics are, for this particular arrangement, the 5x rotor speed and its multiples (Fig. 11).

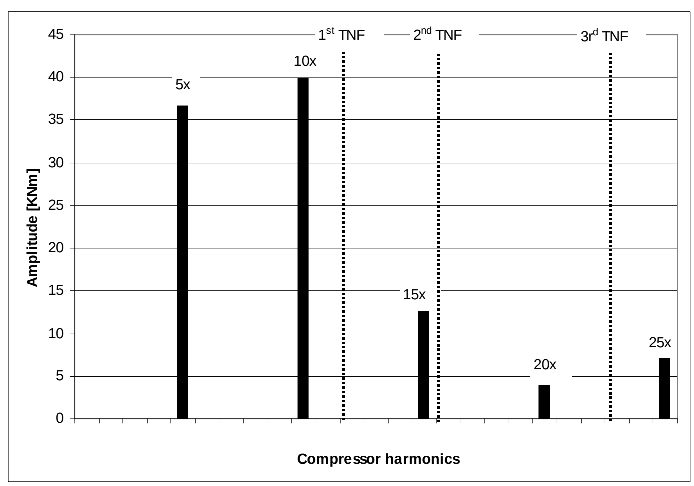

Fig. 11– Global compressor load torque harmonic content

A coupling stiffness variance of about 50% would be necessary for the train to reach resonance (Fig. 12).

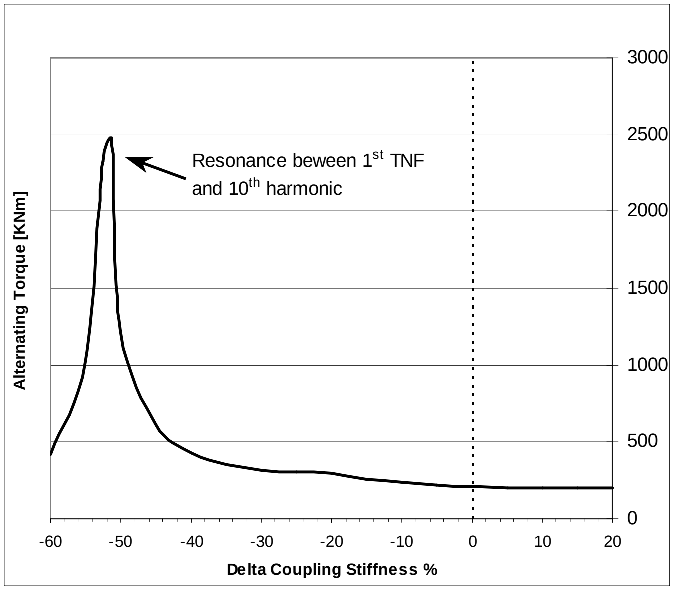

Fig. 12 – Coupling torsional stiffness sensitivity analysis

Steady state analysis

The steady state analysis was performed for all the load conditions and the results were investigated within an interval of ±10% of the compressor speed to cover all possible variances affecting the model data.

Torques and stresses (Fig. 13) related to the worst load condition are plotted over ±10% of train speed.

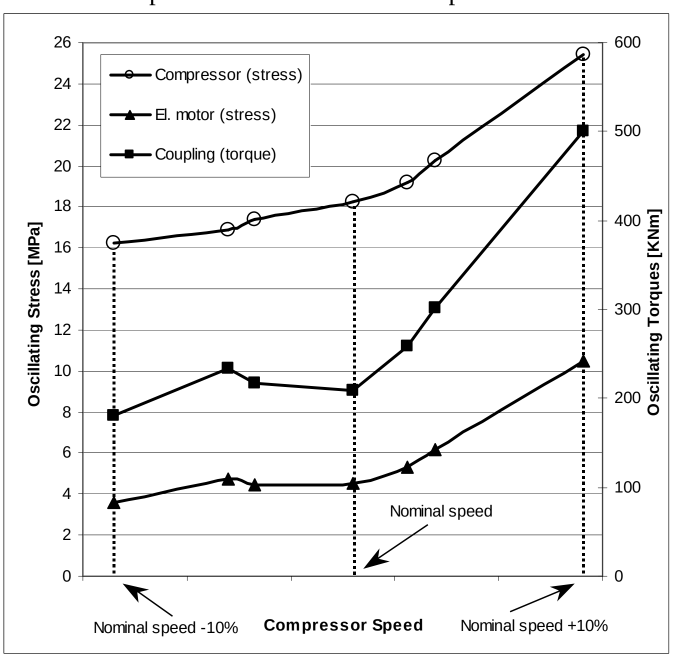

Fig. 13 – Oscillating torques and stresses

Similar diagrams are given for each rotor shaft interval.

The highest oscillating torques and stresses are compared to the endurance limits of each of the respective components.

TRANSIENT TORSIONAL ANALYSIS

In addition to the various operating loads it is important to evaluate the behavior of the whole train under transient situations including short circuit and start-up events.

3-Phase short circuit

A transient analysis using the complete mass elastic model (Fig. 10) was performed to assess the compressor train integrity during a 3-phase short circuit of the electric motor.

The transient event lasts about 0.7 seconds and in the simulation starts after 0.5 seconds to allow the position and speed of the masses to reach steady state under the worst torsional load condition.

The electric motor is subjected to a high frequency and high amplitude oscillating torque, whose peak is about 4 times the nominal value (Fig. 14).

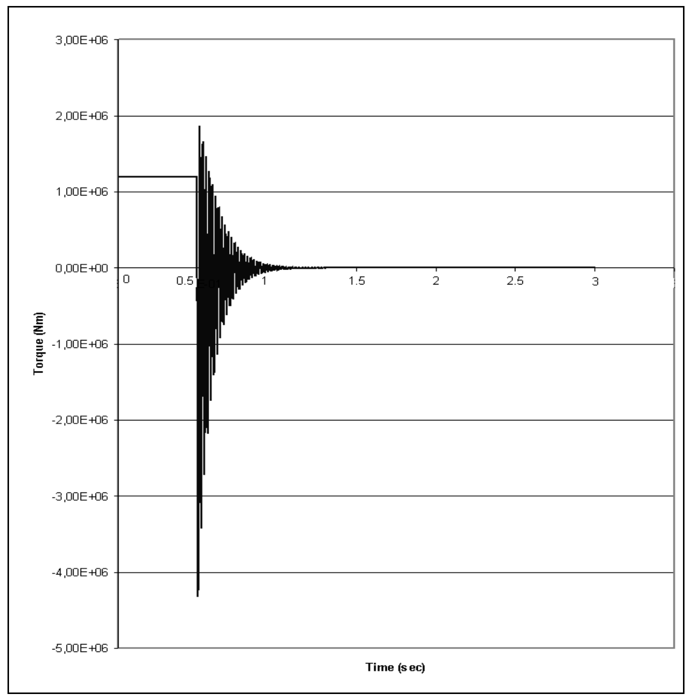

Fig. 14 – 3-Phase short circuit electric motor exciting torque

The solution is time dependent and the TVA results related to the coupling (Fig. 15) and electric motor (Fig. 16) indicate that the torsional transient response is strongly reduced by the system damping. The peak torques and stresses are about 1.5 times their average value.

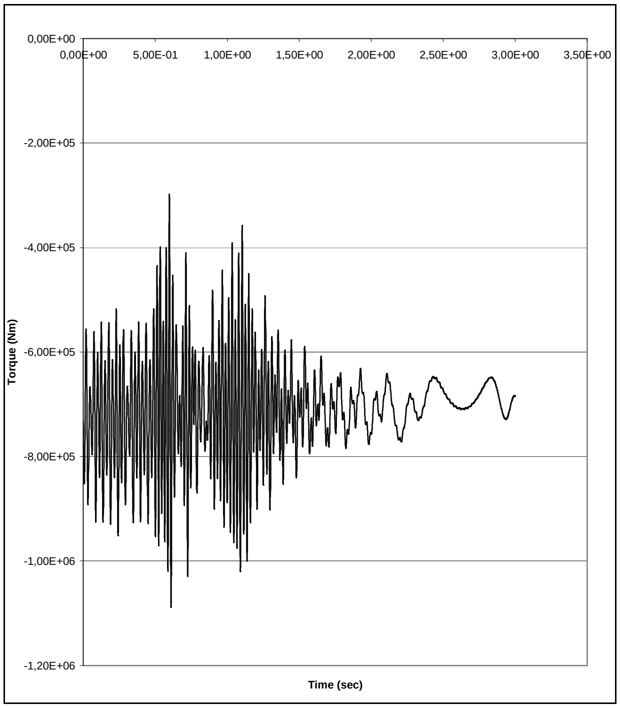

Fig. 15 – Oscillating torques on coupling

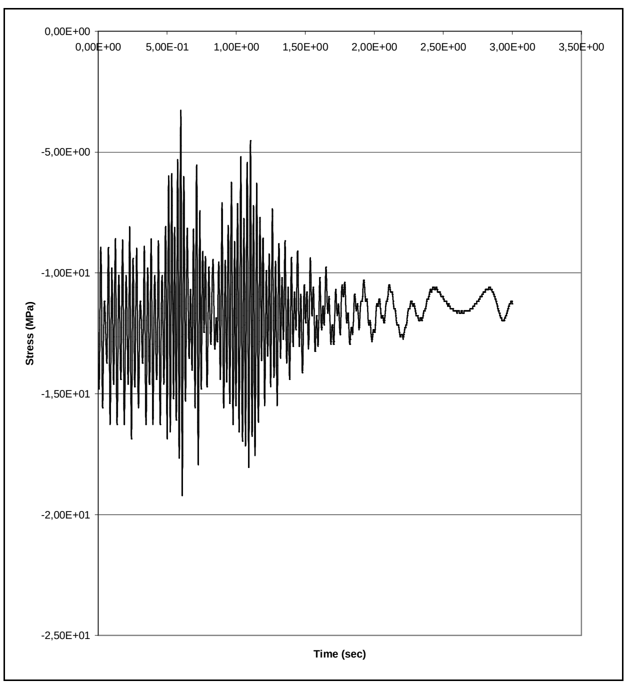

Fig. 16 – Oscillating stress on electric motor shaft

The transient condition deriving from the 3-phase short circuit event acts on the components of the entire train for about 2.5 seconds, creating significant amplification of the stress in the first part of the event. The stress is considerably reduced as the exciting torque abruptly decreases.

Start-up

In addition, a start-up transient analysis is performed to assess the shaft line integrity during the machine start. In this case a reduced model is used in order the make the calculations easier and faster.

The model is obtained by lumping the compressor masses of each frame and finally tuning the compressor-to-flange spring elements for the reduced train 1st and 2nd TNF to match the actual system.

Thus in this case the TVA results are valid only for the unmodified and most critical rotor model intervals (coupling and electric motor).

As the electric motor is of the synchronous type, the start-up load torque consists of a pulsating variable frequency component superimposed on the medium load torque curve (Fig. 17). The compressor is started unloaded and the nominal running speed is reached within 4 seconds.

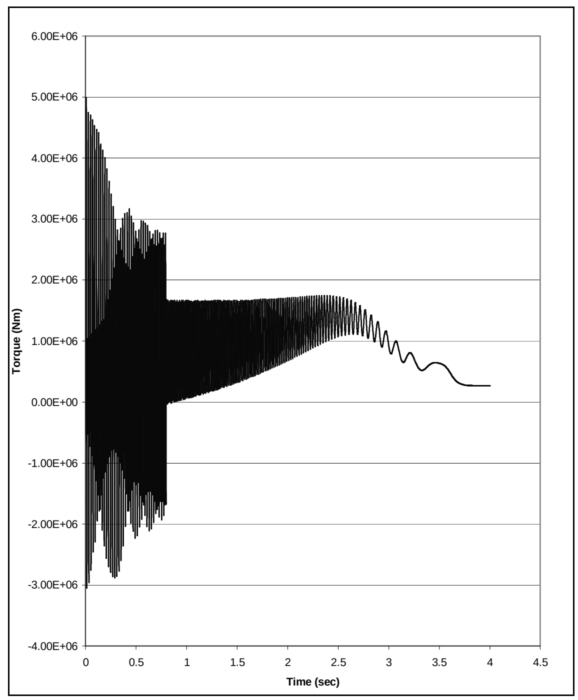

Fig. 17 – Start-up electric motor load torque

Similar brake torques were applied to both compressor lumped masses (Fig. 18).

The TVA results related to the coupling (Fig. 19) and electric motor (Fig. 20) show a high frequency response at the early stage of the machine start.

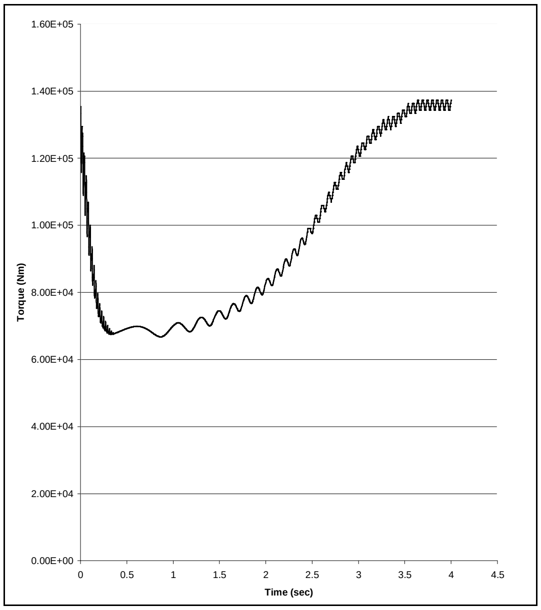

Fig. 18 – Start-up compressor break torque

Since this is a soft start (i.e., unloaded start), the peak stresses do not reach significant amplitudes and are even lower than those encountered under the worst operating condition (see Fig. 15 and Fig. 16 before the transient starts, i.e., within the interval of 0-0.5s).

A local amplification also occurs on all of the train components about 2.3 seconds after the machine start. This phenomenon takes place when the electric motor variable frequency pulsating component crosses the 1st TNF.

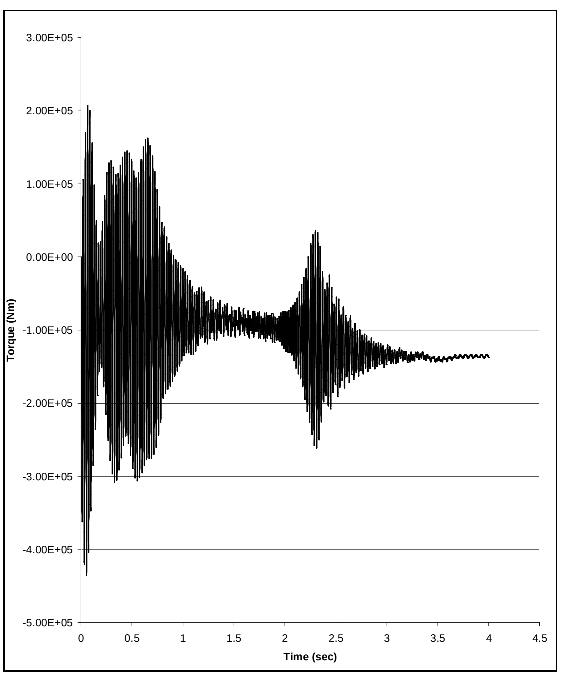

Fig. 19 – Oscillating torque on coupling

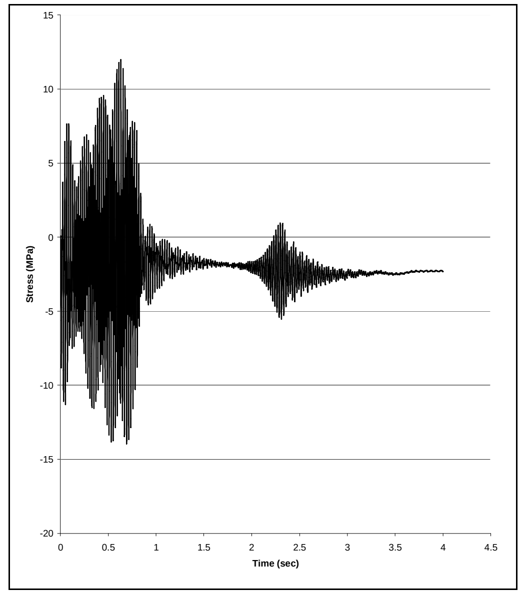

Fig. 20 – Oscillating stress on electric motor shaft

The transient vibration analysis output is given in terms of the peak shear stress and the results relevant to the drive machine and coupling are submitted to the manufacturers for approval, while that for the compressor crankshaft is compared with the material shear yield stress.

CONCLUSIONS

Large compressors, particularly those consisting of many cylinders, require thorough evaluation of the rotodynamics of the whole system.

This paper has discussed the investigation of the steady state and transient conditions for what is currently the largest system (20 cylinder machine) for the production of Ldpe. The analysis, which was conducted using advanced design criteria, indicated proper operation under all conditions assuring safe and reliable performance, and a high level of availability for the hyper-compressors and the entire plant.

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