TORSIONAL-DAMPING—TRANSIENT AND STEADY STATE

by
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ABSTRACT

Couplings play an important role in the solution of many turbomachinery torsional problems. Due to inherently high inertias, many systems have torsional natural frequencies in the units' operating range, and operational problems may result. The coupling is a convenient location in the train to make system torsional changes with a minimum effect on other system parameters. These modifications may incorporate a stiffness correction, the addition of damping, or both, in order to torsionally tune the system for reliable operation.

TORSIONAL PROBLEMS

All torsionally flexible drive trains are subject to nonsteady or oscillatory excitation torques during the normal operation of the system. These excitation torques can be an inherent function of either the driver or the driven equipment and, when superimposed on the normal operating torque, may appear to be of negligible concern. However, when combined with the high inertia loads of many turbomachinery trains and a torsional resonant frequency of the system, these diminutive ripples can result in a tidal wave of problems.

As a result, the traditional role of a coupling to transmit torque with slight misalignment can be expanded to incorporate the tuning and de-sensitizing of torsionally responsive operating systems.

The torsional resonant response of a system is an interaction of all the components in the train. Calculation of torsional natural frequencies is based on the entire system and these frequencies are valid only for that given arrangement. If any component of the train is replaced by an item with torsional characteristics different from the original, the system torsional response must be recalculated and new torsional natural frequencies determined. Occasionally, an original equipment manufacturer is requested to calculate the torsional and lateral critical speeds of the supplied item. Unfortunately, the purchaser is unaware that this request is of limited value, since the torsional response of a single item in a train is meaningless. Likewise, a torsional shop test will yield useless results if the train is not assembled and tested with every item destined for the field.

The interesting aspect of torsional problems in turbomachinery systems is that the first indication of a problem is usually a ruptured shaft or coupling in the field. Silent and deadly, a torsional resonance can lurk at synchronous or nonsynchronous frequencies, and be steady or transient in nature. Once a torsional problem is found in the field and the excitations are determined to be inherent in the system, the only solution available to put the system back online quickly is to decouple the excitation source or to dampen the system response.

Some sources of torsional excitations encountered in the operation of turbomachinery systems are contained in Table 1. A quick review of system torsional response may help explain how a resilient coupling works. A torsional single degree of freedom system, with a disk having a torsional moment of inertia, J, connected to a massless torsional spring, K, is depicted in Figure 1. Newton's law for a rotating body states:

$$T = J\alpha \tag{1}$$

where

T = torque (in-lb)
J = mass moment of inertia in torsion (lb-in-sec^2)
α = angular acceleration (rad/sec^2)

Angular acceleration is the time rate of change of angular velocity, which in turn, is the time rate of change of angular displacement.

Table 1. Typical Sources of Torsional Excitation in Turbomachinery Systems.

Source		Frequency	Mode
Gear	Mesh	No. of Teeth x Gear RPM	Steady
	High Spot	1x Gear RPM	Steady
	Quality	2x Gear RPM	Steady
Steam Turbine	Nozzel Passing	No. of Blades x RPM	Steady
Electric Motor	Variable Frequency Drive	No. of Pulses x Motor Line Frequency	Steady
	Synchronous Start	2x Slip Frequency	Transient
Compressors	Centrifugal Surge	Broad Band	Transient
	Reciprocating	No. of Cylinders x RPM	Steady
Electrical Faults		Varies (See References)	Transient

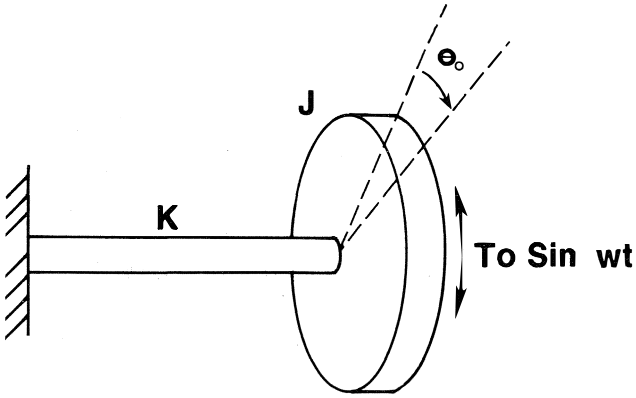

Figure 1. Single Degree of Freedom Torsional Model.

$$\alpha = \frac{d\dot{\theta}}{dt} = \frac{d^2\theta}{dt^2} = \ddot{\theta} \tag{2}$$

where θ = angular displacement in radians

$\dot{\theta}$ = angular velocity in radians/sec = $\frac{d\theta}{dt}$

Assume that an oscillatory excitation torque of $T_o\sin(\omega t)$ is applied to the system in Figure 1. By definition, when the excitation frequency coincides with the torsional natural frequency of the model, all torques will balance and the system will be in a state of resonance.

For equilibrium, the following equation of motion must be satisfied:

$$J\ddot{\theta} + C\dot{\theta} + K\theta = T_o\sin(\omega t) \tag{3}$$

where

$J\ddot{\theta}$ = inertia torque

$C\dot{\theta}$ = damping torque

$K\theta$ = spring torque

In its simplest form, damping is neglected and no external forcing function is applied, resulting in the equation:

$$J\ddot{\theta} + K\theta = 0 \tag{4}$$

Separating variables yields

$$\ddot{\theta} = -\frac{K}{J}\theta \tag{5}$$

The general solution of this second order differential equation is

$$\theta = C_1 \sin \omega t\sqrt{\frac{K}{J}} + C_2 \cos(\omega t)\sqrt{\frac{K}{J}} \tag{6}$$

Assuming the disk is displaced 0_o radians and then released, the following initial conditions apply:

$$\text{at } t=0 \qquad \theta = \theta_o \qquad \dot{\theta} = 0$$

The first condition yields:

$$C_2 = \theta_o \tag{7}$$

Differentiating and substituting the second condition yields:

$$C_1 = 0 \tag{8}$$

The resulting specific solution is therefore:

$$\theta = \theta_o \cos(\omega t)\sqrt{\frac{K}{J}} \tag{9}$$

The period (T) of this vibration is:

$$T = 2\pi\sqrt{\frac{J}{K}} \tag{10}$$

The reciprocal of the period of the vibration is:

$$\frac{1}{T} = \frac{1}{2\pi}\sqrt{\frac{K}{J}} = f_n \tag{11}$$

where f_n represents the torsional natural frequency of the system in cycles per second.

For a complicated multimass system like that shown in Figure 2, the equations of motion become quite complex, especially if a forcing function exists and internal damping is included. Inertial damping (damping to ground) is neglected. The equations of motion for this system would take the form:

$$\begin{aligned}
J_1\ddot{\theta}_1 + C_1(\dot{\theta}_1 - \dot{\theta}_2) + K_1(\theta_1 - \theta_2) &= T_o\sin(\omega t) \\
J_2\ddot{\theta}_2 + C_2(\dot{\theta}_2 - \dot{\theta}_3) - C_1(\dot{\theta}_1 - \dot{\theta}_2) + K_2(\theta_2 - \theta_3) - K_1(\theta_1 - \theta_2) &= 0 \\
J_3\ddot{\theta}_3 + C_3(\dot{\theta}_3 - \dot{\theta}_4) - C_2(\dot{\theta}_2 - \dot{\theta}_3) + K_3(\theta_3 - \theta_4) - K_2(\theta_2 - \theta_3) &= 0 \\
J_4\ddot{\theta}_4 + C_4(\dot{\theta}_4 - \dot{\theta}_5) - C_3(\dot{\theta}_3 - \dot{\theta}_4) + K_4(\theta_4 - \theta_5) - K_3(\theta_3 - \theta_4) &= 0 \\
J_5\ddot{\theta}_5 - C_4(\dot{\theta}_4 - \dot{\theta}_5) - K_4(\theta_4 - \theta_5) &= 0
\end{aligned} \tag{12}$$

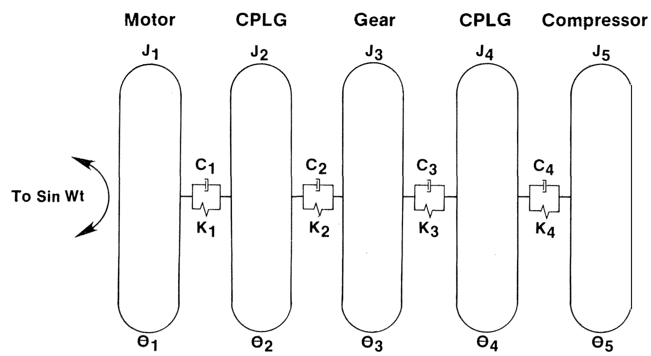

Figure 2. Multi-Mass Torsional Model.

The solutions to problems of this magnitude have been documented by Den Hartog, Doughty, and others. While the rigorous solution to the multimass damped system is difficult, several interesting points should be made:

- An nth degree of freedom system will have (n − 1) natural frequencies.
- Equation (11) indicates that the torsional natural frequencies of a system are a function of the torsional inertias and stiffnesses of the system.
- The natural frequencies of a damped system are essentially the same as for an undamped system for all realistic values of damping.
- The displacements of the system at resonance will be a function of the magnitude of the driving or excitation source and damping.
- Damping in the system represents dissipation of vibratory energy which reduces the amplitudes in the system.
- Damping is a function of the angular velocity change across the damper.

As mentioned previously, the torsional response of a system is a function of the stiffnesses and inertias in the train. While some parameters of the system can be changed, the inertia, J, is usually fixed by the basic process. For example, the J value for a compressor is largely a function of wheel diameters and widths. These, in turn, are set by the required load and flow. Theoretically, the hub and shroud thickness could be varied to tune the system. However, any change in impeller hub thickness, shroud thickness or shaft diameter may significantly alter the lateral response of the unit. These modifications would lead to tradeoffs that probably would not be considered acceptable from a process or operational point of view. As a necessary competitive procedure, some machine manufacturing must proceed in parallel to the engineering analysis. Basic changes to wheels and shafts while torsional problems are worked out would definitely slow the process down and push back delivery. The driver cannot be designed until the system inertia is known, yet the torsional analysis cannot be completed until driver parameters are set. It would appear that an iterative design and analysis procedure would be required and could go on for quite some time. This would complicate things considerably if it were not for the couplings.

ROLE OF THE COUPLING

Most operating people mistakenly believe that couplings are only torque transmitters between different pieces of equipment with a secondary function of handling misalignment. While couplings are typically sized and chosen based on the aforementioned requirements, they can take on another role: that of a "torsional fixer."

Once the driver and driven equipment have been chosen and it is determined that none of the items will be subject to any lateral vibration problems, the system torsional analysis is performed. If a calculated torsional natural frequency coincides with any possible source of excitation (Table 1), the system must be tuned in order to assure reliable operation. A good techniques to add to the torsional analysis was presented by Doughty, and provides a means of gauging the relative sensitivity of changes in each stiffness and inertia in the system at the resonance in question.

Typically, the couplings in a turbomachinery train will be the softest torsional elements in the system. As a result, they represent the controlling factor in the system overall stiffness, since the total system spring rate cannot exceed the stiffness of the softest spring.

Should analysis indicate that a coupling is in a sensitive position, then a small amount of custom design in a relatively standard coupling can accomodate the tuning away from the critical in question. One note of caution: while changes in stiffness of inertia may change a given resonance, their effect on the other criticals must also be determined, since any change in the system will result in a new set of resonant frequencies.

If modification to coupling stiffnesses cannot effectively adjust the system, or if unanticipated excitation frequencies are encountered in the field, the designer has another option to desensitize the train. This last resort approach involves the resilient, or torsionally damped, coupling. These couplings add damping to the system in order to dissipate vibrational energy and effectively reduce twist amplitudes during a resonant condition. While this does not eliminate the source of the problem, it does allow the manageable operation of the unit under resonant conditions.

Once again, the use of a resilient coupling requires that its application in the system during a resonant condition must be at a location sensitive to the applied damping. Since damping is a function of the relative velocities of the coupling hubs, little would be gained by placement at a node or points of small angular velocity changes.

COUPLING TYPES AND SPECIFICATION

A guideline for specification of torsional damping and/or resilient couplings can be found in Appendix D of American Petroleum Institute (API) Standard 671, First Edition, "Special-Purpose Couplings for Refinery Service." The majority of torsional damping couplings currently in use can be classified into five major categories:

- Quill shaft
- Metal-metal resilient
- Elastomer, compression
- Elastomer, shear
- Fluid

The torsionally soft or resilient coupling like the quill shaft coupling and the metal-metal resilient coupling transmit torque and handle misalignment through springy metal strips, coils, discs and diaphragms. They will tune the system by changing the spring constant K. The oil filled metal strip or coil type (Figure 3) will also reduce torsional vibration amplitudes by up to 30 percent with fairly linear characteristics.

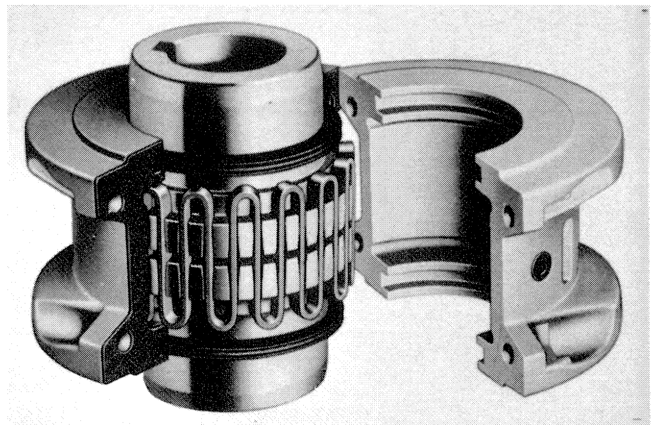

Figure 3. Metal-Metal Resilient Torsional Coupling.

The elastomer (compression) and elastomer (shear coupling) provide both tuning and damping to the system. In some cases the two functions interact, that is, the stiffness (K) or damping (C) may be a function of the other. The elastomer couplings are torsionally softer than the metal-metal resilient couplings, but will introduce higher levels of damping into the system.

There are two major types of elastomer compression couplings. One is the torus type (Figure 4), in which the elastomer is bolted directly to the coupling hubs. The other is the Holset type (Figure 5), with the elastomer held in place by the hub geometry. These couplings will have a higher torque capability than an elastomer shear type coupling, and in the case of the torus design, the bolting of the elastomer to the hub preloads the element, resulting in coupling stiffnesses typically higher than capable with the Holset type coupling.

The elastomer shear type coupling (Figure 6) places the elastomeric element in shear when transmitting torque. The shear load on the elastomeric elements allows the coupling to flex under load, resulting in a softer spring rate than the elastomeric compression coupling for a given torque capability.

The fluid coupling (Figure 7) will torsionally decouple the drive train, isolating the source of excitation. This type coupling also allows a soft start capability, without reduced motor voltage, as well as providing torque overload protection.

Probably the obvious question that arises after the preceding discussion is, "Why not use a soft, highly damped coupling in every case and side step the torsional problem altogether?" As always seems to be the case, with the solution comes some compromise. On the good side, these various couplings provide greater latitudes in the selection of coupling characteristics to solve the torsional problems mentioned earlier. Furthermore, if analysis fails to predict a torsional problem and one arises in the field, the couplings are a quick and inexpensive means of bringing the unit back on line.

On the bad side, many of the elastomeric type couplings are highly nonlinear in their characteristics. The elastomeric

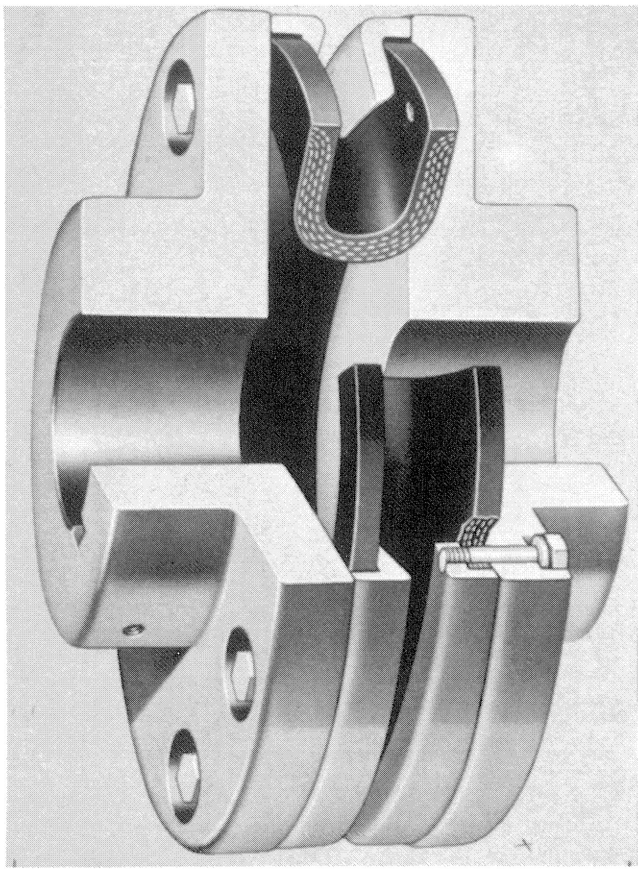

Figure 4. Elastomer, Compression Torsional Coupling.

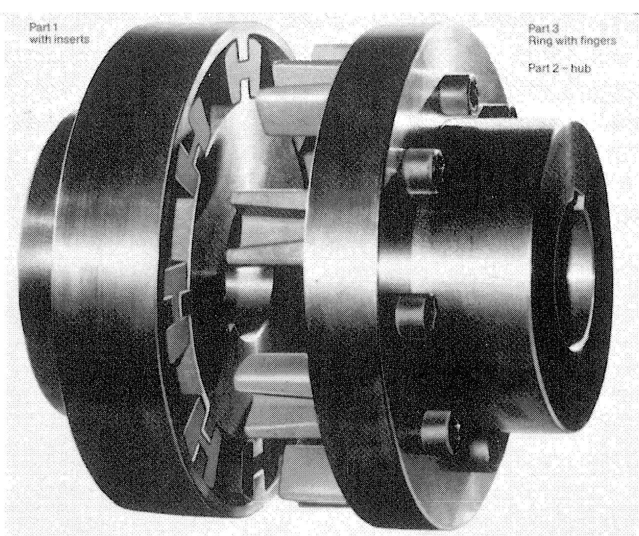

Figure 5. Elastomer, Compression Torsional Coupling.

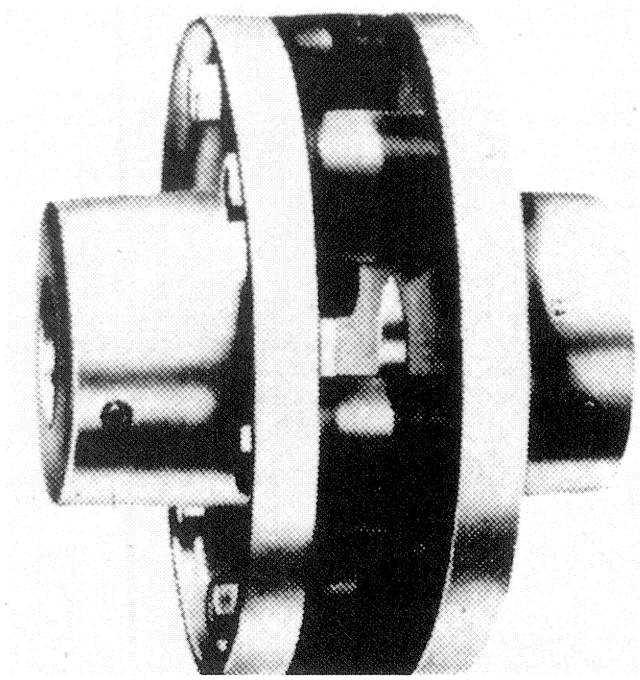

Figure 6. Elastomer, Shear Torsional Coupling.

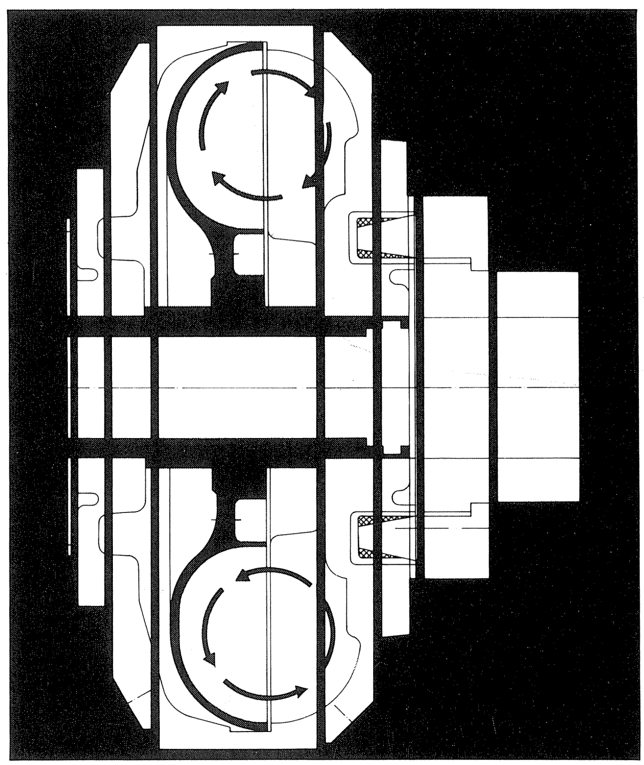

Figure 7. Fluid Coupling

Holset type couplings are very soft at small deflections under low loads, but once the elastomer has filled the available squeeze space, the coupling is effectively rigid. This makes reduction of system response difficult, unless the load and coupling characteristics are well defined prior to installation.

The elastomeric couplings generally do not have a life factor equivalent to a gear or flexible disc coupling. This is further complicated by the fact that if the coupling is to provide damping, the dissipated vibrational energy is converted to heat, which can further shorten the life of the element. Likewise, any elastomer will degrade with time, resulting in a coupling with characteristics that are non-linear with both load and time of service.

The fluid coupling, while inherently isolating and thus decoupling the system, will always have a two to three percent slip or inefficiency under operating conditions. Under high horsepower loads, this requires external cooling of the coupling fluid and the power penalty may be too great to justify this type of coupling.

CASE HISTORY

The remnants of a torsional failure on a 1000 kW motor-gear-compressor train driven by a constant current, variable frequency drive system are portrayed in Figures 8 and 9. Such a failure can bring home the point that torsional failures can be quite dramatic. This system had a torsional critical speed in the variable frequency drive excitation range, and on at least one occasion, had an electric breaker reclosure with the motor at part speed.

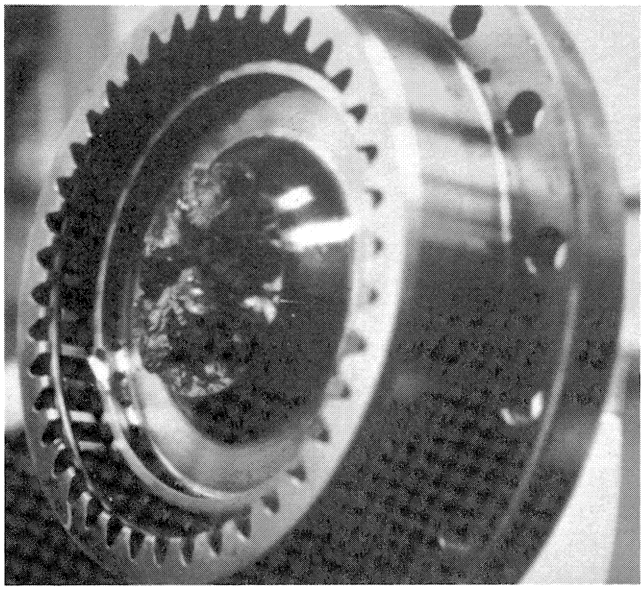

Figure 8. Torsional Rupture at Coupling Hub.

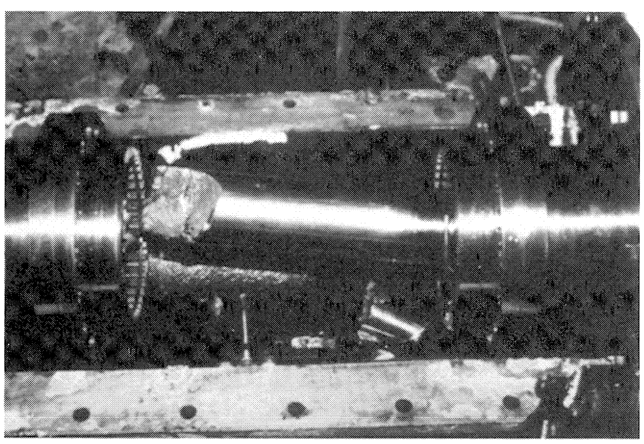

Figure 9. Coupling Torsional Ruptures.

The system was redesigned using an elastomeric compression coupling where stiffness and damping were specified and provisions were made to avoid motor restarts at part speed. Because the cause of failure was never completely explained, the elastomeric coupling may not have been the only solution. However, the system has operated for the last several years without any torsionally related problems.

CONCLUSION

Most torsional vibration problems can be identified prior to installation of a turbomachinery system through a thorough torsional analysis. Typically, the system can be tuned by varying the torsional stiffness of a spacer type coupling with very little modification to a standard design. However, if tuning the system with coupling stiffness alone turns out to be ineffective, or if field operation results in a system failure, a resilient, damping type coupling may be the answer.

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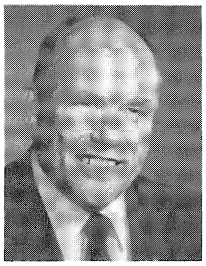

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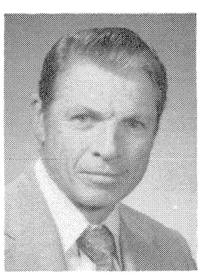

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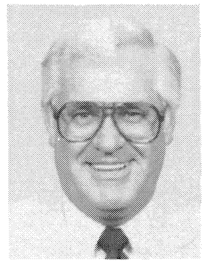

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