and companies that we were visiting in Scandinavia, he periodically kept asking why I was there. I kept giving him the same answer: that I have visited gear and machine tool manufacturers in over a dozen countries during the past five years, and that I always found it interesting to see how other people were doing the same type of work that we do at Arrow Gear. I told him we also reciprocated by permitting foreign visitors to tour our plant, and that it was no longer an Asian/European/USA economy, but a worldwide economy that we are living in, and that we all have to get along together in this environment. I also told him that I was trying to help the AGMA to get its standards accepted in the world marketplace.

After the inverview was over, I asked Mr. Lundstrom why he kept questioning me as to why I was there. He said with a smile, the question wasn't why were we in Scandinavia and why did we visit these specific companies, but why weren't we also visiting their universities and their technical centers as the Japanese visitors do. Not only are the Japanese the No. 1 visitors to Scandinavia, but they spend literally months there visiting all the companies they can and spending weeks at their universities and tech centers. He also said that he and most Scandinavians thought America was a wonderful country, and that it was a shame that the U.S. manufacturing people were not spending more time in Sweden and in other corners of the world, absorbing their intellectual knowledge and bringing it back to the United States as the Japanese do for Japan.

One of the things that has continued to bother me a great deal in my numerous travels around the world, is the number of consumer products, complete assemblies, or component parts that are being made by foreign comanies and end up being sold in the United States. This is not only due to commercial overseas joint ventures by major U.S. corporations, but also by the U.S. government and branches of the Armed Forces buying foreign components and products.

For example, I discovered during this Scandinavian trip alone that the U.S. government buys component parts from Kongsburg Vaapenfabrik, Norway, for the F-16 fighter, ASN Pentagon missile, and that the U.S. Air Force gave them $8 million dollars for the development of an automated machining cell for jet engine rotors. When there are so many U.S. manufacturing companies going out of business daily, we should at least enjoy the business of our own government.

(continued on page 11)

FROM THE

Give Your Gears a Break — Select the Right Coupling!

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How important is the right choice of coupling in determining successful machine design? Consider the following example. A transmission of appropriate size was needed to transfer the speed of the engine driver to that of the driven generator. The transmission was properly selected and sized to endure the rated power requirements indefinitely, but after only a short time in operation, it failed anyway. What happened? The culprit in the case was a coupling. It provided the necessary power and protection against misalignment, but it lacked the ability to isolate the gears from the torque peaks of the diesel engine.

All reciprocating engines produce uneven torque with peaks much higher than the value of the rated torque. The torque fluctuations are caused by the variations in the tangential forces acting on the crankpins. The engine flywheel smooths the fluctuations to some extent; the more inertia it has, the better job it does. Unfortunately, the more inertia a flywheel has, the heavier it usually is, and therefore, invariably, the light and powerful engines of today are "jerky" in this respect. Their flywheels are light to keep the engine mass down. On the other hand, they often drive relatively heavier equipment. The rotating components of the driven equipment may very well have more inertia than the engine flywheel. We will explore the negative effects these features have on the load imposed on the gears in transmissions, power takeoffs, step-up gears, and similar mechanisms and discuss possible countermeasures.

Torque Fluctuations at the Prime Mover Output

One of the highest torque fluctuations is present in diesel engines. The size of the fluctuations depends on many fac-

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tors, the obvious ones being the load on the engine, which determines the torque peaks before the flywheel, and the flywheel polar inertia, which determines how much lower the peaks are on the output shaft. The relative size of the average torque, the maximum instantaneous torque, and the influence of the polar inertia of the flywheel are illustrated here in the case of a common prime mover, the four cylinder diesel engine.

Fig. 1 shows three traces of the instantaneous torque of such an engine sensed at three different locations. The high amplitude line represents the torque at the interface of the crankshaft with the flywheel. The straight line represents the torque that would be sensed after the flywheel if the flywheel had an infinitely large inertia, or if the driven equipment had infinitely low inertia. The line also represents the torque that would be displayed by a dynamometer gauge; i.e., the average torque on the output shaft. The medium amplitude line represents the torque that might be experienced on the output shaft in a real installation.

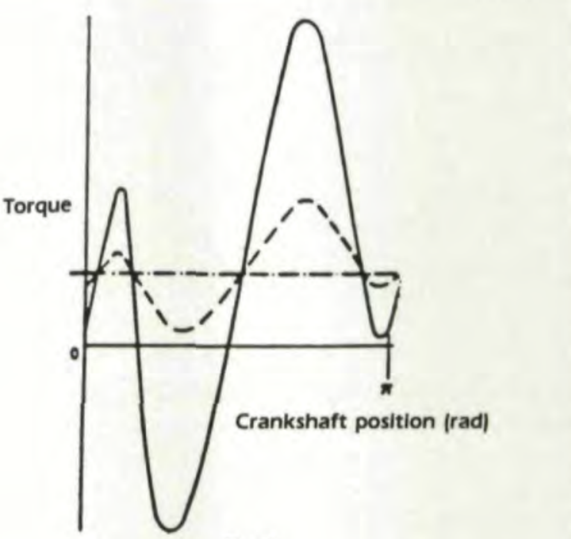

Fig. 1

It is apparent that at a steady load and speed, the peak of the medium amplitude will be higher with higher equipment inertia and lower with lower equipment inertia for a given engine. This consideration has a practical implication for the design of flanges: one ought to be skimpy with flanges on the equipment side and stay with as small diameters as possible. By contrast one may be generous with flanges on the engine side. Inertia added to the engine side benefits the transmission.

Coupling

There is no practical way to make engines with infinite polar inertia, nor can the equipment have zero inertia. Thus, some magnitude of the medium amplitude is inevitably input into the transmission. Obviously, the lowest possible amplitude is desired. Torsionally soft couplings have always been used to insure this condition. Today a wide selection of such couplings exists.

Ideally, the coupling should be so soft that it smooths out the torque fluctuation completely. It should store the excess energy during the time the peaks occur and release it during the time the valleys occur. Such a coupling is, of course, impossible to build. Furthermore, because of hysteresis, the amount of torque fluctuation the coupling can withstand is limited by its capacity to rid itself of heat.

The torsionally soft coupling is engaged to decrease the amplitude that reaches it to the level manageable by the driven equipment. Unfortunately, not all soft couplings are suitable in all applications. There are also applications where a soft coupling does more harm than good to the driven equipment, as in the case where the transmission is relatively small and the inertia of the gears can be neglected.

Resonance

Gear and transmission problems often stem from the overstress due to the occurrence of the torque peaks. The magnitude of the peaks is determined by the influences described so far and also by another factor. Consider that the peaks occur rhythmically. If the frequency of the peaks coincides with one of the natural frequencies of the system, the stress peaks grow with time until a component fails. More couplings and gears fail because of the overstress in resonance than from all other causes combined.

The task, then, is to analyze the whole engine-to-equipment system to predict its natural frequencies. Often, only the first natural frequency needs to be known, as the higher ones can be made to lie outside the operating range by the selection of the coupling stiffness. If the equipment operates at a fixed speed, it is relatively easy to select a coupling whose torsional stiffness is such that resonance does not occur in the vicinity of the operating speed. Many resilient couplings are on the market, and some manufacturers provide information on how to make the right selection. However, when it comes to the wide-speed-range operation, there is no coupling that can run without introducing resonance at some speeds. The task is to select the coupling that covers the widest speed range, learn the speeds where the system could resonate, and avoid operating at those speeds.

Transmission Protection

A torsionally soft coupling should be used wherever possible to protect the transmissions from overload due to excessive torque peaks. The softer the coupling is, the lower the transmitted peaks will be. Furthermore, the softer the coupling, the lower the highest resonant speed will be. A system with a low stiffness coupling will resonate only in the low speed range where the nominal torque is normally very low. Thus, resonance should be experienced only during start-up and shut-down, and the resulting amplitudes should be safe. For the protection of the coupling, these steps should be passed through quickly to limit the time available for the amplitudes to build up. The coupling will last indefinitely if the low speed range is passed through so fast that the stress amplitudes do not reach dangerous levels or the heat does not build up and destroy the resilient material.

A certain amount of inherent damping and nonlinear characteristics in a coupling help to prevent the growth of the amplitudes in resonance. A coupling with these two features should be selected if the operation close to resonance cannot be ruled out. The transmission will experience higher peaks at the steady state load but the increase is generally negligible in comparison to the substantial decrease in resonance.

There are applications where the full torque must be transmitted at a relatively low speed. The very low speed region is safely served by a very stiff coupling. With the stiff coupling, the system can be operated below all significant resonant speeds. The torque peaks transmitted to the transmission are then a function of the exciting pulses of the prime mover and, as discussed earlier, a function of the relative size of the inertias of the engine and the equipment. Operating a transmission in a system with a stiff coupling at higher speeds is not advised unless the resonant speeds are known and can be avoided, or at least crossed over fast at low load. The exact magnitude and the value of the resonant speeds is not easily predictable, and several resonant speeds may be present in the desired operating range.

The most common type of stiff coupling is the ordinary universal joint drive shaft. This type of coupling is often noisy. The noise is usually an indication of the operation in a resonant region. A backlash in the driveline makes the noise, and the stresses in the transmission are further increased with the backlash.

Generally speaking, the use of a torsionally stiff coupling requires that the coupling and the transmission are oversized in comparison to the case when the soft coupling is used. They have to withstand torque values much higher than the values predicted from the power absorbed by the equipment. How much higher depends largely on the relative sizes of the polar moments of inertia of the equipment rotating components with respect to the prime mover rotating components.

Conclusion

For maximum protection of the transmission preference should always be given to the use of a torsionally soft coupling. When selecting a soft coupling, the objective is to find one of the desired physical configuration, torque capacity and allowable speed, which also exhibits the lowest torsional spring rate in its class, and is designed to fail at a load safe for the transmission.

VIEWPOINT

(continued from page 8)

Dear Editor:

The article by Mr. Dale on "Gear Noise and the Sideband Phenomenon" contains some interesting test results, but may have given a slightly deceptive idea of what is currently possible.

The interaction of amplitude and frequency modification with resonances has been recognized for some time, although it is not the only possible way of obtaining highly asymmetric sidebands especially in epicyclic gears.[1] In the case quoted, the effects of pitch errors which only repeat every 559 teeth will give modulation at a vast range of frequencies, all multiples of the basic mesh cycle frequency.

Single flank testing is normally carried out slowly, and the article suggests that it is not possible to carry out transmission error checks at speed; this is, however, done at full speed and full torque on gearboxes. When testing under these conditions, it is easy to use time averaging techniques[2] which are more powerful than simple frequency analysis and have the advantages of separating out pitch errors on the two gears and increasing the accuracy of the results.

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1. P. D. McFadden and J. D. Smith, "An Explanation for the Asymmetry of the Modulation Sidebands About Tooth Meshing Frequency in Epicyclic Gear Vibration." *Proc. Inst. Mech. Eng. 1985, 199* (C1), pp. 65-70.
2. J. D. Smith, *Gears and Their Vibration*. Marcel Dekker, New York and MacMillan, London. 1983.

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