The Effect of Start-Up Load Conditions on Gearbox Performance and Life Failure Analysis, with Supporting Case Study

R. J. Drago

Management Summary

Most gearboxes are rated based on one or more of several criteria, such as peak applied load, nominal operating load, prime mover rated power, driven machine load spectrum, maximum expected overload, etc. These load conditions are used to predict the expected operating life of the gear and bearing system. In many cases—especially where the load and/or speed vary significantly during the normal operational cycle of the gear system—a great deal of attention is paid to the spectrum of load conditions, often resulting in the definition of extensive load/time tables, which are used in a Miner's rule (cumulative damage theory) or Weibull approach to the calculation of the expected life for gear systems under these widely varying, load-speed-time conditions. However, one important factor is very often neglected—the starting load.

If a gear system is run continuously for long periods of time—or if the starting loads are very low and within the normal operating spectrum—the effect of the start-up conditions may often be insignificant in the determination of the life of the gear system. Conversely, if the starting load is significantly higher than any of the normal operating conditions, and the gear system is started and stopped frequently, the start-up load may, depending on its magnitude and frequency, actually be the overriding, limiting design condition. In these cases, failure to account for the start-up load (Ref. 1) conditions in both the basic design of the gear system and in the proper attention to detail of tooth modification, can lead to premature, seemingly unfathomable failures.

This paper addresses the issue of start-up loading and its effect on the performance of a series of gearboxes in an industrial application by virtue of a specific case study. A description of the failures that occurred, and the test program and results that led to a definition of the root cause of the failures and a path of correction, are also presented.

While not addressed in this paper, it should be noted that shut-down load can, in rare cases, also be extremely significant in the design of the gear system due to the very high, short-term loads that can be applied in certain applications. In one case in the author's experience, a gearbox that was used in a high-speed turbo compressor application failed due to very high loads that were caused by an improper shut-down procedure on the driven machine that generated high back pressures and inertial forces sufficient to damage the gears.

Introduction

When gearboxes are used in applications where the connected load has high inertia, the starting torque transmitted by the gearbox can be a great deal higher than the rated load of the prime mover. Power plants often require several evaporative cooling towers or large banks of air-cooled condensers (ACC) to discharge waste heat (Fig. 1). Because of the very large size of the fans used in these applications, they fall into this category of high-inertia starting load devices.

The typical evaporative cooling tower or air-cooled condenser unit is composed of an electric motor that drives a large fan through a gearbox. This assemblage generates air flow for the purpose of removing the waste heat from the power-generation process. In a typical air-cooled condenser application, many identical units (typically 20 to 40) are installed in an array to provide the total cooling required for a large facility. Depending on its type, these geared fan units may operate almost constantly ("base load" plants, e.g.) or very intermittently ("peaking" plants).

The fans in these units are generally very large, and thus have significant inertia (Fig. 3). When started from zero speed, a very high torque is required to accelerate the fan to normal operating speed. If the fan is started infrequently yet run continuously for long periods of time, this high starting torque is of minimal significance. However, when the fan is started and stopped frequently, the number of cycles at the high starting torque can accumulate to a point where they can cause extensive fatigue damage—even if the gear system is adequately rated. Where the gear unit is marginally rated, very early, catastrophic gear failure often results.

As part of the overall investigation of several failures in such gearboxes, we measured starting torque on a typical installation, examined many failed gears and calculated the load capacity ratings for the gearboxes under actual operating conditions. This paper describes the failures observed, the testing conducted, the data analyses and the effect of the high measured starting torque on the life and performance of the gear systems. The test results were surprising, especially during starts where the fan was already windmilling due to natural air flow in the ACC bank. In the final analysis, the importance of appropriate profile modifications is also clearly demonstrated (Fig. 4).

The Initial Incident

A failure was reported in the low-speed gear set of a triple-reduction, single-helical gearbox (Fig. 5). The gearboxes are used in an air-cooled condenser (ACC) facility at a power plant, and had been in service for about four years at the time this investigation was initiated.

The initial failed unit is one of 30 identical units at the same facility, Site A. An identical bank of units, Site B, is also in service for a similar time period at a "sister" power plant in the same state. The first failed Site A unit that we examined had accumulated just over 13,000 hours of loaded service, and during that time had been subjected to more than 7,000 start/stop cycles. All of the gears in these units are carburized, hardened and profile-ground.

Subsequent to this first failure, several additional, very similar failures were also discovered on Site A gearboxes. Additional, ongoing periodic visual inspections of the remaining gearboxes at Site A continue to reveal additional failures. Careful evaluation of the additional low-speed gears showed them to have very similar damage characteristics. It is clear at this point that all of the units at Site A are very likely to eventually suffer similar failures over time.

It was initially reported that, though the systems

continued

Figure 1—Air-cooled condenser bank.

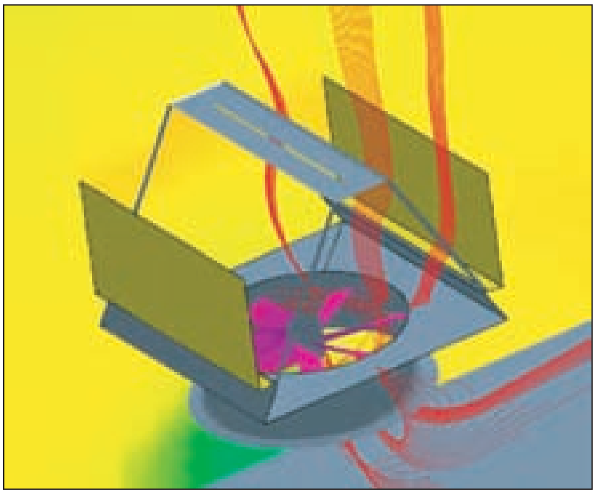

Figure 2—Air-cooled condenser schematic.

Figure 3—Installed air-cooled fan.

Figure 4—Spalled single tooth.

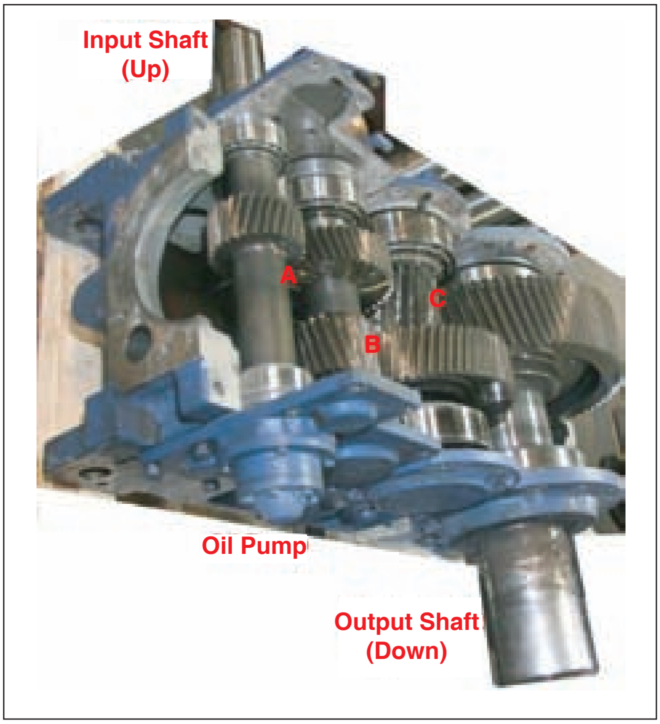

Figure 5—Triple-reduction gearbox (shown in normally installed attitude).

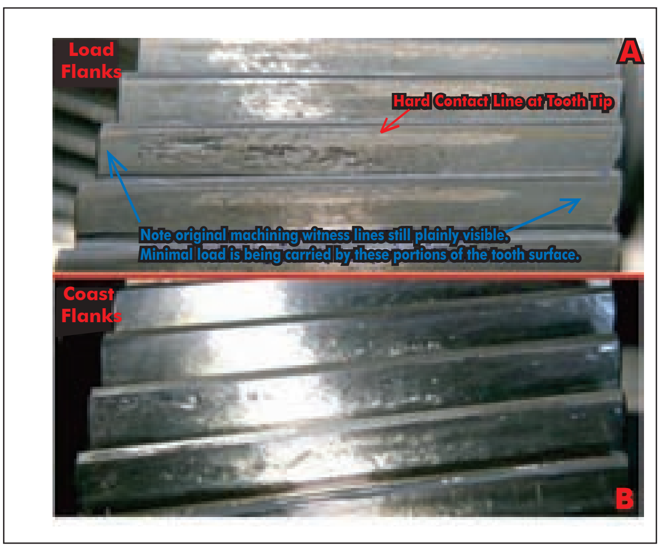

Figure 6—Low-speed gear.

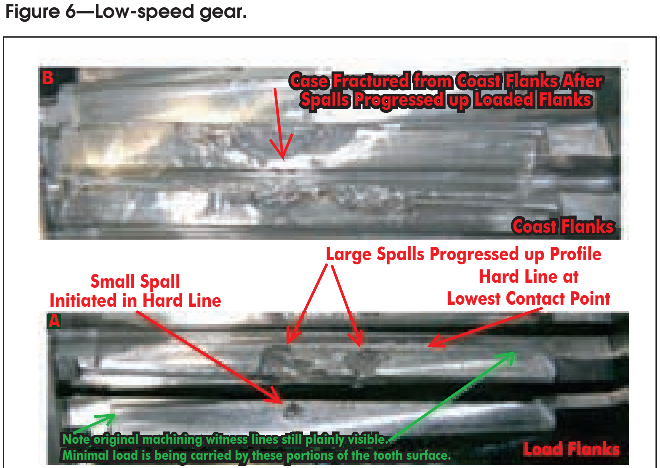

Figure 7—Low-speed pinion.

(gearboxes, fans, motors, controllers, etc.) were identical at the two sites, the gearboxes at Site B had not experienced any failures. As the investigation progressed, however, we did find some indications of very early stage failures of low-speed gears at Site B which were very similar in nature to those observed on Site A. The extent of the damage observed on the Site B low-speed gears was much less than that observed on the low-speed gears on the Site A gearboxes, and to date none of the Site B gearboxes has actually been removed from service due to low-speed gear damage.

Obviously, since the units are identical at both sites, this concentration of failures at Site A and the lack of any apparent failures at Site B was something of a puzzle. Even with the discovery of some very early stage damage on the Site B low-speed gears, it is clear that something is different between the two sites.

After in-depth evaluation of the characteristics of the sites, the only significant difference identified was the fact that Site A was operated as a "peak load" plant, while Site B was operated as a "base load" plant. In the operation of a base load plant, the fans would run almost continuously, with very limited start/stop cycles. In contrast, the operation of a peaking plant involves multiple, frequent start-stop cycles.

While the number of start-stop cycles is different between the two sites, the nominal power draw on the motors at both sites A and B during normal, full-speed, steady-state operation was found to be very much the same.

The Failure

Initial evaluation of the gearbox indicated that the primary damage was on the low-speed gear set. The high-speed and intermediate gear sets did show some damage, but it was largely inconsequential. The low-speed gear (Fig. 6), exhibited a very hard line of contact near the tip of the tooth. This hard, localized region of contact indicates that the tooth did not have sufficient profile modification for the applied loading. The shape of the damaged area suggests that the gear set had some crowning (probably applied to the pinion) to accommodate misalignment. The damage pattern observed on the gear tooth-loaded flanks, however, is slightly heavier on the left end of the face (Fig. 6). The relatively short area of damage indicates that the crown applied to the low-speed gear mesh is likely too large for the applied loading. This tends to concentrate the load in the center portion of the face while unloading its ends. This increases the localized stresses in the heavy-contact region. This is very apparent on the loaded tooth flanks shown in Figure 6.

The location of the damage observed on the low-speed pinion (Fig. 7) corresponds to the damage observed on its mating gear. However, the level of damage on the low-speed pinion is much more severe than that on the mating, low-speed gear. Since the pinion sees many more cycles than the gear, the disparity in the level of damage is expected. The location of the very hard line of contact near the tips of the gear teeth corresponds closely to the similar hard line of contact at the lowest contact point on the pinion teeth. As noted above, this hard contact indicates a lack of adequate profile modification on the pinion or gear or both.

The specific failure mode observed on the pinion is spalling, or surface durability distress. Specifically, spalling is a fatigue mechanism that occurs when very high, local stresses initiate cracks at or near the tooth surface. These cracks progress into the hard case on the tooth surface and progress up along the tooth surface, from the lowest contact point, in the direction of sliding on the tooth surface. As the cracks progress, material on the tooth surface is undermined, and eventually relatively large, somewhat fan-shaped pieces of tooth surface are liberated. This mechanism is clearly shown in Figure 7. As the cracking progresses through the carburized case on the loaded flanks of the teeth, it eventually propagates across the tooth thickness—generally fairly close to the tips of the teeth—and fractures the entire case off the tooth, even extending back onto the coast flanks. Essentially, the carburized case is "peeled" from the loaded and unloaded tooth flanks. (More detail on this failure mode is presented below.)

In addition to the hard line at the lowest contact point, the pinion distress also suggests that the teeth are somewhat over-crowned. This condition is similar to that observed on the mating, low-speed gear, as noted above. Careful examination of both Figures 6 and 7 also shows small areas of the loaded tooth surfaces at both ends of the face width, where the original witness marks from the tooth-finish grinding operation are plainly visible. These areas of relatively light loading further indicate the possible over-crowned condition. In addition, they also indicate that misalignment across this gear mesh is not a major factor in the occurrence of the low-speed gear set failure. While it may be more of a factor on other low-speed sets, it is not the primary causative agent, though it is certainly contributory.

Load Capacity Evaluation

In order to better understand the cause of the failures observed, we calculated the basic load capacity rating of each gear set in the gearbox. The load capacities are best understood by looking at the service factors for each gear mesh, as summarized in Table 1.

As shown by the data in Table 1, although the strength ratings are above the applied power, the durability ratings of the low-speed pinion and gear are less than the applied power. These low power ratings result in durability service factors that are less than unity (1). In an application such as this, we would normally recommend a minimum service factor of at least 2.00. The lower service factor on the low-speed pinion relative to the higher (but still below the recommended minimum) service factor on the mating low-speed gear is consistent with the relatively greater damage experienced by the pinion, as compared to the gear. Based on these ratings, premature durability failures of the low-speed pinion and gear would be expected. The relatively high-strength service factors would allow the units to continue operating for a long period of time after the spalling damage had initiated and progressed, though tooth fracture would ultimately be expected. In this particular application, the power supplied by the motor during normal, steady-state operation is slightly less than the motor nameplate rating; thus the durability service factors for the low-speed gear set are actually slightly higher than shown in Table 1 (and very close to unity).

While the low-durability service factors are certainly of significant concern, of and by themselves they do not fully explain the rather catastrophic failures that occur on the low-speed gear sets at Site A—especially in view of the fact that the durability service factors for the intermediate and high-speed gear sets are only slightly higher (there were no reports of catastrophic failure or even significant surface damage), and there were no catastrophic failures of any of the gears at Site B.

In order to better understand just what is happening, we had to first develop the actual failure scenario explaining the catastrophic damage that occurs on the low-speed gears of the Site A gearboxes.

Failure Scenario

While it may seem obvious that the low durability ratings of the low-speed gear set, particularly the pinion, are fully responsible for the failures observed, they are not the sole cause. Some evidence of a small amount of misalignment is apparent, and this certainly plays a role in the failure as well by generating a load misdistribution across the face that results in high, localized stress levels. Further, it appears that there may be too much crowning on the low-speed set (either by error or design), which also results in load concentration toward the midsection of the face width that increases the unit stress levels in that region. Each of these factors exacerbates the problem of relatively low durability ratings.

Another major factor, however, is at work and is the primary compounding cause, acting in concert with the low durability ratings, of the relatively short-term occurrence of the failures observed.

The specific initiating failure mode observed is spalling (not pitting). The spalling initiated at, or very near, the lowest contact point on the pinion tooth where the loads should be very low and, ideally, virtually zero. This is the point at which the mating gear first makes contact with the pinion. In the case of the subject pinion, the high applied loading on the teeth, relative to their inherent, basic capacity, results in tooth

continued

Table 1—Service Factors

Parameter		Strength	Durability
High Speed	Pinion	2.6	1.2
	Gear	2.9	1.2
Intermediate	Pinion	2.0	1.0
	Gear	2.6	1.1
Low Speed	Pinion	1.5	0.8
	Gear	2.1	0.9

Note:
Ratings based on 175,000 hour required life (24 hours/day, 365 days/year, 20 years) using AGMA Grade 1 materials to motor nameplate power, per AGMA 2001-C95.

deflections, which cause very high loads to exist locally as the tip of the gear makes contact with the pinion. This condition is shown schematically in Figure 8.

These very high local loads, combined with the normal, very high sliding that exists at the lowest contact point on the pinion, cause an abnormal, very high, local stress condition. High starting loads, even though applied for a short duration, greatly aggravate this condition because the tooth deflections (and thus the amount of interference) are increased as compared to normal operation. Though normal operation under the conditions depicted in Figure 8 will eventually result in the formation of a "hard" line near the lowest contact point, frequent high-inertia starts shorten the time required to reach failure. This very high load concentration at the tooth tips and tooth flanks is alleviated by applying appropriate modifications to the tooth profile. Typical profile modifications, which are based on the actual deflected shape of the gear tooth, are shown schematically in Figure 9. The deflection curve shown in Figure 9 is for the low-speed gear set and was calculated at the motor nameplate power. Starting loads, which are significantly higher than the motor nameplate-equivalent torque, will result in proportionately larger tooth deflections.

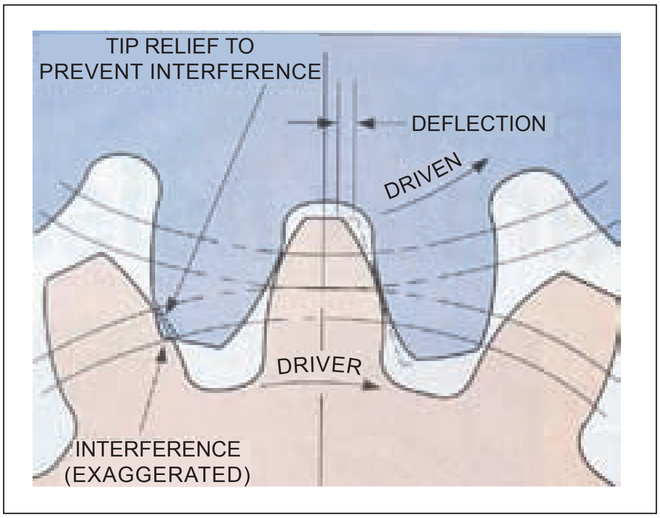

Figure 8—Tip interference due to tooth deflection causes high local loading at lowest contact point.

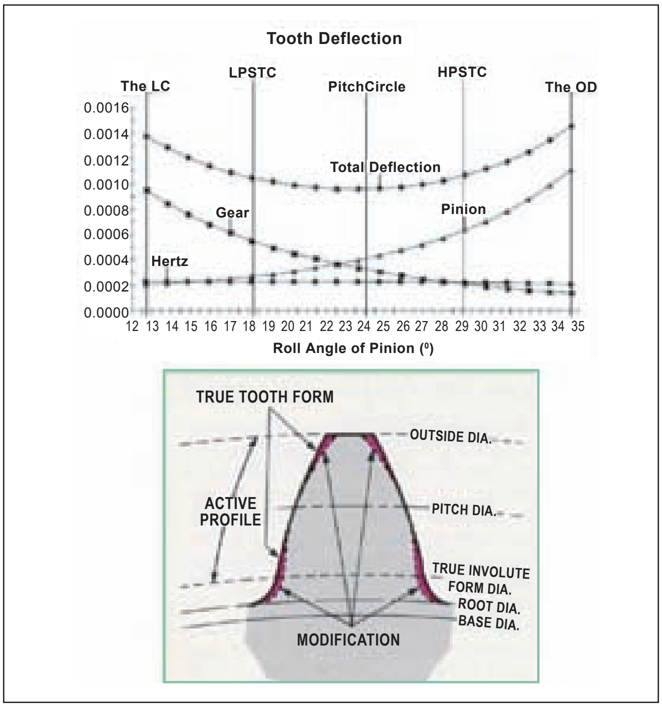

Figure 9—Schematic showing the application of profile modifications to a gear tooth to avoid deflection-induced, high local loading.

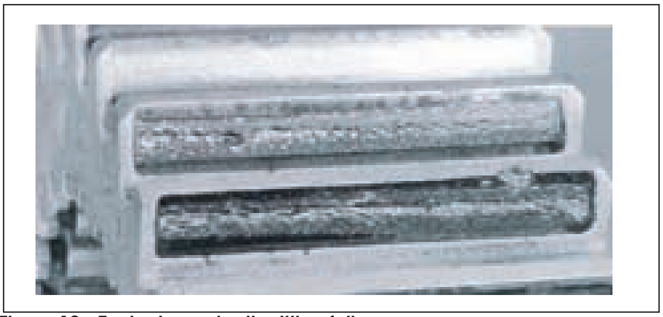

Figure 10—Typical gear tooth pitting failure.

The application of profile modifications does not improve the basic load rating of a gear tooth; however, they do make the tooth much less sensitive to spalling failures, thus allowing the gear set to achieve its inherent life. While the theory is far beyond the scope of our discussion here, spalling failures are not adequately addressed by the durability ratings calculated through the use of the AGMA rating standard. The durability ratings calculated by the standard refer to the pitting failure mode, an example of which is shown in Figure 10, for reference—not the spalling failure mode.

Careful evaluation of the failed pinion (Fig. 7) shows that it did not suffer a pitting failure at all, but rather a spalling failure induced by very high loads at the lowest contact point on the pinion. Note that the damage on the tooth shown in Figure 10 occurs mainly away from the lowest point of contact, and generally in the midsection of the tooth height. This is the typical appearance of classical gear tooth pitting. Conversely, the spalling damage shown on the pinion in Figure 7 starts at the very bottom of the contact zone—in the region of the hard line at the bottom of the tooth—and progresses upward along the tooth profile. This difference is especially significant regarding the remedial actions that are practical for this gear system, as will be explained further below.

As previously noted, spalling is due, among other causes, to inadequate or improper involute profile. The specific nature of the progression of a spalling failure from initiation to complete failure is shown in Figure 11. It is important to note that the very first sign of spalling failure is a relatively innocuous hard line in the region of highest loading, deep in the pinion tooth root. This initial damage is very difficult to see (Figure 11 is shown at high-magnification; thus the damage is obvious). It is possible for the initial stages of spalling to go undetected by a casual observer. The early stages may only be apparent to an experienced, skilled gear engineer. This is an important consideration for Site B (peaking plant), as those gears are subjected to far fewer start-stop cycles, and thus may be in a much earlier stage of progression.

As is the case with the low-speed pinion shown in Figure 7, the initiation of the spalling failure shown in Figure 11 resulted from inadequate profile modification that led to very high loads at the lowest contact point. While spalling is the primary failure mode, its occurrence at a relatively early point in the expected life of these gears was aided by what appears to be excessive crowning, some misalignment across the face width and the inherently low durability rating of the low-speed gear set.

The low-speed pinion also exhibits considerable damage on the coast flanks of the gear teeth (see the top portion of Figure 7). This damage, too, is characteristic of spalling that has progressed to catastrophic failure. To better understand this mechanism, Figure 12 shows a photomicrograph of the cross section of a tooth that has experienced a spalling failure. The cracks—"a"—initiate at or near the surface (Figure 12A) and propagate at an acute angle into the tooth and up along the profile in the direction of sliding. After cracks "a" have propagated sufficiently, the tooth surface is undermined locally, and a relatively large piece of the tooth surface breaks away as the crack "b" progresses to the tooth surface (Fig. 12B). This results in the liberation of relatively large, generally flat chips of tooth material (pitting debris, in contrast, is relatively fine, roundish, small particles) and the characteristic fan-shaped appearance of the damage on the tooth surface (Fig. 12C). As the cracking "a" and "b" progresses, the damaged region on the tooth surface grows ever larger, and the remaining, undamaged, surrounding tooth surface is subjected to ever-increasing unit loads (as the damaged surface spalls away, the remaining surface must carry the full load; thus the unit loading applied to the remaining sound surface increases).

After sufficient surface is destroyed—if the inherent load capacity of the tooth, including especially its bending load capacity, is relatively low (as is the case for the low-speed pinion)—the cracking below the surface will turn in the direction "c," and large portions of the coast tooth flanks will be cracked away relatively quickly. This is the final stage of spalling crack progression, after which the teeth will be

continued

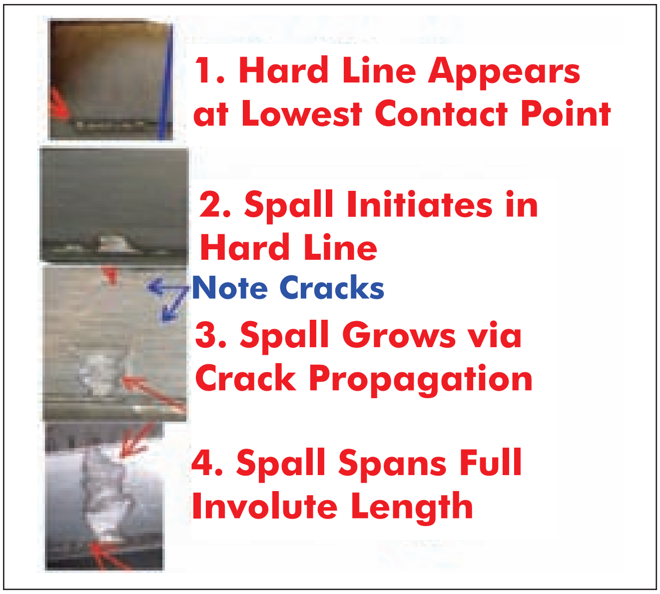

Figure 11—Typical spalling failure progression.

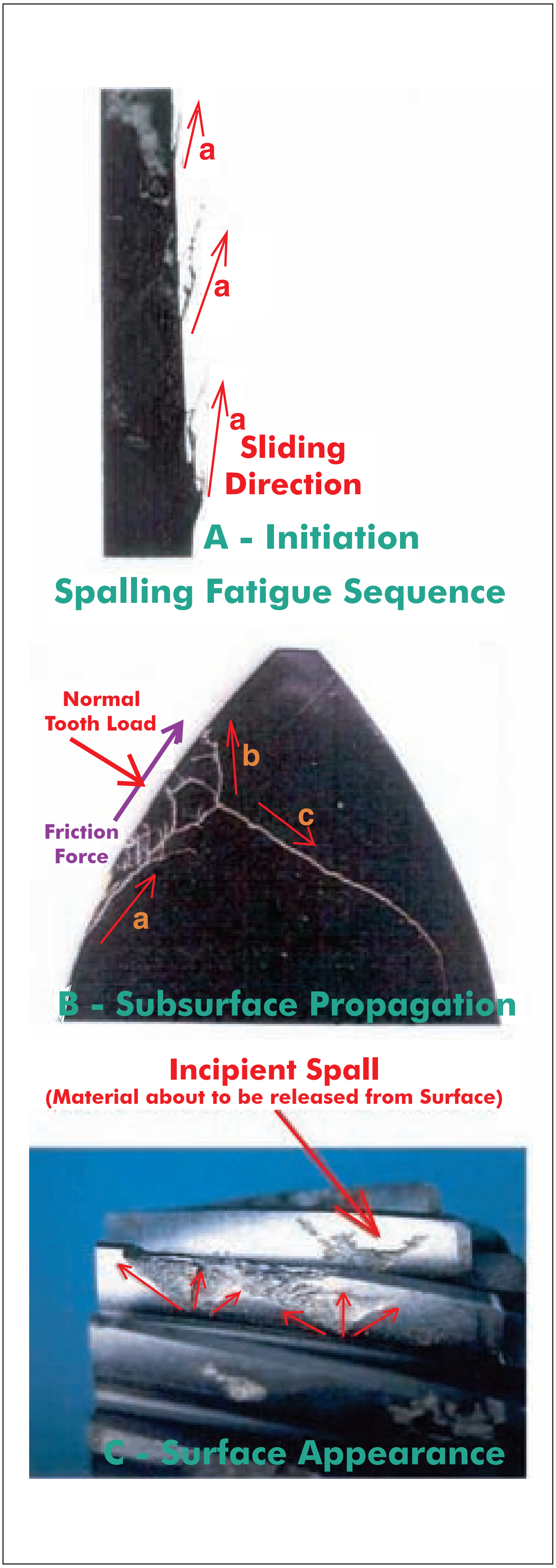

Figure 12—Spalling progression sequence.

completely stripped from the gear. The spalling on the low-speed pinion progressed completely up the loaded flanks (Fig. 7A), wrapped over the tooth tip and down the coast flanks (Fig. 7B). As a result, the low-speed pinion teeth were close to complete tooth fracture and, ultimately, total loss of torque transmission capability.

In order to be sure that our observations on this low-speed gear set were typical of the failures that have occurred on these gearboxes, rather than an isolated incident, we examined another failed low-speed gear set that was selected (by others, randomly) from the "stock" of failed gear sets. A section was removed from approximately the center of the face width of this low-speed pinion (Fig. 13), for laboratory examination.

Careful, close-up examination of a tooth (A) in Figure 15 shows the same mechanism that is presented in Figure 14.

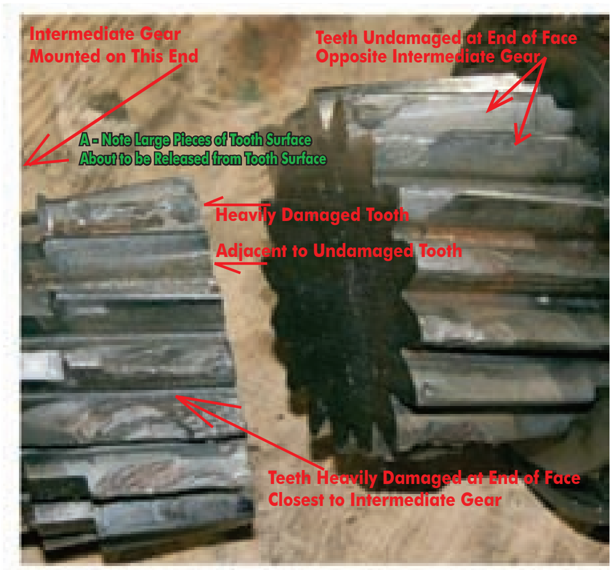

Figure 13—Second low-speed pinion. (Photo taken after section removed for laboratory analysis.)

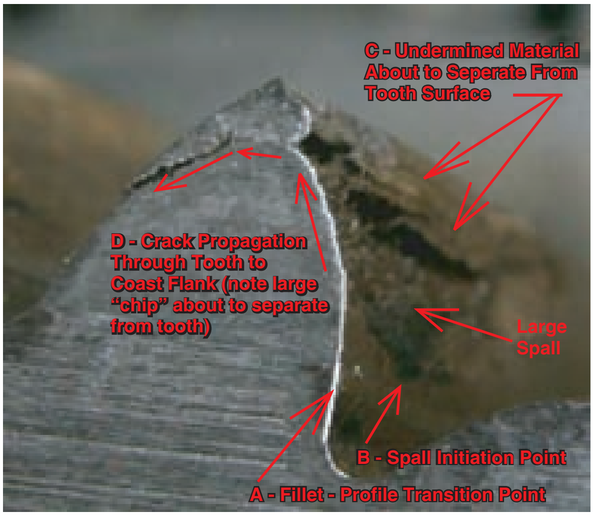

Figure 14—Spall initiation and progression on the pinion tooth (A), enlarged from Figure 11. (Note: Loaded tooth flanks are on the right in this view.)

Because the failure scenario clearly points toward inadequate profile modifications in the presence of high tooth deflections, we had a local vendor conduct a complete inspection (i.e., lead, profile, pitch and pitch diameter runout) of a new, OEM, low-speed gear set before it was installed as a replacement for the damaged, low-speed gear set described above. Of these inspections, the involute chart of the low-speed gear, Figure 15, is of most interest.

This chart shows the gear involute profile to be virtually straight, with no profile modification (the sharp drop-off at the tooth tip is more chamfer than actual tip relief).

The involute chart of the pinion (Fig. 16) shows very similar characteristics. While the pinion does have a very slight amount of tip relief, the flank is completely straight.

Clearly, the same mechanism is at work on the pinion tooth (A) shown in Figures 13 and 14, as that shown in a composite DST image shown in Figure 12. Figure 14 also shows the location of the spall initiation (B) on this tooth in relation to the profile-to-fillet.

The very straight involute chart on the low-speed gear at its tip, combined with the very straight involute chart of the pinion at its flank would, under sufficient load, generate the hard line that was observed at the root of the pinion (Fig. 7). At this critical contact point, there would be about 0.0014 inch of interference due to the total tooth deflection (Fig. 9). Under starting loads, which would be significantly higher than normal operating loads, the amount of interference would be substantially greater.

As noted earlier, the low-speed pinion also appeared to be over-crowned. The lead chart shown in Figure 17 shows this condition as well. Crowning is a very valuable feature, and should be included in any gearbox in an application where some supporting structural deflection can reasonably be expected. This is certainly true of the framework in which these gearboxes are mounted. In addition, some misalignment can also be expected from internal sources, such as bearing clearances and shaft deflections. Still, the amount of crowning and the offset of the crown must all be carefully designed for the specific application. In this case, the crown applied was slightly excessive for the conditions encountered. While this is contributory to the failures observed, it was not a major driving factor.

While all of the pieces of the puzzle fall fairly well into place and point to the combination of a low basic durability rating and lack of adequate profile modification—as the root cause of the failures observed—the significant difference in damage level observed on the low-speed gears at Site A and Site B is still of concern.

As noted above, the most important difference between the two sites involves the number of start-stop cycles experienced by the gearboxes at each site. In order to investigate the loading question further, a program was developed to measure the torque applied to the gearboxes during the start-up cycle and normal loading.

Measured Torque

Site conditions during the testing were what could be called "normal." Wind velocity was not excessive (about 5 knots) and was out of the normal direction. Ambient air temperature was about 68° F. As will be discussed below, the worst-case scenario would be a no-wind condition since this would likely result in a start-up from zero speed. The fans are fitted with an anti-rotation device that prevents them from turning backward, so the worst condition would be a stopped fan condition.

Due to time constraints, only four test runs were made; however, these runs yielded very valuable data.

Test 1 was initiated after the drive had been shut down for several hours. Despite this several-hour down period, when the motor was engaged at the start of test 1, the fan was still turning slowly (windmilling) due to normal cross airflow in the system. Speed at motor engagement was 20.06 RPM for test 1.

Test 2 was conducted quickly after test 1. The power was removed from the drive for several minutes and the fan was allowed to coast down in speed, but the fan did NOT come to zero speed. At the start of test 2, the fan was still turning at a speed of 69.25 RPM when the motor was engaged.

Test 3 was conducted quickly after test 2. After power was removed from the drive for several minutes, the fan was allowed to coast down in speed, but the fan did NOT come to zero speed. At the start of test 3, the fan was still turning at a speed of 24.93 RPM when the motor was engaged.

Test 4 was conducted quickly after test 3 after power was removed from the drive for several minutes and the fan was allowed to coast down in speed, but NOT to zero. At the start of test 4, the fan was still turning at a speed of 26.23 RPM when the motor was engaged.

We examined the data in detail and conducted further analysis so that the results could be presented in graphical form. The results are very interesting and provide a great deal of insight into the loads that are applied to the gearbox during the start-up portion of the cycle.

Figure 18 shows the instantaneous motor torque applied to the gearbox during the motor engagement, while Figure 19 shows the instantaneous motor power delivered to the gearbox during the start cycles measured.

Although the motor comes up to speed relatively quickly, this data shows that during this short period of time, very high torque levels are applied to the gear system. Since the system was not started from zero speed during this testing, we still do not know what the actual "start from zero speed" load would be, but we can make a projection using the data available to us. In order to do so, we plotted starting torque as a function of system speed at engagement (Fig. 20).

We also plotted three straight lines on this figure to represent the most pessimistic (blue), the most optimistic (green), and "nominal" (red) projections to a zero engagement speed starting torque level. (While we have plotted a straight line for simplicity, the actual starting torque will be somewhat exponential, but we do not have enough data to make a better projection at this time. In any case, the data does provide good

continued

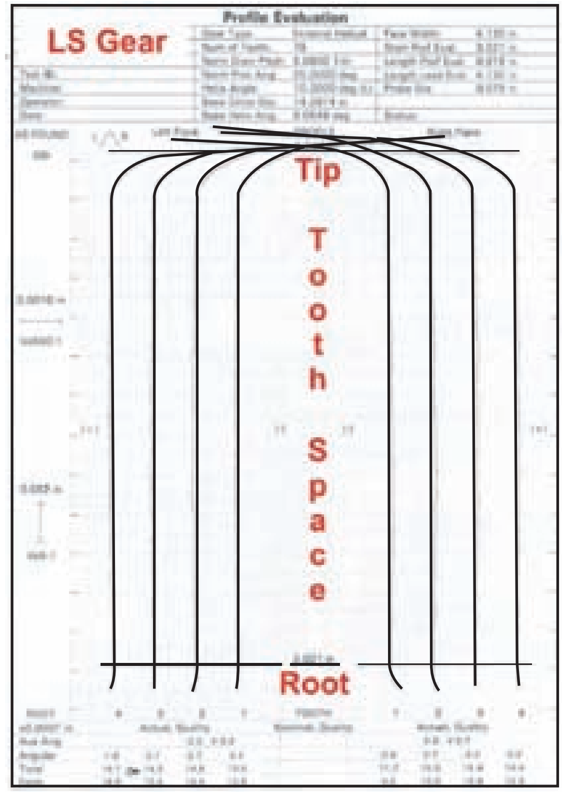

Figure 15—Low-speed gear involute chart.

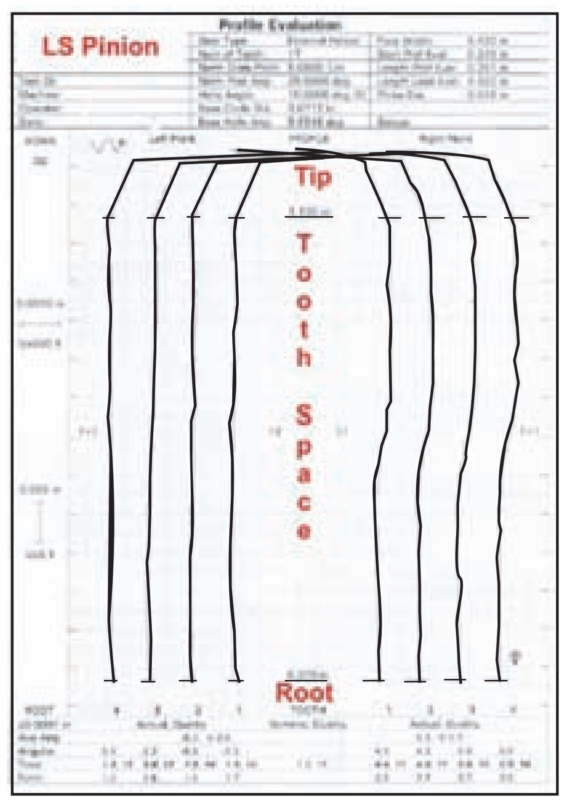

Figure 16—Low-speed pinion involute chart.

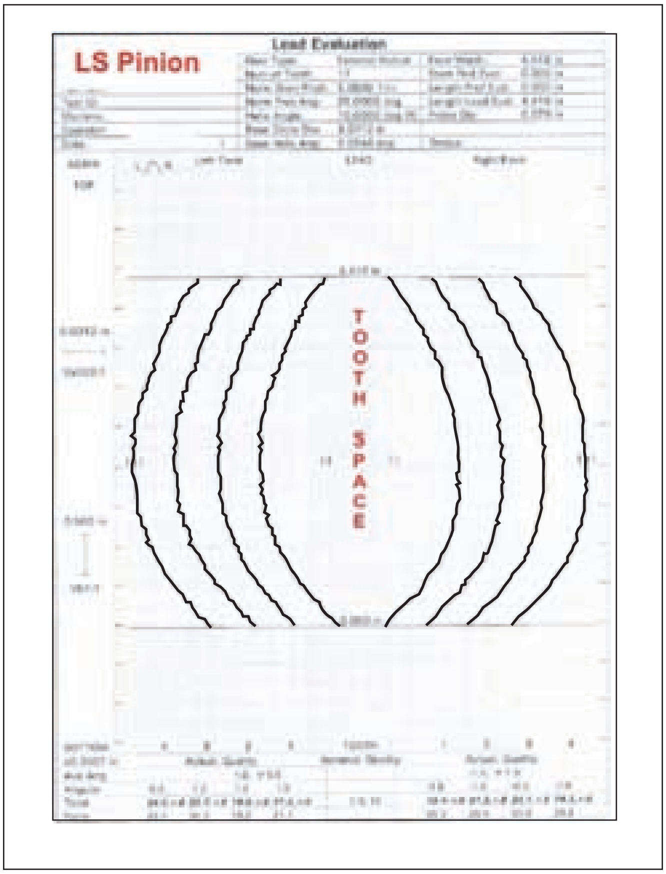

Figure 17—Low-speed pinion lead chart.

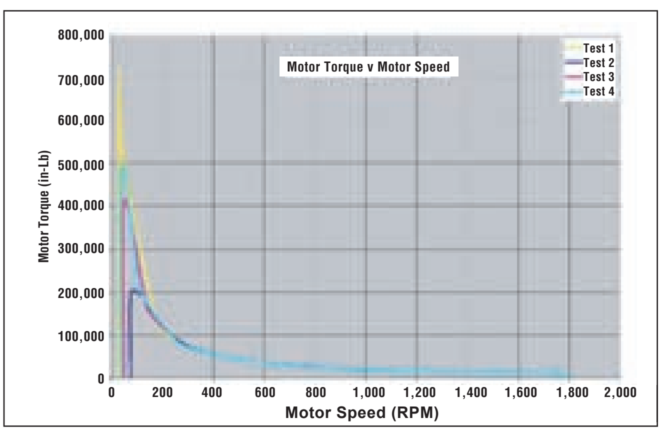

Figure 18—Instantaneous motor torque applied to the gearbox during start-up.

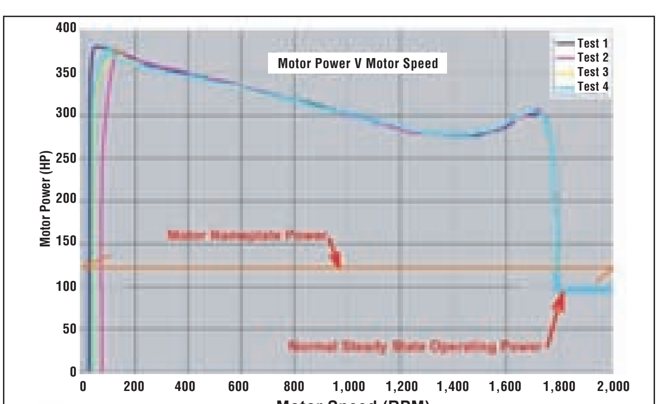

Figure 19—Instantaneous motor power applied to the gearbox during start-up.

insight into the basic operation of the system.) It is possible, but very unlikely, that the zero speed engagement torque might be represented by the pessimistic blue line in Figure 20. We believe that the real value would be somewhere between the most optimistic (green) and the nominal (red) lines. In any case, however, the starting torque is very much higher than the nominal, full-speed running torque provided by the motor. While it is not possible to predict the exact zero engagement speed from the small amount of data available, the data does show that, by any estimate, the start-up loads are very much higher than the operating loads.

It is also both interesting and important to note that, based on the data obtained from this small test program, it may not be the zero speed starting loads that are of most importance. The peripheral observation that the conditions at the site during the testing were relatively benign (no significant wind conditions, 60° F), and that the motor was found to be free-wheeling at about 20 RPM, even after having been shut down for several hours (certainly enough time to coast to a full stop if no other factors were at work), the start-up loads of most interest may be those that occur when the motor is coasting at or above 20 RPM. Even a small amount of freewheeling speed at motor engagement can dramatically reduce the starting torque, as Figure 18 clearly shows. This is a very important observation since it is clear that a consistent starting torque of 1,000,000 in-lbs. (the "pessimistic" projection at zero speed) would most likely have destroyed the gears much sooner than the four years reported here. As the wind velocity increases, I would expect that an unrestrained system would windmill at speeds somewhat higher than the 20 RPM. From the data in Figure 18, even a free wheel speed of 70 RPM (only 4% of full motor speed) drops the zero speed engagement starting torque to 200,000 in-lbs. While this value is still very high relative to the 4,347 in-lbs. normal torque, it correlates better with the observed, approximate four-year "life" of the current gears—especially in view of the very fast, observed ramp time to full speed.

It is also very important to note that the number of cycles that the gears experience during start-up is also relatively low. The data provided is not fine enough to calculate with certainty the number of cycles the gears would experience during typical starts, but based on the information available, the number of motor shaft rotations is probably in the range of 1,000 to 3,000 revolutions between a 'typical" free-wheeling start and full speed. Since the gearbox total ratio is about 22:1, the low-speed pinion would experience somewhere between 45 and 136 revolutions during each start cycle. In addition, the applied torque during start-up decreases as the motor comes up to speed (Fig. 18); thus the maximum start-up load is only applied to the gears for a small portion of the estimated 45–136 revolutions (about 10% to 15%, based on the data available). Over the course of thousands of starts, of course, these cycles would certainly add up; but the total would still be relatively low. The number of recorded starts on the gearbox from which the low-speed gear set detailed in this paper was removed was 7,102 at the time the damage was discovered. Using the estimates developed above, the low-speed pinion teeth in this unit would have been subjected to between 32,000 and 150,000 peak-load, start-up load cycles. (These are rough estimates, but I believe that they are reasonable representations of true numbers.)

These very high starting torque levels cause the involute interference shown in Figure 8, which leads to the spall initiation and progression observed on the low-speed gear set. If these high starting loads are applied very infrequently, they will not significantly reduce the expected life of the gears. If, however, they are applied very frequently—as appears to be the case for the subject drive—they can lead to extremely premature failure of the gear system, as we seem to be observing here.

Additionally, the much lower number of start cycles which the gearboxes at Site B experience explains their significantly longer "life" before damage was observed. And yet, after about six years of operation, the Site B gearboxes did begin to exhibit the early signs of spalling failure that are very similar to those experienced by the gearboxes at Site A. While prediction of "life" based on observed gear condition is difficult at the very best, we estimate that the practical life expectancy of the gearboxes at Site B (base load, infrequent start-stop cycles) is about double that of the gearboxes at Site A (peaking, very frequent start-stop cycles).

Conclusions

This investigation demonstrates the importance of proper involute profile modifications and the extreme influence starting loads can have under certain operating conditions. This is particularly true where the driven load has high inertia characteristics and the number of start cycles is more than trivial.

While the gearboxes were of and by themselves of adequate overall design, the application of these gearboxes in this high starting load environment required special tooth modifications in the form of involute profile modifications of sufficient magnitude to avoid interference at the very high starting load condition.

The gearboxes used in this application are standard "catalogue" type units that have been very successfully used in a variety of applications. In this regard, perhaps the overriding conclusion that can be drawn from this investigation is the need to fully understand the loading and overall operating conditions of any application before selecting a gearbox for use. Often, these conditions require the use of either a custom-designed, single-purpose gearbox or the modification of an available "standard" design to tailor it for use in a particular environment.

The Plan

Work is currently underway to design and manufacture new, set-wise-interchangeable, advanced technology gear sets that will have optimized tooth geometry, improved material characteristics and quality. And, most especially, fully tailored, modified (barrel shaped, tip and flank relief) involute profiles. These gears sets will be used to rebuild all of the gearboxes at Site A and Site B in a phased program over the next several years. While this replacement program will not fully "cure" the problems that have been experienced, it will extend the time to failure for all gearboxes significantly.

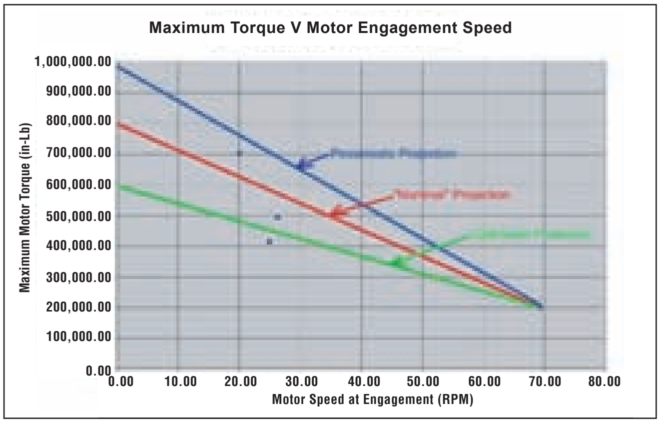

Figure 20—Motor torque vs. motor speed at engagement.

Closing Comment

Obviously, the data provided here is very limited. It would have been very desirable to obtain more extensive data related to other gearbox configurations, operating conditions and environmental variables (wind speed, direction, temperature, effect of operation of units in adjacent cells, etc.). This was, however, a failure analysis effort rather than a research project; thus, the main goal was to understand the root cause of the failures and to develop a reasonable, cost-effective solution. This was accomplished, and the implementation is ongoing. Our purpose in presenting this information in this forum is not the definition of a fully researched study of starting load effects (as welcome as such a study would be), but rather to present useful design and application information in the spirit of the Chinese proverb, "It is better to light a single candle than to curse the darkness." The author hopes that the greater design community reads this material with this limitation in mind.

I can only imagine how much brighter the world of gearing would be if we lit more single candles

In his role as chief engineer of Drive Systems Technology, Inc., **Ray Drago** *is active in all areas of mechanical power transmission. These activities include the design and analysis of drive systems for such diverse areas as large, high-speed paper, printing and cardboard machinery; commercial marine drives; heart pumps; large oil field valves; high-speed cable climbing devices; high-speed gas turbine/generator sets; special automotive racing gearboxes; artificial limbs; mine shaft hoists; air- and water-cooled condensers; miniature gear motors (120 in-oz torque range); automatic bolt torquing devices; very large mining and mill gears; municipal and industrial water and waste water processing system drives and small private helicopter conversions (piston to turbine engines).*