

SPECIAL PURPOSE GEARING—A USER'S APPROACH TO SOUND APPLICATION ENGINEERING

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ABSTRACT

A recent study of Exxon Corporation's world-wide machinery startups indicated that special purpose gear problems have not been critical in terms of project delays. However, the incidence rate of gear problems has been relatively high, with the majority attributable to errors in vendor design or quality control. In an effort to reduce these incidents and resulting startup delays, Exxon Research and Engineering Company has implemented a program of intensive technical follow-up on all new gear units which has resulted in a substantial reduction of gear startup problems, and has practically eliminated special purpose gears as a source of project startup delays. This paper presents a summary of Exxon's gear startup problems over the last 10 years, and outlines specific engineering areas which are subject to Exxon's engineering reviews during the design and manufacturing phases.

The reliability of process machinery has been a continuing major concern to users in the petroleum and petrochemical industries. The rapidly escalating costs of new plants and equipment, emerging shortages of petroleum products, and increasing sizes of individual process units are all exerting strong pressures for prompt startup of new plants and long, trouble-free operation. Although long-term reliability and life of process machinery is generally adequate, many new projects experience significant machinery commissioning problems which delay plant startups by weeks or even months. For large projects, the losses in production and profits may be severe, and can, in some cases, seriously affect overall project economics.

In an attempt to pinpoint the causes and take corrective actions to minimize such losses, Exxon Research and Engineering Company (ERE) made a detailed survey of machinery startup problems and resulting delays on new projects. This study covered a period of approximately 10 years and included a variety of petroleum and petrochemical projects built worldwide for affiliates of Exxon Corporation. Although gears were far from the worst offenders in terms of startup problems, they did contribute as much as 3 days average startup delay per unit. The study also indicated that the majority of the gear problems encountered on new project startups resulted from errors in design or quality control.

Based on these findings, Exxon initiated a program of intensified engineering followup of major new process machinery including gears. Startup experience over the last three years has shown that this engineering effort has been successful, and that startup delays due to gear problems have dropped to about 20 percent of their former levels, which is considered acceptable.

This paper presents a summary of startup problems experienced by Exxon affiliates with special purpose, high speed gearing and a description of those engineering areas where Exxon has intensified its engineering followup to overcome these difficulties.

Gear Failure Summary

This discussion will be confined to gears defined by the American Petroleum Institute Standard 613 as "High Speed, Special Purpose" type. These are precision gears of parallel shaft design with pinion speeds above 3600 RPM, pitch line velocities over 4000 feet per minute, and journal velocities above 1500 FPM. In the process industry, this category of gearing is commonly used to drive centrifugal, axial, and rotary compressors, and also large reciprocating compressors and turbo-generators, in refineries and petrochemical plants. Since this equipment

is usually applied in a single train arrangement and is of critical importance to overall plant operation, very stringent demands are made on this equipment in terms of continuous run lengths and availability.

The petroleum and chemical industries also purchase a large number of small, often slower speed gears for such applications as centrifugal pumps, fans, small reciprocating compressors and blowers. However, this type of equipment is generally spared or non-critical, and downtime of individual units due to mechanical failures has little impact on overall plant reliability. The startup problem study was, therefore, limited to gearing of the high speed, special purpose type; and this discussion is generally limited to this category.

On behalf of its affiliated companies, Exxon Research and Engineering Company is involved in the purchase of approximately 15 to 20 special purpose gear units annually. This equipment is purchased worldwide, and the population covered by this discussion includes equipment manufactured in the U.S., Europe, and, to a lesser degree, also Japan.

The majority of special purpose gearing owned by Exxon affiliates is used as speed increasers or reducers in centrifugal compressor trains and gas or steam-turbogenerators. A special application is the double reduction gearing used on large, turbine-driven reciprocating compressor drives. Typically, single reduction gears range from 1000 up to about 20,000 HP, with pinion speeds from 5-12,000 RPM. Double reduction gears include sizes between 1000 and 9000 HP, and generally operate at output speeds for reciprocating compressors, i.e., about 300 RPM.

Field startup experience with these machinery trains has indicated a surprisingly high number of startup problems due to gear bearing failures, poor oil drainage, weld failures and generally quality control defects, which are summarized in the following discussion.

Inadequate Thrust Bearing Design

For many years, high speed, double helical gears were built with little more than axial locating bearings, the general assumption being that there were no internal thrust forces generated in the gear, and in sleeve-bearing motors the rotor centering forces were generally of low magnitude. Even in single helical gearing, the thrust bearings were generally designed only for the relatively light loads generated in the gear mesh, and bearings were usually of the light duty, flat or tapered land design.

The advent of large-housepower geared trains, however, brought the problem of coupling torque lockup forces to the surface, which is now the most important single factor governing selection and sizing of gear thrust bearings. Just to briefly explain this phenomenon, torque lockup forces are generated in the tooth mesh of gear type couplings (which are standard equipment on high speed compressor trains) due to thermal expansion and contraction of the coupled shafts which may be caused by warmup of the equipment during startup as well as changes in operating conditions. In new, properly lubricated couplings, the shafts are free to expand with only small frictional forces in the coupling. However, as the coupling teeth wear and pit, or oil sludge and dirt are accumulated in the coupling, the friction forces, which

resist the sliding movement of expanding shafts, may increase substantially. This, in turn, may cause equally high, opposite forces on the gear thrust bearing whenever thermal shaft expansion or contraction occurs.

Exxon's first encounter with this problem was on a 10,000 HP speed increasing gear in a large air compression train between two compressor casing operating at approximately 5000 and 10,000 RPM. This gear was equipped with a flat land thrust bearing on the low speed shaft designed for thrust loads based on an assumed coupling friction factor of 0.15. This thrust bearing failed very shortly after startup of the compressor train. The problems was corrected by replacing the flat land bearing with a tapered land design, which in effect doubled the thrust capacity of the bearing, and no further failures occurred.

At approximately the same time, a similar problem occurred on a large double-ended compressor drive turbine. This machine was rated at 30,000 HP and installed in an ammonia synthesis compressor train, driving a single compressor casing on one shaft end and two casings on the opposite end. The high torques experienced on this train, changing thrust directions on each casing, and the maximum size limitation of the existing thrust bearing made this particular problem extremely difficult to analyze. Extensive field testing was, therefore, required before a solution could be developed. This failure was unrelated to gears, except for the following significant findings from field tests about coupling lockup phenomena in general:

- (1) In compressor drives, torque lockup forces are unpredictable in the timing of their occurrence, as well as direction. These forces will occur not only at startup during equipment warmup, but also during normal operation, sometimes seemingly without noticeable change in equipment operating conditions.
- (2) Compressor and steam turbine rotors often alternate their thrust load between active and inactive sides of the thrust bearing. The direction of thrust forces transmitted to the gear through coupling lockup can, therefore, not be predicted—and cannot be assumed always to be in the direction of shaft thermal expansion.
- (3) Thrust measurements by thrust bearing load cells indicated coupling friction factors as high as 0.3, which is about twice the value formerly considered conservative.

Based on the above results, a detailed review was made of high speed gears which were at that time in startup or on order with manufacturers, and it was found that four gearboxes required substantial uprating of the thrust bearings by either increasing bearing size or by installation of double-acting, tilting shoe thrust bearings. As an example of the order of magnitude of this uprate, one 10,000 HP gear unit had a flat land bearing with a 500 lb. maximum load, whereas the maximum load based on field data was close to 5000 lbs. These gear units, as well as other units purchased later, have been equipped with tilting shoe thrust bearings rated for a 0.3 friction factor, and no further failures of gear thrust bearings have occurred.

Poor Oil Drainage and Foaming

Manufacturers usually expend considerable engineering effort in design and component selection of lube oil supply systems. Lube oil drainage, on the other hand, often receives less attention, sometimes with serious consequences. For example, on some units the gear casing design allowed insufficient space between rotating elements and the drain sump, which prohibited proper oil drainage and caused massive foaming. In several cases, including gears as large as 20,000 HP and pitch line velocities as high as 28,000 FPM, sump depths measured only 6-9 inches. Undersized or insufficiently sloped drain lines sometimes aggravated the situation.

As simple as this problem may appear, it is difficult to make corrections in the field except by major modifications or complete redesign of the casing. For this reason, field correction of this problem is rarely completely satisfactory and usually relies on a combination of marginal remedies such as special oil additives, installation of baffles, special venting arrangements, and "fine-tuning" of lube oil flows, pressures and temperatures. In other words, it is an easy problem to detect and avoid during the design phase, but field solutions can be costly in terms of expenses and startup delay.

Faulty Journal Bearing Design

In addition to the normal bearing design aspects of load, stability, and lubrication, gear bearings pose a few special problems related to the effects of gear mesh orientation and forces on bearing operation. Generally speaking, the gear mesh direction (up or down) and bearing sizes are selected to provide stable operation under all anticipated operating conditions. However, certain operating modes are sometimes overlooked, which has led to failures in the field.

A typical example of this type of oversight is a number of double reduction gears which are used on turbine-driven reciprocating compressors. These gears utilize a nested arrangement and quill shafts in order to handle the large torque requirements and to provide predictable torsional characteristics. These units were designed with down-mesh on the pinion shaft to achieve stable operation of both pinion and idler gear at normal load. It was overlooked, however, that the steam turbine drivers could not be accelerated as quickly as motor drivers and required extensive warmup at low speeds. The low speed and load increased the pinion bearing attitude angle sufficiently to place the pinion at the bearing horizontal split line directly over the oil groove. At this point, the bearing area was insufficient to support the pinion shaft, and bearing damage resulted. The problem was solved by rotating the bearing shells 45°, which placed the pinion shaft on a full-width bearing area for both the loaded and warmup condition.

In addition to such occasional oversight, gears often utilize highly loaded bearings in order to avoid instability at low-load conditions. A number of Exxon units operate with plain sleeve bearings loaded to 500 psi, and bearing oil outlet temperatures over 210° F. Although no failures have been directly attributable to these high bearing loads, these bearings are very sensitive to oil supply conditions and contamination with small dirt particles, which reduces their overall reliability. Tilting pad bearings with moderate loading may be an alternative solution in cases

where bearing stability under varying loads is in question.

Welded Web Failures

Gears with high rotational speeds and pitch line velocities are usually machined from solid forgings, either integral or shrunk-and-keyed on the shafts. In general, this type of construction has been completely satisfactory. For lower speed gears, however, costs can be reduced by using fabricated (welded) gear blanks, which have suffered from occasional weld failures. Due to the nature of this application, welding details require a considerable amount of attention. Also, actual stress levels are often difficult to predict due to torque fluctuations and effects of vibration. Errors in predicting these factors often result in a crack which may start in a weld and gradually progresses through the web, ultimately causing catastrophic failure of the gear rotor. A number of such failures have occurred in Exxon units from weld cracks either at the hub or outer rim of fabricated gears.

Quality Control Defects

The last, and probably largest, category of gear defects includes those resulting from improper machining or assembly. Most defects of this type are caught by the manufacturer's quality control or during the mechanical run tests. However, a number of startup problems have occurred due to errors in areas which are not sufficiently covered by normal quality control, or which do not become apparent until the gear is operated at full load at the job site.

Machining errors usually manifest themselves in the form of errors in tooth geometry, which may cause either mechanical or acoustic vibration. Also, such errors may result in poor tooth contact between matching gears, and localized gear overload, pitting and scoring. Following are a few examples of this type of problem:

During the routine inspection of a motor-driven 22,000 HP centrifugal compressor, severe fretting corrosion was observed under the compressor's thrust collar, and under the coupling hub. As might be expected, the reaction to this problem was to blame the light interference fit on both components. Upon further investigation, it was found, however, that the fits on this compressor were identical to those used on a number of similar machines, without any apparent difficulties. This information led to an intensive vibration analysis of the entire train, which ultimately disclosed that a high axial vibration of the gear pinion shaft was being transmitted through the gear-type coupling to the compressor shaft. A complete dimensional check of the gear rotors disclosed high axial runout of the apex of this double-helical unit. Exact measurements of this type of runout are extremely difficult to perform in the field, nor can it be corrected by anything except replacement of the rotating elements. For this reason, the problem was ultimately solved by use of heavier shrink fits on the coupling hubs as well as under the compressor thrust collar.

A second example developed on a single helical speed increasing gear which was purchased for installation between two compressor casings. Per standard API practice, the gear was factory-tested at no load. Subsequent internal inspection indicated that the gear had failed to achieve the 85 percent tooth contact required by the specification. This was attributed to the fact that the tooth profile of

these hardened and ground gears had been compensated for tooth deflection under load, and that full tooth contact would not be achieved on a no-load test. On this basis, the gear was shipped to the compressor manufacturer's shop for a string test of the entire unit. However, this loaded test not only failed to improve the tooth contact per se, but also showed that the actual contact area on the gears oscillated between the two ends of pinion and gear, indicating serious machining errors. The gear had to be removed from the train, returned to the manufacturer and lapped to obtain satisfactory contact at considerable expense and delay in delivery schedule.

In addition to this particular case, Exxon has had a number of gears where insufficient tooth contact remained undetected during the unloaded mechanical run test at the factory and caused initial scoring or pitting of gear teeth in the loaded areas. However, once this initial wear of the tooth surfaces had equalized the contact across the tooth width, the pitting and scoring ceased.

The opportunities for assembly defects in gear units appear rather limited, but they still do occur. For example, bearing liners have been installed backwards, lube oil passages plugged, and bearing lube oil lines not connected. Surprisingly enough, many of these units have passed shop tests and subsequent internal inspection without exhibiting any signs of distress.

Gear Problem Categories

The above problem summary covers only gear startup problems resulting from errors in manufacture or assembly. There is, of course, another category of problems—those caused by either faulty field installation, or maloperation of the equipment. However, the Exxon survey of new plant startup records indicates that, over a 10-year period, equipment failures resulting from manufacturing errors, and resulting delays, accounted for 70 to 80 percent of the total project delays attributable to rotating equipment. In other words, vendor design and quality control errors, and resulting equipment failures, have a significantly greater impact on plant startup schedules than any other category.

This generalization applies to special purpose gearing as well as other high speed rotating equipment such as steam and gas turbines, and centrifugal and axial compressors. Figure 1 shows a general breakdown of machinery startup delays into individual problem categories. The two major areas, design and quality control, total over 70 percent of all potential delays, i.e., delays that were attributed to machinery only and not "camouflaged" by other startup difficulties. The remaining 30 percent of delay is made up of field installation errors, maloperation and miscellaneous other causes.

It should be emphasized at this point that the survey did not indicate a particular failure pattern in terms of manufacturers, geographical origin, or, for that matter, particular design concepts such as single vs. double helical. Gear startup problems are spread throughout the industry, and no special category of gears or suppliers was singled out as being particularly troublesome.

In order to uncover design and quality control errors at the factory before units are shipped to the job site, Exxon initiated an intensive followup program in equipment procurement and engineering on new projects, which

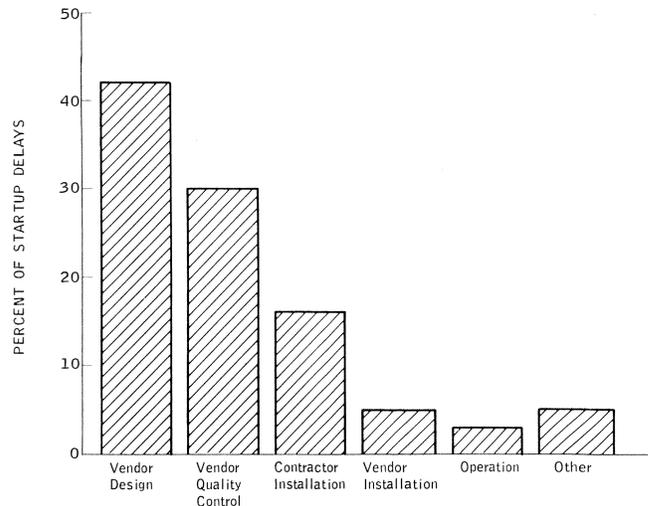


Figure 1. Machinery Startup Problem Distribution—1964-1973.

will be described in the following section.

Exxon Engineering Procedures For Gear Selection and Followup

Before entering into a discussion of Exxon's approach to equipment engineering, a few general comments about the role of Exxon Research and Engineering Company (ERE) are in order.

As the central technical organization of Exxon Corporation, Exxon Research and Engineering Company is responsible for engineering, project management and startup of major new capital projects. In order to utilize its engineering capabilities most effectively, its effort concentrates primarily on conceptual design of new projects, the preparation of technical equipment standards, and followup and supervision of contracting organizations which usually handle the actual equipment procurement and detail engineering. Or translated into machinery terms, the company usually prepares a general performance or duty specification, establishes machinery standards, and appraises the contractor's and vendor's engineering.

On a typical gear application, Exxon's equipment specialists review the contractor's inquiry specification, bid review, and vendor selection. During the detail engineering phase, they normally appraise critical design aspects such as rating calculations, critical speeds, bearing selection and loads, and witness shop tests.

In all of the critical design areas, ERE has developed its own in-house guidelines and criteria for acceptability, which are either contained in technical standards or used during followup to appraise the contractor's or vendor's engineering. By following this procedure throughout the engineering and manufacturing phase, Exxon is thus able to exert strong influence on the overall technical effort going into each machine. The company believes—and has confirmed this by recent field experience—that this approach goes a long way towards reducing the possibility of major design errors in gearing, and associated project downtime. To a somewhat lesser degree, these procedures also have reduced the frequency of quality control problems.

The specific areas emphasized in ERE's engineering followup are summarized in Figure 2.

Invoke API Standards With Amendments

Until approximately six years ago, industry lacked a comprehensive standard covering the overall design of special purpose gearing. Gears were generally purchased on the basis of the AGMA standards, which provide little guidance beyond basic rating and strength requirements. The API Standard 613, "High Speed, Special Purpose Gear Units for Refinery Services," which was issued in 1968, incorporated the AGMA rating standards but expanded its scope to include additional guidelines for design of the gear unit as well as the overall equipment train. It includes specific requirements in such areas as gear sizing, bearing design, vibration and critical speeds, lubrication and testing. The APA standard represents a satisfactory document in the sense that it establishes minimum design requirements. However, it is similar to other industry standards in that it is necessarily a compromise between numerous contributing companies which may have topics, and is often too general. For this reason, Exxon has widely divergent views and requirements on particular topics, and is often too general. For this reason Exxon has found it necessary to supplement the API standard with additional requirements which are issued under the title of "Exxon Engineering Basic Practices" for gears. This document, together with the API standard, constitutes the technical specification for gears and is transmitted to the manufacturer as part of the inquiry and purchase specification. Generally speaking, this Basic Practice adds specific, detailed requirements in those areas of gear design which experience has indicated are critical.

Purchase Proven Equipment

Exxon's plant startup history has demonstrated that the highest risk of major equipment design defects and delays is incurred with equipment which exceeds previous experience in size, horsepower, and speed. In such cases, the designer is forced to extrapolate from previous analytical or test information, often unaware of critical factors which come into play at the uprated speeds and power ratings. For this reason, Exxon attaches great significance to a manufacturer's experience, and requires satisfactory operation on at least two similar units as a precondition

for accepting a bid. On critical units, users of these reference machines are contacted to verify their actual operating history.

The administration of this experience clause in a bid review situation, of course, requires a good deal of engineering judgment. For centrifugal compressor drives, for example, speeds and horsepowers rarely match those of existing machines, and the extent of acceptable extrapolation becomes a matter of personal judgment. However, this requirement does flag to the vendor that the company is willing to pay a premium for proven equipment, and usually results in a more conservative proposal.

Manufacturers have, on occasion, voiced criticism of this approach since it appears to suppress design innovations and stifle creativity. This may be correct to a certain extent, but for equipment users, it remains a fact of life that design innovations have more often than not led to problems in the field. The resulting delays are, of course, much more costly to the user since he assumes the total downtime cost of an entire unit or plant, rather than only the replacement costs of individual machines or components.

Exxon does, of course, occasionally build plants where equipment exceeds present manufacturing and operating experience. In these cases, extrapolations from previous experience are flagged early in the engineering phase, and risks are minimized by added engineering reviews, contingency measures, and additional testing at the manufacturer's shop or in the field prior to actual plant startup.

Critical Speed Analysis

Although gears have not been a major source of torsional and lateral critical speed problems, a sufficient number of cases occurred where predicted values for complete trains differed considerably from those determined on the test stand or in the field. Based on this experience, and the high cost associated with resulting failures, it was considered necessary to develop in-house computer capabilities to calculate torsional and lateral critical speeds, as well as rotor response to imbalance. On critical equipment, these analyses are conducted starting from basic equipment dimensional drawings and using Exxon's own data on bearing stiffness and damping characteristics, and coupling penetration factors. These analytical tools have proved invaluable not only in detecting errors in the manufacturer's studies, but also in exposing sensitivities of the equipment to imbalance, changing bearing clearances, and lube oil temperatures.

Bearing Design Review

Based on failure experience in field startups, Exxon has developed specific application guidelines as well as design appraisal tools for thrust as well as radial bearings. This information is used as the basis for selecting general bearing types and sizes during equipment purchase, and to carry out detail design checks in the engineering phase. Items which would be reviewed during followup would be such design aspects as oil film thickness, temperature rise, and attitude angle (stability) on all sleeve bearings.

For thrust bearings, Exxon's specifications now require double-acting tilting-shoe bearings with load equalization (Kingsbury type) on all compressor drive appli-

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| EXXON GEAR ENGINEERING FOLLOWUP |
| <ul style="list-style-type: none"> • INVOKE API 613 PLUS AMENDMENTS • PURCHASE PROVEN EQUIPMENT • REVIEW CRITICAL SPEED ANALYSIS • REVIEW BEARING DESIGN • REVIEW OIL DRAINAGE • INSTALL VIBRATION MONITORING EQUIPMENT • STANDARD API SHOP TESTS AND INSPECTION |

Figure 2. Exxon Gear Engineering Followup.

cations. Also, these bearings must be designed for a continuous load equal to the sum of the coupling lockup forces based on a 0.3 friction factor, and any additional thrust forces that may be generated within the gear or imposed externally, for example, by motor centering forces. At these loads, the bearings should operate within the following limits:

- Center-pivot (Kingsbury type) — 300 psi load
- Optimum pivot (Michell type) — 400 psi load
- Maximum Pad Temperature — 212° F
- Minimum Oil Film Thickness — 0.001 in.

As an interesting sidelight, some of the test work instigated by Exxon after thrust bearing failures indicated that the rating information published by some bearing manufacturers was optimistic. It was found that some bearing types could not achieve their catalog ratings without exceeding maximum allowable oil temperatures and that maximum loads for particular bearing type resulted in considerably different temperature profiles than was previously assumed.

Exxon has compiled similar calculation procedures and design guidelines for most commercial radial bearing types, i.e., sleeve type, elliptical, pressure dam, and tilting pad types. Although they are not part of Exxon's formal equipment specifications, they generally agree with the manufacturers' design procedure with the exception of oil film thicknesses, which are often below the 1 mil considered acceptable.

Oil Drainage

Oil foaming and windage problems need close follow-up not only in gear procurement but also in the overall installation design. For gears with pitchline velocities above 24000 FPM, it is Exxon's practice to require either an extra deep (30 in.) oil sump below the bull gear, or dual drain openings. For these units, care must also be taken to assure continuous slope and even flow in the oil drain piping. As additional items, independent oil flow regulation to the gear mesh is required and the overall temperature rise in the gear oil is limited to a maximum of 60° F.

Vibration Monitoring Systems

In addition to the engineering followup described above, Exxon invests considerable capital in vibration monitoring and protection systems which are installed on all special purpose gearings. Present practice is to install two non-contacting vibration probes on each gear bearing with one probe connected to a vibration monitor, which contains amplitude readout and alarm and trip circuits. Despite their cost, these systems are vital to long-term reliability not only because they provide early warning of incipient problems, but also for troubleshooting. The vibration probe output signals can be fed into an oscilloscope for visual display, or used for various methods of frequency analysis to determine the cause of abnormal vibrations.

As a further step in this direction, Exxon has now set up facilities for computerized machinery signature analysis. Vibration signals are recorded periodically during the life of the equipment and fed into the computer for comparison to the baseline recording. Any changes in fre-

quency composition or amplitude can provide valuable information regarding internal wear, unbalance, or other factors affecting run length capability of the equipment and scheduling of overhauls. This signature analysis system is presently being set up in the Exxon, U.S.A. refineries and will soon be expanded to overseas facilities.

Inspection and Tests

The specification items and review procedures which have been described are all aimed at detecting and avoiding engineering design errors but have little or no effect in the area of quality control. For the user of special purpose gears, control of manufacturing and assembly quality is a task that must primarily be left to the manufacturer and can only be supplemented by purchaser's inspection at certain critical points. Also, purchaser's specifications rarely include detail material specifications and tolerances, which leaves the inspector without guidelines other than the manufacturer's own quality standards.

Ideally, any significant quality control errors, other than those relating to long-term reliability, should be detected by the mechanical running test at the manufacturer's plant. However, no-load tests are only partly effective in this respect, and few manufacturers have the necessary facilities for load testing. This lack of test facilities can be partly compensated by the use of more sophisticated vibration instruments which can provide more reliable indications of overall quality and potential problems during unloaded run tests.

The Exxon inspection procedures concentrate primarily on basic material standards for casting and forgings, and specific items included in the specifications such as tooth finish, backlash, and tooth contact. Mechanical run tests are specified per API 613, and the casings are subsequently opened for an internal inspection of bearings, seals, and tooth contact patterns. To provide as near a duplication of actual operating conditions as possible during the shop test, the vendor is required to use lube oil of specified viscosity, and to simulate actual weights of couplings and spacers. Contract vibration probes are used, and output signals are analyzed for any half-or-multiple-frequency components. Where possible and warranted, gears are load-tested or are shipped to the manufacturer of the driven equipment for a combined run test.

Followup Pays Off In Reduced Downtime

On the surface, the engineering followup outlined above appears to be a rather extensive effort for a user to undertake, and one could argue that all these engineering checks and analyses are really the manufacturers' responsibility. In an ideal situation, this, of course, is correct. In real life, however, competitive pressure and manpower limitations often restrict vendors in the amount of engineering effort and conservatism they can apply in the design of a particular unit. Also, the gear manufacturer is usually not assigned overall engineering responsibility for the train, which limits his input in engineering the overall train. A third factor, which is of particular significance to international companies, are difficulties in communications between driver, gear, and driven equipment suppliers which are often located in different countries and even in different continents. Because of these factors, experience has confirmed that this engineering effort does, in fact, uncover potential design errors, and is more than justified

by the reduction in new project startup delays.

Returning to the startup problem study for a moment, the average project downtime due to startup problems on special purpose gears is shown in Figure 3. As can be seen, startup delays gradually increased in the late sixties to a peak of 3-1/2 days in 1969. At that time, the intensified followup procedures went into effect, and project delays have been reduced to less than 1 day per unit. This may appear to be a low order-of-magnitude change; however, it must be borne in mind that daily downtime losses on major process units can be quite high and usually more than enough to compensate for the cost of the additional engineering followup.

At this point, Exxon has reached a point of diminishing return in this engineering effort since further reductions in project downtime could only be achieved with a disproportionate increase in effort and cost, and may not be feasible at all. The remaining gear problems are due to occasional quality control defects as well as field erection errors which can be corrected with relatively little downtime. Therefore, the company feels that it has achieved an acceptable level of mechanical startup performance in special purpose gearing and plans to continue the present engineering followup procedures to avoid repetition of earlier problems.

It is recognized that there may be many other methods for reducing commissioning problems with special

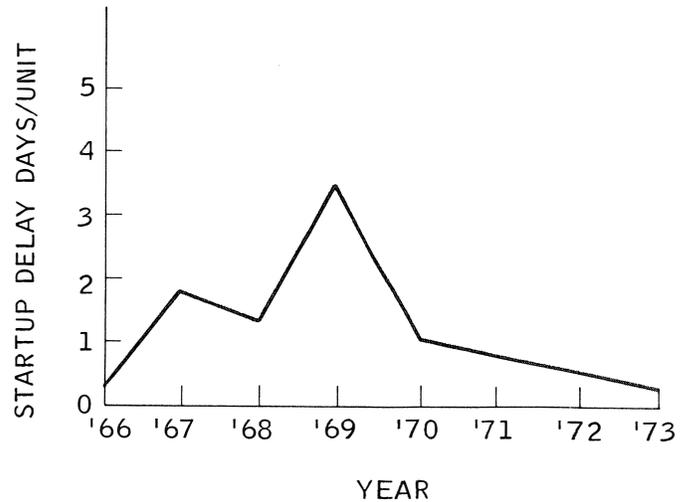


Figure 3. Gear Startup Delays.

purpose gearing in the field. For a variety of reasons, this large company operating with affiliates on a worldwide basis with tight construction schedules has found this approach to sound gear application engineering most effective.