

SELECTION AND APPLICATION OF HIGH SPEED GEAR DRIVES

by

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ABSTRACT

This will be an exploration of the relevant factors involved in successfully applying gear drives to high speed rotating machinery. Relationships of internal design features to external equipment and couplings is defined for use in designing compatible shaft systems. Procedures for installation, check out, maintenance and assurance of continuous service are suggested.

INTRODUCTION

The application of gear drives to large shafting trains involving high speeds and power levels has never been an easy task. With current materials and heat treating techniques, the use of high hardness gearing with tooth loads in the order of 1500 to 2000 pounds per inch of face at pitchline velocities of 20,000 to 30,000 feet per minute is not at all uncommon. In turbine driven test equipment, gear drives have been built with pitchline velocities as high as 55,000 feet per minute and rotational speeds approaching 100,000 RPM. The magnitude of internal forces and material stresses, coupled with the high speeds has resulted in gear drives which are dynamically complicated and sensitive to influences from other components in the system. Parallel increases in the technology of turbines, compressors, couplings and bearings have resulted in components which can exert substantial external forces on connected equipment. It is most important that effects on the gear drives from external system components be understood and examined at the design stage if equipment is to be designed which will avoid severe problems in achieving successful operation.

As the gear drive will be a substantial economic block in a process system it is the subject of considerable cost pressure and is usually designed to very closely suit the stated operating conditions. As a result, it is likely

to be the "fuse" for many kinds of system disturbances whether internally or externally generated. The gear manufacturer's responsibility as defined by the drive warranty should be clearly understood as a gear unit is expressly not warranted against failure or unsatisfactory operation resulting from dynamic vibrations of any form imposed upon it by the drive system in which it is installed, no matter how induced, unless the nature of such vibrations have been fully defined and explicitly accepted as a condition of operation.

This puts the burden of successful operation directly upon the shoulders of the system packager, whether the OEM or user. How then does a potential purchaser of a system evaluate a proposal which includes the use of high powered high speed gearing?

SERVICE FACTOR

The common denominator for major gear quotations is the service factor. This is defined as the minimum ratio between calculated capacity and average transmitted load for any component of the system. In general, one of three criteria will be the controlling influence in a gear drive. These are failure due to tooth surface pitting, wear, or physical loss of teeth due to breakage. Consequences of the three modes of failure differ, particularly in regard to the length of time involved. Wear can continue at a slow rate for a long period of time without affecting the serviceability or reliability of the machinery. Pitting, if progressive, will eventually destroy the working profile of the teeth, altering the thermal characteristics and often rendering the drive unsuitable because of high vibration levels long before the teeth are incapable of carrying load. Loss of a portion of a tooth by breakage has immediate consequences. The balance is immediately and drastically affected by this and in the case of a major tooth breakage the gear will be incapable of further operation. This will necessitate the immediate shut down of the drive with no regard possible of the effects on other related equipment. Any evaluation of a service factor then should determine which of the three modes is involved. Current practice by Falk includes the automatic provision of an additional 50% margin when designing for gear tooth bending. This has the effect of eliminating gear tooth breakage as a primary cause of failure except in the case of severe and unforeseen overloads.

Design against failure by wear under heavy tooth loads will result in the selection of heavy bodied lubricants, generally 150 SSU or more at supply temperature. Pitting failures are the most difficult to provide a margin against as increasing gear size or hardness are the only means of improving capacity and both entail an increase in cost.

The service factor itself is not an overload capacity, per se, as it includes either empirical or theoretical estimates of the effect of such factors as length of service life, torque fluctuations, and reliability level required. The service factors as established by the American Gear Manufacturers Association and published in their standards are intended for application to transmitted load requirements: if substantial overload capacity is planned or allowed for as in the case of an oversize driver, for example, additional gear rating must be included to provide for operation at those levels. Similarly, torque loads resulting from torsional oscillations or faulty operation are outside the scope of the normally applied service factors and must be evaluated and provided for separately. It should be noted that any torque fluctuation which results in separation of the gear teeth at speed will be most difficult to provide for. Impact loads occur during re-engagement and very short service life is a frequent result of operation under these conditions.

TYPE OF DRIVE

Normally, gear drives proposed for turbine driven applications will be of single helical or double helical type with rotors carried in sleeve type bearings.

The blanket exclusion of rolling type bearings from drives of this class may be unwarranted as, particularly in the lower horsepower ranges, bearing ratings can be easily provided such that race and roller fatigue can be ignored as a source of failure. Drives using rolling element bearings may sometimes provide additional design latitude for the gear manufacturer in providing economical high quality equipment for extended service lives. The extensive use of rolling element bearings in contemporary light weight gas turbine designs bears strong testimony to this point.

The choice between single and double helical gearing is sometimes difficult even for an experienced gear designer. Both types of gearing can be made to equal limits of accuracy as control of the accuracy of gearing is a function only of the accuracy and maintenance of the gear generating machinery, machining techniques and operator skill. A hobbled gear is generated in a continuous process by a simple and easily maintained rack form cutter which will produce gearing of extremely high profile accuracy with virtually immeasurable spacing errors and uniform lead. Where both helices of a double helical gear are cut at the same time or sequentially without a change in setup, apex position error will be virtually unobservable either in the Lab or in operation and axial vibration excitation from the mesh will be negligible. The same basic equipment can be used for generating either single or double helical gears although the continuous finishing processes used for double helical gears produce a higher order of accuracy of lead and tooth spacing than does grinding if it is used for finishing a single helical type.

If finish machining of the working flanks of the gear teeth is employed, any operation which is performed upon a single tooth at a time without reference to other teeth on the gear obscures whatever accuracy the teeth may have had prior to that point, and substitutes the indexing and other characteristics of the finishing equip-

ment. Minimum stock removal requirements and cutting pressures involved virtually guarantee an abrupt space error between the first and last tooth finished. The use of any reciprocating machine for finishing of gear teeth to high standards of accuracy can be justified only on the basis of continual maintenance and adjustment of the stops and ways to counteract wear effects of the frequent reversal of massive tool heads.

External thrust loading is a significant problem in the design of any gear unit and effects differ based upon the choice of single or double helical gearing. In either case, an accurate estimate of the thrust loading is required to make an intelligent compensation for it. With double helical gearing, continuous axial loading can be accommodated by a slight increase in capacity to account for the helix load imbalance. The increase in cost and reduction of efficiency thus caused is only a fraction of that incurred when a large diameter high velocity thrust bearing must be mounted on a single helical pinion shaft. As an example, Falk builds double helical gearing for use in single-poster compressors where the induced thrust is carried entirely by imbalance of the helix forces and these gears perform excellently with no visible deterioration of either helix.

Intermittent loading such as that from gear couplings mounted on thermally expanding shafting is an entirely different kind of problem. This is accommodated in a double helical unit by judicious selection of helix angle and coupling size (they directly affect each other) so that axial coupling forces resulting from the transmission of torque are less than the thrust force produced by each helix of the gear. This assures that the coupling will slip to relieve any axial loading and that the balance of power on the two helices will be maintained. This is the design procedure which has been used for high quality marine propulsion gearing for the last fifty years and in most designs where lifetimes of twenty to forty years are anticipated. When high speed gear couplings whose pitch diameter is substantially smaller than that of the pinion are selected, axial forces produced will be high and the combination should be examined very closely as a potential source of trouble.

The range of coupling travel, both lateral and axial, should be established for each shaft and the thrust bearing located so that external thrusts or position shifts will have a minimum effect on the gear meshes. In the case of the single poster compressors above, another entirely satisfactory solution would be to use a thrust bearing on the pinion (compressor) shaft, none on the gear, use a sleeve bearing motor and let the gear and motor rotor follow the pinion bearing.

In single helical gearing all forces externally generated must be added to the thrust produced by the gear itself and the total used on each shaft to select the high speed (and low speed) shaft thrust bearing. The consequence of error in thrust or bearing capacity estimation in this case will be frequent failures of the thrust bearing or the associated shafting.

Single helical gearing, due to the asymmetrical loading from the helix, has two sources of design difficulty which do not exist in double helical gear sets. The effective center of tooth pressure oscillates back and forth

across the face putting substantial alternating loads on the shaft bearings. This results in peak bearing loads substantially larger than those calculated or anticipated and which can lead to early bearing failure if sleeve bearings are used at their expected capacity. In addition, the helix induced thrust force causes the gearing to try to skew in the housing, both unbalancing the bearing loading and forcing the gearing to run out of parallel. One reason why there has been so much attention paid to crowning of single helical gearing is that the effects of end tooth bearing due to shaft misalignment must be alleviated. To reduce this effect, sleeve bearings must be designed with substantially less than optimum oil film clearance leading to higher operating temperatures, greater sensitivity to oil contamination, and in borderline cases, reduced bearing life. If properly designed for, with preloaded or very small clearance rolling element bearing, as in several manufacturers' commercial designs, these effects will be small and single helical gearing can be run quite satisfactorily.

In single helical drives, an additional consideration arises in that torque is related to axial thrust, which is supported by the compliant hydro-dynamic and mechanical characteristics of the thrust bearing and its supports. The torsional and axial response characteristics are effectively coupled which complicates and intensifies dynamic problems as any torsional cyclic variations are automatically translated into axial vibrations.

Noise from an operating gear set is a function of roundness and concentricity of operating elements, both gearing and shafting, accurate balance, and in particular, control of tooth spacing, accurate errors and uniformity of mesh stiffness to reduce meshing frequency excitation. It is significant that for submarine gears, where the ultimate in quietness is essential, the hallmarks are moderate tooth loading, fine pitch, high helix angle, and low pressure angle; all diametrically opposite from the usual and necessary practice for single helical, hardened and ground gearing, which have low helix angle (for minimum thrust), very coarse pitch teeth (to get adequate strength) and high loading (because of the carburize hardening).

As you may gather from the foregoing the double helical high speed gear is the first choice for high reliability applications. The reasons are those which have always selected in its favor in that accuracy of calculating loading, smoothness of operation, and predictable performance render unnecessary any juggling or deviations from easily defined and measured geometry. These gearsets will be more efficient with unmatched reliability if properly applied, providing reasonable intelligence is exercised in selection of the couplings, and will run with less vibration and at noise levels that are often indistinguishable from that of the connected machinery.

INTEGRATING THE SYSTEM

Suitability of new shafting design in terms of critical speed response is a complex problem requiring the maximum in cooperation between the gear designer, the foundation designer, and coupling suppliers as not only shaft stiffness, but bearing stiffness, housing stiffness, foundation stiffness, coupling overhung mass and imbalance are all important factors. The system shafting

shall be designed to operate sufficiently remote from lateral critical frequencies to allow balancing for smooth operation under any normal combination of speed and load. Due allowance must be made in calculations for the effect of changing bearing stiffness due to loading and speed in gear drives with sleeve bearings. Torsional characteristics should be such that no torque ripple in excess of 10% of the mean transmitted load can occur and under no condition should torsional disturbances be permitted which will result in tooth separation under load.

A substantial structure to support the gear drive weight, thrust and torque reactions with minimum load deflections must be provided. At least two dowels for locating each gear housing are required and attention is directed to the need for minimizing housing vibrations from whatever source. Ideally, the structure should be reinforced concrete or steel filled with grout. The inclusion of oil reservoirs in the structure supporting major train components should be avoided as unavoidable thermal changes will have adverse effects on alignment. If a concrete reinforced or a filled structure cannot be provided, resonance due to train component mass and structure stiffness at system rotational frequencies or harmonics should be avoided.

In addition to considerations of the effects of induced or transmitted thrust on connected equipment and any desired torsional or damping characteristics, couplings should be selected with due regard for the effects of long service at the speeds involved. If extended operation is required, flexing discs or diaphragm couplings may be required to secure freedom from batch grease lubrication or oil lubrication sludging difficulties. Standard commercial balance levels are not normally adequate and trim balancing in assembly will usually be required to permit operation with minimum vibration levels. Alignment should be established to account for the axial spring rate of diaphragm type couplings in addition to normal thermal and load induced changes in the structure.

Lubrication piping should be arranged to minimize housing strains due to piping thermal changes. Consideration should be given to providing flexible or slip sections at critical locations to avoid disturbing train component alignment.

As gear tooth surface wear rate is directly affected by lubricant viscosities the gear manufacturer's recommendations should be carefully followed as even minor changes can substantially alter gear performance in critical situations.

PROTECTION BEFORE START UP

Preservation of the gear assembly by the manufacturer should be specified to maintain the drives during the period of installation and storage before use.

Standard procedure at Falk is to seal any openings with pressure sensitive tape, protect internal parts with vapor phase rust inhibitor and coat external machined parts with a firm film forming, polar type rust inhibitor compound. Small hardware is either wrapped in water repellent, vapor phase, rust inhibitor type paper and sealed with pressure sensitive tape or placed in plastic

bags containing vapor phase, rust inhibiting type crystals and sealed. Hardware is then boxed for shipment with the unit. We have adopted this procedure for very good reasons. First, protection is afforded for twelve months stored outdoors, or twenty-four months stored in a dry building; secondly, and we believe more importantly, the gearbox can be installed and operated without the need of disassembly for removal of preservatives. Should the unit be disassembled for removal of preservatives in a dusty or dirty construction site atmosphere, it is almost certain that some grit will remain in the lube system. We have found that handling of gears even though they are made of steel can cause bumps, gouges and other serious surface defects which are harmful to the operation of the gear. Also, since high speed gears are nearly always retained in sleeve bearings, the babbitt of the bearings can easily be upset, scraped and scored. When a gearbox that has been so protected arrives at the job site, it is important that the unit remain sealed until ready to install or additional means be taken to protect it. Should someone open the inspection cover, the protective atmosphere is lost. The same is true of the hardware and spare parts.

SET UP AND INITIAL OPERATION

The mounting surface should be a flat, level, single plane surface of finished steel at a height which will permit the shimming necessary to properly align the gear unit to connecting shafts. The shims should be of a size at least equal to the width of the unit foot pad. Estimate the required amount of shim according to the dimension from the mounting surface to the centerline of the driving or driven equipment and the base to centerline dimension of the gear unit, allowing for thermal growth. Place that amount of shim equally at all foundation bolt locations. Then place the gear unit on the foundation in the approximate required position. Use a feeler gage and check shim packs at each location for tightness and adjust as necessary, to maintain equal support at each shim pack. Uneven supports can distort the gear box and adversely affect the gear tooth contact.

Using a dial indicator mounted from a shaft check alignment to the connecting shaft, adjust shim packs as necessary until the shafts are properly aligned, again allowing for thermal growth. The indicator support should be so rigid that it cannot sag and thus give erroneous readings as the indicator is rotated. Re-check shimming with a feeler gage.

Tighten foundation bolts uniformly then re-check alignment. It may be necessary to repeat the shimming and tightening foundation bolts to obtain final correct cold alignment. Then check tooth contact by lightly coating the gear teeth with Prussian Blue or suitable substitute. Rotate or rock the pinion or lighter element back and forth sharply within the confines of the backlash. It is recommended that a person knowledgeable of proper tooth contact be responsible for the evaluation of the contact so checked, preferably the gear manufacturer's service man.

Alignment of high speed gear units should always be hot checked and adjustments made as necessary. Temperatures vary so greatly throughout the housing

and shafting that it is impossible to accurately calculate a thermal growth and, therefore, the ultimate alignment check is in the hot condition. Methods of checking vary from simple micrometer measurements to sophisticated optical equipment measurements.

When the alignment is complete, the baseplate or bed should be grouted in as close to the gear housing base as possible.

It is mandatory that the lube oil system be thoroughly cleaned before start-up. The usual procedure is to blank off the gearbox and other components to which acid would be harmful and then acid flush the system, followed with a neutralizing flush before flushing with the lube oil. When an auxiliary pump has not been provided, it will be necessary to adopt other means to flush or clean the system. Gear mesh spray nozzles should be checked to be sure dirt has not been pumped through the system and blocked the orifices. This may be accomplished by circulating oil through the system and observing the sprays or by introducing high pressure air into the spray nozzle manifold. During installation, provision should be made to monitor vibrations during start-up and operation.

When possible, gears should be run-in on initial start-up. Speeds and loads should be increased in percentage increments. Lube oil temperature and pressure and bearing temperatures should be observed and adjustments made to the lube system as required. The number of adjustments will depend on the complexity of the system, of course. Of primary importance is the oil pressure. When an auxiliary pump has been provided, oil should be circulated before the actual start. If not, the pump should be primed and the journals wetted with oil. Primer holes are sometimes provided or alternately journals can be oiled through the holes provided for bearing temperature detectors. It is recommended that warning devices be provided to eliminate as much as possible the human error and the setpoints should be checked.

OPERATION AND MAINTENANCE

Sleeve bearing gear units simply will not run without oil. Under normal operating conditions, the oil should be changed every 2500 operating hours or every six months, whichever occurs first. Where operating conditions warrant, this period may be extended; conversely, severe operating conditions may make it necessary to change oil at more frequent intervals. Such conditions may occur with rapid rise and fall in temperature which produces condensation, or when operating in moist or dusty atmospheres, or in the presence of chemical fumes. In any case, the lubricant supplier should be consulted when determining a lubricant maintenance program. It may be necessary to periodically analyze the oil until a reasonable program can be established. Shut-down or a unit which has been operating in a humid atmosphere can result in considerable condensation and subsequent rusting of the gears and shaft journals as well as housing in a very short time. When water contacts clean steel, it begins to etch the steel immediately. When shut-downs in such conditions are necessary, provisions to prevent condensation must be furnished. Some-

times the heat from a lightbulb suspended in the gearbox may be adequate. Alternately, continuing to circulate hot oil may be the answer. The case in point here is that shut-downs as well as start-ups require care and attention.

It is advisable to check bolt tightening during the first days of operation of a new unit. Thermal expansions and vibrations will cause bolts to loosen. Visual inspections are only common sense and can lead to preventive maintenance measures which circumvent the need for costly repairs.

Should it become necessary to service a high gear unit, it is recommended that it be done under the supervision of the manufacturer's service man. He is a specialist, skilled in recognizing many causes of gear problems and their corrective measures. Furthermore, he is

eminently well qualified to rebuild or reassemble a unit as it would be done at the factory and to perform inspection to insure satisfactory operation.

Stocking of spare parts is an important part of any maintenance program. Frankly, the designer of high speed gears is of the opinion that the gears will operate indefinitely when properly installed and maintained. However, accidents do happen and the user should evaluate the cost of down-time and repair and decide what he should invest in spare parts. High speed units are custom designed and built to order; therefore, the manufacturer does not stock parts and months may be required to manufacture replacement parts should there be a failure for some reason. Generally, spare rotating elements and bearings will suffice; however, some users have purchased complete standby units to insure a minimum of down-time should a failure occur.