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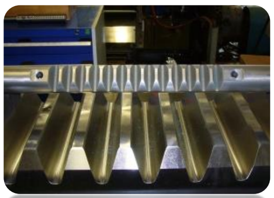


GENERAL OVERVIEW

In power-transmitting gear trains which are loaded unidirectional, backlash (the free movement between gear tooth flanks when set at working center distance) is functionally unimportant, and too much is better than too little. The situation is different where the direction of tooth-loading reverses, in the case of drives required to relate or control rotational/angular position. Rack/pinion drives by their nature are also subject to reversal of tooth loading.

RACK & PINION SYSTEM BACKLASH

The stipulation *"gears to run without backlash"* is almost impossible to achieve. This is due to the effects of various kinds of "errors" i.e., manufacturing tolerances. Such tolerances include – center distance, radial run-out, lateral run-out, pitch and profile. In combination these variances act to produce "tooth-kick", "tooth-to-tooth composite error", tooth thickness variation, and more importantly with rack/ pinion, tooth, and axis alignment errors.

Wherever possible a spring tensioned, and flexibly mounted pinion is recommended. This allows full in-depth mesh (dual flank contact) with provision for the pinion to "ride" errors/tolerances that cause high spots. Another possibility, subject to the torsional loading, is the use of "anti-backlash" pinions.

Rack/pinion systems combining drive loading with positional accuracy, and with the pinion mounted at fixed centers, demand that the control of manufacturing tolerances is held to a minimum, great care is taken in ensuring drive alignment and precise setting of the actual centers occur. This is to achieve consistent minimum backlash with no "tight" engagement anywhere along the rack traverse. Gears, which must operate with minimum backlash, must be mounted so that the tooth alignment under maximum loading is well preserved. With rack/pinion the mounting must also cater for the reversal of loading.

Typically, gears of AGMA 10 quality (BS grade 6) – *"precision cut quality"* or better are necessary for minimal backlash applications. The use of pinions having crowned and barreled teeth can aid problems associated with any lack of rigidity and alignment of the pinion with the rack.

STRAIGHT SPUR RACK & PINION SYSTEM

While straight spur rack/pinion drives can, with the above care, achieve minimal backlash, the effect of manufactured tolerance errors can be evidenced by noise and lack of smoothness of the drive. As many applications are now working with higher speeds (and higher power) other considerations are: the use of pinions having full involute profile contact (where there are low numbers of teeth) and incorporation of profile modification to ensure no under-cutting. Rack/ pinion drives where the pinion is mounted fully in mesh can also benefit from the use of rack teeth with tip-relief and/or semi-topped cut teeth (see separate informational report). This is to avoid the phenomena of rack tooth addendum tip "interference" in the pinion dedendum root. Such applications need individual consideration, and where possible, the avoidance of pinion tooth contact at or near the "base circle diameter".

HELICAL RACK & PINION SYSTEM

The next level of application improvement implements the use of helical pinions of proportions to ensure an over-lap ratio >1. These systems can be operated at higher speeds compared to spur gears. This is because, unlike spurs that have a cycle of one pair, two pair then one pair, two pair then one pair etc. of teeth in contact during and along the tooth contact path cycle, helical pairs with over-lap greater than unity always have two pairs of teeth in mesh. The line of contact along the tooth surface is no longer parallel to the axis as for a spur gear, but is inclined, as a result of the different stages of tooth contact along the tooth face. Thus, the tooth engagement and load distribution are gradual with helical gears, and can be used to performance advantage when the speed is too high for successful application of spur gears.

EFFECT OF HELIX ANGLE

The larger the helix angle (HA), the greater the reaction of side thrust (a function of Tan HA) and the smaller the required face width to achieve an overlap ratio >1 (a function of Cosec HA). The smaller the HA, the smaller the side thrust and the larger the required face width. For example:

1" face, 12NDP, 20 degree NPA, 30 degree HA, minimum face 0.523", overlap ratio 1.91, ratio of side thrust to tangential load 0.577.

If 20 degree HA, minimum face 0.765", overlap ratio 1.31, ratio of side thrust to tangential load 0.364

The latter requires less power to transmit the requested tangential force

LUBRICATION

Many X, Y, Z machine tool rack/pinion applications are in environments not suited to the usual and necessary lubrication requirements, such as where the application of grease/oil etc. could be detrimental to the adhesion and accumulation of swarfe. Such drives must still have some form of lubrication. The operation of drives without any lubrication results in metal-to-metal contact which aggravates the initial run-in period. In this situation "high spots" will "pit" and "score" followed by early wear of tooth flank surfaces.

Such applications should be coated with a suitable "anti-scuffing" agent. These are available as aerosols, are non-sticky and leave a dry film coating that resists accumulation of dust, dirt, and contaminants. This prevents or minimizes potential seizure and "galling".

LUBRICATION

The above synopsis is offered only as a guide to the avoidance of numerous problems that are often associated with *"zero backlash"* requirements. Each rack and pinion system should be fully evaluated to determine the best design components and system configuration for the particular application. However, the general rule of thumb is; wherever possible, some backlash ought to be incorporated in the rack/pinion design.

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