INFLUENCE OF HARDENED AND GROUND GEARS

ON SYSTEM DESIGNS

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Introduction

Case-hardened and ground gears have been manufactured by the Maag Gear-Wheel Company in Zurich for the past fifty years, i.e. since the first grinding machines for generating involutes were completed. At that time carburizing was the only practical case-hardening method. In the following years and up to the present time this method has been subject to continuous development and refinement. The carburizing process is now controlled fully automatically and its costs have been reduced considerably. So it is still the most frequently applied hardening method in our works today, although there is a fair amount of induction hardening and nitriding being exercised. The statements in this paper are therefore based largely on experience with carburized and hardened gears ground on the Maag grinders.

Effects of Surface-hardening on Tooth Load Capacity

What is gained by case-hardening the teeth?

1) Above all, the surface durability is greatly increased. It rises in proportion to the square of the hardness - or of the tensile strength.

2) There is an increase in tooth bending fatigue strength, which, however, is less impressive than that in surface durability. Compared with a through-hardened gear of 300 Brinell, a carburized tooth hardened to 60 Rockwell C attains an endurance bending strength, which is approximately 50% higher. 1

It would only be possible to fully utilize the very high surface durability of case-hardened teeth, if their bending fatigue strength could be raised to a similar degree. Much can be done towards this end by resorting to a coarse pitch. But it must not be forgotten that coarser pitches mean a higher velocity of sliding contact between the flanks, and hence also greater danger of oil film failure.

Fig. 1 is an attempt to represent the limiting loads on a gear in such manner that the influence of the three criteria, i.e. bending fatigue strength, surface durability and oil film resistance is clearly apparent. The curves are drawn over a "tooth size" axis (diametral pitch; module) for a given medium sized gear; through-hardened in the first case and case-hardened in the second case.

The bending fatigue strength (curve 1) increases with the size of the tooth, whereas the scoring resistance (curve 2) decreases.

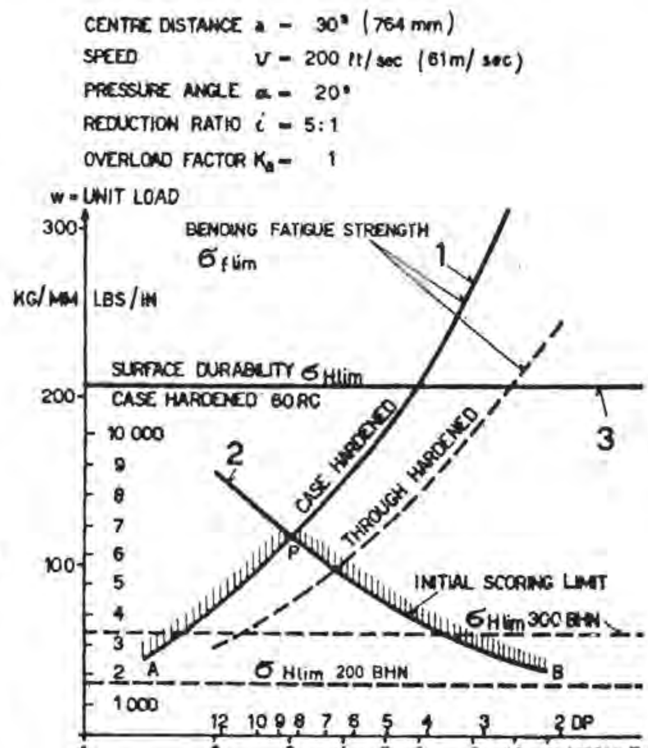


Fig. 1

Limiting loads on a medium sized gear

The surface durability (curve 3), however, is practically independent of tooth size.

In the case of soft gears, e.g. 200 Brinell, we see that the load capacity is governed over the whole practical pitch range by the surface durability, i.e. by the Hertzian pressure. This is illustrated by the lowermost horizontal dotted line. Pitting is therefore the most common defect experienced in soft gears.

But with case-hardened gears, the conditions are quite different. The limiting load is given by the curve APB, governed over the fine pitch range AP by the bending strength, and over the range PB by the resistance to scoring. With this size of gear the surface durability, way above the limit curve APB, no longer affects gear dimensions as it does in soft gears. However, it should be noted that the smaller the gear, the lower the curve of surface durability sinks in relation to the bending strength. This means that with small sized case-hardened gears the Hertzian stress comes into the picture again. These relationships have been confirmed in service and on the test stand. 2
We also see from the curves that optimal load capacity would be achieved with a pitch represented in Fig. 1 by the point "P".

In practice of course, a case-hardened gear is so dimensioned as to keep well to the right of point P. That means that the pitch would be chosen first and foremost to assure adequate safety against tooth fracture. It is therefore the scoring limit which also becomes a governing factor for case-hardened gears.

The scoring resistance is lowest with a new gear. But with increasing number of service hours the tooth flanks become smoother and the scoring resistance increases considerably. Where loads are high, therefore, the problem lies mainly in the avoidance of scoring in the early life of a gear: we call such scoring "initial scoring".

Curve 2 in Fig. 1 represents the load limit where initial scoring must be expected in this particular gear unit when it is new and lubricated by a straight mineral oil. This danger zone of initial scoring can be raised by using an EP oil, or by other means as for instance electrolytic copper plating of the pinion teeth. A copper layer of 1 - 2 ten-thousandth of an inch is very efficient in overcoming initial scoring: and the costs involved are small.

However, the large majority of all case-hardened gears for industrial and marine applications are usually designed with specific tooth loads which are below the danger zone of initial scoring. Therefore most case-hardened gears can be loaded to the maximum designed power right from the start, and no special running-in measures must be taken.

Here it must be pointed out that the kind of running-in often practised with soft gearing, in order to straighten out toothing- and alignment-errors, is not practical with case-hardened gears. The grinding marks on the tooth flanks polish out relatively slowly; they are usually still visible after many years of service. Running-in of case-hardened gears can only smoothen the tooth flank to some extent. For this reason they should be of a high accuracy when they leave the grinding machine, and they should be carefully mounted to ensure optimum tooth alignment. This fact is of great importance, and must be kept in mind when designing systems with hardened and ground gearing.

Influence of Case-hardened and Ground Teeth on the General Gearing Conception

The properties of case-hardened gears described beforehand and illustrated by Fig. 1 lead naturally to certain characteristic features which distinguish such gear drives from the conventional non-hardened type. These features are:

relatively coarse pitch
smaller centre distances and
considerably narrower facewidths.

Facewidths are generally at the most half that of an equivalent non-hardened double-helical gear. This fact in itself is adequate ground for the almost exclusive use of single helical gears in case-hardened trains. But there are a number of other pressing reasons, of technical and economic nature, for working with single helicals.

We shall deal with these various points briefly, for they do not only influence the design of the gear unit itself, but often the layout of the whole transmission system as well.

1) With regard to costs, it is obvious that a single-helical gear will be less expensive. It is more time-consuming and costlier to grind a double-helical gear, since the two halves cannot be ground simultaneously.

2) The effect of small tooth alignment errors in single-helical gears can be eliminated during the assembly stage, simply by adjusting a pinion or wheel bearing accordingly. In the case of double-helical gears, such errors can at best be shared between the two halves.

3) Operational experience shows that single-helical gears with comparatively small helix angle do not excite axial vibration. Such vibration sometimes develops in double-helicals, due to the different composite errors in the two halves.

   Furthermore, single-helical gears which have relatively small helix angles remain largely unaffected by axial vibration induced by external forces. This is an important consideration for marine gearing, for example, where the ship's screw often excites heavy axial vibration, which is transmitted directly to the output gear of the drive unit, and can lead to tooth defects in double-helical gears of large helix angle.

4) From the design point of view the single-helical gear affords scope for interesting solutions, which are not possible with double-helical gears. We shall return to this matter later with a few illustrative examples.

It is often argued that due to the thrust of the single-helical toothing, and the resulting unequal reactions in the bearings, the gear axes become misaligned, impairing the uniformity of the load distribution. But practical experience with gears fitted with journal bearings has shown that there is no noticeable change in the tooth alignment, if the following design rule governing the minimum bearing span is applied:

The product of the two ratios facewidth to bearing span and gear diameter to bearing span multiplied by the tangent of the helix angle should be not larger than 0.16 to 0.22.

$$\frac{F\ d\ \tan \psi}{L^2} \leqq 0.16 \text{ to } 0.22$$

where:

F = facewidth $\quad$ ψ = helix angle

d = gear diameter $\quad$ L = bearing span (centre to centre)

The higher value applies more to speed reduction drives, whilst the lower value applies to speed increasing gears.

The following points should also be kept in mind when laying out and dimensioning case-hardened and ground single-helical gear trains:

Modern grinding machines made by Maag permit tooth flanks to be provided with a longitudinal correction defined by close tolerance limits. This correction is prescribed such as to give a uniform load distribution across the whole facewidth under a given load (usually the average operating load). Hence we can nowadays allow the elastic deflections of a pinion under load to attain the same values in a case-hardened gear unit as are conventionally accepted for through-hardened units.

For industrial and marine gears the design will allow for a total deflection (comprising bending and torsion of the pinion body) no greater than 6 to (8) tenthousandths of an inch. But it is worth noting that case-hardened gear drives for high specific loads have been successfully made, which are subjected to much greater deflections.

Naturally it is imperative in every case that the workshop technique of producing the necessary longitudinal correction is thoroughly mastered.

Comparing case-hardened gears with through-hardened gears, it should also be mentioned that the load distribution factor K_m for the former is favourably influenced by the following circumstances:

1) On account of the narrower facewidths in case-hardened drives, housing distortions - whatever their origin might be - impair the uniformity of load distribution to a much less extent.

2) Smaller gear dimensions also mean low pitch line velocities. In high-speed high-power gear drives, centrifugal forces cause elastic deformation in the gear body, and can affect the tooth bearing conditions. The small centre distance has in such cases an added significance.

3) Small gears can be built into smaller housings. The more compact the latter, the stiffer they can be built and the less will be the effect of external forces and thermal influences.

Broadly speaking, in drives with case-hardened and ground gears the single-helical gearing must definitely be given first preference. The main advantage of the double-helical gear, i.e. that it does not develop an axial thrust, is insufficient to compensate for its expensive, time-consuming manufacture.

Synchronizing Gear between Twin Diesel Engine and Generator

Operational experience has proved carburized and ground gears to be exceptionally resistant to shock loading, a property which makes them ideal for systems subject to heavy torsional vibrations - on diesel driven ships for example or Diesel generator plants.

One such combination, in which vibrations are really extreme, is shown in Fig. 2. The two crankshafts of a twin diesel engine are linked to a generator via a speed increasing gear drive. They are designed for installation in diesel-electric locomotives with a generator output of 2350 HP. The input and output shafts are fitted with rigid couplings. The extraordinary feature is the heavy vibrational torque which the gears must withstand. At standard load and standard speed, the vibrational torque is 300% of the trans-

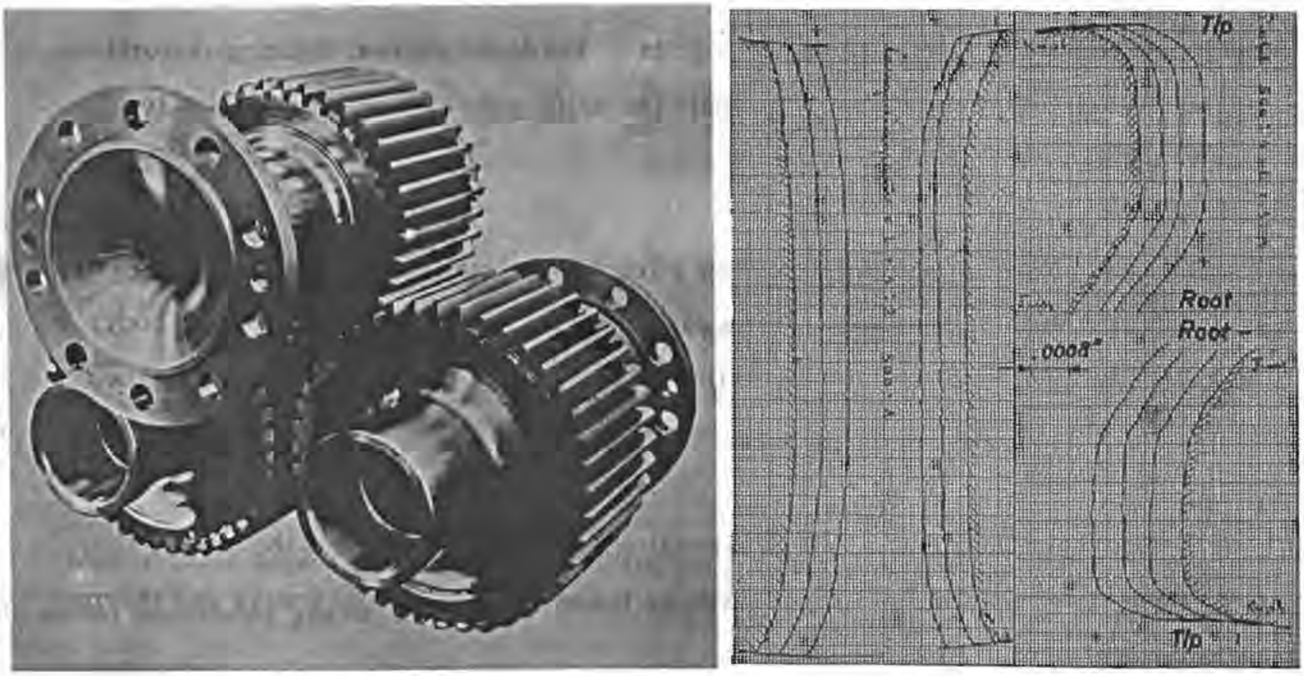


Fig. 2

Highly loaded Diesel locomotive gears with inspection records of tooth profile and lead

mitted torque, which means that the teeth are subjected to incessant shock loading up to 3500 lbs/in of facewidth. Naturally the backlash is kept to an absolute minimum, being 3 1/2 to 5 thousandth of an inch for the centre distance of 16 1/2 inches. It was decided to install case-hardened and ground spur gears. About 750 of these units have as yet been built and put into operation. There remains no doubt that under service condition of this kind the load limit is governed by the oil film resistance. When these gear units were still in the early stage of their development, "initial scoring" trouble was experienced. This was overcome by copper plating the pinion teeth and by maintaining narrow tolerances on tooth form including corrections. An SAE-30 oil with EP additives is used.

Marine Gear for Turbine Driven Vessels

In latter years one has been becoming more and more conscious that not only first costs, but also the size and weight of the propulsion unit in merchant ships are of some economic importance. The trend is towards shorter engine rooms and less weight. Case-hardened and ground gearing and the associated change-over to single-helical gears widen the scope in this direction.

Firstly we refer to Fig. 3 to comment on a few basic constructional features of single-helical marine gears. It concerns a tandem articulated drive - or a TA-type, for short.

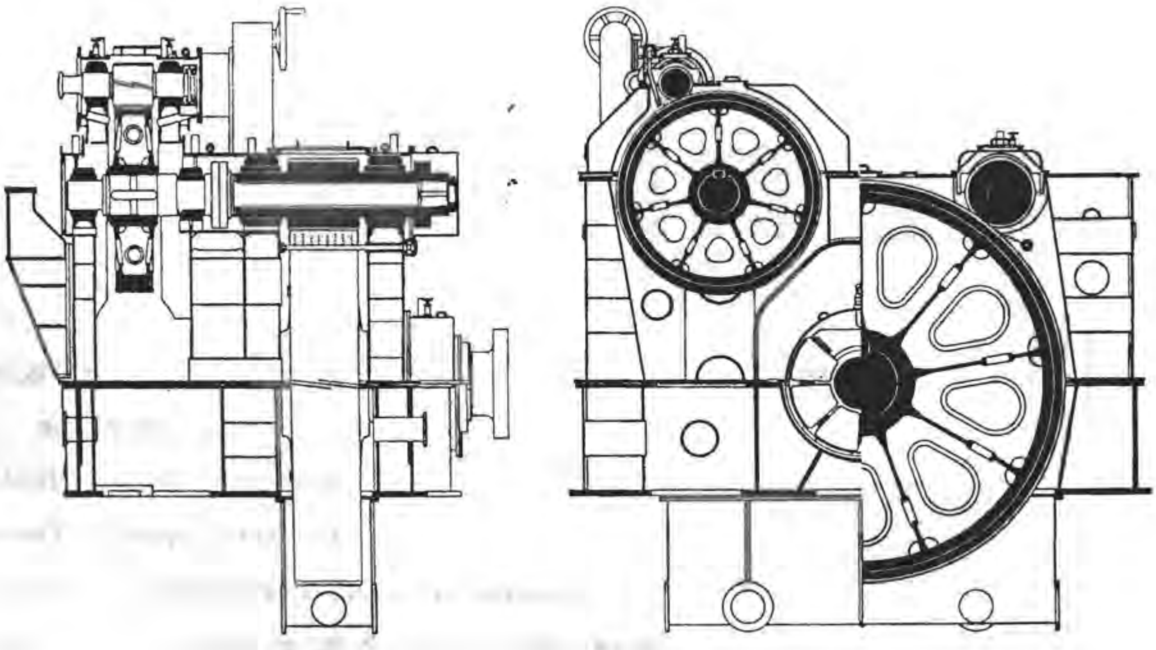


Fig. 3

Tandem articulated marine propulsion gear

The output gear is located axially against the propeller thrust bearing. The secondary pinion is connected via quill shaft and rigid couplings to the primary wheel. The axial thrust of the pinion and wheel are mutually opposed and thereby practically cancelled, permitting the intermediate shaft to be located axially by a simple collar bearing. Primary pinion thrust, however, is taken up by a tilting pad bearing of the Michell or Kingsbury type.

Now there is in fact a feasible method of dispensing with this primary pinion thrust bearing, a method which unfortunately has not yet been applied to large units. It consists in replacing the toothed coupling of the turbine with a quill shaft. Not only does the pinion thrust bearing become superfluous by this move, but the reaction in the turbine thrust bearing is notably reduced by partial absorption in the gear teeth.

Such a design is only advisable using a single-helical gear set, for only single-helicals possess the property of axial self-adjustment, and avoid with certainty the transmission of axial vibration from the prop-shaft to the turbine.

This construction has been applied numberously to turbine generator plants, as will be shown later in connection with epicyclic gears.

Dual Tandem Articulated Gear (DTA-Type)

Exactly the same basic construction is also possible for locked train drives as per Fig. 4. Here too, rigid couplings can be used to connect the quill shafts at both ends to the secondary pinion and the primary wheel, for the single-helical gears are axially self-adjusting. Such a construction is practically impossible with double-helical gears. Two mating double-helical gears are in mutual positive axial location. For this reason there must be a toothed coupling on at least one intermediate shaft, e.g. at point "A", Fig. 4, which can slide axially. The division of load between the toothing halves can nevertheless be impaired by indefinite frictional forces in a toothed coupling, which, after all, can attain appreciable values. If only for this reason, it is better to use single-helicals in locked train drives.

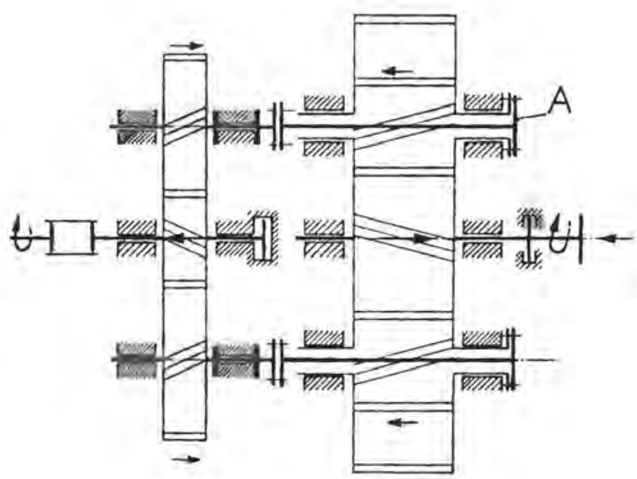


Fig. 4

Dual tandem articulated drive

Striding developments in tanker construction has of late boosted the demand for higher power at simultaneously lower speeds. We have worked out a project for a locked train drive to transmit 40000 HP at a favorable propeller speed of 90 rpm from a twin turbine input. For such high torque transmission, the locked train construction is used to great advantage. A gear drive of this type with case-hardened and ground pinions and through-hardened and ground wheel teeth, would weigh 110 tons. The wheel would have a diameter of 14 feet at a facewidth of 26 inches. From the general design point of view, therefore, the transmission of torque of this order does not present any serious difficulties.

Dual Tandem Self-Adjusting Gear (DTS-Type)

The property of the single-helical gear - that it finds its own axial location - paves the way for a unique type of locked train construction. The quill shafts can be discarded and the primary wheel coupled directly with the secondary pinion.

Thereby the number of intermediate shaft bearings is reduced from 4 to 2. In this manner the unit becomes shorter and lighter, suffers less bearing losses, and is cheaper to manufacture. Obviously a new approach is required to the problem of load division between the two intermediate shafts, for the quill shafts no longer exist. The solution is found by pivot mounting the primary pinion, so that it can float in a certain direction, a feature from which the name of this gear is derived:DTS-Type, standing for Dual Tandem Self-Adjusting.

The manner in which load distribution is established is best explained by Fig. 5. Consider first an epicyclic gear having three planets with fixed axes. The sun pinion, which has no bearings at all, finds its proper location as soon as torque is applied, and load is shared equally between the points of mesh. If this gear is altered by removing one of the planet shafts and replacing the internal toothing by a bull gear of external toothing, we obtain a locked train gear with no quill shafts. However, as shown by the lower sketch, the reaction force P of the removed planet must now be replaced by another force equal in value and direction. Such a force will hold the pinion in its proper location. This is done by adding two bearings to the pinion shaft. These bearings are seated in a lever which is allowed to pivot around a fixed point on the line of action of the reaction force P. By such means, the two tooth loads A and B always remain equal in size, irrespective of the torque applied; in other words the load is always equally divided between the two intermediate shafts.

A test gear of ample size as shown in Fig. 6 has proved the feasability of such a design. The two levers carrying the pinion bearings are rigidly connected with one another, and pivot about two hardened pins.

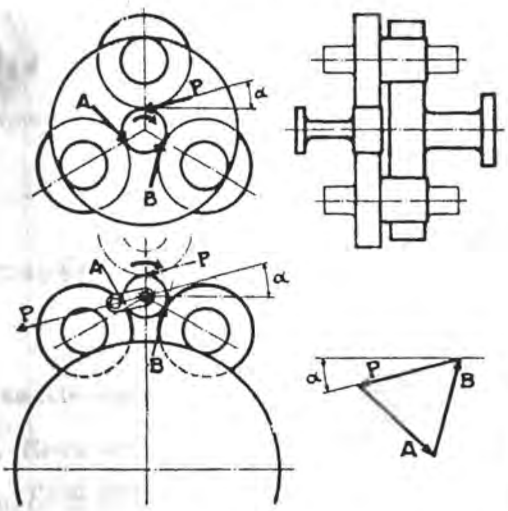


Fig. 5

Gear systems with self-adjusting primary pinion

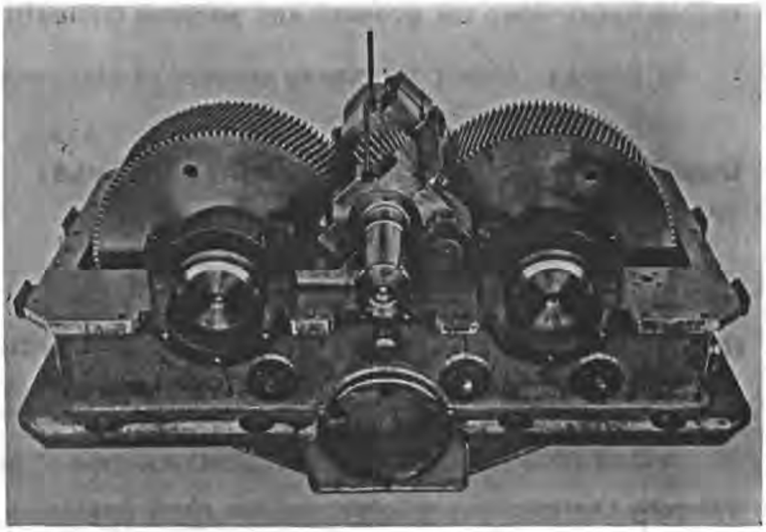
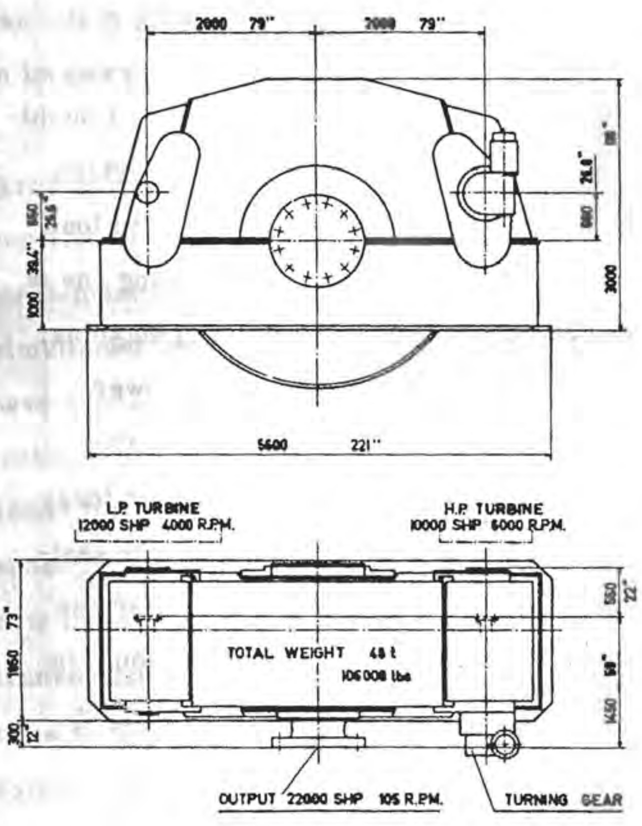


Fig. 6

Test gear of DTS-type



Drawing Fig. 7 shows the main dimensions of a DTS-type marine gear of 22000 SHP and 105 rpm at the screw shaft, designed to suit a low head engine room. All the pinions and the primary gears are case-hardened, and the bull gear has a rim of NiCrMo-steel of 320 - 360 BNH.

Fig. 7

Dimensional sketch of DTS-type marine gear. Output rating 22000 SHP at 105 rpm.

It is the extremely short length of this gear unit which should be noted. Because the first reduction is placed aft of the bull gear, the turbines can be placed very close to the gear, which still leaves adequate length for the tooth couplings or even a quill shaft. The overall length of the propulsion unit can thus be reduced to a minimum. The total weight of this gear unit is 48 long tons.

Case-hardened Gear in a Destroyer Power Package

Because of the very short length of case-hardened and ground marine gears, they are ideal for propulsion units of the kind built by the German manufacturer Messrs. Wahodag in Hamburg. They have designed a system where the reduction gear, condenser and turbines are firmly bolted together to form a rigid, selfsupporting structure.

Fig. 8 shows such a power pack which entered service about two years ago. This gear of Maag construction, however, is still of the conventional Dual Tandem Articulated design

Fig. 8

Destroyer power package with Maag DTA-type propulsion gear

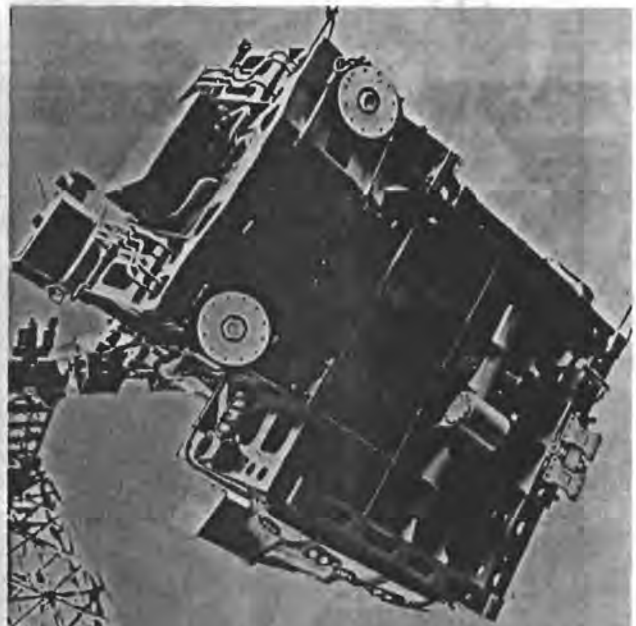


Fig. 9

Destroyer power package with three point support. View from below.

(DTA-Type), which means that quill shafts are fitted. The three turbines (cruising, HP and LP) are mounted on top of the condenser. These units can be put on board ship completely assembled. The output power is close to 40000 HP and the max. output speed is 357 rpm. The weight of the gear unit by itself is 26 long tons.

An interesting feature is the three point support. Fig. 9 gives a view from below. The two round flanges to the left are component parts of the two spherically shaped supports under the gear unit, one on either side of the bull gear. The third point of support is situated on the centre line at the forward end of the condenser. This support can slide to allow free heat expansion.

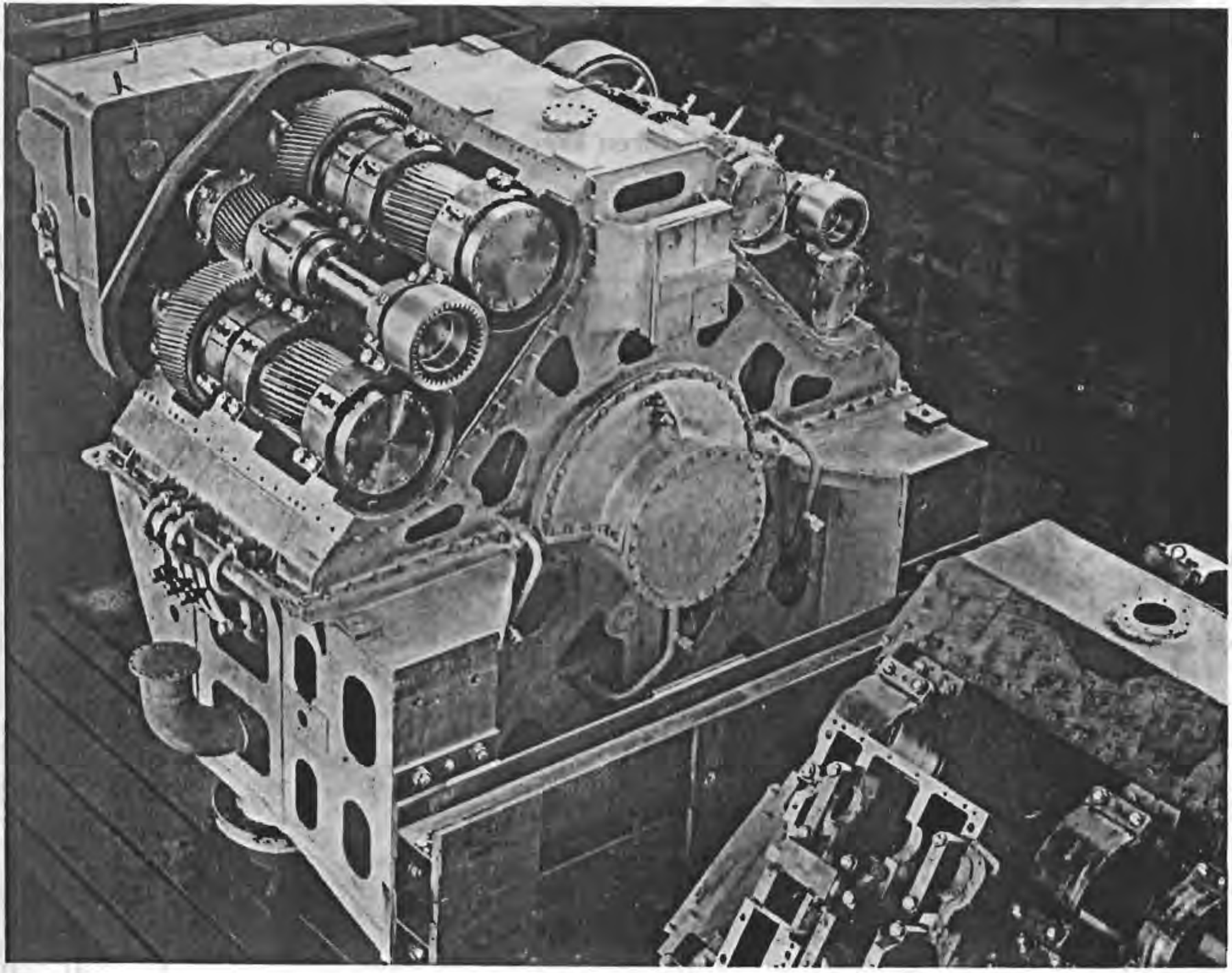


Fig. 10

Maag 2 DTA-240-type gear for destroyer power package. Top covers removed.

In such a case it is important to place the two gear end supports directly under the centre line of the bull gear. This will minimize gear case distortion due to load.

Only a very light foundation is needed for this three point supporting system, which is an important saving in weight. Service experience has been excellent. Although rather heavy hull vibrations occurred during the first trial, there has not been the slightest sign of the tooth contact being affected by gear case distortions.

Fig. 10 shows the reduction gear alone with covers removed. If this gear was changed to the DTS-type, its length could be reduced by more than 25% and its weight by about 15%. Fig. 11 shows the dimensions of a 40000 HP power package with DTS-gear. The overall length is 204 inches.

Studies have been made with a view to finding out whether this design would be suitable for merchant ship propulsion systems. The gear itself presents no problems and similar reductions in weight and gear length are possible.

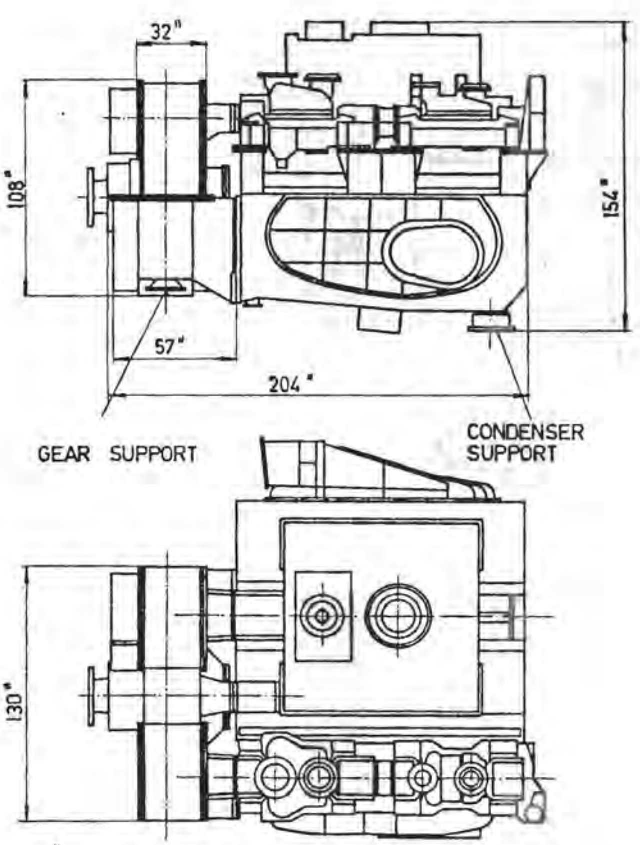


Fig. 11

Dimensional sketch of 40000 HP destroyer power package

Epicyclic Gears

Epicyclic gears have become more and more popular in recent years. Maag designs embody a new approach to the subject. The use of single helical gearing resulted in an extremely simple construction which makes assembly possible without the aid of a specialist. Fig. 12 shows the main features of this epicyclic gear:

- single helical toothing
- fixed annulus
- floating sun pinion
- pinion and planets case-hardened and ground
- annulus through-hardened and ground

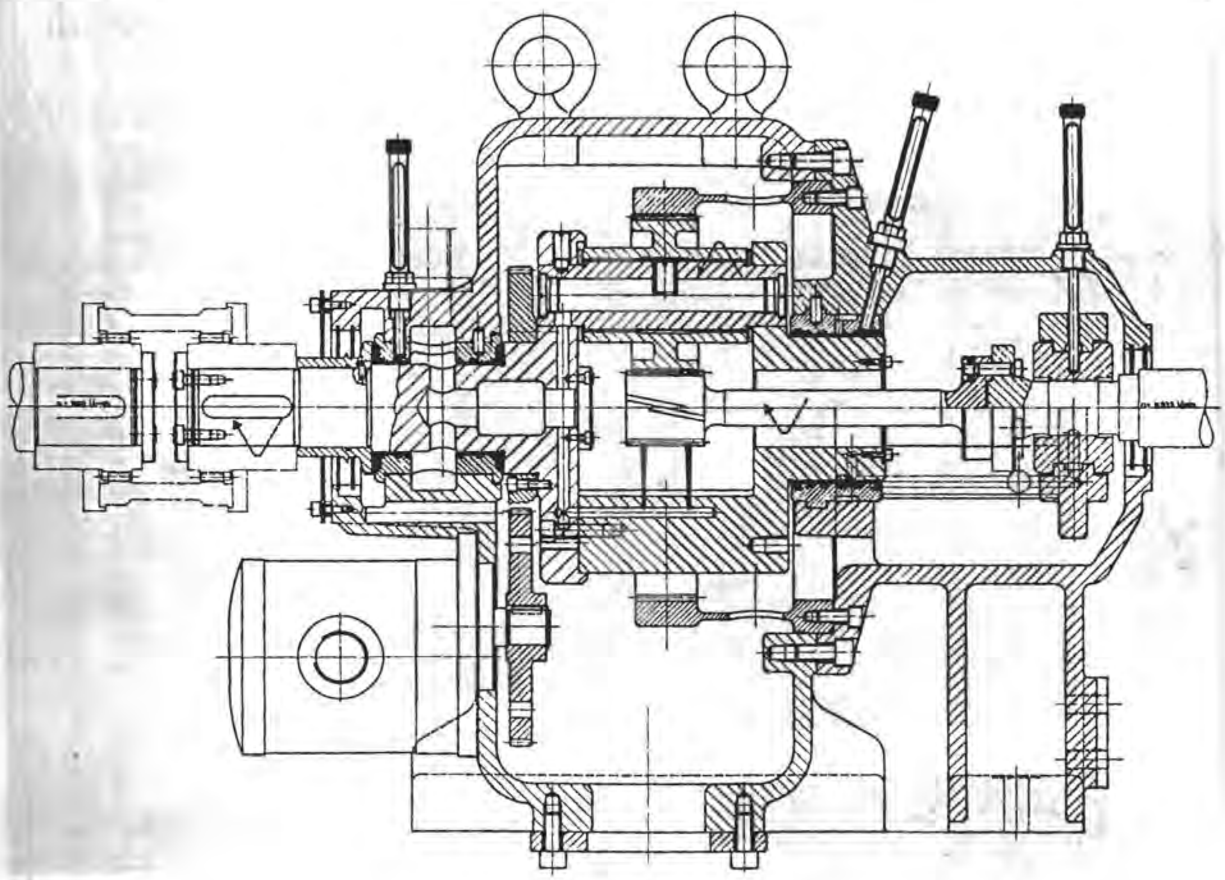


Fig. 12

Epicyclic gear for turbo-generator set incorporating turbine bearing

It was found that if toothings and planet carrier are carefully machined, no flexible mounting of the annulus is needed. For equal load distribution to all tooth engagements, it is adequate to allow the sun pinion to float.

This epicyclic gear is installed between steam turbine and generator. The thrust acting on the annulus is carried by the casing.

Since each planet is nothing more than an idler gear, the thrusts at its two points of mesh are opposed to each other, and therefore there is no thrust on the planet carrier and output shaft. The planets are held axially simply by light locating rings.

The sun pinion in this case is connected to the turbine shaft by rigid flange coupling. The thin flexible shaft between turbine coupling and sun pinion allows the pinion to centre itself in the planets as soon as torque is applied. Thrust from the sun pinion acts in the opposite direction to the turbine thrust. These two thrusts being almost equal, the turbine thrust bearing can be reduced in size considerably.

This, then, is a single helical gear which has no actual thrust bearing. More even - the thrust of the sun pinion is a virtue. It must be realized that with double helical toothing this design is not even possible.

Another interesting application of the single helical epicyclic gear is illustrated by Fig. 13. This is a Freon turbo-compressor unit. The pinion is integral with the compressor shaft. Only one bearing with thrust collars is arranged close to the compressor wheel. The other end of the compressor shaft is centered by the tooth loads at the three meshing points of the sun pinion. The pinion thrust of the single helical toothing opposes the thrust induced on the impeller, which results in a compressor thrust bearing much reduced in size. The gear is totally enclosed and is running in a Freon atmosphere. Bearing losses in this simple drive system are low; the production costs low and maintenance easy.

From these few examples we see that it is important for the gear designer to work in close contact with the manufacturer of driving and driven machinery. Only in this way can the best technical solutions be found - and these are usually the most economical at the same time.

When weighing up the manufacturing costs of through-hardened gears against those for case-hardened gears, it is well to remember the following points:

The costs of machining the teeth of single-helical case-hardened gears (cutting and grinding) average only about 14% of the total costs of the gear unit. Price comparisons must therefore take the total gear unit costs into consideration, the total drive system costs, in fact. Not only does case-hardening and grinding give a resistant tooth flank, but above all light, space-saving gear units in general.

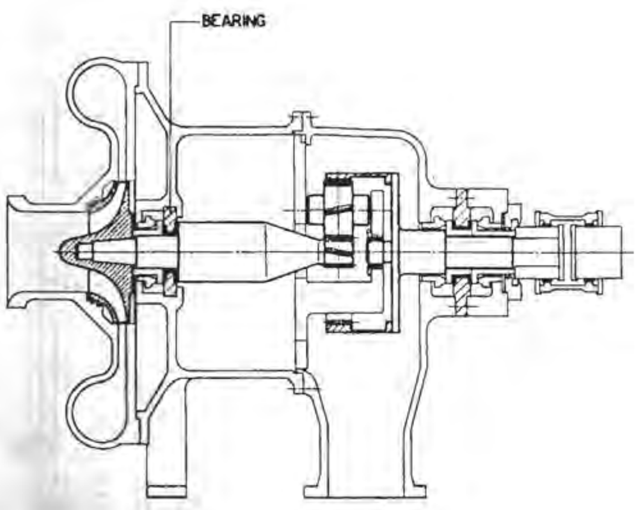


Fig. 13

Single-helical epicyclic gear in

Freon turbo compressor unit

Moreover, with respect to the entire drive system, single-helical, case-hardened gear units can often result in simplified layouts, which again cut overall costs.

Studied in this light, more and more facts reveal that the case-hardened and ground gear is the gateway to the most paying solutions in many of the diverse fields of gear transmission.

Nor should such costing surveys overlook the exceptionally long life potential of case-hardened and ground gears. Designed properly, wear will be practically non-existant.

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