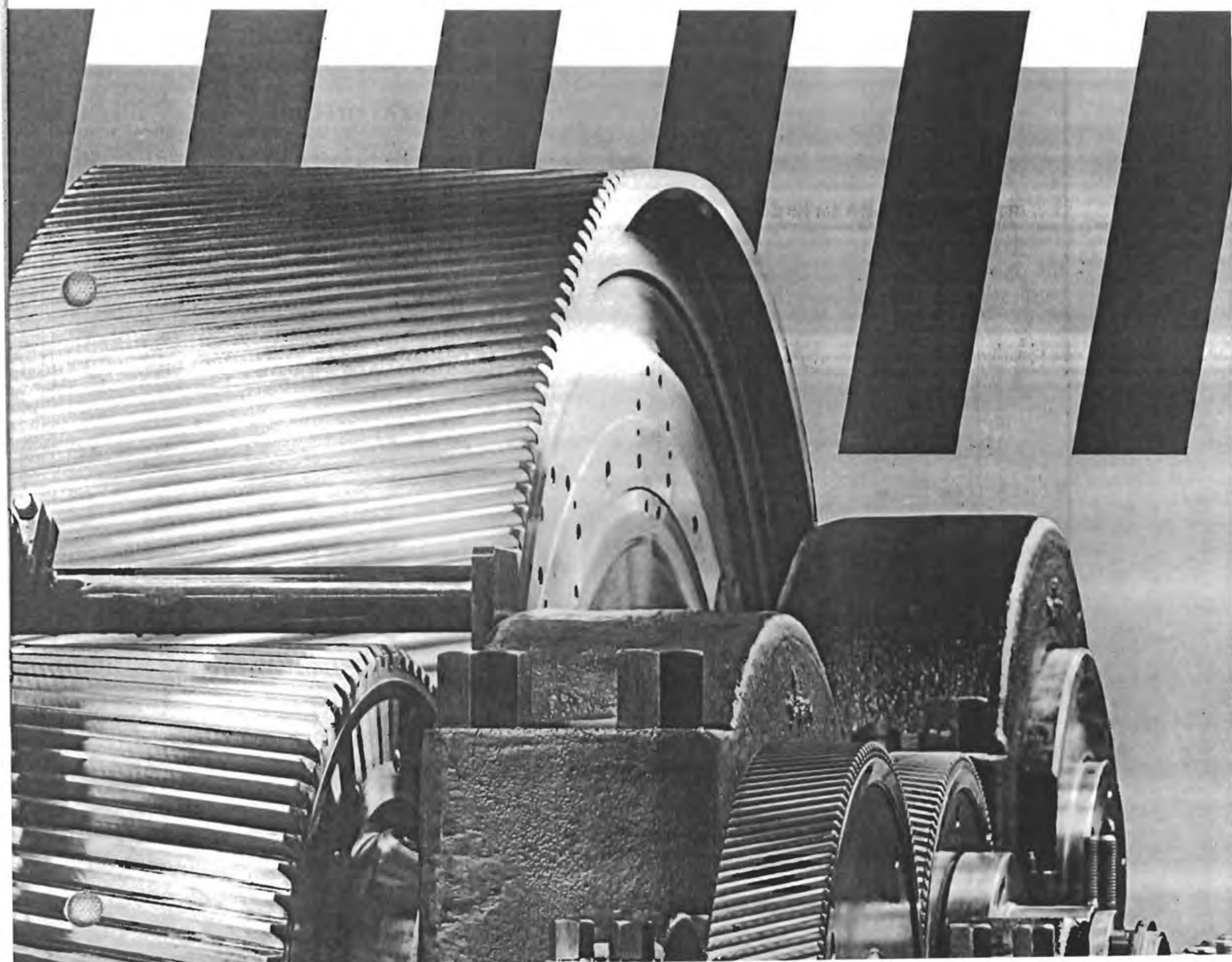
MAAG GEAR DESIGN PRINCIPLES

MAAG



MAAG gear design principles

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The advantages of hardened and ground gears were recognized already very early by the founder of our Company, Dr. h.c. Max Maag. The MAAG design philosophy has since centered on hardened and ground single helical or straight spur gears.

This paper is intended to illustrate the appreciable advantages of these design principles.

1. Hardening and grinding

1.1 Greater load capacity

The load capacity of gears is determined by the following three criteria:

- Bending stress
- Contact stress
- Carrying capacity of the oil film between the tooth flanks.

The bending stress at the root of the tooth is given by the following simplified formula:

$$\sigma_F = \mathrm{Const}_F \cdot \frac{F_t}{b \cdot m_n}$$

where σ_F = bending stress at the root of the tooth

Const_F = combination of all constant or approximately constant factors occurring in ISO or AGMA formulae.

F_t = tangential load at pitch circle

b = face width

m_n = normal module of gear teeth

The contact stress is given by the following, also simplified formula:

$$H = \mathrm{Const}_H \sqrt{\frac{F_t}{b} \cdot \frac{2a}{d_1 d_2}} = \mathrm{Const}_H \sqrt{\frac{F_t}{bd_1} \cdot \frac{u+1}{u}}$$

$$= \mathrm{Const}_H \sqrt{\frac{F_t}{b} \cdot \frac{(u+1)^2}{2au}}$$

Where $_H$ contact stress

$Const_H$ Combination of all constant or approximately constant factors in the ISO or AGMA formulae

F_t = face width

a = centre distance

d_1 = pinion diameter

d_2 = gear diameter

u = gear ratio ($\geqslant 1$)

The contact stress is substantially proportional to the root of the K factor, which according to Lloyd or API is

$$K = \frac{F_t}{bd_1} \cdot \frac{u+1}{u}$$

The carrying capacity of the oil film, for which a formula has been proposed in the new ISO standards (compiled with MAAG collaboration) is reduced as the pitch increases.

The permissible bending stress according to MAAG is

- for steels case hardened to HRC = 60: 500 N/mm^2
- for alloy steels through hardened to HB = 300: 300 N/mm^2

For otherwise identical geometrical conditions, case hardened gears can therefore carry 1.7 times the load of gears through hardened to 300 HB.

The permissible contact stress according to MAAG is

- for steels case hardened to HRC = 60: 1600 N/mm^2
- for alloy steels through hardened to HB = 300: 850 N/mm^2

The ratio of the permissible contact stresses is 1.88. As the permissible load however rises with the square of the permissible contact stress, case hardened gears can carry about 3.5 times as much load as gears through hardened to 300 HB under otherwise identical conditions.

In practice, case hardened gears are designed with somewhat coarser pitches than through hardened gears. If, for instance, the pitch is increased by 50 %, then the previously mentioned 1.7 ratio between the carrying capacities of case hardened gears and gears through hardened to 300 HB is increased to approximately 2.5. This still leaves adequate safety margin in respect of contact stress.

In general it may be assumed that the use of case hardened steels instead of through hardened steels increases the load carrying capacity of gears two- or threefold.

Important in this respect is that for through hardened gears the criterion determining the permissible load is the contact stress, whilst for case hardened gears the criterion is usually the bending stress.

These relationships are shown in Fig. 1. The permissible load per unit face width $\frac{F_t}{b}$ is plotted against the normal module.

Curve 1 Permissible load per unit face width determined by contact stress for a material through hardened to HB = 300

Curve 2 Permissible load per unit face width determined by contact stress for a material case hardened to HRC = 60

Curve 3 Permissible load per unit face width determined by bending stress at the root of the tooth for a material through hardened to HB = 300

Curve 4 Permissible load per unit face width determined by bending stress at the root of the tooth for a material case hardened to HRC = 60

Curve 5 Permissible load per unit face width determined by the carrying capacity of the oil film (not influenced by the choice of material).

The design point for gears through hardened to 300 HB is point A. The smallest possible module (pitch) is chosen for manufacturing reasons. The design point A is far removed from the scuffing limit (curve 5).

The design point for case hardened and ground gears could be situated at point B if it were only a matter of the bending and contact stresses. It must however be moved down to the scuffing limit at point C. In practice, point C is generally moved still somewhat further to the right along curve 5 with further increase of the module m_n to maintain an adequate safety margin where tooth fracture is concerned.

Case hardened gears designed in accordance with Fig. 1 have a great safety margin in respect of pitting, i.e.

in respect of the permissible contact stress.

With decreasing centre distance however, the contact stress increases, as can be seen from the formula for σ_H given previously. The pitting criterion for case hardened gears then becomes important. This applies primarily to aircraft, automotive and small industrial gears. Nowadays such gears are inconceivable without case hardening.

1.2 Smaller dimensions for the same duty

Increasing the load carrying capacity by a factor of 2 or 3 by the choice of case hardened materials logically affords the opportunity of correspondingly reducing the dimensions and weight of the gears for the same duty.

A marine reduction gear is shown in Fig. 2, on the left with hardened and ground gears and on the right with gears of conventional design.

The second reduction of a rolling mill drive transmitting 10,000 hp at 28 rpm is shown in Fig. 3.

The conventional double helical gears with a pinion hardened to 260 HB and a gear hardened to 230 HB requires a gear with a face width of 2 x 750 mm + 200 mm gap.

The single helical version with a pinion case hardened to 60 HRC and a gear through hardened to 340 HB enables the face width of the gear to be reduced to 800 mm with practically the same centre distance. The weight is practically halved from 91 t to 45 t.

The dimensions can be reduced still further by also employing a case hardened gear.

A further interesting example is shown in the photograph in Fig. 4. Here a pair of conventional double helical gears was replaced by a single helical, hardened and ground MAAG gear set, while retaining the original casing. The power tansmitted is 1000 kW at 10,080/1500 rpm. The centre distance is 360 mm and the original face width is 2 x 200 + 55 = 455 mm. The face width of the MAAG gear set is only 160 mm. Of particular interest in this example is the reduction in pinion bearing span, which can be seen clearly in the photograph.

1.3 Special advantages

Apart from the appreciable advantage of either being able to transmit very much more power with gears of the same dimensions or the same power with much smaller gears and of correspondingly less weight, hardened and ground gears also offer the following special advantages.

- Greater stiffness of the gearbox casing

The smaller dimensions enable appreciably stiffer gearbox casings to be designed without excessive increase in weight. This has a beneficial effect on the noise level and the vibration characteristics of the transmission.

- Reduction of rotor deflection

Each rotor, especially the pinion, deflects under tooth and torsional loads, as shown in Fig. 5. For the same load, this deflection becomes smaller as the span is reduced. It then becomes easier to calculate the deflection in advance and to provide compensation by appropriate longitudinal modifications during grinding.

- Lower peripheral speeds

Smaller dimensions result in lower peripheral speeds for the same speeds of rotation. Lower peripheral speeds in turn result in lower windage losses and therefore smaller temperature rises in the gearbox rotors.

Lower peripheral speeds also mean lower centrifugal forces in the rotors.

Smaller temperature rises and lower centrifugal forces reduce rotor deflection and therefore lead to better tooth bearing patterns and hence to higher load capacity.

- Higher permissible contact stress, particularly for small centre distances

As already mentioned in section 1.1, the contact stress tends to increase as the centre distance decreases. Nowadays it is therefore essential to harden small high performance gears and to grind them also for particularly heavy duty application. Gears for aircraft, helicopters, cars, trucks and construction plant are nowadays inconceivable without case hardening.

2. Single helical gears

Conventional transmissions very frequently employ double helical gears, which at one time offered certain advantages with the production methods then current.

Max Maag already advocated the use of single helical gears and pointed the way to their manufacture.

2.1 Advantages

- Simpler manufacture

Each gear has only one set of teeth instead of two. This is appreciably simpler for manufacture, particularly for hardening and grinding.

- Adjustment of tooth bearing pattern is possible

With single helical gears, an accurate tooth bearing pattern can be achieved very easily by deformation of the gearbox casing or by adjustable bearings.

In the case of double helical gears this is appreciably more difficult, as the dimensions are larger. If the helix angles of the four sets of meshing teeth are not exactly correct, an exact adjustment of the tooth bearing pattern is not possible at all and at best it is only possible to minimise the error by a compromise.

- Compact construction

Single helical gears do not require a central gap. This makes them more compact with all the consequent advantages.

- Profile modifications during grinding

The grinding of profile modifications for compensating deflections under load, centrifugal forces and temperature variations is simpler than on double helical gears.

- No axial shuttle

Double helical gears have a tendency to produce an axial shuttle if the index variation (cumulative pitch error) on both halves of the gear does not vary in the same manner. This tendency is absent on single helical gears.

Single helical gears are also insensitive to externally excited axial vibrations.

2.2 Special characteristics

- Tilting moment

On double helical gears the axial thrusts theoretically cancel themselves out. This is not the case on single helical gears. This can result in poor tooth bearing patterns where there are large diameter gears combined with small bearing spans. In such cases the use of pinion thrust collars, which will be discussed subsequently, is recommended. Tilting moments are insignificant where there are large face width to diameter ratios.

- Axial thrust

The axial thrust resulting from single helical gears is in itself no disadvantage. It even can be exploited as an advantage by special arrangements of driving machine - gearbox - driven machine. If an axial thrust is undesirable or not permissible, then the design utilizing pinion thrust collars is recommended.

These thrust collars not only eliminate all tilting moments, but also make the provision of thrust bearings on the pinion and gear shafts capable of sustaining the entire thrust unnecessary and so lead to improved efficiency.

The pinion thrust collars also make it possible to transmit axial thrust from one gearbox shaft to the other. With double helical gears this is only permissible to a very limited extent.

2.3 Examples of hardened and ground single helical gears

A dual tandem articulated marine reduction gear is shown in Fig. 6. The axial thrust of the large gear acts in opposition to the propeller thrust.

The thrusts from the second reduction pinion and the first reduction gear are of equal magnitude and opposed and therefore cancel themselves out. The idler shaft can therefore be coupled rigidly to a flexible shaft. A toothed coupling is not necessary. A simple tapered land collar bearing is adequate as thrust bearing.

The thrust of the first reduction pinion can act in opposition to the turbine thrust. The use of a flexible shaft allows the use of a common thrust bearing.

A gearbox which can be arranged for transmitting either 40 MW at 4500/3600 rpm or 35 MW at 4500/3000 rpm between a gas turbine and a generator is shown in Fig. 7.

The pinion is equipped with thrust collars to take the thrust from the single helical teeth.

The generator is connected axially to the gear by a flexible shaft.

The connection between the pinion and the turbine is also by flexible shaft, so that the entire installation only requires a single thrust bearing, namely that of the gas turbine. This improves the efficiency.

A type GB-47 speed increasing gearbox between a gas turbine and compressor for transmitting 24 MW at 4670/10130 rpm is shown in Fig. 8.

The gearbox is connected to the turbine and compressor by toothed couplings.

Tilting pad thrust bearings take the thrusts from the shafts.

The type GO-98 gearbox shown in Fig. 9 is also a step-up drive and transmits 31,500 hp at 1200/5830 rpm. The power is provided by a synchronous motor in this case. A toothed coupling is fitted to both gearbox shafts.

The thrust on the high speed pinion shaft is taken by a tilting pad thrust bearing, while a simple taper land collar bearing suffices on the slow speed gear shaft.

3. Spur gears

Spur gears have become very fashionable again recently. They are very simple to produce and do not result in any axial thrust.

3.1 Spur gears in slow speed heavy duty planetary gearboxes

MAAG have developed a special, spur gear planetary gearbox design for the use in the cement industry, i.e. for cement mill and cement kiln drives.

A type CPU-30S double reduction gearbox of this type is shown in Fig. 10. The power transmitted is 3200 kW. The speed ratio is 500/16.2 rpm.

The high transmission ratio requires a double reduction design.

Particularly noteworthy are the fabricated construction of planet carriers and gearbox casing and the mounting of the planet wheels on bearing metal coated pins integral with the planet carrier. The gearbox is appreciably smaller and lighter than a corresponding parallel shaft gearbox.

3.2 Spur gears in marine propulsion

A similar design of gearbox is also supplied by MAAG for marine propulsion for use with medium speed Diesel engines.

Two such marine gearboxes mounted on a back-to-back test rig are shown in Fig. 11.

The power transmitted is 13,200 kW and the speed ratio is 400/122 rpm. This speed ratio of approximately 4:1 calls for a design with stationary annular gear and rotating planet carrier. For smaller speed ratios, such as encountered with slower speed engines, the design of planetary gearboxes with stationary planet carrier and rotating annular gear is more advantageous. Such a planetary gearbox allows true power splitting and is therefore appreciably more compact than a corresponding parallel shaft gearbox.

3.3 The use of spur gears in high speed gearboxes

A gas turbine double reduction gear with spur gears is shown in Fig. 12. A deep tooth profile was chosen in the interest of achieving a high contact ratio. The power transmitted is 1700 hp and the speed ratio is 17100/1208 rpm.

A planetary splitter gearbox with "Flexpin" flexible planet wheel pins built under licence from Vickers Shipbuilders Limited, Barrow-in-Furness, England, is shown in Fig. 13. The Flexpin arrangement makes exact load sharing between the individual planet wheels, which may number 5 or more, possible. The power transmitted is 4050 kW with speed ratios of 12020/17412/1500 rpm.

A single reduction planetary gearbox with Flexpin arrangement and spur gears is shown in Fig. 14.

The power transmitted is 4000 kW and the speed ratio is 13471/5271 rpm.

4. Summary

MAAG advocate the building of gearboxes with case hardened and ground gears, which, in contrast with conventional through hardened gears have the advantage of more compact dimensions, smaller peripheral speeds and lower cost. Other features are greater stiffness of the casing and smaller deflections of the gearbox rotors.

The use of case hardening and grinding logically leads to the design of gearboxes with single helical or spur gears, which, compared with conventional double helical gears, have the advantage of simpler manufacture as well as numerous other design and functional advantages discussed in detail in this paper, namely:

- Facility for tooth bearing pattern adjustment
- Compact construction
- Simple grinding of longitudinal and profile modifications
- Elimination of axial vibrations
- Capability of transmitting axial thrusts
- Exploitation of the axial thrust
- Use of pinion thrust collars
- Simpler design of planetary gearboxes.

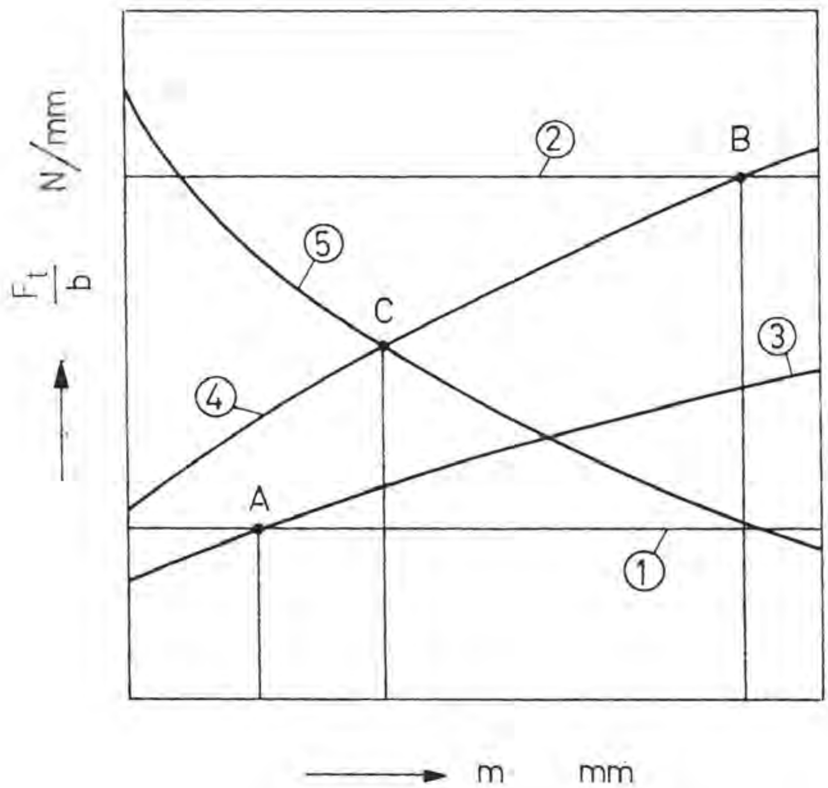

Fig. 1 Criteria for gear design: Tangential pitch line load per unit face width in relation to module.

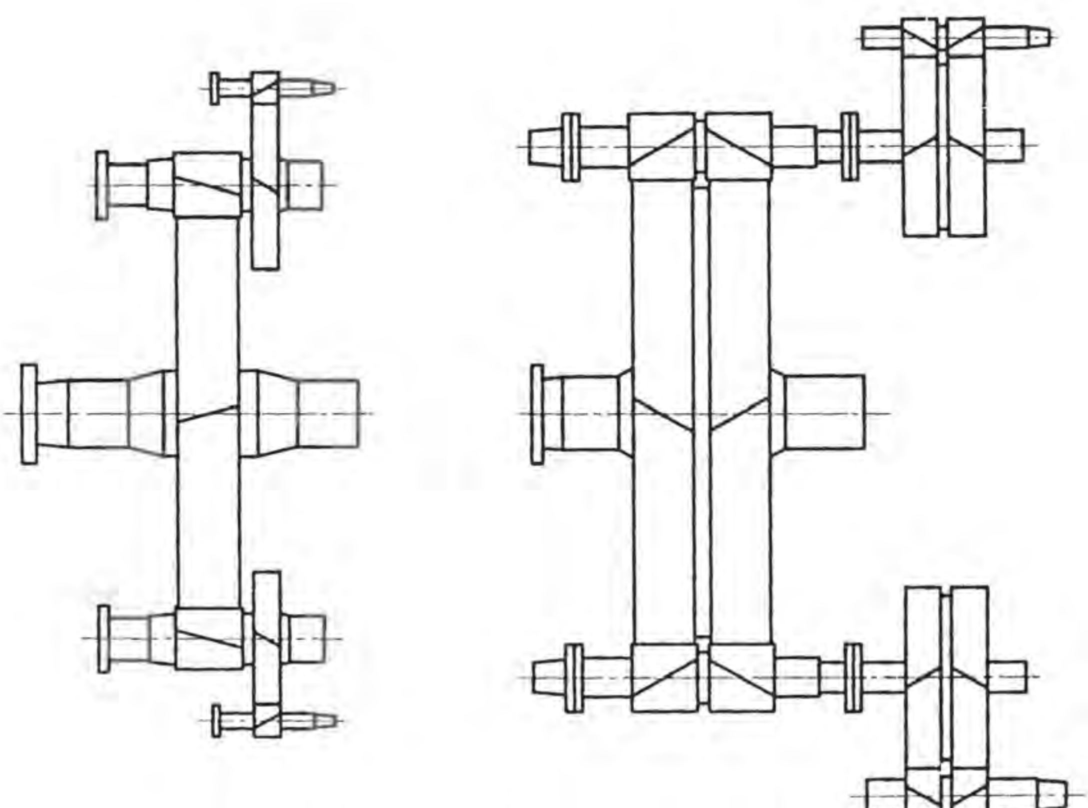

Fig. 2 Marine reduction gears for transmitting the same power, left: hardened and ground;

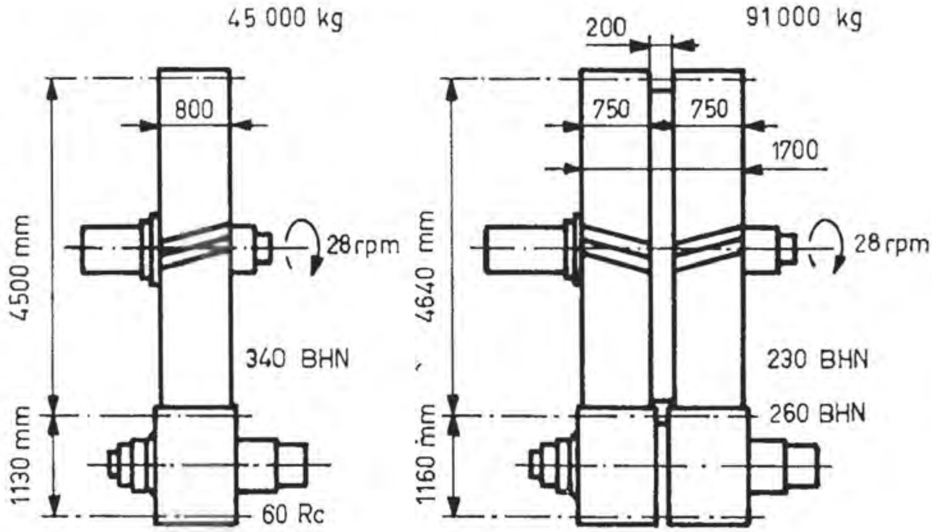

Fig. 3 Second stage of a rolling mill reduction gear, left hardened and ground; right unhardened

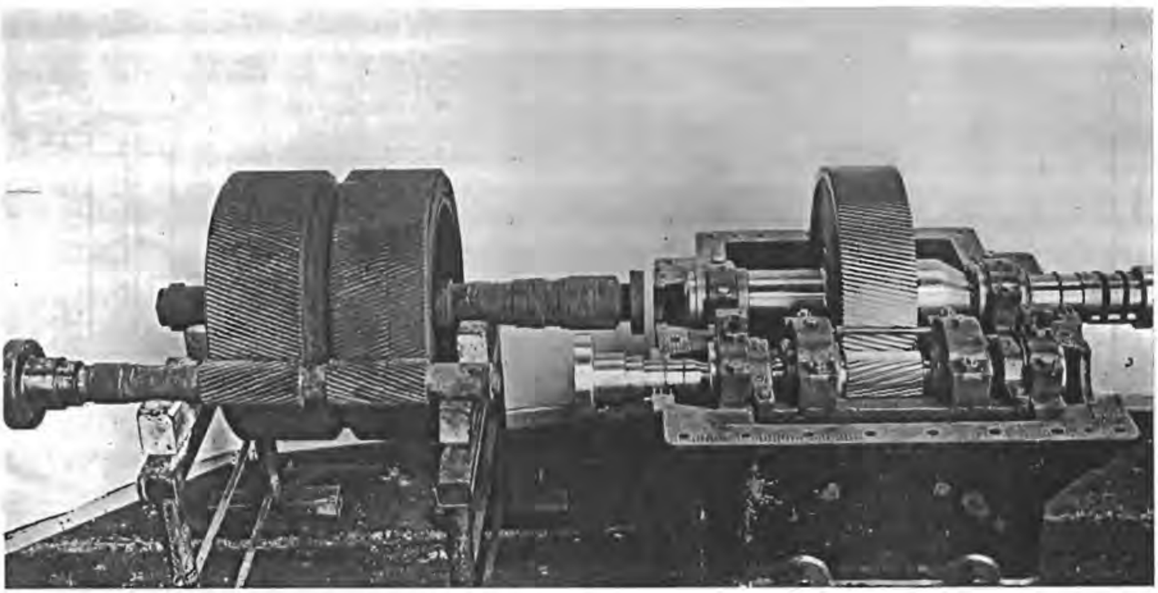

Fig. 4 Replacement of a set of double helical gears by single helical gears in the same casing. 1000 kW at 10080/1500 rpm.

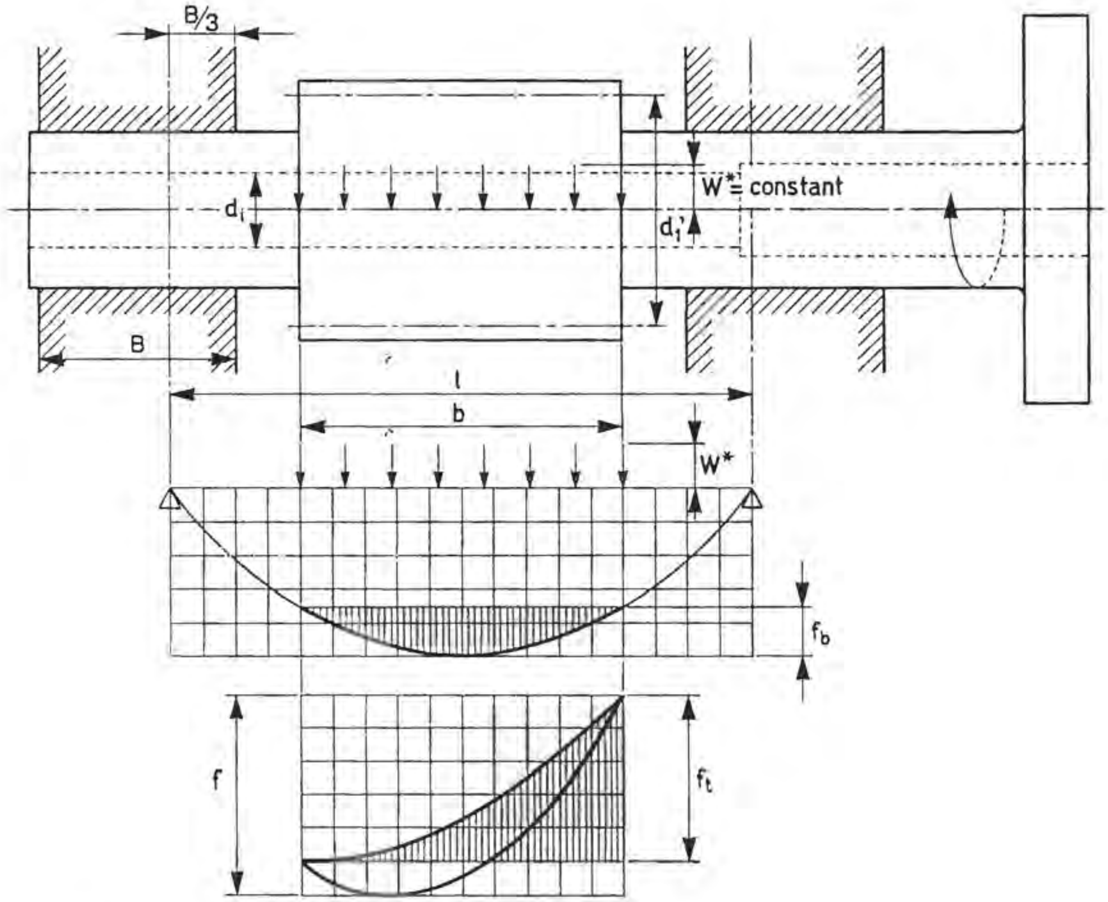

Fig. 5 Elastic deflection of pinion

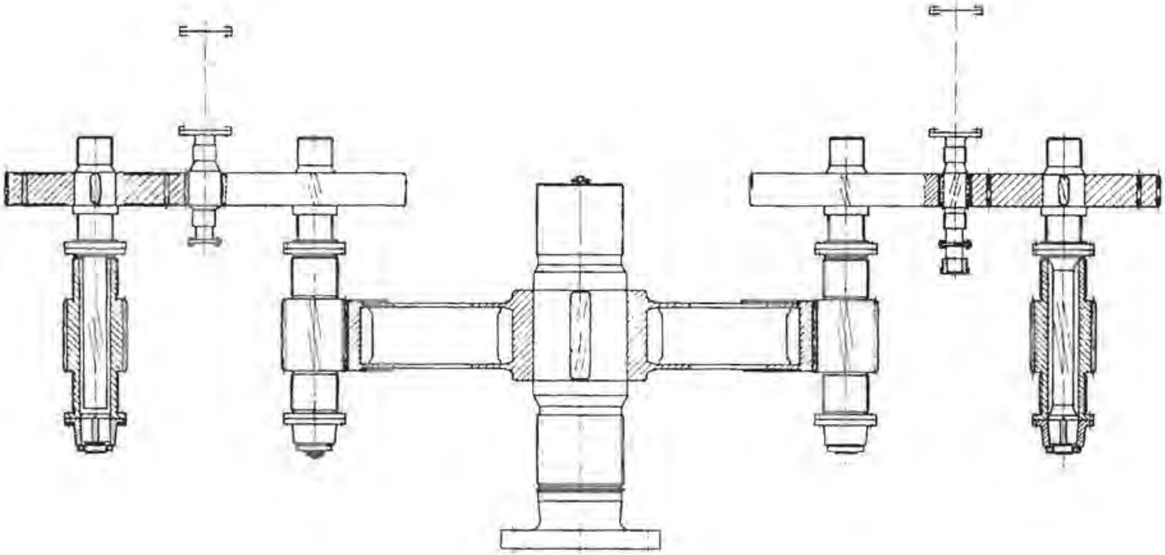

Fig. 6 Marine reduction gear of the dual tandem articulated type

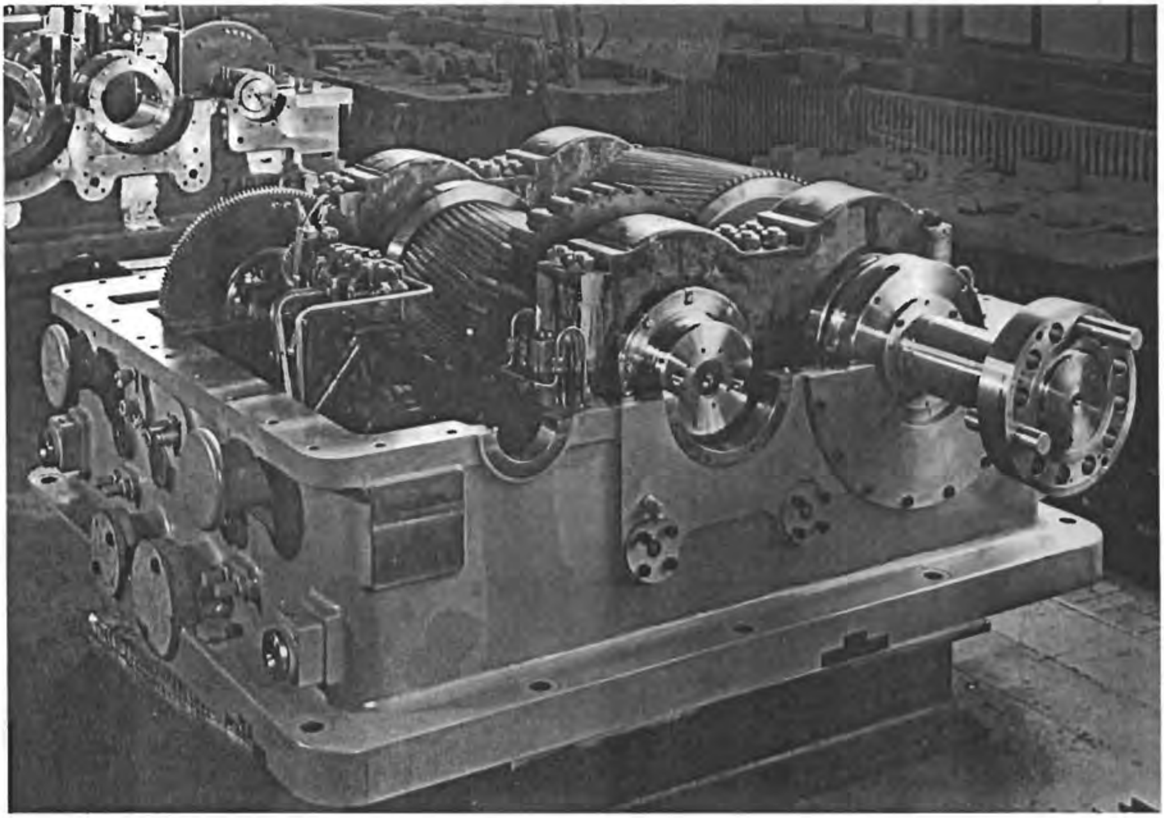

Fig. 7 Gearbox for a gas turbine-generator drive
40 MW, 4500/3600 rpm or 35 MW, 4500/3000 rpm.

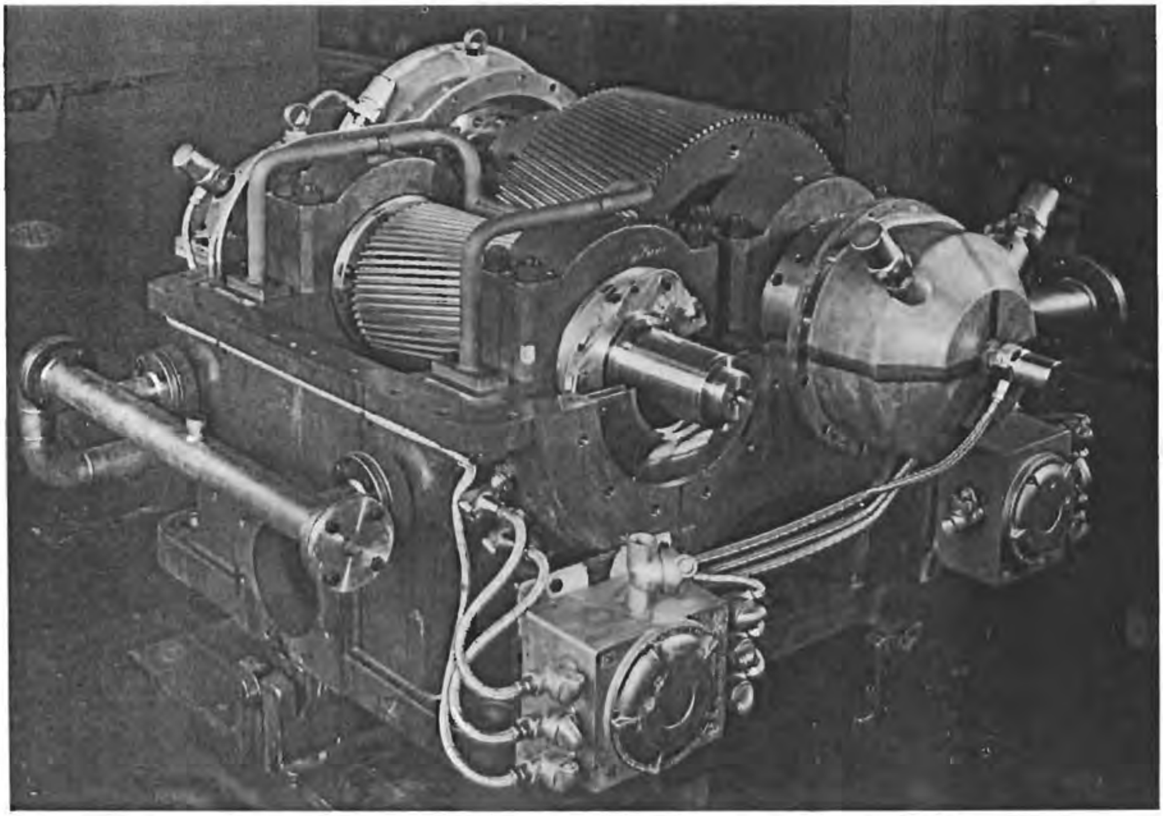

Fig. 8 Speed increasing gearbox between gas turbine and

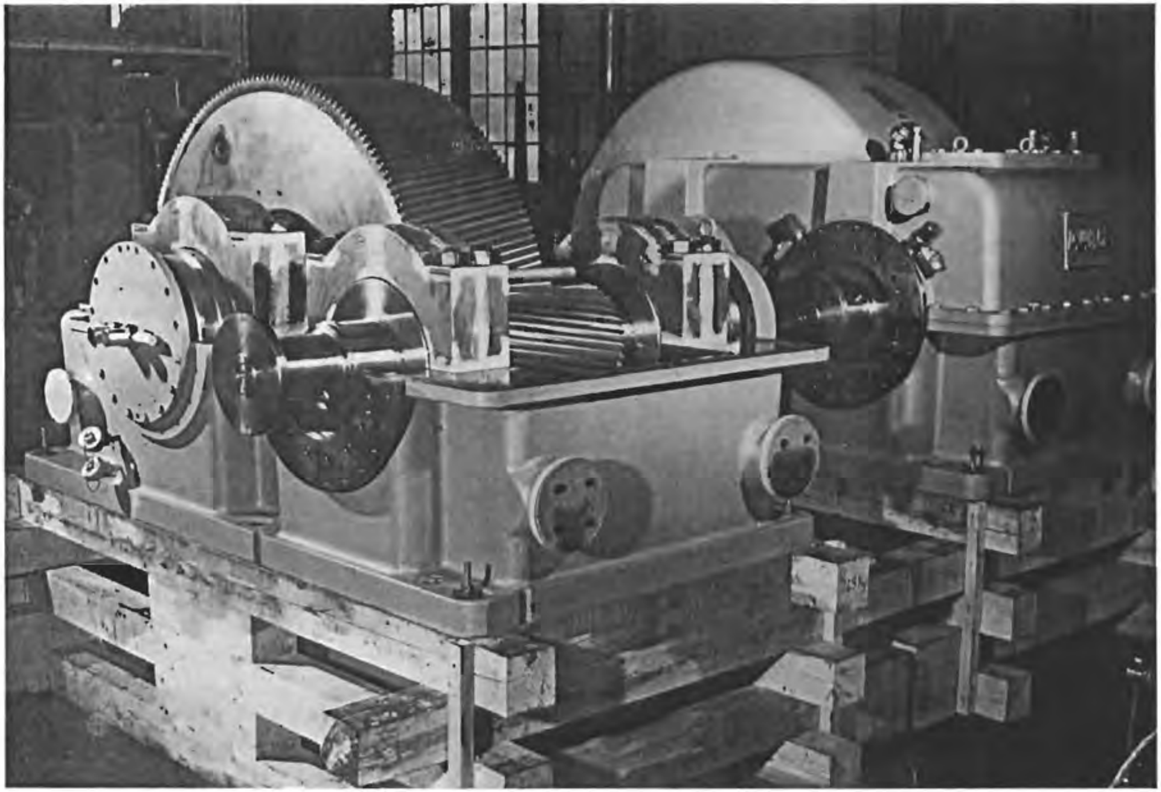

Fig. 9 Speed increasing gearbox between synchronous motor and compressor. 31500 hp, 1200/5830 rpm.

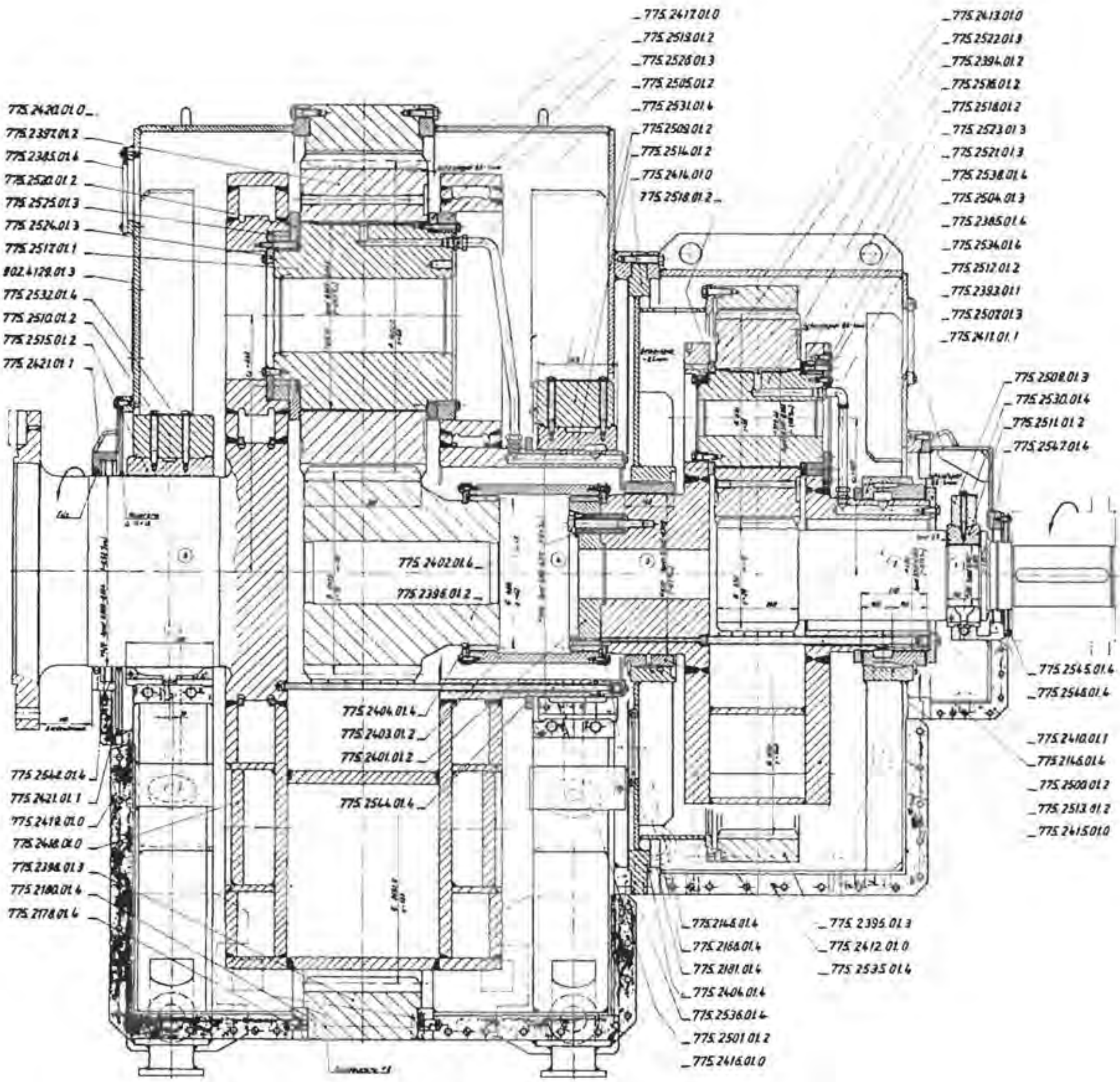

Fig. 10 Type CPU-30S heavy duty planetary gearbox.
3200 kW, 500/16.2 rpm.

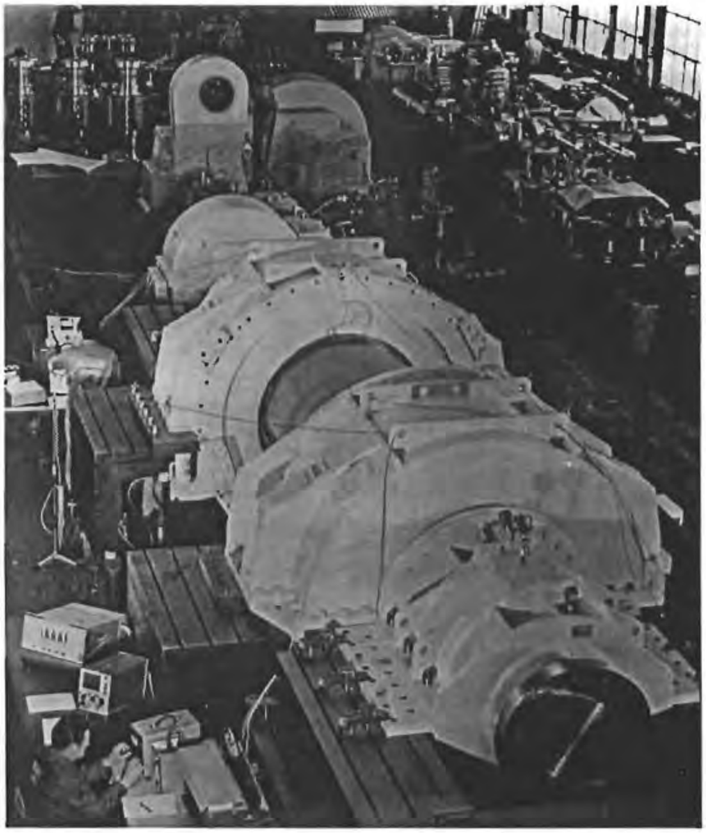

Fig. 11 Marine planetary gearbox on test rig.
13,200 kW, 400/122 rpm.

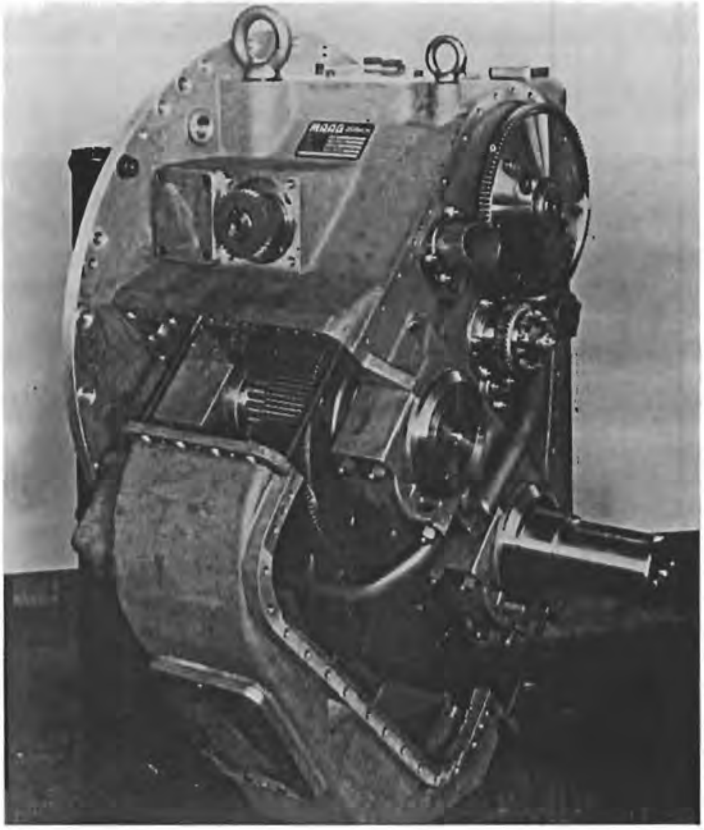

Fig. 12 Gas turbine reduction gearbox

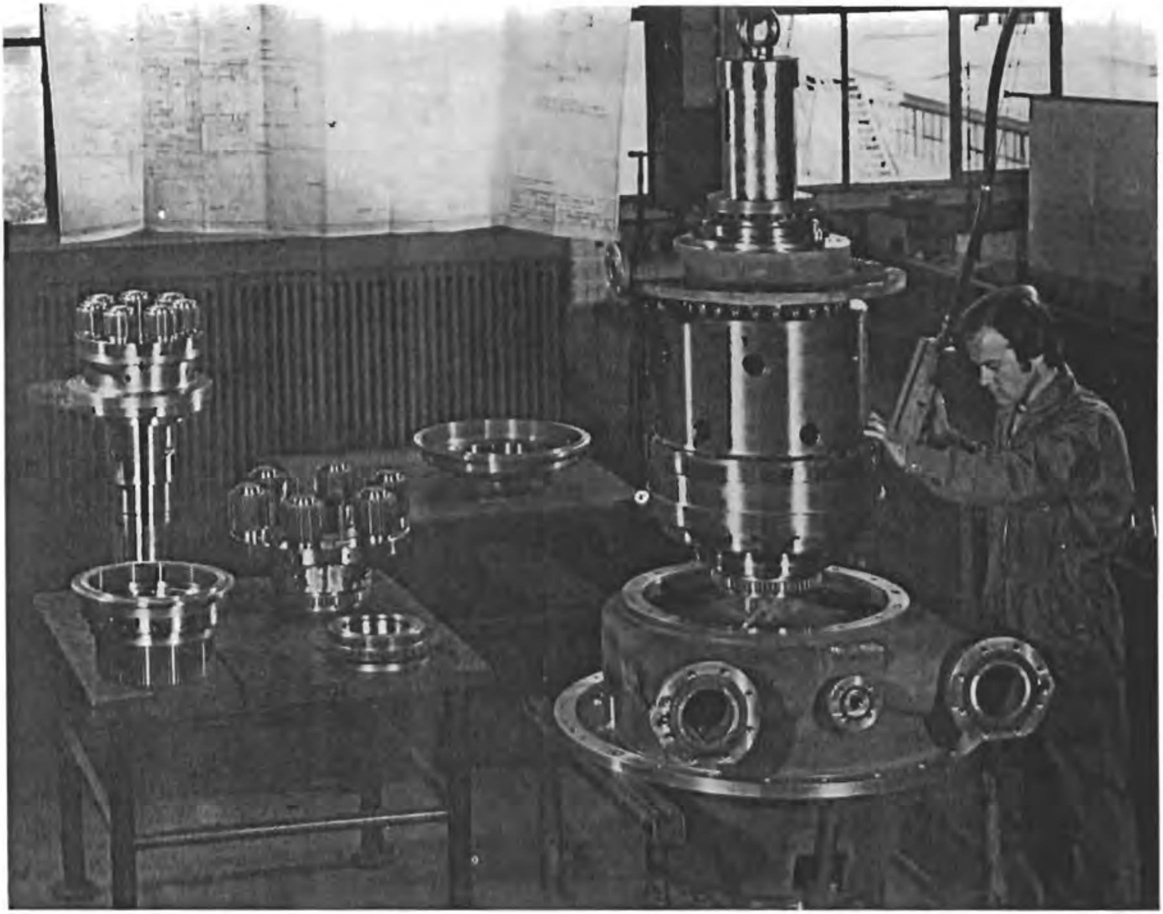

Fig. 13 Splitter planetary gearbox
gas turbine - compressor - generator
4050 kW, 12,020/17,412/1500 rpm.

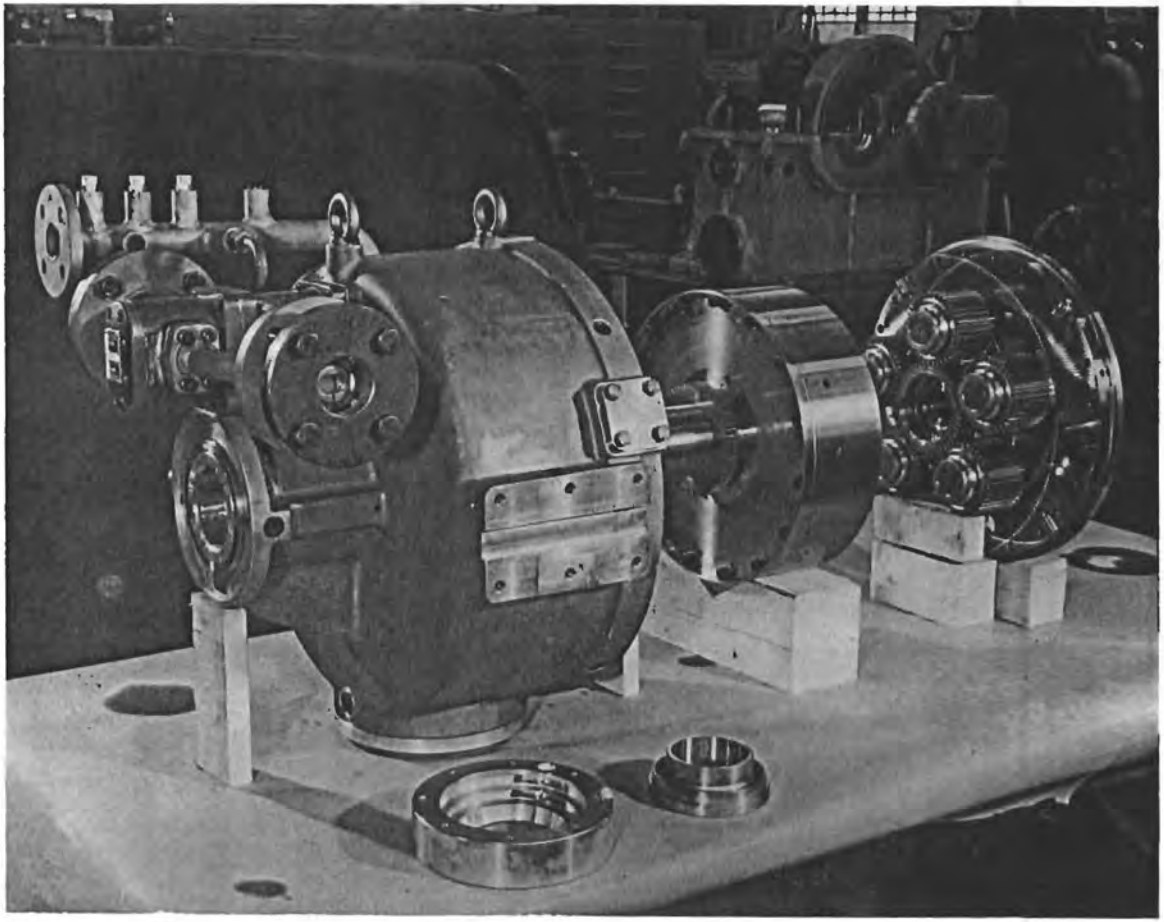

Fig. 14 Single reduction planetary gearbox
5420 hp, 13,471/5,271 rpm.