

Marine gearing—development and technology

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The first part of this paper deals with the development of the surface-hardened and ground marine gear. The accuracy which can be attained today in finish machining of such gears is discussed. Modern gear grinders can now produce two- and three-dimensional tooth corrections of any shape deemed necessary.

The second part describes the more sophisticated combined marine propulsion gear system CODAG and a reverse gear for high-power transmission. Rules for the safe operation of such systems are laid down and the required synchronizing clutches are described.

1 HISTORICAL BACKGROUND

Carburized and hardened gears could be manufactured with a reasonable accuracy only after tooth grinding machines became available. It was in 1919 when the engineer Max Maag designed and built his first grinder and put it into operation. The first orders for hardened gear rotors came from manufacturers of turbomachinery. The high load capacity and shock resistance of such toothings were demonstrated in locomotives.

The first propulsion gear was installed in a diesel-powered ship in 1933. This hardened and ground gear had spur teeth with a face width of one metre!

It was not until after World War II that this type of gear started to draw keen interest in the marine industry. Prototype gears were installed in steam-driven destroyers and frigates of a number of countries. The United Kingdom, France, Holland, the United States and Canada were forerunners, Germany, Italy and India followed a little later. Simultaneously, grinding machines were delivered to the first group mentioned above.

The Royal Navy and PAMETRADA* started an extensive research programme. Load tests were run with full-size marine gears, with conventional hobbled and shaved gears and with the new hardened and ground versions.

The results of this big effort were published at the famous Gearing Conference in London, 1958 (1, 2). They showed clearly that the load capacity of the case carburized, hardened and ground toothing was superior to other techniques.

A special series of test runs demonstrated the importance of tooth corrections. Properly designed and manufactured the load could be increased considerably before the first signs of distress showed.

A few years earlier the lesson about tooth correction had to be learned in a different way. A set of marine gears with true, uncorrected involutes was delivered to the United States. On the test bed the teeth promptly scuffed when only at partial load and the tips had to be hand stoned (tip relief) repeatedly to overcome this problem (3).

* Parsons and Marine Engineering Turbine Research and Development Association.

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A fair number of propulsion gears were built for steam-driven ships, tankers, freighters and passenger liners. These gears had carburized pinions and the gears were made of high-tensile steel. The high load capacity of this combination resulted in small face width, about half that of shaved gears. This promoted the single helical design which was less costly to grind. However, this design was new in merchant shipping. Most ship owners were sceptical. There was a certain scepticism with the Classification Societies as well; the gears had to be oversized. The conventional hobbled and shaved gear dominated the field up to the time when the large bore diesel put a rather abrupt end to steam propulsion.

2 GROUND GEARS FOR NAVAL CRAFT

On naval ships the gas turbine replaced steam propulsion, often in combination with high-speed diesel engines.

With modern ships machinery noise became another important factor. Today, marine reduction gears must be

- Reliable and durable
- Silent
- Accessible (easy to maintain)
- Small in size and weight

in this order of importance. The cost factor can be placed anywhere. Still, in the final stage of a contract it is often the all-important one.

The surface-hardened gear can meet these requirements best. But in the early days the noise level was relatively high. In the 1950s the airborne noise of the Leander Class gearboxes (4) was in the order of 110 dB at full power. Cyclic machine errors and pitch errors were the main causes.

The struggle for lower noise and improved reliability continues until today. Some measures taken to improve the situation are described in the following sections. However, it is not possible to give figures in decibels for each individual measure. Usually too many parameters change from one case to another.

2.1 Tooth geometry

In search of quieter gears a change was made from the standard 20° pressure angle basic rack tooth profile to

what is called a 'deep' tooth form. The first encouraging results were achieved around 1960 with propulsion gears for tankers. Today Maag has standardized this tooth form and applies it to all naval propulsion gears. It has a pressure angle of 17.5° and the addendum is $1.25 \times$ module. In a tooth mesh the tooth contact ratio is larger than 2. The meshing of such toothing is smoother and quieter. Laboratory tests carried out in recent years confirmed this (5).

The deep tooth has a rather slender appearance. However, load tests showed even a slightly higher tooth strength than for the 20° tooth. The bigger leverage is more than compensated for by the larger contact ratio (6).

2.2 Gear design

For turbine propulsion the locked-train gear is favoured because of its small size and weight and good efficiency. The splitting of the torque into two power paths results in smaller specific loads at the tooth meshes which helps to reduce gear noise.

Apart from excellent durability, surface-hardened teeth have a greater strength than through-hardened teeth. Consequently, the tooth size can be made smaller and the tooth meshing condition (overlap ratio) is improved which again lowers the noise level.

Single helical teeth in combination with thrust collars are recommended. The manufacture of a highly accurate toothing with proper tooth corrections is easier with single helicals, and the tilting effect due to the axial tooth load is compensated for by these collars.

These design parameters for low noise levels have evolved from experience over the years and from direct comparisons with conventional double helical, shaved gears:

- Locked-train design
- Single helical, hardened and ground teeth
- Thrust collars
- Diaphragm couplings between first reduction gear and quill shafts
- Thrust bearing at the pinion first reduction and bull wheel
- Sound attenuating coupling to the propeller shaft

2.3 Tooth accuracy

The improvement of the grinding machine accuracies was accelerated after large and reliable tooth measuring machines became available. The largest measuring machine ever built can handle gears up to 5 m diameter and 50 tonnes weight. A complete documentation such as pitch, profile and lead deviations, and surface roughness is delivered. At the same time the actual tooth corrections are recorded.

Eventually gears to British Standard Class AO could well be made on the HSS 460 gear grinding machine. The structure-borne noise with such gears was recently measured on a destroyer gear (Fig. 1).

2.4 State of the art in design and finishing hardened gears

Late in 1988 Maag completed a new CNC (computer numerically controlled) gear grinding machine, type SE-

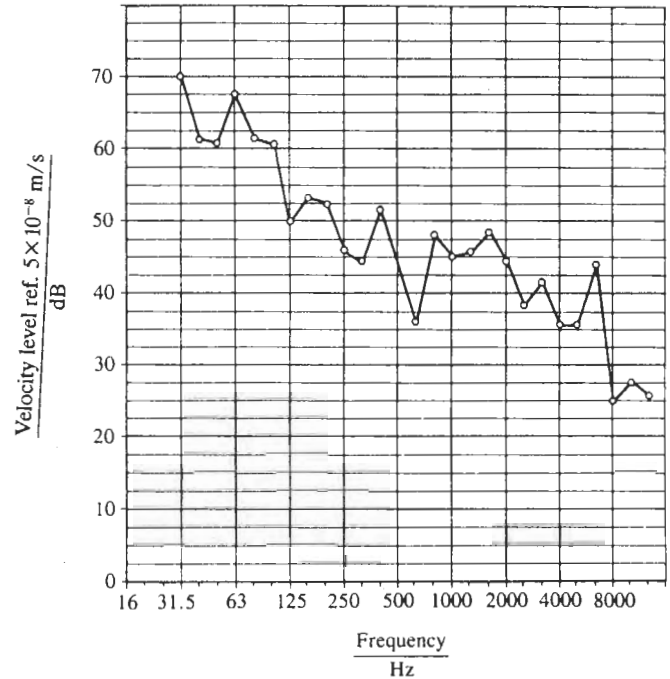


Fig. 1 Structure-borne noise at foot of destroyer gears at 18 kt (~ 4 MW): rigid mounting

202E. This machine is combined with a gear measuring centre (Fig. 2). The first results obtained in production grinding have been most encouraging. Another step toward higher tooth accuracies and productivity has been made. Furthermore the machine is equipped for grinding two- or three-dimensional tooth corrections.

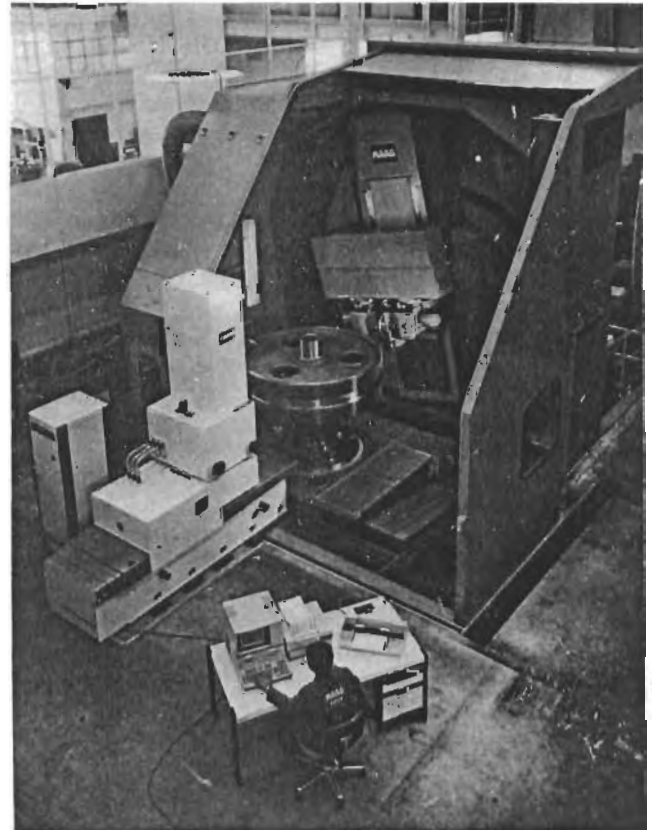


Fig. 2 Tooth grinding and measuring centre SE

All important gear quality elements are measured and recorded on the grinding machine without breaking the set-up. If errors are found to be outside the tolerances corrections of the machine setting can be made quickly without costly interruption of the grinding process. When a gear or pinion is lifted off the machine it is finished, checked and the deviations are documented. The machine data are as follows:

Gear outside diameter	0–2000 mm
Module	3–36
Maximum face width for spur gears	1260 mm
Maximum work load on table	15 000 kg
Grinding wheels	CBN
Coolant oil is applied to the grinding wheels	

Two more machines of this type will be added. One of them will be for outside diameters up to 5000 mm.

2.4.1 Tooth deviations

A series of primary gears for destroyer propulsion units has recently been ground on the old HSS 460 grinder and one gear set on the new SE machine. These gears have the following tooth data:

Pitch diameter	871 mm
Face width	320 mm
Number of teeth	125
Pressure angle	17.5°
Helix angle	16°

The tooth flanks are of true involute shape. All tooth corrections are made on the pinion teeth.

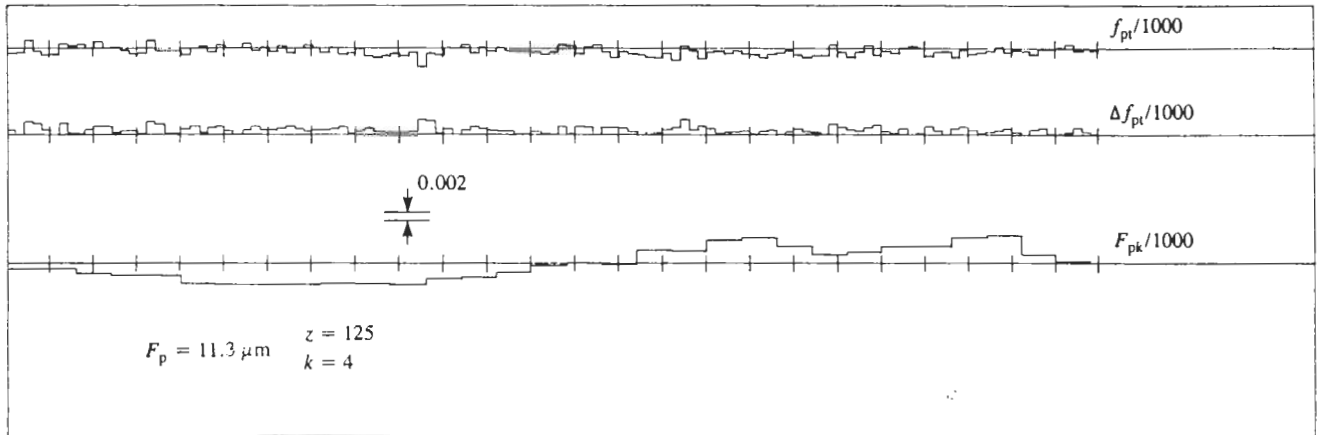
The tooth deviation records are shown on Figs 3, 4 and 5. Each figure contains records from the old and new grinder to make comparison easier. The improvement achieved by the SE machine is quite evident; the quality grade is DIN 1 or ISO 0.

The structure-borne noise levels of Fig. 1 were measured with these gears made on the old machine. With gears ground on the new machine the noise measurements will be carried out on board upon completion of the ships.

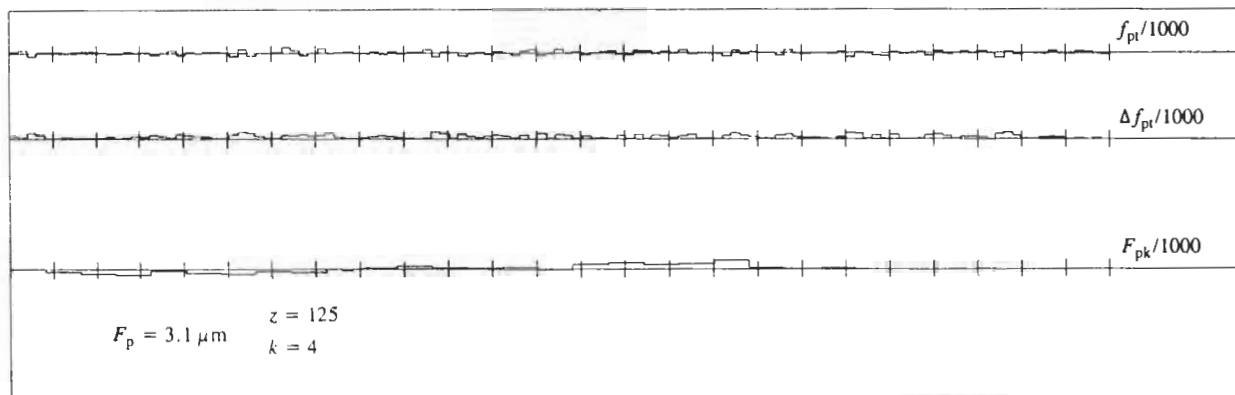
A test gear of roughly twice the diameter was also ground on the SE machine. The tooth data and the quality grades are given in Table 1.

2.4.2 Tooth corrections

Gears and pinions under load suffer elastic deflections: they bend and twist and the meshing teeth get deformed. Friction and quenching in the toothing and friction in the bearings raise the temperature of the rotors. The average temperature of the pinion becomes higher than the gear and the temperature distribution in the gear blanks is uneven. The hottest point of a single helical gear is approximately one-third of the face width from the tooth extremity which enters the mesh last. Consequently, there is a difference in base pitch between pinion and gear which is different at each point across the face width.

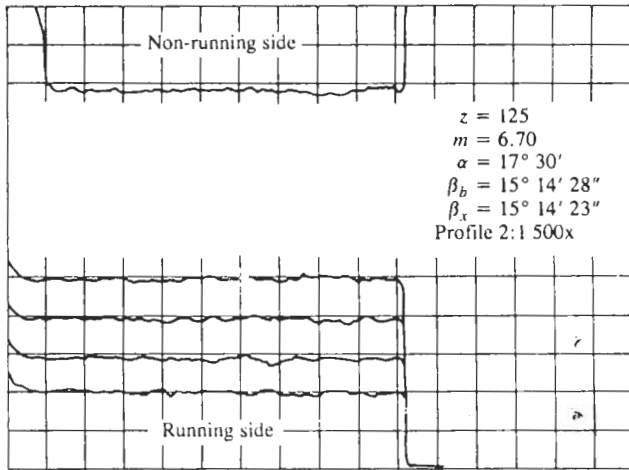


(a) HSS 460

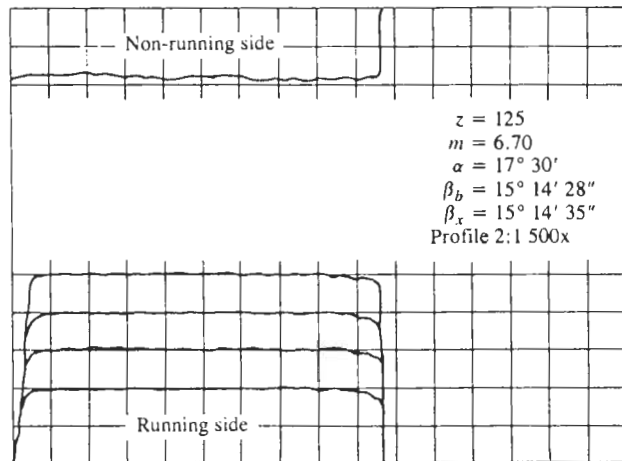


(b) SE 202

Fig. 3 Tooth pitch deviation records HSS 460 and SE 202



(a) HSS 460



(b) SE 202

Fig. 4 Profile deviation HS 460 and SE 202

These elastic deflections and heat expansions influence the tooth meshing unfavourably. Therefore, the theoretical involute at the pinion is corrected during tooth grinding so that at a predetermined load and speed the meshing is close to ideal. Ideal conditions are even load distribution across the face and between the meshing teeth and equal base pitches at pinion and gear at each point across the face. Smooth tooth engagement and disengagement are further considerations.

The complex nature of determining tooth corrections is dealt with in greater detail in the Maag gear book (7).

Such three-dimensional corrections (with variable base pitch across the face) can now be ground by the new SE-202 machine.

A test pinion with extreme corrections was ground with the following data:

Pitch diameter	425 mm
Face width	2 x 300 mm (double helical)
Number of teeth	57
Helix angles	20°

The specified tooth corrections and tolerances have been fed into the computer of the machine. After tooth grinding the integral measuring machine plots the tolerance fields (two lines) as well as the actual shape of the profiles and leads (Figs 6 and 7). Sections outside the

Table 1 Quality grades of test gear ground on SE machine

		Quality grade	
		ISO	DIN
<i>Pitch deviation</i>			
Accumulated	F_p	0	1
Adjacent	f_{pt}	1 (2)	1 (2)
<i>Profile deviation</i>			
Total	F_a	0 (1)	(1) 2
Pressure angle	f_{Ha}	0 (1)	1 (2)
Profile form	f_{ra}	1	2
<i>Helix deviation</i>			
Total tooth alignment	F_β	1	1
Tooth alignment	$f_{H\beta}$	0 (1)	1
Longitudinal form	$f_{r\beta}$	(1) 2	1

Tooth Data:

Pitch diameter	1822 mm
Face width	200 mm
Number of teeth	243
Helix angle	12°
No tooth corrections	

tolerances are corrected by entering new machine setting data into the computer and regrinding.

Figure 6 shows tooth profiles of the same tooth at different sections across the face width. The pressure angle (base pitch) as well as the shape of the profile are different at each section. Figure 7 shows lead diagrams of the same tooth at various heights of the flank. The longitudinal corrections are positive and negative. Figure 8 consists of a multitude of measured tooth flank diagrams plotted across the face width. The three-dimensional character of the tooth correction is clearly visible.

2.4.3 Summary and outlook

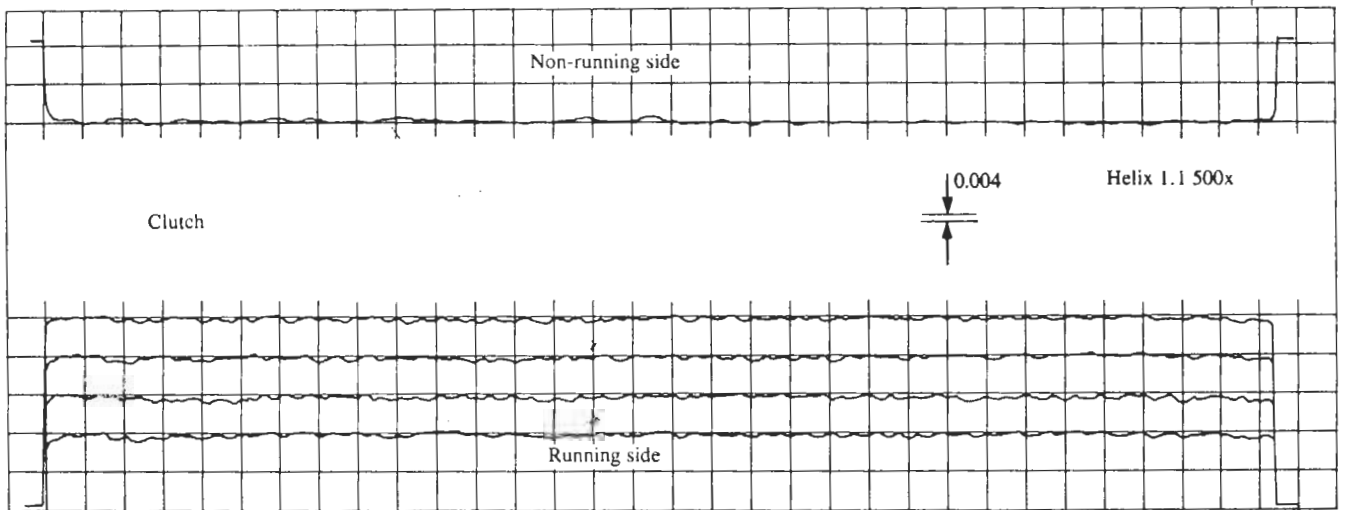
The accuracy of tooth grinding has been further improved. Quality grades between DIN 1 and 2 or ISO 0 and 1 have been achieved. In the near future ground marine gears with DIN 1 quality will be available.

With the new machines involute corrections of practically any shape deemed advantageous can be made. Consequently, the meshing conditions of gears can be improved. This will benefit the load capacity and reduce gear noise. However, a precise prediction of noise reduction is not easy to draw. Theoretical work to this end, which was carried out in the United States (8), is based on known gear tooth deviations and may lead to an answer.

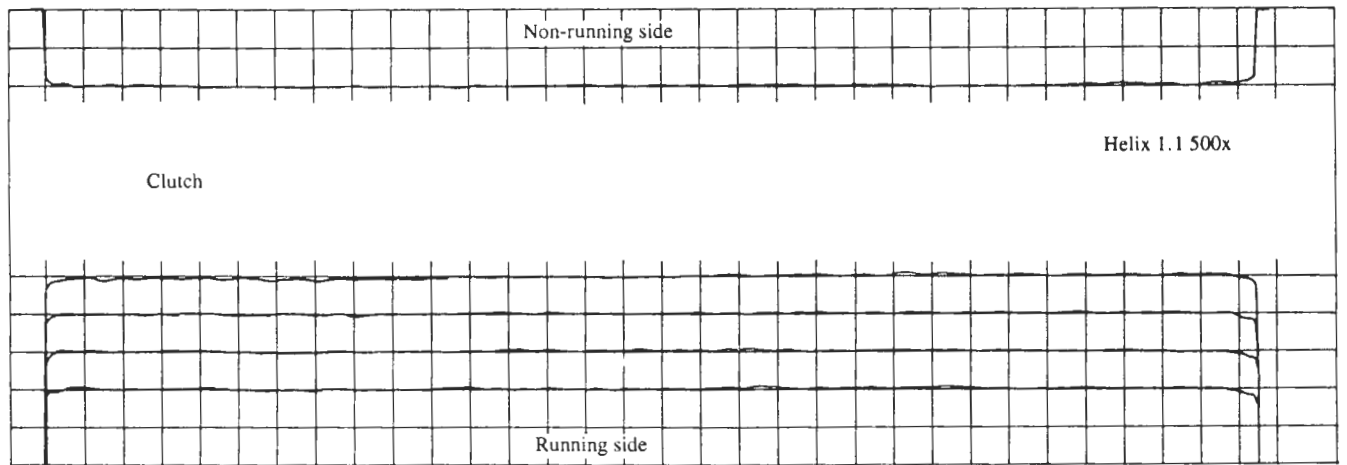
Maag has built a special back-to-back test stand for high-speed gears to find optimal tooth corrections. Eventually, this work will lead to empirical formulae for determining such corrections in the design stage of new projects.

3 GEAR SYSTEMS FOR COMBINED PROPULSION UNITS

Today, major warships are mainly propelled by gas turbines and high-speed diesel engines. Combinations of the two provide good efficiency, with the diesel for cruising and the high-power gas turbine for maximum speed. In some cases the cruise engine is also a gas turbine but of smaller size. We talk of CODOG and COGOG propulsion systems. Either the cruise engine or the booster engine is driving.



(a) HSS 460



(b) SE 202

Fig. 5 Lead deviation HSS 460 and SE 202

Gear systems for such units are relatively simple and experience has shown that they are most reliable. They include synchronizing clutches which in earlier days were considered potentially susceptible to breakdown.

Propulsion systems where booster and cruise engines work simultaneously when sailing full speed ahead are known as COSAG, COGAG and CODAG systems. A well-known representative of the steam and gas combination (COSAG) is *HMS Devonshire* (9). COGAG installations with two gas turbines of equal size driving

one shaft are frequently installed on heavy ships such as large destroyers, helicopter carriers etc.

These two configurations (COSAG and COGAG) present no special problems to the gear designer, at least so long as controllable pitch propellers are used.

Why are CODAG systems not more popular?

Often the cruising power is rather small in percentage terms so that the gain in maximum speed would be in the order of just one knot if diesel power were added. This benefit is considered not worth the additional

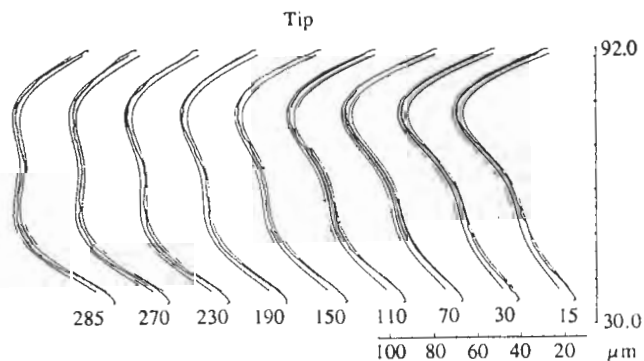


Fig. 6 Profile diagrams of one tooth

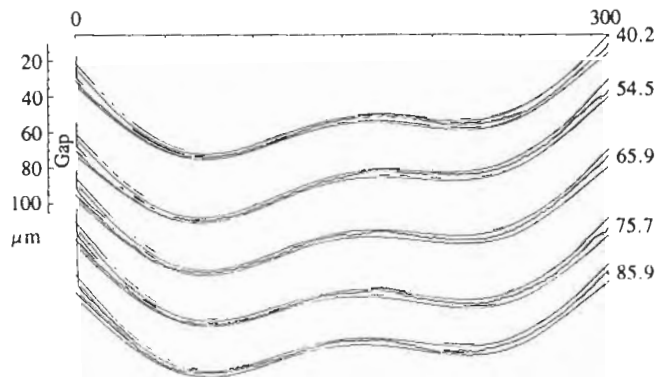


Fig. 7 Lead diagrams of one tooth

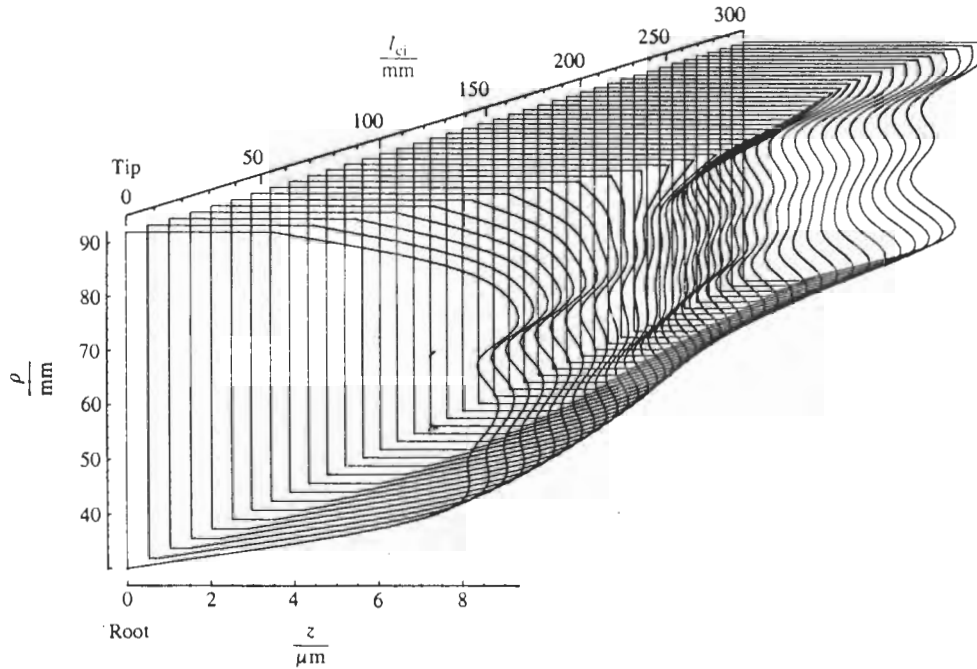


Fig. 8 Tooth flank diagrams plotted across the face width

costs. One reason for shying away from CODAG may be the difficulty in controlling the power-sharing between a large booster and a small diesel. However, modern electronic means should master this problem. Furthermore, in most CODAG systems the speed ratio of the cruising geartrain must be reduced before the large booster power is added. Otherwise the cruise engine would overspeed.

To make CODAG systems equally attractive and reliable as the well-proven CODOG they must meet the following requirements:

1. The changeover from one operating mode to another must be smooth without abrupt changes in speed and torque.
2. Once a new operating mode has been selected by the operator, the changeover must be initiated and performed solely by changing power and torque at the engines in a predetermined way (which is programmed in the controls).
3. The proper actuation of the clutches must under no circumstances depend on external controls such as shaft speed pick-ups, torque indicators etc.
4. Relative shaft speed detectors other than the overrunning mechanism of the clutch itself (pawls and ratchet wheels) must be avoided, that is the clutches must only provide signals to the controls (engaged, locked etc.) but must not depend on any signals from the controls (apart from being informed beforehand of an intended change of operating mode).
5. The clutches must be of a sturdy design and capable of safely withstanding relative shaft accelerations at the synchronization point which is greatly in excess of the acceleration experienced in normal service.

The characteristics of a CODAG system built according to the above rules could somehow be compared to a certain extent with an automatic gearshift in a car. A preselector informs the gearbox of an intended change of operating mode and the changeover itself is per-

formed simply by appropriate changes of power and speed of the prime movers.

In order to meet these requirements two types of synchronizing clutches are needed:

Type MS

Type MS (Fig. 9) is a synchronizing clutch that has been widely installed in CODOG systems. It engages automatically when the speed of the prime mover begins to overrun the pinion speed, and it disengages automatically when the prime mover is slowed down and its torque turns negative.

Synchronization is accomplished by an overrunning mechanism consisting of a ratchet wheel and pawls. The clutch teeth engage in a screw motion.

If required a locking device can be added. This device has helical teeth. Torque transmission is permitted in either direction for any length of time. The well-known SSS clutch has the same basic characteristics. Both clutches are well described in Dr Shannon's book *Marine gearing* (10).

For further clarification Fig. 9 shows the principle of synchronization, the shock absorption at the pawls and the engagement of the clutch teeth. The symbol for this clutch as used in the rotor plans is +MS, the plus sign indicating that engagement takes place when the input shaft overruns the output shaft.

Type DS

Type DS is a double clutch configuration. This clutch has been presented recently and is described at length in various technical magazines (11, 12). Therefore, the following concentrates mainly on its characteristics.

Two identical MS clutches are arranged in line and back to back. They are connected by a tooth coupling sleeve (Fig. 10). One clutch is seated on the output shaft (+MS) and the second one on the input shaft (-MS), the minus sign indicating that this latter clutch will engage when the output shaft speed overruns the input

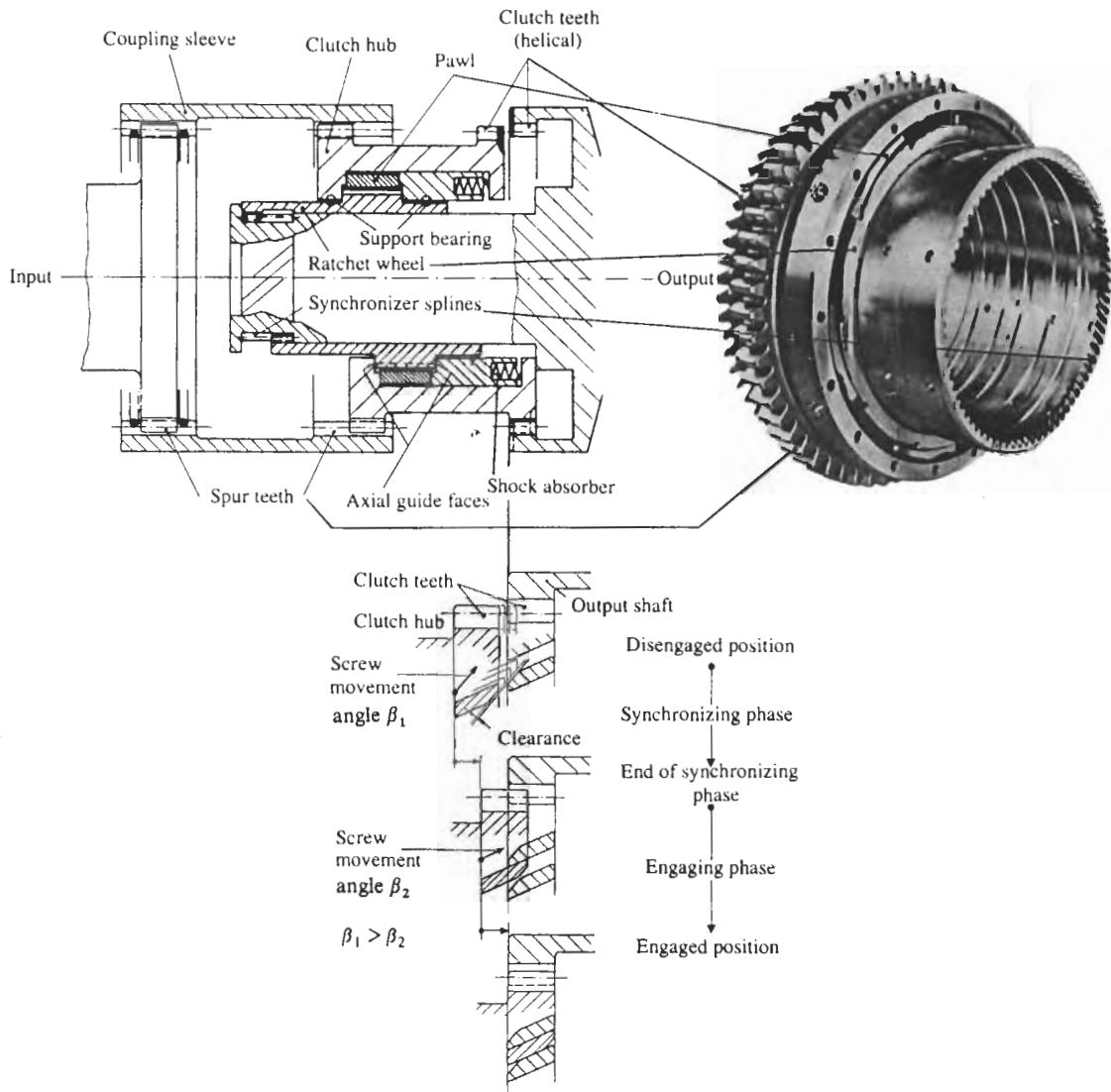


Fig 9 Synchronizing clutch type MS

speed. Either of the two MS clutches may be disengaged but it is made *impossible* by mechanical means that both can be disengaged at the same time, that is 'lost shaft' situations are avoided.

The helical locking devices are operated by an integrated clutch control. This control receives the information from the machinery controls indicating an intended change of operating mode. Two clutch modes can be preselected:

- Mode I 'Engage'
- Mode II 'Freewheeling'

It is important to note that the selection of a new mode does not have any effect on the clutch itself. It only *prepares* the clutch for changing to the new mode as soon as the appropriate speed and torque conditions are met.

The characteristics of the DS clutch are best explained by referring to the graphs of Fig. 11 which show the two alternative engagements:

1. In the 'freewheeling' mode the input and output shaft speeds are free to override each other. At each crossover one side engages while the other side disengages automatically.
2. The 'engage' signal can be given to the clutch control at any time (point 3), no matter which shaft happens to rotate faster at that moment. The DS clutch will engage at the next crossover of the two shaft speeds (point 4).

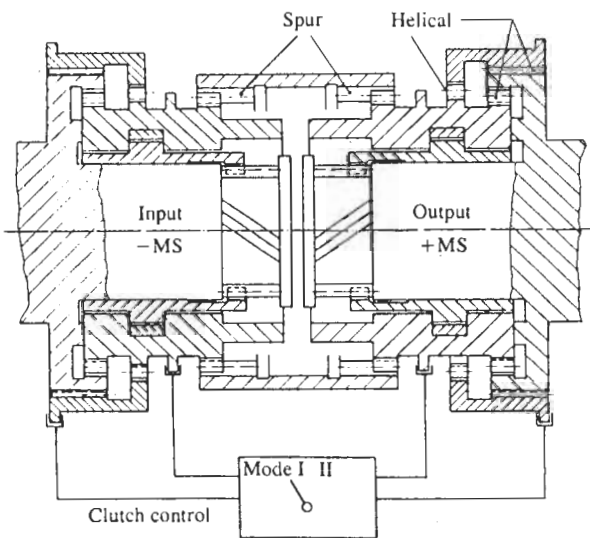


Fig 10 Synchronizing clutch type DS

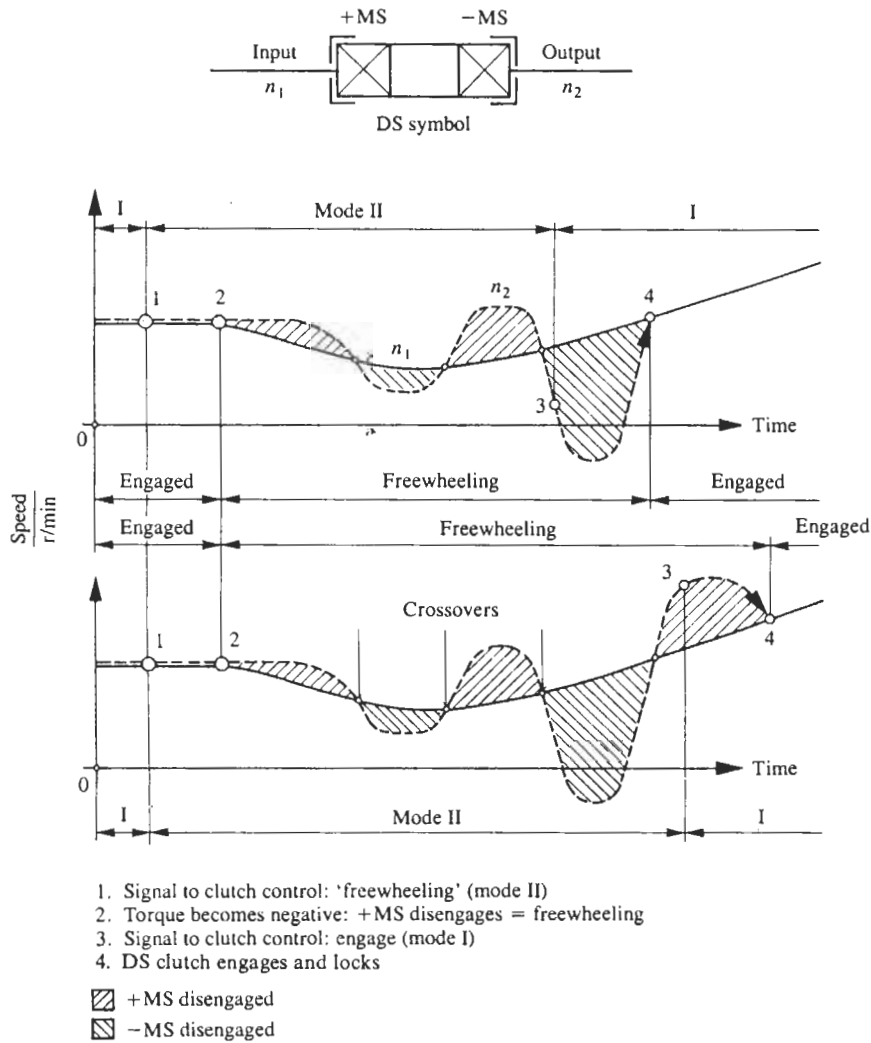


Fig. 11 Characteristics of DS clutch

3.1 CODAG systems

Figure 12 shows a much simplified rotor plan of a CODAG system. The MS and DS clutches provide a solution which meets the requirements established above. This system has been published recently (13). The following remarks are made for additional clarification.

1. The booster drives the main gear via clutch B (+MS).
2. In the cruising mode the diesel is directly connected by clutch C (+MS).
3. When the booster and cruise engine operate together the cruising power is passed through a speed increaser (A gear) and the DS clutch A.
4. The changeovers from full speed to cruising speed and vice versa are described step by step in the Appendix.

Figure 13 shows a propulsion plant comprising one booster and two diesels driving two shafts in a CODAG arrangement. The functioning of this system is basically identical to that described in the Appendix. With this configuration the total cruise engine power is more likely to be sufficiently large to make CODAG really superior to CODOG. Or, if the additional ship speed is

not an issue it may well be possible to install a smaller boost engine.

4 REVERSE GEARS FOR USE WITH GAS TURBINES

Gas turbines are not reversible. The controllable pitch propeller is widely used on gas-turbine-driven ships.

Reverse gears have been less popular, except with the Royal Navy. In the early 1950s PAMETRADA worked on fluid couplings and astern converters for reverse gears. The propulsion systems for the County and Tribal Class are famous designs (9). The development of reverse gears in the United Kingdom was topped by the COGAG installation for *HMS Invincible* (14) which entered service in 1981. Separate fluid couplings were used for ahead and astern manoeuvring.

Inspired by these efforts, the Italian company Franco Tosi built their reversible converter coupling (RCC) which provides ahead and astern drive in one unit. RCCs were first applied in the gears for the Italian cruiser *Garibaldi* (Fig. 14). Like *HMS Invincible*, the *Garibaldi* has COGOG machinery with triple reduction reverse gears. The maximum power per shaft is 50 000 h.p. and the weight of one gear unit is approximately 75 tonnes. The RCCs are placed inside the gearbox

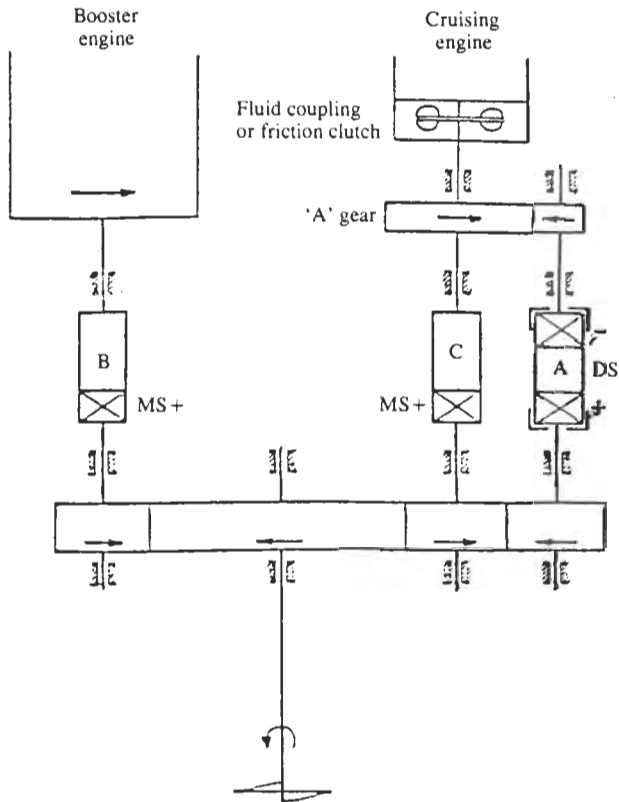


Fig. 12 CODAG gear system

between the first and second reduction. For full power ahead the torque bypasses the RCC and is transmitted by a formfit SSS clutch with lock control and pawl-free position [see Fig. 14.4 of reference (10)].

For manoeuvring the RCCs are filled and the SSS clutches disengaged and moved to the 'pawl-free' position. Astern rotation is accomplished simply by inserting the vanes. The manoeuvring quality is excellent. Shaft speeds down to a stop can be controlled accurately and the gear—and propeller—noise at these low speeds is at a minimum.

Special attention is needed for re-engaging the synchronizing clutch! The slip in the RCC must be overcome. A powerful disc brake is required to reduce the gas turbine speed to a point where the clutch input speed has dropped below the clutch output speed. This clutch engagement is shown graphically in Fig. 15a.

The change from the manoeuvring mode to full speed ahead is thus performed automatically by the engine controls in the following steps:

- Withdraw vanes: ship moves ahead
- Reduce gas turbine power to idling
- Apply brake: clutch input speed drops below output speed
- Shift clutch pawls to the 'pawls active' position
- Lift brake
- Increase gas turbine speed: clutch will engage automatically

A number of options have been studied aiming at a more simple gear design of lower weight and which requires less space. Furthermore it was found that the double synchronizing clutch DS will simplify the controls, shorten the changeover time and improve reliability.

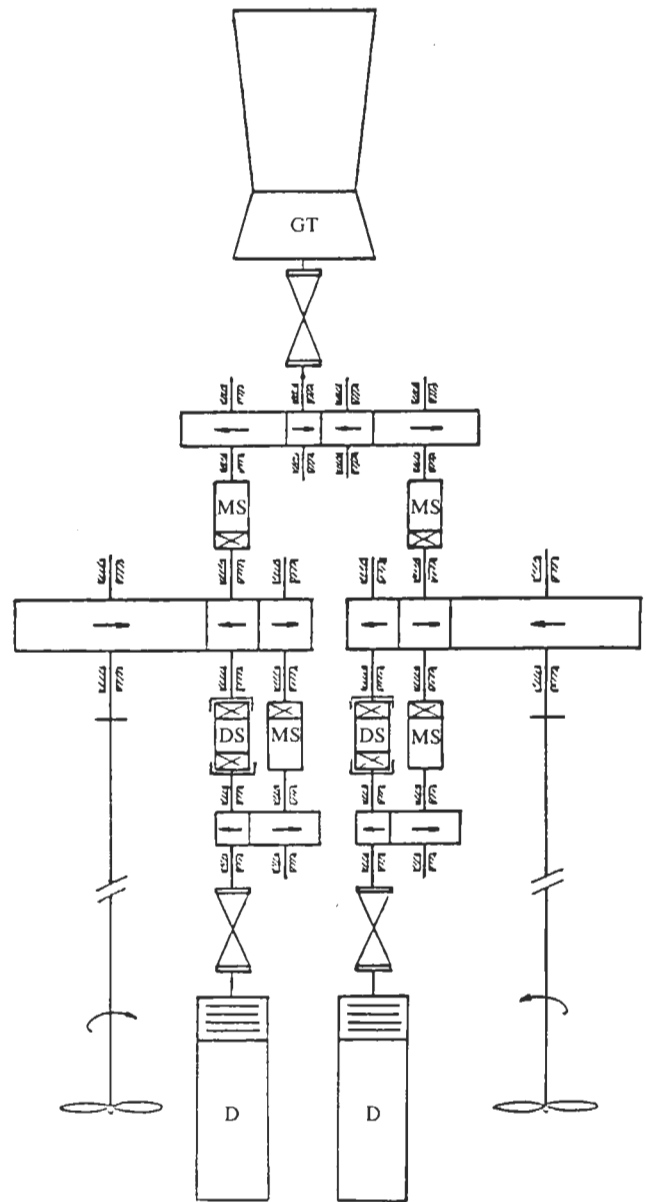


Fig. 13 CODAG system for Corvette

The outcome of this study is shown by the rotor plan of Fig. 16 (13). The following summarizes the important features and some advantages over the *Garibaldi* system.

Full ahead power is transmitted by the locked train gearing with the RCC emptied and the gear and pinion of the reverse shaft idling. The two clutches at the two intermediate shafts (one DS and one disengageable tooth coupling TC) are engaged.

In the *manoeuvring mode* the tooth coupling TC is disengaged and the DS clutch is 'freewheeling'. The power is transmitted to the reverse shaft R via one first-reduction gear and the RCC, which is filled with oil. The propeller shaft rotates astern with the vanes of the RCC *withdrawn* and *ahead* with the vanes *inserted*.

The advantages in comparison with the *Garibaldi* solution (Fig. 14) are:

1. The engagement of the DS clutch is performed as shown by the graph on Fig. 15b. Engagement takes

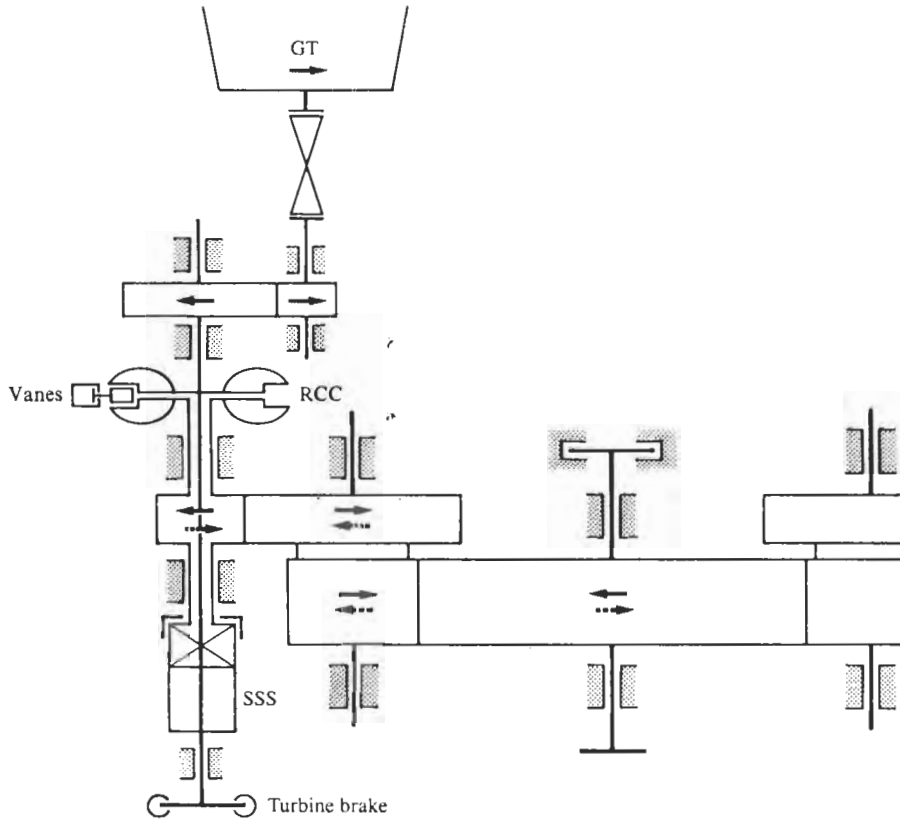


Fig. 14 Franco Tosi reversing gear

- place fast when the two shaft speeds intersect *for the first time*. The controls are less sophisticated.
- No gas turbine brake is needed to re-engage the DS clutch because the RCC (vanes inserted) works as a torque converter when manoeuvring ahead. Of further help in forcing the two shaft speeds of the clutch to cross each other is the speed differential between the intermediate shaft and the reverse shaft.

- The pawls of the DS clutch are *always* active. There is never a 'lost shaft' situation. Reliability is improved.
- Gears and pinions are arranged in two planes only. This permits 'standard' gear casing design and short length.
- The RCC (hot part!) is placed outside the main gear-box.

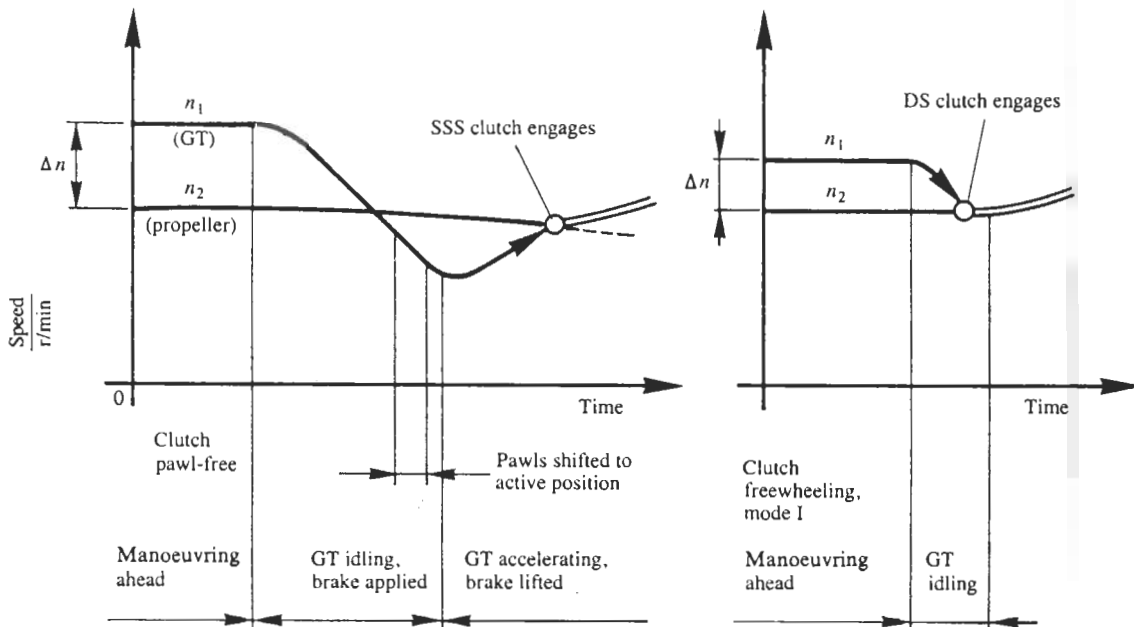


Fig. 15 Re-engagement of synchronizing clutch

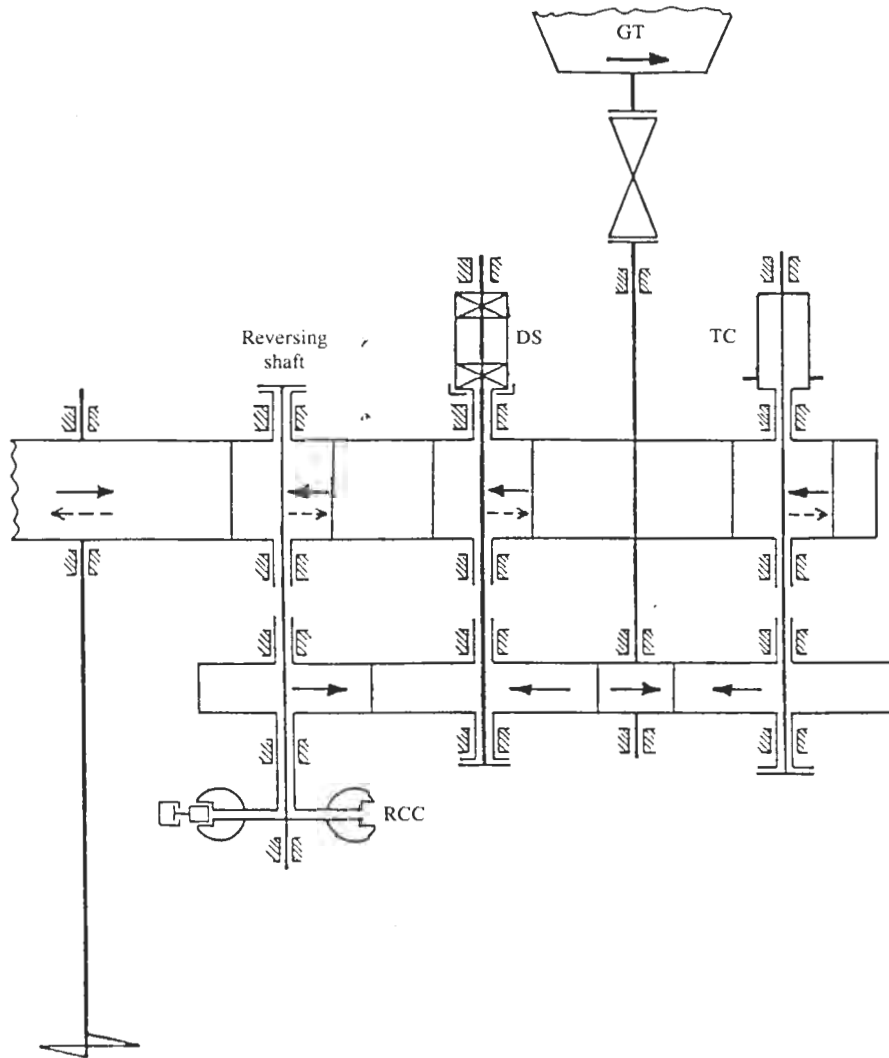


Fig. 16 Proposed reversing gear

6. Clutches are quillshaft mounted which provides excellent accessibility to all vital parts.
7. The overall weight of the gearbox will be reduced to 68 tonnes.

The windage losses in the RCC when sailing full ahead are in the order of 70 h.p. These losses plus the losses in the reverse shafting are more than compensated for by the locked-train gearing and by the fact that only two reduction stages are necessary instead of three.

5 CONCLUSIONS

Modern naval propulsion gears are equipped almost exclusively with surface-hardened gear teeth or high-tensile steels are used for large diameters. New gear grinding equipment provides tooth accuracies close to quality grade DIN 1 and the possibility to grind three-dimensional tooth flank corrections. Corrections of this nature are needed to compensate properly tooth deviations caused by uneven temperature distribution in the gear blanks at a defined speed and power.

Integral tooth flank measuring equipment records tooth deviations with the workpiece still on the grinding machine.

Such gearing, properly laid out and maintained, keeps its shape over an extended length of time and operates at minimum tooth meshing noise.

Propulsion gear systems of a more sophisticated nature (for example CODAG) can be built to provide similar operating reliability as CODOG installations. The application of two types of synchronizing clutches eliminates the need for external controls. The changeovers from one operating mode to another are simply accomplished by appropriate changes of power and speed of the prime movers.

A reverse gear with reversible converter coupling (RCC) is proposed. The use of a double synchronizing clutch for bypassing the RCC results in quick and safe changeovers. Again, no external clutch controls are required.

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APPENDIX

CODAG (Fig. 12)

Changeover from full ahead (gas turbine plus diesel) to cruising (diesel only)

Step 1 At the command 'cruising' the controls of the DS clutch (A) are set to 'freewheeling' whereby the plus side of this clutch is unlocked.

Step 2 Power and speed are reduced. The power of the cruise engine is transferred to the booster. When the

torque at the cruise engine turns negative the plus side of the DS disengages and the minus side unlocks automatically. The DS is 'freewheeling'.

Step 3 The speed at the cruise engine is increased again and the booster power is further reduced. First the plus side of the DS will engage and the minus side will disengage (crossover). Finally clutch C will engage automatically.

Step 4 Power is transferred back from the booster to the cruise engine.

Step 5 Booster is stopped and clutch B disengages automatically.

Changeover from cruising to full ahead

Step 1 At the command 'full ahead' the controls of the DS clutch are set to 'engage' whereby the locking devices at both sides of this clutch are moved to the lock position. (The locking rings are engaged by spring force.)

Step 2 The booster is started and steadily brought up in speed until clutch B engages automatically.

Step 3 Power is transferred from the cruise engine to the booster.

Step 4 Booster and propeller speed increase while the speed at the cruise engine remains approximately constant. First clutch C will disengage automatically. Finally the negative side of the DS clutch will engage and lock automatically.

Step 5 Resume full power at both prime movers.