NAVAL GEARING REQUIREMENTS

BY

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INTRODUCTION

There are several outstanding differences between the requirements for naval and merchant ship gearing:

(*i*) Minimized space and weight is important for naval vessels and relatively unimportant for merchant ships.

(*ii*) Minimized noise and vibration may be a matter of life or death in naval vessels whereas it affects only habitability in merchant ships. Additionally provision has to be made in naval gearing for shock resistance.

(*iii*) The use of combined plants with gas turbines for boost power in naval vessels necessitates more complicated gear boxes, incorporating additional gear trains and clutches, than are at present required for merchant ships.

(*iv*) Cheapness is all important for merchant ships whereas it is less important than the other requirements for naval gearing.

Many people would say that another fundamental difference is that merchant ship gearing has to be designed to survive some twenty years running for over 300 days per year at service power, whereas naval gearing may only have a few hundred hours running at over 80 per cent power in the same period. The author does not regard this as of such importance, since 100 hours at 200 r.p.m. gives over 10^6 revolutions and with gear ratio taken into account, naval as well as mercantile gears have to be able to survive a full fatigue life. With case-hardened gears not subject to pitting and which must be designed to avoid scuffing and tooth breakage, there should be no deterioration however long they are run at high power.

In what follows, concentration will be on aspects (*i*), (*ii*) and (*iii*) above.

TRENDS IN NAVAL MACHINERY INSTALLATIONS

Combined plants designed to make use of the excellent characteristics of gas turbines as boost engines are currently being considered for a wide range of naval vessels throughout the world. These include:

COSAG — steam and gas
CONAG— nuclear steam and gas
CONOG— nuclear steam or gas (gas for 'get-you-home' purposes)
COGAG— gas and gas
COGOG— gas or gas (cruising engine is not used at full ship power)
CODAG— Diesel and gas
CODOG— Diesel or gas (cruising Diesel is not used at full ship power)

Type	*BHP*	*RPM*
High-Speed Diesels	1500 to 3000 2000 to 4000	1500 to 1800 900 to 1200
Medium-Speed Diesels	3000 to 6000 3000 to 9000 3000 to 13500 6000 to 18000	600 500 450 400
Small Gas Turbines*	3000 to 4500	1000 to 2000
Medium Gas Turbines	8000 12000 to 16000	4900 3600 to 6000
Large Gas Turbines	20000 to 30000	3600 to 6000

*Small gas turbines usually have an integral reduction gearbox and so the input speed to the ship's main gearing is usually at an intermediate speed.

FIG. 1—POWERS AND GEARBOX INPUT SPEEDS OF TYPICAL PRIME MOVERS FOR COMBINED MACHINERY PLANTS

The relative advantages and disadvantages of these combinations are discussed in a paper by Good (Ref. 1). Since a satisfactory 'astern' gas turbine has not yet been developed, reversing means have to be provided either in the gearing or in the propeller. In addition to the complications that this may involve in reversing gear trains, provision has to be made in the gearing for clutching in and out the prime mover, and possibly for a gear change for the Diesel drive in 'AG' designs as opposed to 'OG' designs. Another feature that may arise with gas turbines, which have poor part-load performance, is the desirability of a combined gearcase for the two shafts as described in Ref. 1. Where gas power turbines have to be turned when not in use, either because of the risk of brinelling or rotor distortion, a multiplicity of turning gears with their own clutching in and control problems may have to be built into the gearing. Naval gears therefore tend to be complicated as well as demanding minimum space and weight. Hence the need in naval gears to be progressive in cutting down design margins and reducing factors of safety to a larger extent than is justified in merchant ship gearing.

FIG. 1 gives particulars of some gas turbines and Diesel engines that are being incorporated in present designs for various navies.

GEAR DESIGN AND POWER LIMITATIONS

General

The detailed requirements in respect of pinion face width to diameter ratio, tooth pitches, helix angle, gear material combinations and the corresponding tooth loadings are specified in G.R.M.(E)-D.G. Ships/TPSE/108 and need not be reproduced in detail here.

Gear tooth pitches are now specified on the diametral pitch system and, in general, the minimum pitch is preferred.

While a helix angle of less than 20 degrees is preferred for single reduction single helical gears, with double reduction single helical gears considerations of reducing end thrust normally lead to the use of angles in the range 5 degrees to 15 degrees for secondary gears and over 20 degrees for primary gears.

Gear tooth loadings are specified and compared by means of the Lloyd's

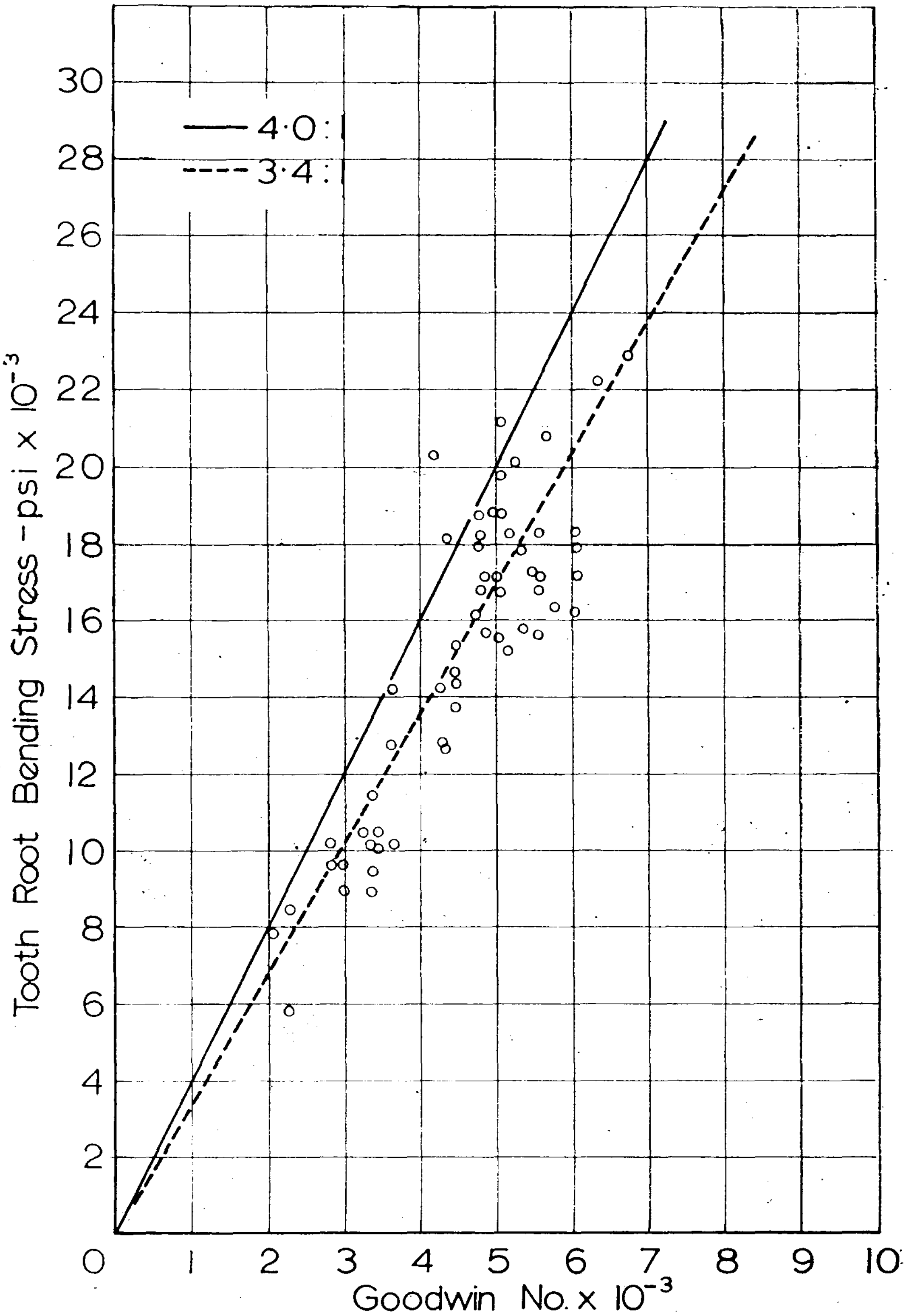


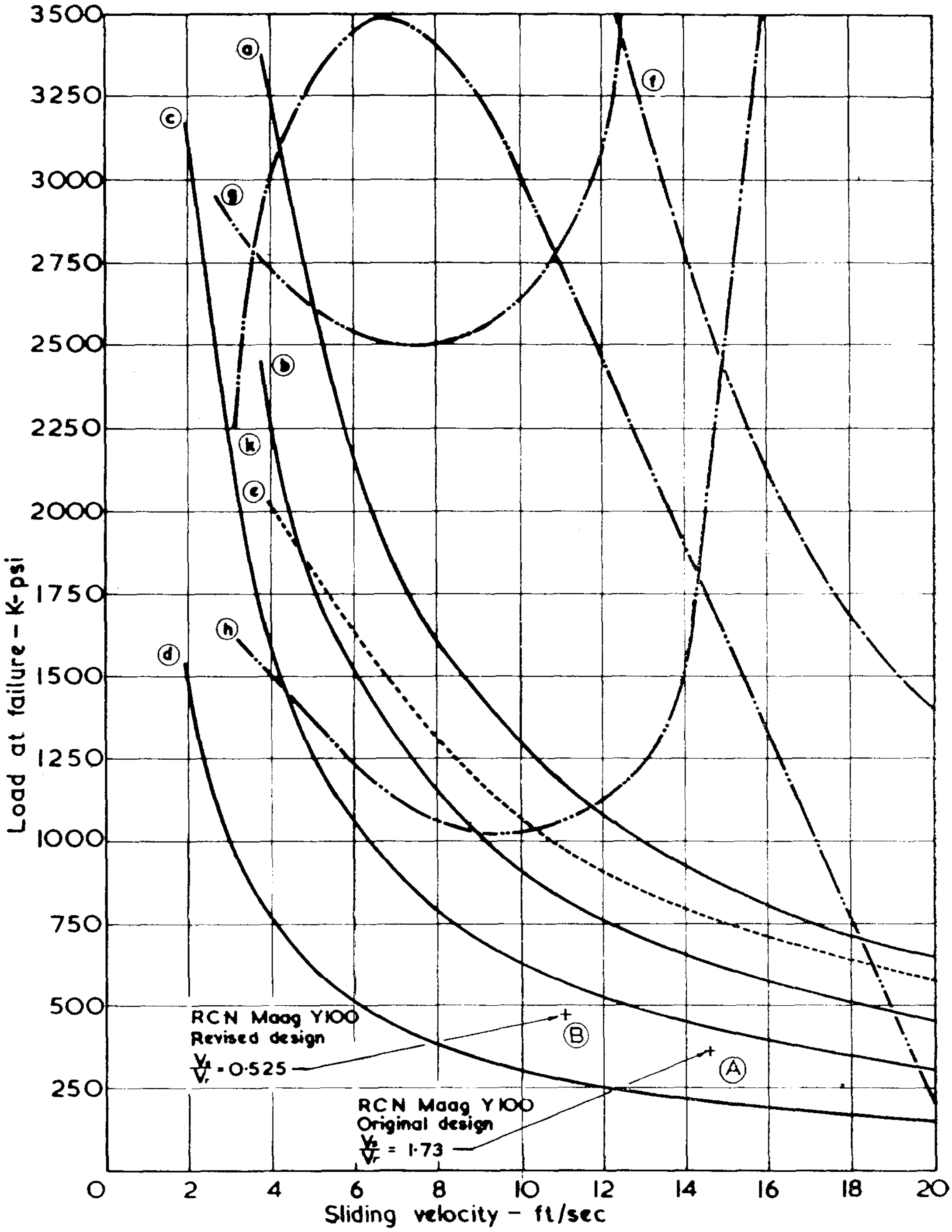
FIG. 2—RELATIONSHIP BETWEEN ROOT BENDING STRESS AND GOODWIN NUMBER FOR GEAR TEETH WITH 20-DEGREE PRESSURE ANGLE

Register K factor which is calculated by the formula $K = \frac{P}{d} \frac{(R+1)}{R}$

where $P =$ Load per unit length of overall face width in lb/inch

$R =$ Reduction ratio $= \frac{\text{No. of wheel teeth}}{\text{No. of pinion teeth}}$

$d =$ P.C.D. of pinion in inches



Symbols used

—— NAVGRA limit $\frac{F_K V_s^2}{V_r} = 2.165 \times 10^4$

(a) $\frac{V_s}{V_r} = 0.839$, $R_r = 1.0$

(b) $\frac{V_s}{V_r} = 1.192$, $R_r = 1.0$

(c) $\frac{V_s}{V_r} = 1.73$, $R_r = 1.0$

(d) $\frac{V_s}{V_r} = 1.73$, $R_r = 2.095$

------ Tuplin limit

(e) $\frac{F_K}{R_r} = \frac{1.5 \times 10^6}{V_s + 100}$

—— PVT limit

(f) $PVT = 2.4 \times 10^6$

—— BSRA test results

(g) En 36/En 36, $\frac{V_s}{V_r} = 0.839$

(h) En 36/En 36, $\frac{V_s}{V_r} = 1.192$

(k) En 26/En 9, $\frac{V_s}{V_r} = 1.192$

FIG. 3—COMPARISON OF DISC TEST RESULTS AND SCUFFING CRITERIA

The maximum permissible *K* factors are 150 to 225 for through hardened gears and 450 to 550 for surface hardened gears, depending upon the material combination selected and whether the gears are for the primary or the secondary reduction.

Tooth Root Bending Stresses

To ensure that all designs for M.O.D. (N) can be compared on a common basis, an agreed method of calculation, known as the NAVGRA method, is specified together with an allowable stress (excluding stress concentration factor) for each particular gear material.

In the initial stages of a design, a rough check that the bending stresses will not be excessive can be made by ensuring that a parameter, which the author's staff have come to call the Goodwin Number, is not exceeded. This useful criterion is calculated by dividing the load per inch of face width by the normal pitch in inches. FIG. 2 shows the calculated values of the bending stress plotted against this criterion for a number of naval gear designs and it can be seen that a mean line through them represents a ratio of 3·4: 1. The ratio assumed for design purposes is 4 : 1 just to be on the safe side.

Scuffing Criteria

Despite the effort that has been devoted over a number of years in many places, there is no very clear guidance on the conditions under which scuffing is liable to occur with either straight mineral or EP additive oil. BSRA recently reported some disc machine results (Ref. 2) with both types of oil and Y-A.R.D. attempted to align these results with the NAVGRA, Tuplin and PVT (General Motors) criteria as shown in FIG. 3. The Maag criterion could not be shown conveniently on this graph.

The only practical gear results available for comparison were those obtained on the Canadian Y. 100 Maag gearing. Point A shows the conditions pertaining when the original design scuffed on OM88, but ran satisfactorily on an EP additive oil and point B shows the conditions after redesign with very much lower $\frac{Vs}{Vr}$ value. The revised design subsequently ran satisfactorily on a non EP oil. Both the BSRA results and the NAVGRA criteria show that scuffing of the original gears was inevitable with OM88 and that the redesigned gears should be safe. However, the difference in shape between the BSRA and NAVGRA curves is disturbing and points to the need for further research work in this field of gear design.

Relationship between Main Wheel Diameter and Shaft Speed

The maximum size of main wheel that can be embodied in a given design will naturally depend on the manufacturing facilities available. FIGS. 4 and 5 show, for single and dual tandem gear arrangements, the relationship between main shaft speed and main wheel diameter for a given set of conditions, as follows:

(*i*) An input power of 25,800 hp from a single gas turbine giving 25,000 *shp.

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*Conversions for powers above or below 25,000 shp at the same input speed of 5,600 rpm, assuming constant F/d ratio and constant K, can be made as follows:

(*i*) At constant main wheel diameter, the main shaft speed varies approximately as $\sqrt{\text{shp.}}$

(*ii*) At constant main shaft speed, the main wheel diameter varies as $\sqrt[3]{\text{shp.}}$

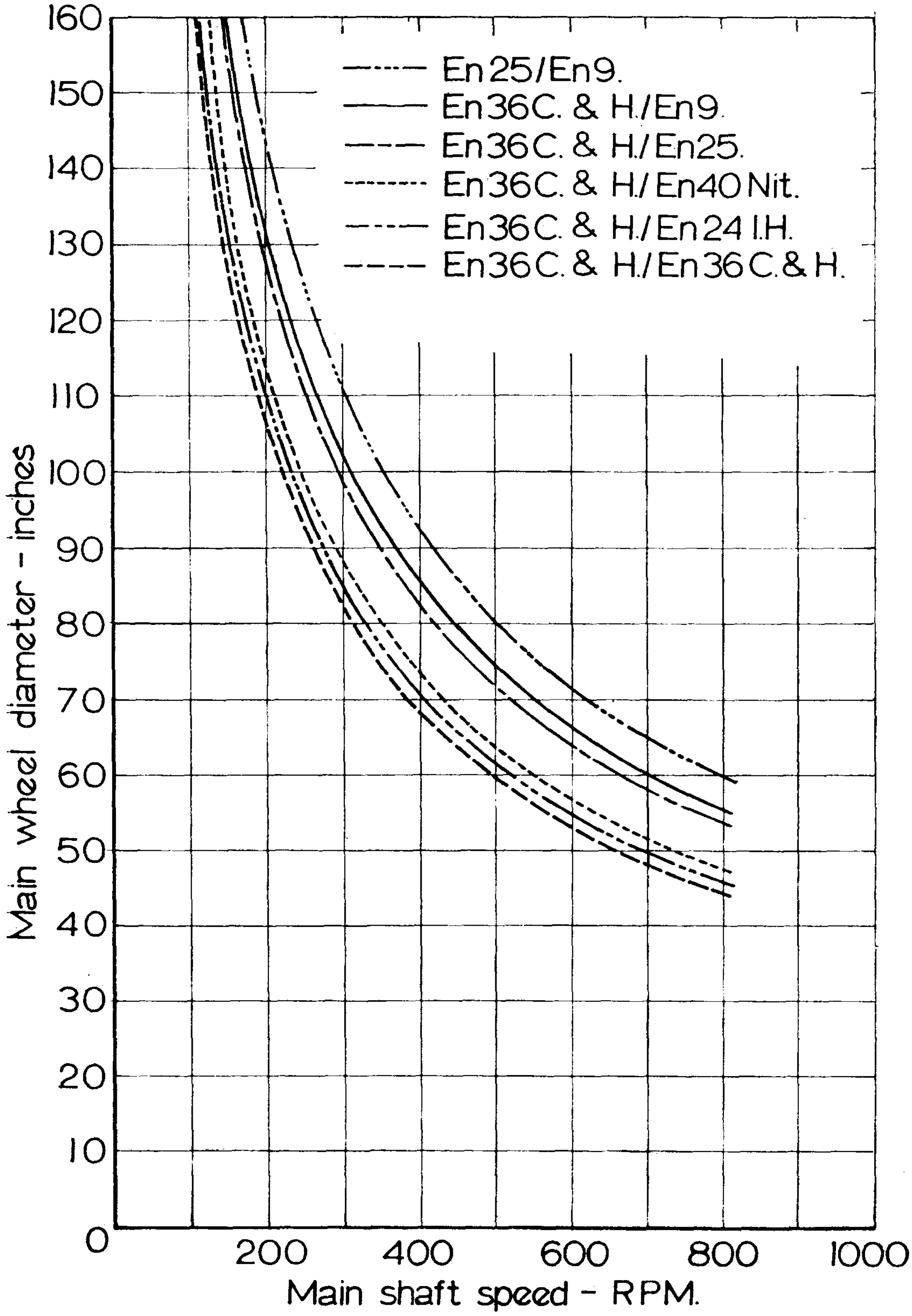


Fig. 4—Limitations for single tandem gear arrangements

(*ii*) An input speed of 5,600 rpm.

(*iii*) A distribution of overall gear ratio between primary and secondary reductions such that the secondary gear ratio is equal to 1·25 times the primary gear ratio.

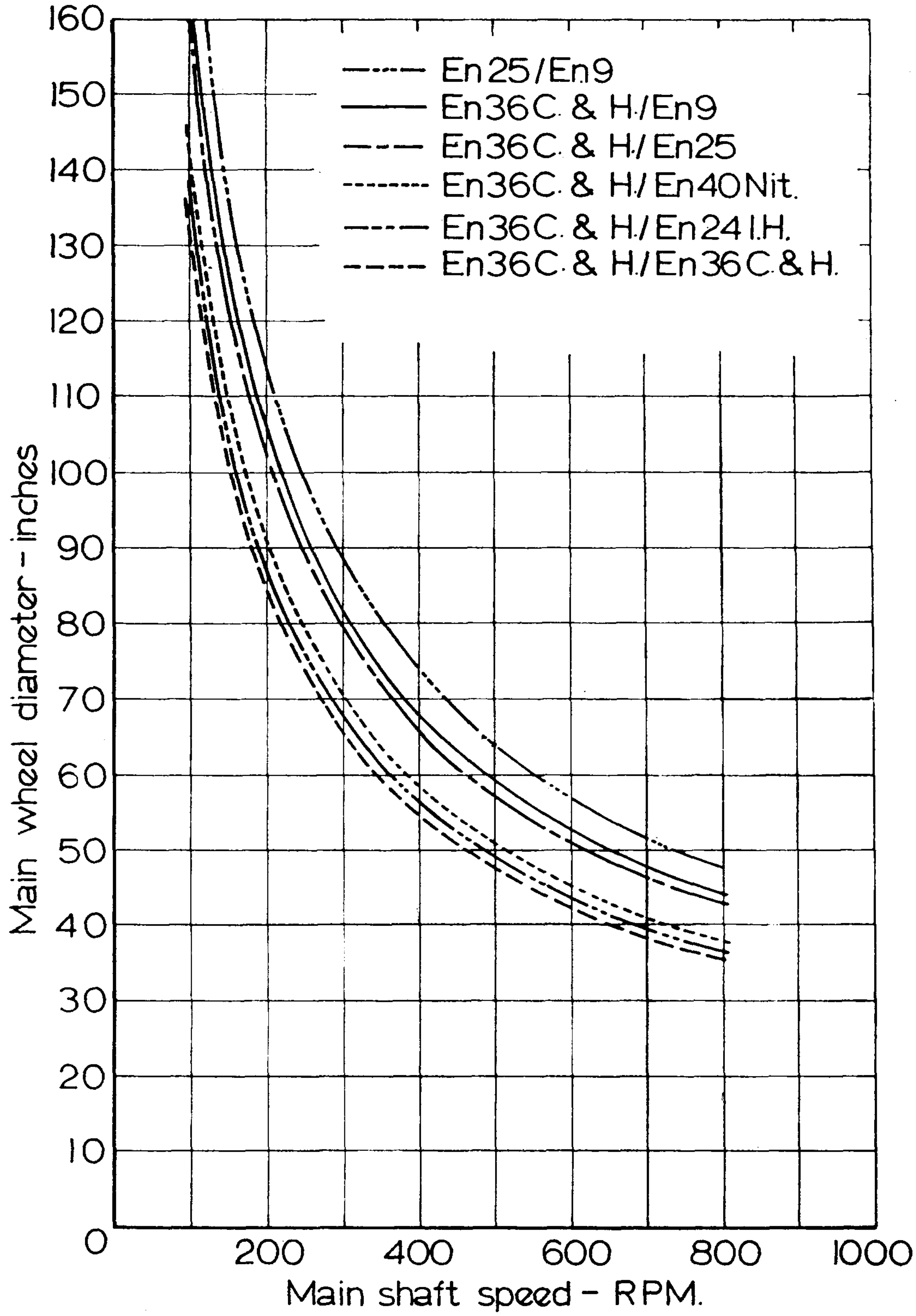


FIG. 5—LIMITATIONS FOR DUAL TANDEM GEAR ARRANGEMENTS

(*iv*) The designs employing through-hardened wheels have been made double helical. The designs employing all surface hardened gear elements have been made single helical.

(*v*) No account has been taken of present limits on manufacturing sizes.

Currently there is a trend towards lower propeller speeds and this may demand further development of the manufacturing facilities. If the complications of triple reduction gearing are unacceptable then there are three possibilities:

(*i*) A larger carburizing furnace

(*ii*) Acceptance of induction hardening for larger sizes

(*iii*) Development of facilities for and acceptance of larger nitrided wheels.

In connection with (*i*) above, it is felt that the carburizing of larger diameter wheels may increase the problems of distortion, and consequently the capital costs of the furnace and subsequent manufacturing costs could be prohibitive. Although there is no inherent difficulty in providing larger plant for induction hardening, this method always leaves a doubt as to complete homogeneity. Nitriding is a comparatively low temperature process with consequential savings in production time and cost due to the almost negligible distortion. It will be seen from FIGS. 4 and 5 that with nitrided main wheels of 10 ft diameter, the main shaft speed could be reduced to approximately 185 rpm for a single tandem and 130 rpm for a dual tandem arrangement and this with a K factor approximately 80 per cent of that of a carburized and hardened gear.

GEARCASE DESIGN

The main function of the gearcase is to keep the gear trains meshing correctly and the stiffer it can be made the better chance it has of achieving this purpose. Quite a large effort has been made over the years to measure the distortions that take place. In the author's view, all that these measurements have achieved is to show that even very stiff, highly loaded, naval gearboxes do deflect in the expected directions as much as 0·030 in. or 0·040 in. in some cases. Therefore, the designer must resign himself to some deflection, and try to arrange his methods of support so that it does not lead to lack of parallelism in the gear trains.

Three-Point Support

For more than ten years the Royal Navy has specified three-point support for main propulsion gears and in practice this results in three-area support. This has been successful in preventing hull distortion causing distortion of the gearcase and provides a useful datum condition during manufacture and erection on board. When the main thrust block is integral with the gearcase, one of the support areas is adjacent to it. The other two areas are positioned on either side of the main wheel so as to take the torque reaction of the final reduction as near as possible to source and thereby avoid gearcase deflections which would affect alignments.

Where Constant Position Mounting Systems, which are described later, are fitted to gearcases, the positioning valves can be arranged to give more truly three-point support. The mounts, of which there may be more than one to each positioning valve, can be arranged to apply the loads in the most advantageous positions to minimize distortion.

Packaging of Lubricating Oil Components

In conventional designs, noise ranging has demonstrated that underwater noise is produced by the oil pump and its motor. When the main gearcase is supported on CPMS to reduce the transmission of gear noise, it becomes attractive to make the main drain tank integral with the gearbox so as to isolate the pump noise also. Once this has been adopted the next step is to mount

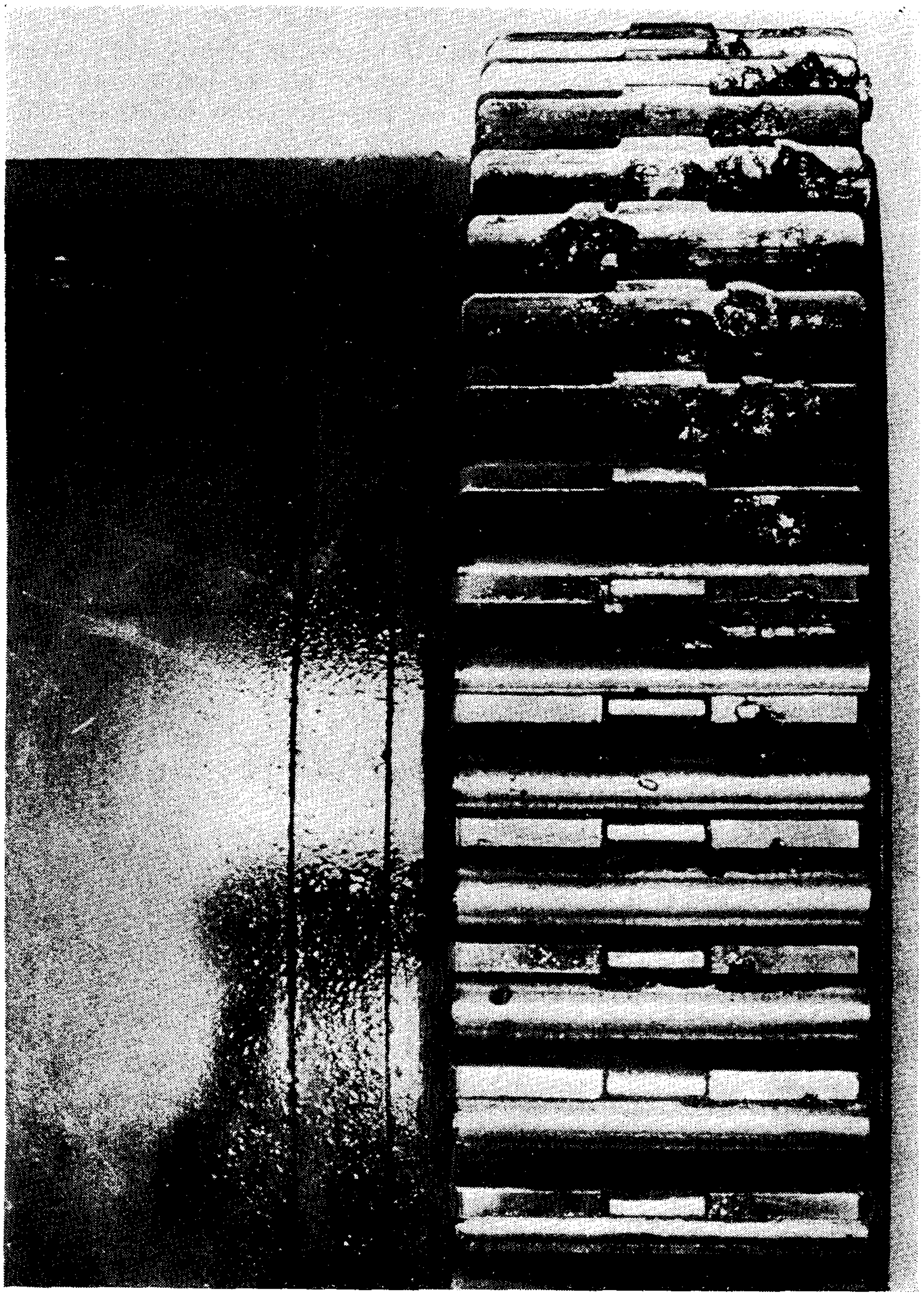


FIG. 6—SHOWING DAMAGE CAUSED TO A NITRIDED FINE TOOTH COUPLING BY TURBINE/GEARING MISALIGNMENT IN A FRIGATE

the L.O. cooler and filter bank on the gearbox. This reduces the number of flexible connections required to the supply and return L.O. pipes for the prime mover and the sea water connections for the L.O. cooler. It is also possible to mount the centrifuges on platforms carried by the gearbox.

This development is in keeping with the modern trend towards packaging or moduling and facilitates the manufacture of lubricating oil systems under 'clean' conditions.

MAIN TURBINE FLEXIBLE COUPLINGS

Maintenance of the alignment between turbines and gearing is the combined responsibility of both the component designers and the installation designer. The installation designer must know the lines of action and magnitudes of the forces on the seatings due to the weights, thrust and torque reaction of the main machinery, before he can design the seating, and for the best result the component designers must know or be advised by the installation designer, where the hull structure can best support the loads they will apply and what deflections the couplings can withstand.

Face crowning, improved lubrication and nitrided surfaces have all been tried for tooth couplings in naval vessels, while Platt and Strachan (Ref. 3) have recommended low tooth loadings and adequate lubrication for merchant designs. The author, however, believes that highly loaded fine tooth couplings have an almost impossible task in providing for axial movement against the friction of the torque load, particularly when some misalignment is almost bound to exist and axial vibrations are liable to occur.

Because the use of a relatively long torque tube reduces the angular misalignment imposed on each flexible coupling under conditions of parallel misalignment, designers often forget that in cases of tilting of turbine or gearcase, the total angular misalignment is imposed at one end of the coupling and is unaffected by the torque tube length.

FIG. 6 is a photograph of a frigate's fine tooth coupling with barrelled, crowned and nitrided teeth, which was found to be welded solid on checking, after reaching about half power on contractors sea trials. This could be removed only by end jacking after the torque tube had been cut through by drilling. In this case, the lubrication arrangements were quite adequate and the failure was later proved to have been due to misalignment when loaded. Measurements of turbine/gearing relative movements were subsequently carried out in a number of ships of the Class and the introduction of an initial cold misalignment produced acceptable conditions at full power.

Angular misalignment of just over 4 minutes of arc, e.g., 1/16 inch in $4\frac{1}{2}$ feet, produced the failure of the fine tooth coupling shown in FIG. 6, whereas a multiple steel disc type coupling has withstood a misalignment of 20 minutes of arc for 10^7 cycles. Disc type couplings have the advantage of:

(*i*) Simplicity—no requirement for lubricating oil supplies
(*ii*) The difficulties of dynamic balancing are eased by the radial stiffness
(*iii*) Ability to withstand high misalignment
(*iv*) Ability to withstand axial vibration without fretting.

CLUTCHES AND BRAKES

Automatic Clutches

Clutches of an elementary type were frequently found in the cruising turbine drives of cruisers in the pre-war Royal Navy. Following on this experience an automatic clutch actuated by a frictional speed sensing device was used in the cruising turbine drive of the Y. 100 machinery. Due to unsatisfactory performance caused by shuttling in and out, a lock-in/lock-out feature was specified for the clutch design for the COSAG powered *Tribal* and *County* Classes, because an unlocked clutch would be liable to shuttling under conditions of torsional vibration and torque reversal such as might occur when manoeuvring astern and due to seaway effects.

It is interesting to note that a simple Sprag type clutch of Borg Warner manufacture is reported to be giving satisfactory service in the '*Peder Skram*' which, however, has a C-P propeller and rotation is therefore unidirectional. It is not known whether as yet there are any marine applications of the Maag clutch which has automatic synchronizing but, unlike the SSS clutch, is not self-shifting and is moved to the engaged position by a high pressure hydraulic thruster. As in the case of SSS, the Maag unit includes a locking feature.

Friction Clutches

The gear type clutches discussed in the previous paragraphs are particularly suitable for high-speed turbine drives. However, friction clutches, which are in general suitable only for the lower range of speeds, are more suitable for the connection of Diesel engines because of the ability to slip during engagement. There are two types of clutch duty to be considered for a Diesel drive. If the ship has a C-P propeller then the clutch is only required to connect the engine either at idling speed or a synchronized speed and there should be no difficulties. However, if there are two Diesels, and it is intended to run one engine ahead and the other astern and manoeuvre by appropriate manipulation of clutches, the clutches may be adequate for manoeuvring at a reduced power level, but a crash astern manoeuvre from a higher ship speed may well wreck the clutch.

While there are catalogues listing friction clutches up to about 15,000 shp at 600 rpm, the author has not come across any existing marine applications at this sort of power. This is probably not surprising as a senior representative of Lloyd's Register stated recently that friction type clutches had not been approved for powers exceeding 3,000 bhp for manoeuvring.

Disc Brakes

There are three potential applications for disc brakes in naval ships with combined machinery plants. The first case concerns ships with C-P propellers where it is prudent to provide a brake to stop the shaft under certain emergency conditions, e.g., to prevent a rope fouling the propeller. In this case where the propeller may be turning at 80 or 90 rpm in zero pitch with the gas turbine idling, the brake has only to deal with the minimum gas turbine torque and the inertia torques of the shaft system. This brake would be designed on the assumption that it will be applied only when the way on the ship is negligible and interlocks can be used to ensure that the gas turbine throttle is reduced to the idling position. In a typical frigate design a torque of 10^5 lb ft would stop the shaft from idling in just under 10 seconds with an energy dissipation of about 3×10^6 ft lb and this duty is feasible. It would not, of course, be the intention to apply this brake every time 'stop' is ordered on the telegraph, but only on the rare occasions when a turning propeller would be dangerous.

The second case is the use of a brake to provide deceleration of a prime mover to secure a torque reversal and thereby actuate a particular kind of SSS clutch known as an 'inverted clutch'. This type may be needed to bring into use the manoeuvring trains when the ship is underway. The estimated energy dissipation required in a *County* Class ship would be of the order of 5×10^6 ft lb in about $1\frac{1}{4}$ seconds and should be feasible.

The third case involves the use of a disc brake to stop the shaft with way on the ship as required during manoeuvring, so that reversal may be effected by a simple gearbox with a layshaft and dog clutches. This does not seem to be a practical proposition as it has been calculated that, to provide for a crash stop, or for 20 minutes of manoeuvring, a *County* Class ship would require a total of about fifty 4·5 ft diameter air cooled disc brakes or alternatively about 16 liquid cooled disc brakes of the same size.

FLUID COUPLINGS FOR REVERSING TRANSMISSIONS

Where the ship is propelled by unidirectional prime movers such as the gas turbine or the high speed non-reversing Diesel engine driving a fixed-pitch propeller, a means for astern manoeuvring has to be provided in the main gearbox. This usually consists of an additional gear train incorporating an idler. In order to engage or disengage the normal ahead train and the additional astern train during periods of rapid manoeuvring, it has been the practice to use a pair of fluid couplings, one associated with the ahead train and one associated with the astern train. The fluid couplings are filled and emptied as appropriate, for ahead and astern propulsion—see Refs. 4 and 5.

The power which can be transmitted by a coupling is dependent upon its speed and diameter, the relation being

$$HP \propto N^3 . D^5$$

There is therefore an incentive to run the coupling in a high speed train to keep the size down. However, centrifugal stresses may impose a speed and hence a power limitation for a given diameter. The final selection of the coupling is a compromise between torque, speed, acceptable slip and diameter and the oil flow required for manoeuvring conditions. During manoeuvring the output half of the coupling may, at some time, be stalled when changing from ahead to astern or vice versa. In this condition, the entire power being developed by the prime mover is absorbed by the oil in the fluid coupling as heat thus raising the temperature of the oil. The oil requirement of the coupling must be based on this condition to prevent overheating of the oil, particularly if there is any possibility that the stalled condition may be unduly prolonged. Unfortunately, this is the condition in which oil is somewhat reluctant to pass through the coupling, so the coupling has to be designed to pass a specific quantity of oil in the stalled condition and the amount of oil which the coupling can and does take in the normal running condition is higher.

The main disadvantage of this type of reversing transmission is the increase in total machinery weight, not just on account of the couplings, but also of the additional gear elements, the increased size of L.O. pumps, coolers, piping, valves, drain tank and storage tank. This must be assessed when considering alternative means of reversing, e.g., controllable-pitch propellers.

BEARINGS AND LUBRICATION

Journal Bearings

As a direct result of the research work on high-speed bearings sponsored by the Royal Navy in the early 1950's, most modern turbine and gearing designs employ journal bearing L/D ratios between $\frac{1}{4}$ and $\frac{2}{3}$ with the majority in the range $\frac{1}{3}$ to $\frac{1}{2}$, accepting loads of up to 500 lb/sq in., thus giving improved efficiency and reliability.

The Royal Navy has specified prefinished steel backed bearing shells (thin wall for general use and medium wall type for main wheel bearings above 12 in. diameter) for more than ten years. The adoption of this type gives many advantages, including low cost and weight, interchangeability, absence of need for bedding in, reduced bulk and weight of spares and permits repair by replacement.

Means for adjustment of at least one bearing housing in each gear element line, excluding the main wheel, are required. This may be achieved by mounting the bearing shells in an eccentrically bored sleeve or by means of a bearing housing which is fitted in a planed location in the gearbox, with adjustment in the horizontal and vertical planes provided by machined spacing pieces.

For naval gearing, the usual temperature limits for safe operation of journal bearings are taken as 180 degrees F. (82 degrees C) oil outlet temperature or 240 degrees F (115 degrees C) whitemetal temperature. To ensure that these values are not exceeded the usual diametral clearance ratios are as follows:

Bearings at HP turbine speed	0·0022 to 0·0025
Bearings at LP turbine speed	0·002
Bearings at intermediate speeds	0·0015
Bearings at propeller speed	0·00075 to 0·001

Main Thrust Bearings

The problem of excessive axial vibration of propeller shafting was first encountered in naval ships and means of overcoming it either by thrust block positioning or by resonance changers are described by Rigby (Ref. 6) and Goodwin (Ref. 7). The same problem has now arisen in the Merchant Navy following the recent rapid advance in installed powers, and Couchman (Ref. 8) has described recent work in this field.

It is usual naval practice to demand the provision of thrust measuring equipment in association with the main thrust bearing. Once this equipment is provided it is prudent to specify that it be suitable for use with resonance changing equipment and the fitting of the latter, should it prove necessary, is comparatively easy. This eliminates the need for positioning the thrust block at otherwise unsuitable positions, together with heavy, stiff structural support work.

Lubrication

There is a tendency in the design of both naval and merchant forced lubrication systems to overspecify the lubricating oil flow which results in:

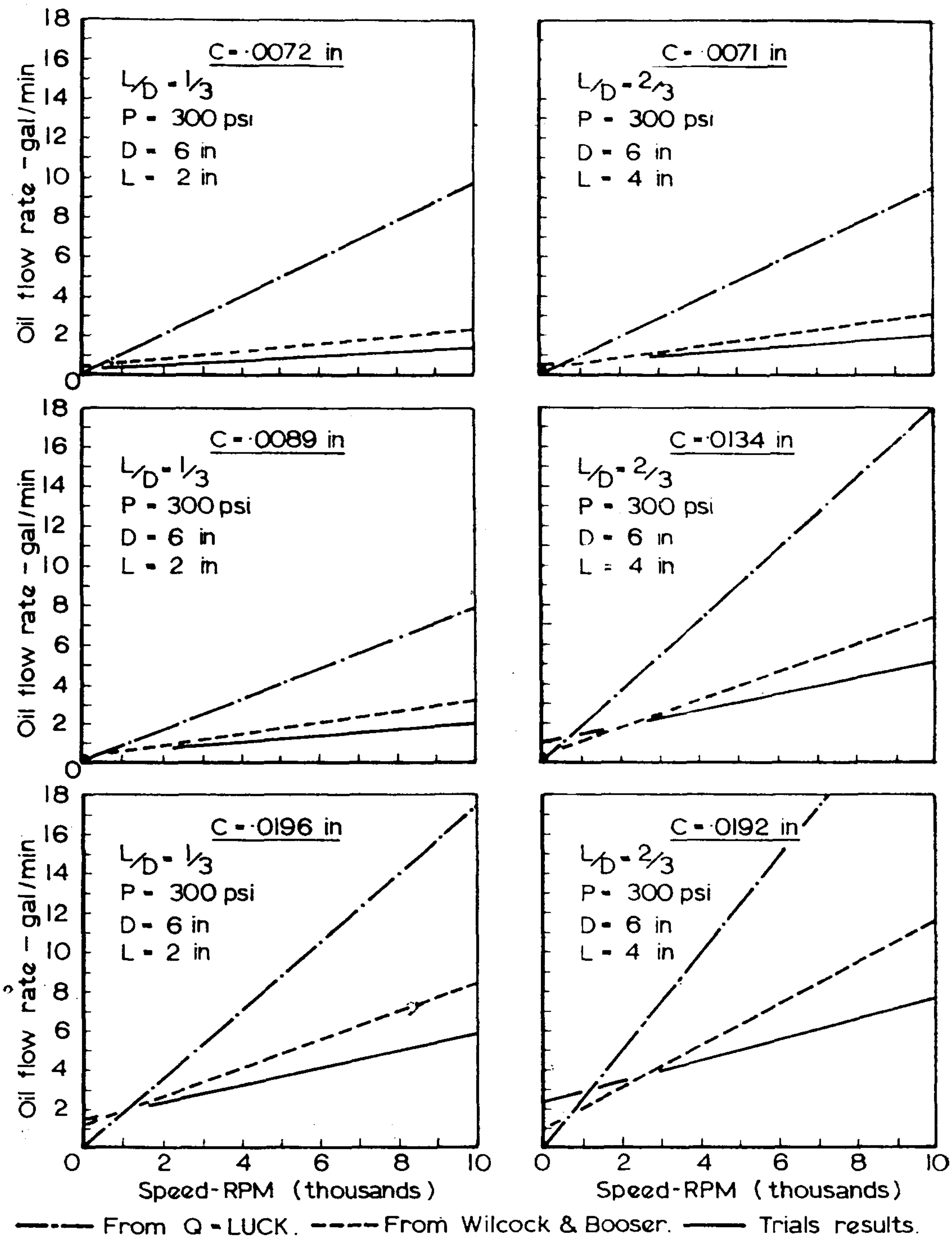
(*i*) Larger and heavier pumps, filters, coolers, pipes and valves
(*ii*) A larger drain tank and correspondingly larger storage tanks
(*iii*) A loss in efficiency
(*iv*) An increase in capital cost.

Normally the drain tank size is specified on the basis of the working level corresponding with 2 minutes' supply of oil at the design flow although several designs have been satisfactory with less than 1 minute's supply, notably the Y.100 type of which nearly 200 shaft sets are fitted throughout the world.

When it is desired to reduce the L.O. requirements, it is usually profitable to review first the thrust bearing requirements. Recent full scale tests sponsored by M.O.D. (N) have shown that when certain thrust bearings were operated with the maker's recommended oil flow, the power losses were about twice the maker's estimate. However, when the bearings were operated under the same conditions but half the recommended oil flow, operation was satisfactory.

There are many varying methods in use in the industry for estimating the oil requirements of journal bearings, thrust bearings and gearing sprayers. In design studies, it is sometimes found that there is 100 per cent difference in the estimated oil quantities by different firms and that the ratio of oil to sprayers to that to bearings may vary from $\frac{1}{3}$ to $\frac{2}{3}$ in designs to meet the same specification. One of the most consistent methods of estimation for journal bearings is that of Wilcock & Booser, but tests have shown that for the type of bearing with which we are concerned the constants need adjustment to give lower quantities at higher rpm and larger quantities at lower rpm. FIG. 7 illustrates this.

The controversial subjects of filtration, flushing arrangements, EP additive oils and the minimum safe inlet pressure to bearings and sprayers are beyond the scope of this article. However, it should perhaps be emphasized that it is the quantity of oil reaching a bearing that matters; this depends largely



Symbols used	
Q = Total oil flow rate - gal / min.	K = 1·7 for L/D ≤ 1/3 1·3 for L/D > 2/3
L = Axial length of bearing - in	P = Bearing pressure - psi
U = Journal speed – ft / sec.	D = Bearing diameter – in
C = Diametral clearance in	L/D = Length / diameter ratio

FIG. 7—CALCULATED AND MEASURED OIL-FLOW/SHAFT-SPEED RELATIONSHIPS FOR VARIOUS BEARINGS

on the bearing design, its running speed and clearance rather than on the oil supply pressure. Bearings have been run satisfactorily even at full power with no oil pressure showing at the inlet; trials have been carried out satisfactorily at low powers on sets of gearing with manifold pressures down to 3 lb/sq in., some pressure being required to supply oil through the gearing sprayers.

REDUCTION OF NOISE

Noise Generation

Noise Related to Design

The load is taken and released by successive gear teeth at tooth contact frequency and this is associated with deflections in the teeth themselves. Unless allowance is made in design for softening the impact of engagement and disengagement, the impacts will tend to cause vibration and noise. Tip and root relief on the profile, as well as end relief on the helix, are important means of reducing the magnitude of these impacts. In single helical double reduction gears, taking into account end relief, the number of overlaps is beneficially as high as possible consistent with satisfactory thrust arrangements and the author believes that this number, taking into account reliefs, should be an integer.

Noise Due to Errors

Any inaccuracies in manufacture will lead to periodic variations in load between the tooth surfaces and give rise to additional noise. The various errors and their effect on noise are summarized as follows:

(*i*) Cumulative pitch errors, giving noise at rotational frequency or multiples of it.
(*ii*) Adjacent and short span errors, giving noise at rotational frequency or multiples of it.
(*iii*) Profile errors that give noise at tooth engagement frequency or multiples of it.
(*iv*) Tooth contact errors, giving hardness of contact at the end of the teeth and leading to noise at tooth engagement frequency or multiples of it.
(*v*) Errors in roundness and concentricity of journals which produce noise at rotational frequency or multiples of it.
(*vi*) Errors in balance, giving noise at rotational frequency.
(*vii*) Errors in surface finish giving noise in a continuous spectrum which may be important if there are easily produced resonant frequencies of structural members of the gearbox.

Adjacent and short span pitch errors are probably more important than cumulative pitch errors since the impact effect is liable to be more severe.

Manufacture of Gearing

Accuracy

The accuracy of all naval propulsion gears transmitting more than 1,000 hp is required to comply with B.S. 1807, Part 1, Grade A1, subject to the amendments recently published by M.O.D.(N) which tighten the requirements for:

(*i*) Concentricity of flanges and journals.
(*ii*) Parallelism of journals.
(*iii*) Roundness and co-axiality of journals.
(*iv*) Limits of wobble of couplings, end faces and thrust faces.

(*v*) Cumulative and short span errors.

(*vi*) Single pitch errors.

(*vii*) Pressure angle as indicated by base pitch measurement.

Stringent requirements are of course specified for the testing and certification of hobbing and grinding machines.

Balancing

Until about ten years ago, only the parts running at turbine speed were subjected to dynamic balancing; the other parts were given only a static balance. In general, the effect of the requirements for static and dynamic balancing specified in Chapter 12 of G.R.M. (E). is to limit the unbalance force at each bearing, expressed as a percentage of the weight of the rotating part, to 0·5 per cent for main gearwheels running below 200 rpm and to 4 per cent for a typical pinion running at 5,600 rpm.

A further refinement is the provision in new designs to permit in-place balancing of the high speed flexible couplings and torque tube assembly between the turbine and its input pinion in the main gearing.

Noise Reduction

Noise Reduction at the Source

Particular attention should be paid to the design of tip and root relief and to end relief, with a view to minimizing the shock of engagement and disengagement of the teeth. Tests are being carried out by NAVGRA on this aspect. The end reliefs should be of the generator type and apply the load gradually over a reasonable length of tooth. Tooth pitches chosen for the primary and secondary gears should be the minimum that can reasonably be adopted bearing in mind root bending stresses. In profile grinding, care should be taken to ensure the best surface finish so as to cut down friction noise to a minimum. Even when the inaccuracy has been confined to the very small limits described above, gearing still remains a 'noisy beast' transmitting vibration through its supports and noise through the air, and something more is required.

Noise Isolation

The ideal method of noise isolation is to mount the complete gearbox on flexible mounts and completely enclose it in a suitable acoustic hood, but this is not often possible due to the space required. Partial enclosure of the gearcase by means of panels attached to stiffening ribs running round the case can result in over 80 per cent of the noise emitting surface being shielded. Theoretically, this can reduce airborne noise by some 10 to 15 dB but more practical tests need to be carried out to verify the best gearcase treatment.

Ideally the flexible mounts should have a stiffness which is high at the very low frequencies corresponding with motion in a sea-way, preferably infinite (zero movement) under torque loading and low at frequencies above about 5 c/s. The mount damping should be high at the natural frequency, preferably falling rapidly above the natural frequency. Alternatively, the damping could be amplitude sensitive, since the amplitude associated with movement at the natural frequency is several orders of magnitude greater than that associated with vibrations at higher frequencies. The seating stiffness should be as high as possible, since however flexible the mountings are, there will be some force transmitted by them.

In the next section a description is given of the Constant Position Mounting System which fulfils the requirements of the third and fourth paragraphs above.

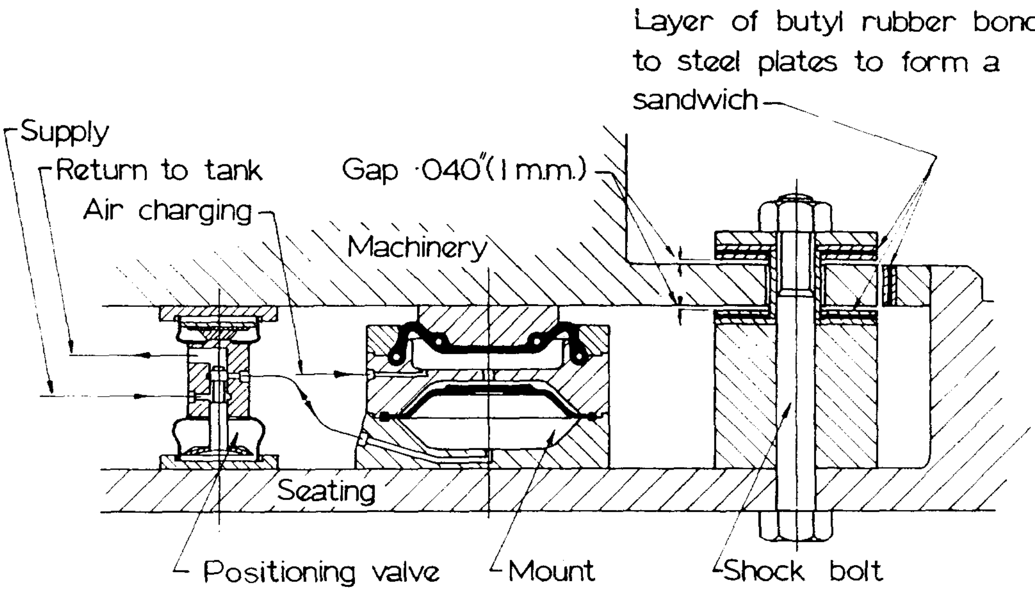


FIG. 8—DIAGRAMMATIC SECTION THROUGH CPM POSITIONING VALVE, MOUNT AND SHOCK STOP

CONSTANT POSITION MOUNTING SYSTEM

The Constant Position Mounting System (CPMS) is an active mounting system which has been under development by Y-A.R.D. since 1961. A number of shore trials are being carried out preparatory to extensive use in a variety of seagoing applications.

System Description

The mount itself, see FIG. 8, is an air mount, the air being sealed into the mount by a specially developed high pressure fabric backed rubber diaphragm, capable of operating up to a maximum normal working pressure of 1,000 lb/sq in. and with a burst pressure in excess of 4,000 lb/sq in. Position control valves sense the position of the mounted item relative to its seating. These valves vary the pressure in the mount such that the force applied by the mount to the mounted item automatically counteracts any tendency to depart from the design relative position.

Both hydraulic (see FIG. 6 of Ref. 9) and pneumatic control systems have been developed, the choice of control system depending on the system requirements. In the former, a pressure-controlled swashplate pump supplies oil to the control valves at the design maximum pressure of up to 1,400 lb/sq in. The control oil is separated from the mount air by a flexible membrane in a separate compressor chamber between the control valve and the mount; this may be incorporated in the mount as shown in FIG. 8, and the air in the compressor chamber is connected to that in the mount through a stabilizing orifice.

With the air-control system the upper system pressure will generally be limited by the availability of suitable air compressors to about 500 lb/sq in., obtainable with a two-stage compressor. Either open-cycle (exhausting to atmosphere) or closed-cycle operation, using the compressor as a boost between lower and upper system pressures, are possible, depending on the system requirements.

Control can be applied in one (vertical), two (vertical and athwartships) or three directions. In general, three-point vertical control, i.e., three control valves each controlling one or more mounts, is used, and should the requirements warrant the additional complexity, athwartships and fore/aft double-acting control valves controlling double-acting differential mount pairs control the position of the mounted item in all six degrees of freedom.

System Operation

When the system is hydraulically controlled, the mounts are first charged to a pre-determined air pressure. The leakage rate is so low that only occasional checking and topping-up is necessary. This pre-charging is, of course, unnecessary when the control system is pneumatic.

Operation of the system is then entirely automatic. On switching on the supply pressure, the mount pressure rises until the mounted item is in its design position, when further flow to the mounts is cut off by the control valve. Movement of the mounted item away from the design position causes the control valves to connect the mount either to supply or exhaust, thus varying the mount pressure to apply a position correcting force to the mounted item.

If the design sensitivity of the control valves exceeds a certain limit, the system becomes unstable, tending to oscillate at the system natural frequency about its equilibrium setting. Much of the original development effort was spent in developing the control system to give great sensitivity, hence resistance to rolling and pitching forces, while retaining an adequate stability margin.

System Characteristics

The outstanding advantages of the CPMS are derived from the variation of stiffness and damping with frequency which the system exhibits. Under static conditions (torque variations and static load variations), the system is infinitely stiff within the pressure capabilities of the mounts. At rolling and pitching frequencies, the stiffness is approximately 10 times that at the 'natural' frequency and, hence, the movements of the mounted item are reduced by a factor of 10 compared with those of a conventionally mounted machine. Above the 'natural' frequency, the attenuation approaches the attenuation which would be obtained from an 'ideal' mount having the same 'natural' frequency. These three features make the CPMS particularly suitable for mounting gearcases, where large misalignments would be intolerable.

The damping is strongly frequency-dependent. While at frequencies just above the 'natural' frequency the damping tends to zero, at the 'natural' frequency the mount is very highly damped having an amplification factor of about 2·5 compared with an amplification factor of about 15 exhibited by a conventional mount. This is the feature which makes the mount specially suitable for mounting the gas turbine that is used in current COSAG installations. When this engine was mounted on conventional rubber acoustic mounts it vibrated so wildly that it destroyed the flexible coupling within seconds before the gas turbine could be shut down.

The CPMS restricts the relative movement of the mounted item under a sea-way (say ± 20 degrees roll) to about $\pm 0{\cdot}02$ in. Adequate margin must be allowed for thermal distortion or growth and ship working, and a clearance of about 0·04 in. is provided before the mount 'bottoms'. When the CPMS is not operating, the mounted item 'sits down' and becomes misaligned by this clearance, which is in general well within normal misalignment tolerances.

In systems subject to shock, parallel snubbing arrangements, incorporating a rubber snubbing layer to prevent impactive loading as shown in FIG. 8, may be used. Alternatively, a larger mount travel can be designed into the mount and parallel shock devices incorporated to provide any desired degree of shock protection. The shock snubbers or parallel shock devices are normally clear of the mounted item by the design clearance so that vibration attenuation is not impaired.

The normal mean operating pressure is 600 lb/sq in. and standard mounts are manufactured in nine sizes between 2·3 in. and 16$\frac{3}{4}$ in. effective diameter. The latter has a mean working load of 130,000 lb. It is, therefore, possible to mount very heavy items, for example, rafted assemblies of main and ancillary machinery with economy in mounting positions.

Summary

A mounting system has been developed and successfully tested under sea-going conditions with the following characteristics:

(*i*) Good vibration attentuation ('natural' frequency 3·5 c/s).

(*ii*) Low amplification (high damping at the 'natural' frequency).

(*iii*) Ease and constancy of alignment, i.e., no 'creep'.

(*iv*) Reduction of relative movement under a sea-way, hence of fatigue of connections to the mounted item.

(*v*) Effectively zero movement under constant force, e.g., torque or thrust.

(*vi*) Compact, high load capacity mount units.

(*vii*) Parallel devices to give small relative shock deflections without impairing the vibration attenuation.

CONCLUDING REMARKS

The features characteristic of modern naval gearing have been discussed and it has been indicated that there are acceptable solutions in respect of:

1. Gearing tooth design criteria, with the exception of scuffing.
2. Gearcase design and support.
3. Main turbine flexible couplings.
4. Automatic synchronizing clutches.
5. Friction clutches and disc brakes for limited powers, the future for higher powers being uncertain.
6. Fluid couplings.
7. Journal and thrust bearings.
8. Transmission of structural vibration and noise.

Directions in which further progress is clearly desirable are as follows:

1. The establishment of facilities for manufacturing nitrided main gear-wheels that will be capable of transmitting higher powers at lower shaft speeds and be cheaper than carburized and ground main gear-wheels.
2. The development of effective and compact acoustic treatments for gearcases to limit the transmission of airborne noise.
3. Better means of specifying lubricating oil requirements leading to economy in capital cost and better efficiency.
4. Defining the limits imposed by the possibility of scuffing.

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