Experience with Hardened and Ground Gearing in the Royal Canadian Navy*

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The Royal Canadian Navy has in commission fourteen destroyer escorts developing 30,000 s.h.p. on two shafts and which are equipped with an advanced design of hardened and ground main reduction gearing having rated tooth loadings of up to 412 K. Eleven of these shipsets of gearing were manufactured in a unique production facility which was set up by the Canadian Government in 1952 to permit the manufacture of hardened and ground marine gearing in Canada.

This paper deals with the service experience obtained with this gearing, the extensive prototype testing of the first units at Pametrada and the manufacturing experience obtained in Canada. An assessment has been made of the various factors influencing the choice of hardened and ground main gearing in naval installations which would also appear to be applicable to mercantile installations, with particular attention being paid to reliability and cost.

It is suggested that hardened and ground gearing is able to provide the maximum reliability at the minimum cost and may therefore be equally attractive for naval and mercantile installations alike.

INTRODUCTION

The International Conference on Gearing held in London by the Institution of Mechanical Engineers in 1958, focused attention on developments affecting all aspects of gearing design, manufacture and lubrication, over a wide range of applications. It was, however, in the marine gearing field that some of the most significant advances in gear design and load-carrying capacity were reported[1 and 2] and this was primarily the result of extensive development and test programmes initiated by the Admiralty over the previous decade. Substantial advancements in naval propulsion machinery demanded increases in main reduction gearing load-carrying capacity which were clearly unobtainable with hobbed and shaved or soft gearing. While the high load-carrying capacity provided by case-hardened and ground gearing was well known in the general engineering field, very little experience was available even after World War II regarding its application in high power marine machinery installations. Admiralty sponsored gear investigation, development and test programmes have since done much to determine many of the design and manufacturing limitations[3 and 4] of case-hardened and ground gearing.

The Royal Canadian Navy has been privileged to benefit considerably from the gearing work undertaken by the Admiralty and has in consequence been able to install hardened and ground main reduction gearing in each of its fourteen 30,000 s.h.p. *St. Laurent* Class destroyer escorts now in commission. Experience with these gear units has been unique in that they are believed to have the highest rated tooth loadings of any main turbine reduction gearing in service, and are also the first large marine gears to be fully carburized and hardened. The most outstanding aspect of this programme, however, is that eleven of the shipsets of gearing were manufactured in a newly established Crown-owned gear plant in Canada, thus bringing the Royal Canadian Navy into closer touch with the problems of manufacture than most marine gearing customers either experience or desire.

The scope of this paper will therefore be to review the considerations affecting the choice of hardened and ground main gearing and to present those aspects of manufacture, testing and service experience which may permit a more rational assessment of the general mercantile application of this type of gearing.

Considerations Influencing the Choice of Hardened and Ground Gearing

The requirement for maximum reliability at minimum cost applies equally well to all items of naval and mercantile machinery alike. Apart from the considerations of bearing design and lubrication, which it is not proposed to cover in this paper, main gearing reliability is largely determined by the margin provided by the gear tooth design with respect to:

a) bending strength—resistance to fracture
b) surface loading—resistance to pitting
c) heat dissipation—resistance to scuffing

Cost is determined by the manufacturing implications of the design requirements and the availability of the necessary production facilities.

For most naval gearing installations, and particularly in classes of the *St. Laurent* type, minimum weight and space is considered to be a requirement which is surpassed in importance only by that of reliability. Unlike mercantile practice, naval gearing is operated at its maximum rated power for only a small percentage of its total life and may well be operated at as low as 10 per cent power for over 80 per cent of its life. It is therefore possible to permit higher design loadings than are acceptable in normal mercantile practice.

Tooth loadings can be increased only to within the limitations of the root and surface strength as determined by the material properties. Providing the teeth are large enough to provide adequate root strength, the load-carrying capacity of soft gears is limited by the fatigue strength of the material under compressive loading. Surface fatigue strength increases directly with the ultimate tensile strength, which in turn is proportional to the surface hardness. The maximum tooth loading which can be transmitted by gearing which is finished by hobbing and shaving is therefore determined by the maxi-

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mum surface hardness which it is practicable to machine. This limiting hardness is generally accepted to be about 350 B.H.N. and is obtained by through hardening high percentage alloy steels. In mercantile gearing installations the Lloyd's K factor for tooth loading would be limited to about 120 K for a material of this hardness, but a peak design loading of up to 250 K might be permissible in a naval gear installation.

A considerable increase of surface fatigue strength and load-carrying capacity is provided by case-hardening the gear teeth, in fact the load-carrying capacity is then no longer limited by the surface fatigue strength. The design loading criterion for case-hardened gears is now recognized(2) as usually being the root strength since the bending fatigue strength, unlike the surface fatigue strength, does not increase with the ultimate tensile strength of the material. The utilization of the high surface load-carrying capacity provided by case-hardened gears depends entirely on the tooth design. In general it is found that a case-hardened gear will permit at least two and a half times the maximum root stress allowable in a hobbed and shaved gear of the same tooth design.

The root strength of carburized and hardened gears is greatly assisted by the compressive stresses created in the hardened layer due to the increased volume of material caused by the formation of martensite. This compressive stress must first be absorbed by the external tooth loading before the tooth roots are subjected to the normal tensile bending stresses. The tooth strength is thereby increased over and above the strength of the hardened case. It should be noted that surface residual compressive stress is not obtained with induction hardened gears.

The third factor influencing reliability is the resistance to scuffing which relates to the ability of a tooth surface to maintain an oil film under pressure and under sliding. The scuffing limit in soft gearing is generally above the surface fatigue limit but, as the surface hardness increases the load-carrying criterion, other than root strength, changes from surface pitting to scuffing. The avoidance of scuffing in hardened and ground gearing is generally a matter of obtaining a tooth design which produces the minimum of sliding and which has a good surface finish.

On the basis of presently accepted design limits the foregoing indicates that the highest degree of reliability and freedom from failure is obtainable from case-hardened and ground gearing.

An important milestone in establishing the value of hardened and ground gearing in naval machinery was the installation of two 27,000 s.h.p. units in H.M.S. *Diana* in 1951. These gears, which were built in Switzerland, were of the double reduction, dual tandem articulated type and were carburized and hardened with the exception of the air-hardened secondary reduction gearwheels. The tooth loadings were up to 260 K in the H.P. primary reduction and 200 K in the secondary reduction. The satisfactory experience obtained with the *Diana* gearing did much to influence the type and design of gearing selected for the *St. Laurent* Class.

The availability of domestic manufacturing facilities is a matter of prime concern in specifying naval main gearing requirements, as indeed for any defence equipment. It was such a consideration which faced the Royal Canadian Navy when it embarked on a destroyer escort programme in 1950 with the technical assistance of the Admiralty. The requirement was for all machinery to be manufactured in Canada other than the initial sets, which were to be built and tested by the designers. There being no manufacturing source of precision marine reduction gearing in Canada at all, the Canadian Government decided to establish a Crown-owned gear plant in Montreal to be operated under contract by a commercial engineering company to produce naval gearing. It was thus possible to select a type and design of gearing best suited to the naval requirement and then to set up the necessary facilities to manufacture on a production basis.

The circumstances regarding the selection of hardened and ground gearing for the *St. Laurent* Class are perhaps rather unique, since the Royal Canadian Navy was not only accepting a most advanced gear design having tooth loadings appreciably higher than the *Diana* gearing for installation in an entire class

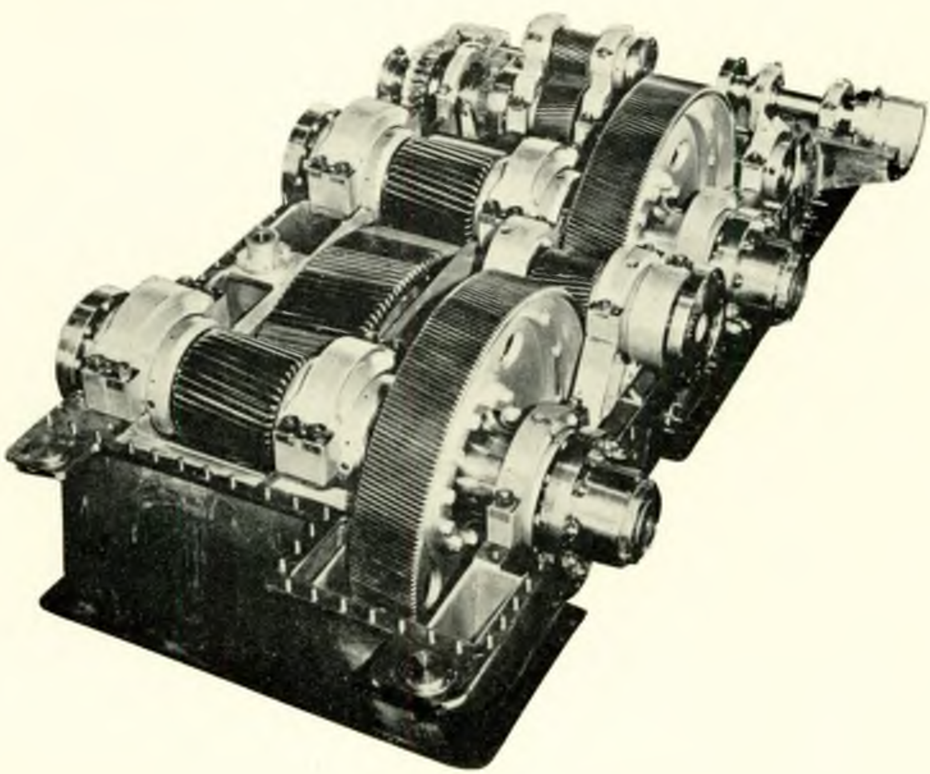


FIG. 1—St. Laurent *Class main gearing port unit*

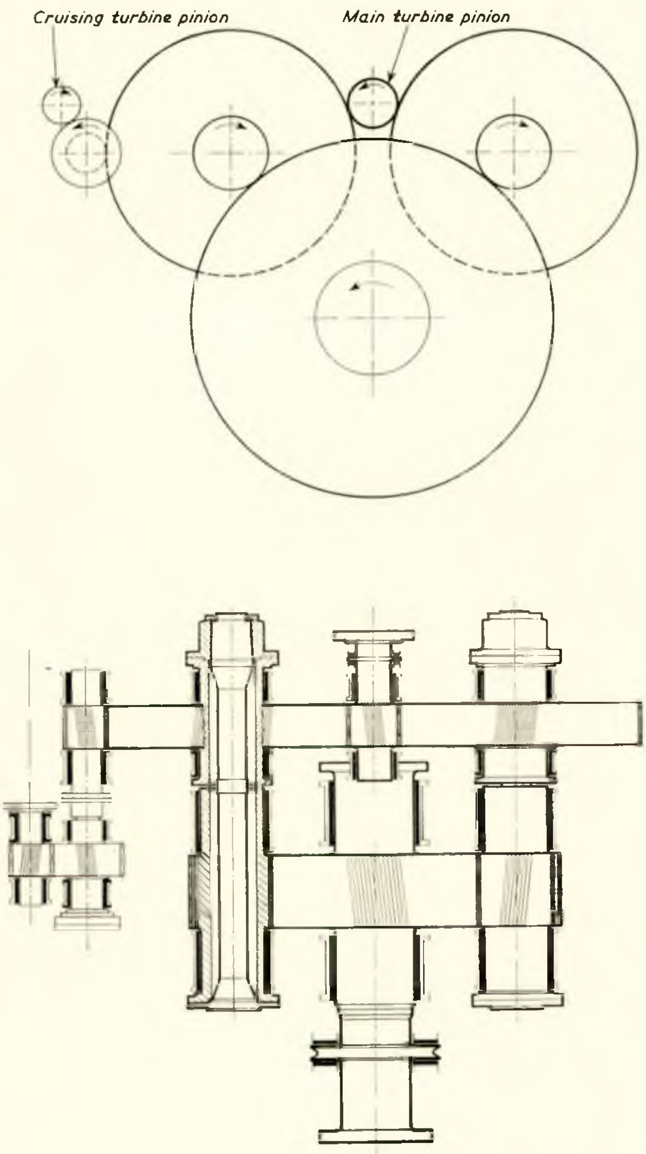


FIG. 2—*Arrangement of* St. Laurent *Class main gearing*

of ships, but it was also committed to arranging for their manufacture in the first gear plant of its type in the world, involving a type of work, techniques and processes which were completely outside all previous experience in North America.

St. Laurent *Class Main Gearing Design*

The *St. Laurent* Class main gearing design (Figs. 1 and 2) is of the double reduction dual tandem articulated single helical type incorporating a triple reduction cruising turbine drive. Each unit transmits 15,000 s.h.p. All pinions and gearwheels are carburized, hardened and ground. Details of the original gear design and materials for the main turbine drive are given in Tables I and II respectively.

Pametrada Shore Trials

In 1951 the port unit of the first shipset of gearing was sent to Pametrada for full power testing under a dynamometer load in conjunction with the complete main and auxiliary machinery installation. The gearcase was installed on simulated engine room seatings and on a three point support. The three chocking areas are shown by the double hatching in Fig. 3(a).

Using an extreme pressure oil, in accordance with the current Admiralty running-in practice, the gearing was gradually run up to about 94 per cent full power when overheating occurred in the secondary reduction gearwheel forward bearing. The condition of the gearing was excellent, but it was decided that further trials were necessary to prove the ability of the gearing to run on standard turbine oil OM88, which had been specified for service. At 85 per cent power, with OM88 oil, light scuffing was observed on the secondary reduction pinion tooth tips (Figs. 4 and 5) and at the roots of the main gearwheel (Fig. 6). The OM88 oil was then replaced with extreme pressure oil for the remaining trials on this set of gearing. The gear unit was gradually brought up to full power loading after which the gear tooth scuffing was found to have polished over at one end but to have extended at the other end. In addition, both the forward and after main gearwheel bearings were found to have wiped.

The failure of the main wheel bearings led to a series of bearing design changes affecting the angular location and number of oil inlets and the type of backing material. The first of the revised design bearings was tested concurrently with a gearcase deflexion test, for which dial indicators mounted on a special frame were used to measure movements of each corner of the gearcase. The measured gearcase distortion, permitted by the three-point support at high powers, was considered to be excessive and consequently a contributory cause of the gear scuffing. It was decided that the after corners of the gearcase should be chocked to reduce the distortion at high powers. Additional torque resisting chocks were fitted under each side of the gearcase, to provide a five-point support as shown in Fig. 3(a) by the single and double hatched areas. The gearing subsequently satisfactorily completed a four hour

TABLE I.—*St. Laurent* CLASS MAIN GEARING DESIGN DATA.

	Original design				Revised design			
	Primary		Secondary		Primary		Secondary	
	Pinion	Wheel	Pinion	Wheel	Pinion	Wheel	Pinion	Wheel
Number of elements	1	2	2	1	1	2	2	1
Number of teeth	37	190	29	143	43	221	38	187
Pitch circle diameter, inch	8·98	46·13	13·61	67·10	8·98	46·14	13·63	67·08
Face width, inch	7·875		13·75		7·875		13·75	
Tangential load/inch face width, lb/in.	2402		4664		2406		4666	
Reduction ratio	5·135		4·931		5·140		4·921	
Overall ratio	25·32				25·29			
K factor	320		412		320		412	
Helix angle, deg.	10		6		10		6	
Normal pitch, inch	0·742		1·463		0·640		1·114	
Normal pressure angle, deg.	15		15		23		20	
Addendum, inch	0·241	0·206	0·584	0·347	0·221	0·181	0·377	0·326
Addendum ratio	1·168		1·685		1·224		1·156	

TABLE II.—GEAR MATERIALS.

	Original material (Swiss)	Modified A.I.S.I. 3310 (D.E.W. 3610)	A.I.S.I. 3310
CHEMICAL COMPOSITION			
Carbon, per cent.	0·08	0·08–0·14	0·08–0·13
Manganese, per cent.	0·44	0·40–0·60	0·45–0·60
Silicon, per cent.	0·25–0·30	0·20–0·35	0·20–0·35
Nickel, per cent.	3·20–3·50	3·00–3·25	3·25–3·75
Chromium, per cent.	0·54–0·60	0·40–0·60	1·40–1·75
Phosphorus, per cent.		0·025 maximum	0·025 maximum
Sulphur, per cent.		0·025 maximum	0·025 maximum
MECHANICAL PROPERTIES (CORE)	After water hardening	After water hardening	After oil hardening
Ultimate tensile strength, lb./sq. in.	143,000–163,000	120,000 minimum	120,000 minimum
Yield point, lb./sq. in.	114,000–134,000	85,000 minimum	85,000 minimum
Elongation, per cent.	8–10 (on L=10d.)	12 min. (on 2 ins.)	12 min. (on 2 ins.)
Reduction in area, per cent.	45 minimum	40 minimum	40 minimum
Izod impact, ft./lbs.	35 minimum	30 minimum	30 minimum

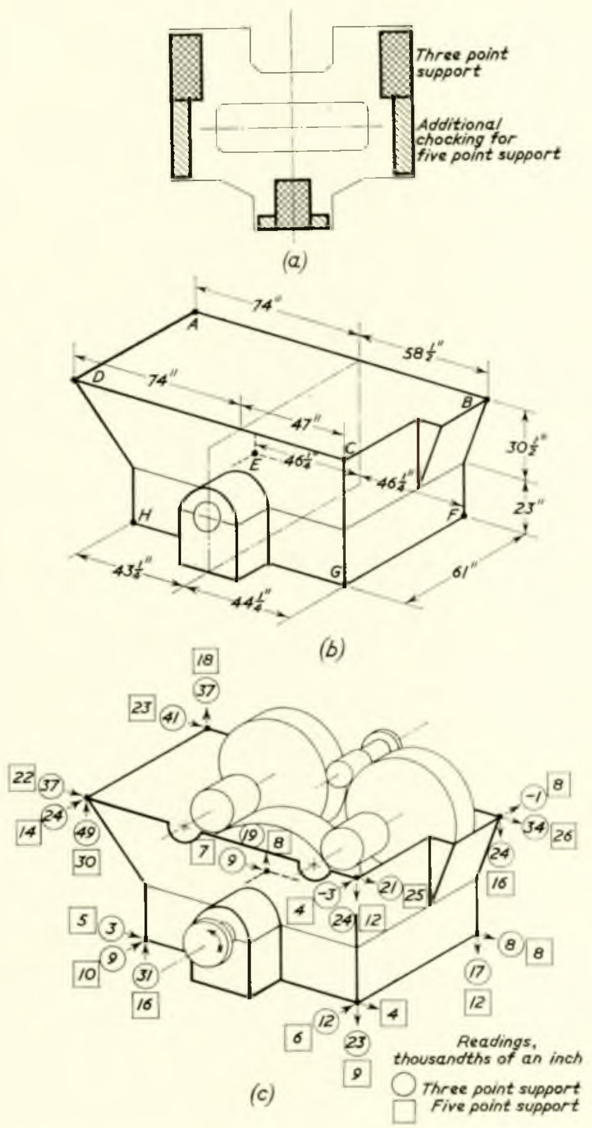


FIG. 3(a)—*Gearcase chocking arrangement*
(b)—*Reference points for measuring gearcase distortions*
(c)—*Gearcase distortions under 130 per cent full power torque*

full power proving trial and a short trial under 130 per cent full power torque.

This first gear set completed more than 400 running hours over the full power range during the course of trials designed to prove the entire main and auxiliary machinery installation. The earlier scuffing on these gears was found to have polished over quite satisfactorily after the completion of all the prototype trials (Figs. 7 and 8).

The ability of this gearing design to run without scuffing on standard turbine oil was rather obscured by the possible effects of the distortion under a three-point support and the wiped main gearwheel bearings. Since this ability could not be conclusively proved on gearing which had in fact already scuffed, it was decided to test the port unit of the second shipset of gearing on standard turbine oil only. The oil selected for these trials was OM100.

After running-in under low power loading for 20 hours the

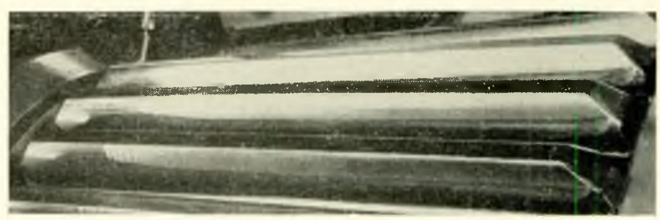


FIG. 4—*Scuffing on inboard secondary pinion of 1st unit after running on OM88 oil*

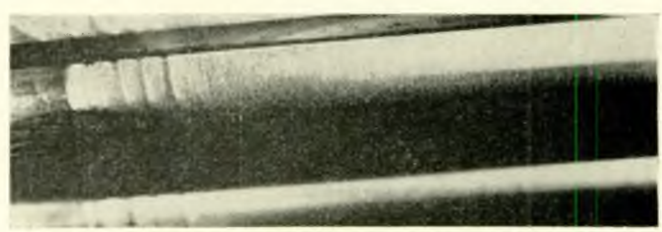


FIG. 5—*Enlarged view of scuffing in Fig. 4*

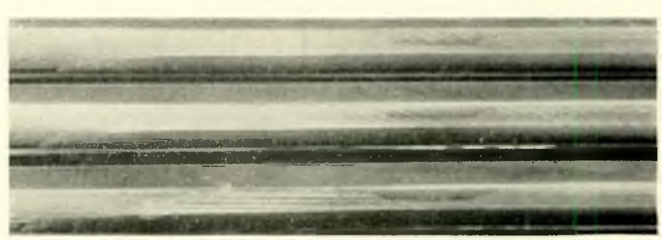


FIG. 6—*Scuffing on main wheel of 1st unit after running on OM88 oil*

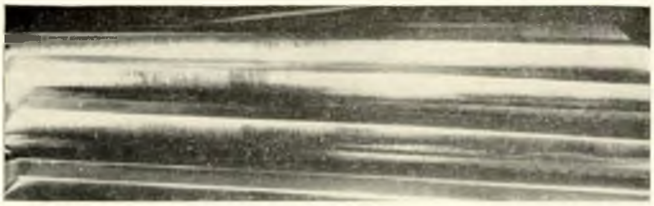


FIG. 7—*Inboard secondary pinion of 1st unit on completion of trials with E.P. oil*

FIG. 8—*Main wheel of 1st unit on completion of trials with E.P. oil*

FIG. 9—*Scuffing on inboard secondary pinion of 2nd unit after running on OM100 oil*

FIG. 10—*Scuffing on main wheel of 2nd unit after running on OM100 oil*

second port unit was worked up to 50 per cent power on a five-point support. Scuffing was observed commencing at the after ends of the secondary reduction pinions. Further increments of loading above the 50 per cent power increased the scuffing until the trial was stopped at 80 per cent power (Figs. 9 and 10).

It was concluded from the trials on the second unit that the gear design was not capable of operation on standard turbine oil without scuffing. While steps were immediately taken to carry out a design investigation, it was necessary to confirm that this highly loaded gear design was capable of completing all trials up to full power without distress when used only with an extreme pressure oil on a five-point support. The port gearing unit from the third shipset was accordingly sent to Pametrada for testing. All trials including the 130 per cent torque trial were satisfactorily completed by this unit on a five-point support using extreme pressure oil. Since both the second and third unit had been operated on a five-point support and that scuffing had occurred only in the unit running with standard turbine oil, it was decided to test the significance of the different gearcase distortions between the three and five-point methods of support. The third gear unit was subsequently run up to full power on a three-point support without any sign of tooth distress, thus proving that the gear design was capable of being run satisfactorily on either three or five-points providing extreme pressure oil was used. The gearcase distortions which were measured during the trials on the third unit with three and five-point supports are shown in Fig. 3(b) and (c).

It may be noted that the gearing in each of the three units tested at Pametrada have given a completely trouble-free performance in service with no sign of scuffing or any other deleterious effects. The second port unit was returned to the manufacturer for inspection and re-grinding of the scuffed components before the gearing was installed in the ship.

Revised Gear Design

Following separate design investigations carried out by the gear manufacturer and the Admiralty Vickers Gearing Research Association, it was concluded that the specific sliding or the slide/roll ratio in approach and recess flanks of the secondary reduction gear teeth were too high for the design tooth loading and peripheral speed. A revised design (Table I) was developed by the manufacturer to permit operation at all powers on standard turbine oil OM100. The design was developed in consideration of scuffing criteria determined by AVGRA from a disc-testing programme and involved an increase in pressure angle and diametral pitch.

Although the Pametrada trials produced no scuffing in the primary reduction gear train when running on OM100, the tooth design, when considered on the same basis as the secondary reduction tooth design, was found to be marginal. A revised primary reduction tooth design was therefore developed, again with an increased pressure angle and diametral pitch. As mentioned later, the revised gear design has been entirely satisfactory in service but the 23 deg. pressure angle in the primary reduction has been found to cause appreciable difficulty in manufacture. Later gear units in the class have been fitted with the revised secondary reduction gear design but with the original primary reduction design.

Crown-owned Gear Plant

The Crown-owned gear plant in Canada was equipped to permit the manufacture of a full range of naval main and auxiliary turbine gearing up to 142in. diameter, which is the capacity of the largest grinding machines. The gears are cut, ground and inspected on precision equipment installed in a temperature controlled, windowless but well lit shop. The plant includes a separate heat treatment shop in which are installed vertical gas carburizing and reheating furnaces of up to 75in. diameter capacity. The large gearwheel furnaces are heated by oil fired radiant tubes while the smaller pinion furnace is electrically heated. The atmosphere and carburizing medium in the furnaces are controlled by propane-fed gas generators of the exothermic and endothermic types. A dew-point indicator is used to control the carburizing potential of the gas.

Manufacture in Canada

General Design Considerations affecting Manufacture. The high load carrying capacity provided by case-hardened gear teeth is, needless to say, by no means solely dependent on the surface hardness. Research which has so far been carried out in this field points to the significance of the hardness curve between the tooth surface and the core, the type of case and the method of hardening as factors influencing the tooth strength and fatigue resistance to bending. It is apparent that tooth root strength, or resistance to shock loading, decreases as the case depth increases. The bending fatigue strength on the other hand increases with the case depth up to a maximum when the case depth/module ratio reaches a value which is estimated by various researchers as ranging from ·07-·23. Effective case depth is the depth of case measured from the surface having a hardness of not less than a certain specified value. The practice in Canada has been to measure effective case depth above the 500 DPH level on the hardness curve.

The consideration which arises is that the case depth must be maintained to within very close limits in order to achieve optimum strength and load-carrying capacity. The process of carburizing and hardening gear rims does inherently involve growth and distortion which must be satisfactorily combated in order to make the process practicable. Excessive gear rim growth and tooth distortions must be removed by grinding, and it is not only necessary to keep grinding stock down to a minimum in the interest of reasonable manufacturing time, but there is also a very definite limit on the amount by which the case can be reduced. In addition to the consideration of maintaining a minimum depth of case, any growth or shrinkage of the gear rim affecting the design outside diameter may also affect the addendum ratio to the point of reducing the active profiles of the mating teeth. Gearwheel addendum changes with the growth or shrinkage of the wheel diameter; thus an oversize gear wheel would have an increased addendum and would necessitate an offsetting reduction in the addendum of the mating pinion. This would result in increased sliding on both the pinion and wheel approach flanks, which could well reduce the resistance to scuffing to a critical level.

The manufacture of hardened and ground gearing will be seem to hinge entirely on the ability to manufacture gearwheels to a high degree of accuracy with a surface hardness and case depth each to within closely prescribed limits. The manufacture of case hardened pinions does not involve the same extent of unpredictable dimensional variations during heat treatment as the gear wheels. The distortions and growths which do occur are sufficiently small or predictable to permit the pinion to be manufactured to suit the finished gear wheels.

It may therefore be of interest to note the principal aspects of the manufacturing procedure used in Canada for the manufacture of the primary and secondary reduction gear wheels for the *St. Laurent* Class main gearing, with a view to providing a basis for assessing the manufacturing problems and economics of marine hardened and ground main propulsion gearing.

The requirement is to produce accurately ground gear wheels and pinions having tooth surface hardnesses ranging from 62 Rockwell C (739 DPH) on the pinion teeth to 59 Rockwell C (675 DPH) on the gear teeth with a minimum acceptable value of 57 Rockwell C (636 DPH). The depth of case measured with respect to 500 DPH is required to be not less than ·040in. and to have a fine grain tempered martensitic micro-structure free from boundary carbides.

Material

In order that the gearwheels may be finish ground as close as possible to the required design dimensions, with the minimum removal of stock, it is necessary that the material be selected not only for its hardenability and provision of a ductile core, but also for its stability and predictable behaviour under heat treatment. The overall properties of the material used in the original Swiss-made gearing for the *St. Laurent* Class programme—a low carbon nickel chrome water-quenching steel—have been hard to match in comparable steels available in North America. The specified composition and properties of this material, which are given in Table II were most closely matched by using a modified AISI 3310 carburizing steel which has been designated as DEW 3610. Forgings of this material, which has been used for the majority of pinions and wheels manufactured to date, have not permitted the degree of dimensional control considered desirable during heat treatment, there even being a significant variation between forgings of the same heat. This experience is now known to be largely due to the inadequate restriction of grain size in the forgings. Recent experience has been with AISI 3310, a standard oil-quenching steel which minimizes the risk of cracking generally associated with water-quenching steels. It is believed that AISI 3310 should reduce distortions but experience with this material has not been sufficiently extensive to date to judge the degree of improvement. The heat treatment described in this paper is that which is currently in use for AISI 3310.

Pre-quenching

The amount that a carburized gear rim will grow in diameter during hardening may vary considerably from forging to forging even where they are from the same heat. To provide a reasonable indication of how much this growth may be, the gear rims are first subjected to a pre-quenching in the condition as received from the forging supplier. AISI 3310 rims are heated to 1,480 deg. F. and held for four hours. The rim is quenched in oil at 120/150 deg. F. and allowed to cool to shop temperature. DEW 3610 gear rims are heated to 1,550 deg. F. and quenched in a 5 per cent caustic soda solution.

Gear Cutting

Gear rims are machined to an outside diameter, prior to gear cutting and carburizing, which makes allowance for the subsequent removal of the carburized case on the tips of the teeth, for a slight shrinkage during carburizing and for the anticipated growth during hardening as determined from the pre-quench. The pre-quenching increases the hardness of the material to the detriment of its machinability, but subsequent annealing has been necessary in only a few instances and only with DEW 3610 material. The gear teeth are cut on rack shaping machines in three principal operations; roughing, finishing and protuberance cutting. Protuberance cutters are used to form the tooth roots and to confine the grinding stock allowance to the tooth flanks.

Carburizing

The gear rim and the control test pieces, which are tooled out of the rim bore, are heated in the carburizing furnace (Fig. 11) to 1,650 deg. F. in an inert gas atmosphere which is passed at the rate of approximately 1,000 cu. ft./hr. Up to about 3 per cent propane is added to the neutral carrier gas at the commencement of carburizing and is reduced to about ½ per cent during the subsequent diffusion. Test pieces are removed from the furnace periodically to check the case depth by penetrascope. Previous practice has been to continue the carburizing until about half the required case depth was indicated on the test piece and then to diffuse until the test piece indicated the required case and an absence of boundary carbides. A modified carburizing procedure is now in use, involving alternate two hour cycles for carburizing and diffusion, which is believed to reduce the presence of free cementite. Secondary

FIG. 11—*Primary reduction gear rim entering carburizing furnace*

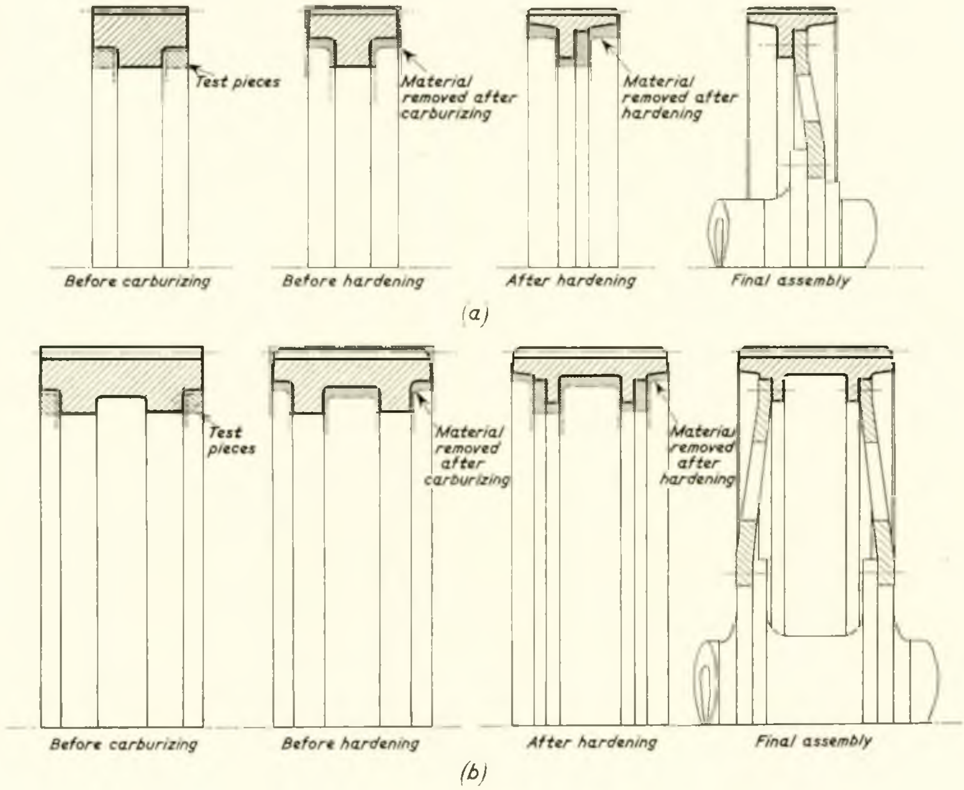


FIG. 12(a)—*Primary reduction gear rim sections up to assembly*
(b)—*Secondary reduction gear rim sections up to assembly*

reduction gearwheels were carburized for about 18 hours and diffused for a further 18 hours.

After cooling in a tempering furnace the gear rims are stress-relieved or annealed by holding at 1,200 deg. F. for five to six hours and then cooling in the furnace to below 600 deg. F. The gear rim distortion which is produced during carburizing is corrected as far as possible before proceding with further machining. Out of roundness and tapering (conicallity) are corrected by applying a jacking spider which uses large radiused pads to stretch the rim. Out of flatness is corrected on a large press. The carburized layer is then machined off all surfaces not required to be case hardened (Fig. 12).

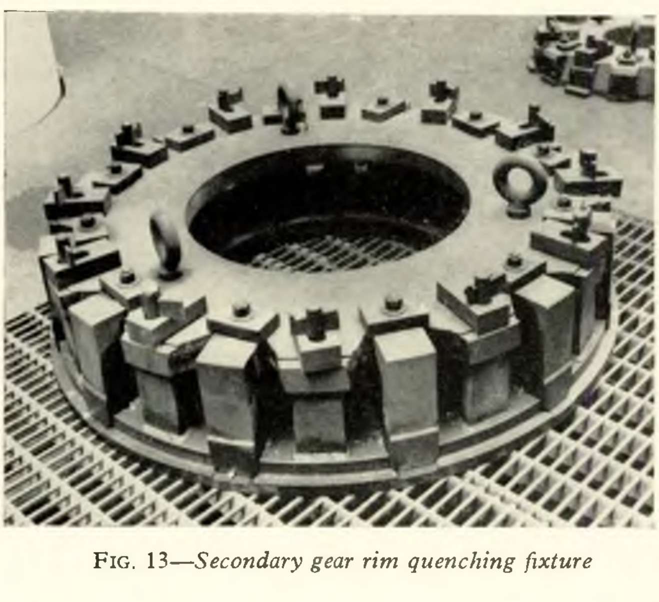


FIG. 13—*Secondary gear rim quenching fixture*

Hardening

The temperature of the gear rim and its test pieces is raised to 1,200 deg. F. and held for 6 hours and then raised to 1,480 deg. F. and held for 2 to 4 hours. The gear rim is then quickly transferred from the furnace to a quenching fixture (Fig. 13) which consists of a heavy cast steel ring fitted with a series of clamps and adjustable taper pads arranged circumferentially and set to a predetermined diameter. The taper pads are arranged to locate the bore of the rim while the clamps hold the rim flat. The gear rim complete with the quenching fixture is quenched in oil at 120/150 deg. F. and held in the oil for 20 minutes. The time taken between emerging from the furnace to quenching is approximately $3\frac{1}{2}$ minutes which includes a 2 minute holding period on the quenching fixture. Following hardening, the gear rim is tempered at 250 deg. F. for 8 to 10 hours after which it is removed from the quenching fixture.

Deep Freezing

The higher alloy content of AISI 3310 and the use of oil quenching introduces a higher susceptibility to retained austenite than did DEW 3610. This condition has been satisfactorily avoided by deep freezing after quenching and tempering. The rim temperature is lowered to -90 to -100 deg. F. in a container of dry ice and alcohol and held for 2 hours. The gear rim is then re-tempered at 250 deg. F. for 8-10 hours as after hardening. Deep freezing produces a slight increase of between 10-20 DPH points of surface hardness.

Case Depth and Surface Hardness

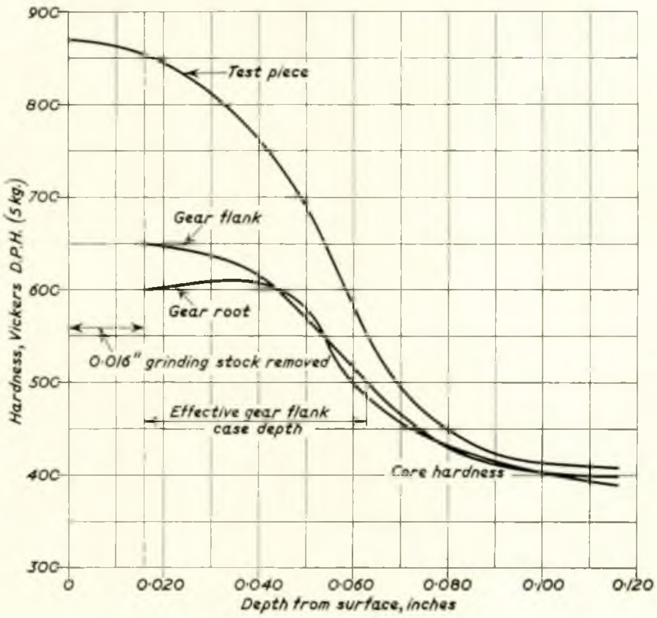
As already stated the duration of the carburizing and diffusion process is determined by the depth and quality of the case obtained on the test pieces which are withdrawn from the carburizing furnace and quenched in oil. The test pieces, which measure $1\frac{1}{2}$in. × $1\frac{1}{2}$in. × $1\frac{1}{2}$in. receive a more severe quench than the gear rims principally on account of the effect of the different masses. The result is that the case hardnesses measured on the test pieces are appreciably higher than on the corresponding gear rim which is subjected to the same heat treatment. For similar reasons the surface hardness obtained in pinions is slightly higher than on the gearwheels. Case depths which are determined from the hardness curves tend to be only slightly higher on test pieces than on the corresponding gear teeth. The depth of case formed round the gear tooth roots is generally found to be slightly less than that on the gear flanks due to the effect of surface concavity. This effect is not considered undesirable since the subsequent grinding operations on the tooth flanks tend to unify the depth of case round the entire tooth. The required case depths are determined in consideration of grinding stock allowances which must be provided to produce the finish ground tooth profiles. The required test piece case depths are given in Table III

TABLE III.—TEST PIECE CASE DEPTHS AND SURFACE HARDNESSES

	Case depth inches	Surface hardness DPH(5Kg) Rc	
Pinions	0·050–0·070	820–942	65–68
Primary gearwheels	0·050–0·070	763–820	63–65
Secondary gearwheels	0·075–0·100	763–820	63–65

together with the corresponding surface hardness values.

Gear rims which have been sectioned have clearly illustrated the difference in case properties between the production test pieces and the actual gear. Fig. 14 shows the mean tooth



	Test piece	Gear flank	Gear root
Surface hardness D.P.H.	870	650	600
Case depth inches (500 D.P.H.)	0·069	0·047	0·044
Visual case depth inches	0·056-0·060	0·052	0·052

FIG. 14—*Hardness gradients for DEW 3610 primary gear rim*

flank and tooth root hardness curves measured from a sectioned DEW 3610 primary reduction gear rim, from which approximately ·016in. of grinding stock had been removed, together with the hardness curve for the corresponding test piece. The gear tooth hardness curves were derived from two diametrically opposite gear rim teeth each of which was sectioned in three locations across the tooth face. One of the etched tooth sections is shown in Figs. 15, 16 and 17, together with the microstructures of its case and core. It will be observed that the

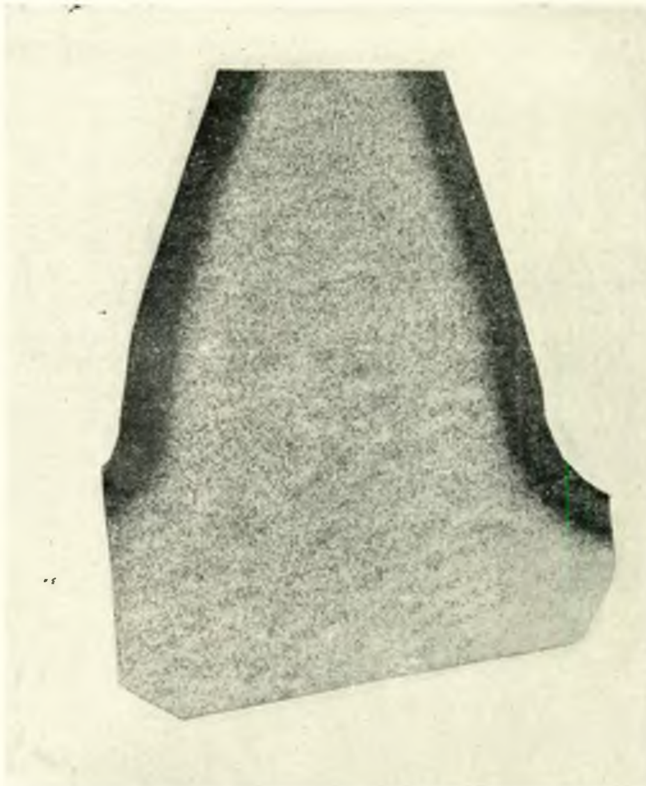


FIG. 15—*Etched section of DEW 3610 primary gear tooth*

surface hardness of the test piece, which in this case was water quenched, was appreciably higher than that of the actual gear. Test pieces for a water-quenching steel are normally oil quenched to give a more realistic indication of gear rim hardness.

Gear Rim Distortion

The straightening operations carried out after carburizing are repeated after hardening. The trueness and roundness of the gear rims are carefully checked before the bores are

FIG. 16—*Case microstructure of DEW 3610 primary gear tooth*

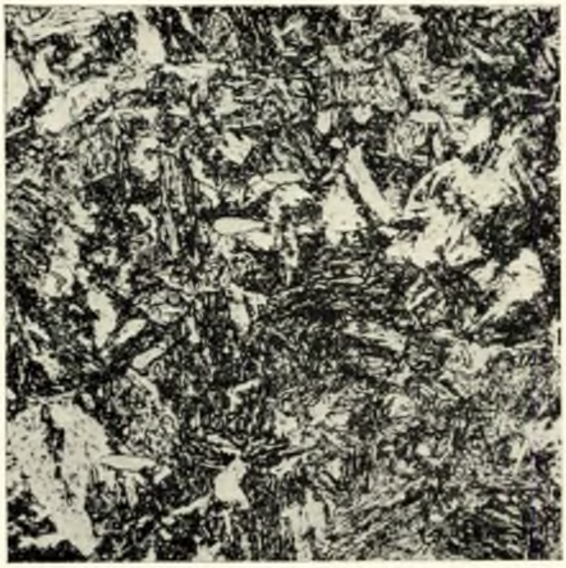


FIG. 17—*Core microstructure of DEW 3610 primary gear tooth*

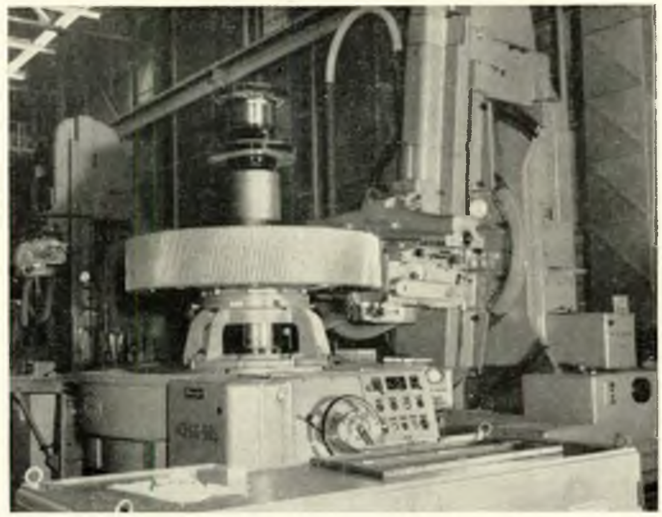


FIG. 18—*Secondary reduction gearwheel on grinding machine*

machined to receive the gear wheel discs which are assembled with a small shrink fit.

The amount by which the gear rim diameters shrink during carburizing is usually less than the growth during hardening which in the case of the 67in. secondary reduction gear were recorded as high as ·160in. This results in a net increase of the tooth root diameter between the cutting and hardening stages. Since the tooth roots are not ground except for blending-in the finish-ground tooth flanks it is most important that the final root diameter does not vary appreciably from the design figure. In practice it has been possible to hold the final diameter to within + ·030in. on the primary wheels and +·070in. on the secondary wheel of the design figures. The outside diameter of the gear rim, which is machined after carburizing and ground after hardening, does not usually vary as much as the root diameter. Variations in outside diameters do, of course, raise a special problem when manufacturing locked trains where it is necessary to obtain pairs of gearwheels of the same dimensions. In dealing with a number of gear units it is possible to select matching pairs of gear rims. Where the size of the gear rim after carburizing indicates that it would be too large after completing the full heat treatment, it has been satisfactory to re-cut and recarburize the gear rims with no detrimental effect to the case or core properties.

Out of roundness in the gear rims after heat treatment has been recorded up to ·070in. for primary wheels and up to ·100in. for secondary wheels. The out of roundness in the majority of gear rims was found to be about half these values prior to being corrected to within ·020in. of the true diameter. Out of flatness is a much smaller problem and is in fact corrected to within ·010in. in most gear rims before machining.

Gear Grinding

The case-hardened gears are ground (Fig. 18) using the Maag grinding process which utilizes the fundamental principle of involute generation by rolling gear teeth over the cutting edges of accurately located dished grinding wheels. The method of applying this principle at the Canadian gear plant is different for pinions and gearwheels. The generating method used for gearwheel grinding requires the two grinding wheels to be set with the grinding planes, formed by the outer edges, inclined to match the flanks of the basic rack cutter Fig. 19(a). The true involute tooth profile is obtained by the relative rolling or traversing of the grinding wheels acting as a rack cutter about the pitch circle diameter of the gearwheel, thus simultaneously generating the ahead and astern flanks of two separate teeth. The pressure angle of the tooth profiles generated by

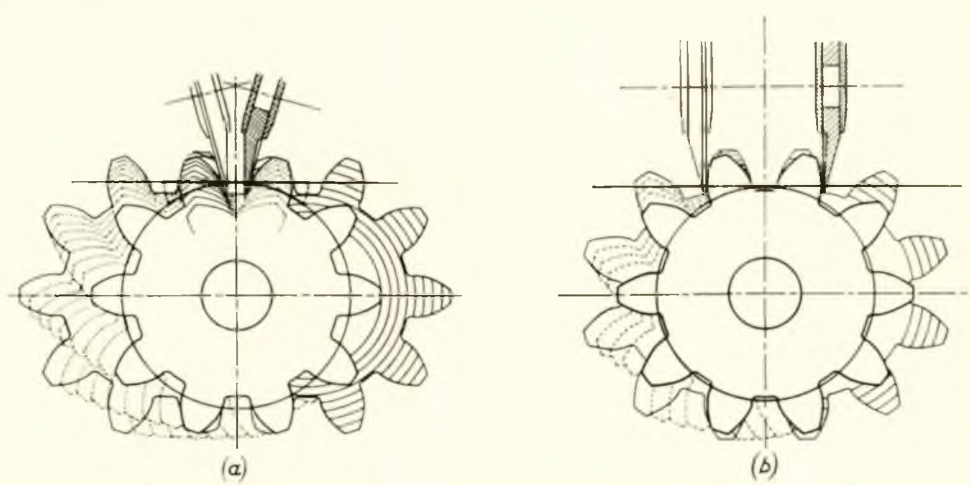


FIG. 19(a)—*Inclined wheel grinding method*
(b)—*Zero degree grinding method*

this process is determined entirely by the inclination or setting of the two grinding planes. The arc of grinding wheel contact with the tooth flanks is small but nevertheless sufficient to produce the characteristic criss-cross grinding arc pattern of this process.

The pinions are ground on machines where the involute profile is generated from the base circle using what is known as the zero degree method. The two grinding wheels are arranged with the grinding planes parallel and face inwards (Fig. 19(b)). The bottom tips of the grinding wheels are set to touch the outside flanks of a suitable small block of teeth on a plane tangential with the base circle, about which the pinion is then rolled and traversed. The pressure angle of the tooth is determined by the base circle diameter and the accuracy of the involute profile is primarily dependent on the setting of the grinding wheel tips with respect to the base circle. The point contact of the grinding wheel in this method permits an exceptionally fine surface finish which is devoid of the criss-cross grinding pattern obtained on the gearwheel teeth. The most important advantage of zero degree grinding is that it permits the most satisfactory method of accurately applying tooth profile corrections. Tip and root relief is applied to the pinion flanks by an additional movement being imparted to the pinion at the end of each generating stroke by a cam-actuated mechanism.

Gear grinding is carried out in the three broad phases of roughing, semi-finishing and finishing. The roughing operation is essentially one of removing the minimum stock necessary to clean up all tooth flanks and must be preceded by careful setting up and followed through with appropriate adjustments to the grinding heads. The stock removed per pass is reduced from about ·002in. during rough-grinding to about ·0005in. during the semi-finishing operation when the tooth profiles and pitch spacing are brought close to the required accuracy. The grinding wheel generating speed is reduced during semi-finishing and is reduced appreciably further during the finish-grinding operation which is allowed to continue until the wheels, which are set to remove no more than ·0001 to ·0002in. of stock, "spark out". The limitation on the amount by which the original case depth, as indicated by the test pieces, may be reduced during grinding is presently accepted by the Royal Canadian Navy as 30 per cent, and is measured from the point on the gear where the grinding wheels first make contact. This is of course an arbitrary limitation but experience has shown it to be realistic and in line with the specified requirement for final surface hardness and case depth.

The gearwheels are usually completely finish-ground before manufacture is commenced on the mating pinions. The diameters to which the pinions are machined prior to gear cutting and the tooth block M measurement are determined to suit the finished gearwheel. The pinions are ground to match the respective wheels exactly by obtaining the same base pitch measurements and by appropriate adjustments following numerous checks in the meshing frame. The pinions are ground to obtain full depth meshing over the entire length of tooth with the axes of the pinion and wheel parallel. Contrary to present thinking, helix angle correction has been applied to all existing units. A torque correction of ·0004in. on the primary and secondary reduction pinions was originally considered necessary and was applied during the final grinding operation and checked on the meshing frame with a ·0004in. adjustment on the parallelism between the pinion and wheel axes.

Grinding Time and Gear Accuracy

In addition to the obvious influence of gear distortion and grinding stock allowance on grinding times, consideration must be given to the effect of accuracy requirements and the ease of achieving the specified tooth form as determined by limitations of the manufacturing equipment. The accuracy specified for the gears made in Canada together with the average values of grinding stock removed and total grinding hours are shown in Table IV.

TABLE IV.—GEAR ACCURACIES AND GRINDING TIMES.

	Pinions	Primary reduction gearwheels (46 in. PCD)	Secondary reduction gearwheels (67 in. PCD)
Maximum profile errors, inches	·00015	·00015	·00015
Maximum tooth spacing errors, inches	·00015	·00025	·00035
Average stock removed per flank, inches	·008–·015	·010–·021	·012–·030
Average grinding time, hours	65–90	160 (15 deg. P.A.) 250 (23 deg. P.A.)	400

The grinding hours which are inclusive of setting-up times and all checking time between grinding operations, reflect a marked increase between the primary reduction gearwheels of the 15 deg. and 23 deg. pressure angle designs. It will be seen from Fig. 19(a) that with a smaller tooth and a higher pressure angle, the tips of the two grinding wheels would have to be thinned in order to extend to the bottom of each tooth flank without interference. The necessary thinning produces flexibility which in turn appreciably lengthens the grinding time, to achieve a given accuracy. The grinding wheels are of course the key between grinding time and accuracy in this process. Since the grinding wheels are in contact with the gear flanks only at one point, or a small arc on the fine dished edge, the requirement for stability and good uniform cutting qualities is critical. The inability of either a single wheel or both wheels to retain uniform cutting properties quickly increases the time required to produce a given accuracy. The degree of gearwheel flexibility is affected by the amount of edge contact with the tooth surface and will consequently vary as the arc contact of the inclined grinding wheel method changes at tooth entry and exit. This variation in grinding wheel flexibility is not so evident with the zero degree grinding method where the variation in edge contact is much less throughout the entire generating and traversing cycle. Early experience showed that grinding times could be influenced far more by the type of grinding wheel than by the amount of grinding stock to be removed. Flexibility and loss of cutting qualities in grinding wheels will extensively prolong the time required to correct very minor deviations from the required profile. Stiffening plates may be used to combat flexibility but will easily give rise to serious grinding burns where the grinding wheel flexure is due to poor cutting qualities.

The problem of grinding wheel flexibility and poor cutting qualities came to a head during the manufacture of the 23 deg. pressure angle primary reduction gearwheels. The Canadian grinding wheel industry was consulted and after considerable experimentation with various grits, grain structures and bonds, a superior grinding wheel was eventually developed which combined all the desirable requirements of uniformity and consistency in cutting properties and strength. Furthermore, the wheels were found to be sufficiently versatile regard-

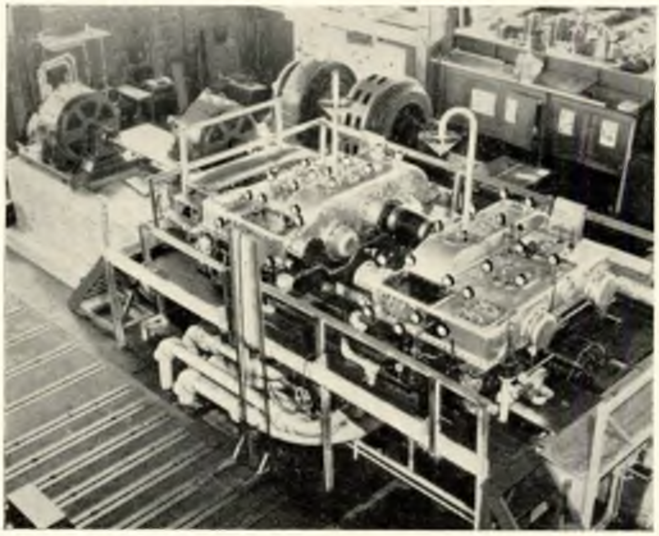


FIG. 20—*Port and starboard main gearing units on back-to-back test*

ing cutting properties and grinding finish to be used for both roughing and finishing operations. It was thus found possible to produce the 23 deg. pressure angle tooth form to the required standard of accuracy, but only at the expense of increased grinding time as shown in Table IV.

The permitted tooth spacing errors will be noted to be slightly outside the values specified for British Standard 1807 Class A1 which is generally applicable to hobbed and shaved gearing and is indeed necessary for most marine steam turbine reduction gearing of this type. Class A1 accuracy is in fact achieved in the Canadian gear plant but there is no evidence to suggest that it is necessary for hardened and ground gearing except for the most stringent of naval requirements relating to quietness of operation. Higher accuracies mean extending the selective grinding operations during the finishing process.

Assembly and Shop Testing

The mating sets of pinions and gearwheels are assembled in gearcases having all bearing housings bored and scraped to ensure parallelism between the axes. Gearcase parallelism between axes is a naval requirement to permit the use of interchangeable concentric bearings. Non-clearance concentric bearings are used for checking the meshing alignment of the pinions and gearwheels in the gearcase. For correct alignment the mating patterns registered with non-clearance bearings are required to compare with the mating patterns obtained on the meshing frame. The dual drives between the primary and secondary reduction gear trains are assembled and locked in a manner to ensure that all ahead flanks are simultaneously in contact at all powers. It is required that any variation between quill shaft torques, due to changing journal attitudes at low power, is never sufficient to cause tooth separation.

Each gearcase is mounted on the basic three-point support during manufacture and assembly on the test bed. A check on the condition of the gearcase from the time of assembly to the ship installation is obtained by sighting collimator targets erected on special pads at each top corner of the gearcase from a datum plane located above the secondary reduction gearwheel after bearing.

Each shipset of gearing is tested up to full power using the power circulation or "back-to-back" method of loading (Figs. 20 and 21). A hydraulic vane-type torque applier is coupled between the forward ends of the two secondary reduction gearwheels which are provided with forward coupling flanges especially for that purpose. The required power to rotate the gearing is provided by two 600 h.p. variable speed motors driving through flexible couplings fitted to the after ends of the main quill shafts in one gearbox. This method of testing permits the gearing units to be loaded independently of the speed but requires one unit to act as a speed increaser, that is with the astern flanks being loaded under ahead rotation. It is therefore necessary for all trials to be repeated with the loading reversed in order that each unit may be tested under the rated power conditions.

The gearing is slowly run up to full power speed with the minimum torque necessary to prevent chatter being applied and using extreme pressure oil. The torque is then increased in stages until full power is attained after a total running period of up to twenty hours. The gearing is run at full power for four hours and then subjected to a further eight hours testing with propeller law loading up to full power. The gear units are finally subjected to a fifteen minute full power trial with 130 per cent torque and a proportionate reduction in speed. The total shop testing time is approximately 60 hours per shipset of gearing.

Shop testing was initially carried out with both the three and five point supports. The results of these tests confirmed the earlier experience at Pametrada, that the gearing is capable of satisfactory operation with either type of support, in spite of the amount of twisting produced with the after corners of the gearcase unsupported. In consequence it has become the approved practice to install the gearing on the test bed

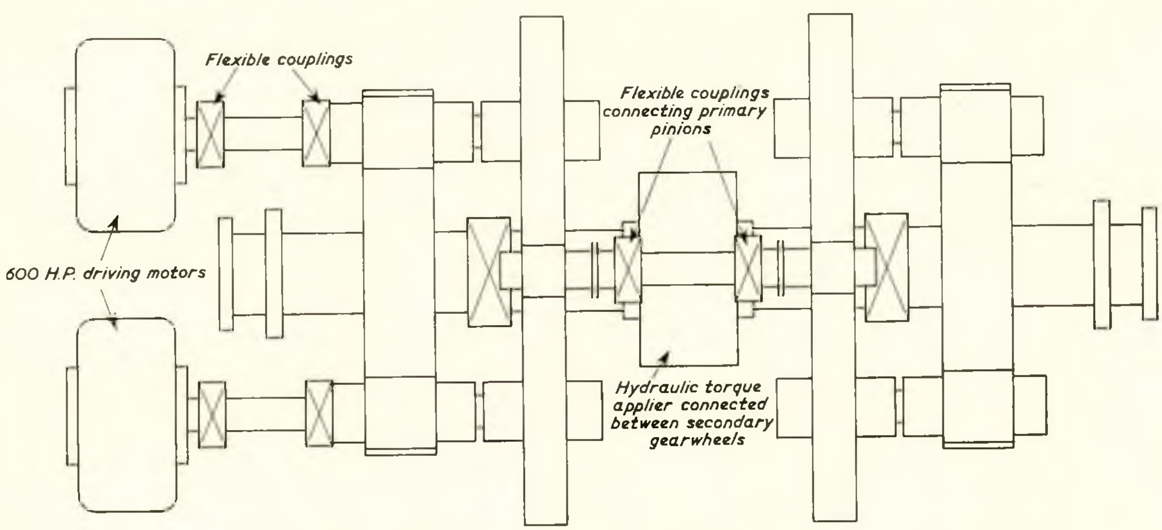


FIG. 21—*Arrangement of power circulation rig*

and in the ship on a three-point support and then to fit the remaining torque resisting chocks.

Service Experience

Service experience with the main gearing in all fourteen ships of the *St. Laurent* Class, over the last five years since commissioning has been excellent. The appearance of the gear teeth differs from the original condition only by a slight polishing of the original grinding patterns.

One of the ships, with the revised gear design, has now been running for almost a year with standard turbine oil after satisfactorily completing a four hour full power trial to prove the capability of the gear design. It may be noted that the ability of a gearing design to permit operation with a standard turbine oil, continues to be a recognized naval requirement, regardless of whether extreme pressure oils are used in service.

Main gearing tooth mating patterns observed both on the test bed and particularly in ship installations have not always compared with those obtained on the meshing frame. Low power tooth contacts have even been observed at the driving ends of the pinions. While the satisfactory condition of the gearing obviated the need for concern, efforts were made to diagnose the cause of what might have been mistaken for gear-case misalignment.

The most obvious factor emerging from this investigation was the effect of propeller shaft alignment on the meshing of the secondary reduction gearwheel. Due to the lightness of the main gearwheel and the absence of a separate thrust shaft, it was found to be most important for the main gearing to be aligned in the ship installation so that the intermediate propeller shaft would not exert a bending moment on the gearwheel and cause it to lift at the forward end. Main gearwheel lifting at low powers affects not only the mating patterns in the secondary reduction but also the load distribution between its mating pinions where they are in a locked train. Again, this effect has not been found to be deleterious to the gear teeth, but in conjunction with other factors can lead to gearcase knocking due to tooth separation at low powers.

A precise analysis of all the conditions affecting tooth mating patterns in single helical gearing—such as journal attitudes in clearance bearings, hydrodynamic influence of the oil film, quill shaft restraint and gearcase torsional and thermal distortion—can best be described as being virtually impossible. The conditions can, however, be analysed, sufficiently to indicate that the conventional basis for determining helix angle correction, in consideration of torsional and bending deflexions, is entirely inadequate for single helical gearing. The slewing effect resulting from the axial thrust component of single helical tooth loads, causes different angular attitudes of the pinion and gearwheel journals between forward and after bearings. The pinion and gearwheel axes are therefore not parallel to their respective bearing axes. If the maximum tooth loading is assumed to be equally distributed along the face width, as for double helical gearing, then it will be found that the pinion and gearwheel axes cannot be parallel to each other. Conversely, in order that the pinion and gearwheel axes may be parallel in the plane necessary to permit full face contact, the centre of tooth loading must necessarily be displaced from the centre of the face width. It therefore follows that a helix angle correction which is calculated on the basis of uniform tooth loading will be increased or decreased depending on whether the centre of tooth loading is displaced away or towards the pinion coupling.

In the case of *St. Laurent* Class main gearing it is now considered that the ·0004in. helix angle correction, which had been applied to all pinions, is not ensuring the uniform distribution of tooth load at full power. In view of the multitude of variables and indeterminates concerning helix angle correction, it is no longer being applied to gear sets now in production. Where prototype shop trials indicate concentrations of tooth loading at full power then helix angle correction will be applied on a trial and error basis.

It may be concluded that the high margin of safety obtainable with hardened and ground gearing has been well demonstrated by their trouble-free service in *St. Laurent* Class where it has been subjected to unknown concentrations of tooth load due to the doubtful application of helix angle correction and the effects of considerable gearcase twisting. It is considered most unlikely that soft gearing, which is believed to be particularly sensitive to malalignment, would be able to withstand these same conditions of operation.

The Mercantile Application for Hardened and Ground Gearing

Although several tankers and at least one ocean liner are now fitted with hardened and ground gearing and have given completely trouble-free service over a period of several years, the general use of this type of gearing in merchant ships is still not widely recognized. Of the many requirements which would conceivably influence the choice of hardened and ground gearing, the need for improved reliability would, it is thought, be high on the list in view of the incidence of gear tooth failures, ranging from premature wear to pitting and breakages, which continue to be reported in soft or hobbed and shaved type gearing. It is apparent that the highest proportion of mercantile main gearing failures are in aft end machinery installations.

The reduction of weight and space obtained with hardened and ground gearing is unlikely to be an important asset in tankers and may in certain cases even create some difficulty in permitting adequate condenser space. It is considered that the condenser problem would only be acute in a two turbine dual tandem articulated gearing arrangement where the height between the turbine and propeller shaft centres would be a minimum. In spite of its higher load carrying capacity it does not appear that hardened and ground gearing can be built to transmit higher powers than soft gears. This is because the torque capacity of the maximum size of gearwheels that can presently be manufactured by either the hobbing and shaving process (200in.) or the grinding process (142in.) are approximately equal.

There are many cases on record of surface failures in gear teeth of the soft hobbed and shaved type being overcome by the fitting of case-hardened and ground pinions. The increased load-carrying capacity of this combination of materials is small but nevertheless effective in cases of marginal design. Significantly higher load-carrying capacity and reliability can only be obtained where the gearwheels are also case-hardened and ground.

Unfortunately although many gear manufacturers are able to provide hardened and ground pinions very few are equipped to supply hardened and ground gearwheels or are even prepared to acknowledge the advantages of this type of gearing. While there can be little doubt that the outlook of the gearing industry on this matter is influenced to no small degree by existing machine tool investments, it is curious to note that hardened and ground gearing is commonly condemned on the basis of slow and costly manufacture. The suggestion of the manufacturing time for marine hardened and ground gearing being excessive has most certainly not been borne out by experience in Canada. Furthermore, in consideration of the tremendous strides which have been made in improved gear hardening methods, particularly the tooth-by-tooth induction hardening process[(4)] which can be applied to medium and large gearwheels, there can be little doubt that hardened and ground marine gearing is beginning to look economically attractive.

The removal of hardened gear material by grinding is unquestionably a much slower process than the removal of soft gear material by hobbing and shaving. However, in consideration of the reduced gearwheel diameters and face width permitted by utilizing the high load-carrying capacity of case-hardened gearing, the overall manufacturing time required to process the smaller gear tooth area would appear to be less than for the larger hobbed gears. In point of fact it is known that a much higher proportion of the total manufacturing time for a set of main gearing is spent on the gearcase and assembly than on the manufacture of the pinions and gearwheels. It might then be stated that the cost of a set of gearing is

TABLE V.—RELATIVE DIMENSIONAL AND COST INDICES FOR HARDENED AND SOFT GEARING.

Gear material		Gear type	Primary reduction				Secondary reduction				Manufacturing time index per unit toothed area	Cost index
Pinion	Gearwheel		K factor	Diameter	Face width	Unit tooth load	K factor	Diameter	Face width	Unit tooth load		
Soft (basis)	Soft (basis)	Double helical	80	1	1	1	70	1	1	1	1	1
Through hardened	Through hardened	Double helical	120	0·87	0·87	1·31	100	0·89	0·89	1·27	1·5	0·96
Carburized and hardened	Induction hardened	Single helical	250	0·77	0·54	2·41	250	0·78	0·47	2·78	1·5	0·73
Carburized and hardened	Carburized and hardened	Single helical	300	0·73	0·51	2·72	300	0·73	0·44	3·14	2·5	0·76

influenced far more by the size of the secondary reduction gearwheel than the gearcutting times.

To obtain an approximate estimate of the order of cost saving, which might be achieved by using case-hardened and ground gearing, a comparison of gear sizes has been made in Table V for the same torque requirement applying the Lloyd's K factors recommended by Page[2] for merchant ship applications. The figures listed are relative to a basic design employing K factors of 80 and 70 in the primary and secondary reductions respectively. Gear manufacturing times per unit peripheral area have been estimated on the basis of comparisons which have been made with respect to the carburized and hardened grinding times reported in Table V. The relative cost indices for complete main gearing units have been derived in consideration of the size of the secondary reduction gearwheels, the peripheral (toothed) areas and the manufacturing time indices.

CONCLUSIONS

The high reliability provided by hardened and ground gearing and its ability to withstand concentrated tooth loads, due to various internal and external factors causing misalignment, without distress has been demonstrated by the experience with *St. Laurent* Class in the Royal Canadian Navy. A high pressure angle is recommended to avoid scuffing but this should not exceed 20 deg. in the interest of ease of manufacture.

The overall manufacturing time required to produce a set of hardened and ground gearing may be expected to be appreciably less than for an equivalent soft gear design for the same power/speed requirement. Experience with induction hardened gearwheels running with carburized and hardened pinions is still very limited but, from the indications of tests so far conducted, it is considered that this type of gearing will become equally attractive for naval and mercantile installations alike on the basis of reliability and cost.

ACKNOWLEDGEMENTS

The author wishes to acknowledge the assistance given by the Resident Naval Overseer, Lachine and his staff in compiling manufacturing data and to thank the following companies and bodies for the use of illustrative material: Department of Mines and Technical Services, Ottawa, Dominion Engineering Works Ltd., Montreal, Pametrada, Wallsend and Maag Gear Wheel Co. Ltd., Zurich.

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REFERENCES

1) NEWMAN, A. D. 1958. "Load Carrying Tests of Admiralty Gearing", International Conference on Gearing.
2) PAGE, H. H. 1958. "Advances in Loading of Main Propulsion Gears", International Conference on Gearing.
3) CHESTERS, W. T. 1958. "Study of the Surface Fatigue Behaviour of Gear Materials with Specimens of Simple Form", International Conference on Gearing.
4) CHAMBERLAIN, A. 1958. "Developments in the Heat Treatment of Large (Marine) Gears", International Conference on Gearing.
5) BRADDYLL, J. R. G., and OLDHAM, M. C. 1959. "High Torque Merchant Ship Gearing". Trans.N.E.C.I.

Discussion in London

TUESDAY, 11TH APRIL 1961

MR. S. ARCHER, M.Sc. (Member) said the Institute was indeed fortunate to be able to include in its TRANSACTIONS such an expert and detailed account, as that given by Mr. Nicholson, of the application of hardened and ground gearing in naval service, embracing as it did design, development, manufacture, testing and service experience. The paper struck him as a model example of an interesting technical story simply and straightforwardly told with no attempt either to dramatize or play down any particular experience or aspect of the project.

The first point he noted was in the author's introductory summary, wherein he referred to the extensive prototype testing carried out at Pametrada on the first units of the series. The value of such shore-based tests was incontestably demonstrated by the experiences so gained and suggested that, also in cases of merchant designs, embodying novel features or advanced practice, recourse to similar prototype testing on shore would be likely to yield dividends. This would, however, usually not be economic unless a sufficiently large number of sets for, say, a standard series of ships was in question.

In the case of the destroyer escort gears under notice, if full power shore-testing facilities had not been available, the possible consequences in loss of time and money might have been depressingly serious. At the same time he thought that all concerned were to be congratulated on the courageous and carefully weighed decision to "change horses almost in mid-stream" (or should he say "mid-ocean"?!) and adopt a radically different tooth form after the early teething troubles.

On page 62 the statement was made that "a case-hardened gear will permit at least 2½ times the maximum root stress allowable in a hobbed and shaved gear of the same tooth design". This, presumably also assumed identical or equivalent through-hardened material having the same tensile strength, etc., as that of the case-hardened core material.

Table I on page 63 showed a very interesting comparison between the original coarse-pitched, 15 deg. design and the revised design with reduced pitch and increased pressure angle. Assuming main shaft revolutions of 220 per minute, he calculated the following values of sliding and rolling speeds, from which it was clear that the design changes adopted reduced the critical conditions of sliding and slide/roll ratios by the following percentages:

Reduction of sliding velocity in primaries and secondaries = 30 to 40 per cent (approximately).

Reduction of slide/roll ratios in primaries = 55 to 60 per cent (approximately).

Reduction of slide/roll ratio in secondaries at pinion approach = 66 per cent (approximately).

Reduction of slide/roll ratio in secondaries at pinion recess = 40 per cent (approximately).

In Table II—Gear Materials—it would seem that on the basis of a Poldi chart the equivalent elongation percentages on 2in. specified for the original (Swiss) material were about 17 to 19, i.e. about 50 per cent greater ductility than the comparable American materials selected. Could the author suggest what was responsible for the more favourable ductility and impact values of the Swiss material despite the higher tensile strength and yield point?

One aspect of the gearing performance not dealt with in any detail by the author was the question of noise level. In particular one would expect quite a deal of tooth contact noise with the relatively coarse pitches and low helical angles adopted. Since several sets presumably were in service with the original coarse tooth design, could the author give any figures, or comment in any way, on the comparative noise levels as between the two designs.

On the technique of carburizing (page 66) it was interesting to note the reduction in free cementite believed to be obtained by the alternate short periods of carburization and diffusion. Presumably this assisted also in reducing the danger of quenching cracks? The use of the deep-freezing technique was also interesting. It seemed to be applied to everything these days,

TABLE VI

Design	Mesh	Position on pinion in mesh	Sliding velocity ft./min.	Rolling velocity, ft./min.			Slide/Roll ratio*
				Pinion	Wheel	Mean	
Original	Primary	Approach	2,520	1,340	3,860	2,600	0·966
		Recess	2,590	5,600	3,020	4,310	0·600
	Secondary	Approach	1,100	90	1,190	640	1·716
		Recess	855	1,715	860	1,290	0·665
Revised	Primary	Approach	1,740	3,730	5,470	4,600	0·379
		Recess	1,560	6,490	4,930	5,710	0·274
	Secondary	Approach	645	790	1,435	1,115	0·580
		Recess	625	1,845	1,220	1,530	0·408

* Based on *mean* rolling velocity of pinion and wheel at point considered, see Appendix I of paper.

"Some Teething Troubles in Post-War Reduction Gears" by S. Archer, M.Sc. (Member), Trans. Vol. lxviii, 1956, pages 324/5.

from kippers to human beings! Presumably there was no increased danger of case cracking.

The implications of Fig. 14 and Table III showing the large differences between gear tooth and test coupon hardnesses and case depths were particularly important for inspecting authorities such as that Mr. Archer represented. Presumably there would be similar but smaller differences between test piece tensile strengths and gear tooth core strengths? From information in his society's records it seemed that tooth surface hardness had a tendency to increase considerably towards the ends of the teeth on account of the more rapid quenching there. Could the author confirm this in his experience and if so, did he think it important?

On page 68 he noted that "straightening operations are carried out after carburizing and repeated after hardening". Presumably the amount of correction in the latter case was so small that there was little danger to the case-hardened surfaces?

The influence on grinding time and cost of the increased pressure angle of 23 deg. in the primary gears was well brought out in Table IV. Presumably no increase in grinding time occurred with the 20 deg. 2nd reduction gears on account of the larger pitch, i.e. no thinning of grinding wheels was required for the secondaries?

It would be interesting if the author could describe in somewhat greater detail the method adopted for equalizing the torques on the two branches of the locked train and how it was measured? Also what maximum percentage torque difference on the two quills was allowable?

Fig. 3(a) showing gearcase chocking arrangements and deflexions under 130 per cent full torque was of profound interest and revealed an astonishing degree of distortion due to the torque reaction. Even with the "five-point" support there was some 12/1,000in. "cross-wind" on the port side at the top of the gearcase. The function of the long quillshafts was thus extremely important if load distribution was to be anything like uniform. Would any additional transverse stiffening be effective? For example, heavy gusset brackets in line with the gearcase end walls might help to resist the torque reaction. Incidentally, it was difficult to be sure from Fig. 1 or Fig. 20, but it was concluded the gearcases were of fully fabricated design? If so, had any cracking of welds been experienced in any of the gearboxes? Having regard to the measured deflexions and bearing in mind that those related to conditions on a firm foundation and not on a moving elastic structure, as in a seaway, it was hardly surprising that the author saw little point in an automatic application of a standard amount of helix correction and preferred an "*ad hoc*" approach in each case! Presumably the practice of arranging the single helix unsymmetrically between extended bearing centres in order partially to cancel bending and torsional deflexions was considered in the design stage as an alternative to helix correction? If so, what were the objections other than slightly increased axial length?

If the author were to design another gearbox of similar type and loading, would he consider the possibility of adjustable bearing housings, as an "*ad hoc*" means of improving tooth bearing as required? If not, why not, please?

It was interesting to note that, owing partly to the lightness of the main wheel, lifting of the forward journal was experienced under light load conditions on unpredictable occasions, thus bearing out some of the conclusions of Zrodowski and Andersen* in their recent paper before this Institute. Was this movement once-per-rev. or was it at propeller blade frequency? Even at higher ratings load sharing seems to have suffered from the same cause. Ideally, it would be interesting to try the effect of say, a Bibby, or similar flexible coupling, in the main drive in order to isolate the gears entirely from such effects. Two small geared turbine merchant ships, 3,600 s.h.p., have given seven to eight years trouble-free service with such drives and have had several propeller damages without affecting couplings or gears; but, of course, 15,000 s.h.p. was another story! Still, it would be fun to try such a solution, for the possible gains were high.

He noted that the author did not consider the increased load-carrying capacity, resulting from the use of case-hardened pinions, would greatly improve the performance and reliability of a soft through-hardened wheel and for a significant improvement case-hardening and grinding of the wheel was essential. This agreed generally with experience in merchant service, since a number of expensive failures and/or cases of severe wheel tooth pitting have ensued from an excessive hardness differential between pinion and wheel material.

On page 72 the author stated that the highest proportion of mercantile "soft" gearing failures were in aft end machinery installations. Admittedly, this was true and doubtless was also influenced in some degree by shaft misalignment effects on the main wheel. Incidentally, the records indicated that the total number of geared turbine installations in merchant service fitted aft was very nearly equal to the number of those fitted amidships.

Could the author give some idea of the extent of full power operation to date with this series of destroyer escorts? If it were possible, for example, to assess this on some such basis as the following, this would be very helpful in estimating the relevance to potential merchant applications of the naval experience reported:

$$C_1 = \frac{\textit{Total ship s.h.p. hours at sea}}{\textit{Total ship operating hours at sea}}$$

and

$$C_2 = \frac{\textit{Total ship s.h.p. hours at full power at sea}}{\textit{Total ship operating hours at sea}}$$

He was unable to agree entirely with the author in his opening sentence of the final section of the paper entitled "The Mercantile Application for Hardened and Ground Gearing", since within the last ten years several cases of turbine gearing failure had been reported to the organization he represented, both with hardened and ground gears as also with other types. To be fair, however, those were in most cases due to material defects which might just as easily have occurred with "soft" gearing.

In his opinion, the best use for case-hardened and ground gearing in mercantile practice would be to reduce face widths *substantially*, thus shortening the gearbox and hence torsional and bending deflexions on the gear teeth, but maintaining almost the same *diameters* as with "soft" gearing, thereby avoiding difficulties with too short centres, such as insufficient space for condensers, etc. The enhanced factor of safety so achieved should promote sounder sleep among superintendent engineers!

He thanked Mr. Nicholson sincerely for a most valuable and stimulating paper.

Mr. A. Sykes said Mr. Nicholson's paper was particularly valuable in that it dealt with a type of gear which, when it was made, was relatively new in the marine field, though there was a certain amount of experience available in other fields. Its value was still further enhanced by the fact that it applied to a considerable number of sets, 14 ships in all, which had been in operation for five years, which was far more useful than a single experimental case. It was always felt that actual service experience was more to be relied on.

The author had mentioned in the early part of his remarks that some experimental work had been done in this country in which a K value of 1,358 had been achieved. Although that figure had been published, it was found later that a mistake had been made and that the actual figure was about 1,000. But even that still left a very comfortable margin over the figure of 412 to which Mr. Nicholson referred. He had referred to the question of root strength resulting from the compressive stress in carburized and hardened gears, but he had thrown some doubt on achieving the same improvement in induction hardened gears. The view was held in England and was supported by a certain amount of evidence, that a similar effect

* "Co-ordinated Alignment of Line Shaft, Propulsion Gear and Turbines", Trans.I.Mar.E., 1960, Vol. 72, pp. 135-185.

had been obtained with induction hardened gears. He believed it was the fact that, with induction hardened materials there was a compressive stress in the direction of the depth of the tooth, although not in the length of the tooth; but fortunately it was the direction of the depth in which there was particular interest, because that was the direction in which the bending stress occurred. Some very good results had been obtained in tests on induction hardened gears, in England and loads had been achieved very considerably in excess of 412K and they had the advantage of requiring very little grinding time.

Mr. Nicholson had suggested that the use of the A.I.S.I. 3310 steel should reduce distortion and he had indicated there was not yet sufficient experience to confirm this. Mr. Sykes very seriously doubted whether any improvement would be obtained resulting from small differences in chemical analysis. Somewhat similar gears had been made in Switzerland, in Great Britain and in Canada, and the results, as far as they were obtainable as regards distortion, had been very similar in all cases. The distortion of large case-hardened gears, as Mr. Nicholson had pointed out, could be of a very serious amount. The figure for distortion of induction hardened gears of the same size was about 1/20th that of case-hardened gears, which did result in a very much reduced grinding time and it was possible that with further experience it would be possible in some cases to eliminate grinding altogether.

In addition to the grinding there were other expenses incurred in straightening, to which the author had already referred, and the cost of quenching fixtures. A further point was that it was fairly regular experience with large case-hardened wheels to leave the pinion until after the wheel has been completed. That, of course, interfered with interchangeability. It meant that each pinion was made to suit its particular wheel.

He was very interested in the author's remarks about Class A1 accuracy, but said he was not quite sure what had been implied. He thought another speaker would mention this. He believed Mr. Nicholson had some doubt at one point as to whether Class A1 accuracy was really very necessary. Accuracy of course had an effect on both load capacity and quietness.

Another field which had not been mentioned was the use of nitrided gears. An experiment in England had shown that nitrided gears would carry a load equal to that of case-hardened gears, though it had to be admitted that in general it was only possible to have a very shallow depth of casing except by an abnormal length of hardening time. There did seem to be a field for both nitrided gears and induction hardened gears, nitrided gears being used for relatively fine pitches where the case depth would not be very large, and induction hardened gears being used for coarser pitches, slow speed gears, in which a depth of hardening comparable to that in case-hardened gears could quite easily be achieved. Members might be interested to know that a nitriding furnace suitable for hardening gears up to 80in. in diameter would shortly be installed in England.

He said he was very interested in the remarks about three-point support for the gearcase. Mr. Nicholson's experience there was identical with what had been found in England. They had started off with a very similar gearcase, the same size and same power, on a three-point support. Whilst that was good enough for assembly and manufacture, when it came to full load running it was found to be necessary to go to a five-point support. It seemed quite impracticable to design a gearcase strong enough to stand full loads without additional support.

The author had dealt briefly with the subject of helix correction, and it seemed that he was rather doubtful about the value of it. In this connexion a very interesting suggestion had been made some time ago by Mr. Rogerson of Cammell Lairds, who had pointed out that the working temperature under normal conditions of a pinion was usually about 10 degrees higher than the corresponding wheel, arising from the fact that the same amount of power was being transmitted to both—they both had to absorb a similar amount of heat arising from friction loss—but the pinion had a very much smaller mass than that of the wheel and therefore its temperature ran higher. The fact that the temperature of the pinion was higher than that of the wheel caused its axial pitch to increase, which in itself could be made to give a helix correction under full load conditions. The difference in axial pitch could, of course, be made to neutralize the effect of the deflexion due to bending and twisting. It was then simply a question as to whether with double helical gears, the helices ran with point forward or point backwards under normal conditions. It was possible to make use of that feature to avoid or mitigate the effects of deflexion by both bending and twisting under full load.

Reference had been made to the slewing effect of single helical gears. That, of course, did occur, but again it could be dealt with fairly well by the method that was being used to a large extent by Messrs. Brown Boveri in which they employed a thrust cone on the pinion. That practice had been tried in England to a certain amount and it appeared to be entirely satisfactory. Frictional losses, moreover, were quite small, and it completely eliminated any slewing effect.

Reference had been made to deep freeezing. People in this country were not altogether enamoured of deep freezing as they thought it led to brittleness, although it could be used to increase hardness if the hardness was rather deficient. But that was regarded as a remedy rather than a normal practice.

Commander E. H. W. Platt, M.B.E., R.N. (Member) commented that both in the paper and in his presentation the author had made it very clear that the time occupied in grinding was the principal factor in controlling the production costs of hardened and ground gearing. On pages 68 and 69 the author had stated that it was possible to hold the net increase of size of wheel rims through the carburizing and hardening processes to within 0·3in. for the primary wheels and 0·7in. for the main wheels in this gear design. This amounted to rather more than half of one per cent for the smaller wheels and just over one per cent for the larger. He asked: Was it to be inferred from this that the percentage net increase in diameter of work was likely itself to increase with size? Also, in stating that the net increase could be held within these limits, did the author infer that no rejections were now anticipated from distortion? If not, could the author indicate the proportion of distortions experienced? Whatever the answers to these questions were, one was very heartened to hear optimistic prophesies by Mr. Sykes on the development of induction hardening and other methods which would get people out of this distortion difficulty experienced in carburizing, and it was hoped that perhaps later speakers, including his successors in the Admiralty, would be able to give a little more information about this.

On page 72 the author had suggested that it was the fact that the gears were hardened and ground, which enabled them, in the case of the *St. Laurent* Class, to tolerate concentrations of load, resulting from doubtful application of helix angle correction and the effects of gearcase flexure. Whilst it was agreed that the very large factor of safety against surface damage inherent in case-hardened and ground gears was of great significance in this context, it occurred to him that the fact that they were single helical, whereas practically all soft marine gears were double helical, might play at least some part in their ability to accept what amounted to a degree of misalignment. He said he would be interested to hear the author's views on this suggestion.

On page 72 the author had summarized the present prospects of hardened and ground gearing in the mercantile application, and the speaker was generally in agreement with his views, although he wanted to comment on some of the opinions which had been expressed. Mr. Nicholson's remark that weight and space considerations were not of primary importance in merchant ship machinery was broadly correct, although in the case of tankers, reduction of engine room length could provide a substantial benefit in increasing the cargo capacity for a given hull. If by ingenious and unconventional approach to the condenser problem, introduced as a result of lowering the turbines when a smaller gearbox was fitted, advantage could be taken of its reduced dimensions to bring down engine room length, a real gain would have been made, and it was suggested that shipbuilders and marine engine designers should find time to give this matter attention.

He supported the author strongly in questioning the validity of the opinion that the manufacture of hardened and ground gearing need be more costly or longer in time than for soft gears. Indeed, it should be a cheaper and quicker method of gear manufacture once the hardening methods which eliminated major distortions associated with carburizing became available, and it would seem that the figures which the author had shown to support this were probably quite correct. He said he would go so far as to suggest that in the end, lower cost for equivalent hardness and reliability would prove to be the principal reason for the introduction of hardened and ground gears in merchant ships.

Nevertheless, in the past ten years or so, the substantial capital investment by engine builders all over the world in machinery for the production of "soft" gears of very high quality which was well able to meet the broad operational requirements of shipowners, might inevitably, and he suggested, legitimately, slow down the widespread adoption of hardened and ground gearing in merchant ships. He felt that on the whole, experience, particularly in England, with what must be called normal gearing was more satisfactory than perhaps the author had indicated. This was stressed particularly in the United Kingdom, which it was thought had a good record over the past few years in this respect. In contrast to warship machinery where maximum exploitation of the load-carrying capacity of hardened and ground gears provided outstanding advantages, the gains from its use in merchant ships did not seem sufficiently striking to warrant accelerated re-equipment of existing facilities. It could well be that the first exploitation of these techniques in commercial service would lie in the use of hardened and ground primaries and possibly also secondary pinions in conjunction with soft main wheels, one reason being that the same machines which were employed for grinding warship main wheels were suitable for merchant ship primaries. In this way one could see the opportunity for full use of these machines during what was hoped would be normal periods of small scale warship production in times of peace. Braddyll and Oldham in their paper, taken as Reference 5 by the author, had shown that "mixed" gear sets of this type were not unattractive. Therefore, in present circumstances one was inclined to confine one's predictions to a gradual realization of potential advantages of hardened and ground gearing in the merchant ship field, and to suggest to manufacturers that when the time came for major re-equipment of their gear production plant they should give serious consideration to these advantages.

Mr. G. A. Kemper (Member) first thanked Mr. Nicholson for his excellent paper and especially for the clear description of the considerations leading to the adoption of hardened and ground gears both for naval and merchant marine purposes. He thought it of interest to mention that during the same period gearings had been designed and manufactured for a series of eight destroyers of the Royal Netherlands Navy, each vessel having two gearings of 30,000 s.h.p. The Royal Company "De Schelde" had been the main contractors. The gearings had been designed in co-operation with Messrs Maag of Zurich and the first two gearings ground and assembled by Maag, as the grinding machines were not at the time available in the De Schelde yard. The gearings were of the dual tandem locked train type, and differed from the Canadian gearings in so far that only in the first reduction both the pinions and wheels were case-hardened and ground. In the second reduction case-hardened and ground pinions worked in conjunction with through hardened and ground wheels. The K factor at maximum load was 310 for the first reduction and 225 for the second reduction. A higher loaded case-hardened main wheel would hardly have reduced the size as the double locked train, with four intermediate wheels, required a certain minimum diameter of the main wheel. These gearings had operated entirely satisfactorily since 1955. They had been lubricated by extreme pressure oil, and no trouble had been experienced with scuffing or pitting; but they had not been tried with normal oil. Considering the K factors of the Canadian design, it was striking that the K factor for the second reduction was higher than that for the first reduction, contrary to normal practice. He asked: Was the choice of the K factor perhaps more guided by geometrical data than by theoretical considerations?

Mr. Nicholson had stated rather high figures for the maximum tooth spacing errors and had suggested that Class A1 accuracy should not be strictly necessary. A lower accuracy, however, firstly, meant higher dynamic loading (thus counteracting the aim of achieving the highest possible effective output) and secondly, an appreciably higher noise level. In merchant marine ships the noise level was now a matter of general interest, and for this reason Mr. Kemper said he would not like to see Mr. Nicholson's suggestion towards lower accuracy adopted, particularly as the modern grinding machines could very well meet the highest requirements.

He said it was not clear to him why a three or five-point support was used during assembly and testing, as for the final support on board further torque resisting chocks had been fitted. It seemed that the casing had as a result to be designed with extra rigidity (which meant it would be heavier and more expensive) whilst no real advantage was obtained, as the final situation was still the same as with the usual application of chocks all around the casing. No advantage could be seen in it being designed as one complete thing instead of splitting it up into two. He asked if Mr. Nicholson could give more information about the reasons for this detail of the Canadian design.

Finally, he was glad to use this opportunity to support Mr. Nicholson's view concerning the future application of hardened and ground gears on merchant marine ships. The speaker's company had supplied 11 gearings of this type for tankers and passenger ships, and cargo ships with outputs ranging from 5,000 to 19,250 s.h.p. per unit. Amongst these the normal articulated as well as the locked train type were represented, and all were operating highly satisfactorily—the first since 1955. He agreed that the combination of saving in weight and space plus reliability would undoubtedly lead more shipowners and gear builders to hardened and ground marine gears. Nevertheless, he was equally sure that double helical gears would for many years to come maintain their well established position.

Mr. W. G. Smith said that in opening the discussion Mr. Archer had so admirably covered most thoroughly nearly all points in the paper, to the extent that it left little scope for further comments. Mr. Sykes had questioned Mr. Nicholson's reference to AVGRA test gears having successfully withstood loads of up to 1,358K, this figure was correct in test gears simulating second reduction gears. The calibration error referred to by Mr. Sykes was relative to tests on gears simulating first reduction gears and these figures had been corrected in the proceedings of the International Conference on Gearing.

He supported Mr. Nicholson in his propaganda for the use of hardened gears in merchant ships. The author had mentioned Mr. Braddyll's paper, in which the use of hardened primary gears was advocated, with a resultant saving in cost. Mr. Nicholson showed that an even greater increase in saving was possible. It was a little disappointing that merchant ship operators were reluctant to seriously consider the use of hardened gears, as it seemed that a much greater risk was being taken by the employment of soft gears, the loading on which was being constantly increased, maybe only slightly, but it was reaching a figure close to the optimum, with a consequent reduction in the factor of safety. The installation of hardened gears would restore this factor to a much safer margin and also reduce costs.

There had been considerable success in this country in the tooth by tooth technique of induction hardening gears up to 72in. diameter and measurements had shown that the resultant distortion was easily within a tenth of that experienced with carburized and case-hardened gears. Consequently the grinding times had also been reduced to about a third, not a tenth as one might expect as the setting up and finishing times were the same in each case. A typical case-hardened contour obtained by this induction hardening process was shown in the diagram.

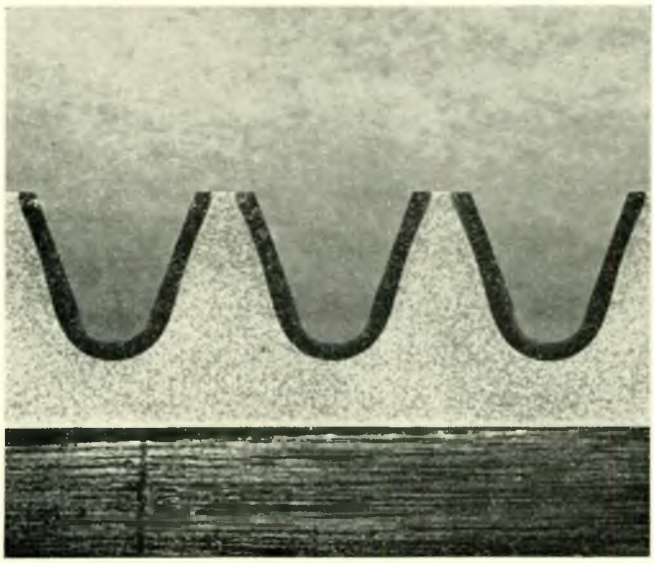

FIG. 22—*Etched section of tooth by tooth induction hardened gear showing uniformity of hardened contour*

He wondered about the grinding time given by Mr. Nicholson for tooth grinding the secondary wheel, 400 hours. Did this refer to the original design in which the secondary wheel had 143 teeth or the revised design with 187 teeth. The experience in this country suggested that as a general rule the time taken for tooth grinding this particular type of gear was in the region of three hours per tooth.

With regard to the application of helix correction to secondary pinions, to compensate for bending, torsional deflexion and temperature, this was further complicated when the attitude of the gears in their bearings had to be taken into consideration and also that the amplitude of correction was only suitable at one particular loading. To overcome these difficulties secondary pinions had been employed in which the drive was taken at the centre of the pinion instead of at one end. This had been achieved by trepanning away the metal at each end of the pinion between the shaft and the rim and so leaving a web at a position in the centre of the face width.

Present design of hardened gearing entailed the use of single helical gears of small helix angle in order to keep the end thrust to a minimum, but the possible employment of thrust cones, would enable helix angles to be increased, with a consequent further reduction in noise.

He agreed with Mr. Sykes that the use of nitrided gears, the distortion of which was so small that tooth grinding could be eliminated, was becoming most attractive. It was known that this type of gear, up to 96in. diameter, was being put to use in the naval field on the continent. In this respect, with the improved technique now being introduced to the tooth by tooth induction hardening process, the use of such gears without tooth grinding would also seem a distinct possibility.

In conclusion he congratulated Mr. Nicholson on presenting such an interesting paper and added that he could vouch for the excellence and quietness of the gearing described, as he had recently had the pleasure of travelling in one of the vessels mentioned.

MR. J. H. GOOCH, M.A., joined with the other speakers in congratulating Mr. Nicholson on presenting a very interesting technical paper. He agreed with what they had already said, that for case-hardened gears probably the future would show more use of nitrided or induction hardened gears in order to get away from the severe distortion of wheels in the carburizing process.

In the section on grinding times and gear accuracy the author had revealed that the tooth spacing errors were slightly in excess of values specified in B.S. 1807 Class A1 and said that there was no evidence to show that this high standard was really necessary for hardened and ground gears. He was ready to believe this, but did not agree that a lower standard of tooth spacing accuracy could be tolerated with highly loaded case-hardened gears than with moderately loaded hobbed and shaved soft gears. Instead, he believed that the higher the tooth loading the more accurate the gear should be. There was plenty of evidence becoming available to show that BS 1807 Class 1 pitch errors were not necessary to ensure excellent operation of hobbed and shaved soft gears, and it seemed that the stringency of this particular item in the Standard was in excess of what was necessary for either shaved or ground gears.

The author's remarks and conclusions about helix correction were very interesting. For hardened and ground gearing Mr. Nicholson and his colleagues had found that the calculated correction when applied was not providing the required uniform tooth loading and so gears at present in production were being made without this correction. But correction was to be applied to the teeth after full power prototype trials to relieve concentrations of tooth loading shown up during the trials. This seemed eminently sensible and was surely the ideal way of applying correction.

Although it was practicable to run prototype naval gearboxes at full power in a power circulating rig such as the author had described, it was not practical or economic to do the same with most normal merchant ships' gearboxes. Therefore, grinding the final corrections to the helices of hardened and ground merchant ships' gears, in the same manner, was not possible and one would have to fall back on using corrections, applied from calculations, tempered with what experience could be gained from existing sets in service. This was one reason why hardened and ground merchant ships' gears would not be used at the same tooth loading as even the most conservative warship loading. Another and more cogent reason was that merchant gears normally worked at full power continously, while warship gears spent only a small proportion of their service life at full power.

On page 62 the author had stated that with a good through hardened steel, hobbed and shaved gears might reasonably be designed for maximum service tooth loads of up to 120K for merchant ships' gears and 250K for naval gears. Mr. Gooch agreed with this, and also on the appropriateness of the tooth loading of 412K for *St. Laurent* Class ships' gears using hardened and ground teeth. But he would like to take issue with Mr. Nicholson on his advocacy of case-hardened gears for merchant ships on the grounds that they would be cheaper. There had been quite a lot of talk about this already. In Table V, and in particular lines 2 and 3, which were the ones which indicated modern practice and the possible alternative, the author clearly presented his assumptions. Bearing in mind what he had just said, Mr. Gooch contended that the figure of 250K in line 3 of that table was too high. If a judicious advance in tooth loading in naval gears was from 250K to 412K, with the addition of prototype testing at full power, then surely a judicious advance in merchant ship gear tooth loading would be from 120K to about 200K for primary gears and from 100K to about 170K for secondary gears, to retain the same sort of order of safety factor, bearing in mind especially that the criterion with case-hardened gears changed from surface stress and surface failure of the teeth to root stress and root failure of the teeth.

Consequently, the proposed economy in using hardened gears for merchant ships would become even more marginal. This marginal economy on paper would in fact be swamped by considerations of grinding machine and heat treatment equipment availability. Therefore, Mr. Gooch maintained that a change to case-hardened and ground gears for merchant ships would not lead to any appreciable economy in manufacture.

There was another reason for not encouraging the use of gears with case hardened teeth in merchant ships, based on considerations of safety and reliability. With conventional through hardened gears the tooth surface stress was more critical than the root stress, and so if any tooth defects occurred, they would appear as surface distress, either pitting or scuffing. These defects developed only over a period of time, were easily seen and recognized, and even in the worst cases the gears could continue to be used though at reduced power, should the ocasion demand it. However, with case-hardened gears, the tooth root stress becomes more critical than surface stress, and so if gear tooth failure occurred it would be by tooth breakage. This was a sudden event which occurred without warning and if the broken parts jam in the mesh the gears could be severely damaged and rendered useless.

Mr. H. H. Page said that Mr. Nicholson's paper had been read with some degree of satisfaction at the Admiralty, as it was seen that the decisions taken some years ago to recommend to the Royal Canadian Navy that they should undertake manufacture of hardened and ground gears had been fully justified.

Commander Weaving, in a written contribution, was dealing with experience on similar gears in the Royal Navy.

Dealing with the paper in some of the minor points, he said that with regard to scuffing it was now fully agreed that the 20 deg. pressure angle with standard tooth depth was satisfactory. It was thought that at the time the 23 deg. pressure angle was considered the problem was rather over-emphasized.

With regard to the deep freeze treatment mentioned by Mr. Nicholson on page 67, this had been found to be a disadvantage rather than an advantage. It had been introduced in the manufacture of gearing for naval service because it was considered that the conversion of the retained austenite would lead to dimensional change. It had subsequently been established when the matter was fully investigated that the danger of dimensional change even at 0 deg. C. was almost negligible. The first difficulty encountered was cracking during grinding. This had been attributed to the excessive hardness, that is 800-850 VPN. A maximum hardness of 775 VPN was now specified. The deep freeze treatment had produced a marked increase in hardness, and it was proposed that this should be overcome by increasing the tempering temperature after the deep freeze treatment above the 250 deg. C. previously in use. Some forgings had therefore been tempered to 300 deg. C. to 350 deg. C. and in a few cases it was found that the izod had fallen to 8ft. lb. due to temper brittleness. In view of the conclusion that the danger of dimensional change was negligible it had been decided to omit deep freeze, except as a possible means of recovering a gear which was too soft. His experience, therefore, appeared to have been contrary to the Canadians, as apparently they did not get the excessive hardness, and consequently they did not have to use the higher tempering temperature. His experience was based on the use of EN36A steel.

Another interesting aspect of this was in the significance of izods. It was of interest to note that as a result of work done on the effect of low izods all the gears were considered acceptable for service. This was a continuation of the work of Chester and Russell ("The Izod Test in Gear Design and Performance", Engineering, 7th August 1953) where the energy required to break a gear tooth type specimen was compared with the izod specimen. Using case-hardened specimens the following results had been obtained:

	Tempering Temperature	
	200 deg. C.	*350 deg. C.*
Standard Izod	95ft. lb.	17ft. lb.
Special tooth type	50ft. lb.	60ft. lb.

It was not clear whether Mr. Nicholson's conclusions that increased case depth reduced the resistance to shock, were based on the standard izod tests; if so, the conclusions were likely to have been different with the gear tooth type specimen.

In assembly and shop testing, the practice of testing each unit up to full power was considered a rather expensive and unnecessary procedure. The practice of his department, once a design was proved, was to only run it up to overspeed on "no load" as part of the turbine trials. It had not been the practice to check the gearcases, during installation, with a collimator, as it was considered that the three-area support at present used obviated the danger of any appreciable distortion.

Helix correction to allow for the side thrust of single helical gears had been discussed at some length, and the general opinion was that it was unnecessary, but service experience was awaited for final confirmation. As Mr. Nicholson had stated, there were many other "unknowns" which could not be taken into account.

The importance of the shafting on the alignment of gearing was underlined by Mr. Nicholson. This coincided with the opinion of the writer's department that every effort should be made to assist the gearing alignment. This was the reason why the department had adopted the practice of siting the first shaft bearing remote from the gearcase and attempting to distribute the shaft weight equally on the gearcase main wheel bearings. He thought this was a much more practical method of approach than a Bibby coupling as suggested by Mr. Archer.

Mr. Nicholson had mentioned in his text that standard concentric bearings were used. It was not quite clear how this was achieved in view of the fact that the bearings had to be scraped in order to achieve the original line. It had been his department's practice to provide means of adjusting the bearings and fitting thin-wall bearings and by fitting standard bearings and using accurate meshing frames it was possible to produce interchangeable gears and interchangeable bearings which had much to offer the users in service.

In conclusion, the general opinion of his organization was that hardened and ground gearing, as had already been mentioned, using nitrided primary wheels and induction hardened secondary wheels and case hardened pinions was more reliable and was cheaper in production than the ordinary through hardened gears.

Mr. R. E. Salthouse, B.Sc.Tech. (Associate Member) said Mr. Nicholson's paper contained a number of points which were exactly paralleled in the experience of hardened and ground gearing designed and manufactured in England, and he thought it might be of interest to list some of them, namely: (i) the satisfactory use of a normal pressure angle of 20 deg., and the difficulty of grinding teeth of higher pressure angles; (ii) the fact that the use of deep freezing could be satisfactory; (iii) the fact that hardened and ground gearing might be economically advantageous in addition to possessing high load carrying capacities; (iv) the fact that helix angle correction, determined on the basis of torque alone, could be misleading, and the fact that the effect of slew might be completely masked by relaxing effects in the positioning of journals and gearcase distortion.

The author had mentioned that helix correction had now been applied when required by regrinding the pinion. But if the pinions were so designed that the effect of torque could be ignored the only remaining correction required would be linear. The simple way of carrying out this kind of correction was to make the bearing housings adjustable.

The paper was a valuable record of considerable experience in the design, manufacture and use of case-hardened ground gearing. There were, however, a number of subjects on which further information would be welcomed. In the paper reference had been made to the necessity for close control of case depth. The ratio of case depth to module was said to lie within the limits of 0·07 and 0·23. This seemed to be quite a large difference, and the two statements appeared to be incompatible. No reference was made to grinding cracks, and it would be interesting to learn from Mr. Nicholson whether he had had any experience of this phenomenon. The importance of main wheel/shaft alignment had been mentioned, and it would again be interesting to know whether the author had measured the effect of misalignment in any way, particularly the misalignment between the propeller shafting and the light main wheel. Prolonged load runs had been made for the shore testing of each shipset, and one wondered whether such protracted running was necessary once the basic design of the machinery had been proved. The author's comments on this would be very much appreciated.

MR. T. I. FOWLE said he had one question on the subject of service experience in connexion with the use of the non-E.P. oil in one ship. There was evidence that a very short period of running on an E.P. oil, 100 hours or so, materially reduced the sensitivity of a particular gear to scuffing. This was perhaps not surprising because it was paralleled by experience with automobile rear axle gears. He therefore asked Mr. Nicholson whether this particular ship had had any running at all on an E.P. oil before it was changed over to non-E.P. oil.

MR. WATERWORTH remarked that following such an array of gear experts there was very little left in the paper on which to comment. On page 71 it was stated, "Class A1 accuracy is in fact achieved in the Canadian gear plant but there is no evidence to suggest that it was necessary for hardened and ground gearing except for the most stringent of naval requirements relating to quietness". Should not this be read with an additional paragraph to the effect that this accuracy was not necessary for K factors of up to and including 412? Considering land applications of gearing and the British Standard allowable surface stress values, the S_c values for the normal gear combinations of soft and through hardened materials were at, or approaching, the maximum, and it was also known that the 10,000 S_c values used for case-hardened En 36 and En 39 are not the maximum values. If this reasoning was applied to the K factor system used in marine applications, and there were no reason why it should not be used, it was not clear whether if at a later date these K factors for En 36 were increased, the author's comments regarding accuracy would still appertain.

On the same basis comment must be made with regard to the three and the five-point gearcase support. Was it to be understood that at 412K, there was so much in hand that the localized increase in tooth loading associated with deflexion in excess of 0·025in., a deflexion which was excessive and still present with the five-point support, was lost or adequately covered in the allowable load carrying capacity of this material?

The author had made some comments regarding the load-carrying capacity of case-hardened pinions mating with soft or through hardened wheels, to the effect that there was little to be gained by using such combinations of material and manufacture. Without being disrespectful, it was questioned whether the author had actual experience on this point or not. The question of accuracy was surely the most important, for there was evidence in this country that taking even a soft material and changing its standard of accuracy, particularly with respect to profile and by as little as a few ten thousandths of an inch, quite a difference in the load-carrying capacity had been made up to the extent of five times. With such confusion of the issue it was wondered if the comment referring to the use of hardened pinions and soft wheels was quite true.

Questioning again, there was a comment at the bottom of page 70 to the effect that the pinions were ground to obtain full depth meshing over the entire length. Was it to be taken that there was no profile modifications to these teeth? If there was profile modification it would be interesting to know its magnitude and disposition down the profile. It would be surprising if there was no modification, for he would have thought that it was as important on the criteria of scuffing as the question of accuracy and the pressure angle.

Correspondence

MR. J. CACCIOLA in a written contribution wished to emphasize that the following opinions were personal, and in no way reflected the views of any official departments.

He thought that the Royal Canadian Navy's application of carburized, hardened and ground reduction gearing for main propulsion was an admirable accomplishment very ably summarized in Mr. Nicholson's most welcome paper. The circumstances which dictated acquirements of such specialized gear production facilities appeared uniquely fortuitous. Other countries already in possession of conventional gear cutting equipment either had not been able to justify the required equipment on an economic basis or wished to avoid the problems entailed in the development of case-hardened and ground designs. U.S. Naval requirements for speed reducers as yet had not demanded the increased load-carrying potential of case-hardened gears or deviation from conventional practice utilizing through hardened steel gears. Indeed the author's claims for increased reliability of case-hardened and ground gearing was undisputed and could be substantiated by test experience of the U.S. Naval Boiler and Turbine Laboratory with load-carrying capability virtually double that of comparable through hardened steel gears.

Difficulties encountered in the experience of the writer's organization with early experimental case-hardened double helical gear designs, caused by excessive distortion in hardening, had not only severely penalized the product by cost and excessive production time, but had materially reduced load-carrying potential by the necessary excess removal of the beneficial carburized surface layer. In addition, several instances of excessive case depth and of grinding damage had demonstrated the unfavourable aspects which had been a deterrent to the adoption of this type of gearing. Recent laboratory experience with experimental single helical destroyer gears had shown that distortion could be held within acceptable limits utilizing separable rim design with quenching performance on a fixture. Maximum stock removal of only 0·005in. was found adequate to clean up and finish the tooth surfaces of these gears which were 24·6in. pitch diameter and 5¾in. active face width. In view of early experience, wherein necessary stock removal exceeded 0·025in. for comparable diameter gears, the latest results appeared quite promising.

The U.S. Navy's experimental gear development programme was continuing to ascertain relative load limitations and the more favourable production techniques for case-hardening gearing. Those tests, using six normal diametral pitch elements, were being made with carburized gears finished by lapping as well as by grinding and with nitrided and induction hardened types. The development of case-hardened gears for naval and marine use was still considered to be in its infancy. The encouraging experience of the Canadian Navy should materially promote further use of case-hardened gears and their eventual universal adoption.

MR. G. H. CLARK (Member) wrote that he found the paper most interesting and that it gave valuable information on the design and manufacture of marine reduction gears employing very high designed load factors.

He would be interested to know why single-helical gears were adopted in place of double-helical as used in British merchant ships and in the Royal Navy. How were the pinions and wheels allowed to take up their correct running position and how was unbalanced end thrust absorbed?

While appreciating the importance of surface hardness in connexion with resistance to scuffing, perhaps the influence of good surface finish might have been stressed. Very high K factors involved high tooth loading, defined in terms of lb./in. of overall face. This assumed contact over the entire length of the tooth face but in practice this could not be achieved due to limitations in accuracy of machining and grinding; in fact, the load was carried on the metal asperities of the mating teeth. Hardened gears could only be finished by grinding, which, if properly carried out, gave a very good surface finish, with smaller asperity amplitude. With hardened and ground gears, therefore, the actual area in contact per inch of tooth length might be greater then with hobbed and shaved "soft" gears. It was possible that even with K factors as high as 412, the true unit loading might be very little—if at all—greater than in less accurately finished merchant ship gears with designed K factors in the order of 90 to 100. The author's comments on this point would be appreciated.

Mention was made of Admiralty turbine oil specifications O.M.88, O.M.100, and to "an extreme-pressure oil" used during testing and in service of the gears in the *St. Laurent* Class destroyers. It would be helpful if details of these oils could be given. What were the characteristics of the E.P. oil and did it meet the Royal Navy O.E.P.90 specification? Even with oils meeting O.E.P.90 there could be a considerable variation in load-carrying properties, depending upon the amount and chemical activity of the extreme pressure additives incorporated. Everything else being equal, the more chemically active the additive, the higher the load-carrying properties, but unfortunately the more prone was the oil to cause corrosion in the oil system.

Shop testing of the gears under load was an excellent practice to adopt. "Running-in" with an extreme pressure oil should result in an excellent surface finish and should go a long way towards prevention of scuffing in service. To the best of the writer's knowledge, this practice had never been used in this country except perhaps for prototype sets of naval gears, and would be well worth while adopting for merchant ship gears—particularly if K factors continued to increase. Except for twin-screw ships, back to back running-in would not be feasible but it might be possible to use a brake in place of one of the gear sets.

CDR. A. J. H. GOODWIN, O.B.E., R.N. (Member) wrote that he regretted that circumstances prevented his attending the presentation of this excellent paper which provided a concise record of the problems met and overcome in the introduction of hardened and ground gears for the R.C.N. during the last decade.

There was, however, one statement in the paper where his own views differed from those of the author. He referred to the statement that it was possible, with hardened and ground gears, to permit higher design loadings than were acceptable in normal mercantile practice because naval gears might operate at low powers for most of their life.

He believed that hardened and ground merchant gears could be taken progressively to naval loading because:

(i) As stated in the paper, the load-carrying capacity is then no longer limited by the surface fatigue strength.

(ii) As stated in the paper, the loading criterion for such gears is the root strength.

(iii) The author omitted to make clear in the paper that Naval gears, although they spend much time at low loads, have run sufficiently long in *St. Laurent* Class to demonstrate that bending fatigue strength is satisfactory.

The experience to date in R.C.N. should lend encouragement to those shipowners who were considering the adoption of hardened and ground gears.

The author had given some indication of the manufacturing facilities provided in the government-owned gear plant in Canada. It would be of interest to know what type of balancing equipment was employed for the gear elements and the limits worked to, and also whether the torque tubes were balanced under torque.

MR. I. S. HILL felt that although it had been made clear that the tooth grinding operation on the gear rims was carried out after final assembly of the rims, discs and shafts, the general accuracy and especially the concentricity of the assembly prior to grinding must be of paramount importance. He wished to ask:

1) What degree of interference was used between the gear rims and discs? Was assembly achieved by heating the rim or freezing the disc?

2) What difficulties, if any, were encountered on assembly of the secondary gears where two discs, fitted one from each side, were used?

3) What sort of fit was used between the disc bore and the shaft location diameter?

4) What means were used of securing the discs to the rim and shaft?

MR. G. JOBLING wished to submit the following points for consideration. He suggested that instead of using a case-hardened steel, a nickel/chromium/molybdenum steel having a carbon content of over 4 per cent should be used. A suitable cast could be selected from British Standards specifications such as EN.24, EN.26 or one similar.

The steel could be carburized and heat-treated in the following manner: The blanks rough-machined but not gear-cut and normalized from 960 deg. C. The gear teeth would be cut, carburized and hardened, and tempered to the required core properties. The gear would then be corrected for distortion, if necessary, and carburized surfaces machined off where desired.

The opposite flanks and roots of teeth would be induction hardened using a suitably shaped heater; during this operation the gear could be immersed in oil. The roots of the teeth would be shot-peened to harden and compress the austenite and improve fatigue properties. The gear would then be tempered at 150 to 180 deg. C. and the flanks ground.

Better core properties should be obtained than with A.I.S.I. 3310, and harder case also could be produced by induction hardening as the cooling was effected by the quench medium as well as the internal mass of the gear inside the case.

The harder case produced should minimize the scuffing experienced. The carburizing time would also be reduced as the percentage of carbon in the steel before carburizing was approximately half that required in the outer layers of the case. If an uneven amount of case had been removed during correction after tempering, an even case pattern should still be produced by induction hardening as the core carbon was very much higher than in the normal carburizing steels, although there might be a slight loss in the overall percentage of carbon over the thickness of the case in places where it had been removed by correction.

It would appear that when flame hardening or induction hardening was considered, this hardening was only confined to the flanks of the teeth and not the roots. By hardening the roots of the gear also, the fatigue strength of the tooth could be improved considerably.

The grinding operation on the teeth might have to be modified owing to the harder case, particularly when roughing out. The hardening of the case would result in a saving of time with very much less distortion, the deep freezing treatment was replaced by shot-peening.

He did not know of any gears having been produced in this manner but could not see any major difficulties in the method.

As a matter of interest, his company had found that when hardening a set of four 21in. diameter and 4in. section carburized steel gearwheels, one on tops of the other, the following distortions occurred:

Top gear moved out 0·075in.
Second gear moved out 0·05in.
Third gear moved out 0·025in.
Bottom gear retained its dimensions.

It would appear that compression influenced the degree of distortion suffered.

After increasing the normalizing temperature from 880 deg. C. to 960 deg. C., the movement was in the order of 0·012-0·015in. after hardening.

Mr. G. Keenan (Associate Member of Council) wrote that the author was to be congratulated on a paper, which, by its nature, must be of great interest to all marine engineers. He had felt that the time had come for a complete re-appraisal of ideas with respect to marine gears, and he thought this paper might well pioneer this.

He noted that in the installations described, that not only had the K factor been increased considerably, but that there was a complete break away from the double helical tradition. The reasons which led to the adoption of this design in the early stages of the introduction of turbine gearing had long ceased to exist but there was always the reluctance to depart from a successful procedure.

The original Grade A gearing specification imposed quite fantastic limits of accuracy, quite beyond the accuracy that could reasonably be expected from cutting tools however rigid the hobbing machines and true the master gear. The hobs were far from ideal cutting tools, and grinding seemed to be a good method to make gears truer to profile and pitch.

It was regretted that the paper did not give details of the methods used in checking the pitch and tooth profiles. Measurements taken without de-greasing the wheels did not make for accuracy, and perhaps Mr. Nicholson could give some details as to the preparation and procedure for checking gears at the Crown gear plant in Canada.

In normal soft gearing not many defects were observed in high speed gears, i.e. first reduction gears or gears for turbo generators, but the story was vastly different for the second reduction or main gears. Here scuffing, pitting, wear or pitch line were all too familiar. He had observed over many years that a good gear presented a speckled appearance, and no trouble occurred until the small bright areas joined up and formed a continuous area of contact. When this happened the usual pitting, scuffing and pitch line indentation and deterioration occurred. Sometimes this reached a certain phase and got no worse. He was of the opinion that the success of this type of gearing owed its success to small undulations which were a product of the gear cutting process. These undulations formed minute wells for the lubricating oil, which could not be dissipated by heat whilst the teeth were passing through the mesh. Whilst the oil was on the tooth the surface (skin) temperature could not rise appreciably, and allow the welding and tearing apart process which was characteristic of tooth pitting. The oil contained in the undulations must be evaporated before the surfaces in contact could rise to a dangerous degree. Shaving improved the surface finish and the tool marks were in the right direction for a sliding tooth contact, this contributing to the success of those gears. However, the need for extreme accuracy made those gears extremely costly to produce. The tooth forms were long and slender and in order to keep the root stress down to an acceptable figure the K factor was kept to a moderate value.

In hardened and ground gears, the hardened surface was not so liable to plastic flow under pressure and generated heat, and it seemed that a highly accurate and polished surface would be a definite advantage, especially if the fine grinding and polishing processes were reasonable in cost. It was noted that rolling contact was preferred to sliding contact; could the cycloidal curve be readily produced by grinding?

Hardened and ground gears had been used for driving heavy industrial plants and no marine application could equal in severity the drive to a billet rolling mill. Here the gears were subjected to sudden shocks, abrupt reversals and very heavy tooth loading, so there did not seem any reason to regard hardened and ground gears as anything but robust.

With regard to the construction of the wheels shown in Fig. 12, were the rims completely finished before assembly to the hub and disc? In Fig. 12a it was noted that the wheel disc was single and dished. Was this to give the rim resilient support? If so he regarded it as a most valuable feature. Cyclic acceleration produced by pitch errors caused shock to the teeth and this could produce heavy contact pressures if the rim was held by a rigid centre. The effect of shocks arising from pitch and profile errors could be minimized if the rim was allowed to "breathe" on the wheel. Had any consideration been given to the probable effect of a "hard" or "soft" centre to gear wheels. A design allowing the gear shroud to slide on the centre core of the wheel, the drive being effected through resilient elements might eliminate hammering arising from pitch and profile errors. The weight of the shroud would be reduced to a practical minimum and the effect of mass reduced.

With regard to noise, one of the causes of noise arose from the fact that the pinions were of sound homogeneous steel and of such physical dimensions to be vibrated in the audio frequency range. In fact one of the best ways to test for suspected tooth root fractures was to strip the pinion completely, suspend it by a hemp rope and strike it with a copper hammer. If sound it would ring and the vibrations would be sustained for a considerable time, so it would seem that a small measure of excitation could produce a ringing noise in the gears. Pinion noise could be stopped by fitting an artificial flaw. The end of the pinion journal could be recessed to take a cast iron piston ring or a circlip. This would detune the pinion. Gear casings were good resonators but the propagation of noise from those could be greatly reduced by lagging, perforated metal sheets, etc.

Fig. 3a showed the effect of deflexion under gear loading and the effect of torque. The case appeared to be sufficiently rigid, but the foundations did not appear to have the necessary stiffness to resist the engine torque. Perhaps this had to be tolerated in warship construction.

It would seem that great attention should be paid to the design of gearcases for stiffness, or at least uniform deflexion of the pinion bearings with reference to the wheel bearings should be the aim. Any design which permitted the axis of the bearings to deflect out of plane due to gear loading will produce extremely heavy local loading on teeth of stiff section.

There was no doubt that the single helical design with hardened and ground teeth was a far more robust proposition than the usual double helical installation.

The design of bearings was interesting and details including lubrication arrangements would have been appreciated. Many main turbine bearings had huge expanses of white metal innocent of any means of introducing oil. Cooling apparently relied on the flow of oil through the bearing clearances. If fine clearances were desired, then oil should be circulated as a cooling medium around the bearing at low pressure and a supply of oil at high pressure but small in quantity to maintain lubrication. Experiments had shown that considerable surface temperatures had developed in normal design turbine bearings. If the design called for close bearing clearances surely some external cooling was necessary.

In a warship, where damage from shock arising from remote massive explosions might be a major source of casualties, surely the thrust block and main shafting should be isolated from the gearing by a flexible coupling. This would eliminate any stresses on the gearing from outside sources.

This paper had been of great interest to him and he wished to express his appreciation to the author for his most valuable paper.

Dr. H. E. Merritt, M.B.E., wrote that Mr. Nicholson's most interesting and valuable paper prompted the following observations, based on experience of other applications in which case-hardened gearing was well established.

The author rightly said that surface fatigue strength was not a limiting factor in case-hardened gears (at least, with the tooth numbers employed in the cases mentioned). The contact-loading criterion S_c = unit contact-line load ÷ relative radius of curvature was approximately equal to 6·3K where K was the usual Lloyds factor; so that the gears in Table I ran at around S_c = 2,000 and 2,500 (nominal) for the primary and secondary gears respectively. The endurance limit of case-hardened steel for surface breakdown, taken at 10^8 cycles, was around S_c = 16,000, and some vehicle transmissions operated at occasional peak stresses of this order, or higher.

Anticipating greater use of case-hardened gearing for marine propulsion, more study of bending stresses was needed, to lead to an accepted bending-stress criterion as a companion to the K-factor for surface loading. This would be desirable even for soft-gear applications. Meantime, what was already known, supported by developments in the aircraft industry, suggested a new approach to tooth proportions for case-hardened marine gears, in the direction of higher pressure angle and reduced depth of tooth.

The author described the difficulties experienced with a pressure angle of 23 degrees. But this reflected merely a limitation of the particular arrangement of grinding wheels employed, and was not fundamental. Given an improvement in this respect, an increase of tooth strength, through changed proportions, of 25 per cent could be expected, combined with lower sliding velocities and reduced liability to scuffing.

In the automobile industry, where profile grinding was not economic, a high degree of uniformity in heat treatment distortion was achieved. Some of the measures employed were control of grain size, control of forging procedure, and normalizing at above the carburizing temperature. In large marine wheels, consistent forging technique might not be feasible, but high temperature normalizing might well be beneficial.

The time of 3½ minutes from leaving the furnace to quenching gave an abnormal opportunity for surface decarburization to occur. The fatigue strength of the material at the surface of the tensile fillet curve (which was very properly untouched during the grinding operation) might be seriously impaired; but shot-peening would be of marked benefit, if carried out under controlled conditions.

In some automotive applications, carburized gear rings were quenched on their final centres, which controlled size and roundness and provided a shrink fit. From his experience, the author would be able to appraise the production problems which would arise in the wheels with which he was concerned if this were attempted, given some re-arrangement of the overall design. The principal technical problem would be that of the shrink-fit tensile stress in the rim, and this might be minimized by individual induction hardening of carburized teeth followed by shot-peening.

The author's comments would be valued on why so much trouble was taken in removing the carburized layer from all except the tooth surfaces. Apart from surfaces which had to be machined after hardening, had this been demonstrated to be necessary? Vast numbers of automotive gears, of comparable tooth dimensions, operating at stress-levels from three to ten times as high as those described in the paper were not so treated.

Mr. A. J. Morton (Associate Member) wrote that the gearcase distortion readings shown in Fig. 3(c) gave rise to some significant design considerations. In the 130 per cent torque condition, the gearbox rotated about a fore and aft axis, the rotation at the after end being greater than at the forward end whichever of the two chocking arrangements was employed. This implied axial twisting of the box, which must have thrown wheel and pinion axes out of parallel and tended to cause concentrations of load at one end of the teeth. This uneven tooth loading would cause the two bearings of an affected wheel or pinion to be loaded differently and therefore to have different oil film thickness, thus offsetting to some extent the effect of twisting of the box. Estimates of these quantities were naturally crude, but it was shown after the trials that they were by no means negligible in comparison with the helix correction applied to the pinion teeth to correct for bending and twisting of the pinion itself. To apply helix correction on a sound basis, therefore, the deflexion of the gearcase itself should be taken into account, and this required more attention to gearcase design than had hitherto been customary.

To predict the degree of twist which would occur in a gearcase supported as in Fig. 3(a) was hardly practicable, but the twist could certainly be minimized by a more logical chocking arrangement. The major cause of the twisting was the torque reaction at the main wheel, which was transmitted through the bearings to the gearcase itself and thence through the chocks and seatings to the hull. The gearcase, being an elastic structure, would naturally twist axially if it was required to transmit a heavy torque from one transverse plane to another, and to prevent this the main wheel torque reaction should be resisted by chocks in the same plane as that in which it was developed. This called for chocks on either side of the main wheel. To support the box adequately a third chocking point was required, and this should naturally be at the other end of the box on the centre line, where it could play no part in resisting axial torsion. The only torque carried by the gearcase would then be the relatively small one generated in the plane of the primary wheels. A three-point chocking system also had the advantage that hull distortion resulted only in bodily movement of the gearcase and not in distortion of it.

It was customary to stiffen gearcases by means of external webs on the side, top and end panels, these webs being always disposed vertically and horizontally, i.e. parallel to the edges of the panels. Twisting of the gearcase must involve both shearing and warping of the panels, and webs arranged parallel to the panel edges were relatively ineffective in resisting either of these actions, as straight lines parallel to the edges were not deformed thereby. By arranging the webs diagonally, much greater strength was obtainable, since shearing of the panels threw the webs into either tension or compression, and warping caused them to bend. It was possible that the much heavier scantlings customary in merchant ships made these points academic, but for light, powerful warships they would appear to be important.

This whole subject of gearcase distortion provided an excellent illustration of the value of thorough test bed trials of prototype machinery under load. Such trials did far more than provide steam rate and efficiency figures, valuable though these were—they made it possible to measure quantities which could not be covered in sea trials, and to do so under special conditions which were very revealing from the design standpoint. Prototype trials of this type, properly planned and analysed, could make a great difference to ultimate reliability, and the *St. Laurent* Class ships undoubtedly benefited very substantially in this way.

Mr. A. D. Newman wrote that Mr. Nicholson was to be congratulated on the very large amount of experience in hardened and ground gearing detail in his paper, experience which had been remarkably free from trouble. He was most interested to see his claims that the time involved in producing such gearing could be appreciably less than for the equivalent design used through hardened steels and that the cost could be lower. In his table of costs he saw that the induction-hardened gear cost index was slightly lower that that of the carburized and hardened gear, and he wondered whether Mr. Nicholson had any views on the relative position of nitrided and ground gears or indeed gears nitrided after grinding and shaving.

In his description of service experience Mr. Nicholson discussed the effect of propeller shaft alignment on the meshing of the secondary gears, and he mentioned this largely from the static aspect of the basic alignment in the ship. His experience with light high-speed twin-screw machinery of a somewhat similar type to that with which Mr. Nicholson was concerned was that the change in alignment caused by the ship manœuvring, especially at high speeds, could be more important than the basic alignment. The differences in immersion between the two propellers when the ship was heeling while manœuvring together with the differences in rotational speed and transmitted torque appeared capable of producing complete changes in attitude of main wheel bearings, particularly the forward bearings, leading sometimes to bearing failure and presumably to gearing misalignment. He wondered whether Mr. Nicholson had experienced this effect?

COMMANDER P. D. V. WEAVING, R.N., in a written contribution wished to congratulate the author on his excellent paper and to thank him for reporting such valuable experience with hardened and ground gears. The decisions to fit such gears in those ships and to set up manufacturing facilities in Canada required sound technical judgement and courage and it was most gratifying to know that this policy had been so amply justified by the excellent performance of the gears in service.

For a similar class of Royal Navy vessels, at about the same time, the decision was taken to fit through hardened, hobbed and shaved gears, the details of which have been given by Page*. Although these gears had given satisfactory service in a number of ships, in others their performance had been disappointing and both pitting and scuffing had been experienced, particularly of secondary pinions and in a few cases excessive wear of tooth profiles. It was believed that the combination of materials (EN.26 and EN.30B) used, had, at this loading (primaries 230K, secondaries 270K) insufficient margin of safety to cover minor errors in gear cutting, assembly and installation in the ships. Later ships of the class had carburized and ground secondary pinions and had been entirely satisfactory, as had the gears of H.M.S. *Diana*, which were still in excellent condition.

The general purpose frigates and guided missile destroyers now under construction were being fitted with hardened and ground gears. The majority of these were carburized and ground but certain ships' sets included a number of nitrided and induction hardened gears. So far, the full scale gear tests carried out by the Admiralty-Vickers Gearing Research Association, the shore trials of prototype machinery sets and sea trials of the first ship also supported the author's opinion that hardened and ground gears offered considerable increases in the margins of safety, even when loaded between two and three times as highly as through hardened, hobbed and shaved gears.

Notwithstanding the increased root strength which was believed to result from carburizing, full scale tests carried out by the Admiralty-Vickers Gearing Research Association indicated that both induction hardened main wheels and nitrided primary wheels would successfully carry loads of the same order as carburized wheels. Since both these hardening processes resulted in very much less distortion than carburizing, with appreciable reductions in grinding times and manufacturing costs, it seemed likely that the use of such gears in the British Mercantile Marine could not be long delayed.

The successful production of more than twenty highly loaded carburized and ground gear sets was no mean achievement and the author's account of the manufacturing facilities and production details was of great interest. Despite certain differences in equipment and procedure the recorded figures for distortion and grinding times appeared to be of the same order as those experienced in the U.K., some of which had already been reported by Chamberlain*. Less distortion should result from a reduction in carburizing time and it seems possible that some reduction in case depth would be acceptable. It would be of interest to know the author's views on this.

The practice of "back to back" testing before installation in the ship appeared to offer the advantages of earlier detection of faults at a time when rectification could be undertaken with a minimum of inconvenience and delay and also permitted some "running in" of the gears before they were exposed to the severe conditions of high torque and low speed which could arise during manœuvring on sea trials. Although R.N. warship gears were run at full speed during shop trials they were not run on load before installation in the ship. It was believed that the time and expense of back to back testing before installation would be fully justified. It would be interesting to know whether or not the author's experience supported this belief.

It was noted that it was a Royal Canadian Naval requirement for gear designs to operate satisfactorily with standard turbine oils. For those Royal Navy ships in which weight and space were at a premium it was, at present, believed that the advantages of hardened and ground gears could not be fully realized unless E.P. oils were used and it was the intention to continue to use such oils in the main and auxiliary machinery of modern Royal Navy ships.

The remarks on the effect of propeller shaft alignment on the meshing of the main gear wheel were of particular interest since it was now believed that the troubles with through hardened gears in certain Royal Navy ships already referred to had resulted, in some cases, from this cause. In ships now building, efforts were being made to ensure that the alignment was such that the intermediate shaft would not impose bending moments on the main gearwheel. The problem was considered in detail by Andersen and Zrodowski† who produced convincing arguments and evidence for their case. Nevertheless, it had been suggested that a frigate hull and shafting installation was so flexible that any bending moments exerted on the main gearwheel were negligible and that, in any case, the changes in alignment which occurred as construction proceeded, in a seaway and in different conditions of draught and trim outweighed the effects of changes in alignment procedure.

It would therefore be of great interest to have more information on the procedure now followed in the Royal Canadian Navy ships and in particular, to know whether an allowance was made for thermal expansion of the gearcase, the condition (i.e. draught and trim, etc.) of the ship when the main gearwheel was aligned to the shafting and whether or not calibrated jacks were used to check bearing reactions.

The author's remarks concerning the slew of single helical gears were of particular interest. In the Royal Navy hardened and ground gear sets now being built, calculations indicated the necessity for helix corrections but in view of the uncertain basis of such calculations it was decided not to correct the gears. Adjustable bearing housings had, however, been fitted. So far, prototype shore trials had indicated that this decision was correct and it had not been necessary to adjust bearings.

* Page, H. H. 1958. "Advances in loading of Main Propulsion Gears", International Conference on Gearing.

* "Developments in the Heat Treatment of Large (Marine) Gears", International Conference on Gearing, 1958.

† "Co-ordinated Alignment of Line Shaft, Propulsion Gear and Turbines", Trans.I.Mar.E., Vol. 70, pp. 135-185.

Discussion in Ottawa

FRIDAY, 26TH MAY 1961

J. J. ZRODOWSKI said that he had read, with great interest, the author's informative paper on the Royal Canadian Navy's experience with hardened and ground gearing. There was no denying that higher gear tooth loads could be used with case-hardened and accurately ground gears. However, he questioned the author's statement, that a peak design loading of up to 250-K might be permissible, in naval gear installations for through hardened, hobbed and shaved gears. Much higher K factors had been successfully used on this type of gearing; for example, on high-powered U.S. Navy destroyer leaders, which had been in active service about five years. The propulsion gear K factors on these hobbed and shaved gears and pinions were 300 on the first-reduction, and 290 on the second reduction. The gear tooth loads per inch of face were 2,050lb., on the first reduction and 2,830lb. on the second reduction. There were no signs of any distress on the pinion and gear teeth after this period of operation. The pinions had a hardness range of 350-400 B.H.N. and the gear rim material has a 300-350 B.H.N. range.

Based on design studies, which he had made on high-powered Navy propulsion gears, there was a saving in weight and size of the gears in using case-hardened and ground single helix gears on the first reduction and through hardened hobbed and shaved gears on the second reduction. The first reduction gears would be made from slick type form rolled forgings, including the gear rim, web and hub of the narrow faced gear. Such a design should have a minimum rim distortion, during the case hardening process and would eliminate the objectionable features of either a bolted or shrunk on gear rim. However, the noise level would be higher than for an accurately hobbed and shaved gear.

There was also some weight and size advantage, in using case-hardened gearing on the first and second reductions of the *St. Laurent* Class destroyer escorts, where the shaft horsepower and overall ratio were comparatively low. It was, however, noted that, even with the small 67in. pitch diameter and narrow 13·75in. face width, an average of 400 hours of grinding time was required, to produce a gear with tooth spacing errors slightly outside the values specified for British Standard 1807 Class A1. For greater powers on naval vessels and merchant ships, where larger diameters and wider face width second-reduction gears were required, the distortions would be greater, with a resulting much longer grinding time, to obtain the same degree of accuracy. There were other problems, encountered with excessive gear rim growth and distortion, which were well defined by the author at top of page 66. The author stated, that it was not only necessary to keep grinding stock down to a minimum, in the interest of reasonable manufacturing time, but there was also a very definite limit on the amount by which the carburized case could be reduced without seriously weakening the tooth.

Since it was not possible to use case-hardened gears, on diameters larger than 70in., because of excessive rim distortions, there was, in his opinion, no advantage in using ground second-reduction gears for large power navy and merchant marine propulsion gearing, where larger than 70in. diameter gears were required. Because of this limitation, all of the ground second reduction gears now in merchant marine service, had oil or air through hardened, instead of case-hardened, teeth. The oil or air through hardened gear teeth had the same degree of hardness used on hobbed and shaved gears. It was the custom, in his company, to precision hob and shave, within British Standard 1807 Class A1, large diameter, double helical second-reduction gears, having forged alloy steel gear rims with a hardness range of 300-350 B.H.N. In so doing they were able to produce a large diameter gear, to high precision accuracy, in less time and without the objectionable features of end thrust deflexions of the gear rim, due to single helix of a ground gear. A single-helix ground gear also had a small helix angle of approximately 10 deg., which approached a spur type gear. A spur gear had no axial crossovers; therefore, it was much noisier in operation, than a double-helical gear, with its many axial crossovers and smoother meshing in the arc of action. It was noted that the paper did not make any reference to the engine room noise level, in the vicinity of the propulsion gears. Did the author have this information and comparable operating data, on noise level of hardened and ground and hobbed and shaved propulsion gears? Quietness of operation of all machinery in the engine room, on both naval and merchant ships, was now more important, as noted by recent naval architects' and shipbuilders' specifications for propulsion machinery, wherein maximum acceptable noise levels were specified.

In regard to Table V on relative dimensional and cost indices for hardened and soft gearing, he did not agree on the relative cost difference of 4 per cent between the 80-70 K factor and the 120-100 K factor gears, as given in the table. Their quoted price would be 10 per cent less for the 120-100 K factor gear. This was based on design cost studies and other cost analyses, assuming that all pinion and gear diameters remained alike, in both cases, to accommodate the cross-compound turbines and condenser. However, the face width of the first reduction pinions and gears would be reduced 40 per cent and the face width of the second reduction pinions and gear would be reduced 30 per cent. The pinion and gear rim material, shafting and gear housing length, in way of the tooth area, would also be reduced by an equal amount. He could not make any factual statements on cost comparison, between a through hardened, hobbed and shaved gear and a hardened and ground gear, but he believed that the relationship, as shown in Table V, would not hold true for gears manufactured in the United States. In fact, in the company which he represented, hardened and ground gearing would require more time and be more expensive to manufacture, because large size gear teeth could be hobbed and shaved to high precision accuracy in high hardness, through hardened alloy steels in less time, on the rugged and highly accurate custom designed and built, hobbing and shaving machines, than by the much slower process on the Maag disc grinding machines.

MR. H. A. SLEDGE (Member) said that the author had presented a detailed and comprehensive paper on the manufacture of hardened and ground gears and he, along with his colleagues, was to be congratulated upon his treatment of the subject. Since the paper purported to be on experience with the gearing in the Royal Canadian Navy, he suggested that reference to coupling design, methods of lubrication, gearwheel thrusts and instrumentation would have added to its general interest. Perhaps the author had reserved these, along with other items which might have been considered as standard equipment, for a future paper which they might look forward to with anticipation.

In the paper recently presented to the St. Lawrence-Ottawa Section by Mr. Kilchenmann, mention had been made of the advantages gained from using modern lubrication oils in the cylinders of large Diesel engines. Mr. Nicholson reported that a gearing unit on test had been run using E.P. oil and the resulting condition of the gear was found to be excellent. The unit had then been run on a standard lubricating oil and scuffing had been observed. In this case the demonstrated advantages of the E.P. oil had not been utilized. No doubt the author had carefully investigated the situation and perhaps he would be prepared to discuss the reasons for the decision.

Operating conditions had not been referred to in the paper in detail, but it was usually accepted that gear efficiency was affected by lubricating oil temperature and that reduction gear horsepower losses increased, with decreases in lubricating oil temperature. Could the author indicate what temperatures were recommended for use with the gear sets under discussion, both when running on E.P. and standard lubricating oils.

The advancement in design of hardened and ground gears would appear to be based upon several factors, but the author had stressed two; these were increased K factors and decreased production costs. K factors, in themselves, would not necessarily indicate to what degree a propulsion gear could be considered conservative or progressive, because they were only an index of the compressive stress at the area of pinion and gear tooth contact. Other important factors were actual tooth loading and root stress.

412 K would be considered comparatively high, but a large power planetary gear using hobbed, shaved and nitrided pinions and planets had been developed, using a K factor of 781. In this case the ring gear was hobbed, shaved and through-hardened. Inasmuch as the Canadian Crown-owned Gear Plant was also involved in development and design, had this type of reduction unit been considered, using hardened and ground gears?

Where a single helical gear, such as was described, was utilized, the provision of additional gearwheel thrusts must have consequent economical repercussions. With a double helical gear, the additional thrusts were not required. Would the author consider the production of double helical hardened and ground gearing units, economically and practically, possible?

A factor of growing importance, particularly in the type of vessel in which the gear units were installed, was that of noise. Undue noise was construed to be that which prevented ordinary conversation between persons standing in the proximity of the gear sets. Would the author give some indication of the relative noise level with the gearing concerned, under operating conditions?

Details of accuracy and ship tests had been covered in detail but no mention of static or dynamic balancing of the finished gearwheels had been made. Would the author confirm whether it was intended to consider these tests as being of standard procedure?

Finally, in the short history of the section, he felt they could be grateful for the high standard of papers presented before them and the paper that evening had been no exception to the rule.

MR. J. LONGHURST, B.Sc.(Eng.) Lond. wished to congratulate the author on the excellent and most informative paper he had written on a subject of great interest to all those concerned with power transmission. He felt, also, that the Royal Canadian Navy had earned no little credit in its decision to accept a main propulsion gearing system, which, as this paper so well demonstrated, was a more advanced development of hardened and ground parallel shaft gearing than had been tackled before.

It had been said that gearing was a necessary evil in the present state of the art of power transmission, where, for the most part, prime movers rotated faster than driven machines. Faced with this "necessary evil", therefore, it was sound practice to make it as inconspicuous as possible. In addition, speeds of prime movers, particularly, pure rotational ones, had been increasing at a faster rate than most of the shafts or machines driven by those prime movers. Obviously, this would mean larger reduction ratios and consequently, bulkier and more unmanageable reduction gears, unless steps could be taken concurrently to make the rotating gear element smaller. One of the ways by which part of both these aims could be achieved was the adoption of tooth surfaces able to withstand greater loading.

If one could attempt to read between the lines of this paper—and could do so with accuracy—development of the hardened and ground main propulsion gearing for the Royal Canadian Navy destroyer escort vessels brought its headaches as well as its triumphs and the solutions of the problems encountered must have required much perseverance by the designers, the manufacturers and the Royal Canadian Navy. He hoped that there would be contributions to this discussion from both the designer and the manufacturer. He wished to confine his remarks, therefore, to the one or two parts of the author's paper with which he had the presumption to take some issue and upon which he sought more information.

The author had covered, quite exhaustively, the various manufacturing processes concerned with the production of the carburized, hardened and ground gear elements. He was not telling the author anything he did not know when he said that there were many advocates today of the use of nitriding steels and the nitriding process for gear components of the size of those in the primary train and secondary pinion at least. In the early 1950's, he heard of one gearing designer and manufacturer who had produced shaved and nitrided gear components of 48in. diameter without measurable gear tooth distortion and, no doubt, larger wheels than this were common today. The author made no mention of any consideration of this method. If considered, why was it rejected? If the use of nitriding steels and nitriding presented few problems for the primary train and secondary pinion, could not the method have been developed for the production of the secondary gearwheel, thus reaping the ensuing manufacturing benefits?

It was encouraging to know that experience with standard turbine lubricating oil had been successful in one vessel, fitted with the revised gear design. As would be known by those who had had experience with some types of lubricating oil with extreme pressure additives, the use of such oils brought its own problems with it and he hoped that the day would not be far off when the Royal Canadian Navy would feel able to revert to the more standard types of lubricating oils.

The author suggested that soft gearing would be unlikely to withstand those same conditions of operation, with particular reference to the doubtful application of helix angle correction and gearcase twisting. He understood that there were a greater number of hobbed and shaved gears in service in similar main propulsion installations in the Royal Navy than there were hardened and ground installations in the Royal Canadian Navy. Could the author give any information on the performance of those Royal Navy's main propulsion gears, with special reference to those problems?

Lest the author felt that he was not an advocate of hardened gearing, he wished to mention that he was privileged to be associated for several years with the development of "Allen-Stoeckicht" epicyclic gearing. That being so, he would remind the author that "experience with hardened and ground gearing in the Royal Canadian Navy" was not confined to this main propulsion gearing. He suggested that this main propulsion duty, today, would be considered a "natural" for Stoeckicht gearing, where the drive was from a single cylinder turbine, through an overall reduction ratio of approximately 25:1, to a single shaft in two trains in series. He was sure that a double-helical epicyclic gear of this design would achieve even further savings in weight and space and would eliminate a considerable number of the manufacturing problems which have been associated with the present design.

In adapting hardened gearing to mercantile marine applications, the author had mentioned the difficulty of obtaining adequate condenser space. If one considered the two-cylinder, cross-compounded, geared, condensing turbine installations common to many of the larger vessels built in Canadian shipyards today for commercial tankers, lakers, ore carriers and other similar applications, the problem was not only one of adequate condenser space but also, or adequate centre distance between the two turbine cylinders. However, some advantage of the smaller size of hardened gear components could be obtained by marrying hardened and soft gearing to reduce the size of the main gearwheel. So far, this marriage of convenience had not been widely consummated, probably because of development costs in a highly competitive market, but perhaps, the recent offer of a transfusion to the life blood of Canada's shipping industry might see the adoption of hardened gearing, even in this limited way, by Canadian marine engineers for mercantile as well as Naval applications.

Mr. J. T. Campbell, on behalf of Mr. G. T. R. Campbell, P.Eng. (Member) and Mr. N. V. Laskey, B.Sc.(Eng.) said they had read Mr. Nicholson's paper with interest, particularly the section dealing with the elaborate and complex metallurgical processes of pre-quenching, carburizing, diffusing, annealing, hardening, quenching, tempering, deep freezing and re-tempering.

The question to be posed in the light of what Mr. Nicholson had said was whether or not it was commercially desirable to subordinate all the advances made in the design and manufacture of hobbed and shaved gears to the somewhat questionable need for the employment of hardened and ground gearing in ships.

The record of gear performance to date in the mercantile service was, by and large, very satisfactory and failures, except in a few isolated cases, could be traced invariably to extraneous causes rather than to a shortcoming in gear design. A few cases in which premature and accelerated tooth distress had been recorded were usually, on analysis, traced to laxity in production control, defective material or unsatisfactory installation.

The complete confidence shown by shipowners today in hobbed and shaved, double helical marine reduction gears was exemplified by the fact that large vessels such as 75,000 d.w.t. tankers were accepted after a low powered dock trial of six hours duration, followed by a sea trial when the propulsion unit was gradually brought up to full power over a period of eight hours. The full power trial was continued for eight hours after which a two hour overload trial was conducted. The astern trial was confined to one hour on the basis of 80 per cent ahead torque at 50 per cent ahead revolutions.

After delivery, those vessels operated at full rated power for 300 days a year without even the use of an extreme pressure lubricant during the first few months of service.

In view of this, there was hardly any likelihood that hardened and ground gearing would supplant hobbed and shaved gears in the mercantile service. The remarkable service reliability, no doubt, reflected the extreme precision obtainable with modern gear hobbing and shaving processes. As an example, it was claimed by one United States gear manufacturer that even with gearwheels as large as 200 inches in diameter an overall accuracy of 0·0002in. could be guaranteed.

In the U.S.A., the General Electric Company had produced for the U.S. Navy compact and light weight, double, locked train gear sets using hobbed and shaved gears operating with a K. factor of 303 in the primary reduction gear train and 294 in the secondary. The gearwheel rims were manufactured from chrome/nickel/molybdenum steel with a Brinell hardness number of 300-350. The pinions were manufactured from alloy steel and heat treated between the hobbing and shaving process in order to increase the Brinell hardness number to a value between 350-400. As a weight saving artifice, the primary and secondary reduction gears were welded integrally with their hollow shafts and the gear-box was fabricated from steel plate and welded throughout.

In contrast to this, the hardened and ground gearing adopted by the Royal Canadian Navy for the fourteen ships of the *St. Laurent* Class had been designed to operate with a K factor of 320 in the primary reduction gear train and 412 in the secondary.

It was necessary, therefore, to examine the ramifications in designing a gear train for these high K. factors and determine what had to be sacrificed from a gear design standpoint to achieve this end.

1) Operating experience since the advent of gearing had established beyond doubt that a double helical gear train with its many axial cross-overs at the point of mesh made for quiet operation. With hardened gearing in which the tooth profile had to be finished by grinding, a double helical configuration could not be employed. Single helical gears had therefore to be used. The use of single helical gears did not provide grounds for objection, but owing to the end thrust which was produced, the helical angle had perforce to be reduced to a minimum. In the gears under consideration, the helix angles in the primary and secondary reduction gear trains were 10 deg. and 6 deg. respectively. From this, it would be appreciated that those teeth closely resembled those of a straight spur gear which were known to be noisy in operation. Another feature in this design which would normally be noise provoking was the very coarse pitch of the teeth. In the original design, diametral pitches of 4·12 and 2·13 were employed in the primary and secondary reduction gear trains in which normal pitch values of 0·742 and 1·463 respectively obtain. Those diametral pitches were altered to 4·79 and 2·79 and the pressure angle modified from 15 deg. to 23 deg. and 20 deg. in the primary and secondary gears respectively.

   This very coarse pitch on the gear teeth and particularly so in the secondary reduction gear train was a design characteristic of those hardened and ground gears dictated, to a large degree, by the high order tangential tooth loading of 4,666lb./in. face width of tooth. The unit load (i.e. diametral pitch × tangential load/inch face width), was 13,018. Alternatively, it might be argued that a coarse pitch had to be selected in order to permit adequate depth of carburized case after finish grinding, as Mr. Nicholson stated that as much as 0·030in. had to be ground from each flank in order to produce the designed tooth form. It would be observed from the curve of hardness shown in Fig. 14, that values of hardness dropped rapidly beyond a depth of 0·040in. from the carburized surface.

2) The axial thrust referred to above had to be accommodated on thrust collars. It was agreed that the axial thrust produced by the single helical gearing could be reduced to a minimum by suitably opposing the thrust of the intermediate rotating elements, but, nevertheless, thrust collars on the pinion and gear shafting with flat land, babbitted thrust bearings on either side of the pinion or gearwheel were notorious for their predeliction to "wipe" for no apparent reason.

3) On Fig. 12(a), the final assembly of the primary

reduction gearwheel was shown. It consisted of a single, circular, offset, flanged plate which was bolted to a flange on the shaft and to a central rib on the inner periphery of the gearwheel rim. This arrangement was asymmetrical to say the least and would not be acceptable in mercantile practice. Bolted gearwheel assemblies had long since been disused in mercantile practice. The reason for this was because a bolted assembly of the type illustrated had always been suspect. Frettage at the conjoint surfaces was unavoidable and this mechanical interaction in association with even the most minute surface asperity in the bolt resulted in a drastic reduction in the fatigue strength of the bolt material at which time crack initiation and propagation took place. In merchant vessels there had been occasions when gearwheel bolt heads had broken off and passed through the gear mesh destroying the gear train. One such case, was brought to light some two years ago. The assembly of the secondary reduction gearwheel was symmetrical, but exhibited the same drawback of a bolted assembly.

In his opening remarks, Mr. Nicholson mentioned that the main considerations influencing choice of hardened and ground gearing were the requirements for maximum reliability at minimum cost. He went on to say at the end of his paper that service experience with the gearing in all fourteen ships of the *St. Laurent* Class over the last five years since commissioning had been excellent. He also stated that naval gearing was operated at its maximum rated power for only a small percentage of its total life and might well be operated at as low as 10 per cent power for over 80 per cent of its life.

The author was therefore asked on what basis was it claimed that the service experience was excellent? Of the five years service, for how many hours had the gears been operated at their rated output of 15,000 s.h.p. and for what duration had the longest run at full power been maintained?

No justification could be seen for the adoption of hardened and ground gears of the type described particularly as the demand for full power in a naval vessel was limited.

With regard to reliability, the facts of the case were examined by way of an example. It had been shown that the "unit load" on the gear teeth of the secondary reduction gearwheel was 13,018. In contrast to this, the unit load of the secondary reduction gearwheel of a 50,000 s.h.p. double reduction, twin drive, locked train, propulsion gear of an aircraft carrier was 9,955. The gear teeth were of the double-helical type, hobbed, peened and shaved, 2·96 diametral pitch, 20 deg. pressure angle and the K factor was 175.

In the light of this, could it be claimed that the aircraft carrier gear with a unit load of 9,955 was less reliable than the gear on the *St. Laurent* Class vessel which was designed for a unit load of 13,000. After the last war, the U.S. Navy purchased from Switzerland a pair of gear-boxes of similar design to that described by Mr. Nicholson and conducted a front to front torque test. Details of this test were given in a paper presented to S.N.A.M.E. on November 13th 1952, by Commander Ivan Monk, U.S.N., Lieutenant-Commander L. J. Thomas, U.S.N.R., and C. C. Atkinson.

In this paper, it was mentioned that the only visible signs of distress which manifested itself in the gear teeth of these hardened and ground gears was "scoring". The consensus of opinion at that time was that this scoring could be eliminated by involute modification or by the use of an E.P. lubricant. The teething difficulties with the gearing described by Mr. Nicholson confirmed this.

Mr. Nicholson in his paper extolled the advantage to be gained by the use of hardened and ground gearing and went on to demonstrate how the cost of such a unit would be less than the conventional type of gear. It appeared that there was only one palpable advantage which could be claimed for such gearing. The saving in weight and space it afforded was the only attractive feature but Mr. Nicholson made no mention of this fact. The cost was prohibitive, although it was claimed that hardened and ground gearing was chosen in this case as it afforded a cost saving.

For the average mercantile vessel, weight and space as a rule did not pose the same problem as in the Navy. However, in a vessel such as a destroyer escort which was closely approaching the size of a destroyer, would it be unreasonable to assume that the space occupied by gearing did not become critical nor did the weight? In the paper presented by Monk, Thomas and Atkinson, a comparison of standard and special destroyer-escort gear arrangements was given in Table 4, part of which was reproduced below to show that the torque transmitted per pound weight (lb. ft./lb.) was 2·5 with a standard gear arrangement at 100 per cent power while with hardened and ground gearing this figure was 4·3 at 100 per cent power.

To revert to cost comparisons, it was not possible to agree with Mr. Nicholson's method of computation. A realistic appraisal of the cost differential between hardened/ground gears and hobbed/shaved gears could only be made by comparing two naval type gears of the same capacity as installed on the *St. Laurent* Class vessel.

With the manufacture of hobbed and shaved gears, the machining of the forgings prior to hobbing and shaving consists of a relatively simple process of "turning". With regard to the gearwheels, the all-welded design simplified assembly and made for cheapness, but nevertheless ensured a very rugged and symmetrical assembly. After this, the hobbing proceeded followed by the post-hobbing process of shaving. The heat treatment of the pinions between the hobbing and shaving process would account for a period of 12 hours for each pinion. However, as the gearwheels in any gearbox were usually the most costly single item to manufacture, the cost comparison would be confined to that involved in the manufacture of the gearwheels.

One large and well known gear manufacturer on this Continent claimed that a 200in. diameter final gearwheel could be completely hobbed in 10 days or 240 machine hours. Shaving of this gear took about 72 hours. In Mr. Nicholson's paper, Table IV, the grinding time on a 67in. P.C.D. secondary reduction gearwheel was 400 hours. No mention was made of the time spent in hobbing the rim prior to the carburizing process. As the hobbing and shaving time for different diameter wheels of equal face width could be connected for estimating purposes, in direct proportion, it would only take

TABLE VII (Part of Table 4 in paper by Monk, Thomas and Atkinson.)

	Standard Gear Arrangement			Special Gear Arrangement	
Test condition per cent full load torque	100 per cent	322 per cent	450 per cent	100 per cent	250 per cent
Total transmitted torque (lb.-ft.)	78,780	253,700	354,560	78,780	196,950
Total transmitted torque/lb .wt. lb. ft./lb.	2·5	8·1	11·4	4·3	10·7
Operational efficiency per cent	95·4	98·2	98·7	95·4	97.5
Maximum compressive stress at pitch line	42,500	70,400	90,200	66,475	105,700
Maximum beam strength (Lewis)	5,800	18,000	29,800	12,000	30,700
Maximum bearing loading (lb./sq. in. of projected area).	150	485	675	210	527

80 hours to hob a 67in. P.C.D. wheel and 24 hours to shave it thereby making for a total hobbing/shaving machine time of 104 hours.

In contrast to this, the hobbing and grinding of a 67in. P.C.D. secondary reduction gearwheel would take about 440 hours and this allowed for a reduction in machine time of 40 hours on the hobbing operation owing to the single helix to be hobbed instead of the double helices in the hobbed and shaved gear.

Over and above this, the heat treatment of the rim did in fact present a costly manufacturing process both in time and money. The hobbed and shaved gearwheel rim required no heat treatment, but was shot peened. The stark reality of the costly heat treatment process could best be appreciated by examining the sequence in a tabular form (Table VIII).

TABLE VIII

Process	Temperature	Heating Time	Holding Time	Cooling Time
1) Pre-quenching	1,480 deg. F.	not stated	4 hours	not stated
2) Carburizing	1,650 deg. F.	,, ,,	18 ,,	,, ,,
3) Diffusing	1,650 deg. F.	,, ,,	18 ,,	,, ,,
4) Annealing	1,200 deg. F.	,, ,,	6 ,,	,, ,,
5) Hardening (first)	1,200 deg. F.	,, ,,	6 ,,	,, ,,
6) Hardening (second)	1,480 deg. F.	,, ,,	4 ,,	,, ,,
7) Quenching	—	,, ,,	0·33 ,,	,, ,,
8) Tempering	250 deg. F.	,, ,,	10 ,,	,, ,,
9) Deep Freezing (minus	95 deg. F.	,, ,,	2 ,,	,, ,,
10) Re-tempering	250 deg. F.	,, ,,	10 ,,	,, ,,

In the above tabulation, the actual holding time at specific temperatures amounted to 78·33 hours and if the heating and cooling times were added to this, the production time for heat treatment of a 67in. P.C.D. gearwheel rim could readily amount to 250 hours which was a conservative estimate. Time was also necessary for correction of the "out-of-roundness" and for flattening, all of which had been described by Mr. Nicholson. From the foregoing, it appeared that with a hardened and ground gear, the production time would be 690 hours as against 104 for a hobbed and shaved gear.

If financial considerations were to be appreciated concomitant with the need for gearing capable of operating at abnormally high K factors, recent advances in the age-old process of nitriding had indicated that hobbed and shaved gears of intermediate sizes could be nitrided economically while distortion in the gear tooth profile could be contained within tolerable limits. This eliminated the need for any corrective profile grinding.

Only recently, it was stated that a large power, lightweight planetary-type ship propulsion gear using a hobbed, shaved and nitrided double helical sun pinion and planets had satisfactorily undergone testing at the U.S. Naval Testing Laboratory and on board ship. The K factor used was 781.

It was not appreciated why the gearing was tested with three-point support. Could it be that the design of the gearbox was intended to provide a rigidity sufficient to withstand the high order of torque loading in the gearing which tended to distort it. Surely, a considerable saving in weight could have been effected if the scantlings of the box had been reduced and five-point chocking adopted in the first instance This had to be done subsequently.

With regard to the chocking of gear-boxes in mercantile vessels, the design objective had always been to provide a very rigid foundation and to securely chock and bolt the box around the perimeter.

With regard to Mr. Nicholson's comments on the application of hardened and ground gears for tankers, the point which he failed to recognize was the reduction ratio to be provided in the gearing. This was usually about 44 : 1. As a consequence, the final wheel diameter was in the region of 14ft. and therefore could neither be hardened nor ground with existing equipment. Mr. Nicholson's remarks about the lack of condenser space was not understood. As already mentioned, a reduction ratio of 44 : 1 was necessary on large tankers using, say, a 22,500 s.h.p. propulsion unit driving a single screw 18ft. in diameter at about 110 r.p.m. The positioning of a condenser below the turbines therefore presented no problem.

MR. J. D'OTTAVIO (Member) said that he did not wish to enter the battle between the shavers and the grinders, however, he would like to be permitted to divert from the highly intellectual brainwork involved in the hardened and ground gearing—in the design and manufacture of the gears and so on—to the toil and sweat involved in chocking. He thought he knew a bit more about that.

The author seemed to be very insistent in having a three-point support. If he understood that correctly, it meant supported at three points only and therefore there was no contact whatsoever anywhere else, outside these points, between the ship and the gearing. He would like to be assured, that the gearing in the ship which had been operating for a year, with standard turbine oil, was chocked-up in this way. If this were the case there was no chocking at all and that was extremely interesting.

MR. E. N. KING, M.Sc.(Durham) (Member) said that the question of noise had been raised several times. He presumed there were two facets to that question. One was the engine room noise and the nuisance value to the personnel. He would be interested to know more about the military aspect. In 1951, at the time this paper began to refer, the auxiliary machinery was considered from a noise aspect. It was an important factor.

Was it the case now that the noise aspect was just as important from the military aspect, or had the modern detection methods ruled that out of consideration.

Author's Reply

MR. D. K. NICHOLSON replying to the discussion said he was most appreciative of the interest shown in the paper and wished to thank all contributors to the valuable discussions which had ensued. In attempting to deal with all points in the order in which they had been raised, he proposed to make his replies embrace all similar or related points which might have been raised by later contributors.

The economy and justification of full power shore testing on all gear sets of the *St. Laurent* Class had been questioned or supported by a number of contributors. The value of shore testing in determining design deficiencies and manufacturing defects or inadequacies in main gearing before reaching the ship installation, was generally appreciated, but as Mr. Archer and Mr. Gooch rightly suggested economic justification would be extremely difficult in the case of small numbers of merchant ship machinery and indeed for small numbers of naval machinery. For advanced design naval machinery full scale protoype trials were a necessity. Therefore once a test rig was set up for the initial shipset of machinery in the manufacturer's works, it would be hard to justify subjecting succeeding units, erected on the same test stand, to a conventional spin test only. In Canada the additional cost of back-to-back testing was relatively small and was considered to be well justified. It was of course a different consideration for a naval programme, which Commander Weaving had in mind, where units of the same design might be produced by a number of manufacturers. The requirement for the full power shore testing of individual gear sets after prototype trials should perhaps be determined on the basis of the particular manufacturer's record and ability.

Mr. Archer had correctly commented that the quoted difference in root stresses permitted by case hardened and through hardened gears assumed similiar core strength properties.

Mr. Archer might be interested to compare his calculated slide/roll ratios with the figures given in Table IX for which a shaft speed of 227 r.p.m. had been taken and allowance made for the increased operational pressure angle due to the spread of centre distances. No adjustment had been made for profile modification.

It was interesting to note that the scuffing experienced in the original design secondary reduction commenced at the pinion tips rather than in the approach flank where the slide/roll ratio was a maximum. As observed by Mr. Archer in his paper,* scuffing was more likely to occur at the pinion tips where the sliding and rolling velocities were opposed to each other.

In answer to Mr. Archer's comments on the improved ductility shown for the original Swiss material in Table II, it must be acknowledged that only the values in the third and fourth columns were minimum specification properties. The values in the second column were now known to be actual test properties. There should in fact, be little difference between the actual test properties of DEW 3610 and the original Swiss material.

Mr. Archer asked about the comparative noise levels between the two gear tooth designs. Although the revised gear design was considered to be slightly quieter, no significant difference in noise levels had been measured. Overall noise levels in which Mr. Zrodowski and Mr. Sledge were also interested, measured in the immediate vicinity of the main gearing, had reached 105 db. while in the frequency octave bands the maximum levels reached 100 db. Those noise levels were not considered to be abnormal for ships of the *St. Laurent* type and were not the subject of any special noise investigation.

The diffusion type carburizing cycle referred to by Mr. Archer had been found to produce a more uniform carbon content gradient over the depth of case and thus assisted in obtaining an effective case with a carbide-free surface. It was claimed that deep freezing reduced the danger of subsequent cracking during grinding and during service by ensuring the controlled transformation of austenite. The tendency to cracking was further reduced by stress relieving after deep freezing. The occurrence of brittleness, grinding cracks and excessive hardnesses, as reported by Mr. Sykes and Mr. Page, had not been experienced in Canada. Could it be that the processing to which they referred involved too high

TABLE IX.—SLIDE/ROLL RATIOS

	Point of contact		Slide/Roll Ratios			Reduction %
			Pinion	Gearwheel	Mean	
Original Design	Primary	Approach	1·131	·531	·831	—
		Recess	·391	·642	·517	—
	Secondary	Approach	2·515	·716	1·615	—
		Recess	·563	1·286	·925	—
Revised Design	Primary	Approach	·359	·264	·312	62·5
		Recess	·247	·328	·288	44·3
	Secondary	Approach	·669	·401	·535	66·9
		Recess	·324	·480	·402	56·5

* "Some Teething Troubles in Post War Reduction Gearing", 1956. Trans.I.Mar.E., Vol. 68.

quenching temperatures? The purpose of the $3\frac{1}{2}$ minute period taken between the furnace and the quenching tank in Canada was partially to achieve the desired quenching temperature.

Experience in Canada indicated that tooth surface hardnesses were very slightly higher at the end which entered the quenching tank. This was not considered to be important other than the possible effect on distortion resulting from non-uniform heat treatment. He agreed with Mr. Archer that the tooth core strengths would be slightly lower than that indicated by the test pieces. The R.C.N. had no experience indicating that the gear rim straightening operations which were carried out after hardening were detrimental to the case.

In answer to Mr. Archer and Mr. Smith, secondary reduction gearwheel grinding times given in Table IV were for the 20 deg. pressure angle. No secondary reduction gear trains to the original design had been made in Canada. An analysis of grinding experience at the Crown-owned gear plant indicated that grinding times were influenced more by the outside diameter and face width of the gearwheel than by the number of teeth, or diametral pitch.

Mr. Archer asked about the method of equalizing the torque in the two branches of the locked train. The main primary pinion was torqued against a locked main gearwheel, through the outboard quillshaft, to achieve ahead flank contact throughout. The inboard secondary pinion was fitted to its quillshaft to achieve ahead flank contact in the inboard branch. Non-clearance bearings were used during lock-up. In view of the appreciable range of journal attitudes over the entire power range it was not possible to equalize the torque in the two quillshafts for all powers. At extremely low powers (less than 3 per cent full power) the torque differential under operational conditions could be as high as 25 per cent whereas at full power the differential would be entirely negligible.

The gearcase deflexions which had been reported had prompted several contributors to question the adequacy of both the gearcase stiffness and the method of support. In the absence of any deleterious effects in the gearing or gearcases, which were fully fabricated, in any ship, he was more inclined to give thought to ascertaining to what extent gearcase stiffness was necessary rather than to determining ways and means of reducing what appeared to be a tolerable amount of deflexion. The use of a transverse chocking arrangement on either side of the main gearwheel, as suggested by Mr. Morton, would be beneficial in restricting axial twist, although it would appear to raise a problem of access for chocking (which would affect the accuracy of chocking). He thought that there was considerable merit in locating the secondary train forward and retaining the three-point support, with the forward two chocking areas under the secondary reduction pinion bearing walls. Although, as Mr. Kemper, Mr. Campbell and Mr. Laskey pointed out, the *St. Laurent* Class main gearcase was designed and proven for a three-point support, it could not be clearly demonstrated that the gearcase was too stiff when chocked for the five-point support. It should be noted that the additional chocking, which produced the five-point support, was now used, not because it was necessary, but because there was no apparent advantage in having these chocks omitted. Mr. Kemper would appreciate that the initial provision in the design for supporting the main gearing on three chocking areas (loosely described as "points") was advocated by many authorities in the early post World War II years when troubles with double reduction gearing were still fairly extensive. The aim was to isolate the gearcase from the effects of hull distortion. Experience had shown that this method of chocking was not necessary in *St. Laurent* Class to ensure satisfactory operation. The three-point support was in fact used only as a preliminary step prior to the final chocking.

The asymmetrical spacing of a pinion helix between its bearings was not considered for the *St. Laurent* Class gearing design. As stated by Mr. Archer this would help to minimize the required helix angle correction, although at the cost of increasing the gearcase length. The use of a centre drive in secondary reduction pinions as described by Mr. Smith would appear to be an attractive method of eliminating the need for helix angle correction, if indeed it could be established that there was a need for it at all. He wished to correct the impression reflected in the comments from Mr. Gooch and Mr. Salthouse that it was the intention of the R.C.N. to apply helix angle correction after observing what amount if any, was required from the test bed trials. While this procedure would of course be followed at any time it were considered necessary, the need for its adoption would indicate an inadequacy in the current manufacturing or alignment requirements. Therefore, except in cases where deviations from specified requirements got by undetected, the need for regrinding a pinion should occur once only, that was at the time the design requirements affecting helix angle correction and so forth were evaluated on the test bed.

Mr. Archer asked whether the use of adjustable bearing housings would be considered for future requirements. While they would certainly be considered, he was not convinced that they were to be preferred to the practice of accurately jig boring and scraping the gearcase housings. Where the effect of gearcase stiffness on internal gearing alignment and the determination of operational alignment requirements became important design considerations, as indeed they should, there was a good argument for using adjustable bearing housings in a prototype gearbox. This procedure would also obviate the need for helix angle correction as mentioned by Mr. Salthouse.

In answer to Mr. Archer's question on the main gearwheel lifting, the forward journal was not moving with the frequency of the shaft revolutions in a manner to cause the gearcase knocking which had been reported. The knocking was at the shaft revolution frequency and was predominant at the shaft speed producing the first order torsional critical. The lifting of the main gearwheel was found to unload the inboard secondary pinion sufficiently to cause tooth separation when the torque fluctuations at the torsional critical were compounded together with the effect of a low power torque differential between the quillshafts. In reply to Mr. Salthouse, the main gearwheel lifting had been measured directly from top and bottom bearing clearances.

The use of a flexible coupling between the main gearing and the thrust block, as suggested by Mr. Archer, would be beneficial in aft end machinery arrangements providing the coupling were really flexible when transmitting high torques. However, for midship machinery arrangements, he supported Mr. Page's view that a long unsupported length of shafting, between the gearing and the first shaft bearing, should be quite satisfactory, providing the shafting/gearing alignment was correct. The incorporation of a flexible coupling in the main shaft line would, of course, rule out the use of an integral thrust block, would require a separate thrust bearing for the single helical main gearwheel and would generally impose a length penalty. In reply to Mr. Keenan, who also advocated the use of a main shaft flexible coupling, to help protect main machinery from underwater shock, he confirmed that the provision of adequate shock resistance, for main machinery, was, of course, a major consideration in modern warship design. In some ships, the use of a main shaft flexible coupling, abaft the main gearing, was used as part of a shock isolating arrangement. It should perhaps be noted that double helical gears were less able to absorb axial shock loads, transmitted along the shafting, than single helical gears.

With regard to the information, requested by Mr. Archer, on the extent of full power operation with *St. Laurent* Class, it was not usual for ships of this type to exceed 10 hours of full power steaming per year. On this basis, the early ships of the Class would be approaching a total of 70 hours of full power operation, inclusive of shore testing and contractor's sea trials. The values of C_1 and C_2 would be of the order of 4,200 and 90 respectively.

He did not agree with Mr. Archer's view that the higher load-carrying capacity, provided by case-hardened gearing, would best be utilized in mercantile practice, by reducing the face width of the conventional soft wheel, and not the diameter. In his opinion, the diameter of the main wheel was the largest single factor which influenced the size and cost of a set of gearing. It should therefore be kept to the smallest possible diameter, as determined by such requirements as the configuration of shaft centres, speed reduction and the provision of adequate condenser space. Having first determined the main gearwheel diameter, the face width should be determined in consideration of pinion and wheel L/D ratios, which provided a suitable overlap ratio and adequate proportions for stiffness and stability. Where the size of the main gearwheel was not primarily subject to the physical limitations of the machinery installation, it should be reduced to utilize the maximum possible load-carrying capacity —which was where case-hardened gears came into their own. In this regard it was interesting to note that Mr. Zrodowski had ruled out of consideration case-hardened gears over 70in. in diameter. While he would readily agree with Mr. Zrodowski if he were referring only to carburized and hardened gearwheels, he could find no grounds for excluding induction hardened gearing, which could be manufactured up to the limitation of available gear grinding machines (142in. diameter). In addition, nitriding had already been successfully applied to gearwheels in excess of 70in. diameter.

The comments made by Mr. Sykes and Commander Weaving on the recent achievements of equal load-carrying capacity, with induction hardened and nitrided gears reaching that for carburized and hardened gears was most encouraging and would prompt him to raise the K factors in line 3 of Table V to 300.

In answer to Mr. Sledge and Mr. Longhurst, the application of nitriding to medium sized gearwheels had been proven only in recent years. It appeared to be eminently suitable for primary reduction gearwheels in naval and mercantile gearing, but because of its shallow depth of case, it might be necessary to confine its use to fine pitch teeth of say, not less than 6 D.P. The prospect of gearwheel induction hardening being perfected to the stage of producing negligible distortion, as mentioned by Mr. Sykes, was indeed a very attractive one, to manufacturers of hobbed and shaved gears. It was understood that this stage had already been achieved in Switzerland, particularly with nitrided marine gearwheels.

Mr. Sykes and Mr. Page suggested that interchangeable pinions and gears for marine gearing were desirable. In the absence of adequate reliability or design margins of safety, he was in full agreement with them. Experience in Canada indicated that it was neither economical nor necessary to make individual, carburized and hardened mating gear components interchangeable. Interchangeability was confined to mating pairs of gears or gear trains and bearings. He very much questioned whether the manufacturing implications of interchangeable gearing components could be accommodated in applications catering for high tooth loadings and quiet operation.

The comments by Messrs. Sykes, Kemper, Gooch and Waterworth, on B.S. 1807 Class A1 accuracy, had resulted from his failure to fully qualify his remarks, regarding the application of this accuracy standard to case-hardened gearing. The importance of gear accuracy, with regard to load-carrying capacity and noise levels, was of course quite undisputed. However, bearing in mind that B.S. 1807 was specifically applicable to hobbed gearing, it would seem that appropriate allowances should be made in applying it to gears ground by the Maag process. Tooth-to-tooth spacing errors, as measured on ground teeth, were inclusive of profile variations which did not arise in hobbed gears. In the case of *St. Laurent* Class gearing, the permitted tooth spacing error of ·00035in., in the secondary reduction gearwheels, had been interpreted as the maximum variation, measured at three positions over the tooth profile and at three positions across the face width. This related to the apparent tooth spacing error. The actual tooth spacing error to which B.S. 1807 referred and which related to cumulative pitch errors would readily fall within the Class A1 requirement. B.S. 1807 could certainly be interpreted in a manner which would produce an unnecessarily high standard of accuracy in hardened and ground gearing. There was surely a need for an appropriate standard of accuracy for hardened and ground gearing, based on the standard methods of measurement which were employed and classified with respect to load-carrying capacity and noise level requirements.

Mr. Sykes might be interested to know that the effects of differential temperatures in pinions and wheels had been the subject of an excellent paper* by W. P. Welch and J. F. Boron, of Westinghouse Corporation.

Mr. Sykes and Mr. Smith both referred to satisfactory experience which had been obtained with thrust cones. This was a most attractive way of permitting higher helix angles in single helical gearing and was an equally attractive method of overcoming the slewing effect.

He was afraid that Commander Platt was not referring to the information in his paper, when he concluded that the time occupied in grinding was the principal factor in controlling the cost of hardened and ground gearing. As he had already stated, the size of the secondary reduction gearwheel would appear to be the principal factor; not the manufacturing implications of the pinions and gearwheels. It was this very consideration which, to his mind, explained why Braddyll[5] could find no significant cost difference, by re-designing a double-reduction gearcase to accommodate single helical, hardened and ground primary reduction gearing in place of double helical hobbed and shaved gearing. Induction-hardened primary gearwheels, with carburized and hardened pinions, were chosen, but no change was made in the secondary reduction train. It was estimated that the manufacturing costs for the pinions and gearwheels seldom exceeded 25 per cent of the total cost of the gearing. Any reduction in grinding times, resulting from the use of induction hardening or nitriding, would affect only this small proportion of the overall cost.

It was agreed that the percentage net increase in gear rim diameters during carburizing and hardening could be expected to increase with size. Commander Platt would presumably have noticed that he had misquoted the gearwheel dimensional and percentage increases from the paper, by enlarging them ten times and would therefore be gratified to learn that no rejections from distortion were anticipated nor had they been experienced over the last six years. The author could see no obvious reason why soft gearing should have a greater susceptibility to failure under malalignment, on account of them being double helical. He would suggest that the percentage increase in tooth loading or load concentrations, due to the effect of malalignment, would be greater as the overall face width increased. In this respect nested gears became particularly susceptible to failure, due to the effect of shaft alignment or hull distortion on the internal alignment. Commander Platt's call for a concerted effort, on the part of the shipbuilders and engine designers, to minimize the influence exerted by the main condenser on the main gearing design, was strongly supported.

In answer to Mr. Kemper, the selection of a lower K factor in the primary reduction, than for the secondary reduction, was influenced mainly by the desired spacing of secondary reduction pinions on the main gearwheel and the resulting primary gearwheel proportions. For a two turbine dual drive naval gearing arrangement, of the type referred to by Mr. Kemper, a primary reduction K factor, appreciably higher than either 310 or 320, would appear to be warranted.

He would like to thank Mr. Smith for confirming the tooth loading (1358K) on the secondary reduction gear tests at AVGRA[1].

It was apparent that Mr. Gooch and Commander Goodwin had somewhat opposing views on the application of naval gear loadings in mercantile gearing. With no more than 70 hours at full power operation, in which ships of the *St. Laurent* Class would have barely reached halfway in the completion of 10^7 secondary pinion cycles, the experience with this Class was not therefore sufficient to justify 412K

* "Thermal Instability in High Speed Gearing", A.S.M.E. 1959.

tooth loads for mercantile use. However, an analysis of the Admiralty gearing load-carrying tests[1], could leave little doubt that present naval gear loading could be endured on a continuous basis. In answer to the comments made by Mr. Gooch and Mr. Sledge on the secondary train design loading of 412K, it should be remembered that this figure was set about twelve years ago. Current naval case-hardened gearing design loadings would be in the 500-600K range and in some cases even higher. It was these values that Mr. Gooch should consider, in assessing the recommended mercantile loading, given in Table V. Rather than the loading of 250K for induction hardened gearing being too high, there were already sound reasons for raising it to 300K, as for the carburized and hardened gearing. In stoutly challenging the ecenomy of case-hardened and ground gearing. Mr. Gooch was following a path which, although well-trod, was nevertheless in danger of petering out. Manufacturing facilities for case-hardened and ground marine gearing were, of course, already available in many parts of the world. Many sets of case-hardened and ground gearing were in use, in both naval and merchant vessels, and the trend was increasing.

Mr. Gooch's final caution, on the use of hardened and ground gearing, was on the basis of avoiding the risk of a catastrophic tooth failure; whereas he inferred that the risk would be confined to pitting and scuffing where soft gears were used. The author did not find this argument very plausible since, to his knowledge, the very type of failure against which the shipowner was to be protected, had in fact been experienced, predominantly in the type of gearing he would be recommended. It might be added that a good proportion of tooth breakages, in soft gearing, originated from pitting which had either gone undetected or which had been permitted to remain in the hope that it would clear up.

His statement regarding the influence of case depth on shock resistance was made in reference to Dr. Ing H. Glaubitz's paper*. Mr. Page would notice that the impact test specimens described in this paper were designed in consideration of gear tooth loadings. With regard to Mr. Page's question, on how standard concentric bearings were used in *St. Laurent* Class, the gearcase housings were accurately jig-bored and then scraped to receive standard bearings and to achieve parallelism between bearing axes. The bearing bores were not scraped.

In reply to Mr. Salthouse, the case depth/module ratio, at which the bending fatigue strength reached a maximum, had been found to have different values by different researchers. All values appeared to fall in the range ·07 to ·23. Reference should be made again to Dr. Ing H. Glaubitz*.

In reply to Mr. Fowle, the ship which had been operated for over a year on standard turbine oil, completed its shore testing and its initial 2½ years of service with E.P. oil. It would be noted that the second gear unit tested at Pametrada, which was run on standard turbine only, scuffed at a much lower power than did the first unit, which was first run with E.P. oil. Although this would support Mr. Fowle's view, that prior running with an E.P. oil would reduce the sensitivity to scuffing on standard oil, it was still questioned whether a sufficient change to the tooth surfaces (as evidenced by polishing) did in fact take place on case-hardened gearing. The value of an E.P. or anti-wear oil, being used initially in newly manufactured soft or through hardened gearing, was clearly acknowledged.

The effects of the gearcase deflexion and chocking arrangement on the gear tooth loading and load-carrying capacity had not been determined and so a satisfactory answer could not be given to Mr. Waterworth on this aspect. The fact that the gear teeth showed no indication of load concentration, would indicate that there was sufficient flexibility provided in the main gearing components and bearings to maintain correct internal alignment, between mating components.

The R.C.N. had no experience with the combination of carburized and hardened pinions, with through hardened gearwheels, but the following points were pertinent to Mr. Waterworth's comments. The tooth loading, obtainable with a through hardened gearwheel, was generally higher when it was mated with a case-hardened pinion, than when it was mated with a through hardened pinion. The question to be examined was how much of this extra load-carrying capacity could be safely utilized, without risking damage to the gearwheel teeth. He knew of no ship in service, either naval or mercantile, which was running with such a combination of materials, where the tooth loadings were higher than those which would have been attainable had all components been through hardened. Since the criterion for loading would pass from the case-hardened pinions to the through-hardened gearwheels, he was forced to conclude that only a limited increase in loading could be permitted, if a failure was to be avoided in the gearwheel teeth.

Also in answer to Mr. Waterworth, full depth meshing was achieved, only over the extent of the uncorrected tooth profiles, which presumably was a contradiction of terms. The tip and root relief on the original design pinions was ·0009in. on the primaries and ·0015in. on the secondaries. These amounts were reduced to ·0006in. and ·0008in. respectively in the revised design.

In reply to Commander Goodwin on balancing, all pinions were dynamically balanced to within less than 1 oz./in. In *St. Laurent* Class, the gearwheels were statically balanced to within 10 oz./in. in the primaries and 50 oz./in. the secondaries. It was presumed that Commander Goodwin also referred to the turbine gearing flexible coupling. These couplings, with the connecting torque tubes, were dynamically balanced in a special supporting rig, with the tooth driving faces in contact. The assembly was not torqued.

Mr. Hill had expressed interest in the assembly of the gearwheels. An interference of about ·015in. was applied between the gear rims and the discs and ·007in. between the discs and the shafts. The rims were heated to about 200 deg. F. No difficulties had been experienced with the assembly of the secondary reduction gearwheels. The procedure was to shrink one disc on the shaft and one in the gear rim. The gear rim, with the disc temporarily bolted, was then re-heated and shrunk on to the shaft and other disc. Both gearwheel assemblies were secured by fitted bolts, but tapered bolts were also used in the secondary gearwheel. All bolts were torqued by a prescribed amount and were secured by centre punching and a nut locking compound. Tack-welding was not used to lock the bolts.

He was interested in Mr. Jobling's suggested heat treatment procedure, but was unable to offer any authoritative comment at this time. It was suggested, however, that a medium carbon alloy steel was not suitable for carburizing, that it would not produce an effective case and would be more susceptible to cracking. It was also questioned whether shot-peening would be effective, in transforming retained austenite below a depth of ·001 to ·002in.

In reply to Mr. Keenan, he regretted that it was necessary to limit the scope of the paper and to omit details of the bearing design and of the method of gear measurement employed. Briefly, the pinions were given comparative profile checks, on equipment used in conjunction with the grinding machines. All pinions and gearwheels were checked with the Maag T.M.E. and T.M.A. instruments for base pitch, profile errors and circular pitch. The meshing frame was used for matching helix angles and measuring helix angle correction. No degreasing was employed, since the Maag system of gear grinding was a dry process. The operations performed after gearwheel assembly, other than the gear grinding, were the grinding of the gear rim faces and outside diameter. The prime purpose of the dished gear discs was to provide stiffness. It would, however, be noted that Mr. Zrodowski considered the design to be subject to end deflexion. If this was so, it had caused no deleterious effects and might well be a desirable factor, in maintaining internal alignment between mating gears. No consideration had been given to the use of resilient gearwheel cores, to damp out vibrations, but a number of the noise attenuating features,

* "The Effective Case Depth of Surface Hardened Gear Teeth" by Dr. Ing H. Glaubitz, Verein Deutscher Ingenieure, Zeitschrift 1958, Vol. 100(8), pp. 216-226.

on the lines described by Mr. Keenan, had in fact been found to be effective and beneficial. These features had been successfully applied to gearing in later ships to the *St. Laurent* Class. He did not think that Mr. Keenan would find too much support for his "small undulation theory" being the mainstay of large soft gearing. While the ability of a tooth surface to maintain an oil film was a desirable property, it would be of little value, if obtained at the expense of increasing the load concentrations on the peaks of the surface undulations and irregularities.

Dr. Merritt was quite correct in stating that the difficulties experienced by the R.C.N. with the 23 deg. pressure angle were not a fundamental limitation. With regard to the 3½ minute period between the furnace and quenching, attention was drawn to the inclusive two minute holding period on the quenching fixture. This holding period permitted the gear rim temperature to drop to the level required for quenching and at the same time allowed the rim to firmly grip the fixture prior to quenching. The danger of surface decarburization was recognized and it was agreed that shot peening, or better still vapour blasting, would be beneficial for restoring the compressive stress. The quenching of gear rim and disc assemblies could not be attempted, in view of the variations which occurred outside the predicted growth. The shrink fit between the disc and gear rim had to be held to within close limits to prevent gear rim dishing.

As Dr. Merritt had observed, the carburized layer was removed from all gear rim surfaces except the gear tooth roots and profiles and the inside diameter which bears on the quenching fixture. This was done to permit the easy removal, after hardening, of material which, by virtue of the increased mass it provided, was beneficial in restricting distortion during carburizing and hardening. In addition, the existence of an unnecessary hardened case would cause an unnecessary stress raiser. Hardened surfaces were retained, on the inner diameter of the gear rim, to assist in withstanding the shrink loading on the quenching fixture.

Commander Weaving's report of the gearing experience, with similar ships in the Royal Navy, prompted him to remark that the range between satisfactory and unsatisfactory service was often extremely small and did not necessarily reflect margins of safety which were synonymous with reliability. The selection of a smaller design case depth would appear to offer an improvement in load-carrying capacity and this might permit an even greater reduction in the applied case, on the basis that smaller gear rim distortions should result from the application of smaller depths of case. With regard to shafting/gearing alignment, it was the present practice, in the R.C.N., to require main gearing units to be installed in a manner which would ensure uniform static loading between the two main gearwheel bearings, with the ship uniformly ballasted to the half oil condition. It was considered that this condition was best achieved by aligning the gearwheel coupling to the forward coupling of the coupled-up line shafting, either face-to-face or with an appropriate *bottom* breakage. The flexibility of the hull and its affect on shafting and gearing alignment was recognized. It was therefore considered important that the initial alignment was achieved with the ship in a mean condition and not an extreme condition, as was often the case, where alignment work was done with the ship on the slips or in drydock. The effect of gearcase thermal expansion on the shafting/gearing alignment was considered to be negligible in the *St. Laurent* Class, particularly since the gearcase in those ships had a dry sump. However, alignment records for shafting/gearing installations were completed, with readings of shaft breakages, bearing clearances, shaft journal attitudes, tank readings and all relevant temperatures inside and outside the ship. Calibrated jacks for checking bearing reactions were used, but not as a routine measure.

Mr. Cacciola, who had been associated with considerable test and investigation work, at the U.S. Naval Boiler and Turbine Laboratory in Philadelphia, on soft, through hardened and case-hardened gearing, deftly reflected the realistic, philosophical outlook on case-hardened and ground gearing, from the viewpoint of a country having substantial manufacturing facilities for conventional type gearing. The United States Navy's experimental gear development programme, to which he referred, should greatly assist in the assessment of the reliability/cost relationship for various types of gearing and gear production techniques.

In answer to Mr. Clark, case-hardened and ground gears were usually designed for single helices. One of the principal reasons was that the gap between double helices would be larger than for hobbed double helical gears and would thus sacrifice some of the potential saving in gearcase length. Michell thrust bearings absorbed the axial load components in the turbine pinions, while the net thrust, resulting from the opposing primary gearwheel and secondary pinion assemblies, was absorbed by thrust faces on the primary gearwheel after bearings. The axial thrust from the main gearwheel acted in opposition to the propeller thrust, thus slightly relieving the load on the main thrust bearing. Specification properties of the Admiralty turbine oils OM88 and OM100 were given in Table 12(i) of Mr. Newman's paper[1]. As Mr. Clarke would know, these were plain mineral oils. The E.P. oils, used during the testing at Pametrada, were special load-carrying oils of a proprietary brand. It was understood that the viscosity characteristics were similar to the standard turbine oils used and that the load-carrying capacities would have met or exceeded that required by the Admiralty oil specification OEP 90, which was of course not in existence at that time.

Mr. Zrodowski had questioned the limit of 250K for through hardened gears in view of his own experience with loadings up to 300K. This was indeed a notable achievement, but it should be appreciated that the limit of 250K, for through hardened gears, was set in consideration of the load-carrying capacity, permitted by surface hardnesses of the order of 340 B.H.N. for gearwheels and 380 B.H.N. for pinions, which were regarded as the practical limit of machineability. It would have been interesting to know Mr. Zrodowski's view on the practicability and economics of using 350 B.H.N. on the gearwheels and 400 B.H.N. on the pinions. Commander Brayley and Mr. Berg in their paper* referred to gears having K factors of more than 300 and with surface hardnesses higher than those quoted by Mr. Zrodowski. Apart from noting that these gears were reported in the paper to have suffered initially from repeated tooth breakages, it was suggested that Mr. Zrodowski and other U.S. gearing designers were pushing the application of hobbed and shaved through hardened gearing, far beyond its practical and economic limitations.

Experience in the R.C.N. indicated nothing to substantiate the view that bolted or shrunk-on gear rims were objectional design features. The design and assembly practice described in his reply to Mr. Hill was considered to be quite satisfactory. On the other hand, it should be taken into account that gear rim welding could be very troublesome, particularly when using the type of high strength alloy steels, which Mr. Zrodowski would require, to cater for his high tooth loadings.

The cost reduction of 10 per cent, which Mr. Zrodowski estimated would be obtained by utilizing the additional load-carrying capacity, provided by through hardened gears in accordance with line 2 of Table V, was most interesting, particularly since it did not involve a reduction in gearwheel diameter. Unfortunately Mr. Zrodowski later inferred that where the load-carrying capacity was increased, by means which precluded hobbing and shaving, such as by case-hardening then the cost differential went the opposite way. This was a view which, of course, he could not support.

Replying to Mr. Sledge, standard turbine oil had been specified for use in service, so it was considered necessary to prove the capability of the design in this regard during the Pametrada prototype trials. Tests had been carried out on *St. Laurent* Class gearing, to determine the effect of lubricating oil inlet temperature on gearing efficiency. The specified

* "Design and Service Experience with United States Naval Gears", International Conference on Gearing 1958.

lubricating oil inlet temperature was 120 deg. F., for both standard and E.P. oils. However, it had been established that, on the basis of maintaining the oil discharge temperature from the gearing at about 140 deg. F., an increased efficiency of approximately 4 to 5 per cent was obtained at low powers, which was of considerable value to naval ships.

With regard to Mr. Longhurst's interest in the experience obtained by the Royal Navy, with the through hardened hobbed and shaved double helical gearing, fitted in similar ships to the *St. Laurent* Class, it would be noted that Commander Weaving had given this information, in his contribution to this discussion.

The application of epicyclic gearing in the main propulsion system was continually under review. Perhaps the only known high power main propulsion application was the installation of 50,000 s.h.p. epicyclic units in U.S.S. *Timmerman,* which was reported to have seen very little service. Epicyclic gearing was quite suitable for primary reductions, but the difficulties, associated with the accommodation of co-axial input and output shafts, made it unsuitable for providing the overall speed reduction in steam turbine installations.

Most of the points made by Mr. Campbell and Mr. Laskey had in fact been answered, either in the preceding discussion or in the paper itself. However, since these contributors had gone to great pains to suggest that the *St. Laurent* Class gearing was of a questionable design, requiring economically prohibitive methods of manufacture and which had given service with which the R.C.N. had no right to be satisfied, he would like to make a somewhat general reply. The *St. Laurent* Class gearing, which was still considered to be an advanced design, 12 years after it was laid down, was in complete accord with a firm trend in naval gearing, which was being followed by the principal navies of the world, with the notable exception of the U.S. Navy. The high load-carrying capacity obtainable with case-hardened gearing, together with a high margin of safety, both considerations being of immense value in naval requirements, had been indisputably established by the gear test programmes, referred to in his paper. The contributors grossly underestimated the significance of five years trouble-free service in naval ships, which although engaging for only a small proportion of the time at full power, were, in fact, frequently subjected to far more vigorous steaming conditions, while manœuvring in exercises, than were experienced by merchant ships. On the basis of the above mentioned test programmes, the "unit" loading of the *St. Laurent* Class secondary reduction gearing, to which Mr. Campbell and Mr. Laskey referred, could be safely increased to at least double the "unit" loading quoted for the aircraft carrier gearing. There was little justification for the economy of case-hardened marine gearing being rejected on the basis of the R.C.N. experience and particularly since technical advancements, in the broad manufacturing field of case-hardened and ground gearing in Europe and the United Kingdom, had been very considerable. It was suggested that Mr. Campbell and Mr. Laskey would be extremely presumptious to condemn a type of gearing, on the basis of cost, before the tenders were called.

In reply to Mr. D'Ottavio, the gearing in the ship, which had been in service for over a year with standard turbine oil, was chocked on a five-point support, as indeed were all ships of the class, now in commission.

Mr. King rightly presumed that the noise of main gearing and other warship equipment was most carefully considered, from both the military and the habitability aspects.

In reply to Mr. Newman, it was his view that nitrided gears would offer little economic advantage over induction hardened gears, regardless of whether they were hobbed and shaved, hobbed and ground or rack-cut and ground. He would therefore be inclined to apply a similar cost index, assuming of course that it were in fact feasible to nitride secondary reduction gearwheels. The cost index for induction hardened gearing shown in Table V would of course be reduced to about 0·68 if the load-carrying capacity was in fact recognized as the same for carburized and hardened gearing by applying a K factor of 300. The internal gearing alignment was, as Mr. Newman suggested, considerably affected during the extreme conditions of high speed manœuvring to which most naval ships were commonly subjected. It was believed that these conditions could best be withstood if the static gearing/shafting alignment was critically established for a mean condition, thus providing the maximum range for deviation at either extreme. Appreciable changes of journal attitude in main gearwheel bearings, due to changes in relative hull and shaft flexures occurring during ahead and astern manœuvring, had been observed in *St. Laurent* Class ships but without any incidence of failure.