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Epicyclic Gears

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Epicyclic gears in various forms have been used in a wide field of engineering for many years. There are many successful applications in automotive engineering, whilst more recently epicyclic gears of considerable horsepower have been used for straight speed reduction, particularly in aircraft. Until recently, few large power industrial epicyclic gears were built in this country but they have been used extensively elsewhere, notably in Germany.

In marine fields, both here and abroad, epicyclic gears have been used as main propulsion reduction gears for powers up to 1,000 h.p. but their application for auxiliaries has been very limited.

In this paper the authors briefly survey some of the historical background of epicyclic gearing development and discuss the basic design principles of some well tried forms likely to appeal to marine engineers. The importance of accurate manufacture to ensure successful operation is stressed.

The advantages of this type of gearing, including the savings of weight and space, are illustrated for a number of applications and some comparisons made between parallel shaft and epicyclic gears. Particular reference is made to the use of epicyclic gearing with marine auxiliaries.

In conclusion, the authors indicate, from present marine, industrial and aircraft experience, future possibilities and trends in marine applications.

INTRODUCTORY AND HISTORICAL

In mechanical engineering it is not unusual to find that many of the basic ideas which are now common practice are based on fundamental principles which have been known for many centuries. There are also numerous cases where ideas have been patented in the very early stages of mechanical engineering development and have lain unused because of the difficulties of practical application or methods of construction, or because the theory has not been fully understood.

The story of epicyclic gearing is no exception to this historical trend and one finds, for instance, that in 1781, James Watt patented a sun and planet gear which he utilized in his early engines, to convert reciprocating motion into rotary motion without infringing existing patents on the crank mechanism. This application is illustrated in Fig. 1 (Plate 1)

The fascination and potentialities of this form of compact gearing are clearly illustrated by reference to the numbers of patents which are continually being taken out in this field.

Epicyclic gearing was freely used during the course of the 19th century on a wide variety of industrial applications but did not keep pace with the demand for larger powers, because improvements in the technique of cutting external gears outstripped developments in the production of internal gears. Further, the difficulties in ensuring that the planet wheels shared the load were either not appreciated or could not be overcome, because it was not possible to make and measure the components with the required degree of accuracy and none of the co-axial members was allowed to float.

The importance of some of these factors was fully realized by the Lanchesters, who referred to this question in their paper[10] on epicyclic gears. They had also grasped one of the most significant facts of mechanical engineering design and construction when they stated: "The success of any mechanism, however well conceived, depends finally upon its correctness in detail".

Early in this century the advantages to be obtained from a compact and "in-line" gear were quickly appreciated by the pioneers of automobile engineering. By way of example, one has only to mention the Model "T" Ford, the early Lanchester cars and, more recently, the gear pioneered by W. G. Wilson which is widely used today. His paper[17] describing his gear is a classic on this subject. It is also interesting to note that in his introductory remarks he mentioned a three-speed compound epicyclic gear constructed in 1920 for a trawl winch, transmitting 170 h.p. at 170 r.p.m., which was still running satisfactorily in 1932 when the paper was read.

In industrial fields, epicyclic gears for high speeds or large torques have not been extensively used in this country until recent years but great interest has been shown in their potentialities lately, not only in this country but also in the United States of America. On the other hand, this form of gear has had quite extensive use on the Continent, particularly in Germany. This is largely due to the wide study of the problem by W. G. Stoeckicht, who has successfully designed and applied epicyclic gearing, not only in long life industrial engineering but also in the aircraft and marine fields. The 3,300 h.p. 3,200 to 1,170 r.p.m. gears for the Jumo 222, the largest piston type aircraft engine developed in Germany, were designed by him. His 5,000 h.p. 3,770 to 580 r.p.m. marine main propulsion gear, which is described and illustrated in his paper[15], "Some Advantages of Planetary Gears", is a remarkable achievement. The gearcase is 3 feet in diameter, 2ft. 6in. long and the total weight of the gear, including the fabricated steel case, is one ton.

Figs. 2(a) and (b) (Plate 1) show two gears of his design which are running in Germany on turbo alternator sets. They are double helical planetary gears and the sun and planet wheels have been nitrided in both cases. It will be noted that in both applications the governor is almost as large as the gearcase to which it is attached.

The advent of propeller gas turbines with their high speeds of rotation has given a further stimulus to the use of epicyclic gearing for aircraft, and whilst one must treat this experience with reserve because of the vitally different conditions obtaining, there are nevertheless lessons to be learned which, with appropriate modification, can be successfully applied to marine engineering. Fig. 3 (Plate 1) shows the compound planet gear for the Bristol Theseus which was the first propeller gas turbine to pass a Ministry of Supply type test.

Small reversing epicyclic gears have been in constant use for some twenty years in conjunction with Diesel engine driven main propulsion machinery and a modern gear in this field is described later. It is not intended to deal with the bevel type of reversing epicyclic gear which is also commonly used.

It is interesting to see in T. W. F. Brown's paper[1] that the marine engineering industry realizes the potential advantages of large marine propulsion reversing epicyclic gears.

It is unfortunate that security restrictions prevent one knowing what other developments in the same direction may be going on, either in this country or abroad.

The theory of design and the methods of determining ratios of the many forms and inversions of epicyclic gear trains have already been studied and discussed at great length by Love[11], Merritt[12], Stoeckicht[15], Wilson[17], and many others, and it is not proposed to go deeply into these matters.

TYPES

The essential difference between an epicyclic and an ordinary gear train is that in an ordinary train the axes of the wheels are fixed but in an epicyclic train at least one axis moves around another axis which is fixed.

The basic elements of the gears described in this section are:—

(1) A central sun wheel.
(2) An internally toothed gear ring or annulus.
(3) A planet carrier in which the planet wheels are supported.
(4) A planet wheel or wheels engaging with the sun pinion and/or annulus.

There are many possible forms and combinations of epicyclic gears. Here it is proposed to describe five types only,

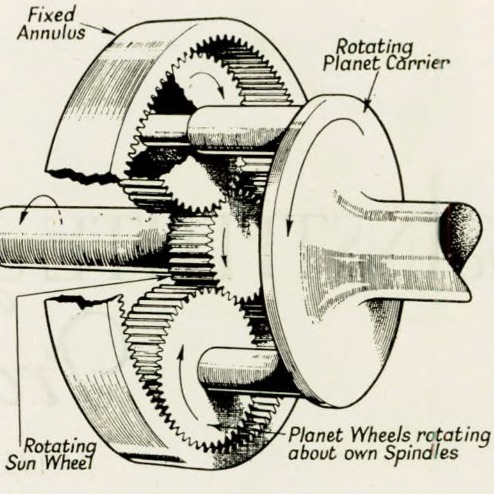


FIG. (4a)—*Planetary gear*

one of which is not a true epicyclic, and one or two of the possible combinations of these.

So far as the authors know, there are no generally accepted terms for the types which are to be described and for the sake of convenience, therefore, these will be referred to as "planetary", "star", "solar", "compound planet" and "double annulus" gears respectively.

Planetary Gears

Fig. 4(a) shows a diagrammatic representation of a planetary gear. The annulus is fixed to the casing and the planet carrier rotates in the same direction as the sun wheel.

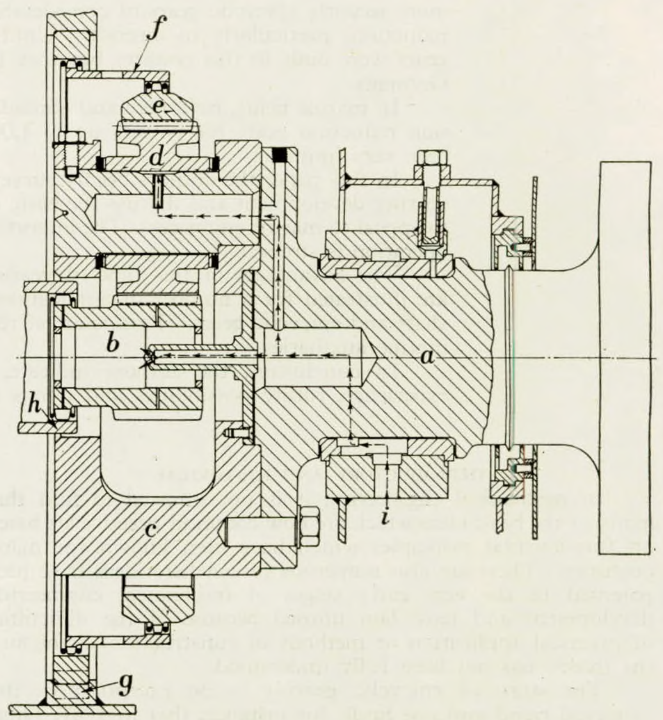


FIG. 4(b)—*5:1 planetary gear*

(a)—Low speed shaft
(b)—Sun wheel
(c)—Planet carrier
(d)—Three planet wheels
(e)—16-in. p.c. diameter annulus
(f)—Double toothed flexible coupling ring fixing annulus to gearcase
(g)—Torque reaction from gear taken by key at horizontal joint of gearcase
(h)—Part of flexible coupling connecting sun wheel to rotating annulus of first train star gear
(i)—Oil inlet

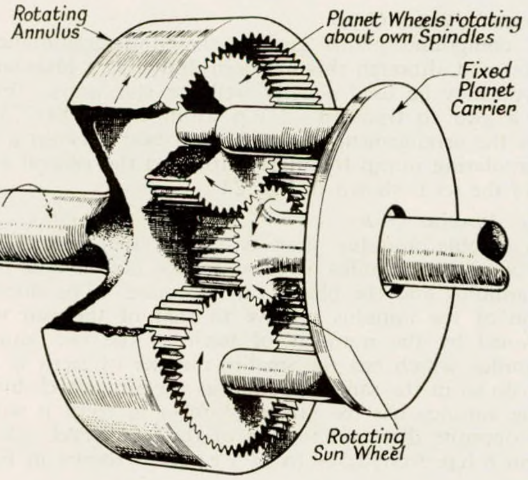


FIG. 5(a)—*Star gear*

Fig. 4(b) shows such a gear to transmit 600 h.p. from 3,000 to 600 r.p.m. The gear shown is the second train of a double reduction gear between a turbine and generator. The sun wheel is connected to the rotating annulus of a star gear which reduces the turbine speed from 16,500 to 3,000 r.p.m. It will be noticed that the bearings which would be necessary for the pinion of a parallel shaft gear have been eliminated.

The planet carrier is bolted up solidly to the armature shaft and the bearing at the right of the illustration, from which the planet carrier is overhung, forms the second bearing for the generator. A general arrangement of the set is shown in Fig. 21.

Star Gears

Fig. 5(a) shows a diagrammatic representation of a star gear and it must be pointed out that, according to the definition given at the beginning of this section, this is not a true epicyclic gear, although it is of the same general form. In this type of gear there is no relative motion between the axes of any of the wheels. The planet carrier is fixed in the casing and the annulus rotates in the opposite direction to that of the sun wheel. Fig. 5(b) shows a gear to transmit 550 h.p. from 1,000 to 625 r.p.m. The gear shown is a reduction gear for the main propulsion reversing gearbox shown in Fig. 25(a).

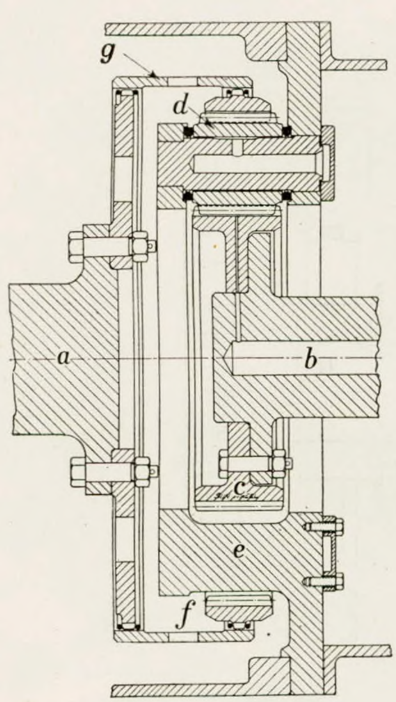


FIG. 5(b)—*1·6:1 star gear*

(a)—Low speed shaft
(b)—High speed shaft
(c)—Sun wheel
(d)—Six star wheels
(e)—Star wheel carrier
(f)—16-in. p.c. diameter annulus
(g)—Double toothed flexible coupling ring fixing annulus to low speed shaft

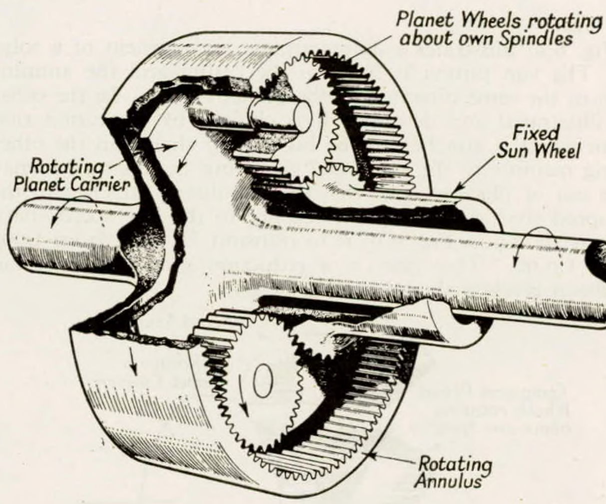


FIG. 6(a)—*Solar gear*

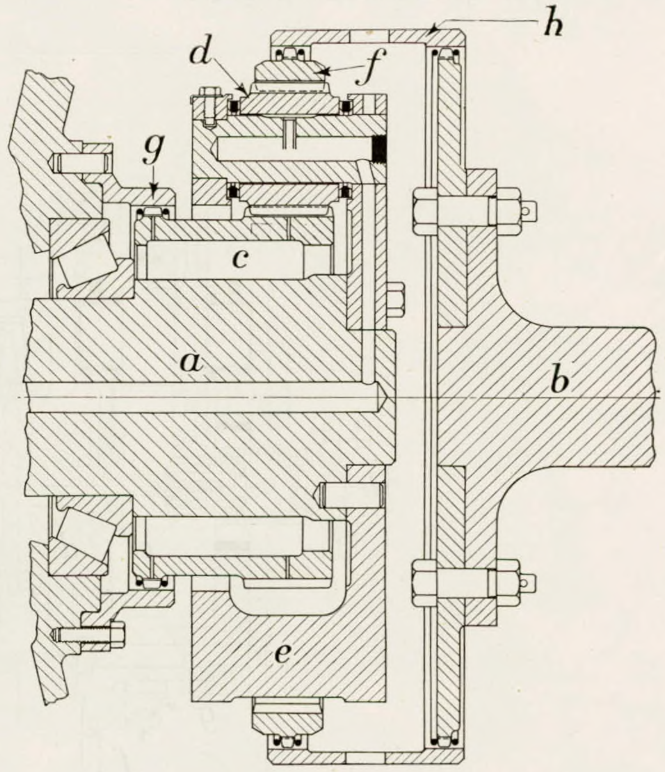


FIG. 6(b)—*1·6:1 solar gear*

(a)—Low speed shaft
(b)—High speed shaft
(c)—Sun wheel
(d)—Five planet wheels
(e)—Planet carrier
(f)—15·625 p.c. diameter annulus
(g)—Single toothed flexible coupling ring fixing sun wheel to gear case
(h)—Double toothed flexible coupling ring fixing annulus to high speed shaft

Solar Gears

Fig. 6(a) illustrates a diagrammatic arrangement of a solar gear. The sun pinion is fixed to the casing and the annulus rotates in the same direction as the planet carrier. In the other gears illustrated and described it is obvious by inspection that the sun wheel is attached to the high speed shaft and the other rotating member to the low speed shaft, but in this type it may not be out of place to state that the annulus is attached to the high speed shaft and the planet carrier to the low speed shaft. The gear shown in Fig. 6(b) is to transmit 550 h.p. from 1,000 to 625 r.p.m. This gear is a reduction gear for the main propulsion gearbox shown in Fig. 25(b).

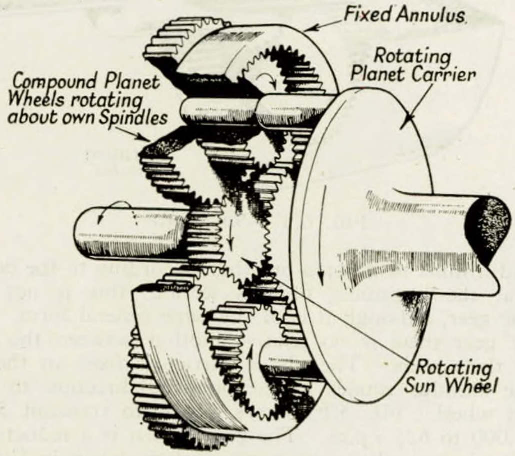


FIG. 7(a)—*Compound planet gear*

Compound Planet Gears

A compound planet gear is shown diagrammatically in Fig. 7(a) and although this has been drawn as a planetary gear, this type may be used equally well for star gears. Fig. 7(b) shows a gear to transmit 72 h.p. from 16,000 to 775 r.p.m. This is the arrangement of a reduction gear between a turbine and circulating pump for marine use and the general arrangement of the set is shown in Fig. 24.

Double Annulus Gears

A double annulus gear is shown diagrammatically in Fig. 8(a). One annulus is fixed in the casing and both the other annulus and the planet carrier rotate. The direction of rotation of the annulus relative to that of the sun wheel is determined by the numbers of teeth in the two annuli. If the annulus which has the smaller number of teeth is rotating it will do so in the same direction as the sun wheel, but if the rotating annulus has the greater number of teeth it will rotate in the opposite direction to that of the sun wheel. A gear to transmit 6 h.p. from 1,500 to 13·5 r.p.m. is shown in Fig. 8(b).

CHOICE OF TYPE

Before leaving this very brief outline of a few types of epicyclic gears, some of their characteristics and the reasons influencing the choice of a particular type should be mentioned.

In addition to the fact that the direction of rotation of the output shaft relative to that of the input shaft is determined by the type of epicyclic gear fitted, there are also limitations with regard to the ratio which may be obtained. Generally speaking, planetary gears are suitable for ratios from 3 : 1 to 12 : 1, star gears for ratios between 2 : 1 and 11 : 1, solar gears for ratios between 1·2 : 1 and 1·7 : 1, compound planet gears for ratios of 11 : 1 to 25 : 1, and double annulus gears for ratios from 25 : 1 to 1,600 : 1.

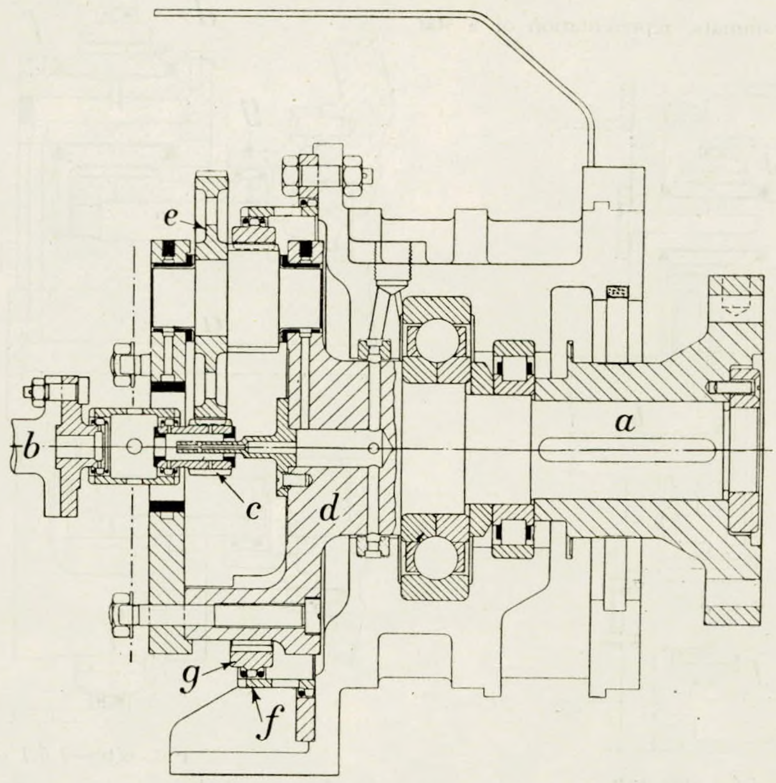


FIG. 7(b)—*20·6:1 compound planet gear*

(a)—Low speed shaft
(b)—High speed shaft
(c)—Sun wheel
(d)—Planet carrier
(e)—Three compound planet wheels
(f)—9·25 p.c. diameter annulus
(g)—Double toothed flexible coupling ring fixing annulus to gear case

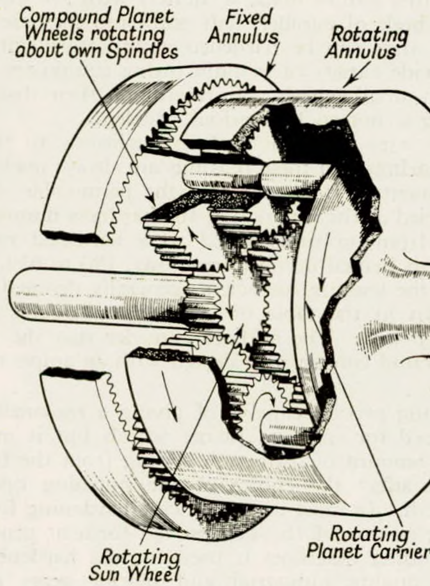


FIG. 8(a)—*Double annulus gear*

It will be noted that in certain cases it is possible to use more than one gear for a given ratio. For example, both the star gear shown in Fig. 5(b) and the solar gear in Fig. 6(b) have the same ratio and each of these forms a reduction train in otherwise identical main propulsion reversing gearboxes. They were designed to give opposite rotation for the two propellers of a twin-screw ship in which both Diesel engines have the same rotation.

In addition to the tooth reaction loads which have to be taken by the planet bearings in planetary and solar gears, there is also a centrifugal load on the planet bearings and for this reason these gears are not suitable for high speeds of rotation of the planet carrier.

Star gears are normally used when the low speed shaft rotates at 3,000 r.p.m., or more. The ratio of a planetary gear is $\frac{A}{S} + 1$ and of a star gear $\frac{A^*}{S}$, and consequently the annulus of a star gear is larger than that of a planetary gear for the same ratio.

Solar gears may be used to obtain lower ratios than those which it is possible to achieve with star or planetary gears.

Care must be taken in the design of compound planet gears because the use of a compound planet wheel gives rise to a tipping moment on the planet wheel bearings. This tipping moment arises because the tooth load at the sunwheel-planet contact is not in the same plane as that of the planet-annulus contact.

Double annulus gears are generally more efficient than worm gears and may be used in their place where the concentric arrangement of input and output shafts is desirable.

DESIGN

General

Amongst the main differences between the tooth working conditions in epicyclic gears and parallel shaft gears, the following should be particularly noted:

(a) The surface stress between internal and external teeth running together is less than that between two external teeth because the relative radius of curvature, which is inversely proportional to the stress, is so much increased. For this reason it is usually found unnecessary to case-harden the annuli used in epicyclic gears even when the sun wheel and planet wheels are case-hardened.

(b) For a given size and number, teeth cut on an internal ring are stronger than those cut on an equivalent external wheel.

Epicyclic gears may be made with straight spur, single helical or double helical teeth but the design of planet bearings for gears with single helical teeth needs special consideration.

The principal advantages which epicyclic gears offer to marine engineers are considerable savings in weight and space and higher efficiency compared with parallel shaft gears. The co-axial arrangement of input and output shafts is also an

* See Appendix 1.

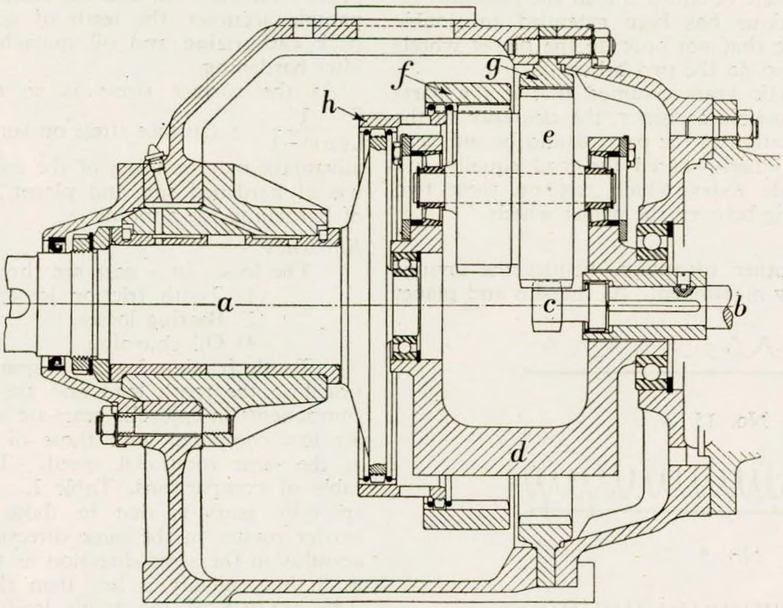


FIG. 8(b)—*111:1 double annulus gear*

(a)—Low speed shaft
(b)—High speed shaft
(c)—Sun wheel
(d)—Planet carrier
(e)—Three compound planet wheels
(f)—Rotating annulus
(g)—14·25 p.c. diameter annulus
(h)—Double toothed flexible coupling ring fixing rotating annulus to low speed shaft

advantage in many cases.

These savings in weight and space are due to the fact that the load is transmitted by several tooth contacts. For example, in a gear with three planets each tooth engagement of the sun wheel would have to carry one-third of the total load, which means, broadly speaking, that the sun wheel would have the dimensions of the pinion of a parallel shaft gear designed for a third of the actual load. This reduction in size leads to further advantages which will be discussed later.

Load Sharing between Planet Wheels

In order to gain the advantages of space and weight saving offered by the use of epicyclic gears, it is essential to ensure that the planet wheels do, in fact, share the load equally. Obviously, there is no point in providing several planet wheels if only one is actively engaged in transmitting power.

If torque is applied to an epicyclic gear with three planet wheels whilst it is prevented from rotating and one of the co-axial members, either the sun wheel or the annulus or the planet carrier, is allowed to float, the tooth forces involved will move the floating member until equilibrium is obtained. This state of equilibrium will be reached when the tooth loads on all the members are equal. This is mentioned by Merritt[12] and, as far as the authors know, Stoeckicht is the first gear designer to have used a floating member as standard practice.

Under running conditions the problem becomes more complicated, since it is impossible to manufacture components which are perfectly accurate. Very small errors can cause considerable accelerating forces to act but experience has shown that by mounting the annulus flexibly in the gearcase, very good load sharing can be achieved.

This has been proved by the many gears made to Stoeckicht's designs which have given long periods of trouble-free service and also by applying strain gauges to the fixed annuli of planetary gears running under load. The method of applying this technique has been described fully by Keig[9] but, briefly, it consists of attaching a number of electrical resistance strain gauges to the outside of the annulus and measuring the strains produced as successive planets pass each gauge.

Fig. 9 shows some typical records from which it will be noticed that the amplitudes of three successive peaks due to the three planet wheels passing a gauge position are almost identical. Similar results were obtained for all the positions on the annulus. This technique has been extended in double helical gears to demonstrate that not only do the planet wheels share the load equally but so do the two helices.

Earlier users of epicyclic gears assumed that if the parts were made with the best possible accuracy, the clearance in the bearings and elastic deformation of the parts would be sufficient to ensure that the planet wheels shared the load equally and there are many gears made today which rely on these two assumptions for load sharing between the planet wheels.

Hardened Components

Epicyclic gears have other advantages besides the savings in weight and space already mentioned. As the sun and planet wheels are small they can be made accurately with less difficulty than the larger wheels of parallel shaft gears and they are easier to handle. Also they can be hardened more conveniently because there is a wide variety of manufacturing equipment available to deal with small wheels. Furthermore, their distortion during hardening is not such a serious problem.

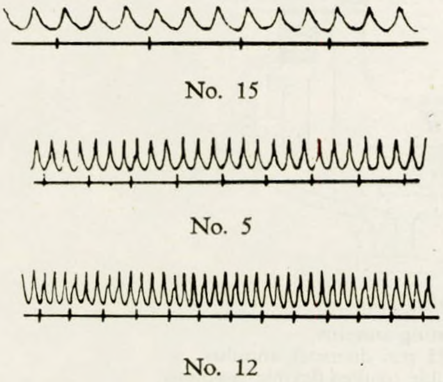


FIG. 9—*Typical strain gauge records*

The surface stress in gear teeth is analogous to that in ball and roller bearings and these bearings are always made with hardened components. It is said that the permissible surface stress may be varied as the square[7] of the hardness number and when changing from an unhardened to a hardened material with an increase in Brinell number from, say, 190 to 630, much higher values of the loading coefficient, generally denoted by K, than those shown in the table of comparisons (Table I) are theoretically acceptable. The authors consider that the figures given represent sound conservative design with an ample margin of safety.

Any hardening process capable of giving a reasonable case depth may be used for sun and planet wheels but it must be realized that the amount of distortion arising from the hardening process will affect the choice of the finishing operation applied to the teeth. In most instances, case hardening followed by gear grinding on any of the well proved form or generating type of gear grinding machines is used for the hardened gear wheels of high quality industrial and marine gears in this country. In the future, induction hardening may offer a means of hardening, with distortion low enough to eliminate the necessity for any post hardening process on the gear teeth.

One method of hardening gear teeth which is used in Germany is nitriding and the authors have experience of successful gears which have been hardened by this process in England. They believe that this method of hardening gear teeth will become more widely used in this country. Some of the points in its favour are the almost negligible amount of distortion which takes place during hardening, adequate case depth blending well into the core and high core strength. After nitriding, a slight "bloom" or softer layer is present on the surface to the depth of about ·001 inch, but this can be quickly removed by light lapping if required. This lapping also gives a slight amount of tip and root relief to the teeth. Gears have been run very successfully without any post-nitriding process on the teeth and the authors have found that for comparable accuracy the teeth of gears which are hardened by pack carburizing and oil quenching must always be ground after hardening.

As the surface stress is so much less on the annulus, $\left[\frac{1}{\text{ratio}-1} \times \text{(surface stress on sun wheel) for a planetary gear}\right]$ advantage may be taken of the reduction in size offered by the use of hardened sun and planet wheels without the necessity of hardening the annulus.

Efficiency

The losses in a gear are threefold:

(1) Tooth friction losses.
(2) Bearing losses.
(3) Oil churning.

Tooth friction losses in gears are approximately proportional to the tooth load and the pitch line velocity. As the components in epicyclic gears are small their pitch line velocities are low compared with those of parallel shaft gears running at the same rotational speed. This is clearly shown in the table of comparisons, Table I. Another point in favour of epicyclic gears is that in those instances where the planet carrier rotates in the same direction as the sun pinion, or the annulus in the same direction as the planet carrier, the relative pitch line velocity is less than the actual pitch line velocity. The product of the tooth load and the relative pitch line velocity is proportional to the "potential horsepower" which is discussed at length by Buckingham[2]. It follows that since the tooth losses are proportional to the potential horsepower, the efficiency of a gear in which the potential horsepower is less than the transmitted horsepower will be higher than that of a gear in which these two are the same.

In solar gears for example, the efficiency is exceptionally high because the potential horsepower is equal to:

$$\frac{\text{ratio} - 1}{\text{ratio}} \times (\text{transmitted horsepower})$$

e.g. in a 1·5 : 1 solar gear the tooth friction losses will be only one-third of those of a parallel shaft gear of the same ratio.

Bearing losses are dependent on the size of the bearings and when input and output shaft bearings are supplied in epicyclic gears they are small because no tooth reaction loads have to be carried. It may be of interest to note that a lightly loaded 2-in. diameter bearing 2 inch long with a shaft running at 7,500 r.p.m., lubricated by a medium turbine oil, will absorb about 1·5 h.p. In epicyclic gears it is often possible to eliminate the high speed sun wheel bearings, either by overhanging the sun wheel from the high speed shaft or by connecting it to the high speed shaft by means of a simple gear tooth type flexible coupling and supporting it between the planet wheels. It is also possible in some cases to eliminate one of the low speed shaft bearings by overhanging the planet carrier, as has been done in the gear illustrated in Fig. 4. Under the above conditions the only bearings in the gearcase are the planet wheel bearings and one low speed shaft bearing. The losses in these are usually less than those in the bearings of a parallel shaft gear, as indicated in the table of comparisons, Table I.

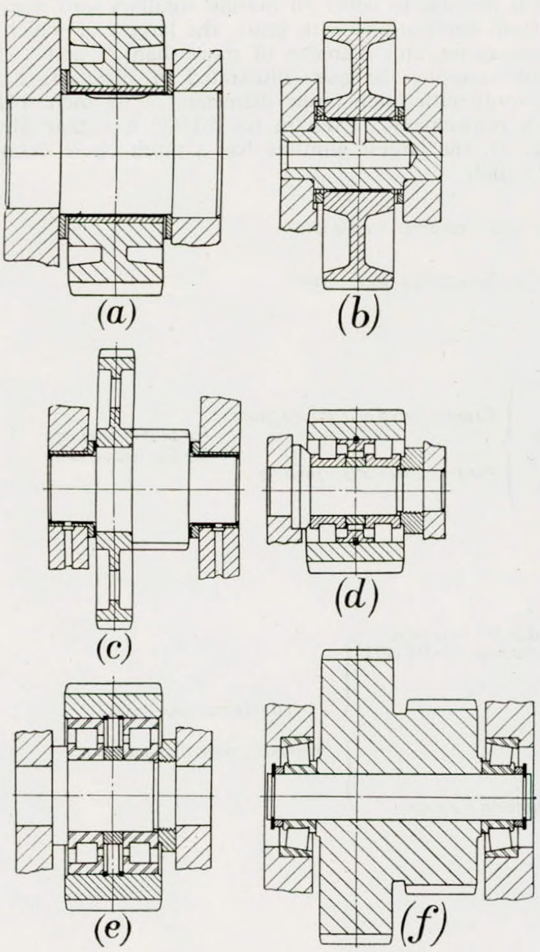


FIG. 10—*Typical planet bearing arrangements*

(a)—Bush fitted in wheel
(b)—Whitemetal faced spindle
(c)—Bushes fitted in planet carrier
(d)—Hardened bore of wheel used as outer race of roller bearing
(e)—Roller bearings in planet wheel
(f)—Taper roller bearings in planet carrier

In a well-designed epicyclic gear, oil churning should play a negligible part in the total losses.

Planet Wheel Bearings

One feature of epicyclic gears which needs special attention is the design of the planet wheel bearings. Plain or ball or roller bearings may be used according to circumstances. Fig. 10 shows some typical planet bearing arrangements. At (a) is shown a planet wheel which has a bush lined with bearing metal pressed into the bore of the wheel. This method is in general use and many gears are running successfully with such bearings, but the arrangement shown at (b) has several advantages. In this arrangement a bearing metal facing is applied direct to the planet wheel spindle. With this arrangement the load is always in the same direction on the bearing metal and consequently there is less chance of the metal failing by fatigue than there is with the arrangement at (a) where the white metal bearing is continuously rotating and is subjected in effect to a rotating load. With the arrangement shown in (b) it is possible to finish the bore of the wheel before the teeth are finished and thus ensure that they are true with the bore. In the arrangement shown in (a) it is necessary to finish bore the bush true with the teeth after it has been pressed into the wheel, which is not an easy operation. Alternatively, the wheel and bush must be made to very close limits so that after pressing into the wheel, both the requisite fit between the bush and the wheel and the required dimension for the bore of the bush are obtained without the need for finish machining the bush in position. If a bush were to move axially in the wheel whilst the gear was running it might have serious consequences, but with a bearing metal faced spindle this possibility cannot arise.

At (c) is shown an arrangement for the bearings of a compound planet wheel. In this instance bushes pressed into the planet carrier are preferred because they spread the bearing centre distance and form the best arrangement to resist the tipping moment, which is present in this type of gear. As the bushes are pressed into the planet carrier and not into the wheel, the load on the bearing metal is always in the same direction.

The authors consider that white metal is the best bearing material for either the bushes or the faced spindles used in planet wheel bearings. It is more than adequate for the loads which have to be carried. Planet wheel bearing pressures up to 750lb. per sq. in. have proved satisfactory but pressures are usually well below this figure.

Fig. 10(d), (e) and (f) shows several ways in which roller bearings may be applied in planet wheels. The wheel at (d) uses the hardened and lapped bore of the planet wheel as the outer race of the roller bearing. This method can be adopted when there is no room for a conventional outer race; (e) shows a bearing where there is room for an outer race; (f) shows the use of taper roller bearings for compound planet wheels.

It is often unnecessary to supply oil under pressure for gears which are fitted with ball and roller bearings, splash lubrication being adequate.

Gearcases and Main Shaft Bearings

The planet bearings alone are subjected to tooth reaction loads and therefore the main shaft bearings have only to support the weight of the components. This weight, the torque reaction from the gear and the thrust, when single helical gears are used, do not impose large forces on to the gearcase, and consequently this can be of light construction.

Planet Carriers

Planet carriers should always be made as rigid as possible and the spacing of the holes for the planet wheel spindles must be of the highest possible degree of accuracy. Fig. 11(a) shows a typical cast iron planet carrier. At (b) in the same figure is shown a typical forged planet carrier in which the low speed shaft coupling is forged integral with the carrier.

Lubrication

A typical method of lubricating an epicyclic gear is shown in Fig. 4. The planet wheels are fed with oil led into the

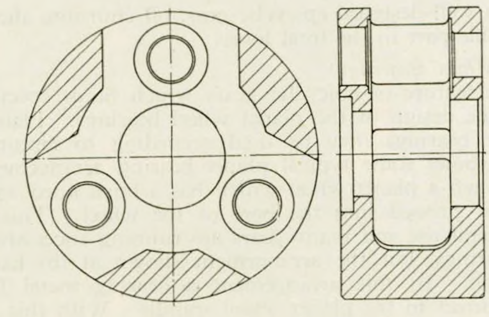


FIG. 11(a)—*Typical cast iron planet carrier*

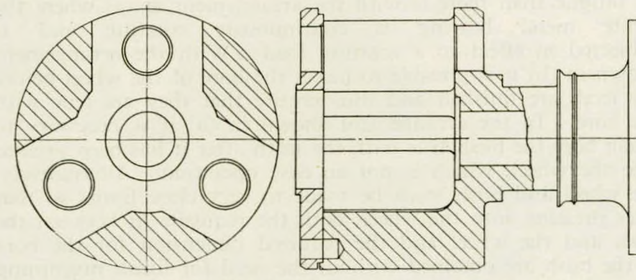


FIG. 11(b)—*Typical forged steel planet carrier*

centre of the low speed shaft and thence by radial holes via the centre of the planet wheel spindles to the bearings. The sunwheel-planet contacts are lubricated by oil, led into the centre of the sun wheel, which flows by radial holes into two tooth spaces. The planet-annulus contacts are lubricated by the oil mist present in the gearbox. The gear tooth type flexible coupling is lubricated by radial holes leading into each tooth space.

It is always desirable to use a filtered oil supply.

Accessibility

In the view of the authors, dismantling should be confined to the minimum required by statutory regulations. As with other machinery, damage can be caused by the introduction of dirt into working parts. When it is necessary to open up machinery for inspection it is always advantageous for this to take the least possible time. Fig. 12(a) (Plate 2) shows the outside view of a gear for a 200 kW. marine turbo generator. At (b) is shown the gear with the cover removed and at (c) the gear stripped so that all the parts, including the planet wheel bearings, can be examined. The time taken by two men to reach stage (c) was $1\frac{1}{2}$ hours, and the gear can be reassembled easily in the same time. With care, epicyclic gears can be designed so that the parts are accessible and, because all the parts are small, they are easy to handle.

Gear Sizes

Parallel shaft gear sizes are usually referred to in terms of the centre distance between the shafts, and in epicyclic gears it is easiest to refer to the gear size in terms of the annulus pitch circle diameter.

It is possible to cover all marine auxiliary and many main propulsion applications with gears, the largest of which would not have an annulus diameter of more than 25 inch.

For example, the gears illustrated in Figs. 23, 20, 25(a), have annuli with pitch circle diameters of $6\frac{3}{4}$ inch, $14\frac{1}{4}$ inch, 16 inch respectively. Even in the 10,000 h.p. gear illustrated in Fig. 31, the largest annulus has a pitch circle diameter of only 56 inch.

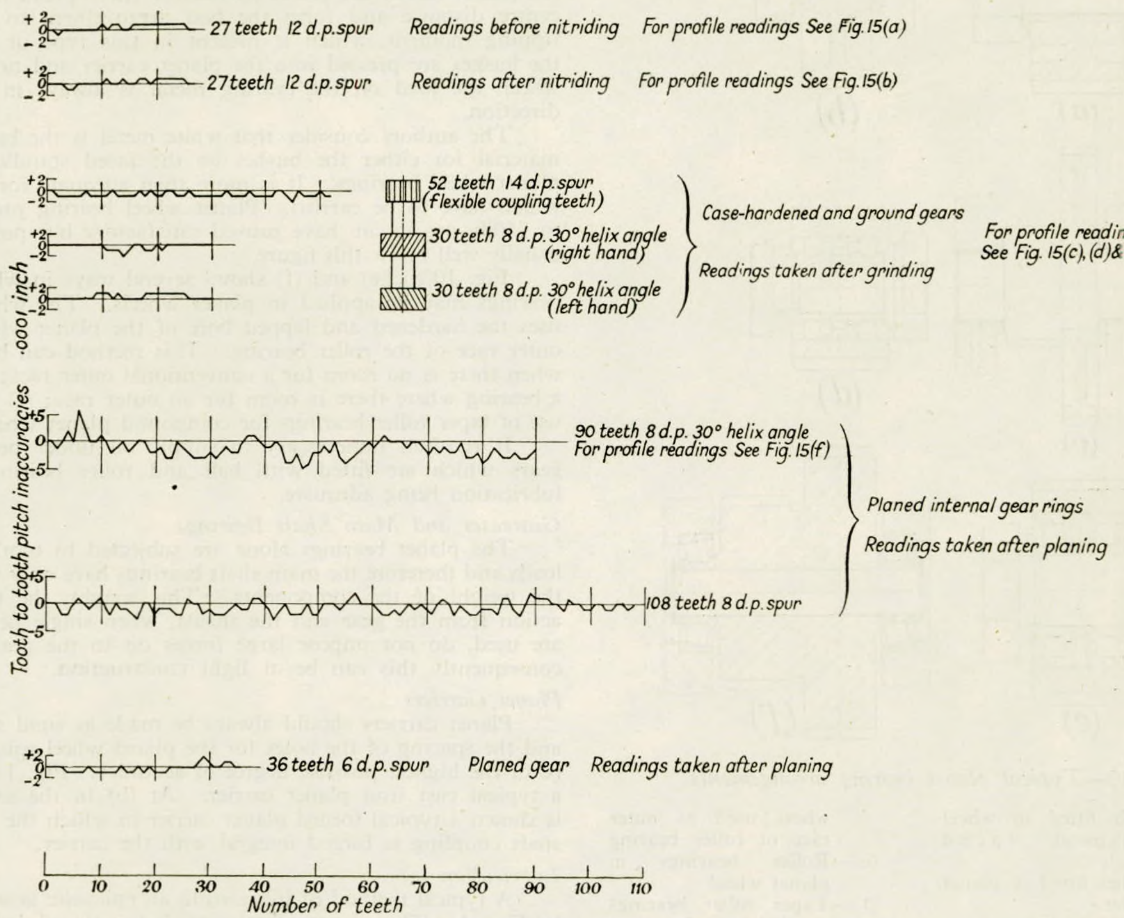


FIG. 14—*Typical pitch readings*

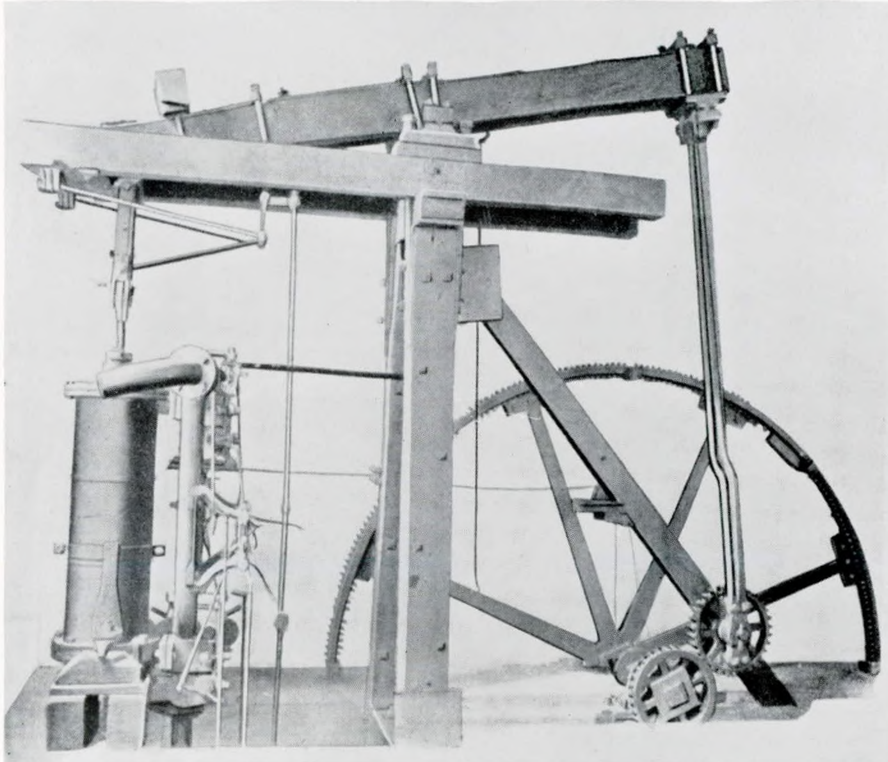


FIG. 1—*James Watt's engine with sun and planet gear*

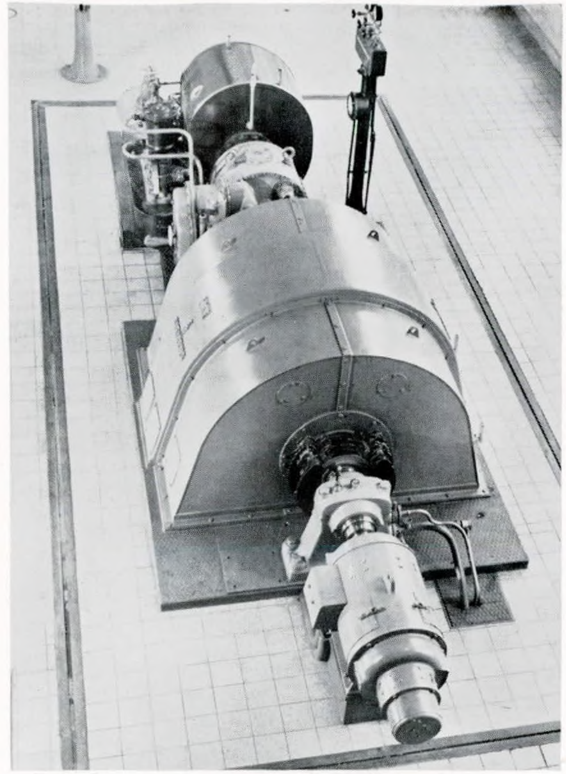


FIG. 2(a)—*1,100 kW. turbo-alternator with 18,000/1,500 r.p.m. epicyclic gear*

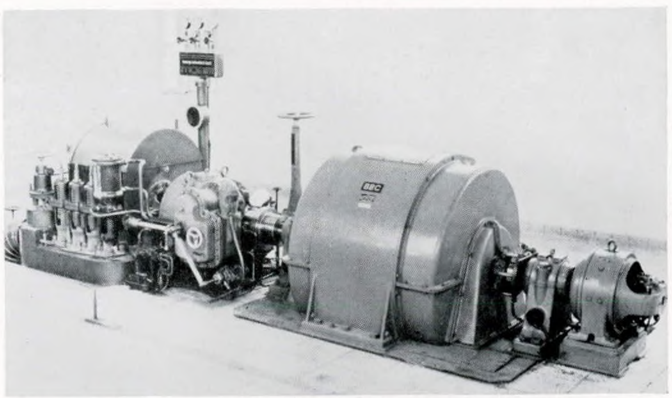


FIG. 2(b)—*865 kW. turbo-alternator with 16,000/1,500 r.p.m. epicyclic gear*

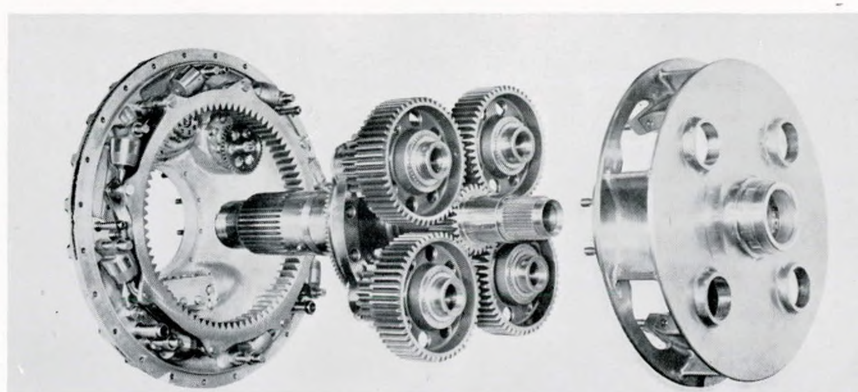


FIG. 3—*2,200 h.p. 9,000/1,070 r.p.m. Bristol Theseus gear*

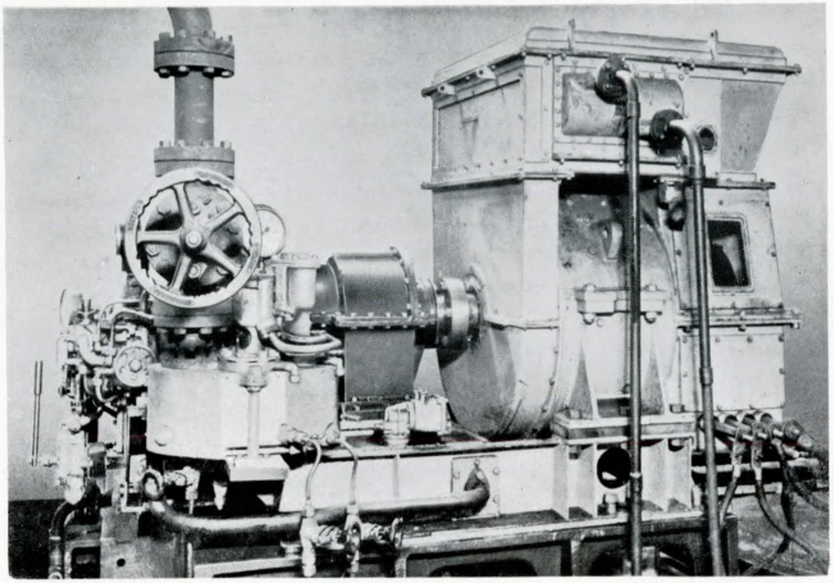


FIG. 12(a)—*200 kW. marine turbo-generator with 9,000/1,250 r.p.m. epicyclic gear*

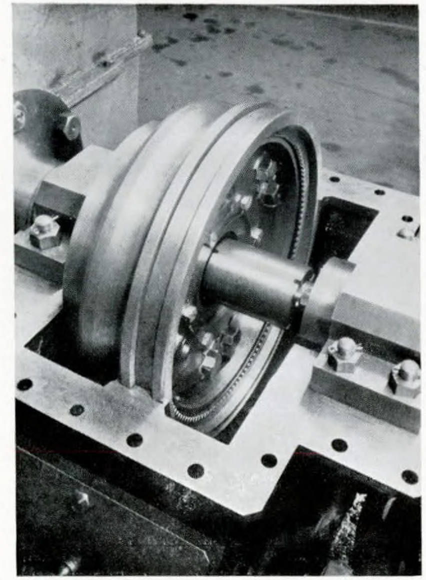


FIG. 12(b)—*View of gear with cover removed*

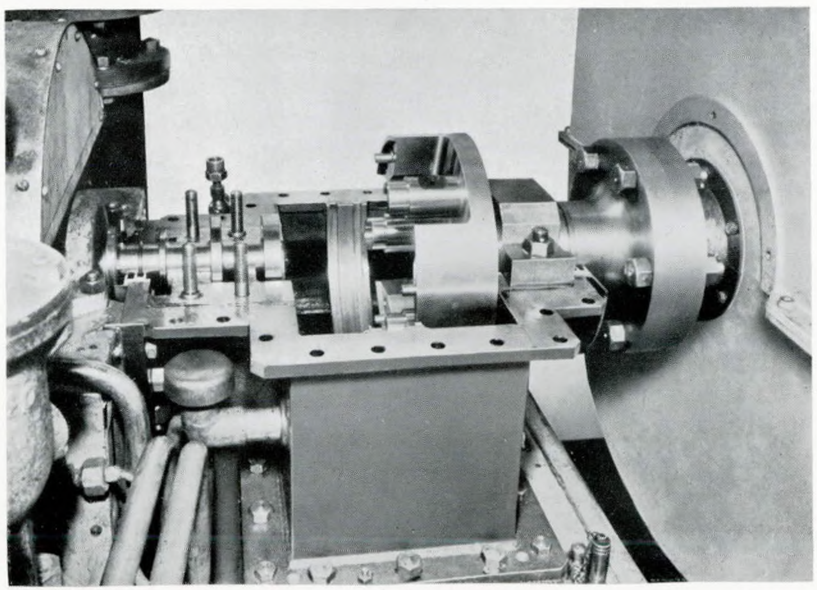


FIG. 12(c)—*View of gearcase and planet carrier after dismantling gear*

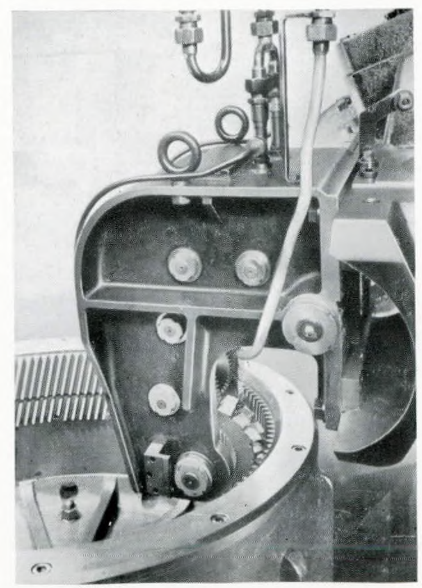


FIG. 13—*Internal teeth being produced on hobbing machine*

MANUFACTURE

General

The load carrying capacity, life and quietness of gears depend not only on experience in design but also on accurate manufacture both with regard to the gear tooth form and the general machining.

It has already been mentioned that within a comparatively small range of gear sizes it is possible to cover all marine auxiliary and many main propulsion requirements.

It follows that all the component parts within such a range are light and easy to handle, and, further, the general machining and gear tooth forming can be carried out on a narrow range of machine tools which are both cheaper in first cost and need less room than machine tool equipment for manufacturing gears of equivalent powers built as parallel shaft gears.

As with parallel shaft gears, a high degree of accuracy in gear tooth formation is necessary for epicyclic gears. The general dimensional accuracy required for the other components is of the same order as may be expected for any other modern precision piece of engineering but, in addition to such dimensional accuracy, it is important to achieve really good positional accuracy. This is especially the case for planet carriers and the bores of the various bearings. Where large numbers are involved, these points are best covered by a substantial degree of jigging or by the provision of special-purpose machine tools but, where a small number only is involved, requirements can be met by a production type of jig boring machine plus general machine tools kept in first class condition.

The production of forgings, castings and fabrications is all normal engineering practice requiring no comment.

The final check that components have been made to the required standard is that they should run satisfactorily when they have been assembled without fitting, and in practice this can be achieved.

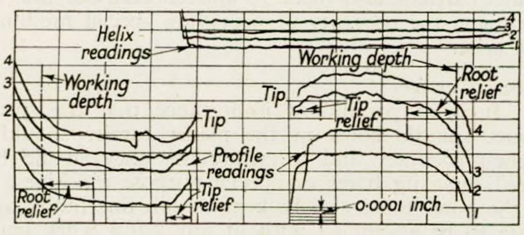
Taking into account the good general finish required, coupled with the comparative lightness of the parts, care in handling must not be overlooked as an essential item in the sequence of manufacture. Finished parts should always be properly protected until they are ready for assembly.

More damage can be done in initial test bed running than in the whole life of the gears if adequate precautions are not taken to clean out all oil ways and if the interior surfaces of the gearbox are not free from dirt and swarf. Special care should be taken with the filtering of oil used during test running.

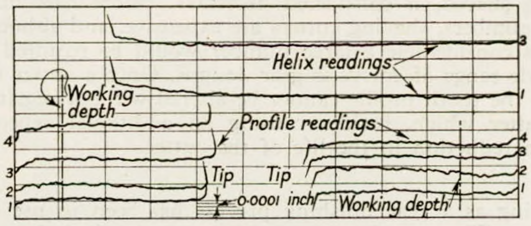
Only those points of special or unusual interest in the manufacture of epicyclic gears are described here and it should be appreciated that most of the components are manufactured by methods normal to other forms of gearing. Further, the methods described are those which, in the experience of the authors, have been found satisfactory. There are, of course, other alternative methods which may be equally satisfactory.

Hobbing and Planing with Special Reference to Internal Gears

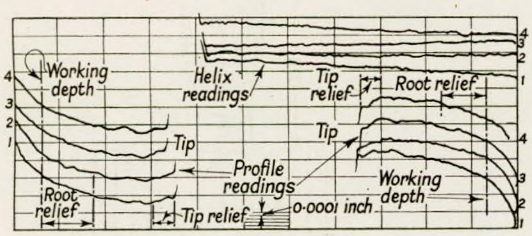
There are a number of factors which affect the choice of machines for cutting the teeth of epicyclic gears. The hobbing process has been developed to such a degree in the last years that the pitch accuracy now obtainable is of a very high order. For this reason alone there is much to be said for hobbing but to cut an internal gear on a hobbing machine requires a cutter which has been designed specially for the gear in question and any alterations in design necessitate an alteration to the form



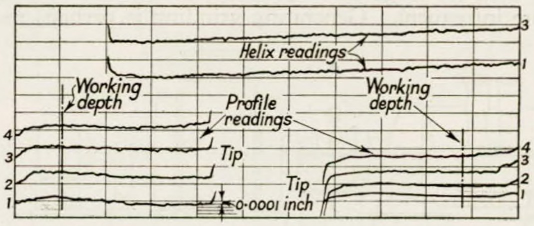
(a)
Readings taken on 27-tooth 12 d.p. spur gear, before nitriding



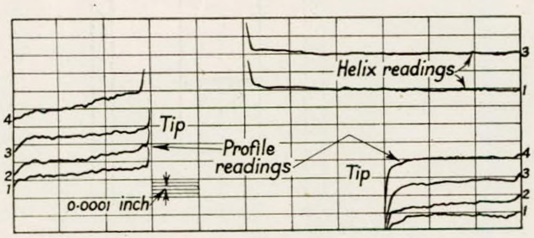
(d)
Readings taken on 30-tooth 8 d.p. 30 deg. helix angle gear (r.h.)



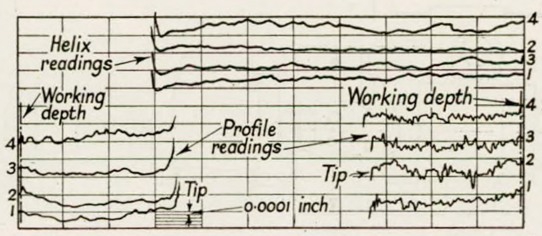
(b)
Readings taken on 27-tooth 12 d.p. spur gear, after nitriding



(e)
Readings taken on 30-tooth 8 d.p. 30 deg. helix angle gear (l.h.)



(c)
Readings taken on 52-tooth 14 d.p. spur gear (flexible coupling teeth)



(f)
Readings taken on 90-tooth 8 d.p. 30 deg. helix angle internal gear ring

FIG. 15

of the cutter. The reason for this is that, when cutting internal teeth, the hobbing machine is being used to carry out a continuous form milling operation and only one of the teeth on the cutter is complementary to the required gear tooth space. Fig. 13 (Plate 2) shows this operation being carried out successfully on a helical toothed annulus.

Gear planing machines, using rack type cutters for external teeth and with provision for cutting internal gears with Fellows type cutters, have been successfully used for epicyclic gear components. The accuracy obtained from high quality planing machines is very good, as can be seen from the graphs shown in Fig. 14. The necessity of changing the cutter for each change in design does not arise when annuli are planed. The designer is restricted in his choice of helix angle by the lead of the guides provided for the machine and the diameter of the cutters but this is not a great disadvantage. The authors have found that high quality work can be produced by planing and in their view it is generally the most suitable method for producing internal gears.

The annulus and coupling rings of the gears illustrated are slight in section and it might appear difficult to keep them in an undistorted condition, but it has been found that this is possible when the correct technique is used. Grinding the bore and outside diameter is inadvisable and annealing after rough machining the blank rings is considered to be essential. The teeth in the rings are always cut in a free condition, i.e. care is taken not to distort the ring when clamping it to the planing machine table.

Shaving

For batch production, an economical method of producing hardened sun and planet wheels is to use nitriding steels, the gears being shaved, if considered necessary, before hardening. For small numbers, shaving cutters are expensive, and although their life is considerable a large number would be required to cover a wide range of epicyclic gear designs. Once a cutter has been made the tooth profile cannot be altered without regrinding the cutter which, besides causing an additional expense, results in a reduction in the life of the cutter.

Grinding

Grinding as a gear finishing process has been in use for many years. The case for its adoption was particularly well stated in H. F. L. Orcutt's paper[13] in 1925.

Both form grinding and generating grinding have their place in the manufacture of epicyclic gears. Form grinding is more often used for large batch production and when changes in design are infrequent. Generating grinding is perhaps more

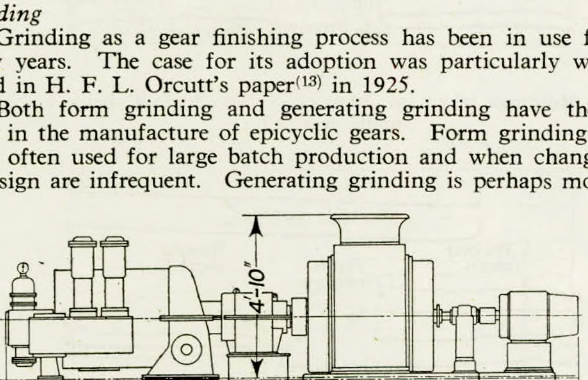


FIG. 16—*750 kW. turbo-alternator with 7,000/1,500 r.p.m. epicyclic gear*

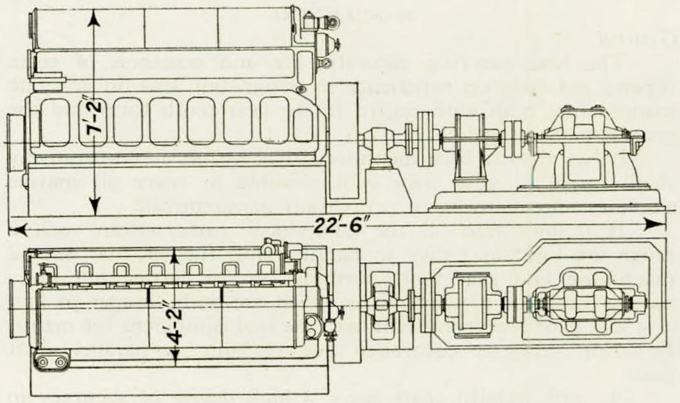


FIG. 17—*241 h.p. Diesel driven pump with 568/1,770 r.p.m. epicyclic gear*

suitable when small numbers of a varied range of designs are required.

Checking of Toothed Components

The authors consider it advisable to use a universal gear checker to enable the helix angle, profile and concentricity to be recorded at one setting. This saves much time and has the advantage that measurements are made from one basis. Some specimen test records are shown in Fig. 15. The comparative smallness of epicyclic gear parts enables the actual profile and helix to be easily measured with certainty and, by using planet carriers, also made to and checked for fine dimensional accuracy, the necessity of providing special meshing rigs does not arise.

Planet Carriers

It is of paramount importance that the spacing of the planet spindle bores in the planet carrier should be of the highest accuracy and to ensure this they should be precision bored on a jig borer or special-purpose machine. Parallelism of the bores must also be kept within fine limits to enable the gears to be assembled without any hand work and to ensure consistent meshing of all the wheels.

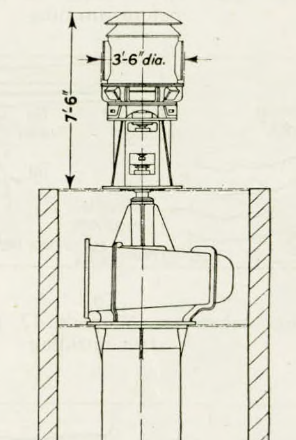


FIG. 18—*140 h.p. motor driven pump with 1,480/285 r.p.m. epicyclic gear*

Planet Wheel Bearings

The design merits of spindles faced with white metal have been discussed and their use has been shown to give advantages over bushed wheels. From the manufacturing point of view such spindles are no more difficult to produce than white-metal lined bushes. With such spindles the bores of the planet wheels may be honed and polished.

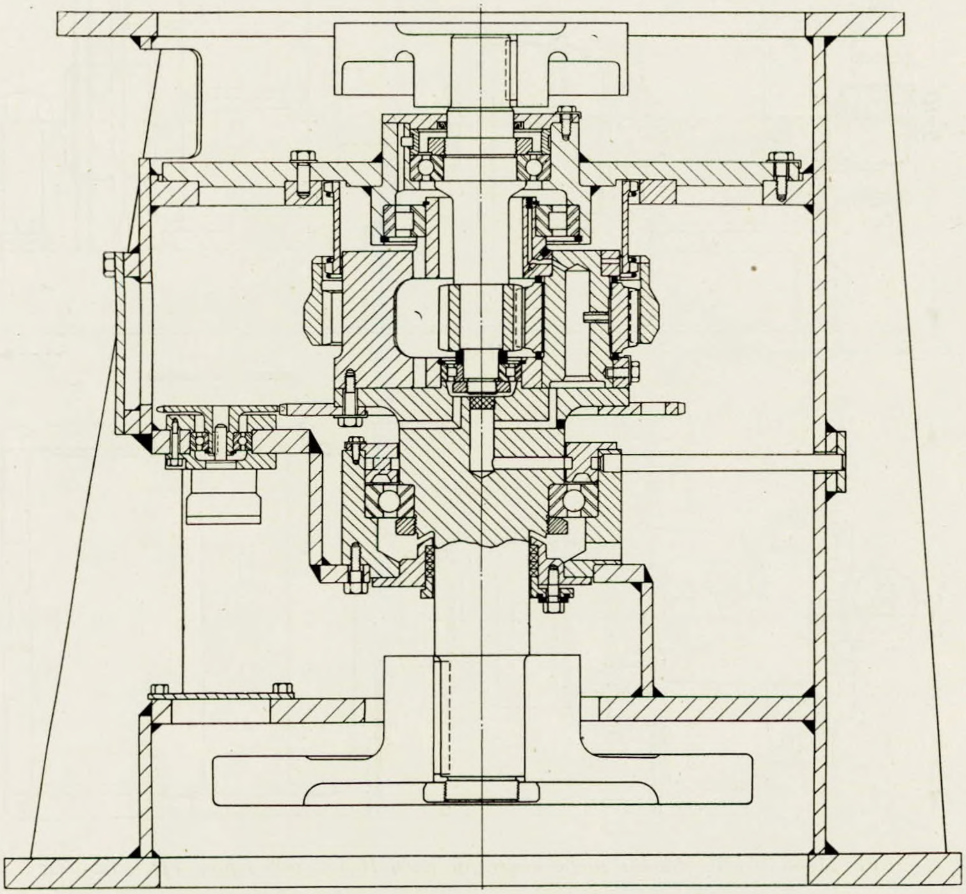


FIG. 19—*Sectional arrangement of gear shown in Fig. 18*

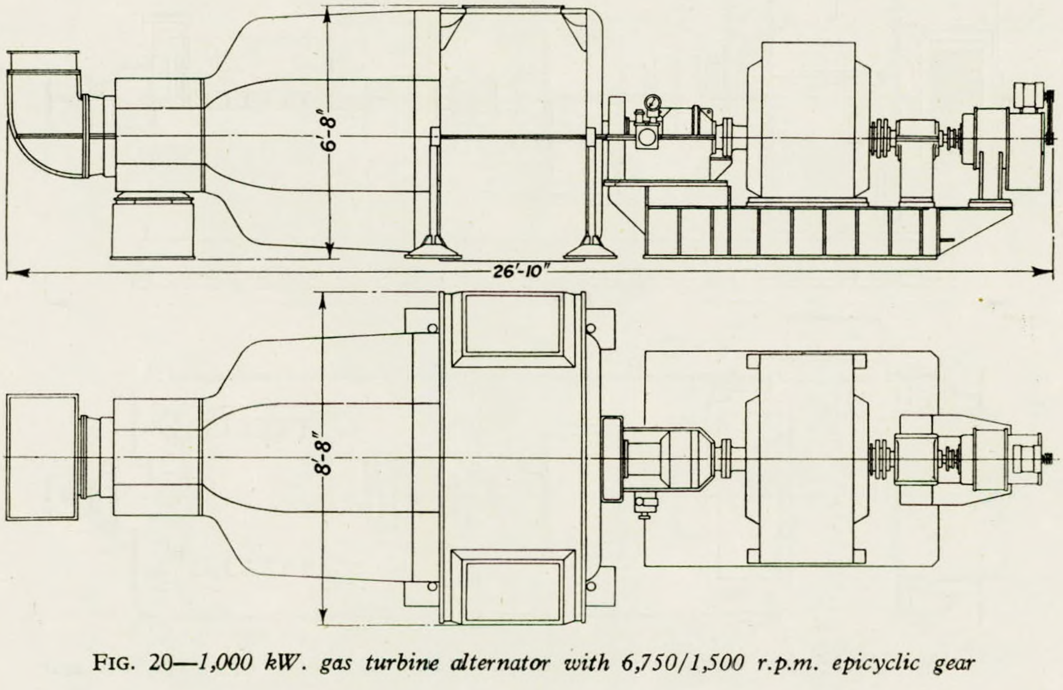


FIG. 20—*1,000 kW. gas turbine alternator with 6,750/1,500 r.p.m. epicyclic gear*

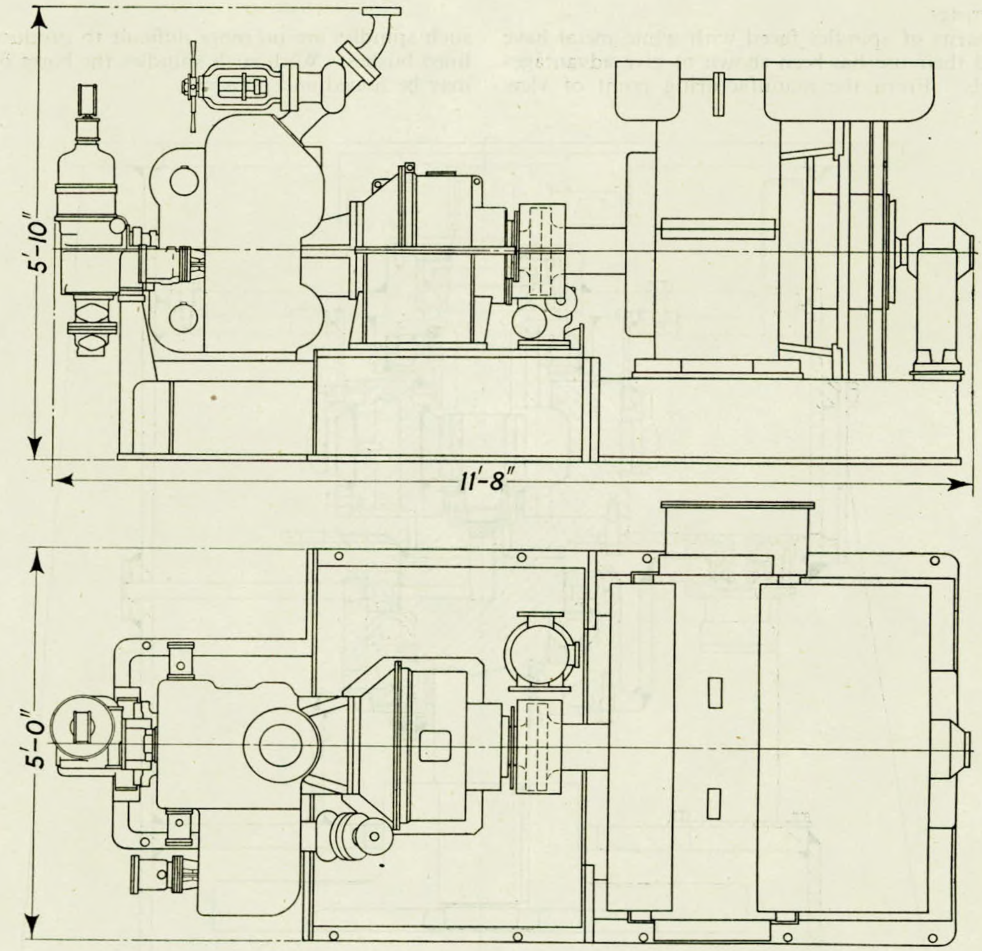


FIG. 21—*350 kW. marine turbo-generator with 16,500/600 r.p.m. epicyclic gear*

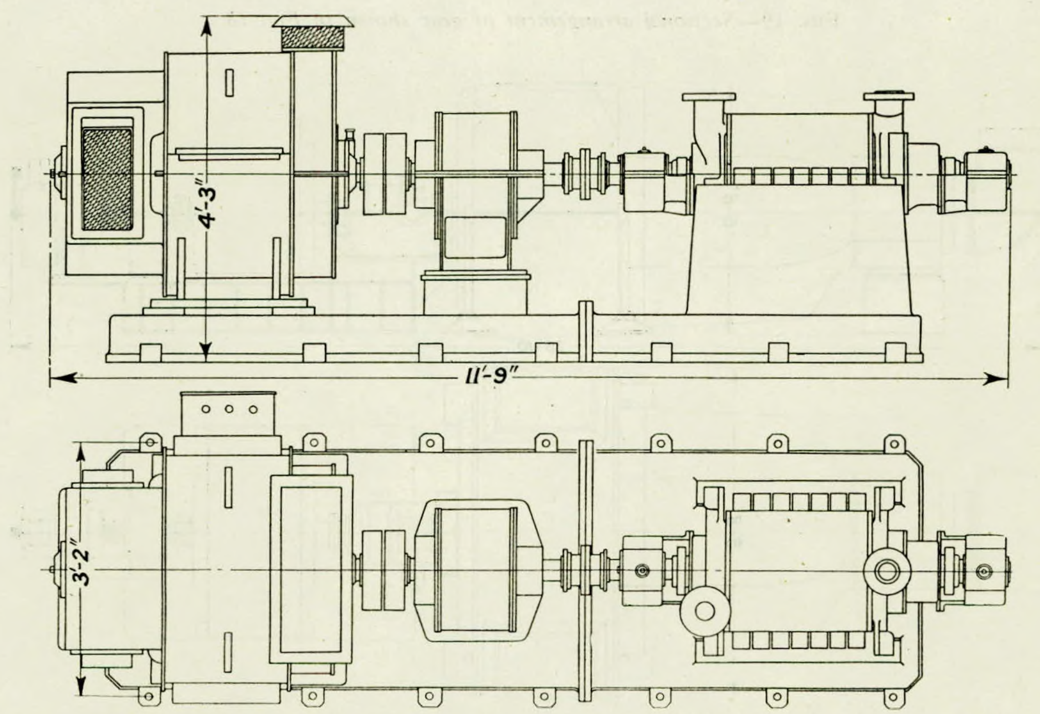


FIG. 22—*130 h.p. marine motor driven pump with 1,000/4,000 r.p.m. epicyclic gear*

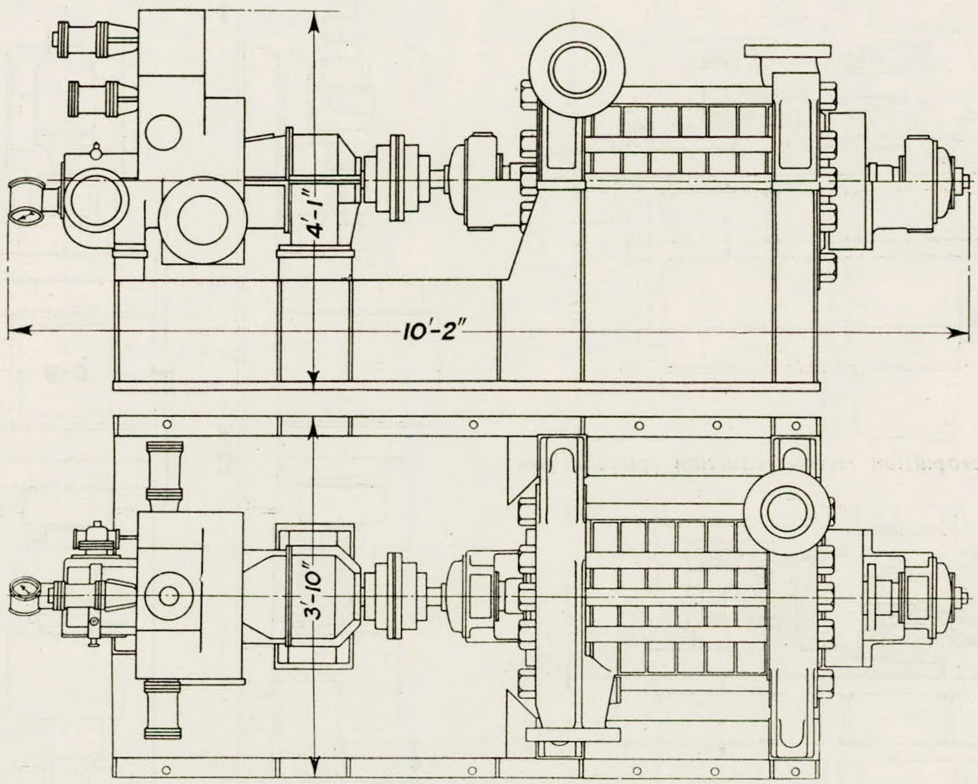


FIG. 23—*380 h.p. marine turbine driven pump with 14,000/3,100 r.p.m. epicyclic gear*

EXAMPLES FROM INDUSTRIAL AND MARINE APPLICATIONS

750 kW. 7,000/1,500 r.p.m. Industrial Turbo Alternator Set

Fig. 16 illustrates the above application and shows clearly the small size of the gear and the "in line" arrangement which results from its use.

241 h.p. 568/1,770 r.p.m. Diesel Driven Pump

Illustrated in Fig. 17 is a speed increasing unit. It is interesting to note that the gear is roughly of the same proportions as the flexible coupling between it and the Diesel engine.

140 h.p. 1,480/285 r.p.m. Vertical Motor Driven Circulating Water Pump

A vertical arrangement is shown in Fig. 18. In this case the epicyclic gear was mounted in the motor pedestal. The use of a 1,480 r.p.m. motor and speed reducing gear gave a considerable saving in head room, which was important in this application.

Fig. 19 shows a cross sectional arrangement through the pedestal and the gear.

1,000 kW. 6,750/1,500 r.p.m. Marine Gas Turbine Alternator

An epicyclic gear used between a gas turbine and alternator is illustrated in Fig. 20. The compactness of the gear is very clearly shown in this figure.

350 kW. 16,500/600 r.p.m. Marine Turbo Generator

Fig. 21 shows the very neat arrangement which can be obtained with an epicyclic gear even with a ratio of 27:1. For the ships in which these sets are to be installed, the generators for both steam and Diesel driven sets are interchangeable, those for the Diesels being direct coupled to engines running at 600 r.p.m. A double reduction epicylic gear fits in well with the high speed steam turbine used for the steam driven sets. The advantages gained by having both steam and Diesel driven generators interchangeable are obvious.

In this gear, the initial reduction is obtained by means of a star gear and the second stage is a planetary gear. The cross sectional arrangement through the second stage is illustrated in Fig. 4.

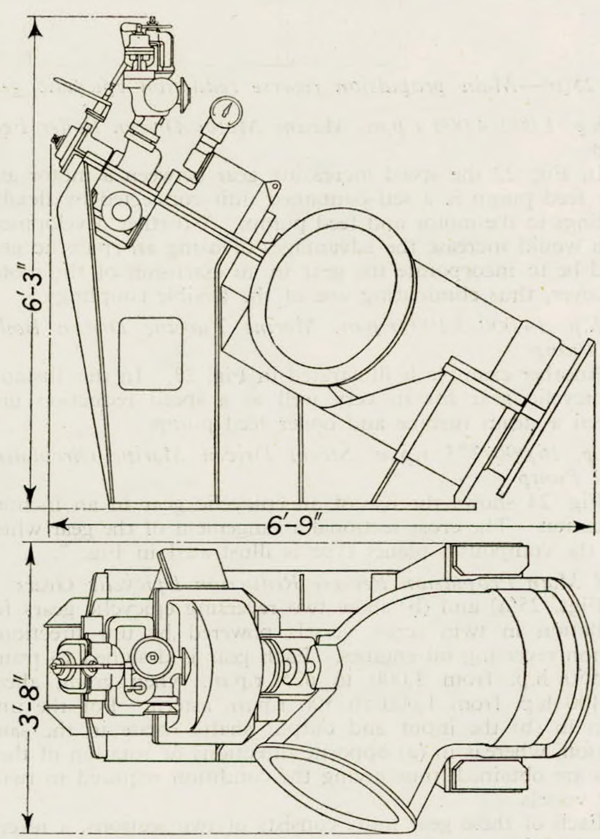


FIG. 24—*72 h.p. marine turbo-circulating pump with 16,000/775 r.p.m. epicyclic gear*

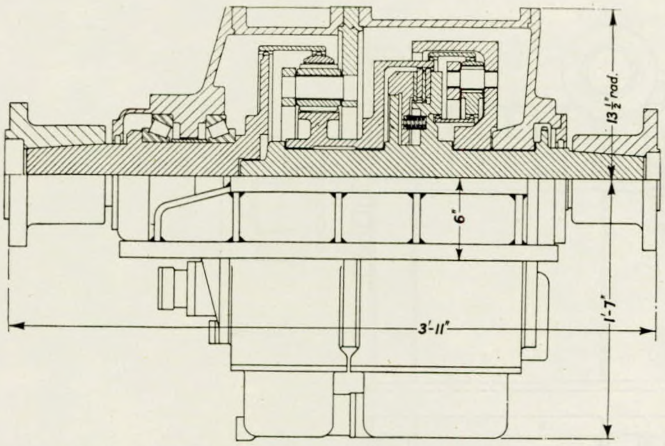


FIG. 25(a)—*Main propulsion reverse reduction epicyclic gear*

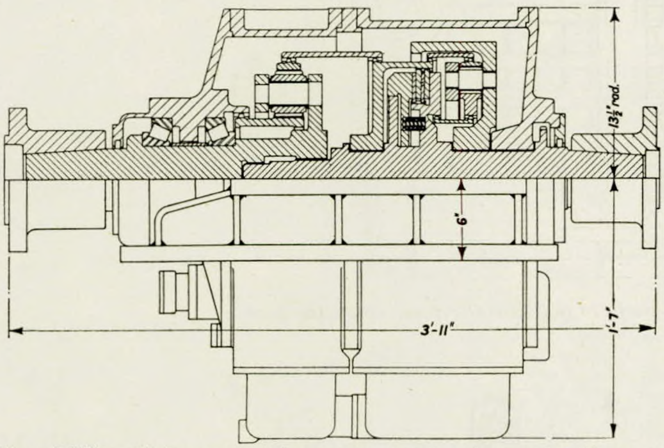


FIG. 25(b)—*Main propulsion reverse reduction epicyclic gear*

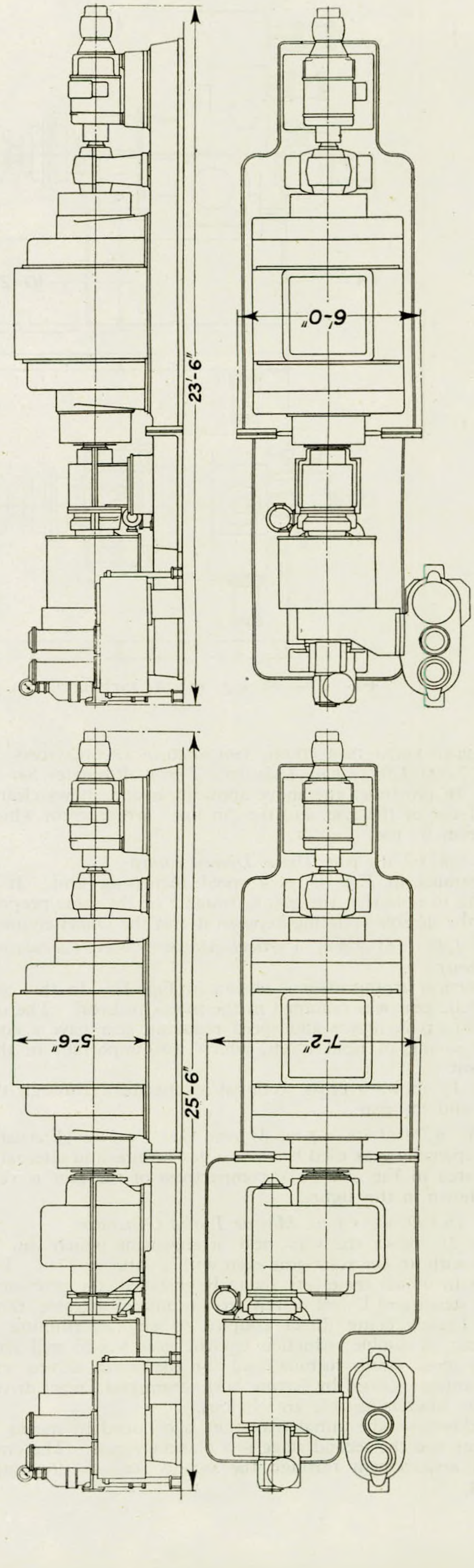


FIG. 26—*2,000 kW. turbo-alternator: gear ratio 4:1*

130 h.p. 1,000/4,000 r.p.m. Marine Motor Driven Boiler Feed Pump

In Fig. 22 the speed increasing gear between a motor and boiler feed pump is a self-contained unit connected by flexible couplings to the motor and feed pump. A further development which would increase the advantages of using an epicyclic gear would be to incorporate the gear in an extension of the motor end cover, thus eliminating one of the flexible couplings.

380 h.p. 14,000/3,100 r.p.m. Marine Turbine Driven Boiler Feed Pump

Another example is illustrated in Fig. 23. In this instance an epicyclic gear fits in very well as a speed reduction unit between a steam turbine and boiler feed pump.

72 h.p. 16,000/775 r.p.m. Steam Driven Marine Circulating Water Pump

Fig. 24 shows the use of an epicyclic gear in an inclined application. The cross sectional arrangement of the gear which is of the compound planet type is illustrated in Fig. 7.

Small Main Propulsion Reverse Reduction Epicyclic Gears

Figs. 25(a) and (b) show two reversing epicyclic gears for installation in twin screw vessels powered by uni-directional and non-reversing oil engines. Each gear is designed to transmit 550 h.p. from 1,000 to 625 r.p.m. when going ahead and 400 h.p. from 1,000 to 500 r.p.m. astern. For the unit shown in (b) the input and output shafts rotate in the same direction, whereas in (a) opposite directions or rotation of these shafts are obtained, thus giving the condition required in twin-screw vessels.

Each of these gear units consists of two sections, a reversing unit and a reduction unit. The reversing unit gives a slight additional reduction when going astern. In (b), the second

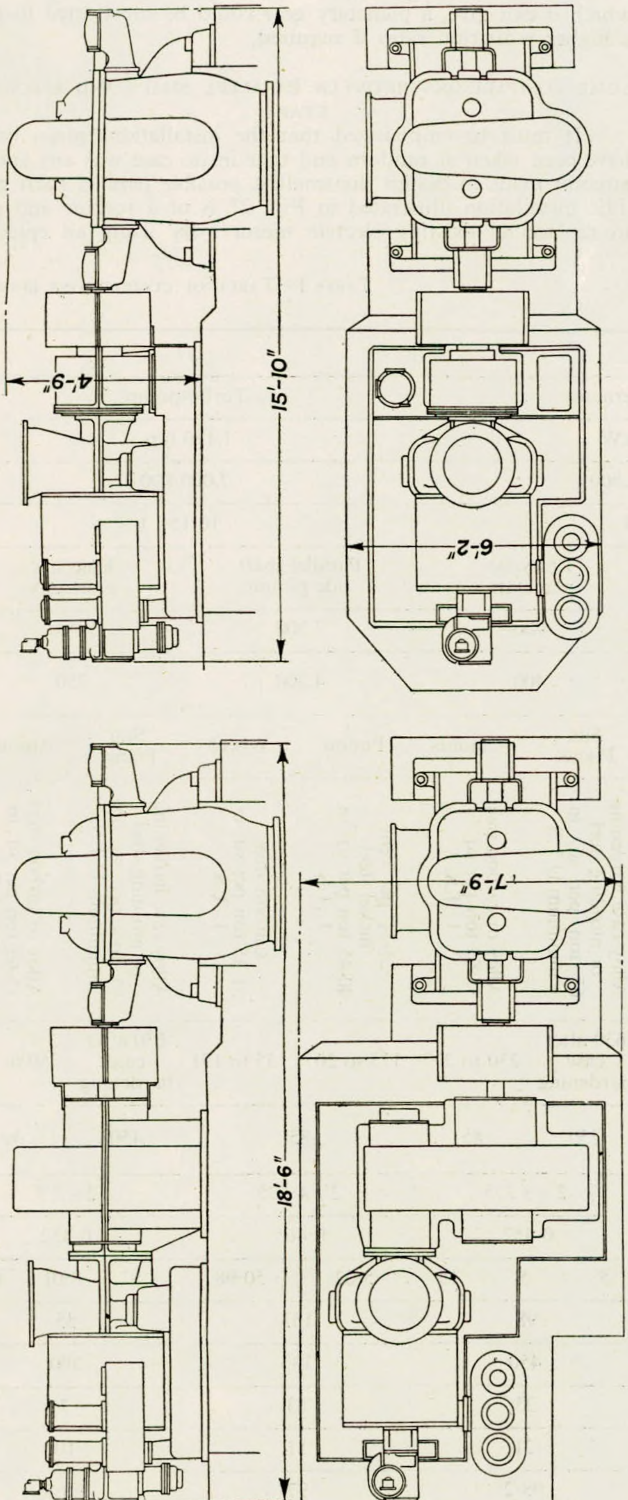


FIG. 27—1,180 h.p. turbo-pump: gear ratio 10·15:1

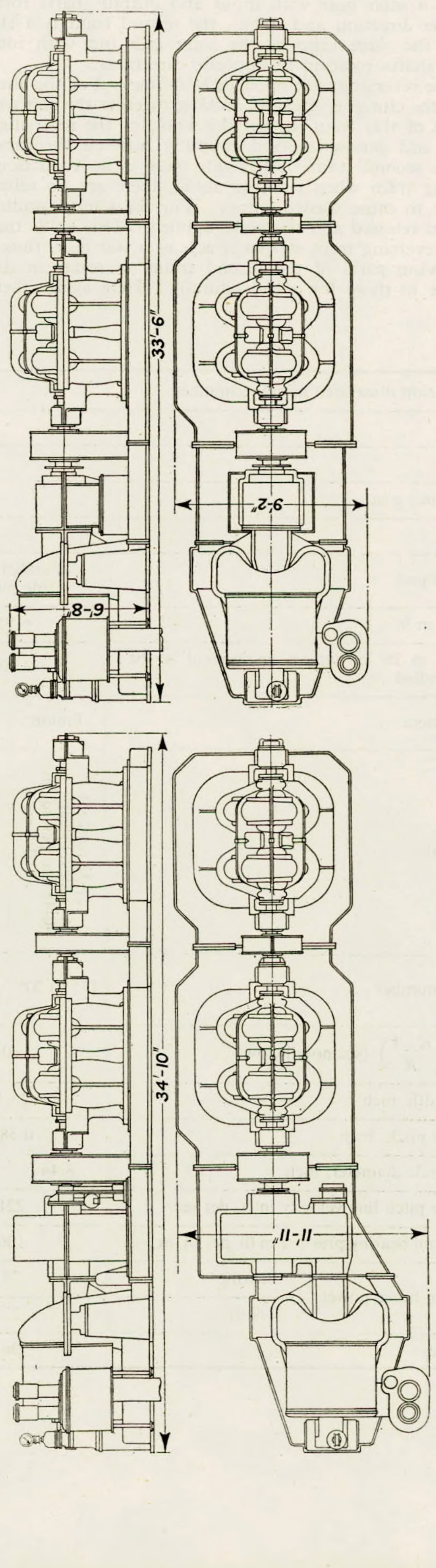


FIG. 28—4,360 h.p. turbo-pump: gear ratio 5·5:1

train is a solar gear with input and output shafts rotating in the same direction and in (a), the second train is a star gear, having the same ratio as the solar gear but with input and output shafts rotating in opposite directions.

The reversing unit operates as follows: For the ahead condition, the clutch is engaged, locking together the sun wheel and annulus of this train so that the whole of the reversing system rotates and acts as a solid coupling between the input shaft and the second train. Although there is no reduction in the reversing train when running ahead, there are no idling components to cause wasted energy. For the astern condition, the clutch is released and the brake applied. This holds the carrier of the reversing train so that it acts as a star gear, thus causing the moving parts of the second trains to rotate in directions opposite to those for ahead running. Here again, there is no waste of power since all the gear components are being usefully employed.

By altering the ratios of the final reduction trains, a wide range of propeller speeds for a common engine speed can be obtained and although the solar gear is restricted in the ratios which it can give, a planetary gear could be substituted to give a higher reduction ratio if required.

SOME COMPARISONS BETWEEN PARALLEL SHAFT AND EPICYCLIC GEARS

It must be emphasized that the installations given below have been taken at random and that in no case was any special attempt made to design the smallest possible parallel shaft gear. The installation illustrated in Fig. 27 is of a turbine and gear to replace an existing electric motor. By using an epicyclic

TABLE I—TABLE OF COMPARISONS BETWEEN

Application illustrated in figure number	26				27			
Duty	Turbo-alternator				Turbo-pump			
Power	2,000 kW.				1,180 h.p.			
Speeds in r.p.m.	6,000/1,500				7,000/690			
Ratio	4 : 1				10·15 : 1			
Type of gear	Parallel shaft side pinion		Epicyclic planetary		Parallel shaft side pinion		Epicyclic planetary	
Weight in lb.	6,500		900		7,300		1,900	
Weight in lb. of largest component to be handled	3,400		400		4,300		750	
Component	Pinion	Wheel	Sun Planet	Annulus	Pinion	Wheel	Sun Planet	Annulus
Materials	$2\frac{3}{4}$-$3\frac{3}{4}$ per cent nickel steel 40-45 ton per sq. in. U.T.S.	Carbon Steel 31-35 ton per sq. in. U.T.S.	Alloy case hardening or nitriding steel 65 ton per sq. in. minimum U.T.S.	Alloy or carbon steel 55-65 ton per sq. in. U.T.S.	$2\frac{3}{4}$-$3\frac{3}{4}$ per cent nickel steel 40-45 ton per sq. in. U.T.S.	Carbon steel 31-35 ton per sq. in. U.T.S.	Alloy case hardening or nitriding steel 65 ton per sq. in. minimum U.T.S.	Alloy or carbon steel 55-65 ton per sq. in. U.T.S.
Brinell number	175 to 205	135 to 151	630 after case hardening	250 to 295	175 to 205	135 to 151	630 after case hardening	250 to 295
$K=\frac{P}{d}\left(\frac{R\pm1}{R}\right)$ (see note below)	80		250	85	85		450	49
Face width, inch	2×8		2×3·375		2×4·625		2×2·5	
Circular pitch, inch	0·589		0·452		0·465		0·452	
Pitch circle diameter, inch	8·44	33·54	5 | 5	15	5·02	50·98	1·97 | 8·01	18·0
Relative pitch line velocity in ft. per sec.	221		98		153		55	
Maximum bearing pressure in lb. per sq. in.	106		450		133		300	
Losses in horse power: Bearing	78		23		23		7	
Losses in horse power: Tooth	27		21		11		10	
Efficiency	96		98·2		97		98·5	

$K=\frac{P}{d}\left(\frac{R+1}{R}\right)$ for two external wheels in contact

$K=\frac{P}{d}\left(\frac{R-1}{R}\right)$ for an external and internal wheel in contact gear, the motor foundations could be used but the use of a side or top pinion parallel shaft gear would have necessitated extensive modifications to the foundations.

All the other installations have given satisfactory service with parallel shaft gears but nevertheless the illustrations and the table of comparisons indicate clearly the advantages which would have been obtained by using epicyclic gears.

2,000 kW. Turbo Alternator 6,000/1,500 r.p.m.

Fig. 26 shows the arrangement of an industrial set as installed and also the arrangement which would be given by the use of an epicyclic gear.

1,180 h.p. Turbo Pump 7,000/690 r.p.m.

Fig. 27 illustrates an industrial turbine and epicyclic gear to replace an electric motor for a pumping installation. The extensive modifications which would have been required if a parallel shaft gear had been used are clearly indicated.

4,360 h.p. Turbo Pump 6,160/1,120 r.p.m.

Fig. 28 shows a large pumping installation.

1,060 kW. 6,000/600 r.p.m. Marine Turbo Generator

Fig. 29 illustrates a marine turbo generator. Also illustrated is the arrangement which would be given by the use of an epicyclic gear.

375 kW. 8,500/1,100 r.p.m. Marine Turbo Generator

Fig. 30 illustrates a marine turbo generator. In the same figure is shown the more compact arrangement which would be given if an epicyclic gear were used.

Table I gives the data for all these gears from which some of the advantages to be gained by the use of epicyclic gears

PARALLEL SHAFT AND EPICYCLIC GEARS

28				29				30			
Turbo-pump				Turbo-generator				Turbo-generator			
4,360 h.p.				1,060 kW.				375 kW.			
6,160/1,120				6,000/600				8,500/1,100			
5·5 : 1				10 : 1				7·73 : 1			
Parallel shaft side pinion		Epicyclic planetary		Parallel shaft top pinion		Epicyclic planetary		Parallel shaft side pinion		Epicyclic planetary	
12,900		2,650		9,500		2,650		2,600		560	
8,500		1,100		5,900		1,100		1,500		200	
Pinion	Wheel	Sun Planet	Annulus	Pinion	Wheel	Sun Planet	Annulus	Pinion	Wheel	Sun Planet	Annulus
2$\frac{3}{4}$-3$\frac{3}{4}$ per cent nickel steel 40-45 tons per sq. in. U.T.S.	Carbon steel 31-35 tons per sq. in. U.T.S.	Alloy case hardening or nitriding steel 65 tons per sq. in. minimum U.T.S.	Alloy or carbon steel 55-65 tons per sq. in. U.T.S.	2$\frac{3}{4}$-3$\frac{3}{4}$ per cent nickel steel 40-45 tons per sq. in. U.T.S.	Carbon steel 31-35 tons per sq. in. U.T.S.	Alloy case hardening or nitriding steel 65 tons per sq. in. minimum U.T.S.	Alloy or carbon steel 55-65 tons per sq. in. U.T.S.	2$\frac{3}{4}$-3$\frac{3}{4}$ per cent nickel steel 40-45 tons per sq. in. U.T.S.	Carbon steel 31-35 tons per sq. in. U.T.S.	Alloy case hardening or nitriding steel 65 tons per sq. in. minimum U.T.S.	Alloy or carbon steel 55-65 tons per sq. in. U.T.S.
175 to 205	135 to 151	630 after case hardening	250 to 295	175 to 205	135 to 151	630 after case hardening	250 to 295	175 to 205	135 to 151	630 after case hardening	250 to 295
74		370	82	76		370	41	89		400	60
2×8·0		2×3·25		2×6·5		2×3·25		2×3·75		2×2·0	
0·573		0·725		0·571		0·515		0·463		0·362	
9·85	54·18	4·44 7·78	20·0	5·82	58·16	2·33 9·33	21	3·69	28·32	1·49 4·25	10·0
265		95		152·5		55		137		48	
126		400		136		300		27		330	
93		31		34		9		24		4	
44		35		15		13		5·5		5	
96·9		98·5		96·8		98·5		94·5		98·4	

Where:—P = load in lb. per inch of face width
d = pitch circle diameter of smallest wheel in train
R = ratio of the speeds of the two wheels in contact

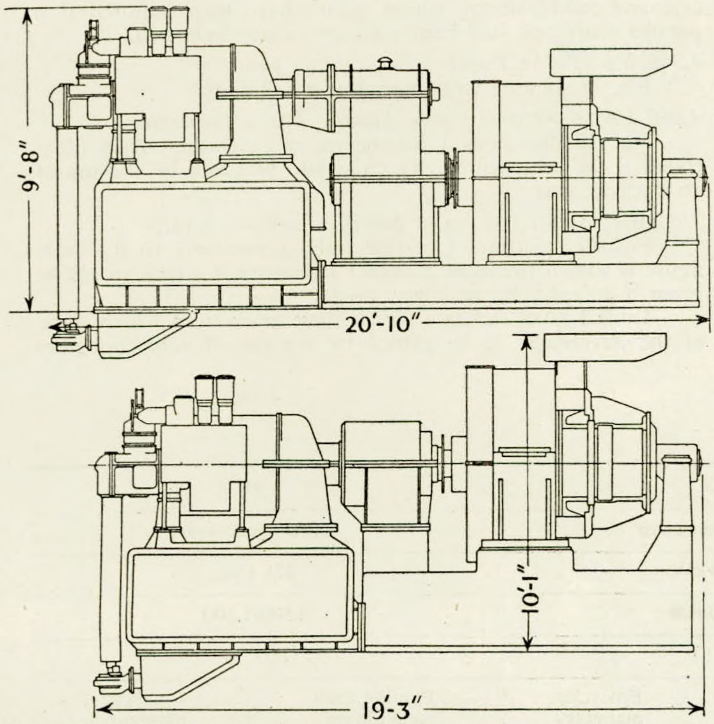


FIG. 29—*1,060 kW. marine turbo-generator: gear ratio 10:1*

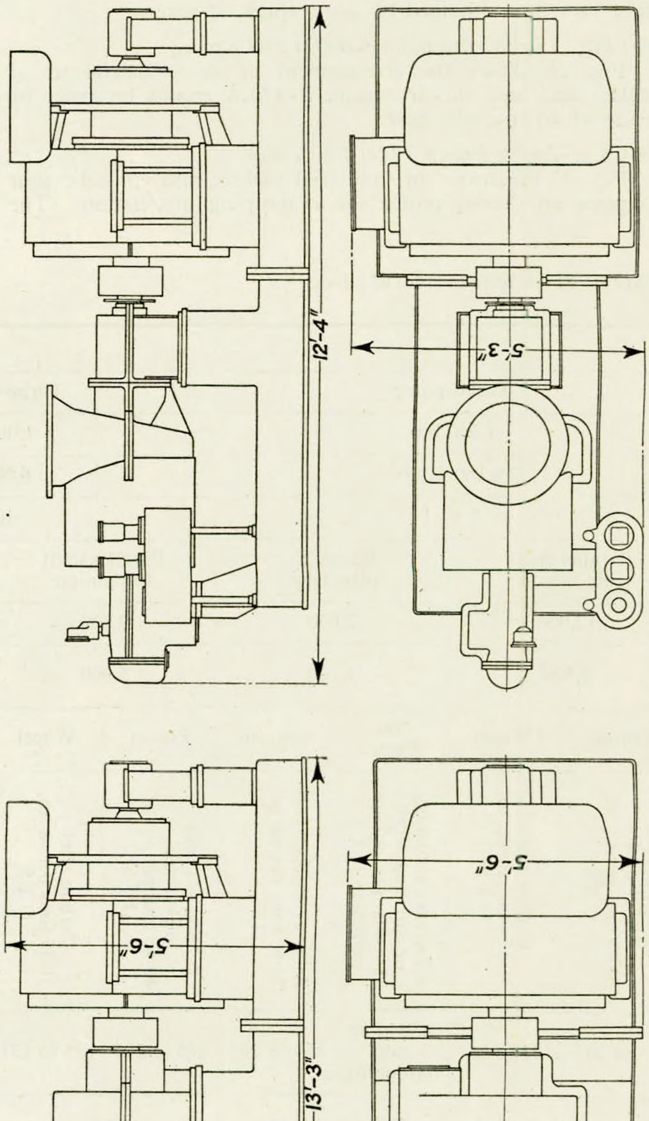


FIG. 30—*375 kW. marine turbo-generator: gear ratio 7·73:1*

are easily seen. These may be briefly summarized as follows:

(1) Smaller dimensions.
(2) Lower weight.
(3) Higher efficiency.
(4) Lower pitch line velocity.
(5) Co-axial arrangement of driving and driven shafts.

FUTURE TRENDS

So far previous history and present applications have been discussed, together with some of the fundamental problems associated with the design and production of epicyclic gears. It is reasonable, therefore, to examine in some detail the future which this form of gearing faces in the field of marine engineering.

Reciprocating engines for larger powers have generally been operated at speeds which allow direct drive to the propeller shaft, but for smaller powers, internal combustion engines run at speeds which demand the incorporation of gearing between the engine and propeller shaft. In particular, the uni-directional engine necessitates the use of a reversing mechanism and it is here that epicyclic gears offer an attractive solution. Whilst epicyclic gears have been employed regularly for small craft, there has been a reluctance to use them for higher powers. The tendency to build oil engines developing up to 2,000 h.p. at speeds well in excess of propeller speed will bring to the fore the use of reversing epicyclic gears.

The increasing use of high speed, high efficiency steam turbines and the continual development of gas turbines for main propulsion units demand an efficient and reliable form of reverse reduction gear. An epicyclic gear would assist greatly towards achieving the maximum possible efficiency in such installations. For example, in a steam turbine running at 4,500 r.p.m. and developing 10,000 s.h.p., there is a constant loss when going ahead of some 200 h.p. due to astern turbine windage. A double reduction gear for this power would involve the loss of a further 300 h.p. A reverse reduction epicyclic gear could limit the total loss to about 200 h.p.

A gear suitable for 10,000 s.h.p., reducing from 4,500 to 110 r.p.m. ahead and 6,700 s.h.p. reducing from 4,500 to 89 r.p.m. astern is illustrated in Fig. 31. In this arrangement, both when going ahead and astern, all the components are doing useful work. The gear comprises three epicyclic trains; one double helical reduction train followed by two spur trains, which incorporate the reversing features. The double helical planetary train gives the initial reduction ahead and astern and the output shaft of this train is connected to the sun wheel of the second train.

When going ahead the brake drum of the third train is held stationary, when the second train operates as a planetary unit and the third as a solar. When going astern the brake of the third train is released and that of the second train applied, causing the second train to operate as a star gear and the third as a planetary. The difference in rotation is obtained, therefore, by changing the character of the second train.

Designed on a very conservative rating, the total weight of this reverse reduction gear, including a hydraulic braking system and control equipment, Michell thrust block and gearcase, is estimated to be about 35 tons and the space required is approximately 12 feet long by 9 feet diameter.

A Pametrada proposal which was illustrated in T. W. F. Brown's paper(1) already mentioned, is reproduced in Fig. 32 by permission of the North-East Coast Institution of Engineers and Shipbuilders. This gear is also for 10,000 s.h.p. 4,500 to 120 r.p.m. ahead and to 110 r.p.m. astern. The weight is estimated to be about 20 tons, including the hydraulic brakes but excluding the thrust block.

The reversing system which is incorporated in the first train comprises a driving sun wheel meshing with three compound planet wheels, the second parts of which mesh with another set of planet wheels mounted in the same planet carrier. These last wheels engage with the second of two annuli, each of which is connected to a brake drum. The planet carrier of this train is the driven member for both running conditions and drives the sun wheel of the final reduction train, which is a planetary gear with compound planets. When going ahead the ahead brake is applied, which makes the first part of the compound planet train a simple planetary system. The single planets and second annulus with its brake drum are allowed to idle, whilst the planet carrier drives the sun wheel of the second train, which in turn drives the planet carrier of the reduction train in the desired direction. For the astern condition the ahead brake is released and the astern brake applied. The second part of the compound planet train now operates as a planetary unit, the single planets acting as idler wheels to achieve the reverse rotation whilst the first part of the compound planets, the first annulus and its brake drum idle.

These examples are but two of the many configurations of epicyclic gears which could be used to satisfy reverse reduction requirements for single or tandem turbines but by them typical principles of operation are well illustrated. The many mechanical problems in reversing gears of such powers would need considerable development but there is every reason to believe that they could be solved.

Another manner in which the "in line" property of epicyclic gears could be utilized is by installing the final stage of a propulsion gear as a self-contained epicyclic unit in the shaft tunnel adjacent to the stern gland. By this means the intermediate shafting could be run at a speed considerably higher than that of the propeller, thus allowing a reduction of intermediate shaft diameter and weight.

There may well be cases where a very attractive overall arrangement of propulsion machinery could be achieved by the combined use of epicyclic and parallel shaft gears. The epicyclic gear could be used either as a first or second reduction.

In many cases it is desirable to reduce the weight and space taken up by machinery and this can be assisted in many instances by the incorporation of epicyclic gears either as speed reducing or increasing units.

Examples have been given of the ways in which epicyclic gears are being used to transmit power to such auxiliaries as generators and pumps. Other possible applications include winches, forced or induced draught fans and blowers, refrigerating gas compressors, oil engine superchargers and turning gear for main engines.

Consideration could also be given to a further standardization of motor sizes and speeds with the help of compact speed increasing or reducing gears to meet the wide range of auxiliary speeds needed. This would result in a substantial saving in electrical spares.

CONCLUSION

The authors realize that epicyclic gears are not necessarily the most suitable for all gearing applications and each case should be carefully considered so that the best overall arrangement can be achieved. This may come about by the use of epicyclic or parallel shaft gears or a combination of both.

Very striking developments have taken place in all forms

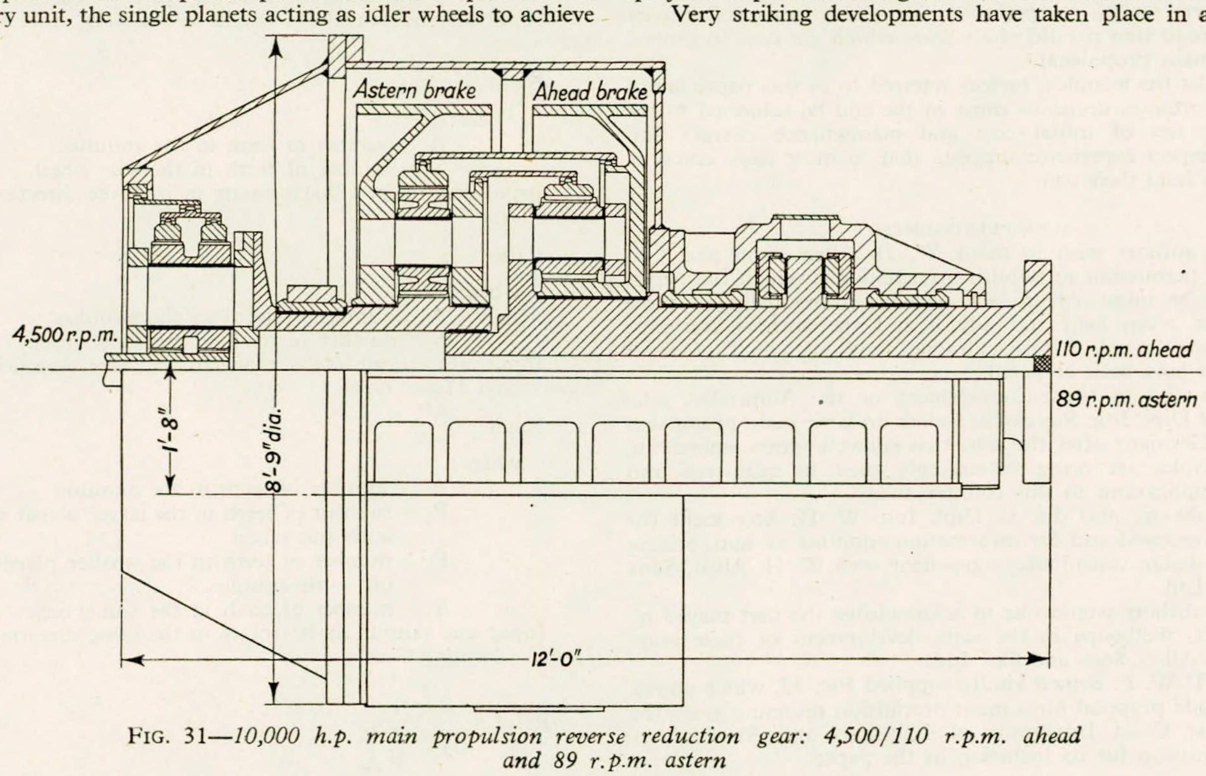


FIG. 31—*10,000 h.p. main propulsion reverse reduction gear: 4,500/110 r.p.m. ahead and 89 r.p.m. astern*

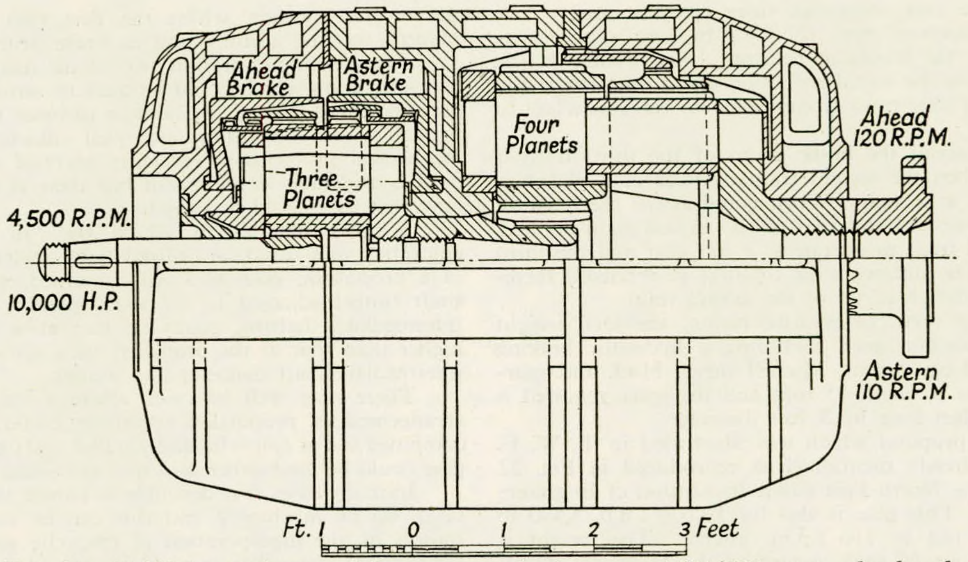


FIG. 32—*10,000 h.p. main propulsion reverse reduction gear: 4,500/120 r.p.m. ahead and 110 r.p. m. astern*

of gearing due to the scientific approach which has been made to the design and manufacturing techniques involved. Some of this work is recorded in the references given in Appendix II. The authors feel that too little attention has been given to the effects of applying these developments to the epicyclic forms of gearing and an attempt has been made to remedy this omission by bringing to the notice of marine engineers some of the possibilities of present-day and future epicyclic gears.

Successful experience shows that such gearing satisfies the marine engineer's first and foremost requirement of reliability. Its compactness alone offers great advantages to be gained by its adoption in a wider variety of applications, as speed reducing or increasing gears, for higher powers than have so far received consideration.

It is the conviction of the authors that with determined development large powered reversing gears will in the future become very serious competitors to the much larger and heavier straight reduction parallel shaft gears which are now in general use for main propulsion.

Whilst the technical factors referred to in this paper are of vital importance, decisions must in the end be subjected to the economic test of initial cost and maintenance charges and in this respect experience suggests that in most cases epicyclic gears can hold their own.

ACKNOWLEDGEMENTS

The authors wish to thank W. H. Allen, Sons and Co., Ltd., for permission to publish information given in the paper. It would be impossible to acknowledge individually all those who have given help and advice and the authors are most grateful for the aid of their colleagues, without which the paper could not have been prepared.

It is due to the encouragement of the Admiralty, who learned of Dipl. Ing. Stoeckicht's work on large main propulsion gears in Germany after the war, that epicyclic gears embodying his principles are being increasingly used in industrial and marine applications in this country today.

Thanks are also due to Dipl. Ing. W. G. Stoeckicht for guidance received and for information supplied by him, arising from his design consultancy agreement with W. H. Allen, Sons and Co., Ltd.

The authors would like to acknowledge the part played by Mr. D. B. Welbourn in the early development of these gears by W. H. Allen, Sons and Co., Ltd.

Dr. T. W. F. Brown kindly supplied Fig. 32, which shows a Pametrada proposal for a main propulsion reversing gear; the North-East Coast Institution of Engineers and Shipbuilders gave permission for its inclusion in the paper.

Thanks are due to the Bristol Aeroplane Company for information about the Bristol Theseus gear and for supplying Fig. 3.

The photographs shown in Fig. 2 were supplied by courtesy of Hamburger Turbinenfabrik G.m.b.H., who built the turbines. They illustrate B.H.S.-Stoeckicht gears manufactured by Bayerische Berg. Hütten und Salzwerke A.G., Sonthofen.

APPENDIX I

FORMULÆ FOR RATIOS OF THE GEARS REFERRED TO IN THE PAPER

Planetary Gear

$$\text{Ratio} = \frac{A}{S} + 1$$

where:

A = number of teeth in the annulus.
S = number of teeth in the sun wheel.

Input and output shafts rotate in the same direction.

Star Gear

$$\text{Ratio} = \frac{A}{S}$$

where:

A = number of teeth in the annulus.
S = number of teeth in the sun wheel.

Input and output shafts rotate in opposite directions.

Solar Gear

$$\text{Ratio} = \frac{S}{A} + 1$$

where:

A = number of teeth in the annulus.
S = number of teeth in the sun wheel.

Input and output shafts rotate in the same direction.

Compound Planet Gear

$$\text{Ratio} = \frac{AP_1}{SP_2} + 1$$

where:

A = number of teeth in the annulus.
P_1 = number of teeth in the larger planet meshing with sun wheel.
P_2 = number of teeth in the smaller planet meshing with annulus.
S = number of teeth in the sun wheel.

Input and output shafts rotate in the same direction.

Double Annulus Gear

$$\text{Ratio} = \frac{1 + \dfrac{A_1}{S}}{1 - \dfrac{A_1P_2}{P_1A_2}}$$

where:

A_1 = number of teeth in the fixed annulus.
S = number of teeth in the sun wheel.
P_1 = number of teeth in the planet meshing with A_1 and S.
A_2 = number of teeth in the rotating annulus.
P_2 = number of teeth in the planet meshing with A_2.

Input and output shafts rotate in the same direction when A_2 is smaller than A_1 and in opposite directions when A_2 is larger than A_1.

APPENDIX II

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Discussion

Dr. G. H. Forsyth, M.B.E. (Member) said he considered it an honour to be asked to open the discussion on this paper, because it was of importance to the engineering industry in general at this stage of development of transmission gears. It was interesting to note on page 79, that the authors quoted a statement by the Lanchesters to the effect that the success of any mechanism, however well conceived, depended finally upon its correctness in detail. This, surely, was applicable to all problems in transmission involving high duty gears and, in particular, it was applicable to epicyclic gears.

A statement was made on page 83 to the effect that "the surface stress between internal and external teeth running together is less than between two external teeth because the relative radius of curvature, which is inversely proportional to the stress, is so much increased". Further, "for this reason it is usually found unnecessary to case harden the annuli used for epicyclic gears, even when the sun and planet wheels are case hardened". This statement required qualifying, for two reasons:—

1. The stress, whether it be taken as compressive or shear, was inversely proportional to the *square root* of the relative radius of curvature; the effect was therefore less than if it were simply inversely proportional. Fig. 33 made this point clear and indicated that the load carrying was proportional to the relative radius of curvature at the point of contact.

2. Fig. 34 indicated the shear stress for an actual gear design, at points distant from the surface, i.e. at the line of contact, both for the sun-planet contact, and for the planet-annulus contact. It was true, as the

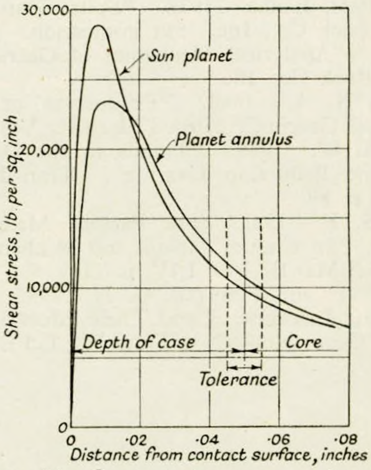
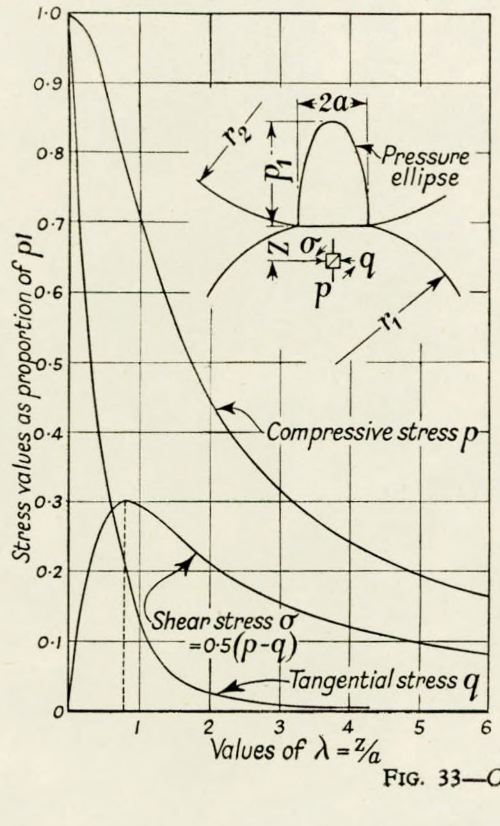


Fig. 34—*Contact stresses: sun planet; planet annulus*



Relative radius of curvature at line of contact, $R = \dfrac{r_1 r_2}{r_1 + r_2}$

Load per inch at line of contact = W in lb.

Compressive stress $p = \cdot 418\sqrt{\dfrac{WE}{R}}$ lb. per sq. in.

Half width of contact $a = \dfrac{0\cdot 2775}{1000}\sqrt{W.R.}$ in.

For the same material and same value of stress $W = k\ R$

This leads to the well known formula used by Parsons where

$P = k \times d$

where P = load per inch face width

d = *p.c.d.* and approximately $\propto$ to the relative radius of curvature for the gear ratios commonly used in marine gears.

Fig. 33—*Contact stresses on gear teeth*

authors pointed out, that the shear stress was greater for the sun-planet contact but the shear stress at the annulus contact was only slightly less. In this case, the sun and planet wheels were made of V.18A steel with a core strength of 65-80 tons per sq. in., case hardened, and the annulus gear from V11.T8 oil hardened and tempered, U.T.S. 55 tons per sq. in.

Perhaps the authors would give their opinion as to how the factor of safety should be determined at each contact. Since the annulus had a reduced U.T.S. as compared with the planet and, further, since it was not carburized and case hardened, the factor of safety could not be high. To date, after full load running, the annulus gear had shown no signs of pitting, but so far as factors of safety were concerned, based on the Hertz equation, some advantage might accrue by the fitting of case hardened annulus gears in high duty epicyclic boxes. There would, however, be a real problem in quenching an annulus gear of sizes useful in marine gears.

He thought the statement on page 84 should read:—

$$\left[\frac{1}{\sqrt{\text{ratio}-1}} \times \text{(surface stress on sun wheel) for a planetary gear}\right]$$

Further, this was only an approximate formula and if applied to the gear to which Fig. 34 referred, would give a stress of 18,000lb. per. sq. in., instead of 23,300lb. per sq. in. when the Hertz stress was correctly derived. Probably Fig. 34 indicated generally the ratio of sun-planet to planet-annulus surface stress conditions. Thus, if the material of the annulus gear were, say, 55 tons per sq. in., and not case hardened, it seemed that the factor of safety for annulus gears would be considerably less than for sun-planet contacts. There was much yet to be learned about the various factors affecting pitting and especially in relation to the material used, conditions of lubrication, etc., and it seemed that practical testing only would give the information required by engineers.

Fig. 35 showed the variation of bending stress and Hertz compressive stress with the value of the normal pitch chosen. It would be noted that as the normal pitch increased, the bending stress decreased. The surface stress, however, varied due to changes in the length of the line of contact, this being a function of the overlap ratio. With a face width between one and two axial pitches this variation was reasonable. The value of the normal pitch chosen for this particular gear was 0·8 inch, giving a stress value of about 18,000lb. per sq. in. The steel in question was V.18A carburized and case hardened. Details

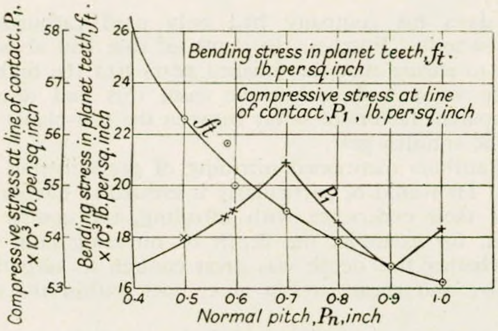


FIG. 35—*Stress at the planet-annulus contact*

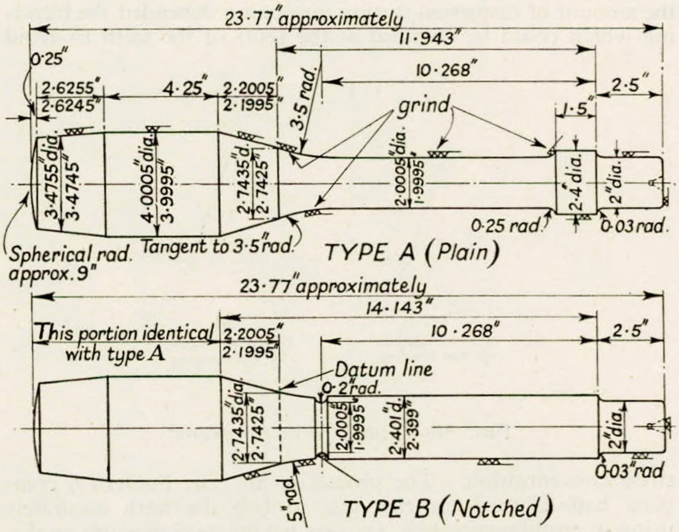
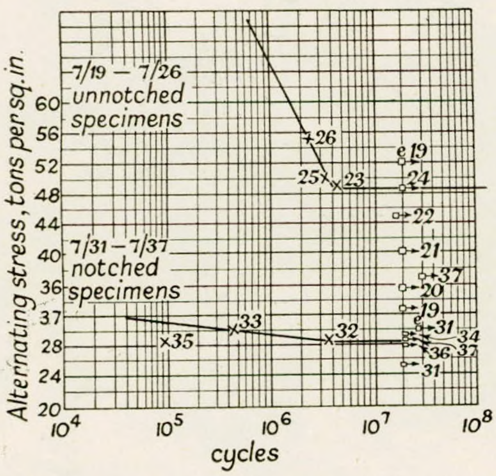


FIG. 36—*Specimens as used in 2-in. Wohler fatigue testing machine*



Remarks × Specimen fractured
×→ Specimen broke in chuck
□→ Specimen not fractured
e□→ Specimen not fractured after load increase

Test no. series: 11
Type of machine: ½ in. Wohler. Machine No. 3. Speed 2,950 c.p.m.
Dimensions at test section: 0·5 in. diameter.
Particulars of notch: Semi-circular 0·05 in. radius
Finish on specimen: Ground

Material	E.N.36 3 per cent Ni, Cr		Test specimen 7A/3	Test specimen 7B/3
Condition	Carburized to give case depth of 0·05 to 0·06 in.	Yield point T.S.I.	53·2	55·6
Analysis	0·11C, 0·43Mn, 0·17Si, 0·026S, 0·022P, 3·13Ni, 0·88Ct	U.T.S.	67·2	70·8
H.T. particulars	Carburize for 20 hours at 900 deg. C.	Elongation on 2 in.	16	12
		Reduction of area, per cent	54·8	54·8
		Hardness Brinell	311	321
		Izod ft./lb.	40, 40, 41	40, 37, 38

FIG. 37—*Fatigue test results*

of fatigue specimens and the fatigue strength of this material were given in Figs. 36 and 37.

Taking the notched specimen as a basis, these indicated a factor of safety of 3·5; with good root blending the factor should be greater than this, the stress concentration being 1·71 for the notched specimen. The bending stress in the planet teeth was, of course, reversed, due to the direction of loading of the two contacts.

Perhaps the authors would like to comment on the value of the bending stress selected in the design of this gear, bearing in mind the importance of guarding at all costs against failure due to bending stress in the teeth.

The authors rightly pointed out on page 84 that the epicyclic gear lends itself to small parts for case hardening and there was thus less risk of serious distortion during quenching. In this connexion it might be mentioned that a planet gear of the form shown in Fig. 10(a) of the paper was much more difficult to quench than that shown in Fig. 10(e). Upon the amount of distortion during quenching depended the blending which could be obtained at the roots of the teeth to avoid

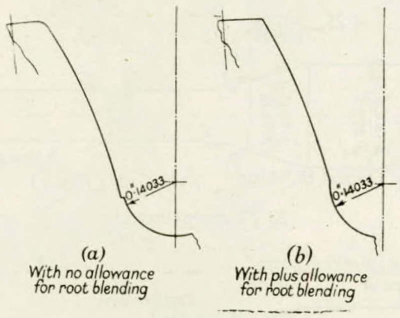


FIG. 38—*Tooth form for planet*

stress concentration. The procedure his (Dr. Forsyth's) company had adopted for this was to hob the teeth accurately using a protuberance hob, i.e. one having two pressure angles, to cross polish or grind the tooth roots after carburizing but before quenching, so as to avoid, as far as possible, grinding cracks and in any case to reveal them should they occur and to design the protuberance hob in such a way that a plus rather than a negative allowance would be left on the finished teeth due to the combined effect of circular pitch errors in the hobbing machine and distortion whilst quenching.

Fig. 38 indicated clearly what could happen if the procedure already outlined were not followed and, clearly, condition (b) was preferable to condition (a).

Fig. 39 indicated an actual case in practice where condition (b) was obtained.

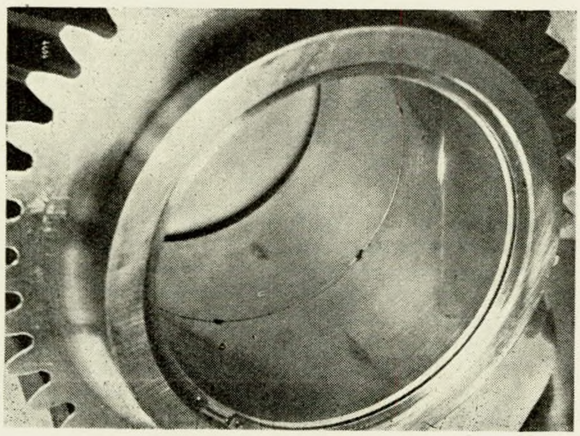


FIG. 39—*Planet wheel, showing correct root blending*

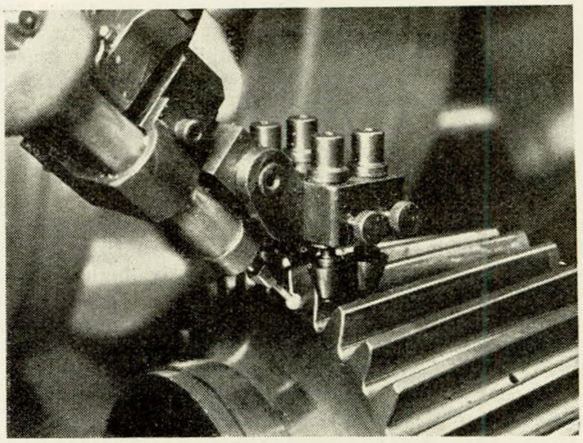


FIG. 40—*Cross grinding tooth roots*

Fig. 40 indicated a machine used for cross grinding the roots of the teeth. The operation was carried out after carburizing but before quenching, as already mentioned, to improve tooth blending and to raise the fatigue strength. As case hardened gears increased in size, it became almost impossible, due to distortion during quenching, to get correct root blending.

To date, his company had only used carburized and hardened wheels. Further, the depth of case had always been sufficient to ensure that the finished gears had the high values of surface stress well within the case; this had resulted in higher apparent factors of safety between the sun-planet contact than at the annulus gear.

The authors mentioned nitriding of gears instead of carburizing. He would be particularly interested in having further details of their experience with nitriding, the quality of the steel used, for example, the depth of nitriding and, in particular, whether this depth was great enough to keep the high compressive and shear stresses at contact within the nitrided depth.

The authors stated on page 83 that epicyclic gears might

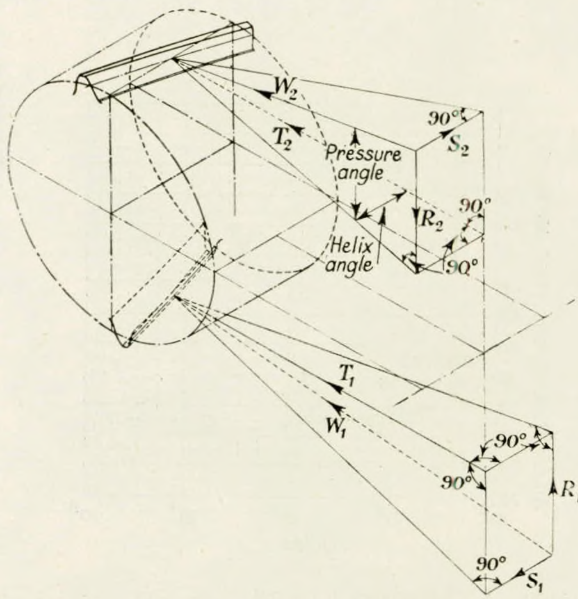


FIG. 41—*Tooth loads on single helical planet*

$T_1 = T_2$ Tangential load
$W_1 = W_2$ Load normal to tooth
$R_1 = R_2$ Radial component load
$S_1 = S_2$ Axial component load

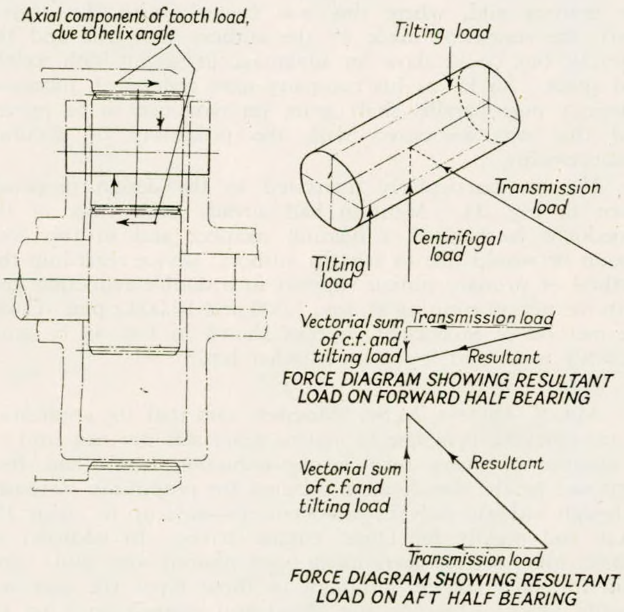


FIG. 42—*Bearing loads on single helical planet bearing pin*

be made with straight spur, single helical or double helical teeth, but the design of the planet bearings with single helical teeth would need special consideration. This reference to single helical gearing was very true because of the fore and aft tilting action on the planet wheels at the annulus and sun contacts caused by forces in opposite directions.

Figs. 41 and 42 indicated clearly the complex system of forces acting. To add to this difficulty of high loading, if the bearing rotated with the planet wheel it was subject, as the authors pointed out, to a fatigue form of loading. For this reason it was considered that where high specific loading was involved of the order of 800lb. per sq. in., it was better to use copper lead bearings. His company had carried out tests with this magnitude of loading with such bearings where the calculated oil film thickness was only about 0·5 thousandths of an inch. Probably the best solution was to use the white metalled spindle type of bearing and reduce loads to about 500lb. per sq. in. His company were at present carrying out back to back tests with such an arrangement. With extremely high duty bearings of this type, the authors correctly pointed out the need for good filtering of oil. This could be achieved with felt type filters.

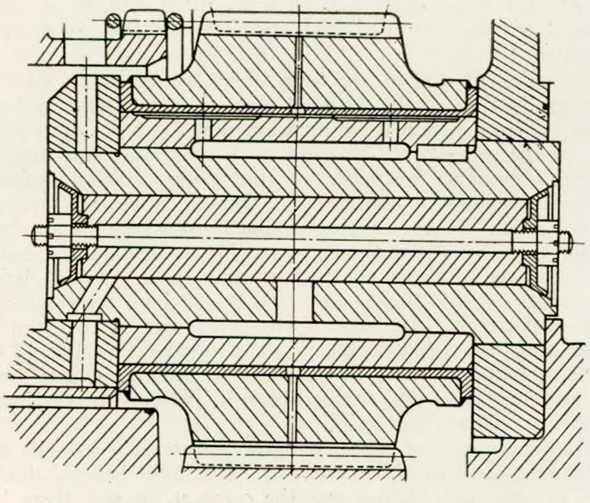


FIG. 43—*Original copper lead bushes which failed*

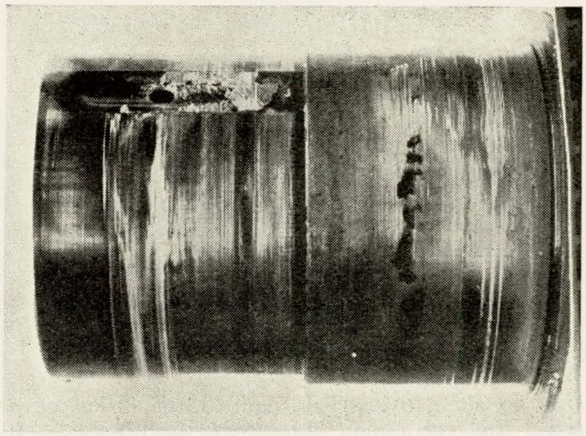


FIG. 44—*Copper lead bush which failed*

Regarding the fitting of bushes, reference Fig. 10(a), it was true, as the authors mentioned, that their fitting was not an easy matter. His company had tried making these bushes to very close limits so that, after pressing into the wheel, both the requisite fit between the bush and the wheel and the required dimensions of the bore of the bush were obtained without the need of finish machining of the bush in position. Unfortunately, the bushes moved axially. Fig. 43 showed this arrangement.

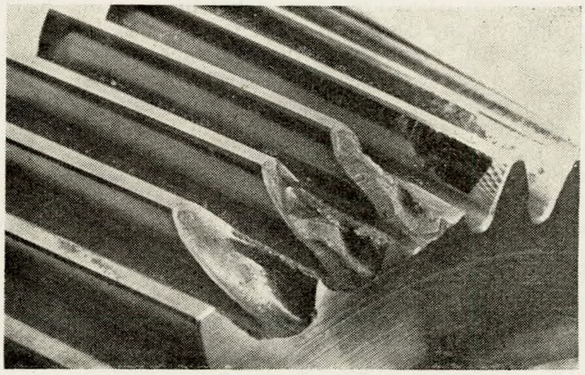


FIG. 45—*Fatigue fracture of planet wheel teeth*

The axial movement of these bushes, which were spot welded to the wheel, caused overheating and these being of steel with copper lead lining, brazing took place and the whole bearing was wrecked. This was shown in Fig. 44.

The planet wheel to which this bush was fitted showed typical fatigue fracture, as indicated in Fig. 45.

Fig. 46 showed the modified arrangement using a spring ring to prevent axial movement of the bearing bush. So far, this had given satisfactory service.

Fig. 47 showed white metalled spindles which were being

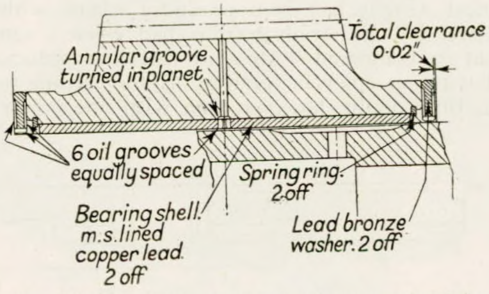


FIG. 46—*Modified arrangement of copper lead bush*

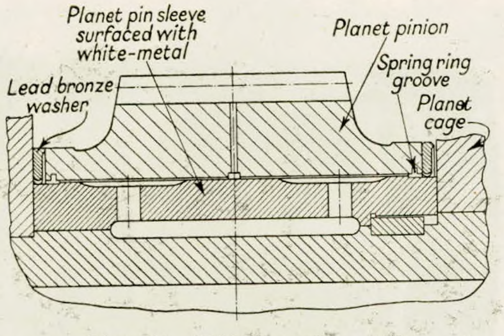


FIG. 47—*White metal bearing*

tried as an alternative. This omitted the fitting of bushes altogether and the pinion bores were superfinished. In the event of failure they would be presented with a bad soldering job rather than with a good brazing job. This reference to brazing and soldering was really an important point because, without any doubt, white metal bearings which moved integrally with the planet pins were a better job. In this connexion it was interesting and heartening to note that the authors stated that loads up to 750lb. per sq. in. had proved satisfactory with this arrangement.

On page 84, the authors mentioned the importance of equality of load sharing between the various planet wheels. Mention was also made of the floating member as put forward by Stoeckicht for achieving this balance. This was easy with the three-planet wheel arrangement but his company had been responsible for the design of epicyclic gears having five-planet wheels. It had been found that machine tools today were sufficiently accurate to get equal load distribution with the Stoeckicht principle, i.e. a pinion without bearings, even with gears of considerable size. The limits of accuracy required to balance the load on five-planet wheels were derived in the first place by calculation and proved in a similar manner to that mentioned by the authors, i.e. by fixing strain gauges round the annulus gears at five equally pitched points and measuring the straining of the annulus gear. Fig. 48 indicated the results obtained.

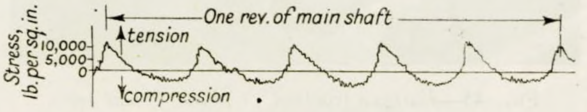


FIG. 48—*Stress value at outer surface of annulus gear ring*

The authors pointed out that some of the principal advantages of epicyclic gears were the saving of weight and space and high efficiency as compared with parallel shaft gears. As regards saving of weight and space, it should be pointed out that this was not always the case and depended entirely on the machinery arrangement. For example, if more than two power turbines were to be geared in and where turbine centre lines could not be coaxial with the low speed shaft, there was little advantage in using an epicyclic gear. In fact, two identical designs had been produced where, with a two-turbine drive, a locked train gearbox had given a similar size and weight as compared with a normal first reduction gear combined with an epicyclic gearbox as the second reduction. There was little doubt, however, that with one power feed to

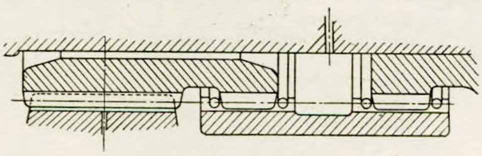


FIG. 49—*Stoeckicht method of flexible pinion support*

the gearbox and, where this was coaxial, with the output shaft, the statement made by the authors was true and the epicyclic box could show an advantage in saving both weight and space. So far as his company were concerned, increased efficiency over parallel shaft gears yet remained to be proved and this was associated with the possibility of accurate measurements.

He was particularly interested in the design proposals given in Fig. 31. Mention had already been made of the Stoeckicht method of a floating member and in this connexion he would like to ask the authors' advice regarding this method of primary pinion support in a double reduction gear with the pinion running at, say, 5,000 and 10,000 r.p.m. Could the method of Stoeckicht support shown in Fig. 49 be satisfactorily used, and if so, up to what h.p.?

MR. S. ARCHER, M.Sc. (Member) said that the application of the epicyclic principle to marine gears was not new and for a number of years now reverse-reduction gears had been approved by the classification societies for propulsion purposes, although hitherto only in small powers—say, up to about 150 s.h.p. and mainly for Diesel engine drives. In addition, of course, small reverse gears using bevel pinions were quite common for such service, although in those types the gear was usually locked when driving ahead and worked on what the authors termed the "star" gear principle when reversing; that was to say, it was not a true epicyclic gear.

The authors had presented in their paper a very fine survey of the past, present and possible future development of this interesting type of gear; as they had so graphically illustrated, it was a remarkably versatile mechanism with an almost unlimited range of reduction ratios (apart from the "blind" spots imposed by considerations of minimum numbers of teeth).

He wished to congratulate the authors on their careful choice of nomenclature for the various types of epicyclic arrangement described in their paper and he hoped those terms might eventually be adopted by engineers generally, as hitherto there did not seem to have been any consistency in that respect.

It would appear from a study of the last part of the paper that a good many of the designs illustrated were either projected or in course of construction. It would perhaps be helpful from a marine engineer's point of view if the authors could state how many of their gear units were at present in actual service at sea, with, if possible, a note of the maximum shaft horsepower so far transmitted in propulsion and auxiliary applications respectively and the corresponding periods of service.

Turning to details of design, it was understood that the 5,000 s.h.p. submarine gear designed by Stoeckicht had a K factor as high as 648 with 6 d.p. teeth. Could the authors state whether those gears were actually fitted and, if so, what length of service they had had and whether they were fully successful?

An excellent idea of the high degree of accuracy attained in the production of the authors' gears was presented in Figs. 14 and 15, from which it would seem that tooth-to-tooth pitch errors of the order of $\pm$ 2/10,000 inch maximum and usually less were obtainable for hardened and ground sun and planets with profile errors of about $\pm$ 1/10,000 inch clear of the tip and root relief. It was concluded that those were representative and not specially selected examples. The corresponding errors shown for the internal toothed annulus were greater, being of the order of $\pm$ 5/10,000 and $\pm$ 3/10,000 inch for pitch and profile respectively. In view of the possibility of the annulus ring tending to act like a bell, might it not be desirable, in the interest of reducing noise to the limit, to attempt to cut down those annulus errors still further, even if, perhaps, it was found necessary to harden and grind?

On the subject of noise, had the authors carried out any comparative tests to show whether the single helical, double helical or spur gear design was the quietest, or was there any other evidence on this point?

It had also been suggested that there was a limit of size above which the double helical gear became more attractive than the single helical. Could the authors comment on that for epicyclic gears?

The two points in favour of the epicyclic gear from the noise point of view were, of course (1) small components capable of manufacture to much closer limits than those of conventional gears; and (2) smaller inertia loading on the teeth as a result of point (1) in conjunction with the generally much lower relative pitch line velocities, as shown in Table I.

As regards service in the driving of marine auxiliary machinery, it would seem that the modern epicyclic gear should be fully capable of holding its own with the conventional parallel shaft gear as to reliability, and its advantages in other directions had been sufficiently pointed out in the paper.

For propulsion gears of sizeable dimensions, however, as the authors had pointed out, there was need for considerable development work before it might be possible to "fit and forget" and "confine dismantling to the minimum required by statutory regulations", to quote the authors' words. In that respect, the question of adequate accessibility would have to be borne carefully in mind. It must first be established that such gears were capable of standing up to fairly severe punishment in the form of shock loading, dirty or water-contaminated oil, and so on. When it was remembered that, in a turbine-driven ship pitching heavily in lightly ballasted condition, it was possible (on account of the high turbine inertia) for the gearing to be subjected over long periods to periodic repetitions of loading of range approaching full load propeller torque, perhaps several times a minute, it would be appreciated how much more onerous were those conditions than for auxiliary drive. In the testing of prototype propulsion gears such as those depicted in Figs. 31 and 32, therefore, he would put in a plea for serious consideration of experimental means to simulate seagoing conditions; for example, by periodic variation of brake torque.

Another problem was undoubtedly the question of thermal distortion of the gears, or, at the worst, the possibility of oil fires in the gearbox during manœuvring, on account of the large amount of kinetic energy to be converted into heat on the brake drums when going from full ahead to full astern as, for example, in a crash manœuvre. Incidentally, as drawn in Fig. 31, it would appear that in the third train the sun pinion could not be assembled, and it was concluded that the planet carrier disc would be made detachable from the shaft.

The authors' suggestion to fit the gearing near the tailshaft and thus save weight in line shafting was attractive, provided the need for keeping thrust block and gearing under close supervision could be dispensed with; alternatively, it might be necessary for an engineer to be stationed continuously at the gears.

One advantage of coaxial gearing not mentioned by the authors was the possibility of using the gearing as a torsionmeter, for example, by making the casing float on cylindrical roller tracks and weighing the torque reaction by suitable means. That might conveniently be done by hydraulic cylinders (an example was seen in the aircraft gear, Fig. 3) in which case the opportunity could perhaps be taken to arrange for those, or other cylinders, to act as dashpots to relieve the gearing of shock loading.

Another possibility in oil engine drives would be to utilize the motion of the casing under torque reaction relative to the hull structure to provide damping, frictional or otherwise. For example, by using leaf springs having a non-linear torque-deflexion characteristic, it should be possible to effect considerable damping of torsional vibration.

Mr. S. A. Couling said that the authors' company were to be congratulated on the extent to which they had developed the use of epicyclic gearing. One might, perhaps, criticize the paper for not emphasizing sufficiently the importance of the Stoeckicht patent because without it, or its equivalent, ensuring equal division of load on the various trains of wheel teeth, one would hardly dare to suggest epicyclic gearing for marine application.

In their conclusion, the authors said that "epicyclic gears are not necessarily the most suitable for all gearing applications". This was very true, especially on *Queen Mary* or *Queen Elizabeth* sets, where several turbines were driving into one main wheel or even where several Diesel engines were driving into one propeller shaft gear wheel.

Where "in-line" drives had proved successful, the conventional way of designing gears had been applied successfully. He was thinking of the locked train or twin drive where there was "in-line" drive and also gear motor units. His company made something like a hundred to two hundred sets a month of "in-line" drives with conventional design with efficiencies of 98 per cent.

The comparisons in Table I appeared to him to be biased and most misleading. For example, a side-by-side gearbox with K values comparable with the epicyclic gear and of the same materials as the epicyclic gear, was easy to conceive; he was certain that the conventional design of gearing would pay dividends. He had chosen the 375 kW. set shown in the last column as an example; that could be designed with 12-in. centres with a maximum total weight of about 1,000lb. and an efficiency of 98 per cent, nothing like the one the authors suggested.

As far as gear cutting was concerned, the conventional gearboxes had only two parts to machine, the pinion and wheel, while the epicyclic gear had a sun wheel, at least three planets, an internal annular ring and two pairs of coupling teeth. Accurate boring of the side-by-side boxes was obviously very much easier than that required for the epicyclic gearing.

Perhaps he should explain that forty years ago he had the privilege of serving his time when gearing was in its infancy. At that time there was no epicyclic gearing, so if he lacked enthusiasm perhaps they had only themselves to blame!

What he had said was not intended as a condemnation of epicyclic gearing but was an attempt to point out the unfair comparison made in the paper and to suggest that the claims made for epicyclic gearing were not in any way the whole story.

Captain(E) J. G. C. Given, C.B.E., R.N. (Member) said that he would confine his brief remarks to the future applications for marine propulsion and to some aspects that struck him particularly.

First of all, the compound planet gear had, to his mind, some resemblance—though it might seem far fetched—to the locked train double-reduction gear if one took the bull wheel, turned it inside out and wrapped it round the secondary pinions.

There appeared to be a lack of torsional flexibility in the gear and it must be remembered that propeller torque was still a variable factor. He would not like to see epicyclic gears go out into the wilderness as double-reduction gears did in this country for a long time, due to a variety of circumstances, but largely to the lack of appreciation of these torsional variations.

Brief mention was made of lubrication, and the authors spoke of mist lubrication of the gear teeth. He could not quite swallow that this would be adequate for large sizes. Probably, though this was not intended, the problems of lubrication would not be easy, especially if one wished to separate supplies for bearing and tooth surfaces.

He would be interested to hear more about the authors' experiences with noise in epicyclic gears, bearing in mind the reference which had already been made to the bell-like nature of the annulus.

The authors discounted very well the lack of accessibility by the small size. He would point out, however, bearing in mind that accidents happened in the best regulated families, that possibly epicyclic gears would have much more catastrophic breakdowns from tooth failure than "in-line" and more conventional types of marine gear. However, that was just another warning note.

Finally, these remarks did not in any way mean that he did not share the authors' view that there was a great possibility of development for these gears in marine propulsion applications. He would, however, emphasize the plea made by Mr. Archer, for a very thorough full-scale testing on shore, simulating seagoing conditions as nearly as possible, before the large-size epicyclic marine gear was planted on the long suffering seagoing engineer who had to take it round the world.

Mr. F. J. Everest, M.Sc., A.C.G.I., D.I.C., said that in the opening section of their very interesting paper the authors had given the impression, despite the past applications that were quoted, that the virtues of epicyclic gearing had largely lain dormant until brought to light by certain recent patents and applications. Nothing, of course, could be further from the truth. The absence of extensive applications of epicyclic gearing in the past was evidence of the appreciation by experienced gear designers and specialists of the inherent disadvantages and difficulties involved in this form of gearing.

Emphasis was made in the paper on a potential saving in weight and space, yet in the early applications cited in the paper these factors played no part. The model "T" Ford, Wilson, Cotal, and similar automobile transmissions were developed primarily to take advantage of the ability to change gear without the use of sliding gears, dog clutches and so on, and such designs were and continued to be generally heavier and more bulky than the corresponding conventional units. Similar applications were to be found in industrial and marine drives where speed change and reversing were the main consideration.

In regard to aircraft applications, the requirement of an "in-line" drive was normally the primary consideration, and this was the feature of epicyclic gearing which undoubtedly provided its main attraction and not *per se* the saving of weight and space.

With reference to the direct comparison between parallel shaft and epicyclic gears, and particularly to the figures given in Table I, the data presented and the conclusions drawn were—he agreed with Mr. Couling—entirely misleading, since the authors had not compared like with like insofar as their epicyclic examples employed case-hardening whilst the parallel gears did not. Worse than that, the parallel gear examples used materials in the design which had not been in common use for twenty years or more. It did not even represent modern practice in regard to the conventional gear unit.

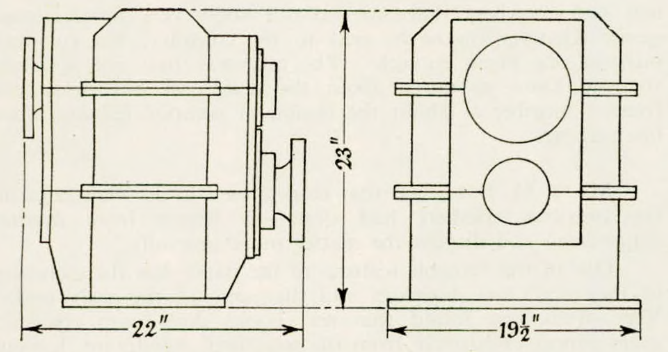
It could readily be demonstrated that the difference in size shown in these examples resulted almost entirely from the difference in material and not from the gearing arrangement. To illustrate the point, a design had been worked out for a unit of each type suitable for a 500 kW. ship's auxiliary generating set. The current British Standard specification was used in both sets of calculations and the comparative figures were given.

It should be noted that the weights given in Table II compared very favourably—560lb. for the parallel shaft unit against 510lb. The relative pitch line velocities were 98·5 and 93·5. These figures should be carefully noted in view of the statement that had been made more than once during the evening about the epicyclic unit inherently resulting in a lower relative pitch line velocity. Losses in horsepower for bearings and tooth losses were also shown. The distribution between these two elements was different in the two cases. The summation of the two was not very far different, yielding an efficiency in either case of just over 97 per cent. The figure of 96 per cent shown in the first example in Table I was definitely on the low side for modern practice. 97 per cent was conservative and 98 per cent was a common figure with parallel shaft gearing.

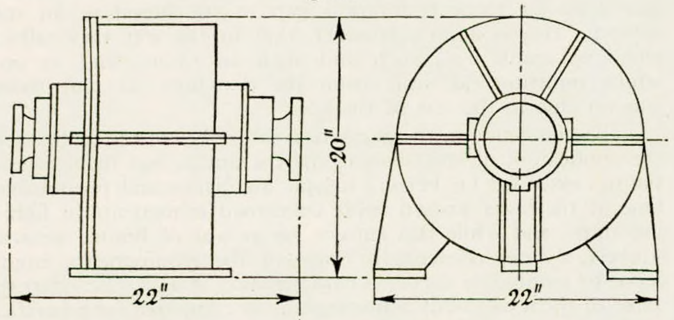
Mr. Allen had stated that the load on the pinion in the case of the conventional gearing was three times that on the sun pinion for an epicyclic unit. It was true that the load was shared between the three planet pinions, the load on any one of them being one-third. But the following statement, that thereby the face width of an epicyclic would be one-

Table II. COMPARISON BETWEEN PARALLEL SHAFT AND EPICYCLIC GEARS.

Duty		Turbo-generator			
Power		500 kW			
Speeds in r.p.m.		6,500/1,800			
Ratio		3·61:1			
Type of gear		Parallel shaft top pinion		Epicyclic planetary	
Weight in lb.		560		510	
Component		Pinion	Wheel	Sun planet	Annulus
Materials		Alloy case hardening steel, 70 tons per sq. in. U.T.S.	Alloy case hardening steel, 50 tons per sq. in. U.T.S.	Alloy case hardening steel, 50 and 70 tons per sq. in. U.T.S.	Alloy steel, 55/65 tons per sq. in. U.T.S.
Brinell No.		601 after case hardening and grinding	601 after case hardening and grinding	601 after case hardening and grinding	241 to 311
Face width, inch		5¾		3¼	
Circular pitch, inch		0·454		0·314	
Pitch circle, diameter, inch		3·47	12·53	4·6 and 3·7	12
Relative pitch line velocity, ft. per sec.		98·5		93·5	
Maximum bearing pressure, lb. per sq. in.		270		520	
Losses in horsepower	Bearing	16·5		14	
	Tooth	5		10	
Efficiency		97·2		97	



500 kW. 6,500/1,800 r.p.m. pinion above wheel; case hardened gears



500 kW. 6,500/1,800 r.p.m. planetary gear; case hardened sun and planets

FIG. 50

third, was entirely erroneous. Reference to B.S.S.436 or any treatise on gearing would show that to be so.

There were two main factors which affected the load-carrying capacity of a gear tooth; one was the load on the tooth and the other was the frequency of the number of repetitions of stress. In the case of the epicyclic unit, the load on the tooth of the sun pinion was one-third, but the stress was repeated at three times the frequency, and one effect tended to cancel out the other with the result that, in the conventional unit worked out, one got 5¾ inch for the face width as against 3¼ inch for an epicyclic, so there was a net reduction in face width in the epicyclic case. As far as the unit as a whole was concerned, that was offset and more than offset, in many cases, by the additional length required to house the double gear tooth coupling. In some cases also it was necessary to accommodate the planet pinion shaft bearings.

As would be seen from Fig. 50, the external dimensions of these two examples, both designs making use of hardened and ground gears, were given. Between the coupling flanges, the figures were 22 inch and the height was 23 as against 20 inch. There was only a matter of inches in it, and only 50lb. in weight.

In the example in question, it was doubtful whether the "in-line" arrangement was, in fact, an advantage. In Fig. 29 the generator had to be lifted on to a high plinth in order to match up to the new centre height. The authors said that this could be got over by designing a shallower condenser, but one would expect that if the condenser were shallower it would automatically have to be wider or longer.

With regard to the bearings, it was claimed both in the paper and frequently elsewhere that the epicyclic design solved bearing problems. Admittedly, it was a convenient feature that the tooth loading on the input and output shaft bearings balanced out, but in its place a worse problem was introduced in the loadings on the planet bearings, where double tooth loading occurred—two contacts which added up to give the double loading on each of the bearings—and there were three of them.

Mr. Allen had said that it was possible with an epicyclic unit to reduce the number of bearings to the three planet bearings and only one output bearing, giving a total of four. It was encouraging to think the number could be got down to the same number as was met with in the conventional unit.

The problem of lubricating the planet pinions was not an easy one. He agreed with Dr. Forsyth and others that it was not insuperable, but one must face up to the fact that it was definitely a difficult problem. The loading on the input pinion bearings in the case for which he had given figures was about 270lb. per sq. in. On the planet bearings it was over 500lb. per sq. in. and the three bearings were in an extremely awkward position to lubricate. As had been pointed out, however, extreme care as to cleanliness of oil, and so on, was essential to give freedom from seizure. Added to that, in certain of the arrangements shown a centrifugal loading was applied to the bearings which added to the double tooth loading, making the problem even worse.

As far as the gears were concerned, single helical tooth gears introduced the problem of tilting and end thrust. Double helical gears could not be conveniently incorporated without splitting one or more of the gears. In fact, it was very difficult to assemble at all without splitting one or more of the gears and that was not desirable. For these reasons, spur gears were commonly used in epicyclic units, and it was common knowledge that spur gears were inherently more noisy than helical gears.

The annulus gear was not an easy manufacturing proposition; it could be hobbed, but only by using the single position very narrow hob, with a formed tooth. Alternatively, it could be cut by the plano-generating process. In neither case would one expect the accuracy to be as good as was obtainable by hobbing or grinding the final wheel of a straightforward reduction gear.

The authors' claim for the accuracy of the planing process seemed rather excessive and could not really be taken too seriously.

To sum up, therefore, epicyclic gear units compared unfavourably with conventional units using comparable materials in regard to simplicity of design, ease of manufacture and maintenance, overall reliability and first cost. Those tempted to seek substantial savings in weight and space by adopting epicyclic gearing should first compare the corresponding dimensions for a conventional unit using similar materials and, if any slight saving should be attainable, they should carefully weigh up the disadvantages which would thereby be introduced.

For the benefit of those who might not be fully informed in regard to these matters, it might be of interest to explain that the necessary technique and plant for manufacturing conventional gear units with hardened and ground gears within the range of ratings referred to in the paper existed in this country.

DR. H. E. MERRITT, M.B.E., said that it was always stimulating to be shown a new approach to some branch of engineering technique which had reached the moribund condition usually described as "accepted practice". A stimulus of this kind, of course, could be pleasant or unpleasant, according to whether it added to one's knowledge or challenged those opinions which one cherished most because one could not prove them. But in either case, one should be grateful to the authors of the paper which was being discussed that evening.

The paper was primarily concerned with one basic type of epicyclic gear and with its application to marine purposes and was directed, quite properly, to marine engineers who might use it, rather than to gear manufacturers or technologists who, by the normal practice of the gear industry, would quite naturally proceed to tear it to pieces.

An epicyclic gear was obviously more complex than a fixed-shaft gear, and there should therefore be solid reasons before it was adopted. There were cases, of course, in which the basic kinematic properties of an epicyclic train provided the only solution. The first of these was the property of combining the motions of three or more rotating shafts. The second was the property of providing a change-speed mechanism in which the gear could be engaged by friction when one member was brought to rest. This was the feature which characterized the reversing trains suggested in the paper; it was not an essential, but a convenient arrangement, since it was usually easier to provide a brake than a clutch.

A third kinematic feature of epicyclic trains was the possibility of providing large ratios of reduction with comparatively few gears. But this could be a pitfall or gin; it was always hard to come by something for nothing. A number of epicyclic gears had been made which had given a good deal of trouble because the designers were too ambitious or too ignorant.

But successful epicyclic gears could be and had been made, the only requirement being that all the technical characteristics should be properly calculated. It was quite possible and all the basic technical design data had already been established by earlier practice. All these things could be done.

The authors had not dealt in as much detail as the gear specialists would like with tooth pressures and so forth which were permissible, but they had mentioned three important items of practical "know how". These were the need for high accuracy; the need for the members to "float" to equalize the loading; and the need for adequate lubrication of the planet bearings.

However, whether a device could be made successfully and whether it ought to be made at all were two entirely different questions. It appeared to him, looking at the matter with a narrow and prejudiced outlook (to which the study of gears so often seemed to lead) that in spite of the sober and laudable attitude expressed in the first paragraph of their conclusions, the authors strained too hard to find uses for inversions of a somewhat narrowly limited arrangement of gears.

Their main thesis was the great saving in overall dimensions made possible by the epicyclic gears they described but this resulted from a simultaneous change of two variables, namely, the gear arrangement and the gear materials. This was, of course, unscientific if one looked at the problem objectively.

Paraphrasing the previous speaker, the change from the conventional type of materials to case-hardened materials would result in itself in a reduction of weight and volume to one-sixth or more of what they would otherwise be. The reasons were, firstly, the much higher permissible stresses in the teeth as between case-hardened and soft steels; secondly, the lower pitch circle velocities; and thirdly—a new point—the greater stiffness of a small gearbox in relation to the elasticity of the gear teeth. Most gear problems arose from inaccuracy and the greater tooth deflexion in relation to inaccuracies. Less margin had to be allowed for maldistribution of tooth loading.

He agreed with the authors about case-hardened materials, but he would be more conservative in the type of arrangement adopted. If he were really anxious to obtain the smallest possible drive, assuming that it was required to be an "in-line" drive, he would prefer—particularly with more than 5 or 6 : 1—a double layshaft co-axial drive in which one had the advantage of case-hardened gears and some sharing of loading on the input and output gears, considerable reduction of loading on the coaxial bearings, much cleaner assembly, easier dismantling, more positive lubrication, and so on.

Under the heading, "Future Trends", the authors' proposals suggested a somewhat sombre reflexion. It was that engineers had so far been defeated in the search for an elegant mechanism for reducing and reversing speed at high horsepowers. The use of astern turbines, or the great monstrosity—as he believed it to be—of electrical transmission reflected a sorry position. He shared the view of the authors that for reversing purposes a combination of epicyclic gears with friction members might provide the necessary solution. He had said elsewhere that one did not know very much about gears, whatever might be said to the contrary, but for this purpose one knew enough. The pity was that one did not seem to know as much about the design of a very simple friction member as about the design of complicated machines like turbines.

Mr. J. M. Ford said that he proposed to do what none of the previous speakers had done—to depart from marine engineering and discuss the matter more generally.

One of the valuable features of the paper was the inclusion of the very clear drawings and diagrams in the early pages. Very often, one found that text books dealt with epicyclic gears almost exclusively from the analytical standpoint, leaving the actual engineering largely to the imagination. By giving such line drawings as those on pages 80 to 83, the authors had gone far towards filling a gap in the literature on the subject. It was a pity, however, that in the text they talked about so many h.p., such and such an r.p.m., and so on, whilst omitting the scale from the drawings, so that there was no clue to the size of the gear.

The tendency in the paper had naturally been to emphasize the application of epicyclics to marine duties, but their potentialities extended far beyond marine auxiliaries and propulsion. One of the lesser known fields concerned armaments in fighting ships, and while this subject *per se* was of limited general interest, a brief reference to some of the requirements might serve to emphasize certain characteristics of epicyclic gearing, some of them distinctly advantageous as compared with parallel shaft gears, others tending to offset the advantages.

In certain applications which had been explored in the Naval Ordnance Department of the Admiralty, it had been clear that in situations where space was at a premium the "in-line" arrangement, as a general rule, led to a much more compact layout than was possible with the parallel shaft type of gear. With the latter, the sideways displacement between, say, an electric motor and a hydraulic pump usually left spaces which could not readily be utilized. In a relatively spacious main machinery space this was generally of little consequence, but in modern armaments where the tendency was to crowd a quart into a pint pot, every cubic foot must be employed to the best advantage. Moreover, ability to dispense with separate bearings for the sun pinion, combined with the relatively small overall length and diameter of the gear assembly, often permitted of the epicyclic being constructed as an integral part of a hydraulic engine or built directly on to the end of an electric motor, to give a much shorter overall unit than was normally practicable with a parallel shaft gear.

It was well known that heavy armaments were sometimes moved by powerful machinery, constituting a servo system. The great attractiveness, from the viewpoint of space and weight, of epicyclics—say, up to 200 h.p.—had led to some examination of their possibilities as reduction gears between the heavy masses to be moved and the driving engines. In this connexion a problem had been encountered which was well known to those associated with servo mechanisms, namely, that of the effect of resilience or springiness in the gear drive and of its backlash on the natural frequency of oscillation and general behaviour of the system. Here there was a minor conflict between the requirements of the application, for which the maximum stiffness and minimum backlash were desirable, and the requirements of the epicyclic gearing, which must of necessity have a certain amount of flexibility or play, if only to ensure that the planets shared the load. As compared with a parallel shaft gear, there was an additional mesh in the active train and, moreover, the necessity for the floating annulus introduced a further source of backlash at the two sets of teeth on the annulus suspension. In short, it appeared at first sight that whenever heavy masses had to be accurately positioned, both under dynamic and static conditions, the epicyclic might be at a disadvantage as a reduction gear. However, so powerful was the urge to save weight and space in these armament applications, that the question was continuing to receive the closest attention.

Perhaps he might say something here about a typical test which was applied to some of these epicyclic gears—reversal at full torque, five times a minute, continuously running for 100 hours So far it had proved impossible to damage the epicyclic gear but it had been difficult to keep the coupling tight on the shaft.

It was clear that high precision in manufacture was necessary if this requirement was to be met and in this connexion it was doubtless the authors' modesty which had prevented them from disclosing the perfection to which they had developed their gear manufacturing techniques.

To come to more general considerations, a previous speaker had referred to the use of epicyclic gears as transmission dynamometers. In the simplest case of a planetary gear, it was only necessary to mount the fixed annulus in such a way that its torque reaction could be measured, in order to know the transmitted torque with a very high degree of accuracy. The reaction could be measured in a variety of ways—hydraulic, electric, strain gauges, and so on. When epicyclics were used in propulsion plants, this technique was probably capable of giving a truer value of the power going to the propeller than any other means. In industrial testing, an epicyclic dynamometer permitted, for example, of testing pump power consumption from, say, a Diesel engine. Incidentally, reverting to the subject of servos, an epicyclic gear immediately following the engine or driving motor and arranged for torque measurement offered possibilities for obtaining a torque or acceleration feedback term.

One further point: on page 83 it was stated that star gears were normally used when the low speed shaft rotated at 3,000 or more r.p.m. Turning to Fig. 5(a) and Fig. 5(b), it would be observed that the floating annulus (f) and the flexible coupling ring (g) revolved with the low speed shaft. It would seem that complete dynamic balance in such cases was theoretically impossible. Admittedly, the parts were relatively light, and they were approximately centred, but at the high angular velocities envisaged, was there not a risk of some vibration?

Finally, on page 86, there was a sentence which should be printed in red, namely: "It is always desirable to use a filtered oil supply". The cost of keeping the oil clean was negligible compared with the cost of the gearing, and the resultant saving in wear and tear could be fully appreciated only when one had seen what could happen without filtering. In machinery of the kind under review, cleanliness undoubtedly came before godliness!

MR. M. G. R. PETTY (Associate) said that there were three points on which he would like to comment.

Firstly noise; it would appear that with highly stressed small components operating at high speeds in totally enclosed housings, which themselves were load carrying members, that sonic frequencies of a fairly high order would be developed, though presumably the oil would have some damping effect. It would be interesting if comparisons were available between epicyclic gears for larger horsepowers and compound trains of simple gearing. Could the authors provide any information on these lines?

Secondly, oil churning; it could be readily understood that with a diversity of small gears carrying the load, that tooth

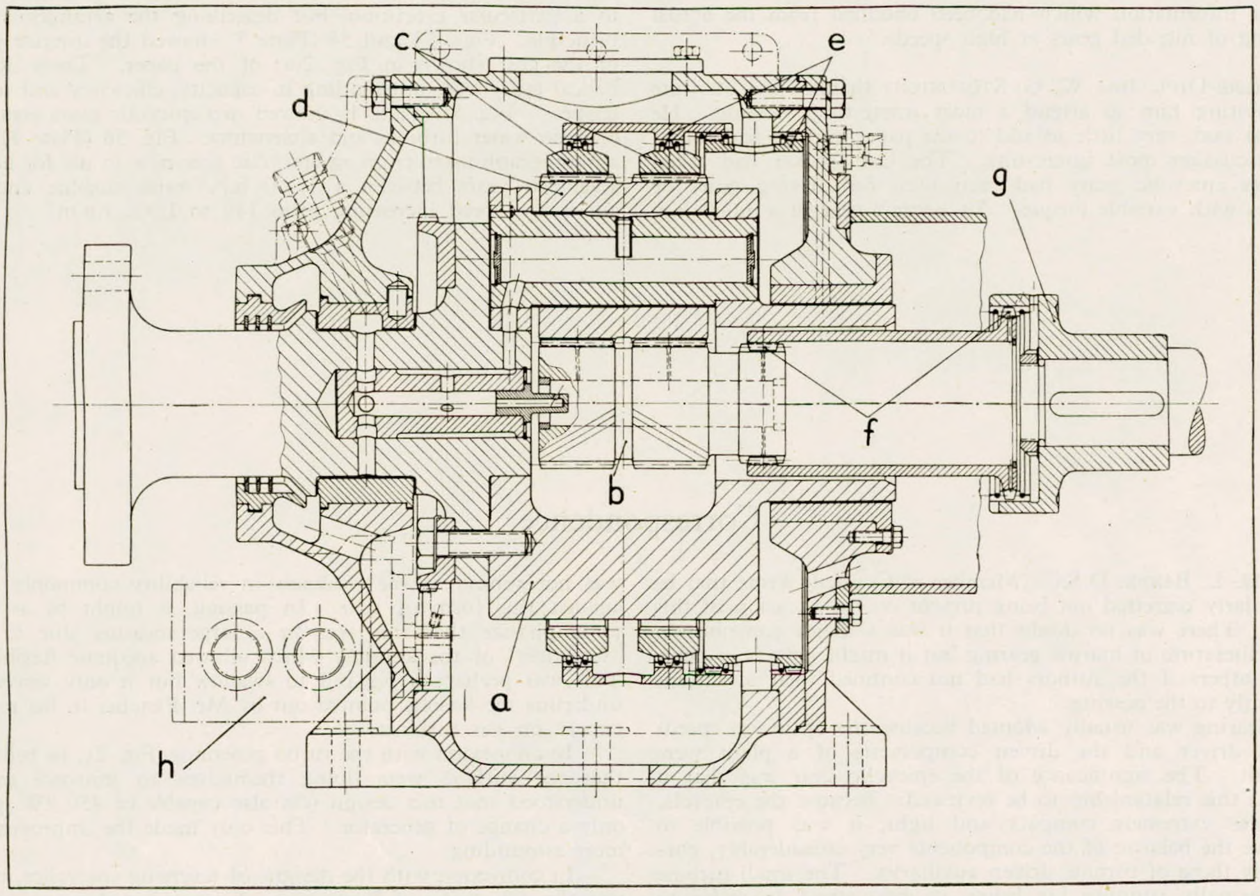


FIG. 52—*Typical double helical Stoeckicht epicyclic gear*

a—Planet carrier
b—Sun wheel
c—Annulus
d—Planet wheel
e—Annulus toothed coupling
f—Sun wheel toothed coupling
g—Coupling flange of driving machine
h—Oil pump

pressures might be of a lower order in installations such as had been described tonight, than in simple "in line" gears, but surely one was faced with the fact that the primary pinion in, say, a gas turbine unit, must have a very high peripheral speed and therefore the resultant centrifugal forces must make the lubrication of the teeth of the primary wheel a matter of some difficulty. It would also appear that wheels located furthest from the centre would collect the most oil and that the mating wheels would have to displace at very high velocities the oil between the teeth. It occurred to him that the work done on this oil might well account for a fair percentage of the overall heat generated. Could the authors say whether this did in fact take place and, if so, to what degree?

Thirdly, the application of epicyclic gears to machines of uneven torque. The installations dealt with in the paper had been either electrical generators or centrifugal pumps, both of which had smooth torque characteristics. The application which interested him most was the driving of multicylinder hydraulic pumps at, say, 800 to 1,000 r.p.m. These machines, of course, had peaks produced by each cylinder in turn coming to its work stroke, although by careful design and the correct selection of the number of cylinders, it was possible to produce a pump with quite a smooth pulse line. It was customary when driving pumps by a Diesel engine, for instance, to interpose a flywheel of suitable proportions to damp both pump and engine. Could the authors tell them whether such an application was suitable for epicyclic gears and, also, whether any data thereon was available.

He was particularly interested in the use of nitrided gears as he had used nitrided rams and liners running at very close limits with considerable success, although final grinding was necessary for these parts. He would be very interested in any further information which had been obtained from the actual running of nitrided gears at high speeds.

HERR DIPL. ING. W. G. STOECKICHT thanked the Institute for inviting him to attend a most interesting meeting. He had, he said, very little to add to the paper and he had found the discussion most interesting. The last speaker had asked whether epicyclic gears had been used for driving gears or engines with variable torque. An aircraft gear of a 3,000 h.p. aircraft engine for which he was responsible had been mentioned in the paper; the torque of this engine varied with each revolution from zero to 200 per cent.

Another application of the epicyclic gear in connexion with a cylinder pump had been seen by the authors. They would remember that there was a cylinder pump with an electric motor. In some quarters it was believed that the epicyclic gear lent itself very well to that kind of drive.

The question had been asked whether the conventional gear made of hardened gear wheels would be a successful competitor to the epicyclic gear. He could say this from his own experience: that, for instance, the 5,000 h.p. gear mentioned in the paper had met such competition and the competitor was a very famous firm in the field of gear design and gear manufacture. They built a competitive conventional gear with hardened and ground gear wheels but they could not obtain either the weight or the dimensions of the epicyclic gear. Worst of all, this gear had failed, it was a failure in the test pit. The result was that the very firm who had been building this competing gear decided to enter the epicyclic field. There was another firm on the continent—the only firm in the world, he believed, who used conventional large industrial and marine gearing with hardened gear wheels; they were a very famous firm and the accuracy of their products was very high. Nevertheless the dimensions of their gears could not compete with the dimensions of properly designed epicyclic gears.

A few photographs were shown in order to give a further explanation of the gears shown in Figs. 2(a) and 2(b) in the paper. These gears were double helical epicyclic gears representing the latest development of such gears in Germany. Fig. 51 (Plate 3) showed a closer view of the gear shown in Fig. 2(b) in the paper. Fig. 52 was a sectional drawing not referring to a particular execution, but describing the arrangement in principle. Figs. 53 and 54 (Plate 3) showed the interior parts of the gear shown in Fig. 2(a) of the paper. These double helical gears were outstanding in capacity, efficiency and noiselessness. Fig. 55 (Plate 4) showed two epicyclic gears arranged between water turbines and alternators. Fig. 56 (Plate 4) was a photograph taken from an epicyclic gear now in use for nearly seventeen years between a 1,000 h.p. water turbine and an alternator, speed increasing from 180 to 1,000 r.p.m.

Correspondence

MR. L. BAKER, D.S.C. (Member of Council) wrote that he particularly regretted not being present on the occasion of this paper. There was no doubt that it was a major contribution to the literature of marine gearing but it might have been helpful to others if the authors had not confined their attentions so strictly to the gearing.

Gearing was usually adopted because the optimum speeds of the driver and the driven components of a plant were different. The significance of the epicyclic gear was that it enabled this relationship to be reviewed. Because the epicyclic gear was extremely compact and light, it was possible to improve the balance of the components very considerably, particularly those of turbine driven auxiliaries. The small turbine was normally running far below its best speed in order to suit the reduction to the capacity of the gear ratios or to suit a compromise speed for the pump where this was direct driven. Tests in this country and service experience abroad had shown that the epicyclic gear constructed on the Stoeckicht principles was not subject to the weakness in reliability commonly held against this form of gear. In passing, it might be as well to emphasize that this was in a large measure due to the "weakness" of the annulus which allowed adequate flexibility. This was perhaps a big pill to swallow but it only served to underline the lessons pointed out by Mr. Fletcher in his recent paper* on gas turbines.

In connexion with the turbo generator, Fig. 21, he believed that the authors were doing themselves an injustice as he understood that this design was also capable of 450 kW. with only a change of generator. This only made the improvement more astounding.

In connexion with the designs of reversing epicyclics, there was the net saving of 3 per cent efficiency but there were two

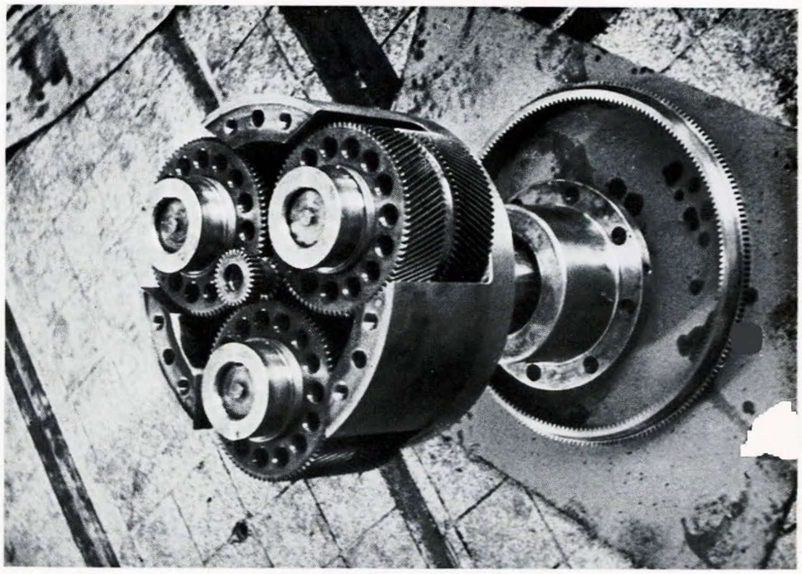
* Fletcher, A. Holmes. "The Marine Gas Turbine from the Viewpoint of an Aeronautical Engineer". Presented at a meeting of the Institution of Mechanical Engineers on 25th January 1952.

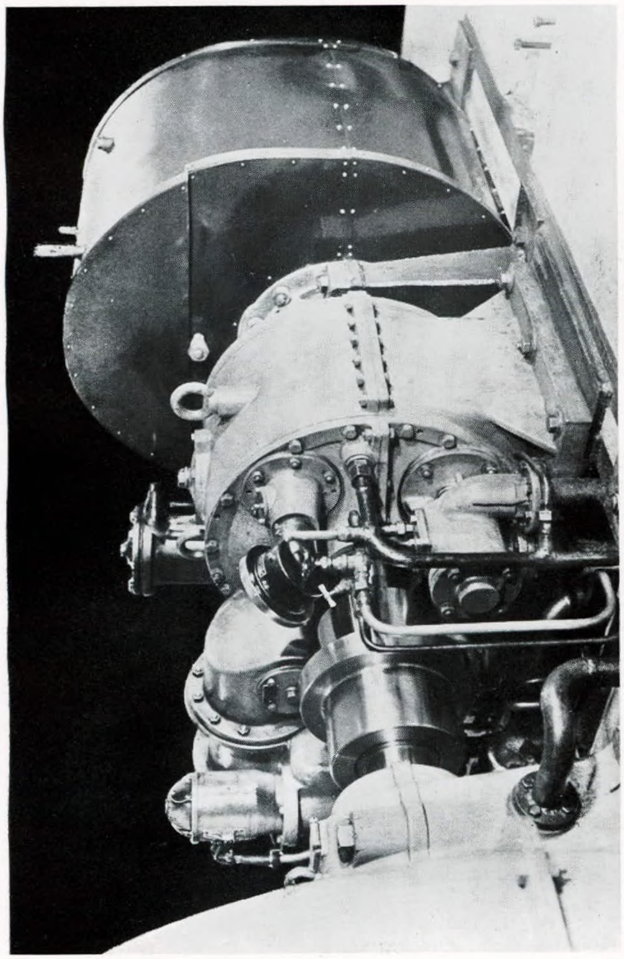
Plate 3

FIG. 51 *(left)—View of gear shown in Fig. 2(b)*

FIG. 53 *(bottom left)—View of gear shown in Fig. 2(a) with cover removed*

FIG. 54 *(below)—View of gear internals of installation shown in Fig. 2(a)*





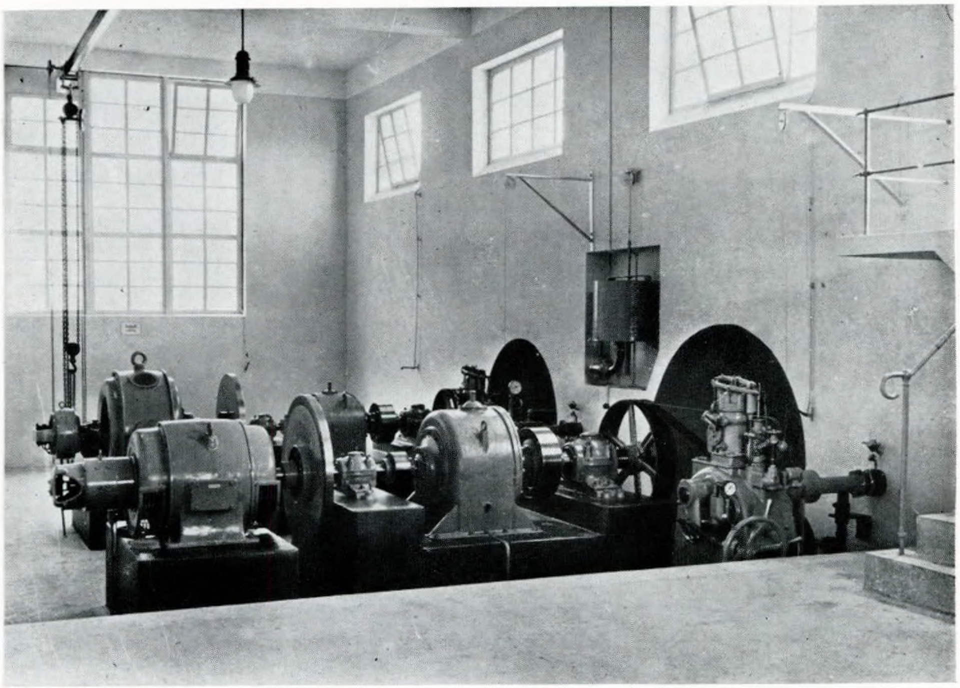


FIG. 55—*Stoeckicht gears between Escher Wyss water turbines and alternators. 160 h.p. speed increasing from 201 to 750 r.p.m. and 282 h.p. speed increasing from 153 to 1,000 r.p.m.*

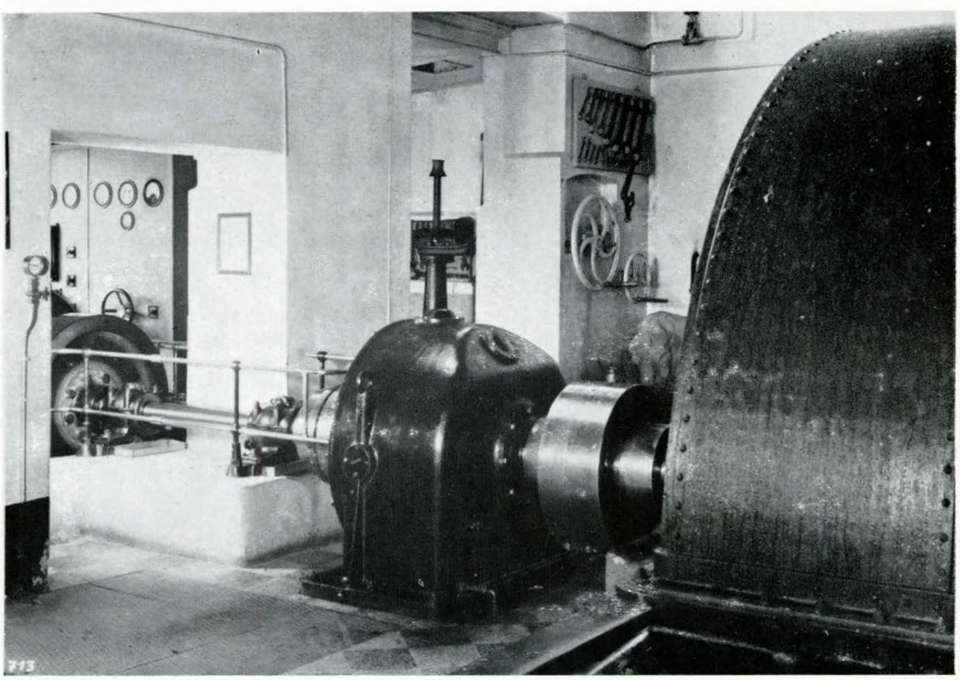


FIG. 56—*Stoeckicht epicyclic gear between 1,000 h.p. water turbine and alternator, speed increasing from 180 to 1,000 r.p.m. In use for seventeen years approximately*

other major points: firstly, the astern power available was much greater than was normally provided with turbine machinery; secondly, and perhaps most important of all, it made possible the considerable gains from reheating the steam to the original steam temperature without the risks of mal-operation normally associated with this scheme where reversing turbines are fitted and without the far greater loss in efficiency of electric coupling where this expedient is adopted to avoid the risk.

PROF. IR H. BLOK wrote that, as he was an ardent advocate of epicyclic gears, he would like to learn more about the practical limitations that must exist in epicyclic gears, just as in any other piece of machinery. Therefore, he posed the following two queries.

When the fullest advantages conceivable were to be taken of the reduction in size offered by the use of hardened sun and planet wheels, the risk of tooth scuffing might possibly be introduced. It seemed that present day design of epicyclic gears was still conservative enough in view of scuffing. Would the authors give their opinion and perhaps some quantitative data about the margin of safety, from this point of view, that is left for the still more advanced designs to be expected in the future? Or did they think that there were other limitations more decisive than those set by scuffing?

Further, it might be queried whether special precautions had proved necessary to prevent fretting corrosion on the teeth of the flexible coupling rings, fixing the annulus to the gear case or to other components. Minute rocking motions, such as were here to be expected, were nowadays generally recognized to result all too readily in fretting corrosion.

MR. T. BRADING had not studied the paper in detail but, referring to page 84, mention was made of nitriding of gear teeth. Presumably the choice of materials for this process had to be very carefully controlled, as the depth of nitriding was necessarily very small and any deflexion of the core would be disastrous.

Could the authors say whether the Stoeckicht gear was noticeably noisier than any other spur epicyclic, as the only example the writer had heard running was very noisy. The nature of the noise led to the opinion that something was eccentric in the whole set-up but opinions were sometimes very dangerous. In this respect his company had recently had an interesting case where a whole batch of gears were found to be noisy. They were of the normal straight tooth spur case hardened type and at least six different "expert" opinions were put forward as to the cause of the noise.

The noise occurred over a very small speed range and disappeared immediately just above or below this range. It was eventually found, after a great deal of study and experiment, that slight inaccuracies were present in the gear tooth form and by regrinding on a generating type of grinding machine the noise was practically eliminated.

Investigation of the form grinding process originally employed showed that there was nothing fundamentally wrong with the process, but closer control was required over the operator.

DR. T. W. F. BROWN, M.Sc. (Member) considered that this valuable paper, particularly for marine propulsion, was read appropriately before the Institute of Marine Engineers.

The diagrams showing the differences between planetary, star, solar, compound planet gear and double annulus gear were very clear and the value was greatly enhanced by the sectional drawings of each of these types of gear included in the figures. In Table I, the K values and sizes for a number of epicyclic gears were contrasted with parallel shaft, side pinion gears. He felt that the value of the paper would be further enhanced if similar figures could be given for the gears shown in Figs. 4(b) to 8(b) inclusive or, alternatively, if one size of a large component could be given, i.e. the p.c.d. of the annulus or the journal diameter from which other sizes could be determined.

The advantages in the use of epicyclic gears as reduction gears for main propulsion turbine drive were well brought out in the paper. If reversing and/or 2-speed elements were also added to the gears, which could be done best in an epicyclic gear box, at once all reversing and manœuvring sequences could be carried out on the gear box alone, the turbine being left when idling to operate on a speed governor at light load, always running in one direction, and the most efficient range of the turbine revolutions could be operated twice in relation to the revolutions of the main propeller, thus giving peak efficiency at cruising and full power conditions without undue complexity. In addition to brake gear, probably with multiple surfaces loaded hydraulically, a hydraulic coupling operating with reasonably high slip and therefore of small dimensions could also be incorporated to take up the main shock of reversing. The synchronization of turbine control and the gear changing or reversing could be carried out as smoothly as in, say, a Daimler car. Parts were all relatively small and were accessible (considerably more so than the actuating gear in a reversible propeller).

The elimination of the astern element in the steam turbine while free from trouble did make a vast improvement in turbine design as the distance between bearing centres of the ahead portion of the turbine alone was greatly reduced and higher speeds could be used again, leading to a smaller size of ahead turbine. Astern steam pipes, receiver pipes and manœuvring valves were also eliminated, together with the ill effects of possible leakage of astern steam when running ahead.

The paper dealt mainly with the principle of epicyclic gears, but it was hoped that it would stimulate interest in this form of gearing for marine application with its great promise of compactness, weight reduction and improved manœuvrability. The gear could also be produced on smaller machines than those required to produce the usual parallel shaft turbine reduction gears used at present in marine propulsion.

MR. A. W. DAVIS, B.Sc. (Member) recollected that his first contact with the theory and design of epicyclic gearing was in the course of a discussion with Dr. Stoeckicht on behalf of the Admiralty in the summer of 1945, and it had been particularly interesting to be able to examine in the clearer detail of the excellently presented paper the various ingenious layouts of epicyclic gears, and the devices by which these had been rendered a practical proposition, and also to read of the further developments and the practical applications sponsored by the authors' firm.

Fundamentally there was little doubt but that, given first-class workmanship, a family of gears had been developed to a stage that it could be regarded as a practical and reasonably reliable mechanism offering attractions in the way of reduced space and weight that no other type of gear was likely to be able to achieve.

Whether the degree of reliability achieved was sufficient to justify the employment of an epicyclic gear for marine main propulsion drives was a matter of doubt, although for employment with auxiliary machinery where smaller units were involved, the authors appeared to have demonstrated that the time had arrived for its adoption.

The larger the unit the greater became the problems associated with the design and performance of the planet wheel bearings and the accuracy of their alignment and position called for a higher degree of skill the larger the unit. Lubrication was another feature which tended to become more temperamental in relatively large rotating units, especially when circumstances might lead to rapid and unpredictable deterioration of the qualities of the oil supply; for example, sludging of the oil could quickly be revealed by centrifugal purifier action affecting lubricating oil channels to planet wheels, while the effect of salt water could rapidly produce in epicyclic gearing symptoms which would be quickly diagnosed on a parallel shaft gear and remedial action taken but which would remain undetected in the bosom of a complex epicyclic unit until irreparable damage transpired.

These considerations reflected upon the serious disadvantages of complex machinery which could not be readily examined. In this connexion it was also necessary to dwell upon the dangers to be associated with the reassembly of complex units having regard to the impracticability of carrying out an effective supervisory examination immediately before closing the casing, and the omission or displacement of any securities might only subsequently be revealed at the post mortem, always assuming that a lee shore had not meantime claimed a victim representing the whole as distinct from the part. This was not to disparage the developments presented by the paper under discussion but, where the safety of a ship was concerned, it was absolutely essential that the common failings of humanity be taken adequately into account.

Table I illustrated the saving in efficiency of the epicyclic in comparison with the parallel shaft gear and it was evident that this was of the order of 1½ per cent when the comparison was made with double reduction parallel shaft gears. The difference was significant but not sufficiently so to claim consideration in relation to the advisability or otherwise of the adoption of a complex mechanism.

The authors drew attention to the fact that the epicyclic gear had particularly interesting possibilities from the point of view of reversibility. In this connexion, it would be recognized that the critical components additional and enshrouding the epicyclic gear itself were the clutch units. In relatively light gears such units did not present mechanical difficulty but the marine application was quite a different matter when large tonnages were affected. The energy to be absorbed in decelerating a ship when the main engines were put over from "full ahead" to a nominal "full astern" was released partly by the propeller and partly in the reversing mechanism. In the case of a reversing steam turbine the astern steam was superheated by windage in the blading and in the case of a reversing Diesel engine the energy was absorbed by compression in the cylinders. Where clutches were involved, a great amount of heat was generated between the opposing clutch units, whatever these might be, and this heat could only be prevented from causing damage when the clutch was designed for the circulation of a heavy flow of coolant. It would be generally felt that such units were not situated to the best advantage when by the nature of their function they were situated to envelop an epicyclic gear box.

It was recognized that the authors accepted these limitations in principle, as they had emphasized in their conclusion, and in drawing attention to certain of the drawbacks of an epicyclic gear in its application to main propulsion on shipboard, the writer did not wish to detract from the brilliant developments presented by the authors in the paper.

MR. G. B. R. FEILDEN, M.A., M.I.Mech.E., wrote regretting that he had been unable to attend the reading of the paper.

When the design of the Ruston and Hornsby 750 kW. gas turbine was begun in 1946, quotations were obtained from a number of builders of parallel shaft gears and from one builder of epicyclic gears, for the supply of the 4 : 1 reduction gear necessary for driving the 1,500 r.p.m. alternator from the 6,000 r.p.m. output turbine. The epicyclic gear proved to be cheaper than any of the parallel shaft gears, and was very much smaller. This point could be clearly seen from the photograph of Fig. 57, in which the gear was marked by the arrow "A". Many engineers who had seen the set had commented on the fact that the gear casing was but little bigger than that of the admittedly conservatively designed alternator pedestal bearing shown by arrow "B".

A set of gear internals designed and manufactured by W. H. Allen, Sons and Co., Ltd., was fitted to the Ruston gas turbine in October 1951, since which date the gear had completed 1,607 hours' running at loads from 800 kW. downwards. The general condition of the gear teeth after this period of use could be gathered from Fig. 58, while Fig. 59 showed

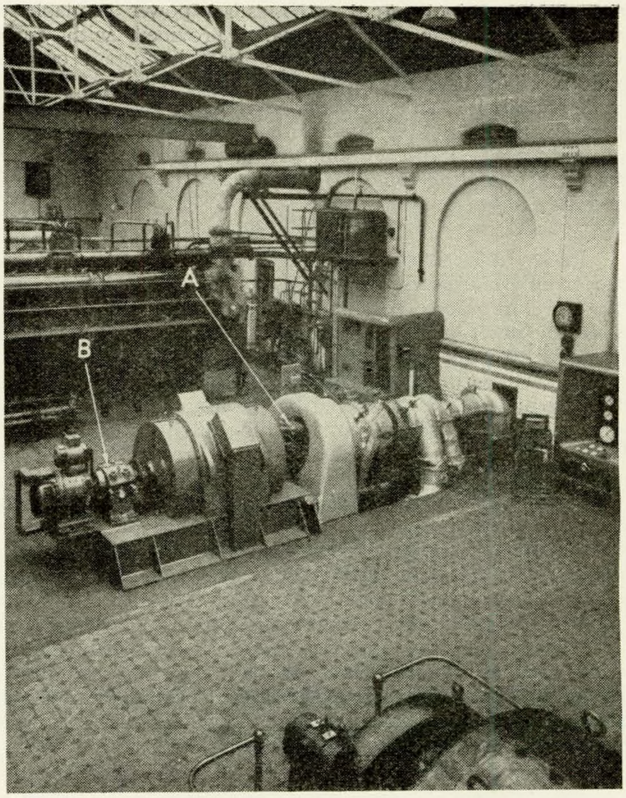


FIG. 57—*General view of Ruston and Hornsby 750 kW. gas turbine, showing installation of epicyclic reduction gear at "A"*

a close-up view of the teeth of the sunwheel. As would be seen from these photographs, polishing of the teeth had taken place, but no wear could be measured when the tooth profiles were checked and this was confirmed by the fact that the original grinding marks were still visible. The only component in the gear which left the least grounds for criticism was one of the annulus rings, in which somewhat uneven bedding was observed, which gave rise to higher tooth pressures at the ends of the teeth than in the middle. Measurement of this annulus showed that the ends of the teeth were, in fact, about ·0002 inch "proud" of their nominal position. This small error

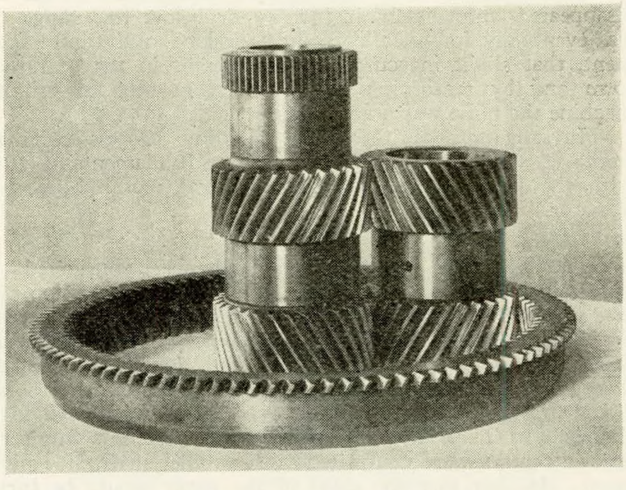


FIG. 58—*Sun wheel, one planet gear and one annulus ring of Allen-Stoeckicht epicyclic reduction gear, after running for 1,607 hours in Ruston gas turbine*

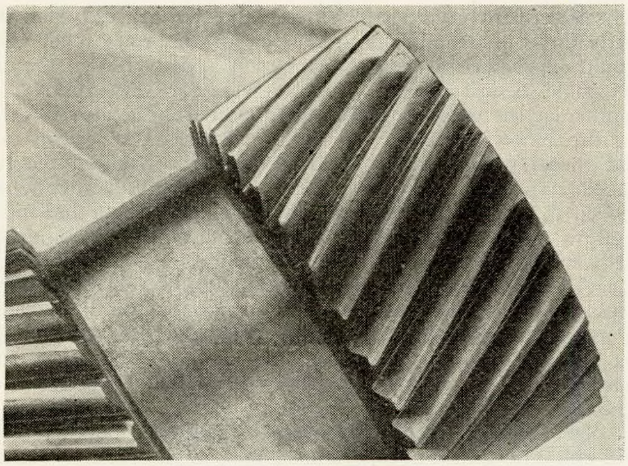


FIG. 59—*View of loaded face (at top) of sun wheel of Allen-Stoeckicht 4:1 epicyclic reduction gear after operation in Ruston gas turbine*
Note even polishing of the loaded faces and the persistence of the original grinding marks on these faces

had made no difference to the running of the gear, which was remarkably smooth and silent throughout the period of testing.

While the accessibility of the gear shown in Fig. 57 left something to be desired, in subsequent machines this point had been greatly improved by a rearrangement of the turbine exhaust trunking, and the writer considered that it might be claimed that, by suitable design, the accessibility of epicyclic gears might be made adequate for practical requirements. Owing to the much smaller size of the running components in an epicyclic gear than in an equivalent parallel shaft design, it was feasible to contemplate removing an epicyclic gear *in toto* from its case, which permitted a more thorough examination to be made than was possible if the inspection had to be carried out with the components in their operating position.

In conclusion, Mr. Feilden wished to congratulate W. H. Allen, Sons and Co., Ltd., on their initiative in embarking upon the manufacture of precision epicyclic gearing in the United Kingdom. For too long engineers had had to accept the large bulk and moderate accuracy which had been available in commercial parallel shaft gears, and Mr. Feilden had no doubt regarding the important rôle which epicyclic gears would play in future marine and land power plant development.

REAR ADMIRAL(E) I. G. MACLEAN thought it was gratifying to see from this most interesting paper the strides that had been taken in the development of epicyclic gearing for marine purposes. He had believed for a number of years that gearing of this type could form a reliable and equally efficient alternative to the more normal type and that it might have considerable advantages in particular applications. The first paragraph of the authors' conclusions seemed to him to be a very fair appreciation.

To be critical, it was perhaps a pity that the comparisons quoted in Table I were between epicyclic gear sets using hardened and ground or nitrided gears in the planetaries and parallel shaft sets having what are commonly referred to as soft gears. If the same advanced manufacturing techniques had been used for the latter, the comparisons would have been much more valuable. It would seem that some at least of the advantages claimed were the result of the efforts which the firm had made in developing the technique necessary for the accurate manufacture of gears in hard material, rather than in inherent advantages of epicyclic gearing. Even if this was so, it was still apparent that the firm had been wise to provide themselves with the means of manufacturing epicyclic types of gear, thus placing themselves in a position to make the most suitable gear for any particular case.

There could be little doubt that it would be to the national advantage were other firms equally well equipped.

MR. R. MICHEL, M.S., considered the paper to be timely and informative. While epicyclic gears had been known in the United States of America for many years, the generality of their use, in his opinion, had not been comparable to the advantages which these gears possessed over fixed axis types.

Planetary gears were employed on U.S. Navy Eagle boats built during World War I. These gears were of about 2,400 h.p. and were used to reduce turbine speed to propeller speed. So far as was known, they operated satisfactorily. Planetary gears were also widely used during World War II in relatively low powered landing craft for reducing Diesel engine speeds to propeller speeds. These gears were of the reversible type.

The advantages of planetary gears for Naval propulsion was first brought to the writer's attention while on a mission to Germany after World War II. At that time data was obtained on the 5,000 h.p. gear designed by Stoeckicht and mentioned on page 80 of the paper. The specific weight in pounds per rated horsepower was about one-sixth of that of a conventional fixed axis single reduction gear of the same power and reduction ratio. This gear was tested and successfully carried a load up to 8,500 h.p. without damage to the gear teeth. At that load, one of the planet wheel bearings failed.

The success of modern epicyclic gearing for relatively large powers was due to built-in flexibility, which ensured uniform load distribution to all gear surfaces. This was obtained in Stoeckicht's designs by the use of a floating member, usually the gear ring. It was of interest to note that the Westinghouse Electric Corporation employed a floating frame for equalizing the load along the surface of pinion teeth of a fixed axis marine gear.

In spite of the flexibility with which modern epicyclic gears were designed, the idea should not prevail that accuracy of manufacture and careful consideration of stresses of the various parts was not required. With the very low specific weights attained and the resultant high utilization of the metal, attention to details of design and manufacture were paramount.

The use of reversible epicyclic gearing for marine and naval propulsion was very attractive, as the reversing element of the prime mover might be omitted. Idling of the prime mover for warming-up was readily obtainable in epicyclic gears.

MR. A. SYKES, B.Sc., Wh.Ex., M.I.Mech.E., considered that the paper made a very notable contribution to the development of the application of epicyclic gears for large powers. The authors stated that there were numerous cases where ideas in the early stages of mechanical engineering development had lain unused because of the difficulties of application or methods of construction. He thought in this case that the emphasis should be placed on the facilities for cutting accurate gears.

One of the difficulties of applying epicyclic gears in the past, particularly for high speed work, was that the effect of pitch errors had a very appreciable influence on the distribution of load and the incidence of vibration, and it was only fairly recently that it had been practicable to produce gears of this type, particularly internal gears, to a degree of precision which would make them practicable in application. The most important feature claimed for the epicyclic gear was that it could be designed to occupy small space and to be of relatively light weight. This claim was not fully established, particularly on gears of small and medium power. The output and input shafts were necessarily in line with each other, although it should be pointed out that this was not always an advantage, as there were some arrangements of plant in which it was more convenient to have the two shafts displaced from each other by a suitable distance.

Examples were given of large planetary gears in use in Germany up to 5,000 h.p. and it would be interesting to know what life had actually been accomplished on these gears and whether their performance had been satisfactory in all respects. Although, as mentioned above, the epicyclic gear was usually compact, it was not necessarily lower in cost than a simple reduction gear; in fact, in small and medium powers the reverse was usually the case and it would generally be agreed that the question of maintenance was a somewhat more difficult problem in the case of the epicyclic gear.

Fig. 46 showed a planetary gear which formed the second train of a double reduction gear. It would be interesting to know why the first reduction gear was not of the planetary type and it would be useful to have a general indication as to where the planetary gear had advantages or otherwise as compared with the other types of gear illustrated and described.

In regard to efficiency, although it was true that, with the normal arrangement of epicyclic gear, the speed of tooth engagement for a given primary speed was a little lower than with a simple parallel shaft gear, it should not be overlooked that the number of tooth engagements in proportion to the speed of output shaft was greater than with the parallel shaft gear. In the latter case, each member made one contact per revolution, whereas with the epicyclic gear for each revolution of the output shaft there was more than one tooth engagement depending on the ratio of the gear. With a very high ratio epicyclic gear, the number of tooth engagements was very large in proportion to the speed of the output shaft and, consequently, the frictional losses and churning losses which were related to the number of tooth contacts were correspondingly high.

With regard to bearing losses it should not be overlooked that the bearing load on the planet pinions was equal to twice the tooth load and the size of these bearings must be designed accordingly with their attendant frictional losses. It was far too sweeping a statement to say that an epicyclic gear would generally be more efficient than a worm gear, as even with high ratios a double or triple reduction worm gear could be designed so as to compare very favourably with an epicyclic gear and would usually be of simple construction.

It was mentioned in the paper that the planetary type of gear was not suitable where the speed of rotation of the planet carrier was high on account of the centrifugal load on the planet bearings. It was also clear that lubrication of the planet bearings in such circumstances was by no means a simple problem.

It would be interesting to know whether any full load efficiency tests had actually been carried out so that they could be compared with corresponding figures for parallel shaft gears, as careful tests had been made on this type of gear and recorded efficiencies were available.

Load sharing between planet wheels was a very important point and he agreed with the authors in emphasizing the necessity of accurate boring of the planet carrier. Any error in spacing of the planet pinions had an effect as regards distribution of load twice as great as a pitch error of corresponding magnitude in one of the gears and when one took into account that there was an accumulation of errors arising from the sun pinion, the planet pinions, the annulus and the spacing of the planet pinions, there was a likelihood of eccentric running of any member which was not located by bearings and even if this was only of small amount it would, at high speeds, undoubtedly involve variations in tooth loading.

He was particularly interested to hear of the success which had been achieved with the nitriding process. As was well known, the case depth with this process was very small and experience with heavily loaded nitrided gears had not been uniformly successful, as it had been found in some cases that the case collapsed under heavy load. The authors stated that lapping after nitriding gave a slight amount of tip and root relief. Could they give figures as to the actual amount of this relief, as it was difficult to see how this could occur to any measurable degree.

In respect of quietness of running, it would be interesting to have the authors' views as to the relative merits of straight, single helical and double helical gears. Straight teeth did, of course, involve the fewest manufacturing difficulties and double helical teeth involved a gap between the two helices which was fairly large in proportion to the size of the gears used. One would expect helical gears to run more quietly but in the case of single helical gears, whilst it was not very difficult to counteract the thrusts on the sun pinion and the annulus, and the thrusts on the planet pinions were balanced as regards total end thrust, the point of application of the load on the two opposite sides of the planet pinion caused a tilting force which needed to be counteracted by suitable bearing design and might tend to affect the uniformity of distribution of load on the teeth and along the length of the bearings.

An interesting point which the paper brought out was the application of white metal to the surface of spindles, instead of to the bores of wheels as was the usual practice. Presumably the authors had some reason for this design. What were considered to be the advantages over the normal bushed type of wheel?

In Table I, giving a comparison between parallel shafts and epicyclic gears, it was notable that in the examples given, the comparison was not on the same basis, as soft materials for parallel shaft gears were compared with hardened materials for the planetary gears. If hardened gears were used for parallel shaft arrangement, much higher loads could be carried and a correspondingly smaller gear could be employed, as well as achieving the attendant simplicity of the arrangement with one wheel and one pinion.

The developments described were of a particularly interesting nature but it was perhaps too early to form a definite conclusion as to their value, and advantage in cost and maintenance charges was doubtful; epicyclic gears would in general only be preferred where small space and light weight were vital factors, but here also the use of hardened and ground parallel shaft gears should not be overlooked, as the difference in weight and space occupied was quite small and the greater simplicity of the parallel shaft gear was unquestionable.

PROFESSOR W. A. TUPLIN, D.Sc., M.I.Mech.E., wrote that the power loss by tooth-friction in epicyclic gears might be estimated without referring to "potential horsepower". This term needed a good deal of explanation, particularly in connexion with the more complicated types of epicyclic gear, and its use was preferably avoided.

The energy dissipated in unit time by tooth friction in any pair of spur, helical or bevel gears was the product of the tooth load acting tangentially to the pitch circles, the speed of the teeth measured at the pitch circles relative to the plane of the axis of the gears and a loss-factor depending on the coefficient of friction between the teeth and some details of their geometry.

An approximate value of the loss factor was:—

$$\mu \left[\frac{1}{d_{01}} + \frac{1}{d_{02}}\right] \left[\frac{a_1^2 + a_2^2}{a_1 + a_2}\right]$$

where μ = coefficient of friction;

$a_1\, a_2$ = distance from pitch point of engagement to tip of gear, measured along line of action (suffixes referred to the two gears);

$d_{01}\, d_{02}$ were the base diameters of the gears.

For normal running conditions in spur, helical or bevel gears, the loss factor was about:—

$$\frac{1}{6}\left(\frac{1}{t} + \frac{1}{T}\right)$$

where t and T were the numbers of teeth in the gears. For an internal gear, T was negative.

In applying this principle to the determination of power loss by friction in any pair of gears in an epicyclic train, however complicated, all that was necessary was to determine the angular velocity of either gear relative to the plane of the axes of the gears, hence to determine the pitch circle speed of the teeth relative to that plane, and finally to multiply this speed by the product of tangential tooth load and loss factor. There was no trouble with algebraic signs; all the quantities concerned were taken as positive.

MR. R. H. WEIR considered that the authors were to be congratulated on producing a paper which provided most interesting reading and, at the same time, an invaluable reference. In spite of the wide field covered, he found little to query and still less to criticize.

His only criticism concerned the comparisons made between parallel shaft and epicyclic gears in Table I. He felt that this table probably exaggerated the differences between the two types of gears and he would like to have seen at least one comparison made using the same design assumptions and materials for both. In fairness to the authors, it should be pointed out that they drew attention to this weakness in Table I. His queries were as follows.

The authors appeared to have set a limit of 10,000 h.p. for epicyclic gears. Did they regard this as an all-time limit or merely the largest h.p. gear which they considered should be tackled in the light of present knowledge and experience?

Everything else being equal (which it never was!) the lower weight of the epicyclic gear would suggest lower cost. The authors did not claim this as an advantage and he would like to have their views on relative costs of truly comparative gears.

It would seem that some of the efficiencies quoted in Table I must be calculated values. If this were so, which of the figures gave a comparison of efficiencies measured on test?

MR. D. B. WELBOURN, M.A., A.M.I.Mech.E., A.M.I.E.E., before making any comments on the technical content of this most interesting paper, thought it might be of interest if he were to recount a chapter of technical history, and say something of how it came to be read at all. The authors had paid tribute to the Admiralty, but the Admiralty was not an amorphous body, but like all other bodies a collection of individuals. Just before the end of the war with Germany, Com'r(E) I. G. Aylen, O.B.E., D.S.C., R.N., knowing that the German development of high speed submarines was based on research work in progress at the Walter Works, near Kiel, occupied them before an armistice was declared with a small detachment of 30 Assault Unit, and, although surrounded by millions, literally, of German troops, preserved the works and its contents for allied investigation. The surprise of those investigating the works could be imagined when it was discovered that one of the hydrogen-peroxide propelled "U" boats contained a gear weighing only 1 ton and just 40 inch in diameter to transmit 5,000 h.p., with a reduction ratio of 3,770 to 580 r.p.m. This discovery was followed up rapidly on the spot by the discovery of its designer, Mr. W. G. Stoeckicht. The importance of the discovery was fully realized by Captain (E) I. G. McLean, O.B.E., R.N. and Com'r(E) W. H. B. Lane, D.S.C., R.N., with the result that in the middle of January 1946, he (Mr. Welbourn), as an R.N.V.R. officer, was on his way to Munich with instructions to discuss with Mr. Stoeckicht the possibility of designing and building such gears in this country. It was, he felt, only fitting that tribute should be paid to the energy with which the responsible officers acted and the longsightedness which they displayed.

Turning to the paper proper, he would like first of all to emphasize the aspect about which the authors had, he felt, been unduly cautious, and that was the economic aspect of this development. At a time such as the present when this country was fighting for its economic existence, the advantages offered by the reduction in quantities of valuable raw materials required for a set of gears, the reduction in the labour content of them, and finally the reduction in the losses in the machinery, of which they formed a part, were factors which must not be overlooked. In this respect, he was a little surprised, turning to Table I, to observe how heavy the sets of epicyclic gearing listed there appeared to be compared with the parallel shaft gears given by comparison, since comparing annulus diameters with wheel sizes and the face widths for the two sets of gears, it would appear that the epicyclic gear should, in every case, weigh not more than one-eighth to one-tenth of the weight of the parallel shaft gear with which it was compared. It was not clear to him where this discrepancy in weights arose unless the epicyclic gear had, in each case, been unduly penalized by the inclusion of a half coupling and unnecessary length of shaft, which might well have been avoided by the use either of a further gear tooth sleeve coupling or of a Hirth coupling.

It was possible that the weight had been penalized to a certain extent by the fact that the loadings on the annulus teeth seemed to be unduly modest in every case compared with the nominal loadings allowed in the parallel shaft gears.

The comparisons given in this table suggested that full advantage had not been taken of the possibility offered by these gears to avoid a three-bearing arrangement on the low speed shaft and the improvement obtainable, particularly in marine sets where the supports were not always as rigid as they might be.

Considering further the data given in Table I, he would be glad if the authors would confirm that the pressure angle was the same for the parallel shaft and for the epicyclic gears, since otherwise the K factors were not strictly comparable, and it seemed to him to be a pity that the K factor was not defined so as to include half the cosecant of twice the pressure angle which would give K values strictly comparable, irrespective of pressure angle. What mattered, of course, was not the nominal K value but the actual K value when the dynamic loads on the gear teeth were included and, since these, following Dr. Tuplin's recent paper* to the Institution of Mechanical Engineers, would be very much smaller in the case of epicyclic gears, such as those described, where the pitchline velocities were relatively low and the elasticity of the parts was high, the K values used in these epicyclic gears could only be described as being excessively conservative, taken at their face value.

This, however, was not the whole story, since it appeared to him that the pinion face widths were, in almost every case, so long that due to the torsional deflexion of the pinion only a portion of the face width would be carrying load, and if this were taken into account the K values were increased to a more realistic level. Incidentally, this would have afforded the opportunity to reduce the dimensions of the gears and, consequently, their weights.

Turning to the allowable bearing pressures, it might seem at first sight that these were high compared with the present practice but closer examination would show that they were also remarkably conservative figures, since the bearings in question were short, with no possibility of misalignment, and calculation along the lines of Cameron's paper would show that the oil film thickness in every case was probably just as great as the oil film thickness in the parallel shaft gear bearings. Again, it was possibly a pity that the authors had not included this criterion.

There was one other comment on the design which he would like to make and that was on the question of annulus design and load sharing between the planet wheels. The saw tooth shape given by the strain gauges could be shown analytically to be due to the three components of load, radial, tangential and the bending moment on the tooth, which, respectively, looked like this:—

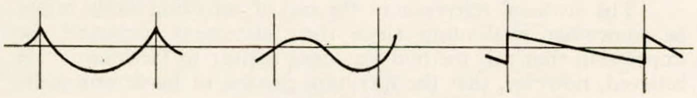


FIG. 60

He could not agree with the authors, however, when they suggested that all that was necessary to obtain a good load sharing was to allow one of the coaxial members to float and that, by mounting the annulus flexibly in the gear case, very good load sharing could be achieved. This would be true for a low speed three-planet gear, but for a high speed gear, or for one with more than three planets, very brief consideration of the acceleration forces involved would demonstrate that,

* Tuplin, W. A. 1950. "Gear-tooth Stresses at High Speed". Proc.I.Mech.E., Vol. 163, p. 162.

unless large dynamic tooth loads were to be called into play, the flexibility of the annulus itself was the determining factor and, if this were sufficiently great, then freedom to float was not essential.

Leading from this, he would like to suggest to the authors that, at any rate in big gears it would be worth while to shave or grind the annulus teeth, even if they were soft, to obtain adjacent pitch accuracies comparable with the extremely high level which they had achieved on the pinions and planet wheels.

In conclusion, he would like to remark that even had he appeared to be critical of certain points of the paper, its presentation had given him great pleasure, and he could only hope that it would be followed by a rapid increase in the rate of their introduction into the marine engineering world.

Mr. A. G. Wilson (Associate) thought that the authors rightly stressed the great advantages of epicyclic gears in affording an economical method of speed reduction associated with high power and high speed prime movers.

It was certainly to be expected that epicyclic gears would afford economies and advantages in these fields that would lead to their wide acceptance, but they had already a place in the less spectacular but possibly more numerous applications where lower speed prime movers of lesser horsepowers were employed. In this field, and particularly where changes of ratio or changes of direction were sought, the epicyclic gear lent itself just as appropriately as it did to a reduction gear.

The authors illustrated a large number of applications varying in power, speed and the amount of reduction afforded, and they showed a series of designs which were both neat and compact, but it was not made clear in their paper how many of these had actually proved themselves in service.

His experience in design and manufacture of epicyclic gears enabled him to endorse the authors' observations with regard to the importance of extreme dimensional accuracy and high quality of finish in the manufacture of epicyclic gears and this particularly applied to gears which had to operate under conditions of high speed and high specific loading.

Under the heading of efficiency, the authors indicated that certain types of epicyclic gearing were inherently more efficient than others, depending on the relative motion of the various gears in the train, and thus depending on how great the so-called potential horsepower is in relation to the transmitted horsepower. He would amplify this by saying that in any epicyclic gear arrangement, either simple or compounded, in which the input and output rotated in the same direction, a certain component of the power transmitted might be regarded as being transmitted directly and the remaining component, which was the potential horsepower referred to, required to be converted by the gearing. Thus, the nearer the gear ratio was to unity, the smaller would be the component to be converted. He might add, therefore, that in an epicyclic gear having a ratio close to unity, not only should the gear losses be extremely low but also a higher specific loading might be carried due to the lower relative speeds of the gear components.

The authors' reference to the use of nitriding steels might be somewhat misleading since their statement conveyed the impression that this method was used mainly in Germany. He believed, however, that the nitriding process of hardening gears had been used in the U.S.A., and to some extent in this country for many years.

It was refreshing to one who had been associated most of his business life with the manufacture and development of epicyclic gears, and who had heard so many unjust condemnations of their principles from uninformed people, to find that such active interest was being taken in their application to the problems which were now coming to the forefront as a result of the development of new high speed prime movers.

Mr. H. A. Wilson (Member) agreed with the authors that the principle of epicyclic gearing had been known for many years. It had been applied in various forms, to speed recording instruments, certain forms of transmission in automobile drives, aeroplane propulsion, etc., but the application to power drives for electric generators, pumps and similar auxiliary equipment in the marine field, was a comparatively recent development.

The main advantages claimed were reduction in space required, also weight, but if, as was brought out in the paper (cf. Table I), this had been accomplished at the expense of higher tooth and bearing pressures, it raised questions as to the prospective life of gears in the epicyclic system.

A point that seemed deserving of some consideration was that, whereas in the parallel shaft arrangement, bearings were all exposed to view and were generally of the split type which could be readily examined and lined up, the planet wheel bearings (and some others) in the epicyclic system were of the solid, i.e. not split, type, and were completely hidden from view, whether running or stopped. It might be advanced that bearings of this kind, which could not be either seen or "felt", did not give trouble. Equally, external bearings should not give trouble, but the unfortunate fact remained that from time to time they did.

Since the first parallel shaft type of gear was introduced, a great many problems arising out of tooth form, clearances, alignment, vibration, etc., had arisen, and he thought it might be said that after a great deal of study these had been practically overcome. It would appear to him, however, that in any extensive change over to epicyclic gears it might be necessary to face up to a new set of problems of a similar kind.

The authors were to be complimented on presenting a most interesting paper, to which they had obviously given most careful thought. There was no doubt that for certain applications of moderate horsepower, there was a considerable field for development, but for appreciably higher powers, particularly where continuous rating was involved, such as in main propulsion, progress, in his view, must necessarily involve a gradual process of evolution built up on actual experience.

Mr. G. L. Dannehower, M.E., wrote that his company had been long familiar with the many advantages of epicyclic gears but greater use of the arrangement in America had been retarded because too little attention had been given to advanced techniques for obtaining hardness, accuracy and smoothness, and uniformity of the planets by finish grinding. Dry grinding without a coolant was necessary to avoid grinding checks and two point generation was necessary. He felt that epicyclic gears would be more widely used when greater accuracy and quieter running was attained by advanced methods of manufacturing the gears and rings.

Authors' Reply

The authors would like to thank all those who had shown an interest in the paper and especially those who had contributed to the discussion, either verbally or in writing. Before making individual replies they wanted to reply in general terms to the comments made on Table I.

This table had been criticized on the grounds that it did not compare like with like. As Mr. Weir had mentioned, this fact had been clearly pointed out in the paper. Table I compared the type of parallel shaft gear in general use with the type of epicyclic gear used for the last twenty years in Germany and now being used in this country. However, to meet this criticism, Table III had been compiled comparing "soft" parallel shaft, hardened parallel shaft and hardened epicyclic gears, taking two examples from Table I and Mr. Everest's example from Table II. In Table III, gears had been specially designed for the application given, whilst in Table I the authors' nearest available standard gear had been used. It was hoped that enough information had been given in Table III to show that, as far as possible, the hardened parallel shaft gears and the epicyclic gears were comparable.

As far as possible K had been made the same in each example and $K=90$ had been taken for soft materials except for example 3, in which the annulus K was much below this figure as the gear was for a high ratio. A common maximum bearing pressure of 270lb. per sq. in. had been taken in all cases. If more usual bearing pressures, of the order of 150lb. per sq. in., were used for the soft parallel shaft gears, their efficiencies would be reduced to 97·3, 97·5, and 97·5 per cent in examples 1, 2 and 3 respectively.

Conversely, the efficiencies of the epicyclic gears would be increased by making their bearing pressures nearer to those given in Table I, which showed more normal practice. It would be seen from Table III that in every case the hardened and ground epicyclic gear had smaller dimensions, weighed less and was more efficient than the equivalent hardened and ground parallel shaft gear. In example 1 the savings were considerably more than those suggested by Mr. Everest and the bearing pressure was half that given in his example.

All the gears in Table III were designed solely for the purpose of compiling the table. Furthermore, the authors would suggest that the gear wheels in examples 2 and 3 would present considerable difficulties in hardening and quenching as well as being large for grinding. The wheel for the 4,360 h.p. gear had a pitch circle diameter of 32 inch and that for the 1,060 kW. gear a wheel of 37-in. pitch circle diameter. The corresponding largest wheels to be hardened and ground in the equivalent epicyclic gears were 8-in. and 10-in. pitch circle diameter respectively. The differences in the epicyclic gears quoted in Table II and Table III arose from the fact that the authors had designed the epicyclic gears in accordance with their usual practice based on successful running experience, whereas Mr. Everest had designed an epicyclic gear from B.S.S.436. The authors would suggest that epicyclic gears had not been considered when this British Standards Specification was drawn up and it did not take into account some of the most important considerations of epicyclic gear design. In fact, it was not even the British Standard for parallel shaft turbine gears.

It was quite evident that there were considerable savings to be made by the use of hardened parallel shaft or epicyclic gears. If, as Mr. Everest said, design experience and manufacturing facilities for hardened parallel shaft gears were available in this country, it was most regrettable that in the interests of national progress and economy wider use were not made of them in those cases where they offered the best solution. The authors were inclined to share Herr Stoeckicht's view that there was only one firm in the world who made large marine and industrial gears with hardened and ground wheels and that firm was not in this country. In this connexion, a point worth considering was that, for example, in a 10:1 hardened and ground parallel shaft gear, the wheel would be ten times as large as the pinion. In a 10:1 planetary gear the planet wheel would be only four times the size of the sun wheel, which itself would be smaller than the pinion of a parallel shaft gear.

Dr. Forsyth's contribution was extremely interesting and he had obviously gone to a great deal of trouble, particularly in preparing slides of what appeared to be a large, and possibly experimental, gear. The authors had attempted to give a broad survey in the simplest possible terms, and they had considered surface stress in terms of the Lloyd's Register of Shipping formula for K as this seemed appropriate in a paper for marine engineers. It gave people a comparison in terms they were used to, but in fact the authors did not use this formula directly when calculating epicyclic gears. K, of course, varied inversely as the relative radius of curvature. In keeping within accepted K values it was known that the Hertz stress was within acceptable limits. Dr. Forsyth would have noticed that in Table I the K values for the annuli were all within Lloyd's specification for "soft" gears. They were in agreement with him that the square root of the relative radius of curvature should be used when considering Hertz stress, and that this stress on the annulus was more correctly expressed as $\frac{1}{\sqrt{\text{ratio}-1}}$ multiplied by the Hertz stress on the sun wheel. Dr. Forsyth's illustrations, Figs. 33 and 34, helped to make this point clear.

The authors had not sufficient information about Dr. Forsyth's gear to give an opinion of its factors of safety, but they could say that the K values given in Table I were conservative. It was the authors' practice to use conservatively rated epicyclic gears which even so had considerable advantages.

Dr. Forsyth's view that failure by bending must be avoided at all costs was fully shared by the authors. A worn tooth would continue to run but a broken one might well prove disastrous in any type of gear.

The machine illustrated in Fig. 40 appeared to be extremely ingenious and the authors would say that for large gears it would provide an excellent means of finishing the roots of the teeth.

They had used various grades of nitriding steel and the position of maximum stress had always been well within the case depth. Their experience had been confined to small gears in this country but they knew of nitrided gears in Germany which had been used very successfully to transmit several thousand horsepower. The authors would shortly be running some 2,000 h.p. gears using this method of hardening.

Figs. 41 and 42 amplified very well the authors' statement about bearings for single helical gears. They always preferred to use spur or double helical gears unless there were special reasons for using single helicals. Figs. 44 and 45 showed what could happen if proper care were not taken in the design and manufacture of single helical planet wheel bearings. Figs. 46 and 47 illustrated infinitely better arrangements. The authors were pleased to see Dr. Forsyth's evidence of more than three planet wheels sharing the load, and Fig. 48 showed how good had been the load sharing between the five planet wheels in Dr. Forsyth's gear designed on Stoeckicht principles.

Although the authors had no experience of the limit of horsepower for the method of pinion support shown in Fig. 49, they could assure Dr. Forsyth that a similar method had been successfully used for speeds up to 18,000 r.p.m. transmitting several thousand horsepower.

Dr. Forsyth's general remarks amplified the authors' conclusions.

Mr. Archer was quite right in assuming that many of the designs shown were in course of construction. Actually all the gears illustrated in the paper, with the exception of Figs. 26 to 30 and 31 and 32, were either building or actually in service. The gear illustrated in Fig. 12 had been running at sea since November 1950 and had been completely trouble free. The following figures of Stoeckicht main propulsion gears for small German craft were possibly of interest. From 1933 to the end of 1950, 2,324 gears for main propulsion were delivered. The total horsepower was 185,000 and the size of the gears varied from about 20 to 850 h.p. These gears were for what might be termed normal duties, and did not include the 5,000 h.p. and one or two other special gears.

TABLE III—TO BE COMPARED

Example		1						
Application originally illustrated in figure number		50						
Duty		Turbo-generator						
Power		500 kW.						
Speeds in r.p.m.		6,500/1,800						
Ratio		3·61 : 1						
Type of gear		Parallel shaft top pinion		Parallel shaft top pinion		Epicyclic planetary		
Weight in lb.		720		560		370		
Weight in lb. of largest component to be handled		430		240		200*		
Component		Pinion	Wheel	Pinion	Wheel	Sun	planet	Annulus
Materials		2¾-3¾ per cent nickel steel 40-45 ton per sq. in. U.T.S.	Carbon steel 31-35 ton per sq. in. U.T.S.	Alloy case hardening steel 65 ton per sq. in. minimum U.T.S.				Alloy or carbon steel 55-65 ton per sq. in. U.T.S.
Brinell number		175 to 205	135 to 151	630 after case hardening				250 to 295
$K=\frac{P}{d}\left(\frac{R\pm1}{R}\right)$ (see note below Table I)		90		254		235		90
Face width, inch		2×4		$2\times2\frac{7}{8}$		2×1		
Circular pitch, inch		0·454		0·454		0·302		
Pitch circle diameters, inch		4·907	17·754	3·464	12·558	4·042	3·271	10·585
Relative pitch line velocity in ft. per sec.		139		98·5		83		
Maximum bearing pressure in lb. per sq. in.		270		270		270		
Losses in horse power	Bearing	7·6		12·6		14		
	Tooth	4·2		4·2		3·1		
Efficiency, per cent		98·3		97·6		97·9		
Gearcase sizes	Length, inch	25		22		17		
	Width, inch	22		17		$16\frac{1}{2}$		
	Height, inch	27		23		16		

* Weight of complete epicyclic gear internals to be removed from gear

In addition, 5,085 Stoeckicht bevel type reversing gears for main propulsion were delivered in the same period.

The authors could not say much about the 5,000 h.p. submarine gears but no doubt Mr. Michel's remarks would be of interest to Mr. Archer.

The errors shown in Figs. 14 and 15 were truly representative and the results were not specially selected.

The way in which the annulus rings were mounted appeared to act as a vibration damper but the authors agreed entirely that every effort should be made to reduce all errors to the absolute minimum practicable.

The authors had no quantitative data on noise but the noise level of all the epicyclics they had made had been satisfactory. Spur epicyclics had been made which were comparable with average helical parallel shaft gears. The gears illustrated in Figs. 2(a) and (b) were both extremely quiet, as were other similar ones in service. In the main, the authors would say that double helicals were the quietest.

Single helical gears were not regarded with favour by the authors and they preferred to use either spur or double helicals, the latter for larger powers, but it was difficult to draw a hard and fast line—each case had to be considered on its merits.

Mr. Archer and Captain Given were expressing the authors' own feelings in suggesting extensive shore trials for large prototype main propulsion reversing gears. Pametrada had excellent testing facilities and would, the authors assumed, be able to simulate sea going conditions.

Mr. Couling was always a pleasure to listen to whether he was speaking for or against a particular subject, and the

WITH TABLES I AND II

2							3						
28							29						
Turbo-pump							Turbo-generator						
4,360 h.p.							1,060 kW.						
6,160/1,120							6,000/600						
5·5 : 1							10 : 1						
Parallel shaft side pinion		Parallel shaft side pinion		Epicyclic Planetary			Parallel shaft top pinion		Parallel shaft top pinion		Epicyclic planetary		
7,660		3,400		2,500			9,000		3,800		2,200		
3,660		1,730		1,600*			5,600		2,100		1,500*		
Pinion	Wheel	Pinion	Wheel	Sun planet		Annulus	Pinion	Wheel	Pinion	Wheel	Sun planet		Annulus
2¾-3¾ per cent nickel steel 40-45 ton per sq. in. U.T.S.	Carbon steel 31-35 ton per sq. in. U.T.S.	Alloy case hardening steel 65 ton per sq. in. minimum U.T.S.				Alloy or carbon steel 55-65 ton per sq. in. U.T.S.	2¾-3¾ per cent nickel steel 40-45 ton per sq. in. U.T.S.	Carbon steel 31-35 ton per sq. in. U.T.S.	Alloy case hardening steel 65 ton per sq. in. minimum U.T.S.				Alloy or carbon steel 55-65 ton per sq. in. U.T.S.
175 to 205	135 to 151	630 after case hardening				250 to 295	175 to 205	135 to 151	630 after case hardening				250 to 295
90		360		360		90	90		410		410		50
2×7½		2×4½		2×3			2×6		2×3½		2×2⅝		
0·605		0·907		0·605			0·605		0·725		0·518		
8·853	48·690	5·774	31·754	4·619	8·083	20·785	5·581	55·811	3·695	36·950	2·474	9·897	22·269
238		155		102			148		96·5		58·5		
270		270		270			270		270		270		
42		46·2		38			14		21		8·2		
26		26		21·4			9		9		8·1		
98·5		98·8		99			98·5		98		98·9		
60		42		36			46½		37		30		
70		48½		33			65		47		33		
54		36		32			72		48		32		

case and then dismantled

authors hoped that Table III would go some way towards satisfying his points.

The authors shared his view on the importance of load sharing but disagreed with his contention that it was easier to bore the two lines in parallel shaft gears than the single line in epicyclic gearboxes, quite apart from the size factor favouring the latter.

With reference to Mr. Couling's remarks about patents, the authors should perhaps have pointed out that whilst many of the gears mentioned were covered by Stoeckicht patents, that illustrated in Fig. 32 incorporated Pametrada patents.

Captain Given's views about the torsional flexibility of compound planet wheels were shared by the authors and this would be borne in mind in the design of large main propulsion reversing gears. He had emphasized some of the points the authors had in mind when they said that large main propulsion gears would require determined development. They hoped that Captain Given's other points had already been covered.

Mr. Everest's reasons why epicyclic gears had not been used for marine and industrial applications were not shared by the authors. To say that disadvantages and difficulties experienced in the past were necessarily inherent, reflected an attitude of mind which could only stultify progress. The authors had tried to demonstrate that the problems had been overcome and had given examples of applications in which such gears were being successfully used. Properly designed epicyclic gears had, in fact, been used for the last twenty years in Germany and there were, as he would see from other contributors' remarks, well qualified engineers who shared the authors' views about early manufacturing and load sharing problems.

Mr. Everest might find Table III of interest and the authors would like to point out that the advantages shown for their redesigned version of his epicyclic gear would tend to increase with the ratio. In considering his example of a 500 kW. turbo generator, it should be borne in mind that turbine speeds for this duty were now within the region of 10,000 r.p.m.

The soft materials used in Table I and criticized by Mr. Everest as being twenty years out of date were to Lloyd's Register's latest specification.

The authors agreed that the face width of an epicyclic gear sun wheel would not be one-third that of the pinion of a parallel shaft gear and no such statement was made in the paper.

Mr. Everest must surely agree that a given load at a given frequency would be preferable to three times the load applied at one-third the frequency. It was interesting to note that, whatever his reasoning, the epicyclic gear sun wheel in his Table II finished up by being smaller than the pinion of a parallel shaft gear in the same material.

The authors agreed that a shallower condenser would be longer and/or wider than that shown in Fig. 29. A longer one could, in fact, be used to great advantage for supporting both turbine and gear.

In pointing out that the total number of bearings was the same for parallel shaft and epicyclic gears, Mr. Everest had completely evaded the vital point that the two highest speed bearings had been eliminated.

It was agreed that the lubrication of planet bearings was important but in practice it had not proved as difficult as Mr. Everest suggested. It was well known that clean oil was important for the bearings of any modern machinery and filtered oil was advocated to reduce wear from abrasive particles. The inference that without filtration any of the bearings would seize was incorrect.

The production of double helical gears offered no difficulties and, far from being undesirable, the independent annuli proved advantageous. As a matter of interest, in addition to those already in service, some fifty double helical gears for turbine driven machinery were now under construction in Germany and this country, totalling 150,000 h.p.

It should be pointed out that the noise level for a given duty depended upon correct design and manufacture and the difference in noise level for the same duty between certain epicyclics with either spur or helical teeth had proved indistinguishable on test.

The authors were amazed that Mr. Everest should query the accuracy they had obtained with the planing process as illustrated in Figs. 14 and 15. These figures were both correct and typical and, although quite good, were not the best recorded. Efforts were constantly being made to improve the average still further.

In his summary, Mr. Everest enumerated five points of comparison, only two of which he had previously mentioned. Simplicity of design, the authors conceded for parallel shaft gears, but he had not substantiated his statements regarding overall dimensions, ease of manufacture, overall reliability and first cost.

Mr. Everest had appeared to be anxious to condemn epicyclic gears at all costs and the authors hoped their reply had demonstrated that some of his statements were, to borrow his own phraseology, "rather excessive and could not really be taken too seriously".

Dr. Merritt would appreciate that in the broad survey given in the paper it had not been possible to go very deeply into the finer points of design; that might well form the subject of another paper.

The authors hoped that Table III would appear a little more scientific than Table I, although they felt that he would agree that gearing design and manufacture were still an art as well as, if not more than, a science.

The authors were only too conscious of the fact that large brakes and clutches still required considerable development, which they hoped would not be long delayed.

Dr. Merritt had mentioned the increased stiffness of small gearboxes and this was indeed a most important point.

The authors were most grateful to Dr. Merritt for his contribution, which brought to the discussion a wealth of authority and experience from the gearing world.

Mr. Ford's contribution was particularly valuable as it recorded a most arduous test which had been carried out by a completely independent authority, from which test the gears had emerged with flying colours. His remarks and those of Herr Stoeckicht were perhaps of considerable interest to other contributors who had wondered about the effects of varying torques on epicyclic gears.

The authors were sorry that the scale of figures 4(b) to 8(b) was so hard to find. It was, in fact, given by the annulus pitch circle diameter which was quoted, and the drawings were to scale.

In view of Mr. Ford's well-known association with torsionmeters, his remarks on the torque measurements possible would be particularly noted by marine engineers.

With regard to rotating annulus rings, the author's personal experience went as far as using them for speeds up to 4,300 r.p.m. The rings in question were about 14-in. diameter and there was no evidence of vibration.

The authors heartily endorsed Mr. Ford's views on the correct order of precedence for cleanliness and godliness for accurately made machinery.

Mr. Petty raised the question of the effect of centrifugal forces on lubrication and the authors could say that for rotational speeds up to 43,000 r.p.m. (which was the maximum speed at which they had run a gear) they had had no trouble. He would notice that the oil was led from the centre of the sun wheel to the teeth and did not have to be forced into the tooth mesh against centrifugal force. By carefully controlling the quantity of oil supplied, the authors had had no difficulty with excessive heating. They hoped that Mr. Petty's other points had already been covered.

Herr Stoeckicht had unrivalled experience in the application of epicyclic gears in marine, industrial and aircraft fields, and the authors were, therefore, particularly grateful to him for coming to England to take part in the discussion and amplifying with such authority some of the points which they had made.

Mr. Baker brought out some of the advantages of epicyclic gears which were not included in the paper, and the authors were pleased to see that he shared their confidence about the reliability of properly designed epicyclic gears. They agreed entirely that the correct annulus section must be used in all designs.

He was quite right in saying that the gear in Fig. 21 was designed for 450 kW. but as the set showed a 350 kW. generator, the authors had not mentioned this point.

Mr. Baker's additional remarks about the advantages of main propulsion reverse gears were matters to which marine engineers would no doubt give considerable thought.

Professor Blok asked about scuffing and the authors confirmed that they had had no trouble with this in their gears but were sorry that they had no quantitative data on the margin of safety. "Scuffing" and its causes were still very controversial subjects. There were many factors besides scuffing to be considered in limit designs but they had no intention of using limit designs for marine or industrial applications.

The phenomenon of fretting corrosion was well known to the authors and they were consequently very alive to this possibility in tooth couplings. They had not experienced trouble from this cause in their epicyclic gears.

Mr. Brading's opinion that the choice of steel for nitrided gears needed careful consideration was shared by the authors, but they had nevertheless run gears successfully with differing grades of nitriding steel. They felt that this success was largely due to the conservative rating of the gears. In their experience, Stoeckicht gears were much more likely to be noticeably quieter than other gears and the engine room staff of the ship in which the gear illustrated in Fig. 12 was fitted had remarked on this in no uncertain terms. This gear, which had spur teeth, had replaced a single helical parallel shaft gear.

Dr. Brown would find that the annulus pitch circle diameter for Figs. 4(b) to 8(b) was given with the description of the components of the gears.

The authors were pleased to see further advantages listed for main propulsion reverse reduction gears and Dr. Brown's remarks about the relative accessibility of epicyclic gears and the actuating mechanism of reversible propellers were very much to the point.

Mr. Davis's view that the time appeared to have come to use epicyclic gears for auxiliaries would be noted with interest as his opinion carried great weight in the marine gearing world.

His remarks about large main propulsion gears emphasized their own view that such gears would need determined development work and that in large reverse gears the clutches and brakes needed very careful consideration. They thought it only fair to add that they had done a considerable amount of design work on the gear shown in Fig. 31, which had led them to the conclusion that the problems could be overcome.

The importance which could be attached to savings of the order of 1½ per cent depended largely on the actual power under consideration. Mr. Davis would recall that, in his reply to the discussion on his excellent paper(4) in 1949, he had suggested that ¼ per cent might be of considerable significance!

Mr. Feilden's remarks about satisfactory running made a most valuable contribution to the paper and his illustrations, Figs. 57, 58 and 59, spoke for themselves. His comments on accessibility and examination answered some of the queries which had been raised about these points. In Table I the authors had shown that the complete epicyclic gear assembly was only one-sixth to one-ninth of the weight of a "soft" parallel shaft gearwheel for the same duty and in Table III it could be seen that it also weighed less than the equivalent hardened and ground parallel shaft gearwheel. The authors fully shared Mr. Feilden's confidence in the increasingly important rôle which epicyclic gears would play in future marine and land power development.

Admiral Maclean's remarks were much appreciated and the authors hoped that Table III went some way towards meeting his wish to see hardened parallel shaft gears compared with hardened epicyclic gears.

Mr. Michel's interesting contribution brought information of American experience with a 5,000 h.p. Stoeckicht gear which in itself answered some of the points raised by other contributors.

Mr. Sykes had amplified the authors' points about early manufacturing difficulties and as far as the general arrangement was concerned they had said that each case should be considered on its merits.

The first train of the gear shown in Fig. 4(b) was a star gear because of the high rotational speed of its output shaft.

It had to be borne in mind that with the larger number of tooth contacts in an epicyclic gear, each contact had less power to transmit than the tooth contacts of a parallel shaft gear.

Typical efficiencies of gears of the type shown in Fig. 8(b) with high speed shafts rotating at 1,500 r.p.m. were:—

Ratio	Efficiency, per cent
100:1	92
500:1	73
1,000:1	59
1,500:1	52

Mr. Sykes would appreciate that bearing load was only one of the factors to be considered in evaluating bearing loss.

Full load efficiency tests had been carried out on epicyclic gears. Some with which the authors had been concerned had special means provided for measuring the high speed bearing losses.

The strain gauge results shown by Dr. Forsyth, as well as their own, showed that load sharing was achieved. The records in Fig. 9 were of a 200 h.p. 30,000/3,000 r.p.m. gear. Record No. 15 was at 9,800 r.p.m., No. 5 at 20,300 r.p.m. and No. 12 at 29,200 r.p.m.

Perhaps the unsuccessful nitrided gears mentioned by Mr. Sykes had been overloaded. There were records of all types of gears failing and one must have the full facts before forming an opinion about the causes of failure.

The tip and root relief was achieved in lapping because the specific sliding was greatest at the tip and root.

The authors hoped that Mr. Sykes's points about quiet running had already been covered, and he would find that they had studied the design of planet wheel bearings in some detail in the paper.

With regard to maintenance, the 1,000 h.p. gear shown by Herr Stoeckicht in Fig. 57 had been in service for seventeen years and the maintenance had been nil.

Professor Tuplin's alternative method of calculating epicyclic gear tooth losses was most interesting. They felt that he would agree that even if the tooth friction losses were calculated by his method, potential horsepower could not always be ignored in the design of epicyclic gears, and it was possible that the unsuccessful designers mentioned by Dr. Merritt had not taken this into consideration.

Mr. Weir might be interested to know that a gear of 10,000 h.p. was being built at the present time in Germany for an industrial application. The authors knew no reason why this should be the limit.

As far as costs were concerned, there were so many factors to be considered that at this stage the authors felt it might be misleading to add to their statement in the last paragraph of the paper.

All the efficiencies given in the table were calculated by methods found successful from previous running experience.

Mr. Welbourn's opening remarks were most appropriate and the authors were delighted to see this acknowledgement to the Naval officers who had taken the initiative in investigating Herr Stoeckicht's work in Germany at the end of the war.

They entirely agreed that at times such as the present every endeavour should be made in every direction to conserve materials.

The authors hoped that the further explanation about Table I and the inclusion of Table III helped to clear up Mr. Welbourn's queries about the sizes of the epicyclic gears. The K values used for the sake of comparison were from the Lloyd's Register formula and any differences in pressure angle there were in the gears would have only a slight effect on them. Where considered desirable, it was the practice of the authors to correct sun wheels for torsional deflexion.

The authors were pleased to have Mr. Welbourn's opinion that their bearing pressures were conservative. In their broad survey, more subtle points such as oil film thickness had to be omitted. They agreed with Mr. Welbourn that the annulus must be made flexible but this necessitated mounting it flexibly in the casing, as by rigidly fixing it in the casing its flexibility would be lost. The authors entirely agreed that in large gears, or for that matter in small, every endeavour should be made to reduce annulus pitch-to-pitch inaccuracies.

Mr. A. G. Wilson's remarks were very welcome coming from an engineer who had had so much experience with the development and manufacture of epicyclic gears. The authors hoped that the additional information already given in replies to other contributors covered the points which he had raised.

Mr. H. A. Wilson would notice that, although the tooth pressures might appear higher, the quality of the material had been increased far more than the pressures. The authors agreed that occasionally any bearing might give trouble and this risk was very much reduced by using a filtered oil supply.

He quite rightly mentioned some of the problems which would have to be faced in the application of such gears for main propulsion and the authors entirely agreed that all these problems required very careful consideration.

Mr. Dannehower's statement that a high degree of accuracy was desirable in epicyclic gears was fully shared by the authors.

INSTITUTE ACTIVITIES

Minutes of Proceedings of the Ordinary Meeting held at the Institute on Tuesday, 11th March 1952

An ordinary meeting was held at the Institute on Tuesday, 11th March 1952, at 5.30 p.m. Mr. James Turnbull, O.B.E. (Chairman of Council), was in the Chair. A paper by Messrs. H. N. G. Allen, M.A., M.I.Mar.E., A.M.I.C.E., M.I.Mech.E., A.M.I.N.A., and T. P. Jones, A.M.I.Mar.E., A.M.I.Mech.E., entitled "Epicyclic Gears", was read and discussed. One hundred-and-one members and visitors were present and nine speakers took part in the discussion.

A vote of thanks to the authors was proposed by Com'r(E) J. I. T. Green, R.N. (Member), and accorded enthusiastically. The meeting ended at 8.10 p.m.

Southern Junior Branch I.N.A. and I.Mar.E.

On Tuesday, 13th May 1952, at 7.30 p.m., a meeting was held at the Technical College, Portsmouth, by the Southern Junior Branch for the presentation of the lecture entitled "The Marine Combustion Turbine", by Mr. J. Calderwood, M.Sc. (Vice-President, I.Mar.E.). Between ninety and one hundred members and guests were present, which was considered to be a very good attendance for the time of the year, when students were preparing for examinations.

Following the lecture, a film entitled "The Gas Turbine Goes to Sea" was shown, and proved to have been produced for a general rather than a technical audience; finally, after a very useful discussion, Mr. Calderwood was thanked for his most interesting and instructive lecture.

OBITUARY

ALFRED COE (Member 6674), born in 1901, was apprenticed from 1917-22 with Cammell Laird and Co., Ltd., Birkenhead. During this time he also attended the Holt Technical School, Birkenhead, and was awarded three first class certificates under Cammell Laird and Company's Technical School Bonus Scheme, and a first prize in 1920 for technical work. From 1922-28 he sailed as engineer in various vessels owned by Alfred Holt and Company and obtained a First Class Board of Trade Certificate.

In 1929 Mr. Coe joined Lloyd's Register of Shipping as an engineer surveyor and was stationed at Sunderland; a few months later he was transferred to Trieste and served subsequently in Falmouth, Bilbao, Genoa, Manchester, Balboa, Durban, and on the outdoor staff in London. In February 1946 he was appointed senior engineer surveyor, in Bilbao until October 1950, and then in Glasgow. He was elected a Member of the Institute in 1931. Mr. Coe died on 30th April 1952.

BERNARD LAWRENCE SILLEY (Member 5865), who died on 14th March 1952, was born in 1903. He was educated at Dulwich and St. Catherine's College, Cambridge, where he read Engineering, obtaining his Bachelor of Arts Degree. He served an apprenticeship with J. and E. Hall, Ltd., of Dartford, and R. and W. Hawthorn Leslie and Co., Ltd., and was at sea for a time with Alfred Holt and Company and the Union Castle Line before entering the service of R. and H. Green and Silley Weir, Ltd., of which company his father, the late Mr. John H. Silley, was chairman and managing director.

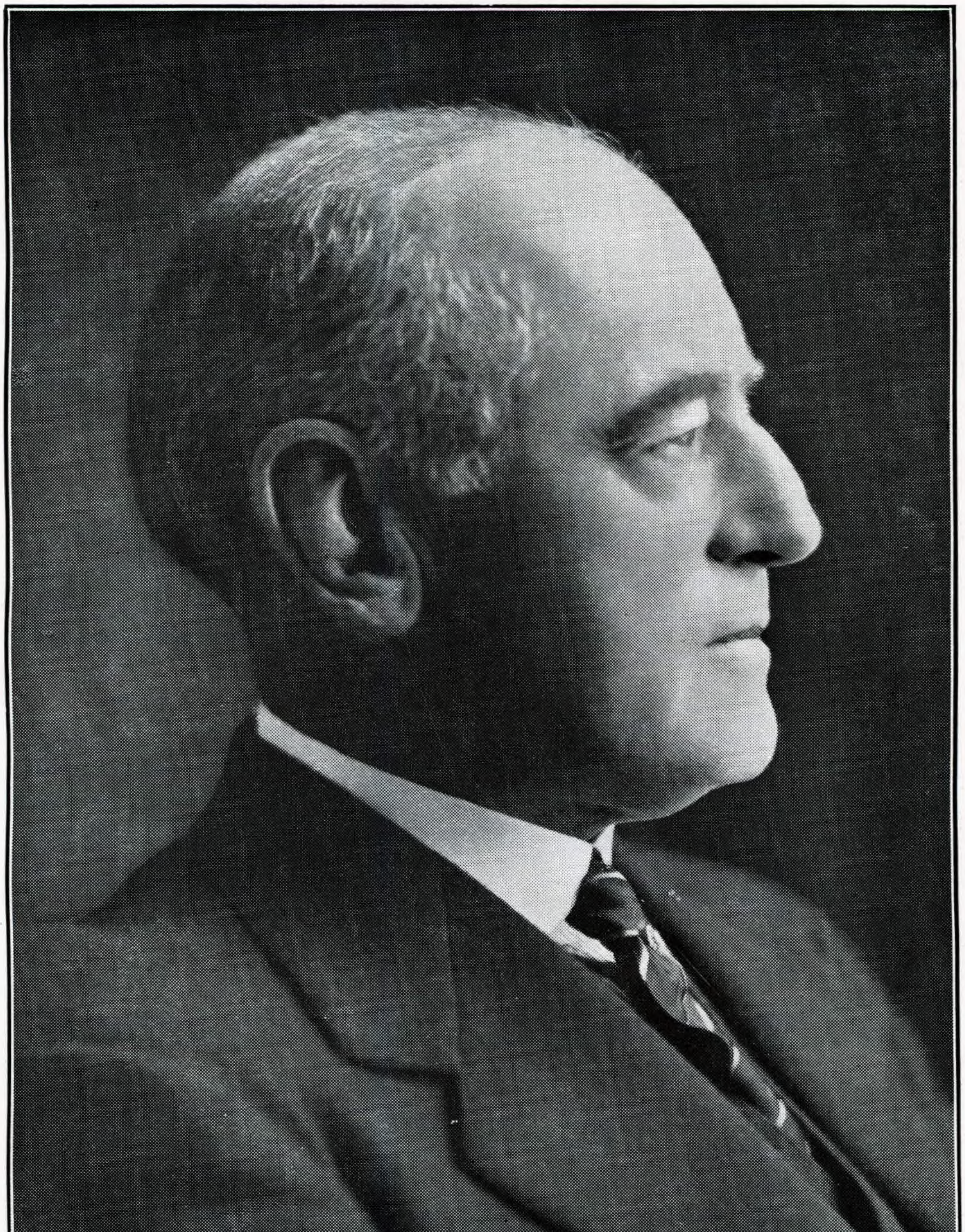
He did pioneer work in connexion with the experiments with pulverized coal, then being carried out by the company, and sailed in the first vessel ever to circumnavigate the world on pulverized coal.

Subsequently he took charge of the up-river section of the company's business, with headquarters at Blackwall Yard, and in 1942 took over the supervision of the building of tank landing craft, then being constructed in readiness for the Normandy invasion in 1944.

In 1943 he received a special commission in the Royal Engineers with the rank of Major and saw service overseas in Burma. It was whilst serving in Burma that he was taken extremely ill and it was more than probable that this illness was responsible for the breakdown in his health which occurred shortly after demobilization and which ultimately led to his early death.

Mr. Silley was a director of Silley, Cox and Co., Ltd., Falmouth Docks and Engineering Company, and a number of companies allied to R. and H. Green and Silley Weir, Ltd. He was elected a Member of the Institute in 1927 and was a generous supporter of the Guild of Benevolence, with which his father had been associated when he was President in 1934-35.

COM'R(E) THOMAS FIFE, R.C.N.(ret.) (Member 10508) was born in 1901 and served an apprenticeship with R. and W. Hawthorn Leslie and Co., Ltd. From 1922-37 he served as third and second engineer in various ships and obtained a First Class Board of Trade Certificate. From 1937-40 he was assistant to the chief engineer of the Bermuda Electric Light, Power and Traction Company. From 1940-45 he served in the Royal Canadian Navy, as Commander(E) from December 1943. At the time of his election to membership in 1945, he was manager of the engineering department of the H.M.C. Dockyard, Halifax, Nova Scotia. He was a member of the Engineering Institute of Canada. Commander Fife died suddenly on 23rd February 1952.



Sir Alan Garrett Anderson, G.B.E.

With acknowledgements to "Modern Transport"

SIR ALAN GARRETT ANDERSON, G.B.E.
President, 1928.

Sir Alan Garrett Anderson, G.B.E., Controller of Railways at the Ministry of Transport and Chairman of the Railway Executive from 1941-45, died in London on 4th May 1952, aged 75.

He was born in 1877, the son of Mr. James George Skelton Anderson and Dr. Elizabeth Garrett Anderson, who had the distinction of being England's first woman doctor. He was educated at Eton, a King's Scholar, and afterwards at Trinity College, Oxford. In 1897 he entered his father's firm, Anderson, Anderson and Company, joint managers at the time with Frederick Green and Company of the Orient Steam Navigation Company, and became a partner in 1900. In 1911 he was elected a director of the Midland Railway and when this later became the London, Midland and Scottish Railway he was appointed to the new board.

Two years after the outbreak of the First World War he was appointed vice-chairman of the Wheat Commission and chairman of the Wheat Executive, which controlled the supply of grain to Great Britain, France and Italy. In 1917-18 he was Controller in the Admiralty. Subsequently he became vice-chairman of the Food Council. When the United States joined the allies in 1917, Sir Alan was a member of Mr. Balfour's mission to Washington, and assisted in setting up the control of wheat in America and Canada. Shortly after his return from the United States he was appointed Controller at the Admiralty. In 1918 he became a director of the Bank of England and held the position of Deputy Governor during 1925-26. In 1923-24 he was vice-president of the Chamber of Shipping and president in the following year; his term of office was characterized by the time and energy which he devoted to the affairs of the chamber. He seconded the important resolution of flag discrimination at the Rome Congress of the International Chamber of Commerce; he was chairman of the Sea Transport Committee, and he figured largely in the deliberations which led to the Maritime Ports Convention. He was also chairman of the committee on wireless telegraph and life-saving appliances of the International Shipping Convention and of the wireless committee of the chamber. Among other important offices he held were those of vice-president and president of the International Chamber of Commerce, and president of the Association of Chambers of Commerce. He also served on a number of Board of Trade committees connected with industrial and economic matters.

Sir Alan was for some years a Governor of Eton College, a member of the Royal Commission on the National Debt, a member of the Fishmongers' Company, honorary treasurer of the London School of Medicine for Women and of the Royal Free Hospital, and a trustee of the Seamen's Pension Fund. He represented the City of London in Parliament from 1935-40.

Sir Alan was created a Knight of the Order of the British Empire in 1917 and Knight Grand Cross of the Order in 1934; he was an Officer of the Legion of Honour.

Commander Sir Robert Micklem, C.B.E., R.N. (ret.).

COMMANDER SIR ROBERT MICKLEM, C.B.E., R.N. (ret.).
President, 1948.

By the death of Sir Robert Micklem, C.B.E., on 13th May 1952, at the age of sixty, the engineering industry has lost a powerful personality.

Sir Robert was the younger son of the late Leonard Micklem and was educated at the Royal Naval Colleges at Osborne and Dartmouth. He served in the Royal Navy from 1903 to 1919, as Lieutenant (E) during the 1914-18 war, of which two years were spent in the Submarine Service. He retired from the Navy in 1919 and after serving several years with various companies associated with Vickers, Ltd., he was in 1944 appointed Deputy Chairman and Managing Director of engineering works and shipyards of Vickers-Armstrongs, Ltd., and Chairman in 1946. At the time of his retirement he was also Joint Managing Director of Vickers, Ltd., as well as Chairman of Cooke Troughton and Simms, Ltd., and Palmers Hebburn Co., Ltd., and was on the boards of several companies outside the Vickers group.

During the 1939-45 war, he served in 1942 as Chairman of the Regional Board, Northern Area, of the Ministry of Production, and in the same year his services were loaned to the Ministry of Supply. From 1942-44 he was Chairman of the Tank Board and also of the Armoured Fighting Vehicle Division of the Ministry of Supply. He was made a C.B.E. in 1942 and was knighted in the New Year's Honours List of 1946.

He was a Member and past Member of Council of the Institution of Mechanical Engineers, a Member of the Institution of Naval Architects, a Member and past Member of Council of the North-East Coast Institution of Engineers and Shipbuilders, and President of the Engineering and Allied Employers' National Federation for two years to February 1951. He was President of the Institute during the 1948-49 session and members associated with him during that period will recall the exemplary way in which he carried out his duties as President. The members of the Sydney Local Section have a special reason for remembering his year of office as they had the pleasure of his company at their inaugural dinner in Sydney on 18th January 1949.

Sir Robert's retirement was announced on 1st May 1952, only two weeks prior to his death. It is generally known that he was suffering from a serious illness for about two years before he relinquished his duties, but during this period he carried on in his usual energetic and cheerful manner.

JOINT PAPER

read before

THE INSTITUTE OF MARINE ENGINEERS *and*

THE INSTITUTION OF NAVAL ARCHITECTS

SOME RECENT STUDIES OF HUMAN STRESS FROM A MARINE AND NAVAL VIEWPOINT

By N. H. MACKWORTH, M.B., Ch.B., Ph.D.

Recent laboratory researches in experimental psychology have analysed the effects of the general surroundings on the working ability of human subjects. The paper deals first with some of the results from research studies aimed at discovering the effects of high atmospheric temperatures. Secondly, the effects of very low atmospheric temperatures on human performance are considered. Thirdly, the influence of prolonged general noise on watchkeeping ability is discussed. The rest of the paper considers the changes in accuracy and speed of work that arise from various local arrangements of the working task in contrast to these influences arising from the general surroundings. It is believed that very much can still be done to improve work design. Instances of such possibilities open to experimental proof have been taken from ways in which visual information can be supplied more effectively—and also from ways in which manual actions can best be utilized. But designing the task need not stop at sensory and motor considerations, and the paper ends with some points taken from researches on decision taking, with some indications on how activities of this kind can be helped or hindered.

Throughout the paper an attempt has been made to illustrate the approach used in considering such problems rather than to try to give any complete survey of the knowledge at present available. This information is now extensive—although it is scattered, rather specific and very incomplete, largely owing to the great complexities of human behaviour.

INTRODUCTION

Speculation on the nature of man has fascinated human beings for many centuries[22] but only for one century has man attempted to base his speculations on quantitative measurements of actual human behaviour. Two world wars have demonstrated this lack of reliable evidence on the effects of stress on human beings but it is only during the last fifteen years that there has been a considerable expansion in the subject of experimental psychology. In this country the attack has been led by Sir Frederic Bartlett from the Department of Experimental Psychology in the University of Cambridge with the help of the late Dr. Kenneth Craik.

Bartlett[3] has described how the Applied Psychology Research Unit of the Medical Research Council was established in the Psychological Laboratory, and he has also indicated the main interests of this Unit, which has received much help from the Royal Navy through the Royal Naval Personnel Research Committee of the Medical Research Council. The basic idea behind these researches is that the intellectual curiosity of most research investigators is aroused by attempting to solve an immediate practical problem. Once efforts have been made to find administrative answers to meet this practical need, further work is undertaken of a more basic nature to try to generalize the results. This link between the practical and the theoretical sides of research can perhaps be illustrated by the diagram shown in Fig. 1.

Should any general principles emerge from this basic research, attempts are then made to check these generalizations by selecting for study other specific experimental situations taken into the laboratory and modelled on real life tasks. This gives an opportunity for testing the appli-

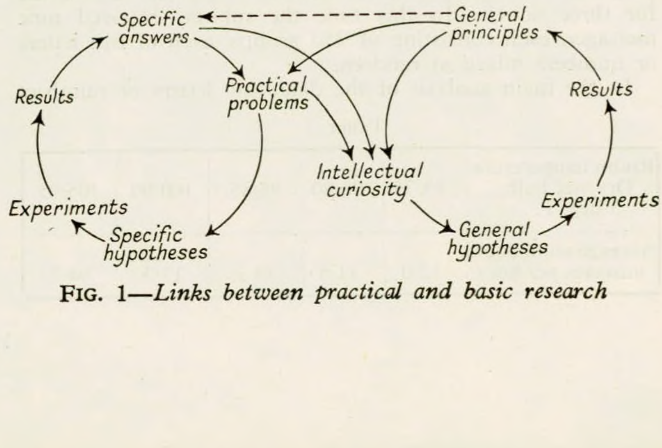


FIG. 1—*Links between practical and basic research*

cability of results which have been obtained on basic studies of forms of work which are unlikely to occur in everyday life. The left-hand circle of Fig. 1 represents practical studies in the real life situation or some simulation of this—and the right-hand circle represents basic studies in the laboratory with their many possible unrealities. There is therefore a cross-check, as it were, as well as a cross-fertilization between practice and theory.

HIGH ATMOSPHERIC TEMPERATURES

The studies of the effects of the thermal environment on human ability give an instance of this approach. Work was started in the Medical Research Council Unit at Cambridge, was continued in the joint Royal Naval and Medical Research Council Tropical Research Unit at Singapore, and is now starting up again in the Cambridge hot-room. In these latter researches support is being given by the Royal Navy and the Royal Air Force as well as by the National Coal Board.

The immediate practical problem was one that arose in the Royal Navy towards the end of the second World War. The problem was the extent to which it was necessary to provide air-conditioning equipment to maintain effective human performance. On what basis should air-conditioning equipment be allocated within ships, since such equipment was not only heavy and bulky, but also very scarce? Did human performance deteriorate at all with high atmospheric temperatures—at levels of the thermal environment in which men could exist? If so, was there any difference between the effects of heat stress on mental as compared with muscular work? If heat stress did have any such effect, how far could the environmental temperature be allowed to rise above the lowest temperature that air conditioning could be expected to produce under tropical conditions—i.e. before this heat stress began to produce an effect on the working ability of healthy young sailors who were dressed only in shorts, and well acclimatized to heat stress?

Many forms of human work have been studied in hot-rooms both in the United States[23] and also in this country[19]. Two typical studies have been selected from the mass of data, since one of these shows the effects on mental work and the other shows the effects on physical work. Fig. 2 (Plate 1) illustrates the task studied in the case of the mental work—the recording of Morse code messages received over the headphones at the high transmission rate of about one letter or number per second. Altogether eleven experienced wireless operators were given this wireless telegraphy test; most of the men had recently returned to Britain after working for several years in tropical countries. On their arrival at the laboratory they were given daily acclimatization spells for two to three months in temperatures ranging up to those experienced in the main tests. During this preparatory period the men also had further daily practice at wireless telegraphic reception. In the main experiments the temperature was set at any one of five atmospheric temperatures between 85/75 deg. F. and 105/95 deg. F. dry/wet bulb, the air movement being always 100 ft. per min. The test period for any one of the test days lasted for three hours. In this time the subjects received nine messages, each consisting of 250 groups, each of five letters or numbers mixed at random.

In the main analysis of the data, any letters or numbers

TABLE 1

Room temperature Dry/wet bulb, deg. F.	85/75	90/80	95/85	100/90	105/95
Average number of mistakes per hour	12·0	11·5	15·3	17·3	94·7

missed or wrong on the written answer sheets were regarded as mistakes. Table 1 shows that the average number of these errors per subject per hour showed a definite increase at the higher room temperatures.

The change in error score between 85/75 and 90/80 could have been a chance effect, but the rise in error score from 85/75 to 95/85 was statistically reliable at the $p = <\cdot 05$ level.*

Even within this group of highly trained men there were some differences in the working ability. The subjects were therefore arranged in order of merit on the basis of their results on the two lowest room temperatures, i.e. most nearly approximating their "normal" working environment. The best three men were arbitrarily classed as being very skilled, the next five as very good and the last three as competent. Fig. 3 and Table 2 show how the level of operating efficiency

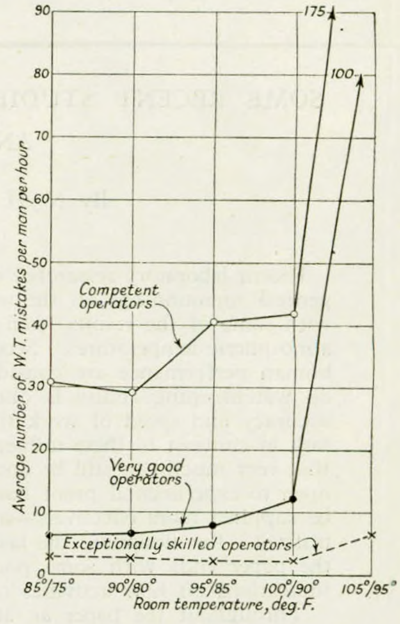


FIG. 3—*Level of ability influencing deterioration in hot and humid atmospheres*

determined the extent of the work deterioration resulting from a given atmospheric temperature, again in terms of the average number of mistakes per subject per hour.

TABLE 2

Category	Room temperature (Dry/wet bulb, deg. F.) 85/75	90/80	95/85	100/90	105/95
Exceptionally skilled	2·9	2·7	2·5	1·9	6·2
Very good	6·2	6·6	7·9	12·0	99·8
Competent	30·9	28·7	40·7	41·8	174·9

It is possible to criticize such studies by pointing out that under these laboratory circumstances the incentive under which the men were working was not sufficiently strong. To gain some evidence on this valid objection, two contrasting levels of incentives were introduced in the analysis of the effects of heat stress on the physical effort task given to thirty Naval ratings. It was, in fact, of great importance to discover the extent to which the harassing

* i.e. the odds are less than five in one hundred that the difference is a chance result due to random factors in the experiment.

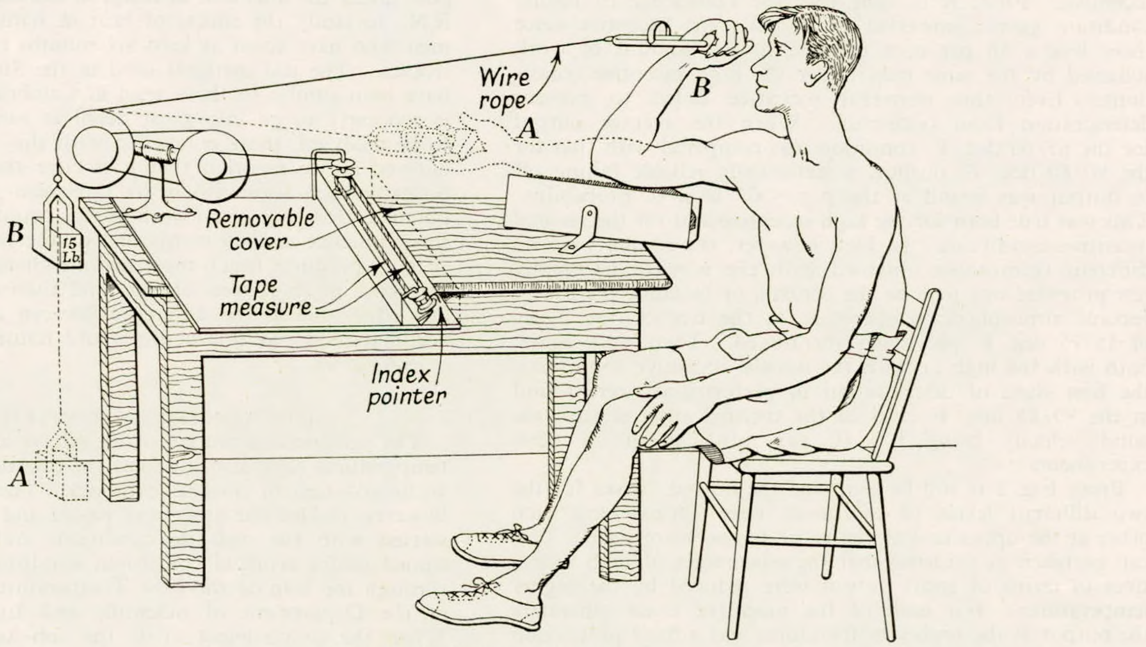


FIG. 4—*Pull test apparatus*

effect of a given environment changed, if at all, when one altered the incentive. Some held the view that a powerful incentive would prevent any such decline in ability occurring under these levels of stress. Others, however, felt that a more likely hypothesis was that the subjects *would* deteriorate even with this powerful incentive, but that they might deteriorate to a lesser extent for the same level of stress. It was therefore possible to regard the resulting experiment devised to test these views as a struggle between the effects of high incentives and the effects of hot and humid atmospheres, the results of which would be expressed in terms of muscular output at a physical effort task.

Fig 4 gives the test situation in which the subjects had to raise and lower a 15 lb. weight by bending and stretching the arm in time to a metronome. The subject was asked to continue to do this until completely exhausted, until he could not lift the weight even a fraction of an inch off the floor. This meant that about 30 ft. lb. of energy were being expended every two seconds in raising the weight. Direct measurement of the respiratory exchange by the bag method confirmed that the task as a whole did entail heavy work since muscular energy was being used at a rate of about 310 Kilo-calories per hour.

The thirty subjects first had a daily practice spell on the apparatus during a period of two weeks before the main investigation. For this period, too, the men had daily acclimatization spells in the hot-room, the room temperature then being between 100/90 deg. and 105/95 deg. on the dry/wet Fahrenheit scale.

For the main investigation each man had twelve experimental runs on twelve successive days. On successive pairs of these test days the men had one of the six given room temperatures with either normal or high incentive conditions. The air movement was always set at 100 ft. per min.

Under the ordinary incentive conditions the subject was asked to work until he felt that he could no longer continue. The experimenter made no comment on the amount of work done and the subject had no knowledge of results. Under the high incentive conditions the subject was given an output target figure which was set by the experimenter at a level 25 per cent higher than his previous best figure. This was expressed in terms of the reading on the simple cumulative adding machine made from a tape measure, shown in Fig. 4. This added up the amount of work done by adding the distance through which the weight had been raised since the start of the task. The experimenter gave a running commentary on the progress being made both in relation to this target figure and the achievements of other men known to the subject.

Fig. 5 summarizes the average results from the thirty men at each of the six room temperatures and the two levels of

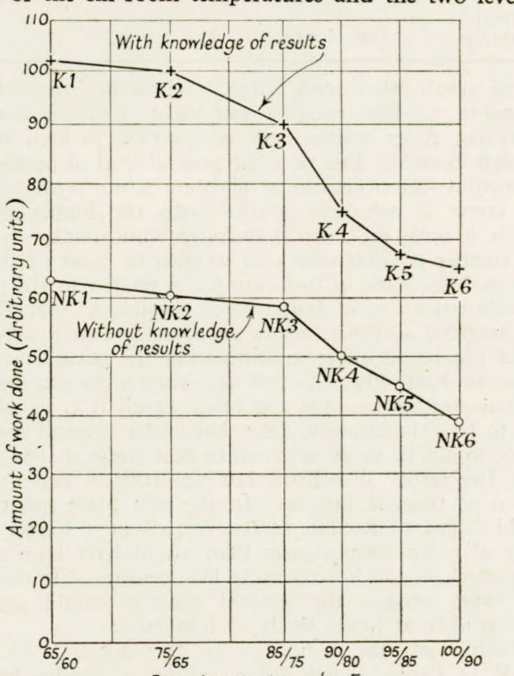


FIG. 5—*High atmospheric temperatures reduce the advantages of strong incentives in physical work*

incentive. First, it is clear that the knowledge of results condition gave a powerful additional spurt incentive since there was a 58 per cent increase in the amount of work achieved by the same men under the high incentive conditions. Even this powerful incentive failed to prevent deterioration from occurring. When the average output for the 65/60 deg. F. condition was compared with that for the 90/80 deg. F. output, a statistically reliable falling off in output was found at the $p = < \cdot 05$ level of probability. This was true both for the high incentive and for the normal incentive conditions. In fact, however, the results were no different from those obtained with the wireless telegraphy test provided one took as the control, or baseline, the lowest feasible atmospheric temperature in the tropics, the figure of 85/75 deg. F. previously mentioned. Then, once again, both with the high or with the normal incentive conditions, the first signs of deterioration in performance were found at the 95/85 deg. F. level of the thermal environment, the wind velocity being 100 ft. per min., as in all these experiments.

From Fig. 5 it will be seen that the output curves for the two different levels of incentives were approaching each other at the upper end of the room temperature scale. This can be taken as evidence that the advantages of high incentives in terms of spurt output were reduced by the higher temperatures. For each of the incentive levels separately the output at the higher temperatures was a fixed proportion of that at the normal room temperature, as will be seen from Table 3, which gives the relative amount of work done at different room temperatures. For each incentive level the amount of work performed is expressed as a percentage of the output at the 65/60 deg. F. room temperature level.

TABLE 3

Incentive conditions	Room temperature, deg. F. 65/60	75/65	85/75	90/80	95/85	100/90
High incentive	100	98	88	73	66	63
Normal incentive	100	96	92	80	71	60

This result, that with either level of incentive the men are going to lose roughly one-third of their output at 65/60 deg. room temperature, means that, in fact, although the high incentive does raise the general level of performance, the output deterioration in absolute units with the same heat stress is definitely greater with the higher incentive than it is with the normal incentive conditions.

A further possible objection to climatic work of this kind done in hot-rooms in this country is related to the problem of acclimatization to heat. It is possible to take the view that artificial acclimatization, such as these men were given, would not be adequate in simulating the natural acclimatization to heat which people are known to acquire under tropical conditions. On the other hand, it is equally possible to take the opposite view that under tropical conditions people might be more sensitive to heat stress if they suffered from the rather ill-defined but nevertheless real condition known as tropical fatigue. In the first place, performance would begin to decline in the tropics at a higher critical range of room temperatures than might have been expected from work in the hot-room in this country. If the second view were correct, the tropical subjects would deteriorate more readily at lower levels of heat stress.

Much work has been done on this question, mostly by Mr. R. D. Pepler[21] after some preliminary studies by Dr. A. Carpenter[8]. Both these Medical Research Council investigators have been working in the joint Royal Naval and Medical Research Council Tropical Research Unit at Singapore under the direction of Surgeon Commander F. P. Ellis, R.N., to study the effects of heat in naturally acclimatized men who have spent at least six months to one year in the tropics. The test methods used in the Singapore hot-room have been similar to those tried at Cambridge. Although it is too early to go into great detail as some data have still to be analysed, those concerned with this work have kindly allowed me to mention that it is clear that the findings in Singapore are substantially in agreement with the evidence obtained from hot-room work in this country. For example, this influence already mentioned of the initial level of skill (Fig. 3) produces much more definite changes in human performance in short tests of this kind than does, for instance, any effect due to the difference between artificial hot-room acclimatization in this country and natural acclimatization at Singapore.

LOW ATMOSPHERIC TEMPERATURES

The performance studies of the effects of low atmospheric temperatures have also involved the question of the possible acclimatization to climatic extremes. These problems were, however, tackled the other way round and the investigations started with the real life conditions and were then continued under artificial cold-room conditions at Cambridge, through the help of the Low Temperature Research Station of the Department of Scientific and Industrial Research. When the psychologist visits the sub-Arctic, he finds no shortage of problems to investigate. Even when he goes there with the intention of bringing back at least some quantitative data, he still finds many problems which he might study. When, however, the investigator has this approach of obtaining data by actual measurement rather than solely by an analysis of expressed opinion, he finds he must concentrate his attention on one small aspect of human behaviour and limit his study to one very small part of the vast range of possible investigations.

Some recent researches have therefore concentrated on the simple fact that human fingers become numb in very cold winds. This impairment of touch by bitterly cold winds is an interesting phenomenon known to everyone. But it is perhaps less obvious that this effect can be put to work to analyse the results of changes in a cold environment. The main aim of the work has been to determine whether an index of finger numbness could assess the effect of the two main physical factors in wind chill—the effects of the low atmospheric air temperatures and the effects of raising the wind speed. The second aim of the research has been to find whether some such numbness index could give any indication of the presence or absence of acclimatization due either to natural or artificial cold conditions.

A biological index of numbness has clearly some meaning in terms of human ability whenever men must work with their bare hands in cold atmospheres. The very tasks for which gloves must be removed are those that need good manual dexterity. Accurate finger movements often depend on good touch sensitivity, particularly when vision cannot guide the fingers at the same time. This absence of vision can clearly arise in many situations, as widely different as darkness, blinding snow or from attempts to repair equipment badly designed for easy maintenance.

The field studies were made in Canada, at Fort Churchill near the Hudson Bay. Tests were given on days with atmospheric temperatures which were termed either *cold* or *very cold*. These conditions were respectively within the range of −25·1 deg. C. to −30 deg. C. or between −30·1 deg. C. and −35 deg. C. On the Fahrenheit scale, *cold air* represented 45 to 54 degrees of frost, whereas the *very cold air* was appropriately named since it represented 54 to 63 degrees of frost. Test days were chosen and the shutters of a primitive wind tunnel adjusted so that any one of

FIG. 2—*Wireless telegraphy test in the hot-room*

FIG. 6—*V-test apparatus in sub-arctic trials*

Plate 2

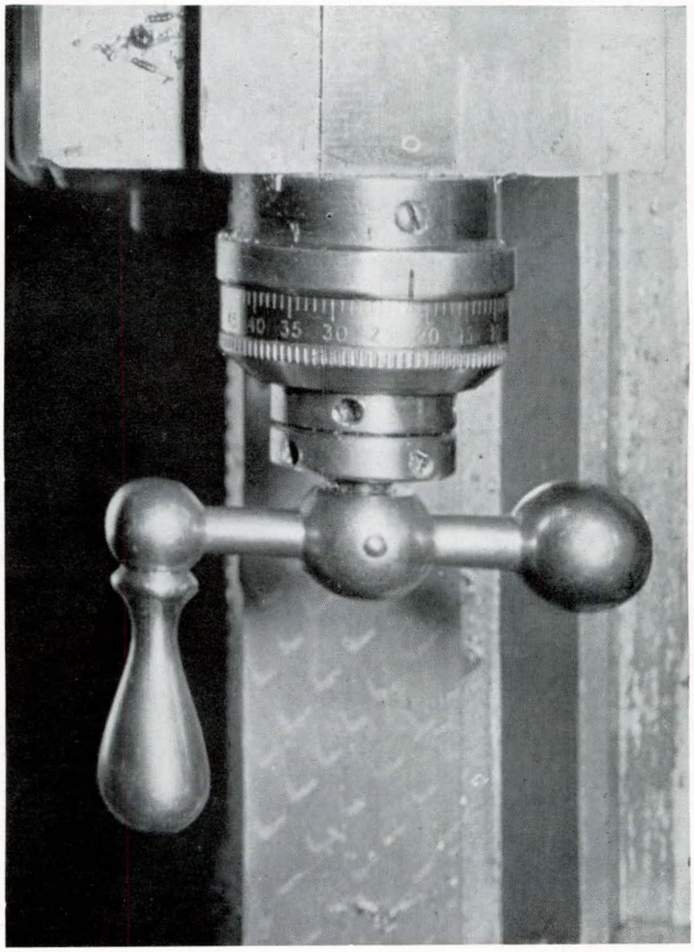


FIG. 10—*Dial face display*

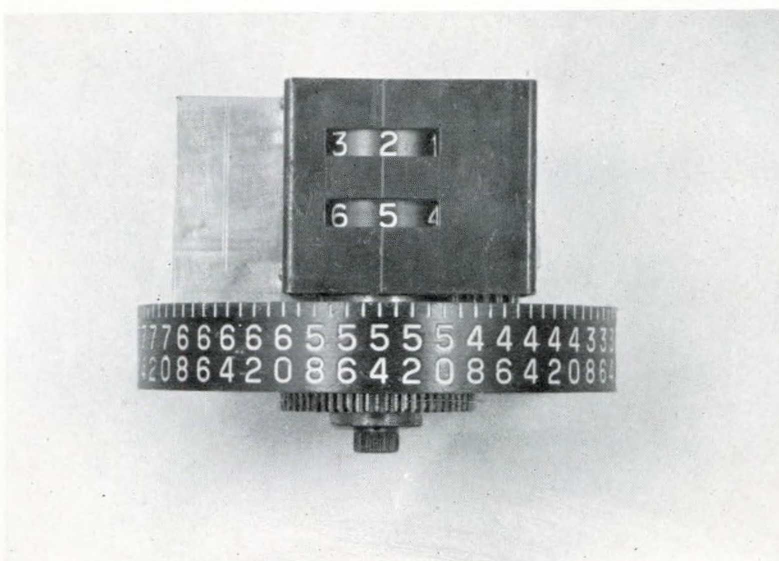


FIG. 11—*Counter display*

five different wind speeds could be experienced. These were in steps of two miles per hour from 0 - 2 m.p.h. to 8 - 10 m.p.h. inclusive.

Thirty-five volunteers were studied in the first series of cold tests. Three-quarters of the men had been in the sub-Arctic for less than a year and the others had been there between one and two years. The experimental apparatus consisted of the V-test device and this measured the size of the smallest gap between two sharp wooden edges which the subject could just detect by touch alone when pressing with the tip of his forefinger while resting in a warm room. The subject then entered the wind tunnel and stood sideways to the wind-stream and held his hand as shown in Fig. 6 (Plate 1) at right-angles to the direction of the wind. This hand was covered by a special woollen glove without any index finger, and therefore only the test forefinger was left entirely bare for the cold exposure, which lasted for three minutes. During this exposure a further series of readings was taken by the V-test device which measured the size of the gap required to experience the feeling by touch alone that any gap was present at all. Each man then returned to the warm test room and a further series of recovery readings was taken for one hour after exposure.

For each person the initial gap size required in the warm room was subtracted from each of the gap sizes required subsequently by that person. The numbness index was therefore the average increase required in the physical gap size to preserve the tactile impression of the gap despite any numbness. The average increase was the mean reading for several people for each environment. The first reading was the average "increase" required in a further test in the warm room and really tested the reliability of the technique since all these first points should have fallen on the zero line.

Fig 7 summarizes the results and this compares the numbing effects of the wind speeds and air temperatures experienced. Changes in the wind speed had just as much effect as did changes in the air temperature, even within the narrow range of wind speeds which could be tested in these trials. The presence of the higher wind speeds intensified the effects of the lower air temperatures. Since the wind could be divided into the *calm* conditions (0 - 6 m.p.h.) and the *breeze* conditions (6·1 - 10 m.p.h.), it was possible to show, as in Fig. 7, the numbness trends found with the four main environmental conditions: calm-cold, calm-very cold, breeze-cold, and breeze-very cold. It is clear that during the actual exposure the drop of a few degrees in the air temperature was much more important under breeze conditions than it was under the calm air conditions. The difference between the average indices during the actual exposure was three times greater with the breeze conditions.

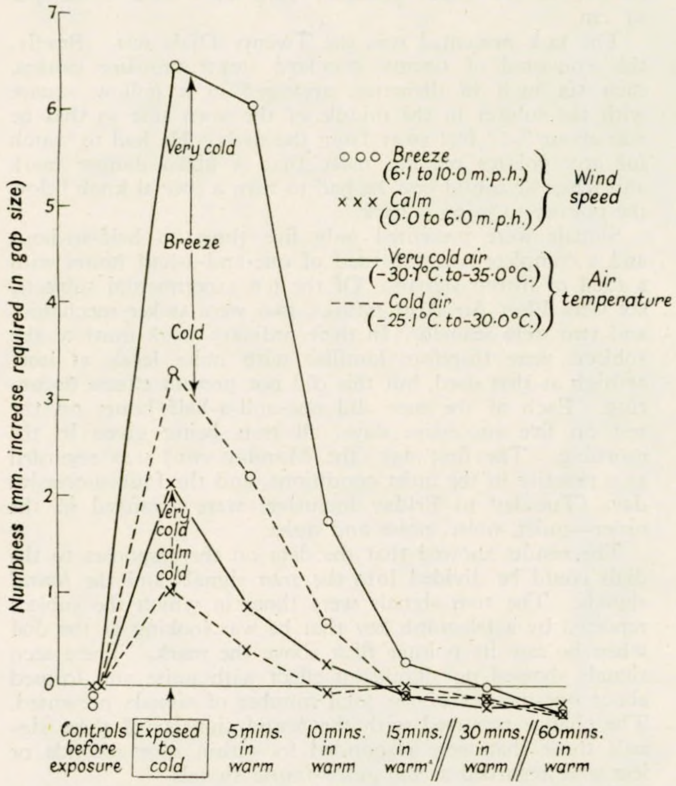


FIG. 7—*Trend effects of wind speeds and air temperature on V-test numbness*

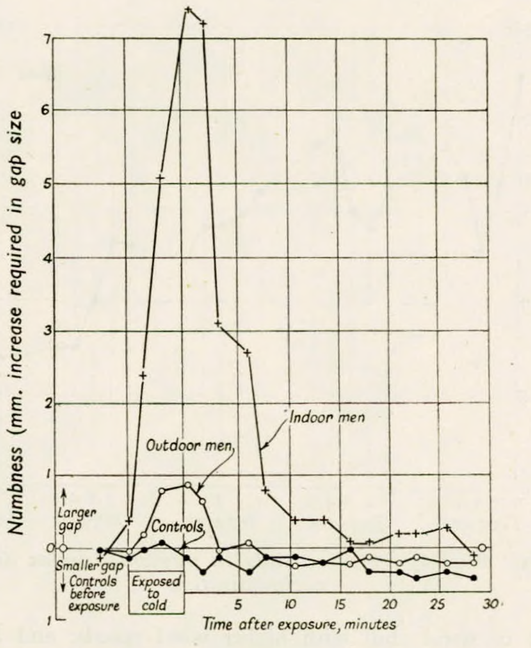


FIG. 8—*Numbness trends from indoor and outdoor men—exposure to slight wind*

As a pointer for the problem of acclimatization to cold it was noted in these tests that, *provided the environmental conditions were not too severe,* there was less numbness in those men whose duties normally required that they should spend more than half their working day out-of-doors or in unheated buildings. A second field experiment was therefore started with a further group of men. Ten of these subjects were not exposed to cold at all and acted as a control group which simply did the V-test repeatedly under warm conditions. Five outdoor men and five indoor men were exposed to a 4 m.p.h. wind under exactly the same circumstances except that the exposure was now six minutes. In men with this outdoor experience it is clear from Fig. 8 that a very considerably reduced amount of numbness was found. The gap increase required during the actual exposure was about one millimeter for the outdoor men and about seven millimetres for the indoor. With these particular environmental conditions it was noted that the outdoor men had a less marked fall in skin temperature and immediately after exposure their index fingers were three degrees warmer than those of the indoor men. It therefore seems possible that there is less marked vasoconstriction in the outdoor men under these wind chill conditions. This acclimatization effect disappears entirely at a more severe

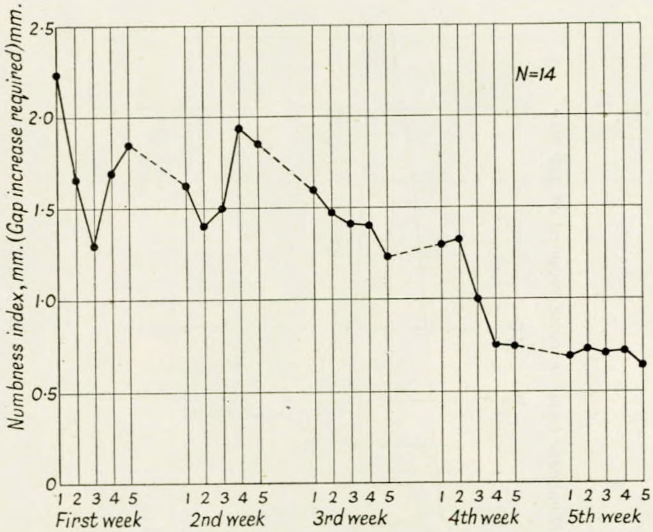


FIG. 9—*Day by day trends in numbness index during acclimatization*

level of wind chill with higher wind speeds, and is also much less marked under still air conditions.

Three possible criticisms can be made of these field experiments and further laboratory studies are now being done in an attempt to meet these objections. It might have been that these outdoor men were self-selected in some way which ensured that there would be a difference in temperament between the outdoor and indoor groups—a difference in temperament which could conceivably have affected the results in some way apart altogether from any possible acclimatization difference between these two groups. Secondly, the outdoor men might have been better at using the slight sensory cues coming from numb hands due to long practice at interpreting this faulty sensory information. Thirdly, the outdoor men may have been less afraid of exposing their hands to cold and therefore had less marked vasoconstriction of the finger blood vessels.

In these laboratory cold-room studies of the effects of daily exposure to artificial cold, fourteen Naval ratings have been exposed to -15 deg. C. two hours a day, five days a week for a total of five weeks. This daily exposure was given to the men dressed in full Naval rig but without any gloves and was always followed by a one-minute exposure of the test hand to a 6 m.p.h. breeze, also at -15 deg. C.

Fig. 9 shows the average readings for these men, in terms, as before, of the gap increase required to keep the tactile impression of a gap after the exposure to the fan, the reading before cold exposure being taken each successive day. It is clear that the numbness index shows a marked reduction during the fourth week of this series of tests. Since this cold-room result was produced in the same group of men, differences in temperament are not an essential factor in the development of this effect.

Evidence against the possibility of the field results being due to practice at using imperfect sensory cues, or to a difference in the amount of fear caused by the procedure, was found in a further series of cold-room studies. Under exactly the same conditions a group of six men was tested in the height of the 1950 summer which had maximum daily outdoor temperatures which were usually in the seventies. The numbness trend which was obtained in the same way as in Fig. 9 showed no downward tendency at all, presumably because acclimatization to the heat prevented cold acclimatization developing. The possibility of increased ability to use faulty sensory cues was further excluded by another series of cold tests in which eighteen men were given the V-test only at the beginning and end of the five week test period and this group showed numbness changes very similar to those in Fig. 9. It is therefore maintained that these differences between indoor and outdoor men found in the field studies were due to a real local acclimatization of the hands to cold. This acclimatization is useful in withstanding moderate amounts of wind chill such as might be experienced with good wind shielding, but it is useless in preventing frostbite in severe wind chill situations.

NOISE

Many authors have agreed, after reviewing the literature, that there is as yet very little evidence of any measurable effects due to prolonged high intensity noise on human behaviour.[5, 16] Some recent research at Cambridge questions this viewpoint mainly because much of the previous work has been on short-lasting samples of human behaviour and the experimental subjects used in previous tests have mainly been drawn from an undergraduate population. The author was greatly indebted to Mr. D. E. Broadbent, who is also a member of the Medical Research Council Unit, for his permission to quote from some very recent researches of his which have not yet been published[6, 7].

The noise was produced in a room sixteen feet square and eight feet high, built of metal plate and loaded to prevent resonance. In three of the walls loudspeakers were inserted, and these speakers were fed from a tape recorder playing over an endless loop of tape. The recording on the tape had been made near rapidly moving ship machinery and the spectrum obtained showed that this was a white noise. On *noise* days this was presented at the level of 100 db. and on *quiet* days it was given at 70 db. to mask any extraneous noises. Both these measurements were taken relative to the usual arbitrary zero of 0·0002 dynes per sq. cm.

The task presented was the Twenty Dials test. Briefly, this consisted of twenty standard steam pressure gauges, each six inch in diameter, arranged in a hollow square with the subject in the middle of the open side so that he was about 7-12 feet away from the dials. He had to watch for any pointer reading more than a given danger mark and when he found one, he had to turn a special knob below the pointer to bring it back.

Signals were presented only five times in half-an-hour and a complete run consisted of one-and-a-half hours with a total of fifteen signals. Of the ten experimental subjects, six were Fleet Air Arm ratings, two were stoker mechanics and two were seamen. In their ordinary work most of the subjects were therefore familiar with noise levels at least as high as that used, but this did not prevent effects occurring. Each of the men did one-and-a-half hours on the test on five successive days, all tests being given in the morning. The first day (the Monday run) was regarded as a practice in the quiet conditions, and the four successive days (Tuesday to Friday inclusive) were arranged in the order—quiet, noise, noise and quiet.

The results showed that the data on the responses to the dials could be divided into the *seen* signals and the *found* signals. The *seen* signals were those in which the subject reported by a telegraph key that he was looking at the dial when he saw its pointer flick above the mark. These seen signals showed no significant effect with noise and formed about one-quarter of the total number of signals presented. The change occurred with the *found* signals; of these signals those that were responded to within nine seconds or less were regarded as the *quick-found* signals.

Control runs under quiet conditions had previously shown that the first run should be regarded as a practice run. The relevant comparison was therefore between the second and fifth run in the quiet, and the third and fourth runs in the noise. The percentage of quick-found signals was 21 per cent on the practice. During the two *quiet* days it was 34 and 38 per cent, whereas on the *noise* days the proportion fell to 19 and 24 per cent. This difference between the percentage of quick-founds under quiet and noise conditions was statistically confirmed since a value of 3·21 was obtained when the $p = <\cdot 05$ was 2·26, and $p = <\cdot 01$ was 3·25.

Broadbent's interpretation of this change is that one form of impairment in continuous performance tasks appears as momentary lapses of attention which are experienced because the signals have occurred at times which the subject cannot predict. A strong competing stimulus such as general noise is thought to increase the tendency towards such lapses. One special feature of vigilance tasks is the subject's inability to tell when it is safe to relax his attention. The monotony of such tasks also makes it more likely that the part of the stimulus situation that is controlling response will change more often. Easier forms of vigilance tasks do not show this change since it is possible for the subject to redeem momentary lapses by a compensating spurt in the visual search provided that the signal remains on view indefinitely. The implications for visual display in noise are that signals should remain on view if full advantage is to be taken of this adaptive behaviour.

VISUAL DISPLAY

Much research has recently been done on the optimum arrangements for visual display. One instance that could be given is that of the experimental tests which have been made of the legibility of capital letters and numbers for the accurate display of information presented in aircraft control rooms. The results have some implications for the shape and size of any rooms in which vital information is displayed on wall boards or tables[2]. Extensive summaries now exist of some of the known facts on the visibility of scales and dials when the criteria are accuracy and speed of reading[9, 10]. Then again, Golds[14] has also recently drawn attention to the need for a performance specification of scales for laboratory and industrial instruments as well as for graphic recorders. He mentions many matters which would require consideration in specifications of this kind, such as the usual reading distance and angle of view, the required degree of accuracy and speed of reading, the dimensions of scale marks, the pointer size, the style and size of the characters on the scale and the colour of the scale and pointer. Several of the matters still need further research but much has already been well established by many workers about the effects of such changes in the display.

A recent experiment on visual display has been done by Gibbs[12] at Cambridge on the relative merits of the cyclometer type of display as compared with the dial type of display normally used in lathes and jig borers. The existing arrangement is usually that shown in Fig. 10 (Plate 2), whereby one complete turn of the dial equals 0·25 inch linear movement of the machine tool and each small graduation on the dial equals 0·001 inch linear travel. This device clearly has to measure both the extensive initial movement of the machine and its final accurate positioning. The cyclometer form of counter shown in Fig. 11 (Plate 2) has the advantage of giving a direct reading over the whole range of this movement, relative to a fixed datum point. It consists of a four-figure counter with manual setting to zero. An early version of this was first tried by being geared to the transverse shaft of a horizontal borer. Inches of travel appeared in the top window, tenths in the next and, finally, hundredths and thousandths of an inch at the bottom.

Ten men (who had had general engineering training, although they were not fully skilled machinists) were each given the task of boring ten holes at varying distances from a fixed datum, using both the dial and the counter methods on different occasions. The average time required to bore the holes was only 3 minutes 47 seconds with the *counter* as compared with 6 minutes 3 seconds with the *dial*. The difference in times mainly arose from the calculation times required to set the dial. When the two methods were compared on the setting accuracy, it was found that there was only one setting error with the *counter* compared with thirty-three setting errors with the *dial*. (Eighteen of these were initial dial setting errors and fifteen were caused by these faulty initial dial settings being repeated on subsequent occasions.) There were, of course, no calculation errors with the counter, but there were four such errors with the dial method. This was initially sponsored by the Human Factors Panel of the Committee on Industrial Productivity. Some studies were made at the Production Engineering Research Association and also at the College of Aeronautics in Cranfield. Much development work has also been done by English Numbering Machines, Ltd.

MANUAL CONTROLS

As with the visual display work, various monographs and summaries have now appeared on some of the factors in the most effective design of manual controls[4, 15]. For a recent illustration of the general experimental approach to problems of this kind I am again greatly indebted to Mr. C. B. Gibbs of the Medical Research Council Unit for permission to quote from unpublished researches on the advantages of pressure operated control levers in a velocity control system[11, 13].

A series of experiments has been done to compare the relative merits of two possible types of joystick control. One joystick was of the normal free-moving type; the other had strong spring centering but could be deflected slightly by applying manual pressure. In both instances the joystick controlled the movement of a spot of light (see Fig. 12) and the operator's task was to keep that light steady on a fixed index against the action of the equipment, which was always trying to deflect the spot away from the mark. In this continuous tracking task accuracy was usually measured in terms of errors of position which were added to one or more of four counters representing "high", "low", "right" or "left" errors.

Thirty-six Naval ratings were divided into two groups of equal tracking ability. Group A was then given the fixed or pressure control whereas Group B had the free-moving joystick. Fig. 13 shows that the average total error scores for Group A with the pressure control were definitely less than the errors made by Group B with the free-moving control during these initial runs on the first three days of

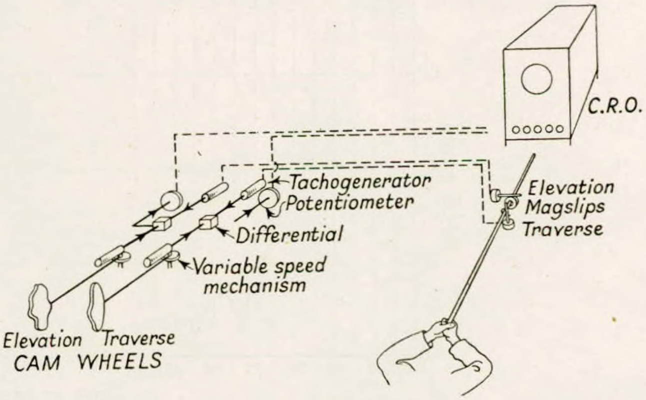


FIG. 12—*Diagrammatic layout of apparatus for joystick experiments*

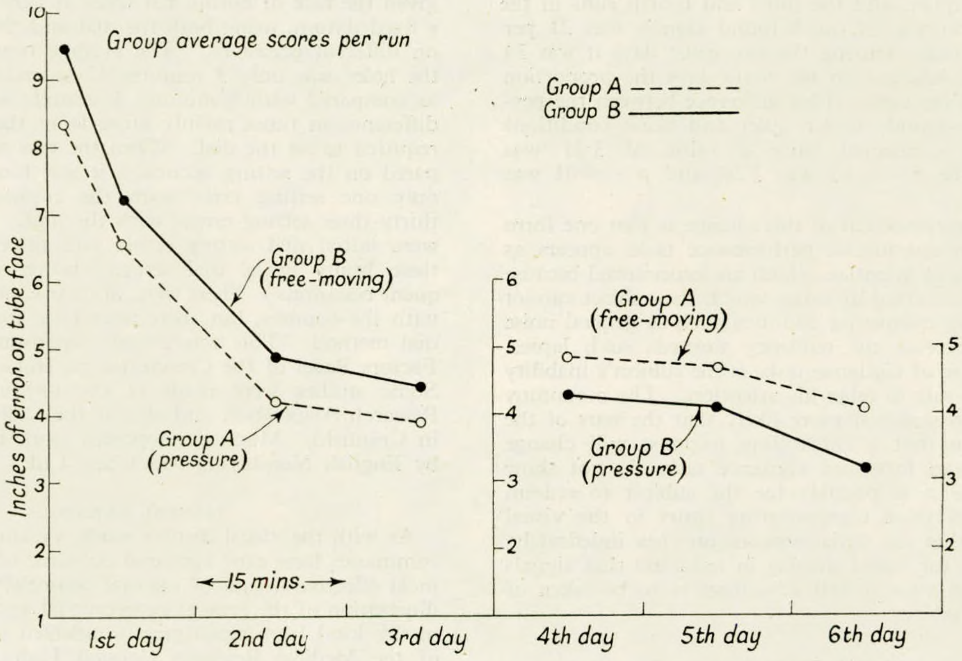


FIG. 13—*Learning curves in pressure and free-moving conditions*

the experiment. The test groups were then reversed, but it will be seen from Fig. 13 that even so the presssure control still held its advantages.

Gibbs has also made a more detailed analysis of the tracking performance from results obtained by continuous graphic records, and this helped to determine which particular parts of the tracking course (set by the machine controlled movements of the spot of light) showed this advantage in favour of the pressure control. It turned out that most advantage was found under two conditions: (1) when the spring loading assisted the human adjustment of the lever back to the reference point on the display, and (2) when large and unexpected changes took place in the target course. Fig. 14 brings out the point that the benefit derived from the pressure control was seen when the target was negatively accelerating.

Since the visual display is exactly the same for the two forms of control, this pressure control advantage is believed to be related to processes arising from sense organs in the muscles and joints. These sense receptors give the "feel" of a manual control and act rather like strain gauges to give information on limb position and movement. Physiological researches have shown that such sensory data are simpler when, as in the pressure control, muscles contract with little or no shortening. It is possible that muscle and joint receptors are relatively more important, and pressure controls have therefore this advantage over free-moving controls, (1) when the visual display lags behind the position of the

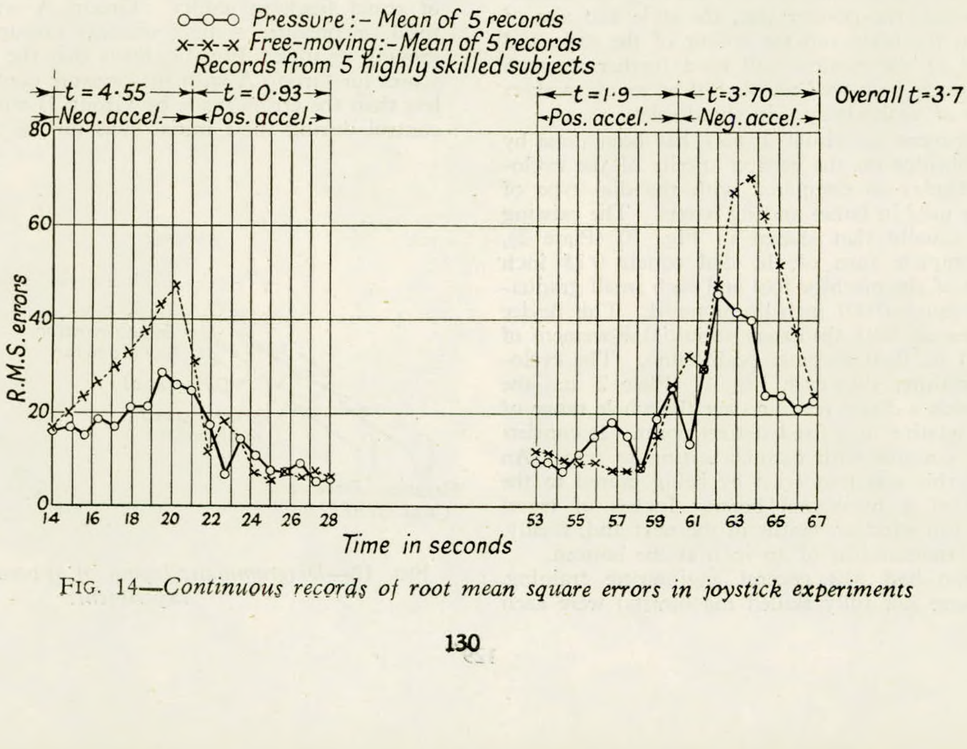


FIG. 14—*Continuous records of root mean square errors in joystick experiments*

control, as in this example where the control system was a velocity rather than a positional control system; (2) these receptors may also take over a more important role relative to vision when, again as in this case, the task is to keep a light on a fixed point rather than as in the alternative arrangement where the machine controlled pointer must be followed by another spot of light which represents the adjustment of the subject's control handle.

DECISION TAKING

The great advances made in recent times by communication engineers have made it possible in real life situations aboard ships to centralize vast masses of detailed information so that the officers taking decisions on such information are faced with tasks of very great complexity and responsibility. In this huge communication network of the modern ship there are, therefore, key points at which it is of the utmost importance that the decisions are correctly made. A great quantity of detailed information can be poured in on such officers from many channels of information—i.e. from many potential sources of demands for immediate action. It is sometimes questionable whether adding more and more information in great detail has any effect other than to obscure the really essential facts already being displayed, especially when some of the extra data are only slightly different versions of the same facts. The engineer should provide simplified summaries of such enormously complex information, or the user's brain will inevitably be overwhelmed with non-essentials.

In researches on problems of this kind psychologists are starting to follow the elegant injunction of Francis Bacon[1] (1605) on the need for a careful consideration of the limits of human ability . . . "It would greatly contribute to the encouragement and honour of mankind to have these tops or utmost extents of human nature collected from faithful history; I mean the greatest length whereto human nature of itself has ever gone in the several endowments of body and mind". A start has been made but the work has so far been little more than extrapolations from common sense in these attempts to define the main causes of difficulty in decision taking. The most obvious cause of such difficulty is clearly an increase in the demanded rate at which the orders have to be issued. The effects of this speed stress on accuracy of decision can readily be demonstrated. More

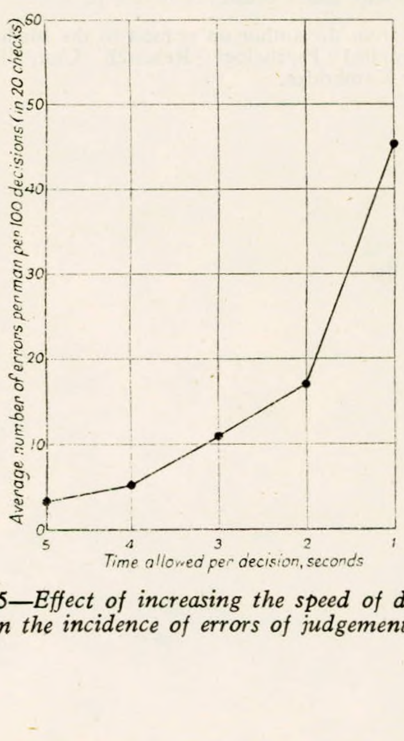


FIG. 15—*Effect of increasing the speed of decision on the incidence of errors of judgement*

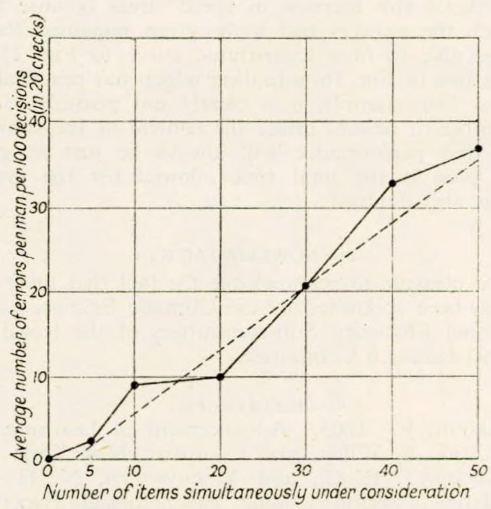


FIG. 16—*Effect of greatly increasing the load on the incidence of errors of judgement*

attention, however, is now being paid to difficulties in decision taking which arise from the fact that the officer is at the convergence point of a large number of incoming channels of information. The more the number of potential sources of demands for instant action, the greater will be the chances of three main difficulties arising in dealing with these immediate demands.

The first is the *crisis* difficulty which is due to increased opportunity for signals demanding instant action to overlap and appear on two or more channels at the same time. A burst of intensive work is followed by a quiet spell so that the speed stress as measured by the number of orders per minute may not appear to be high.

Secondly, serious difficulties can arise from the presence of *irrelevant information* in some of the many channels, i.e. superfluous as regards the immediate decisions. The third difficulty is the *disorder* difficulty when the officer has the problem of determining in which order he should take his decisions. He has so much data to search through that he cannot find the correct order in which to take the required decisions to give most chance of success. So far the deterioration in ability has been shown by failure to deal with the immediate present, but it has also been possible to demonstrate that a heavy mental load in the above sense will cause difficulties in thinking by limiting the range of immediate memory. Similarly, it is believed that improved research methods will make it possible to show handicaps on thinking which have arisen from the effects of a heavy mental load in reducing the range of anticipation span.

To illustrate the effects of changes in speed stress and load stress in decision taking, some results can be given from recent investigations. In this laboratory task the subject had to decide which of several courses of action should be undertaken and to determine which would be the most suitable order in which to take the decisions. The mobile objects under the control of the men doing this work had to arrive at the right places at the right times and this was possible only if the correct instructions were given. Errors were therefore expressed in terms of the average distance which separated the objects from their expected places of arrival at the expected times of arrival. Fig. 15 shows the effects of speed stress. As the time allowed per decision was reduced from 5 seconds to 1 second, there was a considerable rise in the error score. Fig. 16 gives the effects of load stress, i.e. the results of increasing the number of objects or potential sources of demands for action. This was without any increase in speed stress because the rate at which the answers had to be given remained the same. It is possible to fit a logarithmic curve to Fig. 15 and a straight line to Fig. 16, a finding which has practical implications. For example, it is clearly not possible to double the number of objects under the control of the subject and expect that performance will always be just as good as before, even if the total time allowed for the task as a whole is also doubled.

ACKNOWLEDGEMENTS

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Discussion

REAR ADMIRAL(E) F. E. CLEMITSON (Vice-President, I.Mar.E.) said that it was a privilege and a pleasure to open the discussion. At the same time, there must be many who were better qualified to do so than he, for he knew little about what might be called the pure research side. He knew something of the practical side, however, having been from time to time at the receiving end or what might perhaps be described as the hot end! He very well remembered the time when Surgeon Commander Ellis had come out to the Eastern Fleet to find out how hot it was and what could be done to make it a bit less hot. So far as the Royal Navy was concerned, he thought quite a lot had come out of that.

Dr. Mackworth was to be congratulated on a most important and absorbing paper, which had been extremely ably presented. The problems which had inspired the studies were very real practical problems and they were pressing problems for the Royal Navy, whose ships were liable to serve in many and varying conditions under increasingly strenuous conditions, particularly in war but even in peace. These same problems must also be of particular interest to the Merchant Navy. They affected everybody in the ship but other speakers would be better qualified than he to touch on the naval architecture aspect and he therefore proposed to confine his remarks mainly to the engineering aspect.

The engineer in both the Royal Navy and the Merchant Navy must be very concerned with these problems, and particularly with the effect on watchkeepers of high temperatures in machinery spaces. Low temperatures fortunately did not present quite such a problem to the engineering personnel—except those who had to deal with aircraft. At the same time, paradoxically enough, a boiler room in the Arctic could be an extremely uncomfortable place. Those who had experienced tropical conditions in engineering rooms all knew of the detrimental effects. They also knew that attempts to improve conditions, particularly in warships, must generally result in a compromise, because of weight and space limitations. Air-conditioning, for instance, in warship machinery spaces, was normally not possible on this account. Efforts had to be directed to reducing the wild heat by better lagging and stopping steam leaks. Such measures had a double advantage. It was also desirable to improve ventilation as regards both air supply and the amount of hot air exhaust. The value of a good flow of high velocity air directed at the watchkeeper's position could not be overestimated. No doubt Dr. Mackworth would say that it was mainly psychological, but it had a very good effect, even though it was only hot air.

The value of the results obtained by the research workers in this field was that, knowing what the human frame should be able to stand without too much loss of efficiency, one could set limits to design, so that, while avoiding the fitting of unnecessarily powerful and bulky equipment, one could at the same time ensure that the performance of men who had to serve under these high temperature conditions did not deteriorate too far.

He was tempted to hope—although it was probably a vain hope—that research might be able to provide some idea of the length of time which men might be able to serve in ships under such conditions and on hot stations, and how far improvements in off-duty conditions could help them. Probably it was a vain hope; he suspected the problem had far too many variables.

He agreed with Dr. Mackworth that, from the practical aspect, investigations into noise problems were not very far advanced, but they were coming much more to the fore, particularly with the increasing use of Diesel machinery and the problems of high speed boiler fans and advanced steam conditions. The procedure for adapting research to assist the practical problem was really very similar to that in the high temperature field; that was to say, how—with an acceptable penalty in weight and space and possibly in efficiency of machinery—one could ensure a satisfactory level of efficiency in the watchkeepers who had to look after the machinery and work in very noisy spaces.

Dr. Mackworth had said a good deal about visual display, and there was no doubt that work on this subject could be of particular value, both in improving efficiency and in the interrelated problems of heat and cold and even, perhaps, saving of manpower. Considerable work had been done in this direction by the Naval Motion Study Unit as far as the Royal Navy was concerned, not only, of course, on machinery problems, of which he could mention a few, such as graduations of gauges, layout of panels and layout of controls. This gave promise of having value in increasing efficiency and even here and there saving a watchkeeper.

He suspected that as a result of research some of the long-established and preconceived notions about the best way a gauge should read or a hand-wheel should be turned might receive something of a shock, and there might be a lot of "sales resistance". Dr. Mackworth had mentioned the æsthetic question, but this one had a bearing as well.

Finally, there was the question of decision taking. In both warships and merchant ships there was increasing complication, taking the ship as a whole as well as the machinery alone. Those in control had a mass of information fed to them which they had to sort out and on the basis of which they had to produce the right decision or the right answer very quickly. Anything that could be done to assist them to sort it out in the easiest possible way and to take quick and correct decisions must be of peculiar value.

To repeat, this was a most important paper dealing with most important problems. As a result of the research work that was being carried out it would be possible, he felt sure, to learn a lot and to put it to very good use in ships whatever they might be and wherever they might be.

MR. W. MCCLIMONT, B.Sc., said that when he first saw the paper it took him back to a time early in the war when he happened to be in charge of an Army unit of the Royal Engineers, which was taking in recruits. On the first day a standard intelligence test was applied to all the recruits.

On an average, in the three-weekly intakes some four to five per cent were sent off immediately to the Pioneer Corps. In many cases, the results were due to the fact that these people could not read or write. A surprisingly high percentage came into that category.

On one occasion he was staggered to find from the marked papers brought to him that between thirteen and fourteen per cent of the people had to be rejected, most of their papers being blank except for names. That had really alarmed him. However, he discovered on making enquiries, just before these men were sent off, that a batch of Welshmen had been included, and they could not read or write English, so the results were completely astray. Since then, he had been very sceptical about mass observation of subjects and, while very useful work had been done in this field, he still viewed it with some caution.

When Dr. Mackworth spoke about people doing two hours' duty and having deteriorated over that period, he wondered whether the subjects knew how long they had done and when they were due to go off duty. His own recollections went back to Territorial days before the war when one was very much concerned with how long there was still to go and even more with whether the local pubs would be open when one came off duty!

His own interest in the paper lay at the point where the information produced by Dr. Mackworth and others could be used. It had been his job for some time to try to find out to what extent it could be applied to desirable conditions and limits of endurance in merchant ship machinery spaces, and then to consider the conditions prevailing in merchant ships and how they could be improved.

A great deal of useful information was available in the paper but it was a little difficult to comment on it in view of the professed purpose of the paper as set out in the summary: —

> "Throughout the paper an attempt has been made to illustrate the approach used in considering such problems rather than to try to give any complete survey of the knowledge at present available".

No doubt Dr. Mackworth's answer to any comments that might be made as to points omitted from the paper would be that they had been omitted deliberately.

Insofar as merchant ship work was concerned, it might have been more interesting to hear about the limits of desirable conditions than about the upper limts of endurance. With regard to high temperature effects, the emphasis in the paper was on the upper limits of endurance, but one knew full well that in these days it was difficult to keep enough seagoing engineers, without bringing in the idea of upper limits of endurance!

Perhaps it might be of interest to give some idea of the conditions found in engine rooms as compared with the 95/85 deg. F. condition which appeared to be regarded as the borderline where efficiency began to break down a bit. Would Dr. Mackworth agree that above a wet bulb of about 87 the actual dry bulb condition was not particularly important—that with a wet bulb at 87 one might have a dry bulb at 94 or 104 but there was very little difference in effect, so that the operative condition was the wet bulb condition?

The figures which followed related to the outside conditions experienced in the Persian Gulf which, according to statistics compiled by the Meteorological Office, were about 86 deg. F. wet bulb. By way of illustration he had taken some figures obtained on the manœuvring platform where the watchkeeper mainly had to spend time continuously: —

- 5,500 ton single-screw Diesel ship, about 87·4, or about 1·4 wet bulb higher than outside;
- Two 9,700 ton twin-screw Diesel cargo ships, 88·7 and 89, or a little above 85, which appeared to be the limit at which efficiency began to fall off;
- 27,000 ton quadruple-screw liner, again a Diesel, 88·9;
- Two 8,100 ton single-screw Diesel tankers, 89·8 and 88·6.

Thus, conditions in these Diesel ships were a little above outside conditions, though not very much. However, they were getting on to the slope of the curve shown in Fig. 3 of the paper.

Steam turbine ships provided the following data: —

- Two 9,200 ton single-screw turbine ships, 90·4 and 91·7, which was up a bit;
- Two 8,000 ton single-screw turbine cargo ships, 94·8 and 92·6;
- Two 13,000 ton single-screw turbine tankers, 90 and 94·3;
- Two 23,500 ton twin-screw turbine liners, 89·2 and 91·1;
- 20,000 ton twin-screw liner, 90·1;
- One 16,500 ton twin-screw turbine liner had a figure as high as 95·4, which was pretty hot.

With steam reciprocating ships, conditions might be even worse. Conditions varied very widely—as widely as from a 9,800 ton single-screw steam reciprocating tanker at 86·4 to a 10,500 ton twin-screw steam reciprocating cargo liner at 98·1.

The ages of the ships ranged from thirty-one to six years and it was interesting that no relationship could be found between the age of the ship and the conditions. The two single-screw turbine tankers with figures of 90 and 94·3 were built in 1921 and the two single-screw turbine ships with figures showing little appreciable difference—90·4 and 91·7—were built in 1945. The standard appeared to vary very widely.

With regard to acclimatization to high temperature conditions, emphasis had been laid mainly on the question of people who had been acclimatized for twelve to eighteen months, and for most of the experimental work they had been acclimatized before tests were carried out and artificially given this condition. For merchant shipping, data on people who had not been acclimatized would probably be much more useful, because a month was about as long as one would be in extremely hot outside or engine room conditions. In the case of voyages from this country to Australia, for instance, severe engine room conditions would probably be experienced for about a fortnight, and the period in the more temperate climates—though not necessarily very temperate climates—was sufficiently long to nullify any acclimatization there might have been during that period.

There were many other very complex factors to consider in relation to engine room conditions. One factor in a particular case was the effect of radiant heat and high humidity from escaping steam in adding greatly to liability to sickness. This was another factor, therefore, that might be worth considering. In a room at a particular humidity and a particular high temperature, efficiency might not be impaired for normal reasons. But there were many people, even fairly experienced seagoing people, who would suffer from considerable sickness if even moderately rough conditions were superimposed on these temperature conditions.

With regard to the tests on noise, the noise range covered was 70 db. on quiet days and 100 db. on noisy days. To be of interest, this would have to be carried a little higher. It might be quite adequate in Naval practice, but a level of 110 or even 120 would be of interest in merchant practice. In many cases, it would be found that the noise in the generator room went up to 120 db.

MR. J. K. W. MACVICAR (Associate, I.Mar.E.) said that, in spite of the great difficulty of obtaining complete data, owing to the complexity of human behaviour, the author had been able to show trends and reactions which were clearly of value. The contribution which he wished to make fell rather into the physiological field, although it was really very difficult for the layman to know where physiology ended and psychology began; no doubt, if he erred, Dr. Mackworth would be good enough to correct him.

In the marine world, the effects of extreme cold were not quite so interesting as those of extreme heat and excessive noise. While conditions likely to induce heat exhaustion in the Navy affected a wide range of the ship's company, in the Merchant Navy the engine room and stokehold personnel were most likely to suffer from the effects of extreme heat and possibly excessive noise.

As Mr. McClimont had already pointed out, 100 deg. F. dry and 90 deg. F. wet bulb represented a condition which was not at all uncommon in engine rooms of merchant ships passing through tropical waters. In this connexion, it was interesting to note the author's finding that, under such temperature conditions, and, given normal incentive, it was unlikely that human efficiencies higher than 60 per cent as compared with normal conditions would be attained. It would also be interesting to hear comments from superintendent engineers who had occasion to complain to their staffs of sins of omission and commission were they to be referred to Dr. Mackworth's paper and had it pointed out to them that the conditions under which the work had to be done did not permit of the high standard of efficiency which the superintendents had formerly allowed themselves to expect. Perhaps these remarks could be collected and circulated privately since it was doubtful if they would be suitable for general publication!

The conclusions reached were undoubtedly true, however, and they were confirmed by the investigations carried out during the last war by the United States naval authorities.

Electronic equipment was now being introduced on board Naval vessels which gave a degree of control formerly not considered possible and the question now arose whether the human being was efficient enough to match this equipment, particularly when suffering from stress arising from heat exhaustion. This had been recognized as such an important aspect that, in all vital compartments, air conditioning had been introduced in order to maintain the operating personnel at the maximum possible efficiency.

It would be interesting to hear Dr. Mackworth's views on acclimatization in relation to ships where certain working spaces were air conditioned, bearing in mind that the operators were only in the air conditioned compartment whilst actually working, and, during other periods, were subjected to all the rigours of the climate. Where the climate was extreme and conditions below deck even more so, it must frequently be the case that the operators were unable to sleep or rest properly and could not obtain the full benefit or reach the desired high standard of efficiency within the air conditioned compartment as if they had been privileged to live throughout the twenty-four hours in milder conditions.

On the other hand, many competent authorities believed that where a good night's sleep could be ensured by means of air conditioning the living accommodation, the subjects were able to maintain a high standard of efficiency even though the working conditions were extremely hard. Again, the time factor must be taken into account and it would be interesting if the author would care to hazard some views as to human ability under the following headings:—

(1) Subjects living during off-duty periods under extreme conditions and also working under extreme conditions;

(2) Subjects living during off-duty periods under comfortable conditions and working under extreme conditions;

(3) Subjects living during off-duty periods under extreme conditions and working under normal conditions;

(4) Subjects living under comfortable conditions both during off-duty periods and during working periods.

This would enable the various experiments to be compared. It was appreciated that the author had compared efficiency of working, between comfortable and extreme conditions, but the question of acclimatization and the length of time during which the conditions were experienced, was a most important factor. In other words, it was suggested that tests such as had been made for high atmospheric temperatures should, in fact, be carried out in the tropics, as had been done at Singapore in the work carried out under the direction of Surgeon-Commander F. D. Ellis. While the author had suggested that the Singapore tests more or less confirmed the Cambridge tests, it would be interesting to have the Singapore data in some detail.

He pointed out that it was difficult to have full confidence in tests carried out in this country under laboratory conditions since the psychological aspect appeared to intrude. In recent weeks he had had some experience of working in a hot room in which Red Sea and Persian Gulf conditions were maintained. Despite the fact that the conditions were really extreme and would have produced something approaching complete exhaustion if experienced over a long period in these regions without any possibility of relief for some considerable time, no such reaction took place in the hot box, probably due to the fact that the subjects were aware that after a comparatively short period they could obtain complete relief.

On the subject of noise the author had produced some very interesting information but he had confined himself to noise levels, which at the moment were considered normal in ships' engine rooms. He had not increased the noise to the extreme where physical suffering was likely. As a matter of interest, it might be mentioned that in some machinery spaces in ships belonging to a foreign power, the period of duty in the engine and boiler rooms was actually determined by human reactions to the excessive noise which existed. As new types of machinery came into use, conditions might occur which would make research into this field well worth tackling.

In conclusion, he again mentioned that it had not been his intention to touch on the psychological aspect of the subject but it was interesting that, as he had gathered from Dr. Mackworth, when the second engineer gave the juniors a piece of his mind in the general roustabout during the watch, he was not really being rude or boorish but was in fact giving them a "psychological stimulus". This definition should be more widely known so that seagoing members might place a different interpretation on the stimulus, thus avoiding feelings being hurt!

The CHAIRMAN said that Mr. MacVicar had referred to work in the United States, and as it happened a research psychologist from the States was present. He invited him to address the meeting.

DR. HENRY A. IMUS complimented the author on his excellent dual job—the preprinted paper and his extemporaneous explanation of it. He had known Dr. Mackworth for a number of years and was impressed and pleased with the research which was being carried out at Cambridge under Sir Frederic Bartlett. The United States Office of Naval Research was following this research with considerable interest.

The research conducted in the States during the war, particularly on radar and range-finder operators, with which he had been associated, confirmed many of the points that had been brought out during the evening. At the U.S. Navy School for Radar and Range-finder Operators at Fort Lauderdale, Florida, a number of these conditions were studied. Similar work was conducted at the Army Radar Unit at Camp Murphy a few miles away. If a man was kept on watch for a period of twenty to thirty minutes, he achieved his maximum production. After that, he steadily fell off. It was also found that if he were shifted after half-an-hour to another part of the task and then brought back for another twenty to thirty minutes, his efficiency could be kept up. Dr. Mackworth had shown the truth of this by his results.

Not only in radar work but also in range-finder work on anti-aircraft fire-control, it was found that to give a man almost immediate knowledge of the results was extremely helpful in keeping up the maximum efficiency of performance.

Another interesting point that appeared from the Army Radar study was that the addition of one more watcher to the radar staff brought about a considerable improvement in performance when looking for chance targets coming into the view. With three or more men, efficiency fell off. There was, therefore, some advantage in having two men on the job instead of one.

These careful scientific experiments were all very well, but a very practical attitude must be maintained towards them, as was indicated by the mother who carefully taught her boy to cut his finger nails with his left hand because some day he might happen to lose his right hand!

While at the School of Aviation Medicine in Florida, he had had the opportunity of going out on an aircraft carrier on "carrier qualification" operations. The carrier was out for a period of seventy-two hours on continuous operations, during which time some four hundred landings on the carrier were made during the day and night without interruption. At this time, which was right at the end of the war, the fire-room crew was extremely short handed. The men were working four hours on and four hours off in the sub-tropics in the summer. During this long stretch, there was a complete breakdown of the fire-room crew. It was therefore necessary to call off this expensive operation and to return to port, because the engine room could not be kept functioning. It occurred to him that more attention to comfortable working conditions and perhaps the addition of remote controls might do a great deal to help the men when they were working under extremely severe conditions.

This brought him to his third point: that if there were to be changes, they should be made on the design table. They must start with the design engineer, because if one waited until the ship was constructed and the components were in, space and weight limitations would prevent corrective action.

He might give one or two examples of that. One was rather amusing. In the early design stages of the B-29 it happened that there were three design engineers working on the controls in the cockpit. When the controls were tried out at the mock-up stage the pilots could not manage them; everything was awkward and the controls were in the wrong position, and so on. It was discovered later on that the three design engineers all happened to be left-handed. Again, during the war, when work was done on the computing sight to be put on anti-aircraft guns on the stern of tankers or freighters for individual protection, the equipment was designed and built, and was then taken for sea trials. It happened that, in order to get on a target a man had to operate a handlebar gear into which he was strapped, and also keep his eye in line with a reticule on the target. It was necessary for him on various target azimuths to change the intensity of the illumination on the reticule so that he could perform adequately. The rheostat for adjusting the illumination was down by his knee, so the only way to work the control was to have two men, one a giant and the other a midget!

Mr. F. D. Clark (Associate Member of Council, I.Mar.E.) said that he was not a psychologist but a marine engineer with experience in the Merchant Navy and the Navy. He spoke, therefore, as an operator. He would like to congratulate the author on his extremely interesting paper but he would also like to congratulate those members of the respective institutions who had had the forethought to arrange that this subject should be brought to the attention of the members.

During the past thirty-five years many papers had been read at the Institute illustrating the great strides and progress that had been made in the design and performance of ships' machinery. This was the first time, he believed, that there had been a paper dealing with the human factor. They had discussed at great length in the past the stresses and strains that machines and ships would withstand under varying conditions of heat, load and quality of fuel supply, but they seemed to have ignored until now the reaction of the man who must operate this machinery also under varying conditions of heat, load and—if they liked—quality of food supplied. Yet he was just as essential to the smooth running of the ship as the correct machining of the teeth of the main gear wheel.

Fortunately, however, it appeared that those concerned with the science of psychology had also been making progress during the past thirty-five years. Might he suggest that if one took the results of their studies in how to obtain higher efficiency in the human factor—or increased productivity per man hour—under varying conditions, combined with the studies concerned with the layout of ships' machinery spaces, the result would quite certainly be an increase in the efficiency of the ship and personnel?

It might be an achievement to squeeze another three thousand horsepower, say, into a smaller engine room than before, but if by so doing one produced operating conditions which tended to reduce the physical and mental efficiency of the operator, his drop in efficiency would certainly be reflected in the efficiency of the plant under his control.

A great deal of money was now being spent by shipowners on higher wages, improved accommodation and new apprenticeship schemes to encourage the right type of man to go to sea or go back to sea as a marine engineer. He wondered how much of this could have been saved if more thought had been given in earlier days by the designers to the man who had to clamber on boiler tops to shut and open valves in an atmosphere that burned the inside of his nose as he breathed, or to that workshop flat in the top corner of the engine room where he became just a soggy mass working another four hours, doing a field day in addition to the watches he kept on the engine room platform, where it was comparatively cool—say, a mere 110 deg. F.

Looking back on his sea service, he found Fig. 3 on page 124 extremely interesting, and he wondered whether one second engineer, in particular, with whom he sailed, would have been able to plot a similar chart for him. He assumed that the second engineer would have found a better line on his graph had he (Mr. Clark) had the higher incentive of realizing that the field days organized were always necessary and not part of a ritual specially invented for junior engineers. Incidentally, the author's point on results obtained under higher incentive conditions was an important one and illustrated further a fact which had been known and practised in industry for some time; namely, that it was an added incentive for a man to increase his efforts if he could appreciate the usefulness of what he was doing and why he was doing it.

He would have liked to see some mention of recommended clothing under various conditions experienced at sea. In a hot engine room, for instance, he always found that thick boots and socks were far more comfortable than lighter equipment. He would be interested to hear of any research that had been done on the type of clothing that should be worn in an engine room, bearing in mind that protective clothing must be worn to guard against burns through brushing against hot metal parts, either through the machinery of the engine room being closely packed or because of the rolling of the ship. Consideration should also be given on this matter to that common complaint had been a martyr to it himself, and he was certain that the was a martyr to it himself, and he was certain that the efficiency of any man must be reduced to some extent when a boiler suit soaked in his own perspiration clung to his body like a wet rag and with every bodily movement he made he received the equivalent of thousands of pins sticking into every joint.

He was also interested in the author's statements about men becoming acclimatized to conditions of heat or cold. During the war when the ship consisted of a series of closed watertight compartments, the engineer officer doing his rounds in the various compartments, particularly on a warship, was obliged to come out on the open deck from one compartment, and go along and down into the next. As far as he could recall he personally had never felt the sudden change of temperature from the hot engine room to the cold deck and, in point of fact, even in the Merchant Navy, when going from the hot engine room into the refrigerator chambers to take temperatures, he never wore any additional clothing, nor could he recall ever suffering any ill effects from these sudden changes from heat to cold. Would the author say that his body was acclimatized to these extreme and sudden changes, and although he could not recall suffering any ill effects, would he consider that nevertheless this was not a good practice?

With regard to the section on noise, he had found from experience that in an engine room the best signals, particularly alarm signals, were those which used sound as a means of drawing attention. The ears became acclimatized to the noise of the running machinery, and anything slightly different from this general noise was noticed immediately. In fact, it was interesting to note that on first going to sea, one usually had the greatest difficulty in hearing the conversation of the other fellow, and one tended to shout back at him in reply. After one had become acclimatized to the noise, one could carry on a conversation in a normal tone. He must admit that he had never been quite certain whether it was recognition of the spoken word by sound or through lip reading. He would welcome the author's comments on this.

Finally, he would suggest that the Papers Committee should classify this subject as one of extreme interest and put it on the rota as one to be brought up periodically. He would like to see another paper some time in the future, dealing solely with the study of the reactions of the engineer in charge of a ship's engine room who had to combat conditions under all the headings which the author had put into his paper, namely heat, physical work, noise, visual display and manual controls, and very often to make many decisions in a very short space of time.

CAPT.(S) A. D. DUCKWORTH, R.N.(ret.) (Secretary, I.N.A.) said it was encouraging to find that so much attention was being paid to the habitability of H.M. ships. For him, as for many others, these investigations seemed a little late in the day; but they would be none the less welcome to their successors who would need all the habitability on board ship which scientists and constructors could provide for them.

He would invite the author's attention to Fig. 2, where five appropriately unclad telegraphists were seen at work in a kind of synthetic Turkish bath, a condition all too familiar to those who knew what a confined space on board ship could be like in tropical waters. But, in wartime conditions were made still worse, for overalls and anti-flash clothing had to be worn, perhaps for long periods, with an attendant discomfort that had to be endured to be believed. He hoped the author would remember this extra burden, where one's whole body had to be covered, and arms and face protected with anti-flash clothing, leaving only eyes, nose and mouth exposed. Perhaps some experiments with victims similarly swathed would yield further valuable data.

He could confirm what previous speakers had said about temperature and its immediate effect on one's work and vitality, though naturally individuals varied in their reactions. He had always found that he and most people could stand dry heat well, but with only a small increase of humidity one wilted completely, and personal efficiency was apt to suffer.

PROFESSOR G. L. BROWN, C.B.E., F.R.S., said that he had been interested in work of this kind for many years, and he would like to make one or two general observations.

It did not need an experimental psychologist to show that with excessive heat and humidity there was a loss of efficiency. The man in the street or the man in the ship knew that perfectly well. But the purpose of Dr. Mackworth's work, if he might be allowed to explain it, was to measure that loss of efficiency. Admiral Clemitson had put the point very well when he had said that every ship was a compromise. If one put more gear into her one made conditions more unpleasant for the men inside her. For physical work one could say, on the basis of the work done in the last few years, that at a certain temperature there would be a certain accurately measurable loss of performance. With hard physical work, the loss was easily measurable. But with the complexities of modern civilization, hard physical work was becoming more and more rare. Men now used bulldozers instead of picks and shovels, and Dr. Mackworth's purpose, primarily, was to be able eventually—he was nowhere near it yet—to produce an accurate measure of the falling off in performance. For instance, another gun on a ship would increase her hitting power, but would it at the same time so reduce the efficiency of the crew by overburdening the already overburdened space of the ship that the end result would not be a gain in hitting power but a loss? It was quite obvious that this falling off in efficiency could not be assessed numerically as yet. The work of the Cambridge school was the first really serious pioneer work in attempting to reach some accurate measure of human performance, not in simple physical tasks, but in those complicated evolutions which required the higher levels of activity in the brain.

The only other criticism that might be directed against Dr. Mackworth's work was that the great bulk of it had been carried out in the laboratory. There was an obvious reason for this; only in this way could perfectly controlled conditions be secured. He was asked some years ago to investigate the effects of four-hour watches on the performance of Wren ratings, and he was discussing this with a Wren officer. She said, "Oh, yes, the girls who are on four-hour watches suffer very much from lack of sleep and general loss of efficiency; but the reason is perfectly simple. The Wrens on four-hour watches are on communications, and they are the first to make dates with ships coming into harbour".

Dr. Mackworth would be the first to admit that everything he did in the laboratory must eventually be done in ships at sea before the final results were obtained.

MR. D. E. BROADBENT said that as a Welshman he should perhaps start by saying that his own attitude to experimental psychology also dated back to finding some people who could not read or write. This also took place in the forces. He had found that these people had been through a selection procedure of the old type, and they had been posted to the unit of the commanding officer. He found out that they could not read or write only because they had been posted as clerks, special duties! He gave them some intelligence tests and an even more alarming fact emerged, which was that one of them did much better than he (Mr. Broadbent) had done! He was glad to be able to say that this man had afterwards learned to read and write in a fortnight, quite a remarkable achievement. Ever since then he had had a favourable attitude to experimental psychology.

He would like to take up some of the points on noise and communication, that being his own particular line. He was very interested in the suggestion that sound signals were best for some purposes in an engine room. This he believed to be due to the undirected character of hearing. One could not see a visual signal unless one was looking at it. One could hear a signal, no matter what direction it came from. This raised the general point that for some purposes sound was better and for others vision.

The system adopted by the Institute of having a printed paper for the details and an oral introduction brought this out very clearly. It was almost impossible to take in a large number of details by sound but as a worker on hearing he was glad that sound was better for some purposes.

The main question about the work in Cambridge on noise was perhaps the low level of noise employed. For the benefit of those who were not altogether familiar with the db. scale, 100 db. was a very low level compared with the conditions prevailing in ships. It was the sort of noise in which one could just make oneself heard by shouting at the top of one's voice, and levels from thirty to forty times this amount prevailed in ships.

The experiments were carried out at this comparatively low level for a very good reason. If the levels reached in practice had been used, the hearing of the subjects would definitely have been impaired. That being so, any results due to noise might have been affected by the fact that the hearing was becoming worse and therefore the noise reaching the subjects was becoming less. This was a well known fact. There was no question that the levels employed generally in practice affected hearing, quite apart from any effect on efficiency. Therefore, one wanted to reduce the noise, if possible, to the sort of level employed in the experiments.

It also raised the question of communication under these noisy conditions. It was a fact that there was an optimum noise level for communication for any signal/noise ratio. This meant that if one had a certain amount of auditory signal going on, one might hear better if one became slightly deaf. This might be the reason why people who had worked in engine rooms sometimes appeared to understand speech better. It might be rather more economical if they wore ear plugs instead, and that had been suggested.

In general, since the level used in the experiments was so much lower than that prevailing in ships, the question of finding a critical level for noise, as distinct, possibly, from heat, was more and more impracticable. The only answer was to reduce the noise as much as one could and to take other measures to deal with its effects when it had been made as low as possible. It was obvious that heat must be produced, but there was no particular reason why noise should not be cut down more than it was. Nevertheless, the level was bound to be high, and in these circumstances, it was essential to consider the nature of the task a man was expected to perform in noise.

Dr. Mackworth had made a point of the nature of the display. It was possible to remove the noise effect simply by making the display easier, and it was also possible to change the task altogether.

It had been suggested in connexion with air traffic control that the tendency was to get the machine to do the job and the man to sit back and watch it. As Dr. Mackworth's work had shown, that might not be a very good idea, because if the machine broke down the man would not be watching. The suggestion which was made in place of this arrangement was that the man should do the work and the machine should watch him. This was not, of course, as silly as it sounded. The point was that one could have mechanical interlocks on most work, so that the most serious errors would not occur. One was likely to get a higher level of efficiency by designing the mechanical controls to provide safety factors, than by letting the machine do the routine work while the man dealt with emergencies.

That was merely a part of the general point that the critical factor of deciding how human beings performed under stress came in at the design level. The task must be designed originally so that stress would not affect it too much.

LIEUT. COM'R(E) H. T. MEADOWS, D.S.C., R.D., R.N.R. (Member) said that he was prompted to join the discussion to lend emphasis to Mr. Clark's reference to prickly heat. It was his experience that the effects of prickly heat were much more devastating than those of heat exhaustion. In support of this he mentioned that during the early stages of the Far East campaign in the late war, a front line submarine had been immobilized and confined to harbour, not by damage, or engine failure, or lack of stores, or ammunition, but solely due to the fact that more than half of the crew was suffering acutely from prickly heat. While he thought that strictly this was outside the bounds of the paper, and was the work of a doctor of medicine rather than a doctor of philosophy, he hoped that the publicity given this evening to the scourge of prickly heat would help to produce a cure or a preventive treatment which could be made available on both merchant and Naval ships.

There had been much said that evening about the ill effects of heat in enclosed spaces. It might be surprising to the more learned and possibly less practical, that heat and humidity had some medicinal value. In this respect, he quoted the practice of second engineers in the shipping company with which he had served; junior engineers turning to for days work in harbour, and suffering from the effects of alcoholism, were sent to work on the boiler tops. Such treatment never failed to produce a complete recovery by breakfast time!

Correspondence

MR. L. BAKER, D.S.C. (Member of Council, I.Mar.E.) wrote that the introduction of the subject to marine engineering was very timely and no one was better qualified to do this than the author. To those who had seen the work progressing it had been most absorbing and he felt quite sure that, given due consideration, many of the lessons that were being learned by the staff of the Applied Psychological Laboratories could be applied to the benefit of the seagoing marine engineer and thereby to the benefit of the shipowner.

He was somewhat disappointed that Dr. Mackworth had not thought fit to include details of the work that he had done on the design of letters and figures and on the marking of dials. This work was very valuable. The conclusions reached were quite contrary to what one anticipated and yet, so far as he had been able to see, there had been no attempt to apply the results outside the field of aircraft design.

In the same way, designers appeared to have very little interest in producing apparatus that was functionally suitable for the human being. For example, a big majority of "engine order telegraphs" on the bridge were entirely unsuitable for use in times of stress, due to the fact that the range of movement required on the handle was too large and that accurate centering of the transmission could only be achieved by considerable body movement. If a joystick of an aircraft had been designed on the same principle as the handle of an "order" telegraph, he doubted whether any aircraft would have left the ground!

Another common example was the arrangement of pressure gauges and thermometers on control panels. This seldom bore any relation to the requirements of supervision or of the physical limits of the man himself. A typical illustration of a badly arranged dial was shown in Fig. 17, which showed the main control panel of a modern steam

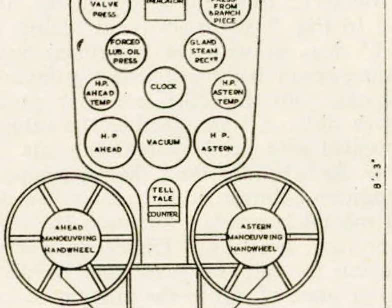

FIG. 17—*Arrangement of main control board in "P" and "H" class ships*

ship with badly grouped controls and, worst of all, the revolution indicator situated so high that no man could look at it for any length of time without moving away from the control wheels. In Fig. 18 the arrangement of a similar

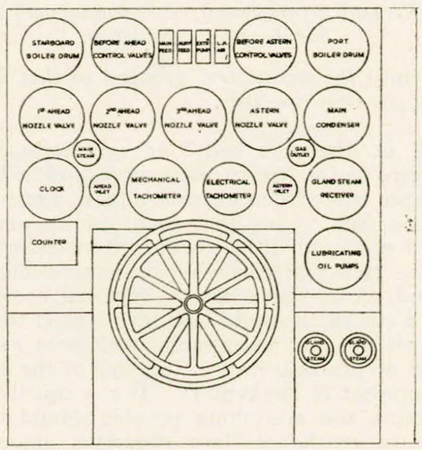

FIG. 18—*Arrangement of main control board in "N" class ships*

panel for a later class of steamers showed how the difficulties had been overcome by careful grouping.

Figs. 19 and 20 showed the same panel for the control of a heating installation in a building; the first was arranged by the heating engineers to fit in with the architectural

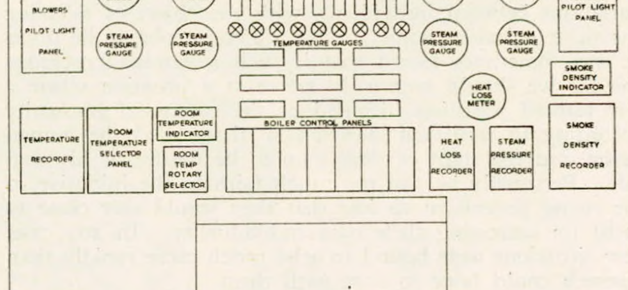

FIG. 19—*Proposed arrangement of thermometers and gauges for a heating installation*

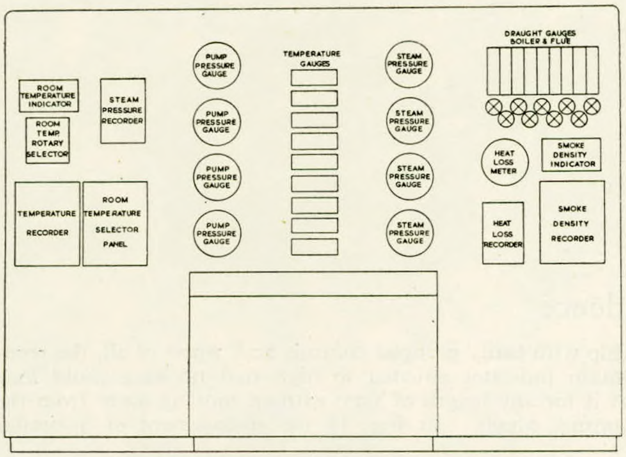

FIG. 20—*Revised arrangement of thermometers and gauges for heating installation*

features, whilst the second was grouped so that functionally related gauges were together.

MR. M. G. BENNETT had long ago learnt to pay the highest regard to Dr. Mackworth's work, and he thought the present paper was of considerable topical interest. It was, perhaps, too trite to say that such studies were specially needed just now with the increasing complexity of modern civilization. New problems were constantly being exchanged for old and life probably always appeared complex at the moment. Certainly it would have been a good thing had the results of all this sort of research work been available and understood by practical men at the time of the great industrial development of last century. It was equally important for the future and everything possible should be done to propagate the results of these researches amongst people with responsibility for affecting the environments in which they all worked.

He imagined that one of the greatest difficulties in spreading the truth about these matters was the feeling that, if one were tough enough one could stand anything, and that individuals varied so much, and were so adaptable, that people could be found to fit any job in any environment. It was, of course, quite true that individuals varied widely and that the will to surmount difficulties helped enormously to overcome them. It must be equally true that it was wise to arrange for work to be done as easily and congenially as possible so as to leave as much energy as possible available for doing the work well.

Complementary to these studies was the study of the variations between individuals, with the object of selecting the most suitable people for the different jobs to be done. It was sometimes feared that if such researches proceeded too far we should eventually arrive at a situation where a few learned scientists disposed of the careers of humanity according to statistical laws which disregarded the human desire and the right of each man to be master of his own fate. Personally he had too much faith in the initiative of the rising generation to fear that they would ever cease to insist on exercising their own individuality. In any case, new situations were bound to arise much more rapidly than research could hope to cope with them.

Meanwhile, the scope for the application of such work as Dr. Mackworth's was enormous and the benefit that could result from this application would be very great. He was not competent to discuss the researches in detail, but he would commend them as strongly as possible to everyone who could exert any influence over the working conditions of their fellow men.

COM'R(E) F. ROBERTS, O.B.E., D.S.C., R.N.(ret.) (Member, I.Mar.E.) wished, on the basis of his own experience, covering thirty-one years of marine and naval engineering and the stress conditions of two wars, to remark on some of the points dealt with in Dr. Mackworth's paper.

There was a large psychological factor in the reaction of the individual to considerable changes from his normal environment. Some individuals were not prepared to deal with the unusual or the unknown and met changes in the normal pattern of their lives with a resentment which induced an incurable loss of mental and physical efficiency. Such individuals could develop no habitude to any unusual conditions of the environment, in spite of long experience.

Again, fear of environment could induce large declines in physical and mental efficiency. This was particularly the case with those individuals who had a fear of the sea but had become seafarers by virtue of patriotism, conscription, parental misguidance or economic necessity.

There were other individuals, however, who were prepared to accept large changes in their environment without resentment, and such individuals rapidly habituated themselves to new conditions, so that after the initial loss of efficiency there came a steady improvement as experience brought the necessary mental and physiological adjustments to the new environment.

Considering this last group of individuals, the effect of a new environment in reducing efficiency had two modes. In the first, concentration upon the task in hand was reduced by physical discomforts such as heavy sweating, shivering, etc., whose incidence ensured insistent distractions. In the second mode, changes in metabolic requirements, which had not been provided for, caused a real fall in mental and physical wellbeing. In regard to this second condition, research which set out to discover the most suitable clothing and diet for given environments could do much to mitigate these effects.

Considering the results of the test carried out at high temperatures and high humidities, the effect of both increase in physical discomfort, and change in metabolic need were clearly seen in the curves of Fig. 3. Increasing physical discomfort (e.g. sweat running into eyes, or dripping on to the signal pad) reduced the efficiency of competent and very good operators, but did not affect the efficiency of the exceptionally skilled operators, whose mental translation of Morse signals into message groups had become so much a reflex operation that it did not involve a high concentration in maintaining accuracy. At temperatures above 100/90 deg., a marked fall in the efficiency of all three groups took place, which was probably due to the fall in the saline concentration in the body fluids occasioned by the profuse sweating required to retain the normal body temperature. In Fig. 5, the more rapid decline in efficiency set in at 85/75 deg., as, with the additional muscular heat, profuse sweating began at a lower temperature and humidity than was the case with the more sedentary work of Fig. 3. Dr. Mackworth did not state whether the subjects of these tests were supplied with drinks containing salt. In his own experience, he had found that the provision of drinks containing sodium chloride could reduce considerably the physical and mental lassitude which was the result of long periods of profuse sweating. Under such circumstances the body craving for salt was such that drinks containing a teaspoonful or more of salt to the pint tasted quite normal and were refreshingly palatable. It was suggested that a series of tests should be carried out on the high temperature working efficiency of groups of subjects when supplied with salted drinks.

With regard to the tests on the effect of low temperatures upon manual skill, it would seem possible that muscular sensitivity (i.e. the reflex appreciation of the balance of tensions in the muscles of the hand and arm) was of greater importance than tactile sense in the acquirement of manual dexterity, for the thickened skin of the working surfaces of the skilled craftsman's hand did not seem to affect his craft skill. Hence a test to determine the reduction in muscular sensitivity on exposure to cold would be of value. Tests of this nature could comprise the measurement of a preset gap with feeler gauges, making a comparison of the diameters of taper plugs against a parallel plug with outside callipers, etc.

Dr. Mackworth did not state whether any attempt was made in the Twenty Dials test to relate pointer changes to changes in the noise spectrum. There was invariably a close relation between changes in the components of the noise spectrum of a group of machines, and changes in the pointer readings of the dials which indicated the variables affecting that group of machines. So, just as the experienced concert-goer could pick out the individual instruments of a large orchestra and appreciate aurally the quality of their separate performances, the experienced marine engineer was audibly conscious of the performance of the machinery in his charge, and was immediately aware of any component which started running out of time or tune. This awareness immediately drew his attention to the appropriate dial. Hence machinery noise aided, rather than detracted from, visual alertness in trained and experienced machinery watch-keepers, though it might reduce the efficiency of subjects when there was no relation between dial readings and machinery noise. And it seemed that, for validity of conclusions in this matter, a test in which the change in pointer reading was invariably related to a change in noise spectrum should be devised and carried out.

Visual display by counter was always preferable to visual display by divided scales or dials, when speed and accuracy in reading the values were necessary. When, however, it was only necessary to keep the value of a variable (e.g. a working pressure) between a maximum and a minimum, a dial with coloured arcs, arranged so that the pointer was vertical in the mean position, was the best choice.

A fault with many manual controls was that the change in the variable controlled was not directly proportional to the movement of the controlling handwheel or lever. As a result of this lack of proportionality, the operator required long experience before he could readily adjust the control to give any desired value of the variable. It should not be difficult to design controls so that their movement was directly proportional to the change in the variable controlled.

Undoubtedly there were limits to the speed and accuracy of the number of decisions which any human being could make in a given time, but it did not appear that automatic mechanisms for the removal of excessive and irrelevant information from communications could be devised. A senior officer could best avoid mental jamming in emergencies by the careful organization and training of his staff. If, by forethought, he could predict and visualize the possible emergencies that could occur, then he could select and train suitable subordinates to deal directly with minor decisions, and filter out all but the important and the immediately relevant in their reports to him. Then, with such a fully trained staff, he was left free to make the decisions of major importance on a basis of properly filtered information, knowing that decisions on minor details had already been made and appropriate action taken. And knowing that if he should become a casualty, some one or more of his staff could take over his function with a reasonable adequacy.

MR. JAMES WOODING commented that in the noise experiment, where the effect of monotony was important, he thought some account should have been taken of the intelligence level of the subjects; the more intelligent ones would be more likely to make mistakes through boredom and therefore subjects should have been chosen having equal intelligence quotients.

Author's Reply

DR. MACKWORTH, replying to the discussion, said that he could deal only with some of the questions he had been asked. The best way to answer them would be to do more research.

Admiral Clemitson had referred to length of stay in the tropics. The best way to study this question would probably be by means of sickness returns of people who had had different lengths of stay. Minor sickness would seem to be one possible index.

In reply to Mr. McClimont, he said that attempts had been made to study the effects of disturbed sleep, and it was known that a difference of ten degrees meant a considerable amount of disturbance in terms of the number of restless movements made by people sleeping in these higher temperatures. There was a fifty per cent increase in the amount of tossing and turning between 75 and 85 deg. F. wet bulb. It was very difficult to show any subsequent effect in terms of performance on, for example, tasks such as the trainer pilot's task on a synthetic aircraft. On the whole, it did not seem to have any effect at all, but he agreed that studies of this kind should be undertaken.

The point about acclimatization was very well made. This was an outstanding question: very little was known about the trend of efficiency and the development of adaptation during acclimatization.

Prickly heat was being studied by those who were working for the Navy and the Medical Research Council at Singapore; they were in a good position there to study that highly important question.

INSTITUTE ACTIVITIES

Minutes of Proceedings of the Ordinary Meeting held at the Institute on Tuesday, 8th April 1952

A joint meeting of the Institute and the Institution of Naval Architects was held at the Institute on Tuesday, 8th April 1952, at 5.30 p.m. Mr. James Turnbull, O.B.E. (Chairman of Council) was in the Chair, supported by Mr. L. Woollard, M.A. (Vice-President of the Institution of Naval Architects).

A paper by N. H. Mackworth, M.B., Ch.B., Ph.D., entitled "Some Recent Studies of Human Stress from a Marine and Naval Viewpoint", was read and discussed. Fifty members and visitors were present and nine speakers took part in the discussion.

A vote of thanks to the author was proposed by Mr. Woollard and was accorded with acclamation. The meeting ended at 7.20 p.m.

Junior Section

On Thursday, 17th April 1952, at 7 p.m., the Junior Section met at the Institute to discuss "The Value of Technical Qualifications to the Marine Engineer Apprentice". Mr. F. D. Clark (Associate Member of Council) was in the Chair and thirty-three members and friends were present.

In opening the meeting, the Chairman informed the Section that an annual award of £5 was available for the best report, submitted during the year by a Graduate or Student, on a junior discussion held at the Institute.

The subject was introduced by Mr. T. W. Longmuir (Member of Council) and was followed by a discussion in which thirteen speakers took part, and which is reported in the Supplement to the current issue (pages 1 to 4). On this occasion, the panel of members to whom questions were referred consisted of Mr. Longmuir, Messrs. S. Hogg and A. Logan (Members of Council) and F. A. Everard.

Mr. P. F. Morgan, M.A. (Graduate) proposed a vote of thanks to the members of the panel, which was accorded by acclamation, and the meeting ended at 9 p.m.

Summer Golf Meeting 1952

On Tuesday, 20th May 1952, in fine, cool weather, the Summer Golf Meeting was held at Hadley Wood Golf Club.

Twenty-six members took part in the morning Stroke Competition for the Institute Cup, which was won by Mr. R. B. Pinkney with a net score of 70. Mr. E. F. J. Baugh was second with 71, and Mr. W. J. S. Glass third with 73.

After lunch, twenty-eight players took part in a Bogie Greensome Competition. E. F. J. Baugh and W. J. Borrowman were first with 3 down, G. M. McGavin and R. K. Craig coming in second with 4 down.

The prizes were presented after tea by Mr. A. Robertson, Convenor of the Social Events Committee. Mr. Pinkney re-

Summer Golf Meeting 1952

ceived a Nell Gwynne clock in addition to the Institute Cup. Mr. Baugh decided to forego the morning prize as he and his partner had won the two first prizes in the afternoon; according to the rules, therefore, the second prize in the morning went to Mr. Glass, who received a heat resisting tray. Messrs. Baugh and Borrowman received cork screw sets, while Messrs. McGavin and Craig received Royal Auction Bridge boxes.

On Mr. Robertson's proposal, a hearty vote of thanks to the Hadley Wood Golf Club Committee was carried unanimously; the arrangements which had been made for the meeting were excellent, including a most enjoyable lunch and tea. He also thanked those members who had so kindly subscribed to the Prize Fund, namely R. K. Craig, J. G. Edmiston, J. H. F. Edmiston, J. A. Goddard, W. Q. Henriques, P. R. Masson, N. M. Niven, R. B. Pinkney, J. A. Rhynas, A. Robertson, W. Sampson, A. C. Smith and W. Tennant.

He announced that the Autumn Meeting would be held in either September or October, and it was hoped to arrange another competition between the Institute and the Institution of Naval Architects. Mr. Craig thought that the meeting should take place as early as possible, and Mr. Sampson suggested the third or fourth week in September. It was also decided that a notice be inserted in the Supplement to the Transactions to the effect that only registered members of the Institute Golfing Society would receive notices of the Autumn Meeting.

The meeting terminated with a vote of thanks to Mr. Robertson.

Sydney Local Section

On Thursday, 29th May 1952, at 8 p.m., the Sydney Local Section held a General Meeting at Science House, Gloucester Street, Sydney. The Chair was taken by the Local Vice-President, Mr. H. A. Garnett, and eighty-two members and guests were present.

Mr. E. Stuart Clarke delivered a most interesting and informative lecture, well illustrated by lantern slides, on "The Manufacture of Doxford Engines at the Commonwealth Engine Works, Port Melbourne". Messrs. Weymouth, Butcher, Robertson, Ross, Petrie, Williams and Recknell took part in the discussion. A vote of thanks to Mr. Clarke was proposed by Mr. H. P. Weymouth, seconded by Mr. G. T. Marriner, and carried by acclamation.

Membership Elections

Elected 12th May 1952

MEMBERS

Henry Beveridge
Walter Joseph Burden
William Burton, Lieut.(E), R.N.
Eric Gordon Harbottle
Raynor Lawrence Lawrence
Peter Dickson Lochtie
William McClimont, B.Sc.
Colin Hill Menzies
Robert Millar
Percy Thomas Shipley, Com'r(E), R.N.
Francis Jardine Welch
Isamu Yamashita

ASSOCIATE MEMBER

Vernon James Morcombe

ASSOCIATES

Harry Alexander Andrews, E.R.A., R.N.
Peter Battison
Soli Khurshed Dhondy
Ian Graham Toyne Duncan
Raymond Leonard Floode
Thomas Harold Griffin
John Hendry
Philip Holland
Alexander McLean Knox
Gavin Hamilton Dempster Martin
William Cyril Merwood
Robert Broadfoot Stitt MacAdam
John Henry Michell
Alexander Thomas Naysmith
Colin Pearson
Ivor Lee Polden
Charles Gerard Purvis
Mohammed Iqbal Qureshi
William Baird Robertson
James William Scott
George William Shotter, Sen'r Comm'd Eng'r, R.N.
Henry Alexander Sledge
Donald Taylor
Ralph George Whitelaw

GRADUATES

Angus Brown
Ronald George Swan

STUDENTS

Peter John Balls
Deba Priya Guha
Piyaman Jayaratne

TRANSFER FROM ASSOCIATE TO MEMBER

Robert Harrison
Arnold Nelson
Frank Norman Tanner
George Beresford Williams

TRANSFER FROM ASSOCIATE TO ASSOCIATE MEMBER

Ronald Mensforth
Vellore Ramanatha Rajagopalan

TRANSFER FROM GRADUATE TO ASSOCIATE MEMBER

Denis Knowles

TRANSFER FROM GRADUATE TO ASSOCIATE

Girgis Fahmy Aziz

TRANSFER FROM STUDENT TO ASSOCIATE

John Leon Wood, B.Sc.

Elected 16th June 1952

MEMBERS

James Ross Bailes
Gordon Clark
John McGibbon Crossan
Harry Stapleton Doggett, Lt.-Com'r(E), R.N.
John Owen Charles Duffy
Arthur Gordon Emery
David Walter Evans
Edward George Gillett, Lieut.(E), R.N.
William David Heggie
Saleem Ihsan-Ullah
James Arthur Kent
John Malcolm McFarlane
George Harry Redmond, Lieut.(E), R.N.
Maxwell Murray Stevenson
Jacques Stockman
Edgar James Terrey

ASSOCIATE MEMBERS

Narendra Bhalla, Lieut.(E), I.N.
Daniel Whyte, B.Sc.

ASSOCIATES

John Anthony Boutland
Maxwell Lyle Bradbury
George Maconachie Brown
William John Cummuskey

John Roderick Dallas
William Ditchburn
Raymond Earl
Eric Bennie Fraser
John Robert Fullick
Bernard Gavin
Leslie Greenacre
Alan George Hall
Arthur Holmes
William Humphries
Stanley Johnson
John Edgar Randle
Peter Craig Roberts
Tom Burnell Snowdon
Walter Turnbull
Dennis Watson
Gilbert Brian Yates
Phiroz Ardeshir Zaveri

GRADUATE
Anthony Andrew Pollock

STUDENT
David Arnold Gutteridge

TRANSFER FROM ASSOCIATE MEMBER TO MEMBER
Eric George Charlton
Thomas Maddison Pallas

TRANSFER FROM ASSOCIATE TO MEMBER
Thomas Charles Bishop
John Armstrong Clay
Philip Henry Hylton
Edgar Albert Jackson
David Donaldson Miller
Sylvester Mathews
Douglas James Rankin

TRANSFER FROM ASSOCIATE TO ASSOCIATE MEMBER
George Frederick Gatward

TRANSFER FROM GRADUATE TO ASSOCIATE MEMBER
Peter James Paxton

TRANSFER FROM STUDENT TO ASSOCIATE MEMBER
Alan William Haughton, Lieut.(E), R.N.
John Edgar Bowell, Lieut.(E), R.N.
George Donaldson Fairley, Lieut.(E), R.N.
Ivan Raoul Jones, Lieut.(E), R.A.N.
Derek Desmond Norton Long, Lieut.(E), R.N.
Donald MacPherson Spiller, Lieut.(E), R.N.

TRANSFER FROM STUDENT TO ASSOCIATE
Neville Dean
Ronald Arthur Rannard, Lieut.(E), R.A.N.R.

TRANSFER FROM STUDENT TO GRADUATE
Robert Henry Crowther, B.Sc.
Peter Guy Edwards, Capt., R.E.

OBITUARY

ALFRED COLYER (Member 9485) was born in 1891 and served an apprenticeship at Chatham Dockyard from 1904-10. He stayed at Chatham until 1915, engaged in work on destroyers of the Tribal class and then spent a year as chargehand of fitters in Malta Dockyard. From 1916-19 he was an engineer overseer for the Admiralty in the Thames district, from 1919-20 he was chargehand of fitters at Chatham Dockyard and from 1920 until his election as an Associate of the Institute in 1942 he was assistant engineer manager at Falmouth for Silley, Cox and Co., Ltd. A few months later Mr. Colyer was transferred from Associateship to full Membership of the Institute.

STANLEY SMITH COOK, B.A., F.R.S. (Member 7151) was born in Canterbury in 1875 and received his general education at the King's School there, from 1889 to 1893, when he went to St. John's College, Cambridge; he graduated in 1896 as Seventh Wrangler in the Mathematical Tripos, and obtained First Class Honours in the Mechanical Sciences Tripos of 1897. He remained at Cambridge until 1898 to study experimental electricity and then went to Newcastle-on-Tyne to serve a three-year apprenticeship with C. A. Parsons and Company. Within a year he had been appointed personal research assistant to Dr. Gerald Stoney, F.R.S., and remained in that position until 1903 when he joined the Parsons Marine Steam Turbine Company. He was associated with Sir Charles Parsons in various researches and in the development of the marine steam turbine and mechanical gearing; in 1910 he was appointed technical manager of the company and a director in 1930. Although he retired from the managership in December 1948, he retained his seat on the board of the company until his death on 21st May 1952.

Mr. Cook was elected a Fellow of the Royal Society in 1928. He was a member of the various technical committees of Pametrada and a member of the research board of the British Shipbuilding Research Association. He was a Member of the Institution of Naval Architects, the Institution of Mechanical Engineers, the Institute of Metals, and a Fellow and Gold Medallist of the North-East Coast Institution of Engineers and Shipbuilders, to which he had belonged for forty years, being a member of their Council and of numerous committees. He was elected to membership of the Institute in 1932.

Mr. Cook contributed numerous papers to these technical societies on steam turbines and their accessories, propellers and mechanical gearing, including the 1938 Parsons Memorial Lecture to the Institution of Mechanical Engineers; of the three papers he read to the Institute between 1934 and 1944, he was awarded the Denny Gold Medal in 1938 for a paper entitled "Modern Marine Steam Turbine Design".

WILLIAM JAMES ELLWOOD (Member 7346) was born in 1866. He spent forty-six years as an engineer at sea, thirty-eight of them with the British India Steam Navigation Co., Ltd., and thirty-five of these as chief engineer. He joined the Institute in 1933 at the time of his retirement. Mr. Ellwood lived with a nephew and his family at Lydiate in Lancashire from 1933 until his death on 20th May 1952.

ARTHUR RODERICK EVANS (Member 11175), who was born in 1887, served an apprenticeship to engineering from 1904-08 with Brecknell, Munro and Rogers, Bristol, and the Tredegar Iron and Coal Company of Tredegar. From 1908-10 he was at sea and obtained a Second Class B.o.T. Certificate in 1910. He was in charge of a repair depot at Bahia Blanca owned by the J. I. Case Company of Wisconsin from 1912-14. During the whole of the First World War, from 1914-19, he served with the Royal Engineers, first as mechanical officer to the Eighth Corps and Fifth Army and later as technical adviser to the Second Army. From 1928-30 he had a business of his own and for the following two years he was without regular employment. In 1931, however, he was appointed plant inspector with H.M. Prison Commission and remained in that employment until his death on 1st June 1952, latterly as an inspector.

Mr. Evans was elected to membership of the Institute in 1947; he was also a Member of the Institute of Fuel and of the Institution of Professional Civil Servants.

ALFRED E. JORDAN (Member 2391) was born in Liverpool in 1875 and served an engineering apprenticeship with the General Steam Navigation Company. For eight years he was employed as an engineer at sea, five of them as chief engineer, and he obtained a First Class B.o.T. Certificate. In 1907 he went to the United States as surveyor for the Salvage Association of London and from 1919-42 he was a consulting engineer, marine surveyor and appraiser in New York, until he joined the States Marine Corporation as vice-president in 1942. He died suddenly on 24th April 1952 while on a visit to Glasgow in connexion with the shipbuilding programme being carried out by the Fairfield Shipbuilding and Engineering Co., Ltd., on behalf of the American corporation.

Mr. Jordan was mayor of the village of Great Neck Estates, Long Island, from 1928-45. He was a Member of the Association of Naval Architects and Marine Engineers, New York, and was elected to membership of the Institute in 1910.

PETER MANSON (Member 1768), born in 1876, was a native of Leith and served his apprenticeship there with William Morrison and Sons. He went to sea in vessels of the British India Steam Navigation Co., Ltd., serving with them until 1904. On obtaining his Extra First Class Certificate in 1905, he went into the drawing office of Hawthorn, Ltd., engineers and shipbuilders of Leith. In 1907 he was appointed engineer and boiler surveyor to the General Accident Insurance Company and left their service on being appointed engineer and ship surveyor in 1908 to the Board of Trade at Glasgow. Mr. Manson remained in Glasgow until 1928, when he was promoted senior engineer surveyor at Liverpool; he also served there as deputy principal officer until July 1941, when he reached retiring age. Because of the war, he continued to serve as engineer surveyor with the Ministry of Transport and latterly with the Director of Merchant Ship Repairs, and retired in December 1949. He died suddenly at home on 12th April 1952. Mr. Manson was elected a Member of the Institute in 1904.

ARCHIBALD VICTOR MONK (Member 10810) was born in 1886 and sailed for seven years, from 1907-14, in ships of the P. and O. S.N. Company, obtaining a First Class B.o.T. Certificate in 1912. From 1914-20 he served in the Royal Navy, mainly in battleships, as engineer lieutenant and engineer lieutenant commander. He was manager of the fuel oil branch of the Anglo-Iranian Oil Co., Ltd., from 1920-41 and again from 1946 until January 1950, when he retired. From 1941-46 he was on the staff of the Engineer-in-Chief of the Fleet. Mr. Monk died on 29th October 1951. He was elected a Member of the Institute in 1946.

LESLIE MOUNT (Associate 13193) was born in 1901. He attended Watford High School and was accepted for training in H.M.S. *Fisgard* as a boy artificer in the Royal Navy in 1917; on completing this apprenticeship in 1921 he went to sea in the H.M.S. *Royal Oak*, serving subsequently in H.M. Ships *Benbow, Carlisle, Frobisher, Hermes, Valiant, Pembroke, Sheffield* and *Bradford*. He left the sea in January 1946, having served as engineer officer in charge since 1941. In August 1946 Mr. Mount was appointed fuel efficiency engineer with the Ministry of Works. He died in the Royal Naval Hospital at Chatham on 23rd April 1952. Mr. Mount was elected an Associate of the Institute in 1951.

GORDON MURRAY (Member 10770) was born in 1904 and apprenticed to Brown Brothers and Co., Ltd., of Edinburgh, from 1920-27. For a year after that he was draughtsman with the Fairfield Shipbuilding and Engineering Co., Ltd., and then until 1931 draughtsman with the Wallsend Slipway and Engineering Co., Ltd. In 1931 he went to sea, serving for seven years as sixth to second engineer with the Eagle Oil and Shipping Co., Ltd.; he obtained a First Class M.o.T. Certificate in 1936. He returned to Brown Brothers and Co., Ltd., as a draughtsman in 1938 and remained with them until 1941 when he volunteered for further sea service and sailed throughout the remainder of the war as chief engineer in various ships. In 1943 the ship in which he was serving was torpedoed, and, with twenty-two other survivors in a lifeboat, it was nineteen days before they reached land. For his "outstanding courage and resource" during these difficult days, Mr. Murray was awarded an O.B.E. in the 1943 Birthday Honours. He was elected a Member of the Institute in 1946.

On finally leaving the sea in 1947, he acted as superintendent engineer for H. M. Thomson, Esq., and established himself in business in Edinburgh, representing R. Hood Haggie and Sons, Ltd., patent wire and hemp rope manufacturers of Newcastle-on-Tyne, W. and J. Leigh, paint manufacturers of Bolton, and White's Marine Engineering Co., Ltd., Hebburn-on-Tyne. On 16th November 1951, however, he became suddenly ill and died two days later.

JOHN PATTIE (Member 7909), who was born in 1881, served a five years' apprenticeship with James Scott of Tayport. He went to sea in the first place with the Currie Line and then served for eleven years in vessels owned by J. P. Bruce of Dundee. During this time he obtained a First Class B.o.T. Certificate. Then for twenty-one years he was employed by the British Tanker Co., Ltd., and when, in 1935, he was elected a Member of the Institute, he was a superintendent engineer for the company, stationed at Swansea. He left the sea and purchased for his son a garage business in Alloa, in which he assisted; however, his son's health broke down and the family moved to Tayport. There, Mr. Pattie's son died in 1948, his wife in 1950 and himself in October 1951.

ALAN GRANT RICHARDSON (Member 5979) served an apprenticeship from 1906-11 with Campbell and Calderwood and for the following year he was a draughtsman with Dunsmuir and Jackson. Throughout the 1914-18 war he served as an officer with the Royal Engineers and was awarded the Military Cross in 1915; later he received a Bar to this decoration and was mentioned several times in despatches. For two years he was in the estimating department of David Rowan and Co., Ltd., of Glasgow, and followed this with two years as outside manager for Rankin and Blackmore, Ltd.

In 1921 Mr. Richardson was appointed chief marine superintendent engineer in Abadan for the Anglo-Persian Oil Co., Ltd.; he remained there until 1927, when he was transferred to the London office of the company and continued in their employment until his retirement, only a few weeks before he died on 15th May 1952. He had been a Member of the Institute since 1928.

WILLIAM WHYTE (Member 8185), who was born in 1901, served an apprenticeship with Harland and Wolff, Ltd., in Belfast and Glasgow. From 1923-30 he was at sea in ships owned by Elder, Dempster and Co., Ltd., and obtained a First Class Motor B.o.T. Certificate. In 1930 he was chief engineer of the vessel *Cementkarrier* owned by the Canada Cement Company and in 1932 he was second engineer of one of the Burns and Laird Lines' ships, s.s. *Lairdsgrove*.

From 1932 until 1935 he worked for the British Broadcasting Corporation as a Diesel shift engineer, first at their Scottish regional transmitter and then at the West regional transmitter. In 1935 he went out to the Gold Coast where he remained until his sudden death while swimming on 26th April 1952. Until 1937 he was shift engineer at the Sekondi power house of the Gold Coast Government Railways and was then transferred to work in the Electrical Department; in August 1949, however, he returned to the Railway Administration on his appointment as superintendent marine engineer of Takoradi Harbour. Mr. Whyte had been a Member of the Institute since 1936.

Obituary

JAN DANIEL WILTON (Member 5218) was born at Rotterdam on 24th April 1877. After serving an apprenticeship and gaining further experience with Earle's Shipbuilding and Engineering Company, Hull, he joined the company, Wilton's Engineering and Slipway Company of Rotterdam, which had developed from the small smithy founded by his father. In 1906 he was elected a managing director, a position which he held until his retirement in 1935. He remained on the board of directors of the company until his death on 16th April 1952.

During his years of service in the company, Mr. Wilton was in charge of the engineering drawing office; he was also particularly concerned with the social welfare and training of their apprentices and was largely responsible for the inauguration of the company's present professional training school for apprentices. He was elected a Member of the Institute in 1924.

LEE WOOD (Member 3635) was born in Birkenhead, where he served an engineering apprenticeship with W. Rowlanson, engineers and millwrights. He joined the Pacific Steam Navigation Company in 1890 as a junior engineer and was promoted chief engineer of the *Sarmiento* in 1906, later holding that rank in the steamers *Oropesa, Bogota, Mexico,* and in the motorship *Lobos*. He was appointed superintendent engineer of the company in 1918 and retired in 1933 after forty-three years' service. He was one of the pioneers in the application and adoption of the oil engine in merchant ships and was prominently identified with the construction of the Pacific Steam Navigation Company's motorships.

After his retirement from the company's service he started business on his own account in Liverpool as a consulting marine engineer and acted for a period as Liverpool representative of Harland and Wolff, Ltd.

Mr. Wood was a past chairman of the Liverpool Marine Engineers' and Naval Architects' Guild, of which he was honorary treasurer and a trustee at the time of his death. He was a Member of the Institute from 1919 and served from 1935 until 1947 as Vice-President for Liverpool.