Some Observations on Marine Gearing*

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The purpose of this paper is to indicate how the loading of marine steam turbine gearing may rise considerably higher than the value calculated on the basis of mean torque. In some instances there are periodic fluctuations of torque having peak values at least 100 per cent in excess of the mean; in other cases the increase is a constant load due to additional resistance to the hull.

There are so many different factors influencing gear tooth loading that a complete analysis is quite beyond the scope of this paper, consequently it is only possible to describe briefly those which are most important.

The relative proportions of the teeth most commonly used for marine installations are shown in Fig. 1, in which all the teeth are drawn to the same scale.

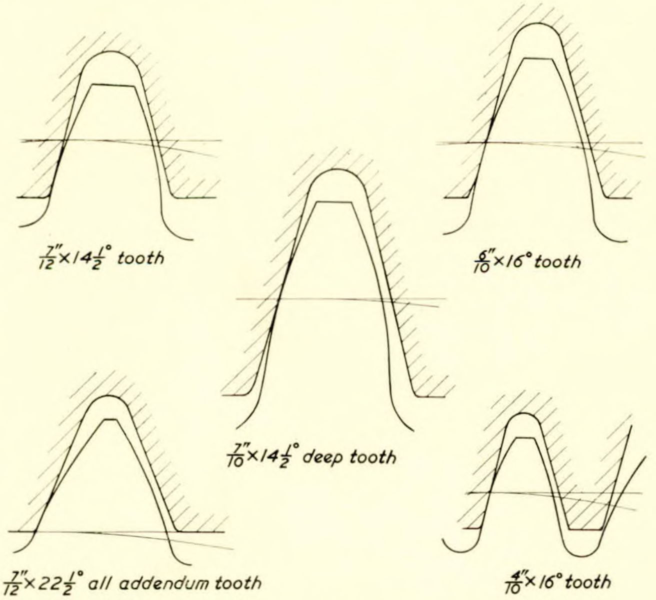

FIG. 1—*Diagram showing relative proportions of teeth in common use for marine gears*

Engagement between a pinion and wheel is shown in Figs. 2 and 3, which serve in particular to illustrate the properties of the line of contact. This line is tangential to the base circles of the involute curves of both the pinion and wheel teeth and passes through the pitch point of the gears; it is also normal to the pressure angle of the teeth. Contact between two teeth

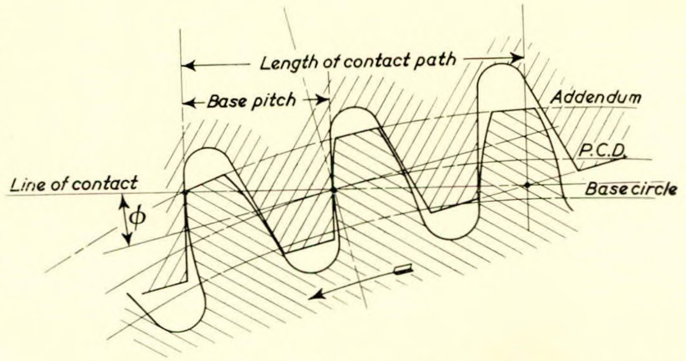

FIG. 2—*14½-deg. ordinary involute; 30-tooth pinion meshing with a rack; 7/12-in. × 14½-deg. tooth form*

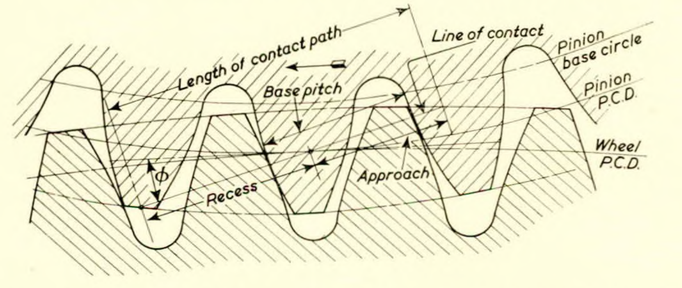

FIG. 3—*6/10-in. × 16-deg. tooth form (modified addendum); 40-tooth pinion engaging with 122-tooth wheel drawn in the circular plane*

in a pair of gears begins at the intersection of this line with the addendum circle of the wheel teeth and is completed at its intersection with the addendum circle of the pinion teeth. The line of contact is therefore divided into parts, the approach and recess portions respectively, while the ratio of its total length to the base pitch represents the number of teeth instantaneously in contact. As is indicated in the diagram, the length of the approach portion is wholly dependent upon wheel dimensions and similarly the length of recess is dependent upon pinion dimensions only, being derived from the addendum and diameter in each case.

The relative motion between two teeth in engagement is pure rolling at the pitch point only; elsewhere it is a combination of rolling and sliding. Since the velocity of rolling is at a minimum at the root of the tooth and increases to a maximum at the tip of the tooth, initial contact between a pair of engaging teeth takes place under conditions of maximum relative sliding velocity. Similar conditions exist at the end of recess where the relative sliding velocity between the disengaging teeth also reaches a maximum value. There is, however, the funda-

* Presented at a meeting of the Merseyside and North Western Section on 7th February 1955.

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mental difference that during engagement the tip of the wheel tooth has a higher rolling velocity than the root of the pinion tooth, while conversely at disengagement the tip of the pinion tooth has a higher rolling velocity than the root of the wheel tooth.

Scuffing normally appears on the addendum of both wheel and pinion teeth, hence it must occur on the wheel teeth during approach, while on the pinion teeth it must occur during recess, in both instances showing a preference for the surface having the higher rolling velocity. In a similar manner it is significant that pitting normally appears at the pitch line and on the dedendum.

Due to the load being transmitted by a pair of gears the teeth in contact are deflected from their normal unstrained position and it follows that the disengaged teeth of the pinion are then in advance of their true position while the disengaged teeth of the wheel are retarded and consequently the engagement of successive teeth, which are unloaded and therefore undeflected, takes place while they are out of pitch by the amount of the double deflection. At the commencement of contact the root of a pinion tooth engages with the tip of a wheel tooth and it is clear that such deflection imposes severe loading conditions during the period of the highest relative sliding velocity. As contact progresses the newly engaged teeth gradually assume their share of the load while the relative sliding velocity diminishes until it disappears when contact coincides with the pitch point of the gears. As engagement continues the contact point moves towards the tip of the pinion tooth and towards the root of the wheel tooth, where conditions again resemble those at the beginning of contact. It is apparent that the bending stress is greatest on wheel teeth during approach and on pinion teeth during recess, and that the most severe conditions of contact occur at the tips of the teeth. Hence it is essential that some tip relief should be provided in order to assist engagement and disengagement. It is also clear that contact conditions will deteriorate very rapidly when deflexion is increased, due to additional loading.

To compensate for the deflexion of the teeth the amount of tip relief provided should equal the combined effect of the circumferential displacement of the two gears when transmitting their full power and should vary with the depth and and root thickness of the teeth. Hence the relatively flexible 7/10-in. deep tooth requires considerably more tip relief than the 7/12-in. normal depth tooth which is, of course, a more rigid form. The normal deflexion of a 7/10 × 14½-deg. tooth is about 0.001 under service power loading; a total tip relief of 0.002 is added to compensate for deflexion of wheel and pinion (Fig. 4).

Excessive relief causes a limitation of the effective depth of tooth and reduces the length of the contact line which is equal,

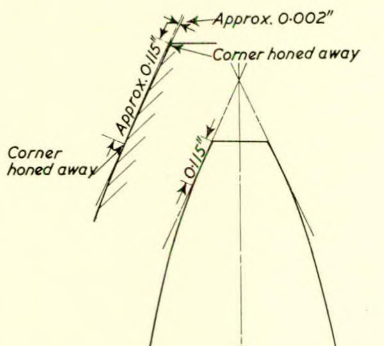

FIG. 4—*Diagram showing extent of tip relief on main wheel teeth 7/10-in.× 14½-deg. secondary gears (normal plane)*

of course, to a reduction in the number of teeth instantaneously in contact. It is, therefore, important that the designed depth of penetration of hobs should not be exceeded.

The fundamental difference between the "Parsons Improved Involute All Addendum" tooth and the standard form is that, in the former, due to the absence of addendum on the wheel teeth, the arc of approach does not exist and consequently the first point of contact between mating teeth is at the pitch point where the rolling speeds of both teeth are equal and sliding does not occur. Another feature of this form of tooth is the increased stiffness and root strength resulting from the adoption of the 22½-deg. pressure angle. In addition, the greater length of addendum on the pinion teeth increases the length of the arc of recess above that obtained with teeth of the normal type having a 14½-deg. pressure angle. This form of tooth is subject to an unusually short length of line of contact (due to the absence of the approach portion) and consequently the increase in the recess portion is most desirable, although it does introduce a higher sliding velocity at the end of recess than occurs at similar speeds of rotation with ordinary gears. Due to the superior rigidity of "AA" teeth, only a limited amount of tip relief is necessary (Fig. 5).

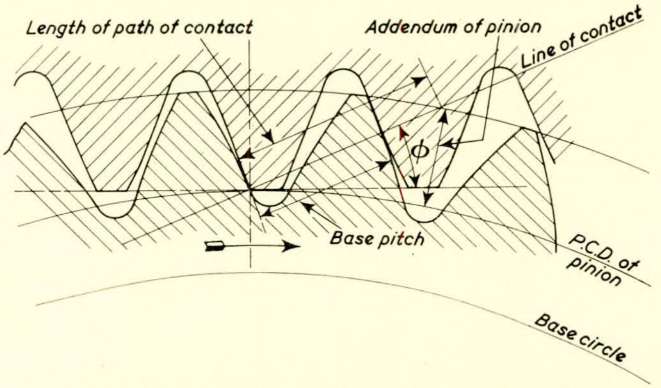

FIG. 5—*7/12-in. × 22½-deg. A.A. tooth form; 30-tooth pinion meshing with a rack*

$$\textit{Contact ratio} = \frac{\textit{contact path length}}{\textit{base pitch}}$$

The curvature of the involute increases uniformly from the base circle to the tip of the tooth and the hob cuts, which are a succession of flats tangential to the involute curve, gradually increase in width of spacing as their distance from the base circle is increased (Fig. 6). Due to the absence of dedendum on a pinion having "AA" teeth, the generation occurs at a distance from the base circle and consequently the tooth profile is produced by considerably less in number cuts of greater width than normally perform the generation of an ordinary tooth and the resulting finish is correspondingly rougher. In view of this inherent disadvantage, it is clearly evident that in order to attain a reasonable degree of finish on this particular type of tooth, it is essential that imposed errors resulting from periodic errors of the hobbing machine should be a minimum.

Some uncertainty exists regarding the maximum pitch line velocity allowable but it is suggested that speeds have not yet approached a limiting value and that failures which have been attributed to this cause were in fact due to additional tooth loading induced by gear hobbing errors in conjunction with the dynamic properties of the system.

Until recently, it has been normal practice to use the 7/12-in. ×22½-deg. flank angle all-addendum tooth for primary reduction gearing and, when all the characteristics of the system are suitably related, the tooth gives very good service and is perfectly satisfactory in all, including the most modern, applications. It is being displaced, however, by the 6/10-in. × 16-deg. pressure angle tooth having an addendum on the pinion tooth equal to 60 per cent of the total meshing depth, the remaining 40 per cent being dedendum. The introduction of this tooth for primary gears represents an attempt to reduce

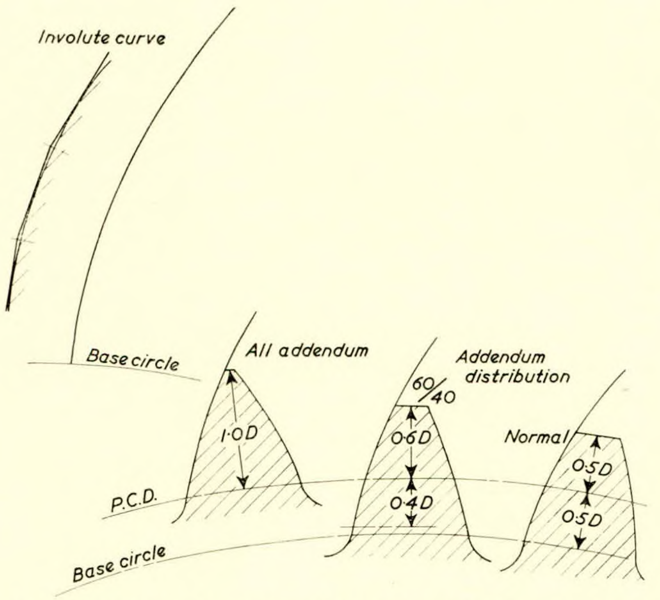

FIG. 6—*Diagram showing relationship between percentage addendum and base circle, indicating the progressive increase in width of cut produced by a hob cutting remote from the base circle.*

Approximate number of cuts forming profile:

Pinions	Ordinary 14½—degree	All addendum 22½—degree
	43	15
Wheels	29	All dedendum 22½—degree
		19

the relative sliding velocity below that normally associated with the all-addendum tooth and on the basis of the foregoing remarks it would appear that it may be more susceptible to pitting than to scuffing. Generation of the profile is accomplished by approximately twice as many hob cuts as form the involute curve of an all-addendum tooth and consequently it is considerably easier to obtain a relatively superior finish on the 6/10-in. × 16-deg. tooth, a factor which enhances its load carrying capacity, especially in the initial stages.

With few exceptions secondary gear units employ the 7/10-in. × 14½-deg. deep tooth having equal addendum-dedendum distribution. The tooth was introduced mainly for use in interleaved installations where it provided a greater degree of flexibility than was possible with the forms of tooth hitherto used. Increased sliding velocity results from the extra depth and, although the tooth is satisfactory, there are conditions attached to its application which must be fulfilled.

Under perfect conditions of gear cutting and in the absence of any extraneous impressed forces acting on the gear teeth, the conditions of loading would be governed by the number of teeth instantaneously in contact and the surface available to share the load. As stated previously the former is related to the depth of the teeth and the flank angle while the surface available to share the load is dependent upon the curvature of the teeth at the point of engagement.

Severe conditions of tooth loading occur in heavy weather and under such conditions the desire to maintain speed is more dangerous with a turbine than with a reciprocating engine, either steam or Diesel. With reciprocating engines the mean pressure in the cylinders remains almost constant irrespective of the revolutions at which the engines are running and, consequently, in rough weather the torque developed will remain constant and the power will be reduced. Under similar conditions, however, the steam turbine will maintain its power essentially, notwithstanding reduced ship speed, and there is a corresponding increase in torque throughout the system.

As a rough guide to the degree of torque variation possible in heavy weather in the Western Ocean, in one vessel a speed fluctuation of 16 r.p.m. has been observed. In this instance the normal speed in smooth water was 110 r.p.m. and, although the steam quality and quantity to the turbines remained unaltered, the speed increased to 116 r.p.m. and decreased to 100 r.p.m. at each cycle of the vessel's pitch. The lower speed occurred when the stern lifted and the bow plunged into a sea. Calculations of the kinetic energy expended by the rotors indicated a torque increase of approximately 25 per cent in the h.p. and 80 per cent in the l.p. secondary pinions, these values being additional to the mean transmission torque.

It appears that these conditions represent the normal to be expected on the North Atlantic run and are, therefore, an indication of the increase in tooth loading which commonly occurs and frequently causes pitting of the gear teeth. It is also a reason why Lloyd's "K" factor for teeth is necessarily a conservative value.

Precession Effects

A further result of heavy weather is the gyroscopic effect on the rotating parts. If a turbine rotor is spinning about its axis and a couple is applied which tends to turn the rotor about another axis perpendicular to the axis of spin, the rotor will endeavour to rotate about a third axis perpendicular to each of the other two, for example, in a horizontal athwartships plane when pitching, or in a vertical fore and aft plane when turning. This tendency to precess is resisted by a couple exerted by the bearings, equal in magnitude and opposite in sense, to the precession couple.

All rotating units having their axis lying fore and aft are subjected to gyroscopic forces when the vessel is either pitching or turning (Fig. 7).

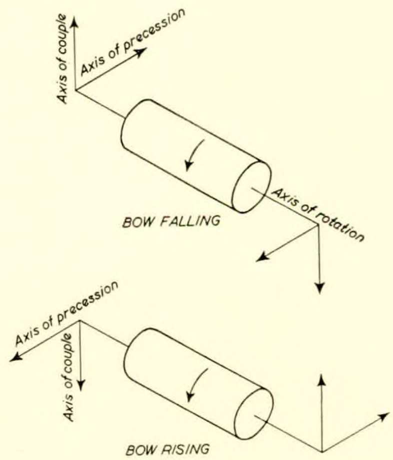

FIG. 7—*Diagram showing effect of precession on a rotating mass; motor rotating anti-clockwise looking aft*

For the determination of the magnitude of the restraining couple the predominating factor is the maximum angular velocity of swing in the plane perpendicular to the axis of spin and is denoted by ω_1.

The maximum angular velocity of the vessel pitching in simple harmonic motion is then given by

$$\omega_1 = \frac{\theta \text{ radians}}{\text{t seconds} \times 0.636}$$

$$= \frac{\text{angle of pitch in degrees} \times 2\pi}{\text{time of } \frac{1}{2} \text{ cycle of pitch in secs.} \times 360 \times 0.636}$$

$$= \text{radians per second}$$

From observation it appears that in the North Atlantic the period of pitching is about 6 seconds and an angle of pitch of 12 degrees is assumed. Then, as an example,

$$\omega_1 = \frac{12 \times 2\pi}{3 \times 360 \times 0.636} = 0{\cdot}11 \text{ radians per second}$$

Assuming that the main gear wheel mass moment of inertia $J = 28{\cdot}0$ tons ft. sec.^{2} units and that its speed when racing is 120 r.p.m., then its maximum angular velocity is equal to

$$\omega_2 = \frac{2\pi \times \text{r.p.m.}}{60} = 12{\cdot}57 \text{ radians per second}$$

The precession couple is equal to $J\omega_1\omega_2 = 28{\cdot}0 \times 0{\cdot}11 \times 12{\cdot}57 = 38{\cdot}7$ tons ft. and assuming that the main wheel bearings are spaced 6 feet apart, then the force at each bearing is $\frac{38{\cdot}7}{6} = 6{\cdot}45$ tons.

The additional bearing pressure caused by the precession of a rotor is usually small but there is a corresponding disturbance of alignment due to the periodic variation in the position of the load line of the bearing. In the case of primary flexible couplings the rotor and pinion are rotating in the same direction and consequently precess in the same direction. This imposes a periodic misalignment between the adjacent rotating elements and is one of the reasons why it is necessary to barrel the teeth of flexible couplings (Fig. 8).

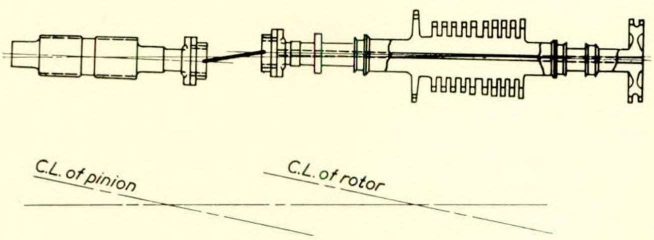

FIG. 8—*Diagram showing effect of precession on flexible couplings*

A pair of meshed gears rotate in opposite directions to each other (Fig. 9) and consequently their precession is of opposite phase. During each oscillation of the vessel the precession couple changes sign and its effect is either to be added to or subtracted from the bearing loading polygon,

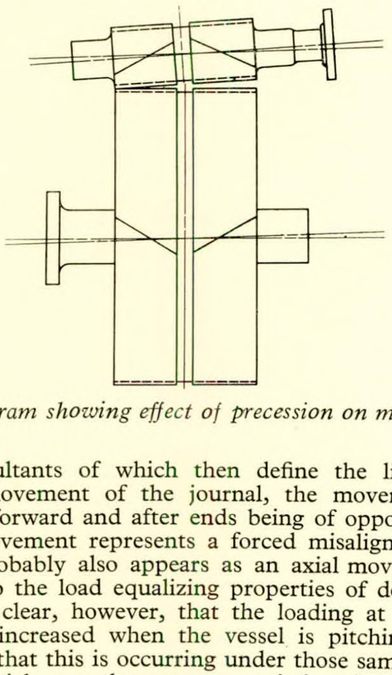

FIG. 9—*Diagram showing effect of precession on meshing gears*

the two resultants of which then define the limits of the oscillatory movement of the journal, the movement of the shaft at the forward and after ends being of opposite sign.

This movement represents a forced misalignment of the gears and probably also appears as an axial movement of the pinion due to the load equalizing properties of double helical teeth. It is clear, however, that the loading at the ends of the teeth is increased when the vessel is pitching and it is equally clear that this is occurring under those same conditions of weather which cause large torque variations in the system.

Rapid Manoeuvring

It is well-known that tooth loading reaches high limits when manoeuvring, the worst conditions being when the vessel is moving full astern and ahead steam is admitted to the turbines. In an impulse turbine, the standstill torque produced when admitting the full quantity of ahead steam to blades which are held stationary is theoretically equal to twice the normal full power torque. In practice, allowing for losses, the value is approximately 1.75.

Hence the high torques produced during manœuvring are occurring under the worst possible conditions of lubrication, while the load-carrying capacity of the teeth is lowest at zero speed. It is unfortunate that, in general, the first movements of the machinery in a new vessel are of this type and occur before the surfaces of the teeth have become work-hardened. It is therefore good practice to have a prolonged basin trial and to restrict engine movement until the vessel is clear of dock gates and in the fairway. Even on a basin trial the torque is higher than it would be at the same revolutions with the vessel moving.

The temperament of the man on the controls contributes largely to the preservation of gearing during the first twelve months of its life and efforts should be made to ensure that he is aware of the forces involved.

Inertia Effects

The relative mass effects of the various rotating components constituting a turbine installation, together with the torsional stiffness of the connecting shafts, are factors which determine the magnitude of the impacting forces resulting from gear tooth errors.

Considering a pitch error between two teeth of a primary pinion (Fig. 10), there will be a corresponding cyclic acceleration and deceleration of the pitch circle velocity at this point in every revolution. The proportional mass effect between rotor and primary wheel, together with the stiffness of the shaft,

FIG. 10—*Typical cumulative pitch errors (main wheel)*

will determine the proportional distribution of the effect of the error. If the mass moment of inertia of the rotor exceeds that of the primary wheel, the rotor will persist rotating at uniform angular velocity, more or less undisturbed by the pitch error in the pinion, which will then reflect in the rotation of the primary wheel. Conversely, when the mass of the primary wheel exceeds that of the rotor the latter is subjected to the fluctuations in speed produced by the errors in the teeth of the pinion.

Similar relationships exist between the primary and secondary gears where the wheel having the greater mass effect controls the rotation of the other.

It will be noted that this is applicable to all kinds of turbines and gearing and also includes all types of irregularities in gear teeth, including eccentricity of pitch circle relative to journals. There is a distinction between articulated gears and interleaved, the quill shafts of the former preclude the transmission of errors between primary and secondary gearing, but rigid connexion of interleaved gears introduces the possibility of the combination of the effects of the primary and secondary errors.

A few examples of each type have been calculated in order to obtain a more complete understanding of the fundamental differences between the articulated and interleaved gearing. The results may be summarized as follows, and are based on gears containing errors equal to Grade A:—

The inertia loading of l.p. primary gears with either type of installation is about 100 per cent in excess of the mean value owing to the large mass effect of the rotor. On secondary gear teeth the inertia loading increase above the mean value is about 15-20 per cent with interleaved gearing. There is no increase at the secondary teeth of articulated gears having normal quill shafts.

Longitudinal Expansion of Gears

It is common practice to cut gears so that the after helix of the main wheel is right-handed, thereby making the inner ends of the teeth form the arrow point when rotating in the ahead direction.

The running temperature of a pinion usually exceeds that of the main wheel by about 10-20 deg. F. so that there is a differential expansion between them. Assuming a pinion 60 inches long having a temperature 15 deg. F. above that of the wheel, there will be a total differential expansion of 0.006 inch, i.e. equivalent to an axial pitch error of 0.003 inch in each helix (Fig. 11).

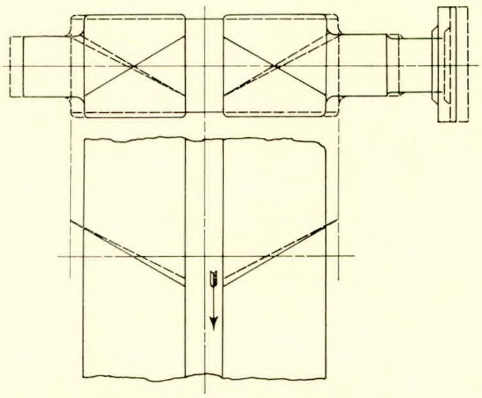

FIG. 11—*Diagram showing effect of differential axial expansion between gears*

Owing to the self-centreing properties of double helical gears, this will result in a concentration of loading on the teeth at the outside ends of the helices. Hence it would appear that if the gears were cut to the opposite hand they might be more satisfactory because then the highest contact pressure would occur at the centre of the gears, adjacent to the gap, where it would tend to compensate the effect of deflexion and twist of the pinion under load.

If this conjecture is valid it should follow that, in a twin-screw vessel having outward turning propellers, the port set of gearing should normally be in a better condition than the starboard set.

A feature which has received some attention is the radial expansion of gears together with the linear expansion of gear cases.

Using 7/10-in. × 14½-deg. hobs of the present design gives a clearance of 0.035 inch between the tips of the pinion teeth and the point of intersection of the profile and root radius of the wheel teeth under ideal conditions (Fig. 12). In normal installations the position of the l.p. pinion load line usually results in that pinion being at the top of its bearings when under full load. At lower powers and when stopping, this pinion gradually approaches the main wheel approximately to the extent of the clearance in the bearings. If this clearance is 0.015 inch, it means that there remains only 0.020 inch to cover the relative expansion between the main wheel and the gear case.

The coefficient of linear expansion of cast iron is 0.0000059 in./in./deg. F. and for steel is 0.0000067 in./in./deg. F., showing that the worst combination would be a steel wheel in a cast iron gear case. Furthermore, the greater mass of the main wheel retains heat for a longer period than the gear case, which may be exposed to the flow of ventilating air. Hence,

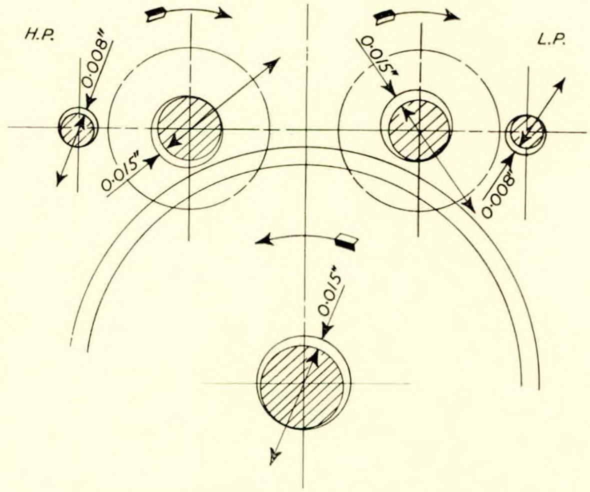

FIG. 12—*Typical double reduction arrangement showing attitude of journals in their respective bearings when running ahead*

when the machinery is slowed down, the different rates of contraction tend to reduce the clearance at the roots of the teeth.

In the case of a main wheel 14 ft. 0 in. diameter, there can be 36 deg. F. temperature difference between a wheel and gear case of similar materials before interference at the roots of the teeth occurs. With a steel wheel and a cast iron gear case, interference begins at a temperature difference of 32 deg. F.

The addition of tip relief delays interference and permits more relative movement between teeth than would otherwise be possible without damage. It is suggested that this may account for the success which has been achieved on many ships by adopting tip relief on the gears and thereby preventing scuffing (Fig. 13).

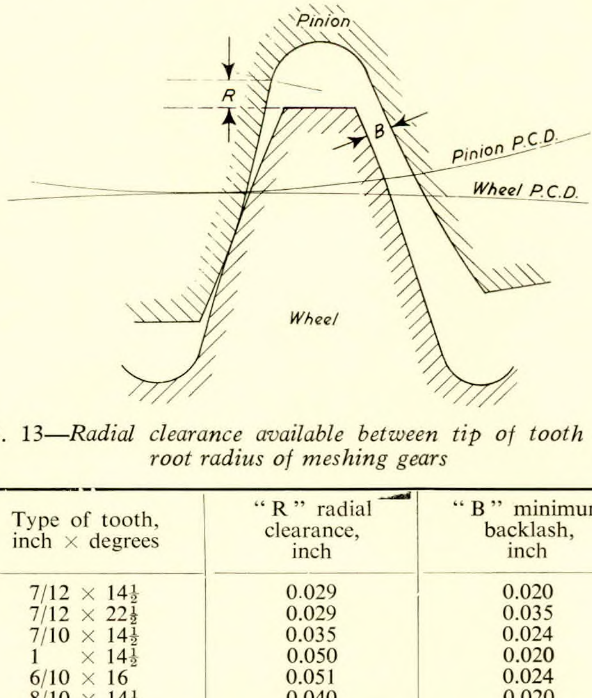

FIG. 13—*Radial clearance available between tip of tooth and root radius of meshing gears*

Type of tooth, inch × degrees	"R" radial clearance, inch	"B" minimum backlash, inch
7/12 × 14½	0.029	0.020
7/12 × 22½	0.029	0.035
7/10 × 14½	0.035	0.024
1 × 14½	0.050	0.020
6/10 × 16	0.051	0.024
8/10 × 14½	0.040	0.020

Axial Movement of Pinions

All pinions shuttle to some extent and until the inaccuracies of gear cutting are eliminated and perfect tooth action becomes possible, axial sliding between mating teeth is inevitable. In addition, shuttling may be due to rotational errors of the main wheel imparting a low frequency surge to the pinions in an axial direction.

A pinion transmitting torque will naturally centralize itself in order that both helices share the load equally and consequently, if the contact pressure on one helix is varied by undulations or irregularities on the tooth surface, the pinion will slide axially until equal load sharing between the helices is re-established.

Another form of gear cutting inaccuracy which causes axial shuttling of the pinions is the combined effect of the cumulative pitch errors on both helices of wheel and pinion (Fig. 10).

Considering first a pinion having a cumulative pitch error on each helix approximating to a sine curve, and in phase with each other, the rotational effect will be that of an eccentric and load sharing between the helices will always be equal. If, however, the sine curves are displaced by any phase angle the effect is the same as a slight change in the helical angle of the teeth and consequently the pinion will immediately slide and re-establish equal loading on each helix. The frequency of axial shuttling is dependent upon the type and form of the curve of cumulative pitch error but is generally once per revolution of the pinion.

It follows that when primary wheels have similar cumulative pitch errors having phase coincidence between the two helices, the effect is impressed on the associated pinion and usually indicates one cycle of error in a number of revolutions of the pinion equal to the gear ratio existing between the gears.

The conditions between the main wheel and secondary pinions are exactly similar and it is usual to find the pinions have a once per revolution combined with a cycle covering the number of revolutions of the pinion equal to the gear ratio.

It may be interesting to note that the once familiar beat which was evident in the noise from the earlier gears was due to the cyclic combination of cumulative pitch errors of the pinions and wheel having their phase relationship changed by the hunting tooth, one cycle of the hunt being performed at each waist of the beat. A secondary effect is produced by the larger pinion of the l.p. turbine which naturally has a different cycle to the others. Periodic coincidence occurs when all the waists appear together and produce the resonant condition which was familiar at frequent intervals in a turbine engine-room. With the improvement in accuracy of gears this has disappeared from the modern engine room.

Thrust Block Seatings

The axial movement of pinions is also associated with the rigidity of the thrust block seating, together with the magnitude of the impulses arising from the propeller blades. In addition to the axial movements mentioned previously, which were a function of gear manufacture, considerable axial movements of gearing have been recorded having a frequency of four per revolution of the propeller. In these cases the propeller blade

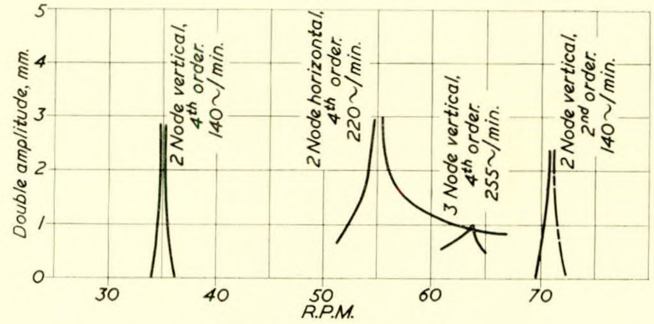

FIG. 15—*Hull vibration*

clearance in the aperture was small and the thrust seatings were not sufficiently rigid in the fore and aft direction.

The recorded movement amounted to about 0.030 inch at a 4 per revolution of propeller frequency, and extended throughout the gears to the primary pinions where it was recorded at the same amplitude as at the main wheel. A similar type of vibration has also been recorded, occurring at a frequency of 8 times per revolution of the propeller.

When gearing is subjected to vibration of this type the wear on the teeth is naturally excessive; their appearance, however, is deceptive as the extremely high polish which results from vibration movement is of course quite different from the finish found on good gears after normal running.

The subject of hull vibration associated with thrust block seating rigidity, strength of gear case and seating, propeller and aperture design, all influence the performance of the main gearing and it is important that, when a critical examination of gearing is being undertaken, these factors should be given consideration. Incidentally, fretting and wear of flexible coupling teeth is most usually associated with vibration of this type.

In order to indicate the relationship of the various frequencies of hull vibration and to show how difficult it is to avoid resonance at some condition of loading, vibration records from two vessels are shown (Figs. 14, 15). Fig. 16 shows the variation in frequency with loading of a vessel and demonstrates the difficulties which are being experienced in attempting to avoid resonance. This vessel is old and it appears

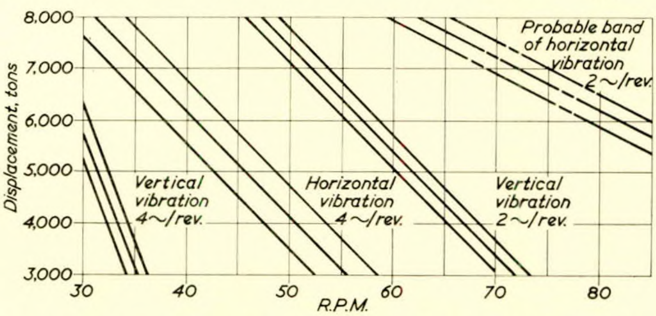

FIG. 16—*Example showing variation in frequency of hull vibration with displacement in an old vessel*

reasonable to presume that originally the natural frequencies were much higher than they are now but that the general softening of the hull has caused a decline to the present level.

Figs. 17 and 18 show examples of records of axial shuttling of gearing due to flexibility of thrust seating and inaccurate primary wheel.

The foregoing examples do not entirely complete the study of the torque fluctuations arising in service; there are other causes which are still more obscure and are not yet fully understood. The information given, however, does indicate that the additional forces produced in bad weather are of a cumulative nature, i.e. the torque increase due to pitching occurs at the same instant as the precession forces are most effective; furthermore, these very conditions might be expected

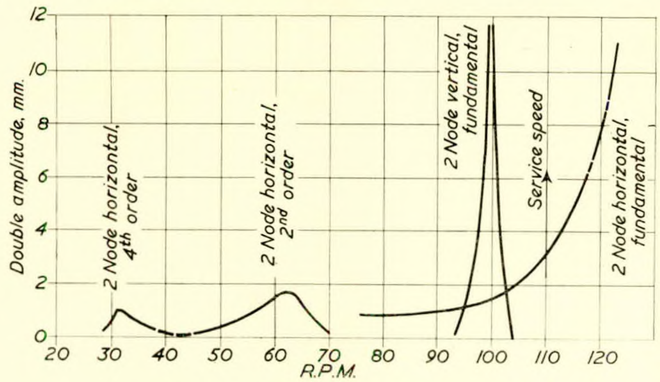

FIG. 14—*Hull vibration*

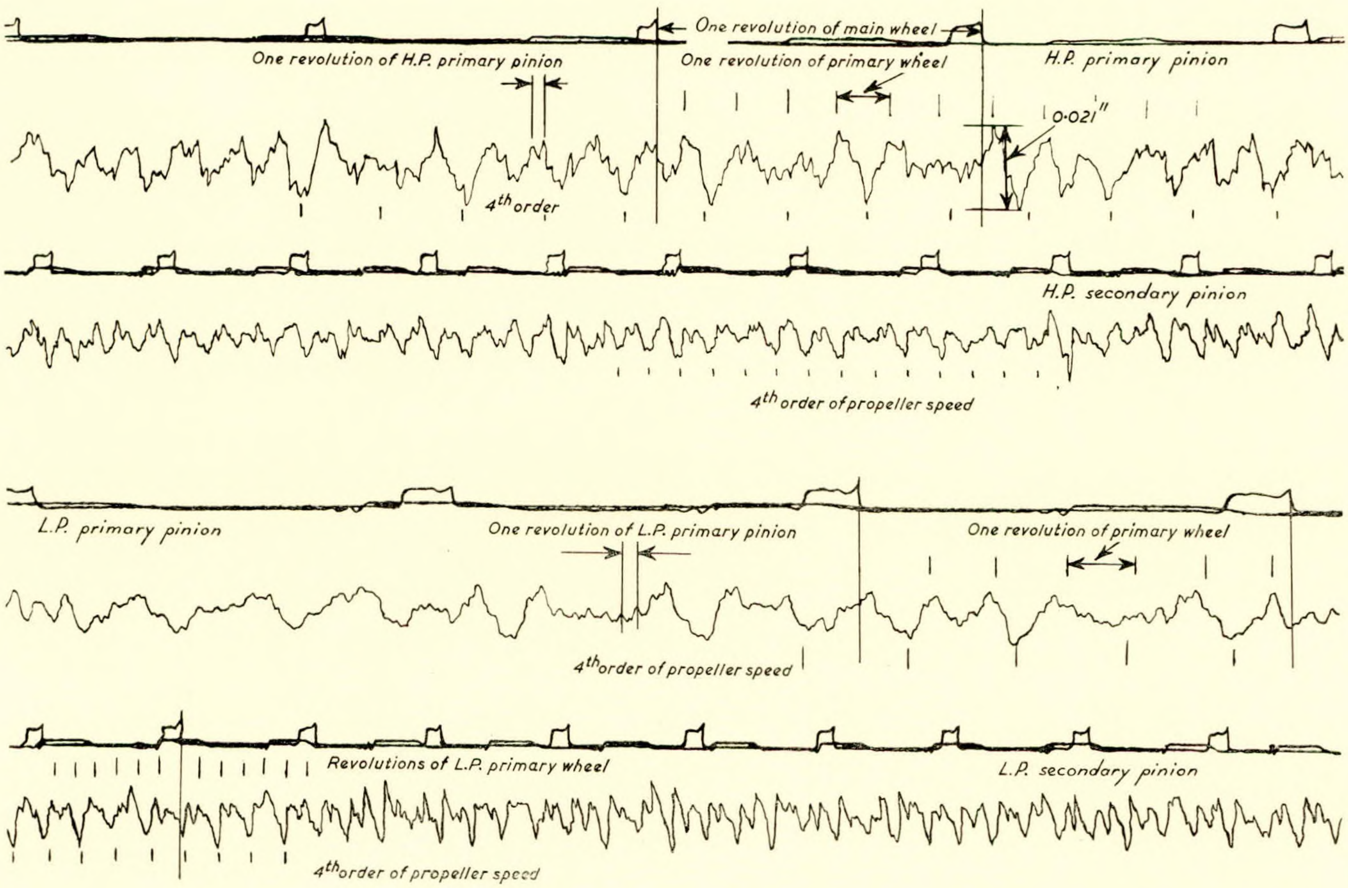

FIG. 17—*Records of axial vibratory movement of gearing*

to increase the temperature within the gear case, thereby accentuating the differential expansion between the gears.

Gearing Design

Reviewing gearing design on the basis of the foregoing remarks, together with a knowledge of the characteristics of installations both satisfactory and unsatisfactory, enables new design to be approached with a degree of confidence.

Under normal circumstances single reduction gearing does not present difficulty, its use being limited to vessels operating with moderate steam conditions and having correspondingly low turbine speeds. The type of tooth normally used is 7/12-in. $\times 14\frac{1}{2}$-deg. ordinary involute form running at pitch line speeds up to about 110 ft. per sec. Inertia forces of the type previously discussed are normally insignificant and are ignored in design calculations. The critical speed of torsional oscillation usually occurs at about 35 r.p.m. in amidship installations of this type and is well below the running speed.

In general, installations of this type are not particularly troublesome and under normal conditions and with reasonable accuracy of gear cutting are likely to give satisfactory service.

Double Reduction Gearing

Double reduction gearing presents many more difficulties than single reduction and some uncertainty exists regarding the respective merits of the interleaved system and the articulated gearing having quill drives, resulting in many instances of the one being employed where the other would have been more advantageous.

The question of choice is primarily based upon the relationship existing between the inertia of the masses, together with a consideration of the torsional vibration characteristics of the system and although it is never detrimental to fit quill drives there are occasions when they are unnecessary and therefore uneconomical. Conversely, interleaved gearing may become troublesome in circumstances where conditions are not altogether suitable for its use.

In general, the torsional vibration aspect concerns only articulated gears having quill shafts, while the relative mass effects of the wheels and rotors in association with the gear cutting errors cause more serious periodic increase of the tooth loading of interleaved gears than of quill drives.

To comply with advanced steam conditions and retain a reasonable diameter of rotor necessitates a high turbine speed and consequently double reduction gearing is essential in order to permit suitable propeller speeds. Due regard must be given to the proportion of the reduction allotted to each of the gear systems and also to the limiting value of pitch line velocity.

Primarily, load carrying capacity increases proportionately with increased pitch line velocity and maximum permissible values of the latter are not yet known with any degree of certainty; however, the following values represent the maximum speed employed in present-day merchant vessel construction in the United Kingdom.

	Pitch line velocity, ft. per sec. Articulated quill drive	Interleaved
Primary gears, 7/12 in. $\times 22\frac{1}{2}$ deg. AA.	224	175
Secondary gears, 7/10 in. $\times 14\frac{1}{2}$ deg. deep	80	80
Single reduction gears, 7/12 in. $\times 14\frac{1}{2}$ deg.	— 110	—

A summary of the formulæ used in the initial determination of gearing dimensions indicates the effect obtainable by variation of any factor.

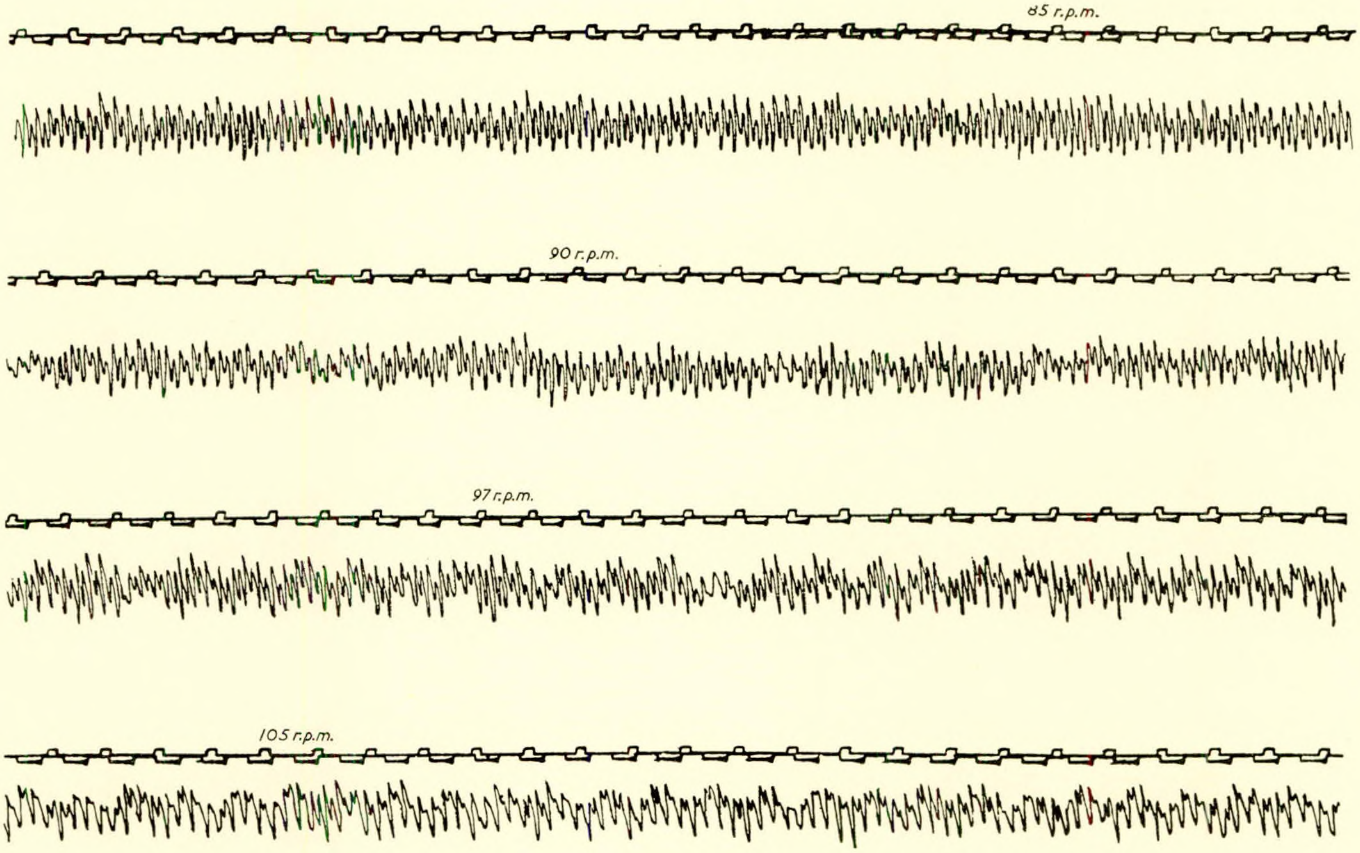

FIG. 18—*Records of axial vibratory movement of main wheel*

F = tooth loading factor
W = tooth loading lb. per inch face width
C.R. = contact ratio
r_c = relative curvature of teeth at pitch point
r = radius of pinion in inches
R = radius of wheel in inches
T = torque in lb. in.
V = pitch line velocity ft. per sec.

Then $T = \dfrac{\text{S.H.P.} \times 63{,}000}{\text{R.P.M.}}$

$$r_c = \frac{R+r}{R \times r}$$

$$V = \frac{\text{R.P.M.} \times \text{P.C.D.} \times \pi}{12 \times 60}$$

$$W = F \times \text{C.R.} \times r_c$$

$$\text{Face width} = \frac{T}{Wr} = \frac{T}{F \times r \times \text{C.R.} \times r_c}$$

To gain advantage from a high curvature factor it is evident that pinions should be as large as possible in diameter but this feature is limited by the permissible pitch line velocity and together these form the main factor controlling the power output from a turbine unit. When using 7/12-in. × 22½-deg. AA teeth, the adoption of forty teeth is preferred as a minimum in a pinion, which in conjunction with a maximum velocity of pitch line of 224 ft. per sec. implies a maximum turbine speed of about 6,000 r.p.m. Since it is desirable that the length/diameter ratio should not exceed 2.25 it is evident, from values of tooth loading, that the power transmitted should not exceed 4,500 s.h.p. (based on a primary gear ratio of 4:1).

If it is essential to increase the power of the turbine, the diameter of the pinion must be increased and the turbine speed reduced to maintain the pitch line velocity at 224 ft. per sec. The increased diameter permits a corresponding increase in the face width of the gear within the limit of the length/diameter ratio.

Secondary wheel dimensions are based upon the propeller shaft speed, together with a maximum pitch line velocity of 80 ft. per sec. Depending upon the particular speed required and the conditions attached to the cutting of the main wheel, this may be reduced in order to allot a greater proportion of the reduction to the primary gears.

A main wheel having a cast iron centre is considered preferable to one of the built type, which is difficult to retain accurate while cutting. There is also an advantage in that the additional mass of the cast type tends to ensure that the node of 1-node vibration does in fact appear in the propeller shaft and not in the l.p. quill shaft.

INSTITUTE ACTIVITIES

Minutes of Proceedings of the Ordinary Meeting Held at the Institute on Tuesday, 19th April 1955

An Ordinary Meeting was held at the Institute on 19th April 1955, at 5.30 p.m., when a paper entitled "Crankcase Explosions in Marine Diesel Engines", by J. H. Burgoyne, D.Sc., Ph.D., F.R.I.C., and Professor D. M. Newitt, M.C., D.Sc., F.R.S., was presented and discussed. Mr. J. P. Campbell (Chairman of Council) was in the Chair. There were 105 members and visitors present and nine speakers took part in the discussion.

A vote of thanks to the authors, proposed by the Chairman, was accorded by acclamation. The meeting ended at 7.40 p.m.

Section Reports

Merseyside and North Western

Mr. G. Kenworthy-Neale (Associate Member), Honorary Secretary of the Merseyside and North Western Section, wishes all correspondence to be sent to:—

54, Aintree Lane,
Aintree,
Liverpool, 10.

South Wales

Fifth Annual Golf Meeting

The fifth annual golf meeting of the South Wales Section was held at the Glamorgan Golf Club on 10th June 1955, an occasion marked by the presentation of a silver cup, to be known as "The David Skae Cup", to be held for one year by the member returning the lowest net score in the competition and of which a replica would be held by the winner.

There was an entry of forty members and guests, the weather was fine and the play was good. The number present was supplemented by non-golfing members and their guests, so that a total of fifty-four sat down to supper in the club house. Mr. Gordon Wickett (Deputy Chairman) took the Chair in the absence of the the Chairman, Mr. H. S. W. Jones.

Mr. F. F. Richardson introduced the Vice-President and donor of the Cup, Mr. David Skae, who stressed the point that, in view of the success of these meetings in the past, the award of a token was definitely warranted. Mr. Skae asked the Glamorgan Golf Club Captain, Colonel Douglas Duncan, to house the Cup on behalf of the Section, and enquired if it would be possible for the Section to erect their Honours Board in the club house, as the Glamorgan Golf Club was now the permanent home for the Section's annual golf meeting. Mr. Skae then presented the prizes to the various winners, as follows:—

David Skae Cup and Replica	B. Watkins (Net 71)
Second Prize (Member) ...	T. Grieve (Net 75)
Best round by Member with over-18 handicap ...	F. R. Watley (Net 83)
Putting Prize (Member) ...	D. Skae (29 putts)
First Prize (Guest)	C. C. Parker (Net 73) (Winner on second nine holes)
Second Prize (Guest) ...	E. Hallam (Net 73)
Putting Prize (Guest) ...	J. F. Barlee (28 putts)

Consolation prizes were won by O. Harrison (Member) and A. Williams (Guest).

Mr. T. O. Buchanan (Member) proposed a vote of thanks to the Glamorgan Golf Club and congratulated Colonel Duncan on the fine state of the course and excellent service of the indoor staff; he also endorsed Mr. Skae's request for an Honours Board.

Colonel Duncan, in responding, said the Glamorgan Golf Club would always be most pleased to see the Institute members and their friends. As Captain of the Club, he said he would be gratified to have the magnificent "David Skae Cup" housed with the Club silverware and the Institute's Honours Board erected in some suitable place.

A vote of thanks to the Chairman was proposed by Mr. A. Weeks, and the Chairman, in replying, concluded his remarks by thanking Colonel Duncan for his permission to erect the Section's Golf Honours Board in the club house.

Sydney

General Meetings

A general meeting of the Sydney Section was held at Science House, Gloucester Street, Sydney, on Tuesday, 31st May 1955, at 8.0 p.m. Eng. Capt G. I. D. Hutcheson, R.A.N.(ret.) (Local Vice-President), was in the Chair and 128 members and guests were present.

Dr. E. P. George gave a lecture, well illustrated by lantern slides and sketches, on nuclear energy and propulsion. The following members and guests took part in the discussion: Messrs. E. T. Buls, J. A. Carson, J. H. Cowell, J. H. Ekem, A. A. Hay-Mackenzie, K. Irvine, J. B. Jones, H. W. Lees, P. Meredith, H. P. Weymouth and Cdr.(E) P. Berry-Smith.

A further general meeting was held at Science House on Friday, 15th July 1955. Ninety-two members and guests attended and Eng. Capt Hutcheson was again in the Chair.

A film of the Snowy Mountains Scheme entitled "Harvesting the Snows" was shown first and a brief introductory address on the scheme was given by Mr. T. A. Lang, Associate Commissioner. Mr. A. N. G. Bray, Executive Engineer, then gave a talk on "Hydraulic Turbines" and was followed by Mr. G. Fekete, who spoke about "Guthega Penstocks", and made particular reference to the quality of steel and the method of fabrication and testing, including X-ray and gamma ray examination of welding. Finally, Mr. G. Fussell, Executive Engineer, lectured on transport, plant and workshop facilities.

A vote of thanks to the speakers was proposed by Mr. C. McLachlan and carried with acclamation.

Meeting for Students and Apprentices

A meeting for students and apprentices was held at Science House at 8 p.m. on Tuesday, 5th July 1955. There was a total attendance of fifty-seven, comprising thirteen members and forty-four students and apprentices.

Mr. H. W. Lees (Member) addressed the meeting on "The Training of Marine Engineers" and stressed the importance of young men developing their powers of observation. A good discussion followed and numerous questions were asked by students and answered by the various senior members present. A vote of thanks to Mr. Lees was proposed by Mr. D. N. Findlay and carried by acclamation.

Membership Elections

Members elected 25th July 1955

MEMBERS

Arthur Ernest Anderson
David Barclay
Duncan Campbell
Thomas Henry Dunsmoir
Edward William Endacott, Sen. Cd. Eng., R.N.
Allen Hay
Reginald Thomas Jones, Cdr., R.N.
Leonard Lionel Lawrie
Matthew Spratt Lisle
John MacGregor
James McCalmont Matthews
James Hay Mitchell
William Murphy
Alfred Vincent Simon
Francis Leslie Tewkesbury, Cdr., R.N.

ASSOCIATE MEMBERS

Patrick Donald Collins, B.Sc.(Eng.)
Arthur Colquhoun
George James Davis
Roland Dodds
Gerald Francom
John McGregor Galloway
David William Gillett
Donald Terry Goodall
Thomas Jaffrey Henderson, Lieut.-Cdr., R.N.
Peter John Irving
William McCubbray
Hubert Alan McDowell
Alex Carroll Mark
Alfred Matthew Martin
Thomas Ollerton
Benjamin Corbett Pester, Lieut.-Cdr., R.N.Z.N.
Bruce Clifford Porter
James Clifford Sanders
James Brown Stewart

ASSOCIATES

Charles Anthony Durden
George Henry Hughes
William Gallon McNaughton
Arvindkumar Dhirajlal Shah
James Sweeney
Robert Winter-Evans

GRADUATES

Ronald Charles Bailey
John Derek Berry
Promoto Kumar Biswas
William Derek Bookless
Douglas Brown
Trevor Albert Blake Butler
Reginald John Carter
Mohammad Mustafa Jaffari
Richard Douglas Vincent Kite, B.Sc.
Jal Noshirwan Kootar
Murray Dunbar McNicol
Samuel John Nelson
Edmund Nurse
Lawrence C. Poole
Ronald Poolton
Robert Charles Rodwell
Abdus Samad
Kenneth Ronald Shaw
Ahmed Soliman Aly
Donald Percy Springfield
Alan Thomas Tindall
Alan Tout
Alewyn Johannes Janse van Rensburg

STUDENT

Thomas Arthur Johnson

PROBATIONER STUDENT

William Alfred Duncan Stevens

TRANSFER FROM ASSOCIATE TO MEMBER

Thomas William Griffin

TRANSFER FROM ASSOCIATE TO ASSOCIATE MEMBER

Gerald Clive Burnan, Lieut., R.N.
Geoffrey Stewart Hall
James Owen Harrison
Harry Leah
Bhim Sain Makhija
Dimitri Nicholaidis
David Shirley Pike
Derek Sydney Richmond
Dharmdas Thakurdas Thadani
Douglas Murray Thompson
Richard Waters
Gordon Graham Watson, Lieut., R.N.
David Wilson
Herbert Dutton Woodhouse
Alan Wyatt, Lieut., R.N.

TRANSFER FROM STUDENT TO ASSOCIATE MEMBER

Keith Dennis Oborn Wake, Lieut., R.N.

TRANSFER FROM STUDENT TO GRADUATE

Bernard James Vaughan

TRANSFER FROM PROBATIONER STUDENT TO STUDENT

Donald Adrian Mieras
Edward James Perry

OBITUARY

Sir Richard William Allen, C.B.E., D.L. (Member 3146) was born at Cardiff in 1867 and educated at Christ College, Finchley, London. He received his technical education privately from Dr. Gisbert Kapp whose work, with others, helped to place dynamo design on a sound scientific basis. Subsequently, he was apprenticed to the firm which his father had founded, W. H. Allen and Company, at the original York Street Works, Lambeth, London, where he was associated in the production of high-speed steam engines, gas exhausters, electric generators and motors. Later he gained further experience with John Elder and Company, Glasgow (now the Fairfield Shipbuilding and Engineering Co., Ltd.) and also with the Naval Construction and Armaments Company, Barrow-in-Furness (now Vickers-Armstrongs, Ltd.). In addition, he obtained seagoing experience as an engineer in the s.s. *Etruria,* owned by the Cunard Line. Thereafter he was engaged in the family business, being managing director from 1894 until 1926, when, on his father's death, he became chairman.

Sir Richard travelled extensively, on one occasion as British delegate for the University of Cambridge, the University of London and the Institution of Mechanical Engineers at the World Engineering Congress held in Tokyo in 1929. He was honoured by the award of the C.B.E. in 1918 and received a knighthood in 1942. He died on 17th July 1955, aged eighty-eight.

He was Past President, Section G, of the British Association for the Advancement of Science, Past President of the Institution of Mechanical Engineers, Trustee and Vice-President of the Engineering and Allied Employers National Federation; he was a Member of the Institution of Civil Engineers, Honorary Life Member of the Japan Mechanical Society, Member of the Institution of Naval Architects, the Iron and Steel Institute, the Institute of Metals, and many other technical societies. He had been a Member of the Institute of Marine Engineers since 1916. He was Deputy Lieutenant for the County of Bedford, a Justice of the Peace for the County of Bedford, High Sheriff of Bedfordshire, a member of the Bedfordshire County Council for several years, a member of the Bedfordshire Standing Joint Committee and a member of the Court of Quarter Sessions.

Harry Halsall (Member 7747) was born in 1896 and died after a short illness on 3rd April 1955. His engineering apprenticeship was served with Howard and Bulloughs, Ltd., Accrington, between 1910-15, and for the next three years he was employed by Simpson Brothers of Hopton, near Burnley. In 1918 he joined the Hall Line and sailed as junior engineer until 1922, when he transferred to the Clan Line, serving them until 1929. In 1929 he joined the Madras Port Trust as assistant dredging master and second engineer, which led to his appointment in 1934 as chief engineer and dredging master with the Port Commissioners, Chittagong, Eastern Bengal, later undertaking the duties of mechanical superintendent as well. In January 1942 he was called up for war service in the Suez Canal area, for two years being seconded to the Royal Navy and Royal Engineers, and working under their direction on the dredger *Patunga* until, in 1944, the dredger was lent to the Nigerian Government and Mr. Halsall served in her at Lagos until 1946, still remaining, however, in the Indian Service. In March 1946 he returned with the *Patunga* to India, resuming his position in Chittagong with the Port Commissioners until July 1949, when he left India on leave prior to retirement in 1951 and joined his wife and family in Dublin. Mr. Halsall had been a Member of the Institute since 1934.

Charlie Matthews (Member 11501), died on 21st December 1954, aged fifty-three years. He received a training as an artificer apprentice in H.M.S. *Fisgard* from 1917-22 and then served as engine room artificer in various ships for the next five years. From 1927-30 he was E.R.A. instructor in charge of the Mechanical Training Establishment Drawing Office and lecturer in marine engineering to apprentices. From 1930-40 he was a warrant engineer at sea, except for three years when he was employed at the Admiralty Fuel Experimental Station at Haslar. For two three-year periods, from 1939 and 1944, he was chief engineer in charge of machinery and stores in H.M. minesweepers and destroyers *Fermoy, Orwell and Zealous,* and from 1941-44 he was on the repair staff of the shore base for maintenance of naval vessels; during these years he was promoted first commissioned engineer and then lieutenant(E). From 1947 until his retirement in 1949 he was in charge of a group of reserve destroyers. On leaving the Royal Navy Mr. Matthews was appointed combustion engineer at Portsmouth Power Station, a position he held until his death.

Arthur Tate (Member 12929) was born in 1903. He was apprenticed to the Parsons Marine Turbine Co., Ltd., Wallsend, from 1918-19 and to John Readhead and Sons, Ltd., South Shields, for the next five years. After spending nine years at sea, 1924-33, as junior to chief engineer, he was appointed engineer in charge of a gas plant for the Anglo-Iranian Oil Co. Ltd., at Abadan; in 1936 he left this post to become assistant to the chief engineer of the Demerara Bauxite Company in British Guiana. From 1938-39 he was superintendent engineer for all War Department vessels stationed at Singapore. Early in 1939 Mr. Tate joined the Singapore City Council and served in the Water Department as Mechanical Engineer of the Island Works until his death on 15th May 1955. He had been a Member of the Institute since 1950 and was an Associate Member of the Institutions of Mechanical Engineers and Water Engineers.

Frank Ernest Urry (Member 14381) died suddenly on 24th March 1955 after being in retirement for only four months, aged sixty-six years. He served an apprenticeship with Ernest Scot and Mountain, Gateshead on Tyne, Swan, Hunter and Wigham Richardson, Ltd., and Duncan and Company of Newcastle on Tyne. He was first employed by C. A. Parsons and Co., Ltd., in 1914 at the Heaton Works as a fitter and erector and in 1925 was transferred to the outside erection staff, remaining in that department until his retirement in November 1934. In the course of his duties he erected and commissioned turbo generators at Shanghai, Penang and Copenhagen, and for over twenty years was on the erection staff in India. During the latter period he was concerned with the erection and commissioning of turbo generators at the Southern, Mulajore and Cossipore generating stations of the Calcutta Electric Supply Corporation, and in many of Martin

Company's power stations in India. Mr. Urry was elected a Member of the Institute in 1953.

WILLIAM EWART GLADSTONE WALLACE (Member 6974) was born in 1899 and died on 20th July 1955 at the Royal Hospital, Sheffield. In the 1914-18 war he joined the Army at the age of sixteen but, due to being under age, was released after being wounded on active service in France. He was soon back in service, however, in the uniform of the Royal Flying Corps and until his demobilization in 1918 he was attached to the Kite Balloon Section. On his return to civilian life he joined the Merchant Navy and was for a time with the Orient Line; he obtained a First Class Board of Trade Certificate and spent several years "tramping" to various parts of the world. Leaving the sea, he went to Nigeria in the service of the Crown Agents for the Colonies at Takoradi Harbour, where he stayed for four years. During the last twenty-three years he had been engineer manager with Thomas Heiton and Co., Ltd., Dublin, and for a time was on the board of management.

Mr. Wallace had been a Member of the Institute since 1932.