PAPERS ON ENGINEERING SUBJECTS.

TRANSMISSION OF POWER FOR MARINE PROPELLING MACHINERY.*

Prior to the advent of the steam turbine as a prime mover for marine propelling machinery the method of transmission of the power to the propeller did not call for any special apparatus, the engine crank-shaft being invariably connected direct to the propeller shaft. The main reason for this lies in the fact that the reciprocating engine depends mainly for its action on the utilisation of the pressure energy of the steam direct into work and although the velocity of the moving parts is involved in the question of power developed, the velocity of flow of the steam is relatively unimportant except in so far as it must be considered in providing suitable steam passages. In any case the efficiency of the reciprocating engine is little affected by the speed of running within wide limits and a mean piston speed of about 1,200 ft. per minute represents the highest speed found desirable in such engines; this limiting speed has only been exceeded to a slight extent in very special cases.

When the steam turbine was adopted, however, a new situation was created, as the working of this type of engine depends on a velocity effect of the steam, the heat and pressure energy of the steam being converted into velocity energy and the steam then impinging on and being deflected by suitable curved vanes attached to the moving parts. Owing to the very high velocities attained by the steam for even comparatively small drops in pressure, velocities of, and exceeding 1,000 f.s. being common practice, which are far above the speed of the moving parts practicable in reciprocating engines and also owing to the fact that to secure efficiency in any design of steam turbine it is necessary to run it at a speed at which the blade speed is a particular ratio of the steam speed, depending on the type of turbine, the features governing the design of such engines are radically different from those governing the design of reciprocating engines. In land installations where weight and space are comparatively unimportant factors, and where turbines are generally used to drive dynamos in which high running speeds can be accepted, it was usually found possible to secure the full advantage of the steam turbine, but in a marine installation where weight and space are of paramount importance, more especially in high speed warships and where the best efficiencies from the propeller point of view are obtained with

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* Substance of a lecture delivered at Chatham on 15th December 1920.

comparatively low revolutions, it was not possible to do so. The desired ratio between blade speed and steam speed for a given number of revolutions can be obtained by increasing the diameter of the rotor or by dividing the pressure drop up into a large number of stages; either of these courses increases the weight and space required, and, without going more fully into the question, it is sufficient to at once state that all such designs depending on a direct drive were of the nature of a compromise in which efficiency was sacrificed, both in the turbines, by running at too low a blade speed, and in the propeller, by running at revolutions too high, to obtain the best overall results. Even so, the efficiency of the steam turbine, apart from the reliability, ease of running, and the large increase of power of individual units together with reduced weight per unit of power that were only attainable with this type of prime mover, was sufficient to enable it to replace the reciprocating engine in vessels exceeding about 20 knots speed. The provision of cruising turbines to secure efficiency at reduced speeds at the expense of additional weight and complication allowed the turbine also to compete satisfactorily with the reciprocating engine under the range of speeds usual in Naval vessels. For the period ranging from about 1906 (H.M.S. " Dreadnought ") to some time in the war period, the direct drive type of turbine was the standard for warships, but the disadvantage of the system was by no means lost sight of, and developments of an intermediate form of transmission were constantly proposed and experimented with. It is with this aspect of the steam turbine drive that is is proposed to deal.

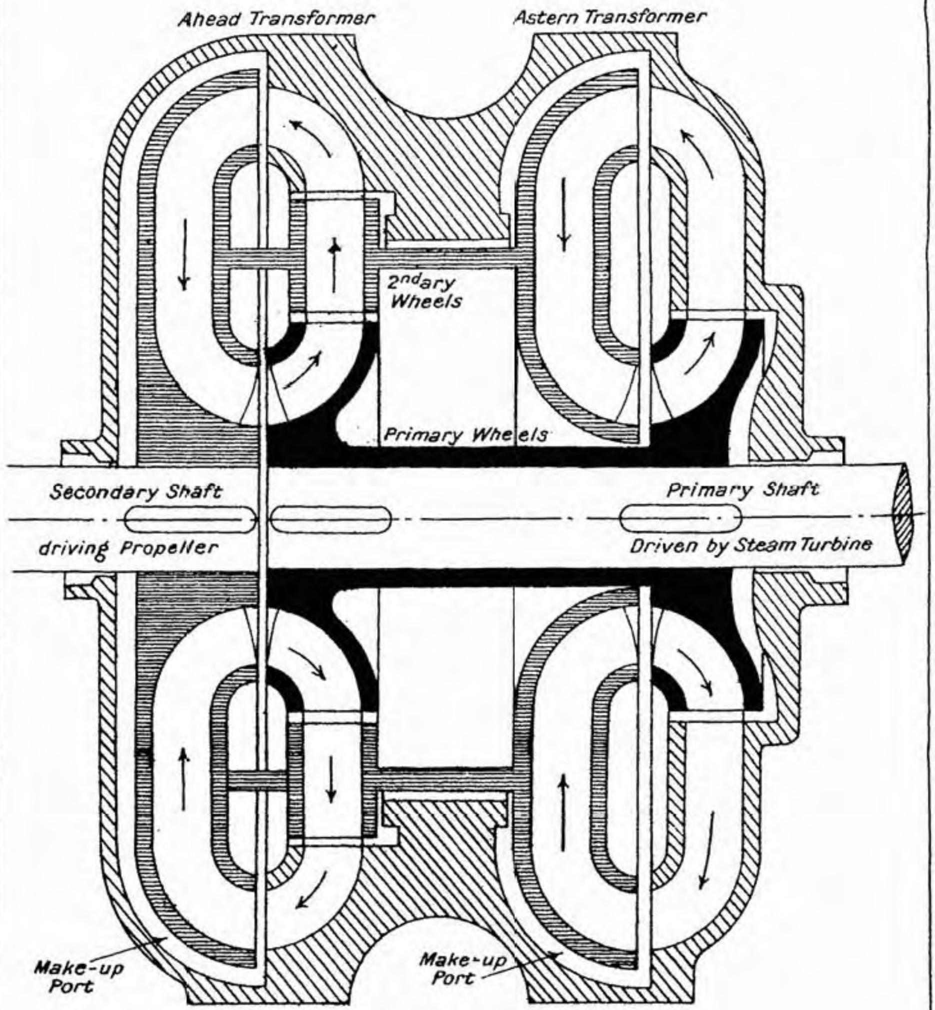
It is clear that to get the best overall results, the turbines and the propellers should be designed independently of one another for the particular case aiming for high efficiency in each part and arranging for an efficient form of transmission of the power from the turbine spindles to the propeller shafts. From what has been said previously, this form of transmission must be in the nature of a reduction device and three practicable methods are so far available for effecting the " gear-down," viz., the hydraulic, mechanical, and electrical, each possessing its peculiar merits and demerits. Up to the present time the mechanical method only has been tried in H.M. Navy for high-powered installations, but some U.S. battleships and battle cruisers have been, and are being, fitted with the electrical system, whilst the hydraulic method found favour in some vessels of the German Navy. The mechanical method has been generally fitted in recent years on a large scale, both in warship and mercantile vessels of several countries.

The main features of these three systems are described below, and from these descriptions it will be simple to follow the advantages, disadvantages, and limitations attendant on each when applied for marine or more particularly for naval purposes.

FÖTTINGER HYDRAULIC TRANSFORMER.

TRANSMISSION OF POWER.

FIGURE 1.



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Hydraulic Transmission.

The hydraulic method was devised and patented by Dr. Fottinger in Germany, and is generally known by his name as the Fottinger system. The principle of action is as follows, and is illustrated in Fig. 1.

A high-speed turbine running continually in one direction has mounted on its shaft a rotary pump, called the Primary, which delivers water with high velocity through suitable guide channels on to the vanes of a water turbine wheel, known as the Secondary, mounted upon the propeller shaft and running at a reduced velocity. The whole of this apparatus is designated the Transformer. For going astern a second transformer is fitted, the vanes of pump, &c., being so designed that the propeller shaft runs in the opposite direction to the pump and steam turbine. The particular transformer in use, ahead or astern, as the case may be, is maintained full of water, the other meanwhile running empty, arrangements being fitted for quickly emptying, charging, and maintaining charged against leakage. The two transformers for ahead and astern working are so designed as to be compactly mounted in one casing, and to obtain good efficiency in the ahead direction the secondary in the ahead is in two stages, suitable guide vanes being interposed between the two stages to correctly change the direction of flow of the water. High efficiency is not so essential when running astern, and here the single stage meets requirements and in any case gives a higher power astern than would be obtained with the usual proportions of the astern portion of the turbine installation. The water runs in a closed circuit, the wheels being so arranged that the water as it leaves the secondary enters the primary direct without any guiding arrangements. This simplifies the arrangement and materially increases the efficiency as there is no loss of energy carried away by exhaust water. An efficiency of transmission as high as 90 per cent. is claimed for this type of reduction device.

The primary and secondary wheels, likewise the guide rings, are of bronze to better stand the effects of corrosion, which would in time roughen the surfaces and lead to additional eddying and frictional losses. In T.B.D. installations the casing is also of bronze, but it is understood that in some very large-powered installations that were projected the casings would have been of cast-iron. Labyrinth packing is fitted where the shafts leave the transformer casing and between the ahead and astern portions, with leak-offs led to the tank carrying the make-up and reserve water for the transformer circuits. This tank is built below and attached to the transformer. Fresh water only is used.

To quickly charge and maintain the systems charged when running a special high-speed turbine-driven pump is fitted with a suction to the tank and discharge to a casing containing the slide valve, which distributes the water to the transformers. The necessary manœuvring is performed by operating the slide-valve, which opens communication between the delivery from the pump and the transformer required to be used, at the same time opening the drain from the other transformer to the tank.

Regulation of speed is effected by altering the speed of the steam turbine in the ordinary way, and the speed of the propeller shaft changes in approximately the same proportion as the turbine.

In the T.B.D.'s of 23,000 S.H.P. (total) arranged on two shafts the maximum revolutions of the steam turbine were 2,500, and of the propeller shaft 520 at full speed, giving a reduction ratio of about 5 to 1.

The main thrust block is fitted abaft the transformers, and it is arranged in the design for the propeller thrust to be approximately balanced by the end thrust on the secondary wheel, whilst the end thrust on the primary wheel serves to approximately balance the steam thrust on the turbine rotor, the difference being taken on a small adjusting block.

The water used in the transformer circuits rises considerably in temperature due to the frictional and eddying actions, and it is arranged for this water to be continously drawn off for boiler feed, the requisite make-up in the tank being supplied from the cold feed circuit. The loss of energy is thereby to some extent recovered. With this installation various safety devices and precautions are necessary to ensure against damage to the steam turbines and transformers during manœuvring. Before the manœuvring slide can be reversed, *i.e.*, a system emptied, it is first necessary to move a lever, which ordinarily locks the manœuvring lever, to a position which by a special device closes down the turbine regulating valve and therefore eases up the turbine. This prevents the turbine running under steam with the transformers empty. When the speed has fallen to a predetermined figure the manœuvring slide is moved across and the interlocking lever, which is itself locked when the manœuvring lever is in mid position, so that steam cannot be opened when transformers are empty, is again moved to open up the main regulating valve and allow the turbine to accelerate and deliver the power required.

An emergency self-closing valve is fitted in the main steam supply to the turbine which is actuated in the following eventualities :—

(1) When a particular pressure in the first stage of the turbine is exceeded (this would mean a steam overload).

(2) When the speed of the turbine exceeds a certain figure (may take place with a shortage of water in a transformer due to the make-up failing).

(3) When the transformer revolutions exceed a certain figure (may take place if the propeller is damaged).

A cruising valve and a special steam valve for running the turbines with circuits empty (as in warming up and trying engines) are also fitted.

The make-up pumps for each set of engines are cross-connected to enable the main engines on both sides to be run in case of a failure of one pump.

From some data that has become available from one of the German T.B.D.'s fitted with this installation during the war, it transpires that considerable trouble was experienced owing to failures of vanes in the transformer wheels necessitating renewals of the same. No advantage in economy was shown over similar powered direct drive installation vessels and it is doubtful if the high degree of efficiency claimed was being obtained or that the gain in efficiency of the turbine was sufficient to overcome the loss entailed in the transmission.

Mechanical Transmission.

With this arrangement the high-speed turbines actuate toothed pinions which mesh with a wheel on the propeller shaft in the case of single reduction, or with wheels which carry further pinions which actuate with the wheel on the propeller shaft in the case of double reduction. The ratio of reduction of revolutions is a fixed figure, being governed by the relative numbers of teeth in the pinion and wheel elements. In all high-speed vessels in H.M. Navy so fitted the reduction is carried out in one stage, but in the slower speed vessels of the Mercantile Marine, where the powers and therefore turbines, are smaller, necessitating higher revolutions to obtain the required peripheral speeds, and where the shaft revolutions are lower, the increased ratio of reduction required is carried out more generally in two stages. The general arrangement as regards the component parts of an "all-geared" set adopted in the first T.B.D.'s so fitted, viz., "Leonidas" and "Lucifer" in 1914, has been adhered to in naval practice to the present day; in this design each set of gearing is driven by two pinions actuated by the high-pressure and low-pressure turbine respectively. In some installations an additional turbine for cruising has been fitted on an extension of the high-pressure turbine spindle. In the Mercantile Service double reduction practice an intermediate pressure turbine is sometimes fitted, the pinion of which actuates the same wheel as the high pressure pinion. In naval practice the astern turbine is incorporated with the low pressure turbine, the high-pressure turbine running idle when the engines are going astern. In the Mercantile practice it is usual to incorporate a high-pressure astern turbine in the H.P. casing (or I.P. casing if so fitted) exhausting to the low-pressure astern turbine incorporated in the L.P. casing.

The maximum ratio of reduction possible with a single reduction gear is about 20 to 1, being governed on the one hand by the minimum size of pinion capable of carrying the load and of being fitted with suitable teeth and on the other hand, by the maximum size of wheel which can be handled in manufacture and accommodated on board. A 4-in. pinion and a 12-ft. wheel are the limiting figures for general cases. This would give a ratio of revolution of 36 to 1, but in a ship with machinery of such a power that could be transmitted through a 4-in. pinion, a gear wheel of 12-ft. diameter, could not be conveniently accommodated with a practicable arrangement of machinery. For ships of speeds above 16 knots, the single reduction gearing is very satisfactory, but when it is desired to gain the full advantage of geared turbines for average slow-speed mercantile craft, double reduction gearing must be resorted to as permitting of high ratios of reduction in a practicable manner. Reductions of 30 to 40 to 1 can be arranged quite conveniently in this manner.

There are two standard arrangements of such gearing generally termed the "three box" and the "interleaved" system respectively, shown in Figs. 2 and 3. Each has its advantages. With the former which is simply three single reduction gears, the H.P. and L.P. in parallel and both in series with the second reduction, the span between the bearings is much less than in the interleaved design, and the possibility of deflection under load prejudicing the efficient working is therefore less. The three-box arrangement, however, occupies rather more space in the engine room, but the smaller casings required with it are more easily handled than the larger casing of the interleaved variety. Both systems have been widely fitted, and the conditions in individual works or the requirements of particular ships appear to have influenced the choice rather than any other factor.

It is essential that the form of tooth used in high speed gearing subject to heavy loads should be such that uniform angular velocity is transmitted. Several forms of tooth satisfy this condition, but of these, the involute possesses several important essential advantages over other forms, the two chief of which are :—

(1) The teeth can be automatically generated in a machine by a tool with straight-cutting edges, which allows it to be easily made and accurately ground.

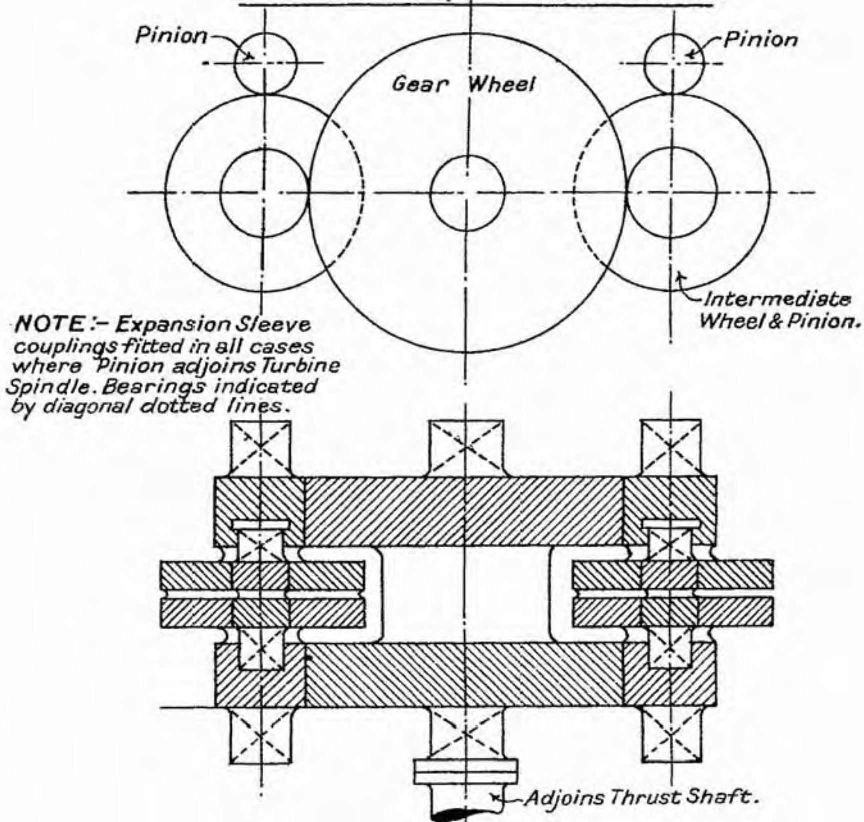
(2) A variation in distance within limits between the wheel centres does not affect the uniform velocity ratio desired.

These two important advantages are such as to render involute teeth indispensable for gearing which has to transmit high powers, seeing that accuracy of tooth form is vital and the necessary clearances and wear in bearings apart from small errors in alignment preclude the distances between wheel and pinion centres being always exactly that designed for.

In straight cut or spur gears at any particular moment the number of teeth in contact, which may be one and, in any case,

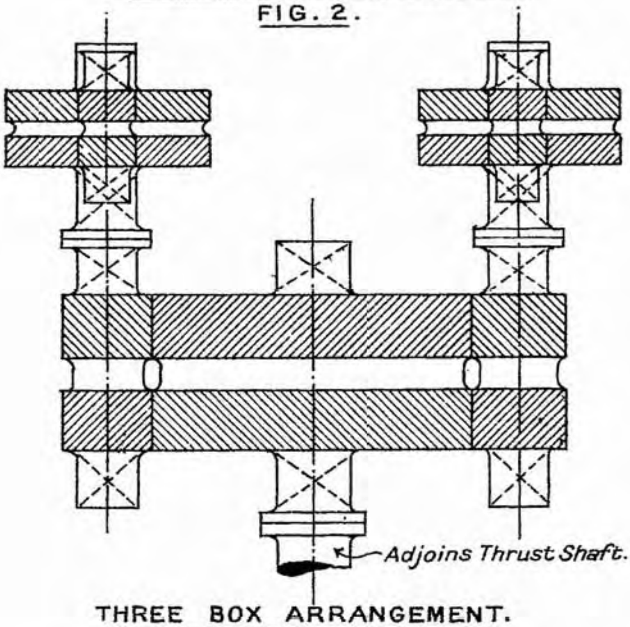
DOUBLE REDUCTION GEARS.

TRANSMISSION OF POWER.



INTERLEAVED ARRANGEMENT.

FIG. 2.



THREE BOX ARRANGEMENT.

FIG. 3.

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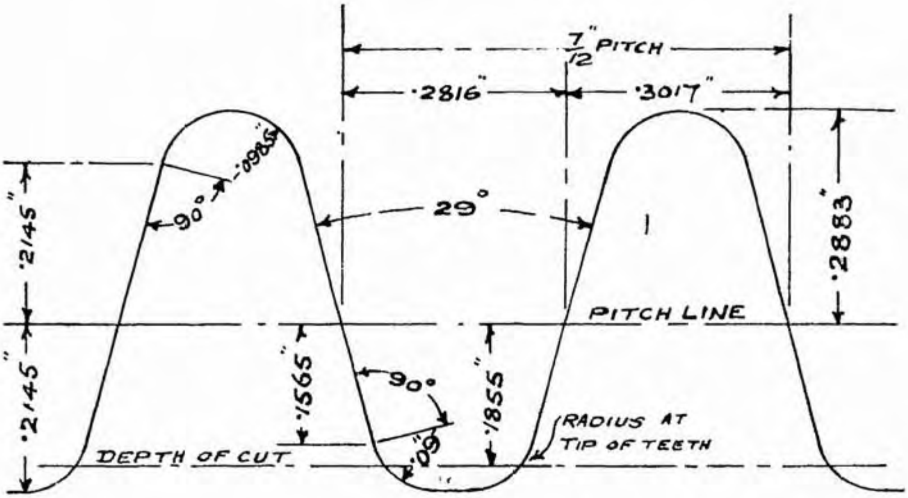
not more than two, imposes a limit to the speed at which high powers can be safely or smoothly transmitted, and therefore for high-power helical cut gears, *i.e.*, gears in which the teeth are inclined to the axes of the wheels, are necessary. In helical teeth gearing the load is shared at any moment by a much greater number of teeth, this number depending on the helical angle other things being the same, and the load on any one length of tooth moreover comes on gradually, increases to a maximum and then decreases as gradually as it comes on. As a result of the inclination of the teeth in helical gears the total normal pressure on the teeth is, for a given torque, greater than the total normal pressure on the teeth in spur gearing. Notwithstanding this, the tooth stress is less owing to the much greater average length taking the load. Smaller teeth can therefore be fitted, and this assists smooth working. All the gearing now in use is of the double helical type, a right and left hand helix being used to balance the axial thrust entailed. The helical angle for some years was 45°, but in the last few years it has been reduced to 30°, and this is now a general standard angle for Admiralty practice. Angles from $22\frac{1}{2}°$ to 45° have been used in commercial practice. The angle is rarely an exact number of degrees as it depends on the available driving and change-wheel mechanism of the cutting machine. Fine pitched gears give more silent running at high speeds and a normal pitch of 7/12 in. has been adopted for the majority of high-power installations, but a pitch of $\frac{3}{4}$-in. was used in the "Courageous" class, and 1-in. in the "Hood." Pitches exceeding 2-ins. have been used in the final reduction wheels of some two-stage mercantile gear, but these large pitches have been abandoned in recent practice.

The proportions of the teeth in relation to the pitch should from various considerations fall within certain narrow limits. The usual ratio in well-designed gears provides for an addendum, *i.e.*, the portion of the tooth outside the pitch line equal to ·3 of the pitch, but this ratio may be reduced in special circumstances. The other tooth proportions are decided by the necessary working clearances required. It is essential that a very pronounced radius should be provided at the tooth root to avoid excessive local stresses, and it is the modern practice to slightly chamfer off the points of the teeth to prevent digging-in and excessive wear at these positions.

The sketch shows a typical hob tooth for a modern 7/12-in. pitch gear; the tooth is a reciprocal of the hob as regards depth, but the tooth face is curved, the curvature depending on the radius of wheel or pinion as the case may be. The angle $14\frac{1}{2}°$, which is the inclination of the hob face to a radial line, is generally adopted for all gears both ashore and afloat, and is known as the tooth angle or the angle of obliquity. A line normal to this face gives the line of action of the resultant force between the teeth, neglecting the frictional effect.



The more important particulars of the gearing fitted in typical warships are shown in Table I., by study of which the various features so far discussed and to be discussed can be better understood. More especially by comparing the two T.B.D.'s "Romola" and "Seafire," although the axial lengths of pinions, &c., are the same, and the number of teeth in contact more in the case of the 45° job, the total length in contact is greater in the 30° case, the normal load is less and consequently the load on the teeth per inch is less for both reasons.

The adequate and efficient lubrication of the teeth is very important, not only to provide a film between an otherwise metallic contact of the surfaces but also to abstract the heat consequent upon the sliding friction between the teeth. The loss due to friction in a good gear including that due to the bearings, is from 1½ to 2 per cent. of the total power transmitted, and this loss of power which appears in the form of heat must be carried away by the oil, excepting the small amount passed on by radiation. It is necessary that the temperature of the working surfaces of the gear should be kept as low as possible both on grounds of efficient working and to maintain the viscosity of the oil, and therefore the amount of oil supplied must be liberal and such that the heat may be carried away with only a small rise in temperature of the oil. The oil is sprayed directly on to the line of contact of the teeth by a number of sprayers pitched a few inches apart, the sprayers being designed to give flat fan-shaped jets so that the whole region of contact is served. In the earlier designs of reduction gear nozzles were fitted to spray oil to the line of contact both on the entering side when running head and also when running astern, but the latter groups are not now fitted as experience shows they are not essential considering the short periods of astern working usual in Naval vessels.

To avoid the same teeth of gear-wheel and pinion coming in contact revolution after revolution and also to ensure uniform wear it is usual to fix the gearing ratio so that it is not only a whole number, but the number of teeth in the wheel and pinion have no factor in common. This is very often most simply done by making the number of teeth in the gear-wheels or pinions prime numbers.

Experience indicates that a high degree of accuracy of tooth-cutting and the maintenance of accurate alignment are necessary to the successful operation of any mechanical reduction gear. All large gears at the present time are cut by what is known as the hobbing process. The tool called the "hob" is of cylindrical form somewhat resembling a tap. The blank to be cut is mounted on the table or face plate of the machine, and is given a motion of rotation bearing the desired ratio of revolution to that of the hob; the combination of this motion with the slow feed of the latter in the direction of the axis of the blank causes an automatic generation and cutting of the true tooth form. Great accuracy is essential in the hob, in the gear-cutting machine details and in the setting of the hob and the blank, and it is arranged for the machine to run continuously during a cutting operation till the whole surface is complete to eliminate as far as possible any source of error due to lost motion in the machine or temperature variations. A difficulty in the early days of gear-cutting for large powers was caused by the fact that the master worm-wheel which rotated the machine table or face plate was not quite accurate; this difficulty led to an addition to the machines of a "creep" mechanism whereby the blank is rotated at a slightly higher speed than the table to which the driving worm-wheel is attached. Any recurring error in the worm-wheel which might be copied on the job being cut is with such a machine no longer in the direction of the axis, but is distributed in helices on the gear-wheel or pinion. With improvements in the accuracy of the manufacture of these worm wheels and the maintenance of the machine in good condition, with all parts well lubricated during a cutting operation, the advantage of the creep has been lessened, and in most cases it is not now fitted.

As regards alignment, it is clearly essential that the driving and driven shafts shall be parallel with each other and their axes in the same plane to ensure uniform contact across the whole width of the gear. In all Admiralty installations the turbine spindles, pinions and gear-wheels are carried on rigid bearings, and the correct alignment is determined by accurate machine work in boring the gear housings and fitting the bearings. In the first all-geared installations the turbines and gear cases were each carried on separate foundations built up from the hull structure. The turbines and gear cases as separate units are of stiff construction and under working conditions do not suffer distortion. The possibility of relative motion between the

TABLE

Particulars of Gearing

—		Battle Cruiser. "Hood."	Light Cruisers. "Courageous."	Light Cruisers. "Danæ."
Horse power	H.P.	17,500	8,500	8,600
	L.P.	18,500	14,000	11,400
Revolutions per minute.	H.P.	1,497	2,660	2,568
	L.P.	1,098	1,420	1,403
	Propeller	210	340	275
Pitch circle diameters, inches.	H.P. Pinion	20·174	10·721	11·755
	L.P. ,,	27·51	20·14	21·514
	Gear-wheel	143·787	83·998	109·789
Addendum of teeth - inches		·266	·231	·1855
Pitch - inches	Normal	·99847	·744	·583
	Circular	1·153319	1·052	·69679
Helical angle		29° 57′	45°	33° 12·4′
Tooth angle	Normal	14° 28·7′	18° 15′	14° 29·5′
	Circular	16° 36′	25°	17° 12′
Number of teeth	H.P. Pinion	55	32	53
	L.P. Pinion	75	60	97
	Gear-wheel	392	251	495
Total width of pinion face - inches		75	54	53
Effective width of pinion face inches		73·25	54	51·25
Number of teeth engaging		36·60	57·3	48·3
Total length of contact - inches	H.P.	128·8	94·5	103·3
	L.P.	132·9	97·0	105·5
Tangential load lbs.	H.P.	73,000	37,500	35,850
	L.P.	77,200	61,800	47,500
Normal load - lbs.	H.P.	87,000	55,900	44,400
	L.P.	92,600	92,000	58,700
Normal load per inch of contact - lbs.	H.P.	675	592	430
	L.P.	697	948	556
Speed at pitch line in ft. per second		132	125	132

I.

of Typical Vessels.

	T.B.D.'s.			P " Boat.	" K " Submarine.
" Dauntless."	" Seafire."	" Romola."	" Leonidas."		
10,000 10,000	6,250 7,250	6,250 7,250	5,750 6,100	875 875	2,250 2,750
1,800 1,800 275	2,985 2,262 350	2,970 2,270 350	3,340 1,815 380	3,500 3,500 300	3,300 3,000 380
15·319 15·319 99·897	10·0694 13·2830 85·9114	10·1283 13·2447 85·961	9·603 17·648 84·348	4·6 4·6 53·862	6·7526 7·5316 58·9552
·1855	·12	·12	·1855	·168	·12
·583 ·67783	·583 ·6730	·583 ·815875	·583 ·8153	·5196 ·600	·583 ·8158
30° 40·3′	29° 58·9′	44° 23·6′	44° 22′	30°	44° 23·6′
14° 29·5′ 16° 44′	14° 29·5′ 16° 37·5′	14° 29·5′ 19° 57′	14° 29·7′ 19° 56′	17° 30′ 20°	14° 29·5′ 19° 57′
71 71 463	47 62 401	39 51 331	37 68 325	24 24 281	26 29 227
60	40	40	29	21	28
58·25	38·25	39·5	27·6	20·5	26·9
51·7	33·0	46·7	33·1	19·71	32·2
123·0 123·0	52·2 52·2	45·3 45·3	48·4 49·9	35·3 35·3	31·0 31·0
45,700 45,700	26,200 30,500	26,200 30,500	20,300 24,040	6,840 6,840	12,700 15,400
55,000 55,000	31,300 36,300	38,000 44,000	29,350 34,800	8.300 8,300	18,400 22,300
449 449	600 695	838 970	606 697	234 234	594 720
120·4	131	131	140	70	97·6

turbines and gear case, however, existed especially in the lightly constructed vessels of the T.B.D. type, and therefore possibilities of misalignment were present. In later designs this relative movement is not possible as the after ends of the turbines are now supported on stiff extensions of the gear case, the whole installation comprising gear case and turbine forming a rigid unit.

The method of ensuring alignment of gearing when fitting up is deserving of brief description. It has been stated that if the distances separating the centres of wheels carrying involute teeth be changed within certain limits the smooth and efficient action of the teeth will not be affected. It is necessary, however, for running reasons to preserve certain clearances between the tips of the teeth and the recess in which they engage; in aligning it is essential when setting this clearance to keep in mind the positions that the pinions and wheel will take up when transmitting power. Due to the obliquity of the teeth the force on the pinion will not be tangential to the pitch line circumference, and it will have components acting radially and tangentially. In addition there is the effect of the weight of the pinion to be considered, which is, however, small in relation to the other forces at the maximum powers. On the gear-wheel itself there will be the forces transmitted from the pinions and the weight of the wheel. Without considering any particular case the net effects will work out somewhat as shown in Fig. 4, and the directions of the resultant forces indicate the direction to which the parts will move when running steadily. Apart from the alignment question this figure also indicates the positions at which the oil should be supplied to the bearings observing it is necessary that the oil should be introduced at a region of low pressure. If two supplies be arranged for each bearing at diametrically opposite positions it is a simple question to guarantee an ample supply under all conditions, and this is always arranged for.

The respective resultant forces will shift the positions of the pinions and gear wheel somewhat when running. On the H.P. pinion the direction of the resultant force will change from OA to OB as the power increases from zero to the full power ahead. On the L.P. pinion it will change as shown from O′C to O′D. On the wheel it will change from O″E to O″F.

When running astern the force on the H.P. pinion will be small, and it will remain in about the same direction as when standing. On the L.P. pinion it will change in direction from O′C to O′G at full power astern whilst on the wheel it will change from O″E to O″H.

For the purposes of alignment the pinion is assumed to lie in a convenient reference position, and in spite of the foregoing reasoning, the pinions are generally assumed to lie out radially from the wheel and to be in contact with the bearings either at the points K and L or at the points through which the resultant

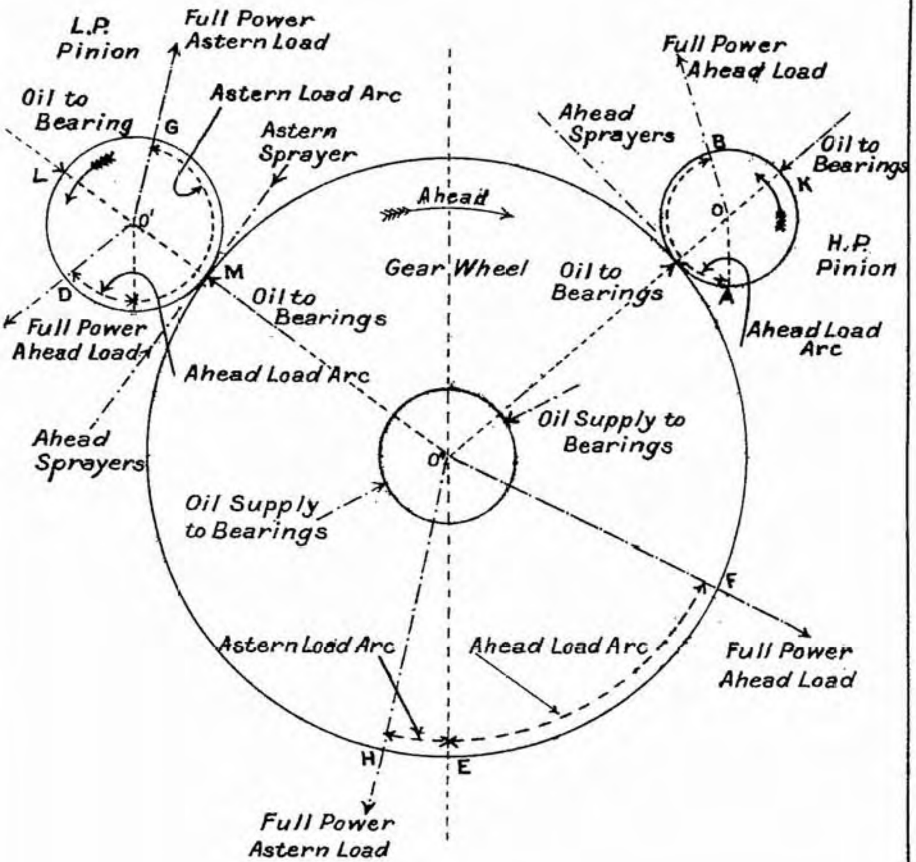


DIAGRAM SHOWING FORCES ON GEARING AND POSITIONS OF OIL SUPPLIES.

TRANSMISSION OF POWER.

FIGURE 4.

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force acts when developing the full power ahead. The gear wheel is taken as lying at E. This assumption does not affect the accuracy of alignment if borne in mind and catered for as seen later.

During erection the gear wheel and pinion bearing housings are first bored out to the designed lines of centres and, in order to ensure accuracy, these bores should be tried with special mandrels made for the particular job. The bearing bushes are now machined externally and bedded in the housings and afterwards bored concentrically to the exact spindle diameter internally thus still maintaining the designed gearing centres. The gear case is now brought up to the turbines and adjusted so that the centre lines of the turbine rotors are truly in line with the lines of centres through the pinion bearings and is then secured.

The pinion and wheel bearings are finally removed and bored to their working diameters, being set eccentrically, and such that no metal is removed from the reference points previously selected. The clearance required for high-speed bearings is a matter of experience, and although some variation is noted with different makers, the following figures give representative practice :—

Journals under 6-ins. diameter	-	20 thousandths.
Journals 6-ins. to 12-ins. diameter	-	25 ,,
Journals 18-ins. to 24-ins. diameter		35 ,,
Journals 24-ins. to 30-ins. diameter		40 ,,

The completed bearings will now have eccentric bores and to act as a guide for remetalling if required at some later time, lines are scribed or a small shoulder turned on the ends of the brasses concentric with the completed bores.

The pinions will with this method of construction be free to move by the amount depending upon the bearing clearances in directions tending to put them into deeper mesh with the wheel and to maintain the requisite tooth clearances radially it is usually necessary to set the hobs so that it cuts a few thousandths of an inch deeper than is indicated on the design. This will lessen the pitch circle diameters slightly and also the thickness of the teeth to a very small extent. The amount of "backlash" or clearance between the teeth is usually 30 to 40 thousandths for a 7/12-in. pitch gear; experience shows that a clearance less than about 30 thousandths is likely to lead to running difficulties.

An advantage of choosing the points K and L for reference as distinct from B and D, apart from their convenience in the machine shop, is that the shock on the bearings with a sudden change of speed or reversal is less.

For any particular intensity of pressure the surface stress on the material of a pinion tooth is greater than that on the gear wheel owing to the greater curvature of surface. The teeth of gearing are said to engage by line contact, but it must be appreciated that contact under a load can no more occur on a line than the load on a ball bearing can occur on a number of geometrical points. Elastic deformation of the surface must take place, and the momentary flattening leads to a surface contact. The width of the flat is necessarily the same on pinion and wheel teeth and must consequently entail a greater stress on the pinion teeth which have the greater curvature. Further, the pinion teeth come into engagement more frequently than the wheel teeth and to equalise this wear and counteract the greater surface stress therefore, the pinions are made of nickel steel, the proportion of nickel being about $3\frac{1}{2}$ per cent. with ·3 to ·35 per cent. of carbon. The tensile strength must be not less than 40 tons per square inch with an elongation of not less than 23 per cent.

These nickel steel forgings require very careful heat treatment to ensure that the material throughout shall be uniform. For this reason all pinions except those of small diameter are made hollow to assist this. Oil tempering is also carried out to obtain the full advantages of the alloy. The pinion blank is made so as to be reduced at least 2-ins. in diameter and 2-ins. on each side of each pinion blank by machining from the finished forging. In some large installations the teeth are formed on separately forged rims shrunk on and pegged to the pinion shaft, which in this case is made of the ordinary forging quality of carbon steel.

Each pinion shaft is supported on three bearings in naval designs to minimise transverse deflections which would affect the contact between the teeth, but in the low-power pinions of mercantile designs which are comparatively short, this is not usual. An expansion coupling is fitted where the pinion joins the turbine spindle so that it may be free to float axially and pick up its best position relatively to the gear wheel, suitable clearances being allowed for between the ends of the pinions and cheeks of adjacent bearings.

The gear wheel is made up of a continuous shaft, of the rim, and of a centre uniting the shaft to the rim. The centre may be made of a solid steel casting or of two or more forged steel wheels or be constructed of plates.

Cast and forged steel wheels are shrunk on and pinned to the shaft and the rim is in turn shrunk on and pinned to them. Plate centres are bolted by fitted bolts to flanges made solid with the shaft and with the wheel rim respectively. The larger designs are arranged with a separate casting keyed to the shaft which carried the flanges for the centre plates. The rim, sometimes called the shroud, is made of forged carbon steel of tensile strength 31 to 35 tons per square inch. The two portions of the rim on which the teeth are cut are made in one piece for those designs in which the rims are shrunk on to wheel centres comprised of steel forgings or castings. In the built-up designs of wheel centres comprising plates, each rim section and the intermediate portion of the rim joining them is made separately and bolted together. This is necessary for constructional reasons.

The gear case is mainly of cast-iron and beside supporting the bearings for pinion and gear wheel also forms the enclosed oil tight chamber required for the gearing and its bearings. The framework for the bearings and for the attachment to the vessel is usually made in two pieces, bolted together by a horizontal joint. For very large designs the lower portion has been made in two pieces united by a vertical joint. In the usual machinery arrangement with underhung condensers where the pinions are both situated above the centre line of the gear wheel, the lower portion of the framework carries the feet for attaching the gear case to the vessel and housing for the gear wheel bearings, while the upper portion carries the housings for the pinion bearings. The lower portion also carries extensions at the forward end for supporting the after feet of the turbines so that the whole system of gear case and turbines forms one solid unit fixed at the gear case, the turbines being fitted with sliding feet at the forward ends, the L.P. carrying the condenser bodily with it as it expands.

Ventilating pipes to allow the escape of vapour are fitted to the highest part of the case, these pipes being fitted with turned-over ends or baffles to prevent the entrance of dirt, &c.

During the early running of gears on service there is often observed on the faces of the teeth an action known as pitting. Small flakes of metal are detached from the otherwise smooth surface of the teeth, leaving small hollow depressions, varying in size from a small speck to as large as $\frac{1}{8}$ in. diameter, and 15 to 30 thousandths of an inch deep. The cause of this pitting has not been entirely satisfactorily accounted for. A lack of homogeneity in the material at the tooth surface or crushing of minute high spots causing failure of small sections of the metal at these high spots are possible reasons which have been advanced. It is noticed that the "pits" are most common in the region of the pitch line where only rolling action occurs, and the effect of mutual pressure in this position may be more intense due to the possibility of metallic contact, since relative sliding velocity between two surfaces is essential for the formation and maintenance of an oil film. From considerations of the flexibility of the teeth, it will be apparent that the greatest intensity of pressure on teeth in contact can be sustained at the pitch point. If, therefore, any "hammering" takes place due to fluctuations of the load at the propeller end, this effect will be felt more intensely at the pitch line. Although it has been demonstrated in more than one case in practice that gears will stand heavy shocks at the propeller end without ill effect, it appears that under rapidly repeated blows, even of small magnitude, the metal of the teeth may shell off and an influence of this nature probably appears the most convincing explanation of this feature.

In any case the action has been found to not seriously prejudice the smooth and efficient working of the gear, and it usually ceases to extend after a short period of running. The pitting will naturally be more intense in the gear in which the pressure is not uniformly distributed along the length of the teeth as in such gears the intensity of pressure will be higher.

Another not uncommon feature in the gears during their early life is a small amount of what is usually described for want of a better term as "wear" as distinct from abrasion, especially on the pinions, the teeth at the regions of higher pressure being rubbed away so that a sharp razor-like edge or small "rag" is formed on the points of the teeth. Slight overheating of the pinion would account for this action, or it may be explained by "flow" of the metal due to excessive pressures in the presence of perfect lubrication; a similar phenomenon has been noted in worm gear subject to high loads. As a general rule, however, it can be stated that once the regions of higher pressure have been levelled and the load is more uniformly distributed, the action does not continue nor is the working of the gear affected, unless there are some errors in the shapes of the teeth, lack of lubrication or other disturbing causes.

Before a vessel is accepted by the Admiralty the gears are opened up after the full-power trials and all surfaces, and especially the roots and ends of the teeth scrupulously examined with a powerful magnifying glass to ascertain whether there are any flaws or signs of cracking present that may lead to subsequent fatigue failures.

In view of the wide application of mechanical gearing for marine propulsion there is now ample evidence regarding its reliability, and it will be of interest to consider how failures may occur in these gears. The types of failures are :—

(1) Excessive wear of the teeth; and—
(2) Fracture of the teeth.

There are also cases where the working of the gearing is accompanied by a good deal of noise, the features producing the noise being apparently not such as to be detrimental to the life or efficiency of the gear.

Of the two types of failure mentioned above, the first is, perhaps, the most serious, as once wear sets in deformation of the surface of the teeth is caused, and the wear continues with increasing intensity. Unless the power is reduced, continued running will eventually lead to the removal of the gear for recutting to correct the form of the teeth or the substitution by another gear. Apart from the reduction of life involved the working of the gear is seriously prejudiced by such wear as it is accompanied by excessive noise and heavy "shivering" which transmits its effects to the turbines and subsidiary fittings and may cause failure of lubricating pipes, &c., or even more serious breakdowns. The failure may be caused by errors in cutting the gear wh·ch have contributed to give an incorrect shape of tooth or errors in alignment either when first built up or caused by working of the vessel, more especially in cases where the turbines and gear cases are separately supported. With the solid construction· now adopted, errors in alignment are not likely to arise if the intention of the design is fulfilled This serious fault has fortunately been very rare in naval gears, and in the very few cases in which it has occurred it was in evidence on the contractors trials and generally corrected before vessel proceeded on service and in the other cases after a short period of service. There have been a few such cases also in the mercantile marine in the double reduction sets.

Fracture of teeth may or may not be a serious failure. In a number of cases an isolated tooth has fractured and the broken piece fallen clear. If however, the fractured portion be caught between pinion and wheel it will cause damage at the contact, and the excessive strain brought on the teeth locally may cause further failure and an extension of the trouble. If not too serious, the fractured teeth may be smoothed off and the gear replaced as good as new and there are cases of gears in the service running to-day with portions of teeth missing. The fracture of the teeth, which usually takes place at the end of a pinion element, may be due either to defective metal or an excessive stress due to the load being badly distributed. For example, a discrepancy in the helical angle of pinion and wheel respectively would cause the load to be taken mainly at one end, and it is of the utmost importance in gear-cutting therefore to ensure that the helical angles of the components are the same; a very small difference in angle is sufficient to give a faulty distribution of load.

If the extreme ends of the teeth were exposed to the load there would be a danger of excessive local stress arising due to the fact that the teeth at this point are not fully supported there owing to the overhang of the teeth. To avoid the load coming on this part the corners of the teeth of the pinions and gear wheel are chamfered off for a length of $\frac{1}{4}$ in. measured along the axis. As a further safeguard to the teeth of the pinions having in view torsional deflections and an unequal distribution of load they are thinned off for a length of $\frac{3}{8}$ in. at each end.

As regards the loads that may be taken on gearing, running experience is the only guide. It is usual to express the load in terms of lbs. per inch axial width of tooth surface, but a better indication is expressed by load in lbs. per inch length of contact, as the length of contact is not always proportional to the tooth width since it depends on the helical angle and tooth proportions. With the 45° angle gears, pressures up to and exceeding 1,000 lbs. have been common practice and have been entirely satisfactory, but with the 30° angle, owing to the increase in contact length, and with the same main proportions as in the 45° gears, 700 lbs. represents the highest pressures now running with this angle in H.M. Service.

It is interesting to point out that the removal of a gear wheel and pinions of a T.B.D. in the case where excessive wear took place did not involve the scrapping of the large and expensive gear wheel. The wear was too great to admit of the teeth being cleaned up, and the same general dimensions being maintained but teeth of a deeper type were cut and new pinions supplied to suit. As the existing gear case had to be utilised and it was not desired to alter its alignment in relation to the turbines, it was necessary to maintain the same distances between centres of wheels and pinions, and it can be seen from the following table that this requirement was fulfilled.

——	Original.	New.
Teeth in wheel	401	401
,, H.P. pinion	47	49
,, L.P. ,,	62	64
P.C.D. of wheel	85·9114″	85·577″
,, H.P. pinion	10·0694″	10·4038″
,, L.P. ,,	13·2831″	13·6175″
Depth of tooth	·355	·4738
Revs. wheel	350	350
,, H.P.	2980	2864
,, L.P.	2260	2193

The slightly reduced turbine revolutions due to the fitting of pinions each with two additional teeth had no noticeable effect on the efficiency.

Electrical Method.

With this arrangement one or more high-speed turbines are used to drive dynamos which generate the electric current necessary to actuate one or more slow-speed motors fitted on each propeller shaft, either direct or through mechanical gearing. The details of the motors are so designed as to give electrically a number of speed reductions somewhat similar in effect as obtains in the change-speed gear of a motor car. The number of vessels so fitted at the present time is small, the highest-powered ships being the American battleships of approximately 30,000 S.H.P., of which the "New Mexico" has now been on service some time. The new American battle cruisers of 180,000 S.H.P. are designed for a similar system, but these are not at present very far advanced. The system was first tried on a moderately large scale in the U.S. collier "Jupiter," of 7,000 S.H.P., and the experience gained in this vessel was utilised for the development of the "New Mexico" design. In the vessels referred to the motors drive the propeller shafts direct, and it is only in the case of low-powered cargo-ships that mechanical gearing has been interposed between motors and propeller shafts. This is necessary or desirable to avoid the use of large motors running at low revolutions. A reduction of revolutions in a typical case is from 3,600 at the electric generators to 1,750 at the motors and 75 at the propeller shafts.

The electrical system of transmission as generally understood has not yet been tried in H.M. Service, but in the submarines, power is in effect transmitted electrically from the oil engines to storage batteries and then to motors when submerged, the motors being the dynamos when batteries are being charged up. Also in the " K " class submarines, which are ordinarily propelled by geared steam turbines on the surface, a separate Diesel engine driven electric generating set is fitted to charge the storage batteries for the motors for submerged running; if required, this engine and dynamo could be used to drive the motors direct when on the surface, these motors driving the propeller shafts through mechanical gearing.

To adequately explain the system as developed for a large high-powered vessel the arrangement as fitted in the " New Mexico " will be described. The main propelling machinery in this vessel consists of two alternating current turbo-generators and four induction motors fitted one on each of the four propeller shafts. As subsidiaries to the above there are two 300 K.W. direct current generators for excitation of the main generator fields and for driving the various motor-driven auxiliaries; two motor-generator boosters, the functions of which are to vary the fields on the main generators without necessitating variation of the voltage produced by the exciters, which must remain constant in view of the auxiliaries supplied by them; a main switchboard carrying the alternating switches and control apparatus generally for the main units; a direct current switchboard; and two ventilating fans for each main motor.

Alternating currents with high voltage are necessary to meet the conditions for electric transmission in its application to ships of high power. The main generators are two-pole, nominally rated at something over 15,000 H.P. each, but have an overload rating of 25 per cent. They are designed to run at 2,100 revolutions per minute at full speed. In their main features these units do not differ from those used in ordinary commercial practice. Each generator is provided with a disconnecting switch which is double-throw; in one position of the switch the windings are arranged in a series connection and in the other position of the switch in a parallel connection. The first position is used when one generator is operating two motors, and under this condition the maximum voltage is 4,242 volts. The second position is used when one generator is operating four motors, and under this condition the maximum voltage is 3,000 volts. This arrangement enables the generator to utilise to the best advantage the double current capacity of the motor circuit when using four motors and improves the overall efficiency of transmission.

The main motors are designed to give together 29,000 S.H.P. at 167 revolutions per minute, equivalent to a ship's speed of 21 knots. The stator windings are arranged to provide for pole changing; one throw of the pole-changing switch will give 24 poles, and the other will give 36 poles on the stator. This gives for synchronous speed with the two-pole generator in the first case a 12 to 1 reduction ratio and in the second case an 18 to 1 reduction ratio between turbine and propeller revolutions. This device enables the turbine to be operated at full revolutions both at 21 knots and at 12/18 or two-thirds speed, *i.e.* approximately 15 knots, and thus improves the economy of the installation at low speeds.

The motor stator is provided with only one winding, and the pole-changing is accomplished by changing the arrangement of this one winding.

The motors are of the double squirrel cage induction type, with two independent sets of windings in two slots which are separated by a deep air gap of narrow section. The squirrel cage in the outer slot is composed of high resistance bars of German silver, and that of the inner slot is composed of low-resistance bars of copper. The object of this construction is to provide a winding which will give the necessary torque required for reversing and at the same time make it possible to use the squirrel cage motor, which is a very simple type of motor and requiring no changes in its rotor winding when shifting from 36 to 24 poles and *vice versâ* in the motor stator. The high torque required for reversing can be obtained in other ways by a further complication in the rotor design, and the use of external resistance and this device is under consideration for some of the later U.S. battleships. The "New Mexico" arrangement has, however, the advantage of simplicity.

The details of the theory of action of this type of motor and the effect of running out of step with the generator cannot be gone into here, but the general action will be understood from what has been said and also when considering the operation of the plant.

It is necessary to slow down the turbines during reversal, otherwise the motor could never pull into synchronism with the generator.

A speed governor is fitted to the turbine with a moveable fulcrum which is used for all speed changes. The governor can be adjusted to give any desired speed on the generator from 700 to 2,200 revolutions, and when once set will maintain that desired speed regardless of changes in load. The governor weights operate a pilot valve of a hydraulic relay, which admits oil pressure from the forced lubrication system to a piston driving a rack. This rack actuates a pinion mounted on a cam shaft, and the motion of the cams opens successive valves in the steam chest of the turbine. The effect of the motion of the speed control lever is to shift the fulcrum about which the governor weights act, and thus give a different speed setting for each position of the fulcrum. In addition, a fitting is provided which limits the motion of the governor for any given setting and thus the amount of steam the turbine can take and the power delivered for that setting. This is necessary on account of the increase of power which can be taken by the inner shafts when the ship is turning. This power might be greater than the generator could give at the speed if the motors are to run at synchronous speed, and the result would be that the inboard motors would drop out of step and stop. With these limit stops the turbine simply slows down and the motors remain in step.

An emergency overspeed governor is also fitted which, on a certain designed speed being reached, trips the main steam and auxiliary exhaust steam admission valves and shuts steam completely off the turbine.

The control arrangements will now be considered.

Control gear for use on shipboard presents conditions different from those obtaining in land power stations. In the design and fitting of all switch-gear, instruments, &c., great care must be taken to exclude moist air, particularly when it is impregnated with salt, and sometimes coal dust. Adequate facilities for inspection must also be provided. In large vessels, where heavy currents at high voltages have to be dealt with, the problem of fulfilling the necessary requirements of the control gear is not an easy one remembering that simplicity, reliability and certainty of operation are absolutely essential.

On the main switchboard are mounted the following :—

(1) The motor disconnecting switches, one for each motor.

(2) The bus-tie or sectionalising switch which connects the two sides of the ship, making it possible to run all four motors from either generator or the two sides independently. This switch may, therefore, be looked upon as performing a similar function as a cross-connection valve in a twin or four-screw steam installation.

(3) The pole-changing switches, one for each pair of motors.

(4) The reversing switches, one for each pair of motors.

(5) The speed levers which are connected by rod mechanism to the generator governors.

(6) The steam limit levers referred to previously.

(7) The field levers for the main generators. These levers control solenoid-operated switches on the direct current exciter board and also the rheostat of the booster which controls the strength of the main field. The range

of voltage given to the generator field is from 30 to 300 volts. This variation is provided in order to obtain the highest efficiency of the generators and motors under the varying conditions of speed and load.

The disconnecting switches of the main generators are mounted on the engine room bulkheads just outboard of the main switch board.

The general arrangement in the five engine rooms containing the main machinery is shown in Fig. 5.

The functions of the various switches will be understood from their nomenclature, but it is necessary to introduce various complications in the way of mechanical and electrical interlocks so that it is impossible to do any damage to the machinery or impair the life of the fittings through mistakes in operation and big rushes of current. These interlocks will be briefly described as they are a necessary feature of electrical transmission at high powers and voltages. It may be noted that these arrangements would be much further complicated in an installation with more than two generators, especially if the generators are fitted with pole-changing arrangements, as has sometimes been proposed; also by the fitting of additional motors on each shaft in the attempt to obtain further "gear-downs" by pole-changing arrangements on the different motors.

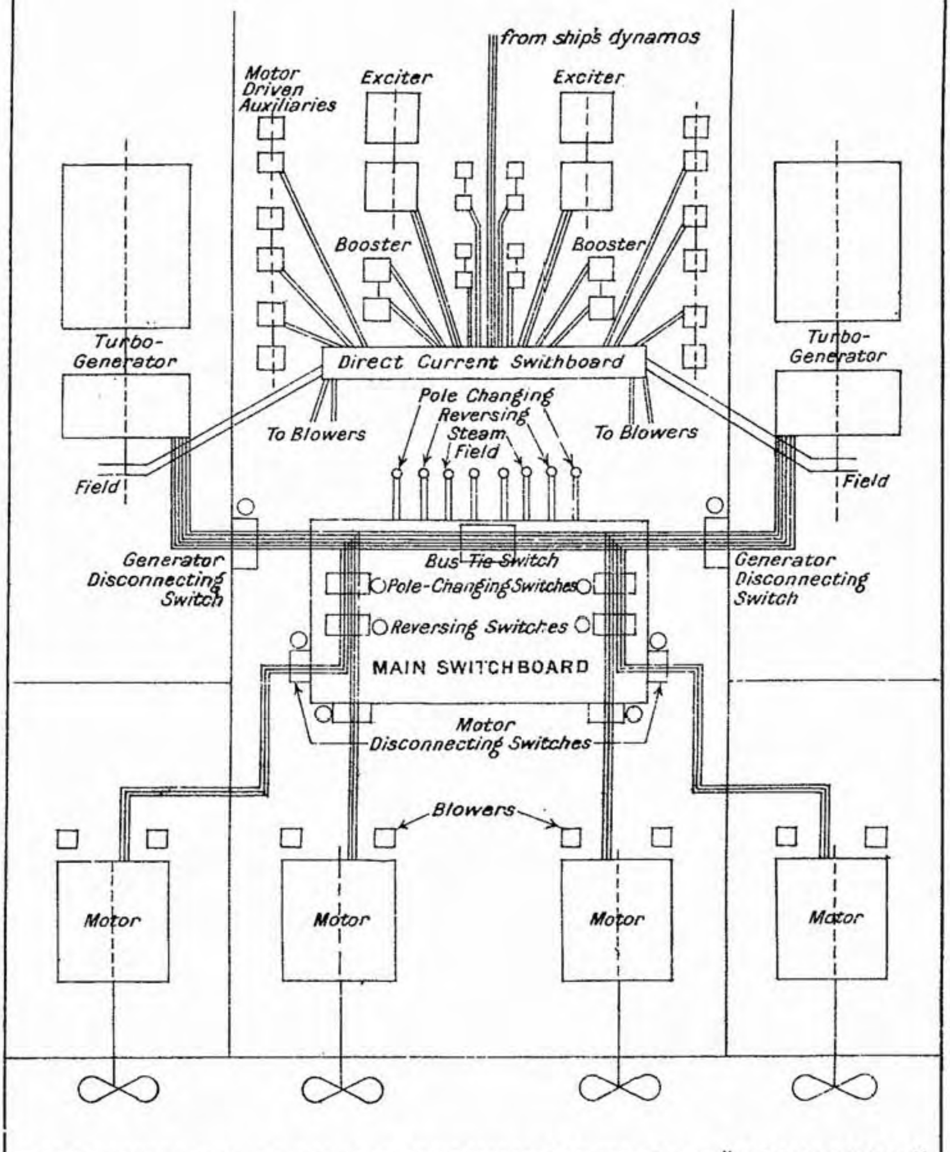
1. The motor disconnecting switches and the generator disconnecting switches are electrically interlocked and cannot be operated unless the generator field switch is open, *i.e.* all circuits are dead.

2. The bus-tie switch which is inside the board structure is inaccessible unless all the circuits are dead.

3. The bus-tie switch and the main generator disconnecting switches are mechanically interlocked so that only two of these switches can be closed at any one time, thus preventing the two main generators being coupled in parallel. They are also interlocked so that the generator disconnecting switches can be thrown only into the high voltage position when the bus-tie is open, and only into the low-voltage position when it is closed, thus ensuring that the generator will always be run with low voltage when operating four motors.

4. The pole-changing and reversing switches are mechanically interlocked with the field lever so that they cannot be moved unless the field lever is open and all circuits are dead. An additional safeguard is fitted so that these switches cannot be opened immediately the field switch is opened but till the current in the main generator of motor circuits has fallen to a predetermined value. This involves a delay of not more than three seconds under normal conditions.

5. The pole-changing and reversing levers are mechanically interlocked so that the reversing levers cannot be put into the astern position unless the pole-changing switch is in the 36-pole



DIAGRAMMATIC PLAN OF MACHINERY IN U.S.S. "NEW MEXICO."

TRANSMISSION OF POWER.

FIGURE 5.

10246. 32374/D33.

Malby & Sons. Lith.

position. This is to insure the maximum torque being available when reversing.

6. The field levers and speed levers are mechanically interlocked so that the speed lever is carried to low speed whenever field is taken off the main generator. A large increase of speed carries the field lever with it. Large increases of speed cannot therefore be made without corresponding increases in the field strength, and the turbine must be reduced in speed before reversal of the main motors is effected.

The operation of the main machinery can now be understood.

1. *Getting underway* with one generator and motors on the 36-pole connection. This assumes the maximum speed will not be over 15 knots. The turbine would be running at low speed and all circuits would be dead. On receiving an order to go ahead—

(*a*) The pole-changing switches would be put in the 36-pole position.

(*b*) The reversing levers would be thrown in the ahead position.

(*c*) The field switch would be closed and brought up to the desired field strength and the motors would start and work up towards 1/18 the revolutions of the generator.

(*d*) The turbine would then be brought up to the desired speed, the motors keeping in step.

2. *Reversing*—

(*a*) Open field switch, the speed lever being carried to low speed at the same time.

(*b*) When current has fallen and the relay operates for releasing the reversing levers, switch over to the astern position.

(*c*) Close the field switches and apply over-excitation till the motors have reversed and come up to synchronous speed and then reduce the field to normal.

(*d*) Increase the speed of the turbine to the revolutions desired.

In case it is desired to make a maximum speed greater than 15 knots but less than 17½ knots, *i.e.* in this case not more than one-half full power, one generator will be sufficient, but the motors will be run on the 24-pole condition. Going ahead will be exactly the same as described above. To reverse, however, it will be necessary, after opening the field switch and bringing the turbine to slow speed, to switch the pole-changers to the 36-pole condition and proceed as before.

For speeds above 17½ knots the two generators will be used, the motors being on the 24-pole condition for going ahead, and changed to the 36-pole condition for astern working. The bus-tie switch will be open and the two sides will work independently.

The limitations of action in this vessel may now be seen, but it has been urged that these limitations are not serious, the extreme flexibility of action with the ordinary geared or direct driven arrangement being more a matter of convenience.

When on one generator and proceeding ahead, all four shafts must run at the same speed ahead. If one side is reversed, the side going astern must run astern at the same revolutions as the other side ahead and before reversing, the side that is to remain ahead must be slowed down before it is possible to reverse the other side. It is also possible to run ahead one side and the other stopped. When all shafts are reversed the full power is not immediately available for going astern as is often claimed, as the motors are in the 36-pole condition to give the necessary torque, and to obtain full power it would be necessary to break the circuits after reversal, ease, and change to the 24-pole condition.

Efficient ventilating arrangements for the generators and motors are necessary in an electric installation, as the loss of power in transmission, which amounts to about 7 per cent. of the energy passed, appears as heat in these details. The generator is fitted with a fan attached to its spindle, whilst separate electrically driven fans are fitted for the main motors.

This indicates a special danger with this system. In case of local overheating of an intensity sufficient to cause ignition of some of the inflammable details of the structure, the draught from the ventilating fans might cause a rapid increase and extension of the combustion and a firing up of the machine. Fire-extinguishing apparatus is specified for some of the electrical details of the new U.S. battle cruisers, and an automatic cut-out of the fan or main switch appears to be indicated in the case of overheating of the circuits.

Although not yet attempted, it may be possible to recover some of the lost power by arranging to lead the hot air to the boiler rooms or in the event of water-cooling being devised, it might be possible to arrange for feed-water heating.

Special care has to be given to the design, support, and the high degree of insulation of the large cables necessary to prevent injury from damp and shocks, and the insulation of the motor and generator circuits requires great care, more especially if these units are working in compartments exposed to steam leakage.

The installation proposed for the battle-cruisers is generally similar in principle to that of the battleships. The number of units is doubled. There are two motors on each shaft. At low speeds any one turbo-generator can be used to actuate all four shafts; at intermediate speeds two generators will be used and at high speeds each shaft will be actuated from its own turbo-generator. The excitation for the main generators is taken direct from the 300 K.W. generating sets fitted exclusively for the purpose, the variation in excitation being obtained by varying the voltage of this generator.

A feature of the electric drive is that most of the auxiliaries are electrically driven at a cost, say, of 15 lbs. of steam per horse power against a moderate estimate of 40 lbs. for direct steam-driving. This advantage can, however, be obtained in other installations by the extension of the use of electrically-driven auxiliaries, entailing the fitting of additional dynamos.

Variable Speed Gear.

Before reviewing the merits and demerits of the three systems discussed, it is as well to note that there is another method of transmitting power hydraulically, that is, by means of the form of variable speed gear on the well-known Williams-Janney principle, by means of which any speed within limits in either a forward or reverse direction without altering the speed or direction of running of the prime mover is possible. Apparatus of this type has not been used for transmission of power for marine propelling machinery except for very small powers, but has been fitted a good deal for auxiliary purposes in some special classes of vessels where intermittent work is required. The efficiency of transmission is comparatively low and does not probably exceed 85 per cent., and the gear is very heavy for the power. Apart from the doubt as to the practicability of the development of this system for high powers, the efficiency is sufficient to place it out of the running for a large power installation running continuously, and further, elaborate arrangements would also have to be fitted to absorb the heat generated in the gear.

Relative Merits for Marine Use.

A brief description of the three practicable forms of transmission at present available having now been given, it is proposed to compare them in certain of their aspects which govern their suitability for marine use, and more particularly for the propelling machinery of war ships.

1. *Efficiency of Transmission.*—With the best design of the respective devices the losses in transmission may be approximately taken as :—

> *Hydraulic.*—Ten per cent., the losses being mainly frictional, eddying, &c., in the transformer. This efficiency includes the heat gain from the water previously referred to.
>
> *Mechanical.*—One and a half per cent. to two per cent. per gear, the losses being frictional at the tooth surfaces, bearing and rotational losses.
>
> *Electrical.*—Eight per cent. being electrical and mechanical losses in the dynamos, motors, and cables.

The mechanical method, therefore, in this important particular possesses a marked advantage over the other devices, an advantage which to a large extent outweighs the advantages possessed by the other methods.

2. *Ratio of Reduction.*—This is a fixed figure at all speeds for the mechanical method. With single reduction gears a figure in an extreme case as high as 28 to 1 has been reached, but 7 to 10 is a more usual standard for naval designs. For double reduction figures of 30 to 40 to 1 are common.

To secure the best efficiency of the turbines over a considerable range of speed it is therefore necessary to design the turbines with cruising arrangements, or to fit special cruising turbines. The ratio of reduction in the hydraulic arrangement is practically fixed, but a figure of about 7 to 1 is the highest limit for this device.

W th the electrical method one or more definite rates of reduction can be given by suitable pole-changing arrangements, and it is possible to arrange for a high speed of turbine at suitable reduced speeds. In the case of the "New Mexico" it has been seen that this is done for 15 knots besides the full speed. Moreover, at reduced speeds only the number of generators required for the power need be used, and it is still possible to work all the propeller shafts. The electrical method alone possesses this advantage, which is important as at reduced powers the load factor, and therefore economy, of the turbines in operation remains high. The advantage of us'ng only part of the generating installation would, however, fall away under war conditions when all machinery was kept avai!able for full power at short notice.

3. *Reversibility and Astern Power.*—Separate astern turbines are necessary with the mechanical method, and considerations of space and weight entail their being of smaller power and low economy as compared with the ahead turbines. Apart from an increase in weight and overall dimensions due to the astern turbines fitted, there is a permanent loss of about one per cent. of power in driving the astern turbine when running ahead.

With the hydraulic method a reverse transformer is necessary and although the turbine is available for full power the reverse transformer is made of a simpler design and a certain amount of efficiency sacrificed.

With the electrical method no astern turbine or motors are necessary, the propeller shaft motors being arranged to work in either direction and a high power astern is available, but other considerations, *e.g.*, the necessity of giving the motors a high-starting torque involving running under certain pole conditions prevent the full power being obtained or availed of normally. Actually in the "New Mexico," the maximum power astern of about one third that ahead was accepted during the official trials as suitable for requirements, which is easily obtained with the ordinary design in geared arrangements.

The fact that the turbines run always in the one direction has been urged as an advantage if super-heated steam is used, and this will probably be the case in most of the future designs from the point of view of economy. The higher temperatures which may arise in turbines during reversal in a superheated installation can be readily provided for or eliminated by suitable precautions in design and operation.

4. *Manœuvring.*—This is pre-eminently simple with the mechanical method as no special apparatus is required in connection with the gearing, it being only necessary to manipulate the turbine manœuvring valves. With the hydraulic method it is a little more complicated owing to the necessity of emptying one and charging the other transformer when reversing. The arrangements have been previously described.

The additional complications with the electrical method have been discussed, and likewise some of the limitations of manœuvring inherent with this device.

5. *Care in Operation.*—Little or no special care is required for the mechanical gearing. Provided the oil supply is maintained, nothing else is required other than that which will be necessary for the care of the turbines themselves.

The hydraulic drive should not require much attention if carefully worked, and all safety arrangements kept in good condition.

The electric drive will, in general, require more attention, as the large electrical units working in steamy or damp compartments require keeping well ventilated and dry. The personnel will require to have a special knowledge of electrical practice.

6. *Weight and Space.*—The electric drive is on the whole heavier than the other systems, but possesses some superiority in utilising space to better advantage, since the turbines are not fixed by the positions of the propeller shafts as in the other designs, thus admitting of a greater amount of latitude as regards arrangement in the ship. The motors can be placed well aft and short propeller shafts used. The generators can be placed close to the boilers, and the length of the main steam piping therefore reduced. The manœuvring gear can be placed in any convenient position.

7. *Durability and Reliability.*—As far as can be at present stated, the advantage in this respect unquestionably lies with the mechanical gearing, sufficient experience being now available under hard service conditions to warrant the statement that well-constructed and attended gearing is one of the most reliable parts of a marine installation. The risk of short circuits in an electrical installation due to mechanical derangements, moisture, &c., is always present in spite of elaborate safety arrangements and under the conditions which may arise in action it is at least doubtful if even the most elaborate arrangements will afford the safeguards which have been yielded in the past in British practice.

8. *Range of Application.*—The mechanical gear can be, and has been, applied to all classes of ships, but as at present developed the electrical drive is only applicable to capital ships of the battleship or battle cruiser type of high power in which economy at reduced speeds and powers is vital, in which the necessary weight and space can be provided, and in which full advantages can be taken of the flexibility as affecting the subdivision and disposition of the plant. For mercantile vessels of high power the system presents certain advantages and in these vessels floor space might be saved by installing the generators above the motors in the generally large unutilised spaces in the hatchways.

In the case of small steamers of the cargo-carrying type the system is not generally so attractive. The steam and fuel consumptions of the Swedish mercantile vessels of the "Mjolner" type of about 1,000 S.H.P. are, however, very good, advantage having been taken of a high degree of superheat in conjunction with efficient Ljungstrom turbines. Mechancial gearing is, however, necessary between motors and propellers in such small-powered and low speed vessels.

9. *Manufacture.*—The capacity for the manufacture of high-powered electrical plant in this country is limited.

Summing up, it may be generally stated that the hydraulic drive in the form of Fottinger transformers is inferior in all essential respects to the mechanical gearing, and it appears unlikely that it will be fitted in any future high-power marine installations. The general indications are all in favour of the mechanical gear holding the field for moderate and high powers in this country, but the more extended use of the efficient Ljungstrom turbine which is inherently suited to electric power generators, may, in the near future, broaden the field of the electric drive for low and moderate powers. As further experience is obtained, developments in the electric drive may be anticipated if the desired requisites on the electrical side can be effectually catered for. Developments on special lines must be mainly looked for in the motors as it is in these special fittings that many changes are required as compared with those worked under shore conditions.

Intermediate forms of transmission are not likely to be required in internal combustion engined vessels as suitable speeds of revolution can usually be arranged with a direct drive, but special cases may arise where lower manœuvring speeds than can ordinarily be obtained with these engines are required, and if the weight could be accepted there may be developments of the electric drive with these engines as complications in the way of manœuvring and reversing of these engines could be thus avoided. Mechanical gearing has not been applied to large internal combustion engines, and is open to one disadvantage at least, *i.e.*, the doubt as to the continued efficiency of the gear when transmitting a variable torque.