be directly compared, and the combined work compared with what would be done by the steam, supposing the whole expansion had taken place in the large cylinder only.

In the tandem type of engine the two cylinders are in direct communication during the whole stroke till compression begins in the small cylinder ; consequently the back pressure line of the top diagram is practically identical during the greater part of its length with the steam line of the lower pressure diagram. The gap between them represents loss of pressure due to the receiver formed by the pipes between the two cylinders and to the resistance of the passages.

Fig. 207 represents a pair of diagrams from a two-cylinder compound engine with receiver, the cylinders having diameters of 46 and 87 inches respectively, and the common stroke of 57 inches, and the ratio of the cylinders 3·6 to 1. The particulars as to the initial and mean pressures and the horse-power developed by each cylinder are given on the diagrams.

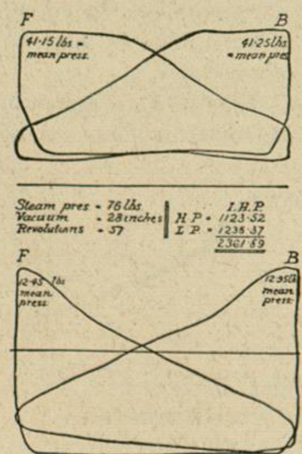

Fig. 207.

It will be noticed that the scales of pressure for each cylinder are quite different, that for the high-pressure being 60 lbs. to the inch, and for the low-pressure 16 lbs. to the inch. In order to make the diagrams comparable, the pressure ordinates ought each to be reduced to the same scale, and the volume ordinates should be in the ratio of the volumes of the two cylinders. This is effected in the following manner Draw the base line AB, fig. 208, corresponding with the zero of pressure. Divide each diagram, fig. 207, by any number of vertical ordinates. Divide the line AB at the

point C into two parts, so that AC : AB :: vol. of small cylinder to vol. of large cylinder :: 1 : 3·6. Divide AC and AB into the same number of parts as there are divisions in the original diagrams, and draw through these points a series of vertical ordinates. Then measuring from the base line AB mark off on these ordinates, to any convenient scale, the pressures as found from the corresponding ordinates on the original diagrams. And through the points thus found draw the new diagrams as shown on fig. 208.

In order to compare the work done by the steam in the compound engine with the work that would be done if the expansion took place all in the large cylinder, we must know first the rate of expansion in the compound engine. This in the present example is as nearly as possible ten. Hence, allowing an admission line DE of one-tenth the stroke, and drawing in the hyperbolic line of expansion EF, we obtain the approximate diagram for this ratio of expansion in a single cylinder, on the assumption that there is no loss from condensation. It will be seen that the two actual diagrams fit fairly well into the single approximate diagram after allowance has been made for the 'drop,' or fall of pressure at the end of the stroke of the high-pressure cylinder, and for the resistance due to passages between the cylinders. Instead of the hyperbola EF, the curve $pv^{1 \cdot 0646} = C$, which represents the adiabatic curve of expansion of dry saturated steam, is often made use of. Some engineers consider that $pv^{1 \cdot 2} = C$ represents more closely what takes place.

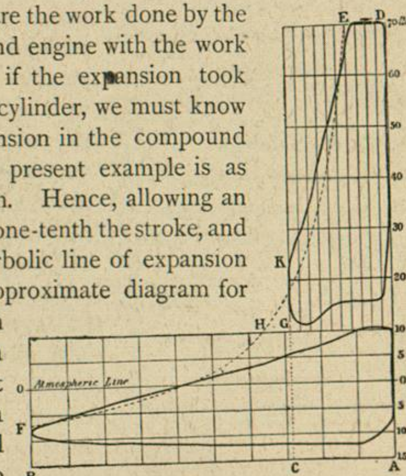

Fig. 208.

Fig. 209 is a set of indicator diagrams taken from a three-cylinder compound engine, having two low-pressure cylinders and cranks set at equal angles. The particulars as to boiler and receiver pressure, vacuum, number of revolutions, horse-power, &c., are given on the diagrams. The engines from which these diagrams were taken belong to the Trans-atlantic steamer the 'Arizona' illustrated in figs. 197, 198.

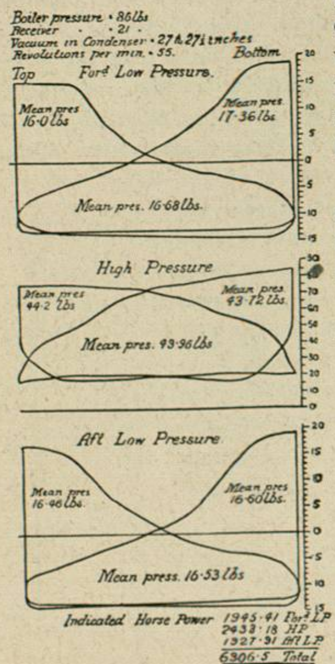

Fig. 209.

Fig. 210 gives the diagrams taken from a set of modern triple expansive marine engines, together with a combination diagram formed in the manner already explained. The dotted line shows the adiabatic expansion of steam with the same ratio of expansion in a single cylinder, and the gaps between the actual diagrams and the adiabatic line show the losses due to 'drop' in the two receivers, to the resistance of passages and to the condensation which takes place in the cylinders.

The mechanical advantages of compound engines.—In addition to diminishing the loss due to condensation in the cylinders, the compound engine possesses mechanical advantages over the older type of engine, in which the expansion takes place completely in a single cylinder. In the latter case there is a great difference between the initial and mean strains on the piston, whenever the rate of expansion is high; whereas in the compound engine the difference between the initial and mean strains in each cylinder is much reduced. Similarly the twisting moment

on the crank shaft is more nearly uniform in the case of the compound engine. Now the dimensions of the moving parts have to be designed to meet the maximum strains; hence

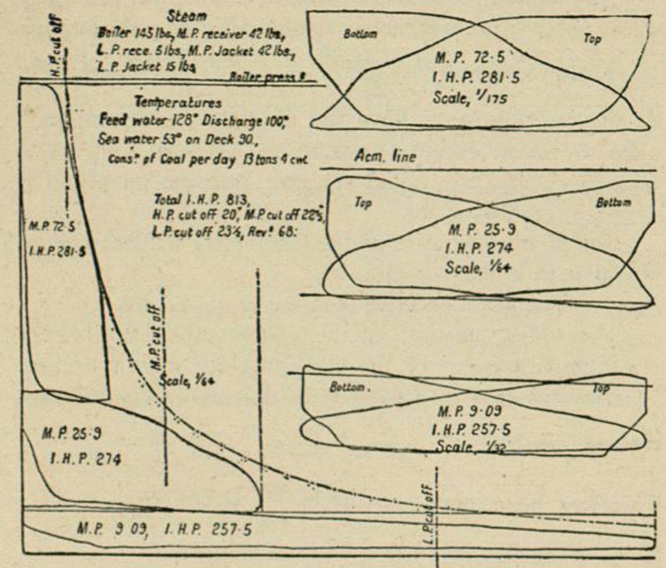

Fig. 210.

with compound engines a saving may be effected in the weight of these parts.

As examples we will compare, first, the case of a single cylinder expansive engine with a tandem compound; and, second, a two-cylinder expansive with a two-crank compound of the receiver type, neglecting clearance and compression and assuming hyperbolic expansion.

Let the initial pressure of the steam be 100 lbs. per square inch, absolute; the ratio of expansion 6, and the area of the piston in the ordinary expansive engine, A. Then, if both engines have the same stroke, the area of the low-pressure cylinder of the compound engine will also be

A, and if the ratio of the two cylinders be, say, 4, the area of the high-pressure cylinder will be $\frac{A}{4}$.

Taking first the simple expansive engine, the theoretical mean pressure for hyperbolic expansion is got by the formula.

$$\text{Mean pressure} = p\frac{1+\log_e E}{E} = 100\frac{1+1\cdot7918}{6} = 46\cdot5 \text{ lbs.}$$

If the back pressure be 4 lbs. absolute per square inch, then the mean effective pressure on piston $= (46\cdot5-4)$ A $= 42\cdot5$ A lbs., the initial effective pressure on piston $= (100-4)$ A $= 96$ A lbs.

Taking now the case of the compound tandem engine, we have, as in the first case,

The mean effective pressure $= 42\cdot5$ A lbs.

The initial pressure on the low-pressure piston equals the terminal pressure on the high, provided there is no loss.

Now the rate of expansion in the high-pressure cylinder $= \frac{6}{4} = 1\cdot5$.

Therefore the terminal pressure $= \frac{100}{1\cdot5} = 66\cdot6$ lbs.

Now the effective initial load on the small piston

$$= (100 - 66\cdot6)\frac{A}{4} = 8\cdot32 \text{ A.}$$

And the effective initial load on the large piston

$$= (66\cdot66 - 4) \text{ A} = 62\cdot66 \text{ A.}$$

Total initial load $= 71$ A, as against 96 A in the first case, the mean pressure being the same in each instance.

Take next the comparison between a two-cylinder simple expansive engine and a two-cylinder receiver compound, the area of the low-pressure cylinder of the latter being, as before, denoted by A, while the piston area of each cylinder of the simple engine is $\frac{A}{2}$. The other data are supposed to be unchanged. The mean pressure in the simple engine is the same as in the first example, viz. $46\cdot5$ lbs. per square inch.

The mean effective pressure on each piston

$$= (46\cdot5-4)\frac{A}{2} = 21\cdot25 \text{ A lbs.}$$

The initial effective pressure on each piston

$$= (100-4)\frac{A}{2} = 48 \text{ A lbs.}$$

Next take the case of the compound engine, and suppose the ratio of the cylinders to be 3 to 1. As the rate of expansion is 6, the cut-off in the small cylinder must be at half-stroke.

In an engine of this description the receiver pressure would probably be 28 lbs. Now to find out the point of cut-off in the low-pressure cylinder in order that this may be the receiver pressure at the commencement of the stroke of the large piston, we must remember that the total rate of expansion is 6, and therefore the final pressure in the large cylinder is $\frac{100}{6} = 16\frac{2}{3}$ lbs. Therefore the cut-off in this cylinder is $\frac{16\frac{2}{3}}{28} = \cdot59$ of the stroke. From these data we deduce the following particulars :—

Mean pressure in small cylinder

$$= 100\frac{1+\log_e 2}{2} = 84\cdot65 \text{ lbs. per square inch.}$$

Mean effective pressure in the small cylinder

$$= 84\cdot65 - 28 = 56\cdot65 \text{ lbs. per square inch.}$$

Mean effective load on small piston

$$= 56\cdot65 \times \frac{A}{3} = 18\cdot88 \text{ A lbs.}$$

Initial effective load on small piston

$$= (100-28)\frac{A}{3} = 24 \text{ A lbs.}$$

Mean pressure in large cylinder

$$= 28\frac{1+\log_e 1\cdot69}{1\cdot69} = 25\cdot2 \text{ lbs. per square inch.}$$

Mean effective pressure in large cylinder

$$= 25\cdot2 - 4 = 21\cdot2 \text{ lbs. per square inch.}$$

Mean effective load on large piston $= 21{\cdot}2 \times A$ lbs.
Initial effective load on large piston $= (28-4)\ A = 24$ lbs.

Hence we see that in the simple engine the initial load is to the mean as 2·16 to 1. Whereas in the small cylinder of the compound engine the ratio is 1·27 to 1, and in the large cylinder 1·13 to 1.

It is unnecessary to pursue this investigation further, for it is evident that in the cases of three-cylinder ordinary compounds, and triple expansive engines the ratios will be more in favour of the compound system than in the two examples given.

The foregoing calculations take no account of the strains caused by the inertia of the reciprocating parts, and they

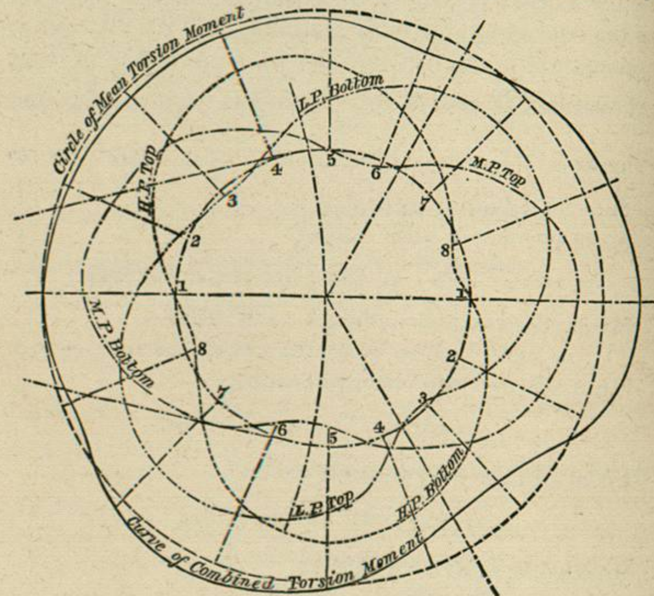

Fig 211.

therefore only apply to the strains when the engine is moving slowly, as, for instance, when stopping and starting. The

proper method of investigating the true pressures on the moving parts, when their inertia and the effects of compression are taken into account, is explained in Chapter V.

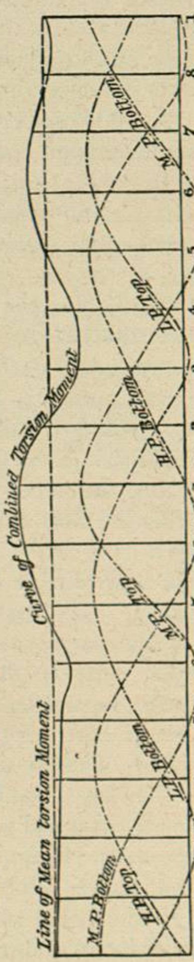

Fig. 212.

The curve of twisting moments on the crank axle is also more uniform in the case of the compound engines, and as the reciprocating parts are comparatively light, the influence exerted by their acceleration and retardation on the curve of moments is less than in the case of simple engines working at the same rate of expansion. Fig. 211 is given as an example of the uniformity of twisting moment which may be obtained with a triple expansive engine having three cranks set at 120°, and fig. 212 is the same diagram drawn on a straight instead of a circular base. The indicator diagrams from which the curves were deduced are also given (see fig. 210).

The relative sizes of the Cylinders in Compound Engines.—Within certain limits the engineer possesses considerable latitude in the choice of the ratio of cylinders in compound engines. The objects to be kept in view are—1, to divide the power as equally as possible between the cylinders—2, to avoid an excessive rate of expansion in either cylinder, which produces a high ratio of initial to mean pressure—and 3, to avoid excessive drop in the receiver.

It is always possible to equalise the power developed by the two cylinders, by altering the point of cut-off in the large cylinder, no matter what the ratio may be between the two. The effect of increasing the rate of expansion in the large cylinder is to increase the receiver pressure and consequently the initial pressure, while at the same time it increases the back pressure in the small cylinder. Consequently the proportion of the power developed in the small cylinder is decreased.

As all marine engines, no matter what the initial pressure may be, expand down to about the same terminal pressure, the size of the large cylinder is fixed solely by the power to be developed, for it must be large enough to contain the whole volume of the steam at the terminal pressure. Hence the ratio of the two cylinders is determined by the size we choose to give the small cylinder. If we start with the assumption that the low-pressure cylinder is to exert half of the total power developed, then the mean pressure in it for a given power is always the same, and as the terminal pressure is also constant, the point of cut-off depends upon the pressure in the receiver.

Now the larger the high-pressure piston relatively, the smaller the mean pressure in it per square inch, for a given power. But as the initial pressure is fixed, we must have a more unfavourable ratio of initial to mean pressures with relatively large high-pressure cylinders than with small. Also the larger relatively the high-pressure piston, the smaller must be the terminal pressure in it, and consequently the smaller also the receiver pressure would be, unless it were kept up by an early cut-off in the large cylinder. Thus we see that when the high-pressure cylinder is relatively large, that is, when the ratio of the two cylinders is small, the initial pressure in each must be high, relatively to the mean pressures. While on the other hand the 'drop' in the receiver will be comparatively small.

As the boiler pressure increases, the rate of expansion should increase also ; hence, either the cut-off must take place earlier in the small cylinder, or else the ratio of the two cylinders must be increased. If the proportions are

such that the cut-off in the small cylinder must take place before half-stroke, it will be necessary to provide this cylinder with a separate expansion valve.

The following[1] table gives the proportions of cylinders most generally in use so as to avoid the use of expansion valves, and to secure a favourable ratio of initial to mean pressures with a moderate 'drop' in the receiver.

Type of Engine		Boiler-pressure absolute								
		85	95	105	115	125	135	145	155	165
Tandem . .	Ratio of large to small cylinder	4 to 3·5	4	4·5	5	—	—	—	—	—
2 cylr. receiver .	Ditto	3	3·75	4	4·5	—	—	—	—	—
3 cylr. receiver .	Combined ratio of low-pressure to small cylinder		3·4	3·7	4	—	—	—	—	—
Triple expansive	Ratio of low-pressure to high-pressure cylinder	—	—	—	—	5	5·4	5·8	6·2	6·6

In the case of triple expansive engines the ratio of the large low-pressure to the intermediate cylinder should be half that of the low-pressure to the high.

[1] These figures are given on the authority of Mr. A. E. Seaton's *Manual of the Marine Engine.*