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Sir ALEXANDER B. W. KENNEDY, LL.D., F.R.S., President,
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“Internal-Combustion Engines for Marine Purposes.”

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THE increasing use of internal-combustion engines on land, and their undoubted economy, have led to considerable interest being taken in their application to marine purposes. Already a large number of such engines have been fitted in small craft, principally on the Continent. In this Paper, however, these small engines will not be considered, except so far as they may furnish lessons in design for larger engines, which will be necessary if the internal-combustion motor should come into use for ordinary sea-going vessels.

The Author was led to give attention to engines of this kind in connection with proposals to fit them in vessels classed with Lloyd's Register; and he desires to thank Messrs. Beardmore & Co., Messrs. Rodger & Co., Messrs. Richardsons, Westgarth & Co., Messrs. Thornycroft & Co., Messrs. Crossley Brothers, Messrs. Hornsby & Sons, and also many of his personal engineering friends, for the information and experience regarding internal-combustion engines which they have placed at his disposal.

On land, internal-combustion engines have been made to use various fuels, namely, petrol, refined oils, heavy hydrocarbon oils, coal-gas, producer-gas, coke-oven gas, and blast-furnace gas. Of these fuels, petrol and refined oil, on account of their cost, are suitable only for very special vessels, and up to the present heavy oil and producer-gas are the only fuels which it is necessary to consider in connection with large marine engines. As regards producer-gas, only that made from coke, or from anthracite coal, seems at present to be available for marine work, but there is a reasonable expectation that a producer will be perfected soon which will enable bituminous

coal to be used without requiring the elaborate washing-arrangements which can be employed on land for freeing the gas from dust and tar. It is not intended, however, in this Paper to deal with the question of producers, but only to discuss the mechanism of the engine itself. Although the requirements for a successful marine engine are very different from those for a land-engine, it will be well first to glance at what is being done on land in the way of large installations.

Where very large powers are not required, the engines are generally single-acting with single cylinders, and as the four-stroke cycle is usually employed, this leads to large cylinder-volumes in proportion to the power generated, and it also necessarily involves the use of heavy fly-wheels. The largest engines of this type with which the Author is acquainted are some made by Messrs. Richardsons, Westgarth & Co., with cylinders $51\frac{1}{4}$ inches in diameter, 55 inches stroke, running at 80 revolutions per minute, and indicating upwards of 800 HP. These makers, however, have ceased to build this kind of engine, and for large powers they make double-acting two-cylinder tandem engines. Of these they have made several sets with cylinders $29\frac{1}{2}$ inches in diameter, $35\frac{1}{2}$ inches stroke, running at 110 revolutions per minute, and giving about 850 I.H.P. per pair of tandem cylinders. Messrs. Crossley Brothers have made engines of almost identical dimensions. The Deutz Gas-Motoren Fabrik have made tandem double-acting engines up to 2,000 I.H.P. with two cylinders. Messrs. Rodger & Co., of Glasgow, have made large vertical engines resembling the marine type. Other makers have also made large and powerful engines, and the experience of detail necessary to make the large cylinders capable of withstanding the shocks of explosion and the high temperatures of the burning fuel is therefore easily available if the other mechanical requirements are complied with.

Much useful information upon the latest experience with land engines is given in three Papers recently contributed to the Iron and Steel Institute.

For marine purposes the largest set the Author is acquainted with has been made by Messrs. Beardmore & Co., of Dalmuir.¹ These engines are intended to use producer-gas, but they have not yet been put to work at sea.

The next in point of size are engines on the Diesel principle using heavy mineral oil: they have been fitted in two vessels trading in the Baltic.

¹ For particulars see Transactions of the Institution of Naval Architects, vol. xlviii (1906), p. 112.

Nearly all of the small craft fitted with internal-combustion engines use heavy oil as a fuel.

The special conditions to be satisfied by a successful marine engine are:—

- (a) The engine must be reversible.
- (b) It must be capable of being stopped quickly and of being started quickly either ahead or astern.
- (c) It must be capable of being promptly speeded to any desired number of revolutions between dead slow and full speed, and of being kept steadily at the required speed for any length of time. "Dead slow" ought to be not faster than one-quarter of full speed, and should be less than this in very fast vessels.
- (d) It must be capable of working well, not only in smooth water, but also in heavy weather in a seaway in which the varying immersion of the propeller causes rapidly changing conditions of resistance.
- (e) All working parts must be readily accessible for overhauling, and all wearing surfaces must be capable of being promptly and easily adjusted.
- (f) The engine must be economical in fuel, especially at its ordinary working-speed.

In small engines condition (a) is not complied with: the engines are always run in one direction, the reversing of the vessel being provided for either by a reversing-bladed propeller or by the intervention of toothed gearing. Condition (b) also, in small engines, is fulfilled by means of a friction-clutch arrangement which permits the engine to be kept running while the screw-shaft is stopped. For large sea-going vessels, however, condition (d) practically puts movable-bladed propellers, clutches, and toothed gearing entirely out of question, and necessitates a reversing engine, with a direct drive on to the screw-shaft, and a fixed propeller. In large engines, also, conditions (b) and (c) prohibit the use of a heavy fly-wheel, and necessitate a fairly equable turning-moment at all working-speeds.

In marine engines the revolutions are practically proportional to the speed of the ship, and as the vessel's resistance increases much more rapidly than the speed, it follows that the mean effective pressure in the cylinders must be capable of being reduced in very much greater ratio than the speed of revolution. For instance, at one-half of full speed the power exerted is less than one-eighth of the full power, so that the mean effective pressure is less than one-fourth

of that at full power; whilst at dead slow, say at quarter-speed, the mean pressure would be only about one-sixteenth of that at full power. This great range of mean effective pressure is very difficult of attainment, especially as a fly-wheel is not admissible, so that the reduction of power by the suppression of, say, alternate fuel-charges is not practicable.

In considering the turning-moment and how it affects the lower limit of speed of revolution, it may be pointed out that in a steam-engine the mean pressure can be reduced in almost any desired degree by throttling the steam-inlet. In an internal-combustion engine, however, there is not nearly the same possibility of range. The pressure may be reduced, but only in a certain ratio, by cutting off part of the fuel-supply though not the air-supply. Below this point the mixture will be too poor to ignite. Even at the critical point at which ignition is just possible, the mean pressure produced

Fig. 1.

is comparatively high. If the ratio of mixture of air and fuel is kept at an inflammable point, and the supply is throttled, a reduction of pressure is obtained; but there is a limit to this also, because the clearance-space is in most engines left filled with burnt gas at atmospheric pressure, and this dilutes a too-greatly enfeebled charge below the ignition-point. The indicator-diagram shown in *Fig. 1* is from a Crossley gas-engine. The heavy diagram represents the working at a full charge, the lighter diagram that with the charge reduced to about as low a point as is practicable. At the full power the mean effective pressure is 81·3 lbs. per square inch, whilst at the low power it is 36·8 lbs.

In a marine engine this reduction of mean pressure would represent a reduction of speed of revolution of only 32 per cent.; in other words, if speed could be reduced only by such reduction of charge as in this instance, in a vessel whose full speed was

10 knots per hour, no speed less than $6\frac{3}{4}$ knots per hour could be attempted without the engines stopping.

The question of the effect of a fly-wheel upon the steadiness of the motion of an internal-combustion engine was dealt with by Mr. James Atkinson, of the firm of Messrs. Crossley Brothers, in January, 1906, in a Paper read before the Manchester section of the Institution of Electrical Engineers. His Paper took full account of the effect of inertia of the reciprocating parts of the machinery, and showed that with the actual weights of the single-acting single cylinder that he was considering, the turning-moment varied from 7.7 times the mean moment in a forward direction to about 1.5 times the mean moment in the reverse direction; in other words, the total variation of load on the shafting was more than nine times the stress due

Fig. 2.

to the mean load. Such an engine is out of consideration for marine purposes.

While the four-stroke cycle is retained, the minimum number of cylinders necessary to give always a positive turning-moment is four single-acting or two double-acting; and even with these numbers of cylinders the turning-moment will vanish at the dead points, and the engine will have all the irregularities of motion of an ordinary single-cylinder steam-engine.

It will be well to examine the turning-moments of the double-acting tandem engine, which is the simplest engine employing the four-stroke cycle that will work without a fly-wheel or its equivalent. *Fig. 2* represents an actual indicator-diagram of a horizontal engine of this type by Messrs. Richardsons, Westgarth & Co.; but, as marine engines are almost invariably made vertical,

this diagram will be taken as representing the cylinder-pressures of a vertical tandem engine of the same diameter and stroke. In such an engine, besides the effect of the inertia of the reciprocating parts, there are also to be considered the weights of the parts themselves, as the weight acts in conjunction with the forward pressure on the down stroke and in opposition to it on the up stroke. Assuming the length of the connecting-rod to be of the usual proportion, namely, four times that of the crank, the upper curve in *Fig. 3* represents the actual forward turning-moment of the engine in each revolution due to the four impulses which occur every two revolutions.

It will be seen that, owing to the effect of the compression, there is actually a slight negative moment just before the two dead points. The ratio of maximum turning-moment to the mean or useful work is 2.14:1. The lower curve in this diagram represents separately the effect of the dead weight and of the inertia of the moving parts which have been included in the turning-moment curve. If two

such engines, having the cranks at right angles, be combined, the power is of course doubled, but at the same time a more equable movement is obtained, there being at no point a negative moment, while the ratio of maximum to mean moment is reduced to 1.75:1.

When three cranks and three pairs of tandem cylinders are used the turning-moments are as shown by the lower curve of *Fig. 4*. The motion is still more equable, the ratio of maximum to mean being reduced to 1.31:1.

When four cranks and four pairs of tandem cylinders are employed, the turning-moments are as in *Fig. 5*, one curve showing the case in which the cranks are set at equal angles of 90° , when the ratio of maximum to mean is 1.48:1, the other curve showing the case in which the cranks are at angles of 0° , 90° , 135° and 225° , when the ratio of maximum to mean is 1.39:1.

It will be seen that the increase from three to four cranks has not been an improvement. This leads to the conclusion that an ordinary internal-combustion engine with three cranks will probably, on the whole, give the best results from the point of view of smoothness of running.

It has been mentioned that a reduction of 55 per cent. of the mean pressure—probably about the greatest that can be obtained in an ordinary internal-combustion engine by the enfeeblement of the charge—leads only to a reduction of 32 per cent. of the speeds of rotation and of the vessel, if the engine works with all the cylinders charged: it will of course be possible, in a three-crank engine with double-acting tandem cylinders, to cut out entirely one set of cylinders, say the top set. This will reduce the effective mean moment by one-half, and in consequence the speed of revolution will be reduced by about 30 per cent. The effect of this on the turning-moment is shown for two revolutions in *Figs. 6*, which give the moment both for one cylinder only and for the combination of the three cylinders. The latter curve shows that the engine will still run with three cylinders working, there being always a positive turning-moment.

If, besides cutting out half the cylinders, it is possible to reduce the mean pressure in the remaining cylinders in the same proportion as in the diagram for the Crossley engine (*Fig. 1*), the result will be a reduction of the effective moment to $22\frac{1}{2}$ per cent. of that at full power, and of the speed of revolution to a little less than half that of full power. This is a better range of speed than is obtained by reduction of pressure only in all the cylinders, but it still is not satisfactory for marine work. It is doubtful, however, whether the

engine would run with so great a reduction of charge. The subject of a greater reduction of speed will be referred to later on.

It will be convenient here to state briefly the principles involved

in the four-stroke-cycle internal-combustion engine, in order to

see what modifications are desirable or possible for introducing it into marine work.

In *Fig. 7*, AB represents the total capacity of the cylinder, AC being the clearance-space, CB the volume swept through by the piston. At the commencement of the cycle there is a volume AC of burnt gases in the cylinder at about the atmospheric pressure; the first stroke draws in a volume CB of combustible mixture also at about atmospheric pressure. The second stroke compresses the mixture of burnt and unburnt gases into the volume DE (= AC) of the clearance-space, the compression increasing the pressure to DC above the atmosphere. At this instant ignition takes place, the heat generated increasing the pressure to CF. On the third stroke the

Fig. 7.

hot burnt gases expand till they fill the whole cylinder-volume, and at the termination of the stroke they have the pressure GB. On the fourth stroke the larger portion of the gases is exhausted from the cylinder, leaving the volume AC at about atmospheric pressure to take part in the next cycle.

During the second stroke an amount of work represented by the area BDC has to be done in compressing the gas, and during the third stroke an amount of work, represented by BCFG, is performed by the gas, the area BDFG representing the net work obtained during the cycle.

It will be observed that at the commencement of the exhaust the gas-pressure represented by GB is still very considerable, and that the energy represented by this pressure is not utilized in the engine. For instance, in the Richardsons-Westgarth engine, which has been referred to, the terminal pressure is 32 lbs. per square inch above the atmosphere, whilst in the Crossley engine it is about 40 lbs. In marine engines it is generally considered that the expansion of the steam may be usefully carried so far that the terminal pressure shown on the indicator-cards is not more than 6 lbs. or 7 lbs. above the back pressure. Greater economy could therefore be realized in the internal-combustion engine if by any means a greater expansion could be obtained when using the same quantity of fuel. For instance, if the volume could be increased to AL additional work represented by the area LBG M would be obtained.

Turning to the other end of the diagram, it will be seen that if the working-volume could be increased by an amount which would reduce the clearance to AK, an addition to the work would be made represented by the area DHNF, whilst the addition to the fuel-charge would be represented by only the volume KC; and as the pressure in this volume KC is always much greater than the mean pressure, it is seen that the useful work done by increased compression is always proportionately greater than the excess of charge used. The smaller the clearance-spaces, therefore, the greater will be the economy, but the greater also will be the work which has to be done during compression, and the greater will be the initial pressure at ignition. Further, as compression is always accompanied by a rise of temperature, an excessive compression might be accompanied by so high a temperature as to cause premature ignition. It is the necessary limitation to the safe initial pressures and compression-temperatures which determines the possible range of economy obtainable at this end of the diagram. In the Richardsons-Westgarth engine to which reference has been made, the initial forward pressure in the cylinder is 450 lbs. per square inch above the atmosphere, and in the Diesel engine a pressure of 600 lbs. per square inch is often reached. Although an increase of economy obtained by reducing the clearance necessitates high pressures within the cylinders, whilst that obtained by increasing the expansion involves only lower cylinder-pressures, yet, on account of the larger area of cylinder that the latter requires, any given increase of area at the right hand of the indicator-diagram involves higher loads on the piston-rods, connecting-rods, and crank-pins than an equal increase of area made on the left hand of the diagram.

As the compression-stage of the cycle always precedes the impulse, some auxiliary power is necessary for starting the engine by giving

the first turn round. In small launch-engines hand-power may be used, but with larger engines mechanical means have to be applied, the most usual being the use of compressed air supplied by auxiliary mechanism. For large engines, therefore, a separate air-compressing engine, combined with adequate storage-capacity, is a necessity, and this in spite of the fact that one-half of the functioning of all the main cylinders is taken up in gas-compression.

If the necessary compression for ordinary working could be performed in a separate cylinder, the compressed gas being stored in a suitable receptacle and admitted to the main cylinders at the commencement of the stroke, the main engine could be made a two-stroke cycle instead of a four-stroke cycle, and the same size of main cylinders would then yield considerably more than double the power obtained where the four-stroke cycle is used. Referring again to *Fig. 7*, the work at each end of the cylinder per two revolutions would in a two-cycle engine be twice the area BCFG, instead of, as in the four-cycle engine, once the area BDFG; but in this case the gas-compressors would have to be capable of doing during the same time the work represented by twice the area BAED. In other words, by separating the work of compression and expansion into two different sets of cylinders one set only will have to be made to withstand the excessive pressures and temperatures of combustion, and this set will be of smaller diameter, for the same net work, than the cylinders employed in the usual way, whilst the other set will have to be made somewhat larger, but will have to withstand only the pressures and temperatures due to compression. The change will be accompanied, however, by a considerable loss of efficiency, due to the excess of work required in forcing the gas out of the cylinder into the receiver, from which it will pass into the clearance-space of the main cylinder without giving up a corresponding amount of work.

As an instance, consider the Richardsons-Westgarth engine again. In *Fig. 4* the lower curve has already been given as representing the turning-moment of a three-crank six-cylinder double-acting tandem set of engines working on the ordinary four-stroke cycle. The upper curve represents the turning-moment of a three-cylinder three-crank double-acting set of cylinders of the same size, in which the compression is performed in a separate set of cylinders. In the former case the ratio of maximum to mean turning-moment is 1.31 : 1; in the latter case the ratio is nearly the same, namely, 1.33 : 1, but the mean turning-moment is increased by 26 per cent., whilst no extra stress is put upon the pistons, piston-rods, or connecting-rods. The loss of economy incidental to this change is,

however, very large, being equivalent to a mean increase of back pressure of more than 25 lbs. per square inch, or a loss of more than 30 per cent. If the separation of the compression from the expansion-cylinders were necessarily coupled with so great a loss of efficiency, it would be useless to discuss it, even although it would entirely solve the problem of being able to run the engine at low speeds. This, however, is far from being the case.

Referring again to *Fig. 7*, this time as an ideal diagram of an engine with separate compressing- and expanding-cylinders, for the compression of the charge and forcing it into a separate receiver the cylinder requires to be made with a working-capacity equal to AB, which is the working-volume plus the clearance-space of an ordinary engine using the same amount of mixture. This cylinder also requires to be made with the minimum amount of clearance, so that the whole of the charge is forced out at each stroke. The work done in compressing the charge up to the pressure of the receiver is BCD, and CAED is the further work done in forcing it out of the cylinder into a receiver. If, in the same way, the expansion-cylinder be made with a working-volume equal to the volume plus clearance of the ordinary type of engine and with practically no clearance, then, by admitting the charge from the receiver into a portion of the stroke equal to AC before ignition, an amount of work is returned by the compressed mixture which is equal to that expended in pushing it out of the compression-cylinder. If it be then fired the remaining part of the diagram is obtained as before. The net result is that the same total useful work is obtained with the same charge as before, or, in other words, there is no loss in economy. Further, as it is no longer necessary to make the main cylinder the same size as the compression-cylinder it may be made still larger—say, with a total volume equal to AL—and then the further work represented by the area LBGM will be obtained from the same charge. This is an actual gain of economy.

Although the changed design involves the use of somewhat larger cylinders, it leads to many incidental advantages. There being no clearance, there will be practically no burnt gases left in the expansion-cylinder, and the amount of the charge can therefore be reduced not only by an earlier cut-off but also by throttling, without the risk of diluting it below the ignition-point. By an earlier cut-off, retaining the same admission-pressure and the same quality of charge, a greater expansion can be obtained and therefore greater economy at powers somewhat below the full power. With an earlier cut-off, accompanied by a reduced admission-pressure, the combustion of the charge can still be accomplished, and at the same time the mean pressure can be

lowered very considerably below that obtainable under any ordinary

working-conditions with the four-stroke-cycle arrangement involving large clearance-spaces; and with this reduction of pressure a far greater range of speed is possible.

Among other incidental advantages may be mentioned the fact that, as the cycle is a two-stroke one, there will be an entire suppression of the 2-to-1 gear for actuating the admission- and exhaust-valves, it being possible to provide for the necessary valve-movements and also for their requisite adjustability, both for variations of speed and for reversing, without the intervention of any toothed gearing. This is a point which will appeal to all marine engineers.

The turning-moment of such an engine as that indicated above is shown in *Fig. 8*, calculated from the indicator-diagram shown in *Fig. 9*, which is the same pressure-volume curve as has been used before, but

with the stroke increased by the clearance required in the engine

from which this diagram was taken. The necessary allowances have been made for the effect of the reciprocating weights due to the altered condition of the design. The peaks shown in *Fig. 8* are of course due to the supposition of the ignition occurring instantaneously, which, however, will not actually happen; but even on this supposition it is seen that the resulting motion is not incompatible with good working, the ratio of maximum to mean moment being only 1.55:1.

With regard to the separate compression-cylinders, *Fig. 2* shows that at the commencement of the compression-stroke the cylinder is not actually full of the charge at atmospheric pressure. In fact, with fast-running engines it is impossible to draw in the full volume of the charge. In the particular case illustrated, the cylinder is only

Fig. 9.

about three-quarters full. With separate compression-cylinders, larger air- and gas-inlet valves are possible, and fuller charges can be introduced, so that in the supposed engine it would be quite sufficient to make the working-volume of the compression-cylinder equal to or even slightly less than that of the ordinary engine with which it is compared, instead of making it equal to the full volume plus clearance. But it will doubtless be found advisable to divide the compression—or at least that of the air—into two stages. By this means a much more equable turning-moment is required for the compression-mechanism than is needed where a high degree of compression has to be obtained by one stage.

In oil-engines it is only necessary to provide for the compression of the air: in gas-engines both gas and air have to be compressed. This may be done after their mixture, but preferably they should be

separately compressed and only become mixed as they enter the cylinder. In the latter case, if the gas is maintained at a pressure slightly higher than that of the air, there will be no danger of an air-leakage into the gas and no risk of explosion by back-fire in ordinary working-conditions.

The compression of the gas or air is necessarily accompanied by a considerable increase in temperature, in fact, in some of the oil-engines in successful use this feature is necessary for the ignition of the charge. Where compressed air is required for industrial purposes the heat produced by the compression is purposely abstracted and is wasted, but in the case under consideration this will not be necessary, and beyond the cooling by the cylinder-walls, the lubrication of which will not present greater difficulties than that of the main engine-cylinders, there need be no abstraction of heat.

It will probably be found to be advisable for the main engine to take its charge directly from the pipe between the compressor and the receiver, not from the receiver itself; the heat of compression will thus be received in the main cylinders practically as soon as it is produced, instead of being dissipated in the receiver. Further, in the opening of the receiver, through which there will be constantly an ebb and flow of some small part of the air-charge, there may, if necessary, be fitted a regenerator which will retain the heat of the hot gas entering the receiver and will raise the temperature of the cooler gas leaving it.

An engine on these lines can be made with the compressors driven from the main-engine crank-shaft, in which case the net available power is the difference between that developed in the expansion-cylinders and that utilized directly in compression. It is, in fact, the same power as is available from the same charges in the ordinary four-stroke-cycle engines, and, as has been shown, the suppression of the clearances enables a much lower speed to be obtained when necessary. By the complete separation of the compression, however, that is, by performing it by means of a separate engine, it will be possible for any required degree of slowness to be obtained by suppressing the whole of the fuel-supply and admitting compressed-air only to the cylinders; and as a supply of compressed-air is always required for starting-purposes, and must be provided by some auxiliary appliance, the most convenient arrangement will be for the compressors to be driven independently of the main engines.

In conclusion, to fix ideas as to required dimensions of cylinders, etc., a three-crank tandem set of engines working on the ordinary

four-stroke cycle will be taken, the indicator-diagram of which is shown in *Fig. 2*. If each of the six cylinders be 30 inches in diameter, the water-cooled piston-rods 5 inches in diameter, the stroke 3 feet, and the speed 100 revolutions per minute, then, with an effective mean pressure of 86.5 lbs. per square inch, there will be obtained, say, 3,250 I.H.P. Such an engine could not be reduced in speed below 65 revolutions per minute without risk of stopping.

In the proposed arrangement of separate-compression engines with the two-stroke cycle any required slowness could be obtained. The mean forward pressure would be 119 lbs. per square inch, and for the same indicated power, namely, 3,250 I.H.P., there would be required, if the same stroke and speed of revolution were retained, three expansion-cylinders 25½ inches in diameter. Allowing a mechanical efficiency of 0.85 for the compressing-gear, about 1,450 HP. will be necessary. If performed in two stages, the compression would require, say, two double-acting air-cylinders about 31 inches in diameter for the first stage, and two double-acting cylinders about 18 inches in diameter, or one double-acting cylinder about 25 inches in diameter, for the second stage, the necessary power being obtained from a pair of ordinary four-stroke-cycle tandem engines with four cylinders 24½ inches in diameter, or one tandem set with two cylinders 34½ inches in diameter, assuming the same piston-speed as before for all the cylinders.

The compressor-engines might be fitted with a fly-wheel, and as they have to work against a comparatively fixed resistance, proportional to the receiver-pressure, they could be speeded as desired without difficulty.

The increase in the number of cylinders due to separating the compression from the main engine is a disadvantage; but although the total cylinder-capacity is greater, the diameters of the cylinders are smaller, and only half of them are subject to the high temperatures and pressures of the burning fuel; so that, on the whole, it is thought that the advantages of the arrangement more than compensate for the increased number of working parts.

The Paper is accompanied by three drawings, one tracing and one sun-print, from which the Figures in the text have been prepared.