(Paper No. 3327.)

"High-Speed Electrical Generating-Plant."

By THOMAS HERBERT MINSHALL.

SINCE the publication of Mr. Willans's Papers on Steam-Engine Trials¹ there has been a continuous succession of improvements in the design of the high-speed engine; and it is intended to consider in this Paper some of its advantages, as a prime mover, at the present day. Although it is believed that a part of the data is novel, the object of the Paper is not so much to present original matter as to endeavour to gather together the most recent available information from a number of different sources, in order that the subject may be fully discussed.

The distinction between high and low speed is more or less indefinite; the general practice at the present time, with respect to angular and piston speeds, is indicated by *Fig. 1*.

Recent development of the high-speed engine has been almost entirely confined to Great Britain, but its earlier history is largely American.² As long ago as 1861, the late Mr. John F. Allen built small engines to run at 160 revolutions to 200 revolutions per minute, and showed at the London Exhibition of 1862 an engine with a cylinder 8 inches in diameter and a stroke of 2 feet, which ran at 165 revolutions to 225 revolutions per minute. Several engines were supplied to Trinity House in 1867, having a speed of 300 revolutions per minute; and in the same year one running at 500 revolutions per minute was exhibited at Paris.³ From this date onward, engines of the same or similar kind were built in considerable numbers, both in England⁴ and in the United States. But difficulties in lubrication prevented any great increase in speed or in size, until the single-acting engines of Willans, and later those of Westinghouse, were brought out.

¹ Minutes of Proceedings Inst. C.E., vol. xciii. p. 128; and vol. cxiv. p. 2.

² Porter, Transactions American Society Mechanical Engineers, vol. xxii. p. 65.

³ Proceedings Institution Mechanical Engineers, 1868, p. 50.

⁴ By Brotherhood and others.

Some comparatively high piston-speeds were attained, however, in locomotive and marine work (*Fig. 1*).

The investigations of Willans led in England to a development of the high-speed engine such as has taken place nowhere else; and resulted in the introduction of a scientific system of designing and testing, previously unknown. The single-acting

Fig. 1.

engines of Willans and those of Westinghouse showed a marked increase in economy: due, in the former, to excellent design and accurate workmanship; and in the latter, possibly to the reduced clearance obtained with the horizontal piston-valve. The development of forced lubrication by Belliss has now made it possible to run double-acting engines as fast as single-acting engines, or faster (*Fig. 1*). Extensive experience of both these kinds of high-

speed engine in this country has served to establish their reputation firmly. The high-speed engine, however, has been little used in America, where it has been looked upon, since the days of Corliss, as an uneconomical and somewhat unreliable piece of machinery, suitable for small power only.¹ But, as will be shown later, there are other reasons for the almost universal adoption of low-speed engines in the power-houses of the United States: a practice which is the more remarkable, inasmuch as it is a well-known characteristic of American engineering to demand the maximum output from machinery in the minimum time, and to attach less importance to its life than is usual in this country.

In England, as soon as it was found that high-speed engines were capable of meeting the requirements of electric-lighting stations, the demand for them rapidly increased, and resulted in a continual improvement in their design. Until quite recently, builders of high-speed engines have practically kept pace with the demand for increased sizes; and in spite of the claims for higher economy and greater reliability made by the advocates of low speed, it is probable that at the present time more electric power is generated in England by high-speed than by low-speed engines. (See Appendix, Table I. B.)

The success of the high-speed engine has resulted in a tendency to increase the speed somewhat, even of Corliss engines, as well as to produce a class of medium-speed engines² (*Fig. 1*). The latter are unable to use trip gear; but a number of excellent positive valve-gears³ have been produced, some of which give extremely good steam-distribution.

The Author proposes to consider the advantages of the high-speed engine with reference to:—

(1) Adaptability.—Dealing with the causes which limit the speeds of engines and dynamos, and with the questions of governing, vibration, inertia, parallel running, etc. “Speed” throughout the Paper means angular, not piston speed.

(2) Efficiency.—Dealing with the effect of clearance, superheating, jacketing, different ratios of expansion, valve-gear, etc.

(3) Cost.—Dealing with the comparative importance of cost with different load-factors, prices of fuel, etc.

¹ There are, of course, exceptions. Standard American high-speed engines are now made up to 750 HP.; and there is a 2,000-HP. mill-engine in New Hampshire running at 230 revolutions per minute.

² C. Day, “Engines for driving large dynamos.” Manchester Association of Engineers, 12 October, 1901.

³ Notably that of McIntosh and Seymour.

ADAPTABILITY OF HIGH-SPEED ENGINES.

Assuming for the present that increased speed gives increased economy, it is interesting to inquire into the causes which limit the ultimate angular velocity of engines;¹ and on this point there would appear to be considerable difference of opinion among designers, which cannot be entirely accounted for² by the necessity of conforming to the speed of the dynamo.

For every kind of valve-gear there is a limiting speed, beyond which satisfactory working is uncertain. At higher speeds than 120 revolutions per minute trip gear is not usual;³ but positively-driven Corliss valves are worked at much higher speeds.⁴ In general, however, medium-speed engines are fitted with flat slide-valves, at least on the intermediate- and low-pressure cylinders; although with superheated steam, which has led to the use of poppet-valves on many low-speed engines, it is becoming not uncommon to find piston-valves on the high-pressure cylinder. For speeds exceeding about 180 revolutions per minute, the use of piston-valves is becoming general in all but the smallest engines. Although engines working at much higher speeds with flat slide-valves are occasionally met with, experience has shown that, as a rule, the wear and tear and the inertia, together with the difficulty of balancing the steam-pressure, outweigh in the larger sizes any advantages of extra steam-tightness the slide-valve may possess. Speaking generally, it may be said that high-speed engines use some form of piston-valve, and hence at present their angular velocity is not limited by the nature of their valve-gear. But there is a limit to the speed at which the valve-gear can be worked satisfactorily, due to the difficulty of obtaining a sufficiently good exhaust with ports and valves of normal size. (See p. 167.)

Vibration.—Although the difficulties of securing quiet and continuous running at high speeds have gradually been surmounted by improved lubrication and workmanship and by a proper study of compression, it appeared at one time as though the question of

¹ See Thurston, "Stationary Steam Engines," New York 1884. Haeder and Powles, "Handbook of the Steam Engine," 3rd edition. London, 1902.

² Porter, "The Limitation of Engine Speed." Transactions American Society Mechanical Engineers, vol. xiv. p. 806.

³ Although drop valves have been used in small petroleum-engines at 200 revolutions per minute and upwards.

⁴ Example: the "Universal" Engine. In certain cases (e.g., Ferranti) the valve-gear is operated at half the speed of the engine.

vibration would set a limit to the speed and size of the high-speed engine.

Considerable trouble attended the use of single- and two-crank engines in early days;¹ but most of it was obviated, even with considerably increased piston-speeds, by the use of three-crank engines having lines of parts of equal weight. The six-cylinder or nine-cylinder three-crank tandem engine gives a very even crank-effort, and probably less vibration than any other kind. The Author's experience tends to show that the three-cylinder three-crank engine of triple-expansion, or even of compound type must be very carefully balanced and adjusted to give equally good results.

It is well known² that a rotating counterweight cannot perfectly balance a reciprocating movement, although fair results can be obtained when the connecting-rod is five or six times as long as the crank, and is of ample weight. Hence in the three-cylinder three-crank triple-expansion engine it has been found necessary to have the three lines of reciprocating parts of equal weight, in order to obtain steady running; and by this means vibration may practically be eliminated at all loads, even without balance-weights. Engines of this kind are being made with three cylinders up to 2,000 HP.; but the Author believes that with larger sizes it will be better to adopt a four-cylinder arrangement, and even possibly to balance it on the Schlick or a similar system; this would reduce the weight of the low-pressure piston, and would thereby partially compensate for the increased length,³ cost, and steam-consumption, which the use of an additional cylinder usually entails.

In all cases the foundations have to be carefully arranged to meet the special requirements of the site; and in certain places it has been found that the size of the site which is necessary in order to insure steadiness approaches that needed for vertical low-speed engines. York stone settings are found far preferable to grouting, and their use has eliminated a great deal of trouble.

Inertia.—The most serious result of higher speed is the increase in the inertia of the reciprocating parts.⁴ These parts, which are at

¹ Robinson, *Journal Institution Electrical Engineers*, vol. xxiv. p. 466.

² *Transactions American Society Mechanical Engineers*, vol. iv. p. 368; vol. viii. p. 191.

³ This is about 15 per cent. See Durston, *Minutes of Proceedings Inst. C.E.*, vol. cxxxvii. p. 207.

⁴ J. H. Dales, "High-Speed Engines," *Feilden's Magazine*, vols. v.-vii.: also Norris and Morgan, "High-Speed Steam-Engines," 2nd edition, p. 7. London, 1902.

rest on each dead centre, acquire during the first half of the stroke, and lose during the latter half, a velocity which near midstroke is equal to that of the crank. The effect of their inertia is greatest at the end of the stroke, where the incoming steam produces a reversal of stress, which, in the absence of cushioning, generally results in a knock.¹ As the speed is increased, however, up to a certain limit called by Dr. Thurston the "natural" speed, the inertia of the moving parts may become of advantage rather than otherwise by producing a more uniform effort upon the crank-pin.² In order to obtain this advantage to the full, it is generally necessary to increase the weight of small pistons beyond that actually required for strength; and it was by such means that Porter engines were enabled to run at speeds previously deemed impossible.

Engines running at high speeds possess the great advantage that the inertia prevents a sudden reversal of stress at the dead centres, and gives instead a gradual reversal, well beyond the dead centres, and quite independently of the amount of cushioning.

A number of diagrams, showing the effect of inertia upon the stress, and the benefit which results from a judicious combination of the forces due to steam and inertia, are given in *Figs. 2 and 3*. In certain cases the increase of pressure due to the inertia is greater³ than that due to the steam; and at high speeds sudden reversal of stress must be avoided altogether.⁴ In single-acting engines it is possible to do this by using cushioning-chambers; but in double-acting engines, when condensing, the amount of compression that can be obtained limits the speed. Moreover, the inertia of the valves in large engines is frequently a cause of knocking; which may again lead to the use of separate cushioning-cylinders.

Lubrication.—The introduction of forced lubrication for bearings has greatly improved the working of double-acting engines; but the problem of cylinder-lubrication requires further investigation. With the intermittent running which is usual in electric-light stations, the use of cylinder-oil cannot be reduced to the extent that is common in marine work, where it is frequently confined

¹ The knock is, as a rule, greater with a cut-off than with a throttle-governor, due to the full boiler-pressure entering the cylinder, in the former arrangement.

² Thurston, "Stationary Steam Engines," p. 65. See also Cotterill, "Applied Mechanics," 5th edition, p. 282. London, 1900.

³ In certain small fan-engines the force required to start the piston is one hundred and fifty times the weight. Cotterill, "Applied Mechanics," p. 224.

⁴ As is done in the Willans engines, for instance.

Figs. 2.

PROGRAMS WITH ABSCISSAE PROPORTIONAL TO CRANK DISPLACEMENT OR TIME

Cylinders, 14½ inches, 22 inches and 34½ inches in diameter, stroke, 16 inches; revolutions per minute, 295. I.H.P. = 11.1 P., 212.7; L.P., 244.7; L.P., 307.9—total, 765 I.H.P.

Effective H.P. = 664; efficiency = 86.7 per cent.

BELLISS ENGINE (No. 1233).—COMBINED STEAM AND INERTIA DIAGRAMS AT FULL LOAD CONDENSING.

[illegible]

to starting. The introduction of superheated steam has added to the difficulty. The use of a mixture of graphite and cylinder-oil, or, where feed-water troubles are anticipated, of graphite alone,

has been found beneficial;¹ but opinion is much divided on this question.

¹ The use of graphite probably assists initial condensation. See Rice, "The Laws of Cylinder Condensation." Transactions American Society Mechanical Engineers, vol. xviii. p. 950.

Dynamos.—Certain difficulties in connection with the design of continuous-current dynamos for high speeds have hitherto prevented manufacturers from keeping pace with the constantly increasing speeds adopted by engine-builders, although the continual advances which are being made show that the task is not impossible. Hence the speed of continuous-current combinations is limited by that of the dynamo, and engines often have to run at 5 per cent. to 12 per cent. below their standard speed, thus reversing the practice of a few years ago, when belts were used (Appendix, Table II. B).

In the United States the increase in the size of units took place more rapidly than in this country; and, instead of endeavouring to overcome the difficulties of design, the makers reduced the speed of the generators to that now common. In England on the other hand, most of the earlier continuous-current machines were comparatively small, and were used for the lighting of limited areas at low voltage; for which purpose the high-speed combination proved to be well suited, on account of the small space available in those days for generating-stations.

Considerable difference of opinion exists amongst designers as to the relative importance of the various difficulties connected with the design of high-speed dynamos; some take the voltage per bar, and others the amount of self-induction of the individual coils, as the chief factor in sparkless running; others consider both of these to be of less importance than a perfect commutator. The question, which is both important and interesting, has been fully discussed by various writers,¹ and will only be mentioned here. Formerly the effects of centrifugal force limited the size of commutators, and gave rise to much trouble; but now the difficulty is rather to prevent the displacement of the segments under the stresses of heating and cooling. The peripheral speed does not as a rule exceed 3,000 feet per minute (Appendix, Table II. A).

Notwithstanding these various drawbacks, high-speed continuous-current dynamos are increasing in size. Three years ago several manufacturers would not undertake to produce 300-kilowatt 500-volt machines to run at 360 revolutions per minute; to-day 700-kilowatt 500-volt machines run sparklessly at 280 revolutions per minute on tramway loads; and Messrs. Siemens have produced 1,200-kilowatt machines running at 200 revolutions per minute. Moreover, there appears to be good reason for expecting the present

¹ See Parshall and Hobart, "Electric Generators," p. 157; Hawkins's "Theory of Commutation"; Hobart, "Modern Commutating Dynamo Machinery," *Journal Institution Electrical Engineers*, vol. xxxi. p. 170.

demand to result in a further increase in the size, and reduction in the cost, of high-speed continuous-current dynamos; although, on account of the difficulties already mentioned, the price will not, apparently for some time at least, be much less than for low-speed machines.

For alternators the only limits to speed are of a mechanical kind; and, as a rule, it will be found that the speed of the engine fixes that of the alternator. Consequently the speed of the alternator is lower than it should be for maximum economy.¹ In the United States, belt-driven 800-kilowatt alternators run satisfactorily at 500 revolutions per minute, and 100-kilowatt alternators run at 2,000 revolutions per minute. In these the reduction in weight, and consequently also of cost, is considerable (Appendix, Table XII.).

Governing.—The commonest objection urged against the use of high-speed engines for electric-traction work is the difficulty of governing; and no argument has been more frequently advanced in favour of American low-speed engines. On examination, however, it is generally found that the close regulation of the latter is largely due to the fly-wheel; and reference to the results of governing-trials (Appendix, Table VIII.), and to the voltage in the charts (*Figs. 4*), shows that the regulation of high-speed engines can be made as close as is necessary for ordinary purposes.

Speed-variation may be conveniently considered under three headings, namely:—

- (a) Permanent change of speed.
- (b) Temporary variation (measured in seconds).
- (c) Instantaneous variation.

(a) The permanent alteration of speed due to change of load can, with most forms of governor, be made as small as is desired by the use of sensitive springs; any tendency to “hunt” being, as a rule, avoided by the use of suitable dash-pots.

In certain cases, as where the engine has been fitted with a combined throttle- and expansion-governor, it has been possible practically to obtain isochronism² with high-speed engines. (See Appendix, Table VIII. B.)

(b) The temporary change of speed due to change of load depends chiefly upon the fly-wheel; and, as is shown later, as much fly-wheel effect as is required can be obtained without

¹ Guilbert “European Types of High-Tension Alternators.” *Electrical World and Engineer*, vol. xxxvii. p. 113.

² With an inertia governor such as the Begtrup, almost instantaneous isochronism is obtainable.

difficulty in the high-speed engine. As already stated, the difference

in this respect between triple and compound engines is only slight.

(c) Even more important than the change of speed under varying loads is the question of parallel running,¹ which has been raised to the highest importance by the increasing use of alternating current and of rotary transforming devices worked through long cables. Here there can be no doubt as to the advantage of high speed.

Experience has shown that "hunting" is usually due to one of two causes; namely, either uneven turning-moment or unsuitable governors. Fly-wheel effect, or the use of a governor with small momentum of parts, counteracts the former;

Figs. 4.

THREE-CRANK TANDEM-COMPOUND HIGH-SPEED ENGINES.

¹ See Emmet, *Electrical World and Engineer*, vol. xxxv. p. 95. Trinks, *Proceedings Engineers' Society of Western Pennsylvania*, vol. xvii. p. 163. See also three Papers before the American Institute Electrical Engineers, 25th October, 1901; and H. C. Leake, *Electrical Review*, vol. 1. p. 543.

the use of less sensitive springs, or, where close regulation is essential, of a properly designed dash-pot, will generally overcome the effect of the latter. Any further trouble can usually be avoided by the use of conductors in or between the pole-pieces, or of an *amortisseur* of the Leblanc or other kind; but the adoption of these, or of the more complex devices employed by some continental makers, tends to lower the efficiency of the machine.¹

A few years ago it was usual to require in engines for electric lighting a much closer speed-regulation than was really necessary; and the extremely sensitive governor-springs used for attaining this end produced hunting, which was often attributed to the uneven turning-moment of the engine. That the latter is seldom the cause of hunting in high-speed machinery can be shown by cutting out the governor and regulating on the stop-valve: in this way it is generally possible to obtain excellent parallel running with a variety of high-speed engines and alternators, under various conditions of load and field-excitation.

For any given machine, however, there is a maximum² angular displacement compatible with good parallel running, and the angle is greater with high-speed than with low-speed engines,³ on account of the greater number of poles in the latter. The need for a heavy fly-wheel arises as much from the necessity of controlling the angular displacement as from the desire for close speed-regulation under variable load.

The actual fly-wheel effect required in various kinds of engines to keep the speed-variation (with normal cut-off) within $\frac{1}{300}$ of the mean during each revolution is given in the Appendix, Table X.⁴ With the same factor of safety, the actual weight of the fly-wheel is naturally much less in the high-speed engine; enabling cast iron to be used without danger,⁵ owing to the disk form of construction. In low-speed engines disastrous experience

¹ See Meyer, *Electrical Review*, vol. xlix. p. 5.

² For rotary converters Hobart gives 2.5°, and Eborall 2°.

³ In America the specified variation of crank-effort is usually dependent on the frequency; in German practice usually no notice is taken of the latter.

⁴ The Author has obtained confirmation of some of these figures by using an oscillograph. The results of the experiments will be published shortly.

⁵ The calculated bursting speed for the fly-wheels of the engines at the Waterloo and City Railway was 2,020 revolutions or 5.32 times normal speed, while the maximum variation of speed under all loads was $\frac{1}{2}$ per cent. See also C. H. Benjamin, "The Bursting of Small Cast-Iron Fly-Wheels," *Feilden's Magazine*, vol. vii. p. 198.

has led to the adoption of built-up and steel wheels at a considerable increase of cost.¹

EFFICIENCY.

Mechanical Efficiency of Engines.—Rankine's remark, made many years ago—that the existing knowledge of the amount of energy lost by the internal friction of engines is vague and indefinite,² still holds good with regard to low-speed engines, in spite of all the careful work done since that time. As a result of the increased use of electrical machinery, however, and of the publication of the Willans investigations, there arose a demand for greater accuracy in engine-testing. Engine-makers found it necessary to ascertain at least the steam-consumption per indicated horse-power, instead of the tons of fuel used per annum, which seems for many years to have been the favourite form of guarantee with the builders of mill- and pumping-engines. In high-speed engines the indicator has been supplemented by the brake³ (Appendix, Table III.), and the results obtained by its use have added considerably to knowledge on the subject.

From the buyer's point of view, however, it is more satisfactory to specify the steam-consumption per kilowatt-hour, which can be determined with considerably greater accuracy, leaving the efficiency of the engine to be determined by the builder. Owing to the difficulty of making brake tests,⁴ the builder has usually assumed the friction-losses shown on the no-load diagram to be the same at all loads, and some of the results shown in Table I. of the Appendix are calculated on this assumption; but, as might be expected, this is not always strictly correct, and the results so obtained usually indicate a higher efficiency than is obtained from the combined test.

In *Figs. 5 and 6* are shown the results of some investigations

¹ Partly in order to avoid the increased cost, in one of the latest American stations (Manhattan Electric Railway), the amount of fly-wheel effect required has been reduced by adopting horizontal and vertical cylinders in the same engine, with a result somewhat similar to that of four cylinders in marine engines. Some engines of marine type are used at Wallsend to drive alternators, from which synchronous motors are worked without trouble.

² "A Manual of the Steam-Engine, etc.," 2nd edition, p. 423. London, 1861.

³ For a description of the Froude dynamometer see *Proceedings Institution Mechanical Engineers*, 1877, p. 237; also Weighton, *Transactions North-East Coast Institute of Engineers and Shipbuilders*, vol. xiii. p. 79.

⁴ The Author is informed by Mr. Carl Sulzer that a series of brake tests on low-speed engines are being made at Winterthur.

upon this point, which seem to indicate an increase of friction with increase of load in high-speed three-crank forced-lubrication engines. Both Mr. E. Reynolds, of Milwaukee, and Mr. C. Sulzer, however, have observed reliable instances of diminished friction with increased load; whereas Dr. Thurston usually found a slight increase, confirming the views of the Count de Pambour. Professor R. C. Carpenter¹ (who made tests at speeds up to 912 revolutions per

Fig. 5.

HIGH-SPEED FORCED-LUBRICATION ENGINE. THREE CRANKS, TRIPLE EXPANSION, 1,000 B.H.P.

minute) and others have found the piston-ring friction² to increase slowly with the mean steam-pressure, and hence with a later cut-off; but otherwise to be practically independent of the load.

From a large number of tests, either made by the Author or

¹ Transactions American Society Mechanical Engineers, vol. x. p. 110.

² As Clausius pointed out, the piston-ring friction does not materially affect the final economy, as the heat it generates is returned to the steam.

published¹ from time to time, it would appear that the average efficiency of low-speed engines may be taken as 88 to 91 per cent.² excluding the air-pump.

Brake tests of ordinary single-acting high-speed tandem engines

Fig. 6.

HIGH-SPEED THREE-CRANK FORCED-LUBRICATION ENGINE, 200 B.H.P.

show losses of 10 to 11 per cent. when stiff (Appendix, Table III.); but in the Author's experience the losses may be reduced by as much as 20 per cent. after some months' working.

¹ Thurston, "The Steam Engine at the end of the 19th Century." Transactions American Society Mechanical Engineers, vol. xxi. p. 181; also *Engineering*, vols. lxxix., lxxx., and lxxxi. Paris Exhibition.

² The higher efficiencies often guaranteed are usually based on the assumption of constant friction-losses. Sulzer and Leavitt have given results as high as 93 per cent. in special cases, but the methods of testing are not stated. Still higher results (up to 96 per cent.) have recently been given in connection with the Glasgow tramway engines.

A somewhat higher initial efficiency and, consequently, a smaller improvement, is obtained from double-acting forced-lubrication engines having two and three cylinders, which when new frequently give efficiencies of 92 to 94 per cent. (Appendix, Table III.). A Belliss engine at Sunderland, tested by Dr. Kennedy, showed an efficiency of 96·3 per cent., and an even more remarkable efficiency at light load, namely, 89·3 per cent.¹ That this is not a solitary instance can be seen by reference to the Appendix, Table XIV., where the low friction in engines working with forced lubrication is shown to be remarkable.²

The results of efficiency tests made by various methods are shown in the Appendix, Tables IX. From the curves in *Fig. 7* it will be

Fig. 7.

(The numbers of the engines refer to Table I.)

seen that the mechanical efficiency of the high-speed engine at light loads is very high—a most important factor for variable-load working.³

Efficiency of Dynamos and Alternators.—Increase of speed in dynamos gives greatly improved ventilation, an increase of 50 per cent. in the speed sometimes resulting in a decrease of temperature of as much as 25 per cent. This does not apply to the commutator, in which the friction and the contact-resistance increase more

¹ Proceedings Institution Mechanical Engineers, 1897, p. 326.

² See also Tables I. and III., Appendix.

³ In the case of superheat there appears to be a slight increase in friction the amount of which, however, has not been actually determined.

rapidly than the ventilation—in fact, almost as the square of the speed.¹ The increase in contact-resistance is in favour of freedom from sparking,² but augments the resistance-losses by between 1 per cent. and 3 per cent. The losses by journal-friction and windage are increased, but not proportionately, on account of the reduced diameters; and the total mechanical losses are generally about 5 per cent. more with high speed. The hysteresis and eddy-current losses also increase; but, chiefly on account of the improved ventilation, the average commercial efficiency of the high-speed dynamo may be $1\frac{1}{2}$ to 2 per cent. higher for the same cost and output.

In alternators, in addition to the improved ventilation there is an increase of efficiency due to the smaller number and greater size of the poles, enabling the copper and iron to be used to greater advantage; and an increase of fully 2 per cent. in the efficiency can be obtained for the same output and cost, although as a rule the cost is considerably reduced (Appendix, Table XII.).

Steam-Consumption.—The effect of higher speed upon the steam-consumption of the single-acting central-valve engine, apart from its mechanical efficiency, was fully investigated by Mr. Willans; and the same general conclusions probably hold good for engines of other kinds.

The results of tests made upon double-acting engines seem to show that initial condensation due to contact with the cylinder-walls—and hence also the benefit derived from jacketing—are small (Appendix, Tables IV. B and IX.). But the effects of water lodged in the cylinder, and consequently the advantages due to superheated steam, are considerable, possibly greater in high-speed than in low-speed engines (Appendix, Table VI.); hence the use³ of internally-polished covers with high speed is hardly justified. Mr. A. L. Rice⁴ and Dr. Thurston found, however, that in low-speed engines the initial condensation increased 40 per cent. with rough covers.

Several causes combine to prevent the economy of high-speed

¹ See Parshall and Hobart, "Electric Generators." London, 1900.

² In some German dynamos a resistance is inserted between the segments of the commutator and the armature-bars.

³ Even although the area of the cover forms a greater proportion of the total surface than in the low-speed engine. See also Callendar and Nicolson, "Law of Condensation of Steam." Minutes of Proceedings Inst. C.E., vol. cxxxi. p. 147.

⁴ Rice, "Cylinder Condensation." Transactions American Society Mechanical Engineers, vol. xviii. p. 950; also vol. viii. p. 479.

clearance-space does not form so large a portion of the total steam as with the earlier cut-off usual in the low-speed engine. Partly on this account, the loss of economy due to clearance is not so great as might be expected—as may be seen from the Appendix, Table V. B, which shows the increase in steam-consumption due to an intentional increase of clearance, effected by raising the cylinder-covers. *Figs. 8* show indicator-diagrams taken in this trial. Partial confirmation¹ of this result is found in the small effect produced by the large clearance in the high-pressure cylinders of triple-expansion engines. The worst effect of large port-clearance is the increased surface offered for condensation.

Superheated Steam.—The economy due to the use of superheated steam appears to be considerably greater in high-speed than in low-speed engines, partly on account of the reduction of leakage past the piston-valves.² The results of some experiments made with a view to test the latter point are given in the Appendix, Table VII.; but it is not certain whether the result is due wholly to the expansion of the metal itself, or to some more complex cause. The small amount of superheat, or even mere drying, which results from the use of the throttle-governor, apparently gives greater economy than is obtained by the use of steam at boiler pressure.

The improvement in economy due to the use of moderate superheat in low- and high-speed engines is shown in the Appendix, Table VI.; but further experience is required before the high degrees of superheat (200° F. to 300° F.) now becoming common³ with low-speed engines can be used with high-speed. The trouble with superheated steam arises apparently from the *variation* of temperature, rather than from the actual maximum temperature; and it may sometimes be advisable to use an automatic by-pass for admitting saturated steam into the steam-chest when the superheat exceeds a certain amount. In general, the use of highly superheated steam should be avoided until the cylinders are worn to a good surface.

¹ See also Ball, Transactions American Society Mechanical Engineers, vol. xiv. p. 1067, who shows that with slow compression the effect of any clearance in compound engines up to 10 per cent. is inappreciable. In certain typical cases (*e.g.*, "Universal" Engine) it is necessary to increase the clearance in order to avoid excessive cushioning.

² Minutes of Proceedings Inst. C.E., vol. cxxviii. p. 62.

³ See Lenke on "Superheated Steam in Engines," Proceedings Institution Mechanical Engineers, 1901, p. 847. Also Patchell, "Notes on Steam Superheating," *Ibid*, 1896, p. 134.

It might be expected that the foregoing results would, to a certain extent, apply to the second and third stages of the expansion; but the results of a number of experiments made by the Author, with reheaters fitted to the intermediate-pressure receivers, have not so far shown sufficient economy to justify the increased expense.¹ It would appear that the advantages of reheating are greater with low-speed than with high-speed engines, on account of the greater volume of receiver-steam and the better facilities for cooling which exist in low-speed engines. But where a degree of superheat in excess of that desirable for use in the high-pressure cylinder is obtainable, it is sometimes advantageous to allow the steam to pass through the high-pressure jacket on its way to the high-pressure cylinder.²

Jackets are not usually required where superheated steam is employed; but the subject is one upon which there is still considerable difference of opinion among designers.

Jacketing.—Some makers of low-speed engines (*e.g.*, Messrs. Escher, Wyss & Co.) jacket all three cylinders in triple-expansion engines; others jacket the low-pressure cylinders only, considering that it is better to supply the additional heat required in the high-pressure cylinder by superheating the steam.

In a number of instances the failure to obtain the results expected from jacketing is probably due to the use of an insufficient quantity of steam in the jackets, or to the cylinder-covers being unjacketed, which is a serious source of loss. Sulzer and Leavitt have obtained some of their best results when using in the jackets 17 to 20 per cent. of the total steam; but tests are often published showing less than 5 per cent. of the total steam condensed in the jackets, which are fitted with steam-traps. Experience has shown, however, that the latter do not give that steady flow of steam through the jacket which should take place, and which is desirable, even if the steam has to be used for auxiliary machinery; and the use of some other device for regulating the flow of steam is advisable.

On the whole, the majority of the results obtained hitherto show

¹ The actual figures obtained hitherto, however, are not sufficiently conclusive to be published. Some results recently obtained by Messrs. Browett, Lindley & Co. tend to confirm the above conclusions.

² See description of Schmidt system, Minutes of Proceedings Inst. C.E., vol. cxxviii, p. 61.

that the steam-jacket has but little effect¹ on the economy of high-speed engines. A series of tests made by the Author in 1893 upon a high-speed triple-expansion engine at the Globe Works, Lincoln, showed an average gain in economy of less than 2 per cent.; and in the tests, made upon a Belliss engine, No. 1,003 (Appendix, Table IV. A), no gain at all was found. This is probably due to the fact that there is not sufficient time for the proper transfer of heat through the cylinder-liner. The air-space of the jacket, however, if kept dry, forms an excellent lagging for the cylinder; and on starting, the jacket-steam is more effective in warming the cylinder than is live steam admitted into it. These advantages are generally sufficient to justify the extra expense incurred in fitting the necessary steam-pipe, inasmuch as it is common in this country to fit liners to most of the cylinders, at least in horizontal and high-speed engines.

Condensing.—In addition to the economies effected by the foregoing means, the use of a condenser with high-speed engines has become common in England, although practically unknown in the United States. The advantages thereby gained are perhaps not quite so great as with low-speed engines, in which it is not uncommon to obtain in the low-pressure cylinder a vacuum only $\frac{1}{2}$ inch less than in the condenser itself; in high-speed engines the difference is more usually 2 to 3 inches, partly in consequence of the distance between the engine and the condensing-plant. The necessity for the separation of the two is chiefly due to the difficulty of working air-pumps at high speed; and, although in marine engines air-pumps are often worked at 400 revolutions per minute, it is desirable, for continuous running, not to exceed 200 revolutions to 230 revolutions per minute. Where, as in large units, the engine does not exceed this speed, there seems to be no objection to direct driving.

In comparing the capital and annual cost of high-speed and low-speed engines, it must be remembered that the latter usually work their own air-pumps, and the cost of the necessary power for driving these must be taken into account.²

¹ Even with low-speed engines the advantages of jacketing are not always evident. A careful test recently made at Wallsend on a 1,000-H.P. engine (100 revolutions per minute) showed practically no difference between the economy of jacketed and unjacketed cylinders (Appendix, Table IV. B), possibly because of low percentage of jacket-steam. Professor Weighton's experiments at Newcastle showed but little benefit from jacketing.

² As has been done in Table XV. (Appendix). From a trial made by the Author the power required for driving the condensing auxiliaries for a 300-kilowatt set was about 10 per cent., divided as follows:—4 per cent. to the Edwards three-throw air-pump, 4 per cent. to the circulating-pump, and 2 per cent. to the fan motor for the cooling-tower. Both the pumps were driven electrically.

A more important difficulty at high speeds is the reduction of

Figs. 9.

SINGLE-ACTING HIGH-SPEED CENTRAL-VALVE ENGINES.

The upper thick dotted line in each diagram gives the consumption with the fixed expansion set so as to give the overload without using the by-pass valves.

vacuum in the cylinder itself, due to the difficulty of exhausting with sufficient rapidity through ports of normal size, because an

early closing of the exhaust ports is required in order to produce the amount of cushioning necessary for quiet running.¹

Expansion-Gear.—The relative economy of throttle and expansion-governing has been the subject of much discussion. With respect to high-speed engines, the question has been fully dealt with by Captain Sankey,² who found that, below a certain load, the advantage lay with the throttle-governor. The latter is quite suitable for the steady loads which occur in electric lighting; but for handling economically the overloads of traction-work, expansion-governing has usually been considered necessary. There is not

Fig. 10.

so much difference as is supposed, however, between the economy of the two methods (*Figs. 9*).

Where throttle-governing is in use, two methods of dealing with the overloads are possible. One is the use of a by-pass to admit high-pressure steam into the low-pressure cylinder; or, in triple-expansion engines, into the intermediate-pressure cylinder: the other method is to give the high-pressure cylinder a sufficiently late cut-off to enable it to take the overload—in other words, to reduce the economy slightly, and increase the size of the engine beyond that necessary for the normal full load.

¹ For example, a reduction of vacuum of 6 inches (from 26 inches to 20 inches) in an 800-I.H.P. triple-expansion high-speed engine gave a loss of 66 HP. in the low-pressure cylinder, or nearly 14 per cent. of the total.

² Proceedings Institution Mechanical Engineers, 1895, p. 161.

The results of some overload tests made upon engines with steady loads of different values are given in *Figs. 10-13*; but the comparative economy in actual practice must depend upon the nature of the load, and is probably different with steady and with variable loads.¹

From the results of some tests made upon a Belliss engine, fitted first with a throttle-governor and then with a combined expansion- and throttle-governor, it would appear that the latter has a certain advantage, although the high mechanical efficiency results in a

Fig. 11.

450-B.H.P. TRIPLE-EXPANSION HIGH-SPEED FORCED-LUBRICATION ENGINE.

very economical performance of the throttle-governed engines at light loads² (*Figs. 11, 12, and 13*).

Triple- versus Double-expansion Engines.—Another debatable subject is the relative merit of triple and of double expansion. Until comparatively recently the triple-expansion engine was confined to marine and pumping work; and even at the present time few

¹ Further experiments on this important point are being made by the Author, with high- and low-speed engines on tramway loads. At present the results obtained point to a difference of about 2 lbs. of steam per kilowatt-hour between steady and variable loads. (See Appendix, Table III. D.)

² Doubtless due partly to the drying of the steam.

builders of low-speed stationary engines recommend them for other purposes. The great majority of electric power-houses in America¹ use cross compound vertical or horizontal engines, which have been adopted in consequence largely of the compact arrangement obtained thereby. It is also claimed by the makers that, except in stations having a particularly high load-factor (*e.g.*, pumping-stations), the extra cost, repairs, and friction-losses in triple-expansion engines practically counterbalance their small saving in steam-consumption over compound engines, especially where superheated steam is used. Accurate figures relative to this point

Fig. 12.

TRIPLE-EXPANSION HIGH-SPEED FORCED-LUBRICATION ENGINE.

are much needed; but from a number of sources the Author has gathered that the gain due to the use of the third cylinder on low-speed engines, where the steam-pressure and vacuum are the same, averages at full load not more than 1 lb. per indicated horse-power, and somewhat less per brake horse-power, and is practically nothing at light loads.

In England and in Germany, where steam-economy has perhaps been studied more than capital cost, triple expansion is used

¹ Of forty power-houses recently visited by the Author only two had triple-expansion engines.

rather more frequently; but the majority of builders of low-speed engines still recommend compound engines for electrical work. In power-houses having a reasonably high load-factor, however, there is a good deal in favour of triple-expansion engines, either of the medium-speed type used at Sheffield and other places, or of the marine type, as at Wallsend: the economy of the latter is particularly high (Appendix, Table IX. D).

Fig. 13.

700 B.H.P. TRIPLE-EXPANSION HIGH-SPEED FORCED-LUBRICATION ENGINE.

With regard to high speed, Mr. Willans found little gain in using triple expansion at pressures lower than 160 lbs. per square inch in condensing engines; and in general the results obtained with double-acting engines seem to bear this out. At or above this pressure the use of the third cylinder is attended with a considerable gain in economy (Appendix, Table I.). This is due not only to the lower temperature-range per cylinder, but also to the fact that the clearance- and leakage-steam which escapes the second

cylinder does work in the third; while the smaller difference of pressure between the opposite sides of the valves and pistons itself reduces the leakage.

It will also be noted that in high-speed double-acting side-by-side engines there is practically no difference between the mechanical efficiencies of double and triple expansion, even at light loads and only about 1 per cent. in single-acting tandem engines. As a result, the light-load economy is relatively considerably higher than in triple-expansion low-speed engines.

It is sometimes stated that the governing of triple-expansion engines is less satisfactory than that of compound engines. But apparently this statement does not apply when the speed is high (Appendix, Table VIII. A.): then there is but little to choose

Fig. 14.

between the governing of the two classes, because any slight differences which exist in the variation of the crank-effort are quite inappreciable at high speeds.

Valves.—In addition to the question of expansion-gear, the use of piston-valves is often urged as an argument against high-speed engines. It is certainly impossible to obtain as rapid a cut-off with a piston-valve of reasonable diameter as with a drop-valve, unless some special form of cam be used; but too much importance may be attached to the appearance of the indicator-diagram, which does not seem to have so much bearing upon the economy as is often supposed. The combination diagrams shown in *Fig. 14* prove that a fair expansion-curve may be obtained with piston-valves working at high speed.

Combined Efficiency.—Although the kind of engines to be adopted

in any particular instance must be influenced by the points already mentioned, the consulting engineer is more interested in the steam-consumption of the combined set in actual practice. The results of a number of recent trials are given in the Appendix (Tables I. and IX.). The majority of the tests have been made at makers' works; but several of them have been repeated on the site, in some instances after many months' service, and a fair agreement of results has been shown.

In those tests which have been carried out by the Author, whether on site or elsewhere, every effort has been made to ensure accuracy; the indicators have been tested from time to time, and the thermometers, wattmeters, and other instruments have been calibrated by an independent authority after each test, and the necessary corrections made. In the majority of the trials the condensers have been tested for leakage. The detailed results of the trials not conducted by the Author have in general been examined by him; and it is believed that the figures given in the Appendix, Table I., may be taken as fairly representative of results which can be obtained in modern practice.

Although the results obtained at full load are in favour of low speed, it will be noticed that, contrary to expectation, the high-speed engine, in spite of its throttle-governor, more than holds its own at the lighter loads: which is a most important point for working under a variable load.

COST OF HIGH-SPEED ENGINES.

Cost of Maintenance.—It is often supposed that almost any accident to a high-speed engine must result in a sudden and complete breakdown, and cause more damage than a similar accident to a low-speed engine; but the experience of the Author and other users of high-speed machinery does not bear this out. As an example, a high-speed engine in London had two cylinder-covers broken by water, and the studs stripped: this, the only damage, was repaired within 24 hours, although new covers had to be obtained from Birmingham. At the Croydon Electricity Works several accidents have occurred during 5 years' working, but in every one there has been time to transfer the load to another engine before shutting down the damaged engine.

Fly-wheel accidents are practically unknown with the high-speed engine: which is a point of considerable importance. In fact, it is the experience of many engineers that the cost of

repairs with the modern high-speed engine is considerably less than with the low-speed engine, assuming that they receive the same amount of attention.

Judging from a number of stations visited by the Author, it would appear that the attendance required by high-speed engines averages 50 per cent. to 65 per cent. of that necessary for horizontal low-speed engines, and 60 per cent. to 75 per cent. of that necessary for vertical engines, where the latter have communicating platforms.

No argument is more frequently adduced in favour of low-speed engines than their long life and the few repairs necessary; and there can be little doubt that a beam-engine making twenty strokes per minute, with a piston-speed of 130 feet per minute, will usually last longer than the engine of a torpedo-boat-destroyer running at, say, 400 revolutions per minute, with a piston-speed of 1,200 feet per minute. In actual practice, however, the constant thrust of the single-acting engine, and the forced lubrication in the double-acting engine, reduce wear to a remarkable extent. Particulars of the actual wear of an engine after 4 years' work are given in the Appendix, Table XI.; and are fairly typical of well-kept engines, judging from a number of measurements made annually by the Author.

It has been impossible to obtain any sufficiently reliable figures from low-speed engines for enabling a good comparison to be made; but such results as have come to the Author's notice seem to show that the re-boring of cylinders and skinning of rods is at least as frequent in low-speed engines as in high-speed engines, and considerably more costly, especially where the cylinders are not provided with liners: the extra expense is due, of course, to increased weight of the various parts to be handled. Dr. Kennedy has informed the Author that the cost of the repairs to the Westminster Electric Supply Corporation's high-speed engines averages 1s. per horse-power-annum; at the Croydon works the average has been about 10 per cent. higher. Table XIII. shows that the difference in the cost of spares for the two types is considerable.

Capital Cost.—In order to make the comparison complete, the annual cost must include the interest on capital outlay; and some rough idea of the latter may be obtained from the particulars given in Table XV., which, however, are necessarily approximate only.

On account of the accurate workmanship required in high-speed machinery, the difference between its first cost and that of low-

speed machinery is not so great as might be expected, especially when the cost of air-pumps is taken into account.

For reasons already mentioned, high-speed dynamos also cost at the present time nearly as much as low-speed machines; but there is a considerable difference with alternators; and also a saving of 30 per cent. to 50 per cent. in foundations. By working at higher speed the saving in the cost of buildings is considerable, and is of the same order as that for foundations,¹ depending, of course, upon whether vertical or horizontal engines are used.

Fig. 15 is a plan of part of the engine-room at the Brighton Electricity

¹ For costs of foundations, see E. H. Sniffin on "The Commercial Aspect of Steam-Turbines." *Electrical World and Engineer*, vol. xl. p. 623.

Works, which is an excellent example of utilizing space with high-speed machinery.

Recently the Author had occasion to prepare estimates for a power-station fitted alternatively with high-speed or with low-speed machinery; and from the tenders and figures so obtained has been prepared Table XV., showing the comparative capital and annual cost under the two systems. The figures therein are based upon the results of working in a number of power-houses; and it is assumed for purposes of comparison—

1. That, in the district in which the power-house is situated, the quality of coal improves, and the consumption falls, with the price, in the ratio shown, the improvement being less marked with the longer hours of working.

2. That the relative economy of high-speed and low-speed plants with a 100 per cent. load-factor is 90 per cent. (*i.e.* the average difference in steam-consumption is about 2 lbs. per kilowatt-hour at full load), and that under practical working-conditions, with load-factors of 60 per cent., 35 per cent., and 12 per cent., the relative economy is 92 per cent., 93 per cent., and 95 per cent. for the 500-kilowatt sets, and 93 per cent., 94 per cent., and 95 per cent. for the 1,500-kilowatt sets, respectively.

3. That the small sets, both high- and low-speed, have a 5-per cent. higher steam-consumption than the larger sets.

With regard to the cost of working the two stations, the relative importance of the running-expenses and interest-charges depends largely upon the load-factor. Where the latter is fairly high, the actual economy of working is naturally of far greater importance than the capital outlay. For example, in Station No. 1 the interest-charges (engine-room only) would, with a load-factor of 12 per cent. (lighting), amount to 0.26*d.* per Board-of-Trade unit; but with a load-factor of 35 per cent. (tramway) would be only 0.089*d.* per unit; hence, the claim of the high-speed engine to be used for the latter class of work must here be based upon the lower cost of working.

In conclusion, it would appear that the difference in steam-consumption between the most recent forms of high- and low-speed continuous-current machinery is about 2 lbs. to 3 lbs. per kilowatt-hour at full load for condensing engines with 180 lbs. per square inch steam-pressure, and is somewhat less for alternating-current machinery. At lighter loads, however, the economy of the high-speed engine, even when it is throttle-governed and especially when it is fitted with expansion-gear, would seem to be quite as high as that of the low-speed engine. Hence, contrary

to the usual belief, high-speed machinery is quite as suitable for traction as for lighting, especially as excellent governing is easily obtainable. The latter fact, together with the increased efficiency of high-speed alternators, also fits the high-speed engine for electrical transmission of power.

In addition to the advantages already mentioned, which are capable of being calculated, high-speed machinery possesses several others, the result of small floor-space, crane requirements, etc. When these are considered in conjunction with the results given in the Appendix, Table I., it would seem that such machinery is deserving of more attention for power-stations than it has received hitherto.

The Author desires to express his indebtedness to the members of his staff for their co-operation, and to those engineers who have assisted him in preparing this Paper by placing at his disposal much useful information.

The Paper is accompanied by twenty-five drawings, from a selection of which the Figures in the text have been prepared; and by the following Appendix.

[APPENDIX.

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[THE INST. C.E. VOL. CLI.]

APPENDIX
TABLE I. A.—SPEED

Type of Engine.	B.H.P.		Steam-Pressure (Lbs. per Square Inch). Revolutions per Minute.		Nature of Valve- Gear.	Percentage of Mechanical Efficiency.				
	Econo- mical.	Maxi- mum.				Max. Load.	Full Load.	$\frac{3}{4}$ Load.	$\frac{1}{2}$ Load.	$\frac{1}{4}$ Load.
Double-acting high-speed triple-expansion	1,000	1,200	170	280	Piston	..	94.0	92.0	90.0	..
Ditto	750	940	160	300	..	..	93.5	92.8	92.7	88.8
Ditto	750	940	180	300	..	..	94.0 <i>e</i>	..	90.1 <i>e</i>	..
Ditto	430	..	180	360	..	93.5	93.6	90.1 <i>e</i>	89.3	80.0 <i>e</i>
Ditto	430	..	165	360	..	..	83.2 <i>e</i>	80.1 <i>e</i>	70.85 <i>e</i>	..
Ditto	300	..	160	420	..	..	..	..	..	..
Double-acting high-speed six- cylinder compound	375	..	160	360	..	..	95.2	90.4	..	84.2
Single-acting high-speed triple	1,000	..	..	..	Central piston	..	..	..	..	..
Reavell compound	..	..	150	400	Central	..	..	..	..	..
English vertical cross compound	1,000	1,300	180	214	Grid slide	..	94.0	92.0	89.0	..
American vertical cross compound	1,000	..	180	90	Triple Corliss	..	91.0	..	..	..
Average English low speed cross compound	1,000	..	180	$\left\{ \begin{smallmatrix} 90 \\ \text{to} \\ 100 \end{smallmatrix} \right\}$	Corliss	..	92.5	..	..	..
American horizontal cross compound	800	..	160	..	Triple Corliss	..	..	..	..	..
English triple expansion	1,000	..	180	90	..	..	85.5	..	..	..
Ditto	..	..	..	..	Piston on H.P., slide on I.P. and L.P.	..	..	92.0	..	..
Sulzer	3,500	4,000	170	..	Drop	..	91.0	89.0	89.0	85.0
Marine type triple-expansion	1,000 I.H.P.	..	192	102.3	..	..	..	..	..	..

Figures in italics are makers' guarantees. Figures marked "e" are calculated

D I X.

AND EFFICIENCIES.

Steam-Consumption. Lbs. per Brake Horse-power-Hour.					Generator.	Volts.	E.H.P. × 100. I.H.P.	Lbs. Steam per Kilowatt-Hour.	Output Kilowatts.	Date of Trial.
Authority.	Full Load.	$\frac{3}{4}$ Load.	$\frac{1}{2}$ Load.	$\frac{1}{4}$ Load.						
..	14.5	15.0	16.0 ¹		Continuous	500				31/10/01
{ No. 1,003 tested by Author	16.08	16.76	17.35	18.84	Alternating	2,118	..	..	557.00	4/10/01
{ No. 1,233 tested at B. and M.	..	..	..	..	..	2,280	87.45	21.21	512.00	19/12/01
{ No. 1,245 tested at B. and M.	15.25	17.11	18.27	19.43	Continuous	560	88.75	22.47	300.00	9/5/01
{ No. 1,246 tested at B. and M.	..	..	..	..	..	526	79.04	22.41	300.00	5/4/01
{ No. 1,284 tested at B. and M.	..	..	..	..	..	250	85.91	24.8	195.00	14/6/01
..	19.2	20.19	..	21.24						
..	..	..	..	..	..	..	..	{ Full Load. 21 22.75		
{ Reavell & Co. }	..	..	..	..	Continuous	500	86.25	27.2	100.00	18/8/01
..	14.5	14.0								
..	15.4									
..	14.7									
{ Tested at City Road Station. }	..	..	..	..	..	..	..	On test. 18.46	186.16	11/10/01
..	14.7	..	..	..	..	..	..	21.2		
..	14.9									
{ Engineering, vol. lxx. p. 557 }	..	9.4 ²	..	..	Alternating	6,000				
..	{ 10.4 per I.H.P. }	..	..	..	..	6,000	79.3	17.6	700.00	6/10/01

from combined tests.

¹ With 20° F. superheat.

² Per I.H.P.-hour.

N 2

TABLE I. B.—PROPORTION OF HIGH-SPEED TO LOW-SPEED PLANT,¹ 1901–1902.

Lighting.	High-Speed. I.H.P.	Percentage of Total.	Low-Speed. I.H.P.	Percentage of Total.	Turbines. I.H.P.	Percentage of Total.	Totals I.H.P.
In operation, Provinces	228,309	70·3	78,050	24·0	18,370	5·7	324,729
“ London	103,982	74·5	35,560	25·5	..	..	139,542
“ Scotland and Ireland	49,407	90·7	5,060	9·25	110	0·05	54,577
Total in operation	381,698	73·6	118,670	22·9	18,480	3·5	518,848
In progress, Provinces and Scotland	8,495	40·3	12,625	59·7	..	..	21,120
“ London	5,000	100·0	..	..	..	..	5,000
Total in progress	13,495	51·8	12,625	48·2	..	..	26,120
Grand total	395,193	72·6	131,295	24·0	18,840	3·4	544,968
Traction.	High-Speed. I.H.P.	Percentage of Total.	Low-Speed. I.H.P.	Percentage of Total.	Turbines. I.H.P.	Percentage of Total.	Totals I.H.P.
In operation	53,968	36·8	91,140	62·0	1,800	1·2	146,908
“ progress	8,480	26·1	23,725	72·83	350	1·07	32,555
Grand total	62,448	31·8	114,865	64·1	2,150	1·1	179,463

¹ Summarized from Tables of Statistics published by *The Electrician*.

TABLE II. A.—PERIPHERAL SPEED OF COMMUTATORS.

Type of Dynamo.	Revolutions per Minute.	Output. Kilowatts.	Peripheral Speed. Feet per Min.	Volts.	Volts per Bar.
Multipolar Railway Generator, 550 volts (a)	360	300	3,200	550	
Multipolar Railway Generator, 550 volts (b)	280	600	3,050	550	
Lighting Generator, 160 volts (a) . . .	360	100	2,190	160	10·0
Lighting Generator, 160 volts (b) . . .	900	30	3,450	160	
Three-Phase Synchro- nous Motor-Genera- tor	240	450	2,480	650	
Two-Phase Synchro- nous Motor-Genera- tor	212	500	2,450	230	
Three-Phase Rotary Converter	250	500	2,580	550	
Three-Phase Asyn- chronous Motor- Generator (a) . . .	400	150	1,760	510	18·9
Three-Phase Asyn- chronous Motor- Generator (b) . . .	300	500	1,880	"	"
Three-Phase Synchro- nous Motor-Genera- tor (a)	480	150	1,760	"	"
Three-Phase Synchro- nous Motor-Genera- tor (b)	300	500	1,880	"	"
Three-Phase Rotary Converter (a) . . .	480	150	2,970	"	12·7
Three-Phase Rotary Converter (b) . . .	300	500	3,100	"	14·1
Rotary Converter (a) .	300	500	"		
" " (b) .	250	750	"		
" " (c) .	250	1,000	"		
" " (d) .	215	1,500	"		
Railway Generator (a)	75	1,500	1,685	600	10·3
Railway Generator (b)	70	2,700	1,920	"	12·1
Rotary Converter . .	250	900	3,540	"	

Average
values.

TABLE II. B.—SPEED OF CONTINUOUS-CURRENT MACHINES, SPECIFIED AS MAXIMUM BY VARIOUS MAKERS.

Firm.	1,000 B.H.P.	1,500 B.H.P.	1,715 B.H.P.	2,000 B.H.P.	Enquiry.	Horse-power.	Specified. Revolutions per Minute.	Belliss Standard. Revolutions per Minute.
Number 1	..	..	..	180	Number 1	1,200-1,500	180	290
"	..	150	..	..	"	1,100	230	300
"	..	120-140	95-120	90-115	"	700-800	280	330
"	300	230	..	200	"	720	270	330
"	230	200	..	200	"	730-1,460	100	290
"	280	240	..	200	"	750	270	330
"	275	200	..	150	"	650-800	280	330
Belliss standard engine speed	300	275	260	240	"	700-1,000	250	330

TABLE III. A.—BRAKE TEST OF DOUBLE-ACTING HIGH-SPEED ENGINE.
800 I.H.P. Triple-Expansion Engine, tested at Birmingham, 4 January, 1901.

Time.	Steam-Pressure. Lbs. per Square Inch.			Vacuum at Engine. Inches.	Revolutions per Minute.	Brake-H.P.	Water-Consumption.			Diagrams Taken.		B.H.P.	Efficiency. B.H.P. I.H.P. Per Cent.
	At Boiler.	At Engine.	L.P.				Time.	Lbs. per Hour.	Lbs. per B.H.P.-Hour.	Time.	I.H.P.		
$\frac{1}{2}$ Load.													
10.0	155	28	5	25.75	300	188.1	..	..	..	..	..	..	..
10.15	155	28	5	25.75	300	188.1	..	..	..	..	..	..	..
10.30	166	25	6	26.0	300	188.1	..	..	..	..	..	..	..
10.45	162	30	6	26.2	300	188.1	..	..	..	..	..	..	..
11.0	165	30	6	26.2	300	188.1	10.53 to 11.34	3,544	18.84	..	..	..	..
11.15	162	30	6	26.0	300	188.1	..	..	..	11.30	211.65	188.1	88.8
11.30	165	30	5	26.4	300	188.1	..	..	..	11.45	211.50	188.1	88.9
11.45	164	30	6	26.3	300	188.1	..	..	..	..	..	..	..
$\frac{1}{2}$ Load.													
12.15	154	63	17	26.0	300	375.0	12.26 to 12.48	6,506	17.35	..	..	..	..
12.30	163	62	17	26.0	300	375.0	..	..	..	12.45	400.13	375.0	93.7
12.45	162	62	17	26.0	300	375.0	..	..	..	..	..	..	..

TABLE III. A.—continued.

Time.	Steam-Pressure. Lbs. per Square Inch.			Vacuum at Engine. Inches.	Revolutions per Minute.	Brake-H.P.	Water-Consumption.		Diagrams Taken.		B.H.P.	Efficiency. B.H.P. I.H.P. Per Cent.
	At Boiler.	H.P.	At Engine.				Time.	Lbs. per Hour.	Time.	I.H.P.		
<i>½ Load.</i>												
2.30	159	95	32	1.5	300	564.0	..	..	..	..	..	..
2.45	165	95	32	1.5	300	564.0	..	..	2.45	607.76	564.0	92.8
3.0	162	96	32	1.5	300	564.0	2.58	..	3.0	611.65	564.0	92.2
3.15	160	96	32	1.5	300	564.0	2.58 to 3.13	9,452	3.15	600.96	564.0	93.8
<i>Full Load.</i>												
3.30	164	129	48	9.0	300	750.0	..	..	3.30	808.37	750.0	92.8
4.45	162	127	47	9.0	300	750.0	..	..	..	..	..	..
4.0	163	126	46	9.0	300	750.0	..	..	..	..	..	..
4.15	158	129	47	9.0	300	750.0	..	..	4.15	795.64	750.0	94.2
4.30	160	128	46	9.0	300	750.0	..	..	4.30	792.16	750.0	94.6
4.45	163	129	46	9.0	300	750.0	4.22 to 4.40	11,867	..	..	..	..
5.0	160	128	47	9.0	300	750.0	..	..	..	..	..	..
5.15	156	127	47	9.0	300	750.0	..	..	..	..	..	..
5.30	158	127	47	9.0	300	750.0	..	..	..	..	..	..
5.45	163	128	47	9.0	300	750.0	5.34 to 5.51	12,135	5.45	792.29	750.0	94.6
6.0	164	128	47	9.0	300	750.0	..	..	..	..	..	..

TABLE III. B.—BRAKE TESTS OF CENTRAL-VALVE ENGINES.

Type.	I.H.P.	B.H.P.	Efficiency.	Mean Pressure. Lbs. per Square In.
Compound	676·0	695·0	Per Cent. 89·5	46·2
Compound	107·5	96·36	89·65	37·58

C.—BRAKE TEST OF DOUBLE-ACTING HIGH-SPEED ENGINE.

1,000 B.H.P. Triple-Expansion Engine tested at Birmingham, 31 October, 1901.

Full Load, Condensing.

Time.	Steam-Pressure. Lbs. per Square Inch.				Vacuum at Condenser.	Revolutions per Minute.	B.H.P.
	At Boiler.	At Engine.					
6.45	170	H.P. 150	I.P. 40	L.P. 6·0	Inches. 25·5	280	969·7
7.0	175	162	45	9·5	26·3	280	„
7.15	174	158	47	10·5	26·2	280	„
7.30	170	156	46	10·0	26·2	280	„
7.45	190	159	46	9·5	26·3	280	„
8.0	195	162	45	9·5	26·2	280	„
8.15	170	159	47	10·0	26·5	280	„
8.30	175	159	47	10·0	26·4	280	„

Water-Consumption.

Tank No.	Time.	Per Hour.	Per B.H.P.-Hour.	Diagrams taken.		B.H.P.	Efficiency B.H.P. I.H.P.
				Time.	I.H.P.		
1	7.31 to 7.53	14,629	15·08	7.40	1036·1	969·7	Per Cent. 93·6
				7.45	1036·0	969·7	93·6
2	8.8 to 8.31	14,682	15·14	8.15	1022·0	969·7	94·8
				8.30	1031·0	969·7	94·0

Mean temperature of steam during steam-consumption trial—

Tank No. 1 mean temperature = 418° F.

„ No. 2 „ „ = 419° F.

TABLE III. C.—*continued.**Three-quarter Load, Condensing.*

Time.	Steam-Pressure. Lbs. per Square Inch.				Vacuum at Condenser.	Revolutions per Minute.	B.H.P.
	At Boiler.	At Engine.					
		H.P.	I.P.	L.P.	Inches.		
8.45	195	121	30	4.0	25.9	280	727.5
9.0	180	123	30	3.5	25.8	280	"
9.15	182	124	31	3.5	25.7	280	"
9.35	185	120	30	3.0	25.8	280	"
9.45	165	122	32	3.5	25.7	280	"
10.0	170	122	31	3.5	25.6	280	"
10.15	185	122	32	4.0	25.5	280	"

Water-Consumption.

Tank No.	Time.	Lbs. per B.H.P.-Hour.	Diagrams taken.		B.H.P.	Efficiency B.H.P. I.H.P.
			Time.	I.H.P.		
1	8.51 to 9.10	15.43	9.0	782.6	727.5	Per Cent. 92.97
			9.15	786.8	"	92.47
2	9.23 to 9.42	15.69	9.45	791.2	"	91.96
			10.0	789.9	"	92.1

Steam temperatures during steam-consumption trials—

Tank No. 1 mean temperature = 415° F.

" " 2 " " = 404° F.

Half Load, Condensing.

Time.	Steam-Pressure. Lbs. per Square Inch.			Vacuum at Condenser.	Revolutions per Minute.	
	At Boiler.	At Engine.				
		H.P.	I.P.	L.P.	Inches.	
2.30	160	84	19	3.0	25.4	280
2.45	162	86	18	4.0	25.8	280
3.0	164	86	18	4.0	25.8	280
3.15	185	86	17	4.5	25.8	280
3.30	180	86	17	4.5	25.8	280
3.45	155	86	18	4.0	25.8	280
3.55	145	86	18	4.0	25.8	280
4.0	135	85	17	4.0	25.6	280
4.15	135	87	18	4.0	25.7	280

TABLE III. C.—*continued.*

Water-Consumption.

Time.	Lbs. per B.H.P.-Hour.	Diagrams taken.		B.H.P.	Efficiency B.H.P. I.H.P.
		Time.	I.H.P.		
3.51 to 4.18	16.86	3.55	542.2	485.2	Per Cent. 89.5
		4.15	552.2	485.2	87.9

Mean temperature of steam taken during half-load consumption = 398° F.

Overload with By-pass Valve.

Time.	Steam Pressure. Lbs. per Square Inch.				Vacuum at Condenser.	Revolutions per Minute.	B.H.P.
	At Boiler.	At Engine.					
		H.P.	I.P.	L.P.	Inches.		
10.30	175	165	70	18.0	25.5	280	1164
10.45	175	169	68	17.5	25.5	280	„
11.0	182	167	67	17.5	25.6	280	„
11.15	170	165	67	18.0	25.2	280	„

Diagrams taken.				B.H.P.	Efficiency <u>B.H.P.</u> <u>I.H.P.</u>
Time.	I.H.P.				
10.30	1244.6			1,164	Per Cent. 93.5
10.45	1243.1			1,164	93.6

TABLE III. D.—VARIABLE-LOAD TEST ON HIGH-SPEED, DOUBLE-ACTING, SELF-LUBRICATING ENGINE AT BIRMINGHAM, 15 NOVEMBER, 1901, WITH FIXED AND VARIABLE CUT-OFF.

The engine was started on full load, and, at a signal given, every half-minute the load was reduced by steps until only a very small load remained. The engine was then loaded again in the same manner, *i.e.*, by steps at half-minute intervals. All the readings—ammeter, voltmeter and gauges—were taken simultaneously in the steady intervals between the changes of load. The readings were continued over 90 minutes with the fixed cut-off and 90 minutes with the variable. The results were :—

With variable cut-off 27.87 lbs. steam per kilowatt-hour.

„ fixed „ 28.37 „ „ „ „

When tested under a steady load approximately equal to the mean of the variable load, the engine required 25.4 lbs. steam per kilowatt-hour.

TABLE IV. A.—TEST OF HIGH-SPEED DOUBLE-ACTING ENGINE AT FULL LOAD, WITH AND WITHOUT STEAM-JACKETS.
800 I.H.P. Triple-Expansion Engine (No. 1,003), tested at Birmingham, 4 January, 1901.

Time.	Steam-Pressure. Lbs. per Square Inch.			Vacuum at Engine.	Revolutions per Minute.	Indicated-HP.	Diagrams Taken.		Efficiency B.H.P. I.H.P.	Water-Consumption.		
	At Boiler.	At Engine.					Time.	I.H.P.		Time.	Per Hour.	Per I.H.P.- Hour.
		H.P.	I.P.									
P.M.												
3.30	164	129	48	9	24.75	300	3.30	803	92.8	..	..	
3.45	162	127	47	9	24.75	300	..	..	..	..	..	
4.0	163	126	46	9	24.75	300	..	..	..	..	..	
4.15	158	129	47	9	24.75	300	4.15	796	94.2	..	..	
4.30	160	128	46	9	24.75	300	4.30	792	94.6	4.22 to 4.40	11,867 Without jackets	
4.45	163	129	46	9	24.75	300	..	..	..	..	15.84 Without jackets	
5.0	160	128	47	9	24.75	300	..	..	..	..	..	
5.15	156	127	47	9	25.0	300	..	..	..	5.8 to 5.25	12,522 With jackets	
5.30	158	127	47	9	25.0	300	5.30	810	92.5	5.25	16.69 With jackets	
5.45	163	128	47	9	25.0	300	5.45	792	94.6	5.34 to 5.51	12,135 Without jackets	
6.0	164	128	47	9	25.0	300	..	..	..	..	16.18 Without jackets	

TABLE IV. B.—TEST OF THREE-CRANK TRIPLE-EXPANSION ENGINE OF MARINE TYPE, AT WALLSEND, 16 OCTOBER, 1901.

Comparison of Performance of Engine with Cylinders Jacketed and Unjacketed.

	Cylinders Jacketed.	Cylinders Unjacketed.
Mean pressure of steam at engine, lbs. per square inch	193	192
„ temperature of steam at engine, °F.	468½	475
„ superheat of steam at engine, °F.	83½	90½
„ pressure in I.P. receiver, lbs. per square inch .	60	62
„ „ „ L.P. receiver, „ „ .	10	9½
„ „ „ H.P. jacket, „ „ .	193	..
„ „ „ I.P. jacket, „ „ .	37	..
„ „ „ L.P. jacket, „ „ .	12½	..
„ vacuum in condenser, inches	26	25·6
„ temperature of condensed steam, °F.	111½	108
Mean temperature of circulating water at condenser inlet, °F. }	82·4	79
Mean rise of temperature of circulating water, °F. .	7·8	8
„ revolutions per minute by revolution-counter .	101·2	102·3
„ I.H.P. developed in H.P. cylinder	343	360
„ „ „ „ I.P. „	314	330
„ „ „ „ L.P. „	373	340
„ total I.H.P.	1,030	1,030
Total weight of steam used by engine, lbs. per hour .	10,663½	10,717½
Proportion used in jackets, per cent.	3·3	..
Total weight of steam per I.H.P.-hour, lbs.	10·35	10·4
Mechanical efficiency of engine, including bearing losses (assuming efficiency of alternator to be 95 per cent.) }	85·7	83·5

In the test with cylinders unjacketed, the mean superheat was a little greater and the mean vacuum a little less than in the test with cylinders unjacketed; so that it can be said that at this load there is practically no difference in consumption per indicated-horse-power, whether the cylinders be unjacketed or jacketed. The mechanical efficiency of the engine is about 2 per cent. lower when unjacketed than when jacketed, and it appears difficult to explain this.

TABLE V. A.—RATIO OF CLEARANCE-VOLUME TO VOLUME SWEEP BY PISTON.

Type of Engine.	Brake H.P.	Clearance-Volume in Percentage of Volume swept by Piston.										Stroke. Inches.	Revolutions or Double Strokes per Minute.
		H.P.			I.P.			L.P.					
		Top.	Bottom.	Mean.	Top.	Bottom.	Mean.	Top.	Bottom.	Mean.			
Belliss triple - crank high-speed ¹ (No. 1,233) (No. 1,245) (No. 1,246)	750	12.5	12.5	12.5	12.5	13.5	13.0	11.5	16.0	13.8	16	300	
	450	20.0	17.0	18.5	19.0	22.0	20.5	14.0	19.0	16.5	12	360	
	1,000	13.3	12.5	12.9	17.5	16.2	16.9	13.7	16.2	14.9	16	280	
	1,060	31.7	22.4	27.0	..	..	..	..	7.2	7.5	7.4	16	..
Belliss compound ¹ TEC/9													
No. 973 and 974													
Willans 9-cylinder ¹ (Bankside)	680	13.0	..	13.0	10.0	..	10.0	6.0	..	6.0	16	..	
Ferranti (Cardiff)	1,650	..	..	2.8	..	..	..	..	..	2.4	30	200	
Manchester	1,400	..	..	5.6	..	..	..	..	..	3.6	18	214	
	900	..	..	4.1	..	..	..	..	..	3.5	21	90	
Corliss	500	..	..	2.5	..	..	2.5	..	..	2.5	..	..	
"	1,500	..	..	1.5	..	..	1.5	..	..	1.5	..	..	
Allen high-speed ¹	200	..	..	15.0	..	..	..	..	..	12.0	8	425	
Tandem-compound ¹	500	..	..	12.0	..	..	..	..	..	10.0	12	350	
Musgrave horizontal triple-expansion	1,280	..	..	4.0	..	..	4.3	..	..	5.0	..	..	
Ditto compound	1,280	..	..	4.0	..	..	..	..	..	5.0	..	..	
Triple marine (s.s. "Iona")	1,000	..	..	12.4	..	..	10.1	..	..	7.6	39	61	
Belliss TCT/9 No. 1,107.	1,000	..	..	12.5	..	..	13.2	..	..	13.8	14	300	
Allis triple pumping	620	..	..	1.4	..	..	..	..	..	..	60	Double strokes, 20.3	
Leavitt triple pumping	520	..	..	1.4	..	..	..	..	..	..	72	50.6	
Lake Erie triple pumping	1,120	..	..	1.4	..	..	..	..	..	..	60	20.8	
Snow triple pumping.	740	..	..	1.8	..	..	..	..	..	..	60	21.5	
St. Louis Waterworks.													
Allis triple	..	..	..	1.036	..	..	1.18	..	..	0.509	..	..	

¹ These engines have piston-valves.

TABLE V. B.—EFFECT OF VARYING CLEARANCE ON THE ECONOMY OF A SELF-LUBRICATING COMPOUND ENGINE (E.C. 5 SIZE).
(Continued on p. 192.)

A	B	C	D	E	F	G	H	I
	Reference Number.	Steam Pressure at Engine, Lbs. per Sq. In.	Vacuum at Engine, Inches.	Revolutions per Minute.	B.H.P.	Water per B.H.P., Hour, Lbs.	Superheat, °F.	Water Calculated for 40° of Superheat, Lbs.
Clearance.	1	100	25	500	96.2	27.9	0	
	2	100	25	500	96.2	24.16	40	24.16
	3	100	25	500	96.2	22.4	80	
	4	100	25	500	98.5	28.1	0	
	5	100	25	500	98.5	24.5	35	24.3
	6	100	25	500	98.5	21.8	100	
	7	100	25	500	98.8	28.9	0	25.25
	8	100	25	500	98.0	29.1	0	25.6
	9	100	25	500	98.0	24.5	75	
Table of Clearance.								
Clearance.	H.P.			L.P.				
	Top.	Bottom.	Mean.	Top.	Bottom.	Mean.		
	Per Cent.	Per Cent.	Per Cent.	Per Cent.	Per Cent.	Per Cent.		
	23.50	16	19.75	11.50	18.6	15.05		
Normal clearance	42.85	..	..	42.85	..	..		
Added "	66.35	16	41.18	54.35	18.6	36.48		
Total "								

TABLE V. B.—continued.

Clearance.	Mean Pressures Lbs. per Square Inch.				I.H.P.			Brake Efficiency.
	H.P.		L.P.		H.P.	L.P.	Total.	
	Top.	Bottom.	Top.	Bottom.				
Normal clearance .	41·5	40·4	25·28	23·95	36·25	66·8	103·05	Per Cent. 93·3.
Extra on H.P. . .	36·9	41·6	28·3	24·8	34·8	72·1	106·9	92·2
Extra on L.P. . .	48·0	41·2	20·8	25·5	39·5	62·7	102·2	92·7
Extra on H.P. and L.P. }	44·0	44·1	23·2	25·0	39·0	65·4	104·4	93·8

The results given are the mean of several trials under each condition. Those in column I (p. 51) were obtained by plotting the results at various degrees of superheat.

SUMMARY.

Adding clearance to H.P. cylinder increased the power and only slightly reduced the economy.
 Adding clearance to L.P. cylinder decreased the power, but had a more marked effect in reducing the economy.
 Adding clearance to H.P. and L.P. cylinders increased the power and reduced the economy.
 Superheating did not much change the nature of these results.
 Increase of power due to extra clearance in H.P. cylinder occurred in L.P. cylinder.
 Decrease of power due to extra clearance in L.P. cylinder occurred in H.P. cylinder.
 The diagram steam-line was much steadied by the additional clearance. See *Figs. 9* (p. 167).

TABLE VI.—GAIN DUE TO SUPERHEAT IN LOW- AND HIGH-SPEED ENGINES.

Engine.	Horse-Power.	Type.	Percentage Saving on Dry Steam due to 10° Superheat.	Remarks.
Low-speed Sulzer	3,000	{Triple-exp. drop-valves}	0·67	{ <i>The Engineer</i> , vol. lxxxix. p. 546. Report of Association of Steam-Users in Alsace, 1895. Grand Junction pumping-engine.
„ „ Corliss	..	Compound	0·78	
„ „ Jas. Simpson & Co. . . . }	..	..	1·0	
High-speed	550	Triple	1·5	Automatic expansion.
„ „	350	„	1·7	Throttling engine.
„ „	300	„	1·0	{Automatic expansion and full loads. Automatic expansion and light loads.
„ „	300	„	1·4	
„ „	135	„	1·5	Throttling engine.
„ „	135	Compound	1·75	„ „
„ „	100	„	1·4	„ „

TABLE VII.—EFFECT OF SUPERHEAT ON LEAKAGE PAST PISTON-VALVE.

Test made on 100-B.H.P. Compound Double-Acting High-Speed Engine.

	Valve, Good Fit, i.e., 0·001 Inch Slack.		0·008 Inch Turned Off, i.e., 0·007 Inch Slack.	
	Wet Steam.	Dry Steam.	Wet Steam.	Dry Steam.
Steam-pressure on range, lbs. per square inch	98	98	108	99
Temperature of saturated steam corresponding to pressure, ° F. }	336·7	336·7	343	337·4
Temperature as noted, ° F. .	..	339·5	..	343·6
Steam - pressure in governor-valve, lbs. per square inch . }	57	57	57	57
Vacuum, inches	26·2	26·2	26·2	26·2
Revolutions per minute . . .	500	500	500	500
Effective HP.	69·32	60·3	60·3	60·3
Lbs. of water per hour . . .	1,532	1,549	1,724	1,591
Lbs. of water per effective HP.-hour }	25·39	25·6	28·5	26·3
Increase per cent.	..	..	12·2	2·7

TABLE VIII. A.—GOVERNOR TESTS.

Type.	Revolutions per Minute.			Foot-tons per E.H.P. Fly-wheel.	Remarks.
	Load on.	Momentary Variation.	Load off.		
<i>Double-acting compound.</i>					
E.H.P. 324	382	391	386	2·87	In alternator only.
<i>Double-acting triple.</i>					
B.H.P. 1,102	240	..	245	0·78	In fly-wheel, no dynamo.
„ 256	420	445	427	0·61	In fly-wheel only.
<i>Single-acting central- valve compound.</i>					
I.H.P. 100	460	477	467	..	..
„ 700	301	312	312	..	..
„ 360	359	369	366	..	..
„ 180	450	466	463	..	..
„ 450	333	343	339	..	..

TABLE VIII. B.—TEST OF 300-I.H.P. DOUBLE-ACTING BELLISS ENGINE.

Load		Revolutions per Minute			Remarks.
From	To	From	To	Settle at	
Full	0	395	430	396	With combined expansion and throttling governor.
0	Full	396	380	394	
Full	0	399	427	396	
0	Full	395	383	399	
Full	0	395	429	394	With throttling governor only.
0	Full	392	380	395	

The governing with the expansion-gearing was quite as sensitive as with the throttle-valve only: practically the same results were obtained, the interval between "Break" and "Settle" being 14 seconds in each case.

TABLE VIII. C.—GOVERNING-TRIALS WITH FOUR-CRANK TRIPLE-EXPANSION ENGINE No. 371, AT WALLSEND POWER-STATION, 9 MARCH, 1901.

Number.	Indicated-HP. changed		Revolutions per Minute. Steady.	Time. Seconds.	Revolutions per Minute.	Time. Seconds.	Revolutions per Minute.	Time. Seconds.	Revolutions per Minute.	Time. Seconds.	Revolutions per Minute. Steady.
	From	To									
1	320	0	100	3	102	20	100	5	101	5	100½
2	0	320	100½	3	99	5	100½	5	100	..	..
3	550	280	100	3	101	5	100	..	..	..	..
4	280	0	100	3	102	20	100½	5	101	..	..
5	0	550	101	3	98	5	100	5	99½	..	..
6	810	425	99	3	101	5	99½	5	100½	..	..
7	425	0	100½	3	102½	25	100	5	101	10	100¾
8	1,130	520	99	3	101½	5	100	5	101	5	100½
9	525	0	100½	3	103	25	101	..	..	..	..
10	0	1,030	101	3	97	5	99½	10	99	..	..
11	1,035	0	99	3	104½	25	101	..	..	..	..
12	0	1,045	101	3	97	5	99½	10	99	..	..
13	1,040	0	99½	3	105	30	101	..	..	..	..

The indicated horse-power has been calculated from the electrical readings by comparison with steam diagrams taken at the same electrical load. The tachometer readings are taken at the moments of highest and lowest revolutions; the time between the readings has been calculated from the revolutions counted during the same space of time.

TABLE VIII. D.—GOVERNING-TRIALS WITH FOUR-CRANK TRIPLE-EXPANSION ENGINE No. 370, AT WALLSEND POWER-STATION, 29 AUGUST, 1901.

No.	Indicated HP. changed		Revolutions per Minute. Steady.	Time. Seconds.	Revolutions per Minute.	Time. Seconds.	Revolutions per Minute. Steady.
	From	To					
1	580	0	100½	3	103½	30	102
2	0	580	102	3	99	5	100½
3	530	0	100½	3	102½	30	102
4	0	800	102	3	99	5	99½
5	800	0	99½	3	105	30	102
This was improved by changing the weights and tried again the next day—30 August, 1901.							
1	0	950	100½	3	98½	5	98¾
2	950	0	98¾	3	103	30	100½

Some alterations reduced the number of fluctuations after changing the load.

TABLE IX. A.—ENGINE NO. 1,233 AND ALTERNATOR. SUMMARY OF TEST AT WORKS OF MESSRS. BELLISS AND MORCOM, BIRMINGHAM, 19, 20, 21 AND 22 DECEMBER, 1900.

Engine: triple-expansion, three-crank; cylinders, 14½ inches, 22 inches, and 34½ inches × 16 inches; 300 revolutions per minute.
 Alternator: revolving-armature, single-phase, sixty cycles, 2,100 volts, 240 amperes.

Load.	Output. Kilowatts	Steam-Consumption.			Temperature of Steam, F.	Superheat, F.	Efficiency Percent.	Sets of Cards.	Energy for Excitation.		Steam-Pressures by Gauge, Lbs. per Square Inch.			Vacuum at Con- denser, Inches.	Remarks.	
		Lbs. per Hour.	Lbs. per Kilowatt-Hour.						Per- cent- age of watts Out- put.	At Boiler.	Initial in Cylinders.					
			Jacket- Steam Omitted.	Jacket- Steam Included.							H. P.	I. P.	L. P.			
Full . .	496.9	11,032	22.19	22.61	389°	19°	87.9	10	3.35	0.67	159	143	64	15.6	25.0	Valves set for 150 lbs.; by-pass valve shut; jacket- steam used.
Half . .	257.5	5,985	23.23	24.04*	378°	8°	83.2	4	3.38	1.31	158	76	26.9	1.7	26.5	Ditto.
Three-fifths	300.4	8,300	27.60	28.30	376°	27°	82.5	4	4.42	1.47	118	74	47	8.5	26.3	Valves set for 150 lbs.; by-pass valve open; jacket- steam used.
Full . .	500.1	10,800	21.56	21.98*	419°	37°	87.3	7	2.83	0.57	186	171	59	14.0	25.4	Valves set for 170 lbs.; by-pass valve shut; jacket- steam used.
Full . .	499.7	11,010	22.00	..	..	..	..	0	3.25	0.65	191	170	60	15.3	25.5	Valves set for 170 lbs.; by-pass valve shut; no jacket- steam.
Five-sixths	432.1	9,850	22.80	23.25*	381°	3°	..	0	2.05	0.48	190	153	50	9.3	25.0	Valves set for 170 lbs.; by-pass valve shut; jacket- steam, but no superheater.

TABLE IX. A.—(continued).

Load.	Output. Kilowatts	Steam-Consumption.		Mean Tempera- ture of Steam, °F.	Superheat, °F.	Efficiency Per cent.	Number of Sets of Cards.	Energy for Steam-Pressures by Gauge. Excitation, Lbs. per Square Inch.			Vacuum at Con- denser, Inches.	Remarks.
		Lbs. per Hour.	Lbs. per Kilowatt-Hour. Jacket- Steam Omitted.					Kilo- watts Out- put.	Per- cent- age of Boiler.	Initial in Cylinders. At H.P. I.P. L.P.		
Half . .	253.9	6,130	24.1	24.9*	395°	11°	2					Valves set for 170 lbs.; by-pass valve shut; jacket- steam. Valves set for 150 lbs.; by-pass valve shut; no jacket- steam. Valves set for 170 lbs.; by-pass valve open; no jacket- steam.
Half . .	253.7	6,270	24.6	..	393°	11°	3	3.281.29	186	97 26.2 2.5 25.1		
Overload .	599.0	13,770	26.3	..	405°	25°	3	4.23 0.71	181 168 101 28.8 24.0			

The readings in this Table are the means obtained from a series of trials.

An asterisk (*) against the steam-consumption denotes that jacket-steam was not recorded, and was taken to be the same as in recorded cases.

Owing to the load being taken by resistance-frames with limited range of adjustment, variations in load were, in many instances, made by varying the voltage. The range of variation was 2,000 to 2,500 volts. This may partly account for the great variation in exciting energy.

The output was worked on the Thomson-Houston Wattmeter (No. 300) readings, as these were approximately the mean of the products of two sets of independent ammeter and voltmeter readings. An integrating wattmeter was also used, and it agreed closely with the above.

TABLE IX. B.—FULL-LOAD EFFICIENCY-TEST OF DOUBLE-ACTING HIGH-SPEED ENGINE WITH ALTERNATOR.
Tested by the Author, 27 September, 1898.

Time.	Revolutions per Minute.	I.H.P.			Output.			Efficiency I.H.P. 2×0.746 $a + b$	Exciting Energy.		Efficiency, allowing for Exciting Energy.	Remarks.			
		Crank A.	Crank B.	Crank C.	(Crank Total.	Volts.	Kilowatts by Weston.		Kilowatts by Watmeter.	E.H.P. by Watmeter.			E.H.P.	Percentage of Output.	E.H.P. Output less Exciting Energy.
		Top of Cyls. only.			(a)	(b)					Per Cent.	Positions of Indicators on Engine.			
11.45	362	134.2	62.8	..	2,020	248	244	329.8	..	2.88	3.86	1.17	..		
12.15	361	133.2	63.8	..	1,970	248	245	6330.8	..	2.92	3.91	1.18	..		
1.0	362	135.1	62.0	..	2,020	246	248	331.0	..	2.97	3.98	1.20	..		
Stopped at 1.34 for about 13 minutes to change over indicator-cocks.															
		Bottom of Cyls. only.										Positions of Indicators on Engine.			
3.0	359	..	64.5	133.0	..	2,000	246	246	329.8	..	3.1	4.15	1.25	..	
3.45	362	..	64.9	133.5	..	2,000	244	246	328.0	..	3.06	4.1	1.25	..	
4.0	364	..	64.7	132.9	..	1,990	246	248	331.0	..	3.04	4.07	1.23	..	
Means	361.6	134.1	127.6	133.1	304.8	2,000	246	3246.2	330.0	83.6	2.98	4.01	1.21	326	82.6

TABLE IX. C.—TEST OF HIGH-SPEED DOUBLE-ACTING COMPOUND ENGINE AND ALTERNATOR.

Cylinders, 8 inches and 14 inches × 10 inches; six cylinders, three cranks.
Tested by Author, 29 November, 1898.

Time.	Revolutions per Minute.	Indicated H.P.	Exciting Energy.		Output. Kilowatts.	Volts.	E.H.P.	Efficiency. E.H.P. I.H.P. Per Cent.
			E.H.P.	Percentage of Output.				
A.M.								
8.0	357	..	..	1.02	250.4	1,960	..	..
8.15	366	..	..	1.03	254.4	1,990	..	..
8.30	367	..	..	1.07	241.6	1,970	..	Mean 87.2
8.45	367	..	..	1.05	244.8	2,006	..	
9.0	368	..	..	1.1	245.6	1,990	..	..
9.30	367	381.0	3.7	1.11	248.0	1,980	333.3	86.7
9.45	367	..	..	1.12	246.4	1,986	..	..
10.0	367	..	..	1.15	240.8	2,000	..	..
10.10	368	371.2	3.88	1.17	247.2	1,980	332.3	88.6
10.30	357	..	..	1.18	256.0	1,960	..	..
10.45	359	..	..	1.28	232.0	2,070	..	..
11.0	360	375.2	3.9	1.2	244.0	1,980	328.0	86.5
11.15	361	..	..	1.21	244.8	1,976	..	..
11.30	360	..	..	1.27	234.4	2,004	..	..
11.45	360	..	..	1.23	242.4	2,000	..	..
P.M.								
12.0	361	..	..	1.24	240.0	2,002	..	..
12.15	360	..	..	1.18	240.0	1,948	..	..
12.30	361	376.8	4.07	1.25	244.0	1,996	328.0	86.2
12.45	361	..	..	1.22	251.2	1,980	..	..
1.0	361	..	..	1.25	246.4	1,990	..	..
1.15	361	..	..	1.26	242.4	2,006	..	..
1.30	360	..	..	1.25	245.6	1,986	..	..
1.53	361	375.0	4.14	1.25	248.0	2,010	333.3	87.9
2.0	361	..	..	1.29	236.8	2,050	..	..

Engine started 7 A.M. Ran light till 7.40, when load was put on. Full load commenced 8 A.M. Load put on by means of acidulated-water resistance in a tank with lead electrodes.

Temperature rise after six hours' run :—Fields, 50° F.; Armature, 42° F.

TABLE IX. D.—TEST OF THREE-CRANK TRIPLE-EXPANSION MARINE-TYPE
ENGINE AT WALLSEND, 6 OCTOBER, 1901.

Duration of trial	4 hours.
Mean revolutions of engine by tachometer	100·2 per minute.
" " " " " revolution counter	102·3 " "
" kilowatts from switchboard integrating watt-meter	609 kilowatts.
Mean apparent kilowatts	628 " "
Power factor	97 per cent.
Mean total indicator-HP., from	1,030 I.H.P.
" temperature of steam at engine	475° F.
Temperature of saturated steam at mean engine-pressure	384·5° F.
Mean superheat of steam at engine	90·5° F.
" temperature of condensed steam	108° F.
Mean temperature of circulating water at condenser inlet	79° F.
" " " " " outlet	87° F.
" pressure of steam at engine	192 lbs. per sq. in.
" vacuum in condenser	25·6 inches.
Total weight of steam used by engine in cylinders	42,869 lbs.
" " " " " jackets	0 " "
" " " consumed by engine	42,869 " "
" " " " " per hour	10,717½ " "
Total weight of steam consumed by engine per I.H.P.-hour	10·4 " "
Total steam consumed by engine per kilowatt-hour	17·6 " "
Combined efficiency $\frac{\text{E.H.P.}}{\text{I.H.P.}}$	79·3 per cent.
Mean power absorbed in excitation-circuit, including } rheostat (51 amperes 240 volts) }	12·2 kilowatts.
Mean power absorbed by circulating-pump motor (79½ } amperes 242 volts) }	19·2 " "

The energy absorbed in excitation and circulating-pump motor is not included in the above figures of consumption and efficiency.

TABLE X.—ENERGY AND WEIGHT OF FLY-WHEELS NECESSARY TO KEEP THE FLUCTUATION OF SPEED DURING ONE REVOLUTION WITHIN 3% OF THE MEAN, CALCULATED FROM THE FLUCTUATION OF ENERGY DUE TO VARYING STEAM-PRESSURE AND ACCELERATION AND WEIGHT OF RECIPROCATING PARTS. SPEED AT CENTRE OF GYRATION, 80 FEET PER SECOND.

Engine.	E.H.P.	Revolutions Per Minute.	Fluctuation of Energy. Per Cent.	Fly-wheel Effect Required. Foot-Tons.		Requisite Weight of Fly-wheel Rim supposing no Fly-wheel Effect supplied by Revolving Parts of Machine. Lbs.	Diameter of Centre of Gyration 80 Feet per Second.	Diagrams of Fluctuation of Turning-Effort.
				Total.	Per E.H.P.			
Three-crank double-acting engine, complete engine on each crank Three-crank triple-expansion, powers 6:5:4	410	350	3.1	51.0	0.124	1,140	4.35	
	410	350	6.6	108.6	0.265	2,444	"	
	410	350	11.5	195.0	0.475	4,410	"	
Three-crank single-acting engine, complete engine on each crank Two opposite cranks . . .	410	350	29.2	485.0	1.180	10,870	"	
	410	180	34.2	795.0	1.940	17,820	8.45	
Horizontal tandem single crank	410							

TABLE XI. A.—WEAR ON DOUBLE-ACTING HIGH-SPEED SELF-LUBRICATING ENGINE AFTER 4 YEARS' RUNNING.

Pins and Bearings.	Largest Diameter.	Least Diameter.	Original Diameter.	Apparent Wear.	Thickness of Crown of Brasses.	Thickness of Side of Brasses.	Apparent Wear.
	Inches.	Inches.	Inches.	Inch.	Inch.	Inch.	Inch.
<i>Crosshead-Pin.</i>							
H.P. . .	2·571	{ 2·705 Middle of pin 2·5705	..	None	NOTE.—These brasses had been "touched up," making gauging unreliable.		
L.P. . .	2·571	{ Middle of pin	..	"			
<i>Main Bearings and Shaft.</i>							
Next fly-wheel	5·000	4·9995 Middle	5·000	"			
Intermediate L.P.	4·9995	4·9995	5·000	0·0005			
Intermediate H.P.	{ 5·000 Taper in pin	{ 4·994 Taper in pin	5·000	None			
Next governor	5·000	5·000	5·000	"			
<i>Crank-Pin.</i>							
H.P. . .	{ 5·000 0·0005 oval	{ 4·9995 0·0005 oval	5·000	"	0·484	0·488	0·004
L.P. . .	5·000	4·999	5·000	"	0·486	0·492	0·006
Eccentric sheave	{ 9·5 0·001 oval	{ 9·499 0·001 oval	9·5	"	..	..	..
Piston-rods and Valve-rods.				H.P. Piston-Rod.	L.P. Piston-Rod.	Valve-Rod.	
				Inches.	Inches.	Inch.	
Diameter of end of working part . . .				2·994	2·996	1·601	} 0·0015 oval
" " middle of " " . . .				2·994	2·996	1·599	
" " non-working part . . .				3·000	3·000	1·613	
Apparent wear of end of working part . .				0·006	0·004	0·012	
" " " middle of working part .				0·006	0·004	0·014	

TABLE XI. A—(continued).

Cylinders and Valve-Chamber.	H.P. Cylinder.	L.P. Cylinder.	Valve- Chamber.
	Inches.	Inches.	Inches.
Original diameter	..	..	8·0
Smallest „	11·9945	11·979	8·000
Largest „	11·998	11·987	8·003
Apparent wear between smallest and largest diameters	None	..	0·003

O.I.Pump.	No. 1.	No. 2.
	Inch.	
Bore of pump	1·507	..
Diameter of plunger	1·505	..
Apparent wear	None	..

Valve-Chambers.	Bore of Chamber.	Diameter of Ring.	Difference.
	Inches.	Inches.	Inch.
H.P. top	8·001	7·996	0·005
H.P. bottom	8·002	7·996	0·006
L.P. top	8·003	7·996	0·007
L.P. bottom	8·001	7·993	0·008

Piston-Rings.	H.P.	L.P.
Description	Rowan rings	Rowan rings
Condition	{ Very good (worn to a good surface)	{ Very good (worn to a good surface)

No adjustments made during 4 years.

TABLE XI. B.—TESTS MADE ON THREE ENGINES
TO SHOW THE EFFECT OF WEAR ON STEAM-CONSUMPTION.

Three identical 175-B.H.P. Compound Engines
tested at Messrs. Belliss and Morcom's Works, Birmingham.

	No. 1.	No. 2.	No. 3.
Brake-HP.	160·0	172·0	176
Water per B.H.P.-hour, lbs.	23·8	23·75	23
Condition of engines . . {	After 3½ years' running. }	New	New
Tests made	Non-condensing.	Non-condensing.	Non-condensing.

The amount of wear of the slide-valve rings of No. 1, as measured after 3½ years' running, was 0·004 inch; *i.e.*, they were 0·006 inch slack, 0·002 inch of which would be slack due to original working fit.

TABLE XII.—WEIGHT AND OUTPUT OF "GENERAL ELECTRIC" ALTERNATORS.

	500 Kilowatts.		1,000 Kilowatts.	
	360 Revolutions per Minute.	90 Revolutions per Minute.	400 Revolutions per Minute.	120 Revolutions per Minute.
<i>Efficiencies.</i>	Per Cent.	Per Cent.	Per Cent.	Per Cent.
Half load	90·0	90·0	92·0	92·0
Three-quarter load . .	92·5	92·5	94·0	94·0
Full load	93·5	93·5	95·0	95·0
<i>Weight.</i>	Lbs.	Lbs.	Lbs.	Lbs.
Armature	9,000	27,600	16,400	36,800
Field	9,000	28,000	17,500	34,300
Base	7,500	4,400	10,600	4,700
Total	25,500	60,000	44,500	75,800
<i>Fly-wheel Effect.</i>				
Lbs.	7,900	28,000	15,000	34,300
Feet radius	1½ foot	4 foot	1½ foot	4 feet
	Ft. Ins. Ft. Ins.	Ft. Ins. Ft. Ins.	Ft. Ins. Ft. Ins.	Ft. Ins. Ft. Ins.
Floor space	9 3 by 7 8	5 by 17 10½	11 3 by 7 8	5 8 by 18 2

TABLE XIII.—COST OF SPARES.

1,000 B.H.P. High-Speed Triple-Expansion Double-Acting Engine.

	Price.	Weight.
	£ s.	Cwt. Qr. Lbs.
One connecting-rod, with crosshead pins, caps and bolts	50 0	3 0 7
One pair of piston-rod crosshead brasses	4 10	0 1 2
One pair connecting-rod crankpins	26 0	2 2 21
One valve-rod with cap, bolts, brasses and nuts	8 0	1 2 0

1,000 B.H.P. Vertical Corliss Cross-Compound Engine.

	Price.	Weight.
	£ s.	Cwt.
One connecting-rod	125 0	3½
One set of brasses for connecting-rod, about	34 0	..
One set of journal brasses for one bearing, about	48 0	12
One spare valve-rod for Corliss valve, for H.P. cylinder	4 10	..
“ “ “ “ for L.P. “	6 10	..

TABLE XIV.—TRIAL OF THE REDUCTION IN FRICTION AT NO-LOAD
DUE TO FORCED LUBRICATION.

120-B.H.P., two-crank, compound, self-lubricating engine, cylinders 9 inches and 15 inches in diameter, stroke 8 inches. Oil-pump driven from slide-valve eccentric-straps. Engine ran without load, condensing, and at a speed of 450 revolutions per minute, during each of the trials.

Trial No. 1 :—

Pressure in oil-pipes 30 lbs. per square inch.
Horse-power indicated 2·128 I.H.P.

Trial No. 2 :—

Pressure in oil-pipes 5 lbs. per square inch.
Horse-power indicated 2·409 I.H.P.

Trial No. 3 :—

The oil-pump was uncoupled altogether, the engine being lubricated with such oil as remained in the oil-pipes from the previous trial.

Pressure in oil-pipes 0 lbs. per square inch.
Horse-power indicated 3·333 I.H.P.

The oil-pump is single-acting, 1½ inch bore by 2¼ inches stroke, and draws the oil from the same level as the pump. The calculated horse-power required to drive it is therefore approximately 0·135 HP. when pumping against 30 lbs. per square inch oil-pressure, and 0·022 HP. when pumping against 5 lbs. per square inch oil-pressure.

NOTE.—If the above figures are taken to indicate the constant friction of the engines as usual with large low-speed engines, the efficiency works out to 98·4 per cent.

TABLE XV.—COMPARATIVE ECONOMY OF HIGH- AND LOW-SPEED STATIONS.

Power-House No. 1.

Load Factor.	Output per Year.	I.						II.	III.		I. + II. + III.				
		Coal.							Wages per Unit.	Interest at 5 per Cent.	Interest per Unit.	A	B	C	
		Lbs. per Unit.		Cost per Unit.											
		Coal at 6s. per Ton.	Coal at 12s. per Ton.	A	B	C									
Per Cent.															
Lighting 12	1,576,800	7.0	6.5	0.224	0.418	0.579	d.	d.	d.	d.	d.	d.	d.	d.	d.
Trams . 35	4,599,000	6.3	5.8	0.202	0.373	0.501	0.150	0.260	0.634	0.828	0.989				
Lighting, power and trams } 60	7,884,000	4.8	4.5	0.154	0.289	0.405	0.090	0.089	0.381	0.552	0.680				
							0.046	0.052	0.252	0.387	0.503				

1

2

Power-House No. 2.															
3	Lighting	12	4,730,400	6.65	6.18	5.7	0.214	0.397	0.550	0.075	On £64,500. 3,225	0.164	0.453	0.636	0.789
	Trams	35	13,787,000	5.99	5.51	4.94	0.192	0.354	0.476	0.045	"	0.055	0.282	0.454	0.576
	Lighting, power and trams	60	23,625,000	4.56	4.28	3.99	0.146	0.275	0.385	0.023	"	0.033	0.202	0.331	0.441
4	Lighting	12	4,730,400	6.31	5.86	5.42	0.202	0.377	0.522	0.10	On £35,200. 4,260	0.217	0.519	0.694	0.839
	Trams	35	13,787,000	5.63	5.18	4.64	0.181	0.334	0.447	0.060	"	0.074	0.315	0.468	0.581
	Lighting, power and trams	60	23,625,000	4.24	3.98	3.71	0.136	0.256	0.358	0.031	"	0.043	0.210	0.330	0.432

1. Four high-speed sets of 500 kilowatts each. Total cost, £34,000. Assumed maximum load, 1,500 kilowatts.

2. Four low-speed sets " " " £47,000. " " " " " "

3. Four high-speed sets 1,500 " " " £64,500. " " " 4,500 " "

4. Four low-speed sets " " " £85,200. " " " " " "

$$\text{Load-factor} = \frac{\text{Output} \times 100}{\text{Maximum load} \times \text{time}}$$