

in the fly-wheel, when the speed becomes too high, actuates the latching arrangement *i-k* and throws the oil pump serving to charge the vaporizer temporarily out of action.

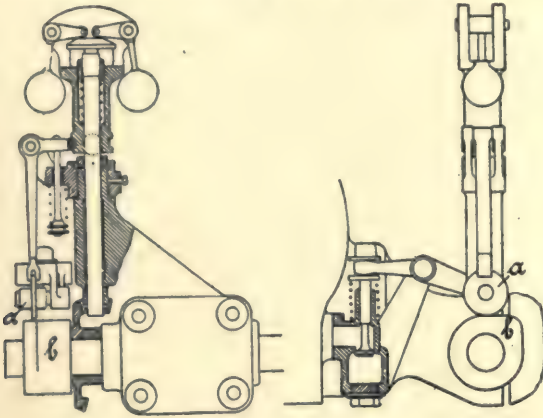


FIG. 14-14.

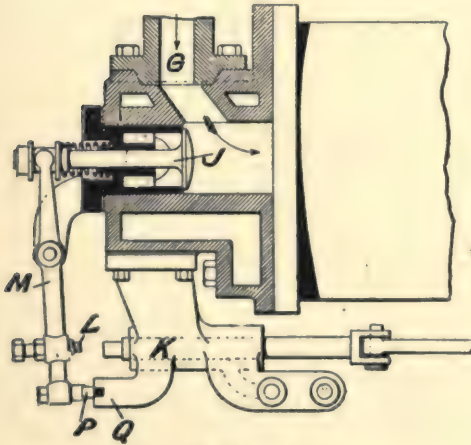


FIG. 14-15.

The Robey governor,* Fig. 14-14, is of the fly-ball type; normally the roller, *a*, under the control of the governor travels on the cam, *b*, thus opening the gas valve at the proper time. Under excessive

* Schöttler, Die Gasmaschine.

speed, however, the roller is shoved to the left on its spindle by the governor linkage, and misses the cam.

The Campbell engine has a fly-ball governor which operates to keep the exhaust valve open when the speed exceeds the normal, Figs. 14-15 and 14-16.* The rising of the governor weights, Fig. 14-16, depresses the end, *O*, of the lever, *N*, thus interposing the plate, *P*, Fig. 14-15, between the end of the exhaust lever, *M*, and the stationary bracket, *Q*. This prevents the exhaust valve from closing until the dropping of the speed causes the governor

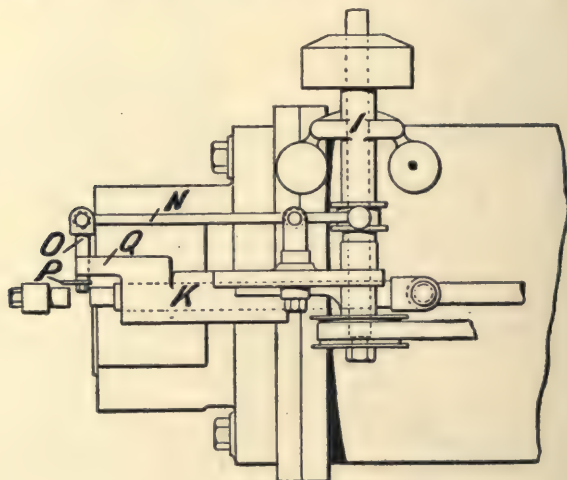


FIG. 14-16.

to withdraw the plate, *P*, when the exhaust valve is again regularly opened by the sliding piece, *K*.

A shaft governor acting to hold the exhaust valve open is used on the Perkins engine, Fig. 14-17.† The exhaust valve is operated by the lever *R*, which in turn is actuated by the block *A*, striking the end block, *B*, of the rod, *C*. As the speed exceeds the normal, the governor weights move toward the circumference of the wheel, and, through suitable linkage, throw the latch, *P*, into a recess in the side of the block, *B*. This prevents the rod, *C*, from returning, and keeps the exhaust valve open.

* Clerk, the Oil and Gas Engine.

† Power Quarterly, Oct. 15, 1900.

(c) QUALITY REGULATION. — The quality method of regulation, as carried out for gas fuel, is best exemplified in the Nürnberg engine. This engine has separate gas and inlet valves. At all

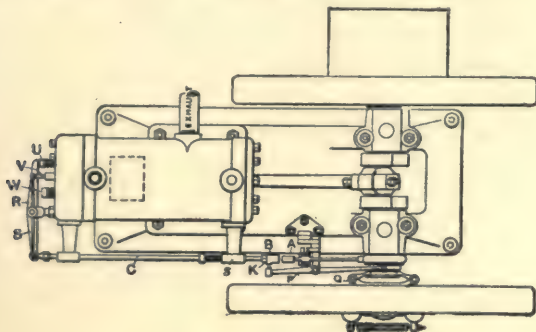


FIG. 14-17.

loads the inlet valve opens and closes at the same time, thus admitting a constant charge of volume. But the gas valve, under governor control, opens later as the load drops, and closes about the same time as the main inlet valve. Thus the composition or

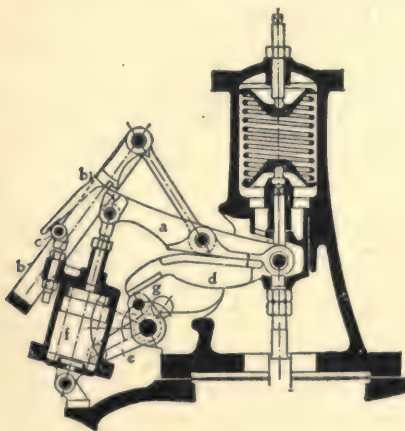


FIG. 14-18.

quality of the mixture changes with every load. A certain disadvantage of this method of proportioning the charge is that the mixture is not only affected by the relative time of opening of

the gas and inlet valves, but also by the relative velocities of the gas and air currents. Thus at low loads, when the gas valve opens, the gas column has to start from rest, while the air column already has its maximum velocity. The time needed for the acceleration of the gas column is practically constant, hence this factor affects the mixture differently for every load. Another point is that unless the gas valve closes quickly there is apt to be so much throttling of gas as to make the mixture drawn in at the end for low loads incombustible. This is especially bad since this mixture is apt to be located around the igniter.

The governor used on the Nürnberg engine is of the ordinary fly-ball type. The trip gear for the gas valve is shown in Fig. 14-18.* The governor determines the position of the bell-crank

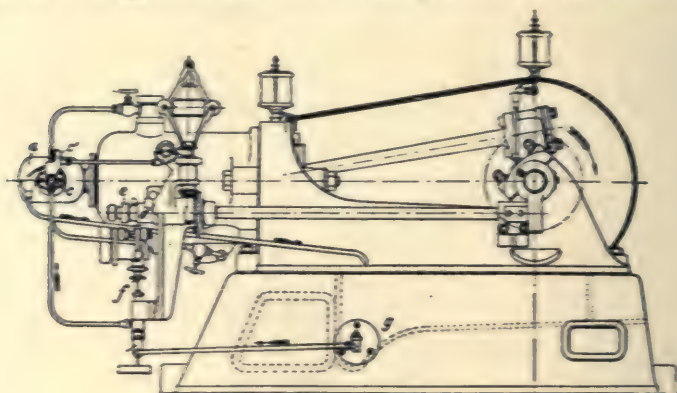


FIG. 14-19.

lever *e-g*, and through this the position of the wiper cam lever *d*. The eccentric rod, *b*, through the latch *b₁*, lifts the gas valve by depressing the valve lever, *a*. The position of *d* determines the time of opening. The valve is closed practically instantaneously by the coil spring shown in the top of the housing as soon as the small roller, *c*, has released the latch *b₁*. The dash pot, *j*, serves to dampen the drop of the valve, making it close without shock.

Many oil engines are governed by what is practically quality regulation. Thus the Hornsby-Akroyd oil engine, Fig. 14-19,† takes a full charge of air every stroke, but the quantity of oil is

* Power, Feb., 1906.

† Güldner, *Verbrennungsmotoren*, p. 119.

changed to suit the load. The quality of the mixture therefore changes for every different load. The proportioning of the oil supply to the load is accomplished as follows. The oil pump supplies a constant quantity of oil every stroke to the vaporizer valve, *c*. This valve, however, is a two-way valve, one exit opening through an atomizing nozzle into the vaporizing chamber, while the other allows some of the oil to flow back into the oil tank. The fly-ball governor, through the linkage shown, controls the size of this overflow opening, *c'*, and thus determines the amount of oil which enters the vaporizer chamber.

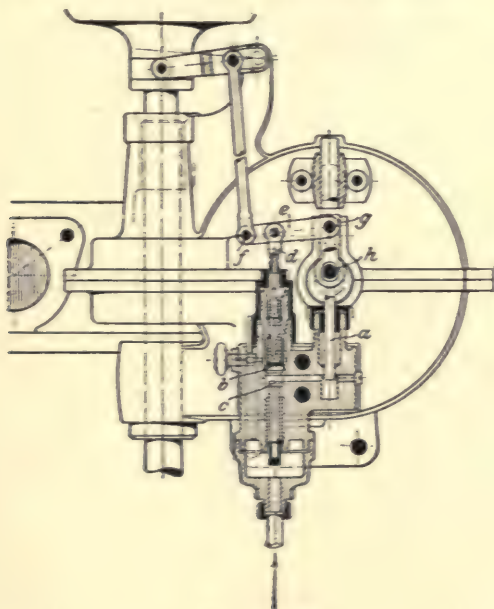


FIG. 14-20.

The governing mechanism of a small Diesel oil engine is shown in Fig. 14-20.* The pump plunger, *a*, is actuated by the lay shaft, *h*. The suction valve is shown at *c* and the automatic discharge valve at *b*. The oil under pressure flows through *b* to the atomizing nozzle. The amount of oil required for the load on the engine at any given time is measured by controlling the time of closing of the suction valve, *c*, forcing more or less of the oil back into the

* Güldner, *Verbrennungsmotoren*, p. 379.

oil supply tank. The motion of valve *c* is controlled through linkage not shown, by the governor lever *e*. At *g* this lever receives an up-and-down motion from the lay shaft, but the manner in which this motion is transmitted to the suction valve depends upon the position of the fulcrum, *f*, which is determined by the governor, thus varying the time of closing of the valve.

(*d*) QUANTITY REGULATION. — Koerting Bros. govern their four-cycle engine as follows, Fig. 14-21*: Air enters through the pipe, *D*, and gas through *B*. The two are mixed in the proportion suited to the particular gas by the mixing valve *A*. This valve

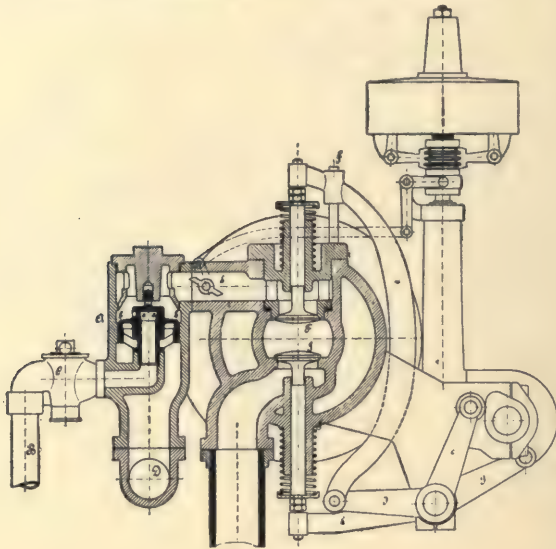


FIG. 14-21.

when once adjusted therefore furnishes a mixture of constant quality. To suit the quantity of this mixture to the load, the governor controls the position of the throttle valve *e* in the passage leading from the mixing valve to the inlet valve. This valve thus throttles the charge during the entire suction stroke.

A very similar arrangement is used in the Westinghouse vertical engine, Fig. 14-22.† *A* is the throttle valve controlling the entrance of the mixture to the cylinder. Its position depends upon the

* Schöttler, *Die Gasmachine*.

† L. S. Marks, *Instruction Paper on Gas and Oil Engines*.

action of the fly-ball governor, *B*. Gas enters the interior of this valve through ports *G* and air through ports *D*. The relative proportions of these ports are fixed by slides *H-H*, so that the proper relation of air to gas for the fuel used can be established and maintained.

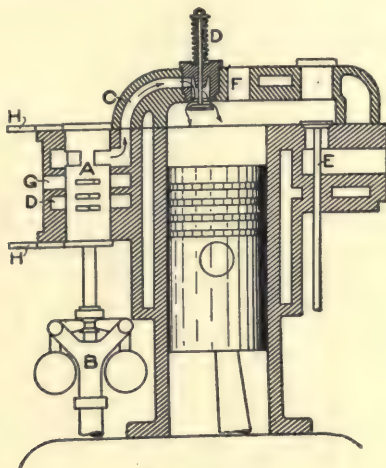


FIG. 14-22.

Fig. 14-23 shows the arrangements for operating and controlling the mixing valve in the Ehrhardt and Sehmer engine, which is a modification of the Deutz engine.* It will be readily seen, by a study of the figure, that the governor, through its linkage, controls the position of the fulcrum, about which the valve lever turns. The farther this fulcrum moves to the right, the greater will be the opening of the inlet-valve and the greater, consequently, also, the amount of mixture, proportioned in passing the valve, admitted to the cylinder.

It will be noted that all of the above examples of quantity governing throttle the mixture throughout the suction stroke. It was pointed out, in the discussion on various systems of governing, that the system employing quantity regulation with cut-off was superior to that which throttles the charge throughout the stroke, and it seems surprising, therefore, that this system is not in more extended use. There is to the writer's knowledge only

* K. Reinhardt, Bi-monthly Bulletin of the Am. Inst. of Mining Engineers, November, 1906, p. 1092

one engine in this country employing the cut-off system, the Sargent complete expansion engine. This engine employs a Rites inertia governor which acts upon the lay shaft driving gear which is loose on the crank shaft. As the load increases or decreases, the governor retards or advances this gear relative to the crank shaft, thus affecting all of the valve events depending upon the lay shaft at the same time. Since near the end of the stroke, however, the piston velocity is comparatively small, a relatively

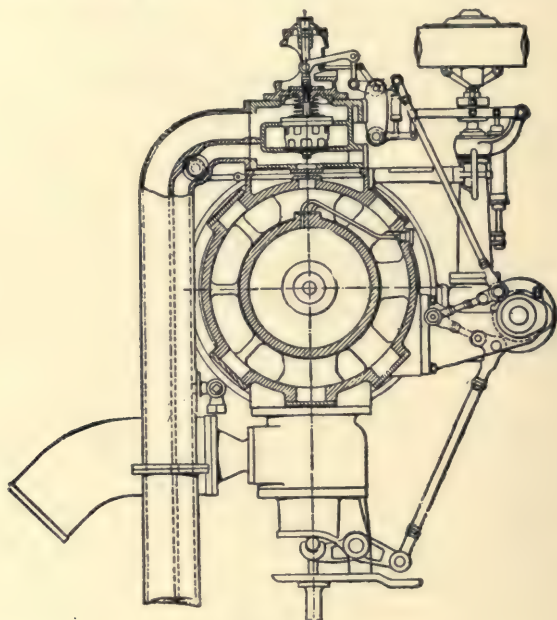


FIG. 14-23.

large advance or retardation of the gear will not affect the exhaust events or the beginning of the suction stroke very much. But the cut-off occurs along the suction line somewhere near the time when the piston has its maximum velocity, and hence a relatively small advance or retardation of the gear is sufficient to change the cut-off materially and hence to control the speed.

(e) COMBINATION QUANTITY AND QUALITY REGULATION. — The purpose of these combination systems has already been

pointed out. Fig. 14-24 shows the method as carried out by Reichenbach.* The governor, through the rod, *c*, controls the position of the bell crank lever, *d*. This lever has two slotted arms, *a* and *b*. The former, by rod and lever, regulates the position of the mixing valve, while the latter, by a similar arrangement, controls the position of the throttle valve. Suppose now that the sliding blocks *a* and *b* are in the position shown in the figure. Under these circumstances, the movement of the throttle valve is insignificant

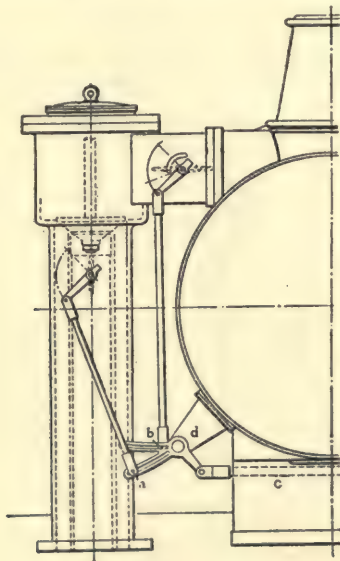


FIG. 14-24.

because *b* is so close to the center. Hence in this position the engine is practically governed by quality regulation. If the positions of *a* and *b* were reversed, that is, *b* at the outer end of its slot and *a* at the inner end, there would be practically quantity regulation. In actual operation the positions of *a* and *b* in their respective slots are determined by trial and depend upon the gas used. The final adjustment is such that for the upper ranges of load the quality of the mixture is changed, while for the lower loads the throttle valve comes into action while the mixture is

* Power, July, 1906.

held at practically constant composition. In addition to this, Reichenbach also puts the point of ignition under governor control, advancing it as the load drops.

Reinhardt's method of governing does not come strictly under any of the heads so far discussed. The inlet valve is shown in Fig. 14-25.* The valve opens and closes with the beginning and end of the suction stroke. Above the valve there are arranged a series of ports, I for gas and II and III for air. The flow of gas

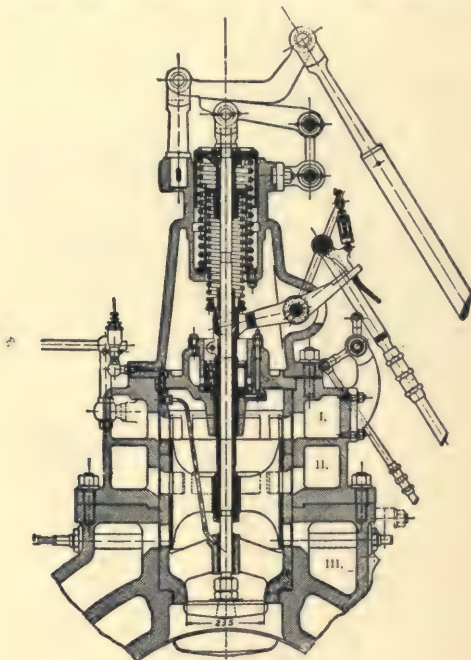


FIG. 14-25.

or air from these ports into the valve chamber proper is controlled by a slide guided by the valve spindle. At the beginning of the suction stroke the slide only frees the ports from chamber III, and hence only air is admitted. At some point in the stroke, however, as determined by the governor, the slide is suddenly released and forced down by springs. This closes ports III and opens gas ports I and air ports II. Mixture of the proper proportion then

* Reinhardt, Bi-monthly Bulletin, Am. Inst. Mining Engs., Nov., 1906.

enters the cylinder until the end of the suction stroke. As far as the admission of a varying quantity of constant mixture is concerned, this system is pure quantity regulation; but since the amount of air drawn from III increases as the load decreases, the mixture in the cylinder as a whole grows leaner as the load drops. It is claimed, however, that combustion is good even under friction load, because although the amount of constant mixture drawn in is small, more or less stratification of the charge ensures the presence of a fairly rich mixture around the igniter. The system has the advantage over pure quantity regulation in that the compression pressure remains practically unchanged.

In general principle the above method of Reinhardt seems to be a modification of the method of Mees patented in 1901.

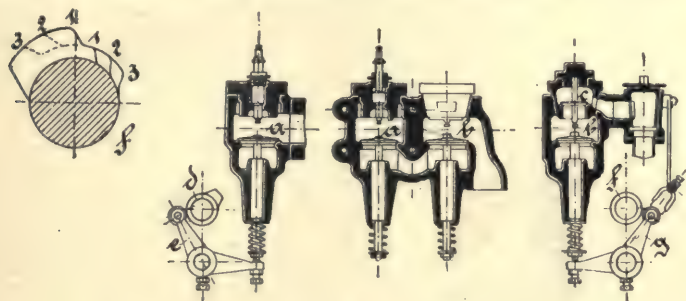


FIG. 14-26.

Letombe, recognizing the serious effect on thermal efficiency of the decrease in compression which accompanies pure quantity regulation, goes one step further and so arranges the valves that the lean mixtures under low loads are more strongly compressed than the rich mixtures at the higher loads. This is thermally an important step in the right direction. In Fig. 14-26,* *a* is the main inlet valve operated at a constant lift by the cam, *d*, through the lever, *e*. Ahead of this inlet valve are the air valve, *b*, and the gas valve, *c*. Valve *b* is operated through the lever, *g*, by the cam, *f*. Gas valve *c* opens only when pushed up by *b*. Cam *f* is shown in greater detail at the left. It has a number of elevations, each of which consists of two steps of different lengths, as shown at 1, 2, and 3. At full load the roller on the lever, *g*, mounts

* Schöttler, Die Gasmaschine.

the higher step of elevation, 1, opening both *b* and *c* wide, and the mixture, of constant proportion, enters the cylinder through *a*, which has been opened at the same time. After awhile the roller drops to the lower step of elevation, 1, air valve, *b*, closes partly, closing gas valve, *c*, completely. Only air is then drawn into the cylinder until the roller, *g*, drops off the cam altogether. Now suppose that the load drops. The governor then pulls over the roller, *g*, until it comes in line with elevation 2 or 3, as the case

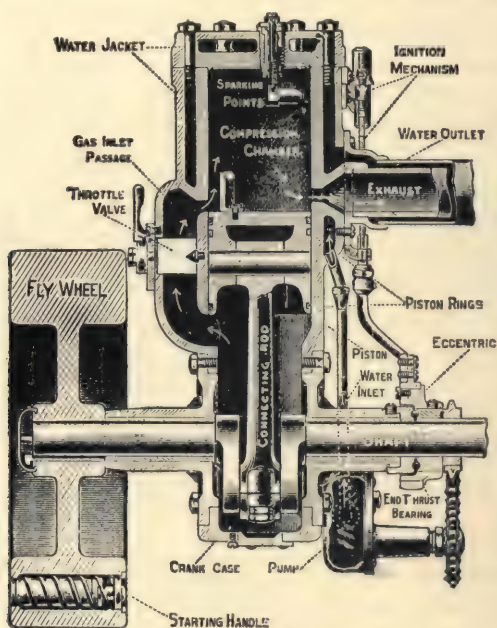


FIG. 14-27.

may be. The time of opening of the gas valve, *c*, is then shortened, because the higher step of the elevation is shorter; but the time of admission of air alone to the cylinder is lengthened, because the lower step of the cam is longer than it was before. Hence, although the mixture is leaner, the total charge volume is greater than at full load, and the leaner mixtures therefore receive a higher compression. Theoretically, this should give increased thermal efficiencies at low loads as compared with other systems

of regulation. In practice, however, for reasons not explained, there does not seem to be much difference.

(f) GOVERNING OF TWO-CYCLE ENGINES. — Fig. 14-27 shows a type of two-cycle engine, the Lozier,* much used for small motor boat work. As in all machines of this kind, the mixture is formed and compressed in the crank-case, and flows from here through a communicating passage into the cylinder, as shown by the arrows. Speed is controlled simply by the throttle valve in the passage, thus controlling the amount of mixture entering the cylinder. This is a very simple case, because the load on the engine is practically constant.

In the new Buckeye two-cycle engine, which, as far as size is concerned, is intermediate between the small two-cycle gasoline engine and the very large two-cycle blast furnace gas engines of the Koerting or Oechelhäuser type, the forward end of the cylinder is used as a mixture pump. Regulation is effected by means of the balanced throttle valve 60, Fig. 12-31; p. 289, which, under the control of the fly-ball governor, regulates both the suction to the pump and the delivery from the pump to the combustion chamber. Thus this engine governs by varying the quantity of the mixture; but since the cylinder is always thoroughly scavenged by fresh air, the mixture will naturally grow leaner as the load drops, while the compression pressure remains practically the same.

The governing details of the Oechelhäuser two-cycle machine vary somewhat, depending upon the firm building them. Fig. 14-28 † shows the scheme adopted by Borsig. It may be remembered that in this machine air and mixture enter the cylinder through separate rings of ports in the side of the cylinder, as soon as the piston uncovers them. The air ports are uncovered first to admit the scavenging air, the mixture ports a little later. Each ring of ports is surrounded by a receiver into which the air and gas pumps deliver their charges under some compression. It is evident that, since the time taken by the piston to uncover and cover the gas and air ports is practically constant, the amount of mixture that can enter the cylinder during a given time must depend upon the pressure in these receivers, and upon the port area.

* *Power Quarterly*, Oct. 15, 1900.

† Riedler, *Gross-Gasmaschinen*, p. 67.

The governor, therefore, is made to act upon the gas and air pumps to control the pressure in the receivers. Further, as the load decreases and approaches the friction load, the amount of mixture entering becomes so small as compared with the amount of air in the cylinder that ignition is likely to become difficult. For this reason the mixture ports are surrounded by a slide under governor control, as shown at A, Fig. 14-28, which is operated in such a way that as the load decreases the ports opposite the igniter are

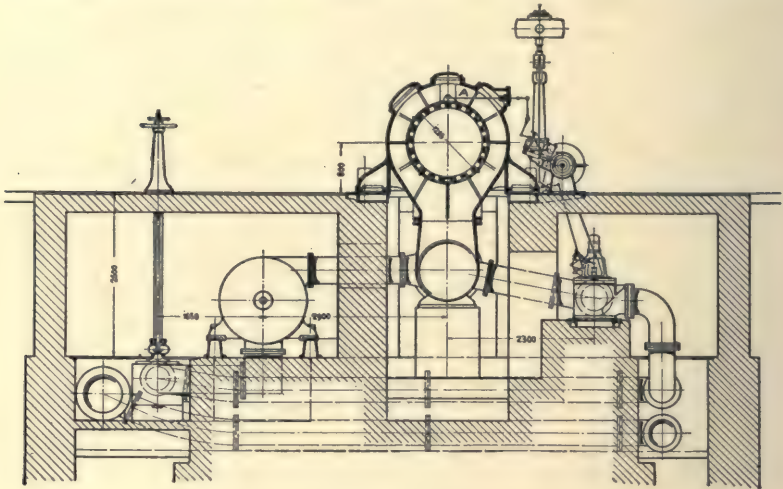


FIG. 14-28.

gradually closed first, thus insuring a comparatively rich mixture around the igniter at all times.

The method of governing the Koerting two-cycle engine also differs somewhat, depending upon the manufacturing firm. In all designs there are a gas and an air pump, of which the air pump delivers its charge into the cylinder from the commencement of its discharging stroke in order to scavenge out the cylinder. The gas pump runs idle for a part of its discharge stroke, generally forcing the gas back into the suction main, until, at a point determined by the governor, the overflow valve closes, and the gas pump delivers the rest of its charge into the cylinder through a valve on top. Fig. 14-29 * shows the pump construction used by

* F. E. Junge in *Power*, November, 1906.

CHAPTER XV

THE ESTIMATION OF POWER OF GAS ENGINES

It is intended in this chapter to point out the various methods in use by which the power that a given engine may be expected to develop can be computed, or, what amounts to the same thing, to determine the cylinder dimensions for any given power.

In steam engine practice this is a comparatively simple matter. It is necessary merely to lay down an ideal indicator card along well-defined lines, and then, by the use of card factors closely fixed by long practical experience, to determine the probable mean effective pressure in the cylinder or cylinders.

Although this method is by some writers advocated also for the gas engine, the determination of the card factor for gas engines is based upon so many component factors, which in turn depend altogether upon the judgment of the designer, that the result is anything but certain. The difficulties attaching to this method are pointed out in greater detail below.

Before describing the various methods of computing cylinder dimensions it is well to examine briefly into the allowable piston speed. The theoretical limit of piston speed of course depends directly upon the time of explosion, and this in turn depends upon the kind of mixture and to a certain extent upon the method of ignition. As far as modern practice goes the upper limit to-day seems to be at about 800 feet per minute. A certain formula, empirical as far as the writer is aware, makes the piston speed depend upon the horse-power of the engine, stating that

$$\text{piston speed} = (660 + 6 \sqrt{\text{B. H. P.}_n}) \text{ ft. per min.} \quad (1)$$

in which B. H. P._n is the normal brake horse-power.

A similar empirical formula for the revolutions per minute

$$\text{is} \quad \text{r.p.m.} = 65 + \frac{1200}{\sqrt{\text{B.H.P.}_n}} \quad (2)$$

For stationary and especially low-compression engines the values of equations (1) and (2) should not be exceeded. For automobile and other high-speed work the revolutions may be increased up to 1.35 times the value computed from equation (2).

I. The First Method of Horse-power Computation is more or less empirical in that it depends either upon the outright assumption of the mean effective pressure, or upon the computation of this factor from empirical or semi-empirical formulæ. Thus Grover bases the determination of the M. E. P. upon the compression pressure, making

$$\text{M. E. P.} = 2 C - .01 C^2 \quad (3)$$

where C = the pressure of compression in pounds above atmosphere. This formula is derived from an examination of a large number of indicator diagrams, but leaves out of consideration the kind of fuel or the quality of the mixture. Further, the formula gives the maximum result when the compression pressure is 100 pounds by gage. Beyond this the M. E. P. drops. The results are therefore at best approximate and in some cases absurd.

S. A. Moss, in an article entitled "Rational Methods of Gas Engine Powering," in *Power*, July, 1906, goes further than Grover and, in fixing upon the probable M. E. P., takes into account the size of engine as well as the kind of fuel. The cooling loss is relatively less in large than in small engines, hence the M. E. P. may be expected to increase somewhat with the size of the engine everything else remaining the same. The kind of fuel has an influence upon the power developed in any given cylinder due to the fact that the best air-fuel ratios, *i.e.*, mixtures, of the different fuels have a different heat content per cubic foot. Some of the mixtures are poorer, some richer than the average, as computed in Chapter X. Presupposing equal completeness of combustion, the richer mixtures yield the higher mean pressures, and hence the greater power. The following table shows the probable M. E. P., as computed by Moss. Only four-cycle engines are considered and the fuel is assumed to be average natural gas or illuminating gas (average heat content 87 B. T. U. per cu. ft. of mixture).

TABLE I. — VALUES OF MEAN EFFECTIVE PRESSURE FOR GAS ENGINES

Compression Pressure, Lbs. per Sq. Inch Gage	Probable Clearance as % of Displaced Volume	MEAN EFFECTIVE PRESSURES, LBS. PER SQ. INCH.						
		Approximate Brake Horse-Power of Engine						
		5 or less	10	25	50	100	200	500
50	40	60	65	70	75			
60	35	65	70	75	80			
70	30	70	75	80	85	85	90	95
80	28	70	75	85	90	90	95	100
90	26			90	95	95	100	105
100	24			95	95	100	100	110
110	22			95	95	100	100	110
120	20					100	100	110

If some other kind of fuel than the two above specified is used, the M. E. P. of Table I should be multiplied by the proper factor from Table II following:

TABLE II. — RATIOS OF HEAT OF COMBUSTION PER CUBIC FOOT OF PERFECT MIXTURE, TO VALUE FOR AVERAGE NATURAL GAS OR MANUFACTURED ILLUMINATING GAS (87).

Oil Gas	1.00
Water Gas (uncarbureted)	1.00
Coke Oven Gas	.93
Air Gas (Siemens Producer Gas)	.79
Carbureted Water Gas	1.05
Anthracite Producer Gas	.80
Bituminous Producer Gas	.87
Blast Furnace Gas	.67
Acetylene Gas	1.28
Gasoline (liquid, vapor, or vaporized and mixed with air)	1.12
Kerosene (vaporized alone or mixed with air), and entering cylinder at about 150° F.	.90

That the maximum compression pressure allowable differs for every fuel has already been pointed out in Chapter IV, where a table giving the compression pressures ordinarily employed will be found.

Having thus determined the probable M. E. P., the indicated horse-power developed at normal load by any given engine may then be computed from the formula

$$I. H. P._n = \frac{(M. E. P.) \text{ } lan}{33000} \quad (4)$$

where l = stroke in ft.

a = piston area in square inch.

and n = no. of explosions per minute = about $\frac{r.p.m.}{2}$ in a four-cycle machine at full load.

It will generally be found that at full load the number of explosions in a hit-and-miss engine is not quite equal to $\frac{r.p.m.}{2}$ or that in a machine governed by any of the precision methods the card given at full load is not quite the maximum card obtainable. This is done in both cases to give the engine some overload capacity, say from 10 to 20 per cent. Assuming an average overload capacity of say 15 per cent, we then have the

$$I.H.P._n = \frac{I.H.P._m}{1.15} \quad (5)$$

where $I.H.P._n$ = normal or full load I. H. P., and

$I.H.P._m$ = maximum I. H. P. obtainable.

Now at full load the mechanical efficiency of an average engine may be assumed to be 80 per cent, so that the normal brake horse-power will be

$$B.H.P._n = .8 I.H.P._n = \frac{.8}{1.15} I.H.P._m = .695 I.H.P._m \quad (6)$$

$$\text{or } I.H.P._m = 1.44 B.H.P._n \quad (7)$$

In the case of an engine to be constructed, $B.H.P._n$ is usually specified, hence $I.H.P._n$ can be found from equation (6). Substituting this in equation (4) leaves as the unknown factors in this equation the factors l , a , and n . The latter depends upon the revolutions, which may also be specified or determined from equation (2). This leaves only l and a to be determined.

It is next necessary to find some relation between cylinder diameter and stroke. An examination of existing engines shows that practice in this regard is pretty well settled. High-speed engines show a ratio of $\frac{\text{stroke}}{\text{diameter}} = 1.1$ to 1.3. In medium speed engines the ratio varies from about 1.2 to 1.6, and large machines vary between 1.5 and 2.0.

To determine the cylinder dimensions, let d be the diameter of the cylinder in feet and let

$$l = x d \quad (8)$$

Then equation (4) may be rewritten for *maximum* power,

$$\begin{aligned} \text{I.H.P.}_m &= \frac{(\text{M. E. P.}) \times x d \times .785 d^2 \times 144 \times n}{33000} \\ &= \frac{(\text{M. E. P.}) x d^3 n}{300} \end{aligned} \quad (9)$$

From this the cylinder diameter is

$$d = \sqrt[3]{\frac{300 \text{ I.H.P.}_m}{(\text{M.E.P.}) x n}} \text{ ft.} \quad (10)$$

EXAMPLES. — 1. To show the method of using these equations, suppose it is desired to determine the required cylinder dimensions for a 100 B. H. P., four-cycle throttling engine, using anthracite producer gas.

From equation (2), the number of revolutions should be about

$$\text{r.p.m.} = 65 + \frac{1200}{\sqrt{100}} = 185$$

From the table of Chapter IV, a compression pressure of 120 pounds by gage is about normal for the fuel used. From Tables I and II we should expect a mean effective pressure for this case of

$$\text{M. E. P.} = 100 \times .80 = 80 \text{ pounds.}$$

From equation (7), $\text{I. H. P.}_m = 1.44 \text{ B. H. P.}_n = 1.44 \times 100 = 144$

The engine being of the throttling type,

$$n = \frac{\text{r.p.m.}}{2} = \frac{185}{2} = 92.5$$

and x may be assumed equal to 1.6.

Substituting these values in equation (10), we finally have

$$\begin{aligned} \text{cyl. dia. } d &= \sqrt[3]{\frac{300 \times 144}{80 \times 1.6 \times 92.5}} = \sqrt[3]{3.65} = 1.54 \text{ ft.} \\ &= \text{Appr. } 18\frac{1}{2}'' \end{aligned}$$

$$\text{Stroke} = 1.6 \times 18.5'' = \text{app. } 29\frac{5}{8}''$$

and $\text{r.p.m.} = 185$

2. A four-cycle gasoline engine has a cylinder diameter of 6", a stroke of $8\frac{1}{2}$ ", and the number of revolutions is 250 per minute. It is governed by hit and miss. What is its probable brake horse-power at normal load?

The M. E. P. at normal I. H. P. from Tables I and II is about

$$1.12 \times 75 = 84 \text{ pounds}$$

assuming that the compression pressure is about 70 pounds by gage. For the maximum indicated horse-power, the number of explosions n may be assumed to be equal to $\frac{\text{r.p.m.}}{2} = \frac{250}{2} = 125$.

Substituting in equation (4)

$$\begin{aligned} \text{I. H. P.}_m &= 1.15 \text{ I. H. P.}_n = 144 \text{ B. H. P.}_n = \frac{84 \times \frac{8.5}{12} \times 785 \times 6^2 \times 125}{33000} \\ &= 6.35 \end{aligned}$$

$$\text{From which} \quad \text{B. H. P.}_n = \frac{6.35}{1.44} = \text{appr. } 4.4$$

II. The Second Method of determining mean effective pressure follows steam engine practice, and is developed in detail by Lucke in his Gas Engine Design. The method consists of drawing first a standard air reference diagram. From this the mean effective pressure is determined by multiplying the mean effective pressure of the standard diagram by a card or diagram factor depending upon the kind of fuel used and the compression carried. Thus fundamentally this method does not differ greatly from that of Moss outlined above.

The standard air reference diagram is obtained by working a pound of air, which is assumed to receive as much heat as a pound of the proper fuel-air mixture to be used in the real engine would contain, through the cycle.

The following table shows the card factors as determined by Lucke from a large number of constructed engines:

Kind of Fuel and Method of Use	Range of Compression by Gage	Card Factor %
Kerosene, when previously vaporized	45-75	30-40
Kerosene, injected on hot bulb, may be as low as		20
Gasoline, used in carbureter requiring a vacuum, depending upon the extent of the vacuum		25-40
Gasoline, with but little initial vacuum	80-130	50-30
Producer gas	100-160	56-40
Coal gas	Av. 80	Av. 45
Blast furnace gas	130-180	48-30
Natural gas	90-140	52-40

NOTE. — Card factors for two-cycle engines may be taken as .8 that for four-cycle machines.

The mean effective pressure of the standard air reference diagram of course increases continuously with the compression. The above table, however, shows that the card factor decreases as the compression increases for all fuels except kerosene. This accounts for the fact that in Moss's Table I the M. E. P. does not increase beyond certain compressions. The thermal and mechanical reasons underlying this fact have been explained in Chap. IV.

The table of card factors shows a large range of variation in the different factors and emphasizes the point made by Lucke that in determining cylinder sizes a great deal must be left to the personal experience and judgment of the designer.

III. Third Method of Determining Cylinder Dimensions. It must be obvious from what has been said that it is not at all easy to predict with fair accuracy the mean effective pressure an engine may be expected to realize. For that reason a method of determining the cylinder dimensions of a proposed engine, or the power of an existing engine, without first determining the M. E. P., should be welcome.

The method, developed by Güldner,* is based upon two well-known facts:

1. The power developed by any gas engine depends directly upon the volume of the mixture it can handle in unit time.
2. The power also depends upon the thermal efficiency with which the engine can handle this volume of mixture.

Regarding the first point, the charge volume in unit time

* Güldner, Entwerfen & Berechnen der Verbrennungsmotoren.

involves cylinder diameter, stroke, and number of revolutions. Two of these factors can usually be fixed upon, which determines the third. The computation of the charge volume for a given time involves the assumption of a value for the volumetric efficiency of the suction stroke. There is now so much experimental data regarding this point that no great error can be made. A table of volumetric efficiencies E_v for various types of engines is given on page 86.

The same is true of the thermal efficiency and fuel characteristics. A large number of tests on various types of machines and different fuels enables us to-day to forecast with a fair degree of accuracy what thermal efficiency may be expected from a given engine when its approximate power and the kind of fuel used are known.

The following derivation of equations for values of diameter, stroke and revolutions per minute is due to Güldner,* as is also the appended table. All metric measurements and values have been transformed to English units:

Let B. H. P._n or N_n = nominal brake horse-power.

n = R. P. M.

d = piston diameter in feet.

l = stroke in feet.

V_h = $.785 d^2 l$ = piston displacement per stroke in cu. ft.

$V = E_v V_h$ = actual volume of mixture in cu. ft. per suction stroke, barometer 28.95", temperature 59 degrees Fahrenheit.

$E_v = \frac{V}{V_h}$ = volumetric efficiency of suction stroke.

L = the volume of air in cu. ft. required for 1 cu. ft. of gas fuel, or 1 pound of liquid fuel, under most favorable practical conditions. This is not the theoretical quantity, but some excess quantity as seems best for the particular fuel.

* The article may be found either in the Zeitschrift des Vereines deutscher Ingenieure, April 26, 1902, or in his book, Entwerfen und Berechnen der Verbrennungsmotoren, pp. 213-215.

L_h = the resulting actual quantity of air in cu. ft. for one explosion, for nominal horse-power.

C_s = the quantity of fuel used per hour, for gases in cu. ft., for liquids in pounds, at nominal horse-power.

C = the same per horse-power hour.

C_h = the same per explosion.

H = the lower heating value of the fuel, for gases per cu. ft., for liquids, per pound, in B. T. U.

$$e = \frac{N_n \times 33000 \times 60}{778 C_s H} = \frac{2545 N_n}{C_s H} = \left\{ \begin{array}{l} \text{thermal efficiency at the} \\ \text{brake} = \text{economic efficiency.} \end{array} \right.$$

From the above we can derive for four-cycle engines

$$C_s = \frac{N_n \times 33000 \times 60}{778 H_e} = \frac{2545 N_n}{H_e} \quad (11)$$

$$C_h = \frac{2 C_s}{60 n} = \frac{2 \times 2545 N_n}{60 n H_e} = \frac{84.8 N_n}{n H_e} \quad (12)$$

$$L_h = \frac{C_s L}{30 n} = \frac{2545 N_n L}{30 n H_e} = \frac{84.8 N_n L}{n H_e} \quad (13)$$

For two-cycle engines equations (12) and (13) should be divided by 2, because there is a charging stroke for every revolution.

ENGINES FOR GAS FUEL

The actual charge taken in by the engine during one suction stroke must be

$$V = C_h + L_h$$

This volume requires a piston displacement of

$$\begin{aligned} V_h &= .785 d^2 l = \frac{C_h + L_h}{E_v} = \frac{84.8 N_n + 84.8 N_n L}{n H_e E_v} \\ &= \frac{84.8 N_n (1 + L)}{n H_e E_v} \text{ cubic feet} \end{aligned} \quad (14)$$

Solving (14) for d , l and n in turn, we have

$$d = \sqrt{\frac{108 N_n (1 + L)}{enHlE_v}} \text{ feet} \quad (15)$$

$$l = \frac{108 N_n (1 + L)}{enHd^2E_v} \text{ feet} \quad (16)$$

$$n = \frac{108 N_n (1 + L)}{elHd^2E_v} \text{ feet} \quad (17)$$

ENGINES FOR LIQUID FUEL

In this case the fuel is introduced either in liquid form or in the shape of vapor. In either case the volume ratios of fuel to air are very much smaller than they are in the case of gaseous fuels. Take for instance the case of alcohol, a fuel of low heating value. Here we find that the vapor does not form theoretically more than 4 per cent of the volume of the charge. In reality it is even less than this on account of the excess air used. In view of this fact we may put for these engines

$$V_h = .785 d^2 l = \frac{L_h}{E_v} = \frac{84.8 N_n L}{n H e E_v} \text{ cubic feet} \quad (18)$$

Solving this again for d , l and n we obtain for engines using liquid fuel:

$$d = \sqrt{\frac{108 N_n L}{enHlE_v}} \text{ ft.} \quad (19)$$

$$l = \frac{108 N_n L}{enHd^2E_v} \text{ ft.} \quad (20)$$

$$\text{and} \quad n = \frac{108 N_n L}{elHd^2E_v} \quad (21)$$

In these equations the quantities to which certain values must be assigned are L , e and E_v . As stated before, there is now so much practical data at hand that the proper determination of these should cause no difficulty. For values of E_v see Table, p. 86. The table on the opposite page gives values of L , C and e , as found for various fuels.

Since the nominal brake horse-power N_n is 18 to 20 per cent below the maximum capacity of engines in ordinary cases, it is well to assume an excess of air of about 30 per cent at the outset, in the case of machines using gas. In the case of liquid fuel

LOWER HEATING VALUE (H) B.T.U. PER		AIR REQUIRED, CUBIC FEET			CONSUMPTION OF FUEL (C) PER BRAKE HORSE-POWER HOUR, AND THERMAL BRAKE EFFICIENCY (e), WHEN $N_n =$																	
					5 B.H.P.				10 B.H.P.				25 B.H.P.				50 B.H.P.				100 B.H.P. and over	
					(C)		(e)		(C)		(e)		(C)		(e)		(C)		(e)		(C)	
Cu. ft.	Lb.	Cu. ft.	Lb.	Actually (L) per	Cu. ft.	Lb.	Cu. ft.	Lb.	Cu. ft.	Lb.	Cu. ft.	Lb.	Cu. ft.	Lb.	Cu. ft.	Lb.	Cu. ft.	Lb.	Cu. ft.	Lb.		
I. Illuminating gas																						
a. Lean		505	5.5	7.5	24.8	20	22.2	22	20.4	24	19.1	26	18.5	27								
b. Ordinary		560	to	to	22.2	20	20.1	22	18.3	24	16.9	26	16.5	27								
c. Rich		618	6.5	10.0	20.4	20	18.3	22	16.9	24	15.5	26	15.1	27								
d. Rich		675	6.5	10.0	18.7	20	16.7	22	15.5	24	14.1	26	13.7	27								
II. Generator gas																						
(a) On basis of																						
anthracite		13400	85	1.1	106.0	17	95.3	19	84.8	21	77.6	23	73.2	24								
(b) Anthracite gas		141	85	1.1	106.0	17	95.3	19	84.8	21	77.6	23	73.2	24								
(c) On basis of coke		12950	to	to	116.0	17	102.0	19	91.8	21	84.8	23	81.3	24								
(d) Coke gas		129	1.00	1.4	116.0	17	102.0	19	91.8	21	84.8	23	81.3	24								
III. Blast furnace gas		106	75	1.0-1.2	116.0	17	102.0	19	91.8	21	84.8	23	81.3	24								
IV. Coke oven gas		505	5.3	7.0	116.0	17	102.0	19	91.8	21	84.8	23	81.3	24								
V. Kerosene		18900	185.0	257-353	116.0	17	102.0	19	91.8	21	84.8	23	81.3	24								
VI. Crude oil (Diesel)		18000	176	288-323	116.0	17	102.0	19	91.8	21	84.8	23	81.3	24								
VII. Gasoline		19800	176	240-323	116.0	17	102.0	19	91.8	21	84.8	23	81.3	24								
VIII. Alcohol (90% vol.)		10300	96.5	128-193	116.0	17	102.0	19	91.8	21	84.8	23	81.3	24								
					116.0	17	102.0	19	91.8	21	84.8	23	81.3	24								
					116.0	17	102.0	19	91.8	21	84.8	23	81.3	24								
					116.0	17	102.0	19	91.8	21	84.8	23	81.3	24								
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					116.0	17	102.0	19	91.8	21	84.8	23	81.3	24								
					116.0	17	102.0	19	91.8	21	84.8	23	81.3	24								
					116.0	17	102.0	19	91.8	21												

engines a greater excess, 50 to 60 per cent, should be used, because these fuels are usually high in heating value, and if any economical degree of compression is to be employed, the fuel mixture should be rather lean. Besides it is not easy to produce a uniform fuel mixture in the case of liquid fuels, and an excess of air would help to overcome this difficulty. These excess air allowances have been made in the table. The values of C and e are for the various fuels and for various sizes of engines, as determined from existing data. The fuel consumed by heating apparatus and ignition flames is not included in C . In the case of engines using generator gas it is supposed that suction generators are employed requiring no fuel for a separate boiler.

To see how the results obtained by Güldner's method agree with those obtained by Moss, the same examples will be taken.

EXAMPLE 1. — Engine to be 100 B. H. P., throttling governor, fuel to be anthracite producer gas. Determine the cylinder dimensions, assuming as before that the number of revolutions is 185 per minute.

For this case any one of the formulæ 15 to 17 may be used. Taking the first, we have

$$d = \sqrt{\frac{108 N_n (1 + L)}{enHE_v}} \text{ feet}$$

in which $N_n = 100$ B. H. P.

L = air used per cu. ft. of the fuel gas = say 1.3 for anthracite gas from the table.

e = thermal efficiency that may be expected = .24 from the table.

n = 185.

H = heating value of fuel = 141 B. T. U. from the table.

l = Stroke = $x d$ = assumed as 1.6 d in previous case.

E_v = volumetric efficiency = say .90 from table p. 86.

$$\text{Then } d^2 = \frac{108 \times 100 \times 2.3}{.24 \times 185 \times 141 \times 1.6 d \times .90}$$

$$\text{or } d^3 = \frac{108 \times 100 \times 2.3}{.24 \times 185 \times 141 \times 1.6 \times .90} = 2.77$$

from which $d = 1.42$ ft. = appr. 17", and

$$l = 1.6 d = 1.6 \times 17'' = \text{appr. } 27\frac{1}{2}''.$$

This method, therefore, apparently calls for a smaller engine.

EXAMPLE 2.—Gasoline engine, cyl. diam. 6", stroke $8\frac{1}{2}''$, r.p.m. 250. Hit-and-miss governor. Determine probable B. H. P.

For this, equation 21 may be rewritten

$$N_n = \frac{nelHd^2E_v}{108 L}$$

in which $n = 250$ r.p.m.

$e =$ thermal efficiency = .19 from table.

$l =$ stroke = $8\frac{1}{2}'' = .71$ ft.

$H =$ heating value of fuel = 19800, from table.

$d = 6'' = .5$ ft., $d^2 = .25$.

$E_v = .75$, from p. 86, since this engine is likely to have an automatic inlet valve.

$L = 250$, for a good rich mixture.

Then

$$N_n = \frac{250 \times .19 \times .71 \times 19800 \times .25 \times .75}{108 \times 250} = 4.65 \text{ B. H. P.}$$

The previous example gave 4.4 B. H. P. so that the agreement in this case is satisfactory.

IV. For determining the power rating of automobile engines, several more or less empirical formulæ are in use, two of which will be cited.

The Association of Automobile Manufacturers has fixed upon the following expression for four-cycle engines:

$$\text{B. H. P.} = \frac{d^2 N}{2.5}$$

where $d =$ diameter of cylinder in inches, and

$N =$ number of cylinders.

This is based on a speed of 1000 r.p.m. In this formula, however, the factor 2.5 results from the contraction of several other factors more or less arbitrarily assumed, and hence as far as design is concerned the formula is of little or no use.

$$\text{B.H.P.} = \frac{d^2 l N n}{14000} \left(.48 + \frac{1}{10 Cl} \right)$$

where d = cyl. diam. in inches.

l = stroke in inches.

N = no. of cylinders.

n = r.p.m., and

Cl = clearance expressed in terms of piston displacement.

The following example shows the application of the methods discussed to the determination of the power of an automobile machine.

EXAMPLE. — Cylinder diameter 4", stroke $4\frac{1}{2}$ ", r.p.m. 1000, no. of cylinders 4. Determine the probable B. H. P.

I. By determination of M. E. P.

Assume compression carried is 80 pounds, which requires about 28 per cent clearance. From Moss's Tables

$$\text{M. E. P.} = 75 \times 1.12 = 84 \text{ pounds.}$$

From equation (4) therefore

$$\text{I. H. P.}_n = \frac{84 \times \frac{4.5}{12} \times .785 \times 4^2 \times 500}{33000} = 6 \text{ H. P.}$$

Since the mechanical efficiency is in the neighborhood of .80,

$$\text{B. H. P.} = .8 \times 6 = 4.8$$

and for four cylinders

$$\text{Total B. H. P.} = 4 \times 4.8 = 19.2$$

II. By Gldner's Method:

$$\begin{aligned} N_n &= \frac{nelHd^2E_v}{108L} \\ &= \frac{1000 \times .20 \times \frac{4.5}{12} \times 19800 \times \left(\frac{4}{12}\right)^2 \times .75}{108 \times 250} = 4.6 \end{aligned}$$

and total B. H. P. = $4 \times 4.6 = 18.4$

III. By Association Formula:

$$\text{B. H. P.} = \frac{4^2 \times 4}{2.5} = \frac{64}{2.5} = 25.6$$

IV. By Rice's Formula:

$$\text{B.H.P.} = \frac{4^2 \times 4.5 \times 4 \times 1000}{14000} \left(.48 + \frac{1}{10 \times .28} \right) = 17.3$$

Of the above results, those obtained by methods I, II, and IV agree very well. Method III gives a result considerably higher, but in view of the fact that the choice of the factor 2.5 is somewhat arbitrary, this discrepancy does not mean much. If the factor had been taken equal to three, for instance, as is sometimes done, the result would be brought down to 21.3 horse-power which is not so far different from the rest.

CHAPTER XVI

METHODS OF TESTING GAS ENGINES

THE actual testing of gas engines, that is, the determination of indicated and brake horse-power, speed, fuel consumption, etc., does not differ materially from the methods long become standard in steam engine practice. But the calculations involved in the proper working up of the data obtained are enough different to have caused engineering societies at home and abroad to set up rules and regulations, so-called codes, for testing internal combustion engines.

Thus in 1898 the American Society of Mechanical Engineers appointed a committee for this purpose which rendered a final report in an Appendix to the Steam Engine Code in 1901. The British Institution of Civil Engineers followed suit, the preliminary and final reports being rendered in March and December, 1905, respectively. Finally the Verein Deutscher Ingenieure adopted a code for the testing of gas engines and gas producers, which was published in the *Zeitschrift* for November 24, 1906. The latter code was translated nearly entirely by Mr. F. E. Junge and will be found in *Power* for February, 1907.

Of these codes the American and German give specific rules for testing, the latter paying particular attention to acceptance tests of both engines and gas producers. The British report concerns itself mainly with establishing efficiency standards.

In describing the methods followed in the testing of gas engines, it was thought best to give the American code in its entirety, supplementing it by most of the references to the steam engine code contained in its original printed form. This is followed by a translation of nearly the entire German code, modified where necessary to suit the American conditions. This code is also given because it not only supplements the American code in some important details, but on account of the rules given for the testing of gas producers.

RULES FOR CONDUCTING TESTS OF GAS AND OIL ENGINES. CODE OF
1901

I. Objects of the Tests. — At the outset the specific object of the test should be ascertained, whether it be to determine the fulfilment of a contract guarantee, to ascertain the highest economy obtainable, to find the working economy and the defects as they exist, to ascertain the performance under special conditions, or to determine the effect of changes in the conditions; and the test should be arranged accordingly.

Much depends upon the local conditions as to what preparations should be made for a test, and this must be determined largely by the good sense, tact, judgment, and ingenuity of the expert undertaking it, keeping in mind the main issue, which is to obtain accurate and reliable data. In deciding questions of contract, a clear understanding in regard to the methods of test should be agreed upon beforehand with all parties, unless these are distinctly provided for in the contract.

II. General Condition of the Engine. — Examine the engine, and make notes of its general condition, and any points of design, construction, or operation which bear on the objects in view. Make a special examination of all the valves by inspecting the seats and bearing surfaces, and note their condition, and see if the piston rings are gas-tight.

If the trial is made to determine the highest efficiency, and the examination shows evidence of leakage, the valves and piston rings, etc., should be made tight, and all parts of the engine put in the best possible working condition before starting on the test.

III. Dimensions, etc. — Take the dimensions of the cylinder, or cylinders, whether already known or not; this should be done when they are hot, and in working order. If they are slightly worn the average diameter should be determined. Measure, also, the compression space or clearance volume, which should be done, if practicable, by filling the spaces with water previously measured, the proper correction being made for the temperature.

IV. Fuel. — Decide upon the gas or oil to be used, and if the trial is to be made for maximum efficiency, the fuel should be the best of its class that can readily be obtained, or one that shows the highest calorific power.

V. Calibration of Instruments used in the Tests. — All instruments and apparatus should be calibrated and their reliability and accuracy verified by comparison with recognized standards. Apparatus liable to change or to become broken during the tests, such as gages, indicator springs, and thermometers, should be calibrated both before and after the experiments. The accuracy of all scales should be verified by standard weights. In the case of gas or water meters, special attention should be given to their calibration, both before and after the trial, and at the same rate of flow and pressure as exists during the trial.

(a) **GAGES.** — For pressures above the atmosphere, one of the most convenient, and at the same time reliable, standards is the dead-weight testing apparatus which is manufactured by many of the prominent gage makers. It consists of a vertical plunger nicely fitted into a cylinder containing oil or glycerine, through the medium of which the pressure is transmitted to the gage. The plunger is surmounted by a circular stand on which weights may be placed, and by means of which any desired pressure can be secured. The total weight, in pounds, on the plunger at any time, divided by the area of the plunger and of the bushing which receives it, in square inches, gives the pressure in pounds per square inch.

Another standard of comparison for pressures is the mercury column. If this instrument is used, assurance must be had that it is properly graduated with reference to the ever varying zero point; that the mercury is pure, and that the proper correction is made for any difference of temperature that exists, compared with the temperature at which the instrument was graduated.

For pressure below the atmosphere, an air pump or some other means of producing a vacuum is required, and reference must be made to a mercury gage. Such a gage may be a U-tube having a length of 30 inches or so, with both arms properly filled with pure mercury.

(b) **THERMOMETERS.** — Standard thermometers are those which indicate 212 degrees Fahrenheit in steam escaping from boiling water at the normal barometrical pressure of 29.92 inches, the whole stem up to the 212-degree point being surrounded by the steam; and which indicate 32 degrees Fahrenheit in melting

ice, the stem being likewise completely immersed to the 32-degree point; and which are calibrated for points between and beyond these two references points. We recommend, for temperatures between 212 degrees and 400 degrees Fahrenheit, that the comparison of the thermometer be made with the temperature given in Regnault's Steam Tables, the method required being to place it in a mercury well surrounded by saturated steam under sufficient pressure to give the right temperature. The pressure should be accurately determined as pointed out in the above section (a), and the thermometer should be immersed to the same extent as it is under its working condition.

Thermometers in practice are seldom used with the stems fully immersed; consequently, when they are compared with the standard, the comparison should be made under like conditions, whatever those happen to be.

If pyrometers of any kind are used, they should be compared with a mercury thermometer within its range, and if extreme accuracy is required with an air thermometer, or a standard based thereon, at higher points, care being taken that the medium surrounding the pyrometer, be it air or liquid, is of the same uniform temperature as that surrounding the standard.

(c) INDICATOR SPRINGS. — The indicator springs should be calibrated with the indicator in as nearly as possible the same condition as to temperature as exists during the trial. This temperature can usually be estimated in any particular case. A simple way of heating the indicator is to subject it to a steam pressure just before calibration. Compressed air, or compressed carbonic acid gas, are suitable for the actual work of calibration. These gases should be used in preference to steam, so as to bring the conditions as near as possible to those which obtain when the indicators are in actual use. When compressed carbonic acid gas is used, and trouble arises from the clogging of the escape valves with ice, the pipe between the valve and the gas tank should be heated. With both air and carbonic acid gas, the pipes leading to the indicator should also be heated if it is found that they are below the required temperature. The springs may be calibrated for this class of engines under a constant pressure, if desired, and the most satisfactory method is to cover the whole range of pressure through which the indicator acts; first, by grad-

ually increasing it from the lowest to the highest point, and then gradually reducing it from the highest to the lowest point, in the manner which has heretofore been widely followed by indicator makers; a mean of the results should be taken. The calibration should be made for at least five points, two of these being for the pressures corresponding to the maximum and minimum pressures, and three for intermediate points equally distant.

The standard of comparison recommended is the dead weight testing apparatus, a mercury column, or a steam gage, which has been proved correct by reference to either of these standards.

When the scale of the spring determined by calibration is found to vary from the nominal scale with substantial uniformity, it is usually sufficiently accurate to take the arithmetical mean of the scales found at the different pressures tried. When, however, the scale varies considerably at the different points, and absolute accuracy is desired, the method to be pursued is as follows: Select a sample diagram and divide it into a number of parts by means of lines parallel to the atmospheric line, the number of lines being equal to and corresponding with the number of points at which the calibration of the spring is made. Take the mean scale of the spring for each division and multiply it by the area of the diagram inclosed between two contiguous lines. Add all the products together and divide by the area of the whole diagram; the result will be the average scale of the spring to be used. If the sample diagram selected is a fair representative of the entire set of diagrams taken during the test, this average scale can be applied to the whole. If not, a sufficient number of samples of diagrams representing the various conditions can be selected, and the average scale determined by a similar method for each, and thereby the average for the whole run.

(d) GAS METERS. — A meter used for measuring gas for a gas engine should be calibrated by referring its readings to the displacement of a gasometer of known volume, by comparing it with a standard gas meter of known error, or by passing air through the meter from a tank in which air under pressure is stored. If the latter method is adopted, it is necessary to observe the pressure of the air in the tank and its temperature, both at the tank and at the meter, and this should be done at uniform intervals during the progress of the calibration. The amount of

air passing through the meter is computed from the volume of the tank and the observed temperatures and pressures.

The volume of the gas thus ascertained should be reduced to the equivalent at a given temperature and atmospheric pressure, corrected for the effect of moisture in the gas, which is ordinarily at the saturation point or nearly so. We recommend that a standard be adopted for gas-engine work, the same as that used in photometry, namely, the equivalent volume of the gas when saturated with moisture at the normal atmospheric pressure at a temperature of 60 degrees Fahrenheit. In order to reduce the reading of the volume containing moist gas at any other temperature to this standard, multiply by the factor

$$\frac{459.4 + 60}{459.4 + t} \times \frac{b - (29.92 - s)}{29.4},$$

in which b is the height of the barometer in inches at 32 degrees Fahrenheit, t the temperature of the gas at the meter in degrees Fahrenheit, and s the vacuum in inches of mercury corresponding to the temperature of t obtained from steam tables.

(e) WATER METERS. — A good method of calibrating a water meter is the following, reference being made to Fig. 16-1.

Two tees A and B are placed in the feed pipe, and between them two valves C and D . The meter is connected between the outlets of the tees A and B . The valves E and F are placed one on each side of the meter. When the meter is running, the valves E and F are opened, and the valves C and D are closed. Should an accident happen to the meter during the test, the valves E and F may be closed,

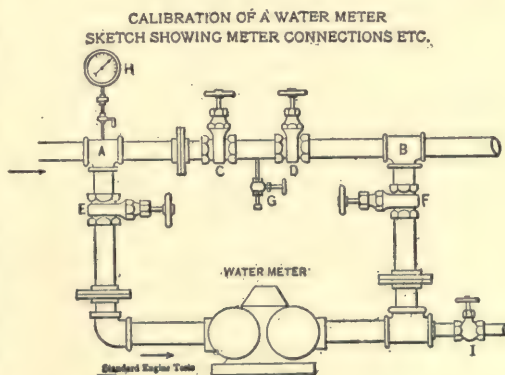


FIG. 16-1.

and the valves C and D opened, so as to allow the feed water to flow directly to the point of use. A small bleeder G is

opened when the valves *C* and *D* are closed, in order to make sure that there is no leakage. A gage is attached at *H*. When the meter is tested, the valves *C*, *D*, and *F* are closed, and the valves *E* and *I* are opened. The water flows from the valve *I* to a tank placed on weighing scales. In testing the meter the rate of flow should be the same as that on test, and the water leaving the meter is throttled at the valve *I* until the pressure shown by the gage *H* is the same as that indicated when the meter is running under the normal conditions. The piping leading from the valve *I* to the tank is arranged with a swinging joint, consisting merely of a loosely fitting elbow, so that it can be swung readily into the tank or away from it. After the desired pressure and rate of flow have been secured, the end of the pipe is swung into the tank the instant that the pointer of the meter is opposite some graduation mark on the dial, and the water continues to empty into the tank. The tests should be made by starting and stopping at the same graduation mark on the meter dial, and continued until at least 10 or 20 cubic feet are discharged for one test. The water collected in the tank is then weighed.

The water passing the meter should always be under pressure in order that any air in the meter may be discharged through the vents provided for this purpose. Care should be taken that there is no air contained in the water. The meter should be tested both before and after the engine trial, and several tests be made of the meter in each case in order to obtain confirmative results. It is well to make preliminary tests to determine whether the meter works satisfactorily before connecting it up for an engine trial. The results should agree with each other for two widely different rates of flow.

VI. Duration of Test. — The duration of a test should depend upon its character and the objects in view, and in any case the test should be continued until the consecutive readings of the rates at which oil or gas is consumed, taken at say half-hourly intervals, become uniform and thus verify each other. If the object is to determine the working economy, and the period of time during which the engine is usually in motion is some part of twenty-four hours, the duration of the test should be fixed for this number of hours. If the engine is one using coal for generating gas, the test should cover a long enough period to determine

with accuracy the coal used in the gas producer; such a test should be of at least twenty-four hours' duration, and in most cases it should extend over several days.

VII. Starting and Stopping a Test. — In a test for determining the maximum economy of an engine, it should first be run a sufficient time to bring all the conditions to a normal and constant state. Then the regular observations of the test should begin, and continue for the allotted time.

If a test is made to determine the performance under working conditions, the test should begin as soon as the regular preparations have been made for starting the engine in practical work, and the measurements should then commence and be continued until the close of the period covered by the day's work.

VIII. Measurement of Fuel. — If the fuel used is coal furnished to a gas producer, the same methods apply for determining the consumption as are used in steam boiler tests.

If the fuel used be gas, the only practical method of measurement is the use of a meter through which the gas is passed. Gas bags should be placed between the meter and the engine to diminish the variation of pressure, and these should be of a size proportionate to the quantity used. When a meter is employed to measure the air used by an engine, a receiver with a flexible diaphragm should be placed between the engine and the meter. The temperature and pressure of the gas should be measured, as also the barometric pressure and temperature of the atmosphere, and the quantity of gas should be determined by reference to the calibration of the meter, taking into account the temperature and pressure of the gas. (See Section V (*d*)).

If the fuel is oil, this can be drawn from a tank which is filled to the original level at the end of the test, the amount of oil required for so doing being weighed; or, for a small engine, the oil may be drawn from a calibrated vessel such as a vertical pipe.

In an engine using an igniting flame the gas or oil required for it should be included in that of the main supply, but the amount so used should be stated separately, if possible.

IX. Measurement of Heat-Units Consumed by the Engine. — The number of heat units used is found by multiplying the number of pounds of coal or oil or the cubic feet of gas consumed, by the total heat of combustion of the fuel as determined by a cal-

orimeter test. In determining the total heat of combustion no deduction is made for the latent heat of the water vapor in the products of combustion. There is a difference of opinion on the propriety of using this higher heating value, and for purposes of comparison care must be taken to note whether this or the lower value has been used. The calorimeter recommended for determining the heat of combustion is the Mahler, for solid fuels or oil, or the Junker for gases, or some form of calorimeter known to be equally reliable. (See Chapter VI, or Poole on "The Calorific Power of Fuels.")

It is sometimes desirable, also, to have a complete chemical analysis of the oil or gas. The total heat of combustion may be computed, if desired, from the results of the analysis, and should agree well with the calorimeter values.

In using the gas calorimeter, which involves the determination of the volume instead of the weight of the gas, it is important that this should be reduced to the same temperature as that corresponding to the conditions of the engine trial. The formula to be used for making the reduction is that already given in Section V (d).

For the purpose of making the calorimeter test, if the fuel used is coal for generating gas in a producer, or oil, samples should be taken at the time of the engine trial, and carefully preserved for subsequent determination. If gas is used, it is better to have a gas calorimeter on the spot, samples taken, and the calorimeter test made while the trial is going on.

X. Measurement of Jacket Water to Cylinder or Cylinders. — The jacket water may be measured by passing it through a water meter or allowing it to flow from a measuring tank before entering the jacket, or by collecting it in tanks on its discharge.

XI. Indicated Horse-Power. — The directions given for determining the indicated horse-power for steam engines apply in all respects to internal combustion engines.

The indicated horse-power should be determined from the average mean effective pressure of diagrams taken at intervals of twenty minutes, and at more frequent intervals if the nature of the test makes this necessary. With variable loads, such as those of engines driving generators for electric railroad work, and of rubber-grinding and rolling mill engines, the diagrams cannot be

taken too often. In cases like the latter, one method of obtaining suitable averages is to take a series of diagrams on the same blank card without unhooking the driving cord, and apply the pencil at successive intervals of ten seconds until two minutes' time or more has elapsed, thereby obtaining a dozen or more indications in the time covered. This tends to insure the determination of a fair average for that period. In taking diagrams for variable loads, as indeed for any load, the pencil should be applied long enough to cover several successive revolutions, so that the variations produced by the action of the governor may be properly recorded. To determine whether the governor is subject to what is called "racing" or "hunting," a "variation diagram" should be obtained; that is, one in which the pencil is applied a sufficient time to cover a complete cycle of variations. When the governor is found to be working in this manner, the defect should be remedied before proceeding with the test.

AUTHOR'S NOTE. — When the engine is governed by the hit-and-miss principle the diagrams taken on one card should in any case cover the series of consecutive explosions, and the mean diagram should be used as the basis of calculations.

The most satisfactory driving rig for indicating seems to be some form of well-made pantagraph, with driving cord or fine annealed wire leading to the indicator. The reducing motion, whatever it may be, and the connections to the indicator, should be so perfect as to produce diagrams of equal lengths, and produce a proportionate reduction of the motion of the piston at every point of the stroke, as proved by test.

To test the accuracy of the reducing motion without making special preparations for a thorough examination, it is sufficient to make a comparison between the actual proportion of the stroke covered and the apparent proportion measured on the indicator, and see how they agree. This may be done on a large engine by making the comparison wherever it happens to stop, and repeating the comparison when it has stopped with the piston at some other point of the stroke. With an engine which can be turned over by hand, or where auxiliary power is provided for moving it, the comparison may be made at a number of equidistant points in the stroke. To make the test properly, a diagram should be taken just before stopping, and this will serve as a

reference for the measurements taken after stopping. The actual proportion of stroke covered is determined by measuring the distance which the piston has moved and comparing it with the whole length of the stroke, making sure that the slack has all been taken up. To obtain the apparent indication from the diagram, the indicator pencil is moved up and down with the finger so as to make a vertical mark on the diagram, and the distance of this mark from the beginning of the diagram compared to the whole length of the diagram is the proportion desired.

It is necessary, of course, to go through these operations without changing in any way the adjustment of the driving cord of the indicator, or any part of the mechanism that would alter the movements of the indicator.

In the manipulation of the indicator it is important to keep the instrument in clean condition and preserve it in mechanically good order. Ordinary cylinder oil is the best material to use for lubricating the indicator piston for pressures above the atmosphere. It is better to have the piston fit the cylinder rather loosely — so as to get absolute freedom of motion — than to have a mechanically accurate fit. In the latter case, extreme care and frequent cleanings are required to obtain good diagrams. No diagrams should be accepted in which there is any appearance of want of freedom in the movement of the mechanism. A ragged or serrated line in the region of the expansion or compression lines is a sure indication that the piston or some part of the mechanism sticks; and when this state of things is revealed the indicator should not be trusted, but the cause should be ascertained and a suitable remedy applied. An indicator which is free when subjected to a steady pressure, as it is under a test of the springs for calibration, should be able to produce the same horizontal line, or substantially the same, after pushing the pencil down with the finger, as that traced after pushing the pencil up and subsequently tapping it lightly. When the pencil is moved by the finger, first up and then down, the piston being subjected to the pressure, the movement should appear smooth to the sense of feeling.

The pipe connections for indicating gas and oil engines should be removed as far as possible from the ports and ignition devices, and made preferably in the cylinder head. The pipes should be

as short and direct as possible. Avoid the use of long pipes, otherwise explosions of the gas in these connections may occur.

Ordinary indicators suitable for indicating steam engines are much too lightly constructed for gas and oil engines. The pencil mechanism, especially the pencil arm, needs to be very strong to prevent injury by the sudden impact at the instant of the explosion; a special gas-engine indicator is required for satisfactory work, with a small piston and a small spring.

See Chapter I for the description of various indicators.

XII. Brake Horse-Power. — The determination of the brake horse-power, which is very desirable, is the same for internal combustion as for steam engines.

This term applies to the power delivered from the fly-wheel shaft to the engine. It is the power absorbed by a friction brake applied to the rim of the wheel, or to the shaft. A form of brake is preferred that is self-adjusting to a certain extent, so that it will, of itself, tend to maintain a constant resistance at the rim of the wheel. One of the simplest brakes for comparatively small engines, which may be made to embody this principle, consists of a cotton or hemp rope, or a number of ropes, encircling the wheel, arranged with weighing scales or other means for showing the strain. An ordinary

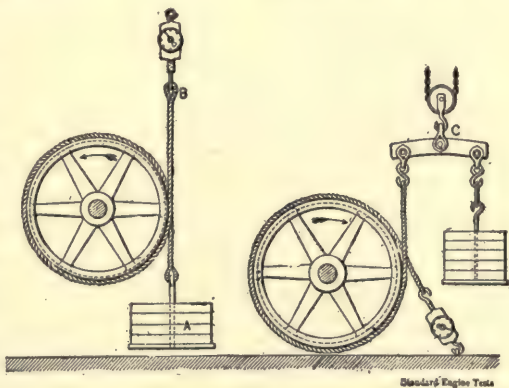


FIG. 16-2.

band brake may also be constructed so as to embody the principle. The wheel should be provided with interior flanges for holding water used for keeping the rim cool.

A self-adjusting rope brake is illustrated in Fig. 16-2, where it will be seen that, if the friction at the rim of the wheel increases, it will lift the weight A, which action will diminish the tension in the end B of the rope, and thus prevent a further increase in the friction. The same device can be used for a band brake of the

ordinary construction. Where space below the wheel is limited, a cross bar, *C*, supported by a chain tackle exactly at its center point, may be used as shown in Fig. 16-2, thereby causing the action of the weight on the brake to be upward. A safety stop should be used with either form, to prevent the weights being accidentally raised more than a certain amount.

The water-friction brake is specially adapted for high speeds and has the advantage of being self-cooling. The Alden brake is also self-cooling and is capable of fine adjustment.

A water-friction brake is shown in Fig. 16-3. It consists of two circular discs, *A* and *B*, attached to the shaft *C*, and revolving in a case, *E*; between fixed planes.

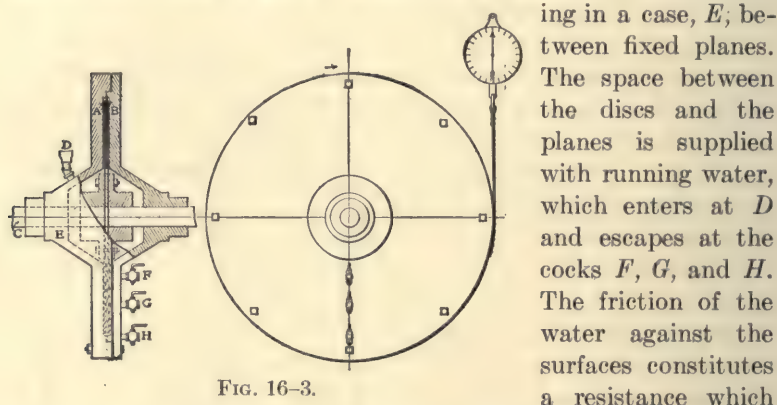


FIG. 16-3.

The space between the discs and the planes is supplied with running water, which enters at *D* and escapes at the cocks *F*, *G*, and *H*. The friction of the water against the surfaces constitutes a resistance which absorbs the desired power, and the heat generated within is carried away by the water itself. The water is thrown outward by centrifugal action and fills the outer portion of the case. The greater the depth of the ring of water, the greater amount of power absorbed. By suitably adjusting the amount of water entering and leaving any desired power can be obtained. Water-friction brakes have been used successfully at speeds of over 20,000 revolutions per minute.

For methods of computing brake horse-power see Chapter I.

XIII. Speed. — There are several reliable methods of ascertaining the speed, or the number of revolutions of the engine crank shaft per minute. The simplest is the familiar method of counting a number of turns for a period of one minute with the eye fixed on the second hand of a time piece. Another is the use of a counter held for a minute or a number of minutes against

the end of the main shaft. Another is the use of a reliable tachometer held likewise against the end of the shaft. The most reliable method, and the one we recommend, is the use of a continuous recording engine register or counter, taking the total reading each time that the general test data are recorded, and computing the revolutions per minute corresponding to the difference in the readings of the instrument. When the speed is above 250 revolutions per minute, it is almost impossible to make a satisfactory counting of the revolutions without the use of some kind of mechanical counter.

The determination of variation of speed during a single revolution, or the effect of the fluctuation due to sudden changes of the load, is also desirable, especially in engines driving electric generators used for lighting purposes. There is at present no recognized standard method of making such determinations, and if such are desired, the method employed may be devised by the person making the test and described in detail in the report.

One method suggested for determining the instantaneous variation of speed which accompanies a change of load is as follows: A screen containing a narrow slot is placed on the end of a bar and vibrated by means of electricity. A corresponding slot in a stationary screen is placed parallel and nearly touching the vibrating screen, and the two screens are placed a short distance from the fly-wheel of the engine in such a position that the observer can look through the two slots in the direction of the spokes of the wheel. The vibrations are adjusted so as to conform to the frequency with which the spokes of the wheel pass the slots.

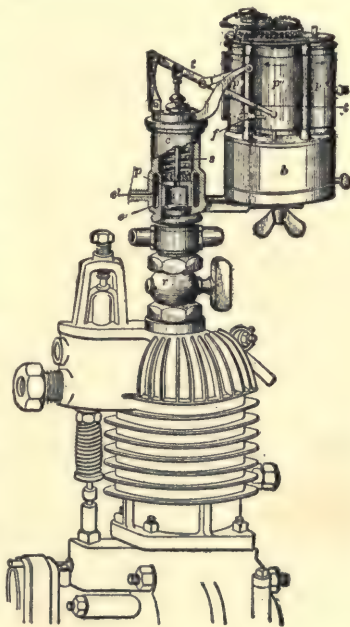


FIG. 16-4.

When this is done the observer viewing the wheel through the slots sees what appears to be a stationary fly-wheel. When a change in the velocity of the fly-wheel occurs, the wheel appears to revolve either backward or forward according to the direction of the change. By careful observations of the amount of this motion, the change of angular velocity during any given time is revealed.

Experiments that have been made with a device of this kind show that the instantaneous gain of velocity, upon suddenly removing all the load from an engine, amounted to from one-sixth to one-quarter of a revolution of the wheel.

In an engine which is governed by varying the number of explosions or working cycles, a record should be kept of the number of explosions per minute; or if the engine is running at nearly maximum load, by counting the number of times the governor causes a miss in the explosions.

One way of mechanically recording the explosions is to attach to the exhaust pipe a cylinder and piston arranged so that the pressure caused by the exhaust gases operate against a light spring and moves a register, which is provided for automatically counting the number.

AUTHOR'S NOTE. — An instrument for this purpose has been devised by R. Mathot. The following description is from his book on "Modern Gas Engines and Producer Gas Plants:"

The instrument, Fig. 16-4, is somewhat similar in form to the ordinary indicator. Its record, however, is made on a paper tape which is continuously unwound. The cylinder *c* is provided with a piston *p*, about the stem of which a spring *s* is coiled. A clock train contained in the chamber *b* unwinds the strip of paper from the roll *p'* and draws it over the drum *p''*, where the pencil *t* leaves the mark. The tape is then rewound on the spindle *p'''*. A small stylus or pencil *f* traces the atmospheric line on the paper as it passes over the drum *p''*. In order to obviate the binding of the piston *p* when subjected to the high temperature of the explosions, the cylinder *c* is provided with a casing *e* in which water is circulated by means of a small rubber tube which fits over the nipple *e'*. This recorder analyzes with absolute precision the work of all engines, whatever may be their speed. It gives a continuous graphic record from which the number of explosions, together with the initial pressure of each, can be determined, and the order of their succession. Consequently the regularity or irregularity of the variations can be observed and traced to the secondary influences producing them, such as the action of the inlet and outlet valves and the sensitiveness of the governor. It renders it possible to estimate the resistance to suction and the back pressure due to expelling the burnt gases, the chief causes of loss in efficiency in high-speed engines. Furthermore, the influence of compression is markedly shown from the diagram obtained.

The recorder is mounted on the engine; its piston is driven back by each of the explosions to a height corresponding with their force; and the stylus

or pencil controlled by the lever *t* records them side by side on the moving strip of paper. The speed with which this strip is unwound conforms with the number of revolutions of the engine to be tested, so that the records of the explosions are placed side by side clearly and legibly.

Their succession indicates not only the number of explosions and of revolutions which occur in a given time, but also their regularity, the number of misfires. The pressure of the explosions is measured by a scale connected with the recorder-spring. By employing a very weak spring which flexes at the bottom simply by the effect of the compression in the engine cylinder, it is possible to ascertain the amount of the resistance to suction and to the exhaust. It is simply sufficient to compare the explosion record with the atmospheric line, traced by the stylus *f*. By means of this apparatus, and of the records which it furnishes, it is possible analytically to regulate the work of an engine, to ascertain the proportion of air, gas, or hydrocarbon which produces the most powerful explosion, to regulate the compression, the speed, the time of ignition, the temperature, and the like.

XIV. Recording the Data. — The time of taking weights and every observation should be recorded, and note made of every event, however unimportant it may seem to be. The pressures, temperatures, meter readings, speeds, and other measurements should be observed every 20 or 30 minutes when the conditions are practically uniform, and at more frequent intervals if they are variable. Observations of the gas or oil measurements should be taken with special care at the expiration of each hour, so as to divide the test into hourly periods, and reveal the uniformity, or otherwise, of the conditions and results as the test goes forward.

All data and observations should be kept on suitable prepared blank sheets or in notebooks.

XV. Uniformity of Conditions. — When the object of the test is to determine the maximum economy, all the conditions relating to the operation of the engine should be maintained as constant as possible during the trial.

XVI. Indicator Diagrams and their Analysis. — **SAMPLE DIAGRAMS:** Sample diagrams nearest to the mean should be selected from those taken during the trial and appended to the tables of the results. If there are separate compression or feed cylinders, the indicator diagrams from these should be taken and the power deducted from that of the main cylinder.

XVII. Standards of Economy and Efficiency. — The hourly consumption of heat, determined as pointed out in Article IX, divided by the indicated or the brake horse-power, is the standard expression of engine economy recommended.

In making comparisons between the standard for internal

combustion engines and that for steam, it must be borne in mind that the former relates to energy concerned in the generation of the force employed, whereas in the steam engine it does not relate to the entire energy expended during the process of combustion in the steam boiler. The steam engine standard does not cover the losses due to combustion, while the internal combustion engine standard, in cases where a crude fuel such as oil is burned in the cylinder, does cover these losses. To make a direct comparison between the two classes of engines considered as complete plants for the production of power, the losses in generating the working agent must be taken into account in both cases and the comparison must be on the basis of the fuel used; and not only this, but on the basis of the same or equivalent fuel used in each case. In such a comparison, where producer gas is used, and the producer is included in the plant, the fuel consumption, which will be the weight of coal in both cases, may be directly compared.

The thermal efficiency ratio per indicated horse-power or per brake horse-power for internal combustion engines is obtained in the same manner as for steam engines, and is expressed by the fraction

$$\frac{2545}{\text{B.T.U. per H.P. per hour}}$$

XVIII. Heat Balance. — For purposes of scientific research, a heat balance should be drawn which shows the manner in which the total heat of combustion is expended in the various processes concerned in the working of the engine. It may be divided into three parts: first, the heat which is converted into the indicated or brake work; second, the heat rejected in the cooling water of the jackets; and third, the heat rejected in the exhaust gases, together with that lost through incomplete combustion and radiation.

To determine the first item, the number of foot-pounds of work performed by, say, one pound or one cubic foot of the fuel is determined; and this quantity divided by 778, which is the mechanical equivalent of one British thermal unit, gives the number of heat units desired. The second item is determined by measuring the amount of cooling water passed through the jackets, equivalent to one pound or one cubic foot of fuel consumed, and

calculating the amount of heat rejected, by multiplying this quantity by the difference in the sensible heat of the water leaving the jacket and that entering. The third item is obtained by the method of differences; that is, by subtracting the sum of the first two items from the total heat supplied. The third item can be subdivided by computing the heat rejected in the exhaust gases as a separate quantity. The data for this computation are found by analyzing the fuel and the exhaust gases, or by measuring the quantity of air admitted to the cylinder in addition to that of the gas or oil.

For methods of making fuel and exhaust gas computations, see Chapter VI.

XIX. Report of Test. — The data and results of a test should be reported in the manner outlined in one of the following tables, the first of which gives a complete summary when all the data are determined, and the second is a shorter form of report in which some of the minor items are omitted.

XX. Temperatures Computed at Various Points of the Indicator Diagram. — The computation of temperatures corresponding to various points in the indicator is, at best, approximate. It is possible only where the temperature of one point is known or assumed, or where the amount of air entering the cylinder along with the charges of gas or oil, and the temperature of the exhaust gases, is determined.

If the amount of air is determined for a gas engine, together with the necessary temperatures, so that the volume and the temperature of the air entering the cylinder per stroke, and that of the gas are known, we may, by combining this with the other data, compute the temperature for a point in the compression curve. In this computation we must allow for the volume of the exhaust gases remaining in the cylinder at the end of the stroke. The temperature at the point in the compression curve where it meets or crosses the atmospheric line will be given by the formula:

$$T = \frac{491.4 V'}{V'' + V''' + V'''} - 459.4 \dots (4);$$

where V' is the total volume corresponding to the point where the compression curve meets or crosses the atmospheric line; V''

the volume of the air at atmospheric pressure entering the cylinder during each working cycle, reduced to the equivalent volume at 32 degrees Fahrenheit; V''' the volume of the gas consumed per cycle reduced to the equivalent at atmospheric pressure and 32 degrees Fahrenheit; and V'''' the volume of the exhaust gases retained in the cylinder reduced to the same basis. To reduce the actual volumes to those at 32 degrees Fahrenheit, multiply by the ratios of $491.4 \div (T' + 459.4)$, where T' is the observed temperature of the air and of the gas used as fuel. For the exhaust gases retained in the cylinder at the end of the stroke T' may be taken as the temperature of the exhaust gases leaving the engine, provided the engine is not of the "scavenging" type.

Having determined the temperature of a point in the compression curve, the temperature of any point in the diagram may be found by the equation

$$T_1 = (T + 459.4) \frac{P_1 V_1}{PV} - 459.4 \dots (B).$$

Here T_1 is the desired temperature of any point in the diagram where the absolute pressure is P_1 and the total volume V_1 , and P and V are the corresponding quantities for the point in the compression line having the temperature T computed from the formula (A).

Formula (B) holds only where the weight of the gases contained in the cylinder is constant. It is also assumed in this formula that the density of the gas compared to air at the same temperature and pressure is the same before and after the explosion.

A second method may be employed, provided the air which enters the cylinder is measured. This will allow for any difference in the density of the gas before and after explosion, and more exact values for temperatures on the expansion curve may be obtained than by the first method.

In this method the density of the exhaust gases compared to air at the same temperature and pressure is computed, assuming perfect combustion, and including the effect of the water vapor present; and from this density the volume of the gases exhausted per cycle is determined. If this volume exhausted per cycle, added to the volume of the gas retained in the clearance space at the end of the stroke, be called V in equation B, and T be the

observed temperature of the exhaust gases, this equation may be used for determining the temperature of any point in the diagram in the way already described. This method is more complicated than the first, as it involves the determination of the theoretical density after explosion, but it possesses the advantage that it may be applied to an oil as well as to a gas engine.

A third method of computing the temperature of the various points in the diagram may be employed where analyses of the exhaust gases as well as of the fuel have to be made. This method is more complicated than the first, but, in common with the second, it possesses the advantage that it may be applied to an oil as well as to a gas engine.

In applying the third method the volume of the exhaust gases discharged per working cycle would be given by the formula:

$$V_2 = \frac{1}{D} (Rw + w) \dots (C)$$

where D is the density of the exhaust gases at their observed temperature, computed from the analysis, assuming the vapor of water produced through burning the hydrogen in the fuel to be in a gaseous state; R the weight of the air which enters the cylinder per pound of fuel consumed per working cycle; the value of R , providing there are no unconsumed hydrocarbons, may be computed by employing the formula:

$$R = \frac{NC}{.33 (\text{CO}_2 + \text{CO})} \dots (D)$$

where N , CO_2 and CO represent the proportions, by volume, of the several constituents of the exhaust gases, and C the weight of carbon consumed and converted to CO_2 or CO per pound of fuel burned, computed from the analysis of the fuel and of the exhaust gases.

Having determined the volume V_2 of the exhaust gases, formula (B) may be used in computing the temperature, in which case T will represent the temperature of the exhaust gases as in the second method, P the pressure of the exhaust, and V the volume of the exhaust gases V_2 discharged per stroke, added to the volume of the gases retained in the cylinder at the end of the stroke.

The value of R given in equation (D) is approximate, on

account of the fact that the percentage of N should be that due to the air alone, and not that due to the air in addition to that contained in the fuel gas. Where extreme accuracy is desired, the value found for R may be used to determine the percentage of N which in the analysis of the exhaust gases is due to the N in the fuel gas, and this value may be subtracted from the total N shown by the analysis of the fuel gases, in order to obtain the correct value of N to be used in equation (C).

TABLE NO. 1

DATA AND RESULTS OF TEST OF GAS OR OIL ENGINE

Arranged according to the Complete Form advised by the Engine Test Committee, American Society of Mechanical Engineers. Code of 1902

1. Made by of
on engine located at
to determine
2. Date of trial
3. Type of engine, whether oil or gas
4. Class of engine (mill, marine, motor for vehicle, pumping, or other)
5. Number of revolutions for one cycle, and class of cycle
6. Method of ignition
7. Name of builders
8. Gas or oil used
- (a) Specific gravity deg. Fahr.
- (b) Burning point "
- (c) Flashing point "
9. Dimensions of engine:
- 1st Cyl. 2d Cyl.
- (a) Class of cylinder (working or for compressing the charge)
- (b) Vertical or horizontal
- (c) Single or double acting
- (d) Cylinder dimensions
- Bore in.
- Stroke ft.
- Diameter piston rod in.
- Diameter tail rod in.
- (e) Compression space or clearance in per cent of volume displaced by piston per stroke ..
- Head end
- Crank end
- Average
- (f) Surface in square feet (average)
- Barrel of cylinders
- Cylinder heads
- Clearance and ports
- Ends of piston
- Piston rod

- (g) Jacket surfaces or internal surfaces of cylinder
 heated by jackets, in square feet
 Barrel of cylinder.....
 Cylinder heads
 Clearance and ports
 (h) Horse-power constant for one lb. M. E. P., and
 one revolution per minute
 10. Give description of main features of engine and plant, and illustrate with
 drawings of same given on an appended sheet. Describe method of
 governing. State whether the conditions were constant throughout
 the test.

Total Quantities

11. Duration of test hours.
 12. Gas or oil consumed cu. ft. or lbs.
 13. Air supplied in cubic feet cubic feet.
 14. Cooling water supplied to jackets "
 15. Calorific value of gas or oil by calorimeter test, determined
 by B. T. U.

Hourly Quantities

16. Gas or oil consumed per hour cu. ft. or lbs.
 17. Cooling water supplied per hour lbs.

Pressures and Temperatures

18. Pressure at meter (for gas engine) in inches of water..... ins.
 19. Barometric pressure of atmosphere:
 (a) Reading of height of barometer "
 (b) Reading of temperature of barometer deg. Fahr.
 (c) Reading of barometer corrected to 32° Fahr. ins.
 20. Temperature of cooling water:
 (a) Inlet deg. Fahr.
 (b) Outlet "
 21. Temperature of gas at meter (for gas engine) "
 22. Temperature of atmosphere:
 (a) Dry-bulb thermometer "
 (b) Wet-bulb thermometer "
 (c) Degree of humidity per cent.
 23. Temperature of exhaust gases deg. Fahr.
 How determined

Data Relating to Heat Measurement

24. Heat units consumed per hour (lbs. of oil or cu. ft. of gas per
 hour multiplied by the total heat of combustion)..... B. T. U.
 25. Heat rejected in cooling water:
 (a) Total per hour "
 (b) In per cent of heat of combustion of the gas or oil
 consumed per cent.
 26. Sensible heat rejected in exhaust gases above temperature
 of inlet air:
 (a) Total per hour B. T. U.
 (b) In per cent of heat of combustion of the gas or oil con-
 sumed per cent.
 27. Heat lost through incomplete combustion and radiation
 per hour:
 (a) Total per hour B. T. U.
 (b) In per cent of heat of combustion of the gas or oil con-
 sumed per cent.

Speed, Etc.

- | | |
|---|------|
| 28. Revolutions per minute | rev. |
| 29. Average number of explosions per minute | |
| How determined | |
| 30. Variation of speed between no load and full load | rev. |
| 31. Fluctuation of speed on changing from no load to full load
measured by the increase in the revolutions due to the
change. | |

Indicator Diagrams

- | | | |
|--|----------|---------|
| | 1st Cyl. | 2d Cyl. |
| 32. Pressure in lbs. per sq. in. above atmosphere: | | |
| (a) Maximum pressure | | |
| (b) Pressure just before ignition | | |
| (c) Pressure at end of expansion | | |
| (d) Exhaust pressure | | |
| 33. Temperatures in deg. Fahr. computed from diagrams: | | |
| (a) Maximum temperature (not necessarily at
maximum pressure) | | |
| (b) Just before ignition | | |
| (c) At end of expansion | | |
| (d) During exhaust | | |
| 34. Mean effective pressure in lbs. per sq. in. | | |

Power

- | | |
|---|-----------|
| 35. Power as rated by builders: | |
| (a) Indicated horse-power | H. P. |
| (b) Brake | " |
| 36. Indicated horse-power actually developed: | |
| First cylinder | " |
| Second cylinder | " |
| Total | " |
| 37. Brake H. P., electric H. P., or pump H. P., according to the
class of engine | " |
| 38. Friction indicated H. P. from diagram, with no load on
engine and computed for average speed | " |
| 39. Percentage of indicated H. P. lost in friction | per cent. |

Standard Efficiency Results

- | | |
|---|-----------|
| 40. Heat units consumed by the engine per hour: | |
| (a) Per indicated horse-power | B. T. U. |
| (b) Per brake horse-power | " |
| 41. Heat units consumed by the engine per minute: | |
| (a) Per indicated horse-power | " |
| (b) Per brake horse-power | " |
| 42. Thermal efficiency ratio: | |
| (a) Per indicated horse-power | per cent. |
| (b) Per brake horse-power | " |

Miscellaneous Efficiency Results

- | | |
|---|--|
| 43. Cubic feet of gas or lbs. of oil consumed per H. P. per hour: | |
| (a) Per indicated horse-power | |
| (b) Per brake horse-power | |

Heat Balance

- | | |
|---|--|
| 44. Quantifies given in per cents of the total heat of combustion
of the fuel: | |
|---|--|

(a) Heat equivalent of indicated horse-power	per cent.
(b) Heat rejected in cooling water	"
(c) Heat rejected in exhaust gases and lost through radiation and incomplete combustion	"
Sum = 100	"
Subdivisions of Item (c):	
(c1) Heat rejected in exhaust gases	"
(c2) Lost through incomplete combustion	"
(c3) Lost through radiation, and unaccounted for	"
Sum = Item (c)	

Additional Data

Add any additional data bearing on the particular objects of the test or relating to the special class of service for which the engine is to be used. Also give copies of indicator diagrams nearest the mean and the corresponding scales. Where analyses are made of the gas or oil used as fuel, or of the exhaust gases, the results may be given in a separate table.

TABLE NO. 2

DATA AND RESULTS OF STANDARD HEAT TEST OF GAS OR OIL ENGINE

Arranged according to the Short Form advised by the Engine Test Committee, American Society of Mechanical Engineers. Code of 1902.

1. Made by of
on engine located at
to determine
2. Date of trial
3. Type and class of engine
4. Kind of fuel used
 - (a) Specific gravity deg. Fahr.
 - (b) Burning point "
 - (c) Flashing point "
5. Dimensions of engine:

	1st Cyl.	2d Cyl.
(a) Class of cylinder (working or for compressing the charge)		
(b) Single or double acting		
(c) Cylinder dimensions:		
Bore	in.	
Stroke	ft.	
Diameter piston rod	in.	
(d) Average compression space, or clearance in per cent		
(e) Horse-power constant for one lb. M. E. P. and one revolution per minute		

Total Quantities

6. Duration of test hours.
7. Gas or oil consumed cu. ft. or lbs.
8. Cooling water supplied to jackets
9. Calorific value of fuel by calorimeter test, determined by calorimeter B. T. U.

Pressures and Temperatures

- | | |
|--|------------|
| 10. Pressure at meter (for gas engine) in inches of water..... | ins. |
| 11. Barometric pressure of atmosphere: | |
| (a) Reading of barometer | " |
| (b) Reading corrected to 32 degs. Fahr | " |
| 12. Temperature of cooling water: | |
| (a) Inlet | deg. Fahr. |
| (b) Outlet | " |
| (c) Degree of humidity | " |
| 13. Temperature of gas at meter (for gas engine) | " |
| 14. Temperature of atmosphere: | |
| (a) Dry bulb thermometer | " |
| (b) Wet bulb thermometer | " |
| 15. Temperature of exhaust gases | " |

Data Relating to Heat Measurement

- | | |
|---|----------|
| 16. Heat units consumed per hour (pounds of oil or cubic feet of gas per hour multiplied by the total heat of combustion) | B. T. U. |
| 17. Heat rejected in cooling water per hour | " |

Speed, Etc.

- | | |
|---|------|
| 18. Revolutions per minute | rev. |
| 19. Average number of explosions per minute | |

Indicator Diagrams

- | | | |
|--|----------|---------|
| 20. Pressure in lbs. per sq. in. above atmosphere: | 1st Cyl. | 2d Cyl. |
| (a) Maximum pressure | | |
| (b) Pressure just before ignition | | |
| (c) Pressure at end of expansion | | |
| (d) Exhaust pressure | | |
| (e) Mean effective pressure | | |

Power

- | | |
|--|-----------|
| 21. Indicated horse-power: | |
| First cylinder | H. P. |
| Second cylinder | " |
| Total | " |
| 22. Brake horse-power | " |
| 23. Friction horse-power by friction diagrams | " |
| 24. Percentage of indicated horse-power lost in friction | per cent. |

Standard Efficiency, and Other Results

- | | |
|---|-----------------|
| 25. Heat units consumed by the engine per hour: | |
| (a) Per indicated horse-power | B. T. U. |
| (b) Per brake horse-power | " |
| 26. Pounds of oil or cubic feet of gas consumed per hour: | |
| (a) Per indicated horse-power | lbs. or cu. ft. |
| (b) Per brake horse-power | " |

Additional Data

Add any additional data bearing on the particular objects of the test or relating to the special class of service for which the engine is to be used. Also give copies of indicator diagrams nearest the mean, and the corresponding scales.

RULES FOR TESTING GAS PRODUCERS AND GAS ENGINES. CODE
OF THE GERMAN SOCIETY OF ENGINEERS *

All metric units have been transposed to English units

The preparation of the following rules for making gas engine and producer tests was undertaken by a committee appointed from the Verein deutscher Ingenieure, in collaboration with the German Society of Engine Builders, with the view of establishing definite general regulations governing such tests. It is desirable, by specifying the important proportions of the examined plants and the conditions under which the results were attained, to insure that these results are not only applicable to a single case, but that they have general value. To attain this end it is necessary that all data should be given uniformly according to a code of regulations such as that here presented.

The execution of such tests should be intrusted only to persons possessing the required expert knowledge and practical experience. These persons must make a trial plan, or schedule, appropriate to the individual case in hand, which, in many instances, will not require that all of the investigations stipulated in the general code are actually carried out. They must further examine the instruments for measuring or recording purposes as to their fitness and must compile the results. The following rules, the adoption or selection of which must be left to the soundness of judgment of the investigator, are intended to serve as a basis on which to proceed.

GENERAL REGULATIONS

Object of Investigation

1. The object of a test made on a producer-gas plant may be to determine:

(a) The quantity, composition, and calorific value of the fuel consumed.

(b) The quantity, composition, and heat value of the gas produced.

(c) The degree of efficiency of the producer-gas plant.

(d) The separate heat losses in the plant.

* Mainly from F. E. Junge's translation in *Power*, Feb., 1907.

(e) The quantity of impurities contained in one cubic meter or one cubic foot of gas (dust, tar, sulphur, etc.).

(f) The moisture contents of the gas.

(g) The water consumption of the producer-gas plant, either total or in the separate parts.

(h) The mechanical work required for operating the plant, including apparatus.

(i) The duration of time required for starting.

(k) The stand-by losses during intervals of shutting down day or night.

2. The object of a test made on an internal combustion (gas) engine may be to determine:

(a) The indicated capacity and the effective output.

(b) The mechanical efficiency.

(c) The fuel consumption and the heat consumption per horse-power hour.

(d) The consumption of lubricants, separately for cylinder and engine.

(e) The consumption of water and the heat conducted to the cooling water.

(f) The fluctuations in number of revolutions.

(g) The composition of exhaust gases.

NUMBER AND DURATION OF TESTS

Admissible Fluctuations

3. The number and duration of trials are determined by the purpose of the test as well as by a consideration of the conditions of installation and operation, and must be settled and previously arranged according to paragraphs four to eight. For trials of special importance the results of which are decisive for acceptance tests, for penalties or for premiums, this item deserves special consideration.

4. Acceptance tests should be made if possible immediately after a plant has been put into actual operation; the manufacturers, however, must be granted a reasonable time for making preliminary trials of their own and for carrying out alterations or improvements then necessary. The length of this time and other conditions are best agreed upon when drawing up the delivery contract.

5. In order to be able to get acquainted with the operation of the plant that is to be tested, to find time for examining the testing devices employed, and to break in observers and assistants, it is desirable that preliminary trials be allowed.

6. If the fuel consumption in gas producers is to be determined, the trial run must be extended over at least eight hours under constant conditions and without interruptions.

7. For determining the consumption of liquid or gaseous fuel and provided the conditions are constant, it is sufficient for the higher loads to extend measurements over an hour, while for finding the consumption at the lower loads, measurements of even shorter duration are sufficient. To ascertain the constancy of the conditions the temperature of the outflowing cooling water must be read from time to time. These rules as to the duration of the tests are made with the provision that no interruption or disturbance of the trial takes place, and that intermediate readings show only slightly varying values for the consumption.

8. If only the mechanical efficiency of an engine is to be determined, trials of short duration under constant conditions are sufficient; but at least ten sets of indicator cards should be taken.

9. For investigations of special importance at least two tests should be made, one after the other. They should be accepted only if no interruptions occurred and if the results show no greater deviations than those due to unavoidable errors of observation. The mean of the two results is to be taken as the final result.

10. The extent to which the capacity and consumption of gas may differ from the guarantee or contract figures, without justifying a claim of breach of contract, is to be clearly stated before the tests (either in the original contract or in the schedule of tests). When no other agreement has been previously arrived at, the capacity guarantee is regarded as fulfilled if the figure obtained in the test is not more than 5 per cent below the value on which the guarantee was based. This margin, however, is allowable only for the maximum output which was promised beyond the guaranteed continuous output. The latter must be rendered by the engine under all circumstances.

The consumption of fuel and water as determined on test should not exceed the guaranteed figures by more than 5 per cent

even if, during the trial, the engine load fluctuated somewhat from the load upon which the guarantee was based, provided that fluctuation do not exceed an average of ± 5 per cent of such load, or a maximum of ± 15 per cent.

Since it is often impossible when making tests to have the internal combustion engine work at exactly the effective (horse-power) capacity on which the guarantee agreed upon in the contract is based, it is recommended that the agreement shall specify the expected fuel consumption for higher and lower outputs. The same provision is perferably made also with gas producers.

UNITS OF MEASUREMENT AND DESIGNATIONS

11. When giving pressure data it must be stated whether absolute pressures or gage pressures above or below the atmospheric are meant. Absolute pressure equals atmospheric pressure plus gage pressure.

12. All temperature and heat measurements refer to the Fahrenheit scale.

13. The mechanical equivalent of heat is taken at 778 foot-pounds.

14. The calorific value of a fuel is to be taken as its lower heating value; that is, the heat which is liberated by the complete combustion of the fuel when the burnt products are cooled down to the original (room) temperature at constant pressure, it being assumed, however, that the water of combustion and the moisture contained in the fuel remain vaporized. The calorific value must be based on the unit quantity or weight of original fuel, without deducting ash, moisture, etc., and is to be expressed in heat units. For both solid and liquid fuels the unit of weight is the pound.

The heat value of gaseous fuels is based on one cubic foot at 32 degrees Fahrenheit, and 760 millimeters barometer pressure, or must be expressed in thermal units as "effective" heat value, that is, reduced to one cubic foot of actual gas used. If not specially stated, it is always understood that the heat value recorded is that of gas at 32 degrees Fahrenheit and 760 millimeters barometer pressure.

In this country the general standard so far recommended seems to indicate for "standard gas" a temperature of 60 degrees

Fahrenheit, and a pressure of 14.7 pounds per square inch, corresponding to the usual atmospheric pressure.

15. The efficiency of a gas-producer plant is the ratio of the latent heat contained in the gas as produced to the heat of combustion of the total weight of fuel consumed in the plant, both items being computed from the lower heating value. In producer-gas plants having a separately fired steam boiler, it is advisable also to determine the ratio of the heat which is chemically bound in the producer gas to the heat equivalent of that portion of the fuel which is consumed in the producer proper for making such gas.

16. The unit of measurement used for the power or work output of an internal combustion engine is the horsepower equal to 33000 foot-pounds per minute. It must be clearly stated whether the indicated power, or the useful or available power, is meant. If not otherwise designated it is understood that the figures refer to the useful or available output.

17. The indicated power of the engine, or the indicated work, is the difference between the total power developed or work done, and the indicated power, or work, which is consumed within the engine; in short, the difference between the positive and the negative indicated power or work.

AUTHOR'S NOTE. — This is the provision which caused considerable discussion among gas-engine experts some time ago. It means as it stands, that in a 4-cycle machine, the indicated horse-power is that determined from the work diagram minus the work shown by the lower loop diagram; and, in a 2-cycle engine, the total indicated horse-power as determined from the diagram of the power cylinder minus the pump work is considered as the indicated horse-power. This view is undoubtedly correct when the mechanical efficiency of the engine itself as a machine is to be determined.

The power required at "no load" is the power indicated when no useful work is rendered by the engine.

18. Mechanical efficiency is the ratio of the useful power to the indicated power of the engine.

19. All consumption figures should be reduced to the hour basis, and if they are to be compared with the output of the engine they must be based on one horse-power hour. If not otherwise agreed upon, these data refer to the useful or available output at full load.

EXECUTION OF TESTS

20. If the quantity of gas made in a producer or the weight of fuel consumed in an engine is to be measured, then all pipes or ducts which are not used in the test must be cut off from the piping which leads to the producer and engine that are to be tested. This is best done by means of blind flanges. The active ducts, pipes, gas holders, etc., must be examined with regard to leakage and made tight if necessary. Unavoidable losses due to leakage must be determined. This holds especially for masonry gas mains.

FUEL CONSUMPTION OF A GAS-PRODUCER PLANT

21. The kind, number, and duration of tests must be agreed upon according to the general rules laid down in paragraphs 1 to 10.

22. The constructive features and the operative conditions of gas-producer plants must be described and illustrated in the report by drawings, so far as this is necessary, to arrive at a clear understanding of the manner of working and of the results obtained.

23. Before making the test the plant should be examined as to whether or not it is in good working order.

24. The quantity of fuel consumed in the gas producer is determined by taking the weight of the fuel which is charged into the producer during the trials in order that the producer may contain at the end of the test exactly the same amount of heat — either liberated, or chemically bound in the fuel — that it contained when starting the test. To meet this requirement it is not sufficient that the depth of the fuel bed be the same at the end as it was at the beginning; it must also be taken into consideration what influence the ash and the slag left in the producer, the location of the incandescent zone, the formation of fissures and cavities, the closeness or density of the producer charge, and the chemical composition of the burning fuel particles exercise on the heat contents of the producer.

In order to comply with this requirement the following rules should be followed:

25. When starting the test the plant should be in the condition of stability or normal working condition, if possible.

This means that after a period of shut-down for cleaning or repairs it should be in active operation for one or more days, running on fuel of the same characteristics and size, with the same depth of fuel bed, the same skill of attendance as regards the charging or feeding of fresh fuel and the removing of slag, and under the same load conditions that will obtain during the test.

26. During the trial the producer should be charged and poked as nearly in accordance with the requirements for attendance as possible. The level of fuel charged must be the same at the beginning and at the end of the tests and should be kept constant during the trial. About half an hour before starting and before stopping a test, the slag and ashes should be removed.

If it is impossible to rake out the ashes during the operation of the producer, the plant must be shut down immediately after stopping the test, the ashes must be taken out at once and the producer refilled up to the same level that existed when starting the test. The weight of fuel used for this purpose must be added to the consumption.

27. The fuel consumed during the trial must be weighed, also the fuel which has not been burnt and remains useful; that is, that portion which drops down from above the grate while raking out the ashes, and that which is culled out from the ashes as unburnt. The weight of the former may be deducted from the consumption, but not the amount which is taken out from the ashes, nor the coal dust which accumulates in the scrubbers and in the flues between the producer and the engine.

28. To be able to determine the quantity of ash and slag produced during the trial, the ash box must be emptied before the test. If this is not possible, as when an inclined grate is used, the refuse in the ash box must be equalized before and after the run.

29. The stand-by losses during intervals of shutting down at day and night must be determined.

30. In order to get a representative sample of the solid fuel, the following course may be pursued: Of every carload, basket, or other measure of fuel, put a shovelful into a covered receptacle. Immediately after the test is over, the contents of the receptacle should be broken, mixed, spread and quartered by drawing the

two diagonals of a square. The two opposite quarters are rejected, the two others broken up finer, mixed and quartered, and the two opposite quarters rejected. This is continued until a sample of some 10 to 20 pounds remains, which is preserved, in well-closed receptacles, for analysis. In addition to this a number of other samples must be put away in air-tight receptacles for use in determining the contents of moisture in the fuel.

31. The composition of the fuel shall be determined by chemical analysis. Its contents in carbon (C), hydrogen (H), oxygen (O), sulphur (S), ash (A), and water (W) must be given in percentage of weight referred to the original fuel. The contents, in the fuel, of nitrogen (N) can be disregarded. The behavior of the fuel when being heated should be determined by a coking test.

32. The calorific value of the fuel must be determined by calorimetric analysis. An approximate determination of the heating value can be made on the basis of the chemical analysis by employing De Long's formula:

$$\text{Heating value} = 145 C + 522.3 \left(H - \frac{O}{8} \right) + 40 S - 9.66 W$$

in which C, O, H, S, and W are expressed in weight per cent.

TESTING AN INTERNAL-COMBUSTION ENGINE

33. Kind, number, and duration of trials to be agreed upon according to the general regulations Nos. 1 to 8.

34. The constructive features and operative conditions of the engine must be so illustrated in the report as to enable one to form a correct idea of the manner of working and of the results of operation. Especially important are the type and capacity of engine, diameter of cylinder and piston-rod, piston stroke, contents of clearance space, and other essential dimensions; the normal rate of revolution and the admissible fluctuations; kind and heat value of fuel for which the engine is intended. The diameter of the cylinder and the stroke should be actually measured if this is possible.

The contents of the compression space are preferably determined by filling with water. If it is impossible to state the cubical contents of the compression space, then the compression

pressure at full load should at least be given. This is done by taking an indicator card while the ignition is interrupted.

35. Before making the test the engine must be examined internally and externally as to whether or not it is in good working order.

36. The number of revolutions of the engine should be determined by a continuous speed counter, the records of which must be noted at certain intervals, and must be checked or corrected from time to time by direct readings. If the speed conditions of the engine are to be investigated it is essential to determine the following items:

(a) The number of revolutions under constant conditions at maximum load and at no load.

(b) The fluctuations in speed at constant load.

(c) The temporary change in the number of turns when the load is suddenly decreased or increased from a given constant load by a prescribed amount. These determinations can be executed with apparatus of the character of the Horn tachograph. The fluctuations of speed during the performance of one engine cycle above and below the mean value, expressed in parts of the latter, should be determined by calculation unless otherwise provided.

The coefficient of fly-wheel regulation is

$$\delta = \frac{N_{\max.} - N_{\min.}}{\frac{N_{\max.} + N_{\min.}}{2}} = 2 \left(\frac{N_{\max.} - N_{\min.}}{N_{\max.} + N_{\min.}} \right)$$

where N = number of revolutions.

37. The useful output can be determined either by brake test or by electrical measurement.

The dimensions and weight of the brake should be determined before the trial.

The electrical measurements can be made on a generator directly coupled to the gas engine. The useful work is computed from the output of the dynamo. The efficiency of the generator should be determined by one of the methods as laid down in the "Rules for Judging and Testing Electrical Machinery and Transformers," published by the association of German electrical engineers. If the efficiency is found approximately by measuring

the determinable losses, then an adequate amount (say 2 per cent of the full load output) must be allowed for losses not accounted for.

The apparatus with which the electrical measurements are executed must be calibrated before and if possible also after the test.

Whether anything besides the 2 per cent above allowed should be credited to the gas engine for increased bearing friction and windage of the generator must be settled in each individual case.

Whether, in case the useful output can neither be determined by brake test or by electrical measurements, the code provision for testing steam engines can be admitted as correct for gas engines, namely, to designate the useful output as the difference between the indicated work at any load and the indicated work at no load, cannot be settled at the present state of development, since results of accurate investigations are not yet available.

38. Indicators must be connected immediately to the combustion chamber without employing long piping with sharp bends, and one indicator must be provided for every combustion chamber. For this purpose each compression chamber must have an opening for three-quarter or one inch Whitworth thread. The same holds true for pump cylinders.

The indicators and their springs must be calibrated before and after the test according to the accepted standards.

39. During the test, cards should be taken quite frequently from every combustion chamber and from the pump cylinders. The cards should be designated by numbers, and the time when each card was taken, the scale of springs used and the number of single cards obtained must be recorded on the cards. At least five diagrams should be taken on one card successively. From time to time diagrams indicated with a weak spring should be taken from the combustion chambers.

The indicated work at no load should be determined immediately after stopping the main test and while the engine is still warmed up ready for operation. Care must be taken that the no-load cards are not taken during an acceleration or during a retardation period of the fly-wheel.

ANALYSIS OF THE GAS GENERATED IN A PRODUCER-GAS PLANT OR
CONSUMED IN AN INTERNAL COMBUSTION ENGINE, OR OF THE
LIQUID FUEL USED

40. The samples for the chemical analysis of the gas must be taken during the trial at regular intervals and as frequently as possible.

They must be either analyzed on the spot or preserved in glass tubes closed by melting the ends. The analysis is to determine, in per cent of volume, the contents of the gas in carbon monoxide (CO), carbon dioxide (CO₂), hydrogen (H₂), marsh gas (CH₄), heavy hydrocarbons and oxygen (O₂).

In addition it is recommended to determine the contents of sulfur. The gas samples should be taken from the gas main between the cleaning apparatus and the engine.

41. The heat value of the gas should be determined as often as possible by calorimetric analysis, and the burner of the calorimeter should be fed from the gas main without interruption. In suction producer plants this can be done by means of a gas pump drawing from the main. If conditions should make it necessary that a sample be taken from the pipe while the calorimeter is shut off, such sample to be later transferred to and burned in the calorimeter, then the quantity of gas so taken should not be less than 300 liters (10.59 cubic feet), in order that the calorimeter may at first be brought into the condition of stability as regards the water of combustion, and in order that at least 100 liters (3.53 cubic feet) remain available for two successive analyses. The suction pump, the gas holder and the piping must be made tight with special care when making a calorimeter analysis of suction gas.

42. The gas meter of the calorimeter in which the heat value of the gas is determined must be calibrated. For determining the temperatures of the calorimeter water, only thermometers with calibration certificates or others compared with such should be used. The scales must be divided at least into tenths of a degree.

On the basis of the chemical analysis the heating value per standard cubic foot of gases which do not contain heavy hydro-

carbons can be computed from the following formula, if a calorimetric analysis cannot be made

$$\text{Heating Value} = 3.42 \text{ CO} + 2.97 \text{ H}_2 + 9.52 \text{ CH}_4$$

where CO, H₂ and CH₄ are expressed in volume per cent.

43. The quantity of gas produced or consumed should be measured by means of a gas holder or a gas meter. The cross-sectional area of the holder should be determined by measurement of its circumference at several places. Consumption tests with the gas holder shall not be made while the latter is exposed to the sun.

44. The gas meter must be calibrated and set level; it must be so filled that the water level corresponds to the normal filling existing during calibration. Between the gas meter and the engine a pressure regulator must be installed or a large suction space provided so that the water level shows only small pulsations during the pressure fluctuations.

45. At intervals corresponding to the duration of test the following readings should be taken: Position of the bell of the gas holder at three places or the records shown by the gas meter; the pressure in the bell or in the gas meter; the temperature of the gas when entering and when leaving the gas holder or the gas meter and before reaching the engine; the barometric pressure.

46. If the temperature of the gas is different when measuring the consumption than when measuring the heat value, the computation must also take into account the increase of volume which is due to the moisture contents of the gas at higher temperatures.

47. The consumption of liquid fuel must be determined either by weight or by measuring its volume. For determining heat value, composition, and specific weight of the fuel one representative average sample is sufficient.

48. When measuring the fuel consumption of internal combustion engines, the consumption of lubricating oil for the cylinder should be determined at the same time.

49. If the consumption at low loads of a double-acting tandem or twin engine is to be determined, it is not allowable to shut off the gas from one or more ends of the cylinders, provided that no other arrangements have been previously agreed upon and

are mentioned in the report, or that the governor acts automatically in the way described.

EXPLANATIONS TO VARIOUS ARTICLES OF THE CODE

The main code is followed by a number of explanations from which the following extracts are taken. The figures refer to the paragraphs of the above code.

1 & 2

In most cases only one or two of the objects of test mentioned are taken into account in any given trial. If in any exceptional case the object of the test should not be any of those mentioned, it should be a simple matter to adapt the rules given.

Under 2 (c) the term horse-power hour is used. It is essential that in any given trial this term be more closely defined, as horse-power may mean indicated brake, or even horse-power developed by pumps.

4

It is extremely desirable that the contract state the time allowed the manufacturer for adjustment and trial runs, because his own interests may make him call sometimes for a long, sometimes for a short period. In the case of a small engine, more or less a commercial stock machine, he may wish to have the period as short as possible, and this the buyer may agree to without danger of loss to himself. If, however, the machine is of a special type, or one provided with special attachments, it is but a matter of justice to allow the manufacturer a reasonable time in which to break in the engine and to give him an opportunity to correct any imperfections that may appear. It is to the interest of the buyer to grant such a period in order to become familiar with the machine before taking over the entire responsibility of operating it. It is also true that many faults appear only after some weeks of operation.

On the other hand, too long a period of adjustment is in many cases not acceptable to the buyer, because any extended work of improvement usually seriously hampers operation; and because in many cases he desires an operative machine, which no longer requires the care of the manufacturer, as soon as possible.

It frequently happens that no acceptance test is agreed upon.

In such cases it sometimes happens that the buyer comes back upon the manufacturer for faults which did not develop until the machine had been in operation some time.

If the manufacturer then agrees to an investigation or a test, a sufficient period should be given him to make any investigation he sees fit or to correct any imperfections that may have appeared before the decisive trial or investigation is made. This sometimes leads to a simple settlement of the matter in that the manufacturer discovers that ignorance or carelessness on the part of the operator have caused the imperfections complained of. The granting of such a period also guards the buyer against any later claim of the manufacturer that during the trial the machine was not in the condition in which he delivered it.

5

Preliminary tests are always desirable, but not absolutely necessary. The cost of any kind of investigation is usually quite high and of course the cost increases directly with the time. The expert called will therefore make such tests when they seem to him essential. But the manufacturer should have the right to call for the time necessary for such trials if he is to present the machine in its best condition.

6

It cannot be denied that eight hours is a rather short time, because it is extremely difficult to determine whether the producer is in the same condition at the end as at the beginning of the test, and because this uncertainty may lead to large errors. On the other hand it is unquestionable that in many cases a longer time would call forth so many difficulties in operation that eight hours would seem the necessary limit.

The rule is mainly framed to prevent trials of so short duration that serious errors can hardly be avoided, but it leaves it to the judgment of the experimenter whether to make the tests longer than eight hours where it seems desirable and is possible to do so.

7

Intermediate readings are recommended without qualification, since they form the best criterion of the constancy of conditions.

With liquid or gas fuels of constant composition, individual readings every five minutes apart sometimes show no variation for hours at a time. In such a case it is useless to extend the time of the trial.

8

In determining the mechanical efficiency of an engine it should not be forgotten that, although the average load may be constant, there may be speed variations due to the inevitable inequality of the indicator diagrams, so that during some cycles work is done in accelerating the fly-wheel, while during others the fly-wheel by retardation gives up some of its kinetic energy.

To minimize any error that this may introduce into the determination of the mechanical efficiency, at least ten diagrams should be taken.

If the conditions are otherwise constant, however, it is not necessary to spread these diagrams over any considerable period of time.

It is self-evident that during the time of taking the diagrams the supply of lubricating oil must not be increased.

Changes in the mechanical efficiency of the engine, as for instance those due to fouling, cannot be detected with certainty even by a long test period; they become noticeable usually only after a period of operation extending over two weeks. The determination of the mechanical efficiency of an engine, after constant conditions of operation are attained, therefore only applies to the engine in its then existing state or condition.

The number of diagrams to be taken on one card cannot be definitely stated. On account of variation in the diagrams, which is less at high than at low loads, care should be had not to take too few. On the other hand it is useless to take more than can be clearly distinguished. The running together of a larger number of diagrams only makes their evaluation more uncertain.

10

In consideration of unavoidable errors of observation, possible errors of the instruments used, etc., it is meet and usual to allow a certain margin between the figures found on trial and those guaranteed. In the steam engine code 5 per cent is allowed for this, and it seems reasonable to assume the same figure in this

case. Only in one point, in the guaranteed normal capacity, does the gas engine call for an exception.

A given steam engine gives its most economical results at a certain cut-off, but a higher capacity can always be obtained at the expense of a little economy, that is, a buyer is certain that even a machine slightly too small will give him sufficient capacity. A gas engine, on the contrary, works with the greatest economy at its maximum load. It is to the interest of the buyer, therefore, to get an engine exactly suited to his needs and not to choose it too large. It is possible for the same reason that any engine, if lacking slightly in guaranteed capacity, may become absolutely useless to the buyer. For these reasons it was thought advisable not to grant the manufacturer any leeway whatever as regards guaranteed capacity.

It is clear, therefore, that the manufacturer must take upon himself any possible inaccuracies in the measurements, unless he can show them up and demand a new trial. For that reason it is well for him to make his guarantee a little on the safe side of what he knows his engine is capable of developing. On the other hand, there is no harm done to the interest of the buyer if the manufacturer underrates the normal capacity of his machine, because the former will always call for an engine of a certain normal capacity to suit his needs. If he fails to do this, but places his dependence in the guaranteed maximum capacity, he is open to the charge of carelessness.

Since during acceptance tests it is often not possible to keep the load quite constant, it became necessary, following the steam engine code, to allow a certain amount of variation, within which no just cause could be found for objection to the trial. There are cases where the variations occurring are much greater, as when a gas engine is used for driving a roll train. But no one set of "Rules" can possibly take into account all such extreme cases, and in such instances the contract should contain the necessary agreements to make any test clear and free from subsequent objections.

The wish has been expressed from several quarters, that the "Rules" should contain a definition of the term "Normal Capacity." On account of the peculiarity of the gas engine above discussed this is not quite feasible. But the term "Maximum

Continuous Capacity" perhaps defines most nearly what is intended in most cases.

14

It is sometimes the case that the heating value of the standard cubic foot, that is, reduced to 32 degrees Fahrenheit and 760 mm. barometer, is so greatly different from the actual value of the gas as used, that any contract which contains only the heating value of the gas stated on that basis does not convey much meaning to the non-technical buyer. If for instance a given gas has a heating value of 135 B. T. U. per standard cubic foot, its effective heating value at a high altitude and in a warm climate, say at 68 degrees and 620 mm. barometer, will only be about 100 B. T. U. per cubic foot. To obviate any misunderstanding, it should be clearly stated that, when the "effective" heating value of the gas is not definitely specified, the heating value at 32 degrees Fahrenheit and 760 mm. barometer is meant.

19

By "full" load is meant the normal capacity, as per paragraph 10.

23

For acceptance tests, and all other tests which are intended to decide any disagreements between manufacturer and buyer, such examination should be carried out in the presence and with the aid of the former, as already mentioned under paragraph 4.

24-26

In all gas producer tests it is hardly possible with certainty to have all conditions exactly the same at the end as at the beginning. But since any difference in the beginning and end conditions may lead to considerable error, which can only be equalized by excessive length of test, the "Rules" are intended to operate to the end that such errors are not in any way magnified by the method of test. Hence the detailed statement in the regulations.

27

Since in actual operation the fuel in the ash or the coal dust in the gas mains are hardly ever utilized, no correction should

be made for these on any trial. In order, however, to prevent the results from being influenced by insufficient cleaning of the producer, any fuel which falls out from above the grate during the cleaning period may be subtracted from the amount charged.

35

See explanation under 23.

37

A brake test of a large engine is in some instances not possible, and in any case a matter of considerable cost. In many cases, however, the larger gas engines are either direct connected to a generator or to some other power consumer, as a blowing cylinder. In the former case electrical measurements, from which the effective horse-power may be determined, are easily made. In the latter case the capacity guarantee will in most instances be based upon the performance of the power consumer, as for example the air compressed by the blowing cylinder. Outside of engines of this type, however, there still remain many cases in which it would be of the utmost value to have some means of determining the effective capacity, and it should not be forgotten that, even in the case of medium-sized machines, a braking of the engine at the place of erection is often, on account of local restrictions, very difficult. The problem has been solved for steam engines by assuming that the difference between the indicated horse-power at any load and the indicated horse-power at no load is the effective or useful horse-power. It is quite possible that in many cases this is not quite correct, but the method is very generally accepted and followed.

On account of the great overload capacity of the steam engine, a small error in this respect does not mean a great deal. But the case of the gas engine is quite different. The data on hand does not warrant the application of the same method to the gas engine, and the consequences of an erroneous conclusion are much more serious on account of the lack of overload capacity.

For these reasons one is compelled in some cases to omit the determination of the effective capacity altogether and to be content with the determination of the indicated power only. It is recommended in such cases that the mechanical efficiency be not

assumed too high and that any guarantees regarding fuel, etc., also be based upon the indicated horse-power.

It is sometimes possible to brake an engine on the test floor of the factory. The mechanical efficiency may thus be previously determined when it is known that no brake test can be made in the final place of erection.

39

The number of diagrams to be taken during any given test cannot be definitely specified. Much depends upon the length of test, and the decision may be left to the judgment of the experimenter.

It is, however, always recommended that a bundle of diagrams, instead of only one, be taken on every card. Thus a series of diagrams are obtained, while, if only a single diagram is taken, it is possible to hit upon the same diagram in the series a number of times. (See under extract 8.)

The work of fluid friction, that is, the lower-loop diagram, cannot be determined with certainty from the full indicator cards. It is best for that reason to ignore the loop when determining the positive work and to find the negative work from special weak spring diagrams.

48

The measurements of the quantity of lubricating oil used is of importance in smaller engines, because the fuel consumption can be favorably influenced by a copious supply of the lubricant.

49

If under low loads, only one end of the cylinder is allowed to work, the fuel consumption would be much lower. But since this is not generally done in operation, the results would be erroneous. If, however, the governor during operation shuts off the individual cylinders or cylinder ends, as the load drops, this is of course also permissible during a test.

CHAPTER XVII

THE PERFORMANCE OF GAS ENGINES AND GAS PRODUCERS

1. As indicated in the rules for testing, the very great majority of tests of engines are made to determine capacity and fuel consumption. In some special cases, as with engines driving generators, tests are also sometimes made of the regulation. These three tests together take care of what may be termed the commercial side of testing. All other tests are special in that they are not often executed in an acceptance test, but form in most cases the object of scientific or laboratory investigation.

Such investigations are in many instances very valuable, and have served to throw a flood of light on the somewhat complex cylinder actions of the internal combustion engine. It was thus found that the temperature of the cylinder walls, *i.e.*, the cooling water conditions, piston speed, ignition, proportion of mixture, compression, etc., all had a more or less marked effect upon engine performance. In the following the results obtained and the conclusions drawn by various experimenters concerning the effect of these various factors are briefly set forth.

The main bulk of this work has been done by Witz in France, Slaby and E. Meyer in Germany, and Burstall and others in England. In spite of the fact that none of these investigations are open to serious objection on the score of inaccuracy, the conclusions arrived at are not always in accord. This is undoubtedly due to the complexity of the cylinder actions, and the interdependence of the various factors entering the problem.

2. **Cooling Water Conditions and Piston Speed.** — It is reasonable to suppose that the higher the wall temperature of the cylinder, *i.e.*, the smaller the temperature difference between mixture and wall and the greater the piston speed, cutting down the time of exposure, the smaller the loss to the jacket.

The heat thus saved, however, may go in two directions: either

a greater thermal efficiency is shown, resulting in greater power developed for the same expenditure of heat, or the heat saved from going into the jacket is lost in the exhaust.

Witz, upon the basis of his experiments, comes to the former conclusion, and says "The action of the jacket is the great regulator of combustion phenomena." He summarizes his results as follows:

1. The efficiency increases with the piston speed and with the temperature of the surrounding walls.

2. The combustion of explosive mixtures is the more rapid the greater the speed of expansion and the hotter the cylinder walls.

The work of Slaby and of Meyer, however, seems to controvert these conclusions. Some of Slaby's tests show that while the gas consumption per horse-power hour decreases somewhat with an increase in the piston speed, there is an increase in the consumption with a rise of jacket water temperature. The following table of figures, quoted by Schöttler from some of Meyer's tests, illustrates the point that the heat saved from the jacket by higher piston speed may go to the exhaust, leaving the thermal return practically unaffected.

Ratio of Com- pression	R. P. M.	MEAN EFFECT- IVE PRESSURE	Ratio Air to Gas	HEATING VALUE OF CHARGE	WORK DONE BY 1 B.T.U.	EXHAUST TEMP.	HEAT DISTRIBUTION IN %		
		No. 1 sq. inch		B.T.U.	Ft. Lbs.	°F	Work	Jacket Water	Exhaust
2.67	187	54.3	7.11	18.5	140	1022	18.0	51.2	30.8
2.67	247	51.7	7.35	17.4	141	1137	18.1	45.6	36.3
4.32	187	69.3	7.43	17.0	190	867	24.4	53.8	21.8
4.32	247	65.2	7.40	16.8	184	992	23.7	49.5	26.8

In tests of this kind there are one or two simultaneous actions, not directly under control, which may serve to modify the final result and account in a measure for the discrepancy appearing in the results above discussed. An increase in the temperature of the walls or an increase in the piston speed both cause a decrease in the charge volume, the former by heating the incoming charge and decreasing the density of the mixture, the latter by increased friction loss in pipes and ports. The direct result of

this is that the effect of the action of the cylinder wall upon the leaner charge is proportionately greater. Thus the beneficial effect of greater piston speed may be partly annulled by the relatively stronger action of the walls. Further it is true that a smaller charge weight means a lower compression pressure, and the comparatively greater admixture of burned gases at high speeds causes a less rapid combustion. Both of these actions tend to decrease the efficiency. There are thus several antagonistic actions, and the final result is consequently in many cases quite problematical.

The *net* result of an increase in the cylinder wall temperature or of the piston speed, or both, in an existing machine, is certainly a decrease of maximum capacity for reasons already pointed out. Further, the effect of a variation in the temperature of the jacket water, while perhaps not quite so marked in engines using gas, is certainly quite noticeable in liquid fuel engines, especially those using kerosene or alcohol. It is quite possible in kerosene engines, by running the jacket too cold, to increase the oil consumption seriously by condensation of the oil vapor on the comparatively cold cylinder surfaces. The same holds true of alcohol. Thus hot walls are in such cases of undoubted benefit. The limits to temperature are, of course, decrease of engine capacity and danger of pre-ignition.

3. Compression. — The theoretical effect of increasing the compression, and the commercial limits to this increase have already been discussed in Chapter III. Much of the increased efficiency of blast furnace and producer gas engines as compared with illuminating gas and liquid fuel engines is directly due to the greater compressions that the former fuels can stand. The above table of Meyer's results gives some idea of the gain that can be made with illuminating gas by increasing the compression. With a compression ratio of 2.67, the average thermal efficiency was 18.05 per cent, with a ratio of 4.32 the average was 24.05 per cent, a gain of $\frac{24.05 - 18.05}{18.05} = 33$ per cent.

Another test made by E. Meyer * on a 10 horse-power engine, which was operated with illuminating and with producer gas, gave the following results:

* E. Meyer, Z. d. V. d. I., July 5, 1902.

COMPRESSION RATIO	INDICATED THERMAL EFFICIENCY	
	With Illuminating Gas	With Producer Gas
4.98	27.1	24.4
4.59	26.5	23.2
3.84	24.8	21.5

Here again the beneficial influence of the higher compression is marked, although the gain is not so great as in the former case, owing to the smaller change in the compression ratio.

Other instances pointing to the same result can be adduced without difficulty. See the next table below, also by E. Meyer, who has done an immense amount of work in the investigation of gas engines. Bánki took advantage of the principle in his gasoline engine, in which, by using water injection, he could employ compression ratios similar to those used in producer gas work and realized thermal efficiencies fully equal to those obtained with the leaner power gases.

4. The Mixture. — The inherent advantage of the use of lean mixtures has already been shown in Chapter III. Burstall * on the basis of his tests for the British Institution of Mechanical Engineers, concludes that the thermal efficiency depends upon the correct choice of the mixture, and that the ratio of air to gas should increase with the compression. His results, however, do not definitely warrant the latter part of this deduction, although it finds some support in the above-mentioned tests by Meyer, † as shown by the following table:

Test No.	Ratio of Compression	Ratio Air to Gas	Gas Consumption Cu. Ft. per I. H. P.-hour
1 }	2.67	6.41	27.1
2 }		8.08	25.4
3 }		6.38	22.6
4 }	3.23	8.07	21.6
5 }		5.93	20.6
6 }	3.87	8.29	18.5
7 }		6.00	19.4
8 }	4.32	8.35	17.9

* Proceedings, 1898, p. 209.

† E. Meyer, Z. d. V. d. L., 1899, p. 361.

This table shows that, whatever the ratio of compression, the gas consumption is less with the leaner mixtures. The cause for this, besides the theoretical reason, may possibly be found in the fact that with leaner mixtures the maximum temperatures in the cycle are lower than with rich mixtures, always assuming, of course, that the mixture contains no excess gas.

How the efficiency of an engine may be affected by careless setting of the fuel valve is well shown in some results quoted by Lucke.* The test cited is on a 10 I. H. P. Otto engine governed by hit and miss. The fuel used was carbureted water gas.

Gas Valve Number	Efficiency
9	16.0
8	16.5
7	18.0
6	19.0
5	10.0

It is quite evident from the figures that the last setting wasted a lot of unburned gas, while the leaner mixtures were inefficient, probably due to sluggish combustion. The same thing was noticed in a series of tests on German alcohol engines, in which it was found that the setting of the fuel needle valve had a pronounced effect upon the economy.

5. Variation of Point of Ignition. — The effect on the appearance of the diagram of varying the point of ignition has already been discussed. To get some idea of the influence of varying time of ignition on engine capacity and efficiency, the following figures are given. The first set † was obtained in determining the range of adjustment of the igniter gear on an 8" x 12" horizontal hit-and miss-engine, running 265 r.p.m. on natural gas.

* Lucke, Gas Engine Design.

† Obtained through the courtesy of Mr. A. B. Gould, of the Wellman, Seaver, Morgan Co., Cleveland.

Card No.	Crank Angle below horizontal at time of ignition	Max. brake load possible at 265 R. P. M.
8	35°	47 lbs. net
9	33°	47 " "
10	32°	47½ " "
11	24°	47 " "
12	23°	44½ " "
13	15°	44 " "

The accompanying indicator cards are shown, much reduced, in Fig. 17-1. The scale of spring was 160 pounds.

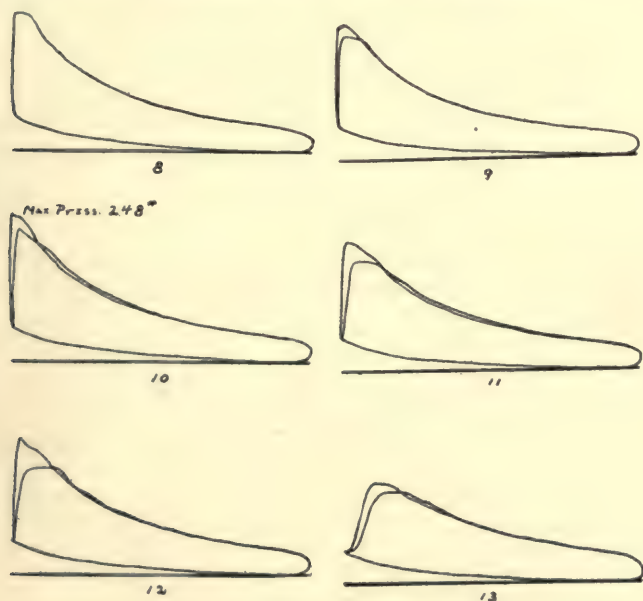


FIG. 17-1.

The second set of figures is due to Mr. J. R. Bibbins, and was published by him in the *Michigan Technic* for February, 1907. They are here reproduced by permission of the author, obtained through Mr. R. D. Day of the Westinghouse Machine Co. The tests were made on a 9 x 11", 2-cylinder Westinghouse gas engine. The load was kept constant at about 70 B. H. P., the speed was held constant at about 300. r.p.m. The gas used was natural

gas with a constant lower heating value of 934 B. T. U. The point of ignition was changed by steps from dead center to 55 degree crank angle ahead of the center, that is, the spark was advanced to that extent. The following table shows the results:

Point of Ignition Degrees early	Load B. H. P.	R. P. M.	Gas per B. H. P. per hour cu. ft.	B. T. U. per B. H. P. per hour	Thermal Efficiency on Brake %	Relative Efficiency	PRESSURES LBS. per sq. inch	
							Max.	Release
0	70.0	292	14.38	13410	19.0	.815	151	36
8	70.8	295	13.34	12470	20.4	.875	168	36
20	71.0	296	12.36	11530	22.1	.947	177.5	33.6
25	71.0	296	12.3	11490	22.2	.951	220.5	31.2
30	71.3	297	11.71	10940	23.3	1.000	252	31.2
35	71.3	297	12.03	11230	22.7	.972	252	28.8
45	71.2	296.5	12.40	11590	21.9	.942	379	28.8
55	70.0	292	15.74	14700	17.3	.742	437	24.0

The results of the test are shown graphically in Fig. 17-2. The best lead angle for the sparking gear appears to be between 30 and 35 degrees, in which this test agrees closely with the re-

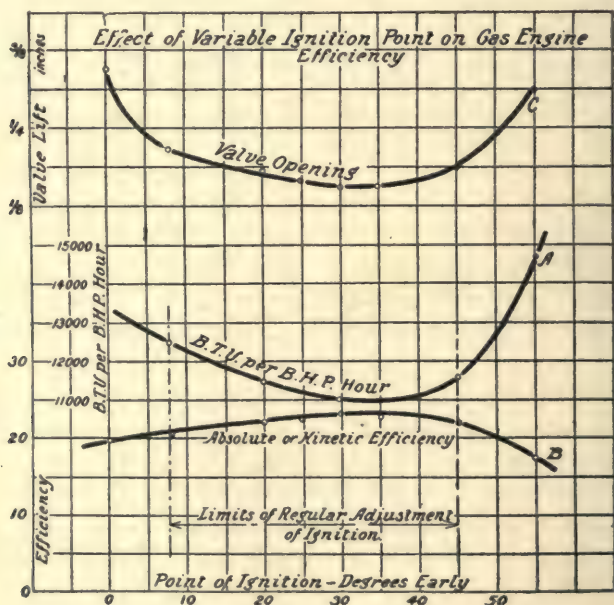


FIG. 17-2.

sults obtained by Mr. Gould on a similar engine. Fig. 17-3 shows the typical indicator card accompanying each igniter position.

6. Engine Economy depending upon Load. — As is the case in the steam engine, the efficiency of a gas engine decreases as the load decreases below the normal. The amount of this decrease varies in different engines, depending mainly upon the

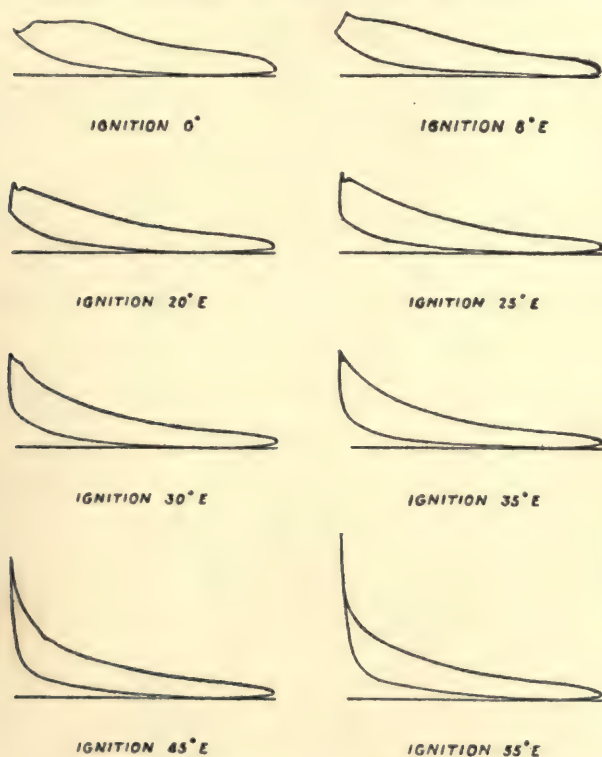


FIG. 17-3.

kind of fuel used and the system of governing employed. The following set of curves, Fig. 17-4, makes this clear. Regarding the range of load above normal, however, it is found that, while a steam engine generally shows a decrease in efficiency for overloads, the gas engine usually shows a greater efficiency at the maximum load than at the normal; in other words, just as long as a gas engine keeps up the normal speed under an increase in

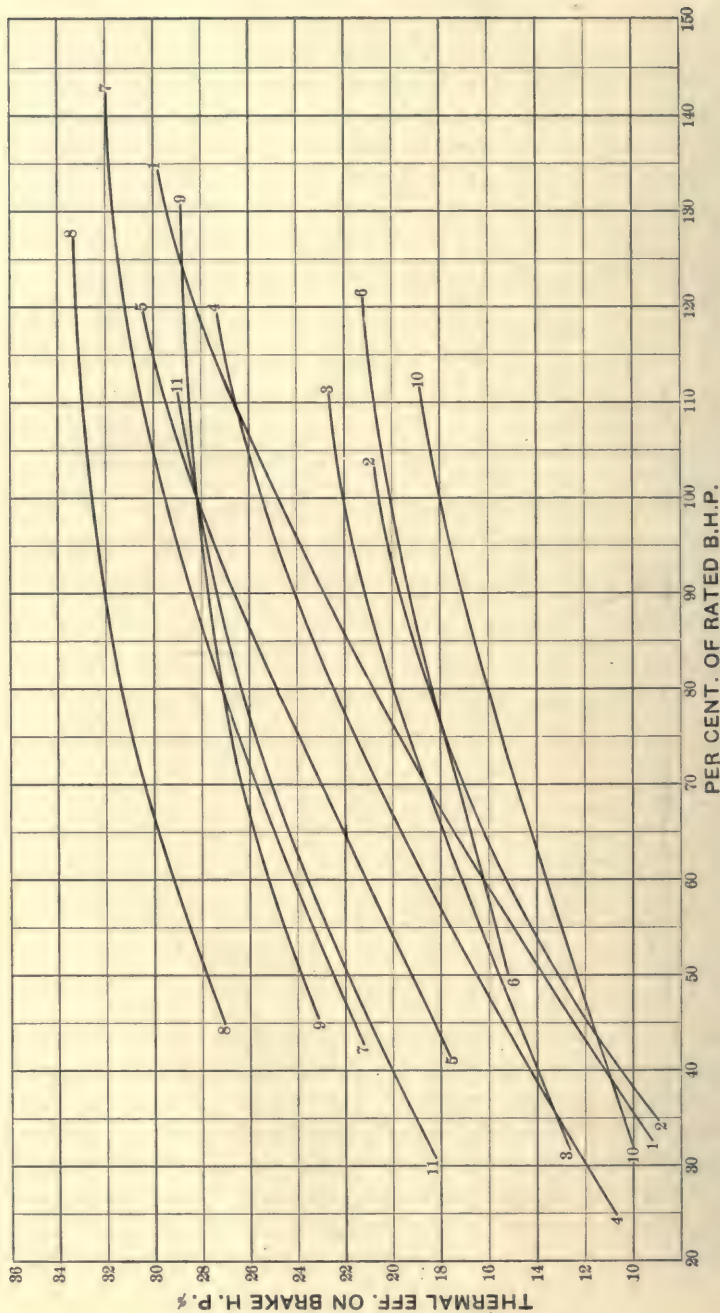


Fig. 17-4.

load the thermal efficiency will rise with the load. A few typical efficiency curves are shown in the figure. The data for these has been collected from various sources, as shown in the accompanying table:

Curve No.	Type of Engine	B.H.P.	R.P.M.	Fuel	Governing	Reference
1	Deutz, Single Cylinder	50	200	Ill. Gas	Throttling Gas.	E. Meyer, Z. d. V. d. I., 1898.
2	Westinghouse 3-cycl. Vert.	100	270	Natural Gas.	Throttling Mix.	Robertson, 1899.
3	Deutz.	450	—	Producer Gas.	—	Guarantee Fig. for entire plant. Josse, 1904.
4	Güldner, Single-cylinder.	35	220	Producer Gas.	Throttling	Güldner, 1906, for plant including generator.
5	Nürnberg.	1200	106	Blast-Furnace Gas.	Throttling Gas.	Riedler, Gross-Gas-Maschinen, 1905.
6	Swiderski, Single-cylinder.	15	235	Alcohol.	Hit & Miss.	E. Meyer, Z. d. V. d. I., 1903.
7	Deutz, Single-cylinder.	12	285	Alcohol.	Throttling.	"
8	Diesel.	70	158	Russian Kerosene.	Cut-off.	"
9	Diesel.	8	275	"	"	"
10	Hornsby-Akroyd	25	202	Kerosene.	Regulating Oil.	Robinson, 1898.
11	Bánki.	25	210	Gasoline with water injection.	Hit & Miss.	Jonas and Taborsky Z. d. Oest. Arch. & Ing V., 1900.

7. The Heat Balance. — Accounting for the heat furnished to a heat engine is called constructing the *Heat Balance*. In an internal combustion engine the heat supplied to the engine is that contained in the fuel furnished to the engine in a given time. For convenience all heat calculations are referred to some standard temperature, usually room temperature. It is usual to account for the heat in four separate items:

1. Heat represented in indicated work.
2. Heat carried off in the jacket water.

3. Heat lost in exhaust.

4. Heat loss due to radiation, conduction, etc.

Of these the first and second items admit of accurate determination, the fourth is nearly always found by difference between the sum of the other three items and the heat supplied. Item three is much more difficult of exact determination. Its calculation involves the determination of the weight of exhaust gases and of the specific heat of these gases at exhaust temperature.

The only accurate way to find the weight of the exhaust gases is by metering or otherwise determining the air supply to the engine. The weight of the exhaust gases is in all cases the weight of air plus the weight of the fuel.

There are two other ways sometimes employed, but either one can only give approximate results. One of these determines the charge weight from the piston displacement. This involves

(a) The volumetric efficiency of the suction stroke, and

(b) Some assumption as to the temperature of the charge at the end of the suction stroke.

It is possible to determine the volumetric efficiency with fair accuracy by means of a weak spring card, but the second point offers much more difficulty. The charge at the end of the suction stroke, taking the case of a gas engine, consists of a certain amount of air, of fresh gas, and of burned gases from the clearance. The entering temperature of air and of gas can be accurately determined, but the temperature of the mixture, as it enters the cylinder, changes, due to contact with the hot walls and to mixing with the clearance gases. Nothing is definitely known of the weight of the clearance gases, for although their pressure and volume are known, nothing is known of the temperature. Hence neither the wall effect nor that due to the clearance gases can be definitely gaged and all temperature computations therefore become approximate. The only positive way of determining the temperature at the end of the suction stroke is by actual measurement. This has been successfully done, but the apparatus is not such as could well be employed in ordinary testing.

It must be evident, therefore, that piston displacement measurement of the weight of the exhaust gases can be approximate only. To cite a case in point, the test of Brooks and Stewart, mentioned in Chapter V, showed an actual ratio of air to gas on

test of 6.63, while piston displacement computation showed 8.32.

The second approximate computation for the weight of the exhaust gases is based upon the exhaust gas analysis. The method of doing this has been explained in detail in Chapter VI. The trouble with this scheme lies in the difficulty of obtaining representative gas samples. But granting even that these are obtained, it is often found that computations based upon the analyses show an excess coefficient smaller than that really used. That is, the weight of exhaust gases so determined is less than the actual amount. Thus, Schöttler, in computation on some of Slaby's tests, shows in one case that, based upon analysis, the excess coefficient was 5.52, while in reality it was 6.2. Schöttler attributes this discrepancy to a change in the analysis due to a burning up of the lubricating oil, which is apt to increase the CO_2 content of the exhaust gas at the expense of the percentage of O. It has been shown that a variation in the supply of lubricating oil may change the fuel consumption under circumstances quite materially, and Schöttler's surmise is therefore probably correct. At the same time, however, any such effect must be more marked in the smaller machines, and the writer believes that, given representative samples of exhaust gas from anything but the small machines, in conjunction with accurate fuel analysis, a very close approximation to the actual weight of the exhaust gases can be obtained. At any rate the method should be more accurate than that based on piston displacement.

Finally, it should not be forgotten that even an accurate determination of the air supply still leaves open the question of the specific heat of the exhaust gases.

The heat balance of the four items above outlined is sometimes shortened to three by combining items 3 and 4 and determining their sum by difference. On the other hand, a balance of many more items can be drawn up. Thus each event of the cycle may be examined by itself and the heat and energy interchanges be determined. A very detailed balance of this kind is given in Schöttler's "*Die Gasmachine*," p. 321. How far this question needs to be entered into depends altogether upon the importance of the test, but an effort should be made in every case to so arrange

the apparatus that at least a heat balance of the kind first discussed can be drawn up from the results of the test.

8. Results of Tests of Engines and Gas Producers. — By this term is here meant the results shown by engines or producers when operated under normal conditions and at or near normal load. The number of tests from which this data can be obtained is at the present writing quite large, and some notable collections of test data have been made. The most extensive is perhaps that contained in Appendix A of Brian Donkin's "Gas, Oil and Air Engines." This collection consists of eleven tables, arranged according to fuel used, containing in all some 280 tests. Another large collection is that found appended to Witz's "Moteurs à Gas et à Pétrole." Such collections are valuable as showing what has been done, and serve as a guide as to what may be expected from an engine under design or construction. The greatest care, however, should be exercised to see that only reliable data is incorporated.

ENGINE TESTS. — Table I (pp. 544 and 545) contains a series of engine test data taken from various sources. The tests are arranged according to the kind of fuel used, this being the most logical way. In some cases not all the data is given in the original report. Wherever possible computations have been made to make the items complete. In many cases, however, not enough information is given to permit of this, and the record necessarily remains incomplete.

TESTS OF PRODUCERS AND PRODUCER PLANTS. — Table II, p. 546, gives some of the results obtained on tests of producers. In some cases the tests refer to producers only, giving no data regarding the engine used; in others the data is fairly complete for the entire plant.

In conclusion, Fig. 17-5 shows a set of curves drawn by C. H. Day in 1905 during an investigation on the economy of producer gas. The curves show the relation between cubic feet of gas used and brake horse-power for various kinds of gas. While it must be understood that they are approximate only, they represent a fair general average of what is accomplished to-day. Many plants show much better fuel consumption, but others show

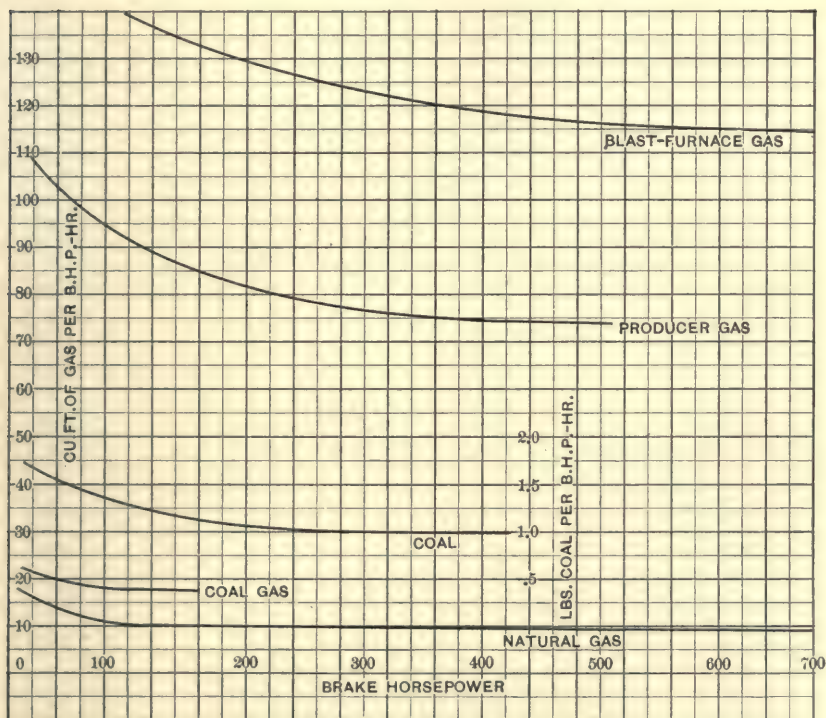


FIG. 17-5.

correspondingly worse. The curves also show in a measure the sizes up to which the various engines are built. Thus an illuminating gas engine of 150 horse-power is a large engine of its type. Producer plants in 1905 apparently were not built much larger than 500 horse-power, while natural gas and blast furnace gas engines were built exceeding 1000 horse-power, and for the latter gas 2000 and 3000 horse-power is not now out of ordinary.

TABLE I.

Kind of Fuel	Name of Engine	DIMENSIONS AND OTHER DATA					Rated B.H.P.	R.p.m.
		Dia. of Cyl. Inch.	Stroke Inch.	No. of Cyls.	2 or 4-cycle	Single or Double Act.		
Illuminating Gas	Koerting	—	—	—	4	S	—	161
Illuminating Gas	Deutz	4.96	5.92	1	4	S	2	260
Illuminating Gas	Güldner	—	—	1	4	S	15	—
Illuminating Gas	Güldner	10	15.7	1	4	S	20	210.7
Illuminating Gas	Bánki	—	—	1	4	S	16	255
Illuminating Gas	Tangye	10	19	1	4	S	—	193.6
Illuminating Gas	Crossley	7	15	1	4	S	—	200
Illuminating Gas	Westinghouse	—	—	3	4	S	—	235
Natural Gas	Westinghouse	25	30	3	4	S	550	150
Natural Gas	Snow	25	48	4	4	—	—	—
Natural Gas	Otto	11.25	18	1	4	S	36	220.4
Natural Gas	Walrath	13	14	3	4	S	75	253.3
Producer Gas	R. D. Wood	25	30	2 (Tandem)	4	S	300	149
Producer Gas	Westinghouse	—	—	3	4	S	—	235
Producer Gas	Crossley	26	36	—	4	S	—	152.4
Producer Gas	Koerting	21.6	37.7	1	2	D	—	101
Producer Gas	Deutz	—	—	—	4	—	—	161.6
Coke Oven Gas	Oechelhäuser (Borsig)	65	37.5	1	2	2 pistons in 1 cylinder	500	110.6
Mond Gas	Crossley	16.9	24	2 (Opposed)	4	S	—	162
Mond Gas	Crossley	26	36	2 (Opposed)	4	S	400	148.5
Blast Furnace Gas	Nürnberg	33.5	43.4	2 (Tandem)	4	D	—	105.6
Blast Furnace Gas	Berlin-Anhalt	16.95	27.5	1	4	S	60	160.6
Blast Furnace Gas	Cockerill	51.2	55.2	1	4	D	—	94.57
Gasoline	Fairbanks	6.5	9	1	4	S	7	300
Gasoline	Springfield	6.5	12	1	4	S	6	230.4
Gasoline	Lozier	5	6	1	2	S	5	513
Gasoline	Westinghouse	5.75	8	2	4	S	10	289.7
Gasoline	Bánki	—	—	1	4	S	—	209.1
Gasoline	Daimler	3.56	5.11	4	4	S	16-20	400
Gasoline	Daimler	3.56	5.11	4	4	S	16-20	600
Gasoline	Daimler	3.56	5.11	4	4	S	16-20	1000
Kerosene	Diesel	10.23	16.16	1	4	S	—	186.6
Kerosene, Russian	Diesel	15.75	23.65	1	4	S	70	158.8
Kerosene, Russian	Diesel	6.65	10.60	1	4	S	8	270.3
Kerosene	Priestman	10.9	14.1	1	4	S	—	172
Kerosene	Grob & Co.	9.07	9.07	1	4	S	8	266
Kerosene	Swiderski	10	10	1	4	S	—	249
Kerosene	Koerting	6.9	10.82	1	4	S	—	222
Kerosene	Blackstone	7	12	1	4	S	—	240
Kerosene	Stevenson	9.5	18	1	4	S	—	200
Kerosene	Hornsby	8.2	14	1	4	S	—	213.8
Kerosene	Hornsby	14.5	17	1	4	S	25	202.6
Crude Oil	Diesel	—	—	—	4	S	225	169.1
Alcohol, 86.1 vol. %	Deutz	8.35	11.8	1	4	S	12	276.9
Alcohol, 86.1 vol. %	Mariefelde	9.95	15.75	1	4	S	14	197.6
Alcohol, 86.1 vol. %	Koerting	6.16	9.95	1	4	S	6	307.3
Alcohol, 87 vol. %	Bánki	—	—	1	4	S	20	225

ENGINE TESTS

H.P. ON TEST		LOWER HEATING VALUE OF FUEL B.T.U. PER		HEAT DISTRIBUTION, %				Mech. Eff. %	Thermal Eff. on B.H.P. %	References and Remarks
I.H.P.	B.H.P.	Lb.	Cu. ft.	Work I.H.P.	Jacket	Exhaust	Rest			
—	108.1	—	500	—	—	—	—	—	28.22	De La Vergne Machine Co., Cat.
2.30	1.72	—	572	21.5	50.4	25.0	4.1	75	16.1	Wimplinger, Zeitschrift d. V. d. I. Sept. 8, 1906.
—	—	—	580	40.5	—	—	—	—	—	Dubbel, Z. d. V. d. I., Nov. 3, 1906.
35.9	—	—	—	42.7	33.2	24.1	—	—	—	Test by Schröter, Z. d. V. d. I., June, 1904.
—	17.03	—	—	—	27.97	—	—	—	31.0	Schimanek, Z. d. V. d. I., Jan. 17, 1903.
28.6	25.4	—	609	—	—	—	—	85	24.5	Witz, 1902.
14	12	—	680	—	—	—	—	86	22	Clerk, 1894.
196.5	147.5	—	597.4	35	—	—	—	76	26.7	Ballinger & Hunt, Sibley College Thesis, 1904.
618	550	—	1000	27.1	—	—	—	89	24.1	J. R. Bibbins, A.I.E.E., Dec. 1903.
736.7	594.5	—	1175(?)	29.4	—	—	—	80.7	23.7	Hastings & Parker, Sib. Coll. Thesis, 1901.
36.3	28.8	—	1086	20.2	—	—	—	79.5	16.1	Hunting, Sib. Coll. Thesis, 1902.
86.7	78.7	—	1041	27.1	49.5	23.4	—	78.7	21.3	Geer & Vanelain, Sib. Coll. Thesis, 1902.
242	169	—	145	24.4	25.0	50.6	—	70	17.1	Goldsmith & Hartwig, Sib. Coll. Thesis, 1906.
173.6	124.8	—	123.8	33	—	—	—	71.8	23.7	Ballinger & Hunt, Sib. Coll. Thesis, 1904.
377.9	313	—	90.13	—	—	—	—	83	21.78	Humphrey, Inst. Mech. Eng., 1900.
481	341	—	129.5	—	—	—	—	71	24.1	E. Meyer, Jour. für Gas Belenchtung, 1900.
81	70.4	—	113	—	—	—	—	87	29	E. Meyer, 1903.
765 (net)	628 (Blowing Cyl.)	—	376	33.4	—	—	—	82.1	27.5	E. Meyer, Z. d. V. d. I., Feb. 1905. Test No. VIIb.
141.6	—	—	135	29.1	—	—	—	—	—	} Humphrey's Test, Schöttler, Z. Jan. 1902.
440	364	—	138	28.6	—	—	—	83	23.7	
1427	1186	—	88	33.9	—	—	—	83.1	28.2	Riedler, Gross-gasmaschinen, p. 158.
79.5	54.5	—	102	—	—	—	—	—	26.2	E. Meyer, Z. 1899, p. 448.
786.16	575	—	99.5	—	—	—	—	—	20.4	Hubert, March, 1900.
11.28	7.23	18200	—	20.5	—	—	—	64	13.15	Elwood, Ford, Garrow, Sib. Coll. Thesis, 1907.
9.63	6.16	20078	—	15.8	—	—	—	64	10.2	Keeley & Spier, Sib. Coll. Thesis, 1900.
6.28	4.92	18520	—	19.8	—	—	—	78.9	15.7	Bayne & Speiden, Sib. Coll. Thesis, 1902.
9.19	5.58	—	—	28.5	—	—	—	60.7	17.3	Glasgow & Powley, Sib. Coll. Thesis, 1902.
—	26.4	—	—	—	—	—	—	—	28	Jonas & Taborsky, Z. Oest. Arch. & Ing. Verein, p. 512, 1900.
—	—	17500	—	19.3	—	—	—	86.1	16.6	} Prof. B. Hopkinson, Cambridge, 1906.
—	—	17500	—	22.0	—	—	—	85.4	18.8	
—	—	17500	—	24.2	—	—	—	79.8	19.3	
30.46	20.81	18604	—	37.7	—	—	—	68.3	25.8	Cat. American Diesel Engine Co.
88	69.6	18610	—	40.3	—	—	—	79.1	31.9	} E. Meyer, Z. d. V. d. I., May, 1903.
11.19	8.6	18610	—	35.8	—	—	—	77.0	27.6	
10.69	10.18	19800	—	—	—	—	—	95.2(?)	13.4	Hartman, Z. d. V. d. I., 1895, p. 586.
11.68	9.22	19800	—	—	—	—	—	79	13.6	Hartman, Z. d. V. d. I., 1895, p. 616.
—	10	19600	—	—	—	—	—	—	15.8	} Schöttler, die Gasmaschine, p. 207.
—	4.15	19699	—	—	—	—	—	—	9.9	
—	3.05	18000	—	—	—	—	—	—	19.7	} Engineering, Vol. 88, 1899.
—	12.44	18000	—	—	—	—	—	—	9.2	
6.07	4.95	18600	—	16	29	55	—	81.5	13	Robinson, 1893, Gas & Pet. Engines, p. 702.
31	26.74	18870	—	21	50	29	—	86.0	18	Robinson, 1898, Gas & Pet. Engines, p. 710.
—	249.7	19460	—	—	—	—	—	—	28.1	Kimberley & Clark Paper Co., Kimberley, Wis., Power, Oct. 1906.
—	16.8	9900	—	—	—	—	—	—	31.6	} E. Meyer, Z. d. V. d. I., April, 1903.
—	19.77	9900	—	—	—	—	—	—	32.7	
—	7.39	9900	—	—	—	—	—	—	21.8	
—	32.13	9700	—	—	32.1	—	—	—	30.18	Schimanek, Z. d. V. d. I., Jan. 17, 1903.

TABLE II. TESTS OF GAS PRODUCERS

Make of Engine	Make of Producer	Type of Producer	Kind of Fuel	Heating Value of Fuel B.T.U. per cu. ft.	ENGINE			HORSE-POWER		Producer Efficiency %	FUEL, LBS. PER HOUR		Heating Value of Gas B.T.U. per cu. ft.	References and Remarks
					Dia. Cyl. Inch.	Stroke Inch.	R. p.m.	Indicated	Brake		L.H.P.	H.P.		
R. D. Wood	Taylor	Pressure	Anthracite	10240	25	30	149	242	169	79.4 (cold gas)	1.28	1.83	145	Goldsmith & Hartwig, Sibley College Thesis, 1905, Zeitschrift des Vereins deutscher Ingenieure, June 25, 1904, } Z. d. V. d. I., June 17, 1905.
Guldner	Guldner	Suction	Anthracite	13630	—	—	220.8	44.8	35.6	—	.692	.875	—	
Mees	Fintsch	Suction	Coke	13270	11	15.7	220	25.76	20.2	—	.766	1.09	—	
Campbell	Fintsch	Suction	Anthracite	14620	11	15.7	220	26.30	21.48	—	.732	.895	—	
Campbell	Campbell	Suction	Scotch Anth. Pea Coal	—	9.5	18.0	192.4	23.73	20.44	—	—	.93	—	
Crossley	Crossley	Suction	"	—	7	12	232.5	9.66	8.25	—	—	1.22	—	
Ind. Eng. Co. Manchester	Ind. Eng. Co. Manchester	Suction	"	—	8.5	14	189.2	18.25	15.35	—	—	.77	—	
National	National	Suction	"	—	8.5	14	204.9	—	9.72	—	—	1.13	—	
National	National	Suction	"	—	7	15	219.80	11.88	9.74	—	—	.84	—	
National	National	Suction	"	—	10	18	193	25.54	20.57	—	—	.80	—	
Tangye	Tangye	Suction	"	—	7	16	224.4	10.86	8.34	—	—	1.25	—	
Nürnberg, Single Act.	Nürnberg, Single Act.	Suction	Anthracite	—	10	19	197.9	23.45	19.98	80	.78	.975	—	
Nürnberg, Single Act.	Nürnberg, Single Act.	Suction	Coke	—	—	—	—	134.4	107.4	75	.93	1.16	—	
Tangye	Tangye	Suction	Scotch & Welsh Anth. Bit. & Pocahontas Slack	—	11	20	186	37.5	110.0	—	—	.709	—	
Westinghouse, 500 H.P.	Loomis-Pettibone	—	Colorado Bit. Bituminous	—	—	—	—	36.34	29.46	—	—	1.42	—	
Westinghouse, 250 H.P.	Taylor	Pressure	Colorado Bit.	9767	—	—	—	—	—	—	—	1.95	—	
Stockport, 250 H.P.	Wilson	Pressure	Bituminous	—	—	—	—	—	—	—	—	1.40	—	
Campbell, 280 H.P.	Dowson	Pressure	Anthracite	—	—	—	—	—	—	—	—	1.12	—	
Westinghouse, 3000 H.P.	Dowson	Pressure	Anthracite	—	—	—	—	—	—	—	—	1.20	—	
Deutz, 250 H.P.	Deutz	Suction	Anthracite	14000	—	—	—	—	—	—	—	.744	—	
Crossley, 90 H.P.	Crossley	Suction	Coke	—	—	—	—	—	—	—	—	.91	—	
Crossley, 300 H.P.	Crossley	Suction	Anthracite	11370	—	—	—	—	—	—	—	.975	—	
Westinghouse, 225 H.P.	Taylor	Pressure	Colorado, No. 1 Lignite	—	19	22	200	—	235.4	64.2	—	1.55	—	
Letombe	Letombe	Suction	—	—	23.6	31.5	135	—	290	—	—	.81	—	

CHAPTER XVIII

COST OF INSTALLATION AND OF OPERATION

1. **Cost of Producers and Engines.** — With the approach to standardization of the design of the internal combustion engine there has also come a tendency toward standard prices. The figures available, however, for anything but small engines are not plentiful. This is no doubt due to the fact that large installations are as yet not numerous and information is correspondingly scarce. The writer has been able to glean the following data from current literature:

PRODUCERS. — The *Gas Engine*, June, 1906, gives the following figures for producers:

Pressure Producers, erected complete but not including freight, \$25 per horse-power for hard coal, and \$27 per horse-power for soft coal producers. These figures are for sizes in the neighborhood of 300 horse-power. The cost is distributed approximately as follows:

- 25 per cent in the producer.
- 15 per cent in the scrubber.
- 30 per cent in the gas holder.
- 30 per cent in piping and fittings.

The increased cost of the bituminous coal plant is mainly due to greater cost of washing apparatus.

Suction Gas Plants will, on a rough estimate, cost only about one-half the above. The reasons for this are greater capacity of producer and the elimination of the gas holder. One maker gives the following approximate figures for the price of suction gas plants from 25 to 250 horse-power.

$$\text{Price per horse-power} = \$11 + \frac{400}{\text{H.P.}}$$

Thus a 25 horse-power plant would cost \$37.00 per horse-power, and one of 250 horse-power \$12.60 per horse-power. In the writer's experience practice agrees fairly well with these figures.

ENGINES. — The cost of stationary gas engines at present seems to range from about \$70 per horse-power for very small engines to in the neighborhood of \$25 per horse-power for large ones. Recent quotations on producer-gas engines for instance showed the cost to be about \$55 per horse-power for a 10 horse-power engine, and \$36 per horse-power for a 50 horse-power machine. The latter is, however, a comparatively low price for an engine of this size. The price of launch engines is frequently lower than this, from \$50 per horse-power for the small engine to about \$20 per horse-power for an engine of 25 to 30 horse-power. The cost of Diesel engines runs somewhat higher, varying probably between \$50 and \$90 per horse-power, depending upon size. None of these figures include cost of erection.

Suction Gas Plants. — Recent quotations for plants of this type have been for a 10 horse-power plant, which is below the ordinary commercial size, \$90 per horse-power for a 50 horse-power plant \$51 per horse-power and again \$61 per horse-power, and for a 75 horse-power plant \$53 per horse-power.

2. Cost of Erection. — The cost of erection varies largely with the size of the foundations. Regarding the latter there seems to be no definite proportionality between it and the size of the engine. In general, of course, vertical engines do not require as large or as heavy a foundation as horizontal engines, but the condition of the ground has a great deal to do with the final cost. Güldner* estimates that from 15 to 20 cubic feet of foundation per brake horse-power of engine should be sufficient in the ordinary case if the ground is safe. For large machines, however, this may even be as low as 8 cu. ft. per B. H. P. The total cost of erection and starting should not exceed about 3 per cent of the purchase price in the case of gasoline and gas engines, and should not be over 5 per cent in the case of producer gas plants. These figures agree fairly well with the estimates made by H. A. Clark in 1903.†

3. Piping and Auxiliaries. — This item is apt to vary within wide limits, especially in producer gas plants, where much depends upon the location of the producer plant with reference to

* *Verbrennungsmotoren*, p. 433.

† H. Ade Clark, *The Diesel Engine*, *Proceedings of Mechanical Engineers*, 1903.

the engine. In a normal case, 5 per cent of the cost price of the engine or plant should cover the cost. This, however, does not include the cost of an air compressor system if this is required to start the plant.

4. **Floor Space and Buildings.**—The cost of buildings is intimately connected with the floor space required by the plant, but outside of this nothing very definite can be said of the cost of the building, since everything depends upon the construction used. The cost based on the square foot of floor space may vary from about \$1.50 for a plain brick building with steel roof, to \$2.75 for a modern steel concrete structure. Among the figures available are the following. A. H. Clark, in the article already mentioned, estimates the cost of building as follows:

Type of Engine	Size of Engine B. H. P.	Cost of Engine, Producer and Accessories	Cost of Building	Ratio Cost of Bldg. Cost of Plant
Producer Gas *	35	\$2,360	\$ 640	.27
Diesel	35	2,860	440	.16
Producer Gas ...	80	4,400	930	.21
Diesel	80	4,400	685	.16
Producer Gas ...	160	7,450	1,150	.16
Diesel	160	8,300	880	.11

Güldner, in making a similar estimate, combines the cost of building and of necessary foundations for the engine and producer. Expressing this combined cost in per cent of the cost of plant and accessories, the following table shows the results:

Type of Plant	Horse-power of Plant													
	5	10	15	20	25	30	40	50	100	125	150	175	200	
Illuminating Gas Engine	27	22	20	19	18	17	16	15	13	13	13	13	13	
Suction Gas Plant Complete	30	26	24	22.5	21	19.5	19	18	15.5	15	15	15	14.5	

Turning next to the floor space required, there is considerable data available in the literature published. Collaborating the

* Pressure producer.

figures given by manufacturers for the floor space required for horizontal single-cylinder gas, gasoline and oil engines, and for suction gas producers, we obtain Curves I and III of Fig. 18-1.

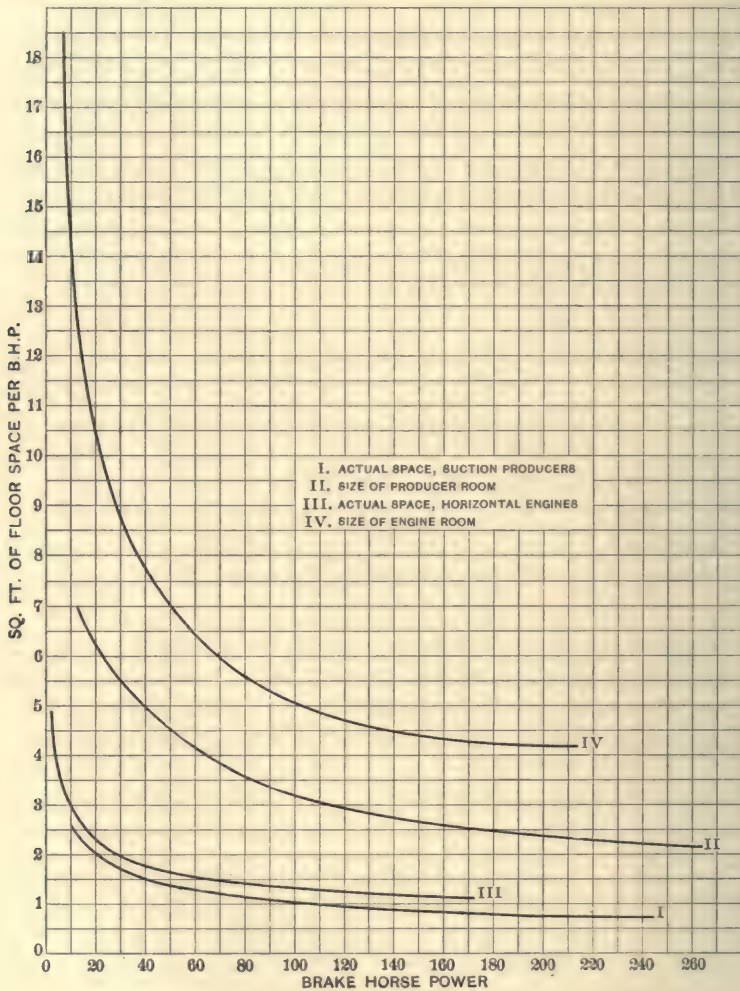


FIG. 18-1.

It should be remembered in connection with these curves, however, that they represent average figures only and they can therefore give only approximate results. For example, a 20 B. H. P.

suction gas producer will, according to Curve I, occupy approximately $20 \times 2 = 40$ sq. ft. of floor space, say a space $9 \times 4\frac{1}{2}$ ft.; similarly, a 200 B. H. P. installation would require $200 \times .8 = 160$ sq. ft.

In like manner Curve III gives the approximate floor space for horizontal engines. Thus a 5 B. H. P. gasoline engine of standard make would probably require $5 \times 4 = 20$ sq. ft., say a space 3 ft. by 6 ft.; while a 150 B. H. P. producer or illuminating gas engine calls for $150 \times 1.5 = 225$ sq. ft. The curve does not include multi-cylinder engines because the available figures for this type of engine are very erratic. For the same reason no curve could be drawn for vertical engines.

While Curves I and III represent the space actually occupied by the plant, they give no idea of the size of the producer or engine room required. In the producer room space must be left for the proper cleaning of the producer, etc., and in the engine room space for dismantling, etc. On this point the only information that seems available is that given by Güldner in his estimates, the results of which are plotted in Curves II and IV of Fig. 18-1. The former shows the approximate room allowance for suction gas apparatus, the latter for horizontal single-cylinder engines, based on one brake horse-power in each case.

The data available for *pressure producers* is not so extensive as that for suction producers. In the former plants much depends upon the size of the gas holder. Outside of this the space required for suction and pressure producers should not vary greatly. The figures at hand seem to indicate that pressure producer plants, including the gas holder, of from 30 to 40 horse-power occupy 75 per cent more space than suction plants of like capacity. As the power increases the difference grows less, a plant of 100 horse-power requiring apparently only about 50 per cent more space, and one of 175 horse-power only 20 per cent more.

The *minimum head room*, i.e., the height of the room required for either type of plant, seems to vary from 12 ft. in the small 20 horse-power plant to 18 ft. in the large 250 horse-power producer.

In Power for April, 1907, L. L. Brewer published the following table of "Practical Data for Modern Gas Engines." The table gives in column 13 figures for the floor space required by engines larger than those so far discussed. The letters in column 7 have

the following meaning: s.c., single cylinder; tw., twin; td., tandem; d. td., double tandem.

1	2	3	4	5	6	7	8	9	10	11	12	13
Brake Horse-Power	Revolutions per Minute	Builder	No. Cylinders	Strokes per Cycle	Single or Double Acting	Cylinder Arrangement	Weight of Engine without Fly-wheel, Lbs.	Weight of Fly-wheel for Blowing and Pumping Service, Lbs.	Weight of Fly-wheel for Dynamo Driving	Weight of Engine and Fly-wheel (9) per B. H. P.	Weight of Engine and Fly-wheel (10) per B. H. P.	Sq. ft. Floor Space per B. H. P.
100	150	Cockerill	1	4	s.	sc.	45,000	9,000	21,100	540	661	2.05
200	105	Cockerill	1	4	s.	sc.	83,000	25,000	58,500	540	706	1.81
250	150	Cockerill	2	4	s.	td.	65,000	10,000	23,400	300	353	1.24
300	120	Deutz	1	4	s.	sc.	83,500	35,000	81,800	295	551	2.07
300	120	Deutz	2	4	s.	tw.	101,000	14,000	32,800	383	447	1.52
300	140	Deutz	4	4	s.	d. tw.	110,000	3,500	8,200	484	394	1.32
600	80	Cockerill	1	4	s.	sc.	207,000	100,000	234,000	512	734	0.99
600	130	Cockerill	2	4	s.	td.	185,000	46,000	107,500	386	487	1.13
800	110	Oechelhäuser	1	2	s.	sc.	143,000	48,000	112,000	318	425	1.23
800	130	Deutz	2	4	s.	tw.	158,000	28,000	65,500	310	371	1.67
800	130	Deutz	4	4	s.	d. tw.	189,000	7,000	16,400	327	342	1.08
800	110	Koerting	1	2	d.	sc.	136,500	18,000	42,200	258	297	1.11
750	90	Nürnberg	1	4	s.	sc.	297,000	115,000	26,900	560	538	1.03
1200	80	Cockerill	2	4	s.	td.	365,000	95,000	222,000	383	488	0.68
1200	130	Deutz	4	4	s.	d. tw.	354,000	14,000	32,800	307	322	1.01
1200	120	Nürnberg	4	4	s.	d. tw.	280,000	16,000	37,400	246	264	0.94
1200	110	Oechelhäuser	2	2	s.	tw.	260,000	16,000	37,400	230	248	0.9
1200	110	Koerting	2	2	tw.	td.	250,000	4,500	10,500	212	218	0.9
1400	110	Cockerill	2	4	d.	td.	374,000	8,600	20,000	170	164	0.42

It will be noted from column 6 that there is only one double-acting four-cycle engine in the list. The large engines cited are all single-acting twin or tandem, or double engines. It would naturally be assumed that the later types of large four-cycle engines, which are nearly always double-acting tandem, or double-acting twin tandem machines, should show a decrease in the floor space required as compared with the figures in column 13. An indication of the saving in both weight and floor space that may be effected by adopting the double-acting principle is given by the figures for the 1400 B. H. P. double-acting tandem Cockerill engine in the last line of the table. The saving there shown is remarkable and the writer has not been able to check it in the case of other engines the design of which has been changed from the twin or double-twin single-acting to the double-acting tandem. The reason for this probably is that opposed single-acting engines usually work without a cross-head,

while double-acting cylinders demand the use of the same. Hence the total length of engine is not changed materially.

5. Cost of Operation. — The total cost of operation consists of the following items:

- (a) Interest on capital.
- (b) Depreciation of Plant and Buildings.
- (c) Insurance.
- (d) Fuel Cost.
- (e) Cost of Cooling Water.
- (f) Lubricating Oil and Waste.
- (g) Attendance.
- (h) Maintenance and Repairs.

Of these, the first three items are usually called the *fixed charges*, and the last five the *operating* or *works cost*. The sum of fixed charges and works cost is the *total operating cost*.

(a) **INTEREST ON CAPITAL.** The usual allowance in this country for interest on capital is 6 per cent.

(b) **DEPRECIATION OF PLANT AND BUILDINGS.** There is little doubt that as far as gas producers are concerned, the wear and tear on this part of the plant is much less than it is on a boiler plant of the same capacity. On the other hand, the stresses in a gas engine are generally higher throughout than those in a steam engine of the same power, and this naturally leads to a shorter life and consequently somewhat higher allowance for depreciation. Taking it altogether, an allowance of from 7 to 10 per cent of the capital outlay for the power plant should cover depreciation, the lower figure for a producer plant, the higher where engines alone are concerned.

Depreciation on the building should not exceed 2 to 3 per cent of the building cost. In case the building is rented, the rent instead of depreciation should be charged against the plant.

(c) **INSURANCE.** This item is usually very small as compared with the rest and is therefore in most cases neglected.

(d) **FUEL COSTS.** This item is very often used as the sole criterion of the economic status of a plant, but in many cases this is not true. Where the cost of fuel is very high, as for instance where illuminating gas or gasoline are used, the fuel cost usually forms the major part of the operating expenses. But where the cheaper fuels are used or the plant is of high efficiency, as is often

the case in producer plants, an analysis will show that what are usually considered the "incidental" expenses in many cases far outweigh the fuel cost.

The fuel cost varies with the load on the engine. The figures given by manufacturers usually represent the best figures obtained at maximum load, but such conditions rarely obtain in practice. At any rate it would not be safe to base computations upon such "parade" figures. Haeder * considers that the economy figure given by an engine at $\frac{7}{10}$ maximum load, which is equal to 85 per cent of rated load if an over-capacity of 20 per cent is assumed, is the best figure on which to base computations. The same writer gives the following table for the variation of the fuel consumption with load. The third line in this table gives the same information regarding a steam plant:

Gas Engine	VARIATION OF FUEL CONSUMPTION WITH LOAD									
	Max. Load	.9	.8	.7	.6	.5	.4	.3	.2	.1 of Max. Load
Hit-and-miss regulation	1	1.03	1.08	1.14	1.23	1.35	1.50	1.8	2.5	3.0
Throttling regulation	1	1.06	1.13	1.21	1.33	1.50	1.75	2.2	3.0	5.0
Steam plant	1.1	1.05	1.02	1.0	.96	1.02	1.07	1.15	1.30	1.60

The table is based upon the assumption that the gas engine operates normally on $\frac{7}{10}$ of its maximum load, and this is put equal to the normal load on the steam engine. It is interesting to note the superiority of hit-and-miss regulation, as far as economy is concerned, over the other method of governing, and the superiority of the steam engine over both as regards small variation in economy over a wide range of load.

Gulder † makes a similar estimate with the following results:

INCREASE OF FUEL CONSUMPTION WITH DECREASING LOAD

Approximate Load	.75	.66	.50	.33	.25	of Normal Load.
Illuminating Gas Engine.	10	20	35	60	90	} per cent higher fuel consumption than at full load.
Suction Gas Engine	20	30	50	75	100	
Diesel Oil Engine.....	10	20	30	55	80	

* Haeder, Die Gasmotoren.

† Güldner, Verbrennungsmotoren, p. 427.

Thus, for example, if an illuminating gas engine uses 20 cu. ft. of gas per B. H. P. hour at full load, we may except it to use $1.35 \times 20 = 27$ cu. ft. per B. H. P. hour at half load, and $1.9 \times 20 = 38$ cu. ft. per B. H. P. hour at quarter load.

The anthracite coal consumption of suction gas plants for various sizes of plants and for different loads is given by Haeder in the following table:

CONSUMPTION OF ANTHRACITE IN SUCTION GAS PLANTS IN POUNDS PER
B. H. P.-hour

Max. Capacity B. H. P. . . .	14	40	70	100	140	210	420
Normal Load B. H. P.	10	30	50	70	100	150	300
Consumption							
pounds per	B. H. P. max.	1.21	1.0	.93	.90	.88	.84
H. P.-hour	.7 B. H. P. max.	1.45	1.19	1.12	1.10	1.06	1.03
based on . .	.5 B. H. P. max.	1.80	1.47	1.38	1.34	1.32	1.28
						1.28	1.25

It is evident from a study of the figures in the above tables that the probable average load at which a prospective plant will operate plays an important part in the estimation of fuel costs, a point which should not be lost sight of.

We find in the engineering literature of the day considerable information regarding fuel consumption and fuel costs. The cost figures given belong to one of two classes: either they are based upon assumptions regarding both consumption of fuel and cost of unit weight of the fuel, or the figures are obtained in actual operation. It goes without saying that the latter class is much more valuable than the former. In the case of assumed costs and consumptions, comparisons are usually made between steam and gas power under various conditions of operation. Such computations are interesting because they show what might be realized in each case, but in attempting to practically apply the information the greatest attention should be paid to the conditions assumed. As regards actual consumption figures, the tables of the previous chapter give considerable information regarding efficiency of operation. From this data it should be easy to compute the fuel costs as soon as the local cost of unit weight is known.

Below are given a few hypothetical computations. In some cases both the fuel consumption and the fuel cost are assumed outright, in others an attempt has been made to determine the

fuel consumption in relation to the size of the engine.* These figures are followed by a few fuel cost figures from actual practice. Additional data for fuel cost from actual operation will be found in the data on total operating costs at the end of this chapter.

The following table shows an estimate taken from the catalogue of a well-known manufacturer.

RELATIVE COSTS OF FUEL WITH DIFFERENT TYPES OF ENGINES

Type of Engine	Class of Fuel	Price of Fuel	Fuel Consumed per B. H. P. per hour	Cost in cents per B. H. P. per hour	Cost per annum 3000 hours per 100 B. H. P.
Simple non-condensing slide valve	Bituminous Coal	\$3.00 per gross ton	5 lbs.	.669	\$2,007.00
Compound condensing	Bituminous Coal	\$3.00	3 lbs.	.402	1,205.00
Steam Turbine	Bituminous Coal	\$3.00	3 lbs.	.402	1,205.00
Oil Engine	Gasoline	14 cents per gal.	0.125 gal.	1.75	5,250.00
Oil Engine	Crude Oil	4 cents per gal.	0.10 gal.	.40	1,200.00
Gas Engine	Illuminating Gas	75 cents per 1000 cu. ft.	19 cu. ft.	1.425	4,275.00
Gas Engine	Natural Gas	20 cents per 1000 cu. ft.	13 cu. ft.	.26	780.00
Gas Engine with suction producer.	Anthracite Coal	\$3.00 per gross ton	1 lb.	.134	402.00
Gas Engine with suction producer.	Coke	\$3.00 per gross ton	1.25 lb.	.167	502.00

The next table, compiled by J. I. Wile, is similar to the above, except that the assumptions differ somewhat. A comparison of the corresponding items in the two tables serves to show the variation in the final results arrived at by computations of this kind.

STATISTICS OF FUEL CONSUMPTION AND COST PER ANNUM OF PRODUCER
GAS POWER AND OTHER POWERS

Type of Engine	Kind of Fuel	Price per Ton	Fuel consumption in lbs. per B. H. P. per hour	Cost in cents per B. H. P. per hour	Cost in dollars per 100 B. H. P. per hour	Cost per 100 B. H. P. per Annum 3000 hours	Gas and Oil Engines Steam Eng. Prod. Gas Eng.
Oil Engine	Gasoline	12 cents per Gallon	1 Pint	1.50	\$1.50	\$4,500	
Gas Engine	Illuminating Gas	75c per 1000 Cubic Feet	18 Cubic Feet per B. H. P.	1.35	1.35	4,050	
Gas Engine	Natural Gas	30c per 1000 Cubic Feet	13 Cubic Feet per B. H. P.	.39	.39	1,170	
Simple Steam Engine...	Bituminous Coal	\$3.00	8 lbs.	1.2	1.20	3,600	
Compound Steam Engine Non-condensing.....	Bituminous Coal	3.00	5	.75	.75	2,250	
Triple Expanding Steam Engine, condensing...	Bituminous Coal	3.00	2	.3	.30	900	
Producer Gas Engine...	Anthracite Coal	5.00	1	.25	.25	750	
Producer Gas Engine...	Gas Coke	3.00	1.25	.1875	.183	565	
Producer Gas Engine...	Bituminous Coal	2.50	1.25	.1565	.1565	470	
Producer Gas Engine...	Anthracite Coal	3.00	1	.15	.15	450	

C. H. Day,* in an investigation on the economy of gas producer engines, compiled the following fuel cost data for the various kinds of prime movers mentioned. The fuel consumption figures were obtained in each case by collaborating the results of a considerable number of tests. The costs per annum were then computed by assuming the cost of unit weight of the fuel. The total time of operation per year is taken at 3000 hours.

ANNUAL FUEL COSTS FOR STEAM ENGINES. COST OF COAL ASSUMED AT
\$3.00 PER TON OF 2000 LBS.

Type of Engine	B. H. P.	Lbs. Coal per B. H. P. per hour	Cost of Coal per B. H. P. per year
Simple, non-condensing	50	5.75	\$25.90
Simple, condensing	100	4.45	20.00
Compound, condensing	200	2.74	12.70
Compound, condensing	600	1.97	8.85
Compound, condensing	1000	1.90	8.55
Triple exp. condensing	2000	1.87	8.40

The data available for *steam turbines* is not as extensive as

*Sibley College Thesis, 1905.

that for steam engines, and the range covered is not so wide as regards capacity. The following figures, also compiled by Mr. Day, show some fuel costs for prime movers of this kind. The working time has again been assumed at 3000 hours per annum.

ANNUAL FUEL COSTS FOR STEAM TURBINES. COST OF COAL ASSUMED AT \$3.00 PER TON OF 2000 LBS.

Number of Tests	Average B. H. P.	Lbs. Coal per B. H. P.-hour	Cost of Coal per B. H. P. per year
27	616	1.740	\$7.83
10	1085	1.735	7.80
5	1359	1.725	7.75
25	1739	1.655	7.44

The computations made on *illuminating* and *natural gas* engines show the following results. Time of operation per year 3000 hours, cost of illuminating gas 75 cents per 1000 cu. ft., cost of natural gas 50 cents per 1000 cu. ft. The latter assumption is high, since in many localities natural gas is sold at 30 or even 20 cents per 1000 cu. ft. It is a simple matter, however, to reduce the figures in the table in the corresponding ratio.

ANNUAL FUEL COST FOR ILLUMINATING AND NATURAL GAS ENGINES. COST OF ILLUMINATING GAS ASSUMED AT 75 CENTS, THAT OF NATURAL GAS AT 50 CENTS, PER 1000 CU. FT.

B. H. P.	Cu. ft. per B. H. P.-hour		Annual Cost of Gas per B. H. P. per year of 3000 hours	
	Illuminating Gas	Natural Gas	Illuminating Gas Engines	Natural Gas Engines
10	22.0	—	\$49.50	\$24.00
20	21.5	16.0	48.30	23.50
30	21.0	15.5	47.20	23.30
40	20.5	15.0	46.00	22.60
50	20.0	14.5	44.80	21.80
75	19.1	13.8	42.80	20.80
100	18.3	13.0	41.10	19.60
200	—	11.4	—	17.10
300	—	10.2	—	15.40
400	—	9.7	—	14.60
500	—	9.4	—	14.10

For *producer gas* engines Day found the following figures for the consumption of gas and of coal, from which the fuel costs per year of 3000 hours are computed, assuming coal to cost \$4 per ton of 2000 pounds.

ANNUAL FUEL COST OF PRODUCER GAS ENGINES

B. H. P.	Cu. ft. of Gas per B. H. P.-hour	Lbs. of Coal per B. H. P.-hour	Annual Cost of Coal per B. H. P.
50	105	1.45	\$8.70
100	96	1.35	8.10
150	89	1.23	7.33
200	83	1.17	7.03
250	79	1.12	6.72
300	76	1.08	6.48
400	73	1.05	6.30
500	72	1.03	6.18

The cost of the gas for *blast furnace* gas engines has in many computations been neglected on the assumption that this gas, if not used in engines, is a mere waste product. Lately, however, it has come to be recognized that some value must be assigned to this fuel since money was expended on it in every case for cleaning it preparatory to making it fit for use in engines. The ordinary method of evaluating this gas is to compare its heat content with that of steam and to assign a value to the gas corresponding to the cost of steam. Thus the cost of the gas will in every locality vary with the cost of coal. As an example, H. Freyn * makes the following computation:

"Let us assume that the price of coal delivered into bins at the plant be \$2.75 per ton, that the coal have a heat value of 13,000 B. T. U. per pound, and, further, that steam of 150 pounds boiler pressure or about 165 pounds absolute pressure be raised by burning this coal under boilers. One pound of steam will then contain 1225 B. T. U. from zero degrees Fahrenheit. Assuming feed water at 70 degrees, there would be required 1155 B. T. U. to generate 1 pound of steam at 150 pounds boiler pressure. In a boiler plant having 65 per cent efficiency, 1000 pounds of coal

* H. Freyn, Available Power and Cost of Operation of a Power Station for Waste Gases from a Blast-furnace Plant. Journal Western Society of Engineers, February, 1906.

could raise $(0.65 \times 1000 \times 13,000) \div 1155 = 7300$ pounds of steam. The value of 1000 pounds of steam would be $2.75 \div (2 \times 7.3) = \$0.188$, or 18.8 c. To this must be added for labor and maintenance approximately 1 c. per 1000 pounds of steam, making the total value of 1000 pounds = 19.8 c. 1000 cu. ft. of blast furnace gas have a heat value of $1000 \times 90 = 90,000$ B. T. U. and are equivalent to $(0.65 \times 90,000) \div 1155 = 51$ pounds of steam, which in turn are worth $(51 \div 1000) \times 19.8 = 1$ c.

"The value of 1000 cu. ft. of blast furnace gas would, therefore, be 1 c."

L. Eberhardt, in an article in the *Zeitschrift d. V. d. I.*,* makes a similar computation, arriving at a somewhat different result. Starting with the assumption that 1000 pounds of steam will cost from 25 to 32 c., according to the price and quality of coal, he finds that 1000 cu. ft. of cleaned blast furnace gas should have a value of from 1.43 to 1.84 c., the gas having a heating value of 102 B. T. U. per cu. ft. This is considerably higher than the value found by Freyn, which is mainly due to the lower grade of coal (11,650 B. T. U. per pound) and the higher grade of gas assumed.

To get some idea of the fuel cost of blast furnace gas power as compared with steam power, Eberhardt, in the article mentioned, gives the following tables. The original tables give the cost per B. H. P. hour, but these have been recomputed to the basis of a year of 3000 working hours, to make them directly comparable with the figures in previous tables.

ANNUAL FUEL COST OF BLAST-FURNACE GAS ENGINES. HEATING VALUE OF COAL TAKEN AT 11650 B. T. U. PER LB. HEATING VALUE OF GAS TAKEN AT 102 B. T. U. PER CU. FT.

Cost of coal per ton of 2000 lbs.....dollars	2.34	2.75	3.18
Cost of 1000 lbs. of steam cents	24.8	28.4	31.8
Cost of 1000 cu. ft. of gas cents	1.43	1.64	1.84

A fair blast furnace gas engine will show the following fuel consumption:

At full load	99 cu. ft. of gas per B. H. P. hour.		
At $\frac{9}{10}$ load	106	"	"
At $\frac{8}{10}$ load	113	"	"
At $\frac{7}{10}$ load	122	"	"
At $\frac{1}{2}$ load	131	"	"

* June 3, 1905.

With these assumptions of gas consumption per B. H. P., and of local cost of coal, the fuel costs per B. H. P. per year of 3000 hours will then be as per the following table:

Cost of Coal per Ton	Per Cent full Load on Engine				
	100	90	80	66	50
	Annual Cost of Blast Furnace Gas per B. H. P. Dollars				
\$2.34	4.26	4.56	4.87	5.25	5.63
2.75	4.85	5.18	5.55	5.97	6.42
3.18	5.46	5.83	6.23	6.72	7.20

The fuel cost of operating on *gasoline* as fuel is of course considerably higher than any of the figures above quoted. The usual assumption made in regard to gasoline engines is that they will require 1 pint of gasoline per B. H. P. hour. This corresponds to a thermal efficiency of about 18 per cent on the brake, a figure which should be reached by a fair-sized engine in good condition. For the smaller machines, however, one pint is perhaps somewhat low, and for say a 2 horse-power machine a consumption of 2 pints per B. H. P. hour is probably a safer assumption. It should also be borne in mind that the purchase price of gasoline varies somewhat with the quantity bought. The following table of fuel costs takes these various points into account.

FUEL COST FOR GASOLINE ENGINES PER B. H. P. PER YEAR OF 3000 WORKING HOURS

Size of Engine, B. H. P.	2	4	6	10	20
Cost of gasoline per gal., cents	20	18	16	14	13
Consumption of gasoline per B. H. P.					
hr., gallons	.25	.20	.18	.15	.12
Fuel cost per B. H. P. per year, dollars	150.00	108.00	86.40	63.00	46.80

(e) COST OF WATER FOR COOLING AND WASHING. Many estimates of total operating costs totally neglect the cost of water for cooling and washing purposes required by the plant, although in many localities this may amount to a considerable item of expense. In general terms, where water has to be brought from city mains, it pays to install cooling apparatus of some kind. Of course for small installations this may be simply a tank from

the surface of which the heat is radiated. The tank is connected at top and bottom with the water jacket of the engine, and the water circulates by convection. Where such a cooling tank cannot be placed in the immediate vicinity of the engine, and connected to the jacket by short straight pipes, a circulating pump of some kind must be used. As the size of the installation grows, cooling towers or cooling ponds have to be resorted to, unless an abundant source of clean water is available without cost, except that of pumping. Under such conditions the cost of cooling water is very materially reduced. With any kind of cooling system the cost then consists of the cost of pumping plus the cost of any clean water that must be supplied from time to time to replace that lost by evaporation. Of course it is not possible to give any definite figures for this cost item, because it depends upon the cost of water in the particular locality, upon the total amount of water circulated, and upon the kind of pumps used. A few figures for the amount of water required or in circulation per B. H. P. hour are given below.

In a producer plant an additional amount of water is required for the producers and scrubbers. The water vaporized for the producers of course is not again available. That used for the washing plant is in general subject to the same considerations as that used for cooling the engines. But where a cooling system is used for both, they should never be combined, because the requirements for clean cooling water for the engines are much more strict. The scrubber water usually carries considerable quantities of sediment and is contaminated with ammonia and sulfur compounds, so strongly in some cases as to give it a very noxious odor. In such cases, settling tanks and ponds, together with a considerable addition of clean water, seem to be the only remedies to recover at least a part of the water.

The approximate quantity of cooling water required may be figured from the following considerations:

One horse-power, assuming an engine efficiency of say 25 per cent at the cylinder, requires the expenditure of $\frac{2545}{.25} = 10,200$ B. T. U. per I. H. P. hour. The jacket loss approximates about 40 per cent of the heat expended, that is, in this case the jacket water must carry off $.40 \times 10,200 = 4080$ B. T. U. per hour.

Assuming the inlet temperature at 70 degrees, the outlet at 150 degrees, we find the number of pounds of water required = approx. $\frac{4080}{150-70} = 51$ pounds = say $6\frac{1}{2}$ gallons per I. H. P. hour at full load.

The following figures from various sources show how this estimate agrees with others and with some data from actual practice.

W. Heym, in the *Gasmotorentechnik*,* states that a producer plant requires per horse-power hour 6.5 gallons for the engine, 4 gallons for the scrubber, and .13 gallons for the producer vaporizer.

Freyn, in the paper already mentioned, makes the following estimates for a blast furnace gas plant of 10,500 B. H. P. Theissen washers are used to clean the gas.

GALLONS OF COOLING WATER REQUIRED

Load on Engines	For the Engines per B. H. P.-hour	For the Washers per 1000 cu. ft. of gas cleaned per minute
Full	8.5	12.0
$\frac{3}{4}$	10.5	14.0
$\frac{1}{2}$	13.0	16.0

Freyn also estimates that if the plant is located near a stream of water, the cost of pumping would probably be in the neighborhood of 2 c. per 1000 gallons.

J. R. Bibbins† on a 51-hour test of a 500 horse-power Westinghouse horizontal engine, reported a water consumption for the engines only of 9.4 gallons per B. H. P. hour at full load. The same authority reports a figure of 5.65 gallons per B. H. P. hour‡ found on similar engines in another plant.

In conclusion it should be said that the water-consumption of gas engines depends somewhat upon the size of the cylinder. Thus a single-acting cylinder of very large diameter usually

* November, 1907.

† J. R. Bibbins, Proceedings A. S. M. E., Mid-November, 1907.

‡ J. R. Bibbins, Gas Driven Electric Power Station, Proc. of the Eng. Soc. of W. Pa.

requires more cooling water than two single-acting or one double-acting cylinder of the same capacity. Skilful attendance also is a considerable factor in the amount of cooling water used, a point which is very often neglected.

(f) OIL AND WASTE. The consumption of lubricating oil per B. H. P. hour depends upon the size of the engine and, as in the case of cooling water, largely also upon the care of attendants. It is also true that new machines may for a time require two or three times the ordinary amount of oil until all the parts have become adjusted to their service. In large and medium sized plants it is usual to employ a gravity or circulating system of oiling, in which case the oil is recovered, filtered, and again used. In such systems the cost of oil is a small item and consists mostly of the cost of oil necessary to replace that unavoidably lost. Of course none of the cylinder oil used is recovered.

Güldner estimates that the consumption of lubricating oil usually varies from .006 to .008 pints of oil per B. H. P.-hour, and that under favorable conditions .003 pints per B. H. P.-hour may be reached. L. L. Brewer finds that the consumption for the large engines quoted in the table, page 552, varies from .0045 to .0055 pints per B. H. P.-hour, and that in double-acting two-cycle engines the consumption may be as low as .0035 pints.

The following table compiled by J. R. Bibbins and published in the paper before the Society of Engineers of West Pennsylvania, already quoted, shows the actual oil consumption in a plant containing two horizontal Westinghouse engines of 500 horse-power each, direct connected to 300 K. W. generators. The figures cover a period of four months and the results should therefore be very reliable. The system of oiling used is of the continuous circulating and filtering type. The consumption amounts to .00202 pints of cylinder oil and .00286 pints of engine oil per horse-power hour, which agrees well with the figures given by Brewer.

OIL CONSUMPTION

4 months ending Sunday, May 23, 1906

	CYLINDER OIL *	ENGINE OIL †
Kind	Imperial Cylinder Oil	Special Gas Engine Oil
Maker	Union Petroleum Co., Philadelphia	
Price (barrel lots)	32 cents per Gal.	18 cents per Gal.
Quantity used	561 Gals.	765 Gals.
Quantity used per month	140½ Gals.	191½ Gals.
Quantity used per (opertg.) day ...	4.68 Gals.	6.38 Gals.
Quantity used per full day	6.07 Gals.	8.27 Gals.
Engine Hours per day	37	37
Engine H.P. Hours per day	18500 *	18500
Oil per Engine Hour	0.127 Gals.	0.172 Gals.
Oil per H.P. Hour	0.000253 Gals.	0.000345 Gals.
Cost per Engine Hour	4.05 cents	3.1 cents.
Cost per H.P. Hour	0.00809	0.00621
Total Cost		0.0143 cents per H.P. Hour

The cost of cotton waste or other cleaning material is an item very hard to estimate, and in any case of little influence on the final result. It is usually combined with the cost of oil, and some figures for this combined cost will be found in the estimates of total operating costs at the end of this chapter.

(g) ATTENDANCE. The statement very often carelessly made with regard to small gas engine installations, that they require no waiting on, is of course not quite the truth. The fact is that a small gas or gasoline engine, if in good order, does not require, after starting, much attending except the proper handling of the lubricators. In this respect the gas power installation has an advantage over the small steam power plant, where at least one man is generally required all the time. For small natural gas or illuminating gas engines, therefore, it is possible to employ the "engineer," to use a familiar term, somewhere else at least part of the time, but naturally, as the size of engine increases, this spare time decreases, and it is doubtful if suction gas plants, no matter how small, can get along with less than one man's entire time.

The general methods of taking care of medium sized and large

*Includes crank case oil for exciter engines. Drainage from glands of main engines collected, mixed with old engine oil and used in crank case. No crank case oil purchased.

†Includes oil consumption of auxiliaries. Sufficient quantity old engine oil drawn from circulating system to supply auxiliaries replaced by fresh oil.

gas engines are perhaps not far different from those used in steam engine practice. It cannot be denied, however, that gas engines have much more opportunity, so to speak, to go wrong. That is, jacket water, lubrication, igniters, valves, etc., all need careful looking after, and any of these things, if neglected, may cause a shut down. While, therefore, steam engine attendants generally accustom themselves to the new service very quickly, it would be wrong to assume that they can in general take care of a gas engine plant without some instruction and practice. Of course the larger the plant, the more important this point becomes.

There seems to be nothing in English engineering literature to give an approximate idea of the time actually required in attending gas engines. Güldner proposes the following equations, apparently based on practical experience:

For illuminating gas, natural gas and oil engines,

$$W = .25 \sqrt{N_n} \text{ hours.}$$

For suction gas plants $W = 1.25 \sqrt{N_n}$ hours

where W = time of attendance required in hours per day of 10 hours and N_n = rated capacity of plant.

Thus, for instance, a 100 horse-power natural gas engine would actually require about $.25 \sqrt{100} = 2.5$ hours in a 10-hour shift. Hence the attendant's time could be largely used somewhere else. On the other hand, a 100 horse-power suction gas plant would actually require $1.25 \sqrt{100} = 12.5$ hours, which means that two men will have to be employed.

Where a station is made up of several smaller units and producers, the cost of attendance naturally increases. Thus if there are n units in a plant of N_n horse-power total, Güldner states that the actual time required per operating day of the plant may be expressed by

$$W = 1.25 \sqrt{n N_n} \text{ hours.}$$

Thus if we take a 1200 horse-power suction gas plant made up of 4 units of 300 horse-power each,

$$W = 1.25 \sqrt{4 \times 1200} = 86.5 \text{ hours,}$$

which would mean the service of 9 men in the 10-hour shift. If

this capacity had been put in one unit, the time would have been

$$W = 1.25 \sqrt{1200} = 43.5 \text{ hours,}$$

which would have called for only 5 men per shift.

If in any of the above equations we introduce the hourly wage scale, we may express the cost of attendance per B. H. P. per hour. Thus if the scale should be 20 c. per hour, we would have for illuminating gas, natural gas, or oil engines, for the 10-hour shift,

$$\text{Cost per B. H. P.-hour} = \frac{20 \times .25 \sqrt{N_n}}{10 N_n} = \frac{.50}{\sqrt{N_n}} \text{ cents.}$$

For suction gas plants,

$$\text{Cost per B. H. P.-hour} = \frac{20 \times 1.25 \sqrt{N_n}}{10 N_n} = \frac{2.5}{\sqrt{N_n}} \text{ cents}$$

and for large suction gas plants of n units,

$$\text{Cost per B. H. P.-hour} = \frac{20 \times 1.25 \sqrt{n N_n}}{10 N_n} = \frac{2.5 \sqrt{n}}{\sqrt{N_n}} \text{ cents.}$$

There is little data available to check the accuracy of these formulæ, but the two or three instances cited by various authorities for the labor cost in suction gas plants check very well with the results of the formula.

(h) MAINTENANCE AND REPAIRS. The expenditures for maintenance and repairs of a gas engine installation should not exceed 3 per cent of the purchase price of the engines and producers. There is no doubt that this item may easily run to 10 per cent and over, especially if in a producer plant insufficient attention is paid to cleaning the gas. But such conditions are not normal and should not occur in good practice.

Freyer states that the following are average figures for repair accounts for a blast furnace gas station:

2½ per cent per year of the price of the engines and electric generators.

7 per cent per year of the price of the cleaning plant.

5 per cent per year of the price of the air compressor used for starting.

2 per cent per year of the price of piping, etc.

To this may be added from 1 to 2 per cent of the cost of the building as maintenance for the building.

Total Operating Costs and Cost as Compared with other Prime Movers — As in the case of fuel costs, the information available in engineering literature on total operating costs is of two kinds. In the first kind the estimates are based entirely upon hypothetical assumptions, in the second the results are those obtained in actual operation. Again, the latter information is of course of much greater value, but except for large installations little of this class has been made public. Especially as regards small installations of gasoline, oil and illuminating gas engines there seems to be an almost absolute lack of data referring to total operating costs. For that reason, in order to get some idea of costs, it will be necessary for the time being to depend upon cost computations based on assumptions. In any concrete case these assumptions should be carefully scanned to see how they agree with actually existing conditions before forming a final idea of operating cost.

The data available from actual practice is mainly due to Mr. J. R. Bibbins* and others, of the Westinghouse Machine Company, and of course relates to Westinghouse engines. This, however, does not preclude their applicability to other similar plants. This information, combined with that from a few English plants, is about all that can be cited at the present writing.

Hypothetical computations are nearly all made on a comparative basis. Of this kind are the tables published by W. O. Webber in the *Engineering News*, August 15, 1907. It is not easy to give a concise abstract of the various factors assumed in the computation of these tables, and for that reason they are given in full, except for electric power which does not especially interest us here.

* The following is a partial list of the valuable papers published:

J. R. Bibbins, Gas Driven Electric Power Systems, Warren & Jamestown Street Railway, Eng. Soc. of W. Pa.; Gas Power for Central Stations, Am. Inst. of E. E., New York, December 18, 1903; Producer Gas Power Plant, Proc. A. S. M. E., December, 1906; Application of Gas Power to Central Station Work, National Elec. Light Assoc., Washington, D. C., June, 1907; Duty Test on Gas Power Plant, Proc. A. S. M. E., Mid-November, 1907; and A. West, Gas Power in Electric Railway Work, American Street & Interurban Railway Assoc., Philadelphia Convention, 1905.

E. E. Arnold, The High Power Internal Combustion Engine and its Fitting for Central Station Work, New England Street Railway Club, March, 1904.

Reprint from Power, December, 1903, A Gas Engine Pumping Station.

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COST OF GASOLINE POWER

Size of plant in H.P.	2	6	10	20
Price of engine in place	\$150.00	\$325.00	\$500.00	\$750.00
Gasoline per B. H. P. per hour..	$\frac{1}{2}$ gal.	$\frac{1}{4}$ gal.	$\frac{1}{2}$ gal.	$\frac{1}{2}$ gal.
Cost per gallon	\$0.22	\$0.20	\$0.19	\$0.18
= cost per 3,080 hours	\$451.53	\$924.00	\$975.13	\$1,386.00
Attendance at \$1 per day	308.00	308.00	308.00	308.00
Interest, 5 per cent.	7.50	16.25	25.00	37.50
Depreciation, 5 per cent	7.50	16.25	25.00	37.50
Repairs, 10 per cent	15.00	32.50	50.00	75.00
Supplies, 20 per cent	30.00	65.00	100.00	150.00
Insurance, 2 per cent	3.00	6.50	10.00	15.00
Taxes, 1 per cent	1.50	3.25	5.00	7.50
Power cost	\$824.03	\$1,371.75	\$1,498.13	\$2,016.50

To these figures should be added charges on space occupied as follows:

Value of space occupied	\$100.00	\$150.00	\$200.00	\$300.00
Interest, 5 per cent.	\$5.00	\$7.50	\$10.00	\$15.00
Repairs, 2 per cent	2.00	3.00	4.00	6.00
Insurance, 1 per cent	1.00	1.50	2.00	3.00
Taxes, 1 per cent	1.00	1.50	2.00	3.00
Total annual charge for space	\$9.00	\$13.50	\$18.00	\$27.00
Total cost per annum.	\$833.03	\$1,385.25	\$1,516.13	\$2,043.30
Cost of 1 H.P. per annum, 10 hour basis.	416.51	239.87	151.61	102.17
Cost of 1 H.P. per hour	\$0.1352	\$0.0780	\$0.0492	\$0.0331

COST OF GAS POWER

\$1.50 per 1000 cubic feet of gas less 20 per cent, if paid in 10 days = \$1.20
net, gas 760 B. T. U.

Size of plant in H.P.	2	6	10	20
Engine cost in place	\$200.00	\$375.00	\$550.00	\$1,050.00
Gas per H.P.-hour in cubic feet.	30	25	22	20
Value of gas consumed, 3080 hours	\$221.76	\$554.40	\$843.12	\$1,478.00
Attendance, \$1 per day	308.00	308.00	308.00	308.00
Interest, 5 per cent.	10.00	18.75	27.50	52.50
Depreciation, 5 per cent.	10.00	18.75	27.50	52.50
Repairs, 10 per cent	20.00	37.50	55.00	105.00
Supplies, 20 per cent	40.00	75.00	110.00	210.00
Insurance, 2 per cent	4.00	7.50	11.00	21.00
Taxes, 1 per cent	2.00	3.75	5.50	10.50
Power cost	\$615.76	\$1,023.65	\$1,387.62	\$2,237.50
Annual charge for space	9.00	13.50	18.00	27.00
Total cost per annum.	\$624.76	\$1,037.15	\$1,405.62	\$2,264.50
Cost of 1 H.P. per annum, 10 hour basis.	312.38	172.86	110.56	143.22
Cost of 1 H.P. per hour	\$0.1014	\$0.0561	\$0.0456	\$0.0367

COST OF STEAM POWER

	6	10	20
Size of plant in H.P.	6	10	20
Cost of plant per H.P.	\$250.00	\$220.00	\$200.00
Fixed charge, 14 per cent	\$35.00	\$30.80	\$28.00
Coal per H.P.-hour, in pounds	20	15	12
Cost of coal at \$5 per ton	\$154.00	\$103.00	\$82.50
Attendance, 3080 hours	75.00	50.00	30.00
Oil, waste and supplies	15.00	10.00	6.00
Cost 1 H.P. per annum, 10-hour basis	\$279.00	\$194.80	\$146.50
Cost of 1 H.P. per hour	\$0.0906	\$0.0832	\$0.0475

ANNUAL COST OF POWER PER BRAKE HORSE-POWER *

B. H. P. of Unit	Steam	Gas	Gasoline
1	\$600.00	\$380.00	\$487.50
2	500.00	312.50	416.00
3	437.50	260.00	350.00
4	375.00	220.00	300.00
5	320.00	192.50	262.50
6	280.00	172.50	240.00
7	250.00	160.00	210.00
8	230.00	152.50	182.50
9	210.00	145.00	165.00
10	195.00	140.00	152.00
12	175.00	132.50	137.50
14	165.00	126.00	122.00
16	157.50	120.00	112.50
18	150.00	116.50	107.50
20	146.00	113.00	102.00
22	140.00	110.00	98.00
24	137.50	107.50	95.00
26	133.00	105.00	92.50
28	130.00	102.50	90.00
30	127.50	102.00	87.50
35	124.00	100.00	85.00
40	120.00	98.00	82.50
50	112.50	96.00	80.00
60	105.00	94.00	78.00
70	100.00	92.00	76.00
80	95.00	90.00	74.00
90	90.50	88.00	72.00
100	86.40	86.00	70.00

Attention is called in these tables to the high allowance for repairs in the case of the gasoline and gas engine and the high coal consumption assumed for the steam engine.

Another interesting comparison is that made by H. A. Clark

* Unit costs: Coal, \$5 per ton; gas \$1.20 per 1000 cubic feet, at 760 B. T. U.; gasoline, \$0.20 per gallon.

in a paper on the Diesel Engine.* In this case the computations are carried out for three sizes of Diesel engine, Crossley Gas Engine and Dowson producer, and high-speed compound condensing steam engine. The conditions assumed are partly as follows, the remainder being given in the table itself.

Cost of fuel delivered:

Diesel engine, crude oil at \$10.90 per ton = appr. 3.8 c. per gallon.

Crossley engine, anthracite at \$5.70 per ton.

Steam engine, coal at \$3 per ton.

The fuel cost is based on the following assumption:

For the Diesel, the consumption has been taken in each case as the mean between full and half-load rates. These rates were actually determined by tests.

For the Crossley, 1.5 to 1.25 pounds of coal per B. H. P. hour plus an allowance for stand-by losses.

For the steam engine, 4 to 3.5 pounds of coal per B. H. P. hour for the lowest and highest powers, assuming an evaporation of 8 pounds of water per pound of coal.

The cost figures in the original table have all been transposed to dollars and cents. The computations are based on a year of 2700 working hours.

In the discussion on Clark's paper, the cost computations were rather severely criticised, mainly on account of the fuel costs assumed. The claim was made that both the coal consumption for the steam engine and the cost of the steam coal was assumed too high, and that the cost of oil at 3.8 c. per gallon could only apply to seaboard towns. As far as American conditions are concerned, the assumption regarding the steam engine would seem to be about right, while the cost of anthracite at \$5.70 per ton for the gas engine is certainly not too low. On the other hand, the price of 3.8 c. per gallon for the crude oil would seem to favor the Diesel engine. It all comes to the point that applies to all computations of this kind, and that is that the results are strictly applicable only to the locality for which they are computed. Allowing for possible difference in the labor costs, however, it seems to the writer that the comparison is quite fair as between the steam and the gas engine.

* Proc. of M. E., 1903, II.

TOTAL COSTS IN POWER PRODUCTION

	Diesel Engine 35	Gas Engine 35	Steam Engine 35	Diesel Engine 80	Gas Engine 80	Steam Engine 80	Diesel Engine 160	Gas Engine 160	Steam Engine 160
1. Means of production	12x20x15	12x20x15	12x20x15	16x24x20	16x24x20	14x24x20	20x27x20	20x27x20	17x27x20
2. Brake horse-power	—	10x20	—	—	11x24	—	12x26	—	—
3. Size of engine house, ft.	—	8x16x12	—	—	10x18x15	15x46x20	10x20x16	—	20x48x20
4. Ground for gas holder, ft.	—	\$ 107.00	\$ 146.00	\$ 73.00	\$ 141.00	\$ 194.00	\$ 199.00	\$ 272.00	\$ 272.00
5. Size of boiler house and gas producer, ft.	—	631.00	1115.00	681.00	925.00	1750.00	1140.00	2040.00	2040.00
6. Cost of land at \$1.70 per square yard	—	73.00	291.00	98.00	136.00	389.00	170.00	487.00	487.00
7. Cost of buildings	49.00	—	—	—	—	—	—	—	—
8. Engine foundations, etc., boiler settings	437.00	—	—	—	—	—	—	—	—
9. Engines, tanks, piping and condenser, etc.	49.00	—	—	—	—	—	—	—	—
10. Boiler or gas producer and accessories	\$2890.00	1360.00	1165.00	4370.00	3080.00	2430.00	8170.00	5550.00	3890.00
11. Total capital cost	—	974.00	1408.00	—	1260.00	1700.00	—	1845.00	2910.00
12. Capital cost per B. H. P.	3425.00	3145.00	4125.00	5222.00	5544.00	6463.00	9319.00	8928.00	9589.00
ANNUAL CHARGES	97.90	90.00	117.70	65.25	69.30	80.80	58.20	55.80	59.90
13. Interest on capital, 4 per cent	\$ 135.20	\$ 126.50	\$ 165.50	\$ 209.00	\$ 222.00	\$ 259.00	\$ 373.00	\$ 357.00	\$ 384.00
14. Building, maintenance and depreciation, 5 per cent	21.90	31.60	55.90	34.10	43.80	87.50	46.20	57.20	104.00
15. Engines and machinery, maintenance and depreciation, 10 per cent	284.50	136.20	116.80	438.00	309.00	243.00	817.00	554.00	389.00
16. Boilers, Maintenance and depreciation, 12 per cent	—	—	170.50	—	—	204.00	—	—	350.00
17. Gas producer and holder. Depreciation, 5 per cent	—	48.70	—	—	63.20	—	—	—	—
18. Fuel	231.00	438.00	632.00	477.00	1000.00	1265.00	904.00	1670.00	2140.00
19. Lubricant, waste, etc.	97.40	97.40	97.40	170.50	170.50	170.50	268.00	268.00	268.00
20. Wages, removal of ash, etc.	365.00	438.00	642.00	365.00	462.00	535.00	426.00	535.00	680.00
21. Total	1133.00	1316.40	1700.00	1693.60	270.50	2764.00	2834.20	3533.50	4315.00
22. Annual cost per B. H. P.	32.40	37.60	48.70	21.19	28.38	34.55	71.72	22.09	26.30
23. Cost per B. H. P. hour, cents	1.08	1.38	1.78	.78	1.04	1.26	.64	.80	.98

As a final example of this kind of hypothetical computation, the following table is given. The figures are due to Mr. L. G. Findlay of the De La Vergne Machine Company of New York. They were presented at the Dayton meeting of the Ohio Society of Mechanical, Electrical, and Steam Engineers, and were published in *Power*, July, 1907. The table is very complete and would repay careful study. Especially interesting is the effect of the load factor on operating costs. This factor is here defined as the average load on the plant divided by the rated capacity of the plant.

In Clark's tables above no mention is made of the load factor and it is evidently assumed to be 100 per cent. That, however, is a condition existing and maintained in very few plants. In fact, if it were, it would be poor engineering, as it gives the plant but very little overload capacity. The marked effect that a decrease of the load factor from 100 per cent to 50 per cent has upon operating costs is very clearly brought out in the table. In the tables given below for actual cost data, the load factor is in most cases even lower than this.

In the discussion on the results of the table following its presentation, the steam men objected to the comparison between the steam and gas plants at the rated load, on account of the much greater overload capacity of the steam engine plant. Lines 36, 37, 38, and 39 of the table are computed on the assumption that 80 per cent of the steam used by the engines is available for heating, but no standing charges are made against the heating plant.

Turning next to data from actual practice, the first of the following tables was given by Mr. J. E. Dowson in a paper in the *Journal of the Institute of Electrical Engineers* April, 1904. The table refers to results obtained in two English plants; the first is equipped with Dowson generators and Crossley engines, the second with the same type of producer and Westinghouse vertical engines of comparatively small size. In spite of the higher cost of coal, the greater capacity of the Walthamstow plants brings down the wage and repair items enough to make the total works costs not far different. The table only gives works cost, and it should be remembered that the results apply to English conditions.

COST OF 200 HORSE-POWER AND 1000 HORSE-POWER STEAM AND GAS PLANTS

Assuming the 1000 horse-power engine is compound condensing and the 200 horse-power is a simple automatic cut-off engine; that the coal with steam plants and bituminous coal producer costs \$2 per short ton, while anthracite is \$4.50 per ton. Bituminous coal averaging 11,000 B. T. U. per pound and the anthracite 13,000 B. T. U., lower heat values.

Load Factor	Steam						Producer Gas					
	1000 H. P.			200 H. P.			Bituminous			Anthracite		
	Full	50 per cent		Full	50 per cent		Full	50 per cent		Full	50 per cent	
1 Cost of engine per rated horse-power		18.00			14.00			40.00			44.00	
2 Cost of piping per rated horse-power		6.00			5.00			3.50			3.00	
3 Cost of condensers per rated horse-power		3.00			..			..			..	
4 Cost of pumps, etc., per rated horse-power		0.50			..			..			..	
5 Cost of air starting app. per rated horse-power								1.00			1.50	
6 Total of engine plant		27.50			19.00			44.50			48.50	
7 Depreciation, 4 per cent on total		1.10			0.76			1.78			1.94	
8 Repairs, 2 per cent on total		0.55			0.38			0.89			0.97	
9 Interest, 5 per cent on total		1.37			0.95			2.22			2.42	
10 Total of lines 7, 8, 9		3.02			2.09			4.89			5.33	
11 Boilers per rated horse-power		10.00			12.00			..			..	
12 Producers per rated horse-power								15.00			18.50	
13 Chimneys and breeching per rated horse-power		7.00			3.50			..			..	
14 Heaters, feed-pumps per rated horse-power		1.50			1.50			..			..	
15 Auto. stokers or coal machinery per rated horse-power		5.00			..			2.00			..	
16 Total of fuel plant		23.50			17.00			17.00			18.50	
17 Depreciation, 8 per cent on total cost		1.88			1.36			1.36			1.48	
18 Repairs, 3 per cent on total cost		0.70			0.51			0.51			0.55	

19	Interest, 5 per cent. on total cost	1.17	0.85	0.85	0.92	0.60	0.70
20	Total of lines 17, 18, 19	3.75	2.72	2.72	2.95	1.92	2.24
21	Coal used in B. H. P.-hr. in lb.	2.00	2.20	4.75	1.10	0.90	0.90
22	Cost of coal per B. H. P. per day of 24 hours	4.80	5.28	11.40	2.64	4.86	4.86
23	Attendance of engine B. H. P. per day of 24 hours	1.20	2.40	3.00	6.00	1.05	2.10
24	Attendance of fuel plant per day of 24 hours	0.65	1.30	1.75	3.50	1.50	3.75
25	Oil, waste and supplies	0.50	0.80	0.50	0.80	0.50	0.70
26	Total daily expense per B. H. P. per 24 hour day	7.15	9.78	16.65	26.52	4.94	7.90
27	Yearly oper. expenses 309 days per B. H. P., dollars	22.09	30.22	51.45	81.95	15.26	24.41
28	Cost of coal per B. H. P. per day of 10½ hours	2.05	2.25	4.86	6.46	1.13	1.54
29	Attendance of engine B. H. P. per day of 10½ hours	0.70	1.40	1.50	3.00	0.40	0.80
30	Attendance of fuel plant per day of 10½ hours	0.28	0.56	0.88	1.75	0.45	0.90
31	Oil, waste and supplies per day of 10½ hours	0.27	0.49	0.26	0.49	0.27	0.46
32	Total daily expenses per day of 10½ hours	3.30	4.70	7.50	11.70	2.25	3.70
33	Yearly oper. expenses 309 days per B. H. P., dollars	10.20	14.52	23.17	36.15	6.95	11.43
34	Yearly expenses lines 10, 20, 27 = 24-hr. day	28.86	43.76	56.26	91.57	22.87	39.63
35	Yearly expenses lines 10, 20, 33 = 10½-hr. day	16.97	28.06	27.98	45.77	14.56	26.55
36	Cost of power, all exhaust steam used for heating	24 -hr. day.	37.61	61.25			
37	Cost of power, same as line 36 but short run	10½-hr. day.	19.07	30.68			
38	Cost of power, 50 per cent exhaust steam for heating	24 -hr. day.	42.41	69.56			
39	Cost of power, same as line 38 but short run	10½-hr. day.	21.11	34.90			
	Total cost, lines 6 + 16 for entire plant	\$51,000.00	\$7,200.00	\$61,500.00	\$13,400.00	\$56,500.00	\$12,500.00
	Water evaporated per lb. coal, at 212°	10.00	8.00				

The second and third tables following are due to Mr. J. R. Bibbins of the Westinghouse Machine Company and were published in a paper on the Application of Gas Power to Central Station Work, read before the National Electric Light Association at Washington, June, 1907. Both of the stations mentioned use natural gas for fuel, but the Bradford plant is equipped with rather small Westinghouse vertical engines, while the Warren and Jamestown station contains two large direct connected units of 500 horse-power each. In both cases the load factor is considerably less than 50 per cent; in the case of the Bradford plant it is only about 19 per cent. The figures for the latter plant are remarkable for the length of time covered, 8½ years.

WORKS COST OF OPERATING GAS PLANTS

	Plant	
	Midland Railway Co., Leicester	Urban District Station, Walthamstow
Engine:		
Capacity tested	300	1500
Number	6	7
Make	Crossley	Westinghouse
Length of run	5 months	12 months
Load factor	—	15.25
K. W.-hour generated	200497	659756
Remarks — service	Arc and incandescent lighting, Dowson gas.	Arc and incandescent lighting, Dowson gas.
Duty of plant: *		
Lbs. per K. W.-hour	3	2
Lbs. per B. H. P.	2.25	1.5
Cost:		
Coal per ton	\$3.75	\$6.50
Fuel per K. W.-hour	.123c.	.17c.
Oil, waste, water	.020c.	.095c.
Wages	.292c.	.015c.
Total works cost power, cents per K. W.-hour ..	.473c.	.445c.
Authority:	R. M. DEELY, Supt. Locomotives.	F. A. WILKINSON, Elec. Engineer in charge.

WORKS COST OF POWER — GAS DRIVEN CENTRAL STATION

BRADFORD ELECTRIC LIGHT & POWER CO., BRADFORD, PA.

Station capacity (five engines)	470 kilowatts
Number of years in operation	8.5
Average station load factor ('03-'06)	19.09 per cent
Average gas consumption per kilowatt-hour ('03-'06)	25.5 cu. ft.
Average heat efficiency at switchboard	14.14 per cent

* Including "stand-by" losses.

Works Cost of Power	Dollars per year. Average 8 years	Cents per Kilowatt-hour. Average 4 years
Fuel, gas, including heating	\$3,121	0.307
Station wages	3,068	0.410
Oil, waste and supplies	454	0.054
Running repairs (total)	848	0.085
Running repairs gas engines	285	0.034
Total operating cost	\$7,776	0.890

Engine repairs — \$0.36 per horse-power per year.

57.00 per engine per year.

.75 per cent on investment.

Fuel — Bradford natural gas.

WORKS COSTS — GAS POWER RAILWAY PLANT

WARREN & JAMESTOWN STREET RAILWAY COMPANY

Capacity of plant (two units)	600 kilowatts
Average time operated per day	18.5 hours
Average output per day	4115 kilowatt hours
Average load, per cent rating	37 per cent

Works Cost of Power	Dollars per day	Cents per Kilowatt-hour
Average First Four Months, 1906:		
Fuel gas	\$12.97	0.315
Wages	12.37	0.300
Oil	2.63	0.064
Repairs and miscellaneous supplies ..	3.29	0.080
Total	\$31.26	0.759

Fuel — "Bradford Sand" natural gas.

In conclusion, to make the figures in the above tables readily comparable among themselves, the cost data has been recomputed to the basis of cost in cents per B. H. P. hour in all cases.

Table I gives the total operating costs for small and medium sized plants as computed from the data of Webber & Clark. The load factor is in all cases 100 per cent. The difference in the cost of steam power arrived at by the two authorities is remarkable, Clark's figures being less than 50 per cent those of Webber. The reason for this is found in two directions, different assumptions as to cost of coal, \$5 per ton as against \$3, and different assumptions as to the coal consumption per horse-power. Webber

assumes as high as 20 pounds of coal per horse-power for a 6 horse-power plant, and charges a 20 horse-power plant up with 12 pounds per horse-power-hour. In fact Webber's figures are very liberal throughout for all three kinds of power.

In Table II, the works and total operating costs per B. H. P.-hour are shown side by side as computed from Findlay's table.

TABLE I
Total Operating Costs per B. H. P.-hour
Small and Medium Sized Plants

Kind of Power	Kind of Fuel	Load Factor	Working hours per year	Size of Plant												Authority
				5	10	16	20	24	30	35	50	80	100	160		
				Cost in cents per B. H. P.-hour												
Steam Engine.	Coal	100	3080	13.9	6.3	5.1	4.7	4.5	4.1	4.0	3.7	3.1	2.8	Webber		
Gas Engine ..	Ill. Gas	100	3080	6.2	4.6	3.9	3.6	3.5	3.3	3.2	3.1	2.9	2.8	Webber		
Gas Engine ..	Gasoline	100	3080	8.5	4.9	3.7	3.3	3.1	2.8	2.7	2.6	2.4	2.3	Webber		
Steam Engine.	Coal	100	2700							1.78		1.26	.98	Clark		
Gas Engine ..	Prod. Gas	100	2700							1.38		1.04	.80	Clark		
Oil Engine, Diesel	Crude Oil	100	2700							1.08		.78	.64	Clark		

TABLE II
Total Operating and Works Cost per B. H. P.-hour.
Computed from Table by Mr. L. G. Findlay.
Works costs in brackets.

Hours in Operation per day	Load Factor	Size of Plant B. H. P.	Kind of Power		
			Steam	Bituminous Producer Gas	Anthracite Producer Gas
			Cents per B. H. P.-hour		
24	100	1000	.39 (.30)	.31 (.21)	.39 (.30)
24	50	1000	.69 (.41)	.54 (.33)	.67 (.49)
10.25	100	1000	.53 (.33)	.46 (.22)	.55 (.33)
10.25	50	1000	.88 (.47)	.84 (.36)	.99 (.56)
24	100	200	.76 (.69)	.40 (.29)	.43 (.38)
24	50	200	1.23 (1.10)	.71 (.49)	.86 (.66)
10.25	100	200	.88 (.73)	.59 (.33)	.69 (.45)
10.25	50	200	1.44 (1.15)	1.11 (.59)	1.26 (.78)

The data from the Bradford and Jamestown plants and from the two English plants shows the following results for the works cost per B. H. P.-hour.

Fuel	Load Factor	B. H. P. rated	Works Cost per B. H. P.-hour cents	Plant
Natural gas	19.09	640	.67	Bradford
Natural gas	37.0	800	.57	Jamestown
Dowson gas	—	400	.35	Midland Railway
Dowson gas	15.25	2000	.33	Walthamstow

From these figures, and those of Table II, the conclusion seems justified that it should be easily possible to produce one brake horse-power in a fair sized plant, say not under 200 horse-power, for a works cost of from .5 to .75 cents per hour, depending upon the conditions involved. This excludes fixed charges and assumes a load factor in the neighborhood of 50 per cent.

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