A Short History of Large Gas Engines

Bryan Lawton
Newcomen Society

In the nineteenth century large volumes of low-energy gas from blast furnaces were wasted, despite improvements such as Neilson's hot blast process, and this prompted B.H. Thwaite, in 1894, to experiment with it as a source of fuel for gas engines. His experiments were successful and gas engines, despite the low energy content of blast furnace gas, produced almost the same power as with coal gas. However, the available engines were too small to make use of the huge quantities being wasted and consequently new and much larger engines were developed very quickly, particularly in Belgium and Germany. Bore diameter increased from about 10 inches to over 60 inches in about twenty years and water-cooled, double-acting, tandem, horizontal engines were adopted. Britain and the United States were content, initially, to manufacture such large engines under licence. Still larger engines would have been made but for high piston temperatures and high thermal stresses causing failures and placing an upper limit on the bore diameter and the power per unit area of piston. British manufactures solved the problem by using large numbers of smaller cylinders, arranged vertically in tandem. Modern gas engines use up to twenty-four smaller cylinders, usually arranged in vee rather than tandem formation.

KEYWORDS: Thwaite, Gas Engines, Waste Heat, Blast Furnaces, Thermal and Mechanical Stress, Performance Limits

Introduction

In 1894, at the Glasgow Iron and Steel Works, B.H. Thwaite discovered that waste gas from blast furnaces could be cleaned and burnt very successfully in gas engines, despite its low energy content.[1] Existing gas engines were too small to make effective use of the vast quantities of such gas and consequently, over the next twenty years, gas engines increased in power from about 100 bhp to over 4,000 bhp, and in cylinder diameter from about 10 inches to 60 inches. Once developed, large gas engines found other applications, but the rapid increase in their cylinder diameter ended as suddenly as it began. This was because they had reached the thermal and mechanical limits of what could be achieved with the available materials. In Britain the manufacture and use of large gas engines was intermittent but on the Continent and in the United States they remained important and successful. The power output of gas engines has continued to increase since the

1920s but their maximum physical size, characterised by the cylinder diameter, has remained constant or decreased. Up to twenty-four smaller cylinders are now employed.

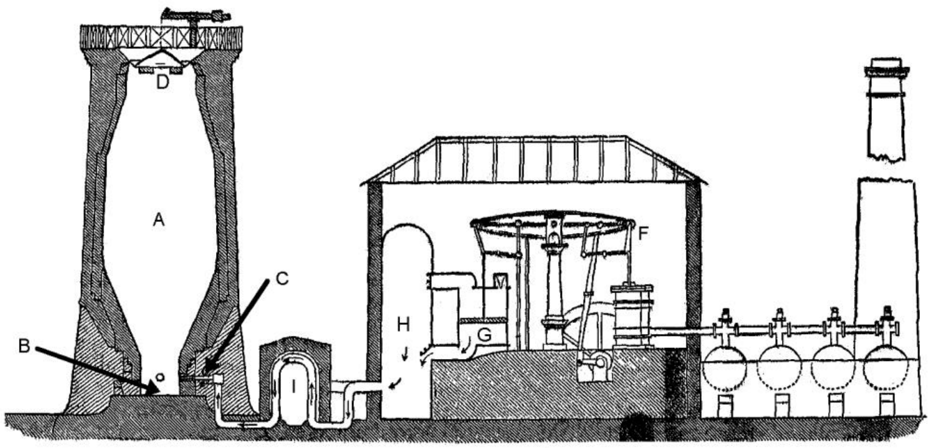
This paper concentrates on the British, Continental and American experience between about 1890 and 1920, when cylinder diameter grew to its largest, and it explains and quantifies the thermal, and mechanical constraints that prevented the development of larger gas engines; constraints that still restrict the development of gas and diesel engines.

Blast Furnace and Coke Oven Gas

In the eighteenth and nineteenth centuries quite enormous quantities of combustible gas were wasted in the exhaust from the manufacture of coke for use in blast furnaces and the combustion of the same coke in the manufacture of pig iron. Initially such waste gases were burnt off at the mouth of the furnace and resulted in some very dramatic night scenes in places like Coalbrookdale, Shropshire, where the night sky was lit up by the red glow issuing from the mouths of a multitude of furnaces. To some it seemed like a vision of hell but others responded to the drama. For example, Arthur Young, in 1783, thought the site was almost sublime and was blind to the pollution it was causing, whereas when Thomas Carlisle visited forty-one years later (1824), he saw a half frightful scene with its pestilential smoke, sooty men, and blackened grain.[2] Such scenes, initially impressive and inspirational, had become repulsive, being both wasteful of fuel and a major source of pollution.

Neilsen, in 1828 at the Clyde Ironworks, was the first to reduce the amount of coke used in blast furnaces. He did this by pre-heating the cold blast by passing it through an oven, heated by solid fuel, before supplying it to the blast furnace. He found that 100 lb of coal burnt in pre-heating the blast was able to save 300-400 lb of fuel burnt in the furnace.[3] The reason for this was that coke in the blast furnace was only partially burned to carbon monoxide whereas in the blast oven the coal was burnt to completion. In 1833 Faber du Faur invented a hot-blast oven consisting of sixteen cast iron pipes united by semi-circular bends, enclosed in a chamber and heated by the waste gases from the blast furnace. Many other designs followed but the main type was based on the principle of intermittent adsorption of heat by masses of firebrick and the subsequent transference of that heat to the blast. Cowper, in 1860, introduced this type.

In Figure 1 a cone *D* was fitted to the top of the blast furnace through which the furnace could be loaded with coke, ore and limestone. As was normal, the air supply (blast) to burn the coke and smelt the ore was supplied under slight pressure through a series of holes C at the base of the furnace (tuyères). The arrangements for supplying this blast were quite complex. A steam engine *F*, supplied from a boiler, drove a blower *G*, and the compressed air was collected in a receiver *H* having a large volume. The purpose of the receiver was to reduce the



Hot-blast Furnace.

Figure 1. Nineteenth century hot blast furnace.
A body of furnace, *B* hearth, *C* tuyères for hot air entry, *D* belt and cone for closing furnace, *F* blowing engine, *G* blowing cylinder, *H* receiver, *I* oven for heating blast-pipes.

magnitude of oscillations in pressure and flow rate from the blower and thus deliver a constant flow rate and pressure to the tuyères. Between the receiver and the tuyères the air passed through an oven *I* which heated the air to 300-550ºC. Various oven designs were used as indicated above. The hot blast supplied to the furnace required less heat than cold air to reach the combustion temperature and thus it reduced the consumption of coke.

But Neilson's hot blast did not use all the exhaust gases from the furnace and much useful energy still went to waste. From about 1850 onwards this was used under steam boilers to raise steam to drive the blowers using a steam engine, as is shown in Figure 1. Before this the blowers were driven by waterpower or by steam raised by burning coal.

The composition of blast furnace gas varies greatly depending on the fuel that is used (charcoal, coke, or coal of different grades) and on the ores that are smelted. Typical compositions are shown in Table 1. Carbon monoxide is the most useful gas and this varies from 20% to 35%. The other constituents are either inert or insignificant. The calorific value was typically in the range 80-120 Btu/ft^3, and with such a low energy (coal gas is typically 520 Btu/ft^3) it was not easy to operate steam boilers, but nevertheless it was done. Another problem was that the gas was very dusty and dirty and contained much tar and ammonia, which had to be removed.

Donkin says blast furnaces using the hot blast yielded 162,000 ft^3 of gas per ton of pig iron and as the world's annual production of pig iron in 1899 was

Table 1 Typical composition of blast furnace gas.

Fuel	N_2	CO	CO_2	CH_4	H_2	C_2H_4
Charcoal	60-63	20-30	6-20	0-1	0-0.4	-
Coke	64	35	1	-	0.1	-
Coal	56	22	15	4	1	2

over 40 million tons it is clear that 6.5 million million ft^3 of gas was produced in blast furnaces annually, even assuming they all operated using a hot blast.[4] However, by 1901 only a portion of this was wasted because, during the third quarter of the nineteenth century, engineers found ways to use the gas without letting it go to waste. Donkin, quoting data from F.W. Lürmann, Table 2, says that of the 162,000 ft^3 of gas produced per ton of iron, 10% was lost by leakage, 28% was used in heating the air blast, 39% was used under boilers to generate steam, and 23% was not utilised. Before the gas could be used under steam boilers it had to be cleaned and purified. This was the position in the best plant then available, but many furnaces still operated with open mouths and wasted all the furnace gas. The unused gas, 37,000 ft^3/ton, would produce if burned in a gas engine, some 280 hp hours per ton. A plant producing 600 tons per day of iron could generate 7,000 hp. If all the gases were burned in gas engines (101,000 ft^3/ton) then the same plant could generate 19,000 hp. Such large engines did not exist before about 1900.

Table 2 Usage of Blast Furnace Gas per ton or iron.

Blast Furnace Gas per ton of iron (ft^3)	Used to Heat Air Blast (ft^3)	Used to Raise Steam for Air Blower (ft^3)	Unused (ft^3)	Lost by Leakage (ft^3)
162,000 (100%)	45,000 (28%)	64,000 (39%)	37,000 (23%)	16,000 (10%)

Development of Gas Engines Using Blast Furnace Gas

Gas engine manufacture was stimulated in Britain in 1890 by the expiration of Otto's patent for the four-stroke cycle. Crossley Bros., of Openshaw, Manchester, had the rights to the invention outside Germany, and by 1888 claimed a production of 27,000 engines;[5] in 1885 Dr Otto claimed only that 5,000 had been produced in Germany and 10,000 elsewhere.[6] But after the expiration of Otto's patent in Britain the number of manufacturers increased, and almost all offered four-stroke engines, rather than Clerk's two-stroke or Linford's six-stroke versions, and with the increased competition the price fell by more than 50%. Figure 2 shows the purchase price of Crossley's Otto and Langen gas engine in 1877, the price of Clerk's two-stroke engine in 1882, and the vastly reduced price of Campbell's four-stroke gas engine in 1893.[7] The price increased roughly as the square root of power, consequently doubling an engine's power increased its selling price by only 40-50%. The newly introduced oil engines, it should be noticed, were considerably more expensive than the same size of gas engine, not because they were more expensive to make, but because the manufacturers needed to recover some of their investment in the new engine.

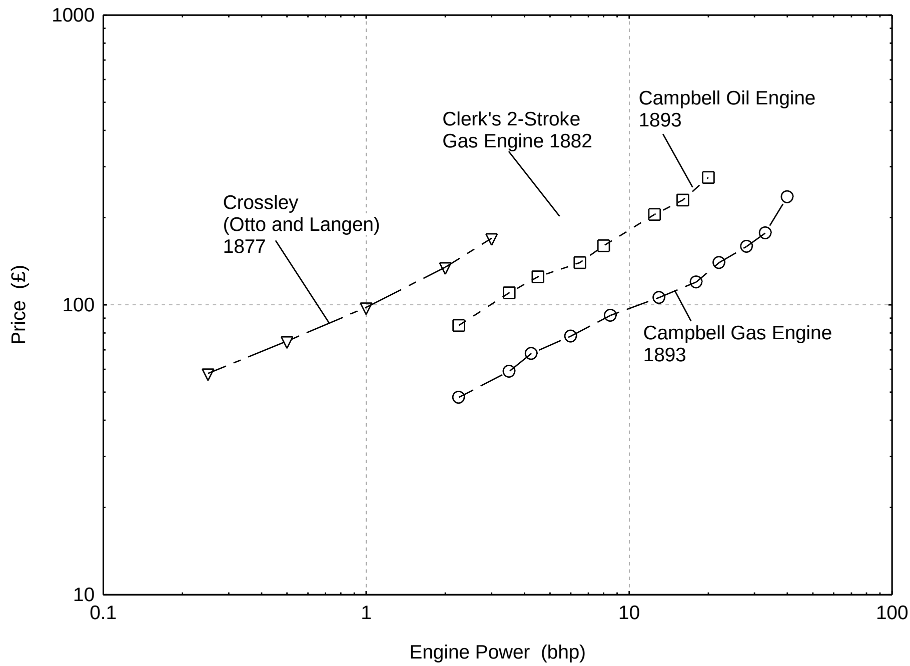


Figure 2. Reduction in the price of gas engines 1877-1893.

Initially gas engines developed less than 10 hp, used coal gas, and were ideal for the small manufacturer, being much simpler and cheaper to run than steam engines. Between 1890 and 1910 they developed quickly, Figure 3, and power increased from about 100 hp to 2,000 hp. For a given mean effective pressure and mean piston speed the power increases as the square of cylinder diameter,

Appendix A Equ. 16, so the easiest way to larger power was by increasing the cylinder diameter, from about 10 inches to about 60 inches. After this period of rapid growth power and cylinder diameter grew more modestly. This was because thermal stress is proportional to cylinder diameter and 60 inches was the limit that cast iron cylinders, pistons, and cylinder covers could withstand.

Before 1889 gas engines were less than 100 hp, and their low capital cost and low running cost made them much more attractive than reciprocating steam engines, which they quickly displaced from the market. They became larger. In

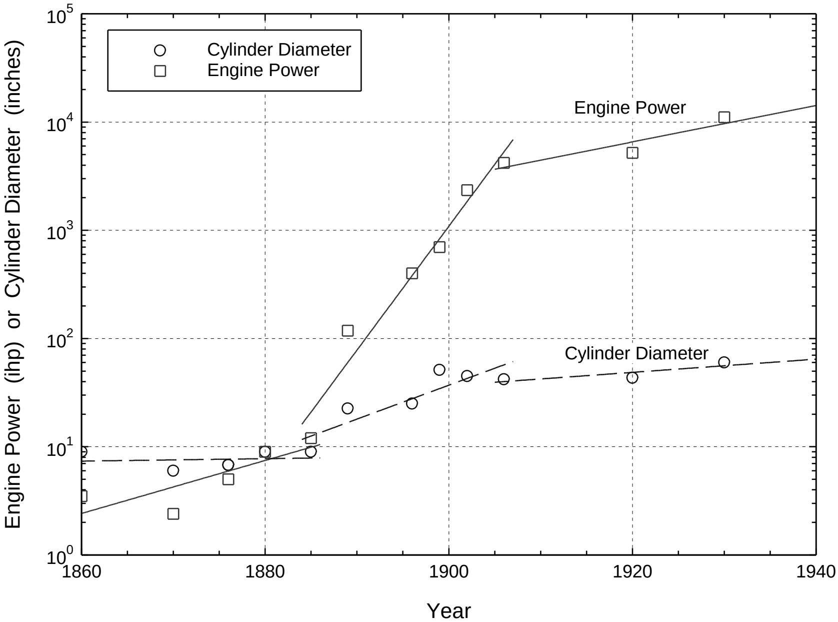


Figure 3. Growth in power and cylinder diameter of the largest gas engines, 1860-1940.

1889 a larger version of Delamare-Deboutteville's successful Simplex engine was made available.[8] The bore was 22.6 inches, the stroke was 37.4 inches, and it developed 100 hp. The speed was not specified but it is likely to be about 100 rev/min. Although of relatively low power (about 0.29 ihp/in^2 of piston area) the engine was of a very considerable size and weight, as may be seen in Figure 4.

In 1896 Messrs J.E.H. Andrews & Co., of Reddish, built the Stockport engine, the largest gas engine in the world at that time (Figure 5). The cylinder bore was 25 inches, the stroke 36 inches, and it developed 400 ihp. The two cylinders were arranged in a rather complicated way in tandem fashion on a bedplate. To avoid passing the piston rod of the outer piston through the combustion space of the inner piston, where it would need to be cooled, the pistons were connected by a system of piston rods, crossheads and side rods on either side of the inner piston. Both pistons were thus connected to a common crankshaft by a common connecting rod. A two-throw crankshaft would have been simpler. Two cylinders on one crank throw were known as tandem construction, and when fitted with double-acting, water-cooled pistons it became the standard layout for Continental and American engines. Each cylinder operated on the Otto cycle but the valve timing was arranged so the cylinders fired on alternate revolutions to give one impulse per revolution of the crankshaft. Clerk says

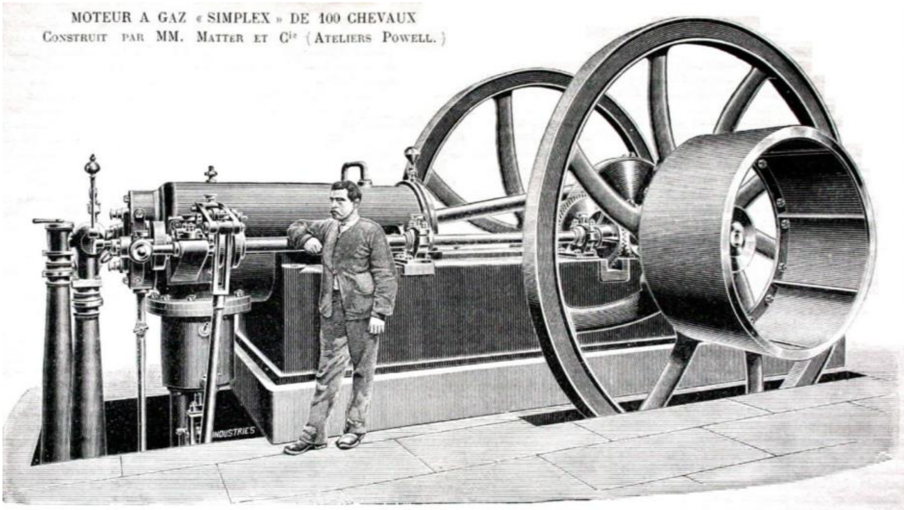
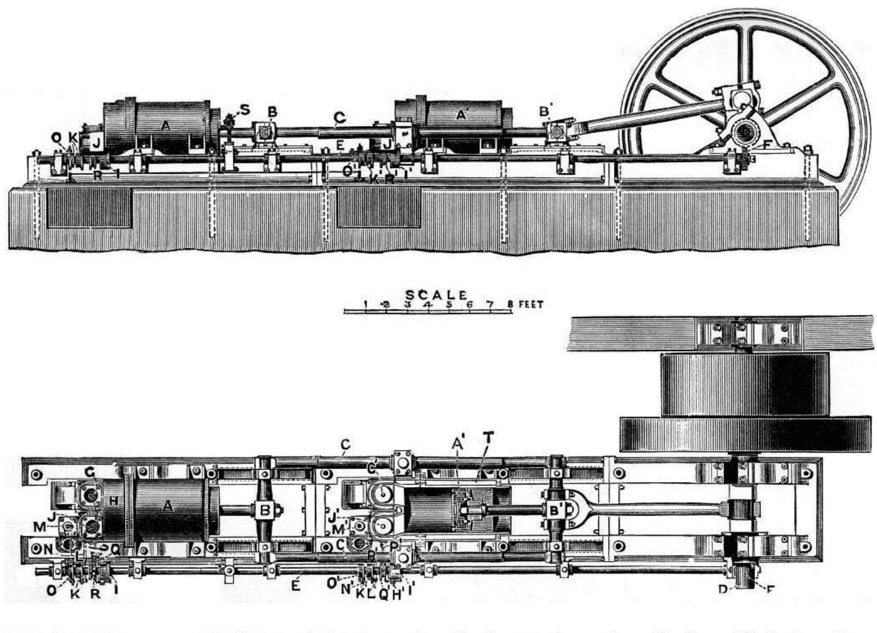


Figure 4. 100 hp Simplex gas engine of 1889.



LEADING DIMENSIONS.—Diam. of cylinders, 25″; length of stroke, 36″; diam. of exhaust valves, 9½″; diam. of feed valves, 9″; diam. of gas valves (Dowson gas), 9″; crank shaft diam. 10½″.

LIST OF PARTS.—Tandem pistons, A A′; crossheads, B B′; crosshead coupling rods, C C; crank shaft, D; valve gear side shaft, E; two to one worm gear, F; exhaust valves, G G′; exhaust valve levers, H H′; exhaust valve cams, I I′; feed valves, J J′; feed valve cams, K K′; feed valve levers, L L′; gas valves (Dowson gas), M M′; gas valve levers, N N′; gas valve cams, O O′; igniter timing valves, P P′; igniter timing valve levers, Q Q′; igniter timing valve cams, R R′; governor, S; water jackets, T.

Stockport Otto Engine, 400 IHP (plan part section).

Figure 5. A 400 indicated horsepower Stockport engine

the engine was used to drive a mill at Godalming and was supplied by a Dowson gas plant, but little else is known of its performance.[9] Indicated power amounted to a respectable 0.407 ihp/in^2 of piston area. We can guess that if the mean piston speed was 800 ft/min (133 rev/min) then the mean effective pressure was 67 lb/in^2, both of which were typical of engines burning Dowson gas. Evidently the customary arrangement of water-cooled cylinder and air-cooled piston was used.

However, large gas engines did not really develop until the very end of the nineteenth century when the possibility of using the vast quantities of waste blast furnace gas was considered. Its low calorific value, only about 20% of the calorific value of coal gas, and its very dusty nature, had discouraged interest in its use for gas engines; Lürmann had considered and rejected the possibility in 1886.[10] Such gas was usually cleaned and used under boilers to raise steam for steam engines, although, as mentioned above, much was wasted. However, the success of gas engines in burning low calorific Dowson gas had removed one objection, and the necessity to clean the gases before they could be used under a boiler had partially removed the second.

Donkin says that Mr B.H. Thwaite, at the Glasgow Iron and Steel Works, Wishaw, Scotland, was the first, in May 1894, to use the gases from the top of a blast furnace to produce power in a gas engine after the gases had been purified.[11] Thwaite later gives some account of his discovery.[12] The gases were drawn from the furnace through a large gas main to a condenser where the tar and ammonia were removed. Part was then used in the normal way to raise steam in boilers and to heat the air blast and part was forced through a scrubber and purifier, to remove dust, before passing into a holder and on to the gas engine. Thwaite used an Acme four-stroke gas engine of 12 inches bore and 20 inches stroke running at 190 rev/min and developing 15 bhp (which seems rather low). It generated power for electric lighting. Booth tested it in August 1896 and found that the calorific value of the blast furnace gas was 126 Btu/ft^3 and contained 27.8% of combustible gases. The furnace yielded 170,000 to 180,000 ft^3 of gas per ton of iron smelted and the consumption of the engine was 84 ft^3 per indicated horsepower per hour (24% indicated thermal efficiency). A very modest production of 100 tons or iron per day would thus give sufficient gas to generate 8,500 bhp. The need for large engines was manifest.

The success at Wishaw prompted similar schemes at Frodingham Ironworks where a 15 bhp engine was started in 1895, using gas of 102 Btu/ft^3 calorific value, and at Barrow-in-Furness, where a larger 160 hp engine was installed and the power used for electric lighting. The seed was sown, and larger gas engines suitable for blast furnace gas started to appear. The largest British engine, running on blast furnace gas, known to Donkin in 1901 was a 250 bhp single cylinder Crossley Premier engine, but he noted that Messrs Cochrane of Middlesbrough were then erecting a 600 hp Cockerill (Seraing) gas engine.

Probably the most important work on the use of blast furnace gases in gas engines was that undertaken on two engines successively worked at the ironworks of the Sociètè Cockerill at Seraing in Belgium.[13] Messrs Bailly and Kraft under the direction of Adolphe Greiner were asked to select the most suitable gas engine, and it was whilst testing a Simplex engine on producer gas that they conceived the idea, independently of the British trials, of using it in conjunction with blast furnace gas. The ironworks at Seraing produced 600 tons of pig iron per day at this time so there was an immense wastage of gas. The engine chosen was particularly suitable because its combustion chamber had a rounded shape and did not have corners where dust could settle. A small experimental 8 hp Simplex engine was started in December 1894. The test was very successful but it was not until April 1898 that a larger 150 hp engine was tested and that too gave excellent results.

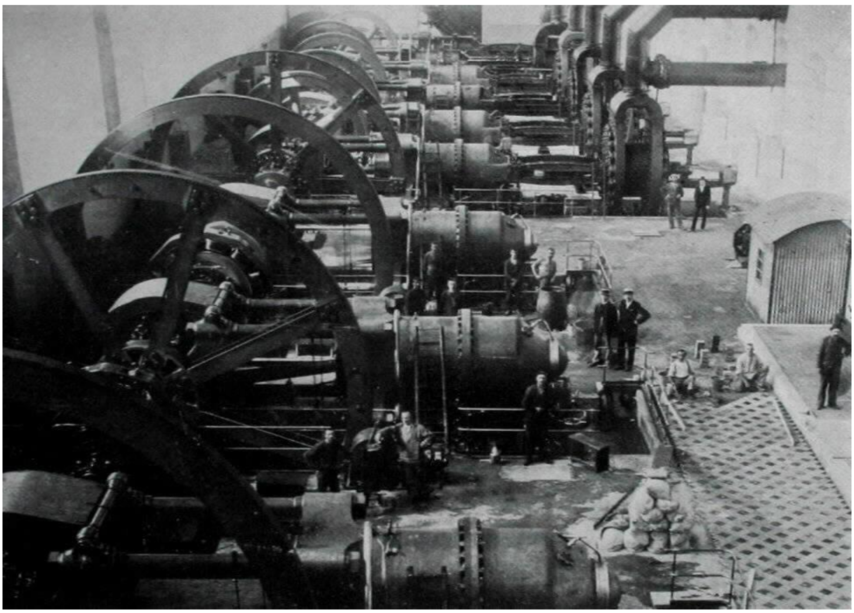


Figure 6. Société Cockerill gas engines at Differdingen. Six driving air blowers and three generating current for electric lighting. 700 ihp each.

Figure 6 illustrates the size that was reached in 1899 by single-cylinder four-stroke engines.[14] Here six Société Cockerill gas engines operating on blast furnace gas were used to drive blowers and three gas engines powered generators for electric lighting. The cylinder diameter was 51.2 inches, the stroke was 55.1 inches, and the weight was 127 tonnes without the flywheel. The enormous flywheels evident in Figure 6 were required to reduce speed oscillation and lighting flicker when used for electrical generation. Usually the flywheel weighed about

20% of the engine weight. It would have been much easier to use a larger number of small cylinders to generate the necessary evenness and power.

Greiner described these engines as the first blowing engine worked by blast furnace gas ever employed in any ironworks. Clearly he was unaware of British experiments, but his claim is justified. Greiner gave full credit to Delamare-Deboutteville, who invented the Simplex engine, and to his own engineers in developing a 600 hp engine which, by 1900, had been running for about a year on un-purified gas taken from the Seraing blast furnaces. By un-purified gas Greiner meant that the gas was not purified more than was normally required for gas burned under steam boilers. The indicated power varied depending on the quality of the gas supply but in tests 700 ihp was the average and this was equivalent to 550 hp in compressed air delivered to the furnace at a pressure of 6.7 to 7.8 lb/in^2. The thermal efficiency was 30%, some 50% of the heat being carried off by the cylinder cooling water and 20% by the exhaust gas. The mean piston speed was 736 ft/min and the engines developed 0.34 ihp/in^2 of piston area. Eminent engineers from Belgium, France, England, and Germany witnessed the trials at Seraing and returned to their homes enthused by what they had seen. The stage was set for the rapid development of large gas engines to take advantage of the new market.

Large Continental and American Gas Engines

At Seraing the designs for a 1200 hp engine were already made when Greiner delivered his paper in 1900 and three such engines had been ordered for a Lorraine ironworks where the furnaces were making 300 tons of pig iron per day. Progress on the Continent was rapid. Roberts-Austin states that by 1906 in some 41 German smelting works there were in operation or in course of erection 391 gas engines with an effective horsepower of 385,000, of which the greater proportion was worked by blast furnace gas.[15] The average power per engine was almost 1,000 hp.

The growth in the market for large gas engines using blast furnace gas was particularly marked on the Continent, as illustrated in Figure 7. Since the introduction of large four-stroke gas engines in 1902 they spread rapidly in Germany, as mentioned above. Their simplicity and ease of operation caused them to be preferred to two-stroke engines, which were also used for driving blowing engines.[16] The aggregate power of large gas engines used in Europe was about one million hp in 1910.

By 1908 the preliminary work at Cockerill had been superseded by larger and more ambitious designs, as exemplified in Table 3. German companies were particularly active.[17] Ehrhardt and Seher had made 59 engines producing an average of 1,183 hp per unit, MAN had made 215 units at 1,192 hp per unit, and Körting had made 198 engines of 837 hp average power. Crossley Bros, the only British manufacturer in the table, had made 57 engines producing an average of

415 hp, which stands in marked contrast to the many thousands of smaller engines they had produced. The total power of large gas engines at work in Europe amounted to about 698,000 bhp in 1908, and as mentioned above, it reached almost a million by 1910.

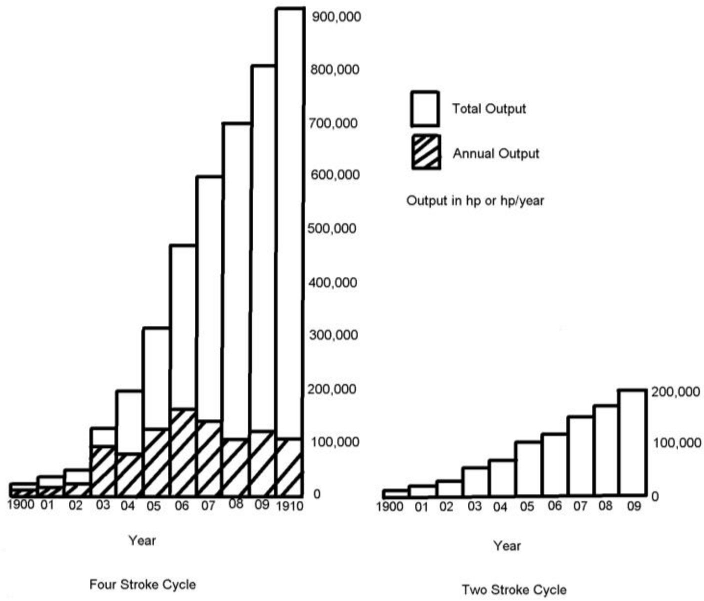


Figure 7. Grown in the installed capacity (hp) and the annual production (hp/year) of large four-stroke and two-stroke gas engines in Europe.

Table 3 European Large Engine Manufacture, c.1908

Manufacturer	**Number of Engines Produced**	**Total Power (bhp)**	**Average Power (bhp)**
Crossley Bros	57	23,660	415
Ehrhardt & Sehmer	59	69,790	1,183
Otto Gasmotoren Fabrik	82	47,400	578
Gebrüder Körting	198	165,760	837
Société Alcacienne	55	23,410	426
Société John Cockerill	148	102,925	695
Société Suisse Winterthur	67	8,620	129
Vereinigtein Maschinenfabrik Augsburg and Nürnberg	215	256,240	1,192
Total	851	697,805	820

Clerk and Burls state that an Erhardt and Sehmer engine generated 1,100 bhp per cylinder, had two crank throws, and on each throw there were two double-acting cylinders arranged in tandem.[18] The four-stroke cycle was used and the timing arranged so that the cylinders fired to give one power stroke every revolution. Thus the engine had four double-acting pistons generating a total of 4,400 bhp. The cylinder bore was 45.25 inch, stroke was 51.25 inch and engine speed was 94 rev/min. Accordingly the mean piston speed was 803 ft/min, and the power per unit of piston area was 0.342 bhp/in^2 (about 0.4 ihp/in^2). Because the pistons were double-acting they, and the piston rod that passed through the combustion chamber to operate the outer piston, had to be water-cooled because air-cooling through an open-ended cylinder was not possible.

MAN engines, Figure 8, were similarly arranged with two water-cooled pistons in tandem and hence equivalent to four single acting pistons.[19] The performance of a 1,200 bhp engine running on blast furnace gas is shown in Table 4. By June 1910 MAN claimed to have produced double-acting gas engines amounting to a total of 444,660 bhp. If Langer's figures are correct, Figure 7, then MAN had produced almost half the total power of large gas engines sold in Europe before 1910.

Table 4 Performance of a 1200 bhp engine MAN on blast furnace gas (88 Btu/ft^3).

Cylinder Bore and Stroke	Speed and Mean Piston Speed	Brake and Indicated Power	Brake and Indicated efficiency	Power per unit piston area
33.46 in x 43.3 in	105 rev/min 758 ft/min	1186 bhp 1427 ihp	28.5% 34.3%	0.406 ihp/in^2

There are two main reasons for German dominance in the large gas engine market. Firstly, German pig iron production had increased from 4 million tonnes/year in 1890 to 17 million tonnes/year in 1913 whereas UK production remained static, increasing from 8 million to 10 million tonnes/year in the same period.[20] German production exceeded UK production after 1905. Thus most German iron works were new and used the most modern methods, whereas most UK iron works were long established and were more difficult to change. Thwaite notes that a furnace of 1789 remained active at Lowmoor into the 1880s.[21] Secondly, German hard coal production was half that of the UK and whereas the UK was a major exporter of coal, Germany, supplemented by a considerable proportion of brown coal, produced barely sufficient for her own needs. Thus blast furnace and coke oven gas were more valuable in Germany than in the UK, where cheap coal removed the incentive to use waste gas more economically.

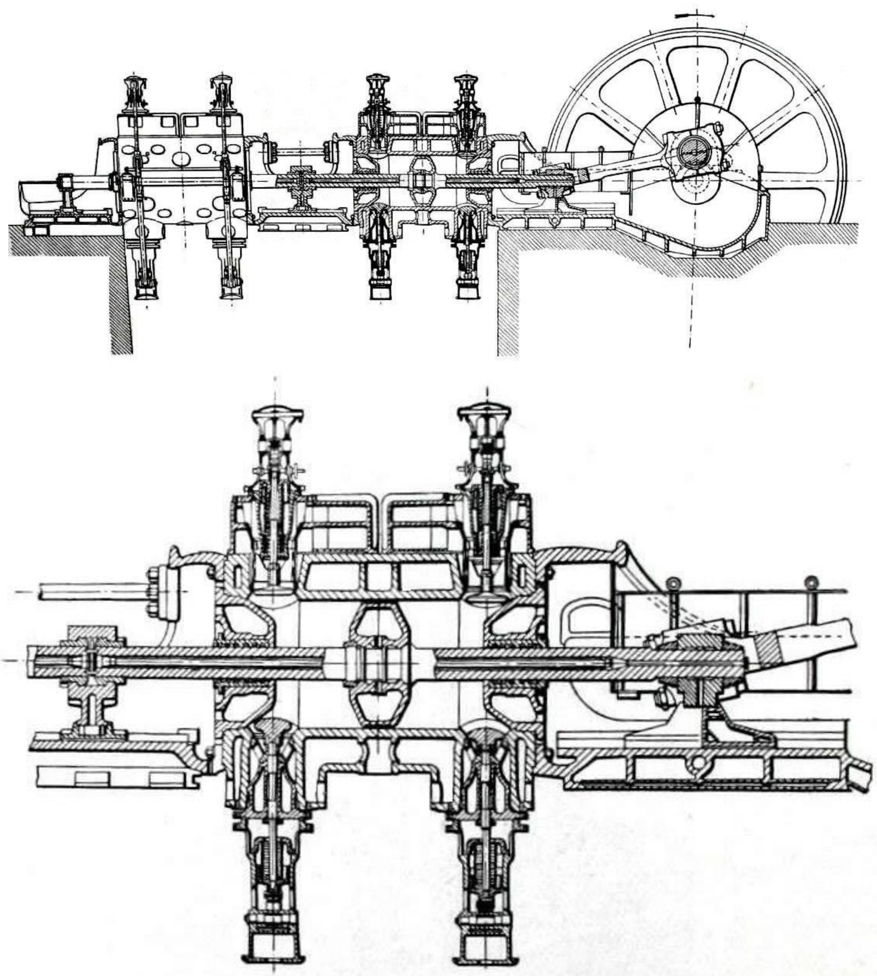


Figure 8. MAN tandem, double-acting, gas engine. Water for piston cooling is supplied along the piston rods.

Gebrüder Körting also built a large number of gas engine suitable for operations with blast furnace gas, but they preferred to use Clerk’s two-stroke cycle rather than the four-stroke cycle. In the United States the first large engines installed were the Körting-Clerk type and were built by De La Vergue Co, NY. Sixteen blowing engines of 2,000 bhp each, and 8 electric driving engines of 1,000 hp each, were installed in 1902.

Clerk and Burls state that the (American) Westinghouse Co., of Pittsburgh, manufactured horizontal, twin tandem engines, having two cranks and four double-acting cylinders per unit. The cylinder diameter was 38 inches and stroke was 54 in. This was equivalent to eight single-acting cylinders and although the power was not stated it should have produced about 3,000 bhp (3,630 ihp at 0.4 ihp/in^2). The Snow Steam Pump Company had a comparable horizontal, tandem engine of 42 inches cylinder diameter and 54 inches stroke. Similarly the William Tod Company, of Youngsten, Ohio, installed a 3,000 bhp four cylinder, twin tandem, double-acting engine at the Carnegie Steel Co., Ohio, having a cylinder bore of 42 inches, a stroke of 60 inches, and running at 75 rev/min. This also appears to be the equivalent of eight single-acting cylinders and thus the power per unit piston area is 0.27 bhp/in^2 (about 0.32 ihp/in^2), which is rather conservative.

Adams estimated the total power of large gas engines at work in the United States at about 350,000 bhp,[22] but Langer thought it was about 500,000 hp.[23] This was a half, or less, than that at work in Europe and Langer attributes this to the failure of two large two-stroke engines installed in the Lackawanna Steelworks, which, he thought, had discouraged investment. Nevertheless, some very large and successful gas engines were in use. The problem with large cylinders was that they tended to crack due to thermal and mechanical stress. Langer states:

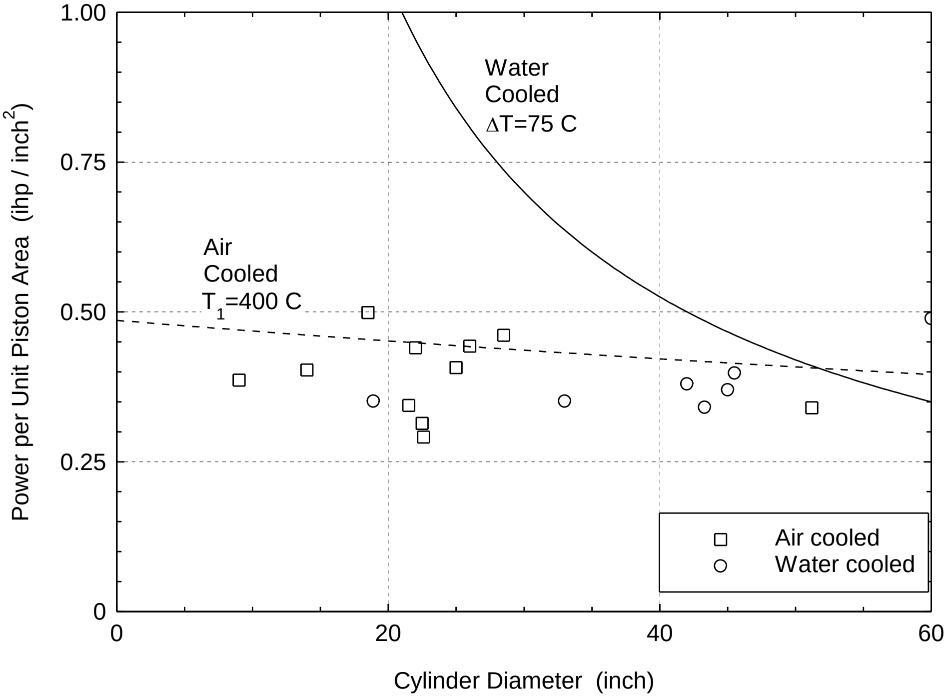
> If, however, it be remembered that a difference of 100 °C in temperature produces a stress of over 14,000 lb/in^2 when the longitudinal expansion of metal is prevented, it can only be a matter of surprise that such cracks are not even more frequent.[24]

He then discussed several alternative cylinder designs intended to avoid thermal stress failure. Cast steel had been tried in place of cast iron but had failed to solve the problem because the modulus of elasticity of steel was too high in relation to its tensile strength.

Thermal Limits of Gas Engines

Reciprocating steam engines were usually lagged to prevent heat loss. This was possible because the components could not become hotter than the steam temperature and the strength and creep properties of cast iron were good up to about 400 °C. If gas engines were un-cooled the components would reach temperatures of 800 C or more, so they had to be cooled to reduce the temperatures below about 400 C. Air-cooled gas engine pistons reach this limit when power per unit piston area reached about 0.5 ihp/in^2 or less (Figure 9).

The theoretical limit, for the assumptions specified below the figure, is shown by the dashed line and is derived in Appendix A. Water-cooled pistons do not reach 400 °C unless power reaches about 4 ihp/in^2, which is not possible with naturally aspirated engines. However, water-cooling was necessary for double acting pistons and was commonly used for cylinders and covers. Such water-cooling caused large thermal stresses and the critical limit in cast iron was reached when the temperature difference across the wall thickness reached about 75 °C, see



Maximum piston temperature T_1= 400 °C, coolant temperature T_c=20 °C, heat transfer coefficient h_c=0.528 Btu/ft^2/min/K (air-cooled) and 26.45 Btu/ft^2/min/K (water-cooled), thermal conductivity k=0.693Btu/(ft K min), indicated thermal efficiency η=0.37, heat flow to wall β=0.025 (air-cooled) or 0.03 (water-cooled), Poisson's ratio μ =0.3, coefficient of thermal expansion α=12x10^{-6} K^{-1}, modulus of elasticity E=16x10^6 lb/in^2, wall thickness t=0.06 of cylinder bore d.

Figure 9. Theoretical and actual power per unit piston area for air-cooled and water-cooled gas-engine pistons, cylinders and cylinder covers.

Appendix A. For a given heat flow the temperature difference increases with size, consequently large pistons are more liable to crack. The relation between power per unit area and thermal stress is derived in Appendix A and the theoretical limit is plotted in Figure 9. Water-cooled pistons, cylinders and covers larger than about 50 inches diameter failed at a lower power than air-cooled pistons and this set an upper limit to cylinder size. This limit had already been reached by 1910 and some large engines were proving to be unreliable. The way forward to higher powers was to use a larger number of smaller air-cooled cylinders. This was the solution adopted by English manufacturers. Modern natural gas engines, and diesel engines, also adopt the same solution except that they have oil or water-cooled pistons to accommodate the increased power generated by higher speeds and turbo-charging.

Large English Gas Engines

In correspondence related to Donkin's 1901 paper, Booth noted that the English ironmasters had practically ignored for the last six years the economy to be gained from Thwaite's invention, despite the successful running of the engine at Wishaw, but that Continental ironmasters had recognised the merit of the invention.[25] It was not until about 1910 that British gas engine manufacturers began to construct suitably large units. They had been content, until that time, to build large engines under licence from Continental manufacturers, or use smaller British engines. However, the English designers took a different approach from their Continental and American colleagues. Clerk and Burls remark on the "English engineers' repugnance to large double-acting cylinders with watered pistons and piston rods" and they note that:

> Most English makers of long experience prefer open cylinder engines with pistons having no watering arrangements, whereas the standard continental type for higher powers is now double-acting, tandem cylinders for the four-stroke cycle.[26]

It was not simply "repugnance" but good design that caused the change. Engines with large diameter cylinders were, as we have seen, liable to thermal failure. Moreover, they were very heavy and costly. Weight per horsepower increased as the square root of power (Figure 10). Single-acting, single cylinder engines were the heaviest but Continental preference for double-acting engines reduced weight per horsepower by a factor of two. The way to produce lighter, and therefore cheaper engines, and to reduce the thermal stress, was to use a larger number of smaller cylinders to generate the necessary power. This was the solution offered by two English companies.

The (English) Westinghouse Company was the first British company to offer large engines suitable for blast furnace gas. They used the simpler air-cooled, single-acting pistons, kept the cylinder diameter to a modest size, and obtained a large power by use of many cylinders. They showed a three-cylinder version at the Franco-British Exhibition in 1908, and Figure 11 illustrates the four-cylinder version. It was a vertical, tandem unit developing 1,000 bhp. Figure 11 also shows the firing order. The cylinder bore was 22 inches (top piston), 21 inches (bottom piston), and the stroke was 24 inches. Engine speed was 200 rev/min and the four-stroke cycle was used. The mean piston speed was 800 ft/min and the power per unit area of piston was 0.344 bhp/in^2 (about 0.4 ihp/in^2). The great reduction in weight (and cost) by this construction is illustrated in Figure 10. Clearly the future lay with such engines, although a vee construction is now preferred to the tandem arrangement.

Vertical, rather than horizontal, cylinders were adopted, probably because tandem construction doubled the unbalanced reciprocating force and horizontal cylinders thus required substantial holding-down bolts. There were no horizontal forces on vertical engines and the foundations were correspondingly simplified.

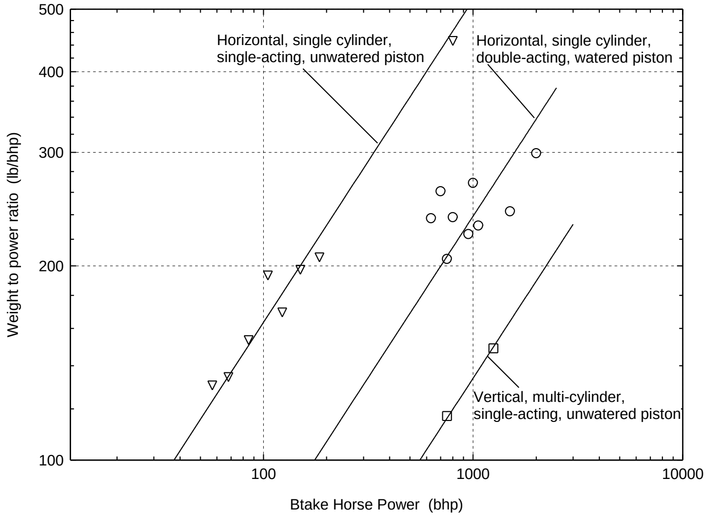


Figure 10. Weight per unit power of gas engines (not including flywheel).

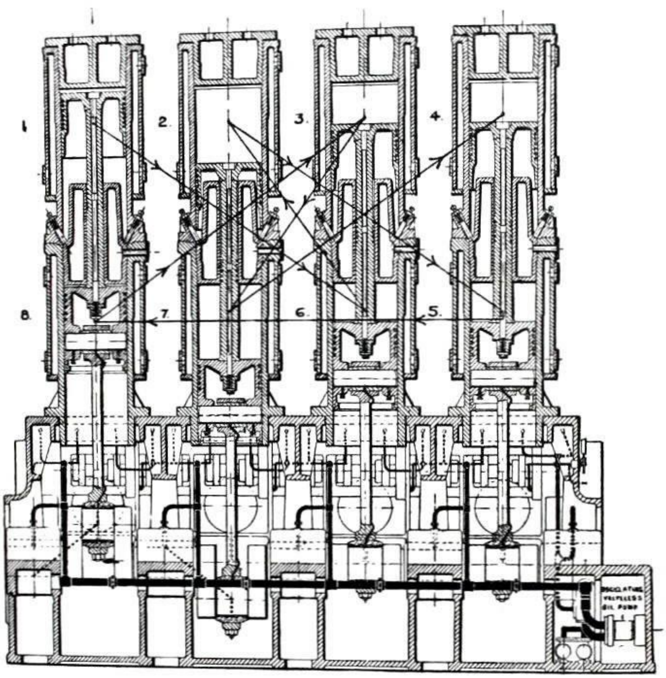


Figure 11. English Westinghouse 1,000 bhp tandem engine, showing firing order of cylinders.

This was not the first vertical gas engine, they were commonly used in the natural gas belt in America, but it was the first UK engine to reach 1,000 bhp. The firing order chosen was that now used for four-cylinder two stroke engines, so there was a firing impulse every 90 degrees of rotation, which dispensed with the need for a heavy flywheel, and also gave perfect primary and secondary balance of reciprocating forces. There were, of course, unbalanced primary and secondary couples, but these were of much smaller magnitude than the unbalanced forces of single cylinder engines of similar power.

The underside of the top piston was a closed space that was pressurised by the descent of the piston, which was said to form a buffer to counteract the inertia of the reciprocating parts. It is true that the buffer pressure relieved the engine bearings of the task of retarding the piston, but the buffer pressure acts equally on the cover of the lower cylinder and thus has no net effect on the reciprocating forces transmitted to the foundations. The rod connecting the bottom and top pistons passes through the bottom combustion chamber, but it is cooled as it passes through the cover of the lower cylinder. Two oil pumps in the sump feed all the main bearings, and through drillings in the crank they feed the big-end bearings, and through a pipe along the connecting rods they feed the crankpins. The camshaft is similarly lubricated. To avoid an unequal distribution of gas and air between the eight cylinders they were supplied as two separate groups of four cylinders.

The National Gas Engine Company put a somewhat similar vertical, tandem engine on the market in 1912.[27] Before 1910 they had built horizontal gas engines ranging from 2 hp to 300 hp, but the sales of the smaller engines were decreasing in the face of competition from small petrol engines and small electric motors. The directors considered that they needed to make larger engines and they invested about a million pounds in building a new and larger factory, claimed to be the world's largest devoted solely to gas engines. They determined on a vertical, tandem, multi-cylinder engine. Figure 12 illustrates two 1,000 bhp engines driving generators at a municipal power station. Amongst their first engines were a 750 bhp and two 1,000 bhp units for the "Festival of Empire" at the Crystal Palace, London, to work on town gas. In 1912 the company supplied three 1,500 bhp engines to Japan, and to municipal power stations, collieries, ironworks, shipyards, cotton mills, and so on. In 1916 the company introduced a range of large blowers for blast furnace work and an order was received for engines to generate 11,500 bhp, of which 4,500 bhp was to drive the blowers, giving 72,000 ft^3/min at a pressure of 12 lb/in^2. The isothermal power required to compress this quantity of air is 2,781 bhp, so the isothermal efficiency of the compressor was about 62%, which is quite respectable.

The arrangement of tandem cylinders is shown in Figure 13. By firing the top and bottom cylinders alternately on the four-stroke cycle there was a working stroke with every down stroke, and a compression stroke with every up stroke. The space formed between the top piston and the intermediate cover was used, as in the Westinghouse engine, as a buffer and the company claimed the moving parts were, therefore, cushioned on both the up and the down strokes, and this, together with the number of cranks, gave an exceedingly even turning effort (ideal for electrical

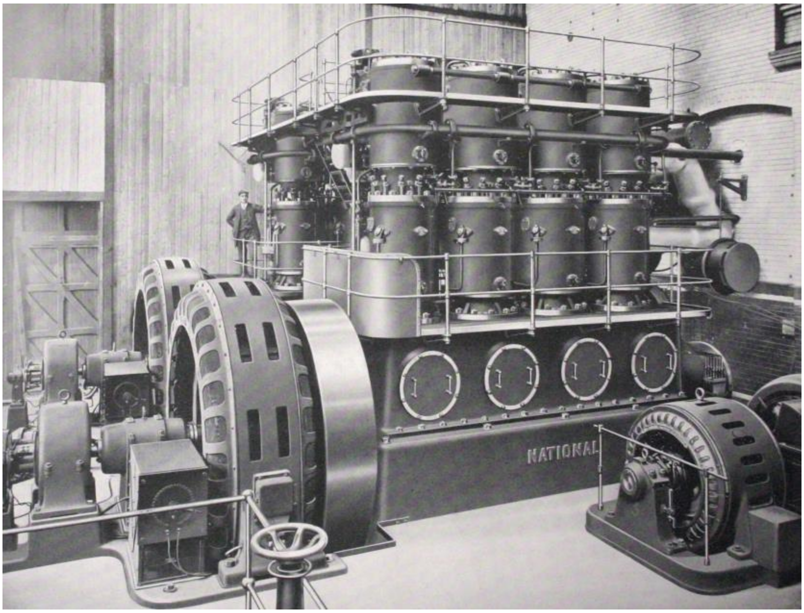


Figure 12. Installation of two 1,000 bhp National gas engines coupled to alternators at a municipal power station. The size may be judged from the man standing on the platform above the generators.

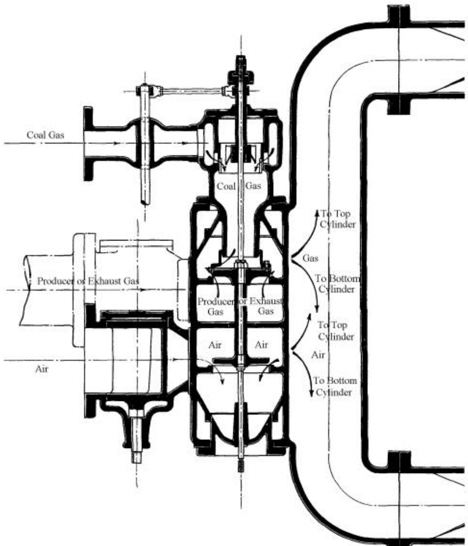
power generation). Setscrews (polescrews) were used to hold down the cover over the bottom cylinder and the joint was made on a small ledge fitted with a copper sealing-ring. Hit and miss governing, which produced great irregularity in speed, was replaced by quantity governing in which a valve throttled both the air and gas. If need be the mixture ratio could be adjusted with changes of load to give partly quantity and partly quality governing. The cylinders were cooled by water but this was not adopted for the pistons, because it was considered important that no moving parts should be water-cooled, presumably because of the difficulty of transferring water to moving parts without leakage and contamination of lubricating oil. Ignition was by means of two high-tension sparking plugs in each cylinder.

The 1,500 bhp engine consisted of 12 pistons (6 cylinders) of 24 inches stroke. The diameter of the upper cylinder was 23 inches, 22 inches for the lower cylinder, and it operated at 200 rev/min. The full-load brake mean effective pressure was 50.6 lb/in^2, which is what might be expected of an engine running on blast furnace gas having a lower calorific value of only 100 Btu/lb. The mean piston speed was 800 ft/min and the power per unit piston area was 0.314 bhp/in^2, about 0.37 ihp/in^2. The fuel consumption was stated to be less than 10,000 Btu/(bhp hour), which is better than 25% brake thermal efficiency.

Figure 13. Arrangement of the tandem cylinder on National gas engines, 1912.

When the engine ran on richer fuels (normal town gas) there must have been a danger of overheating or thermal cracking, which was always a risk on large engines, and consequently the mixture admitted to the engine was diluted with exhaust gas to reduce its energy content. It was an early example of exhaust gas re-circulation, and the change from one gas to the other was made while the engine was running and without any noticeable flicker in the engine speed. The complex valve and pipe arrangements to dilute coal gas with exhaust gas, mix with air, and supply the upper and lower cylinders are displayed in Figure 14.

Engines of this size could not be shipped ready assembled from the factory but the normal procedure was to assemble and test them in the presence of the customer's engineer, and then strip down and package the components for delivery to site. The packaged gas engine was delivered by rail, and perhaps by sea, and the length of train for a single 1,500 bhp gas engine amounted to fifteen or more railway wagons, Figure 15. An American engine was said to require 45 fifty-ton railway wagons for its delivery.



Mixing Arrangements for Rich Gas and Exhaust Gas

Figure 14. Mixing arrangements for coal gas, producer/exhaust gas, and air for a 24 inch stroke National tandem gas engine.

Figure 15. A 15-wagon train loaded with one 1,500 bhp National vertical tandem gas engine outside the company's Large Engine Department.

Subsequent Developments

By 1914 the period of rapid increase in size and power was at an end because larger or more powerful engines proved unreliable and were subject to thermal failures. In 1917 in was reported that in America “there are few large units being built and installed” because of “the decline of natural gas in certain sections; the attraction of low first cost in steam turbines; and the development of diesel and semi-diesel engines”. Moreover:

> some of the gas engines built from the earlier designs were subject to inherent weaknesses and their operation was not entirely satisfactory.” Gas engines would seem to “have had their greatest popularity unless some great improvements can be made in their design, so as to produce cheaper and more reliable machines or to provide more economical operation.[28]

In England the large gas engines built by Westinghouse and by National were quite successful and in 1917 it was reported that National had sold over 100 of their 1,000 and 1,500 bhp engines despite their sales being hampered by the hostilities of 1914-19.[29] Several British companies had offered horizontal tandem gas engines of the Continental type, namely “Richardson, Westgarth & Co., Mather & Platt, W. Beardmore & Co, the Lilleshall Co., and Galloways, who have respectively taken up the manufacture of the Cockerill, Koerting, Oechelhauser, Nürnberg, and Ehrhardt and Sehmer types”.[30]

In 1919 an editorial in *Gas and Oil Power* expressed the hope that British manufacturers

> should be able to make inroads into the large horizontal gas engine trade previously monopolized by Continental firms. In large vertical engines British firms had already taken the lead.[31]

It was not to be. In 1919 both National and Vickers were advertising their large gas engines but by 1926 such advertisements had ceased. Neither Westinghouse nor National produced further designs of vertical types because the market for large gas engines, at least in Britain, declined in the face of competition from diesel engines, and a preference amongst British iron producers for steam. Gas engine makers increasingly turned over their production to diesel engines. As early as 1915 Chorlton had written a paper on the conversion engines from gas to oil[32] and as late as 1940 the National marketed the first dual-fuel engines.[33]

In the United States and Germany gas engines continued in use, particularly in the American oil fields where they were used to burn natural gas and flare gas that would otherwise be wasted. In 1930 a very large American gas engine using blast furnace gas was announced (Figure 16). It was a four-stroke, double-acting, twin-tandem type developing 6,600 kW (10% overload) from 60 inch diameter cylinders of 64 inch stroke and running at 83.3 rev/min (about 0.49 ihp/in^2). The mean piston speed was 889 ft/min the mean effective pressure was a remarkable 72 lb/in^2.[34]

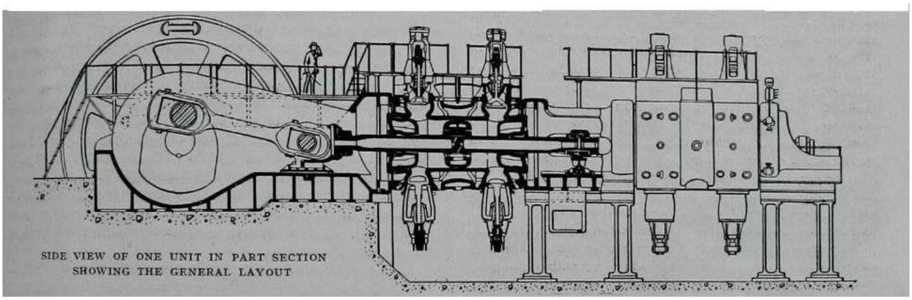


Figure 16. A large American gas engine for blast furnace gas. Sixty inches cylinder diameter and 6,600 kW (10% overload). The man (left centre) is almost lost.

In Britain, although duel fuel and convertible gas/oil engines were developed,[35] the market remained small and it was not until the arrival of natural gas from the North Sea in about 1967 that new engines appeared, based on current diesel engine configurations. Typical of which was the Mirrlees-National KV-Major, which was a 16-cylinder turbocharged vee-engine of 15 inches cylinder diameter, 18 inches stroke, and developed 5,936 bhp at 514 rev/min (2.1 bhp/in^2, 1542 ft/min piston speed).[36]

On the Continent, GE-Jenbacher have recently introduced a new gas engine of 5,362 bhp at 1500 rev/min from 24 cylinders of 7.5 inches diameter and 8.66 inches stroke. The power per unit piston area is a remarkable 5.43 bhp/in^2 and the piston speed is 2,165 ft/min.[37] Clearly, the thermal and mechanical limitations first experienced by large gas engines still apply, and not only to gas engines but also to diesel and petrol engines, neither of which have exceeded the cylinder diameter of large gas engines. However, the use of oil-cooled pistons with steel crowns in conjunction with turbo-chargers has enabled engine power per unit piston area to increase at a steady rate. The British approach, using a large number of smaller, high speed pistons, proved to be the way ahead.

Appendix A. Size Limits of Gas, Diesel, and Oil Engines

The thermal limits are illustrated in the graphical construction shown in Figure 17. *AB* represents the heat transfer through the wall of an air-cooled gas engine piston. The mean temperature of the combustion gas, Tg, and the ratio of the piston's thermal conductivity, k, and the mean heat transfer coefficient of the combustion gas, h_g defines the point *A*. The point *B* is specified by the coolant temperature, T_c, and the ratio of the conductivity, k, and the coolant's heat transfer coefficient, h_c. T_1 and T_2 are the temperatures of the piston surface and t is its thickness.

Increasing the engine power moves point A to point C and the piston temperature rises, as indicated by line CB. If piston temperature rises above 400 C, for cast iron, then the tensile strength falls and creep rate increases. To avoid this the power must be decreased or air-cooling may be replaced by water-cooling. Water-cooling moves point B to point D and the piston temperatures are lowered, as indicated by line CD. However, the temperature difference across the piston is now increased and hence the thermal stress is increased and may cause failure. In cast iron a temperature difference in excess of 75ºC is likely to cause failure. Pistons, and other combustion chamber components, must be designed to avoid these two failure modes.

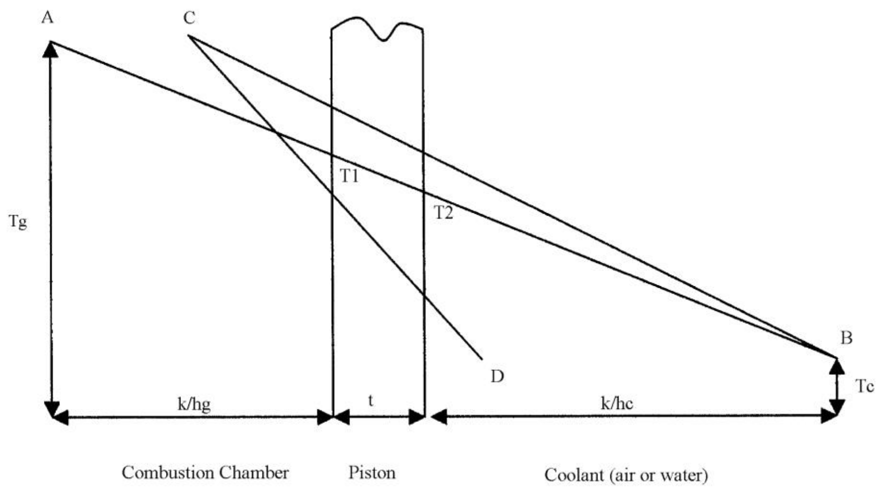


Figure 17. Graphical construction for heat flow through a piston crown.
AB air-cooled, *CB* increased power, *CD* water-cooled.

a. Mechanical Stress

The thickness of the piston crown, t, was usually taken to be the same as the thickness of the cylinder and is, therefore, determined from the well-known relation for the circumferential stress in a thin cylinder:

$$\sigma_p = \frac{pd}{2t}$$

$$\text{that is } t = \frac{pd}{2\sigma_p} = \frac{500 * d}{2 * 5,000} = 0.05d \qquad (1)$$

σ_p is the pressure stress, p is the maximum cylinder pressure, and d is the cylinder diameter. For gas engines the maximum pressure was often taken as 500 lb/in^2 and the allowable stress in cast iron as 5,000 lb/in^2, consequently the thickness is 0.05 of the cylinder diameter. Usually, however a slightly thicker wall, corresponding to 0.06 of the cylinder diameter, was used.[38]

The low stress allowed for cast iron, 5,000 lb/in^2, may be surprising, considering that the tensile strength of the cast iron then used was about 20,000 lb/in^2 to 60,000 lb/in^2, depending on the grade and the size of the test bar (large test bars have a lower tensile strength). But experience had shown that this was needed and we now know that it was necessary to avoid fatigue failure. The analysis shown below (Equ.15) confirms this apparently conservative limit.

b. Thermal Stress

Thermal stress was very important and imposed an upper limit to the bore diameter of the cylinder because as size increased the thermal stress of the combustion chamber's components also increased and eventually exceeded the material's strength. A component subjected to a uniform rise in temperature, and free to expand, is not subject to thermal stress, but if there is a variation in temperature through the material, or the material is not free to expand, then thermal stresses are generated. Timoshenko has shown that the thermal stress in a simple thin-walled plate subjected to a linear temperature gradient through its thickness is:

$$\sigma_{th} = \frac{\alpha E(T_1 - T_2)}{2(1-\mu)} \qquad (2)$$

where σ_{th} is the thermal stress at the outer and inner surfaces (tensile on the cooled side, compressive on the heated side), α is the coefficient of thermal expansion, E is the modulus of elasticity, T_1 is the hot-side temperature, T_2 is the cool-side surface temperature, and μ is Poisson's ratio.[39] As cast iron is weaker in tension than in compression it is the cooled surface that is critical. Langer was most concerned about the risks of cylinder cracking and notes that a temperature rise of 100° C produces a stress of over 14,000 lb/in^2, when the longitudinal expansion of the metal is prevented.[40] The extension of such cracks, he says, was prevented by chiselling, drilling, and stemming with soft copper, but this was unavailing for deep cracks because the high coefficient of expansion of copper tended to burst open the crack.

It is shown below that to prevent fatigue the pressure stress should be a quarter of the tensile strength and the thermal stress should be half of the tensile strength, thus the limiting temperature difference across the piston crown or across the cylinder wall is:

$$\begin{aligned} T_1 - T_2 &= \frac{(1-\mu)TS}{\alpha E} \\ &= \frac{0.7 * 20,000}{12 * 10^{-6} * 16 * 10^{6}} = 73 \ \ \mathrm{C} \end{aligned} \tag{3}$$

TS is the tensile strength. For cast iron this temperature difference is about 75 °C, if the tensile strength is taken to be 20,000 lb/in^2. Air-cooling could not generate such a temperature difference across a plate but water-cooling could, as indicated in Figure 17.

From Fourier's law the temperature difference across the piston crown is:

$$T_1 - T_2 = \frac{qt}{k} \tag{4}$$

q is the heat flux through the wall thickness and *k* is the thermal conductivity. The heat flux may be taken as some fraction (β) of the rate at which energy is released by combustion. Expressed per unit area of piston we have:

$$\begin{aligned} \eta &= \frac{W/A}{Q/A} \\ \text{Thus} \ \ q &= \beta Q/A = \frac{\beta}{\eta}\left(\frac{W}{A}\right) \end{aligned} \tag{5}$$

η is the indicated thermal efficiency, *W* is the indicated power, *A* is the cross-sectional area of the cylinder, and *Q* is the heat supply rate. Substituting into Equ.4 gives the relation between power per unit piston area and the thermal stress, namely:

$$\begin{aligned} \left(\frac{W}{A}\right) &= \frac{\eta}{\beta}\frac{k(T_1 - T_2)}{t} \\ t &= 0.06d \end{aligned} \tag{6}$$

This is plotted in Figure 9. β, we now know, from measurements and from computer studies, is typically about 0.025 for air-cooled pistons and 0.03 for water-cooled pistons.[41] Clearly as the bore diameter of the cylinder increases the thickness also increases (t = 0.06d) and consequently the indicated power per unit area of the piston (*W/A*) decreases. There is a cylinder diameter beyond which it is impractical to run a water-cooled piston/cylinder because power output would be too low.

To increase the power output per unit area one should choose a material with high thermal conductivity and tensile strength, and low coefficient of thermal expansion and modulus of elasticity. Cast iron was almost always used and typical properties are specified below Figure 9.

c. Maximum Piston Temperature

The piston crown is the most difficult area to cool and initially it was cooled only by the motion of air against the underside of the piston as it moved to and fro in an open-ended cylinder. Above about 15 inches cylinder diameter internal radiating ribs were added to improve cooling; and above 20 inches cylinder diameter it was often cooled by water, circulating through telescopic or articulated tubes. The material used for the piston was cast iron and the piston crown was usually made the same thickness as the cylinder wall, namely, 6% of the cylinder bore. Cast iron has a maximum working temperature of about 400 °C; above this temperature its tensile strength declines somewhat, but more significantly, creep becomes important and can lead to the formation of residual tensile stresses that form radial cracks on the hot side of pistons.[42] Thus the power generated above the piston should not be so great as to exceed the maximum allowable piston temperature.

Assuming one-dimensional steady heat flow in the centre of the piston then the piston temperature is defined by the usual heat flow equations (Fourier's law and Newton's law):

$$q = k\frac{(T_1 - T_2)}{t} = h_c(T_2 - T_c) \qquad (7)$$

q is the heat flux, k is thermal conductivity, T_1 is the gas-side piston temperature, T_2 is the coolant-side piston temperature, t is the piston crown thickness (taken to be .06d), h_c is the coolant heat transfer coefficient, and T_c is the coolant temperature. Eliminating T_2 from these equations gives:

$$q = \frac{kh_c(T_1 - T_c)}{h_c t + k} \qquad (8)$$

Equating Equ.5 and Equ.8 and re-arranging gives the relation between engine power and piston temperature:

$$\left(\frac{W}{A}\right)=\frac{\eta}{\beta}\frac{kh_c(T_1-T_c)}{(h_c t+k)} \qquad (9)$$
$$t=0.06d$$

For air-cooled pistons $h_c t << k$, and thus the power per unit area of piston is almost independent of engine size and is usually about 0.5 ihp/in^2 if piston temperature is kept below 400 °C, see Figure 9.

d. Allowable Stress

The permissible thermal and pressure stresses are not independent. The thermal stress has a steady value for an engine running at constant load but the pressure stress fluctuates as the gas pressure fluctuates and thus the components are liable to fail in fatigue. This form of failure was not properly understood at the beginning of the twentieth century and it was found necessary to base design stresses on experience and use large factors of safety, perhaps as high as four or more.

Figure 18 (left) defines the alternating stress and the mean stress in relation to the applied pressure stress and the thermal stress, and Figure 18 (right) shows Goodman's relation between the alternating (cyclic) stress, σ_a, and the mean stress, σ_m.[43] In the normal fatigue test the mean stress is zero and, for cast iron, fatigue failure occurs if the fatigue stress is greater than a third of the material's tensile strength (*TS/3*). When the alternating stress is zero the material fails when the mean stress reaches the tensile strength, as in the normal tensile test. For intermediate mean stresses the allowable alternating stress is assumed to vary linearly between these conditions, thus:

$$\sigma_a=\frac{TS-\sigma_m}{3} \qquad \sigma_m>0 \qquad (10)$$

Now, the alternating stress for a gas engine cylinder is half the pressure stress and the mean stress is the thermal stress plus half the alternating stress (if the pressure stress is tensile), as Figure 18 (left) indicates.

$$\begin{aligned}\sigma_a&=\frac{\sigma_p}{2}\\ \sigma_m&=\sigma_{th}+\frac{\sigma_p}{2}\end{aligned} \qquad (11)$$

Substituting Equ.11 into Equ.10 gives the failure criterion:

$$2\sigma_p + \sigma_{th} = TS \tag{12}$$

From Equ.1, 2 and 6 the thermal stress may be written as:

$$\sigma_{th} = \frac{\beta}{\eta} \frac{\alpha E}{4(1-\mu)k} \left(\frac{W}{A}\right) \frac{pd}{\sigma_p} \tag{13}$$

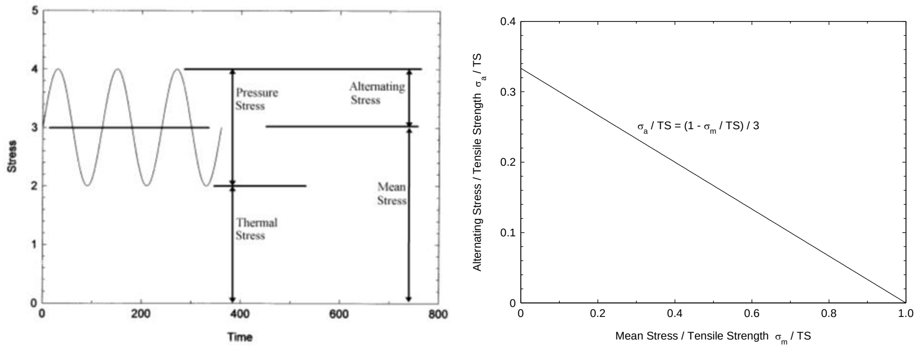


Figure 18 Left: Idealised steady and alternating stresses.
Right: Simple Goodman diagram for cast iron

If f is the factor of safety (the ratio of the allowed thermal stress, Equ.12, to the actual thermal stress, Equ.13) then:

$$f = \frac{TS - 2\sigma_p}{\sigma_{th}}$$

$$f = \frac{4(1-\mu)k\eta\sigma_p(TS - 2\sigma_p)}{\alpha E \beta (W/A) pd} \tag{14}$$

Differentiating Equ.14 with respect to σ_p and equating to zero shows that the factor of safety has a maximum value when the pressure stress is chosen to be a quarter of the allowable tensile strength, and the corresponding thermal stress is then a half of the tensile strength,

Factor of Safety is maximum when:

$$\sigma_p = \frac{TS}{4} \qquad \sigma_{th} = \frac{TS}{2} \tag{15}$$

Thus the practical conservatism of the early gas engine designers in choosing a rather low allowable stress for the cylinder and piston is justified.

e. Application

Equ.1, 9 and 6 may be used to compute the performance limits of gas engines with a reasonable degree of accuracy. However, the heat transfer to the piston, defined by the ratio β, is not well known, although for most engines it has a value of about 0.03. A more accurate approach is to model the performance of a typical gas engine using the well-known emptying and filling equations combined with suitable models for gas flow through inlet and exhaust valves and for convective/radiative heat transfer between the combustion gases and the combustion chamber walls. Such a computer program has been described by the author and confirmed the results illustrated in Figure 9.[44] The principal characteristics used in the computation are summarised below the figure.

f. Power per Unit Piston Area

This is a useful quantity and has been used frequently in the paper. It is related to the mean effective pressure and the mean piston speed. For a four-stroke engine the horsepower is:

$$W = \frac{p_m ALN}{2*33000} \quad \text{and} \quad C_m = 2LN$$
$$\text{Thus} \quad \frac{W}{A} = \frac{p_m C_m}{4*33000} \qquad (16)$$

Both the mean effective pressure, p_m, and the mean piston speed, C_m, are more or less independent of engine size, and so the power generated per unit area of piston is also more or less independent of size. A useful relation (identity) for determining the mean effective pressure is:

$$p_m = \frac{\eta_v \eta E_v}{f_v + 1} \qquad (17)$$

η_v is the volumetric efficiency (0.9), η is brake thermal efficiency (0.25), E_v is the calorific value of the fuel per unit volume (92 Btu/ft^3), and f_v is the volumetric air/fuel ratio (0.7 for stoichiometric combustion of blast furnace gas). Allowing for 30% excess air supply the brake mean effective pressure of a gas engine using the stipulated figures is 58.6 lb/in^2. If the mean piston speed is 800 ft/min then from Equ.16 the brake power per unit area of piston is 0.355 bhp/in^2 (0.4 ihp/in^2).

Notes and References

1. B.H. Thwaite, 'The Blast Furnace as a Centre of Power Production' in *Cassier's Magazine* XXXIII (Nov-Apr, 1907-8), pp. 23-40.

2. H. Jennings, *Pandaemonium* (Papermac, London, 1995), pp. 79, 165.

3. Sir W.C. Roberts-Austin, *Introduction to Metallurgy*, (Charles Griffin & Co., London, 1910), p. 374; T.K. Derry, and T.I. Williams,

A Short History of Technology, (Oxford University Press, 1960), p. 478.
4. B. Donkin, 'Motive Power from Blast-Furnace Gas' in *Proc ICE* CXLVIII (1901), pp. 1-55.
5. Advertising Catalogue 131, (Crossley's Otto Engine), Marks and Clerk Collection, Science Museum Library, Wroughton, Wilts.
6. Anon., 'Legal Intelligence: Otto v Steel', *The Engineer* 4 Dec 1885, p. 433.
7. Advertising Catalogues: 34 (Clerk/Sterne), 85 (Otto and Langen), and 47 (Campbell), Marks and Clerk Collection, Science Museum Library, Wroughton, Wilts.
8. Anon., 'Exposition Universelle de 1889' in *Revue Industielle* 42, 29 July 1889, p.413.
9. D. Clerk, *The Gas and Oil Engine*, (John Wiley & Sons, New York, 1906), pp. 322-5.
10. F.W. Lürmann,, 'Kupolofen mit getrennter Verbrennung des Kohlen-oxydgases' in *Zeitschrift des Vereines deutscher Ingenieure* XXX (1886), pp. 704-5.
11. Donkin, op. cit.
12. Thwaite, op. cit.
13. A. Greiner, 'A Blowing-Engine Worked by Blast Furnace Gas' in *Journal of Iron and Steel Inst.*, 1900, pp. 109-12; L. Greiner, 'The Utilisation of the Waste Gases of Blast Furnaces and Coke Ovens in Metallurgical Works' in *Cassier's Magazine* XXXIII, (Nov-Apr 1907-8), pp. 68-82.
14. D. Clerk, and Burls, *The Gas, Petrol, and Oil Engine*, Vol.1, (Longmans, Green and Co., 1909), p. 48.
15. Sir W.C. Roberts-Austin, *Introduction to Metallurgy* (Charles Griffin & Co., London, 1910), p. 371.
16. Prof. Langer, Prof., 'Recent Experience in Working Large Gas Engines' in *Mechanical World* (30 Sept, 1910), pp. 158-9.
17. Clerk, op. cit., pp. 47-8.
18. Ibid.
19. Ibid., pp. 125-38.
20. B.R. Mitchell, *European Historical Statistics, 1750-1975*, Macmillan Press, 1975, p. 215 and p. 184.
21. B.H. Thwaite, "The Blast Furnace as a Centre of Power Production", Cassier's Magazine, Vol XXXIII, Nov-Apr, 1907-8, p. 26.
22. E.T. Adams, "The Development of Large Gas Engines in America", Cassier's Magazine, Vol XXXIII, Nov-Apr 1907-8, pp. 41-54.
23. Langer, op. cit.
24. Ibid.
25. Donkin, op. cit.
26. Clerk, op. cit., pp. 48, 161.
27. Anon., "National Vertical Gas Engines", Ashton-under-Lyne Public Library Archive, D64/5.
28. Anon., 'The Gas Engine in the U.S.A.', *Gas and Oil Power* 12, (1916-7), p. 57.
29. Anon., "A Big Record", *Gas and Oil Power* 13 (1917-8), p. 115.
30. Anon., 'Large Gas Engines: Evolution of Types and Economic Progress' in *Gas and Oil Power* 10, (1914-15), pp. 124-32.
31. Editorial, *Gas and Oil Power* XV, No. 172, 1 Jan 1919, p. 1.
32. A.E.L. Chorlton, 'Convertible Combustion Engines' in *Proc.I.Mech.E.*, (Feb 1915), pp. 155-69.
33. J. Jones, "The Dual-Fuel Engine", (Diesel Engine Users' Association, Oct 1940), pp. 1-13.
34. Anon., 'Two Notable Blast Furnace Gas Engines' in *Gas and Oil Power* 25 (1930), pp. 175-6.
35. Jones, op. cit.
36. Anon., 'Mirrlees-National K+P-Major Duel-Fuel Engine' in *Gas and Oil Power* (Dec 1968), pp. 298-300.
37. Anon., 'GE-Jenbacher Engine for Tomato Greenhouses' in *Diesel and Gas Turbine World* 37, (Jan-Feb 2005), pp. 26-8.
38. Anon., 'Notes on the Constructional Details of Gas Engines, III; in *Mechanical World* (21 Oct 1910), p. 200.
39. S. Timoshenko, *Strength of Materials* Part II (D. Van Nostrand Co., New York, 1960), p. 91.
40. Langer, op. cit.
41. Papers 1, 2, and 10, Symposium: 'Thermal Loading of Diesel Engines', *Proc.I.Mech.E.* 179, Part 3C (1964-5).
42. D. Fitzgeorge and J.A. Pope, 'An Investigation of the Factors Contributing to the Failure of Diesel Engine Pistons and Cylinder Covers' in *N.E. Coast Instn. Shipbuilders* 71 (Feb 1955), p. 163.
43. Timoshenko, op. cit., p. 476.
44. B. Lawton and D.H. Millar, "Digital Simulation of Rotary Piston Engines",

Conference on Control and Simulation of Mobile Power Plant (I.Mech.E., London, 1973), pp. 25-32.

Notes on Contributor

Bryan Lawton served his apprenticeship at the National Gas & Oil Engine Co Ltd, and at Mirrlees-National Ltd, gaining BSc and PhD degrees at Salford University. He lectured at the UMIST before transferring to the RMCS, Shrivenham where he worked on Wankel and diesel-Wankel engines, heat transfer and wear in engines and in large guns, liquid propellant guns, transient temperature measurement, computer tomography, and the measurement and theory of skin burns. He was, until retirement, Reader in Thermal Power at Cranfield University, Shrivenham and has published numerous papers. He has written or contributed to books on ballistics, trauma, and transient temperatures. Historical works include papers on canal history, the characteristics of technology, large gas engines, and a two-volume history of pre-industrial mechanical engineering.
Correspondence: Bryan Lawton, bandbalawton@talktalk.net.