Journal of Power Sources, 7 (1982) 215 - 256

EFFICIENCIES OF HEAT ENGINES AND FUEL CELLS: THE METHANOL FUEL CELL AS A COMPETITOR TO OTTO AND DIESEL ENGINES

R. W. GLAZEBROOK

Shell Research Ltd., Thornton Research Centre, P.O. Box 1, Chester CH1 3SH (Gt. Britain)
(Received July 3, 1981)

Summary

As the real cost of fuel rises the efficiency of energy conversion devices will become of increasing importance. Efficiency is a variable factor depending *inter alia* on load factor. Whereas heat engines commonly yield optimum efficiencies at near to maximum power, fuel cells yield optimum efficiencies at zero power. Projections based on realistic developments suggest that fuel cells will operate overall with higher efficiencies than heat engines when load factors are below ~45%. Road transportation generally operates at load factors much lower than this and represents a suitable market for fuel cells.

1. Introduction

The fuel cell is an attractive concept as an energy conversion device because it enables work to be obtained from the chemical energy of fuels whilst avoiding the use of thermal cycles. The thermodynamic laws that limit the efficiency of heat engines are thereby circumvented and, in principle, high efficiencies can be obtained. Nevertheless, experience accumulated over many years has shown that the high efficiencies once proclaimed for fuel cells will be hard to achieve. It seems likely that the margin between fuel cells and heat engines will be relatively small and heavily dependent on application.

The rising cost of fuel can be expected to direct attention increasingly to efficiency of use. It is therefore appropriate to consider not only how the fuel cell stands today with respect to heat engines, but how it may compare within the next decade or so in the light of developments in both systems that might be reasonably expected.

A comprehensive review would include heat engines such as the Wankel, Stirling and Brayton and high-, medium- and low-temperature fuel cells. However, for some years the Shell Research effort on fuel cells has been concentrated on the development of the low-temperature, direct methanol cell,

0378-7753/82/0000-0000/$02.75

with a view to automotive use in a market dominated by Otto and Diesel engines. Methanol was originally chosen as the fuel-cell fuel because of its better prospects *vis-à-vis* hydrocarbon fuels which are very inert electrochemically. Nowadays, interest in the methanol fuel cell is sustained by the recognition that in the future the cheapest and most widely available synthetic liquid fuel, from coal, natural gas or wood, will probably be methanol, and serious consideration is currently being given to the various problems involved in its large-scale production, distribution and use.

Of particular interest is the utilization of natural gas from remote oil producing regions where large amounts are flared off. At present it is more economic to ship natural gas in refrigerated liquid form, but concern with the hazards of spillage has refocussed attention on its conversion into methanol.

If methanol does become available in quantity and at prices competitive on an energy basis with hydrocarbon fuels, even a small advantage in efficiency provided by the methanol fuel cell over the heat engine may be sufficient to establish it a place in the market. If, on the other hand, methanol remains, as it is today, a relatively small volume, high priced chemical, the methanol fuel cell will never become competitive, except perhaps in specialized markets.

This paper discusses the thermal efficiencies of Otto and Diesel engines, present and future, using conventional fuels as well as methanol. Comparison is made with the low-temperature, direct methanol fuel cell both in stationary applications, where the load factor tends to be high, and in automotive applications where the load factor is generally low. It will be shown that the methanol fuel cell is likely to be competitive only in the automotive field.

2. Heat engines

2.1. Theoretical considerations

Heat engines of one kind or another at present represent virtually the only practical means of obtaining useful work from the chemical energy of fossil fuels. The conversion of heat into work is governed by the Laws of Thermodynamics, of which the Second Law shows that the maximum efficiency of a reversible engine is given by the relation:

$$\eta = 1 - \frac{T_1}{T_2} \qquad (1)$$

where T_1 and T_2 are the temperatures of the sink (normally ambient) and source, respectively. This is often described as the Carnot cycle efficiency.

The heat of combustion, ΔH, of hydrocarbon fuels in air is sufficient to raise the temperature of a stoichiometric fuel/air mixture at atmospheric pressure to some 2450 K [1]. A reversible heat engine, operating between 2450 K and normal temperature (300 K) would have a corresponding efficiency of 88%. A higher source temperature and a correspondingly higher efficiency would be obtained if oxygen were used instead of air.

The Second Law of Thermodynamics presupposes that the energy source is heat at a given temperature. Fossil fuel energy is not, *ab initio*, heat energy, but can, in principle, be released as heat to a body at any given temperature, raising the temperature of that body still further. Thereafter the second law of thermodynamics applies, but does not necessarily exercise a severe restraint. This point can be illustrated by reference to the theoretical thermodynamic cycles which govern the operations of Otto and Diesel engines.

The Otto, or constant volume cycle is depicted in Fig. 1. Fuel/air mixture is compressed adiabatically in step (a) to (b), the mixture is ignited (*i.e.*, heat added) in step (b) to (c), the gases are expanded adiabatically in step (c) to (d) and exhausted in step (d) to (a). The thermal efficiency, η, is given by the expression:

$$\eta = 1 - \left(\frac{1}{r}\right)^{\gamma - 1} \tag{2}$$

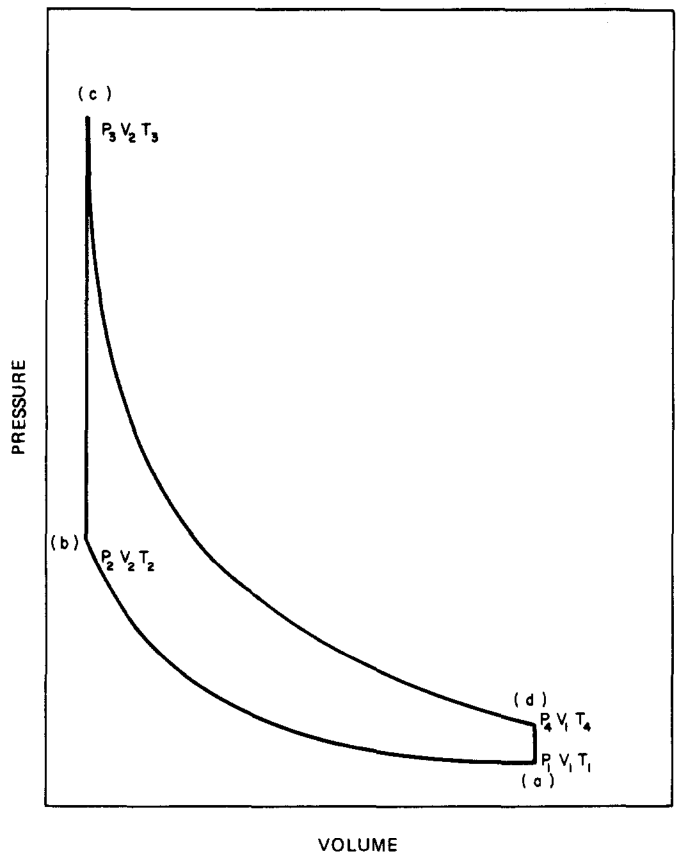


Fig. 1. Constant volume gas cycle. (Theoretical Otto engine.)

where r is the compression ratio V_1/V_2 and γ is the ratio of the specific heats of the working fluid at constant volume and constant pressure, C_V/C_P. Note that this expression does not explicitly involve temperature.

The Diesel or constant pressure cycle, *in its extreme form*, is depicted in Fig. 2. Air is compressed adiabatically in step (a) to (b), fuel is added and ignites spontaneously (heat added) during the constant pressure expansion step (b) to (c), and the spent gases are exhausted in step (c) to (a). The thermal efficiency of this cycle is given by the expression:

$$\eta = 1 - \left(\frac{1}{r}\right)^{\gamma-1} \frac{(r^\gamma - 1)}{\gamma(r-1)}. \tag{3}$$

Equations (2) and (3) are plotted in Fig. 3 both for ideal monatomic gases (γ = 1.667) and for air (γ = 1.402). The constant volume cycle can evidently be very efficient at high compression ratios, exceeding 95% at a compression ratio of 100. The constant pressure cycle is, however, far less efficient, approaching 40% asymptotically.

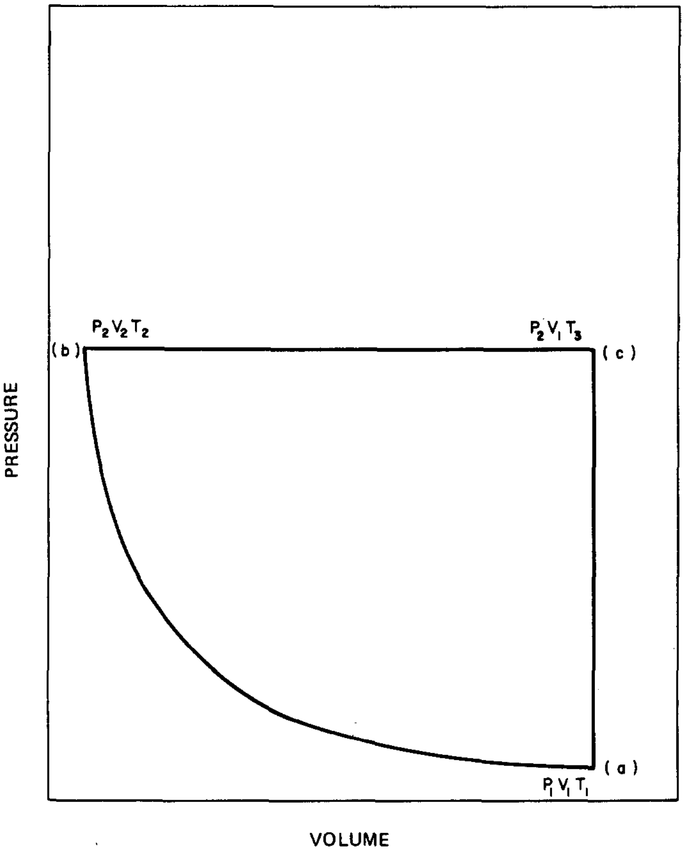


Fig. 2. Constant pressure gas cycle. (Theoretical Diesel engine.)

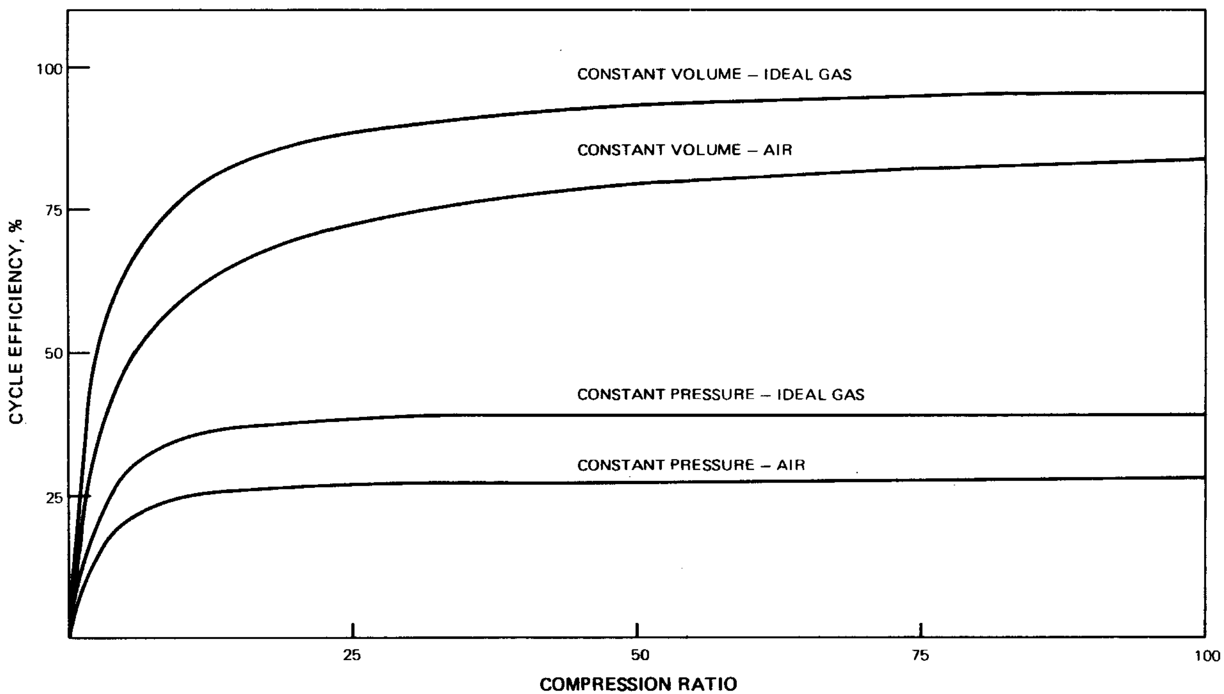


Fig. 3. Theoretical efficiency of constant volume and constant pressure gas cycles. Ideal gas: $\gamma = 1.667$, Air: $\gamma = 1.402$.

As regards the Otto cycle at least, it is clear that the theoretical limitation is of no real consequence. Practical limitations are all-important. In this respect heat engines are not at a disadvantage with respect to fuel cells which equally have an unattainable high theoretical efficiency.

2.2. Practical considerations

The idealized engine cycles depicted in Figs. 1 and 2 do not closely portray the behaviour of real Otto and Diesel engines. In both cases the actual cycle is something of a hybrid of the two, that is to say, each incorporates, in some measure, heat addition at both constant volume and constant pressure, and an adiabatic expansion stage. Such cycles, generally referred to as limited pressure cycles, are depicted in Fig. 4. Otto engine cycles approximate closer to constant volume cycles and Diesel engine cycles to constant pressure cycles but the differences, as observed on actual engine indicator diagrams, are not very striking. Diesel engines are designed to operate at high compression ratios which give rise, in constant volume cycles, to excessively high peak pressures. In order to limit the pressure rise, fuel is injected over a definite period of time such that after the design pressure is reached, further fuel (heat) is added at constant pressure.

The generalized expression for limited pressure cycles is

$$\eta = 1 - \left(\frac{1}{r}\right)^{\gamma-1} \left[\frac{\alpha\beta^{\gamma} - 1}{(\alpha - 1) + \alpha\gamma(\beta - 1)}\right] \tag{4}$$

where α is p_3/p_2 and β is V_3/V_2 (Fig. 4). Constant volume and constant pressure cycles represent extremes where $\beta = 1$ and $\alpha = 1$, respectively, Normally, the constant pressure cycle is considered to have values of β somewhat less than r (*i.e.*, $V_3/V_2 < V_4/V_2$). The smaller the value of β the more closely the efficiency of the limited pressure cycle approaches that of the constant volume cycle.

In practice, then, thermal efficiencies of Otto and Diesel cycles do not differ as much as might have been supposed. The Diesel engine, however, usually yields a higher thermal efficiency than the Otto engine, because it is designed to operate at higher compression ratios that more than compensate for its basically inferior cycle characteristics. The Diesel advantage becomes increasingly pronounced under part-load conditions, as will be discussed.

So far it has been assumed that the working fluid has the properties of an ideal monatomic gas ($\gamma = 1.667$). Air, however, which constitutes some 0.95 mole fraction of the stoichiometric fuel/air mixture is not an ideal gas; γ is 1.40 at room temperature and increases with temperature. The combustion products, moreover, contain large quantities of carbon dioxide and water (~0.25 mole fraction) the specific heats (and γ) of which are also lower and vary considerably with temperature. Other factors influencing thermodynamic efficiency include the change in the number of molecules present before and after combustion and the partial dissociation of combustion products at peak temperatures. The overall effect is that as fuel con-

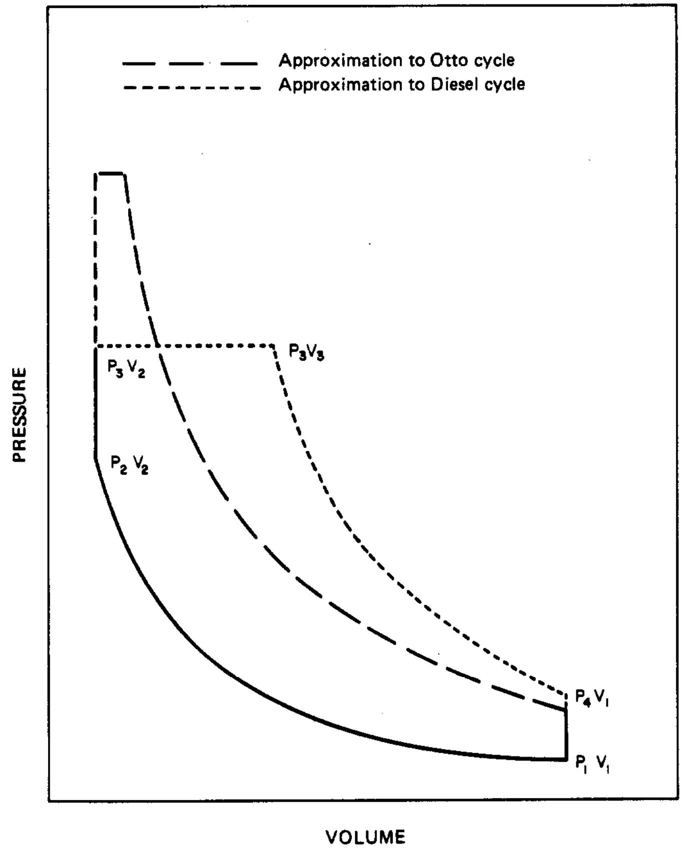


Fig. 4. Limited pressure gas cycles. (Practical Otto and Diesel engines.)

centration increases the cycle departs increasingly from ideality and efficiency falls. Rich mixtures are particularly disadvantageous in this respect. The calculation of efficiency based on strict thermodynamic principles is complex, but simpler, semi-empirical methods can be used [2].

Additional departures from ideality, resulting in a general rounding off of the engine cycle diagram, are due to mechanical features such as the finite rate of exhaust valve opening and piston movement during combustion, and possibly most important of all, heat lost to cylinder walls.

These factors combine to reduce the efficiency of the work done on the piston head. In addition, mechanical work losses further reduce overall engine efficiency. These losses are principally (1) friction losses in pistons and bearings and (2) work done in pumping gases into and out of the engine.

The relative importance of these two distinct components of mechanical work loss depends not only on the type of engine (Otto or Diesel) but on engine load. Whereas in the Otto engine the combined losses increase absolutely at reduced engine loads, in the Diesel engine they tend to decrease. However, in both cases, they increase in proportion to output as load is reduced, more so in the case of the Otto than in the Diesel engine.

Net engine output (brake horsepower) is given by the formula:

$$\text{b.h.p.} = 7.7 \times 10^{-5} \times \text{b.m.e.p.} \times CR \qquad (5)$$

where b.h.p. is brake horsepower, b.m.e.p. is the brake mean effective cylinder pressure in lb/in.^{2}, C is the engine capacity (in litres) and R the engine speed (in rev/min). B.m.e.p. is the true or 'indicated' mean effective cylinder pressure (i.m.e.p.) less mechanical (friction and pumping) pressure losses (f.m.e.p.), *viz.*

$$\text{b.m.e.p.} = \text{i.m.e.p.} - \text{f.m.e.p.} \qquad (6)$$

It is generally advantageous to operate at high i.m.e.p. because, in these circumstances, friction losses become proportionately smaller. However, at very high i.m.e.p. friction losses increase disproportionately, largely owing to increased piston ring pressures, so that brake thermal efficiency is comparatively poor at both low and very high i.m.e.p. At low engine speed heat losses to the cylinder walls increase; at high speed piston displacement during the combustion step increases (with adverse effects on cycle characteristics) and friction losses increase owing to the inertia of moving parts and to the viscous properties of lubricants. Consequently, for every engine there is an operating regime, defined by b.m.e.p. and engine speed, that provides maximum brake thermal efficiency. This is illustrated in Fig. 5 for a modern Otto engine. It can be seen that the most economic specific fuel consumption is achieved at close to, but not generally at, maximum b.m.e.p. and at an intermediate engine speed. This would correspond, in transportation use, to perhaps travelling uphill at wide-open throttle at a relatively low speed in top gear. It is not equivalent to full-power operation. This is achieved only at much higher engine speed such as might be attained during acceleration.

It should be noted that Diesel and Otto engines give qualitatively similar 'engine maps'. An important characteristic of these engines must be appreciated; it is possible to produce any required power output by variable combinations of b.m.e.p. and engine speed, within which specific fuel consumption will vary considerably. Part-load specific fuel consumption figures given without definition of running conditions are accordingly not very meaningful.

The Diesel engine achieves its renowned fuel economy *via* high compression ratios (commonly 14 - 20:1), and must be heavily built to withstand the accompanying stresses. It is, accordingly, a long-lived but expensive unit, having a relatively low power/weight ratio. By contrast, lower compression ratios (commonly 6 - 10:1) in the Otto engine permit lighter construction and higher engine speeds. Accordingly, it is a relatively short-lived but cheaper unit, with a higher power/weight ratio and inferior fuel economy. It is therefore reasonable to expect that the scope for efficiency improvements is greater in the case of the Otto engine than in that of the Diesel engine.

Pollution control legislation introduced in many developed countries is aimed chiefly at the transportation sector of the engine market. Engine

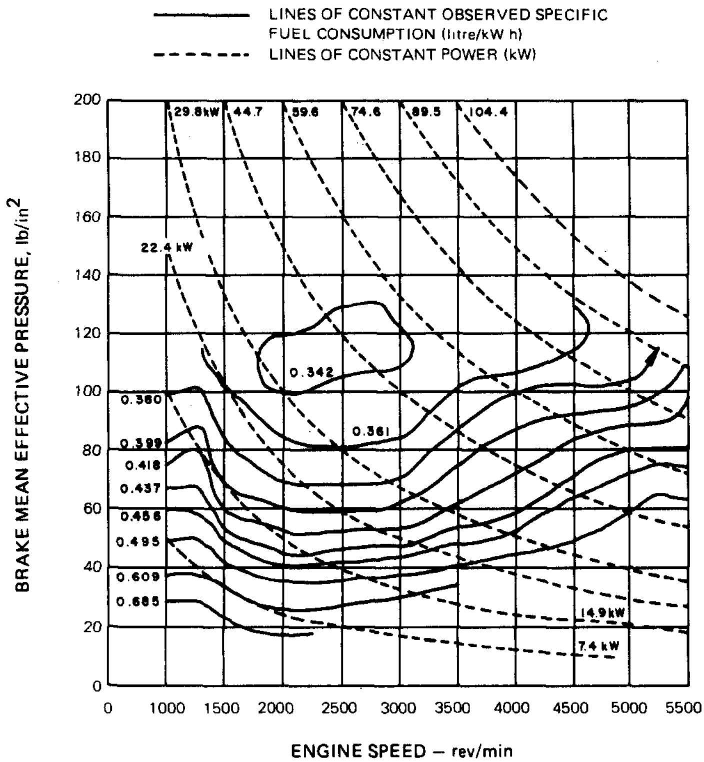


Fig. 5. Fuel consumption map for the BL 2600 gasoline engine.

requirements for pollution control and efficiency are usually in marked conflict with each other and in some countries a sacrifice in efficiency has already been accepted in order to meet exhaust standards. Legislation of this kind may conceivably not be aimed directly at stationary generating sets, but stationary engine design is likely to follow trends in automotive engine design because of the dominant position of the automotive engine in the marketplace. Pollution control is beyond the scope of this paper. It is discussed in detail in ref. 2.

2.3. The Otto (spark-ignition) engine

The Otto engine is normally characterized by two distinctive features: (1) aspiration of the fuel/air mixture and (2) spark-ignition. The combustion process accelerates as the flame front moves outwards from the point of ignition owing to further heating by compression of the unburned portion of the charge. A smooth, rapid pressure rise occurs with a relatively sharp peak.

The Otto engine is subject to two characteristic limitations:

(1) The compression ratio is restricted to about 10:1 by the octane rating of commercial fuels, currently available up to about 98RON (Research Octane Number) with added lead. The refinery cost of producing even higher

octane fuels is unlikely to prove economic and, with the emphasis on reduction of lead content, present octane ratings may not even be maintained [2].

(2) Flame temperature falls on either side of the stoichiometric fuel/air mixture, and the fall is particularly rapid on the lean side. Unduly slow combustion occurs giving rise to inferior cycle characteristics and, in the extreme, misfiring. Because of imperfect carburation and cylinder-to-cylinder variations, 85 - 90% of the stoichiometric quantity of fuel is generally regarded as the present-day practical lean limit.

Part-load operation is achieved by throttling whilst maintaining a more or less constant mixture strength. The effect of this on brake thermal efficiency is invariable adverse, as will be seen.

The theoretical efficiency of the basic engine cycle is not affected by variations in the quantity of either fuel or air supplied to the cylinder provided that the proportions remain constant. However, pumping friction losses, which are negligible at wide-open throttle, increase significantly as the pressure in the induction manifold decreases with decreasing throttle opening. Conversely, mechanical friction losses, largely due to piston ring pressure, decrease as the pressure in the cylinder decreases. Nevertheless, the net effect is that total friction losses increase somewhat at reduced throttle openings. Since the indicated mean effective pressure (i.m.e.p.) falls proportionately with throttle opening, friction losses (f.m.e.p.), as a fraction of i.m.e.p., increase very significantly. Thus, whereas at full-power mechanical losses (mainly piston friction) may reduce brake thermal efficiency to about 83% of the indicated thermal efficiency (*i.e.*, cycle efficiency), at 25% power mechanical losses (now including a significant proportion of pumping work) may reduce brake thermal efficiency to only some 50% of indicated thermal efficiency [2].

The above considerations assume constant speed, and, as already seen, at a given load both low and high speeds adversely affect specific fuel consumption. There is accordingly no direct relationship between load factor and fuel consumption and reference should always be made to engine maps.

Figure 5 represents a typical modern Otto engine. Minimum specific fuel consumption is obtained at near maximum b.m.e.p. (110 lbf/in.^{2}) in a medium speed range (2000 - 3000 rev/min) corresponding to about half (60 b.h.p.) the maximum power output (120 b.h.p.). The specific fuel consumption at maximum power, provided the maximum power point on the map is judiciously chosen, *viz.*, at about 125 b.m.e.p. and 4600 rev/min, can be seen to be not a great deal higher (5 - 6% more). Half-power (60 b.h.p.) can, however, also be obtained in unfavourable regions of the engine map (corresponding to low values of b.m.e.p. and high speeds) that give very much higher specific fuel consumption.

It is possible to select a path on an engine map such as Fig. 5, such that specific fuel consumption does not change appreciably from maximum power down to 25% of maximum. This is not generally a practicable proposition. In automotive applications with fixed ratio gear boxes, engine speed is strictly proportional to road speed whereas power requirement is not.

Widely variable and rapid changes of power outputs may be demanded at any given engine speed. For most stationary applications, however, and especially for electric power generation, a constant speed is normally required, although power requirements can still vary widely.

In order to provide a reserve of power for acceleration and hill climbing the automobile engine is designed to operate for most of the time at low power, in modes represented by the lower left-hand region of Fig. 5. Rapid increases of power (increases of b.m.e.p.) are available instantly by use of the throttle. The poor overall fuel economy of the Otto engine in transportation use is largely a result of this design feature, although frequent cold starts and transmission losses are important contributory factors.

Non-automotive engines have a wide range of application and generalizations of load factors have little significance. On the one hand, standby equipment having a very low (or even zero) utilization factor may be designed to operate at near 100% load factor when in use. On the other hand, a compressor on a construction site may operate at very high utilization but only rarely at a high load factor.

Otto engines for stationary applications are widely available in the range 200 W - 7.5 kW. They are mostly rather smaller than automotive engines and are generally less efficient although there is no fundamental reason why this should be so.

Whereas power is a function of cylinder size, heat loss is proportional to surface area. Other things being equal, multicylinder engines will be less efficient than single cylinder engines of the same gross capacity, and, similarly, the efficiency of single cylinder engines will decrease with decreasing cylinder size. However, the increased heat losses arising with small cylinders can be largely offset by operating at higher compression ratios, because the lower flame temperatures and reduced flame path lengths lessen the tendency to knock. The resulting increased i.m.e.p. can readily be accommodated in small cylinders where temperature gradient stresses are minimal, and although piston ring friction will increase at the higher pressures the net change in brake thermal efficiency need not be significant.

Small engines are subject to lower dynamic stresses than large engines and advantage is often taken of this to increase power output by running at higher speeds, with some sacrifice in efficiency. This is not, of course, an inevitable consequence of miniaturization, but rather reflects efforts to achieve power at low cost without the benefit of volume production on the automobile industry scale.

The poor efficiency of small Otto engines, in practice, can be ascribed to a large extent to poor carburation, which is a consequence of short manifolds giving little opportunity for good fuel/air mixing. In many cases it appears that an appreciable part of the fuel passes through the engine unevaporated and unburned. Whilst such shortcomings could doubtless be improved upon by increasingly sophisticated carburetter design, the higher manufacturing cost would not be considered acceptable.

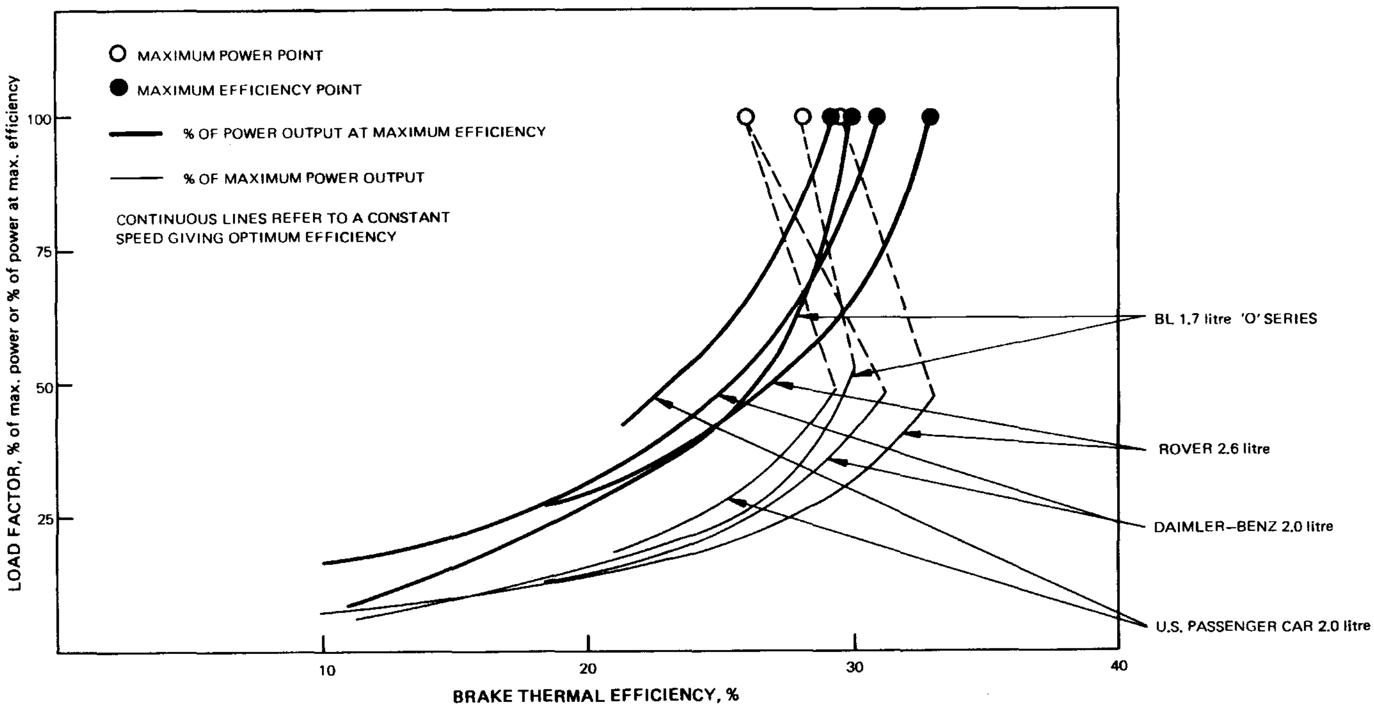


Fig. 6. Part-load efficiency of Otto engines.

The smallest Otto engine known to be commercially available is the 21.2 cm^3 Honda generator, delivering some 110 W at 3800 rev/min [3]. Fuel consumption is not known. Small stationary engines in the range 200 - 500 cm^3 have efficiencies commonly in the range 19 - 20%. By contrast, automobile engines of 1.5 - 2.5 l capacity have maximum efficiencies in the range 26 - 33% (Fig. 6).

Figure 6 compares the performance of some modern automotive Otto engines. The data have been taken from published engine maps and, for the present purpose, have been plotted on the basis of the constant speed that gives the best overall fuel economy for each engine.

The overall thermal efficiencies of commercial Otto engine generator sets in the 1 - 5 kW output range vary between 5 and 18% [4, 5]. These poor efficiencies can be ascribed in part to generator efficiencies (which are generally 70 - 80% in this size range) [4], and in part to poor matching of engine and generator such that the rated generator output is reached at low engine load factor. There is some evidence that the most efficient sets are well-

TABLE 1

Efficiency of the Otto (constant volume cycle) engine
Compression ratio, 9.0:1. Constant speed assumed.

Engine operation	Full power		Half power		Quarter power	
	Indicated thermal efficiency (%)	Brake thermal efficiency (%)	Indicated thermal efficiency (%)	Brake thermal efficiency (%)	Indicated thermal efficiency (%)	Brake thermal efficiency (%)
Theoretical ideal gas cycle (γ = 1.667)	77.0		77.0		77.0	
Theoretical air cycle (γ = 1.40)	58.5		58.5		58.5	
Theoretical stoichiometric fuel/air cycle (γ = 1.275)	45.4		45.4		45.4	
Cycle imperfections (exhaust losses, finite piston speed losses, heat losses* say 14 - 20% of total)	38.5		37		35	
Friction losses (piston, bearings, variable % of total), net efficiency		33		30		27
Pumping losses (variable % of total), net efficiency		32.5		26		18

*Heat losses are difficult to assess. However, because of the reduced cylinder charge at part loads (but the same peak temperatures) heat losses will be increased relative to full load.

matched; these tend to be amongst the largest, *i.e.*, ~7.5 kW. Otto engine generator sets are available in the range 0.5 - 7.5 kW net output with the most common size around 1.5 kW. It is true to say, therefore, that by far the majority of all Otto engine generator sets will run at the quite low efficiencies of 5 - 12%.

Table 1 summarizes the salient features governing the efficiency of Otto cycle engines and indicates what is possible from a good, modern high-compression engine in the automobile size range. A full-load brake thermal efficiency of some 32% can be obtained, reducing to some 26% at 50% of load. The engine depicted in Fig. 5 actually represents a slight improvement on these figures, although on the road the average brake thermal efficiency will generally be quite low, at, say, 16 - 18%, owing to a predominance of use at low load factors.

By contrast, the full-load efficiency of small, stationary Otto engines (<20%) is disappointing. The effects of small scale are not all disadvantageous; use of high compression ratios and moderate speeds can offset the inevitable increased heat and friction losses. It is evident that users of stationary Otto engines have paid little attention to fuel consumption hitherto; initial cost has been of major concern.

2.4. The Diesel (compression-ignition) engine

The Diesel engine is characterized by two distinctive features: (1) injection of a variable quantity of fuel into a full charge of compressed air and (2) compression ignition. After a slight delay combustion proceeds rapidly (giving rise to the characteristic Diesel knock), decelerating towards the end as the residual oxygen concentration decreases. Imperfect fuel/air mixing will give incomplete combustion at high fuel/air ratios and smoke regulations commonly limit mixture strength to about 85% of the stoichiometric quantity of fuel. The ability of fuels to ignite on compression is given in terms of the cetane number, typically 40 - 55 for commercial Diesel fuels. This is readily achieved with aliphatic hydrocarbons, and cetane requirements are not ordinarily restrictive in engine design. The long-term availability of fuels with an acceptable range of cetane numbers, however, does give cause for some concern.

Compression ratios are limited at the lower end of the scale by cold-starting requirements and at the upper end by increased friction losses. In addition there are design and manufacturing problems associated with the production of a relatively small combustion chamber which has to combine adequate clearances with good combustion characteristics. Engine strength, weight, and cost are also increased at high compression ratios. In practice, Diesel engines commonly operate at compression ratios from about 15:1 for direction injection (DI) engines and up to about 24:1 for indirect injection (IDI) engines. Highest compression ratios are generally found in the smallest IDI engines (such as that fitted to the VW Golf at 23.5:1) where it is easiest to accommodate the higher stresses. Large commercial DI engines usually have compression ratios of around 15 or 16:1.

Limited pressure cycle plots show that efficiency does not improve very much at higher compression ratios (>20:1) although (because the ratio i.m.e.p./peak pressure increases slightly) some extra power can be obtained for a given engine strength and size.

Part-load operation is achieved by reducing the amount of fuel injected into the cylinder whilst maintaining a full charge of air. This has three significant consequences: (1) the flame burns more briefly, increasing cycle efficiency (because it approximates more closely to the constant volume cycle), (2) flame temperatures are lower so that heat losses diminish (although remaining approximately constant as a proportion of the heat input), and (3) there is no appreciable change in pumping work losses (which therefore remain negligible at all loads). These characteristics account for the renowned fuel economy of Diesel engines at part loads. At full load, there is little to choose between Diesel and Otto engines as regards efficiency.

Heat losses to the cylinder walls of a Diesel engine amount to a rather larger proportion of the heat input than is the case with the Otto engine, because of increased charge density and increased mixture swirl. They are also the more serious because they represent losses that would otherwise have been converted into work at somewhat higher efficiencies.

Friction losses comprise pumping friction work (1 or 2%) and mechanical friction work. Mechanical friction losses tend to be high because of high piston ring pressures, but remain more or less constant at all loads because cylinder pressures remain more or less constant. Consequently, friction losses increase proportionately as load decreases. The heavy construction of moving components in the Diesel engine enhances friction losses at high speeds through increased dynamic stresses. The Diesel engine is therefore most efficient at relatively low speeds, and is designed primarily to work at such.

The efficiency of small Diesel engines is, in general, worse than that of large engines. Increased friction losses contribute to this effect but not as much as might be supposed because piston ring friction (the main factor) is theoretically independent of cylinder size. Low-cost design is often a contributory cause of high friction losses. A significant factor affecting efficiency is heat loss which increases with cylinder surface/volume ratio. The reduced thermal efficiency is, moreover, accompanied by increased viscous friction losses due to the reduced cylinder wall temperature.

Small, high speed (up to 4500 rpm) Diesel engines designed primarily for automotive use are normally provided with indirect combustion chambers. Since they run at higher speeds than do larger engines there is less time available for the injected fuel to mix with air. To compensate for this, the air in the cylinder is compressed into a small subsidiary chamber and given a strong swirling motion so that the fuel spray is rapidly carried away by the moving air as it is injected, resulting in far better mixing. Power output is increased at the higher speeds thus made possible, but unfortunately efficiency suffers, largely because of the higher heat losses occasioned by the increased surface area of the complex combustion chamber.

Efficiency does not, on the other hand, increase indefinitely with size. To increase durability and reliability, very large engines tend to be conservatively rated, that is to say, they operate at lower temperatures, at lower peak pressures, at lower rates of pressure rise and at lower i.m.e.p. Indicated thermal efficiency and brake thermal efficiency both suffer, offsetting the advantageous lower heat losses.

The lower limit to cylinder size is set by practical considerations. It is difficult to manufacture very small combustion chambers that retain good combustion characteristics with adequate clearances. Difficulties also arise in the manufacture of small yet accurate fuel injector pumps. Fuel quality becomes of increasing importance, and high heat losses increase starting difficulties. Such considerations set the practical lower limit at about 200 cm^3 capacity, yielding some 2 - 2.5 kW output. (By contrast, gasoline engines are commonly built down to about 50 cm^3, delivering some 0.5 - 1.0 kW or so.) Diesel engine generator sets are manufactured with outputs down to 2.5 kW, the most common size being 8 kW.

Diesel engine characteristics are summarized in Table 2. Efficiency does not decrease very much from full to half load, largely because the improvement in indicated thermal efficiency (due to a closer approximation to the constant volume cycle) is offset by proportional increases in friction losses. At lower loadings the gain in theoretical cycle efficiency becomes rather small and no longer adequately offsets the relatively increasing friction losses. The data in Table 2 adequately depict the main features that determine Diesel engine efficiency, though the correspondence with any particular commercial engine performance is not close. The reason is that there are many design variables (especially as regards the combustion chamber) that affect efficiency, and engines differ very much from each other, none being really representative.

Figure 7 illustrates the performance of some modern automotive Diesel engines. As with the Otto engines in Fig. 6, the data have been plotted on the basis of the most economic constant speed for each engine. On this basis, the difference in part-load efficiency between Otto engine (Fig. 6) and Diesel engine (Fig. 7), though real, is not very striking, although a larger sample might perhaps show greater differences.

Engine maps, similar to those applicable to Otto engines (Fig. 5), should be used to give an accurate picture of Diesel engine performance at varying loads and speeds.

Small automotive IDI Diesel engines of about 2 l capacity (suitable for commercial vehicles, taxis, etc.) have full-load brake thermal efficiencies in the range 27 - 33%. Large DI truck engines of 10 - 20 l capacity have significantly higher full-load brake thermal efficiencies, in the region of 36 - 42%. Engines used on generator sets in the range 250 - 1000 cm^3 have brake thermal efficiencies in the range 27 - 36%, similar to those of automotive 2 l engines, indicating that decreasing size down from 2 l does not have as large an effect on efficiency as one might expect, though the smallest units do tend to have the lowest efficiencies. Complete Diesel engine generator sets have net efficiencies in the region of 20 - 25%. With generator efficiencies of 70 - 85%, this suggests that engine and generator matching is rather better than is generally the case with Otto engine sets.

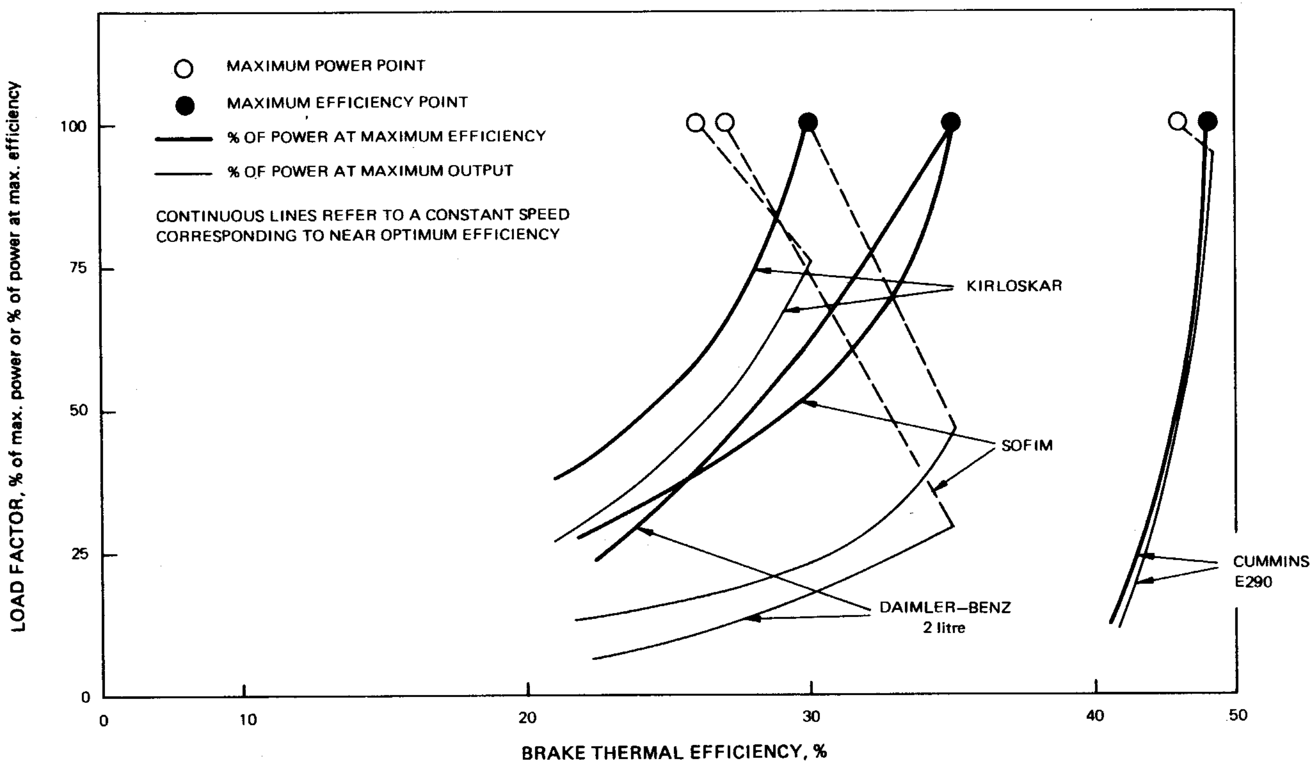


Fig. 7. Part-load efficiency of Diesel engines.

TABLE 2

Efficiency of the Diesel (limited pressure cycle) engine
Compression ratio, 18:1, $\beta = 1.514$, $\alpha = 2$. Constant speed assumed.

Engine operation	Full power		Half power		Quarter power	
	Indicated thermal efficiency (%)	Brake thermal efficiency (%)	Indicated thermal efficiency (%)	Brake thermal efficiency (%)	Indicated thermal efficiency (%)	Brake thermal efficiency (%)
Theoretical ideal gas cycle ($\gamma = 1.667$)	83.4					
Theoretical air cycle ($\gamma = 1.40$)	66.8					
Theoretical stoichiometric fuel/air cycle ($\gamma = 1.275$)	53.2					
Theoretical 85% stoichiometric fuel/air ratio cycle ($\gamma = 1.285$)	54.5					
Theoretical 43.5% stoichiometric fuel/air ratio cycle ($\gamma = 1.324$)			59.1			
Theoretical 28.0% stoichiometric fuel/air ratio cycle ($\gamma = 1.345$)					61.4	
Departure from ideal cycle (prolonged flame, early exhaust opening), net efficiency	46.3		53.2		55.3	
Heat losses (constant 6.5% of heat input), net efficiency		40		47		49
Mechanical losses (constant but increasing in proportion at decreasing loads), net efficiency		33		31		24

2.5. Comparison of Otto and Diesel engines

Although the Diesel engine is invariably regarded as more efficient than the Otto engine the comparison is seldom made under strictly comparable conditions, *i.e.*, using engines of similar power output over similar driving cycles. To produce the same rated power output as a gasoline engine a larger capacity Diesel engine operating at lower b.m.e.p. is required. Fuel consumption data [5, 6] for two such engines having the same rated output and similar power/speed characteristics have been replotted in Fig. 8 to show how the engine efficiencies compare at equal load factors. High load factors

(>75%) can be achieved only at high engine speeds (say >3000 rev/min) and in these circumstances the Otto engine is actually the more efficient. At medium load factors (~50%) the Otto engine is the more efficient at high speeds and the Diesel at low speeds. Low load factors (<25%) can be produced only at relatively low speeds (<2500 rev/min) and here the Diesel is unquestionably more efficient than the Otto engine.

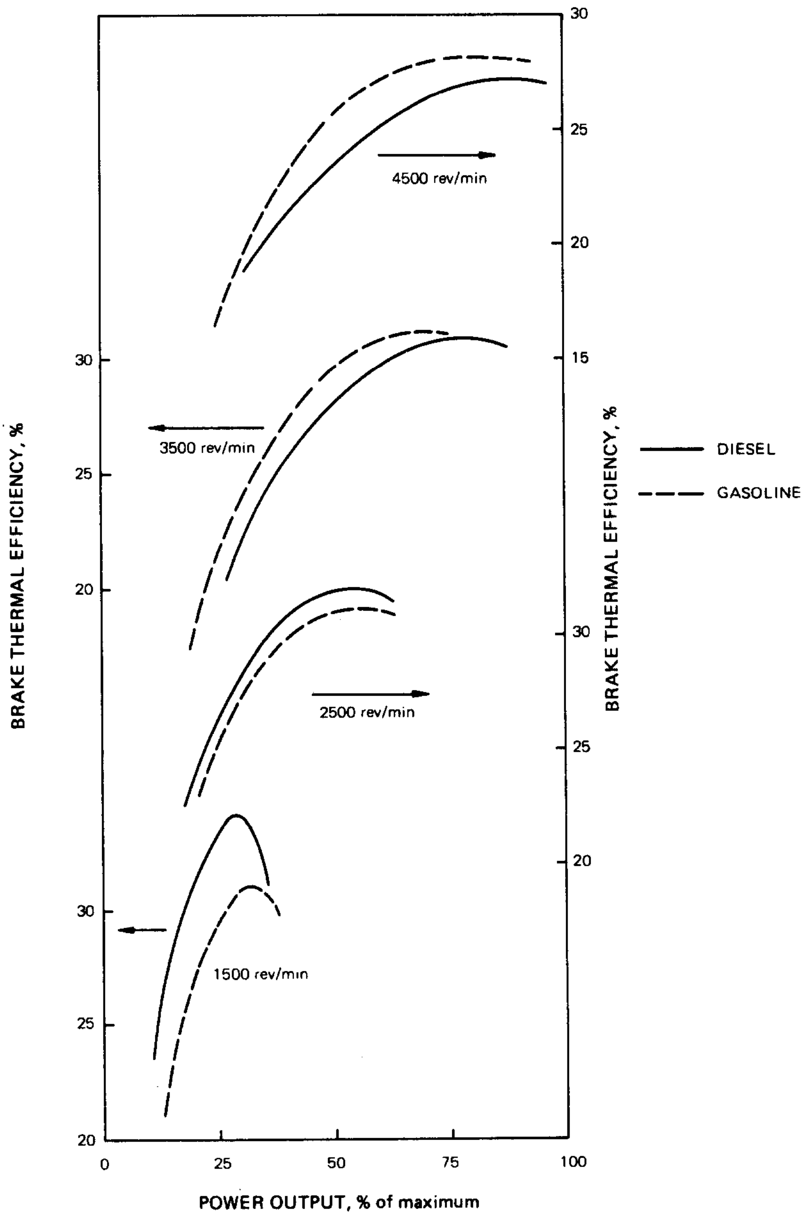


Fig. 8. Brake thermal efficiency of equivalent Diesel and Otto engines (data from ref. 5).

Table 3 gives a similar comparison at part loads at, or close to, the most efficient speed range as appropriate. Efficiency hardly varies from maximum load down to 50% load, but as the load is further reduced the efficiencies of both engines fall, that of the Otto more so than that of the Diesel. Essentially, the advantage of the Diesel in transportation is due to preponderance of part-load operations. In general, the Diesel is claimed to have an advantage of 20 - 30% fuel economy over the Otto engine in automotive use. This economy advantage, viewed against the figures in the last column of Table 3, may well be ascribed to the fact that Diesel-engined vehicles are generally underpowered by comparison with Otto-engined vehicles and in consequence are used at higher load factors. Volumetrically, Diesel fuel contains up to 10% more energy than does gasoline, giving the Diesel vehicle an illusory further advantage when efficiency is related to miles (kilometres) per gallon (litre).

TABLE 3

Brake thermal efficiency of equivalent Otto and Diesel engines under part-load conditions 2.11 l Diesel : 1.62 l Otto. Data from ref. 5.

Load factor, % of maximum output	Speed, rev/min	Brake thermal efficiency (%)		Advantage of Diesel over Otto (%)
		Otto	Diesel	
75	3500	31.0	31.0	0
50	2500	31.0	32.0	5
25	1500	29.5	33.5	14
20	1500	27.5	31.5	15
15	1500	24.0	28.5	19

2.6. Future developments in heat engines

As we have seen, by comparison with the Otto engine the Diesel engine is relatively efficient at part-load, has a lower power/weight ratio, and is more expensive. Efficiency has always been of prime concern during its development, which is now at a mature stage. Without doubt further efficiency improvements will be obtained by paying close attention to factors such as friction losses and by the use of light-weight structural components. Nobody appears to expect that these improvements will be other than evolutionary in kind though they may contribute a useful overall gain in efficiency.

By contrast, the Otto engine has been developed for high power and low cost with some sacrifice in efficiency. Accordingly, this engine might be expected to show the greatest efficiency gains in the next decade or so, and it is perhaps natural that attempts should be made to provide it with the particular characteristic that, in the Diesel engine, yields good efficiency. This is essentially the ability to burn lean mixtures, permitting power modulation to be achieved by variation in mixture strength.

The Diesel engine is a stratified-charge engine, *i.e.*, the mixture is inhomogeneous and burns outwardly from fuel-rich regions into leaner regions. Several stratified-charge Otto engines with similar combustion properties have been developed. Fuel injection, which may be IDI or DI, is normally used. In the indirect system a rich fuel mixture is ignited in an auxiliary combustion chamber, from where the flame spreads into leaner mixture regions in the main chamber. The Honda CVCC engine [6] is of this kind (although it actually uses two carburettors to provide rich mixture to the auxiliary chamber and lean mixture to the main chamber). The direct system achieves a similar effect by judicious shaping of a single chamber, often by providing a well in the piston head, and by a precise alignment of injector nozzle and spark plug. The Ford PROCO is a good example.

In both forms of stratified-charge engine, the aim is to achieve a high degree of swirl to produce rapid combustion without losing the favourable characteristics of the Otto cycle (low noise, high power, and ease of starting). The indirect form suffers, by comparison with the direct form, from higher heat losses and from fluid-friction losses in the hot gases, these effects being associated with the more complex combustion chamber shape. On balance, the direct form may be considered most likely to prove the more efficient in the long run.

The operation of the stratified-charge engine depends on the precise control of conditions in the combustion chamber over a wide range of loads and speeds. The design difficulties remain considerable and success is by no means yet assured.

Such engines are virtually hybrid Otto/Diesel engines and are capable of using wide-cut fuels which, as a general rule, can be produced in the refinery more cheaply and efficiently than high-octane fuels. However, this may or may not prove to be advantageous to the stratified-charge engine since the refiner aims at meeting the requirements of all liquid fuel markets in the most cost-effective way, and this could mean the reservation of the cheap, wide-cut fraction for the aviation market and the supplementation of the road transportation market with aromatic synthetics or alcohols.

As an alternative to stratified-charge engines, homogeneous-charge, 'lean-burn' engines are being developed using, as a rule, a carburetted fuel supply. The extent to which Otto engines can be run lean is limited in practice by cyclic irregularities, bearing in mind the allowances that must be made to avoid the occurrence of knock and misfiring under the most unfavourable circumstances. Some improvement in this respect can be made by better mixture control and electronic ignition timing, especially if these are computer-controlled *via* suitable engine signals. More dramatic improvement can be achieved by combustion chamber and intake port design by means of which a high degree of turbulence is produced [7]. The May Fireball cylinder head exemplifies this principle [8]. Fuel is compressed under the inlet valve and then forced into a deep recess below the exhaust valve where it acquires a violent swirling motion and where spark ignition occurs.

Rapid combustion takes place, but knock is eliminated because of the relatively short flame path and because of the quenching effect on the end gas of the large inlet valve in the main part of the combustion chamber.

Lean-burn implies lower flame temperatures and, hence, reduced susceptibility to knock. In consequence the compression ratio can be raised without a corresponding increase in octane requirement. This enhances efficiency and partly restores the power inevitably lost as a consequence of the reduced fuel intake.

Generally speaking, lean-burn, high compression ratio engines will operate satisfactorily at compression ratios in the region of 12 - 14:1 and with fuel/air ratios down to 75% of stoichiometric. At 14:1 compression ratio such an engine will have a theoretical fuel/air cycle efficiency of ~53% [9]. Figure 9, taken from a recent publication [10], shows that the engine will deliver approximately 10% less maximum power than would a similar capacity, modern 9:1 compression ratio engine operating on a stoichiometric fuel/air mixture (1:15 ratio) at a lower theoretical efficiency of 44%.

There are no good reasons to suppose that exhaust losses and finite piston speed losses in lean-burn engines will be noticeably different from those of engines of today, but heat losses should be a little less because of the lower flame temperatures. Thus, whereas in Table 1, 14 - 20% heat losses were assumed we shall arbitrarily take 10 - 16% for the lean-burn engine.

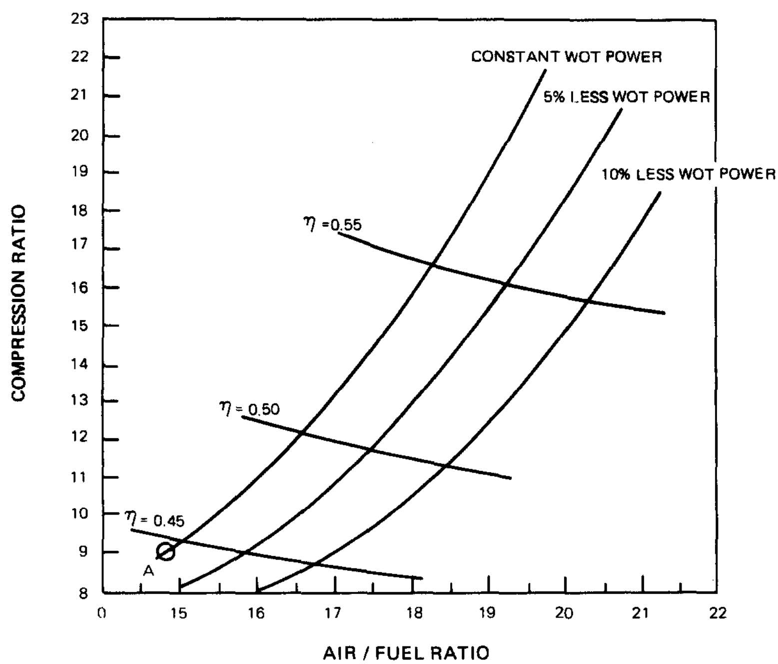


Fig. 9. Dependence of wide open throttle power and efficiency on compression ratio and mixture strength for a constant volume fuel/air cycle with iso-octane fuel. Point A is the operating condition of a typical current engine.

Pumping losses will remain similar. Some future reduction in friction losses can be expected through the use of lower viscosity lubricants and by the use of friction-reducing additives. It has been suggested [10] that this will result in a 10% overall improvement in short trip efficiency, but since friction losses increase relatively as power output decreases it is difficult to put this estimate in perspective. We shall arbitrarily assume friction losses can be reduced in absolute terms to 75% of current levels. Making these assumptions, Table 2 can be recompiled to give an estimate of the likely full- and part-load efficiency of the future carburetted high-compression, lean-burn gasoline engine (Table 4). This suggests an improvement in efficiency of some 30% above that now obtainable from Otto engines (Table 1) over the full range of power output, a little less than, though not out of line with, the improvements predicted elsewhere [10]. The estimate also represents an improvement over present day small IDI automotive Diesel engine performance, especially at full power, but does not match larger DI Diesel engine performance at part load. If the estimated performance for the high-compression, lean-burn engine can be reached, it must be assumed that small Diesel engines will have to do even better if they are to succeed in the market.

No estimate can be made of the likely part-load performance of stratified-charge, fuel-injection, lean-burn engines since it is not clear how combustion will be affected by the excessively lean mixtures appropriate at reduced power output. Presumably a performance at least as good as that of the high-compression, lean-burn engine must be obtained if such engines are to succeed.

TABLE 4

Potential efficiency of the Otto carburetted high-compression lean-burn engine
Compression ratio 14:1. Constant speed assumed.

Engine operation	Full power		Half power		Quarter power	
	Indicated thermal efficiency (%)	Brake thermal efficiency (%)	Indicated thermal efficiency (%)	Brake thermal efficiency (%)	Indicated thermal efficiency (%)	Brake thermal efficiency (%)
Theoretical 75% stoichiometric fuel/air cycle ($\gamma = 1.296$)	54.2		54.2		54.2	
Cycle imperfections (exhaust losses, piston speed losses, heat losses, say 10 - 16% of total)	48.8		47.2		45.5	
Friction losses (piston, bearings, variable % of total), net efficiency		45		41		36
Pumping losses (variable % of total), net efficiency		42		34		24

The disparity between present day efficiencies on the road and actual engine efficiency can be ascribed, to some extent, to inevitable inefficiencies in cold starting and idling, but probably in the main to the provision of high but little utilized power, and to the use of fixed-ratio gear boxes. Possibly the motorist of the future can be persuaded by high fuel prices to accept some sacrifice in power for the sake of improved fuel economy. From the technical point of view the use of an infinitely variable transmission coupled with microcomputer control permitting the engine to remain permanently in the most efficient high b.m.e.p./moderate speed regime would produce, on the road, the efficiencies indicated by Tables 1, 2 and 4 less, of course, transmission losses. This concept has been appreciated for years but the ideal transmission has proved elusive for both technical and economic reasons. Developments continue to take place, however, and a recent version of the DAF Variomatic belt-drive system, with a claimed 98 - 99% efficiency is said to be close to success [11]. Its use would appear to be limited to cars and small commercial vehicles because of its restricted gear ratio range.

These concepts are equally valid for stationary applications and transportation. Thus, there is some limited scope for efficiency improvements as regards Diesel generating sets, but much wider scope as regards Otto generators. One might expect to find well-matched (small) Diesel generator sets providing an overall 25 - 30% full-load efficiency. If it is assumed that the present disparity of some 30% or so in efficiency between small and large Otto engines can be diminished by better design to say 15%, one might expect 200 - 500 cm^3 capacity lean-burn, high compression Otto engines to have full-load efficiencies of about 34% with net electrical output in the range 24 - 27%, even in the absence of any significant improvement in generator performance (already 70 - 80%). With both engines, the provision of an infinitely variable gear box would contribute significantly to the improvement of part-load efficiency. Altogether, it seems not unreasonable to suggest a possible improvement in generating efficiency approaching double that which is obtainable from the best Otto generators available today.

2.7. Methanol as a fuel for heat engines

The physical and chemical properties of methanol that are of relevance to its use as a fuel in heat engines are shown, together with the corresponding properties of diesel and gasoline fuels, in Table 5.

Salient features [1] are:

(1) Methanol has a very low cetane number, far too low for it to be considered as a practical fuel in conventional Diesel engines, except perhaps as a component in a fuel blend.

(2) It has a high Research Octane Number and is therefore intrinsically suitable for use in Otto engines with high compression ratios (*e.g.*, 12 - 14) and correspondingly high thermal efficiencies.

(3) It has a relatively low calorific value (about 50% of that of gasoline and a correspondingly low stoichiometric air/fuel ratio. Thus the volumetric

fuel flow needs to be about twice that of gasoline in similar size engines.

(4) In contrast to gasoline under comparable conditions, combustion takes place at a lower temperature and at a higher velocity. The flame is non-luminous. Consequently the engine can be run under lean-burn conditions, the misfire limit being reached at about 66% of stoichiometric fuel/air ratio [12]. The combination of these factors leads to a considerable reduction of heat loss and an improvement in indicated thermal efficiency.

(5) It has a high latent heat of evaporation. This has a marked cooling effect on the temperature of the fuel/air mixture, which is thermodynamically advantageous but gives rise to cold-starting problems with carburettor systems; they will not generally produce an ignitable methanol/air mixture at temperatures below 5 °C.

(6) On the whole, methanol appears to present few problems with regards to emissions. Lower temperatures and lean-burn minimize NO_x emissions; there are no particulates, lead or sulphur in the exhaust, while CO emissions appear to be comparable with those from gasoline-fuelled engines. On the other hand, aldehyde emissions and the generally toxic nature of methanol do give cause for some concern [13].

Methanol is soluble in water and somewhat corrosive and for these reasons, as well as its low volumetric calorific value and high latent heat, it is quite unsuitable for use except as a blending component in unmodified

TABLE 5

Some physical and chemical properties of methanol, diesel fuel and gasoline

Properties	Methanol	Diesel fuel	Gasoline
Formula	CH_3OH	Mixture of C_{15} - C_{25} hydrocarbons	Mixture of C_4 - C_{12} hydrocarbons
Molecular weight	32.04	200 - 350	60 - 170
Oxygen (wt.%)	49.9	0	0
Stoichiometric air/fuel ratio	6.45	14.7	14.7
Density (g/ml, 15 °C)	0.796	0.82 - 0.92	0.71 - 0.78
Boiling point (°C)	64	180 - 340	27 - 225
Flash point (°C)	11	60	—43
Vapour pressure (mbar @ 38 °C)	320		475 - 1050
Flammability limits (vol %)	6.7 - 36.0		1.4 - 7.6
Calorific value (MJ/kg @ 20 °C)	19.7	41.4	42.8
(MJ/l @ 20 °C)	15.6	34.0	32.1
Latent heat of vaporization (MJ/kg)	1.0	0.19	0.31
Solubility in water	Infinite	Nil	Nil
Research octane number	115 - 145**		98*
Motor octane number	87 - 95		88*
Cetane number	3	45 - 55	

*Typical octane numbers for European premium gasoline.

**Blending octane numbers. Value depends on octane number and chemical composition of base gasoline.

present-day engines. Although not ideal it is an acceptable fuel for use in a purpose-built engine, *viz.*, a high-compression, lean-burn Otto engine or perhaps an ignition-assisted Diesel engine.

How efficient will such an engine be? Although the lean misfire limit is said to be 67% of the stoichiometric fuel/air ratio the most effective operating condition appears to be at 77% of stoichiometry at a compression ratio of 13:1 [14].

Computation of the theoretical fuel/air cycle efficiency from the thermodynamic properties of the working fluid has been done for hydrocarbons [15] but not, so far as is known, for methanol.

Deviations from simple theory are mostly due to the composition of the exhaust gases, which differs considerably from air and contains large quantities of water and carbon dioxide [16]. Different fuels produce different proportions of carbon dioxide and water but their thermodynamic effects tend to be counteractive and hydrocarbon fuels all yield similar theoretical cycle efficiencies. The ratio of water to carbon dioxide produced by the combustion of methanol (2:1) is very much higher than that produced by liquid hydrocarbons (from 0.5 to 1.25:1) but in the absence of contrary evidence it will be assumed that the cycle efficiency will not differ too much from that of hydrocarbons. In this case the carburetted high-compression, lean-burn engine will give similar efficiencies with hydrocarbon and methanol fuels and Table 4 will apply equally to both.

However, the weight of evidence seems to suggest that methanol can be burned leaner than gasoline, so that it may prove to have an efficiency advantage over gasoline under the all important part-load condition. It will also produce more power from a given engine size (due to the cooling effect of the high latent heat of evaporation on the charge drawn into the cylinder). The disadvantages of methanol are principally in potential cold starting and handling problems. Nevertheless, its future as an automotive fuel will, in the end, depend on political and economic rather than on technical factors.

2.8. Ethanol as a fuel for heat engines

The use of ethanol as a fuel is outside the scope of this paper but in view of topical interest a few comments will be made.

Thermodynamically and chemically the properties of ethanol differ only marginally from those of methanol. Thus it has, similarly, a high octane number and latent heat of evaporation and a low cetane number and calorific value. It is also miscible with water. It therefore presents corresponding advantages and disadvantages although, in general, the problems associated with the use of ethanol are less acute.

Ethanol is principally of interest to those countries that have surplus land and a suitable climate for growing fermentable crops such as sugar cane or cereals, and are prepared to support uneconomical indigenous production for political purposes, *e.g.*, the conservation of foreign exchange. On the world market ethanol is unlikely to become available in quantity at prices that would make it competitive with methanol.

3. Fuel cells

3.1. Theoretical

The fuel cell differs fundamentally from heat engines in that the work it produces is electrical and not mechanical. This may or may not be advantageous depending on the form of work output required, since conversion of one to the other invariably involves a 5 - 20% loss.

Heat engines utilize the heat of combustion of the fuel whereas fuel cells utilize the free energy of combustion; in either case the state of the fuel and its oxidation products (whether liquid or gaseous) has to be taken into account. It is usual to assume heat engines emit water as vapour, and the corresponding heat energy is known as the Lower Heating Value (LHV). Self-consistent data have been used to compile Table 6 which shows that in the case of methanol the free energy of combustion is less than the Upper Heating Value (UHV) but is slightly greater than the LHV by some 5%. The fuel cell starts, then, with a slight advantage over the (practical) heat engine.

The efficiency of fuel cells is given by the relationship:

$$\eta = \frac{Vi}{V^0 i_t} \tag{7}$$

where V and V^0 are the actual and theoretical cell voltages and i and i_t are the actual and theoretical current densities.

Current efficiency (i/i_t) is usually high, >90%. In principle, current efficiency losses are attributable to incomplete or side reactions but, in practice, physical loss of fuel is generally more important. Hydrogen, for example, is deliberately vented to remove accumulations of deleterious impurities; methanol is unavoidably vented with exhaust air. Good cell design will minimize these losses.

Voltage efficiency (V/V^0) represents the most serious loss factor and is the focal point of most fuel cell research. The voltage of a fuel cell is a combination of three independent components; the potential at the anode, the potential at the cathode and the voltage drop in the electrolyte (plus resistive losses elsewhere).

TABLE 6

Thermodynamic properties of fuels at 25 °C

Fuel	State	Upper heating value (MJ/kg)	Lower heating value (MJ/kg)	Free energy of oxidation (MJ/kg)
Iso-octane	Liquid	48.2	45.3	
Methanol	Liquid	22.7	20.4	
Methanol	Aqueous, 1 mol/l			21.44

Standard cell voltages (V^0) are directly related to the Standard Free Energy, ΔG^0, of the process by the relationship:

$$-\Delta G^0 = nFV^0 \tag{8}$$

which, by appropriate choice of parameters, applies equally to the half cell reaction (*i.e.*, the electrode reaction) as to the complete cell. Actual electrode potentials are related to standard electrode potentials (E^0) by a polarization term η_v such that:

$$\text{Electrode potential} = E^0 - \eta_v. \tag{9}$$

η_v measures the departure of the electrode process from ideality and must be kept as small as possible for optimum efficiency. It is a complex function of current density. A theoretical but undoubtedly oversimplified relationship is:

$$\eta_v = -\frac{RT}{\alpha nF} \ln \frac{i}{i_o} \times \frac{i_l}{(i_l - i)} \tag{10}$$

where i_o is the exchange current density, i_l the limiting current density and the other symbols have their usual significance. Over a fairly wide range of practical current densities eqn. (10) approximates to the Tafel equation:

$$\eta_v = a + b \log i \tag{11}$$

where a and b are constants.

At an efficient fuel cell electrode i_o will be large, achieved by good catalysis, as will i_l, achieved by a combination of a high fuel (or oxidant) concentration and by a high degree of turbulence.

The voltage drop in the electrolyte, η_r, is given by the relationship:

$$\eta_r = iR \tag{12}$$

where R is the resistance of the electrolyte per unit area of electrode. R depends on the specific resistance of the electrolyte and its thickness. Electrolyte thickness is minimized by close spacing of the electrodes.

The net cell voltage is the algebraic combination of the two independent standard electrode potentials (*i.e.*, fuel and oxidant) together with their respective polarization terms and the resistive term, each being temperature dependent. Generally, resistive and polarization terms decrease with temperature (not always, because of contributory factors such as the solubility of oxygen in the electrolyte which falls with increasing temperature) whilst the standard free energy to which electrode potentials are related may increase or decrease depending on the fuel (Fig. 10) [17]. The net result is that the voltage efficiency of the cell always increases with temperature, but the advantage can, in some cases, be offset by the diminishing ΔG^0 term and the lower cell voltage that is implied. Fortunately this is not so with methanol, although the advantage of high temperature in this case is offset by a host of practical considerations (catalyst stability, fuel stability, pressurization).

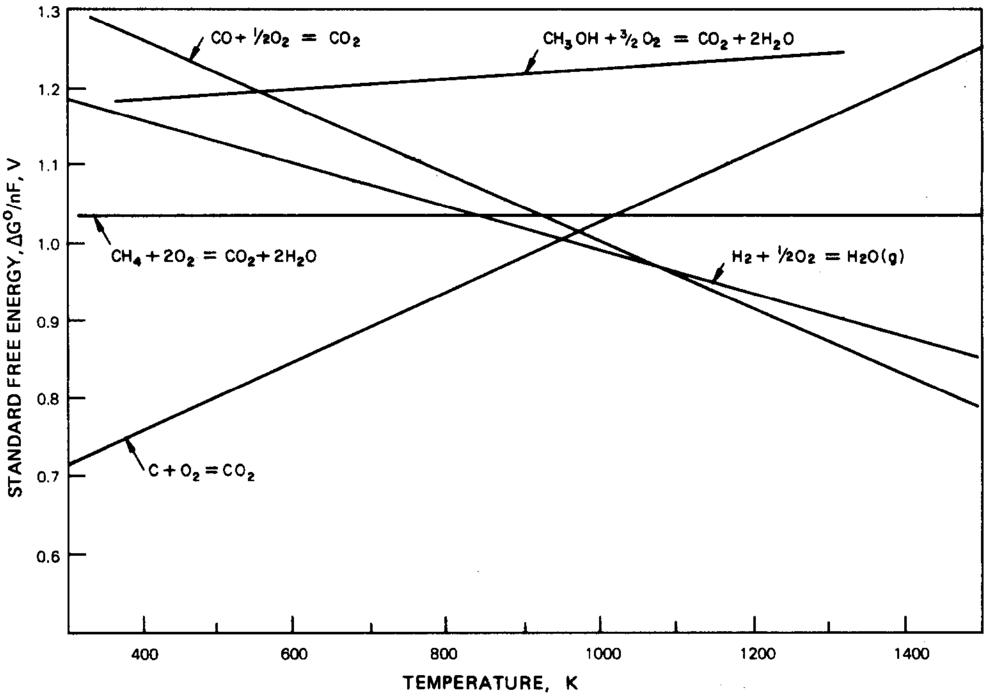
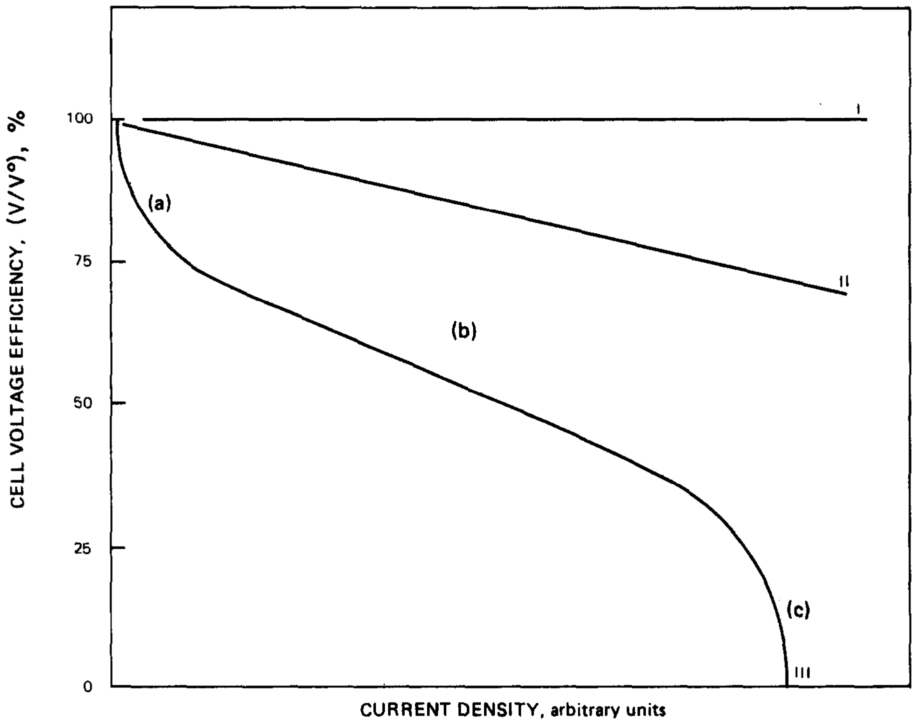


Fig. 10. Variation of free energies of combustion with temperature.

Generalized fuel cell characteristics are given in Fig. 11. At high current efficiencies voltage efficiency gives a close approximation to overall efficiency. At low current densities, the main loss factor is due to 'activation polarization' or poor catalysis of the electron transfer process (section a, curve III); at intermediate current densities resistive losses predominate (section b, curve III); whilst at high current densities 'concentration polarization', or slow mass transfer processes, become limiting. No generalization is possible with regard to the relative importance of each of these three processes since they are mutually independent. In practice, with air as the oxidant, there is always a substantial contribution from activation polarization, which is increased when a relatively poor fuel such as methanol is used. As a rule, and particularly so in the case of methanol, the combination of activation and resistive losses is such that concentration polarization becomes significant only at high current densities where the efficiency has already fallen below practical requirements.

The relationship between load factor and efficiency can be described in general terms. As with the internal combustion engine the load factor can be defined on the basis of maximum power output. Neglecting activation and concentration polarization terms, the fuel cell i/V characteristic will be linear, controlled by resistance terms alone. It is easily shown that if the cell internal resistance is R and the external, or load, resistance is r, the load factor (*i.e.*, power output/maximum power) is given by the expression:



CURVE I. Theoretical 'ideal' fuel cell.
CURVE II. Resistive losses only.
CURVE III. Resistive losses plus electrode polarization losses. Section(a)mainly electrode polarization. Section (b)mainly electrode polarization and resistive losses. Section(c)inclusive of polarization caused by poor mass transport processes

Fig. 11. Schematic representation of fuel cell characteristics.

$$\text{load factor} = \frac{4Rr}{(R+r)^2}. \tag{13}$$

At maximum power, load factor = 1.0, and $R = r$.
Now

$$\eta = \frac{i^2 r}{i^2(R+r)} = \frac{r}{R+r} \tag{14}$$

and when $R = r$, $\eta = 0.5$, giving the important, though often overlooked result that, at maximum power, voltage efficiency is 50%. This, of course, is an optimal result. Inclusion of the activation and concentration polarization terms would have the effect of reducing the efficiency at peak power to below 50%. This is shown in Fig. 12 where curves are plotted for (i) purely resistive losses and for (ii) combined activation polarization and resistive

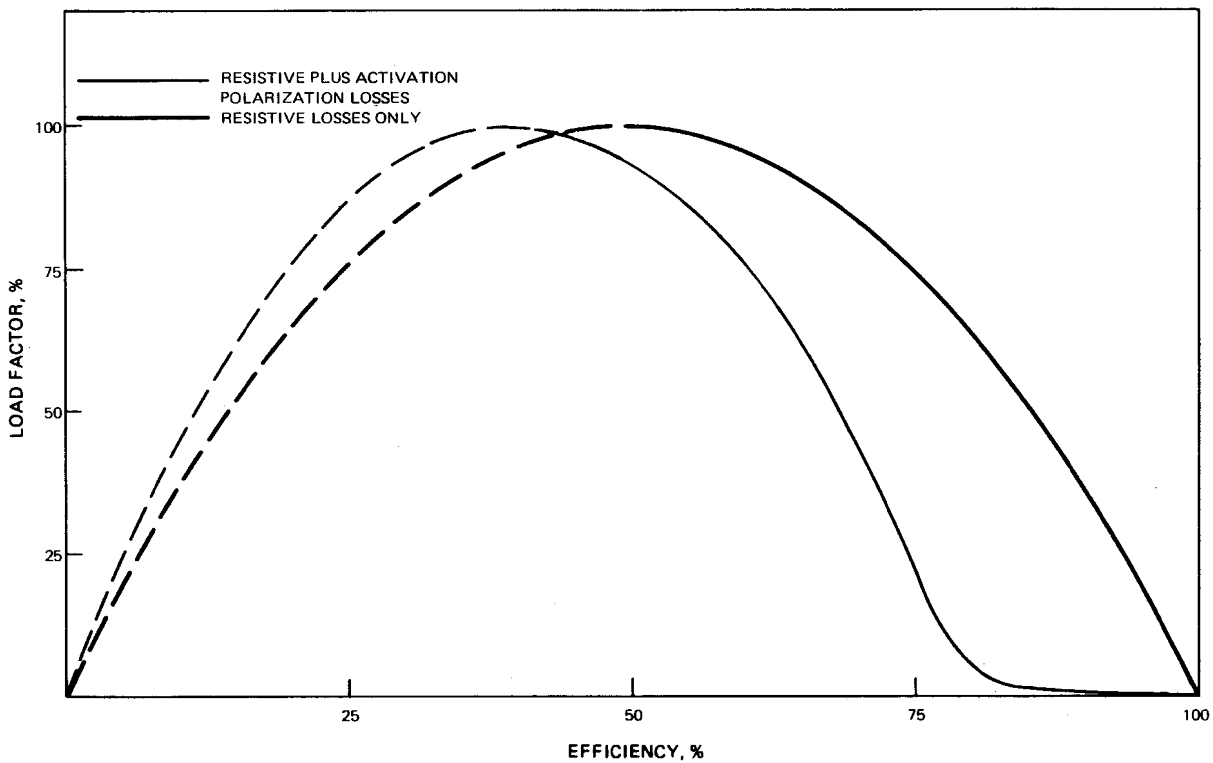


Fig. 12. Efficiency of fuel cells *vs.* load factor.

losses. The latter curve is based on the assumption that the linear portion of the fuel cell characteristic (Fig. 11, curve III, section b) extrapolates back to 1.0 V instead of the theoretical value of 1.23 V, which corresponds closely to the typical behaviour of a hydrogen/oxygen fuel cell. Figure 12 shows that activation polarization exerts its most pronounced effect at low load factors where the fuel cell is most efficient, and emphasizes the importance of developing improved catalysts to minimize this form of loss.

Beyond the peak power point (Fig. 12, dotted portion of curves) power and efficiency both fall and this portion of the fuel cell characteristic is of academic interest only.

Power consumed in auxiliaries, specifically air fans (*e.g.*, for oxygen supply and cooling) and electrolyte pumps, augments the above efficiency losses. Auxiliary losses cannot be confidently assessed and in the author's experience may lie anywhere between 5 and 24% of total output. For simplicity, it will be assumed that auxiliary losses will remain a constant proportion of power output. Even more serious losses may stem from chemical processing of a commercially obtainable fuel into one acceptable to the fuel cell, *e.g.*, naphtha or methanol into hydrogen. Large-scale processes of this kind, *e.g.*, the steam re-formation of methane, take place at thermal efficiencies in the range 55 - 65%. Methanol can be reformed on a laboratory scale with thermal efficiencies approaching 80% [18].

If commercial-scale hydrocarbon reformer efficiencies could be reproduced on a small scale a hydrogen/air fuel cell using reformed hydrocarbons might have a net efficiency within the range 22 - 52% from full load down to almost zero load (exclusive of auxiliary losses).

3.2. *The low-temperature direct methanol fuel cell*

The Shell Research fuel cell programme has been focussed on the low-temperature direct methanol/air cell with the objective of meeting the basic requirements of road transportation. In recent years emphasis has been directed towards improvements in catalysis [19]. The curves drawn in Fig. 13 are based on recent work. They represent best estimates of present performance, practical scale electrodes not having been constructed.

In a complete cell, allowance must be made for resistive losses. The last fully engineered Shell Research fuel cell was constructed in 1969 [20]. This fuel cell consumed hydrazine and air and was designed as a model for the liquid-fuel cell. It had the rather high internal resistance of 1 Ω cm^{-2} not all of which was attributable to the electrolyte. The air electrode in these cells was based on microporous PVC with an equivalent electrolyte thickness of ~2 mm. Allowing for a 1 mm gap between electrodes (less would hardly suffice to permit effective electrolyte circulation), allowing also for space occupancy by inert structural material (packing) and for resistances on electrodes and current collectors, a minimum equivalent electrolyte thickness of 4 mm seems a reasonable estimate. With 3M sulphuric acid at 60 °C this yields a possible minimum cell resistance of 0.33 Ω cm^{-1}.

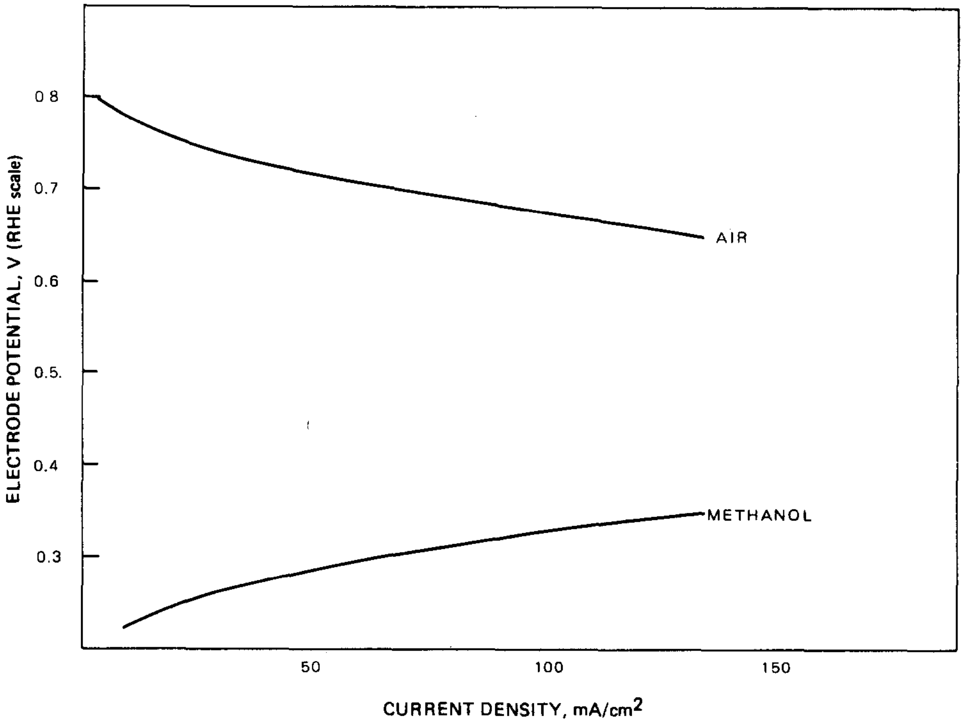


Fig. 13. Basic characteristics of Shell Research methanol and air electrodes. Methanol electrodes: Pt/Ru on carbon paper. Air electrodes: Iridium chelate catalysts. All curves *iR*-free.

United Technologies' fuel cells use catalysed carbon paper electrodes coated on one side with the electrolyte composition (phosphoric acid/silicon carbide/PTFE) and pressed together between moulded graphite current collector/supports [21]. This design represents the most advanced fuel cell technology available today and such a construction may well provide a suitable basis for the low-temperature direct methanol cell. The actual internal resistance of these cells is not known but an estimate of 0.28 Ω cm^{-2} has been made based on published information [22]. The specific conductivity of phosphoric acid at 190 °C is approximately 0.66 Ω^{-1} cm^{-1} (by extrapolation of data in ref. 23). Substitution of phosphoric acid by 3M sulphuric acid in such cells and operation at 60 °C (specific conductivity of 3M sulphuric acid at 60 °C is 1.20 Ω^{-1} cm^{-1}) would yield an internal resistance of some 0.22 Ω cm^{-2}, or rather better than the best estimate for the Shell Research design.

The estimated output of the hypothetical low-temperature direct methanol fuel cell given in Fig. 14 accordingly assumes an internal resistance of 0.25 Ω cm^{-2}. The resultant load factor/efficiency relationship with an allowance for a 15% loss of output attributable to auxiliaries is included in Fig. 15.

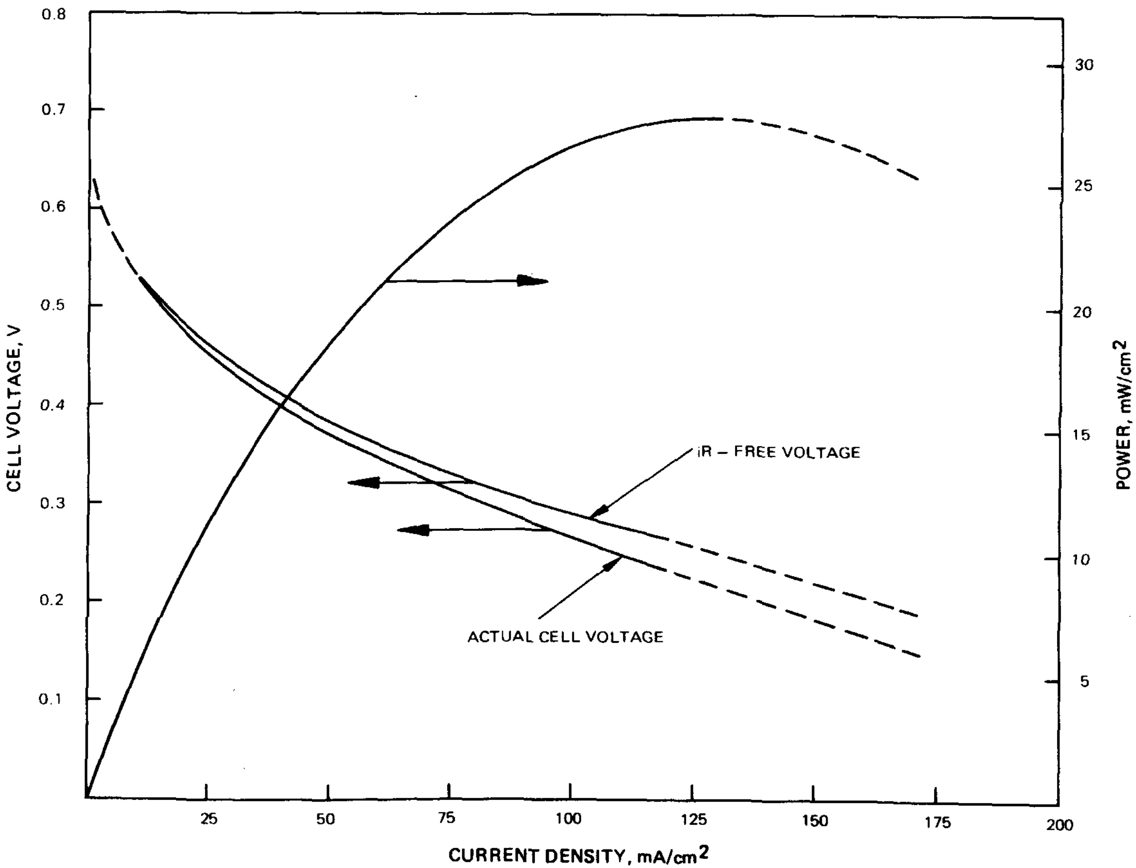


Fig. 14. Predicted characteristics of the Shell Research methanol/air fuel cell.

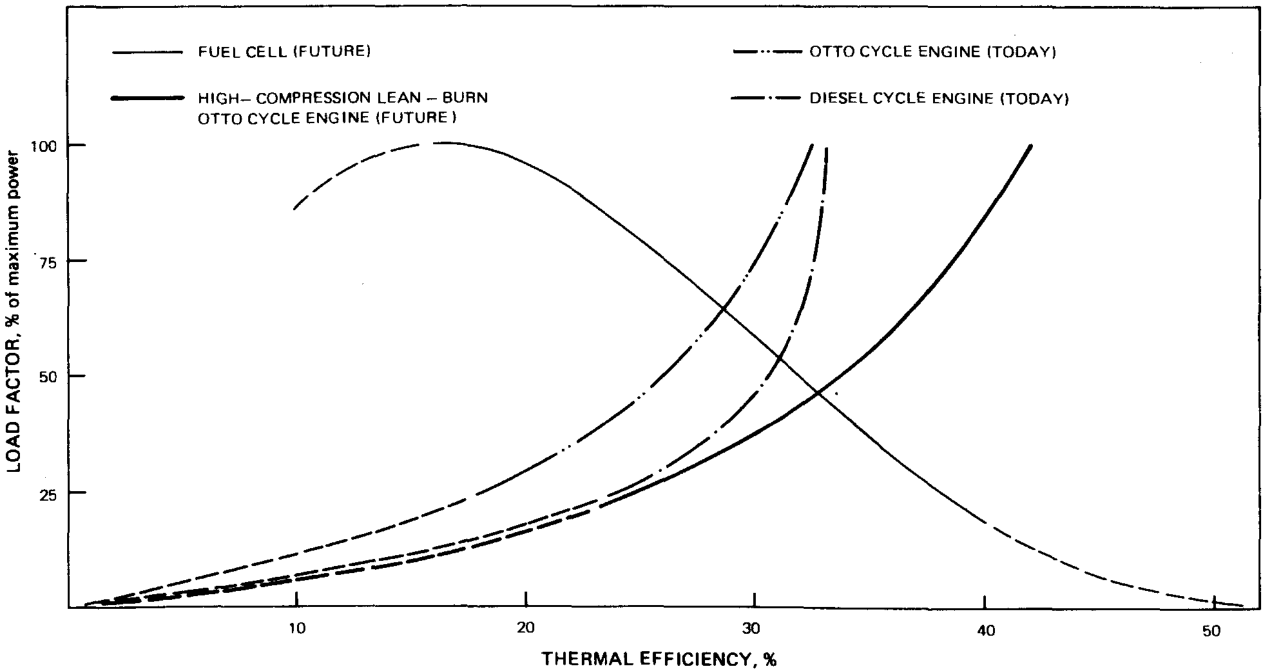


Fig. 15. Load factor/efficiency relationship for the low-temperature methanol fuel cell and heat engines. (Engine data from Tables 1, 2 and 4.)

4. Heat engines and the fuel cell compared

The performance of present day Otto and Diesel engines is compared with the estimated performance of the low-temperature methanol fuel cell and with the lean-burn, high-compression, gasoline or methanol heat engine in Fig. 15. In compiling Fig. 15 efficiency is based, in each case, on the lower heating value of the fuel; in the case of methanol this amounts to the assumption that the theoretical voltage is 1.127 V.

The fuel cell will have a higher efficiency than present day heat engines at load factors less than about 50% and, more importantly, will remain more efficient than the future high-compression, lean-burn engine at load factors of less than about 45%. In transportation the engine is seldom used at such high load factors. Much depends on the vehicle and its usage. Heavy Diesel-engined trucks, for instance, being relatively low-powered, doubtless run at higher load factors than relatively high-powered Otto engine passenger cars.

The type of service the vehicle is used for will also heavily influence the average load factor. The requirements of the VW Otto-engined Golf over the ECE 15 urban drive cycle and a simulated highway cycle are shown in Fig. 16. On the highway cycle, over 50% of the time is spent at <25% load factor and over 75% of the time at <35% load factor. On the ECE 15 urban cycle the power requirement never exceeds 25% of maximum power available.

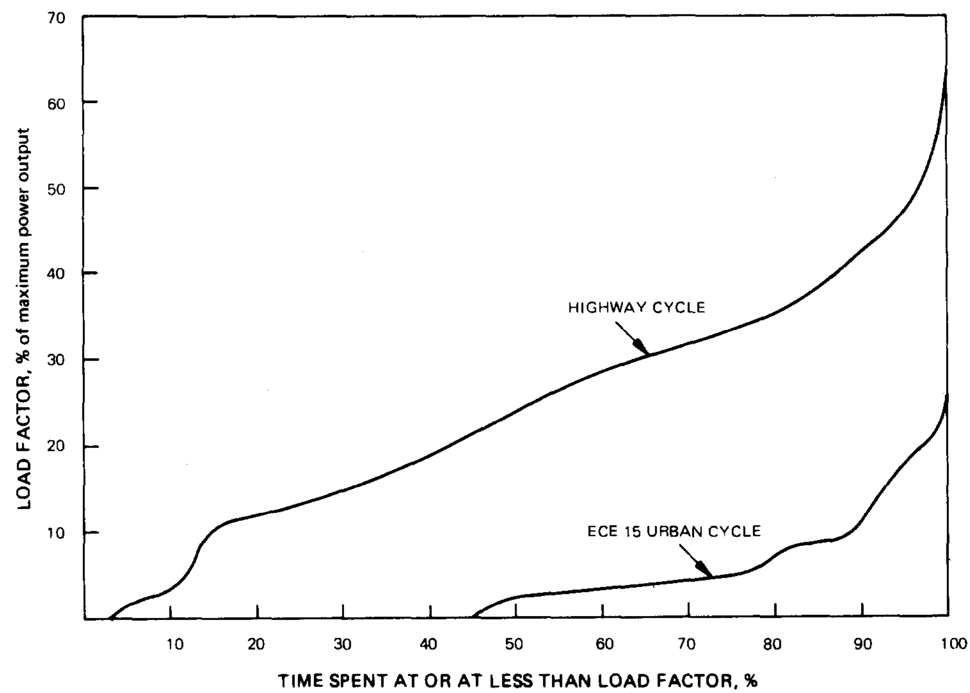


Fig. 16. Load factor requirements of the Otto-engined VW Golf over the ECE 15 urban cycle and a simulated highway driving cycle.

The VW Golf is a small, modern European vehicle with modest performance. Larger and more powerful vehicles will generally operate at lower load factors than the Golf, with correspondingly lower overall efficiencies when driven over similar cycles.

It is of interest to compare the performance of the modern passenger vehicle with the predicted performance of the future high-compression, lean burn, Otto-engined vehicle and with a fuel-celled vehicle. For this purpose the curves in Fig. 15 have been used, although a good deal of the information required must be obtained by extrapolation of the engine curves to very low load factors (<25%). For simplicity it is assumed that the hypothetical vehicles will have the same power/weight ratio as the Otto-engined VW Golf and that the energy and power requirements over the driving cycle will be the same in every case.

The cycle efficiency is given by the formula:

$$\eta = \frac{\int (\text{Power})\, dT}{\int (\text{Power} / \text{Efficiency})\, dT} . \qquad (15)$$

The power/time relationship for the driving cycles can be readily calculated and the efficiency at any given power is derived from Fig. 15 for each power unit.

The results obtained are summarized and compared in Table 7 with test data obtained for the VW Golf in its Diesel and Otto versions. The high-compression, lean-burn vehicle gives results very similar indeed to those of the present day Diesel-engined vehicle over two ECE cycles. This is as expected because these two cycles are carried out at average load factors of ~30% (ECE highway) and ~10% (ECE 15 urban) where, as seen from Fig. 15, the modern Diesel and the future high-compression, lean-burn engines do not differ greatly. The improvement over the modern Otto-engined vehicle is greater in the case of the two highway cycles than it is in the case of the urban cycle, again reflecting the greater difference in efficiency between the two power units (*i.e.*, the modern Otto and the future high-compression lean-burn) at high load factors.

The fuel cell vehicle is considered in two versions: (1) a hybrid consisting of a 20 kW net output fuel cell with lead–acid batteries to provide peak power and (2) a 37 kW net output fuel cell (matching the output of the present-day VW Golf). Examination of the driving cycles has shown that power demand in excess of 20 kW is marginal (<23 kW) and brief (~1% of total cycle time) and only then is required in the case of the ECE 15 cycle. The energy lost *via* the battery is therefore insignificant and can be ignored for the present comparative purposes.

Table 7 indicates that it is preferable to use the larger fuel cell version so as to obtain the efficiency advantage stemming from operation at lower load factors. The advantage of the hybrid vehicle is a reduction in weight and cost (problematic). At the limit the fuel cell in the hybrid vehicle might be made just large enough to provide the average highway cycle power requirement (*e.g.*, some 11.4 kW in the case of the ECE highway cycle), with

TABLE 7

Thermal efficiencies of equivalent small European passenger vehicles over urban and highway driving cycles

Driving cycle	Average load factor (%)	Thermal efficiency (%)				
		VW Golf Otto (37 kW) (a)	VW Golf Diesel (37 kW) (a)	High-compression lean-burn (37 kW) (b)	Fuel cell (37 kW) (b)	Fuel cell (20 kW/ lead–acid) (b)
ECE 15 (urban)	10	11	14.5	12.5	43.2	39.7
ECE highway (const 90 km/h)	30	22.4	25.3	27.5	36.5	30.5
Simulated highway (c)	30	19	—	25.5	36.1	27.9

(a) Test data.
(b) Assumes vehicle to have same power and weight as the Otto-engined VW Golf (small weight differences between Otto and Diesel versions are not taken into consideration).
(c) The simulated highway cycle approximates to the US EPA highway cycle.

sufficient storage batteries provided to meet peak power requirements. Such a hybrid would have a performance more than adequate for the ECE 15 cycle (but might not meet the demand of many users). On the ECE highway cycle the system would operate at close to 100% load factor with an efficiency of ~16%. On the ECE 15 urban cycle the load factor would be 31% and the efficiency ~37%. It would then be more efficient than the high-compression, lean-burn engine on the urban cycle but less so on the open highway. Whether a net fuel saving would result would clearly depend on the proportions of urban and highway use. The hybrid power concept is basically unsound when applied to the fuel cell vehicle. Its advantages are really manifest only in the case of the heat engine which works most efficiently at high load factors.

It must be pointed out that in arriving at the above results, no account has been taken of electric motor and drive efficiencies. Direct current electric motors are available in the power range 10 - 100 kW with efficiencies of ~90% at design speeds and loads. Their performance at part loads, however, depends very largely on the method of power modulation and the provision or otherwise of a variable transmission. The characteristics of a relatively recent d.c. motor made by Joseph Lucas Ltd. [24] are reproduced in Fig. 17. It shows that the highest efficiency (>92%) is obtainable only at close to peak power and maximum speed. Efficiency falls with both decreasing power and speed but remains above 80% down to 15% of maximum power over a wide speed range. Accordingly, the fuel cell data in Table 7 are undoubtedly optimistic, but on the other hand, no allowance has been made for transmission losses with respect to the high-compression, lean-burn engine.

Heat engines are evidently not well matched to transportation uses if efficiency is the sole criterion. They are better suited to stationary applications,

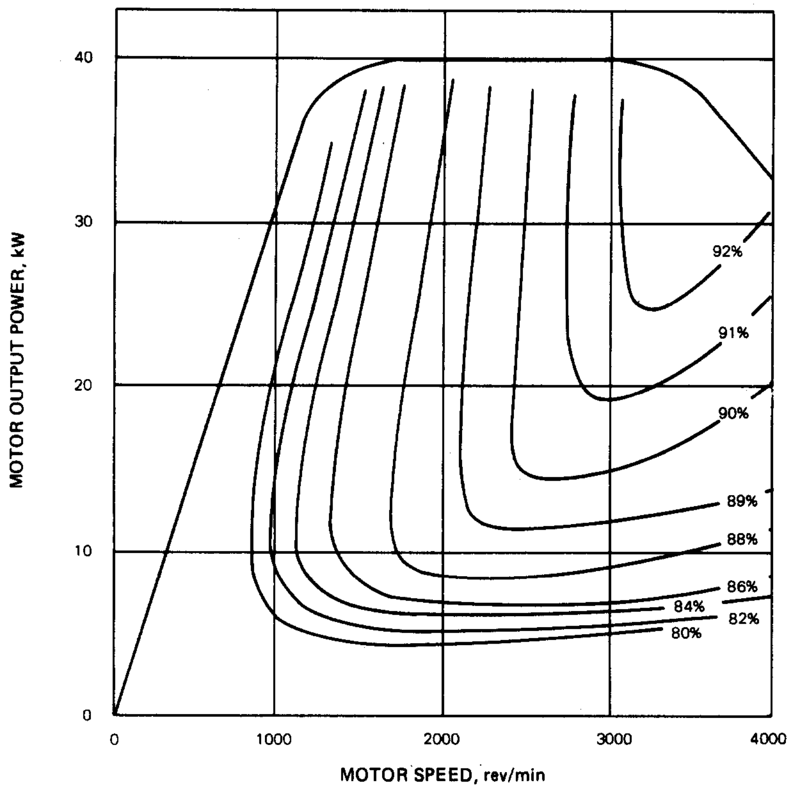


Fig. 17. Efficiency characteristics of a 40 kW series traction motor.

especially those demanding a relatively constant output which can be closely matched to the rated power output of the engine. In cases when average load factors exceed 50% the fuel cell offers no advantage.

The fuel cell is intrinsically better suited to road transportation than to stationary applications. At load factors <30%, efficiency exceeds 38%, comparing favourably with 18 - 24% from present day engines (Tables 1 and 2) and 24% from the high-compression, lean-burn engine. In contrast to heat engines, the greater the reserve of power the more economical it becomes in overall use.

To capitalize on this potential for fuel economy it is vital to operate fuel cells at low load factors. The most critical challenge to fuel cell research is to reduce capital costs to a level at which operation at low load factors becomes economical.

It is not the intention in this paper to discuss costs, but a few comments must be made. Economically affordable capital costs depend on variables such as equipment lifetime, interest rates, fuel costs, maintenance costs and load factor, all of which will be variable not only in time but in relation to competing equipment.

Crude estimates indicate that the hypothetical Shell Research direct methanol fuel cell using precious metal catalysts may cost anything between \$1000 and 5000/kW. Within this wide cost range a figure of \$2000/kW is judged to represent a probable lower cost limit. The hypothetical cell design specification on which these cost figures are based incorporates a current density of 52 mA/cm^2 and therefore the design rating will approximate to a 50% load factor (Fig. 14). If the specification current density had been fixed at the maximum power point we would have had, in effect, a capital cost amounting to \$1000/kW, with a corresponding down-rating of efficiency from 33% to 16% (Fig. 18). Conversely, derating the cell to 25% of maximum power output would double the capital cost to \$4000/kW and increase rated efficiency from 33% to 38%. Figure 18 shows that further gains in efficiency by the derating process would be very costly indeed.

Costs in the range \$1000/kW and above are much too high for road transportation, even if, as seems possible, a substantial fuel saving may result. Costs at this level would mean that the 37 kW power unit for the equivalent of the small modern passenger vehicle would cost \$37,000. Cost reductions of one or two orders of magnitude are still required.

A forecast of the costs and efficiencies of the high-compression, lean-burn engine and the fuel cell in the form of stationary generator sets is given in Table 8. It is assumed that the high-compression, lean-burn engine will cost the same as the present day Diesel, with which it has much in common. The fuel cell offers no competition to the internal combustion engine at load factors above 50%, but at lower levels the advantage in fuel economy can become very significant. The user could, however, save expense by purchasing a small internal combustion engine set and providing peak power (if this

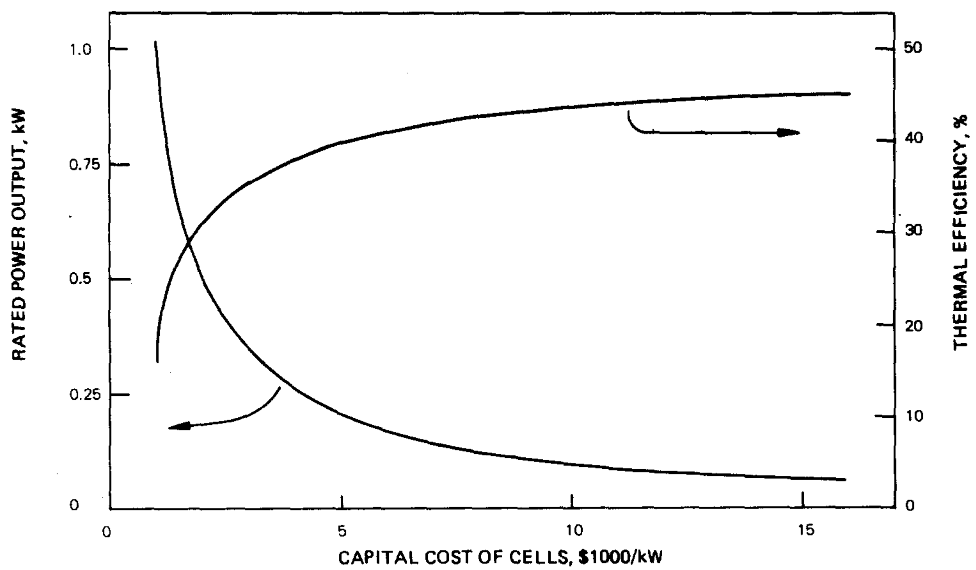


Fig. 18. Cost of fuel cells in relation to power output and efficiency.

TABLE 8

Comparison of present day and future generator sets

	Present		Future	
	Otto engine sets	Diesel engine sets	High-compression lean-burn engine sets	Methanol fuel cell
Price \$ (1980)/kw	200 - 400	400 - 600	400 - 600	1000+
Overall thermal efficiency (%):				
(1) Full load	12 - 14	20 - 26	34	16
(2) 50% full load	10 - 12	18 - 24	27	32
(3) 25% full load	6 - 8	15 - 19	19	38

were required infrequently) by means of batteries. In this way, a small, high-compression, lean-burn set operating at full load would yield an efficiency (34%) only a little inferior to that of the fuel cell running at 25% of output (38% efficient).

Finally, it must be emphasized that capital costs and efficiency do not alone determine the marketability of any device, additional factors such as reliability, portability, environmental acceptability (noise, heat, effluents) may be, in some circumstances, overriding.

5. Conclusions

(1) The Diesel engine, traditionally developed for fuel economy, is mature and unlikely to be dramatically improved over the next decade or two.

(2) The Otto engine, presently inferior to the Diesel engine at part loads, is likely to undergo development in a high-compression, lean-burn form. Its performance at part loads (<25%) will then closely match that of the present day Diesel engine on which it will improve at full load.

(3) The direct methanol fuel cell, using electrodes with a performance capability predicted on the basis of present day laboratory data, will be more efficient than the high-compression, lean-burn engine at load factors less than ~45%.

(4) The fuel cell is essentially best suited to applications demanding varying power outputs where the average load factor is low (say <25%). Transportation represents such a market, where the fuel cell could offer considerable fuel savings.

(5) Drastically reduced fuel cell capital costs, *(e.g., via* better catalysts) are necessary if the fuel cell is to compete with the internal combustion engine for transportation use.

(6) The availability of cheap methanol could stimulate the development of the methanol fuel cell. Conversely, the development of such a cell would increase the attractiveness of producing methanol as a fuel.

(7) Users of small, stationary power packs are not, generally, very concerned about fuel costs. Although this attitude may change as fuel prices rise, many (*i.e.*, those who operate equipment at high load factors) would not benefit from the application of fuel cells. However, if fuel cells become sufficiently cheap, oversize units operating at low load factors could yield very high efficiencies.

Acknowledgements

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