The Atkinson cycle revisited for improved part-load fuel efficiency?

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Abstract

A 15% to 20% reduction in global gasoline consumption could be achieved virtually overnight if the conventional spark-ignition engine throttle was eliminated. The fuel consumption penalty occurs during part-load operation, the condition where gasoline engines operate for most of their life-time. It has been shown that the throttle arrangement is not an inherent feature of the engine and that, theoretically at least, the penalty need not be paid. All that is lacking is an innovative idea.

A number of alternative concepts to the conventional throttle have been considered. The mode of operation of many such concepts involves transforming the Otto cycle at part load into an Atkinson cycle. The extent of the fuel saving that can be realised vary from 15% to 25%, depending on the concept and the engine load.

Of all the concepts, it is proposed that a deliberate attempt to make an Otto-Atkinson hybrid engine would represent the optimum solution. The design criteria call for different stroke lengths between the inlet/compression and the expansion/exhaust strokes. They also require that the inlet/compression stroke length be continuously variable to eliminate the conventional throttle arrangement. Meeting these criteria would ensure that the part-load efficiency is maximised without sacrificing on the full-load efficiency that is currently achieveable by modern spark-ignition engines. One solution to the problem is presented which demonstrates that the concept is technically feasible.

1. Introduction

It is generally true that research investigations dealing with the performance of internal combustion engines are carried out at full load conditions. However, for the greater portion of their lifetime, motor-car engines operate at less than half load, and therefore investigating the engine efficiency at part-load is of great importance.

An analysis of the distribution of the fuel energy is discussed in detail by Taylor [1]. A typical example for an automotive engine operating at full load and at some mid-range speed is given in figure 1. (If the fuel power is assumed to be 100 kW, then the percentage break-down may be equated directly with power in kW.) It may be noted that the fuel energy is roughly divided into three equal amounts: one third of the original fuel energy is lost to the heat of the exhaust; one third is obtained at the crankshaft as useful energy, of which a small portion is needed to drive the auxiliaries; and one third is lost to the coolant, either as direct heat-transfer from the combustion zone or indirectly from mechanical frictional losses which are ultimately dissipated in the form of heat.

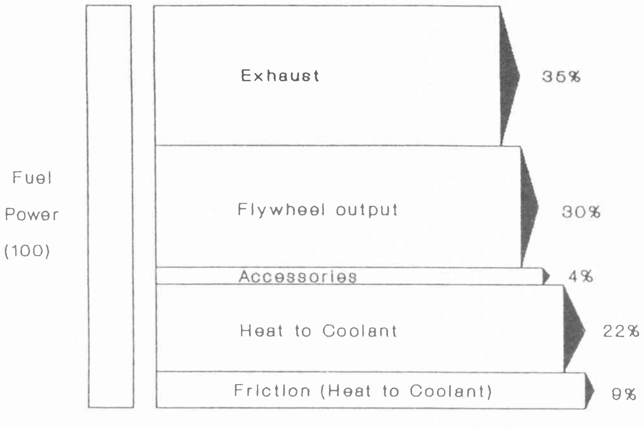

Figure 1 – Fuel power distribution at full load for a typical automotive engine

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At part load, the exhaust heat losses and the combustion heat-rejection losses are reduced in approximate proportion to the fuel power, whilst the mechanical friction and accessory losses are virtually unchanged, assuming that the engine speed is constant in this example. This situation is illustrated in figure 2 (data extracted from [2]), which shows the approximate power break-down for 20% output. The flywheel power has dropped from 30 units to 6 units, and the exhaust and heat-to-coolant have dropped in similar proportion. However, the accessories power has remained unchanged at 4 units and the friction loading has reduced very slightly owing to the reduced bearing loading.

The situation at part load gains a further dimension in the form of throttling loss. This is caused by the restriction imposed by the inlet throttle valve which requires the engine to pump the working fluid (air and fuel mixture) from a low pressure in the inlet manifold to a higher

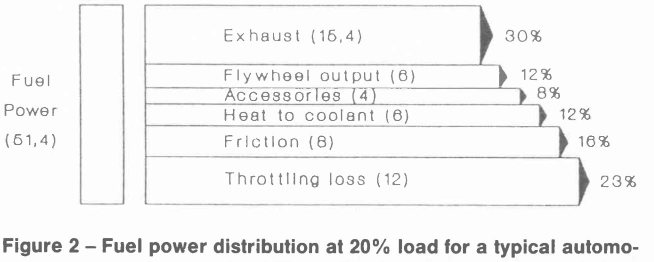

**Figure 2 – Fuel power distribution at 20% load for a typical automotive engine
Numbers in brackets relate to full load scale in fig. 1**

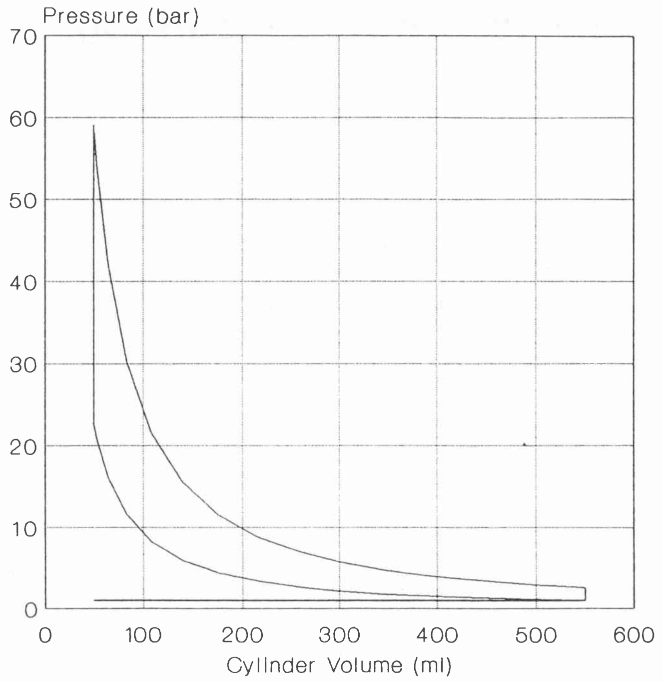

Figure 3 – p-V diagram for a typical spark-ignition engine at full load

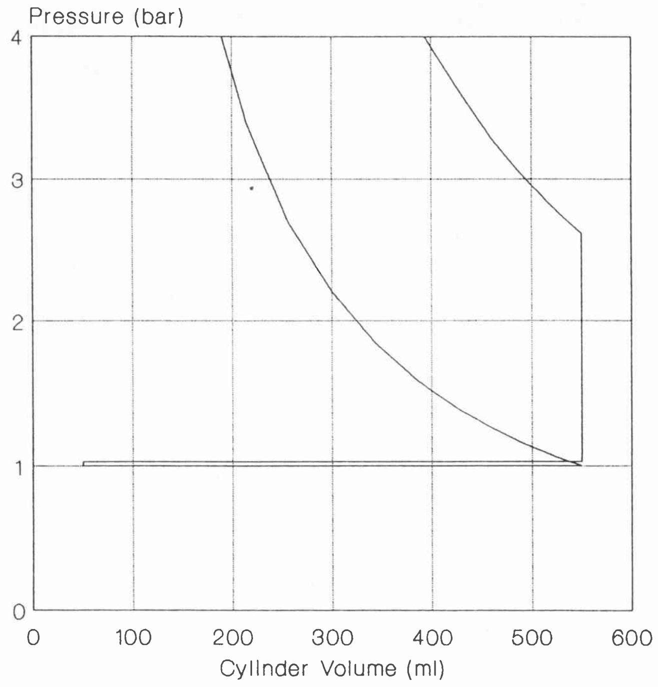

Figure 5 – Close-up of inlet and exhaust strokes of a spark-ignition engine at part load

pressure in the exhaust manifold. As indicated in figure 2, the throttling loss at 20% power output represents some 23% of the total fuel power requirement.

The influence of the engine throttle is well illustrated in the pressure-volume (p-V) diagram. A theoretical p-V diagram for a spark-ignition engine operating at full load is shown in figure 3. Since it is the intake and exhaust strokes that are of interest, this part of the diagram is shown magnified in figure 4. For clarity, the inlet and exhaust strokes are shown slightly separated, although theoretically, they would both take place at atmospheric pressure.

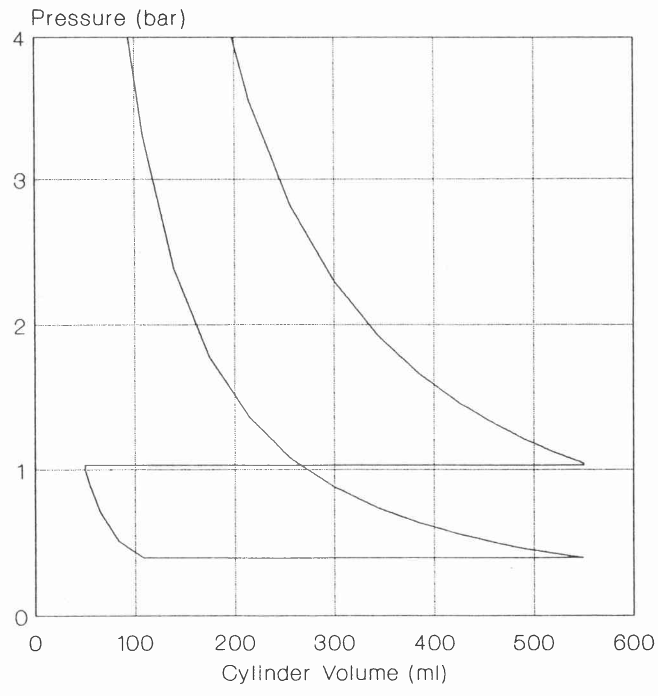

Figure 4 – Close-up of inlet and exhaust strokes of a spark-ignition engine at full load

At part load, the picture is very different. Figure 5 shows the inlet stroke with an inlet manifold pressure of 0,4 bar absolute pressure. The area enclosed by the suction stroke represents work that has to be done by the engine and is in fact the throttling loss.

It has always been recognised that the loss associated with the part-load throttling process is a severe disadvantage of the conventional spark-ignition engine. Since this loss increases as the load is reduced, the magnitude of the throttling loss depends on the style of driving as well as the pattern of driving, i.e., city centre, suburban freeway, commuter traffic or inter-city travel. Despite these uncertainties, it has been estimated that the fuel consumption due to the throttle loss averages out to about 20% to 25% of the total fuel used for the typical urban driving conditions [3]. Theoretically, this energy loss is not an inherent attribute of the spark-ignition engine but rather it is a characteristic of the conventional throttle arrangement which has survived only because of its convenience and simplicity.

2. Technical background

What options exist for the possible elimination of the throttling loss?

The essential requirement of any alternative concept to the conventional throttle is the facility to vary the mass of the induction charge whilst maintaining reliable ignition and combustion within the engine. A variety of concepts have been proposed which attempt to address the problem:

1. The concept of a small engine fitted with a turbocharger or a super-charger effectively results in the engine operating nearer wide-open throttle under urban driving conditions whilst retaining the capability to produce adequate maximum power. Such concepts have been proposed by most of the manufacturers of small, economy vehicles.
2. Concepts such as stratified charging have been attempted, whereby the fuel flow is varied to suit the engine power demand, whilst the inlet air flow remains unthrottled [4]. Expensive fuel injection equipment and problems of reliable ignition over the entire load range have generally prevented such engines from entering the market place.
3. The inlet charge could be expanded by means of a turbine and the expansion work could be extracted to do useful work. Whilst this concept is theoretically possible it does not compare as favourably as option 1 above in terms of fuel-consumption benefit and capital cost.
4. In the case of multi-cylinder engines, there is the possibility for reducing the output power at part load by cutting out one or more cylinders, leaving the remaining active cylinders to function at wide-open throttle. This concept has been investigated and was shown to give up to 15% improved part-load fuel consumption [5, 6]. However, the control of the load regulation was difficult and the engine balance, spark-plug fouling and the possibility of thermal distortion of the engine represented difficult technical obstacles.
5. The engine power could be regulated by varying the inlet valve opening duration or the length of the inlet stroke. In effect, this results in the engine approximating the Atkinson cycle, as will be shown. Although this concept does not enjoy much coverage in the literature, it does raise interesting possibilities, some of which are explored in more detail below.

3. The Atkinson cycle

Before considering the role of the Atkinson cycle in the context of a throttling loss alternative, it may be helpful to describe the thermodynamic cycle itself.

The theoretical Atkinson cycle is depicted in figure 6. It consists of an adiabatic compression process, a constant-volume pressure rise, an adiabatic expansion, and a constant-pressure reduction in volume. The air-standard cycle efficiency of the Atkinson cycle can be shown to be:

Air-standard cycle efficiency

$$= 1 - \nu \left[\frac{r_{v2} - r_{v3}}{(r_{v2})^{\nu} - (r_{v3})^{\nu}} \right] \quad \text{(eq 1)}$$

where r_{v2} = the expansion ratio
r_{v3} = the compression ratio

At the same compression ratio, the Atkinson cycle is more efficient than the Otto cycle owing to the more complete expansion of the working fluid.

A number of commercial Atkinson-cycle engines were built during the period 1906-1910. The motion necessary

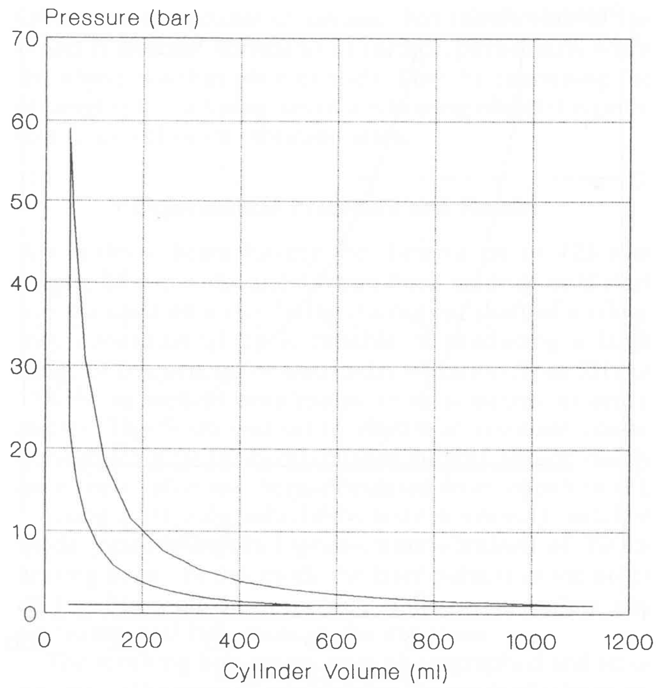

Figure 6 – p-V diagram for the Atkinson cycle

to produce the constant-pressure process was obtained by making the piston drive the crank through a toggle joint. Although the engine was demonstrated to be more efficient than an equivalent Otto-cycle engine, the full extent of the theoretical efficiency was not realised owing to the friction losses associated with the longer piston travel. Moreover, the toggle joint mechanism proved to be problematic and the Atkinson engine was discontinued [7].

3.1 Variable inlet-valve opening duration

An alternative concept to the conventional throttle, variable inlet-valve duration involves either shutting the inlet valve early at some stage during the inlet stroke and allowing free expansion of the trapped mixture for the remainder of the inlet stroke, or lengthening the inlet valve opening duration so that some of the inlet charge is expelled back out of the cylinder before valve closure occurs.

A p-V diagram for a cycle with early valve cut-off is shown in figure 7. It will be seen that the p-V diagram has effectively become the Atkinson cycle, with an expansion and re-compression section that has no effect on the thermodynamic cycle although piston motion occurs during this section. Whilst it should be apparent that the effective compression ratio of this cycle is less than that of the same engine at full load, equation 1 above indicates that the cycle efficiency is nominally the same as the full-load cycle since the expansion ratio has not changed and it is this ratio that dominates the Atkinson cycle efficiency.

A certain amount of research and experimentation has been reported in the literature concerning this concept which has demonstrated that the theoretical fuel savings can be only partially realised in practice.

Saadawi [9] investigated the possible benefits of this concept by manufacturing special camshaft profiles for a single-cylinder test engine that corresponded to fixed

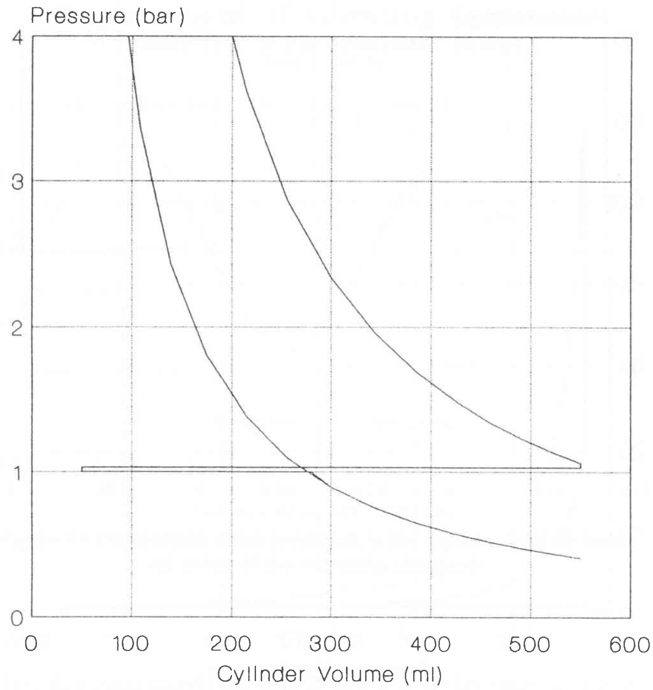

Figure 7 – Theoretical early inlet-valve closure

part-load operation points without engine throttling. He found that the brake-thermal efficiency of the engine was improved by 16,6% at half of full load when compared to the conventional throttle arrangement.

A similar investigation was carried out by Tuttle [2] using a multi-cylinder engine. He also experimented with pre-machined camshafts that corresponded to fixed part-load ratings. A number of benefits were noted which included lower specific fuel consumption, lower cylinder-gas temperature and lower NO_x emissions. At a test-vehicle speed of 64 km/h, the fuel consumption was improved by 10% when compared to a conventionally throttled engine. However, it was found that throttling losses could not be completely eliminated by this concept since the inlet valve could not be instantaneously closed due to inertia considerations. As the inlet valve was closing in the mid-inlet stroke, the valve itself represented an effective throttle. This is illustrated in figure 8 where the inlet valve is constrained to close according to a normal cam profile. It is apparent that some of the benefit of the theoretical cycle has been lost as indicated by the throttling area in the diagram.

Interestingly, a vehicle was built, circa 1912, that embodied this concept [8]. The Sizaire-Naudin had no carburettor throttle and the engine output was regulated by sliding a profiled cam along its shaft which modified the operation of the inlet valve. Unfortunately, the reference gives no further details of this innovative design or its performance.

The practical problem of producing a continuously-variable inlet duration has been addressed by several researchers [10, 11]. The proposed solutions generally take the form of a hydraulic cam follower which can collapse (hydraulically) at any desired state of the induction period. Additional refinements include features to prevent the valve from impacting on the valve seat. Although these ideas have some merit none has yet been demonstrated to compete successfully with the conventional throttle, in terms of the additional complexity justified by the improved fuel consumption. It should also be noted that, at typical part load speeds, about 0,3 kW of the engine power is required to compress the inlet valve springs. This power is normally recovered as the valve closes but, under a hydraulic dumping arrangement, would be simply lost to fluid friction in the hydraulic system.

3.2 Variable inlet stroke

If it were possible to vary the length of the inlet stroke, the quantity of inlet mixture could be regulated and hence the power output. This concept is illustrated for part load in the p-V diagram shown in figure 9. It may be seen that the engine is now following a true Atkinson cycle.

Since the cycle is conceptual at this stage, it may be helpful to list the design criteria of an engine mechanism that would achieve the optimum performance.

1. The length of the inlet/compression stroke must be continuously variable to regulate the power output.
2. As the compression stroke shortens, the clearance volume should be reduced so as to maintain a constant compression ratio.
3. The expansion/exhaust stroke must remain fixed, or at least longer than the inlet/compression stroke to achieve the full benefit of the Atkinson cycle.

This theoretical concept was investigated using a sophisti-

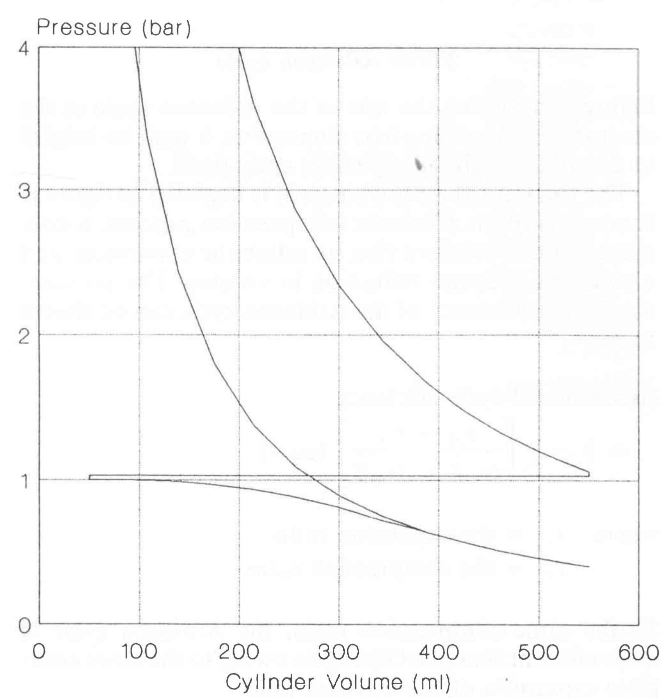

Figure 8 – Early inlet-valve closure with valve inertia effects considered

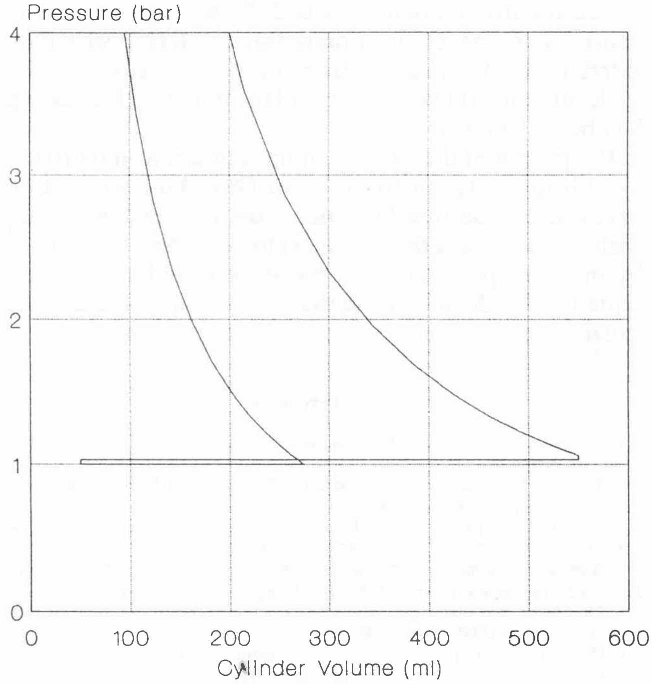

Figure 9 – Part-load diagram for a variable inlet-stroke engine with a (normal) full expansion-stroke

cated computer simulation by Ma and Rajabu [12]. A number of engine parameters were investigated which attempted to represent the bounds of practical possibility. It was concluded that the Otto-Atkinson cycle could deliver substantial fuel economy benefits with virtually no emission penalties. For example, comparing the optimised Atkinson engine with a conventional Otto engine, at 1 500 rev/min, the predicted fuel consumption improvement at a BMEP of 5 bar was 16%, increasing to 20% at a BMEP of 2 bar. The predicted HC and NO_x emissions for both engines were virtually identical.

Whilst Ma and Rajabu offered no suggestions as to how the required thermodynamic cycle might be achieved in practice, one possible mechanism, which is shown in figure 10, is submitted by the author. In effect, the piston motion is controlled by a combination of a primary and auxiliary crankshaft, the auxiliary crank turning at half the speed of the primary crank. A cross-link combines the motion of the two crankshafts, and a sliding member on the cross-link provides the lower pivot for the piston connecting rod.

As it is depicted in the diagram, the position of the sliding member on the cross-link would correspond to about 30% of full load from the engine. During the intake/compression stroke, the two crankshafts are out of phase and the piston motion describes a short stroke. During the expansion and exhaust stroke, the crankshafts are in phase, resulting in a long stroke and hence the overall piston locus achieves the Atkinson cycle criteria. If the sliding member were positioned at the extreme left of its travel, the piston would be virtually linked to the primary crankshaft and the locus of the piston would be that of a conventional Otto-cycle engine, with a full intake and compression stroke followed by a full expansion and exhaust stroke. Thus, the position of the slider on the

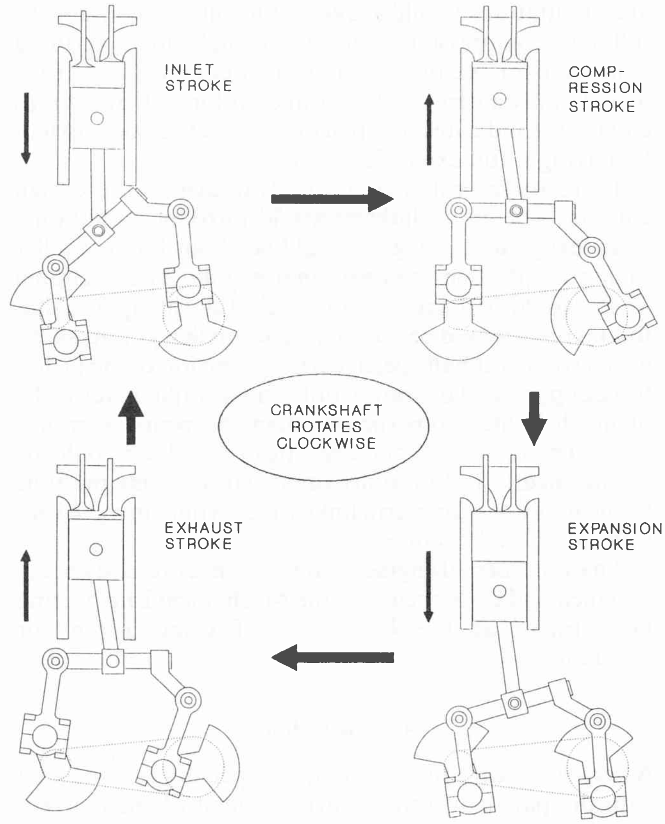

Figure 10 – Variable inlet-stroke engine concept (About 30% inlet stroke illustrated)

cross-link determines the magnitude of the inlet/compression stroke and hence the engine power output.

An analysis of the piston motion achieved by such a mechanism is illustrated in figure 11. Owing to practical considerations, the maximum piston/cylinder head clear-

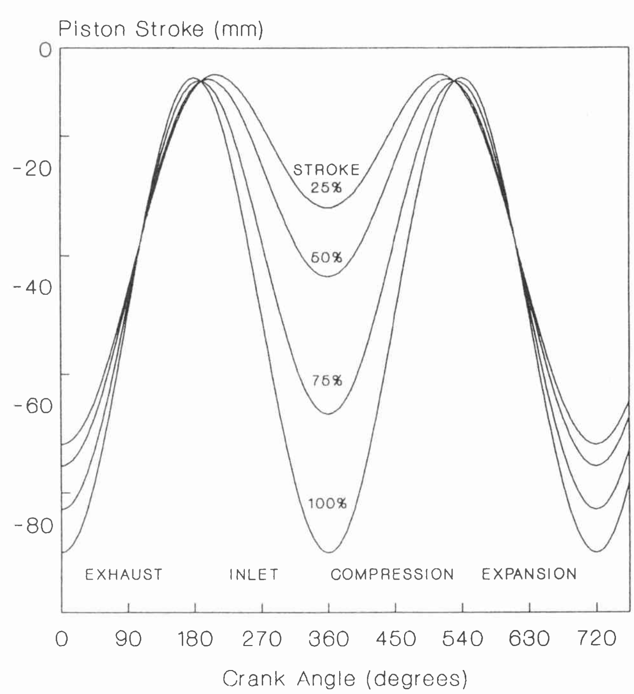

Figure 11 – Locus of piston travel with variable inlet-stroke mechanism

ance limitations would make it difficult to maintain the full-load compression ratio at very light loads (criteria item 2 above), because the short inlet/compression stroke would imply a very small clearance volume. However, as equation 1 indicates, the penalty is offset to some extent by having a full expansion stroke.

There are several obvious disadvantages to this design concept. The crank linkage would introduce additional complexity and the engine would be about 150 mm taller than an equivalent conventional engine. The frictional losses would be increased although it is anticipated that this increase would be minimal. The auxiliary crankshaft, which rotates at half speed, carries a portion of the piston force at part-load operation only and it could therefore be of much lighter construction than the primary crankshaft. The cross-link and the "throttle" slider could be constrained to follow approximate straight-line motions by means of swinging-arm links which would involve very little additional friction.

An experimental engine of this design is presently being designed and constructed by the Mechanical Engineering Department at the University of Cape Town for evaluation.

4. Conclusions

A very significant fuel consumption penalty is associated with the part-load throttle arrangement of the conventional spark-ignition engine. It has been shown that the throttle arrangement is not an integral feature of the engine and that, theoretically at least, the penalty need not be paid.

The extent of the fuel saving that could be realised has been estimated at about 15% to 25%. Various alternative concepts to the conventional throttle have been considered which, in effect, transform the part-load Otto cycle into an Atkinson cycle, and the potential fuel saving has been demonstrated.

It is proposed that the optimum solution would involve a deliberate attempt to make an Otto-Atkinson hybrid engine that is designed for maximum part-load efficiency without sacrificing the full-load efficiency that is achieved by modern spark-ignition engines. One solution is presented which demonstrates that the concept is technically feasible.

5. References

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