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ABSTRACT

Theoretical thermodynamic analysis reveals that, when a fixed amount of heat energy is added into an Otto cycle the thermal efficiency of that cycle can be substantially improved by increasing the expansion ratio while keeping the compression ratio unchanged to achieve a greater net work output. As such, to maximise the cycle work output, the exhaust gas is allowed to expand to atmospheric pressure within the power machinery itself. With this approach, the pressure versus specific volume diagram of this modified cycle at exhaust valve opening is thus minimised and hence waste heat recovery/utilisation such as the implementation of turbocharging system can be eliminated.

This paper presents the development and design considerations of a double helical screw internal combustion engine. The engine consists of two pairs of helical screw type, positive displacement machines in which one pair performs the operation of compressing the working fluid, while the other performs the operation of expansion. The machines, being separate, could be designed in such a way that the effective gas expansion ratio is greater than the compression ratio.

INTRODUCTION

Reciprocating piston engines are the dominant type of engine used in land transportation. Their presence is also widespread in sea and air transport and in fixed installations. The Wankel engine, which offers an interesting departure from convention, has nevertheless failed to capture a significant fraction of the engine market. Despite the technological maturity and reliability of piston engines, they suffer from several fundamental disadvantages [1]*.

The stroke of a piston moving up and down its cylinder is determined by the crank arm length. By virtue of such a design, the compression ratio is equal to the expansion ratio. With the exception of Miller engines, where the effective compression ratio is reduced by early or late closure of the inlet valves [2, 3, 4]. This fundamental constraint limits the thermal efficiency of the piston engine. At the end of the expansion stroke, the pressure of the combustion gasses in the cylinder is still above that of ambient. Thus potentially extractable work output is lost in the exhaust gasses.

The piston and other moving parts suffer extreme acceleration and deceleration while in operation. This limits the maximum operating speed of the engine for a given level of materials technology [5]. Furthermore, the rapidly reciprocating masses produce vibration which is always undesirable, albeit to different extents depending on the application in which the engine is found. The means of overcoming this problem has invariably been the addition of more cylinders and counter weights for dynamic balancing [5, 6]. This further worsens the mechanical complexity of the engine. For example, the four-stroke, six-cylinder piston engine has six pistons and connecting rods, one crankshaft, at least one camshaft and at least twelve valves, giving a total of twenty-six principal moving parts. This compares rather unfavourably with the three principal moving parts in a twin rotor Wankel engine which has comparable vibration and torque fluctuation [1].

The linear force produced by the piston is converted to more useful rotational torque by the crank mechanism. Again by virtue of design, the

* Numbers in brackets indicate references found at the end of the paper.

torque produced is not uniform. It is even negative during the compression stroke. To provide a reasonably smooth torque, the solution has been to add more cylinders, with its consequent disadvantages mentioned above. To prevent the engine from stalling during the compression stroke, a relatively high idling speed is maintained, together with the use of a heavy flywheel [6]. The former results in poor fuel economy in stop-go applications such as land transport and the latter, poor engine acceleration.

The combined consequence of vibration and uneven torque distribution in a cycle leads to a further disadvantages. The mountings and power transmission systems of piston engines need to be more over-designed with greater dynamic factors. This results in a vicious cycle of increased overall weight of the vehicle, which then requires a larger engine, and in turn leads to further increase in the weight of the transmission system [1].

The four-stroke piston engine requires a complex and power consuming valve mechanism which adds to the cost of the engine, reduces its thermal efficiency and introduces a further source of mechanical vibration [1]. While the two-stroke version has no valves or only a simple valve mechanism, it suffers from poor scavenging and poor emissions characteristics [5, 7].

The double helical screw internal combustion engine is conceived with a view to solving the foregoing disadvantages in piston engines, in particular their inability to provide different compression and expansion ratios and their non-rotary output.

The objective of this paper is to describe the design of a prototype double helical screw engine whose compression and expansion ratios are equal and to present the practical experience gained during the development of the engine. Improvements that will be incorporated into future prototypes will also be discussed.

COMPARISON OF OTTO AND ATKINSON CYCLES

The potential advantage of having the expansion ratio greater than the compression ratio can be examined by comparing the Otto and Atkinson cycles. The Atkinson cycle is different from the well-known Otto cycle in that during the expansion stroke, the combustion gasses are allowed to expand to atmospheric pressure [8, 9, 10].

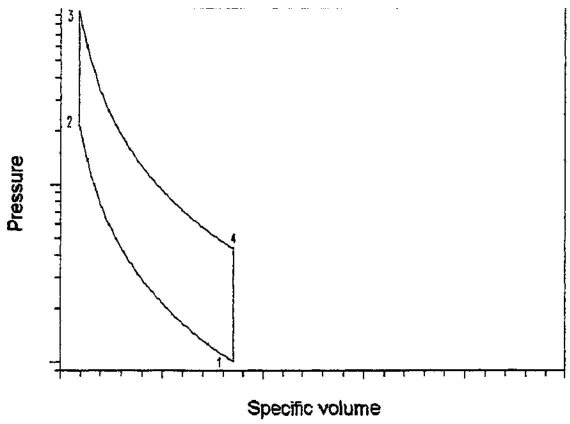

Figure 1. Idealised air-standard Otto cycle

ATKINSON CYCLE - The idealised air-standard Pressure-Specific volume diagram is shown below.

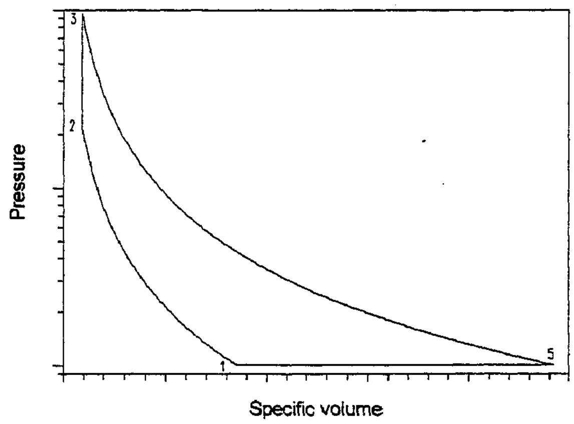

Figure 2. Idealised air-standard Atkinson cycle

COMPARISONS - It is immediately apparent from the above diagrams that for the same negative work (compression work) and heat input, the positive work of the Atkinson cycle is far larger. A comparison of the plots of the thermal efficiencies of the two cycles against compression ratio shows that at a given compression ratio, the efficiency of the Atkinson cycle is higher than that of the Otto cycle. The equations for thermal efficiencies are:

$$\text{Otto cycle}: \eta_O = 1 - \frac{1}{r^{k-1}} \tag{1}$$

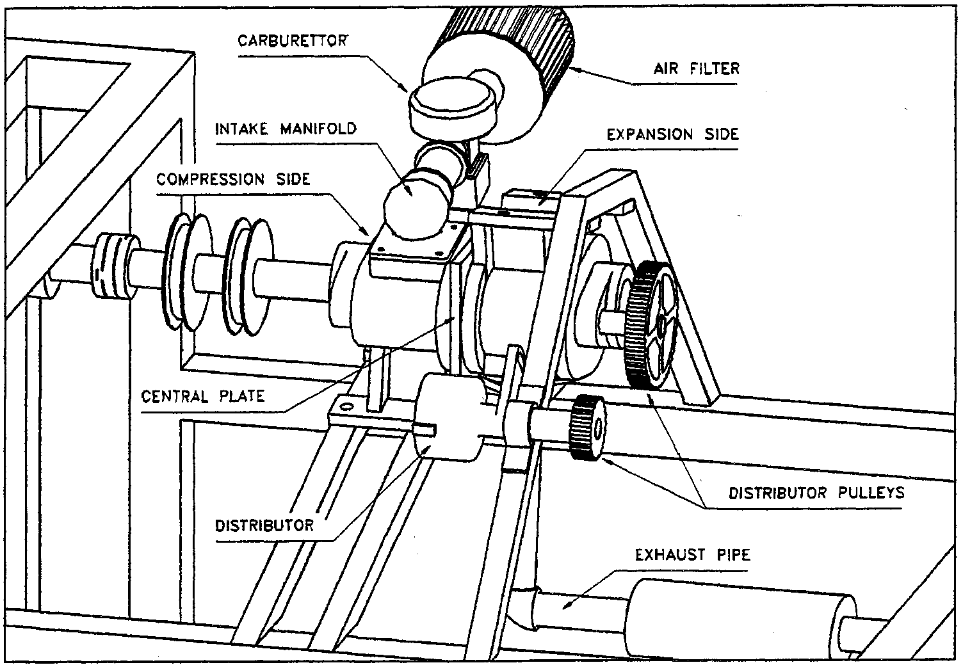

Figure 4. Layout of test rig

$$\text{Atkinson cycle}: \eta_A = 1 - \frac{k}{r^{k-1}}\left[\frac{r_p^{1/k} - 1}{r_p - 1}\right] \quad (2)$$

where

$$r_p = \frac{p_3}{p_2} = \frac{1}{r^k} \cdot \frac{p_3}{p_1}$$

or

$$r_p = 1 + (k-1) \cdot \left[\frac{Q_{in}}{p_1 V_1}\right] \cdot \frac{1}{r^{k-1}}$$

It can be seen that for the Atkinson cycle, the thermal efficiency depends on the normalised heat input Q_{in}/p_1V_1 in addition to the compression ratio. For a typical value of Q_{in}/p_1V_1 = 30, Fig. 3 shows the theoretical improvement in thermal efficiency of the Atkinson over the Otto cycle.

A further comparison of Fig. 1 and Fig. 2 shows that in the Atkinson engine, for complete expansion to ambient pressure, the expansion ratio has to be about three times the compression ratio.

In practice, however, the gain in positive work arising from additional expansion will be slightly reduced by friction. This is especially so during part-load operation [11]. A practical engine must be designed so that the net work output during the entire expansion stroke is never negative. This can be ensured by using a lightly spring-loaded under-pressure valve to admit exhaust gas or atmospheric air (depending on emission requirements) into the expansion chamber when its pressure falls below ambient. Furthermore, the maximum expansion ratio should be less than the theoretically calculated value.

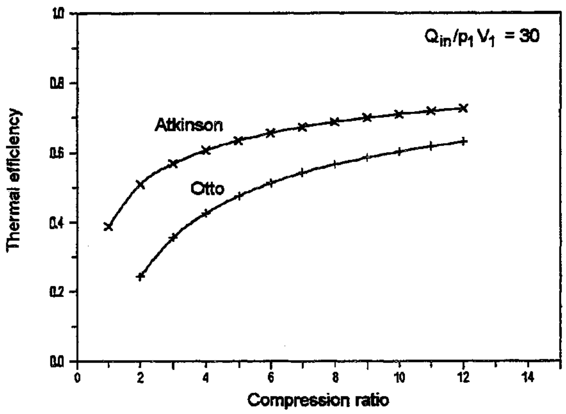

Figure 3. Thermal efficiencies of Atkinson and Otto cycles against compression ratio

DESIGN OF THE DOUBLE HELICAL SCREW ENGINE

GENERAL LAYOUT - The engine consists of two pairs of rotary helical screw type compressors, one rotating in its designed direction and responsible for the intake and compression strokes while the other coupled to the first and arranged to rotate in the reverse direction to perform the expansion and exhaust operations. Figure 4 above shows the test rig layout.

SELECTION OF AIR-ENDS - Commercially available screw compressor air-ends were used to make the principal components of the engine. The primary consideration factors for selecting the air-ends were cost and weight. A lighter engine would be much easier to handle and install on a test rig. The Sigma-0 air-end by Kaeser, one of the smallest and least expensive commercially available, was selected. Figure 5 shows the profile of the rotors. In order to test the feasibility of the double helical screw concept alone without consideration of the Atkinson cycle principle, it was decided that the same size of expansion and compression machine be used.

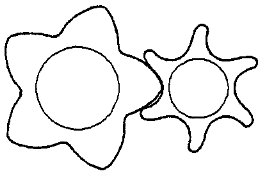

Figure 5. Profiles of male (left) and female (right) rotors

DESIGN OF COMPRESSION SIDE EXIT PORT - As shown in Fig. 4, the central plate is the component linking the compression and expansion sides of the engine. This plate is to provide an exit port for the compression side. One face of the central plate covers the exit end of the compression side. The profile and position of the exit port is shown in Fig. 6 below. Together with the plate valve riveted on the male rotor, the port position could result in a compression ratio range suitable for Diesel applications. This compression ratio range is higher than the normal range for conventional petrol engines as it is anticipated that the leakage losses will reduce the effective compression ratio. Overcoming leakage losses is the major challenge in the process of developing a successful double helical screw engine.

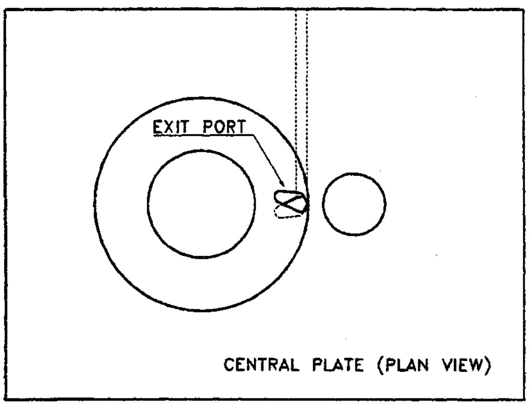

Figure 6. Profile and position of compression side exit port

Figure 7 shows the profile of the plate valve. The plate valve was found to be necessary because the nature of the rotor profile is such that a flow path to the intake side is formed when the exit port is relocated for a higher compression ratio than the manufacturer's own designed ratio. It can be eliminated with custom designed rotors.

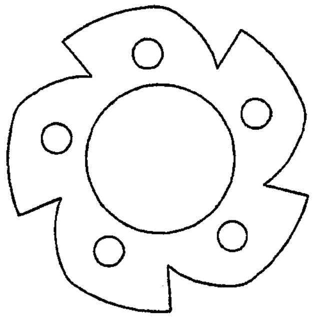

Figure 7. Profile of plate valve

DESIGN OF EXPANSION SIDE ENTRY PORT - The other face of the central plate covers the entry end of the expansion side. The angular position of the expansion rotor relative to that of the compression rotor, the profile and the position of the entry port and the profile of the plate valve were designed such that at the instant when the compression side exit valve is just opened, corresponding expansion space will be at its minimum volume. Continued rotation then results in the compression space emptying its fuel-air mixture via its exit port, through the central plate and via the entry port (leading to the expansion side) into the expansion space. Finally, at the instant when the compression space is at its minimum volume, the expansion side entry port is just closed. In the first prototype, the compression and expansion air-ends were identical and thus the profile and position of the entry port were simply horizontally symmetrical about the centreline of the central plate.

DESIGN OF COMBUSTION CHAMBER - The combustion chamber is formed by the combination of the compression space (compression side) near the end of its compression operation, the central plate, and the expansion space (expansion side) near the beginning of its expansion operation. During the process of combustion, the working fluid is not stationary but in motion from the compression side to the expansion side. Figure 8a shows a section of the combustion chamber at the start of the transfer process of the fuel-air mixture while Fig. 8b shows the end of the transfer process. Thus combustion occurs under near isovolumetric conditions. Also the 'pseudo' top dead centre is not instantaneous but of a finite and design-able duration. One advantage of this is that the combustion process can be close to the ideal constant volume condition without the need for excessive ignition advance, particularly at high engine speed.

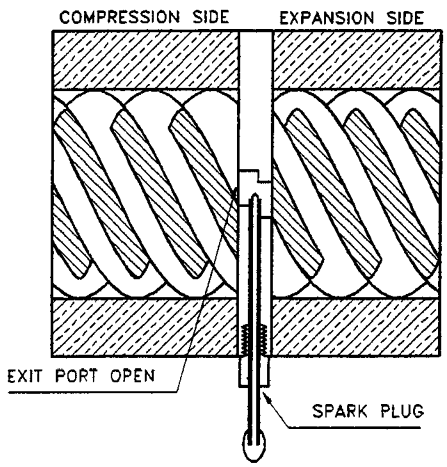

Figure 8a. Combustion chamber, start of transfer process

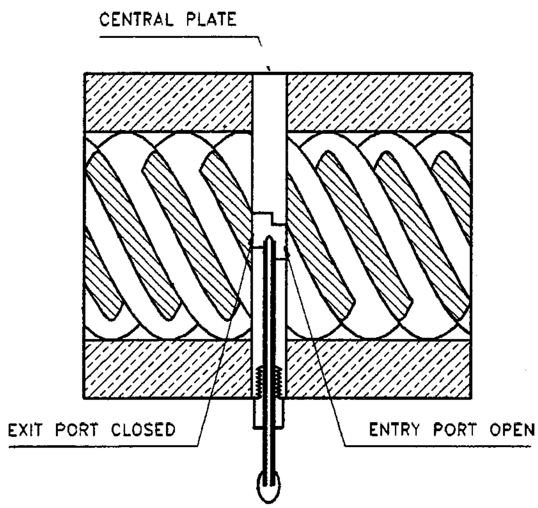

Figure 8b. Combustion chamber, end of transfer process

To minimise the dead volume in the central plate, it was made as thin as possible. Due to the thinness of the central plate, it was impossible to contour the region between the exit and entry ports for smooth fluid flow. This is a disadvantage which will be improved in future designs.

LOCATION OF SPARK PLUG - Figures 8a and b above also show the location of the spark plug. It is easily apparent that an ordinary spark plug would not be able to fit into the 12 mm thickness of the central plate. Thus a custom made spark plug, discharging to the wall of the combustion chamber, was used. Figure 9 shows the designed spark plug.

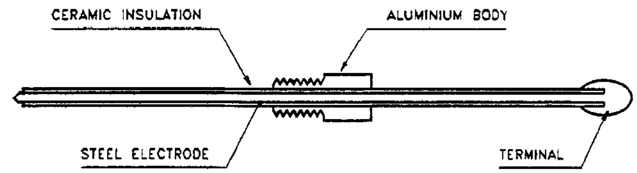

Figure 9. Custom designed spark plug

COOLING AND LUBRICATION - It was originally intended that the expansion rotors be hollowed out and oil circulated within to cool them. For simplicity, it was decided that no dedicated cooling system be provided, leaving the engine to rely only on air blown externally along the casing for cooling. During the development of the engine, care was taken that the rotors did not seize from overheating. Lubrication was provided as in two-stroke engines by feeding oil into the inlet manifold.

VOLUMETRIC CHARACTERISTICS - Figure 10 shows the theoretical volume against shaft angle for one male-female pair of helical space.

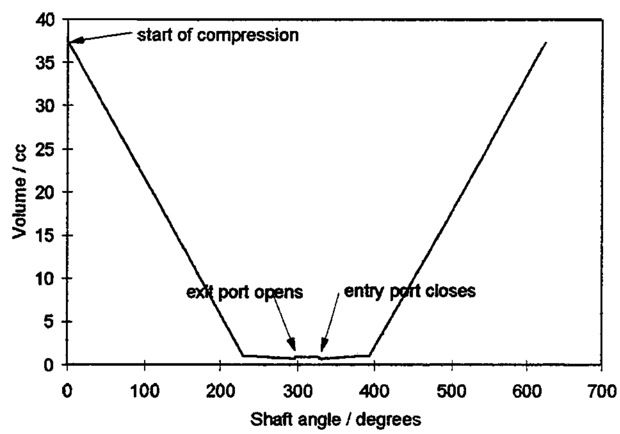

Figure 10. Volume vs. shaft angle (theoretical)

CARBURETOR - An SU-type variable jet motorcycle carburetor, originally designed for feeding 200 cc of 4-stroke displacement was used. The selection was made on the basis of comparison between the free air delivery of the air-end at the desired rotating speed of 3600 rpm and the inlet volumetric flow rate of a 200 cc 4-stroke engine at a typical 8000 rpm. An oil inlet was provided on the intake manifold for injecting lubricating oil into the fuel-air mixture stream.

SPARK CONTROL - A timing pulley attached to the 6-chamber female rotor of the expansion side was used to drive a 3-cylinder conventional distributor at a 2:1 gear ratio. As there was only one spark plug, the rotating arm and the topmost cover of the distributor was removed. The secondary terminal from the ignition coil was connected directly to the spark plug. The contact point gap was adjusted to a smaller distance to cope with the increased rpm at which it has to operate. For example, running the engine at 3600 rpm requires a distributor speed equivalent to that required by a 4-stroke, 3-cylinder engine at 14400 rpm! The centrifugal and vacuum advance mechanisms of the distributor were removed and timing can be adjusted by external means.

EXPERIENCE GAINED

SEALING - It was found that the single most important requirement for the engine to stand any chance of success is proper sealing of the working volume. Despite the high designed compression ratio, the effective compression required for ignition of the fuel-air mixture was not achieved due to leakage.

Until manufacturing processes are able produce parts that do not deviate from the designed profile, flexible seals have to be used with the rotors.

CHARGE VELOCITY - In an attempt to improve the ignitability of the fuel-air mixture, liquefied petroleum gas was used temporarily in place of gasoline. The unsuccessful outcome of this method led to the suspicion that the velocity of the fuel-air mixture during the transfer and combustion process may be too high for successful combustion. Unlike piston engines, the fuel-air mixture in this engine has to be transferred to the expansion side when it reaches the end of the compression stroke. It is in a process of flow during ignition and combustion. Under certain conditions, the flow velocity in the prototype engine may reach an average value that is high enough to blow out the flame.

Through proper consideration of the compression ratio, port position and helix angle of the rotors, the required flow velocity can be designed for the engine. Techniques used to reduce the local air velocity in gas turbines such as the flameholder may also be applied.

FUTURE DEVELOPMENTS

Building upon the experience gained from the first prototype, the second prototype will incorporate the improvements given below.

CUSTOM DESIGNED ROTORS - The need for near perfect sealing renders the Kaeser rotors unusable. They were originally designed to be run immersed in oil and so had relatively large clearances. Custom designed rotors with provision for conformable seals will be used. The rotors will also be designed for proper flow velocity.

ATKINSON CYCLE PRINCIPLE - In the first prototype, the compression and expansion ratios were the same. In the second prototype, the Atkinson cycle principle will be adopted by using a larger set of expansion rotors.

OTHER IMPROVEMENTS - Being a first prototype, the dimensional control and alignment of the engine was poor, thus resulting in high friction and poor performance. Proper engine optimisation will be carried out when building the second prototype.

Other new technologies such as the spark gasket and sputtered dry lubricants will also be tried.

SUMMARY

The double helical screw internal combustion engine was conceived with the hope of overcoming many of the problems inherent in piston engines, in particular, their inability to adopt the Atkinson cycle principle and their non-rotary output.

The first prototype engine built was to test the feasibility of the double helical screw concept without consideration for the Atkinson cycle principle, using two helical screw machines of the same size.

It was found that the commercially purchased machines, originally designed to run immersed in oil, had poor sealing and thus could not achieve the typical compression required for successful ignition of the fuel-air mixture.

It was suspected that the flow velocity during the combustion process may be too high for favourable combustion.

Building upon the experience gained from the first prototype, a second engine will be built. This new engine will use custom designed rotors to eliminate the sealing problem and provide a proper flow velocity. It will also incorporate the Atkinson cycle principle.

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SYMBOLS

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- η thermal efficiency
- r compression ratio
- r_p pressure ratio
- k ratio of specific heats
- p pressure
- V volume
- Q heat

SUBSCRIPTS -

- O Otto cycle
- A Atkinson cycle
- in input