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ABSTRACT

The hypocycloid mechanism, in its basic or modified version, can be used in reciprocating machines as an alternative solution to the conventional crankshaft-connecting rod system. This kinematic device is quite old in concept, but still gives rise to some research since it has some positive mechanical features, and it allows several functions to be easily combined.

In the hypocycloid mechanism in fact the connecting rod has a perfectly sinusoidal straight-line motion, a characteristic that theoretically reduces the piston side thrust to zero and permits a perfect balance of inertia forces without auxiliary rotating shafts. Moreover, by means of a second head with suitable seals that prevent leakage to the crankcase, the lower portion of the cylinder can be used as a second working chamber without using the crosshead thus reducing the dimensions of the double acting machine.

The authors have undertaken theoretical and experimental research in order to evaluate the positive features, in particular those concerning the possibility of the combination of functions, and to compare these characteristics with the evident shortcomings. This has led to the design and construction of a double acting compressor, whose performances have been compared with those of a traditional compressor.

A perfectly balanced two-stroke engine, with the scavenging pump separated from the crankcase, has also been constructed and tested.

INTRODUCTION

The traditional slider-crank mechanism, that is usually used in alternative machines for the transmission of motion between the piston and the crankshaft, is without doubt very simple and efficient. For this reason it has remained almost unchanged since its introduction.

Nevertheless in the past and even nowadays, different kinematic systems have been proposed and studied because of their interesting features which can allow, for example, different functions to be combined and which also present some positive mechanical characteristics.

One of these kinematic systems is a planetary gear which is called the hypocycloid mechanism and which, in the traditional scheme presented in Fig. 1, is composed of an internal gear (a) and a planet gear (b) connected by an arm (c) to the main shaft (d).

As the diameter ratio between the ring gear and the planet is 2:1, each point of the planet reference diameter describes a straight motion; for this reason the big-end axis of the connecting rod is centered on a point of this reference diameter. The straight motion of the connecting rod has the same axis as the cylinder.

The hypocycloid mechanism has some interesting features: first of all, the piston has a perfectly sinusoidal motion, for this reason the second order inertia forces are completely null. This feature allows the perfect balance of the machine with any number of cylinders and without additional countershafts.

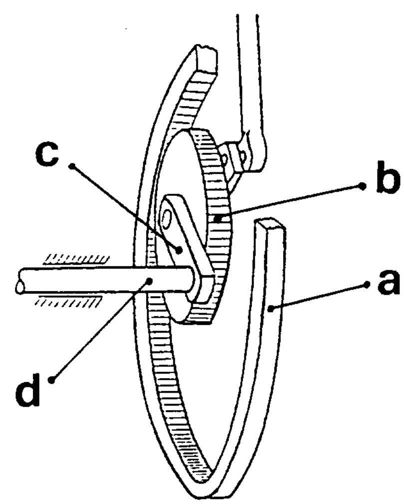


Figure 1. Traditional hypocycloid mechanism.

In [1] and [2], for example, the “Big Bearing” version [3] of this mechanism is used for a perfect balanced two-stroke engine to power a chain saw. Vibrations are very low in comparison to those of a conventional chain saw, even though the noise level is slightly larger.

A “Big Bearing” solution with a lateral slider to reduce the gear strength, is presented in [4]. The authors propose using this mechanism in marine engines in order to eliminate the crosshead and, at the same time, to reduce the dimensions of the engine.

Besides the possibility of perfectly balancing the machine, it has been shown in [5], [6] that sinusoidal spark ignition engines have a slower motion at the TDC and this seems to be positive as far as thermodynamic efficiency is concerned. In fact the piston spends more time in the combustion zone and this can lead to higher peak pressure and temperature.

On the other hand, the absence of piston side thrust, due to the perfectly straight motion of the connecting rod, eliminates the piston slap and allows a remarkable reduction of the friction between the piston and the slider thus augmenting mechanical efficiency [7], [8]. An additional benefit, due to side thrust reduction, is the possibility of simplifying the piston rings, whose crevices are considered to be the main cause of HC emissions [7].

The hypocycloid mechanism has also some shortcomings, that are mainly due to the gear transmission which, in this application, is neither easy to design nor to construct, is highly and unevenly stressed, and is noisy under operative conditions. Moreover, the crank for a machine of the same stroke, is half that of a traditional one and this leads to the fact that the average strength to the crank pin has to be doubled in order to obtain the same torque. Finally, the relative angular speed of the pinion gear-crank pin coupling is double that of the traditional slider-crank mechanism.

Several different solutions were proposed for this mechanism in the past such as the Modified Hypocycloid Mechanism [7] which presents better characteristics as far as gear transmission is concerned but is much more complicated. There are also some kinematic systems without gears such as the trocoidal mechanism [9] which does not have the positive feature of a null side thrust. Finally some solutions have been put forward with eccentric gears such as the Stiller-Smith mechanism [10] and the Ishida mechanism [11], [12], [13], [14], [15].

A very important feature of the hypocycloid mechanism consists of the fact that, as previously mentioned the connecting rod has a perfectly straight-line motion, a characteristic that permits the use of the lower portion of the cylinder as a second working chamber. This feature could also be obtained in a traditional crank-slider by means of a crosshead though this solution presents well known limitations and shortcomings. It is very simple to obtain multiple acting machines with the hypocycloid mechanism (Fig. 2), such as double acting reciprocating compressors (a), two-stroke engines with the scavenging pump separated from the crankcase (b) and four-stroke engines with a built-in surcharging compressor (c).

The positive feature regarding the possibility of the combination of functions without using the crosshead, is analyzed and experimentally tested in this paper. Firstly, a double acting compressor has been designed, built and tested on an experimental bench and its performances have been compared with those of the same compressor during a single acting operation [16]. Secondly, a two-stroke engine, with the scavenging pump separated from the crankcase and with perfect balance, has been constructed and tested. In this machine, as in the previous one, the lower portion of the cylinder is separated from the crankcase by means of suitable seals that prevent leakage. Lubrication of the liner-piston assembly and of the gears and bearings in the crankcase are therefore completely separated; this allows one to significantly reduce the oil quantity in the fuel.

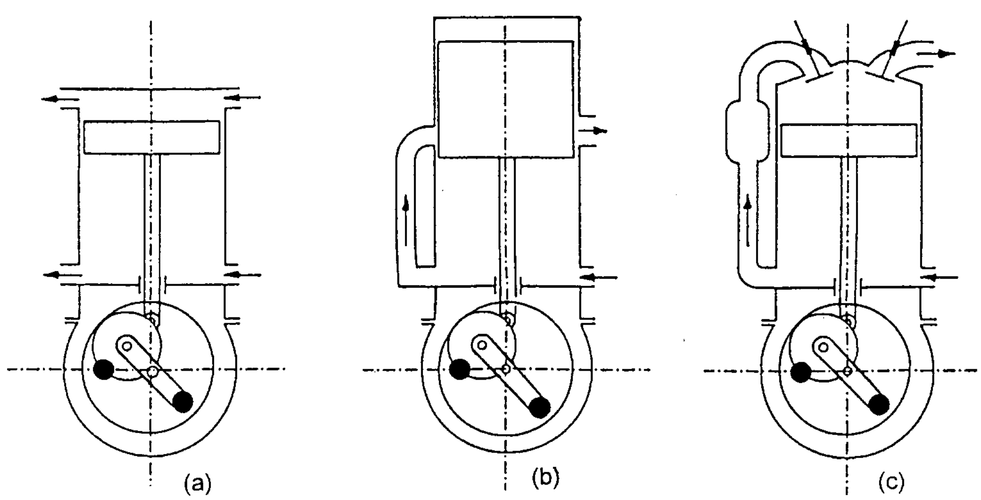


Figure 2. Double acting hypocycloid machines: double acting reciprocating compressors (a), two-stroke engines with the scavenging pump separated from the crankcase (b), four-stroke engine with built-in surcharging compressor (c).

KINEMATICS AND DESIGN

As previously mentioned, in a hypocycloid mechanism (Fig. 3) where the diameter ratio between the ring gear and the planet is 2:1, each point A of the planet reference diameter describes a straight motion whose length is equal to the reference diameter of the ring gear. It is therefore possible to connect point A to the big-end of a connecting rod whose motion is transmitted to the piston of a volumetric machine.

The perfectly straight-line motion of the rod and the presence of the gears theoretically give rise to a null piston side thrust. This feature offers a reduction in the piston assembly friction and piston slap and, from a constructive point of view, allows one to completely eliminate the wrist-pin.

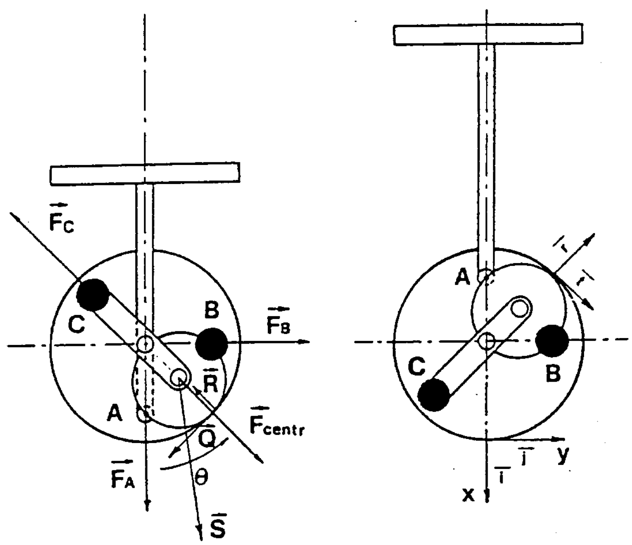


Figure 3. Kinematic scheme of the hypocycloid mechanism.

Furthermore, the piston in a hypocycloid machine does not have the task of the sliding bearing as in a slider-crank engine. For this reason the piston skirt can be greatly reduced or even eliminated for a compressor or a four-stroke engine. As far as a two-stroke engine is concerned, the skirt has the role of a shutter which times the opening and shutting of the inlet, exhaust and transfer ports. For this reason, it can only be lightened and slightly reduced.

From a kinematic point of view, it is important to point out that if the crank turns with an angular speed ω the pinion will turn with an angular speed $-\omega$; on one hand this is negative because the relative angular speed of the pinion gear crank pin coupling is double that of the crankshaft, but on the other it allows a perfect balance of the first order inertia forces. The balancing can be performed by positioning a mass in B equal to the oscillating mass acting in A [16]. The composition of the two inertia forces due to mass A and B, gives a purely rotating inertia force which can be equilibrated by means of a mass C.

The design of the mechanical features of a hypocycloid machine must incorporate careful and precise calculations and dimensions due to the very high and complex loading of the gears, bearings and pins. Firstly, the gear pin load should be taken into consideration. The strength on this pin is composed of the inertia force of oscillating masses and of the gas forces of the upper and lower chambers of the cylinder. The design is even more difficult because the system is undetermined, from a static point of view, due to the multiple bearing points.

The force acting on pin A can be expressed as follows:

$$\vec{F}_A(\theta) = F_A(\theta)\cdot\vec{i} = \left[m_{alt}\omega^2 \frac{c}{2}\cos\theta + \left(p_s(\theta) - p_i(\theta)\right)A_s \right]\cdot\vec{i} \qquad \text{(Eq. 1)}$$

When a perfect balance is required, it is necessary to put a counterweight equal to m_{alt} in B, this mass determines the following inertia force:

$$\vec{F}_B(\theta) = F_B(\theta)\cdot\vec{j} = \left[m_{alt}\omega^2 \frac{c}{2}\sin\theta \right]\cdot\vec{j} \qquad \text{(Eq. 2)}$$

From the summation of the moments, calculated from the crank pin reference axis, the tangential gear tooth load can be expressed as:

$$\vec{Q}(\theta) = Q(\theta)\cdot\vec{t} = \left(F_A(\theta)\sin(\theta) - F_B(\theta)\cos(\theta)\right)\cdot\vec{t} = \left(p_s(\theta) - p_i(\theta)\right)A_s\sin(\theta)\cdot\vec{t} \qquad \text{(Eq. 3)}$$

It is important to point out that the inertia force of mass B annul the positive action of the piston and rod inertia forces which, in the high pressure phase, counteract the thrust due to the gas loads.

The torque exerted on the crankshaft can be expressed by the following expression:

$$T(\theta) = Q(\theta)\frac{c}{2} \qquad \text{(Eq. 4)}$$

The radial components of the gear tooth load and the rotating mass inertia force are:

$$\vec{R}(\theta) = R(\theta)\cdot\vec{r} = -\left|Q(\theta)\right|\cdot tg(\Theta)\cdot\vec{r} \qquad \text{(Eq. 5)}$$

and

$$\vec{F}_{centr} = F_{centr}\cdot\vec{r} = m_c\cdot\omega^2\frac{c}{4}\cdot\vec{r} \qquad \text{(Eq. 6)}$$

To balance the actions of the rotating mass m_c and the oscillating masses m_{alt}, positioned in A and B, it is necessary to design a mass in C whose rotating inertia force is as follows:

$$\vec{F}_C = -\left(m_c + 2m_{alt}\right)\cdot w^2\frac{c}{4}\cdot\vec{r} \qquad \text{(Eq. 7)}$$

Finally, the load on the crank pin (or on the two crank pins in the case where the mechanism is designed symmetrically on two bearings) can be determined by means of the following equation:

$$\left|\vec{S}(\theta)\right| = \sqrt{\begin{array}{l}\left(F_c + R(\theta) + F_A(\theta)\cos(\theta) + F_B(\theta)\sin(\theta)\right)^2 + \\ +\left(Q(\theta) + F_A(\theta)\sin(\theta) + F_B(\theta)\cos(\theta)\right)^2\end{array}} \quad \text{(Eq. 8)}$$

DOUBLE ACTING COMPRESSOR

DESIGN OF THE PROTOTYPE – The single cylinder double acting hypocycloid compressor was designed in order to construct a machine with n=3000 rpm maximum angular velocity, p=9 bar maximum cylinder pressure and V=170 cm^3 single cylinder displacement.

The compressor was constructed using some parts of an existing commercial compressor such as the cylinder block and head, valves, crankcase and the bearings.

A single bearing asymmetrical solution was considered to be suitable to construct this machine because of the light acting load. Furthermore, for the sake of simplicity, only a partial balance of the inertia force was performed.

The design of mechanical parameters has been performed on the basis of the previously presented load analysis starting from the load on the planet pin FA(θ) which was calculated by referring to a theoretical pressure cycle. The compressor was considered in its single acting operation because this case is the most stressing.

The maximum values of the planet eccentric pin load have been used for the bending and shearing stress determination that is necessary for the design of the pin itself and for the design of the connecting rod and rod-piston bracing.

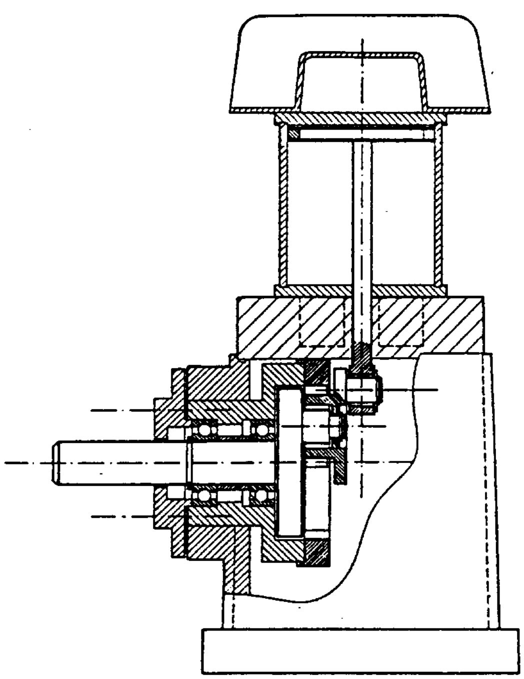


Figure 4. Double acting compressor.

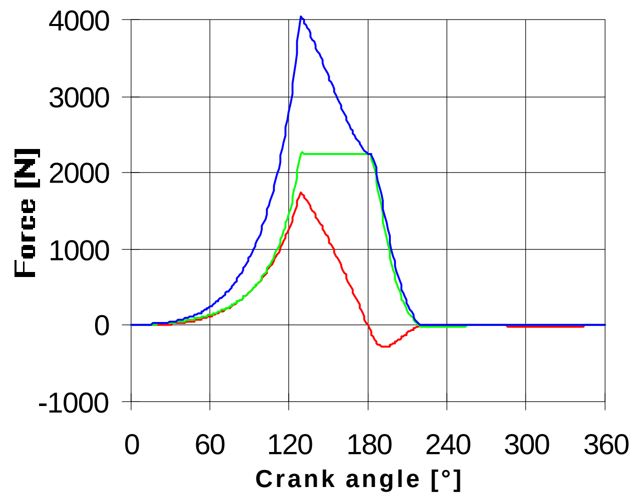


Figure 5. Behaviour of the main forces during a traditional cycle (n=500 rpm, β=9).

The tangential gear tooth load Q(θ) was then determined for the design of the gear module and thickness; finally, the values of the crank pin load S(θ) were determined which, along with the rotating inertia force Fcentr, the planet eccentric pin load $F_A(\theta)$ and the tangential gear tooth load Q(θ) and radial load R(θ), allow the bending and shearing stress determination to verify the crank pin and the shaft bearing.

The maximum load conditions are those of low angular speed and high delivery pressure. Figure 5 shows the behaviour of $F_A(\theta)$, Q(θ) and S(θ) for a theoretical pressure cycle when the angular velocity is n=500 rpm and the compression ratio is b=9.

The final main design parameters are given in Table 1:

Table 1. Compressor Design Parameters

Stroke	60 mm
Bore	60 mm
Displacement volume	170 cm^3
Connecting rod diameter	8 mm
Gear module	1.25
Pressure angle	20°
Reference diameter of the ring gear	60 mm
Reference diameter of the planet	30 mm
Number of teeth of the ring gear	48
Planet number of teeth	24
Face width	10 mm
Oscillating mass	101 g

EXPERIMENTAL TESTS – The experimental tests were firstly oriented to the detection of the performances and of the mechanical efficiency of the simple acting compressor [16]. In the following part of the research, which is reported in this paper, the double acting functioning was thoroughly tested.

The experimental bench used for the tests is schematically represented in Fig. 6.

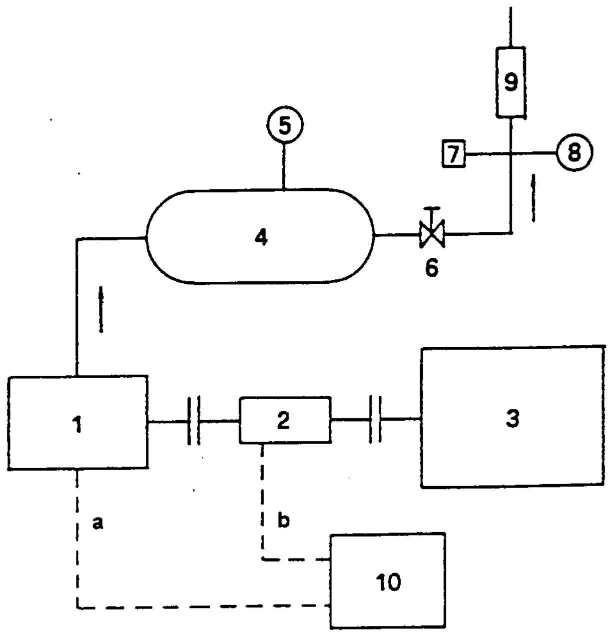


Figure 6. Layout of the compressor experimental apparatus.

This bench is composed of a motor (3) that allows the variation of the driving angular speed, which is controlled by means of a tachometer coupled to a Hottinger T30 FN driving torque transducer (2). The over-all pressure is measured by means of a manometer (5) installed in the plenum (4) while a Kistler 6051 piezoelectric transducer allows the detection of the instant pressure in the cylinder. The over-all pressure can be changed by means of the valve (6) The flow rate is determined by means of a flow meter (9), while the temperature and the pressure of the fluid upstream to the flow meter which are necessary for the determination of the effective mass flow, are detected by means of a manometer (8) and a thermometer (7). The in cylinder pressure (signal a) and the shaft torque (signal b) are sampled and recorded on a computer by means of an acquisition system AVL (10).

Figure 7 shows the compressor operating maps where the pressure ratio versus the corrected mass flow rate is presented; Several tests were performed for different angular speeds ranging from 500 to 2500 rpm. The first map (a) is related to the single acting functioning while the second map (b) is related to the double acting compressor.

Figure 8 permits a comparison of the volumetric efficiency versus the angular speed diagrams for the single acting (a) and the double acting (b) compressor. Each curve is plotted for a constant pressure rate; several pressure rates were considered ranging from β=2 to β=7.

The mechanical efficiency was then determined for each test performed, through the following equation:

$$\eta_m = \frac{P_i}{P} = \frac{P - P_f}{P} \qquad \text{(Eq. 9)}$$

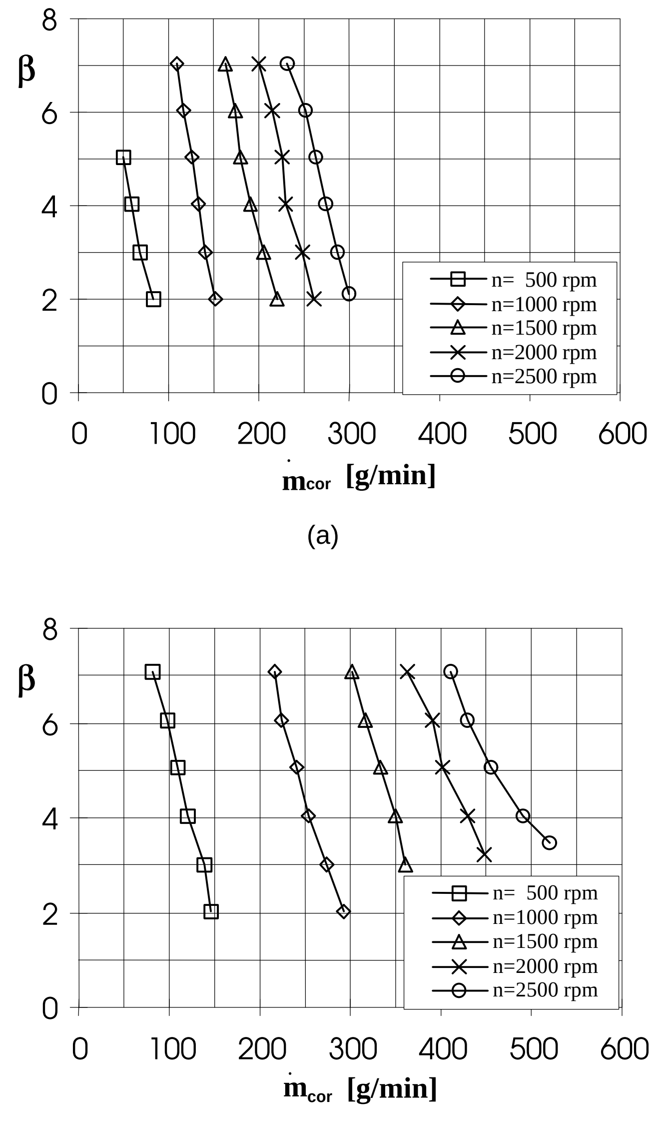
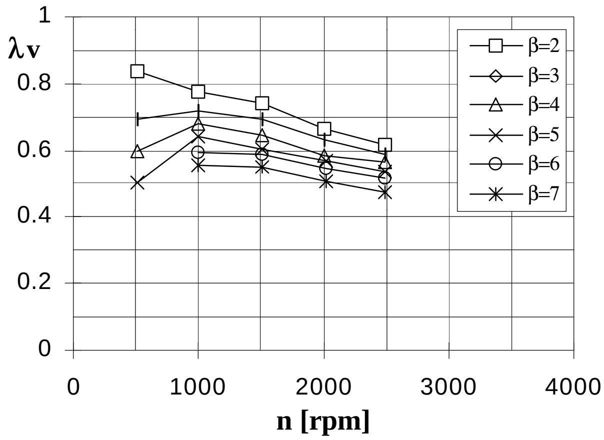


Figure 7. Compressor operating maps. Pressure ratio versus the corrected mass flow rate for the single acting (a) and the double acting (b) compressor.

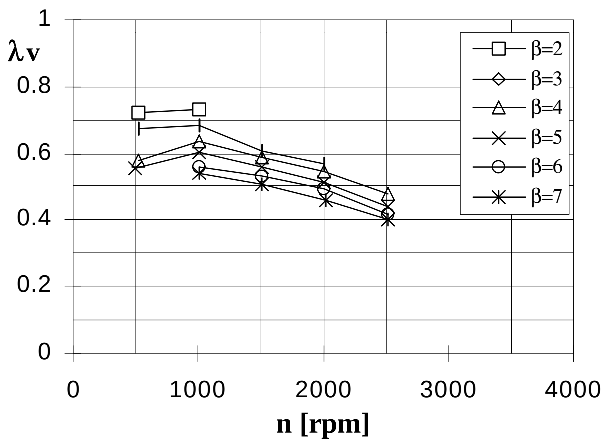
The shaft torque measured with the dynamometer was used to determine the absorbed power (P) while the indicated power (P_i) was calculated by means of the instant cylinder pressure determined by means of the piezoelectric transducer that was mounted in the compressor head.

A comparison between the single acting and double acting compressor mechanical efficiencies is presented in Fig. 9. The figure shows the behaviour of the mechanical efficiency versus the angular velocity for several pressure ratios, ranging from 2 to 5.

From an analysis of the results it is clear that, during the double acting functioning, the power absorbed is almost doubled while the friction power is almost unchanged. For this reason the mechanical efficiency rises remarkably.



(a)



(b)

Figure 8. Volumetric efficiency versus angular speed diagrams for the single acting (a) and the double acting (b) compressor.

One should note that the mechanical efficiency behaviour is almost independent of the delivery pressure and of the angular speed; This is probably connected to the fact that the friction power is due to the gas load, inertia load and Coulomb friction load (principally rings and seals); among these, the gas load is the most important even at 2500 rpm. The inertia forces are very low because the oscillating mass is very small, due to the very light design of the piston and the connecting rod.

TWO-STROKE S.I. ENGINE

DESIGN OF THE PROTOTYPE – A perfectly balanced two-stroke S.I. engine was designed with the scavenging pump separated from the crankcase. As in the case of the compressor, the lower portion of the cylinder is separated from the crankcase by means of suitable seals (Angst Pfister lubriflon, bronze filled modified PTFE) thus preventing leakage. The lubrication of the gears and bearings in the crankcase is not a force-feed system and is actually performed by means of a splash and oil mist lubrication; the lubrication of the liner-piston assembly is obtained by means of oil added to the fuel. As the gears and the bearings are lubricated separately, it is possible to significantly reduce the oil quantity in the fuel.

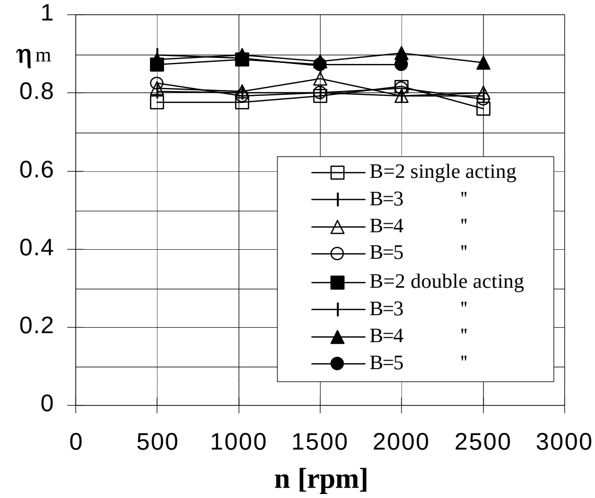


Figure 9. Mechanical efficiency versus angular speed diagrams for single acting and double acting compressor.

A major problem was due to the fact that the oil sump was very small and the natural cooling of the oil was not sufficient for the operating conditions at high angular speed.

The engine was constructed using parts of an existing Cagiva Mito 125 engine, a single cylinder, water-cooled, two-stroke, spark ignition engine.

The parts of the original engine used for the prototype are: the aluminum cylinder block, the aluminum cylinder head, the flywheel, the electronic ignition system and the starting system.

The engine was designed with the purpose of constructing a machine with nmax=10000 rpm maximum angular velocity, p=70 bar maximum cylinder pressure and V≈125 cm^3 of displacement, as in the original Cagiva engine.

On the basis of the load analysis carried out from the theoretical pressure cycle and by means of the same considerations used for the double acting compressor, it was necessary to design the hypocycloid mechanism in its double bearing symmetrical scheme; the second gear transmission, even though not necessary for the delivery of power, is very important for the torsion stiffness of the system and its fatigue behaviour. In fact, even though, on one hand, the axial dimension of the gears increases the span between the main bearings and therefore the bending stress on the big-end pin, on the other this solution permits one to obtain a very small angular clearance with better working conditions of the bearings.

A complete balance of the inertia force was also performed and designed.

The final design parameters are given in Table 2:

Table 2. Engine Design Parameters

Stroke	50 mm
Bore	55.5 mm
Displacement volume	121 cm^3
Connecting rod diameter	13 mm
Gear module	2
Pressure angle	25°
Reference diameter of the ring gear	50 mm
Reference diameter of the planet	25 mm
Number of teeth of the ring gear	24
Planet number of teeth	12
Face width	15 mm
Oscillating mass	370 g

Figure 10 shows a sectional view of the constructed hypocycloid engine; from this figure the symmetry of the crankshaft-gear assembly is evident. It is also possible to see the septum that separates the lower portion of the cylinder from the crankcase. Leakage is prevented by means of two PTFE rings.

The oblique oil ducts for planet bearing lubrication are also represented in Fig. 10.

Figure 11 represents the piston and the rod assembly which are rigidly connected by means of a screw. In this case, but not in the case of the double acting compressor, it was necessary to maintain the skirt which has the role of opening and closing the engine ports.

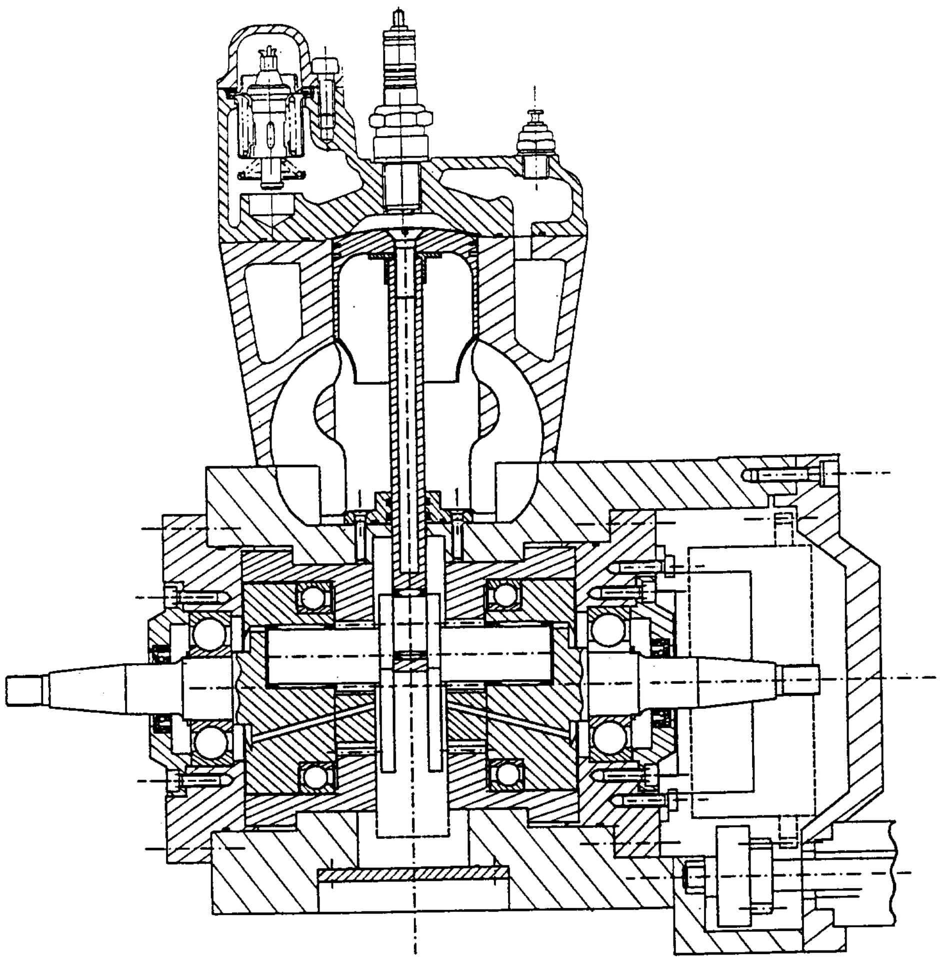


Figure 10. Side view of the two-stroke, S.I., water cooled, hypocycloid engine.

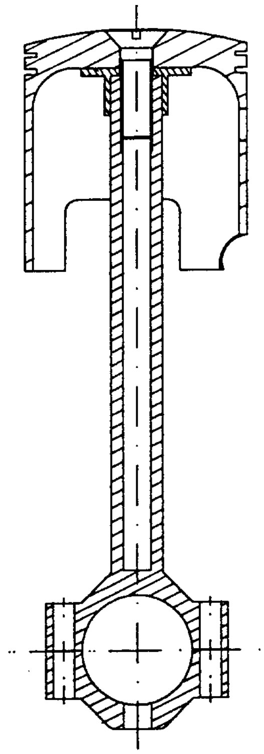


Figure 11. Piston - connecting rod assembly.

The piston was constructed in aluminum and, as it is not subjected to side thrust, the thickness of its skirt was reduced to 1.9 mm (3 mm in the original piston). The weight of piston, rod, rolling bearing and big end pin (m_{alt} =370 g) is remarkably reduced if compared to the weight of the same parts of the original engine.

Figure 12 schematically represents the crank which is composed of the two planet gears and pins and of the big end pin whose axis passes through a point of the planet reference circle which connects the two gears. The big end pin axis is therefore eccentric in comparison to the axis of the planets and its eccentricity is ε =c/4 where c is the stroke of the engine. The mass malt, whose center of gravity is situated in B (see Fig. 3), is made up of two counterweights whose masses are equal to $m_{alt}/2$ each and whose barycenter is at a distance ε from the axis of the planets in a symmetric position with respect to the big end pin axis.

Figure 13 represents the shaft with an eccentric hole where the planet pin is inserted with the support of roller bearings. The starting tests, performed on a dynamometric bench, have shown the correct behaviour of the engine at different angular speeds and different loads. Two different carburetor nozzles of the main metering system were tried for the engine set up: the 148 and the 145 nozzles (where 148 and 145 are the nozzle orifice diameters in 1/100 mm). The 148 nozzle and the trapped compression ratio ρ=5.6 were used in the first two experimental tests performed (Fig. 14). This trapped compression ratio, lower than that of the original Cagiva Mito 125 engine where this parameter assumes the value ρ=6.6, was chosen to limit the engine load during its set up. The tests of Fig. 15 were performed with the original trapped compression ratio (ρ=6.6). This value was obtained by inserting a washer between the piston and the rod.

The test of Fig. 15 (b) was performed up to a revolution speed of n=6000 rpm. In these conditions the lubricating oil in the crankcase rapidly became overheated. It is important to remember that the lubrication of the crankcase mechanism is performed by means of a splash system without any direct cooling of the oil in the sump, therefore it is difficult to avoid thermal overload, especially at high speeds.

In future research it will be necessary to design an efficient oil cooling system in order to test the engine at higher speeds.

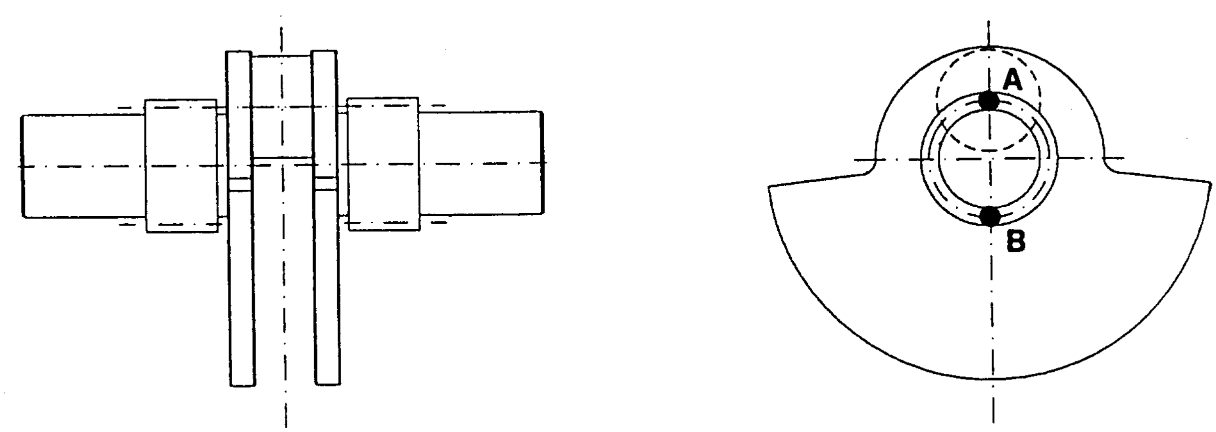


Figure 12. Planet crank composed of two planet gears and the big-end pin.

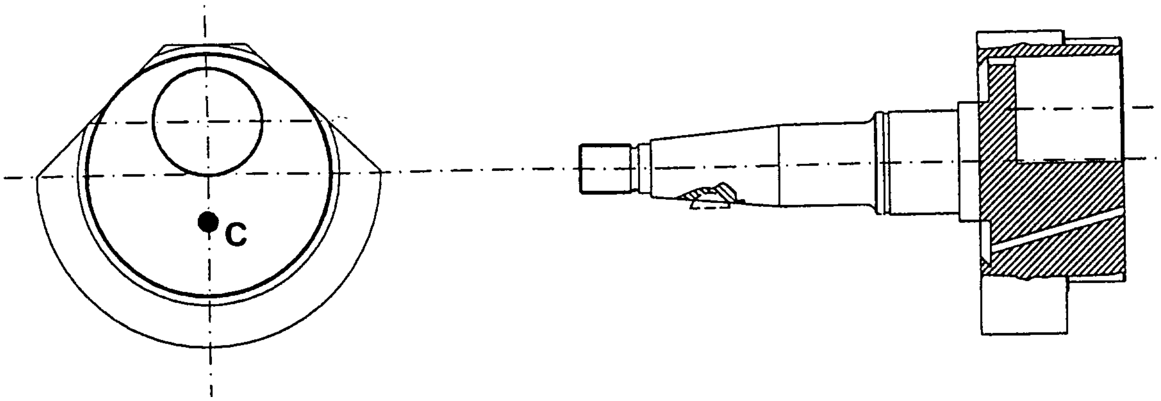
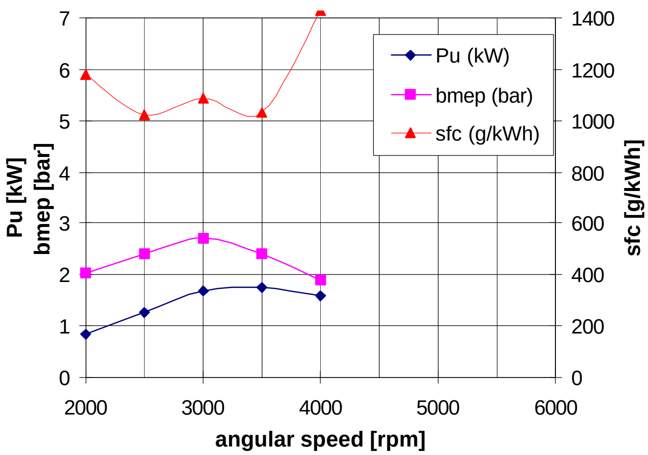
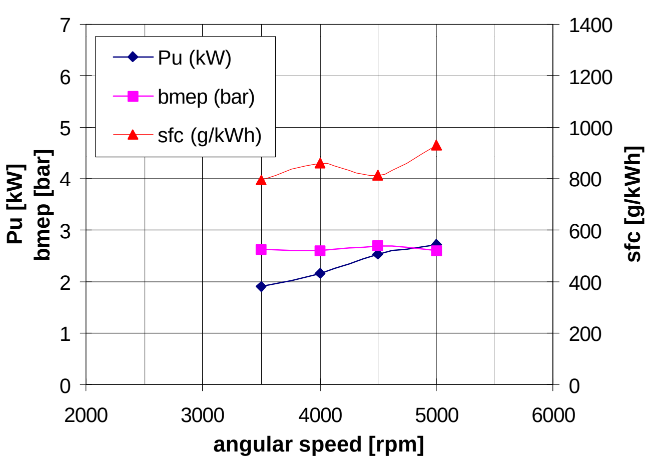


Figure 13. Main shaft with the eccentric hole where the planet pin is inserted.

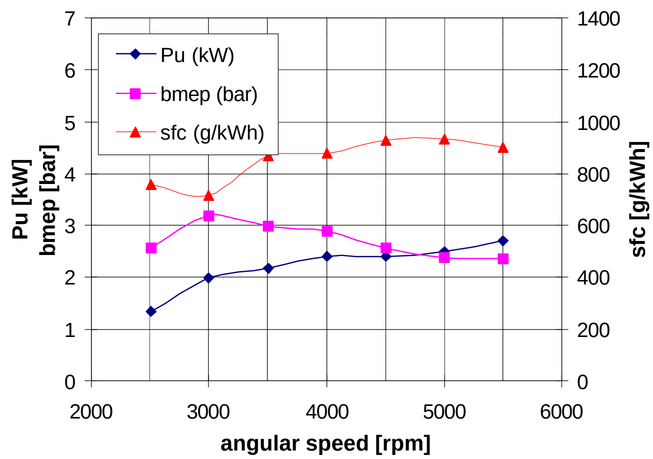


(a)

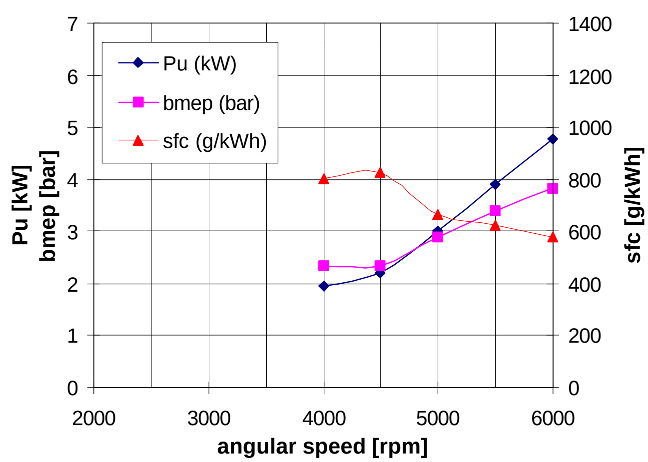


(b)

Figure 14. Carburetor nozzle 148, trapped compression ratio ρ=5.6: (a) low load test, (b) medium load test.



(a)



(b)

Figure 15. Carburetor nozzle 145, trapped compression ratio ρ=6.6: (a) low load test, (b) medium load test.

CONCLUSIONS

The possibility of constructing double acting machines that are simplified thanks to the use of a hypocycloid mechanism has been analyzed in detail in this research. Firstly, a double acting compressor was designed and constructed; this compressor was then tested at different angular speeds and pressures in the chamber. Secondly, a perfectly balanced two-stroke engine, with the scavenging pump separated from the crankcase, was constructed and tested. The possibility of making this engine function with a very small amount of oil in the fuel, which is only necessary for the lubrication of the liner-piston assembly, was investigated.

The compressor tests have shown the good behaviour of the hypocycloid mechanism with a very large increase of the mechanical efficiency in comparison to the single acting compressor. A single bearing asymmetrical solution was used to construct this machine and only a partial balance of the inertia force was performed.

The design of the two-stroke engine presented some difficulties. Most of the problems were due to the seals between the lower part of the cylinder and the crankcase, the design and construction of the gears, the rolling bearings and the lubrication ducts. The engine has shown good performances that are comparable to those of the original Cagiva engine, but some efforts should be undertaken to design an efficient oil cooling system if higher revolution speeds are to be reached.

ACKNOWLEDGMENTS

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NOMENCLATURE

η_m**:** mechanical efficiency
ω**:** angular speed
A_s**:** piston area
c: stroke
b: bore
T: torque
$\dot{m}_{cor}$ **:** standard flow rate
m_{alt} **:** oscillating mass
mc: rotating mass
p_i**:** inferior chamber pressure
p_s**:** superior chamber pressure
β**:** compression ratio
λ**v:** volumetric efficiency
θ**:** shaft angular position
Θ **:** gear pressure angle