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A new Wankel-type compressor and vacuum pump

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Abstract: When the Wankel principles were first published in the early 1950s most of the initial work was aimed at developing a ***compressor***. At that time many of the characteristics appeared to promise a superior machine than hitherto known. However, all the early designs resulted in a high value for the minimum clearance volume (CV) and this problem was never overcome.

Knowledge now gained from the development and manufacture of the Wankel ***engine*** has enabled the evolution of a new compressor concept where the rotor flank, radially very close-fitting over its central area, provides gas sealing with the housing bore. The rotor has an increased radial clearance towards the apices which makes the machine practical to manufacture. The 'nesting' of the rotor flank with the housing bore at the end of the exhaust stroke results in an extremely small CV.

This machine promises to possess an exceptional combination of all the attributes which are important in achieving ***high energy efficiency*** in positive-displacement compressors and vacuum pumps:

- near-zero CV
- low mechanical friction losses
- low internal gas leakage (assisted via oil flooding)
- high volumetric efficiency.

In addition it is compact, lightweight, vibration-free, consists of few components, and can be built in any chamber size.

The Paper discusses the features and characteristics of the design.

1. Introduction

1.1 Origin of the innovation

In January 2014 the **Daily Telegraph [1]** published a half-page article describing a new type of compressor. This article triggered the question why the Wankel principle**,** which possesses so many relevant advantages as a positive displacement machine, has so far only been exploited as an engine and has never progressed towards being a successful compressor.

A search was made through all the Technical Papers and all the published Patent Applications relating to Wankel type compressors dating from about 1953 to 2013. The high value of the CV issue

IOP Conf. Series: Materials Science and Engineering **232** (2017) 012065 doi:10.1088/1757-899X/232/1/012065

dominated the various designs which had been considered and worked upon. The early attempts by Borsig, NSU, Neumag et al all appeared unsatisfactory. In attempting to increase the rotor R/e ratio (R = rotor radius, e = shaft eccentricity) and thereby improve the fundamental geometric compression ratio, thus minimising the CV, these designs all resulted in small diameter eccentric shafts and stationary gears. These components proved to be insufficiently robust for a practical industrial machine **[2]**.

Furthermore, the resulting CV of these designs was still too high for good efficiency, and the high R/e value employed also resulted in the machines not being particularly compact.

The later-developed **Ogura Wankel [3]** of the 1980s, which was manufactured for vehicle air conditioning applications, used a typical engine-design value for the R/e ratio. This resulted in a smaller and more robust machine but it possessed the extremely large and penalising CV value of 16 %. Nevertheless, probably because of its light weight, freedom from radial vibration and good mechanical efficiency **[4],** it secured a market share for a period.

1.2 The new concept

Study has now led to the GW design (designation as in Garside-Wankel), which exploits the principle whereby a rotor with a uniquely shaped flank profile will provide:

- gas sealing between a moving point in the central region of the rotor flank and the epitrochoidal
- bore
- a close-to-zero value for the resulting final CV
- and in combination with greater radial clearance towards the rotor apices will be practical to manufacture.

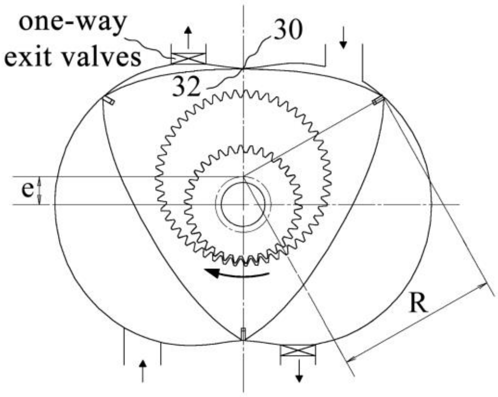

Figure 1 A cross sectional view through the machine with rotor at 'engine' TDC (top dead centre) position.

It will be noted that gear backlash does not affect the radial clearance between 30 and 32 in this TDC position because the point 32 will move merely laterally. At other angular positions the gear backlash plus gear angular location tolerances require that the rotor flank shape is reduced in radial size slightly, to avoid risking impact of the apices with the housing bore.

1.3 A Note on the fundamental sealing quality of the Wankel

Of all the known positive-displacement types of compressor, only the reciprocating and the Wankel possess an adequate sealing system which enables them to operate ***as an IC engine.***

Gas temps of up to 2000 C and gas pressures of around 50 bar exist in such machines.

To create a 4-stroke IC engine, the sliding vane (SV), screw, scroll and Lontra type compressors for example, would generally require two machines built in some type of back-to-back arrangement.
The gas sealing quality of all these designs would be quite incapable of operating efficiently with such temperatures and pressures.
This comment referring to the IC engine is made to illustrate that the fundamental sealing capabilities of the Wankel (whilst not as good as a reciprocating engine using piston rings in a circular bore) are of such quality that it should have a *relatively* easy task when operating as a ***compressor***. Certainly this should be the situation when operating with oil flooded air.

2. Discussion of the major design characteristics

2.1 Minimum Clearance Volume
A study of the Animated Model [5] illustrates how the rotor movement provides close to zero CV because the trailing part of the rotor flank 'nests' with the housing bore 60° ATDC (shaft angle) at the end of the exhaust stroke.

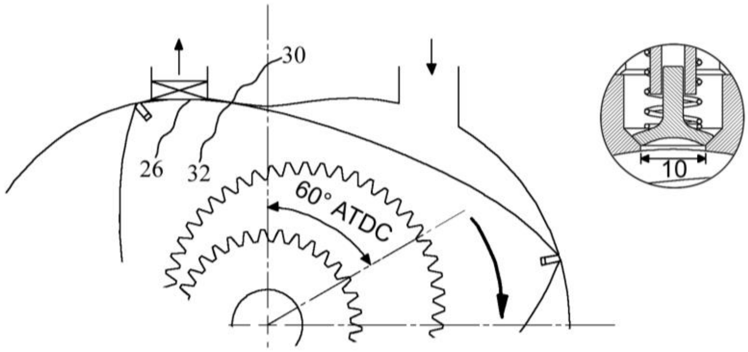

Figure 2. Nesting of the rotor flank with the housing bore at the end of the exhaust stroke; with an inserted view of an exit valve (three are fitted transversely)

The CV (or 'dead volume') in this rotor position consists of the small working clearance between the rotor and housing and small recesses under the outward-opening poppet valves.
The total volume in chamber 26 is less than 0.5% of the intake chamber volume.
In operation, oil is likely to largely fill this cavity

2.2 Gas sealing quality
There are three potential routes for leakage to take place:

2.2.1 Leakage via the radial clearance between the rotor flank and housing bore. The animated model shows how a moving point (32) on the rotor flank continually maintains a close radial clearance with a specifically related moving point (30) of the housing bore during the phase from about 60° BTDC to 60° ATDC of the eccentric shaft movement.
The exhaust stroke is essentially completed by 60° ATDC.
In Wankel ***engines*** the radial clearance between the rotor flank and the bore is not an important consideration, being typically about 0.5 mm . In the central part of the flank there is a combustion recess of course. ***In a compressor*** the clearance needs to be ***much*** smaller, certainly less than 0.1 mm to avoid excessive leakage over the rotor flank during the compression / exhaust stroke.
Calculations indicate that even with 0.05 mm gap the sealing with oil-free air would not be satisfactory but with oil present, as is commonly used in many types of air compressors, leakage per cycle would be less than about 1 % of the free-air intake volume.

With current standard numerically controlled (NC) machines it would not be too onerous to provide a gap of 0.05 ± 0.04 mm. The small leakage over the rotor flank is likely to consist mostly of oil which will have been deposited or centrifuged on to the trochoid bore. Leakage will result largely from the rotor surface movement 'dragging' the oil through rather than viscous flow due to the pressure differential.
From 60° ATDC onwards the rotor flank increased radial clearance to the housing bore no longer affects the gas leakage, this being then controlled by the apex seals.

2.2.2 Leakage via the axial gap between sides of rotor and side plates. A novel design feature in the GW compressor is that the conventional gas side seals, as used in a Wankel engine rotor, are eliminated. The available pressurised oil supplied from the receiver vessel is utilised to completely fill and pressurise the internal cavities of the rotor.
Alternatively, a separate pump may be used to supply the oil.
Centrifugal force on the oil inside the rotor will ensure that the oil pressure at the outer perimeter of the rotor internal cavity will always be higher than the maximum pressure of the air in the working chambers. Some small quantity of this oil, due to the pressure differential, will then continually and intentionally leak ***outwards*** into the working chambers via the axial clearances.
This feature will eliminate ***all*** gas leakage out from the working chambers via the sides of the rotor.
The design has other advantages:

- The manufacturing cost of 30 gas sealing components and their associated machined slots is avoided.
- The mechanical sliding friction of those side seals is eliminated.
- The radial space required by the side seals is not needed. Hence an exceptionally low R/e value for the geometry of the rotor can be selected. This results in a more compact and lighter weight rotor and resulting machine, with further friction savings due to reduced seal travel distances and velocities.

Note: In an Wankel epitrochoidal mechanism the value of R/e can be readily assessed from an axial assy drawing by viewing the closeness of the rotor ring gear PCD to the rotor flank. These coincide when R/e = 5.0 .

2.2.3 Leakage past the apex seals. Similar design apex seals operate satisfactorily in an IC engine. Hence, in an "oil flooded" compressor duty, any gas leakage via the apex seals should be negligible. Wear rates can be expected to be low, as is the case with SV (sliding vane) type compressors, because all metal-to-metal moving contacts will have a hydrodynamic oil film present.

2.3 Mechanical Friction Losses
Comparison with an SV compressor of similar capacity shows that the total surface area swept by the apex seals of the GW in unit time is only about 8% of the area swept by the vanes in the SV (assuming that both are operating at the same rpm). This low figure occurs because the rotor of a GW is only revolving at 1/3 the speed of the eccentric shaft, there are only 3 apex seals (typically 7 or 8 vanes in the SV) and the rotor is axially narrower and employs shorter axial length seals than in the SV type. Furthermore, the lightweight apex seals of the GW are not subjected to significant forces and nominally are stationary in their slots, whereas the heavy vanes of the SV are subject to a high outward centrifugal force. Those vanes are also oscillating radially in their slots with a simultaneous bending moment being imposed.
Each factor of swept area, sliding velocity, radial load, and vane-to-slot movement is more energy consuming in the SV than in the GW.
The GW does have a disadvantageous friction feature in that the rotor gears are operating in 'solid' oil

IOP Conf. Series: Materials Science and Engineering **232** (2017) 012065 doi:10.1088/1757-899X/232/1/012065

which is not ideal. But the gears are open at both axial sides for the 'trapped oil' to escape and the meshing velocities are not high. Special tooth forms can be used.
The likely reduction of gear meshing noise may be valuable.

2.4 Volumetric Efficiency (VE)
Wankel ***engines***, which possess an unrestricted inlet port and a sinusoidal intake volume change, have opportunities for dynamic inlet tuning and typically achieve a VE of over 110% despite the induction air undergoing some heating and expansion during the induction stroke.
As a compressor the VE is likely to be close to 100%, significantly better than all other compressor types which suffer from re-expansion of the CV gas, some leak-back of previously compressed and heated air and generally possess untunable inlet systems.
VE per se does not affect energy efficiency although it does result in a smaller and lighter machine, which possesses a greater throughput for given leakage and friction losses, etc. Consequently those specific loss factors are improved.

3. Minor features of the design

3.1 The primary innovation of the rotor providing gas sealing at a moving point on the rotor flanks does result in unbalanced gas pressure loads on the compressing flank. This imposes a significant cyclic torque load on the gears as illustrated below:

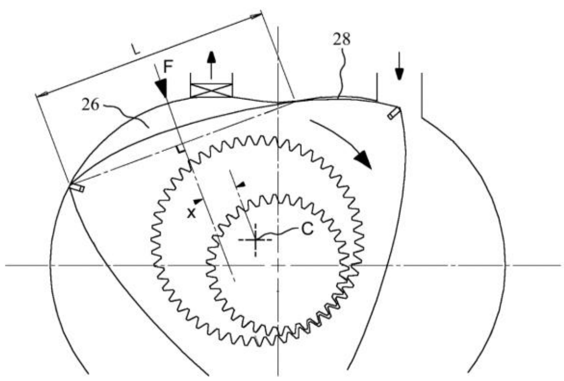

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Figure 3. Intermittent torque loading on the gears results from the uneven gas pressure on the rotor flank. Pressure in chamber 26 is higher than in 28 resulting in Force F producing a torque Fx about the centre C of the rotor

The issues are further discussed in the patent document **[6].**
A twin gear arrangement with gears on either side of the rotor is proposed for higher pressure applications.
Life calculations indicate that:

- a single gear pair of unhardened steel would be satisfactory for up to about 6 bar inlet-to-outlet differential
- use of hard steel would allow up to 12 bar
- and double these values with a twin gear arrangement

3.2 Manufacture of the gears
The first prototype required that twin ring gears were shaped separately from each side.
A later design enables either single or twin gears to be wire cut or broached through in a single pass.

3.3 Exit 1-way valves are required in the GW unlike the SV, screw, and scroll types.
There is sufficient space to install multiple small and light-weight poppet valves capable of passing the required volume flow with fast response and low pressure drop characteristics.
The GW has an advantageous characteristic in that the exhaust stroke occupies a long duration of 330° of eccentric shaft rotation, of which 140° are utilised to eject the final 10% of the chamber volume.
As an example, with 7 bar compression, the exhaust phase in a reciprocating compressor occupies only the final 50° before minimum volume is reached. The GW uses 150° for this exhaust phase, greater by a factor of 3.
The exit valves can therefore be smaller, or involve lower pressure loss, or the operating speed can be higher. For this reason it is considered that the unusual combination of 'oil flooding' in a machine fitted with exit valves is unlikely to present major problems.
The exiting oil will be beneficial in relation to damping the movement of the valves, lubricating and sealing the seats and cooling. Three 10 mm dia poppet valves made from 'Torlon' engineering plastic, which possesses low density as well as high strength plus wear and temperature resistant characteristics, are fitted to the proof-of-concept machine in an axial row. The weight of the valves is only 0.38 gm each.

3.4 Manufacturing cost aspects
The scroll type compressor is currently manufactured economically, despite the requirement for very small radial clearances being essential at six 'sealing points' simultaneously **[7].** Computer-controlled machine tools create precise surface geometry maintaining tolerances measured in microns. The GW only requires a single such 'sealing point'. So, if found to be advantageous, selective assembly perhaps based only on housing bore, rotor OD and shaft throw dimensions could be used to achieve an extremely small gap at this single point.
The scroll and GW machines are basically similar in that they both involve a rotor mounted on an eccentric shaft rotating inside a housing. The manufacturing costs of the scroll compressor are regarded as being low. Hence the costs of the GW should also be acceptable.

3.5 A 'free' addition to the swept volume
The feature of the GW design which gives a near zero CV results from the rotor continuing to rotate past the normal 'engine TDC' position and then nest with the housing. A favourable outcome is that the TDC clearance volume (illustrated in Fig 1) is now ***added to*** the conventional rotary engine 'swept volume' calculation to give an increased volume to the intake chamber.
To clarify, there is an addition of 7.5% intake volume bonus for a given size of machine. Again, this offers size, weight and small energy efficiency advantages.
The 7.5% figure is the fundamental TDC clearance of the 2:3 type geometry when the value of R/e = 5.3 **[8]** .

3.6 Part-load operation.
Throttling the intake of the SV type compressor, in particular, involves a significant reduction in energy efficiency. The GW has space available on the intake port sides to incorporate two (or more) alternative-position later-closing inlet ports, a more efficient way of reducing the output to match varying flow requirements. This may allow use of a lower cost and more efficient fixed-speed electric drive motor for some applications.

4. Application of the GW as a vacuum pump
The attributes of the GW are particularly advantageous for some vacuum pump duties:

- the near-zero CV combined with good gas sealing will allow a high vacuum to be achieved in a single stage machine

- the low mechanical friction losses become more significant in contributing to energy

- efficiency when the air work energy may be very small.
- the machine can be built in unlimited size without incurring vibration problems.

An early type of rotary piston vacuum pump is shown below:

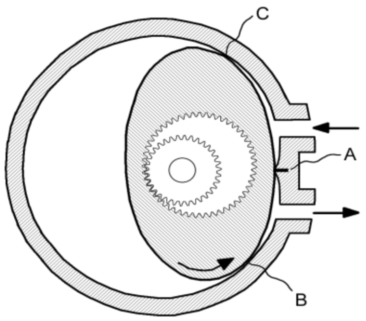

Figure 4 Axial view of a ***hypotrochoidal*** type rotary piston vacuum pump

There are no gas seal pieces in the rotor, simply one 'stationary' seal (A) in the housing at the 3 o'clock position above. Hence it is essential that the rotor has very close 'sealing' clearances with the housing bore ***simultaneously at two points,*** B & C, a difficult manufacturing requirement (particularly in earlier days) bearing in mind that both the gear backlash and the angular positioning accuracy of the gears would affect these clearances. It was perhaps inevitable that this machine was found to be uneconomic to manufacture and it is no longer produced.
However, the descriptions of this machine **[9]** did state that it had advantages over alternative vacuum pumps in that:

- the volume of the pumping chamber with respect to the volume of the entire machine (the so-called 'specific volume') was twice that of the competing 'plunger piston type' vacuum pumps
- it could run at twice the speed because of the low vibration and thus was about four times smaller
- the low vibration eliminated the need for heavy foundations.

5. Some comparisons of the GW with existing compressors

5.1 The CV (or 'dead volume')
This is smaller than other types of competing compressors.

5.2 Gas sealing
The sealing quality potential of the GW offers superior sealing to existing rotary type compressors and may match the best reciprocating type. It should be superior to the screw type, which possesses very long sealing lines, as well as having the 'blow holes'. It should be superior to the scroll compressor which has many more leakage paths. It certainly should be superior to the SV, which has leakage past the axial ends of the rotor and of the vanes, and under the vanes.

5.3 Mechanical friction losses

a) ***Reciprocating compressors:*** An advantage of the GW is the absence of piston side thrust, as well as no requirement for inlet valves with the associated pumping loss. As described in 2.2.2 the GW has zero rotor side mechanical friction, only a small viscous shearing loss. The size and weight of the GW will be lower and consequently will use smaller bearings. Overall friction will be lower, as assessed in more detail in **[6].**

b) The ***scroll compressor*** rotor has a high axial load combined with continual axial face sliding movement which will result in significant energy loss.
c) The ***screw type*** has very long sealing line lengths incurring high speed viscous oil shearing.
If no oil is present at any point along these sealing lines, there will be leakage losses.
d) The ***SV*** is discussed in 2.3

The reduction of mechanical friction in the GW and therefore energy / heat input will lower slightly the compression index and hence provide a small efficiency gain.

6. Testing

In 2014 the design and detailing of a first proof-of-concept 7.5 kW, 7 to 10 bar, 2 x 178cc machine was completed by ECL (Epitrochoidal Compressors Ltd).
At 2900 rpm this machine has an intake swept volume of 1.11 m^3 per minute.
Agreement was reached with a UK compressor manufacturer that they would:
a) Procure the parts to build one machine
b) Test as a 6 to 10 bar air compressor on their existing test rig.

This testing is still in hand.
Procurement of 3 further sets of parts by ECL is also now in hand in order to allow broadening of the testing effort. One of these machines will be configured as a vacuum pump.

7. Conclusions

- ***As an IC engine,*** the Wankel has major advantages in comparison with the reciprocating engine in relation to size, weight, vibration, mechanical and volumetric efficiency; but a ***major*** deficiency in gas sealing quality, plus an inferior combustion chamber shape.
  As a compressor, the GW design retains all the above advantages but evades the two deficient areas. In fact it is the inferior *'combustion chamber shape'* which the GW design has advantageously exploited to create the important near-zero CV feature.
- Published Papers **[10]** compare the energy efficiency of SV versus screw, reciprocating versus scroll; etc., etc. . The conclusions appear to be that no one type is much superior in this aspect to any other. Analysis of the CV, gas sealing, friction and volumetric efficiency attributes indicates that the GW may possess a 15 to 25 % higher energy efficiency than the SV; and also therefore similarly higher than other types of existing air compressors.
  A 7.5kW, 8 bar machine may therefore expect to achieve 75 % or so isentropic efficiency.
  To achieve this performance will require development; and probably the high machining precision as is routinely used for the manufacture of scroll compressors.
- A viewing of the area of the GW intake chamber's maximum-size axial profile (noting that there are two intakes per shaft revolution) compared to the complete machine (the ratio which indicates the so-called 'specific volume'), emphasises how much smaller the GW machine will be relative to other positive displacement machines.
- The GW machine is lighter-weight and consists of fewer components than most other compressor types. Hence, in the longer term and when it is produced in volume, it may well be lower cost to manufacture. .

Acknowledgements
A GB patent application was filed in July 2014 (granted as B2528309); and the corresponding PCT version (GB2015 / 052040) in July 2015.
Note that the patent describes both the 2:3 (two lobed housing with 3 cornered rotor); and 1:2 type machines.
This Paper concentrates on the former only, this being the version with the larger number of potential applications.
ECL is a UK private limited company
75% of the shares are owned by D W Garside,
25% of the shares are owned by J C Thurner,

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