

This book describes a number of original mechanisms and systems conceived by the author along the very productive decades of his life. The degree of development of these concepts is variable; from being just an idea to the attainment of an industrially mass produced machine. Being advanced in years, the author will not be able to bring some of the promising concepts to a favorable outcome, or to discard those that turn out to be impractical. For these reasons, he better opted to capture them in a volume that might inspire young readers to accept this challenge. The topics, presented in eleven chapters, comprise five positive displacement rotary machines –four pumps and one compressor–, a piston engine that follows the Brayton cycle, a magnetic gear transmission, an inertia flywheel for energy storage in vehicles, a passively pressurized journal bearing, a water and heat economizer for showers, and a mango fork.

SOME UNCOMMON MECHANISMS AND SYSTEMS

Legacy of a mechanical engineer

Ricardo Chicurel-Uziel



INSTITUTO
DE INGENIERÍA
UNAM

**SOME UNCOMMON MECHANISMS AND
SYSTEMS**

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Ricardo Chicurel-Uziel
IIUNAM

March 2019

Some uncommon mechanisms and systems.

Legacy of a mechanical engineer

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Preface

For over half a century, I have accumulated ideas for novel mechanisms and systems. Many of these have remained just that: ideas. Some, I have discarded after my initial enthusiasm evaporated when discovering serious flaws or realizing that they were not new. A few have advanced by various degrees towards what I consider the ultimate goal: becoming useful contributions to society. Over the years, it has become increasingly clear that, to decide which of these ideas are worth developing further, and develop those that are, is unattainable during my lifetime; however, I can preserve them in a book hoping that others might be interested in them.

I am indebted to the many people that have helped me one way or another in the various projects on which I have worked during the long space of time chronicled in this book. They are associated with academic and industrial institutions, including Virginia Polytechnic Institute and State University (VPI), where I was a professor of Engineering Mechanics from 1959 to 1970, and the Universidad Nacional Autónoma de México where I have been a professor and researcher for the past 48 years. Special recognition for the support received in recent years are due to the following colleagues: Gabriel Ascanio, Filiberto Gutiérrez, Germán Carmona, Francisco Godínez, José León, Ernesto Terrazas, Roberto Reséndiz, Gustavo Cadena, Rodrigo Cárdenas, my brother, Enrique Chicurel, and, Cristina Verde, who encouraged me to write the book and was always available to help me with various unexpected obstacles along the way. I have received valuable contributions in ideas and day to day help in preparing drawings, performing calculations, attending to administrative tasks, etc., from the following students: Omar Tenorio, Francisco Isaac López, Fo-Lyn Yong, Luis Enrique Calderón. My research has been dependent on the technical assistance of various people which include David Santoyo, Raúl Elizalde, Javier Hernández, Rigoberto Nava.

My wife, Carmen Helbing, deserves special thanks. On countless occasions, knowing her fabulous intuition, I have interrupted her in the middle of important tasks or entertaining activities to ask for her judgement on segments of the book. Invariably, she calmly turns her full attention to my request and usually, without delay, surprises me with a very valuable comment which makes me say to myself “why didn’t I think of that?”

Finally, as I conclude the writing of this book, I am continuing to work in the ideas contained in Chapters 4, 7 and 10.

Keywords: [Positive displacement rotary machines, magnetic gears, energy saving, journal bearings, flywheels]

Resumen

Esta obra agrupa varios conceptos novedosos de mecanismos y sistemas ideados por el autor durante varias décadas. El grado de desarrollo de estos conceptos es variable; desde ser solamente una idea, hasta el logro de una máquina en producción industrial masiva. El autor, ya avanzado en años, no podrá adelantar mucho en la identificación de los conceptos prometedores para llevarlos a una conclusión exitosa, o bien descartar los que no resulten prácticos; así, optó por plasmarlos en un volumen que pueda inspirar a lectores jóvenes que quieran aceptar este desafío.

Los temas, expuestos en once capítulos, incluyen 5 máquinas rotatorias de desplazamiento positivo incluyendo 4 bombas y un motor, un motor de pistones de ciclo Brayton, una transmisión de engranes magnéticos, un sistema de recuperación de energía de volante de inercia para vehículos, una chumacera con presurización pasiva, un sistema para ahorro de agua y calor para regaderas y un tenedor para mangos.

Palabras clave: [Máquinas rotatorias de desplazamiento positivo, engranes magnéticos, ahorro de energía, chumaceras, volantes de inercia]

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1. SPROCKET PUMP

The sprocket pump is a positive displacement mechanism with some similarity to internal gear pumps

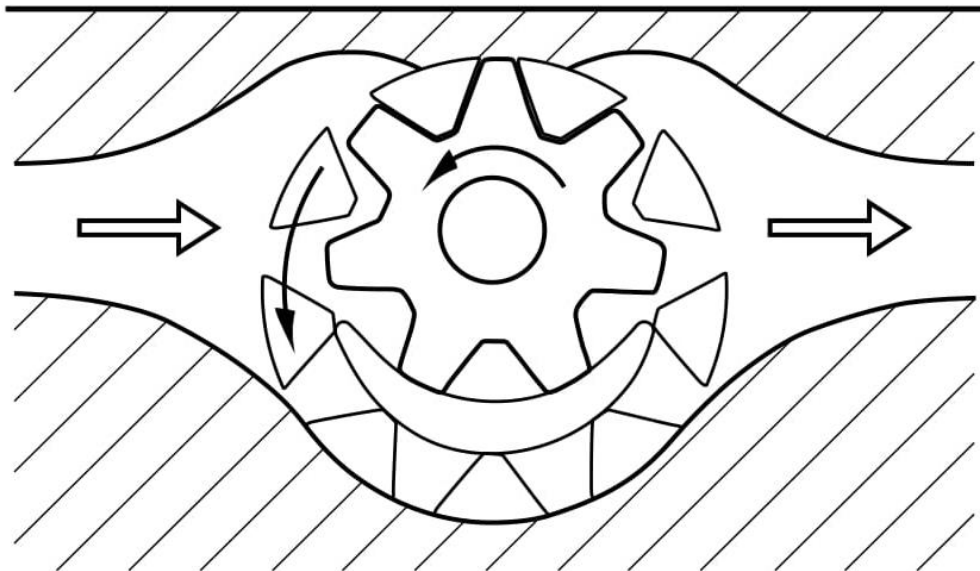


Figure 1.1 Internal gear pump

In Fig. 1.1 is shown schematically an internal gear pump, and in Fig. 1.2 an early version of the sprocket pump, such as described in the first patent (Chicurel, 1988) and in references (Chicurel & , 1981) (Chicurel & , 1982) The latter results essentially by substituting the smaller gear of the internal gear pump by a sprocket (3 in Fig. 1.2) and the large gear with internal teeth, by a disk (1) having a number of pins (2) that mesh with the sprocket. As in the internal gear pump, there is a crescent shaped abutment (7) separating the high and low pressure zones. The disk with pins is driven by means of the pump's shaft which extends from the opposite side of the pins. Fluid is propelled by the pins along the

pumping corridor (4). All machined surfaces in the sprocket pump are either flat or circular cylindrical, which is advantageous from a constructive point of view. The resultant force of the hydraulic pressure acting on the sprocket surfaces is always directed toward the sprocket axis, and therefore has zero turning moment. Thus, the power required to drive the sprocket is only that required to overcome frictional losses. In contrast, the resultant of the hydrostatic pressure on the driven gear in a gear pump produces a moment which must be resisted by a force transmitted by the driving gear. This gives an advantage of the sprocket pump with respect to efficiency and wear of moving parts, although the claim has not been experimentally confirmed. Another unproven claim is that the squeezing action on the fluid in a gear pump as it approaches the mesh point does not occur in a sprocket pump, an advantage with regards to efficiency particularly when pumping highly viscous fluids.

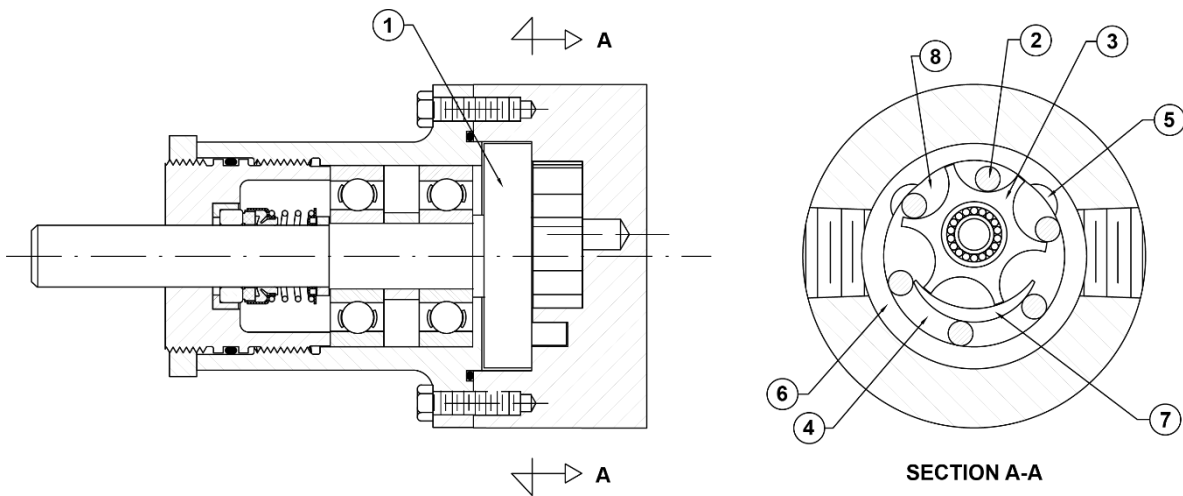


Figure 1.2 Early version of sprocket pump



Fig. 1.3. Partially assembled pump for pumping molasses

A distinguishing feature of the sprocket pump is that it delivers a strictly constant volumetric flow without pulsations or ripples. This characteristic is shared by another pump conceived by the same inventor and described in Ch. 3. Fig. 1.3 is a photograph showing a partially assembled sprocket pump for pumping molasses.

Sprocket pumps have been manufactured by the DYFIM, S.A. Company in Mexico, headed by José León-Garza. He and his former partner, Roberto Reséndiz-Núñez, both mechanical engineers and good friends of mine, enthusiastically embarked in the sprocket pump project in the 1980's. They contributed with many good ideas. Another fellow engineer who has participated in the development of the sprocket pump is Filiberto Gutiérrez-Martínez, researcher at the Engineering Institute of UNAM. He and José are co-inventors of the latest improved version of the pump (Chicurel, et al., 2008). Following is a discussion of those improvements.

Figure 1.4 shows the improved version of the pump.

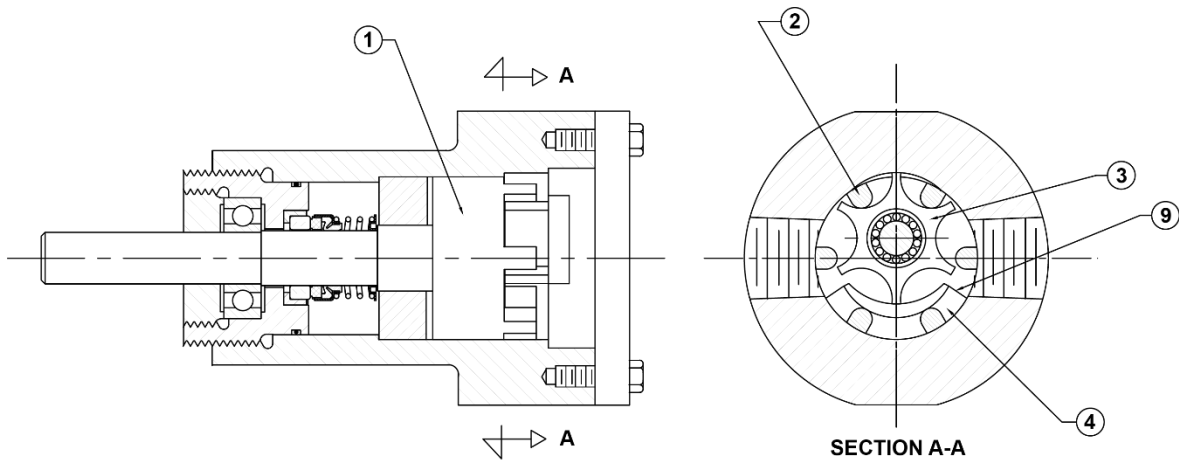


Figure 1.4 Improved version of sprocket pump

In the improved version, instead of a rotor having a disk with pins press fitted into holes, there is a rotor with prismatic cantilevered pumping elements all in one piece (1). The prismatic elements have a cross section having an outer circular arc which is part of the rotor's periphery and provides a surface, instead of a line seal along the outer boundary of the pumping corridor. Originally, the rotor disk provided a flat surface seal in the annular zone around the pins. Because of the non-symmetric pressure distribution on this area, an undesirable bending moment on the shaft was produced. This moment is greatly reduced in the improved version. Instead of the flat surface seal, a cylindrical surface seal, which also operates as a bearing surface, distinguishes the new version.

Another difference between the two versions of the pump is the solution given to trapping of fluid in a space of variable volume, labeled as number 8 in Fig. 1.2. Originally, relief bays, labeled 5 in the same figure, were machined in the annular surface 6. In the new version, the sprocket height is greater than the length of the prismatic pumping elements, requiring a greater depth of the cavity for the sprocket than that of the pumping corridor. In this manner, the fluid in space 8 has an outlet under the pumping elements.

Still another difference introduced in the new version is that the mating surfaces between the rotor's bearings housing and the body supporting the sprocket is displaced so

that the latter body is reduced to a flange from which protrudes the crescent shaped barrier. Thus, the interior and exterior walls of the pumping corridor are in separate parts for easier machining. The new version is covered in a second patent (Chicurel, et al., 2015). Other variations of the sprocket pump which have been proposed for improving its performance are described in references (Chicurel, et al., 1986) (Chicurel, 2004)

2. SINGLE VANE PUMP

As mentioned in a paper presented in 2007 (Chicurel, 2007), I came upon this concept thinking that it was a novel idea, then I found that a patent had already been awarded to Cornelio Seno back in 1991 (Seno, 1991), and then I discovered that, non other than Felix Wankel, had catalogued this mechanism in his treatise on rotary machines of 1965 (Wankel, 1965). My contribution is a variation of the original concept aimed at improving the seal of the variable volume chamber bounded by the rotor, the stator and the vane. Following is a description of this modified concept.

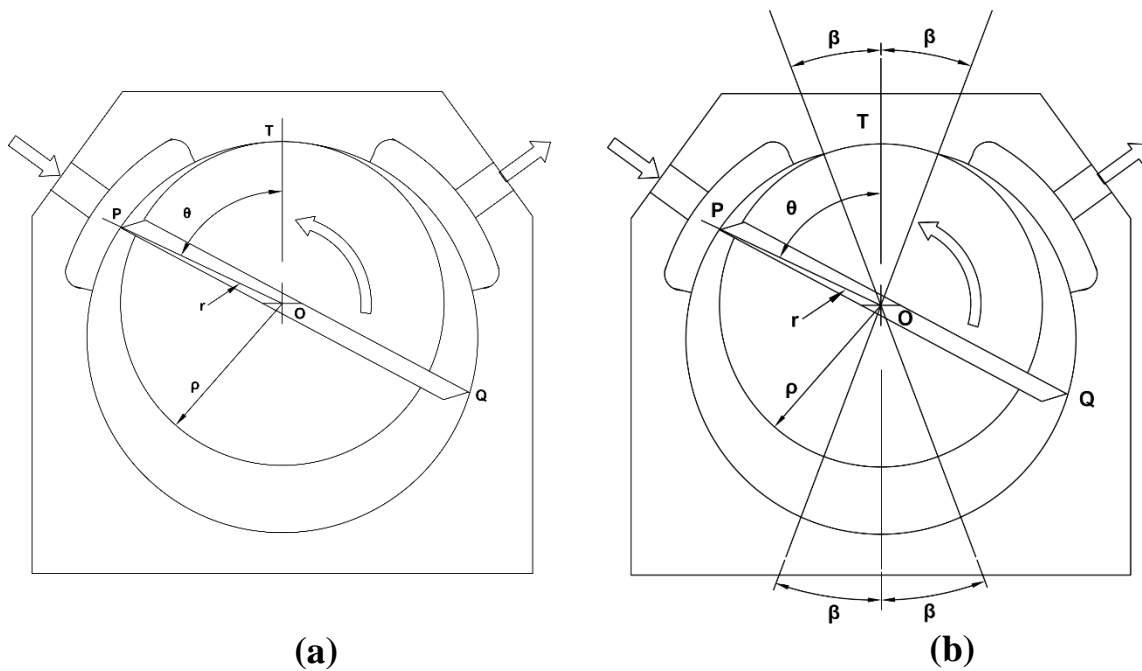


Figure 2.1 (a) Original single vane pump, (b) Modified single vane pump

Fig. 2.1a is a diagram of the original mechanism, equivalent to that described in Seno's patent. The cylindrical rotor has a diametrical groove with respect to which the vane PQ slides. The stator, or cam ring, has a contour which both vane's tips follow at all times. It is defined by the following equation:

$$r = f(\theta) = (L - \rho)(1 - \cos\theta) + \rho, \quad (2.1)$$

where r, θ are polar coordinates, $2L$ is the length of the vane, and ρ the radius of the rotor.

The low and high pressure spaces are separated by the line seal at the tangency point, T, between the rotor and the cam ring. The modified mechanism which I proposed is shown in Fig. 2.1b. The result of this modification is that, instead of the line seal mentioned previously, a seal over an area of tangency extending over an angle β on either side of T is obtained. The equation defining the modified contour is:

$$\begin{aligned} r &= \rho, & -\beta \leq \theta \leq \beta \\ r &= f\left[(\vartheta - \beta) \left(\frac{\pi}{\pi - 2\beta}\right)\right], & \beta \leq \theta \leq \pi - \beta \\ r &= 2L - \rho, & \pi - \beta \leq \theta \leq \pi + \beta \\ r &= f\left[(\vartheta - \pi - \beta) \left(\frac{\pi}{\pi - 2\beta}\right) + \pi\right], & \pi + \beta \leq \theta \leq 2\pi - \beta \end{aligned}$$

In the modified mechanism, the condition that both vane's tips follow the cam ring contour at all times, is preserved.

Both the original and modified versions of the pump have the following advantages over the conventional multiple vane variety: (1) simpler design less costly to manufacture and, (2), the center of mass of the single vane is always near the rotor's axis of rotation resulting in a reduced centrifugal force and reduced contact force between its tips and the stator ring.

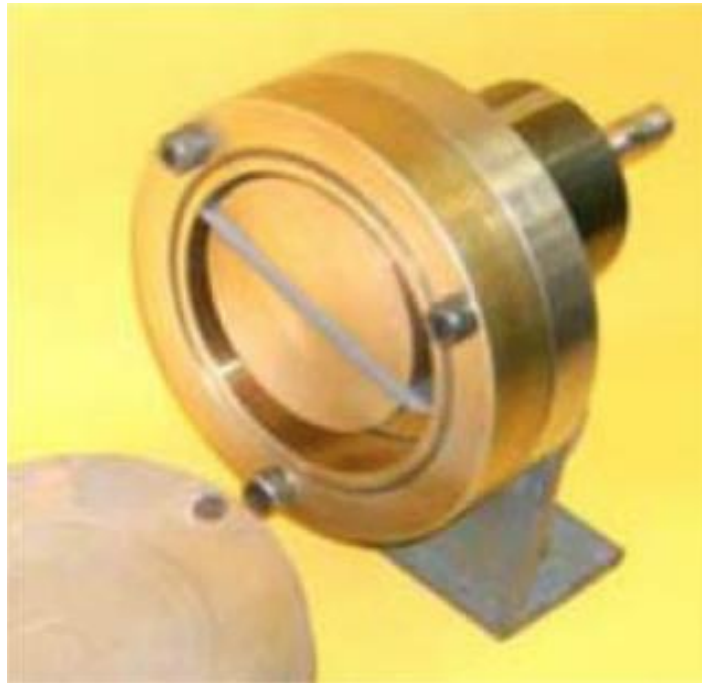


Figure 2.2. Single vane pump

A small pump having a displacement of 31 cm^3 was designed in which the parameter β for the cam ring was chosen equal to 300. A photograph of one of two pumps built is shown in Fig. 2.2. The unit, driven by a 1,750 rpm motor, was used in an application requiring pumping water at a pressure of 0.36 MPa.

3. GYRATING PUMP

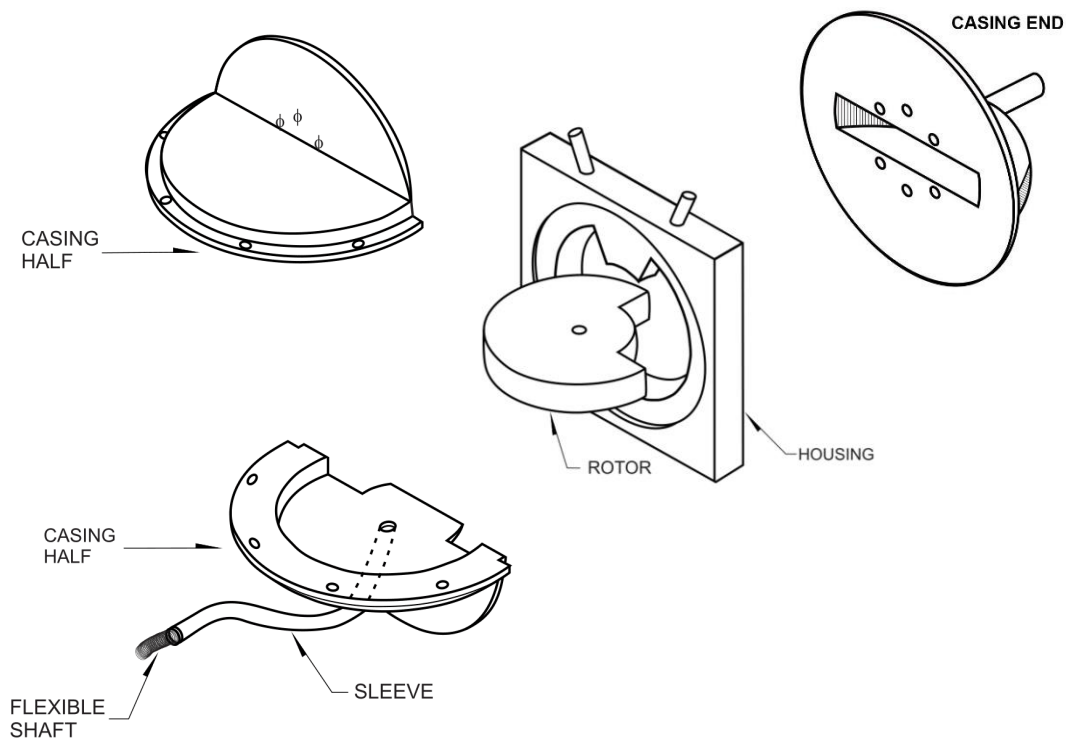


Figure 3.1 Exploded view of gyrating pump

This is a positive displacement pump unlike any mechanism I can think of. Fifty years have elapsed since I was granted a patent (Chicurel, 1967) and, despite my conviction that there must be an ideal application for it, I haven't yet found it. It is distinguished by these characteristics: (1) the output is a constant, ripple free stream, (2) Sealing of the pumping cavity is between flat or spherical surfaces of equal radii, there being no line seals, and (3) the two moving parts rotate at constant speed. Fig. 3.1 is an exploded view of the pump showing a rotor which is contained between two half-casings and a casing end. The half-casings are bolted together. Each has a boss with a spherically contoured periphery terminating in a flat surface onto which the end casing is bolted.

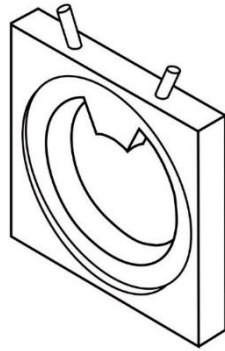


Figure 3.2 Gyrating pump housing

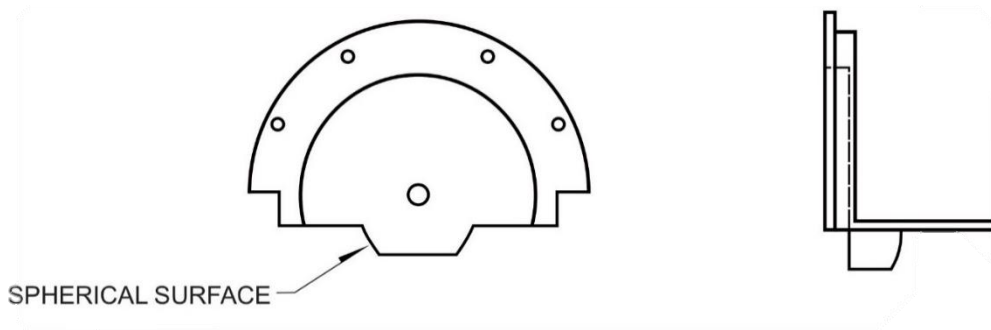


Figure 3.3 Gyrating pump half-casing

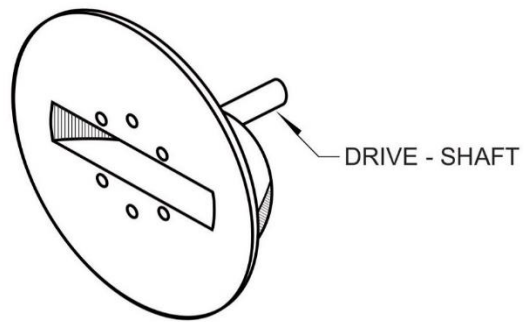


Figure 3.4 Gyrating pump casing end

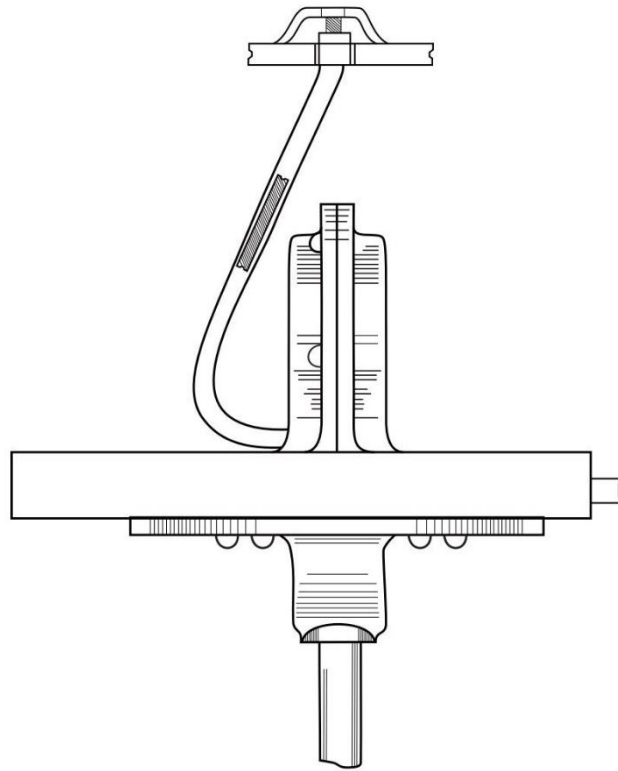


Figure 3.5 Side view of gyrating pump

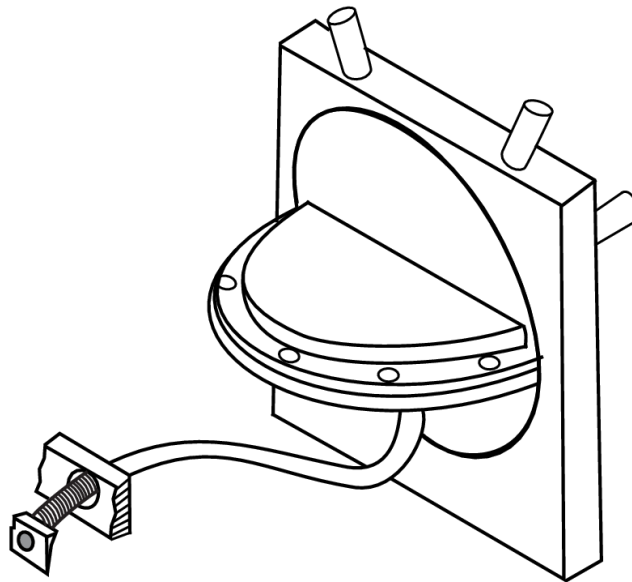


Figure 3.6 Isometric view of gyrating pump

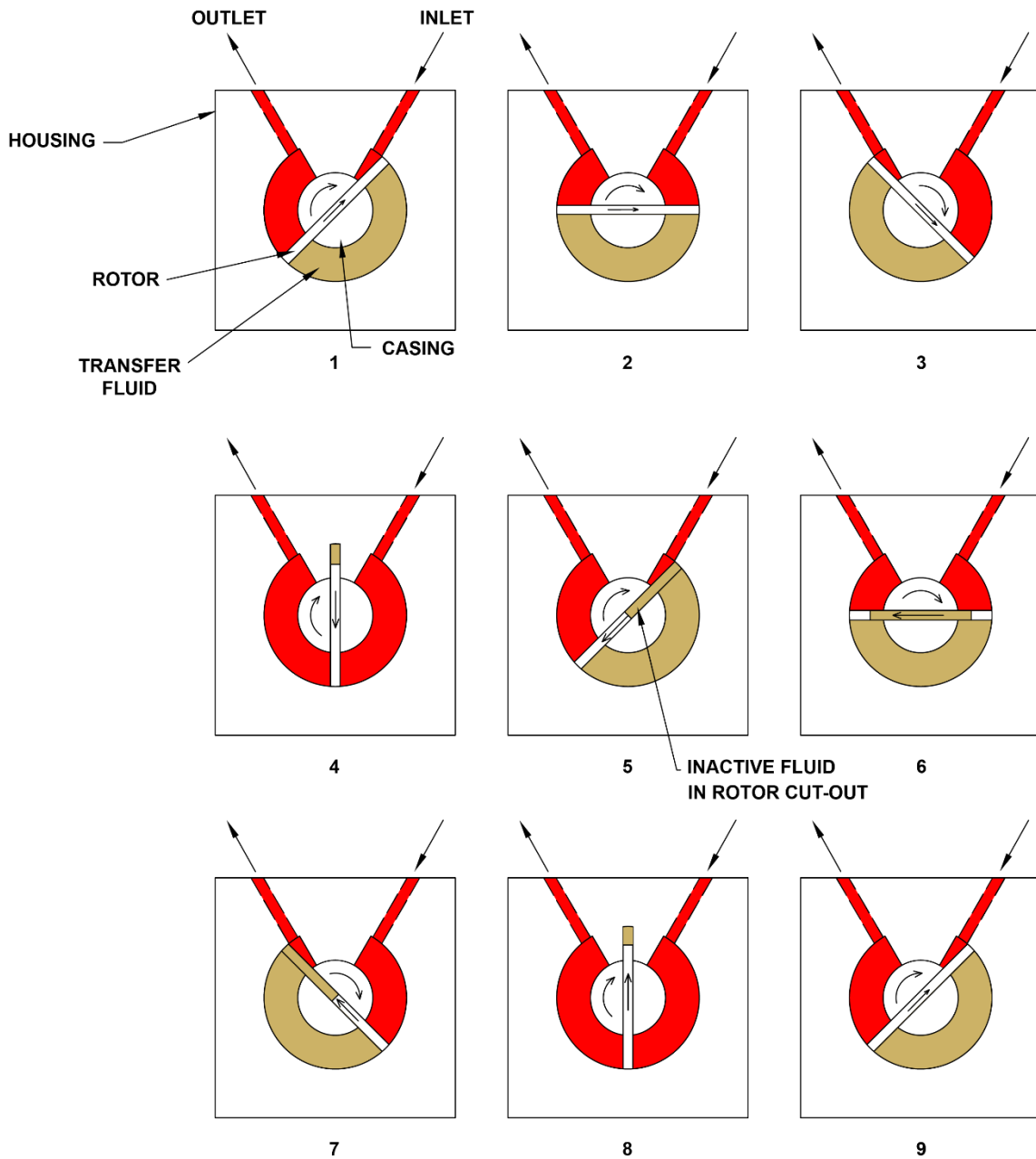


Figure 3.7 Sequential diagram of gyrating pump operation L.E.

Figures 3.2, 3.3 and 3.4 are drawings of the housing, a half-casing and the casing end respectively, and figures 3.5 and 3.6 are side and isometric views respectively of the assembled pump. The pump is driven by means of the shaft on the casing end. The 3-piece casing and rotor within rotate together; however, as will be explained further on, the rotor

also spins inside the casing about its own axis. The rotor is a flat disk with a sectoral cutout. Its peripheral surface is spherical, matching that of an interior raceway of the housing structure. The latter raceway is interrupted by a tooth protruding inwardly. Also, the circular arc portion of the rotor's sectoral cutout has a spherical surface matching the surface at the end of the tooth. The rotor is forced to spin inside the casing by a flexible shaft whose other end is attached to a fixed support and maintained in alignment with the rotor's drive shaft. This action results from the torsional rigidity of the flexible shaft, which maintains its bent configuration by being constrained inside a sleeve. Thus, the rotor rotates about its axis once for every rotation of the casing. Its cutout portion is positioned so that it coincides with the tooth of the housing structure as the plane of the rotor sweeps past that tooth, an event occurring twice per revolution. The rotation of the plane of the rotor imposed by the casing, displaces the fluid in the ring chamber - bounded by the interior raceway of the housing, the casing flanges and the rotor faces - toward the outlet port, creating suction on the inlet port by the receding face of the rotor. A sequential diagram of this process is shown in Fig. 3.7. A note on this pump was published in Machine Design Magazine in 1964 (Anon., 1964).

4. RESONANT VANE VACUUM PUMP

This invention was derived from the single vane pump described in Chapter 2. As was explained there, a vane executes a reciprocating motion with respect to a grooved rotor (Fig 2.1) and, if $\beta=0$, the motion is simple harmonic. Now, consider the addition of a spring to provide a restoring force on the vane toward the center of its stroke. If the spring constant is chosen so that the natural frequency of the resulting spring-mass system relative to a rotating frame of reference coincides with the frequency of the harmonic motion imposed by the constraining stator, a resonant condition results. Viewing this as a forced vibration brought about by contact forces on the vane's tips, with a very large amplification factor, leads to the conclusion that those contact forces are minimized. Thus, friction power loss and wear are reduced. This is particularly suited to a vacuum pump application since other losses caused by fluid drag are minimal in that case.

To establish the equation of motion of the vane's center of mass, we note that Eq. (2.1) represents the trajectory of a blade tip. By subtracting from r half the vane's length, L , and putting $L - \rho = e$, we obtain the expression for the radial coordinate, R , of the center of mass:

$$R = -e \cos \theta = -e \cos \omega t \quad (4.1)$$

where ω is the rotor angular velocity and t the time.

The equation of motion for the mass center in the radial direction is:

$$m(\ddot{R} - \omega^2 R) = -kR \quad (4.2)$$

where m is the vane's mass, k the spring constant, and the mass of the spring is neglected.

Thus, the natural frequency of vibration of the vane with respect to its guiding groove is

$$\sqrt{\frac{k}{m} - \omega^2}$$

For resonance, this frequency must equal the frequency ω of the rotation. Writing this relationship and solving for k :

$$k = 2m\omega^2.$$

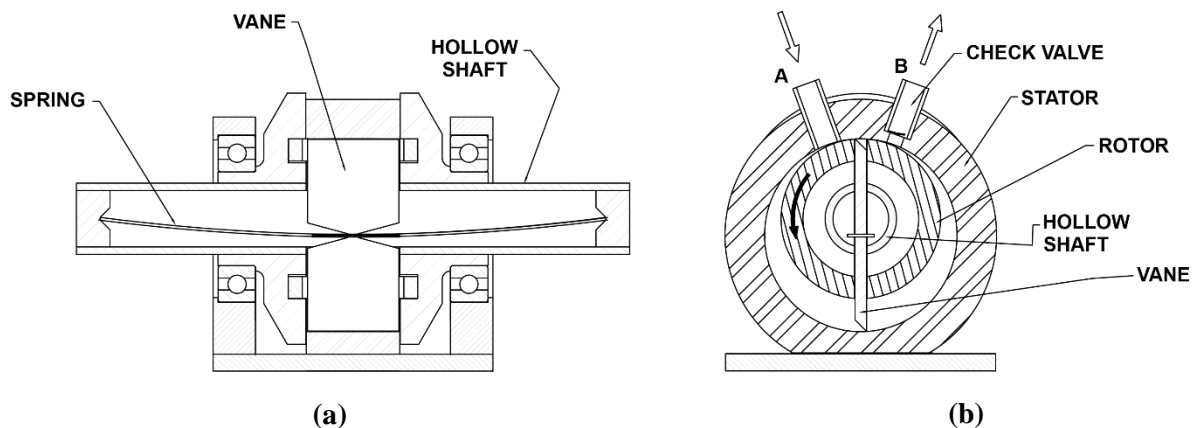


Figure 4.1 Resonant vane vacuum pump, a) longitudinal midsection, b) transverse midsection

A design problem that is encountered is finding the space to accommodate a suitable spring capable of sustaining elastically the required deformation and having the appropriate spring constant. A solution is illustrated in Fig. 4.1 (a), which is a longitudinal midsection of the pump. The spring is a long reed simply supported at its ends and extending inside a hollow shaft. The vane is attached to the spring at its center point. It consists of two identical halves, one on each side of the spring. The tubular rotor shown has diametrically opposed grooves guiding the vane in its reciprocating motion. Fig. 4.1 (b) is a transverse cross section of the mechanism. Air is drawn from a space to be evacuated through inlet A and discharged to the atmosphere through outlet

B after passing through a check valve. A prototype pump is shown in the photograph of Fig. 4.2. In designing this prototype, the distributed mass of the spring was not neglected.



Figure 4.2 Resonant vane vacuum pump prototype

5. ROTARY ENGINE

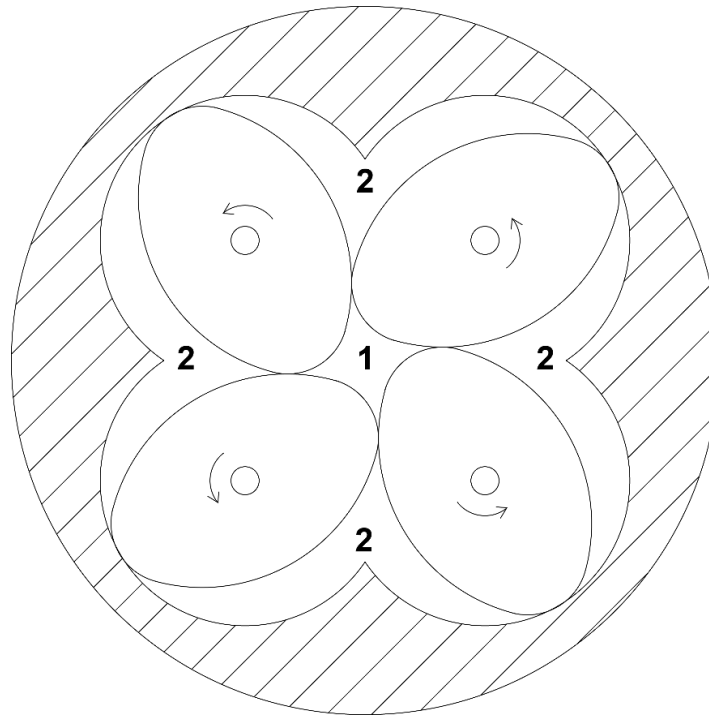


Figure 5.1 Four-rotor rotary engine mechanism. Main chamber: 1; external chambers: 2

This chapter refers to a positive displacement rotary machine that may be conceived as an internal combustion engine or a steam expander and is based on a concept at least 80 years old. Hopkins patent (Hopkins, 1937), issued in 1937, describes the geometry and basic operation of the mechanism, shown schematically in Fig. 5.1. There are four identical rotors, each mounted on a shaft with a gear that meshes with a central gear on the output shaft causing the rotors to be synchronized in their rotation. The centers of rotation of the rotors lie at the vertices of a square. The chamber at the center of this arrangement, bounded by the rotors and two cover plates, has a variable volume going from minimum to maximum and back to minimum twice in each revolution of the rotors. Also shown in Fig. 5.1, are four

additional external chambers of variable volume which appear in some patents related to this mechanism. Other patents, besides the Hopkins patent mentioned before, using the same basic concept, are those of W. A. Homan (1920), R. H. Wise (1966) and D. K. Campbell Jr. (1969). Even though Homan's patent antecedes that of Hopkins, mentioned earlier, by nineteen years, it differs from it in that the rotors have an elliptical contour, which is an approximation to the correct geometry consisting of four circular arcs, as specified by Hopkins. As in Homan's patent, the rotors in both Wise's and Campbell's patents are elliptical, despite the fact that they were issued many years after the Hopkins patent. Campbell should be given credit however, for proposing seals for preventing leakage between adjacent rotors. Wise's patent refers to a pump while the other three refer to Otto cycle internal combustion engines. In Homan's patent, intake and exhaust poppet valves open directly to the work chamber. Hopkins describes a system with ports formed by grooves on the side plates and rotor faces. In Wise's patent, rotary valves are employed, and in Campbell's there are intake and exhaust ports that are periodically uncovered by the rotors.

There is apparently no record of any engines or pumps built according to any of the patents previously mentioned.

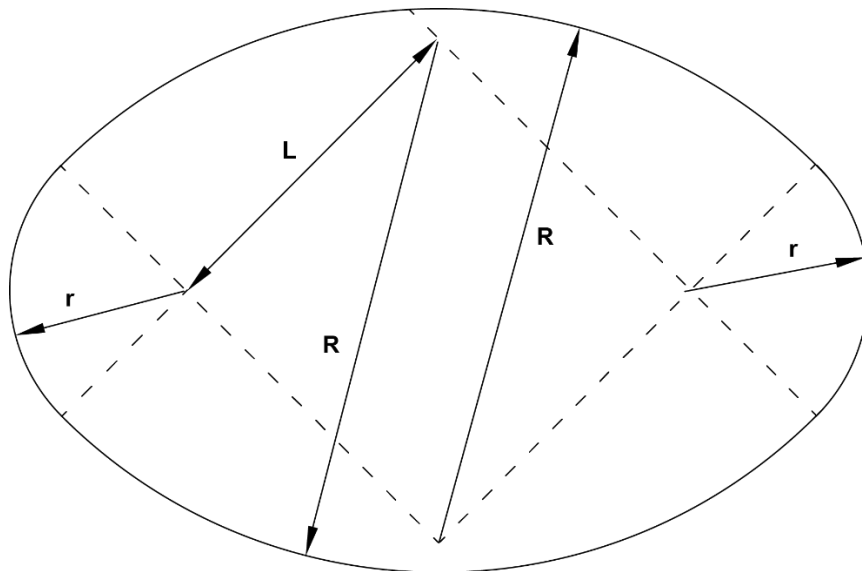


Figure 5.2 Rotor geometry

Before discussing my incursion in this topic, it is convenient to define the rotor geometry, as established by Hopkins and shown in Fig. 5.2. The rotor contour consists of four circular arcs, each subtending 90° and centered on the vertices of a square with sides of length L . The radii of the arcs centered at two opposite vertices is R , and the radii of the other two arcs is r , where $R > r$, and $R - r = L$. The centers of rotation of the four rotors of the engine lie at the vertices of a square having sides of length $R+r$. The minimum and maximum chamber volumes are respectively $(4 - \pi)R^2h$ and $(4 - \pi)r^2h$, where h is the rotor height.

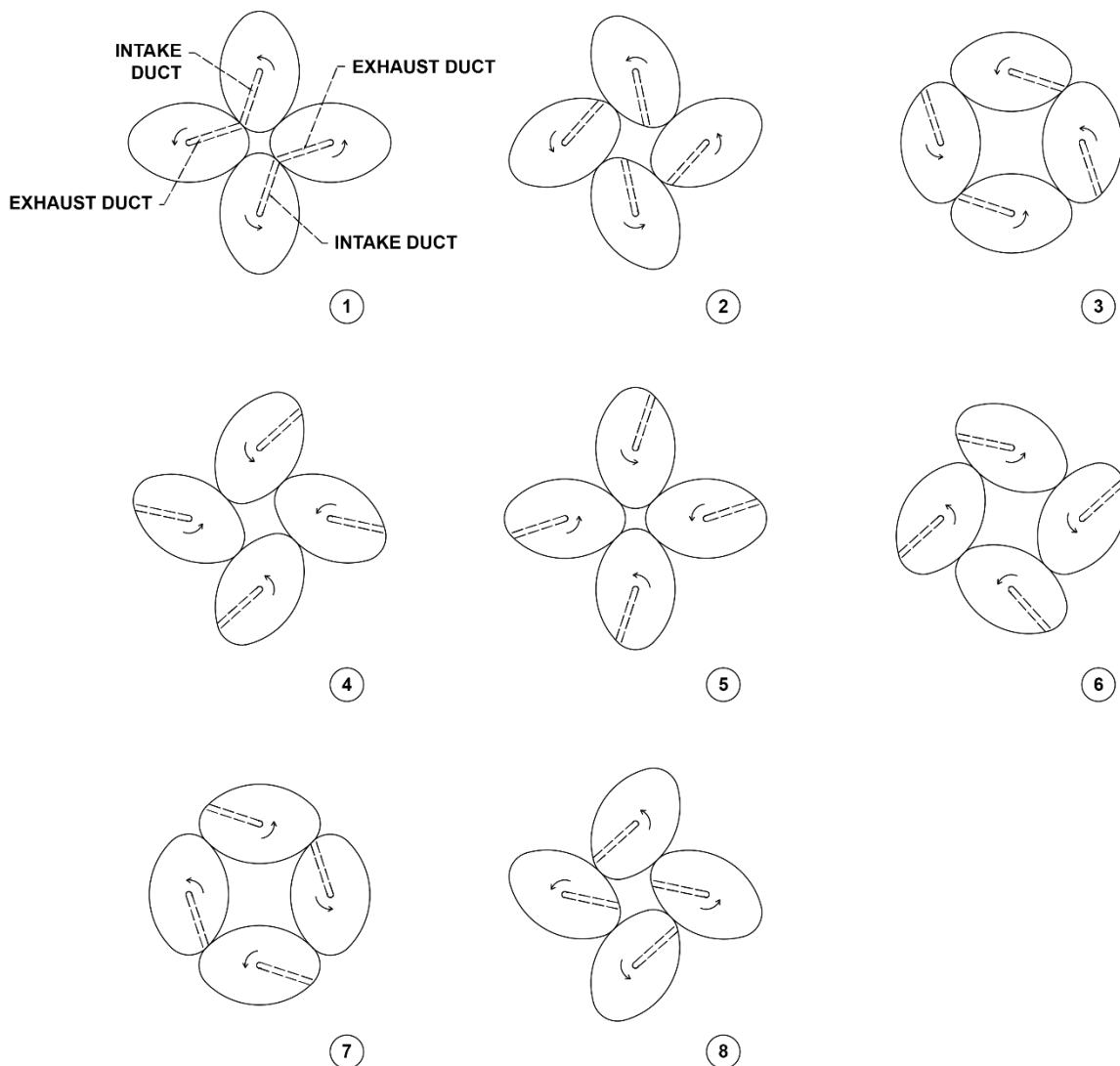


Figure 5.3 Sequence of operation of rotary internal combustion engine. Intake: (1) - (2), compression: (3) - (5), ignition: before (5), expansion: (5) - (7), exhaust: (7) - (8) - (1)

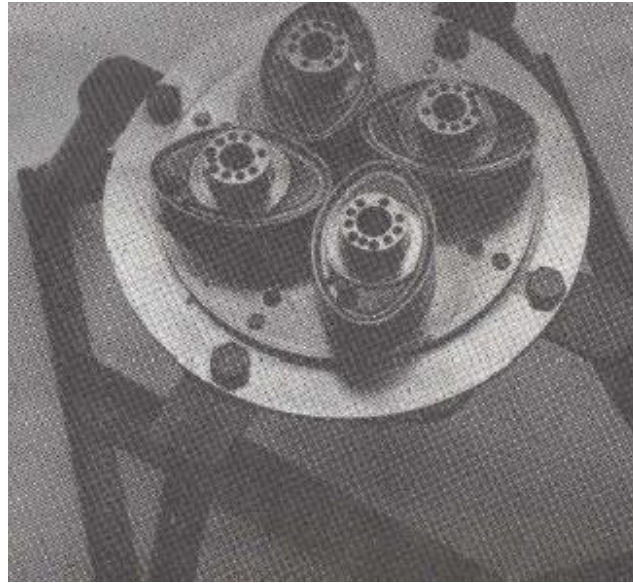


Figure 5.4 Partially assembled rotary internal combustion engine prototype

My interest in this machine has persisted for many years, starting with an internal combustion engine concept from the 1970's. In that design, the rotors themselves doubled as rotary valves. They had radial ducts communicating with axial passages in the rotors' shafts. Two opposing rotors performed as intake valves and the other two as exhaust valves. The intake axial passages in the rotors' shafts were fed from an intake manifold and the flow out of the exhaust axial passages discharged into an exhaust manifold. Stationary tubes within these passages blocked the flow except at appropriate times determined by the position of windows.

The prototype engine that was built had a compression ratio of 9. The sequence of operation is depicted in the diagram of Fig. 5.3. As seen, a four-cycle operation is achieved in one revolution of the rotors. The partially assembled engine is shown in the photograph of Fig. 5.4. Motoring tests to obtain compression curves were carried out at speeds up to 5,800 rpm (Chicurel, Sept. 1975) at which speed a gage pressure of 0.74 MPa was reached. The experimental points agreed well with theoretical curves based on leakage clearances of 0.04mm. The altitude at which the tests were conducted was 2,200 m where barometric pressure is only 0.8 atmospheres. We had hoped that the engine would run with no seals of

any kind. Tests were made in which the engine was driven by an electric motor through a one way clutch, but a self-sustaining operation was not achieved. A minor consolation from this experience is that the engine did run ahead of the driving motor for short periods.

Around that time, interest was growing in small steam engines for converting waste or solar derived heat into mechanical power. It seemed fitting to adapt the rotary engine concept for operation as a Rankine expander. In the first version of this machine, flow in and out of the work chamber was controlled by the same type of rotary valves as in the internal combustion version (Chicurel, Sept. 1975) (Chicurel, Sept. 1976) (Chicurel, 1979). The rotors however, were quite different. Instead of $r/R = 3$, used for the internal combustion engine, that ratio changed to 0 , resulting in "pointed rotors". The operating sequence for the Rankine expander is shown in Fig. 5.5. To allow greater expansion of the steam after admission cut off, and to reduce the pressure losses in the admission and exhaust passages, another design, shown schematically in Fig. 5.6, was implemented. Two opposed rotors act as admission valves. These have disks that fit in recesses in a cover plate. The disks have windows coinciding with rotor channels that discharge into the work chamber. As these windows pass under similar shaped openings in the cover plate, steam flows into the work chamber. Tubular carbon seals prevent leakage of high pressure steam at the interface between the disks and cover plate. A diagram of this arrangement is shown in Fig. 5.7. Exhaust ducts in the four rotors direct the spent steam directly to the atmosphere without turns. Fig. 5.8 is a photograph of the partially disassembled engine, and Fig. 5.9 a photograph of the same engine in operation with steam generated by a solar parabolic trough collector.

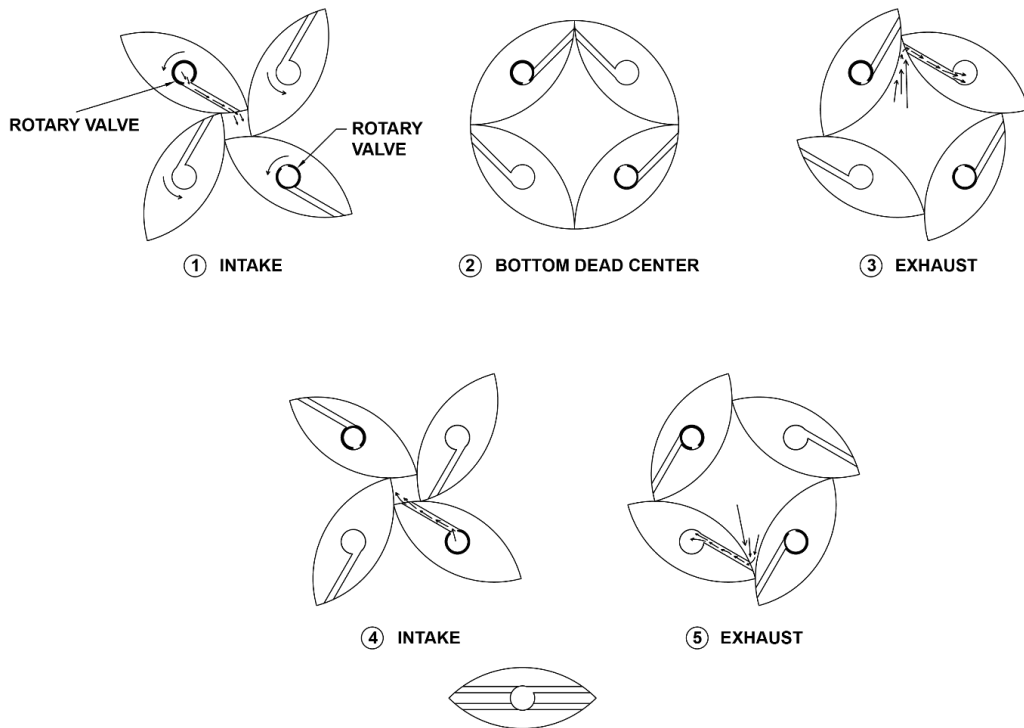


Figure 5.5 Operating sequence of Rankine cycle expander

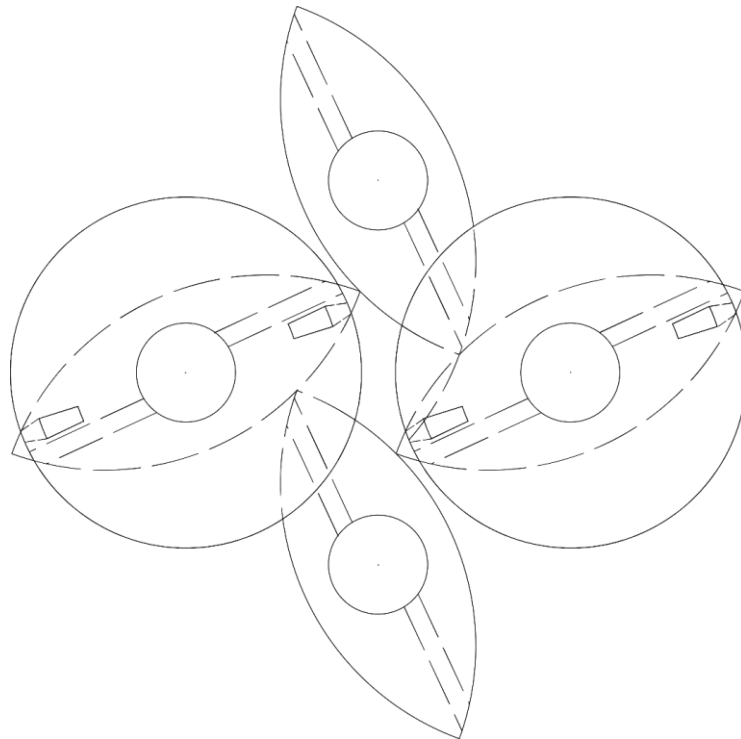


Figure 5.6 Modified version of Rankine cycle expander

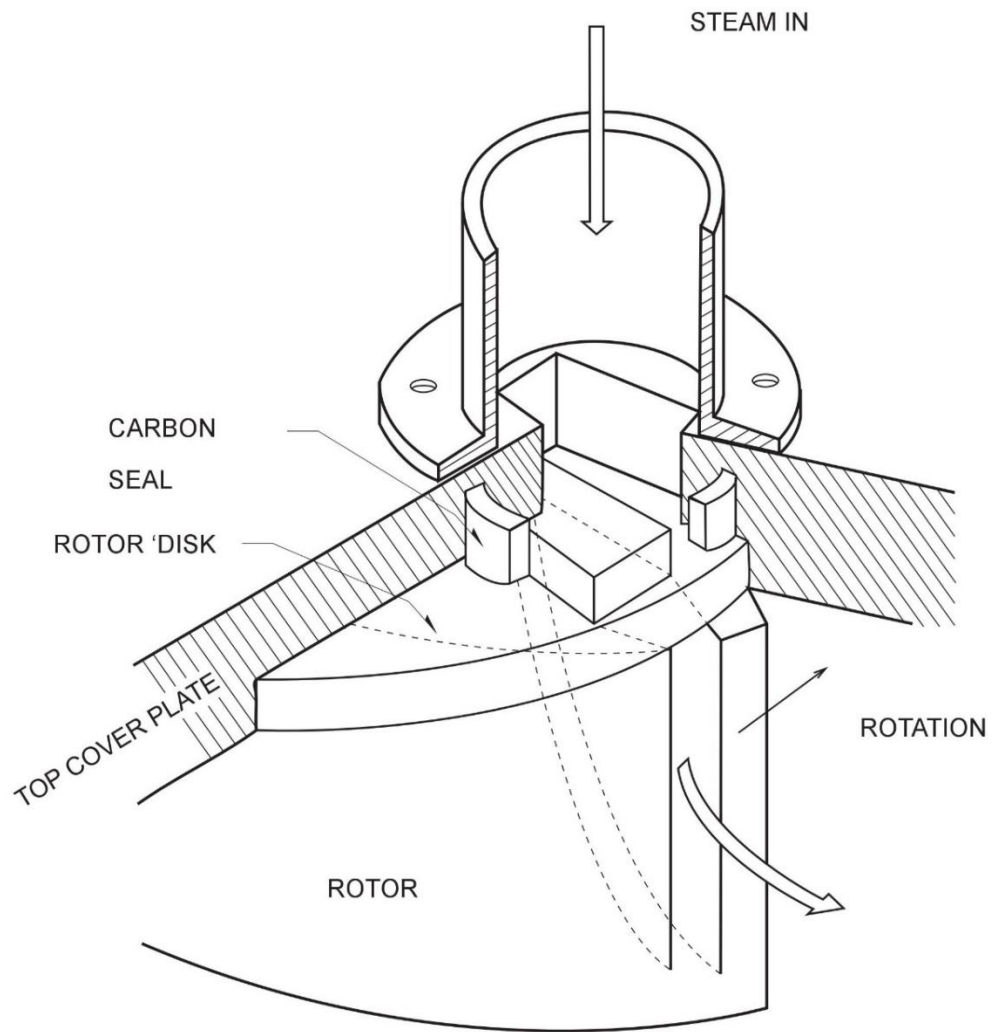


Figure 5.7 Diagram of steam inlet system

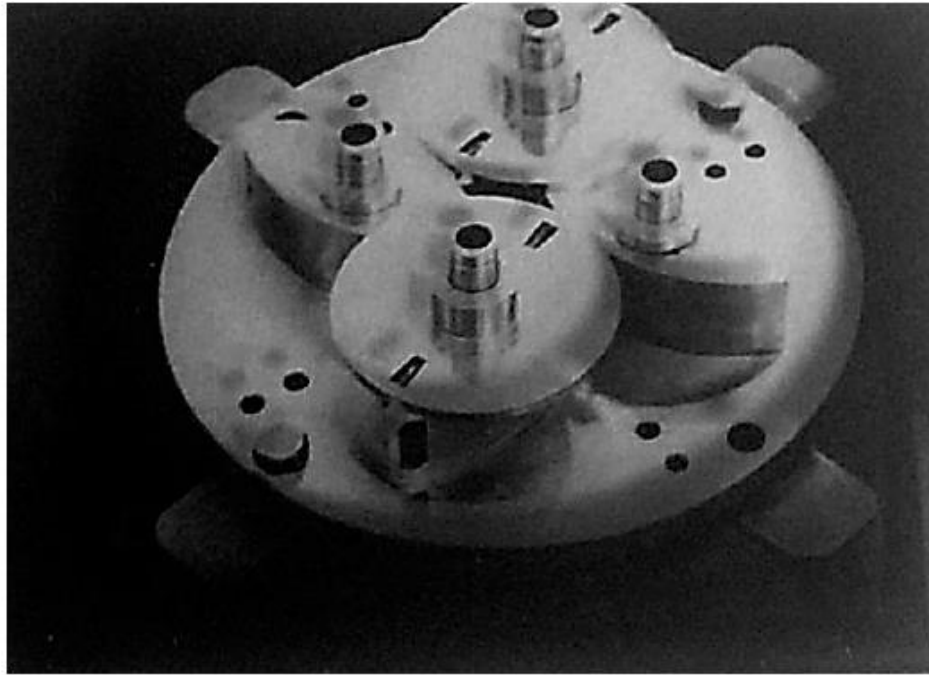


Figure 5.8 Partially disassembled rotary steam engine



Figure 5.9 Rotary engine operating with solar generated steam

A limited testing program to evaluate the performance of both versions of the engine using compressed air at 0.4 and 0.6 MPa, instead of steam, was carried out and reported in (Chicurel, Sept.1980). A maximum peak power of 140 W at the higher pressure was obtained with the first version of the engine at around 900 rpm. The maximum pressure in the work chamber dropped sharply with speed indicating poor “breathing”. The steam expander project was abandoned in the 1980’s, but a new effort to revive it was launched in 2007. The main focus this time was to reduce pressure losses in the admission system. The result was a modified design in which high pressure steam is admitted through a central port in one of the side covers and controlled by means of a balanced reciprocating valve driven by a crank connected to a rotor shaft (González, et al., 19-25 June 2011). Another objective of the revived project was to develop apex and face seals for the work chamber (Chicurel, et al., 2010). Fig. 5.10 is a simplified diagram showing the engine work chamber during exhaust. A spring plus centrifugal force act on each apex seal to guarantee its contact with the flank of a neighboring rotor. The seals are not aligned with the rotor tips, but offset to one side, the side that is exposed to the work chamber during exhaust. Thus, the high pressure during admission-expansion does not act on the seals, avoiding their being pushed into their grooves. The extreme outward position of the seal is limited by a stop. Crucial to the apex seal design is that it allow a smooth passage as the tips of two rotors come together, which occurs twice per revolution, at minimum and maximum work chamber volume. As seen in Fig. 5.10, the seals have a chisel shaped tip. Thus, for example, when the seal of rotor B wipes the flank of rotor A, it encounters a small step causing only a brief loss of contact.

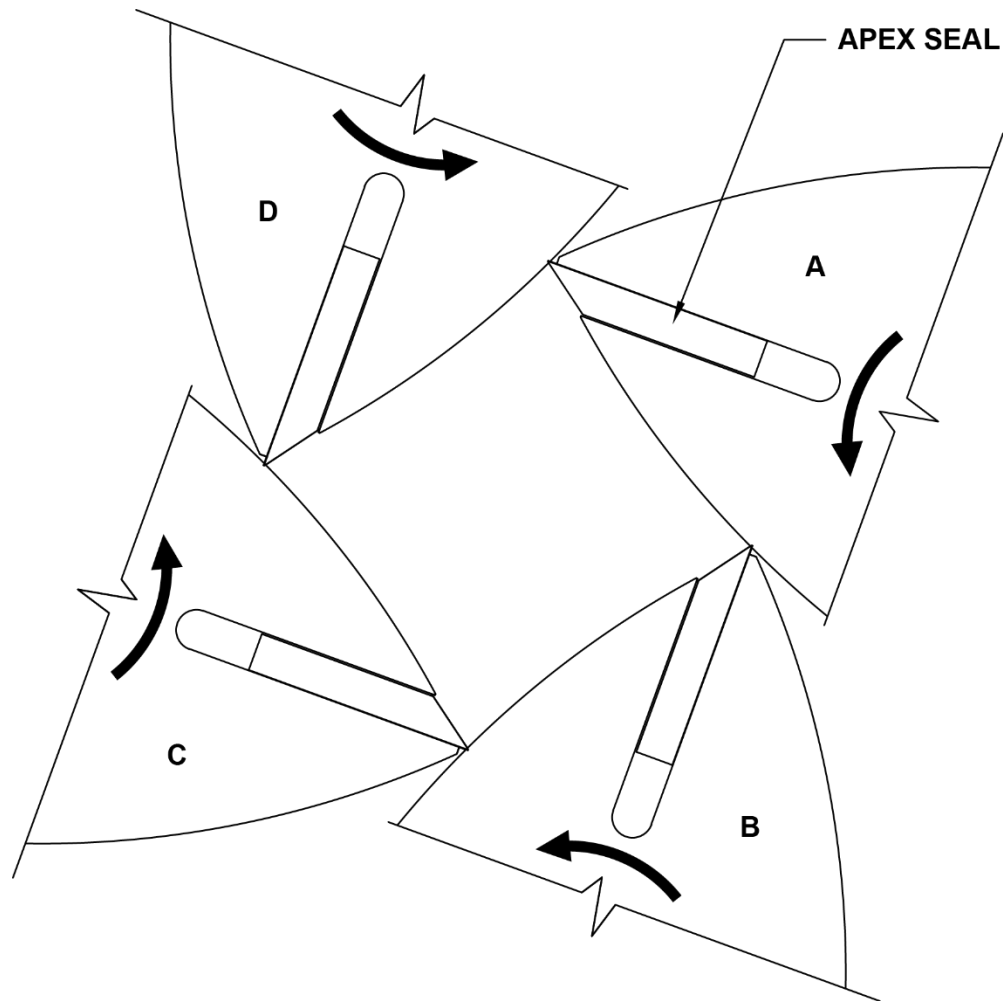


Figure 5.10 Apex seal design

Fig. 5.11(a) shows a face seal and Fig. 5.11 (b) a diagram showing the cross section of a rotor with this type of seal on top and bottom faces. The seal fits in a depression bordered by slant surfaces extending along the edges of the rotor face. The seal is made from a thin plate with grooves causing it to act as a spring to press against the slant surfaces. Thus, a slight pressure from a cover plate on its periphery generates pressure against the slant surfaces and produces an in-plane contraction. A dynamometer test with compressed air, of the modified design incorporating the reciprocating admission valve and a full set of seals, yielded encouraging results: the indicated power calculated from work chamber pressure curves, was 79.5% of the ideal cycle. Fig. 5.12 is a photograph of the engine.

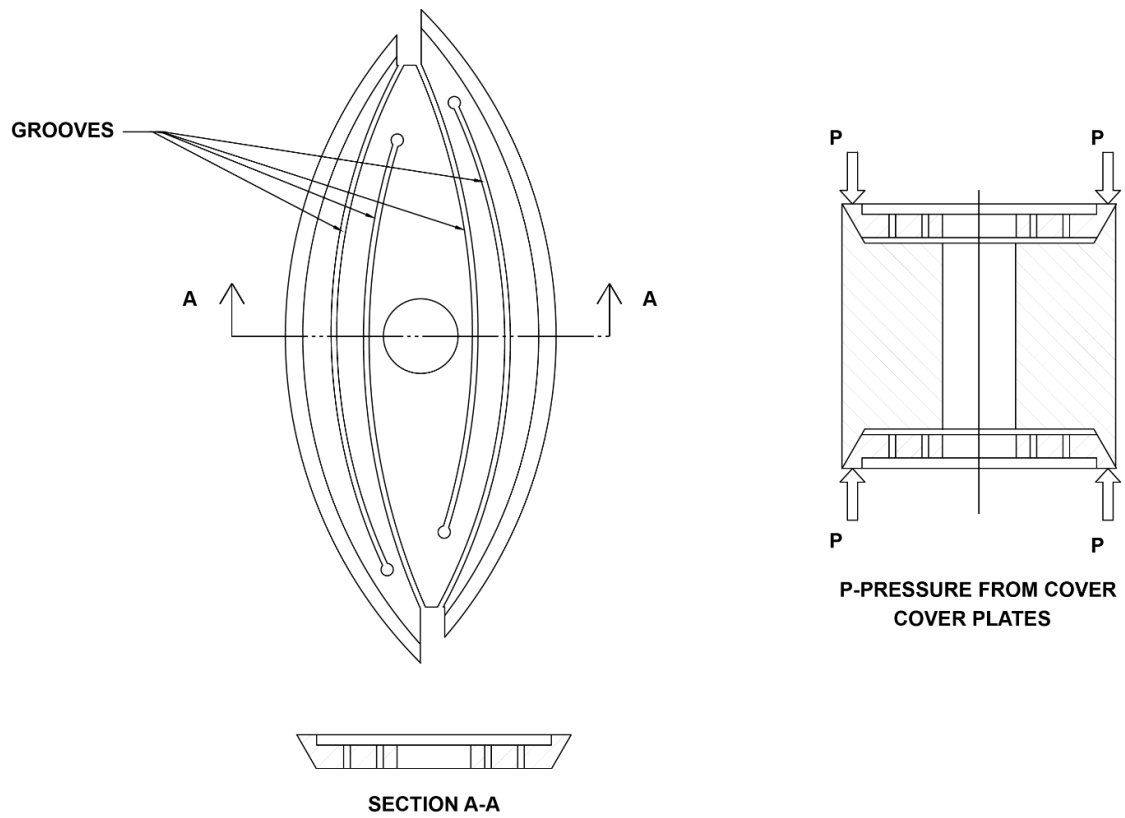


Figure 5.11 a) Face seal b) Rotor mid cross section with face seals



Figure 5.12 Engine with reciprocating admission valve and full set of seals

6. SINGLE CYLINDER BRAYTON CYCLE ENGINE

Whenever “Brayton cycle” is mentioned, one thinks of the gas turbine; however, reciprocating piston engines based on this cycle might, in some cases, compete with the popular internal combustion engine. Interestingly, the original machines built by George Brayton were, in fact, reciprocating piston engines.

In a gas turbine, the working fluid, air, is first compressed, then enters a combustion chamber where fuel is injected and then expands delivering power in the turbine stage. In a Brayton cycle piston engine, the compression and expansion stages occur respectively in compression and expansion cylinders.

J. R. Rosa published calculations (Rosa, 1983) showing that a closed Brayton cycle piston engine is competitive in efficiency with internal combustion engines. Such an engine, receiving heat by external combustion, may operate with a wide variety of fuels. Another engine that shares this advantage, is the closed cycle Stirling engine; however, to obtain the high efficiency for which it is noted, it is necessary to use hydrogen or helium at high pressures as working fluid, leading to a complicated technical challenge. Fryer et al (Fryer, et al., 1973) made a detailed cycle analysis of a closed Brayton cycle piston engine, and designed and built such an engine which they tested. Although its performance suffered from the technical difficulties encountered in a first prototype, it provided convincing evidence that the Brayton cycle piston engine does have a future. More recently, Holtzapple et al (Holtzapple & Rabroker, 2005) patented a Brayton cycle engine concept with piston or “gerotor” (a positive displacement rotary machine) compressor and expander.

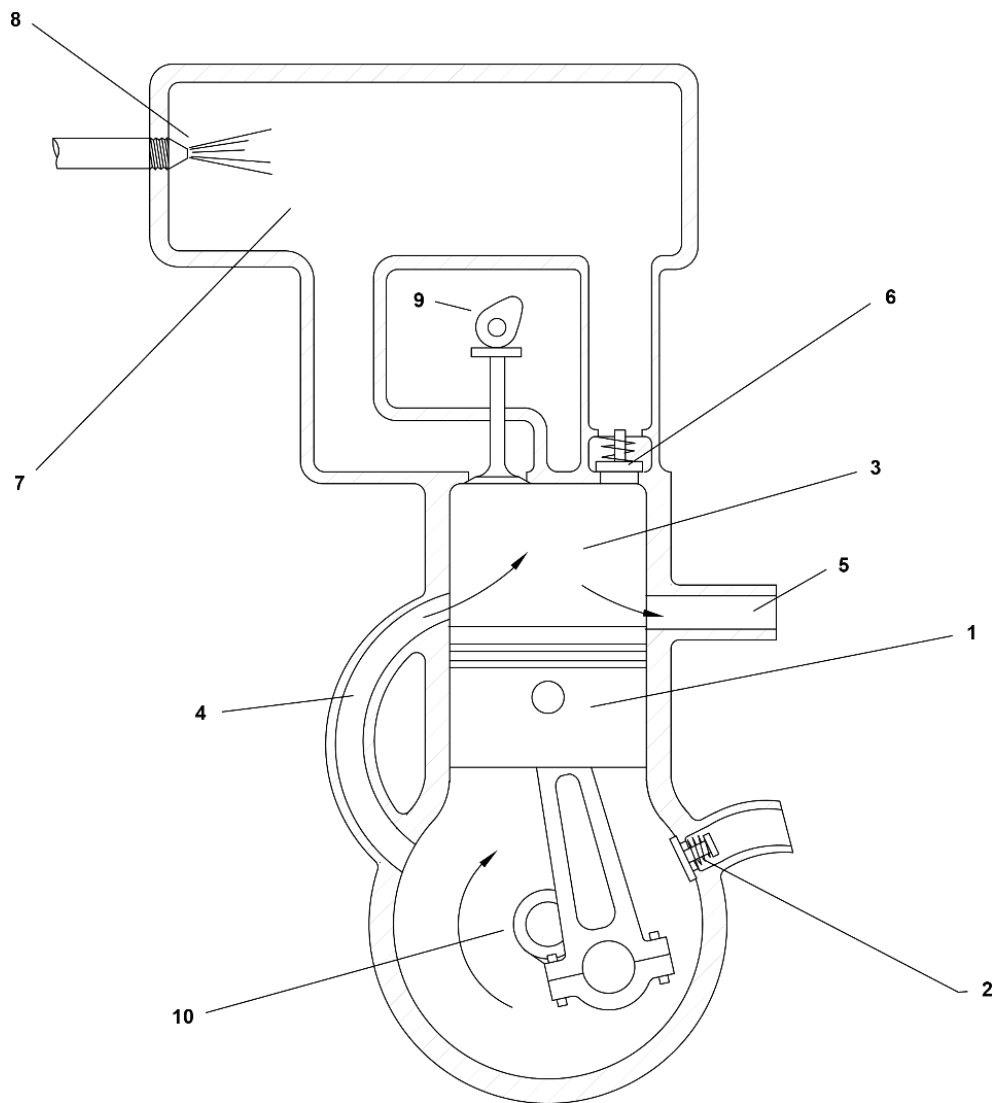


Figure 6.1 Single cylinder Brayton cycle piston engine

In the mid 1970's my friend and colleague, Gustavo Nájera-Díaz, and I were granted a Mexican patent (Chicurel, 1976) for a preliminary concept of an open Brayton cycle engine having a single cylinder in which both processes take place. The concept is an adaptation of the two-cycle internal combustion engine and is depicted in Fig. 6.1. As the piston 1 moves up, it produces suction in the crankcase below causing the check valve 2 to open and allow air to enter, rather than a fuel-air mixture, as in the case of the internal combustion engine.

As the piston then moves downward, the air is slightly compressed and displaced into the cylinder chamber 3 through duct 4. The rush of fresh air, in turn, displaces expanded hot gases from a previous cycle out the exhaust pipe 5. As the piston begins its upward motion, it blocks both intake and exhaust ports and, as it continues its upward travel, it compresses the trapped air until check valve 6 opens allowing the flow of the compressed air into the combustion chamber 7, where fuel is being injected continuously by means of nozzle 8. Starting at top dead center position of the piston, cam operated valve 9 is opened briefly to allow a charge of high pressure gas to flow into the cylinder where it then expands delivering power to crankshaft 10.

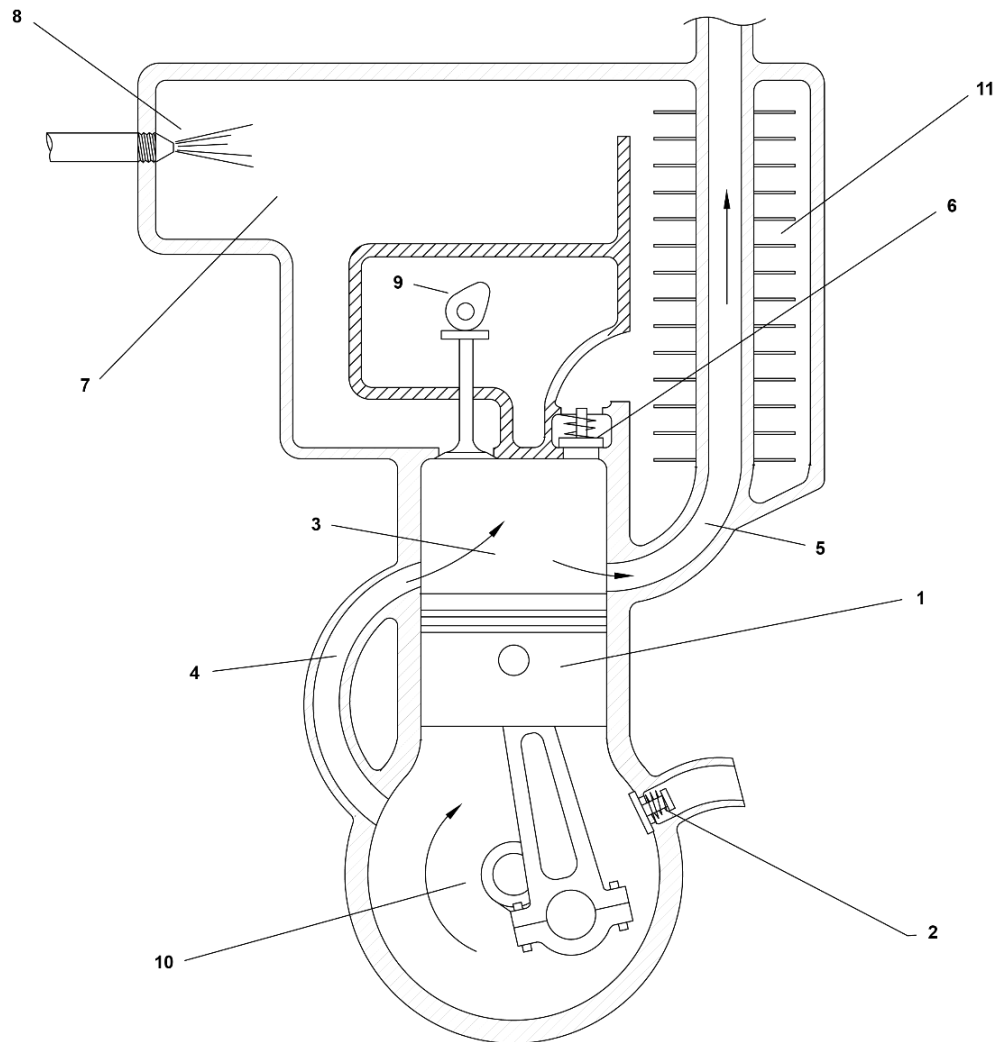


Figure 6.2 Single cylinder Brayton cycle piston engine with regenerator

The simple concept described above lacks a regenerator to transfer heat from the exhaust gasses to the compressed air before entering the combustion chamber, a minimum necessity to obtain an acceptable efficiency. Fig. 6.2 is a diagram of the engine incorporating such a regenerator, labeled 11.

It must be stressed that this is only a preliminary concept that has the appeal of simplicity, but I cannot claim much else.

7. TRANSMISSIONS WITH MAGNETIC GEARS

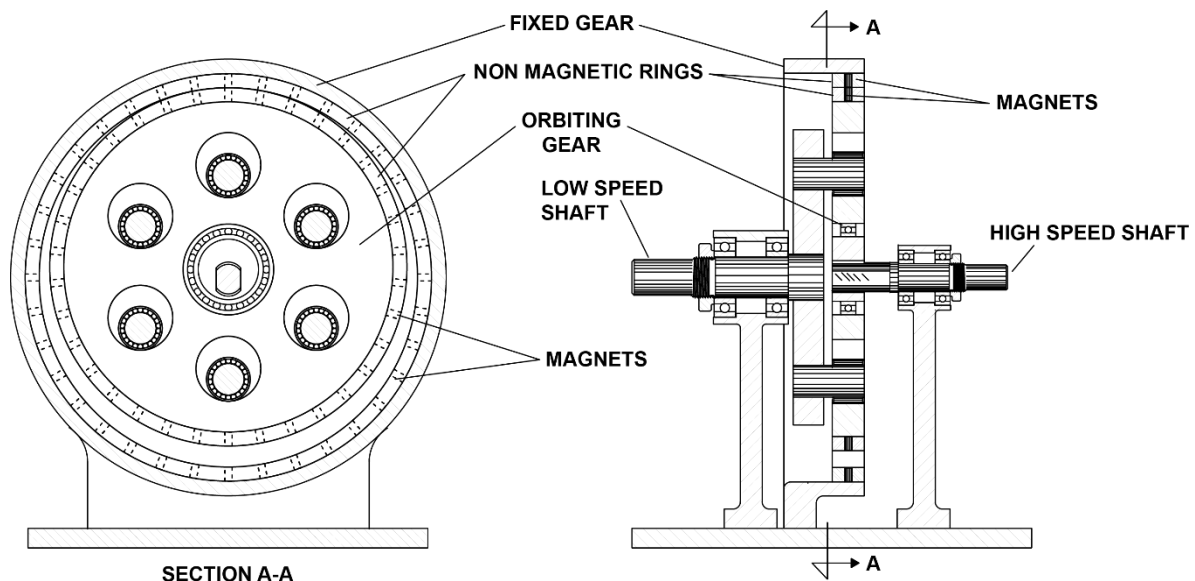


Figure 7.1 Cross sectional view of cycloidal magnetic gear speed reducer L.E.

The use of magnets for the transmission of mechanical power is rapidly growing, with great gains having been made so far in the twenty first century. There are two types of magnetic transmissions: (1) those which have “magnetic gears”, kinematically equivalent to conventional gears, but with magnets that act as teeth in that they lock the relative positions of two such gears in mesh, and (2) those that depend on a commutating action in which the fields of magnets on a wheel are channeled by permeable elements to interact with magnets on another wheel. The contributions that I shall discuss now fall in the first of these categories. In this work, there have been a number of collaborators. One of them is Prof.

Gabriel Ascanio-Gasca, who has participated since the start in 2013. Three theses, authored by Luis Alberto Vázquez-Pérez, Jesús Omar Sánchez-Morfín, and Francisco Isaac Lopez-Alegría, were based on magnetic gear projects, and three more are being written by Omar Tenorio-Fuentes, Fo-Lyn Yong-Primerero and Luis Enrique Calderón-Escutia.

Three prototype cycloidal speed reducers have been built. Fig. 7.1 is a cross-sectional view of the first. It has an orbiting gear with external “teeth” driven by the high speed input shaft by means of a tongue and groove coupling equivalent to a crank. This gear meshes with a stationary gear with internal “teeth”. Each “tooth” consists of a pair of permanent magnets having opposite polarities exposed. The orbiting gear has 52 magnets and the stationary one 54 resulting in a speed reduction ratio of 26:1. A photograph of this unit is shown in Fig. 7.2. The second speed reducer built is a modified version of the first one having rectangular cross section magnets with 2.5 greater pole area than those of the first prototype. Due to difficulties in positioning the magnets correctly, further work on that unit was suspended. A third prototype is a portable demonstration unit, weighing 3 kg, with a speed reduction ratio of 12:1 and having 24 and 26 magnets respectively in the orbiting and stationary gears. The housing parts were produced by 3D printing using ABS polymer. A photograph of this speed reducer is shown in Fig. 7.3.



Figure 7.2 Cycloidal magnetic gear speed reducer prototype

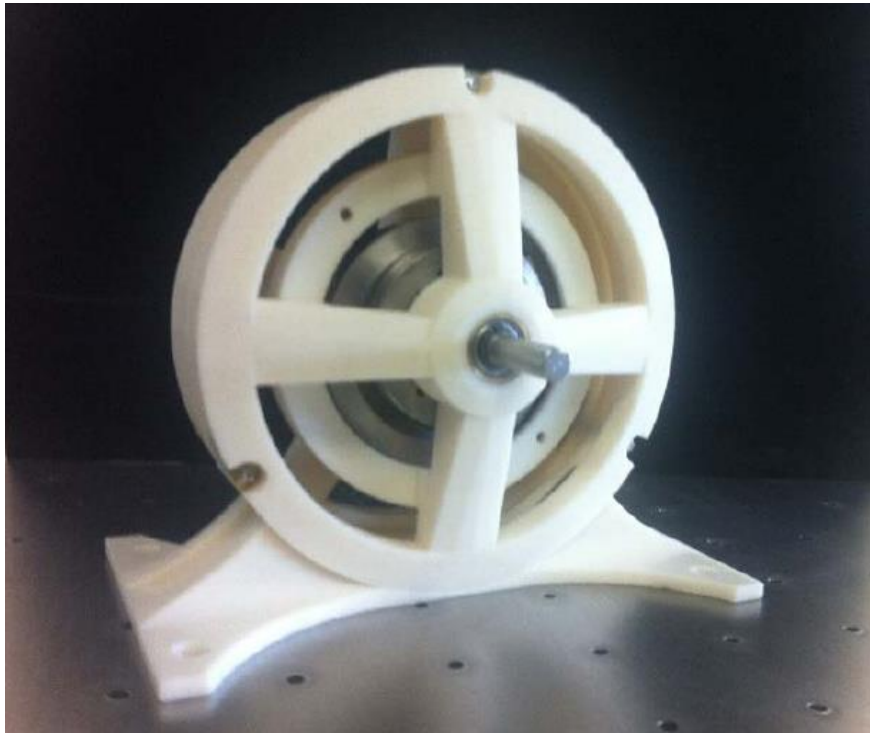


Figure 7.3 Light cycloidal magnetic speed reducer

Unlike other transmissions with magnetic gears, in our speed reducers there is no gap at the mesh point of two gears; the magnets are embedded in nonmagnetic rings that contact one another allowing one gear to roll on the other. Moreover, one of the two gears is unrestrained in the direction of the force of attraction exerted by the other thus giving rise to an equal and opposite contact force. This patented feature (Chicurel, et al., 2015) (Chicurel, et al., 2018) has the following advantages: (1) The normal contact force is accompanied by a friction traction force, increasing the capacity to transmit torque, (2) because the magnets are inserted into holes of the nonmagnetic rings leaving the smallest margin possible so they do not protrude therefrom, the magnetic attraction forces are maximized, (3) the complications in machining and assembly of parts associated with maintaining small gaps at mesh points are avoided, and (4) the contact force between gears balances the attraction exerted by each other, thus eliminating the load that would otherwise have to be resisted by bearings. As is usual in cycloidal transmissions having standard toothed gears, an arrangement for transferring the slow rotation of the orbiting gear to an output shaft was used

here. It consists essentially of a disk of the output slow shaft with pins that penetrate into holes of the orbiting gear, filtering out the high speed circular translation. An improvement introduced in this mechanism, included in the claims of the patent for the speed reducer, is the introduction of needle bearings to reduce frictional losses at the pin-hole interfaces.

The first speed reducer prototype was subjected to limited dynamometer testing (Chicurel, 2013) that showed promise as to acceptable efficiency and torque capacity. The third prototype has been used in reverse, i.e., as a speed increaser, to convert the slow rotation of a small wind turbine to a high speed output to drive a small generator. This test was carried out in a wind tunnel (López-Alegría, et al., Sept. 2015). The experience motivated a study to determine under what conditions is a cycloidal magnetic speed reducer reversible. Such a study is included in (Chicurel, et al., s.f.).

A ramification of this work consists in the evaluation of gears with magnets having their axes oriented axially rather than radially. A particularly interesting case is that of a gear having a single ring magnet. Such a gear has a layered construction with two identical side-disks made of a magnetically permeable material, such as iron or carbon steel, and sandwiched in between, a circular ring permanent magnet of smaller diameter, magnetized in the direction of its geometric axis, thus having north and south poles on its faces. An alternative to the ring magnet is a number of cylindrical smaller magnets, also magnetized in the axial direction, and distributed circumferentially. The outer side-disks have a number of uniformly spaced teeth around the periphery. One such gear may drive another one also having a ring magnet or circumferentially distributed magnets of reversed polarity, by a magnetic “meshing” action causing an alignment of the teeth of both gears in proximity of one another. The teeth provide a path for the magnetic field, which extends to the teeth of the other gear. Thus, magnetic lines of force originating in the north pole of the magnet(s) in one gear extend to the south pole of the magnet(s) in the other gear, and vice-versa (Chicurel, et al., 2016). It is even possible to eliminate the magnet(s) in one of the gears by substituting it (them) by a steel ring that provides a path for the magnetic field originating in the other gear. A photograph of a small demonstration unit in with only one ring magnet in the larger gear is shown in Fig. 7.4. A Mexican patent application for magnetic gear

transmissions with axially oriented permanent magnets, has been filed in 2017 (Chicurel, et al., 2017)



Figure 7.4 Demonstration unit of transmission having only one ring magnet

8. FLYWHEEL ENERGY STORAGE WITH MECHANICAL INPUT-OUTPUT

The system proposed in this study (Chicurel, 2014) resulted from the search of a very simple and inexpensive manner of recovering braking energy of a vehicle via a flywheel. In stop and go traffic, typical of congested cities, a modest amount of kinetic energy associated with the vehicle's motion may be transferred to a flywheel for braking. The energy thus stored may be reused after a short time to contribute to accelerate the vehicle once again. Each cycle is similar to the previous one. Operation of the system is controlled by clutches. For braking, a cable winds onto a pulley of the vehicle's propulsion driveshaft while unwinding from a pulley coupled to the flywheel. The cable windings are in one plane so that the effective radius of the first pulley increases during this operation while that of the flywheel pulley decreases. A continuously variable transmission is effectively obtained. When it is required to accelerate the vehicle, the kinetic energy of the flywheel is transferred to the propulsion driveshaft in a reverse operation, that is, the cable winds onto the flywheel pulley while unwinding from the driveshaft pulley. As may be appreciated, an increasing driven-to-driving pulley velocity ratio results during braking or accelerating periods. The components of this system are simple and widely used in various mechanisms. Because the flywheel in this system is an energy storage device with mechanical input-output, with no energy conversion taking place, a high efficiency of the system is anticipated. When the system is not active, the pulleys' angular positions, and therefore their effective radii and speed ratio, are monitored. If this speed ratio differs from the one dictated by the driveshaft and flywheel instantaneous speeds, the pulleys are rotated by means of small servo motors to correct the discrepancy. In this manner, when the system is brought into action by engaging clutches, no slippage or cable shock loading occurs. The system is not very flexible, that is, once it is started for braking or for accelerating the vehicle, a deceleration or acceleration event that follows a fixed course is initiated. In order to circumvent this

problem, the approach that will now be described, was introduced. First, the acceleration and deceleration excursions that the system provides are chosen on the basis of the most frequently used. Then, if it is desired to decelerate at a lower rate than the flywheel system would provide, the system is not used and a conventional dissipative brake is applied. If, on the other hand, a higher deceleration rate is desired, the system is brought into operation and the dissipative brake provides only the excess needed. Similarly, if the acceleration required is less than that which the flywheel system would provide, the power would be supplied by the vehicle's engine alone, otherwise the engine would provide the excess over that.

9. PASSIVELY PRESSURIZED BEARING

Bearings may sustain load by means of self-generated oil films that prevent metal to metal contact between a rotating shaft and a stationary journal. The conditions under which this is possible are well established. Alternatively, oil may be pumped by an external means to maintain a pressurized zone resulting in a force on the shaft opposite to the load.

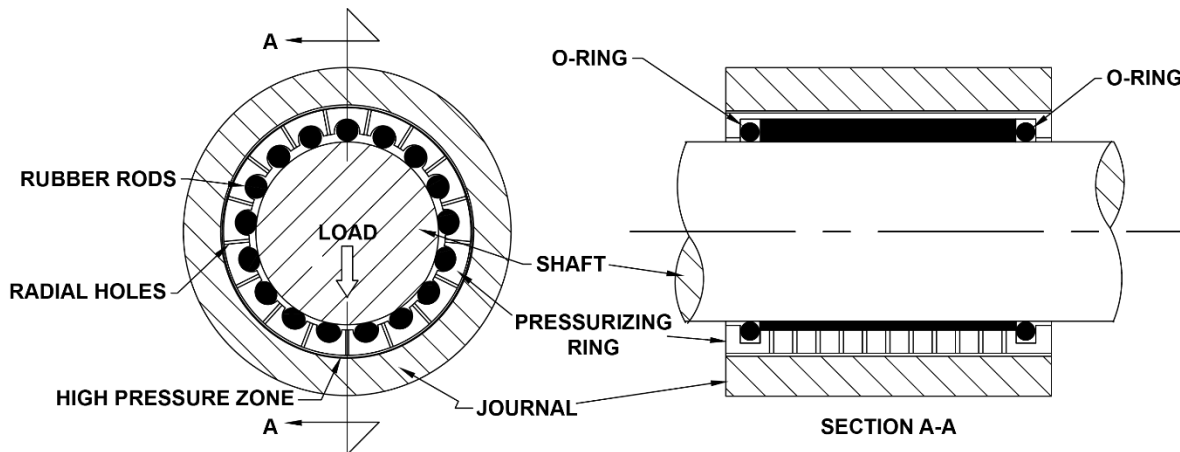


Figure 9.1 Passively pressurized bearing

A method for pressurizing the bearing in a passive manner by introducing what may be called an internal pump, will now be described. The concept is illustrated in the cross sectional diagram of Fig. 9.1. As the shaft rotates, it drags along a pressurizing ring with rubber inserts by means of the friction between such inserts and the shaft. The inserts are rods which fit loosely in longitudinal slots. As may be appreciated, small chambers are created between adjacent inserts in the space between the shaft and the ring. These chambers communicate with the space between the ring and the journal cavity by way of holes along the length of each chamber. The spaces between shaft and ring, and between ring and journal cavity are flooded with lubricant, which may be fed as in a conventional bearing. As the shaft and ring rotate together, the load on the shaft compresses the rubber inserts in the zone under the load, squeezing the oil out of the chambers in that zone. As may be appreciated,

a high pressure zone between ring and journal is created under the load. This zone is continuously being replenished with oil as more chambers enter the loaded zone. In order to reduce the flow of oil out of the pressurized zone at the ends of the bearing, side pressurizing ring lips, as shown in Fig. 9.1, may be provided.

10. WATER AND HEAT ECONOMIZER FOR SHOWERS

The system that is the subject of this chapter is based on a fairly recent idea that I submitted in 2014 to the call for patentable ideas *Primera Convocatoria de la Iniciativa Regional de Patentes para el Desarrollo CAF* (First Call for Proposals to the CAF Regional Initiative for Patents for Development) issued by the bank *Corporación Andina de Fomento*, based in South America. The concept was among the winning proposals, which were entitled to payment of the expenses for a U.S. patent application. The patent was awarded in 2018 (Chicurel, 2018). Hopefully, this will lead to a wide scale application of the invention and of the expected benefits. Following is a brief account of the concept.

When hot water is used for showering or for bathing in a tub, it is usual to let the water run until the water from a heater displaces the cold water remaining in the line. Besides the waste of water that this entails, there is also heat wasted because, as the hot water valve is shut, the hot water remaining in the line, will gradually cool losing the sensible heat it had originally gained in the water heater. In an attempt to avoid the waste of water, users sometimes place a container, such as a pail, to collect the purged water for some appropriate later use instead of letting it go down the drain. Besides the chore involved in doing this, the waste of heat is not eliminated. The economizer prevents the waste of both water and heat by means of an automatic system that collects the purged water and pumps it back into the heater forcing the hot water in the line to return to said heater. This concept is quite different from other schemes to save heat, such as transferring heat from the spent water to preheat water supplied to the cold water shower inlet, as in US patent no. 4,372,372, or recirculating part of the spent water to the shower head after elimination of live microorganisms, as in US

patent no. 8,656,526. These other schemes do not conflict with the economizer here described and may be applied simultaneously to achieve additional savings.

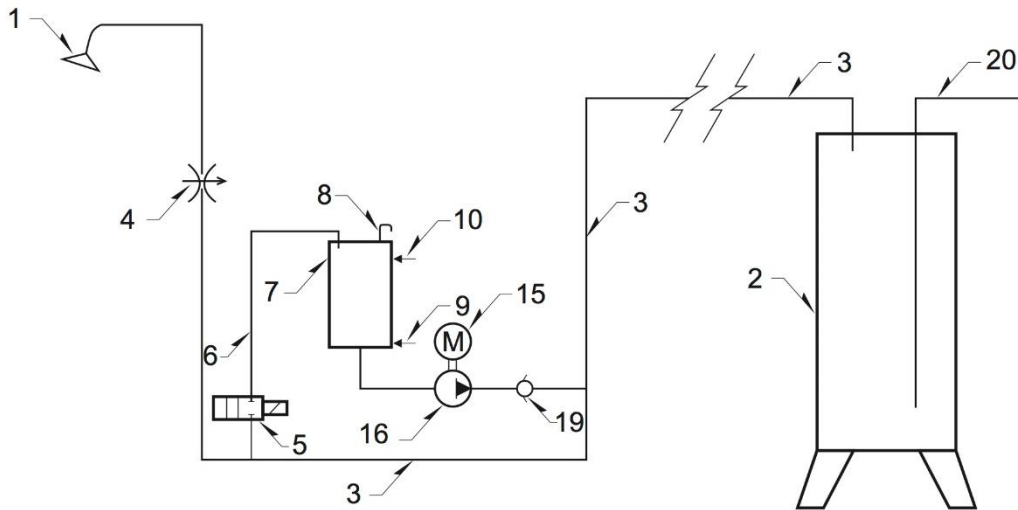


Figure 10.1 Economizer system diagram

In the diagram of Fig. 10.1 is shown a shower head 1 receiving hot water from a water heater 2 through the pipeline 3. Manual control of the flow of hot water is effected by valve 4. When solenoid valve 5 is activated, water from the pipeline 3 flows through line 6 to the holding tank 7, which has a vent to the atmosphere 8. Sensor 9 detects when the water level in the tank is at the lowest point of the operating cycle and sensor 10 when it is at the highest point. In Fig. 10.1 is also shown pump 16 driven by motor 15. Check valve 19 allows flow only in the direction in which pump discharges.

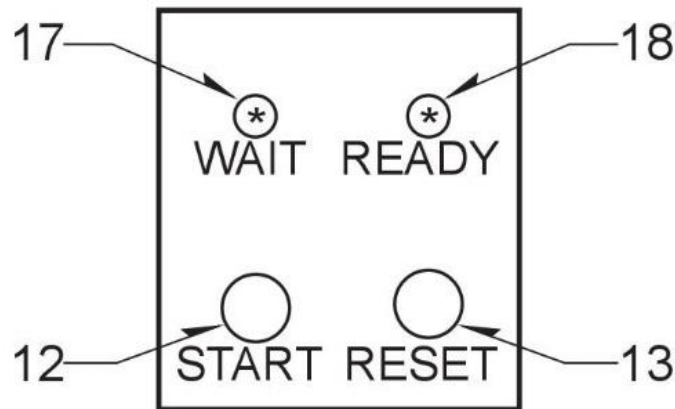


Figure 10.2 Control box for economizer system

Figure 10.2 shows a control box having a display which constitutes the interface with the user. User inputs are by means of the “start” and “reset” buttons. Besides the user inputs to the control box, there are inputs from sensors 9 and 10, and outputs which are required to control solenoid valve 5 and for starting and stopping the pump. To initiate a cycle, the “start” button is pressed, causing solenoid valve 5 to open resulting in the flow of water from line 3 into holding tank 7 through line 6. At the same time, a red pilot light 17 is turned on, signaling the beginning of a waiting period while the holding tank fills up. When the level in that tank reaches the maximum point, a signal from sensor 10 results in the shutting off of solenoid valve 5, while at the same time the red pilot light 17 is turned off and the green pilot light 18 is turned on, signaling to the user that the shower is ready for use. In addition, an audible signal of short duration may be incorporated at this time. Once finished bathing and shutting valve 4, the user presses the “reset” button, causing the pump to start. At the same time, green pilot light 18 is turned off and red pilot light 17 turned on. When the water level in the holding tank reaches its minimum point, sensor 9 sends a signal causing motor 15 to stop while at the same time turning off red pilot light 17. During this last phase of the cycle, the flow direction in pipeline 3 is reversed from the usual direction as cold water from holding tank displaces the hot water in line 3 before a significant drop in its temperature occurs. This hot water enters the water heater at the top displacing an equal volume of cold water at the

bottom which is forced to return to the supply source through pipeline 20. The system is then ready for another cycle of operation.

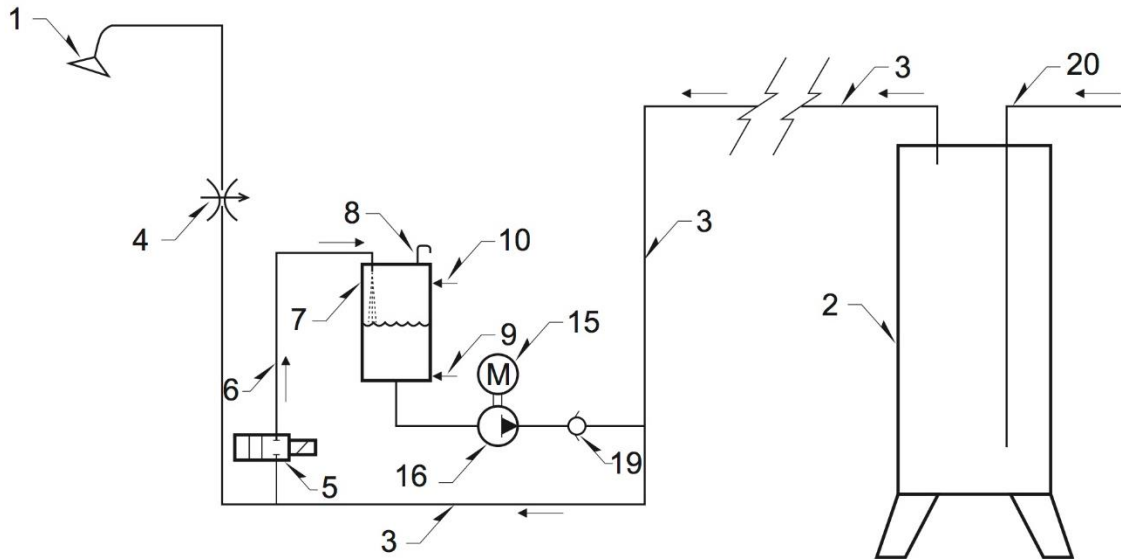


Figure 10.3a Initial phase of sequence

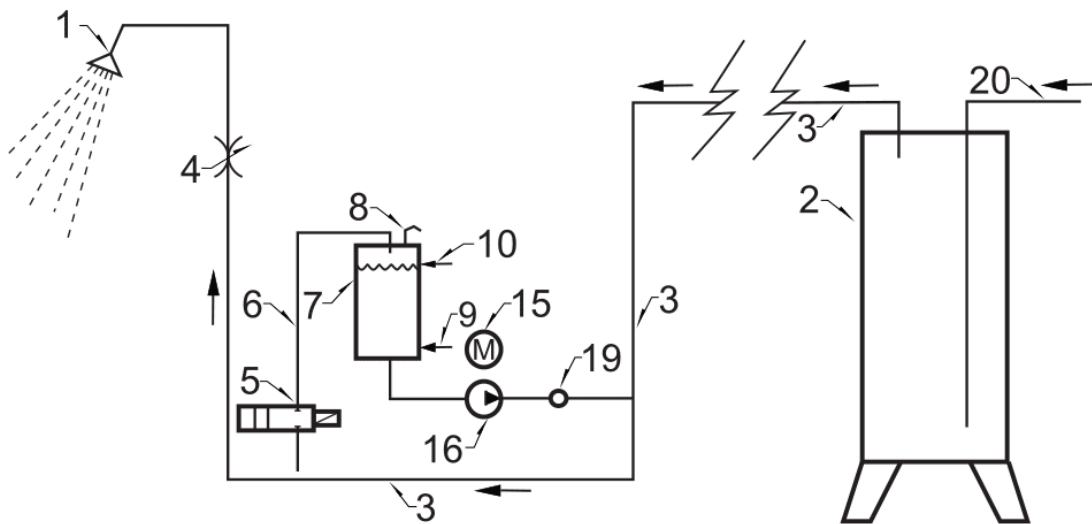


Figure 10.3b Showering phase

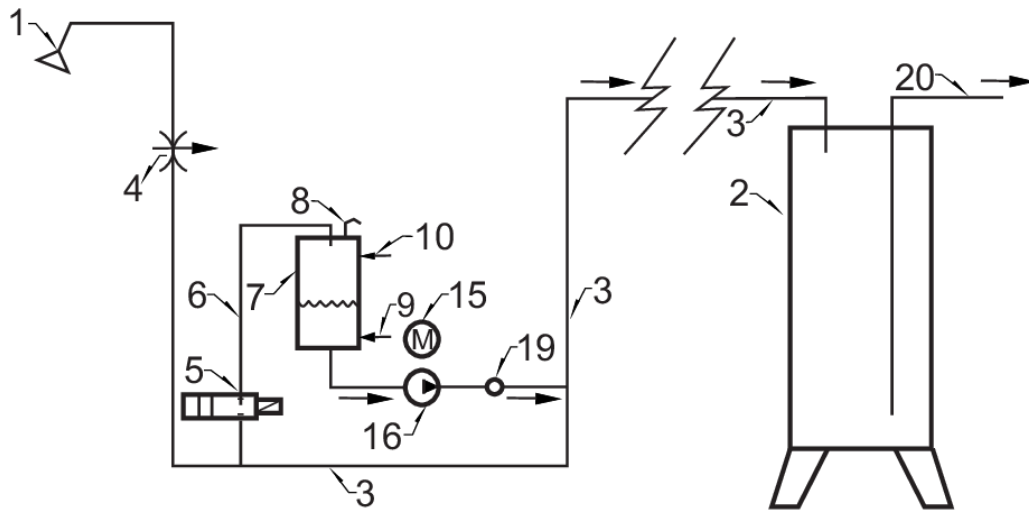


Figure 10.3c Final phase of sequence

Figures 10.3a to 10.3c show, in diagrammatic form, the sequence described above. Fig 10.3a, which corresponds to the initial phase, shows holding tank being filled with water from pipeline 3 as a result of solenoid valve being in the open position. Arrows alongside pipelines indicate the direction of flow. In Fig 10.3b, the solenoid valve has been shut after the water level in holding tank 7 reached its maximum point; also, valve 4 has been opened by the user causing the shower head to deliver hot water from the water heater. In Fig 10.3c is shown the final phase of the sequence in which valve 4 has been shut off by the user and water is being pumped out of the holding tank, establishing a reversed flow in pipeline 3 so that hot water remaining in it is forced back into the water heater, causing cold water in the bottom part to flow out into pipeline 20.

Another way to save water that is worth mentioning, is by means of hot water recirculating systems consisting basically of a loop of insulated piping in which hot water is continuously pumped. The loop passes near the points where hot water is extracted, so that it becomes available without delay. These systems are available commercially. Presumably the economizer system, being simpler, will be less costly.

11. MANGO FORKS

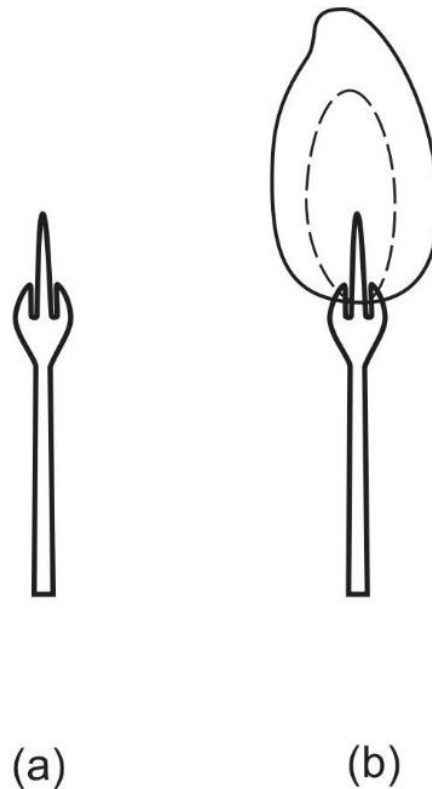


Figure 11.1 Standard mango fork. (a) Traditional mango fork. (b) Fork inserted in a mango

Mangoes are a delicious fruit. I can assert that quite confidently because I haven't yet met a person who has tried them and not liked them. But eating them may be awkward and messy. The standard method has been by the use of a special fork that looks like the one shown in Fig. 11.1a. Fig11.1b shows the fork correctly inserted in a mango. Achieving the insertion may require a few attempts and be frustrating because the fork's pointed middle prong has to be directed quite precisely and firmly to pierce the

seed right on the middle of a narrow edge. Assuming that you successfully completed this operation, you can proceed to peel it, and then enjoy eating it.

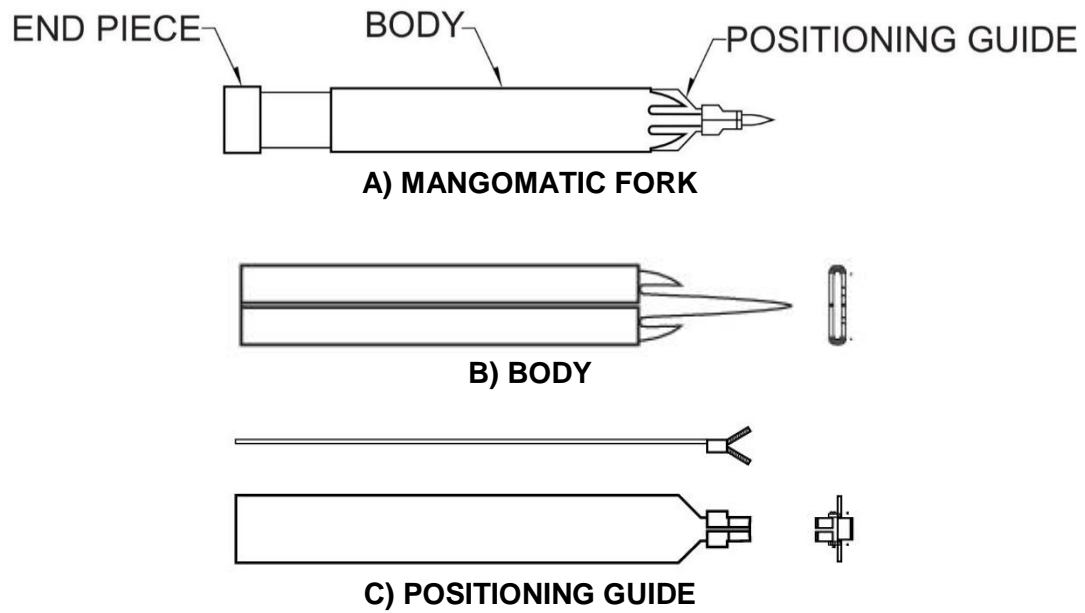


Figure 11.2 Mangomatic fork diagram



Figure 11.3 Photo of mangomatic fork

Then you are faced with the problem of removing the seed to discard it. Some persons hold the fork with the left hand and use a knife to push out the seed, others hold the seed with their teeth while they pull on the fork by hand. Either method is energy intensive and may lead to frustration, and sloppy results. Thus, I decided to take on the challenge of making a better mango fork. The result is the “Mangomatic fork”, illustrated in Fig. 11.2. It consists of a main body, a positioning guide, and an end piece. Fig. 11.3 is a photograph of one of the few dozen produced. Figures 11.4. show how to use the fork. First, as in Fig.11.4a, one holds the mango with the left hand. Grasping the fork with the right hand while pushing with the palm on the end piece, the “V” end of the positioning guide is advanced onto the seed until it seats firmly on its edge. Then, releasing the pressure on the end piece, one grasps the body of the fork and pushes firmly to pierce the seed with the central prong, advancing until a complete penetration is achieved (Fig. 11.4b). The mango is now ready for peeling with a knife and then eating it directly from the fork, or after cutting into slices. Finally, Fig. 11.4c illustrates the operation to discard the seed, which consists in pushing the end piece with the right hand while holding the body of the fork with the left hand.

An attempt to market this product was not highly successful. My wife, Carmen, was supportive of the idea, but commented “this is something for the very rich... and very strong... and also very intelligent”. My son, Edgar, who was to be in charge of sales, reported that “the demand for the forks is very good ... as long as we give them away”. Despite the previous statements, I, and a number of members of my extended family, still believe that the Mangomatic fork provides so far the best option for going about consuming this delicacy.

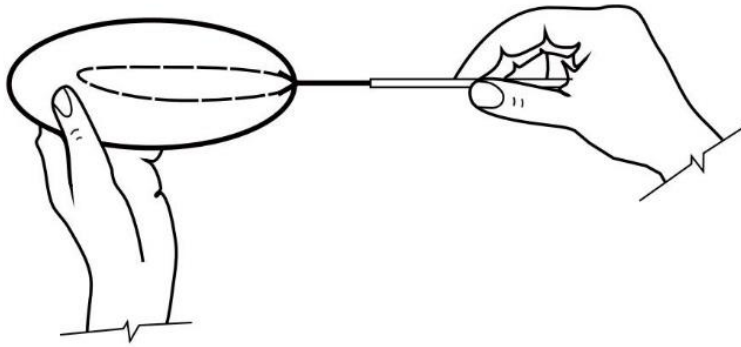


Figure 11.4a Initial operation in the use of the mangomatic fork

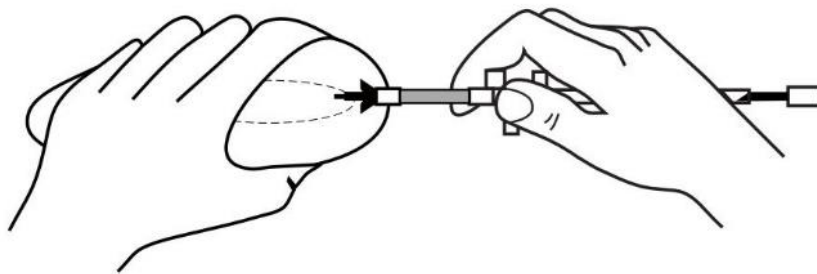


Figure 11.4b Seed piercing operation

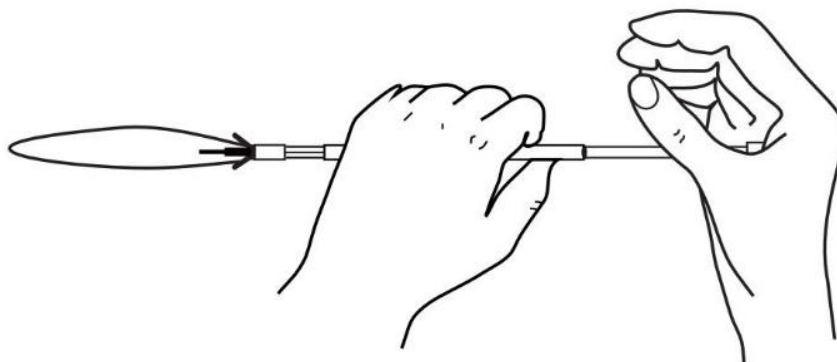


Figure 11.4c Seed ejection operation.

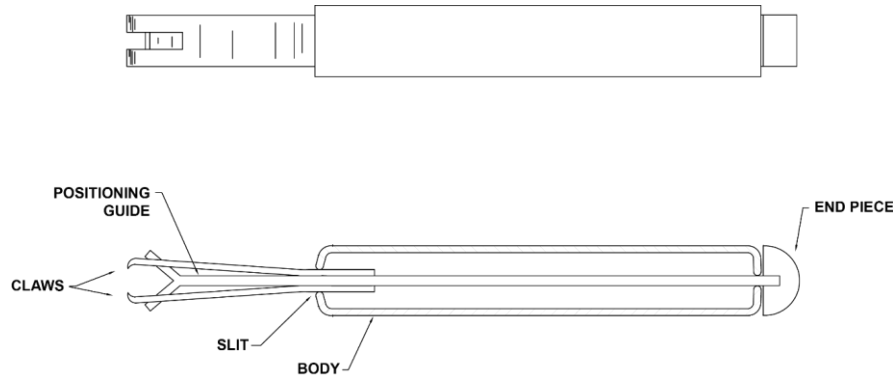


Figure 11.5 Seed gripping fork

In order to make it easier to attach the fork to the mango seed, I have explored the possibility of a gripping mechanism. A fork using this concept is illustrated in Fig. 11.5. As can be appreciated in the longitudinal midsection diagram, there is a positioning guide as in the previous design. This guide consists of a central member with a “V” end and two spring steel claws spot welded to it.

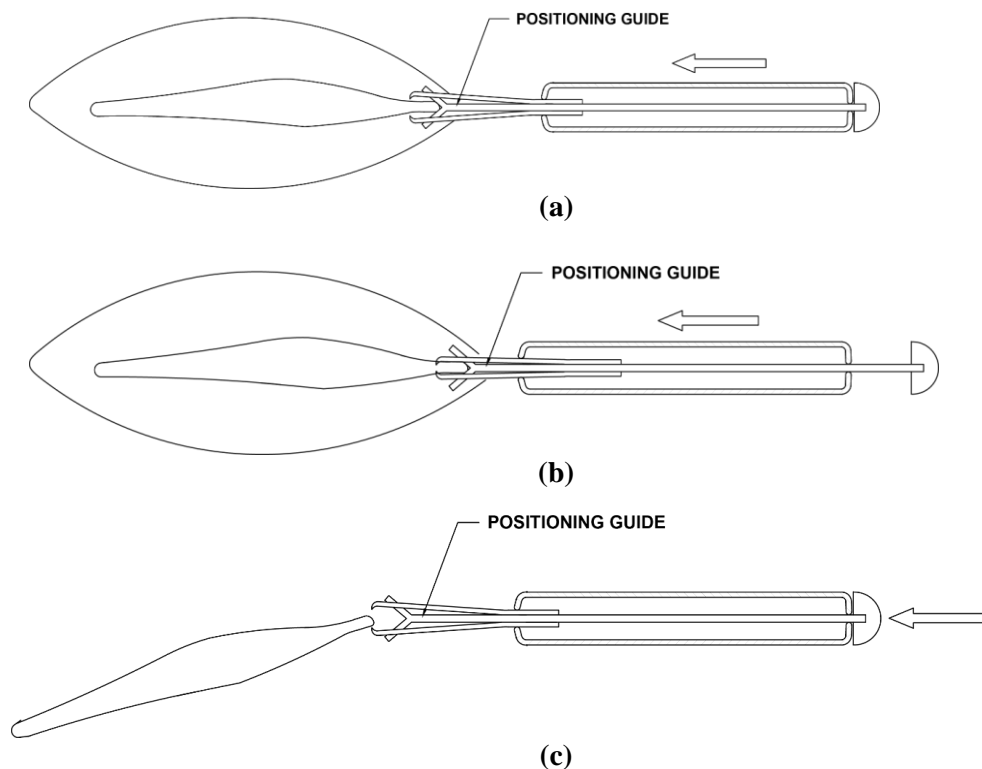


Figure 11.6 a) Initial operation in the use of the seed gripping fork, b) Seed gripping operation, c) Seed ejection operation.

As shown in Fig. 11.6, the fork is used in much the same manner as described before. Again, while the mango is held with the left hand, one pushes on the end piece with the palm of the right hand until the “V” end is brought to bear on the seed base. The claws of the positioning guide are in an open condition at this point (Fig. 11. 6a). The next step is to grasp the body of the fork and push it towards the fruit. This action causes the claws to close in on the seed as the slit from which they protrude is advanced (Fig. 11.6b). The mango may now be peeled and eaten. To discard the seed, the end piece is pushed, as before, with the right hand, while the body of the fork is held with the other hand (Fig. 11.6c). Shown in the photograph of Fig. 11.7 is a somewhat crude prototype of the gripping fork. Although the claws are not as rigid as they should, it works acceptably well. A second prototype should have heavier gage steel claws. Compared to the mangomatic concept first described, the gripping mechanism seems to reduce the force, both to insert the fork and to discard the seed.



Figure 11.7 Mango seed gripping fork

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