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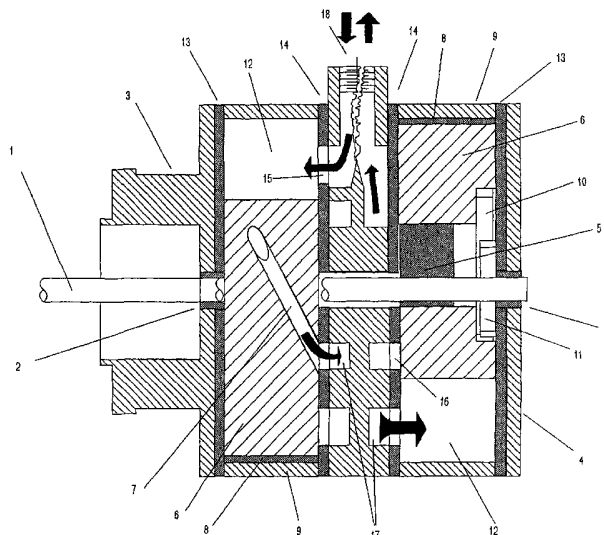
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(54) Title: A STIRLING MACHINE UTILIZING A DOUBLE ACTION PLANETARY MACHINE



(57) Abstract: A plurality of double-acting planetary rotary positive displacement machines with porting apertures (7) through the rotors (6) are employed to provide an efficient and compact apparatus to implement a Stirling thermodynamic engine cycle. The porting apertures permit peak volumetric efficiency and allow each working chamber of a planetary rotary machine to operate independently with few geometric constraints. Independently operating chambers permit double-action expansion and compression and constant volume motion through a regenerator necessary for efficient implementation of a Stirling cycle. Topping and bottoming engine cycles are also contemplated in combination to enhance the regenerative process and expand the work available. Further, self-lubricating apex seals and gas transfer plates permit zero clearance sealing for completely dry, unlubricated operation.



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A STIRLING MACHINE UTILIZING
A DOUBLE ACTION PLANETARY MACHINE

PRIORITY/CONTINUATION DATA

Priority is claimed from U.S. Provisional Appl. 60/308,413 filed July 27, 2001 of this name. For U.S. purposes, priority is claimed from and this is a continuation-in-part of Provisional Appl. 60/308,413 filed July 27, 2001, and of a U.S. Utility Application of the same name filed on July 27, 2001.

BACKGROUND

Technical field of the invention

This invention relates to Stirling cycle heat engines using planetary rotary pumps. Specifically, the invention relates to engines which deliver shaft work when heat is applied and those that act as heat pumps, including refrigerators, when shaft work is applied.

SUMMARY OF INVENTION

The present invention employs a double-acting planetary pump for use in a unidirectional flow, i.e., not reversing, Stirling machine. Specifically, the planetary pump employed uses apertures formed through the rotor that allows high volumetric efficiency and independent operation of each working chamber. Together with zero clearance self-lubricating seals, in tandem as a hot rotor and a cold rotor, these planetary pumps provide a compact means to provide the isothermal expansion and compression and the constant volume movement through a regenerator that yields an efficient Stirling cycle machine. The cycle is further enhanced by adding topping and bottoming cycles to expand the available work from the machine.

SUMMARY OF THE INVENTION

The preferred embodiment utilizes the double-acting planetary pump described in U.S. Prov. Appl. S/N 60/308,413 filed July 27, 2001, published in March, 2002 as the prime motivator of working fluid for a Stirling machine. By adding heat, the machine acts as an engine. By adding work, the machine acts as a refrigerator but the invention is not limited to these two types of devices.

The preferred embodiment uses two planetary rotary machines (generally referred to as ("rotary machines")) for use as pumps and expanders but an integral number of 2 can easily be employed. The basic two machine system is shown in Figure 6 and the corresponding engine cycle it embodies is shown in Figure 7.

Description of the Related Art

Invented by the Reverend Robert Stirling in 1816, the Stirling cycle is the only practical thermodynamic cycle capable of achieving the ideal thermodynamic efficiency of a heat engine, the Carnot efficiency. The Stirling cycle is composed of an isothermal expansion, which provides the work output, followed by a cooling by heat extraction in a regenerator, followed by an isothermal compression, which requires work input, followed by a heating using heat from the regenerator. The Stirling cycle is often referred to as an external combustion engine in that ignition of the fuel and the generation of heat is in a controlled environment outside of the piston or planetary rotor that extracts the work. To the extent efficient combustion can be achieved, internal combustion engines require perfect timing of mechanical components. Still, internal combustion engines are plagued by noxious emissions and are often encumbered by emission controlling devices. The Stirling engine typically uses an external combustion chamber to completely control the combustion process. The result is a low emissions, low noise, multi-fuel, high efficiency power plant. As such, research into creating a viable Stirling engine has been extensive. A vast array of prior art exists in using linear piston machines to provide the expansion and compression phases of the Stirling cycle. The present invention uses planetary rotary machines for the expansion and compression phases and so avoids the complicated rhombic drives and swash plates typically employed to convert the linear motion of the pistons to rotational motion and create the volume dwell necessary in some implementations of the Stirling Cycle.

Wahnschaffe, et al. (1974 3,800,526) created a Stirling engine using a single planetary rotary machine to move the working fluid back and forth through a cooler-regenerator-heater arrangement and extract work from the hot expansion phase. The unique element of their machine is that it maintains the reciprocating fluid motion seen in linear piston machines but without a separate displacer. Juge (1975 3,869,863) utilized the classic Wankel planetary rotary machine in an external combustion (non-Stirling) engine but in an effort to achieve double-action, where each lobe of a two-lobe pump acts independently, Juge's pumps suffer from very poor volumetric efficiency because the intake and exhaust ports to the compression and expansion chambers can not be set at optimum positions due to geometric constraints exactly like Maillard's (GB Patent 583,035 1947) original double-action rotary pump. Whitestone, U.S. Pat. 3,998,054, 1976 used a planetary rotary pump and envisioned its use in a Stirling engine.

Whitestone overcame some of the geometric constraints faced by Juge and Maillard by using grooves along the side of the rotor to admit flow movement, at nearly arbitrary positions, to exposed intake or exhaust ports in the side gas transfer plates. The use of grooves still constrains the position of the intake and exhaust ports. Stirling cycle engines using rotary pumps are not limited to the planetary type. Kelly, U.S. Pat. 4,044,559, 1977, employed multiple rotary vaned pumps in series to achieve a uni-directional flowing Stirling cycle; however use of vaned pumps limits pressure ratio and the non-contact vanes employed can admit significant leakage resulting in poor performance.

Another of the touchstones for this invention is the motion of the Wankel type engine. Technically such an engine is a planetary motion machine, which one inventor characterized as: "a rotating piston arrangement where a motor is guided by a gear mechanism meshing with a toothed reaction wheel in such a way that the rotor can move into or out of one or more consecutively following work chambers which accommodate rotor and are in a stationary casing." F. Jernaes, U. S. Pat. No. 3,221,664, Dec 7, 1965.

The preferred embodiment utilizes the double-acting planetary pump described in U.S. Prov. Appl. S/N 60/308,413 filed July 27, 2001, published in March, 2002 as the prime motivator of working fluid for a Stirling machine. Said provisional application is adopted by reference into this application and an understanding and appreciation of the workings and operation of the double acting planetary pump in Serial Number 60/308,413, is important to an understanding of the contemplated machines using Stirling cycles.

A planetary motion machine offers the benefit of fewer moving parts than a typical machine using cyclical motion, valves, or conversion from rotary to linear motion or vice versa to exert or receive pressure. A planetary motion machine may be a pump (that is taking in a fluid stream and compressing it to be exhausted at higher pressure), or a turbine (utilizing pressure to drive a rotor circularly to a lower pressure exhaust, and generating rotary mechanical power in a rotating shaft). A planetary motion machine has less eccentric motion than a typical straight piston machine. It has fewer moving parts in part because the machine is inherently a rotary machine and need not convert linear motion to rotary motion. Its disadvantages are that traditionally the classic planetary motion machine has only one compression per rotor cycle, and at high speed, there can be problems maintaining a seal of the compression chambers.

The present invention employs the double-acting planetary pump for use in a unidirectional flow, i.e., not reversing, Stirling machine. Specifically, the planetary pump or planetary rotary machine employed uses apertures formed through the rotor that allows high volumetric efficiency and independent operation of each working chamber. Together with zero clearance self-lubricating seals, in tandem as a hot rotor and a cold rotor, these planetary pumps provide a compact means to provide the isothermal expansion and compression and the constant volume movement through a regenerator that yields an efficient Stirling cycle machine. The cycle is further enhanced by adding topping and bottoming cycles to expand the available work from the machine.

The preferred embodiment uses two planetary pumps for use as pumps and expanders but an integral number of 2 can easily be employed. The basic two pump system is shown in Figure 6 and the corresponding engine cycle it embodies is shown in Figure 7.

A classic planetary motion machine is illustrated in Figure 2. The basic shape of the chamber, looking at the chamber from the "top" parallel to the axis of the rotating parts, is that of a symmetric peanut, though the "waist" of the peanut is barely narrowed. The peanut shape is called a peritrochoid in mathematics. The rotor looks like an equilateral triangle with symmetric bulged sides. In essence, the rotor, to use a layperson's description, rolls around in the inside of the peanut with each apex in contact with the peanut. If an engine is placed on the drive shaft of the planetary machine, it will cause the rotor to spin, and the action of an alternating increase and decrease in volumes of the working chambers in combination with alternate occlusion and exposure to intake and exhaust ports will cause fluid to be pumped. Alternatively, if pressurized fluid is allowed into a chamber to force the rotor to turn, then the drive shaft will be forced to rotate and will produce mechanical power at the shaft. Similarly, if pressurized fluid is allowed into a chamber to force the rotor to turn, by changing the position of the intake and exhaust ports for a different chamber, that different chamber can be used to compress fluid, effectively permitting the rotary machine to be a compressor and turbine simultaneously. The fluid can be liquid or gas or a combination.

In order to make a planetary machine attractive, scientists have sought to have more than one chamber simultaneously performing compression/exhaustion while another chamber performs induction/expansion during each rotation of the rotor, and at the same time minimize the number of moving parts, and minimize the speed of what parts are moving. The machine in

the present invention is a double pumping or double action planetary machine, meaning that for each planetary cycle, the machine can have one chamber perform a function of compression/exhaust or intake/expansion, while another chamber performs another function of either compression/exhaust or intake/expansion, and therefore the cycle of at least one chamber consists of a) two motions of intake/compression/exhaust, b) two motions of intake/expansion/exhaust or c) one action of each of intake/compression/exhaust and intake/expansion/exhaust.

In 1976, Whitestone, U.S. Pat. 3,998,054, Dec. 21, 1976, was issued a patent for a "Rotary Mechanism with Improved Volume Displacement Characteristics." While claiming improved displacement characteristics, and using ports in side plates, his rotor did not use the device of a duct through the rotor face and thence to a side port, nor did his pump contemplate a two-lobe peritrochoidal cavity. The effect of not using this duct or aperture through the rotor face and the lack of two-lobe peritrochoidal cavity is that for any given planetary cycle, the pump fails to achieve the swept volume and compression ratio (maximum volume to minimum volume) that the present invention achieves. This can be seen by reviewing Figures 1 through 8 in Whitestone '054. The advantage of the present invention is that a working chamber is nearly totally evacuated from a maximum volume. In Whitestone, particularly as the geometry of his proposed rotor veered away from the three lobed rotor in a square cavity in Figure 2, Whitestone's invention faces one of two efficiency difficulties. First, there is a large permanently retained minimum volume 25f as in Figure 9E, which minimizes the compression ratio of the maximum to minimum volume. Alternatively, second, there is a relatively small maximum volume with a somewhat smaller but substantial minimum volume 12af as in subfigures CF and DF of Figure 13, but no port available for exhaust in Whitestone's '054 invention. Whitestone's porting, shown in Whitestone '054 Figure 9a, which is the identical rotor position to Whitestone '054 subfigure CF of Figure 13, particularly for a solid rotor which eliminates volume 25f of Figure 9E, shows the traditional geometric difficulty faced by Maillard, United Kingdom (British) Pat. No. 583,035 issued 2 Jan 1947, and prior art rotary pumps of either a) maximizing intake volume for the beginning of compression, but also enlarging the volume being compressed at time of exhaust, as in Whitestone '054, or b) lessening intake volume for the beginning of volume, and lessening the volume being compressed at time of exhaust. An example of the latter is Maillard UK Pat. 583,035 and Juge, U.S. Pat. 3,869,863, Mar. 11, 1975.

A rotary pump was proposed in an unpublished project proposal at the University of Calgary, Alberta, Canada referred to as a Zwaiuer-Wankel configuration of rotary Stirling engine, for which a figure is shown at p. 79 of G. Walker, *Stirling Engines*, Clarendon Press, Oxford 1980, Library of Congress Call No. TJ765.W35, and is described at p. 115 of that book, Walker, *Stirling Engines*. In G. Walker, et al, *The Stirling Alternative: Power Systems, Refrigerants and Heat Pumps*, p.78, (Gordon and Breach Science Publishers 1994), the same author remarks that the Zwaiuer-Stirling rotary engine is an “arrangement [that] could provide a compact high specific output machine but although proposed over 20 years ago it has not been reduced to practice so far as is known.” From the drawing, Zwaiuer appeared to use a solid rotor form with porting after the fashion of Maillard or Whitestone ‘054, and in any event did not contemplate the use of a duct through the rotor and corresponding porting arrangement.

There are several other planetary machines which do not achieve double action where ducts through the rotor are contemplated, and/or where the maximum to minimum volume (the compression ratio) is not particularly useful for efficient fluid flow, and/or there are sealing problems. However, no art utilizes a system set out in this invention involving ducting, porting and the relative position of the rotor, duct and ports for the basic pumping or turbine action of the planetary machine to achieve double action with a superior volumetric efficiency without seal loss, double action in a three vaned-two lobed pump meaning two compressions and two expansions of fluid per planetary cycle. Maillard, , United Kingdom (British) Pat. No. 583,035 issued 2 Jan 1947, recognized the geometric constraints of his design, but absent a fluid passage through the rotor and proper design of ports and proper location of such a fluid passage, he could not overcome the geometric constraints. The present invention successfully hurdles the geometric constraints and achieves double action which none of the prior art has achieved, and with a minimization of moving parts. See, for example, ducts through the rotor, but no double action: Child, U.S. Pat. 4,986,739, Jan. 22, 1991, White, Jr., U.S. Pat. 4,872,819, Oct. 10, 1989, Nakayama, U.S. Pat. 4,345,886; ducts for lubrication or cooling: Miles, U.S. Pat. 4,097,205, Jun. 27, 1978, Nakayama, 4,345,886, Aug. 24, 1982 (using retractable vanes in the housing).

Apex seals may be kept in close contact with a roughly orthogonal surface using centrifugal force as seen in Kaatz, 3,191,852, June 29, 1965, and Bishop, U.S. Pat. 5,181,844, Jan. 26, 1993, U.S. Pat. 4,820,140, or using a technique of feeding pressured air in behind the vanes as seen in Smart et al, U.S. Pat. 4,804,313, Feb. 14, 1989. Springs can also be used.

Optimum self-lubricating composites can be seen in any number of patents using polytetrafluoroethylene (PTFE), or better yet using carbon fiber reinforced polyetheretherketone (PEEK), particularly continuous carbon fiber reinforced PEEK. Other materials usable as self-lubricating materials are set out in Davies et al, 5,750,620, May 12, 1998.

The term continuous carbon fiber reinforced PEEK is focused on polyetheretherketone, and a close material cousin PEKK, polyetherketoneketone, but the term includes a compound selected from the group of polyaromatic compounds having amorphous crystal structure corresponding in intermolecular distance to the intermolecular distance of continuous carbon graphite crystal structure such that upon melting of said polyaromatic compound having amorphous crystal structure in the presence of continuous fiber carbon graphite, said combination results in carbon crystal lattice reinforcement of said polyaromatic compound.

OBJECTS OF THIS INVENTION

This invention has three major features yielding improved performance.

First, the creation of a duct through the rotor to the curved face-- that is, diagonally from the side of the rotor through the rotor to the curved face of the rotor--yields, in combination with carefully arranged ports, double pumping action and enhanced inlet or exhaust porting. By proper arrangement of the location of the inlet parts, duct and exhaust parts, no new moving parts are introduced beyond the classic rotary machine design, yet a double action pump is created with substantially improved compression ratio. If pressured air is delivered to the invention with a differential lower pressure on the "opposite" side of the pump, a double pumping turbine yielding power to a drive shaft results with a favorable compression ratio.

Second, one of the chronic problems of Stirling machines relates to edge effects. The invention proposes to use volumetric matching of chambers to optimize efficiency in the constant temperature and constant volume phases of the Stirling cycle. Work is extracted and heat inserted (while preserving constant temperature by expansion) only in the constant temperature phases.

Third, the planetary machine's performance is enhanced by the use of modern self-lubricating polymeric composites, particularly continuous carbon, mineral or glass fiber reinforced polymeric composites, to achieve better sealing.

The invention achieves a variety of objectives by this design. The invention can be a pump when an engine or other rotating device is connected to the machine and causes the rotor to rotate, forcing fluid through the parts of the machine. The invention may be a turbine when pressurized fluid drives the machine, or an engine when combustible mixture is ignited in the working chambers. The invention will be described in terms of a pump, understanding the claims are not limited to a pump and that if, a pressure differential between the intake and exhaust side of the pump exists, the machine will function as a turbine.

DESCRIPTION OF FIGURES and INVENTION

A generic rotor with the features of the present invention utilized as a pump is presented in FIG. 1. The letter "A" denominates the depiction of the aperture through the rotor with its entrance on the rotor face "B," and the aperture's exit on the rotor side opposite to the point indicated by the letter "D" in this embodiment of the invention. The letter "C" indicates the journal bearing hole into which an eccentric drive shaft (normally made eccentric by a cam) is placed which provides power to the rotor. The letter "E" is the annular timing gear which meshes with a stationary sun gear attached to a side plate and guarantees the planetary motion within the peritrochoid.

FIG. 2 illustrates the locations of the typical intake ports and exhaust ports within the peritrochoid shape. For the ducting as shown in the figures, where exhaust is through the duct through the rotor and then to the exhaust ports E, and intake is directly through intake ports I into the working chambers, the intake ports must at least intermittently be within the outer bounds of the trace of the rotor face, and at least intermittently outside the interior trace of the rotor face. The exhaust ports are always within the inner bounds of the trace of the rotor face, and the exhaust duct through the rotor face is intermittently exposed to the exhaust ports in this embodiment.

FIG. 3 displays four positions of the rotor with the intake and exhaust ports and the manifold of apertures (three) overlaid. At a driving shaft position of 500 degrees Before- Top-Dead-Center (BTDC), working volume A, which is defined by the housing and rotor face A, is beginning the intake stroke as the intake port I is just starting to be uncovered. The exhaust duct adjacent to the rotor face corresponding to working volume A is not juxtaposed to the exhaust port so as working volume A expands, fluid will be admitted at the ambient pressure at the intake port.

Meanwhile, working volume B is in the midst of a compression and exhaust stroke as a clear path exists from volume B to the exhaust port E via the aperture and duct through the rotor.

Volumes A and B are sealed from each other by a zero clearance apex seal and the rotor being placed sealingly adjacent to the side plate of the pump. Also, volume C is completing its intake stroke as the working volume is near maximum and intake port I is beginning to be occluded as the rotor side slides over it.

At 370 degree BTDC, working volume A is midway through its intake stroke. The working volume B is completing its compression and exhaust stroke. Working volume C is just beginning its compression stroke with the exhaust port just beginning to be exposed to the exhaust duct through the rotor.

At 240 degrees BTDC, working volume A is near maximum volume and the intake port is now blocked by the rotor side. Working volume B is still expanding and the rotor side has just begun to close off the intake port adjacent to rotor face adjacent to working volume B while working volume C is nearing its minimum volume point. The invention allows the designer to guarantee that the exhaust port is not open while the intake port is open so timing can be completely optimized for maximum performance.

After nearly one drive shaft revolution and one third of a rotor revolution, at 110 degrees BTDC, working volume A is midway through its compression and exhaust stroke. The intake port is almost occluded by the rotor side as to working volume B while the exhaust port is not yet exposed to working chamber B. The rotor side adjacent to working volume C is just uncovering the intake port and the expanding volume admits air from the intake volute.

FIG. 5 shows a cross section along the line of the driving shaft.

Description of the mechanical parts

A description of the interrelationship of the parts is as follows referring to the numbers in FIGURE 5: The driving shaft (1) transmits the mechanical power from an engine or other power source to the pump or rotary machine. The shaft is supported by at least two bearings (2) composed of any self-lubricating material such as PEEK or PTFE or scintered bronze impregnated with lubricant. The bearings are set in the side plate (4). Fixed to the driving shaft are two cams (5) which ride inside continuous carbon reinforced PEEK bearings fit into each rotor and which drive the rotor rotation. Each rotor (6) has three-lobes with the claimed invention of an aperture (7) connecting each rotor face with the rotor side. Each apex of each rotor contains

the claimed invention of an apex seal composed of continuous carbon fiber reinforced PEEK (8). The apex seals are in sliding contact with a two-lobe peritrochoid shroud (9) and forced against the shroud by means of small compression springs. The planetary motion of the rotor is maintained by an annular gear (10) fixed to each rotor by means of screws or pins and a stationary sun gear (11) fixed to the side plate by means of screws. The working volumes of the pump/rotary machine (12) are then formed by the rotor face and the side plates composed of continuous carbon fiber reinforced PEEK (13) and (14). For the second embodiment of the invention, the inner-most side plates (14) contain the intake and exhaust ports (15) and (16), respectively. The intake and exhaust ports expose portions of the claimed invention of intake and exhaust volutes (17) which deliver and collect air from the working volumes to and from the separate intake and exhaust external rotary machine connections (18). The entire unit is held together by means of bolts symmetrically placed about the driving shaft and parallel to it. The preferred embodiment uses two planetary rotary machines for use as pumps and expanders but an integral number of 2 can easily be employed. The basic two rotary machine system is shown in Figure 6 and the corresponding engine cycle it embodies is shown in Figure 7.

DESCRIPTION OF THE PREFERRED EMBODIMENT

The general characteristic of the preferred embodiment of the rotor is somewhat like a two rotor NSU-Wankel internal combustion engine found in some automobiles and aircraft.

The center of each cam is displaced eccentrically from the center of the driving shaft. Each cam rotates within a hole machined into the center of each rotor and drives, in the preferred embodiment, continuous carbon fiber reinforced PEEK bearings fit into each rotor, which action in turn causes rotor rotation as later described.

While reference is made to continuous carbon fiber reinforced PEEK in the preferred embodiment, PEKK (polyetherketoneketone) has similar properties. More broadly, the invention preferably utilizes for either the bearings and/or or the rotor apex tips reinforced polymeric compositions referenced in Davies, U.S. Pat. No. 5,750,620, which will be referred to collectively as carbon fiber reinforced polymeric compositions. Materials such as scintered bronze impregnated with PTFE along with carbon fiber reinforced polymeric compositions, or even hydrocarbons in certain applications, will be the broadest category of suitable materials and will collectively be called self-lubricating materials. All of these may be used, but the optimum selection for use is a continuous carbon fiber reinforced polyaromatic compound such as

continuous carbon fiber reinforced PEEK. The self-lubricating composites are reviewed further later in the description in this invention whether reinforced by glass, mineral or carbon fiber.

The preferred embodiment of each rotor has three apices, and therefore three faces corresponding to the number of apices. Each set of two adjacent apices and the intervening face can be referred to generically as a lobe and will have a working chamber of varying volume opposite that lobe which will be moving rotationally and varying volume simultaneously. The rotor on a smaller scale application is composed of or coated with hardened aluminum, e.g., 6061-T6 and machined to the desired contour of three triangularly placed arcs. Each of the three faces of said rotor is penetrated by one of the important innovations of the claimed invention: namely a single duct machined or molded through the rotor face which pierces the side of the rotor which is diagonal to the face or vane between the apices of the rotor and diagonal to the side of the rotor adjacent to the side plate. The duct then forms an aperture through which gas flows undisturbed when both ends are not obstructed. The rotor also contains an annular timing gear affixed to either side. This annular gear meshes with a stationary sun or spur gear fixed to the non-rotating forward and rear side plates of the rotary machine and constrains the rotor motion to the desired planetary cycle, much like the Wankel design (The gears could be replaced by a guide similar to Greys invention U.S. Patent No. 3,884,600, May 20, 1975).

The peritrochoid shrouds are made of hardened aluminum like 6061-T6, preferably with hard-coat anodizing, and with the next-described side plates form the cavity within which each rotor rotates. The peritrochoid shroud and rotor lie between two side plates, either of which may be ported. The side plates are disposed in conjunction with the shroud such that the side plates are in sliding contact with the rotor. The side plates on which are disposed the stationary sun gears are also made of aluminum and mate with the peritrochoidal shrouds. The side plates could be made of or coated with a self-lubricating material such as PEEK, particularly where there is relatively high speed relative motion between the side plates and the rotor. The side housing could be of PEEK, but this is a less desirable equivalent than the vanes being made of PEEK which are much smaller, and the side housing not being made of PEEK. The sun gears, peritrochoidal shrouds, annular gears and rotors are specifically oriented such the planetary motion of the rotor apices is exactly contained by the shroud. To maintain low friction, the side plates, including the port plate(s) can be made from continuous carbon fiber reinforced PEEK

similar to the apex seal material. In this way, all sliding surface contacts use low friction self-lubricating material.

Discussion of embodiment particular to use of rotary machine for Stirling engine with tracking slot

The first Stirling heat engine embodiment uses essentially continuous communication from chambers that are contracting in the rotary machines through exhaust ports from exhaust ducts through a tracking slot in the side plate for each chambers' exhaust duct(s) on each rotary machine. The exhaust and intake ducts are preferably reasonably short to reduce dead space, as are the plena and ducts that are the means for gas to flow. Similarly, the same embodiment of the heat engine uses essentially continuous communication from chambers that are expanding through intake ports from intake ducts through a tracking slot in the side plate for each chambers' intake duct(s) on each rotary machine.

The intake ducts to the working chamber and the exhaust ducts proceed to opposite side plates of the rotary machine. The duct corresponding to each face trace a different peritrochoidal track on the outside of the rotary machine, so for a two lobe housing with a three-lobe rotor (classic Wankel engine) there are three tracks on one side plate of each rotary machine and three on the other side plate. The tracks are located inside the inner periphery traced by the vane tip, and outside the annular track, preferably. Each track is sealed from the other and will be designated a, b, and c. The slot enables communication to a plenum from which gas in a loop flows to a regenerator unique to each A1+A2 chamber, and each B1+B2 chamber and each C1+C2 chamber in a second rotary machine, as described in more detail momentarily. The concept is to have each rotary machine have each of three chambers alternately expanding and contracting, and continuous communication of the chamber with the exterior of the rotary machine, though not with the chamber rotary machine open to the exterior for exhaust purposes at the same time as the chamber is open to the exterior for intake purposes, and vice-versa, and have a corresponding chamber in rotary machine 2 contracting as a chamber expands in rotary machine 1.

Viewing the peritrochoidal shape of the rotary machine from the side, and arbitrarily for explanation sake that the tip of the planetary rotor is at a point we shall call top dead center, meaning centered on the waist and the peritrochoidal cavity, the chamber opposite the designated top dead center apex of the rotor of the first rotary machine, which is almost totally evacuated,

will be designated as A1, and the vane face of the rotor facing that chamber as vane face a1. The chamber opposite the apex of the second rotary machine, which will be out of phase as later described with the first rotary machine, which at substantially larger and virtually maximum volume, will be designated as A2, and the vane face facing that chamber will be designated as a2. In a three lobe rotary machine, the typical cam and rotor combination consume 1080 degrees for a full cycle. The first rotary machine should be out of phase with the second rotary machine by 360 degrees if the rotary machine operation is viewed as a 1080 degree cycle. If the first rotary machine runs clockwise, the next face to an expanding chamber will be designated B1 and the last contracting chamber C1. If the second rotary machine is also running clockwise on the same shaft as the first rotary machine B2 will be the contracting chamber and C2 the expanding chamber. The sum of chambers A1+A2 is thus always reasonably constant, the sum of chambers B1+B2 is thus always reasonably constant, and the sum of chambers C1+C2 is thus always reasonably constant. Operation of these paired chambers through a regenerator or heat economizer establishes the constant volume heating and cooling processes integral to the Stirling cycle.

Further, the two rotary machines are separate with one rotary machine dedicated to isothermal expansion and the other dedicated to isothermal expansion. Designating rotary machine 1 as the "hot rotary machine", the fluid flow in the system is set up generally as follows: first, assume the fluid is at its highest pressure and temperature and smallest volume in the Stirling process. Fluid proceeds into the working chamber of the hot rotary machine which chamber is expanding and the apertures are open and accepting intake. During the expansion, heat is added through the housing or in the vicinity of the intake port sufficient to maintain isothermal expansion. Upon reaching maximum volume, the chamber intake is occluded by the rotor motion and the fluid starts to exhaust as the working chamber begins to contract and the exhaust port opens and communicates with the chamber via the aperture. At the same time, rotary machine 2, designated as the "cold rotary machine" has a paired chamber that is expanding in phase with the contracting hot rotary machine. The fluid flows between these paired chambers through a heat exchanger, economizer or machine which either draws heat or work, respectively, or both from the working fluid and transfers it to the "cold side" of such heat exchanger or machine which will input heat or work, respectively, or both, into the working fluid. The working fluid will be cooled after passing through such a device. Once the working

fluid is cooled and moved into the “cold rotary machine”, an isothermal compression begins as the working volume contracts and heat is rejected through the housing or in the vicinity of the exhaust port. Excess work from the expansion is employed to drive the compression through a mechanically connected or an electro-mechanically connected driving shaft.

The exhaust port on the cold rotary machine opens after sufficient compression and, while a matching volume begins to expand in the hot rotary machine, the compressed flow is pushed through the heat exchanger, economizer or machine which either draws heat or work at constant volume so that the heat addition raises the fluid’s temperature and pressure.

The delicate timing required to accomplish match the volumes to the cycle is facilitated by the apertures through the rotor face and the periodic opening and closing of the intake and exhaust ports and independence of each working volume chamber. The peritroichoidally “concentric” annular rings through which the inlet and exhaust ducts for each chamber work enable there to actually be three cycles occurring. Independence of each working volume is also achieved in the preferred embodiment by using zero-clearance continuous carbon fiber reinforced polyetheretherketone (PEEK) apex seals and side gas transfer plates. Cams or gears can be used to tune the chamber volumes to match and to overcome idiosyncrasies caused by the regenerators or minor differences in gas loop volumes.

If a heat exchanger only is used, it is essentially a regenerator. If a machine only is used it is a device that transfers work rather than heat and therefore is a bottoming cycle in a combined Stirling-Brayton cycle. In combination, excess heat due to inefficient regeneration can be expanded through a turbine to drive a compressor that pumps to higher pressure the heated flow leaving the regenerator and is therefore a topping cycle. The benefit can be a small increase in work and efficiency when properly implemented.

Connection between the first rotary machine and the second rotary machine may be by mechanical coupling of a shaft, or gears or cams may be used, or a mechanical or electromechanical coupling may be used.

The three gas loops proceed in a heat engine from a heated rotary machine in which expansion occurs converting heat to rotating work, thence to the hot side of a regenerator at

which energy, in the preferred embodiment in the form of heat is incrementally lost to the cool side of the regenerator, then to a second rotary machine whose volume is opening to accept the “transmitted” gas, after which admission contraction driven by the first rotary machine occurs, heat is gained in the cool side of the regenerator, and then the cycle recommences as gas admitted to the first rotary machine is expanded.

Discussion of embodiment particular to use of rotary machine for Stirling engine without tracking slot

The claims are referenced as showing not only the cooperation of the parts but assisting one of ordinary skill in the art in understanding and being able to make the invention herein.

Opposite and parallel to the side plates of the rotary machine are the port plates which contain two intake ports symmetrically placed about the central axis coincident with the driving shaft and the shroud longitudinal center line and two exhaust ports also symmetrically placed about the central axis. The intake and exhaust ports are of sufficient cross sectional area that the air flow will not choke (reach Mach 1) during normal operation which would reduce performance. The position of the ports is determined to maximize the flow rate performance but generally, in a rotary machine where the fluid will be exhausted from a working chamber and out through a duct in the rotor face to an exhaust port in the side plate, the intake port on a side plate is positioned and configured in such a way that:

- a) the intake port is covered by the rotor side at all times except between the “intake port open” and the “intake port closed” rotor position at which time there exists an unobstructed path for air to flow from the intake volute to the working volume formed by the shroud, the side plates, and the rotor face exposed to the intake port. The ports in this configuration are located inside the outer bound of the rotor, but outside the innermost trace of the face of rotor during the rotation cycle.
- b) the “intake port open” rotor position is that rotor position where the working volume is near its minimum and the exhaust port is closed or occluded.
- c) the “intake port closed” rotor position is that rotor position where the working volume is near its maximum and the exhaust port is closed or occluded.

The exhaust port is positioned and configured in such a way that:

- a) the exhaust port is covered by the rotor side at all times.
- b) between the “exhaust port open” and “exhaust port closed” rotor position, the exhaust port is aligned with the rotor side aperture formed by the claimed invention of a duct piercing the rotor face previously exposed to the intake port. The alignment is such that an unobstructed path is formed for air to flow from the working volume to the exhaust port.
- c) the “exhaust port open” rotor position is a position after the working volume is near its maximum and the intake port is closed, and some contraction of the working volume has occurred so that the desired pressure is created.
- d) the “exhaust port closed” rotor position is that where the working volume is near its minimum.

The actual operation of the rotary machine as a fluid movement device begins with the expansion of a chamber rotating the rotor and a particular rotor face towards the “intake port open” position. A review of a pump characterization simplifies the thinking about how the slots in the rotary machine are placed. The intake port is uncovered by the rotor side exposing the minimum working volume and a trailing rotor face to the intake volute. The rotor rotation produces an expanding volume which in turn produces a lower-than-inlet pressure which pulls air into the working volume through the intake volute. Air ceases to flow into working volume as the intake port is occluded by the rotor side prior to the working chamber volume contraction due to rotor rotation. As rotor rotation continues, the air is compressed in a now fully enclosed working chamber until the “exhaust port open” position when a clear path forms from the working volume to the exhaust port via the duct from the rotor face to the rotor side. Air continues to flow out of the contracting volume through the duct and into the exhaust volute until the “exhaust port closed” position is reached. This sequence also occurs in the second lobe of the peritrochoid shroud, albeit out of phase. Since the apex seals and side plates produce nearly zero clearance or actually zero clearance, there is little flow communication between the two lobes. Thus, with the claimed invention of an aperture or duct through the rotor face, the intake and exhaust ports can be utilized or be occluded based on maximizing volumetric efficiency rather than observing the geometric constraints found in the Maillard, United Kingdom Pat No. 583,035, 2 Jan 1947 and Schwab, U.S. Pat. 4,551,073, Nov. 5, 1985 designs.

The intake ports, instead of being in the side plates, could be in the shroud, but the volumetric efficiency of the machine is significantly less.

For a turbine used in a heat pump, there is less need to wait to create access to the exhaust port until after a period of contraction of a particular working chamber. The turbine can accept fluid to an expanding chamber immediately after minimal volume is achieved, cease accepting fluid to that chamber at maximum volume or in desired quantity, and have the chamber commence access to an exhaust port after an intake port is occluded, and after maximum volume has been achieved. Exhaustion of a chamber can continue until just before an apical tip is at a position where minimal volume is achieved.

The system can be a two rotor system which is statically balanced, and/or counterweights or cams may be added for dynamic balance. These counterweights can be fixed to the driving shaft beyond the forward and rear side plates. Multiple rotor combinations can be used to avoid large counterweights.

If the invention is to have each lobe have a separate exhaust stream, then each lobe must have its own separate exhaust duct and port; the above description of porting locations applies for each chamber, but to separate the exhaust streams, there must be more planning of the relative location of the exhaust ducts. Each duct must intersect the rotor side on a separate peritrochoidal track so that a particular duct only vents to a particular track.

There is no requirement in the invention that the duct through the rotor face be used for exhaust. The construct of the planetary machine may be inverted. The intake ports may be designed to be covered by the rotor side at all times, and located to be alternately exposed to an intake duct from the rotor side to the rotor face to a working chamber, with the exhaust ports alternately exposed to the working chamber when the intake ports are not exposed to the duct to the working chamber.

As a turbine, the invention has superior wear properties as a result of the continuous carbon fiber reinforced PEEK used.

Focusing on the Stirling embodiment, utilizing the double-acting planetary pump described in U.S. Prov. Appl. S/N 09/308,413, and summarized above, the second embodiment proposes to use that planetary pump as discussed herein for a Stirling machine. If power is applied to the driving shaft of the planetary pump as used in this invention, working fluid is forced to move, and the system can operate as a refrigerator or cooling machine, more generically called a heat pump.

The planetary rotors employed in this embodiment use apertures through the rotor face to the side of the rotor where the apertures periodically open and close when aligned with ports in the gas transfer side plates and expose the working volume of the pump to the various positions in the Stirling cycle.

It is suggested that there be at least two planetary rotary machines. Six rotary machines comprehends all relative motions of the eccentricity of the rotor and gearing mechanisms. The rotary machines may use ducts through the rotor face to the side plates for both inlet and outlet from the working chamber of the planetary rotary machine, with the ducts as short as possible, meaning the duct should be located preferably close to the side plates of the rotary machine to minimize dead space in the ducts.

The concept in the invention is to have each rotary machine have each of three chambers alternately expanding and contracting, and continuous communication of the chamber with the exterior of the rotary machine, though not with the rotary machine open to the exterior for exhaust purposes at the same time as it is open to the exterior for intake purposes, and vice-versa, and have a corresponding chamber in rotary machine 2 contracting as a chamber expands in rotary machine 1.

Designating rotary machine 1 as the "hot rotary machine," the fluid flow in the system is set up generally as follows: first, fluid passes through a heat stage, which can be an external combustion engine. Fluid proceeds into a working chamber A1 of the hot rotary machine which chamber is expanding at this point and accepting intake. Upon reaching maximum volume, the chamber A1 intake is occluded and the fluid is exhausted. Next, the fluid flows from such exhaust to the "hot side" of a heat exchanger or machine which either draws heat or work, respectively, or both from the working fluid and transfers it to the "cold side" of such heat exchanger or machine which will input heat or work, respectively, or both, into the working fluid. The working fluid will be cooled in such heat exchanger or machine. If a heat exchanger only is used, it is a regenerator; if a machine only is used it is a turbine operating by expansion on the "hot side" thereby cooling and decompressing the working fluid, and a turbine on the same shaft operating by compression on the "cool side" thereby pressuring and warming the working fluid. Such a machine will be ordinarily be a Brayton machine. For the moment, they will be collectively referred to as a regenerator.

1 Upon leaving the cool side of the regenerator or Brayton machine, the working fluid
2 enters a cooling stage of the invention, which exchanges heat to the outside atmosphere which
3 is cooler than the working fluid temperature in the heat stage. Upon exit from the cooling
4 stage, chamber A2 of the second rotary machine which is at the instant described
5 (corresponding to the minimum volume of chamber A1), at a substantially larger volume.
6 The working fluid is exhausted from the second rotary machine to the “cool side” of the heat
7 exchanger or Brayton machine, and thereafter back to the heat stage. This heat stage, rotary
8 machine, “hot side” of the regenerator, cool stage, rotary machine, “cold side” of the
9 regenerator and back to the heat stage is cycle “a.” If the total planetary cycle is considered to
10 be 1080 degrees, cycle “b” is offset by 360 degrees in a three lobe rotor and cycle “c” by
11 another 360 degrees.

12 The system takes advantage of the inexorable flow of hot working fluid from the
13 heater stage to the cooling stage. In the process, work is done as the gas expands and cools in
14 rotary machine 1, and then, at the maximum volume of chamber A1, the fluid is moved along
15 through the regenerator or Brayton machine because as chamber A1 is contracting, the fluid is
16 being literally sucked into chamber A2 through the cool stage, such that the volume of the
17 ducting and chambers is reasonably constant for A1+A2. Phasing of slots to enable the
18 volume being contracted in rotary machine 1 to be received by rotary machine 2 is necessary
19 with respect to each chamber. Such phasing can be made easier if more rotary machines of a
20 multiple of 2 with more chambers are available.

21 Intake and exhaust ducts may be used carry fluid to or from intake and exhaust ports, rather
22 than having ports opening during parts of the cycle directly to a working chamber. In this
23 mode of invention, all ports will then be located inside the innermost trace in each chamber of
24 the face of the rotating rotor.

25 If the lobes have their own separate exhaust duct and ports from each other, as suggested in
26 the prior paragraph, the exhaust streams are separated, and if in the same way as the exhaust
27 streams were separated the intake streams are separated, then the rotary machine can be set up
28 by appropriate porting to be a pump and turbine, meaning one working chamber is pumping
29 (intake from lower pressure and exhaust at higher pressure), while another is acting as a
30 turbine (intake from higher pressure and exhaust at lower pressure). In essence, the pumping

1 side will have an early close of intake in the rotor face motion for the working chamber acting
2 as a pump and later opening and closing of exhaust, while the turbine side will have a
3 relatively later close of intake in the rotor face motion for the working chamber acting as a
4 turbine and later opening and closing of exhaust. More likely, the separation of the exhaust
5 streams are separated, and, the intake streams are separated, there can be independent inputs
6 and outputs for each respective working volume for specialized applications.

7 Alternatively, the intake and exhaust ports for one chamber can each have their own fluid
8 source and exhaust outlet, and the intake and exhaust ports for an opposite chamber can each
9 have their own fluid source and exhaust outlet. In that instance, one "side" or chamber can be
10 acting as a compressor, with the other side acting as a turbine using the same previously-
11 described principles for locating ports to achieve these effects.

12 **Discussion common to both embodiments**

13 Entropic losses, though not mandatory in heat transfer, tend to be hard to avoid in heat
14 transfer unless transfer takes place at close temperature, in which situation, heat transfer
15 occurs slowly. What a Carnot cycle of four curves requires is that there be a temperature drop
16 and increase in two curves. Temperature can fall by pressure reduction or volume increase,
17 and temperature can increase as pressure is increased or volume is decreased.

18 Stirling's genius in inventing the Stirling engine, formerly known as a heat engine,
19 was that he recognized that the regenerator could substantially improve efficiency. Much
20 effort has been spent on improving regenerators which function by transferring heat from the
21 right vertical curve of a Stirling cycle on a P-V diagram to the left vertical curve. This
22 invention can utilize that method, but entropic losses can also be minimized by using a
23 Brayton cycle methodology.

24 The Brayton cycle uses a multi-stage turbine and compressor on a common shaft to
25 transfer work from the right vertical curve of the P-V diagram to the left vertical curve. If
26 curve 1 is defined as the curve from the highest temperature point downward to the right
27 where pressure drops and volume increases to a second point, then curve 2 is a curve from
28 this second point at relatively constant volume with pressure and temperature dropping to a
29 point 3 on the P-V diagram. Curve 3 will be defined as the curve on a P-V diagram
30 proceeding to the left with volume decreasing and pressure and temperature rising relatively

1 slightly to a point 4. Curve 4 is the curve on a P-V diagram with volume relatively constant
2 and pressure and temperature rising.

3 It is important to recognize that conformance with the Carnot cycle is key, not
4 necessarily with the Stirling cycle. The reason this is important is that temperature and
5 pressure both drop in a Carnot cycle, and volume increases. In essence, curve 1 and curve 3
6 need to be isothermal or efficiency is lost, but curves 2 and 4 simply need to have as little heat
7 transfer into unavailable heat as possible.

8 A multi-stage turbine to accomplish the result desired in Carnot curve 2 enables
9 incremental expansion of the working fluid and drop in pressure and temperature yielding
10 work by expansion of the volume across the turbine. Work is generally equal to PdV . Such
11 work is transmitted to a shaft sealed from the opposite side of such shaft on which will be
12 located a compressor. Such compressor is also multi-stage, likely in a larger number of stages
13 than the turbine to facilitate increase in pressure and temperature with minimum entropy
14 losses. Volume would shrink across the curve 4.

15 Waste heat from the combustion process could also be used to generate work to
16 enhance the rise in temperature and pressure on curve 4 by a Brayton machine or as set out in
17 the next paragraph, as to an additional Stirling machine. Traditionally, waste heat from
18 combustion has only been used to pre-heat combustion fluid(s), and not to enhance Stirling
19 performance.

20 Additionally, an additional Stirling machine of smaller scale could be used to increase
21 the pressure and temperature into the traditional corner of curve 4 and curve 1, recognizing
22 that that corner is important because the Carnot and Stirling efficiency is the high temperature
23 less the low temperature divided by the high temperature.

24 Conceptually, work is yielded from the entire cycle when high temperature working
25 fluid is depressurized and volume decreases on curve 1. Work is transferred within the cycle
26 by work done on the turbine in curve 2. These gains are offset by the necessity on curve 3 of
27 decreasing the volume from an external source by cooling, and the work in curve 2 simply
28 increases efficiency by achieving the high temperature-high pressure starting point more
29 efficiently using the work (or heat in a regenerator) from curve 2 to achieve curve 4.

Another concept to improve the prior art Stirling machine in order to enhance the temperature and pressure rise at the top left corner of the PV diagram is to not only utilize what waste heat remains after pre-heating oxidizer and fuel for combustion process, as shown in the prior art, but also to enhance the pressure rise in the 4th curve of the PV diagram. This can be done by expanding the waste heat in a turbine, which further drives the shaft tied to the Brayton turbine which is transferring work from the 2nd curve to the 4th curve of the PV cycle. Alternatively, the heat could be used in a Stirling cycle to create more heat for the end of the driving shaft tied to the Brayton turbine which is transferring work from the 2nd curve to the 4th curve of the PV cycle. In any event, the waste heat should be used to enhance the efficiency of the machine by using it to enhance the height of the 4th curve by increasing the temperature and pressure at relatively constant volume. A heat exchanger from the waste heat would be the lowest level enhancement.

Description of the rotor shape

The equations which describe the shape of the peritrochoid and the faces of the rotor are well developed in the open literature, Kenichi Yamamoto, Rotary Engine, Sankaido Co. Ltd. (1st ed. 1981), therefore only the results as they pertain to this embodiment are presented. The shape of the peritrochoid can be represented in orthogonal coordinates x and y by:

where α is the position angle of the main driving shaft and generates periodic motion every 1080 degrees of driving shaft rotation, e is the eccentricity, meaning the amount the rotor axis

1 is displaced from the driving axis, and R is the radius of the rotor, meaning the distance from
2 the rotor axis to the rotor apex.
3

4 The outer bounds of the shape of each rotor face in the preferred embodiment of a three
5 lobe rotor can be represented by:
6
7

8
9
10
11 where the further variable D is found by the following equation as x varies from -30 to $+30$
12 degrees for each face and y is rotated in such range symmetrically about the rotor axis:
13

14
15
16 As the eccentricity e , in the limit, approaches zero, the three faces become closer to being
17 arcs of a circle connecting the apices; however, the ideal compression ratio declines. The
18 machine can also have three lobes.
19
20

21 The invention has other advantages as a result of the thermodynamic and kinetic
22 effects of the fluid being handled and the arrangement and shape of the ports. The ports may
23 be varied to avoid, or to encourage "choking", where fluid speed has reached Mach 1, and to
24 smooth or vary the characteristics of fluid flow through the machine. Those reasonably skilled
25 in the art will recognize that because of kinetic and thermodynamic effects, there are alternate
26 modes available for operation, and while the working chamber is expanding, there could in

1 fact be a short interval of compression, and conversely, while the working chamber is
2 contracting, there could in fact be a short interval of expansion. The invention does not link
3 the entire compression phase with contraction of the working chamber, nor does the invention
4 link the entire expansion phase with increase in volume of the working chamber. Rather,
5 three fluid action phases are referred to. The arbitrarily selected first phase is the expansion
6 phase which would include an intake phase and a compression phase which would include an
7 exhaust phase, and an interphase at which there would be no intake or exhaust. However,
8 there may be fluid expansion even while there is no intake, or while the working chamber is
9 contracting. Similarly, there may be fluid contraction even while there is no exhaust or while
10 the working chamber is expanding.

11 Also contemplated are ducts through the rotor that enable fluid pressure to be applied
12 behind sliding apical tips or springs behind sliding apical tips if greater pressure of the apical
13 tips against the side housing is desired.

14 Another mode of the invention particularly useful where it is important to separate fluid
15 streams flowing through the planetary machine uses independently "tracked" exhaust and
16 intake ducts for each vane face. By placing the ports for intake ducts inside the trace of
17 innermost peritrochoidal trace made by the vane face, including the edge of the vane face
18 having the apical seal, and outside the outermost trace of the annular gear and cam
19 mechanism so proper sealing is maintained, and by placing the ports for exhaust ducts inside
20 the trace of innermost peritrochoidal trace made by the vane face, including the edge of the
21 vane face having the apical seal, and by using peritrochoidal track segments to enable
22 continuous or at least virtually continuous porting of the ports to separate plena at the desired
23 portions of the cycle, the rotary machine yields a novel feature of double pumping of three
24 separate streams of fluid. While all ports could be on one side of the rotor and the six tracks
25 can be fit to correspond to the intake and exhaust porting arrangement, it is easier to have one
26 side be the intake side and the other the exhaust side.

27
28 The demonstration of the mechanics of the air flow in a Stirling cycle can be described
29 as follows: for a first vane face, starting when the face has a minimal working chamber
30 volume, which we will designate as top dead center (TDC), as the chamber expands, fluid

1 flows in from intake duct 1 which flows to the working chamber corresponding to the first
2 vane face. When the chamber is fully expanded, the side of the rotor obstructs further flow
3 from the intake track into the duct into that working chamber. However, at that point a
4 second vane face will correspond to a working chamber of minimal volume which will be
5 exposed to this or another intake track enabling repetition of the just described first portion of
6 the cycle.

7 Returning to the first vane face, depending on the compression desired, the chamber
8 can be foreclosed from the intake or exhaust duct for that face communicating with any
9 ambient fluid. After selected compression, if any, the exhaust duct can communicate with the
10 exhaust track corresponding to the exhaust port, and as the chamber contracts to minimal
11 volume, the fluid inside the working chamber corresponding to the first vane face is
12 exhausted. On exhaust, the second face, will be ready to turn again to the compression and
13 exhaust phases.

14 The invention as described is particularly useful for external combustion engines,
15 including Stirling engines, or Stirling heat pumps, because the planetary rotary machine
16 converts heat energy to rotational work in a simple mechanism in the expansion phase of the
17 working chamber. The volumetric characteristics of this planetary machine are such that
18 combined with a tandem and like machine, the machines cooperating together can have a
19 corresponding working chamber working in tandem such that the sum of the volumes of
20 working chamber 1 in the first machine plus an arbitrarily selected working chamber 1 in the
21 second machine can be set to be a virtual constant. The invention also enables effective
22 sealing because of the PEEK and more precise machining and cam function the lack of which
23 effective parts has been the traditional impediment to utilizing a planetary machine for an
24 external combustion cycle machine.

25 Although several embodiments of the self-lubricating composite, particularly,
26 continuous carbon reinforced PEEK-based material for rotary planetary machines have been
27 disclosed in the foregoing Detailed Description, it will be understood that other embodiments
28 and modifications are possible without departing from the scope of the invention. The
29 invention contemplates that of any two adjacent moving surfaces, one should be made of
30 continuous fiber reinforced polymeric composite (referred to in this paragraph as "CFRPC").

1 Only a part of a surface in contact with a moving part not made of CFPRC need be made of
2 CFPRC, meaning the side transfer plates could be made of CFPRC for the part in contact with
3 the rotor and the circumferential portion in contact with the CFPRC sliding vanes could be
4 made of metal. If the radial velocity for a certain circumference(s) of the rotor was
5 sufficiently low, only part of the rotor need be in contact with self-lubricating material. As
6 another alternative, with more expense involved, the housing can also be made of CFPRC-
7 containing material, but the preferred embodiment is to use a metallic housing with the
8 CFPRC sliding vanes interior to the housing. Because the rotor speed relative to the side
9 transfer plates may be sufficiently low, the side transfer plate could be made of only metal so
10 long as the sliding vanes are made of CFPRC. An alternative application of CFPRC would be
11 as a thin self-lubricating liner fitted into a metallic housing for use with metallic or other
12 apical tip material, but the preferred embodiment is to use a metallic housing with the
13 CFPRC apical tips interior to the housing. The self-lubricating composite is also useful as a
14 bearing material.

15 A person of ordinary skill in the art of Stirling machines will recognize that the
16 preferred embodiments described for heat machines can be reversed for heat pump machines
17 or refrigerators. The claims describe the reverse variation and serially describe the gas loop.

18 Some other pointers to construction of the Stirling machines in this invention are to
19 use a sufficiently sized regenerator minimize eddying and constriction in the regeneration
20 portions of the cycle, that is, to err on the larger rather than the smaller size. As a general
21 proposition, need a relatively larger regenerator for a Stirling refrigerator than for a Stirling
22 engine operating at comparable temperature ranges. Generally, in the preferred mode, a
23 regenerator, which when used to transfer heat is really a heat exchanger, is preferably a cross-
24 flow integrated exchanger because such an exchanger is most likely to have least external
25 losses.

26 A person of ordinary skill in the art of Stirling machines and heat exchangers can
27 make the calculations for the amount of heat to be exchanged and the necessary size of heat
28 exchanger. A person of ordinary skill in the art of Stirling machines and Brayton machines
29 can determine the relative size of turbomachinery to transfer work from the "hot" side of a
30 regenerator to the "cold side."

1 The embodiments represented herein are only a few of the many embodiments and
2 modifications that a practitioner reasonably skilled in the art could make or use. The
3 invention is not limited to these embodiments nor to the versions encompassed in the figure
4 which is intended as an aid to understanding the invention and is not meant to limit the
5 disclosure or the claims. Alternative embodiments and modifications which would still be
6 encompassed by the invention may be made by those skilled in the art, particularly in light of
7 the foregoing teachings. Therefore, the following claims are intended to cover any alternative
8 embodiments, modifications or equivalents which may be included within the spirit and scope
9 of the invention as claimed.

10

1

1 CLAIMS
23 We claim:
4

5 1. A Stirling cycle machine comprising:

6 at least two rotary machines utilizing a planetary cycle, each said rotary machine
7 having a housing and having a rotor, said rotor having at least N apices, N being equal to or
8 greater than 3, each said rotary machine having at least N apex faces, each said apex face
9 being between each two of said at least N apices, each said apex face corresponding to an
10 alternately expanding and contracting chamber between each said apex face and said housing,
11 there thus being N chambers, said chamber expanding and contracting throughout said
12 planetary cycle of said at least two rotary machines;

13 each said apex face having at least one duct penetrating each apex face of said rotary
14 machine to enable communication from each chamber corresponding to each said apex face;

15 at least one port for gas communication exterior to each said at least two rotary
16 machines from each said duct for each of said N chambers, each said at least one port being
17 designated as an exhaust port;

18 at least one port for gas communication exterior to each said at least two rotary
19 machines from each said N chambers, each said at least one port being designated as an intake
20 port;

21 gas flowing in said Stirling cycle machine in at least one closed loop;

22 said Stirling cycle machine having a heat sink for cooling gas, said heat sink being
23 adjacent to an exterior means of cooling;

24 said Stirling cycle machine having a source for heating said gas, a means of flowing
25 heat from said source for heating said gas to said intake ports and expanding volume, a means
26 flow of said heated gas from said source for heating gas to said chambers through said intake
27 ports for each said chamber in the first of said at least two rotary machines in which said first
28 rotary machine heated gas expands generating work to a driving shaft, a regenerator for each
29 said first of said at least two rotary machines, said regenerator having a first portion from
30 which energy in the form of heat or work is extracted from said higher temperature expanded
31 gas and communicated to a second portion of said regenerator, a means of flow of said gas

1 from said chambers through said exhaust ports for each said chamber of said first of said at
2 least two rotary machines to said first portion of said regenerator, a means of flowing heat
3 from said exhaust ports and each said chamber in the second of said at least two rotary
4 machines to a heat sink for cooling said gas after extraction of heat or work in said first
5 portion of said regenerator, a means of flow from said first portion of said regenerator to said
6 heat sink, a means of flow of said gas from said heat sink to said chambers through said intake
7 ports for each said chamber of said second of said at least two rotary machines to contract said
8 gas flowing from said heat sink to said source for heating gas and move said gas toward said
9 source for heating gas by contracting one of said chambers in said second of said at least two
10 rotary machines, a means of flow from said chambers through said exhaust ports for each said
11 chamber of said second of said at least two rotary machines to a second portion of said
12 regenerator receiving energy from said first portion of said regenerator, a means of flow from
13 said second portion of said regenerator to said source for heating gas;

14 said first of said at least two rotary machines having a rotating shaft;

15 said rotating shaft of said first of said at least two rotary machines being coupled to
16 said second of said at least two rotary machines so that said heat flow from said source for
17 heating gas to said heat sink through said first rotary machine causing rotation causes rotation
18 in said second of said at least two rotary machines to contract said gas flowing from said heat
19 sink to said source for heating gas and move said gas toward said source for heating gas;

20 all of said first of said at least two rotary machines being coupled to rotate in the same
21 direction resulting from expansion of said gas, and all of said second of said at least two
22 rotary machines being coupled so that rotation of said second of said at least two rotary
23 machines causes contraction of said chamber containing said gas flowing from said heat sink
24 to said source for heating gas;

25 at least one each said chamber of said first of said at least two rotary machines, in
26 correspondence with one each said chamber in said second of said at least two rotary
27 machines, said regenerator of each of said first of said at least two rotary machines, said heat
28 source, and said heat sink, and all of said means of flow being connected to enable
29 communication of gas such that the tendency of heat to flow from hot to cold is utilized to
30 cycle gas through said Stirling cycle machine in a closed loop, utilizing the movement of gas

1 from heat to cold to generate work and to efficiently facilitate the return movement of cooled
2 gas to the heat source through the second portion of the regenerator, said at least two rotary
3 machines being offset in their planetary cycles to provide a continuous loop of equal volume
4 when the loop is static and empty for each said chamber in correspondence.

5 2. The Stirling cycle machine according to claim 1, further comprising:

6 said heat sink being an ambient temperature heat exchanger for inhibiting heating of
7 said second of said at least two rotary machines.

8 3. The Stirling cycle machine according to claim 2, further comprising:

9 said source for heating gas utilizing combustion elements; and

10 a means of utilizing waste heat from said source for heating gas to pre-heat at least one
11 of said combustion elements.

12 4. The Stirling cycle machine according to claim 3 further comprising:

13 a supplementary pressure pump powered by at least a portion of waste heat from said
14 source for heating gas to increase pressure and temperature of gas prior to entry of gas into
15 said first of said at least two rotary machines to enable achievement of highest temperature
16 and inhibit the rounding off of the pressure-volume curve in a Stirling cycle machine.

17 5. The Stirling cycle machine according to claim 4, further comprising:

18 said driving shaft and said rotating shaft being mechanically coupled.

19 6. The Stirling cycle machine according to claim 5, further comprising,

20 said rotating shaft of said first of said at least two rotary machines being mechanically
21 coupled to said second of said at least two rotary machines, all of said first of said at least two
22 rotary machines being coupled by said driving shaft with gears.

23 7. The Stirling cycle machine according to claim 5, further comprising:

24 said rotating shaft of said first of said at least two rotary machines being mechanically
25 coupled to said second of said at least two rotary machines, all of said first of said at least two
26 rotary machines being coupled by said driving shaft with cams.

27 8. The Stirling cycle machine according to claim 7, further comprising:

28 said cams enabling variable speeds between said rotor of said first of said at least two
29 rotary machines, and said rotor of said first of said at least two rotary machines.

30 9. The Stirling cycle machine according to claim 5, further comprising:

1 said rotating shaft of said first of said at least two rotary machines being mechanically
2 coupled to said second of said at least two rotary machines, all of said first of said at least two
3 rotary machines being coupled by said driving shaft to rotate in the same direction.

4 10. The Stirling cycle machine according to claim 5, further comprising:

5 said tips for said at least N apices, said tips being composed of a self-lubricating
6 composite.

7 11. The Stirling cycle machine according to claim 10, further comprising:

8 said parts of said housing in relative motion to said rotor being composed of self-
9 lubricating composite.

10 12. The Stirling cycle machine according to claim 11, further comprising:

11 said tips for said at least N apices being composed of self-lubricating composite
12 selected from the group of continuous carbon fiber reinforced PEEK or PEKK.

13 13. The Stirling cycle machine according to claim 12, further comprising:

14 said parts of said housing in relative motion to said rotor being composed of self-
15 lubricating composite selected from the group of PEEK or PEKK.

16 14. The Stirling cycle machine according to claim 13, further comprising:

17 said gas flowing in said machine containing compounds having lubricating
18 characteristics for reducing friction in said at least two rotary machines.

19 15. The Stirling cycle machine according to claim 5, further comprising:

20 a supplementary pressure pump in said loop to increase pressure and temperature of
21 gas after exit of gas from said first of said at least two rotary machines to enable achievement
22 of highest pressure and lowest volume and inhibit the rounding off of the pressure-volume
23 curve in said Stirling cycle machine.

24 16. The Stirling cycle machine according to claim 5, further comprising:

25 said exhaust ports of said first of said at least two rotary machines being truncated to
26 increase pressure and temperature of gas upon exit of gas from said second of said at least two
27 rotary machines to enable achievement of highest pressure and lowest volume and inhibit the
28 rounding off of the pressure-volume curve in said Stirling heat pump machine.

29 17. The Stirling cycle machine according to claim 15, further comprising:

1 a mechanical means for transferring energy from said first portion of said regenerator
2 to said second portion of said regenerator, said mechanical means for transferring energy
3 being coupled by gears to said rotating shaft to rotate and thereby transfer energy in the form
4 of work at a rate consistent with the pressure times the change in temperature contemplated in
5 the pressure-volume diagram of said Stirling heat pump machine contemplated.

6 18. The Stirling cycle machine according to claim 17, further comprising:
7 said mechanical means for transferring energy being a Brayton machine.

8 19. A Stirling heat pump machine utilizing a Stirling cycle comprising:

9 at least two rotary machines utilizing a planetary cycle, each said rotary machine
10 having a housing and having at least N apices, N being equal to or greater than 2, each said
11 rotary machine having at least N apex faces each said apex face corresponding to an
12 alternately expanding and contracting chamber between each said apex face and said housing,
13 said chamber expanding and contracting throughout said planetary cycle of said at least two
14 rotary machines;

15 each said apex face having at least one duct penetrating each apex face of said rotary
16 machine to enable communication from each chamber corresponding to each said apex face;

17 at least one port for gas communication exterior to each said at least two rotary
18 machines from each said duct for each of said N chambers, each said at least one port being
19 designated as an exhaust port;

20 at least one port for gas communication exterior to each said at least two rotary
21 machines from each said N chambers, each said at least one port being designated as an intake
22 port;

23 gas flowing in said Stirling heat pump machine in at least one closed loop;

24 said Stirling heat pump machine having a heat sink for cooling gas, said heat sink
25 being adjacent to an exterior means of cooling;

26 a means of flow of gas to said chambers through said intake ports for each said
27 chamber of the first of said at least two rotary machines, said first of said at least two rotary
28 machines being driven by a driving shaft by an exterior power source and in which said first
29 of said at least two rotary machines said gas is compressed by contracting one of said
30 chambers in said first of said at least two rotary machines, a means of flowing heat from said

1 exhaust ports and said contracting chamber in said first of said at least two rotary machines to
2 said heat sink, a means of flow from said chambers through said exhaust ports for said
3 chamber in said first of said at least two rotary machines to said heat sink, said heat sink for
4 removing heat from said compressed gas, a regenerator for each said first of said at least two
5 rotary machines, said regenerator having a first portion from which energy in the form of heat
6 or work is transferred from said gas in preparation for the expansion phase of the cycle, a
7 means of flow from said heat sink to said first portion of said regenerator, a second of said at
8 least two rotary machines, a means of flow to said chambers through said intake ports for each
9 said chamber in said second of said at least two rotary machines wherein said gas is expanded
10 in said chamber in said second of said at least two rotary machines, a "heat source" being the
11 object which is the subject of cooling and from which object heat will be extracted and
12 therefore cooled by the Stirling heat pump machine, a means of flowing heat from said "heat
13 source" to said exhaust ports and said chamber which is expanding in said second of said at
14 least two rotary machines, a means of flow from said chambers through said exhaust ports for
15 each said chamber of said second of said at least two rotary machines to said "heat source"
16 being cooled, a means of flow to said second portion of said regenerator, said second portion
17 of said regenerator being for the purpose of introducing energy into said gas, a means of flow
18 from said second portion of said regenerator to said means of flow to said chambers through
19 said intake ports for each said chamber of said first of said at least two rotary machines;
20 said first of said at least two rotary machines having a rotating shaft;
21 said rotating shaft of said first of said at least two rotary machines being coupled to
22 said second of said at least two rotary machines so that heat flow from said "heat source"
23 being the object which is the subject of cooling reduces the power to operate said Stirling heat
24 pump machine;
25 said first of said at least two rotary machines and said second of said at least two
26 rotary machines being coupled to rotate in the same direction;
27 at least one each said chamber of said at least two rotary machines, in correspondence
28 with one each said chamber in said second of said at least two rotary machines, said
29 regenerator of each of said first of said at least two rotary machines, said heat source, and said
30 heat sink, and all of said means of flow being connected to enable communication of gas,

1 utilizing the input of work to reverse the Stirling heat engine cycle to extract heat and dispose
2 of it externally thereby cooling an object or area, said at least two rotary machines being
3 offset in their planetary cycles to provide a continuous loop of equal volume when the loop is
4 static and empty for each said chamber in correspondence.

5 20. The Stirling heat pump machine according to claim 19, further comprising:

6 said heat sink being an ambient temperature heat exchanger for inhibiting heating of
7 said first of said at least two rotary machines.

8
9 21. The Stirling heat pump machine according to claim 20, further comprising:

10 said driving shaft and said rotating shaft being mechanically coupled.

11 22. The Stirling heat pump machine according to claim 21, further comprising,

12 said rotating shaft of said first of said at least two rotary machines being mechanically
13 coupled to said second of said at least two rotary machines, all of said first of said at least two
14 rotary machines being coupled by said driving shaft with gears.

15 23. The Stirling heat pump machine according to claim 21, further comprising:

16 said rotating shaft of said first of said at least two rotary machines being mechanically
17 coupled to said second of said at least two rotary machines, all of said first of said at least two
18 rotary machines being coupled by said driving shaft with cams.

19 24. The Stirling heat pump machine according to claim 21, further comprising:

20 said cams enabling variable speeds between said rotor of said first of said at least two
21 rotary machines, and said rotor of said first of said at least two rotary machines.

22 25. The Stirling heat pump machine according to claim 21, further comprising:

23 said rotating shaft of said first of said at least two rotary machines being mechanically
24 coupled to said second of said at least two rotary machines, all of said first of said at least two
25 rotary machines being coupled by said driving shaft to rotate in the same direction.

26 26. The Stirling heat pump machine according to claim 21, further comprising:

27 said tips for said at least N apices, said tips being composed of a self-lubricating
28 composite.

29 27. The Stirling heat pump machine according to claim 26, further comprising:

1 said parts of said housing in relative motion to said rotor being composed of self-
2 lubricating composite.

3 28. The Stirling heat pump machine according to claim 26, further comprising:

4 said tips for said at least N apices being composed of self-lubricating composite
5 selected from the group of continuous carbon fiber reinforced PEEK or PEKK.

6 29. The Stirling heat pump machine according to claim 28, further comprising:

7 said parts of said housing in relative motion to said rotor being composed of self-
8 lubricating composite selected from the group of PEEK or PEKK.

9 30. The Stirling heat pump machine according to claim 29, further comprising:

10 said gas flowing in said machine containing compounds having lubricating
11 characteristics for reducing friction in said at least two rotary machines.

12 31. The Stirling heat pump machine according to claim 21, further comprising:

13 a supplementary pressure pump in said loop to increase pressure and temperature of
14 gas after exit of gas from said first of said at least two rotary machines to enable achievement
15 of highest pressure and lowest volume and inhibit the rounding off of the pressure-volume
16 curve in said Stirling heat pump machine.

17 32. The Stirling heat pump machine according to claim 21, further comprising:

18 said exhaust ports of said first of said at least two rotary machines being truncated to
19 increase pressure and temperature of gas upon exit of gas from said second of said at least two
20 rotary machines to enable achievement of highest pressure and lowest volume and inhibit the
21 rounding off of the pressure-volume curve in said Stirling heat pump machine.

22 33. The Stirling heat pump machine according to claim 21, further comprising:

23 a mechanical means for transferring energy from said first portion of said regenerator
24 to said second portion of said regenerator, said mechanical means for transferring energy
25 being coupled by gears to said rotating shaft to rotate and thereby transfer energy in the form
26 of work at a rate consistent with the pressure times the change in temperature contemplated in
27 the pressure-volume diagram of said Stirling heat pump machine contemplated.

28 34. The Stirling heat pump machine according to claim 33, further comprising:

29 said mechanical means for transferring energy being a Brayton machine.

30 35. A Stirling cycle machine comprising:

1 at least two rotary machines utilizing a planetary cycle, each said rotary machine
2 having a housing and having at least N apices, N being equal to or greater than 2, each said
3 rotary machine having at least N apex faces each said apex face corresponding to an
4 alternately expanding and contracting chamber between each said apex face and said housing,
5 there thus being N chambers, said chamber expanding and contracting throughout said
6 planetary cycle of said at least two rotary machines;

7 each said apex face having at least one duct penetrating each apex face of said rotary
8 machine to enable communication from each chamber corresponding to each said apex face;

9 at least one port for gas communication exterior to each said at least two rotary
10 machines from each said duct for each of said N chambers, each said at least one port being
11 designated as an exhaust port;

12 at least one port for gas communication exterior to each said at least two rotary
13 machines from each said N chambers, each said at least one port being designated as an intake
14 port;

15 gas flowing in said Stirling cycle machine in at least one closed loop;

16 said Stirling cycle machine having a source for heating the first of said rotary
17 machines in which said first rotary machine gas expands generating work to a driving shaft, a
18 regenerator for each said first of said at least two rotary machines, said regenerator having a
19 first portion from which energy in the form of heat or work is extracted from said gas and
20 communicated to a second portion of said regenerator, a means of flow of said gas from said
21 chambers through said exhaust ports for each said chamber of said first of said at least two
22 rotary machines to said first portion of said regenerator, a means of flow from said first
23 portion of said regenerator to said chambers through said intake ports for each said chamber
24 of the second of said at least two rotary machines, a heat sink for cooling said second of said
25 at least two rotary machines, in which said second of said at least two rotary machines, gas is
26 moved toward said second portion of said regenerator by contracting one of said chambers in
27 said second of said at least two rotary machines, a means of flow from said chambers through
28 said exhaust ports for each said chamber of said second of said at least two rotary machines to
29 a second portion of said regenerator receiving energy from said first portion of said
30 regenerator, a means of flow from said second portion of said regenerator to said chambers

1 through said intake ports for each said chamber of said first of said at least two rotary
2 machines;

3 said rotating shaft of said first of said at least two rotary machines being coupled to
4 said second of said at least two rotary machines so work in said first rotary machine causing
5 rotation causes rotation in said second of said at least two rotary machines to contract said gas
6 flowing in said second of said at least two rotary machines and move said gas back to said
7 first of said at least two rotary machines;

8 said rotating shaft of said first of said at least two rotary machines being coupled to
9 said second of said at least two rotary machines so that said heat flow from said source for
10 heating gas to said heat sink through said first rotary machine causing rotation causes rotation
11 in said second of said at least two rotary machines to contract said gas flowing from said heat
12 sink to said source for heating gas and move said gas toward said first of said at least two
13 rotary machines;

14 at least one each said chamber of said first of said at least two rotary machines, in
15 correspondence with one each said chamber in said second of said at least two rotary
16 machines, said regenerator of each of said first of said at least two rotary machines, said heat
17 source, and said heat sink, and all of said means of flow being connected to enable a
18 continuous flow and communication of gas such that the tendency of heat to flow from hot to
19 cold is utilized to continuously cycle gas through said Stirling cycle machine in a closed loop,
20 utilizing the movement of gas from heat to cold to generate work and to efficiently facilitate
21 the return movement of cooled gas to the heat source through the second portion of the
22 regenerator, said at least two rotary machines being offset in their planetary cycles to provide
23 a continuous loop of equal volume when the loop is static and empty for each said chamber in
24 correspondence .

25 36. The Stirling cycle machine according to claim 35, further comprising:

26 said heat sink being an ambient temperature heat exchanger for inhibiting heating of
27 said second of said at least two rotary machines.

28 37. The Stirling cycle machine according to claim 36, further comprising:

29 said source for heating gas utilizing combustion elements; and

1 a means of utilizing waste heat from said source for heating gas to pre-heat at least one
2 of said combustion elements.

3 38. The Stirling cycle machine according to claim 37, further comprising:

4 a supplementary pressure pump powered by at least a portion of waste heat from said
5 source for heating gas to increase pressure and temperature of gas prior to entry of gas into
6 said first of said at least two rotary machines to enable achievement of highest temperature
7 and inhibit the rounding off of the pressure-volume curve in a Stirling cycle machine.

8 39. The Stirling cycle machine according to claim 38, further comprising:

9 said driving shaft and said rotating shaft being mechanically coupled.

10 40. The Stirling cycle machine according to claim 39, further comprising,

11 said rotating shaft of said first of said at least two rotary machines being mechanically
12 coupled to said second of said at least two rotary machines, all of said first of said at least two
13 rotary machines being coupled by said driving shaft with gears.

14 41. The Stirling cycle machine according to claim 39, further comprising:

15 said rotating shaft of said first of said at least two rotary machines being mechanically
16 coupled to said second of said at least two rotary machines, all of said first of said at least two
17 rotary machines being coupled by said driving shaft with cams.

18 42. The Stirling cycle machine according to claim 41, further comprising:

19 said cams enabling variable speeds between said rotor of said first of said at least two
20 rotary machines, and said rotor of said first of said at least two rotary machines.

21 43. The Stirling cycle machine according to claim 39, further comprising:

22 said rotating shaft of said first of said at least two rotary machines being mechanically
23 coupled to said second of said at least two rotary machines, all of said first of said at least two
24 rotary machines being coupled by said driving shaft to rotate in the same direction.

25 44. The Stirling cycle machine according to claim 39, further comprising:

26 said tips for said at least N apices, said tips being composed of a self-lubricating
27 composite.

28 45. The Stirling cycle machine according to claim 43, further comprising:

29 said parts of said housing in relative motion to said rotor being composed of self-
30 lubricating composite.

1 46. The Stirling cycle machine according to claim 44, further comprising:

2 said tips for said at least N apices being composed of self-lubricating composite
3 selected from the group of continuous carbon fiber reinforced PEEK or PEKK.

4 47. The Stirling cycle machine according to claim 46, further comprising:

5 said parts of said housing in relative motion to said rotor being composed of self-
6 lubricating composite selected from the group of PEEK or PEKK.

7 48. The Stirling cycle machine according to claim 47, further comprising:

8 said gas flowing in said machine containing compounds having lubricating
9 characteristics for reducing friction in said at least two rotary machines.

10 49. The Stirling cycle machine according to claim 39, further comprising:

11 a supplementary pressure pump in said loop to increase pressure and temperature of
12 gas after exit of gas from said first of said at least two rotary machines to enable achievement
13 of highest pressure and lowest volume and inhibit the rounding off of the pressure-volume
14 curve in said Stirling cycle machine.

15 50. The Stirling cycle machine according to claim 39, further comprising:

16 said exhaust ports of said first of said at least two rotary machines being truncated to
17 increase pressure and temperature of gas upon exit of gas from said second of said at least two
18 rotary machines to enable achievement of highest pressure and lowest volume and inhibit the
19 rounding off of the pressure-volume curve in said Stirling heat pump machine.

20 51. The Stirling cycle machine according to claim 39, further comprising:

21 a mechanical means for transferring energy from said first portion of said regenerator
22 to said second portion of said regenerator, said mechanical means for transferring energy
23 being coupled by gears to said rotating shaft to rotate and thereby transfer energy in the form
24 of work at a rate consistent with the pressure times the change in temperature contemplated in
25 the pressure-volume diagram of said Stirling heat pump machine contemplated.

26 52. The Stirling cycle machine according to claim 51, further comprising:

27 said mechanical means for transferring energy being a Brayton machine.

28 53. A Stirling heat pump machine utilizing a Stirling cycle comprising:

29 at least two rotary machines utilizing a planetary cycle, each said rotary machine
30 having a housing and having at least N apices, N being equal to or greater than 2, each said

1 rotary machine having at least N apex faces each said face corresponding to an alternately
2 expanding and contracting chamber between each said apex face and said housing, said
3 chamber expanding and contracting throughout said planetary cycle of said at least two rotary
4 machines;

5 each said apex face having at least one duct penetrating each apex face of said rotary
6 machine to enable communication from each chamber corresponding to each said apex face;

7 at least one port for gas communication exterior to each said at least two rotary
8 machines from each said duct for each of said N chambers, each said at least one port being
9 designated as an exhaust port;

10 at least one port for gas communication exterior to each said at least two rotary
11 machines from each said N chambers, each said at least one port being designated as an intake
12 port;

13 gas flowing in said Stirling heat pump machine in at least one closed loop;

14 said Stirling heat pump machine having a heat sink for the first of said at least two
15 rotary machines, said heat sink being having an ambient temperature heat exchanger for
16 inhibiting heating of the first of said at least two rotary machines, said first of said at least two
17 rotary machines being driven by a driving shaft by an exterior power source and in which said
18 first of said at least two rotary machines said gas is compressed by contracting one of said
19 chambers in said first of said at least two rotary machines, a means of flow from said
20 chambers through said exhaust ports for each said chamber in said first of said at least two
21 rotary machines to a regenerator for each said first of said at least two rotary machines, said
22 regenerator having a first portion from which energy in the form of heat or work is transferred
23 from said gas in preparation for the expansion phase of the cycle, a means of flow from said
24 heat sink to said first portion of said regenerator, a means of flow to said chambers through
25 said intake ports for each said chamber in the second of said at least two rotary machines
26 wherein said gas is expanded in said chambers in said second of said at least two rotary
27 machines, a "heat source" being the object which is the subject of cooling and from which
28 object heat will be extracted and therefore cooled by said second of said at least two rotary
29 machines, a means of flow from said exhaust ports of said second of said at least two rotary
30 machines to said second portion of said regenerator, said second portion of said regenerator

1 being for the purpose of introducing energy into said gas, a means of flow from said second
2 portion of said regenerator to said means of flow to said intake ports of said first of said at
3 least two rotary machines;

4 said first of said at least two rotary machines having a rotating shaft;

5 said rotating shaft of said first of said at least two rotary machines being coupled to
6 said second of said at least two rotary machines so that heat flow from said "heat source"
7 being the object which is the subject of cooling flows and reduces the power to turn said
8 driving shaft;

9 said first of said at least two rotary machines and said second of said at least two
10 rotary machines being coupled to rotate in the same direction;

11 at least one each said chamber of said at least two rotary machines, in correspondence
12 with one each said chamber in said second of said at least two rotary machines, said
13 regenerator of each of said first of said at least two rotary machines, said heat source, and said
14 heat sink, and all of said means of flow being connected to enable communication of gas,
15 utilizing the input of work to reverse the Stirling heat engine cycle to extract heat and dispose
16 of it externally thereby cooling an object or area, said at least two rotary machines being
17 offset in their planetary cycles to provide a continuous loop of equal volume when the loop is
18 static and empty for each said chamber in correspondence.

19 54. The Stirling heat pump machine according to claim 53, further comprising:

20 said heat sink being an ambient temperature heat exchanger for inhibiting heating of
21 said first of said at least two rotary machines.

22
23 55. The Stirling heat pump machine according to claim 54, further comprising:

24 said driving shaft and said rotating shaft being mechanically coupled.

25 56. The Stirling heat pump machine according to claim 55, further comprising,

26 said rotating shaft of said first of said at least two rotary machines being mechanically
27 coupled to said second of said at least two rotary machines, all of said first of said at least two
28 rotary machines being coupled by said driving shaft with gears.

29 57. The Stirling heat pump machine according to claim 55, further comprising:

1 said rotating shaft of said first of said at least two rotary machines being mechanically
2 coupled to said second of said at least two rotary machines, all of said first of said at least two
3 rotary machines being coupled by said driving shaft with cams.

4 58. The Stirling heat pump machine according to claim 55, further comprising:

5 said cams enabling variable speeds between said rotor of said first of said at least two
6 rotary machines, and said rotor of said first of said at least two rotary machines.

7 59. The Stirling heat pump machine according to claim 55, further comprising:

8 said rotating shaft of said first of said at least two rotary machines being mechanically
9 coupled to said second of said at least two rotary machines, all of said first of said at least two
10 rotary machines being coupled by said driving shaft to rotate in the same direction.

11 60. The Stirling heat pump machine according to claim 55, further comprising:

12 said tips for said at least N apices, said tips being composed of a self-lubricating
13 composite.

14 61. The Stirling heat pump machine according to claim 60, further comprising:

15 said parts of said housing in relative motion to said rotor being composed of self-
16 lubricating composite.

17 62. The Stirling heat pump machine according to 60, further comprising:

18 said tips for said at least N apices being composed of self-lubricating composite
19 selected from the group of continuous carbon fiber reinforced PEEK or PEKK.

20 63. The Stirling heat pump machine according to claim 62, further comprising:

21 said parts of said housing in relative motion to said rotor being composed of self-
22 lubricating composite selected from the group of PEEK or PEKK.

23 64. The Stirling heat pump machine according to claim 63, further comprising:

24 said gas flowing in said machine containing compounds having lubricating
25 characteristics for reducing friction in said at least two rotary machines.

26 65. The Stirling heat pump machine according to claim 55, further comprising:

27 a supplementary pressure pump in said loop to increase pressure and temperature of
28 gas after exit of gas from said first of said at least two rotary machines to enable achievement
29 of highest pressure and lowest volume and inhibit the rounding off of the pressure-volume
30 curve in said Stirling heat pump machine.

1 66. The Stirling heat pump machine according to claim 55, further comprising:

2 said exhaust ports of said first of said at least two rotary machines being truncated to
3 increase pressure and temperature of gas upon exit of gas from said second of said at least two
4 rotary machines to enable achievement of highest pressure and lowest volume and inhibit the
5 rounding off of the pressure-volume curve in said Stirling heat pump machine.

6 67. The Stirling heat pump machine according to claim 55, further comprising:

7 a mechanical means for transferring energy from said first portion of said regenerator
8 to said second portion of said regenerator, said mechanical means for transferring energy
9 being coupled by gears to said rotating shaft to rotate and thereby transfer energy in the form
10 of work at a rate consistent with the pressure times the change in temperature contemplated in
11 the pressure-volume diagram of said Stirling heat pump machine contemplated.

12 68. The Stirling heat pump machine according to claim 67, further comprising:

13 said mechanical means for transferring energy being a Brayton machine.

14 69. A Stirling cycle machine comprising:

15 at least two rotary machines utilizing a planetary cycle, each said rotary machine
16 having a housing and having at least N apices, N being equal to or greater than 2, each said
17 rotary machine having at least N apex faces each said apex face corresponding to an
18 alternately expanding and contracting chamber between each said apex face and said housing,
19 there thus being N chambers, said chamber expanding and contracting throughout said
20 planetary cycle of said at least two rotary machines;

21 each said housing of each said rotary machine having side plates;

22 each said apex face having at least one intake duct penetrating each apex face of said
23 rotary machine to enable gas communication from each chamber corresponding to each said
24 apex face to an intake port on said rotor adjacent to said side plate;

25 each said intake port for a particular chamber on each said rotary machine being offset
26 from each other so that all intake ports for each said chamber trace the same track on said side
27 plate of said rotary machine to which each said intake port is adjacent, and trace a different
28 track from all ports for any other chamber on said rotary machine;

29 a separate intake port tracking slot in said side plate corresponding to said separate
30 track traced by all said intake ports for each said chamber on each said machine;

1 each said intake port tracking slot having an exclusive plenum exterior to said side
2 plate for said intake port tracking slot so that during said planetary cycle of said rotary
3 machine, each said chamber can communicate gas exclusively with said exteriorly located
4 plenum for each said chamber so that as each said chamber expands, gas can flow into each
5 said chamber separately from any gas flow into or out of any other chamber of said machine;

6 each said apex face having at least one exhaust duct penetrating each apex face of said
7 rotary machine to enable gas communication from each chamber corresponding to each said
8 apex face to an exhaust port on said rotor adjacent to said side plate;

9 each said exhaust port for a particular chamber on each said rotary machine being
10 offset from each other so that all exhaust ports for each said chamber trace the same track on
11 said side plate of said rotary machine to which each said exhaust port is adjacent, and trace a
12 different track from all ports for any other chamber on said rotary machine;

13 a separate exhaust port tracking slot in said side plate corresponding to said separate
14 track traced by all said exhaust ports for each said chamber on each said machine;

15 each said exhaust port tracking slot having an exclusive chamber plenum exterior to
16 said side plate for said exhaust port tracking slot so that during said planetary cycle of said
17 rotary machine, each said chamber can communicate gas exclusively with said exteriorly
18 located chamber plenum for each said chamber so that as each said chamber expands, gas can
19 flow into each said chamber separately from any gas flow into or out of any other chamber of
20 said machine, so that there are a total N chamber plena for intake and N chamber plena for
21 exhaust and each said chamber has an intake chamber plenum and an exhaust chamber
22 plenum;

23 gas flowing in said Stirling cycle machine in at least one closed loop;

24 said Stirling cycle machine having a source for heating the first of said at least two
25 rotary machines, a regenerator for each said chamber of said first of said at least two rotary
26 machines, said chambers in each machine being designated from 1 to N, each said regenerator
27 having a first portion from which energy in the form of heat or work is extracted from said
28 gas and communicated to a second portion of said regenerator, and each set of two of said at
29 least two rotary machines having a means of flow of said gas from said exhaust plenum for
30 said chamber designated number 1 through N of said first of said at least two rotary machines

1 to an intake plenum for said chamber designated 1 through N on said second of said at least
2 two rotary machines through said first portion of said regenerator for said chambers 1 through
3 N, respectively, a means of flow of said gas from said exhaust plenum of said chamber 1
4 through N of said second of said at least two rotary machines through said second portion of
5 said regenerator to said intake plenum for said chamber 1 through N of said first of said at
6 least two rotary machines, respectively, thereby forming a series 1 through N of pairs of
7 connected chambers in a series of gas loops 1 through N;

8 said each chamber of said second of said at least two rotary machines being linked to
9 said selected chamber of said first of said at least two rotary machines being selected to have
10 substantially equal volumes for the sum of volumes of each pair of connected chambers in
11 said series 1 through N at any given position of said planetary cycle of said rotary machines
12 utilizing offset planetary cycles as between said first and second of said at least two rotary
13 machines;

14 said series of gas loops 1 through N having substantially equal volumes throughout
15 said planetary cycle of said rotary machines;

16 said first of said at least two rotary machines having a rotating shaft;

17 said rotating shaft rotating from work resulting from expansion in each said chamber
18 in said first of said at least two rotary machines;

19 said rotating shaft being coupled to said second of said at least two rotary machines so
20 work in said first rotary machine causing rotation causes rotation in said second of said at
21 least two rotary machines to move said gas flowing in said second of said at least two rotary
22 machines back to said first of said at least two rotary machines;

23 thereby forming a series of N closed loops through chambers 1 through N for gas in
24 each said loop to pass from each said first of said at least two rotary machines which is being
25 heated through said first portion for each said regenerator for each said loop and then through
26 said second of said second of said at least two rotary machines returning to said first rotary
27 machine through said second portion of said regenerator for each loop and thereby utilizing
28 the tendency of heat to flow from hot to cold to generate work to said driving shaft in said
29 first of at least two rotary machines, to cycle gas through said Stirling cycle machine in said

1 closed gas loops, and to efficiently facilitate the return cycling of cooler gas to said heat
2 source through said second portion of the regenerator.

3 70. The Stirling cycle machine according to claim 69, further comprising:

4 said second of said at least two rotary machines having heat sink for inhibiting heating
5 of said second of said at least two rotary machines.

6 71. The Stirling cycle machine according to claim 70, further comprising:

7 said heat sink being an ambient temperature heat exchanger for inhibiting heating of
8 said second of said at least two rotary machines.

9 72. The Stirling cycle machine according to claim 71, further comprising:

10 said source for heating gas utilizing combustion elements; and
11 a means of utilizing waste heat from said source for heating gas to pre-heat at least one
12 of said combustion elements.

13 73. The Stirling cycle machine according to claim 72, further comprising:

14 a supplementary pressure pump powered by at least a portion of waste heat from said
15 source for heating gas to increase pressure and temperature of gas prior to entry of gas into
16 said first of said at least two rotary machines to enable achievement of highest temperature
17 and inhibit the rounding off of the pressure-volume curve in a Stirling cycle machine.

18 74. The Stirling cycle machine according to claim 73, further comprising:

19 said driving shaft and said rotating shaft being mechanically coupled.

20 75. The Stirling cycle machine according to claim 74, further comprising,

21 said rotating shaft of said first of said at least two rotary machines being mechanically
22 coupled to said second of said at least two rotary machines, all of said first of said at least two
23 rotary machines being coupled by said driving shaft with gears.

24 76. The Stirling cycle machine according to claim 73, further comprising:

25 said rotating shaft of said first of said at least two rotary machines being mechanically
26 coupled to said second of said at least two rotary machines, all of said first of said at least two
27 rotary machines being coupled by said driving shaft with cams.

28 77. The Stirling cycle machine according to claim 76, further comprising:

29 said cams enabling variable speeds between said rotor of said first of said at least two
30 rotary machines, and said rotor of said first of said at least two rotary machines.

1 78. The Stirling cycle machine according to claim 74, further comprising:

2 said rotating shaft of said first of said at least two rotary machines being mechanically
3 coupled to said second of said at least two rotary machines, all of said first of said at least two
4 rotary machines being coupled by said driving shaft to rotate in the same direction.

5 79. The Stirling cycle machine according to claim 74, further comprising:

6 said tips for said at least N apices, said tips being composed of a self-lubricating
7 composite.

8 80. The Stirling cycle machine according to claim 79, further comprising:

9 said parts of said housing in relative motion to said rotor being composed of self-
10 lubricating composite.

11 81. The Stirling cycle machine according to claim 80, further comprising:

12 said tips for said at least N apices being composed of self-lubricating composite
13 selected from the group of continuous carbon fiber reinforced PEEK or PEKK.

14 82. The Stirling cycle machine according to claim 81, further comprising:

15 said parts of said housing in relative motion to said rotor being composed of self-
16 lubricating composite selected from the group of PEEK or PEKK.

17 83. The Stirling cycle machine according to claim 82, further comprising:

18 said gas flowing in said machine containing compounds having lubricating
19 characteristics for reducing friction in said at least two rotary machines.

20 84. The Stirling cycle machine according to claim 74, further comprising:

21 a supplementary pressure pump in said loop to increase pressure and temperature of
22 gas after exit of gas from said first of said at least two rotary machines to enable achievement
23 of highest pressure and lowest volume and inhibit the rounding off of the pressure-volume
24 curve in said Stirling cycle machine.

25 85. The Stirling cycle machine according to claim 74, further comprising:

26 said exhaust ports of said first of said at least two rotary machines being truncated to
27 increase pressure and temperature of gas upon exit of gas from said second of said at least two
28 rotary machines to enable achievement of highest pressure and lowest volume and inhibit the
29 rounding off of the pressure-volume curve in said Stirling heat pump machine.

30 86. The Stirling cycle machine according to claim 84, further comprising:

1 a mechanical means for transferring energy from said first portion of said regenerator
2 to said second portion of said regenerator, said mechanical means for transferring energy
3 being coupled by gears to said rotating shaft to rotate and thereby transfer energy in the form
4 of work at a rate consistent with the pressure times the change in temperature contemplated in
5 the pressure-volume diagram of said Stirling heat pump machine contemplated.

6 87. The Stirling cycle machine according to claim 86, further comprising:

7 said mechanical means for transferring energy being a Brayton machine.

8 88. The Stirling cycle machine according to claim 74, further comprising:

9 a means for gas communication within each said tracking slot to enable continuous gas
10 communication within each said tracking slot.

11 89. The Stirling cycle machine according to claim 88, further comprising:

12 said means for gas communication within each said tracking slot to enable continuous
13 gas communication within each said tracking slot being a grill.

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17 90. A Stirling heat pump machine comprising:

18 at least two rotary machines utilizing a planetary cycle, each said rotary machine
19 having a housing and having at least N apices, N being equal to or greater than 2, each said
20 rotary machine having at least N apex faces each said apex face corresponding to an
21 alternately expanding and contracting chamber between each said apex face and said housing,
22 there thus being N chambers, said chamber expanding and contracting throughout said
23 planetary cycle of said at least two rotary machines;

24 each said housing of each said rotary machine having side plates;

25 each said apex face having at least one intake duct penetrating each apex face of said
26 rotary machine to enable gas communication from each chamber corresponding to each said
27 apex face to an intake port on said rotor adjacent to said side plate;

28 each said intake port for a particular chamber on each said rotary machine being offset
29 from each other so that all intake ports for each said chamber trace the same track on said side

1 plate of said rotary machine to which each said intake port is adjacent, and trace a different
2 track from all ports for any other chamber on said rotary machine;

3 a separate intake port tracking slot in said side plate corresponding to said separate
4 track traced by all said intake ports for each said chamber on each said machine;

5 each said intake port tracking slot having an exclusive plenum exterior to said side
6 plate for said intake port tracking slot so that during said planetary cycle of said rotary
7 machine, each said chamber can communicate gas exclusively with said exteriorly located
8 plenum for each said chamber so that as each said chamber expands, gas can flow into each
9 said chamber separately from any gas flow into or out of any other chamber of said machine;

10 each said apex face having at least one exhaust duct penetrating each apex face of said
11 rotary machine to enable gas communication from each chamber corresponding to each said
12 apex face to an exhaust port on said rotor adjacent to said side plate;

13 each said exhaust port for a particular chamber on each said rotary machine being
14 offset from each other so that all exhaust ports for each said chamber trace the same track on
15 said side plate of said rotary machine to which each said exhaust port is adjacent, and trace a
16 different track from all ports for any other chamber on said rotary machine;

17 a separate exhaust port tracking slot in said side plate corresponding to said separate
18 track traced by all said exhaust ports for each said chamber on each said machine;

19 each said exhaust port tracking slot having an exclusive chamber plenum exterior to
20 said side plate for said exhaust port tracking slot so that during said planetary cycle of said
21 rotary machine, each said chamber can communicate gas exclusively with said exteriorly
22 located chamber plenum for each said chamber so that as each said chamber expands, gas can
23 flow into each said chamber separately from any gas flow into or out of any other chamber of
24 said machine, so that there are a total N chamber plena for intake and N chamber plena for
25 exhaust and each said chamber has an intake chamber plenum and an exhaust chamber
26 plenum;

27 gas flowing in said Stirling heat pump machine in at least one closed loop;

28 said Stirling heat pump machine having an ambient temperature heat exchanger for
29 inhibiting heating of the first of said at least two rotary machines, a regenerator for each said
30 chamber of said first of said at least two rotary machines, said chambers in each machine

1 being designated from 1 to N, each said regenerator having a first portion from which energy
2 in the form of heat or work is extracted from said gas and communicated to a second portion
3 of said regenerator, and each set of two of said at least two rotary machines having a means of
4 flow of said gas from said exhaust plenum for said chamber designated number 1 through N
5 of said first of said at least two rotary machines to an intake plenum for said chamber
6 designated 1 through N on said second of said at least two rotary machines through said first
7 portion of said regenerator for said chambers 1 through N, respectively, a means of flow of
8 said gas from said exhaust plenum of said chamber 1 through N of said second of said at least
9 two rotary machines through said second portion of said regenerator to said intake plenum for
10 said chamber 1 through N of said first of said at least two rotary machines, respectively,
11 thereby forming a series 1 through N of pairs of connected chambers in a series of gas loops
12 1 through N;

13 said each chamber of said second of said at least two rotary machines being linked to
14 said selected chamber of said first of said at least two rotary machines being selected to have
15 substantially equal volumes for the sum of volumes of each pair of connected chambers in
16 said series 1 through N at any given position of said planetary cycle of said rotary machines
17 utilizing offset planetary cycles as between said first and second of said at least two rotary
18 machines;

19 said series of gas loops 1 through N having substantially equal volumes throughout
20 said planetary cycle of said rotary machines;

21 said first of said at least two rotary machines having a rotating shaft;

22 said rotating shaft being coupled to said second of said at least two rotary machines so
23 work in said second rotary machine reduces required work to cause rotation in said first of
24 said at least two rotary machines;

25 thereby forming a series of N closed loops through chambers 1 through N for gas in
26 each said loop to pass from each said first of said at least two rotary machines, in which
27 heating is inhibited by said heat exchanger, through said first portion for each said regenerator
28 for each said loop and then through said second of said second of said at least two rotary
29 machines returning to said first rotary machine through said second portion of said

1 regenerator for each loop and thereby cooling said second of said at least two rotary
2 machines.

3 91. The Stirling heat pump machine according to claim 90, further comprising:

4 said heat sink being an ambient temperature heat exchanger for inhibiting heating of
5 said first of said at least two rotary machines.

6 92. The Stirling heat pump machine according to claim 91, further comprising:

7 said driving shaft and said rotating shaft being mechanically coupled.

8 93. The Stirling heat pump machine according to claim 92, further comprising,

9 said rotating shaft of said first of said at least two rotary machines being mechanically
10 coupled to said second of said at least two rotary machines, all of said first of said at least two
11 rotary machines being coupled by said driving shaft with gears.

12 94. The Stirling heat pump machine according to claim 93, further comprising:

13 said rotating shaft of said first of said at least two rotary machines being mechanically
14 coupled to said second of said at least two rotary machines, all of said first of said at least two
15 rotary machines being coupled by said driving shaft with cams.

16 95. The Stirling heat pump machine according to claim 92, further comprising:

17 said cams enabling variable speeds between said rotor of said first of said at least two
18 rotary machines, and said rotor of said first of said at least two rotary machines.

19 96. The Stirling heat pump machine according to claim 92, further comprising:

20 said rotating shaft of said first of said at least two rotary machines being mechanically
21 coupled to said second of said at least two rotary machines, all of said first of said at least two
22 rotary machines being coupled by said driving shaft to rotate in the same direction.

23 97. The Stirling heat pump machine according to claim 92, further comprising:

24 said tips for said at least N apices, said tips being composed of a self-lubricating
25 composite.

26 98. The Stirling heat pump machine according to claim 93, further comprising:

27 said parts of said housing in relative motion to said rotor being composed of self-
28 lubricating composite.

29 99. The Stirling heat pump machine according to 97, further comprising:

1 said tips for said at least N apices being composed of self-lubricating composite
2 selected from the group of continuous carbon fiber reinforced PEEK or PEKK.

3 100. The Stirling heat pump machine according to claim 99, further comprising:
4 said parts of said housing in relative motion to said rotor being composed of self-
5 lubricating composite selected from the group of PEEK or PEKK.

6 101. The Stirling heat pump machine according to claim 100, further comprising:
7 said gas flowing in said machine containing compounds having lubricating
8 characteristics for reducing friction in said at least two rotary machines.

9 102. The Stirling heat pump machine according to claim 92, further comprising:
10 a supplementary pressure pump in said loop to increase pressure and temperature of
11 gas after exit of gas from said first of said at least two rotary machines to enable achievement
12 of highest pressure and lowest volume and inhibit the rounding off of the pressure-volume
13 curve in said Stirling heat pump machine.

14 103. The Stirling heat pump machine according to claim 92, further comprising:
15 said exhaust ports of said first of said at least two rotary machines being truncated to
16 increase pressure and temperature of gas upon exit of gas from said second of said at least two
17 rotary machines to enable achievement of highest pressure and lowest volume and inhibit the
18 rounding off of the pressure-volume curve in said Stirling heat pump machine.

19 104. The Stirling heat pump machine according to claim 92, further comprising:
20 a mechanical means for transferring energy from said first portion of said regenerator
21 to said second portion of said regenerator, said mechanical means for transferring energy
22 being coupled by gears to said rotating shaft to rotate and thereby transfer energy in the form
23 of work at a rate consistent with the pressure times the change in temperature contemplated in
24 the pressure-volume diagram of said Stirling heat pump machine contemplated.

25 105. The Stirling heat pump machine according to claim 104, further comprising:
26 said mechanical means for transferring energy being a Brayton machine.

27 106. The Stirling cycle machine according to claim 92, further comprising:
28 a means for gas communication within each said tracking slot to enable continuous gas
29 communication within each said tracking slot.

30 107. The Stirling cycle machine according to claim 106, further comprising:

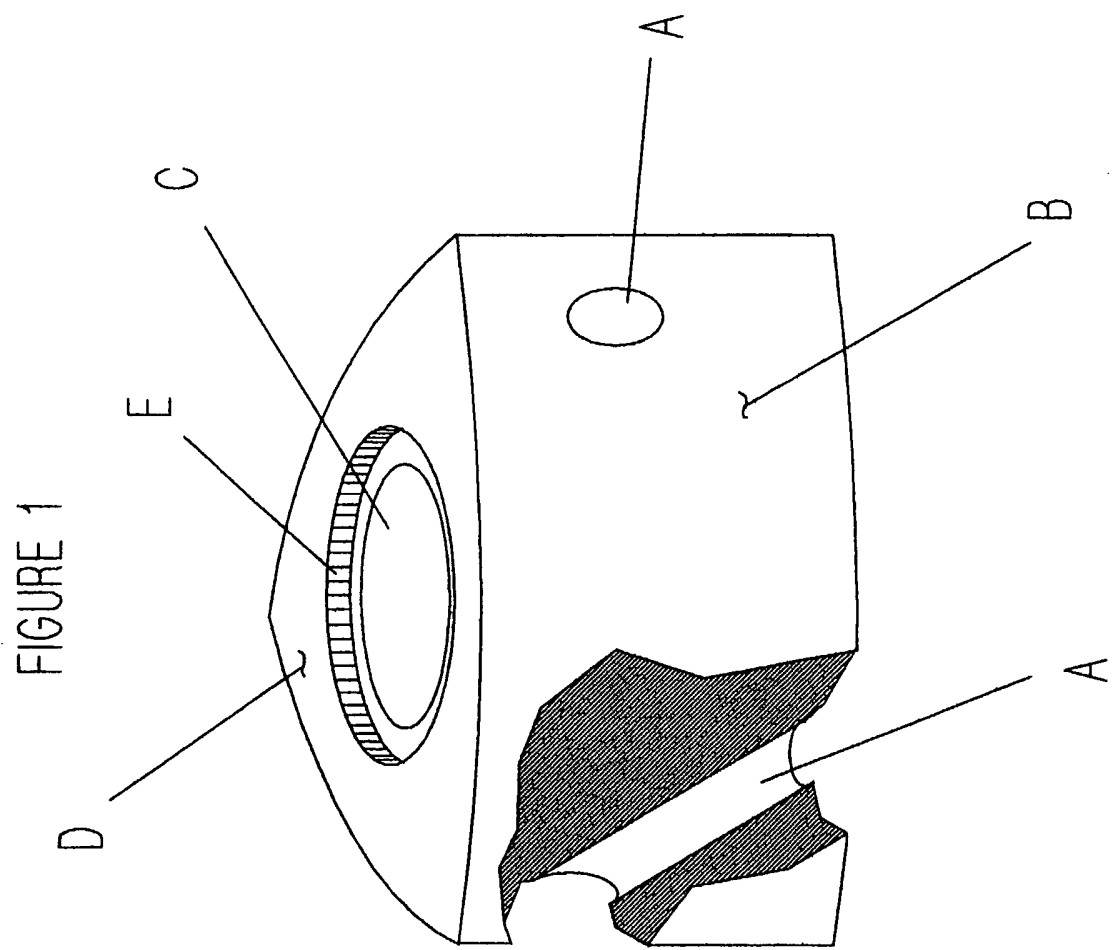
1 said means for gas communication within each said tracking slot to enable continuous
2 gas communication within each said tracking slot being a grill.

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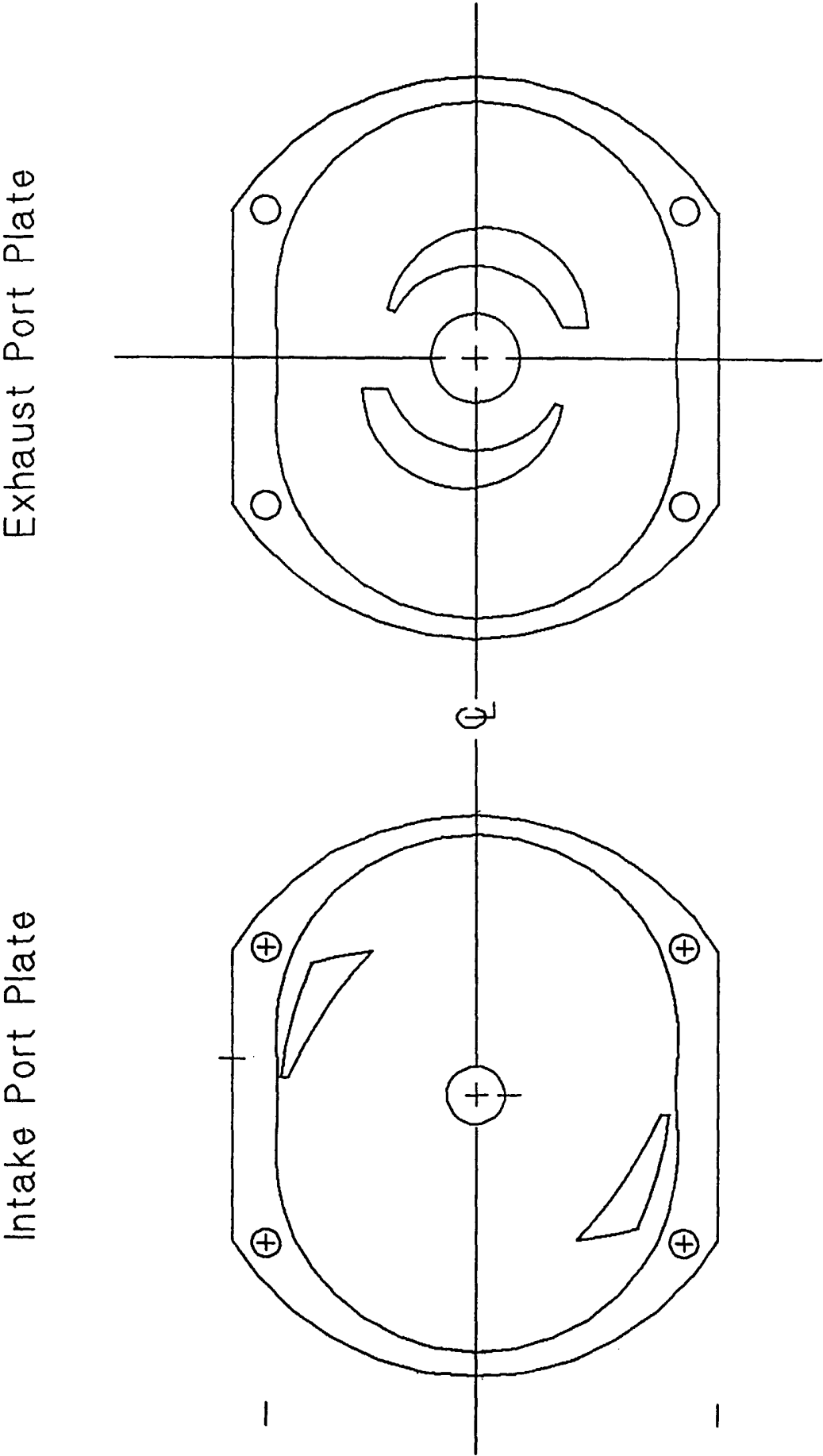
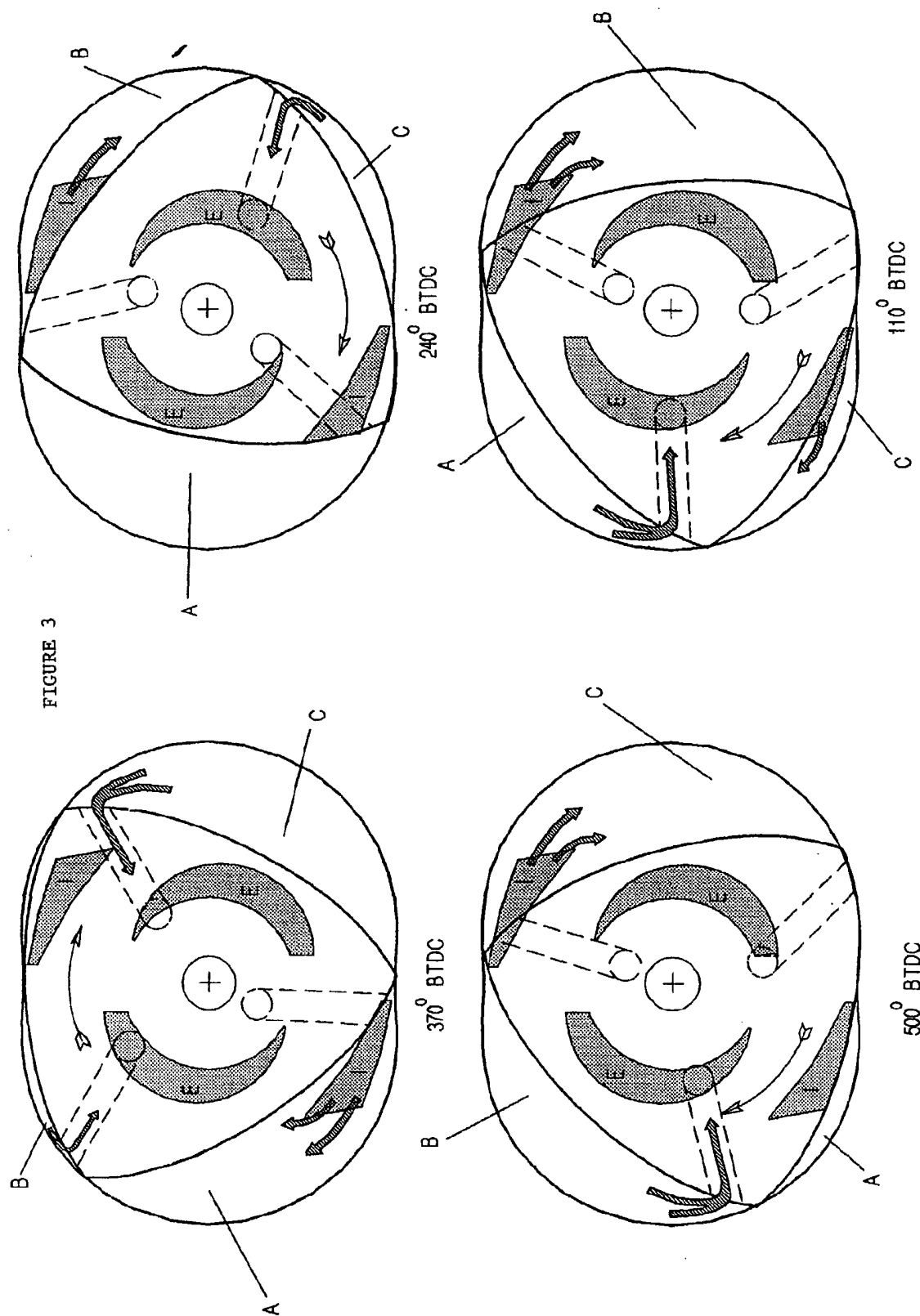


FIGURE 2



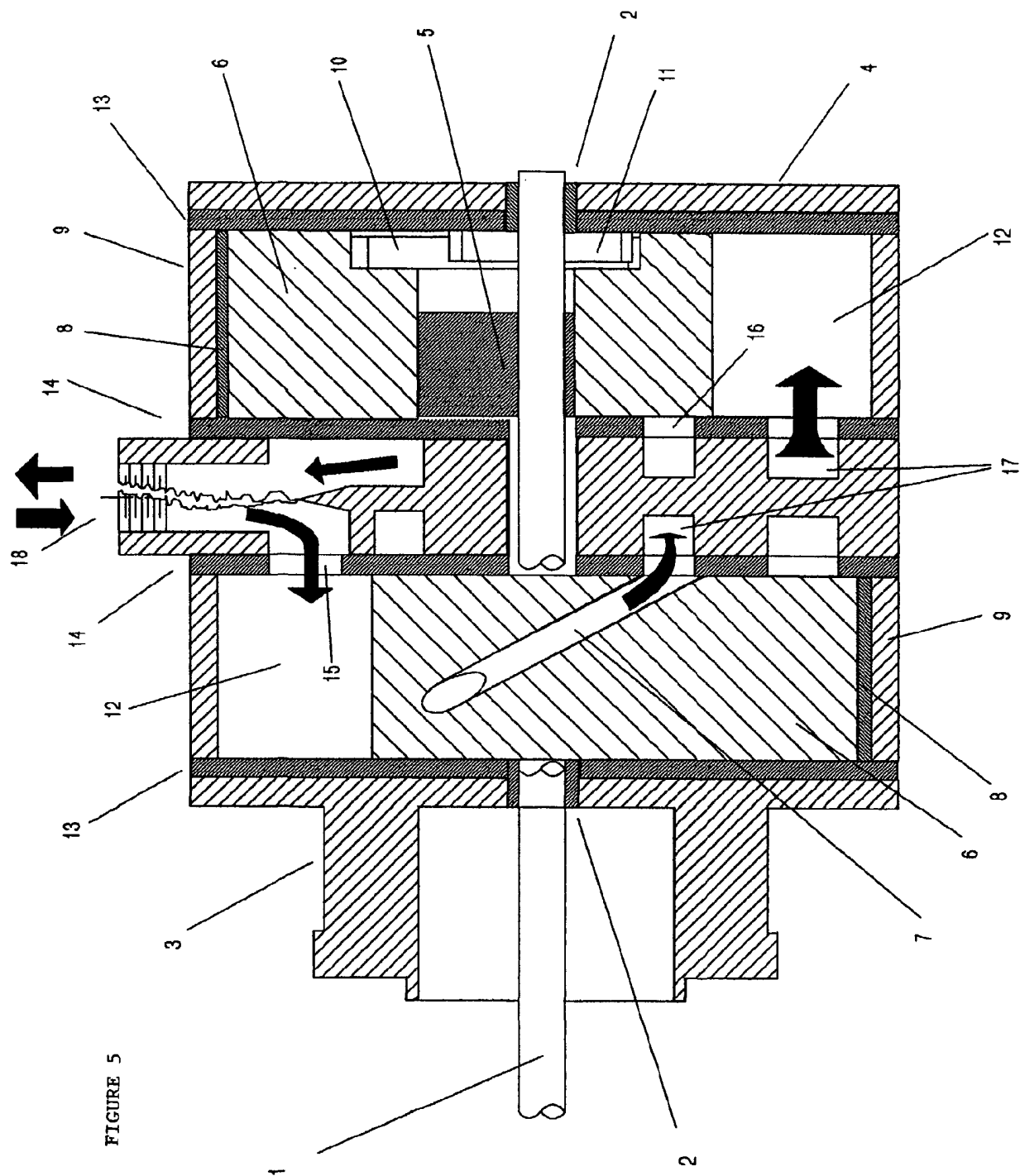


FIGURE 5

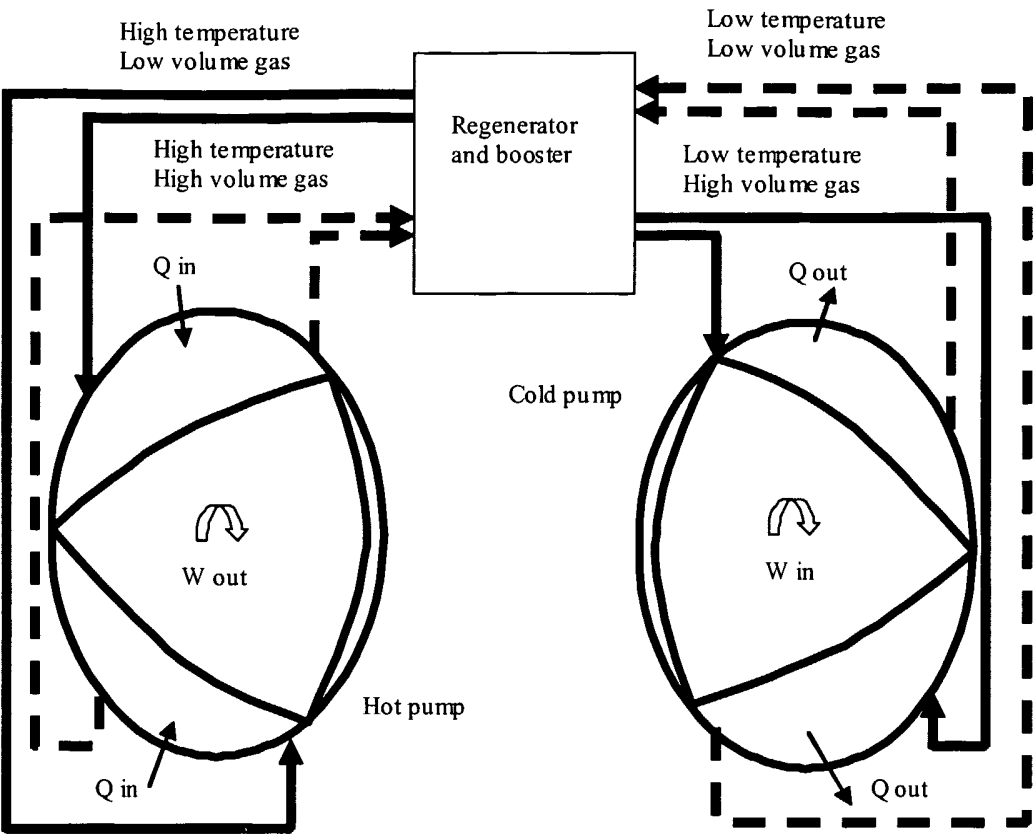
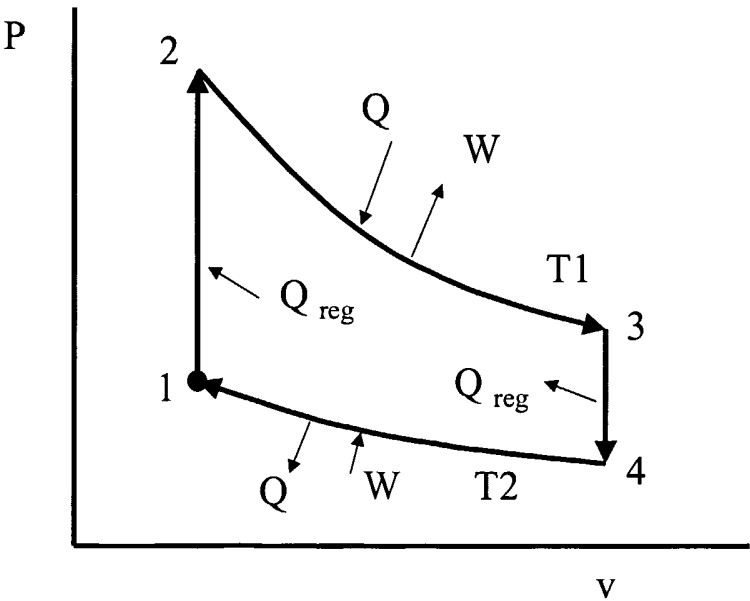


Figure 6. Two rotor Stirling Engine



Ideal Stirling Engine cycle

Figure 7. Ideal Stirling Cycle.

INTERNATIONAL SEARCH REPORT

International application No.
PCT/US02/23966

A. CLASSIFICATION OF SUBJECT MATTER

IPC(7) : F01B 29/10

US CL : 60/517, 519, 526

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

U.S. : 60/517, 519, 525, 526

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched
NONEElectronic data base consulted during the international search (name of data base and, where practicable, search terms used)
NONE**C. DOCUMENTS CONSIDERED TO BE RELEVANT**

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	US 5,211,017 A (PUSIC) 18 May 1993, see figure 1.	1-107
A	US 5,335,497 A (MACOMBER) 09 August 1994, see figure 1.	1-107
A	US 4,179,890 A (HANSON) 25 December 1979, see figure 1.	1-107



Further documents are listed in the continuation of Box C.



See patent family annex.

* Special categories of cited documents:	"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention
"A" document defining the general state of the art which is not considered to be of particular relevance	"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone
"E" earlier document published on or after the international filing date	"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art
"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)	"&" document member of the same patent family
"O" document referring to an oral disclosure, use, exhibition or other means	
"P" document published prior to the international filing date but later than the priority date claimed	

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15 NOVEMBER 2002

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