

April 27, 1965

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3,180,559

MECHANICAL VACUUM PUMP

Filed April 11, 1962

6 Sheets-Sheet 1

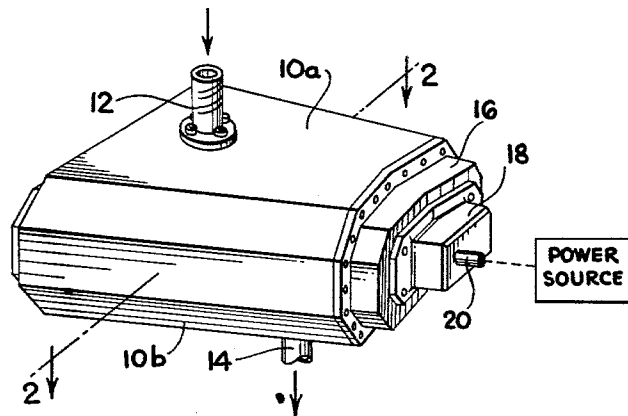


FIG. 1

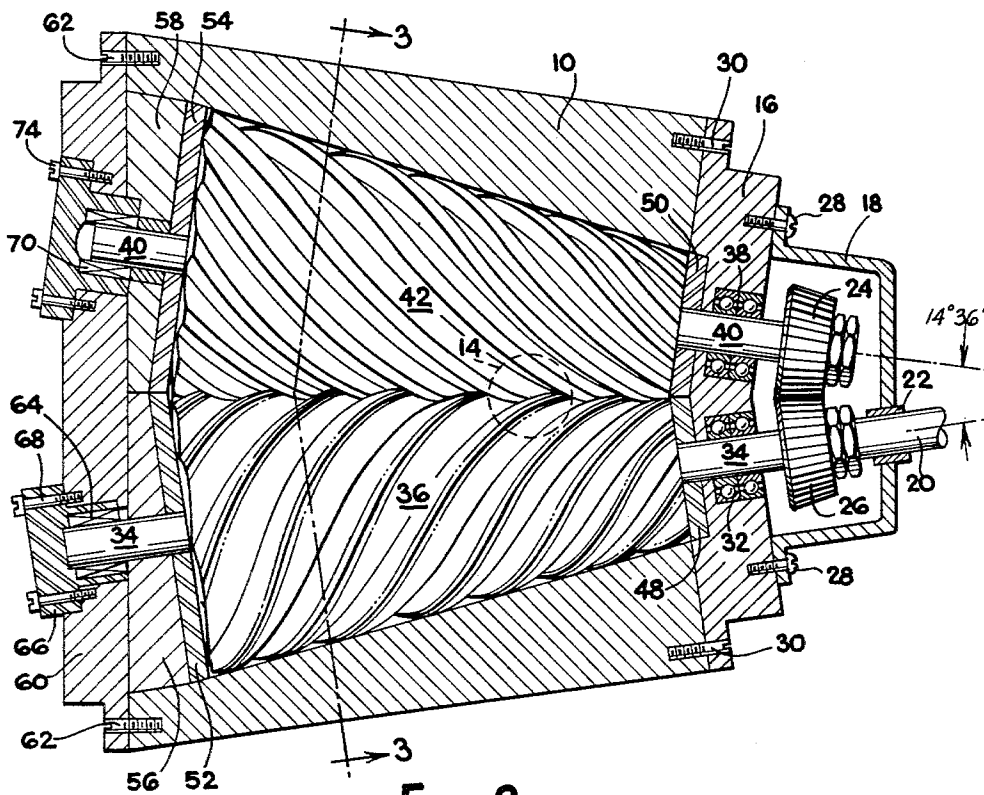


FIG. 2

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6 Sheets-Sheet 2

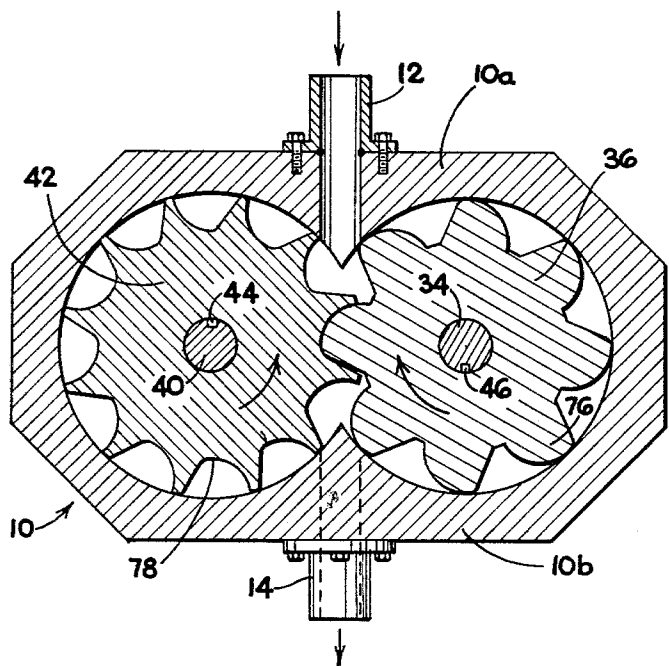


FIG. 3

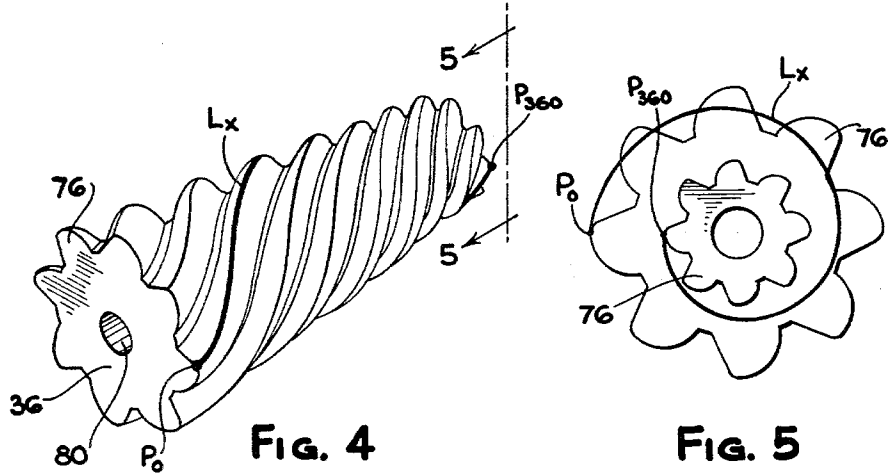


FIG. 4

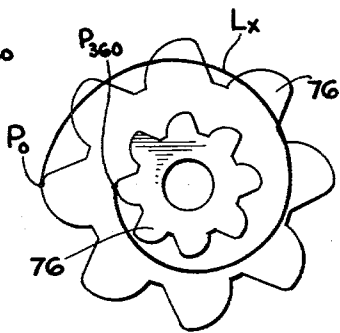


FIG. 5

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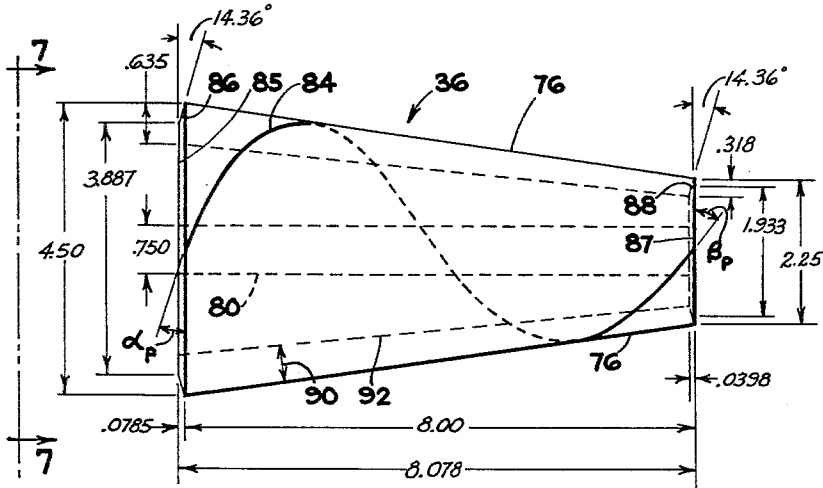


FIG. 6

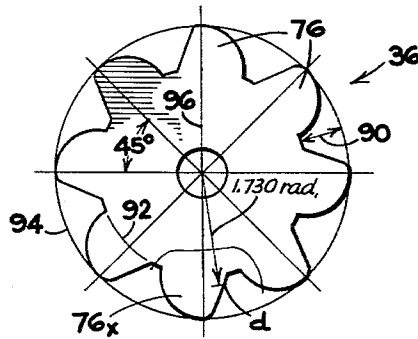


FIG. 7

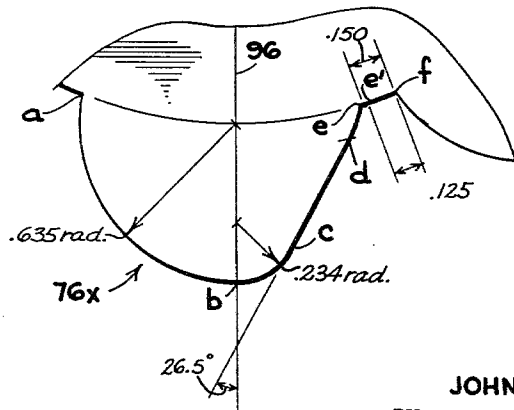


FIG. 8

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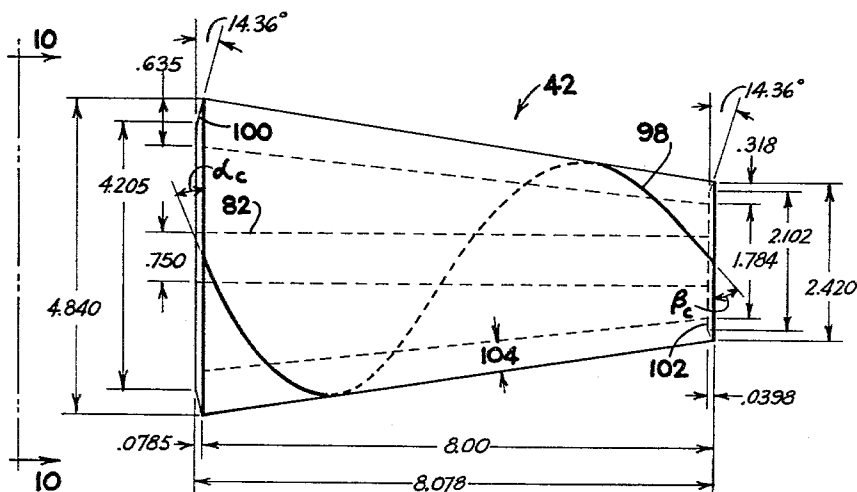


FIG. 9

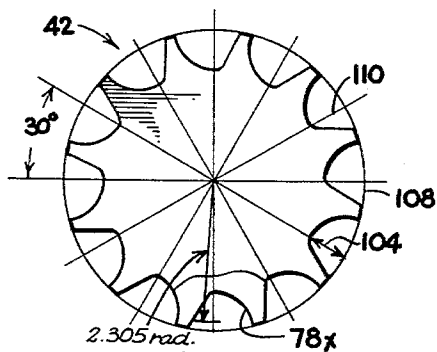


FIG. 10

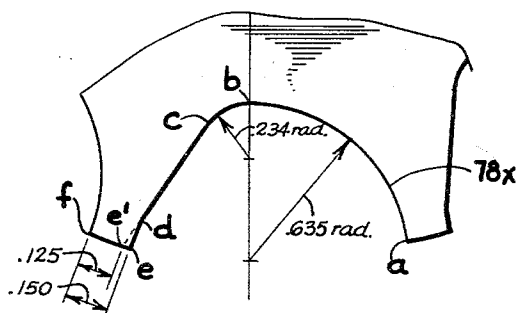


FIG. 11

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6 Sheets-Sheet 5

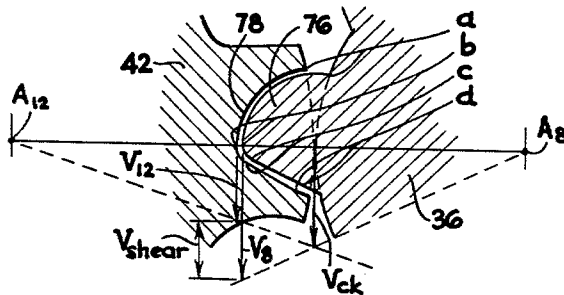


FIG. 12

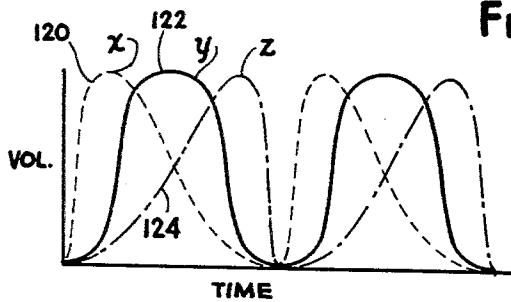


FIG. 13

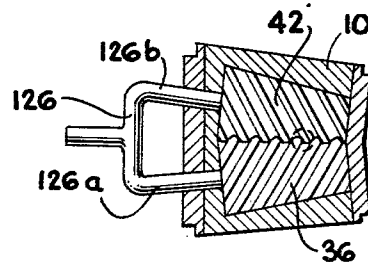


FIG. 14

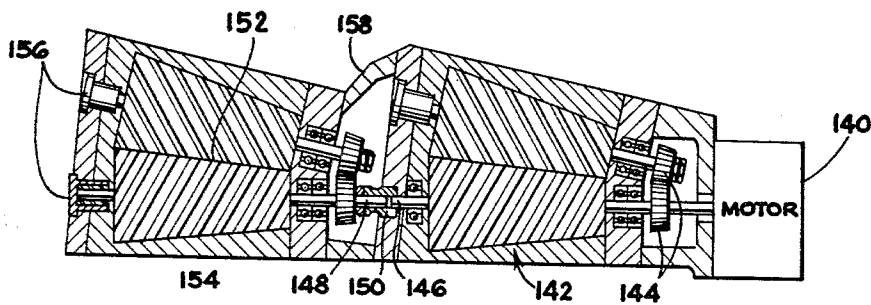


FIG. 15

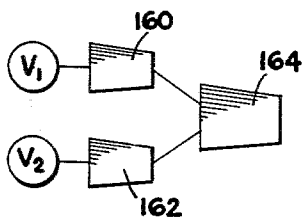


FIG. 16

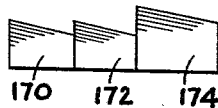


FIG. 17

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6 Sheets-Sheet 6

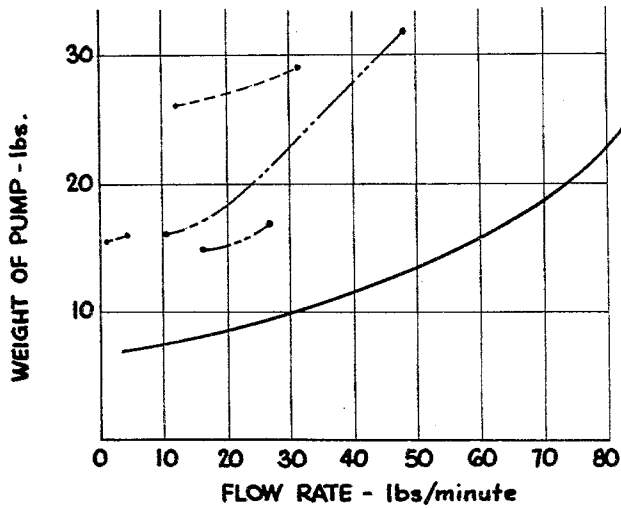


FIG. 18

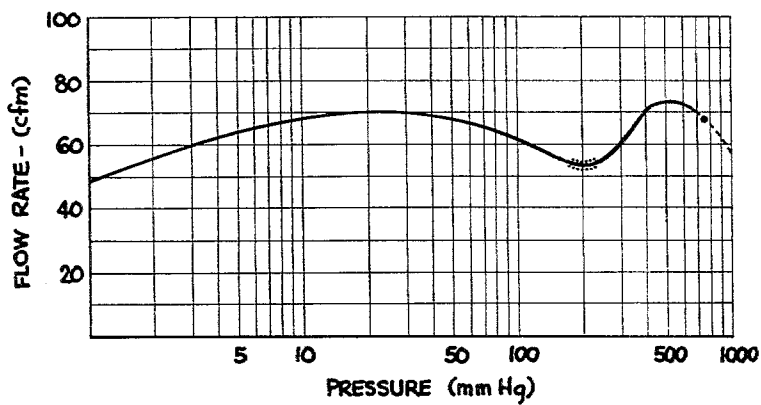


FIG. 19

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MECHANICAL VACUUM PUMP

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Filed Apr. 11, 1962, Ser. No. 187,461
30 Claims. (Cl. 230-142)

The present invention generally relates to mechanical vacuum pumps and more particularly concerns an improved vacuum pump comprising two intermeshing and synchronized rotors.

Although the invention will be described from the standpoint of its embodiment as a vacuum pump device, it will be appreciated that certain features of the apparatus are equally applicable to equivalent mechanical devices.

Vacuum pump technology has been directed towards the objective of achieving a maximum vacuum condition in a minimum amount of time and at minimum expense.

A major problem in this regard has been the necessity of creating a seal to adequately enable the creation of a maximum vacuum condition, and towards this end the prior art has shown a substantial reliance upon oil seals of one type or another. Thus, oil-diffusion pumps and oil-sealed mechanical pumps are conventional in many types of vacuum systems. It will be appreciated, however, that oil molecules even in very small concentration create very serious problems in most types of vacuum processes. Modern vacuum technology in certain areas requires a very clean vacuum, particularly for operations such as vacuum-tube processing, precision thin film deposition, particle accelerator evacuation and the like.

Vacuum pump design within the limitations to which the present invention is directed has usually been based upon the ejector principle, the piston-cylinder principle, or the eccentric rotary vane principle. More recently, the molecular pump has been employed. Each of these devices appears to be characterized by one or more problems related to oil contamination of the vacuum system, low flow rates, pulsation characteristics, shortened life, or expensive production.

More basically, these devices employ a design structure which inherently results in a relatively slow vacuum formation (excepting the ejector unit which is not appropriate for maximum vacuum conditions unless an exotic working medium such as mercury is used) whereby undesirable fluid leakage and heat transfer may occur.

One object, therefore, of the present invention is to provide an improved vacuum pump which is inherently designed to have a minimum degree of fluid leakage and heat transfer, particularly at the point in its cycle wherein it approaches the maximum vacuum condition.

Another object of the present invention is to provide an improved vacuum pump requiring no oil seals or other fluid seals of any type.

Another object of the present invention is to provide an improved vacuum pump which enables a relatively high flow rate and yet a minimum total weight for the unit, which requires no valves, and which acts to create its own molecular seal because of supersonic flow conditions attained.

Still another object of the present invention is to provide an improved vacuum pump which is particularly designed for operation in the vacuum spectrum between 10^{-4} or lower Torr and atmospheric pressure, and which may be produced at a relatively low manufacturing cost to have relatively long life with minimum maintenance.

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Still a further object of the present invention is to provide an improved vacuum pump which is characterized by compression ratios ten to twenty times greater than equivalent conventional units, and yet which may be constructed in a compact design to comply with the foregoing objects.

These and other objects of the present invention are generally achieved by providing an improved fluid vacuum pump comprising a casing having an inlet thereto and an outlet therefrom for drawing in and ejecting the medium being pumped, respectively.

A piston rotor and a cylinder rotor are mounted for rotation within the casing, and the rotors are designed to have intermeshing lobes and grooves co-functioning with each other and with the casing to effect ejection of the fluid.

The terms "piston" and "cylinder" are used throughout the specification and claims in describing the lobed rotor and grooved rotors, respectively, since these terms indicate functionally the action of these respective rotors.

As an important feature of the invention, each of the lobes and the grooves is characterized, respectively, by a radially extending mating flat surface designed to effect maximum displacement of fluid with a minimum rotational movement of the rotors.

In addition, some type of driving means for effecting rotation of the rotors is provided.

A better understanding of the present invention will be had by reference to the drawings, disclosing merely one illustrative embodiment, and in which:

FIGURE 1 is a perspective view of the improved vacuum pump, according to the present invention, disclosing in part an inlet and outlet therefrom for passage of fluid therethrough and also disclosing schematically the connection to a source of power;

FIGURE 2 is a cross section through the vacuum pump of FIGURE 1 taken in the direction of the arrows 2-2;

FIGURE 3 is a cross-sectional view taken in the direction of the arrows 3-3 of FIGURE 2 disclosing the intermeshing of the rotors and the disposition of same within the casing;

FIGURE 4 is a perspective view of the piston rotor shown in FIGURES 2 and 3 on a somewhat different scale;

FIGURE 5 is a view taken in the direction of the arrows 5-5 of FIGURE 4 indicating the spiral of the lobes of the piston rotor;

FIGURE 6 is a side elevation of the piston rotor shown in FIGURES 2, 3, and 4 also disclosing the helical nature of the lobes formed thereon;

FIGURE 7 is a view taken in the direction of the arrows 7-7 of the piston rotor of FIGURE 6 disclosing certain geometrical relationships embodied in the piston rotor, including the left-hand helical spiral;

FIGURE 8 is an enlarged detailed view in cross section of one of the lobes of the piston rotor of FIGURE 5;

FIGURE 9 is a side elevation of the cylinder rotor shown in FIGURES 2 and 3 disclosing the right or left hand helical-spiral form of the cylindrical grooves embodied therein;

FIGURE 10 is a view of the rotor of FIGURE 9 taken in the direction of the arrows 10-10 disclosing in more detail the geometrical relationships of the grooves formed therein;

FIGURE 11 is an enlarged cross-sectional view of one of the grooves of the cylinder rotor shown in FIGURE 10;

FIGURE 12 is an enlarged cross-section of an intermeshing lobe and groove of the piston and cylinder rotors illustrating velocity vector relationships;

FIGURE 13 is a time versus volume chart indicating degree of volume change per unit time rate;

FIGURE 14 is a schematic representation of the vacuum pump of the present design illustrating a preferred manner of drawing fluid into the unit;

FIGURE 15 is a schematic representation of the vacuum pump of the present invention embodied in a two-stage arrangement;

FIGURE 16 schematically indicates two pumps in parallel connected to one pump in series, each embodying the present invention;

FIGURE 17 schematically indicates three stages in series combined;

FIGURE 18 is a flow rate versus weight of pump diagram disclosing advantages of the present construction in relationship to these factors; and,

FIGURE 19 is a vacuum achieved versus flow rate curve indicating the degree of vacuum attained for any given flow rate.

Referring now to the drawings, there is shown in FIGURE 1 an improved vacuum pump, according to the present invention, embodying a tapered casing 10 of generally rectangular cross section (see FIGURE 3). The casing 10 has connected thereto and communicating with the interior thereof an inlet 12 and an outlet 14. The inlet 12 is shown in FIGURE 1 in an alternative position for illustrative purposes, the preferred construction being shown in FIGURE 14 to be described hereafter. The inlet 12 is designed to receive fluid from the space or chamber being evacuated, said fluid being drawn into the casing 10 and then expelled or ejected through the outlet 14 after pumping action has occurred.

Preferably, the casing 10 has coupled thereto at its right hand end (as viewed in FIGURE 1) an end bracket 16 which in turn is partially enclosed by a gear housing 18 through which extends an input shaft 20. The shaft 20 is designed to be coupled to any type of suitable driving mechanism or source of power as schematically indicated, although an electrical motor is preferred.

The internal construction of the vacuum pump may be more clearly understood by reference to FIGURE 2. The shaft 20 is journaled in a bearing 22 in the gear housing 18 and connects in a conventional manner with bevel gears 24 and 26. The bevel gears are in the ratio of 2 to 3 with the driving gear or gear 26 having in one embodiment thirty-eight teeth and the driven gear fifty-seven teeth.

The gear housing may be connected with bolts or screws 28 into tapped holes in the end bracket 16. Similarly, the end bracket 16 as such may be connected with Allen screws or the like 30 into tapped holes in the main casing 10.

Disposed within the end bracket 16 is a duplex bearing 32 journaled a shaft 34 which may be integral with shaft 20 or which may be coupled in some manner thereto. The shaft 34 passes through the axial center of and is designed to drive the piston rotor 36. Similarly, a duplex bearing 38 is also provided within the end bracket 16 within which a shaft 40 is journaled, the latter being designed to drive a cylinder rotor 42. The gear 24 is, of course, coupled to the shaft 40 as is the gear 26 coupled to the shaft 34 such that the input shaft 20 will effect rotation of the rotors 36 and 42 in opposite directions.

In one form, the shaft 40 may be keyed at 44 to the cylinder rotor 42, while the shaft 34 is keyed at 46 to the piston rotor 36, as clearly shown in the view of FIGURE 3.

In order to maintain the close tolerances necessary for the high degree of vacuum desired, a disc type and seal 48 is provided in axial interposition between the rotor 36 and a recess in the bracket 16; similarly, another sealing disc 30 is axially interposed between the end bracket

16 and the cylinder rotor 42. These discs are preferably of a semi-metallic composition so as to have the characteristics of long wear and low friction, thereby suitably sealing off the end faces of the rotors 36 and 42 while at the same time not creating any substantial frictional force upon the rotors 36 and 42. Similar disc-like seals of the same composition may be provided at the opposite ends of the rotors 36 and 42 as indicated by the numerals 52 and 54.

It will be evident from the view of FIGURE 2 that the rotors 36 and 42 have their axes disposed at an angle one to the other such that they are not in parallel alignment. Thus, in the embodiment shown and described, the angle approximates $14^{\circ}36'$ although variations of plus or minus seven degrees are believed feasible. From the standpoint of manufacturing and precision construction, it is desirable that some means be employed to positively index the rotors in position within the casing 10. Inserts 56 and 58 are provided for this purpose. Inserts 56 and 58 are designed to have their inner sides positioned adjacent the seals 52 and 54 with their outer sides being against the end bracket 60 on the left hand end of the unit as viewed in FIGURE 2. The inner faces of inserts 56 and 58 are canted with respect to their outer faces at precisely the same angle as the axes of the rotors 36 and 42 diverge from each other such that precise positioning of these rotors may be effected to obtain optimum vacuum conditions as will be hereafter described.

The end bracket 60, as such, may be fastened with screws or bolts 62 to the casing 10.

Disposed within the end bracket 60 is a bearing 64 having an enclosing bearing cap 66 provided with fastening disc 50 is axially interposed between the end bracket to be received within bearing 64.

Similarly, the left hand end of shaft 40 is received within bearing 70 which is enclosed with a bearing cap 72 secured with screws or bolts 74. Conventional bearings suitable for the particular design of the unit may be employed.

It will be noted from the view of FIGURE 1 that two sides of the casing 10, sides 10a and 10b are parallel to the axes of the rotors 36 and 42. On the other hand, as clearly shown in FIGURE 2, the other sidewalls of the casing 10 are designed to be tapered in accordance with the angle defined between the axes of the rotors 36 and 42 which in turn follows from the taper characterizing the rotors as such. Thus, the sidewalls 10a and 10b are tapered in a direction parallel to the axes of the enclosed rotors, respectively. As a consequence of this construction, gauging and precision construction during manufacture can be simplified. In this regard, it is of interest to note that the precision gauging of the pump requires free running clearances of approximately 0.003 inch.

A more detailed explanation of the construction of the piston and cylinder rotors 36 and 42, respectively, may now be had by reference to FIGURES 3-10.

In the cross section taken in FIGURE 3, it will be seen that the piston rotor 36 is provided with eight lobes or semi-pistons while the cylinder rotor 42 is provided with twelve grooves designed to mate with the lobes of the piston rotor 36. The lobes of the piston rotor 36 have been designated generally by the numeral 76 while the grooves of the cylinder rotor 42 have been generally designated by the numeral 78.

As heretofore mentioned, the bevel gears 26 and 24 are in a two to three ratio; similarly, as heretofore mentioned, the lobes 76 are in a two to three ratio with the grooves 78, i.e., 8:12. In consequence, the diameter on the pitch line of the piston rotor 36 is in the ratio of two to three with the diameter on the pitch line of the cylinder rotor 42. Although other ratios may be employed, it has been found that this particular ratio is advantageous for maximum vacuum efficiency insofar as the design of the illustrative unit is concerned.

As will be evident from the view of FIGURE 4, the piston rotor 36 is provided with lobes 76, as heretofore

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identified, which twist to the left in the form of a spiral-helix or screw (see also FIGURE 5) to complete one turn of 360 degrees from the larger diameter end of the rotor to the smaller diameter end, as indicated by lobe line L_x and points P_0 and P_{360} . In the view of FIGURE 4, the shaft is not shown in the unit, whereby the internal axial bore 80 is indicated. The cylinder rotor is similarly provided with a bore 82, as shown in FIGURE 9. The cylinder rotor 42 (not shown in perspective) is also provided with grooves 78 which complete approximately one 240 degree twist (in the same ratio of 2:3 heretofore discussed) throughout the length of the rotor; however, in the case of the cylinder rotor, the twist is to the right or in a clockwise direction as viewed from the large end of the rotor. These twists are better shown and described in conjunction with FIGURES 6 and 9.

As shown in FIGURE 6 and heretofore mentioned, the piston rotor 36 is provided with lobes forming a left-handed 360 degree twist as indicated schematically by the line 84. In actual construction, it is also desirable to form the larger end 85 of the rotor with beveled lobe edges 86, the bevel being at an angle of $14^\circ 36'$ with respect to a plane normal to the axis of the rotor 36. This bevel is provided for manufacturing purposes and to obtain maximum vacuum conditions.

Similarly, the small end 87 of the rotor is provided with an angled or beveled indentation or recess 88 for the same purpose. It will be noted that the depth of the groove or lobe is measured from the pitch circle or point at which the angled edge merges into the rectilinear end surface (85 or 87) to the outer periphery of the unit as indicated by the numeral 90. Thus, the beveled edge 86 at the smaller end and the beveled edge 88 at the larger end are merely indicative of the ends of the lobes 76.

These beveled lobe ends 86 and 88 assist in approximating a spherical surface to in turn maximize the vacuum action of the unit.

Although precise dimensions do not appear to be critical, dimensions have been indicated in FIGURES 6-11 of one experimental model built. In this regard, it should be noted (as heretofore mentioned) that the preferred angle between the axes of the piston and cylinder rotors is $14^\circ 36'$ or the same angle as the bevel 86 and 88. The angle between the axis of the cylinder rotor 42 and the center line of the unit is somewhat greater than the angle between the axis of the piston rotor 36 and the same center line.

It is to be noted that the angle $\alpha_p < \beta_p$ or that the lobes become more and more parallel to the axis towards the smaller end of the piston rotor 76. As a consequence of this construction, once a given force is exerted on the particular fluid medium, the resistance to movement thereof is gradually decreased so that added acceleration is given to the fluid as it reaches the outlet 14. In the illustrative embodiment, the difference between α_p and β_p is approximately 40° although the angle is not believed critical.

Referring now to the view of FIGURE 7, it will be seen that the pitch circle is defined by the inner points of the lobes 76, as indicated at 92. In order to form the lobes 76, the outer circular periphery 94 of the rotor 36 is divided into 45 degree sections or lines 96. Thus, looking at the view of FIGURE 8, the line 96 at its point of intersection with the pitch circle 92 defines a point forming a radius to form that part of the lobe extending from the points a to b . This radius in the particular construction illustrated is .635 inch. Thereafter, a radius is used of .234 inch, said radius being measured along the 45 degree line 96 radially inwardly and being employed to form the portion $b-c$ of the lobe 76X. Thereafter, between the points $c-d$, a straight line is used. This straight line or rectilinear portion 76X forms the flat portion of the lobe which functions very importantly in the ultimate degree of vacuum produced for the work expended. The point d is found by a radius of 1.730 inches (see FIGURE 7) from the axis of the unit being drawn to connect

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with the surface formed by the .234 radius between points b and c . In one form of the invention, the flat or rectilinear portion $c-d$ would continue to e' , the latter connecting with f on the pitch circle 92. In a preferred form of the invention, however, the pitch circle portion extends between $e-f$ or .025 inch further such that the portion $d-e$ is slightly angled with respect to the portion $c-d$.

It will thus be seen that the lobe 76X is formed predominantly of a radiused portion $a-b$ and a flat portion $c-d$ with connecting portions $b-c$, $d-e$, and $e-f$. It is believed from experiments that the flat portion $c-d$ is the portion of the lobe that lends the greatest novelty to the present invention insofar as the intermeshing construction of the rotor lobes and grooves are concerned. Test runs indicate that as this flat portion mates with the congruous flat portion of the cylinder rotor groove that the maximum compression of the medium occurs.

The construction of the cylinder rotor 42 and the grooves thereof may now be described by reference to FIGURES 9-11. It will be found that most of the elements are analogous for cofunctioning operation.

Thus, looking at the view of FIGURE 9, it will be seen that the cylinder rotor 42 is provided with an approximate 240 degree twist to its grooves, as indicated schematically by the numeral 98. In this instance, as previously stated, the twist 98 extends radially towards the right or clockwise as viewed from the larger end of the cylinder rotor 42. Also, the cylinder rotor is provided with a beveled edge 100 at its larger end and a recessed beveled edge 102 at its smaller end, these edges forming the marginal edges of the grooves 78 and functioning similarly to lobe ends 86 and 88. Thus, the groove depth may be indicated by the numeral 104. It will also be noted that $\alpha_c < \beta_c$ analogously to α_p being less than β_p as heretofore mentioned in connection with the piston rotor.

Referring to the view of FIGURE 10, it will be seen that the cylinder rotor 42 is formed in a somewhat similar manner as the piston rotor 36. In this instance, the pitch circle is indicated at 108, while the 30 degree lines are indicated at 110. The same radii are employed between similarly designated points. Thus, as seen in FIGURE 10, the portion $a-b$ is formed by a radius taken from the 30 degree line 110 of .635 inch. Thereafter, a radius of .234 inch from the point b along the same line 110 of 30 degree line is used to form the surface $b-c$ and identical measurements as heretofore described in conjunction with FIGURE 8 are used to form the portion $c-d$, $d-e$, $e'-f$ and $e-f$.

Thus, the groove 78X which is exemplary of all the grooves 78 of the cylinder rotor 42 is also formed with a flat portion $c-d$ functioning to create the high vacuum area with its mating flat portion $c-d$ of the lobe 76X. Again, the groove 78X is formed with a large radiused portion $a-b$ which mates with the portion $a-b$ of the lobe 76X. Other connecting portions are similar.

It has been found with this type of construction that the mating lobes and grooves tend to create a superharmonic motion or cycle in the ejection and drawing induction of the fluid medium. It is believed that this action occurs as a consequence of the face-to-face relationship of the flat portion $c-d$ of the lobes 76 and grooves 78 whereby a very high nearly complete fast vacuum condition is effected along a center line of the apparatus as contrasting the four link slider action of a piston-cylinder, for example, which creates a vacuum very slowly. Furthermore, it is also of interest to note that this flat-to-flat arrangement simplifies the manufacture by enabling more simple gauging of the spiral angle of twist.

As a consequence of the 2:3 ratio characterizing the diameters, number of lobes versus grooves, and degree of twist of the piston-cylinder rotor combination, an advantageous geometrical relationship occurs which tends to assist in the displacement of fluid media and the accompanying vacuum action.

Thus, in FIGURE 12, there is shown one of the lobes

76 of piston rotor 36 intermeshing with a groove 78 of cylinder rotor 42. The axis of the cylinder rotor 42 is indicated by point A_{12} and the axis of the piston rotor 36 by point A_8 . If lines are drawn from points A_{12} and A_8 to the end of a vector V_{ek} (indicating the kinetic velocity common to both rotors 36 and 42 on the engaging pitch line thereof), a relative indication of the velocity of tip of lobe 76, as shown by V_8 , may be had as against the velocity of the inner surface of the groove 78, as shown by V_{12} . Thus, V_8 represents the vector drawn to intersect the line from A_8 , while V_{12} represents the vector drawn to intersect the line from A_{12} . The difference or $V_8 - V_{12}$ represents the shear velocity V_{shear} or a molecular drag action tending to increase the vacuum characteristics of the unit.

Although applicant does not desire to be bound by the theory explained herein and set forth in the illustrative charts, reference may be made to FIGURE 13 for a further explanation of the type of super-harmonic motion created by applicant's novel rotor device. In this view, the numeral 120 designates a curve indicating applicant's approximate variation in volume measured against time. In this curve, the portion $a-b$ may represent the intermeshing radiused portion and the portion $c-d$ may represent the flat-to-flat intermeshing. It will be seen that the flat portion $c-d$ rises much more closely to the vertical than does the equivalent portion of the curves 122 and 124. Curve z designated by numeral 124 may indicate partial super-harmonic motion, while the curve y designated by numeral 122 may indicate simple harmonic motion such as would characterize a piston-cylinder arrangement of a vacuum pump. Thus, with applicant's improved vacuum pump, a very quick increase in volume occurs (see FIGURE 13) during the flat-to-flat intermeshing of portion $c-d$ with a gradual breaking away during the radiused mating portions $a-b$ of grooves 76 and lobes 78 of the respective rotors.

Of course, the general principle of screw types of rotors for compression is well known in the art, and applicant makes no claim to same as such.

In the use of the improved vacuum pump, it is desirable that the inlet 12, as indicated in FIGURE 1, preferably be formed as a Y-shaped inlet 126, as indicated in FIGURE 14. Preferably, the two legs 126a and 126b of the Y-shaped inlet should be formed so as to be aligned with or open into the interior of the casing 10 with a particular lobe and groove intermeshing with each other approximately one-sixth to one-half the length of the rotors measured from the inlet end of the rotors or the large ends thereof. A preferred point is approximately one-third the length.

The main factor in this regard is to have the inlet open into the center line of the apparatus at a point at which a substantially maximum vacuum is attained. Thus, it is not desirable to have the inlet open directly into the end of the unit since maximum vacuums are not attained in that portion of the pump. Furthermore, it is not preferred to have the inlet come into the upper part of the unit, as shown in FIGURE 1, since the possibility of back pressure exists as determined experimentally although such inlet could connect with the same rotor-groove combination.

The schematic illustration of FIGURE 15 has merely been provided to indicate that the vacuum pump of the present invention may be readily produced to be employed in multistage applications. Thus, looking at the view of FIGURE 15, there is schematically indicated a motor 140 designed to drive generally designated vacuum pump 142 which may be provided with bearings 144. The vacuum pump 142 may be of the construction according to the present invention.

The vacuum pump 142 has a shaft 146 extending therefrom to appropriate bearing means to be coupled to a shaft 148 through a high speed coupling 150.

The shaft 148 in turn drives another vacuum pump 152

which may be provided with bearings 154 and 156 as indicated. A center casing 158 may serve to enclose the high speed coupling 150 and maintain the vacuum conditions therein.

It will be appreciated that a similar construction only further extended may be employed for further stages if desired. Thus, FIGURE 16 illustrates diagrammatically two pumps 160 and 162 connected in parallel to a third pump 164 in series. Designations V_1 and V_2 merely indicate two separate sources of media or fluids. FIGURE 17 illustrates schematically three stages in series, as by pumps 170, 172, and 174. Construction details may be similar to those disclosed in FIGURE 15.

FIGURES 18 and 19 are merely included to give further indication of the characteristics of one experimental model of the improved vacuum pump constructed in accordance with the present invention. It is, of course, expected that improved results will be obtained as further refinements are made.

It is believed that the operation of the improved vacuum pump, according to the present invention, has been clearly described in connection with the foregoing specification. Instrumentation indicates that the flow velocity is such that supersonic speeds are attained whereby the fluid or gas tends to create its own seal with the interior of the casing. In consequence, the oil seals associated with conventional apparatus are eliminated although the unit is still capable of achieving high vacuum conditions. Pressure ratios have already been measured in the experimental prototype built of one to twenty thousand at the inlet 12.

At the present time, it cannot be precisely stated the extent to which the various dimensions may be varied, particularly with respect to the shape of the lobes 76 and the grooves 78. However, it can be stated that it is preferred to have the diameter of the cylinder rotor greater than the diameter of the piston rotor and the cylinder rotor to have a greater number of grooves than the piston rotor has lobes. In addition, it is believed essential to the present invention that the mating lobes 76 and grooves 78 have a flat-to-flat intermeshing portion.

Although the precise location of the fluid inlet may be varied as heretofore described, its position at approximately one-third the length of the unit measured from the larger end thereof is desirable. The precise positioning of the outlet may be varied. Also, as mentioned, the included angle between the axes of the grooved rotor and lobed rotor may possibly be varied to a certain extent, but it is preferred that it be within plus or minus 7 degrees of the angle shown.

The pump as illustratively described has—as heretofore indicated—been constructed and operated. The rotors have been turned as fast as 25,000 r.p.m. with satisfactory results. It will be noted that the tapered rotors in combination with the duplex bearings yield a configuration characterized by a relatively low torsional resonant frequency. In this regard, it will be appreciated that the center of gravity of each rotor is considerably closer to the larger end.

It will be appreciated that other portions of the improved vacuum unit, such as the driving means, the gearings, the bearings, the inserts, the seals, and other components which may be associated with the final assembly may be varied in construction and dimensions. The dimensions shown herein are therefore not to be thought of as limiting, but merely illustrative for the purpose of giving a complete teaching of the invention in one of its embodiments.

It will be evident to those skilled in the art that various modifications and changes may be made in certain portions of the improved vacuum unit without departing from the spirit and scope of the invention thereof; such changes and modifications are deemed to fall within the scope of the following claims.

What is claimed is:

1. An improved fluid vacuum pump comprising:
 - (a) a casing;
 - (b) an inlet to said casing for said fluid;
 - (c) an outlet from said casing for said fluid;
 - (d) a piston rotor and a cylinder rotor mounted for rotation within said casing;
 - (e) said piston and cylinder rotors having intermeshing spiraling lobes and grooves, respectively co-functioning with each other and with said casing to effect induction and ejection of said fluid;
 - (f) each of said lobes and said grooves being characterized, respectively, by a radially extending mating rectilinear flat surface designed to effect maximum displacement of fluid with a minimum movement of said rotors; and,
 - (g) means for synchronized driving of said rotors.
2. An improved fluid vacuum pump, according to claim 1, in which said lobes and grooves, respectively, define a helix on the periphery of said piston rotor and cylinder rotor.
3. An improved fluid vacuum pump, according to claim 1, in which said rotors are each tapered in the same direction and are of frusto-conical shape such that the axes thereof are non-parallel and converge to define an angle therebetween.
4. An improved fluid vacuum pump, according to claim 3, in which the overall diameter of each of said rotors decreases in a direction from said inlet to said outlet.
5. An improved fluid vacuum pump, according to claim 1, in which:
 - (a) said lobes and grooves, respectively, define a helix on the periphery of said piston rotor and cylinder rotor; and,
 - (b) in which said rotors are each tapered in the same direction and are of frusto-conical shape such that the axes thereof are non-parallel to define an angle therebetween.
6. An improved fluid vacuum pump, comprising:
 - (a) a casing;
 - (b) an inlet to said casing for said fluid;
 - (c) an outlet from said casing for said fluid;
 - (d) a piston rotor and a cylinder rotor mounted for rotation within said casing, said rotors being designed to, respectively, have intermeshing lobes and grooves co-functioning with each other and with said casing to effect induction and ejection of said fluid, said piston rotor being characterized by a smaller overall diameter than said cylinder rotor at any given axial position, and defining fewer lobes than said cylinder rotor defines mating grooves;
 - (e) each of said lobes and said grooves being characterized, respectively, by a radially extending mating rectilinear flat surface designed to effect maximum displacement of fluid with a minimum rotation of said rotors; and,
 - (f) means for synchronized driving of said rotors.
7. An improved fluid vacuum pump, according to claim 6, in which said rotors are each tapered in the same direction and are of frusto-conical shape such that the axes thereof are non-parallel and converge to define an angle therebetween.
8. An improved fluid vacuum pump comprising:
 - (a) a casing;
 - (b) an inlet to said casing for said fluid;
 - (c) an outlet from said casing for said fluid;
 - (d) a piston rotor and a cylinder rotor mounted for synchronized rotation within said casing, said rotors being designed to, respectively, have inter-meshing helically formed lobes and grooves co-functioning with each other and with said casing to effect induction and ejection of said fluid, each of said lobes and said grooves being characterized, respectively, by a radially extending mating rectilinear flat surface

- designed to effect maximum displacement of fluid with a minimum movement of said rotors; and,
- (e) each of said rotors being tapered in the same direction so as to be of frusto-conical shape whereby the axes thereof are non-parallel and converge to define an angle therebetween; and,
- (f) means for driving said rotors.
9. An improved fluid vacuum pump, according to claim 8, in which said lobes characterizing said piston rotor effect approximately a 360 degree twist from the large end of said piston rotor to the small end of said piston rotor; and, in which said grooves characterizing said cylinder rotor effect approximately a 240 degree twist from the large end of said cylinder rotor to the small end thereof.
10. An improved fluid vacuum pump, according to claim 8, in which said inlet is located so as to communicate with a center line lobe-groove combination approximately one-sixth to one-half the axial length of said rotors measured from the large ends thereof.
11. An improved fluid vacuum pump comprising:
 - (a) a casing;
 - (b) an inlet to said casing for said fluid;
 - (c) an outlet from said casing for said fluid;
 - (d) a piston rotor and a cylinder rotor mounted for synchronized, intermeshing rotation within said casing;
 - (e) said rotors being designed to have helically formed lobes and grooves co-functioning with each other and with said casing to effect induction and ejection of said fluid;
 - (f) each of said lobes and said grooves being characterized respectively by a radially extending mating rectilinear flat surface designed to effect maximum displacement of said fluid with a minimum movement of said rotors;
 - (g) each of said lobes on said piston rotor defining an approximate 360 degree twist in one direction from the large end thereof to the small end thereof, and each of said grooves on said cylinder rotor defining an approximate 240 degree twist in an opposite direction from the large end thereof to the small end thereof;
 - (h) said piston rotor and said cylinder rotor being tapered in the same direction and each being of frusto-conical shape such that the axes thereof are non-parallel and converge to define an angle therebetween; and,
 - (i) means for driving said rotors to effect induction of fluid to said inlet and ejection of fluid through said outlet.
12. An improved fluid vacuum pump, according to claim 11, in which the inlet to said casing is located so as to communicate with a lobe-groove combination axially disposed approximately one-sixth to one-half the axial length of said rotors from the large ends thereof.
13. An improved fluid vacuum pump, according to claim 12, in which said inlet is located so as to communicate with a lobe-groove combination one-third the axial length of said rotors measured from the large ends thereof.
14. An improved fluid vacuum pump, according to claim 11, in which said piston rotor at any given axial position thereof has a lesser cross-sectional area than said cylinder rotor at a corresponding cross section thereof.
15. An improved fluid vacuum pump, according to claim 11, in which said piston rotor has an overall diameter two-thirds the diameter of said cylinder rotor at any given axial point and in which said piston rotor has two-thirds as many lobes as said cylinder rotor has grooves.
16. An improved fluid vacuum pump comprising:
 - (a) a casing;
 - (b) an inlet to said casing for said fluid;

- (c) an outlet from said casing for said fluid;
 - (d) a piston rotor and a cylinder rotor mounted for rotation with said casing;
 - (e) said rotors being designed to have intermeshing lobes and grooves co-functioning with each other and with said casing to effect induction and ejection of said fluid;
 - (f) each of said lobes and said grooves being characterized, respectively, by a radially extending mating rectilinear flat surface designed to effect maximum displacement of fluid with a minimum movement of said rotors;
 - (g) each of said lobes and said grooves defining a helix on the surface of said rotors, respectively, the angle of said helix changing from one end of each of said rotors to the other end thereof; and,
 - (h) means for driving said rotors in synchronization.
17. An improved fluid vacuum pump comprising:
- (a) a casing;
 - (b) an inlet to said casing for said fluid;
 - (c) an outlet from said casing for said fluid;
 - (d) a piston rotor and a cylinder rotor mounted for rotation within said casing;
 - (e) said rotors being designed to have intermeshing lobes and grooves co-functioning with each other and with said casing to effect induction and ejection of said fluid;
 - (f) each of said lobes of said piston rotor and each of said grooves of said cylinder rotor being characterized, respectively, by a radially extending mating rectilinear flat surface designed to effect maximum displacement of fluid with a minimum movement of said rotors;
 - (g) said piston rotor and said cylinder rotor being tapered in the same direction and being of frusto-conical shape such that the axes thereof are non-parallel and converge to define an angle therebetween;
 - (h) each of said lobes of said piston rotor and each of said grooves of said cylinder rotor defining a helix about the body of the respective rotors, said helix being in the form of an angularly decreasing twist from the large end to the small end of said rotors, respectively; and,
 - (i) means for driving said rotors so as to effect induction of said fluid through said inlet and out through said outlet of said vacuum pump.
18. An improved fluid vacuum pump, according to claim 17, in which said twist characterizing said piston rotor approximates 360 degrees from the large end of said piston rotor to the small end thereof, and in which the twist characterizing said cylinder rotor approximates 240 degrees from the large end of said cylinder rotor to the small end thereof.
19. An improved fluid vacuum pump, according to claim 17, in which the inlet of said casing is located so as to communicate with a mating lobe-groove combination intermeshing at a point one-half to one-sixth the axial length of said rotors measured from the large ends thereof.
20. An improved fluid vacuum pump, according to claim 17, in which said piston rotor has fewer lobes than said cylinder rotor has grooves, and in which said piston rotor has a smaller overall diameter than said cylinder rotor has at any given axial point on the center line of said pump.
21. An improved fluid vacuum pump comprising:
- (a) a casing;
 - (b) an inlet to said casing for said fluid;
 - (c) an outlet from said casing for said fluid;
 - (d) a piston rotor and a cylinder rotor mounted for rotation within said casing;
 - (e) said rotors each being tapered in the same direction and being of frusto-conical shape such that the axes thereof are non-parallel and converge to define an angle therebetween;

- (f) said casing having a pair of opposing sidewalls parallel to the respective axes of said piston rotor and said cylinder rotors;
 - (g) said rotors being designed to have intermeshing spiralling lobes and grooves co-functioning with each other and with said casing to effect induction and ejection of said fluid;
 - (h) each of said lobes and said grooves being characterized, respectively, by a radially extending mating rectilinear flat surface designed to effect maximum displacement of fluid with a minimum movement of said rotors; and,
 - (i) means for driving said rotors in synchronization so as to effect induction of said fluid through said inlet and ejection of said fluid through said outlet of said pump.
22. An improved fluid vacuum pump, according to claim 21, in which said lobes and grooves, respectively, define a helix on the periphery of said piston rotor and said cylinder rotor, respectively.
23. An improved fluid vacuum pump, according to claim 21, in which the overall diameter of said cylinder rotor and said piston rotor decreases in a direction from said inlet to said outlet.
24. An improved fluid vacuum pump, according to claim 21, in which the angle of twist characterizing said spiralling lobes and grooves, respectively on said piston rotor and said cylinder rotor decreases in a direction from the large end to the small end of said rotor, such that said helix more closely approaches being parallel to the axes of said rotors, respectively.
25. An improved fluid vacuum pump, according to claim 21, in which said angle of twist of said piston rotor is approximately 360 degrees from the large end to the small end thereof in one direction and in which the angle of twist for said cylinder rotor is approximately 240 degrees from the large end to the small end thereof in an opposite direction.
26. An improved fluid vacuum pump comprising:
- (a) a casing;
 - (b) an inlet to said casing for said fluid;
 - (c) an outlet from said casing for said fluid;
 - (d) a piston rotor and a cylinder rotor mounted for rotation within said casing;
 - (e) said rotors being designed to have intermeshing spiralling lobes and grooves co-functioning with each other and with said casing to effect induction and ejection of said fluid;
 - (f) each of said lobes and said grooves being characterized, respectively, by a radially extending mating rectilinear flat surface designed to effect maximum displacement of fluid with a minimum movement of said rotors, and a radiused mating portion embracing approximately a quarter circle; and,
 - (g) means for driving said rotors in synchronization.
27. An improved fluid vacuum pump, according to claim 26, in which said rotors are each tapered in the same direction and are of frusto-conical shape such that the axes thereof are non-parallel and converge to define an angle therebetween.
28. An improved fluid vacuum pump, according to claim 26, in which said casing defines opposing sidewalls parallel, respectively, to the respective axes of said rotors.
29. An improved fluid vacuum pump, according to claim 26, in which the lobes of said piston rotor define a helix on the periphery thereof and the grooves of said cylinder rotor define a helix on the periphery thereof.
30. An improved fluid vacuum pump, according to claim 29, in which the helix characterizing said piston rotor and the helix characterizing said cylinder rotor decreases in its angle relative to the axes of said respective rotors from the large end to the small end thereof.

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