Aug. 9, 1966 F. G. WEIS 3,265,009

SEWAGE PUMPING SYSTEM

Filed Aug. 7, 1963 2 Sheets-Sheet 1

Fig. 1.

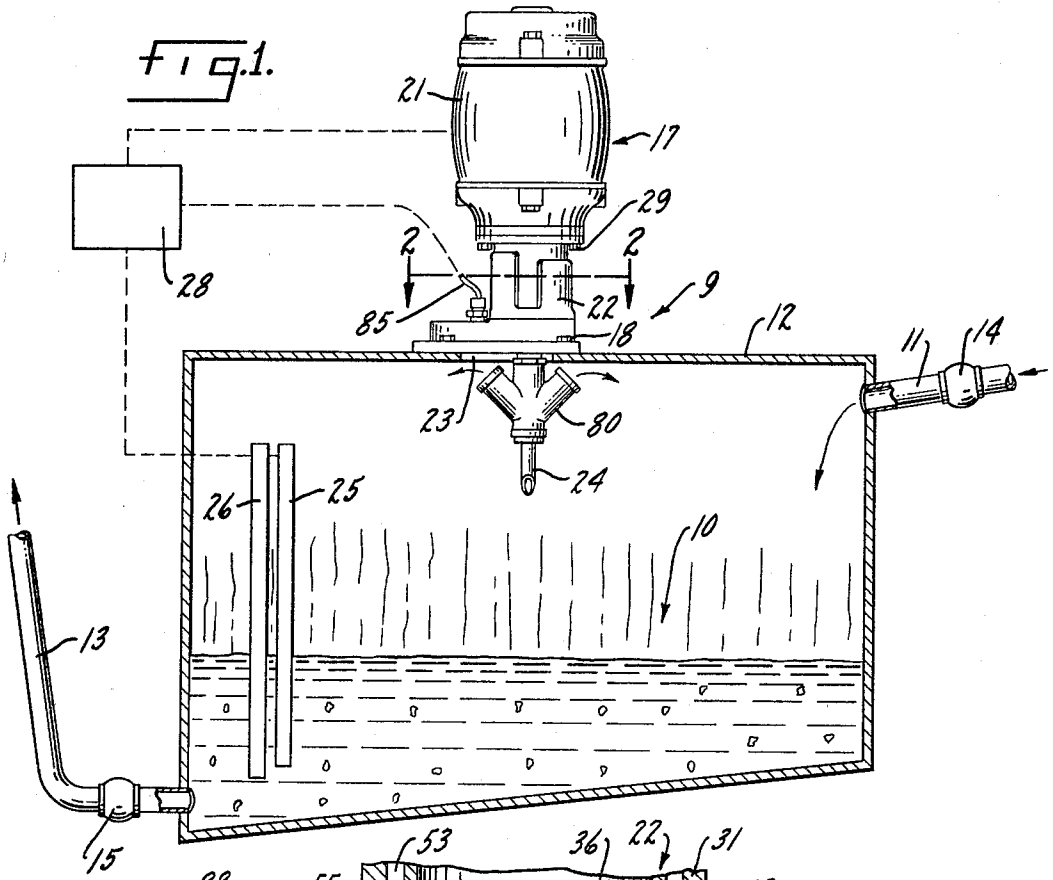
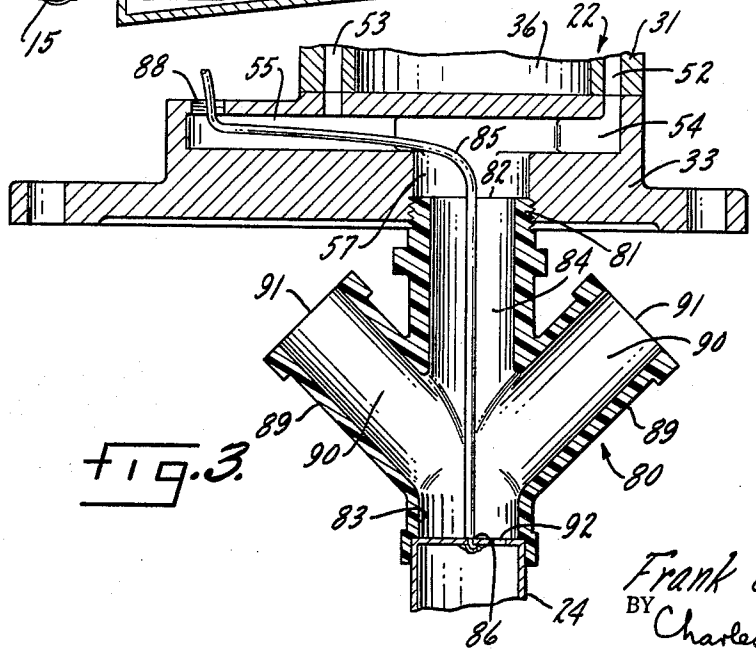


Fig. 3.



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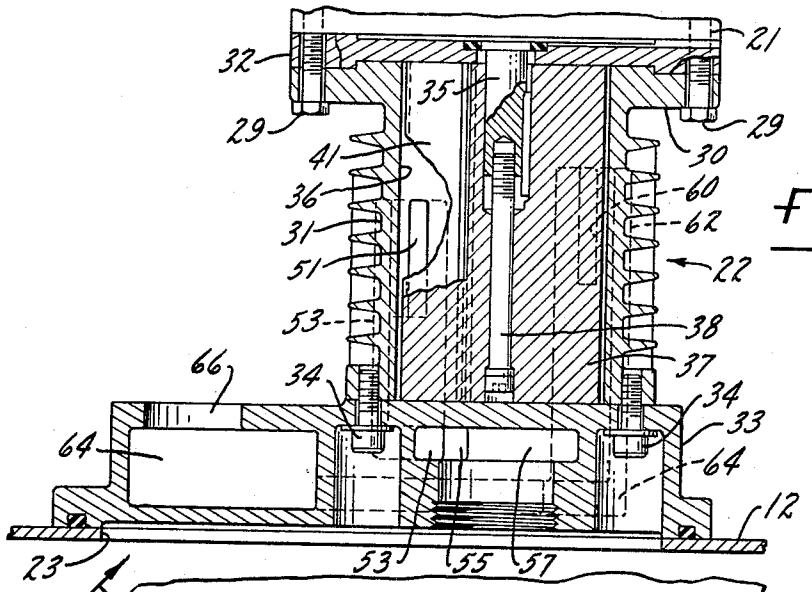


Fig. 4.

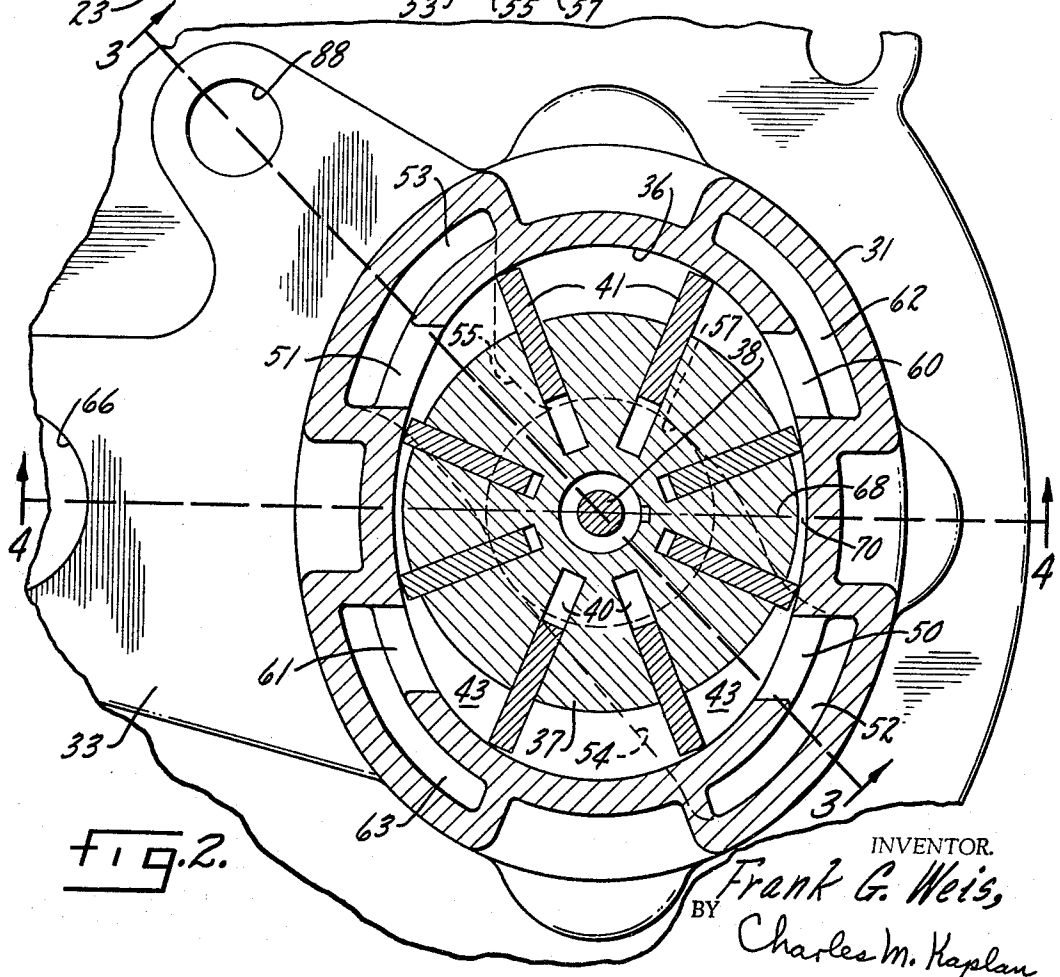


Fig. 2.

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3,265,009

SEWAGE PUMPING SYSTEM

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13 Claims. (Cl. 103—235)

This invention relates to sewage pumping systems employing reversible compressors, and more in particular to an improved, balanced rotor gas compressor constructed to reduce or prevent breakage of vanes.

It is frequently advantageous for a compressor and its driving motor to be mounted so that their axes of rotation are coincident. This enables the use of a single shaft for both the motor and the compressor and eliminates the need for coupling drives such as pulleys and belts.

The usual type of compressor that has been mounted coaxially with its driving motor has a rotor offset from the center of the rotor housing or the rotor has an odd number of vanes. This causes the forces acting on different areas of the rotor to be unequal, and results in what is known as an unbalanced rotor design. When the compressor is to be used to pump a relatively large volume of gas, as for example more than 6 to 8 cubic feet per minute at or above about 5 p.s.i., the forces on an unbalanced rotor type compressor prevent its being used in the coaxial motor-compressor combination described above. The reason is that the forces of unbalance on the rotor cause bending of the shaft connecting the rotor and motor. Deflection of the shaft causes the rotor to bind against the sides of the rotor housing, and this is obviously harmful.

To eliminate the above difficulty, a coaxial motor-compressor unit should employ a compressor of the type in which the rotor has an even number of vanes and the rotor is mounted in the center of a generally oval or elliptical housing having diametrically opposed discharge ports and diametrically opposed intake ports. In this arrangement the forces acting on different areas of the rotor are essentially equal so the rotor can be regarded as balanced. Such compressors can have a relatively tall rotor without the dangers of shaft deflection inherent in unbalanced rotor units.

The use of the above described balanced rotor compressors in systems in which solids or liquids are carried by the gas being compressed has not been wide spread in the past because the rotor vanes are frequently broken. Such vanes are ordinarily made from a carboniferous material because such materials are self-lubricating and thus have a relatively long life even though the vanes continually wear against the inside of the rotor housing. However, such materials are relatively frangible, and condensed vapors or other liquids, and solid particles entering the rotor housing shear off the ends of the vanes when such particles are compressed in the areas of minimum clearance between the rotor and its housing. The reason for this is that heretofore it was believed that the efficiency of such compressors would be reduced below an acceptable value if the total diametrical spacing between the rotor and the housing in the area of the minor axis of the elliptical cavity was greater than about .005 inch. This invention is based in part on the discovery that the minimum spacing between the rotor and its housing can be

increased to a much greater magnitude without loss of efficiency and without breakage of vanes, provided that the spacing is held within certain critical limits.

Accordingly, it is an object of the invention to provide an improved gas compressor eliminating or at least reducing vane breakage caused by liquids or solids being entrained in the gas being compressed.

Another object is to provide an improved gas compressor of normal to increased efficiency in which vane breakage is reduced.

A further object is to provide a balanced-rotor gas compressor employing carboniferous vanes in which the spacing between the rotor and its housing is increased so as to reduce vane breakage caused by trapped liquids or solids, but without reducing the efficiency of the compressor.

A further object is to provide a sewage pumping system in which a balanced-rotor compressor pumps gas that has contacted sewage, but breakage of compressor vanes is reduced notwithstanding solid and liquid particles of sewage entering the compressor.

A further object is to provide an improved sewage pumping system employing a high capacity, relatively efficient gas compressor requiring little maintenance.

A further object is to provide an improved system employing a gas compressor to pump sewage in which shut down of the system to repair broken compressor vanes is minimized.

Other objects and advantages of the invention will be apparent from the drawing, specification, and claims, and the scope of the invention will be pointed out in the claims.

Briefly stated, according to one aspect of the invention, a reversible, balanced-rotor, gas compressor has a housing defining a generally elliptical cavity in which a gas intake port and a gas discharge port are located on opposite sides of the cavity adjacent the minor axis of the cavity, and a rotor is supported so that its axis is coincident with the center of the cavity. A plurality of diametrically opposed carboniferous vanes are slidable in radial slots of the rotor. The compressor can compress gases containing discrete amounts of liquid and solid matter without breaking the vanes and without loss of efficiency if the diameter of the rotor is from about .025 to about .085 inch less than the length of the minor axis of the cavity.

In the drawing:

FIG. 1 is a schematic, partially cross sectional view of an embodiment of the invention.

FIG. 2 is an enlarged cross sectional, partially broken away view taken along the line 2—2 in FIG. 1.

FIG. 3 is a cross sectional, partially broken away view on a reduced scale taken along the line 3—3 in FIG. 2.

FIG. 4 is a cross sectional, partially broken away view on a reduced scale taken along the line 4—4 in FIG. 2.

Referring to the drawing, FIG. 1 shows a system 9 for pumping sewage 10 in accord with the teachings of the invention. A sewage inlet pipe 11 is connected adjacent the upper end of a sewage receiving chamber or tank 12, and a sewage outlet pipe 13 is connected adjacent the bottom of tank 12. A conventional check valve 14 in pipe 11 permits sewage to flow into tank 12 but not out of tank 12, and a similar check valve 15 in pipe 13 permits sewage to flow out of tank 12 but not into tank 12.

A combined motor compressor unit **17** for pumping sewage is attached to tank **12** by bolts **18**. The unit **17** comprises a conventional reversible electric motor **21** connected to a suitable source of electric power and a gas compressor **22** in accord with the teachings of the invention. Compressor **22** communicates with the interior of tank **12** through a hole **23** so that it can withdraw air therefrom as sewage is flowing into tank **12** or compress air thereinto to eject the sewage.

When the system starts up, tank **12** is empty and compressor **22** runs in a direction that withdraws air therefrom. When the sewage level in tank **12** reaches a predetermined high level determined by a hollow, open-ended electrode **24**, motor **21** is caused to reverse and thereby run compressor **22** in a direction that compresses air into tank **12**. As air pressure builds up in tank **12**, it forces the accumulated sewage out through pipe **13**. When the sewage recedes to a predetermined low level determined by the electrodes **25** and **26**, motor **21** stops. The electrodes **24–26** may be interconnected in any sewage ejector electrical control circuit **28** known to the art, it being understood that the specific details of such a circuit form no part of the present invention.

As shown in FIGS. 1 and 4, motor **21** is attached to compressor **22** by bolts **29** which pass through openings in lugs **30** on the compressor housing **31**, and through openings in compressor lid **32**; bolts **29** screw into tapped holes in the casing of motor **21**. Housing **31** is attached to a base member **33** by bolts **34**. The output shaft **35** of motor **21** passes through a hole in lid **32** and is coaxial with the center of an elliptical cavity **36** defined by the inside wall of housing **31**.

A generally cylindrical rotor **37** is mounted for rotation in the center of cavity **36** by having its center axis coincident with shaft **35**. Rotor **37** is secured to shaft **35** by a bolt **38** passing through its center and threading into a tapped hole in shaft **35**.

A plurality of radial slots **40** are equally spaced around the periphery of rotor **37**. A radially slidable vane **41** made from a carboniferous material such as resin treated electro graphite occupies each slot **40**. As rotor **37** turns, centrifugal forces cause vanes **41** to move outwardly into contact with the inside surface of cavity **36** and thereby define chambers **43** of increasing or decreasing size for causing either pressure or suction at the ports in housing **31**. For proper performance of the compressor and sewage pumping system, the number of slots **40** should be equal to $6n-4(n-1)$, with n being a positive whole number. The preferred range for values of n is from 1 to 4, with 2 being the most preferred value for n. When $n=2$, there will be 8 slots in the rotor, as shown in FIG. 2.

A first pair of diametrically opposed ports **50** and **51** in housing **31** communicate with cavity **36**. Ports **50** and **51** are interconnected through channels **52** and **53** and mating passages **54** and **55** in base member **33** with an opening **57**, which communicates with the interior of tank **12**. A second pair of diametrically opposed ports **60** and **61** in housing **31** communicate with cavity **36**. Ports **60** and **61** are interconnected through channels **62** and **63** and mating passages, such as **64** in base member **33**, with an opening **66** which is vented to the atmosphere. For proper operation of the compressor, ports **50** and **60** on one side of cavity **36** and ports **51** and **61** on the other side of cavity **36** must be spaced far enough apart for at least one vane **41** to separate adjacent ports at all times. As shown in FIG. 4, ports **60** and **51** are staggered vertically; port **61** is at the same elevation as port **60**, and port **50** is at the same elevation as port **51**. This arrangement enables vanes **41** to wear more evenly than when all ports are at the same elevation, and thus increases the life of the vanes.

Turning of rotor **37** in a counter-clockwise direction causes compression at ports **50** and **51** and suction at ports **60** and **61**; in this situation air is drawn in through

opening **66**, compressed in cavity **36**, and then pumped through ports **60** and **61** and discharged through opening **57** into tank **12**. Correspondingly, rotation of rotor **37** in a clockwise direction causes compression at ports **60** and **61** and suction at ports **50** and **51**. This causes air to be withdrawn from tank **12** and pumped to the atmosphere.

Air drawn from tank **12** into cavity **36** frequently contains liquid and solid particles of sewage as well as moisture vapor. Compression of such air in elliptical cavity **36** causes deleterious accumulations of solid and liquid matter in the area adjacent the minor axis **68** of the cavity. To prevent breakage of vanes **41** resulting from their impact with such accumulations, the radial spaces **70** between housing **31** and rotor **37** can be increased to values which were heretofore considered unacceptable. It has been discovered that when the difference in length between the minor axis **68** and the diameter of rotor **37** (i.e., twice the radial spacing **70**) is from about .025 to about .085 inch, vanes **41** are not broken by solids and liquids accumulating in cavity **36**, and the efficiency of the compressor remains at an unexpectedly high level. The preferred range of such diametrical differences in length is from about .050 to about .085 inch, with the preferred value being about .065 inch.

Tests were run to determine the effect on compressor efficiency of increasing the spacing between rotor **37** and its housing **31**. A gas compressor of the same general structure as is shown in the drawing had a cylindrical rotor 4.981 inches in height contained in an elliptical cavity in a housing 5.001 inches in height; eight vanes 4.996 inches long made from Speer Carbon Co. Grade 9116 Graphite were slidable in slots uniformly spaced around the rotor. The length of the minor axis of the cavity was 3.058 inches, and the rotor was initially 3.053 inches in diameter, thus leaving a minimum total spacing of .005" between the rotor and the wall of the housing at the minor axis of the cavity. The inlet and outlet ports on each side of the housing adjacent the minor axis were 1½ inches apart at their closest edges.

The compressor was run at a constant speed of 1750 r.p.m., first at an outlet pressure of 6 p.s.i., and then at an outlet pressure of 10 p.s.i., and its efficiency was determined in the manner described below. Thereafter, the rotor was removed and its diameter was machined down to various smaller sizes, and the efficiency tests were repeated under the same conditions.

The efficiency of a compressor is its output horsepower divided by the electrical input horsepower, which was measured by wattmeter. It can be shown that for isentropic compression, compressor output horsepower is equal to:

$$\frac{QP_1\left[\left(\frac{P_2}{P_1}\right)^{(0.283)}-1\right]}{64.85}$$

where:

Q=output volume of gas flow in cubic feet per minute,
P_1=inlet pressure in pounds per square inch,
P_2=outlet pressure in pounds per square inch,

Q was measured by placing an orifice plate across the discharge outlet of the compressor in accord with ASME Power Test Code for Displacement Compressors, Blowers, and Vacuum Pumps PTC 9–1954. All pressures were measured with a mercury manometer, with inlet pressure being atmospheric pressure at all times, and outlet pressure being kept constant by a throttling valve at the compressor outlet. Ambient temperature was essentially constant at about 72° F.

The results of the efficiency test for each spacing be-

tween the rotor and housing at the minor axis of the cavity are presented in the table below:

TABLE

Spacing (inches) [1]	Q (Compressor Output Capacity in c.f.m.)	Output (Horsepower)	Electrical Input (Horsepower)	Efficiency (Percent)
At 6 p.s.i. Outlet Pressure				
.005	28.3	0.660	1.66	39.7
.025	28.0	0.652	1.59	41.0
.045	28.0	0.652	1.59	41.0
.065	27.3	0.635	1.51	42.0
.085	26.9	0.627	1.56	40.1
.105	26.6	0.619	1.74	35.5
At 10 p.s.i. Outlet Pressure				
.005	25.2	0.904	2.46	36.7
.025	24.6	0.884	2.39	36.9
.045	24.6	0.884	2.39	36.9
.065	23.8	0.885	2.27	38.0
.085	22.6	0.812	2.36	34.4
.105	21.6	0.775	2.56	30.2

[1] Minor axis 68 of elliptical cavity 36 minus diameter of rotor 37.

The table shows that contrary to expectation, the efficiency of the compressor did not drop sharply as the spacings **70** between rotor **37** and housing **31** were increased above the prior art maximum of about .005 inch. On the contrary, compressor efficiency rose slightly until maximum efficiency was reached at a spacing of about .065 inch. Efficiency did not drop sharply below that found for the .005 inch spacing until the spacing was increased above about .085 inch.

The tests described above were expected to show that compressor efficiency drops sharply as the spacing **70** was increased. The discovery that compressor efficiency did not drop sharply in the spacing range between about .025 and about .085 inch was highly unexpected. This has the effect of permitting the spacings **70** to be increased to a value at which vanes **41** will not be broken with no less of compressor efficiency. Subsequent tests have shown that when a compressor constructed as described herein was run under field conditions in a sewage ejector, vanes **41** did not break; the spacing between the rotor and its housing was .065 inch. Under essentially the same field conditions, vane breakage had been a severe problem with spacings of about .005 inch.

Turning now to FIGS. 1 and 3, hollow electrode **24** is shown connected to compressor **22** by a unitary insulator-nozzle element **80**, which is made from a suitable corrosion resistant electrical insulating material, such as polyethylene. Threaded end **81**, which has a first opening **82**, screws into opening **57** in base **33**, and the closed end of electrode **24** is secured by any suitable means in a second opening **83** at the other end of the element **80**. Openings **82** and **83** are connected by a first passage **84**. An insulated electrical conductor **85** is soldered to electrode **24** at **86**. Conductor **85** passes through first passage **84**, opening **57**, passage **55** in base **33**, and finally out through an opening **88**, and is then connected into motor control circuit **28**.

One or more air discharging nozzles **89** are defined by second passages **90** which intersect first passage **84**. Nozzles **89** project outwardly from element **80**, and their discharge ports **91** are oriented to expel air from compressor **22** upwardly toward the top of tank **12** and away from electrode **24**. When compressor **22** is evacuating tank **12**, nozzles **89** draw air into compressor **22** from adjacent the top of tank **12** and thus reduce the amount of solid and liquid material drawn into the compressor. A drain hole **92** may be provided in the closed end of hollow electrode **24** to enable moisture that condenses in compressor **22** or element **80** to drain through the open end of the electrode into tank **12**. The drain **92** should be small enough to prevent any significant amount of air from being discharged therethrough.

Heretofore, in sewage ejectors the practice has been to force air through a nozzle or a hollow electrode that expels the air downwardly towards the sewage. This caused splashing of sewage upwardly onto electrode **24** when the sewage is at or near its highest level. The result was that electrode **24** often became so covered with crud that it was rendered inoperative with a consequent failure of the sewage pumping system. The combined insulator-nozzle element **80** solves this problem by expelling air in a direction that prevents splashing of sewage.

It will be understood that while the form of the invention herein shown and described constitutes a preferred embodiment, it is not intended herein to illustrate all of the equivalent forms or ramifications thereof. It will also be understood that the words used are words of description rather than of limitation, and that various changes may be made without departing from the spirit or scope of the invention herein disclosed, and it is aimed in the appended claims to cover all such changes as fall within the true spirit and scope of the invention.

What is claimed is:

1. In a reversible, balanced rotor, gas compressor having:

(1) a housing defining an elliptical cavity having a minor axis of predetermined length,

(2) there being a gas intake port and a gas discharge port in opposite sides of said housing in the area adjacent said minor axis,

(3) a circular rotor supported rotatably in said housing so that its axis is coincident with the center of said elliptical cavity, said rotor having a diameter of predetermined length,

(4) there being an even number of slots in said rotor,

(5) a breakable vane slidable in each slot, and

(6) said slots and said ports being oriented so that a vane is always located between the intake port and discharge port on the same side of said housing;

the improvement enabling said compressor to efficiently compress gases containing discrete amounts of liquid and solid matter without breaking said vanes comprising:

said length of said rotor diameter being from about .025 to about .085 inch less than said length of said minor axis.

2. The invention of claim **1** wherein said length of said rotor diameter is from about .050 to about .085 inch less than said length of said minor axis.

3. The invention of claim **2** wherein said length of said rotor diameter is about .065 inch less than said length of said minor axis.

4. The invention of claim **1** wherein the number of slots in said rotor is equal to $6n-4(n-1)$, with n being a positive whole number.

5. The invention of claim **4** wherein n is from 1 to 4.

6. The invention of claim **6** wherein $n=2$.

7. A sewage pumping system comprising:

(1) chamber means into which sewage is drawn as air is expelled therefrom and then from which sewage is ejected by the pressure of air being compressed thereinto, said air containing condensible moisture vapor and solid particles from said sewage as it comes from said chamber means, and

(2) a balanced rotor, reversible, motor-compressor unit for withdrawing from said chamber or compressing thereinto at least six cubic feet per minute of air, comprising:

(a) a reversible motor having shaft means coaxially aligned with and directly coupled to a generally circular rotor at the center thereof, said rotor having a diameter of predetermined length,

(b) wall means defining a generally elliptical cav-

ity receiving said rotor, said elliptical cavity having a minor axis of predetermined length,

(c) the central axis of said rotor being aligned with the center of said cavity,

(d) said rotor having radial slots receiving slidable vanes made from a material breakable by condensed moisture and solid particles of sewage trapped between said rotor and said wall means, and

(e) said length of said rotor diameter being from about .025 to about .085 inch less than said length of said minor axis, whereby sufficient clearance is provided to permit passage of said condensed moisture and solid particles between said rotor and said wall without breaking said vanes.

8. The invention of claim **7** wherein said length of said rotor diameter is from about .050 to about .085 inch less than said length of said minor axis.

9. The invention of claim **8** wherein said length of said rotor diameter is about .065 inch less than said length of said minor axis.

10. A sewage pumping system comprising:

(1) chamber means into which sewage is drawn as air is expelled therefrom and then from which sewage is ejected by the pressure of air being compressed thereinto, said air containing condensible moisture vapor and solid particles from said sewage as it passes from said chamber means, and

(2) a balanced rotor, reversible, motor-compressor unit for withdrawing from said chamber or compressing thereinto at least six cubic feet per minute of air, comprising:

(a) a reversible motor having shaft means coaxially aligned with and directly coupled to a generally circular rotor at the center thereof, said rotor having a diameter of predetermined length,

(b) a housing defining a generally elliptical cavity receiving said rotor, said elliptical cavity having a minor axis of predetermined length,

(c) the central axis of said rotor being aligned with the center of said cavity,

(d) there being a gas intake port and a gas discharge port in opposite sides of said housing in the area adjacent said minor axis, the intake ports being diametrically opposed to each other and the discharge ports being diametrically opposed to each other,

(e) said rotor having an even number of radial slots receiving slidable vanes made from a material breakable by condensed moisture and solid particles of sewage trapped between said rotor and said housing,

(f) said slots and ports being oriented so that a vane is always located between the intake port and the discharge port on the same side of said housing, and

(g) said length of said rotor diameter being from about .025 to about .085 inch less than said length of said minor axis, whereby sufficient clearance is provided to permit passage of said condensed vapors and solids between said rotor housing wall without breaking said vanes.

11. In a combined motor-compressor unit having:

(1) a reversible electric motor including an output shaft, and

(2) a balanced rotor gas compressor having:

(a) a housing defining an elliptical cavity, said elliptical cavity having a minor axis of predetermined length,

(b) there being a gas intake port and a gas discharge port in opposite sides of said housing in the area adjacent the minor axis of said elliptical cavity, the intake ports being diametrically opposed to each other and the discharge ports being diametrically opposed to each other,

(c) a circular rotor supported rotatably in said housing so that its axis is coincident with the center of said elliptical cavity, said rotor having a diameter of predetermined length,

(d) said motor output shaft being coaxially aligned with and directly secured to said rotor,

(e) said rotor having radial slots in which are slidable vanes made from a material breakable by condensed moisture and solid particles compressed in said cavity, and

(f) said length of said rotor diameter being from about .025 to about .085 inch less than said length of said minor axis, whereby sufficient clearance is provided to permit passage of said condensed vapors and solids between said rotor and cavity wall without breaking said vanes.

12. In a combined motor-compressor unit having:

(1) a reversible electric motor including an output shaft, and

(2) a balanced rotor gas compressor having:

(a) a housing defining an elliptical cavity, said elliptical cavity having a minor axis of predetermined length,

(b) there being a gas intake port and a gas discharge port in opposite sides of said housing in the area adjacent the minor axis of said elliptical cavity, the intake ports being diametrically opposed to each other and the discharge ports being diametrically opposed to each other,

(c) a lid bolted to the upper end of said housing for closing one end of said cavity,

(d) said housing being bolted to a supporting base member at its other end,

(e) a circular rotor supported rotatably in said housing so that its axis is coincident with the center of said elliptical cavity, said rotor having a diameter of predetermined length,

(f) said motor output shaft passing through a hole in said lid and being coaxially aligned with and connected to said rotor by a coaxial bolt passing through the center of said rotor and screwing into said shaft,

(g) said rotor having radial slots in which are slidable vanes made from a material breakable by condensed moisture and solid particles compressed in said cavity, and

(h) said length of said rotor diameter being from about .025 to about .085 inch less than said length of said minor axis, whereby sufficient clearance is provided to permit passage of said condensed vapors and solids between said rotor and cavity wall without breaking said vanes.

13. In a reversible, balanced rotor, gas compressor comprising:

(1) a housing defining an elliptical cavity, said elliptical cavity having a minor axis of predetermined length,

(2) there being a gas intake port and a gas discharge port in opposite sides of said housing in the area adjacent the minor axis of said elliptical cavity,

(a) the intake ports being diametrically opposed to each other and the discharge ports being diametrically opposed to each other,

(b) the intake ports and discharge ports respectively being staggered axially with relation to each other,

(3) a circular rotor supported rotatably in said housing so that its axis is coincident with the center of said elliptical cavity, said rotor having a diameter of predetermined length,

(4) there being an even number of diametrically opposed radial slots in said rotor,

(5) a radially slidable carboniferous vane in each slot, and

(6) said slots and said ports being oriented so that a vane is always located between the intake port and discharge port on the same side of said housing;

the improvement enabling said compressor to efficiently compress gases containing discrete amounts of liquid and solid matter without breaking said vanes comprising:

said length of said rotor diameter being from about .025 to about .085 inch less than said length of said minor axis.

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