Sept. 14, 1965 B. K. WOOLFENDEN ETAL 3,206,106

COMPRESSOR

Filed July 26, 1961 4 Sheets-Sheet 1

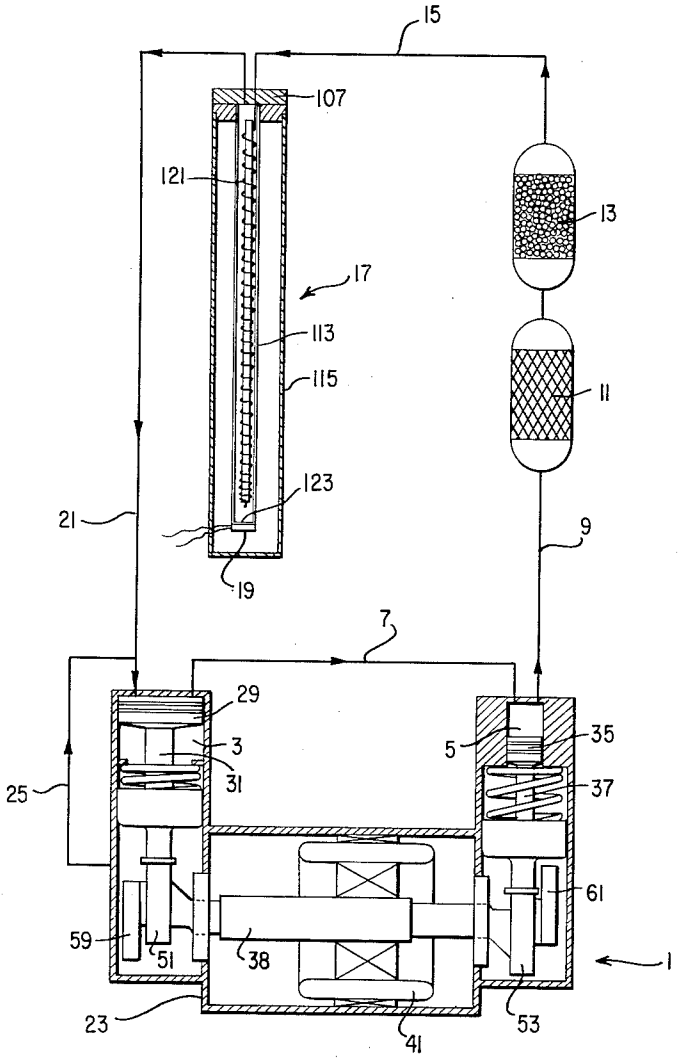


FIG 1

INVENTORS
BRIAN K. WOOLFENDEN
PAUL I. PETERSEN

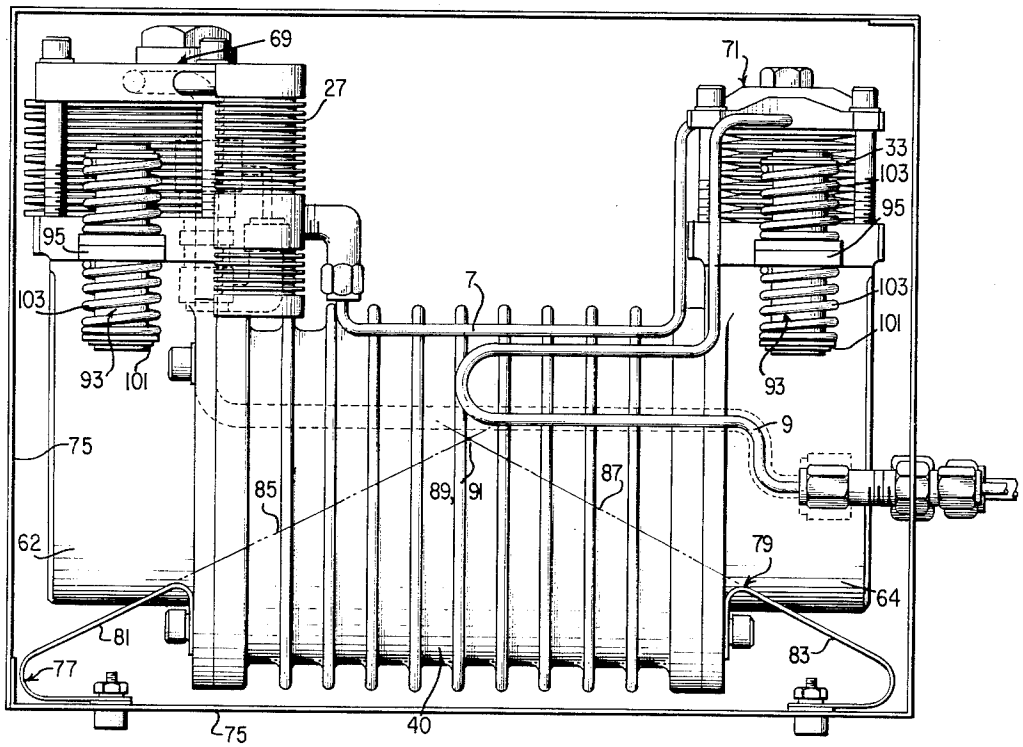
BY Shaney & O'Neil
ATTORNEYS.

Sept. 14, 1965 B. K. WOOLFENDEN ETAL 3,206,106

COMPRESSOR

Filed July 26, 1961 4 Sheets-Sheet 2

FIG 2



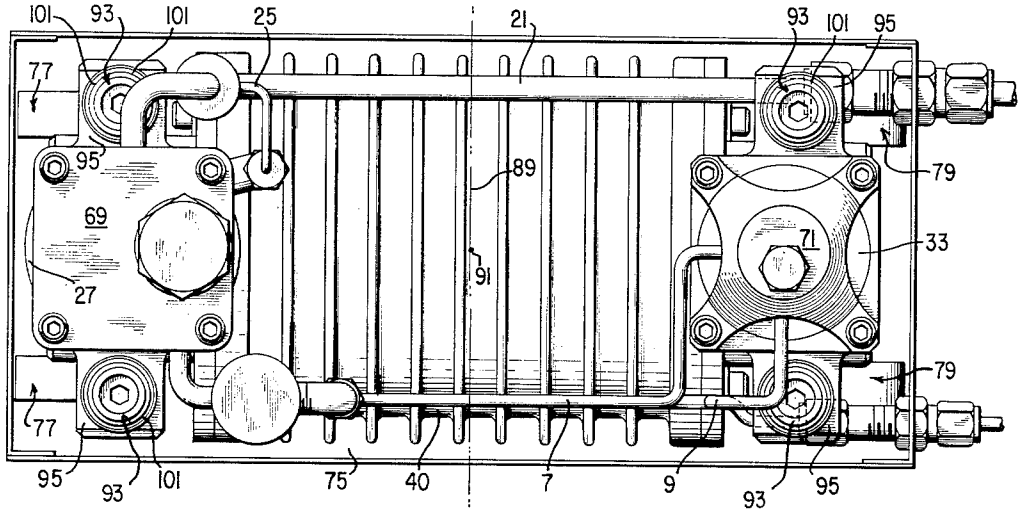


FIG 3

INVENTORS
BRIAN K. WOOLFENDEN
PAUL I. PETERSEN
BY Stanley & O'Neil
ATTORNEYS.

Sept. 14, 1965 B. K. WOOLFENDEN ETAL 3,206,106

COMPRESSOR

Filed July 26, 1961 4 Sheets-Sheet 3

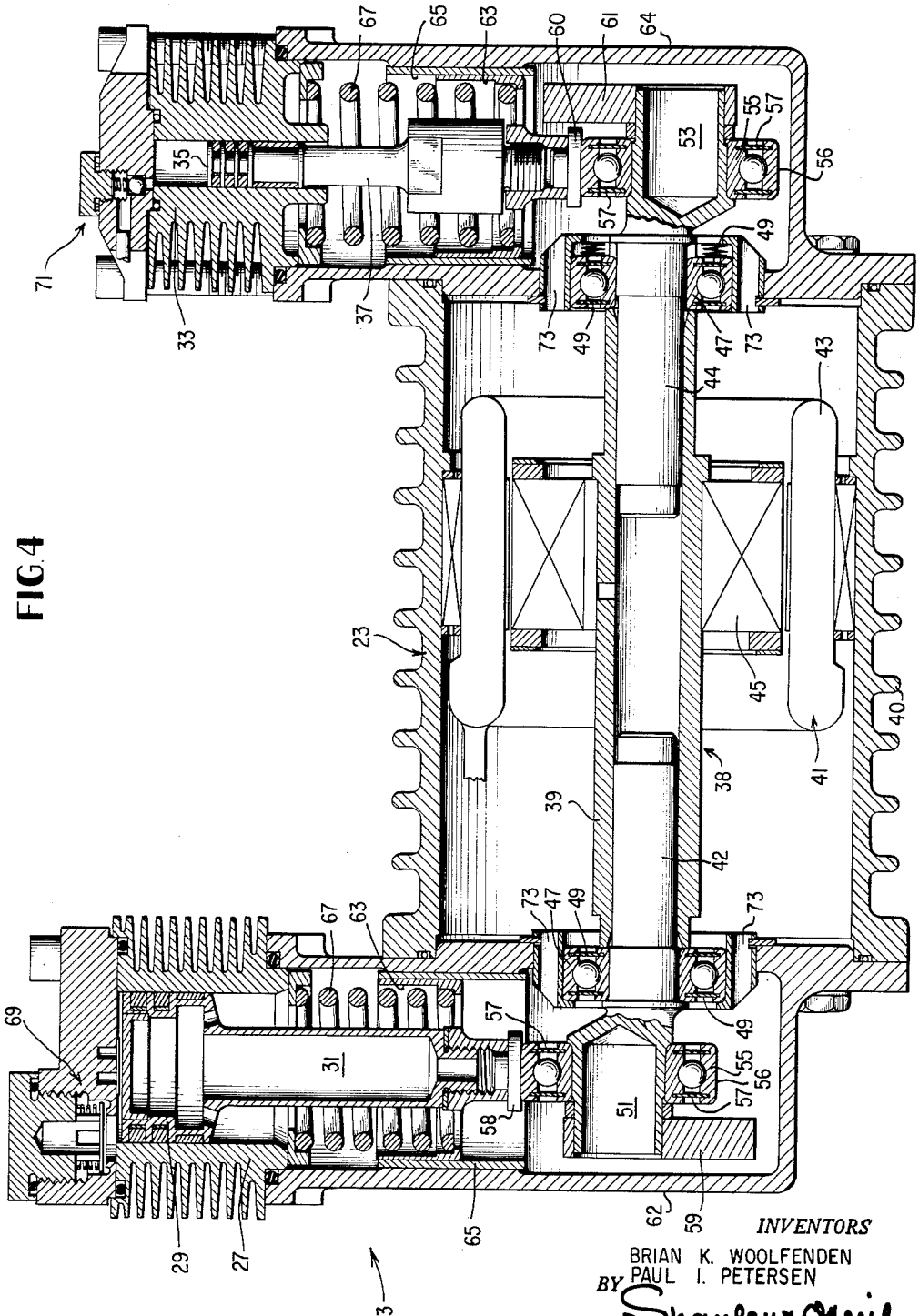


FIG. 4

INVENTORS
BRIAN K. WOOLFENDEN
BY PAUL I. PETERSEN
Shanley & O'Neil
ATTORNEYS.

Sept. 14, 1965 B. K. WOOLFENDEN ETAL 3,206,106

COMPRESSOR

Filed July 26, 1961 4 Sheets-Sheet 4

FIG 6

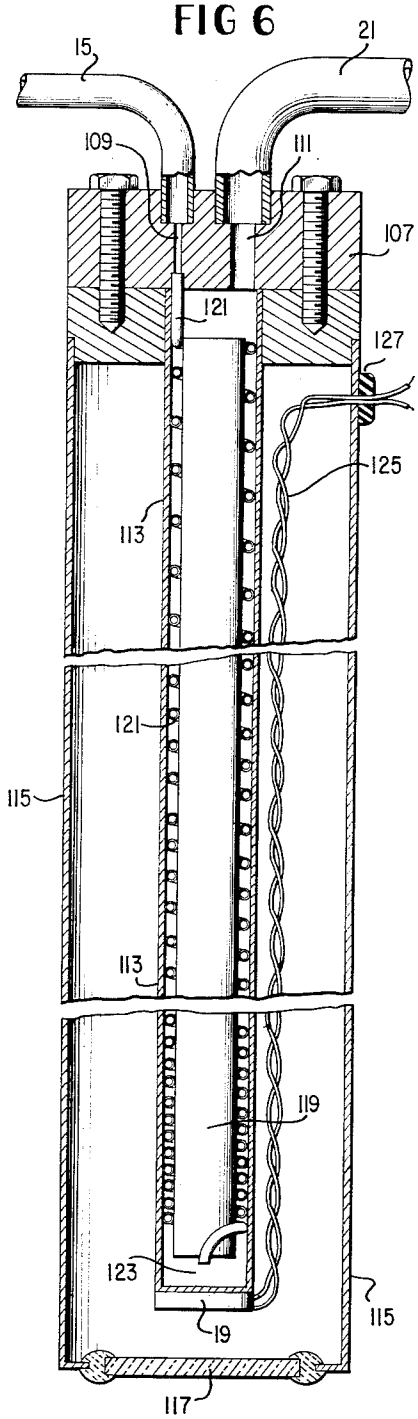


FIG 5

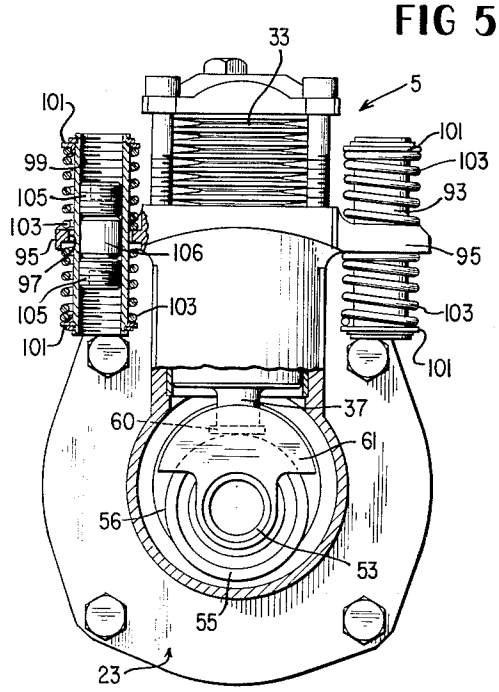
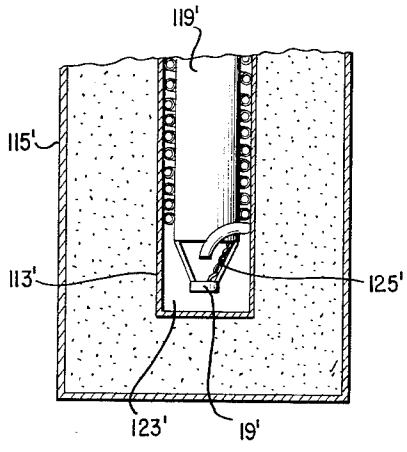


FIG 7



INVENTORS
BRIAN K. WOOLFENDEN
PAUL I. PETERSEN
BY Shanley & O'Neil
ATTORNEYS

United States Patent Office 3,206,106 Patented Sept. 14, 1965

3,206,106 COMPRESSOR

Brian K. Woolfenden, Allentown, Pa., and Paul I. Petersen, Riviera Beach, Fla., assignors to Air Products and Chemicals, Inc., a corporation of Delaware

Filed July 26, 1961, Ser. No. 126,955
8 Claims. (Cl. 230—58)

The present invention relates to compressors and more particularly to gas compressors for use in a low temperature refrigeration system.

It is an object of the present invention to provide a gas compressor well adapted to be miniaturized.

Another object is the provision of a gas compressor which reduces to a minimum contamination of the working fluid.

Another object is the provision of a gas compressor of the reciprocating type which reduces to a minimum contamination of the working fluid and permits recirculation of piston blow-by with the working fluid.

Still another object of the present invention is the provision of a gas compressor that can be operated in any desired bodily orientation.

Another object of the present invention is the provision of a gas compressor which has high dynamic stability.

Finally, it is an object of the present invention to provide a compressor adapted to be miniaturized and adapted for use in compact low temperature refrigeration systems, of the hermetically sealed type, which is of rugged construction and capable of long periods of continuous operation.

Other objects and advantages of the present invention will become apparent from a consideration of the following description, taken in connection with the accompanying drawings, which disclose one embodiment of the invention.

In the drawings, in which similar reference characters denote similar elements throughout the several views:

FIGURE 1 is a diagrammatic representation of a low temperature refrigeration system embodying principles of the present invention;

FIGURE 2 is an elevational view of a compressor for use in a low temperature refrigeration system according to the present invention, with the side of the compressor casing open;

FIGURE 3 is a plan view of the structure of FIGURE 2, with the top of the casing open;

FIGURE 4 is a side cross-sectional view of the compressor of FIGURE 2 without the casing;

FIGURE 5 is an end elevational view of the compressor of FIGURE 4 with parts broken away;

FIGURE 6 is an enlarged view in cross-section of a portion of the refrigeration system of FIGURE 1; and

FIGURE 7 is an enlarged fragmentary view, in cross-section, of a modified form of the invention.

Referring now to FIGURE 1 of the drawings in greater detail, there is shown a refrigeration system constructed in accordance with the principles of the present invention comprising a gas compressor indicated generally at **1**, of the two-stage type, having a relatively low pressure stage **3** and a relatively high pressure stage **5** and an interstage conduit **7** by which the output of the low pressure stage is fed to the intake of the high pressure stage. The high pressure gas leaving the high pressure stage passes through a conduit **9** to an absorber **11** and thence to a filter **13** in which foreign material is removed from the working fluid. This foreign material may take the form of dirt and dust and it may also include hydrocarbons introduced from the operation of the compressor.

The cleaned working fluid then passes from conduit **15** to a liquefier **17** in which it is cooled, by expansion and by countercurrent heat interchange with expanded fluid, to or below its liquefaction temperature. The liquefied working fluid is used in liquefier **17** as refrigerant for maintaining at low temperature a component **19**, for example, a small electronic amplifier or an infra-red cell or the like. The low pressure working fluid, warmed by the heat interchange, leaves the warm end of liquefier **17** by way of conduit **21** and returns to the intake of low pressure stage **3** of the compressor. The motor and both stages of the compressor are housed in a hermetically sealed casing **23**, and blow-by from both stages is returned to the working fluid circuit just upstream of the low pressure stage intake through conduit **25**.

Referring now more particularly to the compressor portion of the system, as seen in FIGURE 4 the low pressure stage **3** comprises a low pressure cylinder **27** that is a portion of hermetically sealed casing **23**. A relatively large piston **29** is axially reciprocable in the cylindrical bore of cylinder **27** and has an axially extending piston rod **31** thereon. High pressure stage **5**, at the other end of the compressor, includes a high pressure cylinder **33** which forms a portion of hermetically sealed casing **23**. A relatively small piston **35** is axially reciprocable in the bore of cylinder **33** along with its attached piston rod **37**.

Extending between and drivingly interconnecting the two stages of the compressors is drive shaft **38** mounted for rotation about its axis and disposed entirely within casing **23**. Drive shaft **38** may include a tubular member **39** carrying the rotor **45** of an electric motor **41** having a stator **43** secured to the interior of housing **40** which forms a part of the hermetically sealed casing **23**. The drive shaft **38** includes portions **42** and **44** secured to opposite ends of the tubular member **39** and supported in axially aligned longitudinally spaced apart sealed ball bearings **47** carried by the interior of casing **23** to mount the drive shaft for axial rotation. The ball bearings **47** are sealed in the sense that seals **49** are provided on both sides of the ball race to assure that the lubricant of the ball bearings does not escape from the ball bearings during operation and contaminate the gas within casing **23** except to the extent discussed below.

At the end of drive shaft **38** adjacent low pressure stage **3** the portion **42** includes an integral eccentric **51** and at the other end of drive shaft **38**, adjacent high pressure stage **5**, the portion **44** has an integral eccentric **53**, the eccentrics **51** and **53** being 180° out of phase relation to each other. Eccentrics **51** and **53** each carry sealed ball bearing assemblies **55** the axes of which are parallel and spaced apart by the sum of the eccentricities of eccentrics **51** and **53**. Ball bearing assemblies **55** are sealed in the same sense and for the same purpose as ball bearings **47**, namely, by seals **57** to prevent contamination of the gas within casing **23**. Also mounted on eccentric **51** is a counterweight **59** and on eccentric **53** a counterweight **61**. Counterweights **59** and **61** are 180° out of phase with each other and with their respective eccentrics. Moreover, the counterweights dynamically balance their respective eccentrics and ball bearing assemblies **55**, that is, the mass times the distance of the center of gravity of counterweight **59** from the axis of drive shaft **38** is equal to the mass times the distance from the axis of drive shaft **38** of the center of gravity of eccentric **51** plus the associated ball bearing assembly **55**, and the same relationship holds true for counterweight **61** relative to eccentric **53** and the associated ball bearing assembly **55**. The outer cylindrical surfaces **56** of the ball bearing assemblies **55** comprise cam surfaces which contact cam followers connected to respective piston rods. In particular, the cam surface **56** of the ball bearing assembly on the eccentric **51** contacts a cam follower **58** connected to the piston rod **31** and the cam surface **56** of the ball bearing assembly on the eccen-

tric 53 contacts a cam follower 60 connected to the piston rod 37.

The movement of pistons 29 and 35 and piston rods 31 and 37 is purely rectilinear and axial. The bores of the respective cylinders maintain the movement of the pistons rectilinear and axial, and the means for assuring that the movement of all portions of the piston rods is rectilinear and axial comprises cup-shaped guides 63 joined in concentric relation with respective piston rods and having cylindrical outer bearing surfaces in slidable bearing contact with cylindrical guide sleeves 65 fixed to the internal sidewalls of housing portions 62 and 64 joined to opposite ends of the housing 40 and respectively connected to the low pressure cylinder 27 and the high pressure cylinder 33 to complete the hermetically sealed casing 23. The interface between members 63 and 65 runs dry, that is, there is no lubricant sump in the compressor. Instead, sleeves 65 are of porous bronze impregnated with polytetrafluoroethylene or any other suitable material, for lubricity. Similarly, pistons 29 and 35 run dry in their respective cylinder bores, but the pistons are encompassed by a plurality of piston rings of polytetrafluoroethylene filled with finely divided carbon, and this lubricating material in conjunction with the radiator fins on the exterior of the cylinders permits the pistons to run dry in the cylinders without excessive buildup of frictional heat. Coil compression springs 67 surround piston rods 31 and 37 and act between stationary portions of the compression stages and the cup-shaped guides 63 to maintain cam followers 58 and 60 of the piston rod assemblies in continuous contact with cam surfaces 56 of respective ball bearing assemblies 55 so that the position of the pistons 29 and 35 is determined by the angular position of respective eccentrics 51 and 53.

Low pressure stage 3 is provided with inlet and outlet valve assemblies 69, communicating with conduits 21 and 7, respectively, while high pressure stage 5 is similarly provided with inlet and outlet valve assemblies 71, communicating with conduits 7 and 9, respectively, these valve assemblies opening the inlet of each stage upon the suction stroke of the piston and closing the inlet and opening the outlet of each stage upon the compression stroke of the piston, in the usual manner. The gas that blows by the pistons is in free communication with all portions of the interior of the casing 23, passages 73 are provided adjacent the mountings of ball bearing assemblies 47.

As best seen in FIGURES 2 and 3, the compressor is mounted on a support 75 by means of two pairs of leaf springs 77 and 79. The leaf springs of the pair 77 are each secured at one of their ends to the support 75 and at the other of their ends to opposite sides of the casing 23 adjacent the low pressure stage end of the compressor, that is, the left hand end as viewed in the drawing. The leaf springs of the pair 79 are connected in a similar manner but in reversed position relative to springs 77, between the support 75 and opposite sides of the casing 23 at the high pressure stage end of the compressor. The leaf springs of the pair 77 have central uniplanar portions 81 of substantial extent that are coplanar with each other, and the leaf springs of the pair 79 have similar central uniplanar portions 83 which are likewise coplanar. The portions 81 and 83 are disposed at acute angles with respect to the plane of the support 75 so that the portions 81 and 83 lie in oppositely inclined planes 85 and 87, respectively, which intersect in a straight line 89 parallel to the plane of the support 75 and perpendicular to and spaced between axes of the piston and piston rod assemblies of the low and high pressure stages. The line 89 passes through the center of gravity 91 of the compressor which lies on or close to a plane passing through the axes of the piston and piston rod assemblies.

In order to counteract the unbalancing effect of the acceleration and deceleration of the pistons and piston rods, the compressor is provided with a plurality of dynamic balancers 93 that act on axes parallel to each other and to the axes of the piston and piston rod assemblies. As shown more clearly in FIGURE 5, each balancer 93 is mounted on a bracket 95 which is rigidly secured to an adjacent compressor stage. Each bracket 95 has an opening 97 therethrough in which is disposed a cylindrical tubular sleeve 99 of less external diameter to provide clearance between the opening and the sleeve throughout the circumference of the sleeve. The sleeve 99 has abutment shoulders 101 at each of its ends and coil compression spring 103 is positioned about the sleeve 99 on each side of bracket 95 and the springs 103 act between the bracket 95 and abutment shoulders 101. The interior of sleeve 99 is screw-threaded, and externally screw-threaded plugs 105 are threadably supported within the sleeve 99. The plugs 105 retain therebetween a block 106 of the proper mass so that each balancer 93 may be tuned or adjusted to balance the compressor assembly.

Balancers 93 are four in number and each is placed as close as convenient to the axis of its respective piston and piston rod assembly. As seen more clearly in FIGURE 3, a pair of balancers 93 are mounted closely adjacent opposite ends of the compressor with the balancers of each pair being located on opposite sides of a plane passing through the axes of the piston and piston rod assemblies of the low and high pressure stages. The arrangement is such so that the two balancers shown in FIGURE 2 tend to balance each other, and so that the two balancers shown in FIGURE 5 also tend to balance each other.

The unique compressor structure described above greatly contributes to the maintenance of the operating fluid of the refrigeration system at a very low level of contamination. The provision of dry lubricated bearing surfaces and the use of sealed ball bearings obtain the result of practically no contamination by lubricants of the gas within the casing of the compressor. This makes it possible to continuously utilize piston blow-by as a portion of the operating fluid. Recycling blow-by to the low pressure stage intake makes it possible to operate the device for long periods of time without renewing the charge of operating fluid. This, in turn, makes it possible to provide a refrigeration cycle for use in environments in which the compressor must remain untended for long periods of time, such as in very high altitude vehicles for example. Moreover, the preservation of the initial charge intact for long periods of time eliminates the need for incorporating recharging equipment in the refrigeration system of the present invention, and thus further aids in miniaturizing the assembly.

The compressor is designed for operation at relatively high speeds and it is essential for long life to properly balance the compressor and eliminate undesirable oscillations and vibrations. Also, in accordance with the present invention the compressor is employed in a system including sensitive mechanical or electronic devices and it is essential that operation of the compressor does not produce vibrations that would damage or affect the sensitivity of such devices. This is accomplished in accordance with the present invention by the combination of specific details of the running gear of the compressor and the mounting means of the compressor. The compressor is designed so that the counterweights on the eccentrics need balance only the eccentrics and the associated ball bearing assemblies; the piston rods have no lateral component of motion imparted to them by the eccentrics, but only axial movement. As a result, the locus of the center of moments of each eccentric and its associated ball bearing assembly is circular, thus enabling the complete dynamic balance of each eccentric and ball bearing assembly by means of a single counterweight of predeterminable shape and mass. As the eccentrics and ball bearing assemblies can be completely dynamically balanced by the counterweights, it follows that eccentrics and ball bearing assemblies of any desired mass or eccentricity can be used, and that an appropriate counter-

weight can be chosen which will attain complete dynamic balancing in all positions. In the illustrated embodiment, the eccentricities of the eccentrics at opposite ends of the drive shaft are equal, so that the stroke of both pistons is the same. However, if desired, it is of course possible to make the pistons of the several pressure stages more nearly the same size and obtain the desired difference in displacement by making the eccentrics of different eccentricities.

As the counterweights can perfectly counteract what would otherwise be the unbalancing effect of the eccentrics and their associated ball bearing assemblies, the only other moving parts that could unbalance the compressor are the piston and piston rod assemblies. This latter unbalancing influence, however, is taken care of in a combination of two ways. In the first place, the leaf springs on which the compressor is mounted have portions **81** and **83** of substantial extent disposed in planes that intersect in a line **89** on which the center of gravity of the device lies and which line is also perpendicular to and spaced between the reciprocatory axes of the piston and piston rod assemblies. As a result, the compressor will oscillate and can oscillate only about line **89**, because the center of gravity of the device lies on line **89**, because the axes of pistons **29** and **35** are disposed on opposite sides of and are perpendicular to line **89**, and because the planes of the uniplanar portions **81** and **83** of the leaf springs on which the compressor is mounted intersect in line **89**. Thus, the compressor has a tendency to vibrate or rapidly oscillate when in use, but this vibration is entirely oscillatory about line **89**. As a result of the disposition and arrangement of the leaf springs on which the compressor is supported, the imposition of any forces other than forces parallel to or coincident with the axes of the piston and piston rod assemblies will not be able to move the compressor. Accordingly, the compressor is rigidly mounted except for its freedom to oscillate about line **89**.

In the second place, the dynamic balancers **93**, by acting along axes parallel to and closely adjacent their respective piston and piston rod assemblies, can exactly counteract the unbalancing effect of the piston and piston rod assemblies.

In this way, and for the first time, the running gear of a compressor has been completely and accurately balanced by breaking down the running gear into two dynamically independent system: the eccentrics and their associated ball bearing assemblies, and the piston and piston rod assemblies, and then providing distinctively different types of balancing equipment thereby precisely to balance each of the two major portions of the running gear.

The low contamination level of the compressor makes it possible not only to return piston blow-by to the intake of the low pressure stage, but also to discharge from the compressor a highly compressed gas at an unprecedentedly low contamination level. As a result of this extremely low contamination level, filters and adsorbers **11** and **13** can be made much smaller, for they need contend only with greatly reduced quantities of contaminant. Also, the very low contamination level of the operating fluid as it leaves the compressor make it possible to use the same filter and adsorber units for long periods without switching or changing them, and this in turn is another advantage when the refrigeration system must operate unattended for long periods of time, as noted above.

The miniaturization of the system does not end with filters and adsorbers **11** and **13**, for the very low contamination level of the working fluid also makes it possible to miniaturize the liquefier **17**. In the past, liquefiers for low boiling point gases could not be made very small, for they were subject to plugging with contaminants from the working fluid. But a feature of the present invention resides in the fact that a new miniaturized liquefier has been devised which can be operated for much longer periods of time wtihout plugging than was heretofore possible. The form of liquefier by which this new feature of combination is achieved is more particularly shown in FIGURE 6 of the drawings.

Referring now to FIGURE 6, the liquefier includes a head **107** to which inlet and outlet conduits **15** and **21**, respectively, are secured. An inlet opening **109** in registry with inlet conduit **15** and an outlet opening **111** in registry with outlet conduit **21** both extend through head **107**. An inner cylindrical shell **113** and an outer cylindrical shell **15** are in concentric spaced apart relationship and are both secured at one of their ends to the head **107** and are both closed at their other ends. The space between shells **113** and **115** is evacuated. The bottom of shell **115** is closed by a transparent window **117** of a material such as sapphire so that component **19**, which is an infra-red cell in FIGURE 6, may "see" its environment.

A central mandrel **119** has a cylindrical outer contour concentric with and disposed within inner shell **113**. An extremely long, generally helically arranged heat exchange tube **121** extends from inlet opening **109** and is wrapped about mandrel **119** in a multiplicity of turns and terminates at its other end adjacent the closed end of shell **113** which forms a liquid receiving space **123**. The working fluid flowing through the tube **121** is cooled by countercurrent heat interchange with relatively cold working fluid as described below and the pressure drop through the length of tube **121** and upon emerging from the tube **121** further cools the working fluid by adiabatic expansion to partly liquefy the working fluid. The liquefied working fluid collecting in the space **123** cools electronic compartment **19** which is contiguous to the cold end of shell **113**. Component **19** performs its functions at the temperature of the liquefied working fluid and is connected with the exterior of the liquefier through wires **125** that extend through outer shell **115** through a hermetic seal **127** which may be of plastic, ceramic or metallic composition.

It is especially important to note, in connection with liquefier **17**, that the heat exchange tube **121** is nonorificed, that is, it discharges working fluid at its open end into the space **123** without causing the working fluid to pass through a marked constriction adjacent the point of discharge. Indeed, the inside diameter of the tube **121** adjacent and at the discharge end of tube **121** is not substantially different from the inside diameter of tube **121** for a substantial distance back from the discharge end. As a consequence a substantial proportion of the pressure drop between the inlet and outlet ends of tube **121** is distributed over the length of the tube instead of being all concentrated at an orifice as in prior constructions. The tube is thus quite long relative to its inside diameter, and the velocity of the working fluid through the tube is extremely high. Tube **121** should have a length at least 2,000 times its inside diameter, for example around 5,000 times, and the working fluid should pass through at at least the speed of sound under the conditions of temperature and pressure of the working fluid. The pressure drop through tube **121** may for example be around half the total pressure drop between the inlet and outlet ends of tube **121**, the remainder of the pressure drop being experienced upon the emergence of the working fluid from the open end of tube **121** in evaporator space **123**.

The working fluid flowing through the tube **121** is cooled by countercurrent heat interchange with expanded working fluid which may include unliquefied working fluid discharged from the tube **121** and vapor resulting from vaporization of liquefied working fluid in the space **123**. The expanded working fluid flows through a helical passageway defined by the spaced turns of the tube **121**, the external surface of the mandrel **119** and the inside

surface of the shell 113 and passes through opening 111 to conduit 21 at substantially ambient temperature.

A somewhat modified form of the liquefier is shown in FIGURE 7 in which the component 19' is a small transmitter which does not need to "see" anything, so the closed end of shell 115' is opaque and the space between shells 113' and 115' is filled with a powdered insulating material, for example, pyrogenic silica having a particle size range of about 0.005–0.02 microns, which is available commercially from the Geoffrey L. Cabot Company under the trademark "Cabosil." In this case, component 19' can be placed inside shell 113' in evaporator 123' and can be removable as a unit with the mandrel and tubing. A major significance of the liquefier of FIGURE 6 or 7 in the refrigeration system of the present invention is that it avoids plugging at contamination levels at which conventional liquefiers would plug with contaminants. Specifically, it has been found that plugging in the usual type of liquefier having an orifice at the cold end of the heat exchange tube of a diameter less than the inside diameter of the heat exchange tube occurs adjacent the orifice and is due to the presence of the orifice. Elimination of the orifice in the liquefiers of the present invention allows the liquefiers to function for much longer periods of time without plugging than would a liquefier in which working fluid having the same contamination were passed through an orifice.

Another factor that contributes to plugging of liquefiers is the diameter of the tube. The larger the diameter, the longer the liquefier can operate on working fluid of a given contamination level. By providing a nonorificed liquefier according to the present invention, it has now become possible to miniaturize the liquefier and still keep it operating for long periods of time without plugging with contaminant.

It might at first be thought that the solution to the problems of contamination in low temperature refrigeration systems is completely to eliminate contaminants from the working fluid prior to charging the system. However, such is not the case, for as a matter of fact it is impossible to prevent the system from adding contaminants to itself. Specifically, the parts of the system continuously add carbon dioxide and hydrocarbons to the refrigerant, despite all precautions that can be taken.

For example, the metal walls of the system and the motor windings exposed to the refrigerant are continuously "out-gassing," that is, emitting gas. The metals emit carbon dioxide and the motor windings emit both carbon dioxide and hydrocarbons despite the fact that they may have been specially selected for low out-gassing. The motor shaft bearings and eccentric bearings are lubricated with grease, and the grease performs its lubricating action by releasing oil from a soap bond. This oil partially vaporizes, and the vapor joins the refrigerant stream despite all efforts to provide adequate seals about the lubricated bearings. Indeed, hydrocarbons from this source are the primary source of contamination of the system. Moreover, hydrocarbon contaminants inherently tend to give greater difficulty than does carbon dioxide, for they solidify at higher temperatures than carbon dioxide and they become more and more viscous until they solidify and thus have more tendency to cause plugging. Indeed, it is estimated that even with the best compressor and motor design, as in the present invention, hydrocarbon contaminants are still added to the refrigerant passing through the crankcase at a rate such that the contaminants in the compressor discharge at about two parts per million.

Moreover, hydrocarbonaceous contaminants render objectionable those quantities of carbon dioxide which otherwise would be harmless. The carbon dioxide tends to precipitate as snow, which to some extent is blown through the system. But hydrocarbon contaminants tend to narrow the channels through which the carbon dioxide snow may pass, whereupon the carbon dioxide contaminant plugs the equipment at levels of carbon dioxide content which would be tolerable in the absence of hydrocarbons.

Just as the removal of contaminants from the charge is not a complete answer to the problem of contamination, so also the provision of adsorbers in the cycle is not a complete answer. Adsorption is a surface phenomenon, and the quantity of contaminant that can be adsorbed is a function of the size of the total surface of absorbent. The contaminant is adsorbed on the adsorbent to a depth substantially no greater than one monomolecular layer of adsorbed contaminant. The need to provide enormous surfaces of absorbent results in adsorption equipment that is quite large. In a closed cycle system, the volume of adsorbent may be and quite often is several hundred times the volume of the closed cycle working fluid. However, the special means of the present invention for reducing contamination of the working fluid, coupled with the special liquefier of the present invention, makes it possible first to keep the contamination level low and second to accommodate that low contamination level for remarkably long periods without plugging the liquefier. At the same time, the low contamination level makes it possible to use an absorber greatly reduced in size. Indeed, remarkably low ratios of adsorber volume to working fluid volume under conditions of standard temperature and pressure of less than about 5% and preferably less than about 2% are practical in connection with the present invention, and this of course makes it possible to miniaturize the adsorber as well as the other components of the system. The significance of these small ratios at standard temperature and pressure will be even more apparent when it is realized that most of the refrigerant in the closed cycle is either under very high pressure or at a very low temperature, so that its volume is very much less than under standard conditions. Accordingly, the importance of achieving the lowest possible ratio of adsorber volume to working fluid volume will be clear, especially when it is remembered that according to the practices of the prior art, the adsorber of a miniaturized refrigeration system would have to be larger than all the rest of the system combined. A miniaturized low temperature refrigeration system constructed in accordance with the present invention included a charge of 0.2 s.c.f. nitrogen and an adsorber vessel having a volume of 5.3 cubic inches,

It will therefore be apparent that the various components of the refrigeration system of the present invention coact in a uniquely novel way to provide a miniaturized system that can operate for long periods of time without plugging by contaminant. The compressor contributes to this system in that the pistons run dry without lubricant and the drive shaft has sealed bearings. Contamination of the compressor output by lubricants is thus kept to an absolute minimum. Also, the compressor contributes to this system in that provision is made for returning piston blow-by to the compressor intake. If this blow-by were discharged from the system, it would be necessary continuously to recharge the system, and the recharging equipment would further increase the size and weight of the system.

The low contamination level of the compressor output makes it possible to use very small filter and adsorber units downstream of the compressor. Nevertheless these filter and adsorber units are effective even further to reduce the contamination level of the working fluid, so that the fluid entering the liquefier is unprecedentedly pure.

The liquefier in turn is designed to run for the longest possible time without clogging at a given contamination level, and when this contamination level is made so extraordinarily low by the equipment of the present invention upstream of the liquefier, it is not only possible to run the liquefier for long periods without clogging, but also to make the liquefier quite small.

The feature of the present invention of providing communication between the chamber defined by the hermetically sealed casing and the compressor inlet permits the use of a chamber as an accumulator during startup and thus makes it unnecessary to provide an additional device for this purpose, thereby further decreasing the overall size of the equipment. During startup, when the system is warm, excess gas in the system passes through the conduit **25** to the chamber defined by the hermetically sealed casing and as the system cools down gas as is required flows from the chamber through the conduit **25** in the opposite direction to the compressor inlet and when normal operating conditions are reached gas flowing from the chamber through conduit **25** comprises piston blow-by.

There is thus provided by the present invention a novel compressor and a novel refrigeration system of which the components coact to enable miniaturization of the entire system, and to make the system capable of being used in any bodily orientation for long periods of time without adjustment or maintenance.

Although the present invention has been described and illustrated in connection with preferred embodiments, it is to be understood that modifications and variations may be resorted to without departing from the spirit of the invention, as those skilled in this art will readily understand. Such modifications and variations are considered to be within the purview and scope of the present invention as defined by the appended claims.

What is claimed is:

1. A gas compressor comprising a motor and a cylinder in unitary assembly with each other, a piston reciprocable in the cylinder along a first line of reciprocating motion, a drive shaft drivingly connecting the piston with the motor adjacent one end of the drive shaft on one side of the motor, weight means mounted for reciprocation out of phase with the piston along a second line parallel to said first line, the drive shaft drivingly connecting the weight means with the motor adjacent the other end of the drive shaft on the other side of the motor, and means mounting the compressor for oscillation about a line passing through the center of gravity of the compressor and perpendicular to the common plane of said first and second lines.

2. A gas compressor as defined in claim **1** in which the weight means comprises a second piston reciprocable in a second cylinder in unitary assembly with the motor.

3. A gas compressor as defined in claim **1** in which the mounting means comprises a pair of leaf spring means supporting the compressor one adjacent each end of the drive shaft, each leaf spring means having a portion of substantial extent that is uniplanar, the planes of said uniplanar portions intersecting in said line passing through the center of gravity of the compressor.

4. A gas compressor as defined in claim **1** including a pair of second weight means spring mounted on the compressor one adjacent each said line of movement for limited reciprocatory motion relative to the compressor parallel to said lines of movement.

5. A gas compressor comprising a motor and a cylinder in unitary assembly with each other, a piston reciprocable in the cylinder and having an axially extending piston rod, a rotary drive shaft drivingly connected with the motor, the drive shaft having a first offset portion adjacent one end of the drive shaft on one side of the motor, means acting between the first offset portion and the end of the piston rod opposite the piston to translate rotary motion of the drive shaft into reciprocatory motion of the piston, means limiting movement of the piston rod to reciprocatory motion axially of the piston, weight means mounted for reciprocation parallel to but out of phase with the piston, the drive shaft having a second offset portion adjacent its other end on the other side of the motor, means acting between the second offset portion and the weight means to translate rotary motion of the drive shaft into reciprocatory motion of the weight means, means limiting movement of the weight means to reciprocatory motion of the weight means along a line parallel to the axis of the piston, and means mounting the compressor for oscillation about a line passing through the center of gravity of the compressor and perpendicular to the common plane of the lines of movement of the piston means.

6. The gas compressor as defined in claim **5**, the weight means comprising a second piston reciprocable in a second cylinder in unitary assembly with the motor.

7. A gas compressor as defined in claim **5** in which the mounting means comprises a pair of leaf spring means supporting the compressor one adjacent each end of the drive shaft, each leaf spring means having a portion of substantial extent that is uniplanar, the planes of said uniplanar portions intersecting in said line passing through the center of gravity of the compressor.

8. A gas compressor as defined in claim **5** including a pair of second weight means spring-mounted one adjacent each of the lines of movement of said piston and first weight means for reciprocatory motion relative to the compressor parallel to said lines of movement.

References Cited by the Examiner

UNITED STATES PATENTS

1,508,805	9/24	Shaw et al.	230—58
1,640,634	8/27	Wise	74—604
1,798,846	3/31	Kennedy	248—20
1,855,570	4/32	Edison	74—604
1,859,039	5/32	Joyce	103—111
1,895,508	1/33	Fowler	230—58
2,420,452	5/47	Strachovsky	248—15
2,751,146	6/56	Moseley	230—58
2,783,615	3/57	Gooch et al.	103—174
2,801,596	8/57	Sewell	103—174
3,033,124	5/62	Wilson	103—228

KARL J. ALBRECHT, *Primary Examiner.*

LAURENCE V. EFNER, JOSEPH H. BRANSON, JR., *Examiners.*