Feb. 1, 1966 P. P. RICE ETAL 3,232,524

FLUID COMPRESSOR

Original Filed Aug. 9, 1963 4 Sheets-Sheet 1

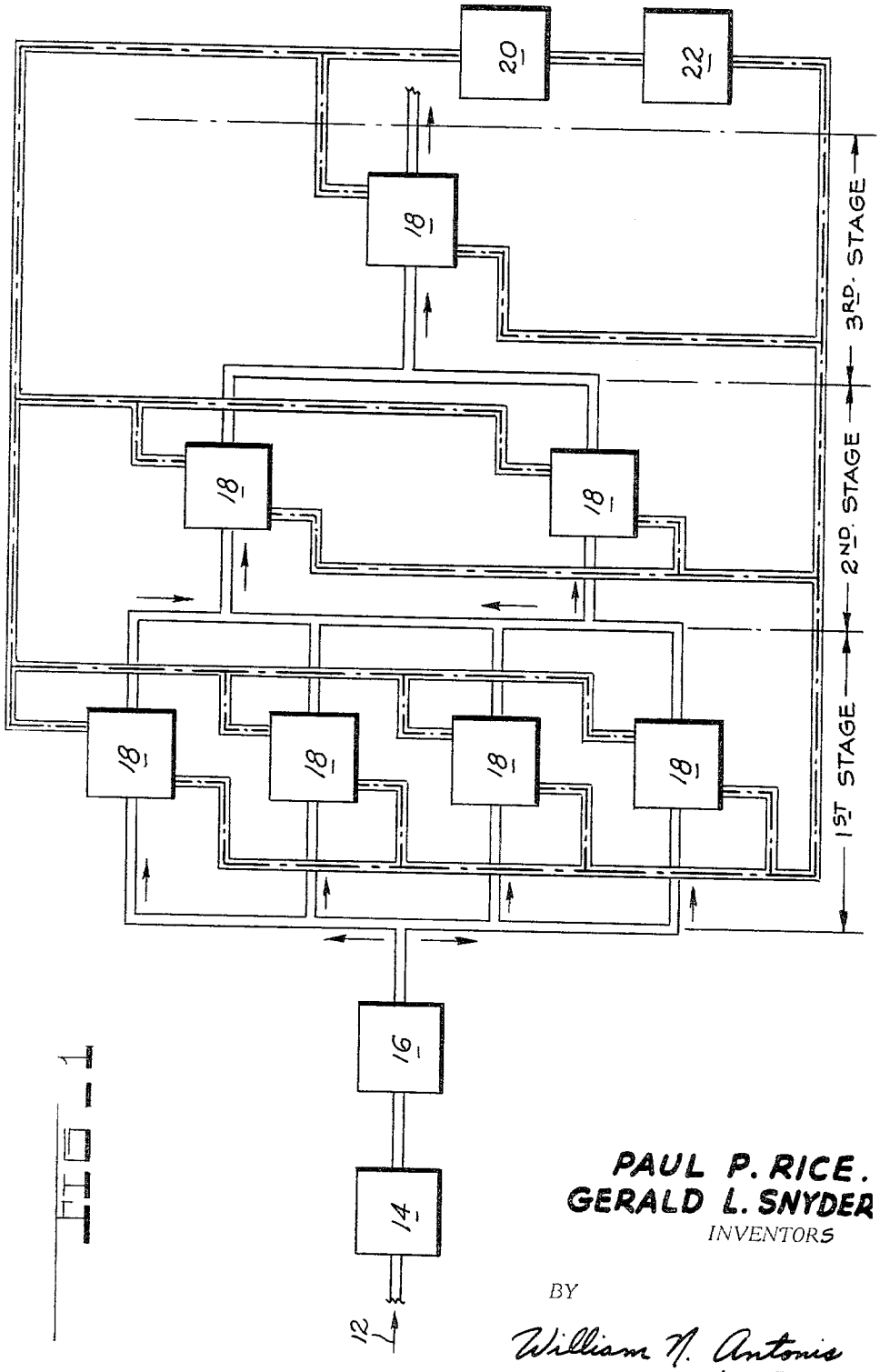


FIG. 1

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Original Filed Aug. 9, 1963 4 Sheets-Sheet 2

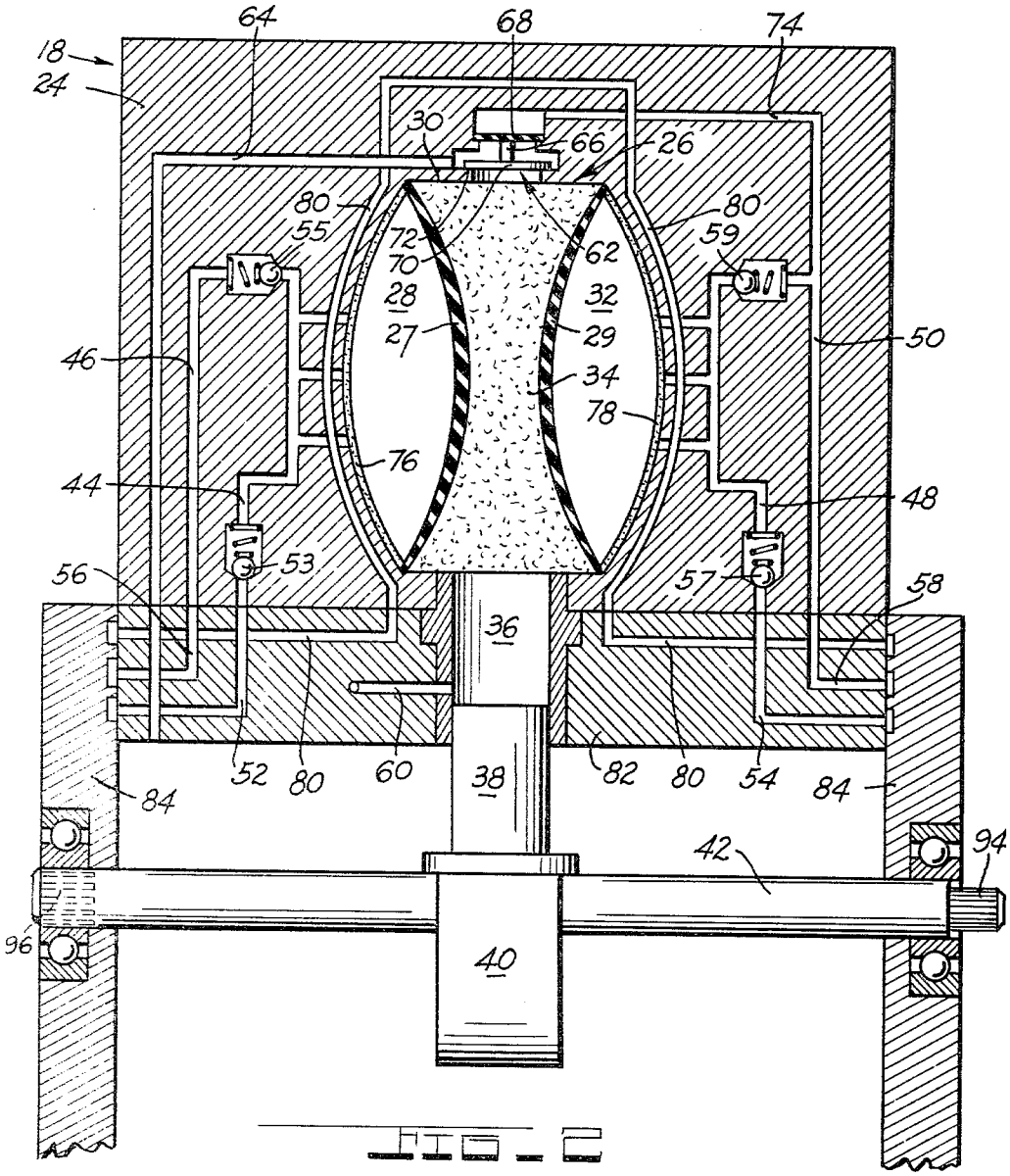


FIG. 2

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FLUID COMPRESSOR

Original Filed Aug. 9, 1963 4 Sheets-Sheet 3

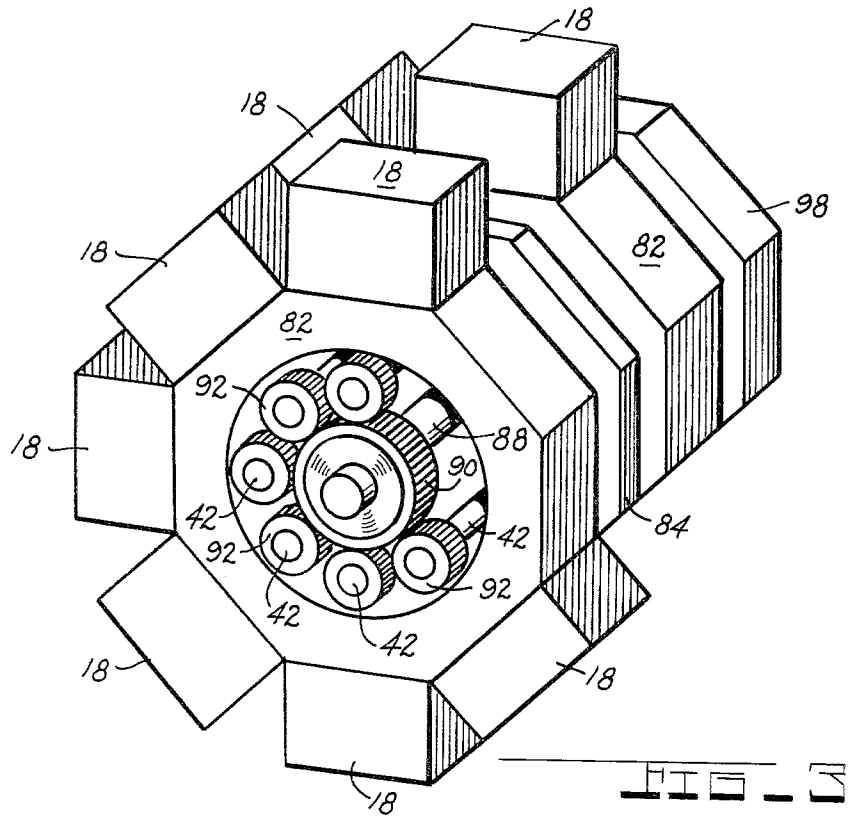


FIG. 3

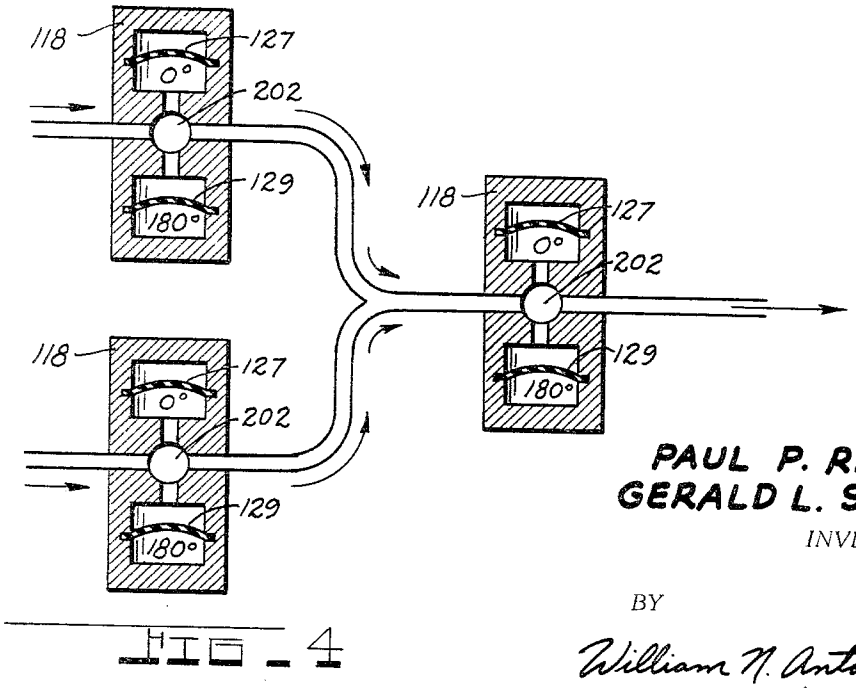


FIG. 4

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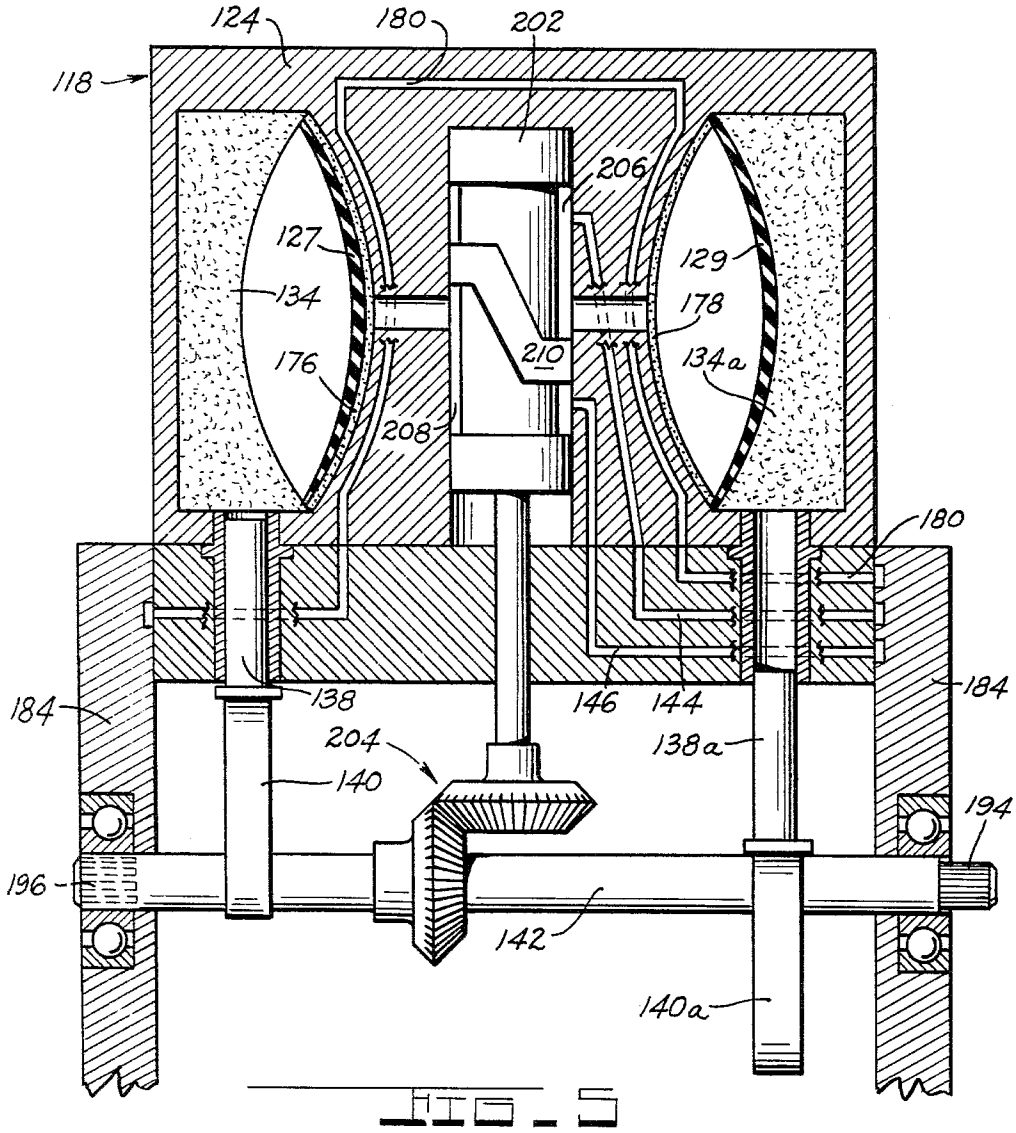


FIG. 5

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United States Patent Office

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FLUID COMPRESSOR

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Original application Aug. 9, 1963, Ser. No. 301,014. Divided and this application Feb. 10, 1965, Ser. No. 431,678

3 Claims. (Cl. 230—162)

This application is a division of our copending application Serial No. 301,014 filed August 9, 1963.

This invention relates to a fluid compressor and more particularly to a high pressure isothermal modular air compressor system.

One of the objects of this invention is to provide a lightweight high pressure gas compressor which can reliably supply high pressure contamination free air or other gas at substantial flow rates.

Another object of this invention is to provide a high pressure gas compressor which is based on a concept of multistage compression wherein a high frequency hydraulic drive is utilized to flex elastomer diaphragms in relatively small chambers, said diaphragms serving as barriers between the hydraulic drive fluid and the gas which is being compressed.

A further object of this invention is to provide a unit which not only has a high volumetric efficiency but which is significantly smaller and lighter in weight than compressors with comparable performance.

A still further object of this invention is to provide a high pressure gas compressor which permits high cycle operation, high volumetric and thermodynamic efficiencies and outstanding size and weight per capacity ratios.

Normal compression techniques involve adiabatic compression with the gas being cooled after each stage of compression. However, the minimum amount of work required for compression is expended when a gas is compressed isothermally. Accordingly, it is an important object of this invention to provide a high pressure gas compressor which compresses and pumps gaseous fluid at a substantially constant temperature through the use of highly efficient porous heat exchanger material which is located within each compression chamber and the use of a circulating coolant through the compressor housing in close proximity to the heat exchanger material.

More specifically, it is an object of this invention to utilize a thin porous metallic liner in the compression chamber through which the gaseous fluid must pass before being discharged from the unit in order to more effectively permit the cooling system to carry away the heat which is generated during the compression cycle.

Another object of this invention is to provide a porous metallic liner which will permit overpressurization of the diaphragm without danger of rupturing the diaphragm or causing it to be extruded into the inlet and outlet valve ports communicating with the pumping chamber.

A further object of this invention is to provide a high pressure gas compressor which utilizes a plurality of substantially similar modules arranged in a predetermined parallel-series arrangement to achieve a desired volumetric compression ratio, said modules being packaged in a polygonal arrangement in order to achieve the desired increase in pressure without utilizing external piping or fittings between compression stages.

The above and other objects and features of the invention will become apparent from the following description of the mechanism taken in connection with the accompanying drawings which form a part of this specification and in which:

FIGURE 1 is an illustration of a typical module parallel-series flow diagram utilized in connection with the invention;

FIGURE 2 is a sectional view of a fluid compressor constructed in accordance with the present invention;

FIGURE 3 is a partially exploded diametric view which illustrates the packaging concept of this invention;

FIGURE 4 is an illustration of a typical two-phase module flow diagram wherein dual pumping chambers within a module are driven 180° out of phase; and

FIGURE 5 is a sectional view similar to that of FIGURE 2 which illustrates a two-phase module which could be utilized in the flow diagram illustrated in FIGURE 4.

Referring to the drawings, it will be seen that the high pressure compressor system includes three major sections, namely, the compression section, the drive mechanism, and the cooling section. The compression section draws in a gas, for example, air at ambient pressure, and boosts it to a given desired pressure. In addition the compression section includes the requisite blowers, filters and purifiers to cleanse the air, and compressors of the diaphragm type. The drive mechanism includes a hydraulic drive system for each compressor, the necessary drive shafts and a source of power such as an electric motor. The cooling section is a closed-loop system utilizing water or other type of refrigerant, a pump and the requisite fluid passageways for circulating the refrigerant through the compressors.

The flow diagram illustrated in FIGURE 1 shows an ambient gaseous fluid input indicated by the arrow 12, an optional blower 14 which could constitute the first stage of compression, and a suitable cold trap and/or purifier 16 for eliminating moisture, oil vapor or any other contaminants from the gaseous fluid. The blower, if needed, could be of the centrifugal compressor type. In any event this invention is predominantly concerned with the concept of utilizing identical or similar twin-cylinder module type compressors in a parallel-series arrangement to provide any desired flow rate in conjunction with any desired pressure rise. It will be seen in FIGURE 1 that the first stage of compression includes four identical parallel connected modular type compressors 18, of the type shown in FIGURE 2, discharging into a second stage of compression which includes two more identical parallel connected modular type compressors 18, said second stage compressors in turn discharging into another identical modular type compressor 18 which constitutes the third stage of the system. Since, assuming perfect gas laws, pressure rise is inversely proportional to stage volumes under constant temperature conditions, this particular parallel-series arrangement which is illustrated will provide an overall compression ratio of 8:1, if each module 18 is considered to be of unit volume (V_1). It will be understood that in multistage operations of this type, alternate stages will be driven 180° out of phase. Other arrangements, including multiple phases, multiples of unit volume, and variations in number of modules per stage to provide any desired flow rate in conjunction with any pressure rise are within this concept.

In order to provide a substantially constant temperature a constant flow refrigerated cooling system is utilized which includes a pump 20, a condenser 22 and the requisite conduits for flowing the coolant directly through each of the compressors 18 so as to carry off the heat of compression and maintain the gaseous fluid being compressed at a substantially constant temperature.

Referring to FIGURE 2, it will be seen that each of the modular type compressors 18 will include a housing 24 having a chamber 26 therein, and two flexible imperforate diaphragms 27 and 29 peripherally secured in the chamber in a suitable manner so as to separate it into three subchambers 28, 30 and 32. Located between the diaphragms and in subchamber 30 is a porous metallic former 34 which limits flexure of each of the dia-

phragms and also provides contoured support for each of the diaphragms. Within bore 36, which is located in the housing and extends into subchamber 30, is a piston 38 which is movable therein. Movement of the piston is caused by a cam 40 which is suitably connected to and rotated by a secondary power shaft 42. Located in the housing are inlet and outlet passages 44 and 46 which permit ingress and egress of a gaseous fluid to and from subchamber 28, and inlet and outlet passages 48 and 50 which permit ingress and egress of the gaseous fluid to subchamber 32. Suitable inlet passages 52 and 54 and outlet passages 56 and 58 are provided in the center section 82 for connecting a compressor module in parallel or series with another compressor module. Located in each of the inlet and outlet passages 44, 46, 48 and 50 are suitable check valves 53, 55, 57 and 59 which function in a conventional manner.

Located in bore 36 and subchamber 30 is a suitable hydraulic pumping fluid which is confined between the piston 38 and diaphragms 27 and 29 for causing movement of said diaphragms upon reciprocation of the piston. In the event of a deficiency in the quantity of hydraulic pumping fluid, it will be replenished on the suction stroke of piston 38 through conduit 60 which communicates with a lube pump (not shown). Excess hydraulic pumping fluid is permitted to return to the sump (not shown) through a suitable differential pressure bleed valve 62 and bleed passage 64. The differential pressure bleed valve includes a movable element 66 having two spaced discs on its ends, one of which is a resilient diaphragm 68 and the other of which is a metal disc 70 which is seatable on a valve seat 72. Since valve diaphragm 68 communicates with the outlet passageway of the compressed gaseous fluid via passageway 74 and disc 70 is subject to the hydraulic fluid pressure in subchamber 26, the bleed valve 62 will be unseated to permit bleeding of excess hydraulic fluid only when the hydraulic pressure rises a predetermined value above the gaseous pressure.

Located within each of the subchambers 28 and 32 are porous metallic liners 76 and 78 which assist in the extraction of heat from the gaseous fluid flowing therethrough during compression of the fluid. The porous liners may be formed of sintered powdered metal and should be of good conductivity. Such a liner will have more surface area and will create greater turbulence than conventional bare or fin extended surface exchangers of the same package size. Because of such increased surface area and turbulence it is possible to achieve increased heat transfer within the same envelope or conversely a smaller envelope for the same heat rejection rates. Located within the housing and in close proximity to each of the liners are passages 80 which carry a coolant such as water or other refrigerant for carrying off the heat of compression from the gaseous fluid to thereby maintain same at a substantially constant temperature. In addition to serving as heat exchangers, the liners 76 and 78 also serve to limit flexure of the diaphragms 27 and 29 and provide contoured supports therefor which prevent extrusion of the diaphragms into the plurality of outlet ports communicating with the compression chambers. Although for drawing purposes the volumetric capacity of the passages between the inlet and outlet check valves is shown in a somewhat exaggerated manner, it should be understood that this volume would in actual practice be as small as practicable.

The operation of the unit shown in FIGURE 2 will be as follows: With the hydraulic piston 38 in its bottom position, the diaphragms 27 and 29 will be in contact with the center porous former 34, as shown in the drawing. With the diaphragm in this inward position subchambers 28 and 32 will be filled with the gaseous fluid transmitted through inlet passageways 44 and 48. As the piston 38 moves in an upward direction, the hydraulic fluid in bore 36 and subchamber 30 will flow through the porous center former 34 and will force the diaphragms

from the inward position, which is shown, to an outward position in which the diaphragms will be in contact with the porous liners 76 and 78. As the gaseous fluid is being compressed into the voids of the porous liner, a portion of the heat of compression is rapidly conducted away. The large surface area ratio of the porous liner to its void volume permits rapid transfer of heat without undue sacrifice of clearance volume with associated loss of volumetric efficiency. As the outlet check valves 55 and 59 open, the gas will flow through the porous liner being further cooled. Since a refrigerant is passed through the compressor housing, the entire module will act as a heat sink. Downward motion of the piston 38 will draw gas into the subchambers 28 and 32 and the cycle will be repeated.

Variations of hydraulic volume due to tolerances, leakage temperature changes, etc., could cause either under or over hydraulic fluid displacement, thereby causing, respectively, reduced pumping efficiency or excessive overpressure conditions. In order to compensate for these variations, piston displacement is deliberately oversized by a small amount over the nominal subchamber displacement to insure that the diaphragms will bottom out on their respective porous liners. Since the hydraulic pressure will rise above the gas pressure when the diaphragms bottom out, excess hydraulic fluid will be relieved through the differential pressure bleed valve 62 in the manner previously described for return to the sump. As previously stated, makeup hydraulic fluid is provided by a lube pump through conduit 60 which is opened when the piston 38 is in its bottom position. Thus, this hydraulic fluid circuit provides automatic fill and bleed characteristics.

One of the major advantages of this invention resides in the overall packaging concept which is shown in FIGURE 3. Referring to this figure, it will be noted that a polygonal shaped center section 82 provides a framework for the plurality of compressors 18 each of which is suitably connected to a face thereof, and also provides the necessary porting for fluid passageways as shown in FIGURE 2. The combination of one center section with its compressor modules may be defined as a unit. Units can be stacked in any reasonable number to provide flexibility or increased capability. Separator plates 84, as shown in FIGURES 2 and 3, provide means for routing fluids from module to module and unit to unit. Plugs may be utilized in the passageways of the separator plates to readily select parallel or series circuitry. Thus, the separator plates 84 in conjunction with the modules 18 and center sections 82 can contain all the fluid passageways, thereby eliminating all external plumbing and fittings.

A secondary power shaft 42 is provided for each module 18 to cause reciprocation of piston 38 via cam 40. A single primary power shaft 88 is centrally located for driving the secondary power shafts through a common gear 90 which meshes with gears 92 operatively connected to each of the secondary power shafts. Power may be transmitted to succeeding stacked units by means of male and female splines 94 and 96 on the shafts. In conformance with the parallel-series and multiple phasing concept, the mesh of the gears and splines will provide means for readily timing the desired phase relationship between stages. An end bell 98 may be provided for closure of the last stacked unit. It should be understood that this packaging concept permits the attachment of a refrigeration pump to the center section 82 in the same manner as the compressor modules 18. Such a refrigeration pump can be identical with or similar to the compressor modules 18.

FIGURES 4 and 5 show another mode of operation wherein the diaphragms 127 and 129 are driven 180° out of phase by two hydraulic pistons 138 and 138*a* which are controlled by 180° out of phase cams 140 and 140*a*. Since the components and operation of this embodiment are essentially the same as the previous embodi-

ment, like parts are identified with like numerals plus 100. It will be noted that only one intake-exhaust valve is necessary for this type of an arrangement and that a timed rotary valve **202** is shown for this purpose. The rotary valve is suitably connected to a secondary power shaft **142** through suitable gearing **204** which will cause timed rotation of the valve so that inlet and outlet channels **206** and **208**, which are separated by land **210**, will sequentially communicate the inlet and outlet gaseous fluid passageways **144** and **146** with the compression side of the diaphragms **127** and **129**. Although the gas and hydraulic sides of the diaphragms have been structurally reversed, this embodiment still utilizes porous formers **134** and **134***a* and porous heat exchanger liners **176** and **178** which are in close proximity to refrigerated coolant passage means **180**. Variations in hydraulic fluid volume, although not shown, would be compensated for in the same manner as shown in the FIGURE 2 embodiment. A typical two-phase module flow diagram, which would utilize modules of the type shown in FIGURE 5 wherein dual pumping chambers within a module are driven 180° out of phase is shown in FIGURE 4. Since the compression and cooling cycles of the FIGURE 5 embodiment are essentially the same as that of the FIGURE 2 embodiment, the operation of these cycles will not be described again.

The several practical advantages which flow from our high pressure isothermal modular air compressor and system therefor are believed to be obvious from the above, and other advantages may suggest themselves to those who are familiar with the art to which this invention relates.

Furthermore, although our invention has been described in connection with certain specific embodiments, it will be obvious to those skilled in the art that various changes may be made in the form, structure and arrangement of parts without departing from the spirit of the invention. Accordingly, we do not desire to be limited to the specific embodiments disclosed herein primarily for purposes of illustration, but instead desire protection falling within the scope of the appended claims.

Having thus described the various features of the invention what we claim as new and desire to secure by Letters Patent is:

1. In a high pressure modular gaseous compressor system, a polygonal shaped center section having a central bore therein, a plurality of hydraulically actuated compressors operable in a predetermined sequence, each of said compressors including a housing which is fixedly connected to one of the faces of the center section, flexible diaphragm means located in each of said housings for dividing the interior thereof into two chambers, the first of which is adapted to receive a gaseous fluid to be compressed and pumped and the second of which is adapted to receive a confined hydraulic fluid for causing flexure of the diaphragm to thereby effect compression and pumping of said gaseous fluid, reciprocating piston means located in each of said housings for producing pulsations in said actuating hydraulic fluid and causing flexure of the associated diaphragms, rotatable cam means operatively connected to each of said piston means for causing reciprocation thereof, a plurality of secondary power shafts located in said central bore, one of which is operatively connected to each of said cam means for causing rotation thereof, a single primary power shaft located in said central bore and operatively connected to each of said secondary power shafts for causing rotation thereof, and inlet and outlet passage means located in said housings and said center section for connecting the gaseous fluid chambers of some of the compressors in parallel and of other compressors in series.

2. A high pressure modular gaseous compressor system, as defined in claim 1, wherein said system includes an integral refrigeration pump having a housing which is fixedly connected to one of the faces of the center sections and passage means located in said center section and compressor housings for circulating a refrigerated coolant capable of extracting the heat of compression from the gaseous fluid being pumped to thereby maintain said gaseous fluid at a substantially constant temperature.

3. In a high pressure modular gaseous compressor system, a plurality of structurally interconnected hydraulically actuated compressors operable in a predetermined sequence, each of said compressors including a housing, a flexible diaphragm in said housing dividing the interior thereof into two chambers, the first of said chambers being adapted to receive a gaseous fluid to be compressed and pumped and the second of said chambers being adapted to receive a hydraulic fluid for actuating the flexible diaphragm to effect compression and pumping of said gaseous fluid from its associated first chamber, reciprocating piston means for producing pulsations in said actuating fluid and causing flexure of said diaphragm, rotatable cam means for causing reciprocation of said piston means, a secondary rotating power shaft operatively connected to said cam means for causing rotation thereof, a primary power shaft operatively connected to each of the secondary power shafts associated with each compressor for transmitting motion thereto, and inlet and outlet passage means for connecting the gaseous fluid chambers of some of the compressors in parallel and of other compressors in series.

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