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COMPRESSOR

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1 Claim

ABSTRACT OF THE DISCLOSURE

A compressor, designed chiefly for air conditioning and/or refrigeration, is operated from a rotary shaft through a Scotch yoke.

The Scotch yoke drive makes possible a compact arrangement of cylinders in which the operating stroke is made of the order of four times as great as the cylinder diameter. The drive shaft can operate at slow speed without sacrifice of output, a feature which greatly reduces wear and noise.

The ratio of maximum to minimum working space is very high, of the order of 350 to 1 or even more, and the efficiency is correspondingly high. The pistons and yoke combine to form a long, continuous, rigid structure which maintains perfect alignment of pistons with cylinders, so that very fine tolerances may be provided for sealing against leakage of gas past the pistons. This effect is promoted by providing inner anti-friction facing for the cylinders, and neoprene U-rings in ring grooves of the pistons.

This invention relates to compressors in general, but more particularly to compressors for use in refrigerating and/or conditioning systems. The invention has for its primary object to provide a compressor which is better adapted to the requirements of the indicated service than any structure previously utilized or proposed.

Among the important advantages achieved by the novel compressor are:

(1) A light, simple, rugged, compact and efficient structure is provided, involving few parts and low labor costs, and permitting easy installation and servicing.

(2) The compressor provides for a high rate of output at low driving speed, with consequent reduction of friction, wear and noise, and of frictionally developed heat.

(3) The compressor provides for a precise and stable maintenance of alignment of pistons with cylinders, with low lateral pressure, so that very fine tolerances are practical, and increased assurance against leakage of gas past the pistons is realized.

(4) The compressor provides assurance against the condensation of gas in objectionable quantities in the cylinders as a result of the cooling of the compressed gas at condensing pressure and temperature after shut-off.

(5) Absolute protection is provided against the occurrence of damage by the accumulation of condensed gas in the crank case.

(6) An improved heat dissipating capacity of the compressor cylinders is realized in relation to the work performed.

In a preferred embodiment of the invention for attaining the foregoing advantages, the compressor is made to include two aligned pistons both of which are made rigidly unitary with a piston driving Scotch yoke or cross-head, and through the Scotch yoke are made rigidly unitary with each other. The pistons are disposed in opposed coaxially arranged cylinders.

The piston stroke and each cylinder are made several times as long as the piston diameter—a feature of primary significance. This high length-to-diameter ratio of the

cylinders is made possible because the Scotch yoke obviates the need for connecting a piston to a driving crank through the conventional wrist pin and oscillating piston rod. With the Scotch yoke drive the stroke may be increased without unduly increasing the space between cylinders.

The Scotch yoke regulates the stroke precisely, so that dimensions can be chosen which cause each piston to be driven almost into contact with the structure at the closed end of its cylinder.

Inlet and outlet ports are provided through the closed cylinder ends, rather than through the sides of the cylinders, in association with inlet and outlet valves which require a minimum of dead space.

The crank pin and Scotch yoke cannot materially tilt the pistons nor subject them to friction-producing lateral pressure. The Scotch yoke, together with the pistons, provides a continuously rigid structure having a combined length considerably greater than two long working strokes, and since the distal ends of the pistons, at least, always fit snugly in their respective cylinders, and there is always a total piston length in the two cylinders greater than the full length of a stroke, the two pistons constantly cooperate in maintaining both piston axes in exact coincidence with the aligned cylinder axes. This manner of securing and maintaining alignment makes it feasible to provide closer tolerances between the pistons and the cylinders than has heretofore been possible. In fact, no measurable clearance is required.

Closer tolerances help in blocking the leakage of gas past the pistons and into the crank case. This effect is promoted by providing the cylinders with continuous, substantially friction free, Teflon liner sleeves, by making each piston the full internal diameter of the cylinder for a length as great as the piston stroke, and by providing O-rings or U-rings of neoprene in circumferential ring grooves of the pistons.

The purpose of the compressor is to increase the pressure of the refrigerant gas to a point at which it will be liquefied when cooled to a temperature below, or in any event, not substantially above, atmospheric temperature, depending on the means utilized for cooling the condenser. When operation of the compressor is arrested before the termination of a stroke, gas which has been compressed to the discharge pressure, but which remains in the cylinder, ought not to be allowed to liquefy as it cools. This is prevented by utilizing a driving mechanism for the input shaft which can be driven backward for a fraction of a stroke by the trapped compressed gas until the diminishing pressure of the gas in the cylinder whose compression stroke was interrupted is substantially balanced by the increasing pressure of the gas in the cylinder whose intake stroke was interrupted. This lowers the pressure of the trapped gas well below a liquefying pressure. All possible damage to the equipment which might result from the presence of the liquefied gas in a cylinder is thereby positively avoided.

There will inevitably be some leakage of gas past the pistons in the crank case, and the crank case must therefore be made airtight. In compressors of the prior art, there have been instances of the crank case becoming completely filled with condensed refrigerant gas, or so nearly filled that during the operating cycle, because of variation of the combined capacity of the crank case and the adjoining cylinder ends, the liquefied gas has suddenly opposed an irresistible force to the completion of a stroke, causing grave damage to the compressor. There is no reason to anticipate accumulation of liquefied gas in any such quantity in the crank case of the present compressor. In any event, the capacity of the crank case space never varies, so that no damage would result even if the crank case did become filled with condensed gas.

Other objects and advantages will hereinafter appear. In the drawing forming part of this specification

FIG. 1 is a diagrammatic view of an illustrative refrigerating or cooling system in which the novel compressor is incorporated;

FIG. 2 is a view in side elevation of an illustrative compressor which embodies features of the invention;

FIG. 3 is a view partly in bottom plan and partly in section, of the compressor of FIG. 2, the section being indicated by the staggered line 3—3 of FIG. 2;

FIG. 4 is a fragmentary sectional view on a larger scale than FIGS. 2 and 3, showing the right hand piston and cylinder with the piston at its extreme right hand limit of movement;

FIG. 5 is a view in elevation of the inner face of a valve and valve plate assembly employed at the outer end of each cylinder; and

FIG. 6 is a view in elevation of the outer face of the valve and valve plate assembly of FIG. 5.

Referring first to FIG. 1, the refrigerating or cooling system is shown as comprising a two cylinder compressor 10, having a drive shaft 12 through which pistons are driven in opposed cylinders 14 and 16. The two pistons are always in exactly opposite phase, one executing an intake stroke as the other executes a compression stroke.

The two pistons draw in refrigerant gas (generally a Freon) from the low pressure side of the system through intake pipes 18 and 20, respectively, and discharge the compressed gas into the high pressure side of the system through output pipes 22 and 24, respectively.

Compressed gas, whose temperature has been raised by compression, is transmitted through pipes 22 and 24 to a pipe 26, and is delivered by pipe 26 to a condenser 28. The condenser is externally cooled by any suitable means, to a temperature at which the compressed gas is liquefied. From the condenser 28 the liquefied gas passes through a pipe 30 to a storage receiver 32.

The condensed gas is retained in the high pressure side of the system by a thermostatically controlled valve 34 until it is required to be utilized for cooling or refrigerating service, whereupon it is progressively released to the low pressure side of the system for evaporation in a heat exchanger 36. From the heat exchanger 36, the evaporated gas is drawn through a pipe 38 and thence, in alternation, through intake pipes 18 and 20 into the cylinders 14 and 16 respectively. The thermostatic means which controls the valve 34 also controls the driving means for the shaft 12. When the valve 34 is closed the shaft driving means is idle, and when the valve 34 is opened the shaft driving means is activated. The novelty of the present invention lies in the compressor, itself, and in the cooperation of the compressor with the usual features of conventional systems.

Any suitable driving means may be provided for the shaft 12, provided it does not interfere materially with the driving backward of the shaft for a fraction of a turn by either piston when the driving means has been deactivated. The drive shaft 12 passes into a casing 40 through a lateral wall 42 thereof, being supported externally and internally of the casing by rugged bearings 44 and 46. The shaft 12 has fast upon it a crank arm 48 which carries a crank pin 50 upon its outer end.

The crank pin 50 is received in the slot 52 of a rigid and substantial Scotch yoke 54 for driving the Scotch yoke from end to end of the casing 40. The Scotch yoke 54 has fast with it two aligned pistons 56 and 58 which play in the aligned, opposed cylinders 14 and 16, respectively.

The casing 40, which is desirably composed of aluminum, or of a magnesium-aluminum alloy, consists of a five sided rectangular box 60 and an end closure plate 62, the plate being affixed to the box by bolts 64 in an airtight manner. One of the cylinders, 16, is attached to the closure plate 62 by bolts 66, while the opposed cylinder, 14, is secured to the box wall opposite the plate

62 by bolts 68. The cylinders are of identical construction, so that a description of one, say 16, will suffice for both.

The cylinder 16 comprises a thick-walled sleeve portion which includes an external circumferential flange at its inner end, through which flange the attaching bolts 66 pass. The sleeve 70 is desirably composed of aluminum or of a magnesium-aluminum alloy, and is grooved circumferentially to provide a multiplicity of fins 72 for increasing the heat dissipating capacity of the structure. At the outer end of the sleeve 70 a valve plate 74 and a cap 76 are attached to the sleeve by headed bolts 78.

Nipples 80 and 82 are provided externally of the sleeve 70 at the top and bottom sides of the sleeve, respectively, and near the outer end of the sleeve, for connection with intake pipe 20 and with the discharge pipe 24. L-shaped passages 84 and 86 extend through the nipples 80 and 82 and into the thick wall of the sleeve 70. These passages turn outward and extend through the outer end of the sleeve. The passage 84 communicates with a passage 88 in the plate, in which a wire mesh strainer 90 is lodged, and the passage 86 communicates with a passage 89 in the valve plate. The cap 76 is divided into upper and lower chambers 92 and 94 by a partition wall 96 which bears continuously against the valve plate 74, for sealing the chambers against communication with one another. A suitable gasket, not shown, may be provided between the valve plate and the cap. Other gaskets, not shown, may be included where useful. The passage 88 communicates with the upper chamber 92.

A very thin, flat, resilient, metallic reed valve 98, fixed at its lower end upon the inner face of the valve plate by pins 100 (one shown), normally lies smoothly and snugly in contact with the inner face of the valve plate, covering and sealing an intake passage or port 102 through the valve plate 74, to bar the escape of gas from the cylinder into the chamber 92 on the compression stroke. On the intake stroke the reed valve 98 yields, permitting gas to enter the cylinder from the chamber 92.

A plate valve 104, located in the lower chamber 94 of the cap 76, covers and seals a port or passage 106 through the valve plate on the intake stroke, but yields to permit discharge of gas from the cylinder into the chamber 94 on the compression stroke. The plate valve is carried on two headed pins or bolts 108, each of which is surrounded by a compression coil spring 110. An abutment plate 112, impaled on the shanks of both pins and bearing against the pin heads, confines the springs in bearing engagement with the plate valve 104.

With the arrangement shown and described, the valve plate assembly can be readily removed and replaced without disturbing the connections to pipes 22 and 24.

Each cylinder sleeve 70 is provided with an anti-friction lining sleeve 114 which may be of brass or of other material, but is preferably of Teflon, because of the anti-friction properties of the latter. The outer end of each piston is formed with a plurality of circumferential grooves in which U-rings 116, desirably of neoprene, are lodged.

It is important that the maximum to minimum volumetric ratio of each cylinder be made large in the interest of efficiency. Contributing to this result are the long stroke of the piston and the very limited volume of the dead space in the cylinder. In a pump having a seven inch piston stroke and a waste space in the end of the cylinder equivalent to .020 inch of cylinder length, the dead space amounts to no more than $\frac{1}{350}$ of the maximum capacity of the cylinder chamber. If the pressure maintained in the high pressure side of the system is five times that maintained in the low pressure side, all but one-seventieth, or about 1.4 percent of the gas which fills the cylinder at the start of a compression stroke will have been expelled at the conclusion of that stroke.

By contrast, if the stroke were 1.75 inches long, and the waste space amounted to the equivalent of .035 inch

of cylinder length, with a five to one compression ratio, one tenth of the compressed gas would remain in the cylinder to be reexpanded and recompressed after each stroke.

The high ratio of stroke length to cylinder diameter is made feasible by the utilization of the scotch yoke drive. I am not aware, however, of any previous compressor employing a Scotch yoke drive in which the Scotch yoke drive has been taken advantage of to provide an exceptionally long piston stroke as compared to the cylinder diameter. If the conventional oscillating piston rod and wrist pin connection to the piston were used, a four to one ratio of stroke to cylinder diameter would require a very wide spacing of the cylinders from one another, and very long piston rods in order to avoid conflict between the piston rods and the cylinder walls. With the Scotch yoke the cylinders may be spaced from one another only by the amount of the piston stroke plus the thickness of the Scotch yoke.

Where a typical compressor drive shaft may run at twelve hundred revolutions per minute to drive a piston through a stroke equal to the diameter of the cylinder, the present machine, of the same cylinder bore can accomplish more work, more smoothly and quietly and with much less than one-fourth the wear, at three hundred revolutions per minute. This will be evident from the following comparison.

At first glance it would appear that four of the short stroke cycles could be executed with the same energy expenditure as one of the long stroke cycles, and that the amount of gas compressed would be the same.

Four short cycles, however, would require four times as many revolutions of the drive shaft in a given time, and four times as many reversals of direction of the pistons. Each piston would have to be accelerated to the same maximum speed four times as often and four times as fast, with a great waste of energy, and particularly with a wasteful conversion of energy to heat through friction, so there would be much more heat to be dissipated.

The heat dissipating capability of the shorter cylinders, because of their relatively small superficial areas, would be only about one-fourth that of the longer cylinders, so that if only the heat due to the compression of the gas tended to raise the temperature of the compressor, the short stroke compressor would operate at a considerably higher average temperature than the other. The friction produced heat of the short stroke compressor aggravates this situation, forcing the operating temperature of the short stroke compressor still higher.

A cylinder full of gas at a given pressure and a comparatively low temperature, has considerably more mass than the same cylinder full of gas at the same pressure but at a much higher temperature. The result is that the short stroke compressor, operating at four times the rotary speed of the other, delivers the same volume of gas at the same pressure in a given time, but at a much higher temperature. It therefore delivers a considerably smaller mass of gas than the long stroke compressor, and at a temperature which throws a much greater cooling load on the condenser.

The fact that the dead space of each cylinder is so restricted in relation to the stroke in the illustrative long stroke compressor presents a problem which requires special attention. Ordinarily the pressure in the high pressure side of the system will be that which causes the gas to be condensed at the temperature to which it is cooled in the condenser.

If the low side pressure is two atmospheres and the high side pressure is ten atmospheres, the gas in one of the cylinders will stand at ten atmospheres when the compression stroke is four-fifths completed. If the compressor is stopped in that position and the piston cannot back up to relieve the pressure, the gas in the cylinder may, during a period of idleness, cool to the liquefying

temperature for ten atmospheres, and most of it will be condensed.

Assuming that the freon having the formula CCl_2F_2 , which has a molecular weight of 121 is used, the gas under ten atmospheres will weigh about 1210 grams per 22.4 liters, or fifty four grams per liter, whereas the liquefied gas will weigh 1,335 grams per liter, representing a shrinkage of volume in the ratio of one to 24.7. The gas which stood in the cylinder with 1.4 inch of stroke to be completed would, in condensing shrink to the equivalent of .05668 inch of stroke, considerably more than the volume of the dead space in the cylinder end. Since the liquid is substantially incompressible, and it would arrive suddenly at the point of completely filling the discharge end of the cylinder when the compressor is again started up, serious damage could result.

By making the shaft drive reversible, however, so the shaft can be driven backward by the gas pressure for a fraction of a stroke when the drive is shut off, the pressure of the gas can be automatically relieved. At the assumed stopping point the cylinder under compression contains a quantity of gas which would amount to one cylinder full at two atmospheres, and the intake cylinder would contain four-fifths of a cylinder full at two atmospheres. Since the combined content of the two working chambers is always equal to the maximum content of one cylinder, we have the equivalent of nine-fifths cylinder full at two atmospheres or one cylinder full at about 3.6 atmospheres. This will prevent condensation of the residual gas under all usual circumstances.

It is possible for the drive to be terminated so near the end of a stroke that the crank pin and the Scotch yoke engage one another within the angle of repose; i.e., so near dead center that the Scotch yoke cannot drive the crank pin backward. Should this occur, however, there would not be enough gas left in the high pressure cylinder to fill the dead space of the cylinder when condensed, and no harm could result.

The making of the drive reversible merely requires that no worm and worm wheel or similar driving means be resorted to. This restriction is of no importance in presently conventional compressors, because the ratio of dead space to stroke is relatively so large, but it is important in the compressor shown and described as illustrative of this invention.

The invention has been illustratively shown and described as embodied in a two cylinder compressor having its cylinders directly opposed to one another. The invention is not limited, however, to any specified number of cylinders, nor to the arrangement of cylinders in directly opposed relation to one another, so long as the piston and cylinder axes are disposed at right angles to the slot of the Scotch yoke.

I have described what I believe to be the best embodiment of my invention. I do not wish, however, to be confined to the embodiments shown, but what I desire to cover by Letters Patent is set forth in the appended claims.

I claim:

1. A compressor adapted for compressing refrigerant gas in a cooling or refrigerating system, comprising, in combination,

(a) opposed cylinders having their distal ends closed, and their proximate ends open and facing one another at a distance apart at least as great as, but not substantially greater than, the length of the working stroke, and having intake and discharge passages at their closed ends,

(b) intake and discharge valves associated, respectively, with the intake and discharge passages at the closed end of each cylinder,

(c) a casing rigidly connecting the opposed cylinders in fixed relation to one another,

(d) driving means which includes a drive shaft and a crank and crank pin carried by the shaft,

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- (e) a Scotch yoke located in the casing and driven to and fro by the crank pin,
- (f) pistons mounted in the respective cylinders and rigidly connected to the Scotch yoke at right angles to the slot thereof so that the Scotch yoke and pistons form a unitary, composite, reciprocating, rigid structure, each cylinder having a working length which is at least several times as great as the internal diameter of the cylinder, and each piston having an outside diameter substantially equal to the interior diameter of the cylinder for a length not substantially less than the length of the associated cylinder, and
- (g) each cylinder including a Teflon liner which extends the full length of the cylinder.

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