Aug. 9, 1938. S. B. REDFIELD ET AL 2,126,279

AIR COMPRESSOR

Filed March 17, 1937 2 Sheets-Sheet 1

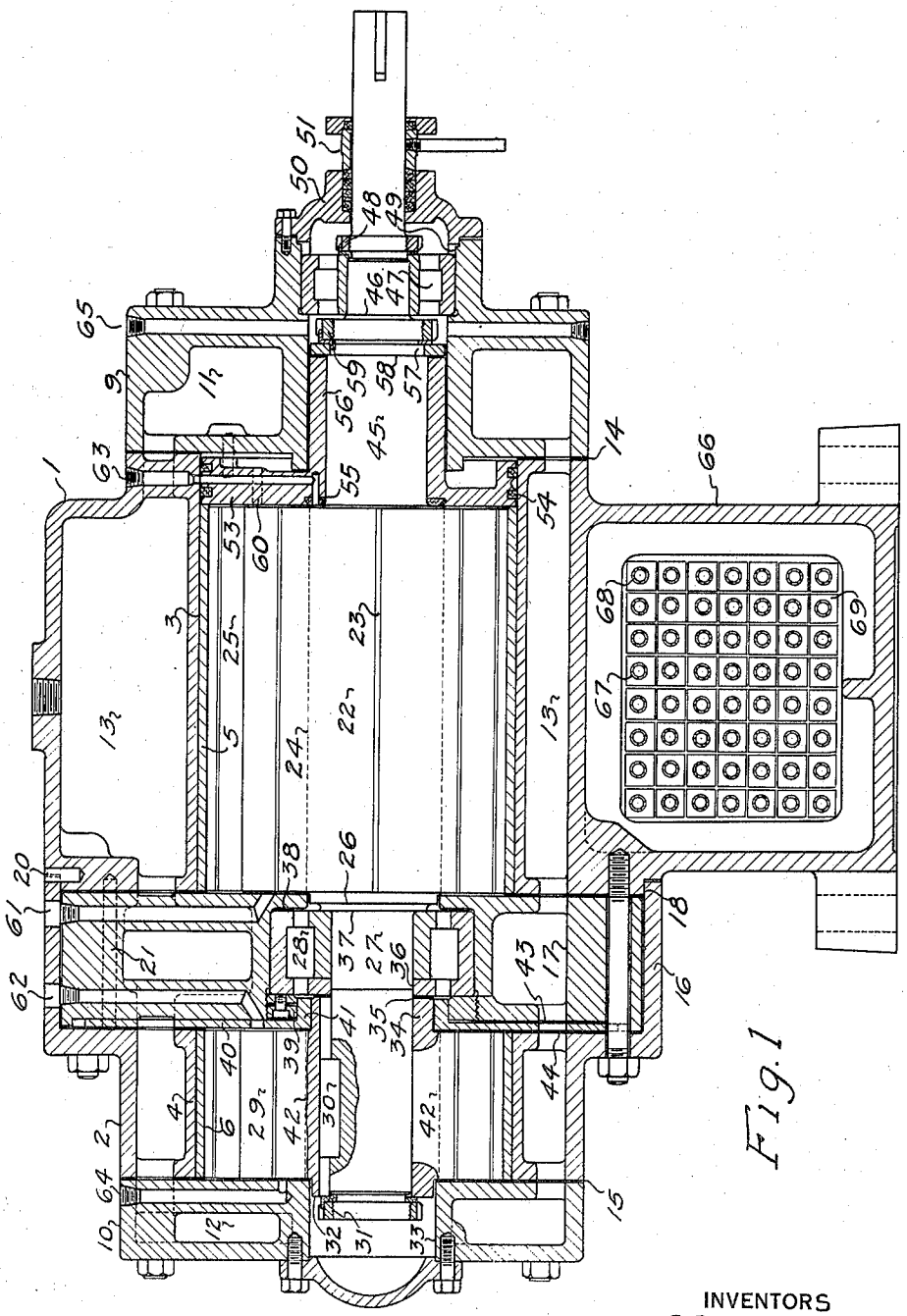


Fig. 1

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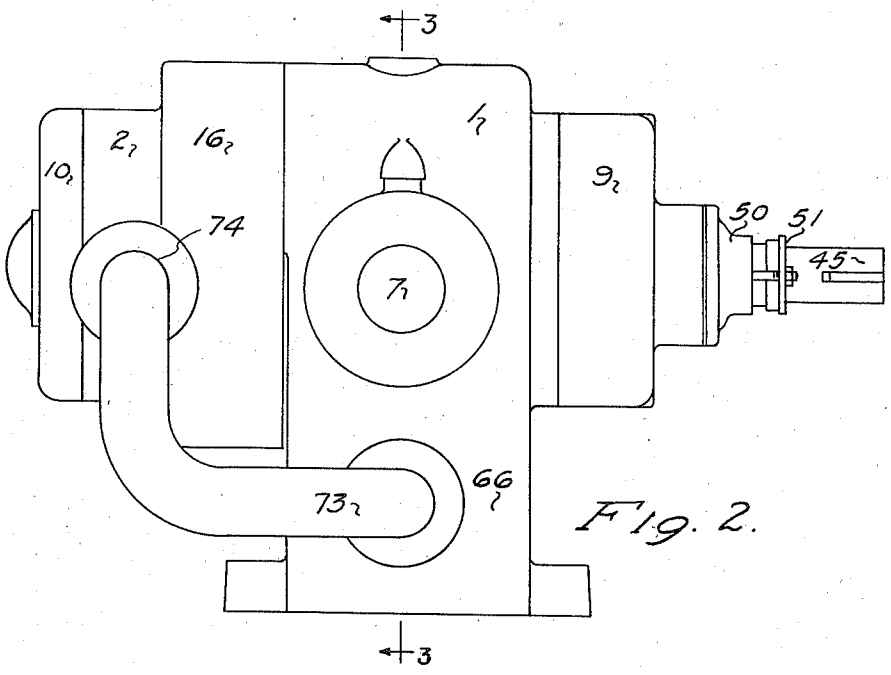


Fig. 2.

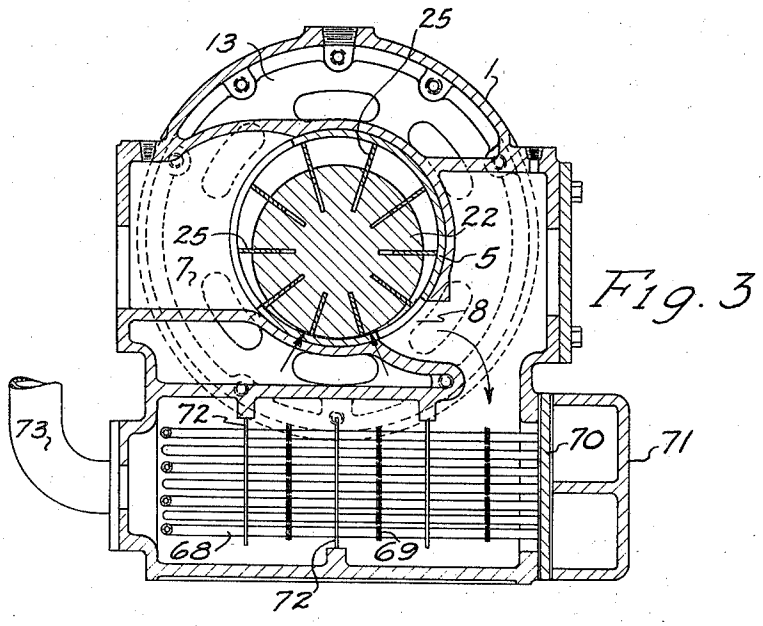


Fig. 3.

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UNITED STATES PATENT OFFICE

2,126,279

AIR COMPRESSOR

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Application March 17, 1937, Serial No. 131,304

4 Claims. (Cl. 230—158)

This invention relates to two-stage rotary compressors and vacuum pumps of the type in which each stage comprises a stator casing having a generally cylindrical bore, within which a cylindrical rotor is arranged eccentrically to provide a crescent shaped space, divided into pockets or cells by a multiplicity of vanes, the cells diminishing in volume from the intake to the discharge port of the casing. The invention has especially to do with apparatus for operation at working pressures in the neighborhood of 100 pounds gauge pressure, or vacuums in excess of 29" of the barometer. To simplify the description, the apparatus will be referred to hereinafter as an air compressor.

It is among the purposes of the invention to provide a compact, unitary apparatus, only slightly larger, in its space requirements, than the low-pressure stage of conventional two-stage compressors, such as disclosed in Pfeiffer Patent No. 1,890,003, but of at least equal efficiency. It is a further purpose to simplify the design and construction whereby the advantages of two-stage compression efficiency can be obtained at relatively low cost.

A principal loss in efficiency in commercial rotary compressors, of the class described, is due to air slippage through the clearance spaces that must be provided between the ends of each rotor and the inner surfaces of the cylinder heads. Such clearance spaces are essential in view of the high operating speed of these compressors, wherein actual metal-to-metal contact would result in scoring, burning or welding of the fixed and moving surfaces. As the stator casing and cylinder heads are water-jacketed for effective cooling, the longitudinal expansion of the rotor, and its trunnion shafts, is considerably greater than that of the cooler casing and, accordingly, adequate clearance must be provided to allow for the expansion of the rotor at the temperature corresponding to the highest discharge pressure at which the machine may be operated.

It is customary to fix a clearance between one end of the rotor and the inner surface of the adjacent cylinder head, by providing a thrust bearing arrangement to restrain longitudinal thrust due to expansion of the rotor, whereby expansion of the latter is forced to take place in the direction of the opposite cylinder head. The clearance at the "fixed end" remains substantially constant, regardless of temperature, increasing only slightly with temperature rise, in accordance with the length of the rotor shaft between the end of the rotor and the thrust bearing. The clearance at the opposite or "expansion end" of the rotor must be sufficient to prevent contact between the rotor and the cylinder head at the temperature corresponding to the highest operating pressure.

In the present apparatus, the rotors of the low and high pressure stages form a unitary assembly on a common shaft and a thrust bearing is located between the rotors so that fixed rotor clearances can be established at adjacent ends of the rotors. With this arrangement, the rotors are forced to expand in opposite directions from the bearing, instead of from an end of the rotor assembly, whereby each expansion end clearance need be established only with relation to the length of the corresponding rotor. Expressed in another way, there are two fixed and two expansion clearances, rather than one fixed and three expansion clearances, the last of which would be based upon expansion of the entire assembly.

In general, the apparatus comprises a water-jacketed casing formed of two sections enclosing, respectively, the low and high-pressure rotors. The section enclosing the low-pressure stage is the larger and is provided with a hollow base, to support the apparatus and to enclose an intercooler communicating directly with the discharge port of the low-pressure stage. The bores of both sections are generally cylindrical and preferably of the same diameter, and are separated by a sealing and bearing assembly which prevents substantial flow of air from the high to the low-pressure stage. The rotors are also preferably of the same diameter, the low-pressure rotor being provided with trunnion shafts one of which is extended and carries the high-pressure rotor. This shaft passes through the sealing and bearing assembly in which it is supported for rotation in a combined thrust and radial bearing, the bearing fixing the position of the rotor assembly so that the rotors expand in opposite directions from the bearing as a center. The other trunnion shaft may conveniently be supported for rotation in a plain roller bearing, the inner race of which is displaced relatively to the outer as the rotor expands or contracts.

The advantages of providing rotors of the same diameter are that the short high-pressure rotor can be secured to a trunnion shaft of the low-pressure rotor and the radial load of the former can be carried by the bearing between the two rotors; expansion of the high-pressure rotor is low and close clearances are permissible, and the large diameter contributes to a high centrifugal force factor, permitting the use of light vanes

and normal speeds. The arrangement described above has a further advantage in reducing the number of machining and assembly operations, whereby the apparatus can be manufactured at relatively low cost. The casing sections can be assembled, with the sealing and bearing assembly omitted, and bored, counter-bored and broached from one setting and sections located, as by means of a dowel, whereby exact alignment is preserved when the complete apparatus is finally assembled.

In the preferred form of the apparatus, however, the sections of the casing are bored as true cylinders and fitted with renewable wear-resisting liners. The two liners can be machined as a single piece, and afterward bored, counterbored and broached from the same setting and then cut to the proper lengths to form the two linings.

The "expansion end" clearance of the low-pressure rotor is preferably held to a substantially constant minimum by means of a "floating" cylinder head such as that disclosed and claimed in a co-pending application of Redfield, Serial No. 131,305, filed March 17, 1937, or in Redfield Patent No. 1,994,786.

For a better understanding of the invention reference is made to the accompanying drawings, in which:

Fig. 1 is a sectional elevation,

Fig. 2 is a side elevation, and

Fig. 3 is a cross-section in line 3—3 of Fig. 2.

Referring to the drawings, the apparatus will be seen to comprise a stator casing formed of two sections, 1 and 2, having cylindrical bores 3 and 4, preferably of the same diameter, enclosing respectively the low and high-pressure stages. The walls of the bores are protected by renewable, wear-resisting liners 5 and 6, the liners being preferably counter-bored and broached, between and as indicated by the arrows in Fig. 3 between the intake port 7 and discharge port 8 of the low-pressure stage, the counter-bore being a segment of a cylinder concentric to the rotor assembly, to be described hereinafter, and cooperating therewith to prevent substantial air slippage below the rotors.

The ends of the casing are closed by cylinder heads 9 and 10, provided with water jackets 11 and 12 which communicate with water jackets 13 of the casing to permit adequate circulation of cooling water. The usual compressible gaskets are provided as at 14 and 15 to prevent leakage, and as will be described more fully hereinafter, the number and thickness of gaskets 15 establish the "expansion end" clearance of the high-pressure stage.

Casing section 2 is provided with an annular flange 16 to form a chamber enclosing a sealing and bearing assembly 17, the end of the flange overlapping a cylindrical shoulder 18 on the casing section 1. Alignment of the sections 1 and 2 is maintained both during machining and assembling by a dowel forced into a bore 20 drilled into both sections. In like manner, a dowel 21, inserted in a bore in casing section 2, sealing and bearing assembly 17 and extending into casing section 1, provides for correct alignment of the sealing and bearing assembly and prevents partial rotation.

The rotor assembly comprises a low-pressure rotor 22, provided with vane slots 23, the depth of the slots being indicated, in Fig. 1, by the dotted lines 24. The vanes 25 may be made of Swedish steel or of a composition such as phenolic condensation product, preferably asbestos cloth impregnated with phenolic resin and compressed and heat treated to avoid swelling in the presence of moisture and oil, and are arranged to slide freely, due to the effect of centrifugal force, into contact with the inner surface of liner 5.

Rotor 22 is provided with a shoulder 26, within the circumference of the bottoms of the vane slots, the shoulder extending into a closely cleared opening for the trunnion shaft 27 in the sealing and bearing assembly 17, to seal the opening against excessive air slippage losses. Trunnion shaft 27 is supported for rotation in a combined radial and thrust bearing 28, and extends beyond it to carry the high-pressure rotor 29. Rotor 29 is bored centrally and is secured to the shaft by the key 30 and a lock-washer and nut 31, on the threaded end of shaft 27, the lock-washer and nut bearing against a shoulder 32 of rotor 29, the shoulder extending into and closely clearing the wall of a bore 33 in cylinder head 10, to serve as a seal against excessive slippage. The opposite side of rotor 29 is provided with a shoulder 34 which bears against one or more shims 35 between it and the collar 36 of bearing 28, the inner race of the bearing being held thereby against shoulder 37 of rotor 22.

The outer race of bearing 28 abuts one or more shims 38 which establish its longitudinal position in the bearing chamber in sealing and bearing assembly 17, the race being secured by lock-nut 39. Substantial air slippage across the end of the rotor 29 and through bearing 28, from the high to the low-pressure stage, is prevented by a filler plate 40, arranged in a circular recess in sealing and bearing assembly 17. The filler plate is provided with a hub collar 41, closely clearing the shoulder 34, the inner wall of the collar being within the circumference of the bottom of the vane slots, indicated by the dotted lines 42. The usual compressible gaskets 43 and 44 are provided between the surfaces of the filler plate, the sealing and bearing assembly and the adjacent surface of casing section 2, to prevent leakage of cooling water.

As the inner race of bearing 28 is rigidly secured to shaft 27, the "fixed end" clearance between the smooth inner surface of the sealing and bearing assembly 17, which serves as a cylinder head of the low-pressure stage, and the end surface of rotor 22, is conveniently established by the number and thickness of shims, 38, which fix the longitudinal position of the outer race. This clearance need only be a running clearance plus the "play" of the bearing, as the effect of expansion is to increase the clearance, but to a negligible degree because of the short distance from the end of the rotor to the inner surface of bearing collar 36. Similarly, the "fixed end" clearance between the end of high-pressure rotor 29 and the adjacent surface of filler plate 40 is established by the number and thickness of shims 35 which locate the position of the rotor on shaft 27.

The "expansion end" clearance of the high-pressure rotor is established by the number and thickness of the gaskets 15, which have the additional function of preventing leakage of cooling water and air. The actual clearance may be checked, in the usual manner, by measuring the thickness of a lead wire pressed between the end of the rotor and the inner surface of cylinder head 10, and it will be understood that by employing a rotor of large diameter and short length, instead of the conventional narrow, long rotor, the expansion is negligible and the efficiency is high, as little clearance is necessary. It will also be understood

that the entire radial load of the rotor can be supported by bearing **28**, eliminating the necessity for the usual expansion end bearing.

The expansion end of low-pressure rotor **22** is provided with a trunnion shaft **45** having a shoulder **46** to serve as an abutment for the inner race of a plain radial roller bearing **47**, the race being secured by a lock-washer and nut **48**, threaded on the shaft. The outer race is secured within a chamber in cylinder head **9** by internal fingers **49** on bearing cover **50**. Shaft **45** passes through the usual packing gland **51** and is provided with a key-way to secure a driving coupling (not shown).

To provide for a minimum and substantially constant expansion end clearance, we prefer to employ a "floating", internal cylinder head of the type described and claimed in the co-pending application of Redfield, referred to previously. This device comprises a disc **53** located within the bore **3** of casing section **1**, beyond the end of liner **5**, the outer edge of the disc being sealed, as by means of conventional internal combustion engine piston rings **54**. The inner surface of the disc is under-cut and provided with an annular bearing **55**, preferably of babbitt, which bears against a shoulder at the end of the rotor within the circumference of the bottoms of the vane slots, to provide the necessary minimum clearance between the end of the rotor and the inner surface of disc **53**, and the location of the bearing serving to prevent air slippage around the shaft. The disc is provided with a sleeve **56**, concentric to the shaft **45**, the sleeve clearing the shaft by a running fit. The outer surface of the sleeve is preferably smoothed by grinding, to permit free longitudinal movement in the bore of cylinder head **9**, but is fitted with sufficient freedom to prevent binding due to expansion. Relative longitudinal movement of the floating head and shaft **45** is restrained by a rotating collar **57** which is secured against a shoulder **58** on the shaft and bears against the outer end of sleeve **56**, the collar being secured to the shaft by lock-washer and nut **59**. Disc **53** is also preferably provided with a hole **60**, drilled at an intermediate pressure point, partially to balance the effect of unequal pressure on its inner surface, and to maintain bearing **55** in contact with the end of the rotor. It will be seen from the foregoing that disc **53** moves with the rotor as the latter expands and contracts, to maintain a substantially minimum clearance.

The compressor is lubricated in the usual manner, similarly to the apparatus of the Pfeiffer patent, the oil being injected by a force-feed pump of any suitable type. Ducts **61** and **62** deliver the oil to the adjacent ends of the rotors at intermediate points between the bottoms of the vane slots and the surface of the rotors, a part of the oil entering the slots but the major quantity being distributed by centrifugal force and the action of the vanes to form films on the surfaces of the liners. Ducts **63** and **64** deliver the oil at or near the bottoms of the slots, so that the pumping action of the vanes, as they move outwardly in the intake zone, will spread the oil on the surfaces of the slots and vanes. Duct **63** opens through bearing **55** to lubricate it and to add to the air sealing effect. A duct **65** is provided to introduce oil to the radial bearing **47**.

Casing section **1** is provided with a hollow base **66**, formed as an integral part of the casting, to support the apparatus and to enclose an intercooler **67**. The intercooler comprises U-tubes **68** provided with fins indicated at **69** to improve heat transfer, the ends of each tube terminating in a tube sheet **70** enclosed by a water manifold **71**. The air discharges directly into the intercooler from the discharge port **8** of the low-pressure stage and is forced by baffles **72** to pass several times through the bank of tubes **68**, the cooled air flowing through the pipe **73** to the intake **74** of the high-pressure stage.

It will be understood that the floating head **53** is preferred to provide for maximum efficiency over a wide range of operating pressures and temperatures, but that the apparatus can be simplified, if the discharge pressure is to be substantially constant, by substituting a conventional stationary cylinder head, of the type shown in Pfeiffer Patent, and establishing the expansion end clearance by the use of gaskets **14** of the necessary number and thickness. It will also be understood that it is within the scope of the invention to employ a relatively long high-pressure rotor, with an extended shaft supported for rotation in a plane radial roller bearing, similar to the bearing **47** and to take advantage of the sealing and bearing assembly, whereby expansion will take place in opposite directions from a thrust bearing between the two rotors.

We claim:

1. A two-stage rotary air compressor of the class described comprising the combination of a stator casing formed of two sections having generally cylindrical bores enclosing respectively a low and a high pressure stage, cylinder heads closing the casing at opposite ends of the bores, a sealing and bearing assembly between the sections and closing adjacent ends of the bores, the opposite surfaces of the assembly serving as cylinder heads for said stages, a cylindrical low pressure rotor arranged eccentrically in one of said bores, the rotor having trunnion shafts, one of the trunnion shafts being supported for rotation in a radial bearing arranged in the cylinder head closing one end of the bore, the other shaft extending through said assembly and being supported by a thrust bearing arranged in and having an outer race secured to the assembly and an inner race secured to said shaft, means to adjust the longitudinal position of the outer race to fix a substantially constant clearance between the low pressure rotor and the adjacent surface of the assembly, a high pressure rotor arranged in the other bore and secured to said shaft, and means to adjust the longitudinal position of said high pressure rotor on said shaft to fix a substantially constant clearance between an end of said high pressure rotor and the other surface of the assembly.

2. A two-stage rotary air compressor of the class described comprising the combination of a stator casing formed of two sections having generally cylindrical bores enclosing respectively a low and a high pressure stage, cylinder heads closing the casing at opposite ends of the bores, a sealing and bearing assembly between the sections and closing adjacent ends of the bores, the opposite surfaces of the assembly serving as cylinder heads for said stages, a cylindrical low pressure rotor arranged eccentrically in one of said bores, the rotor having trunnion shafts, one of the trunnion shafts being supported for rotation in a radial bearing arranged in the cylinder head closing one end of the bore, the other shaft extending through said assembly and being supported by a thrust and radial bearing arranged in and having an outer race secured to the as-

sembly and an inner race secured to the shaft, means to adjust the longitudinal position of the outer race to fix a substantially constant clearance between the low pressure rotor and the adjacent surface of the assembly, a high pressure rotor arranged in the other bore and secured to said shaft, the radial load of said high pressure rotor being carried by said thrust and radial bearing, and means to fix a substantially constant clearance between an end of said high pressure rotor and the other surface of the assembly.

3. A two-stage rotary air compressor of the class described comprising the combination of a stator casing formed of two sections having generally cylindrical bores enclosing respectively a low and a high pressure stage, cylinder heads closing the casing at opposite ends of the bores, a sealing and bearing assembly between the sections and closing adjacent ends of the bores, the opposite surfaces of the assembly serving as cylinder heads for said stages, a cylindrical low pressure rotor arranged eccentrically in one of said bores, the rotor having trunnion shafts, one of the trunnion shafts being supported for rotation in a radial bearing arranged in the cylinder head closing one end of the bore, the other shaft extending through said assembly and being supported by a thrust and radial bearing arranged in a chamber in the assembly and having an outer race separated from a wall of the chamber by shims, a lock-nut to secure the race against the shims, the bearing having an inner race abutting a shoulder on the shaft, a high pressure rotor arranged in the other bore, said high pressure rotor being bored centrally and secured to the shaft on the opposite side of the assembly and having an annular shoulder surrounding the shaft to secure the inner race, shims between the last-named shoulder and the last-named race, and a filler plate having an opening and an annular shoulder surrounding and concentric to the shoulder on said high pressure rotor to enclose said bearing to prevent substantial air flow from the high to the low pressure stage.

4. A two-stage rotary air compressor of the class described comprising the combination of a stator casing formed of two sections having generally cylindrical bores enclosing respectively a low and a high pressure stage, cylinder heads closing the casing at opposite ends of the bores, a sealing and bearing assembly between the sections and closing adjacent ends of the bores, the opposite surfaces of the assembly serving as cylinder heads for said stages, a cylindrical low pressure rotor arranged eccentrically in one of said bores, the rotor having trunnion shafts, one of the trunnion shafts being supported for rotation in a radial bearing arranged in the cylinder head closing one end of the bore, the other shaft extending through said assembly and being supported by a thrust bearing arranged in and having an outer race secured to the assembly and an inner race secured to said shaft, the position of the outer race fixing a substantially constant clearance between the low pressure rotor and the adjacent surface of the assembly, a high pressure rotor arranged in the other bore and secured to said shaft, and means to fix the longitudinal position of said high pressure rotor on said shaft to establish a substantially constant clearance between an end of said high pressure rotor and the other surface of the assembly.

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