Aug. 9, 1938. W. P. EPPERS 2,126,247

AIR COMPRESSOR

Filed March 17, 1937 2 Sheets-Sheet 1

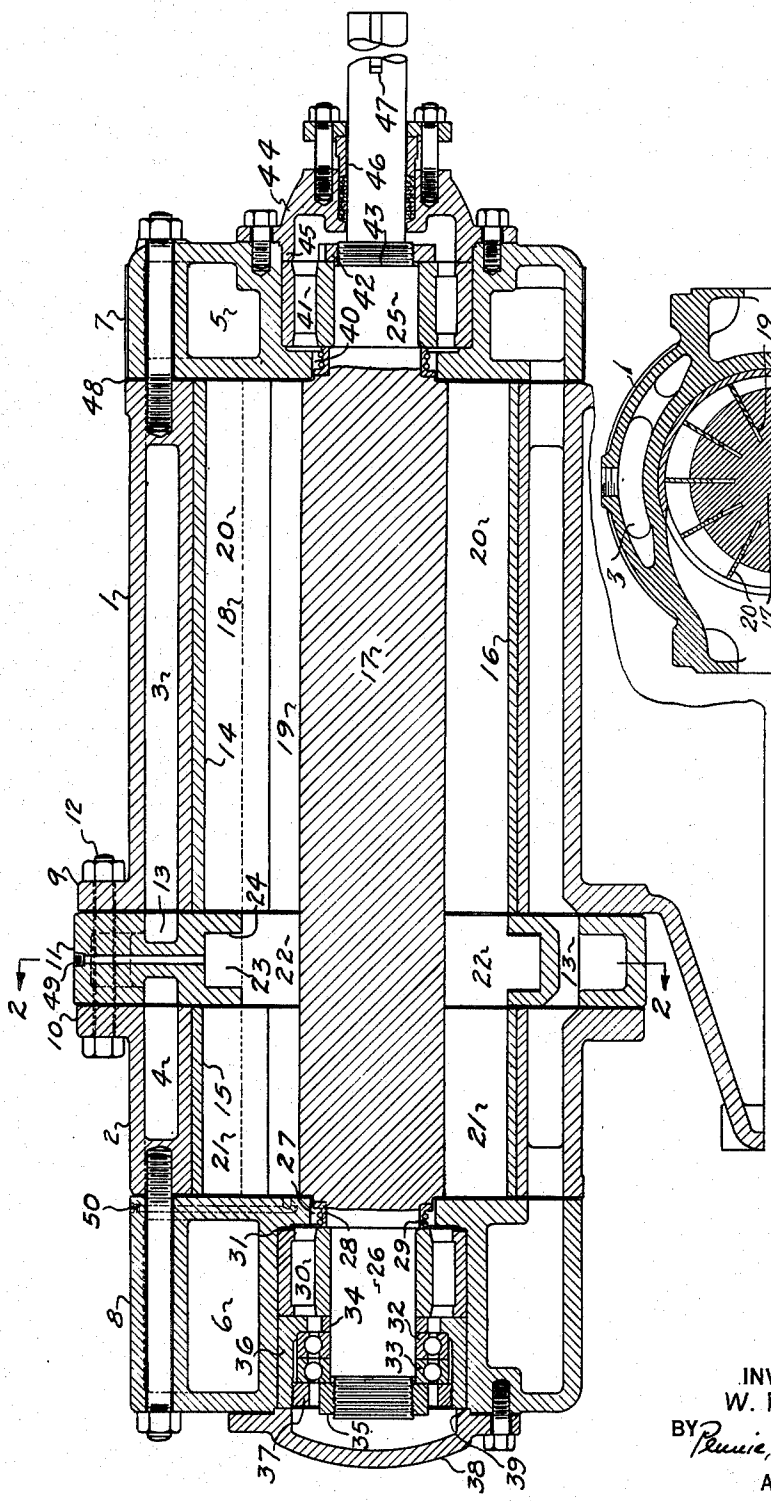

Fig. 1

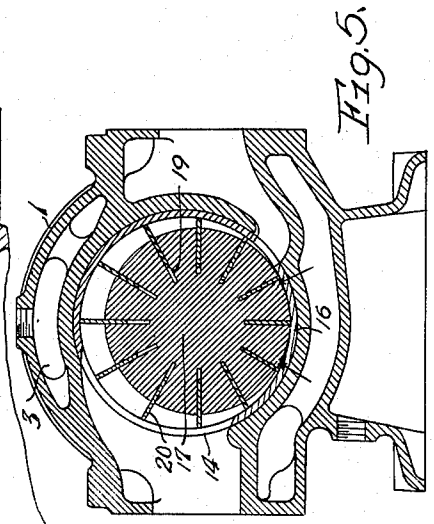

Fig. 5.

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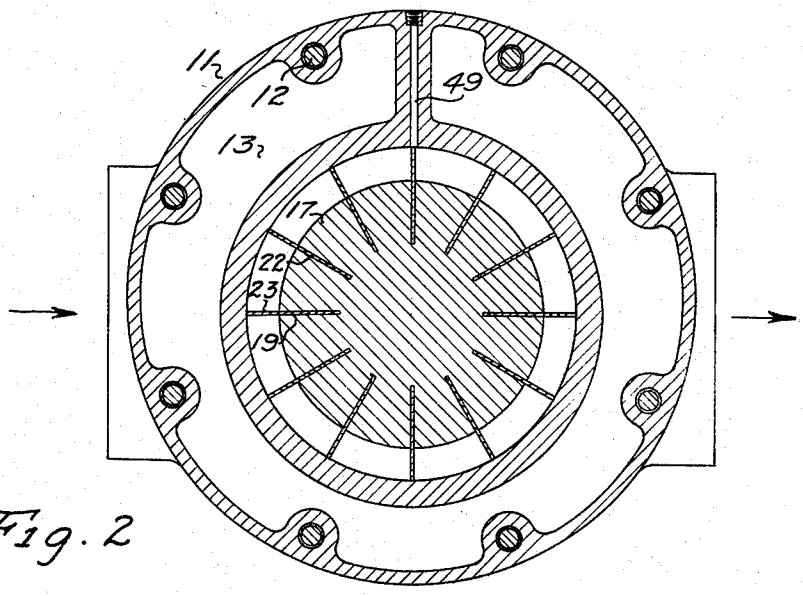

Fig. 2

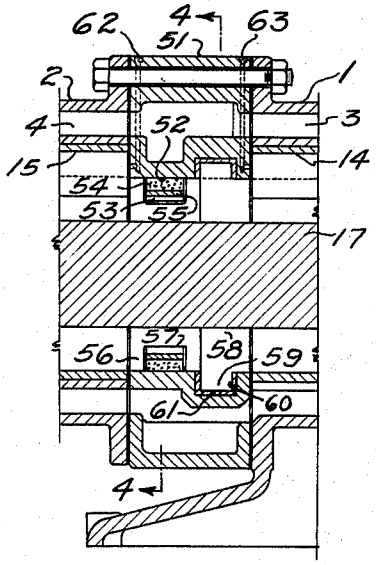

Fig. 3

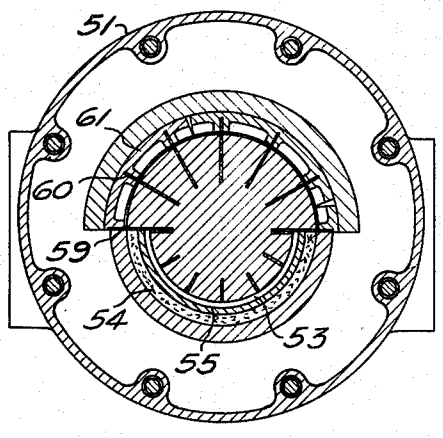

Fig. 4

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UNITED STATES PATENT OFFICE

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AIR COMPRESSOR

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Application March 17, 1937, Serial No. 131,297

10 Claims. (Cl. 230—158)

This invention relates to multiple stage rotary compressors and vacuum pumps for air and other gases, and will be referred to hereinafter as an air compressor to simplify the description. It has especially to do with the type of compressor in which each stage comprises a cylindrical rotor arranged eccentrically within a substantially cylindrical bore, the rotor being provided with a plurality of vanes which pocket the air in cells, to compress the air by decrease in the volume of the cells as they move from the inlet to the discharge outlet. More specifically, the invention relates to improvements in the compressor described and claimed in Pfeiffer Patents 1,895,816 and 1,890,003.

The invention is especially directed to a compressor, of the type described, in which two or more stages are enclosed within the same casing or cylinder; a single rotor, supported for rotation in anti-friction bearings arranged in opposite cylinder heads, serves both stages, and the stages are separated by a division wall provided with sealing devices to prevent substantial loss of air from the high to the low pressure stage. It is among the purposes of the invention to obtain the advantages of two-stage compressor efficiency with a compact, unitary apparatus of the same diameter and only slightly longer than the low pressure cylinder disclosed in the Pfeiffer patents. It is a further purpose to reduce the number of parts and the machining operations necessary to produce them, whereby the cost of manufacture is reduced substantially below that of multiple stage compressors of previous types for equivalent service.

In general, the apparatus comprises a water-jacketed stator casing, provided with a central bore, the casing being divided into two or more sections, each having an air inlet on one side and an outlet for compressed air on the opposite side. The bores of each section are generally cylindrical and preferably of the same diameter, for reasons which will be explained more fully hereinafter, although a portion of the cylinder wall may be undercut to form the usual pressure seal between the inlet and outlet of each stage, as described in Pfeiffer Patent No. 1,890,003. A solid cylindrical rotor, preferably of constant diameter is provided with integral sub-shafts mounted in anti-friction bearings arranged in cylinder heads closing the ends of the casing, the rotor being arranged parallel with but eccentrically to the center line of the bore of the casing. If a pressure seal is provided, as described above, the undercut portion of the bore of the casing is concentric to the rotor and a minimum safe running clearance corrected for working temperatures, is provided.

The sections of the stator casing are separated by a division wall to provide compartments forming the two stages. The division wall is provided with a central bore concentric to the rotor, the bore having a diameter only sufficient to provide a running clearance at the maximum operating temperature. The rotor is provided with a plurality of substantially radial slots running the full length of the rotor, in which thin vanes in each stage are arranged to slide freely and to move outwardly by centrifugal force into contact with the inner wall of the casing. Each slot carries at least three vanes, one in each compartment, and a third cooperating with the division wall to seal the high from the low pressure stage against loss of air through the slot. The third vane is of substantially the same length as the width of the division wall and its width is sufficient to close the distance between the base of the slot and the inner surface of the division wall less the necessary running clearance. The outer edge is provided with a tongue which travels in a corresponding groove in the division wall, the tongue bearing against the edge of the groove adjacent to the low pressure stage, due to the pressure differential between the high and low pressure stages. The third vane, accordingly, serves to prevent air leakage along the slots, and to guide the vanes in the low and high pressure stages, especially in their inward positions.

Although some unavoidable air slippage occurs from the high to the low pressure stages along the rotor, through the space between it and the inner surface of the division wall, the losses are compensated for by the elimination of two of the four end clearance spaces, between the ends of the rotors and inner surfaces of the cylinder heads of conventional type two stage compressors, and in spite of the slightly increased expansion end clearance of the present apparatus.

As the volumetric displacement of the high pressure stage is necessarily considerably less than that of the low pressure stage, I prefer to make both sections of the casing of the same internal diameter and to provide a rotor of constant diameter, for three principal reasons. First, and in contrast to the high pressure stage of the apparatus of the Pfeiffer patents, for example, the greater radius increases the centrifugal force factor and permits the use of shorter vanes while maintaining the necessary centrifugal force to cause the vanes to move outwardly,

without increasing the weight of the vanes or the speed. Second, a shorter rotor can be used, thereby decreasing slippage losses as less clearance need be provided between an end of the rotor and the adjacent cylinder head to provide for a safe running clearance at the maximum operating temperature. It will be realized that as the stator casing is water-jacketed, whereas the rotor is not cooled, the latter expands relatively to the casing, and a safe clearance must be provided. By fixing the rotor in a thrust bearing in the cylinder head adjacent to the high pressure stage, expansion takes place toward the opposite cylinder head, where the clearance described involves the least loss, the clearance at the fixed end of the rotor can therefore be held to a minimum. As the high pressure stage is short, relatively little vane end clearance need be provided in the high pressure stage, making the latter unusually efficient. Third, the cost of manufacture is greatly reduced because the operation of cutting the slots is the most expensive in the manufacture of rotary compressors of this type, and the present design permits a single operation in cutting each slot throughout the length of the rotor. Likewise, the sections of the stator casing can be assembled without the division wall and both sections bored and broached as single operations.

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

Fig. 1 is a longitudinal section of the preferred form of the apparatus,

Fig. 2 is a cross-section on line 2—2 of Fig. 1,

Fig. 3 is a partial longitudinal section, showing a modified form of division wall;

Fig. 4 is a cross-section on line 4—4 of Fig. 3; and

Fig. 5 is a cross-sectional view through the middle of the low pressure stage of the apparatus.

Referring now to the drawings and first to Fig. 1, the stator casing is shown to consist of two sections 1 and 2, enclosing respectively the low and high-pressure stages. The sections are provided with water jackets 3 and 4 which communicate with water jackets 5 and 6 in cylinder heads 7 and 8, closing opposite ends of the stator casing. The adjacent ends of the sections are provided with flanges 9 and 10, having machined surfaces, and are separated by a division wall 11, the parts being secured by through-bolts 12. The division wall is chambered, as indicated at 13, to communicate with the water jackets 3 and 4 to permit the free flow of cooling water. The bores of sections 1 and 2 are cylindrical and are preferably lined with wear resisting liners 14 and 15. The inner surfaces of the liners are cylindrical excepting for a segment, at the bottom and between the intake and discharge ports shown generally by the flow arrows in Fig. 2, the segment being underbored as indicated by the thinner section 16, in the usual manner, to provide a surface concentric to the rotor 17, to form a pressure seal between the latter and the liner. A minimum safe running clearance at the maximum operating temperature is provided between these surfaces. The rotor 17 is arranged eccentrically within the bores in the usual manner, its width being indicated by the dotted line 18 (Fig. 1). It is preferably of uniform diameter and passes centrally through a bore in the division wall 11, the inner surface of the bore clearing the rotor a minimum safe distance to prevent substantial flow of air from the high to the low pressure stage along the surface of the rotor.

The rotor is provided with a plurality of equidistant, preferably radial slots 19, the slots being cut throughout the length of the rotor. Each slot accommodates three vanes, the vanes 20 in the low pressure stage and the vanes 21 in the high pressure stage being free to move outwardly due to centrifugal force, in the usual manner. Between each of these pairs of vanes a third vane 22 is provided. The length of this vane is preferably substantially the same as the width of the division wall 11 plus a slight addition for running clearance and the width is sufficient to fill the slots, i. e., to the surface of the rotor. Vanes 22 are provided with tongues 23 arranged in a corresponding groove 24, the side-walls of which cooperate with the tongues to guide the vanes and prevent their substantial lateral displacement. As the effect of pressure will cause the tongues to bear or run against the right wall of the groove, the running clearance of the tongue is the slight addition to the length of vanes 22, referred to above. The vanes 22 are restrained from substantial outward movement, due to the effect of centrifugal force, by the inner surface of the bore of division wall 11 and have no function in elevating pressure. They serve, however, to seal the slots 19 to prevent substantial air flow from the high to the low pressure stage. Their side edges serve to form continuations of the sides of division wall 11 to guide the vanes 20 and 21 radially in their outward movement, thereby preventing tilting and breakage of vanes or scoring the inner surfaces of the cylinder heads 7 and 8. As the combined length of the three vanes should be at least equal to the length of rotor 17, to prevent leakage or end slippage, the vane 22 permits the rotor to expand freely without displacing vane 20 toward the right of the figure, as would be the case if the slots were closed in the zone of division wall 11, by fixed means.

The rotor 17 is carried by trunnion shafts 25 and 26 in bearings in the cylinder heads 7 and 8. To minimize slippage losses, the "fixed end" of the rotor is carried by the head 8 adjacent to the relatively short high pressure stage. Cylinder head 8 is provided with an eccentric bore 27 which opens into a chamber to accommodate the bearings, the bore preferably being partially sealed by a split collar 28 secured to the shaft by garter springs 29. Adjacent to the collar, shaft 26 is shouldered to form an abutment for the inner race of a radial roller bearing 30, the outer race of which bears against one or more shims 31 which space it from the inner wall of cylinder head 8. The purpose of shims 31 is to fix a desired running clearance between the end of the rotor 17 and the inner surface of cylinder head 8, the number and thickness of shims 31 determining the degree of clearance. Roller bearing 30 carries the entire radial load of that end of the rotor. The thrust load, particularly that due to expansion and contraction of the rotor is carried by a thrust bearing, preferably comprising a pair of ball thrust bearings 32 and 33, to carry the thrust load in both directions. The inner race of bearing 32 is separated from the inner race of bearing 30 by a rotating spacer-ring 34, the inner races being secured in position by a lock-nut 35 secured to the threaded end of trunnion shaft 26. The outer races of bearings 32 and 33 are located within a stationary spacer-ring 36 which bears against the outer race of radial bearing 30, the outer races of bearings 32 and 33 being secured

by a lock-nut 37. The bearing assembly is held in position by a bearing cover 38 provided with an annular boss 39 which bears against spacer-ring 36. It will be understood from the foregoing that the running clearance between the end of rotor 17 and the inner surface of the cylinder head 8 can be fixed by means of shims 31 to a minimum in accordance with the lateral play or looseness of thrust bearings 32 and 33 which fix the position of the rotor and force it to expand in the direction of cylinder head 7.

The "expansion end" of the rotor is carried by the cylinder head 7, having a similar eccentric bore 40 which communicates with a chamber to accommodate the radial bearing 41. The inner race of the bearing is secured to the trunnion shaft 25 by locknut 42 secured to a threaded shoulder 43 on the shaft. The outer race of the bearing is secured by the bearing cover 44 having an annular boss 45 which bears against the outer race of the bearing. Bearing cover 44 is provided with the usual packing gland 46 through which the shaft passes, the shaft being provided with a key-way 47 for connection to the usual flexible coupling.

The expansion end clearance, between the inner face of the cylinder head 7 and the end of rotor 17, is fixed by one or more oiled paper gaskets 48, the number and thickness of which establish the clearance in accordance with the expected expansion of the rotor at the temperature corresponding to the highest operating pressure. The rotor is free to expand or contract as the inner race of bearing 41 is free to move axially.

As section 2 of the stator casing surrounding the high pressure stage is short, the relative expansion of rotor 17 to the casing in this zone is low and accordingly the vanes 21 can be of sufficient length to provide a minimum running clearance at the maximum expected temperature, thereby reducing slippage losses at their ends. The rotor does not displace the vanes 22 as it contracts or expands because these vanes are guided by the tongues 23 in cooperation with the walls of groove 24. By fixing the position of vanes 22 their side edges serve as guides for the radial movement of vanes 21 and 20. This assists in permitting the use of vanes 20 of maximum length, thereby avoiding excessive end clearance. If the slots in the vicinity of the division wall were merely closed, as by welding or otherwise securing pieces in them, it will be realized that these pieces would be longer than the vanes 22, and would have to extend a short distance into the high pressure stage to prevent the corners of vanes 21 from being caught under the inner surface of the division wall 11 when the apparatus is operated at less than maximum pressure, and when operating at high pressure they would be displaced into the low pressure stage by the expansion of the rotor, thereby displacing vanes 20 toward the right. It will accordingly be understood that vanes 22 permit the use of vanes of maximum length in both the low and high pressure stages.

The oil used for lubricating the apparatus is so admitted that it performs the additional function of sealing the surfaces above described against air slippage losses. To this end, the division wall 11 is provided with a duct 49 opening into the bore about centrally of the tongues 23. The greater quantity of the lubricant is admitted at this point, the vanes 22 and their tongues 23 distributing the oil circumferentially and the pressure differential between the high and low pressure stages causing a longitudinal flow toward the low pressure stage wherein the radial movement of the vanes 20 completes the distribution. The relatively thick film thus formed between the inner surface of the bore of the division wall 11 and the surface of rotor 17 accordingly forms an effective seal against air losses in this zone. It will be understood that a part of the oil enters the slots 19 in the vicinity of vanes 22 to assist in the sealing of the slots. The remainder of the oil is admitted through a duct 50 provided in cylinder head 8, the duct opening through the inner surface of the head near the bottom of the slots 19. It has been found that radial movement of the vanes 20 and 21 provides for sufficient distribution to lubricate the bearings adequately.

A modified form of sealing arrangement is shown in Figs. 3 and 4. A division wall 51 is provided with a bore 52 concentric to rotor 17 and the latter is provided with a groove 53 to accommodate compressible packing 54, the packing being lightly held against the inner surface of the bore 52 and the side-walls of groove 53 by an annular spring 55, the packing serving to seal the clearance between the surface of the rotor and the surface of bore 52, as well as the side-walls of the groove. Sealing vanes 56 are provided with openings 57 surrounding the packing and spring. Vanes 56 are restrained from displacement toward the low pressure stage, due to the elevated pressure of the high pressure stage, by abutting vanes 58, the latter being guided by tongues 59 inserted in slots 60 in a rotating guide ring 61. The guide ring is preferably formed of a plurality of segments whereby it may be freely inserted in a corresponding groove in the walls of bore 52.

The lubricating oil is employed in a manner similar to that described with reference to the preferred form to assist in sealing the parts against substantial air losses. The division wall 51 is provided with a duct 62 opening into the high pressure stage, whereby a part of the oil is distributed by the vanes 21 and forms a film moving toward the low pressure stage between the surface of rotor 17 and the bore of the division wall. An additional duct is provided at 63 to complete the sealing effect and the lubrication of the low pressure stage.

The operation of the apparatus will be apparent from the foregoing, and it will be understood that trunnion shaft 25 may be directly coupled, through the usual flexible coupling, to an electric motor or internal combustion engine. The air discharging from the low pressure stage is conducted through any suitable type of intercooler, forming no part of the present invention, and is delivered to the intake of the high pressure stage in the usual manner. The apparatus is intended for normal operation at working pressures up to 125 pounds gauge pressure.

In the construction of the apparatus, the design decreases the number of machining operations. After the inner surfaces of flanges 9 and 10 have been machined, the sections 1 and 2 may be bolted together, without the division wall, and bored together to the same diameter. Similarly, the liners may be machined as a single piece, i. e. they may be turned, bored, counter-bored and broached together and afterward cut to form the two liners 14 and 15. By this procedure exact similarity and proper alignment can be provided at a minimum manufacturing cost.

I claim:

1. A two-stage rotary air compressor of the multi-vane type comprising the combination of a water-jacketed 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 single rotor arranged eccentrically within the bores and provided with vane-slots extending throughout the length of the rotor a division wall between the sections having a bore closely surrounding and concentric to the rotor, means in the slots and surrounded by the wall to prevent substantial flow of air from the high to the low pressure stage, the rotor being provided with trunnion shafts, and bearings in each cylinder head to support the shafts.

2. A two-stage rotary air compressor of the multi-vane type comprising the combination of a water-jacketed 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 single rotor arranged eccentrically within the bores and provided with trunnion shafts, thrust and radial bearing means for one of the shafts in the cylinder head closing the bore of the high pressure stage, a radial bearing for the other shaft in the other cylinder head, a division wall between the sections having a bore closely surrounding and concentric to the rotor, the rotor being provided with vane-slots extending throughout the length of the rotor, each slot carrying a low and a high pressure vane and a sealing vane, the sealing vane cooperating with the division wall to prevent substantial flow of air from the high to the low pressure stage.

3. A two-stage rotary air compressor of the multi-vane type comprising the combination of a water-jacketed 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 single rotor arranged eccentrically within the bores and provided with trunnion shafts, thrust and radial bearing means for one of the shafts in the cylinder head closing the bore of the high pressure stage, a radial bearing for the other shaft in the other cylinder head, a division wall between the sections having a bore closely surrounding and concentric to the rotor, the rotor being provided with vane-slots extending throughout the length of the rotor, each slot carrying a low and a high pressure vane and a sealing vane, the sealing vane being of substantially the same length as the width of the division wall and being surrounded thereby.

4. A two-stage rotary air compressor of the multi-vane type comprising the combination of a water-jacketed 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 single rotor arranged eccentrically within the bores and provided with trunnion shafts, thrust and radial bearing means for one of the shafts in the cylinder head closing the bore of the high pressure stage, a radial bearing for the other shaft in the other cylinder head, a division wall between the sections having a bore closely surrounding and concentric to the rotor, the rotor being provided with vane-slots extending throughout the length of the rotor, each slot carrying a low and a high pressure vane and a sealing vane, the sealing vane being of substantially the same length as the width of the division wall and being surrounded thereby, and means to prevent substantial longitudinal displacement of said vane when the rotor expands or contracts longitudinally.

5. A two-stage rotary air compressor of the multi-vane type comprising the combination of a water-jacketed 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 single rotor arranged eccentrically within the bores and provided with trunnion shafts, thrust and radial bearing means for one of the shafts in the cylinder head closing the bore of the high pressure stage, a radial bearing for the other shaft in the other cylinder head, a division wall between the sections having a bore closely surrounding and concentric to the rotor, the rotor being provided with vane-slots extending throughout the length of the rotor, each slot carrying a low and a high pressure vane and a sealing vane, the sealing vane being of substantially the same length as the width of the division wall and being surrounded thereby, said vane having a tongue extending into a circumferential groove provided in the bore of the wall to prevent substantial longitudinal displacement of the vane.

6. A two-stage rotary air compressor of the multi-vane type comprising the combination of a water-jacketed 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 single rotor arranged eccentrically within the bores and provided with trunnion shafts, thrust and radial bearing means for one of the shafts in the cylinder head closing the bore of the high pressure stage, a radial bearing for the other shaft in the other cylinder head, a division wall between the sections having a bore closely surrounding and concentric to the rotor, the rotor being provided with vane-slots extending throughout the length of the rotor, each slot carrying a low and a high pressure vane and a sealing vane, the sealing vane being at least as long as the width of the division wall, the outer edge of the vane having a tongue arranged in a groove provided in the wall to prevent substantial longitudinal displacement of the vane.

7. A two-stage rotary air compressor of the multi-vane type comprising the combination of a water-jacketed 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 single rotor arranged eccentrically within the bores and provided with trunnion shafts, means to fix a substantially constant clearance between the cylinder head closing the high pressure stage and the adjacent end of the rotor including a thrust bearing having an outer race arranged in and secured to said cylinder head and an inner race secured to the corresponding shaft, a radial bearing in the other cylinder head and supporting the other shaft, a division wall between the sections having a bore closely surrounding the rotor, the rotor being provided with vane-slots extending throughout its length, a low pressure and a high pressure vane in each slot, a sealing vane in each slot between said vanes and surrounded by the wall, the sealing vane being of a width substantially equal to the depth of the slot and having a central tongue, the wall being provided with an internal groove surrounding the rotor to accommodate the tongue

to prevent substantial longitudinal displacement of said vane.

8. A two-stage rotary air compressor of the multi-vane type comprising the combination of a water-jacketed casing formed of two sections having generally cylindrical bores of substantially the same diameter enclosing respectively a low and a high pressure stage, cylinder heads closing the casing at opposite ends of the bores, a single cylindrical rotor arranged eccentrically within the bores and provided with trunnion shafts, thrust and radial bearing means for one of the shafts in the cylinder head closing the high pressure stage, a radial bearing supporting the other shaft in the other cylinder head, a division wall between the sections having a bore closely surrounding and concentric to the rotor, the rotor being provided with vane-slots extending throughout its length, a low pressure and a high pressure vane in each slot, a sealing vane in each slot between said vanes and surrounded by the division wall, and means to prevent substantial longitudinal displacement of the sealing vane including a central tongue formed on its outer edge, the wall being provided with a groove to accommodate the tongue.

9. A two-stage rotary air compressor of the multi-vane type comprising the combination of a water-jacketed 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 single rotor arranged eccentrically within the bores and provided with trunnion shafts, thrust and radial bearing means for one of the shafts in the cylinder head closing the bore of the high pressure stage, a radial bearing for the other shaft in the other cylinder head, a division wall between the sections having a bore closely surrounding and concentric to the rotor, the rotor being provided with vane-slots extending throughout the length of the rotor, each slot carrying a low and a high pressure vane, two sealing vanes between said vanes, and surrounded by the division wall, one of the sealing vanes being provided with an opening near its outer edge corresponding to a circumferential groove provided in the rotor, packing in the opening and groove, a spring holding the packing against the rotor and the sides of the groove, the other sealing vane having a tongue and a segmental rotating guide ring arranged in a groove provided in the division wall, the ring being provided with a slot to accommodate each tongue to prevent longitudinal displacement of the sealing vanes.

10. A two-stage rotary air compressor of the multi-vane type comprising the combination of a casing having a generally cylindrical bore, cylinder heads closing the casing at opposite ends of the bore, a single cylindrical rotor of substantially constant diameter arranged eccentrically within the bore and provided with trunnion shafts, thrust and radial bearing means for one of the shafts in one of the cylinder heads, a radial bearing for the other shaft in the opposite cylinder head, a fixed division wall dividing the bore into a low and a high pressure stage, the division wall having a bore closely surrounding and concentric to the surface of the rotor, the rotor being provided with vane slots extending throughout its length, each slot carrying a low and a high pressure vane, and means in each slot between the vanes and surrounded by the bore of the division wall to close the slots against substantial flow of air from the high to the low pressure stage.

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