March 28, 1967 C. J. SURDY 3,311,291

HELICAL SCREW COMPRESSORS

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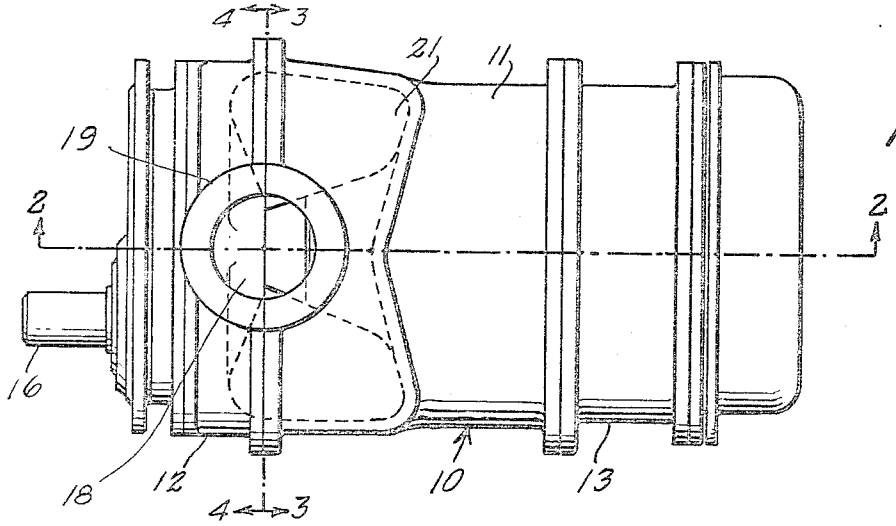


FIG. 1

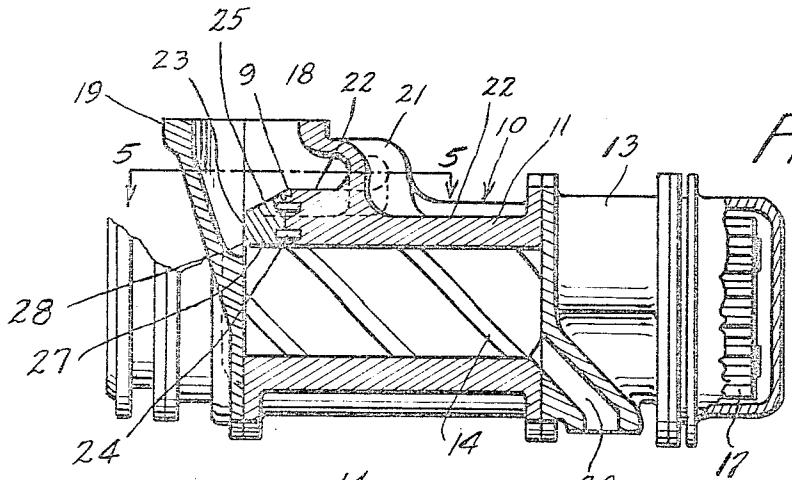


FIG. 2

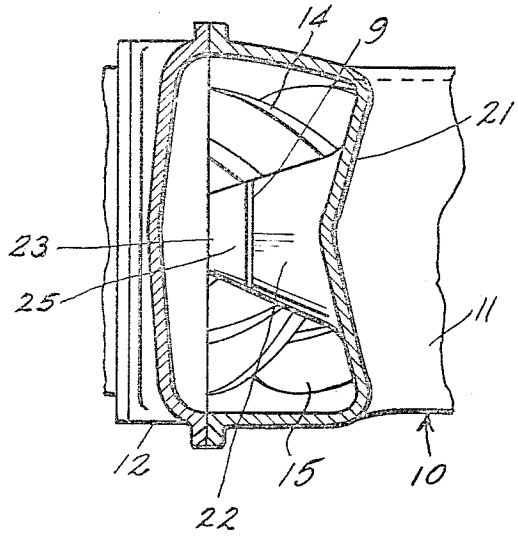


FIG. 5

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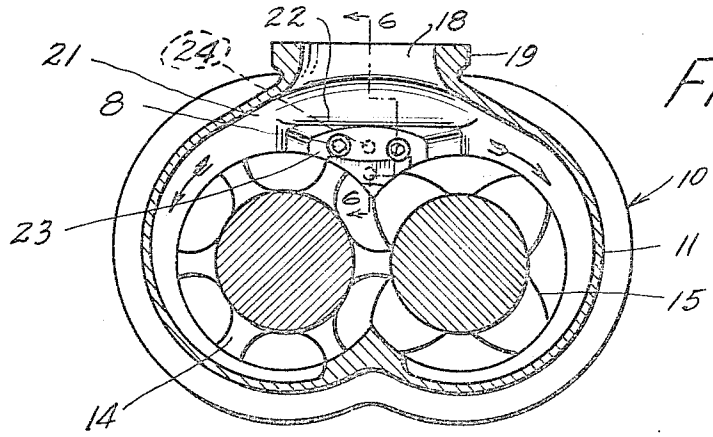


FIG. 3

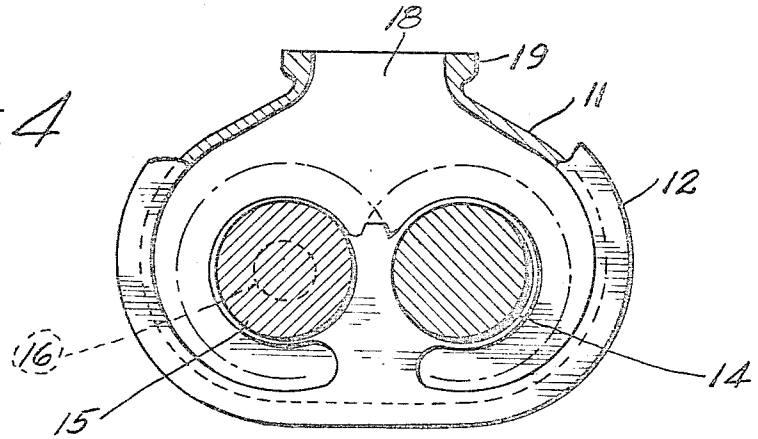


FIG. 4

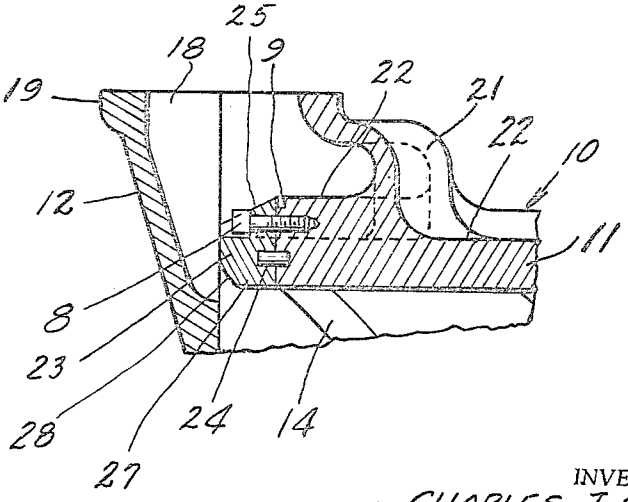


FIG. 6

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FIG. 7

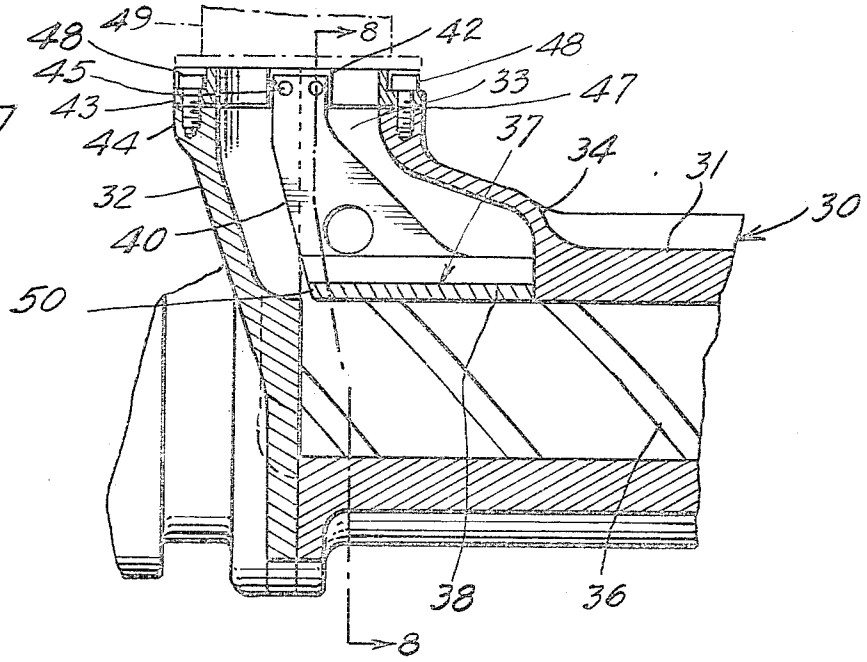
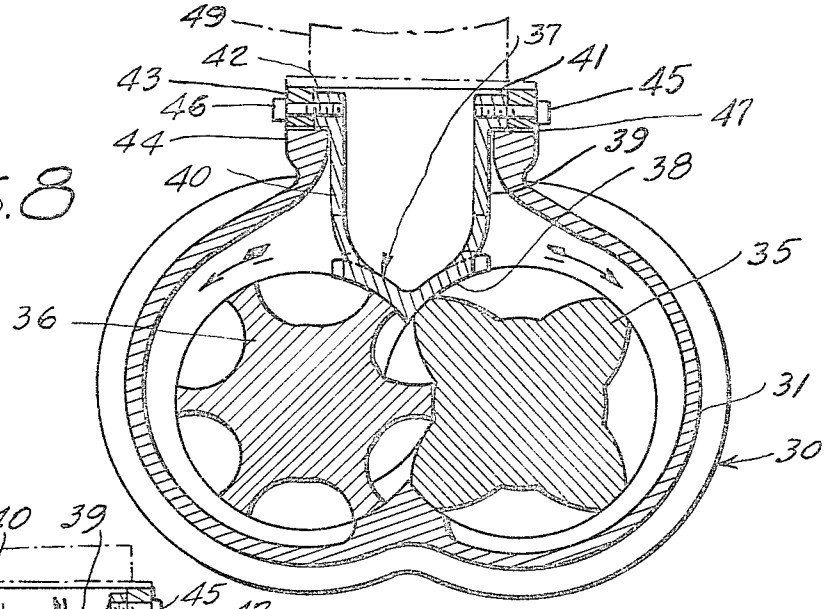


FIG. 8



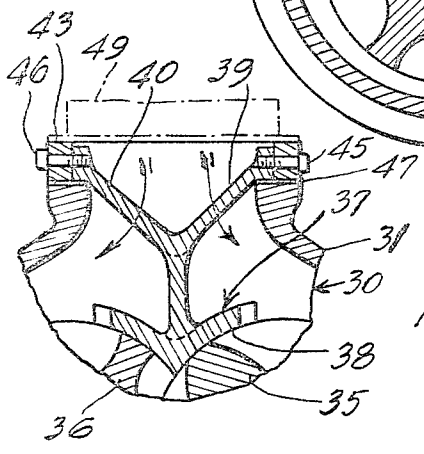


FIG. 9

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3,311,291

HELICAL SCREW COMPRESSORS

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Filed Sept. 16, 1965, Ser. No. 487,832
3 Claims. (Cl. 230—143)

This invention relates to rotary helical screw compressors of the positive displacement type that include two or more helical intermeshing rotors.

The rotors are mounted in a suitable housing provided with appropriate suction and discharge ports, and as the rotors revolve the medium to be compressed is advanced from the suction to the discharge ports in chevron-shaped pockets that diminish in volume as they approach the discharge port. The rotors are not normally in physical contact, small running clearances are provided by timing gears.

Two type rotors are in general use. One type uses generated rotors as shown in Patents 2,287,716 and 2,642,003, granted to J. E. Whitfield. The other type rotors involve use of intermeshing lobes largely circular in cross section. The Patent 2,622,787, granted to H. R. Nilsson, and 3,073,514, granted to W. Bailey et al., illustrate the circular profile rotor form.

It is the object of this invention to improve the intake end of a screw compressor through provision of a removable deflector element embracing a portion of intermeshing screws.

The operation of compressors using either rotor form is greatly improved through the use of the present invention, whether the compression is on a "dry" basis or through an approach to an isothermal process as in W. Bailey et al. Patent 3,073,514.

Referring now to the drawings:

FIG. 1 is a plan view of the compressor suitable for carrying out the object of the invention;

FIG. 2 is a section, generally along the line 2—2 of FIG. 1, with certain parts broken away shown in elevation;

FIG. 3 is a section along the line 3—3 of FIG. 1, with the compressor housing in section;

FIG. 4 is a section along the line 4—4 of FIG. 1 with portions of the compressor housing in section and other portions in elevation;

FIG. 5 is a section along the line 5—5 of FIG. 2;

FIG. 6 is an enlarged section of a portion of FIG. 2;

FIG. 7 is an enlarged view similar to FIG. 2 showing a modification;

FIG. 8 is a section along the line 8—8 of FIG. 7; and

FIG. 9 is a view similar to a portion of FIG. 8, illustrating another modification.

Referring now to FIGS. 1 through 6, inclusive, the compressor 10 comprises the main housing 11, a suction end cover 12, the discharge end cover 13 and helical screws 14, 15. Preferably, the screws 14, 15 are of the generated rotor form with a six-fluted female member 14 and a four-lobe male member 15.

A drive shaft 16, extending from the male screw member 15 rotates this member in the direction of the arrow of FIG. 3. Timing gears 17 transmit power to the female member 14 for rotating the female screw member in the direction of the arrow of FIG. 3.

The suction port 18 is partly in the main housing 11 and partly in the suction cover 12, with suction ends of the screws 14, 15 terminating in a plane normal to the axis of the screws and midway of the circular suction port 18 which has a connecting flange 19.

The discharge port 20 is located at the end of the compressor 10 diagonally opposite the suction port 18 and is partly in the housing 11 and the discharge end cover 13. The air, or other medium to be compressed, enters compressor 10 through the suction port 18 and is advanced by the screws 14, 15 to a position of closure of the suction. Further rotation of the screws advances the gas toward the discharge port 20, and upon sufficient rotation communication is established to discharge the compressed gas through the port 20.

The device described thus far is of conventional form, well known to those familiar with the art.

In order to eliminate or to minimize losses at the suction port that are unrelated to the clearances between rotor screws and rotor screws and housing members, it has been proposed in Whitfield Patent 2,642,003 to cover the suction part of the intermeshed screws by a suitable deflection. This construction, in actual practice, has two main objections.

In the first place, covering of the entire suction portion of the intermeshed rotors with a permanently positioned deflector allows no possibility of checking the clearances between rotors and related parts. Further, such a construction limits operation of the screw compressor to a relatively narrow speed range since at slow rated speed and low pressure, the deflector represents a loss in capacity and consumes additional power. Its addition as a permanent structure is costly and especially burdensome when the compressor is selected for low speed/low pressure operation.

In actual operation it has been determined by tests that at rotor tip speeds less than 40 feet per second, a deflector is not necessary, nor desirable. However, at speeds from 40 feet per second to 150 feet per second, the deflector is required if high losses are to be avoided. These losses are not leakage losses, but losses due to centrifugal pressure created by the rotors which prevents air from entering the rotor pockets. While other minor losses are also present in screw compressor operation, the main ones are those related to clearances and the pressure ratio (leakage losses), and filling losses. It has been observed that when the rotational tip speed velocity of the male rotor is between 40 to 150 feet per second and the corresponding centrifugal pressure exerted by the rotors is in the range from 320 to 4500 microns of mercury above normal suction pressure and contra the inlet flow pattern, significant losses in capacity occur that equal or exceed the normally expected leakage losses.

The critical point is at about 70 feet per second tip velocity of the male rotor, with a corresponding centrifugal pressure of 980 microns of mercury. It is evident that at this speed, axial filling of the rotor pockets is incomplete due to low inspirating effect, while centrifugal pressure is sufficiently high to prevent supplement filling of the rotors through radial gas admission. The removable nature of the deflector contemplated herein will provide a much needed improvement in the screw compressor field, and those skilled in the art will determine from experience when it should be used or eliminated.

A very convenient method of determining the centrifugal pressure in microns of mercury is to take 20% of the square of the tip velocity, expressed in feet per second (ΔP in microns $Hg = 0.20(U^2)$, where U is tip speed of the male rotor in feet per second).

In a well constructed screw compressor, oil flooded for 100 p.s.i.g. service, the leakage or slip losses are approximately 10% to 15% of displacement, with the larger percentage loss in small size compressors.

Losses due to lack of filling of rotor pockets, in the prior construction, account for 15% to 20% of displacement, over and above leakage losses. Therefore, it is evident that elimination of losses unrelated to leakage is a most productive area for screw compressor improvement.

Accordingly, a portion of the main housing 11 is enlarged at the suction end, as at 21 and the top wall 22 is

extended toward the suction end cover 12 but terminates a substantial distance short thereof so that the intermeshed rotors 14, 15 can be exposed for checking purposes and also, so that the suction area for relatively slow speed and low pressure operation may not be unnecessarily restricted, as will presently appear.

Securely fixed by bolts 8 to the main housing wall 21 is terminal section 23. Dowels 24 may be used for mastering the terminal section 23 to the wall 21. The terminal section 23 may be made in varying lengths, but in the embodiment shown, its end remote from wall 21 terminates in the plane common to the ends of the screws 14, 15. The top wall 25 of the terminal section 23 is beveled as at 26, while its lower portion 27 may be chamfered as at 28 to allow for maximum air flow into the screws end-wise.

A groove 9 is formed at the juncture of wall 21 and terminal section 23 for inserting of a suitable tool in order that the terminal section 23 may be moved longitudinally along dowels 24, for subsequent removal of the terminal section 23 through the suction port 18.

It is evident that the terminal section 23 requires careful configuration to aid in the flow of air, and its removable nature facilitates any hand work required to produce the best shape. Dimensionally, the terminal section 23 is so made that its removal through the suction port 18 can be made at will, for the purposes herein set fort.

In FIGS. 7, 8 and 9, modifications are shown that have as their object the application of this invention to existing compressors.

Referring first to FIGS. 7 and 8, the compressor 30 has a main housing 31 and an inlet end cover 32 cooperating with the main housing to form an intake port 33. The housing top wall 34 terminates a substantial distance short of inlet end cover 32 to expose the screw rotors 35, 36 for normally radial admission of air contra the rotation of the rotors 35, 36, best shown by the directional arrows in FIG. 8.

Such a construction dictates relatively slow rotation of the screw rotors and low pressure for most efficient operation; and in order to improve operations without the necessity of disassembly of compressor 30, a deflector 37 is inserted through the intake port 33 so that the arcuate lower wall 38 of said deflector 37 rests upon the outer periphery of screw rotors 35, 36. The deflector 37 has a pair of extensions 39, 40, the outer ends of each of which fit into respective recesses 41, 42 of an adaptor ring 43 secured to the flange 44 of the main housing 31 and inlet end cover 32.

With the deflector 37 correctly positioned upon the screw rotors 35, 36, the outer ends of extensions 39, 40 are secured by bolts 45, 46 to the adaptor ring 43. Then, a suitable shim 47 is inserted between adaptor ring 43 and the flange 44 to provide a running fit between the arcuate lower wall 38 and the periphery of the screw rotors 35, 36. Bolts, such as 48 secure the adaptor ring 43 and deflector 37 carried thereby, to the main housing 31 and inlet end cover 32. A suction throttle valve 49 is indicated by dot and dash lines. The forward edge 50 of wall 38 may stop short of the inner surface of inlet cover 32, depending upon air flow considerations.

FIG. 9 shows a modification of the deflector 37 in that support for the arcuate lower wall 38 is substantially Y-shaped. In some cases, better air flow may be secured in this manner without impairing rigidity of the deflector 37.

When a properly designed deflector 37, or terminal section 23 is used, the spaces formed between rotors, and rotors and housing are completely filled with air, or medium being compressed, and the compressor operates at the design compression ratio. Failure to maintain a full charge in the pockets causes expansion of the gas to a sub-atmospheric condition resulting in operation of of the compressor at a substantially higher than design compression ratios, with consequent losses.

From the foregoing, it will be apparent that by the present invention, a correctly proportioned deflector element can be positioned through the intake port without major disassembly of the compressor. At the same time the deflector element may be removed for proper check of important rotor clearances, and its use dispensed with for low speed operation, increasing the filling cycle of the rotors, reducing horsepower and the cost of manufacture.

I claim the following:

1. In a rotary compressor, a housing with a pair of cylindrical intersecting chambers, each adapted respectively to receive intermeshing helically lobed male and helically grooved female rotors cooperating with each other and the housing to form chevron shaped pockets that progress from one end of the housing to the other end during rotation of the rotors, gas intake and discharge ports in opposite end portions of said housing, said intake port normally exposing a substantial portion of the intermeshed helically lobed male and helically grooved female rotors, the normal rotational speed of said rotors being sufficient to produce a centrifugal pressure equivalent to at least 1,000 microns of mercury above inlet gas pressure and contra inlet flow, the improvement which comprises a deflector element having arcuate walls embracing respective portions of said male and female rotors and providing a running fit with said rotors and being located with respect to said rotors and the intake opening of said housing to restrain radially outward gas flow by centrifugal pressure exerted by said rotors, and means for removably securing said deflector element to said housing, said deflector element being insertable through said intake port for use when centrifugal pressure exerted by said rotors exceeds 1,000 microns of mercury above intake pressure, and said deflector element being removable from said intake port for compressor operation at rotational speeds producing centrifugal pressure substantially less than 1,000 microns of mercury above intake pressure.

2. The improvement as set forth in claim 1 in which the deflector element is positioned in the housing short of an end wall thereof.

3. The improvement as set forth in claim 1 in which the deflector element is provided with extensions for removably securing said deflector in position in said housing.

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