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ELECTRICALLY OPERATED OSCILLATORY COMPRESSORS

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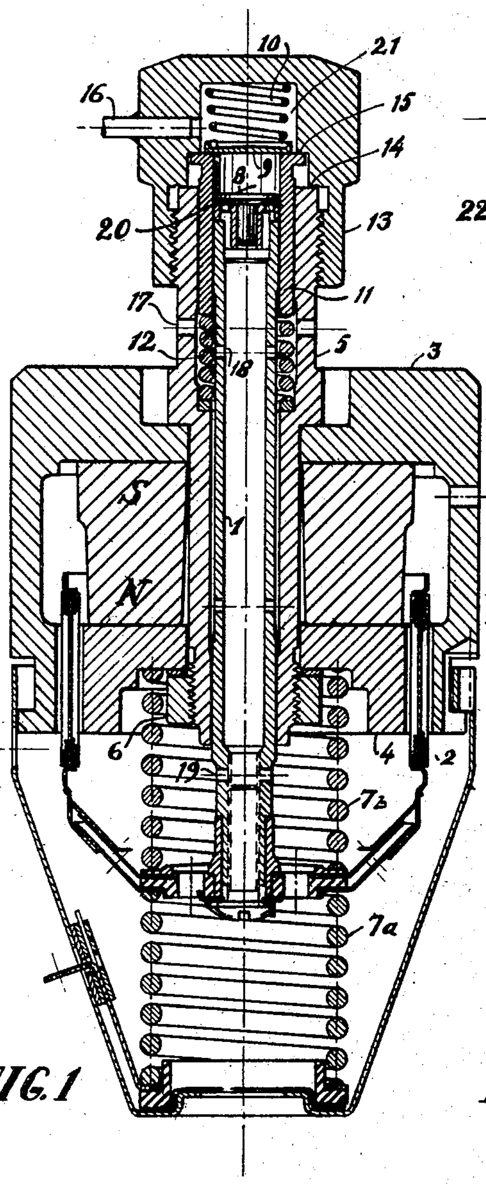

FIG. 1

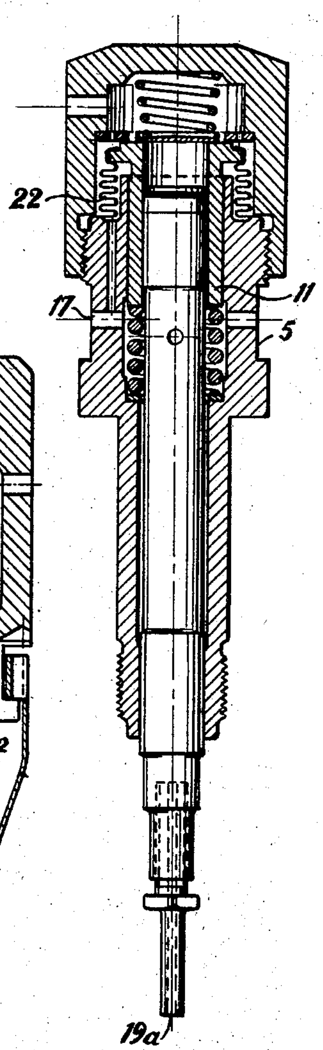

FIG. 2

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ELECTRICALLY OPERATED OSCILLATORY COMPRESSORS

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3 Claims. (Cl. 230—21)

This invention relates to electrically operated oscillatory compressors, i.e. compressors adapted for electromagnetic or electrodynamic operation, having a single-acting pump piston.

In all such oscillatory devices the amplitude of motion of the drive element tends to decrease as the electric voltage decreases, or as the compressor load is raised. The aim of the invention is to provide means for automatically compensating for the adverse effects of such variations, particularly when related to load changes.

The invention will presently be described in conjunction with the accompanying drawing in which Fig. 1 represents a longitudinal section through an embodiment of an electrodynamic oscillatory compressor unit and Fig. 2 represents an alternative embodiment of a corresponding sub-assembly in Fig. 1.

In Fig. 1 numeral **1** designates a hollow piston which is to be axially oscillated by means of a coil member **2** which resembles, and acts like, the moving coil in an electrodynamic loudspeaker. The coil member is movable in an air gap which is magnetized by means of a stationary permanent magnet denoted by letters NS. The magnetic lines of force are directed transversely through the coil member by means of soft steel bodies **3** and **4**. Said bodies are secured to the permanent magnet by means of a central non-magnetic cylinder member **5**, preferably made of a bronze alloy, and a nut **6**. The oscillatory system comprising the piston **1**, the coil member **2**, and a hub member is to be tuned approximately to the frequency of the driving voltage, i.e. normally to 50 or 60 cycles per second, by means of the coil springs *7a* and *7b*. At its upper end the cylinder body **5** is widened to accommodate a cylinder sleeve **11** which is axially movable. When the compressor is at rest, and no gas pressure is built up, the sleeve is held, by means of a coil spring **12**, in continuous contact with a stop at **15**. This stop is effected by an offset in a threaded cap member **13** which covers the top of the cylinder body **5** and tightens against it at **14**. The cap member has a central well **21** which serves as a high pressure chamber and to which a conduit **16** for compressed gas is attached. Against the top end surface of sleeve **11** a pressure valve **9** seats; it is kept in place by means of a valve spring **10**.

The gas to be compressed (the low pressure gas) enters the compressor by the hollow piston through aperture **19**, and also through the passage **17** and **18**. When the compressor is in operation the low pressure gas is drawn into the compression space through a ring type suction valve the ring disk of which is too thin to be shown, considering the scale of the drawing. This disk is kept in location by a valve holder **8** which is firmly secured to the piston. The compression space is essentially that space **20** which is situated between the flat head of member **8**, and the pressure valve **9**.

The device shown in Fig. 2 corresponds to the sub-assembly comprising members **1, 8, 9, 10, 12**, and **13** in Fig. 1. It differs from said sub-assembly essentially to the extent that a metal bellows **22** is provided for sealing against gas leakage between the cylinder body **5** and the cylinder sleeve **11**. At the bottom end of the assembly a central suction aperture *19a* serves essentially the same purpose as aperture **19** in Fig. 1.

The compressor works as follows. Let it first be assumed that the coil spring **12** has sufficient stiffness to keep the cylinder sleeve in permanent mechanical contact with the end cap at **15**. Let is also be assumed that the working voltage is normal and the compressor load comparatively small. The compressor should then preferably be so adjusted that member **8**, at the extreme topward position of the piston, lifts away the pressure valve **9** from its seat a small distance, sufficiently small not to cause any disturbing noise owing to the impact between members **8** and **9**, or between the valve **9** and its seat. Now if the load on the compressor is increased, say by causing the unit to build up a higher pressure in the high pressure chamber **21**, then the amplitude of piston motion will be reduced by an amount approximately corresponding to the increase in electric current demand multiplied by the electric resistance of the moving coil. The amplitude reduction attendant to a high enough load increase prevents member **8** from fully reaching the pressure valve **9**, i.e. causes a dead clearance to form between said elements, with a resultant and more or less sudden drop in unit performance. In addition to, and simultaneously with, the amplitude reduction there will also take place a downward "drift" of the mean position of the piston. The magnitude of said drift may be computed from the piston area, the mean pressure increase in the compression space, and the combined stiffness of the springs *7a* and *7b*. Said drift will for analogous reasons cause a further drop in the unit performance.

Now if the stiffness and initial force of the confined coil spring **12** are more adequately chosen the working mode of the compressor will alter as follows. When only a low overpressure prevails in the high pressure chamber **21** there will be no change, however. But as the high pressure is increased over and above a value determined by the initial spring force the mechanical contact at **15** ceases, and the sleeve commences to drift downwardly, under the influence of the increase in the high pressure. The amount of sleeve drift will, however and for obvious reasons, vary to some extent during the piston cycle. The sleeve oscillation amplitude may be reduced at will by enlarging the annular cross sectional area of the sleeve and of the coil spring section. What is more essential for the proper unit operation is, however, the actual sleeve position during the gas expulsion period, i.e. when the pressure on the opposite sides of the pressure valve **9** are approximately equal. During said phase of operation the drift of the sleeve is determined by the balance between the gas pressure differential acting upon the annular cross sectional area of the cylinder sleeve and the oppositely directed force from spring **12** (not counting the weak spring **10**). By gas pressure differential should be understood the difference between the high and low (inlet) pressure.

By the method described the amplitude reduction, as well as the piston drift, may be compensated for, at least under the condition that the compressor load increase depends solely on a rise in the high pressure alone, such as will be the case in ordinary air compressors. If the unit is to be used for refrigeration, however, load variations will be caused also by changes in the low pressure. In such cases it is advisable to employ a slightly softer compensating spring **12**, since the operating conditions in a refrigerating compressor are likely to be such that a high condenser temperature (higher full pressure) is attended by an increase also in the evaporator temperature (higher low pressure).

It might be understood from the foregoing that it should be advisable to let the sleeve drift compensate for the larger part of load changes whereas the valve lift action (according to prior art; compare U.S. Patent 2,054,097) should provide the remainder of the compensation. Although it should be possible to let also the operating voltage exert an influence on the sleeve position it does not appear practical to do so, at least not in small size units. It is therefore preferred so to adjust the unit that at normal operating voltage the pressure valve is lifted away from its seat a distance equal to, say, 5% of the amplitude of the piston. Thus a voltage drop up to and including 5% will not give rise to any performance reduction attributable to undue cylinder clearance.

By splitting up the compensation between a drift in the sleeve position, and a valve lifting action according to prior art, that advantage will be gained that the mechanical action upon and from the pressure valve will always remain small, and that, as a consequence, the compressor will run quietly, no matter how—within reasonable limits—the load conditions and operating voltage may change.

The choice of bronze or the like for the cylinder body material is explicable not only by the advisability of employing for this part a non-magnetic substance (lest the field from the permanent magnet be partially short circuited). It is also of importance to use a material the thermal expansion coefficient of which is slightly larger than for the steel employed in the piston and sleeve which run at a higher temperature.

Any operational difficulties owing to gas leakage between the cylinder body **5** and the sleeve **11** may be obviated by resorting to the bellows seal **22** according to Fig. 2.

I claim:

1. In an oscillatory electrically operated single acting, single piston compressor in which the piston stroke varies

with the operating voltage and decreases when increasing the compressor load, in combination, a tubular shaped piston having a suction valve at its high pressure end and inlet ports at points removed from said high pressure end, and axially movable piston sleeve in which the high pressure end portion of the piston is arranged to travel, and a stationary housing surrounding said sleeve; said sleeve having an annular end face exposed to the high pressure gas, and an opposite annular face exposed to the low pressure gas, a compression spring encircling said piston and acting on said low pressure face of said sleeve, and a pressure valve seating against said high pressure end face of said sleeve, the axial position of said sleeve and hence of said pressure valve thus being determined during the gas expulsion period substantially by the difference in high and low gas pressure acting upon the annular sleeve area and by the resiliency of said spring.

2. An oscillatory compressor in accordance with claim 1 wherein a stationary cap member attached to the stationary housing serves as a high pressure chamber, and wherein a stop is provided in said cap member to limit the movement of the sleeve, part of the spring force being exerted against said stop member during the gas expulsion period when the compressor is being operated at light loads.

3. An oscillatory compressor in accordance with claim 2 wherein a bellows is provided between the sleeve and the surrounding stationary housing, and wherein the effective annular sleeve area is the difference between the mean cross-sectional area of said bellows and the cross-sectional area of the piston.

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