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High Efficiency Reciprocating Compressors

D. R. Riffe

HIGH EFFICIENCY
RECIPROCATING COMPRESSORS

Delmar Ray Riffe, Senior Research Scientist
Westinghouse Electric Corporation
Research Laboratories
Pittsburgh, Pennsylvania 15235

A 0.52 cu.in. single main bearing, reciprocating refrigerator freezer compressor utilizing a unique unitary connecting rod wrist pin and a unique notched piston and cylinder was designed, built and tested. The compressor developed 1067 Btu/hr with an EER of 4.78 at the standard test point.

INTRODUCTION

Higher energy costs, electrical scarcities, and cases of government intervention have had an effect on the thinking of equipment designers, particularly the reciprocating compressor designer because refrigerator freezer, air conditioner, and heat pump compressors are high energy consumption devices.

It is suspected, and to a limited extent born out by observation, that this energy crisis is not severe enough to make a significant change in the average consumer's buying habits. He will continue to buy a refrigerator or freezer primarily on cost, appearance, convenience, and personal preference with at most only slight consideration given to operating cost. Industrial and commercial users are a little more aware of the cost of the power required to operate their equipment, however. But to a large extent they also buy on initial cost. It is with this philosophy in mind that a two-phase refrigerator freezer compressor development program was undertaken.

One phase of the program was to try to better understand what contributes to compressor efficiency[1] and what can be done to increase this efficiency and another was to cost reduce the device. The real thrust of the program, however, was an attempt to do both simultaneously. By making some revisions that resulted in substantial cost reductions with essentially no effect on efficiency, by making other revisions that resulted in substantial improvements in efficiency with little or no cost increase, and by making some revisions that resulted in a substantial improvement in efficiency but with a substantial cost increase, a model was developed that was much more efficient (40) than the best current production compressor. It had fewer parts and a simplified assembly procedure resulting in a fundamentally lower cost compressor once final production design details are perfected. A 0.52 cu.in. refrigerator freezer compressor was used for the major part of the test program, and 3.5 cu.in. compressor was used to verify the single main bearing unitary connecting rod-notched cylinder concept in the larger size compressors.

[1]For the purpose of this discussion, as is often done, the term compressor efficiency is actually taken to mean the refrigeration system energy efficiency ratio (EER) with no attempt to measure or even define a real compressor efficiency.

THE COMPRESSOR

As can be seen from Figs. 1 and 3, the compressor is basically a single main bearing, single cylinder reciprocating compressor with a unique unitary connecting rod-wrist pin mechanism and a unique piston and cylinder. Single main bearing reciprocating compressors as used in the past have had some undesirable features, such as either a rather complex separate bolted thrust bearing assembly, a rather complex split and bolted crank bearing assembly, or an excessively large crank bearing large enough to permit the main shaft to be threaded through it. The arrangement used herein does not have any of these undesirable features and is much simpler than any other reciprocating compressor, resulting in both substantial savings in material cost, machining labor cost and assembly labor cost. As can be seen from Fig. 1, the shaft is inserted into the housing from the top and the rotor pressed or heat shrunk into place. Next, the piston is inserted into the cylinder (in this case the ring used is incidental to the general design concept but it is not necessarily incidental to some of the more specific design details). With the crank and piston at or near BDC, a unitary connecting rod-wrist pin mechanism is inserted in place. This operation is made possible by a notch in the cylinder plus a notch part way through the piston. The connecting rod-wrist pin is held in place by a thrust washer and a retaining ring at the crank end of the connecting rod.

The housing is a cast aluminum alloy with a cast iron cylinder liner. The main shaft is fabricated from cast iron and heat treated prior to final

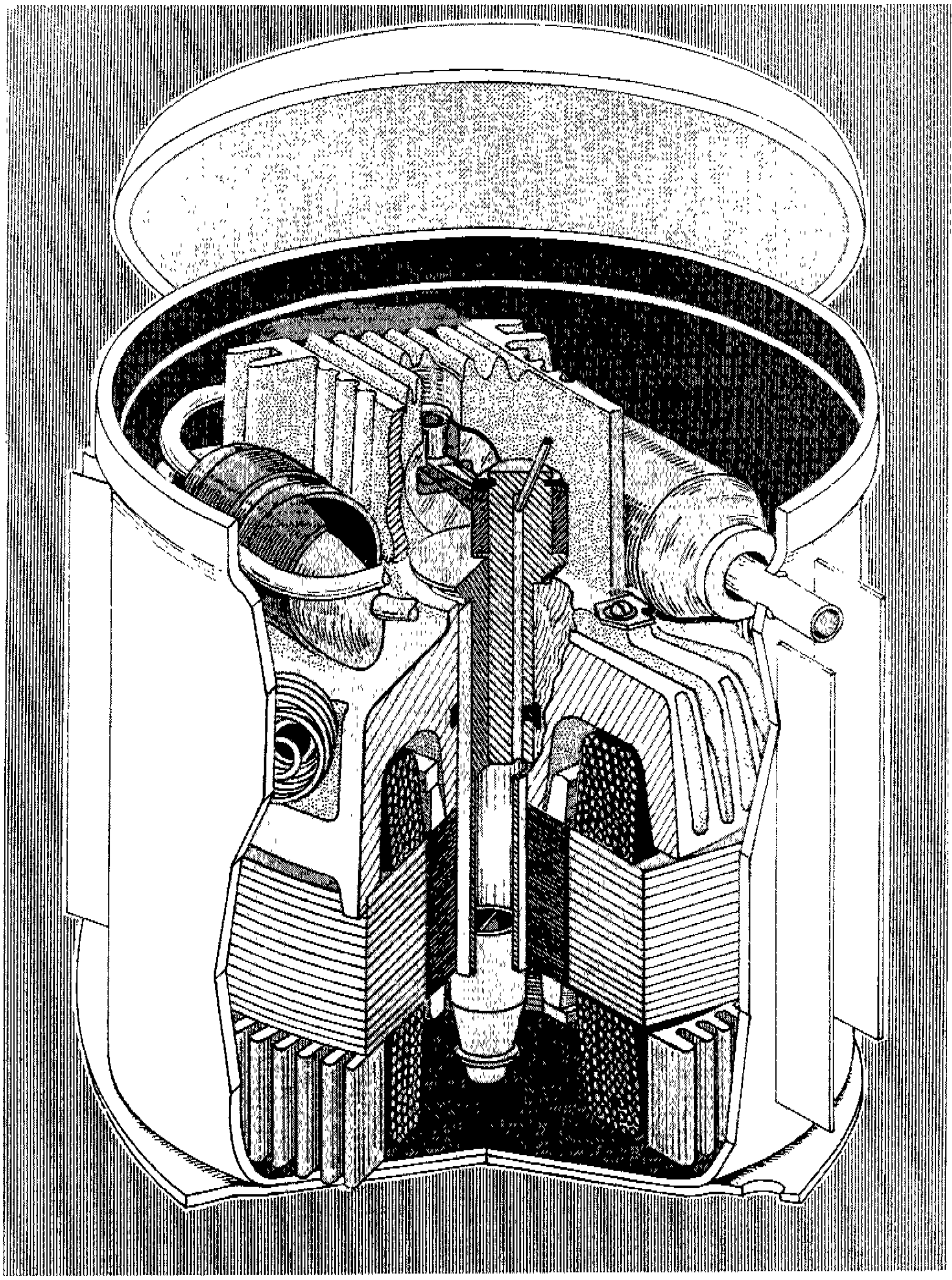

Fig. 1

notching of the piston or the cylinder. However, as mentioned, a ring was used on the piston with a piston diametrical clearance of about 0.002 in.

The single main bearing is 2.00 in. long with a 0.875 in. diameter. Oil is fed up the shaft and through a radial hole in it leading to a circumferential groove around the bearing. The groove is located axially along the bearing at a point near where the load is zero on this cantilever bearing. For purposes of calculating bearing loads and lubrication, it was assumed that the shaft was a cantilever beam supported on a resilient foundation with supporting forces varying linearly with distance from the zero load point where the circumferential groove is located. An axial oil groove runs upward from this circumferential groove to the thrust bearing. (The groove is located radially about 20 deg beyond TDC). Similarly, an axial groove runs downward from the circumferential groove almost all the way to the bottom of the bearing. (This groove is located radially 20 deg beyond BDC.) Room temperature diametrical clearance of this bearing is roughly 0.0005 in. The crank bearing is a conventional design and is lubricated by conventional techniques.

finish machining. The connecting rod-wrist pin assembly is fabricated primarily of 380 aluminum, but two heat-treated tool steel sleeves are pressed onto the yoke at the wrist pin end. These sleeves were turned to finished dimensions before the heat treatment except for a very light grinding operation performed on the sleeve OD after the heat treatment but prior to pressing the sleeves onto the yoke.

The piston is 380 aluminum[2] and contains a standard type piston ring. No dimensional stability or machining problems were encountered in either

The oil pumping mechanism (the mechanism for pumping oil that is circulated within the shell - not any incidental oil that is pumped with the gas through the system) is designed to give as large a flow rate as practical. Observations of the compressor running unloaded in the shell but with the

[2]In the case of one model, a 390 aluminum plug piston was used directly into a 380 aluminum bore with a room temperature diametrical clearance of roughly 0.0004 in. Only a minimum of work was done in this area. Compressor power was somewhat high, and the flow rate was not better than the model using the piston ring.

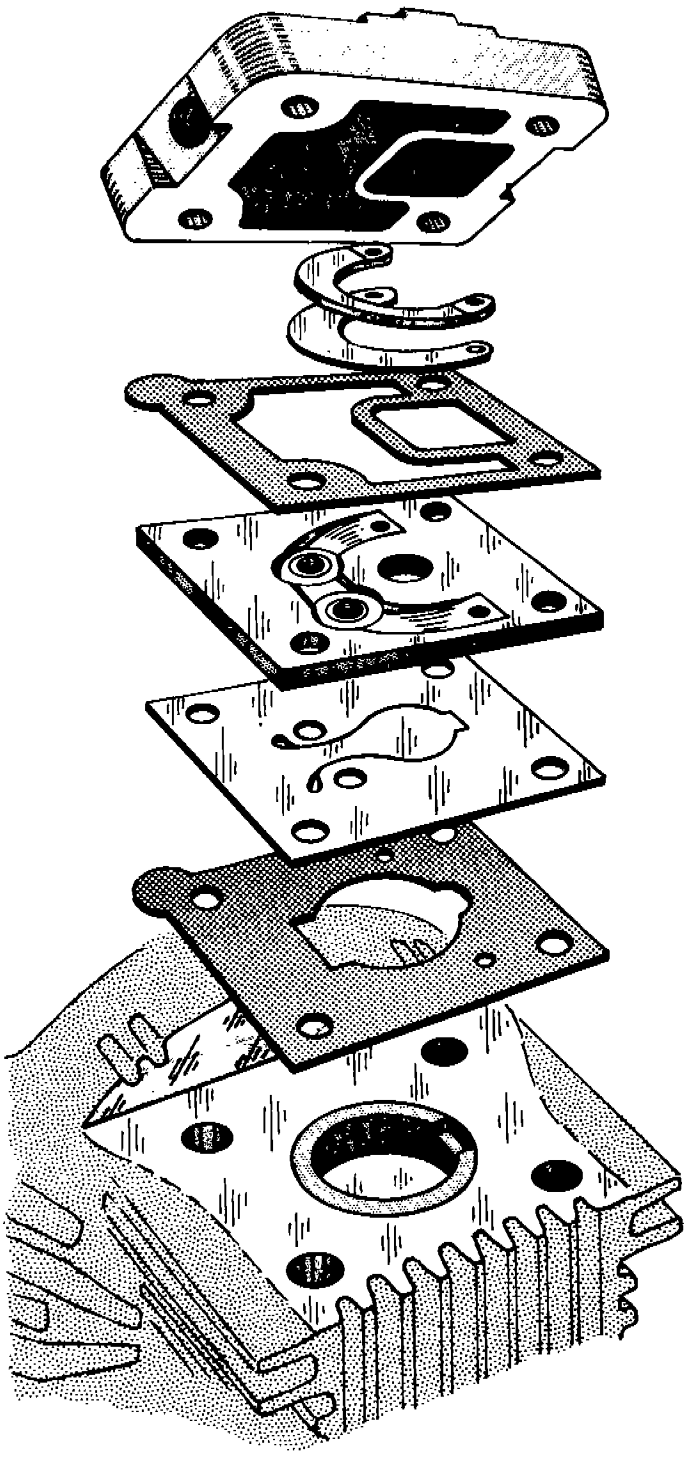

Fig. 2

shell cover removed verifies that there is an abundance of oil slung into the area of the piston wrist pin keeping it well flooded. A strobe light is used to observe this oil flow pattern. The disc on the oil pump adaptor increases the oil pumping rate a little by helping to prevent the inlet from being starved by minimizing the effect of the viscous force plus the centrifugal forces slinging the oil away from the oil inlet hole.

The suction muffler, as can be seen in Fig. 1, is different than conventional mufflers. It was suspected, and at least to some extent substantiated by analysis, that in the case of the high compression ratio, low mass flow compressor most of the compressor cooling is accomplished by the oil carrying the heat from the compressor housing and stator to the shell, as opposed to the heat being carried away by the gas, as this flow rate is not adequate to be effective in cooling the compressor. It was further suspected that in the case of conventional compressors, the gas is heated considerably by the compressor and motor before it enters the cylinder, thus decreasing the volumetric efficiency considerably.

Any proposed correlation between volumetric efficiency and system EER is questionable. However, it was surmised that in the case of the low mass flow, high compression ratio compressor, a large change in volumetric efficiency would have a substantial effect in system EER. The muffler arrangement (shown in Fig.1) was first evaluated in an otherwise standard compressor. It was found that it increased the system EER by 10% at the low suction conditions typical of most refrigerator freezer applications and some low ambiant temperature heat pump conditions. The muffler was not evaluated at high suction conditions. A plastic bushing is used in the end of the muffler to prevent the suction tube from banging against the muffler during starting and particularly during stopping, thus minimizing starting and stopping noise.

The valve mechanism (as shown in Figs. 2 and 3) was nonconventional. First the suction valve, although generally similar in operating principle to current valves, was different in detail. It was formed by a photo-chemical process whereby virtually any geometry, especially very narrow slits, could be produced. Also there is no undesirable edge condition that could lead to fatigue failure. Therefore, there is no need to design a configuration with edges that can be honed. As a result, the re-expansion volume resulting from wasted volumes in the suction valve is very small. The advantages of recessing the discharge valve seat area to reduce the re-expansion volume is, of course, obvious. The particular geometry used here, believed to be unique, permits this seat area to be recessed by a relatively straightforward machining process. After the valve plate is rough-machined, it is mounted on a wedge-shaped

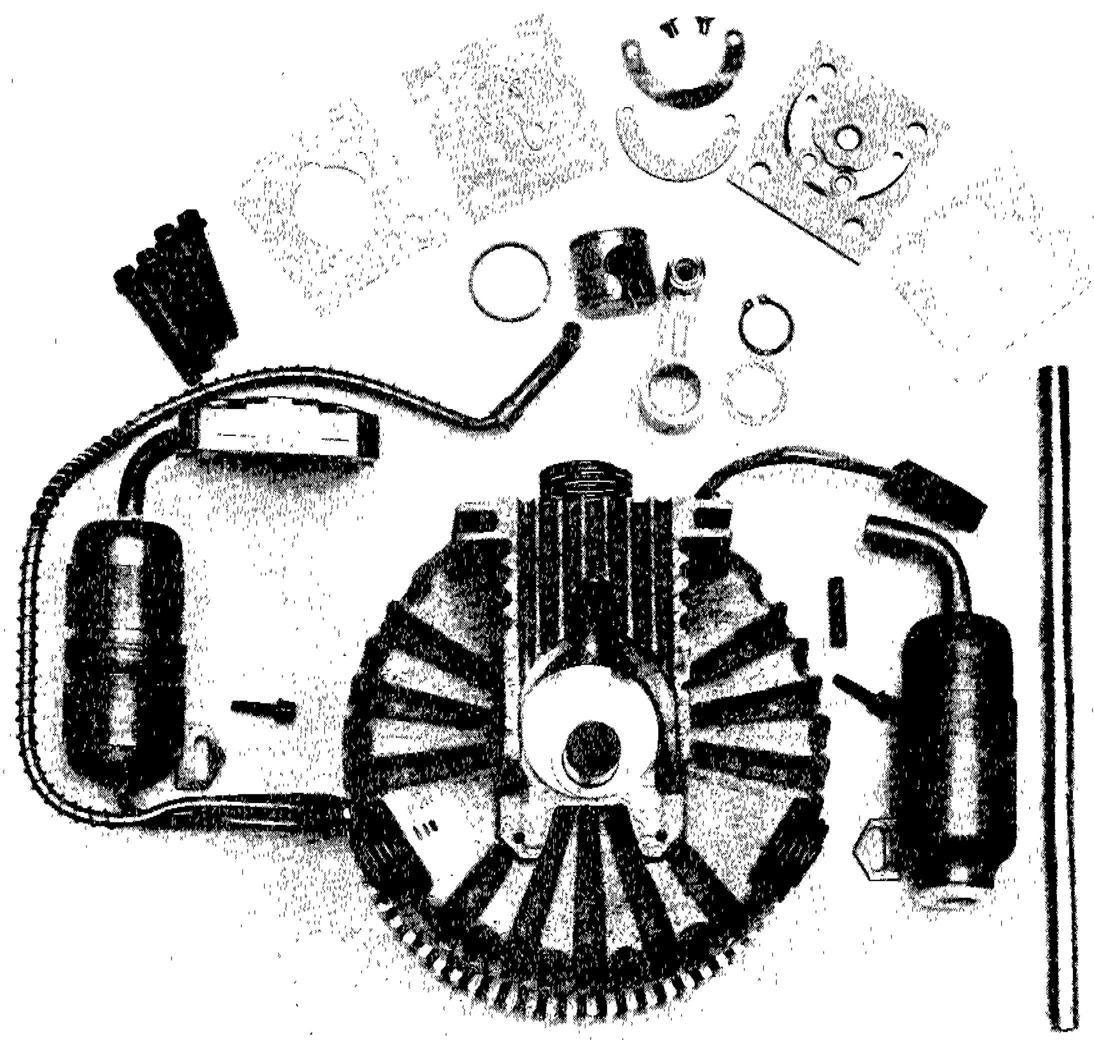

Fig. 3

fixture so that the final sloping seat area itself, as opposed to the valve surface, is horizontal. The valve plate and fixture are rotated about the geometric center of the seat area, while a very thin grinding wheel, not more than 3 in. in diameter, grinds the seat area. The relief on the outer edge of the seat area provides clearance for the grinding wheel to run out. The greater the relief, the larger the diameter the wheel can be.

EFFICIENCY TESTING

It is, of course, well known that the single-phase capacitance motor is roughly 10% more efficient than the split phase induction motor normally used with the refrigerator freezer compressor. This motor contributed about 10% of the 40% improvement in the compressor efficiency; not at a cost of a run capacitor, a start capacitor, and a voltage relay as is normally required, however. A 15-μf run capacitor was used in the auxiliary winding with a PTCR (Sprague Ceroc A260) Shunt. This arrangement was just as efficient running as the more conventional two capacitor and voltage relay arrangement, and the starting torque appeared to be adequate. When tested in a conventional refrigerator, including the simulation of a momentary power failure while operating under peak load condition, the system worked quite satisfactorily. A conventional thermal and current protect device was used. The system would cycle a few times with the current protect device cutting it off. After about three minutes the system would start.

The compressor was evaluated extensively and rather precisely from a performance point of view. The data shown in Table 1 is exactly as taken on the model after the compressor had been run for about 100 hr. Within ±4% the data fits seven identical compressors built as a group. There is no reason to suspect that there is any problem using the compressor in a refrigerator freezer. When testing with the start capacitor and voltage relay, the starting torque is extremely high. It is considerably lower when the PCTR is used, yet it appears to be adequate.

TABLE 1

Cylinder volume	0.52 cu.in.
Speed	3390 rpm
Suction pressure	19.2 Psia (-10F)
Discharge pressure	195.7 Psia (130 F)
Return gas temp.	90F
Liquid temp. corrected to*	90F
Compressor current	2.0 amps
Compressor line voltage	115.0V
Compressor power	223.2W
Compressor PF	0.97
Flow rate	17.21 lb/hr
Capacity*	1067.27 Btuh
EER*	4.78 Btu/whr

*Actual liquid temperatures normally were about 115F and all results are corrected to a liquid temperature of 90F.

Evaluation of the number of parts, number and complexity of machining operation, and particularly the number of assembly steps, leads one to conclude that the compressor could be built at a relatively low cost.

SOUND MEASUREMENTS

Only a minimum of sound level measurements were made. From a balance, structural vibration and 60-cycle sound level point of view, the compressor is excellent. More optimizing is needed in the gas noise end of the spectrum, but the compressor does have a lower total DBA sound level than some production compressors of about equal output. The compressor does have, however, a 0.120-in. thick shell with an 0.180-in. thick cover which is used more for thermal reasons than for acoustic reasons.

Both the housing and the shell are designed with large surface areas in order to improve the compressor heat transfer characteristics. The one-piece shell base serves as a mounting surface but is designed primarily as a cooling surface.

SIZING

It was considered more important to obtain a high efficiency and good reliability than it was to excessively miniaturize the compressor. However, a reasonable profile was obtained. The compressor shell is 7-1/8 in. high and 7 in. in diameter, except it is 7-1/4 in. in diameter over the weld flange. The shell fins, mounting base, electrical bushing, suction, discharge and service tubes all project out from two opposite sides so that the compressor with all its accessories will fit in a space 7-1/4 in. wide by 7-1/8 in. high.

SUMMARY

Based on available test data, indications are that the compressor reliability should be excellent. It has fewer parts than most compressors and runs cooler than most. The wrist pin piston bearing is a little different than most bearings and is difficult to analyze mathematically. In order to test this bearing a standard production compressor was modified to use an earlier model of this connecting rod-wrist pin bearing. The compressor was run at very high pressure ratio (340/19.2) and at a high temperature until the compressor failed after about 4,000 hr. The main bearings had worn so badly that the rotor ultimately rubbed the stator, thus causing failure, but there was no observable wear or damage to the wrist pin bearing.

Seven identical compressors were tested on test stands from about 100 hr each up to 17,000 hr on one compressor with no failure, or degradations in performance or observable wear. One using the PTCR is currently operating in a standard 18-cu.ft. refrigerator under normal consumer environments. It completed the first 700 hr. in the laboratory and has now completed about 12,000 hr. in a home environment.

Although the consumer would probably not notice it directly, the power factor of 0.97 compared to a power factor of 0.65 for a compressor using the more conventional split phase induction motor normally used represents an additional real savings in the cost of energy consumed.

Performance measurements are relatively easy to make. The compressors described herein have been evaluated extensively from a performance point of view. The data presented in Table 1 is believed reliable within a very few percent.

Most of the increase in efficiency is obtained at little or no cost. Also substantial cost improvements can be made just by simplifying geometries.

Improvements in many industrial and commercial refrigeration and air-conditioning applications may not be as great as the 40% obtained for the refrigerator freezer compressor, but all reciprocating compressors can be improved in both efficiency and cost.

ACKNOWLEDGMENT

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