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The Development of Energy Efficient Compressors for Refrigerators and Freezers

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THE DEVELOPMENT OF ENERGY EFFICIENT COMPRESSORS*
FOR REFRIGERATORS AND FREEZERS

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ABSTRACT

An experimental development program to improve the efficiency of compressors used in refrigerator/freezer applications is described. Prototype compressors were constructed and tested with a mean energy efficiency ratio of 5.0 BTU/WH for nominal capacity sizes of 600, 800, 1000 and 1200 BTU/HR The compressors described are entirely suitable for mass production which can occur at the completion of field reliability tests.

INTRODUCTION

This paper outlines the accomplishments of a compressor development program carried out by Columbus Products Company and supported by the Department of Energy through the Union Carbide Corporation. The program involved the development of a highly energy efficient compressor for refrigerators and freezers. The compressor described is significantly more energy efficient than others which are currently being manufactured. A number of prototypes have been built, calorimeter tested and successfully life tested. The project is currently moving into the field test stage. Fourteen hundred refrigerators will be fitted with the energy efficient compressors and placed in the field for reliability verification.

At the inception of this development effort, the energy efficiency ratio of the regrigerator/freezer compressors produced by most manufacturers were on the order of 3.4 BTU/WH. The 17 latest generation prototype compressors constructed have been running on calorimeter tests with a mean coefficient of performance of 5.0, a 47% improvement. The estimated increased factory cost associated with this efficiency improvement cannot be given in absolute terms at this time, but is on the order of 14% as compared to the 3.4 BTU/WH compressor.

While the primary program goal was to improve compressor efficiency, it was clear that because of the highly competitive nature of the compressor business any efficient design developed had to be cost effective and entirely suitable for high volume manufacture. These considerations were used as a basis for many of the design decisions during the course of the development program.

The increase in compressor energy efficiency was achieved in part by returning to a more efficient version of an old idea; the four-pole hermetic motor. Other design changes which have resulted in improved performance were the adoption of a plastic suction muffler, a redesigned discharge muffler and associated tubing, a plastic oil stirrer and revised lubrication system.

The following paragraphs describe the experimental program which resulted in the improved efficiency compressor. In addition to the feature changes and associated energy savings discussed in this paper, numerous other compressor experiments were carried out in an effort to improve efficiency. Many of these results are not reported here since either the energy savings expected did not materialize, or the design feature changes required were not practical from a cost or manufacturability standpoint.

EXPERIMENTAL DEVELOPMENT PROGRAM

A program of evolutionary changes to an existing compressor design was implemented. The alternative approach of developing a completely new compressor was considered, but later rejected because of the extremely high capital investment requirements and the far greater time period required for eventual commercialization. The development methodology involved experimentally evaluating the numerous design changes that were expected to improve efficiency. These changes were applicable in such a

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great variety of combinations that it was impractical to test each and every combination. It was therefore necessary to concentrate on changes or combinations which were judged to be the most promising. The basis for selection in most cases was actual test data from calorimeter testing. In some cases, the suggested changes were considered impractical for high volume manufacture.

Many of the changes that were expected to improve efficiency were fairly easily adaptable to the compressor design configuration existing at the inception of this project. Figure 1 shows a cutaway view of the efficient compressor. It can be seen that the cylinder head, both the discharge and suction muffler and their associated tubing are separate parts. Thus, individual component or complete new subassemblies were easily substituted, using the production compressor as a test vehicle. Similarly, various experimental motors were assembled and trade-off tested to determine the most efficient alternative.

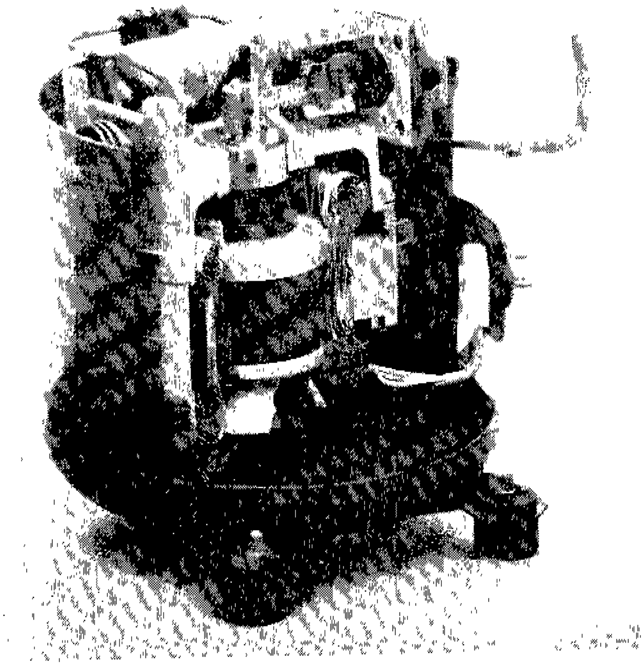

Figure 1: Cutaway of Efficient Compressor

Motor Efficiency Improvement

The simplest, most direct and most effective approach to improve motor compressor efficiency is by way of increasing the efficiency of the driving motor. In compressors applied to refrigerators and freezers, a resistance start induction run (RSIR) motor has been commonly used in the past. The load point efficiency for this type of motor has been historically on the order of 72%. By changing to copper windings and optimizing other motor design features such as stack length increase and reduction of rotor resistance, it was possible to construct and test in compressors, sample RSIR motors with 77% load point efficiencies. When calorimeter tested, however, compressors utilizing these motors did not achieve the coefficient of performance desired, thus experimental work on other motor types was undertaken.

The most common and least costly motor and start system applied to a small refrigerator is a RSIR motor of the type discussed above. This type of motor runs on the main winding only and employs a starting winding with a relay or other switching device to cut out the start winding during normal operation. A simple electrical diagram of the RSIR motor as well as the PSC motor, to be discussed, is shown on Figure 2.

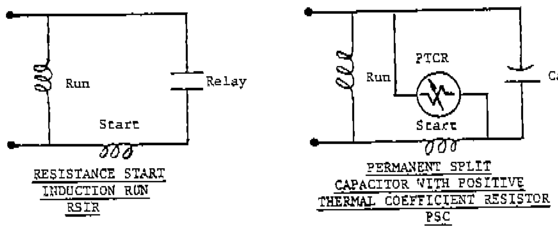

Figure 2: RSIR and PSC Motors

Two or three phase motors are more efficient than single phase but are not suitable for motor compressors because the normal residential power supply is single phase only. Single phase efficiency can be improved, however, by using a permanent split capacitor (PSC) motor. In this arrangement, a capacitor is connected in series with the start winding to introduce a phase displacement so as to partially simulate two phase operation. This simple PSC arrangement is suitable for room air conditioner compressors, but does not provide sufficient starting torque for refrigerator/freezer usage. This difficulty can be overcome by using a device called a positive temperature coefficient resistor (PTCR) connected in parallel with a capacitor. The PTCR has low resistance when cold, so that it allows a momentary high current flow for starting. The current flow through the PTCR, however, causes its resistance to rise rapidly to a very high value, so that in effect, it energizes the capacitor for efficient operation during normal running. Adaptation of the PTCR made the use of a PSC motor feasible for refrigerator/freezer applications. Efficient sample motors of this type were designed and fabricated, and when tested, produced load point efficiencies of 82%. When assembled to compressors which had been otherwise modified to improve efficiency, coefficients of performances on the order of 4.7 were achieved.

The two motors discussed thus far, are both two-pole and at 60 Hertz operate slightly below synchronous speed at about 3500 RPM. For many years most manufacturers of small displacement motor compressors for refrigerators and freezers utilized four-pole motors. A four-pole motor operates at about 1750 RPM from a 60 Hertz source. The maximum efficiency in a manufacturable four-pole motor that may be attained is somewhat less than in a comparable strength two-pole motor due to at least two considerations. These are harmonic content of the magnetic flux and magnetizing force required by the air gap.

The typical two-pole motor will have twelve slots per pole, allowing a concentric winding with five sets of coils that may be distributed to produce a sinusoidal magnetomotive force (mmf measured in ampere turns) resulting in a sinusoidal magnetic flux. Physical geometries of fractional horsepower motors limit the number of slots that may be produced in a four-pole motor, with eight or nine slots per pole the most commonly used compromise. This results in a three, or at the most, four part concentric winding, and as a result the flux wave will be richer in harmonics that tend to represent loss increases or strength reductions of the motor.

In an induction motor, the flux wave produced by a pole crosses the air gap into the rotor then recrosses the air gap to a pole of opposite magnetic polarity returning through the steel of the stator to complete the magnetic circuit. Since the number of air gap crossings are equal to the number of poles and physical limitations cause the air gap length to be similar in two and four-pole motors it can be seen that more ampere turns will be required to pass flux across the air gap in four pole than in two-pole motors of equivalent strengths. The resulting increase in ampere turns in a four-pole motor will produce more current requirement than in a two-pole, thus increasing winding loss.

Calorimeter tests have indicated that the slightly lower efficiency of the four-pole motor is offset by the improved performance of the compressor at four-pole speed. The improvement results from several factors. First, in a given bore configuration, the four-pole models have a longer stroke, which provides better volumetric efficiency because the unavoidable clearance volume is a smaller percentage of the displacement. In addition, valve action may be improved because the intervals during which the valves are opening and closing, and gas flow is restricted, form a smaller percentage of the total cycle. While the gas flow elsewhere in the compressor will be at a similar rate on a steady flow basis, the pressure pulsations are more rapid in the two-pole version, which tends to increase fluid flow losses, since they are approximately proportional to the square of the velocity.

An incidental advantage of the four-pole design is the lower noise level. While this does not in itself reduce power consumption, it does permit the consideration of energy-efficient changes which might otherwise lead to an unacceptable noise level. The first generation four-pole motor samples constructed were of the RSIR type. The load point efficiency of these motors, however, was only 64.5% and even though flow losses were reduced, the resulting compressor coefficient of performance was not high enough to warrant further experimentation with this motor type.

The last motor design considered was the one which was ultimately adopted for use in the efficient compressor prototypes. Both the higher load point efficiency characteristics of a PSC motor design and improved compressor performance resulting from slow speed operation were desired. These factors led to the design, development and testing of a PSC four-pole motor. Figure 3 shows a torque versus efficiency curve for both the RSIR and PSC four-pole motors developed.

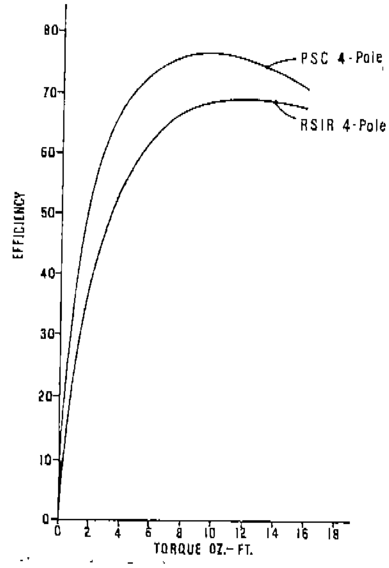

Figure 3: Torque Versus Efficiency of Four-Pole Motors

<u>Mechanical Efficiency Improvements</u>

The advantage associated with the substitution of rolling element bearings for all the plain bearings in the compressor was investigated. Design calculations showed that it should be possible to save about 7 to 10 watts at compressor running speed with the use of needle bearings.

A number of compressors utilizing needle bearings were constructed and tested. The bearing housings were machined to bearing manufacturers tolerance recommendations. When these compressors were calorimeter tested, a power requirement reduction on the order of 12% was achieved. Noise was a serious problem with all the needle bearing compressors.

In order to evaluate the endurance properties of the needle bearing compressors, two samples were subjected to continuous run life tests. The first failed, due to excessive wear of the crankshaft after 2000 hours of operation. The second completed the test run, but on examination at the end of the test, the crankshaft was found to be badly worn. The crankshaft material was pearlitic malleable iron hardened to Rc 58/60.

In summary, adapting needle bearings to a refrigerator compressor has two major drawbacks. The most significant, and probably the most difficult to overcome is the excessive noise characteristic. Secondly, a much harder crankshaft will be required to achieve the necessary compressor reliability characteristics.

Because it had been so clearly demonstrated that a reduction in bearing friction would enable a significant power savings, other attempts were made at reducing the friction of the plain bearing design. It was determined that an increased flow of oil through the bearing resulted in a power savings. Numerous oil pump designs were tested and evaluated based on the quantity of oil pumped per unit of time. Further experiments on the most effective oil pump design showed that when the vent hole in the crank pin was plugged, more oil was forced through the bearings and into the cylinder bore area. An increase in coefficient of performance on the order of 0.2 BTU/WH was measured on a number of test sample compressors with plugged crank pins. The design of the crankshaft oil pump has a second vent hold in the lower end, thus eliminating the possibility of entrapped refrigerant gas in the oil passage.

The function of the oil stirrer is to provide impedance to the sound transmission through the oil in the sump. This impedance is created by the generation of bubbles in the oil. Unfortunately, an effective stirrer from a turbulence standpoint, also utilizes more power. Numerous oil stirrer designs were evaluated in an effort to determine the optimal oil stirrer from a noise and efficiency standpoint. The conical shaped plastic stirrer seen in Figure 1, was effective from a noise standpoint and enabled a two or three watt power reduction as measured on calorimeter tests at 1750 RPM.

Improve Volumetric Efficiency by Reducing Re-expansion Volume

Any compressed refrigerant gas which is not totally exhausted through the discharge valve during the compression stroke is referred to as re-expansion gas. In the compressor under discussion re-expansion volume is found in the following areas:

1. Discharge ports of the valve plate
2. Volume above the piston ring
3. Ports and clearance area of suction reed
4. Piston clearance provided by head gasket
5. Suction valve stop volume
6. Suction valve trepan volume

Experiments were carried out in each of the above areas in an effort to reduce the re-expansion volume to the minimum amount possible. Piston rings with heavier walls were tested. The addition of .005 of an inch to the radial dimension enabled the rings to occupy a larger portion of the ring groove thus reducing the re-expansion volume in this area. Calorimeter test results indicated an increase of 20 BTU/HR was achieved with this change on an 800 BTU/HR compressor.

The suction valve stop was eliminated on a number of cylinder blocks and a special photo chemically etched suction valve with no tip was fabricated. These components were assembled into test compressors and run on the calorimeter at standard rating point conditions. The data from this test series is shown in the following table:

TABLE I

ELIMINATION OF VALVE STOP

REED TYPE	COMPRESSOR DISPLACEMENT	BTU	WATTS	BTU/WH
Production	.52	780	224	3.47
No Valve Tip (or Stop)	.52	918	267	3.44

This data suggests that there was no measurable efficiency improvement associated with the elimination of the valve stop. It is quite possible, however, that this valve reed design was suboptimal from an efficiency standpoint. Because of the entirely different loading characteristics and associated stress patterns of a suction valve with no stop, a lengthly valve reliability verification program would be needed. Considering the apparent lack of efficiency improvement, there was no justification for undertaking such a program.

Flow Loss Reduction

At standard calorimeter rating point conditions the temperature of the suction gas returning to the compressor is 90°F. This temperature is measured before the gas enters the shell. The temperature of the gas entering the cylinder is, of course, much higher. This is because the suction gas which enters the shell circulates freely around various hot parts of the mechanism, is then drawn into the suction muffler and passes through the cylinder. At each point along this path, its temperature increases. The fact that increasing the suction gas superheat reduces the actual amount of refrigerant pumped has been demonstrated both analytically and experimentally in the past. An experimental test series was therefore conducted to investigate the available means of accomplishing this superheat reduction.

An external suction muffler which would be cooler than the internal type was first considered and a sample was constructed. The design, however, presented problems in manufacturing, handling and in returning lubricating oil circulating with the refrigerant to the sump of the shell. Attempts were made to attach a flexible tube to conduct the incoming gas directly to the suction muffler intake. This combination prevents the gas from circulating among hot components. With this combination, it was necessary to include a small vent in the muffler to equalize pressure, and allow the return of oil to the sump. Based on calorimeter test results an increase in capacity and compressor efficiency was realized. The directly connected suction muffler scheme was not incorporated in the prototype because of potential manufacturing problems.

A redesigned suction muffler of a thermoplastic polyester was adopted instead. The muffler inlet was configured with a bell-mouth and positioned to allow suction gas to travel directly from the suction tube at the shell into the muffler. Only the amount of flow restriction necessary to produce the required sound levels was incorporated. The plastic muffler was adopted for a variety of reasons. Irregular curves could be more easily molded in plastic. A large number of samples could be quickly made on a model shop basis for engineering evaluation. It also became apparent that production costs of the muffler would be substantially less than our existing furnace brazed assembly. The most important reason for adoption of the plastic muffler, however, was its contribution to efficiency improvement.

The production discharge mufflers are composed of two stages separated by a solid baffle. A small tube inserted into the baffle provides for the gas flow between the first and second stage of the muffler. One of the first modifications experimented with in an attempt to increase flow, was the installation of a larger diameter tube. Tests on this configuration resulted in no change in either capacity or efficiency. When a second tube was inserted into the baffle and the muffler was tested on a compressor, the efficiency increased 0.1 BTU/WH. This efficiency improvement, however, was obtained at the penalty of an increase in noise. As a result of this marginal improvement in performance, coupled with the increase in noise, these changes were not incorporated in the energy efficient compressor.

An additional flow loss reduction was achieved by substituting a larger diameter discharge tube. Testing showed that a reduction of about 1 or 2 watts could be achieved with the use of this tube. Comparative tests indicated that the slight increase in noise because of reduced discharge tube flexibility still resulted in four-pole compressor noise levels below that of two-pole types of equivalent capacity.

An extensive test series was conducted in an effort to determine the optimal suction and discharge port diameters. It was very difficult to separate the contribution of the discharge and suction port diameter changes to overall compressor efficiency. Because the discharge valve is on top of the valve plate, the discharge port contributes to the reexpansion volume and has a negative effect on efficiency. If the suction port area is increased without an increase in discharge port area then the full improvement in flow losses may not be realized due to increasing losses in the discharge ports.

The optimal suction and discharge port area as well as the number of ports and their location was determined experimentally by testing numerous configurations on the calorimeter. The two-pole and four-pole compressors have different valve port arrangements even though the capacity in BTU/HR may be equivalent.

The foregoing describes the experimental investigations and subsequent evaluation of test data for the areas previously noted as having potential for increasing efficiency. A complete description of all these tests and associated data, is too lengthy for presentation at this time. These test results were utilized, along with all the experimental work previously discussed, as a guide for the development of the efficient compressor described in the following section.

THE ENERGY EFFICIENT COMPRESSOR

The component parts of the energy efficient compressor are shown in Figure 4. The most significant change as compared to the present production compressor is the adoption of a PSC four-pole motor. As was previously discussed, the use of four-pole motors, while not a new idea, resulted in compressor efficiency increases even though the motor efficiency was lower than that of an equivalent two-pole motor.

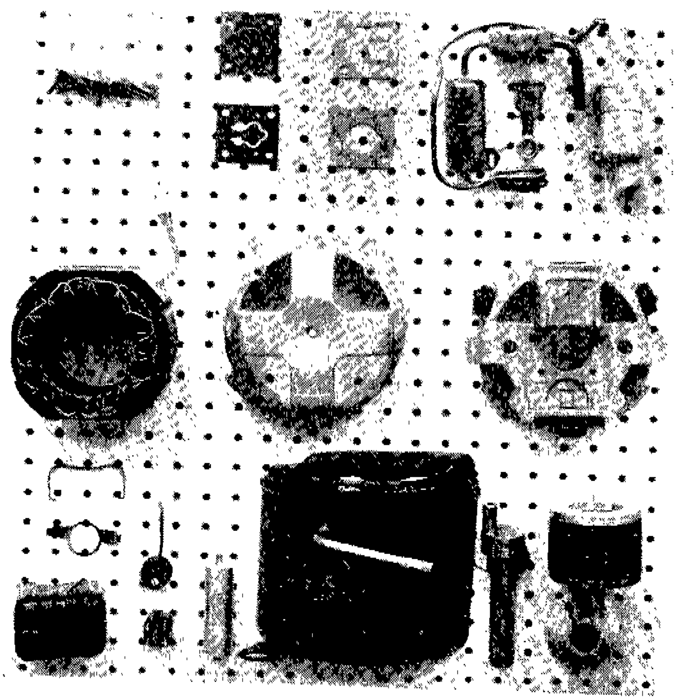

Figure 4: Energy Efficient Compressor Components

The use of the four-pole motor, which runs at half the speed of the two-pole, has necessitated the development of a new oil pump. The present pump would not supply oil to the compressor bearings at an adequate rate when the compressor runs at half speed. In the new pump, the oil is drawn up from the sump through a hole in the center of the crankshaft and then exits through a radial hole into the anular groove shown in the crankshaft on Figure 4. An exit radial hole at the end of the groove allows oil to proceed to a long axial hole up the shank and onto the crank pin bearing. The hole in the end of the crank pin is plugged, forcing more oil into the bearings, resulting in a reduction of bearing friction losses and required power.

The use of a thermoplastic suction muffler has been previously discussed. Of the many configurations constructed and tested, the wide bell-mouth style shown on Figure 4, has exhibited superior performance characteristics and was used on the efficient compressor prototypes.

Experimental four-pole compressors have been built in 600, 800, 1000 and 1200 BTU/HR sizes. In the four-pole design, a capacity size greater than 800 BTU/HR requires a bore larger than the 1.0" diameter currently in production. The bore of these larger displacement compressors is 1.218" diameter. This has, of course, necessitated the development and optimization of different (but similar) valves. Table 2 lists the bore/stroke combinations utilized in the four displacement sizes. The energy efficient prototype compressors have been given the letter W as a model designation.

TABLE 2

MODEL	BORE	STROKE	DISPLACEMENT
W-60	1.000	.820	.64 in^3
W-80	1.000	1.000	.78 in^3
W-100	1.218	.876	1.02 in^3
W-120	1.218	1.000	1.16 in^3

TEST RESULTS

When calorimeter tested at the standard rating point -10°F evaporator and 130°F condensor, the W compressors produced the test data shown in Table 3. There were a total of 17 compressors tested in this group. Motor efficiencies at this load point are also given for the various compressors. Performance curves for the W compressors at 130°F condensor are shown in Figures 5, 6, 7, and 8.

TABLE 3

TYPICAL CALORIMETER TEST DATA

COMPRESSOR	BTU/HR	WATTS	BTU/WHR	MOTOR EFF.
W-60	644.4	129.7	4.97	72.8%
W-80	828.8	164.7	5.03	75.9%
W-100	1068.3	215.0	4.97	74.0%
W-120	1290.5	261.8	4.93	71.5%

Return gas temperatures were held at 90°F., and liquid temperature was corrected to 90°F. The motor efficiencies given are with a winding temperature of 75°C.

Our experience has indicated that if a compressor continues to operate at 30 psig suction and 200 psig discharge on a calorimeter at 98 volts 60 Hertz, it will generally perform adequately in a refrigerator, during a low voltage pulldown test. In order to maximize operating efficiency, the motors utilized have the minimum required breakdown torque. Although two of the compressors exceed the 98 voltage pullout criteria, they do perform satisfactorily in our refrigerators.

The minimum starting voltage was determined for each compressor with balanced pressures of 30 psig and a shell temperature of 150°F. Three minute intervals were allowed between attempts at compressor restarting since this is the minimum off-time of the motor protector. Actual off-times will be greater at lower voltages. It has been determined that the compressors should start at 85 volts locked rotor or less under these conditions, to insure good systems starting performance. Table 4 shows pullout voltage and starting voltage locked rotor of the efficient compressors.

TABLE 4

PULLOUT AND STARTING VOLTAGES

COMPRESSOR	PULLOUT VOLTAGE	STARTING VOLTAGE
W-60	96.7	77
W-80	99.1	70
W-100	101.0	79
W-120	95.5	80

In using a PTCR starting device sufficient off-time, following a protector trip, must be allowed for PTCR resetting. Testing has indicated that approximately 180 seconds are needed for the resistance of the PTCR to be low enough to restart the compressor. In addition to allowing PTCR resetting, the protector must not permit the winding temperatures to exceed 300°F under low voltage conditions. The protectors selected for the W-line compressors exceed 180 seconds reset time at 120 volts locked rotor and stabilized temperatures. In no case do the windings exceed 300°F at 75 volts locked rotor.

Sound power level measurements were made on all capacity models of the efficient compressor. Since test conditions vary within the industry, it is not appropriate to list dBA readings. Under our conditions and in our sound room, the compressors' dBA levels were as low or lower than our production models. The efficient compressors also were as quiet as production compressors of other manufacturers tested under the same conditions.

The efficient compressors were life tested in accordance with established methods. The tests were conducted on load stands in both continuous run and cycling modes. The continuous run test pressures were 10 psig suction and 385 psig discharge. The duration of this test was 3000 hours. The cycling life tests were set up with an equalized pressure of 30 psig. The timing intervals were two minutes running followed by a two minute equalization period. At the end of the two minute running period, the suction pressure reaches a level of 0 to 5 psig and discharge pressure reaches 185 psig. When a total of 100,000 cycles has been accumulated, the test is concluded. Both types of life tests are conducted in a 25°C ambient. A total of 11 compressors were run on the continuous type life test. All capacity sizes were represented. The compressors were calorimeter tested before and after the 3000 hour run to check the effect of this test on performance. None of the samples showed any significant performance degradation.

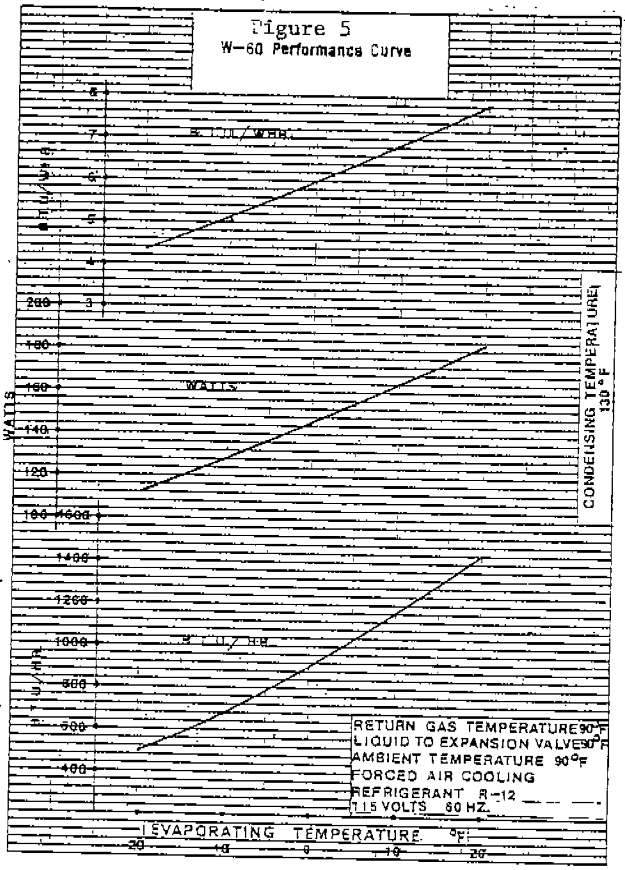

Figure 5
W–60 Performance Curve

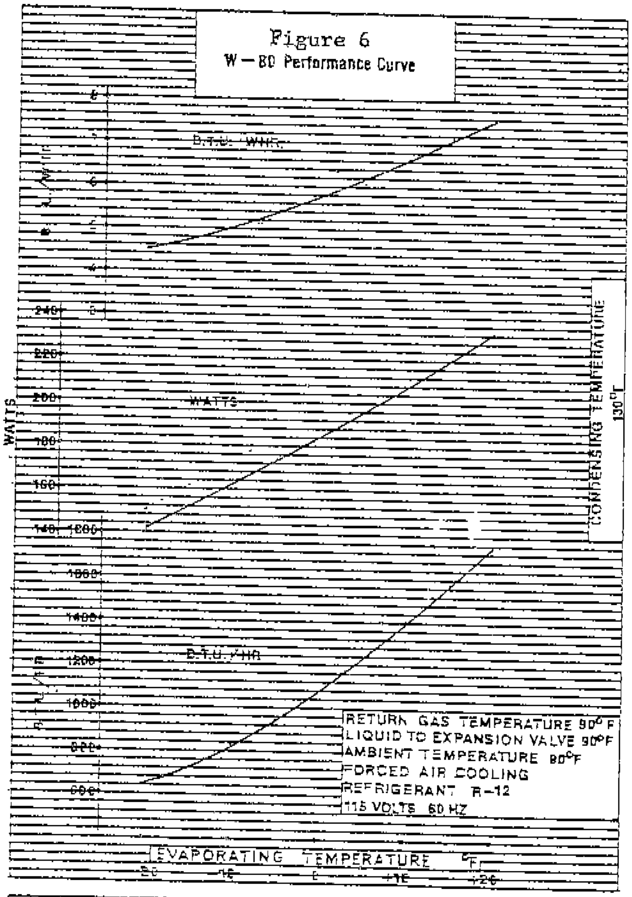

Figure 6
W–80 Performance Curve

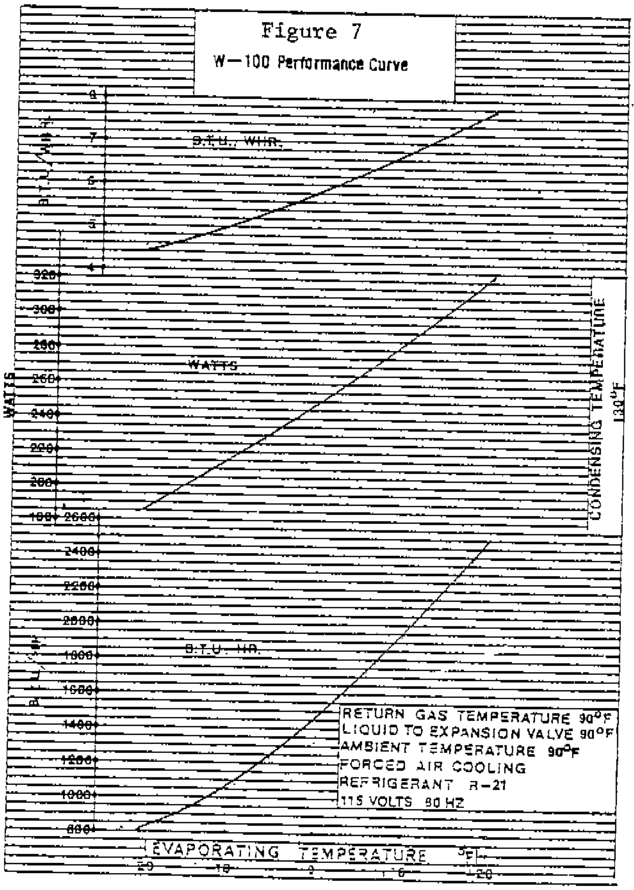

Figure 7
W–100 Performance Curve

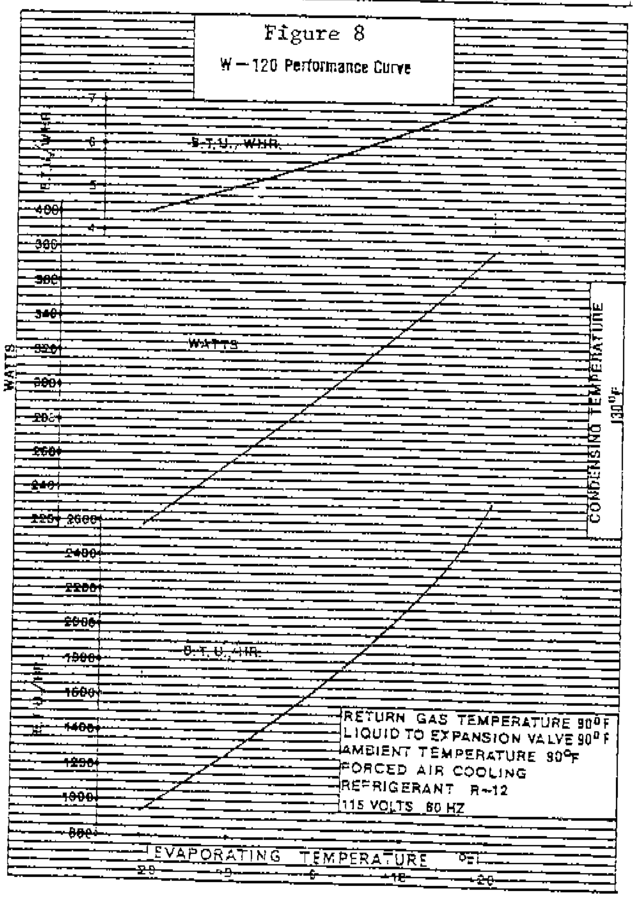

Figure 8
W–120 Performance Curve

The final step in the continuous run life test procedure was to cut open the compressor shells and examine the internal parts for evidence of any changes which may have occurred during the test run. Particular attention was paid to the bearings and to the valves, since these are the parts most likely to show evidence of wear or degradation. The examiner's subjective judgment on the condition of the parts was the basis for conclusions regarding reliability. Stated another way, the samples were judged to have survived the test successfully if the component parts show no more evidence of wear, discoloration or deformation than the component parts of production compressors which have been subjected to the same test. Using these criteria, all the prototype samples passed the test. Four compressors were installed on cycling life test stands. None, at this time, have completed the test but all have accumulated more than 70,000 cycles to date. One of the group has completed over 95,000 cycles.

Calorimeter tests on sample compressors are valuable for measuring the effect of design changes on performance, but complete system tests are necessary for the final evaluation of any efficiency improvements made to the refrigerator. On the calorimeter, the compressor operates at a steady state at standardized test conditions, whereas in a system, these conditions change constantly as the compressor cycles on and off. The system tests which were used to measure the performance of the prototype design have been made per AHAM Std. No. HRF-2-ECFT, or sections of it. In order to obtain relative system data, a standard refrigerator designated RT-18, was tested using a production compressor with a nominal capacity of 800 BTU per hour, and an EER of 3.9 BTU/WHR. The test was then repeated using a prototype sample designated W-80. The test results plotted in accordance with the AHAM Standard are shown in Figure 9 and Figure 10. It can be seen from these Figures that a power consumption reduction of 17.7% was achieved by the use of the W-80 compressor.

A smaller capacity high efficiency compressor, the W-60 was tested in a refrigerator designated RT-12. This refrigerator has only one cold control and does not incorporate an anti sweat heater which can be switched on and off. When the W-60 compressor was tested in this system, power consumption was 59.7 KWH/month. This refrigerator when fitted with the production compressor (now obsolete) used 84.0 KWH/month. A power consumption reduction of 28.9% was achieved. Further systems tests on larger capacity compressors are planned.

ACKNOWLEDGEMENTS

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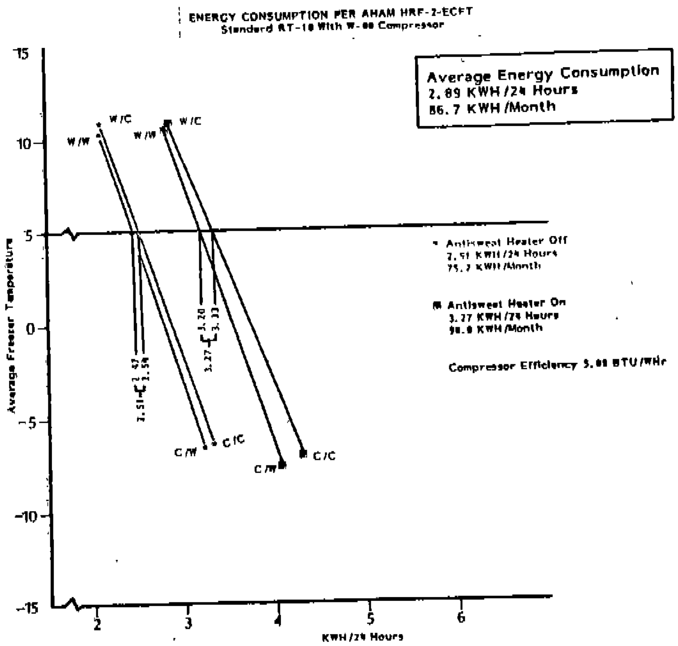

Figure 9

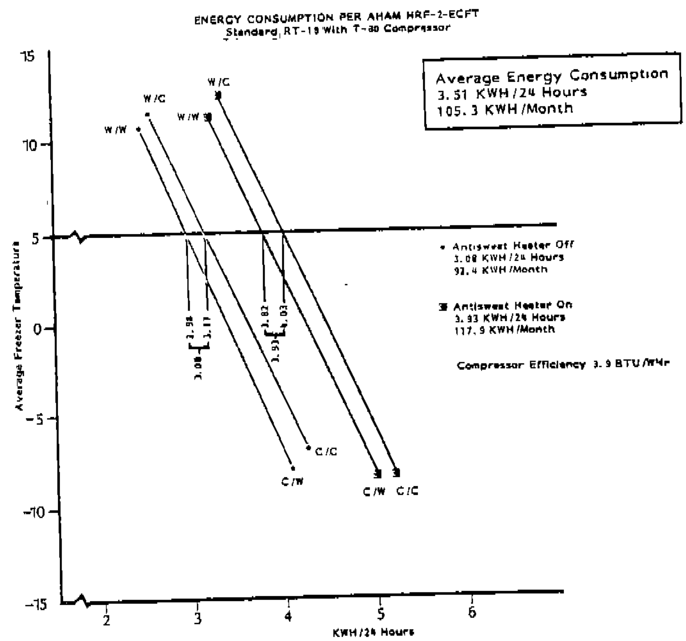

Figure 10