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ENERGY

CONSERVATION

RESEARCH AND DEVELOPMENT OF ENERGY-EFFICIENT APPLIANCE MOTOR-COMPRESSORS

FINAL REPORT

VOLUME 1—EXECUTIVE SUMMARY

Prepared by

Richard T. Nelson

Parker W. MacCarthy

COLUMBUS PRODUCTS COMPANY
300 Phillipi Road
Columbus, Ohio 43228

September 1980

Work performed for

OAK RIDGE NATIONAL LABORATORY

Operated by

UNION CARBIDE CORPORATION

for the

U. S. DEPARTMENT OF ENERGY

Office of Buildings and Community Systems

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The market study was accomplished by a team of second year graduate students from the Carnegie-Mellon University Graduate School of Industrial Administration working under the guidance of Dr. Richard Staelin, Associate Dean and Dr. Robert P. Redinger, Assistant Professor. Ms. Janet K. Felmeth, of the White-Westinghouse Appliance Company was the Project Marketing Advisor. Drs. Staelin and Redinger wrote the Market Study Report..

The Columbus Products Company Engineering Team consisted of Mr. Roger W. Smith and Mr. Marc G. Middleton. Parker W. MacCarthy was the Program Consultant. Mr. Lloyd B. Wilson of the Athens Products Company designed and supervised construction of the efficient hermetic motors. Mr. Richard T. Nelson was the Program Manager and Mrs. Patricia Malone typed this report.

ABSTRACT

The Columbus Products Company, a division of The White Consolidated Industries, has been under contract with Union Carbide Corporation, a prime contractor to the U. S. Department of Energy, for the development of a highly efficient compressor for refrigerators, freezers and room air conditioners. The project included the following major elements:

- Experimental development of highly efficient compressors
- Building and testing of prototypes
- Potential market study
- Formulation of a detailed program plan for a field test

An important goal of the program was that the efficient compressors ultimately reach commercialization. The prototype refrigerator compressors attained an energy efficiency ratio in excess of 5 BTU/Watt Hr. The improvement in operating efficiency, over those which were in production at the inception of the project, is on the order of 44%. In the case of motor-compressors for use in room air conditioners, the best prototype sample to be constructed and tested ran with an energy efficiency ratio of 9.5 BTU/Watt Hr., a 19% improvement.

The purpose of the investigation into the potential market was to gain a better understanding into the process consumers use to decide which model refrigerator/freezer or room air conditioner to buy. The overall conclusion was that consumers were willing to spend a considerable amount on energy savings once they were made aware of the appliance's operating costs and provided with a strategy for incorporating these costs into an investment decision.

An informative presentation by sales personnel was the most important influence on the customer's response. The pay-back period concept was shown to be a valid quantitative mechanism for predicting customer response.

A field test plan for the primary purpose of verifying reliability is outlined. The construction of about 1300 refrigerators with the high efficiency compressor is recommended. A major portion of these will be sold through the White-Westinghouse distributors. Arrangements will be made to monitor the performance of the refrigerators from a reliability standpoint for a period of three years after installation.

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1.0 INTRODUCTION

This report outlines the accomplishments of a compressor development program carried out by Columbus Products Company (CPC), a division of The White Consolidated Industries, and Oak Ridge National Laboratory (ORNL), operated by Union Carbide Corporation Nuclear Division for the U. S. Department of Energy. The program involved the development of highly energy efficient compressors for refrigerator/freezers and room air conditioners (RAC).

The complete project report series consists of three volumes. In this volume, the Executive Summary Report, an overview of the accomplishments and findings of the entire program is presented. The Market Evaluation, implemented by the Carnegie-Mellon University (CMU) Graduate School of Industrial Administration under subcontract to CPC, is shown in Volume II. The technical accomplishments of the project and a recommendation for field trials are shown in Volume III. The essential steps in the project are listed in the Scope of Work, which may be found in Appendix A in this volume.

2.0 REFRIGERATOR/FREEZER MOTOR COMPRESSOR EXPERIMENTAL DEVELOPMENT

The following sections highlight the accomplishments of the refrigerator/freezer motor-compressor development program. The experimental program involved making numerous incremental changes to an existing compressor design. The alternative approach of developing a completely new compressor was considered, but was rejected because of the extremely high capital investment requirements and the far greater time period required for eventual commercialization.

2.1 The Improvement Achieved

The existing compressor model chosen as the base line for efficiency improvement program was designated T52. Typical calorimeter rating point test results for this model, under the test conditions listed in Sec. 2.2 are:

Capacity, BTU per Hr.	780
Watts	225
EER BTU per W. Hr.	3.47

The improvement program resulted in the development of a high efficiency model designated W80. Typical results on model W80 under the same conditions are:

Capacity, BTU per Hr.	800
Watts	160
EER BTU per W. Hr.	5.00

This represents an improvement in EER of 44%.

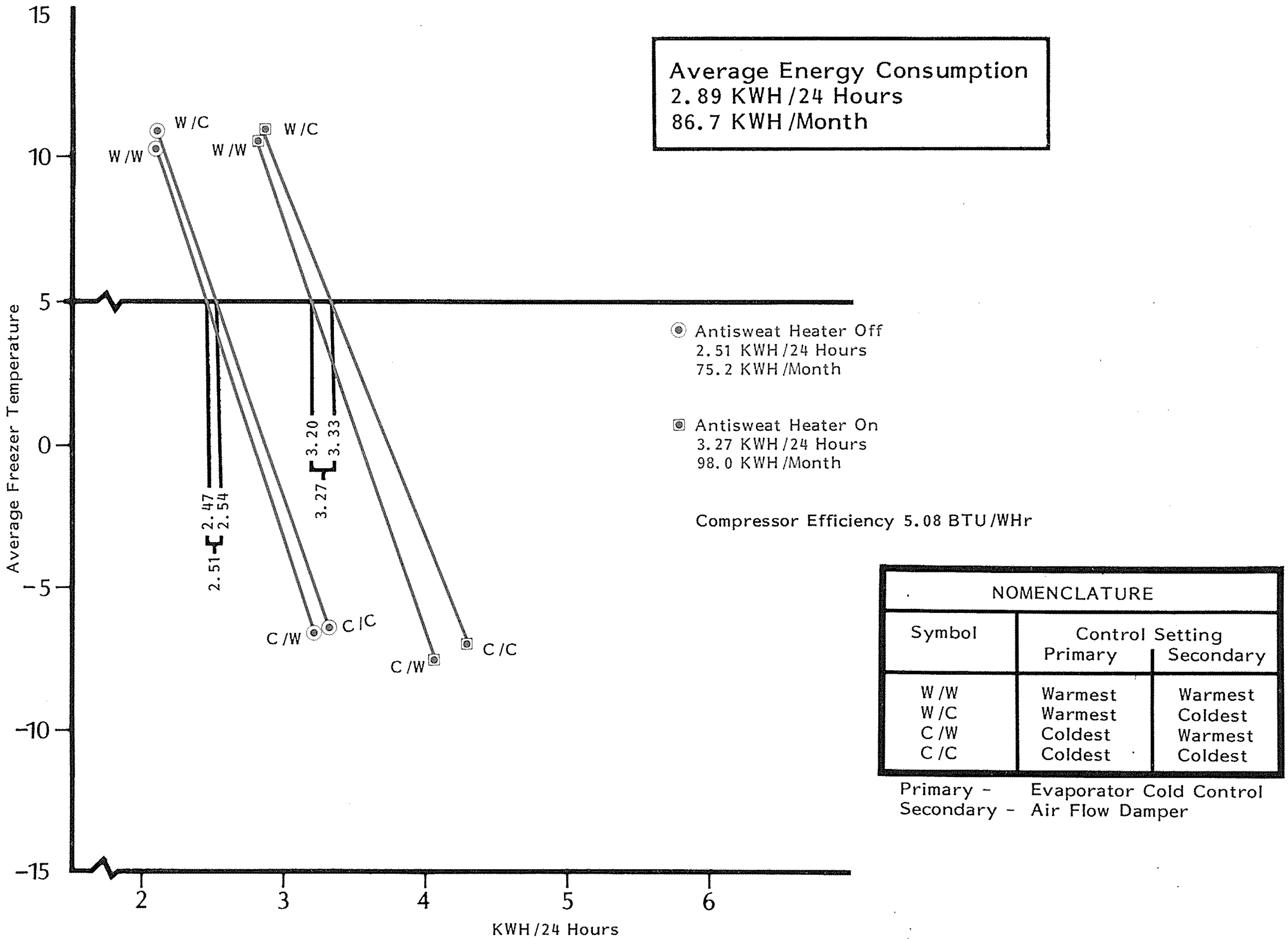
2.1.1 Performance Improvement In Complete Refrigerators

In order to evaluate the efficiency improvements obtainable with the prototype compressors, several tests were made in complete systems. The following combinations were tested. The components are identified as follows:

- T52 Original compressor design at start of project.
- T80 Current compressor in production at CPC, embodying some improvements over T52.
- W80 High-efficiency prototype design.
- RT18 (orig.) Cabinet design in production at start of project.
- RT18 (imp.) Current cabinet design.

The average energy usage of various combinations of the above elements is listed in Table A and on Fig. 2-1, 2-2, 2-3, and 2-4.

ENERGY CONSUMPTION PER AHAM HRF-2-ECFT
Improved RT-18 With W-80 Compressor



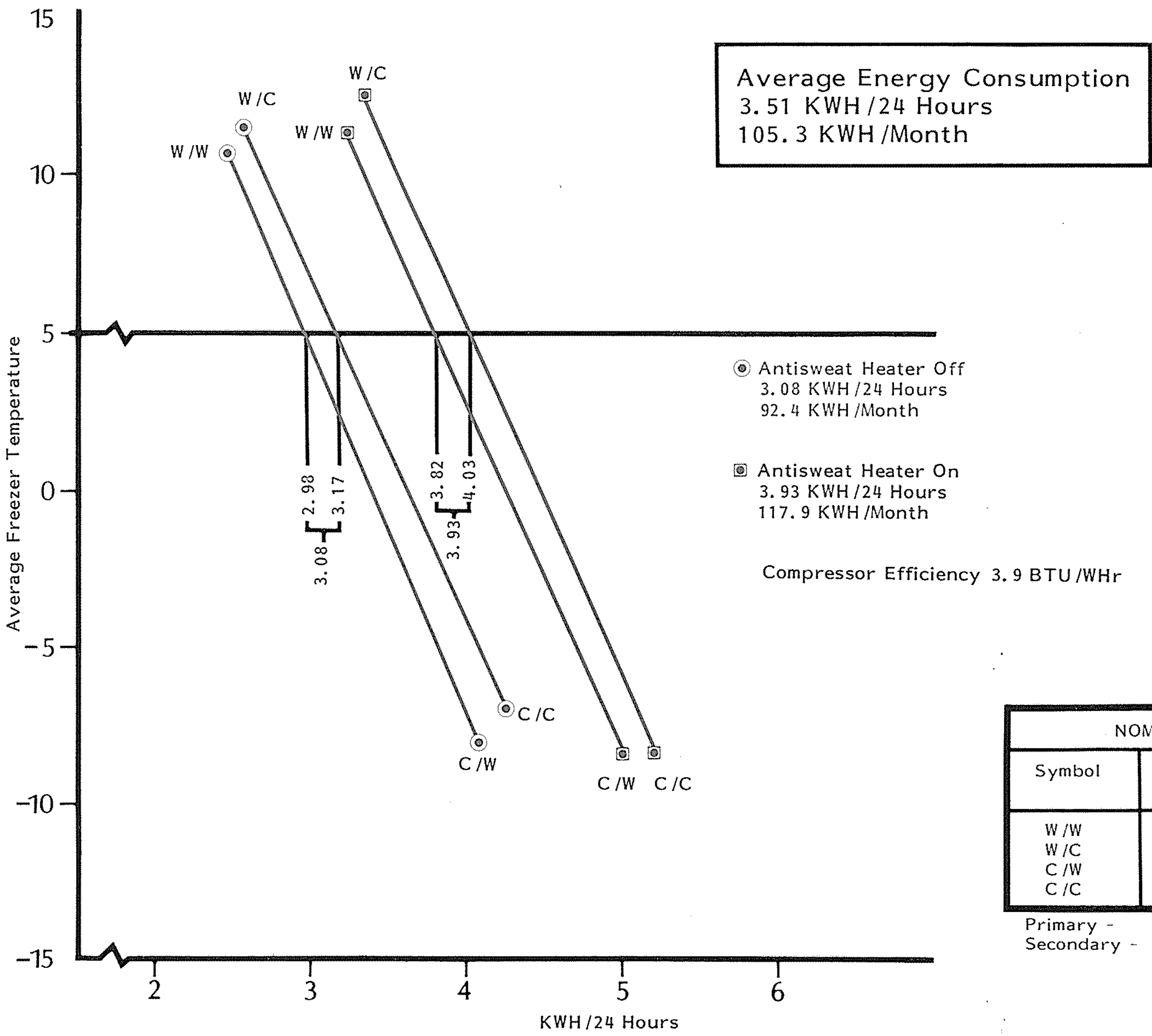
NOMENCLATURE

Symbol	Control Setting Primary	Control Setting Secondary
W/W	Warmest	Warmest
W/C	Warmest	Coldest
C/W	Coldest	Warmest
C/C	Coldest	Coldest

Primary - Evaporator Cold Control
Secondary - Air Flow Damper

Figure 2-1

ENERGY CONSUMPTION PER AHAM HRF-2-ECFT
Improved RT-18 With T-80 Compressor

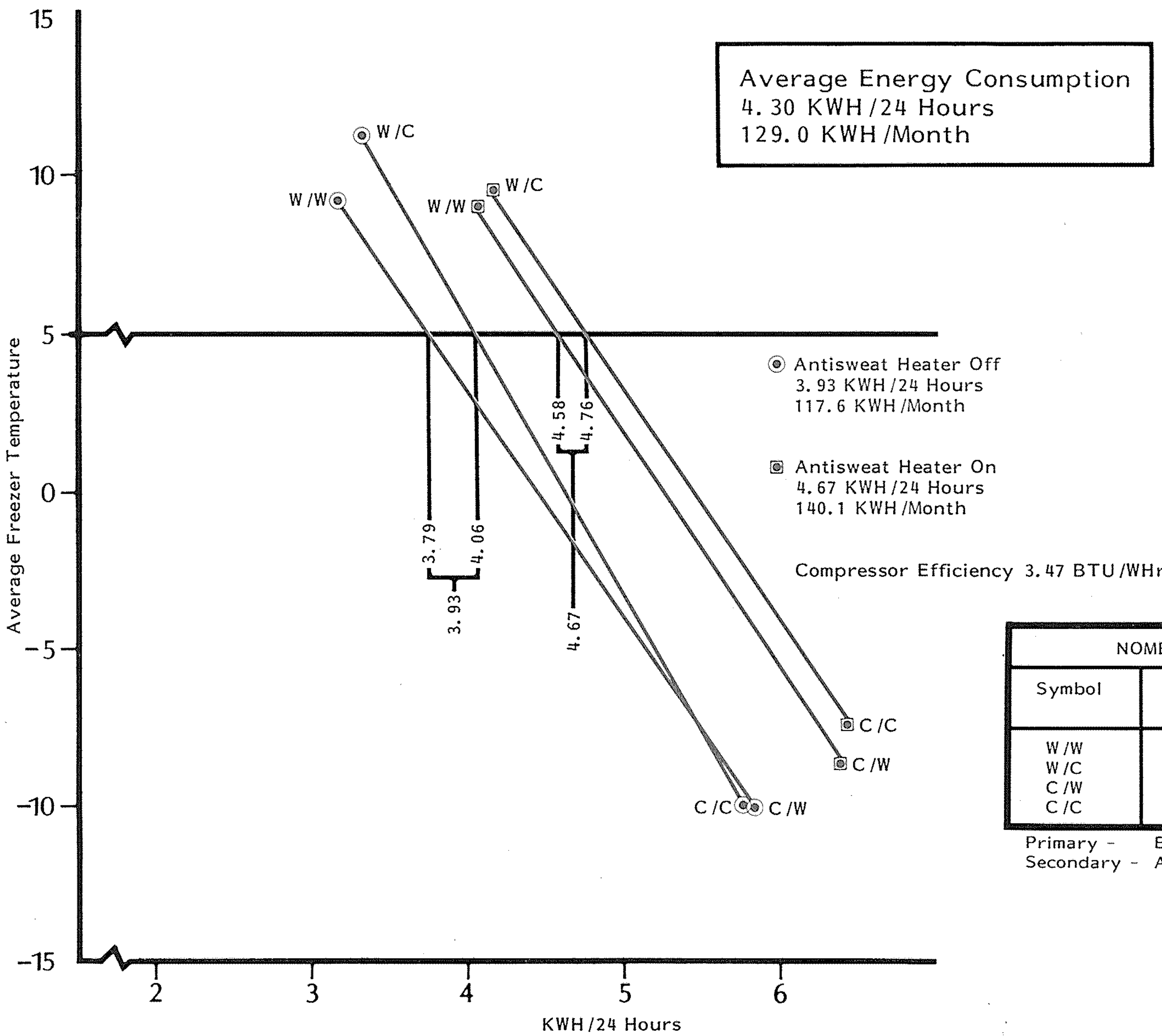


NOMENCLATURE		
Symbol	Control Setting Primary	Secondary
W/W	Warmest	Warmest
W/C	Warmest	Coldest
C/W	Coldest	Warmest
C/C	Coldest	Coldest

Primary - Evaporator Cold Control
Secondary - Air Flow Damper

Figure 2-2

ENERGY CONSUMPTION PER AHAM HRF-2-ECFT
Standard RT-18 with T-52 Compressor

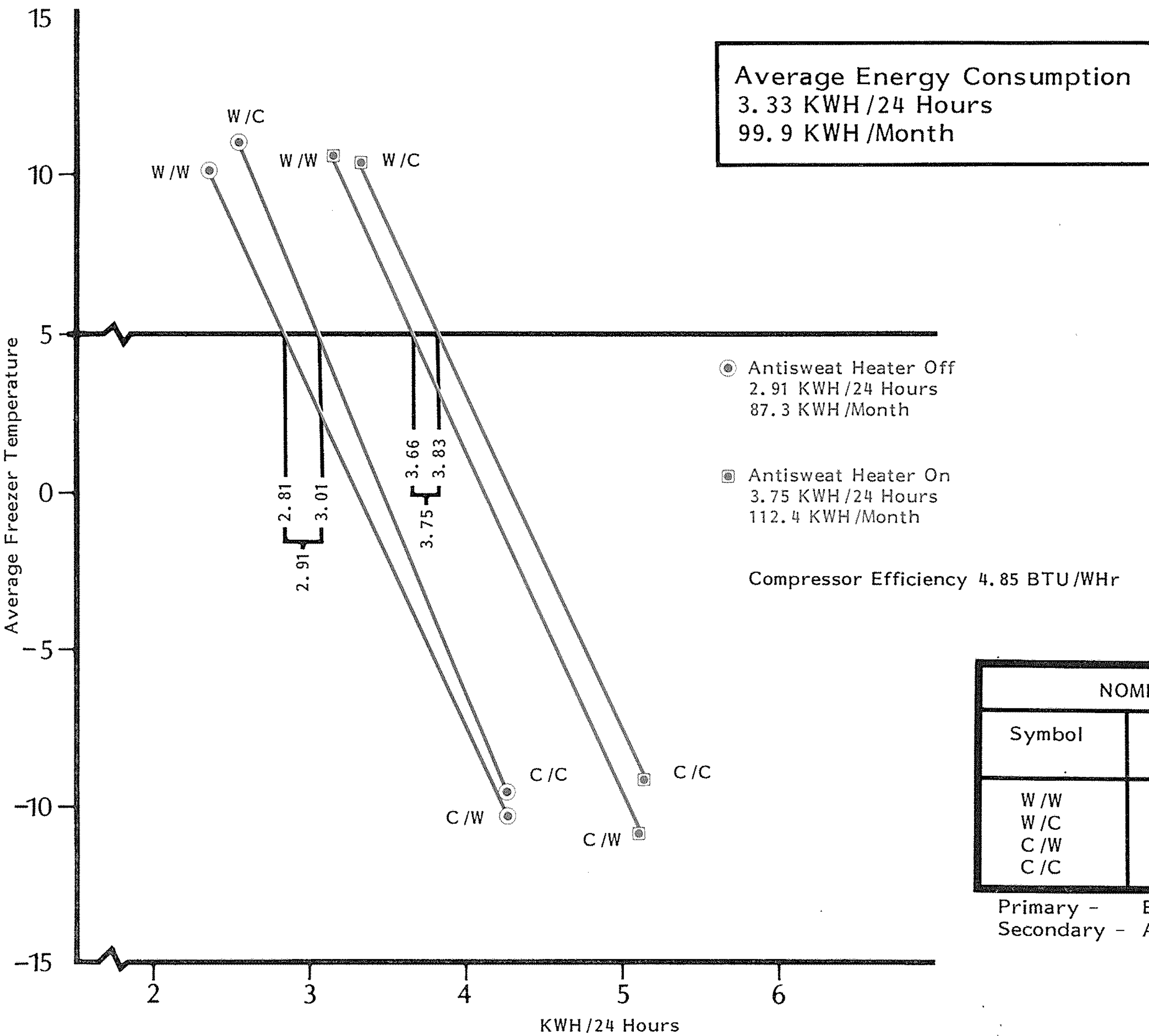


NOMENCLATURE		
Symbol	Control Setting Primary	Control Setting Secondary
W/W	Warmest	Warmest
W/C	Warmest	Coldest
C/W	Coldest	Warmest
C/C	Coldest	Coldest

Primary - Evaporator Cold Control
Secondary - Air Flow Damper

Figure 2-3

ENERGY CONSUMPTION PER AHAM HRF-2-ECFT
Standard RT-18 With W-80 Compressor



NOMENCLATURE		
Symbol	Control Setting	
	Primary	Secondary
W /W	Warmest	Warmest
W /C	Warmest	Coldest
C /W	Coldest	Warmest
C /C	Coldest	Coldest

Primary - Evaporator Cold Control
Secondary - Air Flow Damper

Figure 2-4

TABLE A

Ref. Fig. No.	Compressor Model	Compressor EER	Cabinet Model	Av. Energy Usage KWH/24 Hr.	Av. Energy Usage KWH/Yr.
(a) 2-3	T52	3.47	RT18 orig.	4.30	1569.5
(b) 2-2	T80	3.85	RT18 imp.	3.51	1294
(c) 2-4	W80	4.85	RT18 orig.	3.33	1215.5
(d) 2-1	W80	5.0	RT18 imp.	2.89	1054.8

The gain in (c) vs. (a) of 354. KWH per year represents the improvement due to the prototype compressor alone, compared to the state of the art at the start of the project. For estimating market potential, the gain in (d) vs. (b) is more realistic since it reflects a comparison of the high efficiency model with current CPC production. This average gain is 226.4 KWH per yr.

At the national average electric rate of 4.97¢ per KWH, the average annual energy cost saving in comparing (d) vs. (b) is $11.25. The estimated factory cost increase is $2.75. Assuming that the market price is 2.5 times the factory cost, the increase in market price will be $6.88. For these estimated figures the pay-back period will be approximately 7 1/3 months. The actual pay-back period will vary from place to place depending upon the local electric rate. In any case, based upon the findings of the market survey, the pay-back period should be attractive to the large majority of well informed customers.

2.2 Prototype Development

The prototype development method involved an experimental evaluation of the numerous design changes that were expected to improve efficiency. These changes were applicable in such a great variety of combinations that it was impractical to test 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 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 easily adapted to the compressor design configuration existing at the inception of this project. Fig. 2-5 shows a cutaway view of the efficient compressor. Individual components or complete new subassemblies were easily substituted, using the production compressor as a test vehicle. Similarly, various experimental motors were assembled and tested to determine the most efficient alternative. The bulk of the testing was carried out on a calorimeter at standard conditions.

For refrigerator-freezer compressors, the standard rating point conditions are as follows:

Evaporator temp.	$-10^{\circ}F$
Condenser temp.	$130^{\circ}F$
Return gas temp.	$90^{\circ}F$
Liquid to exp. valve	$90^{0}F$

2.2.1 Motor Performance Improvements

The simplest and most direct approach to improved 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 most often been used in the past, because of its low cost. This motor uses a relay to energize the start winding during start-up. After the motor has accelerated to operational speed, the relay drops out, switching out the start winding.

Two or three phase motors are more efficient than single phase, but are not suitable for motor compressors because the normal

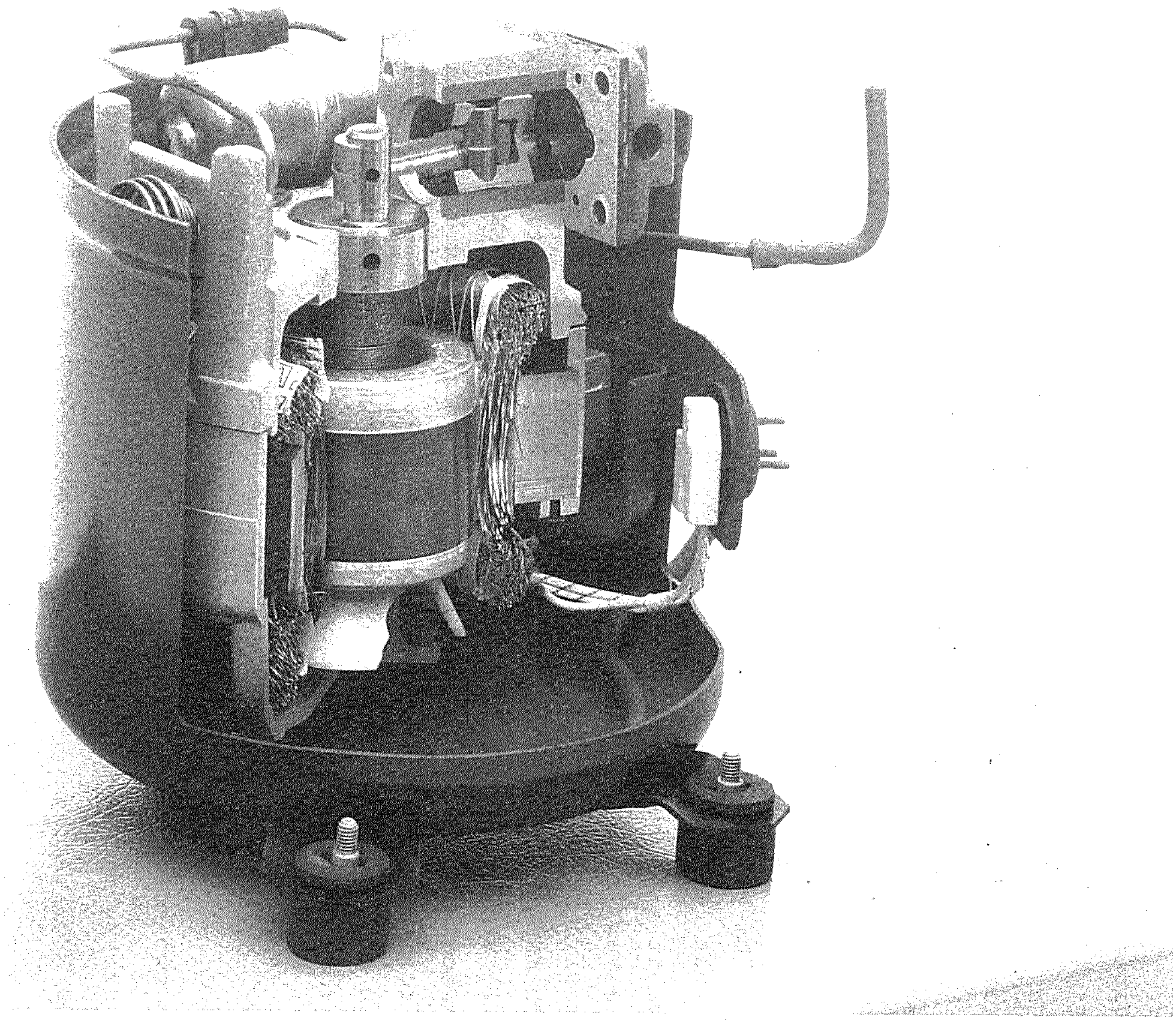


Figure 2-5 W-80 Improved Energy Efficient Compressor

residential power supply is single phase only. Single phase efficiency can be improved by using a permanent split capacitor (PSC) motor. In this arrangement, a capacitor is connected in series with the start winding to induce a phase displacement so as to partially simulate two phase operation. The 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 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, fabricated, and when dynomometer tested, produced load point efficiencies of 82%. When assembled to compressors which had been otherwise modified to improve efficiency, coefficients of performance on the order of 4.7 were achieved.

EER ?

The two motors discussed thus far are both two-pole, and at 60 Hertz, operate slightly below synchronous speed at about 3500 RPM. At one time 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 practical four-pole motor is somewhat less than in a comparable two-pole motor. Calorimeter tests have indicated, however, that the slightly lower efficiency of the four-pole motor is offset by the improved performance of the compressor at four-pole speed. These experimental results have been reinforced by several recently presented theoretical studies. 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 small percentage of the total cycle. Finally, 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 the RSIR type and had a load point efficiency of only 64.5%. Even though flow losses were reduced, the resulting compressor EER 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 and improved compressor performance resulting from slow speed operation were desired. These factors led to the design and development of a PSC four-pole motor which runs at 75.3% load point efficiency.

2.2.2. Volumetric Efficiency Improvements

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, which reduced available re-expansion volume, were tested. Calorimeter test results indicated an increase of 20 BTU/HR was achieved with this change on an 800 BTU/HR compressor.

At standard calorimeter rating point conditions the temperature of the suction gas returning to the compressor is 90°F. 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 finally, 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 a reduction in superheat.
The only effective method which could be found involved the use of a redesigned suction muffler of a thermoplastic polyester material. The muffler inlet was configured with a wide bell-mouth and positioned to allow suction gas to travel directly from the suction tube at the shell into the muffler. The

amount of flow restriction was limited to the point at which sound levels became excessive.

A reduction in flow loss was achieved by increasing the diameter of the discharge tube which runs from the discharge muffler to the shell. Testing showed that a reduction of one or two 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. Because the discharge valve is on top of the valve plate, the discharge port contributes to the re-expansion 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.

2.2.3 Mechanical Efficiency Improvements

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

A number of compressors utilizing needle bearings were constructed. 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 compressor samples tested. In addition, life testing demonstrated that the needle bearings were lacking in durability. Because of the extensive engineering time which would be needed to perfect the needle bearing concept, no further effort was expended in this area during the program.

2.3 Prototype Sample Features

The prototype component parts are shown on Fig. 2-6. The prototypes were given the model designation "W". Although most of the development work was carried on with samples of 800 BTU per hour, nominal capacity samples of other ratings were also built and tested. The result was a proposed list of models as follows:

Model No.	Nominal Capacity BTU Per Hour
W-60	600
W-80	800
W-100	1000
W-120	1200

2.4 Prototype Test Results

When calorimeter tested at the standard calorimeter rating point, the W compressors produced the test data shown in Table B. There were a total of 17 compressors tested in this group. Motor efficiencies at this load point are also given for the various compressors, at a winding temperature of 75°C.

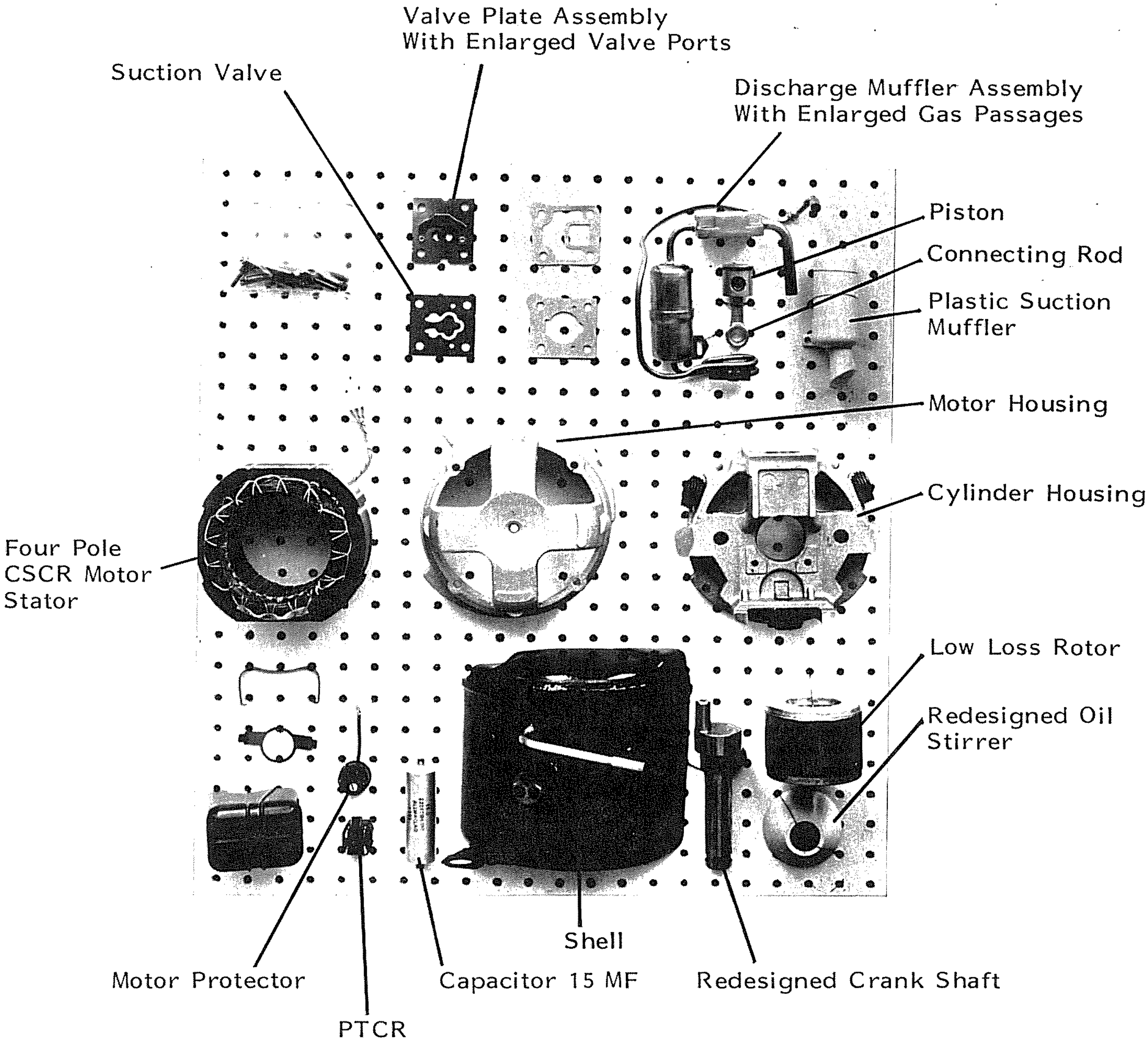


Figure 2-6 Component Parts - Energy Efficient Compressor

TABLE B

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%

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 in the prototypes have the minimum required breakdown torque. Although two of the compressors slightly exceeded the 98 voltage pullout criteria, they did 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^{O}F. 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 C shows pullout voltage and starting voltage locked rotor of the efficient compressors.

TABLE C

PULLOUT AND STARTING VOLTAGES

COMPRESSOR	PULLOUT VOLTAGE	STARTING VOLTAGE
W-60	96.7	77
W-80	99.1	70
W-100	101.1	79
W-120	95.5	80

In using a PTCR starting device, sufficient offtime, 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^{O}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^{O}F at 75 volts locked rotor.

Sound power level measurements were made on all four (600, 800, 1000, 1200 BTU/Hr) 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.

2.4.1 Prototype Life Tests

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 and four were run on the cycling 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.

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. 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.

3.0 ROOM AIR CONDITIONER EXPERIMENTAL DEVELOPMENT

Improved efficiency of the RAC compressor was obtained by implementing a series of evolutionary changes to an existing design.

3.1 The Improvement Achieved

The room air conditioner compressor model chosen for improvement (T66) had the following principal performance figures:

Capacity 6000 BTU/HR
Power Consumption - 750 Watts
EER - 8.0 BTU/WHR

Based on the results of the numerous tests and subsequent evaluation of proposed changes, the design features which contributed to increased efficiency and were practical for mass production were singled out. Prototype compressors incorporating these features were constructed and tested. Typical principal performance figures for these prototypes are as follows:

Capacity 5865 BTU/HR
Power Consumption - 617 Watts
EER - 9.5 BTU/WHR

Comparing the calorimeter test results at standard rating point conditions with the base line model resulted in an EER improvement of 19%.

3.2 Prototype Development

The calorimeter test was the principal test used for the evaluation of RAC motor-compressor performance. The RAC compressor models essentially employ the same mechanical component parts as the refrigerator compressors. The major difference is that RAC compressors require a stronger driving motor because of the greater power requirement brought about by the use of refrigerant 22 and the typically higher evaporator temperatures of an RAC system. Additionally, the lower specific volume of R-22 and greater latent heat of evaporation result in a much greater refrigerating capacity per unit of time than R-12, for a given compressor displacement.

The factors which contribute to the RAC compressor volumetric and mechanical efficiencies are essentially the same as those which have been previously discussed in connection with the refrigerator compressor. RAC compressor motors have historically been of the PSC type and therefore operate at fairly high efficiency levels. Fig. 2-5, the refrigerator compressor cutaway, can be referred to regarding RAC compressor design geometry. The differences between the RAC and the refrigerator prototypes cannot be readily distinguished in Fig. 2-5, but they are discussed in the following sections.

3.2.1 Motor Performance Improvements

RAC compressors operate at a compression ratio of 3.44:1 while the refrigerators operate at 10.5:1; both at rating point conditions. Because of this reexpansion, volume reductions in RAC compressors will have little effect, hence motor improvements constitute a large proportion of potential efficiency improvements in this application.

Since the RAC motor is already of the PSC type, the large improvement in performance obtained by switching from the less efficient RSIR motor is not available. The application of a four-pole motor to the RAC compressor was not a practical alternative. Computer design studies showed that the lamination stack height would have to be significantly longer than that which the longest motor housing has been designed to suit. This would make the compressor too high for many of its present system applications. Improvements can be made, however, by reducing copper losses, reducing rotor resistance, or by optimizing the match of the motor to compressor load requirements. The results of testing improved motor samples are reported in the following sections.

The first group of motor samples were built with a 2.250" lamination stack height. The normal stack height for the CPC production RAC compressor is 2.0". Special crankshafts, motor housings and shells were therefore required to test these motor samples. The characteristics of these motors, compared to the standard RAC motor are shown below:

	Full Load Efficiency	Locked Rotor Torque
Standard T-66 (2.00" Stack)	78.2%	5.12 Oz. Ft.
High Efficiency (2.250" Stack)	81.4%	3.3 Oz. Ft.

Calorimeter test results are compared below to the values for the standard production RAC compressor of the same displacement.

Compressor	BTU/Hr.	Watts	BTU/WH12
Std. T-66	6000	750	8.0
2.50" Stack Motor	5949	713	8.34

Continued testing involving running the above motors on compressors with reduced displacement, was accomplished by substituting various crankshafts with shorter throws. An analysis of the data from this test series led to the design of a new motor with a 2" stack length and slightly higher torque. The implication of previous test results was that a .750 in^3 displacement compressor would be the most compatible for use with these motors. In addition, the test compressor was built up with a special discharge muffler and discharge tube with enlarged flow passages and a valve plate with enlarged ports. The principal compressor performance figures for this configuration are 5492 BTU/Hr. and 615 watts, resulting in an EER of 8.93 BTU/WH. With the completion of this test series and an analysis of all the data, it was determined that this second generation motor exhibited optimal performance characteristics and that further experimentation should be concentrated on other compressor design features.

3.2.2 Miscellaneous Efficiency Improvements

A plug type piston was substituted for the ring type in a number of compressor samples which were subsequently calorimeter tested. There was no clear performance advantage exhibited by either piston type. Other changes which might reduce clearance volume such as a further limitation on head clearance and use of special discharge valves which fill the discharge ports have been tried previously and found to be impractical.

Since refrigerant mass flow in an RAC compressor is much greater than that in a refrigerator/freezer compressor, it might be expected that the potential for efficiency improvements through reduction of flow losses would be greater. The amount of restriction which can be removed from the flow path is, however, limited by compressor geometry and by noise level. The major sources of gas flow restriction are the suction and discharge muffler baffles and tubes.

As the first step in reducing flow losses, an injection molded plastic suction muffler was substituted in place of the normally used steel assembly. The plastic suction muffler has the added advantage of having a lower coefficient of heat transfer than the steel muffler and allows less heat to be absorbed by the returning gas for the same muffler configuration. After numerous experiments, a plastic suction muffler was developed which produced significant performance improvement. When this feature was combined with those previously listed, the following results were obtained:

BTU per Hr.	5814
Watts	626
EER, BTU per W. Hr.	9.28

All of the features included in the above samples were adopted for the prototype design.

An attempt was made to limit heat transfer from the hot compressed gas into the low pressure refrigerant within the compressor shell. As has been previously discussed, raising the temperature of the incoming suction gas and the motor winding both decrease efficiency. It is preferable, if possible, to reject this heat to the atmosphere outside the compressor shell. To this end, the discharge muffler was removed from the compressor and placed in the discharge line external to the shell. The gap thus caused was filled by extending the discharge loop to the cylinder head. The entire loop within the shell was then insulated with pieces of neoprene to reduce heat transfer. Calorimeter test results show that no appreciable change in efficiency occurred when the discharge loop was insulated.

Final tests on prototype samples embodying the improvements selected gave the following test results.

BTU per Hr.	5865
Watts	617
EER, in BTU per W. Hr.	9.5

No tests have been made on high efficiency prototype samples in complete room air conditioners. Since the proposed changes should have no effect on compressor reliability, no life tests have been conducted.

4.0 MARKET STUDY

The purpose of the market study was to investigate the consumer thought processes in deciding what kind of major appliances to buy. More specifically, the objective was to evaluate the consumer's reaction to models with lower operating costs but higher initial prices, and to develop a method for estimating the potential market for high efficiency models.

4.1 Marketing Experiments

Two experiments, referred to as the Field Experiment and the Lab Experiment, were conducted to collect survey data.

4.1.1 The Field Experiment

The field experiment consisted of a survey of 337 people in four different cities. San Francisco, California; Athens, Tennessee; Edison, New Jersey and Burlington, Massachusetts. Each subject was asked to indicate his or her preference rating for a set of 20 hypothetical models of refrigerators or air conditioners. The appliances varied in price, annual operating costs, and in two other attributes (for refrigerators the other features were the presence or absence of an icemaker or glass shelves, and for air conditioners they were the length of warranty and noise characteristics).

The data obtained was studied using conjoint analysis to arrive at an estimate of the weight the individual placed on each feature, and to calculate a payback period for each person; the payback period being the length of time in which the added initial cost would be recovered in savings in energy cost.

4.1.2 The Lab Experiment

The primary goal of the lab experiment was to observe the reactions of customers in a situation closely resembling an actual shopping environment. Specific purposes were to determine the effects of three marketing variables (government energy labels, energy promotional material, and efficiency sales push) on the customers' choices and to collect more data on the payback period concept and other influences on their decisions.

The experiment was a simulated shopping task in which 123 people from the Pittsburgh area were asked to make a purchasing decision in a controlled showroom setting. The consumers were divided in groups, of which some received promotional information on energy and some on other product attributes. In the sales room, they were assisted by trained sales people who concentrated on "selling energy saving to half of the subjects.

4.1.3 Conclusions

The general conclusions derived from the experiments were as follows:

a) The promotional approach used by the sales person is the most important influence on the customer's choice.

b) Government energy labels alone are not effective in increasing the demand for efficient appliances.

c) The concept of a pay-back period is initially unfamiliar to many customers, but when it is explained to them by the salesman they are able to use it making their decisions.

d) Many customers approach an appliance purchase with a preconceived price ceiling, and compensate for buying the efficiency feature by omitting some other feature.

e) Applying the pay-back concept to the estimated energy savings and product cost increase for the prototype design is a valid method of forecasting market potential.

f) The apparent market potential for various pay-back periods is shown on the following table.

TABLE D

APPLIANCE'S PAY-BACK PERIOD*	MARKET POTENTIAL	
	REFRIGERATORS	AIR CONDITIONERS
Over 10 years	43.2%	38.8%
Over 6 years	55.8	48.5
Over 5 years	61.3	53.2
Over 4 years	66.1	58.6
Over 3 years	73.5	66.1
Over 2 years	80.9	72.2
Over 1 year	92.3	83.8

*This refers to the number of years of cost savings to recover the initial price increase.

5.0 PHASE II FIELD DEMONSTRATION PROGRAM

A program for field demonstration is recommended as the next step leading toward ultimate commercialization of the high efficiency compressor for refrigerator-freezer applications. The purpose of Phase II is to verify the performance and reliability of the design, and to collect additional information on customer reactions. The RAC model is not included in the present Phase II program because the improvements in the RAC design do not affect reliability. The principal steps in the Phase II program are described below.

5.1 Manufacture Pilot Lot

The plan calls for manufacture of 1502 compressors. These are to be produced insofar as possible with regular plant facilities. Of the 1502 compressors, 1320 will be installed in refrigerators, a quantity of 50 will be subjected to in-plant life testing, and the remainder will be held in reserve.

5.1.2 Distribution of Pilot Lot

The plan calls for the installation and reliability monitoring of 500 refrigerators in U. S. government housing projects. The remaining 800 will be sold through regular White-Westinghouse distribution channels. Another lot of 20 will be installed in the homes of CPC employees for close observation.

5.1.3 Monitoring of Pilot Lot

Arrangements will be made to obtain regular reports on the performance of the samples, on a monthly basis up to six (6) months and at longer intervals thereafter.

APPENDIX "A"

SCOPE OF WORK

Phase I. Prototype Development and Testing

Task I.1

Submit a detailed project plan for review and approval by the ORNL Technical Manager (TM). This plan shall indicate, in more detail than the proposal program plan, final allocation of financial and personnel resources, timing of principal events that are to occur during execution of the project, decision points and milestones, technical approach, and other items of direct relevance to timely and successful accomplishment of the project objectives.

Task I.2

Perform the studies necessary to determine the potential markets for highly energy efficient motor-compressor units.

Task I.3

Using the approach presented in the Seller's proposal, specify the compressor and perform the work necessary to develop, fabricate, and test prototype samples. Engineering evaluations should be made of the trade-offs between performance of the unit and operating factors such as size and noise output of the unit, the reliability and cost-effectiveness of the units, and modifications required to adapt the equipment to the potential market. Testing should be performed under conditions which are realistic to the chosen application in its target market and which are compatible with the NBS-DOE-FTC* testing and labeling requirements for the chosen application.

In performing Task I.3, the Subcontractor shall, as a minimum, include the following Subtasks.

* National Bureau of Standards - Department of Energy - Federal Trade Commission

I.3.1 Perform the work necessary to maximize motor performance which includes analytical and experimental investigations of permanent split capacitor, capacitor start, capacitor run, and four pole motors, and their relations to efficiency and load.

I.3.2 Perform the development work necessary to determine the motor-compressor improvements by increasing volumetric efficiency. These include mechanical changes to reduce cylinder clearance volume, redesign of discharge gas passages, and reduction of clearance volume relating to suction valve such as eliminating the trepan on the suction parts.

I.3.3 Perform the development work necessary to determine the motor-compressor performance improvements related to flow loss reductions and thermal improvements. These include revised cylinder heads, suction mufflers, discharge mufflers, and valve ports.

I.3.4 Perform the development work necessary to determine the motor compressor performance improvements related to mechanical changes which include reduction of bearing friction and rapid closing discharge valves.

I.3.5 Perform the development work necessary to determine the motor-compressor performance improvements which may result from auxiliary cooling, alternate refrigerants, and increased mass moment of inertia to reduce cyclic speed variation.

I.3.6 Prepare drawings, construct and test prototype compressors which incorporate design improvements. The testing includes performance, life, and system tests.

Task I.4

Submit a detailed Phase II project plan for field demonstration units to be tested and evaluated. The plan for demonstration should be adequate to obtain credible information on energy consumption and efficiency, reliability, performance, safety and cost. Energy efficiency and cost information should be consistent with NBS-DOE-FTC labeling and efficiency rating methods.

Task I.5

Prepare a final report which shall include (a) a summary (executive-type) report covering all aspects of the Phase I work, reflecting resolution of comments from the ORNL TM based on review of draft copy.

Phase II. Demonstration

Task II.1

Based on the results of Phase I, develop final specifications and engineering design of the highly energy motor-compressor units for the capacities and appliance types agreed upon. Establish the manufacturing facility necessary to produce the demonstration units.

Task II.2

Manufacture motor-compressor units, install in application models, and conduct and evaluate a 6-months field demonstration in accordance with the approved project plan (Task I.4). Refine the Task I.2 market analysis.

Task II.3

Prepare a final report which shall include (a) a summary (executive-type) report covering all aspects of the Phase II work, reflecting resolution of comments from the ORNL TM based on review of draft copy.

Task II.4

Conduct and evaluate long-term performance of the demonstration units at a reduced, but adequate, level of surveillance to document the important characteristics, such as annual savings, problems, reliability, maintenance, and user acceptance of appliance. The duration of this task is for 30 months beyond the initial 6-month field demonstration. Prepare annual reports of the Task II.4 evaluations. Each annual report shall report cumulative results to the current time.