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ENERGY CONSERVATION

RESEARCH AND DEVELOPMENT OF ENERGY-EFFICIENT APPLIANCE MOTOR-COMPRESSORS

FINAL REPORT

VOLUME III—DEVELOPMENT AND FIELD TEST PLAN

Prepared by
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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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OAK RIDGE NATIONAL LABORATORY
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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 per WHR. 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 per WHR, 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 appliances operating costs and provided with a stategy 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.

PREFACE

This report outlines the accomplishments of a compressor development program carried out by Columbus Products Company, a division of The White Consolidated Industries, and Oak Ridge National Laboratory 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.

The complete project report series consists of three volumes. Volume I, the Executive Summary Report, presents an overview of the accomplishments and findings of the entire program. The Potential Market Evaluation is shown in Volume II. This volume contains a comprehensive discussion of the technical accomplishments of the project and the recommendation for the field test plan.

ACKNOWLEDGMENTS

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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. Mr. 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.

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LIST OF ABBREVIATIONS

APC — Athens Products Company (WCI Motor Plant)

CMU — Carnegie-Mellon University

CPC — Columbus Products Company (WCI Refrigerator/Compressor/Dishwasher Plant)

BDC — Bottom dead center rotational position of crank pin

EER — Energy efficiency ratio - BTU/WHR

HBP — High back pressure compressor (used in water cooler, dehumidifier, and vending machines)

LBP — Low back pressure refrigerator compressors

LR — Lock Rotor - a condition in which the rotor is locked to prevent starting.

LRA — Lock Rotor Amps

LRV — Lock Rotor Volts

M — Model designation of compressor currently in production at Americold Compressor Corp.

PSC — Permanent Split Capacitor - a type of motor characterized by a capacitor in series with the phase windings while running.

PTC or PTCR — Positive Temperature Coefficient (resistor) - a solid state starting device used to replace a current relay in RSIR applications or to shunt the capacitor in PSC designs during start. Resistance of this device increases with temperature.

RAC — Room Air Conditioner

RSIR — Resistance Start Inducation Run - a type of motor which operates on its main winding only, which is assisted by an auxiliary winding during start conditions. No capacitors are used with this type motor.

SB — Single bearing compressor design employing a bolted on hub mounted to a cylinder housing. Number following identifies displacement (i.e. SB52).

TDC — Top dead center rotational position of crank pin

W — Model designation of four-pole prototypes

WCI — White Consolidated Industries

1.0 INTRODUCTION

Under the terms of Contract No. 7229 between the Columbus Products Company (CPC), a division of The White Consolidated Industries, and The Union Carbide Corporation - Nuclear Division (UCC-ND) acting on behalf of the Department of Energy (DOE), CPC has been engaged in the development of high-efficiency compressors for refrigerator, freezer, and room air conditioner applications. The scope of work for the entire project may be found in Appendix A. This is substantially the same as the scope of work appended to the original contract agreement, except that provisions for progress reports deliverable during the course of development have been omitted. The final report on Phase I of the project is divided into several volumes. Volume I is an executive summary report; Volume II is a detailed report on the potential market analysis. This volume reports on the technical accomplishments of the development program and recommends a plan for field test evaluation of the efficient compressors, which is referred to as Phase II.

1.1 Project Objectives

The basic objective of the project was to investigate the technical and commercial feasibility of reducing energy consumption by the development and utilization of high-efficiency compressors for refrigerators, freezers, and room air conditioners.

1.2 Technical Portion of Project

By means of a program of theoretical analysis, development, and testing of samples, it was found that significant improvements could be made in the energy efficiency ratio (EER) of hermetic motor-compressor assemblies. The high efficiency designs resulting from the development program are believed to be suitable for quantity production without excessive facilities cost, to have acceptable levels of performance and reliability, and to be producible at costs which will make them commercially attractive.

The steps involved in the development of the improved compressor design are described in detail in Sections 5.0 through 7.2 of this report. The major purpose of Phase II of the project is to verify the reliability of the high-efficiency designs by means of a field demonstration program.

1.3 Marketing Survey and Analysis

The marketing survey was undertaken to collect data on the decision making processes of prospective appliance purchasers and to apply this information in judging customer acceptance of high-efficiency appliances.

A complete report on the market survey, comprises Volume II of the Phase I project report. A synopsis of the results, as they apply to the commercial acceptability of the high-efficiency compressor design, appears in Sec. 2 below.

1.4 Summary of Results

1.4.1 Compressor Development Results

	Refrig./Fr. Model	RAC Model
BTU/WH, Existing Design	3.47	8.00
BTU/WH, High Eff. Design	5.00	9.50
% Increase in BTU/WH	44.1	18.7
Estimated Increase in Factory Cost*	$ 2.75	$ 1.28

The performance improvement found in tests on complete refrigerators using high-efficiency compressor samples is reported in Sec. 7.3.

*Compared to comparable standard motor compressor produced at the time of this writing.

1.4.2 Market Survey and Analysis Results

The survey and analysis indicates that there is a substantial market for high-efficiency appliances using compressors having the characteristics tabulated in Sec. 1.4.1 if an informative sales approach is used, and that the high-efficiency segment of the market should be commercially attractive to the manufacturers.

1.5 The Compressor Industry

The manufacture of small hermetic compressors is a high volume activity. Production in the U. S. has been about 7,000,000 per year in recent years. Most of this volume has been concentrated in a relatively small number of plants. Two basic types have been produced; rotary and reciprocating. For most applications the two types have been able to compete on an even basis. The designs produced by CPC have all been of the reciprocating type, and the remainder of this report relates only to this type.

1.6 Design Improvement Strategy

The basic strategy chosen at the beginning of the project was to investigate evolutionary improvements which could be made in the existing design, rather than initiating a radical new design. The evolutionary approach has a number of advantages. It facilitates the procurement of test samples. It greatly reduces the time and expense required for the start-up of production in commercial quantities. It reduces the effort required for the verification of product reliability. In addition, there is no prior assurance that a completely new design would be superior.

The selection of features to be investigated was based partly on past experience and partly on the thermodynamic theory of gas compression. The general theory of compressor design may be found in various text books and is not addressed in detail in this report. A very brief discussion of the theory as it applies to improvements in efficiency is however, included in Appendices C and D. Since it is seldom possible to predict the effect of compressor design changes, the evaluation of the proposed improvements required an extensive test program. The principal test procedures used are described in Sec. 4. The features which were proposed as efficiency improvements are discussed in detail in Sections 5.0 through 6.2.2.

2.0 RESULTS OF MARKET STUDY

Volume II of this report contains the complete market study report prepared by CMU. A synopsis of the report and certain conclusions drawn from it are presented here as an aid in assessing the potential market for high efficiency appliances.

2.1 Market Study Procedure

The experimental part of the market study was divided into two parts referred to as the Field Experiment and the Lab Experiment. In the Field Experiment 337 people in four separate geographic locations were asked to give their preference ratings for a set of 20 hypothetical refrigerator and room air conditioner models with a varied combination of features, one of which was high efficiency. The Lab Experiment was implemented under simulated sales room conditions and consisted of asking a sample of 123 consumers to make a selection among various refrigerators on display. Prior to making the selection, the potential customers were divided into groups and were supplied with different types of promotional information so that the effect of various sales approaches could be evaluated.

2.2 Market Study Conclusions

The single most important influence on customer decision making relating to the selection of energy efficient appliances is the presentation made by the sales person. The concept of pay-back period (that is, the length of time required to recover the higher initial cost of high efficiency models from energy cost savings) as a quantitative factor in decision making, was not originally familiar to the majority of consumers. With adequate explanations, however, most of them were able to grasp and apply the concept.

Information on energy consumption of various models, without further explanations on how to apply the information, was not effective in influencing customers to select high efficiency models. Most customers approached a purchase decision with a budgetary price ceiling in mind, and were therefore inclined to regard high efficiency as a feature to be considered as an alternative to various convenience features. The approximate relationship between high efficiency pay-back period and market potential is given in Table 2-A below.

TABLE 2-A

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.

The general conclusion to be drawn from the survey is that the pay-back concept is a valid quantitative mechanism for estimating customer acceptance of high efficiency appliances. This mechanism will be used to forecast customer acceptance from actual estimates of product cost and energy savings in later sections of the report. It should be emphasized again that this conclusion is largely dependent upon an effective presentation by sales personnel.

An informative presentation by sales personnel is 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.

3.0 COMPRESSOR EFFICIENCY IMPROVEMENT, GENERAL PLAN

As discussed in Sec. 1.6, the strategy chosen for the development of a high efficiency motor-compressor unit was to make evolutionary changes to an existing design. The general guidelines observed in the improvement process are given in the following sections, and the previously existing baseline models are described in Sec. 3.2.

3.1 Design Guidelines

3.1.1 Performance Improvement

Since the main purpose of the project is to develop a line of compressors of improved efficiency, testing effort was concentrated on items which, on the basis of analysis and past experience, appeared most likely to reduce power input. The items selected for use in the prototypes were found by performance tests to produce an improvement.

3.1.2 Estimated Cost

The total increase in cost will result in a market price acceptable to prospective customers, as judged by the market survey reported elsewhere. Manufacturing costs listed are based on CPC Engineering Department estimates

3.1.3 Service Reliability

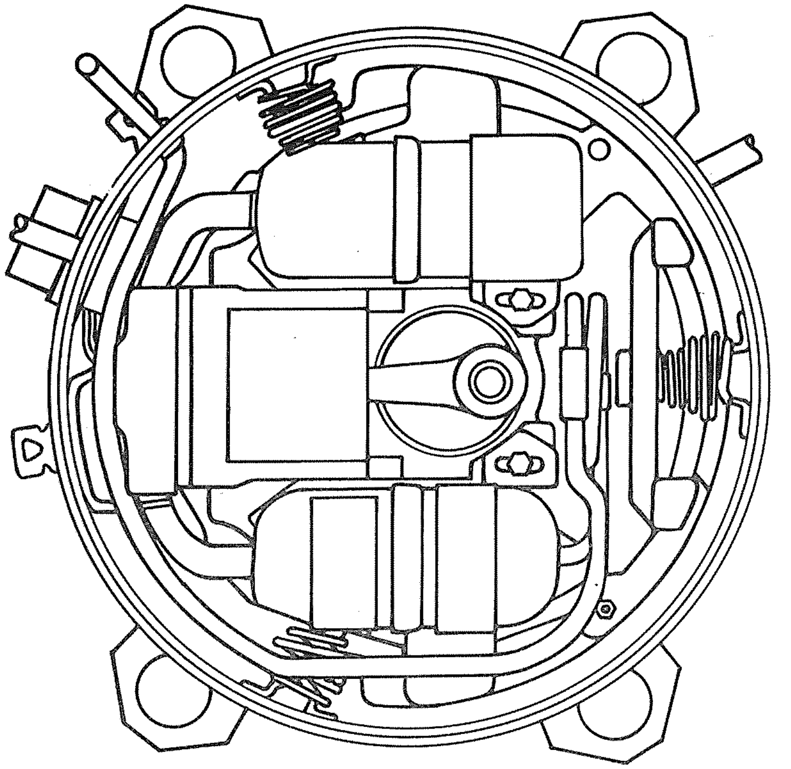
In the reliability area, it has been necessary to rely mostly on engineering judgment, based on the performance of previous designs in the field and on life tests. Recent life tests on the four-pole prototype design have given favorable results, as reported in Sec. 6.3.1. An extended life test program in the field has not as yet been implemented. A few design features which failed in the course of performance testing were, of course, discarded or redesigned. The reliability of the final improved design will be verified later both by more extensive life testing and by field testing. This program is described in Sec. 13.0.

3.1.4 Manufacturing Feasibility

Features which were not readily producible for test sample, prototypes, or the demonstration samples (to be built in Phase II of this project) were excluded. In addition, since a design which could not be made in quantity production would be of very limited utility in reducing energy consumption, future manufacturing feasibility was considered in each case. It is believed that all the features chosen are suitable for manufacture, although some additional development may be needed to adapt them for mass production.

3.2 Existing Compressor Line

The present CPC line of motor-compressor units is designated the T-line. The compressor is of the single cylinder reciprocating type, with a vertical crankshaft. The pump assembly is internally spring mounted inside a welded steel shell. A simplified top view and cross section of a typical model are shown in Fig. 3-1. All present models have a two-pole single phase motor which normally runs at about 3400 rpm. A passage drilled through the crankshaft acts as an oil pump, and delivers oil from the sump in the bottom of the shell to the various points which require lubrication. In current refrigerator-freezer models, a portion of the oil pumped is sprayed on the interior shell walls to assist in heat transfer away from the pump and motor. The motor-compressor assembly is designed to function for at least 10 years without any service or adjustment whatever.



Top View With
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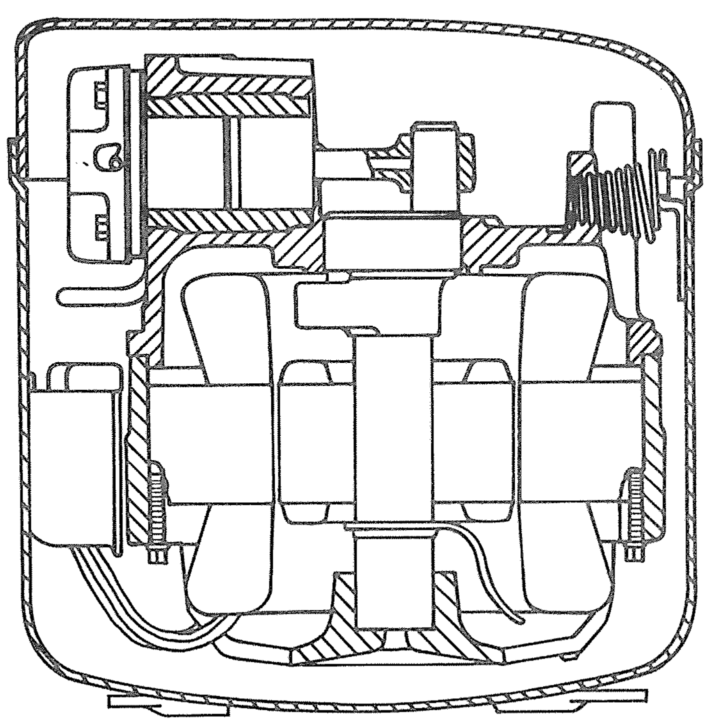


Cross Section

Figure 3-1 Typical "T" Compressor

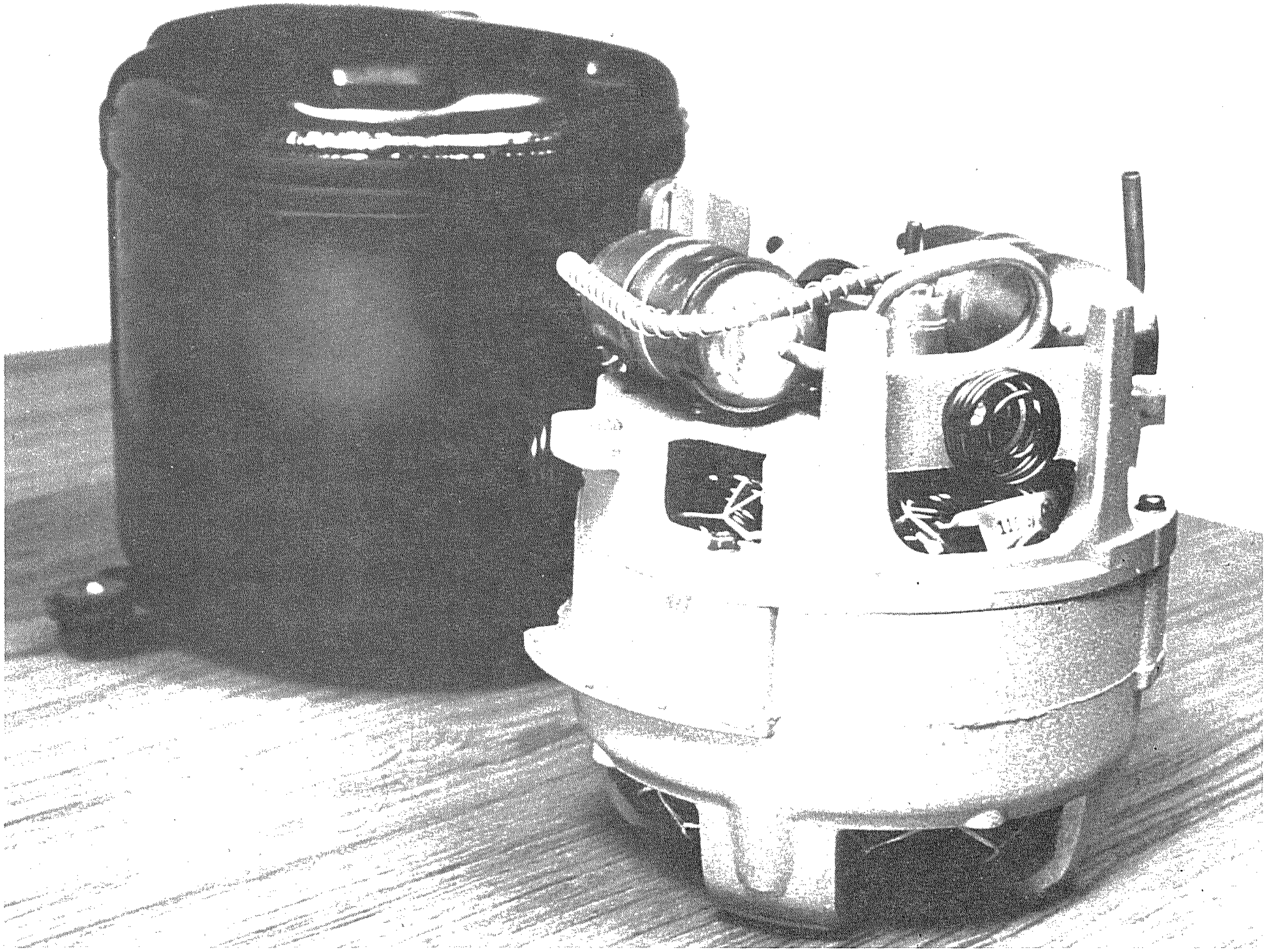


Figure 3-2 Columbus Products Company T 52 Motor Compressor

The T-line models for refrigerator-freezer and RAC applications, respectively, are very similar in mechanical design. In the electrical area, refrigerator-freezer models have been supplied with resistance start induction run (RSIR) motors and a starting relay, while RAC models have had permanent split capacitor (PSC) motors with no relay. Refrigerator-freezer models are designed for operation with refrigerant 12 at low back pressure, while RAC models use refrigerant 22 at high back pressure.

3.3 Baseline Models

3.3.1 Refrigerator-Freezer Baseline

The model refrigerator-freezer chosen for improvement is designated T52. The principal performance figures for this model under standard calorimeter rating point conditions are as follows:

Capacity	- 780 BTU per HR
Power Consumption	- 225 Watts
EER	- 3.47 BTU per WHR

Pictures of the T52 shell and pump assembly may be seen in Fig. 3-2. The improved efficiency compressor developed is shown in the cut-away view in Fig. 3-3.

3.3.2 RAC Baseline

The RAC model used as a baseline for efficiency improvement is designated T66. The typical performance figures for this model at the standard RAC calorimeter rating point are:

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

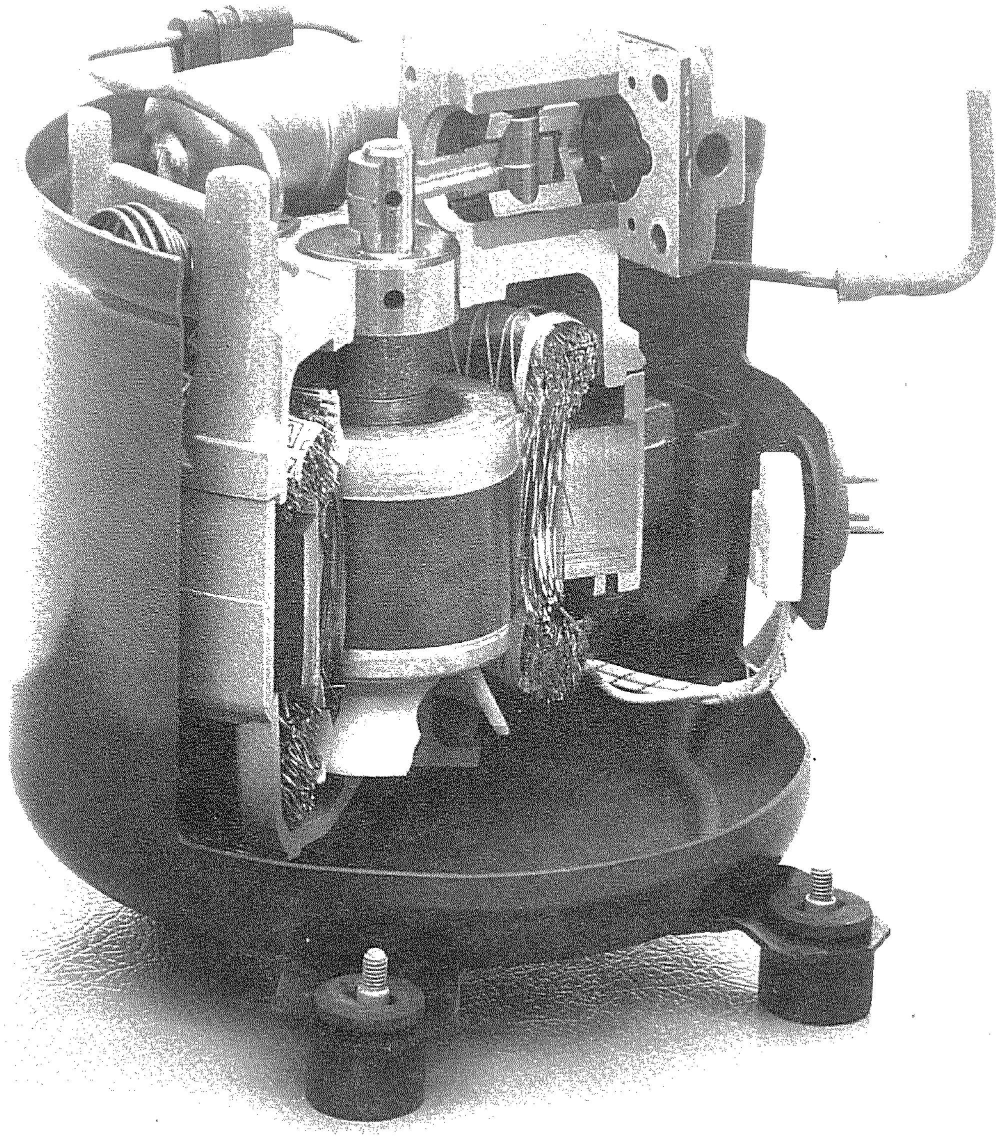


Figure 3-3 W-80 Improved Energy Efficient Compressor

3.4 Sequence of Development Steps

In the actual development program the improvement of the refrigerator-freezer model was undertaken first. The succeeding Sections 4 through 8 relate principally to this model. Corresponding information on the RAC models will be found in Sections 9 through 12.

4.0 TEST PROCEDURES

The various test procedures and apparatus for compressor performance testing are described in the following sections. The compressor test facilities are located in the CPC compressor development laboratory. Systems testing was performed in the environmentally controlled test rooms located at CPC in Columbus, Ohio. All of the motor tests were performed by the motor supplier, Athens Products Company, Athens, Tennessee.

Calorimeter testing is the principal method used to measure compressor efficiency. Other tests include:

- Start Test
- Pullout Test
- Noise Test
- Locked Rotor Test
- Ultimate Trip Test

4.1 Calorimeter Test

The calorimeter test is used by CPC for the measurement of compressor capacity and efficiency. The type of calorimeter used at CPC has a water cooled condenser, an automatic expansion valve, and evaporator enclosed in a secondary refrigerant system with electric heaters. The CPC calorimeter is supplied with pressure gauges, electrical meters, and a resistance bridge for winding temperature reading by thermocouple. Connected with it is a duct in which the compressor is mounted, and in which conditions in an actual installation in an appliance can be simulated. The CPC calorimeter has a supply of regulated 115 volt 60 Hertz power. A functional diagram of a typical calorimeter used by CPC is shown on Fig. 4-1.

Most calorimeter testing is performed under standard rating point conditions which are chosen to represent typical service conditions on an appliance. The actual test conditions vary

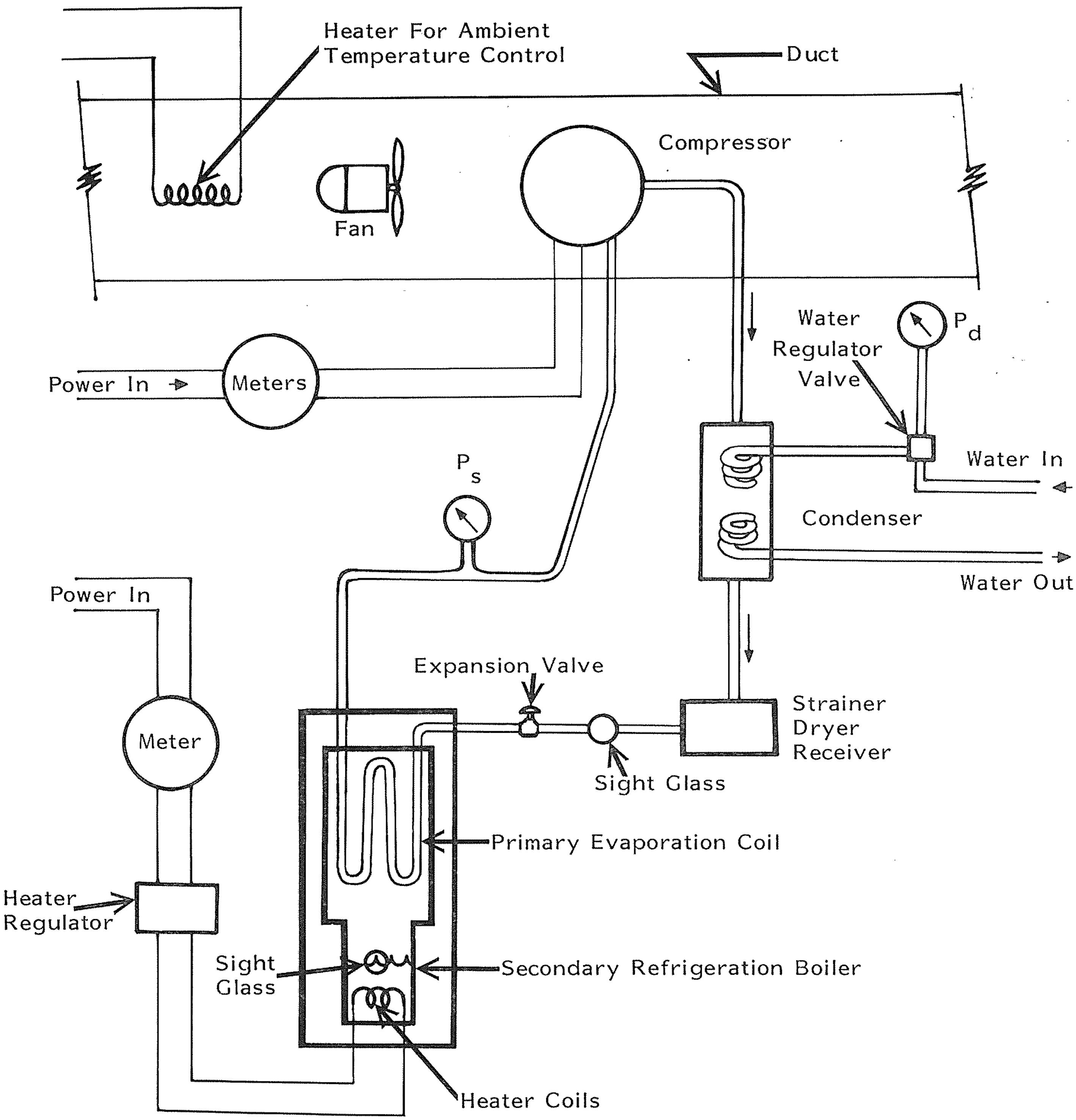


Figure 4-1 Functional Diagram of a Typical Calorimeter

depending on the type of compressor to be tested and its application.* Conditions given in Table 4-A are for compressors using R-12 refrigerant and intended for low back pressure applications. These conditions were utilized during the testing of high efficiency samples for refrigerator-freezer applications.

TABLE 4-A

COMPRESSORS STANDARD CALORIMETER RATING POINT CONDITIONS

Compressor Type Refrigerator/Freezer	Low Back Pressure R-12
Evap. Temp ($^{\circ}$F)	-10
Evap. Press. (psig)	4.5
Cond. Temp. ($^{\circ}$F)	130
Cond. Press. (psig)	181
Return Gas ($^{\circ}$F)	90
Liquid to Exp. ($^{\circ}$F)	90

For a more complete definition of compressor performance than that provided by the standard rating point test, a series of tests may be made at the following evaporator temperatures, in $^{\circ}$F: -20.0, -10.0, 0, + 20.0. Tests may also be run at condensing temperatures of 110.0°F and 150.0°F in addition to the standard 130.0°F level. It is customary to plot the results of tests of this kind as a family of curves, and the procedure is referred to as "curve testing".

During the calorimeter test the ambient temperature around the compressor, which is in the calorimeter duct, is 90.0°F. Forced air flow is directed at the compressor through the duct. The duct area used at CPC for R-12 compressor testing is 144 in^2. Forced air is supplied at a flow rate through the duct of 100 ft^3/min. Thermocouples are supplied and monitored at various locations. Typical thermocouple locations are shown in the following table:

*For a more complete discussion regarding calorimeter testing accuracy see Appendix E, page 172.

TABLE 4-B

TYPICAL THERMOCOUPLE LOCATIONS FOR CALORIMETER TESTING

Suction line 6 in. from compressor
Discharge line 6 in. from compressor
Compressor shell top
Compressor shell under protector
Compressor shell bottom
Condenser inlet
Condenser outlet
Evaporator
Liquid line to expansion valve
Condenser cooling water in
Condenser cooling water out

In operation, cooling water flow to the calorimeter condenser is adjusted to provide the standard discharge or condensing pressure. The expansion valve is set to a standard evaporator or suction pressure, depending on the compressor to be tested, and the heat input to the evaporator is regulated to give standard superheat of the suction gas. The refrigerating effect is then calculated from the measured electric energy supplied to the evaporator during a fixed period of time. Since the observed liquid temperature during the test will not in general equal the standard temperature, the recorded load is adjusted by making a suitable correction based on the enthalpy of the liquid. While the test is in progress, the energy supplied to the compressor is recorded. The EER (energy efficiency ratio) in BTU/WHR is then calculated.

As was noted earlier, calorimeter test conditions differ significantly from the usual conditions in a complete refrigerator system. Calorimeter conditions of operation are stabilized at standard levels, while on a complete system, which runs only part of the time, the conditions vary throughout the running cycle.

4.2 Start Test

This test is made for the purpose of ascertaining the minimum voltage at which the compressors will start and come up to the required speed. The voltage is measured at the compressor or starting relay terminals. The locked rotor voltage, rather than line voltage, is measured since that is the initial state of the compressor. Since the duration of this condition is very short, it is necessary to stall the compressor by some means while adjusting the voltage. This is normally done by the use of a manual valving arrangement to supply R-12 under pressure to the high side of the compressor.

Before starting the actual test procedure, the compressor is charged with the specified quantity of refrigerant oil and the test system is charged with the proper refrigerant. A thermocouple is attached to the bottom of the shell for temperature measurement. The electrical accessories, such as capacitors, overload protectors and start devices are connected into the circuit.

The start test is normally run at a room ambient of 70^{o}F to 75^{o}F. The compressor is started and allowed to circulate the refrigerant through the test system until the shell bottom reached 150^{o}F. After warm up, the compressor is stopped, and the pressure allowed to equalize. The actual test is performed by observing the ability of the compressor to start at successively lower steps in voltage, beginning at a level well above the minimum. The lowest voltage at which the sample will start consistently is reported as the minimum starting voltage. The voltage is measured in each case, not with the sample running, but with it stalled as described above.

4.3 Pullout Test

The pullout test measures the voltage at which a compressor stalls under the conditions described below, and is used to determine if it has sufficient torque for heavy load operation. The pull-out test is usually performed following a calorimeter test since it is performed while the compressor is mounted on a calorimeter. The test condition for a low back pressure, R-12 compressor, is 200 psig discharge pressure with 30 psig suction pressure. Both the return gas temperature and the ambient temperature are set at $90^{\circ}F$.

Testing consisted of dropping the line voltage in successive steps of five volts. After each drop in voltage, the compressor is allowed to stabilize. At stabilized temperature at each step, volts, amps and watts are recorded. The results reported are the lowest voltage at which the compressor runs and the voltage at which pullout occurs.

CPC uses two other variations of this test. The first is designed for greater accuracy. Instead of reducing the voltage in 5 volt steps reductions are limited to 2 volts. Secondly, for a fast test the voltage may be reduced gradually until the compressor stalls. The only time that is allowed is that which is necessary to record the volts amps and watts at 5 volt intervals. Results reported from the quick test are the lowest voltage at which the compressor will run steadily and the voltage at which the compressor pulls out.

4.4 Noise Test

The noise test does not contribute directly to improved efficiency. Any proposed design, in order to have consumer acceptance, must have an acceptable noise level. The noise test must be run because design changes, intended to increase efficiency, usually also change the noise level. Design changes to reduce flow losses are an example. With the compressor running under specified conditions, readings are taken at fixed points with a sound

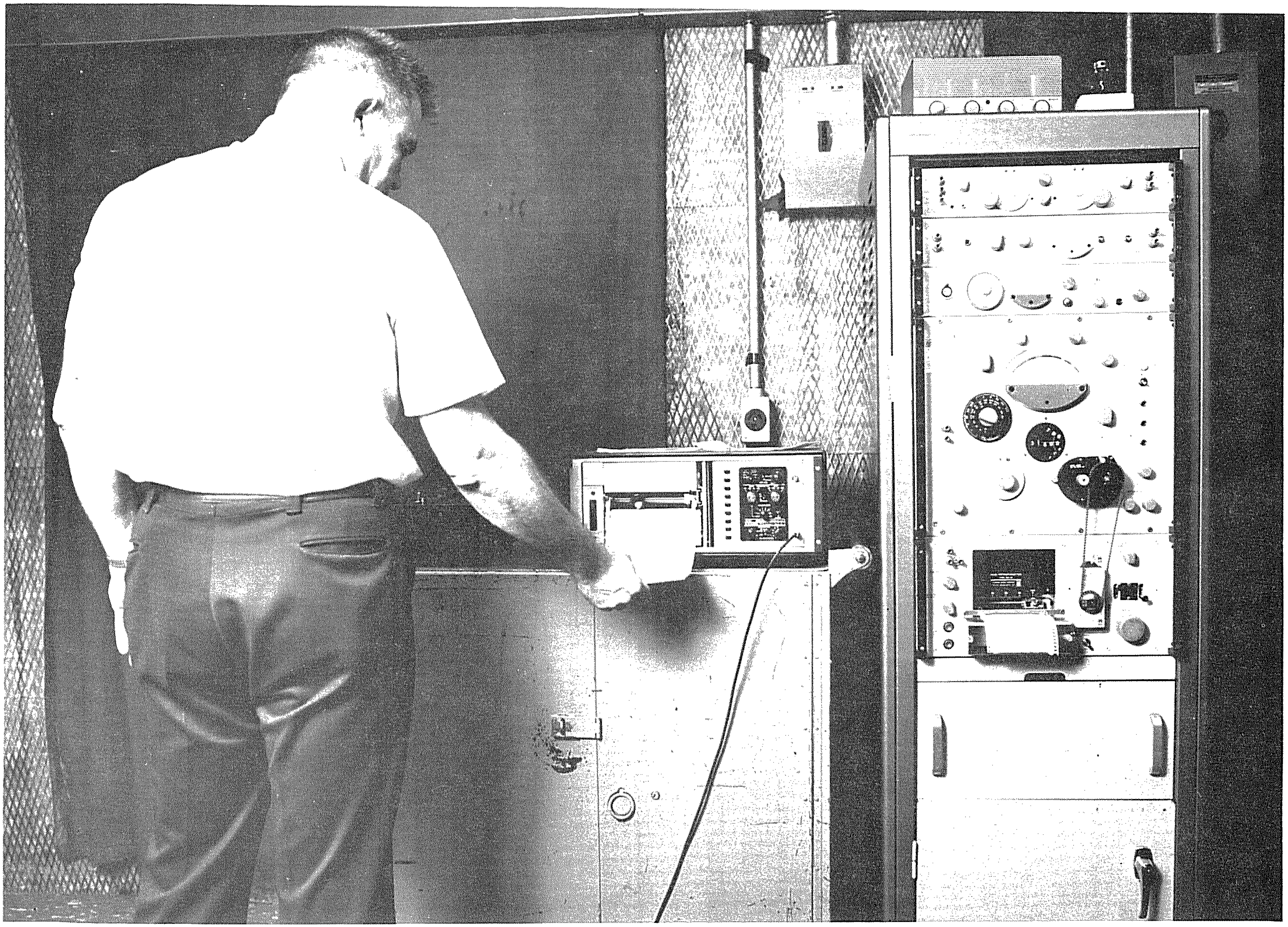


Figure 4-2 Noise Frequency Band Analyzer

level meter. In addition, a noise spectrum, or sound level frequency distribution is recorded. Frequency distribution is analyzed and recorded with the use of 1/10 octave General Radio band analyzer shown on Fig. 4-2.

The CPC test specifications require a test room isolated from outside sources of noise and vibration. The room has non-reflective walls. The CPC test room has a typical background noise level of about 28dB. A refrigerator under test in the CPC test room is shown on Fig. 4-3. A simple calorimeter or water-cooled load stand is used as a refrigeration system when the compressor alone is under test. The calorimeter or load stand usually produces flow noises, therefore it is often isolated from the compressor under test. Compressors under test are mounted on a firm and rigid stand. Rubber mounting grommets, such as those used in refrigerator application, are used to support the compressor on the stand.

Typically, three reading positions are used in the test. Both DbA and DbB readings are taken. One of the microphone positions used at CPC is shown on Fig. 4-4. Readings are taken at compressor startup and at 5, 10, and 30 minutes after starting. Before the final 30 minute reading, the compressor must be stabilized at pressures of 181 psig discharge and 4.5 psig suction. In addition, the return gas temperature must be 90^{o}F. If conditions have not stabilized in 30 minutes, the test must continue running until stabilized, and then final readings taken. The microphone of the sound level meter should be in the same horizontal plane as the end head. CPC utilizes a mastic pad on the end head for normal T-line tests, as well as on production compressors. The mastic pad is also utilized on all the efficient compressor samples.

4.5 Locked Rotor Test

The principle purpose of a locked rotor test is to aid in the selection of a thermal overload protector which will prevent the

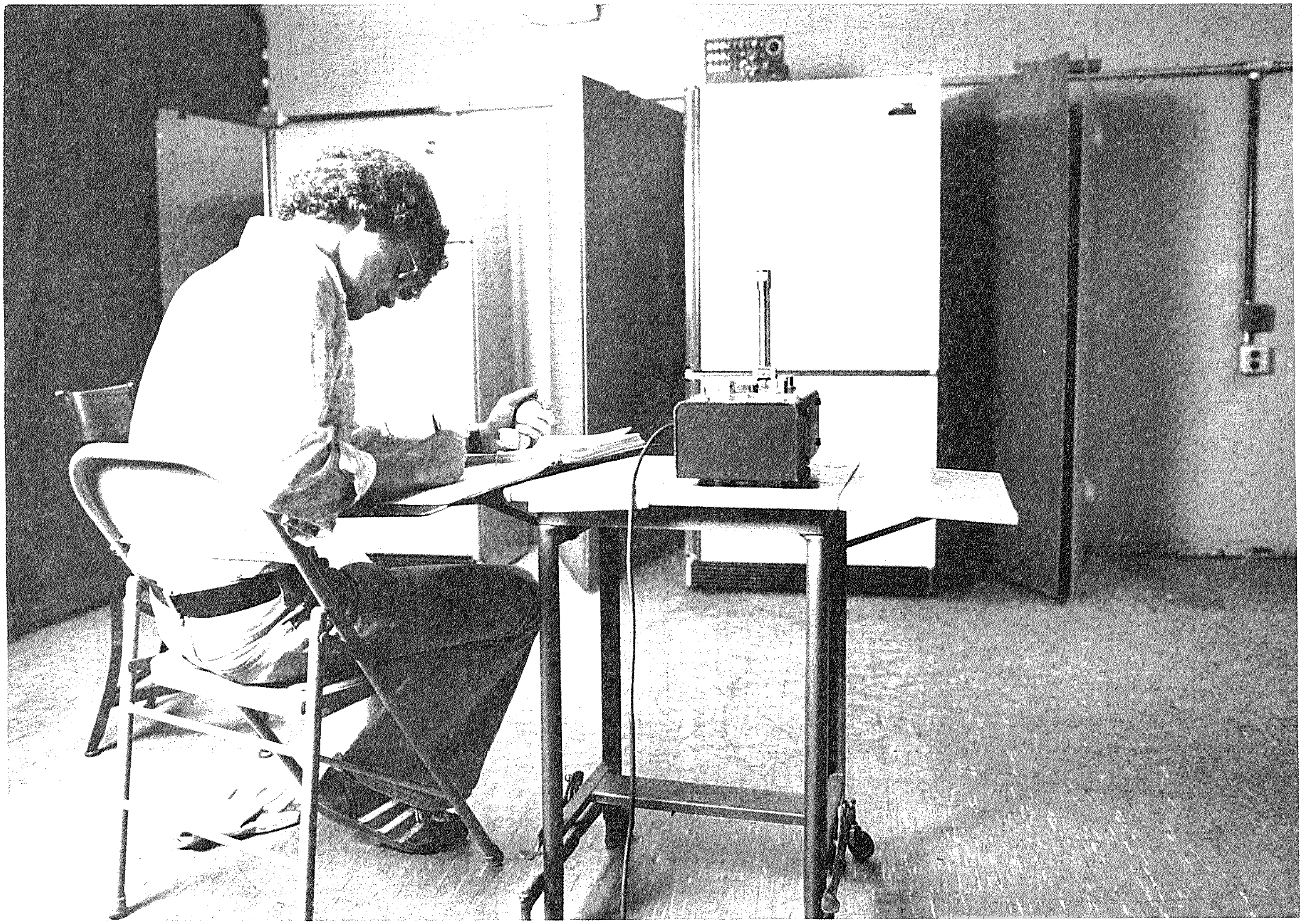


Figure 4-3 Noise Test on Refrigerator

occurrence of excessive temperatures in the motor windings if the compressor fails to start with voltage applied. The most common problem occurs when there is a momentary interruption to the power supply, so that the compressor stops, but cannot restart until pressures have equalized. The equalization process in a refrigerator or freezer often takes 4 or 5 minutes because of the capillary tube restriction between the high and low side. The locked rotor test is not related to compressor efficiency improvement, but it nevertheless must be successfully carried out for the purpose of selecting an overload protector for any proposed design which will be put into service.

Locked rotor tests are performed at 120, 90 and 75 volts, 60 Hz. Testing under these various voltage conditions insures adequate protection under low voltage conditions such as those encountered in a brown out.

Shell temperatures are also monitored during the test. This data is used to insure that the compressor can meet Underwriters Laboratories, Inc. specifications. Underwriters Laboratories specification UL 984 states that the shell of the locked rotor assembly may not exceed 150^{o}C (302^{o}F) when 120 volts, 60 Hz. is applied to the assembly. (The specification only requires voltage regulation of 95% which means that UL may run the locked rotor assembly as low as 114 volts).

Off time of the overload protectors is also monitored. After a protector trip, sufficient time must be allowed to let the motor cool so that ample starting torque will be available during restart conditions. In the case of the PSC motor design using a PTCR starting device, as in the efficient compressor design, sufficient off-time is required to let the PTCR cool down to a low resistance. If the off-time is too short, the compressor will continue to trip the overload protector, since it cannot start without the auxiliary winding.

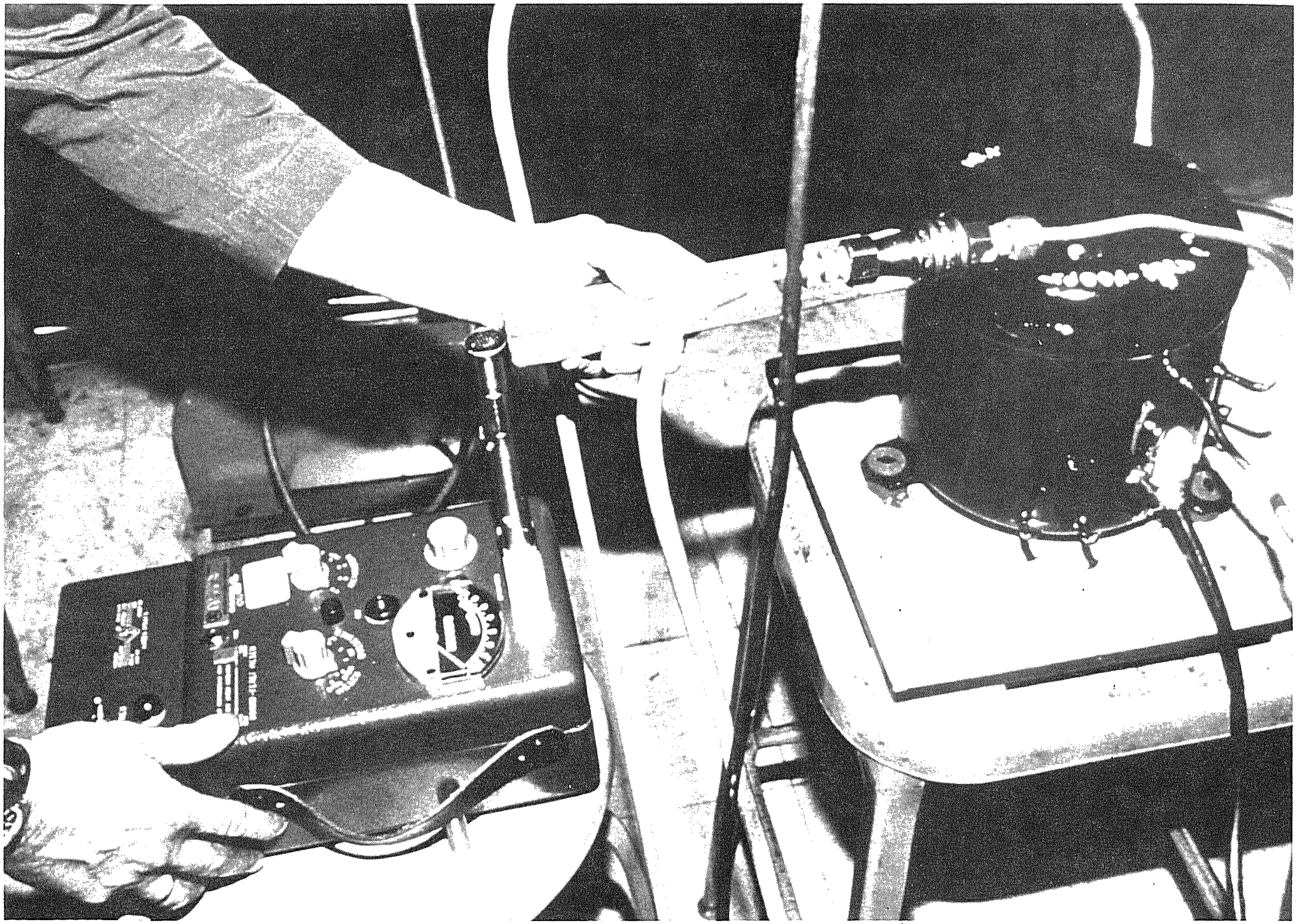


Figure 4-4 Typical Compressor Noise Test Microphone Position

4.6 Ultimate Trip Test

The ultimate trip test is a refrigerator-freezer evaluation procedure. The purpose of the test is to determine if the design of the motor protector is effective in limiting the compressor main windings to an acceptable maximum temperature under conditions of gradually increasing load. This test is the complement of the locked rotor test which verifies motor protecting under stalled or locked conditions. Ultimate trip testing insures that the protector safeguards the motor under running conditions.

Preparation of the cabinet for testing includes installation of proper instrumentation such as thermocouples, gauges, resistance switches and clock. The cabinet is then run for 24 hours to insure its proper operation. Test cabinets are then "soaked" for a minimum of 16 hours prior to the actual test. Temperature of the test room may be 110°F or 120°F. At the beginning of the test, temperature measurements are taken. All temperatures must be within $\pm$ 3°F of the ambient. If temperature readings are not within these limits, additional soaking is required to insure proper temperature stabilization. Suction and discharge pressures are recorded. An automatic ultimate trip tester is then placed in the test cabinet freezer compartment. The cabinet is then run for six hours or until the temperature has stabilized.

After cabinet stabilization the data is read and recorded as "zero time". Readings taken are thermocouple measurements, volts, system amps, watts, compressor amps, suction pressure and discharge pressure. The automatic trip tester is then adjusted and started so that a trip will occur within eight hours. When the trip occurs, the automatic ultimate trip tester will cause a data acquisition readout.

When the approximate trip point of the protector is known, the test is repeated, but more slowly for accurate data points.

During the last hour, the rate of rise of the main winding and motor protector temperatures shall be no greater than $2^{o}F$ per hour for a "trip" to be considered good. In some cases, to obtain good data, it may be necessary to disconnect the ultimate trip tester and increase temperatures by blockage of the compressor and/or condenser cooling air. Any abnormalities during testing are noted and reported. Temperatures, pressures and electrical measurements are then recorded.

4.7 Life Test

Life tests are conducted to evaluate the reliability and durability of complete compressors, subassemblies, or components. Tests are often made to evaluate chemical stability as well as mechanical endurance. The tests are made on "load stands" designed to give accelerated or intensified results. Compressors may be tested in either continuous operation or under cycling conditions.

The load stands consist of the electrical wiring necessary to start and run the compressor and to power the forced air cooling fan. In addition, a clock is used to record total running time. The stands have a refrigerant circuit consisting of a reservoir, expansion valve and interconnecting copper tubing. Gauges to measure suction and discharge pressures are provided. Winding temperature measurements may be made as required during the test by using a portable running bridge.

In application, the amount of refrigerant in the refrigerant system, and the setting of the expansion valve are varied to give the proper test conditions. Suction pressure normally is set to 10 ± 1 psig, while the discharge is set to 385 ± 10 psig. Under normal conditions, shell bottom temperatures of $200^{o}F$ to $225^{o}F$ are expected. While these are not essential test conditions, any significant deviation from normal should be investigated. The test is run continuously, except shut down one night (approximately 16 hours off), per week. Normal duration of the continuous run test is 3,000 hours.

The load stand used for cycling life test is similar to the stand used for the continuous run life test. The primary difference is a timing device with contacts set for 30 seconds on and 30 seconds off. This timer actuates a relay which turns power on and off to the compressor, and a solenoid valve which allows the load test stand to rapidly equalize. After solenoid valve opening, the equalized pressure normally is 16 to 20 psig. At compressor cutoff, the discharge pressure is to be 190 $\pm$ 10 psig, and the suction pressure is 5 $\pm$ 1 psig.

In addition to the continuous run and cycling life test there is a special short duration test called the 100 Hour Test. This test is similar to the continuous run test except that no fan cooling is provided; discharge pressure is 500 psig, suction pressure is 5 psig and the test is run for only 100 hours. The 100 Hour Test is run with extremely heavy load conditions, to give a quick indication of bearing durability. The test conditions also give a very high compression ratio which has been found to accelerate wear at the wrist pin. The usefulness of this test is strictly in the area of quick screening for new designs.

5.0 TECHNICAL INVESTIGATION

The features which were considered for incorporation into the improved refrigerator-freezer compressor design, and which were evaluated on the basis of the guidelines in Sec. 3, are listed in Table 5-A. The table indicates which features were ultimately selected for use in prototype samples, and includes the approximate gain in watts for each item. It also contains a reference to the section in the text of this report where the feature is discussed in detail.

In order to obtain a degree of flexibility until late in the development program, it seemed desirable to prepare two prototype design configurations. One version contained a four-pole motor and a two bearing crankshaft. The other, considered to be a secondary approach, contained a two-pole motor, a single bearing shaft, and a group of other new parts dictated by the requirements of the main features. Development work was carried out on both designs until it became apparent that the four-pole, two bearing version would more properly suit the goal of early commercialization and was also the more efficient of the two. A description of both designs can be found in Sec. 6.

5.1 Motor Performance Improvements

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, an 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% (see efficiency curve for sample PA-5021 on Fig. 5-1). 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 (see curve for sample P-5701 on Fig. 5-1).

TABLE 5-A

HIGH EFFICIENCY MOTOR COMPRESSOR UNITS
FOR REFRIGERATOR-FREEZER APPLICATIONS

FEATURES EVALUATED IN DEVELOPMENT PROGRAM

Feature	Improvement per Cal. Test Watts	Adopted In Final Design Yes/No	Detailed Discussion Sec. No.
Four Pole Motor	13	Yes	5.1.3
Copper Wire	8	Yes	5.1.4
Longer Stack	4	Yes	5.1.5
PSC Motor	9	Yes	5.1.2
Lower Rotor Res.	5	Yes	5.1.6
Lower Loss Core Mat.	6	Yes	5.1.7
Reduced Motor Torque	5	Yes	5.1.8
Reduced Rotor Air Gap	-	No	5.1.9
Reduce Valve Plate Reex.	-	No	5.2.1
Reduce Suc. Valve Reex.	-	No	5.2.2
Reduce Head Clear.	-	No	5.2.3
Omit Suc. Valve Stop	-	No	5.2.4
Use Plug Piston	-	No	5.2.5
Cyl. Head Insulation	-	No	5.3.2
Supercharging	-	No	5.3.3
Internal Cooling Fins	-	No	5.3.4
Insulated Dis. Area	-	No	5.3.5
Compressor Cooling	-	No	5.3.6
Plastic Suc. Muffler	3	Yes	5.3.7
Modified Val. Pl. Ports	4	Yes	5.4.1
Enlarged Dis. Gas. Path	2	Yes	5.4.2
Anti-Friction Brgs.	-	No	5.5.1
Improved Oil Stirrer	5	Yes	5.5.2
Two-Bearing Model	-	No	6.1
Four-Pole Crankshaft	-	Yes	6.2
Plug Crankshaft Vent	5	Yes	6.2
Total Gain In Watts (At A Capacity Of 780 BTU/HR)	69		

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.

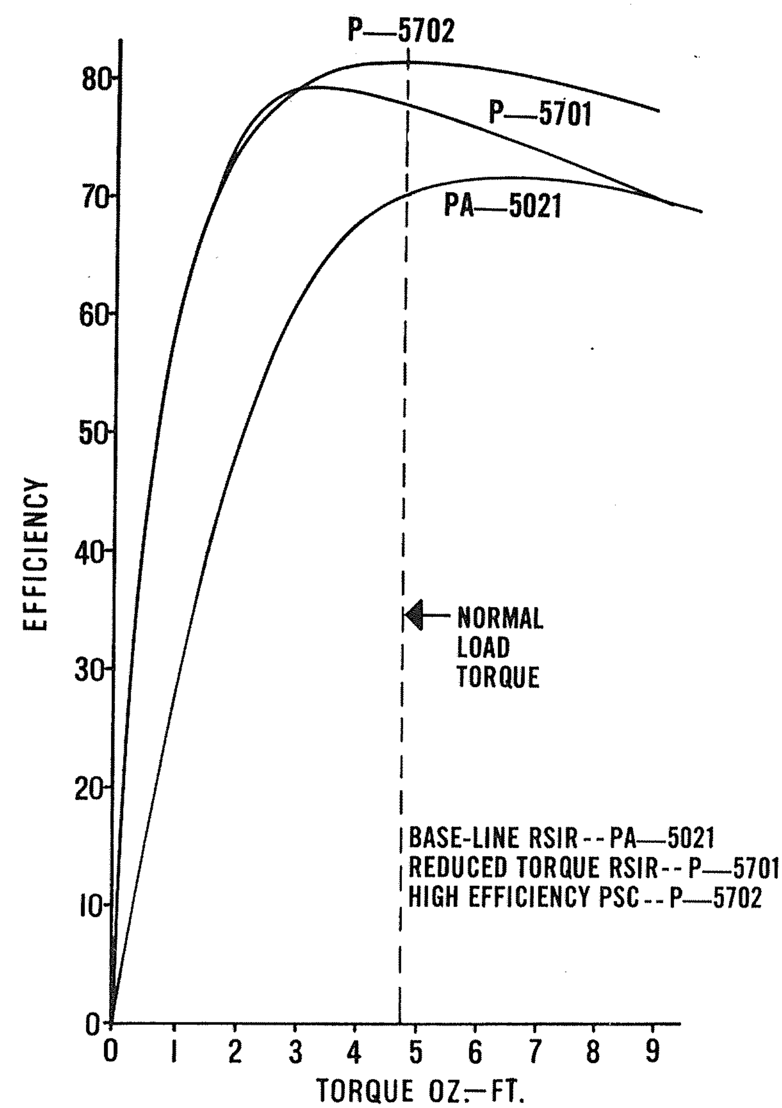
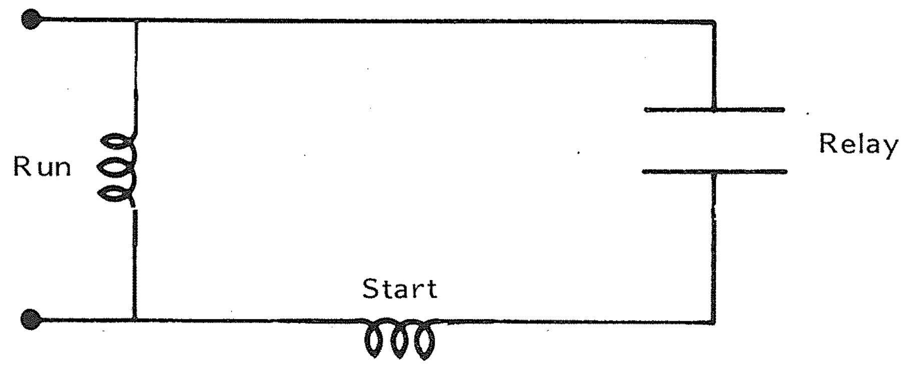


FIG. 5.—1. TWO-POLE MOTOR EFFICIENCY CURVES.

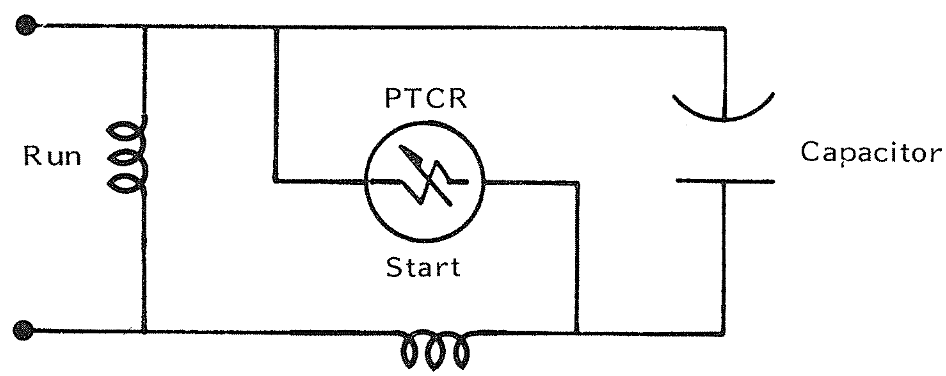
5.1.1 Resistance Start Inducation Run Motor

The most common and least costly motor and start system applied to a small refrigerator is an 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 Fig. 5-2.



Resistance Start
Induction Run
RSIR



Permanent Split
Capacitor With Positive
Thermal Coefficient Resistor
PSC

Figure 5-2 RSIR and PSC Motor Diagrams

5.1.2 Permanent Split Capacitor 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

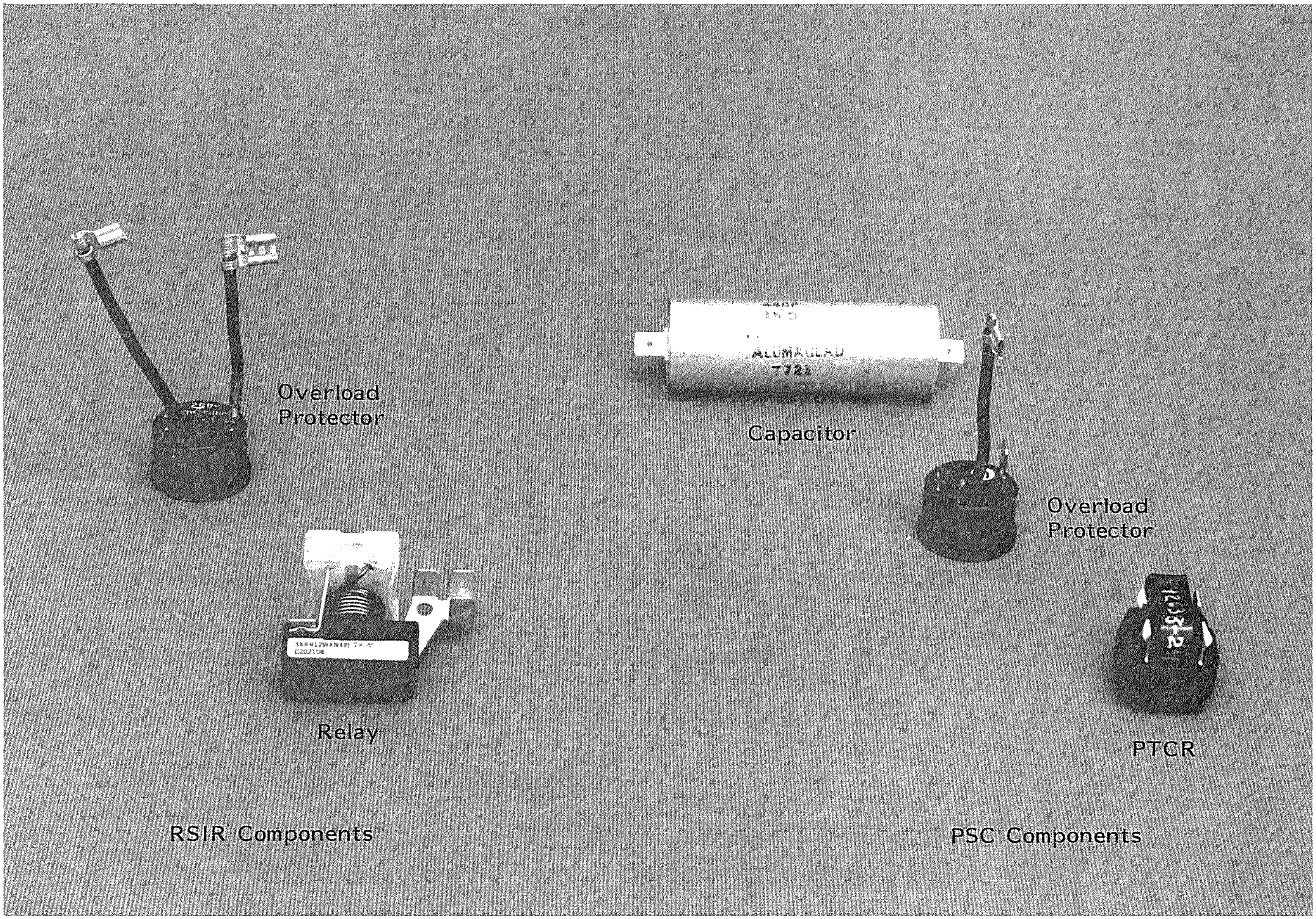


Figure 5-3 Starting and Overload Protection Components

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% as shown on efficiency curve for sample P-5702, Fig. 5-1. When assembled to compressors which had been otherwise modified to improve efficiency, coefficients of performances on the order of 4.7 were achieved. Hermetic motor starting and overload protection components are pictured on Fig. 5-3.

5.1.3 Two-Pole Vs. Four-Pole Motors

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.

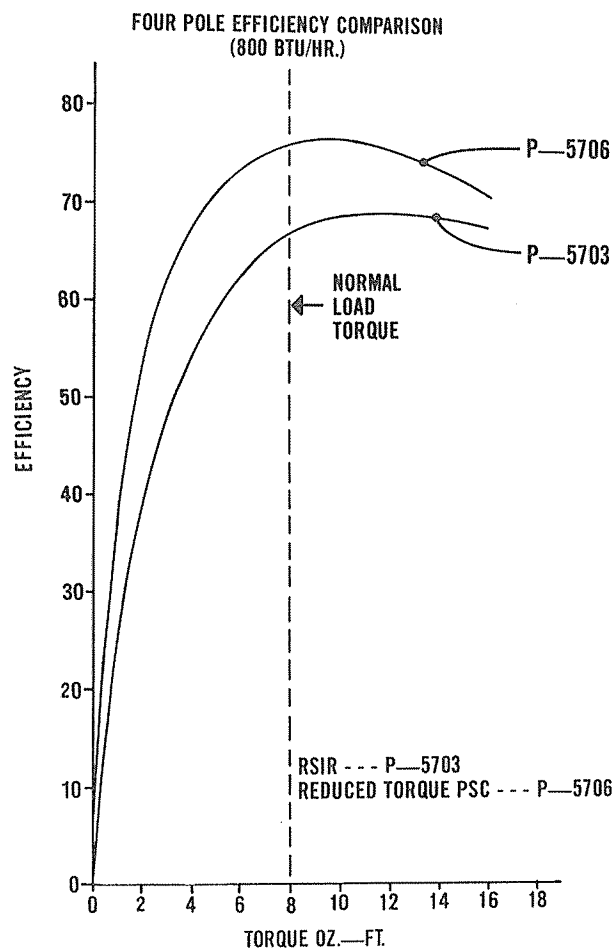
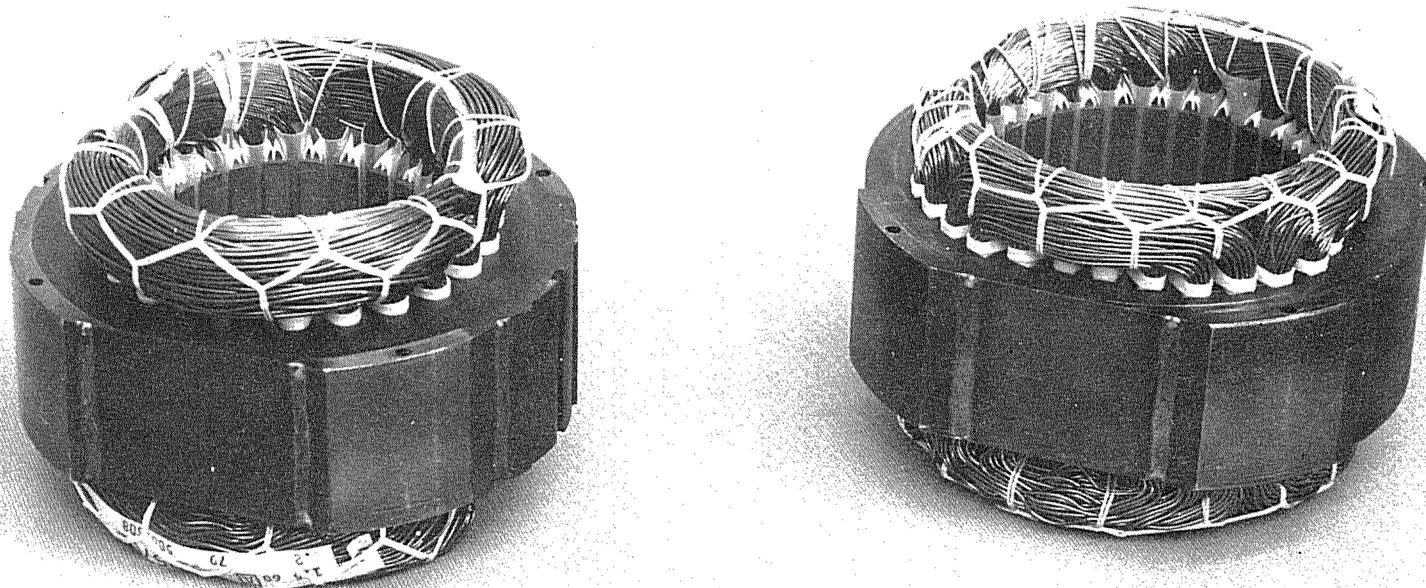


FIG. 5.—4. FOUR-POLE MOTOR EFFICIENCY CURVES.



Two Pole RSIR Stator

Four Pole PSC Stator

Figure 5-5 Hermetic Motor Stators

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. Fig. 5-4 shows a torque versus efficiency curve for both the RSIR and PSC four-pole motors developed. Pictures of typical two-pole and four-pole motor stators appear in Fig. 5-5.

5.1.4 Copper Wire in Place of Aluminum

Aluminum wire has a conductivity 0.618 that of copper, therefore, a greater cross section has to be used with aluminum to obtain the same resistance. Although the increased slot fullness initially caused manufacturing problems, the use of aluminum led to a cost reduction and it has been widely adopted for small hermetic motors. It is generally possible, however, to design a more efficient motor using copper wire, particularly if the latest methods for increasing slot fullness are applied, because the winding resistance is lower, and "copper" losses ($I^2 R$ losses) are reduced. Copper windings are used in the prototype motors, with a resulting improvement of approximately 8 watts in the final four-pole design.

5.1.5 Longer Stack Motor

Increasing the amount of steel in a motor by increasing the stack length reduces the operating flux densities in the various parts of the magnetic circuit. This reduces the ampere turn requirement, thus reducing copper ($I^2 R$) loss and core loss in the steel. The stack length in the CPC PA5021 with aluminum wire is 1.62 in. In the prototypes this was increased to 1.75 in. in the four-pole samples and to 2.00 in. for two-pole. The increase in stack length in the four-pole design contributed approximately 4 watts of the improvement obtained by motor redesign.

5.1.6 Lower Rotor Resistance

Reduction of rotor resistance is accomplished by increasing the amount of aluminum cross section area in the rotor bars and end rings of a cast squirrel-cage rotor. This is limited by the physical area available for end rings and the area for the bar slots that will allow a reasonable magnetic circuit. Reduction of rotor resistance is also limited by starting torque. Benefits of lower rotor resistance are realized in an ability to reduce operating densities of the magnetic circuit and in reduction of rotor losses, thus improving efficiency for a given strength motor. An added benefit is realized as a slight increase in speed at operating points. The rotors of the improved compressors are lower in resistance than current production designs, and this change results in a gain of approximately 5 watts.

5.1.7 Lower Loss Lamination Steel

The lamination material originally used in hermetic motor cores was silicon steel. Some years ago much of the material for the smaller models was changed to carbon steel to reduce cost. Improvements in the permeability of carbon steel made it possible to obtain levels of performance considered adequate at that time. Motor efficiency can still be improved, however, by reducing core losses without a transition back to silicon steel. This can be done by close control on chemistry in steel melting, by careful processing of both steel and laminations, and by reducing lamination thickness. Lower loss steel was used in the improved motors for this project, and the lamination thickness was reduced from .023 to .018 in. Any further reduction in thickness introduces serious manufacturing difficulties. The above changes result in an improvement of approximately 6 watts over the original T52 design. Both changes are being adopted by CPC for current production compressors.

5.1.8 Lower Maximum Torque Motor Designs

The requirement that compressors be able to operate under extreme conditions (minimum voltage, maximum temperature, maximum load) has caused the motors to be designed with higher torque capability than that required for normal operation. This has led to a reduction in efficiency at normal loading. The prototype motors have been designed for lower torque, so that the peak efficiency corresponds more closely to normal loading. This procedure has been facilitated by other changes which have improve mechanical and volumetric efficiency. This improvement is credited with a gain of approximately 5 watts.

5.1.9 Reduced Rotor to Stator Air Gap

Small motors used in refrigeration compressors are typically manufactured with a nominal air gap between stator and rotor of about ten thousandths of an inch. In theory, reduction of the air gap to a smaller value will reduce ampere turns required to pass the flux wave across the air gap. The change, however, requires a significant reduction in tolerances in parts manufacture and assembly to avoid interference (rubbing in the air gap).

Sample motors built and tested with as much as three thousandths reduction in air gap did not produce discernible improvements on dynamometer tests. CPC has, therefore, concluded that the air gap cannot be reduced enough to yield a significant improvement within the area of manufacturing feasibility. The air gap in the improved designs remains within the current production range.

5.2 Volumetric Efficiency Improvement by Reducing Reexpansion Volume

In any positive-displacement compressor, a certain amount of compressed gas, occupying small spaces not swept by the piston, such as a discharge port, is not exhausted through the discharge valve at the end of each compression stroke. This gas is expanded to suction pressure during the intake stroke of the piston. It is often referred to as reexpansion gas and the space occupied by it when compressed is called reexpansion volume. The reexpansion gas diminishes the capacity of the compressor, and since much of the energy expended in compressing it is not recovered, its presence detracts from compressor efficiency. In CPC compressors of the "T" line, reexpansion volume is found mainly in the following areas:

- Discharge ports in the valve plate.
- Volume above and behind the piston ring.
- Ports and tongue clearance areas of the suction reed.
- Piston clearance provided by the head gasket.
- Suction valve stop in cylinder wall.
- Suction valve port trepan,

5.2.1 Reduce Volume of Discharge Ports

The diameter of the discharge ports is determined mainly by flow requirements. (See also Sec. 5.3.7) On the other hand, the length of the ports cannot be reduced by decreasing the thickness of the whole valve plate, because of loss of the stiffness needed to maintain a gasket seal. These considerations led to experiments with a type of plate in which a local recess is cut into the surface in the discharge valve area, so that the effective length of the ports is decreased. A plate of this type is shown in Fig. 5-6, Item 8. This style of plate was used in an early experimental compressor which attained high efficiency levels.

Preliminary tests with plates of this type gave encouraging results. In preparing samples for test it was found to be very difficult to produce a satisfactory sealing surface on the valve seats, located in the bottom of the recess, even by hand methods in the CPC model shop. A review of the design with the machine tool manufacturers and parts suppliers yielded the unanimous opinion that it was not suitable for quantity production. The idea was therefore abandoned.

Another approach to the problem of reducing clearance volume in the discharge port involves the use of a poppet-type valve with a conical seat. Valves of this general type have been used in engines and in large compressors, but not in small hermetic compressors. Calorimeter tests on a sample gave the results below, in comparison with a T52 sample of the base-line design:

Valve System	Compressor Displacement	BTU	Watts	EER BTU/WHR
Production	.52 in^3	824	237.8	3.47
Disc	.52 in^3	912	262.8	3.47

Although capacity increased 10.6%, no increase in EER was observed. A sample placed on life test failed after 30 days, which is about one third of the normal test run. No standard noise test was made, but a high noise level was apparent on samples running in the laboratory. The poppet valve idea was accordingly discarded.

Trepanning around the suction port, on the valve plate, contributes to the reexpansion volume of the compressor. This volume is, however, very small when compared to other volumes contributing to reexpansion. In addition, the elimination of the trepanning around the suction port has resulted in poor suction valve sealing on various CPC experiments conducted in past years. It was judged that this effect would more than offset any gains resulting from reduced reexpansion volume. This idea was therefore not pursued any further.

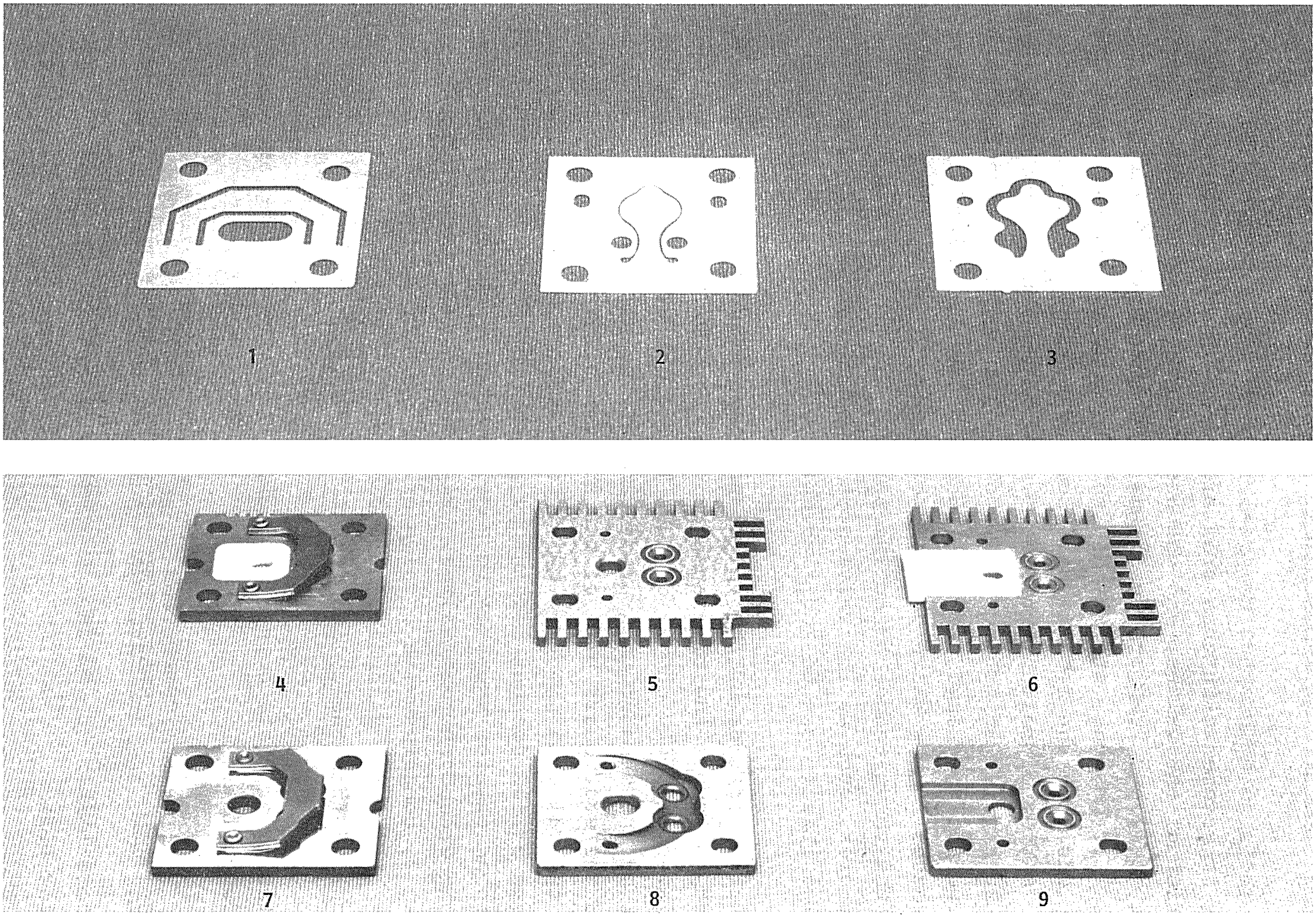


Figure 5-6 Valves and Valve Plates

5.2.2. Reduce Reexpansion Volume in Suction Valve Reed

There is no doubt that compressor valve design affects performance, but improvements in this area are beset by difficulties. The action of the valves, and the associated stress patterns, are very complex. Computer programs have been written for certain specific designs, but involve so many assumptions and simplifications that the results have been of limited practical value. The matter of reliability, or freedom from fatigue failure, is of such vital importance that compressor engineers are generally reluctant to change proven designs in any way which could affect stresses in the valve during operation. This applies particularly to the valve tongue or active part of the valve. Non-moving features, such as the valve ports, and the static areas of the valve itself, can be modified more readily.

Several features of the valves and associated parts can be modified to reduce the clearance or reexpansion volume, which increases capacity and also tends to improve efficiency, although not by the same amount. For example, the slit between the tongue and the stationary part of the suction valve can be made narrower if the valve is produced by a photochemical etching process instead of the present mechanical punching method. CPC contracted with an outside supplier to make up several configurations of photo-etched suction valves. Fig. 5-6 shows a photo-etched valve, Item 2, compares with a valve of the present design, Item 3. The reeds were assembled into compressor samples with valve stops machined into the cylinder bore.

Calorimeter tests both before and after life testing are tabulated below:

TEST	COMPRESSOR DISPLACEMENT	BTU	WATTS	EER BTU/WHR	NO. OF SAMPLES
Before Life	.52 in^3	808.5	233.8	3.45	4
After Life	.52 in^3	856.7	239.3	3.58	4

Before life testing, the samples showed no improvement in EER over the baseline design. A small improvement appeared after life testing. Improvements of this kind are not unusual after prolonged testing. They are presumed to result from "running in" of the sealing or bearing surfaces.

It has been found necessary to finish the edges of valve reeds produced by stamping operations to remove surface defects and stress concentrations, which increase the hazard of fatigue failure in service. This is done by an abrasive tumbling process. It was originally thought that photo etched valves would not require edge finishing, but recent research by the Uddeholm Steel Co. showed that this is not the case, since corrosion pits on the chemically etched edges could cause fatigue failure. It is not, however, possible to finish the edges of the narrow slit in the proposed suction valve design by the abrasive tumbling method now in use, because the particles in the abrasive medium are too large to pass through the slits.

Since the improvement found with the narrow-slit valve was marginal, and there is no readily available way to finish the edges, this feature is not recommended for the prototype design.

5.2.3 Cylinder Head Clearance

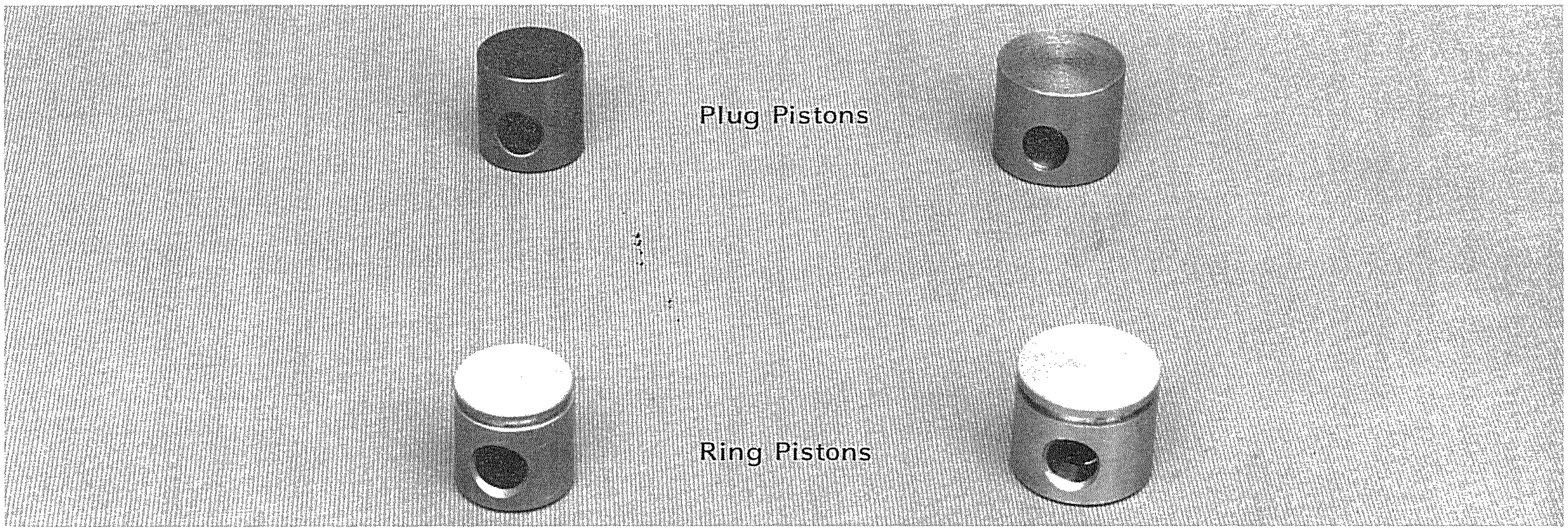
A deck gasket is utilized between the suction valve and the cylinder housing. Normally, these gaskets are selected to provide .003" to .005" clearance between the piston, at top dead center, and the housing. The volume associated with this clearance contributes to the total compressor reexpansion volume. Several tests were made to determine if this clearance could be reduced, thus reducing the reexpansion volume. When compressors were built and tested with reduced head clearance, a knocking noise developed. This noise was probably caused by the piston hitting the suction reed. In addition to the unacceptable noise, this condition would no doubt eventually cause suction valve reed failure. For these reasons, reduced head gasket clearance is considered not to be a practical expedient for the purpose of reducing reexpansion volume.

5.2.4 Reduction of Valve Stop Volume

Omitting the suction valve stop in the cylinder bore decreases clearance volume. Several test compressors were built using suction reeds with the tip removed. The cylinder housings were machined in the CPC model shop with no valve stop cavity in the bore. Two sample compressors were tested using two different reeds with no tips. The first reed tested was a production "T" type which had the tip cut off. When tests were attempted with this reed, it was thought that burrs may have been causing leakage back through the suction port. Photo-etched reeds with no tip were then procured from an outside manufacturer. These valves were installed in the test compressors and run on the calorimeter. The results obtained from the tests can be seen below. Nominal production performance figures have been included for comparative purposes.

REED	COMPRESSOR DISPLACEMENT	BTU	WATTS	EER BTU/WHR	NO. OF SAMPLES
Production	.52 in^3	780	224	3.47	Production
Tipless Production	.52 in^3	857.7	241.3	3.55	2
Photo-Etched	.52 in^3	916.7	266.8	3.44	2

The above test results show an apparent discrepancy between the production reed with no tip and the photo-etched reed of a similar design. The reason for this is uncertain. There is no significant increase in efficiency in either case.



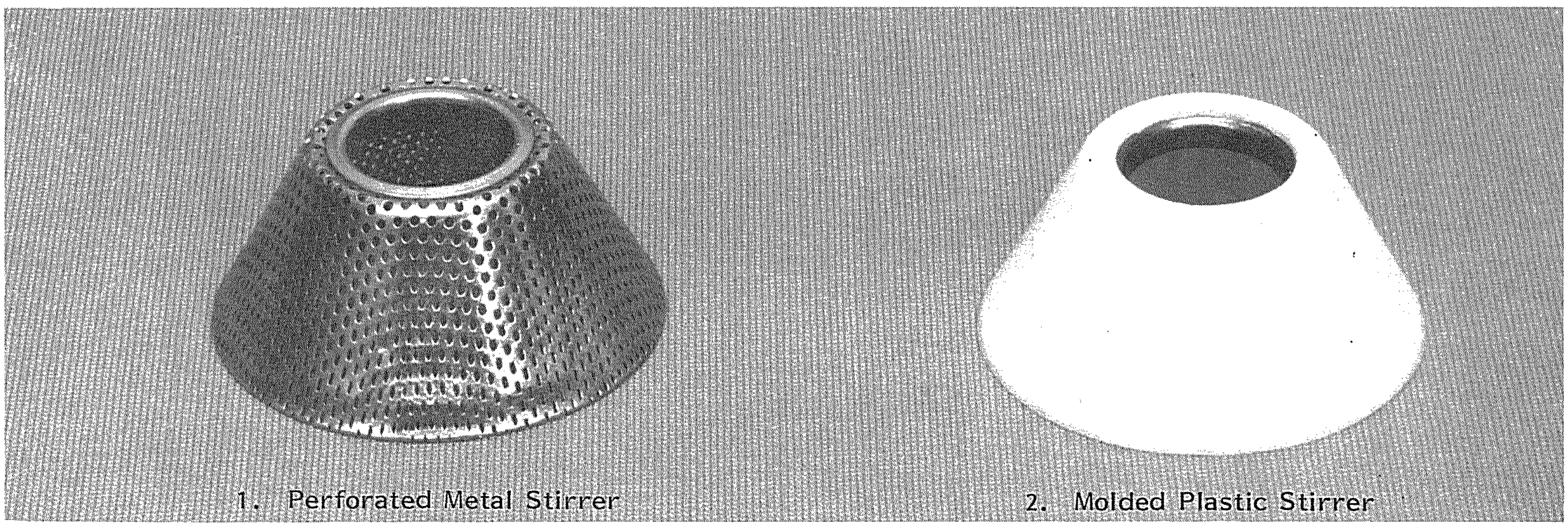


Figure 5-7 Piston and Oil Stirrers

Suction valves operated without a stop in the cylinder wall may be expected to have higher bending stresses and could not be adopted without a long and extensive reliability test program. Since the calorimeter test results were not encouraging, and there are potentially serious reliability problems, stopless suction valves will not be used in the high efficiency design.

5.2.5 Plug Piston Vs. Ring Type Piston

Two types of pistons have been used in small refrigerator compressors. One type uses a piston ring. The other, generally called the plug type, does not use a ring, but operates with a close clearance, usually in the area of .0004 to .0006 in. on the diameter, to minimize blow-by losses. The two types of pistons are pictured in the upper part of Fig. 5-7. The plug piston must be made of the same type of material as the cylinder, to avoid problems with differential thermal expansion. In practice, both piston and cylinder are nearly always made of ferrous alloys. The usual practice in the industry is to produce the close clearance between piston and cylinder by selective assembly of the parts.

In the original design of the "T" line of compressors at CPC the ring type piston was selected because the piston could be made of easily machined aluminum, and because the selective assembly procedure was eliminated. The cylinder consists of a cast iron insert or sleeve in the die cast aluminum cylinder housing.

In the ring-type design there is a small amount of unoccupied space in the ring groove, which acts as part of the reexpansion volume and tends to detract from compressor performance. In the early stages of the efficient compressor development, tests were made to compare the performance of the two types of pistons in typical two-pole samples. These tests showed an apparent gain of about 10 watts. As a result, the plug piston was for some time thought to be the preferred type for the high-efficiency designs.

More recent development have led to a review of this situation. The adoption of four-pole motors has required changes in the lubricating system, as described in Sec. 6.2. Among the results of these changes is more copious lubricating of the piston. Tests on plug pistons compared with copiously lubricated ring-type pistons in four samples gave the results tabulated below.

SAMPLE NUMBER	PISTON TYPE	CAPACITY BTU/HR	WATTS	EER BTU/WHR
1013-1	Plug	767.1	165.7	4.63
1012-1A	Ring	811.7	161.5	5.03
1013-2	Plug	810	170.4	4.75
1012-2A	Ring	821.6	167.2	4.91
1013-3	Plug	748.9	159.3	4.70
1013-3A	Ring	811.2	165.4	4.90
1237-A	Plug	854	180	4.74
1237-B	Ring	831.2	162.4	5.12

In each sample the ring type showed superior efficiency. Tests on plug pistons with increased lubrication did not show a corresponding improvement. As a result of these tests, a decision was made to incorporate the ring type in the high efficiency design. This has the additional advantage of eliminating the cost of selective assembly facilities for the field demonstration project and for future mass production.

5.3 Improve Volumetric Efficiency by Reducing Suction Gas Superheat

In a reciprocating piston compressor (or any other positive displacement type), the ideal displacement, or the volume swept by the piston on each stroke is fixed by the mechanical dimensions of the parts. The actual amount of refrigerant pumped, however, is affected by the pressure and temperature of the suction gas, which are dependent in turn on a number of design features. Design changes which allow more mass of refrigerant to be drawn into the cylinder per stroke will directly increase capacity. There may be an increase in power consumption also, but it will not increase in proportion to the capacity increase, so that there will be a net gain in EER. The capacity can always be adjusted to the appropriate level by dimensional changes within the preferred design framework.

At standard calorimeter rating point conditions the temperature of the suction gas returning to the compressor is 90°F. This temperature is measured just before the gas enters the compressor shell. The temperature of the gas entering the cylinder is, of course, much higher. It is well known that compressor performance can be improved by reducing suction gas superheat. In order to obtain concrete figures on the amount of improvement, CPC ran a series of calorimeter tests in which the suction gas was artificially cooled by an auxiliary refrigerant system. The temperature in the suction cavity in the cylinder head was measured by a thermocouple. Performance at a wide range of temperatures is tabulated below:

SUCTION PLENUM °F	BTU/HR	WATTS	EER BTU/WHR
211	840	231.2	3.63
165	861	231.6	3.71
125	918	240.4	3.81
123	870	228.4	3.81
97	909	230.4	3.95
85	915	235.2	3.89
72	942	236.8	3.98
68	961	241.6	3.98

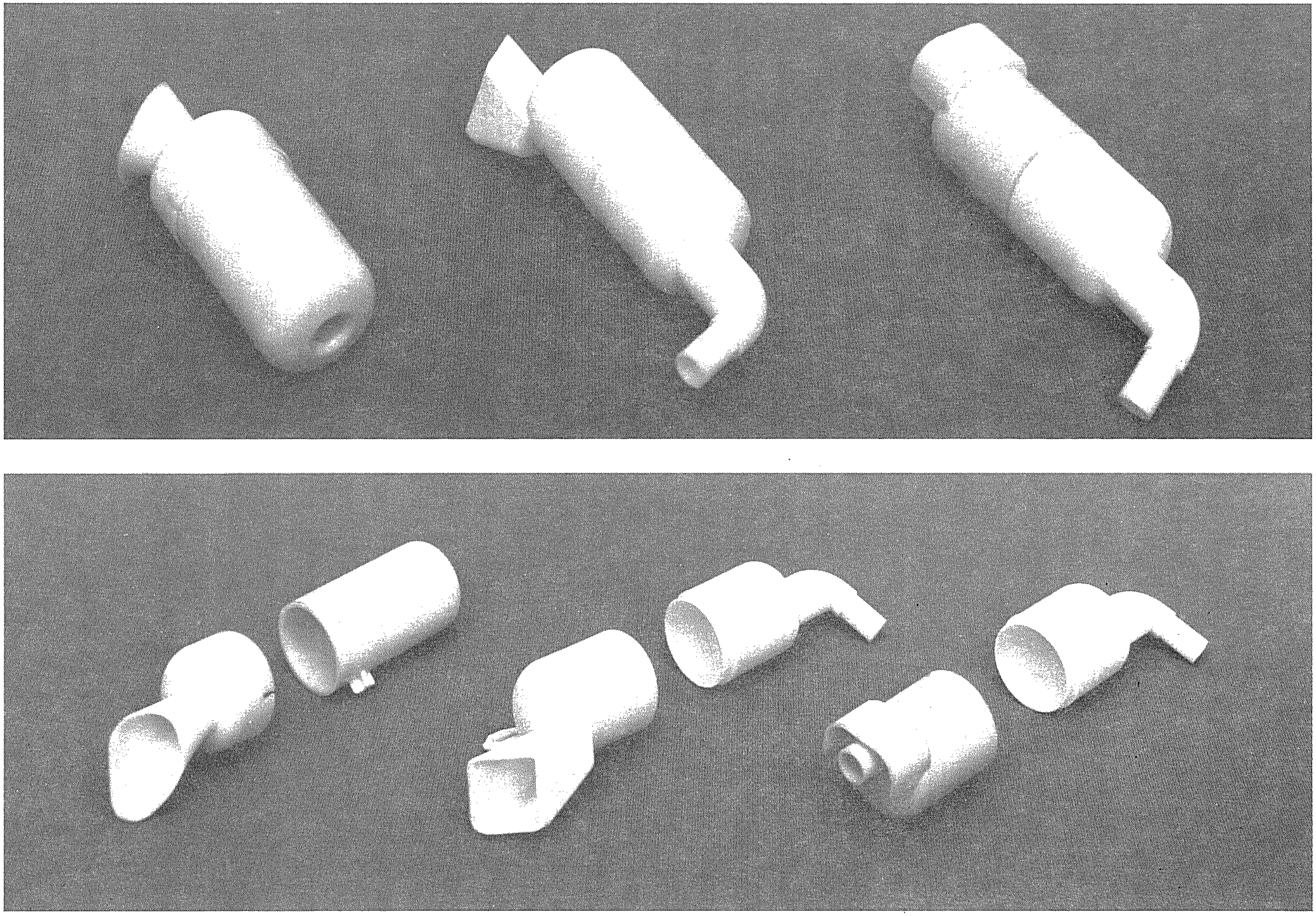


Figure 5-8 Thermoplastic Suction Mufflers Assembly and Parts

The first point on the table is similar to operating conditions with the present design. While it is impossible to reach the lower range shown in a practical compressor design, the investigation demonstrates that a significant improvement can be made with a feasible reduction in superheat.

5.3.1 Guided Suction Muffler Intake

Having demonstrated the advantages of a reduction in suction gas superheat, the next step was to investigate available means of accomplishing it. In the current CPC compressor, and in other reciprocating piston designs in the industry, 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 a chamber in the cylinderhead before entering the cylinder. At each point along this path, its temperature increases. Accordingly, consideration was given to design changes which would tend to isolate the gas from heat sources.

An external suction muffler, which would be cooler than the present internal type, was first considered and a sample was constructed. The design, however, presented problems in manufacture and subsequent handling, and also added complications in returning lubricating oil circulating with the refrigerant to the sump in the compressor shell. The external muffler was, therefore, discarded, and an internal muffler made of a less heat conductive thermoplastic polyester was adopted instead. Three different designs of thermoplastic suction muffler are pictured in Fig. 5-8. Along with this, a flexible tube was added to conduct the gas directly from the entrance to the muffler intake. This combination prevents the gas from circulating around hot components, while the polyester muffler transmits less heat than the present metal type. It is necessary to include a small vent in the muffler to equalize pressures and to allow the return of oil to the sump. Pressure equalization

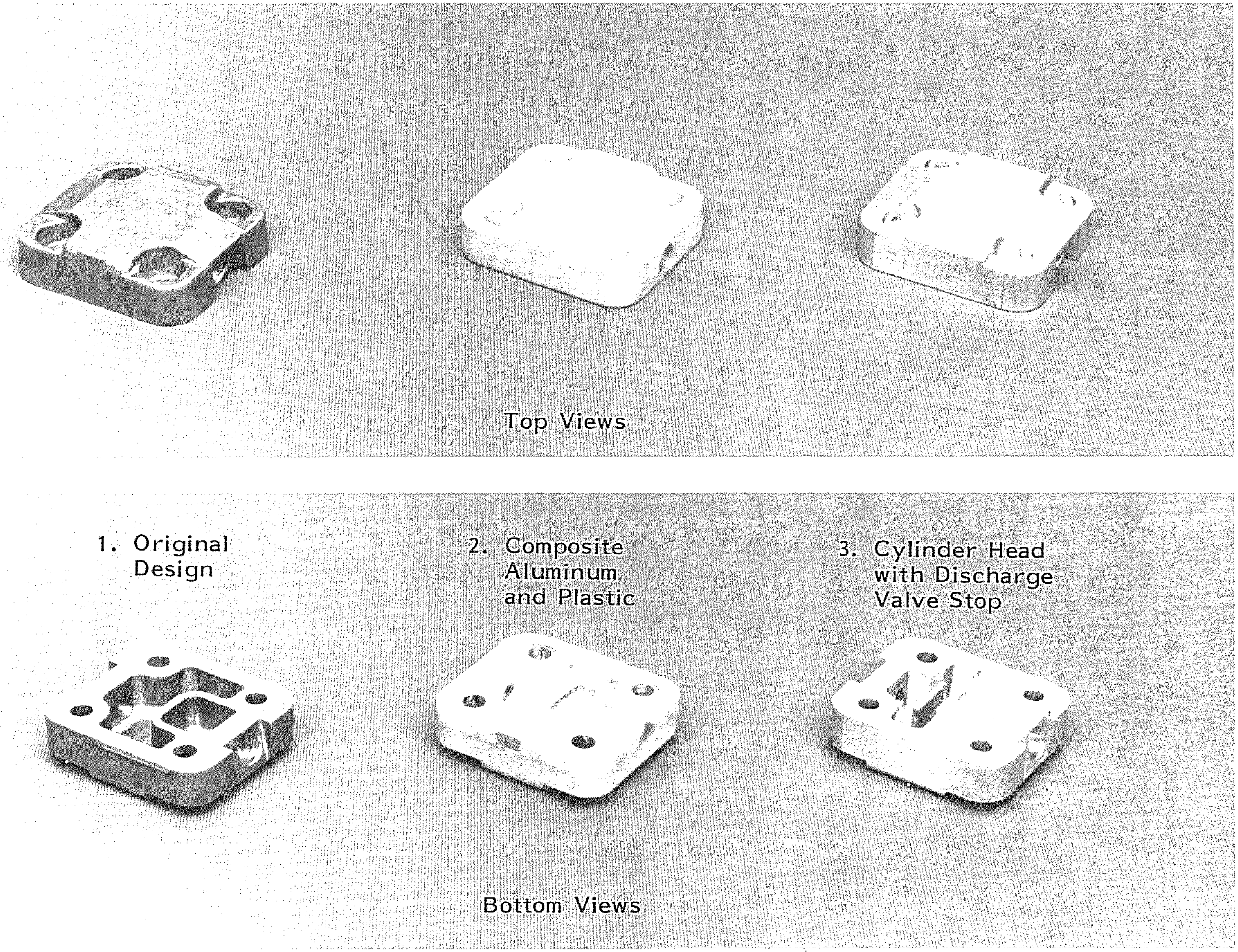


Figure 5-9 Experimental Cylinder Heads

is required because there is a certain amount of blow-by past the piston. If the blow-by gas is not able to return to the suction side of the system, a pressure approaching the discharge pressure will build up inside the shell. This condition will interfere with starting, and may collapse the flexible tubing in the direct suction connection. An oil return vent is needed because a small amount of the oil which lubricates the piston is picked up and circulates with the refrigerant. If this oil cannot return to the sump, the quantity circulating will become large enough to interfere with heat transfer in the evaporator, while the amount remaining in the oil sump may become inadequate for lubrication and cooling.

In order to economize on calorimeter testing, the polyester muffler and direct suction connection were tested in combination with the insulated discharge muffler and discharge line described in Section 5.3.5. The average test results which are included in a tabulation in that section, show a gain in EER from 3.61 to 3.74 when the plastic muffler and direct connection were added. The direct suction connection, however, presents problems from an assembly standpoint. Further, later experiments indicated that a similar improvement in efficiency is produced by a short stub tube from the shell pointing into the suction muffler entrance, but not connected to it. Based on these experiments and manufacturing considerations, it has been determined that the efficient compressor should have guided flow of the suction gas into the muffler intake, but not a direct connection. Additional information on the plastic suction muffler will be found in Sec. 5.3.7.

5.3.2 Cylinder Head Insulation

Another way of preventing suction gas heating is to thermally insulate the suction chamber from the discharge chamber in the cylinder head. This may be accomplished by a variety of means.

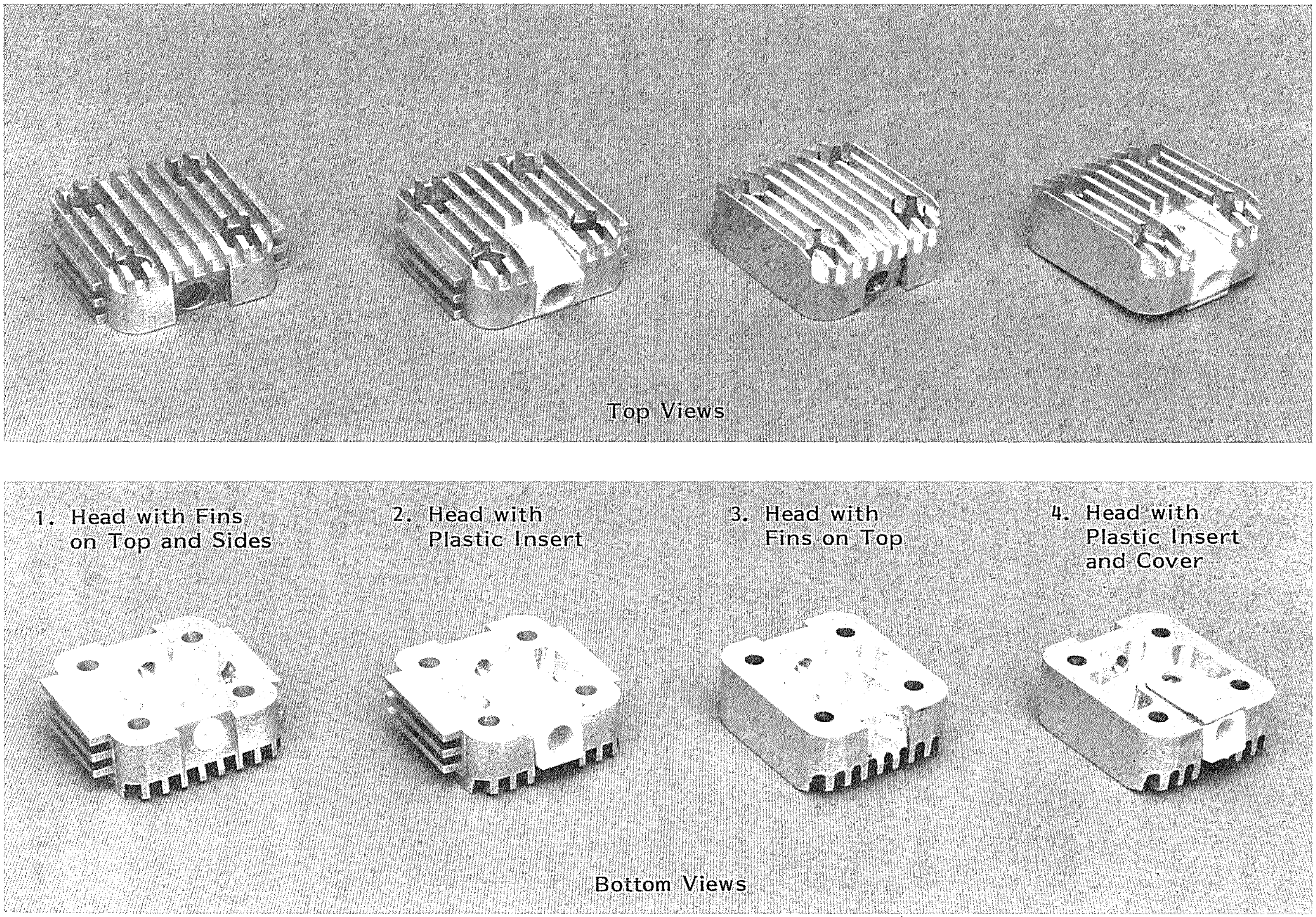


Figure 5-10 Experimental Cylinder Heads With Fins

The possibilities include the formation of a cylinder head out of an insulating material. Plastic or ceramic would be candidates. Attempts to use polyester molded heads have proved unsuccessful due to the poor mechanical properties of the material. Composite plastic and aluminum heads have also yielded disappointing results. A sample composite head is shown in Fig. 5-9, Item 2.

In order to avoid the disadvantages of plastic, while at the same time retaining its insulating features, an inserted polyester liner was used in an aluminum head for calorimeter test purposes (see Fig. 5-10, Items 2 and 4). Experience with the test samples has shown that it is difficult to hold a polyester insert firmly in place without inducing leakage problems. Molded in place liners may prove to be the answer to the problem. This design calls for the head to serve as the female half of the mold. Placed in a holding fixture, a male mold is utilized to inject a plastic into the cylinder head, creating a liner which adheres to the head. This concept appears to be more feasible and less costly than other insulating schemes. Several samples of this type have been constructed and have undergone functional evaluation. In order to investigate the effect of enclosing the suction chamber completely in plastic, some tests were made with inserts in the valve plate as shown in Fig. 5-6, Page 43, Items 4 and 6.

Calorimeter tests on the polyester liners by themselves have not shown any benefits. It seemed possible, however, that an improvement would have resulted if the liners were used in conjunction with the polyester muffler and direct suction connection, thus providing insulating for the whole path of the suction gas. The combination was tested and the results analyzed. There was no apparent advantage associated with the use of plastic suction chamber liners. The efficient compressor, therefore, will not utilize these features.

5.3.3 Investigation of Supercharging

It has been known for a long time that the performance of a refrigeration compressor improyes as the suction pressure increases. Under usual operating conditions, there is no way to take advantage of this fact because the suction pressure is determined by the evaporator temperature. If, however, some auxiliary feature could be added to a compressor to raise the pressure before the gas is drawn into the cylinder, without much expenditure of energy, the efficiency should increase. The use of a turbosupercharger to improve the performance of internal combustion engines is an analogous expedient.

An attempt was made to apply this principal by enclosing the crankcase area on a T-line compressor to form an auxiliary chamber in which the suction gas would be partially compressed by the downward stroke of the piston before entering the cylinder head. The gas entered the enclosed chamber through a check valve similar to the regular suction valve.

Tests on a sample constructed as described above showed no improvement in performance. It was concluded that the crankcase chamber was equivalent to a cylinder with the same displacement as the regular cylinder and a very high reexpansion ratio, which made it incapable of pumping enough gas to supercharge the main cylinder. The idea was therefore discarded, although the theoretical concept of supercharging has merit.

5.3.4 Fins on Internal Pump Parts

In considering suction gas superheating, it appeared to be possible that superheat could be reduced by the use of cooling fins on the surface of the cylinder head valve plate, or cylinder block. Experimental cylinder heads with fins are shown on Fig. 5-10, valve plates on Fig. 5-6, (Page 43), Items 5 and 6, and cylinder blocks with fins on Fig. 5-11. Tests on this idea, however, showed that it was not beneficial. Test results on samples with and without fins on the cylinder block were as follows:

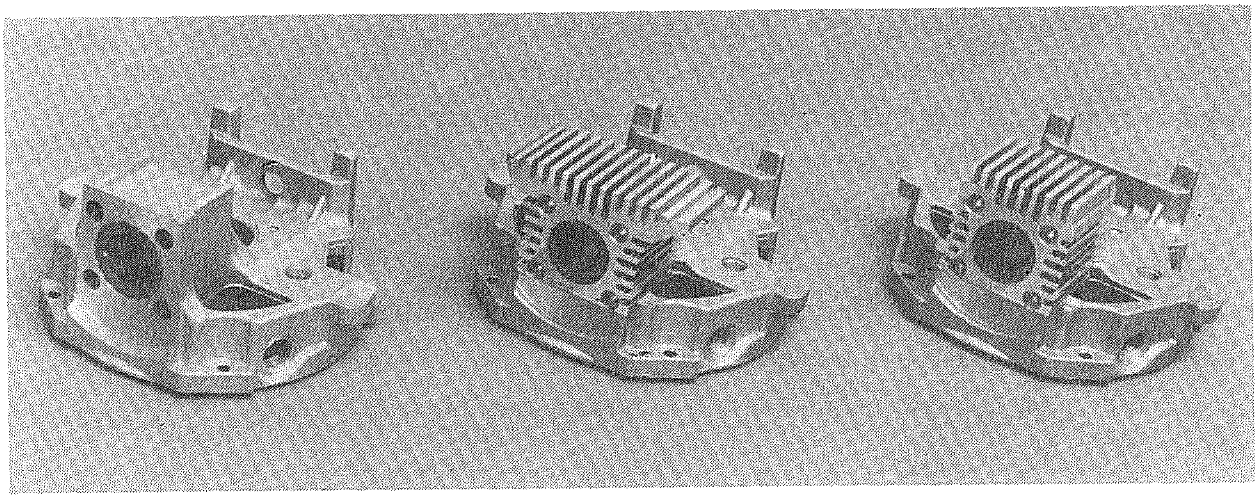


Figure 5-11 Experimental Cylinder Housings

SAMPLE TYPE	BTU PER HR.	WATTS	EER BTU PER WHR
Std. configuration, no fins	780	244	3.19
With fins on cylinder	779	242.8	3.20
Same, retest	777	244.8	3.17

In order to secure more data in this area, a sample was built with a water cooled coil attached to the face of the cylinder-block. This sample was tested at several temperature levels, with the following results:

INLET WATER TEMP. (^{O}F)	CYL. TOP TEMP. (^{O}F)	BTU/HR	WATTS	EER BTU/WHR
164	191	858	252.8	3.39
91	172	876	259.2	3.37
55	147	876	260.8	3.35

The failure of the fins to produce an improvement is explained by the fact that heat emitted by the fins is absorbed by suction gas which circulates around them before it enters the muffler. The heat is merely recirculated and the final temperature of the suction gas is not reduced. Supplementary cooling of the cylinder which does reduce the temperature of the parts without transferring heat to the suction gas, produced a minor increase in capacity, but no efficient improvement. As a result of these test findings, the idea of using internal fins was discarded, and they will not be included in the prototypes.

5.3.5 Insulated Discharge Gas Area

Consideration of the test results reported in the previous section led to the hypotheses that it would be beneficial to transmit as much of the heat of compression as possible to the condenser. In this way, there would be less heat released inside the shell where it could raise the temperature of the suction gas. To carry out this idea, samples were built with a shield enclosing the discharge muffler and a plastic sleeve around the discharge tube leading to the shell. Test results are included in the following table.

TYPE OF SAMPLE	AVERAGE TEST RESULTS		
	CAPACITY BTU/HR	WATTS	EER BTU/WHR
Present Standard Design Samples	813.2	229.4	3.54
With Insulated Discharge Muffler	855.8	237.2	3.61
Same with Plastic Suction Muffler and Direct Connection	899.8	240.6	3.74

The tests reported above were made by adding features progressively to the same set of samples to minimize individual variations. In the first step, the insulated discharge muffler and tube were added, with a resulting improvement in EER from 3.54 to 3.61. The next features added were the polyester suction muffler and direct suction connection discussed in Sec. 5.3.1. This addition led to further improvement in EER to 3.74. It will be noted that these changes also resulted in significant increases in capacity. Later tests with the insulated discharge muffler showed that the increase in EER obtained was not repeatable. Since the implementation of this design from a manufacturing standpoint would be difficult and the improvements to be expected were at best marginal, the insulated discharge muffler idea was not pursued any further.

5.3.6 Compressor Cooling

The operation of the motor and compressor results in the generation of substantial amounts of heat. A portion of this is transmitted to the condenser with the compressed gas, but the major part of it must be dissipated to the atmosphere from the surface of the shell. If temperature levels inside the motor-compressor unit are allowed to become too high, the results may be eventual thermal degradation of the motor insulation, or oil breakdown in the discharge valve area leading to deposits of carbon, usually referred to as "coking", which prevent proper seating of the valve. In small compressor models, the surface of the shell is able to dispose of the heat adequately, but in large capacities, it may be necessary to add cooling fins, use an air circulating fan, or add cooling tubes in the oil.

There has been a general intuitive feeling that motor-compressor efficiency would improve if overall temperatures were reduced below current operating levels. In order to verify this idea, calorimeter tests were made on samples* which could be artificially cooled by circulating water, or refrigerant from a separate system, through an oil cooler tube in the bottom of the shell. The test results are tabulated as follows:

*Standard Columbus Products T-52 Compressors

SAMPLE NO. 1

TEMPERATURES, °F							
MAIN WINDING	SUCTION MUFFLER	SUCTION PLENUM	CYLINDER HEAD	DISCHARGE MUFFLER	BTU/HR	WATTS	EER BTU/WHR
173	157	180	211	231	791	245.6	3.22
163	140	164	195	214	804	254	3.16
153	144	169	-	220	792	253.2	3.12
143	135	160	191	210	822	258.8	3.17
133	126	151	-	203	822	260.4	3.15

SAMPLE NO. 2

TEMPERATURES, °F					
WINDING	SHELL TOP	SHELL BOTTOM	BTU/HR	WATTS	EER BTU/WHR
177	154	148	834	240	3.44
150	124	114	861	259.8	3.32
135	114	105	888	259.4	3.41

These results indicate that added oil cooling is effective in reducing temperatures throughout the motor-compressor unit. There is an increase in capacity, attributed to lower suction gas superheat, but the power input in watts also increases so that there is no net gain in EER. Since general temperature reduction does not improve energy efficiency, the efficient compressors will not include any added cooling features. Oil coolers could be adapted later, of course, if their use was dictated by systems or ambient operating conditions.

It is true that the motor by itself is more efficient at lower temperatures, but the change is small. This is shown by the Athens Products Company test data tabulated below, for a typical motor at 77 and 167^{O}F (25 and 75^{O}C), which is a wider range than any to be encountered in normal compressor operation.

Temperature ^{O}C	77	25
Temperature ^{O}F	167	77
Watts	187	186
RPM	3490	3510
Efficiency %	77.8	78.7

5.3.7 Suction Muffler

Suction mufflers were injection molded at CPC from several grades of a thermoplastic polyester manufactured by The General Electric Company under the trade name of "Valox". The plastic muffler was designed so that its mouth is close to a suction tube coming through the shell. In order to accomplish this the location of the latter tube in the shell had to be changed. Several experimental suction muffler configurations are shown on Fig. 5-8, Page 51.

The use of a thermoplastic muffler permits molding of numerous forms that would have been very difficult to fabricate out of steel. Since it was though that unreinforced thermoplastic might deform at compressor temperatures, samples were molded from a 30% glass filled material. Compressors utilizing these mufflers exhibited an improvement in capacity. A typical capacity increase was 35 BTU/HR from 780 to 815 BTU/HR. No increase in power consumption was noted. Further testing indicated that an efficiency increase of about 0.15 BTU/WHR could be expected. Corrected to 780 BTU, the results can be interpreted as a reduction in power consumption of about 3 watts.

Life tests using this muffler were carried out. Unfortunately, the glass filling of the Valox was too abrasive for this application and contributed to wear of the aluminum parts during the life test. An unfilled thermoplastic material was substituted and subsequent test have yielded no problems in the areas of deformation or wear. Plastic suction mufflers are used in the prototype samples.

5.4 Reduced Refrigerant Flow Restrictions

At a number of points in a compressor restrictions in the refrigerant flow path detract from operating efficiency. In particular, restrictions in the discharge side of the pump lead to an increase in peak discharge pressure in the cylinder and thus require power input with no corresponding gain in output. Investigations in this area are discussed in the following sections.

5.4.1 Valve Plate Port Area Investigation

The selection of the optimal area of the valve ports contributes to compressor efficiency. The location of the ports may be seen on the sample valve plates shown on Fig. 5-6, Page 43. The optimal area of the discharge ports is a compromise, because too small a port restricts flow, while too large a size detracts from performance because the port forms part of the reexpansion volume. In addition, the relative sizes of the suction and discharge ports must be properly coordinated.

The original CPC design used a single suction port with a diameter of 0.265 in. and a pair of discharge ports of 0.125 in. and 0.140 in. diameter respectively. On the basis of calorimeter test results, the ports have been changed on the four-pole high efficiency design to two suction ports of 0.260 in. diameter and two discharge ports of 0.140 in. diameter. These dimensions have been found to be suitable for all models in the proposed high efficiency line.

The increase in port areas contributes an improvement of approximately 4 watts in the case of the W80 model.

5.4.2 Discharge Muffler Investigation

Present CPC discharge mufflers, for domestic refrigeration, are composed of two stages separated by a baffle. A tube inserted into the baffle provides for the gas flow between the first and second stages of the muffler. The first modification tried in an attempt to improve 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, the efficiency increased 0.1 BTU/HR. This efficiency increase, however, was obtained at the penalty of an increase in noise of about 4.5 dBA. As a result of the marginal improvement in performance, coupled with a dramatic increase in noise level, the above changes were determined to be inadvisable.

Tests were also conducted using a larger diameter discharge line. A discharge line of this type is currently used in production room air conditioning compressors, and was used at one time in the "T" line refrigerator compressors. A noise reduction program led to a reduction in the diameter of the discharge tube.

More recent testing has shown that a reduction of approximately 2 watts could be achieved with the use of a larger diameter discharge tube. Since the four-pole design runs at a lower noise level than a two-pole compressor of the same capacity, the larger diameter discharge tube appears to be a worthwhile efficiency modification.

The current large diameter RAC discharge line has two loops to absorb the vibration of the compressor assembly. Efforts are currently underway to modify this line from two loops to one as a cost reduction. Testing has shown that the attendant small change in total line length will not greatly alter the efficiency gains or the sound level of the compressor. If a single loop discharge tube is ultimately adopted in current CPC compressor designs then it will also be incorporated in the high efficiency compressor

5.5 Improved Mechanical Efficiency

The mechanical efficiency of a compressor is the ratio of the work done on the refrigerant gas when it is compressed to the work delivered to the shaft by the motor. In a small reciprocating compressor it is on the order of 85%. The major contributor to a reduction in mechanical efficiency is the friction developed at the bearings and around the piston. Consideration was given to reducing friction in all the interfaces contributing to the overall power losses. These interfaces are:

a) Piston and cylinder
b) Wrist pin/conn rod
c) Crankpin/conn rod
d) Crankshaft/cylinder housing
e) Crankshaft/motor housing (radial)
f) Crankshaft/motor housing (thrust)

When these items are considered individually, it appears that the only feasible means of reducing friction losses at (a) and (b) is by supplying more copious lubrication, as described in Sec. 6.2. The remaining items were originally believed to be possible candidates for the use of rolling element bearings. A report on the investigation of this application is given in Sec. 5.5.1 on the next page.

5.5.1 Rolling Element Bearings

Of the various types of rolling element bearings currently available, only needle bearings were mechanically feasible within the space limitations available.* 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 on the crankshaft, where the application presents fewer mechanical problems than it does for the wrist pin and crankpin. The connecting rod/wrist pin bearing is not a good candidate for needle bearing application because the space inside the piston is so confined. The crankpin application is feasible except for lubrication problems.

In the present design, lubricating oil rising through the crankshaft is delivered progressively to the upper main bearing, the crankpin, and the wrist pin by a series of passages in the shaft and connecting rod. If a needle bearing is installed at any location, however, oil reaching that point will escape through the space around the needles, thus being diverted away from points farther along the passageway. Tests indicated that the installation of a needle bearing in the main upper bearing location resulted in inadequate lubrication of the crankpin unless oil seals were added to enclose the main bearing. Similarly, a needle bearing in the crankpin would deprive the wrist pin of lubrication. For this reason, adaptation of needle bearings to the crankpin is impractical.

An extensive test program was carried out with needle bearings on the crankshaft. Tests were made in samples similar to the present T-line, and also in samples similar to the single-bearing design shown in Fig. 6.2. The latter type included roller thrust bearings as well as journal bearings. Gains ranging from 10 to 15 watts were observed with the various samples. There were, however, serious problems. It was difficult to obtain the correct bearing clearances. Oil supply to the crankpin and wrist pin tended to be scanty. The samples were initially, or soon became, excessively noisy. Several samples which were put

*Patent application filed by CPC.

on life test failed due to severe wear.

As there appeared to be no practical way to overcome all of these difficulties, needle bearings were not included in the high efficiency design.

5.5.2 Oil Stirrer Experiments

Present T-line compressors manufactured at CPC include a device on the lower end of the crankshaft which causes agitation and a certain amount of foaming of lubrication oil during operation. The main purpose of the oil stirrer is to reduce noise transmission to the compressor shell, as the agitated oil conducts sound energy less efficiently than it would in an undisturbed state.

The original oil stirrer design, used in both refrigerator-freezer and RAC compressors, was the finger type, as shown in Fig. 5-12. This type is still used in RAC models, but was superseded several years by a conical stirrer made of perforated metal, pictured as item 1 in the lower part of Fig. 5-7 (page 47).

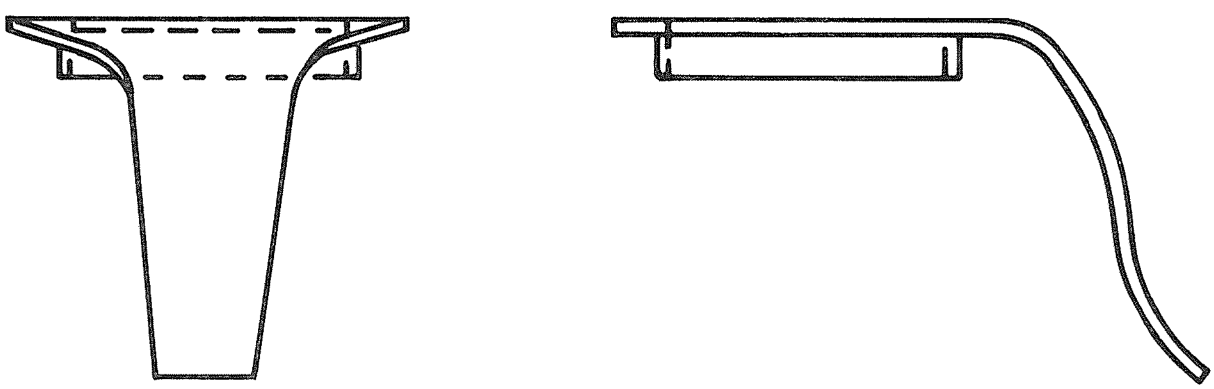


Figure 5-12 Finger Type Oil Stirrer

Calorimeter tests were conducted to determine the potential power savings made possible by elimination or change in design of the oil stirrer.

The initial, and potentially most significant change was to eliminate the oil stirrer entirely. On the four-pole compressors this resulted in a savings of about 3 watts and on the two-pole samples, about 6 watts. The noise level of the two-pole samples increased by 8 dBA when the stirrer was removed and the four pole samples increased by about half that amount. It was therefore concluded that an oil stirrer was definitely required because of noise considerations. Experimentation was continued in an effort to develop an oil stirrer which would be effective in reducing noise and utilize the minimum possible power.

Additional calorimeter and noise testing showed that the perforated oil stirrer was extremely effective in reducing noise. By changing from the finger style oil stirrer to the perforated cone the noise generated by the compressor sample was decreased by 4 dBA. This noise reduction was consistent on both two and four-pole motor samples. Our tests showed that the use of the perforated oil stirrer also resulted in an increase in power consumption of approximately 3 watts.

In an attempt to achieve a noise reduction with a minimum power loss, a conical thermoplastic oil stirrer of the type shown in Fig. 5-7, item 2 was constructed. This oil stirrer was similar in form to the perforated metal style but had no small holes to increase drag in the oil sump. A steel eyelet was molded in place so that the stirrer could be pressed on the crankshaft. Calorimeter tests of sample compressors with this new oil stirrer showed an improvement of about 3 watts compared to perforated type at only a 0.5 dBA noise increase. An incidental advantage is that the injection molded thermoplastic stirrer can be made at a reduced cost as compared to the perforated style.

CPC is planning to use the unperforated thermoplastic oil stirrer in the high efficiency design and started to introduce it in regular production about the middle of 1980. For the four-pole efficient design, the change will result in a gain of approximately 5 watts compared to the basic T52 two-pole model with the perforated stirrer.

6.0 PROTOTYPE DESIGN CONSIDERATIONS

The recommended prototype design for the high efficiency compressor was developed by following the strategy of making evolutionary changes in the existing CPC models.

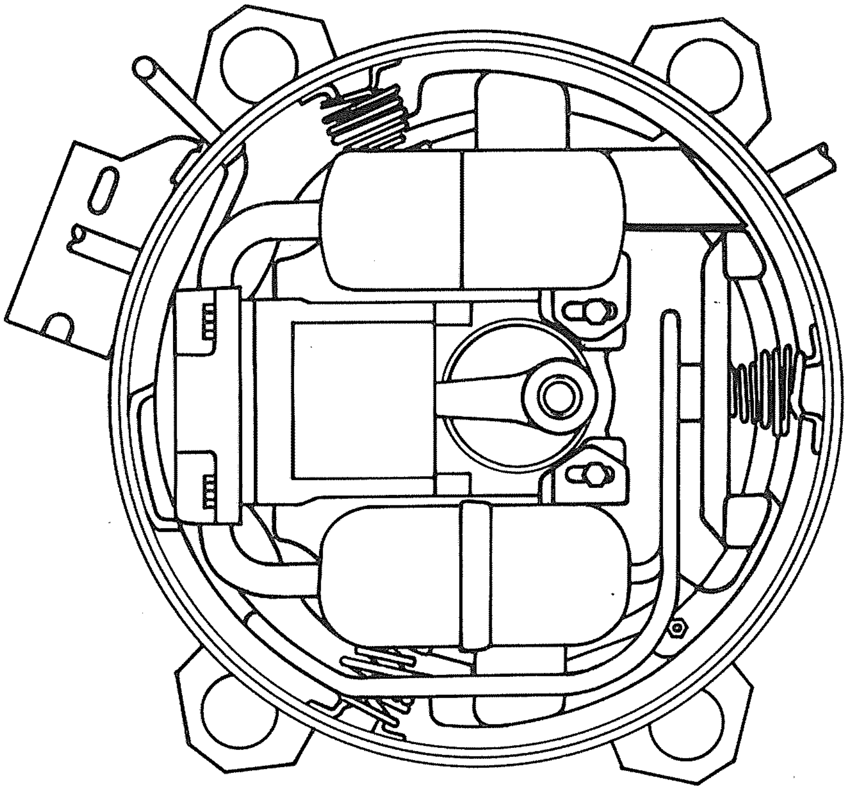
6.1 Two-Bearing Vs. Single-Bearing Designs

In the early stages of development program two separate types of design were investigated. One, like the present CPC design, was of the two-bearing type, with an upper bearing located in the cylinder housing and the lower bearing in the motor housing. An outline of this version is shown in Fig. 6-1. The other was a single-bearing type, as shown in Fig. 6-2. Preliminary prototype samples of the two-bearing design were built with four-pole motors, while single-bearing samples contained improved two-pole motors.

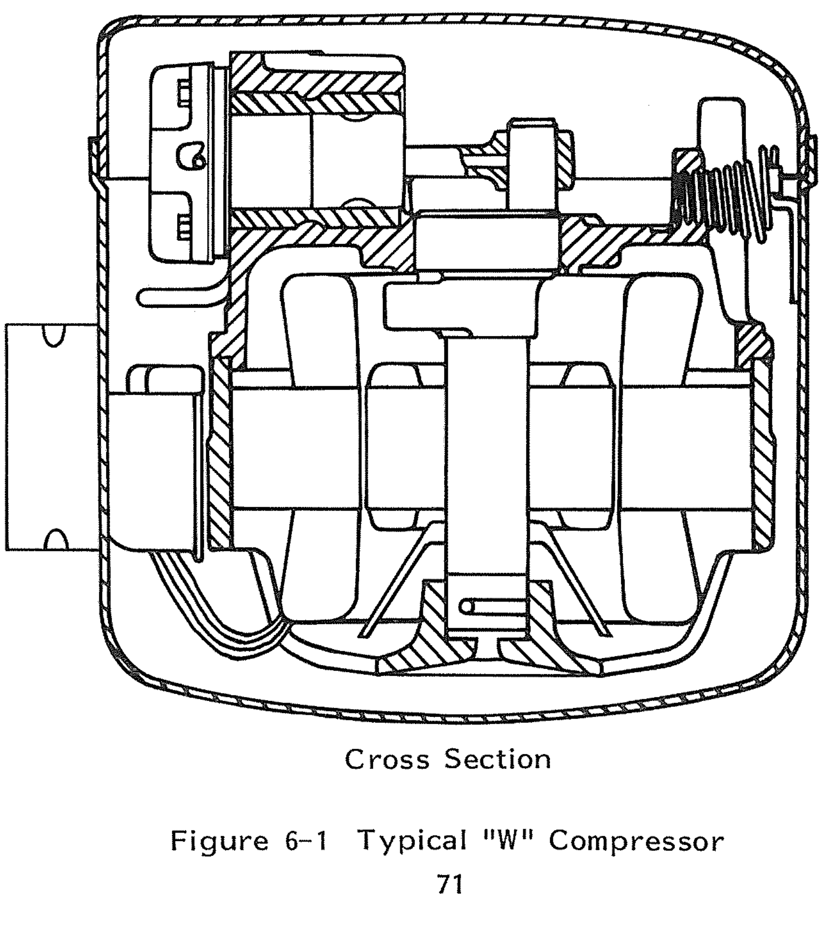
In the course of calorimeter testing it was found that the single-bearing was not superior in mechanical efficiency to the present two-bearing version. Since production of a single-bearing design, for field demonstration and subsequent quantity production, would involve major facilities expenses, the two-bearing version, with a four-pole motor, was chosen for the final prototype design. Models of this design are designated "W".

The single bearing compressor prototype which was designed, constructed, and tested on a "low level of effort" basis, was designated "SB". In order to obtain test data for comparison with the "W" prototypes, "SB" samples utilizing two-pole, low torque PSC motors of the highest available efficiency were built and calorimeter tested. Typical results were as follows:

BTU per HR	800
Watts	172
EER, BTU per WHR	4.65

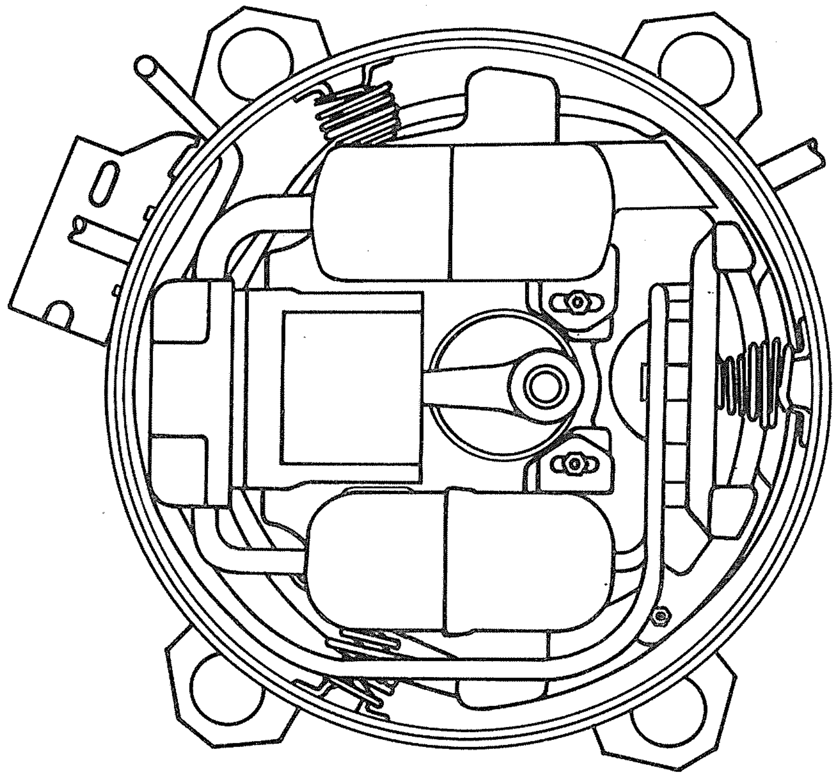


Top View With
End Head Removed

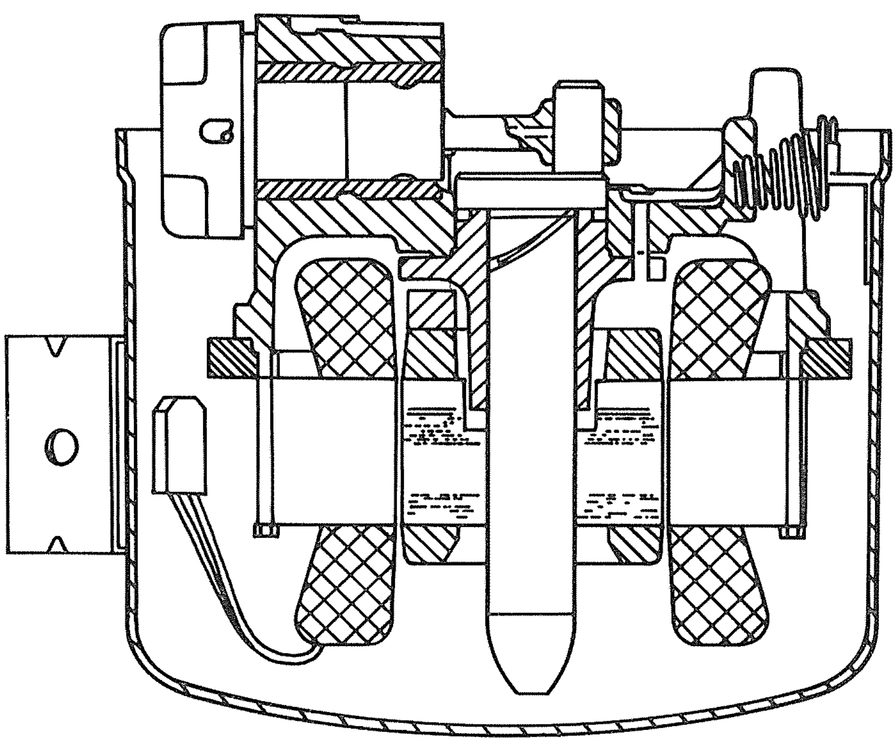


Cross Section

Figure 6-1 Typical "W" Compressor



Top View



Cross Section

Figure 6-2 Single Bearing (S.B.) Compressor

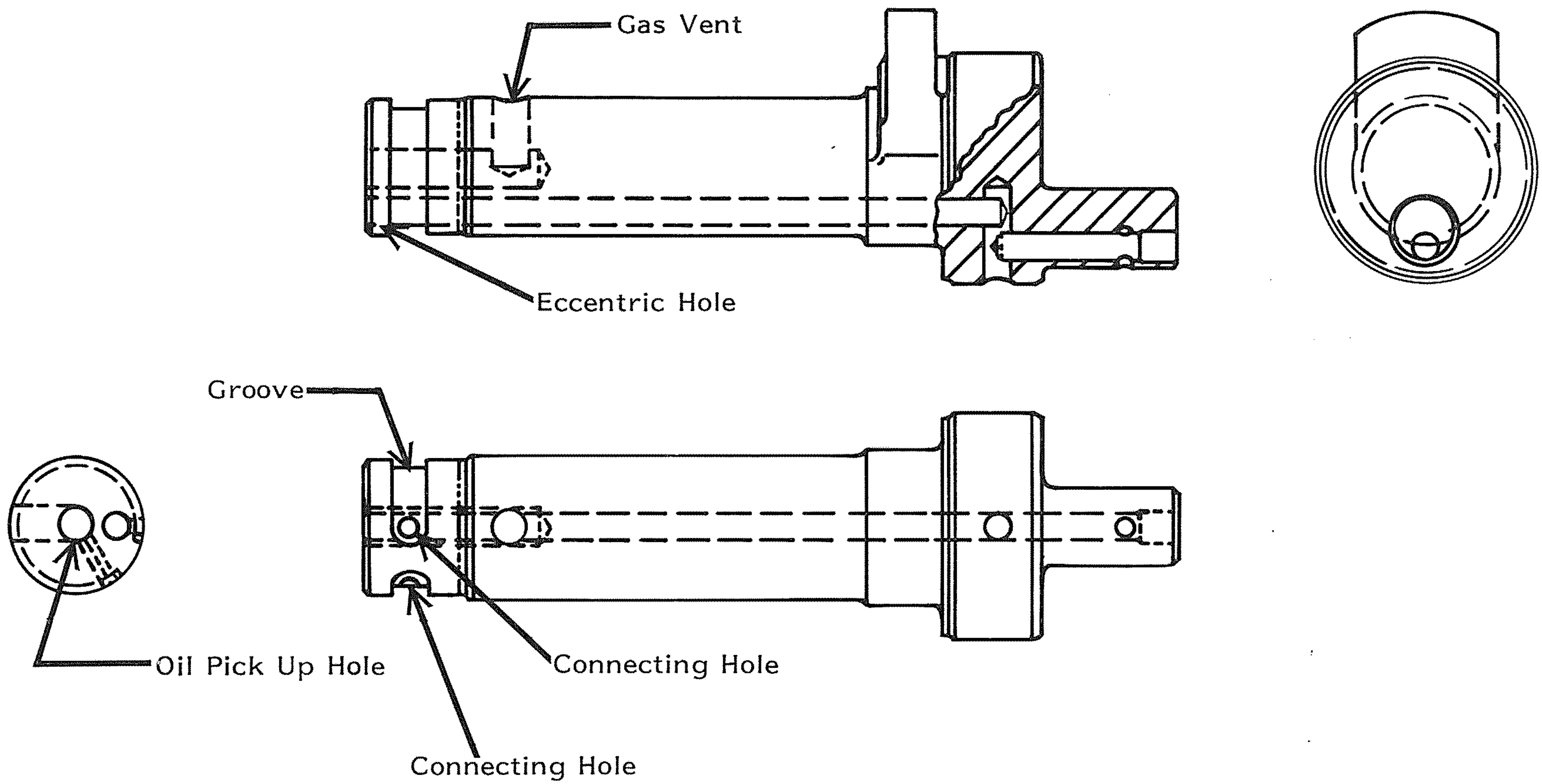


Figure 6-3 Four Pole Crank Shaft

While these samples are much higher in efficiency than the baseline design, they are not as efficient as the four-pole prototypes.

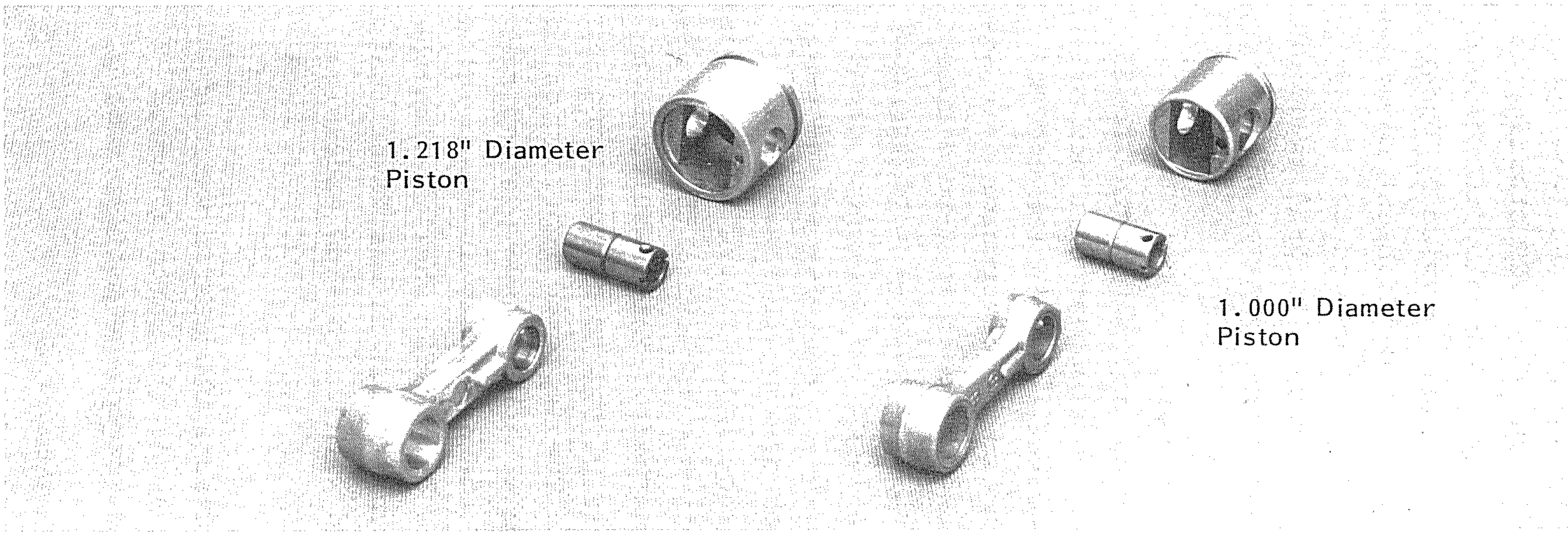
The remainder of this report will relate only to the two-bearing, four-pole models.

6.2 Final Prototype Design Details

Table 5-A (page 30) lists the features which were considered for incorporation in the prototype design. Evaluation of the individual features is described in Sec. 5. Additional considerations involved in the application of these features to an integrated prototype design are discussed in the following paragraphs.

The most significant change as compared to the CPC "T" line production compressor is the adoption of a four-pole permanent split capacitor motor. The visual differences between a four-pole and a two-pole motor can be seen on Fig. 5-5 (page 37). The theoretical differences between two and four-pole motors was discussed in Sec. 5.1.1. The efficiency curves for two types of four-pole motor curves are shown on Fig. 5-4 (page 36). The use of four-pole motors, while not a new idea, resulted in compressor efficiency increases even though the motor efficiency was slightly lower than that of an equivalent two-pole motor. Referring to Fig. 5-4, the motor identified as P5706, which is a reduced torque PSC four-pole will be utilized in the pilot lot field test samples to be constructed during Phase II of this program.

The method of overcoming the inherent low starting torque of a PSC motor by using a PTCR connected in parallel with the capacitor has been previously discussed in Section 5.1.4. It is, however, useful to consider several additional factors related to the use of a PTCR with a PSC motor.



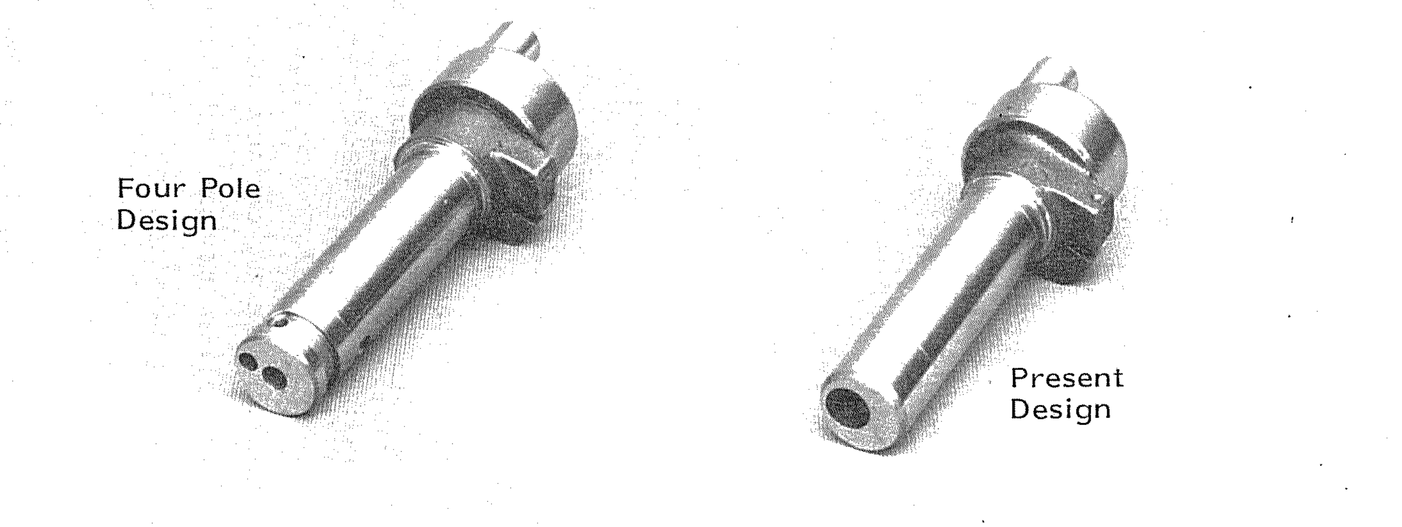


Figure 6-4 Upper: Piston and Connecting Rod Details
Lower: Crankshafts

In operation, the PTCR must draw power so that it can maintain high resistance. The PTCR starting devices* now on the market draw about 5 watts of power during operation, making them less energy efficient than the conventional relay. PTCR starting devices which only use 2 watts of power are soon to be introduced by the manufacturer and if approved for field service by Underwriters Lab, they will be utilized in the CPC efficient compressor run. The physical differences between the PTCR and the relay which is utilized on RSIR motors is shown on Fig. 5-2* (page 32) PSC motor operation, of course, requires the addition of a capacitor to the system.

The use of a four-pole motor, which runs at one half the speed of the two-pole, has necessitated the development of a new oil pump.** The type of pump which is currently used in the CPC "T" line compressor would not supply oil*** to the compressor bearings at an adequate rate when the compressor runs at four-pole speed. Both the new oil pump and the current type form part of the compressor crankshaft. Functional details of the four-pole crankshaft appear in the drawing in Fig. 6-3. Pictures of the new crankshaft and the present production design are shown in the lower part of Fig. 6-4. Another change incorporated in the prototype design to improve lubrication involves closing the oil spray orifice in the upper end of the crankshaft. This results in a more copious and more positive supply of oil to the rotating bearings and to the cylinder walls. The improved lubrication is responsible for the good performance with ring-type pistons reported in Sec.5.2.5 The function of the oil spray in the original "T" design was to convey heat to the compressor shell. With the more efficient motors in the prototype design it was found that the cooling feature could be omitted without causing excessive temperatures.

The recommended prototype design reflects changes in various component parts of the compressor. Among the parts affected are the suction muffler, oil stirrer, discharge muffler tubing,

* 4EA PTCR Start Device Texas Instruments, Inc.
**Patent application filed by CPC.
***Texaco Capella B refrigeration oil

and valve plate. The nature of the changes to these items is described in Sec. 5, while the amount of the resulting improvement is listed there and on Table 5-A, (page 30). Sec. 5 also includes a discussion of other changes which were investigated but were not adopted.

The flow of refrigerant gas through the CPC efficient compressor can be understood by referring to Fig. 6-5. It will be seen that the cool suction gas is directed into the suction muffler intake as rapidly as possible to reduce super-heating. Once through the muffler, the gas travels into the cylinder through the two suction ports and on the next piston up stroke exits through the enlarged discharge ports.

As was noted previously, the use of a four-pole motor has necessitated the design of a new oil pump. The travel of lubricating oil through the compressor can be seen on Fig. 6-6. The oil is drawn up from the sump through the crankshaft and on to the various bearing and journal interfaces.

The electrical components are shown on Fig. 6-7. The use of a PSC motor requires that a capacitator be mounted external to the compressor shell. The PTCR start device will be mounted adjacent to the motor protector.

6.2.1 Manufacturing Considerations

A detailed plan for the manufacture of a limited quantity (about 1,500) of the highly efficient compressors can be found in Sec. 13 of this report. The manufacture and commercial sale of a much larger quantity, after the design has been field tested, is planned. The actual quantity to be produced will depend on market acceptance. The efficient compressor can be produced, for the most part, on already existing CPC capital equipment. The new crankshaft and four-pole motor do, however, require some new equipment.

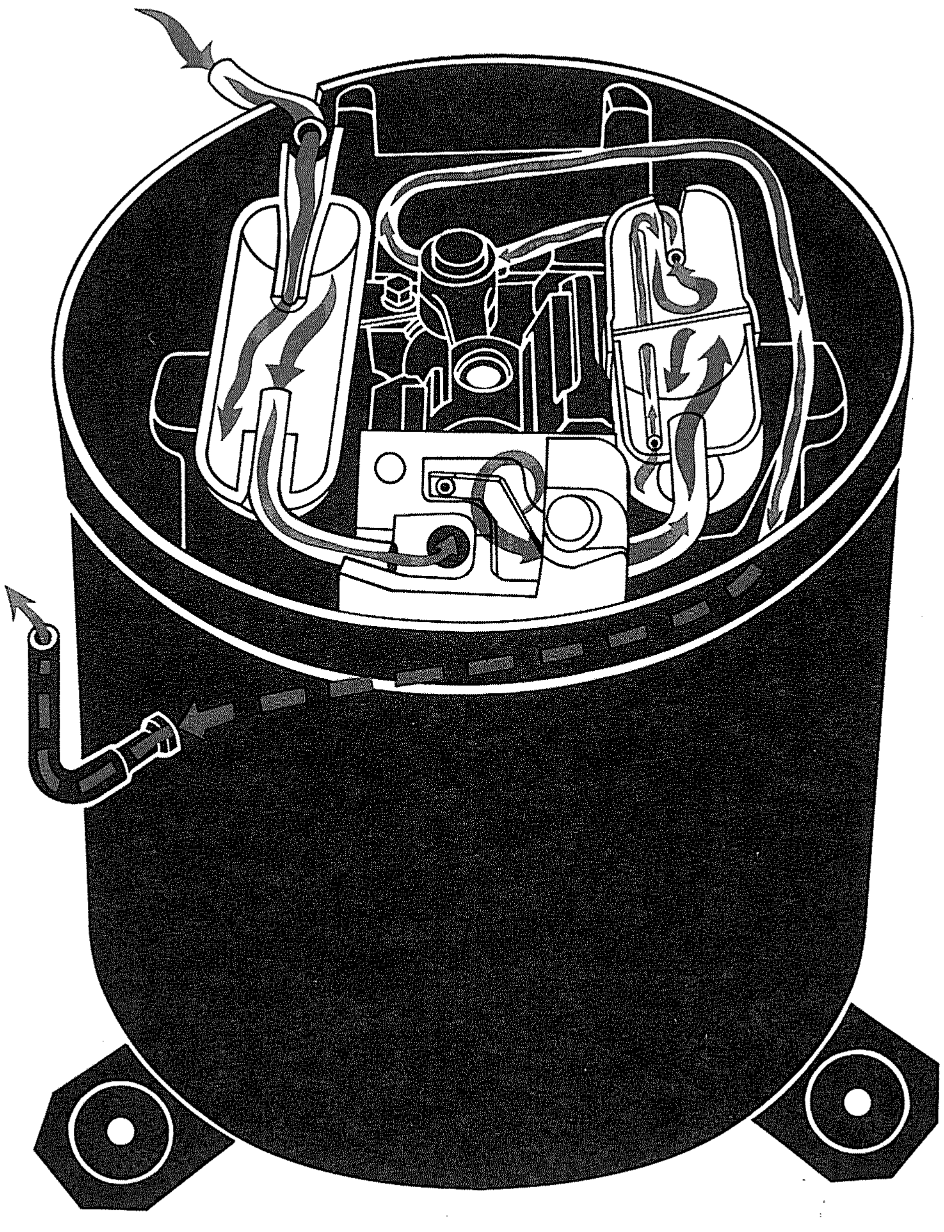


Figure 6-5 Path of Gas Flow Through Compressor

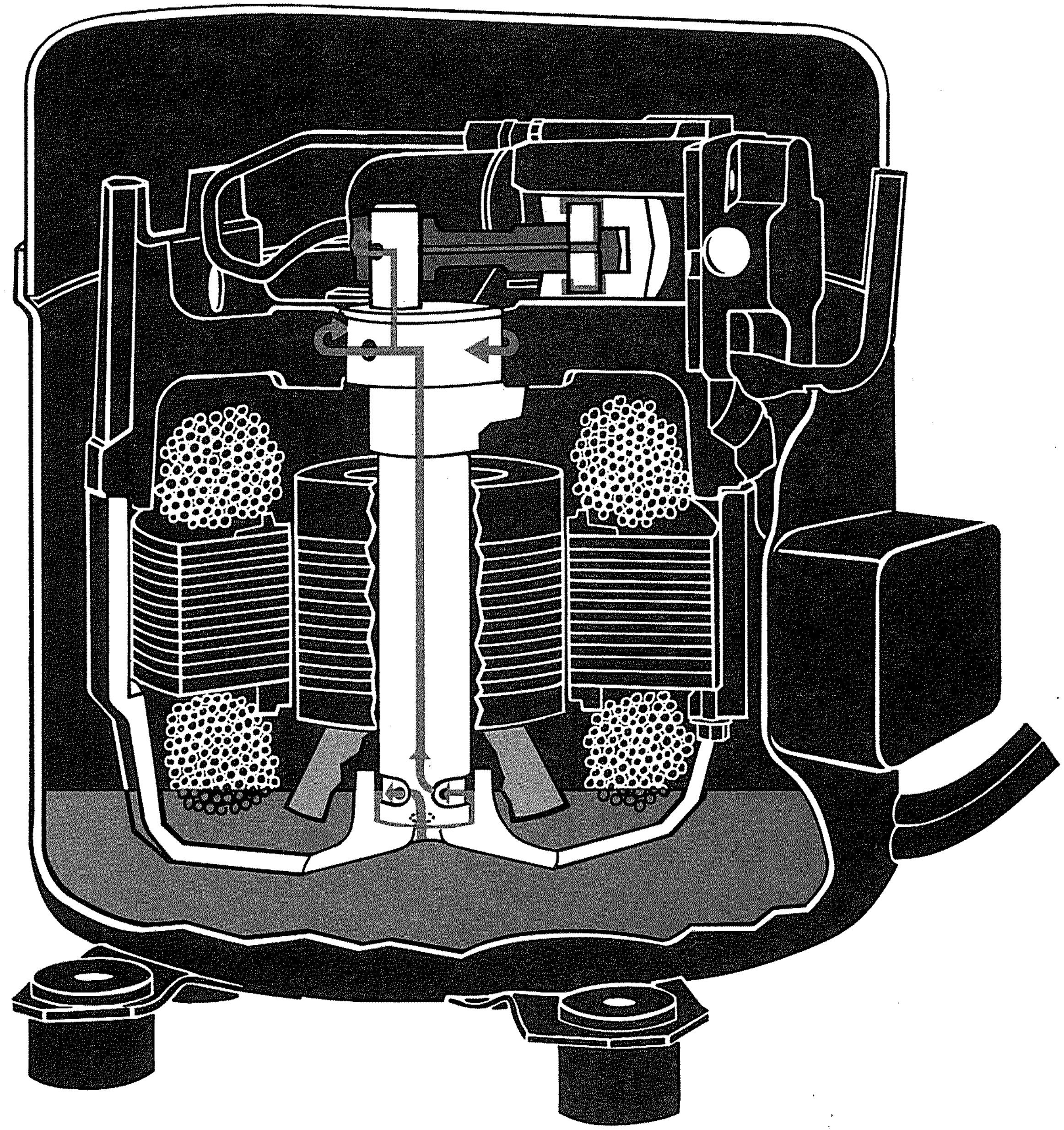


Figure 6-6 Oil Flow Through Compressor

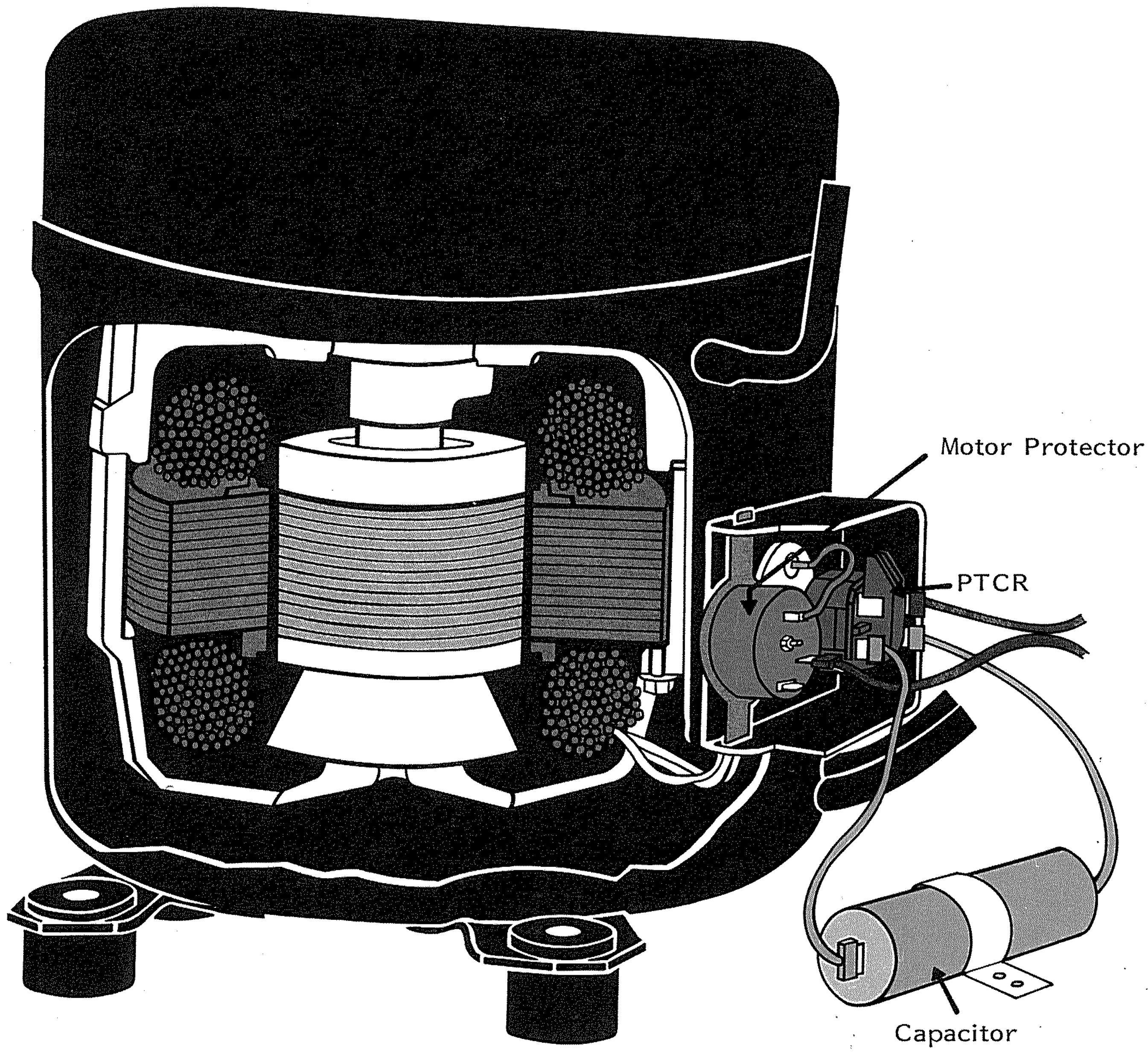


Figure 6-7 Electrical Components of PSC Compressor

The crankshaft is made from the existing casting and can be partially finished on existing grinding equipment. The annular groove and radial holes in the shank can be machined on manual or semi-automated equipment. A special drill is required for the long oil hole in the crankshaft shank. A comparison of the CPC production crankshaft and the new design was shown in Fig. 6-4.

6.2.2 Capacity Size Line Up

Most of the experimental work has been carried out on the 800 BTU/HR. capacity size. When these efficient compressors are manufactured in large quantities, other capacity sizes will be built. Generally as the displacement and therefore the capacity increases, the EER increases. This is because the frictional losses tend to remain constant and the bore stroke relationship is more favorable from a reexpansion standpoint. This later effect is offset by the need to increase the bore diameter in the four-pole compressors for capacity sizes larger than 800 BTU/HR. Since each capacity size requires a different motor design, there may be some variation in the efficiency level attained by the motors.

Experimental four-pole compressors have been built in 500, 600, 800, 1000 and 1200 BTU/HR. sizes. Building and testing numerous capacity sizes aided in the motor design refinements. In the four-pole (1750 RPM) design, as was noted, a capacity size greater than 800 BTU/HR. requires a bore larger than the standard CPC 1.000" diameter. The 1000 and 1200 BTU per hour compressors have been constructed with a 1.218" diameter bore machined and honed into a special sleeve cast in the cylinder housing. Pistons of 1.000" and 1.218" diameter with their associated wrist pins and connecting rods, are pictured in the upper part of Fig. 6-4. The larger diameter necessitates the use of different (but similar) suction valves. The following table lists some of the bore and stroke combinations that were tested during experimentation with various capacity ratings.

SELECTED TEST RESULTS FROM VARIOUS FOUR-POLE BORE/STROKE COMBINATIONS

TABLE 6-A

	Bore	Stroke	Displ. In^3	BTU	EER	Motor
W120	1.218	.499	1.16	1199	3.76	RSIR Motor
W100	1.218	.410	.95	985	4.13	RSIR Motor
W80	1.000	.499	.78	814	4.88	PSC Motors
W60	1.000	.410	.65	590	4.65	PSC Motor
W50	1.000	.375	.59	508	4.33	PSC Motor

The samples did not include all of the improvements used in the recommended prototype design. For example, some of the tests were made with RSIR motors. As a result the EER values were lower than those found in later tests. The test data does, however verify the feasibility of obtaining the desired range of capacities. For test results on later samples of the models listed (except W50) refer to Table 7-A (page 87).

The refinement of a complete capacity line up will take place during Phase II and is discussed further in Sec. 13 of this report.

7.0 PROTOTYPE PERFORMANCE TESTING

Most of the compressor performance tests reported in the preceding sections were conducted to evaluate proposed features or intermediate combinations. When the final high efficiency design was established, prototype samples were built, and were subjected to both compressor performance tests and system tests. The test results are reported in the following sections.

7.1 Compressor Performance Tests

A detailed discussion of CPC compressor performance tests and equipment may be found in Sec. 4.

7.1.1 Prototype Calorimeter Testing

Calorimeter testing has been the basis for many of the design decisions made throughout the development program. There is some variation of results in a repetitive test series because of wearing in of the compressor bearings, changing barometric pressure, or inherent variability in reading meters and pressure gages. Numerous tests have been run on the various high efficiency four-pole W-line compressors. Average performance figures are listed in Table 7-A below.

TABLE 7-A

Average Calorimeter Test Results on W-Line Samples

Compressor Model	Capacity BTU/HR	Input Watts	EER BTU/WHR
W60	644.4	129.7	4.97
W80	828.8	164.7	5.03
W100	1068.3	215.0	4.97
W120	1290.5	261.0	4.93

The results on the W80 are more reliable than those on the other models because larger numbers have been tested. An analysis of the probable range of distribution of calorimeter test results may be found in Appendix E. The W80 samples selected for test on complete refrigerators, as described in Sec. 7.3, had an EER of 5.00 or higher.

Curve tests, at a condensing temperature of 130.0°F were performed on prototype samples of models W60, W80, W100, and W120. The resulting curves are shown on Fig. 7-1 through 7-4.

7.1.2 Pullout Testing

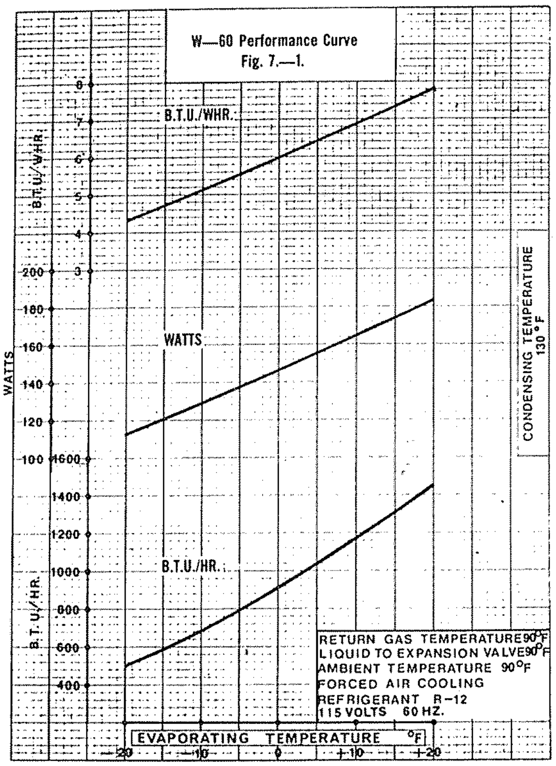
A description of the pullout tests can be found in Sec. 4.3. Historical experience at CPC indicates that a compressor with a pullout of 98 volts or less will function under most systems conditions. Actual verification of system suitability is ultimately required. The W-line compressors and their respective pullout voltages are shown as follows:

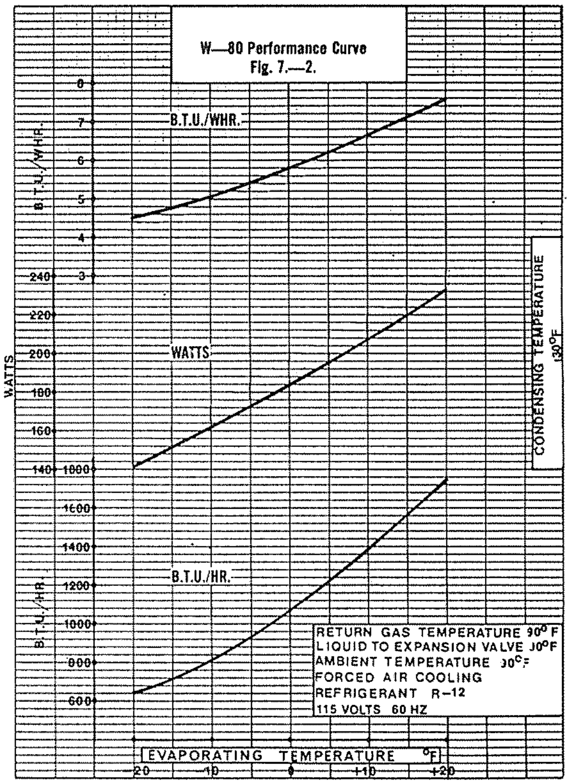
Compressor	Pullout Voltage
W60	96.7
W80	99.1
W100	101.1
W120	95.5

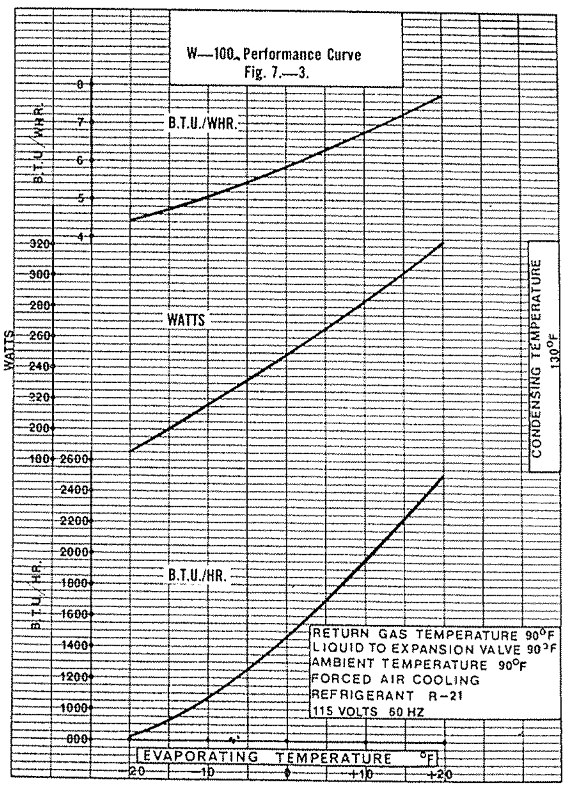
Although two of the models exceed the 98 volt limit, they were found to perform successfully on system tests.

7.1.3 Start Testing

Start tests, which are described in Sec. 7.1.3, are performed to indicate if a compressor will have ample starting torque in a system application. There is no single maximum allowable voltage requirement for all compressors. The maximum allowable starting


W—60 Performance Curve
Fig. 7.—1.
B.T.U./WHR.
WATTS
B.T.U./HR.
CONDENSING TEMPERATURE 130 °F
RETURN GAS TEMPERATURE 90°F
LIQUID TO EXPANSION VALVE 90°F
AMBIENT TEMPERATURE 90°F
FORCED AIR COOLING
REFRIGERANT R-12
115 VOLTS 60 HZ.
EVAPORATING TEMPERATURE °F
-20
-10
0
+10
+20


W—80 Performance Curve
Fig. 7.—2.
B.T.U./WHR.
WATTS
B.T.U./HR.
CONDENSING TEMPERATURE 130°F
RETURN GAS TEMPERATURE 90°F
LIQUID TO EXPANSION VALVE
AMBIENT TEMPERATURE
FORCED AIR COOLING
REFRIGERANT R-12
115 VOLTS 60 HZ
EVAPORATING TEMPERATURE °F
20
10
0
+10
+20


W—100 Performance Curve
Fig. 7.—3.
B.T.U./WHR.
WATTS
B.T.U./HR.
CONDENSING TEMPERATURE 130°F
RETURN GAS TEMPERATURE 90°F
LIQUID TO EXPANSION VALVE
AMBIENT TEMPERATURE 90°F
FORCED AIR COOLING
REFRIGERANT R-21
115 VOLTS 60 HZ
EVAPORATING TEMPERATURE °F
-20
-10
0
+10
+20

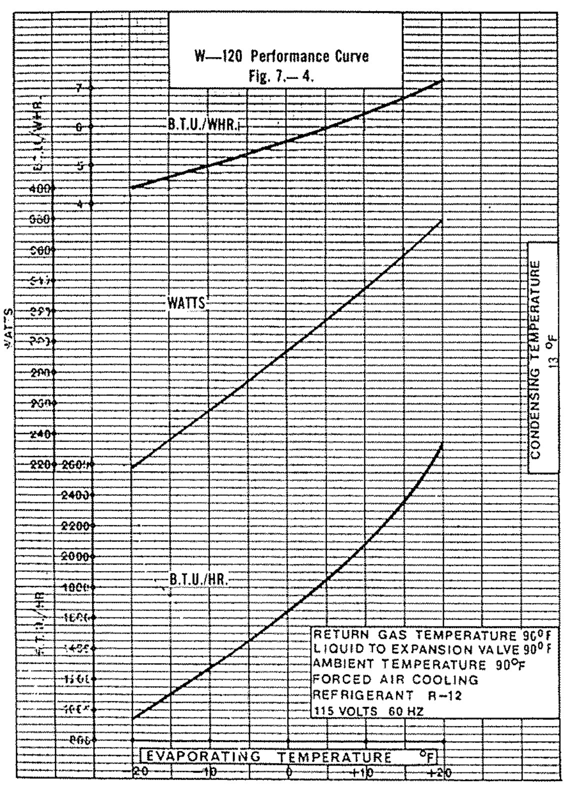

W—120 Performance Curve
Fig. 7.— 4.
B.T.U./WHR.
WATTS
B.T.U./HR.
CONDENSING TEMPERATURE
RETURN GAS TEMPERATURE
LIQUID TO EXPANSION VALVE
AMBIENT TEMPERATURE 90°F
FORCED AIR COOLING
REFRIGERANT R-12
115 VOLTS 60 HZ
EVAPORATING TEMPERATURE °F
20
10
0
+10
+20

Figure 7-1

voltage varies for each compressor, depending on the locked rotor current value for that model. Results of the locked rotor test are in the table below.

Compressor	Start Voltage	Maximum Allowable Start Voltage
W60	77	85.5
W80	70	84.0
W100	79	82.5
W120	80	81.0

As can be seen in the table, all models were under the maximum allowable start voltage.

7.2 Prototype Life Testing

The compressor life test procedures used by CPC are described in Paragraph 4.7.

In order to evaluate the endurance properties of certain proposed features of the prototype designs, 16 samples were installed on continuous-running test stations to be subjected to a test of 3000 hours duration, with a nominal discharge pressure of 385 psig and a nominal suction pressure of 10 psig.

In two of the samples, the crankshaft was supported by roller-type (needle) bearings. The first of these samples failed due to excessive wear of the needle bearings and crankshaft after 2000 hours of operation. The second completed the 3000 test run, but on examination at the end of the test the bearings were found to be badly worn.

On the basis of these results, and earlier disappointing performance tests, no further consideration will be given to needle bearings in the current program.

Thirteen samples of the four-pole motor configuration have completed the 3000 hour test. In addition, a single-bearing, two-pole sample, designated SB52, representing a proposed alternate prototype configuration, has completed the same life test. The nominal capacity in Btu/Hr. of these compressors is listed below:

Designation	Nominal Capacity BTU/HR	Number of Samples
W50	500	2
W60	600	2
W80	800	7
W100	1000	1
W120	1200	1
SB52	800	1

Aside from the roller-bearing specimens referred to above, the samples which have completed the 3000 hour run were all functioning normally at the end of the test. The samples were calorimeter tested before and after the 3000 hour run to check the effect of the test on performance. The results are shown on Table 7-B. None of the samples showed any drastic change, and the average of the whole group does not show any significant trend. The samples were put on test before prototype development work was completed, and therefore they do not include all the refinements found in the final recommended design. As a result, the calorimeter test results are inferior to those obtained later in the program.

After completion of the 3000 hour run and calorimeter testing, the hermetic compressor shells were cut open and the internal parts were examined 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

TABLE 7-B

CALORIMETER TEST PERFORMANCE ON
LIFE TEST SAMPLES BEFORE AND AFTER
3000 HOUR ENDURANCE TEST

MODEL	SAMPLE NUMBER	RELATION TO LIFE TEST	CAPACITY BTU/HR	WATTS	EER BTU/WHR
W50	719-1B	Before After	526 523	116.4 118.4	4.52 4.42
W50	719-2B	Before After	489 465.3	118.0 111.8	4.14 4.16
W60	815-1B	Before After	566 580.8	120.8 127.6	4.69 4.55
W60	815-2B	Before After	610 614.3	128.4 130.5	4.75 4.71
W80	713-1C	Before After	731 755.2	156.8 158.6	4.66 4.76
W80	713-3C	Before After	767 791.1	164.8 161.7	4.65 4.89
W80	713-4A	Before After	721.9 762.5	168.4 167.4	4.28 4.55
W80	925-1	Before After	754.8 756.0	160.8 167.8	4.69 4.51
W80	925-2	Before After	769.4 767.0	157.7 167.6	4.88 4.58
W80	925-6	Before After	743.9 777.2	163.3 170.5	4.55 4.56
W80	925-8	Before After	755.0 754.5	162.6 171.2	4.64 4.41
W100	841-B	Before After	906 926.9	238 228.9	3.80 4.05
W120	753-C	Before After	1080 1140.2	279.6 315.2	3.86 3.62

TABLE 7-C

PROTOTYPE PART ANALYSIS

(FOLLOWING 3000 HOUR LIFE TEST)

FEATURES OF SAMPLE	W50 719-1B	W50 719-2B	W60 815-1B	W60 815-2B	W80 713-1C	W80 713-3A	W80 713-4A	W100 841-B	W120 753-C
Piston Type	Plug	Plug	Plug	Plug	Plug	Plug	Plug	Ring Type	Plug
Oil Stirrer	Plastic	Plastic	Plastic	Plastic	None	None	RAC Type	RAC Type	Plastic
Suction Muffler	"	"	"	"	Plastic	Plastic	Plastic	Plastic	"
Components Examined									
Piston & Cyl. Bore	Very Light Wear	Normal Wear	Very Light Wear	No Wear	Very Light Wear	Slight Wear	Very Light Wear	Normal Wear	More Than Normal Wear
Wrist Pin & Brg.	Normal Wear	Wear Little Above Normal	Normal Wear	Wear Little Above Normal	More Than Normal Wear	Heavy Wear	Slight Wear	Heavy Discoloration	Normal Wear
Crankpin & Rod Brg.	Normal Wear	Light Wear	Normal Wear	No Wear	Slight Wear	Normal Wear	More Than Normal Wear	Normal Wear	Normal Wear
Crankshaft Main Brgs.	" "	" "	Some Galling	" "	Slight Scratching	Scratches	Light Scratches	Light Scratches	Very Light Scratches
Thrust Brg.	" "	More Than Normal Wear	Normal Wear	No Wear	No Wear	"	No Wear	No Wear	No Wear
Valves & Plate	Normal Coking	Light Coking	Normal Coking	Very Light Coking	Normal Coking	Heavy Coking	Very Light Coking	Slight Coking	Normal Coking
Suction Muffler	No Change	No Change	No Change	No Change	No Change	No Change	No Change	No Change	No Change
Oil Stirrer	" "	" "	" "	" "	" "	" "	" "	" "	" "
Misc. Comments		Light Scratches On Brgs.	Galling on Shaft Due to Foreign Particles	Scratches in Crank-pin Brg.		Evidence of Excessive Temp. Perhaps due to poor oil Circulation		Evidence of Slight Discharge Valve Leak	Piston Wear on One Side only.

most likely to show evidence of wear or degradation. The examiner's judgment on the condition of the parts is largely subjective. There are no rigid standards for the appearance of any of the parts. In effect, the samples are judged to have survived the test successfully if the parts show no more evidence of wear, discoloration or deformation than parts of a design known to be satisfactory do, after being subjected to the same test.

On this basis, these preliminary prototype samples passed the test. The parts, following the test, were found to be similar in appearance to parts of current designs which have demonstrated good endurance in regular service in the field.

The major findings derived from the examination of the parts after the 3000 hour test are listed in detail for eleven individual samples in Table 7-C. The descriptions given are relative to the general average of previous tests on the current production design, which is assumed to be normal. It may be noted that a few cases of more than normal wear or discoloration are reported. In none of these was the condition severe enough to lead to an expectation of premature failure in service.

A cycling endurance test on additional samples is still in progress. The results will be reported later. It is generally believed that the cycling test gives a more reliable indication of service durability than the continuous running test reported above.

The successful completion of these life tests represent the first steps in the reliability verification of the high efficiency design. Subsequent steps involve factory life testing of larger quantities, and field demonstration testing, which are included in the Phase II program described in Sec. 13.

7.3 System Testing

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 improvements made. The main reason for this is that on the calorimeter the compressor operates in a steady state at standardized conditions whereas in a system, conditions change constantly as the compressor cycles on and off.

The system tests used to measure the improvement in energy consumption with the prototype designs were made per AHAM Std. No. HRF-2ECFT. Most of the tests were made in RT18 refrigerators, which have fresh and frozen food storage, a dual contol system, and a switchable antisweat heater. The final result obtained in the test is the average power consumption in KWH per 24 hr. at a 5°F freezer temperature. To see how this result is obtained, refer to Fig. 7-5 as a typical example, and note that power consumption and freezer temperatures are first measured, with the antisweat heater off, at four points indicated by the following nomenclature:

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

Lines are drawn to connect pairs of test points. The average KWH/24 HR. values corresponding to the intersection of these lines with a line at a freezer temperature of 5°F is taken as the standarized power consumption. The test is then repeated with the antisweat heater on. The average of the figures for heater off and heater on is a reasonable approximation of the rate of usage for a year. The results of the system test are usually given in terms of KWH per unit of time (day, month, or year). Since the dimensions of the compressor EER, in BTU per WHR, are the inverse of the system performance dimensions, percentage improvements cannot be compared directly.

7.3.1 Test Results

The principal compressor and system combinations tested are listed below, together with the power consumption in KWH/24 HR. for each one, and a reference to the number of the figure on which the test data is plotted.

(a) A T52 compressor in an RT18 refrigerator of the original production design. This represents the performance baseline in effect at the beginning of the DOE compressor improvement project. Average KWH/24 HR. -- 4.30. Fig. 7-2

(b) A T80 compressor in an improved RT18 cabinet. This was put into production late in 1979, and represents the results of the ongoing improvement in products at CPC. Average KWH/24 HR. -- 3.51. Fig. 7-3

(c) A W80 compressor in the original RT18 cabinet, reflecting improvements obtainable through the use of the prototype compressor design along. Average KWH/24 HR. -- 3.33. Fig. 7-4

(d) A W80 compressor in the improved RT18 cabinet. Average KWH/24 HR. -- 2.89. Fig. 7-5

In explanation of the above combinations, the T80 compressor is similar to the original baseline T52 but has a more efficient motor and certain minor improvements. The improved RT18 cabinet has superior insulation, and more efficient fan motors for evaporator and condenser air circulation.

Tests were also made on several supplementary combinations, listed below:

(e) A T52 compressor (a different sample from (a) above) the original RT18 cabinet. KWH/24 HR. -- 3.96 (antisweat heater off). No diagram shown.

(f) An SB80 sample in the same cabinet as (e). The SB80 is the two-pole, single-bearing prototype. KWH/24 HR.-- 3.38 (antisweat heater off). Fig. 7-6.

Combinations (e) and (f) were tested only with the antisweat heater off. The results therefore cannot be compared directly with the average results of tests on (a) through (d).

(g) An MLO-90 compressor in an improved RT18 cabinet. The MLO-90 compressor represents a model currently being produced by Americold, another division of W.C.I. While this test was not directly related to compressor efficiency improvement, it did show the relative improvement of the W80 prototype over compressors currently available. Average KWH/24 HR. -- 4.01. Fig. 7.7.

(h) A T52 compressor sample in a RT12 cabinet, a 12 cu. ft. design which has a single control and incorporates an antisweat heater which cannot be switched off. KWH/24 HR. -- 2.80. Fig. 7.8.

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

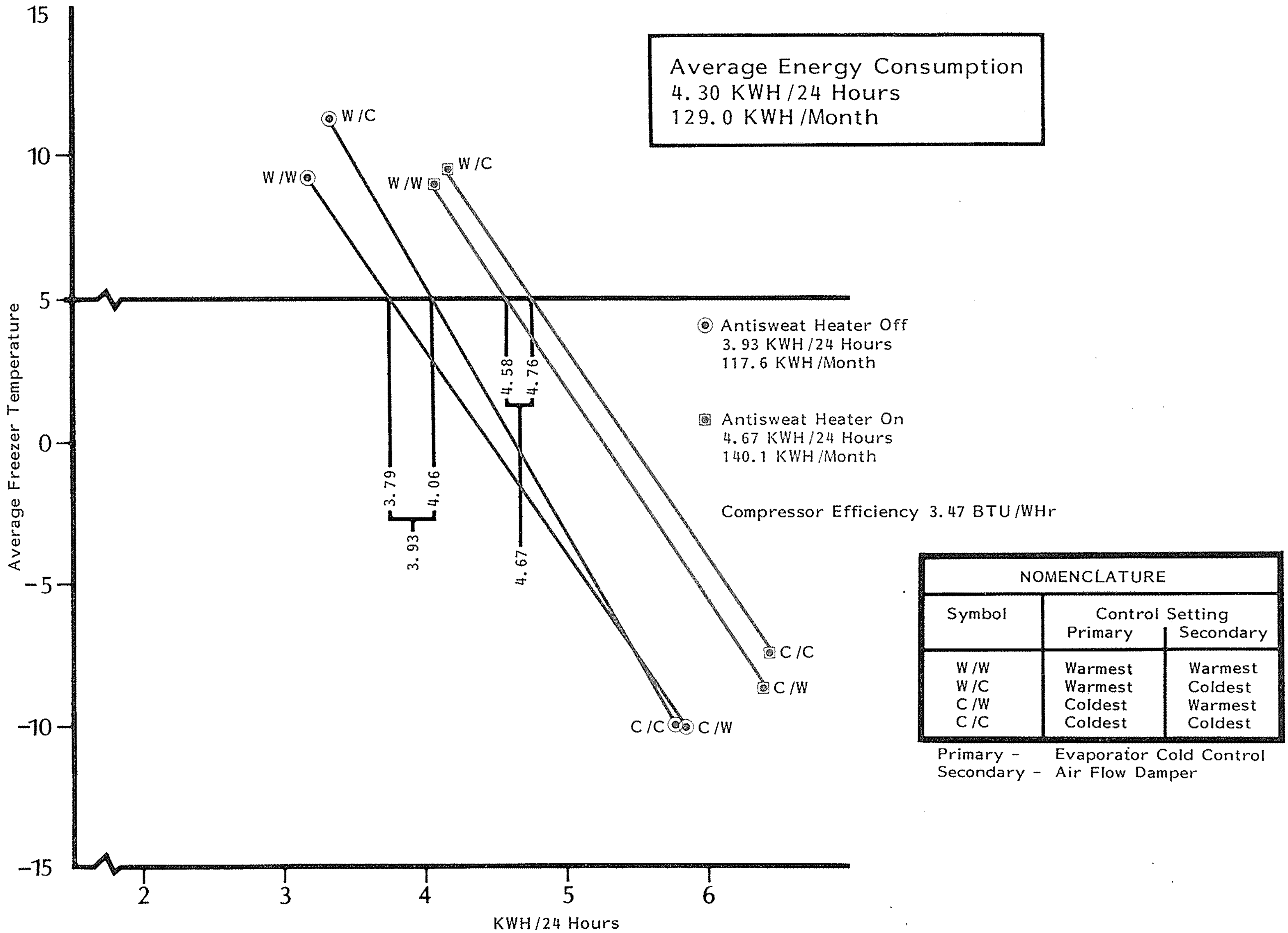
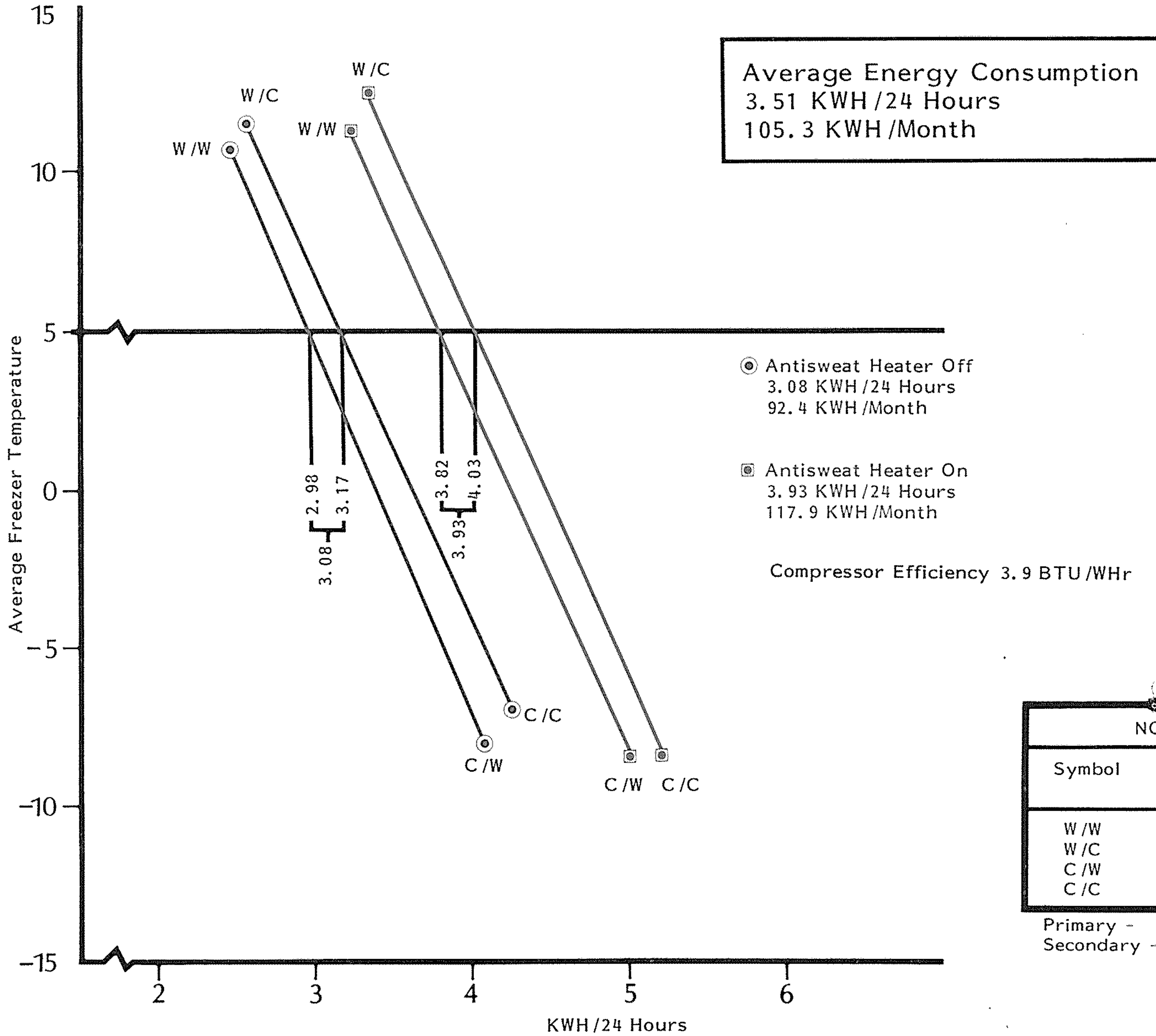


Figure 7-2

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



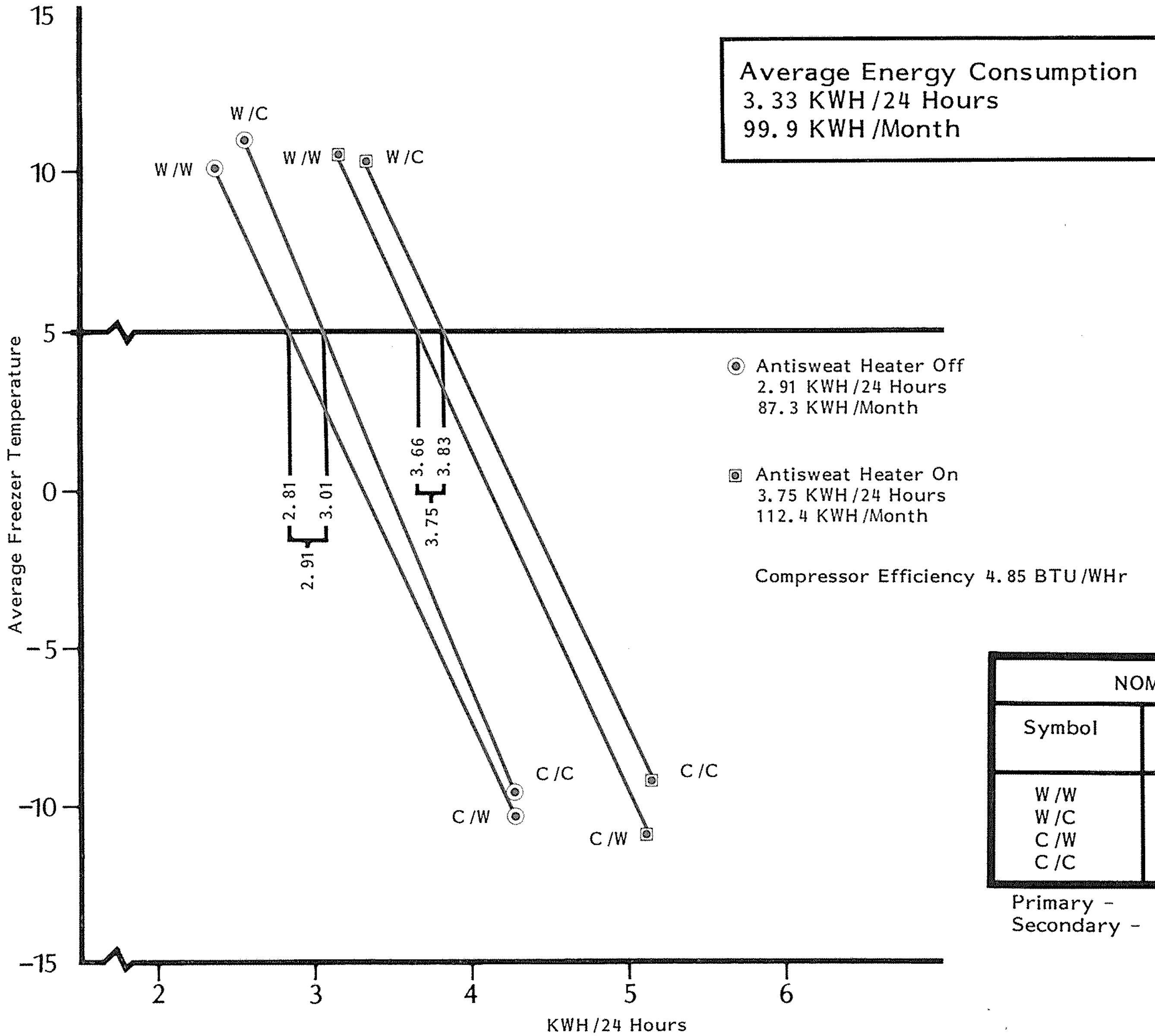
NOMENCLATURE

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 7-3

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

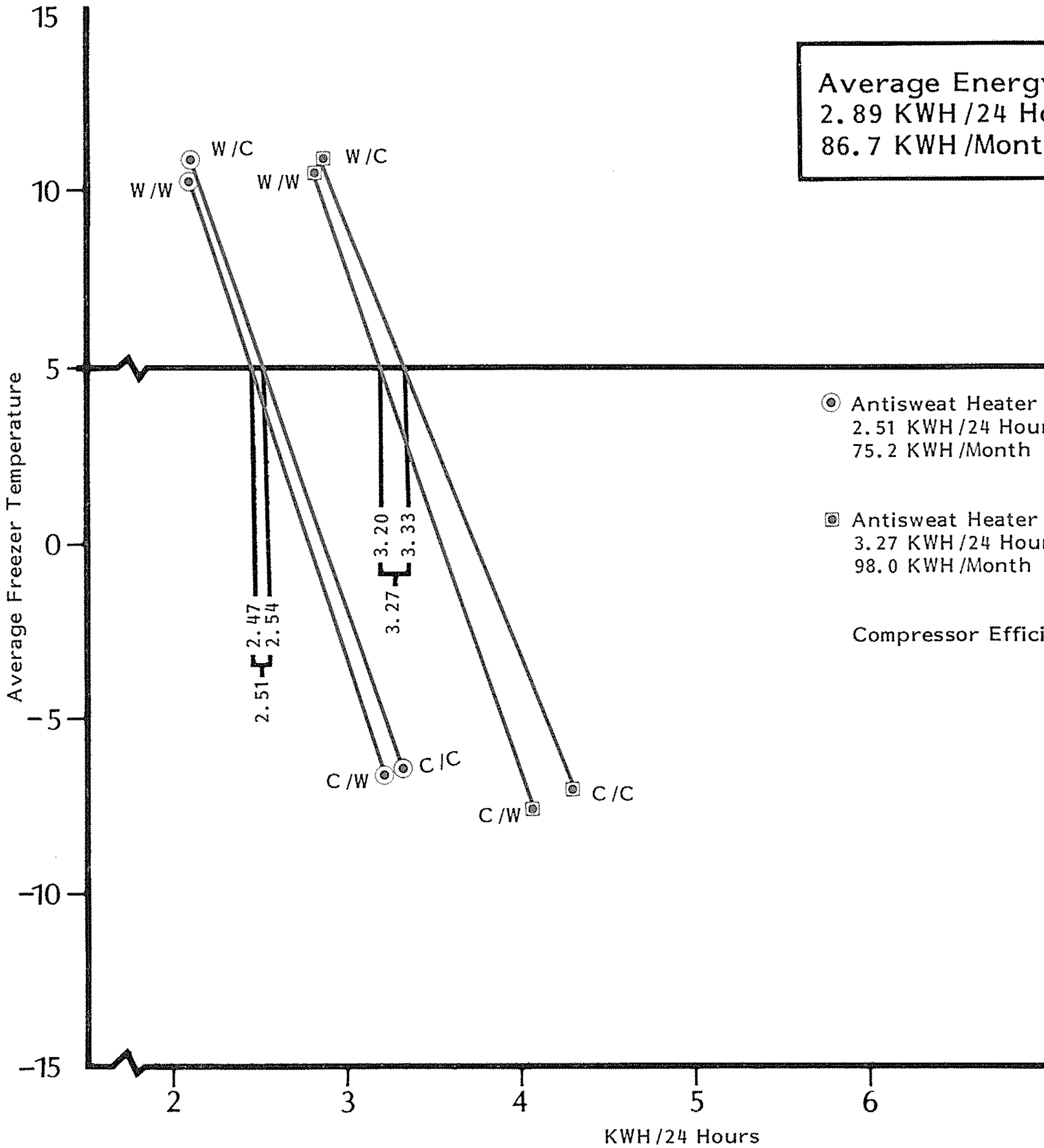


NOMENCLATURE		
Symbol	Control Setting Primary	Secondary
W/W	Warmest	Warmest
W/C	Warmest	Coldest
C/W	Coldest	Warmest
C/C	Coldest	Coldest

Primary - Evaporator Cold Control
Secondary - Air Flow Damper

Figure 7-4

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



Average Energy Consumption
2.89 KWH/24 Hours
86.7 KWH/Month

◉ Antisweat Heater Off
2.51 KWH/24 Hours
75.2 KWH/Month

▣ Antisweat Heater On
3.27 KWH/24 Hours
98.0 KWH/Month

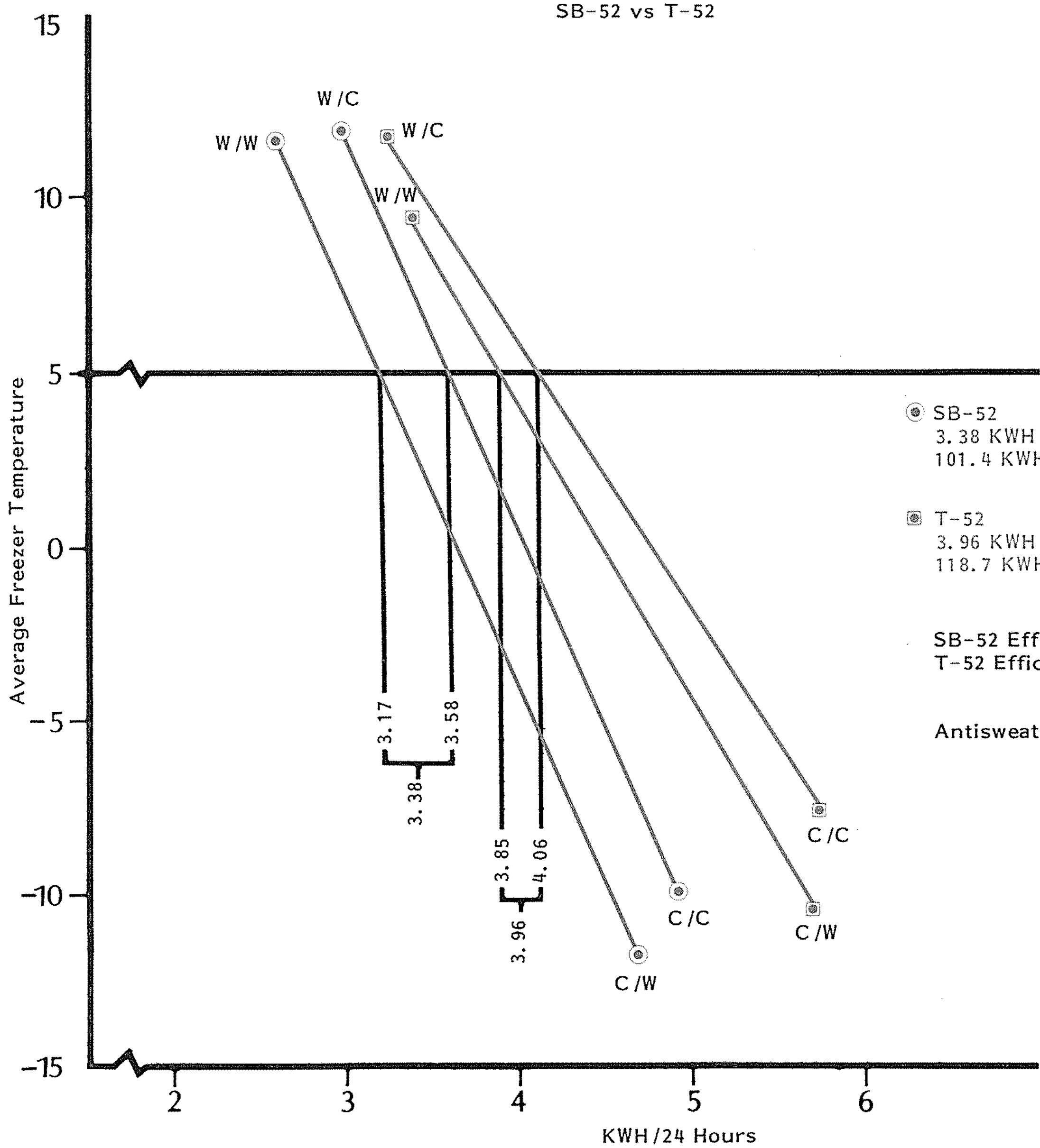
Compressor Efficiency 5.08 BTU/WHr

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 7-5

ENERGY CONSUMPTION PER AHAM HRF-2-ECFT
Standard RT-18
SB-52 vs T-52



SB-52
3.38 KWH /24 Hours
101.4 KWH /Month

T-52
3.96 KWH /24 Hours
118.7 KWH /Month

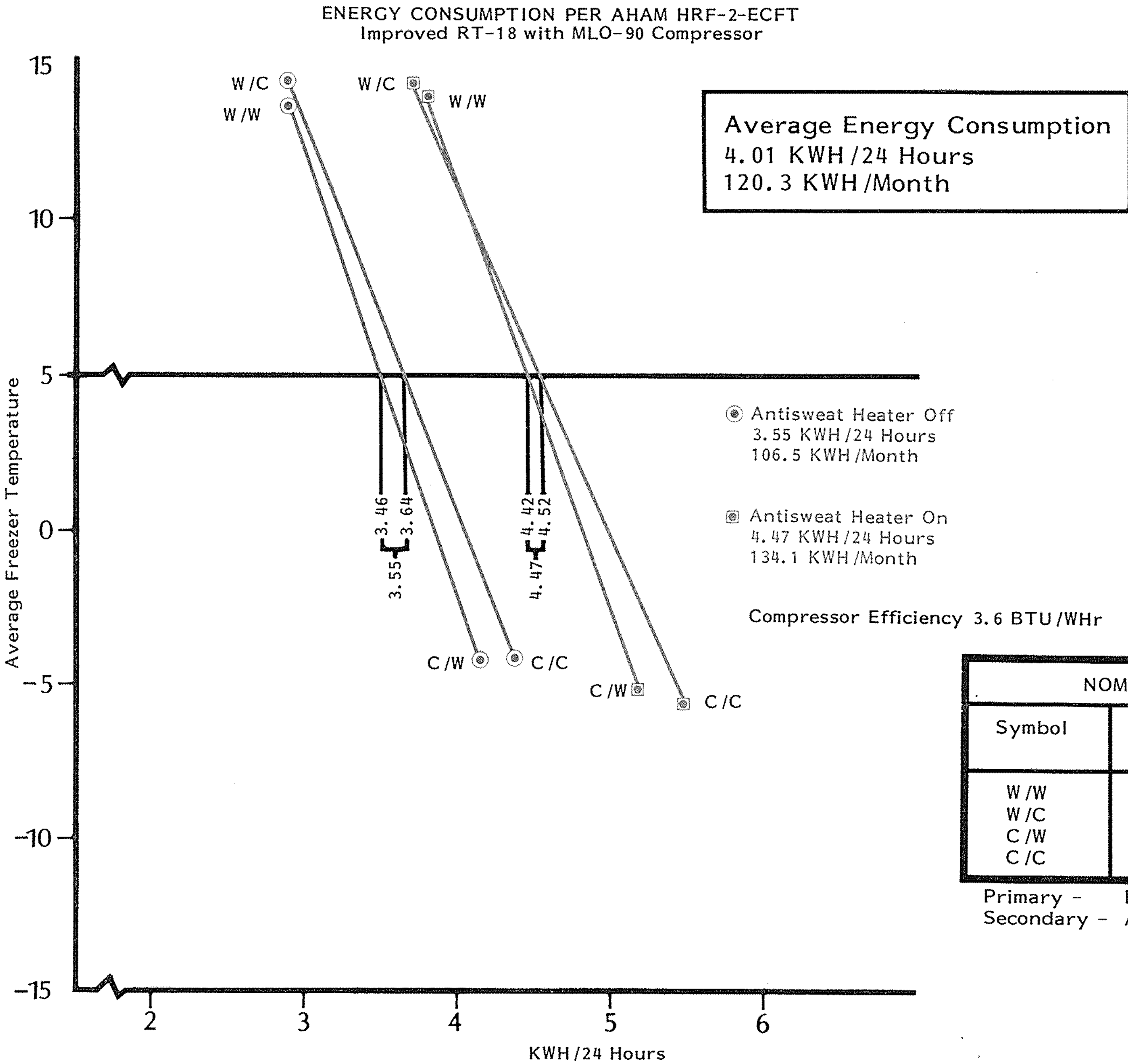
SB-52 Efficiency 4.65 BTU /WHr
T-52 Efficiency 3.47 BTU /WHr

Antisweat Heater Off

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 7-6



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 7-7

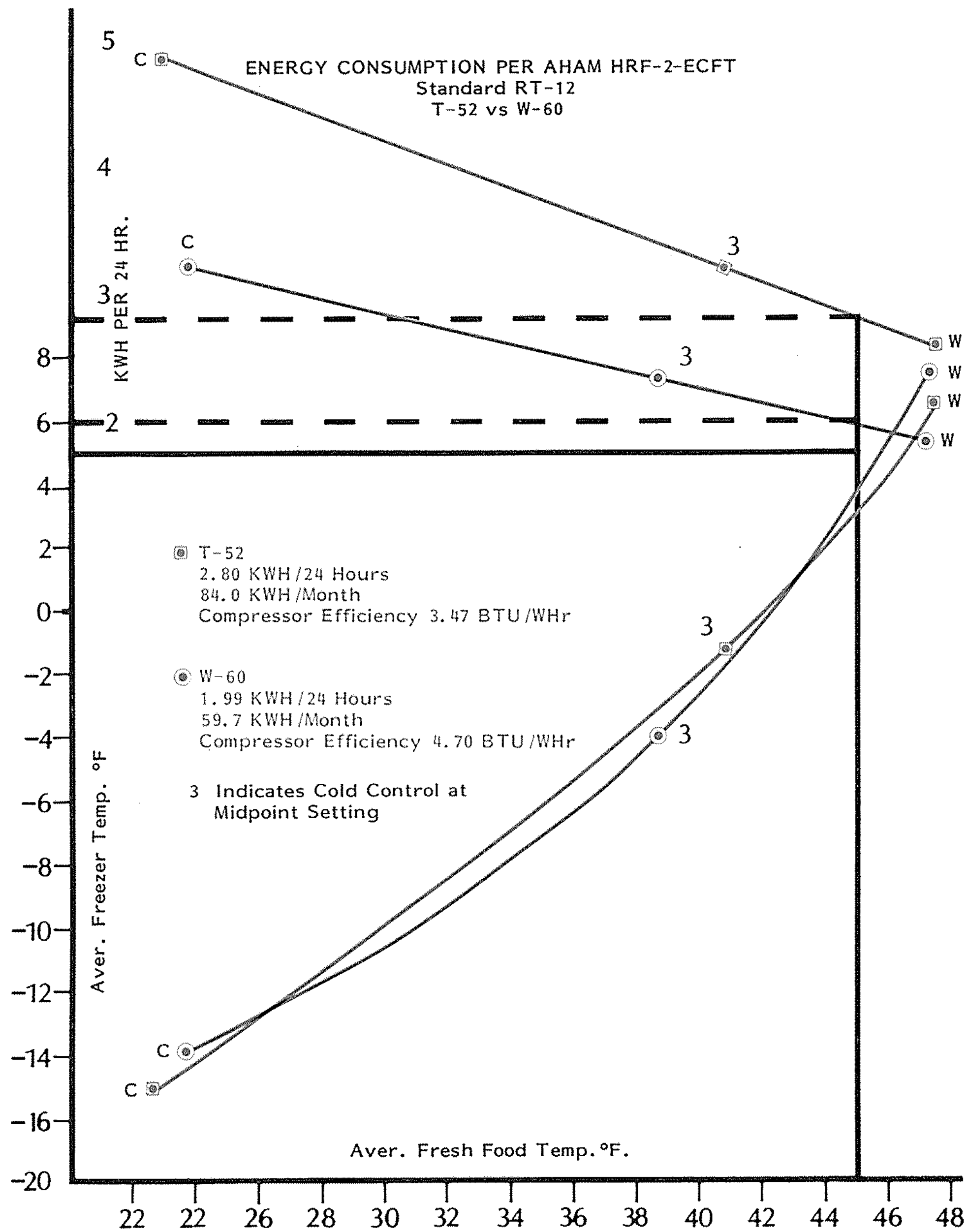


Figure 7-8

NOMENCLATURE	
Symbol	Primary Control Setting
C	Coldest
W	Warmest

TABLE 7-D

SYSTEM TESTS WITH HIGH EFFICIENCY PROTOTYPE COMRESSORS

COMB-INATION NO.	CABINET		COMPRESSOR			SYSTEM KWH/24 HR			KWH/YR		PERF. CURVE FIG. NO.
	MODEL	TYPE	MODEL	EER	% IMP.	HEATER OFF	HEATER ON	AVG.	AVG.	% DEC.	
(a)	RT18	std.	T52	3.47	-	3.93	4.67	4.30	1569.5	-	7-2
(b)	RT18	imp.	T80	3.89	12	3.08	3.93	3.51	1281.2	18	7-3
(c)	RT18	std.	W80	4.85	40	2.91	3.75	3.33	1215.5	23	7-4
(d)	RT18	imp.	W80	5.08	46	2.51	3.27	2.89	1054.8	33	7-5
(e)	RT18	std.	T52	3.47	-	3.96	-	-	-	-	
(f)	RT18	std.	SB52	4.05	34	3.38	-	-	-	-	7-6
(g)	RT18	imp.	MLO-90	3.60	4	3.55	4.47	4.01	1463.6	7	7-7
(h)	RT12	std.	T52	3.47	-		2.80	-	-	-	7-8
(j)	RT12	std.	W60	4.97	43		1.99	-	-	-	7-8

EXPLANATION

1. Compressor EER. -- BTU/WHR, per calorimeter test described in Sec. 4.1.
2. System KWH/24 HR tested per AHAM Std. HRF-2-ECFT.
3. Percentage improvement in EER and decrease in system KWH/YR based on combination (a).
4. Cabinets: Std., models in production through early 1979

   Imp., models with improved insulation and efficient fan motors, introduced late in 1979.
5. Compressor: T52. Baseline model, production through early 1979.
   T80. Improved version of T52, introduced late in 1979.
   W80. High efficiency prototype design with 4-pole PSC motor.
   SB52. Prototype design with 2-pole PSC motor and single bearing.
   MLO90. Americold production model, 1979.

(j) A W60 prototype sample in the same cabinet as (h). KWH/24 HR. -- 1.99. Fig. 7-8.

On account of the absence of a secondary control and switchable heater, the testing of combinations (h) and (j) does not involve the averaging procedures applied in the other combinations.

Table 7-D contains a summary of the test data on all of the combinations listed above. It includes the system power consumption in KWH/24 HR. and in KWH/YR, the compressor EER, the percentage improvement over the base line models, and a reference to the figures showing the system performance data. In the interpretation of the various figures presented for percentage improvement, keep in mind the considerations mentioned earlier:

- Calorimeter test results and system performance tests are not directly related because the compressor operating conditions are different.

- In the forms usually reported, percentage changes in EER and system performance are not comparable.

The combination of the W80 compressor installed in an improved RT18 cabinet (Combination (d)) closely approximates the appliance model proposed for the major quantity of the refrigerators to be included in the field demonstration part of the program. Four additional tests were run to secure more data on the performance of this combination. The figures for average energy consumption of these four samples were 3.13, 2.99, 2.97, and 3.10 KWH per 24 HR. The average for the group of five was 3.02 KWH per 24 HR.

Combination (a) represents the baseline design which was in production when the efficiency improvement program was initiated. The gains in efficiency shown for Combination (c) indicate the energy savings attained by compressor improvement alone. Combination (b) represents the efficiency level of present commercial production at CPC, while (d) represents the design to be used for field demonstration testing in the Phase II program. A comparison of Combinations (d) and (b) will be used in analyzing the marketing implications of the high efficiency design. This analysis is shown in the following section.

8.0 MARKETING IMPLICATIONS OF HIGH EFFICIENCY DESIGN

The discussion of the data obtained in the market survey, as reported in Volume II, and summarized in Sec. 2 of this volume, led to the conclusion that a pay-back concept could be used in estimating the market potential for high efficiency appliances. In applying this principal, a comparison of projected energy cost saving vs. estimated increase in market price will be made for Combinations (b) and (d) as described in preceding Sec. 7. Combination (b) represents current commercial production at CPC and (d) the design to be used for field demonstration samples.

The energy saving in KWH/YR, based on the test data reported in Table 7-A, page 87, is as follows:

		Average KWH/YR
(b)	T80 in improved RT18,	1281.2
(d)	W80 " " " ,	1054.8
	Improvement	226.4

The increase in factory cost is a CPC Engineering Department estimate. The increase in retail price is derived by assuming that the market price of a refrigerator is 2.5 times the factory cost.

	Estimated Increase	
	Factory Cost	Retail Price
Combination (d) vs. (b)	$ 2.75	$ 6.88

Consumer energy costs vary widely in different parts of the country. Savings based on costs of 3.0, 5.0, 7.0 and 9.0 cents per KWH will be used in this report. (The 5.0 figure is approximately equal to national average of 4.97¢ used on energy labels. The savings for the case, for periods up to 36 months are shown on Curve 8-1.

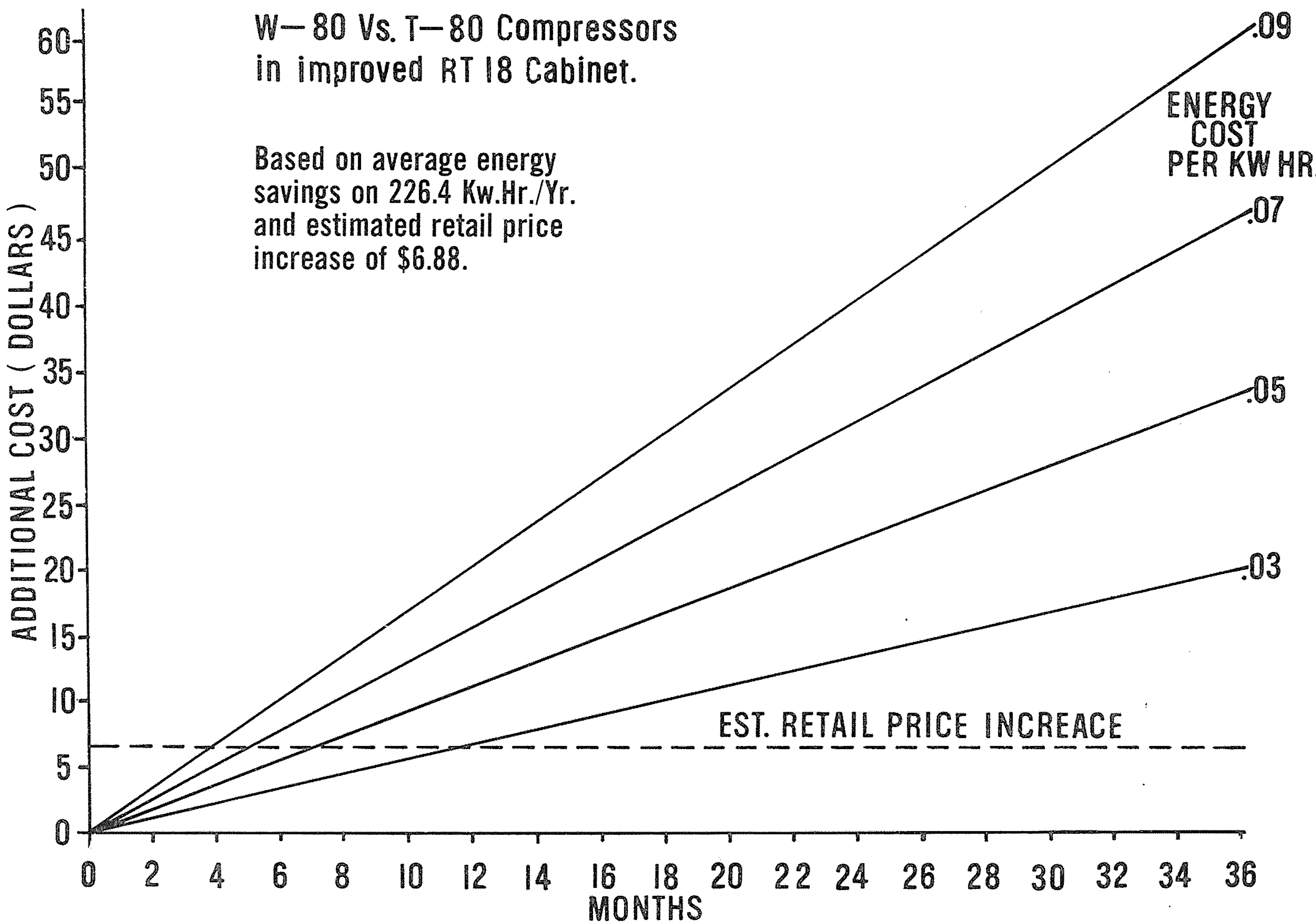

W— 80 Vs. T— 80 Compressors
in improved RT 18 Cabinet.
Based on average energy
savings on 226.4 Kw.Hr./Yr.
and estimated retail price
increase of $6.88.
ENERGY
COST
PER KW HR.
.09
.07
.05
.03
EST. RETAIL PRICE INCREACE
ADDITIONAL COST (DOLLARS)
0
5
10
15
20
25
30
35
40
45
50
55
60
0 2 4 6 8 10 12 14 16 18 20 22 24 26 28 30 32 34 36
MONTHS

Figure 8-1

The corresponding estimated retail price increase of $6.88 is plotted as a horizontal line on the same curve. The intersection of this horizontal line with the energy cost line defines the time required to "pay off" the added price. For the small dollar amounts and short times involved, interest and carrying charges have been disregarded. An inspection of the curve will show that the high efficiency design will "pay off" in 5 to 15 months, depending upon the energy cost, or in 9 months at an energy cost of 5.0 cents per KWH, which closely approximates the national average.

According to the market study results shown in Table 2-A, page 6 a pay-off period of 12 months will be attractive to 92.3% of the potential customers. Even allowing for substantial variations in customer response, it seems safe to conclude that high efficiency refrigerator models using the W80 compressor design will be attractive to most buyers.

9.0 DEVELOPMENT OF HIGH EFFICIENCY COMPRESSOR FOR ROOM AIR CONDITIONER APPLICATIONS

9.1 Introduction

The principal tasks in the development of a high efficiency compressor for room air conditioners are listed in the Scope of Work, Appendix A. The general introductory material, market study results, and development plan found in Sec. 1, 2, and 3 of this report apply to both RAC and refrigerator-freezer models.

The RAC compressor model chosen as a base line for the development effort is designated T66. The T66 is of the reciprocating type, with a two-pole PSC motor. It is very similar in mechanical details to the refrigerator-freezer models described in earlier parts of this volume. The refrigerant used is R-22.

The improved compressor model resulting from the development program has been designated W600. The principal performance parameters of the base line and improved models are listed below:

	Base Line T66	Improved W600
Capacity BTU/HR	6000	6000
Power Consumption Watts	750	632
EER, BTU/WHR	8.0	9.5

The improvement was achieved mainly by the use of a motor of higher electrical efficiency, more closely matched to the compressor torque requirements, and by reductions in hydrodynamic flow losses. An attempt to apply a four-pole motor was unsuccessful.

The use of pay-back period concept for estimating market potential, as outlined in Sec. 2, is also valid for RAC compressors.

On this basis room air conditioners using the improved compressor should be attractive to most customers.

Since the improvements made in the RAC compressor do not affect compressor reliability, a Phase II field demonstration program is not proposed at this time.

9.2 RAC Compressor Testing

The compressor test procedures described in Sec. 4 are applicable in general to RAC models. The standard calorimeter test conditions (refer to Sec. 4.1) are different, however, because of the use of Refrigerant 22. The RAC conditions are listed below:

STANDARD RATING POINT FOR RAC COMPRESSORS

REFRIGERANT 22 - HIGH BACK PRESSURE

Evap. Temp. $^{\circ}F$	45.0
Evap. Press. psig.	76.0
Cond. Temp. $^{\circ}F$	130.0
Cond. Press. psig	296.8
Return Gas $^{\circ}F$	95.0
Liquid to Exp. V. $^{\circ}F$	115.0

The ambient conditions to which the RAC compressor is subjected during the test are also different. The compressor is mounted in a duct with an area of 210 sq. in. The ambient temperature is 90.0°F, and the air circulation rate is 400 cu. ft. per min.

10.0 TECHNICAL INVESTIGATION RAC COMPRESSORS

The following sections describe the evaluation of electrical, hydrodynamic* and mechanical efficiency features proposed for the RAC compressor. Some features which had previously been tried and found to be unsuitable in the course of the refrigerator-freezer compressor improvement program, discussed in Sec. 5, were not retested for the RAC design.

10.1 Electrical Efficiency Improvements

Improvement efforts in this area were directed toward the development of more efficient motors and closer matching of motor characteristics to compressor torque requirements. For a discussion of the factors which affect motor efficiency, refer to Sec. 5.1. The Athens Products Co.**(APC) undertook the development of more efficient motors and supplied samples for test.

10.1.1 Improved Efficiency Two-Pole Motor

The first samples of high-efficiency RAC motors received from APC were designated P-5705. The stack height of this motor was 2.25 inches rather than the standard 2.0 inches, requiring that special crankshafts, motor housings, and shells be constructed for testing. Characteristics of the P-5705 are compared to those of the standard T66 motor below.

Motor	Full-Load Efficiency	Locked Rotor Torque
Standard T66	78.2	5.12 oz-ft
P-5705	81.4	3.3 oz-ft

The P-5705 motors were built into four T66 compressors which were of standard construction except for the special parts previously noted. The compressor were tested, and the results are compared to the values quoted for standard production T66 compressors in the tabulation below:

*All hydrodynamic testing involved the use of Texaco Capella B B refrigeration oil.

**The White Consolidated Motor Plant in Athens, Tennessee

Compressor	BTU/HR	Watts	BTU/WHR
STD. T66	6000	750	8.0
P-5705	5949	713	8.34

The next test series to be implemented involved reducing the stroke, thereby reducing capacity. This was done in order to determine whether the motor was sized to be most efficient at the operating point of this compressor capacity or at some other capacity. The original stroke was .820 inches, the same as the standard T66. It was subsequently reduced in two steps, first to .785 inches, then to .750 inches. The test results are reported below, again using the P-5705 motor in all cases:

Stroke	BTU/HR	Watts	BTU/WHR
.820	5949	713	8.34
.785	5759	670	8.60
.750	5630	625	9.01

These results indicate that the P-5705 motors were undersized for the T66 application. Another explanation for the improvement, at least in part, may be that the reduced refrigerant flow with a shorter stroke also reduces the flow losses through any restrictions in the refrigerant passages. It is believed, however, that the improving motor efficiency produced the major portion of the compressor efficiency improvement.

The second samples of high-efficiency RAC motors received from APC were designated P-5710. The motor lamination stack height was reduced to 2 inches, the same as the standard production motor, thereby allowing use of standard parts in the test compressors. This also makes the ultimate high volume production of a high efficiency compressor much less costly in terms of capital equipment investment. The characteristics of this motor are compared to those of the P-5705 below:

Motor	Full Load Efficiency	Locked-Rotor Torque
P-5705	81.4	3.3 oz-ft
P-5710	81.8	3.4 oz-ft

These results, contrary to expectations, indicate that the shorter stack motor is more efficient than its predecessor. It is also slightly stronger, so that its most efficient operating point should be closer to that of the T66 compressor.

Four T66 compressors were built up using the P-5710 motors. The results are compared to those previously given for standard T66 and P-5705 motor samples.

Compressor	BTU/HR	Watts	BTU/WHR
STD. T66	6000	750	8.0
P-5705 (Avg)	5949	713	8.34
P-5710 (Avg)	6012	701	8.58

The P-5710 samples had the highest efficiency in this series of tests.

The next tests with P-5710 motors were conducted in samples with the stroke reduced to .750 in. These samples also had discharge loops and mufflers with enlarged flow passages, and valve plates with enlarged ports. The results, compared to samples with an .820 in. stroke, and other parts standard are as shown below:

Motor	Compressor Stroke	BTU/HR	Watts	BTU/WHR
P-5710	.820	6012	701	8.58
P-5710	.750	5492	615	8.93

How much of the above improvement can be attributed to reduced stroke and how much to reduced flow losses due to the special parts is impossible to say without additional testing. It is likely that both of these changes did contribute, however.

On the basis of the above tests, the P-5710 motor was selected for use in the prototype design. It is superior in performance, and the fact that the core length is 2.00 in., compared to 2.25 for the R-5705 leads to lower cost, a saving in material, and greatly facilitates adaptation to the present line of compressors.

10.1.2 Four-Pole RAC Motor

Due to limitations of the T compressor as to maximum bore and stroke,*four-pole motors were not initially considered. The largest bore, 1.218 inch diameter, and the longest stroke, 1 inch, would produce a compressor with an estimated capacity of around 5500 BTU/HR, which was not considered adequate as a replacement for the present T66. Further enlargement would require major redesign of the compressor. Additionally, if the motor stack should need to be increased to give sufficient torque, as was thought likely, new parts would be required and the height of the entire compressor would increase, probably making it unsuitable for its present applications. Later, in view of the marked improvement in efficiency observed in refrigerator-freezer samples at four-pole speeds, it was decided that an RAC test was warranted.

Sample motors, designated P-5711 were supplied by APC. The motor characteristics are compared to those of the standard T66 motor below.

Motor	Full Load Efficiency	Locked-Rotor Torque
STD T66	78.2	5.12 oz-ft
P-5711	75.8	5.2 oz-ft

The lower motor efficiency was expected. The refrigerator-freezer four-pole motors also had lower efficiency than two-pole type. The inherent lower efficiency of four-pole motors has been

*Without incurring extremely high facilities expenses.

more than made up by other factors resulting in increased BTU/WHR of compressors when calorimeter tested.

Four sample compressors were built and readied for calorimeter testing. Each behaved in the same manner; after normal start up, the watts slowly increased until, after about fifteen minutes of running, the compressor stalled. This behavior is characteristic of seizure due either to lack of lubrication or loss of clearance as parts heat up. There followed a series of tests in which the oil level was adjusted, the crankshaft was vented, and numerous open air oil pumping tests were performed. Also, initial clearances were rechecked, out-of-roundness of heated pistons and cylinders was measured to verify adequate clearance when heated. Undersized pistons and rings were tried. This investigation did not reveal any mechanical reason for the stallings. Meanwhile, APC analyzed all the test data and concluded that the motors were too weak. As the motor heats up, the torque curve shifts and the motor reaches breakdown torque and stalls. A test at higher voltage showed that the compressor would run a longer period before stalling. Eventual reduction of the stroke confirmed the weak motor theory as successive reductions caused the stall to be postponed for several hours and then to be eliminated entirely, although efficiency was very poor.

Further studies at APC indicated that it would be necessary to increase stack height, perhaps to 2.75 inches, to obtain the needed torque. This would make the compressor unsuitable for its present system applications. As a result of these unfavorable indications, work on four-pole RAC motors was discontinued, and they will not be used in the prototype samples.

10.2 Improved Volumetric Efficiency

Volumetric efficiency improvements are difficult to realize in RAC compressors because of the low compression ratios at which they operate. The RAC compressor operates at about 3.4:1, compared to about 11:1 in refrigerator/freezer models. Because of this efforts to reduce clearance volume in the RAC samples were confined to tests with plug pistons.

The four compressors which were built to test the P-5705 motors were rebuilt and run with cast iron plug pistons. This eliminates the clearance volume which exists between the smaller aluminum piston and cylinder wall above the piston ring. Average results are reported below.

Compressor	BTU/HR	Watts	BTU/WHR
Ringed Piston	5971	717	8.33
Plug Piston	6130	724	8.47

The compressors showed a small capacity and efficiency improvement with the plug pistons. Some of the efficiency improvement in these compressors may have resulted from additional run time. Later, after these compressors had been rebuilt with a .75 inch stroke, they were run with both standard and plug pistons. The results are shown below.

Compressor	BTU/HR	Watts	BTU/WHR
Ringed Piston	5630	625	9.01
Plug Piston	5644	632	8.93

This time, the compressors did not show a significant change in performance when a plug piston was used. The plug piston is therefore not included in the prototype design.

Other changes which might reduce clearance volume, such as reducing head clearance and the use of special discharge valves which fill the discharge ports had been tried in refrigerator/freezer compressors and found to be impractical. These changes were not tried in RAC samples.

10.3 Reduced Discharge Flow Losses

Refrigerant mass flow in an RAC compressor is much greater than that in a refrigerator/freezer compressor. The increased mass flow is brought about by the use of R-22 refrigerant. In comparison with R-12, which is normally used for refrigerator/freezer applications, R-22 has a much larger latent heat of evaporation and a lower specific volume. Thus, for a given volume of saturated refrigerant, R-22 has a much greater refrigerating capacity. This increase in capacity with a given displacement makes R-22 ideally suited for RAC applications where reduced displacement and unit size are important design criteria. In order to accommodate high mass flow rates, RAC compressors are normally designed with gas flow paths as free from restrictions as possible.

The T66 compressor, which was chosen as a base line for development was originally designed such that the gas flow path was consistent with desired capacity and noise levels for a given displacement. Many of the tests to be described in this section actually show that the gas flow path of the T66 compressor was reasonably close to the geometric restrictions of the design.

Four compressors were built with P-5705 motors, the performance characteristics of which have been previously described. With the exception of plug pistons, all other components of these samples were standard T66. Using these compressors as test vehicles a series of experiments relate to the high, or discharge side, of the compressor. An experimental discharge muffler with two rather than three chambers, and a revised interconnecting tube arrangement was constructed. Valve plates with larger discharge ports and discharge tube loops with slightly larger internal diameters were fabricated. These parts, along with the P-5705 motors, were assembled along with T66 component parts. The resulting sample compressors, when calorimeter tested, yielded the following calorimeter test data:

Compressor	BTU/HR	Watts	EER BTU/WHR
Reference Test (Motor change only)	6179	720	8.58
Reduced Flow Loss Sample	6140	699	8.78

The improvement in EER corresponds to a decrease of 16 watts referred to a nominal capacity of 6000 BTU/HR.

Another pair of tests were run with three of these compressors. The stroke was reduced to .75 inch. Plug pistons were used, but the other parts were standard in the first test in this series. For the second test, the same low-loss mufflers and discharge loops were used again. The valve plate had enlarged ports, but also used a plate-type discharge valve as described later in this report. Based on other tests, this discharge valve probably had little influence on these results:

Compressor	BTU/HR	Watts	EER BTU/WHR
First Test	5665	633	8.95
Reduced Flow Loss	5360	599	8.94

This pair of tests showed no significant change in EER. Because the plate-type valve has produced more test variations within a group of compressors, the above results should be considered of less significance than the first group.

No other incremental tests involving these variables were conducted. The discharge muffler, discharge loop, and discharge port restrictions were varied simultaneously in the tests because restriction at any point will prevent improvements made at another point from showing up in the tests. It is felt that the present T-line model T66 is fairly well balanced in these areas. The changes in valve plate, muffler, and discharge loop are included in the prototype design.

10.4 Reduced Suction Flow Losses

Tests were made on sample compressors with several types of plastic suction mufflers, to evaluate the effect of reduced flow losses in the path of the gas entering the cylinder. Since the plastic material has a lower coefficient of heat transfer than steel, the plastic muffler should have the further advantage of causing less super heating of the suction gas. The first samples of plastic mufflers were of similar configuration to the production muffler, but did have flow passage differences, largely due to the design constraints dictated by the injection molding process.

Four compressors were tested with standard suction mufflers, then with the plastic mufflers in their original configuration. These compressors used the P-5705 motor and .786 inch stroke. The results follow:

Compressor	BTU/HR	Watts	EER BTU/WHR
Std. Muffler	5759	670	8.60
Plastic Muffler	5480	676	8.11

During the course of the muffler testing program, it was reasoned that the tube connecting the top of the muffler to the head was a possible source of flow losses. The muffler was therefore extended in length until it reached the suction inlet in the head. A stub length of tube about one half inch long was used to connect the head and muffler. The top of the muffler was reshaped to provide a smooth flow of gas into the head. The lower end of the muffler was the original plastic design for gas inlet through the top of the shell. The oil drain slot was nearly blocked. The suction cavity of the head was built up with epoxy to provide a streamlined flowpath for refrigerant leaving the muffler tube and entering the suction ports. These alterations were tested utilizing a compressor with the P-5710 motor and .75 inch stroke. The oversized discharge tube loop and discharge muffler with reduced restrictions were also adopted. As can be seen below, this combination produced favorable results, although not as dramatic as had been originally predicted.

Compressor	BTU/HR	Watts	EER BTU/WHR
Std. Suction Muffler	5630	614	9.16
Special Plastic Suction Muffler	5814	626	9.28

The above improvement in EER corresponds to a decrease of about 9 watts, when referred to the nominal capacity of 6000 BTU/HR. The plastic suction muffler will be included in the prototype design.

10.5 Reduced Heat Transfer From Discharge Gas Path

An attempt to limit heat transfer from the refrigerant gas was made on the discharge side of the cylinder. The purpose of this is to limit 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 of the motor winding temperature 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. The compressor used for this test was built of standard T66 parts except that the cylinder bore was enlarged to 1.218 inch diameter and the stroke reduced to .53 inch. The cylinder head and valve plate assembly were from a previous production compressor model T101. The results shown below indicate that no appreciable change in efficiency occurred when the discharge loop was insulated.

Compressor	BTU/HR	Watts	EER BTU/WHR
Std. Disch. Loop	5423	636	8.52
Insulated Loop External Muffler	5334	624	8.54

The prototype design will not incorporate any insulation of the discharge line.

10.6 Plate-Type Discharge Valve

The plate-type discharge valve* was designed to be both a cost reduction and possible performance improvement. The present discharge valve consists of a rather squared-off, wide horseshoe shaped reed with short legs. The valve is riveted to the valve plate near the end of these legs and is allowed to bend back against the smooth curve of a valve stop of the same shape which is riveted on along with the reed. In RAC compressors, a second reed is included between these two items, pre-set away from the valve plate, which further cushions the impact of reed against stop.

The plate-type valve was similar to the one shown on Fig. 5.6, Item 1 and consists of a plate of valve steel the same size as the valve plate. It is punched so that a reed of the same configuration as the present valve, but with longer legs, is supported in the same position as the present valve and clamped between the valve plate and cylinder head by the head bolts. With this arrangement, any valve stop must be made a part of the cylinder head. The backup reed can be duplicated by use of a second plate type valve, pre-set away from the first.

The plate-type valve eliminates the riveting operation, rivet holes, and separate valve stop. It was hoped the longer legs would reduce the spring rate, allowing quicker opening and less flow resistance on the discharge stroke. This theory was tested both by performance testing compressors and by observing the trace of cylinder pressure produced by a compressor equipped with a pressure transducer attached to an oscilloscope.

The following results are from tests of compressors using the

*Patent application filed by CPC.

P-5705 motor. The stroke was reduced to .75 inch. In both cases, the valve plate had discharge ports enlarged to .295 inch diameter with no trepanning. The first test compressors had standard discharge valve configurations while the second test compressors used the plate-type valve with no valve stop. The first test results are an average of four compressors, the second only three due to valve breakage in one sample.

Compressor	BTU/HR	Watts	BTU/WHR
Std. Valves	5461	606	9.01
Plate-type Disch.	5326	585	9.10

The test results show a minor improvement in EER. Unfortunately, several other plate-type discharge valves broke during testing, both with and without valve stops. This valve design is therefore not acceptable for the RAC prototypes.

The behavior of the plate-type discharge valve while running was investigated in a test compressor equipped with a 0-400 psia. pressure transducer mounted in the cylinder housing in such a way that it communicated with the valve stop machined into the top of the cylinder bore. The transducer screwed into a close fitting well, which was connected to the valve stop by a very small hole. The small hole damped the pressure fluctuations slightly and added minimum clearance volume to the cylinder. The trace obtained, very closely followed cylinder pressure during a cycle of the compressor. Prior to testing, the transducer trace was zeroed by allowing the cylinder to come to atmospheric pressure (0 psig.), and then calibrated by raising it to a measured value (checked by test gauge) and setting the gain of the oscilloscope to produce the corresponding desired deflection. Inductive pickups were used to produce pulses at top and bottom dead center, to mark these events on the oscilloscope and to trigger the oscilloscope on each revolution so that a continuous trace could be obtained.

A comparison of the traces obtained for a standard valve and a plate-type discharge valve configuration produced the following results and conclusions:

- There is not enough difference in the timing of events to be detectable on the trace, indicating that the plate-type valve does not open appreciably sooner.

- The peak pressure developed is not lower with the plate-type valve, indicating that it does not open appreciably sooner or with appreciably less restriction than the standard valve.

- A slight "bump" which occurs on the trace shortly after top dead center as pressure is falling in the cylinder is more pronounced on the plate-type valve trace. This is interpreted as the discharge valve bouncing on its seat, allowing some high pressure gas to leak back into the cylinder. The greater bounce would be indicative of greater leak-back and lower efficiency.

These observations help to explain why the performance of the plate-type valve is similar to that of the present design, but do not provide information on the cause of the premature failures.

10.7 Mechanical Efficiency

The approaches tried in reducing the mechanical losses in the refrigerator/freezer compressors, and later proven to be unsuccessful, were not repeated in the RAC program. The use of rolling element bearings was ruled out in the refrigerator compressor program because of reliability problems. A single bearing design was not considered for the RAC compressor. The bearing loads are higher and the unsupported length of crankshaft is greater in an RAC compressor.

The oil stirrer design is difficult to evaluate. Compressors built with no oil stirrer have been found to be excessively noisy. The two designs in contention for use on RAC compressors are the present RAC stirrer with a single metal blade which rotates in the oil, and the new conical oil stirrer used in the refrigerator/freezer compressors with a plastic cone rotating in the oil. It is difficult to determine the power required by each design. Refrigerator compressor testing has been less than conclusive in this area. An unloaded compressor running on the bench has been used to try to determine the difference in watts, and the blade type is believed to be slightly better. It has, therefore, been used in all the RAC tests, although the plastic type could be used and would probably not have a significant effect on the performance.

11.0 PROTOTYPE DESIGN RECOMMENDATIONS

The features finally chosen for the RAC prototype design are discussed briefly in the following paragraphs:

The motor will be replaced with the improved efficiency model P-5710 furnished by APC. This has shown the highest efficiency of the samples tested. After all other changes to the compressor, of course, it will require start and pullout tests and motor protector selection and testing, as well as complete system testing for final verification of the application.

The stroke will be reduced from .820 inch to .750 inch. Again, when all other changes have been made, the stroke can be adjusted up or down to find a more efficient operating point.

The valve ports and discharge tube loop will be enlarged and the discharge muffler modified as described in the report to reduce flow losses. All of these changes might increase noise to an unacceptable level and this will have to be determined by noise testing before manufacturing production quantities.

The suction muffler will be plastic, of a design similar to the handmade version tested late in the program. This muffler extends to the side of the cylinder head and uses minimum tubing to connect it to the head. It is shaped for smooth flow into the connecting tube. The suction cavity in the cylinder head will be reshaped for smooth gas flow into the suction ports of the valve plate.

A prototype sample with all of the above features has been constructed and tested. The results, as tested and as referred to a nominal capacity of 6000 BTU/HR, were as follows:

	As Tested	At 6000 BTU/HR
Capacity, BTU/HR	5865	6000
Watts	617	632
EER, BTU/WHR	9.5	9.5

11.1 Design Verification

In the opinion of CPC, the changes incorporated in the RAC prototype design do not affect service reliability. No Phase II program of field demonstration is proposed at this time. A program of factory life testing, however, would be in order prior to manufacture in quantity. Following successful conclusion of the factory life tests, and of the RAC system application tests mentioned in Sec. 11.0, it is our opinion that the prototype design would be ready for quantity production without further testing.

12.0 COMPLETE AIR CONDITIONER IMPROVEMENT

Prototype samples of high efficiency RAC compressors have not been tested in complete room air conditioners. An estimate of the potential energy saving in an air conditioner rated at 6000 BTU per hr. may be made by assuming:

(a) That the room air conditioner duplicates the calorimeter at rating point conditions insofar as compressor operation is concerned. Thus, an EER improvement from 8.0 to 9.5 BTU/WHR with a corresponding reduction of 118 watts is expected.

(b) That the running time per year is 750 hours, which is the average usage in a band acorss the center of the country.

(c) That the average energy cost is 4.97¢ per KWH.

Calculated in this way, the average energy saving per year is $4.42.

The estimated increase in factory cost for the improved model is $1.28. Assuming that the increase in selling price is 2.5 times the increase in factory cost, the added cost to customer would be $3.20. giving an approximate pay-back period of 9 months. According to the results of the market study, this would be attractive to a large majority of customers.

13.0 PHASE II PROGRAM PLAN

13.1 Synopsis of Phase II Program

MAJOR GOALS OF PHASE II

Evaluate Compressor Reliability

Lubrication system (bearings)

Large bore ring piston in aluminum housing with cast iron cylinder sleeve

Evaluate Energy Efficiency

Limited quantity

Correlation of lab. energy tests with actual in-home use

PHASE II TASKS

Develop Design To Include Full Range of Capacity Sizes

800 & 1000 BTU/Hr. (and other sizes as required)

Systems tests & selection of electricals

Life Test

Establish Manufacturing Facilities & Produce 1552 Demonstration Compressors

Many parts are standard

Some special parts to be made partially on existing equipment

To be assembled on existing line

Assemble Demonstration Compressors To 1320 Refrigerators

Coordinate refrigerator model production with appropriate compressor

Distribute Field Test Samples & Collect Market Data

Sell 500 to government, U.S. Dept. of Housing & Urban Development (HUD)

Sell 800 through White-Westinghouse distributors

Utilize sales technique recommended in market study

Select geographic location (NSEW)

Collect market data during sale

Place 20 in CPC/WCI employees homes

Implement Field Trial - Six Months

Monitor reliability on monthly basis

Monitor reliability & energy usage

CPC/WCI lot

Implement Long Term Field Surveillance

Reporting - Monthly, Annual & Final As Required By UCC-ND/DOE

13.2 Introduction

Phase I of the development program comprised the evaluation of numerous proposed design changes intended to improve efficiency. The selection of the most effective changes with regard to energy improvements, and the building of prototype samples with these features were included in the Phase I work. In accordance with the Phase I Program Plan, a prototype energy efficient compressor with an EER of 5.0 BTU/WH was delivered to the ORNL-TM (Technical Monitor).

The demonstration part of this project has been designated Phase II. The major steps in Phase II are listed in the Scope of Work shown in Appendix B. A program developed by CPC for the implementation of the refrigerator compressor portion of Phase II will be found in the following sections. The construction of a total of 1502 high efficiency refrigerator compressors is recommended by CPC. CPC will assemble these samples on actual production machinery to the maximum extent possible. Most of the component parts of the recommended compressor design can be produced and assembled on already existing equipment. There are few parts, however, which require the purchase of additional capital equipment.

The implementation of field trials has, for many years, presented special problems to CPC and most other high volume manufacturers. It is difficult to trace the field samples from the factory through the distribution network. Additionally, the placement of samples in a desired geographical area and locating cooperative consumers who are willing to provide performance information have long presented problems.

CPC recommends utilizing three different methods for placing the high efficiency refrigerator compressors in the field. The compressors will be assembled to refrigerators at CPC utilizing production type factory equipment. The refrigerators will be placed in the field in the following manner:

- WCI and CPC employees homes
- Sold through White-Westinghouse distributors
- Sold to Federal agency: U.S. Housing and Urban Development (HUD)

A complete explanation of the implementation plan relating to the above channels of distribution can be found in Section 13.5.

13.3 Phase II General Approach

13.3.1 Relationship to Phase I

Phase I of the refrigerator portion of the project has involved the procedure of selecting and evaluating changes which appeared likely to increase efficiency, and which were judged to be capable of being manufactured without prohibitive cost increases. Since it is seldom possible to predict the effect of proposed individual changes on overall compressor performance, the evaluation process has required an extensive program of testing.

Because of the extensive testing required, most of the refrigerator compressor development work has been in the direction of improvements to the existing CPC Model T52 (800 BTU/HR) for refrigerator-freezer applications. CPC believes that it would not be feasible to manufacture the high efficiency compressor unless a full capacity lineup was available. With this in mind, CPC has been working, to a lesser extent during Phase I on other capacity sizes. The experimental work on the capacity sizes necessary to constitute a complete line of high efficiency compressors must be completed during Phase II.

13.3.2 Features to Be Verified in Field Demonstration

The efficient refrigerator compressor which was developed during

Phase I, utilizes a four-pole PSC motor. The four-pole PSC motor has been the single largest contributor to the overall efficiency improvement achieved in the 5.0 EER compressor. The four-pole motor however, runs at one half the speed of the present two-pole design. The quantity of compressor lubricating oil pumped varies with the angular velocity of the rotating crankshaft. Tests showed that the oil pumped in the four-pole compressor was reduced to the extent that adequate bearing lubrication was no longer possible with the existing T-line oil pump design. During the course of the Phase I work, CPC has developed a new oil pump specifically for use with four-pole compressors. Oil pump performance testing on the four-pole compressor has been carried out in the laboratory. The quantity of oil pumped per unit of time was determined with the use of a specially constructed test device employing a collector. This laboratory test cannot be carried out under actual hermetic operating conditions. Partial verification of the performance under hermetic conditions can be obtained by successfully running compressors on life tests. Some of the life tests have been completed. Others will continue beyond the completion of the Phase I project. Field tests are however required for complete verification.

A review of the principles of bearing design indicate that operating speed is an important design characteristic. The bearing lubricant film thickness, the quantity of oil flow and the temperature rise are all dependent variables of the rotational speed. Again, life testing is only an indicator of the design suitability over the long term.

CPC recommends that a field test be conducted for both reliability verification and comparative energy use under actual operating conditions. The recommended procedure for placing units in the field involves the purchase of high efficiency units by a government agency for use in federal installations, and the purchase of units by consumers through the normal distribution channels. Twenty refrigerators will be placed in CPC-WCI employees' homes. The entire field placement plan is described in Section 13.5.

13.3.3 Refrigerator Models Recommended for Field Demonstration

While the Phase I development program was directed mainly toward the 800 BTU/HR. rating, some work was also done on models with capacities of 600, 1000, and 1200 BTU/HR. CPC intends to complete the development of these additional models.

On the basis of prototype testing, the refrigerator models listed below have been selected for field demonstration.

- RT18 (18 cu. ft. top mount, forced air system, foam insulation, 800 BTU/HR compressor)

- RT21 (21 cu. ft. top mount, forced air system, foam insulation, 1000 BTU/HR compressor)

For each refrigerator model used in the field demonstration program, system tests must be made to verify the performance of the compressor and its associated electrical components. Approval of the Underwriters' Laboratories must be secured for each compressor and refrigerator.

13.4 Establishment of Manufacturing Facilities Necessary to Produce the Demonstration Samples

CPC recommends that 1502 energy efficient compressors be manufactured at Columbus. The capacity sizes to be constructed and the planned distribution are shown on Table 13-1. As previously noted, many of the component parts of the recommended design can be produced and assembled on existing equipment. Some of the component parts developed during Phase I will have been introduced into production at CPC prior to the completion of the Phase II demonstration. These parts are presently being tooled for mass production. CPC has elected not to tool up for mass production some of the efficient compressor parts, but recommends their manufacture, partially at least, on a model shop basis. One of the parts included in this group is the special crankshaft required with the four-pole motor.

The 1502 prototype compressors will be assembled in the CPC automatic assembly facility. The special component parts will be stockpiled prior to assembly and integrated at the proper time.

The manufacture and assembly of the special high efficiency compressors must be carefully planned so that potential confusion and losses associated with the interruption of normal CPC production activities can be minimized. The special compressors will be identified and shipped to an appropriate staging area at CPC for assembly to the refrigerators. When the refrigerators of the desired size are assembled, the special compressors will be integrated into the assembly operation. The special high efficiency refrigerators will be placed in easily identifiable cartons and identified with special part numbers before being stored for later retrieval.

13.5 Distribution of Field Test Samples

Five hundred highly energy efficient refrigerators will be sold to the U. S. Department of Housing & Urban Development. These will be the CPC model RT18 and will use a compressor with a capacity of 800 BTU/HR and a nominal EER of 5.0. These refrigerators will be placed, if possible, in four different areas of the country, (N,S,E,W) at various federally subsidized public housing complexes.

Eight hundred highly energy efficient refrigerators will be sold through White-Westinghouse dealers. This group will include 400 of the CPC model RT188 and 400 of the RT218. The entire field test lot construction and distribution plan is shown on

PHASE II PROGRAM SCHEDULE (REFRIGERATORS)

PHASE II TASKS

2.1.1 Develop final engineering design & specifications
2.1.2 Establish facilities necessary to produce 1502 compressors
2.1.3 Establish facilities necessary to produce 1320 refrigerators
2.1.4 Final spec & eng. des. report
2.2.1 Manufacture 1502 compressors
2.2.2 Manufacture 1320 refrigerators
2.2.3 Conduct & evaluate 6 months field demonstration
2.2.4 Refine market analysis
2.2.5 Report summarizing manufacturing & demonstration experience
2.3 Final report on all Phase II work
2.4.1 Phone survey & field site visits
2.4.2 Prepare & submit annual reports

Figure 13-1

Table 13A. Arrangements will be made with the dealers to place various capacity models of the highly energy efficient refrigerator on the sales floor. Appropriate point of sale literature will be utilized. Sales bonuses or some other inducement would be offered to the dealer to increase the probability of the sale of the efficient product. Only when the customer makes the purchase decision would he be told that the refrigerator is a field test model. A free service package for an additional year could be offered to the customer for taking part in the field trial. The following sales and service plan is recommended:

- Sell the energy efficient units at a premium price reflecting the actual marked up factory cost increase. This enables the collection of actual market data. (It has been suggested by the CPC marketing consultant that a price increase substantially higher would still generate sales if the proper sales strategy were used).

- Solicit marketing feedback data from consumer and dealer (to be utilized as reinforcement to the Carnegie-Mellon market study).

- Guarantee the sealed system for five years (parts and labor) as is done at present.

Forty refrigerators will be placed in WCI and CPC employees' homes. Of the refrigerators in this group, 20 will contain the energy efficient compressor and 20 will contain the standard production compressor, but will be otherwise comparable. The refrigerators will be instrumented for recording energy usage. The employee will be responsible for providing periodic energy use and reliability data. It is anticipated that this field placement plan will require the least time to implement. Because of this, most of the reliability and energy usage data during the initial six month field trial, will be obtained from this group.

TABLE 13-A

COMPRESSOR AND REFRIGERATOR

PRODUCTION PLAN

REFRIGERATOR		COMPRESSOR				DISPOSTION
Model	Quantity	Model	Approx. Cap.	Quantity		
			BTU/WHR	W80	W100	
RT18EA	500	W80	800	500		HUD
RT21ED	400	W100	1000		400	White-Westinghouse
RT18ED	400	W80	800	400		" "
RT21ED	10	W100	1000		10	Field Evaluation
RT18ED	10	W80	800	10		" "
-	-	W100	1000		66	Reserve
-	-	W80	800	66		"
-	-	W100	1000		25	Life Test
		W80	800	25		" "
TOTALS:	1320			1001	501	
TOTAL COMPRESSORS: 1502						

13.6 Six Month Field Test

The reasons for running a field test are to obtain reliability verification and to obtain comparative energy usage data. Reliability verification information will be obtained by placing sample compressors assembled in refrigerators in the field and monitoring their performance. It is recommended that the field test sample performance data be obtained by asking the purchaser, or in the case of HUD, the user, to fill out a monthly performance evaluation questionnaire. In the event that the cards are not filled out and sent to CPC, a follow up will be made by phone.

The reliability verification information is envisioned by CPC to be partly subjective and partly objective in nature. For example, noise will be judged according to the individual respondent's sensitivity to sound intensity at various frequencies. Further, the refrigerator's physical location in the home has an effect on the noise as perceived by the owner. Obviously, any failure will have to be reported to CPC and the failed compressor retrieved and analyzed. In the case of the refrigerators sold through the White-Westinghouse organization, the failures will all be traced through the existing service organization.

The problem of obtaining comparative energy usage data has long been discussed in the industry. If it is assumed that two refrigerators be placed in the field, one with an energy efficient compressor and one with a standard compressor, the task of relating the energy used by both to some common base line is difficult. This is because both can never be exposed to the same environmental conditions. Energy usage can be affected by the following:

- Frequency and duration of door openings
- Ambient temperature and humidity
- Control setting

- Cleanliness of condenser
- Fresh and frozen food load

The energy usage information will be obtained in the AHAM HRF-2 test when run in an environmentally controlled room at CPC. This test is likely to yield more accurate comparative energy usage results than the field test between the production and energy efficient compressors. The problem of correlating the lab testing with actual field test data still exists, but the information gathered from those units tested in a home environment will aid in establishing a better correlation.

After consideration of the above, CPC recommends that energy usage data be collected only for the group of field test samples placed in homes of WCI and CPC employees. Kilowatt hour meters will be mounted to the refrigerators behind the front toe plate and the respondents will be asked to read the meter monthly. Twenty of the group will be standard production compressors and twenty will have the energy improved compressor. Reliability and energy usage data for this relatively small group will be obtained by mail or personal contact.

13.7 Long-Term Field Performance Evaluation (30 Months Beyond Initial Demonstration)

Energy monitoring of samples in WCI and CPC employees' homes will be continuing throughout the extended 30 month period. The respondents will be asked to read the kilowatt hour meters provided on a monthly basis. Reporting by mail, over the 30 month period, will be the preferred method, but in the event no report is forthcoming, a follow up will be made by telephone or personal contact.

The remainder of the field samples will be monitored only for reliability purposes. The names and addresses of purchasers of the field test units sold through the White-Westinghouse distributors will be noted and kept on record at CPC. Questionnaires will be sent out to these purchasers on a quarterly basis. The remaining group to be supplied with refrigerators with energy efficient compressors is HUD. It is anticipated that these refrigerators will be placed in various public building complexes throughout the county. Contacting the users of these refrigerators to obtain reliability information may be difficult due to changes of occupany. It is suggested that the monitoring of these units be carried out by the building engineer or the normal repair staff that maintains all of the federally owned appliances. The quarterly questionnaire and/or follow up telephone call approach will be utilized. Annual reports summarizing the reliability information will be prepared and submitted to UCC-ND.

13.8 Refinement of Market Analysis

It is planned that 800 refrigerators with energy efficient compressors will be marketed through the White-Westinghouse sales organization. CPC recommends that these refrigerators be marketed with appropriate point of sale literature and salesman's presentation. The Phase I market study states that the salesman's role is predominant in inducing the purchaser to pay more money up front to achieve a life cycle savings because of higher appliance operating efficiency.

The efficient refrigerators sold should carry a price which has been adjusted to reflect the increase in cost of the energy efficient compressors versus that which is otherwise available in the White-Westinghouse line. Additional marketing data will thus be collected, which will either reinforce or modify the findings of the Phase I market study.

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 Monitor (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.

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

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

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 motos, 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 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.

APPENDIX "B"

SCOPE OF WORK

Phase II. Demonstration Plan

Task 2.1

Finalize design and establish manufacturing facilities.

Task 2.1.1 Develop final engineering designs and specifications. Based on the results of Phase I, prepare engineering design information for high efficiency motor compressor units for demonstration samples, in capacity ratings of 600, 800, 1000 and 1200 BTU per hr. Complete application work on PTCR and motor overload protectors. Repeat performance tests in cabinets.

Prepare samples and secure U.L. approval on compressors and complete refrigerators.

Release drawings, specifications and parts lists, and enter parts lists into CPC production control system.

Task 2.1.2 Establish manufacturing facilities needed to produce demonstration compressors.

Prepare and implement detailed plan for the manufacture of high efficiency compressors. Locate sources of parts and materials. Procure special facilities required. Develop production schedule for special compressors in factory assembly area.

Task 2.1.3 Prepare detailed manufacturing plan for demonstration refrigerators. Locate sources of any special parts or material required. Develop production schedule for assembly of compressors into refrigerators on regular production lines.

Task 2.1.4 Submit report on final engineering design and manufacturing plan, covering Tasks 2.1.1, 2.1.2, and 2.1.3 above.

Task 2.2

Manufacture samples, perform life tests and field tests. Refine market analysis.

Task 2.2.1 Procure materials, parts, and facilities. Manufacture compressors and place in temporary storage.

Verify compressor quality by means of a complete inspection of 10 representative sample conducted by CPC Quality Control Department.

Verify compressor performance by calorimeter test on 25 samples in Compressor Engineering Laboratory. Hold samples for use in sample refrigerators where compressors of known performance are needed.

Build 50 stands for compressor life testing. Conduct life test on 50 compressor samples. Samples to be given calorimeter and noise tests both before and after life testing. After life test, the samples are to be torn down and inspected.

Task 2.2.2 Manufacture demonstration refrigerator samples using high efficiency compressor previously produced.

Verify quality by means of complete inspection of representative samples by CPC Quality Control Department.

Performance test 4 refrigerators in Engineering Laboratory, including one of each capacity rating.

Task 2.2.3 Conduct and evaluate results of 6 month field demonstration.

Make arrangements for distribution of refrigerators in field. Intended distribution plan is to place 500 samples in cooperation with HUD in public housing projects in various parts of the country with different climatic conditions, and to distribute 800 with regular White-Westinghouse sales channels. Arrangements are to include keeping records of the location of each sample in the field, and provision for obtaining a monthly report on each. The report will note any service failures in the field, and will also supply information on the user's impression on noise level and general performance. Any compressors which fail in service will be returned to Columbus for analysis.

In addition, 20 high efficiency samples and 20 similar current production models are to be installed in locations where their performance can be monitored more closely, such as CPC employees' homes in the Columbus area. These samples will be provided with instrumentation to record monthly power consumption.

Task 2.2.4 Refine Phase I market analysis data by the integration of data obtained from sales of 800 samples through normal distribution outlets.

Task 2.2.5 Prepare and submit monthly progress reports covering manufacturing, marketing, and field demonstration experience in the six month field test, including Tasks 2.2.1 through 2.2.4, inclusive.

Task 2.3 Prepare and submit a comprehensive report covering all aspects of the Phase II program through the completion of the six month field demonstration. (To be an updated compilation of earlier reports.)

Task 2.4 Conduct, evaluate, and report on long term field demonstration. (Thirty months beyond initial six month test.)

Task 2.4.1 Collect data on user reaction and on comparative power consumption from 40 samples provided with kilowatt hour meters, on a monthly basis. Obtain information on general performance and reliability from other users on a quarterly basis, by mail survey, plus telephone follow up.

Task 2.4.2 Prepare and submit monthly progress reports on the results of the long-term field demonstration program.

Task 2.5 Prepare and submit annual reports on the Phase II long-term field demonstration program. Each annual report shall include cumulative results up to the time of issue.

APPENDIX "C"

REVIEW OF THE GENERAL ASPECTS OF COMPRESSOR DESIGN

The purpose of the compressor in a refrigeration cycle is to increase the pressure of the vapor so that its corresponding condensation temperature will be above the temperature of condenser cooling medium. Refrigerator and freezer condensers are air cooled so the heat rejected by the condenser is transferred to the ambient air in the kitchen. Accomplishing this pressure build-up with the least expenditure of energy is the task of the compressor.

There are two basic types of mechanical refrigeration compressors. These are the rotary and the reciprocating. The advantages of each of these design types have long been debated by compressor engineers. Both rotary and reciprocating compressors have been manufactured for many years for refrigerator and room air conditioner application.

The rotary compressors* are characterized by their circular or rotary rather than reciprocating motion. Two common types of rotary compressors are the rolling piston and the rotating vane. The rolling piston type employs a roller mounted to an eccentric shaft and a stationary, spring loaded blade mounted in a circular housing. The blade, which acts as a separator between the high and low side, reciprocates as the roller rotates. In the other type of rotary compressor, the blades rotate in an off center rotor against a stationary housing. In both cases internal leakage is controlled by hydrodynamic sealing. Rotary compressors are characterized by high volumetric efficiencies because of the small reexpansion volumes associated with their design geometries. Frictional losses associated mostly with the blade action, are believed to be greater hence the rotary compressors do not have a particularly high energy efficiency ratio relative to reciprocating types.

* ASHRAE Handbook and Product Directory Equipment (New York: Society of Heating, Refrigeration and Air Conditioning Engineers, Inc. 1975) PP 12.10 - 2.11.

The apparent simplicity of a rotary and its favorable size in relation to the driving motor diameter have always looked attractive to reciprocating compressor manufacturers. These advantages, however, are offset by the high degree of precision required to produce many of the component parts of the rotary.

A reciprocating compressor, as the name implies, compresses refrigerant gas by the action of a reciprocating piston. There are two types of piston/connecting rod mechanisms commonly used to transfer the rotary motion of the crankshaft. The "Scotch Yoke" incorporates a piston assembly and slide block and has been used for many years by several manufacturers. The great majority of compressors in the market place utilize the common connecting rod and wrist pin arrangement.

The path of refrigerant gas through a hermetic compressor can best be described by starting with the suction gas as it enters the hermetic shell through a tube at a point very close to the suction muffler entrance. After traveling through the suction muffler, the gas fills the suction plenum in the cylinder head. When the suction valve opens, gas is taken in the cylinder through the ports in the valve plate. When the piston is just beyond bottom dead center (BDC) the suction valve closes and the high pressure discharge gas is pumped out through the discharge muffler and coiled discharge tube. From the discharge tube exit, the refrigerant gas continues on through the system.

This cycle can be traced on Fig. A-1 shown on the next page. This figure is a schematic representation of a reciprocating compressor cycle. The BDC and top dead center (TDC) positions are indicated on the sketch. At point A the cylinder pressure is below suction pressure and the suction valve is open. The compression process is described by curve BC. At point B the suction valve closes because of rising cylinder pressure and thereafter the cylinder volume decreases and pressure increases. At point C the cylinder pressure exceeds the discharge pressure, thus forcing open the discharge valve. At point E the gas is again at discharge pressure and the suction stroke begins. Curve EF describes the gas expansion until the pressure drops below the suction pressure enough to force open the suction valve. The schematic representation shown in Fig. C-1 (crank angle vs. cylinder pressure) is very close to a typical oscilloscope trace which can be obtained by mounting a pressure transducer in the cylinder of a reciprocating compressor.

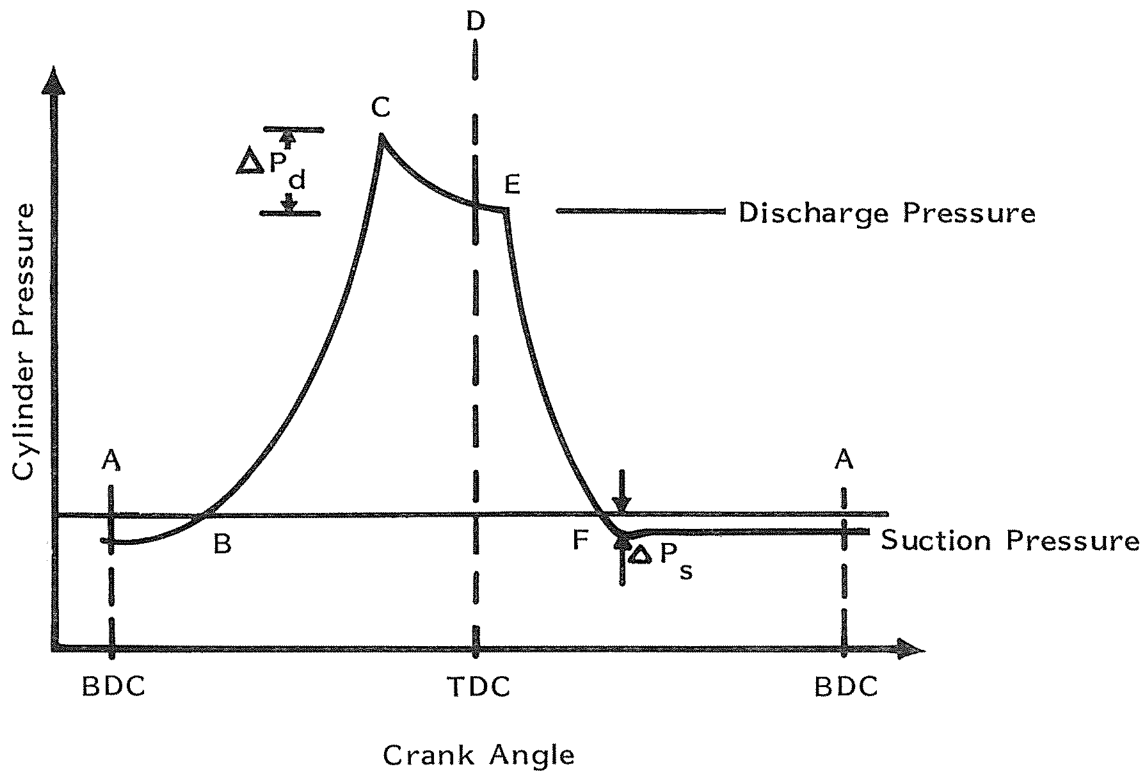


Figure C-1 Reciprocating Compressor Cycle

APPENDIX "D"

REVIEW OF REFRIGERATOR AND COMPRESSOR PERFORMANCE INTER-RELATIONSHIPS

The performance efficiency of a refrigerator or freezer is measured in terms of the energy consumption of the entire system rather than the compressor alone. The efficiency ratio, can be both a measure of refrigeration, as well as compressor efficiency. Referring to Fig. B-1 the basic refrigeration cycle, the refrigeration effect 4 to 1 is the amount of heat absorbed in the evaporator. The compressor cycle 1 to 2 is the work input. Thus, the energy efficiency ratio can be expressed as:

$$EER = \frac{\text{refrigeration effect}}{\text{work input}} = \frac{h_1 - h_4}{h_2 - h_1}$$

It should become apparent, however, that Fig. D-1 represents an ideal refrigeration cycle. Actually, losses occur during each process. Fig. D-2 more closely represents the actual refrigeration cycle. As before, the first process from 1 to 2 is compression. If the compression is truly isentropic, there is no deviation from constant entropy. Actually losses occur during the compression process and it is precisely these losses which are examined in detail in Section 5 of this report. The process from 2 to 5 is where heat rejection occurs. The losses during this process occur as a result of both pressure and temperature gradients. The pressure drop from 2 to 3 approximate losses through the compressor discharge ports and the subcooling from the saturation line at 4 to point 5 related to suction line capillary heat exchange. The process 6 to 1 describes the addition of heat and again, as in the condenser, there are losses due to pressure or temperature gradients. Because of suction line and capillary tube heat exchange as well as suction gas pre-heating occurring in the compressor, superheating of the suction gas occurs reducing the gas density. The EER of this actual cycle is:

$$\text{EER} = \frac{\text{refrigeration effect}}{\text{work input}} = \frac{h_7 - h_6}{(h_2 - h_1) + W_e}$$

In the above equation* W_e represents electrical energy used by refrigerator components other than the compressor. Thus, the compressor coefficient of performance reduces to:

$$\text{COP}_c = \frac{\text{refrigeration effect}}{\text{work input}} = \frac{h_7 - h_6}{h_2 - h_1}$$

It should be noted that the test conditions under which the performance efficiency of a refrigerator is determined are quite different from those under which the compressor is run. Thus, the enthalpy values and, therefore, the EER number obtained are not the same.

The operating test conditions for compressors are set at an arbitrary level and do not deviate as in actual refrigerator applications. The condensing temperature, which is the discharge temperature of the compressor, is adjusted to provide a standard setting. The expansion valve of the calorimeter is also adjusted to a standard evaporator temperature. The standard values used by CPC and most of the industry are shown in Table 1-A on page 164.

*J. Benjamin Horvay "Household Refrigerator and Food Freezer Performance Characteristics". Proceedings of the Confrencion Major Home Appliance Technology for Energy Conservation U.S. Dept. of Energy Div. of Buildings and Community Systems Feb. 27, March 1, 1978 pp. 91-95

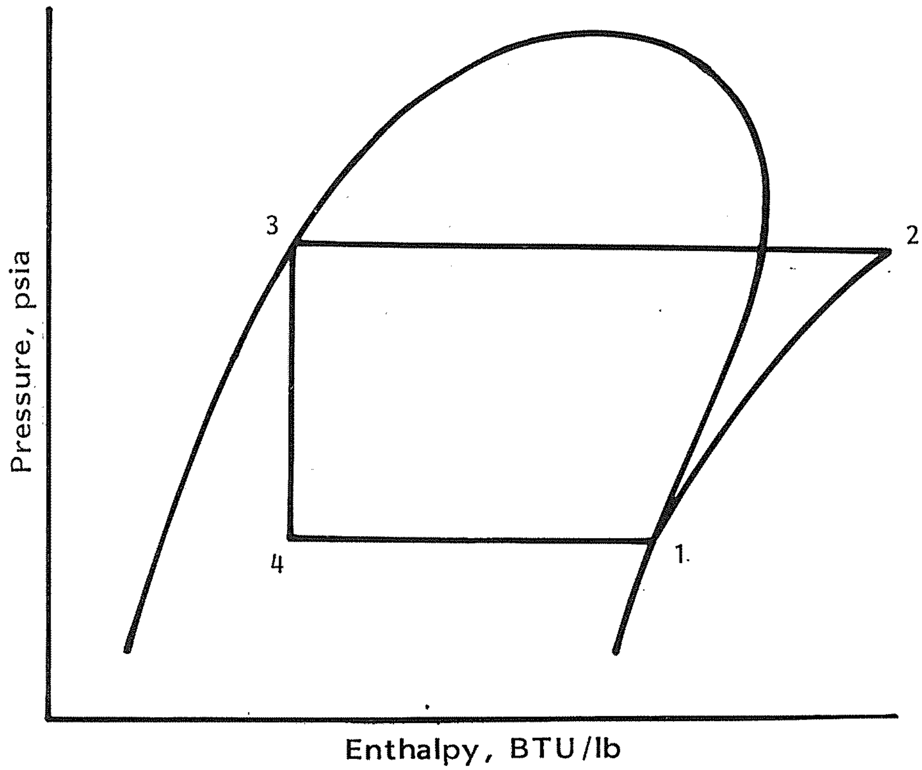


Figure D-1 Ideal Refrigeration Cycle

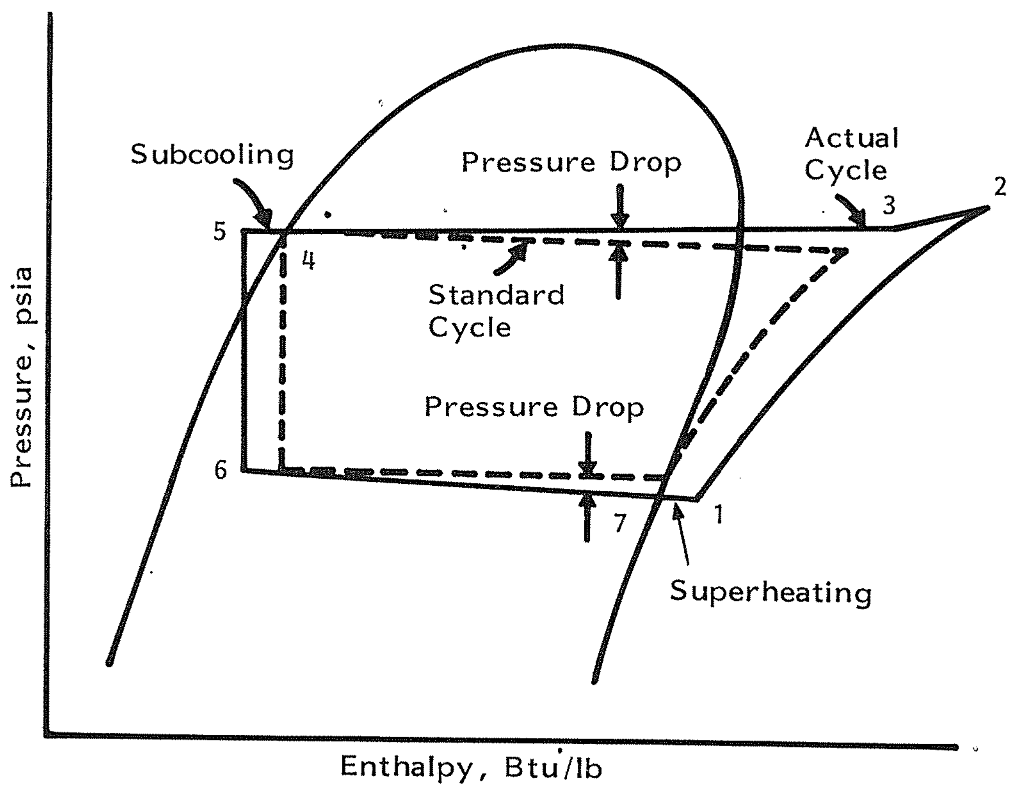


Figure D-2 Actual vs Ideal Refrigeration Cycle

TABLE 1-A

Standard Rating Point for Refrigerator Compressors

Compressor Type Refrigerant	Low Back Pressure P-12
Evap. Temp. (°F)	-10
Evap. Press (psig)	4.5
Cond. Temp. (°F)	130
Cond. Press. (psig)	181
Return Gas °F	90
Liquid to Exp. V. °F	90

The calorimeter test is the principle test used to determine compressor performance. This test was used repeatedly for the evaluation of new design ideas for individual component changes and for the final evaluation of the high efficiency compressor prototypes. Only after compressors performed well on the calorimeter were they assembled to a refrigerator and the combination evaluated as a system.

APPENDIX "E"

CONFIDENCE LEVELS IN HERMETIC REFRIGERATOR COMPRESSOR TESTING

OVERVIEW

When a new or modified design of motor-compressor assembly for refrigerated appliances is being developed, it is important to know how closely it conforms to the design objectives. Later, when the new design is put into quantity production, it is equally important to know the range of variation which may be expected in the product. These considerations are especially significant in the case of compressors designed for the purpose of reducing energy consumption.

COMMON COMPRESSOR RATING PARAMETERS

The most important characteristics used to evaluate compressor performance is the EER in BTU per watt hour. This figure gives a measure of the energy input required to produce a unit of refrigeration effect. Other compressor characteristics, such as watts input, capacity in BTU per hour, noise level and starting voltages, provide useful information for the designer, but are of less importance to the customer. The remainder of this discussion of confidence levels will be mainly confined to EER testing. The values of EER to be discussed are those measured on a compressor calorimeter. It has been found that there is a good degree of correlation between these results and the performance of the compressor in a complete appliance. The apparatus used for calorimeter testing and for other compressor tests are described elsewhere.

COMPARISON OF LABORATORY AND FACTORY TESTING

During a compressor development program, a limited number of samples are built in the engineering laboratory. In general, all of these are calorimeter tested. The assembly and testing work is performed by experienced technicians. As a result, variations are expected to be less than in later factory production, but the number tested is relatively small.

In regular production, on the other hand, large quantities of compressors are assembled, and a wider range of variation in parts will be experienced. At the same time, calorimeter testing is performed on a small number of samples as a quality control procedure. Only about 0.3% of the quantity produced is tested under normal conditions. There is likely to be more test variation in the factory tests because time is not available for analyzing and repeating the tests. Although the sampling percentage in factory testing is small, the daily testing program eventually builds up data on a large population of samples.

In the course of assembly in production lots each compressor is subjected to a simple test designed to eliminate those which do not run or which have gross defects, but these 100% tests do not provide any quantitive information on performance levels. The test is performed by connecting the pump assembly to a source of electrical power and verifying that it starts, pumps and is free of leaks.

CAUSES OF VARIATIONS IN TEST RESULTS

Variations in calorimeter test results arise from two causes. The first consists of errors in instrument readings and procedure during testing. This class of variation would be present if all the samples tested were identical. The second cause of variation is to be found in the manufacturing tolerances in the compressor components. This class of variation would be observed in the test results even if the calorimeter test procedure were error-free.

In the following paragraphs quantitative values are assigned to the major recognized causes of variable results. These values are then used to forecast their effect on calorimeter test results, and to correlate these figures with test data currently available at CPC.

COMMERCIAL COMPRESSOR RATINGS

It has been common practice among manufacturers of small hermetic compressors to publish capacity ratings without any tolerances. A tolerance of ± 5% has usually been assumed. From practical statistical considerations it can also be assumed that the tolerances mean that approximately 95% of the product will fall within the ± 5% range, since a distribution with 100% compliance is improbable.

With a normal distribution, approximately 2.5% of the product will be above the upper band limit and 2.5% beyond the lower limit. In case of appliance efficiency ratings, the user will normally be concerned only about the lower 2.5%.

STATISTICAL ASSUMPTIONS

In estimating the effect of variations in component parts on compressor pumping capacity, the following assumptions will be made:

a) The manufacturing operations on the parts are under control statistically.

b) The actual dimensions of the parts display a normal distribution, symetrically located about the nominal figure.

c) For any particular dimension or performance parameter approximately 95% of the product will fall within a band width of twice the standard deviation on either side of the normal value.

d) The cumulative effect of several variations can be estimated by the root-sum-square method.

That is, if a, b, c, d, e, etc. represent a number of variations, expressed in consistent units, the cumulative effect will be:

$$\sqrt{a^2 + b^2 + c^2 + d^2 + e^2 ---------.}$$

EFFECT OF COMPONENT TOLERANCE ON COMPRESSOR PERFORMANCE

In order to estimate the expected variation in compressor EER, it will be necessary to consider first the effect of component part tolerances on capacity, then the range of variation in motor efficiency, and finally the cumulative effect of both.

For simplicity, the following estimates will be confined to the CPC compressor model T52, which has a rated capacity of 780 BTU per hour. Variations in capacity are caused principally by changes in clearance, or reexpansion volume.

The items which contribute to clearance volume are tabulated in the following table. The information in the Table is arranged as follows:

Column 1. Identification of clearance volume segment.
Column 2. Maximum volume of segment, based on drawing tolerances.
Column 3. Minimum volume of segment, based on drawing tolerances.
Column 4. Spread, from maximum to minimum.
Column 5. Spread of segment, as decimal fraction of total.
Column 6. Fraction in Column 5., squared.

COL. 1	2	3	4	5	6
CLEARANCE SEGMENT	MAXIMUM CU. IN.	MINIMUM CU. IN.	SPREAD CU. IN.	FRACTION	$(FRAC.)^2$
GASKET	.00541	.00379	.00162	.16701	.02789
VALVE RELIEF	.00008	0	.00008	.00824	.00006
PISTON	.00624	.00239	.00385	.39691	.15753
SUCTION TREPAN	.00185	.00029	.00156	.16082	.02586
SUCTION VALVE	.00312	.00312	0	0	0
DISCHARGE PORT	.00583	.00372	.00211	.21753	.04731
VALVE STOP	.00216	.00168	.00048	.04948	.00248
TOTAL	.02469	.01499	.0097	.99999	.26113

The sum of the squares of the fractional tolerance spreads in the table is .26113. The square root of this total, or .51101 is a measured of the probable accumulated range of clearance for 95% compliance. A figure of .511 will be used in subsequent calculation.

An estimate of volumetric efficiency may be calculated from the formula:

$$V.E. = 1 + C - C\left[\frac{P_2}{P_1}\right]^{\frac{1}{k}} \quad \text{(or } 1/k\text{)}$$

Where V.E. = volumetric efficiency

C = clearance, as a decimal fraction obtained by dividing clearance volume by displacement

P_1 = suction pressure 19.2 psia

P_2 = discharge pressure = 195.7 psia

k = ratio of specific heats of refrigerants

For refrigerant 12, $1/k = .87$ and $(P_2/P_1)^{1/k} = 7.535$

and $V.E. = 1 - 6.535C$

The capacity ranges shown in the table below are calculated from the above relationship for the full tolerance spread, and also for the spread reduced by the factor of .511 to give the probable range for 95% compliance.

	FULL RANGE			PROBABLE RANGE		
CLEARANCE CONDITION	MAX.	NOM.	MIN.	MAX.	NOM.	MIN.
CLEARANCE CU. IN.	.02469	.01984	.01499			
CLEARANCE "C" (DECIMAL FRACTION	.04779	.03840	.02901	.04320	.03840	.03360
V.E.	.688	.749	.810	.718	.749	.780
CAPACITY, EST. (BTU PER HR)	716	780	844	747	780	812

The above calculations for estimated capacity range do not take into account losses due to leakage past the piston and around the valves. The nominal capacity figure of 780 BTU per hour does allow for the average amount of leakage, since it is based on the average of a substantial number of calorimeter tests. The estimated capacity ranges in the Table, however, do not include an allowance for variation in leakage from sample to sample. Unfortunately, the variation in leakage is not susceptible to calculation, since it is dependent upon surface conditions which cannot readily be measured or related quantitatively to the rate of leakage. A comparison of the capacity ranges in the preceding table with the actual calorimeter test data, to be discussed later, indicates that the probable effect of leakage is to increase the capacity band by 15 BTU per hour in both directions for the full range of variation and ± 8 BTU per hour for the probable range.

VARIATIONS IN POWER CONSUMPTION

The power consumption of a motor-compressor assembly is affected by variations in motor efficiency. Motor efficiency, in turn, is affected by the various components used. Variations in the items listed below are considered to be most significant.

a. Wire diameter
b. Lamination grade
c. Air gap
d. Rotor skew angle
e. Rotor resistance (affected by voids in conductors and by aluminum conductivity)

The Athens Products Company, which supplies motors to CPC, advised that motor efficiency is expected to have a commercial tolerance of ± 1.0%. As in the case of compressor ratings, this is assumed to mean that 95% of production will fall within a band width of 1.0% on either side of the nominal value. Since the efficiency is measured at a fixed load or torque point, the range of variation in watts input will also be ± 1.0%.

The power input required by a motor-compressor unit is also affected by variations in mechanical friction losses. This factor resembles the factor of leakage loss in the amount of variation and can neither be calculated nor measured directly. Based upon an analysis of calorimeter test results, to be discussed later, the probable variation due to friction is ± 16 watts for the full range of variation or ± 8 watts for a probable range.

ACCURACY OF CALORIMETER TESTS

A method of estimating calorimeter test accuracy for compressor performance has been previously published* with these results, capacity $\pm$ 0.87%, watts $\pm$ 1.0% and EER $\pm$ 1.3%. While the author cautions that the data for sensitivity used are for illustrative purposes only, they appear to be typical of common practice and the estimate of $\pm$ 1.3% will be used. This is understood to apply to engineering laboratory testing. For factory tests, the spread is expected to be larger.

COMBINED UNCERTAINTY FIGURES

According to Table II, the probable range of capacity in BTU per hour volume is 747 to 812 to cover 95% output, including $\pm$ 8 BTU for the estimated effects of leakage, gives a range of 739 to 820. This is equal to a spread of $\pm$ 5.2%. Combined with $\pm$ 0.87% for calorimeter uncertainty gives $\pm$ 5.3%, calculated as $\sqrt{5.2^2 + .87^2}$.

The probable range of watt readings may be estimated as follows:

Nominal watts	225
Motor $\pm$ 1.0% =	$\pm$ 2.3 watts
Friction, est.	$\pm$ 8.0 "
	$\pm$ 10.3 = $\pm$ 4.6%

Combined with $\pm$ 1.0% for measurement uncertainty gives $\pm$ 4.7%, $4.6^2 + 1^2$. On this basis, the estimated range of EER values for 95% compliance is $\sqrt{4.6^2 + 4.7^2} = \pm$ 6.6%.

COMPARISON WITH FACTORY QUALITY CONTROL TEST RESULTS

The available test results for capacity can be compared most directly with the forecasts in the preceeding paragraphs. Using

*J. L. Schlafer, "Procedure for The Error Analysis of a Secondary Refrigerant Compressor Calorimeter" Proceedings of The 1978 Purdue Compressor Technology Conference a nominal capacity of 780 BTU per hour and a range of ± 5.3% gives limits of 739 and 821. In comparison, data for 73 model T52 samples tested in 1979, show only one below 739. The distribution of results for 490 tested in 1974 has been preserved. In this case, the agreement is not as good. The spread in readings must be increased to about 9% to insure 95% compliance. It is suggested that the difference reflects an improvement in manufacturing consistency.

Test data for watts input cannot be related directly to the estimated values because the watts depend on pumping capacity of the compressor as well as on motor characteristics. The 1979 data for EER in BTU per watt hour is in very good agreement with the forecast spread of ± 6.6% for 95% compliance, if the range is computed from the mean value of about 3.20. This later figure however, is not in good agreement with the generally used nominal of 3.47. It appears that the consistency of the data is as expected, but that the nominal value was somewhat optimistic for the T52 model. This problem will not occur on a new model if a realistic value is chosen for the nominal EER.

The capacity data for 1974 show a total range of 166 BTU per hour, equivalent to ± 83. The estimated total spread, due to variations in clearance volume, from Table II, is ± 64 BTU per hour. It is somewhat arbitrarily assumed that of the 19 BTU, 15 are the result of variations in leakage, and the remainder to unrecognized variations. The 15 BTU figure was used previously. In a similar manner, the variation in watts due to friction, quoted as ± 16 watts was arrived at.

COMPARISON WITH RECENT ENGINEERING LABORATORY TEST DATA

A group of 4 samples of compressor model W80 was assembled and tested in the Engineering Laboratory in March, 1980. Model W80 is a high-efficiency design having the same displacement as the former production model T78. In each case the test was repeated after additional run-in time. The test results are shown in this table.

SAMPLE NUMBER	RUN-IN TIME HR.	CAPACITY BTU PR. HR.	MOTOR WATTS	EER BTU PR. W/HR.
1013-1A	1	814.3	164.4	4.95
"	21	824.	160.8	5.12
"	22	809.	162.8	4.96
"	22.5	804.	160.8	5.00
1013-2A	3.5	804.	168.4	4.77
"	20	821.6	167.2	4.91
1013-3A	3.5	817.6	165.2	4.95
"	6	801.3	165	4.96
"	24	814.6	166	4.91
1237-B	2	838.	168.8	4.96
"	19	838.	162.4	5.16
"	20	824.4	162.4	5.08
MEAN		817.6	164.5	4.97

Repeated tests on the same sample show slightly more spread than was forecast in the paragraph on the accuracy of calorimeter tests. Departures from the mean however, fall well within the band width predicted as they should for a limited number of samples assembled and tested under laboratory conditions.

CONCLUSIONS

Motor-compressor assemblies of the basic design and displacement of models T52 or W80, in production quantities, may be expected to have EER, as measured on a calorimeter, such that 95% will lie within $\pm$ 6.6% of the mean or nominal value. Only 2.5% of production may be expected to fall below a level 6.6% below the nominal. (i.e., 4.67).