ENERGY

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ORNL/Sub/78-7229/5

Research and Development of Energy-Efficient High Back-Pressure Compressors

Marc G. Middleton

R. Stuart Sauber

KELVINATOR COMPRESSOR COMPANY
1545 Clyde Park, S.W.
Grand Rapids, Michigan 49509

Work performed for

OAK RIDGE NATIONAL LABORATORY
operated by
UNION CARBIDE CORPORATION
for the

U. S. DEPARTMENT OF ENERGY

OFFICE OF BUILDINGS ENERGY
RESEARCH AND DEVELOPMENT

Printed in the United States of America. Available from
National Technical Information Service
U.S. Department of Commerce
5285 Port Royal Road, Springfield, Virginia 22161
NTIS price codes—Printed Copy: A05 Microfiche A01

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ORNL/Sub/78-7229/5
Dist. Category UC-95d

RESEARCH AND DEVELOPMENT OF ENERGY-EFFICIENT HIGH BACK-PRESSURE COMPRESSOR

SEPTEMBER 1983

Marc G. Middleton
R. Stuart Sauber

Report prepared by

KELVINATOR COMPRESSOR COMPANY
1545 Clyde Park, S.W.
Grand Rapids, Michigan 49509

under
Subcontract 7229
for
OAK RIDGE NATIONAL LABORATORY
Oak Ridge, TN 37830

operated by
UNION CARBIDE CORPORATION
for the
U. S. DEPARTMENT OF ENERGY
under Contract No. W-7405-eng-26

PREFACE

This report describes the work performed in a compressor development program conducted by Kelvinator Compressor Company, a Division of White Consolidated Industries, Incorporated. This work was performed for Oak Ridge National Laboratory which is operated by Union Carbide Corporation for the U.S. Department of Energy. This project was primarily concerned with the development of improved energy-efficient compressors for high back-pressure applications.

Initially, Columbus Products Company, also a Division of White Consolidated Industries, Incorporated, worked on a project to improve the efficiency of compressors for refrigerator-freezers and room air conditioners (RACs). This led to the work in the high back-pressure applications. As worked progressed, it became obvious that some improvements worked equally well in all these applications.

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ACKNOWLEDGMENTS

This work was conducted with the support of the Office of Buildings Energy Research and Development of the U.S. Department of Energy through Oak Ridge National Laboratory. The project technical monitor at Oak Ridge was Mr. Edward A. Vineyard.

Work performed in the market study was supervised by Mr. Henry Preher of Americold Compressor Company. The motors used in this development program were designed by Mr. Lloyd B. Wilson of Athens Products Company.

Engineers involved in this contract at Kelvinator Compressor Company were Mr. Robert W. Earley and Mr. Marc G. Middleton. Mr. Robert D. Spalink contributed his time and energy in acquiring parts and services needed for the performance of this work. Mr. Ralph E. Niven supervised the construction and testing of the compressor. Mr. R. Stuart Sauber was the program manager. Mrs. Kathleen L. Warren typed this report.

ABSTRACT

Improved-efficiency compressors were developed in four capacity sizes. Changes to the baseline compressor were made to the motors, valve plates, and mufflers. The adoption of a slower running speed compressor required larger displacements to maintain the desired capacity. This involved both bore and stroke modifications. All changes that were made to the compressor are readily adaptable to manufacture.

Prototype compressors were built and tested. The largest capacity size (4,000 BTU/hr.) was selected for testing in a vending machine. Additional testing was performed on the prototype compressors in order to rate them on an alternate refrigerant.

A market analysis was performed to determine the potential acceptance of the improved-efficiency machines by a vending machine manufacturer, who supplies a retail sales system of a major soft drink company.

1. INTRODUCTION AND SUMMARY

The basic objective of the project was to investigate the technical and commercial feasibility of reducing energy consumption of high back-pressure compressors which are typically found in vending machines, dehumidifiers, and water coolers.

Two refrigerants, R-12 and R-500, are used with these compressors. R-12 has historically been used in all the above applications. Within the last five years, however, R-500 has been used in some dehumidifier applications. As it has not been economical to develop specific compressors for each of these refrigerants, the same compressors have been dual rated. For a given compressor rated for R-12, the use of R-500 will increase the capacity and motor torque requirements.

Through a program of development and testing of samples we found that an important improvement could be made to the energy-efficiency ratio (EER) of a hermetic compressor. The design, which we adopted, should be suitable for large quantity production without excessive manufacturing and tooling costs. In addition, it is expected that the cost per unit, assuming full production, would be reasonable.

The project was conducted along the lines of an evolutionary change to the production T-line compressor rather than a completely new design. This approach reduced the time and capital required for a start-up of production if desired. An additional benefit is that the amount of work required for verification of product reliability is greatly reduced.

This research and development effort would be useless if there was no potential for utilizing the reduced energy consumption compressor. Thus, an important part of the program was the market study in Section 2. Section 3 is a description of the baseline compressor used as a comparison. The process of testing the various design changes is detailed in Section 4. The results of applying the improved design to a vending machine may be found in Section 5 while Section 6 provides information on how the tests were conducted.

1.1. SUMMARY RESULTS

1.1.1. ENERGY CONSUMPTION IMPROVEMENT

Table 1-1 illustrates the EER improvement for the four models designed for this program.

TABLE 1-1

PRODUCTION		NEW DESIGN		
Model	EER (BTU/WHr)	Model	EER (BTU/WHr)	EER Improvement
TH147	7.0	WH145	8.97	28%
TH210	7.0	WH210	9.36	34%
TH310	7.0	WH310	9.06	29%
TH400	7.0	WH410	7.35	5%

1.1.2. APPLICATION EXAMPLE

The following is an example of the improvement in energy reduction on a typical beverage can cooler.

The vending machine is presently produced with an AE4441A compressor. The equivalent Kelvinator Compressor Co. model is the TH400. Table 1-2 displays the summarized results of the three compressors applied to the beverage can cooler.

TABLE 1-2
Can Vending Machine
Energy Consumption

Compressor	KWHr/Day	KWHr/Year	KWHr/Saved
AE4441A	8.09	2952.85	------
TH400	7.56	2759.40	193.45
WH410	6.97	2544.05	408.80

1.1.3. COST SAVINGS

Table 1-3 displays the cost savings in KWH/year as applied to a typical beverage can vending machine.

TABLE 1-3
Can Vending Machine
Cost of Operation

Compressor	Energy Consumption (KWHr/Year)	$.05/KWHr	Savings/Year
AE4441A	2952.85	$147.64	------
TH400	2759.40	$137.97	$ 9.67
WH410	2544.05	$127.20	$20.44

1.1.4. COST PAYBACK

The cost premium to apply the WH410 design to the machine is $9.00. The payback in energy savings is 5.3 months.

2. MARKET STUDY

The market study was conducted by the Engineering Department of Kelvinator Compressor Company. All participants were manufacturers of refrigerated vending machines and were located in different states across the U.S. With only one exception, the survey was done entirely by telephone.

2.1 METHOD OF EXECUTION

Based on the objectives of the project, a questionnaire was used to gather the necessary information. The document was titled such that it could be mailed to any participant who did not have the time to talk over the phone.

A list of possible participants was gathered from Mr. Henry C. Preher, Vice President of Sales, Americold Compressor Company, Cullman, Alabama, and The Air Conditioning, Heating, and Refrigeration New 1981 Directory Issue.

Each telephone conversation began with an introduction of the nature and title of the project. This included the 21% increase in EER and the increased cost of approximately $3.00 per compressor. At this point, the participant was asked to respond to the questionnaire shown in Figure 2-1.

2.2 RESULTS

Working through the list of possible participants, there were actually five firms whose product would use high back-pressure compressors in

U.S. DEPARTMENT OF ENERGY

SUBCONTRACT 7229

MARKET STUDY

1. Do you think this is a product your company would be interested in purchasing?

2. If yes, would you be able to give me any idea of a volume figure?

3. If this product were purchased, would it constitute an additional line of product for you or would it be incorporated into your present line?

4. Can you see this new product increasing sales volume?

5. Would you think the energy cost savings would be used to reduce the consumer cost or used as a sales incentive?

6. May I ask what your sales of refrigerated vending machines were for 1981?

Figure 2-1. Questionnaire

the capacity range of those investigated. All of these firms responded to the questionnaire. The responses to each question will be dealt with individually.

Number 1. Three respondents gave a definite affirmative response. The other two cited as reasons the additional cost to them, and the fact (in their opinion) that the industry as a consumer was more oriented toward initial price than energy cost savings.

Number 2. Only one participant could give an idea of volume; this was several thousand per year.

Number 3. Every respondent said they would incorporate such a compressor into their present line.

Number 4. Here, only one respondent saw any measurable increase a possibility. The other responses were negative, citing the increase in cost as the major reason.

Number 5. All participants emphatically negated the idea of a reduction in consumer cost, however, there was no clear response in favor of the sales incentive approach.

Number 6. Here, the responses are listed: 75,000, 40,000, 5000, 1000-5000, not allowed to divulge.

2.3 CONCLUSIONS

While 60% of the respondents stated they would be interested in purchasing higher energy efficient, high back-pressure compressors, only one of them could give a measurable volume to use as a potential market share percentage figure. The percentage of his use would be 40 - 50% of the 1000-5000 range.

Forty percent of the respondents replied that the industry is currently more oriented toward the actual purchase price than the possible energy cost savings. This was also reflected in the fact that every participant mentioned that because the consumer was paying for the energy, there was no great concern for energy savings by the manufacturer.

All participants responded that they would incorporate the energy-efficient compressor in their current model line. There was no concern for marketing the fact that the compressor has a better energy efficiency.

It is the opinion of the author that there is a need for an education on energy cost savings at the consumer level. If the owner or renter of the vending machine understood the difference, he might demand higher efficiency equipment from the manufacturer.

3. BASELINE COMPRESSOR

The existing T compressor has been engineered to fit a series of applications. These applications have generally been broken into three major classes; room air conditioning using R-22, low back-pressure systems (such as refrigerators and freezers) using R-12, and high back-pressure systems (dehumidifiers, water coolers, and vending machines) using R-12 and R-500. It is the third class of compressors that is used for the baseline for this project.

In Table 3-1 data is given for the high back-pressure T-line. Capacity is stated for the compressors using both R-12 and R-500. The EER values are approximate. On R-12, the compressors produce slightly lower EERs than they will on R-500.

TABLE 3-1

High Back-Pressure T-Line Data

Model	R-12 Capacity (BTU/Hr.)	EER (BTU/Whr.)	R-500 Capacity (BTU/Hr.)	Displacement (In^3/Rev)
TH147	1470	7.0	1650	0.269
TH210	2100	7.0	2500	0.369
TH310	3100	7.0	3600	0.515
TH400	4000	7.0	4600	0.687

These single-cylinder, reciprocating compressors have a vertical crankshaft containing a centrifical oil pump. The pump assembly is internally spring mounted inside a welded shell. Both the suction and discharge gas flow through mufflers. A cylinder head secures the

valve plate to the cylinder housing. The mufflers are also secured to the cylinder housing. A second housing contains the motor. The cylinder housing and the motor housing, when bolted together, form the basic pump assembly. Bearings in both housings provide support for the crankshaft. A simplified top view and cross section of a typical model are shown in Figures 3-1 and 3-2.

3-1. MOTORS

The motors used in the baseline compressors are all two-pole, split-phase designs. Typically, these motors operate at about 3440 rpm. Since these are two-pole motors and operate from a 60 cycle per second line source, the synchronous speed would be 3600 rpm. The exact speed of the compressors depends on the load and the supply voltage. All motors operate at 115 volts. Listed in Table 3-2 are load point data, torque characteristics, and locked-rotor amperes.

TABLE 3-2

Performance Characteristics of T-Line Motors

	LOAD POINT				TORQUE (oz-ft)		
Model	Torque (oz-ft)	Speed (RPM)	Eff (%)	Current (Amps)	Start	Pullout	Locked-Rotor Current (Amps)
TH147	3.0	3523	68.0	1.95	4.5	10.5	18.5[a]
TH210	4.7	3506	69.0	2.94	4.6	14.4	25.0[a]
TH310	8.1	3530	71.7	4.38	8.4	21.2	32.0
TH400	11.0	3480	71.2	6.10	9.6	31.0	40.0

a - 5 ohms in series with auxiliary winding

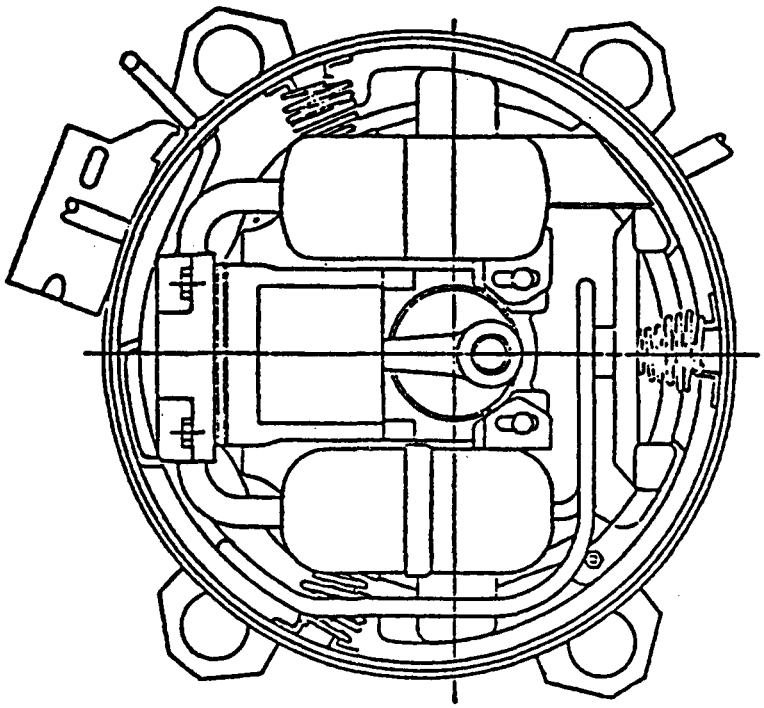

Figure 3-1. Top View With End Head Removed

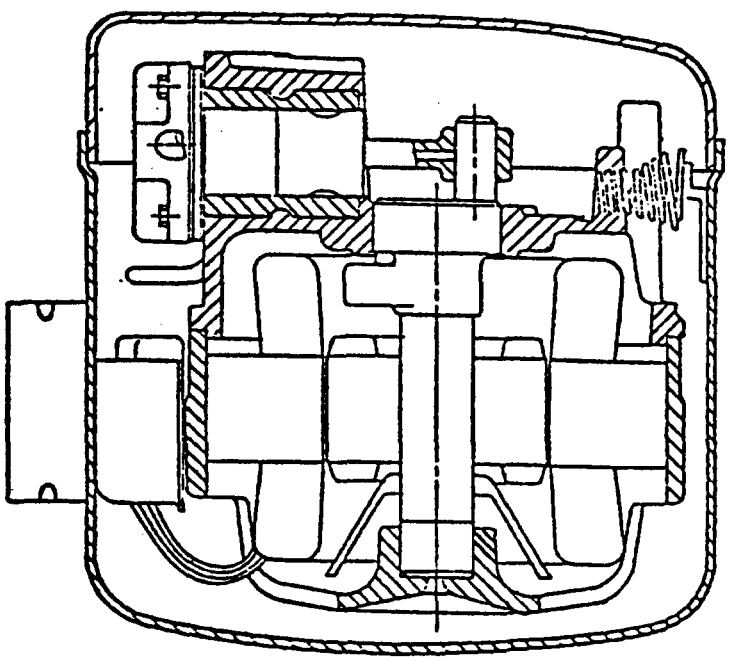

Figure 3-2. Cross Section of Typical Compressor

Locked-rotor current and starting torque for the TH147 and TH210 are with 5 ohms in series with the start winding. This is because these compressors are started with a positive-temperature-coefficient resistor (PTCR) which has a resistance of approximately 5 ohms. The TH310 and TH400 compressors are started with a current relay. Starting-torque data and locked-rotor current are given without any resistance since a current relay provides none.

Physically, the motors are similar. All of the winding end turns are blocked the same. Rotor diameter is the same on all models. The stator bolt hole diameter is 5.48 in. Stack heights vary according to motor size, as can be seen in Table 3-3.

TABLE 3-3

Motor Stack Heights

Model	Motor Stack Height (in.)
TH147	1.325
TH210	1.320
TH310	1.500
TH400	1.750

Stack heights are adjusted primarily for pullout torque-requirements and the corresponding characteristics that are necessary at the load point.

Motor-efficiency data and curves for these units are shown in Figures 3-3 through 3-24. The load point, indicated in these figures, is based on the motor manufacturer's rating point and is not the same point that the motor operates at during a calorimeter test at standard conditions.

DFT K. SHELI	APP. CODE	THERMO	ROT	L SPEC	REF. PA	STACK
ENGR			CCW	912525	P-5707	1.75 5CKB17KBFL

HERMETIC MOTOR IDENTIFICATION

ATHENS PRODUCTS CO.
2000 TELLICO AVE. ATHENS, TENN.
One of the White Consolidated Industries
WCI

STYLE NO 325P122
RATING .15 HP 115VOLTS 60 HZ
CUSTOMER COLUMBUS
CUSTOMER STYLE NO WGO - WH145

PART IDENTIFICATION

	CUSTOMER PART NO.	ATHENS PRODUCTS CO. PART NO.	OUTLINE DWG.
STATOR	C140904	5006S01	1C51F01
ROTOR	C140801	5503R01	1B29F05

MOTOR PERFORMANCE DATA

	MAIN	AUXILIARY
1ϕ RESISTANCE (25°C).....	4.99 ± 7 %	12.8 ± 7 %
TYPE WIRE.....	COPPER	COPPER
WIRE SIZE.....	20 AWG .0320 DIA	24.5 AWG .0190 DIA

3ϕ RESISTANCE (25°C) LINE TO LINE _____ AVERAGE ± _____ %

CAPACITOR: RUN 15 MFD. 170 VOLTS
START _____ MFD. _____ VOLTS

PROTECTOR: _____

LEADS: COMMON 18 AWG MAIN 18 AWG AUX 18 AWG

	COMMON	MAIN	AUX
VOLTS.....	115		
FREQ.....	60		
TEMPERATURE..... °C	75		
LOAD POINT (oz FT)	7		
AMPS.....	1.32		
WATTS.....	144		
RPM.....	1746		
EFF (%).....	75.3		
PF (%).....	94.9		
CAP. VOLTS...	161		

BREAKDOWN TORQUE (oz FT) 16.6
3000 RPM TORQUE (FT) _____

LOCKED ROTOR		4OHMS PARALLEL WITH RUN CAP
VOLTS.....	115	115
AMPS.....	7.6	12.4
TORQUE (oz FT)	2.4	5.8
CAP VOLTS....	125	16.0

RATE OF RISE
MAIN °F/SEC: .78
AUX °F/SEC: _____
ROTOR °F/SEC: _____

PERFORMANCE CURVES
673081
THRU
673085

RELAY DATA
P.U. _____ AMPS-VOLTS D.O. _____ AMPS-VOLTS
CONTINUOUS COIL VOLTS _____ MAX RUPTURE AMPS _____

325P122

APC-176

Figure 3-3. WH145 Motor Performance Data

CURVE NO. 673081

SPEED TORQUE & RELAY APPL:
126.5 VOLTS, 60 HZ 1 Ø
.15 HP TYPE FL
75°C WDG. TEMP.

P-5707
BOOK: 280709
DATE: 3-15-79
L-912525

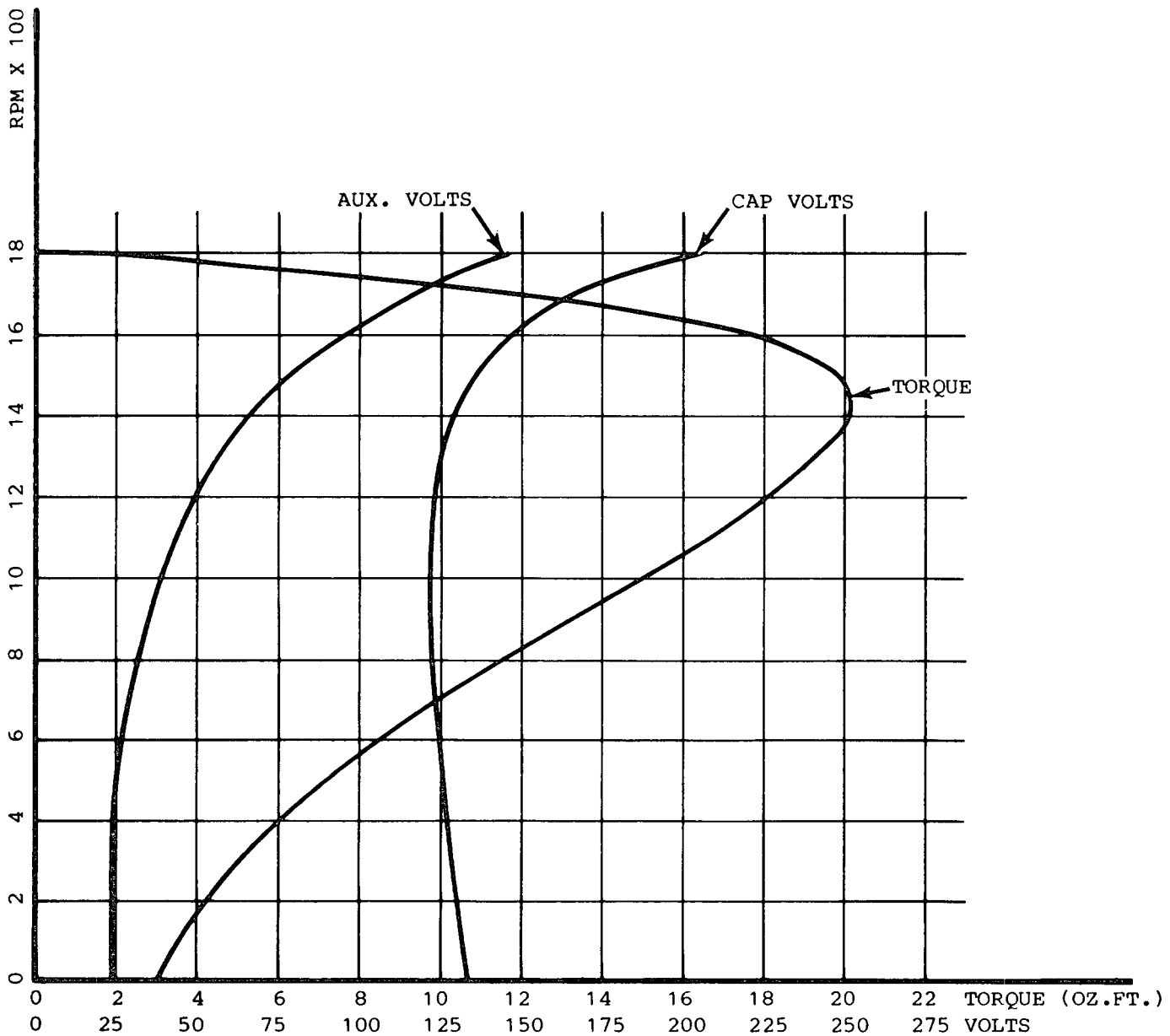

Figure 3-4. WH145 Performance Curve

CURVE NO. 673082

SPEED TORQUE & RELAY APPL:
115 VOLTS, 60 HZ 1 Ø
.15 HP TYPE FL
15 MFD. RUN CAP.
21-25 MFD. START CAP.
75°C WDG. TEMP.

P-5707
BOOK: 280709
DATE: 3-15-79
L-912525

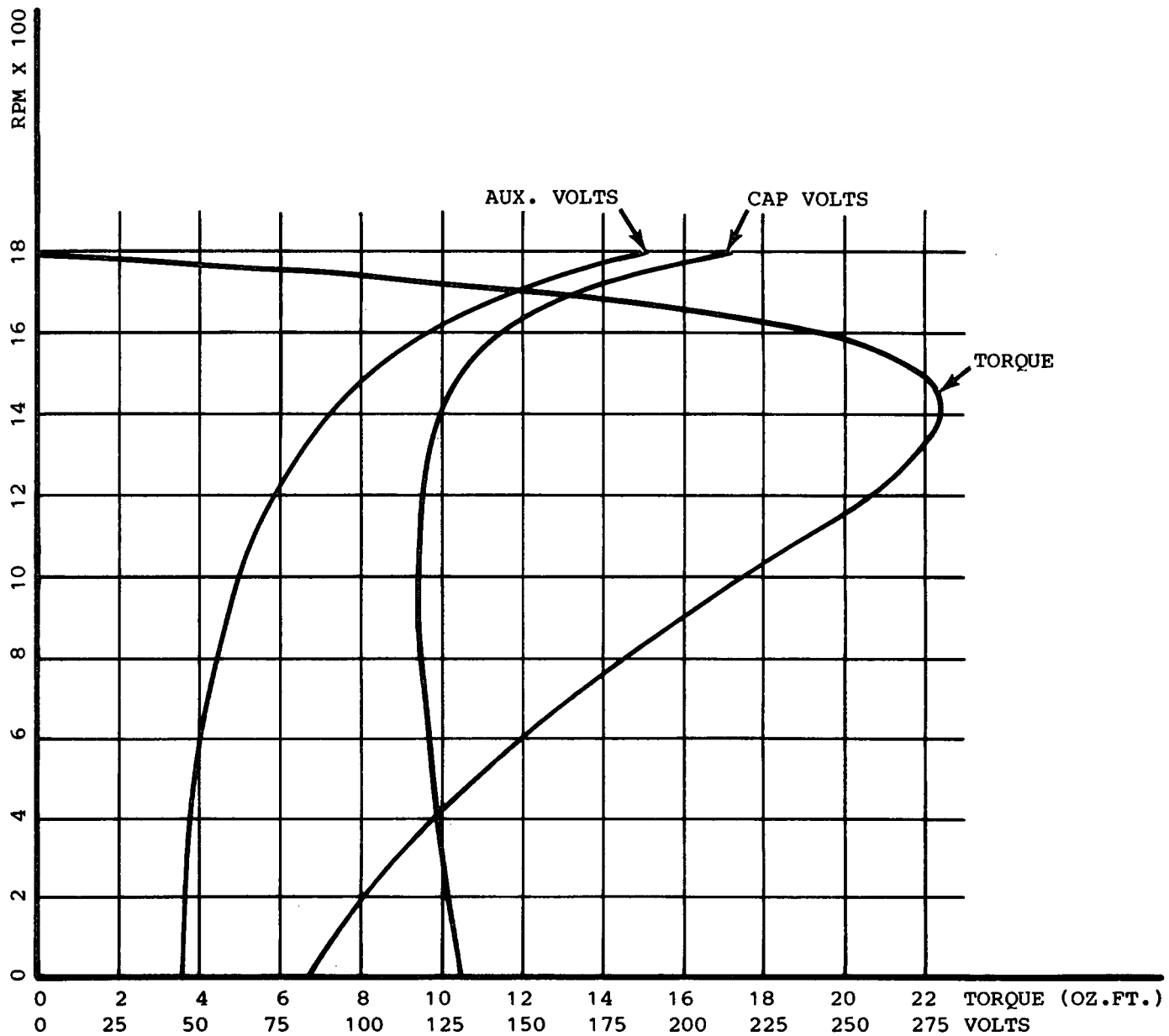

Figure 3-5. WH145 Performance Curve

CURVE NO. 673083

SPEED TORQUE & RELAY APPL:
115 VOLTS, 60 HZ 1 Ø
.15 HP TYPE FL
15 MFD. RUN CAP.
75°C WDG. TEMP.

P-5707
BOOK: 280709
DATE: 3-15-79
L-912525

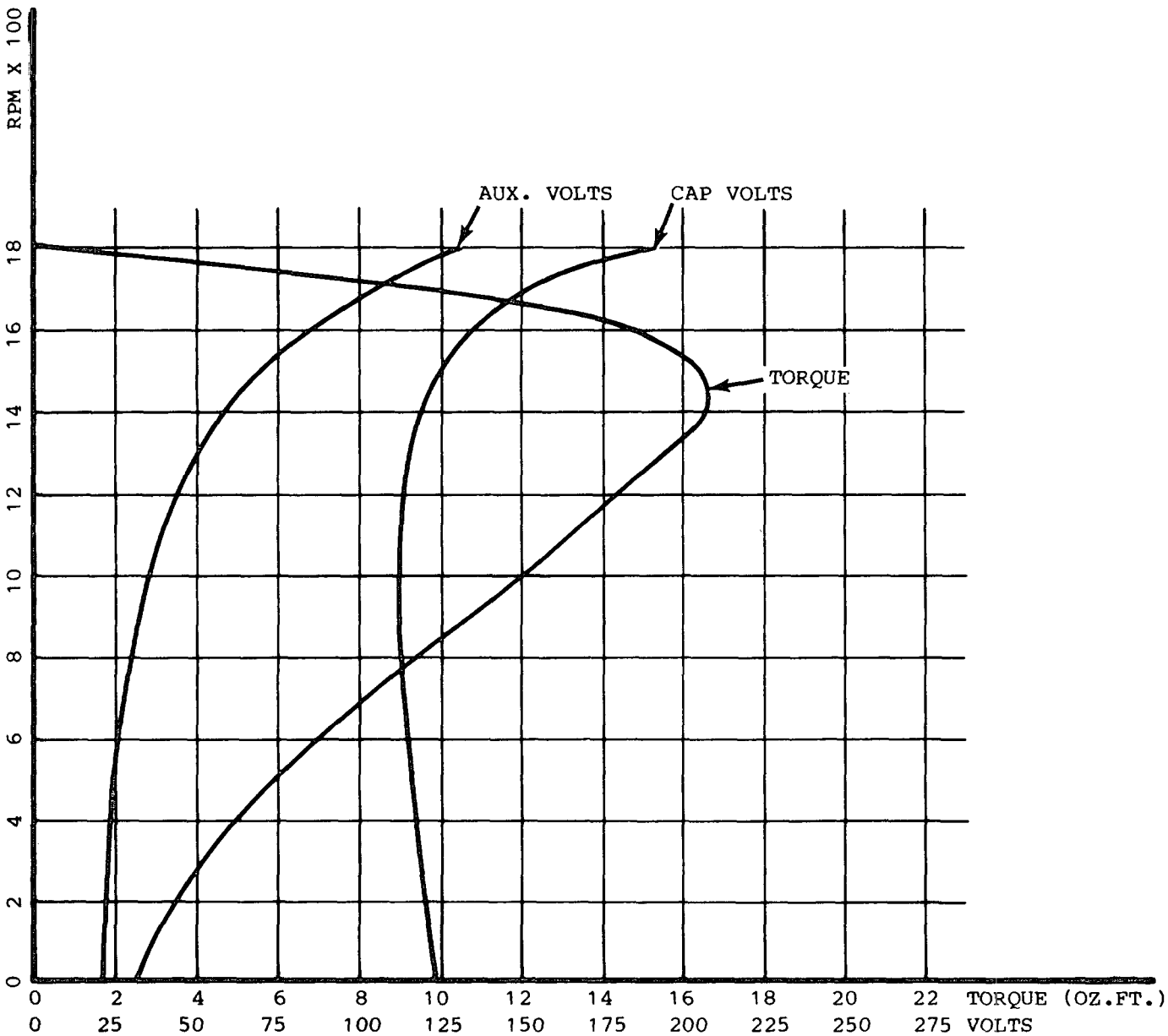

Figure 3-6. WH145 Performance Curve

PERFORMANCE TEST
115 VOLTS, 60 HZ 1 Ø
.15 HP TYPE FL
15 MFD. RUN CAP.
75°C WDG. TEMP.

P-5707
BOOK: 280709
DATE: 3-15-79
L-912525

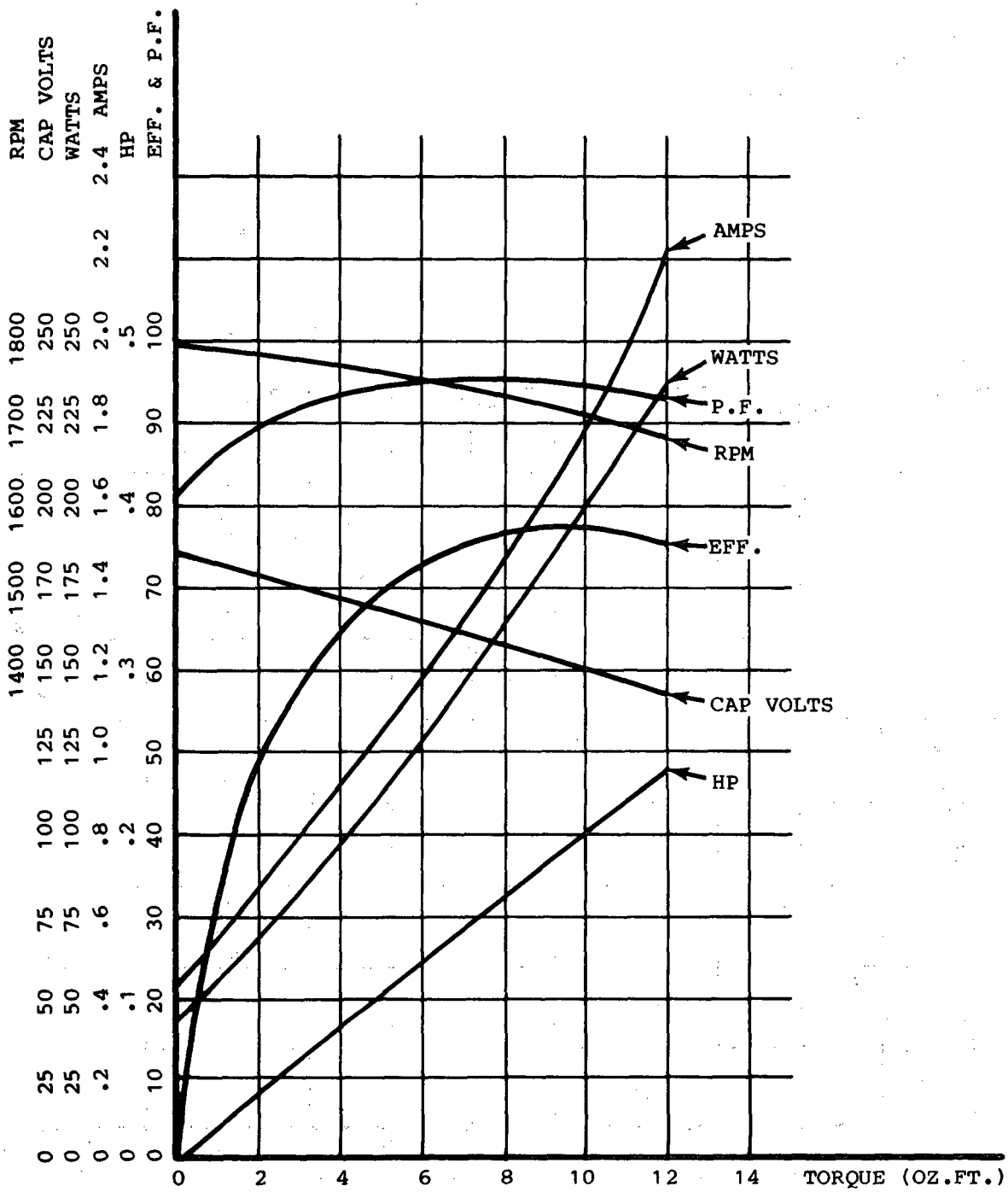

Figure 3-7. WH145 Performance Curve

CURVE NO. 673085

SPEED TORQUE & RELAY APPL:
95 VOLTS, 60 HZ 1 Ø
.15 HP TYPE FL
15 MFD. RUN CAP.
75°C WDG. TEMP.

P-5707
BOOK: 280709
DATE: 3-15-79
L-912525

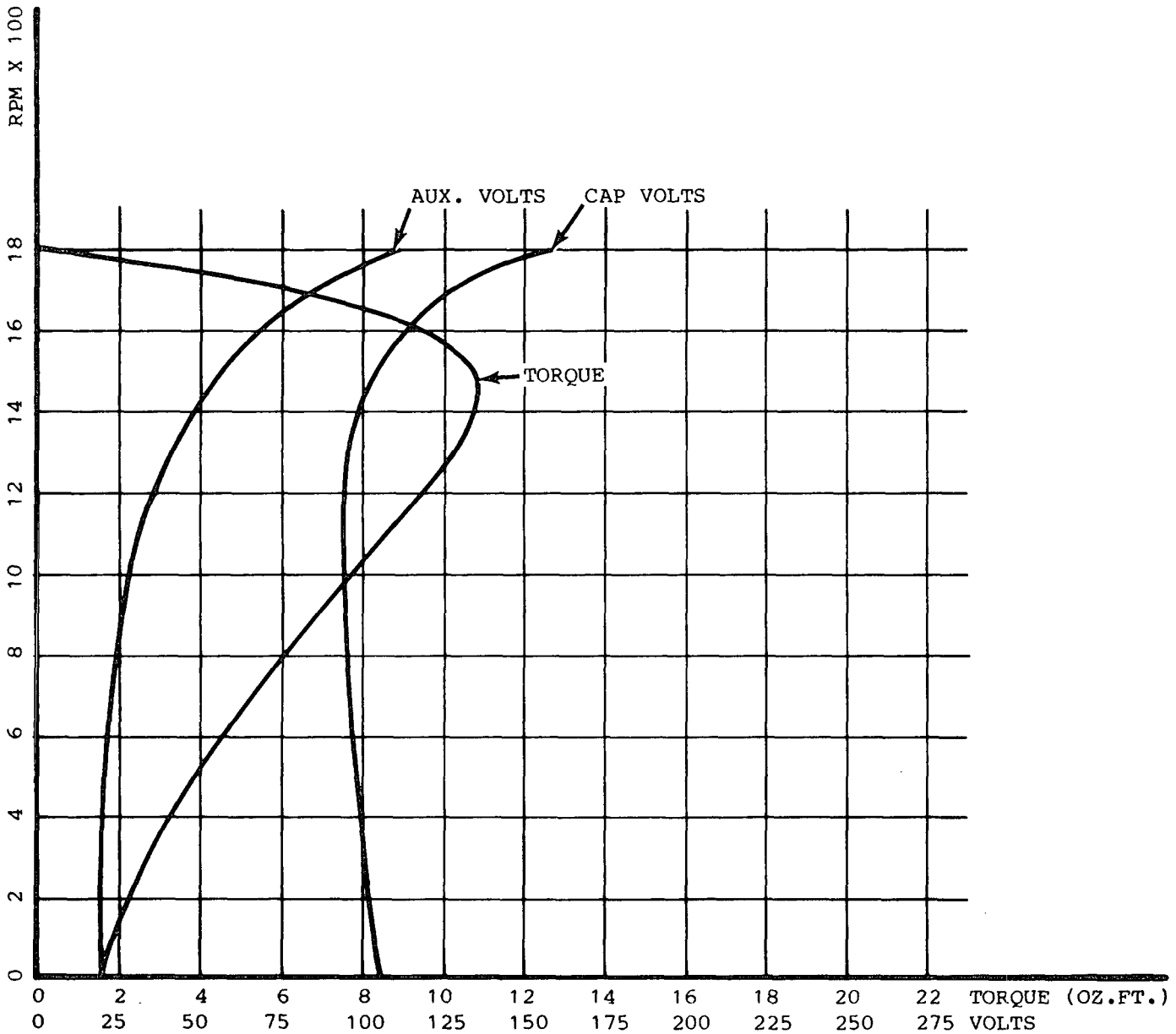

Figure 3-8. WH145 Performance Curve

DFT K. SHELL 4-1-80	APP. CODE	THERMO	ROT	L SPEC	REF. PA	STACK
ENGR. 4-3-80			CCW	912524	P-5706	1.75" 5CKB17KB

JED CUST. P/N'S

HERMETIC MOTOR IDENTIFICATION

ATHENS PRODUCTS CO.
2000 TELLICO AVE. ATHENS, TENN.
One of the White Consolidated Industries

STYLE NO 325P121
RATING .20 HP 115VOLTS 60 HZ
CUSTOMER COLUMBUS
CUSTOMER STYLE NO W80 - WH210

PART IDENTIFICATION

	CUSTOMER PART NO.	ATHENS PRODUCTS CO. PART NO.	OUTLINE DWG.
STATOR	C140903	5006S02	1C51F02
ROTOR	C140801	5503R01	1B29F05

MOTOR PERFORMANCE DATA

	MAIN	AUXILIARY
1ø RESISTANCE (25°C).....	3.91 ± 7 %	8.93 ± 7 %
TYPE WIRE.....	COPPER	COPPER
WIRE SIZE.....	19.5 AWG .0340 DIA	24 AWG .0201 DIA

3ø RESISTANCE (25°C) LINE TO LINE _____ AVERAGE ± _____ %

CAPACITOR: RUN 15 MFD. 170 VOLTS
START _____ MFD. _____ VOLTS

PROTECTOR: _____

LEADS:	COMMON 18 AWG	MAIN 18 AWG	AUX 18 AWG
VOLTS.....	115		
FREQ.....	60		
TEMPERATURE..°C....	75		
LOAD POINT (oz FT)	8		
AMPS.....	1.71		
WATTS.....	167		
RPM.....	1747		
EFF (%).....	75.2		
PF (%).....	83.9		
CAP. VOLTS...	149		

BREAKDOWN TORQUE (oz FT) 20.5
3000 RPM TORQUE (FT) _____

LOCKED ROTOR

		WITH 4OHMS PARALLEL WITH RUN CAP
VOLTS.....	115	115
AMPS. .4 SEC.	10.0	16.3
TORQUE (oz FT)	2.45	9.0
CAP VOLTS....	122	25

RATE OF RISE
MAIN °F/SEC: 1.54
AUX °F/SEC: _____
ROTOR °F/SEC: _____

PERFORMANCE CURVES
673076
THRU
673080

RELAY DATA
P.U. _____ AMPS-VOLTS D.O. _____ AMPS-VOLTS
CONTINUOUS COIL VOLTS _____ MAX RUPTURE AMPS _____

325P121

APC-176

Figure 3-9. WH210 - Motor Performance Data

CURVE NO. 673076

PERFORMANCE TEST
115 VOLTS, 60 HZ 1 Ø
.20 HP TYPE FL
15 MFD. RUN CAP.
75°C WDG. TEMP.

P-5706
BOOK: 280708
DATE: 3-14-79
L-912524

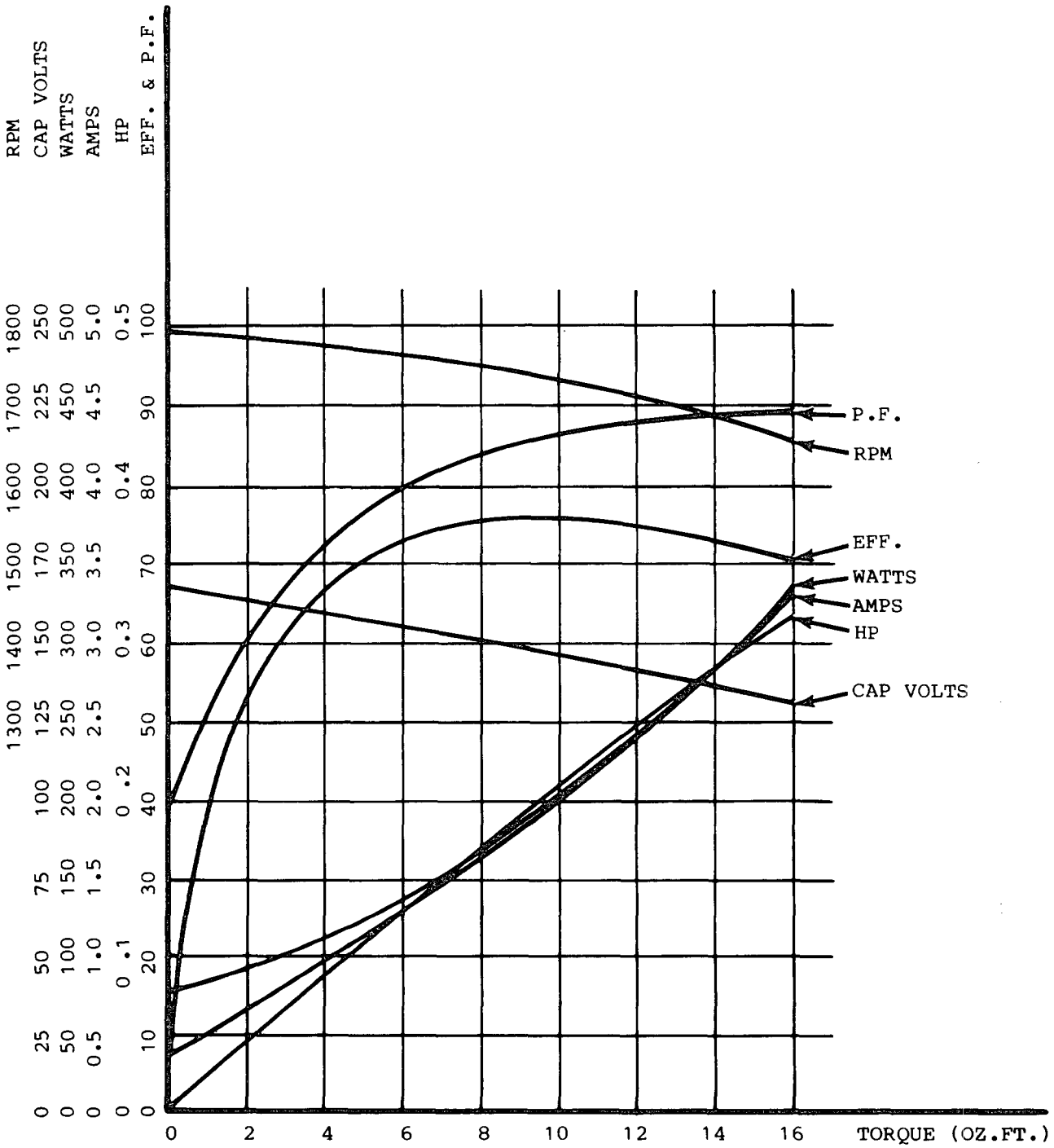

Figure 3-10. WH210 Performance Curve

SPEED TORQUE & RELAY APPL.
115 VOLTS, 60 HZ 1 Ø
.20 HP TYPE FL
15 MFD. RUN CAP.
21-25 MFD. START CAP.
75°C WDG. TEMP.

P-5706
BOOK: 280708
DATE: 3-14-79
L-912524

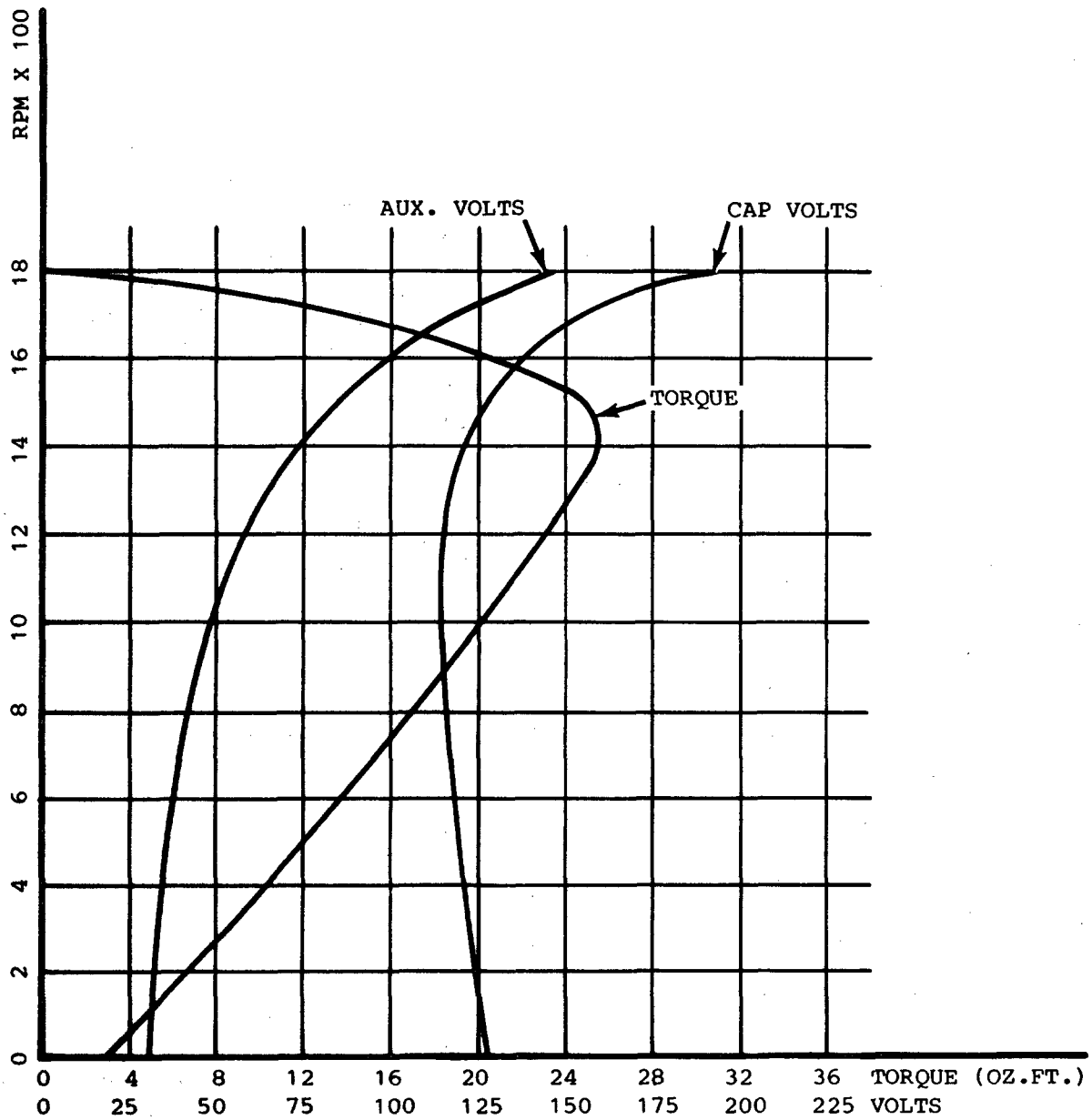

Figure 3-11. WH210 Performance Curve

SPEED TORQUE & RELAY APPL.
95 VOLTS, 60 HZ 1 Ø
.20 HP TYPE FL
15 MFD. RUN CAP.
75°C WDG. TEMP.

P-5706
BOOK: 280708
DATE: 3-14-79
L-912524

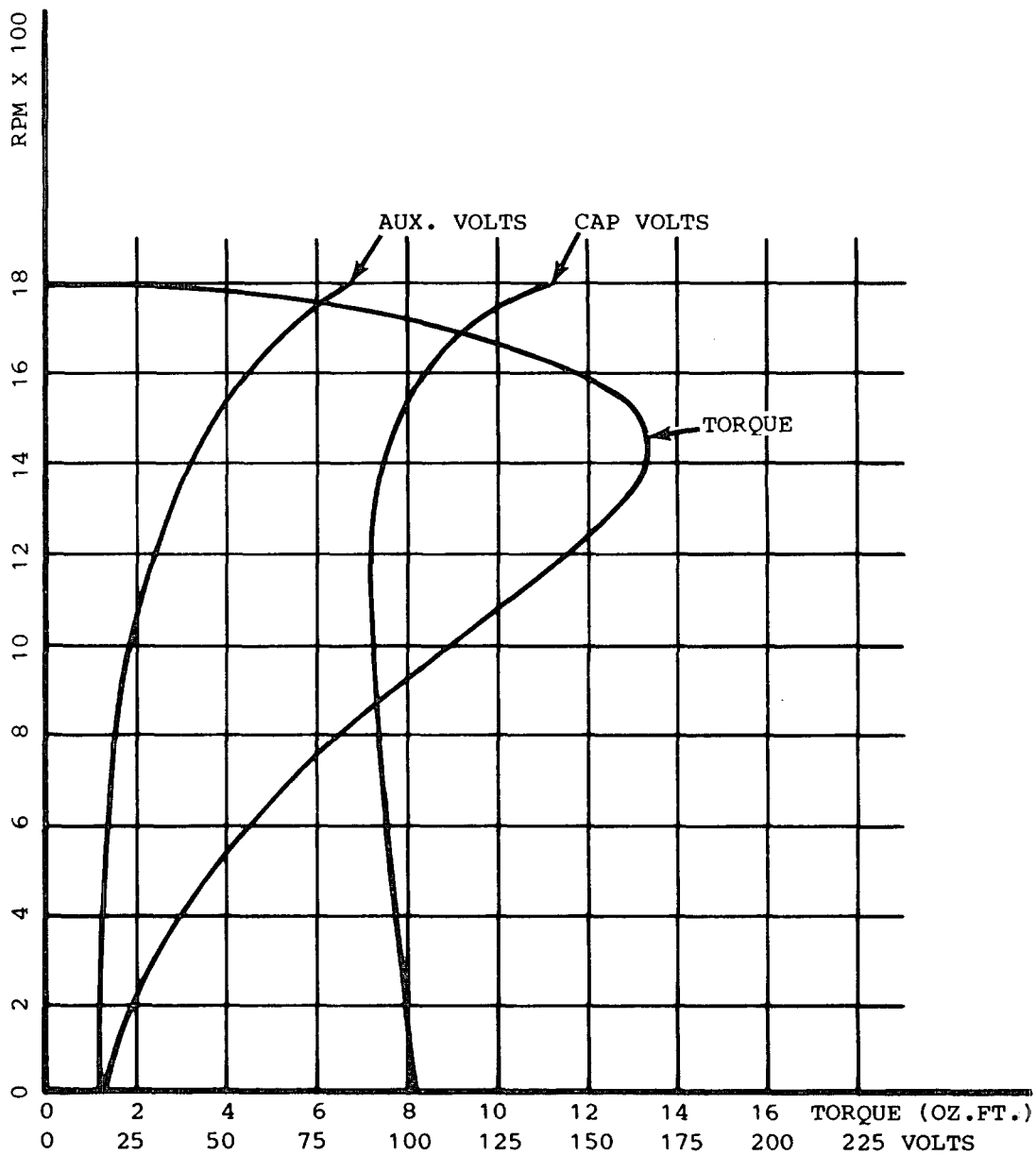

Figure 3-12. WH210 Performance Curve

SPEED TORQUE & RELAY APPL.
126.5 VOLTS, 60 HZ 1 Ø
.20 HP TYPE FL
15 MFD. RUN CAP.
75°C WDG. TEMP.

P-5706
BOOK: 280708
DATE: 3-14-79
L-912524

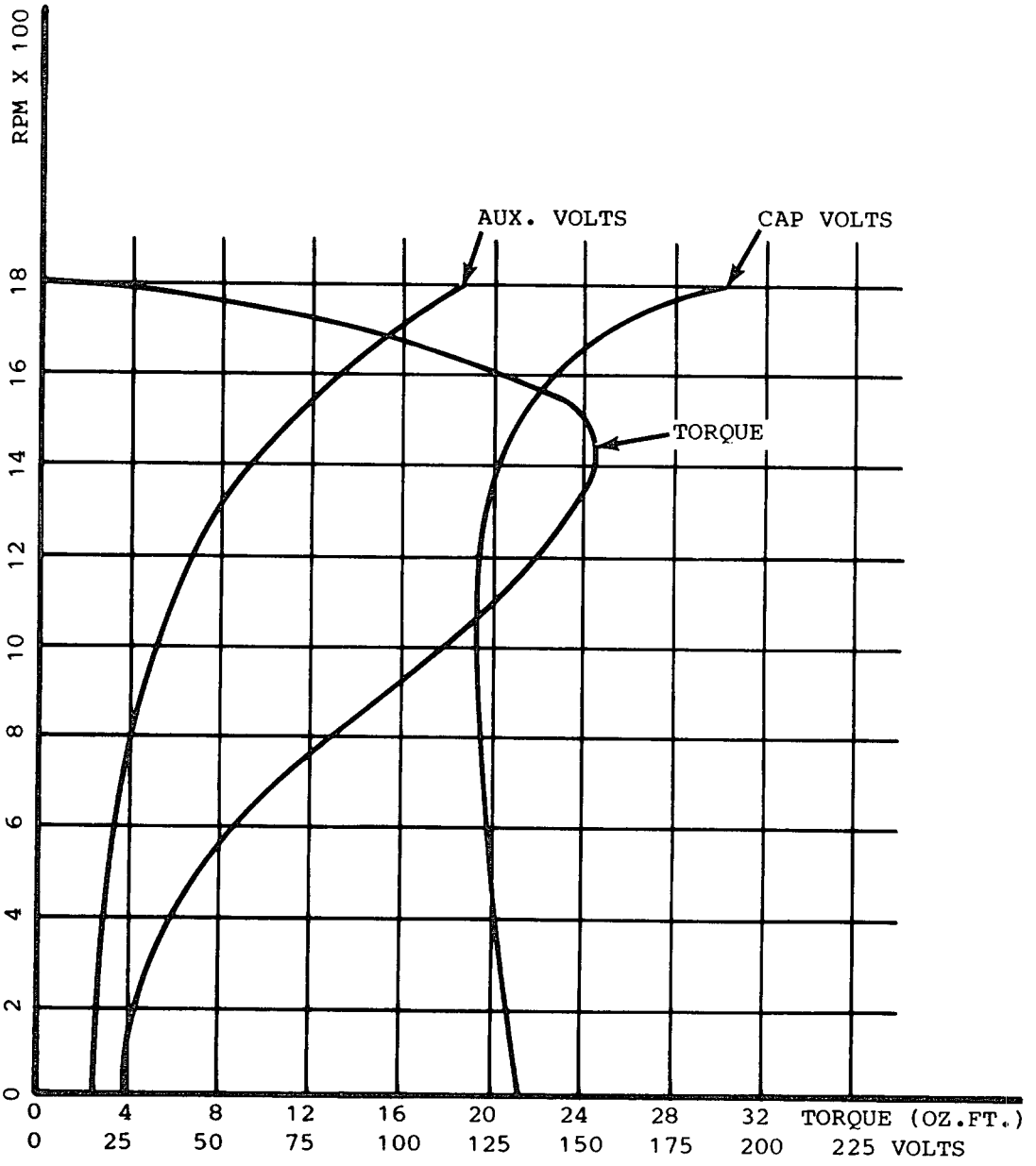

Figure 3-13. WH210 Performance Curve

SPEED TORQUE & RELAY APPL.
115 VOLTS, 60 HZ 1 Ø
.20 HP TYPE FL
15 MFD. RUN CAP.
75°C WDG. TEMP.

P-5706
BOOK: 280708
DATE: 3-14-79
L-912524

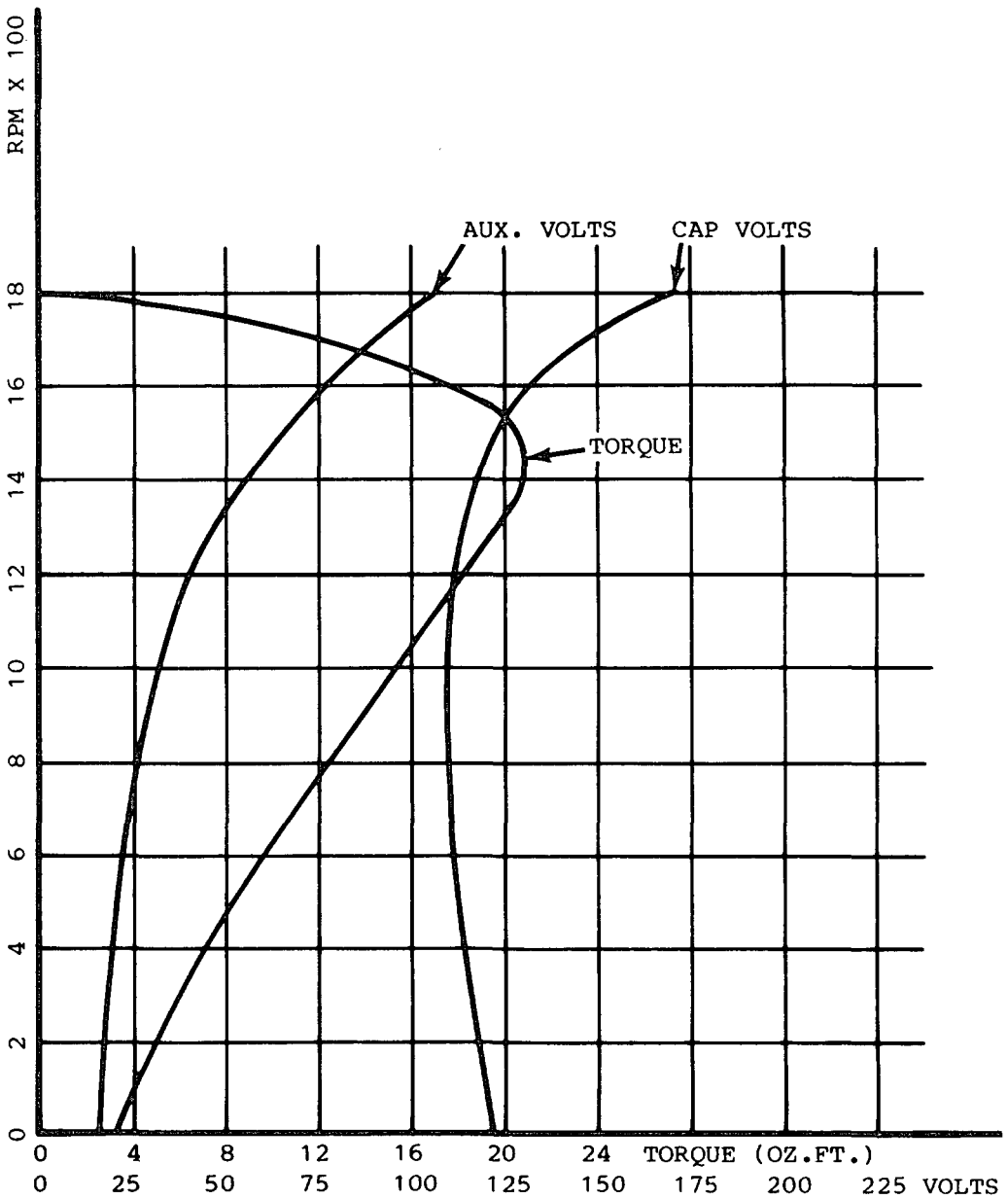

Figure 3-14. WH210 Performance Curve

DFT K. SHELL 4-1-80	APP. CODE	THERMO	ROT	L SPEC	REF. PA	STACK
ENGR 4-3-80			CCW	912542e	P-5709	1.75" 5CKB17KBFL

DDED CUST. P/N'S. 30
6-20-80
6/27/80 2
6-10
X RES. WAS 13.1.
R.002272
9-30-80
9-30-80 3
9-22

HERMETIC MOTOR IDENTIFICATION

ATHENS PRODUCTS CO.
2000 TELLICO AVE. ATHENS, TENN.
One of the White Consolidated Industries
WCI

STYLE NO 325P127
RATING .25 HP 115 VOLTS 60 HZ
CUSTOMER COLUMBUS
CUSTOMER STYLE NO W100 - WH310

PART IDENTIFICATION

	CUSTOMER PART NO.	ATHENS PRODUCTS CO. PART NO.	OUTLINE DWG.
STATOR	C140902	5006S03	1C51F03
ROTOR	C140801	5503R01	1B29F05

MOTOR PERFORMANCE DATA

	MAIN	AUXILIARY
1ϕ RESISTANCE (25°C).....	2.69 ± 7 %	13.6 ± 7 %
TYPE WIRE.....	COPPER	COPPER
WIRE SIZE.....	18.5 AWG .0381 DIA	24.5 AWG .019 DIA

3ϕ RESISTANCE (25°C) LINE TO LINE _____ AVERAGE ± _____ %

CAPACITOR: RUN 15 MFD. 220 VOLTS
START _____ MFD. _____ VOLTS

PROTECTOR: _____

LEADS:	COMMON AWG	MAIN AWG	AUX AWG
VOLTS.....	115		
FREQ.....	60		
TEMPERATURE °C.....	75		
LOAD POINT (oz FT)	12		
AMPS.....	2.43		
WATTS.....	246		
RPM.....	1746		
EFF (%).....	75.6		
PF (%).....	88.0		
CAP. VOLTS...	184		

BREAKDOWN TORQUE (oz FT) 30.2
3000 RPM TORQUE (FT) _____

LOCKED ROTOR
VOLTS..... 115
AMPS..@.4.SEC.14.2
TORQUE (oz FT) 3.5
CAP VOLTS.... 123

RATE OF RISE
MAIN °F/SEC: 2.05
AUX °F/SEC: _____
ROTOR °F/SEC: _____

PERFORMANCE CURVES
673218
THRU
673221

RELAY DATA
P.U. _____ AMPS-VOLTS D.O. _____ AMPS-VOLTS
CONTINUOUS COIL VOLTS _____ MAX RUPTURE AMPS _____

325P127

APC-176

Figure 3-15. WH310 Motor Performance Data

CURVE NO. 673218

PERFORMANCE TEST
115 VOLTS, 60 HZ 1 Ø
.25 HP TYPE FL
15 MFD. RUN CAP.
75°C WDG. TEMP.

P-5709
BOOK: 280729
DATE: 9-5-79
L-912542

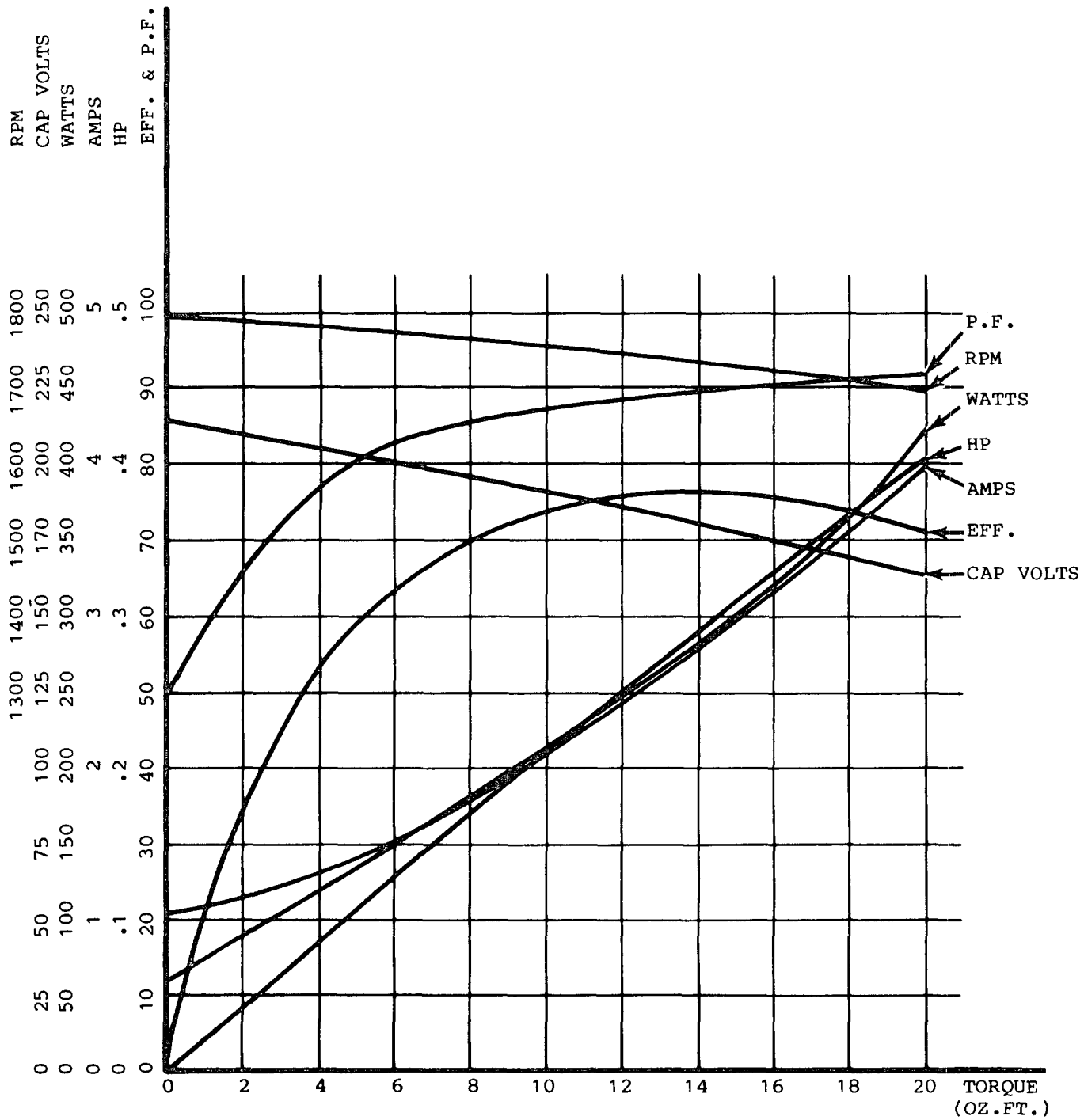

Figure 3-16. WH310 Performance Curve

SPEED TORQUE & RELAY APPL.
115 VOLTS, 60 HZ 1 Ø
.25 HP TYPE FL
15 MFD. RUN CAP.
75°C WDG. TEMP.

P-5709
BOOK: 280729
DATE: 9-5-79
L-912542

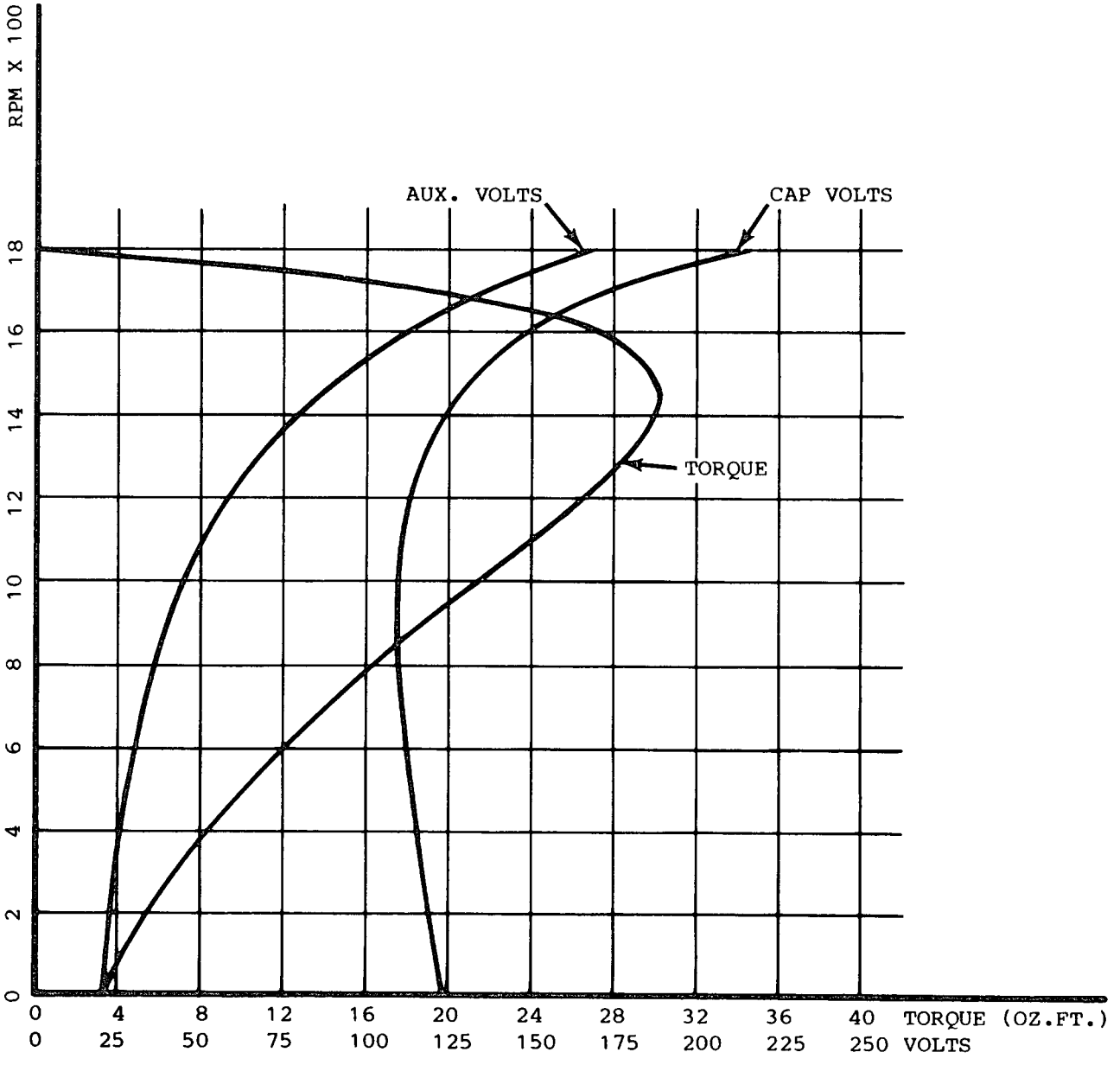

Figure 3-17. WH310 Performance Curve

CURVE NO. 673220

SPEED TORQUE & RELAY APPL.
95 VOLTS, 60 HZ 1 Ø
.25 HP TYPE FL
15 MFD. RUN CAP.
75°C WDG. TEMP.

P-5709
BOOK: 280729
DATE: 9-5-79
L-912542

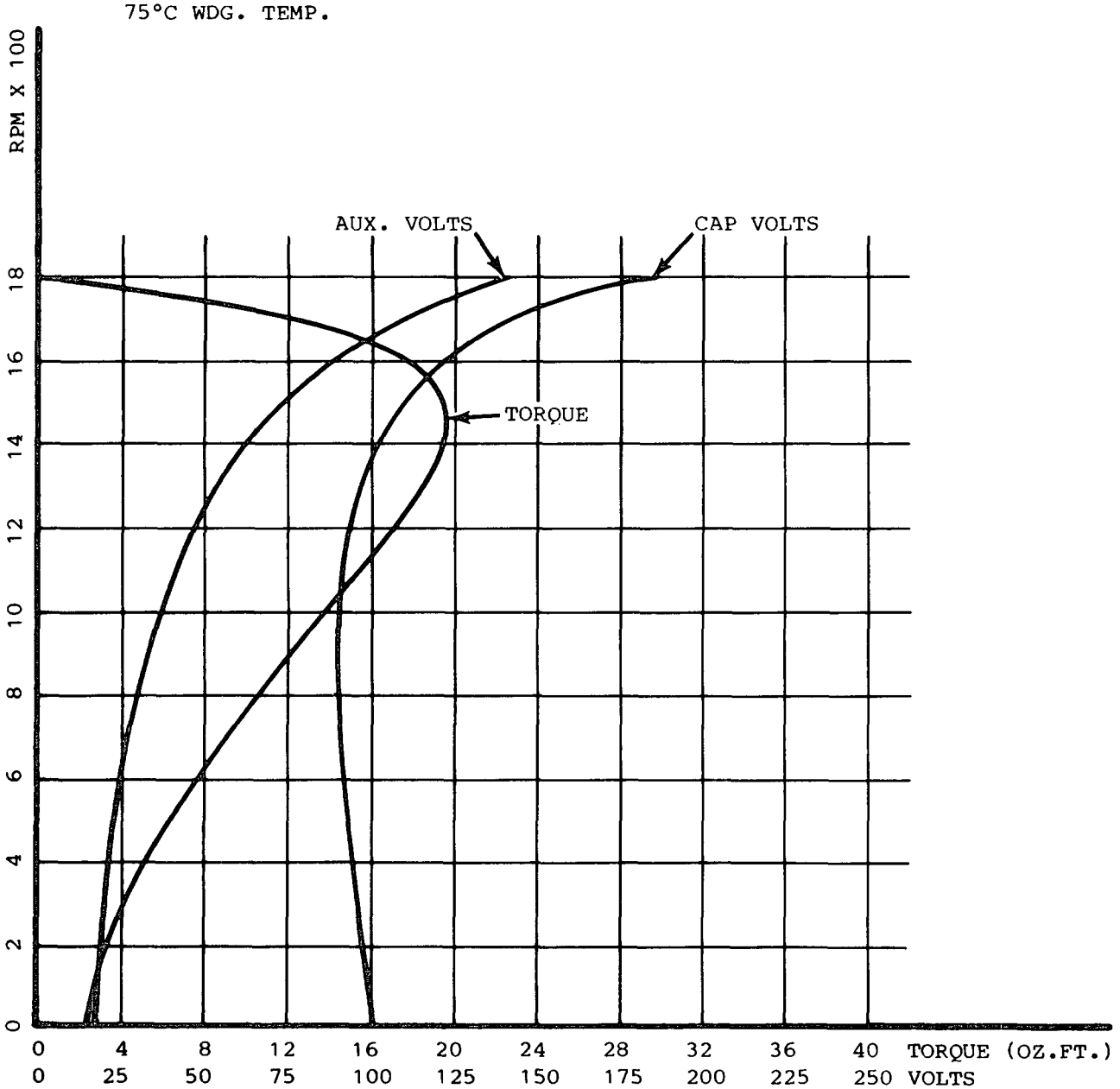

Figure 3-18. WH310 Performance Curve

SPEED TORQUE & RELAY APPL.
126.5 VOLTS, 60 HZ 1 Ø
.25 HP TYPE FL
15 MFD. RUN CAP.
75°C WDG. TEMP.

P-5709
BOOK: 280729
DATE: 9-5-79
L-912542

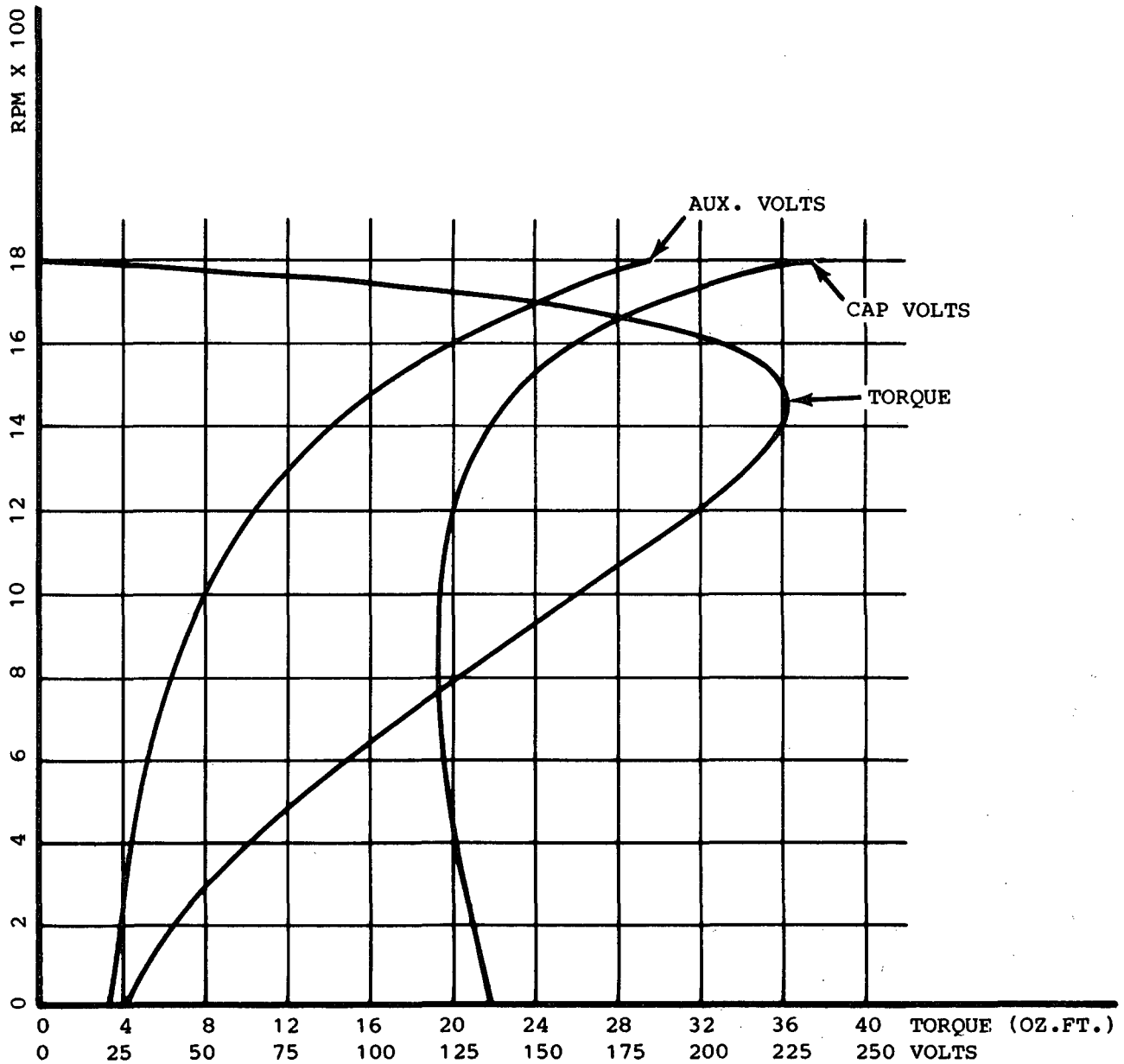

Figure 3-19. WH310 Performance Curve

DFT SW 11-2-79	APP. CODE	THERMO	ROT	L SPEC	REF. PA	STACK
ENGR JB Wilson			CCW	912575		2.025"

HERMETIC MOTOR IDENTIFICATION

ATHENS PRODUCTS CO.
2000 TELLICO AVE. ATHENS, TENN.
One of the White Consolidated Industries
WCI

STYLE NO. P-5711
RATING .60 HP 115 VOLTS 60 HZ
CUSTOMER COLUMBUS PRODUCTS
CUSTOMER STYLE NO 4 POLE FOR T66 APPLICATION - WH410

PART IDENTIFICATION

	CUSTOMER PART NO.	ATHENS PRODUCTS CO. PART NO.	OUTLINE DWG.
STATOR		P-5711	1C51F06
ROTOR		P-5711	1B29F06

MOTOR PERFORMANCE DATA

	MAIN	AUXILIARY
1ϕ RESISTANCE (25°C).....	1.63 ± 7 %	14.1 ± 7 %
TYPE WIRE.....	COPPER	COPPER
WIRE SIZE.....	17.5 AWG .0428 DIA	24.5 AWG .019 DIA

3ϕ RESISTANCE (25°C) LINE TO LINE _____ AVERAGE ± _____ %

CAPACITOR: RUN 15 MFD. 220 VOLTS
START _____ MFD. _____ VOLTS

PROTECTOR: _____

LEADS:	COMMON AWG	MAIN AWG	AUX AWG
VOLTS.....		115	
FREQ.....		60	
TEMPERATURE.....°C		75	
LOAD POINT (OZ FT)		30	
AMPS.....		6.02	
WATTS.....		602	
RPM.....		1715	
EFF (%).....		75.8	
PF (%).....		87.0	
CAP. VOLTS...		214	

BREAKDOWN TORQUE (OZFT) 50.5
3000 RPM TORQUE (FT) _____

LOCKED ROTOR
VOLTS..... 115
AMPS..... @ 4 SEC, 24.7
TORQUE (OZ FT) 5.2
CAP VOLTS.... 126

RATE OF RISE
MAIN °F/SEC: 5.21
AUX °F/SEC: _____
ROTOR °F/SEC: _____

PERFORMANCE CURVES
673273
THRU
673276

RELAY DATA
P.U. _____ AMPS-VOLTS D.O. _____ AMPS-VOLTS
CONTINUOUS COIL VOLTS _____ MAX RUPTURE AMPS _____

P-5711

APC-176

Figure 3-20. Four-Pole For T66 Application - WH410

SPEED TORQUE & RELAY APPL.
95 VOLTS, 60 HZ 1 Ø
.125 HP TYPE FH
75°C WDG. TEMP.

WITH 5 OHM SERIES AUX. RES.

P-5490 & 6084
BOOK: 2805536
DATE: 7-12-77
L-912394

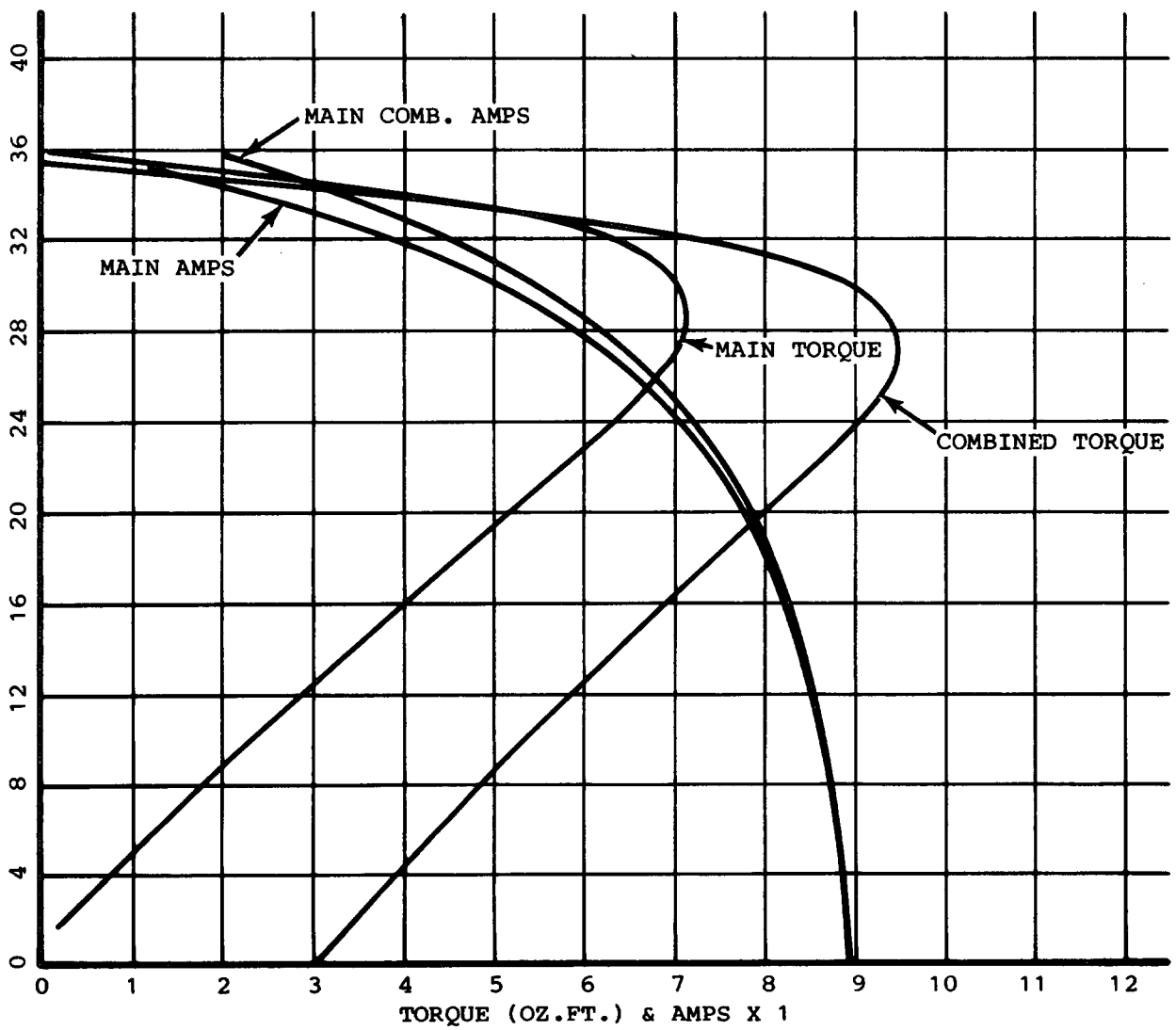

Figure 3-21. WH410 Performance Curve

SPEED TORQUE & RELAY APPL.
95 VOLTS, 60 HZ 1 Ø
.125 HP TYPE FH
75°C WDG. TEMP.

P-5490 & 6084
BOOK: 2805536
DATE: 7-12-77
L-912394

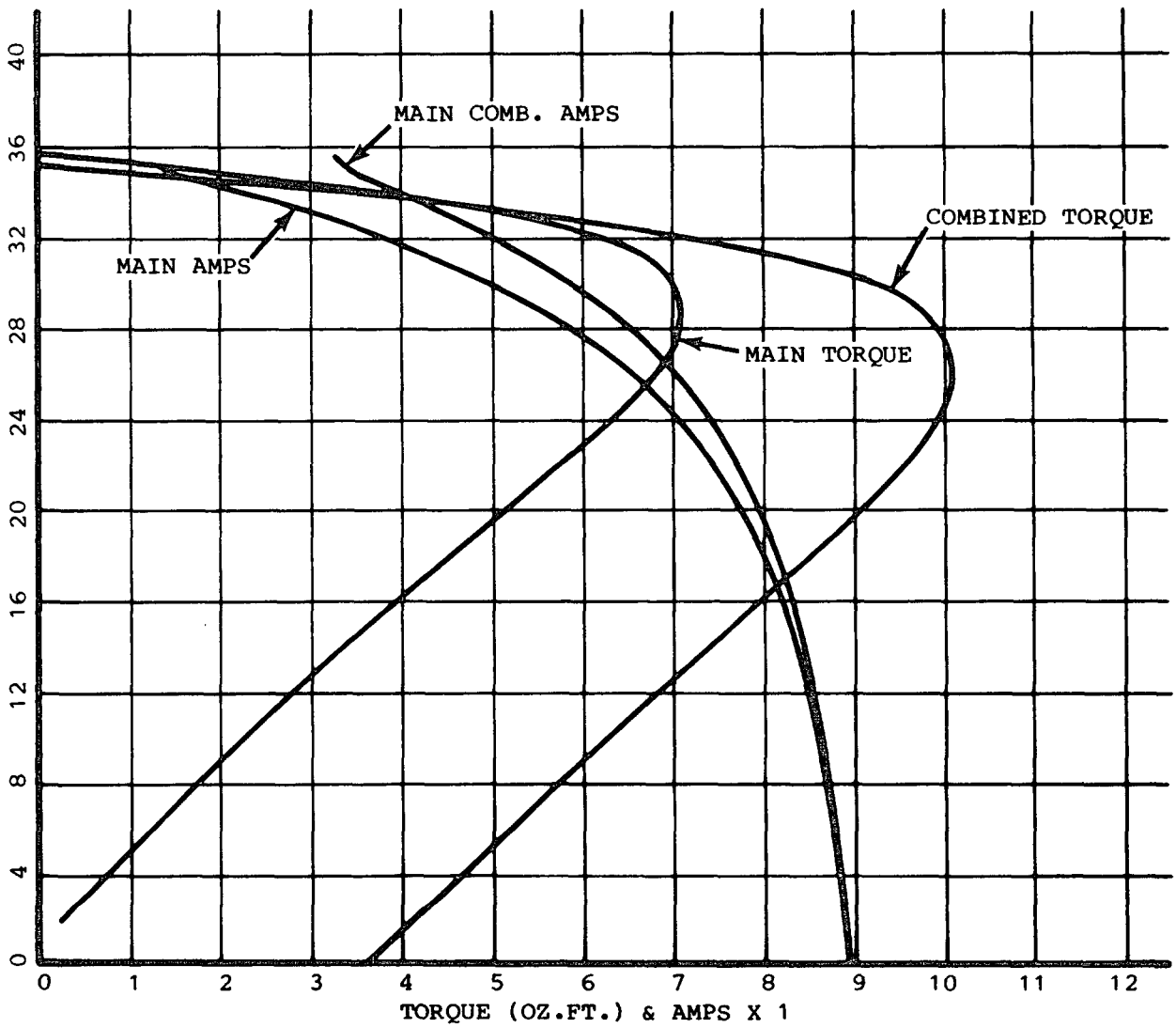

Figure 3-22. WH410 Performance Curve

CURVE NO. 672275

SPEED TORQUE & RELAY APPL.
115 VOLTS, 60 HZ 1 Ø
.125 HP TYPE FH
75°C WDG. TEMP.

P-5490 & 6084
BOOK: 2805536
DATE: 7-12-77
L-912394

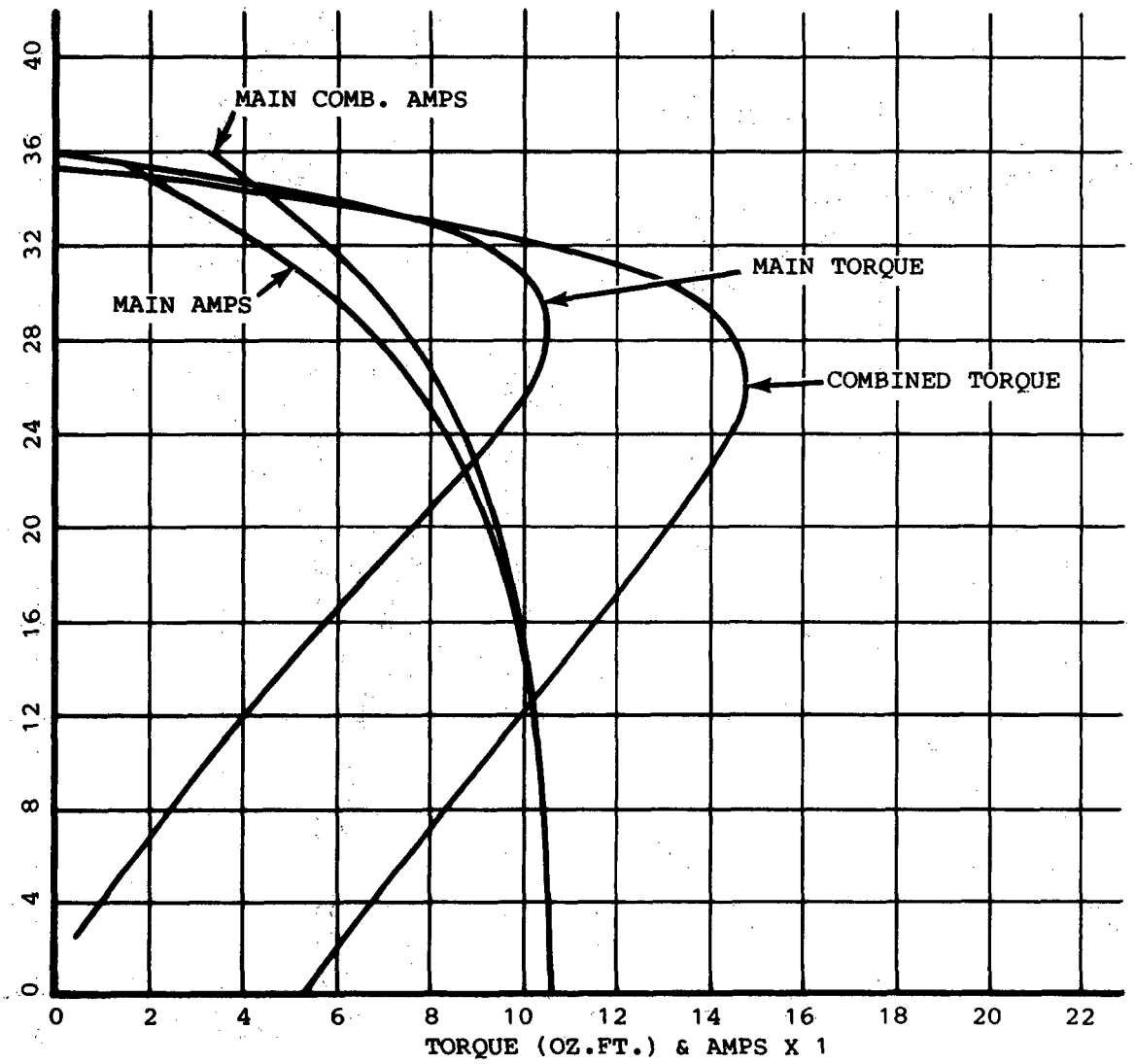

Figure 3-23. WH410 Performance Curve

CURVE NO. 672276

SPEED TORQUE & RELAY APPL.
115 VOLTS, 60 HZ 1 Ø
.125 HP TYPE FH
75°C WDG. TEMP.

WITH 5 OHM SERIES AUX. RES.

P-5490 & 6084
BOOK: 2805536
DATE: 7-12-77
L-912394

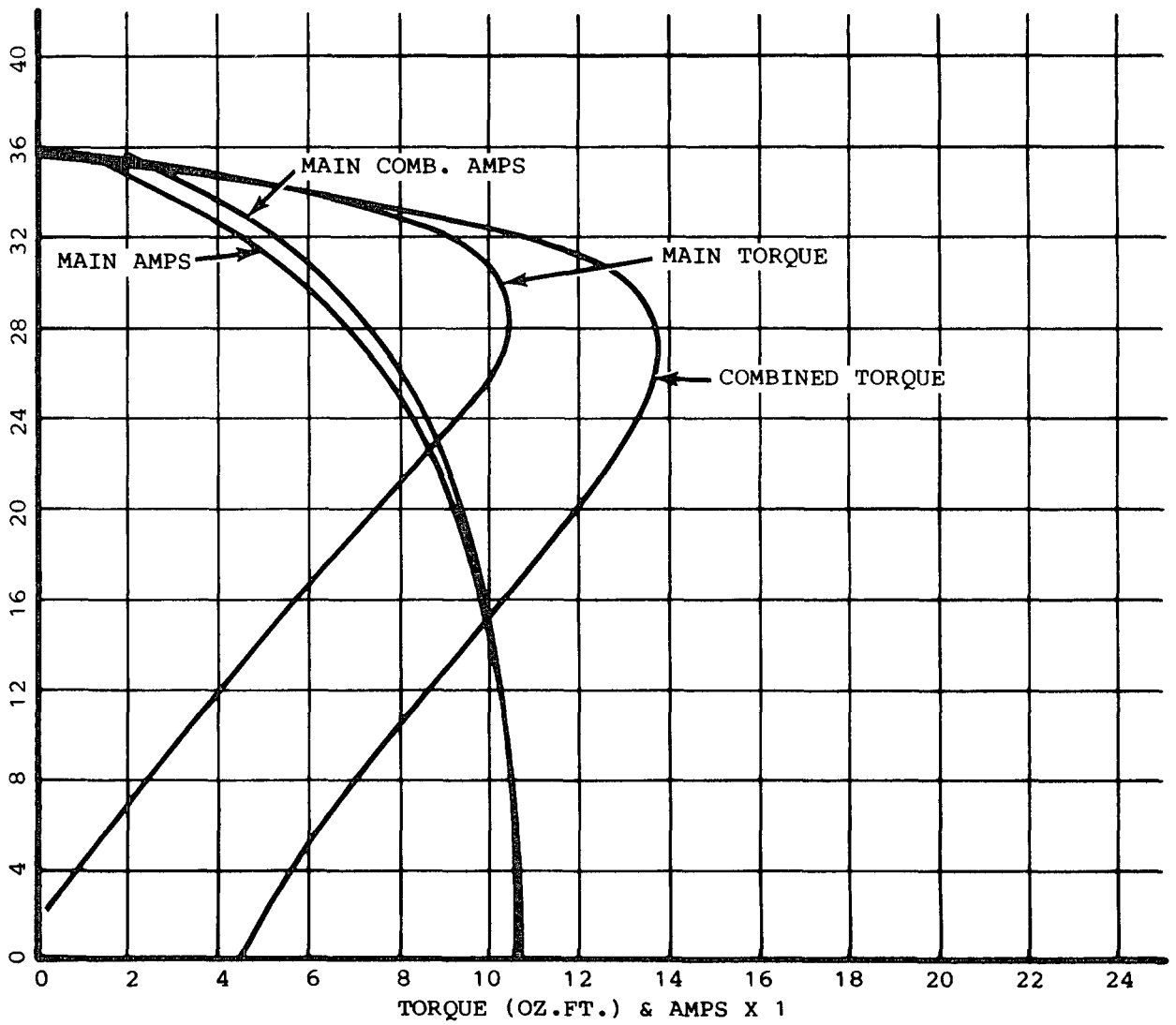

Figure 3-24. WH410 Performance Curve

3.2 DISPLACEMENT

T-line compressors, in the original designs of the early 1960's, were intended to be constant bore with the stroke varied to give different capacities. This is historically true of low back-pressure and high back-pressure compressors. Table 3-4 represents the bore, strokes, and displacement for the high back-pressure TH series.

TABLE 3-4

Bore, Stroke, and Displacement Data

Compressor	Bore (in.)	Stroke (in.)	Displacement (in^3/rev)
TH147	1.0	0.342	0.269
TH210	1.0	0.468	0.369
TH310	1.0	0.756	0.515
TH400	1.0	0.976	0.687

3.3 VALVE SYSTEM

T-line valves consist primarily of a suction valve, a valve plate, and a discharge valve and stop which are riveted to the valve plate.

3.3.1 SUCTION VALVE

Suction valves used on hermetic compressors in this capacity range are usually flapper valves or reeds. These reeds are usually stamped and then tumbled to improve their edge finish. The suction reed used on all the T-line high back-pressure compressors is 0.010 in. thick. This reed has been used with valve plates that contain a single suction port and plates that use two suction ports on low back-pressure

and high back-pressure compressors. The maximum capability of this valve is two 0.260 in. diameter suction ports. A valve stop is machined in the cylinder housing.

3.3.2 VALVE PLATE

Valve plates on all T-line high back-pressure compressors are nominally 1/8 in. thick. External dimensions of width and length are the same. Rivet hole location is the same on all plates. Differences between the plates occur in the number and diameter of suction ports and the diameter of the discharge ports. Table 3-5 lists the variations.

TABLE 3-5

Valve Plates

	SUCTION PORTS		DISCHARGE PORTS DIAMETER	
Model	Number	Diameter (in.)	Port 1 (in.)	Port 2 (in.)
TH147	1	0.310	0.140	0.124
TH210	1	0.310	0.140	0.124
TH310	1	0.310	0.140	0.140
TH400	2	0.260	0.190	0.190

3.3.3 DISCHARGE VALVE

The discharge valve is also a flapper valve or reed. This reed is 0.010 in. thick and is somewhat similar in appearance to a horseshoe. A prebend is put in the reed so that it is in a normally closed position. A stop, similar in shape to the reed, is also provided. The

reed and stop are secured to the plate by riveting. On the largest model (TH400) a second discharge reed is used to cushion the impact in stopping the reed. The discharge reed is applied in the normal manner to the plate but a second reed is reversed and laid on top of it before they and the stop are riveted.

3.4 MUFFLERS, CYLINDER HEAD, AND DISCHARGE TUBE ASSEMBLY

All of the TH series compressors use identical suction mufflers, cylinder heads, discharge mufflers, and discharge tubes. A brief description of these components is contained in the following sections. With the exception of the discharge tube, all of these parts are furnace brazed together.

3.4.1 CYLINDER HEAD

The cylinder head currently being used is a cast-iron version of the aluminum head that was previously used in production for many years. This item was implemented as a cost reduction and does not differ from the previous head in performance.

3.4.2 SUCTION MUFFLER

Two drawn-shell halves comprise the main body of the suction muffler. At the entrance to the suction muffler a shield prevents oil from being drawn into the intake. A mounting foot is brazed to the inlet half so that the muffler may be screwed to the cylinder housing. The suction muffler inlet tube has an inner diameter of 0.319 in. A suction muffler outlet tube with an inner diameter (ID) of 0.319 in. connects the muffler to the head.

3.4.3 DISCHARGE MUFFLER

The discharge muffler is also comprised of two drawn-shell halves. These sections are separated by a baffle which contains a roll pin. The roll pin has an ID of 0.120 in. A mounting foot is brazed to the outlet half so that, like the suction muffler, it may be screwed to the cylinder housing. A tube connects the head to the discharge muffler. Over most of its length, this tube has an ID of 0.194 in. At the muffler connection it is swaged down to 0.100 in.

3.4.4 DISCHARGE TUBE ASSEMBLY

The discharge tube is silver soldered to the discharge muffler. Eventually it is intended to make this connection by furnace brazing. The tube is made of steel and comes with an extension tube of copper brazed to it. The copper extension tube arises is needed to allow soldering the tube to the compressor shell. The ID of the discharge tube is 0.132 in.

4. PROTOTYPE COMPRESSORS

4.1 MOTORS

The simplest way to improve compressor efficiency, from the standpoint of the compressor manufacturers, is to increase the efficiency of the motor. In an attempt to do this we acquired sample motors from Athens Products Company. These samples were designed for two capacity sizes, the 1450 and 4000 BTU/Hr. models. In each of these capacity sizes, three types of motors were provided for each model. The types included a resistance-start induction run (RSIR) aluminum winding, a RSIR copper winding, and a permanent split capacitor (PSC) copper winding. All of these motors were of a two-pole configuration. Sample motor designations and load point efficiencies are shown in Table 4-1.

TABLE 4-1

Sample Motors and Efficiencies

	CAPACITY			
	1450 BTU/Hr.		4000 BTU/Hr.	
Type	Sample No.	Eff.(%)	Sample No.	Eff.(%)
RSIR Al	P-5712	73.1	P-5715	71.6
RSIR Cu	P-5713	76.2	P-5716	75.4
PSC Cu	P-5714	81.5	P-5717	81.5

Since compressor efficiency was our primary goal, we incorporated the P-5714 and P-5717 samples into TH145 and TH400 respectively. Calorimeter testing provided the following results, shown in Table 4-2.

TABLE 4-2

Calorimeter Data With High-Efficiency Motors

Compressor	Capacity (BTU/Hr.)	Input (Watts)	EER (BTU/WHr.)
TH145 With P-5714	1472	181.0	8.13
TH400 With P-5717	5009	619.2	8.09

At about the same time we were conducting tests on the Athens' samples, we incorporated two production low back-pressure motors into high back-pressure compressors. This is possible to do because high back-pressure compressors deliver more energy than low back-pressure compressors at a much lower compression ratio.

From past experience, we knew that the motor torque requirements were about the same for the 2100 BTU/Hr. high back-pressure machines as they were for low back-pressure units in the 800 to 900 BTU/Hr. range. A similar relationship exists between the 3100 BTU/Hr. high back-pressure machine and low back-pressure units in the 1100 to 1300 BTU/Hr. range. Since we had two fairly good motors in production, TEA090 and TEA130, we decided to apply them to the high back-pressure units. Both of these RSIR motors are two-pole with copper run windings and deliver 76.9% efficiency at load point. The calorimeter and pullout results are in Table 4-3.

While all of the tests showed improvement in the EER, the results were less than we had hoped for. The apparent discrepancies in EER values attained between the PSC 145s and 400s (81.5% efficiency motors) and the RSIR 210s and 310s (76.9% efficiency motors) may be explained by the actual production motor efficiencies, which varied from a low of 65.6% for the TH310 to a high of 71.2% for the TH400.

TABLE 4-3

Calorimeter and Pullout Data for
TH210 and TH310 Compressors With Low Back-Pressure Motors

Compressor	Capacity (BTU/Hr.)	Input (Watts)	EER (BTU/WHr.)	Pullout (Volts)
TH210 With TEA090 Motor	2329	279.0	8.35	78
TH310 With TEA130 Motor	3332	411.6	8.10	88

Our initial work in four-pole, high back-pressure compressors was based on the same technique that was used with the TH210s and TH310s. We took low back-pressure motors and applied them to high back-pressure units. The first four-pole, high back-pressure compressor was a W80 unit fitted with a W120 motor. This compressor was given the designation WH80. The compressor was calorimeter and pullout tested.

A second WH80 was also built and tested. The compressor was the same as the first one except it used the motor from a W100 unit. The pullout test on the first WH80 showed that there was more than enough motor torque. In using the smaller W100 motor, it was hoped that we would be further up the efficiency curve and produce a higher EER. However, there was virtually no change as can be seen in Table 4-4.

TABLE 4-4

Calorimeter and Pullout Test Data for the
WH80 Compressor

Motor	Capacity (BTU/Hr.)	Input (Watts)	EER (BTU/WHr.)	Pullout (Volts)	Motor Eff. (%)	Motor Type
W120	2857	307.1	9.30	78	73.8%	PSC
W100	2794	300.1	9.31	88	73.6%	PSC

It should be noted that the WH80 had a displacement of 0.785 in^3/rev. Since we were looking for about 3100 BTU/Hr., it was felt that this compressor displacement could not fit in the model line. Immediately following the building and testing the WH80s, we ran a W60 compressor as a high back-pressure model, Table 4-5. We were sure that at these conditions there would not be enough motor to perform satisfactorily on the pullout. Despite this shortcoming, information was gained on the required motor strengths and capacity available from the W60s displacement of 0.644 in^3/rev.

TABLE 4-5

Calorimeter and Pullout Data for the W60 Compressor

Capacity (BTU/Hr.)	Input (Watts)	EER (BTU/WHr.)	Pullout (Volts)	Motor Eff. (%)	Motor Type
2193	232	9.45	102	75.6	PSC

The tests on these three compressors settled the question of whether we would get higher efficiencies out of two-pole or four-pole motors. From these tests on we concentrated our efforts on the four-pole machines. Due to the large amount of four-pole motor work done in the refrigerator-freezer and RAC compressor development phases of this project, no new motors were developed for the high back-pressure compressors. Three of the four motors were applied to high back-pressure compressors from the refrigerator-freezer compressor. The fourth was a RAC motor that proved to be too small for the 6000 BTU/Hr. application, but worked very nicely in the WH410. Table 4-6 describes the motors selected and some information on them.

TABLE 4-6

Data for WH Line Motors

Model	Motor Type	Stack Height (in.)	Winding Material Main	Winding Material Aux.	Breakdown Torque (oz - ft)	Load Point* Efficiency (%)	Previous Use
WH145	PSC	1 3/4	Cu	Cu	16.6	75.3	W60
WH210	PSC	1 3/4	Cu	Cu	20.5	75.2	W80
WH310	PSC	1 3/4	Cu	Cu	30.2	75.6	W100
WH410	PSC	2	Cu	Cu	50.5	75.8	RAC

*Load Point Efficiency From Motor Manufacturer's Rating Point. (Refer to Figures 3-3 thru 3-24.)

Other features which are found in the motor are discussed individually in the following subsections.

4.1.1 PERMANENT SPLIT CAPACITOR OPERATION

A common and cost effective motor and start system applied to high back-pressure compressors is a split-phase induction motor. This type of motor runs on the main winding only and employs a starting winding with a current relay or other switching device to cut out the start winding during the running operation.

Two- or three-phase motors are more efficient than single-phase motors, but were not suitable for this program because the normal power supply is single-phase. Single-phase efficiency can be improved, however, by using a PSC motor. In this arrangement a capacitor is connected in series with the starting winding to introduce a phase displacement so as to partially simulate two-phase operation.

A circuit diagram of a PSC motor is shown in Figure 4-1. The phasor diagram of voltages and currents are also shown in this figure.

The auxiliary-phase usually has a different number of turns than the main-phase since it is subjected to a different voltage. As long as the main-phase voltage leads the auxiliary-phase by 90°, performance is comparable to a two-phase motor. When the PSC motor is operating as a two-phase motor it is referred to as being in balanced operation. One phase of a two-phase motor will have the voltage V_m and the current I_m. The other winding should have a voltage V_a leading by 90° and a current I_a leading I_m by 90°. Since the capacitor voltage must always lag its current I_a by 90°, the capacitor phasor is constructed parallel to I_m terminating at V_m. Where the phasor voltage V_c intersects the vertical phasor, V_a determining the capacitor volts to give balanced operation, I_a must equal I_m/K where K is the winding ratio yet to be determined. Also V_a must equal K x V_L where V_L is the line voltage. Since we have two equations with K and the capacitance to be determined, one can find the K and capacitance to give balanced operation when the main winding is carrying I_m. Any change in I_m results in a different K and a different capacitance. Whenever the above conditions are met, the motor will run and have an efficiency of a two-phase motor at the balance point.

This simple PSC arrangement is suitable for RAC applications, but does not provide sufficient starting torque for high back-pressure usage. The difficulty can be overcome by using a device called a positive temperature coefficient resistor (PTCR) connected in parallel with the capacitor. The PTCR has a low resistance when cold, so that it allows a momentary high current for starting. The current 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.

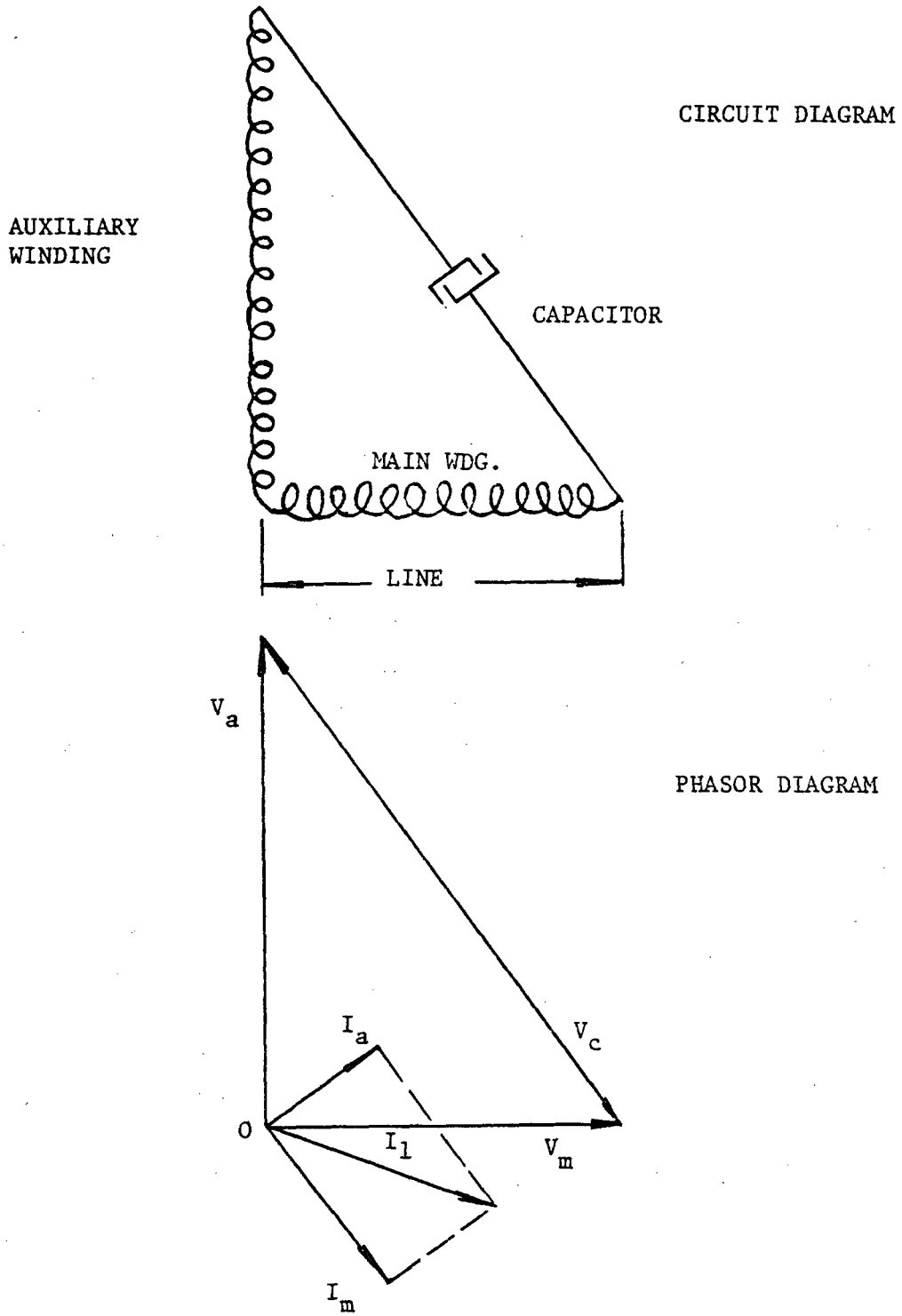

Figure 4-1. Permanent Split Capacitor Motor

4.1.2 LOWER ROTOR RESISTANCE

Reduction of rotor resistance is accomplished by increasing the cross section of the aluminum 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 required for a reasonable magnetic circuit. Reduction of rotor resistance is also limited by starting-torque requirements, since reducing the rotor resistance also reduces the starting torque. Benefits of lower rotor resistance are realized in an ability to reduce operating densities of the magnetic circuit and 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-efficiency compressors are lower in resistance than current production designs.

4.1.3 LOW-LOSS LAMINATION STEEL

Some years ago much of the motor lamination material used in small hermetic motors was changed from silicon steel to carbon steel because of lower cost. Improved permeabilities of carbon steel allowed adequate performance of these motors; however, efficiency improvement can be accomplished by using low-loss steel. This can be done without the transition back to the silicon steels by controlled chemistry in the melting of the steel, and careful processing both by the steel manufacturers and by the lamination manufacturers. Core losses can be further reduced by reducing steel thickness to the minimum consistent with the ability to manufacture and thereby minimizing the eddy-current portion of core loss. The motor-lamination thickness of the four-pole motor has been reduced to 0.019 in. and low-loss steel has been used throughout.

4.1.4 LOWER MAXIMUM TORQUE MOTOR DESIGNS

Compressors must be able to operate under extreme conditions (minimum voltage, maximum temperature, maximum load) which has resulted in motor designs with higher torque capability than that required for normal operation. This has led to a reduction in efficiency at normal loading. The WH-line motors have been designed for lower torque and higher efficiency under normal conditions.

4.1.5 WINDING MATERIAL

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 conductivity. Although the increased slot dimension initially caused manufacturing problems, the use of aluminum led to a cost reduction and it has been widely adopted for hermetic motors. It is generally possible, however, to design a more efficient motor using copper wire because the winding resistance is lower and I^2R losses are reduced. Copper windings are used in all the WH-line motors.

4.2 DISPLACEMENT

The reduction of speed from about 3,440 to 1,720 rpm, as a result of the motor change, required changes in the bore and stroke. The resulting change in displacement, along with the percentage increase is given in Table 4-7. Note that this did not require a doubling of compressor displacement primarily because the volumetric efficiency is substantially better.

TABLE 4-7

Displacement Change
Baseline Versus Prototype Compressors

BASELINE COMPRESSORS			PROTOTYPE COMPRESSORS			
Model	Capacity R-12 (BTU/Hr.)	Displacement (in^3/Rev.)	Model	Capacity R-12 (BTU/Hr.)	Displacement (in^3/Rev.)	Adjusted Increase In Displacement (%)
TH147	1470	0.269	WH145	1340	0.460	0.876
TH210	2100	0.369	WH210	2100	0.644	0.745
TH310	3100	0.515	WH310	3050	0.874	0.725
TH400	4000	0.687	WH410	4070	1.227	0.755

Since the capacities for the improved efficiency prototypes were not identical to the baseline compressors, the increase in displacement values were adjusted to provide the same capacity in order to estimate the percent increase in displacement.

Careful examination of Table 4-8 shows that capacity increases faster than volume displaced until a certain peak is reached and then begins to decline. This relationship is true on both the baseline and prototype units. The cause of this relationship is mostly due to the discharge muffler and tube. The breakpoint occurs at approximately the same point in capacity of both compressor series. As the strokes and bores are different for the same capacity ranges and valve systems have been modified for the WH series, it seems that this is most likely the cause. The discharge mufflers and lines are quite similar for both series of compressors. Figures 4-2 and 4-3 show the typical part in the WH design.

TABLE 4-8

Capacity Ratio
Baseline Versus Prototype Compressors

BASELINE COMPRESSORS		PROTOTYPE COMPRESSORS	
Model	Capacity/Displacement (BTU/Hr. - in^3/Rev.)	Model	Capacity/Displacement (BTU/Hr. - in^3/Rev.)
TH147	5464	WH145	2913
TH200	5691	WH210	3260
TH310	6019	WH310	3490
TH400	5822	WH400	3317
Average	5749	Average	3245

4.3 VALVE SYSTEM

The valve system of the TH-line compressors has been modified for the WH-line. The system is still comprised of a suction valve, discharge valve and stop, and a valve plate. The TH valve system is described in Section 3.3.

4.3.1 SUCTION VALVE

The same flapper valve that is used on the TH compressors is used on the WH145 and WH210 units. The larger bores of the WH310 and WH410 require that the tip on the reed be extended. During the suction or downward stroke of the piston, the tip impacts a machined stop on the cylinder housing preventing excessive stress of the root area of the flapper. This extended tip suction reed is also 0.010 in. thick. Since the basic flapper is identical to the production, the maximum diameter suction ports is 0.260 in.

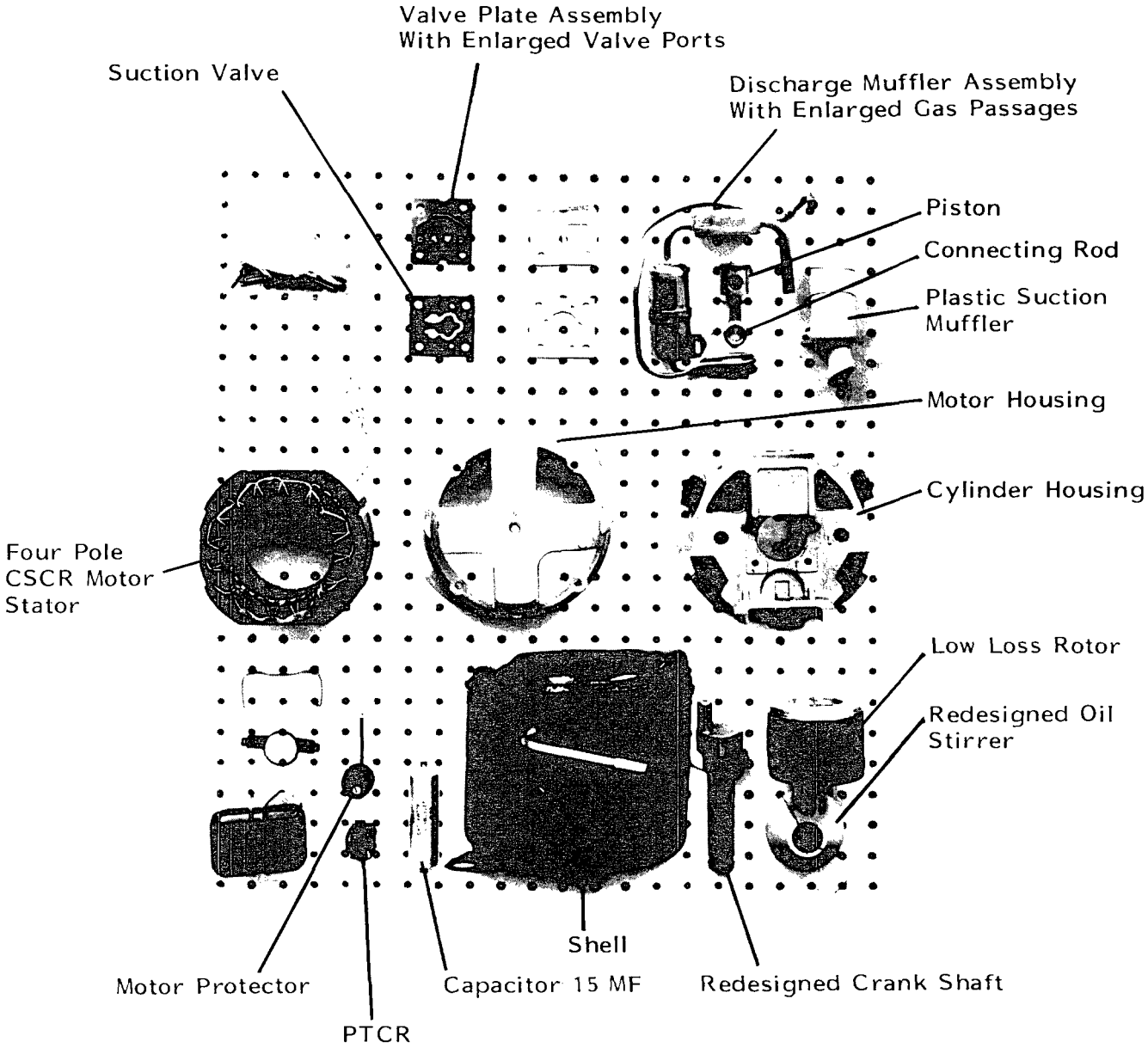

Figure 4-2. Component Parts - Energy Efficient Compressor

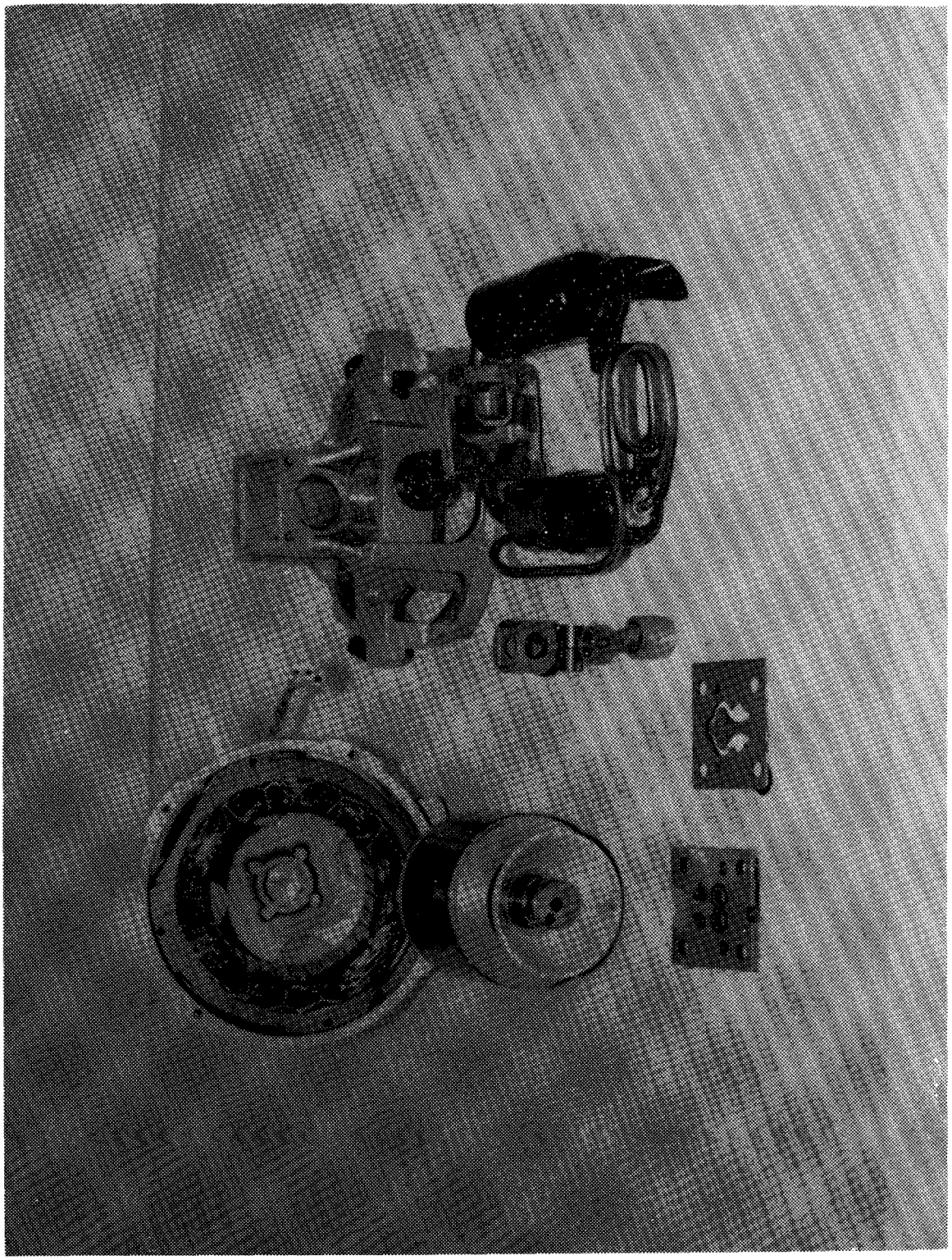

Figure 4-3. Component Parts - Energy Efficient Compressor

4.3.2 VALVE PLATE

The valve plates used on the WH compressors are the same as the high back-pressure T-line compressors. The only difference is the port sizes that are employed. The WH145 and WH210 use the same plate as the TH400 compressor. A valve plate that was used on the T66, a room air-conditioning compressor, is employed on the WH310 and WH400 machines. Table 4-9 gives the port sizes on the WH valve plates.

TABLE 4-9

WH Valve Plates Data

	SUCTION PORTS		DISCHARGE PORTS	
Model	Number	Diameter (in.)	Number	Diameter (in.)
WH145	2	0.260	2	0.190
WH210	2	0.260	2	0.190
WH310	2	0.260	2	0.260
WH410	2	0.260	2	0.260

The larger discharge port diameters have the negative effect of adding to the re-expansion volume. A reduction in flow loss through the ports, however, offsets this and adds to efficiency.

4.3.3 DISCHARGE VALVE

The discharge valve and stop are identical to those for the TH-line. The WH410 uses the same stop and back-up discharge reed as the TH400.

4.4 MUFFLERS, CYLINDER HEAD, AND DISCHARGE TUBE ASSEMBLY

All of the WH prototype compressors use the same suction muffler, cylinder head, discharge muffler, and discharge line. The discharge tube assembly is identical to the one found in the TH compressors. A discussion of this part is found in Section 3.4.4. An illustration of internal gas flow is shown in Figure 4-4.

4.4.1 CYLINDER HEADS

The prototype compressors were all built with aluminum heads. During the transition from aluminum to cast-iron, there was more flexibility to experiment with the aluminum head. Now that the change is completed, the same capabilities exist in the cast-iron. An economic advantage is gained in using the cast-iron version with no adverse performance characteristics expected.

4.4.2 SUCTION MUFFLER

A thermoplastic suction muffler has been adopted for use on the prototypes. This muffler replaces the drawn steel muffler used on the regular production version. A Valox resin is the material used. Initially, when this muffler was used on the improved efficiency low back-pressure compressors, a melting problem developed in the pilot production. It was solved by three changes. First, the grade of Valox was changed from the original pure 325 to a 15% glass-filled resin designated DR-51. Second, the bellmouth muffler inlet was modified to increase the distance from it to the compressor shell. Third, the manufacturing process employed at Kelvinator Compressor Company is different from that used at Columbus Products Company where the problem was encountered. Shells are welded at room temperature and then baked and dehydrated. At Columbus Products Company the shells and pump assemblies came from the oven and were welded at an elevated temperature.

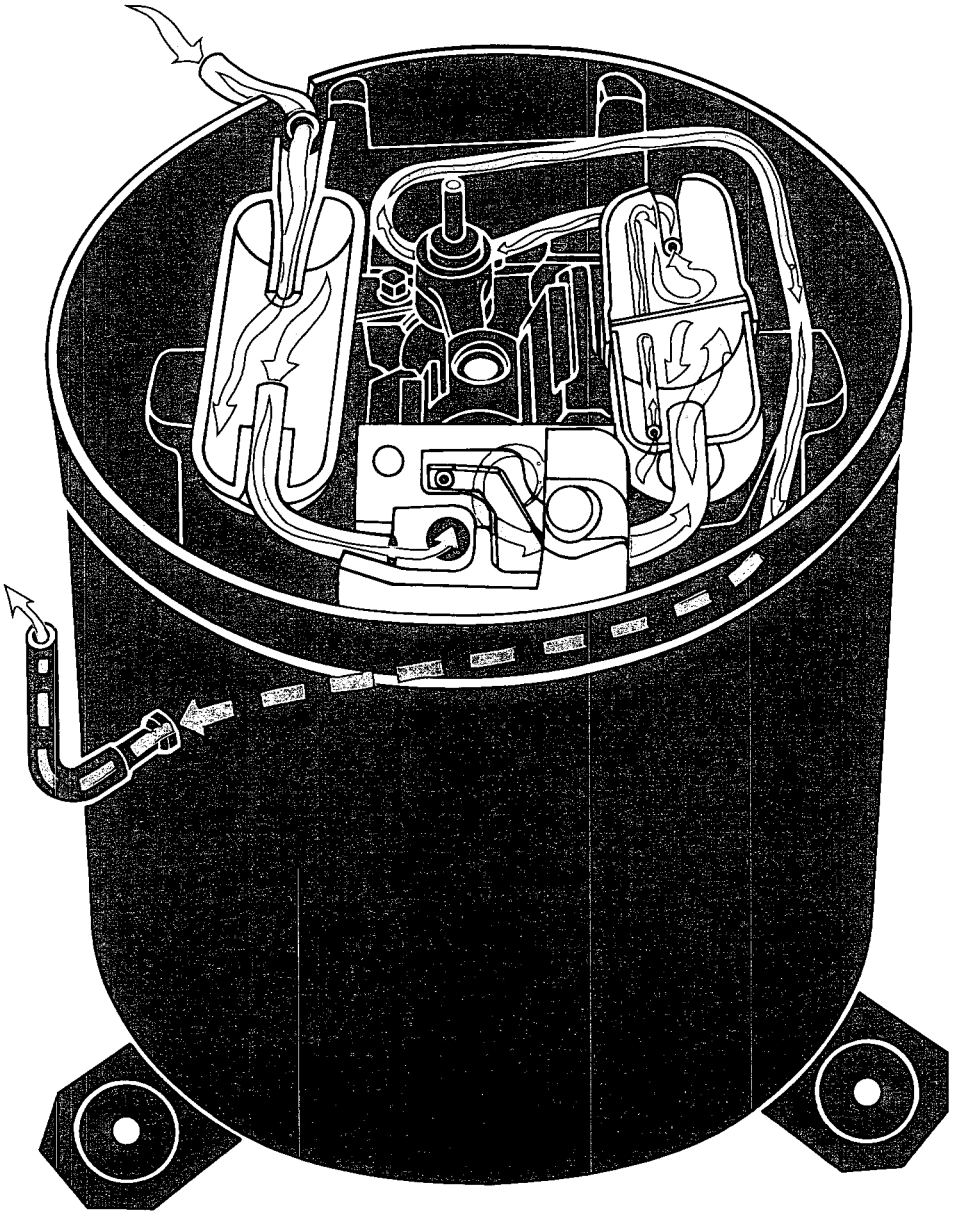

Figure 4-4. Internal Gas Flow

4.4.3 DISCHARGE MUFFLER

The discharge muffler used on the prototype is the same as those used on the production compressor, see Section 3.4.3. A change has been made, however, in the discharge muffler inlet tube. The tube, which has an inner diameter of 0.194 in., has had the 0.100 in. swage removed. Eliminating the swage reduces the flow loss through the high pressure gas path.

4.4.4 DISCHARGE TUBE ASSEMBLY

The discharge tube assembly used on the prototype is the same as those used on the production compressor, see Section 3.4.3.

4.5 CRANKSHAFT

The use of a four-pole motor has required the development of a new oil pump. The oil pump is incorporated as part of the crankshaft. Since the rotational speed was halved when the four-pole motors were adopted, a different crankshaft was needed. The same problem was encountered in the low back-pressure compressor development and the crankshaft developed for it was used on these high back-pressure prototypes.

In the new pump, the oil is drawn up from the sump of the compressor shell through a hole in the center of the crankshaft. It then exits through a radial connecting hole into an annular grove as shown in Figure 4-5. An exit radial hole at the end of the groove allows oil to proceed into a long eccentric hole and up the crankshaft. At the top of the crankshaft a series of holes feed oil into the main journal bearing and the crankpin bearing. The hole in the crankpin is plugged. By doing this a larger amount of oil is supplied to the bearing and through the connecting rod to the bore. The function of the oil spray in the original T design was to convey heat to the

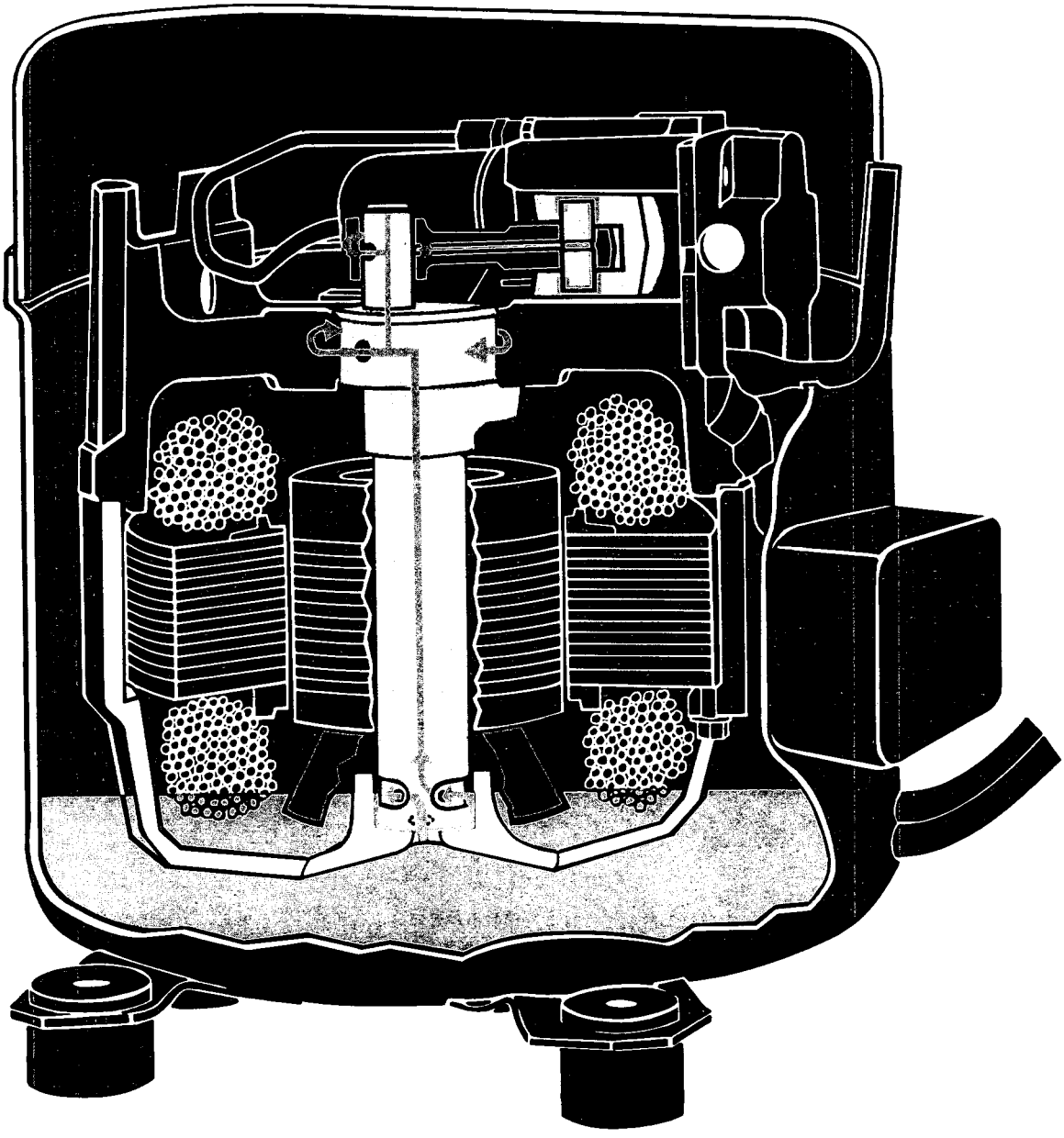

Figure 4-5. Lubrication Flow

compressor shell. With the more efficient motors in the prototype design, it was found that the cooling requirements were greatly reduced and the spray was not necessary.

4.6 TEST RESULTS

After final design and intermediate testing was concluded, a group of seventeen compressors were built for evaluation testing.

The tests were as follows: calorimeter, pullout, start, sound, locked-rotor, ultimate-trip, and life. The methods of testing are described in Section 6 - "Test Procedures".

4.6.1 TESTING PRIOR TO LIFE TEST

The data obtained before the life test from these compressors are given in Table 4-10. As these were judged to be typical of the compressor design, they were used for the life tests.

4.6.2 TESTING AFTER LIFE TEST

After the life tests, the compressors were removed from the test stands and again tested. These tests consisted of calorimeter and sound tests. Table 4-11 lists the data collected in these tests.

4.6.3 COMPARISON OF DATA

A comparison of averaged data, obtained before and after the life test, is shown in Table 4-12. From this data, we can see that for each compressor size, there was a slight decrease in capacity and a small (except for WH310) increase in EER after the life test.

4.6.4 PHYSICAL EXAMINATION OF LIFE TEST

After completion of the tests, the compressors were cut open, torn down, and examined. The normal wear of running parts compared favorably with standard production compressors. The WH145s and WH210s were in very good condition, showing no wear, but this is typical of the smaller displacement compressors. All the compressors had a high degree of dirt, which was attributed to the method of build-up the compressor went through during fabrication. The WH310-005E-B compressor showed heavy carbon deposits on valves and the piston indicating the temperatures and/or pressures during the life test were excessive for some period of time. Also WH410-003E-8 compressor had excessive wrist-pin wear. Examining this compressor further, the conclusion was that it had a very high dirt content. All bearing surfaces showed considerable scratching which did not show on the other WH410 compressors.

TABLE 4-10

Calorimeter, Sound Level, Pullout, and Start Data for Compressors (Prior to Life Test)

No.	Capacity (BTU/Hr.)	Power (Watts)	EER (BTU/WHr.)	Pullout (Volts)	Sound Level (dBA)	Start (Volts)
			WH145			
WH145-001E-A	1357	140.8	9.64	81	50.2	73
WH145-002E-1	1289	144.6	8.91	79	55.0	68
WH145-002E-2	1283	152.0	8.44	81	49.6	69
WH145-002E-3	1265	142.2	8.90	77	51.0	66
AVERAGE	1299	144.9	8.96	80	51.5	69
			WH210			
WH210-001E-A	2059	214.0	9.62	87	52.3	70
WH210-002E-1	1972	216.4	9.11	87	57.2	65
WH210-002E-2	1989	215.4	9.23	89	54.0	75
AVERAGE	2007	215.3	9.32	88	54.5	70
			WH310			
WH310-005E-B	2750	330.2	8.32	91	56.0	91
WH310-006E-B	2776	337.4	8.24	95	56.3	84
AVERAGE	2763	333.8	8.28	93	56.2	88
			WH400			
WH410-003E-1	3937	556.2	7.08	89	N/A	85
WH410-003E-2	4015	541.4	7.42	91	N/A	85
WH410-003E-3	3958	553.2	7.15	93	N/A	88
WH410-003E-4	4011	539.2	7.44	89	N/A	92
WH410-003E-5	3945	537.2	7.34	89	N/A	98
WH410-003E-6	3950	531.6	7.43	89	N/A	96
WH410-003E-7	3933	547.0	7.19	91	N/A	96
WH410-003E-8	3984	539.0	7.39	89	N/A	96
AVERAGE	3966	543.0	7.30	90	N/A	92

TABLE 4-11

Calorimeter and Sound Level Data (After 3,000 Hr. Life Test)

No.	Capacity (BTU/Hr.)	Power (Watts)	EER (BTU/WHr.)	Sound Level (dBA)
		WH145		
WH145-001E-A	1302.6	141.8	9.19	47.5
WH145-002E-1	1272.5	144.2	8.82	49.0
WH145-002E-2	1270.8	142.4	8.92	51.0
WH145-002E-3	1271.1	142.4	8.93	51.0
AVERAGE	1279.3	142.7	8.97	49.6
		WH210		
WH210-001E-A	1996.1	214.0	9.33	51.0
WH210-002E-1	1918.6	205.6	9.33	50.0
WH210-002E-2	1945.8	206.4	9.43	56.0
AVERAGE	1953.5	208.7	9.36	52.3
		WH310		
WH310-005E-B	2645.7	298.0	8.88	62.5
WH310-006E-B	2792.7	302.6	9.23	55.0
AVERAGE	2719.2	300.3	9.06	58.8
		WH410		
WH410-003E-1	3820.0	531.0	7.19	59.5
WH410-003E-2	3884.1	528.6	7.35	55.0
WH410-003E-3	3898.2	525.6	7.42	56.5
WH410-003E-4	3842.1	519.0	7.40	59.0
WH410-003E-5	3835.2	522.4	7.34	55.0
WH410-003E-6	3872.3	519.2	7.46	57.5
WH410-003E-7	3897.1	538.0	7.24	59.5
WH410-003E-8	3832.4	515.6	7.43	59.0
AVERAGE	3860.2	524.9	7.35	57.6

TABLE 4-12

Comparison of Calorimeter Test Data Before and After Life Test

	Capacity (BTU/Hr.)	Power (Watts)	EER (BTU/WHr.)	Sound Level (dBA)
		WH145		
Before Life Test	1299.0	144.9	8.96	51.5
After Life Test	1279.3	142.7	8.97	49.6
Change (%)	98.5%	98.5%	100.1%	96.3%
		WH210		
Before Life Test	2007.0	215.3	9.32	54.5
After Life Test	1953.5	208.7	9.36	52.3
Change (%)	97.3%	96.9%	100.4%	96.0%
		WH310		
Before Life Test	2763.0	333.8	8.23	56.2
After Life Test	2719.2	300.3	9.06	58.8
Change (%)	98.4%	90.0%	110.1%	104.6%
		WH410		
Before Life Test	3966.0	543.0	7.30	N/A
After Life Test	3860.2	524.9	7.35	57.6
Change (%)	97.3%	96.7%	100.7%	N/A

5. SYSTEMS TEST

While the calorimeter testing of compressors gives data on compressor capacity and efficiency at a fixed set of conditions, it is the cycling systems test that provides a more realistic estimator of actual energy savings. There are systems tests which will determine the compressor ability to run at low voltage and meet heavy load conditions. Other tests determine how well the compressor operates at high-temperature conditions.

Testing was done on a Cavalier Corporation canned drink vending machine. The particular machine we used had the designation C6-469-266. We were able to load this machine with 483 cans of soft drinks. This vendor would be regarded nominally as being a 20 case machine. A Tecumseh Products Company compressor came as original equipment. Testing was done with the Tecumseh compressor, a standard Kelvinator TH400 compressor, and the improved efficiency Kelvinator WH410.

5.1 SYSTEM TEST DESCRIPTION

Our basic testing consisted of two tests, the pulldown test and the cycling test.

5.1.1 PULLDOWN TEST

In this test the vending machine is placed in a test room that is held at 90°F. No humidity control was provided. Cans of soft drinks are loaded into the machine. The machine is allowed to stabilize, along with its load, at 90°F. A fan is used, with the vendor's door open to ensure that the cans in the back of the compartment are at equilibrium conditions. Thermocouples, located within cans of soft drinks are monitored before the start of the test to ensure that the machine is completely stabilized. The fan is then turned off and the door is closed. Kilowatthour and run-time meters are then read. The machine

is then started up at 115 volts. During the process of pulling down the load, the amperes, watts, and temperatures are monitored and recorded periodically. The period of monitoring starts out at every minute for the first ten minutes and lengthens to every hour after two hours. Readings are continued until the compressor cycles on the temperature control. Additional information is gained by reading the kilowatthour and run-time meter 24 hours after the start of the pulldown.

A variation on this test included conducting the test at a higher ambient temperature (110°F).

5.1.2 CYCLING TEST

Generally, the cycling tests were conducted primarily to gain a measure of the power consumption in actual use. The test was run from 24 hours after the beginning of the pulldown test for a period of 24 hours. The run-time meter and kilowatthour meter were read before and after the test. Due to the large refrigerated volume it appeared that even after 24 hours from the beginning of a pulldown, the unit had not stabilized. An unstabilized unit would not yield reliable test data. For this reason the cycling test was repeated three days after the start of the pulldown.

A cycling test was also run at 95 volts in a 90°F test room. This test was conducted to to determine whether operation in brown-out situations was adequate.

5.2 TEST RESULTS

Table 5-1 gives the power consumption and percent run-time for three tests with the original Tecumseh compressor, our Kelvinator TH400, and our WH410 prototype in a 90°F room temperature.

TABLE 5-1

Pulldown And Cycling Tests for Canned-Drink Vending Machines With Three Different Compressors

0 to 24 Hours After Start of Pulldown

Compressor	Energy Consumption (KWHr/day)	Run-Time (%)
AE4441A	10.05	73.3
TH400	9.87	79.0
WH410	9.16	74.8
AE4441A (Recheck)	10.90	73.8

Cycling Test

24 Hours to 48 Hours After Start of Pulldown

AE4441A	8.00	53.3
TH400	7.60	57.3
WH410	7.32	54.1
AE4441A (Recheck)	8.49	51.5

Repeat of Above Test - Fully Stabilized

AE4441A	----	----
TH400	7.56	51.6
WH410	6.97	48.0
AE4441A (Recheck)	8.09	48.4

The Tecumseh AE4441A was reinstalled in the vending machine following the testing of the TH400 and WH410 to return it to its original condition and testing was performed to ensure proper operation. The difference in power consumption for these two tests probably resulted from a difference in charge. It is quite possible that the performance of the TH400 and WH410 could be improved by adjusting the refrigerant charge or the capillary tube length to optimize them.

All of the compressors pulled down successfully in both 90°F and 110°F room temperatures with the machine four inches from the wall. All of the compressors operated successfully at 95 volts. The TH400 even operated successfully at 85 volts which is more than is needed.

It is important to note that all of the testing discussed was after installation of a kit, supplied by Cavalier Corporation, to improve the cabinet air circulation over the soft drink cans. Additional detailed information on these tests may be found in Appendix A.

5-3 COMPRESSOR TEST (FOR VENDING MACHINE SYSTEM)

After running the pulldown and cycling tests on the vending machine, it was observed that the actual operating point in the system was significantly removed from the standard rating point. For this reason we modified the test conditions so that we could learn more about the operation of the compressor in the system. The test that was developed was run at a 5°F evaporator, 110°F condenser, 115°F liquid, 95°F suction gas, with the compressor in a 90°F ambient. The standard rating point for high back-pressure compressors, operating on R-12, is given in Section 6.

Following the systems test, the compressors were calorimeter-tested at both the regular rating point and this special rating point. The results of these tests are given in Table 5-2.

TABLE 5-2

Calorimeter Test Results of Vending Machine Compressors

Compressor	Capacity (BTU/Hr.)	Current (Amps)	Input (Watts)	EER (BTU/WHr.)
Standard HBP Rating Point				
AE4441A	4136	6.80	624	6.63
TH400	4133	6.95	614	6.73
WH410	4099	5.60	499	8.21
Special HBP Rating Point				
AE4441A	2083	5.45	400	5.21
TH400	1759	5.30	350	5.02
WH410	2049	4.00	325	6.31

It is clearly shown, in Table 5-2, that the WH410 is significantly more efficient at the standard point and the special point. This combination yields energy savings during pulldowns, such as when the vending machine is loaded with soft drinks, and during daily cycling operation.

6. TEST PROCEDURES

The various test procedures and apparatus for compressor testing are described in the following sections. All of the compressor test facilities and equipment are located in the Kelvinator engineering department. Systems testing was performed in the temperature controlled test rooms located at Columbus Products Company in Columbus, Ohio. All of the motor tests were performed by the motor supplier, Athens Products Company, Athens, Tennessee.

The following sections describe the tests used in evaluating compressor designs and design changes. These tests include:

- Calorimeter Test
- Pullout Test
- Start Test
- Sound Test
- Locked Rotor Test
- Ultimate Trip Test

6.1 CALORIMETER TEST

The calorimeter test is used to measure compressor capacity and efficiency. The calorimeter has a water-cooled condenser, an automatic expansion valve, and an evaporator enclosed in a secondary refrigerant system with electric heaters. The calorimeter has pressure gauges, electrical meters, and a resistance bridge (for winding temperature). The compressor is mounted in a duct in which conditions can be simulated. The motor circuit in the calorimeter is wired for regulated 115 volt, 60 hertz power. A functional diagram of the calorimeter is shown in Figure 6-1.

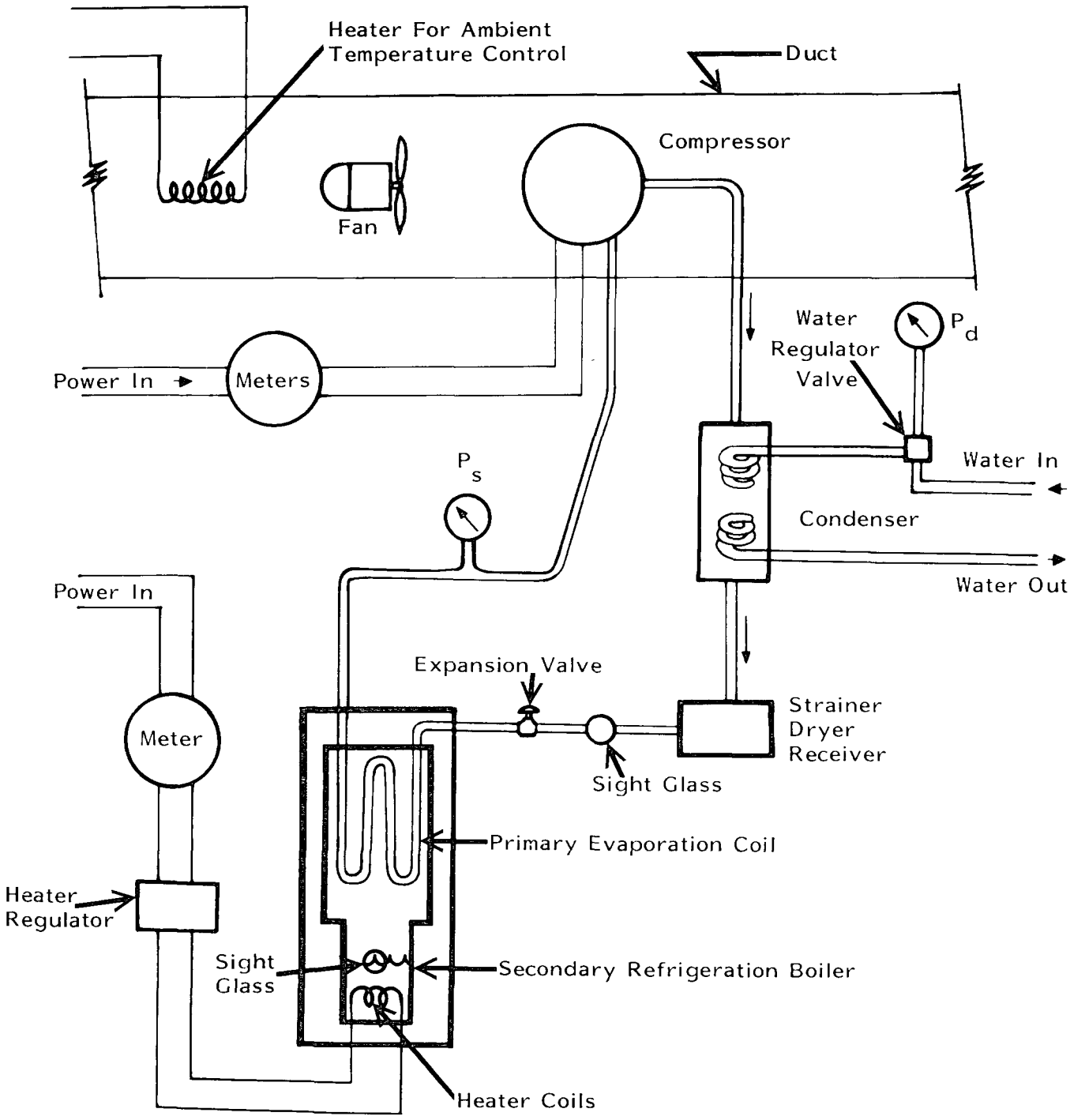

Figure 6-1. Functional Diagram of the Calorimeter

Most calorimeter testing is performed under standard rating point conditions which are chosen to represent typical operating conditions. Conditions given in Table 6-1 are for compressors using R-12 and R-500 refrigerant and intended for high back-pressure applications. These conditions were utilized during the testing of the improved-efficiency compressors for this program.

TABLE 6-1

Standard Calorimeter Rating Point Conditions

Condition	R-12	R-500
Evap. Temp. (°F)	45	45
Evap. Press. (psig)	56.4	66.5
Cond. Temp. (°F)	130	130
Cond. Press. (psig)	181	218
Suction Gas Temp. (°F)	95	95
Liquid to Exp. Temp. (°F)	95	95

For a more complete definition of compressor performance than that provided by the standard rating point test, a series of tests may be made to develop a compressor performance curve. Evaporating temperatures are set at 35°F, 45°F, and 55°F with condensing temperatures held at 110°F, 130°F, and 150°F. It is customary to plot the results of these tests as a set of curves, and the procedure is referred to as "curve testing". Such a set of curves were plotted on the final design of the WH compressors which can be seen in Appendix B.

During the calorimeter test the ambient temperature around the compressor, which is in the calorimeter duct, is 95°F. Forced air flow is directed at the compressor through the duct. Thermocouples are supplied and monitored at various locations.

In operation, cooling-water flow to the calorimeter condenser is adjusted to provide the desired discharge or condensing pressure. The expansion valve is set to an evaporator or suction pressure. The heat input to the evaporator is regulated to give 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 corrected by making a suitable mathematical calculation based on the enthalpy of the liquid. While the test is in progress, the energy supplied to the compressor is recorded. The EER in BTU/WHr. is then calculated.

The test that we employ is a primary test as defined by the American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Incorporated. No secondary test procedure was used.

6.2 PULLOUT TEST

The pullout test measures the voltage at which a compressor stalls under the specified conditions and is used to determine if it has sufficient torque for heavy load operation. The pullout test is usually performed following a calorimeter test while the compressor is mounted on a calorimeter. The test conditions for a high back-pressure compressor with R-12 is 200 psig discharge pressure and 41.7 psig suction pressure. Both the suction gas temperature and the ambient temperature are at 95°F. For R-500, the suction pressure is 52 psig while the discharge pressure is 242 psig. Suction gas and ambient temperatures remain at 95°F.

Testing consisted of reducing the line voltage in successive steps. After each decrease in voltage, the compressor is allowed to stabilize and the 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.

6.3 START TEST

This test is made to ascertain the minimum voltage at which the compressor 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.

Before starting the actual test procedure, the system 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 compressor 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°F to 75°F. The compressor is started and allowed to circulate the refrigerant through the test system until the bottom of the shell reached 150°F. After warm up, the compressor is stopped, and the pressure is allowed to equalize. The 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.

6.4 SOUND TEST

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

The test specifications require a test room isolated from outside sources of noise and vibration. The room has non-reflective walls. 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. Microphone positions used are shown on Figure 6-2. Readings are taken at compressor startup and at 5, 10, and 30 minutes after starting. Before the final reading, the compressor must be stabilized. If conditions have not stabilized in 30 minutes, the test must continue running until stabilized, before final readings are taken.

6.5 LOCKED-ROTOR TEST

The principle purposes of a locked-rotor test is to aid in the selection of a thermal overload protector which will prevent the occurence of excessive temperatures in the motor windings, if the compressor fails to start with voltage applied. The most common problem occurs when there is momentary interruption to the power supply, so that the compressor stops, but cannot restart until pressures have equalized. 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.

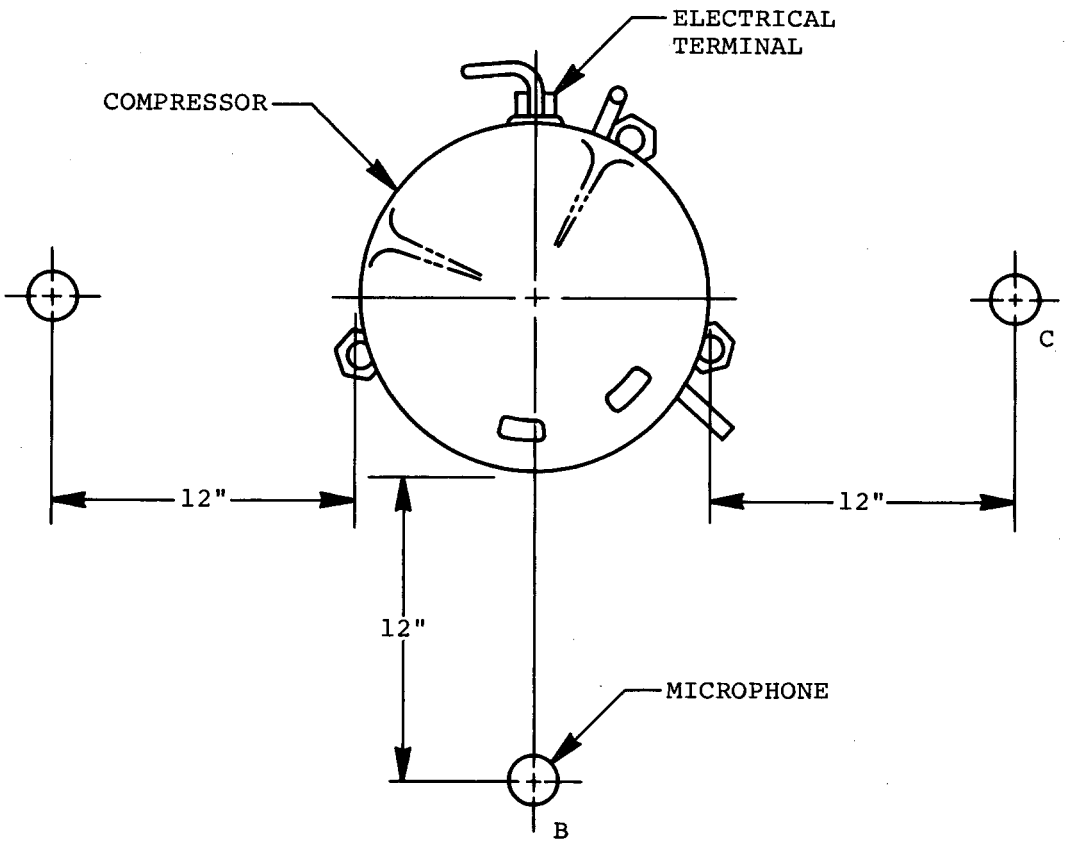

Figure 6-2. Microphone Positions

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

Compressor shell temperatures are also monitored during test. This data is used to ensure that the compressor can meet Underwriters Laboratories, Incorporated specifications. Underwriters Laboratories' specification, UL 984, states that the shell of the locked-rotor assembly may not exceed 150°C (302°F) when 120 volts, 60 Hz is applied to the assembly. (The specification only requires a 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.

6.6 ULTIMATE-TRIP TEST

The purpose of the ultimate-trip 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 tests ensure that the protector safeguards the motor under running conditions.

Ultimate-trip tests may be performed on the system the compressor is intended for, or on the calorimeter. In either case the compressor should be in a 115°F ambient. Temperatures, voltage, amps, wattage, and pressures are continually monitored throughout the test. The compressor is then started and allowed to pulldown the system or calorimeter. As the amps and watts increase, the temperatures increased until a trip occurs. Readings are then noted and recorded. During the last hour of running, the rate of rise of the main winding shall be no greater than 2°F for the trip to be considered good.

6.7 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, Figure 6-3. 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 cooling fan. In addition, a clock is used to record total run-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 51 psig, while the discharge pressure is set to 241 psig. Normal duration of the continuous run test is 4,000 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 2 minutes on and 2 minutes 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 is normally 70 to 75 psig. At compressor cutoff, the discharge pressure is to be 235 $\pm$ 10 psig, and the suction pressure is 45 $\pm$ 5 psig.

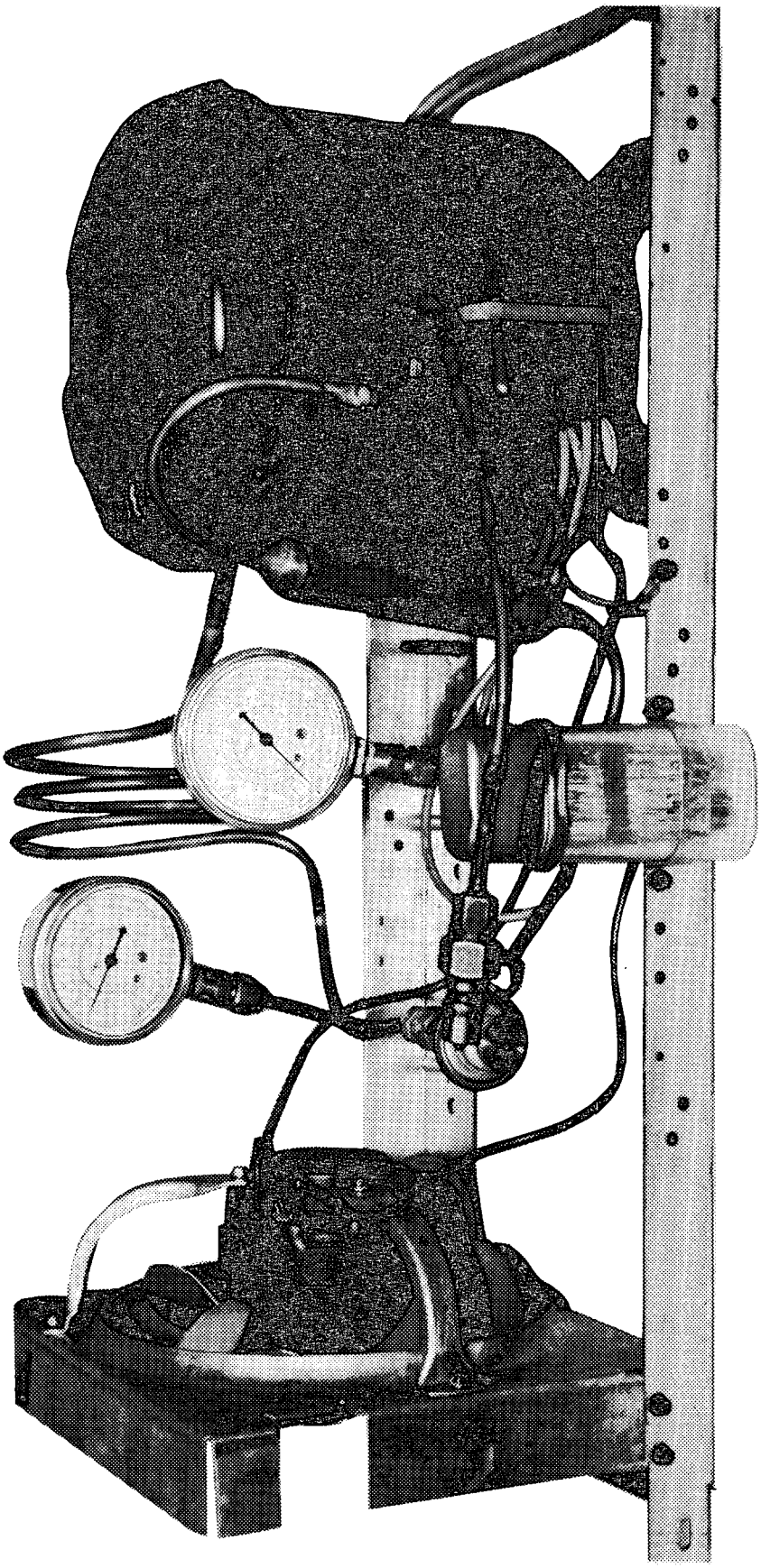

Figure 6-3. Typical Compressor on a Life Test Stand

APPENDIX A

Test Summations From Testing

Of Cavalier Corporation Can Vending

Machine C6-469-266

TABLE A-1

SUMMARY OF TEST DATA FOR A CAVALIER VENDING MACHINE

MODEL C6-469-266 WITH COMPRESSOR WH410

		90°F Initial Pulldown After 24 Hours			90°F Room Cycling After 72 Hours		110°F Pulldown	
Item	Start	Cut On	Cut Off		Cut On	Cut Off	Start	After 3 Hours
Can Temperature, °F								
Bottom Right Rear	92.2	30.2	30.4		30.5	30.	107.0	63.6
Bottom Right Front	92.0	31.3	31.3	Avg.	31.3	30.9	107.7	66.0
Bottom Left Rear	91.9	30.9	30.9	32.0	30.9	30.5	108.2	62.5
Bottom Left Front	92.0	35.6	35.8		35.0	34.5	108.3	69.2
Middle Center Rear	92.0	54.9	53.8		43.4	42.4	108.4	105.0
Middle Center Front	92.0	52.8	52.5		42.4	41.8	109.0	103.0
Top Right Rear	92.3	80.6	79.9		61.7	61.4	109.0	108.2
Top Left Front	92.4	80.4	79.6		63.3	62.5	109.0	108.0
Energy, KWH/Day			9.16			7.32		6.97
% Running Time			74.8			54.1		48.0
Peak Current, Amps.		6.1					6.55	
Peak Power, Watts		538					610	
Time to Peak, min.		20					45	
Peak Discharge Temp., °F		200					233	
Peak Condenser Midway Temp., °F		123.7					142.5	
Time to Peak, min.		30					45	
Control Cut On Temp., °F		35.5			34.5			
Control Cut Off Temp., °F			16.4			17.0		
Time to First Cut Off, hrs.		8.58 Hrs.						

Low Voltage Cycling - OK at 95 Volts

TABLE A-2

SUMMARY OF TEST DATA FOR A CAVALIER VENDING MACHINE MODEL C6-469-266 WITH COMPRESSOR TH400

Item	Start	90°F Initial Pulldown After 24 Hours: Cut On	90°F Initial Pulldown After 24 Hours: Cut Off		90°F Room Cycling After 72 Hours: Cut On	90°F Room Cycling After 72 Hours: Cut Off	110°F Pulldown: Start	110°F Pulldown: After 3 Hours
Can Temperature, °F								
Bottom Right Rear	93.2	29.3	29.5		29.2	20.3	107	59.0
Bottom Right Front	93.2	30.1	30.2	Avg.	29.8	30.1	106.8	61.8
Bottom Left Rear	93.2	30.0	29.9	30.2	29.6	29.2	107	59.0
Bottom Left Front	92.5	31.5	31.4		31.2	31.7	105	63.4
Middle Center Rear	93.4	55.2	55.1		42.1	42.2	106.8	103.1
Middle Center Front	93.5	53.1	52.8		41.3	41.4	107.2	101.2
Top Right Rear	93.6	81	80.9		62.3	62.3	109.3	108.0
Top Left Front	93.7	81.2	81.1		61.8	61.7	109.2	107.8
Energy, KWH/Day			9.87		7.60			7.56
% Running Time			79.0		57.3			51.6
Peak Current, Amps		7.68					8.46	
Peak Power, Watts		612					725	
Time to Peak, min.		20					20	
Peak Discharge Temp., °F		218					259	
Peak Condenser Midway Temp., °F		125					143	
Time to Peak, min.		45					45	
Control Cut On Temp., °F		35.3		35.0				
Control Cut Off Temp., °F			15.5			15.3		
Time to First Cut Off, hrs.		10 Hrs.						

Low Voltage Cycling - Compressor Starts and Runs at 85 Volts; Lights are erratic

TABLE A-3

SUMMARY OF TEST DATA FOR A CAVALIER VENDING MACHINE

MODEL C6-469-266 WITH COMPRESSOR AE4441A

Item	Start	90°F Initial Pulldown After 24 Hours: Cut On	90°F Initial Pulldown After 24 Hours: Cut Off		90°F Room Cycling After 72 Hours: Cut On	90°F Room Cycling After 72 Hours: Cut Off	110°F Pulldown: Start	110°F Pulldown: After 3 Hours
Can Temperature, °F								
Bottom Right Rear	90	31.3	31.3		31.4	31.5	106.2	59.9
Bottom Right Front	90.1	32.2	31.9	Avg.	32.2	32.1	106.2	62.0
Bottom Left Rear	90.3	31.4	31.2	31.7	31.5	31.2	106.2	59.4
Bottom Left Front	90.4	32.3	32.0		32.2	32.0	106.2	63.0
Middle Center Rear	90.5	37.3	36.3		37.0	36.3	108.4	103.0
Middle Center Front	90.6	36.9	36.2		36.5	36.0	106.5	100.6
Top Right Rear	92.2	80.0	80.0		78.57	77.8	107.6	107.0
Top Left Front	92.3	80.0	80.0		78.2	78.0	107.3	106.7
Energy, KWH/Day			10.05					8.0
% Running Time			73.3					53.3
Peak Current, Amps		7.55					8.2	
Peak Power, Watts		627					720	
Time to Peak, min.		20					30	
Peak Discharge Temp.. °F		223					261	
Peak Condenser Midway Temp., °F		124					147	
Time to Peak, min.		90					60	
Control Cut On Temp., °F		-			No Trips			
Control Cut Off Temp., °F		-						
Time to First Cut Off, hrs.		8 Hrs.						

Low Voltage Cycling - OK at 95 Volts; Would Not Start at 90 Volts.

TABLE A-4

SUMMARY OF TEST DATA FOR A CAVALIER VENDING MACHINE MODEL C6-469-266 WITH COMPRESSOR RECHECK AE4441A

Item	Start	90°F Initial Pulldown After 24 Hours			90°F Room Cycling After 48 Hours	
		Cut On	Cut Off		Cut On	Cut Off
Can Temperature, °F						
Bottom Right Rear	90.1	30.0	30.0		30.5	30.0
Bottom Right Front	90.0	31.0	31.0	Avg.	31.0	30.8
Bottom Left Rear	90.1	30.2	30.0	31.7	30.2	30.0
Bottom Left Front	90.0	36.0	35.6		35.5	35.0
Middle Center Rear	90.2	53.0	53.0		45.3	45.1
Middle Center Front	90.2	51.	51.		44.5	44.2
Top Right Rear	90.2	78.	78.		67.7	67.3
Top Left Front	90.1	78.	78.		67.3	67.0
Energy, KWH/Day		10.90		8.49		8.09
% Running Time		73.8		51.5		48.4
Peak Current, Amps		8.05				
Peak Power, Watts		685				
Time to Peak, min.		15				
Peak Discharge Temp., °F		221				
Peak Condenser Midway Temp., °F		124				
Time to Peak, min.		60				
Control Cut On Temp., °F		35.5			33.5	
Control Cut Off Temp., °F			14.5			14.5
Time to First Cut Off, hrs.	8.83 Hrs.					

APPENDIX B

Calorimeter Data Curves

for the

WH Compressor

Final Design

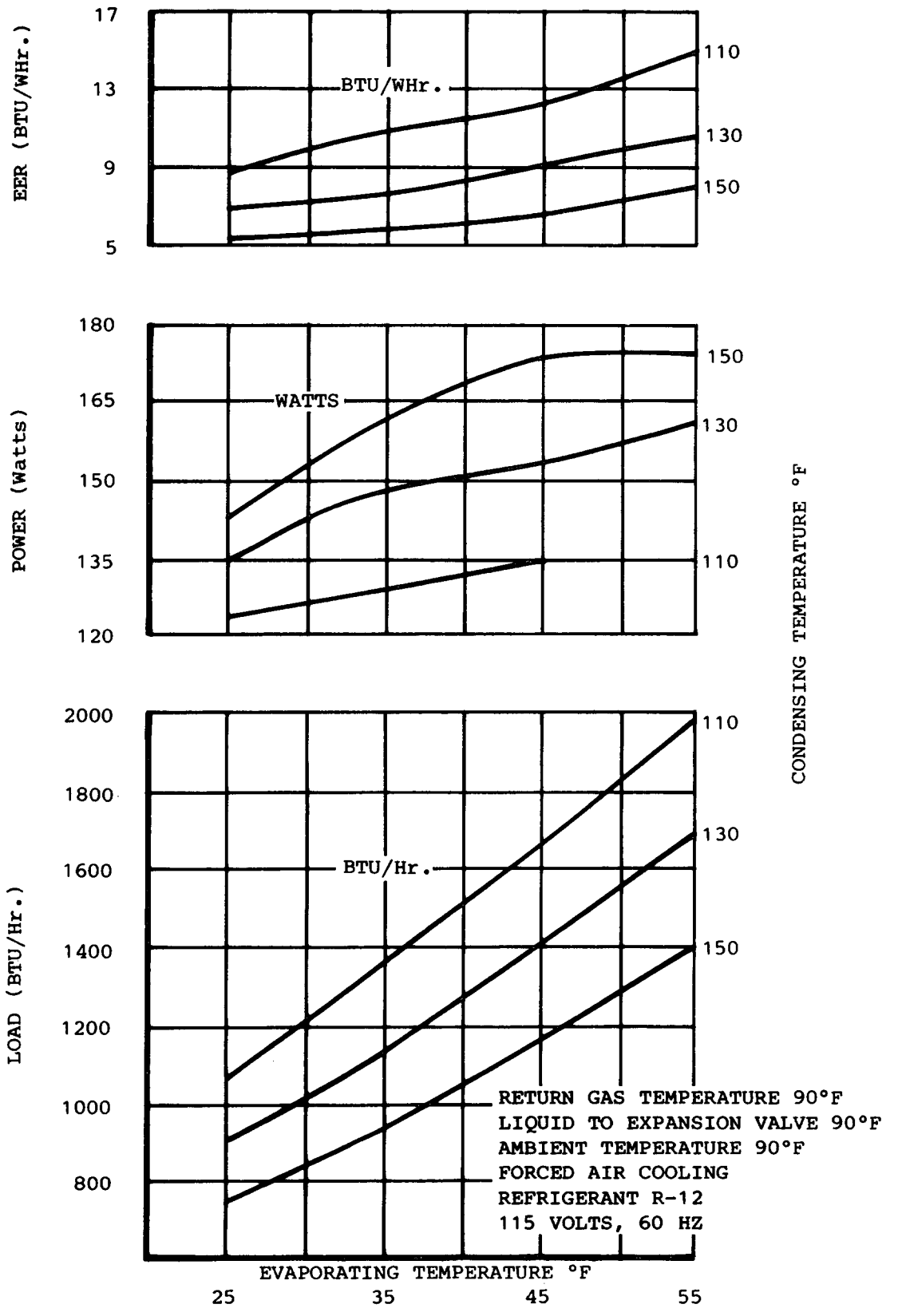

Figure B-1. Calorimeter Data Curve for the WH145 Compressor

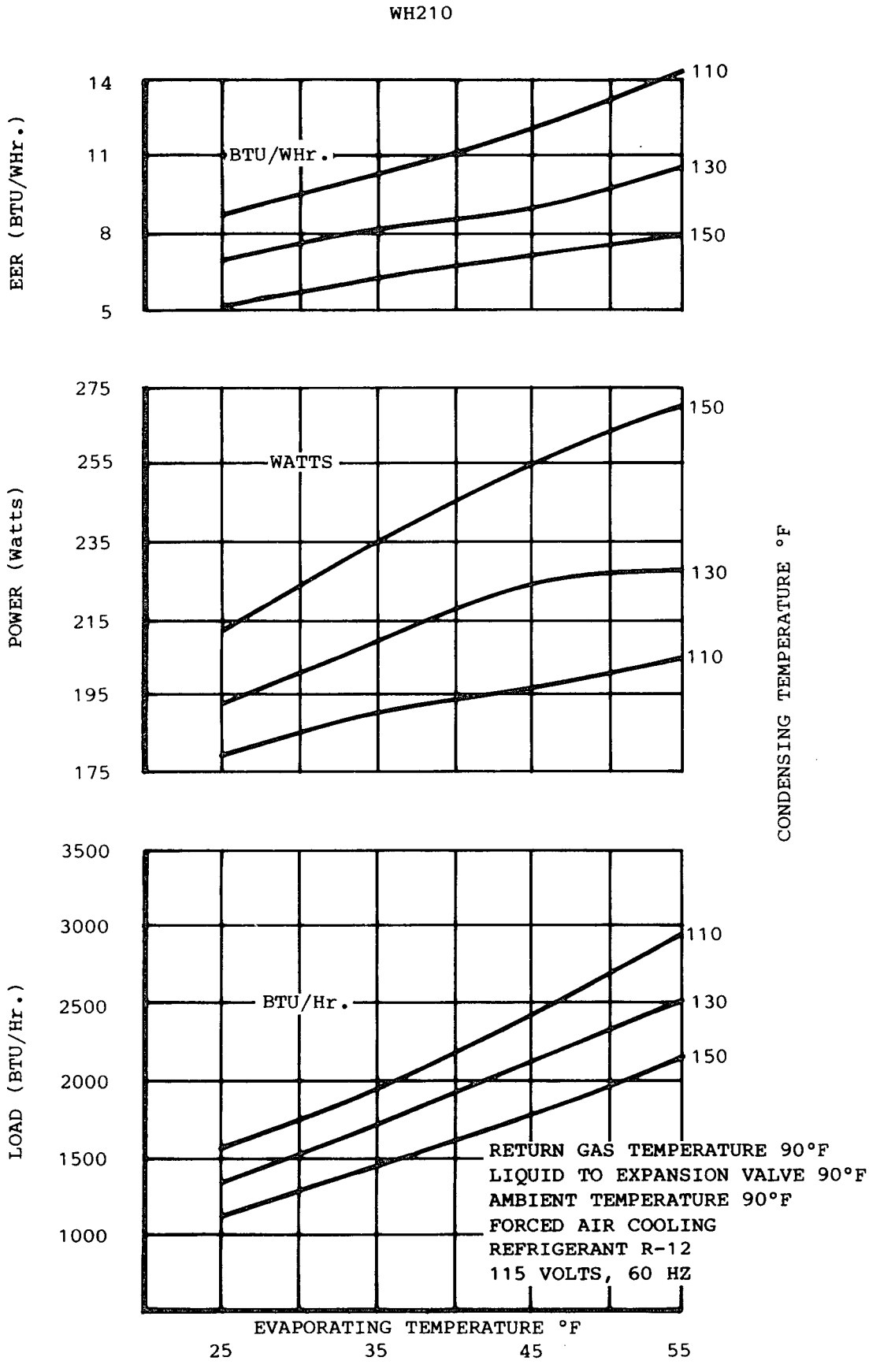

Figure B-2. Calorimeter Data Curve for the WH210 Compressor

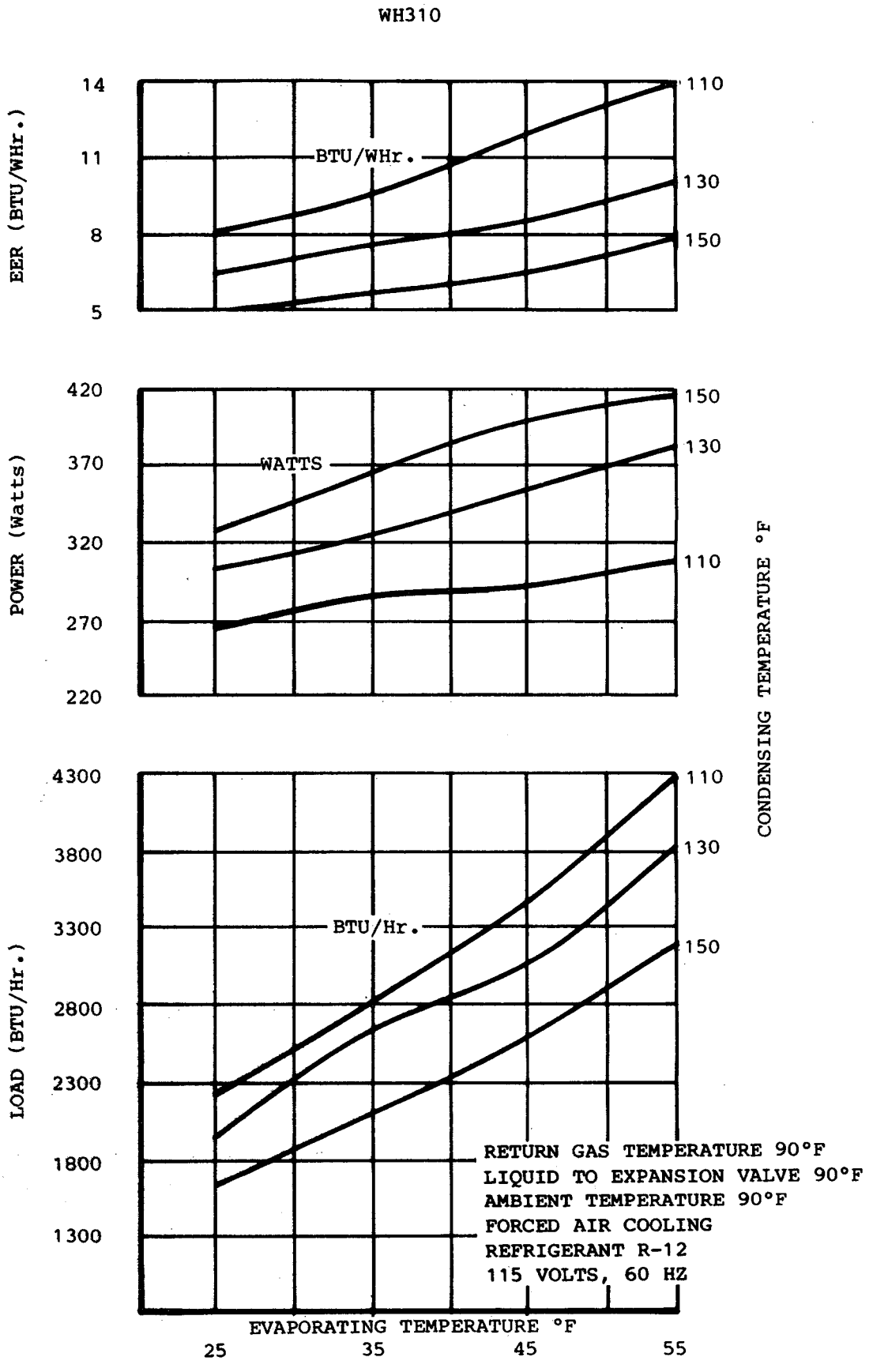

Figure B-3. Calorimeter Data Curve for the WH310 Compressor

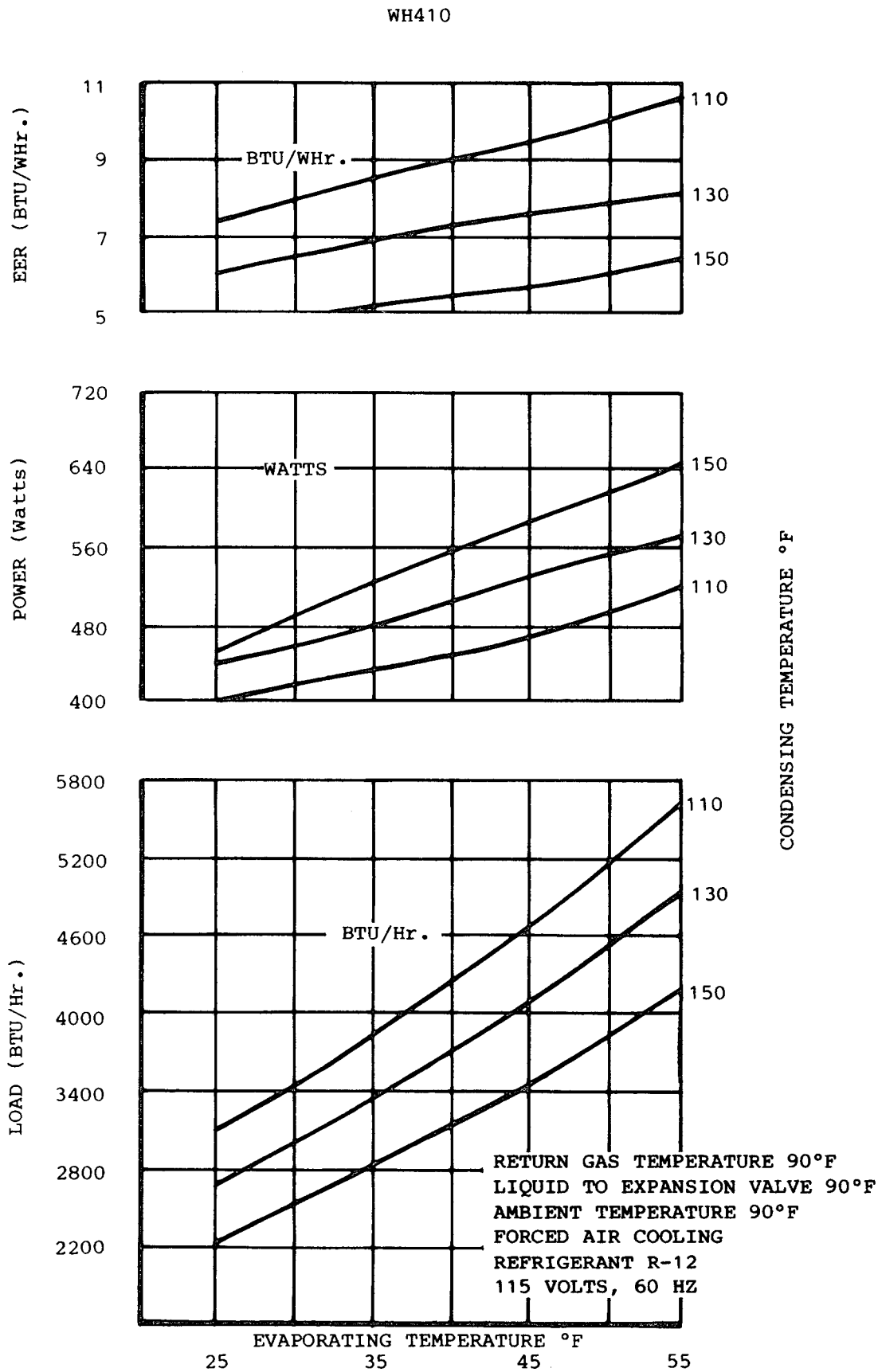

Figure B-4. Calorimeter Data Curve for the WH410 Compressor