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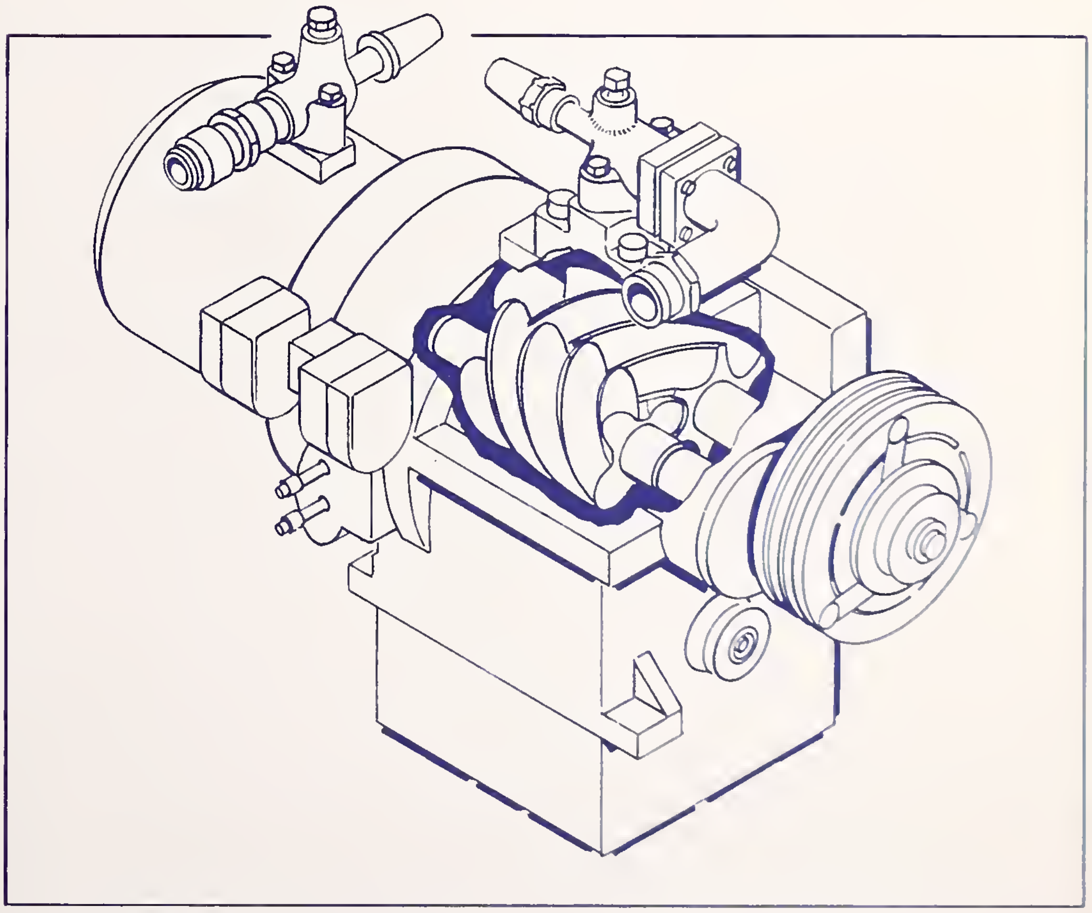
of Transportation
Urban Mass Transportation Administration

Alternative Technology For Transit Bus Air Conditioning: The Rotary Screw Compressor

Office of Technical Assistance
Office of Bus Technology

November 1984
Final Report



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16. Abstract

This report summarizes the results of the test and evaluation of a prototype rotary screw compressor design. The UMTA-funded R&D program consisted of two phases. The objectives of the first phase were to ascertain the extent of the problems with current bus air conditioning systems and to determine the feasibility of adapting alternative compressor technology for use in transit buses. This work was carried out by the (Garrett) AiResearch Manufacturing Co. of Torrance, California and Dunham-Bush, Inc. of W. Hartford, Connecticut and has been documented in UMTA Report No. CA-06-0145-80-1 (NTIS No. PB-215-502). The second phase was to test a prototype alternative compressor under a wide range of simulated, and actual, bus revenue service environmental conditions and was also conducted by Dunham-Bush, Inc. It is the results of this effort that are documented in this report.

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PREFACE

This document provides information on the testing and evaluation of a prototype rotary screw compressor for a bus air conditioning system. This work represents the second, and final, phase of an UMTA program to develop a bus climate control system with improved reliability and durability, as well as lower overall costs.

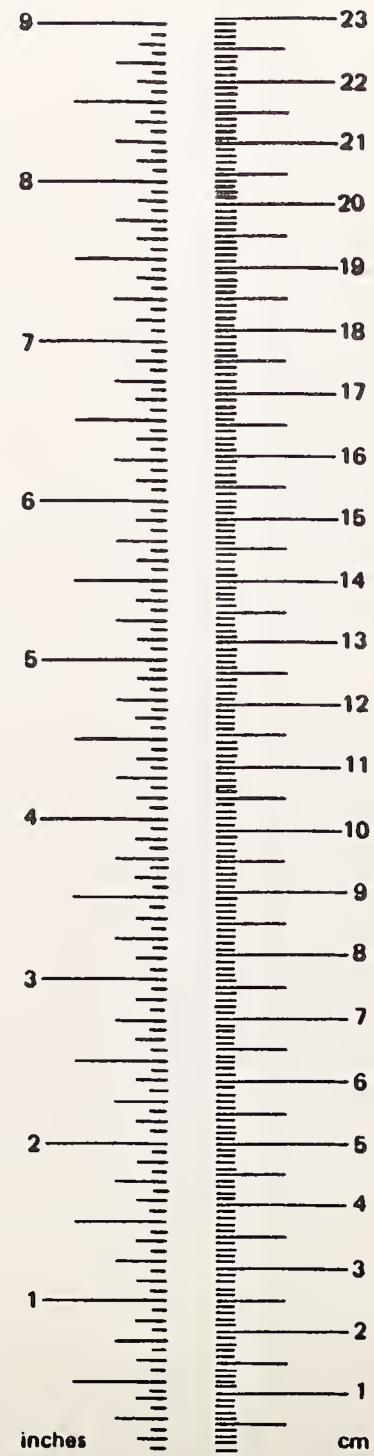
The Transportation Systems Center wishes to acknowledge the fine efforts by Dunham-Bush, Inc. in conducting the test program described in this report. Special thanks are extended to John Halay, Project Engineer, and Robert Gabelmann, air conditioning technician. In addition, gratitude is expressed to personnel at all levels at VIA Metropolitan Transit in San Antonio, Texas. The fine spirit of cooperation exhibited by Wayne Cook, General Manager, Wayne Hale, Manager of Maintenance, and Marvin Bielefeld, Maintenance Coordinator, in particular, contributed to the successful completion of this project.

METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

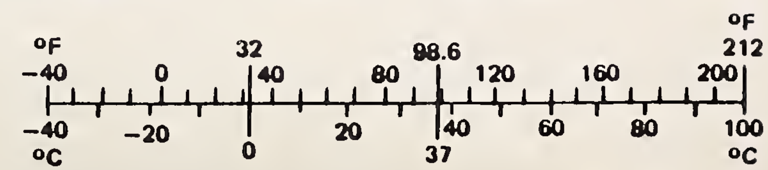
Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C

*1 in. = 2.54 cm (exactly). For other exact conversions and more detail tables see NBS Misc. Publ. 286. Units of Weight and Measures. Price $2.25 SD Catalog No. C13 10 286.



Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	36	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



EXECUTIVE SUMMARY

As part of its bus and paratransit technology program, the Urban Mass Transportation Administration (UMTA) is providing technical assistance to the transit bus industry to help solve serious equipment problems. The principal objective of this assistance is to reduce the recurring costs necessary to operate the U.S. transit bus fleet by developing and evaluating improved subsystem technologies that can be adopted by operators and manufacturers in the near future. Consequently, UMTA's Office of Technical Assistance has sponsored a project aimed at improving the reliability of transit bus air conditioning systems, one of the most troublesome problems confronting urban transit operators today. Air conditioning system breakdowns are having significant adverse effects on general fleet reliability, maintainability, efficiency and, consequently, operational costs.

This UMTA-sponsored project was aimed at reducing those air conditioning system failures that resulted from breakdowns of the refrigeration compressor by developing a different type compressor for use in transit bus air conditioning systems. Today, virtually all air conditioned buses are equipped with a reciprocating (piston-type) compressor which was adapted for transit use from the relatively benign environment of stationary applications. As a result, this type of compressor has had a history of generally unreliable operation at many transit agencies due to the transit bus' hostile environment of shock, vibration, dirt, high temperature and a constantly varying-speed drive mechanism.

A non-reciprocating compressor has been developed to better withstand these severe operating conditions. This compressor contains a pair of counter-rotating, screw-profiled rotors that mesh together to compress the refrigerant gas in an axial direction with compressor capacity being varied by means of an axial slide valve. This design, called a "rotary screw" compressor, has fewer moving parts than the reciprocal type and appears much less prone to breakdowns than the reciprocating equipment currently in use. Preliminary testing conducted during the first phase of the project determined the feasibility of adapting existing ("large-size") rotary screw compressor technology to the smaller size required for transit bus applications. Expanded laboratory and field tests were then conducted in the second phase to establish the capability of the rotary screw compressor to perform in the transit bus environment.

This report documents the activity and results that were obtained during the second phase of the project. Measurements of compressor performance and reliability were obtained under simulated environmental conditions in a laboratory setting, and under actual revenue service conditions. The laboratory testing was composed of a 500-hour performance test and a 2000-hour durability test, both conducted under conditions of heat, shock and a cooling load simulating the transit bus revenue service environment. The field test involved installing and running a prototype compressor on a GMC RTS-II bus at San Antonio, Texas, for 30 days of actual summer revenue service. Both the laboratory and field test units were fully instrumented, thereby providing an accurate picture of the ability of the rotary screw compressor to perform under the imposed conditions.

Results obtained from the tests show that a rotary screw compressor can adequately and reliably perform in the transit bus environment while also providing a small improvement in fuel economy. No failures or maintenance problems have been encountered with the field test compressor which is still in revenue service as of this writing. With the initiation of full-scale revenue service applications of the manufacturers' production units at several transit agencies during the up-coming cooling season, additional data will be obtained on the performance and cost effectiveness of this alternative to the conventional transit bus air conditioning compressor.

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1.0 Introduction

American transit authorities have been plagued for years by the high maintenance and the high operating costs of bus air conditioning systems. Since their widespread introduction in 1959, these systems have utilized a conventional reciprocating (piston) compressor which is difficult to maintain in the transit bus application because of such factors as dirt, vibration, and the inherent problem of matching compressor capacity to the constantly varying cooling load.

Because these systems have required excessive maintenance and exhibited a low degree of reliability, the Urban Mass Transportation Administration, (UMTA), Office of Technical Assistance, initiated a co-operative R&D program, in 1979, with private industry, to develop a more reliable alternative to the reciprocating compressor and to test and evaluate this alternative under actual revenue service conditions.

This report summarizes the results of the test and evaluation of a prototype rotary screw compressor design. Such information should be of value to transit operating and maintenance personnel who are concerned with improving reliability of bus air conditioning systems.

2.0 Background

The UMTA-funded R & D program consisted of two phases. The objectives of the first phase were to ascertain the extent of the problems with current bus air conditioning systems and to determine the feasibility of adapting alternative compressor technology for use in transit buses. This work was carried out by the (Garrett) AiResearch Manufacturing Co. of Torrance, California and Dunham-Bush, Inc. of W. Hartford, Connecticut and has been documented in UMTA Report No. CA-06-0145-80-1 (NTIS No. PB-215-502).

The second phase had the objective of testing a prototype alternative compressor under a wide range of simulated, and actual, bus revenue service environmental conditions and was also conducted by Dunham-Bush, Inc. It is the results of this effort that are documented in this report.

2.1 Problem Identification

In the first phase of the program, the extent of the air conditioning problem was developed from questionnaires to transit agencies, through consultations with agency personnel, and through a study of data supplied by the American Public Transit Association (APTA). Table 1 provides information thus obtained showing high rates of compressor-related breakdowns, primarily due to the high operating temperatures and system pressures found in the harsh environment of transit. An additional factor affecting reciprocating compressor performance is its characteristic inability to be cycled off and on in response to variations in cooling load. Instead, the conventional system reacts to reduced cooling loads, initially, by re-heating the cooled air. Further capacity reduction, if required, is accomplished by unloading two or more cylinders, thereby imposing an unbalanced load on the driving mechanism. This method of capacity control is responsible for increased wear on the compressor system as well as fuel penalties on the air conditioned bus. Finally, while the older, "New-Look" buses have always had openable windows, thereby providing a measure of relief from air conditioning system failures, the fixed-window feature of the newer ADB buses required their removal from revenue service whenever air conditioning system failures occurred. Therefore, a key conclusion of this portion of the program was to focus research on an alternative concept for compressor design.

TABLE 1. COMPRESSOR CLUTCH ASSEMBLY REPLACEMENT RATES ON "NEW-LOOK" BUSES FOR ONE YEAR*

Transit Agency	Percent Replaced Per Year
CTA - Chicago, Ill.	24
VIA - San Antonio, Tex.	43
PTS - Phoenix, Ari.	60
OCTD - Garden Grove, Cal.	63
DTS - Dallas, Tex.	68

* 1978-80 time frame

2.2 Feasibility Study

The second portion of the Phase I effort addressed the feasibility of adapting rotary screw refrigeration compressor technology as an alternative to the conventional compressor for transit bus air conditioning systems. Rotary screw compressors have been employed for a number of years in large (50 ton plus) stationary applications for air conditioning and refrigeration in the chemical, food processing and environmental control industries. However, although the reliability and durability of these larger sized units has been confirmed, development work was required to evaluate a scaled-down (nom. 10 ton) "bus-size" screw compressor.

A reduced-size, rotary screw compressor was designed, "hand-built" and bench tested at the Dunham-Bush, Inc. facilities on an ASHRAE* - approved refrigeration gas-cycle test stand. While measured performance results were somewhat lower than predicted due to the larger-than-normal machining tolerances and assembly clearances of the "hand-built" unit, satisfactory results were obtained over a wide range of test conditions, thus demonstrating the technical feasibility of the smaller compressor.

Based on these bench-test results, it was recommended that the smaller, "bus-size", rotary screw compressor, using R-12 refrigerant, and a solid state microprocessor/controller be further developed and tested as a replacement for the conventional systems currently being supplied in transit buses. Designing the rotary screw compressor to operate with lower pressure R-12 refrigerant would result in a reduction of the high temperatures and pressures being experienced with the use of R-22 in conventional systems. Additional benefits could be obtained through the use of a microprocessor/controller which would take advantage of the inherent capacity-varying capability of the screw compressor to more efficiently match cooling loads and eliminate reheat requirements except under occasional light load, high humidity conditions.

* American Society of Heating, Refrigerating and Air-Conditioning Engineers

2.3 Rotary Screw Compressor System

The rotary screw compressor differs significantly from the piston type in both configuration and operation. It uses two mating helical rotors (Fig. 1) that simultaneously carry the suction gas circumferentially while squeezing it axially toward the discharge end, similar to the way in which a kitchen meat grinder works (Figs. 2 & 3). It operates in a smooth, non-pulsating, continuous motion, since the intake and discharge cycles of several pairs of rotor lobes overlap. This provides an even flow of gas and a uniform shaft torque loading.

The compressor output (capacity) is varied by use of a slide valve which alters the effective length of the rotors by moving axially to open or close bypass ports, thus changing the volume of discharge gas (Fig. 4). The slide valve can be positioned at one (1) of three (3) discreet settings, i.e., 30%, 60% or 100% of compressor capacity, in response to changes in return air temperature and to excessive humidity as well as to suction and discharge pressure variations. Control of the slide valve is accomplished by means of a microprocessor which uses electronic sensors and digital logic to provide control functions and fail-safe compressor operation.

The rotary design of the compressor, coupled with its fewer moving parts (rotors, slide valve and bearings) allow it to be stopped and started (at 30% load) at any engine speed as required, thereby eliminating most reheat requirements. Overall system reliability should be improved due to the inherent capability of the solid state controller to better withstand the harsh transit environment and by the simpler design of the compressor itself.

FIGURE 1. COMPRESSOR ROTOR SET

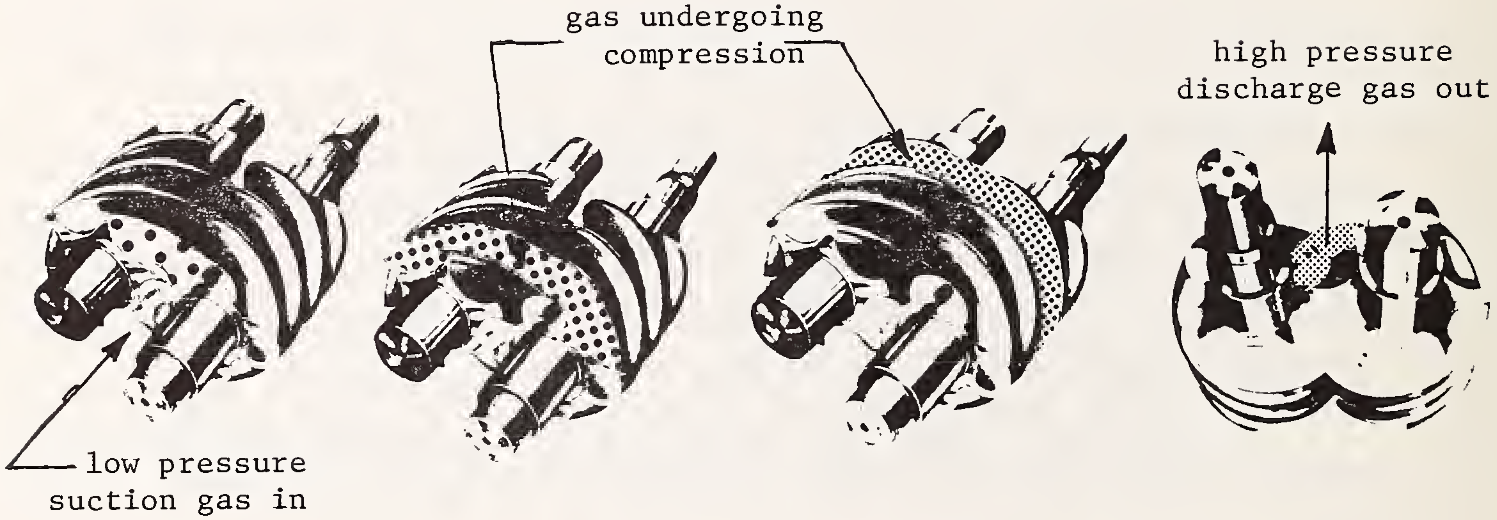


FIGURE 2. WORKING PHASES IN A SCREW COMPRESSOR

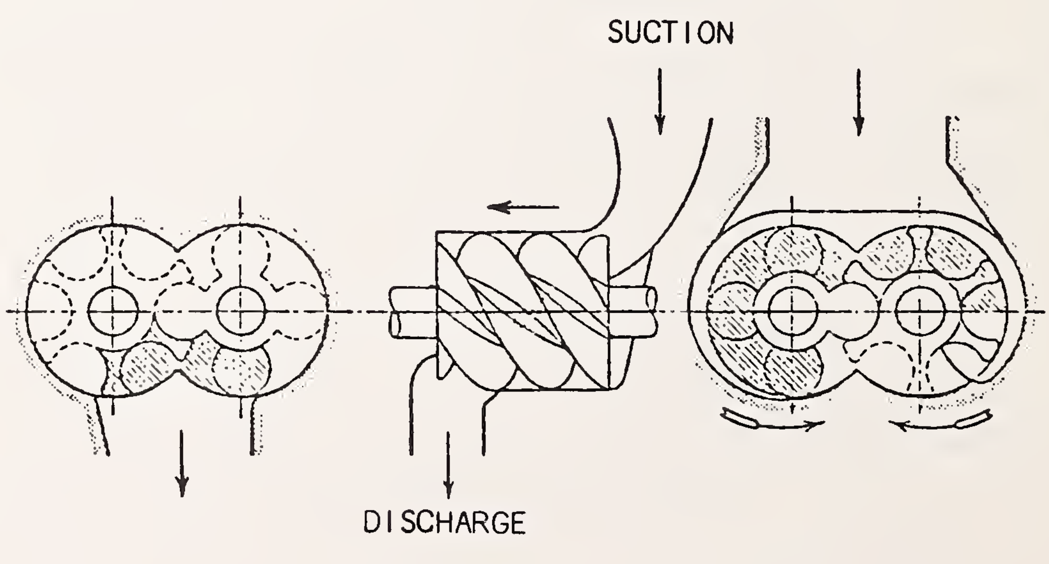
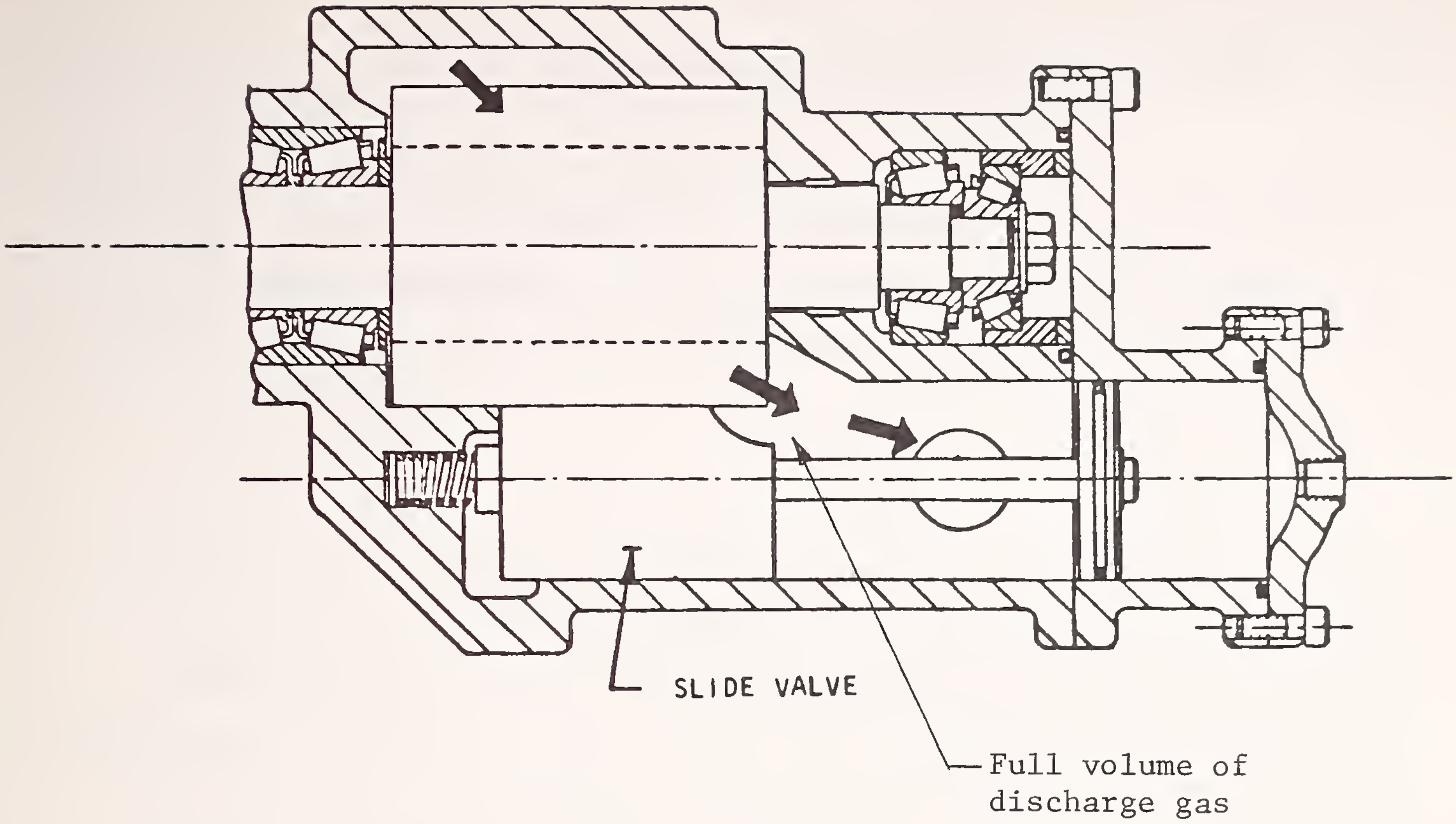
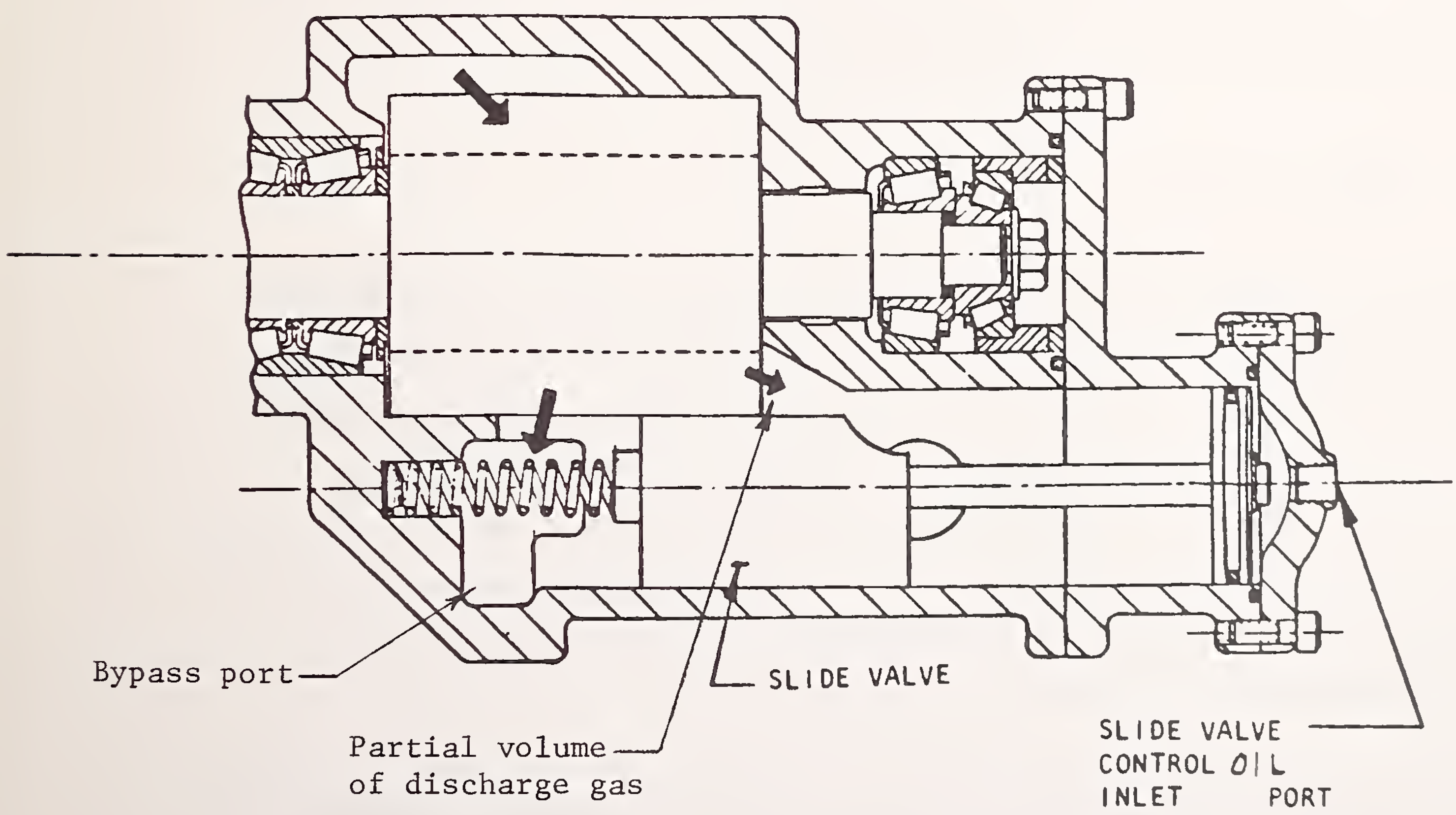


FIGURE 3. RADIAL AND AXIAL FLOW IN A SCREW COMPRESSOR



(a) Fully Loaded Position



(b) Unloaded Position

FIGURE 4. OPERATION OF SLIDE VALVE CAPACITY CONTROL

3.0 Phase II - Prototype Test and Evaluation

The purpose of the Phase II effort was to perform extensive developmental testing of prototype rotary screw compressors to establish the capability of these units to perform under a variety of environmental conditions, particularly those associated with transit bus revenue service.

3.1 Test Program

The first portion of the testing program for the second phase of the air conditioning study consisted of a 500-hour performance test and a 2000-hour continuous operation durability test of a prototype screw compressor/controller package, both of which were conducted in controlled, laboratory settings beginning during the early months of 1982. Problem-free compressor performance during the early stages of these laboratory tests subsequently allowed for the retrofit, as scheduled, of a prototype compressor and controller into a transit authority bus for the "acid test" of actual revenue service during the summer of 1982. These laboratory and field tests were all conducted for AiResearch Mfg. Co., under funding from UMTA, by Dunham-Bush, Inc.

3.1.1 500-Hour Test

This test was conducted in an "artificial climate" room at Dunham-Bush, Inc. laboratories where the temperature and humidity could be varied as desired. An urban run profile, typical of a downtown business district bus run (the most demanding test of a transit bus air conditioning system), was used to simulate actual operating cycles. GMC RTS-II-04 bus air conditioning system components were used in the laboratory set-up to confirm the compatibility of the rotary-screw compressor with representative O.E.M. transit bus hardware.

More than 250, simulated 5-mile route cycles were run which included over 10,000 stop/door open sequences. Different sets of environmental conditions--outdoor and bus interior temperature and humidity levels--were imposed on the system and held for various lengths of time in order to observe the capability of the compressor to react to, and handle, different load situations including minimum loads for on-off operation.

System pressures and temperatures were recorded and coefficients of performance for the system were calculated. After debugging, no operational difficulties were encountered and the compressor and controller proved capable of performing satisfactorily under all of the environmental conditions and loads imposed during the test period. Table 2 outlines the 500-hour test segment of the program. Additional details can be found in Appendix A.

TABLE 2. 500-HOUR TEST

Objectives

- To demonstrate the capability of a screw compressor to operate a bus air conditioning system.
- To demonstrate the compatibility of a screw compressor/control system with the remainder of the conventional bus air conditioning system components.

Test Conditions

- Laboratory setting-environmental chamber.
- Integrated into OEM bus air conditioning system.

Parameters

- Simulated "5-mile" urban run (CBD) profile - 250 cycles.
- Varying ambient and bus interior temperatures and humidity levels.

Results

- Satisfactory operation of compressor/controller package with bus air conditioning system hardware.
- Return air (bus interior) temperature controlled without excessive cycling of capacity control elements.

Recommendations

- Increase high temperature cut-out and liquid injection temperatures to avoid nuisance shut-downs and low discharge gas superheat at high condensing temperatures.
- Increase low pressure cut-out, coil temperature sensor and load-up signal delays to avoid nuisance shutdowns and eliminate a low pressure switch.

3.1.2 2000-Hour Test

For the 2000-hour test, also conducted at Dunham-Bush, Inc. laboratories, the compressor was programmed to run through a complete duty sequence - on at 30%, 100%, 30%, unload (declutch) - every 35 minutes (over 3400 cycles) for the duration of the test. In addition, the compressor/clutch assembly was subjected to a 260^{0} F ambient temperature for more than 600 of the 2000 hours, and to a shock loading of 3.5g within 50 milliseconds every 6 minutes throughout the entire test period. After replacement, during the early stages, of a marginally designed clutch and a non-high temperature resistant shaft seal with properly matched components, the compressor completed the test satisfactorily with no other component failures or problems.

All relevant data, e.g., temperatures, pressures, electrical inputs, etc., were recorded periodically and subsequently used with a computer program to calculate a number of compressor/system performance parameters. Specific volume, enthalpy, cooling capacity, volumetric and energy efficiencies, and performance factors were calculated at intervals throughout the test. After a total running time of almost 2850 hours, the compressor was disassembled and inspected. Careful examination of the slide valve, rotors, bearings, seals and other wear surfaces disclosed only minimal amounts of wear in all areas except for portions of the rotor tips. Here, the evaluation indicated that a modification in the heat treating process was required and this change has been implemented for all production units. Table 3 outlines this durability test section of the program. Additional details and sample performance results calculations can be found in Appendix B.

TABLE 3. 2000-HOUR TEST

Objectives

- To demonstrate the capability of a screw compressor to operate reliably under severe environmental conditions.
- To initiate an investigation of the operating life of a screw compressor under such conditions.

Test Conditions

- Laboratory setting - gas loop.
- Test stand modified to provide heat and shock loadings.

Parameters

- Simulated urban run (CBD) profile - 3400 cycles.
- Programmed for off/on - max. capacity - off loadings, 260^{o}F environmental temperature and periodic 3.5g shock load.

Results

- No compressor failures during 2850 hours of testing.
- Minimal wear on all moving parts.

Recommendations

- Redesign clutch for increased torque capacity.
- Change shaft seal and slide valve load piston seal materials for extra-high temperature service.
- Change rotor tip heat-treating process to reduce work stresses.

3.1.3 Revenue Service Test

For the field test, a prototype compressor and controller were retrofitted into a GMC RTS-II-04 coach at VIA Metropolitan Transit in San Antonio. Replacement of the cage assembly in the evaporator expansion valve (for use with R-12), and insertion of a tee after the dehydrator for liquid injection were the only other modifications to the bus air conditioning system.

A digital print-out Data Logger was installed on the bus to automatically record bus temperatures and control system functions. Manual recordings were made of additional data such as passenger loadings, numbers of stops/door openings, mileage, fuel consumption, etc. Data was recorded for more than 250 hours of actual operation during 2 months of revenue service and totaled approximately 100,000 data points. Figure 5 shows, in graphic form, conditions observed during typical revenue service duty.

Except for a piping change to the compressor and some minor control and wiring problems, the only other difficulties encountered were concerned with the instrumentation components. Once these were solved, the system maintained comfortable passenger compartment temperature levels throughout the duration of the field test.

At the conclusion of the data gathering portion of the revenue service test, the instrumentation hardware was removed from the test bus. The rotary screw compressor/controller package was left in the bus and continues to operate satisfactorily. Table 4 outlines the revenue service test portion of the program. Additional details can be found in Appendix C.

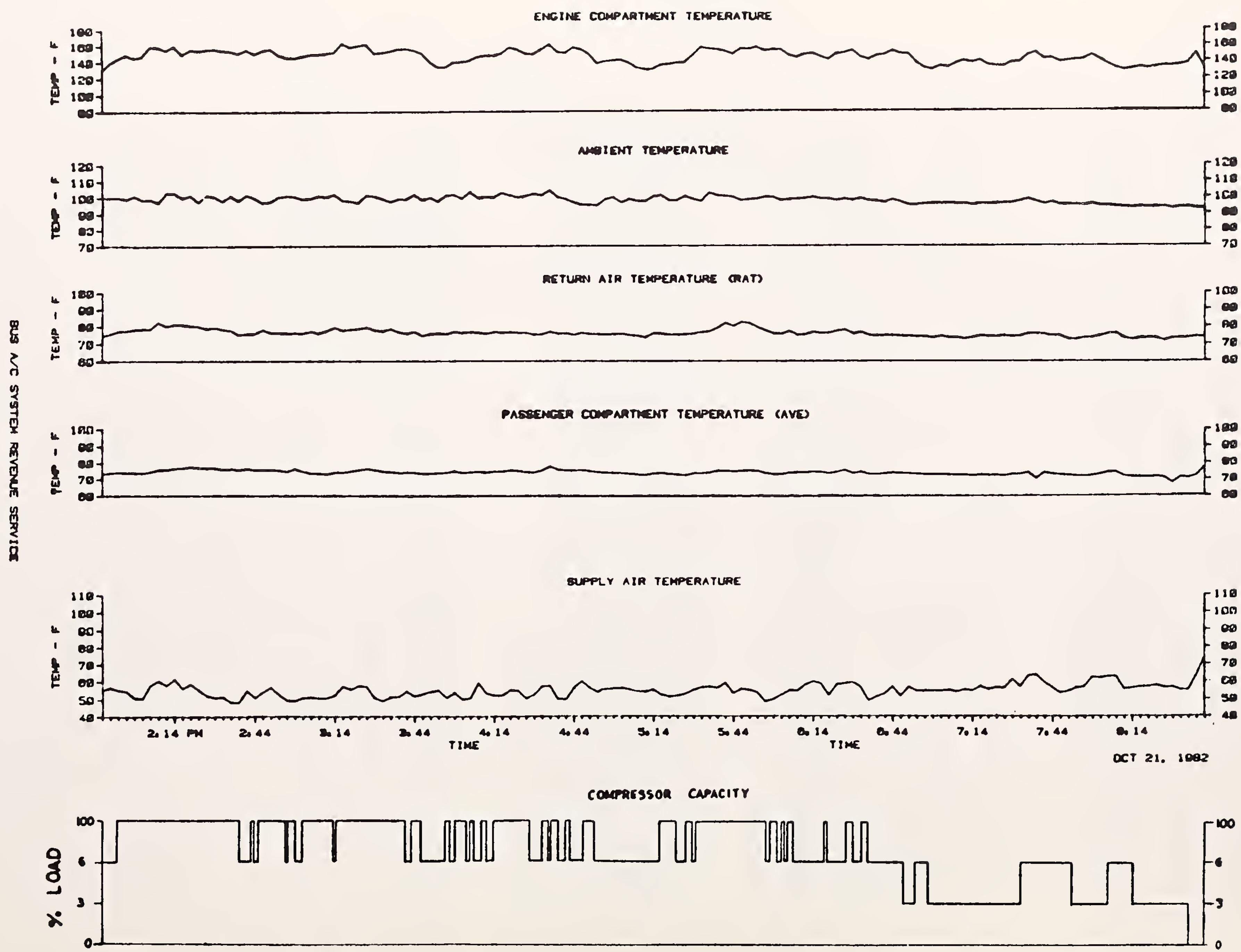


FIGURE 5. BUS AIR CONDITIONING SYSTEM OPERATIONAL TEST CONDITIONS IN REVENUE SERVICE

TABLE 4. REVENUE SERVICE TEST

Objectives

- To demonstrate the capability of a screw compressor to keep a transit bus cool while in actual revenue service.
- To determine the savings in fuel consumption obtained with the operation of a screw compressor versus a reciprocating compressor.

Test Conditions

- Retrofit of compressor/controller package into a GMC RTS-II-04 coach.
- Urban/suburban duty cycles.

Parameters

- Typical 14 hour day, southern U.S. summer climate, actual revenue service.

Results

- Bus interior maintained within comfort range at all times.
- 3% improvement in fuel consumption during test period between test bus and control bus.
- Low vibration and transmission noise levels.
- No mechanical problems.
- No maintenance required.

Recommendations

- Change operation of reheat control to timed pulses to prevent cycling of compressor to 0% and 100% steps under light load/high humidity conditions.

3.2 Evaluation

Results of the 500-hour test show that the rotary screw compressor can successfully operate in a representative bus air conditioning system. The 2000-hour test demonstrated that the screw compressor can operate reliably even under extreme environmental conditions.

The revenue service test showed that the prototype screw compressor/controller package was reliably able to keep a transit bus cool under all of the various environmental conditions encountered during typical, southern climate, summer service. Temperatures in the passenger compartment were maintained well within "White Book" (ADB) specifications throughout the test. While occasional minor corrective maintenance was required for the test bus air conditioning system, the compressor itself required no maintenance at all during the test period.

Limited comparison data shows that the test bus exhibited almost a 3% savings in fuel consumed versus a standard coach during the test period. Dunham-Bush, Inc. expects that, due to the rotary screw compressor control strategy, especially during the transition seasons, annual fuel consumption associated with bus air conditioning systems could be reduced by approximately 50%.

Based on the foregoing, it appears that rotary screw technology can provide a reliable alternative to the conventional, transit bus air conditioning system, refrigeration compressor.

4.0 Current Status

As of this writing, the prototype compressor installation has operated incident-free for two years. On-going evaluation efforts on prototype and preproduction compressors are being conducted at both transit agencies' and bus manufacturers' facilities. The latest (production) version of the Dunham-Bush screw compressor features an integral oil separator and a housing-supported, heavy-duty clutch assembly. Transit authorities in Philadelphia and Washington, D.C. have retrofitted a number of buses with these production units for the purpose of revenue service testing and evaluation during the 1984 cooling season. Other transit agencies, including those in Houston and Milwaukee, have also expressed an interest in purchasing rotary screw compressors for both retrofit and new bus applications. Further information can be obtained from Dunham-Bush, Inc. in West Hartford, Conn. Additional firms currently marketing rotary screw compressors for transit bus applications include the Trane Co. in Montgomery, Ala. and Suetrak Sales Corp., Boulder, Colo.

APPENDIX A

500-Hour Test

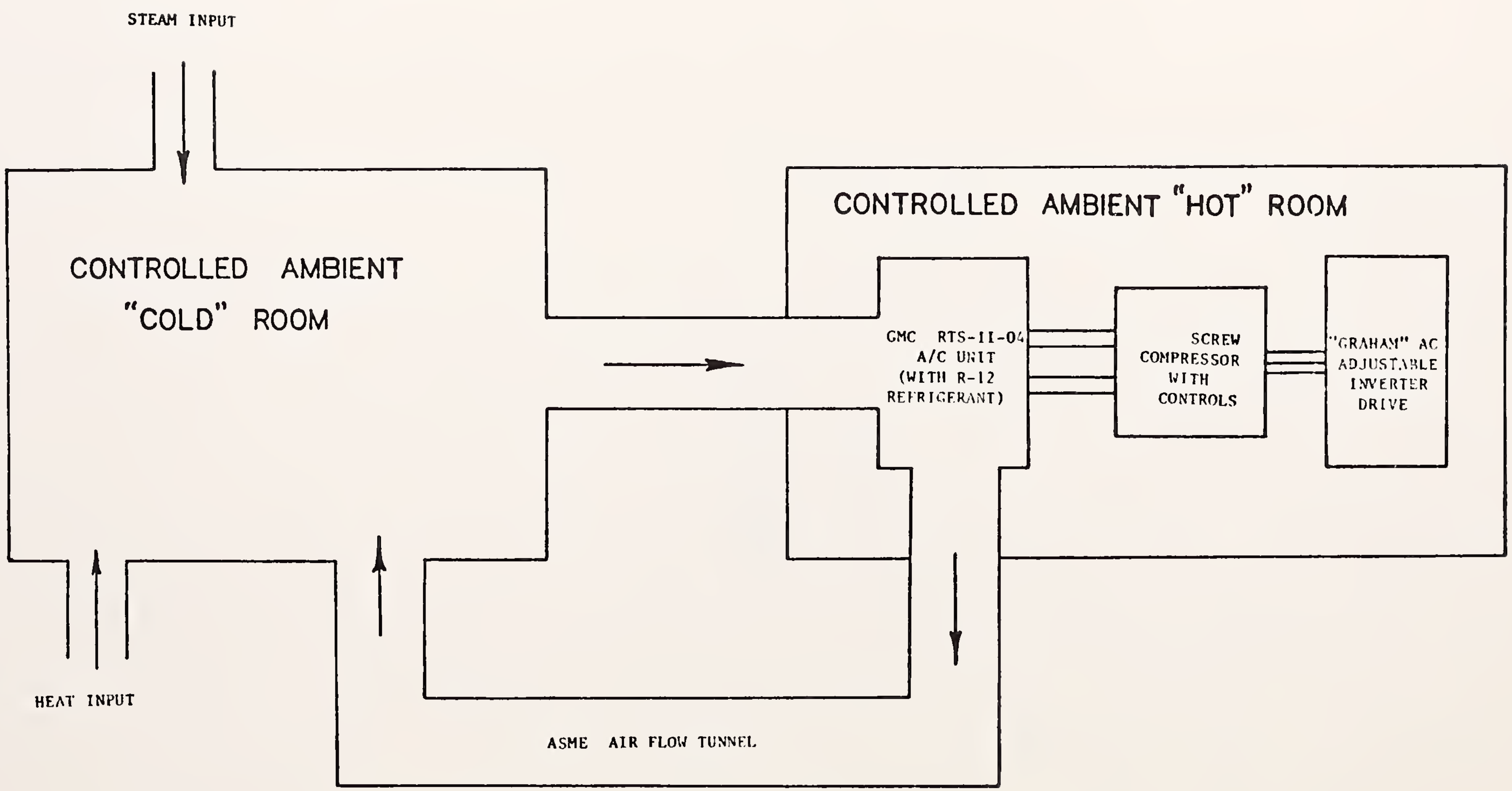


FIGURE A-1. 500-HOUR DEVELOPMENT TEST, LABORATORY CONFIGURATION

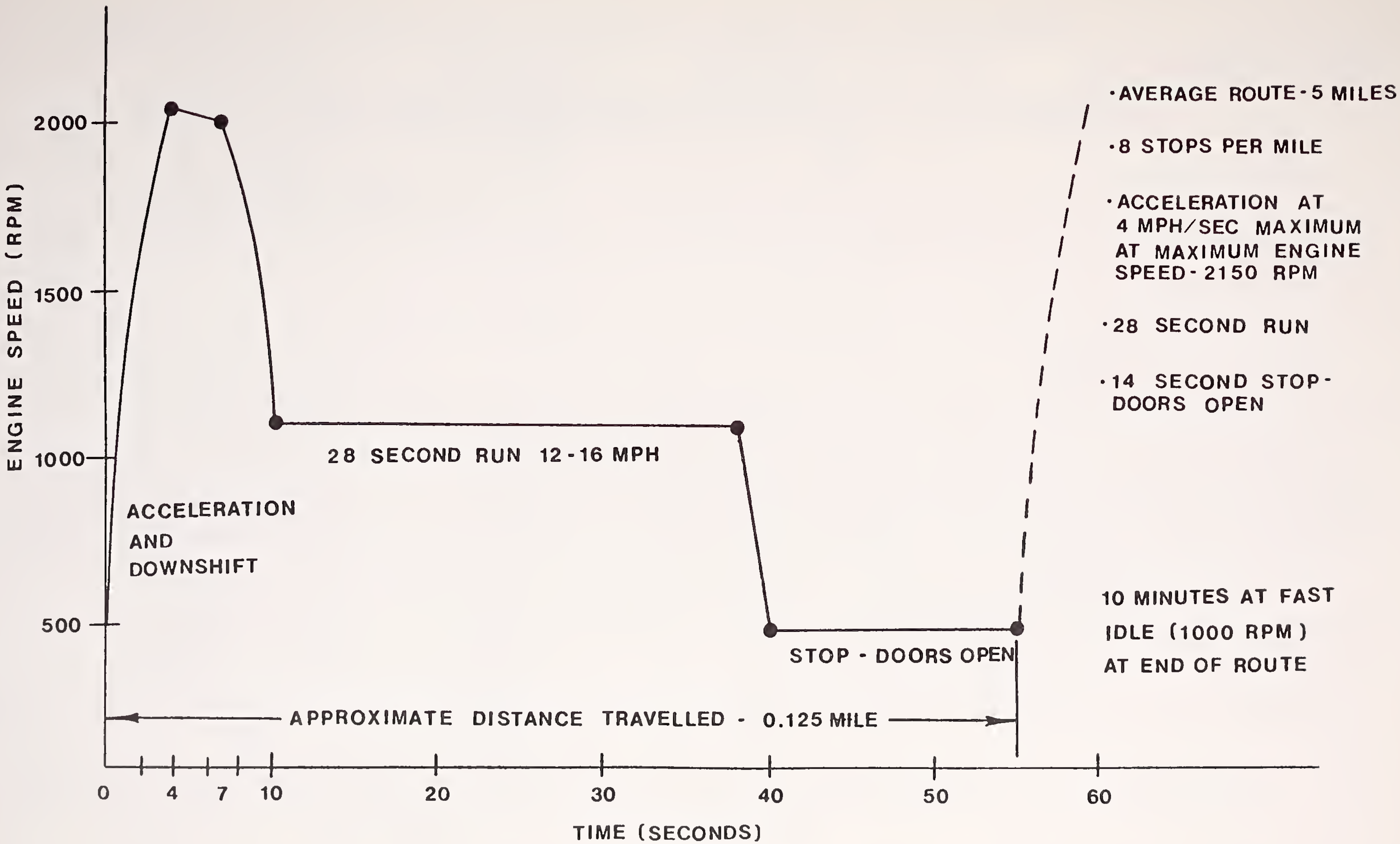


FIGURE A-2. 500-HOUR TEST, AVERAGE URBAN RUN PROFILE TYPICAL OF DOWNTOWN BUSINESS DISTRICT

	'COLD ROOM'		'HOT ROOM'		
TEST NO.	HEAT LOAD	DRY BULB/WET BULB(°F.)	AMBIENT TEMP.	DURATION	DESCRIPTION
1	(1) Maximum	80/67 80/67 86/70	100°F 115°F 115°F	40-50-HRS.	Maximum load condition
2	45,000 BTU/HR.	75/63	90°F	40-50 HRS.	Nominal load condition
3	15,000 BTU/HR.	70/68	65°F	40-50 HRS.	Humidity control evaluation
4	15,000 BTU/HR.	71/61	65°F	40-50 HRS.	On/off cycle evaluation
5	15,000 BTU/HR.	71/61	65°F	40-50 HRS.	Re-heat cycle evaluation
6	(2) 45,000 BTU/HR. 15,000 BTU/HR.	130 @ start/na.	115°F	25 HRS.	Evaluate pulldown ability from 130°F 'cold room' start

(1) Maximum heat load that will prevent <u>any</u> compressor unloading as determined by return air temperature.

(2) To be run at 1000 rpm engine speed (simulated) and designed to evaluate system capability under pulldown loads and temperatures. The 15,000 BTU/HR. heat load was also evaluated in the cycling mode.

*Initial portion of test period used for run-in/control system check-out and development.

FIGURE A-3. 500-HOUR DEVELOPMENT TEST, SUMMARY OF TEST LOAD CONDITIONS

TEST NO.	CONDITIONS EDB/EWB/CEA*	HEAT LOAD BTU/HR.	COMPRESSION ENERGY HP.-MIN. FOR 30 MIN.	CYCLE COP** HEAT REMOVED ÷ COMP. ENERGY
1 Max.Load	80/67/100	74,000	507	1.72
	80/67/115	57,000	545	1.23
	86/70/115	73,000	574	1.50
2 Nominal	75/63/90	48,500	312-536	1.83-1.07
3 High Humidity	70/68/65	20,300	226	1.05
4 on/off	71/61/65	17,900	155	1.36
5 reheat	71/61/65	17,900	191	1.10

*Evaporator dry bulb/Evaporator wet bulb/Condenser entering air (°F.)

** Cycle COP = $\left(\frac{BTU}{HR.} \times .01179\right) \div$ HP.-Min.

FIGURE A-4. AVERAGE URBAN RUN PROFILE EFFICIENCY RESULTS

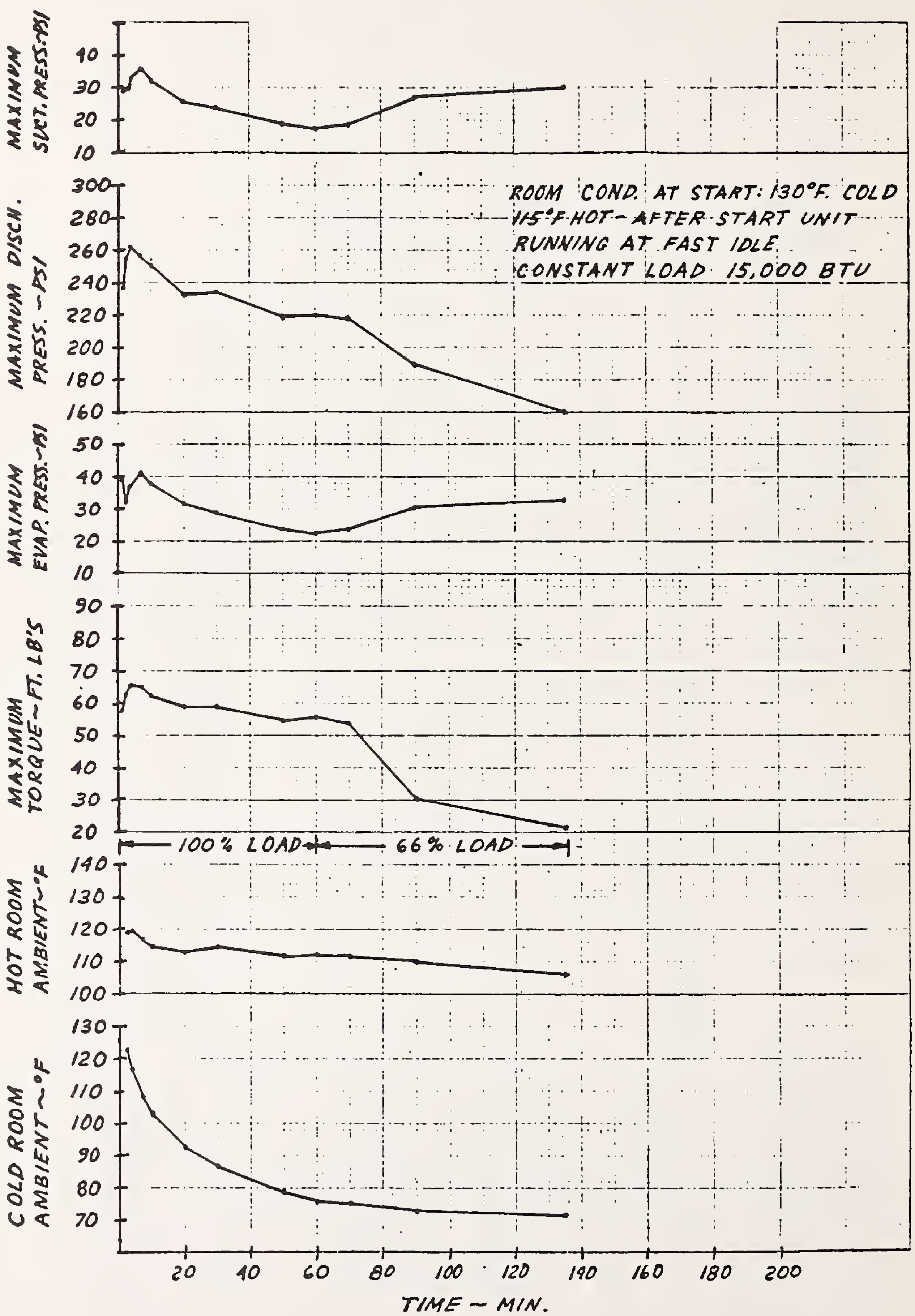


FIGURE A-5. 500-HOUR DEVELOPMENT TEST, PULL DOWN TEST-TEST NO. 6

APPENDIX B

2000-Hour Test

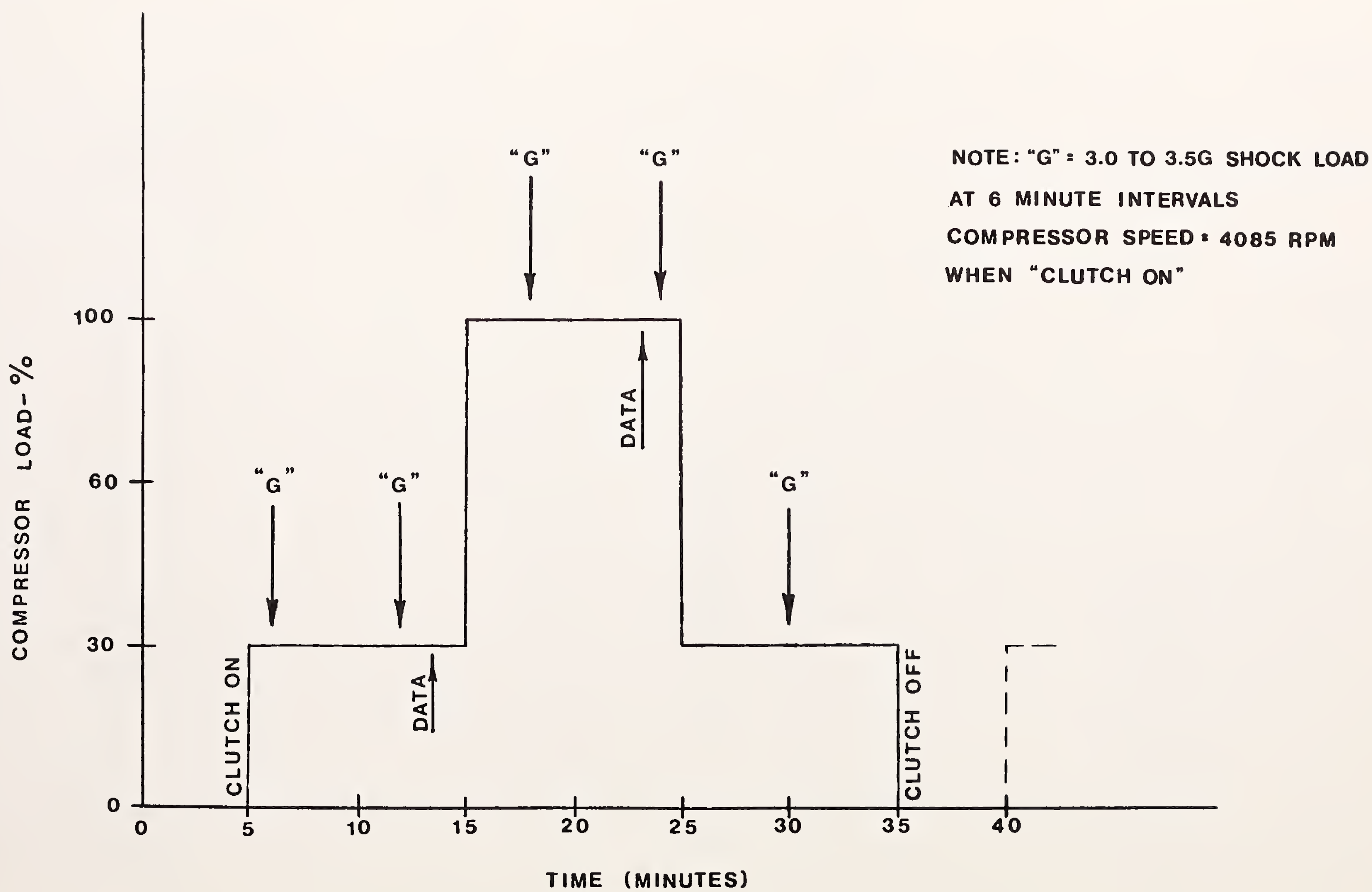


FIGURE B-1. LOAD CYCLE 2000-HOUR DURABILITY TEST

TABLE B-1. 2000-HOUR TEST PERFORMANCE CALCULATIONS

```
RDBUS12H    11:39   04/26/82  MONDAY     I02-20

IS THE COMPRESSOR AT: (1) FULL LOAD, (2) 2/3 LOAD OR (3) 1/3
ENTER 1  2  OR 3
? 1


? 30.05


? 187 220

GAS CONDITIONS AT THE DISCHARGE ORIFICE:
SPECIFIC VOLUME = 0.196 FT3/LBM
ENTHALPY        = 98.96 BTU/LBM

? 79 62.5

GAS CONDITIONS AT THE SUCTION ORIFICE:
SPECIFIC VOLUME = 0.548 FT3/LBM
ENTHALPY        = 86.29 BTU/LBM


? 303 .625,491  700

DISCHARGE FLOW RATE = 0.937 LBM/SEC
SUCTION FLOW RATE   = 0.855 LBM/SEC


? 71.41 5.31.2 37 64 3715.190 235

SUCTION GAS CONDITIONS AT THE COMPRESSOR:
SUCTION VOLUMETRIC FLOW = 39.24 CFM
SPECIFIC VOLUME = 0.765 FT3/LBM
ENTHALPY        = 86.08 BTU/LBM


*************************************************************
*                                                           *
*  CONDITIONS:                                              *
*                                                           *
*  150.2 F CONDENSING,  45 F SAT SUCTION, 3715 RPM. FULL LOAD *
*                                                           *
*************************************************************
                                        RAN WITH
OUTPUT/INPUT FLOW CORRELATION = 15.5 %  LIQ. INJ.
SUCTION COOLING CAPACITY = 130379. BTU/HR
INPUT POWER TO MOTOR      = 31 2 KW
SHAFT POWER               = 37.6 BHP
VOLUMETRIC EFFICIENCY     = 52.3 %
PRESSURE RATIO            =  4.44
PERFORMANCE FACTOR (CAP IN WATTS/BHP IN WATTS) = 1.36
ENERGY EFFICIENCY RATIO (EER)   = 4.65 BTU/WATT-HR
ENTHALPY OF SAT LIQ @ COND TEMP = 43.721 BTU/LBM
```

TABLE B-2. 2000-HOUR TEST PERFORMANCE CALCULATIONS

```
RUN RDBUS12M

RDBUS12M     10:45    06/07/82  MONDAY      I02-20

IS THE COMPRESSOR AT: (1) FULL LOAD, (2) 2/3 LOAD OR (3) 1/3 LOAD??
ENTER 1, 2, OR 3
? 3


? 29.8144


? 195,229

GAS CONDITIONS AT THE DISCHARGE ORIFICE:
SPECIFIC VOLUME = 0.192 FT3/LBM
ENTHALPY        =100.19 BTU/LBM

? 63,45

GAS CONDITIONS AT THE SUCTION ORIFICE:
SPECIFIC VOLUME = 0.702 FT3/LBM
ENTHALPY        = 84.62 BTU/LBM


? 50,.625,72,.700

DISCHARGE FLOW RATE = 0.412 LBM/SEC
SUCTION FLOW RATE   = 0.311 LBM/SEC


? 70,42,18.4,22.19,3790,190,232

SUCTION GAS CONDITIONS AT THE COMPRESSOR:
SUCTION VOLUMETRIC FLOW = 14.11 CFM
SPECIFIC VOLUME = 0.757 FT3/LBM
ENTHALPY        = 85.90 BTU/LBM


****************************************************************
*                                                              *
*  CONDITIONS:                                                 *
*                                                              *
*  149.1 F CONDENSING, 45 F SAT SUCTION, 3790 RPM,  1/3 LOAD   *
*                                                              *
****************************************************************

OUTPUT/INPUT FLOW CORRELATION = 32.5 %
SUCTION COOLING CAPACITY =  47466. BTU/HR
INPUT POWER TO MOTOR      = 18.4 KW
SHAFT POWER               = 22.2 BHP
VOLUMETRIC EFFICIENCY     = 13.4 % (COMPRESSOR UNLOADED)
PRESSURE RATIO            =  4.35
PERFORMANCE FACTOR (CAP IN WATTS/BHP IN WATTS) = 0.84
ENERGY EFFICIENCY RATIO (EER)    = 2.87 BTU/WATT-HR
ENTHALPY OF SAT LIQ @ COND TEMP = 43.441 BTU/LBM
```

APPENDIX C

Revenue Service Test

TABLE C-1. BUS A/C OPERATIONAL FIELD TEST TYPICAL WEEK'S SERVICE

DAY	TEMP. HI/LO	DATE	BUS IN SERVICE	TOTAL RUNNING TIME	DATA COLLECTED
THURSDAY	96°/73°	7-22-82	START: 5:30pm END: 8:30pm	3.0 hours	3 hours
FRIDAY	97°/76°	7-23-82	START: 5:49am END: 20:43pm	14.90 hours	14.9 hours
SATURDAY	98°/75°	7-24-82	START: 6:02am END: 20:47pm	14.75 hours	14.75 hours
SUNDAY	99°/75°	7-25-82	START: 5:42am END: 20:40pm	14.97 hours	14.97 hours
MONDAY	99°/74°	7-26-82	START: 5:44am END: 20:43pm	14.98 hours	14.98
TUESDAY	99°/75°	7-27-82	START: 5:43am END: 20:39pm	14.93 hours	14.93
WEDNESDAY	100°/73°	7-28-82	START: 7:50am END: 20:37pm	12.78 hours bus out of service- dent/body work	12.78

C-2

TABLE C-2. FUEL ECONOMY COMPARISON OF COMPRESSOR REPLACEMENT BUS (ROTARY SCREW) AND CONTROL BUS

Group	Bus No.	MARCH		APRIL		MAY		TOTAL MILES	AVERAGE MPG
		Miles	MPG	Miles	MPG	Miles	MPG		
A	#351	4695	3.72	5355	3.62	5091	3.50	15,141	3.61
A	#352	4431	3.52	4334	3.89	4684	3.63	13,449	3.68
		JUNE		JULY		AUGUST			
		Miles	MPG	Miles	MPG	Miles	MPG		
B	#351	1439	2.95	2292	2.94	857	2.83	4588	2.91
B	#352	1460	3.18	2272	2.97	1200	3.02	4932	3.05

Bus #352: Test Bus

Bus #351: Control Bus

Group A: Pre-test (Base) Period

Group B: Test Period

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