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ROTARY VANE POSITIVE DISPLACEMENT AIR CYCLE MACHINE FOR ARMY EC--ETC(U)

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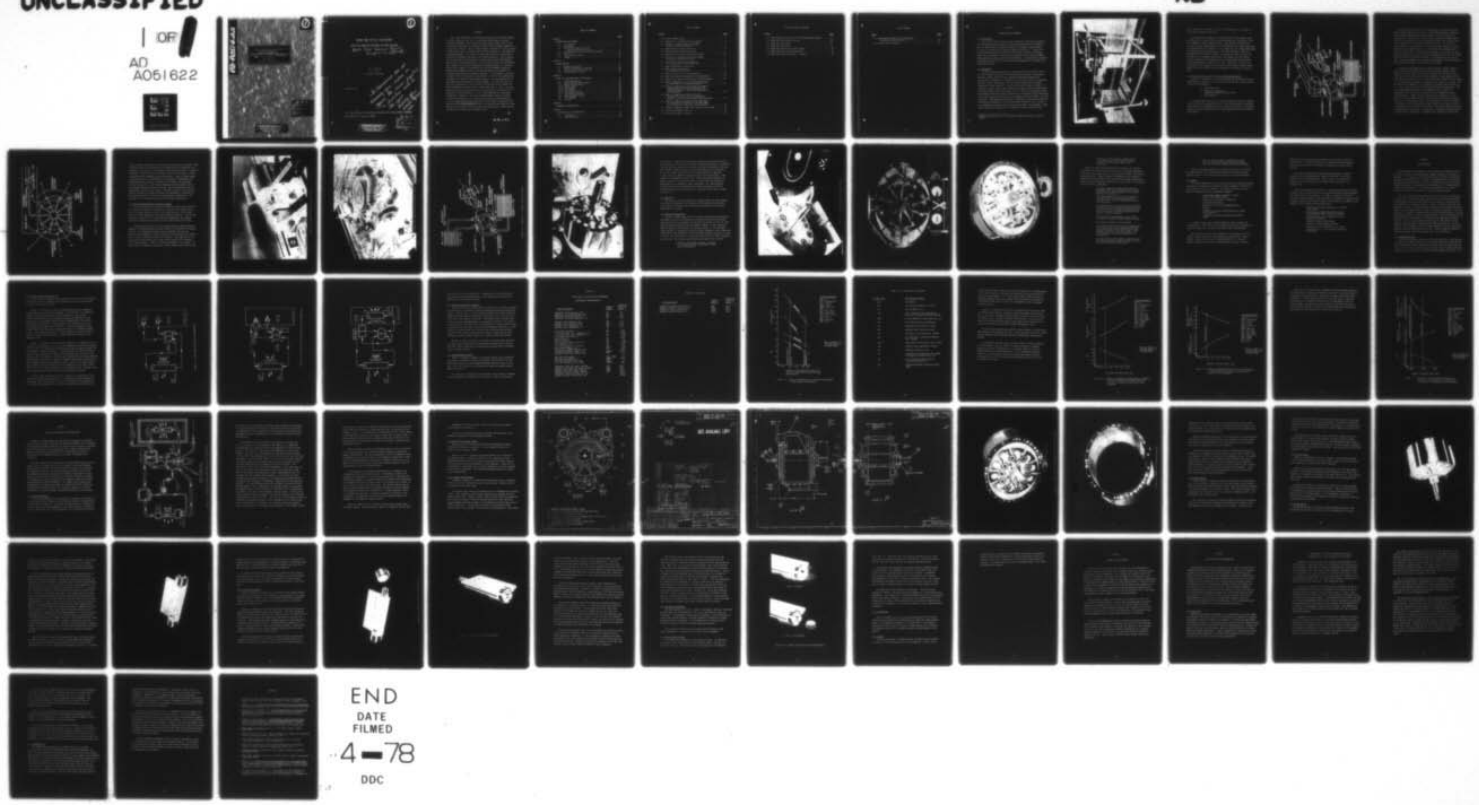

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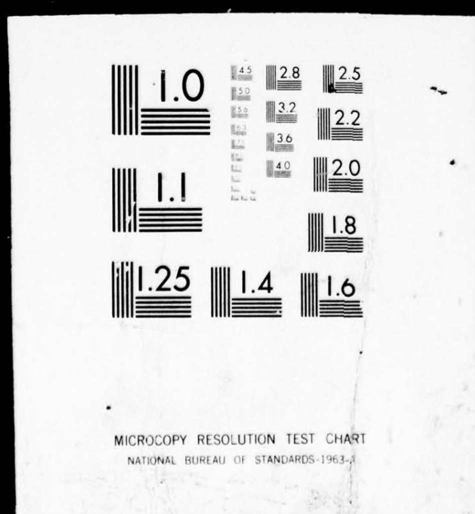

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ROTARY VANE POSITIVE DISPLACEMENT
~~AIR CYCLE MACHINE FOR~~
~~ARMY ECS APPLICATIONS~~
ROVAC, Tech. Report, Compressor-Expander

Final W. R. Bomstad
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AIR CYCLE MACHINE FOR ARMY ECS APPLICATIONS

ROVAC Final Technical Report
Compressor-Expander

W. R. Bomstad

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The ROVAC Corporation

September 1977

The Government does not necessarily agree, endorse or approve the claims, projections or predictions made in this report. Contracting Officer's Technical Representative

U.S. ARMY MOBILITY EQUIPMENT RESEARCH AND DEVELOPMENT COMMAND

FORT BELVOIR, VIRGINIA 22060

ABSTRACT

This report documents an exploratory research and development program to design, fabricate, and test an engineering demonstration prototype positive-displacement rotary-vaned air cycle machine (Model 606) for the United States Army Mobility Equipment Research and Development Command at Ft. Belvoir, Virginia. Existing ROVAC computer software was used to perform system analysis and to establish machine design parameters. The Model 606 compressor-expander (Circulator) was fabricated using conventional means and special techniques developed by The ROVAC Corporation. The Circulator and the associated air conditioning system evolved through modifications from an open cycle system design to a closed cycle system with regeneration. These changes in the system configuration and modifications in Circulator design were performed to incorporate into the 606 breadboard system all promising improvements which are continuously derived from ROVAC's internally funded advanced development programs. The performance testing of the Model 606 air conditioning system and the results of that testing are included in the "Test and Demonstration Report" which was supplied to the Army immediately following the testing program. Under the Army environmental test conditions of 125°F outside, 90°F db and 67°F wb inside, the ROVAC demonstration system showed a steady state COP of 0.544. This performance compares very favorably with an overall COP on the order of 0.35 produced by a well-developed air cycle system employing well-developed turbomachine technology. The "Test and Demonstration Report" contains calculations, based on intra-Circulator heat transfer improvements and the use of well-developed heat exchangers, that provide a sound basis for expecting a (dry air) COP on the order of 1.5 ~~of greater than 1.2~~ for a further-developed ROVAC air cycle system at the specified operating conditions using dry air. Moist air can substantially increase this performance level.

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INTRODUCTION AND SUMMARY

1.1 Introduction

This is the final technical report presenting an exploratory research and development program conducted by The ROVAC Corporation of Florida for the U.S. Army Mobility Equipment Research & Development Command (MERDC), Ft. Belvoir, Virginia under contract DAAG53-76-C-0052 (1)[1]. During this program a ROVAC positive displacement rotary vane Circulator called the Model 606 was designed, fabricated and integrated into an existing MERDC/ ROVAC system for engineering test and evaluation. The program also included performance testing of the system which was accomplished at the ROVAC Test Laboratory Facility.

1.2 Background

In 1967 at Purdue University, an alternate means for producing direct air cycle refrigeration and air conditioning was conceived. This system embodied a simple unitary rotary combination compressor-expander air cycle machine (Circulator) in conjunction with an air-to-air heat exchanger. Analysis, design, fabrication, and test of the system proved the basic feasibility of the concept (2). The first test unit, shown in Figure 1.1 produced 1.6 KW of cooling while operating at ambient ground conditions with an overall coefficient of performance of 0.75. The first generation system, while demonstrating basic system feasibility, also defined certain mechanical design requirements related primarily to mechanical friction reduction in the Circulator. Subsequent design, analysis, fabrication, and testing confirmed the mechanical operability of an anti-friction Circulator unit. As well, new thermodynamic processes were discovered, analyzed, and qualified. These results indicate the possibility of producing a rotary vane direct air

[1] Numbers in parentheses refer to similarly numbered references listed on page 62.

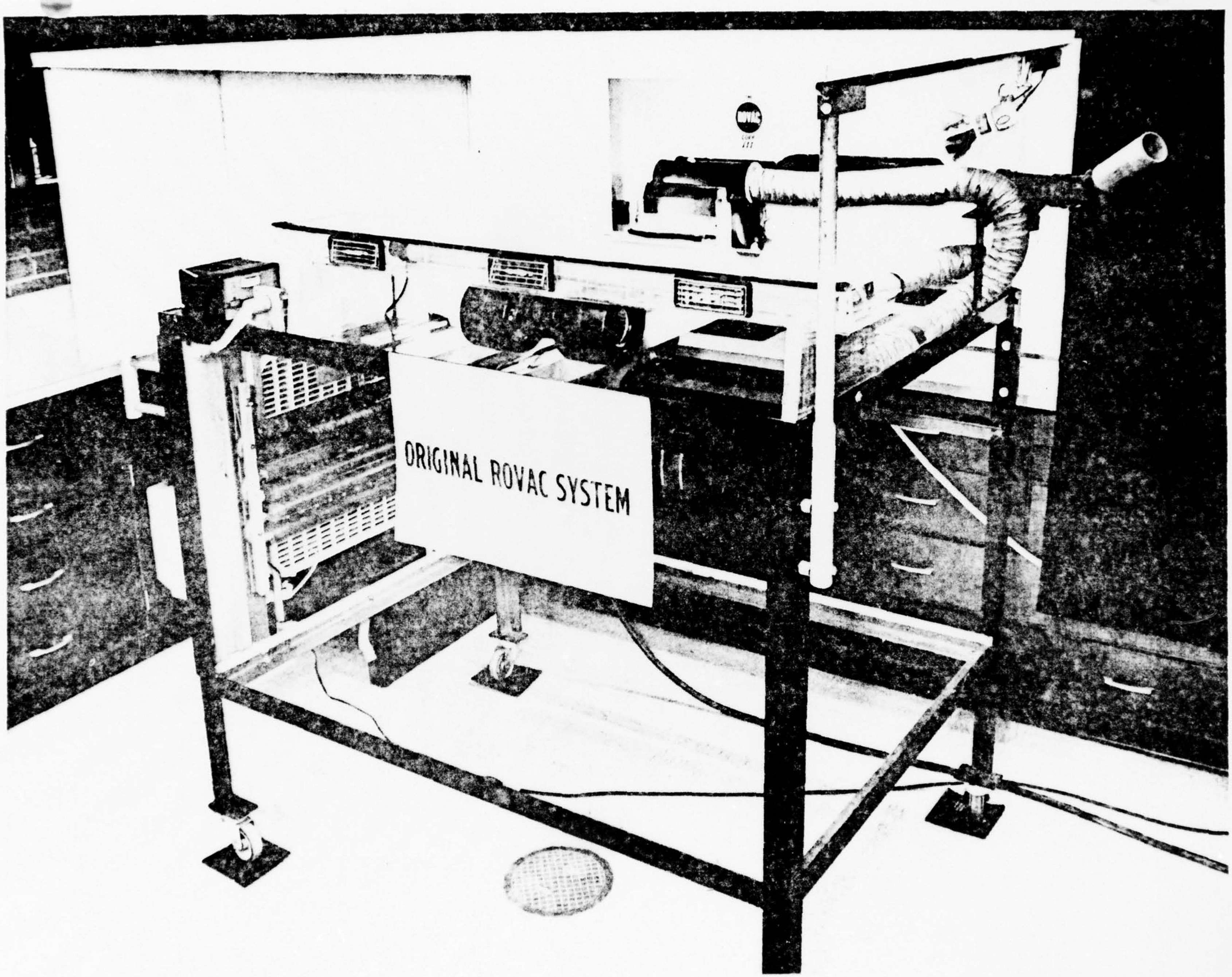


Figure 1.1 -- Original ROVAC System

cycle system with an overall coefficient of performance on the order of 3.0 at general ambient conditions.

The possibility of employing the ROVAC positive displacement rotary vane Circulator in Army thermal management systems was recognized in 1973 by the MERDC air conditioning engineers. This ultimately resulted in U.S. Army contract number DAAK02-74-C-0288 granted to The ROVAC Corporation for the development of a ROVAC air cycle window type air conditioning system for engineering test and evaluation. The ROVAC Circulator, Model 209, developed for this system is similar to the machine shown in Figure 1.4. Although, due to technical difficulties, this system only had limited success in meeting the performance goals it demonstrated that the ROVAC system had potential in Army fields of use. Therefore, to further the development of this promising technology the Army granted contract number DAAG53-76-C-0052 to ROVAC for the development of an advanced Circulator design.

1.3 Rotary Vane Direct Air Cycle Cooling System Description

Figure 1.2 illustrates in a generalized fashion the substance of the ROVAC Open-Cycle System. In the open-cycle form the ROVAC System consists of three basic components:

a) Input-Output Duct

b) Rotary Vane Combination Compressor-Expander-Circulator Unit

c) Heat Exchanger

The Input-Output Duct serves to provide a path for both the incoming flow of warm air and outgoing cold air. This ducting system is equipped with various processing media such as filters, sound control, or moisture separation means, depending upon the actual application of the particular ROVAC System.

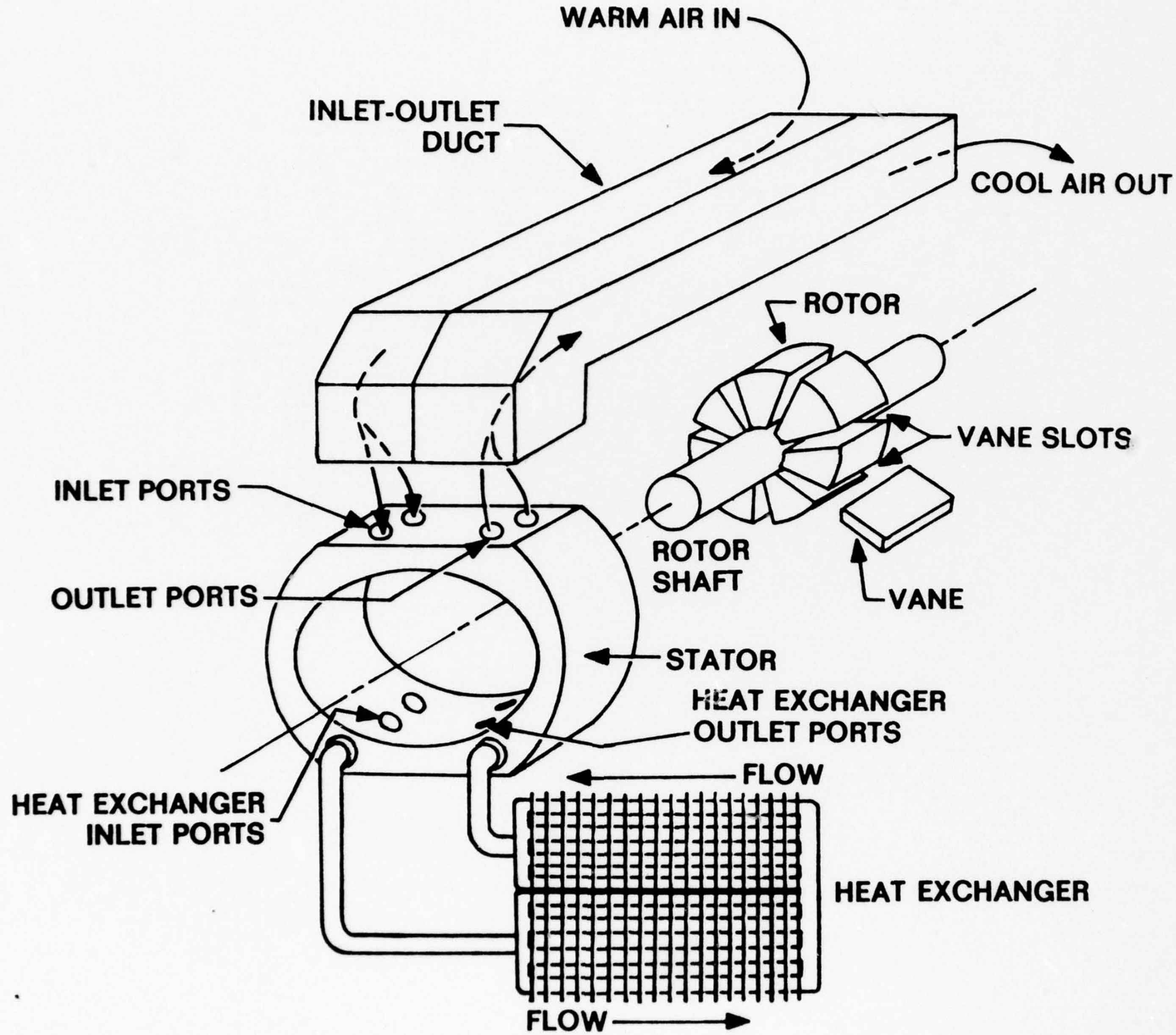


Figure 1.2 -- Components to a ROVAC Open-Cycle System

The Circulator Unit, which can be considered to be the heart of the system, consists mainly of two items: the stationary portion, termed the stator, and the rotating rotor-vane assembly. The stator housing is equipped with a substantially elliptical axial cavity and suitable ports which serve to control the inlet and outlet air flow. The stator housing is also equipped with two endcaps which hold the rotor in its axial position and aid in containing the air that circulates through the machine. The rotor-vane assembly consists of a cylindrical rotor equipped with a series of sliding vanes. These vanes are arranged so that they can slide inside the rotor slots as the rotor-vane assembly rotates within the stator, and therby, maintain continuous contact or near-contact with the stator wall. The Heat Exchanger, also termed the Intercooler, serves to dissipate the heat generated during the operation of the system.

In order to understand how the system functions, consider Figure 1.3 which illustrates the ROVAC System in cross-sectional schematic form. Imagine air (any gas will work, however) entering the system through the input duct. As the rotor turns counter-clockwise in this example, air is drawn through the inlet leg of the duct and into the expanding inlet volume V_i. As rotor motion continues, the maximum inlet volume V_1 is filled. The air which is now trapped is then compressed by further rotor rotation to V_2. At V_2 the air has reached an elevated pressure and temperature. The air is then pumped through the heat exchanger, and it cools to a temperature near that of the surroundings, thus, transferring heat to the surroundings. (When the ROVAC System is being employed as a heat pump, the heat exchanger warmth is transferred to the area being heated.) The cooler air then emerges from the heat exchanger (still at an elevated pressure) and enters the expanding volume segment, V_4, Finally, this air is expanded to V_5 by continued rotor motion. During this expansion process, a large fraction of the work of compression is recovered, thereby greatly reducing the air temperature. Continued rotor rotation then forces the

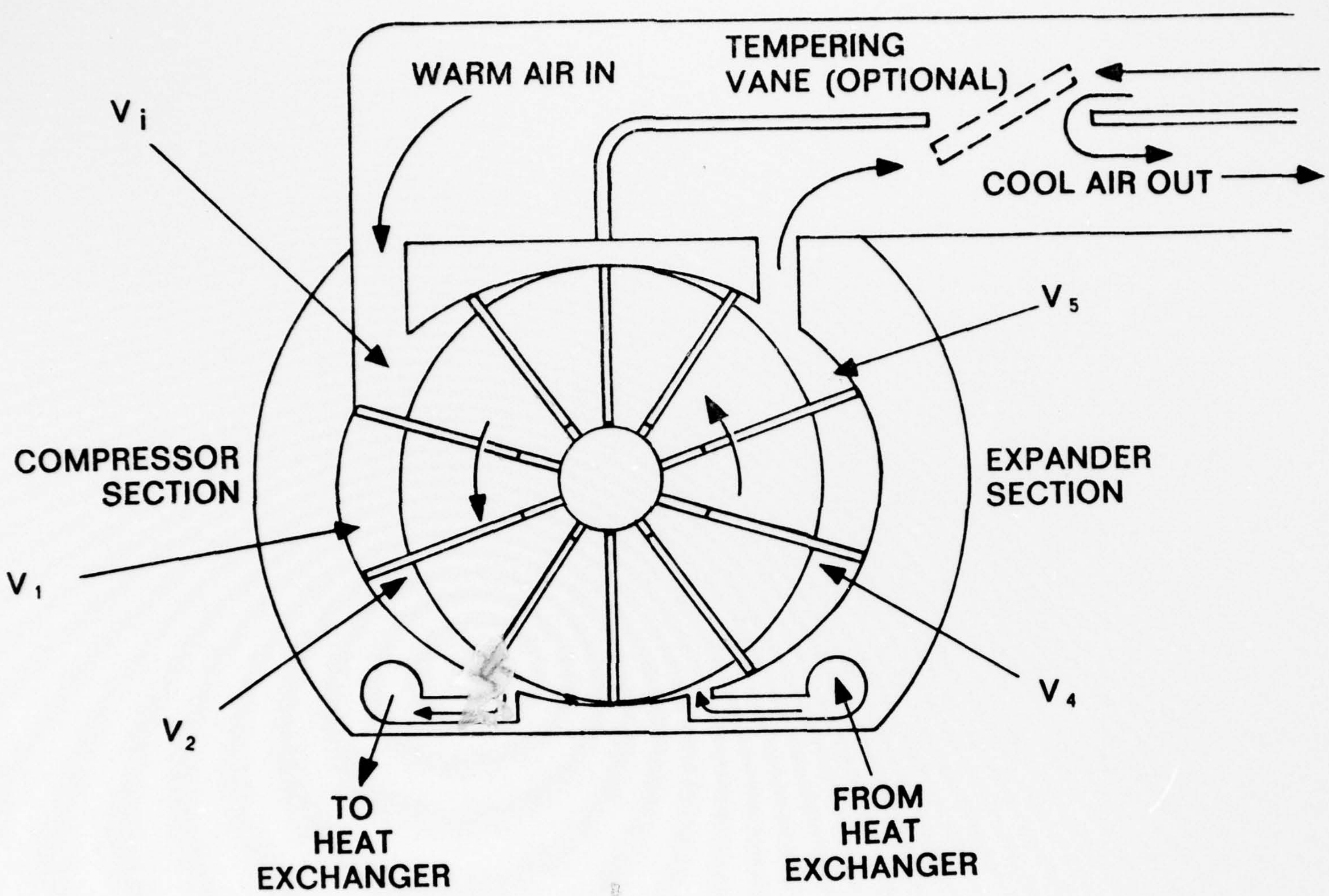


Figure 1.3 -- Schematic of ROVAC Compressor-Expander
Showing How System Functions

cold air through the output ports and down the output leg of the duct. When the ROVAC System is being employed as an open-cycle air conditioner or refrigerator, this cold air is circulated to the area being cooled. Figure 1.4 illustrates a ROVAC open-cycle Circulator under test and Figure 1.5 shows a ROVAC open-cycle automotive system installation. When the ROVAC System is being employed as a closed-cycle air conditioner or refrigerator the input-output ducting system is replaced by a secondary heat exchanger as shown in Figure 1.6. The cold air emerging from the expander outlet is now pumped through this heat exchanger, thus, absorbing heat from its surroundings (the area being cooled). The same Circulator unit and high pressure intercooler (called the Primary Heat Exchanger), as employed in the open-cycle system, is used in the closed-cycle configuration. The advantages to the closed-cycle system are discussed subsequently.

1.4 Anti-Friction Mechanical Circulator Design

It is well known in the design of conventional rotary vane machinery that the major contributor to mechanical friction is the rubbing of the tip of the vanes against the wall of the stator housing. This is true even for machines employing full liquid lubrication systems. The vane tip friction, in addition to its large individual magnitude, also induces significant rubbing forces between the vane sides and the rotor slots.

It was mentioned earlier that the second generation ROVAC Circulator embodied an improved mechanical design related to the reduction of machine friction. This improved mechanical design eliminates the vane tip friction loss altogether by simply preventing vane tip-stator wall contact. This is done by "caming" the motion of the vanes. The kinematics of the vanes are dictated directly by the cam profile existing in the endcaps of the stator housing. Figure 1.7 shows a photograph of this system. As can be seen, the vanes are fitted with rollers pinned to both ends of each

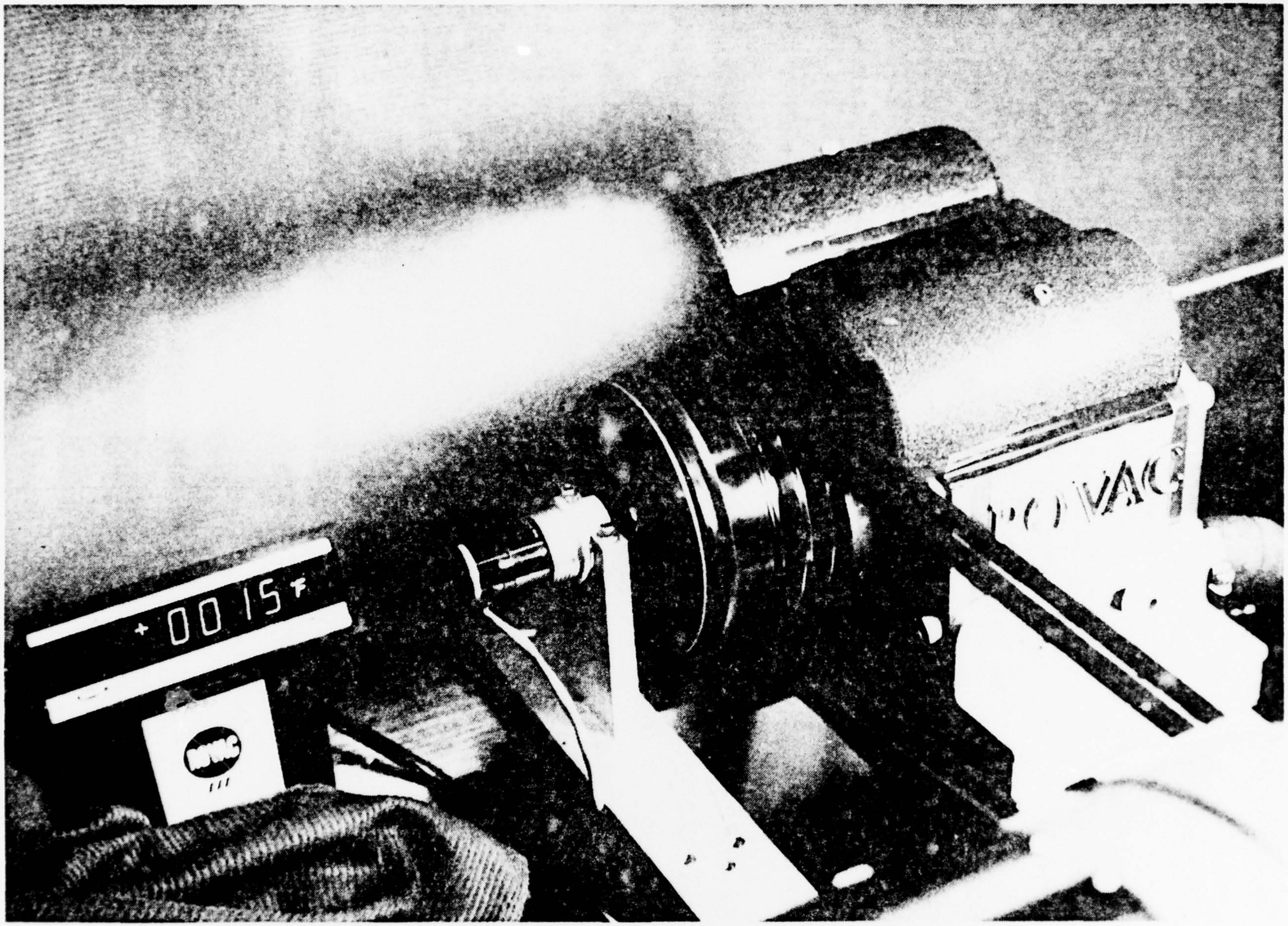


Figure 1.4 -- ROVAC Open-Cycle System Under Test

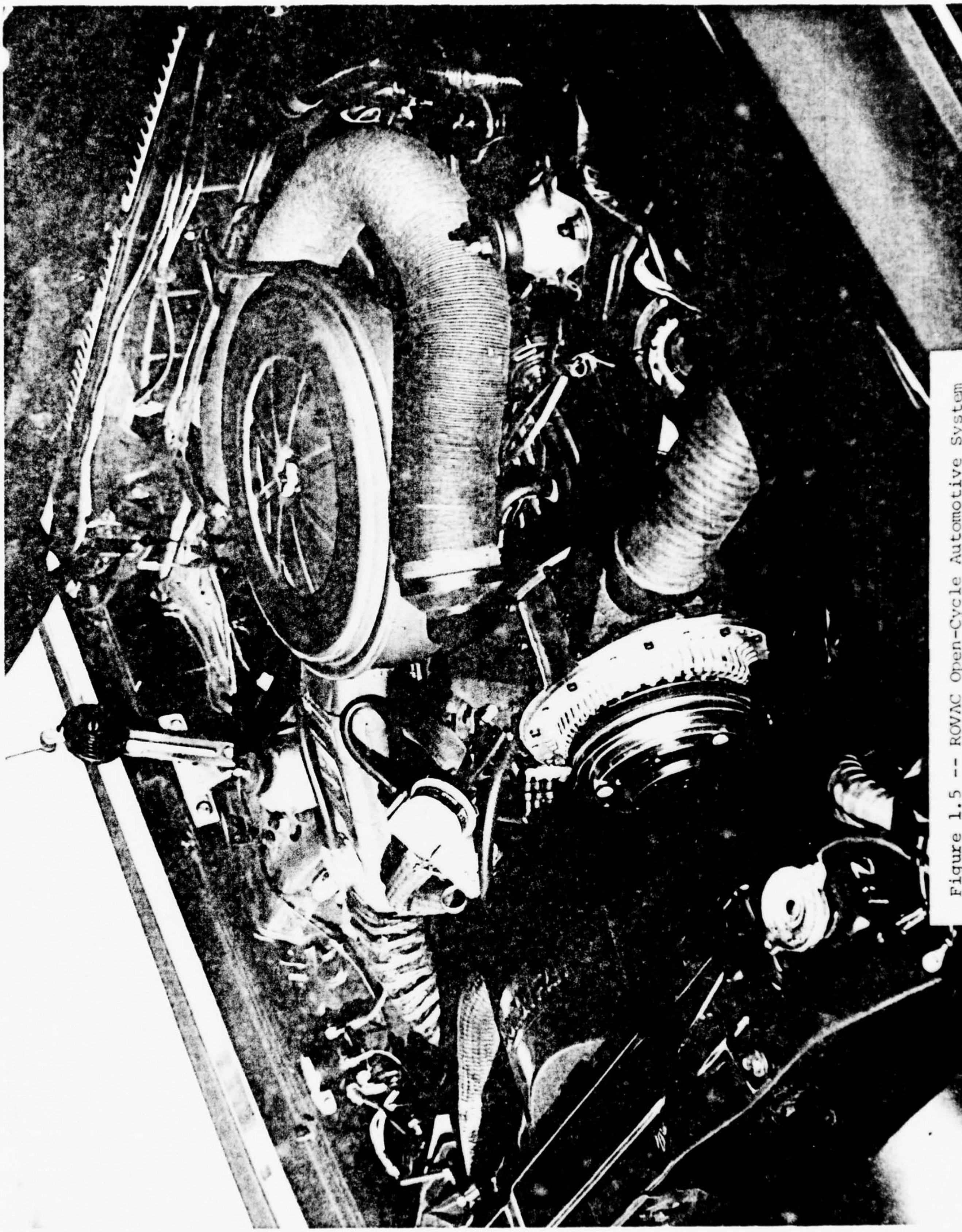


Figure 1.5 -- ROVAC Open-Cycle Automotive System

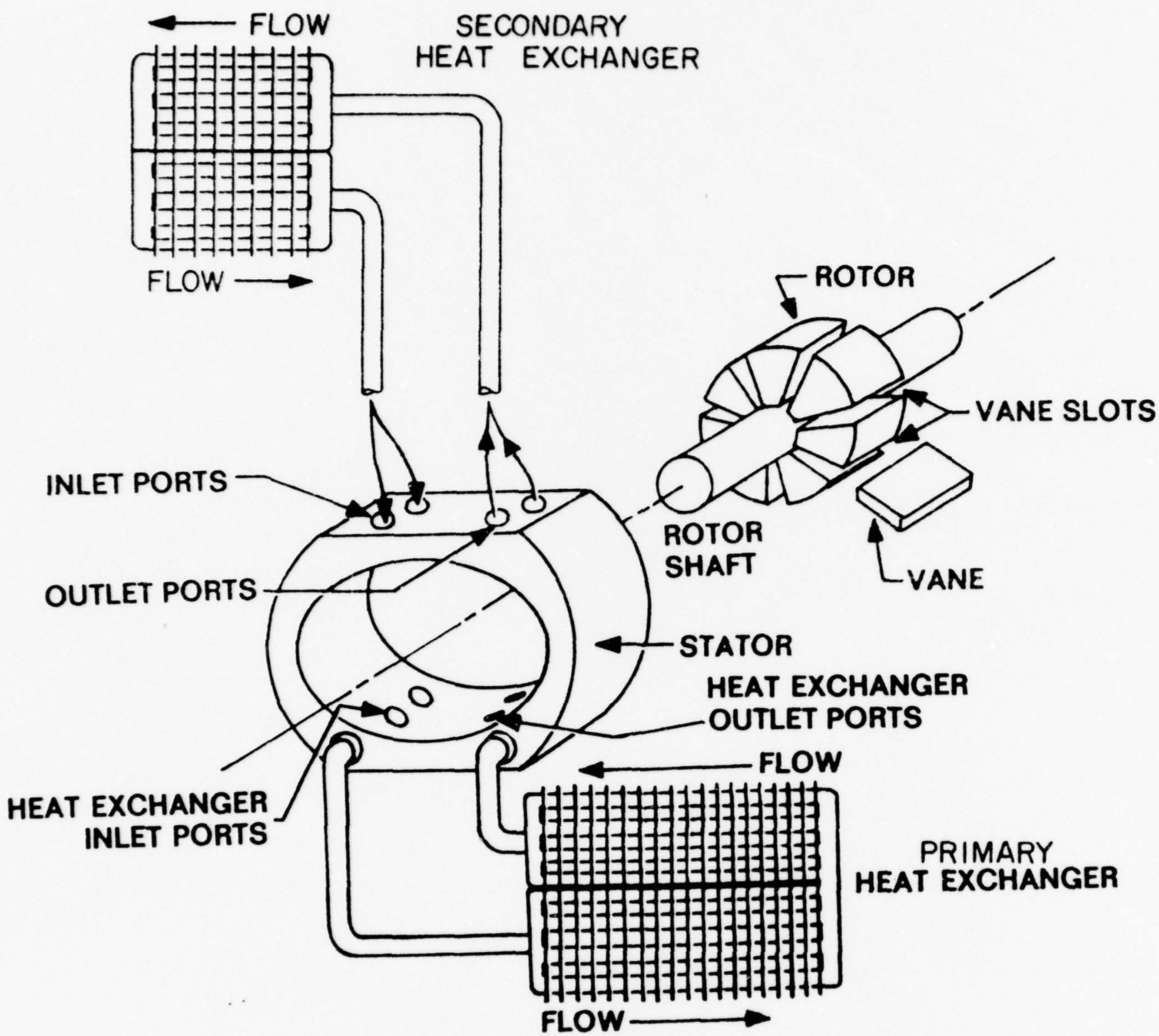


Figure 1.6 -- Components to a ROVAC Closed-Cycle System

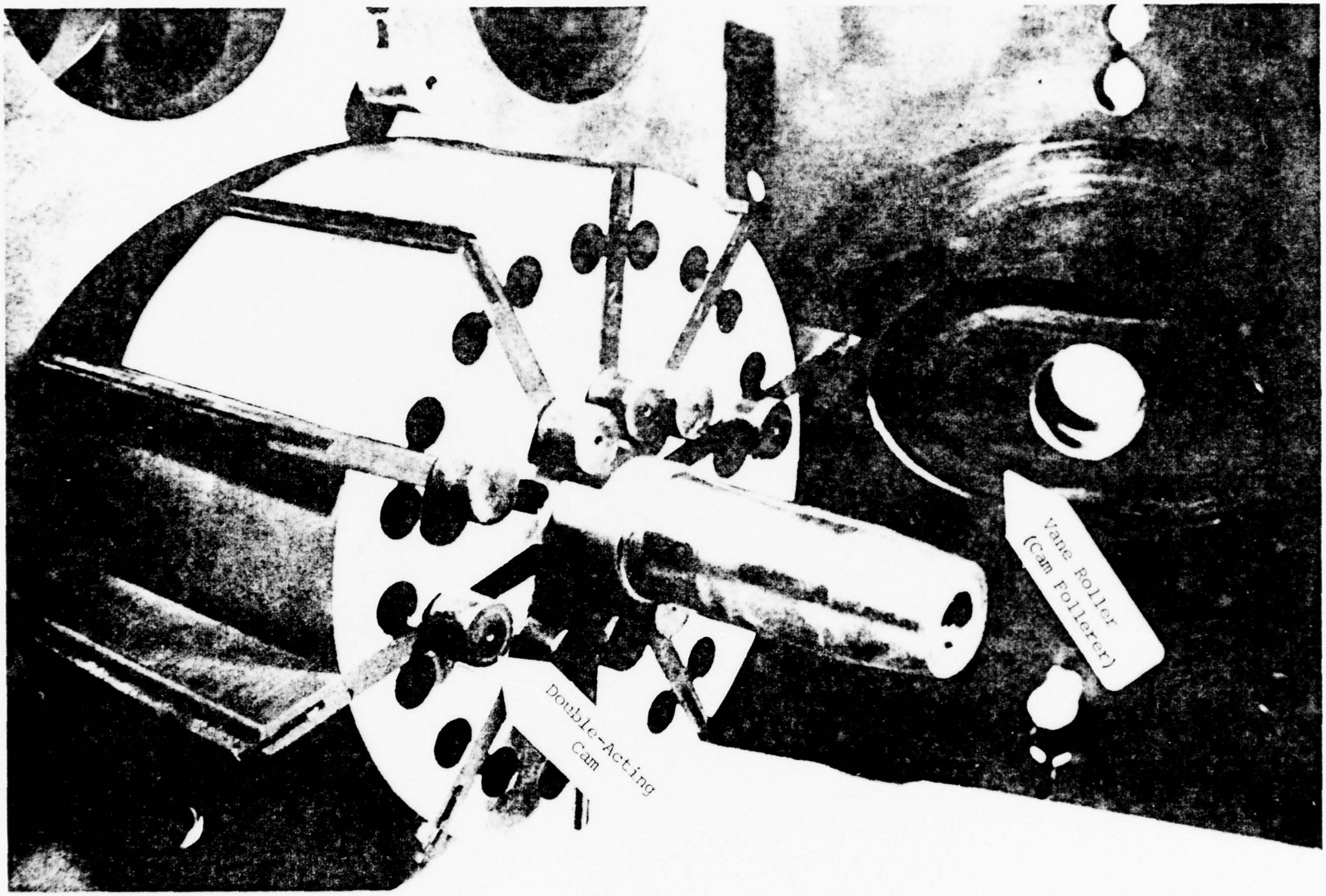


Figure 1.7 -- Anti-Friction Mechanical Circulator
Design Employing a Double-Acting Cam

of the vanes. These rollers follow the contour of the cam profile which is synchronized with the ellipse profile and thereby maintains a controlled vane tip clearance from the stator wall of several thousandths of an inch. In the machine shown, the cam is of the double-acting type (push-pull). In the later ROVAC Circulators only a single-sided cam is employed in order to permit uniform rotation direction of the cam rollers. This concept is illustrated in Figure 1.8. In the early model machines, as shown in Figure 1.9 the roller contact with the outer cam profile was maintained with an expandable inner spring band. The later machines successfully rely on centrifugal forces alone to keep the rollers in constant contact with the cam profile. A machine of this type is shown in Figure 1.10.

1.5 Objective

The objective of this program was to advance ROVAC air cycle machine technology for Army mobile applications through design and fabrication of an advanced Circulator for installation and testing in the existing MERDC/ROVAC 209 system.

1.6 Contract Modifications

Through the ROVAC Corporation's internally-funded advanced Circulator development efforts this program benefited from an advancing technology which paralleled the development of the ROVAC Circulator for the U.S. Army Mobility Equipment Research and Development Command. As this advancing ROVAC technology solved problems and revealed others, it was necessary, where possible, to incorporate certain contract Statement of Work (SOW) modifications to take advantage of the latest ROVAC design innovations in order to produce the most efficient and reliable state-of-the-art ROVAC Circulator for the Army. The following modifications were made in the contract:

- Rework of system and Circulator to operate in a variable pressure, lubricated, closed cycle configuration.

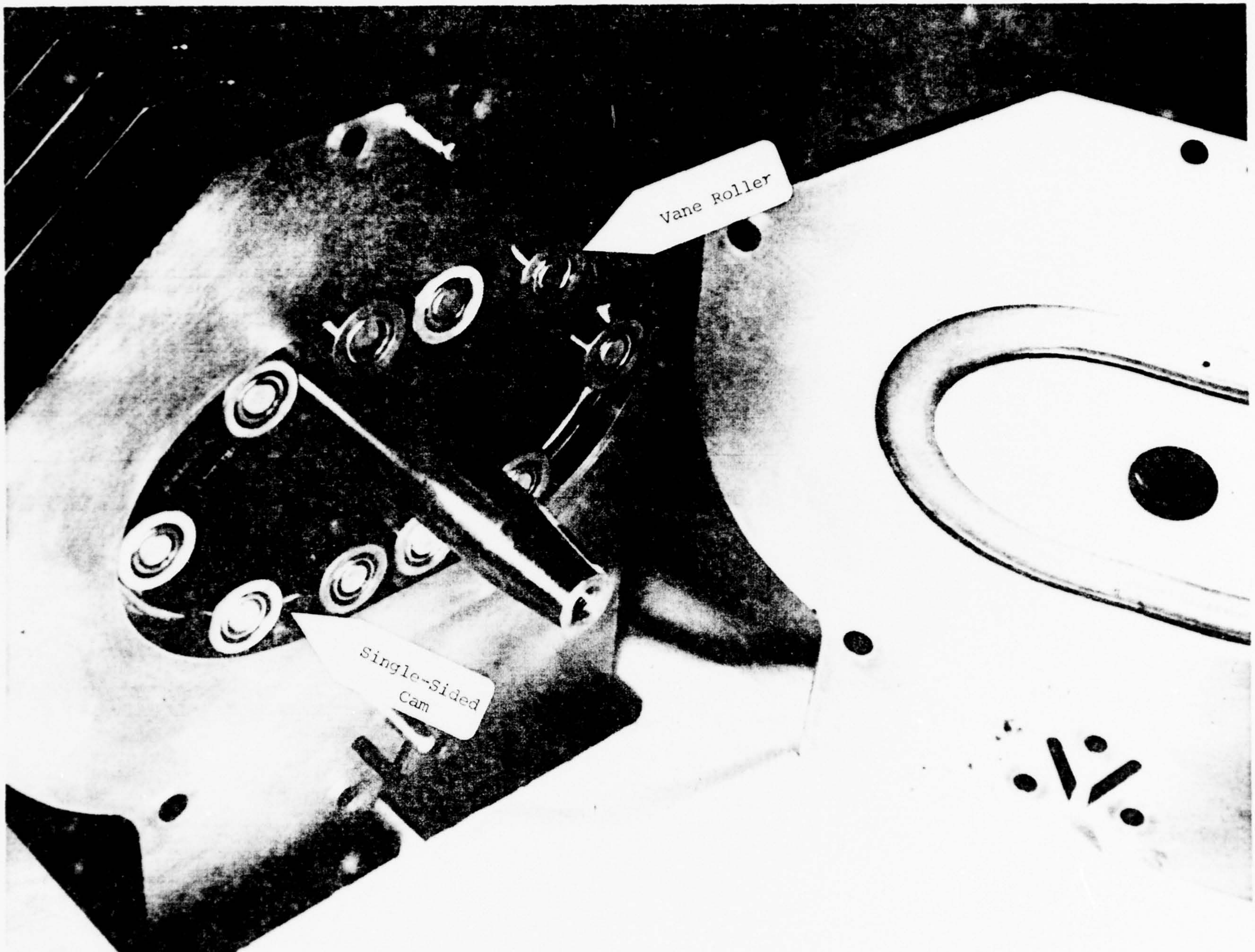


Figure 1.8 -- Anti-Friction Mechanical Circulator Design Employing a Single-Sided Cam

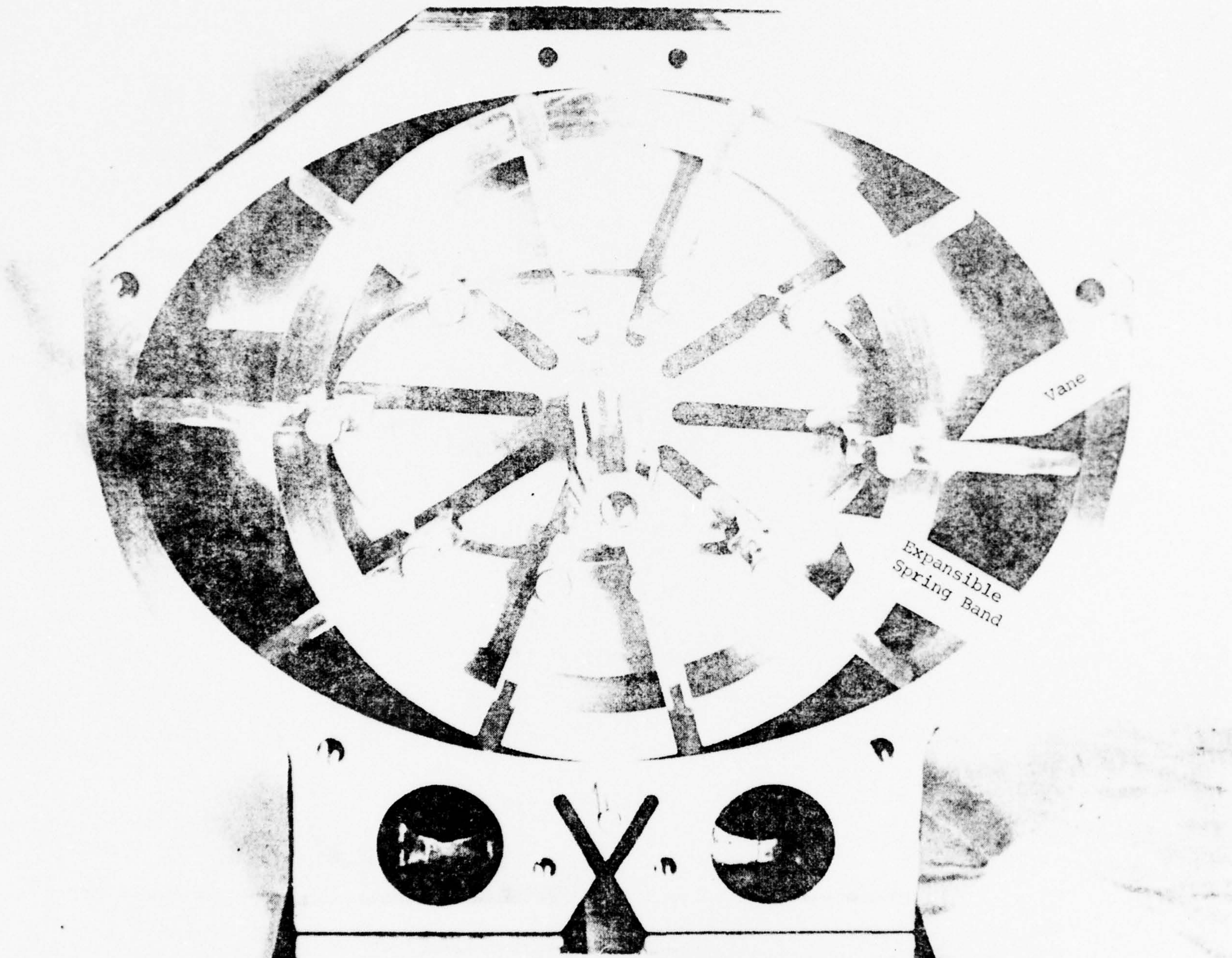


Figure 1.9 -- ROVAC Spring Biased Vane Design

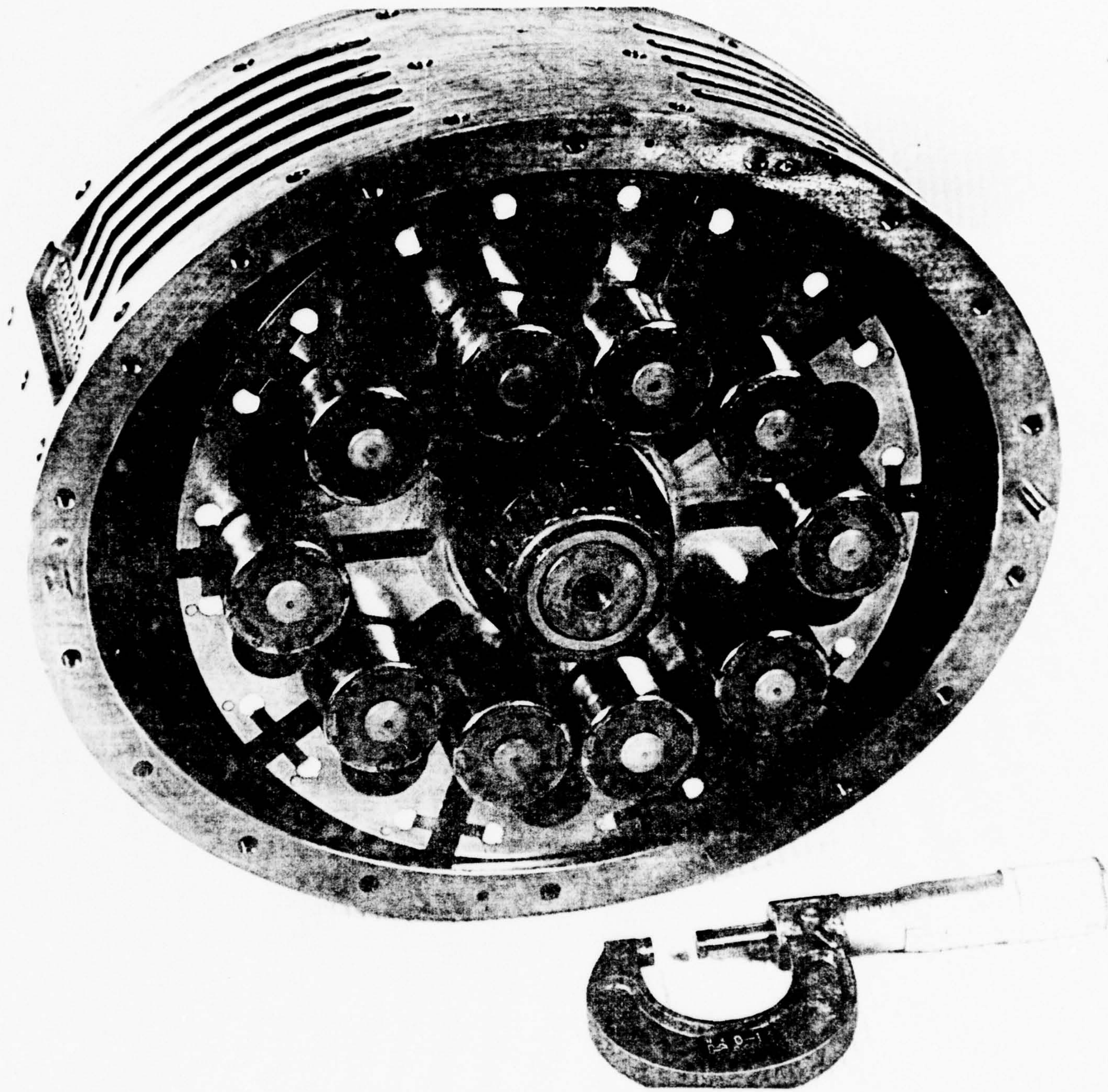


Figure 1.10 -- Modern Rovac ACM Employing Centrifugally Biased Vanes

- Inclusion of a performance testing program to demonstrate the cooling capacity and coefficient of performance (COP) of the Circulator.

Based on a ROVAC contract modification proposal, the contract requirements were modified from an open cycle low density compressor inlet air conditioning system to a closed cycle variable inlet density lubricated system. ROVAC's advanced development of the automotive closed loop variable density inlet Circulator and related engineering studies clearly revealed many advantages to this system. The following is a list of these advantages.

- The higher compressor inlet density (which is possible in the closed system) yields a higher mass flow rate (cooling capacity), thus reduces machine size and weight.
- The closed system can be fully oil lubricated, thus minimizing vane and bearing wear and friction. This results in increased efficiency and reliability.
- The lubricating oil also increases the volumetric efficiency of the Circulator by providing a more effective seal during compression and expansion. This also makes it possible to manufacture the machine to looser tolerances.
- The closed loop pressurization can be varied resulting in a variable cooling capacity system.
- The closed loop system is much quieter.
- Since higher mass flow rates can be accomplished in the closed loop a lower compressor pressure ratio can be used yielding lower internal vane loading and decreased port flow losses. This further enhances the machine efficiency and reliability.
- The closed loop system provides a positive means for dehumidification in much the same way a conventional fluorocarbon system does.

- With the closed system it is possible to inject water or some other phase changing component to the cycle to further enhance the system performance.

Based on The ROVAC Corporation's proposal (No. P76-2) the contract was again modified. This modification added the performance testing program outlined in the original proposal prior to the contract grant.

1.7 Summary

The program documented in this report includes the following sequence of events as specified in the modified contract Statement of Work to design, fabricate and test a ROVAC air cycle machine for the U.S. Army Mobility Equipment Research & Development Command:

- Use of ROVAC computer software in system analysis and machine design.
- Design of the Circulator - Model 606
- Circulator fabrication.
- Modification of existing MERDC/ROVAC 209 system.
- Circulator and system modifications for closed-loop operation.
- Performance testing and improvement modifications.

A computer model called Program ROVAK2 was used to perform the system analysis. The results of this analysis were used to optimize the Circulator design and to evaluate alternate system configurations.

The Model 606 Circulator was designed for an air flow rate of 8.5 lbs/min at 2500 rpm and 14.7 psia compressor inlet pressure. Stator housing, endcaps and rotor were constructed from steel. The vane material ultimately chosen was an advanced graphite fiber reinforced composite and the vane axles were designed to accept either roller or ball bearings. The outside dimensions of the machine measured approximately 8 inches long by an essentially elliptical 8 x 10 inch cross-section.

Circulator fabrication and system modification began in ROVAC's previous office and manufacturing complex in Maitland, Florida and finished in the current Rockledge, Florida plant. The precision internal elliptical contouring was performed by Cam Technology, Inc., Elmsford, N.Y.

Based on analytical studies and test results from related Circulator designs, several Circulator design and system modifications became apparent during the course of the contract. In order to provide the Army with the latest ROVAC technology it was necessary to incorporate as many of these changes as possible into the 606 system. The following is a summary of the changes to the original 606 open cycle design.

- System and Circulator conversion to a closed cycle system.
- Vane material changed from a PPS to an advanced graphite fiber reinforced composite.
- Conversion from a roller bearing to a ball bearing on the vanes.
- Addition of vane biasing ramps.
- Addition of endcap cooling jackets.
- Addition of an oil separator to the system.
- Installation of a 7 1/2 horsepower calibrated drive motor.

SECTION 2

SYSTEM ANALYSIS

The system analysis presented in this section was performed using the analytical capability in existing ROVAC computer software as described in Reference 3. The objective of this analysis was to optimize the design of the Model 606 Circulator and to predict its performance over the specified operating boundary conditions. Circulator design optimization was limited to machine geometry, vane dimensions, vane bearing size, estimated machine volume and port locations. From information gathered in this study a candidate Circulator was selected for detailed component design.

Since the previously discussed conversion from an open cycle system to a closed cycle system did not take place until after the Model 606 machine had been analyzed, designed and fabricated, the analysis contained herein was performed on an open cycle system. However, since the differences between the two modes of operation from a Circulator system analysis and basic machine design standpoint are so slight, the analysis presented is valid for both cases. (However, subsequent analysis disclosed shortcomings in the heat transfer calculations in the main computer model. These shortcomings involve intra-Circulator heat transfer. See the "Test and Demonstration Report", pp 32-35.) In the closed system configuration the compressor inlet and expander outlet pressures (considered the "reference pressure") can vary, however, over the reference pressure ranges employed by the 606 system, the analysis and basic design choices (such as: internal geometry, materials, operating pressure ratio, design speed, etc.) are not effected.

2.1 Boundary Conditions

Operational boundary conditions for the 606 system were established by the contract Statement of Work. The cooling load conditions were specified to be 90°F drybulb and 67°F wetbulb and the heat rejection temperature was set at 125°F. At these conditions the Circulator cooling capacity goal was specified as 8,000 Btuh with a coefficient of performance (COP) goal of 1.0.

2.2 System Configuration Selection

Several system arrangements were investigated for use in the 606 system. Of these, the two most promising configurations for the open cycle system are shown in Figures 2.1 and 2.2.

Figure 2.1 illustrates a simple system employing as basic components a ROVAC Circulator and one heat exchanger. As can be seen, the compressor section of the Circulator receives warm air from the cooling load at ① and compresses it to an elevated pressure and temperature at ②. This hot air is now pumped through the heat exchanger where the heat rejection process takes place. Near ambient (sink) temperature air, still at a pressure near that of state point ②, enters the expander section of the Circulator at ③ and is expanded back down to the ambient pressure at ④. This expansion process, of course, produces the cold air which is directed to the cooling load where it accepts heat and continues to the compressor inlet.

Figure 2.2 illustrates this same basic system with the addition of a regenerative heat exchanger to offset the 35°F temperature differential between sink and load. This improves the system capacity and performance because now the expander can begin expanding at a lower temperature at ③ and thus producing a lower temperature at ④ for the same system mass flow rate. The penalty paid is the addition of another heat exchanger which creates additional system pressure drop between compressor discharge and expander intake and between expander discharge and compressor intake. However, the small pressure drop penalty usually associated with a heat exchanger of this type is well worth it to get the lower temperature to the expander intake. Therefore, this system arrangement was chosen for the open cycle system analysis.

A third system configuration, shown in Figure 2.3, came about as a result of the closed cycle adaptation. It is basically the same regenerative arrangement that was chosen for the open cycle system with the addition of a secondary heat exchanger which is used to transfer heat from the cooling

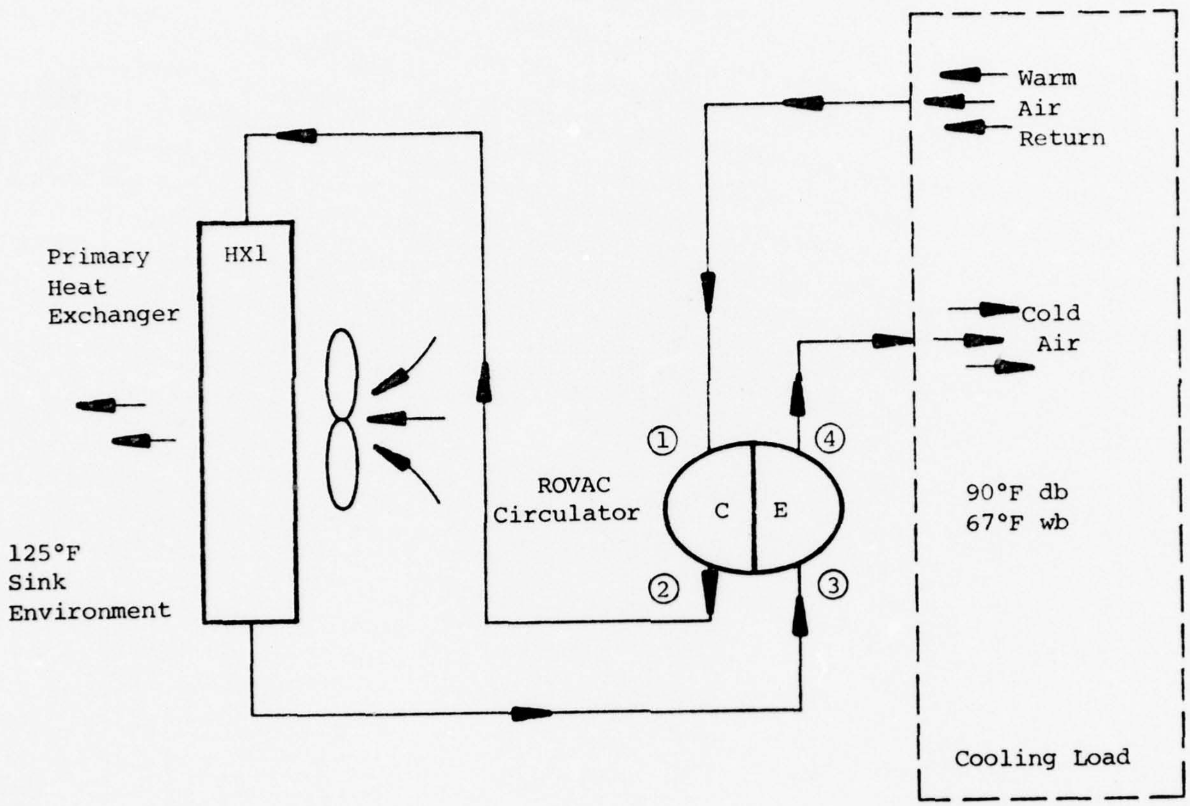


Figure 2.1 -- Basic Open Cycle System Configuration

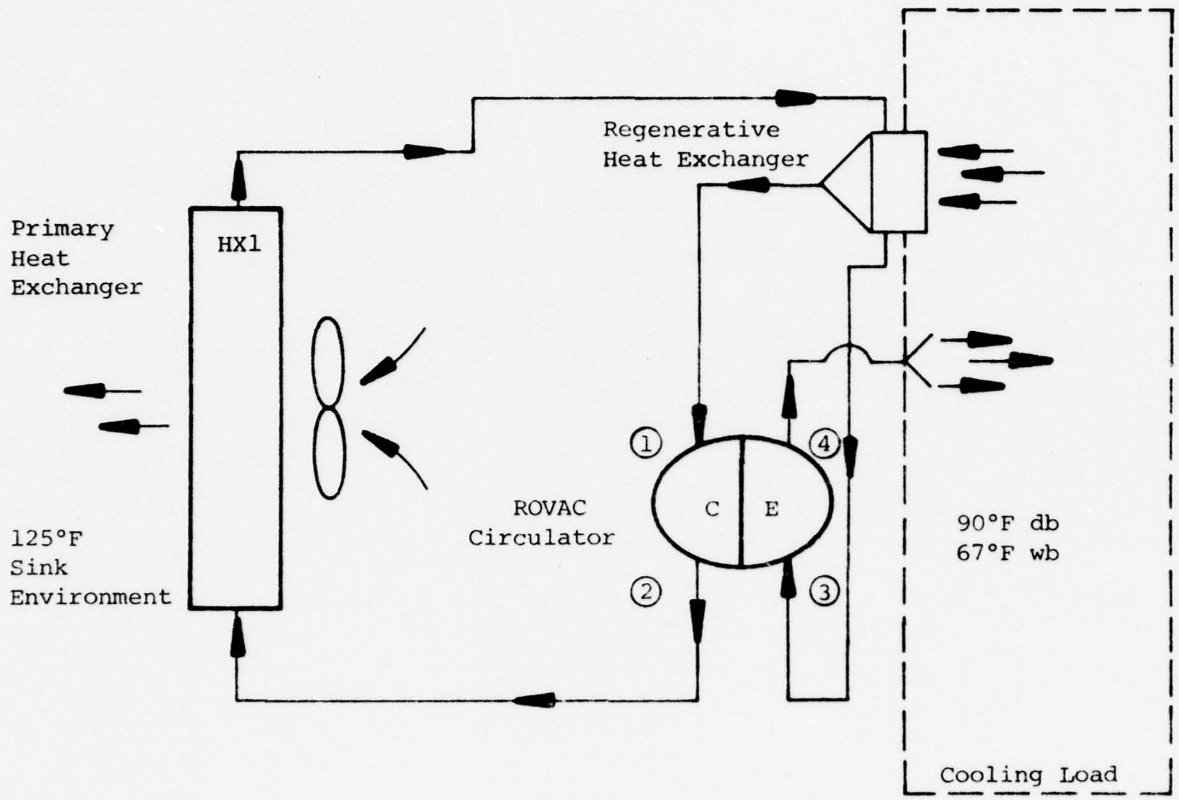


Figure 2.2 -- Open Cycle System Configuration with Regeneration

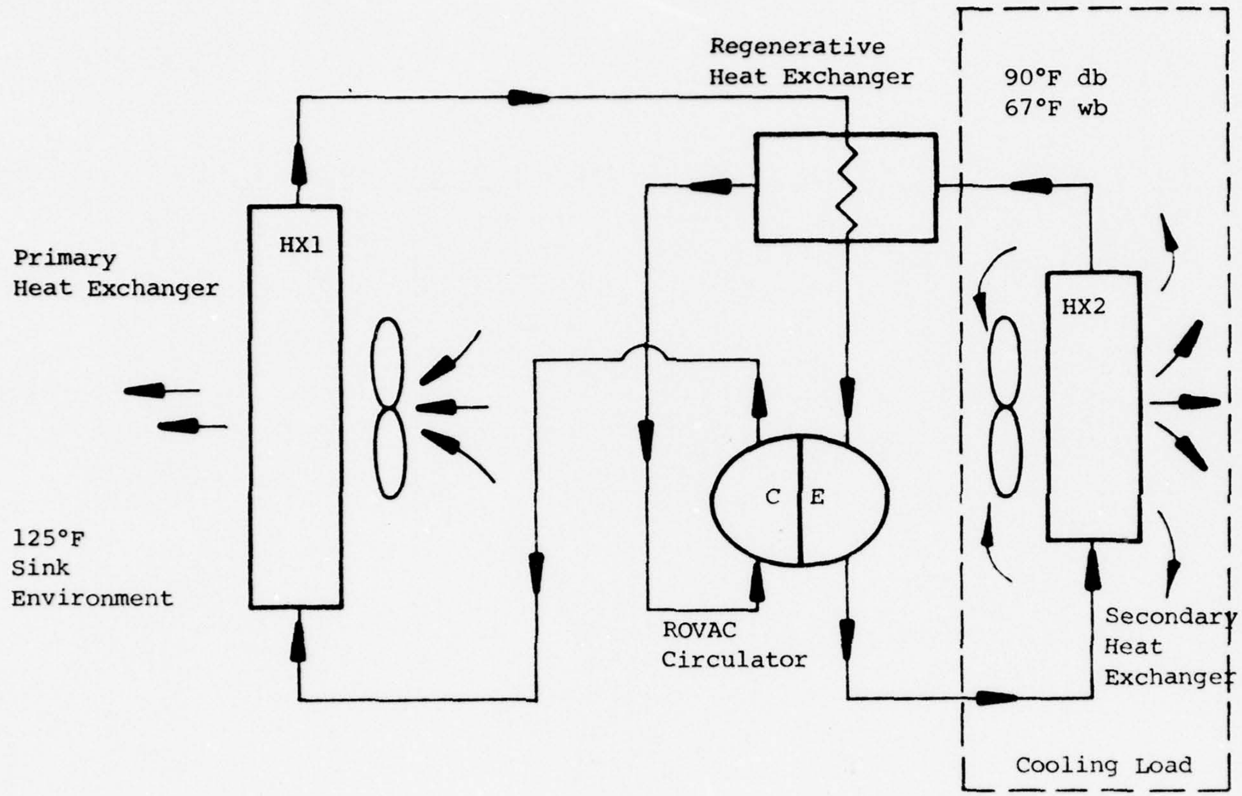


Figure 2.3 -- Closed Cycle System Configuration with Regeneration

load to the air in the closed loop. Ultimately, this is the 606 system configuration which was manufactured. More details on this system are presented in the following sections.

2.3 Model 606 Circulator Selection

The first task in the system analysis effort was to determine a machine size with which to perform the computer predicted performance optimization studies. The goal was to choose a machine geometry which would provide an adequate air flow rate at a compressor inlet pressure of 14.7 psia (open-cycle) and a rotor speed of 2500 rpm. Preliminary studies indicated that a machine having a maximum major axis (compressor section) of 9.0 inches, a minor axis of 7.281 inches and a rotor length of 4.0 inches would provide a sufficient mass flow rate at the specified operating conditions. It was found that a machine with this housing cavity cross-sectional geometry (ratio of minor to major axis) would not require a spring band to help maintain a positive contact between vane roller and cam track surface.

Parametric studies conducted in Reference 4 indicated that generally the ROVAC rotary vaned machine performs most favorably between 2000 and 3000 rpm. ROVAC's experience in designing and testing ROVAC Circulators tends to support this (3).

2.4 System Analysis Results

Table 2.1 presents a summary of the basic machine design parameters and computer predicted operational performance characteristics for the Model 606 Circulator at the recommended maximum operating speed of 2000 rpm. This data is based on the regenerative system arrangement shown in Figures 2.2 and 2.3

The effect the regenerative heat exchanger has on computer predicted system performance is illustrated in Figure 2.4. With no regeneration,

TABLE 2.1

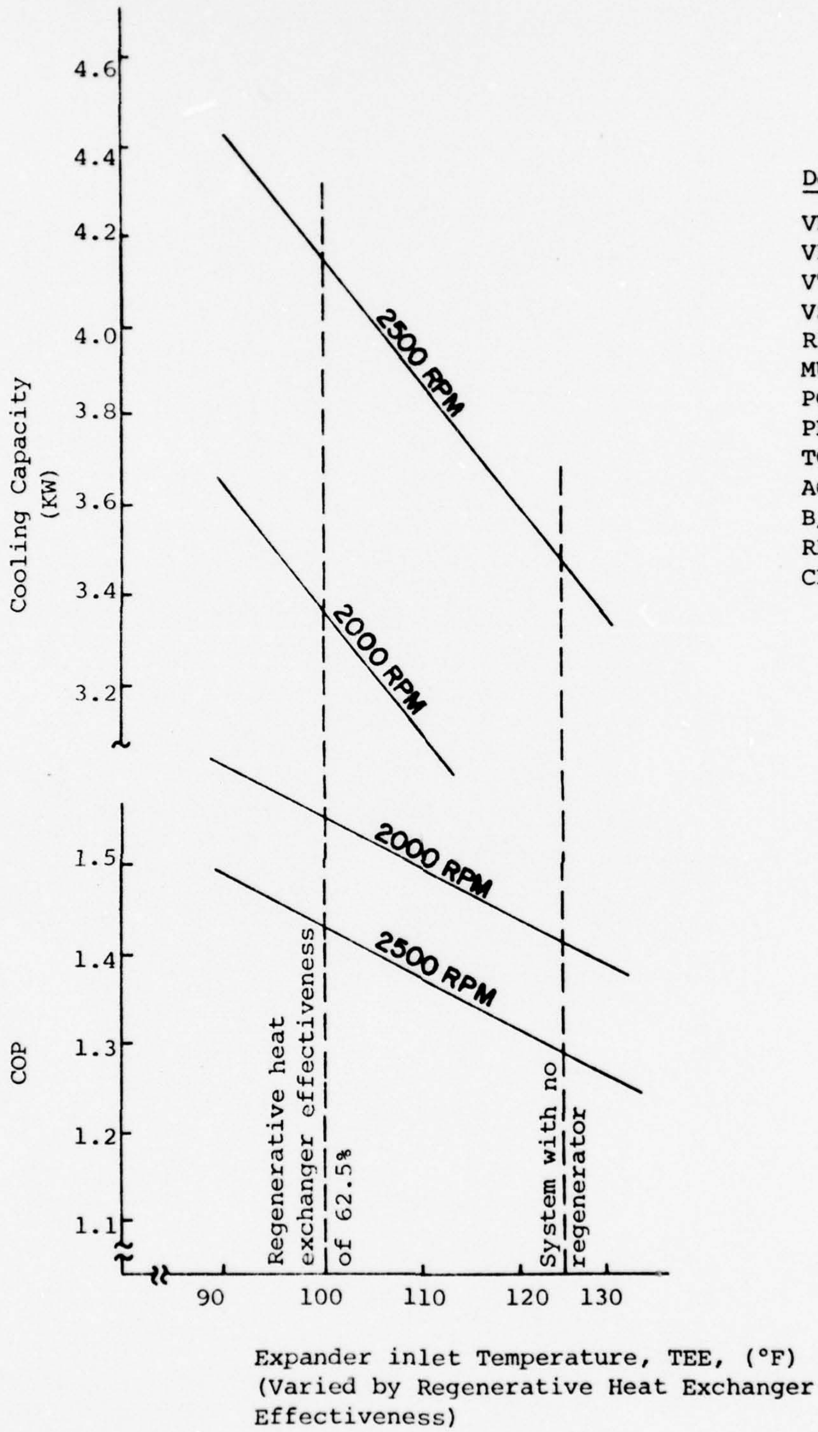
ROVAC MODEL 606 BASIC DESIGN PARAMETERS

PERFORMANCE CHARACTERISTICS

Item Description	Computer Symbol	Design or Predicted Value
Compressor inlet pressure, psia	PCI	14.7
Compressor discharge pressure, psia	PCD	44.1
Compressor inlet temperature, °F	TCI	90.0
Compressor discharge temperature, °F	TCD	310.8
Expander inlet pressure, psia	PEE	43.1
Expander outlet pressure, psia	PEO	14.7
Expander inlet temperature, °F	TEE	100.0
Expander outlet temperature, °F	TEO	5.0
Rotor speed, rev/min	RPM	2000
1/2 Ellipse major axis, Compressor, in.	AC	4.50
1/2 Ellipse major axis, Expander, in.	AE	4.361
1/2 Ellipse minor axis, in.	B	3.60
Rotor length, in.	RLG	4.00
Number of vanes	VN	10
Vane material density, lbs/cu.in.	VRO	0.066
Vane thickness, in.	VT	0.25
Vane sliding coeff. of friction	MU1	0.10
Vane roller diameter, in.	RIMD	1.125
Vane bearing dynamic capacity, lbs.	CD	2200
Estimated Circulator weight, lbs	SUMWT	61.7
Estimated Circulator volume, cu.ft.	PKV	0.18
ACM flow rate, lb/min	FLOC, FLOE	6.57
ACM power requirement, hp	HPREQ	2.92
Vane bearing group life, hours	MTBF	-
Vane wall average PV, psi x fpm	VWPVAV	5546
Compressor inlet port angle, degrees	PACI	-18.0
Compressor discharge port angle, degrees	PACD	60.9
Expander inlet port angle, degrees	PAFE	-63.8
Expander outlet port angle, degrees	PAEO	18.0
Compressor ellipse eccentricity, deg	-	36.87
Expander ellipse eccentricity, deg	-	34.36

TABLE 2.1 (Continued)

Item Description	Computer Symbol	Design or Predicted Value
Compressor adiabatic efficiency, %	AEFC	97.2
Expander adiabatic efficiency, %	AEFE	98.3
Compressor overall efficiency, %	COMEF	90.2
Expander overall efficiency, %	EXPEF	88.8



Design Conditions*

VN = 10
VRO = .066 lb/in^3
VT = 0.25 in.
VSR = 1.49
RIMD = 1.125 in.
MU1 = .1
PCI = 14.7 psia
PEO = 14.7 psia
TCI = TLO = 550°R
AC = 4.5 in.
B/A = 0.8
RPM = 2500 rpm
CPR = 3.0

*Refer to Table 2.2 for description of variable names

Figure 2.4 -- Effect of Regeneration on Predicted Performance of ROVAC Model 606 Circulator

TABLE 2.2 -- Description of Variables

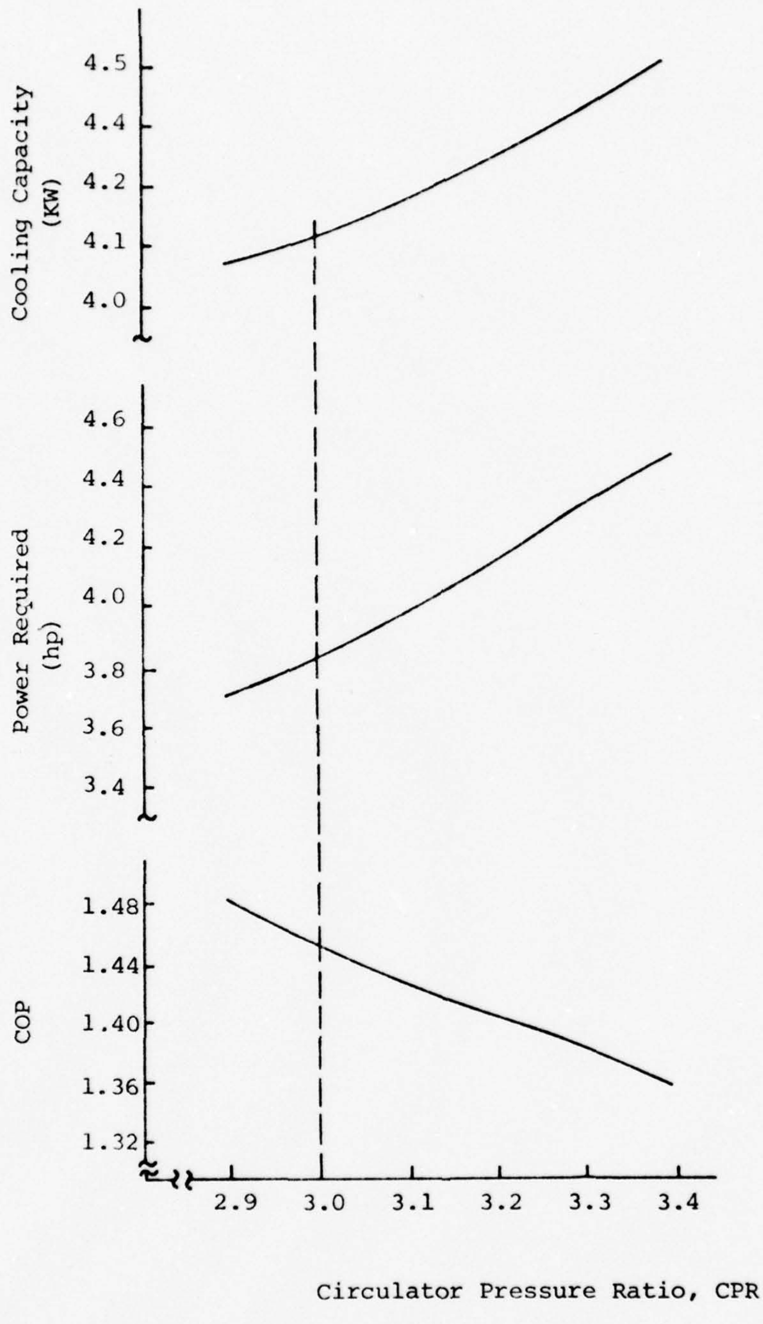
Variable Name	Description and Units
VN	Vane number
VRO	Density of vane material, (lb/in^3)
VT	Vane thickness, (in)
VSR	Ratio, length of vane remaining in rotor slot at major axis position/(A-B)
RIMD	Outside diameter of cam roller rim, (in)
MU1	Vane sliding friction coefficient
PCI	Compressor inlet pressure, (psia)
PEO	Expander outlet pressure, (psia)
TCI	Compressor inlet temperature, (deg.R)
TLO	Air temperature leaving load compartment, (deg.R)
TSNK	Temperature of intercooler sink, (deg.R)
TEE	Expander entry temperature, (deg.R)
CPR	Compressor pressure ratio
AC	Compressor side half major axis lenght of elliptical cylinder bore, (in)
B/A	Ratio of minor to major axis of elliptical cylinder bore
RPM	Compressor/Expander rotational speed, (RPM)

at 2500 rpm Circulator speed, the predicted Circulator COP is approximately 1.29. However, with a 62.5% effective regenerator the predicted COP increases to approximately 1.43. The map also shows a marked increase in system performance when the Circulator speed is decreased from 2500 rpm (Circulator design speed) to 2000 rpm (recommended Circulator maximum operating speed).

Figure 2.5 illustrates the effect of compressor pressure ratio (CPR) on system cooling capacity, Circulator power requirement and Circulator COP. As expected, the system cooling capacity increases with an increase in pressure ratio. However, Circulator COP decreases. This occurs as a result of increased vane and bearing loads (which cause higher friction) at the higher pressure ratios and increased thermodynamic work.

In selecting a pressure ratio (CPR) for the 606 design a trade-off was made between cooling capacity and COP. It was felt that a CPR of 3.0 would provide adequate cooling capacity and COP for performance demonstration testing.

The compressor pressure ratio also has an effect on the reliability of the Circulator. This is evident in Figure 2.6 where the computer predicted vane wall PV (Pressure, psi, x Velocity, fpm) and mean time between failures is plotted against CPR. At the compressor pressure ratio of 3.0 (as previously selected) the mean time between failures is predicted to be 13,750 hours and the predicted vane wall PV is 7,750 psi ft/min at 2500 rpm.

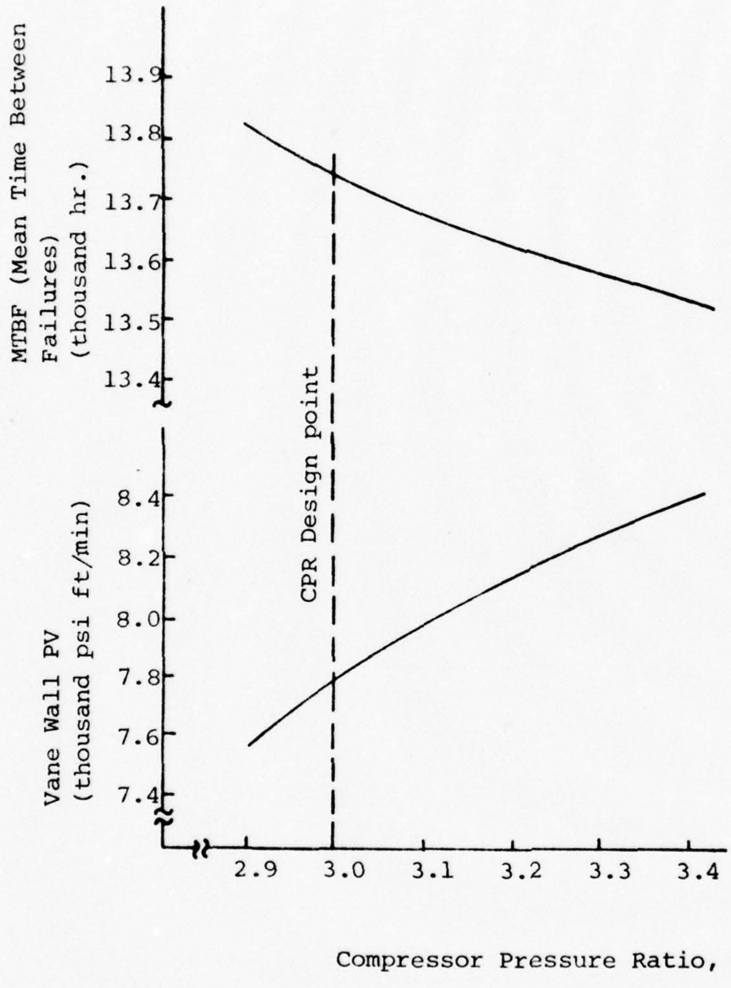


Design Conditions*

VN = 10
VRO = .066 lb/in^3
VT = 0.25 in.
VSR = 1.49
RIMD = 1.125 in.
MU1 = 0.1
RPM = 2500 rpm
PCI = 14.7 psia
PEO = 14.7 psia
TCI = TLO = 550°R
TSNK = 585°R
TEE = 560°R
AC = 4.5 in.
B/A = 0.8

*Refer to Table 2.2 for description of variable names

Figure 2.5 -- Effect of Compressor Pressure Ratio on Computer Predicted Performance for ROVAC Model 606 Circulator (System with Regenerative Heat Exchanger)



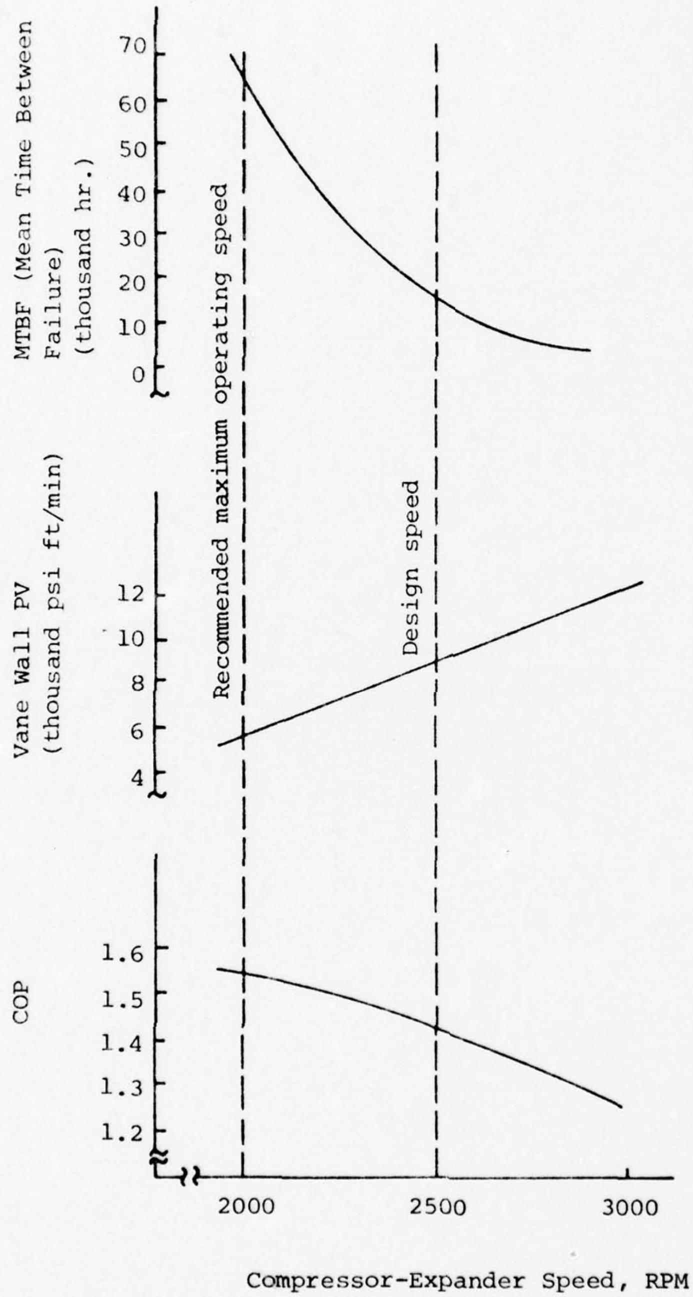
<u>Design Conditions*</u>

VN = 10
VRO = .066 lb/in^3
VT = 0.25 in.
VSR = 1.49
RIMD = 1.125 in.
MU1 = 0.1
RPM = 2500 rpm
PCI = 14.7 psia
PEO = 14.7 psia
TCI = TLO = 550°R
TSNK = 585°R
TEE = 560PR
AC = 4.5 in.
B/A = 0.8

*Refer to Table 2.2 for description of variable names

Figure 2.6 -- Effect of Compressor Pressure Ratio on Reliability of ROVAC Model 606 Circulator (System with Regenerative Heat Exchanger)

Selection of 2000 rpm as the recommended Circulator maximum operating speed was based on studies investigating the effect of rotational speed on Circulator reliability and COP. The results of these studies are shown in Figure 2.7. The greatest benefit in the 500 rpm drop in Circulator speed shows up in reliability where there is an approximate 5-fold increase in MTBF. As previously shown in Figure 2.4 and again here there is a favorable increase in COP as a result of decreasing the maximum operating speed from the design point of 2500 rpm to 2000 rpm.



Design Conditions*

VN = 10
VRO = .66 lb/in^3
VT = 0.25 in.
VSR = 1.49
RIMD = 1.125 in.
MU1 = .1
AC = 4.5 in.
B/A = 0.8
PCI = 14.7 psia
PEO = 14.7 psia
TCI = TLO = 550°R
TSNK = 585°R
TEE = 560°R
CPR = 3.0

*Refer to Table 2.2 for description of variable names

Figure 2.7 -- Effect of Circulator Rotational Speed on Performance and Reliability of ROVAC Model 606 (System with Regenerative Heat Exchanger)

SECTION 3

MODEL 606 CIRCULATOR AND SYSTEM DESIGN

Design of the ROVAC Model 606 Circulator was conducted in accordance with the results of the system analysis and the performance goals stipulated in the contract Statement of Work. the objective was to design a Circulator which would perform as close as possible to the computer predicted performance characteristics generated by the system analysis (Section 2) when operated in a breadboard system at the Army environmental test conditions.

The Army supplied to the Contractor the existing ROVAC 209 open cycle system (with the exception of the 209 Circulator), which was developed under a previous contract, for modification to accept the 606 Circulator for performance testing. This system underwent extensive modification to implement closed cycle operation and to add the regenerative circuit. Since it was not within the scope of the contract to develop or optimize the system design, system modifications and hardware additions were made with commercially available hardware. For example, the secondary heat exchanger used to transfer heat from the cooling load to the closed loop air stream was a modified vapor cycle evaporator instead of a well-developed air to air heat exchanger designed specifically for that purpose.

3.1 System Description

As discussed previously the 606 System is a prototype closed-loop air cycle variable cooling capacity air conditioning system with regeneration. As illustrated in Figure 3.1, the system consists mainly of the Model 606 prototype ROVAC positive-displacement rotary vane Circulator, Circulator electric drive motor, three heat exchangers, an oil separator and electric

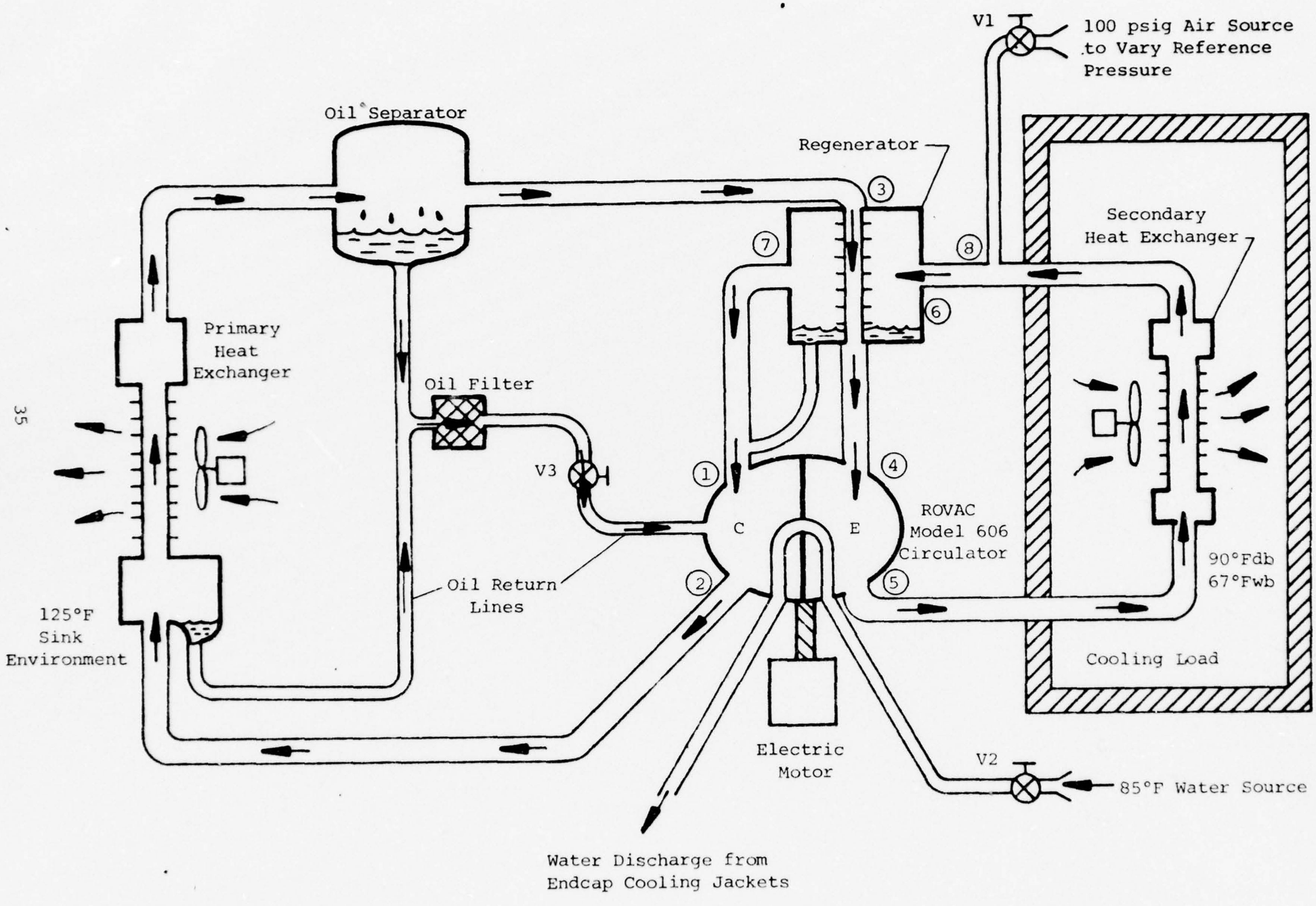


Figure 3.1 -- Schematic Diagram of ROVAC Model 606 Air Cycle System

driven fans to pass air across the primary and secondary heat exchangers. The system was designed for full oil lubrication of the Circulator which is accomplished by oil reclamation by the oil separator and primary heat exchanger tank and the pumping action of the system's internal pressure differentials.

To describe how the system functions, imagine air entering the compressor section of the Circulator at 1 wherein it is compressed polytropically to an elevated pressure and temperature at 2 by rotary motion of the Circulator internal rotating components, the rotor-vane assembly. This hot air is then pumped, by continued Circulator motion, to the primary heat exchanger where the heat is rejected to the 125°F sink environment. The near ambient (sink) air, still at an elevated pressure, then flows through an oil separator wherein a large percentage of the Circulator lubricating oil is separated from the air stream. (The lubrication system will be discussed subsequently.) This relatively oil free air then enters the high pressure side of the regenerator at 3 where it is further cooled down by the cooler (approximately 88°F) air returning from the secondary heat exchanger. At this point the total pressure has been reduced slightly by the pressure drops associated with the primary heat exchanger, oil separator, regenerator and the connecting ductwork. In a well developed system this pressure drop should be small, perhaps around 1.5 psi. Continuing, high pressure cooled air now enters the expander section of the Circulator at 4 wherein it is expanded polytropically to near that of the reference pressure at 5 . During the expansion process a large percentage of the work of compression is recovered thereby producing cold air at 5 . This cold air is then pumped through the secondary heat exchanger, wherein heat is absorbed from the cooling load, thus a cooling effect is achieved. The relatively warm (approximately 88°F) air emerging from the secondary heat exchanger is then pumped through the low pressure side of the regenerator wherein the air accepts additional energy from the relatively hot (approximately 130°F) air entering the regenerator at ③ . Air emerging from the regenerator at ⑦ is then pumped to the compressor section of the Circulator where this discussion began. A small pressure differential exists between points ① and ⑤ due to the flow restrictions caused by the secondary heat exchanger, regenerator and connecting ductwork.

Since the system is closed, the reference pressure (level of pressure in the low pressure circuit, between expander discharge and compressor intake) can be varied. This was accomplished in the 606 breadboard system by connecting a 100 psig air source to the reference pressure duct at ⑧ . Since the cooling capacity of the system is directly proportional to the density of the air in the loop, varying the reference pressure (valve V1) proportionately varies the cooling effect.

As illustrated in the diagram, lubricating oil is returned to the Circulator from three collection points: oil separator, primary heat exchanger and regenerator. The oil returning from the oil separator and primary heat exchanger is passed through a filter where solid contaminants are separated from the oil stream. This oil is returned to the center of the Circulator via the rotor drive shaft. The pressure differential between the heat exchanger and oil separator and the rotor cavity supplies the necessary driving force to move the oil. A small mechanical valve, V3, regulates the flow rate. To prevent excessive oil buildup in the regenerator, an unregulated oil return line was connected from the regenerator to the compressor intake manifold.

Water was supplied to the Circulator endcap cooling jackets from a 85°F water source. A small mechanical valve, V2, regulated the flow rate.

Advantages of the closed cycle system over the open cycle system were discussed in Section 1.6.

The following paragraphs present the design and description of the components employed in the 606 breadboard system.

3.2 Model 606 Circulator Design

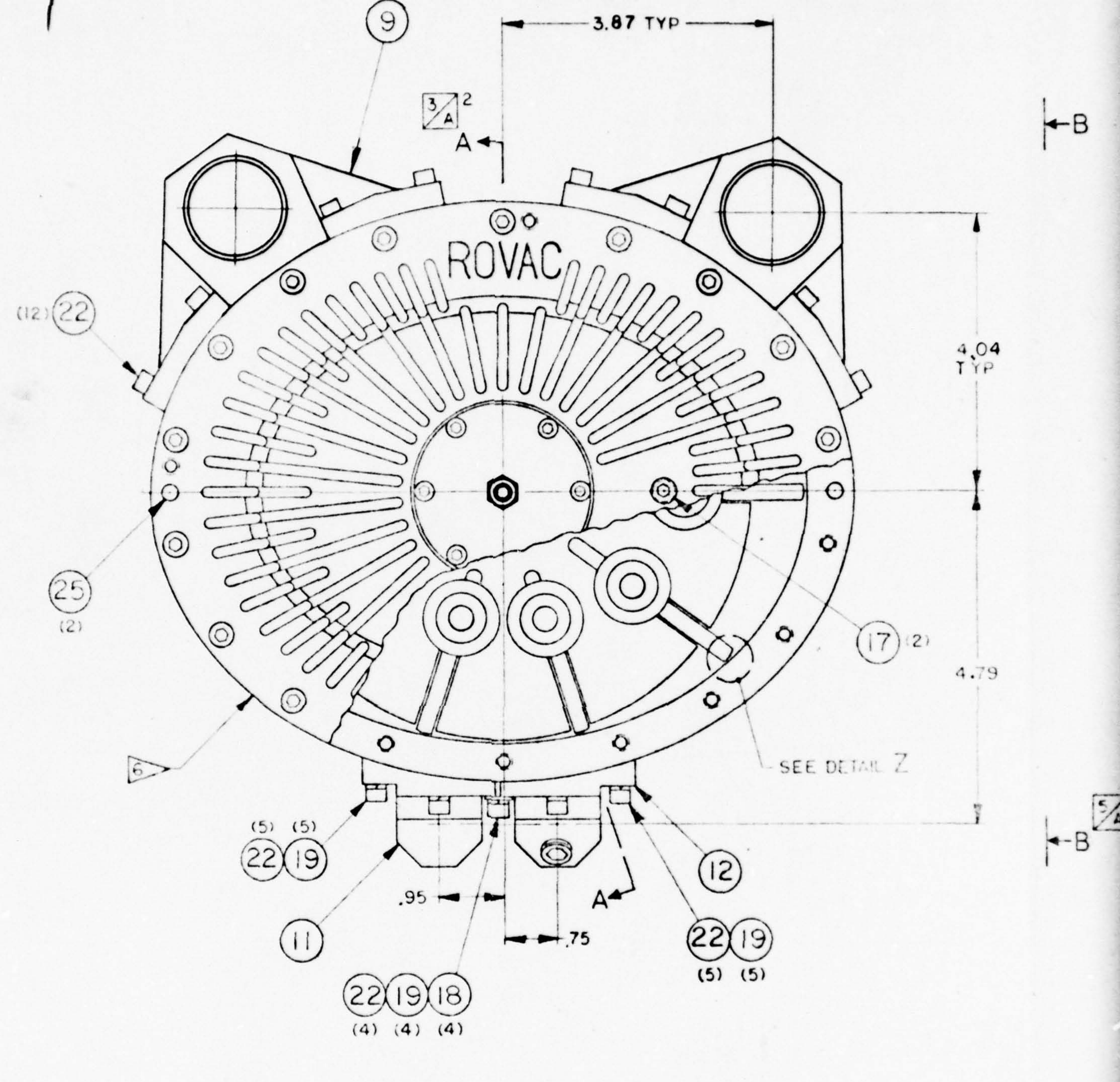
Basic operation of the Circulator and the thermodynamic processes involved were discussed in Section 1.3. This Section contains discussion on the Circulator design.

Figures 3.2 and 3.3 present engineering drawings of the Model 606 Circulator assembly. Included in Figure 3.2 is a component and sub-assembly breakdown and the set of detailed engineering drawings supplied to the Army contains detailed drawings of each component. Figure 3.4 is a photograph of the machine with one endcap removed giving visual exposure to the rotating components. The following paragraphs discuss the design of the subcomponents.

3.2.1 Stator Housing Design

As shown in Figure 3.5 the Model 606 stator housing has an elliptical cross-section with a generally constant wall thickness. Material used in its design was low carbon steel.

The housing inner cavity was designed to have an asymmetrical elliptical contour. This asymmetry exists between compressor and expander sections, about the minor axis. However, symmetry about the major axis was maintained. This elliptical mismatch allows the maximum compressor intake and final maximum expander discharge volumes to be established uniformly about the respective major axis. At these two locations, the Circulator internal volume change is at a minimum (it goes to zero instantaneously). The primary



7. IMPERIAL - EASTMAN CORP, CHICAGO, ILLINOIS

6. TRUE ELLIPSE WITH AN 10.125 MAJOR AXIS, 8.324 MINOR AXIS.

5. PRESS IN PLACE AS SHOWN BOTH ENDS.

4. PRELOAD F/N 15 200 ± 50 LBS TO DETERMINED THKNS OF F/N 4 AND 8.

3. THE TIMKEN CO. CANTON, OHIO 44706.

2. CRANE PACKING CO., MORTON GROOVE, ILLINOIS 60053.

1. REMOVE BURRS AND BREAK SHARP EDGES.

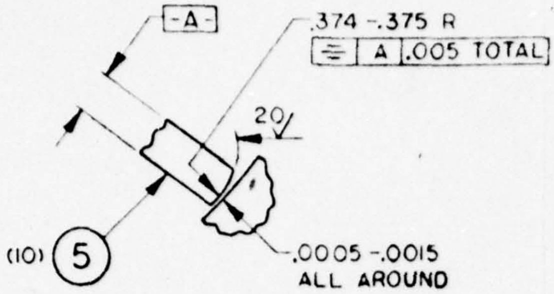
NOTES:

A	A
SH 2	SH 1

REVISION STATUS OF SHEETS

LTR	ZONE	DESCRIPTION	DATE	APPROVED
A		REVISED FROM OPEN LOOP SYSTEM TO A CLOSED LOOP SYSTEM SH 1 AND 2.	7-12-76	[illegible]

BEST AVAILABLE COPY



DETAIL Z
SCALE: 2/1
10 PLACES

QUANTITY/DASH NO	FIND NO	PART OR IDENTIFYING NO	NOMENCLATURE OR DESCRIPTION	STOCK SIZE	MATERIAL OR VENDOR	MATERIAL SPECIFICATION	FINISH OR CODE IDENT	REV
	27							
1	26		SEAL, SHAFT		SEE NOTE 2			
2	25		DOWEL PIN	.2495 -.2500 DIA X 1.25				
36	24		SCREW SOC HD	1/4 -28 X 3/4	STEEL			
6	23		" " "	10 -32 X 3/4	"			
44	22		" " "	10 -32 X 5/8	"			
	21							
36	20		WASHER, LOCK, SPLIT	1/4				
26	19		WASHER, LOCK, SPLIT	NO. 10				
4	18		WASHER, PLAIN	NO. 10				
3	17		PLUG, HEX, PIPE	1/8 -27 DRYSEAL NPTF	STEEL			
1	16		KEY	3/16 SQ X 1 1/2	STEEL	SAE1008 - 1020		
2	15	LM12749	CONE, BEARING		SEE NOTE 3			
AR	14	44 - P - 1/4	TUBING, POLY - FLO					
1	13	261P / 1/4 TUBE	NUT AND SLEEVE ASSY		SEE NOTE 7			
1	12	C4 - 1011 - 2	MANIFOLD, INTAKE					
1	11	C4 - 1011 - 1	" , DISCH					
1	10	C4 - 1014 - 2	" , OUTLET					
1	9	C4 - 1014 - 1	MANIFOLD, INLET					
1	8	A - 1124 - 1	COVER PLATE, SEAL					
1	7	A - 1094 - 1	ROTOR					
1	6	A - 1052 - 1	COVER PLATE					
10	5	A - 1125 - A	VANE ASSY					
1	4	A - 1046 - 1	SPACER					
1	3	A - 1126 - 1	OIL ROD FITTING					
1	2	6063790040A02 - A	CAM COVER ASSY					
1	1	" A01 - 1	STATOR HOUSING					
	A	6063790040A00 - A	CIRCULATOR ASSY					

ODD DASH NO. SHOWN — EVEN DASH NO. OPP — PARTS LIST

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES AND ARE AFTER PLATING
TOLERANCES ON FRACTIONS ± — DECIMALS .X ± — .XX ±.03 .XXX ±.010 ANGLES ± —
MATERIAL

CONTRACT NO.
DRAWN BY A A C DATE 7-12-76
CHECKER
GR ENGR
PROGRAM APPROVAL
CUSTOMER APPROVAL

THE ROVAC CORPORATION
105 BLDG., CANDACE DR., MAITLAND, FLORIDA 32751

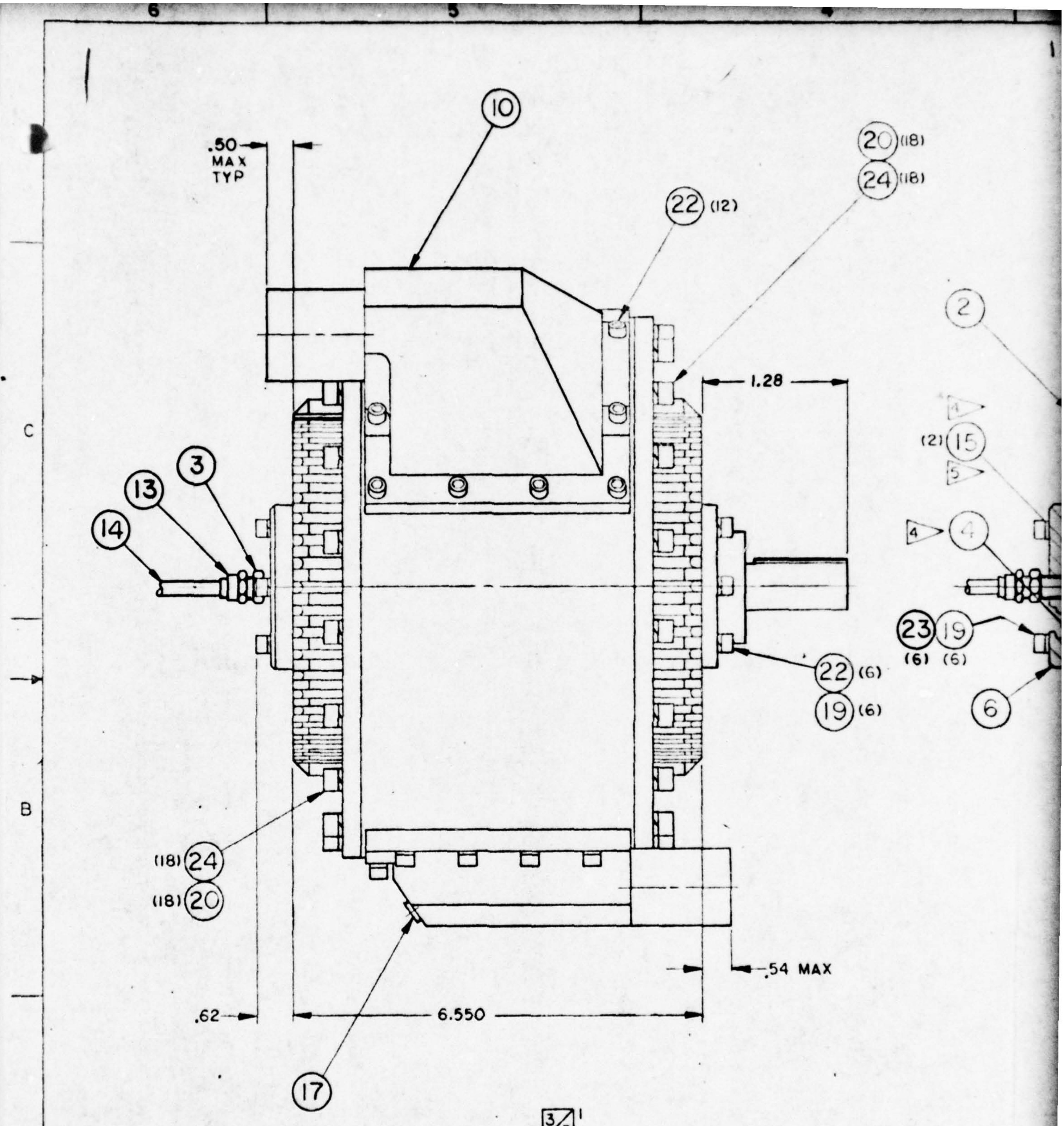
CIRCULATOR ASSEMBLY

Figure 3.2

SIZE D — CODE IDENT NO — DRAWING NUMBER 6063790040A00
SCALE 1/1 — SHEET 1 OF 2

A	A
SH 2	SH 1

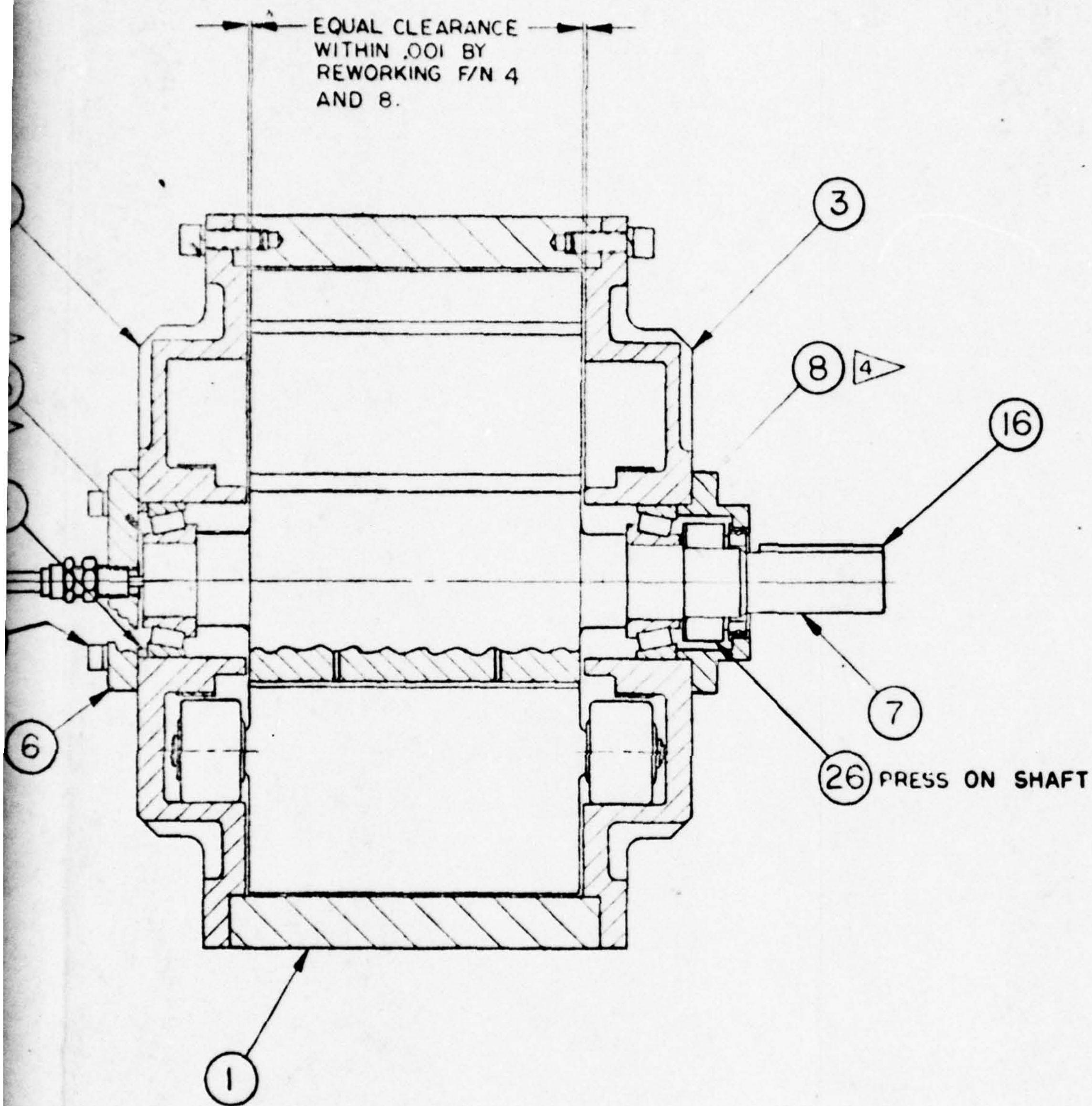
STATUS
CALC WT — DASH NUMBER — NEXT ASSY — NEXT ASSY — FINAL ASSY — TEST
APPLICATION — QTY REQD



6
5
4
C
B
A
10
.50
MAX
TYP
20 (18)
24 (18)
22 (12)
2
1.28
4
(2) 15
5
3
13
14
4
23 19
(6) (6)
22 (6)
19 (6)
6
(18) 24
(18) 20
.54 MAX
.62
6.550
17

VIEW B - B 3/B 1

MF	LTR	ZONE	DESCRIPTION	DATE	APPROVED
	A		REVISED FROM OPEN LOOP SYSTEM TO CLOSED LOOP SYSTEM SH 1 AND 2	7-12-76	WKL 7/12/76



SECTION A A 5/D 1

Figure 3.3

SIZE	CODE IDENT NO.	DRAWING NUMBER
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SCALE 1/1		SHEET 2

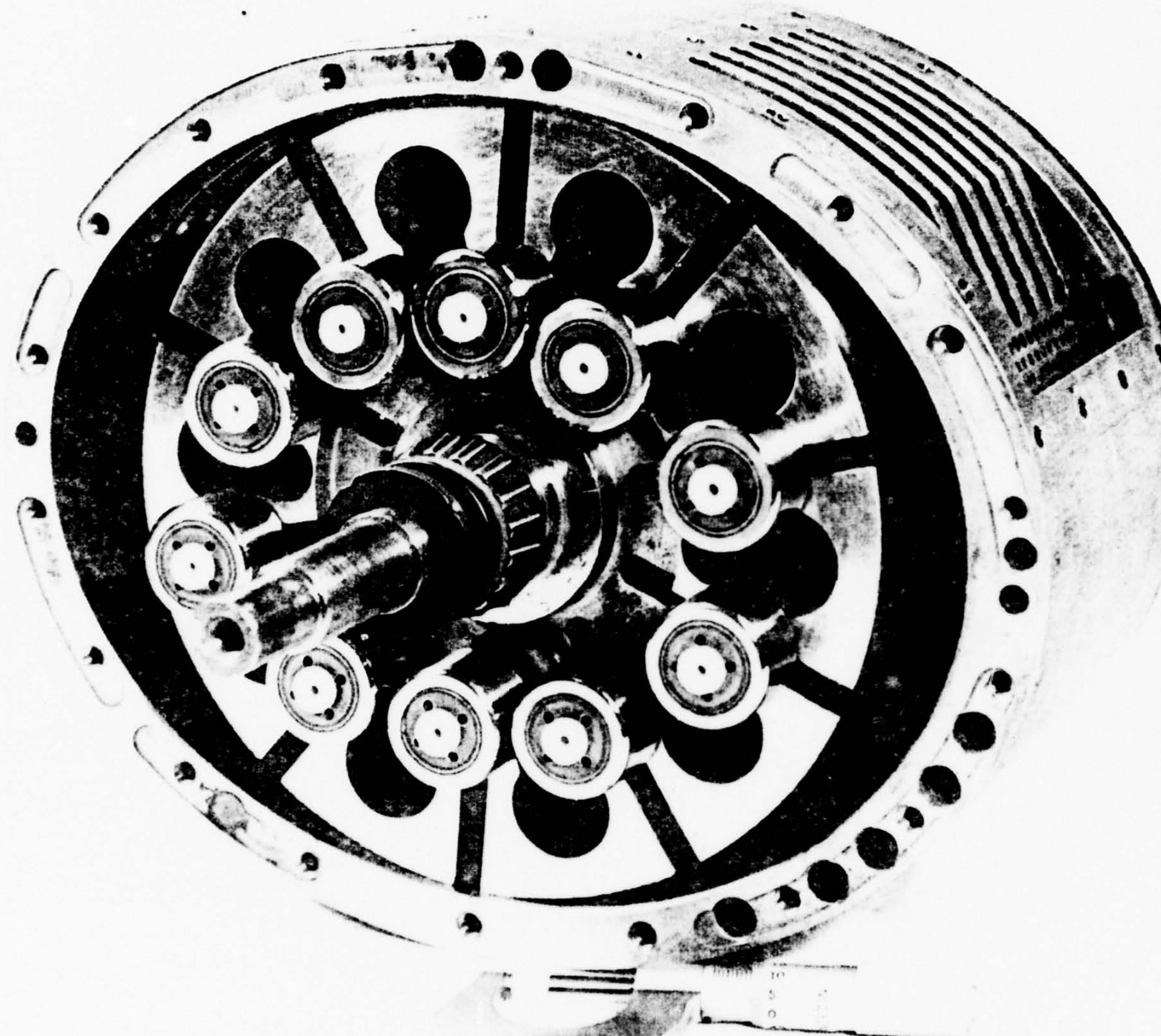


Figure 3.4 -- ROVAC Model 606 Circulator With Forward Endcap Removed

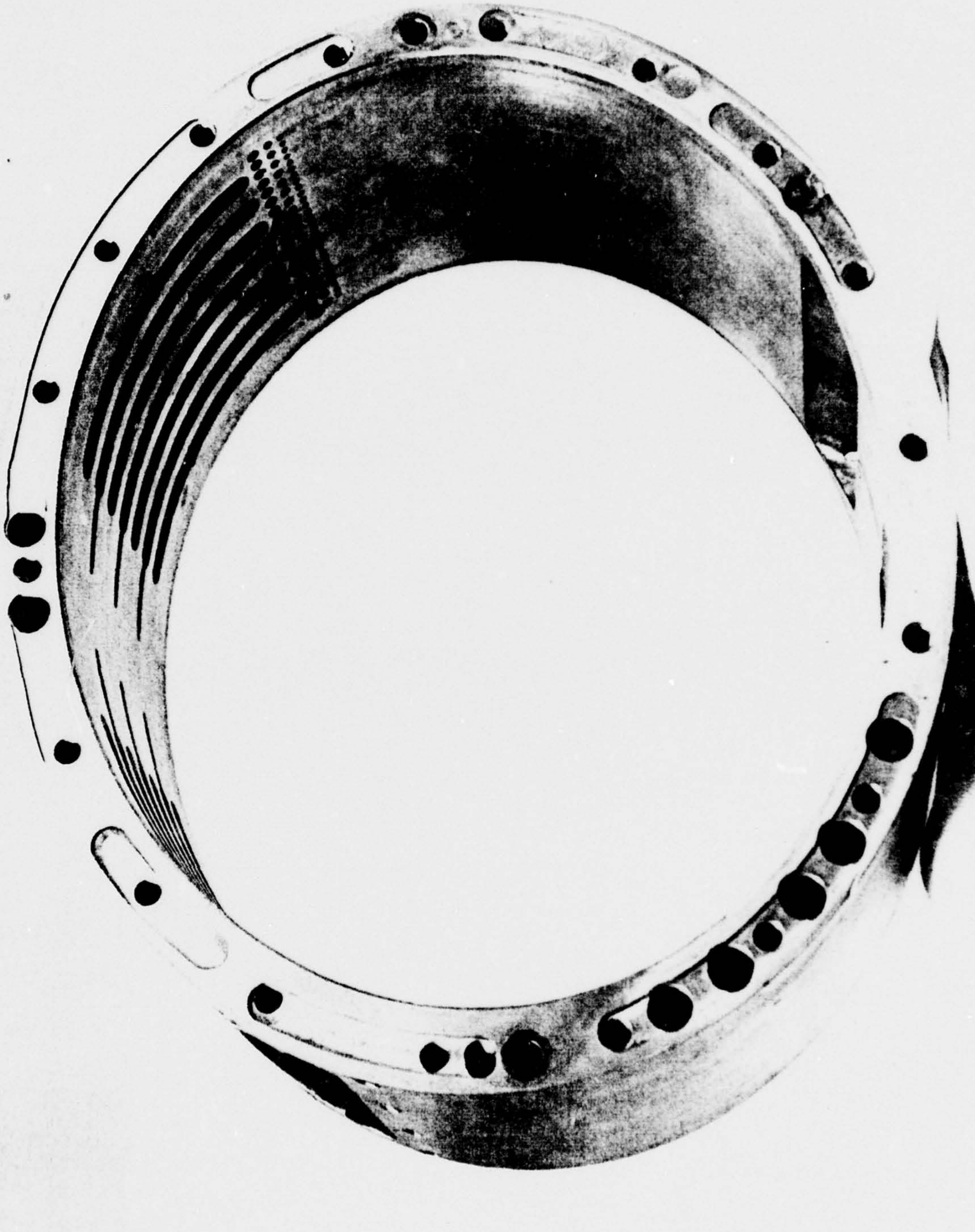


Figure 3.5 -- Model 606 Stator Housing Design

advantage to start moving gas into or out of the machine at this moment is that the air flow pressure drop is minimized since the velocity is low. Other advantages are: noise reduction, reduced vane forces, improved vane cam tracking and reduced machine size.

Aluminum manifolds attached to the outer periphery of the housing serve to direct the air flow from the machine port openings to the duct system. The expander discharge manifold is insulated from the inside to minimize heat transfer effects during cold air discharge.

During bench testing of the 606 machine it was determined that the steady state performance was being degraded in part by heat transfer from the compressor section of the housing to the expander section. To minimize this problem a series of axial holes were bored lengthwise through the housing and the housing to endcap contact surface area was reduced by a milling operation. The holes serve to reduce the conductive heat transfer area in the direction along the housing contour to the expander. Likewise, the milled reliefs in the housing reduce the conductive area for heat transfer between housing and endcaps.

3.2.2 Endcap Design

The 606 endcap is a one-piece design, housing both rotor bearing bore and cam track surface. Because of the cam track surface bearing requirements the entire endcap was constructed from a steel alloy and then hardened to a Rockwell hardness of 58 on the "C" scale. The outside surface of the endcap was finned and cored to aid in transferring compression heat to the environment and to reduce weight.

The Contractor's internally funded development program and preliminary testing of the 606 machine dictated that excessive heat transfer to the cold expander section can be conducted by the endcaps. Hot sources for this heat transfer effect are the compression process and the friction heat being generated by the vanes and vane bearings. To minimize this problem each endcap was fitted with aluminum water cooled jackets. The jackets were designed to enclose the cam track regions and the compressor section of the endcap.

Implementation of the endcap cooling jackets and conductive area reductions made on the housing were attempts to reduce the heat transfer problem which exists in the 606 design. Future development effort should be directed towards a thorough investigation of the heat transfer mechanisms and designs which will alleviate the problem. (See pp 32-35 of the "Test and Demonstration Report".)

3.2.3 Rotor Design

Figure 3.6 illustrates the 606 rotor design. It was fabricated from a solid cylindrical block of low carbon steel. Each of the ten pie-shaped rotor segments were cored-out for weight reduction.

Timken tapered roller bearings were employed as the rotor main support bearings. Each bearing (cone #LM12749 and cup #LM12711) has a basic load capacity of 1280 pounds at 500 rpm for 3000 hours L10 life (5). The bearings have an axial preload of approximately 200 pounds force to minimize rotor axial play.

During bench testing at the Contractor's facility, the rotor periphery was equipped with a dynamic pressure transducer. Used in conjunction with an oscilloscope, this device will give an accurate pressure versus rotation angle history of the ROVAC machine. This information is valuable in analyzing the performance of the machine in regards to proper port location, noise generation and leakage.

3.2.4 Vane Design

Through the Contract's internally funded development program it was determined that the carbon and steel axle vane configuration, which was

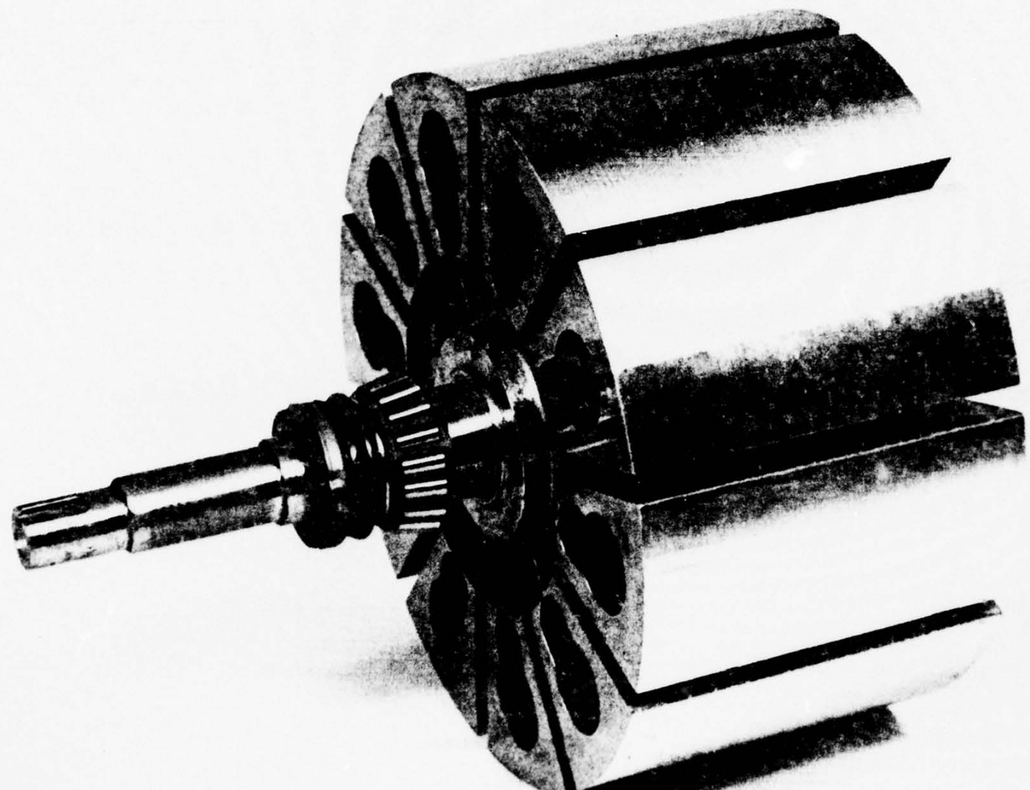


Figure 3.6 -- Model 606 Rotor Design

the first choice for the 606 machine, was not a sound design. Since carbon is brittle and very weak in tension, repetitive failures in the region where the vane body was relieved to accommodate the axle were occurring in test machines. Of course, at this time, the 606 vane design was changed.

Since the carbon/steel vane design problems were associated with the carbon vane body it was decided to change vane body material and still utilitize the steel axle design. The material chosen for this second vane design was a special formulated 40% graphite fiber reinforced polyphenylene sulfide (PPS) plastic which had a flexural modulus of approximately 4×10^6 psi, had favorable tensile properties and was not brittle (6). A one-piece vane assembly was formed by injection molding the plastic around the steel axle. Destructive structural tests made on this design showed that there was an exceptionally good bond between the axle and the vane body and the physical properties of the plastic material indicated that the vane should perform favorably. However, after several bench test sessions, during preliminary system performance testing, several of the vanes showed signs of failure. Cracks were beginning to appear along what is commonly called the "mold packing lines". These lines look much like the annual rings in a piece of lumber. Apparently, after prolonged exposure to the oil saturated environment inside the Circulator the vane body soaked up sufficient oil to cause these mold lines to weaken and crack. This problem could be solved by optimizing the molding operation. However, this process could take several months and be very costly. Instead of trying to perfect the molding process it was decided to model the 606 vane after a design which was being developed and tested favorably in the Contractor's advanced development program.

The final 606 vane design which was modeled after a successful design is shown in Figure 3.7. This is a 3-piece design employing an advanced graphite fiber reinforced composite material (GY70) for the vane body. This material has a tensile strength of approximately 18,000 psi, a flexural

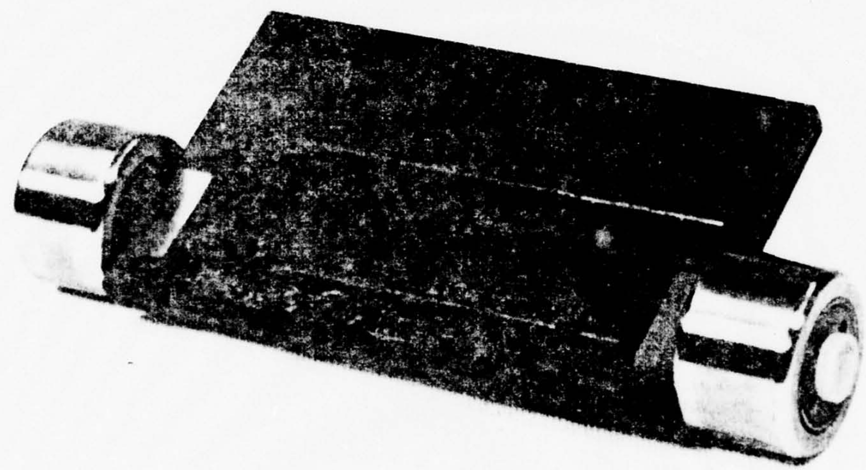


Figure 3.7 -- Model 606 Vane Design

modulus of 12 x 10^6 psi, a density of 0.060 lbs/cu.in. and a coefficient of thermal expansion of approximately 0.8 in/in/°F, making it an excellent vane material (7)(8). The two hardened steel axles located on each end of the vane are held in place by an adhesive and a thru draw-bolt.

To provide for an even distribution of oil across the vane surfaces while sliding in the rotor slot a pattern of shallow reliefs were machined in the vane flanks. Also, to provide a labyrinth type seal between the vane edges and endcap surfaces, shallow grooves were machined in each vane edge.

3.2.5 Vane Bearing Design

Figures 3.8 and 3.9 illustrate the two different vane bearing designs employed in the 606 machine. Both designs are interchangeable on the same vane/axle design. However, due to an improvement modification which was made to the endcaps, the 606 machine will only accept the vane bearing design shown in Figure 3.9.

Figure 3.8 illustrates the original 606 vane bearing design which employed Torrington WJ-081210 caged roller bearings. The steel cylindrical portion of the vane axle, called the axle stub, serves as the inner race of the bearing and the steel vane bearing rim (cam follower) serves as the outer race. Each bearing has a basic dynamic capacity of 2200 pounds with a limiting speed of 34,600 rpm (9). The bearing/roller assembly is retained on the axle stub by a nylon pin and sleeve assembly. Analysis and limited reliability tests on similar machines shows that this is a very conservative heavy duty vane bearing design.

Through the Contractor's internally funded development programs and bearing tests conducted by the Air Force it was determined that a conservative approach to vane bearing design can have a degrading effect on

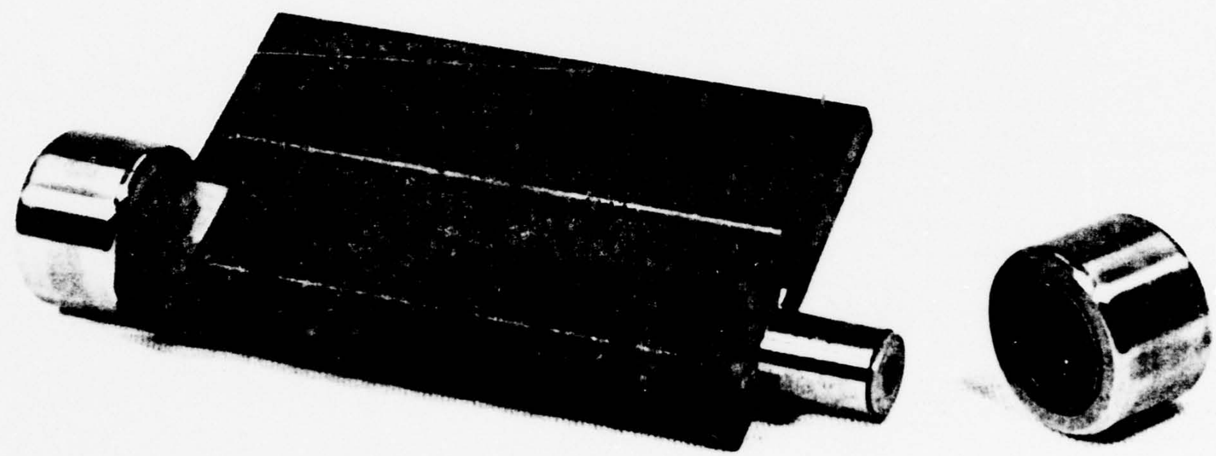


Figure 3.8 -- Model 606 Initial Vane Bearing Design

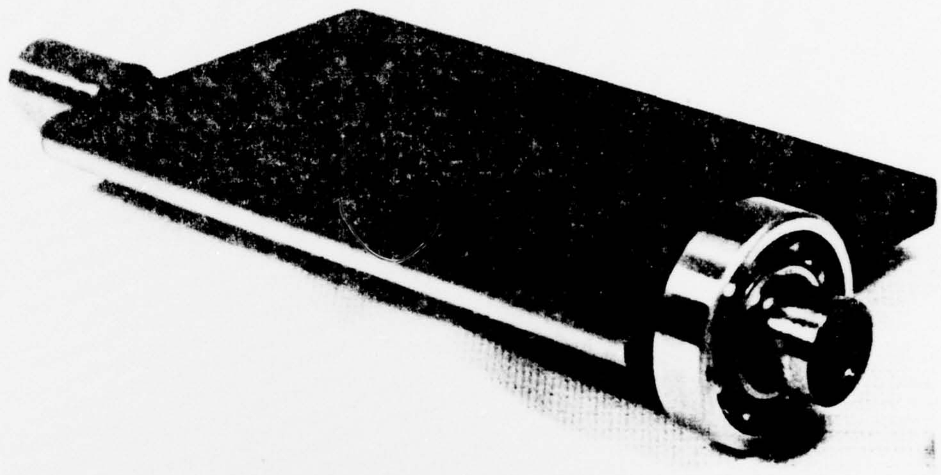


Figure 3.9 -- Model 606 Final Vane Bearing

machine performance. This is because bearing power dissipation increases with increased bearing size and load carrying capabilities. Also the larger and therefore heavier bearings create more of a load on the vane assembly thus increasing its frictional power consumption and heat generation. Additionally, the use of a long roller bearing requires that precise alignment be maintained betwen axle, bearing rim and cam track, otherwise axial loading and binding can occur. This causes additional friction heat and power consumption.

A step away from conservatism and towards a more optimized vane bearing design is the use of a short roller bearing or a ball bearing with less load carrying capacity. Therefore, in order to incorporate into the Army machine the latest technology a change was made in the 606 vane bearing design. However, due to size constraints and commercially available bearing hardware limitations a somewhat less than optimum design could be achieved.

The efficiency improved vane design which was ultimately employed in the 606 machine is shown in Figure 3.9. The bearings are Fafnir S5K ball bearings with a steel retainer. Each bearing has a basic load rating capacity of 805 pounds for 1500 hours minimum life at 33 1/3 rpm (10). This load capacity is not considered adequate for the life expectancy goal for a well-developed Circulator. However, it was the only size available (except for the long roller bearings, of course) for testing in the existing 606 internal geometry and it was felt that these bearings would be adequate to provide meaningful test data during this program.

Since these ball bearings have an outer hardened race which was ground to the required rim diameter dimension the bulky rim required with the roller bearing was eliminated. In future designs the hardened steel axle stub could be left in the annealed state because the ball bearing also has a hardened inner race. By press-fitting the ball bearings on the axle stub the nylon pin and sleeve retaining assembly is also eliminated.

During bench tests it was observed that the vane bearing (or cam follower) assemblies were not maintaining continuous contact with the cam track surfaces at elevated reference pressures (closed loop pressurization). When there is zero reference gage pressure (open cycle operation) or when low (below approximately 5 psig) reference pressures are employed this "tracking" action is maintained mainly by centrifugal forces. However at higher reference pressures these centrifugal forces are overcome by lateral friction and vane tip pressure force fields at certain areas during one revolution and the vane tends to "float". (This same phenomenon was observed in the Contractor's in-house closed cycle test machines.) This problem was corrected in the 606 machine by placing small drawn-cup needle bearings on the excess axle stub resulting from the conversion to the narrow ball bearings, as shown in Figure 3.10. These small bearings mate with a plastic contour called a vane ramp which is attached to each endcap in the cam track cavity. This design provides for a double-acting cam track in the regions where the vanes tend to "float". Thus, continuous vane tracking is insured at the higher system reference pressures.

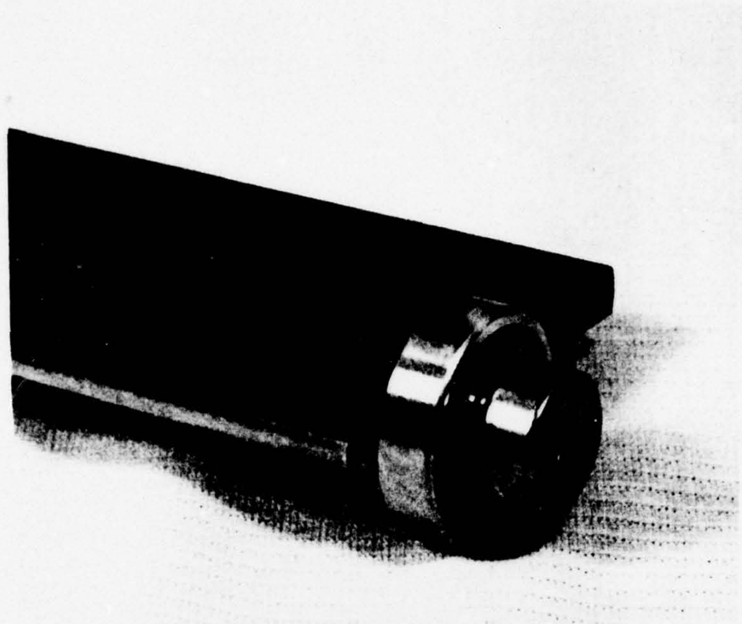
3.3 Circulator Drive Motor

The Army supplied a 220 volt, 3 phase, 5 horsepower, 1800 rpm, calibrated electric motor to the Contractor for use in the 606 system to drive the Circulator. Its calibration curves were also supplied. After conversion to closed cycle system operation it became necessary to replace the 5 horsepower motor with a calibrated 7 1/2 horsepower motor of similar design in order to be able to reach the higher reference pressures.

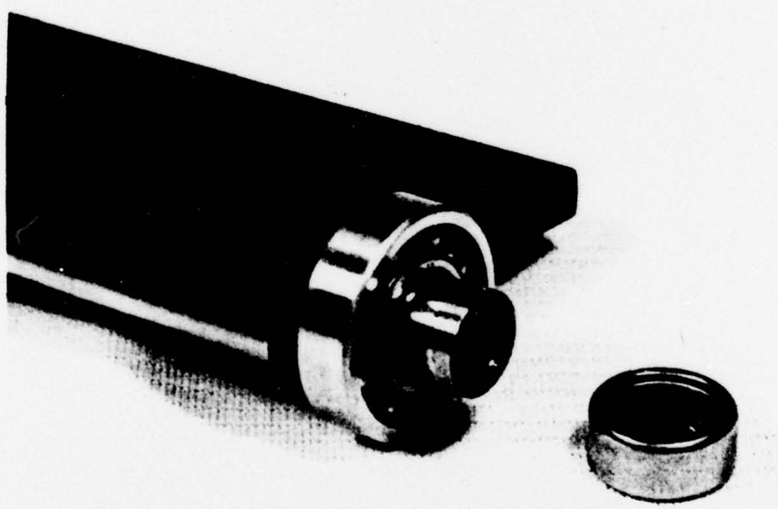
The Circulator was driven by the electric motor through a V-belt pulley arrangement. A single belt was used in this arrangement.

3.4 Heat Exchanger Designs

The primary heat exchanger was supplied by the Army. It came from the previous ROVAC 209 system which was developed under a previous contract with the Army. This heat exchanger was designed for 600 lbm/hour,



3.10a -- Assembled



3.10b -- Disassembled

Figure 3.10 -- Model 606 Positive Vane Biasing Design

300°F inlet air. With 3000 cfm, 110°F, ambient cooling air flow it was predicted to provide an effectiveness of 96 percent. Two 1/15 horsepower electric fans were used to provide the required cooling air flow.

The secondary heat exchanger was produced from two automotive vapor cycle evaporators. To minimize pressure drop to the ROVAC closed loop circuit, the two evaporators were connected in parallel. To minimize cross flow area for installation purposes with respect to the cooling load external air flow, the two cores were placed in series. This created an abnormally high static pressure drop to the external air flow. Therefore, to supply the proper amount of air flow a pressure blower was used.

The regenerative heat exchanger was construct[illegible] from two surplus aircraft oil coolers. They were manifolded together in a cross-counter-flow arrangement. This adaptation of available hardware proved to be only approximately fifty percent effective. Its presence in the system because of its pressure drop to two air flows probably did not contribute a great deal to increasing the performance as it should have done if it were a more effective regenerator.

3.5 Oil Separator

The oil separator used in the high pressure circuit of the closed system was constructed from an approximate one cubic foot volume pressure vessel. This component was added to the system after it was observed during bench testing that the presence of large amounts of oil entering the expander intake prior to expansion degrades the performance of the machine.

3.6 Controls

Since the system was a breadboard system for demonstration purposes all control of the system operation was done manually. Control of the system consisted of a main electrical power on-off switch, mechanical valves to regulate the lubricating oil flow and the water flow to the cooling jackets, and a regulator to vary the closed cycle reference pressure. Procedures for controlling the system are given in the "Test and Demonstration Report".

SECTION 4

LABORATORY DEMONSTRATION

Performance testing of the ROVAC Model 606 air cycle demonstration system to determine a coefficient of performance (COP) while operating under the Army specified environmental test conditions was conducted by the Contractor for the Army. The testing was performed at the Contractor's test facilities employing an environmental test chamber constructed and instrumented in accordance with ASHRAE standards and Army specifications. Specifics of the testing effort including test procedures and performance data evaluation are contained in the "Test and Demonstration Report" which was supplied to the Army immediately following conclusion of the testing effort (11). The following paragraphs provide a summary of the system performance.

Under the Army standard environmental test conditions of 125°F outside and 90°F db/67°F wb inside (cooling load) the ROVAC Model 606 system demonstrated a steady state COP of 0.54. The cooling capacity at this COP was 7200 Btuh. This performance compares very favorably with an overall COP on the order of 0.35 which was produced by a well-developed turbomachine equipped air cycle system under the same test conditions.

The "Test and Demonstration Report" contains calculations, based on intra-Circulator heat transfer improvements and the use of well-developed heat exchangers, that provide a sound basis for expecting a COP of greater than 1.2 (i.e: closely approaching computer predictions) for a further-developed ROVAC air cycle system at the specified operation conditions employing dry air. The presence of moisture would further increase the performance.

SECTION 5

CONCLUSIONS AND RECOMMENDATIONS

As presented in the previous sections of this report, the exploratory research and development work performed during the course of this contract resulted in the design, analysis and fabrication of a ROVAC Circulator (Model 606) and the fabrication and performance testing of a demonstration air cycle cooling system (employing the Model 606 Circulator) for the Army. Although this system employed a then state-of-the-art Circulator design and somewhat less than optimized heat exchanger designs the demonstrated COP at the Army test conditions was 0.544. This performance compares very favorably with an overall COP on the order of 0.35 which was produced by a well-developed turbomachine-equipped air cycle system under the same test conditions. Calculations, based on intra-Circulator heat transfer improvements and the use of well-developed heat exchanger, provide a sound basis for expecting a COP of greater than 1.2 for a further-developed ROVAC air cycle system at the specified operating conditions.

5.1 Conclusions

Noting the significant advancement in the performance of the 606 system as contrasted with that of the previous 209 system it is important to generate a meaningful projection of expected performance in a further-developed ROVAC Circulator and system. It has already been pointed out that the under-developed heat exchangers and ducting hardware imposed a system adversity. However, the heart of the system, the Circulator, bears the largest potential for overall system performance improvement. Detailed Company-funded research and testing on various present-generation Circulators has concluded that there are two specific areas of future Circulator design improvement:

1. Minimization of Intra-Circulator Heat Transfer
2. Maximization of High pressure Port Performance

Knowledge of the second item, high pressure port performance, has been in hand for some time, however, it was learned after fabrication of the 606 machine. While both the compressor discharge port over-pressure and the expander inlet port drop are both important, the dominant loss is the expander port. This is because a loss here counts twice: first as a reduction in expander energy recovery and then as a reduction in expander flow temperature drop. The compressor discharge over-pressure, on the other hand, only counts once: added compressor work.

It has been theoretically and experimentally determined through in-house development that the major shortcoming of the present 606-type of expander ports are couched in their variable velocity - nature. Thus, steps have since been taken to devise a constant-velocity (constant cross-sectional flow area) port geometry. A manifestation of this effort has been the so-called "Baby-Bottom" geometry that consists basically of a centrally located radially-fed inlet region that fans axially and circumferentially to maintain an approximately constant area flow path. This geometry has been tested in an in-house automotive machine (Model 217) and is showing improved performance.

Knowledge in the area of intra-Circulator heat transfer on the other hand, is relatively recent and also a result of company-funded advanced Circulator development activities. With respect to heat transfer, it has been learned that expander efficiency has been eroded by two major factors: high frequency transient heat flow from the rotor surface to the expander as well as very low frequency stator shell (and endcaps) heat conduction from the compressor section to the expander section.

The transient heat transfer arises because of the combination of the relatively high temperatures reached by the air during compression and the subsequent high film coefficient arising from the high velocity tangential air flow "scrubbing" past the rotor during the compressor discharge process. This combination forces the flow of heat into the rotor periphery during the compressor discharge process. The real problem arises, of course, when this hot rotor surface passes into the expander intake region because the relatively cool expander inlet flow again effectively "scrubs" the heat back off the rotor into the expander inlet flow.

The very low frequency transient heat flow is fundamentally a heat conduction phenomena which is aggravated by the turbulence-induction caused by the rapidly moving vane tips as they pass very near to the expander (and compressor) stator walls. This induced turbulence serves to increase the net convective film coefficient arising between the expanding air and the hot expander walls.

The first heat transfer factor is to be dealt with via rotor segment oil cooling and a very low diffusivity thin coating applied to the rotor periphery. This provides a relatively low coating substrate temperature and confines the transient rotor thermal activity to an extremely thin region at the rotor surface. In concert with this rotor configuration, a "cold scrubber plate" is to be installed at the minor axis of the stator housing on the high pressure side. This "cold plate" will provide a sink for the thermal pick-up at the rotor surface and by-pass it to the heat rejection system, thereby eliminating the short-circuiting of the thermal path. (The cold plate will employ a single cross-flow of ambient oil in order to flush away the rotor surface heat. Film coefficient on the order of 10,000-20,000 Btu/ft^2-°F-hr. can be produced across a thin film of oil between the moving rotor surface and the cold plate itself.)

The stator housing conduction flow will be stopped by an approximately 1/16" thick coating of plastic insulation applied to the endcap surfaces. This coating, due to its very low conductivity, will effectively stop stator housing thermal flow from the compressor to the expander. However, to further enhance the effectiveness of this coating, the endcaps should also be cooled by the same flow of viscosity/cooling oil passing through the rotor.

With these basic improvements, the "Test and Demonstration Report" presents calculations (based on the demonstrated 606 performance) which predicts a COP of greater than 1.2 for a further-developed ROVAC air cycle system at the Army environmental operating conditions.

Due to the many potential and demonstrated advantages of the closed cycle system over the open cycle system, such as: variable cooling capacity, increased efficiency and reliability, and reduced weight and size, it is concluded that the closed cycle system arrangement is best suited for Army applications. This has also been the conclusion the Air Force has come to regarding air cycle environmental control systems aboard military aircraft (12).

5.2 Recommendations

The exploratory development work performed under this contract, the Contractor's internally-funded development and testing programs, and ROVAC air cycle machines that have been tested by the Air Force have clearly demonstrated that the state-of-the-art ROVAC air cycle systems have performance far superior to existing well-developed turbomachine air cycle systems and are approaching the performance of well-developed fluorocarbon systems. Also, calculations contained in the "Test and Demonstration Report", based on certain realistic Circulator and system improvements, show that the ROVAC system can have performance levels meeting or exceeding that of the well-developed fluorocarbon systems. Since the ROVAC system has fewer and less complicated parts and uses only ambient air (and perhaps water) as a refrigerant it has the potential to have a lower first cost and be less costly to maintain. Therefore, it is recommended that a follow-on program be initiated to advance the development of a ROVAC non-fluorocarbon environmental control system for the Army. It is further recommended that such a program be well funded to accelerate the development of this technology in order to reach a production level as early as possible.

There are specifically four areas of recommended future development concerning the ROVAC Circulator for Army applications. These are optimization of port flow, vane bearing design, reduction of intra-Circulator heat transfer and materials selection for possible weight reduction. Due to the Contractor's internally funded development programs a number of important advancements have already been made in these areas since the completion of the Army 606 Circulator. For example, port flow has been improved by the "Baby Bottom" port design and specially designed "short-roller" vane bearings have been successfully tested in related Circulator designs.

The most important recommended future Circulator development activity is in the area of intra-Circulator heat transfer reduction. As discussed in the Conclusions section, there are several design techniques to be employed in future Circulator design to reduce the amount of heat flow from the compressor to the expander.

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