FOREWORD

This study was initiated by the Biomedical Laboratory of the 6570th Aerospace Medical Research Laboratories, Aerospace Medical Division, Wright-Patterson Air Force Base, Ohio. This study was performed in support of Project No. 6373, "Equipment for Life Support in Aerospace," and Task No. 637302, "Respiratory Support Equipment." The research was conducted by the A. J. Sawyer Co., Inc., Strykersville, New York under Contract No. AF 33(657)-9019. Mr. Charles C. Jennings was the principal investigator.

Mr. Irving H. Lantz, Respiratory Equipment Branch, Biotechnology Division, served as contract monitor for the 6570th Aerospace Medical Research Laboratories. The research sponsored by this contract was started in April 1962 and was completed in November 1963.

ABSTRACT

This report covers one type of circulating device for use in a closed circuit respiratory system. Circulation is required in a closed system to remove carbon dioxide, add oxygen, remove moisture, and cool recirculating gases.

The device is a single-stage, centrifugal-type blower capable of discharging 12 CFM of oxygen against a back pressure of 25 inches of water. It is powered by a 28-volt D.C. Motor. Power supply is a silver cell battery. Flow control is maintained electrically by a rheostat.

PUBLICATION REVIEW

This technical documentary report is approved.

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LIST OF SYMBOLS

D	Impeller diameter	in.
g	Acceleration due to gravity	32.2 ft. per sec.^{2}
H_{ad}	Adiabatic Head	ft.
$HPad$	Adiabatic horsepower	
K	Overall head coefficient	
k	Ratio of specific heats	1.394 for O_2
n	Revolutions per minute	r.p.m.
P_a	Atmospheric pressure	14.7 PSIA
p	Absolute back pressure	PSIA
Q	Volume flow rate	CFM
R	Gas constant (for O_2)	48.25 ft. lb. per deg.
T	Absolute temperature	(460+t) °F
t	Ambient temperature	°F
W_{O_2}	Weight flow of O_2	lb. per sec.

GREEK LETTERS

γ (gamma)	Specific weight of O_2	lb. per ft. 3
Δ (delta)	Small change (a prefix)	
ϵ (epsilon)	Pressure ratio	

INTRODUCTION

A centrifugal blower consists essentially of one or more impellers equipped with vanes, mounted on a rotating shaft and enclosed by a housing. Gas enters the impeller axially at the center and has energy, both kinetic and potential, imparted to it by the vanes. As the gas leaves the impeller at a high velocity, it is collected in a volute or series of diffusing passages which transforms the kinetic energy into pressure. Then the denser gas is discharged from the unit.

The application of this particular unit is for use in a closed circuit respiratory system on a "Back Pack." It will be subjected to space environment such as weightlessness and lack of atmosphere. No air will be available for forced convection cooling of the motor. Cooling by conduction will be limited to the "Back Pack" frame; hence, motor cooling must be accomplished by the circulating gas in the system. This gas is oxygen at a system pressure of 5 PSIA. Since oxygen has the potential hazards of fire and explosion, the motor must be explosionproof, with brushes suitable for altitude conditions. Also a shaft seal must be utilized to preclude the possibility of the circulating oxygen being contaminated by lubricant vapors from the motor bearings. The system parameters for the Circulating Device are:

Flow: 12 CFM oxygen
Ambient Pressure: 5 PSIA - 14.7 PSIA
Back Pressure: 25 inches of water
Continuous Operation: 40 hours
Flow Control: 4 1/2 CFM to 12 CFM
Acceleration: 0-12g
Power Source: 28 volt D.C. battery
Control: Rheostat

SECTION I

TWO-STAGE BLOWER DESIGN AND MANUFACTURE

Because the gas-circulating device is to be mounted on a "Back Pack" Life Support System, its overall envelope dimensions are of paramount importance. Of the two major dimensions, that is length and diameter, the diameter is the one dimension that is most desirable to be held at a minimum. This is to minimize the pack depth protruding behind the wearer. With this consideration in mind, it was decided to build a two-stage centrifugal blower. Though the two-stage design would necessarily yield a longer unit, the reduced diameter appeared to justify the decision.

The basic system parameters are:

Blower Type - Two-stage centrifugal
Motor Speed - 18,000 RPM
Gas Flow - 12 CFM oxygen
Ambient Pressure - 14.7 PSIA - 5 PSIA
Back Pressure - 25 in. H_2O (0.9 PSIG)

Calculating for oxygen at S. T. P.

$$\epsilon = \frac{Pa + p}{Pa} = \frac{14.7 + 0.9}{14.7} = 1.06 \qquad (1)$$

Total Adiabatic Head (H_{ad}):

$$H_{ad} = RT\left[\frac{k}{k-1}\right]\left\{\epsilon^{\left[\frac{k-1}{k}\right]} - 1\right\} \qquad (2)$$

$$H_{ad} = (46.25)\,(540)\left[\frac{1.394}{1.394-1}\right]\left\{1.06^{\left[\frac{1.394-1}{1.394}\right]} - 1\right\} = 1534 \text{ ft.}$$

Adiabatic Head per Stage (H_{ad}/stage):

$$H_{ad}/\text{stage} = \frac{1534}{2} = 767 \text{ ft.}$$

Specific Weight of Oxygen (γ):

$$\gamma = \frac{Pa}{RT} = \frac{144\,(14.7)}{48.25\,(540)} = 0.0813 \text{ lb/ft.}^3 \qquad (3)$$

Weight Flow (Wo_2):

$$Wo_2 = \frac{Q\ \gamma}{60} = \frac{12\,(0.0813)}{60} = 0.01625 \text{ lb/sec} \qquad (4)$$

Adiabatic Horsepower per Stage (HPad):

$$HPad = \frac{Wo_2\ Had/stage}{550} = \frac{(0.0162)(767)}{550} = 0.0226 \qquad (5)$$

$$\text{Total Horsepower} = 2(0.0226) = 0.0452 \text{ HP} \qquad (5a)$$

Impeller Diameter (D):

$$D = \frac{1300}{n}\sqrt{\frac{Had/stage}{K}} = \frac{1300}{17000}\sqrt{\frac{767}{0.55}} = 2.86 \text{ in.} \qquad (6)$$

(Note: 1) Motor speed with shaft seal approx. 17,000 RPM
2) Assume overall head coefficient of 0.55)

Temperature Rise (ΔT):

$$\Delta T = T \left\{ \epsilon^{\left[\frac{k-1}{k}\right]} - 1 \right\} = 540 \left\{ 1.06^{\left[\frac{1.394-1}{1.394}\right]} - 1 \right\} = 8.96^{\circ}F \qquad (7)$$

When selecting the final impeller design consideration must be given to the various losses inherent to any centrifugal machine. The main losses (refs. 1 and 2) are (1) disc friction, (2) head, (3) leakage, (4) mechanical, (5) turbulences, and (6) interstage. At this point in centrifugal blower design the engineer must rely upon his experience and the "art" that he has thereby attained in selecting the dimension of his blower impeller. For example, in this case calculations yield an impeller diameter of 2.86 inches. Because of the losses mentioned an impeller diameter of 3.50 inches was selected.

Two considerations affect the final selection of the blower motor. First is the calculated value of adiabatic horsepower. Second is motor heat rise. As noted in equation 5a, it should require 0.0452 horsepower to deliver oxygen at the required pressure and flow. With the known losses involved this value is obviously marginal. Also, since motor cooling is accomplished by passing the circulating oxygen over the motor, the motor heat is imparted to the circulating medium and contributes to the amount of heat that must be dissipated by the system. The direct relationship of heat quantity versus size and weight of heat exchanger is obvious. It is, therefore, clearly advantageous to keep the motor heat rise at a minimum. Within the limits of size and weight, it is best to use a motor with such power rating that it will actually operate at a load less than its design rating.

With these considerations in mind, a motor rated at 0.15 HP was chosen. There was no sacrifice in size incident to the selection, and the motor weight was also essentially unchanged.

With the motor and impeller selection established, the final detail design of the circulating device was accomplished. Its configuration is shown in Figure 1.

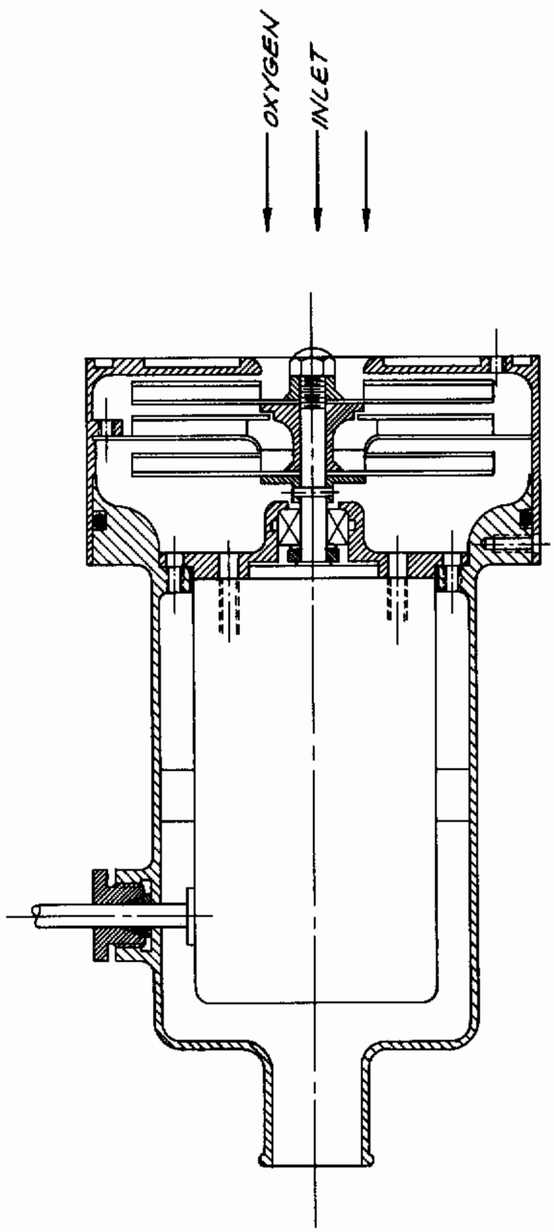

FIGURE NO. 1: TWO-STAGE CENTRIFUGAL CIRCULATING DEVICE

SECTION II

TESTING AND EVALUATION

When manufacture was completed, tests of pressure versus flow were conducted. The results are tabulated in Table 1. It can be seen that the device delivered 1.4 cfm and 9 inches H_2O less than required. A check on all calculations showed that the sizes and power rating selected should perform as desired; however, satisfactory results were not attained.

TABLE 1

TEST RESULTS FOR TWO-STAGE CONTRIFUGAL CIRCULATING DEVICE

MODEL	FLOW(cfm)	PRESSURE In. H_2O
Original Design	10.6	16
Model "A"	10.8	32 *
Model "A" with Diffuser	10.6	17.2
Model "A" with reworked Diffuser	10.8	19.0

* Pressure taken at periphery of second stage impeller

Finally, the basic design itself was questioned. First, the clearance between the inlet eye and the first stage impeller were reduced to an absolute minimum, as was the clearance between the back of the first stage impeller and the inlet to the second stage. Virtually no change in performance was noted.

Next, a pressure reading was taken at the periphery of the second stage impeller. At this point, a pressure of 32 inches of water was obtained, indicating that the impeller section was performing as originally planned. A pressure reading taken at the shaft-end of the motor showed a pressure of only 18 inches of H_2O. This indicated that the diffuser section was not functioning. As shown in Figure 1 there is a void between the second stage impeller and the motor. It was concluded that any diffusion action that might be obtained from the unit was destroyed in this region; therefore, a cone was made to fill this void. Essentially, it had a profile which paralleled the internal profile of the the outside housing. This was the first of numerous attempts to provide correct diffusion to the device. It was possible to "move" the high pressure point of 32 in. H_2O from the periphery of the second stage impeller down onto the diffuser itself, but regardless of design the pressure drop between diffuser and outlet was such that a pressure greater than 19 in. H_2O at the outlet was never attained. Also, at no time was the full 12-cfm flow attained. It was concluded that the entire region from second stage impeller to outlet had to function as a diffuser for correct flows and pressures to obtain. An original premise of this design was that this region would serve somewhat as a plenum chamber with minimal losses across its length. Thus its true function as a diffuser was not originally appreciated.

Based upon the above conclusion, it was considered not beneficial to pursue the two-stage design further. This decision was reinforced by the knowledge that the overall device diameter would have exceeded the diameter of a conventional single-stage centrifugal blower with the same rating. Also, being a two-stage device, the unit would inherently be heavier, however slight the weight penalty might be. Since the entire exhaust section from the second-stage impeller to the outlet was in fact a diffuser, it was implicit that this section had to be an annular diffuser.

An annular diffuser for a centrifugal blower has an outside diameter generally 150-180% greater than the impeller diameter. In this case the O.D. would be at least 5.25 inches, not counting the diffuser housing thickness. This would negate all advantages the two-stage unit originally offered. (Note: A guide-vane diffuser is feasible, but the complexity of guide vanes plus the added weight precluded this approach as a practical solution.)

SECTION III

SINGLE-STAGE BLOWER DESIGN AND MANUFACTURE

To determine the impeller diameter for a single-stage impeller equation 6 is used. The value for H_{ad} derived from equation 2 is used for this calculation.

Single Stage Impeller Diameter:

$$D = \frac{1300}{17000} \sqrt{\frac{1534}{.55}} = 4.05 \text{ in.}$$

Because the original 2-stage impeller section produced 7 in. H_2O pressure over the specification it was decided that a slight increase in dimension over and above that calculated was justified. Therefore, an impeller diameter of 4.125 inches was selected.

Figure 2 is a cross-sectional view of the device. Oxygen gas is drawn in axially over the motor as it is routed to the eye of the impeller. The gas is discharged via a conventional volute scroll, with a discharge nozzle at 90° to the flow path.

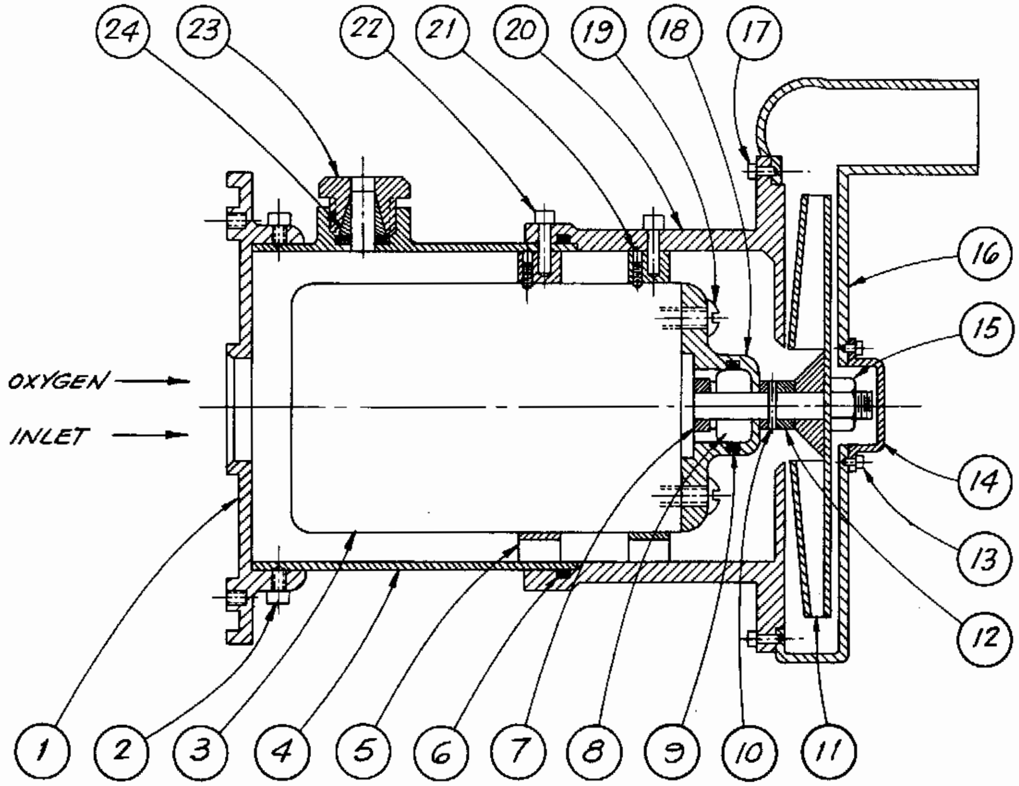

FIGURE NO. 2: SINGLE-STAGE CENTRIFUGAL CIRCULATING DEVICE

ITEM	PART NAME
24	COLLET
23	NUT
22	SCREW, CAP
21	SCREW, SET
20	END HOUSING
19	SCREW
18	FLANGE
17	SCREW, CAP
16	SCROLL
15	NUT
14	CAP
13	SCREW, CAP
12	FLANGE
11	IMPELLER ASS'Y.
10	PIN
9	"O" RING
8	SHAFT SEAL
7	RING
6	"O" RING
5	CLAMP ASS'Y.
4	SLEEVE
3	MOTOR
2	SCREW, CAP
1	END COVER
	BLOWER ASS'Y.

SECTION IV

TESTING AND EVALUATION

A circulating device was designed and built in accordance with Figure 2. Its performance is shown in Table 2. This device met all the performance requirements as originally specified with the exception that its weight was 3 lb. 12 oz.

Table 2 shows that the total temperature rise for Serial No. X-1 was 35°F after 30 minutes of continuous operation. Serial No. X-2 has essentially the same characteristics. From equation 7 the theoretical adiabatic temperature rise for the gas itself is approximately 9°F. Generally a 65 - 75% adiabatic efficiency is normal. Therefore the actual adiabatic ΔT is between 12°F and 14°F, thus the increase in temperature of the outlet gas due to the motor is on the order of 20°F.

TABLE 2

TEST RESULTS FOR PROTOTYPE SINGLE-STAGE CENTRIFUGAL CIRCULATING DEVICE

Serial No. X-1

ELAPSED TIME (Min.)	FLOW (CFM)	PRESSURE (In. H_2O)	AMB. TEMP. (°F)	FRAME TEMP. (°F)	OUTLET TEMP. (°F)
Start	12	26.2	80	80	94
5	12	26.2	83	106	104
10	12	26.2	83	112	114
20	12	26.2	82	115	115
30	12	26.2	82	118	117

Serial No. X-2

ELAPSED TIME (Min.)	FLOW (CFM)	PRESSURE (In. H_2O)	AMB. TEMP. (°F)	FRAME TEMP. (°F)	OUTLET TEMP. (°F)
Start	12	27.2	83	83	90
5	12	27.2	84	109	110
10	12	27.2	84	112	114
20	12	27.2	83	114	116
30	12	27.2	84	114	117

SECTION V

FINAL DESIGN AND MANUFACTURE OF SINGLE-STAGE BLOWER

With the performance success of the single-stage blower a new case and scroll was designed. This new case is of thin-walled, one-piece construction. Because of the one-piece construction of the case, a new motor mounting system had to be developed. Basically, it is a tripod spider and is attached to the motor with set screws. The motor and spider assembly is then secured to the scroll with long screws. The case is separately attached to the scroll. Since motor loading is not directly applied to the case, the thin-walled design is feasible.

The scroll is essentially unchanged, except that its walls were made as thin as possible consistent with structural strength. The new configuration conforms to Figure 3.

The new design has slightly superior performance and provides an appreciable weight reduction. This final unit weighed 2 lb. 15 1/4 oz.

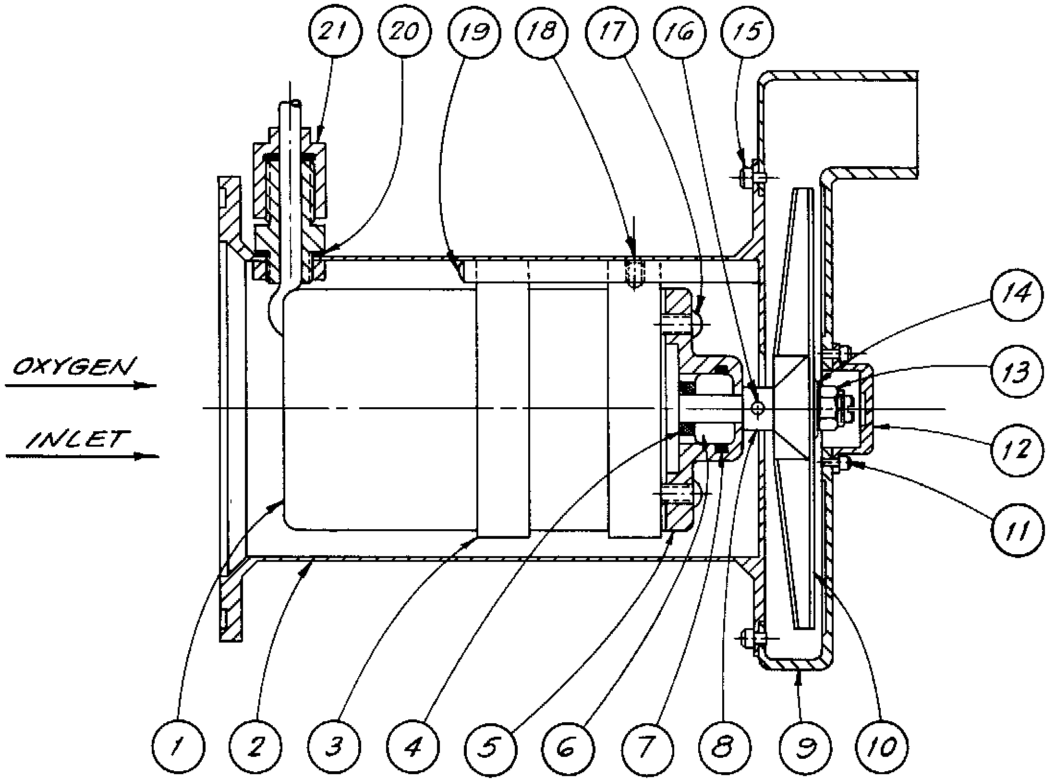

FIGURE NO. 3: SINGLE-STAGE CENTRIFUGAL CIRCULATION DEVICE
(PRODUCTION CONFIGURATION)

ITEM	PART NAME
21	MOTOR LEAD SEAL ASS'Y.
20	"O" RING
19	SCREW
18	SCREW, SET
17	SCREW
16	PIN
15	SCREW, CAP
14	WASHER
13	NUT
12	CAP
11	SCREW, CAP
10	IMPELLER
9	SCROLL
8	FLANGE, RETAINING
7	"O" RING
6	SEAL
5	FLANGE
4	RING
3	CLAMP ASS'Y.
2	HOUSING ASS'Y.
1	MOTOR
	BLOWER ASS'Y.

SECTION VI

FINAL TEST RESULTS AND ANALYSIS

The performance of this final design is noted in Table 3. In this table, complete flow, pressure, and temperature data is recorded, including the ambient conditions at the time of test.

It is interesting to note the inverse relationship that exists between flow rate and temperature. For example, at a flow rate of 3.5 CFM the inlet temperature is 99°F, whereas at a flow rate of 12 CFM (the design point) the inlet temperature is 88°F. In both cases the ΔT between inlet and outlet is essentially the same. This seeming paradox, that is, low flow related to high temperature, and vice versa, points up the major role that forced convection cooling plays in this type system.

A factor that should be considered in the design of closed-circuit system is emphasized by this situation. That is, if a closed-circuit system has flow control capability it is imperative that the heat exchanger be so designed that it is capable of removing sufficient heat under the worst possible conditions of temperature, humidity and flow. Obviously, there are many other factors, some of them interrelated to this one, that must be considered, but this is one not to be overlooked. By inference then, the characteristics of each system component must be carefully investigated under all expected conditions of temperature, humidity and flow, before the maximum total heat load of the system may be accurately determined.

TABLE 3

FINAL TEST RESULTS FOR SINGLE-STAGE CENTRIFUGAL CIRCULATING DEVICE

Serial No. X-1

FLOW (CFM)	PRESSURE (In. H_2O)	TEMPERATURE (°F) Inlet (compressor)	Body	Scroll	Outlet
3.5	36.0	96	97	130	136
7.1	32.1	88	94	122	126
10.6	30.0	84	93	117	121
12.0	29.2	84	92	114	119
14.2	27.0	83	91	110	116

Serial No. X-2

FLOW (CFM)	PRESSURE (In. H_2O)	TEMPERATURE (°F) Inlet (compressor)	Body	Scroll	Outlet
3.5	37.8	99	104	125	130
7.1	34.0	95	98	120	125
10.6	30.6	92	97	117	122
12.0	29.6	88	96	109	120
14.2	26.7	86	94	106	116

CONDITIONS AT TIME OF TEST

Barometric Pressure - 29.5 In. Hg
Ambient Temperature - 82°F
Relative Humidity - 89%

SECTION VII

SUPPORT EQUIPMENT

In support of the circulating device is a silver cell battery system and a control rheostat.

The battery is a special design SilverCel © consisting of 19 individual cells. Its overall performance is:

A. 28 Volts, D.C.

B. 8.5 Amperes discharge rate

C. 40 hours operation at 8.5 amperes discharge rate

D. Approximately 50 charge - discharge cycles

E. Operating Life - 1 to 1 1/2 years

The control rheostat is a conventional commercial model mounted into a metal portable case. The portable case also contains a power control switch and a mating receptacle to the circulating device power lead. All system components may be seen in Figure 3.

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SECTION VIII

SUMMARY, CONCLUSIONS AND RECOMMENDATIONS

The final design selected, that is a single-stage centrifugal blower with an axial flow inlet, and a volute scroll with a 90° axial discharge outlet elbow, performs as required by the specification. The final configuration is a reliable design that will give safe service when in contact with oxygen. The only maintenance required is periodic brush changes, such maintenance being inherent with brush-type D.C. motors.

The above device, though designed for space environments, is a "laboratory-type" unit. Also, as an R & D prototype, it is virtually "hand-made," and could be refined should production quantities be required. These refinements fall into two general categories: (1) minor changes, and (2) major changes.

The minor changes generally are associated with careful selection of materials to gain the maximum weight savings possible consistent with reliable operation. Also, assembly details might be altered in production designs to yield a better overall configuration. These changes are ones that generally evolve when converting from prototype to production configurations.

There is one major refinement that might yield a significantly superior unit for actual space applications:

In common with all conventional D.C. motors, this unit has brushes that must be changed periodically. For aerospace application, the unit must be very carefully sealed with epoxy resin after the change to preclude any leakage or the possibility of igniting the system by electrical arcing. Thus, if the brushes could be eliminated a more maintenance free and safe design would result.

Work has been done recently on brushless D.C. motors that could eliminate this weakness. It is felt that a 28-volt, 400 cps, single-phase motor rotating at 22,000 rpm could be used in conjunction with a specially designed solid-state D.C. to A.C. converter. The result would be beneficial from two aspects: (1) The device would be maintenance free, and (2) With the 22,000 rpm motor, a smaller-diameter impeller could be used, reducing the overall envelope dimension of the device.

If a life support system requiring a circulating device is ultimately selected for the Air Force, it is recommended that a brushless, D.C. motor be investigated.

REFERENCES

1. Berry, C. Harold, Flow and Fan, The Industrial Press, New York 13, New York, pp 41, 43, 63, 99, 1954.

2. Church, Austin H., Centrifugal Pumps and Blowers, John Wiley & Sons, Ltd., London, pp 11, 29-31, 40, 1953.

3. Compressed Air and Gas Institute, Compressed Air Handbook, McGraw-Hill Book Company, Inc., New York, 1954.