UNCLASSIFIED

R. Maitland

DC 58-1-89

MASTER

DC 58-1-89

REPORT ON AIR MOTOR

F. W. Schnorr

CONTROL MECHANISM DEVELOPMENT

"PRELIMINARY REPORT"

This report is preliminary and informal in nature and was prepared for use at the Aircraft Nuclear Propulsion Department, General Electric Company in the course of work under AEC contract AT(11-1)-171, U. S. Air Force contract AF33(038)-21102, or U. S. Air Force contract AF33(600)-38062. Views, opinions, conclusions or proposals expressed in the report are those of the author(s) only. This report is subject to revision upon further evaluation or availability of additional data.

January 8, 1958

GENERAL ELECTRIC-AIRCRAFT NUCLEAR PROPULSION DEPARTMENT-CINCINNATI 15, OHIO

UNCLASSIFIED

DISCLAIMER

CONTROL MECHANISM DEVELOPMENT UNIT

TASK 1213

DESIGN OF A HIGH TEMPERATURE AIR MOTOR

Author: Fred W. Schnorr Jr.
Fred W. Schnorr, Jr.
Control Mechanism Development

Approved: John C. Blake
John C. Blake, Supervisor
Control Mechanism Development

JANUARY 8, 1958

DISTRIBUTION

K. W. Arnold
H. E. Bell
J. C. Blake
J. R. Colston
T. A. DeRosier
J. L. DeWeese
G. E. Gorker
N. Haberman
R. T. Hardin
N. W. Hart
M. C. Leverett
L. E. MacFadden
R. Maitland
L. H. McCarty
J. M. Messingschlager
S. H. Proffitt
K. B. Rave
F. W. Schnorr
A. L. Spivak
R. E. Tallman
J. W. Tenhundfeld
P. R. Webb (3)
L. C. White
Idaho Library
Document Control (3)

AUTHOR – COMPONENT	SUBJECT CATEGORY	DOC. NO. DC 58-1-89
F. W. Schnorr - Control Mechanism Development	Technical	DATE 1/8/58
TITLE Design of a High Temperature Air Motor		CLASS. Unclassified
		NO. PAGES 45

ABSTRACT

ANP-608 (8-56)

This report covers the basic design and development concepts of a six vane air motor intended for use at elevated temperatures. (ANP Patent Docket #24D-A-46, submitted by inventors J. C. Blake and F. W. Schnorr, Jr. on November 7, 1956)

723 03

I. INTRODUCTION

As the temperature restrictions on control system components increased, the need for a new power source for actuators becomes apparent. This power source must withstand high temperatures, be economical to operate, and be light in weight. These restrictions led to the choice of an air motor. The availability of compressed air from the gas turbine compressor removed the need for auxiliary power units. The use of graphite and stainless steel materials allows the motor to be operated to temperatures of 1000°F. The availability of better bearing materials will allow even higher operating temperatures.

The development of such an air motor is contained herein.

II. REFERENCES

A. Refer to Appendix A for a complete description of the motor as contained in the patent application.

B. Refer to Appendix B for the physical dimensions of motor prototype number 2.

C. Refer to DC 57-11-76 for an "Analysis of Vane-Type Air Motors" by B. Kaplan.

III. DESIGN

The original conception of this motor was to drive a ball screw type actuator. The ball screw was to be contained within the rotor shaft. Refer to accompanying sketch for details of assembly.

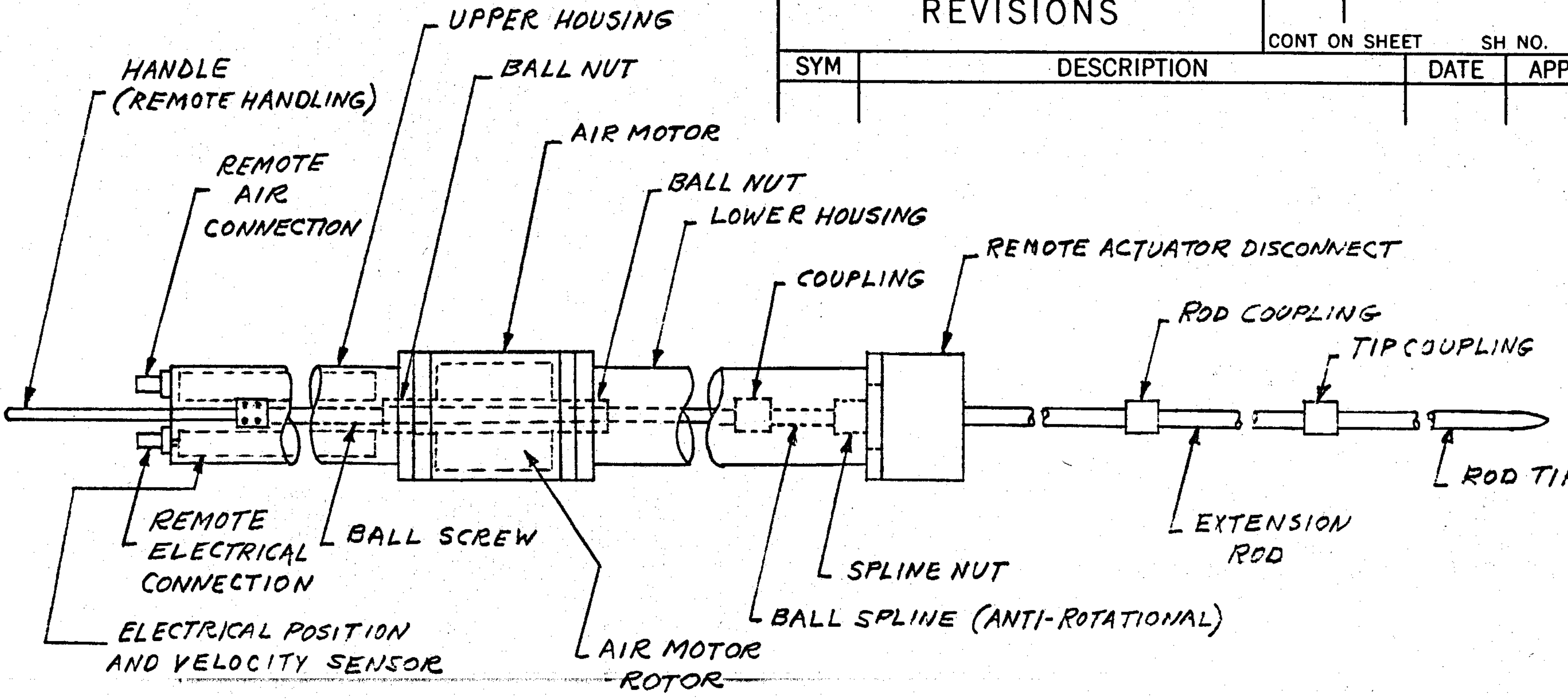
A. Description

The air motor is a rotor type motor with six sliding vanes (see Figure I). The vanes are held against the outer housing by means of air pressure, thus forming six individual power chambers per revolution. The motor is a two cylinder double opposed cam design using secondary air expansion for greater efficiency. A primary and secondary exhaust port removes the necessity of expending energy to exhaust air from the cylinders.

B. Construction

The prototype motors are fabricated from stainless steel and graphite. The motor case and end bells are made of 347 stainless steel, the rotor is from 309 stainless steel. The lesser coefficient of expansion of 309 stainless steel over 347 stainless steel insures the opening of motor clearances as operating temperature increases. Graphite was

REVISIONS				SIZE	REV
				CONT ON SHEET	SH NO.
SYM	DESCRIPTION	DATE	APPROVAL		



723 06

PRINTS TO

UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN INCHES
TOLERANCES ON:
FRACTIONS ± DECIMALS ± ANGLES ±
ALL SURFACES ✓

MATERIAL
———

DRAWN Schnorr
DATE 6-9-56
APPD Fuvl
ISSUED

APPROVED	DATE
ENGR.	
MFG.	
MATL.	

DESIGN STUDY OF AIR MOTOR DRIVE FOR BALL SCREW ACTUATOR

SCALE
WT CALC ACTUAL

GENERAL ELECTRIC
ANPD
DIV OR DEPT
EVENDALE
LOCATION

SIZE A
SKETCH

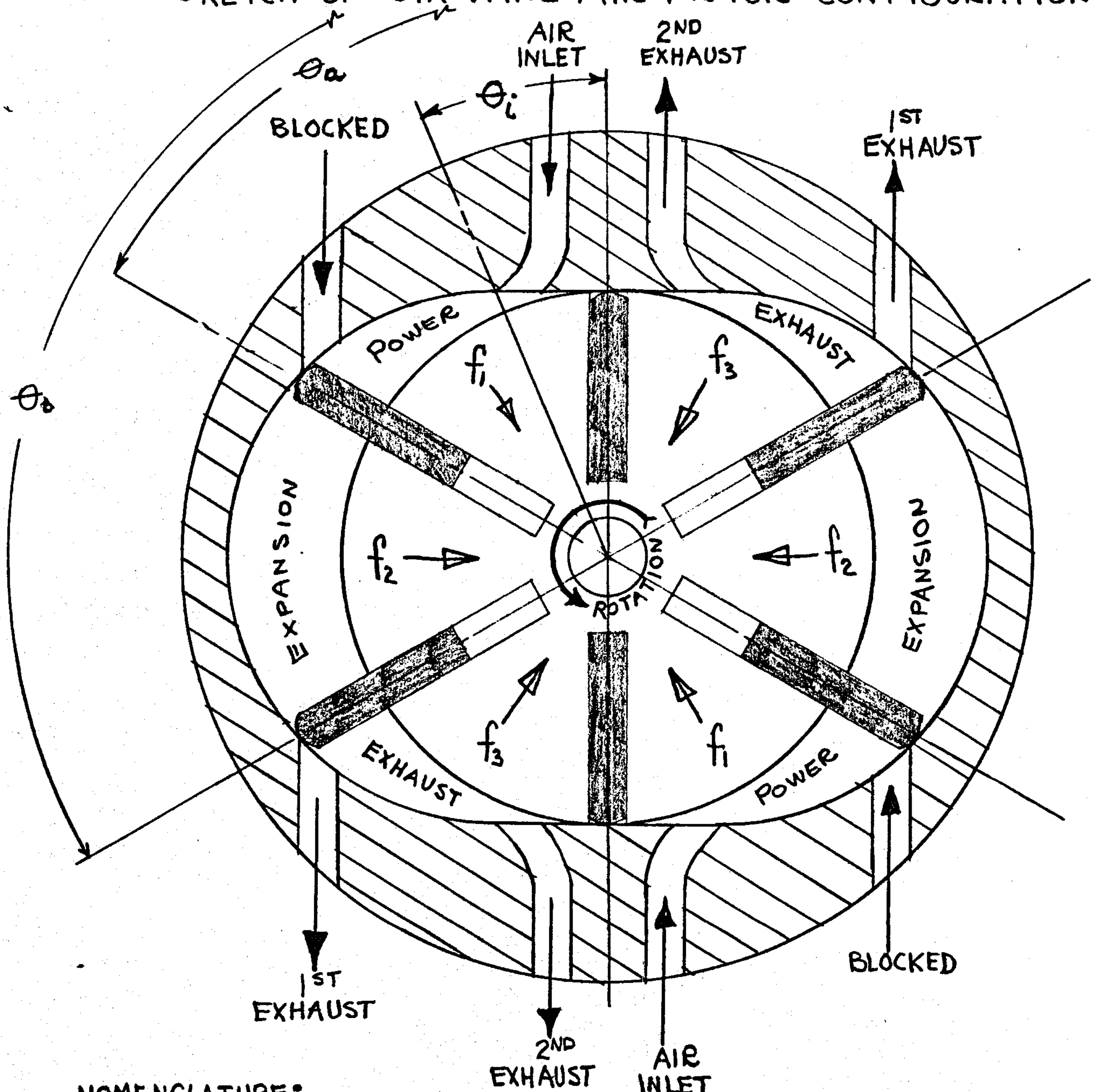
CONT ON SHEET SH NO.

-9-

PP-750 7500 (1-54) rev. PRINTED IN U.S.A.

Fig. 1

Sketch of Six Vane Air Motor Configuration



NOMENCLATURE:

θ_i = ANGLE OF TRAILING EDGE OF INLET PORT (CONTROLS EXPANSION)

f_1 = FORCE COMPONENT DUE TO INLET PRESS.

f_2 = FORCE COMPONENT DUE TO SECONDARY PRESS.

f_3 = FORCE COMPONENT OF EXHAUST AIR.

NOTE: MOTOR HAS BALANCED INTERNAL FORCES.
MOTOR HAS BI-DIRECTIONAL OPERATION.
VANES ARE SUPPORTED BY AIR PRESS.

θ_a & θ_b = VANE ANGLES WHEN EXHAUST PORT OPENS SCHNORR chosen as the bearing, vane, and thrust plate material to insure that no metal-to-metal contact would be possible at elevated temperatures. L-56-HT carbon graphite (Pure Carbon Co.) has a temperature limit of 1000°F in an oxidizing atmosphere.

IV. DEVELOPMENT

A. Prototype Motor #1

The first motor built from the basic design data had a $2\frac{1}{4}$ inch diameter rotor with an oval case offset $\frac{1}{4}$ inch maximum at each end. The bearings ($\frac{1}{2}$ inch long), vanes and thrust plates were fabricated from common reactor grade graphite. The shaft diameter was set at 1.1075 inches so that the ball screw would pass through the center (refer to rotor print, Appendix B).

The completed motor was operated for a 16 hour break-in period with water injection for lubrication. The use of water relieved the high coefficient of friction of the unpolished graphite, but proved detrimental to the operation. Graphite powder mixing with the water formed a gummy deposit on all interior surfaces of the motor, and was of sufficient quantity to completely stall this motor. The complete motor was vapor-degreased and then tested.

Test Data from Prototype #1

(Room Temperature - Dry)

	1	2	3	4
Motor Pressure (psig)	40	60	80	90
Vane Pressure (psig)	9	14.5	13	12
Air Flow (SCFM)	6	7.5	8.5	10
Speed (RPM)	1230	1700	2200	2400
Load (Pounds)	None	None	None	None

This data indicates the motor was performing well, but there is a large amount of power lost in overcoming internal friction. This is apparent from the low speeds and high inlet pressures. This fact became apparent when Prototype Motor #2 was assembled and built.

B. Prototype Motor #2

The original motor case and rotor was reassembled using L-56-HT carbon graphite, supplied by the Pure Carbon Co., as bearings, vanes, and thrust plates. L-56-HT is suitable for use in reducing atmospheres up to 1000°F.

This motor configuration was operated for a twenty hour break-in period to obtain a high polish on graphite surfaces to reduce friction.

Prototype #2 was then completely tested. One set of test data will show improved operation:

Test Data of Prototype #2

(Room Temperature - Dry)

No.	Line Pressure	Vane Pressure	Speed	Load
1	10 psig	3 psig	1300	None
2	20 psig	8 psig	2030	None
3	30 psig	10 psig	2550	None
4	40 psig	14 psig	2980	None
5	50 psig	15 psig	3250	None
6	54 psig	16 psig	3330	None

The above data shows a more than double speed increase due to the better grade of graphite used in assembly.

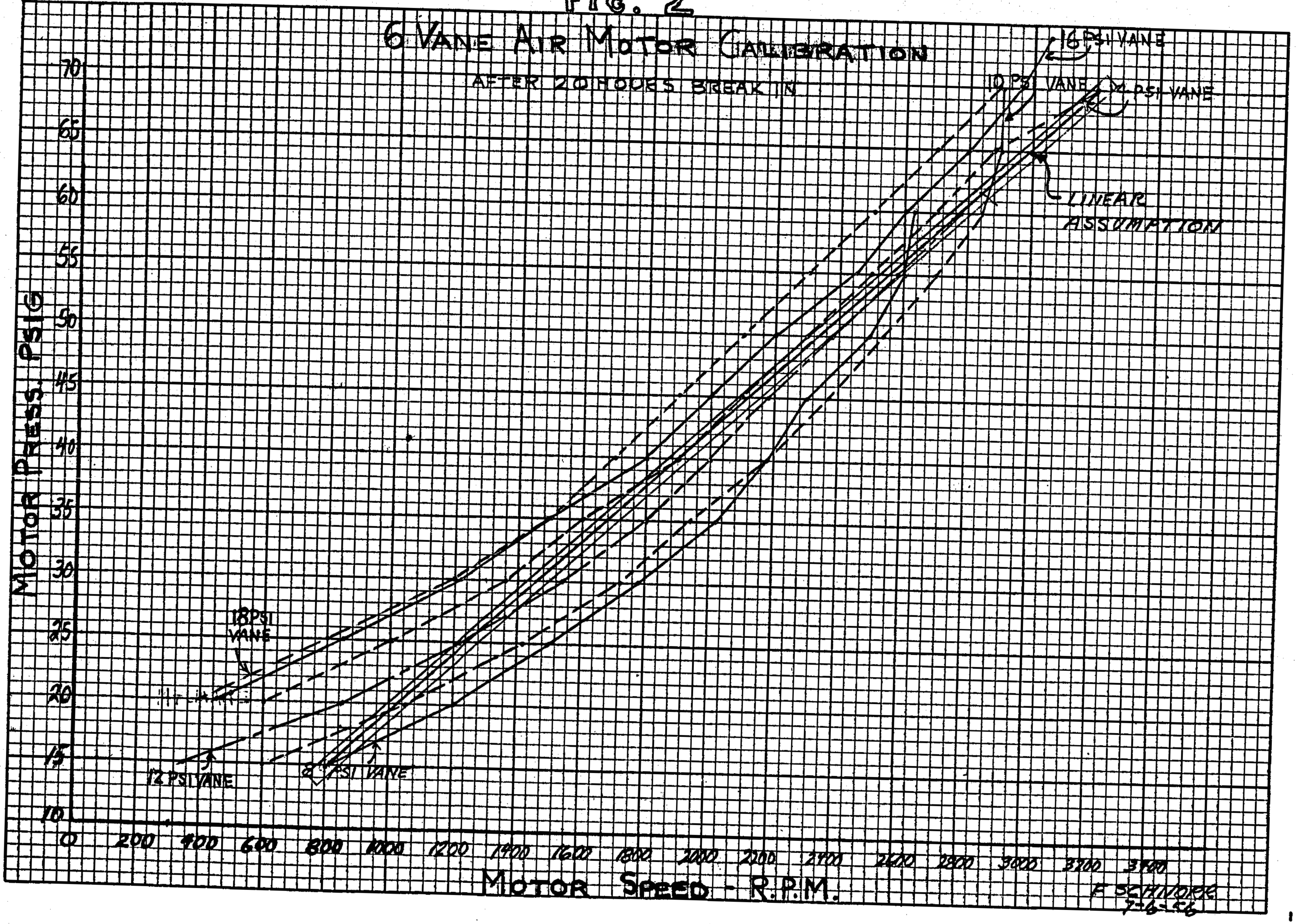
Prototype #1 @ 40 psi = 1230 RPM

Prototype #2 @ 40 psi = 2980 RPM

(Efficiency Increase) $N = \frac{2980}{1230} = 243\%$ increase

Refer to Figure II for the complete test results of Prototype #2. This is a plot of motor pressure vs. motor speed for various values of vane loading pressure. The non-linearity of these curves is indicative that vane loading pressure is an important motor parameter. Figure 2 data was cross plotted on Figure III to observe the effect of vane pressure on motor

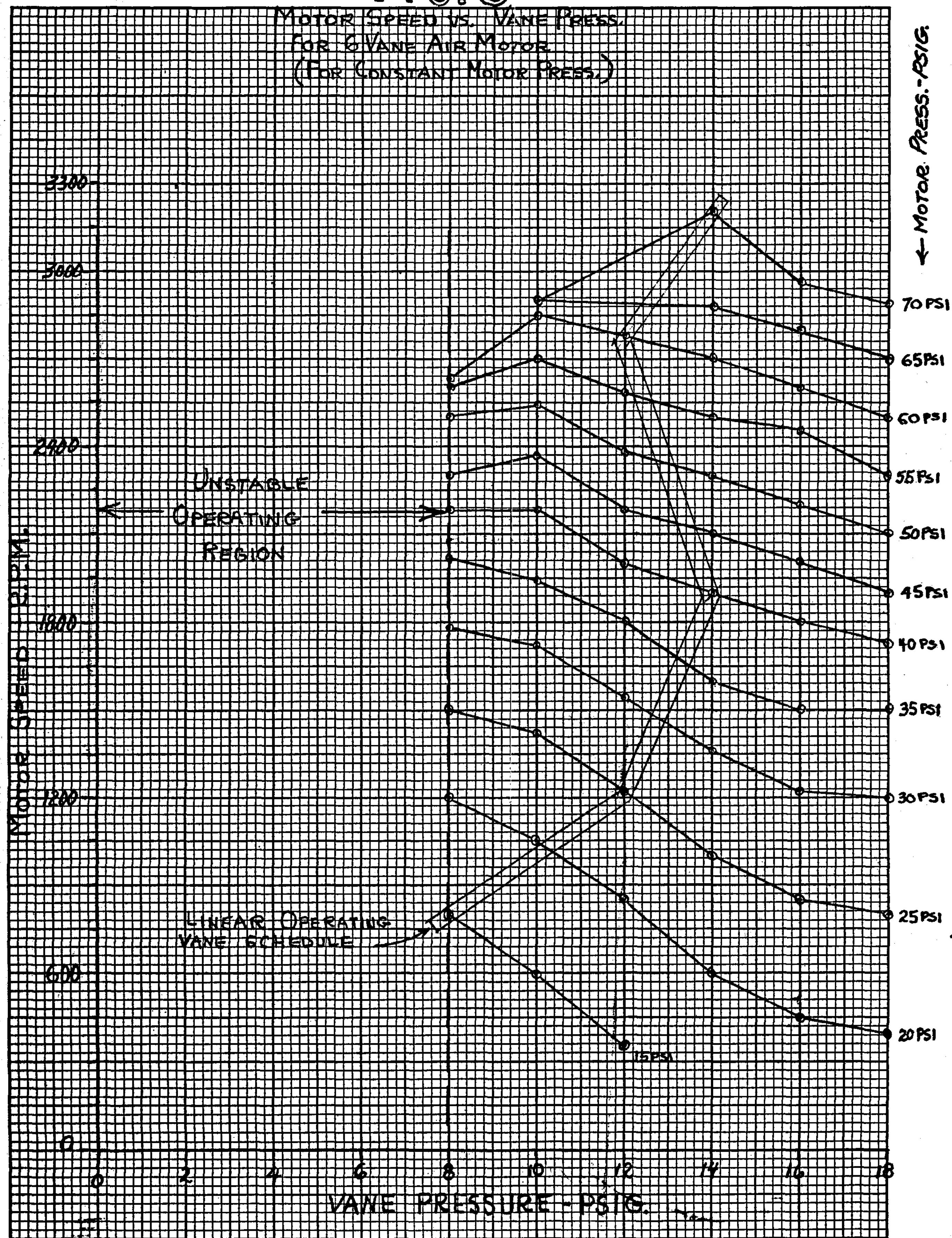
Fig. 2



6 VANE AIR MOTOR CALIBRATION
AFTER 20 HOURS BREAK IN
MOTOR PRESS. PSIG
10
15
20
25
30
35
40
45
50
55
60
65
70
MOTOR SPEED - R.P.M.
0
200
400
600
800
1000
1200
1400
1600
1800
2000
2200
2400
2600
2800
3000
3200
3400
16 PSI VANE
10 PSI VANE
14 PSI VANE
LINEAR ASSUMPTION
18 PSI VANE
12 PSI VANE
8 PSI VANE
F SCHNORR
7-6-56

723 11

Fig. 3



SCHNORR

speed, for a constant motor inlet pressure, which shows a definite need for controlling this parameter. The linearized assumption made on Figure II was transferred to Figure III and resulted in a definite non-linear schedule for vane pressure.

A comparison of Prototypes #1 and #2 is made graphically on Figure IV.

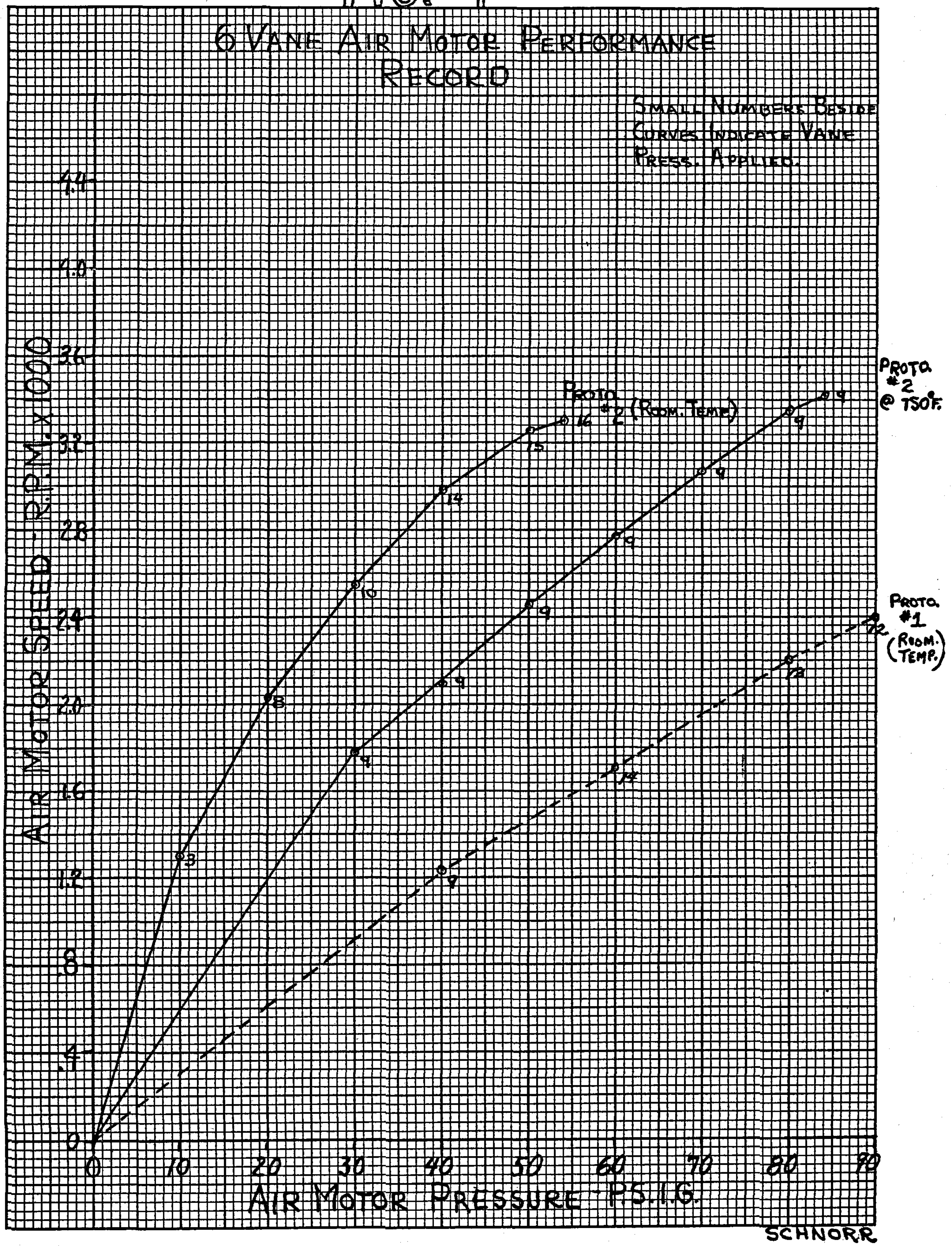
C. Furnace Testing

Prototype Motor #2 was put on life test in furnace at 750°F. The test parameters are as follows:

1. Furnace temperature - 740°F
2. Inlet air temperature - 490°F
3. Motor bearing temperature - 670°F and 680°F
4. Motor case temperature - 650°F
5. Motor pressure - 40 psig
6. Motor vane pressure - 9 psig
7. Motor speed - 2000 RPM

Testing was concluded at 120 hours because of an unforeseen equipment failure. The failure occurred when a threaded tube fitting vibrated loose from the motor case. The missing air supply resulted in a pressure unbalance of 52 pounds on the bearings supporting the rotor.

FIG. 4


6 VANE AIR MOTOR PERFORMANCE RECORD
SMALL NUMBERS BESIDE CURVES INDICATE VANE PRESS. APPLIED.
AIR MOTOR SPEED - R.P.M. x 1000
0
.4
.8
1.2
1.6
2.0
2.4
2.8
3.2
3.6
4.0
4.4
AIR MOTOR PRESSURE - P.S.I.G.
0
10
20
30
40
50
60
70
80
90
PROTO. #2 (ROOM. TEMP.)
PROTO. #2 @ 750°F.
PROTO. #1 (ROOM. TEMP.)
3
8
10
14
15
16
9
12
SCHNORR

This force shifted the end bells by an amount equal to the locating pin clearance at 750°F (0.0010"), which added to the bearing wear of 0.0023" (out of round) allowed interference between rotor and case at the seal point. The interference caused galling and scoring of the rotor and case which broke blades and stalled the motor. The normal rotor clearance at seal point is 0.0025" to 0.0030".

D. 28 Hour Test Data

The following data w... taken after 28 hours of running at temperature to stabilize the motor operation:

Furnace Temperature - 740°F

Inlet Air Temperature - 500°F

Supply Pressure	Vane Pressure	Motor Speed
30 psig	9 psig	1775 RPM
40 psig	9 psig	2090 RPM
50 psig	9 psig	2450 RPM
60 psig	9 psig	2775 RPM
70 psig	9 psig	3075 RPM
80 psig	9 psig	3350 RPM
84 psig	9 psig	3420 RPM

A graph of the 28 hour data is on Figure 5, and the complete speed vs. time curve for all 120 hours is shown on Figure 6.

E. 120 Hour Inspection Data

After the failure occurred the motor was taken to Inspection and the following wear noted:

1. Vanes: 0.0001 in/hr (average wear between vane and case)

   0.00001 in/hr (average wear between vane and rotor slot)

   Estimated vane life: 1550 hours (maximum)

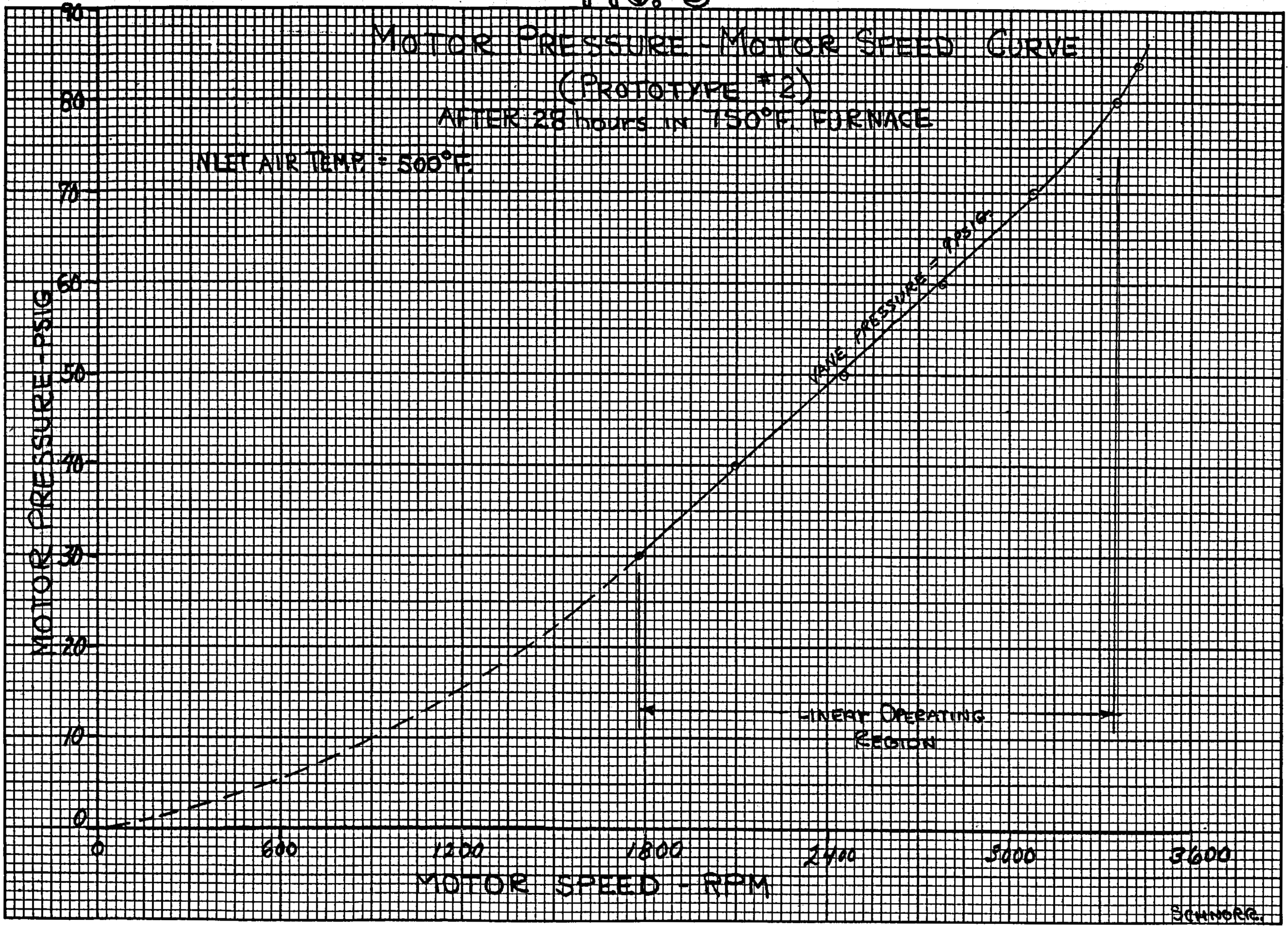
2. Bearings: (designed for 500°F, limited to 800°F)

   The rotor bearings were loaded with a composite load consisting of:

   a. 2.4 pounds rotor weight (balanced)
   b. 4.5 ounces external shaft load (one end) of which 1.5 ounces was eccentric from centerline of motor by 0.040 inch.

The resultant load was pulsating at a frequency equal to one half of the motor speed (1000 cycles/minute). The load summation varied from 0.747 pound to 1.363 pounds on the front bearing (Bearing "A"), and from 0.852 pound to 2.432 pounds on the rear bearing (Bearing "B"). The increased load on the rear bearing resulted from lever arm amplification by the rotor shaft ($2\frac{1}{2}$ inches).

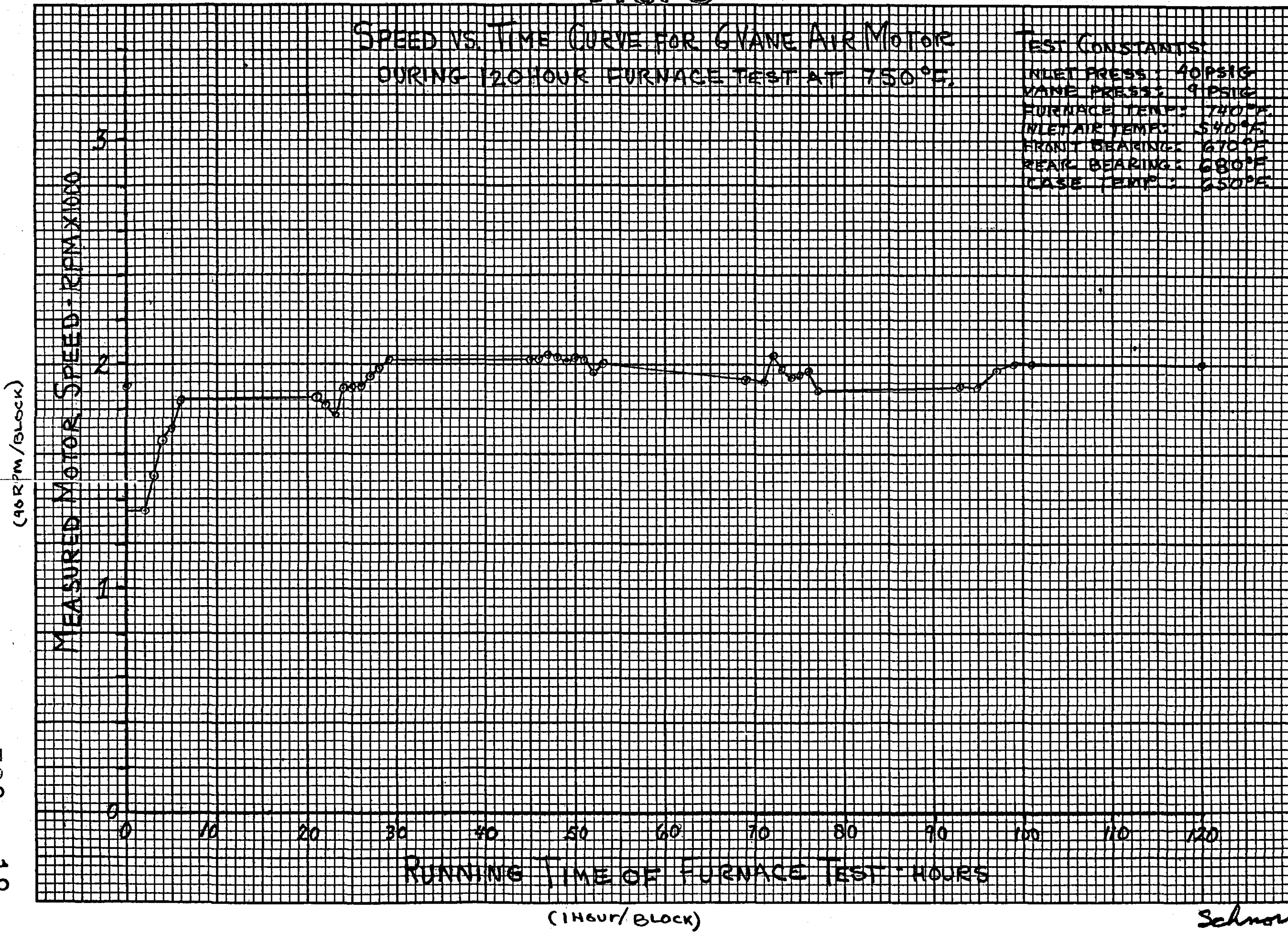
Fig. 5



MOTOR PRESSURE - MOTOR SPEED CURVE
(PROTOTYPE #2)
AFTER 28 hours in 750°F FURNACE
INLET AIR TEMP. = 500°F.
VANE PRESSURE = 7 PSIG.
INERT OPERATING REGION
MOTOR PRESSURE - PSIG
MOTOR SPEED - RPM
0
10
20
30
40
50
60
70
80
90
0
600
1200
1800
2400
3000
3600
SCHNORR.

723 17

Fig. 6



Schnorr

723 18

Bearing Load Differential: "A" - 0.616 pound

"B" - 1.580 pounds

$\frac{\text{Load on "B"}}{\text{Load on "A"}}$ = 2.56 (load factor)

Actual bearing wear:

Bearing "A" = vertical wear = 0.002"

side wear = 0.0009" to 0.0014"

Bearing "B" = vertical wear = 0.0045"

side wear = 0.0023" to 0.0037"

Calculated bearing wear from load factor:

Bearing "A" = 0.002" vertical wear x 2.56 load factor = 0.0051"

(Should correlate with 0.0045" wear "B")

Bearing "A" = 0.0012" side wear x 2.56 load factor = 0.0031

(Should correlate with 0.003" average wear "B".)

Estimated bearing life under imposed conditions:

Bearing "A" = 500 hours

Bearing "B" = 200 hours

Shaft wear = unmeasurable

Thrust Plate Wear = 0.0007" to 0.0009" from initial condition (negligible)

V. CONCLUSIONS

Recently the damaged motor was repaired to operating condition. This motor was run on a dynamometer to determine a value of developed horsepower.

The dynamometer results were:

Speed = 1000 RPM

Torque = 13.5 in-lbs.

Motor Pressure = 82 psig

Vane Pressure = 50 psig

Developed Horsepower = 1/5

The air leakage was very large in this motor indicating low efficiency; so the developed horsepower is only indicative and is not a maximum figure.

The author feels that when this motor is returned to good operating condition, the motor will develop the full 3/4 horsepower for that it was originally designed (line pressure = 100 psig). The new bearing design will give the motor an operating range of -65°F to +1000°F when it has been perfected.

APPENDIX A

Patent disclosure for a High Temperature Air Motor.

Patent Docket Number 24D-A-46

The motor described in the enclosed disclosure is designated as Prototype #3, because a new design of bearings suitable for 1000°F operation has been incorporated. These bearings have been built, and the internal diameter checks the design standards at room temperature and at 1000°F. The mid-temperature range must be improved, because the stainless steel used has a non-linear expansion curve.

AIR MOTOR

I. PURPOSE

The purpose of this invention is to provide a new and improved air motor capable of operation at high temperatures (500°F-1000°F). It is also a purpose to provide an air motor having a plurality of vanes and air ports to facilitate operation in either direction and provide increased efficiency. Another improvement is provided by the use of air loaded vanes to eliminate the use of springs and improve the pressure loading of the vanes upon the motor case at various speeds. Still another improvement lies in the use of balanced forces (double opposed cam design) to lengthen bearing life. Another improvement is the fact that this motor is equipped with bearings that are capable of operation in the temperature band of 500°F to 1000°F. Another improvement is the fact that this motor is equipped with a shuttle piston that allows bi-directional operation with two air lines.

II. ADVANTAGES

One advantage of this construction is high starting torque.

Another advantage is an even power flow for 360° of rotation.

Another advantage is the fact that the motor has no dead spots and will start with full power from any rotor position.

Still another advantage is the fact that, due to the secondary expansion air cylinder, a maximum amount of power is secured from the compressed air before discharge from the motor.

Another advantage is the use of graphite to stainless steel construction to insure a low friction operation at elevated temperatures with a minimum of wear.

Other advantages will become apparent in the following description and drawings of one embodiment of my invention in which:

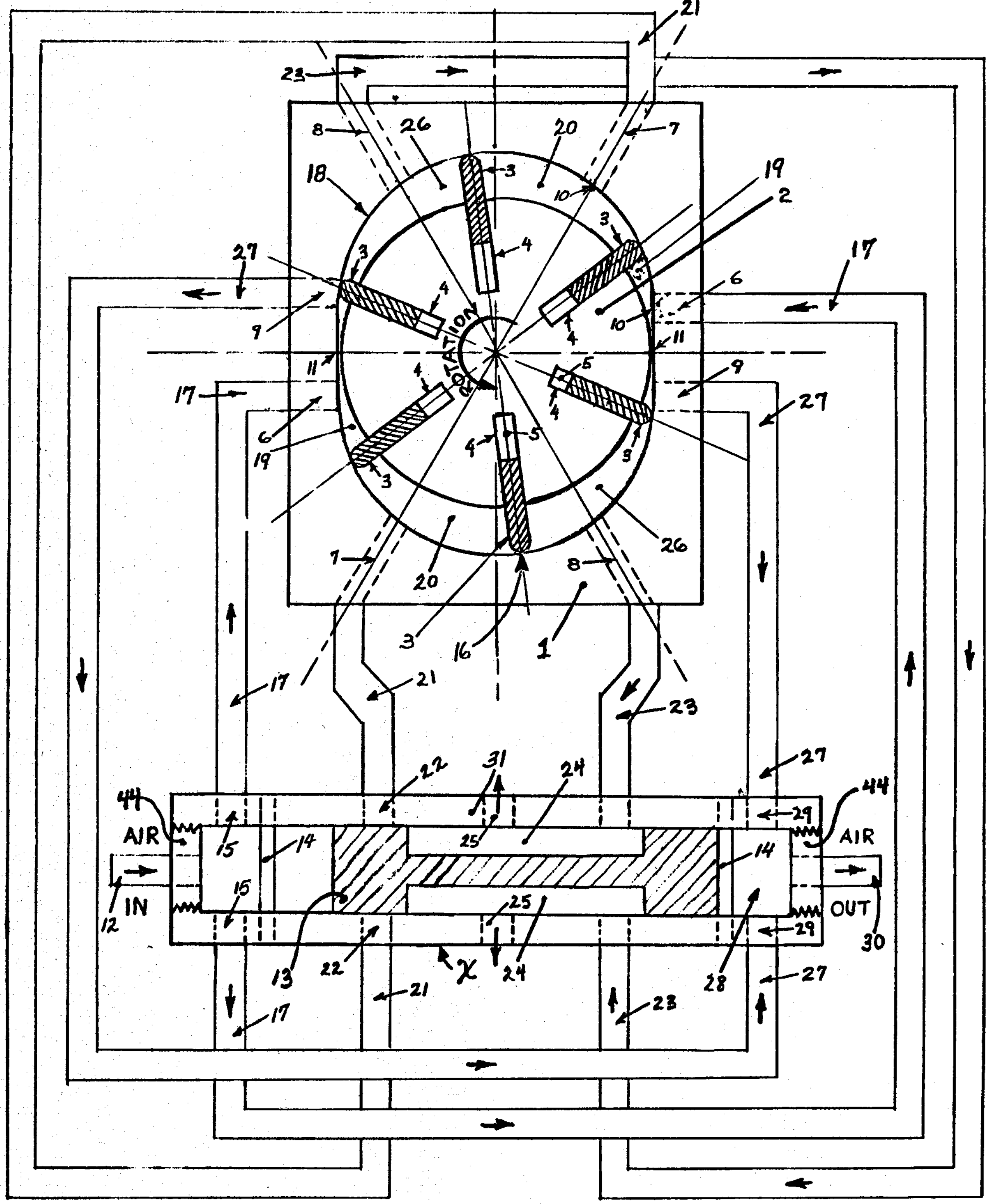
Figure I	is an end sectional view through air motor
Figure II	is a side sectional view through air motor
Figure III	is a face view of an end bell for air motor
Figure IV	is a diagram of motor force balance
Figure V	presents sketches of possible inlet air channel configurations

III. DESCRIPTION

The air motor as illustrated in Figures I and II is provided with a body 1 and a rotor 2 that revolves inside of the body 1. The rotor 2 is equipped with six equidistant vanes 3, that slide in slots 4 in the rotor 2. The motor body 1 is preferably constructed of high temperature stainless steel (AISI type 347 or equivalent). The cylinder 19, 20, and 26 is of double [illegible] opposed cam design (balanced rotor forces, see Figure IV), which can either be oval, as shown, or eliptical in form.

AIR MOTOR

FIG. I



SECTIONAL VIEW THRU AIR MOTOR AND SHUTTLE VALVE
(ARROWS SHOW AIR FLOW FOR COUNTER-CLOCKWISE ROTATION)

FIGURE II

CROSSECTIONAL VIEW OF MOTOR LENGTHWISE WITH END BELLS IN PLACE

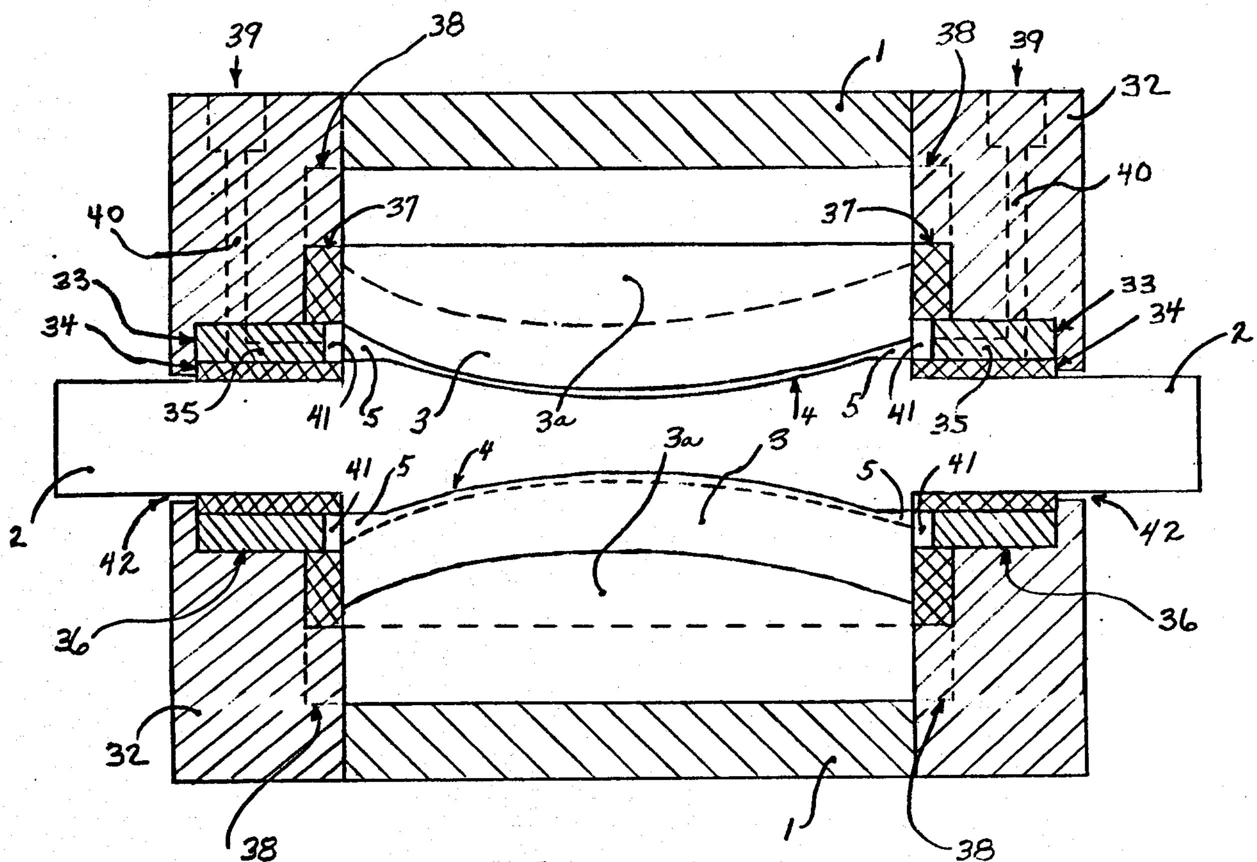
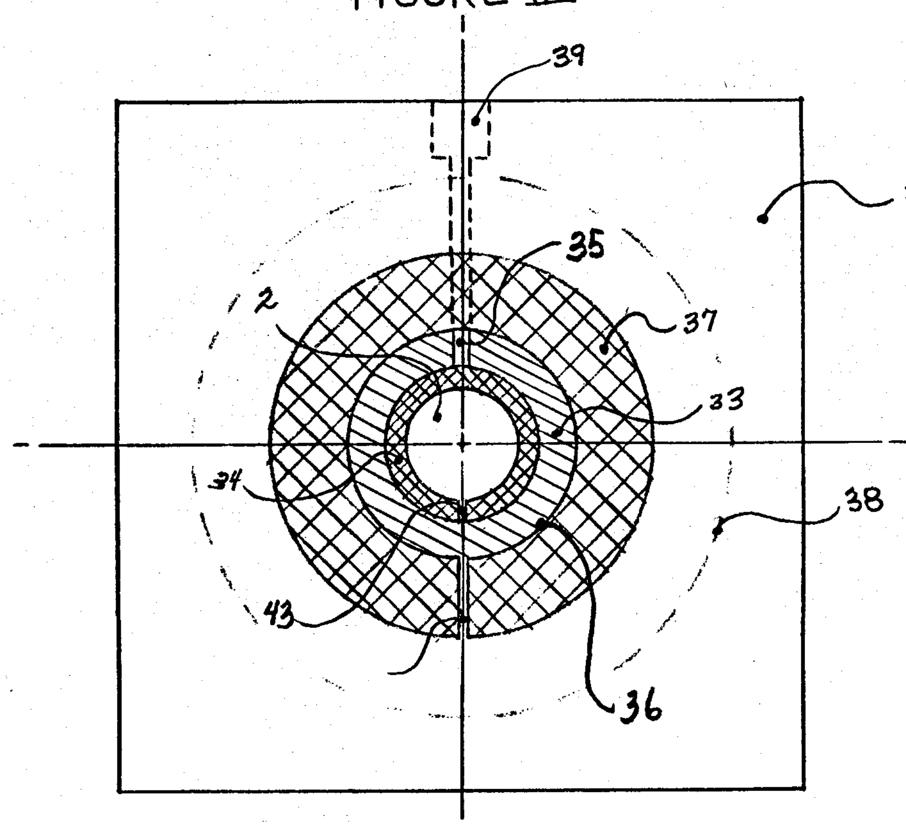


FIGURE III



FACE VIEW OF END BELL AND BEARING

FIGURE IV

BALANCED MOTOR FORCES

P_1 = P.S.I. INLET
P_2 = SECONDARY EXPANSION PRESSURE
P_3 = EXHAUST PRESSURE
W = ROTOR WEIGHT

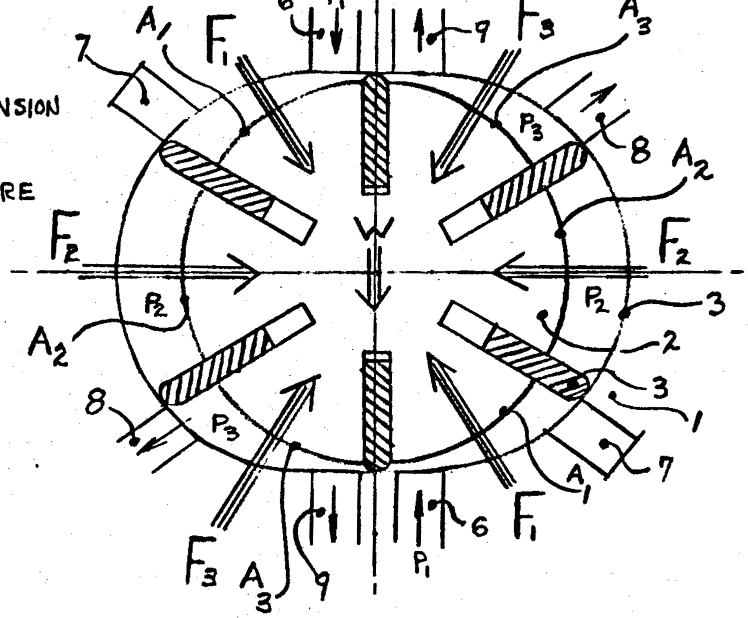
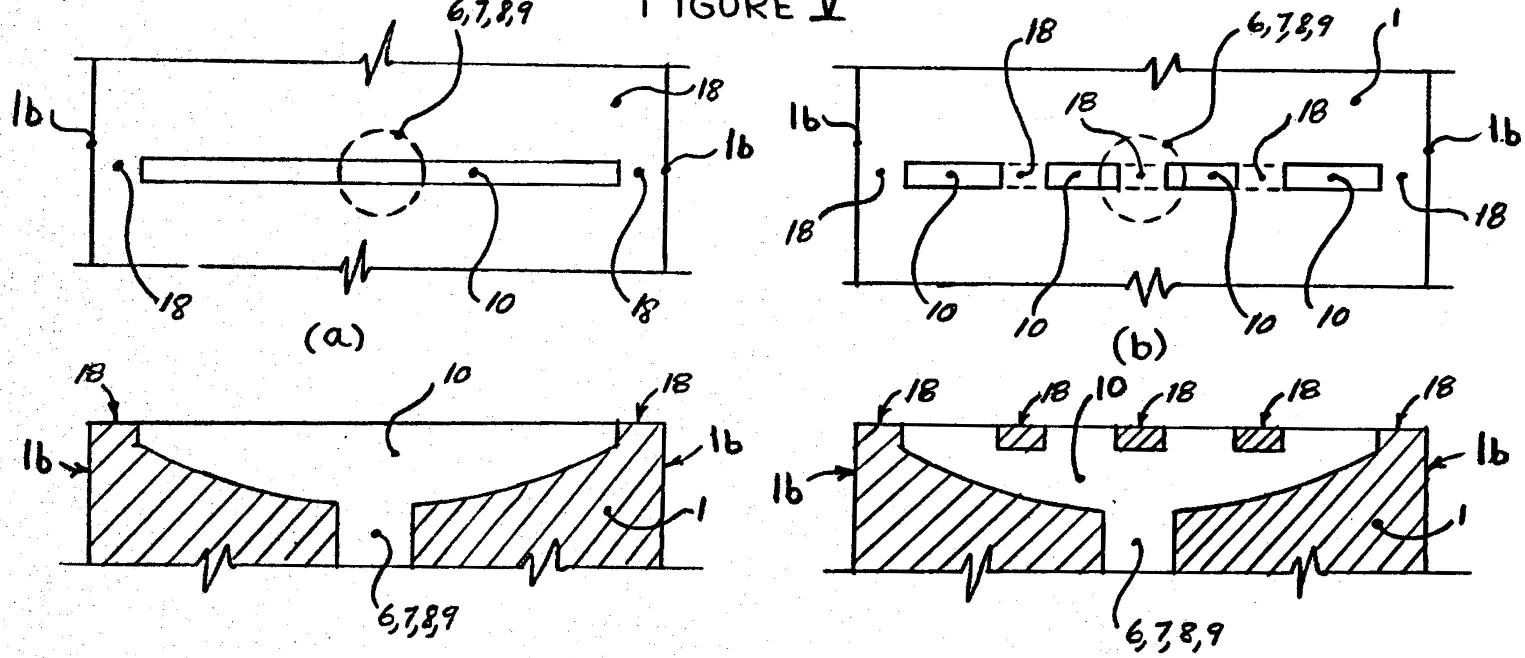


FIGURE V



The rotor 2 is preferably constructed of high temperature stainless steel (AISI type 309 or equivalent) of a lesser coefficient of expansion than the body 1, so as to provide an increasing internal rotor clearance 11 with increasing temperature. This clearance increase provides for better operation at elevated inlet air temperatures.

The rotor 2 is supported by end bells 32 that are preferably constructed of the same material as the body 1. These end bells are provided with high temperature, low friction bearings 33, 34, and thrust plates 37, 38. The material suggested for these surfaces is a high temperature Graphite (Pure Carbon Co. Type L-56-HT) or other suitable material.

The rotor 2 is provided with vanes 3, 3a constructed of the same material as the bearings. The vanes 3, 3a are to operate with a smooth sliding fit within the rotor slots 4 after an initial break-in period to perform a polishing operation if graphite is used. These vanes 3, 3a are provided with clearance 5 at the bottom of the slots 4 to provide an air pocket at 5 for pressure loading of the vanes 3 to position 3a from the end bells 32. This pressure loading provides a variable vane pressure loading at 16 against the motor body 1, and also provides cushioning for vanes 3 constructed of graphite to prevent shock and breakage.

The motor body 1 is provided with a plurality of ports to secure bi-directional operation. For the direction of rotation shown in Figure I, the dual ports 6 become primary air inlet ports. The dual ports 7 become blocked secondary expansion ports. The dual ports 8 become primary air exhaust ports. The dual ports 9 become secondary air exhaust ports. The above sequence will be completely reversed for rotation counter to that which is indicated. The reversal is accomplished by changing the inlet air supply on the shuttle valve from port 12 to port 30. The port opening 10 within the body 1 shall have a width (with rotation) exactly equal to the width of a vane 3. This fact provides for positive shut-off of all ports during cylinder change-over to prevent air leakage between cylinders.

The ports 6, 7, 8, and 9 on one half of the motor shall be offset by one vane width with respect to the other half of the motor. This fact will allow no dead spots and starting with full power on the motor will be possible. The length of the port openings 10 shall be a factor less than the vane length to provide a bearing surface for the vanes 3 when in this position. This will prevent vane chatter and possible breakage or chipping. Suggested configurations appear in Figure V.

The rotor 2 will preferably have a clearance of .002" to .004" between the body 1 and the rotor 2 at the seal point 11, so that a minimum of leakage will occur between the primary ports 6 and the secondary exhaust ports 9.

The vanes 3 may be of solid construction with a radius on the outermost surface 16 to afford an even mating surface to the internal contour 18 of the body 1, or the vanes 3 may be of other appropriate designs.

The shuttle valve x as illustrated in Figure I consists of a body 31; with inlet air ports 12 and 30 primary air ports 15, secondary expansion ports 22, primary exhaust ports 23, and secondary exhaust ports 27. It is equipped with a smoothly operating and tight fitting piston 13 that operates between stops 14 within the body 31. The body 31 is completely sealed with end caps 44 to prevent air leakage. The shuttle valve x can be operated by a four-way valve (not shown) or by two three-way valves (not shown).

The cross-sectional view of Air Motor as illustrated in Figure II, consists of a body 1, a rotor 2, and end bells 32. Vanes 3 are shown in the extreme position of their travel as 3a. The end bells 32 are provided with air inlets 39 to provide air pressure to position the vanes from 3 to 3a as necessary. The air enters at point 39 proceeds through channel 40 into the open slot area 35 of the split bearing housing shell 33. The air then distributes through the annulus 41 and into the inlet guide 5 under each of the vanes 3. The pressure which is trapped in the loading clearance in slot 4 will move the vanes to position 3a as necessary. The end bells 32 are equipped with split bearings 34, split bearing housing shells 33, and thrust plates 37, or 38. A clearance 42 is provided between the rotor 2 and the end bell 32 for high temperature operation.

The Face View of End Bell and Bearing as illustrated in Figure III consists of an end bell 32, containing a bearing 34 split at 43. This bearing is contained in a bearing housing shell 33 which is split at 41. The bearing housing shell 33 is contained within the bore 36 of the end bell 32. The end bell body 32 is provided with a thrust plate 37 that provides a high temperature thrust and bearing surface for the rotor 2. This thrust plate can be extended to 38 to allow a solid break-free surface for the vanes 3 to operate against. The end bell 32 has an air inlet port 39 that connects through a channel to the split 41 in the bearing housing shell 33 for vane loading within the motor (refer to Figure II). The end bell 32 of body 1 is preferably constructed of the same material as the motor body 1 (Figure I). (AISI type 347 stainless steel or equal.) The split bearing housing shell 33 is preferably constructed of high temperature material with a coefficient of expansion as predicted from bearing calculations. The thrust plate 37 or 38 and the split bearing 34 are preferably constructed of high temperature graphite (Pure Carbon Co. type L-56-HT or equivalent).

Description of Force Balance (Figure IV). The Figure IV consists of a rotor of weight W with 2 vanes at the neutral point within the body 1 of the motor.

The inlet pressure P, entering through port 6 and acting over area A of the rotor, results in the force components "F" cancelling each other. This fact is provided for by port 7 being closed off by the shuttle valve. The pressure P_2 of the expanding air in the expansion cylinder applied over area A_2, results in the force components "F_2" cancelling each other. The decreasing pressure P_3 of the exhaust cylinder, leaving through ports 8 and 9, and acting over area A_3, results in the force components "F_3" cancelling. The resultant bearing load is composed of only "W" the rotor weight.

Figure V describes two possible methods of channelling the inlet air flow to, or away from, the vanes. The body 1 with edge 1b is shown with air inlet holes 6, 7, 8, or 9 changing contour to fan shape 10. Vane support is provided by areas 18 as shown in Figure (a) or Figure (b).

IV. OPERATION

Figure I: The air motor is provided with a shuttle valve x having a piston 13 to provide a reversing mechanism for the motor. The air inlet 12 to the shuttle valve x is shown for one direction of rotation of the motor. For opposite rotation the inlet 12 and exhaust 30 ports must be reversed.

Air enters the shuttle valve housing 31 through port 12 and slides suitable piston 13 to stop 14. The air then proceeds through outlet ports 15 into suitable connecting tubes 17 to the power inlet ports 6 of the motor body 1. The air then enters the power cylinders 19 pushing upon the vanes 3, that are in power position, causing the rotor 2 to move through 60° of rotation. At this point the air is then transferred to a new set of vanes 3 that have been moved into the power position.

The power cylinder 19, before mentioned, has been moved into the position for the secondary air expansion to take place in the secondary expansion cylinder 20. This air then expands for 60° of rotation, due to the fact that the shuttle piston 13 has blocked the exhaust ports 7 of the secondary expansion cylinder 20, through the connecting tubes 21 and the inlet ports 22 to the shuttle valve housing 31. Therefore the air expansion cylinder 20 has removed as much work as possible from the inlet air, and dropped the original pressure of the inlet air supply to a value that is practical to remove it from the motor. This is accomplished by the opening of the primary exhaust ports 8 as the vanes 3 move by. The expanded air is then discharged through suitable connecting tubes 23 into the collecting chamber 24 and out to atmosphere through the primary exhaust ports 25 of the shuttle valve housing 31. The vanes 3 then proceed toward the secondary exhaust ports 9 of the motor, pushing the amount of air left in the primary exhaust cylinder 26, after dump, out of the secondary exhaust port 9. This port prevents any work being done in moving the vanes 3 to a new power stroke position 19. The secondary exhaust port 9 vents the air through the suitable connecting tubes 27, into the collecting chamber 28, and through the outlet port 30 of the shuttle valve, to be returned to the four-way valve (not shown).

The vanes 3 have moved through 180° of travel and have accomplished one half of the total number of power strokes per revolution. This type of operation gives rise to 12 primary power strokes per 360° of revolution and 12 secondary air expansion strokes per revolution. This efficient use of air provides the motor with even power distribution, constant torque, and a high starting torque.

The high temperature operation of the motor is provided for by the bearings 34 and thrust plates 37, 38 shown in Figure II and Figure III.

The air motor shown in Figure II consists of a rotor 2 supported by bearings 34 and provided with graphite thrust plates 37 or 38, or other suitable material. There is no possibility of a metal to metal contact within the motor at elevated temperatures. The vanes in position 3 and 3a are provided with a graphite metal seal 37 or a solid graphite seal 38 at the ends.

Figure III illustrates the high temperature bearing provided for the air motor. The bearing 34 with split 43 is manufactured oversize before splitting and then compressed diametrically and inserted into the bearing housing shell 33. The bearing housing shell 33 likewise is manufactured oversize, split at 35, and compressed for insertion into the bore 36 of the end bell 32. This compression of the composite bearing insures that they will follow the confining O.D., which is the bore 36 in the end bell 32 through the complete temperature range of the motor. Due to the split in both the bearing 34 and the shell 33, the wall expansion is the only factor contributing to the thermal characteristics of the I.D. of the bearing 34. This bearing I.D. can be made to follow the thermal expansion of any size shafts by controlling the size of the bore 36 in the end bell 32 and the wall thickness of the housing shell 33 and the bearing 34. The relation equating these factors is as follows:

Shaft O.D. at Temperature Z + Clearance = End Bell Bore at Temperature Z - 2 (Shell Wall at Temperature Z + Bearing Wall at Temperature Z)

An equation has been developed by Pure Carbon Co. for determining the optimum wall thickness of graphite so that the subject bearing 34 will perform as desired through the total temperature band of the air motor. The wall thickness established by this equation will provide the bearing 34 with the necessary properties of diametrical compression or expansion which will allow the bearing to be compressed for insertion without breakage.

This formula is as follows:

$$\overline{\Delta D} = \frac{D^2}{t} \times \frac{2}{3} \lambda$$

Definitions

$\overline{\Delta D}$ = the allowable diametrical compression or expansion of a ring of given wall thickness before breakage due to exceeding the allowable stress at elastic limit in tension or compression.

(This compression or expansion is the factor that insures that the bearing will follow the confining O.D. through the temperature range.)

$\overline{D}$ = mean bearing diameter (free condition) at room temperature

t = wall thickness at room temperature

λ = allowable stress at elastic limit in tension or compression for the particular graphite grade used in the bearing. (units = inches/inch). Exceeding this value with graphite results in breakage.

λ for L-56-HT = .0023 inches/inch in tension (diameter compression)

An expansion of this formula made by the writer for simplicity of solution is as follows:

General Formula:

$$(1000^{o}F \text{ mean diameter} - 70^{o}F \text{ mean diameter}) = \frac{(\text{mean diameter } (70^{o}F))^2}{\text{Wall @ } 70^{o}F} \times \frac{2}{3}\lambda$$

Specific Conditions:

$$(\text{O.D. @ } 1000^{o}F - (\text{wall thickness} + \text{expansion @ } 1000^{o}F)) - (\text{O.D. @ } 70^{o}F - \text{wall thickness @ } 70^{o}F = \frac{\left(\frac{\text{O.D. @ } 70^{o}F + \text{I.D. @ } 70^{o}F}{2}\right)^2}{\text{Wall @ } 70^{o}F} \times \frac{2}{3}\lambda$$

This equation reduced to quadratic form using a $2/3\,\lambda$ equal to .001 inches/inch for L-56-HT graphite results in:

$$1.5\,X^2 + (\overset{(70^{o}F-1000^{o}F)}{\text{mean diameter change}} \times 10^3 + \overset{@\ 70^{o}F}{2 \text{ mean diameter}})\,X - (\text{mean diameter @ } 70^{o}F)^2 = 0$$

X = optimum wall thickness desired.

This formula can also be used for calculating the wall thickness of the bearing housing shell 33. Supposing that the bearing housing shell is constructed of 347 stainless steel, the allowable stress in tension without exceeding the elastic limit or allowing appreciable creep at 1000°F results in $2/3\,\lambda$ = .001 inches/inch fiber stress.

V. NOTE

The bearings 34 and shells 33 are to be manufactured oversize of the top temperature conditions, so that a spring action will still be present to insure that confining O.D., or end bell bore 36, will control the bearing I.D., due to the fact that the bearing 34 and shell 33 will maintain a pressure against this O.D.

The oversize bearings 34 will have sufficient split to insure that all circumferential change during "close in" will be absorbed by the gap and not completely close same.

The oversize shells 33 will have sufficient split to insure that all circumferential change during "close in" will be absorbed by the gap and result in a sufficient gap for air loading passage 35 to the vanes.

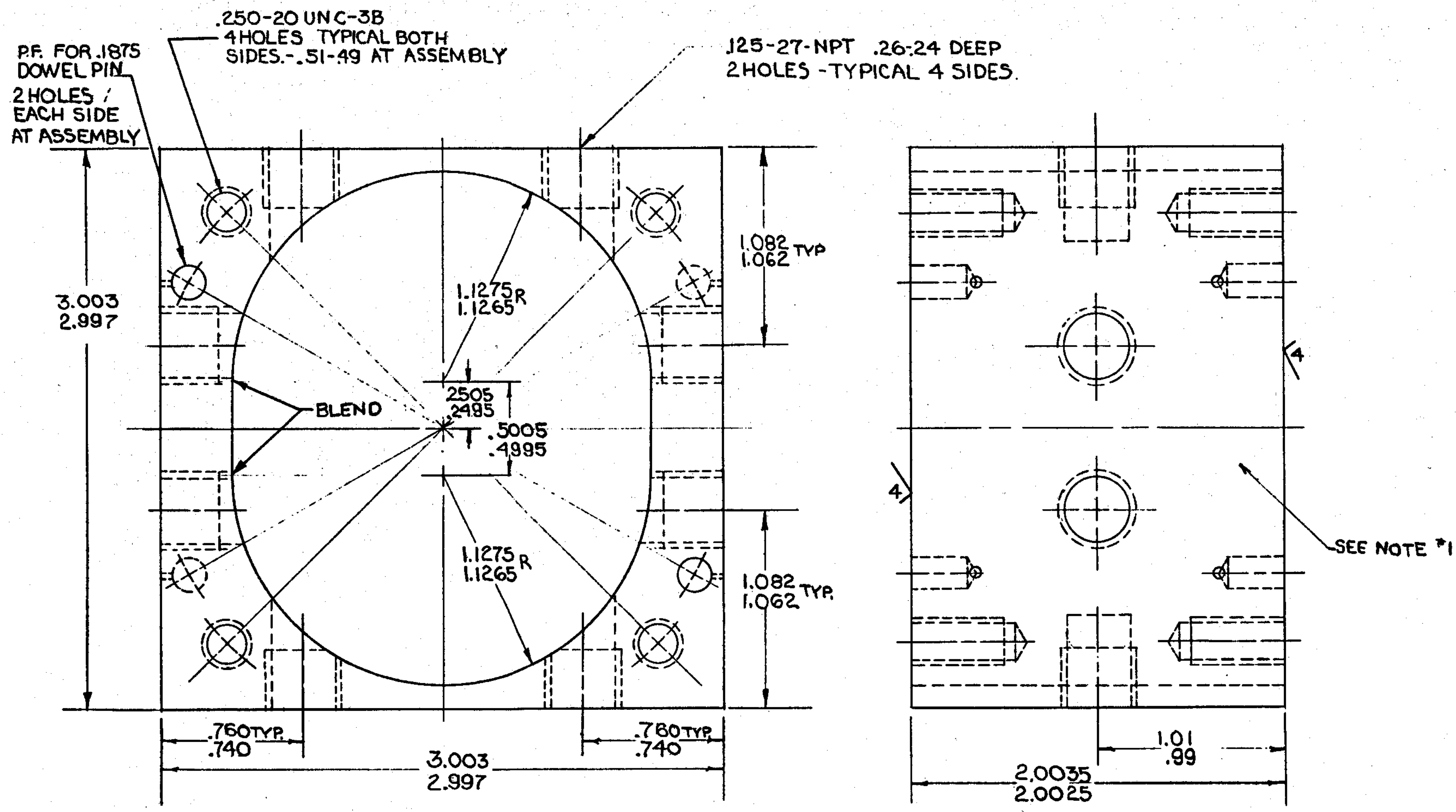
A hard chrome plate (flash) .0005" thick may be applied to the O.D. of the bearing housing shells 33 to provide a bearing surface for insertion and high temperature operation of the shells 33 within the end bell bores 36.

The operation of Figures IV and V are apparent with the description provided.

Although I have presented one embodiment of my invention, other forms may be had without departing from the scope of my invention.

APPENDIX B

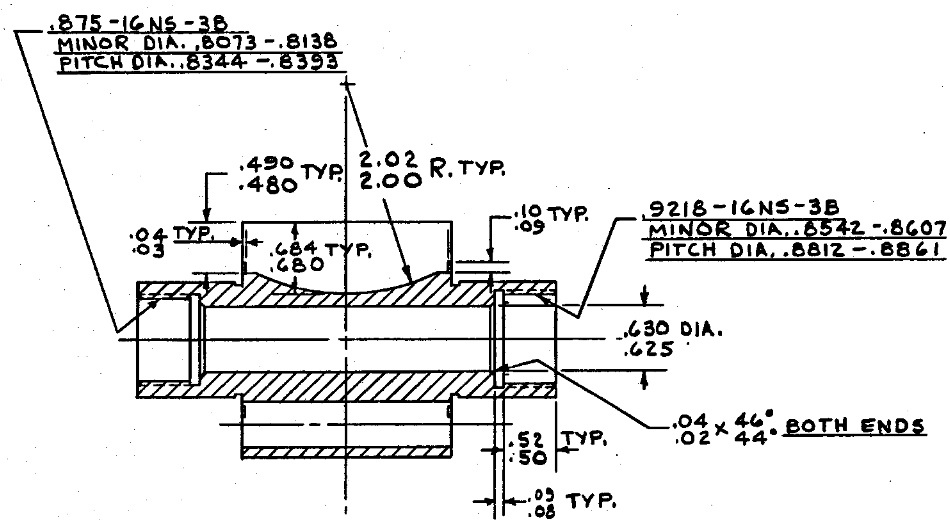
Manufacturing prints for Prototype #2.



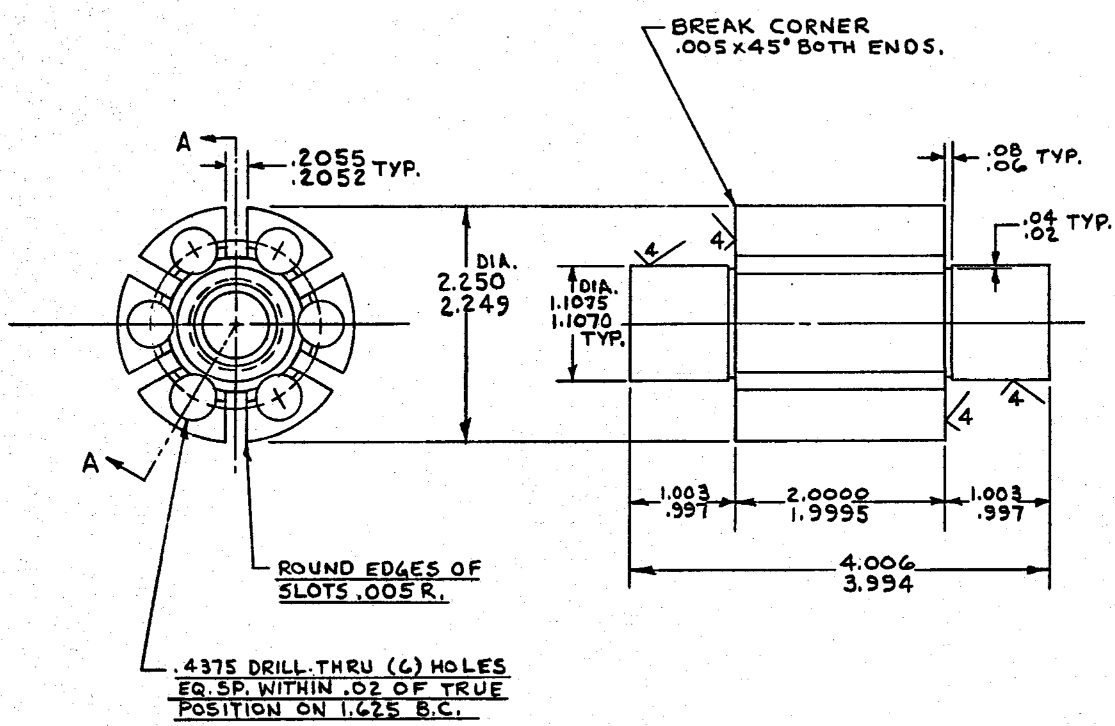
2 BREAK ALL SHARP EDGES .005 R.
1 STAMP PART "GE 641C183 P1 REV_" PER. AMS. 2800.

NOTES:

Housing (641C183)

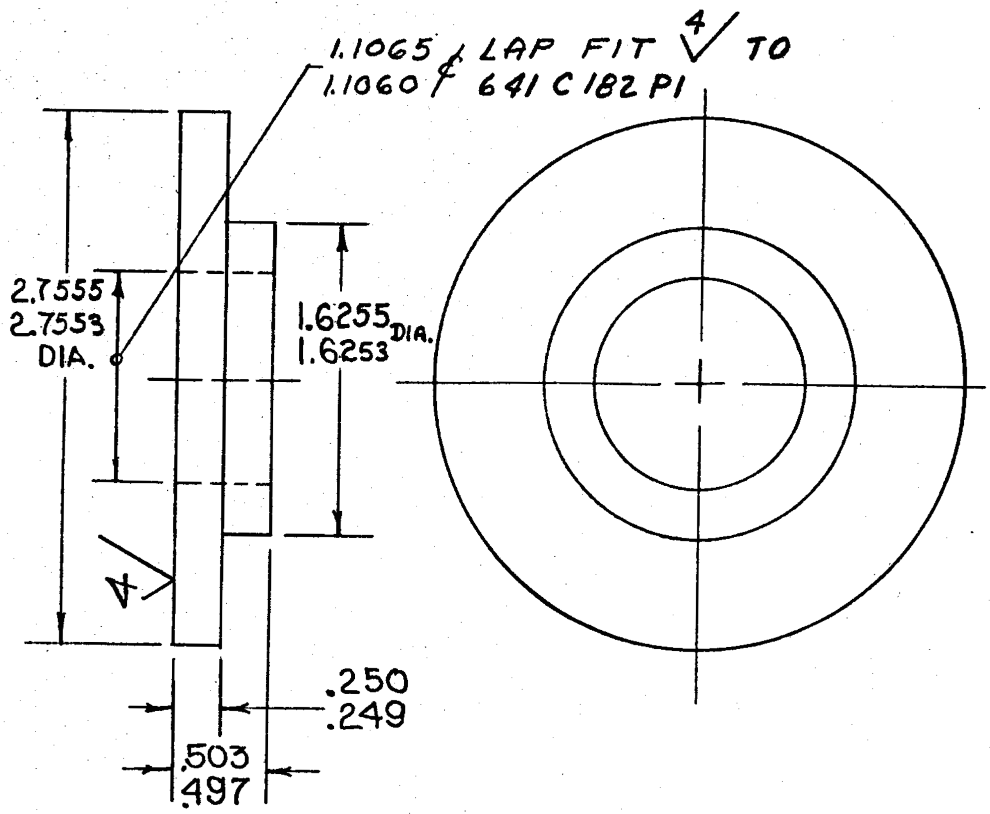


SECTION "A-A"



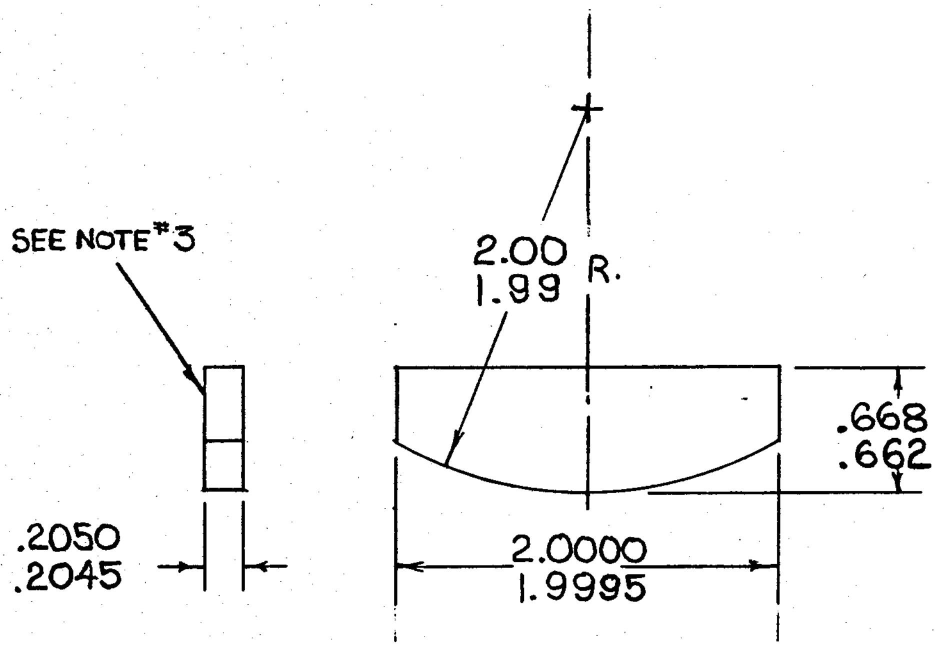
2. BREAK ALL SHARP EDGES .005 R.
1. TAG GE641C182P1 REV__'.
NOTES:

Rotor (641C182)



2 BREAK ALL EDGES .01-03 R.
1 TAG "GE 105A5105 P1. REV_".
NOTES:

Bearing (105A5105)



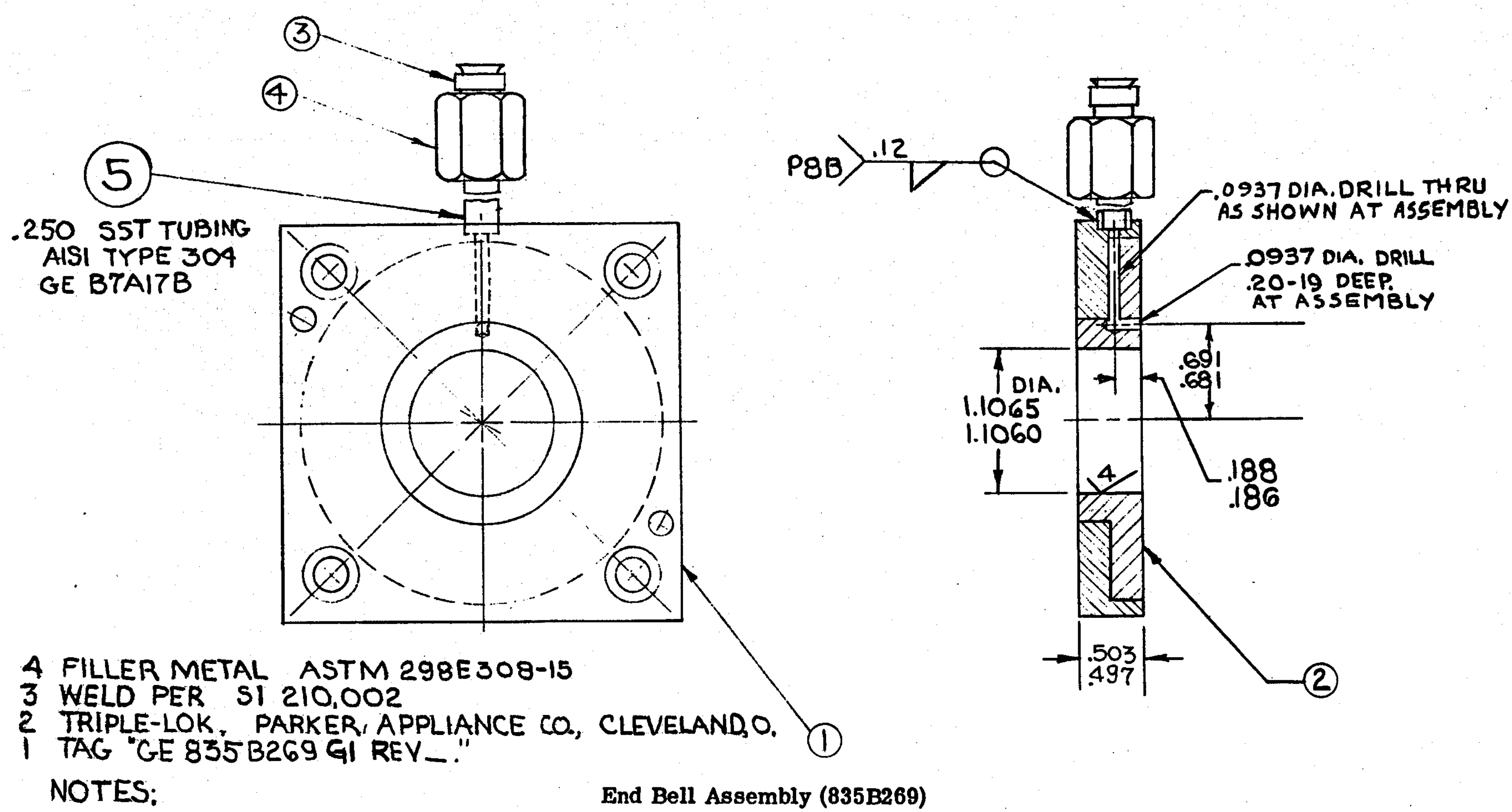
3 SIDES TO BE PARALLEL WITHIN .0005 F.I.R.

2 BREAK ALL EDGES .005 X 45°

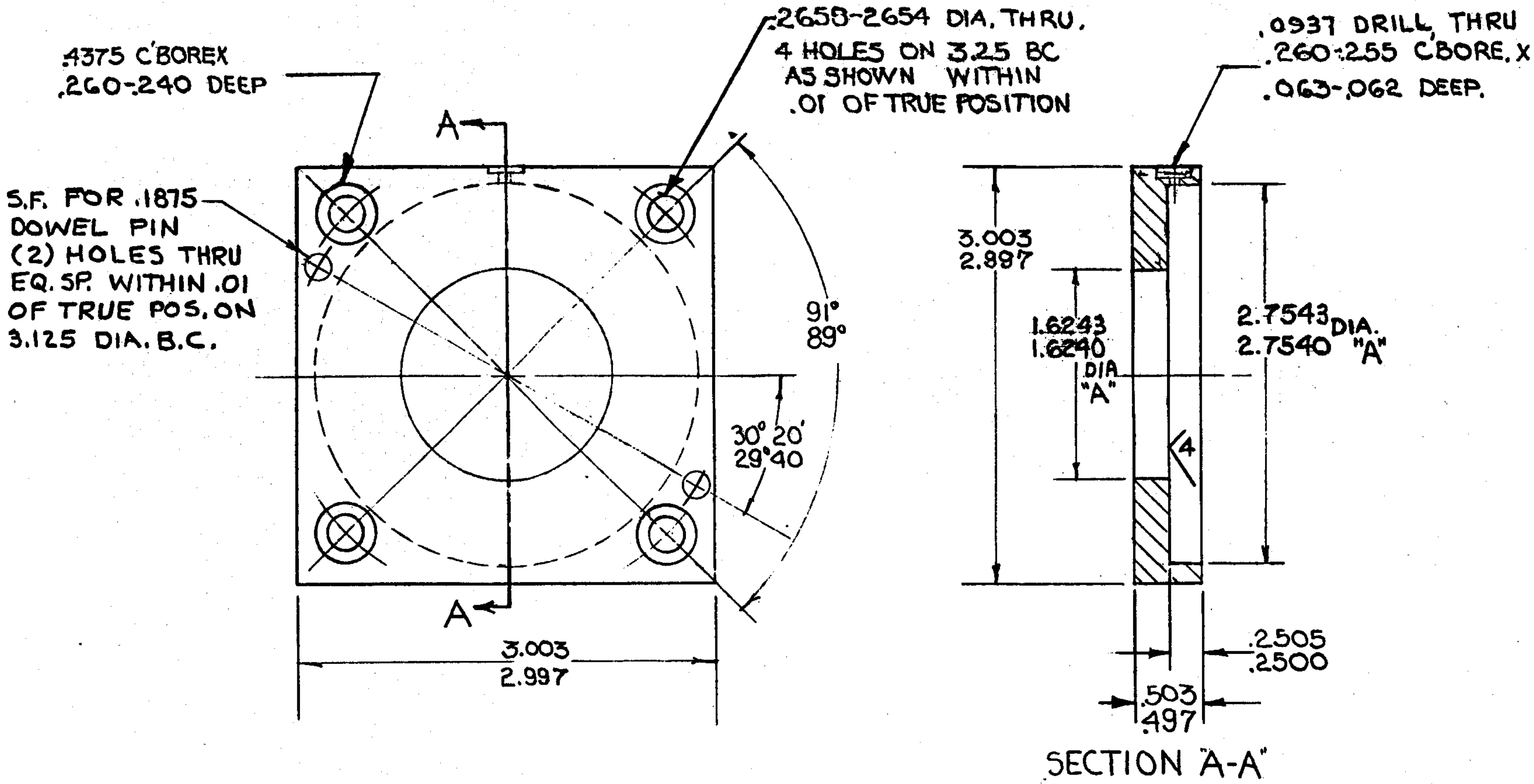
1 TAG "GE 105A5104 P1. REV_"

NOTES:

Vane (105A5104)

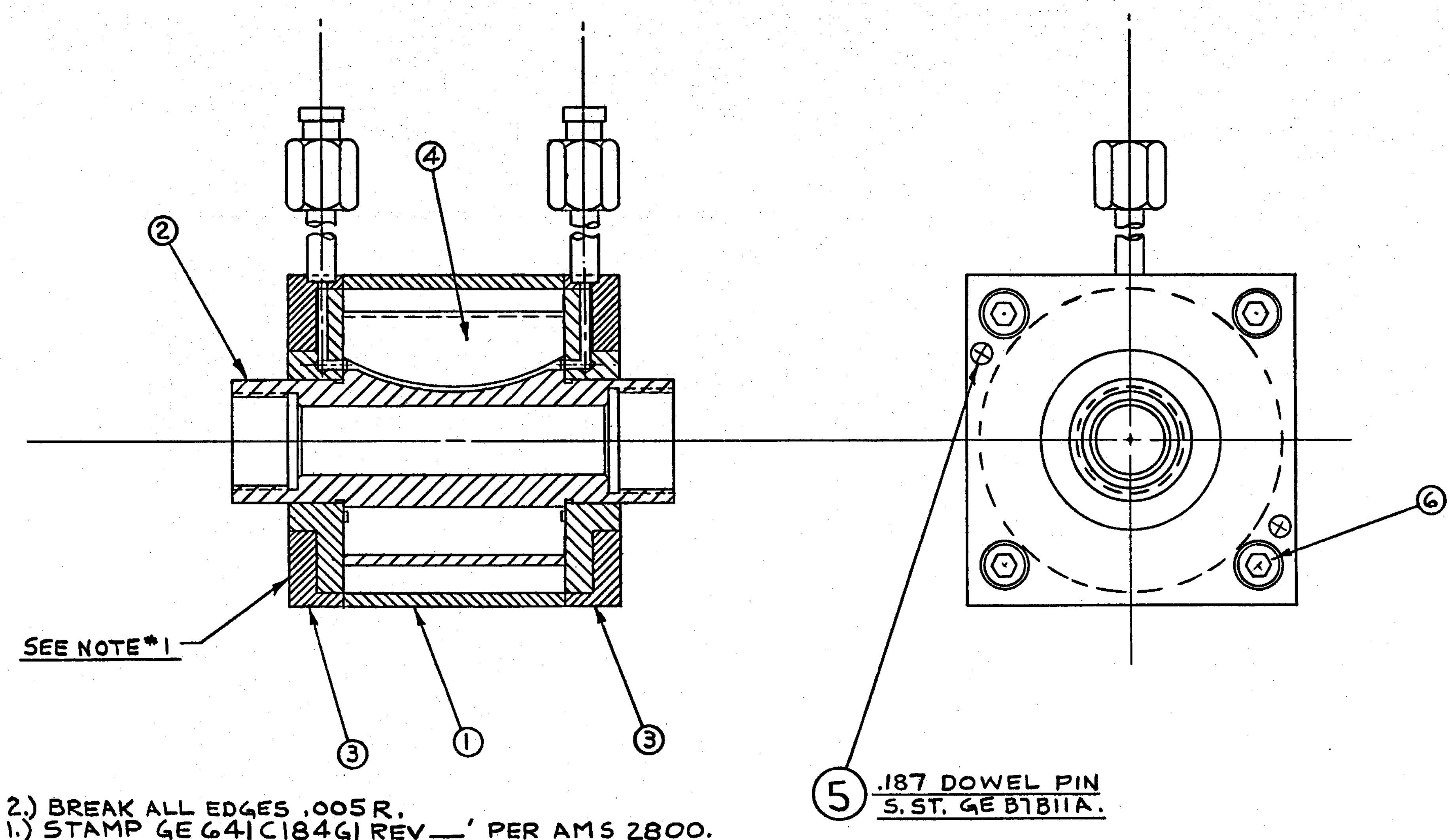


End Bell Assembly (835B269)



3 DIA'S "A" TO BE CONCENTRIC WITHIN .001 F.I.R.
2 BREAK ALL EDGES .005R
1 TAG "G-E 835B268 P1 REV__."
NOTES:

End Bell (835B268)



Air Motor Assembly (641C184)