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GAS FLOW CONTROL DEVICE FOR USE IN AEROSPACE THERMAL AND ATMOSPHERE CONTROL SYSTEMS

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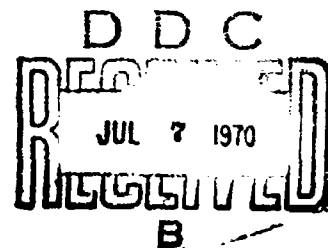
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SUMMARY

The objective of this work was the development of a small, light-weight gas flow controller able to deliver an oxygen flow of 2.65 ± 0.05 , -0.00 lb/hr at a nominal pressure of 200 psig in the presence of $\pm 2\%$ fluctuations in inlet (upstream) pressure during operation in an ambient temperature range of -40 F to $+200$ F.

The scope of this work included the selection of a flow control method and the detailed design, fabrication and testing of a flow control unit. Low weight and small envelope, though emphasized during the design phase, were secondary to demonstrating the performance capabilities of the chosen flow control technique. Consequently, the extensive design work necessary to optimize the dimensional characteristics was beyond the scope of this effort.

The flow controller uses an orifice whose area decreases proportionately with changes in upstream pressure and increases with ambient temperature. Since the mass flow rate delivered by a choked orifice is directly proportional to the product of orifice area and upstream pressure, and inversely proportional to the square root of gas temperature, it is possible, with an appropriate design, to maintain flow within the desired limits.

The variable orifice is the gap created by the axial motion of a partially flattened rod through a sharp edged hole. The flattened portion of the rod is inclined to the axis of the rod to make the area of the gap a function of the axial position of the rod. For small motions of the rod, this relationship is virtually linear. The rod is spring loaded against a diaphragm having an axial deflection determined by the pressure, P_θ , of a trapped volume of gas, and upstream pressure, P_u . By the perfect gas law, P_θ is proportional to the temperature of a gas trapped in a constant volume. Consequently, the orifice area will proportionately increase with ambient temperature and decrease with upstream pressure.

To provide adequate compensation to meet the flow control requirements, the various proportionality constants had to be carefully selected. This is accomplished by specifying the inclination angle of the flat on the rod after the diaphragm stiffness and the orifice discharge coefficient had been experimentally determined. To obtain the correct sensitivity to thermal changes, the nominal value of P_θ was set to one half the nominal value of P_u . Precise setting of the flow rate was accomplished by adjusting the axial position of the rod with the aid of a spacer.

The major design features of the flow controller were selected on the basis of a tradeoff between low diaphragm stress level and large diaphragm deflection and are as follows:

- . weight: 1.5 lb
- . diameter: 2.5 in.
- . length: 1.5 in.
- . diaphragm stress level: 70,000 psi
- . diaphragm deflection: 0.0085 in.
- . orifice hole diameter: 0.060 in.
- . inclination angle of flat: 12.7°
- . upstream pressure: 450 psig

Materials of fabrication were selected on the basis of compatibility with high pressure oxygen, minimum expansion properties and durability. The diaphragm is made of Ni-Span-C nickel steel alloy to minimize changes in Young's Modulus with temperature. Ni-Span-C is also used for other parts of the flow controller to minimize thermal expansion and corrosion. The precision orifice plate and control rod are made of Tungsten Carbide to provide extra stability and durability.

The flow controller was tested with oxygen at ambient temperatures of -40 F, +70 F and +200 F with upstream pressure fluctuations of $\pm 2.5\%$. For an upstream pressure of 450 psig, the low temperature caused a flow change of 2.5% and the high temperature increased flow by 2.1%. At each temperature, $\pm 2\%$ changes in upstream pressure resulted in less than $\pm 0.75\%$ change in flow. The resulting maximum and minimum values of flow were 2.766 and 2.667 lb/hr, respectively. Though this 0.1 lb/hr range is larger than the design goal, these results were obtained without preliminary trial and error "tuning" of the flow controller. The above accuracy might be substantially improved if the optimum value of the inclination angle of the flat were empirically found and used. The scope of the present contract did not permit this to be pursued.

FOREWORD

This report was prepared by Foster-Miller Associates, Inc. and the Aerospace Medical Research Laboratory, on Contract No. F33615-69-C-1686 in support of Project No. 7164, "Thermal Effects and Altitude Protection of Aerospace Crewmen," and Task No. 716411, "Aerospace Pressure Suit and Environmental Control Technology." The work was performed under the direction of the Aerospace Medical Research Laboratory, Aerospace Medical Division, Wright-Patterson Air Force Base, Ohio and was monitored by Clemens M. Meyer, Altitude Protection Branch, Environmental Medicine Division.

The work reported here was performed from 15 June 1969 to 30 November 1969 by the Engineering Studies Division of Foster-Miller Associates, Inc. under the direction of Mr. John Howland. Dr. Seth R. Goldstein was the project engineer, and Mr. Andrew Harvey and Dr. Herbert Richardson provided valuable technical assistance during the design phase of the program.

This technical report has been reviewed and is approved.

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SECTION I

INTRODUCTION

To reduce the power requirements of extravehicular manned space enclosures it would be desirable to use the available potential energy of stored high pressure oxygen. This energy could be harnessed to operate turbine-driven gas or liquid circulating devices to provide temperature control in small volume enclosures. The expended oxygen would, in turn, supply the metabolic needs of the man while maintaining the required carbon dioxide level in a semi-open or open breathing loop. This technique would obviate the power requirements of electrically-powered blowers or pumps.

Previously, pressure regulators have been used to reduce pressures to the required level for driving these circulating devices. Although this regulation maintained a relatively constant inlet pressure, it did not provide the required constant flow when affected by changes in ambient temperature. This suggested that a flow controller should be stationed between the regulator and circulating device to improve overall performance by compensating for these changes.

This report summarizes the results of a technical program conducted to develop the desired flow controller.

SECTION II

EVALUATION OF FLOW CONTROL METHODS

INTRODUCTION

The problem of devising a flow control method to supply oxygen to small gas or liquid turbine-driven circulating devices for atmosphere and thermal control in aerospace enclosures was evaluated. A representative application is illustrated in Figure 1.

Oxygen at high pressure (7500 psig) flows from a storage tank through a regulator, where the pressure is reduced to P_u (450 psig), and into a flow controller. The oxygen leaves the flow controller at a pressure P_d (200 psig), and finally flows into a gas or liquid circulating device. P_u and P_d designate pressures upstream and downstream, respectively, of the flow controller. During this process, the oxygen flow is to be maintained constant within specified limits in the presence of fluctuations in ambient temperature between -40 F and +200 F and in upstream pressure of $\pm 2\%$.

Numerous techniques for maintaining a constant flow of a compressible gas were considered in this study. Three of the more promising methods considered are discussed in this section. The major criteria for evaluation were:

- . ease of control
- . ease of compensating for changes in temperature and pressure
- . simplicity and reliability

METHOD I

In this approach a flow is forced through a constant area choked¹ orifice by an upstream pressure controlled to compensate for changes in the ambient temperature. To generate a pressure that is proportional to the square root of the gas temperature, a mass of gas is trapped in a compliant space having a volume directly proportional to pressure. The equation for an ideal compliant volume is

$$V = C P_0 \quad (1)$$

¹ Choked flow in an orifice occurs when the gas velocity at the point of minimum flow area is equal to the speed of sound. Under these conditions, changes in downstream gas state cannot be detected upstream of the orifice so that flow no longer depends on downstream load conditions.

High-Pressure
Oxygen Tank

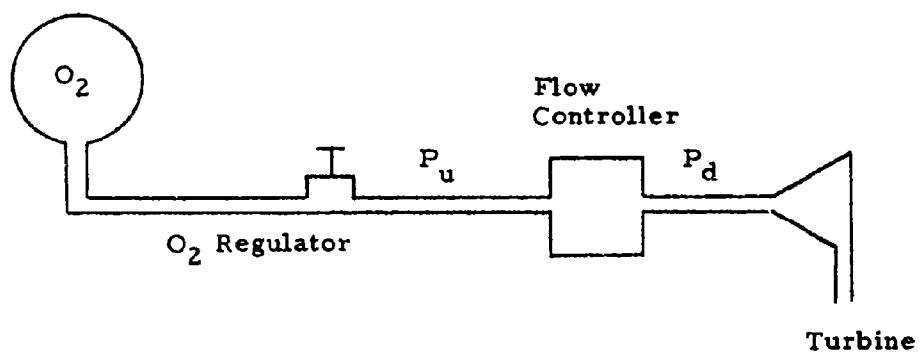


Figure 1. Oxygen Flow Circuit

where C = compliance of the elastic volume containing the trapped gas
 P_{θ} = gas pressure
 V = gas volume

From the perfect gas law, P_{θ} is given by

$$P_{\theta} = \frac{m R \theta}{M V} \quad (2)$$

where m = mass of trapped gas
 M = molecular weight of the gas
 R = universal gas constant
 θ = temperature of the gas

If Equations (1) and (2) are combined and solved for P_{θ} , then

$$P_{\theta} = \sqrt{\frac{m R}{M C}} \sqrt{\theta} \quad (3)$$

The mass, m , of gas contained in the trapped volume is determined by the pressure, $P_{\theta f}$, and temperature, θ_f , when volume, V , is filled and sealed. If Equation (2) is evaluated for these conditions, solved for m , and combined with Equation (3), then

$$P_{\theta} = \left(\frac{P_{\theta f}}{\sqrt{\theta_f}} \right) \sqrt{\theta} \quad (4)$$

In practice, there will always be some parasitic volume, V_o , when the gas pressure, P_{θ} , is zero. To compensate for V_o , it is necessary to subtract a volume from V by inserting a volume of material, V_m , into the space occupied by the gas. The equation for V then becomes

$$V = C P_{\theta} + V_o - V_m \quad (5)$$

To generate a pressure proportional to the square root of temperature as shown in Equation (4), the volume, V_m , must be precisely equal to V_o .

Figure 2 represents a conceptual design of a flow controller using Method I, where pressure, P_{θ} , is proportional to the square root of the temperature of a trapped gas.

The compliant volume (5) is the space swept out by a flat plate (7) when it is deflected by P_{θ} (11). The outer edge of this plate is supported by a cylindrical flexure (13). To compensate for the parasitic volume, V_o , shimstock, representing V_m , is inserted into the space behind the plate

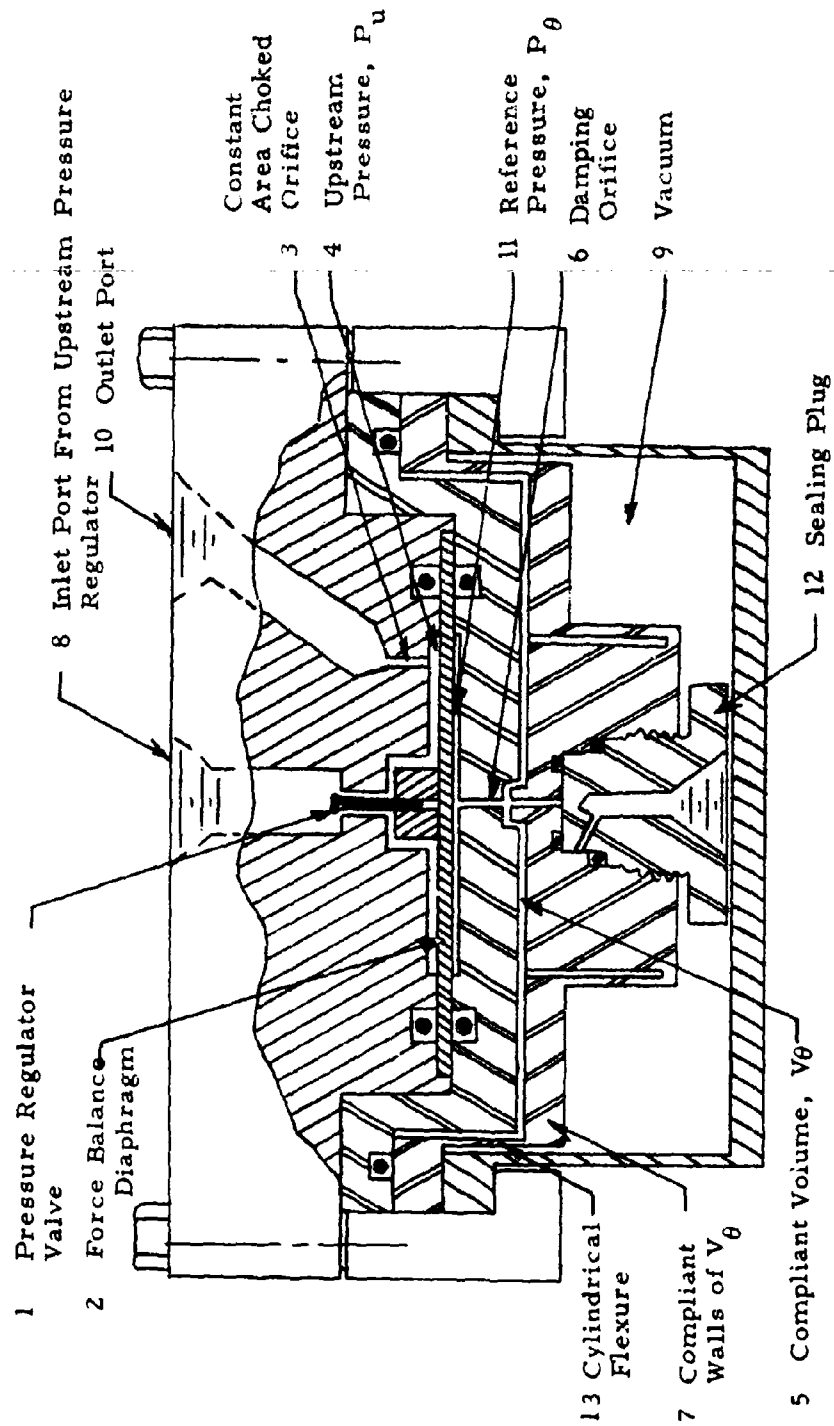


Figure 2 Conceptual Design of a Flow Controller Using Method I

before the device is assembled. The compliant volume is initially filled through a sealing plug (12).

The internal pressure regulator consists of a pressure regulating valve (1) positioned by a force balance diaphragm (2) so that pressure P_u (4) equals the reference pressure, P_θ (11). If P_u does not equal P_θ , the resulting pressure drop across the diaphragm acts to reposition the valve so that this pressure difference is eliminated.

Oxygen from an upstream pressure regulator enters through an inlet port (8), passes through the pressure regulator valve and a fixed metering orifice (3), and finally leaves through an outlet port (10). A space (9) on the outer surface of the plate is evacuated so that changes in ambient temperature and pressure do not affect the compliant volume. Another orifice (6) provides damping to insure stable pressure regulation.

The compliant volume must be large enough for P_θ not to be significantly affected by a deflection of the force balance diaphragm. The force balance diaphragm, in turn, must be large enough that

the spring constant of the diaphragm must not significantly affect the force balance, and adequate damping must be available

Calculations indicated that the overall size of this device would be excessive and it would be extremely sensitive to minute leakage from the compliant volume. To eliminate leakage by welding the volume shut, would make the parasitic volume compensation extremely inconvenient to perform. An additional inconvenience stems from the requirement for a vacuum seal to be maintained around the compliant volume.

For these reasons and its generally complex nature, this flow control method was not chosen for implementation.

METHOD II

This method uses a fixed restriction and maintains the gas temperature and pressure constant. However, since potentially large ambient temperature changes must be accommodated, thermal control must be provided by some external means. This approach is not desirable for the required application and was not given further consideration.

METHOD III

This method is based upon the concept of supplying the gas through a variable restriction (choked orifice) having a predictable and well-controlled pressure-flow characteristic. The restriction is regulated in a prescribed manner to maintain a constant flow rate by compensating for changes in gas temperature and pressure.

An orifice restriction has the great advantage of being practically independent of viscosity. Laminar restrictions were eliminated from consideration because they are difficult to control and are proportional to viscosity which varies by 30% over the ambient temperature range of interest. Turbulent pipe flow restrictions were also eliminated because the required long length of extremely fine diameter tube was considered impractical.

The mass flow rate of a perfect gas expanding isentropically through a choked orifice is given by the following relationship: *

$$w = \frac{K A P_u}{\sqrt{\theta}} \quad (6)$$

where

w	=	mass flow rate
A	=	orifice area
P_u	=	gas pressure upstream of the orifice
θ	=	gas temperature upstream of the orifice
K	=	a proportionality constant dependent upon orifice geometry, gas molecular weight and ratio of specific heats.

Perfect temperature and pressure compensation can be obtained if A is made to vary as

$$A = \frac{C_1 \sqrt{\theta}}{P_u} \quad (7)$$

where C_1 is a proportionality constant. Unfortunately, it is mechanically difficult to implement Equation (7) since the relationship between A , θ , and P_u is nonlinear.

Because of its relative ease of mechanical implementation, the following approximation to Equation (7) was considered.

$$A = A_n + K_\theta (\theta - \theta_n) - K_p (P_u - P_{un}) \quad (8)$$

or

$$\Delta A = K_\theta \Delta \theta - K_p \Delta P_u \quad (9)$$

where subscript n refers to the nominal design conditions, and

* Blackburn, J. F., Reethof G., and Shearer, J. L., "Fluid Power Control", Technology Press of MIT and John Wiley and Sons, Inc., New York, 1960.

θ_n = midpoint of the ambient temperature range

ΔA = change in A from A_n

$\Delta \theta$ = change in θ from θ_n

ΔP_u = change in P_u from P_{un}

K_θ = proportionality constant between change in area and change in temperature

K_p = proportionality constant between change in area and change in upstream pressure.

If K_θ and K_p are set equal to

$$K_\theta = \frac{1}{2} \frac{A_n}{\theta_n} \quad (10)$$

$$K_p = \frac{A_n}{P_{un}} \quad (11)$$

then the fractional change in flow becomes (see Appendix)

$$\frac{\Delta w}{w_n} = \frac{1}{8} \left(\frac{\Delta \theta}{\theta_n} \right)^2 - \left(\frac{\Delta P_u}{P_{un}} \right)^2 + \frac{1}{2} \left(\frac{\Delta \theta}{\theta_n} \right) \left(\frac{\Delta P_u}{P_{un}} \right) + \epsilon \quad (12)$$

where

w_n = design value of flow

Δw = change in flow from w_n

ϵ = sum of negligible higher order terms

From the flow controller requirements, the maximum allowable fractional change in flow and ambient temperature are

$$\left(\frac{\Delta w}{w_n} \right)_{\max} = \pm 0.0093$$

$$\left(\frac{\Delta \theta}{\theta_n} \right)_{\max} = \pm 0.22$$

A good quality pressure regulator, when used in the flow circuit of Figure 1, should be capable of maintaining P_u to within 2% of P_{un} . Equation (12) solved for these values indicates that the predicted fractional change in flow is less than the specified maximum allowable error. Consequently, this method of linearly varying the orifice area in response to changes in θ and P_u was selected as the basis for the ultimate design of the required flow controller.

SECTION III

PRELIMINARY DESIGN OF A FLOW CONTROLLER USING METHOD III

DESIGN PROBLEMS

The major design challenge arises from the extremely small required value of A_n . Equation (6) solved for an upstream pressure of 450 psig indicates that $A_n = 0.000094$ square inches. This is the area of a hole 0.0113 inches in diameter. To achieve $\pm 1\%$ accuracy, the flow controller must be capable of controlling changes in A that are 1% of A_n . A preliminary study indicated that the most promising way to accomplish this would be with orifice geometries controlled by a precise linear motion. One configuration that was considered uses an annular orifice controlled by the motion of a tapered rod inside a sharp edged hole. A small (e. g. 5°) taper angle allows the use of axial motions substantially larger than the desired change in radial gap. However, to keep the annulus thickness larger than 0.001 inches, the rod diameter cannot exceed 0.020 inches. This small diameter was considered undesirable for machining a precision taper angle.

To increase the rod diameter, the geometry was modified to include a sharp edged orifice and a straight rod with a flat on its periphery. Because of the close diametral fit between the rod and orifice, most of the flow passes through the small space or gap created by the flat and the edge of the orifice. Since the flat is precision ground at a slight inclination to the axis of the rod, the gap between the flat and the edge of the orifice becomes proportional to the position of the rod. The orifice area will also be affected by minute changes in the width of the flat inside the orifice. However, an approximate error analysis (see Appendix) indicated that this affects the flow control accuracy by only 1%. Consequently, for small motions of the rod, changes in orifice area are virtually proportional to changes in the rod position. Since the modified geometry allowed a threefold increase in the rod diameter, it was selected for the final flow controller design.

To implement the area variation described by Equation (9), the controller must produce a linear motion having components proportional to ambient temperature and upstream pressure. The pressure dependent motion is obtained by allowing P_u to deflect a diaphragm. Several configurations using metallic thermal expansion were considered for producing a motion proportional to temperature. However, because of large size and uncertainties in the coefficient of thermal expansion, this concept was discarded in favor of using a trapped volume of gas.

If a fixed mass of gas is trapped in a constant volume, the pressure of this gas, according to Equation (2), is directly proportional to the gas temperature. (For the mathematical treatment of this concept see Method I under Section II). If P_g acts on a diaphragm, a motion proportional to the temperature of the gas is obtained.

A study was made of ways to obtain the desired linear motion with different diaphragm configurations. The selected method uses a single diaphragm pressurized by P_θ on one side and P_u on the opposite side. The resulting net pressure force acts against the diaphragm spring constant to produce a linear motion which moves the control rod in the orifice, thereby implementing Equation (9).

PRELIMINARY DESIGN

A preliminary design of a flow controller using Method III is shown in Figure 3. Flow at pressure P_u enters the inlet port (5), passes through the flow control orifice, and leaves by the exit port (6) at pressure P_d . The flow control orifice consists of the orifice plate (2) containing the sharp edged hole, and the control rod (1) with an inclined flat machined on its periphery. The rod is spring loaded against the diaphragm (3) to insure that its axial motion follows that of the diaphragm. The right side of the diaphragm is exposed to the pressure P_u immediately upstream of the variable orifice. The figure shows that an increase in P_u deflects the diaphragm to the left decreasing the orifice area A . Conversely, a decrease in P_u deflects the diaphragm to the right increasing A . The left side of the diaphragm is exposed to P_θ , the pressure of the trapped volume of gas (4). Since the volume swept out by the diaphragm deflection is negligible compared to the remaining volume, the volume of trapped gas can be considered to be constant, and P_θ becomes proportional to the temperature of the trapped gas.

Because the thermal time constant of the flow controller is much smaller than the time scale of the ambient temperature fluctuations, the trapped gas temperature will be equal to the ambient temperature. Since the gas does no work and has negligible heat transfer while flowing between the storage tank and the controller, it will be at ambient temperature at both locations. Consequently, P_θ , and therefore the diaphragm deflection are directly proportional to the gas temperature upstream of the flow control orifice. An increase in ambient temperature increases P_θ which deflects the diaphragm to the right increasing A . Conversely, a decrease in ambient temperature results in a reduction in A .

This discussion demonstrates that the flow controller configuration shown in Figure 3 is capable of producing an orifice area variation qualitatively similar to that described in Equation (9). To meet the flow control requirements, the gain values given by Equations (10) and (11) must be established. This can be accomplished by satisfying the following two relationships:

$$\alpha = \tan^{-1} \frac{A_n}{P_{un} C S} \quad (14)$$

and

$$P_{\theta n} = \frac{P_{un}}{2} \quad (15)$$

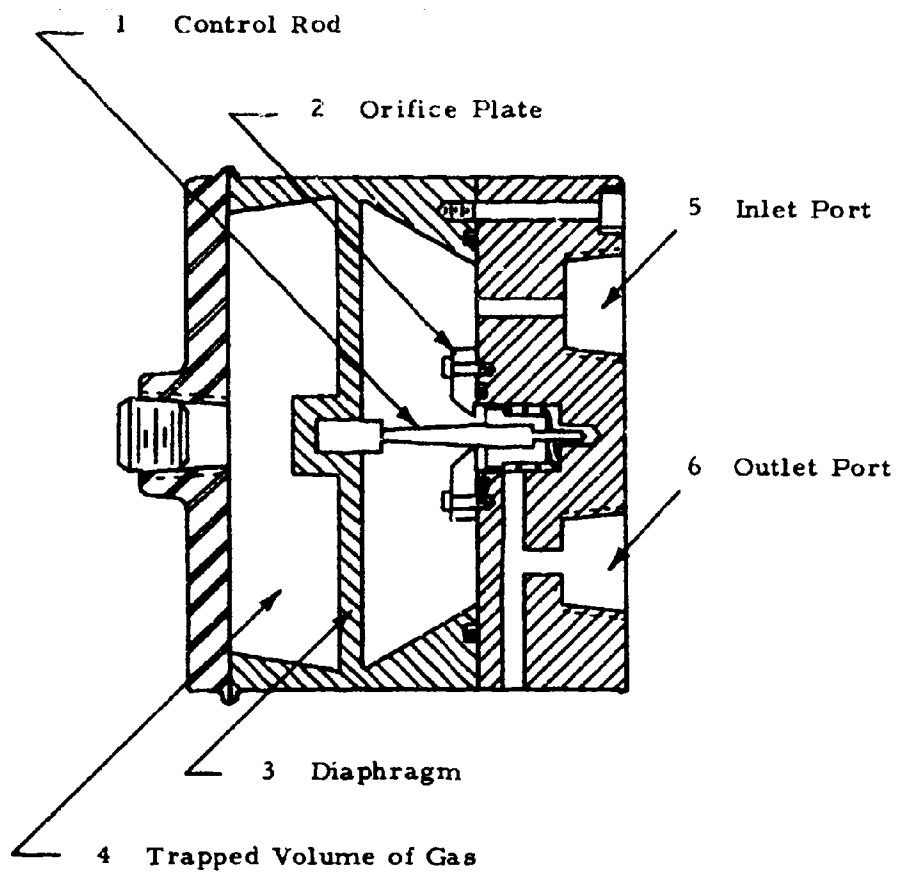


Figure 3. Preliminary Design of a Flow
Controller Using Method III

where

- α = inclination angle of the control rod flat
- A_n = nominal orifice area
- P_{un} = nominal upstream pressure
- C = diaphragm compliance
- S = rate of change of orifice area with respect to gap height
- $P_{\theta n}$ = nominal pressure of trapped gas at temperature, θ .

DERIVATION OF CRITICAL DESIGN VALUES

The flow controller using Method III maintains a constant flow by adjusting its control orifice area to compensate for changes in ambient temperature and upstream pressure. The relationships between these variables may be derived in terms of the physical characteristics of the flow controller, which, in turn, may be used to derive the desired values of control rod inclination and trapped volume gas pressure given in Equations (14) and (15).

CONTROL ROD INCLINATION ANGLE (α)

Figure 4 shows a cross sectional view of the control orifice in a plane perpendicular to the control rod axis. The orifice area consists of two parts:

- . the variable area bounded by the flat and the dotted line, and
- . the constant area annulus having a radial clearance g .

The height of the variable area is h and the total orifice area is given by

$$A = 2 \pi r g_o + r^2 (\beta - \sin \beta \cos \beta) \quad (16)$$

where $\beta = \cos^{-1} \left(\frac{r-h}{r} \right)$.

For small changes in h , changes in orifice area are given by

$$\Delta A = S \Delta h \quad (17)$$

where

- ΔA = change in area A
- Δh = change in h
- S = dA/dh , the rate of change of area with respect to h .

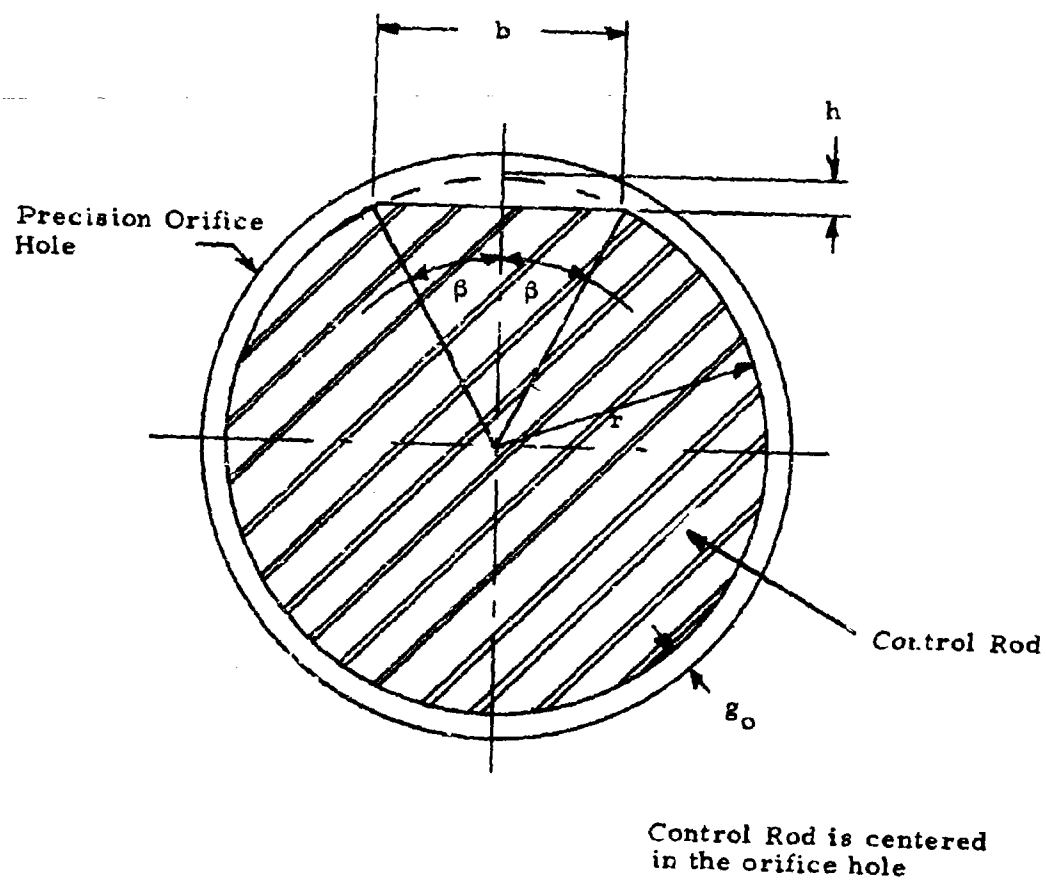


Figure 4. Control Orifice Area

An approximation to S can be computed with the aid of Equation (16). The change in h is related to an axial motion of the control rod, ΔZ , as follows:

$$\Delta h = -\Delta Z \tan \alpha \quad (18)$$

where α is the angle between the flat on the rod and the rod axis. The relationship between ΔZ and a change in differential pressure across the diaphragm is

$$\Delta Z = C \Delta P \quad (19)$$

where C = compliance of the diaphragm

ΔP = change in differential pressure

If the change in differential pressure, ΔP , results from a change in upstream pressure, ΔP_u , then a combination of Equations (17), (18) and (19) yields

$$\Delta A = S \tan \alpha C \Delta P_u \quad (20)$$

or,

$$\frac{\Delta A}{A_n} = - (S \tan \alpha C \frac{P_{un}}{A_n}) \frac{\Delta P_u}{P_{un}} \quad (21)$$

The desired relationship between ΔA and ΔP_u is obtained if the quantity in brackets in equation (21) is unity. This requirement, expressed by equation (14), determines α .

NOMINAL PRESSURE IN TRAPPED VOLUME ($P_{\theta n}$)

Since the pressure in the trapped volume, P_{θ} , is proportional to temperature, θ , changes in these quantities, from a nominal design point, are designated as ΔP_{θ} and $\Delta \theta$, respectively, and are related as follows:

$$\Delta P_{\theta} = P_{\theta n} \frac{\Delta \theta}{\theta_n} \quad (22)$$

where subscript, n, refers to nominal design values. A variation in ambient temperature causes a differential pressure, ΔP , across the diaphragm. Combining Equations (18), (19), (20) and (23) results in the following incremental change in orifice area:

$$\Delta A = (S \tan \alpha C) (P_{\theta n} \frac{\Delta \theta}{P_{un}}) \quad (23)$$

A combination of Equations (23) and (14) yields

$$\frac{\Delta A}{A_n} = \left(\frac{P_{\theta n}}{P_{un}} \right) \quad (24)$$

Ambient temperature compensation is obtained by making the fractional change in area equal to one-half the fractional change in temperature. This is accomplished by the implementation of Equation (15), repeated below

$$P_{\theta n} = \frac{1}{2} P_{un}$$

This value of $P_{\theta n}$ was used to set the pressure in the trapped volume to provide the desired thermal compensation.

SECTION IV

FINAL DESIGN OF THE FLOW CONTROLLER

The final design of the flow controller is shown in Figure 5. It consists of a precision orifice assembly held by a housing and controlled by a diaphragm assembly.

PRECISION ORIFICE ASSEMBLY

The precision orifice assembly (Figure 5) consists of the following components: an orifice plate (2) with its sharp edged precision hole, a control rod (1) with an inclined flat on its periphery, a spacer rod (6) that fits into a hub in the diaphragm (3), a sleeve (7) which fits over the control rod, and a spring (8) that engages the sleeve and is compressed against the housing (4).

The orifice plate is shown in Figure 6. The sharp edged orifice hole is 0.0601 inches in diameter and has a land less than 0.005 inches long. This short land insures an orifice-type flow restriction independent of viscosity at the design Reynold's number of 38,000. The land also supports one end of the control rod. The orifice plate is held to the housing by four countersunk 2-56 screws with split lockwashers. A counterbore in the housing provides radial location for the plate.

The control rod is shown in Figure 7. It has a diameter of 0.0600 inches, and the 0.00005 inch radial clearance between it and the orifice plate hole results in an area only 10% of the nominal orifice area of 1.02×10^{-4} square inches. The rod has a 0.060 inch long flat ground on its periphery at an inclination angle of 12.7° to its axis. This flat is deepest at the end of the rod that touches the spacer rod. When the control rod is centered within the hole and at the axial position corresponding to nominal downstream pressure and ambient temperature, the gap between the center of the flat and the edge of the orifice hole is 0.0044 inches. At this point on the rod, the flat is 0.031 inches wide. Since the angle subtended by the machined portion of the rod periphery is 62° , radial motion of the rod away from the hole centerline is limited to approximately twice the radial clearance, or 0.0001 inches. This motion does not change the orifice area, and the slight geometrical change is not expected to significantly alter the flow pattern.

The end of the rod opposite the flat is supported inside a hole in the housing. Near this support point, a shoulder is machined on the rod to engage the sleeve. The preloaded spring is compressed inside the housing and pushes with a 4 lb force against the flange on the sleeve forcing the rod to maintain contact with the spacer rod. The spacer rod is nominally 0.27 inches long and slides within a $3/32$ inch counterbore in the diaphragm hub. The desired orifice area can be set by grinding or lapping the spacer rod length to change the axial position of the control rod. This sets the desired flow rate.

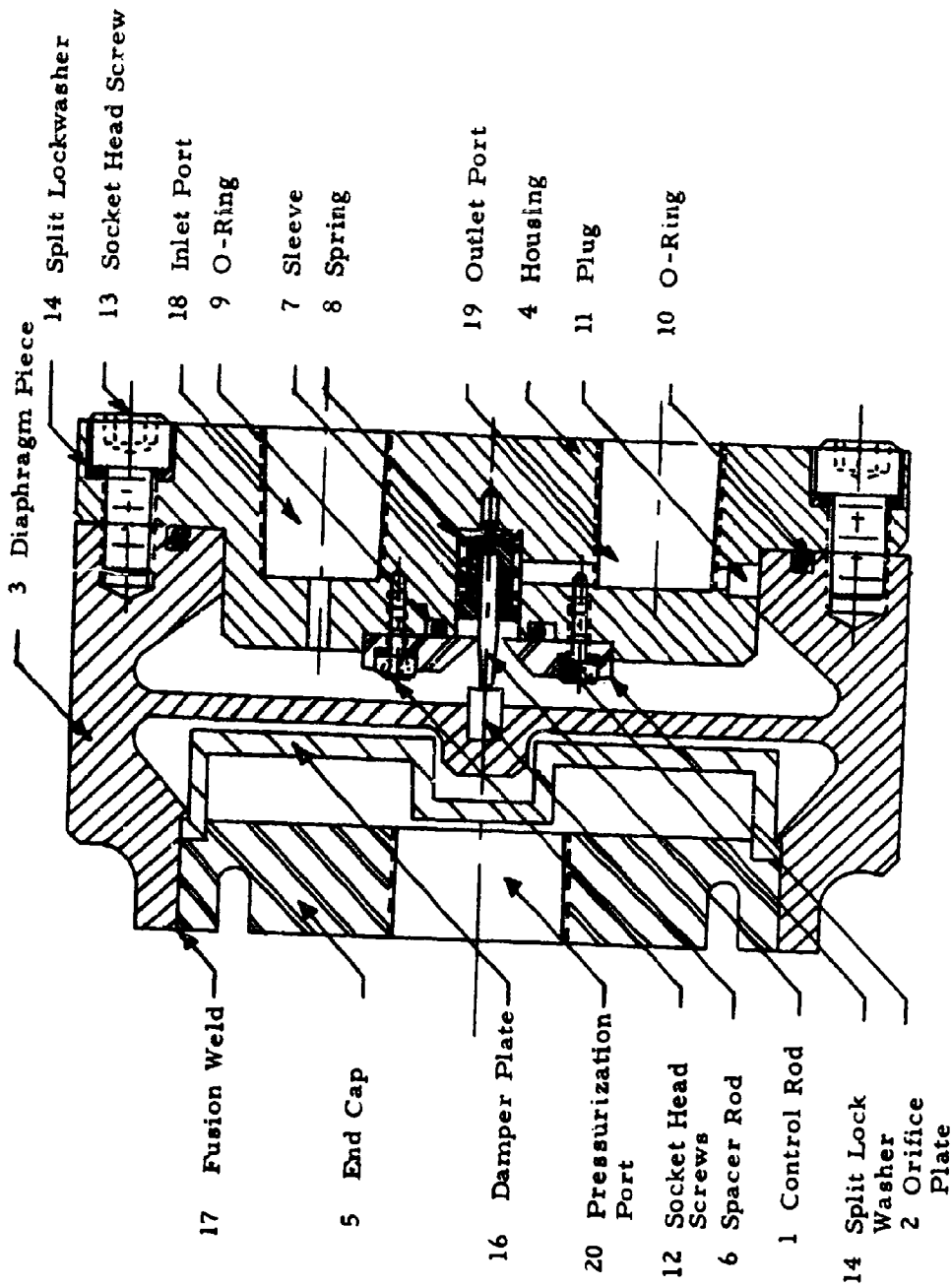
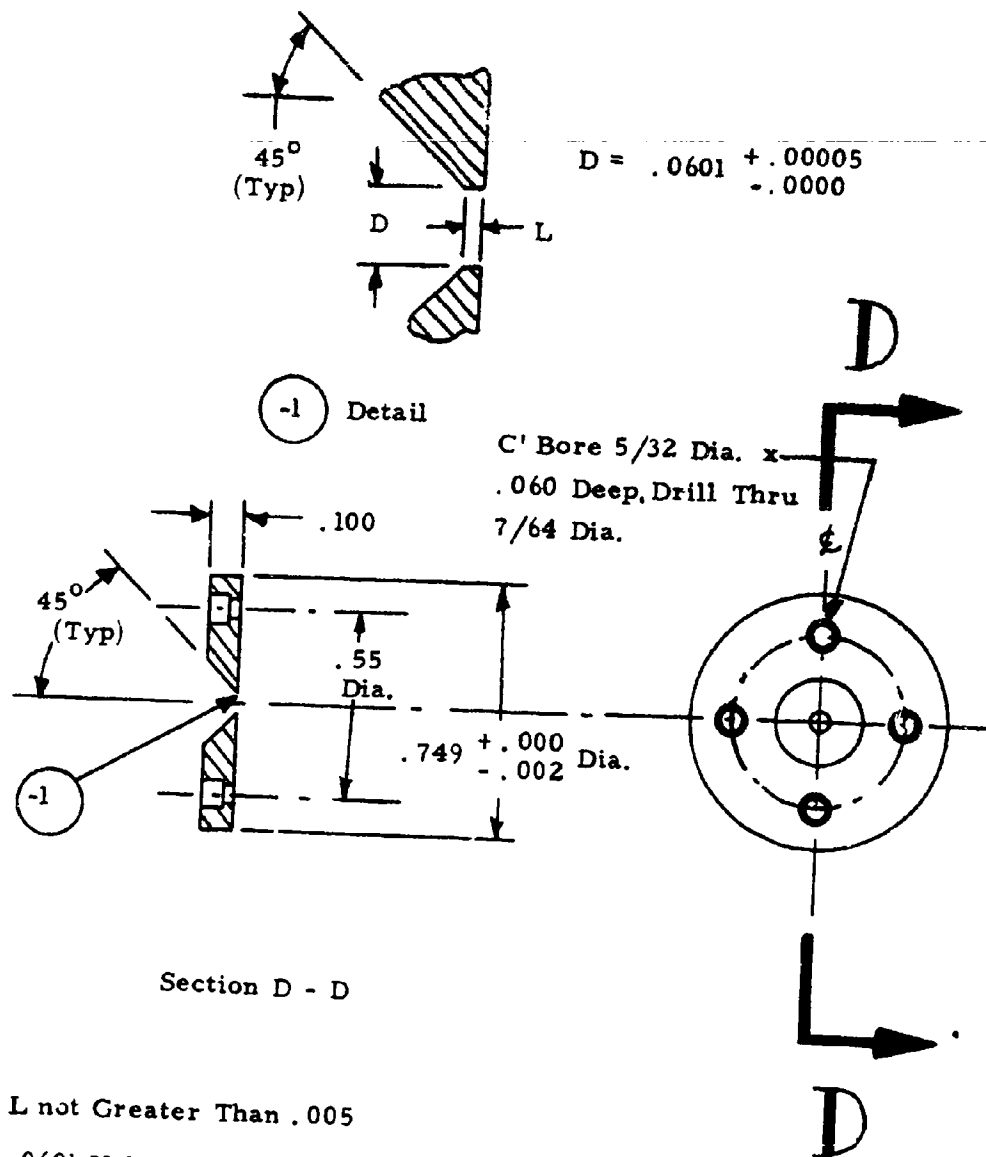
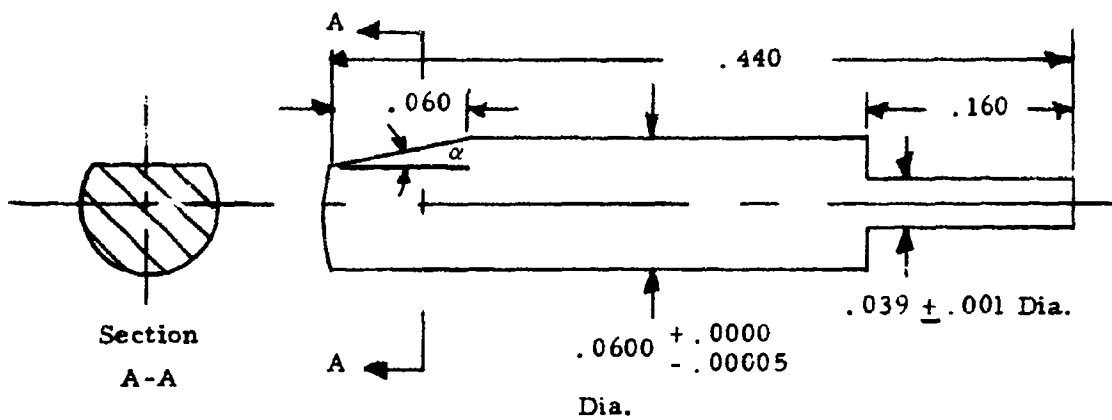


Figure 5. Assembly Drawing of Flow Controller



- 1 L not Greater Than .005
- 2 .0601 Hole to be Concentric with O. D. to Within .002 T. I. R.
- 3 Material - Tungsten Carbide

Figure 6. Orifice Plate



$$\alpha = 12.7^{\circ}$$

Figure 7. Control Rod

HOUSING

The 2-1/2 inch diameter cylindrical housing fits into the diaphragm and seals off the area adjacent to the diaphragm. The housing contains 1/8 NPT pipe tap inlet and outlet ports both of which are offset from the centerline by one-half inch. The spring and the sleeve are housed inside a 3/16 inch counterbore containing a deeper concentric 1/16 inch counterbore for supporting the control rod.

After passing through the flow control orifice, the oxygen flows into the 3/16 inch counterbore past the sleeve and spring and enters a 1/8 inch diameter hole leading to the outlet port. The far end of this hole is sealed off by a plug (11).

The housing is attached to the diaphragm piece by six 8-32 counter-sunk socket head screws (13) with split lockwashers (14). The connecting joint is sealed by an O-ring (10). Another O-ring (9) prevents the gas flow from bypassing the flow control orifice. For experimental convenience, elastomeric O-rings were used at these two face seals. If the O-ring groove depths are slightly reduced, the elastomeric O-rings can be replaced by metal O-rings for optimum compatibility with oxygen.

DIAPHRAGM ASSEMBLY

The cylindrical diaphragm assembly consists of a diaphragm piece, an end cap and a damper plate (16). The diaphragm piece contains a diaphragm that is machined integral with a mounting surface for the housing and a lip for welding to the end cap. The diaphragm diameter and thickness were selected on the basis of a design compromise among the desired qualities of large deflection, low stress and small diameter. For the selected thickness of 0.050 inches and a diameter of 2-1/8 inches, the predicted stress is 72,000 psi and the deflection is 0.011 inches. A 1/4-inch diameter hub in the center of the diaphragm contains a receptacle for the spacer rod.

The trapped volume for thermal compensation is obtained by fusion welding the end cap to the diaphragm piece. The volume is filled through a 1/4-inch diameter copper tube hard soldered to a stainless steel pipe plug. The plug threads into a 1/4 NPT pipe tap (20) in the end cap. After the volume is pressurized, the tubing is temporarily sealed with a pinch-off tool. The excess tubing is cut away and the exposed tube end is hard soldered for a permanent seal. If required, subsequent pressurizations can be performed with a new pipe plug and tubing assembly without disturbing the end cap or its weld joint. For mounting the flow controller, four 8-32 tapped holes equally spaced in a 1-inch diameter bolt circle are provided on the end cap.

The damper plate is attached to the end cap with a light press fit prior to welding. It is held in place inside the trapped volume by shoulders

on the diaphragm piece and end cap. The purpose of this plate is to damp out diaphragm oscillations in the unlikely event that an instability arises from the interaction of the flow controller and the upstream pressure regulator. A minute 0.014 inch diameter orifice hole in the center of the plate is provided to dissipate fluid energy arising from gas displaced by diaphragm oscillations. To maintain a high damper cutoff frequency, the damper plate is contoured to fit around the diaphragm hub, resulting in a small volume between the plate and the moving diaphragm.

SECTION V

FABRICATION OF FLOW CONTROLLER

MATERIALS OF FABRICATION

The flow controller is fabricated from nontoxic materials that will not corrode, peel, crack, or chip when used with high pressure oxygen. Metallic materials are used throughout the device with the exception of two elastomeric O-rings that can be replaced with metal O-rings if a minor modification to the O-ring grooves is made. The flow controller is shown in Figures 8 and 9.

The control rod and orifice plate are made of tungsten carbide because this material has exceptional wear and corrosion resistance and dimensional stability. These properties are very desirable because of the short land of the sharp edged orifice, the close diametral fit between the control rod and the hole in the orifice plate, and the extreme accuracy required for the flat on the periphery of the rod.

The diaphragm piece is made from Ni-Span-C nickel steel alloy. This material when properly cold worked and heat treated, has a nearly zero thermoelastic coefficient,² resulting in negligible change in diaphragm stiffness throughout the specified ambient temperature range. If it were made from conventional stainless steel, this stiffness would vary several per cent over the operating temperature range so that acceptable flow control accuracy could not be maintained. To avoid thermal expansion effects other major parts of the flow controller are also made of Ni-Span-C. Stainless steel is used for the spring, all screws and lockwashers, and the pipe plug in the end cap.

FABRICATION PROCEDURES

The Ni-Span-C alloy was obtained in the form of 25% cold worked bar stock. The diaphragm piece was rough machined, heat treated at 800 F in hydrogen for 5 hours, slowly cooled, and finish machined. This procedure produces a thermoelastic coefficient of $2 \times 10^{-6}/F$ which results in a 0.05% change in Young's Modulus over the complete ambient temperature range.* The 0.2% offset yield strength obtained with the heat treatment is 140,000 psi. The other Ni-Span-C parts were not heat treated since the value of their thermoelastic coefficient is not critical. After final machining, all Ni-Span-C pieces were given a 0.0001 inch thick electrolysis nickel plating for added corrosion resistance.

² Rate of change of Young's Modulus per unit change in temperature, expressed in parts per million per degree F.

* "Engineering Properties of Ni-Span-C Alloy 902" Technical Bulletin T-31, Huntington Alloy Products Division of the International Nickel Co. Inc., Huntington, West Virginia, 1963.



Figure 8. Disassembled Flow Controller

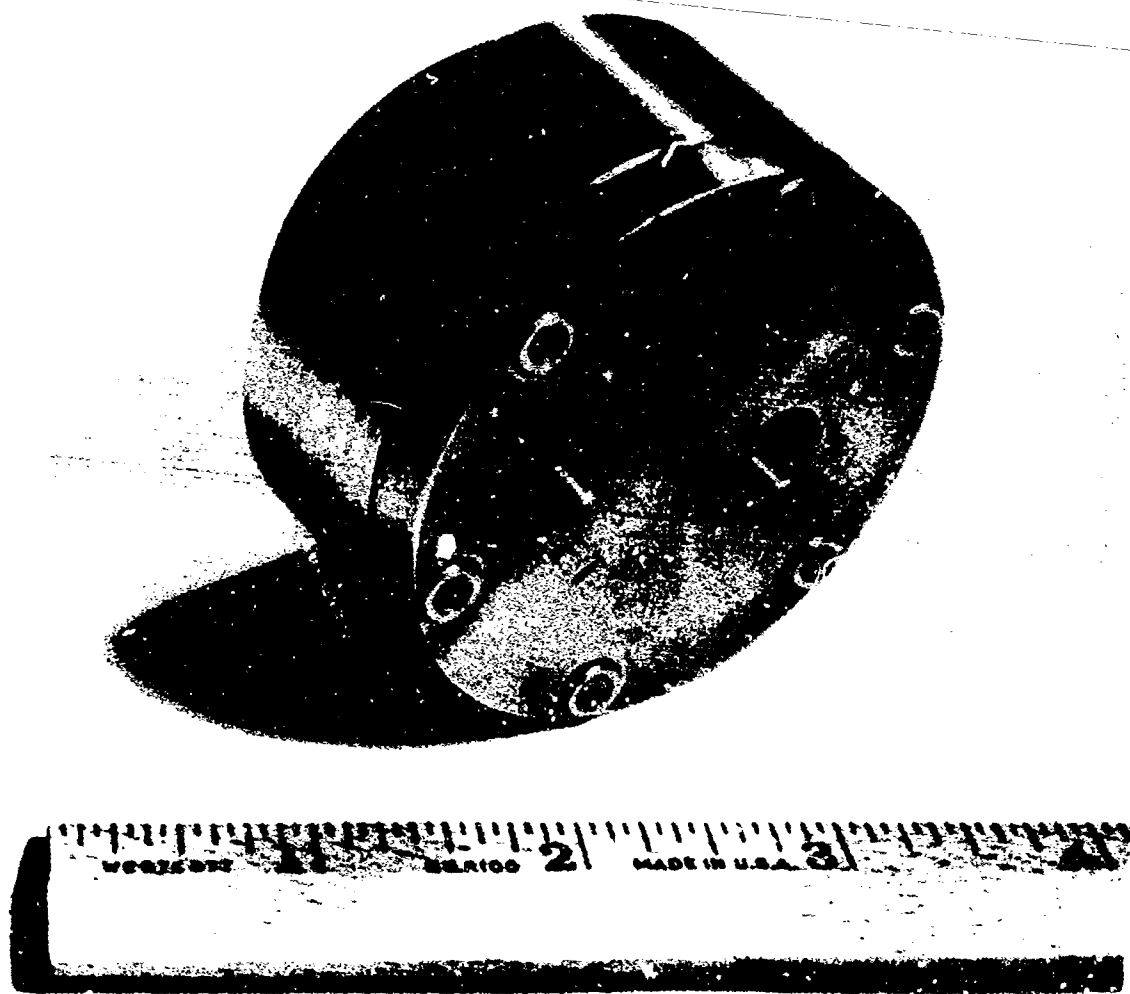


Figure 9, Assembled Flow Controller
25

The end cap was attached to the diaphragm piece by fusion welding. To minimize the possibility of high temperatures altering the desired thermoelastic coefficient or causing thermal distortion of the diaphragm, the weld lip on the diaphragm piece was machined on a thick portion of the flange extending from the root of the diaphragm. This extra mass of metal reduced the rate of temperature rise by acting as a heat sink, and allowed the fusion welding process to be completed before the flange reached an undesirably high temperature.

SECTION VI

FLOW CONTROLLER TESTS

COMPONENT EVALUATION

The individual flow controller components were separately evaluated to insure proper operation and to provide information to complete the flow controller design prior to final assembly and test.

To complete the flow controller design, it was necessary to determine the desired value of α , the inclination angle of the flat on the control rod. From Equation (14), α depends on C , the diaphragm compliance, and A_n , the nominal orifice area. However, the nominal orifice area required to pass the flow is dependent on C_d , the orifice discharge coefficient. Therefore, it was desirable to measure C and C_d to analytically determine α and obtain the proper proportionality between diaphragm deflection and change in orifice area. In addition, the other critical values determined were

- . leakage from the trapped volume
- . effects of internal flow passages, and
- . spacer length required to set the design flow rate.

DIAPHRAGM COMPLIANCE

The diaphragm compliance was determined by measuring the deflection caused by a known differential pressure applied across the diaphragm. The housing and diaphragm pieces were disassembled, and a 0.0001 inch displacement indicator was placed in contact with the spacer rod. Pressures between 0 and 250 psi were applied to the compliant volume, V_0 , and measured with a precision bourdon gage. Diaphragm deflections induced by three cyclic loadings were repeatable and linear to within ± 0.0004 inch. This small deflection, which corresponds to a 0.25% change in orifice area, is within the experimental uncertainty of the measurements. The diaphragm compliance, C , from these measurements, was determined to be 0.346×10^{-4} in./psi. This value is 15% less than the predicted value. The discrepancy is probably due to a combination of the diaphragm thickness being several thousandths of an inch greater than desired, and Young's Modulus being slightly larger than predicted.

ORIFICE DISCHARGE COEFFICIENT

The discharge coefficient of the choked flow control orifice can be determined by measuring the upstream pressure, temperature, mass flow rate, and the orifice area. By substituting the equivalence, $C_d C_2$, for the proportionality constant, K , in Equation (6) and solving for C_d , we have

$$C_d = \frac{w\sqrt{\theta}}{C_2 A P_u} \quad (25)$$

where $C_2 = 0.56 (^{\circ}F)^{1/2}/\text{sec. for oxygen.}$

Unfortunately, because it was not possible to obtain an accurate determination of A, the orifice area, C_d could not be successfully measured. The uncertainty in the value of A was due to a 0.001 inch uncertainty in the axial position of the control rod relative to the knife edge orifice hole and a 0.0002 inch uncertainty in the diametral clearance between the rod and hole. Since these uncertainties could not be resolved within the scope of the present contract, C_d was estimated to be 0.75.

TRAPPED VOLUME LEAKAGE

To determine if there was gross leakage from the flow controller, the outlet port was plugged, the trapped volume and inlet port were pressurized to 500 psig, and the unit immersed in water. The absence of bubbles indicated that the O-ring seal between the housing and diaphragm piece and the weld joining the end cap to the diaphragm piece were satisfactory. To detect smaller leakage rates from the trapped volume, it was necessary to monitor P_{θ} over an extended time interval. A precision bourdon pressure gage was temporarily attached to the tee fitting in the tubing used to pressurize the compliant volume. No measurable leakage could be detected during a two week period. With the pressure gage removed, it is possible to monitor P_{θ} to approximately 1% accuracy by disassembling the housing from the diaphragm piece and measuring the distance between the center of the diaphragm and the mounting surface. Significant changes in P_{θ} will cause diaphragm deflections that measurably alter this distance.

EFFECT OF INTERNAL FLOW PASSAGES

If choked conditions exist in the control orifice, the flow rate will not be influenced by changes in the downstream pressure, P_d . To establish choked conditions, the pressure immediately upstream of the control orifice must be greater than 1.89 times the pressure immediately downstream of the control orifice. If any internal pressure losses occur in the flow passages upstream or downstream of the orifice, the pressure ratio across it will be less than the ratio of P_u/P_d , and under certain circumstances, the flow might unchoke.

The flow rate was measured at room temperature and a 450 psig upstream pressure for different downstream pressures ranging from 0 to 200 psig. The flow rate increased by 1% when P_d was lowered to 175 psig, and 2.3% when P_d was 75 psig. No increase in flow rate was experienced with further decreases in P_d . This effect could likely be eliminated in future work by enlarging all internal flow passages. For final testing, the flow was set to its nominal value when the downstream pressure was 200 psig.

SPACER ROD LENGTH REQUIRED TO ESTABLISH NOMINAL FLOW RATE

The nominal flow rate was set by a trial and error process. Initially, the spacer rod was intentionally made too long, and the resultant enlarged orifice area passed too high a flow. The spacer rod length was gradually lapped or reduced until the controller passed the desired flow.

Flow rate was measured by determining the pressure and temperature of the gas upstream of a specially prepared calibrated orifice. A precision bourdon pressure gage and a mercury thermometer inserted in the gas stream were used to determine these quantities. The accuracy and resolution of these measurements are estimated to be 1% and 0.5%, respectively.

PRELIMINARY TEST

Based upon the measured value of diaphragm compliance, and the estimated value of the orifice discharge coefficient, the desired value of the inclination angle, α , was determined to be 12.7° . A tungsten carbide rod was machined with a flat inclined at this angle. Preliminary room temperature testing of the flow controller was performed with nitrogen gas. The flow rate was measured for different supply pressures to determine if α had been set at the proper value. A 5.5% increase in P_u resulted in a 0.37% increase in flow and a 5.5% decrease in P_u caused a .18% increase. Because of the 0.25% uncertainty in flow measurements, it was not possible to analytically predict a more accurate value of α . Time did not permit an extended trial and error test with different values of α . Since 12.7° appeared reasonably close to the optimum value, it was used for all subsequent testing.

FINAL TEST

Procedure

The flow controller was finally tested with oxygen at ambient temperatures of -40 F, +70 F and +200 F. The supply pressure was varied $\pm 2.25\%$, and $\pm 4.5\%$ about a nominal design value of 450 psig. To simulate the downstream load, a needle valve was placed at the outlet of the flow controller in the temperature environment. This valve was adjusted to make the downstream pressure 200 psig at the nominal design conditions of 70 F ambient temperature and 450 psig upstream pressure. Once this initial adjustment was made, the valve position was not changed for the remainder of the tests. A bourdon gage was attached to the trapped volume of gas during initial pressurization so that P_0 could be monitored.

The experimental setup consisted of three separate temperature conditioning chambers. The first chamber controlled the inlet oxygen temperature, the second the temperature of the flow controller body, and the third the oxygen temperature at the inlet to the flow meter.

These temperatures were sensed with thermocouples and continuously monitored with a potentiometric recorder. During tests, the flow controller body temperature was maintained within 1 C of the oxygen inlet temperature, and the flow meter inlet temperature within 1 C of its calibration temperature.

All pressures were measured with precision bourdon gages. The nominal flow rate was measured with a laminar flow element having an inclined water manometer readout. Due to the small flow rate, the manometer readings were only a small fraction of its full scale. Consequently, meniscus and parallax effects resulted in a measurement uncertainty of approximately 1%. To increase flow measurement accuracy, a specially prepared calibrated orifice was used together with a high resolution bourdon pressure gage. This procedure enabled the detection of 0.25% fluctuations in flow rate.

Results

A summary of the flow rates obtained at the different temperatures and upstream pressures is given in Table I. When the upstream pressure was maintained at its nominal value of 450 psig, the flow increased by 2.5% at -40 F and 2.1% at +200 F. At each temperature, $\pm 2\%$ changes in upstream pressure resulted in less than $\pm 0.75\%$ changes in flow; and $\pm 4.5\%$ changes in pressures caused less than 1.25% change. For $\pm 2\%$ changes in upstream pressure the maximum and minimum flows were 2.766 lb/hr and 2.667 lb/hr, respectively. This is a slightly higher level of flow and a larger spread than required. A slight reduction in spacer rod length will reduce the level of flows.

The above accuracy was obtained without tuning of the control rod inclination angle, since only one angle value was used throughout the test program. Significant improvement in flow control accuracy might result from an intensified effort to determine the effect of control rod inclination angle on flow control accuracy.

TABLE I
FLOW CONTROLLER TEST RESULTS

Oxygen Flow Rate in lbs/hour for various gas temperatures and upstream pressures.

Upstream Pressure, P_u , psig	Gas Temperature, θ , Degrees Rankine		
	420	530	660
430	2.780	2.683	2.734
440	2.766	2.683	2.734
450	2.746	2.678	2.734
460	2.730	2.667	2.724
470	2.715	2.657	2.719

SECTION VII

CONCLUSIONS AND RECOMMENDATIONS

An oxygen flow controller using passive temperature and pressure compensation was developed and operated satisfactorily. The following concepts for use in a low flow gas controller were successfully demonstrated:

- . Achieving temperature compensation by means of a trapped volume of gas which acts on a diaphragm to produce a control valve motion
- . Accurate control valve provided by the axial motion of a precision rod with an inclined flat moving through a knife-edged orifice
- . Use of a diaphragm to superpose the effects of upstream pressure variation and temperature variation, providing the correct control valve position for constant flow

Flow control accuracy of $\pm 2\%$ was obtained using the inclination angle for the rod flat. To obtain maximum flow control accuracy, tuning for the optimum inclination angle would be required. This would require a trial and error test program.

The following recommendations are made for future work using the developed flow controller in its present form;

- . determine the effect of control rod inclination angle on flow control accuracy. Maximum accuracy could be achieved by empirically selecting the optimum value of the angle.
- . modify the spacer rod length to provide the range of flow rates required
- . enlarge the internal flow passages to insure choked conditions at the control orifice for all conditions.

In addition, the weight for future flow controllers using this design concept could be reduced by allowing higher stress levels in the housing pieces, eliminating the damper plate, and reducing the weld lips. These measures should result in weight savings of over 50%.

APPENDIX

FLOW CONTROL ERRORS AND PROPORTIONALITY CONSTANTS

This appendix derives the flow control errors that result from using the linear orifice area variation of Equation (7), instead of the ideal nonlinear area variation of Equation (8) required for perfect flow control.

The set point mass flow rate through the control orifice is given by

$$w_n = \frac{K A_n P_{un}}{\sqrt{\theta_n}} \quad (26)$$

This flow is established by setting P_u to its nominal value, and then adjusting area, A , until the flow is at its nominal value when θ is at the midpoint of the ambient temperature range. If P_u and θ change from their nominal values by ΔP_u and $\Delta \theta$, the area will change from its nominal value by ΔA , and a change in flow rate, Δw , will occur. For this condition, Equation (26) can be rewritten as

$$w_n (1 + \epsilon_w) = \left(\frac{K A_n P_{un}}{\sqrt{\theta_n}} \right) \left(\frac{(1 + \epsilon_A)(1 + \epsilon_p)}{\sqrt{1 + \epsilon_\theta}} \right) \quad (27)$$

or

$$(1 + \epsilon_w) = \frac{(1 + \epsilon_A)(1 + \epsilon_p)}{\sqrt{1 + \epsilon_\theta}} \quad (28)$$

where the fractional changes in flow rate, area, upstream pressure, and temperature are expressed as ϵ terms defined as follows:

$$\begin{aligned} \epsilon_w &= \frac{\Delta w}{w_n} \\ \epsilon_A &= \frac{\Delta A}{A_n} \\ \epsilon_p &= \frac{\Delta P_u}{P_{un}} \\ \epsilon_\theta &= \frac{\Delta \theta}{\theta_n} \end{aligned}$$

The square root term in Equation (28) can be rewritten using the binomial expansion as follows:

$$\frac{1}{\sqrt{1 + \epsilon_\theta}} = 1 - \frac{1}{2} \epsilon_\theta + \frac{3}{8} \epsilon_\theta^2 + \epsilon_1 \quad (29)$$

where ϵ_1 is a sum of higher order terms in ϵ_θ .

If Equation (29) is substituted into Equation (28) and the multiplications performed, the fractional error in flow is given by

$$\epsilon_w = \epsilon_A + \epsilon_p - \frac{1}{2} \epsilon_\theta + \frac{1}{8} \epsilon_\theta^2 - \epsilon_p^2 + \frac{1}{2} \epsilon_\theta \epsilon_p + \epsilon \quad (30)$$

where ϵ is a sum of negligible higher order terms in ϵ_θ and ϵ_p .

The flow controller is designed to vary the control orifice area so that

$$\epsilon_A = -\frac{1}{2} \epsilon_\theta - \epsilon_p \quad (31)$$

Combination of Equations (30) and (31) yields Equation (12), repeated below.

$$\frac{\Delta w}{w_n} = \frac{1}{8} \left(\frac{\Delta \theta}{\theta_n} \right)^2 - \left(\frac{\Delta P_u}{P_{un}} \right)^2 + \frac{1}{2} \left(\frac{\Delta \theta}{\theta_n} \right) \left(\frac{\Delta P_u}{P_{un}} \right) + \epsilon$$

Comparison of Equations (8) and (31) yields the values of K_θ and K_p given in Equations (10) and (11), repeated below.

$$K_\theta = \frac{1}{2} \frac{A_n}{\theta_n}$$

$$K_p = \frac{A_n}{P_{un}}$$

The above expressions were used to adjust the flow controller orifice area variation to provide maximum compensation for changes in upstream pressure and ambient temperature.

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The purpose of this work was to develop a small, lightweight oxygen flow control device able to deliver 2.65 ± 0.05, -0.00 lb/hr oxygen at a nominal outlet pressure of 200 psig when supplied by an upstream pressure regulator and operated in ambient temperatures from -40°F to $+200^\circ\text{F}$. The flow controller meters the oxygen through a variable area choked flow orifice. The orifice assembly consists of a rod, spring-loaded against a diaphragm and allowed to move axially inside a sharp edged hole. The variable orifice is formed by the gap between the edge of the hole and a flat machined on the periphery of the rod and inclined to the rod axis. Axial motion of the rod varies the gap, and therefore the orifice area. One side of the diaphragm is exposed to the upstream pressure of the orifice, and the opposite side is exposed to a trapped constant volume of gas having a pressure proportional to the gas temperature. The diaphragm deflection and the orifice area are linearly related to the pressure and temperature of the gas at the device inlet. Proper selection of the critical design values enables the variation of orifice area to compensate for changing inlet conditions. The flow controller weighs 1.5 lbs, is 2.5 inches in diameter and is 1.5 inches long. It maintains flow constant to within ± 0.05 lb/hr over the specified ambient temperature range in the presence of $\pm 2\%$ changes in upstream pressure. This accuracy might be improved with a program of trial and error "tuning" of the control rod inclination angle.		

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14. KEY WORDS	LINK A		LINK B		LINK C	
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