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SERVOVALVE CONTROLLED MECHANISM

Filed Sept. 2, 1960

2 Sheets-Sheet 1

Fig. 1

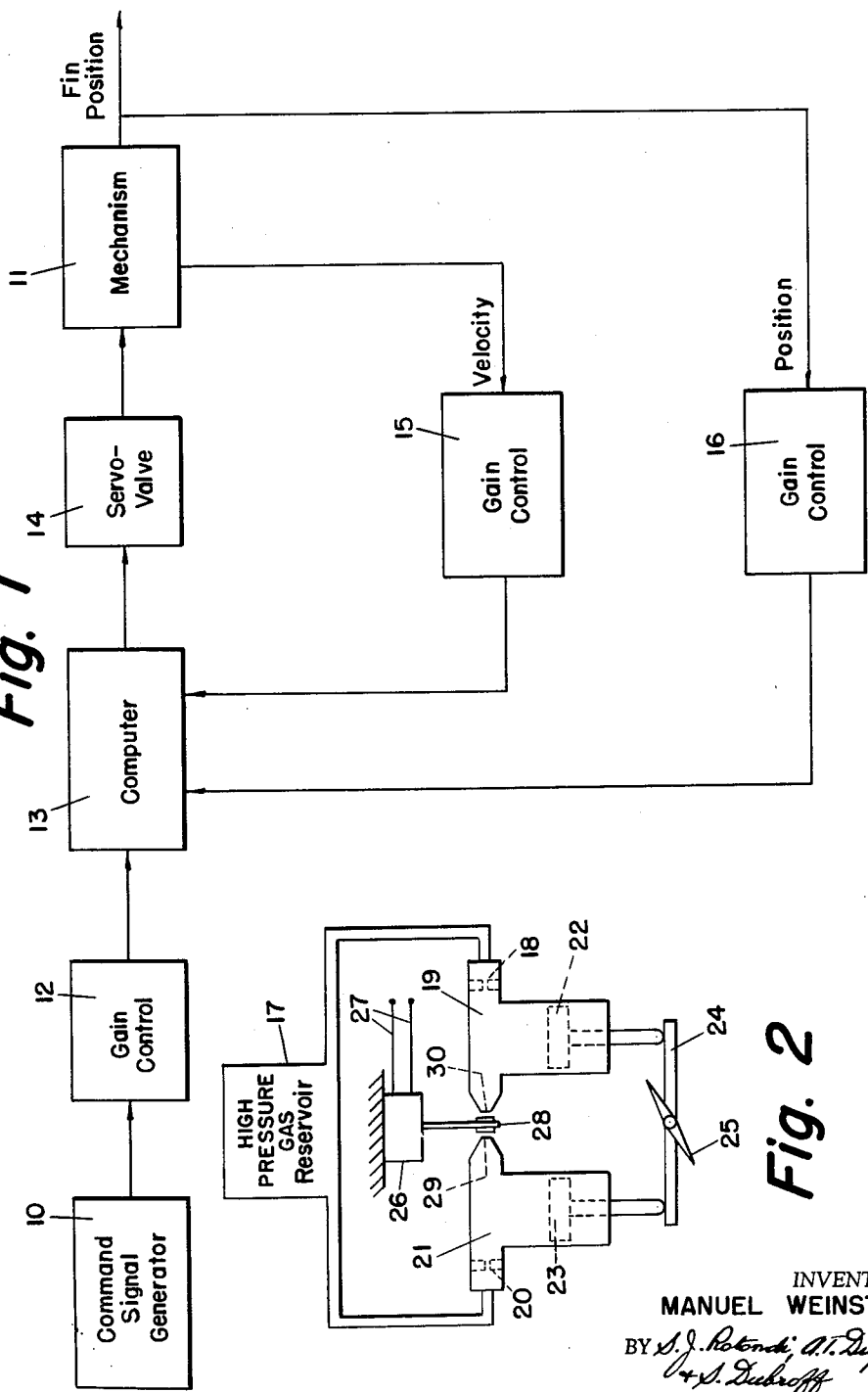


Fig. 2

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Fig. 3

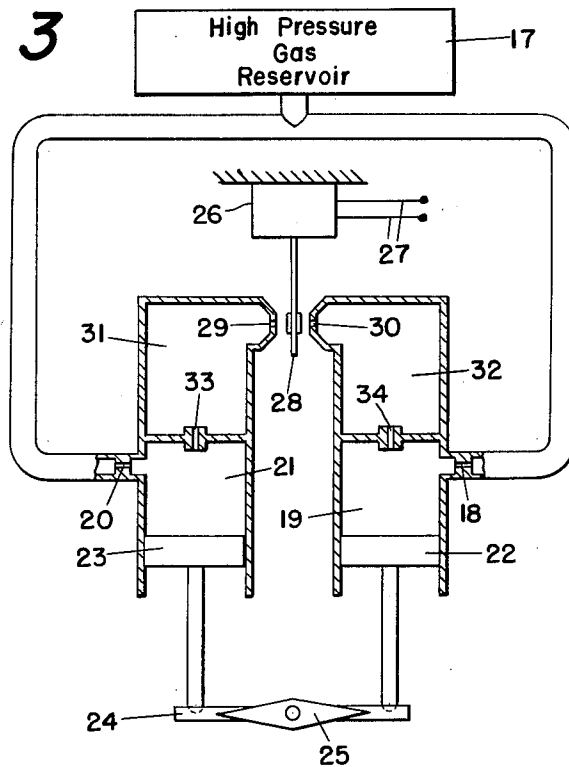
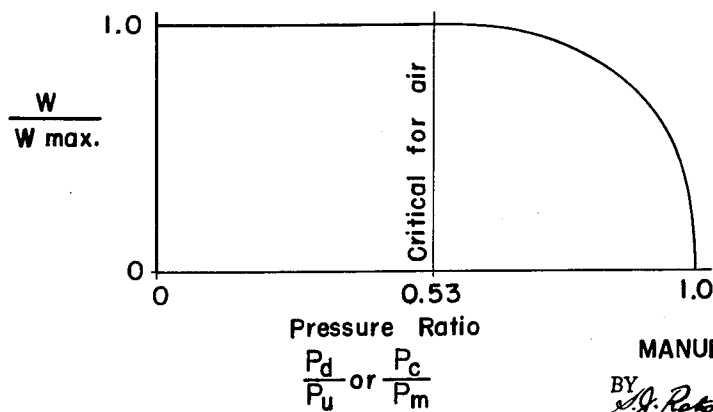


Fig. 4



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SERVOVALVE CONTROLLED MECHANISM

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2 Claims. (Cl. 121-41)

(Granted under Title 35, U.S. Code (1952), sec. 266)

The invention described herein may be manufactured and used by or for the Government for governmental purposes without the payment to me of any royalty thereon.

This invention relates to mechanisms which are operated by a pneumatic or gas pressure, and more particularly to a pneumatic pressure operated mechanism which is controlled by a servovalve motivated in accordance with a command signal.

Presently available is a load positioning system wherein a source of pneumatic pressure is connected to a pair of motor chambers having outlet ports, the effective areas of these ports being controlled by a member which is positioned in accordance with the command signal. This well known system has the disadvantage that its operation involves a continuous discharge of working gas through the outlet ports.

The present invention minimizes this discharge of working gas by interposing between the motor chambers and the signal positioned member control chambers which open into the motor chambers through orifices operating at greater than critical pressure ratios. The net results of this change in the construction of the mechanism are that (1) the required weight of operating gas is very considerably reduced, and (2) less force is required to position the signal responsive member thereby permitting a considerable reduction in the weight of the electrical components of the mechanism.

The invention will be better understood from the following description when considered in connection with the accompanying drawings and its scope is indicated by the appended claims.

Referring to the drawings:

FIG. 1 is a box diagram depicting a load positioning system in which the mechanism of the present invention may be utilized,

FIG. 2 illustrates a load positioning mechanism on which the present invention is an improvement,

FIG. 3 illustrates the load positioning mechanism of the present invention, and

FIG. 4 is a curve showing the relationship which exists between the weight flow ratio through an orifice and the pressure ratio across that orifice. The symbols of FIG. 4 are defined as follows:

W =weight rate of flow in pounds per second

W_{max} =maximum weight of flow in pounds per second

P_c =control chamber pressure in pounds per square inch

P_d =downstream pressure in pounds per inch

P_m =motor chamber pressure in pounds per square inch

P_u =upstream pressure in pounds per square inch

The curve shows that the weight flow ratio is independent of the pressure ratio (that is—constant) for pressure ratios less than critical and dependent on the downstream pressure for pressure ratios greater than critical.

Briefly described, the fin positioning system of FIG. 1 includes a command signal generator 10 which is connected to a fin positioning mechanism 11 through a gain control device 12, a computer 13 and a servovalve 14. Connected between the mechanism 11 and the computer 13 are a velocity feedback including a gain control device 15 and a position feedback including a gain control device 16. The operation of this system to position the fin of a missile is too well known to those skilled in the

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art to require detailed explanation. It is sufficient to say that the servovalve 14 is motivated by a torque motor which is energized in accordance with a command signal delivered to it from the computer 13.

FIG. 2 illustrates the mechanism 11 as it was constructed prior to the present invention. In this form, it includes a high pressure fluid reservoir 17 which is connected through an orifice 18 to a motor chamber 19, and is connected through an orifice 20 to a motor chamber 21. Pistons 22 and 23 movable in the chambers 19 and 21 respectively, are pivoted to the opposite ends of a rocker arm 24 which is rotatable about its center together with a fin 25.

The servovalve is of the open center type and is motivated by a torque motor 26 to which the command signal is applied through leads 27. It includes a flapper 28 which is positioned between the ports 29 and 30 of the motor chambers in accordance with the signal received through the leads 27. This positioning of the flapper 28 varies the pressures in the motor chambers 19 and 21 so that the pistons 22 and 23 are moved to position the fin 25 in accordance with the received command signal.

As is well known, the operation of the prior art mechanism of FIG. 2 involves a continuous flow of working fluid through the ports 29 and 30 at less than critical pressure ratios P_d/P_u . This continuous flow of working fluid is at the pressure existing in the motor chambers 19 and 21 and has the disadvantage that it results in an unnecessarily large loss of working fluid and imposes on the motor 26 a load larger than would be required if the working fluid was discharged from the ports 29 and 30 at a lower pressure.

The present invention, as exemplified by FIG. 3, minimizes these difficulties by the provision of control chambers 31 and 32. In this arrangement, the chamber 31 is connected between the port 29 and an orifice 33, and the chamber 32 is connected between the port 30 and an orifice 34. As previously indicated, the orifices 33 and 34 are operated with greater than critical pressure ratios P_c/P_m so that any change in the control chamber pressure caused by positioning of the flapper 28 will cause a corresponding change in the motor chamber pressure.

The manner in which this control is achieved can be better understood in connection with FIG. 4 which is a curve of weight flow ratio for air through an orifice as a function of the pressure ratio existing across the orifice. It can be seen from this curve that for pressure ratios greater than critical the weight flow through the orifice is a function of the upstream and downstream pressures. Thus if the orifices 33 and 34 are operated with greater than critical pressure ratios, any change in the pressure within the control chambers 31 and 32 will cause a corresponding change in the pressures within the motor chambers 19 and 21.

The effective areas of the ports 29 and 30 are determined by the position of the flapper 28. When the flapper is in its midposition, the pressures in the control chambers 31 and 32 are equal, the pressures in the motor chambers 19 and 21 are equal and the rocker arm 24 and fin 25 are in their illustrated positions. When the flapper 28 moves away from the port 30, the effective area of this port is increased and the effective area of the port 29 is decreased.

Increase in the effective area of the port 30 (1) increases the flow through port 30, (2) decreases the pressure in the control chamber 32, (3) decreases the pressure ratio across the orifice 34, (4) increases the flow from the motor chamber 19 into the control chamber 32, and (5) decreases the pressure in the motor chamber 19. As a result, the pressure upon the piston 22 is decreased.

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Decrease in the effective area of the port 29 (1) decreases the flow through the port 29, (2) increases the pressure in the control chamber 31, (3) increases the pressure ratio across the orifice 33, (4) decreases the flow from the motor chamber 21 into the control chamber 31, and (5) increases the pressure in the motor chamber 21. As a result the pressure upon the piston 33 is increased.

Similar but opposite effects are produced when the flapper 28 moves from its midposition closer to the orifice 30. In each case, these piston pressures, exerted in push-pull fashion, function to position the rocker arm 24 and fin 25 in accordance with the position of the flapper 28. Since the flapper 28 is positioned in accordance with the command signal applied to the torque motor 26, the fin 25 is also positioned in accordance with the command signal.

The provision of the control chambers 31 and 32, as described above, materially reduces both the forces that must be overcome by the flapper 28 and the weight of the operating gas. This reduces the required weight of the electrical components of the mechanism. Thus the net result is a reduction in both the weight of the fluid required to operate the mechanism and in the weight of the mechanism itself. As can be readily appreciated by those skilled in the art, this is a very important consideration where, as on board missiles it is essential that the weight in excess of the payload be reduced to a

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minimum. While the servovalve has been illustrated as of the open center flapper type, it is to be understood that it may be of any other suitable type, one such type being disclosed in a copending application of Manual Weinstock and Raymond C. Sutter, Serial No. 136,407, filed September 1, 1961 for Rotary Servovalve Controlled Mechanism.

I claim:

1. In a system to be operated by a command signal, a mechanism including means forming a pair of motor chambers, means including a load device operable by a difference between the pressures within said motor chambers, a compressible fluid pressure source connected to said motor chambers through different orifices each of which operates at less than its critical pressure ratio, means forming a pair of control chambers, each having an outlet port and each connected to a different one of said motor chambers through a separate orifice, each of which operates at greater than its critical pressure ratio, means movable to vary the effective areas of said ports, and means for positioning said movable means in accordance with a command signal applied thereto.

2. A mechanism according to claim 1 wherein said movable means is a flapper driven by a torque motor.

References Cited in the file of this patent

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