June 6, 1961

YAO TZU LI

COMBUSTION STABILIZER

2,986,883

Filed Dec. 19, 1955

5 Sheets-Sheet 1

Fig. 1

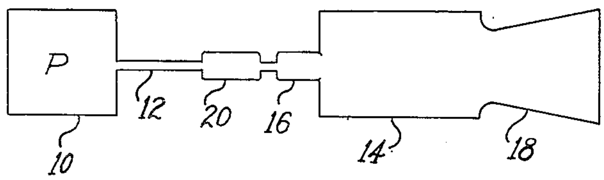


Fig. 2

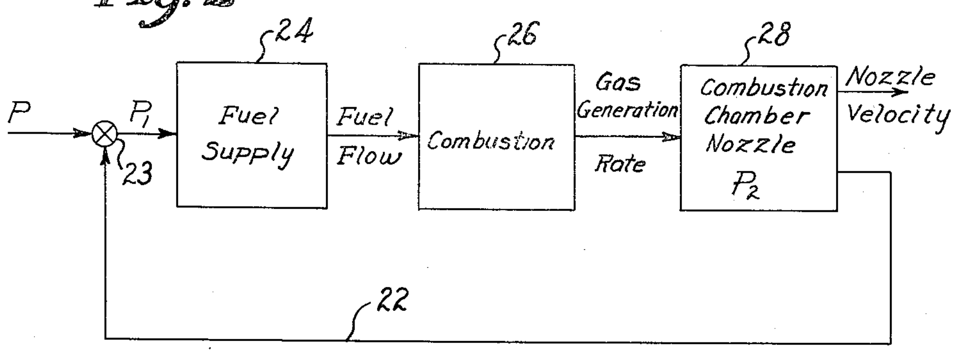


Fig. 2a

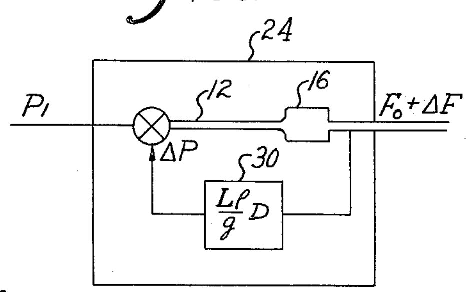
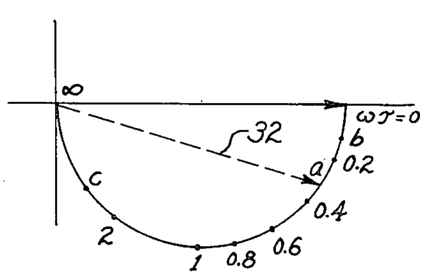


Fig. 3



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June 6, 1961 | YAO TZU LI | 2,986,883

COMBUSTION STABILIZER

Filed Dec. 19, 1955 | 5 Sheets-Sheet 2

Fig. 4

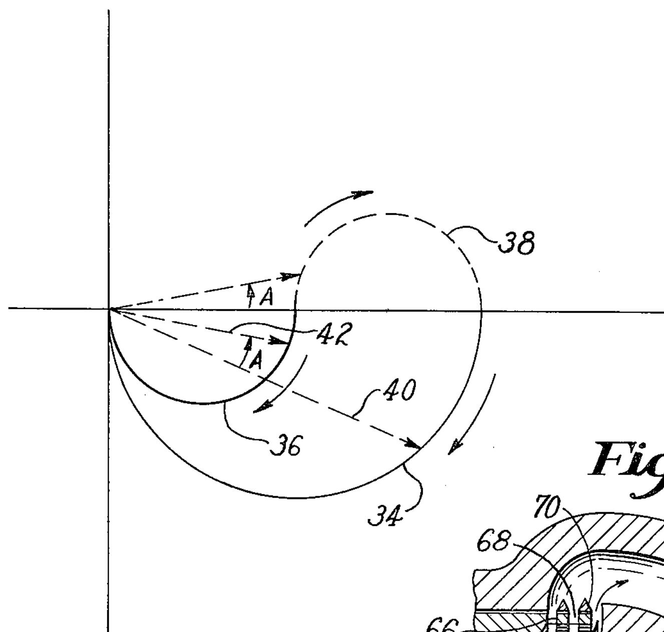


Fig. 9

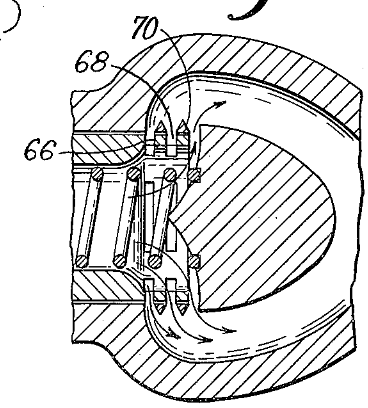
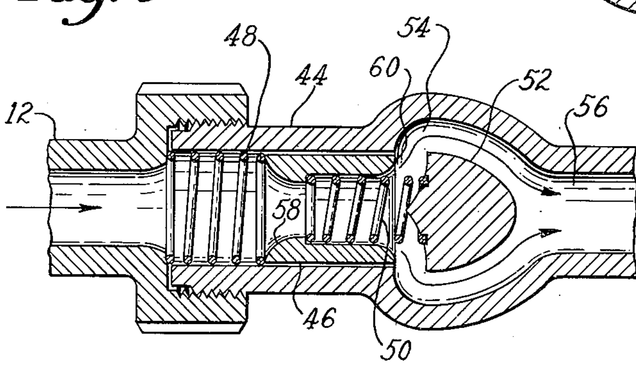


Fig. 5



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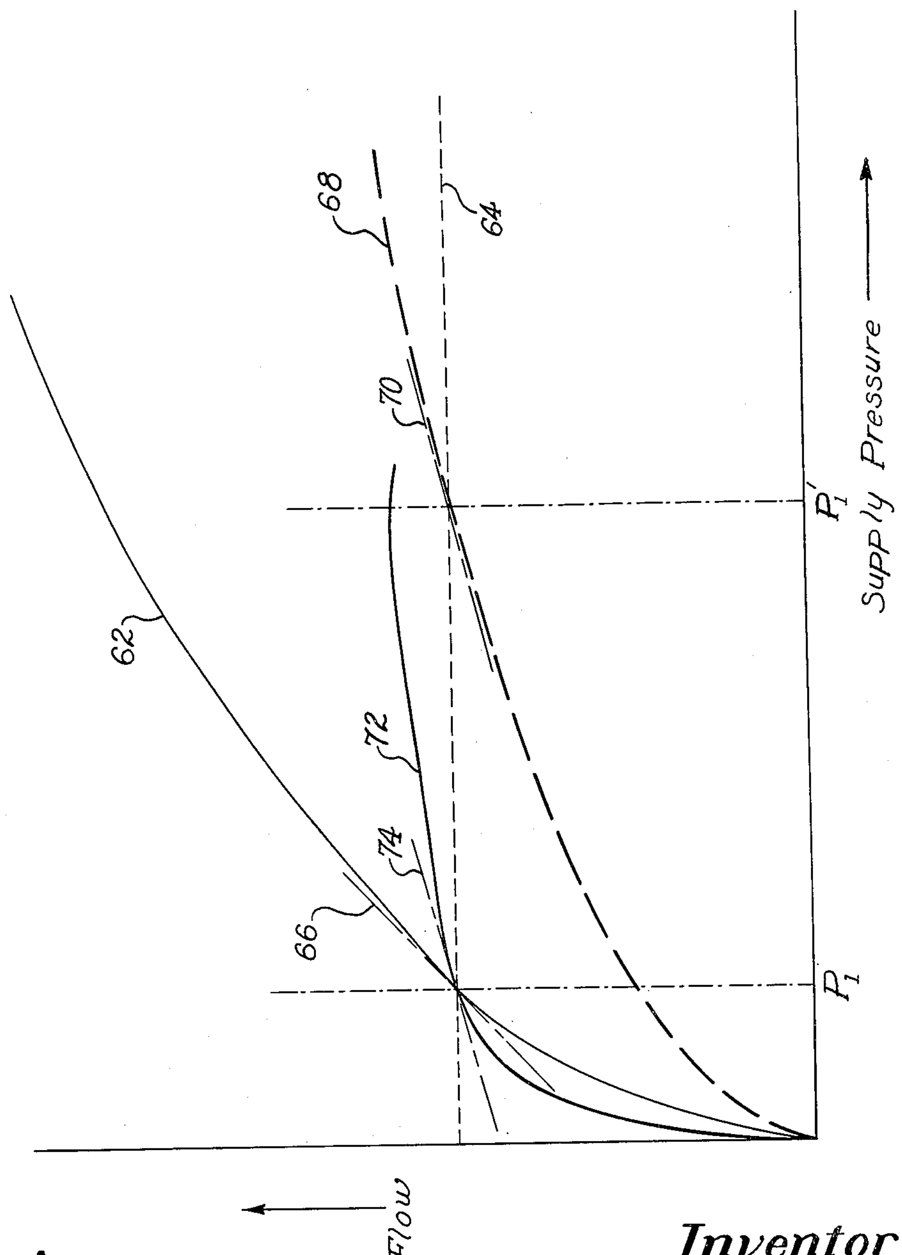

Fig. 6

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Fig. 7

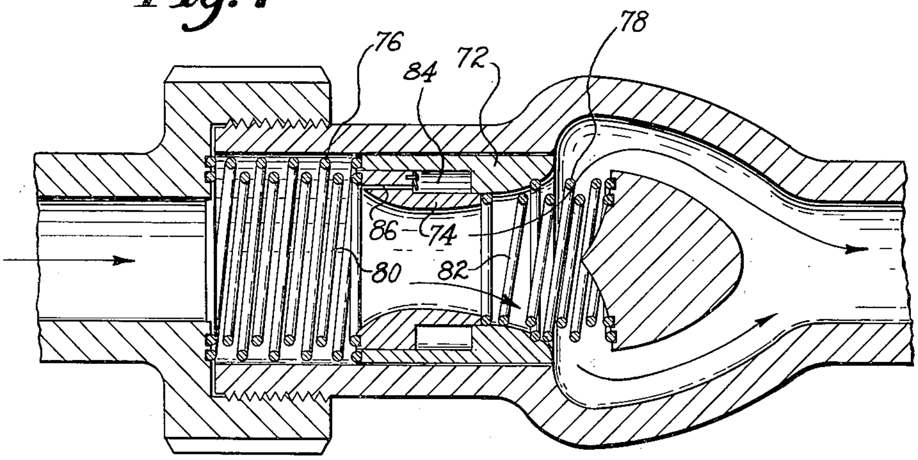


Fig. 7a

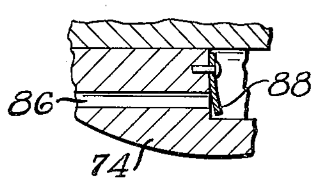
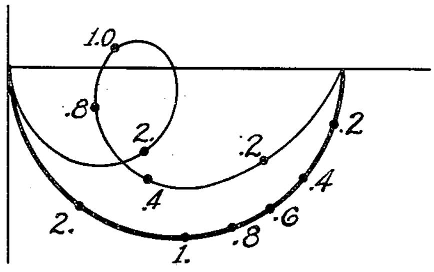


Fig. 11



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2,986,883

COMBUSTION STABILIZER

Filed Dec. 19, 1955

5 Sheets-Sheet 5

Fig. 8

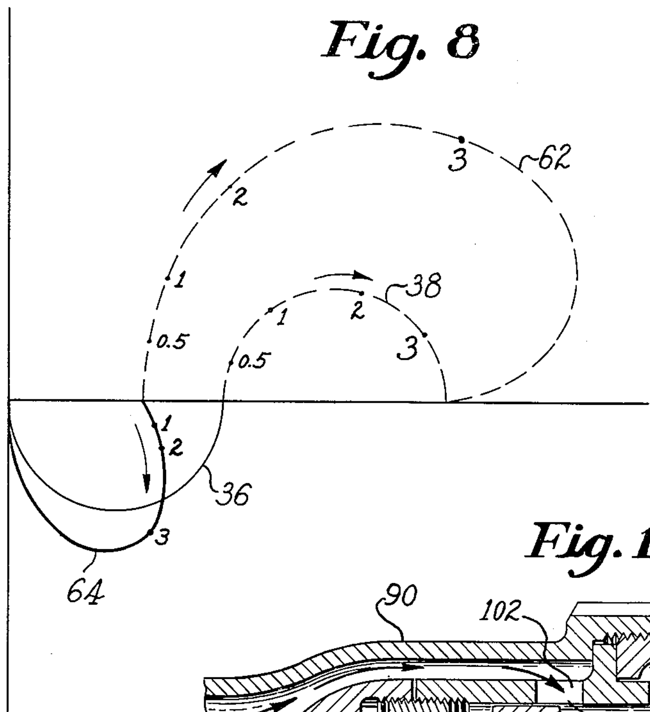
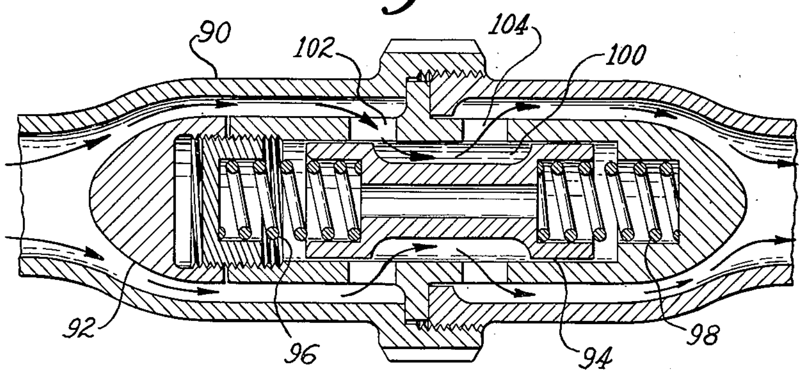


Fig. 10



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COMBUSTION STABILIZER

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Filed Dec. 19, 1955, Ser. No. 553,771
5 Claims. (Cl. 60—39.69)

The present invention relates to apparatus for stabilizing combustion effects and more particularly to apparatus for preventing oscillations in the combustion rate in liquid fuel equipment as, for example, in liquid propellant rockets.

Liquid fluid burners, of which the liquid propellant rocket may be taken as an example, have a tendency toward oscillation because of the coupling between the fuel supply system and the combustion process. In rockets this occurs as low frequency oscillations, commonly known as chugging. The rate of fuel flow between the fuel source and the combustion chamber depends primarily on the pressure difference between the source and the chamber. On the other hand, the gaseous combustion products, because of their great volumetric expansion must be expelled through the nozzle with a high velocity and this requires the building up of a considerable pressure in the combustion chamber itself. Therefore, the rate of gas generation is a function of the fuel flow rate, the chamber pressure is a function of the rate of gas generation, and the fuel flow rate is a function of the chamber pressure. These constitute the characteristics of a "feedback" or closed cycle system. For purposes of theoretical treatment such a system may be considered in the same manner as any feedback system, except that an analysis of the complete rocket system is made very complicated by reason of non-linear effects and because of insufficient knowledge of many of the factors involved.

The low frequency oscillations, or chugging, appearing in the rocket have been recognized as a serious disadvantage, not only because of noise, but because of attendant inefficiency and uncertainty in operation. The oscillations are usually in the neighborhood of a few hundred cycles per second. Various schemes have been proposed in an effort to eliminate the chugging. One such arrangement, based on a theoretical analysis of the entire system, involves the use of a compensating "network" involving a pressure pick-up in the combustion chamber together with a suitable servo to control the fuel pressure. As a practical matter, such arrangements have not been satisfactory because present day servos cannot be made to operate at a sufficiently high frequency to be effective at the chugging frequency. Furthermore the complication of such apparatus introduces serious problems of weight, cost and reliability.

The simplest arrangement for eliminating chugging is to introduce additional restriction into the fuel line, thereby increasing the damping in the system. If the damping is sufficiently great the chugging can be eliminated or reduced, but this requires a high supply pressure. Although this method is widely used in test-stand operation, it is not satisfactory for airborne devices because of the excessive weight and decreased reliability. The effectiveness of the damping method does indicate, however, that it is possible to eliminate chugging by operations acting on the fuel supply part of the system alone, without the necessity for use of elaborate closed-loop compensating mechanisms.

It is therefore the object of the present invention to provide a simple and effective means of stabilizing combustion by elimination of oscillations, especially useful in liquid fuel rockets.

To this end and with other objects in view as will hereinafter appear, the principal features of the invention comprise the introduction of a stabilizer or compensator into the fuel supply line, and responsive to the oscillations in the fuel supply, but without introduction of appreciable resistance to the steady state flow. My improved stabilizer may take either of two forms as will hereinafter be described, one of which responds to the rate of fluid flow and the other to the acceleration of the fluid.

Other features of the invention consist of certain novel features of construction, combinations and arrangements of parts hereinafter described and particularly defined in the claims.

In the accompanying drawings:

FIG. 1 is a diagram of a rocket system.

FIGS. 2 and 2*a* are block diagrams of the system illustrating the conditions by which oscillations occur;

FIGS. 3 and 4 are diagrams showing the general theory of the present invention;

FIG. 5 is a sectional elevation of one form of apparatus according to the present invention;

FIG. 6 is a diagram illustrating the performance of the apparatus of FIG. 5;

FIGS. 7 and 7*a* are sectional elevations of modified forms of the apparatus of FIG. 5;

FIG. 8 is a polar plot showing the operation of the apparatus of FIGS. 5 and 7;

FIG. 9 is a sectional elevation of a further modification of the apparatus of FIG. 5;

FIG. 10 is a sectional elevation of another, and in some respects preferable, embodiment of the invention; and

FIG. 11 is a diagram illustrating the operation of the apparatus of FIG. 10.

Theory

Before going into a description of apparatus according to the present invention I shall first describe the general theory on which my invention depends. In FIG. 1 is shown a diagram of a typical liquid rocket. There is a liquid fuel supply source **10** maintained under a pressure indicated as P, a fuel supply line **12**, a combustion chamber **14**, an injector **16** for injecting the fuel into the combustion chamber, and finally a nozzle **18**. Also shown at **20** in the fuel line is a stabilizer, which, for purposes of general description, may be merely a damper, or one of stabilizers of this invention to be presently described in detail.

For purposes of theoretical analysis there is shown in FIG. 2 a block diagram illustrating the feedback connections in a standard rocket, that is, one not involving the use of the stabilizer **20**. This diagram is similar to the standard analytic diagrams for feedback systems generally and illustrates the coupling arising from the fact that the fuel supply rate depends upon the pressure in the combustion chamber which, in turn, depends upon the fuel flow rate itself. The diagram illustrates that, while the supply pressure is P, the system pressure difference is represented by P_1 and is, in fact, the pressure difference between the supply pressure P and the combustion chamber pressure P_2. The subtraction of the chamber pressure from the supply pressure to obtain the pressure difference P_1 is illustrated diagrammatically by the feedback link **22** and the mixer **23** as is standard in servo analyses. The link **22** therefore represents at least a part of the internal coupling of the system by which oscillations are produced. The fuel flow rate is a function of P_1 and depends upon the performance functions of the several parts of the system. The block **24** represents the fuel supply system itself, namely, the pipe **12** and the injector **16**, and it also involves the mass of the liquid in the supply line.

The block **26** in FIG. 2 represents the combustion characteristics in the chamber **14**. The fuel flow rate, therefore, is a function of the characteristics of the blocks **24** and **26**. The gas generation rate depends not only on the combustion, but also upon the structure of the nozzle and the dynamic characteristics of the combustion chamber and nozzle as indicated by the block **28**. As heretofore stated, the effect of the combustion chamber pressure is manifested through the diagrammatic feedback link **22**.

The complete theoretical analysis of the quantities involved in the diagram of FIG. 2 is exceedingly difficult, if not impossible. However, my invention is concerned only with the characteristics of the fuel supply system itself, as distiguished from the whole system, because as heretofore noted, my principal objective is to correct the chugging condition by operations only on the fuel supply and without the necessity for a complex servo control system. I have discovered in this way that a substantial simplication of the essential theory may be attained.

In FIG. 2*a* I show the internal characteristics of the fuel supply system alone as represented by the block **24** by itself. If the flow were a steady flow only, a flow rate F_o would be obtained. However, because of the mass of the fluid existing in the pipe **12** and the fluctuations in fluid by reasons of the conditions heretofore mentioned, there is a pressure ΔP at chugging frequency and there is a variable flow rate. ΔF superposed on the steady state flow F_o. The conditions are as if there were an internal feedback loop **30** connected from the outlet side of the injector **16** back to the beginning of the line **12**. The characteristics of the feedback loop may be represented by the expression

$$\frac{L\rho}{g}D$$

where D is the differential operator, L is the length of the supply pipe, ρ is the density of the fluid and g is the gravitational constant. In other words, the effect of this loop is to introduce a change of pressure ΔP dependent on the rate of change of fluid flow through the pipeline **12**. If the flow were perfectly steady ΔF would be zero and the differential operator would operate only on the constant F_o and in that case ΔP would be zero. In the actual case the operation of D upon the flow rate results in an acceleration which is converted into pressure by the term

$$\frac{L\rho}{g}$$

In feedback analysis it is customary to obtain a "performance function" involving the frequency scale, using the quantity D as a Laplacian operator. The part-system of FIG. 2*a* is a first order system, in which the performance function (PF) is a simple expression, as follows:

$$(PF)=\frac{S_o}{1+\tau D} \qquad (1)$$

where the sensitivity S_o and the time constant τ are related to the system constants in the following manner:

In general, the sensitivity S is the derivative of the characteristic of flow rate against the pressure difference across the system, namely dF/dP_d, which can be shown to be $F/2P_d$ on the assumption that flow rate is proportional to the square root of pressure difference. S_o represents $F_o/2P_{do}$, namely, the sensitivity at a particular operating point. The time constant is given by the following expression:

$$\tau=S_o\frac{L\rho}{g} \qquad (2)$$

The time constant is therefore a measure of the lag introduced into the system by reason of the fluctuations of the flow in the fuel-supply line.

In keeping with the usual analytical technique of diagramming characteristic functions, I show in FIG. 3 a polar plot of Equation 1 illustrating the performance function for different frequencies. This is in accordance with the usual technique of setting $D=j\omega$, where j is the imaginary operator, and ω is frequency. The plot is made non-dimensional by plotting $\omega\tau$ rather than ω. If there were no oscillations in the line the characteristic would be represented by the arrow lying along the axis of reals. However, the actual characteristic is represented by an arrow such as is shown at **32**, indicating a phase lag at all frequencies other than zero. Since the system plotted in FIG. 3 is a first order system, it does not by itself indicate any oscillation, since a true first order system operating by itself is not oscillatory. However, it is known that the cause of oscillations in any feedback system is the accumulation of lags in the system as a whole (if there is sufficient gain in the system). It is also known from experience that a fuel supply system having the theoretical characteristics of FIG. 3 can result in chugging oscillations of the system as a whole. Such chugging will exist because of the phase lag and gain represented by the characteristic **32** contribute to the phase lags and gains of other parts of the system to satisfy the criteria for oscillation.

The present invention comprises the provision of means for modifying the characteristic of FIG. 3 in such a manner that chugging oscillations will not be maintained in the system as a whole. The invention comprehends two specific embodiments, one of which comprises the insertion of a sufficient phase lead to prevent occurrence of sustained oscillations in the system. The second embodiment of the invention comprises the reduction of gain of the system so that oscillations cannot be sustained. The first embodiment involves a decrease in the time constant and corresponds to a shift of the operating point from *a* to *b* of FIG. 3, while the second embodiment may be constructed to give either a decrease of the time constant, or an increase thereof, involving a shift from *a* to *c*. In the latter case there is an increase in phase lag which by itself is undesirable, but which is more than balanced by the reduction in amplitude.

It is known that an increase of damping by the provision of a restricted orifice in the fuel line results in the elimination or reduction of the chugging effect. Such a restriction requires an increase of pressure for the steady state flow and is hence undesirable as a practical operating means. It is known, however, that the addition of such a restriction decreases the time constant of the system and therefore introduces a lead and also reduces the gain. As indicated in FIG. 4, the characteristic **34** represents the original characteristic similar to that shown in FIG. 3, while the curve **36** represents the characteristic as modified by increased damping. The characteristic of the added damping device itself is indicated by the dash curve **38**. If a rocket has been found to oscillate at a chugging frequency represented by the arrow **40**, the modified performance function upon the introduction of an increased restriction is represented by the arrow **42** which terminates on the characteristic **36**. The phase lead indicated by the angle A has been introduced by the restriction and the amplitude has also been reduced.

Rate-actuated stabilizer

According to the first embodiment of the present invention (shown in principle in FIG. 5) there is provided a compensator to be introduced into the fuel line which produces a phase lead and a gain reduction at least as great as that of a damper. In other words, if the amount of restriction necessary to eliminate chugging is known, a device of the type shown in FIG. 5 may be employed which gives at least as good a modifying characteristic as the damper at or near the chugging frequency. It is important to note, however, that the device shown in

FIG. 5 does not materially impede the steady state flow and hence does not require the use of equipment for providing exceptionally high pressure at the source. In other words, the apparatus provides a frequency-dependent restriction factor so that sufficiently large damping can be obtained in the vicinity of the chugging frequency.

The pipe 12 leads into the body 44 of the compensator within which is mounted a hollow cylindrical valve member 46 adapted for sliding motion within the body. The valve is centralized by means of opposed compression springs 48 and 50, the former bearing against one end of the pipe 12 and the latter bearing against a boss 52 which is of a shape to form outlet passages 54 around it. The fluid from the pipe 12 passes through the interior of the sliding valve 46 and then turns substantially at right angles to go through the passages 54 into the outlet 56 leading to the injector and rocket motor. The valve 46 is formed at its entrance end with a smoothly curved converging surface 58 and is formed at its exit end with a smoothly curved surface 60. The interior of the valve therefore constitutes a nozzle which serves as a flow sensing element. A difference of pressure exists between the entrance and exit ends of the nozzle and such pressure depends upon the flow rate. Thus the valve tends to move into a position determined by the flow rate. An increase in flow rate will tend to diminish the nozzle exit passage 54 and thereby impose a restriction in the flow. Under steady state conditions the nozzle assumes a position partly closing the passage so that the valve can move one way or the other upon an increase or decrease of the fuel rate. The size of the passage under steady state conditions is such that no undue restriction occurs, but the resistance to flow under dynamic conditions is greatly increased.

Under idealized conditions, if the mass of the piston 46 and the resistance to its movement by friction or damping are neglected, it can be shown that the apparatus of FIG. 5 can be made to have a characteristic similar to the simple restriction characteristic 38 of FIG. 4, except that a high pressure difference is not needed at the operating point. The restriction factor R_o of a simple orifice is given by:

$$\frac{1}{R_{or}}=\left(\frac{dF}{dP}\right)_{or}=\frac{F_{or}}{2P_{or}} \qquad (3)$$

where F_{or} and P_{or} represent the flow rate through the orifice and the pressure across it respectively. For the idealized stabilizer of FIG. 5, the corresponding restriction factor R_s is given by:

$$\frac{1}{R_s}=\left(\frac{dF}{dP}\right)_s=\frac{1}{2}\,\frac{F_s}{P_s(1+\sqrt{P_sS_v})} \qquad (4)$$

where S_v is the valve port sensitivity, taking account of the width of valve port 54 and the constants of the springs 48 and 50. With a small valve port and light springs, S_v may be made large and hence R_s of (4) may be made equivalent to R_{or} of (3) with only a moderate pressure drop P_s across the stabilizer. In other words, a stabilizer characteristic like 38 or better may be obtained without imposing excessive restriction to steady state flow.

The conditions are illustrated by the plot of FIG. 6. The curve 62 is a typical characteristic for a fuel supply line with a small restriction. For a given flow F represented by the horizontal dash line 64 a pressure of only P_1 is required. The dynamic characteristic is represented by the derivative of F with respect to P, namely, by the slope of the tangent as indicated at 66. With such a small restriction oscillations at some frequency in the system as a whole are likely to occur. The addition of a further restriction results in a characteristic 68 showing that for a given flow rate F a greatly increased supply pressure P_1' will be required. The slope of the tangent is much less, as is indicated at 70, showing that oscillations may be damped out at the expense of the increased pressure P_1'. In the construction of FIG. 5, however, it is possible to obtain a static characteristic similar to the curve 62 and a dynamic characteristic similar to that of the curve 68. The static characteristic is given by the curve 72 which passes through the same static operating point as the curve 62 so that an average flow F is obtained with only a pressure P_1. The dynamic sensitivity is given by the slope of the tangent 74 which corresponds approximately with 70. The general shape of the characteristic 72 as shown in FIG. 6 can be confirmed by the fact that at low flows there is little or no restriction due to the valve, whereas at high flows the valve moves toward a closed position.

The theoretical discussion concerning the rate-actuated stabilizer in FIG. 5 has neglected the mass of the piston and that of the fluid included within the piston, as well as the damping or friction against the motion of the piston. In taking account of these factors it will be seen that the apparatus of FIG. 5 constitutes a second-order system. Stated in another way, since the system has mass, its dynamic characteristic is represented by a differential equation which must be at least of the second order. Such a system has a more complicated characteristic than a simple orifice, and in a theoretical treatment it is necessary to take account of the undamped natural frequency of the stabilizer itself.

In FIG. 8 I have repeated the characteristic 38 of a simple orifice together with the modified fuel line characteristic 36. Values of $\omega\tau$ are indicated on these curves. At 62 I show a second-order characteristic for a typical device shown in FIG. 5 and at 64 the fuel line characteristic as modified by the stabilizer characteristic 62.

The polar plot 64 is a plot of the following performance function representing the relation between pressure and flow rate as a function of frequency:

$$(PF)=\frac{S_1}{1+j\tau\omega+\dfrac{K}{1+2j\zeta\dfrac{\omega}{\omega_n}\left(\dfrac{\omega}{\omega_n}\right)^2}} \qquad (5)$$

where τ is as defined before, ω is the "forcing" frequency, ω_n is the undamped natural frequency of the stabilizer, and ζ is a "damping coefficient." The factor K is ratio S_o/S_{so}, where S_o is the sensitivity of the unstabilized injector system as previously defined, while S_{so} is the static sensitivity of the stabilizer.

The modified fuel line system characteristic 64 depends to a considerable extent on the undamped natural frequency of the stabilizer. The particular curve 62 shown in FIG. 8 is for an undamped frequency ω_n such that $\omega_n\tau=4$, and $\zeta=0.5$. On these assumptions it will be seen that for an actual frequency such that $\omega\tau=1$ the rate stabilizer provides more lead and more gain reduction than the simple orifice 36, but for higher frequencies, such as $\omega\tau=3$, the rate stabilizer shows an increase in gain over the simple orifice, although it still shows an advantage in phase lead. If the chugging frequency is in the neighborhood of

$$\frac{1}{\tau}$$

the rate actuated stabilizer would provide a system which is more stable than would a restriction of the type indicated in FIG. 8.

This indicates that the undamped natural frequency of the stabilizer should be high, which means that the mass of the moving parts should be small and the springs should be relatively stiff. On the other hand, the valve port sensitivity should be high, which means that the valve area should have a high rate of change with respect to flow rate. In order to obtain a high valve port sensitivity the construction shown in FIG. 9 is preferred. In this construction instead of merely using the piston 60 to close off the port 54, a sleeve valve 66 is provided having a number of openings 68. The valve cooperates with lands 70 which are of a width comparable to that of each of the openings **68**. It will be seen that by the provision of a sufficient number of openings, the total valve width may be made substantially the same as that of a single passage, but the valve area may be completely closed off with a relatively small movement of the piston. By this means it is possible to obtain a high undamped natural frequency, and still have the stabilizer operate with a sufficient sensitivity to provide an overall characteristic for the system which is at least as stable as that obtainable by a simple orifice type restriction, but without the necessity for providing a high supply pressure.

A modified, and in some respects improved, form of rate stabilizer is shown in FIG. 7. In this construction the piston valve is made of two concentric sleeves and the object is to provide a frequency sensitive device which will make the stabilizer effective near the chugging frequency and substantially ineffective under steady state conditions, whereby no substantial restrictions are imposed against a steady flow. To this end there is an outer nozzle valve **72** and an inner sleeve **74** adapted to slide longitudinally relative to one another. The shape of the combined parts **72** and **74** is similar to that of the valve **46** of FIG. 5. The outer sleeve **72** is centralized by two springs **76** and **78** while the inner sleeve is centralized by two springs **80** and **82**. The sleeves are constructed to provide a cavity **84** between them. One or more passages **86** between the sleeves provide for flow of fluid into or out of the cavity. A simple reed valve **88** overlies the inner end of each passage **86**, as shown on an enlarged scale in FIG. 7*a*. Under steady conditions the outer sleeve **72** remains stationary, or nearly so, because the principal force due to the flow is exerted on the inner sleeve **74** and there is little or no shearing force between the two sleeves. However, when the sleeve **74** oscillates because of fluctuating flow in the system, fluid is pumped into the cavity **84** past the direction sensitive valve so that a force is transmitted from the inner sleeve to the outer sleeve to tend to close the valve.

Acceleration-actuated stabilizer

The apparatus shown in FIG. 10 also provides stabilization by actions occurring only in the fuel supply line, but on a somewhat different principle from that of FIGS. 5 and 7. In the line is mounted a body **90** having an internal stationary member **92** suitably streamlined at front and rear. Within the member **92** is a sliding piston **94** centralized by springs **96** and **98**. The piston is reduced in diameter to provide a flow passage in the form of a cavity **100** longitudinally of its outer surface. An inlet port **102** and an exit port **104** in the member **92** communicate with the passage **100** in the piston **94**. The inlet and outlet ports **102** and **104** are arranged so that the flow is guided to enter and leave the cavity **100** in directions perpendicular to the axial motion of the piston. Under these conditions the inlet and outlet flow stream does not exert any force on the piston in the axial direction. The axial force acting on the piston is entirely due to the change of momentum of the fluid in the cavity **100**.

In the arrangement shown in FIG. 10 the piston is shown in its neutral position, the inlet port being partially covered and the outlet port being uncovered. Whenever there is an increase in the rate of flow the piston will move to the left and therefore widen the inlet port. The resulting effect is to decrease the pressure drop through the stabilizer whenever the flow rate increases; that is, whenever there is an increase in the flow rate.

An alternative construction is to centralize the piston **94** in its normal position such that the inlet port is completely uncovered while the outlet port is partly covered. This condition may be represented in the drawing simply by considering that the flow directions in FIG. 10 are reversed, whereby **104** becomes the inlet port and **102** the outlet port. In such a case the opening of the port would be narrowed on an increasee of flow rate.

Thus the device of FIG. 10 provides an acceleration term in the performance function, which term is positive for the flow direction actually shown in FIG. 10 and negative if the apparatus is set up to produce a flow in the reverse direction. Either the positive or negative term may be used, as will be described, although the actual quantities that are involved differ somewhat in the two cases.

In either case the acceleration term is coupled with a flow-rate term because of the fact that a certain pressure difference is necessary to force the fluid through the ports **102** and **104** even when there is no acceleration of the fluid. However, if the sensitivity to acceleration is high, as it may readily be made, the flow-rate term may be neglected.

An analysis similar to that of Equation 5 may be carried out, and it is found that the performance function for the entire fuel supply system is as follows:

$$(PF)=\frac{1}{1+j\tau\omega\pm\dfrac{jK\omega}{1+2j\zeta\dfrac{\omega}{\omega_n}\left(\dfrac{\omega}{\omega_n}\right)^2}} \qquad (6)$$

where the quantities are as defined before, except that K is a quantity involving the sensitivity of the apparatus of FIG. 10. The plus-or-minus sign is used to distinguish the positive and negative acceleration terms, depending on whether the pressure drop increases or decreases upon an acceleration of the fluid.

A typical plot of the conditions is given in FIG. 11. This is for the case of the positive acceleration term, as represented by the flow direction indicated in FIG. 10, and is for the condition $\omega_n\tau=1$ and $\zeta=0.3$. The plot for the system modified by the acceleration-actuated stabilizer of the present invention is at **110**, and for comparison I show a plot **112**, similar to FIG. 4, for a system modified by a simple restriction. From this comparison it will be seen that if the natural frequency ω_n is chosen to be about equal to or greater than the chugging frequency, the acceleration actuated stabilizer will produce more phase lead and less gain than the simple restriction.

It will be noted that the rate-actuated stabilizer plotted in FIG. 8 showed desired stabilizing characteristics for a natural frequency several times the chugging frequency, while the acceleration-actuated device gives satisfactory stabilization if designed for a natural frequency approximately equal to the chugging frequency. The acceleration-actuated stabilizer is preferred over the rate-actuated stabilizer because of its lower natural frequency.

If the negative term of (6) is used, as represented by a flow direction reversed from that of FIG. 10, the system polar plot would show. That the device introduces additional phase lag over a wide range of frequency, as compared with a simple restriction. While increased lag is in itself objectionable, the amplitude is greatly reduced, and hence chugging is eliminated by the fact that the overall gain around the system is insufficient to maintain oscillations.

In the acceleration-actuated device, the valve-sensitivity should be kept high, as in the rate-actuated device. Therefore, a valve arrangement like that of FIG. 9 is preferred; and further sensitivity may be obtained with multiplying linkages.

Having thus described the invention, I claim:

1. In a liquid fuel system having a fuel source, a combustion chamber and a fuel supply line, said system being subject to oscillations because of the effect of the combustion chamber pressure on the fuel supply, the combination of a compensating device having a valve member to control a variable-area passageway in the supply line, said valve member having a normally open passage to introduce no substantial restriction to steady flow, centralizing springs for the valve member to hold the valve member in a position to determine a steady flow

rate under non-oscillating conditions, and means operated by variations in flow for reciprocating the valve to vary the passageway at a frequency to alter the dynamic characteristics of the fluid flow to suppress said oscillations.

2. In a liquid fuel system having a fuel source, a combustion chamber and a fuel supply line, said system being subject to oscillations because of the effect of the combustion chamber pressure on the fuel supply, the combination of a compensating device having a valve member to control a variable-area passageway in the supply line, said valve member having a sliding piston with a passage therethrough leading to the variable area passageway, centralizing means for the piston to hold the piston in a position to determine a steady flow rate under non-oscillating conditions, and means for turning the path of fluid at an angle upon exit from the piston, whereby the piston reciprocates at a frequency dependent on the variations in flow to suppress said oscillations.

3. Apparatus according to claim 2, in which the piston comprises two parts arranged to move with respect to each other, said parts having a cavity between them, and a one-way valve operable under oscillating conditions to cause fluid to be pumped into the cavity.

4. In a liquid fuel system having a fuel source, a combustion chamber and a fuel supply line, said system being subject to oscillations because of the effect of the combustion chamber pressure on the fuel supply, the combination of a compensating device having a valve member in the supply line, said valve member having a normally open passage to introduce no substantial restriction to steady flow, said valve member to control a variable-area passageway in the supply line, said valve member having a sliding piston with a longitudinal fluid passage, means for conveying fluid into and out of said passage, the valve member having valve ports arranged to be variably

covered by the piston and constituting the variable area passageway, and a centralizing mounting for the piston to hold it in a position to determine a steady flow rate under non-oscillating conditions, and means for turning the path of fluid at an angle upon exit from the piston whereby the piston is caused to oscillate in accordance with accelerations of the fluid under fluctuating conditions to vary said ports and thereby suppress the fluctuations in the fluid flow.

5. In a liquid fuel system having a fuel source, a combustion chamber and a fuel supply line, said system being subject to oscillations because of the effect of the combustion chamber pressure on the fuel supply, the combination of a compensating device having a valve member to control a variable-area passageway in the supply line, said valve member having two relatively sliding pistons having a passage therethrough, and separate sets of centralizing springs for the pistons to determine a steady flow rate, means operated by variations in flow for reciprocating the pistons to vary the flow rate at a frequency to suppress said oscillations, the pistons having a cavity between them, and a one-way valve to control passage of fluid into and out of the cavity under conditions of relative movement of the sliding members.

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