Aug. 14, 1945. A. P. SHEPARD ET AL 2,381,932

SPEED GOVERNING MECHANISM FOR METAL SPRAY GUNS OF THE WIRE FEED TYPE

Filed Dec. 1, 1941 6 Sheets-Sheet 1

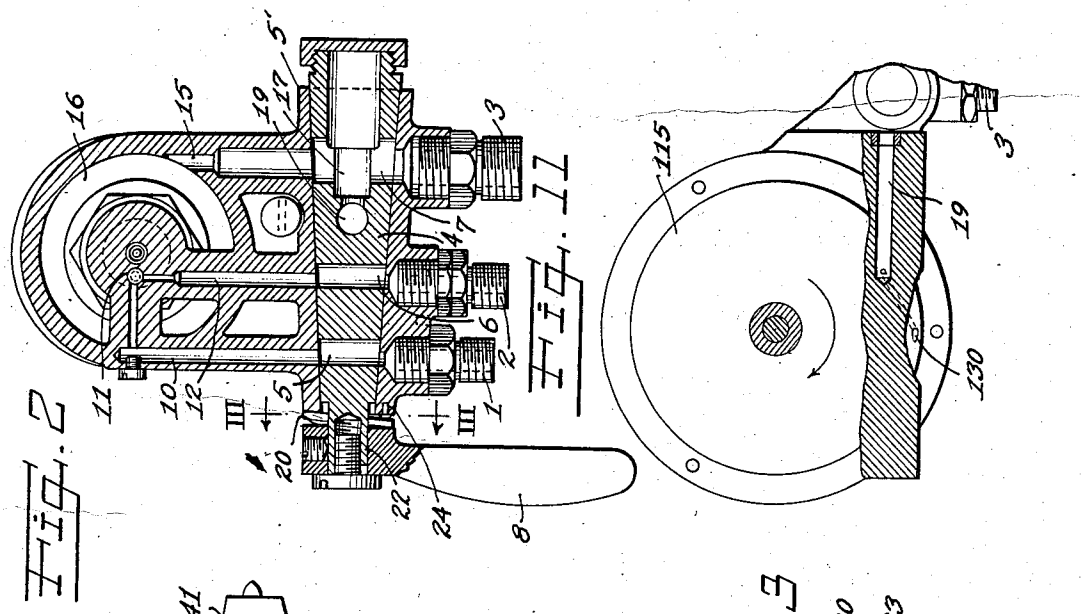


Fig. 2

Fig. 11

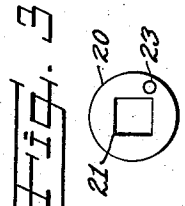


Fig. 3

Fig. 1

INVENTORS:
Arthur P. Shepard
Herbert S. Ingham
BY Burgess & Dinklage
ATTORNEYS

Aug. 14, 1945. A. P. SHEPARD ET AL 2,381,932

SPEED GOVERNING MECHANISM FOR METAL SPRAY GUNS OF THE WIRE FEED TYPE

Filed Dec. 1, 1941 6 Sheets-Sheet 2

Fig. 5

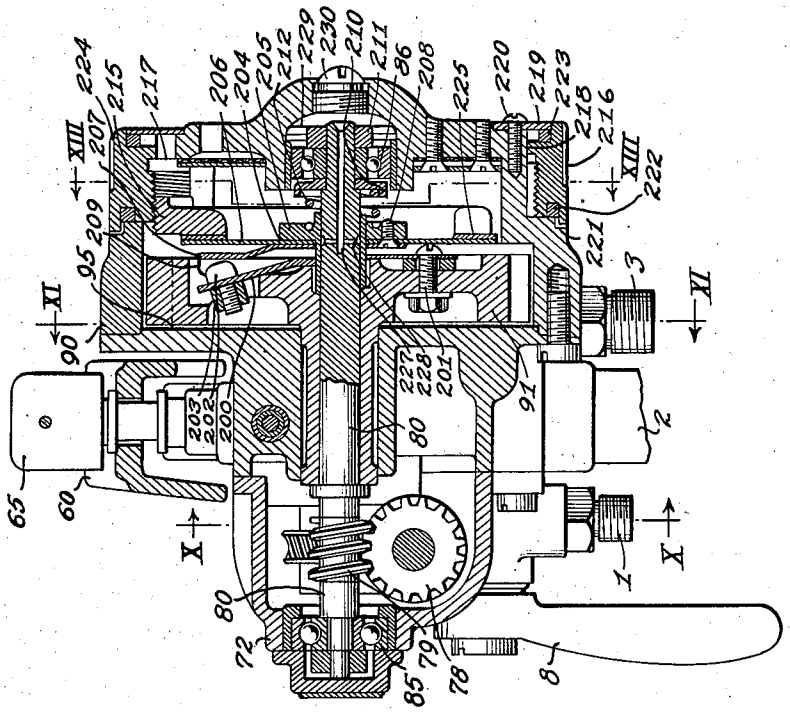
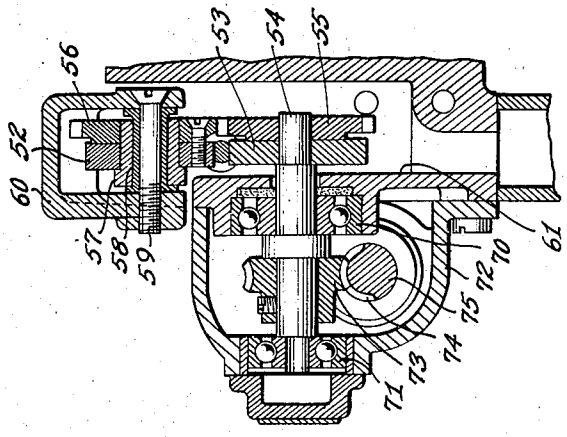


Fig. 4



INVENTORS:
Arthur P. Shepard
Herbert S. Ingham
BY Burgess & Dinklage
ATTORNEYS

Aug. 14, 1945. A. P. SHEPARD ET AL 2,381,932

SPEED GOVERNING MECHANISM FOR METAL SPRAY GUNS OF THE WIRE FEED TYPE

Filed Dec. 1, 1941 6 Sheets-Sheet 3

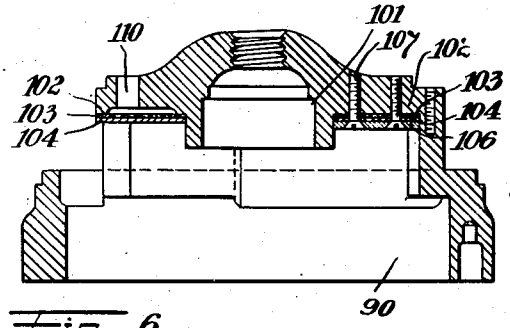


Fig. 6

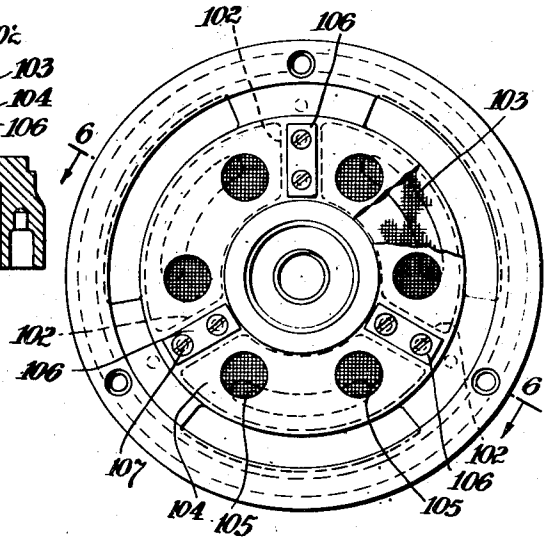


Fig. 6A

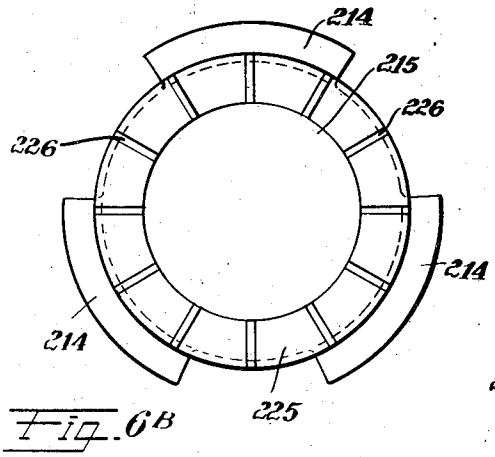


Fig. 6B

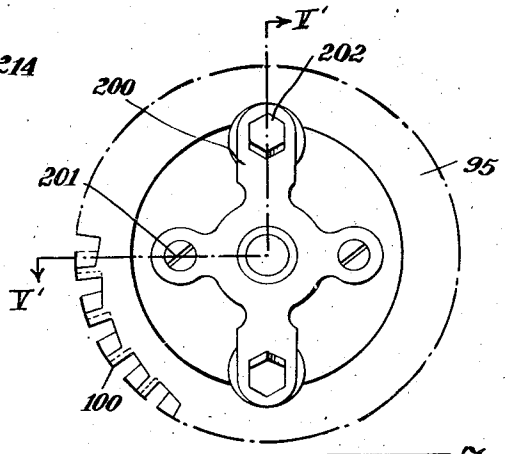


Fig. 7

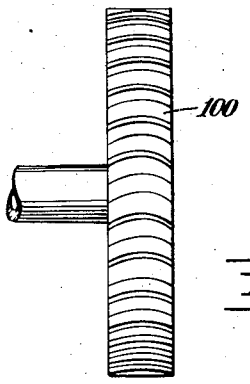


Fig. 8

INVENTORS:
Arthur P. Shepard
Herbert S. Ingham
BY Lewis Bruce
ATTORNEY

Aug. 14, 1945. A. P. SHEPARD ET AL 2,381,932

SPEED GOVERNING MECHANISM FOR METAL SPRAY GUNS OF THE WIRE FEED TYPE

Filed Dec. 1, 1941 6 Sheets-Sheet 4

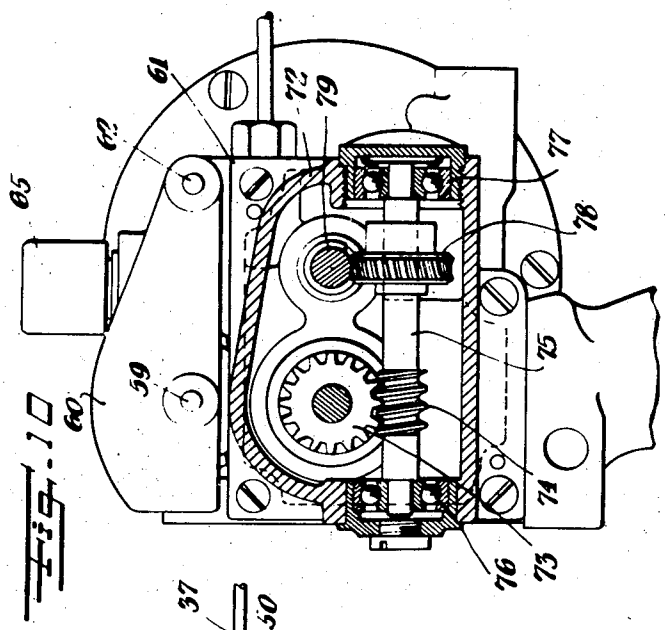


Fig. 10

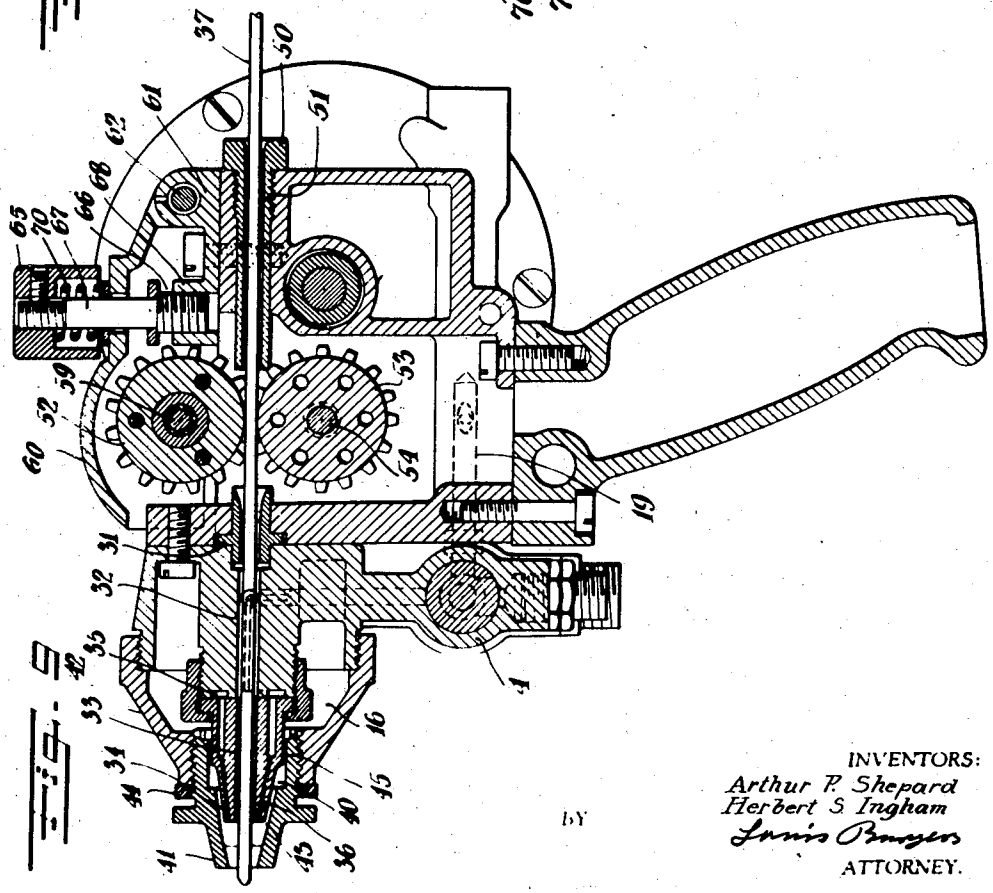


Fig. 9

INVENTORS:
Arthur P. Shepard
Herbert S. Ingham
BY Louis Bucklen
ATTORNEY.

Aug. 14, 1945. A. P. SHEPARD ET AL 2,381,932

SPEED GOVERNING MECHANISM FOR METAL SPRAY GUNS OF THE WIRE FEED TYPE

Filed Dec. 1, 1941 6 Sheets-Sheet 5

Fig. 12

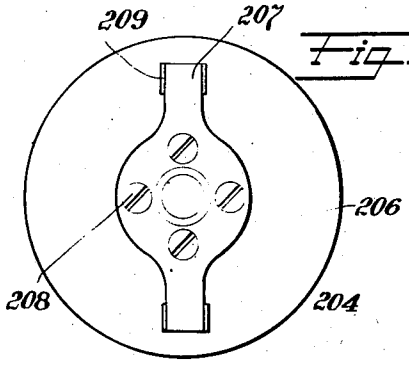
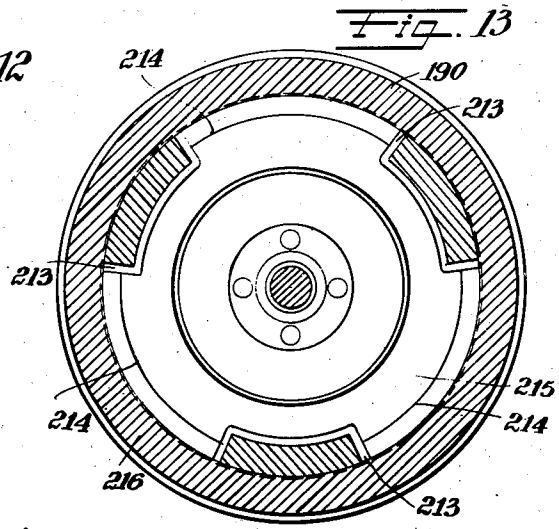


Fig. 13



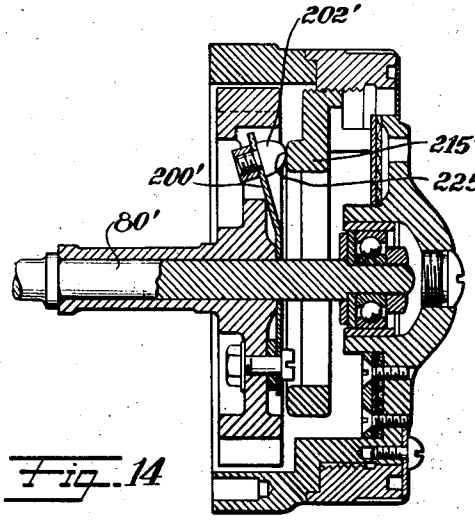


Fig. 14

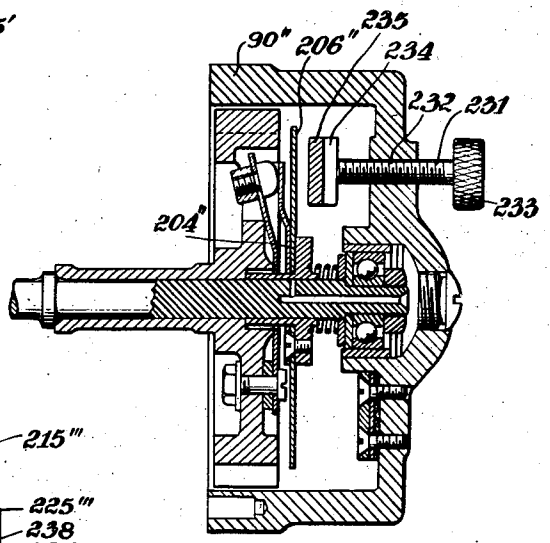


Fig. 15

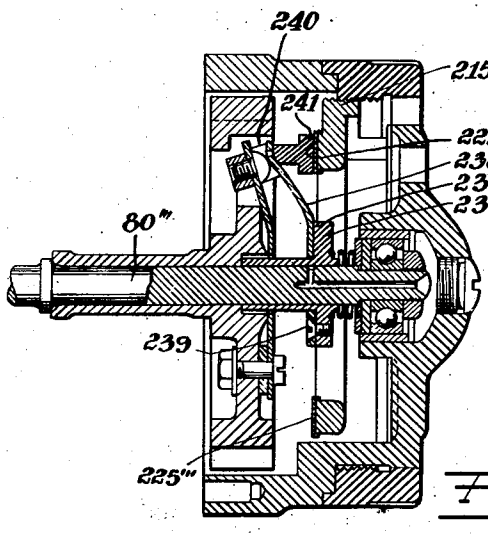


Fig. 16

INVENTORS
Arthur P. Shepard
Herbert S. Ingham
BY
Louis Bucher
ATTORNEY

Aug. 14, 1945. A. P. SHEPARD ET AL 2,381,932

SPEED GOVERNING MECHANISM FOR METAL SPRAY GUNS OF THE WIRE FEED TYPE

Filed Dec. 1, 1941 6 Sheets-Sheet 6

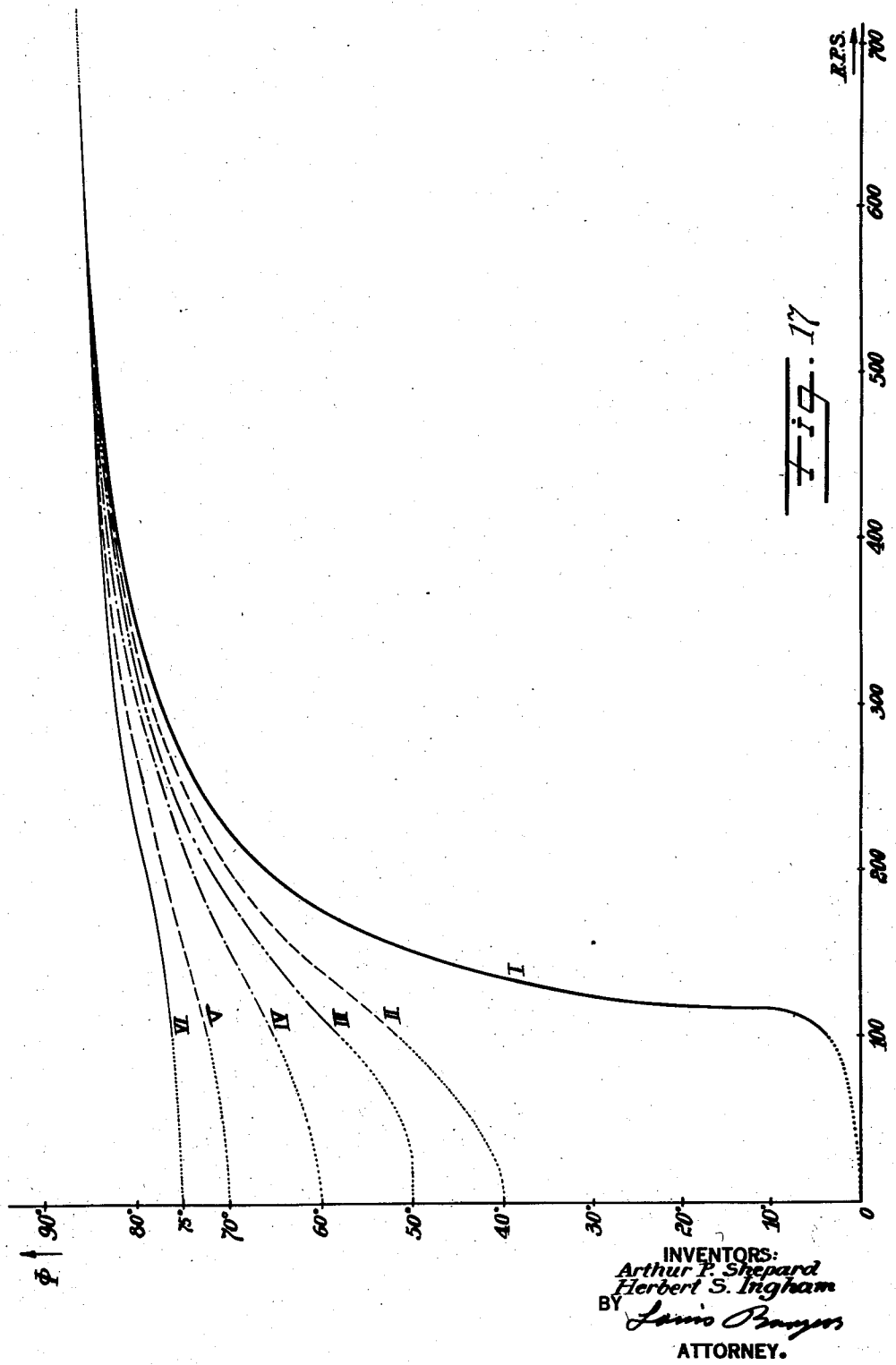


Fig. 17

INVENTORS:
Arthur P. Shepard
Herbert S. Ingham
BY Louis Bangs
ATTORNEY.

Patented Aug. 14, 1945 2,381,932

UNITED STATES PATENT OFFICE

2,381,932

SPEED GOVERNING MECHANISM FOR METAL SPRAY GUNS OF THE WIRE FEED TYPE

Arthur P. Shepard, Forest Hills, and Herbert S. Ingham, Flushing, N. Y., assignors to Metallizing Engineering Company, Inc., Long Island City, N. Y., a corporation of New Jersey

Application December 1, 1941, Serial No. 421,195

23 Claims. (Cl. 188—180)

This invention relates to new and useful improvements in metal spray guns of the wire feed type.

Metal spray guns of the wire feed type are devices in which a metal rod or wire is continuously fed into a melting zone, from which zone the metal is propelled in finely subdivided form by suitable means such as a blast of air or other gas. The rod or wire is fed into the melting zone by suitable rod or wire feeding means, such as knurled burs pressing against opposite sides of the wire. These wire or rod feeding means are driven, preferably operating through reduction gears, by a compressed gas motor, i. e., a motor, the rotor of which is rotated by the force of compressed gas. The load on the motor varies from time to time due to the changes in the position of the operator, kinks in the wire, etc. Since it is essential for successful spraying operations that the ratio of feed of the wire be uniformly maintained, it is likewise essential that the motor have a comparatively stable speed of operation, i. e., that its speed of operation be affected as little as possible by variations in the load. One common form of compressed gas motor is a compressed gas turbine and our invention shall be hereinafter described by way of illustration but not of limitation, in connection with metal spray gun constructions of the wire feed type having a compressed gas turbine.

In the operation of metal spray guns, it is necessary to maintain different rates of feed when spraying different metals. For example, low melting point metals may be fed and sprayed more rapidly than higher melting point metals and larger diameter wires must be fed and sprayed more slowly than smaller diameter wires of the same material.

It is the usual practice to attempt the approximation of the proper conditions for the particular metal to be sprayed by providing for replaceable gearing designed to permit the gun to be operated at the required rate of wire feed and to permit the turbine to operate within a range of stable operation. Changing the gearing, however, involves loss of time and the possibility that metal particles will be picked up by the gears in handling and interfere with the operation of the gun. When attempting to eliminate gear changes by reducing the turbine speed through a throttling of the blast gas supply, the turbine becomes unstable and a uniform wire feed is no longer maintained as the then available power is insufficient to compensate for load increases.

This operating instability of the ordinary gas blast operated turbine particularly at lower speeds, has been a serious disadvantage in metal spray gun constructions hitherto used. Although it may be possible in a given case to maintain a satisfactory wire feed, the ordinary gas turbine is very susceptible to variations in the operating load. The ordinary gas turbine has sufficient power for feeding the wire under normal conditions as long as a sufficient quantity of gas is supplied to the turbine rotor at a sufficiently high pressure. The power requirements of the turbine, however, necessitate the feeding thereto of sufficient impelling gas to cause the turbine to operate at a very high speed. At whatever speed, however, the turbine operates with a given inflow and pressure of gas, all the power is used up and no excess power is available to take care of variations in the operating load.

The governing mechanism in accordance with the invention is variably adjustable to procure different operating speeds and yet is capable at any given speed to make excess power practically instantaneously available when required by sudden overload.

Our governing mechanism broadly comprises at least one first and one second element substantially facing each other, one rotatable with the rotor of the compressed gas motor, said first element being movably mounted for operative actuation by suitable centrifugal speed responsive means to frictionally bear against the other to thereby absorb power, and means for variably adjusting the relative position between said second element and the initial position of said first element within a range of movement of the first element defined by the operating positions of said speed responsive means between minimum and maximum operating speeds of said rotor.

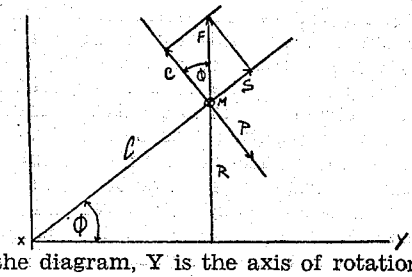
In general, the selection of any particular centrifugal speed responsive device or the particular construction thereof depends, inter alia, upon consideration of design, practicability, desired accuracy and sensitivity, intended capacity of the metal spray gun etc., but foremost upon the speed range within which such gun is to operate. The latter factor is of importance as many metal spray guns of the hereinmentioned type normally operate within a speed range where the maximum operating speed is in excess of 200% of the minimum operating speed.

Within the preferred embodiment of our invention, therefore, cognizance is taken of the fact that the rotor of the gun must be governed accurately with a sensitive speed governor allowing for variations in speed at which it must be

governed over a wide range of operating speeds. For instance, the speed range for the active operation of one particular gun of this type may be from 6,000 revolutions per minute (100 R. P. S.) to almost 40,000 revolutions per minute (666 R. P. S.). This is a speed increase from lowest to highest usable speed of over 500%. With any governor utilizing centrifugal force, this condition presents a particularly acute problem because of the fact that centrifugal force varies as the square of the rotating speed. This means that if the ordinary centrifugal governor were added to the rotor of a metallizing gun, which operates through a 6 times speed range, the force operating on the governor weights would vary 36 times from lowest to highest speed and a mechanical movement actuating the governor mechanism would be subject to equally great variations so that it is in such case practically impossible to obtain the necessary sensitivity at each and every speed throughout the range. In a specific case, for instance, if the total movement of the governing mechanism, which controls the speed, is .150'' between lowest and highest operating speed of the gun and if, a change from low speed to one tenth higher speed produces a movement of .075'' of the governing mechanism then the remaining nine tenths of the speed range could only actuate the mechanism through a total movement of .075'' which is the remaining distance. This would be an insufficient movement of the mechanism for so wide a speed range, and the governor would not be sensitive, nor perform satisfactorily.

The governor in accordance with our preferred construction makes it possible for the first time to utilize a governor of the centrifugal type to govern the rotor of a metal spray gun of the wire feed type through a wide speed range whereby the governing action, i. e., the mechanical movement of the actuating mechanism is substantially approximately proportional to the change in speed of the rotor throughout the speed range covered by the governor.

The principle underlying the preferred construction of the centrifugal speed responsive mechanism in accordance with the invention will be more fully understood from the following diagrammatic representation and derivation developing governor weight positions for given speeds of rotation and balanced by appropriate spring resistance at such positions.



In the diagram, Y is the axis of rotation of a governor element having an arm of the length l and carrying at its end the mass M and pivotally connected at X. R is the distance of M from the axis of rotation Y, and ϕ the angle of deflection of the arm l from the axis of rotation Y.

When the governor element rotates at an angular velocity ω about its axis Y, centrifugal force acts on the mass M in a direction perpendicular to the axis Y. This force is indicated by the vector arrow F. The force F can be considered to be resolved into a component force indicated by the vector arrow C, perpendicular to the arm l, and a component force indicated by the vector arrow S, in the direction of the arm l. The component force C tends to deflect the arm and, assuming the latter to move outwardly against a spring resistance, indicated by the vector arrow P, is balanced by the spring resistance P at any given operating position of l. This spring resistance P is defined by $K\phi$ in which K is the unit spring force in dynes for unit angular deflection of l, i. e., for each degree of deflection of l, through a given range of deflection or corresponding value of ϕ.

$$F=M\omega^2R \tag{1}$$

$$\omega=\frac{2\pi}{T} \tag{2}$$

T is the time required for one revolution and may be expressed in R. P. S.

$$T=\frac{1}{\text{R.P.S.}} \tag{3}$$

$$\omega=2\pi\,\text{R.P.S.} \tag{4}$$

$$R=l\sin\phi \tag{5}$$

$$F=M4\pi^2\overline{\text{R.P.S.}}^2 l\sin\phi \tag{6}$$

$$C=F\cos\phi \tag{7}$$

$$C=M4\pi^2\overline{\text{R.P.S.}}^2 l\sin\phi\cos\phi \tag{8}$$

$$2\sin\phi\cos\phi=\sin 2\phi \tag{9}$$

$$C=M2\pi^2\overline{\text{R.P.S.}}^2 l\sin 2\phi \tag{10}$$

As C is balanced by the spring resistance of arm l.

$$K\phi=M2\pi^2\overline{\text{R.P.S.}}^2 l\sin 2\phi \tag{11}$$

assuming $l=1$ and $m=1$

$$K\phi=2\pi^2\overline{\text{R.P.S.}}^2\sin 2\phi \tag{12}$$

(ϕ may be expressed in radians or degrees if K is appropriately chosen)

Assuming as a representative example an operating range of from 6,000 to 40,000 R. P. M. (100 to 666 R. P. S.) and a total maximum deflection of 88° at a maximum speed of 666 R. P. S., the value of K for equilibrium at that deflection may be found by inserting the various values in Equation 12 resolved for K.

$$K=\frac{2\pi^2\overline{\text{R.P.S.}}^2\sin 2\phi}{\phi} \tag{13}$$

$$K=\frac{2\pi^2\overline{666}^2\sin 176°}{88°}=883.7$$

Having established the unit value K for each degree of deflection through a range from 0° to 88° with a maximum of 666 R. P. S. at 88°, Equation 12 may be resolved for the determination of R. P. S. for different deflections of l or values of ϕ.

$$\text{R.P.S.}=\sqrt{\frac{K\phi}{2\pi^2\sin 2\phi}} \tag{14}$$

The R. P. S. values obtained for different values of ϕ are listed in the following Table I.

Table I

[Deflection range 0°–88°]

ϕ	0	10	20	30	40	50	60	70	80	85	88
R. P. S.	0	118	118.02	124.53	134.8	150.8	176.1	220.8	331.1	468.1	666

When plotting the curve defined by Equation 12 in accordance with the values given in Table I

the curve has a slope as demonstrated by curve I in Fig. 17.

When using instead of an arm controlled by a spring force acting throughout the entire range of 0° to 88°, an arm controlled by a spring force acting through a range beginning with a predetermined initial position at the angle γ of the arm l with respect to the axis of rotation Y, a different value for K controls. This value is defined by Equation 13 except that the value for $\phi=(88°-\gamma)$

$$K=\frac{2\pi^2\overline{\text{R.P.S.}}^2 \sin 2(88°-\gamma)}{(88°-\gamma)} \quad (15)$$

The values of K obtained for different values of γ are listed in the following Table A.

Table A

γ	40	50	60	70	75
K	16,200.6	20,464	27,772.5	42,909.2	59,817.7

The R. P. S values in accordance with Equation 14 for predetermined angles of γ are determined by

$$\text{R.P.S.}=\sqrt{\frac{K(\phi-\gamma)}{2\pi^2 \sin 2(\phi-\gamma)}} \quad (16)$$

The R. P. S. values for different values of ϕ and for each series with constant predetermined values for γ and corresponding constant values for K are listed in the following Tables II, III, IV, V, and VI.

Table II

[Deflection range γ to 88°]

$\gamma=40°$

ϕ	40	41	50	60	70	80	85	88
R. P. S.	0	28.8	91.3	137.7	195.7	309.8	461.2	666

Table III

[Deflection range γ to 88°]

$\gamma=50°$

ϕ	50	51	60	70	80	85	88
R. P. S.	0	32.56	109.41	179.6	309.8	461.2	666

Table IV

[Deflection range γ to 88°]

$\gamma=60°$

ϕ	60	61	70	80	85	88
R. P. S.	0	40.7	147.9	286.8	450.0	666

Table V

[Deflection range γ to 88°]

$\gamma=70°$

ϕ	70	71	75	80	85	88
R. P. S.	0	59.6	147.9	252.9	434.8	666

Table VI

[Deflection range γ to 88°]

$\gamma=75°$

ϕ	75	76	80	85	88
R. P. S.	0	80.3	210.5	417.7	666

The curves plotted for each series of values for ϕ and R. P. S., given in the foregoing tables, are illustrated in Fig. 17 as II, III, IV, V, and VI corresponding respectively to the tables of the same designation.

Analyzing these curves, it will be seen that in each curve there is a section or sections the slope of which roughly approximates a constant, i. e., for approximately equal increments in R. P. S. there are approximately uniform increments in angular deflection of the arm l. This permits the speed governing mechanism to be variably adjusted to any desired R. P. S. throughout the R. P. S. range defined by the particular constant slope curve section and to thereby attain an approximately accurate, uniform and sensitive speed governing operation.

It will be seen however that the different constant slope curve sections of the various curves differ in the R. P. S. range covered by such curve section. Thus, for instance, the section of curve I between 10° and 70° angular deflection, though roughly approximating a constant in its slope, has a very steep pitch with respect to the abscissa and covers for a total of 60° deflection an R. P. S. range of only from 115 to 220 R. P. S. that is a total R. P. S. range of only 105 R. P. S. As to the curve sections of curve I between 70° and 85° angular deflection and 80° and 88° angular deflection, the pitch with respect to the abscissa of these sections is not as steep. As to the section between 70° and 85° angular deflection of curve I, there is an R. P. S. range of 220 to about 465, i. e., a total R. P. S. range of only 245 R. P. S. for a total deflection of 15°. With respect to the section of curve I between 80° and 88° angular deflection, the R. P. S. range is from 330 to 666 or a total of only 333 R. P. S. for 8° deflection.

It is thus seen from the analysis of curve I that there is no single constant slope section in this curve defining anywhere an R. P. S. range covering in excess of 200% speed variations. Though a governor element operating within the range defined by curve I may be used within the limitations inherent therein, said governor element is ordinarily not satisfactory in the case of a metal spray gun of the wire feed type normally operating within a speed range where the maximum operating range is in excess of 200% of the minimum operating range.

The dotted outlines of the various curves represent the plotted slopes below minimum operating R. P. S., i. e., 100 R. P. S. for the device exemplified in this illustration.

Analyzing the curves based on a predetermined initial angular position of the arm l (curves II, III, IV, V, and VI) it will be seen that curves II and III do not possess any section above minimum operating R. P. S. of 100 for which the slope roughly approximates a constant and which defines an R. P. S. range covering in excess of 200% speed variation. In curves IV, V, and VI, it will be seen that these curves roughly approximate a constant all the way between minimum operating R. P. S. of 100 and the maximum operating R. P. S. of 666 which is equivalent to an R. P. S. range covering in excess of 500% speed variation. In these cases the approximation of a constant increases with the increase in the predetermined initial angular position of the arm with respect to the axis of rotation, and, although for practical operation a predetermined angle of 60°, as exemplified by curve IV, will give satisfactory results, higher angles as illustrated by curves V and VI are preferred. Although the constancy of the slope improves with higher initial angles, it has been found that the governing element

with too high a predetermined initial angle may lose sensitivity. For best results an arm having its spring set for a predetermined initial angular position between 70° and 80° and preferably at approximately 75° with respect to its axis of rotation, is preferred. This is exemplified in its operating range by curve VI. "Initial position" or similar expression as used herein in connection with the arm or arms in accordance with our invention is intended to connote the angular position assumed by the arm or arms at zero speed.

Although in the foregoing equations and the various curves illustrated in Fig. 17 a minimum operating range of 100 R. P. S. and a maximum operating range of 666 R. P. S. with an assumed maximum deflection of 88° have been selected, it is understood that these values are merely used by way of exemplification of the principles underlying the preferred embodiment of our invention. The results are not substantially different and the same type of curves result if other values or ranges are substituted for those hereinabove used.

The above derivation shows a relationship between governor weight positions and speed at which equal increments in speed produce approximately equal increments in angular positions of the governor arm. It is now possible to combine a lever arm and weight structure, embodying this principle, with speed control means and with means for adjustably positioning such speed control means to produce a governor operating satisfactorily over a wide speed range.

In the actual operation of the governor there is a governing force acting to control the speed and in the illustrations shown in the drawings, this governing force is parallel to the axis of rotation. For a complete analysis of the forces involved during governing, this governing force must be taken into account, as well as the centrifugal force and spring force. This governing force, however, need not be considered in developing the relationship between governor weight positions and speed at which equal increments in speed produce approximately equal increments in angular positions of the governor arm.

The variable speed governing mechanism, in accordance with the preferred construction of the invention, for a metal spray gun of the wire feed type, having a compressed gas motor, comprises at least one arm, rotatable with the rotor of such motor and centrifugally deflectable for angular deflection with respect to its axis of rotation, against spring resistance, from a predetermined initial position and preferably from a predetermined initial position of at least 60° with respect to such axis; speed control means, composed of at least one first and one second element, the first element being positioned and adapted to be operatively acted upon by the arm or arms, upon centrifugal actuation thereof to frictionally bear against the second element to thereby effect speed control of the rotor by power absorption; and means for variably adjusting the relative position between the second element and the initial position of the first element within a range of mechanical movement of the first element defined by the angular deflection of the arm or arms between minimum and maximum operating speeds of the rotor. The "initial position" of the first element of the speed control means referred to herein, designates that position of such first element as is controlled by the deflectable arm or arms at its or their initial position.

The metal spray gun of the wire feed type embodying our invention does not require any change of gearing. The same is capable of maintaining any number of practical wire feeding speeds and may be shifted from one speed to another by a simple adjustment of the preferably manual control means. Furthermore, the motor or turbine of such metal spray gun is at all times maintained in a stable operating condition. The particular construction in accordance with the preferred embodiment of our invention permits the use of these metal spray guns under substantially stable operating conditions, over a much wider range of wire feed speeds than is otherwise possible.

The invention will be more fully understood and further objects thereof will appear from the following description read in conjunction with the drawings in which:

Fig. 1 is a side view of the metal spray gun illustrating one embodiment of a construction in accordance with the invention;

Fig. 2 is a vertical section through the construction shown in Fig. 1 on the plane indicated by II—II;

Fig. 3 is an illustration of the stopwasher as seen in the plane III—III of Fig. 2;

Fig. 4 is a vertical section through the construction shown in Fig. 1 on the plane indicated by IV—IV;

Fig. 5 is a vertical section through the construction shown in Fig. 1 on the plane indicated by V—V except for the construction of the structure illustrated in Fig. 7 which is shown on the plane indicated by the section line V'—V' in Fig. 7;

Fig. 6 illustrates a view of part of the section shown in Fig. 5;

Fig. 6A is a plane view of the interior of the construction shown in Fig. 6;

Fig. 6B is a view of one element of the construction shown in Fig. 5;

Fig. 7 is a side view of the rotor element of the spray gun;

Fig. 8 is a view of part of the construction shown in Fig. 7 at right angles thereto;

Fig. 9 is a central vertical section parallel to the construction shown in Fig. 1;

Fig. 10 is a vertical section through the construction shown in Fig. 5 on the plane indicated by X—X;

Fig. 11 is a sectional view of Fig. 5 in the plane indicated by XI—XI;

Fig. 12 is one of the governor elements in accordance with our invention;

Fig. 13 is a vertical section through the construction shown in Fig. 5 on the plane indicated by XIII—XIII;

Figs. 14, 15, and 16 illustrate sectional views of variations in the construction of a spray gun embodying our invention.

Fig. 17 is a graphic representation of the principle underlying the preferred embodiment of the invention.

The centrifugally actuable arm element in accordance with our preferred structure may be any one or a multiple number of suitable spring force controlled arm or arms; we prefer a construction, however, where the arm element is a spring element such as for instance one or more spring arms, i. e., arms made of spring material, set at the desired predetermined initial angle.

In case of a substantially rigid arm, we prefer to obtain deflectibility thereof by pivotally mounting the arm with respect to its axis of rotation. When using one or more spring arms, pivotable mounting thereof may be dispensed with as a rule since these as such will usually permit satisfactory deflection.

Referring to the drawings 1 (Fig. 2) indicates the inlet for oxygen or other combustion supporting gas, 2 the inlet for acetylene or other combustible gas, and 3 the inlet for air or other gas for atomization of the metal, projection of the metal spray and driving turbine. When plug 4 of valve 5' is in the position shown, each of the inlets registers with a corresponding hole in the plug, these holes being indicated by numerals 5, 6 and 7 respectively. In this position, oxygen flows through duct 12 to mix with the oxygen in duct 11 and the air flows through duct 15 into chamber 16 and also through the side connection 17 into turbine manifold 19. Openings 5, 6 and 7 in plug 4 are so arranged that as handle 8 is turned from the off position, which is at a right angle to the showing in Fig. 2, first some combustible gas passes into duct 11 and thence to the burner outlet to enable the burner to be lighted. Some air passes simultaneously into manifold 19 to enable the turbine to come up to speed. Alternatively, all the valve passages may be opened but at such rates of flow as to establish favorable lighting conditions which are different from the conditions obtaining when the gun is in operation. The washer 20 (Fig. 3) defines a hole 21 of rectangular cross-section which fits closely the shank 22 (or corresponding section) (Fig. 2) of the plug 4. The washer 20 is formed with the depression 23 (Fig. 3). This washer is spring pressed and this depression slips onto the head of pin 24 when handle 8 is in the correct position for lighting the burner. This offers sufficient resistance to indicate to the operator the lighting position. After the burner has been lighted a slight pressure against handle 8 forces the depression 23 out of engagement with the head of the pin 24. A further movement of handle 8 causes oxygen to flow through duct 10 which establishes a melting flame with the ignited gas and the final movement of handle 8 to the position shown in Fig. 2 permits air to flow into duct 15 and thence into chamber 16 to project the sprayed metal upon the surface to be covered.

The construction of that part of the gun by which the rod or wire is melted and projected will be explained by reference to Fig. 9. The wire moves forward to guide 31 and through duct 32 to the interior 33 of the burner tip 34. The mixture of air and oxygen moves forward through the duct 11, which is immediately behind duct 32 (the arrangement is shown in Fig. 2) and into the annular space 35. From this annular space 35 the combustible mixture moves forward through a number of holes to be discharged through convergent orifices 36 against the wire. This forms a zone of gases undergoing combustion, whereby the wire 37 melts as rapidly as it is progressively advanced into the zone, for which reason this zone is referred to as a melting zone. The air from the chamber 16 advances through the annular space surrounding burner tip 34 and is projected by air nozzle 41 in such a way as to sub-divide and propel the molten metal. The air tip 41 is threaded to the outer shell 42 of the burner so that the orifice 43 defined by conical interior of air tip 41 and conical exterior of burner tip 34 may be adjusted with corresponding variations in the characteristics of the air blast. When a satisfactory adjustment has been made, the top 41 is locked in position by the lock nut 44. It will be noted that the air in passing forward from the chamber 16 goes through the constricted annular space 45 which exerts a definite control over the volume of air passing. As a result of this construction and the orifice effect thereby created, the adjustment of air tip 41 modifies the characteristics of the air blast without so great a modification of the volume of air passing thereto as would otherwise result, which is decidedly advantageous in the adjustment and operation of the gun.

The wire 37 (Fig. 9) enters the gun through the annular guide 50 of hardened material in which is the duct 51. The upper and lower surfaces of the wire are engaged respectively by the burs 52 and 53. Bur 53 is carried by shaft 54, which shift is driven by an air turbine through suitable intermediate gearing which will be hereinafter described. Shaft 54 (Fig. 4) also drives the gear 55 in mesh with gear 56, which in turn drives the upper bur 52. Both gear 56 and bur 52 are secured to the tubular member 57 which rotates on the spool 58 carried by pin 59 (Fig. 4). The screw 59 is carried by the saddle 60 and this saddle is pivotally secured (Fig. 9) to frame 61 of the gun by the hinge 62. When cap 65 is turned the thread member 68 which is a part of frame 61 and the spring 70 exerts pressure on the saddle 60, thereby forcing the upper bur 52 toward the lower bur 53 and thereby causing the burs to engage and advance the wire 37. Conversely, when cap 65 is turned in the reverse direction, pressure of spring 70 on saddle 60 is released and the burs move freely without engaging and advancing the wire.

The shaft 54 (Fig. 4) which drives burs 53 is mounted in ball-bearings 70 and 71. Bearing 70 is held in frame 61 and bearing 71 is held in the housing 72 which is attached to frame 61. The shaft 54 is driven by the worm gear 73, which in turn is driven by the worm 74, carried by the shaft 75. Shaft 75 (Fig. 10) is carried by ball-bearings 76 and 77 mounted in the housing 72. Shaft 75 is in turn driven through the worm gear 78 by the worm 79. The worm 79 (Fig. 5) is integral with the shaft 80 carried by bearings 85 and 86. Bearing 85 is mounted in housing 72 and bearing 86 is mounted in the cap or cover 90 of the turbine 91.

Turbine 91 (Fig. 5) includes the turbine rotor drum 95 secured to shaft 80. Details of the blading 100 are apparent from Figs. 7 and 8. As evident from Fig. 6, cover 90 includes the mounting 101 for the ball-bearing 86 and three ridges 102 (Figs. 6 and 6A) radially arranged about mounting 101 on the interior surface of the cover. The washer-shaped member 103 is made of fine wire mesh and rests directly upon ridges 102. The washer 104 rests directly on the washer 103. The washer 104 defines the perforations 105 (Fig. 6A). The washers 103 and 104 are held in contact with each other and with the ridges 102 by the strips 106, which in turn are secured to the ridges 102 by the screws 107. One result of this construction is that the exhaust from the turbine flows through perforations 105 in the washer 104, thence through openings in the fine screen of which washer 103 is composed, and thence through exhaust ports 110, in cover 90 (Fig. 6 and

Fig. 1) thereby resulting in more quiet operation of the turbine.

The nozzle arrangement operating the turbine is shown in Fig. 11. Face 115 together with the housing 90 defines the space in which the turbine drum rotates; (the drum is not shown to facilitate inspection of the nozzle structure). A nozzle 130 is permanently connected to the manifold 19. Hence, part of the air or other gas which enters through connection 3 and into manifold 19 can flow through jet 130 and hence propel the turbine. All of these connections are made large enough to propel the turbine at the maximum speed at which it will be required to operate, and hence an excessive amount of gas is supplied for every speed except the maximum.

Specifically referring to Figs. 5, 6, 6A, 6B, 7, 8, 12, and 13, illustrating an embodiment of a centrifugally operated power absorption governor in accordance with the invention, rotor 95 carries governor in spring element 200 mounted thereon by means of screws 201. This spring element is set with respect to the axis of shaft 80 at an angle of approximately 75°. The weights 202 in this case consist of screws clamped through a hole in each end of the spring 200 by the nuts 203.

When the rotor 95 is at rest, the weights 202 take the position shown. When the rotor 95 rotates, the centrifugal force acting on the weights 202 tends to pull them away from the axis of the shaft and in so doing tends to deflect the arms of the spring 200 forwardly towards a perpendicular position to the axis of shaft 80. This tendency of the springs to straighten under the centrifugal force, acting on the weights is increased by the weight of nuts 203, which, because they extend beyond the bent spring exert a twisting force due to the action of centrifugal force which twisting force is also in the direction to deflect the spring arms 200 nearly perpendicular to the axis of shaft 80. When the rotor 95 is operating at the lower end of its speed range, the spring arms 200 are but slightly deflected from their initial position. In this speed range a slight change of speed causes a motion of the buttons or weights essentially in the direction of axis of shaft 80, due to the change in the amount of centrifugal force acting on weights 202 and nuts 203. The deflection of the spring is not proportional to the square of the speed of rotation. As the speed increases and the buttons 202 move approximately parallel to the axis of shaft 80, they deflect the spring arms 200 to a position more nearly perpendicular to the axis of shaft 80. The buttons 202 therefore operate in and out essentially parallel to the axis of shaft 80 throughout the entire large speed range required for a metallizing gun, and the position of the buttons 202 at any speed represents a measure of that speed.

Slidably mounted on shaft 80 is brake disc sub-assembly 204. This sub-assembly consists of hub 205, disc 206 and bent clip 207. Bent clip 207 and disc 206 are mounted on hub 205 and secured thereto by screws 208. The ends of the arms of the piece 207 are bent up to form ears or prongs 209. As assembled these ears 209 fit either side of each weight button 202, hence, as rotor 95 rotates, brake disc sub-assembly 204 also rotates being driven by the contact of weight buttons 202 with ears 209 of spring clip 207. Mounted on shaft 80 next to bearing 86 is washer 210. Nut 211 which screws on to threaded end of shaft 80 clamps bearing 86 and washer 210 against the shoulder of the shaft 80. Assembled between washer 210 and the hub 205 is the compression coil spring 212. This coil spring 212 normally tends to hold the brake disc sub-assembly 204 towards the rotor 95, so that the weight buttons 202 contact the ends of spring clip 207.

In operating, as the rotor 95 speeds up, the weight buttons 202 move parallel to the axis of shaft 80 towards the brake disc sub-assembly 204, and by the contact of the buttons 202 with the spring clip 207 force the brake disc sub-assembly 204 to slide along the shaft 80 in opposition to the coil spring 212. The turbine housing 90 is cut away at three sections 213, so as to permit the three segments 214 of spider or friction ring 215 to fit loosely through the resulting openings. The outer periphery of the three segments 214, of spider 215 are threaded. Ring nut 216, substantially forming an extension of the turbine housing 90, is threaded in its bore to engage the threads of spider 215. This thread extends on the bore of ring nut 216 to a neck at the end of the thread which is terminated by shoulder 217. When the nut 216 is turned the spider 215 travels in and out on the thread and is stopped at one end of its travel by shoulder 217, and at the other end of its travel by forcing brake disc sub-assembly 204 to the limit of its travel towards the rotor 95. Ring nut 216 is rotatably mounted on housing 90 and is located by bearing surface 218. Disc 219 is secured to housing 90 by screws 220, and acts to restrain the ring nut 216 on the housing 90 between the housing shoulder 221 and itself so that the ring nut 216 is free to rotate but cannot move longitudinally. Ring nut 216 is provided with grooves 222 and 223, which extend around its faces and which are packed with a packing material such as cork or graphite impregnated cotton string. This construction serves the double purpose of keeping dirt out of the mechanism and also of providing a necessary amount of friction so that ring nut 216 will not turn accidentally. A section of the outer periphery of ring nut 216 is knurled at 224 to provide an easy grip to the fingers for turning. As that portion of the housing 90 which has not been cut away to allow space for the spider segments 204 straddles these segments of the spider 215, the spider 215 is not permitted to rotate, but is permitted to move parallel to the axis of shaft 80. By screwing the ring nut 216 one way or the other the spider 215 may be located longitudinally in any desired position within the limits of its motion. The ring shaped face 225 of spider 215 (Fig. 6B) represents a braking surface to contact the face of the brake disc 206. Brake disc 206 is preferably made of hardened steel and polished on its contacting surface. The braking surface 225 of spider 215 is preferably made of sprayed metal, most preferably of sprayed bronze. For smooth operation it is important that the coefficient of friction, between these two surfaces remain as constant as possible throughout varying conditions of service, as well as throughout varying conditions of lubrication. Sprayed metal is particularly adapted for this purpose because of its constant coefficient of friction and because due to its porosity it will absorb lubricant and hence operate under at least semi-lubricated conditions when it might be otherwise operating dry. The braking face 225 may be divided into segments if desired by cutting a number of radial shallow slots 226. Such slots 226 will tend to collect foreign matter and prevent scratching of the

braking surface. Shaft 80 is provided with lubricant holes 227 and 228, which extend from the lubricant chamber 229 behind the bearing to the surface of the shaft beneath hub 205. Housing 90 is provided with the grease plug 230 which may be removed for the introduction of grease to chamber 229 and to the holes in the shaft 227, 228 to the sliding bearing surface between shaft 80 and hub 205. As grease is added through chamber 229, a small amount of grease will leak past bearing 86 and washer 210 and be thrown upwards by the rotating washer 210 in such a direction that some of the grease will locate on the threads of ring nut 216 and some of the grease will locate on the inner periphery of spider 215 and find its way to the bearing surface 225. Hence, the whole governor mechanism may be lubricated, as well as the bearing 86 by the introduction of grease into chamber 229.

Bent clip 207 performs the double function, first of providing a driving means through the ears 209, the buttons 202, the spring 200 and the rotor 95 between the rotor and the brake disc sub-assembly 204. The other function is that of absorbing by its resiliency the pounding action of weight buttons 202. When operating at high speeds, there is a natural tendency for the weight buttons 202 which are mounted on the ends of spring arms 200 to enter into a vibratory motion. This motion is quickly absorbed by the resiliency of the arms of bent clip 207. It will be understood that bent clip 207 may be eliminated from the sub-assembly 204 and that the weight buttons 202 may then be made to bear directly against brake disc 206. If bent clip 207 is eliminated, however, some other means such as keying, gearing, or other drive connection should be used to drive brake disc sub-assembly 204 and cause it to rotate with rotor 95. When bent clip 207 is used, the brake disc 206 may be made lighter than would otherwise be possible, as piece 207 also serves to apply thrust load from the weight buttons 202 more nearly at the center of disc 206, hence minimizing the distortion due to an unevenly applied load.

In operation ring nut 216 is turned to locate spider 215 in any desired position corresponding to a desired speed. As a full volume of air under full pressure is provided at all times to the turbine rotor 95, this rotor will tend to increase its speed, causing the weight buttons 202 to move toward brake disc sub-assembly 204 and cause the brake disc sub-assembly to move toward the friction surface 225 of spider 215. As the desired speed is reached the brake disc 206 brushes against friction surface 225, generating heat by friction, thus absorbing any excess power which may be developed by the turbine rotor 95. Any heat which is thus developed is absorbed and carried away by the exhaust gas from the turbine which must pass around the spider 215 before finally exhausting through ports 110. The speed of the rotor is, therefore, accurately established by the position of the spider 215, for if the rotor 95 tends to travel faster, increased pressure is exerted between brake disc and braking surface 225, causing an increase in friction. If due to a kink or bend in wire 37 or some other cause an increased load is imposed on the driving mechanism of the metallizing gun, this load causes the rotor 95 to slow down very slightly. However, even a very slight reduction in speed is sufficient to cause the weight buttons 202 to move slightly away from the brake disc sub-assembly 204 and hence release the pressure between the friction surfaces 206 and 225. As this pressure is released less friction occurs and less heat is generated and hence an extra amount of power is released for active use of the turbine and driving mechanism, and hence no further reduction in speed occurs. On the other hand, if the load on the driving mechanism reduces suddenly or even gradually, the rotor 95 tends to speed up very slightly. Even a slight increase in speed is sufficient to cause this sub-assembly to press harder against the friction surface 225, which results in a greater amount of power being absorbed by friction; hence, only a very slight increase in speed is possible even though the load on the driving mechanism would be released entirely. The action of the governor is practically instantaneous so that a very steady wire feed drive is provided, which can easily be adjusted to any given speed over a wide range of speeds by the simple expedient of turning ring nut 216. This constancy of wire feed is maintained even with variations in the amount of pressure of the blast gas supplied for the operation of the turbine.

The construction shown in Fig. 14 is similar to that described above, except that brake disc sub-assembly 204 has been eliminated. In this construction weight buttons 202' operate the same as previously described, except that as they move parallel to the axis of shaft 80' under the action of centrifugal force, they are caused to contact directly with the friction surface 225' of spider 215'. With this construction the friction surface 225' is smooth and continuous and is not slotted as at 226 in the previously described construction. Friction in this case is procured directly between the action of weight buttons 202' and friction surface 225' but the adjustments in operation are otherwise the same as previously described.

In the construction shown in Fig. 15 the rotor spring and weight button construction and the brake disc sub-assembly construction are the same as that described in connection with Fig. 5. In this construction, however, the turbine housing 90'' is not cut away as at 213 (Fig. 13) as no spider nor ring nut is used. Instead, screw 231 is provided to operate through a threaded hole 232 in housing 90''. The outer end of screw 231 is provided with a knurled knob 233 and at the inner end with a small disc 234 which is faced with friction material 235, preferably made of sprayed metal. The operation is the same as that described in connection with Fig. 5 except that as the rotor speeds up, brake disc sub-assembly 204'' is forced to contact friction surface 235 until sufficient friction is generated and absorbed to permit the turbine to arrive at and maintain its pre-determined speed. In this case, the speed is pre-determined or adjusted by the location of friction surface 235 which is obtained by screwing the screw 231 in or out by means of the knurled knob 233.

The construction shown in Fig. 16 is similar to the construction described in connection with Fig. 5 except that brake disc assembly 204 (of Fig. 5) has been replaced by friction button sub-assembly 236. This assembly consists of a hub 237, slidably mounted on shaft 80''' and a bent clip 238 similar to bent clip 207 of Fig. 5, which is secured to hub 237 by screws 239. Ears 240 are provided at the ends of bent clip 238 for the same purpose as described in connection with bent clip 207 of Fig. 5. On each end of bent

clip 238 is mounted a friction button 241. As the turbine speeds up, clip sub-assembly 236 is moved along shaft 80 as described in connection with sub-assembly 204 in Fig. 5. Buttons 241 hence contact friction surface 225''' to cause the turbine to operate at a predetermined speed. The control and operation of the mechanism with this construction of Fig. 5, as the friction button subassembly 236 replaces the friction disc subassembly 204 of Fig. 5.

As aforementioned the speed governing mechanism in accordance with the preferred embodiment of our invention broadly comprises a first friction element, capable of mechanical movement induced by the operations of the arm or arms, and a second friction element, cooperating with the first friction element to effect any desired speed variably adjustable within the entire speed range. Thus, for instance, referring to Fig. 5, brake disc 206 constitutes the first friction element operated by the movement of the spring arms 200 by way of the buttons 202 and clip 207. The second friction element constitutes in this case the friction face 225 of spider 215 variably adjustable within the range of mechanical movement of the brake disc 206 as controlled by the deflection of the spring arms 200 between minimum and maximum speeds of the rotor 95.

Alternatively as seen in Fig. 14, the first friction element may be defined by the friction button 202' operated by the deflection of the spring arms 200', the second friction element in this case being the friction face 225' operating and variably adjustable similar to the second element described in connection with Fig. 5. Further modifications of first and second friction elements in construction and operation similar to those exemplified by Figs. 5 and 14 are illustrated in Figs. 15 and 16.

Our invention is not limited to the construction of a spray gun illustrated in the foregoing example and may be used in connection with any other suitable gun constructions of the gas blast wire feed type, including those having other heating or melting means for the wire, such as an arc or the like and further including those constructions of this type in which multiple wires are fed to the heating zone.

Although within the preferred embodiments exemplified in the foregoing figures the speed control mechanism is illustrated in connection with a motor of the air turbine type, it is within the scope of our invention to use our novel speed control construction to govern other types of compressed gas motors in wire feed type metal spray guns.

The present application broadly covers a speed governing mechanism having speed control means effecting speed control of a rotor of a metal spray gun of the wire feed gas blast type by friction elements cooperating by mechanical movement and actuated by centrifugal means; our co-pending application Serial No. 421,194 covers a speed governing mechanism having speed control means comprising speed control elements cooperating by mechanical movement and actuated by centrifugally operated and angularly deflectable arms; and our co-pending application Serial No. 421,196 covers a speed governing mechanism having speed control means regulating the compressed gas supply to the turbine of such metal spray guns.

The foregoing specific description is for purposes of illustration and not of limitation and it is therefore our intention that the invention be limited only by the appended claims or their equivalents wherein we have endeavored to claim broadly all inherent novelty.

We claim:

1. In a metal spray gun construction of the wire feed type, having a compressed gas motor, the improvement comprising a rotor for said motor, first means rotatable with said rotor and comprising at least one arm centrifugally deflectible with respect to its axis of rotation, against spring resistance, from a predetermined initial angular pre-set position with respect to said axis, second means composed of at least one first and at least one second friction element, one of said first and said second friction elements being rotatable with said rotor and the other mounted for braking action with said rotatable friction element upon frictional engagement therebetween, said first friction element being positioned and adapted to be operatively acted upon by said first means upon centrifugal actuation thereof for total mechanical movement defined by the centrifugal movement of said first means between minimum and maximum operating speeds of said rotor to frictionally bear against said second friction element to thereby effect speed control of said rotor by power absorption, and means for variably adjusting the relative position of distance between said second friction element and the initial position of said first friction element throughout the range of said total mechanical movement of said first friction element.

2. A variable speed governing mechanism for a metal spray gun construction of the wire feed type, having a compressed gas motor, comprising at least one arm, rotatable with the rotor of said motor and centrifugally deflectable for angular deflection with respect to its axis of rotation, against spring resistance, from a predetermined initial position of at least 60° with respect to said axis, speed control means composed of at least one first and at least one second friction element, one of said first and said second friction elements being rotatable with said rotor and the other mounted for braking action with said rotatable friction element upon frictional engagement therebetween, said first friction element being positioned and adapted to be operatively acted upon by said arm upon centrifugal actuation thereof for total mechanical movement defined by angular deflection of said arm between minimum and maximum operating speeds of said rotor to frictionally bear against said second friction element to thereby effect speed control of said rotor by power absorption, and a separate mechanism for variably adjusting the relative position of distance between said second friction element and the initial position of said first friction element throughout the range of said total mechanical movement of said first friction element.

3. A variable speed governing mechanism in accordance with claim 2 in which at least one of said friction elements defines, at least substantially on the path of frictional travel of the other friction element when in frictional contact, a friction surface substantially constituted of spray metal.

4. A variable speed governing mechanism for a metal spray gun construction of the wire feed

type having a compressed gas motor, comprising at least one arm rotatable with the rotor of said motor and centrifugally deflectable for angular deflection with respect to its axis of rotation, against spring resistance, from a predetermined initial position of at least 60° with respect to said axis, speed control means composed of at least one first friction element and one second friction element said second friction element being mounted for braking action with said first friction element upon frictional engagement therebetween, said first friction element being rotatable with said rotor and mounted for total mechanical movement defined by angular deflection of said arm between maximum and minimum operating speeds of said rotor to contact said second friction element and for operative actuation by said arm upon centrifugal actuation thereof to bear frictionally against said second friction element to thereby effect speed control of said rotor by power absorption, and a separate mechanism for variably adjusting the relative position of distance between said second friction element and the initial position of first friction element throughout the range of said total mechanical movement.

5. A variable speed governing mechanism in accordance with claim 4, except that said arm is centrifugally deflectable from a predetermined initial position of from 70 to 80° with respect to said axis.

6. A variable speed governing mechanism in accordance with claim 4, in which said arm carries a weight adjacent the end thereof.

7. A variable speed governing mechanism in accordance with claim 4, in which said first friction element is defined by a weight carried by said arm adjacent the end thereof and in which said second friction element is defined by at least one friction surface along the rotary path of travel of said weight.

8. A variable speed governing mechanism for a metal spray gun construction of the wire feed type having a compressed gas motor, comprising at least one arm carrying a weight adjacent the end thereof, said arm being rotatable with the rotor of said motor and centrifugally deflectable for angular deflection with respect to its axis of rotation, against spring resistance, from a predetermined initial position of at least 60° with respect to said axis, speed control means composed of at least one first friction element and one second friction element said second friction element being mounted for braking action with said first friction element upon frictional engagement therebetween, said first friction element being rotatable with said rotor and mounted for total mechanical movement defined by angular deflection of said arm between maximum and minimum operating speeds of said rotor to contact said second friction element and for operative actuation by said arm upon centrifugal actuation thereof to bear frictionally against said second friction element to thereby effect speed control by power absorption, at least one resilient clip adapted and positioned to rotate with and bear against said arm, and a separate mechanism for variably adjusting the relative position of distance between said second friction element and the initial position of said first friction element throughout the range of said total mechanical movement.

9. A variable speed governing mechanism for a metal spray gun construction of the wire feed type having a compressed gas motor, comprising at least one arm carrying a weight adjacent the end thereof, said arm being rotatable with the rotor of said motor and centrifugally deflectable for angular deflection with respect to its axis of rotation, against spring resistance, from a predetermined initial position of at least 60° with respect to said axis, speed control means composed of at least one first friction element defining at least one first friction surface rotatable with said arm and one second friction element defining at least one second substantially stationary friction surface aligned with the rotary path of travel of said first friction surface said second friction surface being mounted for braking action with said first friction surface upon frictional engagement therebetween, said first friction surface being slidably mounted substantially coaxial with the axis of rotation of said arm for total mechanical movement defined by angular deflection of said arm between maximum and minimum operating speeds of said rotor to contact said second friction surface and for operative actuation by said arm upon centrifugal actuation thereof to bear frictionally against said second friction surface to thereby effect speed control of said rotor by power absorption, and a separate mechanism for variably adjusting the relative position of distance between said second friction surface and the initial position of said first friction surface throughout the range of said total mechanical movement.

10. A variable speed governing mechanism in accordance with claim 9 except that said arm is centrifugally deflectable from a predetermined initial position of from 70° to 80° with respect to said axis.

11. A variable speed governing mechanism for a metal spray gun construction of the wire feed type having a compressed gas motor, comprising at least one arm rotatable with the rotor of said motor and centrifugally deflectable for angular deflection with respect to its axis of rotation, against spring resistance, from a predetermined initial position of at least 60° with respect to said axis, a first friction surface substantially coaxial with the axis of rotation of said arm, a second friction surface, substantially coaxially rotatable with said arm and slidably mounted intermediate said arm and said first friction surface for operative actuation by said arm, against the force of a spring and for total mechanical movement defined by the total angular deflection of said arm between maximum and minimum operating speeds of said rotor to frictionally bear, upon centrifugal actuation of said arm, against said first friction surface, said first friction surface being mounted for braking action with said second friction surface upon frictional engagement therebetween and a separate mechanism for variably adjusting the relative position of distance between said second friction surface and the initial position of said first friction surface throughout the range of said total mechanical movement.

12. A variable speed governing mechanism in accordance with claim 11 except that said arm is centrifugally deflectable from a predetermined initial position of from 70 to 80° with respect to said axis.

13. A variable speed governing mechanism in accordance with claim 11 in which said first friction surface, at least substantially on the path of frictional travel of said second friction surface, is constituted of spray metal.

14. A variable speed governing mechanism in accordance with claim 11 in which said first friction surface carries at least one radial groove substantially on the path of frictional travel of said second friction surface.

15. A variable speed governing mechanism in accordance with claim 11 in which said second friction surface is operatively actuated by said arm by means of a springy clip connected with one end to said second friction surface and bearing with the other end against said weight.

16. A variable speed governing mechanism in accordance with claim 11 in which additional means are provided which are positioned and adapted to direct exhaust gas from said motor to the proximity of at least one of said friction surfaces.

17. A variable speed governing mechanism for a metal spray gun construction of the wire feed type having a compressed gas motor, comprising at least two substantially coaxially aligned, symmetrically arranged, radial spring arms, carrying weights adjacent the free ends thereof and rotatable with the rotor of said motor and centrifugally deflectable for angular deflection with respect to their axis of rotation, against their spring resistance, from a predetermined initial position of at least 60° with respect to said axis, a first friction surface substantially coaxial with the axis of rotation of said arms, a second friction surface, substantially coaxially rotatable with said arms and slidably mounted intermediate said arms and said first friction surface for operative actuation by said arms, against the force of a spring, to frictionally bear, upon centrifugal actuation of said arms, against said first friction surface, a ring member in axially substantially fixed position with respect to the said axis, carrying said first friction surface in thread-engagement therewith, and rotatable around said axis, and means preventing rotation of said first friction surface when rotating said ring member.

18. A variable speed governing mechanism in accordance with claim 17 in which said second friction surface is operatively actuated by said arms by means of spring clips, at least one for each of said arms and each connected with one end to said second friction surface and bearing with the other end against the weight of its arm.

19. A variable speed governing mechanism for a metal spray gun construction of the wire feed type having a compressed gas motor, comprising a shaft in direct drive connection with the rotor of said motor, an outer surface threaded ring member defining a substantially annular first friction surface in substantially coaxial alignment with said shaft, at least two substantially coaxially aligned, diametrically opposed, radial spring arms, carrying weights adjacent the free ends thereof, mounted on said shaft, bent away from said first friction surface in a predetermined initial position of from 70° to 80° with respect to said shaft, a spring, a substantially annular second friction surface, coaxially mounted on said shaft, intermediate said arms and said first friction surface, in axially slidable engagement with said shaft, against the force of said spring, spring clips, at least one for each of said arms and each connected with one end to said second friction surface and fitting with the other end in radially slidable engagement over the weight of its arm, an inner-surface threaded ring member, mounted in longitudinally substantially fixed, rotatable position with respect to said shaft, carrying said first friction surface in thread-engagement therewith, means preventing rotation of said first friction surface when rotating said ring member and means for passing cooling gas to the proximity of said friction surfaces.

20. A variable speed governing mechanism in accordance with claim 17 in which said first friction surface is constituted of spray metal, in which said shaft carries a washer positioned and adapted to centrifugally supply lubricating material, fed to said washer, to at least one of said friction surfaces and in which said first friction surface carries a multiple of radially arranged grooves.

21. In a metal spray gun construction of the wire feed type, having a compressed gas motor, the improvement comprising a rotor for said motor, at least one first and one second friction element substantially adjacent each other, one rotatable with said rotor and the other mounted for braking action with said rotatable friction element upon frictional engagement therebetween, said first element being mounted for operative actuation by centrifugal speed responsive means and for total movement defined by the operating positions of said speed responsive means between minimum and maximum operating speeds of said rotor to frictionally cooperate with the other to thereby absorb power, a separate mechanism for variably adjusting the relative position of distance between said second element and the initial position of said first element throughout the range of said total movement of said first element, and means positioned and adapted to direct cooling gas to the proximity of at least one of said friction elements.

22. In a metal spray gun construction, the improvement in accordance with claim 21 in which the said last mentioned means are positioned and adapted to direct exhaust gas from said motor to the proximity of at least one of said friction elements.

23. In a metal spray gun construction, the improvement in accordance with claim 21 in which at least one of said friction elements defines, at least substantially on the path of frictional travel of the other friction element when in frictional contact, a friction surface substantially constituted of spray metal.

HERBERT S. INGHAM.
ARTHUR P. SHEPARD.