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SPRAY CHARACTERISTICS

OF

A POPPET TYPE VALVE

A THESIS

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the Faculty of the Graduate Division

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OF

A POPPET TYPE VALVE

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LIST OF ABBREVIATIONS AND SYMBOLS

Chapter II

d = Drop size diameter

D = Orifice diameter

$\frac{d}{D}$ = Relative drop size in dimensionless study

$\frac{l}{D}$ = Ratio of length to diameter of the atomizer's orifice

v = Jet velocity

v_1 = Absolute viscosity of liquid

v_a = Absolute viscosity of air

$\theta = 2\alpha$ = spray cone angle

ρ_l = Density of liquid

ρ_a = Density of surrounding air

γ_l = Surface tension of liquid

f, F, ϕ, ψ = Function symbol.

Chapter III

c = Coefficient of viscous damping

$D = \frac{d}{dt}$ = operator

F_1 = Inertia force

F_2 = Viscous damping force

F_3 = Spring force

F_4 = External force

f = Frequency

g = Acceleration due to gravity

k = Spring constant

m = Poppet valve mass + 1/3 spring mass

m_w = Mass flow rate

n = Root of the characteristic equation

Q = Volume flow rate

T = Time per cycle

V_1 = Exit velocity

V_2 = Valve velocity

x = Variable distance above or beyond x_o

x_o = Average distance from the nozzle during vibration

$\dot{x}, \ddot{x} = \frac{dx}{dt}, \frac{d^2x}{dt^2}$

γ = Specific weight of water

ω = Angular velocity.

Chapter VI

A = Outlet flow area at the orifice

A = Cross-section area at the valve gland

A_1 = Orifice area

A_2 = Cross-section area of the stem

a = Equilibrium elongation of the spring

C = Volume of liquid flow through the nozzle in 1 second, when the valve is separated one unit from the nozzle

C_d = Coefficient of discharge

E = Bulk modulus of elasticity

h = Pressure head

p = Pressure drop across the orifice

p = Pressure inside the nozzle

p_o = Average pressure at average distance x_o

V = Jet velocity = $\sqrt{2gh}$

$\bar{V}$ = Volume of the chamber.

SUMMARY

The purpose of this thesis is threefold: (1) the analytic study of valve vibration (2) evaluation of valve performance and (3) the determination of the spray characteristics.

Spray characteristics and their influences are summarized as the background of the study. Then the analysis of valve vibration is performed by postulating the system with some assumptions for the purpose of simplification. An outline to the vibration design is also suggested. Two approaches of analysis are made, one by direct derivation and the other, as a correction, is by the analogy to that derived by Den Hartog on the diesel-engine injection-nozzle. The system is found to be self-excited under influence of the compressibility of fluid which causes pressure variation inside the nozzle.

Experiments were performed for the evaluation of frequency of vibration and determination of the flow rate and pressure relation. A sequence of photographs was taken to study the spray development and some spray characteristics can also be observed. When one design of valve failed to vibrate longitudinally, another was designed in the hope that such vibration would occur. This valve, too, failed to vibrate. After consideration of the analytical study, conclusions are reached as to the cause of failure.

Results obtained from the work done show that:

1. The characteristic curve between flow rate and pressure is independent of vibration.

2. Concerning the spray characteristics, better atomization results by high pressure injection and, as spray is fully developed, spray cone angle depends on the valve angle.
3. Valve vibration could not be obtained from the experiment due to its complexity and lack of time but vibration seems possible.

Also, it is suggested that, spray obtained from a vibrating valve will have better characteristics especially in spray fineness and homogeneity which are suited to the requirements of the combustion process of the internal combustion engine.

CHAPTER I

INTRODUCTION AND OBJECT

Power produced in an engine cylinder does not depend entirely on the amount of fuel drawn into it, but also depends upon other characteristics of the mixture such as its distribution and homogeneity. The air-fuel mixing device is known as a carburetor, a term which is misnomer in the scientific sense, implying as it does something which supplies or adds carbon: a carburetor does not do so. The functions of the carburetor are the mixing and supplying of the air-fuel mixture at almost a constant ratio and in proportion according to the requirements of the engine at any speed. It is an extremely important part in the complex problem of spark ignition engines.

In 1936, Bendix Aviation Corporation of America came out with a newly developed type of carburetor, the Injection Carburetor, the method employed is the injection of fuel under low pressure from a spray nozzle into an air stream passing through an induction manifold. The amount of fuel is controlled by the pressure drop across the venturi tube. The possibility exists that this type of carburetor is a means for improving fuel metering. The greater the atomization of liquid fuel the greater the homogeneity of the mixture and therefore the more effective the combustion should be. Combustion determines, to a large degree, the fuel economy, smoothness of operation, acceleration characteristic and the engine performance.

This circumstance suggests a study directed toward examining the possibility of spray improvement under low pressure.

Placing a spring loaded poppet valve on the outlet side of the spray nozzle and allowing the valve to vibrate under influences of the flow provided a means for carrying out the experiment. Preliminary observations indicated that the spray obtained from this unit gave better atomization.

This thesis, then, was concerned with (1) analytic studies of the valve vibration, (2) evaluation of valve performance and (3) the determination of spray characteristics.

CHAPTER II

BACKGROUND ON SPRAY CHARACTERISTICS

Previous work has been done in the study of the following spray characteristics:

<u>Spray performance</u> - This is the study of spray shape from the nozzle into its overall configuration. Examination of this characteristic is for the utility of spray prediction.

<u>Spray penetration</u> - This term is used for the determination of the concentration and compactness of the spray from a nozzle.

<u>Spray cone-angle</u> - This is one of the most important spray characteristics. The angle is measured by drawing a tangent line from the orifice to the spray envelope.

<u>Spray dispersion</u> - This characterizes the distribution quality of spray and is associated directly with the spray cone-angle. The measurement of this characteristic is the ratio of the volume of liquid to the volume of air in the spray cone. Its decrease denotes a better dispersed spray.

<u>Droplet size</u> - This is another important property in defining the spray fineness. Due to their variation throughout the spray, then the concept of "mean drop size" is established to qualify the degree of atomization.

<u>Spray uniformity</u> - This term indicates the variation of the drop size and homogenity of the spray.

Observation and investigation to find the effects upon these spray characteristics have been performed by many persons either analytically or experimentally. Conclusions have been drawn and curves plotted to explain the spray phenomenon under each influence factor. Typical effects are summarized briefly here.

Atomizer Design.--This is the primary factor, for the quality of a spray in a given set of conditions determines the atomizer efficiency. Two essential types of atomizers have been designed, the plain atomizer and the swirl atomizer, depending on the goal of application. In a plain atomizer, the flow from the nozzle is considered to be in an axial direction for laminar flow and in both axial and radial for turbulent flow, while a swirl atomizer is considered to have axial, radial and tangential direction.

The orifice shape affects the spray appearance and the coefficient of discharge which will affect the velocity of discharge and hence the atomization. It has been found that the coefficient of discharge for the plain atomizer is higher than in the swirl type and the fineness of atomization depends on what type of atomizer is being used. The plain atomizer gives a narrow cone angle, large droplet size but penetrating spray. The swirl type creates a wide cone-angle, soft spray and a better degree of spray dispersion. Results from experiments have shown that a ratio /D (ratio of length to diameter of the atomizer's orifice) also affects the spray penetration and cone angle. An increase in orifice size results in the increasing of size and number of biggest droplets. Unfortunately, the importance of the atomizer becomes less when a high pressure system is used in the injection and high density medium is introduced.

Physical Significance of the Liquid.--Physical effects of the spray characteristics have also been studied extensively from different standpoints. It has been found that the specific gravity or density, surface tension and viscosity are the most important variables. Among these three variables, viscosity seems to be of the greatest influence, for example, the approximate ratio of specific gravity and surface tension of kerosene and light fuel oil are $\frac{8}{9}$ and $\frac{24}{27}$ respectively, but it is $\frac{1}{40}$ for the viscosity ratio.

In an atomization process, beginning with a flow from an orifice, high viscosity means less tendency for the jet to disintegrate. Then, breaking up of the fluid into separate droplets is due to the magnitude of surface tension force. Finally, subdivision of droplets to atomization is under influences of many dependent variables, such as specific gravity, viscosity, surface tension and jet velocity.

An increase in viscosity results in getting a more compact spray and increase in droplet size, but a decrease in spray cone angle, distribution and spray uniformity. Viscosity, internally, retards the rate of flow within the nozzle, and, externally, decreases the rate of breaking up of ligaments and the rate of splitting up of droplets. Sizes of spray angles are influenced largely by the viscosity. Surface tension has the tendency to oppose jet distortion and disintegration of ligaments. An increase in density is a means of spray penetration but it makes difficult the dispersion into the surrounding medium.

Surrounding Medium.--Besides the atomizer itself and the fluid properties, another important factor affecting spray characteristics is the condition of the surrounding medium into which the fluid is being injected. Experiments show that increasing air density will improve the spray up to an optimum degree and further increase of the air density has very little effect. Greater effect of air resistance is expected at lower injection pressure and higher viscosity of the fluid.

Spray penetration and droplet velocity are reduced by a high pressure medium. Advantages are the increasing of spray dispersion, cone angle and tendency to splitting up of droplets. Also, it improves the disintegration of spray. Increasing of air temperature will increase the spray velocity which has some effect on spray cone angle except in the case of high pressure injection. Faster rate of evaporization and smaller droplets size are resulted in using high temperature of the surrounding medium. Investigation of moving medium, introducing of turbulent air will break up spray better than still air, but it will slightly decrease the spray uniformity and penetration.

Injection Pressure.--This is the most important single influence upon atomization of liquid and therefore in turn on the combustion process. Considering the case of combustion, atomization is the process of breaking up fluid into a great number of small droplets providing a great surface area to make the preparation more combustible.

High pressure has special effect upon spray fineness. It will affect the spray cone angle within a certain range until the spray is fully developed, beyond this, increase in pressure will have no effect. It was believed that increasing pressure would increase spray penetration but experiment showed that this was not true. An explanation for this phenomenon is this: as pressure is increasing, there is a tendency to decrease droplet size which reduces spray penetration, the increase of penetration by pressure on one hand and the decrease by influence of drop size on the other hand will balance each other, hence no effect results.

In the field of dimensional analysis for the study of relative drop size (1) useful variables are chosen:

Drop size diameter	d
Orifice diameter	D
Jet velocity	v
Spray cone angle	$\theta = 2\alpha$
Density of liquid	ρ_l
Absolute viscosity of liquid	v
Surface tension of liquid	γ_l
Density of surrounding air	ρ_a
Absolute viscosity of air	v_a

A relationship may be expressed by writing

$$f(d, D, v, \theta, \rho_l, v_l, \gamma_l, \rho_a, v_a) = 0$$

Using the three fundamental units, mass M, length L and time T, and writing down all variables concerned in terms of these primary units, then solving by dimensional analysis (2) the results are:

$$F\left(\frac{d}{D}, \frac{Dv\rho_l}{v_l}, \frac{D\rho_l\gamma_l}{{v_l}^2}, \frac{\rho_l}{\rho_a}, \frac{v_l}{v_a}, \theta\right) = 0$$

or

$$\frac{d}{D} = \psi\left(\frac{Dv\rho_l}{v_l}, \frac{D\gamma_l \rho_l}{v_l^{\ 2}}, \frac{\rho_l}{\rho_a}, \frac{v_l}{v_a}, \theta\right)$$

where,

f, F and ψ represent function symbol.

$\frac{d}{D}$ represents relative drop size.

By squaring the first dimensionless group of the right hand side and dividing by the second, we obtain another dimensionless group, $\frac{Dv^2\rho_l}{\gamma}$, and replacing this group instead of the second, we have

$$\frac{d}{D} = \phi\left(\frac{Dv\rho_l}{v_l}, \frac{Dv^2\rho_l}{\gamma_l}, \frac{\rho_l}{\rho_a}, \frac{v_l}{v_a}, \theta\right)$$

These six dimensionless groups represent the functional relationship equation. The analysis by this method only tells which dimensionless groups are involved. The direct equation will be found only experimentally. Two important groups, $\frac{Dv\rho_l}{v_l}$, Reynolds Number, and $\frac{Dv^2\rho_l}{\gamma_l}$, called the Weber Number, are considered to be important. The former represents the flow phenomenon, to justify that whether it is laminar or turbulence. The latter indicates the ratio of inertia force to the surface tension force of fluid.

CHAPTER III

VIBRATION STUDY

Analysis of Vibration.--The concept of vibration is defined as the behavior of a body under the influence of variable forces. Its motion is periodic, repeats itself in equal intervals of time and its characteristic is a complex one. In many cases, vibration has resulted in failure of machine parts, undesirable noise and improper operation. However, it is more desirable in some other case as in musical sounds, vibration producing machine and in our problem.

In order to perform the analytical solution, it is necessary to make some assumptions to our system for the sake of simplicity; referring to Figure 1, our assumptions are:

a. that the flow is steady

b. that the vibration is of one degree freedom in the direction of the flow through the nozzle,

c. and that all forces acting downward are positive forces.

Let x_o be the average distance from the nozzle, and all variable distances above or beyond x_o is assigned by x (can be plus or minus).

The analysis starts with an examination of all forces in the system with respect to their magnitude and direction.

Inertia force, F_1, is the force due to mass of the valve and the spring (usually adds one-third of the springs mass), acting opposite to the direction of motion. It is equal to $-m\frac{d^2x}{dt^2}$.

Viscous damping force, F_2, is the resisting force which is directly proportional to the velocity of motion, and acts upward when motion is downward and vice-versa. According to our assumption, this force will be $-C\,\frac{dx}{dt}$.

Spring force, F_3, is the restoring force which pulls upward on the mass when the displacement is downward. Thus this force is expressed by $-k(x_o + x)$.

External force or exciting force, F_4, is the force acting on the valve by the flow of fluid from the nozzle in positive direction.

At any position of x, summation of forces are zero, then

$$-m\,\frac{d^2x}{dt^2} - C\,\frac{dx}{dt} - k(x_o + x) + F_4 = 0 \qquad (1)$$

One more assumption will be made for the utility of calculating the force F_4: that there should be no fluid leaving the column until it strikes the valve.

By momentum theory, a force acting on a body is equal to the time rate change of linear momentum of the body in direction of the flow. Let V_1 and V_2 be the exit flow velocity from the nozzle and the valve velocity respectively, and m_w be the mass flow rate of the fluid column, we may write

$$F_4 = m_w(V_1 - V_2)$$

Introducing $m_w = \frac{Q\gamma}{g}$, where Q and are volume flow rate and specific weight respectively, hence

$$F_4 = \frac{Q\gamma}{g}\,(V_1 - V_2) \qquad (2)$$

The velocity V_1 is considered to be constant for a particular flow rate and the velocity V_2 is a function of displacement x which permits us to write $\frac{dx}{dt}$; then equation 2 becomes

$$F_4 = \frac{Q\gamma}{g}\left(V_1 - \frac{dx}{dt}\right) \tag{3}$$

Substitution of equation 3 into equation 1, we get

$$-m\frac{d^2x}{dt^2} - C\frac{dx}{dt} - k(x_o + x) + \frac{Q\gamma}{g}\left(V_1 - \frac{dx}{dt}\right) = 0$$

By rearranging terms and simplifying, we obtain

$$\frac{d^2x}{dt^2} + \frac{1}{m}\left(C + \frac{Q\gamma}{g}\right)\frac{dx}{dt} + \frac{k}{m}x = \frac{1}{m}\left(\frac{Q\gamma V_1}{g} - k x_o\right) \tag{4}$$

The two constant terms of the right hand side will vanish for they balance each other, then equation 4 reduces to

$$\frac{d^2x}{dt^2} + \frac{1}{m}\left(C + \frac{Q\gamma}{g}\right)\frac{dx}{dt} + \frac{k}{m}\ x = 0 \tag{5}$$

This is the expression of our vibrating system in the differential form which can be solved by the operator method (3). Denoting $\frac{d}{dt}$ by D, equation 5 gives

$$D^2x + \frac{1}{m}\left(C + \frac{Q\gamma}{g}\right)Dx + \frac{k}{m}\ x = 0$$

or

$$\left[D^2 + \frac{1}{m}\left(C + \frac{Q\gamma}{g}\right)D + \frac{k}{m}\right] x = 0$$

The characteristic equation of the system is

$$n^2 + \frac{1}{m}(C + \frac{Q\gamma}{g})n + \frac{k}{m} = 0$$

then

$$n = \frac{-\frac{1}{m}(C + \frac{Q\gamma}{g}) \pm \sqrt{\left[\frac{1}{m}(C + \frac{Q\gamma}{g})\right]^2 - 4\frac{k}{m}}}{2}$$

or

$$n = -\frac{1}{Cm}(C + \frac{Q\gamma}{g}) \pm \sqrt{\left[\frac{1}{2m}(C + \frac{Q\gamma}{g})\right]^2 - \frac{k}{m}} \qquad (6)$$

It is necessary to have the value of $\sqrt{\frac{1}{2m}(C + \frac{Q\gamma}{g})^2 - \frac{k}{m}}$ be an imaginary part for instability of the system which brings up to the criterion

$$\left[\frac{1}{2m}(C + \frac{Q\gamma}{g})\right]^2 < \frac{k}{m}$$

or

$$K > \frac{1}{4m}(C + \frac{Q\gamma}{g})^2 \qquad (7)$$

thus the angular velocity is

$$\omega = \sqrt{\frac{k}{m} - \frac{1}{2m}(C + \frac{Q\gamma}{g})^2} \qquad (8)$$

Time for one completed cycle is defined as

$$T = \frac{2\pi}{\omega} = \frac{1}{f} \qquad (9)$$

Hence

$$f = \frac{1}{2\pi}\sqrt{\frac{k}{m} - \left[\frac{1}{2m}(C + \frac{Q\gamma}{g})\right]^2} \qquad (10)$$

Referring to the foregoing analysis, many assumptions had been presumed in an attempt to show the possibility and circumstance under which vibration will occur: One must keep in mind that, any analytic solution of a practical engineering problem will have an accuracy and reliability only as good as the original assumptions demand.

Outline of Design.—Equation 7 and 10 are used as the fundamental basis in the design consideration. The term C in equation 7 depends on properties of the system, such as geometrical arrangement, fluid properties and material used. No general value can be determined for this term, since it varies from one system to another, therefore it is a factor which makes difficulty in the design problem. Trial and error is used in the design on the arrangement of spring constant (k) and mass (m) until it produces vibration. The frequency obtained from different values of the flow rate (Q) and all known values are substituted in equation 10 to solve for the values of C. Having determined the average value C, then the second design can be determined formally on the basis of the vibration criterion.

CHAPTER IV

APPARATUS

Pressure Gage.--

Model A

Range 0-100 psi

Make Foxboro, Mass., U.S.A.

Flowrator Meter:

Model	3-1815/1
Connection Pipe Size	$\frac{1}{2}$ inch
Make	Fisher and Porter, Hatboro, Pa., U.S.A.

Atomizer.--Two atomizers were used in the experiment, see Figure 2 and 3. The first, consists of a nozzle through which fluid is being injected, a coil spring of which one end attaches to an adjustable screw and the other end fixes to a poppet valve's rod. The spring holds the valve to seat on the outlet. The atomizer body provides a connection for tapping to the pipe line, using city water supply as the injection fluid. The second one resembles the first, differing in the arrangement of the valve and spring adjustment, the passage of the flow into the atomizer and the geometric configuration.

CHAPTER V

PROCEDURE

Before the experiment was performed, the pressure gage was tested to find the correction curve. The flowrator was also recalibrated in order to read G.P.M. of water on the millimeters scale. Their plots are shown in Appendix C.

Figure 4 shows the schematic diagram of the apparatus arrangement. The first atomizer, A, was tested at several pressure levels. The spring constant was 2.2 $\frac{lb_f}{m}$ and the valve weight 0.00242 lb_f. Rate of flow was recorded against each indicated pressure. Two different spring forces were used in this test, resulting in Table 1. Unfortunately, the frequency of vibration could not be observed by a stroboscope, for the vibrations were irregular both in motion and direction. In order to obtain vibration in one direction and eliminating off the other motions, then, the second atomizer, B, was designed, having the valve motion up and down along the guide rod with an adjustable spring tension externally. The experiment was carried on with the same procedure as before, but instead of using the flowrator, rate of flow was measured by the weight obtained on the balance within an interval time. Three values of spring constant were used. Again pressure and corresponding rate of flow were recorded, with no sign of any vibration.

A sequence of pictures were taken to study the spray development from the nozzle, see figure 6.

CHAPTER VI

DISCUSSION OF RESULTS

After the experiments were performed and data plotted, the results were found unsatisfactory from the standpoint of valve vibration. We shall discuss the reasons now.

Rate of Flow-Pressure Relation.--Figure 8 is the plot between the rate of flow from the nozzle and the gage pressure, when a spring constant of 0.287 $\frac{lb_f}{in.}$ is used. The spring force of curve I and II are 0.01795 lb_f and 0.0448 lb_f respectively. The plots show that they start from the points where the pressure force is just equal to the spring force, or

$$pA = ka \tag{11}$$

where

p = pressure inside the nozzle

a = equilibrium elongation of the spring

The reason why they are curves may be explained by the equation

$$Q = C_d AV$$

$$= C_d A \sqrt{2gh}$$

$$= C_d A \sqrt{2g \frac{p}{\gamma}}$$

$$Q = C_1 \sqrt{p} \tag{12}$$

where

C_d = coefficient of discharge

A = outlet flow area at the orifice in^2

h = pressure head in. of water

p = pressure drop across the orifice = $h\gamma$ $\frac{lb_f}{in^2}$

= specific weight $\frac{lb_f}{cu.ft.}$

$C_1 = C_dA\sqrt{\frac{2g}{\gamma}}$ = constant for a particular flow rate.

As pressure is increased, the curves I and II tend to have less deviation from one another, suggesting that, when pressure is high enough these two curves will join together at the point where the influence of pressure upon the spring constant has vanished. Beyond this point, they will always coincide.

Figure 9 and 10 show the same characteristic curves as in Figure 8 for different spring constant and spring forces, and also does Figure 7.

The experiment of the vibrating spray nozzle by Sliepcevich, Consiglio and Kurata (4) were done at rather high pressure and their results show the same shape of curves plotted between flow rate and gage pressure. And, at the point where the curves coincided, they explained that, "at this pressure, the head of the valve stem is sufficiently removed from the orifice seat so that it no longer affects the flow characteristics through the annular discharge."

A comparison of the characteristic curve of the non-vibrating valves obtained by the author and of vibrating valves obtained from the experiment of Sliepcevich, Consiglio and Kurata, indicates that there will be no effect upon the characteristic curve between the flow rate and the pressure so far as vibration concerned.

Spray Development.--Figure 6 is a photographic series for different stages of spray from the nozzle, using a spring constant 0.287 $\frac{lb_f}{in.}$ and set-up spring force of 0.036 lb_f. As soon as the pressure was sufficiently large to overcome the spring force, the valve opened forcing the water through the nozzle.

At the earliest stage when pressure was very small, the water wetted the valve and flowed along the wetted surface. The spray tended to spread out as the pressure was increased. At a pressure of 4 psi., Figure 6-a, the spray started to form a bulbar shape under the influence of primarily surface tension and viscosity. The increase of pressure is the cause of the increase of radial force which tries to overcome the surface tension force pulling the spray outward. At a pressure of 4.5 psi., Figure 6-b, the spray formation changed from the bulbar shape becoming tulip-like. With a further increase of pressure, Figure 6-c and d, the spray gradually spread out characterized by the opening of the tulip. The full development of the spray, as shown in figure 6-e, occurred around a pressure of 55 to 60 psi., and had no further effect to the spray formation after this point. Also, at this stage, the spray cone angle depends on the valve angle and the seat. A mist of spray drops were observed at rather high pressure.

Self-Excited Vibration.--A close examination of the result obtained from the vibration equation by the author, considering in the standpoint of momentum theory, brings out the criterion

$$k > \frac{1}{4m}\left(c + \frac{Q\gamma}{g}\right)^2 \tag{7}$$

It indicates that vibration is limited by the flow rate for particular values of k and m. When Q is great enough to make the right side greater than k, vibration will stop, a result which does not seem intuitively correct. Hence, there might be some controversy concerning the original assumptions.

This brings up the question as to the justification of making those original assumptions. The greatest weakness in the plan of attack was the ignoring of the pressure variation inside the nozzle.

Another solution of our vibrating system may approach by analogy to that derived of Den Hartog (5), for diesel fuel injection valve, see Figure 5. He made the assumptions

(a) that the in-flow to the nozzle is constant

(b) and that the out-flow from the nozzle is varying depending on the distance of the valve lifted from the seat.

Having made the two assumptions, he then explained the physical phenomenon by pressure variation inside the nozzle due to the inequality of the in-flow and out-flow. (Another explanation has been stated by Langharne (6) by the theory of the pressure wave propagation of fluid). The vibration resulting from this phenomenon, he called self-excited or self-induced vibration. This type of vibration was defined by Baker (7) as "the phenomenon in which the alternating forces furnishing the energy to the vibration are controlled by the motion in contradistinction to a force vibration, where the force depends on time only."

The derivation is verified in the following steps.

Let x_o be the average distance from the nozzle during vibration, and p_o the average pressure at the distance x_o.

The general equation of motion may be written in the form

$$m\ddot{x} + c\dot{x} + kx_o + kx - p_oA - pA = 0 \tag{13}$$

The term p_oA and kx_o are equal in magnitude and opposite in direction, hence

$$m\ddot{x} + c\dot{x} + kx - pA = 0 \tag{14}$$

Employing the definition of elasticity (8) which is "the fraction of the change in volume per unit volume per unit change in pressure", mathematically written as

$$E = \frac{\frac{dv}{v}}{\frac{1}{dp}}$$

or

$$\frac{\dot{p}}{E} = \frac{\dot{v}}{v} \tag{15}$$

where $\dot{v}$ and $\dot{p}$ represent $\frac{dv}{dt}$ and $\frac{dp}{dt}$ respectively.

Hartog also made the approximation that the velocity of oil flowing out through the nozzle is proportional to the distance of the valve from the nozzle.

The rate of change of volume inside the nozzle is made of two consecutive values, the excess volume of oil flowing in the nozzle per unit second and the volume change in oil chamber due to the valve being in motion, they are

$$\dot{v} = -(Cx + A\dot{x}) \tag{16}$$

where

C = volume of oil flow through the nozzle in 1 second when the lift is unity.

A = cross-section area of the stem at the gland.

A negative sign means the contraction in volume due to compression on fluid.

Hence, equation 15 becomes

$$\dot{p} = \frac{E}{v}(Cx + A\dot{x}) \tag{17}$$

Differentiation of equation 14 with respect to time and the substitution of equation 17 into it, gives

$$m\dddot{x} + c\ddot{x} + (k + \frac{A^2E}{v})\dot{x} + \frac{AEC}{v} \; x = 0 \tag{18}$$

Employing method of operator integration, leads to the criterion of vibration possibility, provides that

$$\frac{c}{m}(\frac{k}{m} + \frac{A^2E}{mv}) < \frac{AEC}{mv}$$

or

$$c < \frac{CE}{v} \; \frac{mA}{(k + \frac{A^2E}{v})} \tag{19}$$

The analogy of this equation to our system, see figure 3, results

$$c < \frac{CE}{v} \frac{m(A_1-A_2)}{k + \frac{(A_1-A_2)^2E}{v}} \tag{20}$$

and the frequency equation is

$$f = \frac{\omega}{2\pi} = \frac{1}{2\pi}\sqrt{\frac{CE}{v}\,\frac{m(A_1-A_2)}{k + \frac{(A_1-A_2)^2E}{V}} - c} \tag{21}$$

where

A_1 =	orifice area	in.^{2}
A_2 =	cross-section area of the stem	in.^{2}
c =	viscous damping coefficient	$\frac{lb_f\text{-sec.}}{in.}$
C =	volume of liquid flow through the nozzle in 1 second when the valve is one unit apart from the nozzle	$\frac{in.^2}{sec.}$
m =	the mass of the valve + 1/3 (spring mass)	lb_m
v =	volume of the chamber	in.^{3}
E =	bulk modulus equal to 300,000 $\frac{lb_f}{in.^2}$ for water (9)	
k =	spring constant	$\frac{lb_f}{in.}$

CHAPTER VII

CONCLUSIONS AND RECOMMENDATIONS

Time was not available to obtain all data that the author wished to secure because of the difficulties which had been discussed in the foregoing chapters. However, conclusions and recommendations can be made as a result of the work done on this thesis.

From the data obtained, it is evident that the characteristic curve between rate of flow and pressure is independent of vibration.

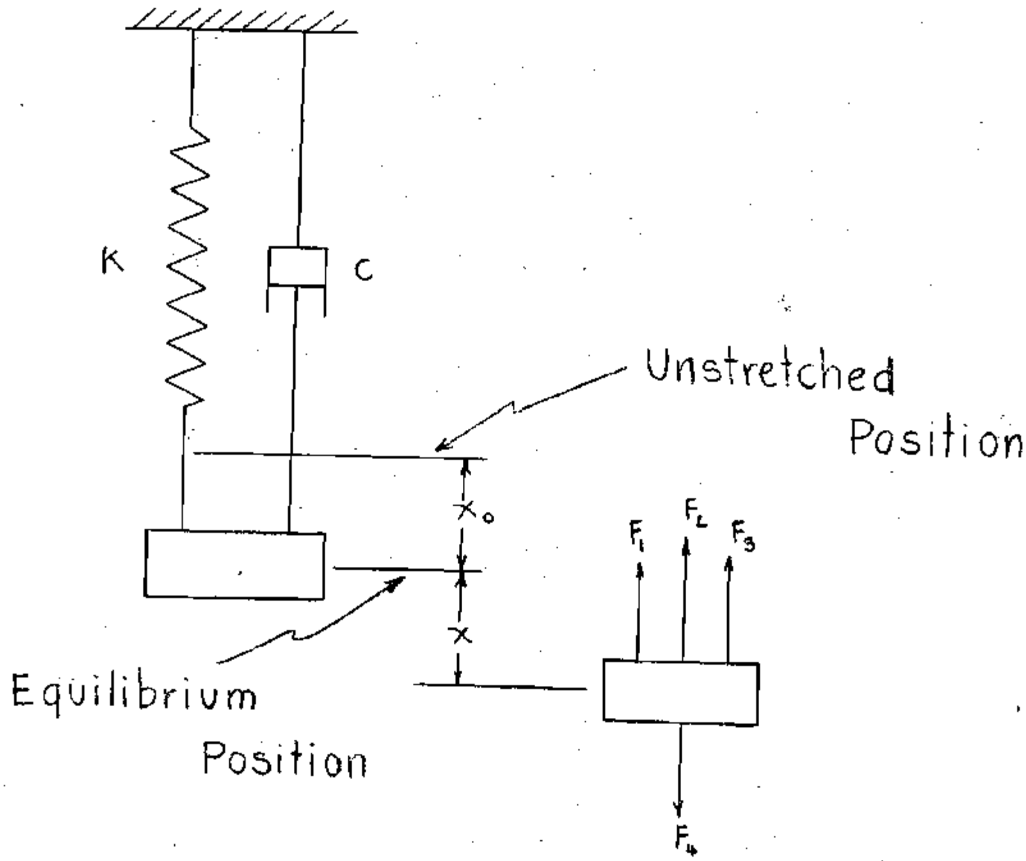
It is believed that atomization resulting from vibration phenomenon will have better characteristics especially in spray fineness, uniformity and capability of evaporization. Improvement of spray cone angle can be made mostly by increase of valve angle.

It is also recommended that the vibrating type of atomizing nozzle is feasible to construct for operation from low pressure to high pressure system. Complication occurs only at low pressure where simplification may be made by using some mechanism to produce pressure variation of the flow in the nozzle, the system will change from self-excitation, which is rather complicated, to forced vibration of a single degree of freedom by a known controlling force, such as by putting an oscillator-valve between the pipe line and the nozzle.

The vibrating type nozzle can be advantageous to suit any application that involves spray formation such as spray dryers and petrol injection carburetors (10) which operate at low pressure. When using in the petrol injection carburetor, the outstanding feature is a remarkable homogeneity of the mixture, besides the other advantages of the injection carburetor, which make it superior to the ordinary carburetor.

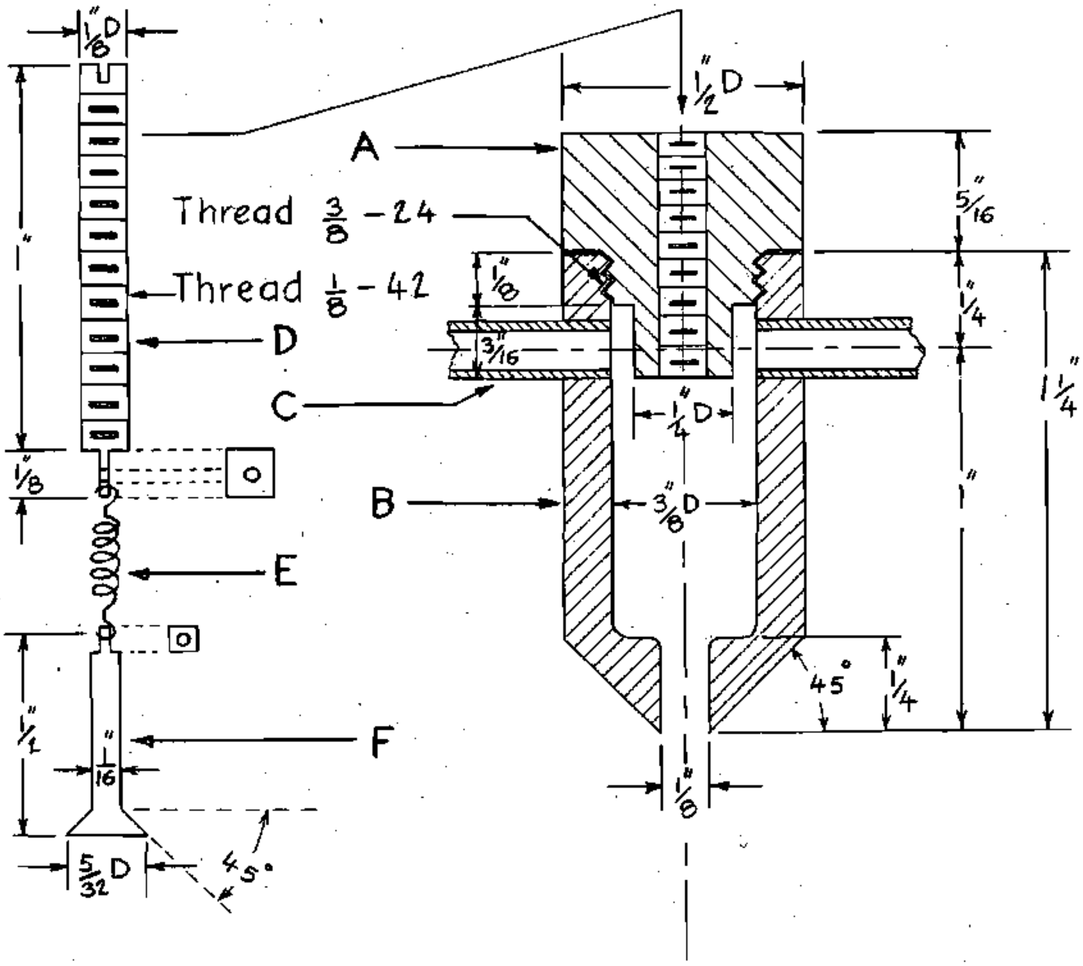
APPENDIX A

ILLUSTRATIONS



Force Vibration with Viscous Damping

FIGURE 1



A Bushing

B Nozzle

C 1/8 inch Standard Copper Tube

D Adjustable Screw

E Coil Spring

F Poppet Valve

FIGURE 2

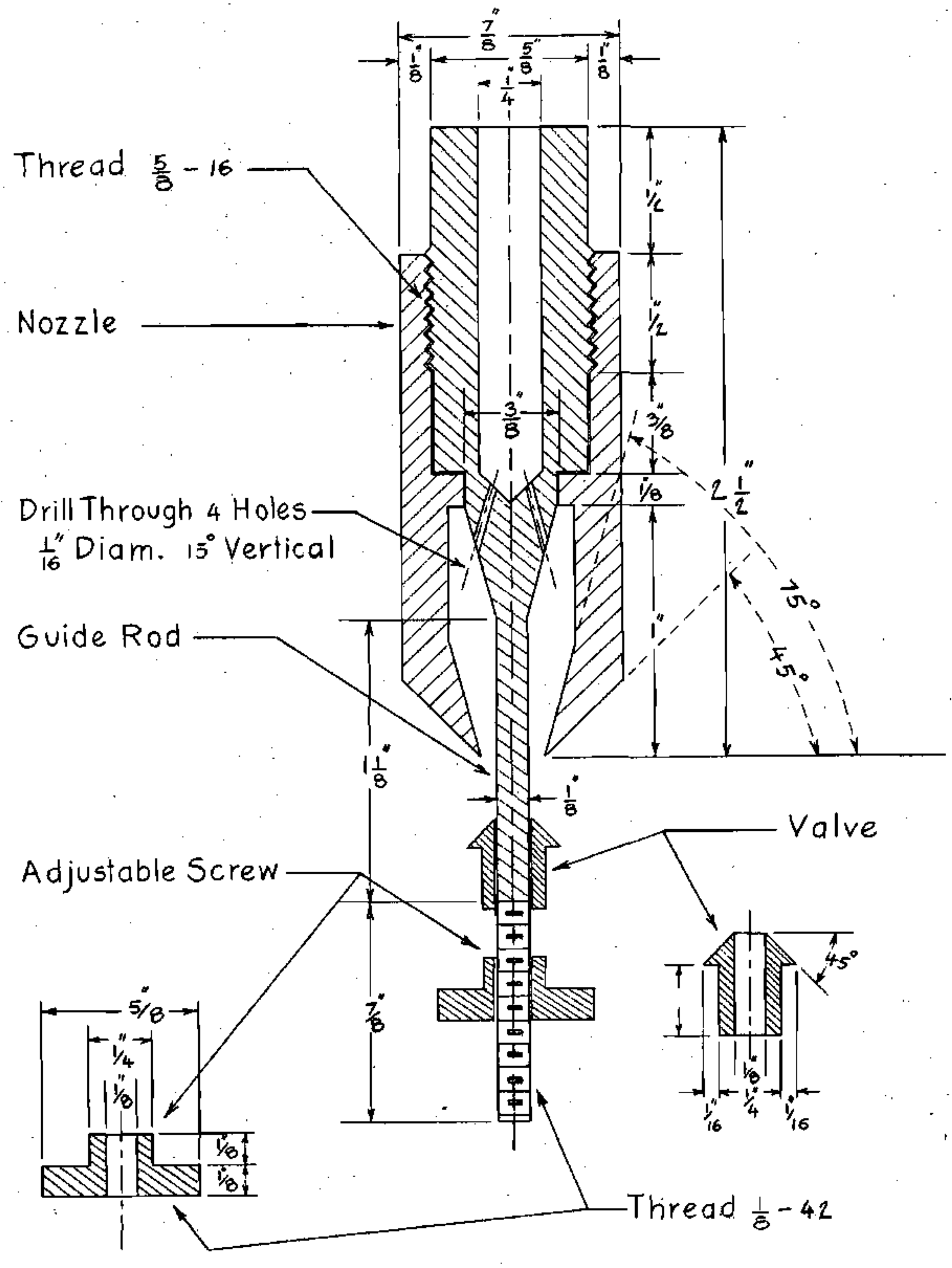
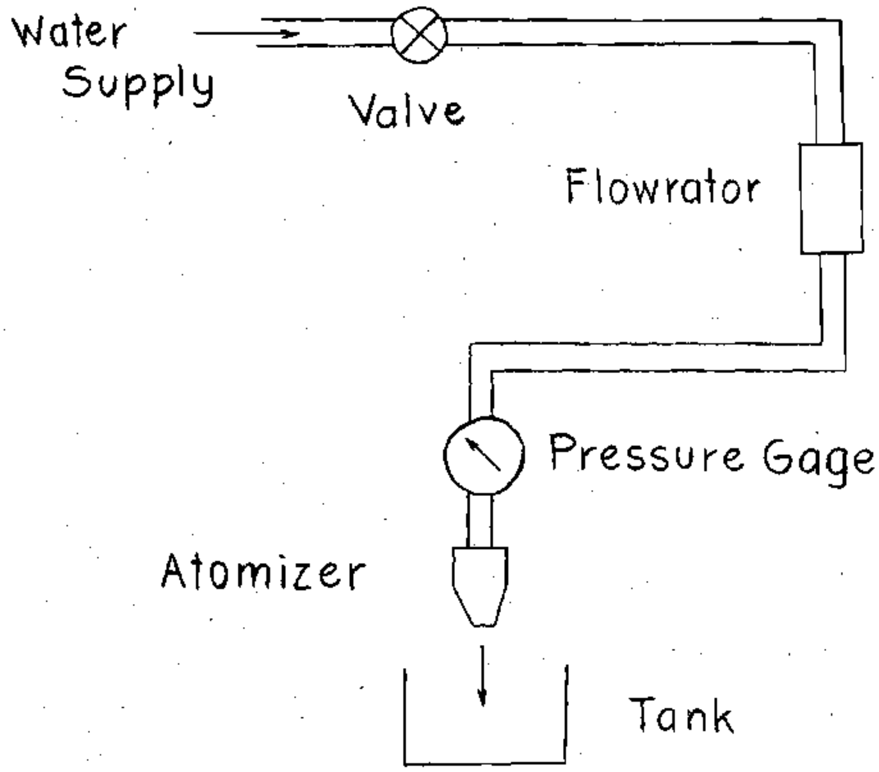
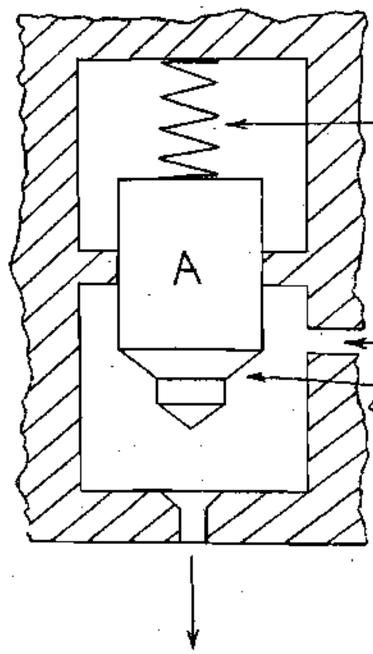

Thread 5/8 - 16
Nozzle
Drill Through 4 Holes
1/16" Diam. 15° Vertical
Guide Rod
Adjustable Screw
Valve
Thread 1/8 - 42
7/8"
5/8"
1/8"
1/4"
3/8"
1/2"
1 3/8"
2 1/2"
1"
1 1/8"
75°
45°
1/16"

FIGURE 3



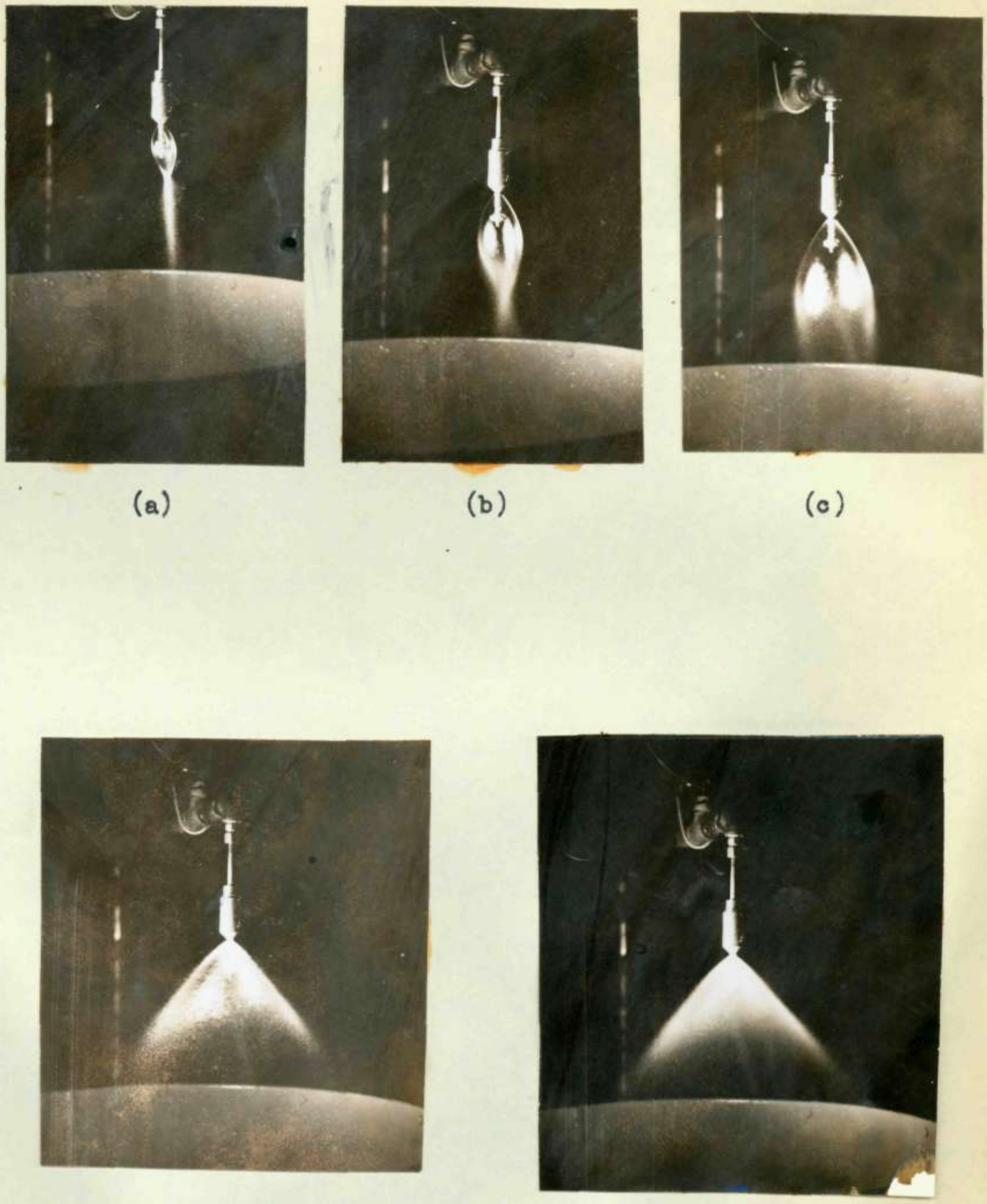
Schematic Diagram of the Connections

FIGURE 4



Diesel Engine Fuel-Injection Valve

FIGURE 5



(a) (b) (c)

(d) (e)

Figure 6. Spray Development

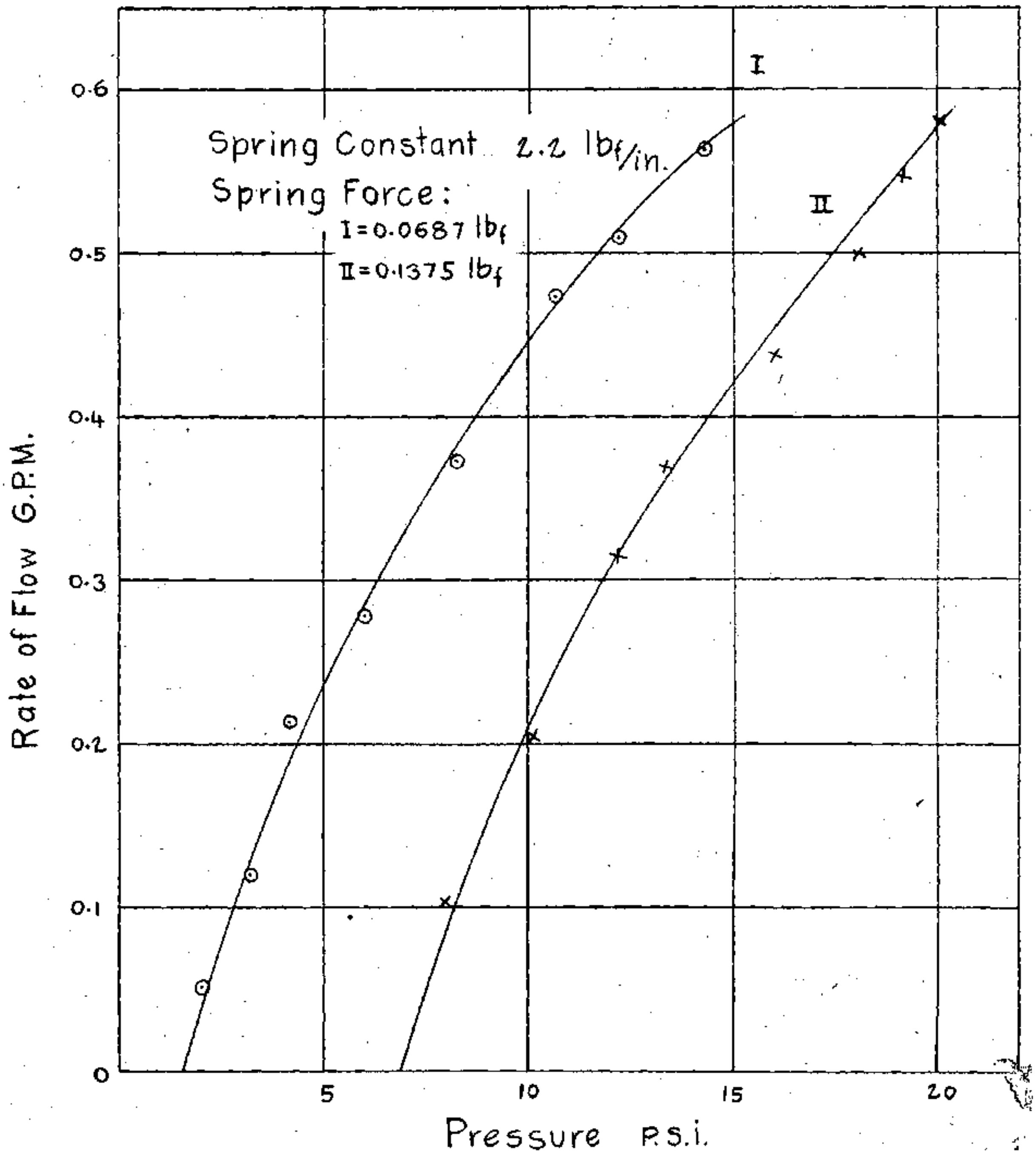

Spring Constant 2.2 lbf/in.
Spring Force:
I=0.0687 lbf
II=0.1375 lbf
I
II
0.6
0.5
0.4
0.3
0.2
0.1
0
5
10
15
20
Rate of Flow G.P.M.
Pressure P.S.I.

FIGURE 7

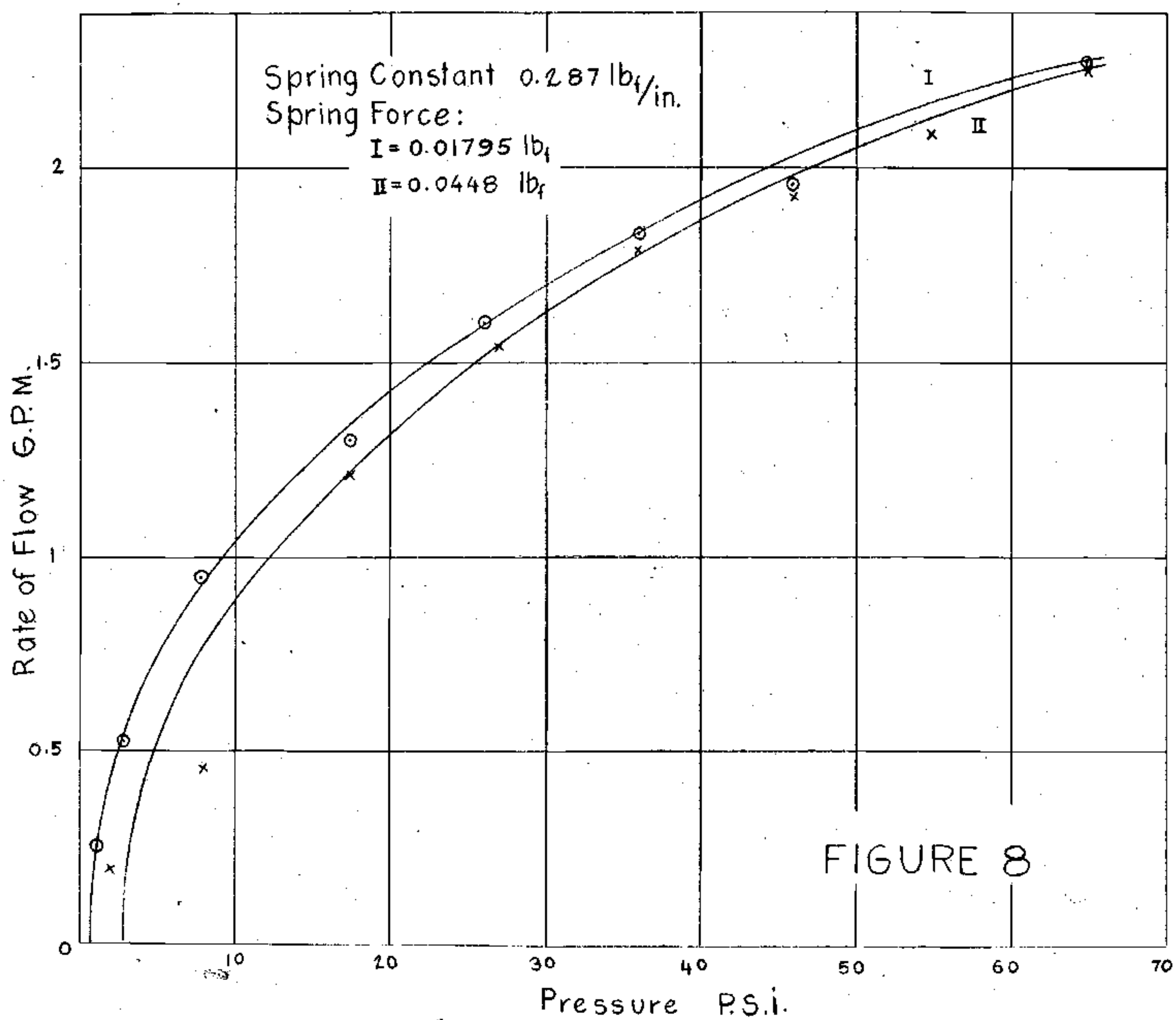

Spring Constant 0.287 lbf/in.
Spring Force:
I = 0.01795 lbf
II = 0.0448 lbf
I
II
Rate of Flow G.P.M.
Pressure P.S.i.
0
0.5
1
1.5
2
10
20
30
40
50
60
70

FIGURE 8

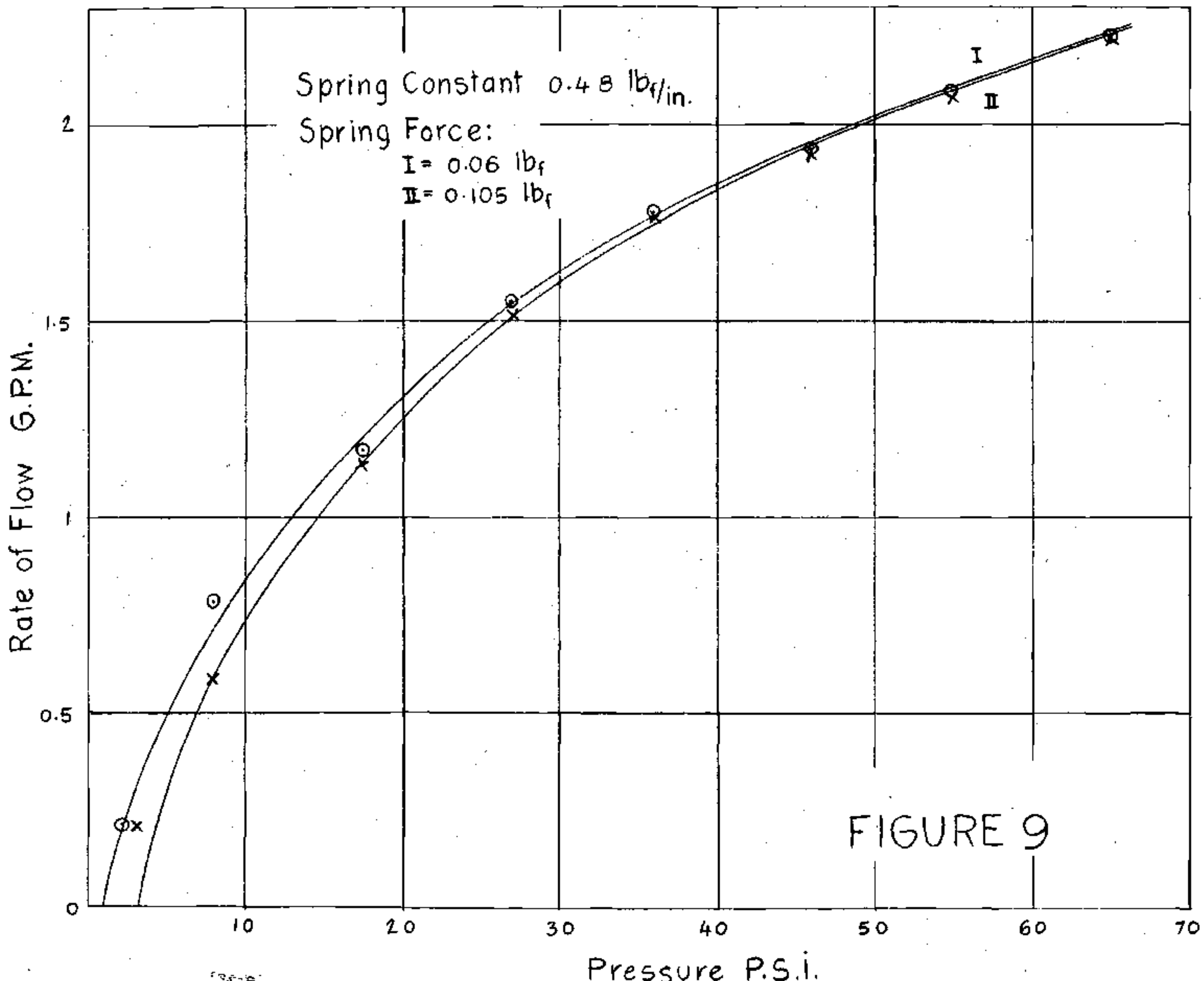
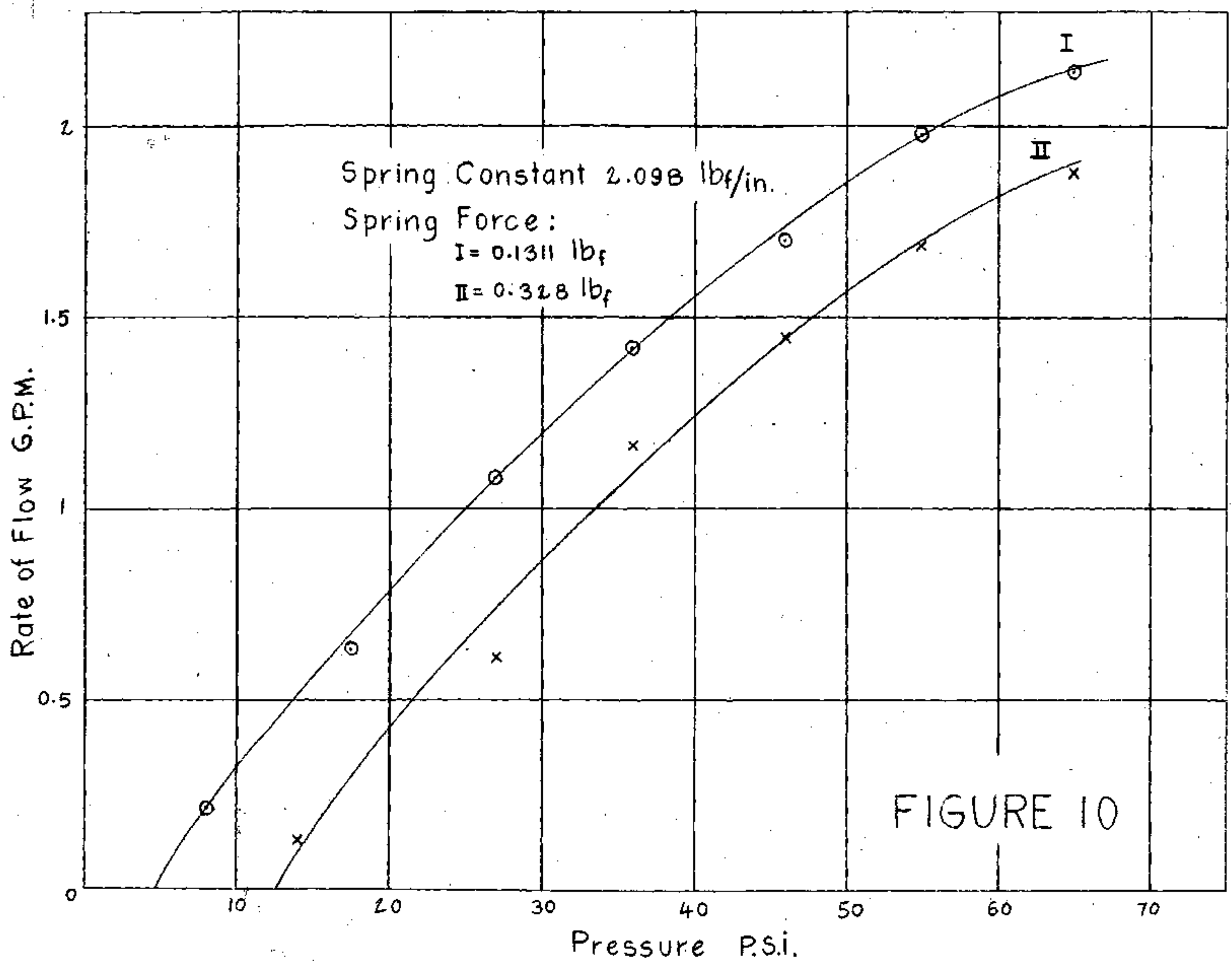

Spring Constant 0.48 lbf/in.
Spring Force:
I = 0.06 lbf
II = 0.105 lbf
I
II
Rate of Flow G.P.M.
Pressure P.S.I.
0
0.5
1
1.5
2
10
20
30
40
50
60
70

FIGURE 9



Spring Constant 2.098 lbf/in.
Spring Force:
I = 0.1311 lbf
II = 0.328 lbf
I
II
Rate of Flow G.P.M.
Pressure P.S.i.
0
0.5
1
1.5
2
10
20
30
40
50
60
70

FIGURE 10

APPENDIX B

TABLES

Table 1. Experimental Data for Atomizer A

Spring Constant 2.2 lb_f/in.

Spring Force 0.0687 lb_f and 0.1375 lb_f

Test No.	Pressure Gage Reading psi.	Correct Pressure psi.	Water Flow mm.	Water Flow G.P.M.	Vibration Cycle/min.
1	3.8	2	19	0.052	
2	5	3.2	50	0.12	
3	6	4.2	89	0.213	
4	8	6	117	0.278	
5	10.2	8.2	157	0.374	
6	12.5	10.6	200	0.475	
7	14.5	12.2	219	0.52	
8	16.5	14.3	236	0.563	Unjustify
1	10	8	43	0.104	
2	12	10.1	86	0.206	
3	14.5	12.2	132	0.315	
4	15.2	13.3	157	0.374	
5	18.5	16	184	0.437	
6	20.5	18	210	0.5	
7	22	19.1	230	0.546	
8	23	20	243	0.58	

Table 2. Experimental Data for Atomizer B

Spring Constant 0.287 lb_f/in.

Spring Force 0.01795 lb_f and 0.0448 lb_f

Run No.	Pressure Gage Reading psi.	Correct Pressure psi.	Water Flow lb.	Time min.	Water Flow G.P.M.
1	3	1.2	8.25	4	0.249
2	5	3	16.75	4	0.505
3	10	8	28.25	4	0.852
4	20	17.5	43.5	4	1.3
5	29	26	53	4	1.6
6	40	36	59.75	4	1.83
7	50	46	64.75	4	1.955
8	60	55	69.75	4	2.103
9	70	65	75.25	4	2.27
1	4	2.2	6.5	4	0.196
2	10	8	15	4	0.453
3	20	17.5	40.25	4	1.215
4	30	27	51	4	1.54
5	40	36	59.5	4	1.797
6	50	46	64	4	1.93
7	60	55	69	4	2.08
8	70	65	75	4	2.26

Table 3. Experimental Data for Atomizer B

Spring Constant 0.48 lb_f/in.

Spring Force 0.06 lb_f and 0.105 lb_f

Run No.	Pressure Gage Reading psi.	Correct Pressure psi.	Water Flow lb.	Time min.	Water Flow G.P.M.
1	4	2.2	7	4	0.2116
2	10	8	26.25	4	0.792
3	20	17.5	39	4	1.178
4	30	27	51.5	4	1.555
5	40	36	59	4	1.78
6	50	46	64	4	1.93
7	60	55	69	4	2.08
8	70	65	73.5	4	2.22
1	5	3.2	7	4	0.2116
2	10	8	19.5	4	0.588
3	20	17.5	37.75	4	1.14
4	30	27	50	4	1.52
5	40	36	58.75	4	1.773
6	50	46	64	4	1.93
7	60	55	68.5	4	2.068
8	70	65	73	4	2.21

Table 4. Experimental Data for Atomizer B

Spring Constant 2.098 lb_f/in.

Spring Force 0.1311 lb_f and 0.328 lb_f

Run No.	Pressure Gage Reading psi.	Correct Pressure psi.	Water Flow lb.	Time min.	Water Flow G.P.M.
1	10	8	7	4	0.2116
2	20	17.5	20.75	4	0.625
3	30	27	35.75	4	1.08
4	40	36	47	4	1.42
5	50	46	56.25	4	1.7
6	60	55	65.5	4	1.98
7	70	65	71	4	2.14
1	16.5	14	4.25	4	0.1283
2	30	27	20.5	4	0.61
3	40	36	39	4	1.176
4	50	46	48	4	1.45
5	60	55	56	4	1.69
6	70	65	62.25	4	1.88

APPENDIX C

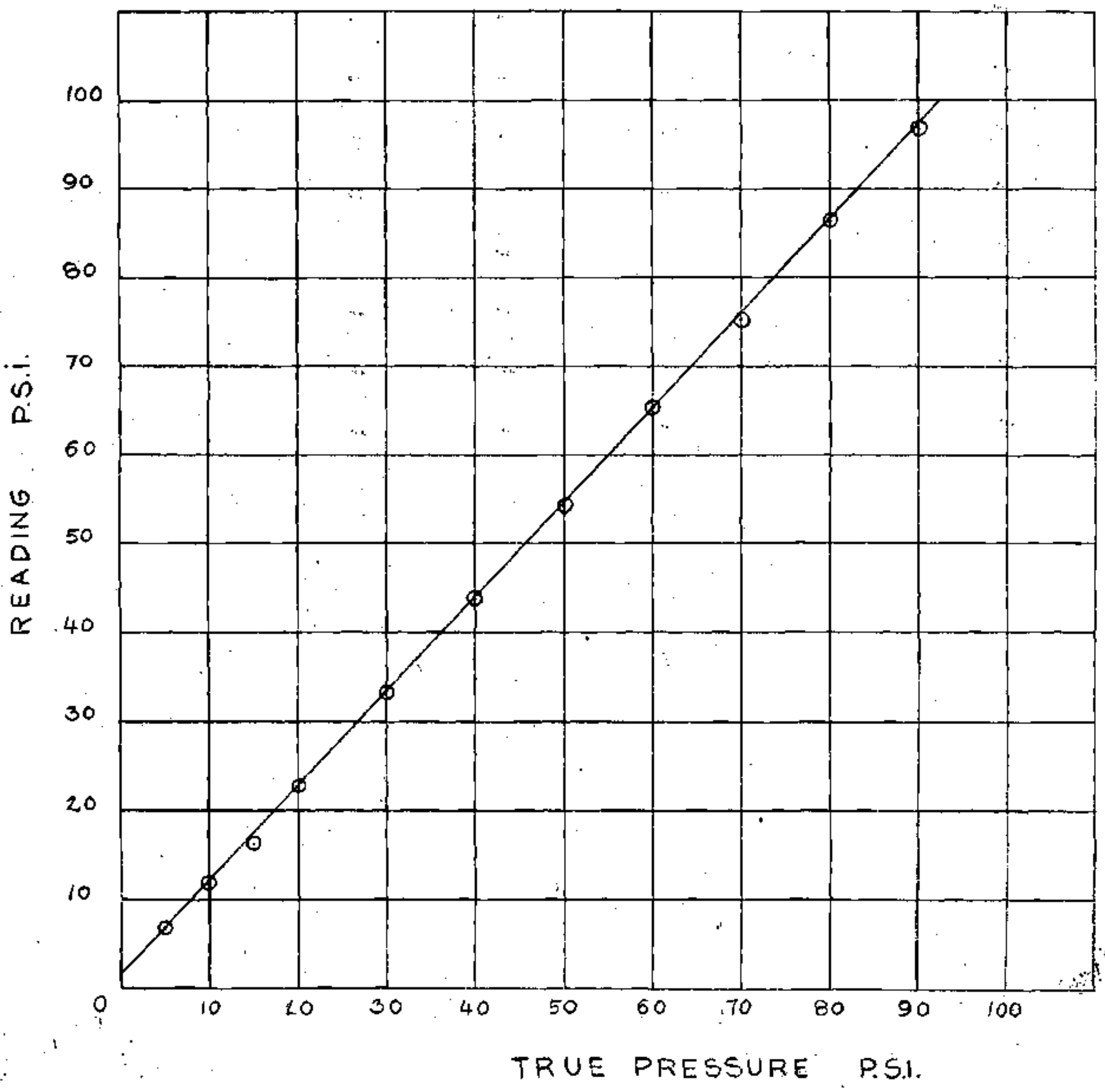
GAGE CORRECTION AND FLOWRATOR CALIBRATION

PRESSURE GAGE CORRECTION

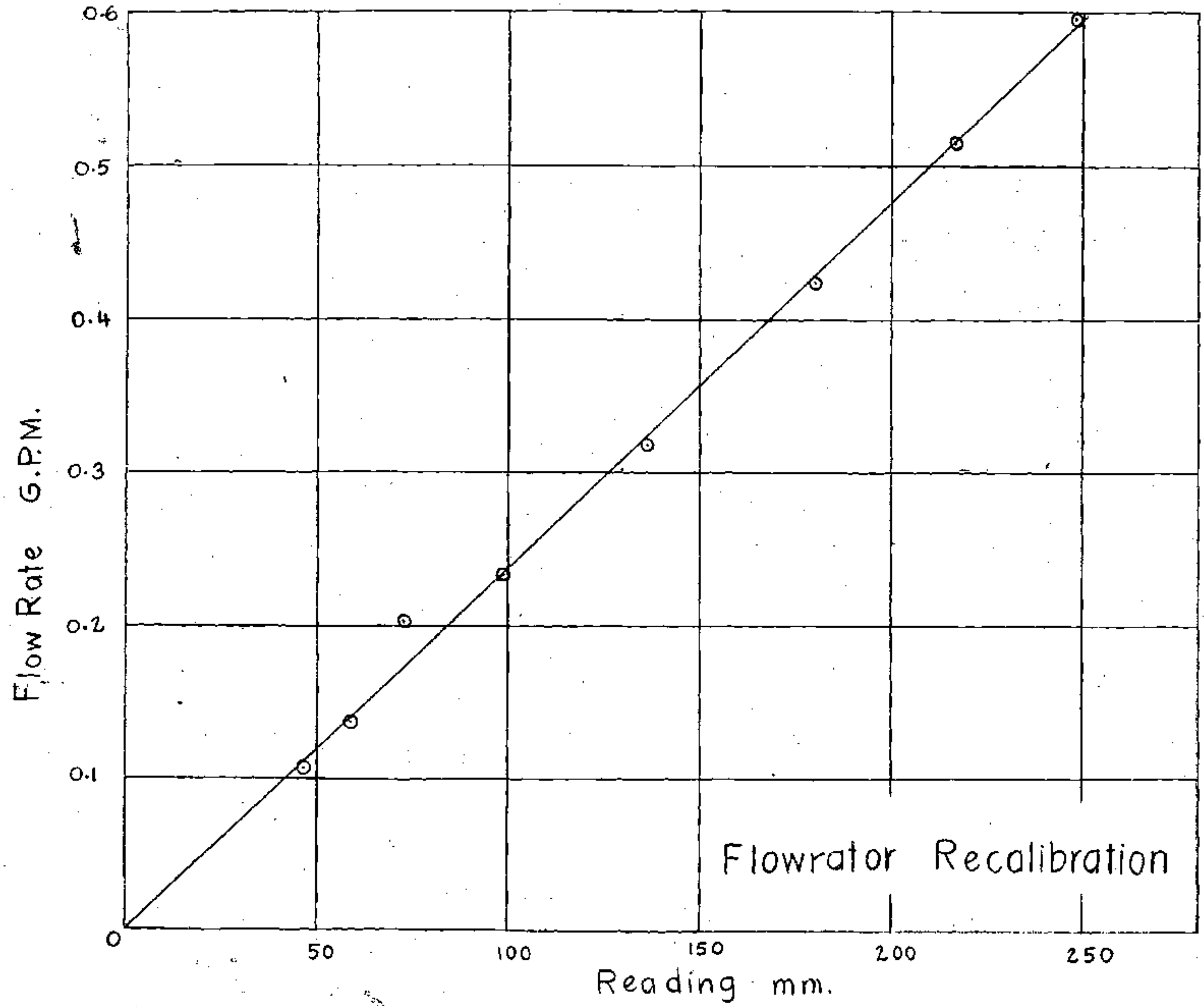
Increasing Weight lb.			Decreasing Weight ib.			
Test No.	Gage Reading psi.	True Pressure psi.	Test No.	Gage Reading psi.	True Pressure psi.	Average Gage Reading psi.
1	7	5	22	7	5	7
2	12	10	21	12	10	12
3	16	15	20	17	15	16.5
4	22	20	19	24	20	23
5	33	30	18	34	30	33.5
6	44	40	17	44	40	44
7	54	50	16	55	50	54.5
8	65	60	15	66	60	65.5
9	75	70	14	76	70	75.5
10	86	80	13	87	80	86.5
11	97	90	12	97	90	97

FLOWRATOR RECALIBRATION

Test No.	Time min.	Weight of Water lb. Before	After	Net Weight	Reading mm.	Water Flow G.P.M.
1	4	108.25	111.75	3.5	46	0.1058
2	4	108.75	113.25	4.5	59	0.136
3	4	108.75	115.5	6.75	72	0.204
4	4	109.25	117	7.75	98	0.234
5	4	110	120.5	10.5	136	0.3175
6	4	110.5	124.5	14	180	0.423
7	4	110.75	127.75	17	217	0.514
8	4	111.5	131.25	19.75	248	0.596



Pressure Gage Calibration



0.6
0.5
0.4
0.3
0.2
0.1
0
50
100
150
200
250
Flow Rate G.P.M.
Reading mm.
Flowrator Recalibration

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