FUNDAMENTAL DESIGN PRINCIPLES OF PRESSURE REGULATING VALVES

by

HENRY LEE BURNS

A THESIS

submitted to

OREGON STATE COLLEGE

in partial fulfillment of
the requirements for the
degree of

MECHANICAL ENGINEER

June 1947

APPROVED:

[redacted]

Head of Department of Mechanical Engineering

[redacted]

Chairman of School Graduate Committee

[redacted]

Associate Dean, Graduate Division

ACKNOWLEDGMENT

The author is especially indebted to the following established manufacturers of pressure regulating valves who supplied descriptive and performance information in the form of advertising literature.

Air Reduction Sales Company, New York, N.Y.

Alar Products Inc., Cleveland, Ohio

American Chain & Cable Co., d'Este Div., Bridgeport, Conn.

The Bastian-Blessing Company, Chicago, Ill.

The Black Manufacturing Co., Baltimore, Md.

Dockson Corporation, Detroit, Mich.

The Foxboro Company, Foxboro, Mass.

General Controls Co., Glendale, Calif.

The Harris Calorific Co., Cleveland, Ohio

The Linde Air Products Company, New York, N.Y.

National Bureau of Standards, Washington, D.C.

Pittsburg Equitable Meter Div. of Rockwell Manufacturing Co., Pittsburg, Pa.

Reliance Regulator Corporation, Alhambra, Calif.

Republic Flow Meters Co., Chicago, Ill.

Spence Engineering Company Inc., Walden, N.Y.

CONTENTS

FUNDAMENTAL DESIGN PRINCIPLES OF PRESSURE REGULATING VALVES

INTRODUCTION

During the past several years the author of this thesis has had numerous occasions to design, purchase, or recommend for purchase pressure regulating valves of various types and uses. In the course of action on these problems it was soon discovered that there was no published information as to the fundamental engineering principles of pressure regulating valve design. Although numerous companies manufacture properly functioning regulators, the design procedure is kept within the organization, and even there it is frequently hand-me-down, rule-of-thumb, or trial and error. If an engineer is purchasing, designing, or using a piece of equipment, it is to his best advantage to understand its operating principles, range of use, and limitations. It would seem then, that a ready reference from which information could be obtained regarding basic pressure regulating valve design would have a place in available engineering literature. It is the purpose of this thesis to make a start in that direction. This thesis is an attempt to crystallize the author's experience in this field, and discussions contained herein are from a more practical standpoint than theoretical.

Pressure regulating valves are referred to as reducing valves, as gas governors, and frequently as just "regulators". The definition of a pressure regulating valve as adhered to in this thesis is that it is an automatic valve which will tend to maintain a predetermined fluid pressure at the valve outlet regardless of fluid pressure at the valve inlet and regardless of rate of fluid flow through the valve. The degree with which a pressure regulating valve maintains a fixed pressure level in a system is a function of the regulator design. There are several other methods of regulating pressures such as by controlling motor and pump speeds, or operating dampers, or actuating switches, etc., but discussions in this thesis are limited to those pressure regulators incorporating a valve means within their structure. For the sake of brevity throughout the following discussions, a pressure regulating valve will be simply referred to as a "regulator".

Regulators are frequently used and are vital to the following equipment; fuel gas transmission and control, gas welding, hydraulic power and control, vacuum systems, refrigerant control, gas therapy, steam regulation, and numerous other operations where an automatic control of gases and liquids is indicated. Although each specific application calls for a different regulator design, the same fundamental engineering principles will be found to apply to them all. In the course of the following discussion, the functions, mechanical description, performance characteristics, design principles, and performance tests of regulators will be considered.

FUNCTIONAL DESCRIPTION

The expressed purpose of a pressure regulating valve is to maintain a predetermined pressure level at its outlet by means of the throttling or modulating action of a valve. Use or need of a regulator precludes a condition of either non-uniform inlet pressure or irregular outlet flows, or both. Since no additional energy is imparted into the system by the regulator it may also be stated that there is always a condition of inlet pressure being reduced and regulated to a lower pressure level. In a fluid system where the available source of fluid pressure under varying conditions is to be used at uniform pressure under varying flow conditions, there will be numerous forces tending to prevent uniformity of pressure. Variations in inlet pressure will vary the force which a valve must work against and will change the flow through a fixed valve opening. Variations in outlet flow will cause changing pressure drops through valve openings. Variations in outlet pressure will also affect valve forces and flow through the valve. It is the mechanical purpose of a regulator to amplify forces produced by the condition to be controlled and to minimize all other forces. This is done in the usual manner of giving a high leverage to the condition to be controlled. With pressure, the controlled condition, the basic leverage system used is the ratio of areas affected by one pressure or the other.

A regulator is made up of five components; a valve means, a pressure sensitive means, a pressure control means, linkages, and some form of housing. Figure one shows the relative location of these parts. The valve functions as a variable orifice between inlet conditions and controlled outlet conditions. Under conditions

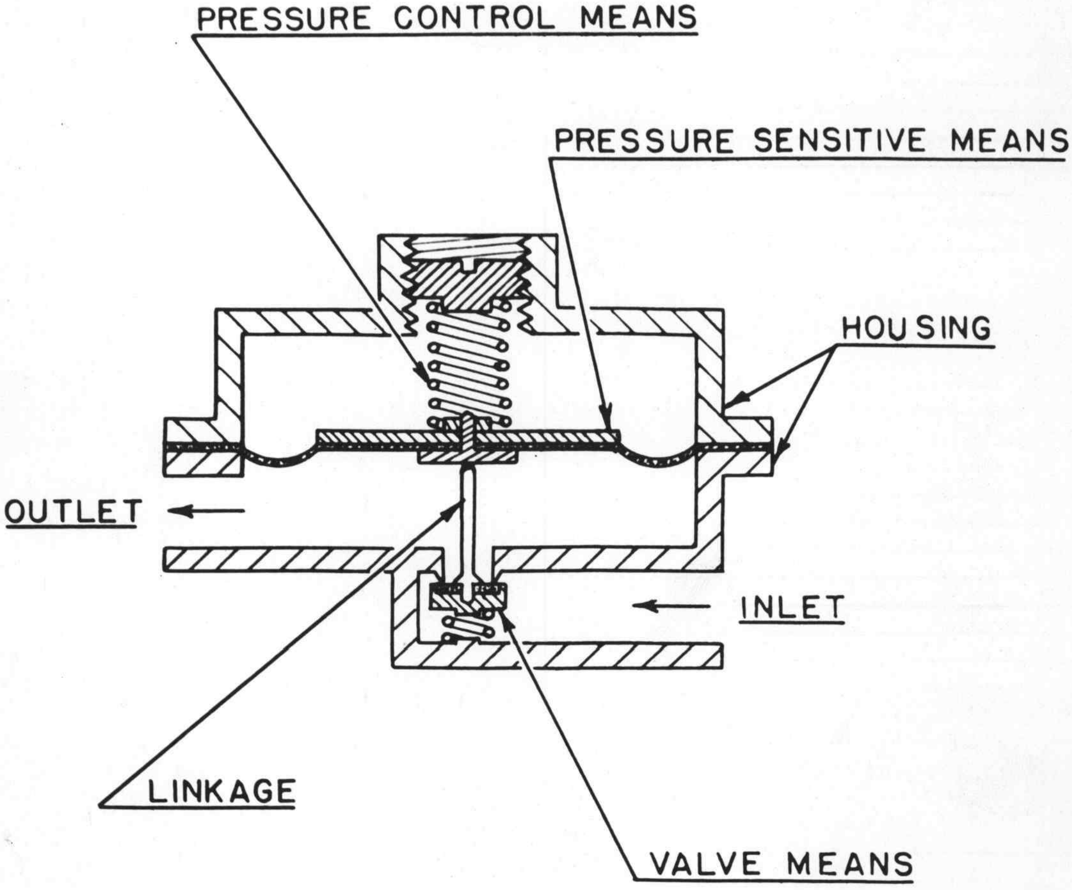


FIGURE 1.

BASIC COMPONENTS OF A PRESSURE REGULATING VALVE of no flow, the valve must close so that outlet pressures will not rise to the higher inlet pressure level; and under flow conditions where fluid is being removed from the regulator outlet it must open a sufficient amount to allow supply fluid at the valve inlet to make up the deficit. Forces acting on the valve will be a function of pressure differential across it, and its exposed area.

The pressure sensitive means is an element which will respond in a mechanical manner to slight changes in pressure. This pressure sensitive means is located in such a position that it will be exposed to the pressure which is to be controlled, and the forces acting on it will be a function of the controlled pressure and area. The area of this pressure sensitive means is made large with respect to the valve area, in order to have a corresponding force advantage over the valve, which is acted on by forces not to be controlled.

In order to fix the pressure level at which the pressure sensitive means will respond, some method of counterbalance or pressure control is required. The pressure control means will oppose the controlled pressure through the medium of aforesaid pressure sensitive means. The direction of forces acting on the pressure sensitive means will control the direction in which it will move.

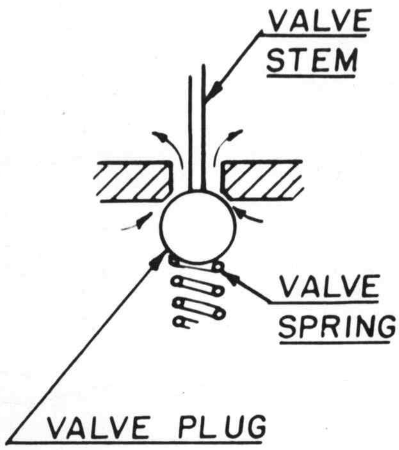
The pressure sensitive means is connected to the valve through some suitable linkage. Therefore any movement of the pressure sensitive element due to unbalancing of the controlled pressure opposed by the pressure control means, will cause a corresponding movement of the valve. The valve will continue to move in one direction or another, allowing more or less fluid to pass, until forces on the pressure sensitive means are again balanced, thereby tending to regulate the pressure. Linkages between elements include various levers, yokes, pivots, and guides that might be necessary.

In order to make the necessary connections, separations, and support, a housing is provided. This housing may incorporate other functions as will be brought out in subsequent discussions.

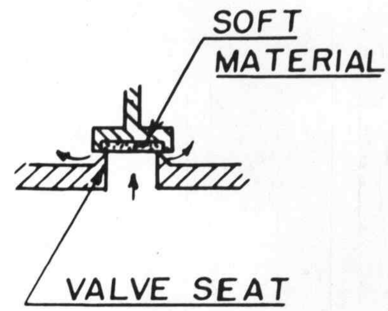
MECHANICAL DESCRIPTION

Valve assembly. With a device which has so many uses and manufacturers, one is bound to find numerous mechanical arrangements for accomplishing the desired purposes. Types of valves are best identified by their relation to the pressure forces acting upon them. A valve is usually composed of two elements--an orifice and a plug. The orifice segment is termed the valve seat and the plug segment is commonly called the valve or plunger. One or the other of these elements must move to give the desired throttling action; the moving element is almost always the valve, although there is no functional reason why the valve seat cannot be the moving element.

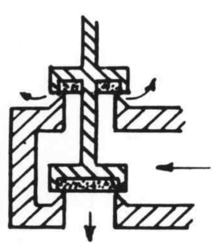
Since the major pressure drop occurs across the valve assembly, there are bound to be forces on the valve tending to seat or unseat it, depending on the position of the plug in relation to the orifice and direction of pressure differential. If the valve plug is on the high pressure side of the seat it is said to close with pressure, and if it is positioned on the low pressure side it is said to close against pressure. A simple round orifice seat and corresponding valve plug is the most common, and both types of valve closure are frequently used. Another common feature of this basic type of valve assembly is the feature of having one or the other of the elements fabricated in a soft or non-metallic material. There are designs however, which have a so called metal to metal seat. Exact shapes of the elements for the above type of valve are varied in appearance and combinations. The valve plug may be spherical, tapered, or flat, and the seat element may be ridged, countersunk, or square edged.



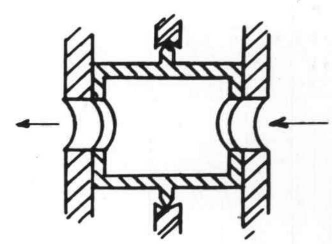
CLOSING WITH PRESSURE



CLOSING AGAINST PRESSURE



BALANCED VALVE



ROTATING VALVE

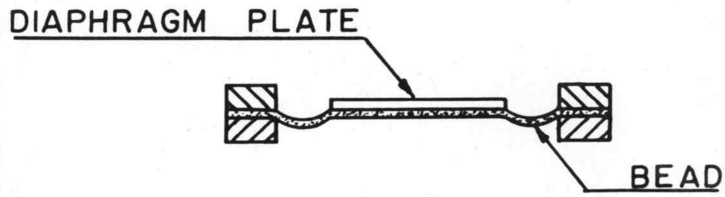
FIGURE 2.

BASIC VALVE TYPES

There are several special valve designs which are arranged in such a manner to eliminate or counter-balance directional effects of aforesaid pressure differential forces. Figure two shows some of the basic valve types used in regulators. Two simple plug and orifice valves tied together in such a way that one seats with pressure and the other against pressure, form what is commonly known as a balanced valve. Design variations in balanced valves are found in the manufacturing measures used for location of the two seats and tying the two valve plugs together. Manufacturing practice sometimes uses variations of sliding pistons, sleeve valves, or rotating plug valves. Naturally a valve with the valve orifice opened and closed by sliding actions, will be found in a great variety of appearances, although the basic functions will be the same.

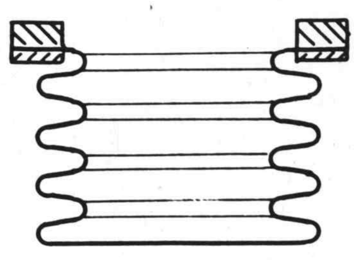
Pressure sensitive means. The element of a regulator which is to respond most to changes in pressure will naturally be the largest single element in a regulator. Numerous methods are used to provide large pressure sensitive areas within the regulator confines and to meet mechanical and performance requirements. Since these elements respond to pressures, a condition common to all of them is the existance of a pressure differential with the pressure sensitive means as the barrier. The most common form of pressure sensitive means is that of a flat sheet or diaphragm. Designs of diaphragms will vary depending on the material, degree of movement required, control pressures, etc. If a diaphragm is to have a large movement or is constructed of a non-elastic material, the usual practice is to incorporate a bead or roll in its construction which will allow the necessary movement. Simple flat diaphragms are most frequently

SIMPLE DIAPHRAGM



BEADED DIAPHRAGM

WAFER DIAPHRAGM (METAL)



BELLOWS (METAL)

NOTE:

ALL ELEMENTS SHOWN HAVE 1 SQ.IN. EFFECTIVE AREA

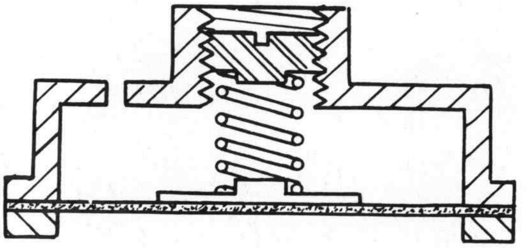
FIGURE 3.

BASIC TYPES OF PRESSURE SENSITIVE MEANS

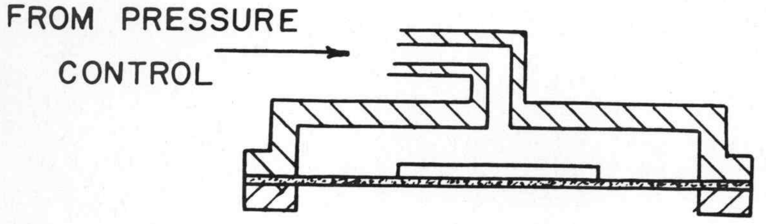
used when the required flexing can be provided by the elasticity of diaphragm material. When the movement or "stroke" is very small, flat metallic diaphragms are used with success. A corrugated or beaded metallic diaphragm is commonly called a wafer diaphragm. For reasons which will be brought out in later discussions, the center portion of diaphragms are made rigid by an added plate or a non-flexing center section.

Pressure sensitive elements are not limited to the diaphragm shape. Conditions may arise where the motion of a piston in a cylinder could best produce the desired action. Sometimes the use of a flexible bag is utilized to provide an exceptionally large sensitive area in a small space. When pressures under control are referenced to absolute pressure an evacuated capsule is used as the sensitive element. The most popular form of metallic pressure sensitive means is that of a bellows, which has considerable flexibility yet retaining the strength of metal. The most common types of diaphragms are shown in figure three.

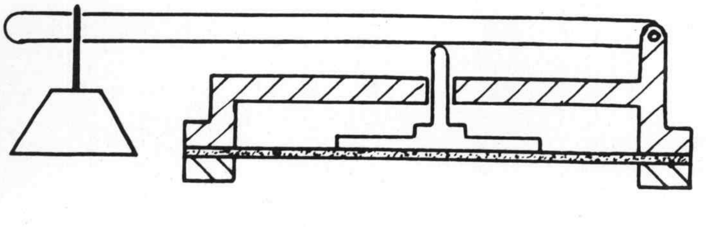
Load control means. The utility of a pressure regulator will be a function of the accuracy with which the user can adjust or predict the pressure level at which the unit will regulate. By far the most popular pressure control means is the common helical spring. Various methods of mounting helical springs are used but their installation can be classified as either fixed or adjustable. Adjustment is made by some form of adjusting screw acting to compress or decompress the spring. Fixed adjustment would naturally be of use in those regulator applications where pressure levels are predetermined. In the case of soft flexible diaphragms the rigid



SPRING LOADED



FLUID LOADING



DEAD WEIGHT LOADING

FIGURE 4.

TYPES OF LOAD CONTROL MEANS

diaphragm plate serves to distribute the high unit pressure that would result from the small end area of a spring. As will be discussed later it is desirable to have low spring rates and this spring property tends to require more space. Numerous variations are used to get a combination of minimum spring rate and minimum space. Springs within springs, torsional springs, various forms of leaf springs, etc., are all used by regulator designers to accomplish this and other special purposes. Another important means of pressure control is the use of a fluid pressure to counterbalance the fluid under regulation. Obviously, with a diaphragm like element between the fluid under regulation and pressure control means, the actual form of the control means has no effect on the regulated fluid other than that of a reacting force. A fluid type of load control means may utilize the same fluid that is being regulated. The usual procedure in this case is to bleed off some of the gas or liquid being controlled and allow it to pressurize a cavity on the opposite side of the pressure sensitive means, and controlling its pressure by a valve which might even be another regulator of small capacity. Fluid from another source may be used conveniently, especially in a system where several regulated pressures are to be controlled together.

In cases such as the metallic bellows, where the pressure sensitive means has considerable spring rate of its own, additional springs are sometimes unnecessary. When vacuum is to be controlled, an evacuated capsule incorporating its own spring rate plus that of a built-in spring is used. Examples of the various load control systems are shown in figure four.

A dead weight is frequently used as the pressure control means. Weights are either applied directly to the pressure sensitive means or indirectly through a leverage system. By means of variable lever ratios or weight position adjustment of pressure level is obtained with the dead weight system.

Linkage. Included in the category of linkages are the various leverages, connections, and guides necessary to integrate functions of each component of a regulator. Linkage between pressure sensitive means and valve means may be either a direct or an indirect connection through the medium of a lever arrangement. Levers are usually arranged to reverse the direction of valve means over that of the pressure sensitive means. Lever linkages are usually designed to give the lever advantage, and therefore the greater stroke, to the pressure sensitive means. Linkages between diaphragm and valve are usually fastened to the diaphragm by means of the rigid portion or diaphragm plate in the center of the pressure sensitive means. These linkages are not always attached to valve means, and valve return springs are utilized to cause the valve elements to follow the linkage. Since the valve element is usually free from any actual connection with other elements, valve guides are provided to keep valve elements in proper relation with valve seat elements.

Housing. The housing of a regulator is made of one or more parts which serve several purposes. Besides being the overall integrating container, the housing contains within its structure various passageways and supports. Provision for inlet and outlet connections with respective passageways is made in the housing.

Housings are usually divided into two parts separated by the pressure sensitive or diaphragm means. The pressure control means is supported, contained, or guided by the half which is also a protective cover for the diaphragm. The other half provides housing and guides for valve elements and the pivots or supports required by linkages. Sometimes the valve seat is machined directly out of the housing. Various stops and movement limits are provided within the housing structure and relative positions of regulator elements are controlled by spacings and shapes machined in the regulator body. In regulator applications where excessive back pressures are apt to occur diaphragm covers are so arranged as to prevent overextension and to provide a support against such overpressures.

A regulator housing can usually be divided into four chambers, an inlet passageway, a delivery pressure chamber, outlet passageway, and the counterbalance, or load control housing chamber. The delivery pressure chamber is that space on the active side of the pressure sensitive means, and the fluid under pressure control communicates with this chamber in order that the pressure sensitive means might respond. Frequently the outlet connection comes directly from this delivery pressure chamber to eliminate the outlet passageway. Sometimes the delivery pressure chamber and outlet chambers are so arranged that a false pressure is provided under the pressure sensitive means--the most common usage of this is to incorporate an ejector whereby outlet fluid flows create a reduction of pressure in the delivery pressure chamber.

Another common item included with the housing but having no real functional purpose on the regulator, is a relief valve built into the body in such a position that it would relieve excessive pressure caused by valve failure.

<u>Multiple Stage Regulator</u>. On regulator installations where the ultimate delivery pressure is very small in relation to the supply or inlet pressure, manufacturers have provided what is known as multistage regulators. A stage of regulation as used here is a complete pressure reducing regulator embodying all of the aforementioned regulator parts. The stages are commonly numbered in the direction of fluid flow--thus the stage where the fluid first enters under the supply conditions is the first stage. The next stage taking the reduced and somewhat regulated fluid from the first is called the second stage, and so on. Rarely, however, does a multiple stage regulator exceed two stages. Delivery pressure adjustments are usually made on the second stage with the first stage having fixed adjustment. For the purpose of compensatory performance characteristics opposite valve actions are sometimes used in the two stages.

Usually regulating stages are so arranged that all of the fluid passing through one passes through the other and both stages are sometimes built into the same body or housing. A special type of multistage regulator having a pilot valve or pilot regulator to provide a fluid counterbalance means, frequently has the pilot regulator as a separate unit and only a small portion of the total flow passes through the pilot stage.

PERFORMANCE DESCRIPTION

Standard Performance Curve. In line with the accepted engineering practice of using performance curves to show the overall characteristics of equipment, there has been established a more or less standard form of plotting regulator performance data for regulators. This so called standard performance curve is obtained by plotting flow vs. delivery pressure at conditions of uniform supplies of inlet pressure. Delivery pressure is used as the vertical ordinate and flow as the horizontal ordinate. Flow data are always referred to a common pressure and temperature condition if a compressable fluid is being regulated. The curves shown in figure five represent the standard arrangement with some typical performance curves for the various types of regulators. Of course there are other methods used to show regulator performance, but the one discussed seems to be most common and most satisfactory.

Performance of Valve Means. The performance characteristics for each individual component of a regulator may be analyzed separately and its effect on overall regulator performance can be predicted with reasonable accuracy. The regulator valve makes a major contribution to the net regulator performance. The major influencing factors in valve performance characteristics are the cross sectional area of the valve opening and the reaction forces set up by the pressure differential across the valve acting on the exposed or effective valve area. The term, effective area, is used to take into account apparent increases in valve area due to velocity effects, etc. The reaction forces are those forces transmitted to other parts of the regulator thereby influencing overall

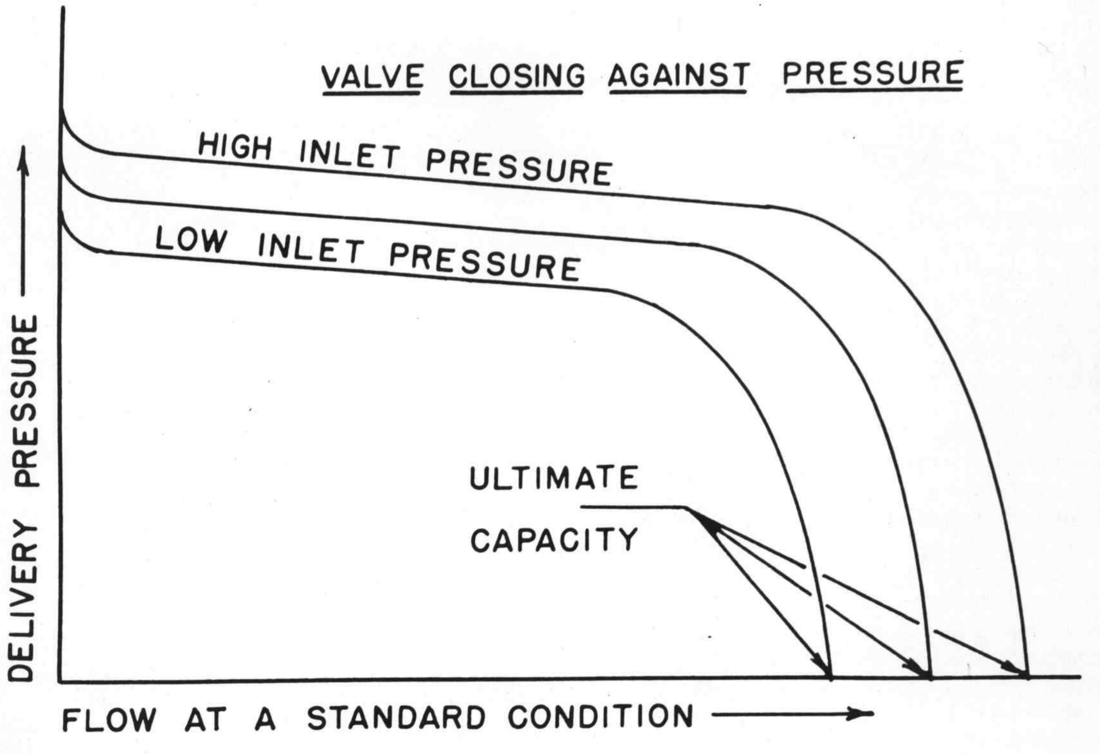


FIGURE 5(a)

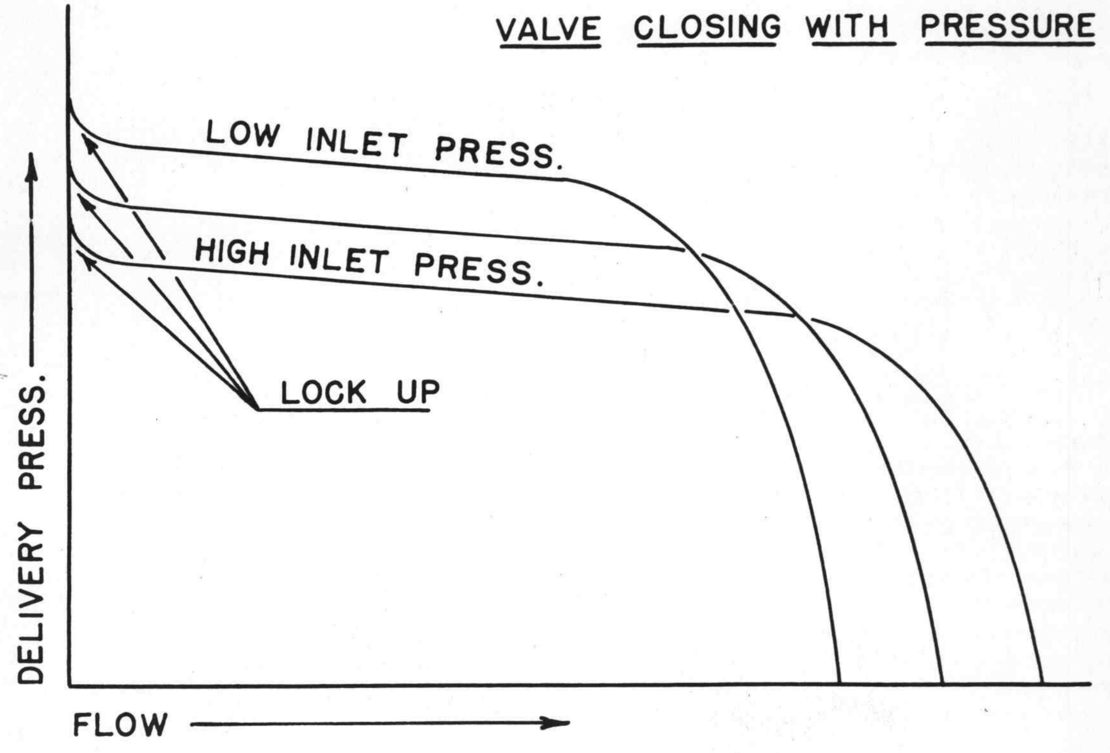


FIGURE 5(b)

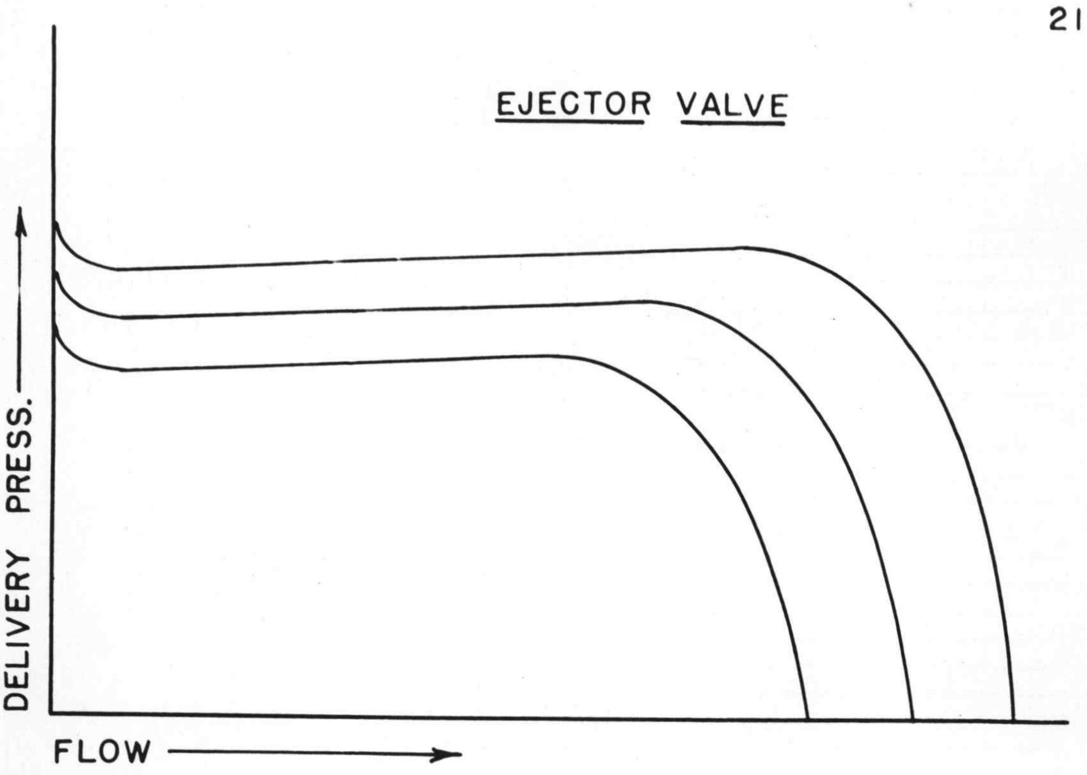


FIGURE 5(c)

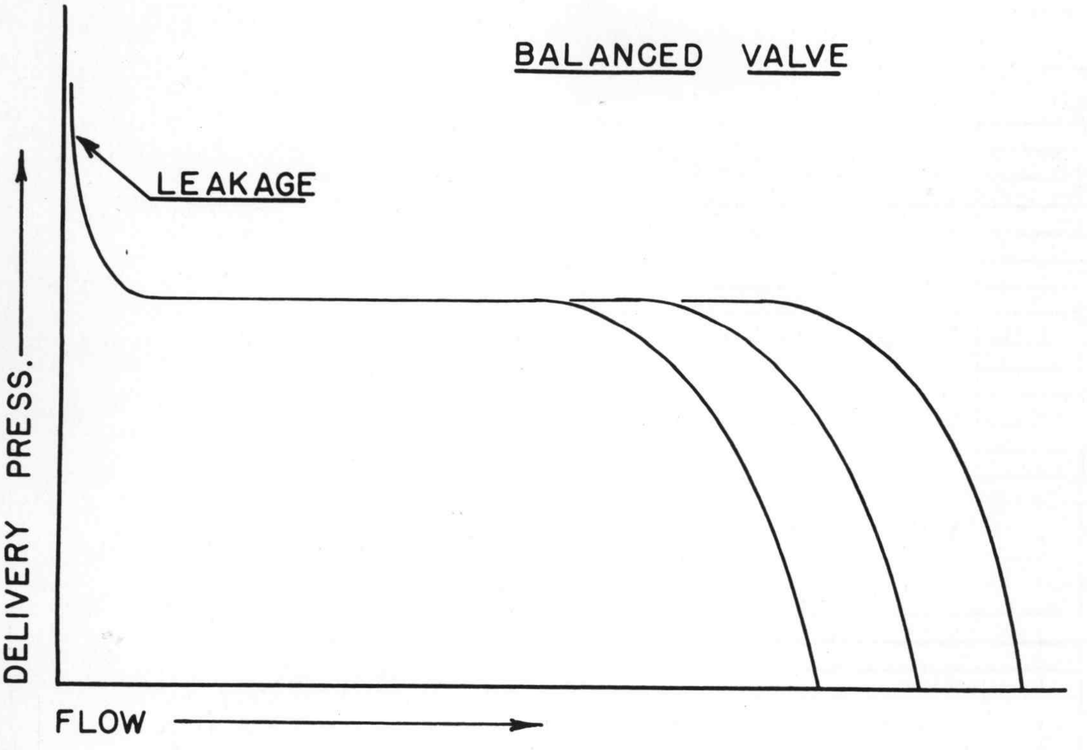


FIGURE 5(d)

performance.

The forces acting on the valve means are a function of the pressure differential times the effective valve area in addition to the force applied by other components of the regulator. For example a pressure differential of one hundred pounds per square inch across a valve of one hundreth square inch area being held in place by a one-half pound spring would produce a reaction force of one-half pound. The direction in which this force will act and its influence on the rest of the regulator is a function of valve design. Assuming pressure differential to be the only variable a valve which works against pressure by closing onto the downstream side of the valve seat would have a reaction force that would cause it to move toward the seat with decreasing inlet pressure. Such a valve would produce a curve distribution such as that shown by figure five(a) of the illustrated performance curves. In like manner a valve which closes with pressure onto the upstream side of the valve seat will move away from the seat with decreasing pressure to produce a curve distribution that would be illustrated by the performance chart of figure five (b).

It is desirable to have a regulator which has the same delivery pressure curve for all inlet pressures. It is this feature which has given rise to the balance, rotating, and sliding valve designs. These valves reduce the effective valve area to zero by counterbalancing the inlet pressure forces on two valves, one acting with and the other against pressure, or by having the pressure forces act at right angles to valve motion and absorbing them in the bearing surfaces of cylinder walls or pivots so that no reaction is transmitted to other components of the regulator. A possible balanced valve curve is shown in figure five (d).

The effective area of a valve is apt to change relative to degree of opening due to the fact that as fluid passes through all of the pressure drop does not occur at one point but will be distributed across the space between the valve and seating perifery of the valve seat. In addition to this, velocity pressure of the fluid striking the valve will produce and increasing effective valve area with increasing flow. Of course the degree of change in effective area with valve opening will be greatly influenced by valve design. This changing area will show up in the performance curve by influencing the slope of the straight part of the delivery pressure lines. Change in area due to pressure distribution across the valve seat will have its major effect at low flows with its corresponding narrow valve opening, and will have an assisting effect on both the valves closing with and against pressure.

Naturally when there are large pressure differentials across a valve seat there will be considerable fluid flow through extremely small openings. This condition will in effect produce a flow after the valve has actually contacted the seat, unless the valve and seat are perfectly matched and there is no dirt present. Conditions of perfect valve construction and absence of dirt are rare, and there is almost always some irregularity between the valve and its seat. The manufacturing practice of making either valve or valve seat of a soft material arises from this condition. To effect complete shutoff of flow will then require a little added force from the actuating parts of the regulator to deform and press the valve into the seat far enough to stop all leaks. This shows in the performance curve as a little up sweep of the delivery pressure lines as zero flow is approached. The increase in delivery pressure of low flow is sometimes referred to as the lock-up pressure. Some lock-up pressure is to be expected on regulators of all designs.

Lock-up pressures will be of similar magnitude on balanced valves as on single seated valves unless one of the valves seats a little ahead of the other in which case the elevation of lock-up pressure will be aggravated and an excessive deformation of one seat is necessary before the other will seat. Since it is mechanically difficult to get the two valves of a balanced valve timed just right, balanced valves are seldom recommended for installations where one hundred per cent shutoff is necessary. It is even more mechanically difficult to get a rotating or sliding valve to shut off completely. The performance curve for a regulator having such a leaking valve would have a delivery pressure curve that would rise parallel with the vertical coordinate at the leakage flow, as shown in figure five (d).

Flow capacity of a regulator valve influences the performance in two ways. During the period when the reaction forces of other parts of the regulator can adjust the valve opening to meet the flow demands on the regulator the size or capacity of the valve will determine the amount of valve stroke to produce an increment of flow change. For regulators having springs or other elements that have a spring rate this stroke will change the reacting forces in proportion to the spring rate and will tend to produce a negative slope in the delivery pressure lines as shown in the sample performance curve. When flow demands on a regulator reach the point where an increased valve stroke no longer opens up more area to fluid flow the ultimate capacity of the regulator will have been reached. This will be a function of the valve seat opening as though it were a fixed orifice. The ultimate capacity will show up in the performance curve as an abrupt break downward to zero pressure. The lines for higher inlet pressure will naturally carry out farther before the ultimate flow is reached as shown by the upper lines for valves closing against pressure and bottom lines for valves closing with pressure for inlet pressures arranged as is indicated.

Pressure Control Means. It is the pressure control means which must react most to forces which tend to unbalance a regulated state. Forces set up by the pressure control or diaphragm element are a function of delivery pressure times effective area of the diaphragm. Since the diaphragm area is relatively large, small changes in pressure will produce a large reaction force which tends to stabilize conditions, and thus outweigh those changing forces coming from conditions at the valve. The diaphragm element functions both as a pressure control means and as a mechanical seal across which a pressure differential and some reference pressure, usually ambient pressure, is set up to provide the necessary forces. This condition plus provisions made for flexibility will reduce the area actually affected by pressure. An exception to this would be in the case of a piston being used as the pressure sensitive means. If very large strokes are required there will be some change in effective area of a flexible diaphragm due to a change in the angle at which the pressure is acting.

The influence the pressure sensitive means has on the overall performance characteristics is strictly a function of its leverage over that of the valve and this leverage is usually a matter of relative areas. Therefore for the same valve and valve to diaphragm linkage a larger diaphragm would bring the parallel segments of the distribution lines closer together and a smaller diaphragm would spread the lines. If no provision is made in the diaphragm design for flexing the full stroke without stretching or bending there will be a spring rate of the pressure control means to be included in the reacting forces. This spring rate multiplied by the stroke per increment of flow will tend to produce a negative slope in the delivery pressure lines of the performance curve.

Load Control Means. The regulator component which has the most influence on the slope of the delivery pressure lines is the load control means. When springs are used to control the pressure level it is obvious that their inherent spring rate will cause a variation in control forces as the valve and diaphragm members change position to meet flow and pressure reduction demands and the greater the spring rate the more negative the delivery pressure slope. When fluids are used to control the load back of a pressure sensitive means then the resulting spring rate will be a function of the pilot valve or source of fluid supply, or the compressibility and volume of the fluid used.

The use of a dead weight as a method of loading the pressure sensitive means eliminates any slope to the delivery pressure line caused by load control. However, this type of control does impose a position limitation on the regulator to which it is attached, since the force exerted will be in the direction of gravity. Any regulator design which provides for small stroke of moving parts and low spring rate in load control means will have the least reduction in delivery pressure with increasing flow.

Effect of Linkages on Performance. The linkages between regulator components have a definite influence on regulator performance since they transmit all the inter-reacting forces. Naturally any spring rate of the linking mechanism will add to the overall spring rate with corresponding effect on the slope of delivery pressure curves. Most regulators have a valve return or counterbalancing spring which will influence this overall spring rate directly. If linkages involve mechanical leverages which increase the stroke of one of the regulator components a corresponding effect on performance curves will be produced. If the material from which linkages are constructed is flexible it will have an appreciable effect on delivery pressure slope under conditions of high force transmissions.

Hysteresis in the performance characteristics will be evident if the mechanical construction allows play or slipping at linkage joints. This condition will give different delivery pressures at the same flow and inlet conditions when flow is increasing than when flow is decreasing. Age and wear of regulator parts would naturally aggravate this condition.

Influence of the Housing on Performance. The housing contributes to the overall regulator performance by its control over the path which the fluids will travel in passing from inlet to outlet. This is especially true on the delivery side where velocities are high, and slight changes in effective pressures under the diaphragm element due to velocity effects will result in a response by the regulator. This condition is used to advantage by making use of a venturi effect created by fluid passing directly from valve opening to regulator outlet at right angles to the chamber under the pressure control means. In this manner an increasing flow will tend to reduce the pressure under the diaphragm which will react to maintain regulation and will produce a tendency to make the slope of delivery pressure lines positive as illustrated in figure five (c). By careful designing this effect can be made to just balance out negative slope tendencies of inherent spring rates. This venturi arrangement will have no effect on the ultimate capacity of the valve. Other conditions of regulator housing design may create an artificial positive pressure under the diaphragm with opposite effects on performance characteristics.

The housing influences overall regulator flow capacity in that the passageways provided add to the pressure drop through the entire unit.

Performance Characteristics of Two Stage Regulation. The purpose of two stages in a regulator is to decrease the spread in delivery pressure without using excessively large diaphragms. The net performance curve will be the summation of the effects of the two stages. The reduction in range of pressure variation produced in the first stage will supply fluid to the second stage at small enough variation for it to produce a final delivery pressure regulation that would approach a single delivery pressure line on the characteristic curve for wide ranges in supply pressure. By using a valve that closes with pressure in one stage and against pressure in the other; the summation of effects where one stage gives increasing delivery pressure with decreasing supply pressure and the other decreasing with the supply, and a tendency toward even more uniform outlet pressures.

However, this arrangement of two flow restricting valves in series is not conducive to high overall regulator capacity. If the valves are made large enough to compensate for the reduction in capacity the undesirable spread in regulated delivery pressures will again appear.

Regulator Performance Under Dynamic Conditions. The preceding discussion was based on uniform conditions for which data could be obtained and performance curves plotted. However, the very necessity for and use of a regulator precludes a tendency toward a non-uniform condition which requires a regulating means to maintain an uniformity in the phase of the system being used. Changing conditions may be slow as inlet pressures would be in the case of a diminishing supply from a fixed volume or they may be quite fast as outlet flows would be in the case of on and off operations. It is the rapid changes which require special attention in regards to regulator performance.

With rapid on and off operation the regulator mechanism may be slow in its reaction and produce a lag and over-shooting operation which may be self stimulating and produce a low frequency vibration or so-called hunting. When conditions of supply pressure and flow are such that the valve will be operating at a very small clearance between plug and seat there may be a condition where the effective area of the valve will change easily and quickly from one valve to another producing changing forces and valve positions. This condition will usually produce a high frequency vibration.

If components of a regulator are heavy or of such a design to possess considerable inertia there will be a tendency toward a condition of hunting back and forth over the regulation level whenever conditions are changed. Regulators that are loosely constructed and have hysteresis may hunt on rapidly changing loads. Also conditions within the system to which the regulator is attached may produce hunting. A system which has the point of use somewhat removed from the discharge opening of the regulator may set up pressure drop in the delivery line which will surge up and down with flow changes to produce an irregular regulator response. Conditions within a particular regulator construction which tend to produce varying and uncontrolled velocity pressure effects may also cause a low or high frequency vibration.

Performance of the Ideal Regulator. One might say that an ideally performing regulator is one which has one delivery pressure curve for all inlet pressures, a zero slope to the delivery pressure curve, no increase in pressure as zero flow is approached, and having no tendency to vibrate. Actually though, the regulator whose performance comes within the requirements of the system into which it is incorporated would have ideal performance. Usually an appreciable range in pressures can be tolerated by most systems requiring a regulator. As long as the condition of lock-up pressure increase occurs at flows below the range of use, the system will suffer no handicap. Even vibration can be tolerated if the condition which brings it on is outside the range of the system's operation.

For an instrument such as a pressure regulating valve where its degree of perfection in performance is a direct function of size, complexity, and cost it would be as much an incorrect design if it had super-performance as it would be if its performance fell below the system requirements.

REGULATOR DESIGN

Valve Design. With so many possible variations in design of each regulator component the problem of working out the best design is based on the simplest design which will meet the desired performance requirements. Choice of a valve means which closes with pressure is indicated when there is a relatively large pressure differential across the valve. This is the one form of valve means which allows all components to move in the same direction thereby providing the most simple design. The resulting direct action also provides for stability of adjustment, a minimum of hysteresis, and ease of valve to valve seat alignment. With the assistance of pressure a more or less positive seating action is obtained. The main disadvantage of the seat with pressure type of valve is the necessity of a valve stem down through the valve opening which reduces the ultimate flow capacity. In order to transmit forces necessary to open and close the valve, the valve stem must have rigidity and strength and will therefore occupy valve opening space that would ordinarily be available to fluid flow, and the valve area acted on by the pressure differential is not reduced accordingly. The seating force available is limited by the pressure differential since it is mechanically difficult to fasten the valve stem member to the diaphragm.

A valve means which closes against pressure is good design when a large capacity, with low delivery pressure variation, is desired. Large forces produced by slight over-pressures on the diaphragm are mechanically easily transmitted to the valve with subsequent reduction in leakage. The major design disadvantage of a valve which seats against pressure is the necessity for reversal of direction of valve action over that of the diaphragm. This reversal of direction requires the use of some leverage or yoke arrangement with subsequent vulnerability to hysteresis and increased spring rate.

Balanced valves are used when there are excessivly wide variations in inlet pressures and when little or no variation in delivery pressure can be tolerated. The very nature of the balance type of valve balances out all forces set up by the inlet pressures and consequently allows large valve openings with correspondingly large capacities. Difficulties encountered with balanced type of valves are mostly mechanical. One cannot rely on both valve members tightening up on their respective seats at the same time which condition will either cause leakage or excessively high lock-up pressures. When a balanced condition is obtained by a rotating or sliding valve action, the mechanics of obtaining positive seating are even more difficult. Suspension of the balanced type of valve within the regulator presents mechanical complications with reference to guiding or pivoting.

Materials of construction for valve mechanisms are divided into two classifications--one a hard valve on a hard seat and the other with either valve or seat being of a relatively soft material. The hard valve and hard valve seat combination is usually a metal to metal valve and has the advantage of accurate and stable valve adjustment with corresponding consistancy of performance. However, with valve components hard, it is extremely difficult to guarantee a leak tight valve. The purpose of a valve means having one of its components soft is to allow a deformation of the softer part around any imperfections of the seating surface due to dirt or non concentricity of parts. The choice as to whether the valve seat or the valve shall be soft is a function of ease of manufacture or assembly for the construction used. Difficulties encountered arise from the possibility of a permanent set being taken by the soft member plus a tendency to flow thereby changing the original valve adjustment. Numerous non metallic substances are used for the soft member with rubber being the most popular although some of the plastics have proven themselves suitable, especially nylon. Care must be taken to use materials that will not produce a sticky valve under heat or exposure to the fluid being regulated. Consideration must also be made of the possibility of heat of compression when inlet pressures are suddenly applied. This condition may cause spontaneous ignition of the non metallic material when oxygen is the fluid being regulated.

The ultimate flow capacity of a regulator valve means cannot be accurately predetermined due to the inherent irregularity of shape and flow passage. Actual capacity can only be determined by test; however, fairly close estimates can be made before the valve is tested or constructed, and by allowing a ten to twenty per cent safety factor, the required capacity should be within the estimate. The basic principle of obtaining a satisfactory estimate is that of resolving the free area of the valve into an equivalent area fixed orifice and proceding with the accepted principles for calculating flow through sharp edged orifices. The free area is the minimum cross section through which the fluid must flow in passing through the valve minus whatever space is taken up by valve stem, etc.

This is not the same area as the aforementioned effective area used to calculate the force created by pressure differentials across the valve.

Regulator design is simplified by the use of a valve spring to return the valve toward the seat on reduction of the flow demand. The other alternative is to tie the valve stem or lever directly to the diaphragm component which complicates adjustment and assembly. Spring rate of the valve stem spring is added directly to the overall spring rate. Valve guides are necessary to assure concentric and uniform seating of the valve member on its seat. It is good guide practice to provide a minimum bearing contact and a maximum range of contact. An example of a satisfactory guide arrangement would be a square guide in a round cylinder which would provide four relatively long widely spaced line contacts as bearing surfaces. Guides must also be light and free so there will be no appreciable addition to the regulator inertia.

In order to prevent vibration due to changing valve area it is common valve design practice to make the valve seat relatively sharp and to elevate it above its surroundings. This condition will fix the region of pressure drop along this sharp perifery. An absolutely sharp edged valve seat is not practical because of possible cutting of the soft material in the valve and the vulnerability of this edge to damage.

Care must be taken in designing the valve means that one does not put more size and complication into the assembly than is called for by the use to which a regulator is put.

<u>Pressure Control Design</u>. Design requirements of a

diaphragm means are that it shall be flexible, have low or consistant spring rate, have large effective area in relation to its total area, be leakproof, be inert to the fluid under regulation, and be able to withstand any pressures to which it might be exposed. The simplest pressure control means is a flat sheet of flexible material such as rubber or impregnated cloth or leather clamped into the proper position. Such a diaphragm is limited in stroke and would have a relatively high spring rate since any flexing away from the normal flat position would require that the material stretch. If a larger stroke or less spring rate is desired a bead or fold can be molded into the diaphragm to provide the extra material necessary to make the desired stroke with little or no tension in the diaphragm. Metallic diaphragms can be used in situations where an appreciable stroke is desired by using several folds or convolutions as in a bellows. Appreciable spring rates cannot be eliminated in the metallic elements but they can be made uniform and constant in nature.

The material to be used for diaphragms will be influenced by the pressure levels and the fluid to which it is exposed. Obviously the diaphragm must be of such strength and thickness that it will not burst and at the same time its weight and flexibility must be commensurate with the regulated pressure level in order to provide sensitivity without gravity effects. Likewise the material used must be impervious to chemical attack by the fluid being regulated. A combination of materials is sometimes needed to provide the required overall characteristics. For example, the impermeability of rubber may be combined with the strength of cloth or leather. Also considerable strength may be given to the flexible material by backing up all but a small flexing portion with heavy metal and providing stops at the end of the desired stroke. Regulation of hydrocarbon gases and fluids present a diaphragm material problem because of the solvent action on rubbers; however some of the new synthetic rubbers seem to be satisfactory. Leather diaphragms must be protected against dry rot.

Calculation of the effective area of a diaphragm cannot be made absolutely correct but can be approximated on the basis of effective diameters. The effective diameter is measured from the center of the flexing portion on one side to the center of the flexing portion on the other side. Thus if the entire diaphragm is flexing from the center to the outside the effective diameter would be one half the actual diameter and the effective area would then be one fourth the total area exposed. In order to achieve the maximum pressure leverage in a minimum of space it is desirable that the effective area be a larger fraction of the total area. This is done by placing a stiffening plate in the diaphragm center to push the flexing portion out toward the perifery. Thus a diaphragm with a center plate of one half its diameter would have an effective diameter of three fourths for an effective area of a little over one half. Utilization of a bead or fold in the diaphragm allows this flexing area to be close to the perifery and still have a suitable stroke. In cases where the diaphragm material itself is stiff as in the metallic ones, there will be no need for a stiffening plate and the effective diameter will be from the center of the bead or convolution on one side to that on the other.

<u>Design Considerations of the Load Control</u>. From the standpoint of ideal performance the load control should possess a very low spring rate, but from a mechanical standpoint this is very difficult. In order for a simple helical spring to have a low spring rate it must be long and large in diameter, which makes available space the limiting factor. For the fluid type of loading, change in pressure due to volume fluctuation as the diaphragm moves represents a form of spring rate. The dead weight loading means does give a zero spring rate but at the expense of a limited operating position.

The force that a load control means must exert on the pressure control or diaphragm means is the product of the regulated pressure times effective area of the diaphragm. With fluid loading it is a case of equal pressures with the diaphragm merely acting as a separator between the controlling fluid and the fluid being regulated. A change in diaphragm position will cause a change in loading force due to spring rate of regulator parts and will affect the regulating pressure accordingly.

The major design problem concerning load control is that of getting a minimum spring rate into a minimum space. By utilizing two helical springs, one within the other, the net spring rate can be reduced considerably. Tension springs and leaf springs can sometimes be arranged to give fairly low spring rates without occupying too much space.

<u>Linkage Design</u>. Since the main purpose of linkage within a regulator is to transmit the reacting forces of the other regulator components, it is a good practice to design a minimum of joints

and to keep all action in a straight line. The actual shape and type of the linkage element is more or less fixed by the type of valve and diaphragm means. The use of a leverage as the linking means may be used to advantage to minimize the forces acting on the valve, but the corresponding increase in diaphragm stroke may create a larger variation in outlet pressure due to the spring rate. When levers are used, care must be taken to include in reaction calculations any possible variation in leverage ratios due to changing fulcrums and bearing points. Any sudden changes in leverage ratios will set up a hunting or vibrating action. Linkage joints must be free but at the same time any undue sloppiness will produce hysteresis.

Housing design. In working out the shape and size of a regulator housing the main object is to encase and support all components of the regulator with a minimum size, weight, and number of parts. The more elements that can be attached to single housing piece the more uniform and stable the adjustments will be. Fluid flow passageways in the housing must be of sufficient cross sectional area to have no appreciable effect on the ultimate capacity of the regulator. By keeping the minimum diameter of passageways handling the full flow, at least twice the valve diameter, the contribution of the housing to the overall pressure drop will be negligible. As stated before these passageways must be arranged so that there will be no possibility of undesireable velocity effects.

Since all adjustments are based on relative positions of regulator components it is essential that the housing does not change in dimension. The utiliaation of several parts to make up a housing will lead to adjustment irregularities, especially if gaskets are used, since they will compress with use and change the relative position of components. The material used in housing construction must be commensurate with good mechanical design for the pressures to be encountered. From the production and maintenance standpoint the housing must be so arranged that vital parts such as valves can be removed easily.

OVERALL CALCULATIONS, DESIGN, AND PREDICTION OF PERFORMANCE

In order to integrate the preceding discussions of regulator design and performance, the following example of simple regulator design procedure is given. Assume that a special automatic control system is being installed on a production machine. This control system requires four cubic feet per minute of air at ten plus or minus one pound per square inch pressure, and a source of air is available at one hundred to two hundred pounds per square inch gage pressure.

The first step would be to check the requirements against performance of commercially available regulators. However, in this case it is assumed that such a regulator is not available and that one must be designed and built to meet the requiements. A regulator which would meet these requirements might be designed in the following way.

1. The valve size will be a function of the flow required, namely, four cubic feet per minute at ten pounds per square inch (assumed to be at seventy degrees Fahrenheit and dry). The density of dry air at 70°F.= 0.075 pounds per cubic foot. A safety factor of twenty per cent is to be applied. Since the minimum absolute inlet pressure of 114.7 pounds per square inch is well over the critical pressure, the weight of air flowing will be directly proportional to the upstream pressure in accordance with the standard formula, $W = 0.533 \frac{CAP_1}{\sqrt{T}}$ (1)

where; W = weight air flowing in lb./sec.

C = discharge coefficient (assume equal to that of a sharp edged orifice or 0.61)

A=orifice area in square inches.

P_1=inlet pressure in lb/sq. in. absolute.

T=air temperature in degrees Fahrenheit absolute.

By requirement the weight (W) of air flowing is equal to $4 \times 1.20 \times 0.075 \times \frac{24.7}{14.7} \times \frac{1}{60} = 0.010$ lb./sec.

Then from equation (1)

$$0.010 = \frac{0.533 \times 0.61 \times A \times 114.7}{\sqrt{460 + 70}}$$

A=0.00616 sq. in. which is the valve area which must be free to the flow of air.

Overall Calculations. For the sake of simplicity a valve that closes with pressure will be used in conjunction with a 1/32 inch diameter valve stem. The cross sectional area of this pin will be 0.00076 square inch so that 0.00616 + 0.00076=0.00692 square inch overall valve area which will be a 3/32 inch diameter valve.

2. The diaphragm size will be a function of the pressure leverage necessary to keep the delivery pressure variation within the plus or minus one pound per square inch allowable variation. Figuring some safety factor and allowing some variation due to spring rate, calculations will be on the basis of plus or minus one half pound per square inch or a total variation of one pound per square inch. Reaction forces arising from inlet pressure will be equal to valve area multiplied by the pressure differential, and the variation in these forces will be the difference between the force at two hundred pounds per square inch and one hundred pounds per square inch inlet, or, 0.0069 (200-10) - 0.0069 (100-10)=0.69 pound variation in force acting through the valve.

Therefore a one pound per square inch pressure change acting on the diaphragm must produce a reacting force of 0.69 pound and since this force change will equal diaphragm area times pressure change the area will equal $\frac{0.69}{1.0} = 0.69$ square inch effective area required of the diaphragm. By incorporating a slight safety factor and increasing this area to 0.7854 square inch; the effective diameter of the diaphragm will be one inch. Let the center disc or diaphragm plate be three fourths inch diameter then the outside diameter of the diaphragm will be one and one fourth inches and the flexing section will be one fourth inch wide.

3. The spring size and spring rate will be a function of the total load and extent of allowable variation due to spring rate. The total stroke of or movement of regulator parts in going from zero flow to the ultimate capacity will be a function of valve lift necessary to produce a periferal area equal to the free flow area of the valve plus the distance the valve must imbed itself into the valve seat to stop all flow. When one pound per square inch of the allowable two pounds per square inch variation is used up, and applying a twenty five per cent safety factor, the pressure variation allowable as a result of spring rate and lock up will be one half pound per square inch. The distance the valve must lift to produce the free flow area will be $\frac{0.00616}{3/32 \times \pi} = 0.0209$ inch. Assume the valve will move into the valve seat enough to produce lock up, then the total stroke to produce less than 0.5 pound per square inch pressure variation will be; $2 \times 0.0209 = 0.0418$ inch. The overall allowable spring rate will be; $\frac{0.5 \times 0.7854}{0.0418} = 9.4$ pounds per inch. Assume ten percent of the overall spring rate is due to elasticity of the diaphragm and ten percent is due to spring rate of a valve

return spring then the spring rate of the load control spring will be: 9.4 x 0.80=7.5 pounds per inch. The maximum working load on this spring will be a product of the regulated pressure times diaphragm area plus force on the valve at maximum inlet pressure plus spring tension on the valve return spring (which will be taken or 0.5 lb.) Then: 10 x 0.7854 + (200-10) 0.0069 + 0.5=9.66 pounds maximum working load.

To summarize the above sample regulator design problem the design specifications are as follows:

a. A 3/32 inch diameter valve closing with pressure, and having a 1/32 inch valve stem, and a 0.5 pound return spring of less than 1 lb./ in. spring rate.

b. A $1\frac{1}{4}$ inch diameter diaphragm with a 3/4 inch center disc, and having a spring rate of less than 1 lb./in. in a range of motion of approximately plus or minus 1/32 inch from the neutral or horizontal position, and capable of withstanding at least 20 lb./sq. in. air pressure without bursting.

c. A load control spring having less than 7.5 pounds per inch spring rate and a working load of approximately ten pounds.

d. A housing which will incorporate an inlet and an outlet connection of at least 1/8 inch diameter; provision for installing a valve assembly, a flanged arrangement for retaining and sealing the diaphragm in place, and a means of centering and adjusting the spring.

The estimated performance of a regulator made in accordance with the above specifications would be as follows:

a. An ultimate capacity of greater than 0.6 pound per minute ($0.6 \div 0.075$) or 8 cu. ft./min. at atmospheric pressure and 70°F. for inlet pressures over 100 pounds per square inch.

b. A total variation in delivery pressure of not more than plus or minus 3/4 lb./sq. in. for inlet pressure variations of 100 lb./sq. in. and flows from zero to eight cubic feet per minute.

c. The type of performance curve would be similar to that illustrated in figure five (b).

Since the above performance estimations come within the specifications set forth by the stated conditions of use the next step is to fabricate this regulator. In fabricating the regulator it is best to use as many standard parts as available. For example, standard ball bearings make good valves when rested on either metal or hard non metallic seats, and in this particular instance a standard automobile tire tube core valve meets the required design specifications and provides a complete ready made valve assembly. The diaphragm could be a thick flat sheet of rubber, a sheet of rubber impregnated canvas, or rubber backed with leather. The load control spring would probably have to be wound specially, and the housing would be a special fabrication job.

PERFORMANCE TESTS

Since some of the above design procedures are approximations and not absolutely accurate, it becomes necessary to conduct performance tests on any regulator where performance requirements encroach upon the safety factors used. Also the application of a regulator to different conditions than those for which data are available may indicate some forms of performance tests. With this in mind it is felt that a brief discussion of regulator test methods and procedures is within the scope of this paper.

The main conditions to be measured are flow and pressure. Measurement of temperatures, densities, and perhaps viscosities might be needed, but in general they will be known in advance and will not vary appreciably during tests. As described in the preceding chapter on regulator performance the characteristics of a regulator may be determined from data in these variables: inlet pressure, delivery pressure, and flow referred to some standard condition. Inlet and delivery pressures are measured with some calibrated pressure gage commensurate with the pressure levels used. For high pressures the standard Bourdon gages are most applicable and for the lower pressure ranges some form of water or mercury manometer might be necessary for the desired accuracy. Flow data may be obtained by means of any standard flow meter calibrated in the flow range involved. If a suitable flow meter is not available the flow may be measured on a pressure, volume, temperature, and time basis. A fixed volume high pressure supply cylinder lends itself to easy calculation of flow by pressure drop applied to the standard gas laws.

The performance characteristics curve is easily plotted from data obtained simultaneously on inlet pressure, delivery pressure, and flow. When these data are plotted in accordance with the example shown in figure five (b) with flow data converted to some convenient standard, the overall performance of the regulator becomes apparent.

Tests for performance under rapidly changing conditions are difficult and require the more complicated and expensive electronic pressure and flow pickup and recording instruments. However, any tendency for a regulator to start vibrating or hunting may be brought out by subjecting it to mechanical vibration or by imposing sudden changes in flow demand.

CONCLUSION

It is hoped that the preceding discussions will serve as a start toward making regulator design an engineering operation. It has been the author's experience that the mere classifying and recognition of the engineering fundamentals involved in an instrument leads to a simplification of design. Thus it is felt that continued study based on the material presented will lead to better regulator design and utilization.

Future improvements in regulator designs will be aided materially by the recognition of a standard nomenclature and standard form for the characteristic curve. The nomenclature used in this thesis is not necessarily the best, but wherever possible the terms used were those prevailing in most of the industry. The same is true for the performance curves presented.

Besides the work yet to be done in tabulating and establishing the known principles of regulator design there is considerable room for research and development in new methods of accomplishing the desired regulating functions. The development of more satisfactory means of balancing valves without sacrificing lock up is needed. Completely satisfactory systems of load control with nearly zero spring rates are yet to be developed. Considerable work is yet to be done on causes and cures for regulator vibration.

Answers to a large portion of the above problems are undoubtedly available in the cumulative knowledge of the experience of others working on pressure regulating valve design, and it is the sincere wish of the author of this thesis to stimulate these others to crystallize and publish a record of their regulator experience.