

technical monograph 27

Fundamentals of Gas Pressure Regulation

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Introduction

Gas pressure regulators have become very familiar items over the years, and nearly everyone has grown accustomed to seeing them in factories, public buildings, by the roadside, and even in their own homes. As is frequently the case with many such familiar items, we all have a tendency to take them for granted. Even the gas man who handles regulators every day as part of his job frequently tends to view the regulator simply as a piece of hardware which fits in the line and regulates pressure. The fact that it will do precisely that, for months on end without human intervention, makes it easy to maintain such a view. It's only when a problem develops or when we are selecting a regulator for a new application, that we need to look more deeply into the fundamentals of the regulator's operation.

Fundamental Essential Elements

The primary function of any gas regulator is to **match the flow of gas through the regulator to the demand for gas placed upon the system**. At the same time, the regulator must maintain the system pressure within certain acceptable limits.

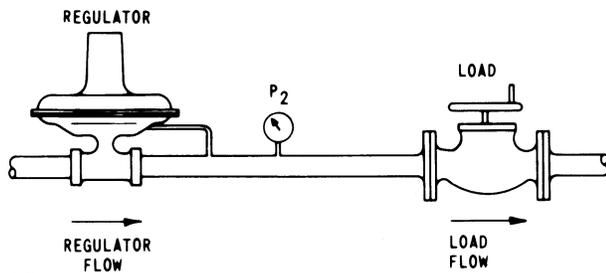


Figure 1. Typical Regulator System

A typical gas pressure system might be similar to that shown in Figure 1, where the regulator is placed upstream of the valve or other device that is varying its demand for gas from the regulator.

If the load flow decreases, then the regulator flow must decrease also. Otherwise, the regulator would put too much gas into the system and the pressure P_2 would tend to increase. On the other hand, if the load flow increases, then the regulator flow must increase also

in order to keep P_2 from decreasing due to a shortage of gas in the pressure system.

From this simple system it is easy to see that the prime job of the regulator is to put exactly as much gas into the piping system as the load device takes out.

If the regulator were capable of **instantaneously** matching its flow to the load flow, then we would never have any major transient variation in the pressure P_2 as the load changed rapidly. From practical experience we all know that this is normally not the case, and in most real-life applications we would expect some fluctuations in P_2 whenever the load changes abruptly. How well the regulator is capable of performing under these dynamic situations is one of the questions that we would ask ourselves when selecting a regulator for a given application.

Since the regulator's job is to modulate the flow of gas into the system, we can see that one of the essential elements of any regulator is a **RESTRICTING ELEMENT** that will fit into the flow stream and provide a variable restriction that can modulate the flow of gas through the regulator.

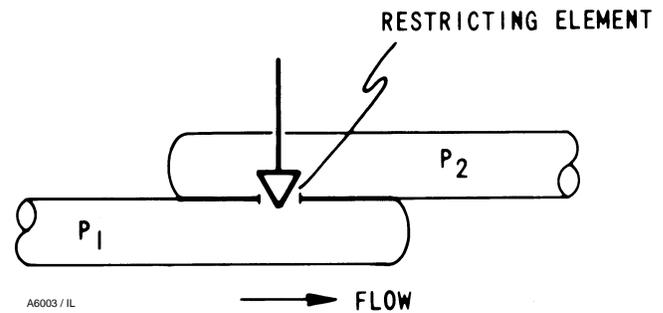


Figure 2. Typical Restricting Element

Figure 2 shows a schematic of a typical regulator restricting element. This restricting element is usually some type of valve arrangement. It can be a single-port globe valve, a cage style valve, butterfly valve, or any other type of valve that is capable of operating as a variable restriction to the flow.

In order to cause this restricting element to vary, some type of loading force will have to be applied to it. Thus we see that the second essential element of a gas regulator is a **LOADING ELEMENT** that can apply the needed force to the restricting element. The loading element can be one of any number of things such as a

weight, a handjack, a spring, a diaphragm actuator, or a piston actuator, to name a few of the more common ones.

A diaphragm actuator and a spring are frequently combined, as shown in Figure 3, to form the most common type of loading element. A loading pressure is applied to a diaphragm to produce a loading force that will act to close the restricting element. The spring provides a reverse loading force which acts to overcome the weight of the moving parts and to provide a fail-safe operating action that is more positive than a pressure force.

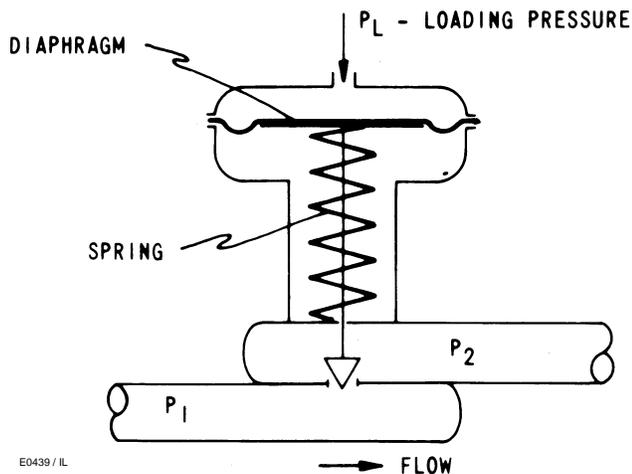


Figure 3. Typical Loading Element

So far, we have a restricting element to modulate the flow through the regulator, and we have a loading element that can apply the necessary force to operate the restricting element. But, how do we know when we are modulating the gas flow correctly? How do we know when we have the regulator flow matched to the load flow? It is rather obvious that we need some type of MEASURING ELEMENT which will tell us when these two flows have been perfectly matched. If we had some economical method of directly measuring these flows, we could use that approach; however, this is not really a very feasible method.

We noticed earlier, in our discussion of Figure 1, that the system pressure (P_2) was directly related to the matching of the two flows. If the restricting element allows too much gas into the system, P_2 will increase. If the restricting element allows too little gas into the system, P_2 will decrease. We can use this convenient fact to provide a simple means of measuring whether or not the regulator is providing the proper flow.

Manometers, Bourdon tubes, bellows, pressure gauges, and diaphragms are some of the possible measuring elements that we might use. Depending

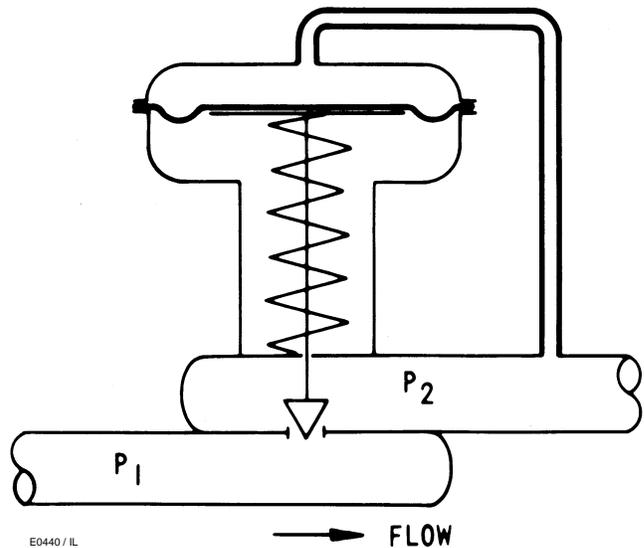


Figure 4. Adding a Typical Measuring Element Completes the Regulator

upon what we wish to accomplish, some of these measuring elements would be more advantageous than others. The diaphragm, for instance, will not only act as a measuring element which will respond to changes in the measured pressure, but it will also simultaneously act as a loading element. As such, it will produce a force to operate the restricting element that varies in response to changes in the measured pressure.

If we add this typical measuring element to the loading element and the restricting element that we selected earlier, we will have a complete gas pressure regulator as shown in Figure 4.

Let's review the action of this regulator. If the restricting element tries to put too much gas into the system, the pressure P_2 will increase. The diaphragm, as a measuring element, responds to this increase in pressure and, as a loading element, produces a force which compresses the spring and thereby restricts the amount of gas going into the system. On the other hand, if the regulator doesn't put enough gas into the system, the pressure falls and the diaphragm responds by producing less force. The spring will then overcome the reduced diaphragm force and open the valve to allow more gas into the system. This type of self-correcting action is known as negative feedback.

This example illustrates that there are three essential elements needed to make any operating gas pressure regulator. They are a RESTRICTING ELEMENT, a LOADING ELEMENT, and a MEASURING ELEMENT. Regardless of how sophisticated the system may become, it still must contain these three essential elements. Since these elements are so essential to the

operation of any gas pressure regulator, perhaps we should analyze each of them in more detail.

Restricting Element

The restricting element that we use will undoubtedly be some type of valve. Regardless of the style of valve which we use, we must remember that its basic purpose is to form a restriction to the flow. It forms a bottleneck which allows us to convert from a high pressure system to a low pressure system.

The pressure differential that exists across the valve represents a difference in potential energy that causes the gas to flow, just as water flows downhill. Our common sense tells us that if we increase this pressure differential across the valve, then we should thereby increase the flow of gas through the valve. In actual practice, this is true only up to a certain critical point.

If we look at a typical regulator installation, such as in Figure 5, we can analyze how the pressure varies along the length of the valve under steady-state flow conditions.

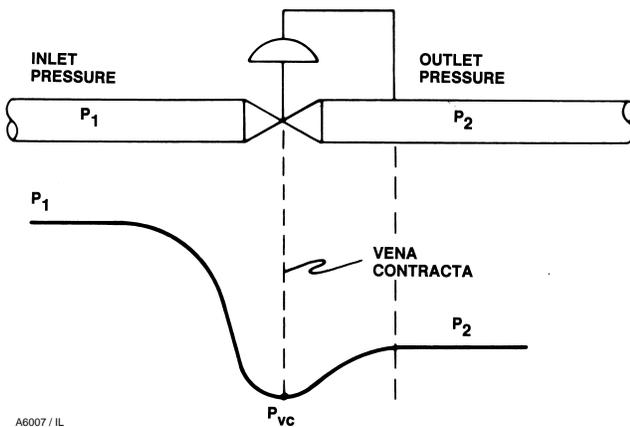


Figure 5. Profile of Pressure Across a Valve

The valve forms a major restriction in the line, and consequently its flow area is considerably smaller than that of the surrounding pipe. If we are to maintain a steady-state flow through this system, we must have just as much gas flow through the small area at the restriction as we have in the larger pipe area. It is obvious then that the **velocity** of the gas must be much greater at the restriction in order to maintain the flow of gas. The velocity will be the greatest at the point of maximum restriction of the flow stream. This point is called the VENA CONTRACTA and is normally located a short distance downstream of the actual orifice.

This increase in velocity represents an increase in kinetic energy that must come at the expense of the potential energy which is represented by pressure. Thus, in Figure 5, we see that the pressure decreases to a minimum at the vena contracta where the velocity is the greatest. As the gas slows down again in the larger downstream piping, we gain back some of the pressure that we lost. This is called pressure recovery.

If we try to increase the flow through this valve, we must, of course, increase the velocity of the gas at all points in the system. As we continue to increase the flow, we will eventually reach a point where the gas velocity reaches the speed of sound at the vena contracta. Since we cannot normally make the gas travel faster than this limiting sonic velocity, we have in effect reached the point where we can no longer increase the volume rate of gas flow through the valve simply by lowering the outlet pressure.

The point where we reach sonic velocity and the flow becomes limited is known as CRITICAL FLOW, and the pressure drop that exists across the valve at that point is known as CRITICAL PRESSURE DROP. If we were to increase the pressure drop beyond the critical point by lowering the outlet pressure of the valve, we would get no further increase in flow through the valve. The actual value of the critical pressure drop varies considerably depending upon the style and flow geometry of the valve.

As the gas passes through the valve, a certain amount of turbulence occurs which results in an energy loss within the valve. Part of the kinetic energy of the gas is changed into heat energy and part of it is changed into noise energy. A valve such as a ball valve or butterfly valve, that has a fairly streamlined flow pattern has a minimum of energy loss in the valve. This results in a relatively high downstream pressure recovery. This type of valve would be called a high recovery valve and is shown in Figure 6 as the solid curve. On the other hand, a valve, such as a double ported globe valve, which has a relatively turbulent flow pattern, will have a fairly high energy loss which means poor downstream pressure recovery. This type of valve would be called a low recovery valve and is shown in Figure 6 as the dashed curve.

Those who specialize in the theory of fluid flow tell us that the pressure differential between the inlet and the vena contracta ($P_1 - P_{vc}$) is a direct measure of flow regardless of the style of valve; whereas, the pressure drop observed across the valve ($\Delta P = P_1 - P_2$) is highly dependent upon valve style. Thus, if the two valves plotted in Figure 6 have **equal flow areas**, then we see that they would also have **equal flow**, since the pressure drop that determines the flow ($P_1 - P_{vc}$) is the same for both valves even though the observed pressure drops across the valves ($P_1 - P_2$) are quite different. The observed pressure drop across the valve

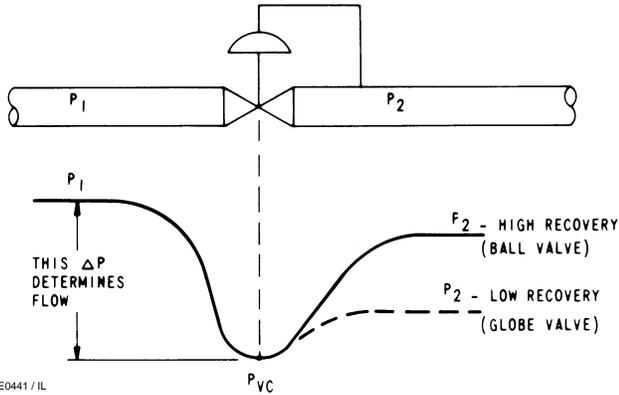


Figure 6. High and Low Recovery Valves

$(\Delta P = P_1 - P_2)$ is a direct measure of the pressure loss in the valve. Since it is quite difficult for us to measure the pressure at the vena contracta, we must, as a practical matter, resort to using the ΔP across the valve as a measure of the flow instead of $(P_1 - P_{VC})$. Since the ΔP across the valve is highly dependent upon valve style, it is obvious that experimental means must be used to find the relationship between this pressure drop and the flow for any given style of valve. This is the reason for the many tables of valve sizing data published by the valve manufacturer.

We said earlier that in general we could increase the flow by increasing the pressure drop across the valve. We have just discussed the fact that when we increase this pressure drop by lowering the downstream pressure, we eventually reach a critical condition where the flow no longer increases. We have reached critical flow because we have achieved the limiting sonic velocity at the vena contracta.

As it turns out, we can continue to increase the flow through the valve even after we have reached the critical flow condition. We do this by increasing the inlet pressure (P_1) to the valve. We would still have sonic velocity at the vena contracta and we would still have critical flow through the same flow area. What we have done, however, by increasing P_1 is increase the **density** of the gas entering the valve and have thereby packed more **standard** cubic feet of gas into each actual cubic foot of volume passing through the regulator. In effect, we have not changed the cubic feet per hour of flow, but we have increased the SCFH, or standard cubic feet per hour. As you will recall, a standard cubic foot of gas is that amount of gas that will occupy a volume of one cubic foot under standard conditions of 60°F and 14.73 psia.

Loading Element

Except for a few of the older weight loaded units, almost all gas regulators have springs. In fact, springs and diaphragms are by far the most universally used loading elements in modern gas regulators.

From a design point of view, there are several spring factors that are important such as type of material, wire diameter, spring diameter, free length, and the number of coils. From the strictly operational point of view of the gas man, however, there is only one factor of real significance. That factor is known as the spring rate.

Spring rate (k) is defined as the number of pounds of force (F) necessary to compress a spring by one inch. If a certain spring requires a force of 60 pounds to compress it one inch, then we would say that the spring rate was 60 pounds per inch. Since the spring rate is a linear relationship within its normal operating range, this same spring would require 90 pounds to compress it 1.5 inches.

If we took another spring and found that it would compress 2 inches under a force of 100 pounds, we could determine the spring rate by dividing the 100 pounds force by the 2 inch compression to obtain 50 lbs./in. This example suggests that we could define a simple formula that would describe this relationship.

$$k = \frac{F}{X}$$

where:

F = force (lbs.)

X = compression (in.)

k = spring rate (lbs./in.)

We can easily manipulate this equation into another common form ($F = kX$) that will allow us to find the spring force developed for any given spring compression. This form of this simple equation is very useful in the study of gas regulators.

As useful as a spring is, it can provide a loading force in one direction only. Energy must be supplied to compress the spring in the other direction. This energy usually comes from the force developed by pressure acting upon a diaphragm.

A diaphragm is simply a piece of fabric coated with some type of rubber-like material. The coating provides a seal to contain the pressure while the fabric gives it enough strength to withstand the pressure. Some diaphragms are molded into special shapes but most are simply cut from flat sheets.

For economic reasons the coating on the diaphragm may be much thinner on one side than the other. The

side with the thinner coating is usually identified by having some type of design or pattern printed on it. When a diaphragm of this type is used, it should be installed so that the pressure is applied to the non-patterned side or the side with the thicker coating.

When we speak of pressure, many of us are inclined to refer to it incorrectly as so many “pounds” of pressure. This is not only wrong, but it tends to obscure the real meaning of pressure. In reality, pressure is a force (lbs.) that is uniformly distributed over an area (in.²). Thus, we should more correctly refer to pressure as so many pounds per square inch. We can define a simple formula that will describe this relationship for us.

$$P = \frac{F}{A}$$

where:

F = force (lbs.)

A = area (in.²)

P = pressure (lbs./in.² or psi)

Again, we can easily manipulate this equation into another common form ($F = PA$) that will allow us to find the force developed by a pressure acting upon an area such as a diaphragm. This equation is also very useful to us in the study of gas regulators.

We don't have to be high powered mathematicians to make a rather interesting and informative study of gas pressure regulation. In fact, just an understanding of the two simple equations that we have described so far is enough to gain a rather comprehensive knowledge of the fundamentals of gas pressure regulation.

Measuring Element

A variety of devices is available for us as gas pressure measuring elements. Two of these, the manometer and the pressure gauge, are widely used as gas pressure measuring elements; however, they do not readily lend themselves to most control applications, so they are of little interest to us in our study of gas pressure regulation.

Bourdon tubes and bellows are gas pressure measuring elements that can be used for automatic control. They both can be made quite accurate and reliable; however, they also have the disadvantage that they require some type of auxiliary equipment to supplement their use.

By far the most universally used gas pressure measuring element is the diaphragm. There are several reasons that account for this. The diaphragm is simple, very economical, highly versatile, easy to maintain, and has the definite advantage of not requiring any additional equipment to supplement its action. In other words, the same diaphragm that serves as a measuring element can also serve as a loading element with no intermediate hardware being needed.

Because of its extreme simplicity, the diaphragm is frequently taken for granted and does not achieve the theoretical attention that it deserves. One important factor that the gas man must be concerned with regarding diaphragms is a proper determination of the **area** that the pressure will be acting upon.

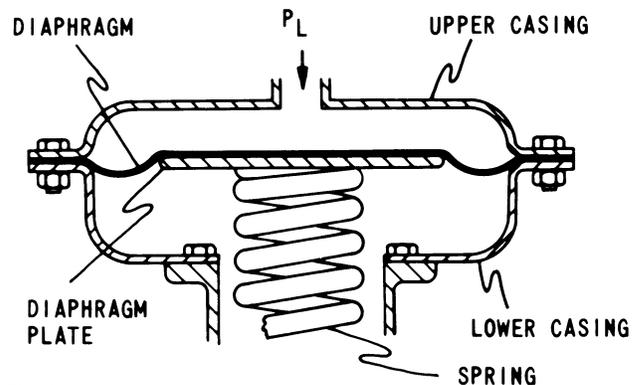
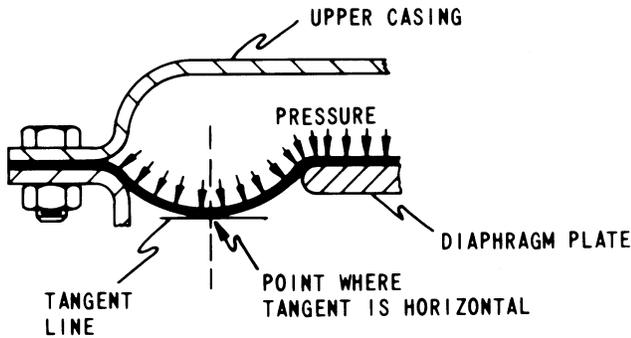


Figure 7. Actuator Diaphragm Assembly

If we take a look at Figure 7, we can see that the loading pressure (P_L) acts over the entire exposed surface of the diaphragm. The diameter of the exposed diaphragm area which the pressure acts upon is the same as the inside diameter of the upper casing.

If we are not careful, we can lead ourselves into a trap at this point. Just because the pressure acts over the entire exposed surface doesn't mean that all of that area is useful to us in terms of providing a loading force. In fact, once the question is raised, we might analyze the situation again and conclude that the only area which is useful is the area of the diaphragm plate, since that is the only place where the pressure is really pushing down on the spring assembly.

If we allow ourselves to come to this conclusion, we would be wrong again! The actual answer lies somewhere between these two extremes, but where? To find out, let's take a close-up look at the small portion of the diaphragm where it is unsupported between the diaphragm plate and the case flange. Because the diaphragm is unsupported here, the pressure acting upon it takes up the slack in the



E0443 / IL
Figure 8. Close-up of Diaphragm Convolution Section

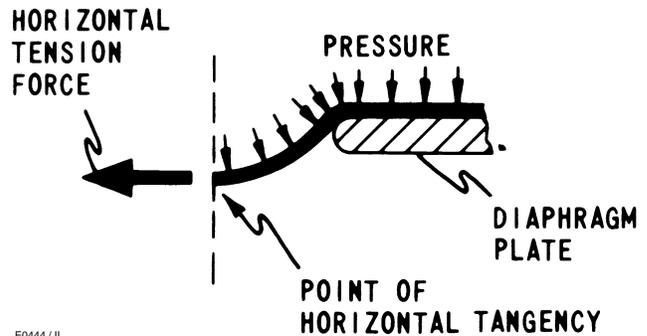
diaphragm and shapes it into what is known as a convolution.

Figure 8 provides a close-up look at how the loading pressure is uniformly distributed over the diaphragm convolution. We have to remember that this pressure is not only uniformly distributed, but also acts perpendicular to the diaphragm surface at every point.

As the pressure shapes the diaphragm convolution, we can visualize a line drawn tangent to the diaphragm at any point along this convolution. As we do this, we will discover that there is one, and only one, point where the tangent line is horizontal. We can identify this as the point of horizontal tangency. A vertical dashed line is drawn through this point in Figure 8 to divide the diaphragm convolution into two sections.

At this point we need to emphasize the fact that a diaphragm is basically a rubber coated fabric. The extreme flexibility of this type of material means that it can support neither shear nor compressive forces. The only force that can be sustained by any flexible material such as this would be a tension force that is always acting directly parallel to the fabric at any point. If we follow this line of reasoning in Figure 8, we can see that a **horizontal** tension force will exist in the diaphragm at the point of horizontal tangency.

The pressure acting to the left of the horizontal tangent point can only be transmitted to the diaphragm plate through tension in the diaphragm material. At the horizontal tangent point, this tensile force is horizontal, as shown in Figure 9, and therefore can contribute nothing to the up and down motion or the thrust of the actuator. In other words, the area outside of the point of horizontal tangency is not effective area. Thus, when we use the formula ($F = PA$) for the force developed by a pressure acting on an area, we must use the **effective** area in the calculation. As we have just shown, this effective area is calculated using the diameter between the points of horizontal tangency on the convolutions.



E0444 / IL
Figure 9. Tensile Force in a Diaphragm

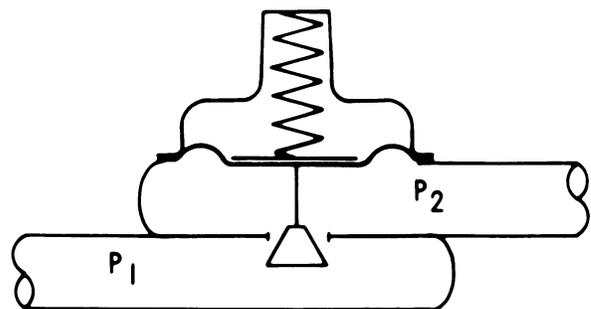
But let's look out! There is another trap here that we must avoid. It would be easy to assume that the point of horizontal tangency falls in the center of the convolution. This is only true for the one position in the stroke where the diaphragm plate is level with the flange. At other positions in the stroke, the effective area changes as the point of horizontal tangency moves inward or outward.

It is easy to visualize how this effective area changes if you hold a slack piece of string between your outstretched hands to simulate the diaphragm convolution. As you move one hand up and down to simulate the motion of the diaphragm plate, you will discover the following relationship.

As the diaphragm moves to COMPRESS the spring, the effective area DECREASES. As the diaphragm moves to RELAX the spring, the effective area INCREASES. This relationship will remain at your fingertips if you remember that the effective area decreases to its least amount just when it is needed most to help compress the spring.

Spring and Diaphragm Effect

Now let's use some of the fundamentals that we have developed about the three essential elements in order to study the operational performance of gas regulators.



A6005 / IL
Figure 10. Typical Self Operated Regulator

Let's look at a typical self operated regulator, such as that in Figure 10, when it is operating under steady-state conditions. In this situation the valve plug is in equilibrium with the pressure force exactly balancing the spring force. For simplicity, and to illustrate a point, the unbalance force on the valve plug is temporarily ignored. The pressure force is developed by the sensed pressure (P_2) acting against the diaphragm area. The spring force is developed from the compression that exists in the spring. We can express this relationship in equation form using the formulas that we developed earlier.

$$P_2 A = kX$$

We can rearrange this equation into a more convenient form by solving for P_2 .

$$P_2 = \frac{kX}{A}$$

Let's use this simple formula now to make some interesting observations about a regulator such as that shown in Figure 10. An example should serve to illustrate the basic principles. Assume that the regulator has the following values for its parameters.

$k = 160 \text{ lbs./in.}$	(spring rate)
$A = 80 \text{ in.}^2$	(effective diaphragm area)
$T = 2 \text{ in.}$	(total valve travel)

Furthermore, we are going to assume that the spring is adjusted so that it has one inch of compression even in the fully extended wide open valve position. This means that when the valve plug travels its full two inches to the closed position, the spring compression will be three inches.

Now apply our formula to the conditions at each end of the valve plug travel. Under very **low** load flow conditions the regulator will not be required to supply much gas to the system and the valve plug will be essentially in the closed position where the spring compression is three inches. The downstream controlled pressure, which is acting upon the diaphragm, can be determined from our formula.

$$\begin{aligned} P_2 &= \frac{kX}{A} \\ &= \frac{(160 \text{ lbs./in.}) (3 \text{ in.})}{(80 \text{ in.}^2)} \\ &= 6 \text{ psig (low load flow)} \end{aligned}$$

If the demand on the system were to change now so that the regulator must supply a **high** flow, the valve plug would have to open up to its wide open position. If we look carefully at our system in Figure 10, we see that it is the controlled pressure (P_2) that is actually

holding the valve plug closed. In order for the valve plug to open, P_2 must decrease, thereby allowing the spring to push it open. When the valve plug gets open, there will again be a balance of forces, and we can find the new value of P_2 . Remember, the spring still has one inch of compression in this open position.

$$\begin{aligned} P_2 &= \frac{kX}{A} \\ &= \frac{(160 \text{ lbs./in.}) (1 \text{ in.})}{(80 \text{ in.}^2)} \\ &= 2 \text{ psig (full load flow)} \end{aligned}$$

Thus, we see that the pressure had to decrease from 6 psig at low load to 2 psig in order to open the valve plug sufficiently to pass the full load flow. Furthermore, since this 2 psig is the pressure that will just hold the valve plug in the open position, this value of the pressure will have to continue as long as the high load flow condition exists. The pressure will only return to the original 6 psig when the load flow demand returns to its original low value.

If you wish to change our assumption regarding the amount of initial spring compression in the wide open condition, you can quickly verify, by similar calculations, that the actual values for P_2 will change also, but the decrease in P_2 will always be the same 4 psi. The actual magnitude of the decrease in controlled pressure required to open the valve is a function of the design parameters for the given regulator and is caused by the required change in spring compression. This is why it is occasionally referred to as spring effect.

Adjusting the amount of initial compression will change the value of the pressure at which we will operate under any given load condition. As a matter of fact, this is exactly how we adjust the set point pressure on our regulator. As we have seen, however, adjusting this spring compression does not change the amount of spring effect.

The decrease in controlled pressure that occurs as we increase the load is called **droop**. In the gas industry, the amount of droop that occurs in the controlled pressure when we go from a low load flow condition to a full load flow condition is defined as **PROPORTIONAL BAND**. This is frequently designated as **PB**.

In the last example we saw that the proportional band of the regulator was caused directly by the spring effect. In that example we assumed that the effective area of the diaphragm was constant. We know from our work in a previous section that this assumption is not always true. The effective area of the diaphragm may be 80 in.² when the valve plug is closed and the spring has its maximum compression, but when the

spring is in its most relaxed state and the valve plug is fully open, the effective area of the diaphragm will have to be something greater than 80 in². A typical value might be 100 in².

When we previously assumed a constant effective area, we found that our example regulator had a 4 psi proportional band. Let's see how the change in effective area would affect the proportional band.

$$\begin{aligned}
 PB &= (P_2)_{low\ load} - (P_2)_{high\ load} \\
 &= \frac{kX}{A}_{low\ load} - \frac{kX}{A}_{high\ load} \\
 &= \frac{(160\ lbs./in.) (3\ in.)}{(80\ in.^2)} - \frac{(160\ lbs./in.) (1\ in.)}{(100\ in.^2)} \\
 &= 6\ psig - 1.6\ psig \\
 &= 4.4\ psig
 \end{aligned}$$

We see that the proportional band of our regulator increased an additional 0.4 psi as a direct result of the change in effective area of the diaphragm. Thus, droop and proportional band are caused by both spring effect and diaphragm effect, as we can see from the plot in Figure 11.

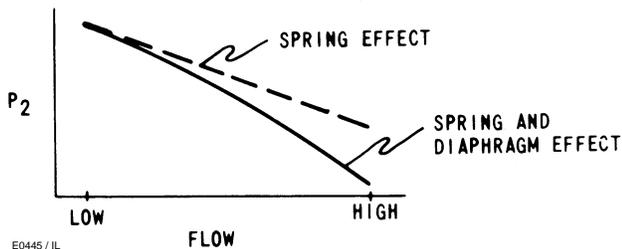


Figure 11. Plot of Droop Due to Spring and Diaphragm Effect

From a performance point of view, it is desirable to keep the proportional band as narrow, or small, as possible. One way that will help is to minimize or eliminate the diaphragm effect. In actual practice this is done quite frequently by using a molded diaphragm with deep convolutions rather than a simple flat-sheet diaphragm that has quite shallow convolutions. You can show yourself quite easily why the deeper convolution minimizes the effective area change by performing the string trick again that was suggested in the last section. As you move one end of the string up and down, you will notice that for a given amount of valve travel, the change in the position of the horizontal tangent point of the convolution is much less with a deeper convolution. This means that there will

be much less change in effective area with valve travel, and consequently, the proportional band of the regulator will be smaller. The major disadvantage of this approach is the higher cost of the molded diaphragm.

We can also narrow the proportional band by installing a spring with a **lower** spring rate, but we might introduce problems with valve plug chattering because of the lower stiffness. Another way might be to install a larger valve size so that the valve travel can be **reduced**. We could also install a different regulator whose effective diaphragm area is **greater**. This would theoretically reduce the proportional band, although we would almost certainly run into problems of exceeding the spring's capacity for travel or initial compression. The last two methods also have the disadvantage of requiring a major change in hardware.

Pilot Operated Regulators

So far we have only discussed self operated regulators. This is the name given to that class of regulators where the measured pressure is applied directly to the loading element with no intermediate hardware. There are really only two basic configurations of self operated regulators that are practical. These two basic types are illustrated in Figures 4 and 10.

If the proportional band of a given self operated regulator is too great for a particular application, there are a number of things that we can do. From our previous examples we recall that spring rate, valve travel, and effective diaphragm area were the three parameters that affected the proportional band. In the last section, we pointed out the way to change these parameters in order to improve the proportional band. If these changes are either inadequate or impractical, the next most logical step is to install a pressure amplifier in the measuring or sensing line. This pressure amplifier is frequently referred to as a pilot.

A typical pilot amplifier consists of a double diaphragm assembly that is rigidly fastened together as shown in Figure 12. This complete diaphragm assembly moves up and down under the action of the spring, the loading pressure (P_L) and the controlled pressure (P_2). As the diaphragm assembly moves, it modulates the flow through the supply pressure nozzle according to changes in P_2 . The maximum opening of this variable orifice must be larger than the fixed orifice shown as a hole through the upper diaphragm.

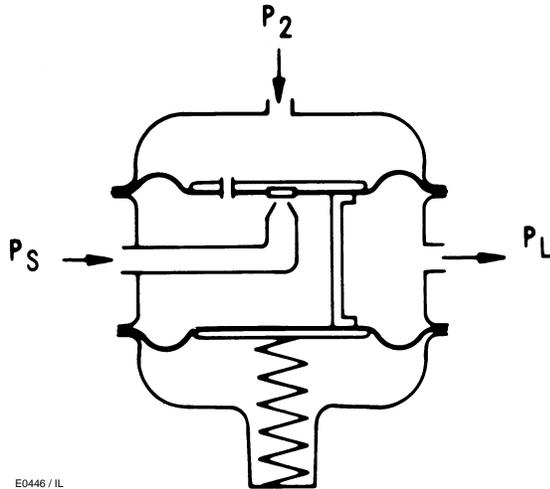


Figure 12. Pilot Amplifier

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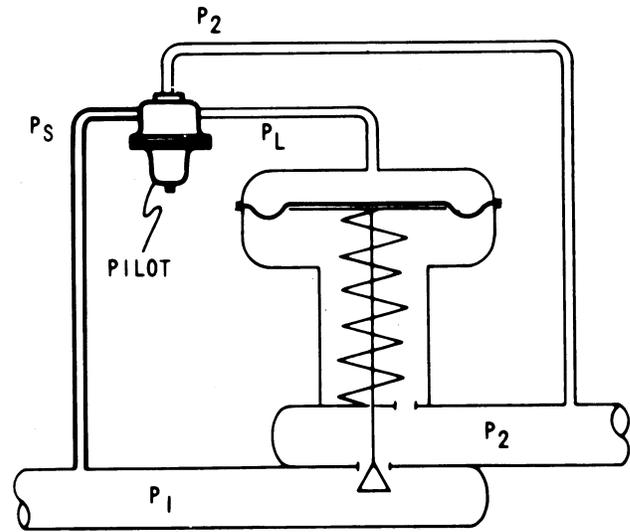


Figure 13. Pilot Operated Regulator

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When we introduce a source of supply pressure (usually P_1 from upstream of the regulator) through the nozzle or variable orifice, the loading pressure (P_L) builds up between the diaphragms because it can't escape fast enough through the fixed orifice. If we increase P_2 to cap off the nozzle then P_L will decrease since flow through the fixed orifice is now greater than through the nozzle. This means that the loading pressure is inversely proportional to P_2 ; i.e., P_L increases when P_2 decreases and vice versa.

A pilot amplifier such as this can be easily designed so that it takes very little change in P_2 to cause the double diaphragm assembly to move far enough to completely open or close the variable nozzle orifice. This means that a very small change in P_2 will result in rather large changes in P_L . In other words, the pilot amplifier has high gain. A typical pilot amplifier gain might be nominally about twenty.

Now, let's see how this pilot amplifier would operate when installed in the regulator sensing line as shown in Figure 13. Because of the pilot's inverse relationship between P_2 and P_L , you will note that it has been necessary to change the direction of the valve plug action in Figure 13. This is important because we still want an increase in P_2 to result in a closing action of the valve plug.

The pilot amplifier that we saw in Figure 12 had three pressure connections. As shown in Figure 13, one of these connections is for the controlled pressure (P_2). The second connection is for the supply pressure which we are obtaining from the upstream pressure (P_1). Finally, the output connection of the pilot is for the loading pressure (P_L) which is applied directly to the regulator diaphragm.

The purpose of the pilot amplifier is to sense changes in the controlled pressure and **amplify** them into larger changes in loading pressure on the diaphragm. The amount of amplification that we get is called the **gain** of the pilot. If we have a pilot amplifier with a gain of 20, then a one psi change in P_2 will cause a 20 psi change in the loading pressure on the diaphragm.

When the system load flow increases, the valve must come wide open just as before in the case of the self operating regulator. This is accomplished by decreasing the diaphragm pressure by the same amount as before. Previously, this amount of diaphragm pressure change showed up directly as droop in the controlled pressure. With our gain of 20 in the pilot amplifier, however, the controlled pressure only needs to droop one twentieth as much as before in order to get the same pressure change on the diaphragm. Thus, we have reduced the proportional band of our regulator by twenty, which is the amount of gain in the pilot.

In our prior example of the self operated regulator, we had a proportional band of 4.4 psi. If we were to install a pilot with a gain of 20 in that regulator, the proportional band would be only 0.22 psi. A rather impressive difference!

With the rather sensational improvement in proportional band that we can achieve with pilot operated regulators, one might wonder why all regulators aren't made this way. The answer is twofold, economics and stability. Pilot operated regulators are more expensive than similar self operated regulators, and the improvement in proportional band may not be sufficiently necessary to justify the increased cost. On the other hand, the gain of the pilot amplifier increases the gain or sensitivity of

the entire pressure regulator loop. If this loop gain is increased too much, the loop can become unstable and the regulator will oscillate or hunt.

When we discussed the supply pressure source for the pilot amplifier in Figure 12, we mentioned that this supply pressure is usually obtained from the gas upstream of the regulator. Since this is true, the pilot is normally designed so that when we bleed P_L down, the gas does not exhaust to the atmosphere but returns instead to the lower controlled pressure downstream of the regulator. Here its basic value as a source of energy can still be utilized.

In order to cause the loading gas to vent downstream, we have to make certain that the loading pressure (P_L) is always greater than the downstream pressure (P_2). One of the easiest ways to accomplish this is to provide some way for the downstream pressure (P_2) to act on the underside of the regulator diaphragm as we have done in Figure 13. Thus, the loading pressure (P_L) will always be forced to operate at a level greater than P_2 in order to be able to stroke the valve.

So far we have discussed spring and diaphragm effect and shown how this results in droop of the controlled pressure of a gas regulator. When we are dealing with strictly proportional control, as is the case in nearly all of the common gas regulators, this droop is a fact of life we must deal with. There is no way we can eliminate it. We can only minimize it with high gain in our regulator loop. We have also seen that the use of a pilot amplifier is one way to reduce this droop.

Service Regulators

A pilot amplifier is very effective in reducing droop in a regulator, but it tends to be expensive. Another method we can use to overcome droop is known as **velocity boosting**. In general, this method is not as good as the pilot amplifier method, but it has the advantage of being less expensive and can be made to work very well under the proper circumstances. Velocity boosting is most frequently used in house service types of regulators.

If we refer back to Figure 5, which shows the pressure profile across a typical regulator, we recall that the vena contracta pressure occurs just a short distance downstream of the actual restriction. Further downstream this pressure recovers to the value of P_2 , which is the controlled pressure and the one which is normally applied to the diaphragm.

A regulator which employs velocity boosting is designed so that the controlled pressure (P_2) is no longer applied to the diaphragm. Instead, a pitot tube, such as shown in Figure 14, or some similar design is

arranged so that the lower pressure near the vena contracta acts upon the diaphragm rather than using the higher P_2 .

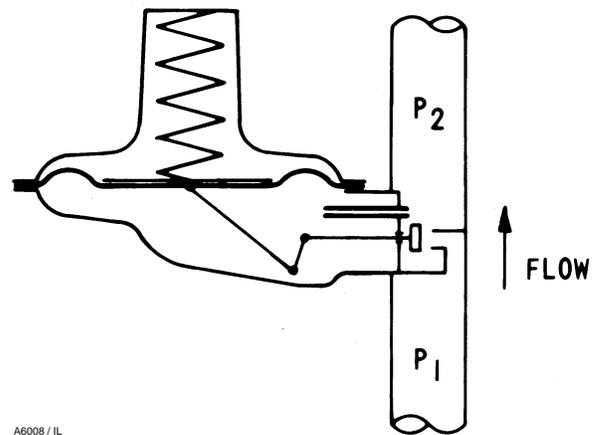


Figure 14. Velocity Boost with a Pitot Tube

As the load flow starts to increase, the sensed pressure at the pitot tube begins to droop just as P_2 does. Since the sensed pressure is near the vena contracta, and the gas velocity is greater there, this pressure, which is applied to the diaphragm, decreases more than P_2 . Consequently the valve is allowed to open slightly wider than it would if P_2 were acting on the diaphragm. This has the effect of keeping P_2 relatively more constant and thus preventing a large droop with high load flow.

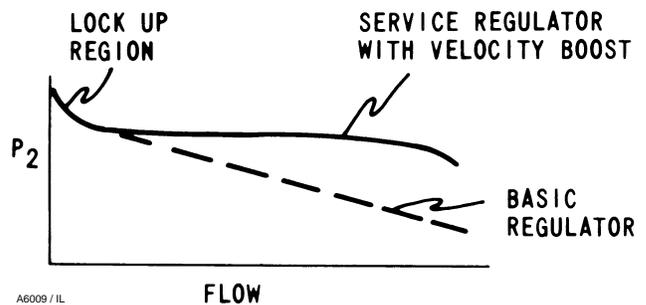


Figure 15. Performance Curves Showing Effect of Velocity Boosting

Spring and diaphragm effect still cause a drooping pressure at the sensing point similar to the basic regulator shown in Figure 15, but this is overcome by the subsequent downstream pressure recovery of P_2 .

On any service regulator, such as that shown in Figure 16, there is always the danger that the customer may shut off all of his appliances including the pilot lights. If that happens, the regulator must shut off for zero flow. This is called **lock up**.

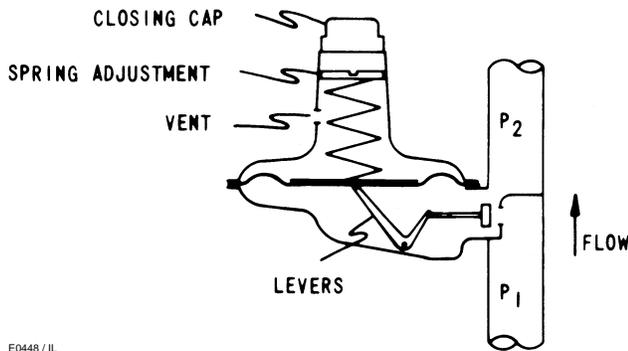


Figure 16. Basic Service Regulator Without Velocity Boost

To get tight shutoff the regulator uses soft seats which must be tightly compressed. This compression force can only come from the action of P_2 acting on the diaphragm, which means that P_2 must rise enough at zero flow to hold the seal tight. This lock up pressure can be clearly seen on the performance curves in Figure 15. The amount of pressure rise that we get in this lock up region is directly dependent upon the stiffness of the elastomeric material on the valve plug.

For economic reasons the house service regulator is usually a single port valve as shown in Figure 16. If we look at the upstream or inlet pressure (P_1) we can see that it is producing a force that is trying to open the valve. The controlled pressure (P_2) is acting on the back side of the valve plug trying to close it, but in a typical application, P_1 is much greater than P_2 . This leaves us with a rather large unbalanced force that is acting to open the valve plug. The only way that we have to balance this force is by P_2 acting on the diaphragm. If we get any kind of variation in the inlet pressure, then we can see that we will also get a variation in our controlled pressure.

One easy way that we can decrease this variation in P_2 , due to fluctuations in P_1 , is to use a mechanical lever ratio connecting the diaphragm assembly to the valve plug. In a typical service regulator, this lever ratio is about three to one. This means that P_2 only has to vary one third as much as before in order to balance fluctuations in P_1 . The lever only reduces the effect of valve unbalance, it does not eliminate it.

Since the magnitude of the unbalanced force on the valve is directly proportional to the valve area, it is to our advantage to select an orifice size which is no larger than necessary to pass the required flow of gas.

The magnitude of the unbalanced force on the valve also increases when P_1 increases. Thus, even though the lever system is helping to reduce the effect of valve unbalance, we still must have an increase in P_2

in order to balance either an increase in orifice size or an increase in P_1

A family of curves is shown in Figure 17 for a given orifice size which clearly illustrates that there is a different outlet pressure curve for each inlet pressure on a typical service regulator.

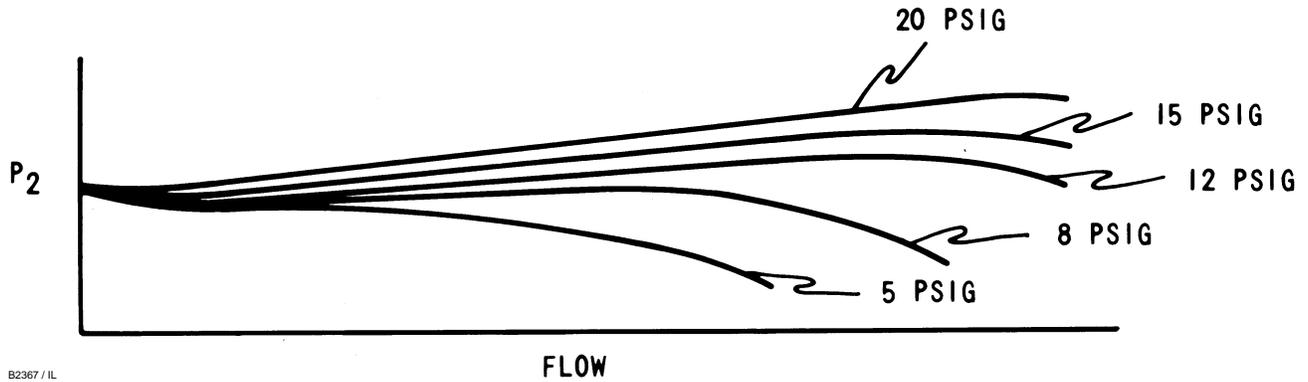
Most house service regulators control pressures in the vicinity of a few inches of water column, yet since each individual gas company's pressure setting may be slightly different, we need some method of adjusting the setpoint pressure.

We know from our previous discussions that P_2 acting on the diaphragm must provide a force that will balance the spring compression force. We can use the adjusting screw on top of the spring, as shown in Figure 16, to increase the total compression force in the spring. P_2 must then also increase in order to balance the larger spring force and hold the valve plug in the proper position. Thus, by changing the initial compression in the spring, we can adjust P_2 to operate at any setpoint value that we wish within a certain range defined by the load limits on the spring.

Another factor that we frequently need to take into consideration when working with service regulators is the weight of the moving parts in the regulator. When installed with the regulator spring on top, the weight of the spring and diaphragm assembly produces an additional downward force that must be balanced by an additional increase in P_2 . If this same regulator was turned upside down, the weight of the moving parts would then operate in the opposite direction and the setpoint pressure would change. Since, in this upside down position, the weight is opposed by the spring instead of the pressure, we can use the initial compression in the spring to support the weight of the parts.

The upper casing of the regulator is designed so that it will hold the spring properly, but it also protects the regulator parts from exposure to the weather, dirt, etc. Just as important, it also keeps a cushion of air above the diaphragm. As the diaphragm assembly moves up and down, the air in this upper casing must move in and out through the vent hole, otherwise, the compression and rarefaction of air in the upper casing would interfere with the diaphragm movement.

If we restrict the flow of air in and out of the upper casing by just the right amount, we can damp the tendency of the diaphragm assembly to go into a sustained oscillation. This type of oscillation is called hunting or buzzing. This is why a regulator will sometimes buzz when we remove the closing cap to adjust the spring. When we close the cap again, the regulator stops buzzing. The same thing can sometimes happen if the vent hole is enlarged for some reason.



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Figure 17. Variation of Controlled Pressure with Changes in Inlet Pressure on a Typical Service Regulator

Conclusion

It should be obvious at this point that there are a great many important fundamentals to understand in order to properly select and apply a gas regulator to do a specific job. Although these fundamentals are profuse in number and have a sound theoretical base, they are relatively straightforward and easy to understand.

As you are probably aware by now, we made a number of simplifying assumptions as we progressed. This was done in the interest of gaining a clearer

understanding of these fundamentals without getting bogged down in special details and exceptions. By no means has the complete story of gas pressure regulation been told. The subject of gas pressure regulation is much broader in scope than can be presented in a single document such as this, but it is sincerely hoped that this paper will help the gas man to gain a working knowledge of some fundamentals that will enable him to do a better job of designing, selecting, applying, evaluating, or troubleshooting any gas pressure regulation equipment.

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